MODERN MARINE ENGINEER'S MANUAL

Volume I

THIRD EDITION

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For the Regiment of Midshipmen
at Kings Point, New York

In memory of Jay

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Foreword

The term "global economy" has been much overused to describe many things-the rise and fall of financial powers, the redistribution of wealth, streamlining, downsizing, and more-all suggesting a relatively new phenomenon. However, global economy is still largely about international commerce-trade across the seas-as it has been for centuries. And transport across those seas continues to be done by high-capacity ships capable of moving cargo over long distances. Commerce always involves a balance of factors such as superior materials, skilled labor, low cost, and state-of-the-art technology. These factors can be readily modeled and analyzed using computer software costing as little as a semester at a local college. The interaction of variables has been examined sufficiently so that patterns of trade or specifics of vehicle selection can be fine-tuned for viability. Sea trade remains the favored method of global transport; the overwhelming majority of international commerce is still carried by ships, as it has been for nearly three millennia.

Over the last quarter-century, technical developments and economies of scale have resulted in dramatically lower costs for building, propelling, and loading ships; the attendant labor cost components are becoming almost the sole determinant of competition among the world's fleets. Materials for shipbuilding can now be produced, refined, and shipped anywhere at such low cost that sourcing differences are insignificant. Much of the credit for this progress goes to advancements in areas closely aligned with and including marine engineering.

Ships-those remarkable, self-contained, floating cities-still face the timeless challenges associated with the sea. No other mode of carrier transport is required to function reliably and continuously at full power for long periods and have the capability for adequate maintenance and repair to be done in-situ. Ships must be designed so that basic ship functions can be carried out at untoward angles of trim or heel and despite the alternating orientations caused by pitching or rolling. Engineering considerations are complicated by ever-increasing engine cycle temperatures further compounded by salt-laden combustion air; engineers must continually strive for the economy to be gained from using lower-quality fuels.

New challenges appear at a brisk pace. While ship operating economy continues to favor lower-paid and potentially less-skilled crews, these same individuals are still expected to be knowledgeable about the increasingly complex ships they sail. The extensive use of shipboard electronics and the expansion of international safety criteria have significant potential for worldwide benefit, but they also present additional concerns regarding the day-to-day operations of ships of all sizes and types. These operating realities further challenge those who design, build, and manage ships.

This book offers the fundamental elements required to help engineers stay current on ways to benefit from the technological advances occurring in these rapidly changing times. The third edition of volume 1 of the *Modern Marine Engineer's Manual* is a superb up-to-date reference for students and practicing professionals alike. It incorporates state-of-the-art changes that have been implemented since the publishing of the second edition, with emphasis in appropriate areas. Written by experts in marine engineering and the relevant academic fields, it is authoritative and contains a significant amount of high-caliber input.

David A. O'Neil
President, 1997-1998
The Society of Naval Architects and Marine Engineers

Preface to Third Edition

The first edition of volume 1 of *Modern Marine Engineer's Manual* was published on the eve of World War II to provide a useful and practical text for the engineering officers, students, port engineers, and ship repair specialists of a rapidly expanding American merchant marine. The second edition, published twenty-four years later, provided useful updates of the original text. Due to dramatic changes in all aspects of ship machinery and ship operations during the past thirty years, this third edition is not a revision of past editions but an entirely new text written in the tradition of earlier editions. All the contributing editors are experts in the areas for which they have prepared chapters. Many are employed as consultants; others hold academic appointments in their fields.

The diesel engine-now the most popular form of main propulsion system-is covered in volume 2. The third edition of volume 1 remains primarily a source of information on steam and gas turbine power plants. The reciprocating steam engine is no longer covered in the text, and the material on steam turbine propulsion has been reduced. However, gas turbine main propulsion has been covered in detail in the expectation that this type of power plant will become increasingly popular as new environmental regulations continue to require the use of higher-quality fuel, contributing to improved economics for the gas turbine. It is possible that gas turbines combined with heat recovery steam generators and steam turbines may become the most popular propulsion system for cruise liners.

Also covered in this newest edition is the personal computer, which is rapidly becoming essential shipboard equipment for many tasks, including spare parts management, maintenance programs, vibration analysis, power plant analysis, management systems for quality and safety, communications, and record keeping.

International, national, and local laws and regulations concerning the protection of the environment and the safety of shipboard personnel and property plus the rules of classification societies and flag states all combine to provide a new challenge to ships personnel. Shipboard systems designed to comply with many of these requirements (including ISO 9002 and the ISM code) are described in the text.

Pumps, pumping systems, and heat exchangers, which are found on all types of ships, are given extensive coverage.

Petroleum fuels are frequently treated chemically and processed mechanically on modern ships. The characteristics of fuels, fuel chemical treatment, fuel mechanical processing, and the implications of such treatments and processes for the maintenance of both internal combustion engines and boilers are presented.

Since shipboard equipment in most ofthe world is manufactured to the metric system, metric measurements are used along with the traditional American units of measure.

In recognition of the use of the text by students, each chapter includes review questions as well as references to materials for further study.

The editors wish to thank all the companies and organizations who gave permission for the use of illustrations and other material in this edition. The names and locations of these companies are acknowledged at the end of each chapter.

Preface to the First Edition

The expansion of shipbuilding made evident about four years ago that there was need of an American textbook on marine engineering that would adequately explain the design and operation of all the general types of marine equipment and at the same time should be written simply, to be easily understood. Because marine engineering was, and is advancing and changing so rapidly, it was necessary that a considerable amount of theory be included in order that the student be prepared to understand future developments in the field of marine engineering. There was the thought too, that for effective use in this time of stress, it would have to be widely distributed among the shipyard and seagoing personnel. This meant that the price ofthe book had to be such that the men could pay.

At this point, it may be mentioned that methods of study of a technical book are very important if useful results are to be obtained. A certain time should be set aside each day for study. This may be interfered with by outside emergencies, but every effort should be made to adhere to it. A short section of the book should be read through completely each day. Then it should be re-read and important words underlined in pencil. The drawings may then be copied in the notebook.

It may also be mentioned that many men "look, but see not." Every man in the "black gang" should be able to sketch on paper the position of every important piece of equipment in the engine room of his ship and know the position of every important control and valve.

More and more marine engineering design is breaking up into specialties and this is the reason that this book is written by a number of men. The authors of the various chapters of the book are specialists, each on his subject, some are engineers of the U.S. Maritime Commission and others are engaged in various outside branches of the maritime industry. Anyone of the authors of the manual will be glad to answer any difficult point that may be brought up in regard to his specialty. Should the student wish to reach one of them he should write care of the Cornell Maritime Press.

It will be noted by those familiar with the subject that a large use has been made of the instruction pamphlets of the U.S. Navy. This use was made both because of the short time available to prepare this manual and because of the excellence of the Navy material.

The experienced marine engineers will notice the omission of many excellent pieces of marine equipment from these pages. This was due to the sharp necessity for conserving time and printed space. In regard to printed space the editor believed that a full description of a single type of equipment to be greatly preferred to cursory and inadequate descriptions of the products of all the various manufacturers. That a piece of equipment is presented in this book does not mean that the author prefers it to some other piece of equipment that may not be mentioned. It may but illustrate the point of the subject better.

The thanks of the editor go out to the splendid cooperation he has received from the authors of the chapters, from the publishers, from all branches of the marine industry without exception and from his superiors in the U.S. Maritime Commission.

Grateful acknowledgment is hereby made to those companies which have supplied us with data and illustrations concerning their products: ....

[The remaining paragraphs of the preface were devoted to an extensive list of companies and organizations that were of help to the editor.]

September 2,1941

A.O.
[Alan Osbourne]

MODERN MARINE ENGINEER'S MANUAL

Volume I

CHAPTER 1

Thermal Sciences and Engineering

JAMES A. HARBACH

The thermal sciences include the closely related sciences of thermodynamics, fluid mechanics, and heat transfer. Thermodynamics is the study of energy in its various forms and the transformation of energy from one form into another. Fluid mechanics is the study of the flow of liquids and gases. Heat transfer is the study of the transfer of heat in various situations as the result of a temperature difference.

The analysis and design of energy systems require the use of all three of the thermal sciences. For example, the overall performance of a refrigeration system can be analyzed using only the principles of thermodynamics, but the design of that system's thermostatic expansion valve relies heavily on the use of fluid mechanics, and determining the surface area of its condenser and evaporator requires the use of heat transfer principles.

THERMODYNAMICS

Definitions and Units

Before a study of thermodynamics can begin, a brief review of some basic definitions and unit systems is necessary.

SUBSTANCES

Most energy systems are based on the use of a working substance to carry energy from one location to another. A pure substance is one that has the same chemical and physical composition at all points. A steam power plant uses water, a pure substance, in its operation. Air is used as the working substance in many systems, for example, an open-cycle gas turbine. Air is not a pure substance but a mixture of oxygen, nitrogen, and several other inert gases.

PHASES

A substance may exist as a solid, a liquid, or a gas. There are specific terms used to describe the transition from one phase to another. Freezing occurs as a liquid becomes a solid. Boiling occurs when a liquid becomes a vapor (gas). Sublimation occurs when a solid turns directly into a gas.

PROPERTIES AND STATE OF A SUBSTANCE

Thermodynamic properties are observable characteristics of a substance such as pressure, temperature, and density. The condition, or state, of a substance can be described by its properties. For example, knowing two independent properties of water, a pure substance, defines its state. This allows all the other properties of the water to be determined.

CLOSED AND OPEN SYSTEMS

A thermodynamic system is defined as a collection of matter or space. If a given mass of substance is defined as the system, this is referred to as a closed system. Air expanding in a piston-cylinder would be most easily analyzed as a closed system. If a given volume is defined as the system, this is referred to as an open system. A section of pipe with water entering and exiting is an example of a fixed-volume or open system. Note that in a closed system, only energy crosses the system boundary. In an open system, both energy and mass cross the system boundary. It is important to note that the selection of the system boundaries determines whether the system is open or closed. These boundaries can be selected based on the analysis to be performed.

PROCESSES AND CYCLES

A process is the change in the state of a system from one point to another. There are an infinite number of ways of getting from state one to state two. The intermediate states that the system goes through describe the path. Two examples of common processes are constant-temperature and constant-pressure. In the first, the temperature is the same constant at all system states. In the second, the pressure is unchanged. A series of processes can be put together to return to the same system state. Such a series of processes is called a thermodynamic cycle (see fig. 1-1).

UNIT SYSTEMS

Engineers work with numbers daily. Dimensions are normally associated with those numbers, defining the value of properties such as length, pressure, and temperature. The two measurement systems in common use are the English engineering system (U.S. customary units) and the *Systeme International D'unites* (SI) system.

Units are either fundamental or derived. The fundamental units are defined and form the basis ofthe unit system. All other units are then de-

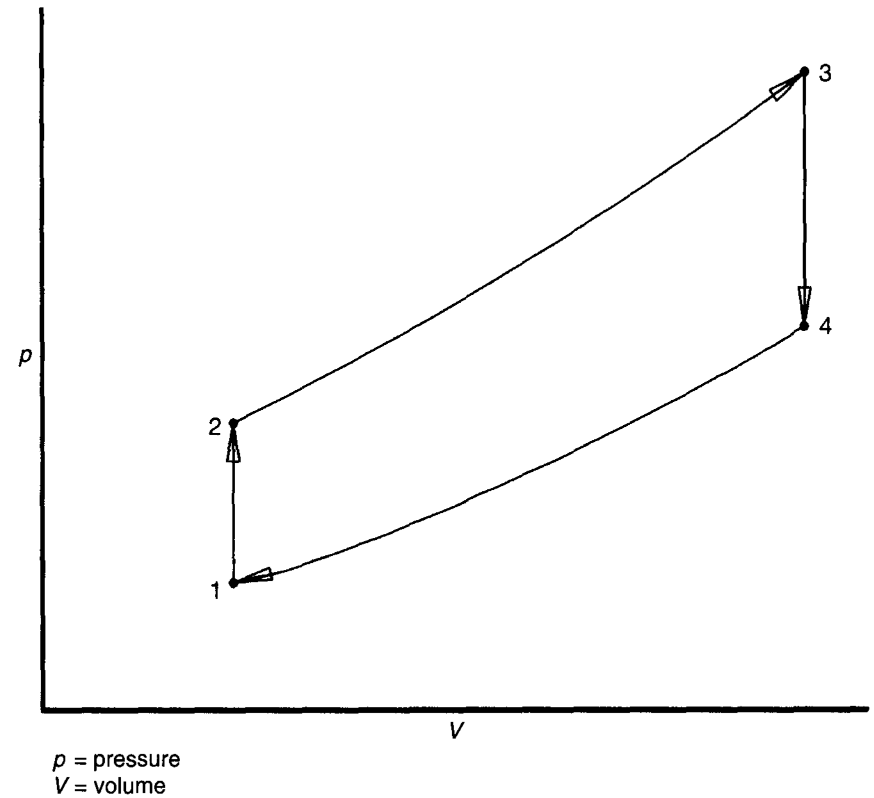


Figure 1-1. A thermodynamic cycle

rived from the fundamental units. Common fundamental units are length, time, and mass or force. In the English engineering system, force is a fundamental unit, and mass is derived using Newton's second law. A mass of one pound is defined as that which exerts one pound of force under the acceleration of standard gravity (32.174 ft/sec^2):

$$F = ma/g_c \qquad \text{lbf} = \frac{(\text{lbm})(\text{ft}/\text{sec}^2)}{(32.174\ \text{lbm-ft}/\text{lbf-sec}^2)}$$

In the SI (metric) system, mass is a fundamental unit, and force is defined. One newton (N) is defined as the force necessary to accelerate one kg of mass at a rate of 1 m/sec^2:

$$F = ma \qquad \text{N} = (\text{kg})(\text{m}/\text{sec}^2)$$

Note that in the SI system, a conversion constant (g_c) is not required to achieve consistent units. Since the standard acceleration of gravity is $9.807\ mlsec^2$, one kg of mass "weighs" 9.807 newtons at sea level.

TEMPERATURE

Temperature scales were originally based on the freezing and boiling points of pure water at standard atmospheric pressure. In the English engineering system, water freezes at 32°F and boils at 212°F. In the SI system, water freezes at ooe and boils at lOOoe.

A temperature scale can also be defined with absolute zero (the lowest possible temperature) as a zero reference. For the Rankine absolute temperature scale, each degree division is the same as the Fahrenheit scale. Since absolute zero is -459.67°F, water freezes at 491.67°R and boils at 671.67°R. The absolute SI temperature scale is Kelvin. Water freezes at 273.15°K and boils at 373.15°K. A comparison of the four temperature scales is shown in figure 1-2.

PRESSURE

Pressure is defined as the normal force per unit area. The most common unit of pressure in the English engineering system is pounds of force per

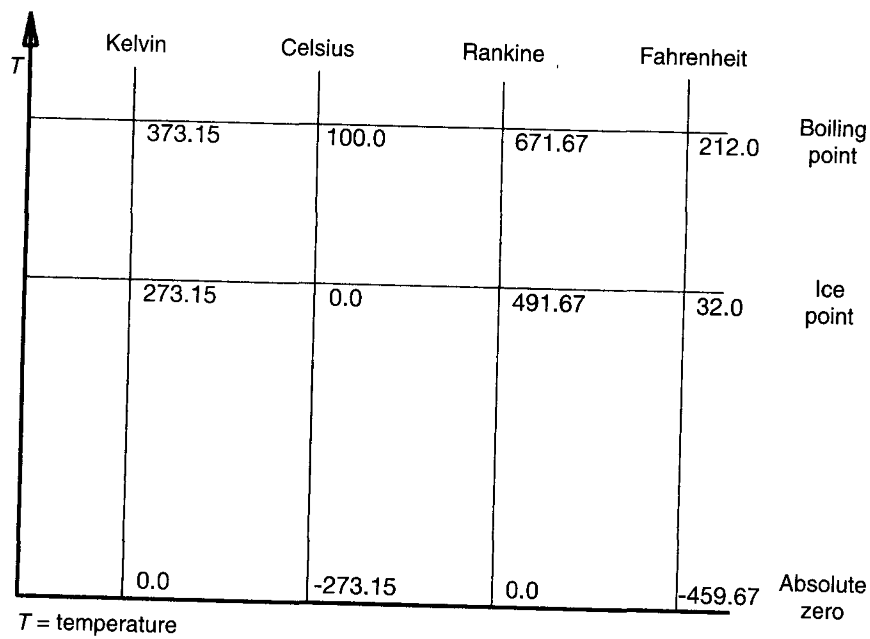


Figure 1-2. Comparison of temperature scales

square inch (psi). In the SI system, the basic unit of pressure is newtons per square meter, or pascals (Pa). Thus,

$$1\ \text{Pa} = 1\ \text{N/m}^2$$

Because 1 Pa is a very small change in pressure, pressure in SI units is commonly presented in kPa, where 1 kPa = 1,000 Pa.

Pressure scales commonly use either the pressure of the surroundings or a perfect vacuum as a reference. Gauge pressure is defined as pressure above atmospheric pressure. Vacuum is defined as pressure below atmospheric. Absolute pressure is the pressure above a perfect vacuum. For positive pressures, the absolute pressure is thus the sum of the gauge pressure and the atmospheric pressure:

$$p_{\text{abs}} = p_{\text{gauge}} + p_{\text{atm}}$$

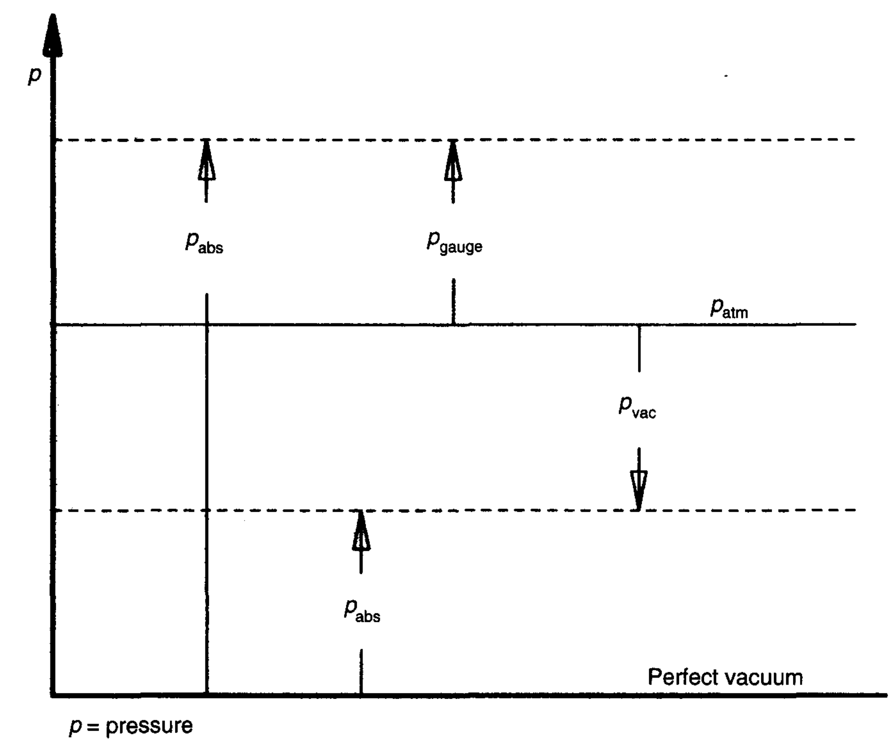


Figure 1-3. Absolute pressure, gauge pressure, and vacuum

For negative pressures, the absolute pressure is the atmospheric pressure minus the vacuum:

$$p_{abs} = p_{atm} - p_{vac}$$

These relationships are shown graphically in figure 1-3. Note that standard atmospheric pressure at sea level is 14.696 psia or 101.325 kPa.

SPECIFIC VOLUME AND DENSITY

Specific volume is defined as the volume of a substance divided by its mass:

$$\upsilon = \frac{V}{m} \ \ \mathrm{m^3/kg} \ \ \left(\mathrm{ft^3/lbm}\right)$$

The inverse of the specific volume is the density, defined as the mass divided by the volume:

$$\rho = \frac{m}{V} \ \ \mathrm{kg/m^3} \ \ \left(\mathrm{lbm/ft^3}\right)$$

CONVERSION FACTORS

A table of factors for converting units between the English and SI systems and some useful constants are provided as table 1.1.

TABLE 1-1

Factors for Converting between SI and English Systems

Area *(A)*	1 ft^2 = 0.0929 m^2 1 in^2 = 645.16 mm^2
Density (ρ)	1 lbm/ft^3 = 16.018 kg/m^3 1 slug/ft^3 = 515.379 kg/m^3
Energy *(E, H, U, PE, KE)*, Heat *(Q)*, Work *(W)*	1 Btu = 1.01551 kJ 1 IT cal = 4.1868 J 1Btu = 778.169 ft-lbf
Entropy *(S)*	1 Btu/R = 1.8992 kJ/K
Force *(F)*	1 lbf = 4.4482 N 1 dyne = 1 × 10^{-5} N
Heat transfer coefficient ($\bar{h}_c$)	1 Btu/hr = 5.678 W/m^2-K
Length *(L)*	1 ft = 0.3048 m 1 m = 39.37 in 1 in = 2.54 cm = 0.0254 m 1 mile = 1.6093 km = 5,280 ft
Mass *(m)*	1 kg = 2.20462 lbm 1 tonne = 1,000 kg 1 slug = 32.1739 lbm 1 short ton = 2,000 lbm
Power ($\dot{W}$), Heat flow rate ($\dot{Q}$)	1 Btu/hr = 0.2931 W 1 hp = 745.7 W 1 hp = 2,544.5 Btu/hr 1 Btu/sec = 1.0551 kW 1 hp = 550 ft-lbf/s 1 kW = 1.341 hp
Pressure *(p)*	1 psi = 6.8948 kPa 1 inHg = 0.4912 psi 1 bar = 100 kPa 1 mmHg = 0.1333 kPa 1 atm = 101.325 kPa = 14.696 psi = 760 mmHg = 29.92 inHg
Specific energy *(q, e, h, u, pe, ke)*, Specific work *(w)*	1 Btu/lbm = 2.3261 kJ/kg
Specific entropy *(s)*, Specific heat (c_p, c_υ), Gas Constant *(R)*	1 Btu/lbm-R = 4.1868 kJ/kg-K 1 Btu/lbm-R = 778.169 ft-lbf/lbm-R
Specific volume (υ)	1 ft^3/lbm = 0.062428 m^3/kg
Temperature *(T)*	*T*[R] = 1.8 *T*[K] *T*[C] = 5/9 (*T*[F] – 32) T[C] = *T*[K] – 273.15 *T*[F] = 9/5 *T*[C] + 32 *T*[F] = *T*[R] – 459.67
Thermal conductivity *(k)*	1 Btu/hr-ft-F = 1.731 W/m-K
Thermal diffusivity (α)	1 ft^2/s = 0.0929 m^2/s 1 ft^2/s = 2.581 × 10^{-5} m^2/s
Velocity *(V)*	1 ft/s = 0.3048 m/s 1 mph = 0.44703 m/s
Viscosity, dynamic (μ)	1 lbm/ft-s = 1.488 N-s/m^2 1 centipoise = 0.001 N-s/m^2
Viscosity, kinematic (υ)	1 ft^2/s = 0.0929 m^2/s 1 ft^2/s = 2.581 × 10^{-5} m^2/s
Volume *(V)*	1 m^3 = 35.3147 ft^3 1 m^3 = 1,000 l 1 ft^3 = 7.481 US gallons 1 US gallon = 3.7853 l

First Law of Thermodynamics

There are two parts to the first law of thermodynamics. The law of conservation of mass states that the total mass is a constant. The law of conservation of energy states that energy can be neither created nor destroyed. These are sometimes described as "bookkeeping" laws since their use typically involves accounting for the mass and energy going into or out of a system. The first law does not indicate whether an exchange or conversion is possible. That is left to the second law.

CONSERVATION OF MASS: THE CONTINUITY EQUATION

For an open system, the change in mass of the system is equal to the difference between the mass entering and the mass leaving. In the case of a steady flow, the change in mass is zero, and the flows of mass into and out of the system are identical:

$$\dot{m}_{\text{in}} = \dot{m}_{\text{out}}$$

A useful equation in determining the mass flow in many practical situations is the continuity equation. It expresses the mass flow in terms of cross-sectional area A, velocity V, and density ρ, or specific volume v, all measurable quantities:

$$\dot{m} = \rho A V = \frac{AV}{v}$$

For a steady-flow open system where the fluid enters at state "i" and exits at state "o," the conservation of mass equation becomes

$$\rho_i A_i V_i = \rho_o A_o V_o$$

EXAMPLE 1-1: Water with a density of 62.4 lbm/ft^3 is flowing through a 2-inch inside-diameter pipe at a velocity of 5 ft/sec. What is the rate of flow in (a) lbm/sec and (b) gallons/minute (gpm)?

Solution: Calculate the pipe cross-sectional area.

$$A = \pi D^2/4 = \frac{(3.1416)(2\text{ in})^2}{(4)\left(144\text{ in}^2/\text{ft}^2\right)} = 0.02182\text{ ft}^2$$

Use the continuity equation.

$$\dot{m} = \rho A V = \left(62.4\text{ lbm/ft}^3\right)\left(0.02182\text{ ft}^2\right)(5\text{ ft/sec})$$

(a) $$\dot{m} = 6.8068\text{ lbm/sec}$$

Convert to gallons per minute (gpm).

$$\text{gpm} = \frac{(6.8068\text{ lbm/sec})(60\text{ sec/min})\left(7.481\text{ gal/ft}^3\right)}{\left(62.4\text{ lbm/ft}^3\right)}$$

(b) $$\text{gpm} = 48.96$$

CONSERVATION OF ENERGY: ENERGY FORMS

In order to apply the law of conservation of energy, it is first necessary to understand some of the forms in which energy can exist. We will discuss only those forms most commonly of interest to engineers studying power plants.

The first energy form to be defined is work. Work is defined as the product of a force F acting through a distance x:

$$W = Fx$$

For example, if a crane lifts a pallet weighing 2,000 lbf vertically 30 feet, 2,000 x 30 = 60,000 ft-lbf of work would be done. Note that if the crane applied only 1,900 lbf, the pallet would not move and no work would be done.

The next energy form to be discussed is heat. In thermodynamics, heat is defined as the energy crossing a system boundary due to a temperature difference between the system and its surroundings. Heat and work are similar in that they have meaning only when they cross a system boundary. A container can be filled with a very hot liquid, but we would not use the term *heat* to describe that energy. *Heat* would only be used to describe the energy transferred from the container to the surroundings by conduction, convection, and radiation. This thermodynamic definition is somewhat different from that in common usage. Heat is commonly given the symbol Q.

Potential energy (PE) depends on the position of a mass in a gravitational force field. First, an arbitrary reference elevation is decided on, say sea level. In order to raise a mass from the reference elevation, work must be done since a force (the weight of the mass) is acting through a distance (the distance above the reference elevation). If the distance above the reference is z, then potential energy is calculated as follows:

$$\begin{aligned} PE &= mgz \quad (\text{N-m}) \\ &= m\frac{g}{g_c}z \quad (\text{ft-lbf}) \end{aligned}$$

The kinetic energy *(KE)* of a mass is defined as the work necessary to increase its velocity from rest. For example, if a car accelerates to 55 mph from 40 mph along a flat highway, its kinetic energy has increased. Note that since there was no change in elevation there was no change in potential energy. Kinetic energy is calculated as follows:

$$KE = \frac{m}{2}V^2 \quad (\text{N-m})$$
$$= \frac{m}{2g_c}V^2 \quad (\text{ft-lbf})$$

One of the more difficult forms of energy to understand is internal energy. Internal energy is a measure of the internal activity of the molecules in the system. It includes such things as the vibrational and rotational movement of the molecules. The difficulty is that it is impossible to measure these molecular energies directly. They must be measured indirectly, by measuring such things as the temperature or phase of the fluid. A liquid has more internal energy when hot than when cold. Steam at 212°F has more internal energy than liquid water at the same temperature. Internal energy is usually given the symbol U.

Another energy term that must be considered is flow energy. This is the work done to push mass into or out of an open system. Flow energy is the product of pressure and volume:

$$W_{flow} = pV = pmu$$

In most open-system problems, both internal and flow energy terms appear together. It is convenient to define a new property called enthalpy, which is merely the sum of the two:

$$H = U + pV$$

or for a unit mass:

$$h = u + pu$$

Enthalpy is commonly tabulated for various fluids such as steam and refrigerants. There is a temptation to think of enthalpy as heat, which it is not. It is merely the sum of u and pu.

The following example illustrates the use of many of the energy terms just defined and their application to the first law of thermodynamics.

EXAMPLE 2: A steam turbine is supplied with 5,000 lbm/hr of steam at an enthalpy of 1,400 Btu/lbm and a velocity of 100 ftlsec. The steam exhausts at an enthalpy of 950 Btu/lbm and a velocity of 900 ft/sec. Determine the horsepower produced.

Solution: The first-law equation for the open system on a per-unit-of-mass basis is

$$h_1 + ke_1 = h_2 + ke_2 + w$$
$$w = (h_1 - h_2) + (ke_1 - ke_2)$$
$$h_1 - h_2 = 1{,}400 \text{ Btu/lbm} - 950 \text{ Btu/lbm} = 450 \text{ Btu/lbm}$$
$$ke_1 - ke_2 = \frac{(V_1^2 - V_2^2)}{2g_cJ}$$
$$= \frac{(100^2 - 900^2 \text{ ft}^2/\text{sec}^2)}{(2)(32.174 \text{ lbm-ft/lbf-sec}^2)(778.16 \text{ ft-lbf/Btu})}$$
$$= -15.97 \text{ Btu/lbm}$$
$$w = 450 - 15.97 = 434.03 \text{ Btu/lbm}$$
$$\dot{W} = w\dot{m} = \frac{(5{,}000 \text{ lbm/hr})(434.03 \text{ Btu/lbm})}{(2{,}544.5 \text{ Btu/hp-hr})}$$
$$= 852.9 \text{ hp}$$

Properties of Pure Substances

Most engineering systems employ a working fluid of some kind to transport energy from one location to another. If the fluid has a constant composition, regardless of phase, it is referred to as a pure substance. The most common pure substance used as working fluid in engineering systems is water. The water can exist as a liquid, a vapor (steam), or a solid (ice). In all situations it is still H20.

In addition to the three phases, a pure substance like water can exist as a mixture of two phases. A glass of ice water has H20 in both the liquid and solid phases. Imagine starting with 1 pound of ice at OaF and standard atmospheric pressure (14.696 psia, or 101.325 kPa). Let's add heat to the ice. Initially, as heat is added, the temperature of the ice increases. At 32°F (O°C) something different happens-the ice begins to melt. Energy is added but the temperature stays constant as the water undergoes a phase change from solid to liquid. It takes 144 Btu of energy to completely melt the 1 pound of ice.

As further heat is added to the 32°F (O°C)liquid, the temperature begins to rise again, until a temperature of 212°F (lOO°C)is achieved. As further heat is added, a second phase change takes place. The water boils, again undergoing a phase change at constant temperature. Mter the addition of another 970 Btu, the water has been completely turned to vapor at 212°F-"saturated steam." If we now add more energy to the steam, its temperature will begin to rise again, resulting in "superheated steam."

It is important to note that the water in the example above would have boiled at a different temperature if the pressure had not been 14.696 psia

(101.325 kPa). If the pressure had been 1 psia, it would have boiled at 101.6°F. If the pressure had been 100 psia, the water would have boiled at 327.8°F. This relationship between saturation temperature and pressure is very important and forms the basis for the operation of many systems. It allows us to make distilled water from seawater in a flash evaporator and boil refrigerant-12 in the evaporator of a refrigeration system and keep meat frozen.

STEAM TABLES

Tables 1-2, 1-3, and 1-4 provide the properties of saturated and superheated steam. Table 1-2 presents data for saturated vapor and liquid with saturation temperature as the input variable. Table 1-3 presents the saturated vapor and liquid data with the saturation pressure as the input variable. Note that properties with the subscript "f" are for saturated liquid, properties with the subscript "g" are for saturated vapor, and those with the subscript "fg" are for the difference between the two. For example, h_{fg} is the difference between the enthalpy of saturated vapor and saturated liquid at the tabulated temperature or pressure. Table 1-4 presents data for superheated steam and compressed water with pressure and temperature as the inputs.

In order to determine the properties of a mixture of saturated vapor and liquid, the quality x must be used. Quality is the ratio of mass of vapor to the total mass of the mixture. For example, the enthalpy of a mixture can be determined as follows:

$$h = h_f + xh_{fg}$$

Similar equations can be written for other properties:

$$v = v_f + xv_{fg}$$

$$s = s_f + xs_{fg}$$

$$u = u_f + xu_{fg}$$

EXAMPLE 1-3: Steam is expanded in a turbine from 900 psia and 900°F to 1 psia. The exhaust has 10 percent moisture. Determine the energy extracted from each pound of steam in Btu.

Solution: Using table 1.4 for 900 psia and 900°F,

$$h = 1{,}452.2 \text{ Btujlbm}$$

Using table 1.3 for 1 psia and $x = 0.9$

$$h_f = 69.73 \text{ Btu/lbm}, \quad h_g = 1{,}036.1 \text{ Btujlbm},$$

$$h = 69.73 + (0.9)(1{,}036.1) = 1{,}002.2 \text{ Btujlbm}$$

The change in enthalpy across the turbine is thus

$$\Delta h = 1{,}452.2 - 1{,}002.2 = 450.0 \text{ Btu/lbm}$$

TABLE 1-2

Properties of Saturated Steam and Saturated Water (Temperature)

Temp F	Press. psia	Volume, ft³/lb			Enthalpy, Btu/lb			Entropy, Btu/lb, F			Temp F
		Water v_f	Evap v_{fg}	Steam v_g	Water h_f	Evap h_{fg}	Steam h_g	Water s_f	Evap s_{fg}	Steam s_g	
32	0.08859	0.01602	3305	3305	−0.02	1075.5	1075.5	0.0000	2.1873	2.1873	**32**
35	0.09991	0.01602	2948	2948	3.00	1073.8	1076.8	0.0061	2.1706	2.1767	**35**
40	0.12163	0.01602	2446	2446	8.03	1071.0	1079.0	0.0162	2.1432	2.1594	**40**
45	0.14744	0.01602	2037.7	2037.8	13.04	1068.1	1081.2	0.0262	2.1164	2.1426	**45**
50	0.17796	0.01602	1704.8	1704.8	18.05	1065.3	1083.4	0.0361	2.0901	2.1262	**50**
60	0.2561	0.01603	1207.6	1207.6	28.06	1059.7	1087.7	0.0555	2.0391	2.0946	**60**
70	0.3629	0.01605	868.3	868.4	38.05	1054.0	1092.1	0.0745	1.9900	2.0645	**70**
80	0.5068	0.01607	633.3	633.3	48.04	1048.4	1096.4	0.0932	1.9426	2.0359	**80**
90	0.6981	0.01610	468.1	468.1	58.02	1042.7	1100.8	0.1115	1.8970	2.0086	**90**
100	0.9492	0.01613	350.4	350.4	68.00	1037.1	1105.1	0.1295	1.8530	1.9825	**100**
110	1.2750	0.01617	265.4	265.4	77.98	1031.4	1109.3	0.1472	1.8105	1.9577	**110**
120	1.6927	0.01620	203.25	203.26	87.97	1025.6	1113.6	0.1646	1.7693	1.9339	**120**
130	2.2230	0.01625	157.32	157.33	97.96	1019.8	1117.8	0.1817	1.7295	1.9112	**130**
140	2.8892	0.01629	122.98	123.00	107.95	1014.0	1122.0	0.1985	1.6910	1.8895	**140**
150	3.718	0.01634	97.05	97.07	117.95	1008.2	1126.1	0.2150	1.6536	1.8686	**150**
160	4.741	0.01640	77.27	77.29	127.96	1002.2	1130.2	0.2313	1.6174	1.8487	**160**
170	5.993	0.01645	62.04	62.06	137.97	996.2	1134.2	0.2473	1.5822	1.8295	**170**
180	7.511	0.01651	50.21	50.22	148.00	990.2	1138.2	0.2631	1.5480	1.8111	**180**
190	9.340	0.01657	40.94	40.96	158.04	984.1	1142.1	0.2787	1.5148	1.7934	**190**
200	11.526	0.01664	33.62	33.64	168.09	977.9	1146.0	0.2940	1.4824	1.7764	**200**
210	14.123	0.01671	27.80	27.82	178.15	971.6	1149.7	0.3091	1.4509	1.7600	**210**
212	14.696	0.01672	26.78	26.80	180.17	970.3	1150.5	0.3121	1.4447	1.7568	**212**
220	17.186	0.01678	23.13	23.15	188.23	965.2	1153.4	0.3241	1.4201	1.7442	**220**
230	20.779	0.01685	19.364	19.381	198.33	958.7	1157.1	0.3388	1.3902	1.7290	**230**
240	24.968	0.01693	16.304	16.321	208.45	952.1	1160.6	0.3533	1.3609	1.7142	**240**
250	29.825	0.01701	13.802	13.819	218.59	945.4	1164.0	0.3677	1.3323	1.7000	**250**
260	35.427	0.01709	11.745	11.762	228.76	938.6	1167.4	0.3819	1.3043	1.6862	**260**
270	41.856	0.01718	10.042	10.060	238.95	931.7	1170.6	0.3960	1.2769	1.6729	**270**
280	49.200	0.01726	8.627	8.644	249.17	924.6	1173.8	0.4098	1.2501	1.6599	**280**
290	57.550	0.01736	7.443	7.460	259.4	917.4	1176.8	0.4236	1.2238	1.6473	**290**
300	67.005	0.01745	6.448	6.466	269.7	910.0	1179.7	0.4372	1.1979	1.6351	**300**
310	77.67	0.01755	5.609	5.626	280.0	902.5	1182.5	0.4506	1.1726	1.6232	**310**
320	89.64	0.01766	4.896	4.914	290.4	894.8	1185.2	0.4640	1.1477	1.6116	**320**
340	117.99	0.01787	3.770	3.788	311.3	878.8	1190.1	0.4902	1.0990	1.5892	**340**
360	153.01	0.01811	2.939	2.957	332.3	862.1	1194.4	0.5161	1.0517	1.5678	**360**
380	195.73	0.01836	2.317	2.335	353.6	844.5	1198.0	0.5416	1.0057	1.5473	**380**
400	247.26	0.01864	1.8444	1.8630	375.1	825.9	1201.0	0.5667	0.9607	1.5274	**400**
420	308.78	0.01894	1.4808	1.4997	396.9	806.2	1203.1	0.5915	0.9165	1.5080	**420**
440	381.54	0.01926	1.1976	1.2169	419.0	785.4	1204.4	0.6161	0.8729	1.4890	**440**
460	466.9	0.0196	0.9746	0.9942	441.5	763.2	1204.8	0.6405	0.8299	1.4704	**460**
480	566.2	0.0200	0.7972	0.8172	464.5	739.6	1204.1	0.6648	0.7871	1.4518	**480**
500	680.9	0.0204	0.6545	0.6749	487.9	714.3	1202.2	0.6890	0.7443	1.4333	**500**
520	812.5	0.0209	0.5386	0.5596	512.0	687.0	1199.0	0.7133	0.7013	1.4146	**520**
540	962.8	0.0215	0.4437	0.4651	536.8	657.5	1194.3	0.7378	0.6577	1.3954	**540**
560	1133.4	0.0221	0.3651	0.3871	562.4	625.3	1187.7	0.7625	0.6132	1.3757	**560**
580	1326.2	0.0228	0.2994	0.3222	589.1	589.9	1179.0	0.7876	0.5673	1.3550	**580**
600	1543.2	0.0236	0.2438	0.2675	617.1	550.6	1167.7	0.8134	0.5196	1.3330	**600**
620	1786.9	0.0247	0.1962	0.2208	646.9	506.3	1153.2	0.8403	0.4689	1.3092	**620**
640	2059.9	0.0260	0.1543	0.1802	679.1	454.6	1133.7	0.8686	0.4134	1.2821	**640**
660	2365.7	0.0277	0.1166	0.1443	714.9	392.1	1107.0	0.8995	0.3502	1.2498	**660**
680	2708.6	0.0304	0.0808	0.1112	758.5	310.1	1068.5	0.9365	0.2720	1.2086	**680**
700	3094.3	0.0366	0.0386	0.0752	822.4	172.7	995.2	0.9901	0.1490	1.1390	**700**
705.5	3208.2	0.0508	0	0.0508	906.0	0	906.0	1.0612	0	1.0612	**705.5**

TABLE 1-3

Properties of Saturated Steam and Saturated Water (Pressure)

Press. psia	Temp F	Volume, ft³/lb Water v_f	Evap v_{fg}	Steam v_g	Enthalpy, Btu/lb Water h_f	Evap h_{fg}	Steam h_g	Entropy, Btu/lb, F Water s_f	Evap s_{fg}	Steam s_g	Energy, Btu/lb Water u_f	Steam u_g	Press. psia
0.0886	32.018	0.01602	3302.4	3302.4	0.00	1075.5	1075.5	0	2.1872	2.1872	0	1021.3	**0.0886**
0.10	35.023	0.01602	2945.5	2945.5	3.03	1073.8	1076.8	0.0061	2.1705	2.1766	3.03	1022.3	**0.10**
0.15	45.453	0.01602	2004.7	2004.7	13.50	1067.9	1081.4	0.0271	2.1140	2.1411	13.50	1025.7	**0.15**
0.20	53.160	0.01603	1526.3	1526.3	21.22	1063.5	1084.7	0.0422	2.0738	2.1160	21.22	1028.3	**0.20**
0.30	64.484	0.01604	1039.7	1039.7	32.54	1057.1	1089.7	0.0641	2.0168	2.0809	32.54	1032.0	**0.30**
0.40	72.869	0.01606	792.0	792.1	40.92	1052.4	1093.3	0.0799	1.9762	2.0562	40.92	1034.7	**0.40**
0.5	79.586	0.01607	641.5	641.5	47.62	1048.6	1096.3	0.0925	1.9446	2.0370	47.62	1036.9	**0.5**
0.6	85.218	0.01609	540.0	540.1	53.25	1045.5	1098.7	0.1028	1.9186	2.0215	53.24	1038.7	**0.6**
0.7	90.09	0.01610	466.93	466.94	58.10	1042.7	1100.8	0.3	1.8966	2.0083	58.10	1040.3	**0.7**
0.8	94.38	0.01611	411.67	411.69	62.39	1040.3	1102.6	0.1117	1.8775	1.9970	62.39	1041.7	**0.8**
0.9	98.24	0.01612	368.41	368.43	66.24	1038.1	1104.3	0.1264	1.8606	1.9870	66.24	1042.9	**0.9**
1.0	101.74	0.01614	333.59	333.60	69.73	1036.1	1105.8	0.1326	1.8455	1.9781	69.73	1044.1	**1.0**
2.0	126.07	0.01623	173.74	173.76	94.03	1022.1	1116.2	0.1750	1.7450	1.9200	94.03	1051.8	**2.0**
3.0	141.47	0.01630	118.71	118.73	109.42	1013.2	1122.6	0.2009	1.6854	1.8864	109.41	1056.7	**3.0**
4.0	152.96	0.01636	90.63	90.64	120.92	1006.4	1127.3	0.2199	1.6428	1.8626	120.90	1060.2	**4.0**
5.0	162.24	0.01641	73.515	73.53	130.20	1000.9	1131.1	0.2349	1.6094	1.8443	130.18	1063.1	**5.0**
6.0	170.05	0.01645	61.967	61.98	138.03	996.2	1134.2	0.2474	1.5820	1.8294	138.01	1065.4	**6.0**
7.0	176.84	0.01649	53.634	53.65	144.83	992.1	1136.9	0.2581	1.5587	1.8168	144.81	1067.4	**7.0**
8.0	182.86	0.01653	47.328	47.35	150.87	988.5	1139.3	0.2676	1.5384	1.8060	150.84	1069.2	**8.0**
9.0	188.27	0.01656	42.385	42.40	156.30	985.1	1141.4	0.2760	1.5204	1.7964	156.28	1070.8	**9.0**
10	193.21	0.01659	38.404	38.42	161.26	982.1	1143.3	0.2836	1.5043	1.7879	161.23	1072.3	**10**
14.696	212.00	0.01672	26.782	26.80	180.17	970.3	1150.5	0.3121	1.4447	1.7568	180.12	1077.6	**14.696**
15	213.03	0.01673	26.274	26.29	181.21	969.7	1150.9	0.3137	1.4415	1.7552	181.16	1077.9	**15**
20	227.96	0.01683	20.070	20.087	196.27	960.1	1156.3	0.3358	1.3962	1.7320	196.21	1082.0	**20**
30	250.34	0.01701	13.7266	13.744	218.9	945.2	1164.1	0.3682	1.3313	1.6995	218.8	1087.9	**30**
40	267.25	0.01715	10.4794	10.497	236.1	933.6	1169.8	0.3921	1.2844	1.6765	236.0	1092.1	**40**
50	281.02	0.01727	8.4967	8.514	250.2	923.9	1174.1	0.4112	1.2474	1.6586	250.1	1095.3	**50**
60	292.71	0.01738	7.1562	7.174	262.2	915.4	1177.6	0.4273	1.2167	1.6440	262.0	1098.0	**60**
70	302.93	0.01748	6.1875	6.205	272.7	907.8	1180.6	0.4411	1.1905	1.6316	272.5	1100.2	**70**
80	312.04	0.01757	5.4536	5.471	282.1	900.9	1183.1	0.4534	1.1675	1.6208	281.9	1102.1	**80**
90	320.28	0.01766	4.8777	4.895	290.7	894.6	1185.3	0.4643	1.1470	1.6113	290.4	1103.7	**90**
100	327.82	0.01774	4.4133	4.431	298.5	888.6	1187.2	0.4743	1.1284	1.6027	298.2	1105.2	**100**
120	341.27	0.01789	3.7097	3.728	312.6	877.8	1190.4	0.4919	1.0960	1.5879	312.2	1107.6	**120**
140	353.04	0.01803	3.2010	3.219	325.0	868.0	1193.0	0.5071	1.0681	1.5752	324.5	1109.6	**140**
160	363.55	0.01815	2.8155	2.834	336.1	859.0	1195.1	0.5206	1.0435	1.5641	335.5	1111.2	**160**
180	373.08	0.01827	2.5129	2.531	346.2	850.7	1196.9	0.5328	1.0215	1.5543	345.6	1112.5	**180**
200	381.80	0.01839	2.2689	2.287	355.5	842.8	1198.3	0.5438	1.0016	1.5454	354.8	1113.7	**200**
250	400.97	0.01865	1.8245	1.8432	376.1	825.0	1201.1	0.5679	0.9585	1.5264	375.3	1115.8	**250**
300	417.35	0.01889	1.5238	1.5427	394.0	808.9	1202.9	0.5882	0.9223	1.5105	392.9	1117.2	**300**
350	431.73	0.01913	1.3064	1.3255	409.8	794.2	1204.0	0.6059	0.8909	1.4968	408.6	1118.1	**350**
400	444.60	0.0193	1.14162	1.1610	424.2	780.4	1204.6	0.6217	0.8630	1.4847	422.7	1118.7	**400**
450	456.28	0.0195	1.01224	1.0318	437.3	767.5	1204.8	0.6360	0.8378	1.4738	435.7	1118.9	**450**
500	467.01	0.0198	0.90787	0.9276	449.5	755.1	1204.7	0.6490	0.8148	1.4639	447.7	1118.8	**500**
550	476.94	0.0199	0.82183	0.8418	460.9	743.3	1204.3	0.6611	0.7936	1.4547	458.9	1118.6	**550**
600	486.20	0.0201	0.74962	0.7698	471.7	732.0	1203.7	0.6723	0.7738	1.4461	469.5	1118.2	**600**
700	503.08	0.0205	0.63505	0.6556	491.6	710.2	1201.8	0.6928	0.7377	1.4304	488.9	1116.9	**700**
800	518.21	0.0209	0.54809	0.5690	509.8	689.6	1199.4	0.7111	0.7051	1.4163	506.7	1115.2	**800**
900	531.95	0.0212	0.47968	0.5009	526.7	669.7	1196.4	0.7279	0.6753	1.4032	523.2	1113.0	**900**
1000	544.58	0.0216	0.42436	0.4460	542.6	650.4	1192.9	0.7434	0.6476	1.3910	538.6	1110.4	**1000**
1100	556.28	0.0220	0.37863	0.4006	557.5	631.5	1189.1	0.7578	0.6216	1.3794	553.1	1107.5	**1100**
1200	567.19	0.0223	0.34013	0.3625	571.9	613.0	1184.8	0.7714	0.5969	1.3683	566.9	1104.3	**1200**
1300	577.42	0.0227	0.30722	0.3299	585.6	594.6	1180.2	0.7843	0.5733	1.3577	580.1	1100.9	**1300**
1400	587.07	0.0231	0.27871	0.3018	598.8	576.5	1175.3	0.7966	0.5507	1.3474	592.9	1097.1	**1400**
1500	596.20	0.0235	0.25372	0.2772	611.7	558.4	1170.1	0.8085	0.5288	1.3373	605.2	1093.1	**1500**
2000	635.80	0.0257	0.16266	0.1883	672.1	466.2	1138.3	0.8625	0.4256	1.2881	662.6	1068.6	**2000**
2500	668.11	0.0286	0.10209	0.1307	731.7	361.6	1093.3	0.9139	0.3206	1.2345	718.5	1032.9	**2500**
3000	695.33	0.0343	0.05073	0.0850	801.8	218.4	1020.3	0.9728	0.1891	1.1619	782.8	973.1	**3000**
3208.2	705.47	0.0508	0	0.0508	906.0	0	906.0	1.0612	0	1.0612	875.9	875.9	**3208.2**

TABLE 1-4A

Properties of Superheated Steam and Compressed Water (Temperature and Pressure)

Abs press. lb/sq in. (sat. temp)		Temperature, F 100	200	300	400	500	600	700	800	900	1000	1100	1200	1300	1400	1500
	v	0.0161	392.5	452.3	511.9	571.5	631.1	690.7								
1	h	68.00	1150.2	1195.7	1241.8	1288.6	1336.1	1384.5								
(101.74)	s	0.1295	2.0509	2.1152	2.1722	2.2237	2.2708	2.3144								
	v	0.0161	78.14	90.24	102.24	114.21	126.15	138.08	150.01	161.94	173.86	185.78	197.70	209.62	221.53	233.45
5	h	68.01	1148.6	1194.8	1241.3	1288.2	1335.9	1384.3	1433.6	1483.7	1534.7	1586.7	1639.6	1693.3	1748.0	1803.5
(162.24)	s	0.1295	1.8716	1.9369	1.9943	2.0460	2.0932	2.1369	2.1776	2.2159	2.2521	2.2866	2.3194	2.3509	2.3811	2.4101
	v	0.0161	38.84	44.98	51.03	57.04	63.03	69.00	74.98	80.94	86.91	92.87	98.84	104.80	110.76	116.72
10	h	68.02	1146.6	1193.7	1240.6	1287.8	1335.5	1384.0	1433.4	1483.5	1534.6	1586.6	1639.5	1693.3	1747.9	1803.4
(193.21)	s	0.1295	1.7928	1.8593	1.9173	1.9692	2.0166	2.0603	2.1011	2.1394	2.1757	2.2101	2.2430	2.2744	2.3046	2.3337
	v	0.0161	0.0166	29.899	33.963	37.985	41.986	45.978	49.964	53.946	57.926	61.905	65.882	69.858	73.833	77.807
15	h	68.04	168.09	1192.5	1239.9	1287.3	1335.2	1383.8	1433.2	1483.4	1534.5	1586.5	1639.4	1693.2	1747.8	1803.4
(213.03)	s	0.1295	0.2940	1.8134	1.8720	1.9242	1.9717	2.0155	2.0563	2.0946	2.1309	2.1653	2.1982	2.2297	2.2599	2.2890
	v	0.0161	0.0166	22.356	25.428	28.457	31.466	34.465	37.458	40.447	43.435	46.420	49.405	52.388	55.370	58.352
20	h	68.05	168.11	1191.4	1239.2	1286.9	1334.9	1383.5	1432.9	1483.2	1534.3	1586.3	1639.3	1693.1	1747.8	1803.3
(227.96)	s	0.1295	0.2940	1.7805	1.8397	1.8921	1.9397	1.9836	2.0244	2.0628	2.0991	2.1336	2.1665	2.1979	2.2282	2.2572
	v	0.0161	0.0166	11.036	12.624	14.165	15.685	17.195	18.699	20.199	21.697	23.194	24.689	26.183	27.676	29.168
40	h	68.10	168.15	1186.6	1236.4	1285.0	1333.6	1382.5	1432.1	1482.5	1533.7	1585.8	1638.8	1992.7	1747.5	1803.0
(267.25)	s	0.1295	0.2940	1.6992	1.7608	1.8143	1.8624	1.9065	1.9476	1.9860	2.0224	2.0569	2.0899	2.1224	2.1516	2.1807
	v	0.0161	0.0166	7.257	8.354	9.400	10.425	11.438	12.446	13.450	14.452	15.452	16.450	17.448	18.445	19.441
60	h	68.15	168.20	1181.6	1233.5	1283.2	1332.3	1381.5	1431.3	1481.8	1533.2	1585.3	1638.4	1692.4	1747.1	1802.8
(292.71)	s	0.1295	0.2939	1.6492	1.7134	1.7681	1.8168	1.8612	1.9024	1.9410	1.9774	2.0120	2.0450	2.0765	2.1068	2.1359
	v	0.0161	0.0166	0.0175	6.218	7.018	7.794	8.560	9.319	10.075	10.829	11.581	12.331	13.081	13.829	14.577
80	h	68.21	168.24	269.74	1230.5	1281.3	1330.9	1380.5	1430.5	1481.1	1532.6	1584.9	1638.0	1692.0	1746.8	1802.5
(312.04)	s	0.1295	0.2939	0.4371	1.6790	1.7349	1.7842	1.8289	1.8702	1.9089	1.9454	1.9800	2.0131	2.0446	2.0750	2.1041
	v	0.0161	0.0166	0.0175	4.935	5.588	6.216	6.833	7.443	8.050	8.655	9.258	9.860	10.460	11.060	11.659
100	h	68.26	168.29	269.77	1227.4	1279.3	1329.6	1379.5	1429.7	1480.4	1532.0	1584.4	1637.6	1691.6	1746.5	1802.2
(327.82)	s	0.1295	0.2939	0.4371	1.6516	1.7088	1.7586	1.8036	1.8451	1.8839	1.9205	1.9552	1.9883	2.0199	2.0502	2.0794
	v	0.0161	0.0166	0.0175	4.0786	4.6341	5.1637	5.6831	6.1928	6.7006	7.2060	7.7096	8.2119	8.7130	9.2134	9.7130
120	h	68.31	168.33	269.81	1224.1	1277.4	1328.1	1378.4	1428.8	1479.8	1531.4	1583.9	1637.1	1691.3	1746.2	1802.0
(341.27)	s	0.1295	0.2939	0.4371	1.6286	1.6872	1.7376	1.7829	1.8246	1.8635	1.9001	1.9349	1.9680	1.9996	2.0300	2.0592
	v	0.0161	0.0166	0.0175	3.4661	3.9526	4.4119	4.8585	5.2995	5.7364	6.1709	6.6036	7.0349	7.4652	7.8946	8.3233
140	h	68.37	168.38	269.85	1220.8	1275.3	1326.8	1377.4	1428.0	1479.1	1530.8	1583.4	1636.7	1690.9	1745.9	1801.7
(353.04)	s	0.1295	0.2939	0.4370	1.6085	1.6686	1.7196	1.7652	1.8071	1.8461	1.8828	1.9176	1.9508	1.9825	2.0129	2.0421
	v	0.0161	0.0166	0.0175	3.0060	3.4413	3.8480	4.2420	4.6295	5.0132	5.3945	5.7741	6.1522	6.5293	6.9055	7.2811
160	h	68.42	168.42	269.89	1217.4	1273.3	1325.4	1376.4	1427.2	1478.4	1530.3	1582.9	1636.3	1690.5	1745.6	1801.4
(363.55)	s	0.1294	0.2938	0.4370	1.5906	1.6522	1.7039	1.7499	1.7919	1.8310	1.8678	1.9027	1.9359	1.9676	1.9980	2.0273
	v	0.0161	0.0166	0.0174	2.6474	3.0433	3.4093	3.7621	4.1084	4.4505	4.7907	5.1289	5.4657	5.8014	6.1363	6.4704
180	h	68.47	168.47	269.92	1213.8	1271.2	1324.0	1375.3	1426.3	1477.7	1529.7	1582.4	1635.9	1690.2	1745.3	1801.2
(373.08)	s	0.1294	0.2938	0.4370	1.5743	1.6376	1.6900	1.7362	1.7784	1.8176	1.8545	1.8894	1.9227	1.9545	1.9849	2.0142
	v	0.0161	0.0166	0.0174	2.3598	2.7247	3.0583	3.3783	3.6915	4.0008	4.3077	4.6128	4.9165	5.2191	5.5209	5.8219
200	h	68.52	168.51	269.96	1210.1	1269.0	1322.6	1374.3	1425.5	1477.0	1529.1	1581.9	1635.4	1689.8	1745.0	1800.9
(381.80)	s	0.1294	0.2938	0.4369	1.5593	1.6242	1.6776	1.7239	1.7663	1.8057	1.8426	1.8776	1.9109	1.9427	1.9732	2.0025
	v	0.0161	0.0166	0.0174	0.0186	2.1504	2.4662	2.6872	2.9410	3.1909	3.4382	3.6837	3.9278	4.1709	4.4131	4.6546
250	h	68.66	168.63	270.05	375.10	1263.5	1319.0	1371.6	1423.4	1475.3	1527.6	1580.6	1634.4	1688.9	1744.2	1800.2
(400.97)	s	0.1294	0.2937	0.4368	0.5667	1.5951	1.6502	1.6976	1.7405	1.7801	1.8173	1.8524	1.8858	1.9177	1.9482	1.9776
	v	0.0161	0.0166	0.0174	0.0186	1.7665	2.0044	2.2263	2.4407	2.6509	2.8585	3.0643	3.2688	3.4721	3.6746	3.8764
300	h	68.79	168.74	270.14	375.15	1257.7	1315.2	1368.9	1421.3	1473.6	1526.2	1579.4	1633.3	1688.0	1743.4	1799.6
(417.35)	s	0.1294	0.2937	0.4307	0.5665	1.5703	1.6274	1.6758	1.7192	1.7591	1.7964	1.8317	1.8652	1.8972	1.9278	1.9572
	v	0.0161	0.0166	0.0174	0.0186	1.4913	1.7028	1.8970	2.0832	2.2652	2.4445	2.6219	2.7980	2.9730	3.1471	3.3205
350	h	68.92	168.85	270.24	375.21	1251.5	1311.4	1366.2	1419.2	1471.8	1524.7	1578.2	1632.3	1687.1	1742.6	1798.9
(431.73)	s	0.1293	0.2936	0.4367	0.5664	1.5483	1.6077	1.6571	1.7009	1.7411	1.7787	1.8141	1.8477	1.8798	1.9105	1.9400
	v	0.0161	0.0166	0.0174	0.0162	1.2841	1.4763	1.6499	1.8151	1.9759	2.1339	2.2901	2.4450	2.5987	2.7515	2.9037
400	h	69.05	168.97	270.33	375.27	1245.1	1307.4	1363.4	1417.0	1470.1	1523.3	1576.9	1631.2	1686.2	1741.9	1798.2
(444.60)	s	0.1293	0.2935	0.4366	0.5663	1.5282	1.5901	1.6406	1.6850	1.7255	1.7632	1.7988	1.8325	1.8647	1.8955	1.9250
	v	0.0161	0.0166	0.0174	0.0186	0.9919	1.1584	1.3037	1.4397	1.5708	1.6992	1.8256	1.9507	2.0746	2.1977	2.3200
500	h	69.32	169.19	270.51	375.38	1231.2	1299.1	1357.7	1412.7	1466.6	1520.3	1574.4	1629.1	1684.4	1740.3	1796.9
(467.01)	s	0.1292	0.2934	0.4364	0.5660	1.4921	1.5595	1.6123	1.6578	1.6990	1.7371	1.7730	1.8069	1.8393	1.8702	1.8998

TABLE 1-4B

Properties of Superheated Steam and Compressed Water (Temperature and Pressure)

Abs press. lb/sq in. (sat. temp)		Temperature, F 100	200	300	400	500	600	700	800	900	1000	1100	1200	1300	1400	1500
	v	0.0161	0.0166	0.0174	0.0186	0.7944	0.9456	1.0726	1.1892	1.3008	1.4093	1.5160	1.6211	1.7252	1.8284	1.9309
600	h	69.58	169.42	270.70	375.49	1215.9	1290.3	1351.8	1408.3	1463.0	1517.4	1571.9	1627.0	1682.6	1738.8	1795.6
(486.20)	s	0.1292	0.2933	0.4362	0.5657	1.4590	1.5329	1.5844	1.6351	1.6769	1.7155	1.7517	1.7859	1.8184	1.8494	1.8792
	v	0.0161	0.0166	0.0174	0.0186	0.0204	0.7928	0.9072	1.0102	1.1078	1.2023	1.2948	1.3858	1.4757	1.5647	1.6530
700	h	69.84	169.65	270.89	375.61	487.93	1281.0	1345.6	1403.7	1459.4	1514.4	1569.4	1624.8	1680.7	1737.2	1794.3
(503.08)	s	0.1291	0.2932	0.4360	0.5655	0.6889	1.5090	1.5673	1.6154	1.6580	1.6970	1.7335	1.7679	1.8006	1.8318	1.8617
	v	0.0161	0.0166	0.0174	0.0186	0.0204	0.6774	0.7828	0.8759	0.9631	1.0470	1.1289	1.2093	1.2885	1.3669	1.4446
800	h	70.11	169.88	271.07	375.73	487.88	1271.1	1339.2	1399.1	1455.8	1511.4	1566.9	1622.7	1678.9	1735.0	1792.9
(518.21)	s	0.1290	0.2930	0.4358	0.5652	0.6885	1.4869	1.5484	1.5980	1.6413	1.6807	1.7175	1.7522	1.7851	1.8164	1.8464
	v	0.0161	0.0166	0.0174	0.0186	0.0204	0.5869	0.6858	0.7713	0.8504	0.9262	0.9998	1.0720	1.1430	1.2131	1.2825
900	h	70.37	170.10	271.26	375.84	487.83	1260.6	1332.7	1394.4	1452.2	1508.5	1564.4	1620.6	1677.1	1734.1	1791.6
(531.95)	s	0.1290	0.2929	0.4357	0.5649	0.6881	1.4659	1.5311	1.5822	1.6263	1.6662	1.7033	1.7382	1.7713	1.8028	1.8329
	v	0.0161	0.0166	0.0174	0.0186	0.0204	0.5137	0.6080	0.6875	0.7603	0.8295	0.8966	0.9622	1.0266	1.0901	1.1529
1000	h	70.63	170.33	271.44	375.96	487.79	1249.3	1325.9	1389.6	1448.5	1504.4	1561.9	1618.4	1675.3	1732.5	1790.3
(544.58)	s	0.1289	0.2928	0.4355	0.5647	0.6876	1.4457	1.5149	1.5677	1.6126	1.6530	1.6905	1.7256	1.7589	1.7905	1.8207
	v	0.0161	0.0166	0.0174	0.0185	0.0203	0.4531	0.5440	0.6188	0.6865	0.7505	0.8121	0.8723	0.9313	0.9894	1.0468
1100	h	70.90	170.56	271.63	376.08	487.75	1237.3	1318.8	1384.7	1444.7	1502.4	1559.4	1616.3	1673.5	1731.0	1789.0
(556.28)	s	0.1289	0.2927	0.4353	0.5644	0.6872	1.4259	1.4996	1.5542	1.6000	1.6410	1.6787	1.7141	1.7475	1.7793	1.8097
	v	0.0161	0.0166	0.0174	0.0185	0.0203	0.4016	0.4905	0.5615	0.6250	0.6845	0.7418	0.7974	0.8519	0.9055	0.9584
1200	h	71.16	170.78	271.82	376.20	487.72	1224.2	1311.5	1379.7	1440.9	1499.4	1556.9	1614.2	1671.6	1729.4	1787.6
(567.19)	s	0.1288	0.2926	0.4351	0.5642	0.6868	1.4061	1.4851	1.5415	1.5883	1.6298	1.6679	1.7035	1.7371	1.7691	1.7996
	v	0.0161	0.0166	0.0174	0.0185	0.0203	0.3176	0.4059	0.4712	0.5282	0.5809	0.6311	0.6798	0.7272	0.7737	0.8195
1400	h	71.68	171.24	272.19	376.44	487.65	1194.1	1296.1	1369.3	1433.2	1493.2	1551.8	1609.9	1668.0	1726.3	1785.0
(587.07)	s	0.1287	0.2923	0.4348	0.5636	0.6859	1.3652	1.4575	1.5182	1.5670	1.6096	1.6484	1.6845	1.7185	1.7508	1.7815
	v	0.0161	0.0166	0.0173	0.0185	0.0202	0.0236	0.3415	0.4032	0.4555	0.5031	0.5482	0.5915	0.6336	0.6748	0.7153
1600	h	72.21	171.69	272.57	376.69	487.60	616.77	1279.4	1358.5	1425.2	1486.9	1546.6	1605.6	1664.3	1723.2	1782.3
(604.87)	s	0.1286	0.2921	0.4344	0.5631	0.6851	0.8129	1.4312	1.4968	1.5478	1.5916	1.6312	1.6678	1.7022	1.7344	1.7657
	v	0.0160	0.0165	0.0173	0.0185	0.0202	0.0235	0.2906	0.3500	0.3988	0.4426	0.4836	0.5229	0.5609	0.5980	0.6343
1800	h	72.73	172.15	272.95	376.93	487.56	615.58	1261.1	1347.2	1417.1	1480.6	1541.1	1601.2	1660.7	1720.1	1779.7
(621.02)	s	0.1284	0.2918	0.4341	0.5626	0.6843	0.8109	1.4054	1.4768	1.5302	1.5753	1.6156	1.6528	1.6876	1.7204	1.7516
	v	0.0160	0.0165	0.0173	0.0184	0.0201	0.0233	0.2488	0.3072	0.3534	0.3942	0.4320	0.4680	0.5027	0.5365	0.5695
2000	h	73.26	172.60	273.32	377.19	487.53	614.48	1240.9	1353.4	1408.7	1474.1	1536.2	1596.9	1657.0	1717.0	1777.1
(635.80)	s	0.1283	0.2916	0.4337	0.5621	0.6834	0.8091	1.3794	1.4578	1.5138	1.5603	1.6014	1.6391	1.6743	1.7075	1.7389
	v	0.0160	0.0165	0.0173	0.0184	0.0200	0.0230	0.1681	0.2293	0.2712	0.3068	0.3390	0.3692	0.3980	0.4259	0.4529
2500	h	74.57	173.74	274.27	377.82	487.50	612.08	1176.7	1303.4	1386.7	1457.5	1522.9	1585.9	1647.8	1709.2	1770.4
(668.11)	s	0.1280	0.2910	0.4329	0.5609	0.6815	0.8048	1.3076	1.4129	1.4766	1.5269	1.5703	1.6094	1.6456	1.6796	1.7116
	v	0.0160	0.0165	0.0172	0.0183	0.0200	0.0228	0.0982	0.1759	0.2161	0.2484	0.2770	0.3033	0.3282	0.3522	0.3753
3000	h	75.88	174.88	275.22	378.47	487.52	610.08	1060.5	1267.0	1363.2	1440.2	1509.4	1574.8	1638.5	1701.4	1761.8
(695.33)	s	0.1277	0.2904	0.4320	0.5597	0.6796	0.8009	1.1966	1.3692	1.4429	1.4976	1.5434	1.5841	1.6214	1.6561	1.6888
	v	0.0160	0.0165	0.0172	0.0183	0.0199	0.0227	0.0335	0.1588	0.1987	0.2301	0.2576	0.2827	0.3065	0.3291	0.3510
3200	h	76.4	175.3	275.6	378.7	487.5	609.4	800.8	1250.9	1353.4	1433.1	1503.8	1570.3	1634.8	1698.3	1761.2
(705.08)	s	0.1276	0.2902	0.4317	0.5592	0.6788	0.7994	0.9708	1.3515	1.4300	1.4866	1.5335	1.5749	1.6126	1.6477	1.6806
	v	0.0160	0.0164	0.0172	0.0183	0.0199	0.0225	0.0307	0.1364	0.1764	0.2066	0.2326	0.2563	0.2784	0.2995	0.3198
3500	h	77.2	176.0	276.2	379.1	487.6	608.4	779.4	1224.6	1338.2	1422.2	1495.5	1563.3	1629.2	1693.6	1757.2
	s	0.1274	0.2899	0.4312	0.5585	0.6777	0.7973	0.9508	1.3242	1.4112	1.4709	1.5194	1.5618	1.6002	1.6358	1.6691
	v	0.0159	0.0164	0.0172	0.0182	0.0198	0.0223	0.0287	0.1052	0.1463	0.1752	0.1994	0.2210	0.2411	0.2601	0.2783
4000	h	78.5	177.2	277.1	379.8	487.7	606.9	763.0	1174.3	1311.6	1403.6	1481.3	1552.2	1619.8	1685.7	1750.6
	s	0.1271	0.2893	0.4304	0.5573	0.6760	0.7940	0.9343	1.2754	1.3807	1.4461	1.4976	1.5417	1.5812	1.6177	1.6516
	v	0.0159	0.0164	0.0171	0.0181	0.0196	0.0219	0.0268	0.0591	0.1038	0.1312	0.1529	0.1718	0.1890	0.2050	0.2203
5000	h	81.1	179.5	279.1	381.2	488.1	604.6	746.0	1042.9	1252.9	1364.6	1452.1	1529.1	1600.9	1670.0	1737.4
	s	0.1265	0.2881	0.4287	0.5550	0.6726	0.7880	0.9153	1.1593	1.3207	1.4001	1.4582	1.5061	1.5481	1.5863	1.6216
	v	0.0159	0.0163	0.0170	0.0180	0.0195	0.0216	0.0256	0.0397	0.0757	0.1020	0.1221	0.1391	0.1544	0.1684	0.1817
6000	h	83.7	181.7	281.0	382.7	488.6	602.9	736.1	945.1	1188.8	1323.6	1422.3	1505.9	1582.0	1654.2	1724.2
	s	0.1258	0.2870	0.4271	0.5528	0.6693	0.7826	0.9026	1.0176	1.2615	1.3574	1.4229	1.4748	1.5194	1.5593	1.5962
	v	0.0158	0.0163	0.0170	0.0180	0.0193	0.0213	0.0248	0.0334	0.0573	0.0816	0.1004	0.1160	0.1298	0.1424	0.1542
7000	h	86.2	184.4	283.0	384.2	489.3	601.7	729.3	901.8	1124.9	1281.7	1392.2	1482.6	1563.1	1638.6	1711.1
	s	0.1252	0.2859	0.4256	0.5507	0.6663	0.7777	0.8926	1.0350	1.2055	1.3171	1.3904	1.4466	1.4938	1.5355	1.5735

Second Law of Thermodynamics

There is nothing in the statement ofthe first law of thermodynamics that infers there is any limitation on the direction of energy flow or the conversion of one form of energy to another. It is the second law that provides such limitations. For example, a hot and a cold metal block are placed into an insulated box. Ifwe opened the box an hour later, we would not be surprised to find two warm blocks. But the opposite is not true. Ifwe put two warm blocks in the box and opened it an hour later, we would not expect to find one hot block and one cold block.

Another important limitation of the second law concerns the conversion of work to heat, and vice versa. Ifwork is supplied to a system, it is possible for it to be completely converted to heat. But the opposite cannot happen. It is impossible to devise a system that will convert all the heat supplied to it into work. Any engine must reject or exhaust some of the heat supplied to the surroundings. In other words, some of the energy supplied to the engine is "unavailable" to perform work. Another way of stating this implication of the second law is that the thermal efficiency of a heat engine must be less than 100 percent.

THE CARNOT CYCLE

The Carnot cycle represents a hypothetical heat engine that provides an upper limit on the conversion of heat supplied into work. A real engine operating between the same two temperature limits can have a thermal efficiency that approaches, but does not exceed, that ofthe Carnot cycle.

The Carnot cycle, shown in figure 1-4, consists of the following four processes:

- 1-2 reversible constant-temperature heat addition
- 2-3 reversible adiabatic expansion
- 3-4 reversible constant-temperature heat rejection
- 4-1 reversible adiabatic compression

The thermal efficiency of the Carnot cycle can be calculated using the following equation:

$$E_{\text{th}} = \frac{W_{\text{net}}}{Q_{\text{in}}} = \frac{T_H - T_L}{T_H}$$

where

T_H and T_L = the cycle temperature limits in R or K.

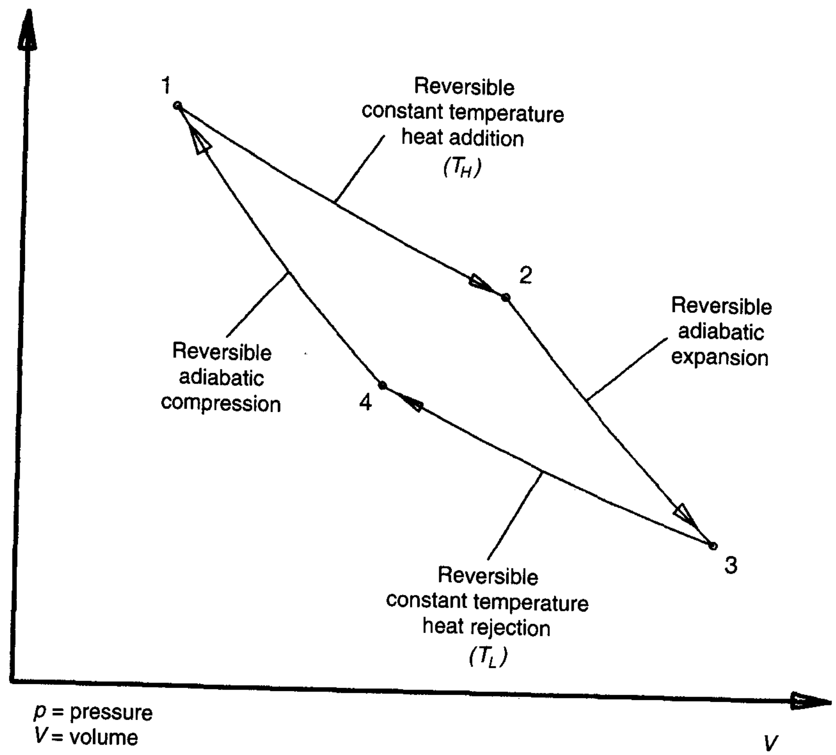


Figure 1-4. The Carnot cycle

EXAMPLE 1-4: An engine operates on the Carnot cycle. It receives 1,000 Btu per minute at 1,040°F and rejects heat at 540°F. Determine the cycle efficiency and the power being produced.

Solution: Determine the cycle efficiency.

$$E_{\text{th}} = \frac{T_H - T_L}{T_H} = \frac{1{,}500^\circ\text{R} - 1{,}000^\circ\text{R}}{1{,}500^\circ\text{R}} = 0.333$$

The power produced is

$$\begin{aligned} W_{\text{net}} &= (E_{\text{th}})(Q_{\text{in}}) = (0.333)(1{,}000 \text{ Btu/min}) \\ &= 333 \text{ Btu/min} = 7.85 \text{ hp} \end{aligned}$$

ENTROPY AND THE MOLLIER CHART

Entropy is a property just like pressure, temperature, and enthalpy. Unfortunately, entropy is not as easy to define as some other properties. One

way to think of entropy is that it is a measure of the reversibility of a process. All real processes involve losses of some kind, and are therefore irreversible. The irreversibilities in the process result in an increase in entropy. The more irreversible the process, the greater the increase in entropy.

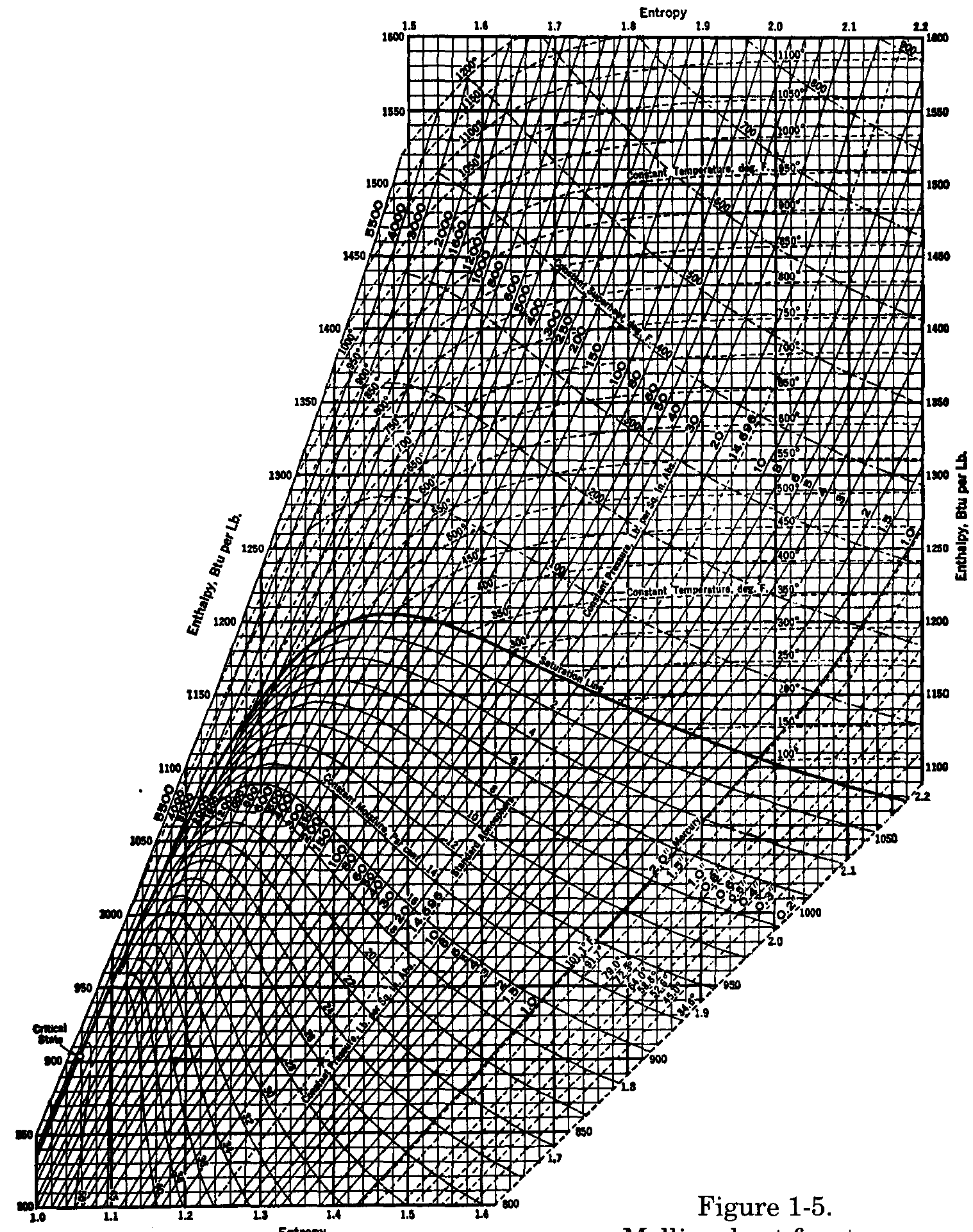


Figure 1-5.
Mollier chart for steam

A useful tool in analyzing processes is the Mollier chart, a plot of the enthalpy of a substance versus its entropy. Figure 1-5 is a Mollier chart for steam. The dark curved line in the middle of the graph is the saturation line. The region above this line is the superheated steam region. Mixtures of saturated steam and liquid lie below this line. Lines of constant temperature and pressure in the superheat region and lines of constant temperature and moisture in the saturated steam and liquid region aid in locating points on the chart.

EXPANSION PROCESS

Assume a fluid is being e:wanded in a turbine, producing power. If the expansion process is ideal, with no losses (Le., reversible) and with no heat transfer to the surroundings (adiabatic), the entropy of the fluid at the outlet will be the same as that at the inlet. This is referred to as an isentropic (constant entropy) process. It will plot on the Mollier chart as a vertical line. See the process 1 to 28 in figure 1-6.

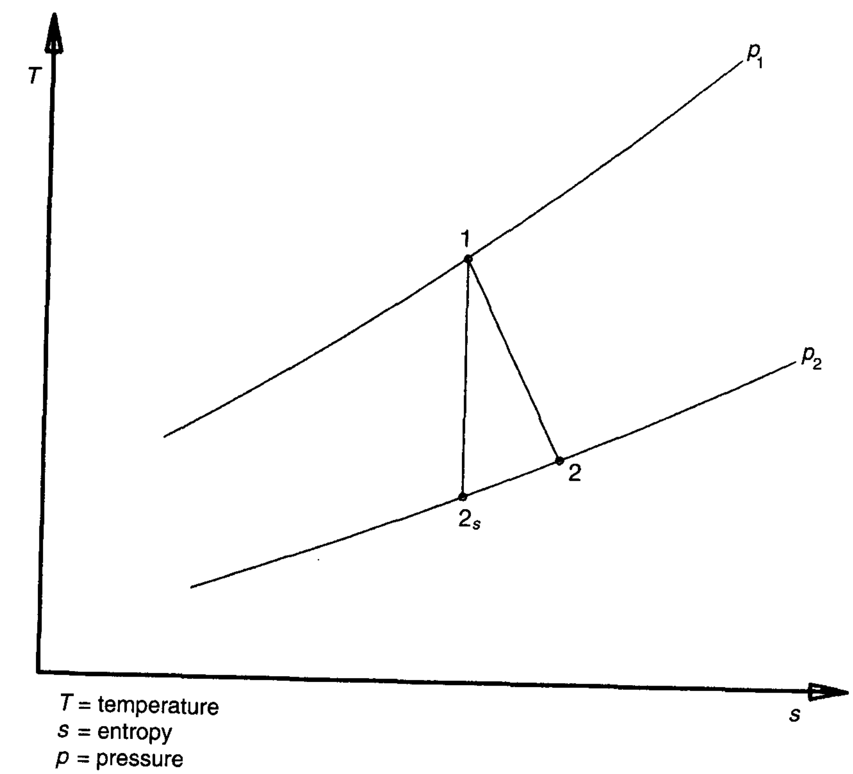


Figure 1-6. Expansion process

Any real expansion process will involve losses and/or heat transfer to the surroundings. This will result in an increase in entropy of the fluid from inlet to outlet. See the process 1 to 2 in figure 1-6.

It is customary to define the efficiency of an expansion process by comparing the enthalpy change for the isentropic expansion to the enthalpy change for the actual expansion process. For the processes shown in figure 1-6, the expansion efficiency can be defined as follows:

$$E_{\text{exp}} = \frac{h_1 - h_2}{h_1 - h_{2s}}$$

COMPRESSION PROCESS

A compression process has a lot of similarities to the expansion process described above. The ideal compression process is reversible adiabatic, i.e., isentropic. Like the ideal expansion process, the ideal compression process will plot as a vertical line on the Mollier chart. Any real process will involve

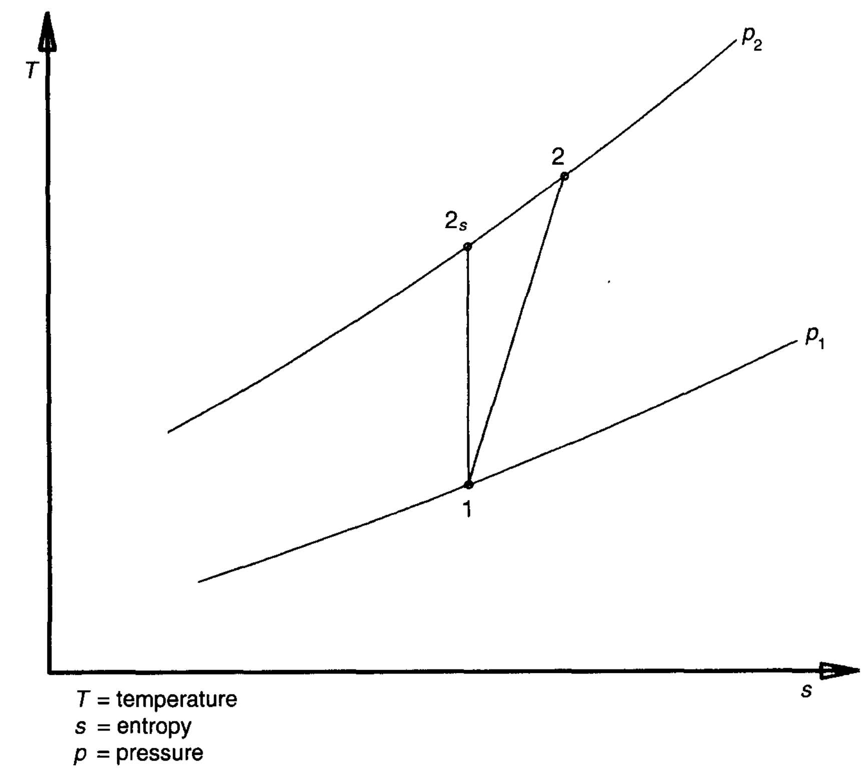


Figure 1-7. Compression process

losses of some kind, and the work of compression will be higher than for the ideal isentropic process (see fig. 1-7). The compression process efficiency can be defined in a manner similar to the efficiency of an expansion process:

$$E_{comp} = \frac{h_1 - h_{2s}}{h_1 - h_2}$$

THROTTLING PROCESS

Throttling is the uncontrolled expansion of a fluid from a high pressure to a lower pressure. An example is flow through a partially closed valve. No work is done and there is negligible heat transfer to or from the surroundings. The resulting high kinetic energy is dissipated in fluid friction. Assuming little or no change in the potential energy or kinetic energy from inlet to outlet, the enthalpy of the fluid remains constant ($h_1 = h_2$). A throttling process will plot as a horizontal line on the Mollier chart (see fig. 1-8).

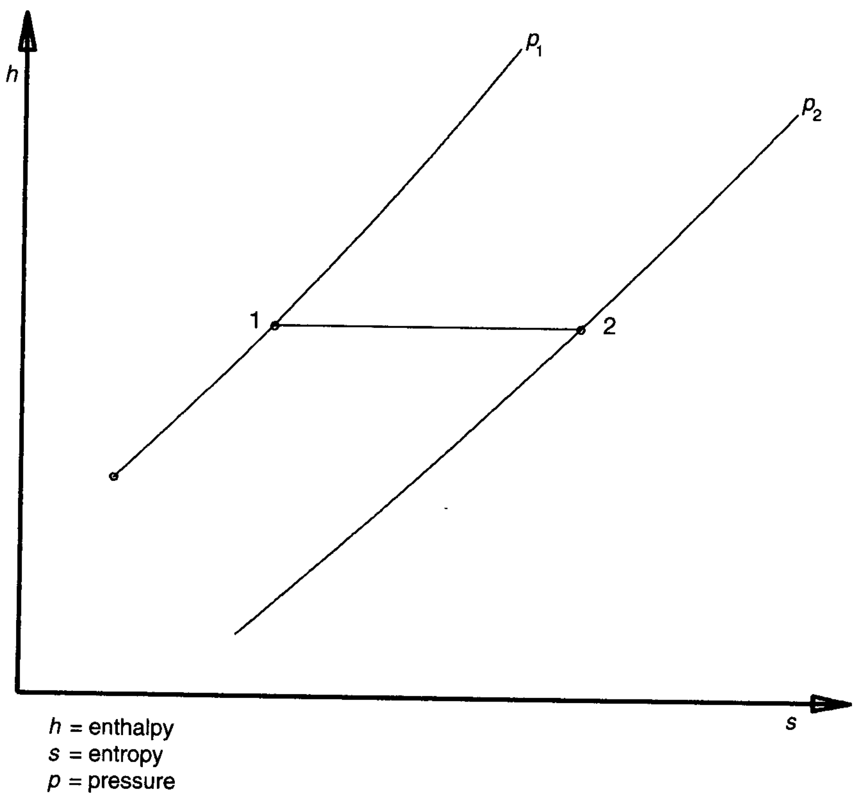


Figure 1-8. Throttling process

A calorimeter is a device for measuring the enthalpy of steam. A common variety used for measuring the enthalpy of wet steam with a small amount of moisture is the throttling type. One of the difficulties with wet steam is that the temperature and pressure are not independent properties. One additional property is necessary to determine the state of the steam. Figure 1-9 shows a typical throttling calorimeter. Wet steam under pressure is expanded into a chamber at atmospheric pressure. As the steam is throttled, it becomes superheated. The enthalpy of the superheated steam can now be determined using the superheated steam tables with the steam's temperature and pressure. Since the process across the valve is throttling, the enthalpy of the superheated steam in the chamber is the same as that of the wet steam in the line. The moisture (or quality) can be calculated using the following equation:

$$h = h_f + x(h_{fg})$$

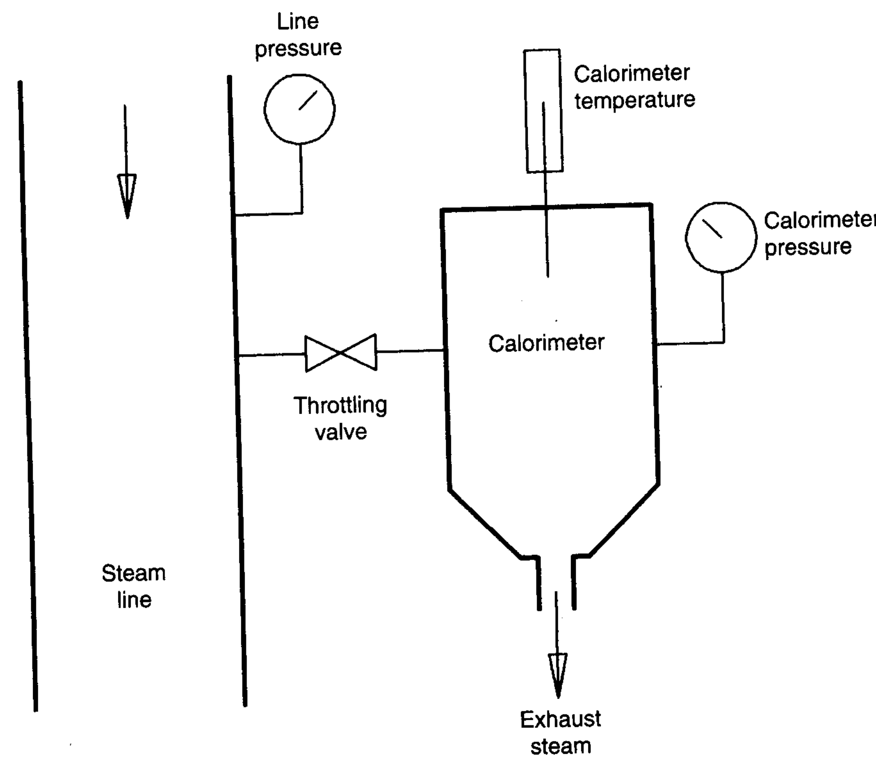


Figure 1-9. Throttling calorimeter

Throttling calorimeters can be used only for steam under moderate to high pressure with a low moisture content. It is necessary for the steam to become superheated after throttling for the enthalpy and quality to be determined.

EXAMPLE 1-5: A throttling calorimeter is connected to a line carrying wet steam at 400 psia. The pressure and temperature in the calorimeter body are 14.7 psia and 250°F. Determine the enthalpy and quality of the steam in the line.

Solution: The enthalpy of the wet steam in the line is equal to that of the superheated steam in the calorimeter. From the superheated steam tables at 14.7 psia and 250°F,

$$h_2 = h_1 = 1{,}168.8 \text{ Btu/lbm}$$
$$h_1 = h_f + x h_{fg}$$

From the saturated steam tables at 400 psia, h_f = 424.2 Btu/lbm and h_{fg} = 780.4 Btu/lbm,

$$1{,}168.8 = 424.2 + (x)(780.4)$$
$$x = 0.954 \text{ or } 95.4\% \text{ (4.6\% moisture)}$$

Ideal Gas Laws

Gases are pure substances in the vapor phase that are highly superheated. There are many situations that involve gases. For example, air at atmospheric pressure is a mixture of gases, primarily nitrogen and oxygen, though it commonly is modeled as a single substance.

The behavior of gases under varying conditions of pressure, temperature, and volume can be predicted quite accurately using the ideal gas equation of state. This equation or law is

$$pV = mRT$$

where

p = absolute pressure
V = total volume
m = mass
R = individual gas constant
T = absolute temperature

The value of R for various gases is tabulated in table 1-5. If both sides of the equation are divided by the mass, the following results:

$$pv = RT$$

where

v = specific volume

The value of the individual gas constant can also be determined using the following:

$$R = \frac{\overline{R}}{M}$$

where

$\overline{R}$ = the universal gas constant
M = molecular mass

The universal gas constant has a value of

$$\overline{R} = 1{,}545.32 \text{ ft-lbf}/p\text{mol-}R = 8.3143 \text{ kJ/kgmol-K}$$

Values for the molecular mass of some common gases are tabulated in table 1-5.

EXAMPLE 1-6: An oxygen cylinder of 2 ft³ has a pressure of 3,000 psig and a temperature of 60°F. The oxygen is gradually used until the pressure is 400 psig. The temperature remains constant. Determine the mass of oxygen used.

Solution: Use the ideal gas law to determine the mass in the tank before and after the removal of the gas.

$$m_1 = \frac{pV}{RT} = \frac{(3{,}014.7 \text{ psia})(144 \text{ in}^2/\text{ft}^2)(2\text{ft}^3)}{(48.29 \text{ ft-lbf/lbm-}R)(520° \text{R})} = 34.58 \text{ lbm}$$
$$m_2 = \frac{pV}{RT} = \frac{(414.7 \text{ psia})(144 \text{ in}^2/\text{ft}^2)(2\text{ft}^3)}{(48.29 \text{ ft-lbf/lbm}-R)(520° \text{R})} = 4.76 \text{ lbm}$$

The mass of oxygen used is thus the difference between the two:

$$\text{mass used} = m_1 - m_2 = 34.58 - 4.76 = 29.82 \text{ lbm}$$

Specific Heat

The specific heat of a substance is defined as the quantity of heat required to raise one unit of mass one degree, i.e., Btu/lbm-R or kJ/kg-K. For a gas, the specific depends on the process the gas undergoes during the heat addition process. It is common to tabulate the specific heat for gases for

TABLE 1-5

Gas Constants and Specific Heats at Low Pressures and 25°C (77°F)

Gas	M (lbm/pmol)	c_p (Btu/lbm-R)	c_p (kJ/kg-K)	c_v (Btu/lbm-R)	c_v (kJ/kg-K)	k	R (ft-lbf/lbm-R)	R (kJ/kg-K)
Acetylene C_2H_2	26.036	0.4048	1.6947	0.3285	1.3753	1.232	59.35	0.3195
Air	28.97	0.24	1.0047	0.1714	0.7176	1.4	53.34	0.287
Ammonia NH_3	17.032	0.499	2.089	0.382	1.5992	1.304	90.73	0.4882
Argon Ar	39.95	0.1244	0.5208	0.0747	0.3127	1.666	38.68	0.2081
Carbon dioxide CO_2	44.01	0.2016	0.844	0.1565	0.6552	1.288	35.11	0.1889
Carbon monoxide CO	28.01	0.2487	1.0412	0.1778	0.7444	1.399	55.17	0.2968
Chlorine Cl_2	70.914	0.1144	0.4789	0.0864	0.3617	1.324	21.79	0.1172
Ethane C_2H_6	30.068	0.4186	1.7525	0.3526	1.4761	1.187	51.39	0.2765
Ethylene C_2H_4	28.052	0.3654	1.5297	0.2946	1.2333	1.24	55.09	0.2964
Helium He	4.003	1.241	5.1954	0.745	3.1189	1.666	386.04	2.077
Hydrazine N_2H_4	32.048	0.393	1.6453	0.33	1.3815	1.195	48.22	0.2594
Hydrogen H_2	2.016	3.419	14.3136	2.434	10.190	1.4	766.54	4.125
Methane CH_4	16.043	0.5099	2.1347	0.3861	1.6164	1.321	96.33	0.5183
Neon Ne	20.183	0.246	1.0298	0.1476	0.6179	1.666	76.57	0.4120
Nitrogen N_2	28.016	0.2484	1.0399	0.1775	0.7431	1.399	55.16	0.2968
Oxygen O_2	32	0.2194	0.9185	0.1573	0.6585	1.395	48.29	0.2598
Propane C_3H_8	44.094	0.3985	1.6683	0.3535	1.4799	1.127	35.05	0.1886
Sulfur dioxide SO_2	64.07	0.1487	0.6225	0.1177	0.4927	1.263	24.12	0.1298
Water vapor H_2O	18.016	0.4454	1.8646	0.3352	1.4033	1.329	85.77	0.4615
Xenon Xe	131.3	0.0378	0.1582	0.0227	0.0950	1.666	11.77	0.0633

constant-pressure (c_p) and constant-volume (c_v) processes. The values of specific heat for various ideal gases are included in table 1-5. The heat added (or removed) in a process can be calculated using the following:

$$Q = m\, c_x (T_2 - T_1)$$

where

c_x = the specific heat for the process

For ideal gases, it can be shown that the changes in internal energy and enthalpy depend only on temperature. The change in internal energy for an ideal gas with constant specific heat can be found using the following equation:

$$U_2 - U_1 = m\, c_v (T_2 - T_1)$$

and the change in enthalpy can be found using the following:

$$H_2 - H_1 = m\, c_p (T_2 - T_1)$$

The above equations are valid for any ideal gas process: reversible, irreversible, constant-pressure, constant-volume, adiabatic, isentropic, and others. The only limitation is that the gas behave as an ideal gas and the specific heat be a constant.

It is important to note that the values of c_p and c_v for real gases vary with temperature. The values listed in table 1-5 are valid for normal atmospheric temperatures only. Figure 1-10 shows the variation of constant-pressure specific heat for some common gases.

Since liquids are incompressible, the specific heat does not depend on the process (constant pressure, constant temperature, etc.). The values of specific heat for some common liquids at 20°C are given in table 1-6. Like gases, the value of specific heat for liquids varies with temperature.

TABLE 1-6

Specific Heat of Some Common Liquids at 20°C

Substance	c_p *kJ/kg-C*	c_p *Btu/lbm-F*
Water	4.182	1.000
Lubricating oil	1.880	0.450
Refrigerant-12	0.966	0.230
Ethylene Glycol	2.380	0.570
Ammonia	4.800	1.150
Carbon Dioxide	3.140	0.750
Glycerine	2.390	0.570

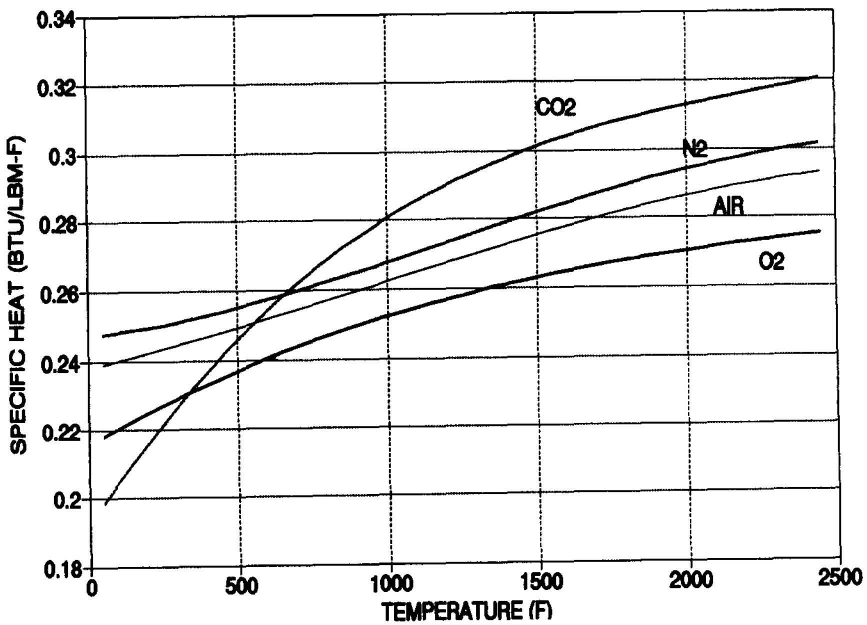


Figure 1-10. Specific heat of common gases

FLUID MECHANICS

Fluid mechanics is a science dealing with the behavior of fluids at rest and in motion. Fluid mechanics is based on the application of some basic principles to the design and analysis of systems in which a fluid is the working medium. There are many examples of such systems that are of interest to the marine engineer.

Bernoulli's Equation

The application of Newton's second law ($F = ma$) to the steady flow of an ideal fluid along a streamline results in Bernoulli's equation:

$$\frac{p}{\rho} + \frac{V^2}{2g_c} + \frac{gz}{g_c} = \text{constant (ft - lbf/lbm)}$$

where

p = pressure (in lbf/ft^2)
ρ = density (in lbm/ft^3)
V = fluid velocity (in ft/sec)
z = elevation (in ft)
g = local acceleration of gravity (in ft/sec^2)

The three terms in the equation are commonly called the pressure head, the velocity head, and the static head. It is traditional in engineering practice to refer to the units of head in Bernoulli's equation as "feet." Note that this involves the cancelling of lbf and lbm. This is clearly incorrect. It is important to remember that each term in the equation represents a quantity of energy (in ft-lbf) per unit of mass.

For SI units, Bernoulli's equation becomes

$$\frac{p}{\rho} + \frac{V^2}{2} + gz = \text{constant (J/kg)}$$

It is important to remember that Bernoulli's equation is valid only for the following conditions:

- steady flow
- incompressible flow
- frictionless flow
- flow along a streamline

Do not attempt to apply Bernoulli's equation to other flow situations (such as transient flow, viscous liquids, the flow of gases if pressure and density change, and the like).

Static, Stagnation, and Dynamic Pressure

The pressure used in Bernoulli's equation is commonly called the static pressure. The static pressure is that which would be measured by an instrument moving with the fluid flow. For fluid flowing in a pipe, this is obviously inconvenient. However, if a small hole is made in the side of the pipe perpendicular to the flow, a pressure instrument will measure the static pressure.

If a moving fluid is decelerated to zero velocity by a frictionless process, the pressure that would be measured is the stagnation (or total) pressure, p_o. For incompressible flow, Bernoulli's equation can be used to calculate the stagnation pressure:

$$\frac{p_o}{\rho} = \frac{p}{\rho} + \frac{V^2}{2g_c}$$

or

$$p_o = p + \frac{\rho V^2}{2g_c}$$

The last term in the above equation ($\rho V^2/2g_c$) is commonly called the dynamic pressure. If the static and stagnation pressures of the fluid flow are measured, the fluid velocity can be calculated (see fig. 1-11).

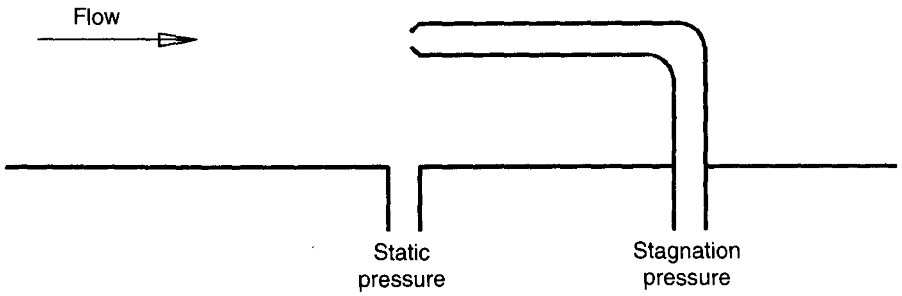


Figure 1-11. Measuring static and stagnation pressure

Pump Head

One useful application of Bernoulli's equation is in the definition of pump head. Pump head is defined as the difference in total head at the pump discharge minus the total head at the pump suction. Using Bernoulli's equation,

$$\text{pump head} = \left(\frac{p_2}{\rho} + \frac{V_2^2}{2g_c} + \frac{g\,z_2}{g_c}\right) - \left(\frac{p_1}{\rho} + \frac{V_1^2}{2g_c} + \frac{g\,z_1}{g_c}\right)$$

or

$$\text{pump head} = \frac{(p_2 - p_1)}{\rho} + \frac{\left(V_2^2 - V_1^2\right)}{2g_c} + (z_2 - z_1)\frac{g}{g_c}$$

Again, it is important to remember that pump head in "feet" is really units of ft-lbf/lbm. This is specific energy or energy per unit mass. Thus pump head represents the energy imparted to each unit of mass as it passes from pump suction to pump discharge.

Liquid Horsepower

The power being transferred to the fluid by the pump is referred to as the liquid horsepower (lhp). It is simply the product of the pump head times the mass flow rate. Using the definition of horsepower 1 hp = 33,000 ft-lbf/min,

$$\text{lhp} = \frac{\dot{m}\,H_{\text{pump}}}{33{,}000}$$

where

$$\dot{m} = \text{lbm/min}$$
$$H_{\text{pump}} = \text{ft-lbf/lbm}$$

Since the capacity of pumps is commonly given in units of gallons per minute, the above equation can be written as follows:

$$\text{lhp} = \frac{\text{gpm}\ H_{\text{pump}}\ sg}{3{,}960}$$

where

sg = the fluid specific gravity, i.e., its density divided by the density of cold water (62.4 lbm/ft^3).

In many situations, there is little or no change in the elevation between the suction and discharge of the pump, and little or no difference in the size of the suction and discharge connections. This means the change in static and velocity head can be neglected, and the liquid horsepower can be estimated using the following equation:

$$\text{lhp} = \frac{\text{gpm}\ \Delta\text{psi}}{1{,}714}$$

where Δpsi = the pressure increase across the pump in psi. It is important to remember that if there is a change in static or velocity head across the pump, one of the first two equations should be used.

The efficiency of a pump is defined as the liquid horsepower (lhp) divided by the brake horsepower (bhp) being delivered by the motor or prime mover.

$$\text{Efficiency} = \frac{\text{lhp}}{\text{bhp}}$$

Example 1-7: A centrifugal pump is delivering 200 gpm of cold water. The discharge gauge reads 45 psig and the suction gauge reads 5 in Hg vacuum. The pump suction diameter is 3 inches and the discharge is 2 inches. Determine the pump head and the liquid horsepower. What is the pump efficiency if the motor is supplying 7.5 hp?

Solution: Convert the gauge readings to psia.

$$p_{\text{suct}} = 14.7 - (5)(0.491) = 12.245\ \text{psia}$$
$$p_{\text{disch}} = 14.7 + 45 = 59.7\ \text{psia}$$

Solve for the suction and discharge velocities.

$$V_{\text{suct}} = \frac{(\text{gpm})/\left[(7.481\ \text{gal/ft}^3)(60\ \text{sec/min})\right]}{(\pi D^2/4)}$$

$$= \frac{(200)/[(7.481)(60)]}{(\pi/4)(3/12)^2} = 9.077\ \text{ft/sec}$$

$$V_{\text{disch}} = \frac{(200)/[(7.481)(60)]}{(\pi/4)(2/12)^2} = 20.42\ \text{ft/sec}$$

The pump head is thus:

$$H_{\text{pump}} = \frac{(P_{\text{disch}} - P_{\text{suct}})}{\rho} + \frac{(V_{\text{disch}}{}^2 - V_{\text{suct}}{}^2)}{2g_c}$$

$$= \frac{(59.7 - 12.245)(144)}{62.4} + \frac{(20.42^2 - 9.077^2)}{(2)(32.2)}$$

$$= 114.7\ \text{ft-lbf/lbm}$$

Determine the liquid horsepower and the efficiency.

$$\text{lhp} = \frac{\text{gpm}\ H_{\text{pump}}\ sg}{3{,}960} = \frac{(200)(114.7)(1)}{3{,}960} = 5.793\ \text{hp}$$

$$\text{Eff} = \frac{\text{lhp}}{\text{bhp}} = \frac{5.793}{7.5} = .772 = 77.2\ \text{percent}$$

Viscosity

Viscosity is the resistance of a fluid to being deformed when a shearing force is applied. Since molasses exhibits a much greater resistance to deformation or flow than water, it is said to be more viscous or to have a higher viscosity.

Figure 1-12 shows a fluid between two plates, one stationary and one being pulled to the right at a constant velocity U by a constant force F. The fluid in contact with the upper plate will move at velocity U and the fluid in contact with the lower plate will be stationary. In between, the fluid velocity will vary linearly. Experiments have shown that for many common fluids, the force F varies directly with plate area, directly with the velocity U, and inversely with the distance y. The shear stress τ is the applied force F divided by the plate area A. The constant that relates the force per unit of plate area to the fluid shearing velocity is the absolute (or dynamic) viscosity (μ):

$$\frac{F}{A} = \tau = \mu \frac{dV}{dy}$$

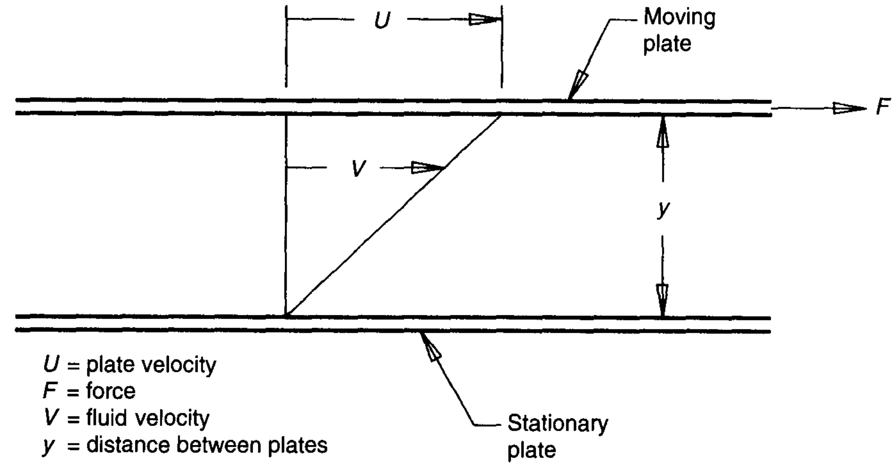


Figure 1-12. Definition of viscosity

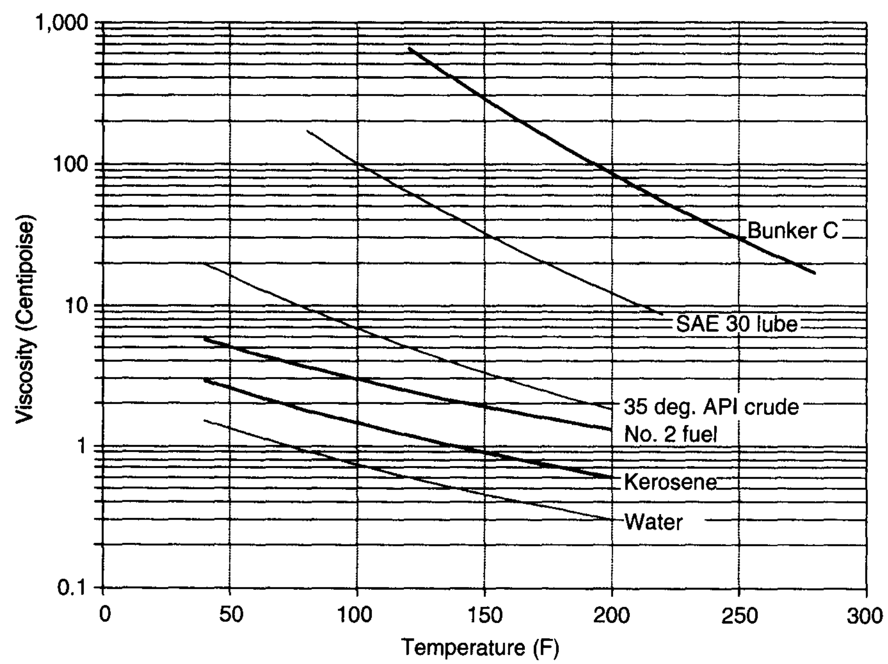


Figure 1-13. Viscosity of common liquids

TABLE 1·7

Viscosity Conversions

Saybolt Seconds Universal	*Centipoise*	*Saybolt Seconds Furol*	*Seconds Redwood No.1*	*Seconds Redwood No.2*
35	2.71		32.1	
40	4.25		36.2	5.10
50	7.68		44.3	5.83
60	10.3		52.3	6.77
70	13.1	12.95	60.9	7.60
80	15.7	13.70	69.2	8.44
90	18.1	14.44	77.6	9.30
100	20.5	15.24	85.6	10.12
150	31.9	19.3	128.	14.48
200	43.0	23.5	170.	18.90
250	53.8	28.0	212.	23.45
300	64.6	32.5	254.	28.0
400	86.2	41.9	338.	37.1
500	108.	51.6	423.	46.2
600	130.	61.4	508.	55.4
700	151.	71.1	592.	64.6
800	173.	81.0	677.	73.8
900	194.	91.0	762.	83.0
1,000	216.	100.7	896.	92.1
1,500	324.'	150.	1,270.	138.2
2,000	432.	200.	1,690.	184.2
2,500	539.	250.	2,120.	230.
3,000	648.	300.	2,540.	276.
4,000	862.	400.	3,380.	368.
5,000	1,079.	500.	4,230.	461.
6,000	1,295.	600.	5,080.	553.
7,000	1,510.	700.	5,920.	645.
8,000	1,726.	800.	6,770.	737.
9,000	1,942.	900.	7,620.	829.
10,000	2,160.	1,000.	8,460.	921.

$$\text{Centistokes} = \frac{\text{Centipoise}}{\text{Specific Gravity}}$$

Above the range of this table, the following approximate relationships can be used:

$$\text{SSU} = \text{Centistokes} \times 4.365$$
$$\text{SSF} \times 10 = \text{SSU}$$
$$\text{Redwood No. 1} \times 1.095 = \text{SSU}$$
$$\text{Redwood No. 2} \times 10.87 = \text{SSU}$$

The units of absolute viscosity in the English system are Ibf-sec/ft². In the metric system, the basic unit of absolute viscosity is the poise (poise = gm/cm-sec).

In fluid mechanics, the ratio of absolute viscosity 11 to density p often arises. This ratio is called the kinematic viscosity v. Since density is mass divided by volume (length^3), the usual unit of kinematic viscosity in the English system is ft^2/sec, and in the metric system is the stoke (cm^2/sec).

Over the years, a number of other units of viscosity have become common in engineering practice. Many of these are based on a particular device developed for measuring viscosity. For example, the Saybolt viscosimeter uses the time it takes a 60-cc sample offluid to pass through an orifice of a particular size. The units are SSU (Saybolt Seconds Universal) and SSF (Saybolt Seconds Furol). Other similar units are Redwood No.1 and No.2 and degrees Engler. Table 1-7 provides some useful conversions among some common units of viscosity. The viscosity of most fluids varies significantly with temperature. Figure 1-13 shows the viscosity of some common liquids as a function of temperature.

HEAT TRANSFER

Heat transfer-the flow of thermal energy from one location to another-plays an important role in the operation of many engineering systems. If the exchange of energy is the result of a temperature difference, the transfer of heat is said to have taken place. Ifheat is being transferred from one body to another, the first law of thermodynamics requires that the internal energy given up by the first body be equal to that taken up by the second. The second law ofthermodynamics requires that the heat flow be from the hot body to the cold body.

The processes that govern heat transfer are commonly termed conduction, convection, and radiation. Almost all situations encountered in engineering practice involve two, or sometimes all three, modes of heat transfer. *Conduction* is the term applied to the exchange of internal energy within a body, or from one body in contact with another, by the exchange of energy from one molecule to another by direct communication. In conduction, the heat transfer takes place within the boundaries ofthe bodies and there is no observable movement of the matter making up the bodies. *Convection* is the term applied to heat transfer caused by the mixing of one portion of a fluid with another as the fluid moves. While the exchange of energy from one molecule to another is via conduction, the energy is transported from one location to another by the movement of the fluid. *Radiation* is the term applied to the exchange of thermal energy from a warmer body to a cooler body through electromagnetic radiation. The important difference between radiation and the two modes discussed above is that the heat transfer can occur without the need for a medium of transport between the bodies. For example, the earth receives heat from the sun across millions of miles of vacuum by thermal radiation.

Conduction

Conduction was defined above as the movement of thermal energy within a substance with no displacement of the material. While conduction can occur in a fluid, it is most commonly observed in solids. A simple example of heat transfer by conduction is the steady-state flow of heat through a solid wall. Figure 1-14 shows a wall with a thickness of Δx with constant

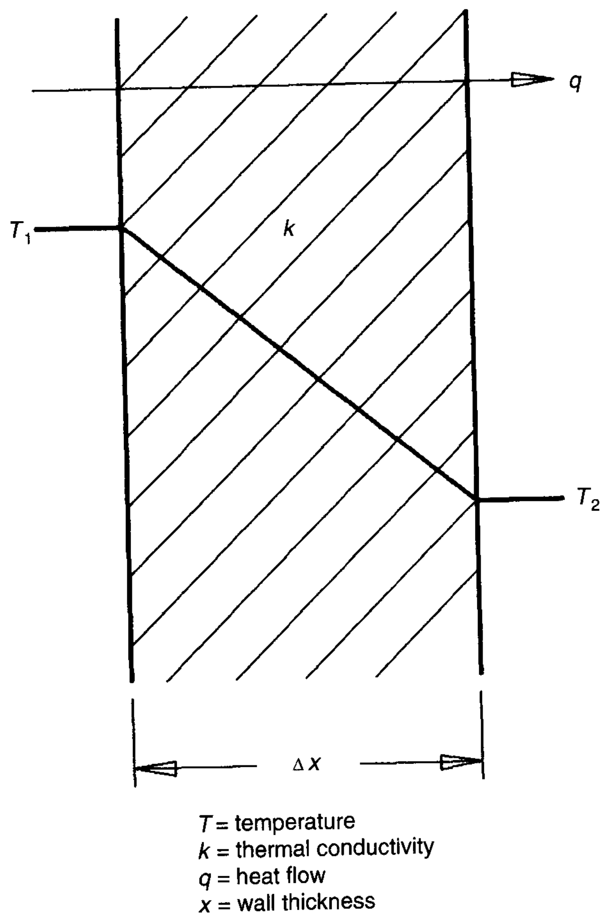


Figure 1-14. Conduction through a wall

temperatures T_1 and T_2 on its two faces. The equation that governs this situation is

$$q = kA\frac{T_1 - T_2}{\Delta x}$$

where

k = thermal conductivity, Btu/hr-ft-F (W/m-C)
A = wall area, ft^2 (m^2)

The thermal conductivity is a property of the material making up the wall. It is dependent on the chemical composition of the substance, its phase, and its molecular structure. Materials with low values of thermal conductivity are referred to as thermal insulators, and those with high values are referred to as thermal conductors.

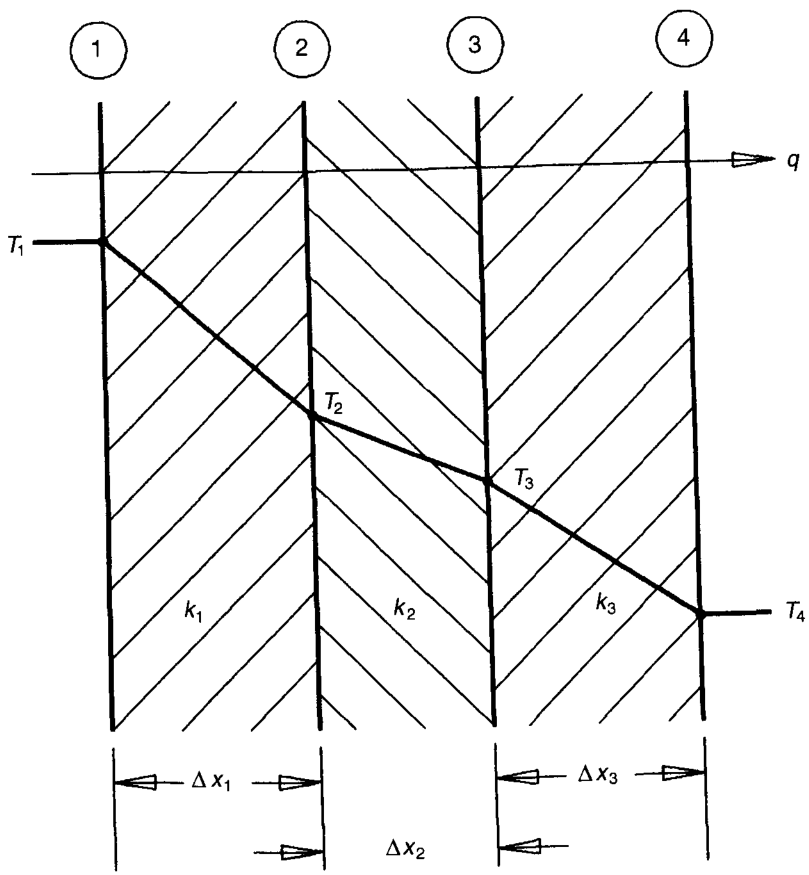
TABLE 1-8

Thermal Conductivity of Some Common Substances (at 1 atmosphere and 20°C)

Substance	*k* *W/m-C*	*k* *Btu/hr-ft-F*
Gases		
Refrigerant-12	0.0093	0.0048
Air	0.026	0.015
Carbon Dioxide	0.048	0.0277
Liquids		
Engine Oil	0.145	0.084
Ethylene Glycol	0.24	0.139
Water	0.57	0.33
Solids		
Silica Insulation	0.023	0.0133
Fiberglass Insulation	0.038	0.022
Wood, Oak	0.17	0.098
Clay	1.3	0.75
Glass Plate	1.4	0.81
Stainless Steel	14.	8.1
Steel	42.	24.3
Iron	80.	46.
Aluminum	237.	137.
Copper	401.	232.

Table 1-8 lists the thermal conductivities of some common substances. The vast range of values is immediately obvious. Some observations that can be made are that liquids are better conductors than gases, solids are better conductors than liquids, and solid metals are better conductors than nonmetals. It should be remembered that the thermal conductivity is not constant; for most materials it will vary with temperature and sometimes pressure. Use caution in applying tabulated values at temperatures different from those listed.

A situation commonly encountered is a wall composed of several layers of different materials of different thicknesses. Figure 1-15 shows a wall constructed ofthree layers of different materials. Under steady-state conditions, the heat flow through each layer must be the same.



T = temperature
q = heat flow
k = thermal conductivity
x = wall thickness

Figure 1-15. Conduction through multilayer wall

$$q = k_1 A \frac{T_1 - T_2}{\Delta x_1} = k_2 A \frac{T_2 - T_3}{\Delta x_2} = k_3 A \frac{T_3 - T_4}{\Delta x_3}$$

Combining the above three equations and eliminating T_2 and T_3

$$q = A \frac{T_1 - T_4}{\Delta x_1/k_1 + \Delta x_2/k_2 + \Delta x_3/k_3}$$

EXAMPLE 1-8: A house wall consists of an outer layer of 10 cm of common brick (k = 0.69 W/m-C), 1.25 cm of sheathing (k = 0.05 W/m-C), and 1.25 cm of sheetrock (k = 0.75 W/m-C). The outside brick surface is 0°C and the inner wall surface is 25°C. How much heat is lost through a wall 2 m high by 5 m long?

Solution: Substitute into the equation above:

$$q = (2)(5)\frac{(25-5)}{0.10/0.69 + 0.0125/0.05 + 0.0125/0.75}$$

$$q = 485.9 \text{ watts}$$

Convection

Convection was defined above as the transfer of energy within a fluid by fluid motion. There are two basic types of convection: free convection, in which density differences cause the fluid motion; and forced convection, where a pressure difference, typically created by a fan or pump, causes the fluid motion. The basic relation of convective heat transfer is Newton's law of cooling:

$$q = hA(T_s - T_f)$$

where

h = convectivefilm coefficient,Btu/hr-ft^2-F or W/m^2-C
A = surface area, ft^2 or m^2
Ts = surface temperature, F or C
Tr = fluid temperature, F or C

Figure 1-16 shows the velocity and temperature profiles of a cold fluid flow across a heated plate. This thin region close to the surface area where the temperature and velocity vary is called the boundary layer. Note that the fluid velocity is zero at the solid surface and increases to the free stream velocity at the outer edge. The heat transfer at the surface is thus by convection, and is dependent on the mixing ofthe fluid within the boundary layer.

The value of the convective film coefficient depends on a variety of factors. Some of these include the properties of fluid involved, the phase of the fluid (liquid or gas), the velocity of fluid flow, whether or not there is a phase change of the fluid, and the shape of the heat transfer surface. The values of the film coefficient cannot be simply tabulated, but must be either calculated or estimated based on experimental data.

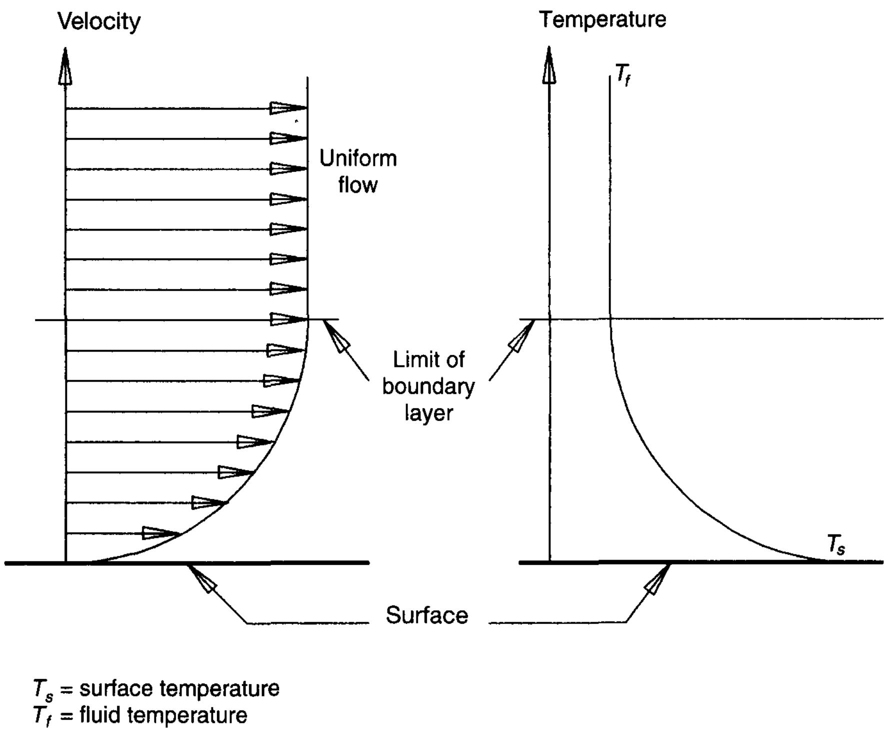


Figure 1-16. Convective boundary layer

Table 1-9 lists some typical values of the convective film coefficient for air and water in common situations. Note the wide range of values listed. Some general observations can be made. First, the film coefficient is higher for water (a liquid) than for air (a gas). Second, the film coefficient is higher when a phase change is taking place (boiling or condensing) than when there is only a change in fluid temperature. Third, the higher velocities of forced convection result in higher film coefficients than free convection. Determining the value of the film coefficient to use in a particular situation requires judgment and experience. There are a number of good texts and reference books on heat transfer (including *Fundamentals of Heat Transfer* by Chapman and *Introduction to Thermal Sciences* by Schmidt et al., cited at the end of this chapter) which can be consulted for further information.

TABLE 1-9

Typical Convective Film Coefficients

Fluid Conditions	*h* W/m^2-*C*	*h* *Btu/hr-ft*^{2}-*F*
Air and gases, 100–200 kPa	80–125	14–22
Air and gases, 1 MPa	250–400	44–70
Air and gases 10 MPa	500–800	88–140
Water	5,000–7,500	880–1,300
Light distillate oil	750–2,000	130–150
Lube oil and heavy fuel oil	60–750	10–130
Steam condensing, 10 kPa	8,000–12,000	1,400–2,000
Steam condensing, 100 kPa	10,000–15,000	1,750–2,650
Steam condensing, 1 MPa	15,000–25,000	2,650–4,400
Water boiling, < 500 kPa	3,000–10,000	525–1,750
Water boiling, 500 kPa–10 MPa	4,000–15,000	700–2,650

Radiation

It was observed above that heat transfer can occur from or to a body by electromagnetic radiation. Unlike heat transfer by conduction or convection, there is no need for a physical medium to be in contact with the body. The basic relationship governing radiative heat transfer is the Stefan- Boltzmann law, which states that the rate of radiative emission is proportional to the surface area times the absolute surface temperature to the fourth power. Thus,

$$q = \sigma A T^4 \text{ W or Btu/hr}$$

The Stefan-Boltzmann constant, σ, is 5.67×10^{-8} W/m^2-K^4 or 0.1714×10^{-8} Btu/hr-ft^2-R^4.

The Stefan-Boltzmann relationship above is based on an ideal blackbody. A blackbody is one that absorbs all the radiation that strikes it. Real materials are not perfect absorbers or radiators. One way of handling the radiative characteristics of a real body is to define emissivity, *E*. The emissivity will range from 1.0 for an ideal blackbody to 0.0 for an ideal reflector. Some typical values of emissivity for some common materials are listed in table 1-10.

TABLE 1-10

Emissivities of Common Materials

Substance	*Emissivity*
Aluminum, oxidized	0.16–0.33
Brass, polished	0.03
Brass, oxidized	0.60
Wrought iron, polished	0.28
Iron, rusted	0.60–0.70
Steel, rough plate	0.94
Steel, stainless	0.24–0.39
Paint, flat black	0.96
Paint, color	0.70–0.96
Brick, red rough	0.93
Brick, refractory	0.29–0.38

A common situation is a body exchanging heat by radiation with its surroundings. The net radiant energy exchanged between the real body and the surroundings is

$$q = AE\sigma\left(T_s^{\,4} - T_b^{\,4}\right)$$

Note that the heat transfer will be positive, i.e., into the body, if the temperature of the surroundings is higher than the body temperature. Negative heat transfer would be interpreted as heat transfer from the body to the surroundings.

EXAMPLE 1-9: A steam pipe with an 8-inch outside diameter has a surface temperature of 500°F. The emissivity of the pipe is 0.5. Calculate the heat transferred by radiation to the 80°F surroundings from a 100-foot length of the steam pipe.

Solution:

$$A = \pi DL = \pi \frac{8}{12} 100 = 209.4 \text{ ft}^2$$

$$q = (209.4)(0.5)\left(0.1714 \times 10^{-8}\right)\left(960^4 - 540^4\right)$$

$$q = 137{,}161 \text{ Btu/hr}$$

Overall Heat Transfer Coefficient

Many common heat transfer situations include a combination of conduction and convection. For example, heat is being lost through the window of a house on a cold winter night. Heat is transferred from the warm room air to the inside window surface by convection, through the window glass by conduction, and from the outside window surface to cold outside air by convection. An overall heat transfer coefficient, U, can be defined which combines the effects of all three processes. The heat transferred is thus

$$q = UA(T_i - T_o)$$

If the glass thickness x and the convective film coefficients on the inside h_i and outside h_o of the window are known, the overall heat transfer coefficient can be calculated as follows:

$$U = \frac{1}{1/h_i + x/k + 1/h_o}$$

Another common situation is the loss of heat from an insulated pipe carrying steam or another hot fluid. The overall heat transfer is a combination of convection on the inside of the pipe and the outside of the insulation and conduction through the pipe wall and the insulation (see fig. 1-17). An additional complication is that the heat transfer area is not a constant. The surface area is larger on the outside of the insulation than it is on the inside of the pipe. If the heat transfer coefficient is defined based on the outside surface area, the heat transferred is

$$q = U_o A_o (T_i - T_o)$$

and the overall heat transfer coefficient, U_o, can be calculated using the following equation:

$$U_o = \frac{1}{\dfrac{r_4}{r_2 h_{12}} + \dfrac{r_4 \ln(r_3/r_2)}{k_{23}} + \dfrac{r_4 \ln(r_4/r_3)}{k_{34}} + \dfrac{l}{h_{45}}}$$

Note that each of the terms in the denominator of the above equation has been modified to adjust for the varying area. This equation should be used for heat transfer through a cylinder whenever there is a significant difference between the inside and outside diameters.

Log Mean Temperature Difference

Heat exchangers are used in a wide variety of marine engineering systems. They are used to cool lubricating oil, to heat fuel oil, and to condense steam. Consider a tube-in-tube heat exchanger with one fluid flowing in the center tube cooling the fluid flowing in the annulus between the tubes. The fluids are arranged to flow in the same direction, or parallel-flow (see fig. 1-18). The heat transferred across the tube area is equal to heat lost by the hot fluid and also equal to heat gained by the cold fluid:

$$q = UA\Delta T_m = \dot{m}_h c_{ph}(T_{hi} - T_{ho}) = \dot{m}_c c_{pc}(T_{co} - T_{ci})$$

It is necessary to use a mean temperature difference across the tube since, as the hot fluid cools and the cool fluid warms, the temperature difference across the tube changes. For this situation, the difference is called the log mean temperature difference *(lmtd)* and can be calculated from the following:

$$lmtd = \frac{\Delta T_1 - \Delta T_2}{ln(\Delta T_1/\Delta T_2)}$$

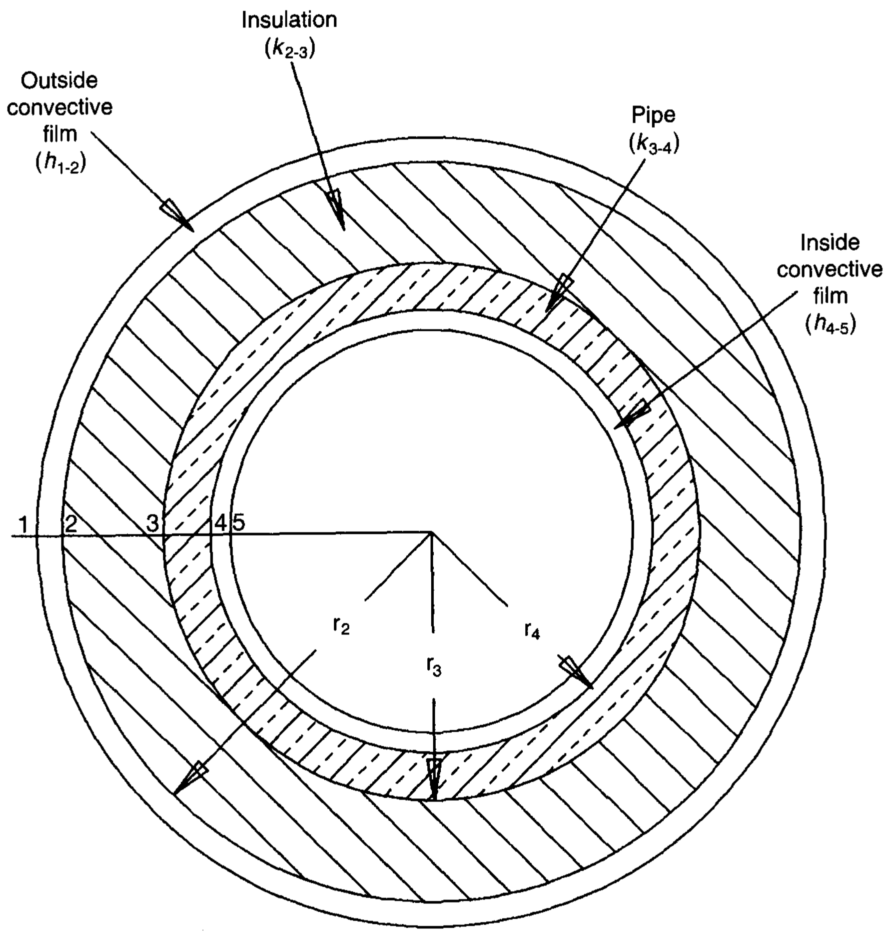


Figure 1-17. Combined heat transfer through insulated pipe

If the direction of flow of one of the fluids in the tube-in-tube heat exchanger is reversed, it becomes counter-flow. Figure 1-19 shows a counter-flow heat exchanger with its temperature distribution. Like the parallel-flow heat exchanger, the mean temperature difference across the tube is the *lmtd*. This is also true for heat exchangers where one or both of the fluids are undergoing a phase change as in a steam turbine condenser or in the evaporator of a refrigeration system. For many other configurations,

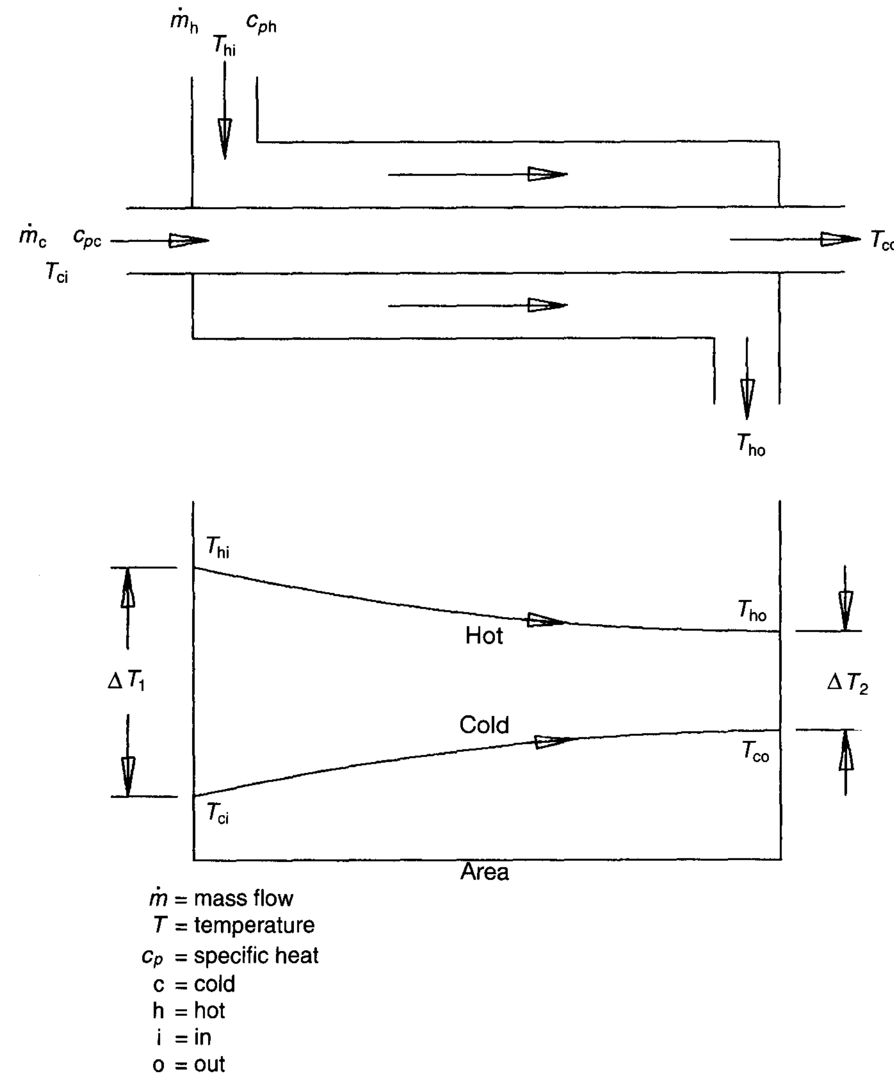


Figure 1-18. Tube-in-tube parallel-flow heat exchanger

such as a shell-and-tube heat exchanger with multiple tube passes, a correction factor must be applied to the *lmtd*. Any text on heat transfer or heat exchanger design will contain procedures for calculating the *lmtd* correction factor for common heat exchanger configurations.

EXAMPLE 1-10: A counter-flow heat exchanger is to cool 10,000 lbm/hr of oil with a c_p of 0.8 Btu/lbm-F from 190°F to 120°F. Water is used to cool the

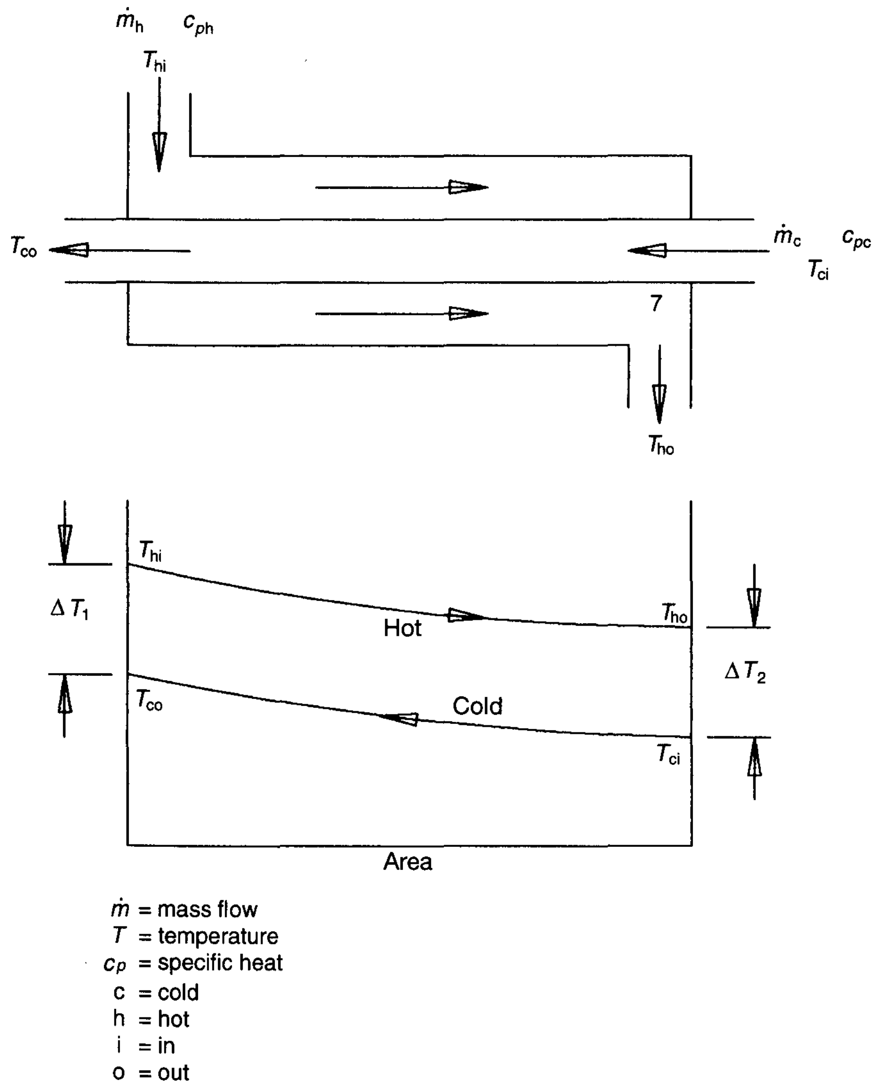


Figure 1-19. Tube-in-tube counter-flow heat exchanger

oil, entering at 75°F and leaving at 90°F. The overall heat transfer coefficient is 40 Btu/hr- ft^2-F. Determine the heat exchanger area and the cooling water flow.

Solution:

$$lmtd = \frac{(190-90)-(120-75)}{ln[(190-90)/(120-75)]} = 68.9^{\circ}\text{F}$$

$$A = \frac{Q}{U\,lmtd} = \frac{\dot{m}c_p\Delta T}{U\,lmtd} = \frac{(10{,}000)(0.8)(190-120)}{(40)(68.9)} = 203.2\ \text{ft}^2$$

$$\dot{m}_h c_{ph}(T_{hi} - T_{ho}) = \dot{m}_c c_{pc}(T_{co} - T_{ci})$$

$$(10{,}000)(0.8)(190-120) = \dot{m}_c(1.0)(90-75)$$

$$\dot{m}_c = 37{,}333\ \text{lb/hr} = 74.6\ \text{gpm}$$

REVIEW

1. Explain the difference between weight and mass.
2. Define absolute pressure, gauge pressure, and vacuum.
3. How are density and specific volume related?
4. What is the continuity equation for steady flow?
5. How does power differ from work?
6. In your own words, define the difference between heat, internal energy, and enthalpy.
7. What is the importance of the Carnot cycle?
8. What is a Mollier chart?
9. Briefly describe the three modes of heat transfer.
10. What is the overall heat transfer coefficient? What two modes of heat transfer does it combine?

REFERENCES

Burghardt, M. D., and J. A. Harbach. 1993. *Engineering Thermodynamics*, 4th ed. New York: HarperCollins.

Chapman, A. J. 1987. *Fundamentals of Heat Transfer*. New York: Macmillan.

EI-Wakil, M. M. 1984. *Powerplant Technology*. New York: McGraw-Hill.

Fox, R. W., and A. T. McDonald. 1992. *Introduction to Fluid Mechanics*, 4th ed. New York: John Wiley & Sons.

Schmidt, F. W., R. E. Henderson, and C. H. Wolgemuth. 1984. *Introduction to Thermal Sciences*. New York: John Wiley & Sons.

CHAPTER 2

Engineering Materials

WALTER M. MACLEAN

ENGINEERING MATERIAL REQUIREMENTS

Historically, many different, naturally occurring materials have been used to develop devices and components of engineering systems. Examples include early use of wood to make beams, shafts, rods, nails, or pins and pipes; stone for building gravity structures such as foundations, columns and arches; and fibers to make flexible materials such as rope, fabric, orjoint sealants such as caulk. Copper, lead, iron, and tin were also used in ancient times for making pipes and tools; lead pipes were used in Roman water systems, copper provided the basis for bronze using tin, and iron carburized in fires, or alloyed and tempered, was used to make early steels for tools, swords, armor, and other war implements. It is known that lead was used about six thousand years ago in Egypt. Writings of early civilizations refer to the use of iron, a rare metal derived from ores as early as three to four thousand years ago. In modern times, earlier generations learned that important structural elements, such as ship's hull structure and rigging, engine components, foundations and casings, pressure vessels and fittings, shafts, rods, plates and beams, and the like, must be constructed from materials that are durable, readily worked, and have properties allowing flexible and reliable service. Because of the marine environment's severity, knowledge ofthe nature of materials and their important physical, mechanical, thermal, chemical, electrical, and nuclear properties (nuclear properties will not be treated in this chapter), and understanding how to use that knowledge to develop reliable engineering constructions, have been major engineering concerns essential to successful developments.

In this chapter, consideration of engineering materials is restricted to solids, materials that can be used structurally to resist, support, isolate, or transfer loads. Fluids are engineering materials as well, their application also designed in engineering practice and their properties precisely defined in order to plan for their efficient use. However, fluids do not resist shearing forces, i.e., forces acting transverse to a body's axis, undergoing large deformations if so loaded, making fluids unsuited for resisting or transferring many applied loadings. Thus, only solid materials are treated here.

On first studying physics, students treat bodies as being rigid; they assume bodies do not deform when acted upon by forces. Actually, all bodies deform when acted upon by forces. To be suitable for engineering systems, loaded-material deformations must be generally small and predictable. As naturally occurring materials are found inadequate in many respects (in strength, toughness, ductility, or corrosion resistance) there has been a continuing thrust to develop new materials more suitable for systematic use in component and system designs. To select materials that will be satisfactory for resisting, supporting, or transmitting applied forces over a system's expected lifetime, mechanical and related material properties must be known and duly considered.

All force applications are dynamic to a greater or lesser degree, even when the forces are applied for long time periods and generally considered static. In addition to force magnitude, the speed or frequency with which forces are applied, and the duration of their application, influence material response. These effects, too, need to be understood and considered in system design.

In engineering systems, energy is converted from one form to another. Chemical energy can, for instance, be converted into thermal energy, then into mechanical energy, then into electrical energy, and back into mechanical energy. Materials respond to thermal as well as mechanical loadings. They expand or contract when heated or cooled, and if heated to high temperatures or cooled to low temperatures their mechanical properties generally change significantly as well. Metals lose strength and melt if heated to high enough temperatures and may become brittle and more readily fracture if cooled to very low temperatures. In designing thermal systems then, the effect of temperature on material properties needs to be known and considered.

It is generally more difficult and expensive to produce and use materials of high purity. Pure materials may not have the properties sufficient for particular applications; the properties may in fact be undesirable in some way, readily oxidizing or undesirably interacting with other materials, for instance. By understanding the chemical characteristics of materials, it is possible to mix them to form alloys to obtain the most desirable performance in specific applications.

Chemical interactions of materials typically involve sharing or exchange of charged particles and free electrons. This may result in electrical current flows between and through materials. It is important then to understand and quantify material electrical characteristics so consideration can be given to them in system design and undesirable interactions can be prevented from causing system performance deterioration. Component isolation is necessary to produce thermally efficient systems and to prevent unwanted thermal exchanges between system elements.

Material weight (expressed as density or in terms of specific gravity) is another important physical factor. All weights must be sustained by lift forces; for displacement-type ships, these forces are called buoyancy. But providing lift is not without cost. Thus lighter-weight materials such as balsa wood, aluminum, titanium, and glass-reinforced plastic (GRP) have inherent advantages due to their light weight, high strength, and competitive cost in certain applications.

Engineering material property investigations are routinely conducted and their results are systematically presented in technical literature. Selection of materials suitable for application in a system design should consider the following (not necessarily in this order):

- available material types
- material properties (density, mechanical, thermal, chemical, electrical)
- material quality, consistency of characteristics, and ease of evaluation
- service requirements materials must meet (strength, toughness, corrosion resistance)
- material workability and susceptibility to manufacturing processes
- material performance reliability in the system environment
- relative costs of finished materials

ENGINEERING MATERIAL TYPES

The list of materials available for application is extensive, with properties that are wide ranging in density, stiffness, strength, ductility, malleability, machinability, joinability, notch toughness, conductivity, interactivity, and more. Materials are typically grouped into metals, nonmetals, and composites. Metals in the pure element forms are frequently undesirable for direct use and are therefore typically combined with other elements to form alloys; due to the importance of iron as a widely used material, metal alloys are further classed as ferrous or nonferrous. Alloys are formed with characteristics making them suitable for typical or special-service environments: hot or cold, wet or dry, static or dynamic. Production processes to prepare materials for use, and a material's final shape, can affect service performance as well. Thus, most metals in marine services are alloys, whose

TABLE 2-1

Metal Element Physical Properties

Metal	*Symbol*	*Specific Gravity*	*Melting Point (F)*	*Melting Point (C)*	*Boiling Point (F)*	*Boiling Point (C)*	*Crystal Structure*[1]	*Young's Modulus*	*Shear Modulus*	*Poisson's Ratio*	*Yield Stress*
								$\times 10^6$ *psi*	$\times 10^6$ *psi*		$\times 10^3$ *psi*
Aluminum	Al	2.71	1,215	657	4,226	2,330	FCC	11.0	4.2	0.33	5.0
Antimony	Sb	6.68	1,166	630	2,624	1,440	rhomb.	11.3	4.0		
Cadmium	Cd	8.65	610	321	1,413	767	HCP	10.0	3.5	0.30	
Chromium	Cr	7.14	2,822	1,550	4,500	2,482	BCC	36.0			
Cobalt	Co	8.90	2,720	1,493	6,368	3,520	HCP	30.0			
Copper	Cu	8.94	1,981	1,083	4,680	2,582	FCC	16.0	6.7	0.35	10.0
Gold	Au	19.30	1,945	1,063	4,820	2,660	FCC	11.4	4.0	0.42	
Iron (α)[2]	Fe	7.86	2,802	1,539	5,072	2,800	BCC	29.3	11.4	0.28	19.0
Lead	Pb	11.34	621	327	3,182	1,750	FCC	2.1	0.8	0.43	1.3
Magnesium	Mg	1.74	1,202	650	2,059	1,126	HCP	6.5	2.4	0.34	12.0
Manganese	Mn	7.40	2,239	1,244	3,789	2,087	BCC	23.0			
Mercury	Hg	13.55	–38	–39	674	357	rhomb.				
Molybdenum	Mo	10.20	4,730	2,610	8,679	4,804	BCC	46.0	17.4	0.324	82.0
Nickel	Ni	8.85	2,651	1,455	5,072	2,800	FCC	30.0	11.0	0.31	20.0

Continued on next page

TABLE 2-1—*Continued*

Metal	*Symbol*	*Specific Gravity*	*Melting Point (F)*	*Melting Point (C)*	*Boiling Point (F)*	*Boiling Point (C)*	*Crystal Structure*[1]	*Young's Modulus*	*Shear Modulus*	*Poisson's Ratio*	*Yield Stress*
Platinum	Pt	21.45	3,218	1,770	7,250	4,010	FCC	24.2	9.3	0.38	
Silver	Ag	10.50	1,761	961	3,979	2,193	FCC	11.2	4.1	0.37	
Sodium	Na	0.97	208	98	1,677	914	BCC	1.3			
Tin	Sn	7.31	449	232	4,239	2,337	tetra.	6.5	2.4	0.33	3.9
Titanium	Ti	4.50	3,294	1,812	5,666	3,130	HCP	16.8	6.3	0.34	62.0
Tungsten	W	19.30	6,116	3,380	10,166	5,630	BCC	50.0	19.5	0.28	
Vanadium	V	5.96	3,146	1,730	6,386	3,530	BCC	18.0	6.7	0.34	
Zinc	Zn	7.14	788	420	1,665	907	HCP	12.0	5.0	0.11	

Notes

1 BCC = body-centered cubic
FCC = face-centered cubic
HCP = hexagonal close-packed
rhomb. = rhombohedral
tetra. = tetragonal

2 See section below on iron.

elements are chosen to enhance parent metal properties and provide for easier producibility and improved in-service performance.

In addition to traditional metals, metal alloys, and such naturally occurring nonmetals as wood and stone, a variety of plastics and ceramics are now available. Newly available also are composites, which are composed of more than one material. Glass- or fiber-reinforced plastic (GRP or FRP), for example, is a modern class of material being used in a widening range of applications; the fibers are immersed in a resin matrix fixing the composite's geometry. The fibers provide the strength while the matrix provides fiber adhesion affording axial force transfer between fibers by shear. Available fibers include, in addition to glass, more exotic products such as Dacron, Mylar, Kevlar, and carbon fiber. Table 2-1 lists a number of elements, along with their approximate specific gravities, melting and boiling points, crystalline structure, elastic moduli, and other characteristics, which are used sometimes in pure form but more typically in alloyed forms.

Basic Elements

Most metals used in marine engineering systems are alloys of aluminum, copper, iron, lead, nickel, tin, titanium, and zinc, although pure forms of these metals are often used as well. Copper, gold, and silver are sometimes called "noble" metals as they do not oxidize readily; gold and sometimes copper are found in the free state, as is platinum. Pure metals tend to be more expensive to produce, typically requiring refining to obtain the desired purity, and may be inadequate in strength; however, they often possess good ductility, conductivity, and other desirable properties. To obtain materials better suited for specific engineering use, these basic metals are typically combined with small amounts of other metal elements such as antimony, chromium, cobalt, magnesium, manganese, and tungsten to improve strength, machinability, corrosion resistance, and other properties, while not seriously reducing ductility, toughness, or other attributes and increasing fatigue resistance if possible. In addition to the metal elements, such nonmetals as carbon, silicon, and sulfur are used in developing alloys that have desired properties.

In the pure state, the metallic atoms arrange themselves in randomly oriented crystal structures as they solidify into grains, with many dislocations developing between the crystal grains. Of particular significance to marine materials are the face-centered cubic (FCC), body-centered cubic (BCC), hexagonal close-packed (HCP), and tetragonal crystalline structures. Antimony forms in a rhombohedral lattice. Figure 2-1 illustrates several crystal structure atomic arrangements. It is noteworthy that all the FCC metals have generally good ductility, a desirable characteristic for engineering system components, whereas HCP and some BCC elements are brittle or become so at low temperatures. The minimum elongation of materials before rupture, a measure of material ductility, is frequently specified as a criterion for material selection.

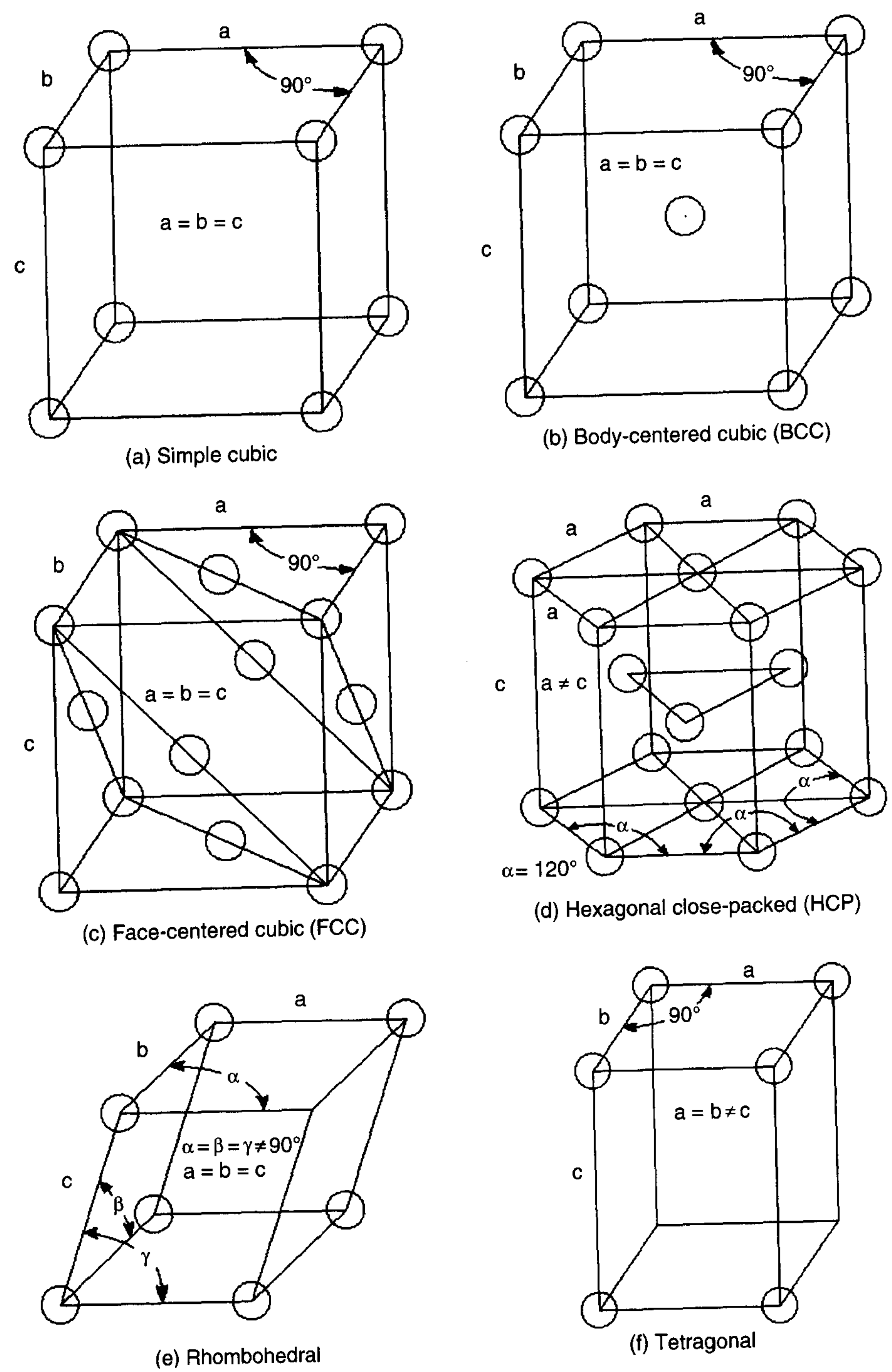


Figure 2-1. Crystal atomic structure arrangements

Commercially useful metals are usually found in igneous rock ore formations composed of mixtures of metallic and mineral compounds. Some typically occurring, commercially exploitable mixtures such as nickel-cobalt-silver, iron-magnesium, zinc-cadmium, lead-chrome, lead-molybdenum, and aluminum-silicate, are found in particular regions of the world. The metals are usually found in such compound forms as arsenates, sulfides, and oxides or in combinations with other metals. Such metals as copper, gold, platinum, and silver are frequently found or may occur only in a free state. Meteorites are about 90 percent iron and 6 percent or so nickel, and it is believed the earth's core has a similar content.

Reduction of ores to prbduce metals typically entails crushing, washing, concentrating, smelting, roasting, fusing, dissolving, precipitating, electrolytic reduction, or refining. To carry out these processes, suitable fluxes, oxidizers, reagents, and other chemical substances must be brought into play. Some limited notes on these are given in the following sections. More detailed information on processes and material properties can be found in inorganic chemistry reference texts such as those listed at the end of this chapter (the older works are still useful in understanding materials and their properties). Many errors and inaccuracies are found in the literature and no complete reference is known; only approximate values are given here to provide readers with reasonably reliable information.

ALUMINUM

Aluminum is the most abundant metallic element in the earth's crust, being the third element in abundance in igneous rocks at about 8.1 percent. Aluminum oxide is the fourth most abundant compound in the earth's crust. The element is typically found in nature combined with others, such as potassium, in the form of silicates. Bauxite, Al2nH20, is an important component in the metal's production, which is usually by electrolysis of its aluminum oxide; aluminum production facilities are typically located near hydroelectric or coal-fired power plants where electric power is less expensive. As produced, aluminum is nearly pure, about 99+ percent, and lacks many structural abilities that would make it most desirable. At the atomic level, its crystalline structure is face-centered cubic (FCC) and this promotes certain properties: it is very ductile, even at very low temperatures, and very malleable (only gold is more malleable). Aluminum can be rolled into very thin sheets, called "foil," and is easily extruded into specially shaped sections. This lends itself to production of structural and architectural shapes, as for trim items. *Mter* extrusion, the shape is typically passed through straightening rolls, removing twists or curvatures. Due to its softness and low melting temperature, aluminum is readily cast and is found in many shipboard outfitting items, though usually as an alloy.

Relatively soft, aluminum has a Brinell hardness of about 20 and a relatively low yield stress, about 5,000 psi (34.47 MPa), and ultimate strength,

about 13,000 psi (89.63 MPa) in its annealed condition. Being quite ductile, it undergoes a 35 to 45 percent elongation before rupture. Due to its low strength and light weight (specific gravity = 2.7), the pure metal cannot be used to full advantage. When alloyed, aluminum can be heat treated or mechanically worked to obtain more desirable properties. However, the strength of heat-treated or cold-worked material tends to revert to the annealed condition if heated above its recrystallization temperature. Easily machined, it takes on a high initial polish; the surface rapidly develops an oxide coating which then retards further oxidation.

Nearly pure aluminum (about 99.5 percent) is used for electrical conductors. Having less electrical conductivity (63 percent, i.e., electrical resistivity is 1.6 times greater) than copper, which has a specific gravity of 8.94, but with lighter weight, it is pound-for-pound about twice as good a conductor. However, its thermal expansion coefficient is 1.55 times that of copper; if in-service thermal variations are significant, care must be taken to keep electrical connection devices tightened.

At ambient temperatures, aluminum oxidizes readily, forming an oxide coating on its surface that protects (shields) the metal from further oxidation, even when the metal is melted. At higher temperatures, however, the oxide, in the presence of the liquid metal, may be ignited by an arc or flame. Thus, if electrical arcs are formed, as in a faulty electrical connection, the metal will readily melt and can ignite (undergo extremely rapid oxidation), initiating fires in adjacent combustible materials. More discussion of this metal and its properties is given below in the section on alloys.

ANTIMONY

Known of in ancient times, antimony was not clearly identified until the seventeenth century. While not rare, its occurrence is still only about 0.002 percent, about the same as for cadmium. It is mainly obtained from its sulfide mineral, stibnite, by crushing and roasting. An inexpensive, relatively hard (about 2.2 times as hard as tin), but easily scratched and brittle metal of bluish-white color, antimony flakes readily. Having a low melting temperature, the metal alloys easily with most metals and is used as a hardener of tin in making babbitt, antifriction metal, pewter, and white metal, and of lead in making hard metal, type metal, and battery plate. It expands on solidifying and is thus useful in reducing shrinkage, better filling mold volumes when cast. Tin-antimony solder, with a melting point of about 450°F (232°C), is suitable for soldering copper tubing joints in potable water plumbing systems. Antimony's compressibility is about the same as that of lead.

CADMIUM

Typically, cadmium is found in zinc ores in a concentration of about 0.5 percent, though it is sometimes found in a more pure form as a cadmium sulfide. It is often produced as a by-product of zinc production as it is more volatile and more easily reduced in the zinc smelting process; most cadmium, however, is produced by the electrolysis of zinc. Cadmium is a silver-white metal that is not as hard as but more ductile and malleable than zinc. It polishes to a high luster but tarnishes gradually as the surface oxidizes. It is often used for anticorrosion plating of wire, tools, and other steel products. If the items are subsequently heat-treated, the cadmium alloys with the iron, giving about the same corrosion resistance as is found in zinc plating (galvanizing). Cadmium becomes rather brittle at higher temperatures and is frequently used as a substitute for tin in bearing metals. It is sometimes used in small amounts to.strengthen copper (otherwise relatively weak), as small amounts do not significantly increase copper's resistivity.

CHROMIUM

A highly corrosion- (oxidation-) resistant material, chromium is used extensively to electroplate steel and other metals to reduce surface oxidation and to make corrosion-resistant chromium (stainless) alloys of steel for service at normal as well as elevated temperatures. As a pure metal, it has a luster similar to that of platinum, taking on a brilliant polish. Chrome ore is found in igneous rocks in a concentration ofless than 0.04 percent, most importantly in the form of a ferrous chromite, and it is usually obtained by first fusing the ore with sodium carbonate to form a sodium chromate. It can then be reduced in an electric furnace. Harder than iron and annealed nickel and a little less so than copper, chromium has the effect ofincreasing the hardness of steel when the steel is rapidly quenched, and it increases the wear resistance of cutting-tool steels when 1.5 to 2.5 percent chromium is added to the steel.

COBALT

Cobalt, a silver-gray metal, is ductile and malleable and is found mainly in nickel-cobalt-silver ores, although common cobalt minerals such as those with arsenic and sulfur, like CoAsS, are more generally found with iron ores. The ore is smelted with a suitable flux in a blast furnace, which produces the unrefined liquid silver metal and an amalgam of arsenides of several metals such as cobalt, iron, or copper. This amalgam is then subjected to various roastings, using salt to form a chloride of cobalt which can then be extracted with water. Cobalt can be precipitated as a hydroxide, which in turn can be reduced to the metal with carbon. It has important uses in making high-strength, structural alloys for use in high-temperature applications. As an alloy with chromium and tungsten, it is important in making high-speed cutting tools.

COPPER

Copper is sometimes found in its elemental state, certainly providing the basis for its use by early civilizations in making bronzes. Copper also (in some

regions predominantly) occurs in the form of sulfide and oxide ores. Crushed native copper ores can be mechanically concentrated and then purified by melting with an appropriate flux. Oxide ores are heated in a furnace and smelted with coke and a suitable flux. The sulfide ores are more difficult to reduce as there is typically some iron present. The purest copper can be obtained by electrolytic processing to produce nearly 100 percent pure copper. Copper has a face-centered cubic crystalline structure and retains its ductility through a broad temperature range, having an elongation of 45+ percent at ambient temperatures before rupture.

As a pure element, its thermal and electrical conductivities are very good, though not as good as those of silver. Copper is competitively priced in world markets and is extensively used in electrical applications such as conductors and motor windings, but primarily in its nearly pure form, as small amounts of impurities can significantly reduce its electrical conductivity.

Although copper is quite ductile, in its annealed state it is quite weak and soft, with a yield stress of less than 10,000 psi (68.95 MPa) and a Brinell hardness number of about 40. Because of its low yield stress and ductility, it is easily extruded into tubes and rolled into sheets. It work-hardens when plastically deformed, raising its yield strength, but becomes more brittle and less fatigue resistant, making it unsuitable for dynamic loading as in a vibratory environment. Because of copper's relatively low recrystallization temperature (about 250°F [121°C] depending on how much plastic straining it has experienced), work-hardened items can be annealed if desired by heating above the recrystallization temperature and quenching in water, thus allowing grain growth and restoring its softness and ductility. Because of its very good thermal conductivity, it is extensively used in heat transfer components, particularly tubing products.

Copper is highly resistant to corrosion under ambient conditions and is extensively used for alloys. As discussed below, the alloys typically retain this corrosion-resistant property.

GOLD

Produced for over six thousand years, gold is a relatively rare metal, constituting only about 5 parts per billion in the earth's crust, and with quite uneven distribution. It frequently occurs in rich concentrations in native form as flakes and nuggets in lode and alluvial deposits. It is found as a compound only in various telluride forms or in quartz veins. Much metal is recovered as a by-product of extraction processes during production of copper, lead, nickel, silver, and zinc. Most countries have some gold; almost half have some production. Seawater also contains some gold, about 0.014 micrograms per liter. Having a distinctive deep yellow color and an FCC crystal lattice, it is relatively heavy, strong, and soft and is the most malleable and ductile of metals (one cubic inch of pure metal can be rolled or hammered into a sheet of about 1,628 square feet, making it suitable for thin-coating objects as in gold leaf applications).

Pure gold is so soft and ductile that it can be cut with a knife. However, its ductility is destroyed by as little as a 0.05 percent addition of antimony, bismuth, or lead; it is usually hardened for commercial uses by adding silver or copper, its fineness then being expressed as a percentage of gold in the alloy (24 carats represents 100 percent gold). A good electricity conductor, its resistivity is about 40 percent greater than that of copper. It does not readily corrode or tarnish and is readily solderable, making it useful for coating electrical contacts wpere stable contact resistance is important, as in electronic circuits. Its emissivity is quite low, about the same as tungsten and less than platinum, but about 20 percent higher than silver. Very chemically inert at ambient temperatures, it does react with aqua regia, a hydrochloric and nitric acid mixture.

IRON

The most commercially important metal, iron has a high and widespread occurrence in the earth's crust, at about a 5 percent concentration in igneous rocks. Its concentration may be higher in lower-lying rocks; in the earth's core, it is believed that iron is the predominant element. Meteors are about 90 percent iron. It is the fourth most prevalent element and the second most prevalent metal after aluminum. Known to prehistoric man, who found meteorites, iron was obtained from ores as early as three to four thousand years ago. Iron is present in a wide range of oxide and sulfide minerals, but is usually obtained from the oxide ores hemetite (Fez03) and magnetite (Fe304), as they are plentiful and readily broken down into iron and oxygen when properly heated. Iron takes on several crystal structures, alpha (a), beta (P), gamma(y), and delta (8), depending upon temperature. Pure iron at room temperature is very ductile and strains up to 45 percent in a 2-inch gauge length before rupture. The iron alpha crystalline structure is body-centered cubic (BCC) at ambient temperatures, but at temperatures above 1,670°F (9lO°C), it changes to an FCC structure as gamma iron with increased ductility. It has a tensile strength of about 30 ksi and a Brinell hardness of about 60. At low temperatures, pure alpha iron becomes brittle and somewhat unsuitable for engineering uses, thus the need for alloying iron with toughening elements.

Alpha iron at room temperature is called ferrite, a relatively soft, tough, highly paramagnetic, grayish-white metal. It can be polished to a high finish, making it look like platinum, but it soon reacts with its environment, forming a surface oxide and losing its luster. As its temperature is raised, it transforms to beta iron, losing most of the paramagnetism while maintaining its BCC crystal structure. As gamma iron, it readily combines with iron carbides. The gamma iron structure can be maintained at room temperature, if rapidly quenched, but this will be discussed further under alloys when considering steel. If the temperature is raised further still, it transforms into delta iron, returning to the BCC lattice structure. Thus, temperature control is very important in obtaining desired properties in iron and its alloys.

Pure iron, 99.9+ percent, has but limited commercial use due to a number offactors. It is difficult to produce (thus costly), it has a high reactivity with other elements such as oxygen, and its physical properties are less attractive than those of the various steels. Pure iron, though, is quite ductile under ambient conditions, is easily forged or drawn, and can be joined by welding when raised to a white-heat temperature. A small amount of iron is produced for special uses, such as catalysts and special magnets. Pure iron production is typically carried out by reducing the oxide in a blast furnace in the presence of carbon (as in coke) and limestone (which takes out such noniron elements as silica) to produce the molten metal and slag which floats on its surface. In the blast furnace, alternate layers of ore, coke, and limestone are charged and air is forced upward through the layers, providing oxygen, bringing the coke to a high heat, and melting the ore. The molten metal forms at the bottom of the furnace and is drawn off periodically and cast into "pigs," thus the source of the term *pig iron*. This product is the raw material from which most iron and steel products are developed. Most iron today is finally produced as steel, an alloy of various compositions, depending upon the amounts ofthe particular alloying elements introduced. Continuous casting production of steel ingots is a highly efficient method in use today for production runs of structural steels. This will be further discussed below in the section on alloys.

LEAD

Although lead rarely occurs naturally, and only to about 0.0002 percent (2 ppm) in the earth's igneous rock crust, it is easily won from its ores and has been produced for nearly six thousand years and on all continents. A relatively soft (Brinell hardness number 4.2), ductile (64 percent elongation in a 2-inch gauge length), bluish-gray metal of low melting point, the commercial importance oflead is exceeded by only a few metals such as aluminum, copper, iron, and zinc. Having a shiny surface when first cut, lead oxidizes readily, forming a black, lead oxide film, PbzO. Produced mainly from its sulfide, PbS, or from lead sulfate, PbS04, it is crushed, roasted to produce an oxide, and then heated again with more lead sulfide to produce lead and sulfur dioxide gas.

Lead has commercial value in making lead-acid batteries, babbitt, solder, type metal, pewter, radiation shielding, and sheathing for electric cables (though this is being largely displaced by plastic sheathing), and other uses. Because of its corrosion resistance, lead is used for making linings for tanks, piping, and equipment in chemical plants and systems. Pure lead has but few structural uses as both its tensile strength, at about 2,000 psi

(13.97 MPa), and its fatigue limit are quite low. When stressed to as little as 200 psi (1.397 MPa), it will exhibit creep, making it unsuitable for resisting sustained loads. Lead is very malleable, can be rolled into sheet form, and is easily cast due to its low melting point. Its structural properties can be significantly improved by modest additions of antimony and tin, as when being used in solders and babbitts, and it is often added to steel, brass, and bronze to improve castability, machinability, and corrosion resistance. Lead compounds, such as lead dioxide, lead carbonate, lead chromate, and lead sulfate, have uses in paint pigments, glazes, and litharge for crystal making. However, the hazard oflead poisoning to the workforce and consuming public has caused lead to be displaced by nontoxic materials over recent years.

MAGNESIUM

The lightest weight structural metal, magnesium has a gray-white luster when freshly cut but oxidizes quickly, causing the surface to tarnish. Relatively plentiful, at a concentration of about 2.1 percent in igneous rocks (about one-fourth that of aluminum), it is the third most abundant structural metal, next to aluminum and iron. Although somewhat brittle, and difficult to shape at room temperature, by heating to 4000-6500F (200°-340°C), magnesium can be extruded into tubes, rods, and various structural shapes and hammered and rolled into sheets and plates. It is readily machined at high speed and can be cast or fabricated using all known methods. It has good stiffness, a little more than half that of aluminum, and good resistance to impact and shock loadings. Used in alloys of aluminum and zinc to improve malleability, machinability, and weldability, it easily alloys with aluminum, as their melting points are very close.

Magnesium is chemically active with other metals and occurs in many mineral compounds, magnesium oxide combining with calcium carbonate to form dolomite, and magnesium chloride combining with potassium chloride to form carnalite, for instance. Magnesium makes up about 0.13 percent of seawater, thus there is a plentiful worldwide supply in addition to that in the minerals. Magnesium is extracted from seawater by treating it with lime, to produce a magnesium hydroxide, followed by treatment with hydrochloric acid. The resulting magnesium chloride is dried and reduced to the liquid metal by electrolysis. Liquid magnesium is typically cast into ingots for further processing into final shapes or use in making alloys. An alternate production method, called the Pidgeon process, mixes crushed and baked dolomite, producing calcium and magnesium oxides, with ferrosilicon. The mixture is pressed into blocks and heated, driving off vaporized metal into a condenser from which the liquid metal is obtained.

Since magnesium oxidizes readily, openly burning when heated, its powder oxidizes extremely rapidly. It has been used extensively to make

incendiary products, such as flares, as it gives off a brilliant bluish-white light. Keeping shop areas clean of magnesium dust is vital.

MANGANESE

Hard, brittle, and ranked ninth in abundance of metals, manganese occurs in a concentration of about 0.1 percent in igneous rocks, principally as manganese oxide with iron and other impurities. However, some ore deposits have concentrations as high as 40 percent. It is difficult to obtain the metal by reduction with carbon, as it forms a series of carbide solid solutions. It is reduced by grinding the oxide ore and mixing it with powdered aluminum, the mixture then being burned in a furnace to produce the metal and a slag. A manganese of higher purity is produced by an electrolytic process in which ore is first treated with sulfuric acid to produce a manganese sulfate that becomes the process electrolyte. The electrolytic cell typically has a lead alloy anode and a steel alloy cathode, producing manganese of99.9 percent purity.

Manganese, reddish-gray in color and softer than iron, is an important element in production of most steels, increasing hardness, improving toughness, increasing impact load resistance, and improving the ability to form and hot-work steel in rolling and forging operations. It helps control adverse effects of sulfur and somewhat increases steel's ability to harden. When manganese concentrations are raised above the 1.6 to 1.9 percent of 13xx manganese steels, say up to 12 percent, wear resistance of steel is increased, though such steels are expensive to produce. It is also an important element in making copper alloys such as manganese bronze; most aluminum and magnesium alloys contain manganese. Manganin, a copper-nickel alloy with 12 percent manganese, has a low temperature coefficient of resistance and is important in making electrical instruments.

MERCURY

Mercury is a silvery-white, "noble" metal that is in a liquid state under ambient conditions, having a low melting, but a high boiling temperature. It is principally derived from mercury sulfide ore (HgS or cinnabar), although it also occurs as amalgams of gold and silver and in other complex minerals such as chlorides, selenides, and tellurides. Roasting of cinnabar produces a mercury vapor which can be condensed, filtered, and washed to produce the liquid metal. Liquid mercury has a low vapor pressure, a nearly constant expansion coefficient in the ambient range, and a high surface tension. It doesn't wet glass, forming a convex upper surface (meniscus) when in a glass tube. Its high density, along with the above properties, makes it useful in mercury barometers, thermometers, manometers, and other scientific instruments. Its liquid and vapor characteristics and electrical conductivity make it useful in electrical switches, mercury vapor lamps, and even as the working fluid of heat engines, though mercury vapor is difficult to contain and it is toxic. Mercury fulminate, Hg(CNOh, is used in percussion caps as it explodes when struck. Amalgams of gold, silver, and tin are used significantly in dentistry.

MOLYBDENUM

Molybdenum does not occur in the free state; it is most frequently found as a sulfide (MoS_2 or molybdenite), and it also occurs as lead or iron molybdenates ($PbMoO_4$ or $Fe_2[MoO_4]_3 \cdot 7.5 H_2O$). It is a hard, silver-white metal having a high melting point, only tungsten being higher, and is tough though ductile with a high modulus of elasticity and tensile strength (up to 350 ksi [2.41GPa] when drawn into wire). These properties make it useful in toughening and strengthening steel and thus it is sometimes used as an alternate to tungsten, its cost being relatively competitive. Its ductility is seen by its 35 percent elongation in a 2-inch gauge length at rupture, although when drawn into wire this is reduced to as little as 2 to 5 percent. Upon being annealed, the recrystallization temperature is about 1,650°F (900°C), and the ductility of the wire can be improved to yield elongations of 10 to 25 percent.

The metal is generally produced by roasting the oxide with lime to convert it into calcium molybdate, which can be used directly in producing steel alloys. The pure metal is produced by reducing the oxide with aluminum or hydrogen. The metal is produced in the form of bars, billets, foil, sheet, strip, wire, and seamless tubing. Its importance to steel production can be appreciated by noting that a 1 percent addition to low-carbon steel will double the steel's strength at 900°F (482°C), the nominal temperature of main steam lines in marine power plants.

Molybdenum has good resistance to corrosion by such substances as sulfuric acid, where even glass-lined equipment may fail, though it oxidizes rapidly at elevated temperatures and must be protected. It bonds well when hot-sprayed on steel and most aluminum alloys and it can be welded, but, when finished, its toughness makes it essentially unmachinable, so finishing must be done by grinding.

NICKEL

Although nickel occurs in the earth's crust as only about 0.02 percent of igneous rocks, it constitutes about 6 percent of iron meteors and it is believed that this is approximately its content in the earth's molten core. The two most important sources of nickel are in deposits of nickel sulfide mixed with iron sulfide and in nickel silicate. Copper ores typically contain some nickel, which also occurs in other complex sulfides, sulfates, silicates, arsenides, arsenates, and oxides. The ore is smelted with a suitable flux in a blast furnace, which typically produces a matte of iron, copper, and nickel sulfide. A converter process can be used to separate out the nickel as nickel sulfide, which in turn can be roasted into an oxide and then reduced to

metal by carbon. The nickel can be electrolytically refined by acting as an anode in a nickel sulfide electrolyte.

Nickel has an FCC crystalline structure with very good ductility over an extensive temperature range. It is highly resistant to corrosion and is consumed in large quantities by the nickel plating industry (one of the larger consumers). It is most extensively used as an alloying element in making steels, and to a lesser extent in making nonferrous alloys such as monel. As an alloying element for steel, it is used to make 9 percent nickel steel, an austenitic steel, which has good ductility at very low temperatures, as in cryogenic applications for liquified natural gas (LNG) tank plating. It is also used with other elements such as chromium and molybdenum.

PLATINUM

A silvery-white, FCC, nonmagnetic, precious metal having medium hardness (Brinell hardness of 42), good ductility, and malleability, platinum is highly resistant to oxidation under ambient conditions, making it suitable, even at elevated temperatures, in corrosive environments that oxidize metals. Platinum has been found in nickel ore deposits that often contain gold, copper and other ores, but it is typically found in elemental form occurring as grains or nuggets, similar to gold, and is obtained by washing and concentrating sands. It is a very heavy metal, having a specific gravity twice that of lead and 10 percent greater than tungsten, and a very high boiling temperature (only molybdenum and tungsten of those listed in table 2-1 are higher). Platinum can be separated from some impurities by flotation and from others by melting and evaporation. After this, chemical reactions can be used to eliminate remaining impurities.

Although used in making jewelry, its engineering importance lies in making instruments, as high-temperature thermocouples (with one wire platinum, and the other 90 percent platinum, 10 percent rhodium for instance), electrical contacts, and heating elements, as a catalyst in inducing chemical reactions, and often to make an anode in ship cathodic protection systems. Its coefficient of thermal expansion is close to that of glass, so it has been used in making electronic elements in tubes, with the platinum encased in glass.

SILICON

A very important element in the mineral world, silicon is never found in the free state even though it constitutes about 87 percent of the earth's crust. Its occurrence is next to the element oxygen, forming almost 26 percent of the outer portion of the earth's surface. In crystalline form, it takes on a gray luster and dissolves readily in many molten metals, having a melting temperature of about 2,570°F (1,410°C). Silicon occurs largely in oxide form as silicon dioxide, or silica. It is also found as quartz glass, sand, flint, and agate and has much importance to the steel industry, though it is produced and used primarily there as a ferrosilicon, obtained by reduction of siliceous iron in an electric furnace. Silicon is added to manganese steels in the amount of 1.8 to 2.2 percent to make spring steel and is added to nickel and chrome-nickel steels in amounts of.5 to .75 percent to produce heat- and corrosion-resistant steels. Silicon also is added to white cast iron to produce the more commonly used gray cast iron.

SILVER

A metal frequently referred to as "precious," silver has been produced since ancient times, even though it constitutes only about 0.05 parts per million in the earth's crust. Silver is usually won from ores found as silver sulfide, with sulfides of copper, lead, nickel, and zinc. Metal production comes mostly as a by-product of refining these more plentiful and less costly metals. In electrolytic purification of copper, a sludge residue is produced that contains silver. Smelting oflead from its sulfide also provides a source of silver sulfide. The silver can be obtained by forming an amalgam with mercury and then distilling the mercury, or by converting the silver sulfide into a silver chloride, roasting it with salt, then carrying out a leaching process. Other processes are also available.

Silver, the whitest of all metals, takes on a high luster and holds a fine polish. In the polished state, the emissivity of silver is quite low, lower than that of aluminum, copper, tungsten, or gold. Accordingly, it is extensively used for electroplating reflective surfaces like mirrors and Thermos bottles, where reflectivity and emissivity are most important. It has very low resistivity, even lower than copper, and is very ductile. Pure silver is so soft, in fact, that its pure form is not structurally useful, its hardness being only a little greater than that of gold. Silver fineness is defined as ten times the silver percentage. Sterling silver, for instance, at about 93 percent silver, has a fineness of 930, the remaining portion being of another metal, typically copper. Jewelry silver often contains as much as 20 percent copper, yielding a silver fineness of 800.

SODIUM

A soft (can be cut by hand with a knife), silvery-white and lightweight metal with a specific gravity ofless than one, sodium floats on water. The sixth most abundant element, it is highly reactive, readily forming compounds with its environment. Due to its reactivity, the metal cannot be kept in a normal atmosphere, but must be stored in a nonreactive environment, such as in kerosene. It is found not only as salt in seawater, but also in dry lake beds and salt domes in many places of the world. Combined with other solid elements such as aluminum, boron, carbon, and gases such as flourine, nitrogen, and oxygen, it forms such commercially important

minerals as borax, saltpeter, and soda ash. It is produced by electrolysis of its chloride salt but has few direct practical uses as a metal. Since it oxidizes readily, however, it is used to deoxidize such metals as chromium, copper, lead, iron, and their various alloys. The atomic submarine *Nautilus* originally used liquid sodium as its reactor's primary coolant.

SULFUR

A nonmetal, sulfur constitutes only about 0.1 percent ofthe earth's crust, though there are numerous mineral combinations with heavy metals, other than gold and platinum. The most extensive mineral deposits are those of iron sulfides. Sulfur also occurs as free deposits in severallocations, such as salt domes. As an element, sulfur is carefully controlled in steel making, being generally kept to less than 0.055 percent except for use in a class oflow-carbon, free-cutting steels where the sulfur content is increased to as much as 0.3 percent in order to improve the steel's machinability. An important use of sulfur is in production of sulfuric acid, though there are a number of other important industrial uses, such as in vulcanizing rubber and for dusting agricultural crops and horticultural plants.

TIN

Tin, a silvery-white, soft, ductile, and malleable metal that does not work-harden, has been used since ancient times. It is easily rolled, spun, and extruded and looks like aluminum when freshly cut. Tin takes on a high polish with a slight bluish tinge due to its oxidation and its development of a protective surface film. With this oxide film developed, tin is quite resistant to further oxidation in a normal atmosphere, though it reacts to strong acids and bases. Constituting only about one part per million in the earth's crust, it occurs in workable tin oxide deposits (SnO_2, or cassiterite) and in alluvial deposits.

Cassiterite is more than twice as heavy as the sands it is found in, so sedimentation allows initial separation and concentration. Production of tin from cassiterite entails roasting with coke to produce tin with some impurities, followed by refining. In electrolytic refining, impure tin constitutes the anode and the refined tin collects on the cathode. Several complicating factors are involved in the smelting process because tin is very fluid at smelting temperatures. When produced, the tin is cast into ingot form (each ingot weighing about 75 pounds), a common form for shipping to users. Ingots of greater or lesser weights are also commercially available. The American Society for Testing Materials (ASTM) has set various tin grade specifications in which the levels of impurities are specified. For instance, grade AAA is electrolytically purified and is 99.98 percent pure.

Cast and used as tin anodes in the electroplating process, tin is extensively used in plating steel, foil, and packaging materials. Detinning of scrap tinplate is an important source of tin. Having a relatively low melting temperature and good wetting ability when liquid, it is widely used in hot-dipped metal coating. Wetting power also makes its use in joining metals possible; in solder, a 50-50 mixture of tin and lead provides the strongest joint onow-melting solders. In addition to use in making babbitts and bearing alloys, tin is used to create coatings on a wide variety of fittings and fixtures; these highly polished surfaces on brass, aluminum, and steel maintain decorative appearances in normal atmospheres.

TITANIUM

A very light, strong, ductile (28 percent elongation in a 2-inch gauge length), and weldable metal of silver-white color, titanium occurs at about 0.43 percent concentration in the earth's crust, primarily as a titanium oxide (TiO_2 or rutile), and as a ferrous oxide of titanium ($FeTiO_3$ or ilmenite). Eighth in abundance among metals in igneous rocks, and fourth most abundant structurally useful metallic element, titanium does not occur in a free state but is found in compounds of calcium, magnesium, manganese, and lead in soils, clays, sand, water, oil, coal, and atmospheric dust. Titanium is prepared with difficulty by reduction of its oxides with carbon and chlorine gas to produce a titanium tetrachloride, which is then mixed with good oxidizers like magnesium or sodium. When the titanium tetrachloride is mixed with magnesium, it is then heated in a mild steel vessel, as an electric arc furnace, under an argon or helium atmosphere, to produce melted metal which is then cast in ingots.

As a structural metal, its abundance is exceeded only by aluminum, iron, and magnesium. Its strength is comparable to that of steel and its alloys are much more corrosion resistant than aluminum, exceeding even that of 18/8 nickel-chrome stainless steels, particularly in the salt spray conditions of the marine environment. Titanium transforms from HCP to BCC at 1,560°F (849°C), allowing heat treatments comparable to steels. Improvements in production methods have reduced its cost and, with its high strength/weight ratio, have made it of increasing importance in high-performance marine vehicle structures and components.

TUNGSTEN

A metal having a higher melting point than any other, tungsten is quite tough and is used to make electric lamp filaments and to increase the toughness of steel. It does not Occur freely in nature, but primarily as oxides with iron and manganese at a concentration of about 0.00005 percent. It is also found as oxides with calcium, lead, and copper. It is typically produced by being fused with sodium carbonate, extracted with water, dissolved in hydrochloric acid, and precipitated in tungstic acid form. Reduced by heating with carbon and compressed, the particles are sintered with an electric current to form metal rods. When drawn into metal wire, tungsten has a tensile strength greater than any other metal. When alloyed with carbon to form tungsten carbide, it is extremely hard, though brittle, and is useful for making cutting tool bits. To reduce brittleness, it is frequently embedded in cobalt to make the hardest of metals.

VANADIUM

A gray or silver-white metal that oxidizes but does not tarnish readily, vanadium is used extensively to add tensile strength to and harden steel by removing oxygen from the molten metal. Vanadium occurs in igneous rocks only to a very small extent (about 0.00017 percent) as oxides in compounds with other elements such as lead, zinc, copper, and potassium. It is leached from its ore with hydrochloric acid, which in turn is mixed with ammonium chloride and then roasted to obtain an oxide. This can then be reduced in an electric furnace with carbon to produce ferrovanadium. The steel industry uses this ferroalloy in concentrations ranging from 35 to 55 percent in producing killed steels, making the additions to the molten steel while still in the ladle in order to reduce and thus control the potential of vanadium to oxidize. Because it is such a powerful oxidizer, the vanadium content of steels is generally kept below about 0.2 percent.

ZINC

Zinc ores are primarily composed of sulfides and carbonates and most contain small amounts of cadmium and some lead. Most zinc is produced by smelting in a reduction process with carbon, though chemical leaching followed by electrolytic reduction is a common production process as well. In the smelting process, the ores are roasted to convert them into oxides, which are then mixed with coal and heated to about 2,200° to 2,400°F (1,205° to 1,315°C). As this temperature is well above the boiling point of zinc (about 1,665°F, or 907°C), the liquid metal is vaporized and then condensed as it passes from the retort. Ores typically have some compounds of lead included, the metal produced also typically containing some lead, as it is difficult to separate it out. If zinc of high purity is required, it may be obtained through the electrolytic process. In this process, the sulfide ore is carefully roasted to convert the sulfide to a sulfate, which is then leached with a weak sulfuric acid and mixed with a small quantity of zinc dust to help precipitate nobler metals. The electrolysis uses a high current to cause the deposition of zinc in a very pure state. This high-purity zinc has good commercial value.

Zinc has a white metallic surface when polished, but it oxidizes quickly to a blue-grayish color. At ambient temperatures it has a hexagonal close-packed crystalline structure. It is hard and somewhat brittle, but it has a recrystallization temperature of 50°F, so that at temperatures of 100° to 150°F (38° to 65°C), grain growth takes place and the metal can be rolled or drawn into desirable forms such as plate and rod. At higher temperatures, however, it becomes quite brittle again. Plate and rod are the forms most frequently found in marine applications, where they are used as anodes in cathodic protection systems. With the melting temperature of zinc being relatively low, it is used extensively for "hot dip" galvanizing of steels to give corrosion protection. In some cases, the steels are double-dipped to improve protection.

Zinc oxide is an important white pigment of anticorrosion paints and is produced by oxidation of the metal or by heating oxidized ore mixed with carbon in an air blast. The carbon reduces the ore, producing zinc vapor, which reoxidizes and then condenses as a dust in a flue filter bag. This white zinc oxide dust is then reclaimed and made available for use as pigment.

Alloys

Most metallic materials used in marine engineering applications are alloys, base metals that have had greater or lesser amounts of other elements mixed with them. Alloying elements are added to enhance particular base metal properties to make them more suited for specific applications. Because excessive amounts of certain alloying elements may deteriorate base metal properties, it is important to control alloys carefully. Shipbuilding steel, for instance, needs to have strength, ductility, and toughness to resist cracks and their propagation throughout its ambient operating temperature range, and must be weldable and fatigue resistant. Ship classification societies, such as the American Bureau of Shipping (ABS), set down chemical and mechanical requirements the steels must meet and oversee the metal production processes. ABS grades are substantially linked to the material specifications of the ASTM as set forth in the ABS *Rules for Building and Classing Steel Vessels.*

ALUMINUM

As produced, aluminum is nearly pure, about 99 percent, and lacks many of the structural abilities that would make it more desirable. That is, it has a relatively low yield stress, about 4,000 to 5,000 psi (27.6 to 34.5 MPa), and ultimate tensile strength of about 12,000 to 13,000 psi (82.7 to 53.7 MPa), though it is quite ductile, having a 35 to 45 percent elongation before rupture, depending upon test sample thickness. When alloyed and tempered or heat treated, its relative strength-to-weight ratio is significantly improved-as much as three times better than steel's. By alloying aluminum with small percentages of other elements, such as chromium, copper, iron, magnesium, manganese, silicon, titanium, or zinc, then mechanically

or thermally treating the material, mechanical properties can be significantly improved, making it more suitable for many marine applications, but most commonly ornamental and outfitting items rather than main structural purposes. It should be noted that several large passenger vessels and a number offreight vessels and naval vessels have had aluminum deckhouses. The primary purpose for such construction has been to save topside weight and improve vessel stability. Low maintenance needs of aluminum, when properly isolated and protected, also offer significant benefits. Aluminum is easily cast, drawn, or extruded into shapes and rolled into useful products such as plates. It can be joined by bolting and riveting as well as by inert gas welding; welding, however, generally degrades its temper and yield strength in the heat-affected zone, causing that portion to revert to the annealed level for the alloy.

A particularly valuable property of aluminum is its retention of ductility at very low temperatures. Thus, it has been used extensively in construction of spherical and rectangular tanks used in near-atmospheric pressure LNG transportation and storage (-260°F or -162°C) and liquid hydrogen storage tanks (-423°F or -253°C). The 5083 aluminum alloy was developed particularly for these purposes; the 7039 alloy has 20 percent better welded strength yet.

A primary difficulty in the marine application of aluminum is its low melting point. As the metal temperature approaches its melting point, the material's strength decreases sharply. It will melt and undergo rapid oxidation at temperatures typical of shipboard fires. Thus, thermal isolation of aluminum, when used in ship structural applications, is an essential requirement of U.S. Coast Guard regulations (see 46CFR72.05-5D. A further difficulty lies in the fact that aluminum readily creates a galvanic cell with other materials such as steel when current paths are available for the electron flows between the two metals; sea salt in a humid atmosphere creates such a path. In combined use, careful isolation of the two materials must be established and maintained, using dielectric materials between mating surfaces and high-integrity coatings, such as epoxy-based paints, to seal the surfaces.

Another special consideration is in respect to aluminum's thermal expansion and contraction, which is about twice that of steel. In the case of the large temperature changes that take place in LNG and hydrogen storage tanks, special consideration must be given to large dimensional changes in the tanks as they are cooled from ambient to operating temperatures of the LNG or liquified hydrogen.

The aluminum alloys are designated by a four-digit number system CXXXX), beginning with the primary alloying element, followed by letters (o, T, H) designating a kind of temper or heat treatment, and one to three added numbers (YYY)indicating the treatment level. Table 2-2 explains the first digit and the letter.

TABLE 2-2

Aluminum Alloy Nomenclature System

Digit	*Primary Alloying Element*
1	Aluminum of 99 percent purity or higher
2	Copper
3	Manganese
4	Silicon
5	Magnesium
6	Magnesium and silicon
7	Zinc

Letter	*Temper*
O	Annealed
T	Heat treated
H	Strain hardened

The letter T indicates a level of temper ranging from 3 to 9. As an example, T4 is a common temper designating solution heat treatment followed by natural aging at room temperature, T6 designates solution heat treatment followed by artificial aging, and T9 designates solution heat treatment followed by artificial aging and then strain hardening. The H designation indicates a temper obtained by cold-working the metal; if a 1 is added to the H, as HI, the metal has been cold-worked while H2 indicates a cold-worked metal that has been partially annealed; H3 means the metal has been strain hardened and stabilized. A second number can follow this first one to indicate further treatment, for instance, H18 is a commercial, fully hardened temper, while a designation of H19 is an extra-hard temper. Even a third digit may be added to the H to indicate special treatment to obtain particular properties for some special use. A special grade, EC, is given to 99.45 percent pure aluminum, which is used for electrical conductors.

For a complete explanation of the aluminum alloy nomenclature system, contact the Aluminum Association or the American Society for Metals (ASM).

Most of the alloys noted in table 2-3 are available in a broad range of forms, including sheet, plate, wire, rod, bar, rolled shapes, extruded shapes, extruded and drawn tubes and pipe, forgings, and rivets. Extruded shapes are produced from ingots which are forced through dies in extrusion presses having capacities in excess of 4,000 tons, producing extruded

lengths to 85 feet (25.9 meters) or section areas to 17 square inches (110 square centimeters). Plates can be cold-rolled in widths up to about 150 inches (3.8 meters), and stretched in giant stretching machines capable of exerting forces up to 15,000 tons and producing elongations up to 4 feet (1.2 meters). The 7072 alloy is used to clad the 3000 and 6000 series alloys. The 5052 alloy has been used in small watercraft and larger yacht production, and the 5083 alloy in producing spherical tanks for storage of LNG and liquified hydrogen. The 6061 alloy is generally suitable for marine applications and is frequently found in a T4 or T6 temper. A 6061 alloy with these tempers has yield strengths of 21 and 40 ksi and ultimate strengths of 35 and 45 ksi, respectively, making it structurally desirable in many applications. In order to avoid low-strength problems due to welding aluminum, weld seams should be located in low-stress regions of the fabrication, or aluminum plates and shapes should be riveted or bolted, as in aircraft work. This latter method offabrication, however, is more costly and is used less often in marine applications.

TABLE 2-3

Typical Aluminum Alloys

Alloy	*Si%*	*Fe%*	*Cu%*	*Mg%*	*Mn%*	*Ch%*	*Zn%*	*Ti%*	*Other*
1100	1.00 Si+Fe		0.20		0.05		0.10		0.15
2024	0.50	0.50	3.80-4.90	1.20-1.80	0.30-0.90	0.10	0.25		0.15
3003	0.60	0.70	0.20		1.00-1.50		0.10		0.15
5052	0.45 Si+Fe		0.10	2.20-2.80	0.10	0.15-0.35	0.10		0.15
5083				4.45	0.80	0.10			0.15
6061	0.40-0.80	0.70	0.15-0.40	0.80-1.20	0.15	0.15-0.35	0.25	0.15	0.15
7072	0.70 Si+Fe		0.10	0.10	0.10		0.80-1.30		0.15

BABBITT

A class of alloys primarily composed of various proportions of tin, antimony, lead, and copper, and limited amounts of zinc or other elements, babbitt is used as a bearing metal for rotating or sliding machinery components. It has a low Brinell hardness number (14-28), a low compressive yield strength at operating temperatures (1,200 to 3,200 psi, 8.27 to 22.1 MPa), and a relatively low melting temperature (360° to 480°F, 218° to 284°C), making it easy to cast into tinned bronze or steel bearing shells.

The proportions are such that these main elements do not enter into solid solution and are thus able to provide the combination of hard and soft particles necessary (1) to support bearing loads and resist wear and (2) to adjust the hard particle positions in the softer matrix to meet the bearing surface conditions. Softer particles are able to wear down, allowing small depressions to develop in the bearing surface that can hold or maintain residual amounts of bearing lubricants, keeping a supply available to minimize metal-to-metal contact that could cut or score the bearing surface. ASTM specifications are used as guides to composition and application, identifying by alloy number the various amounts ofthe constituent elements. The higher the lead content, the lower the compressive yield strength and lighter service the metal is suited for. Copper and antimony generally increase hardness, while tin increases compressive strength and decreases brittleness.

Babbitt's ASTM alloy numbers range from 1 to 19. Alloy number 3, which has tin, antimony, and copper contents of83.3, 8.3, and 8.3 percent, respectively, is the hardest babbitt and offers the best service if properly fitted and not subjected to impact loading. If this babbitt is slowly cooled, the tin and copper form a crystal which is quite hard; accordingly, it should not be used as a bearing material for a soft metal shaft or slipper. This is a more expensive babbitt, and for large, low-speed-engine, crank-pin bearings, for instance, it is more usual practice to use babbitts having a more economical lead base, such as number 7, containing tin, antimony, and lead in 10, 15, and 75 percent portions. Here, the tin and antimony combine to give the support strength. For lightly loaded bearings, such as for ship's line shafting, alloy number 11 would be more economical yet. Its content is antimony and lead in 15 and 85 percent amounts, respectively; due to its high lead content it is suited for supporting shafts of the softer steels or where embedding, conforming, and low friction are most important. However, for more reliable line shaft bearing performance, the greater strength oftin-based babbitt is preferred, particularly if the bearing is to be centrifugally cast to produce a higher-quality product with better bonding between the shell and babbitt.

BRASS

An alloy of copper and zinc in which the copper content may range from 60 to 90 percent, brass is produced in cast and wrought form in billets, bars, and sheets. Cast brass typically has 30 to 40 percent zinc, 35 percent being the most common. Often, a small amount of tin or lead (1 or 2 percent maximum) is added to cast brass to improve the alloy's hardness or machinability, respectively. This brass is referred to as yellow brass due to its bright yellow color. It polishes well, resists corrosion, and has an emissivity about the same as for aluminum. When the percentage of copper in the brass is between 76 and 88 percent, with tin between 2 and 6 percent and lead between 1.5 and 6 percent, the brass takes on a reddish color and is known as red brass. It is readily cast and machined. A variation of red brass having 85 percent copper and 5 percent each of aluminum, silicon, and zinc is often used to make castings of valve bodies, pump casings, bearing shells, and steam fittings. This composition has good tensile strength (27 to 33 ksi, or 186 to 227 MPa), toughness (16 to 20 percent elongation), and Brinell hardness (50 to 60).

A number of brass compositions have been developed that show particular strength, castability, resistance to corrosion and pitting, ability to harden, or other utility and are given names to distinguish them from others. For instance, Muntz metal, having a 60 percent copper and 40 percent zinc composition, can have its properties modified when heated. It is rQlled at 1,100°F (590°C) and quenched to become hardened. Naval brass, almost the same as Tobin bronze, contains about the same amount of copper as does Muntz metal (59 to 62 percent), but with about 1 percent tin, not more than 0.1 percent iron, and not more than 0.3 percent phosphorus added. It is quite resistant to corrosion, hence the name indicating its marine suitability, and is often used as propeller shafting for small craft. Another composition, admiralty metal, having about 70 percent copper, 29 percent zinc, and 1 percent tin content, can be cold-rolled to produce higher tensile strengths of 50 to 95 ksi (345 to 655 MPa), depending upon its hardness in the as-rolled state. Aluminum brass, obtained by adding 2 percent aluminum to 22 percent zinc and 76 percent copper, is easily cast and produces smooth-surfaced castings of high strength, with a tensile strength in the range of 40 to 50 ksi (276 to 345 MPa) and an elastic limit in the range of30 to 38 ksi (207 to 262 MPa). This brass gives good performance in marine service as it is relatively tough and resists impingement pitting.

BRONZE

While true brasses are compositions of copper and zinc, true bronzes are compositions of copper and tin. As can be noted above, it is typical for other elements to be added in order to make desired adjustments in the alloy's properties, but the resulting material is, typically, still called a brass or a bronze. Thus, some confusion in terms is to be expected when discussing these alloys, although the basic definition should be maintained to the extent possible. Further, the composition of commercial alloys can be expected to have some variation within a range, specific percentages for a particular alloy seldom being precisely obtained in any production run. Thus, if the elemental content is critical, it should be set forth in a material specification.

So-called gear bronze probably has the simplest composition, being 89 percent copper and 11 percent tin (nominally), with a maximum of 0.3 percent iron. It is readily cast and machined and has good resistance to abrasion when being subjected to heavy-duty use in gea:.s and worms. Bronzes containing lesser amounts of tin (5, 8, and 10 percent) are in most common use. These also typically contain small amounts of phosphorus (up to about 0.4 percent) and are sometimes called phosphor bronzes, suggesting a larger content and more importance of the phosphorus than actually exists. This name is therefore somewhat misleading. The phosphorus is needed, however, to deoxidize the alloy, and as long as its content is less than about 0.25 percent, it has no significant effect on the alloy's properties. The bronzes have a good range of elastic response, having yield points ranging from 20 to 80 ksi (138 to 552 MPa), depending upon the amount of cold-working and heat treatment performed. It is important to note that the ductility of bronze reduces with cold-working, as high as 50 to 60 percent before and as low as 3 to 5 percent after the rolling and heat treatment processing, depending upon how extensively the processing is carried out. The elastic modulus of these bronzes is typically about 15 Mpsi (103 GPa); as significant lead additions are made, it falls to 12 to 10 Mpsi (82.7 to 68.9 GPa) or less.

Other "bronze" compositions are made not with tin, but with aluminum, manganese, or silicon as the primary alloy element, silicon bronze alloy being a copper-silicon-manganese composition. Similar to the regular copper-tin bronze, the aluminum bronzes come in 5, 8, and 10 percent aluminum (the remainder being copper) compositions. They have yield strengths of about 40 ksi (276 MPa) and tensile strengths ranging from 52 to 78 ksi (358 to 534 MPa) for soft alloys and 100 to 125 ksi (690 to 861 MPa) after being hardened. As with tin bronzes, their ductility decreases sharply when hardened, reducing from 60 percent to 4 percent elongation for the 8 percent aluminum bronze rod when the soft alloy is cold-worked and heat treated. The cast aluminum bronze has not only high strength but great resistance to corrosion, shock, and fatigue. This makes it suitable for tough service, as in production of ship's propellers.

Manganese bronze resembles a 60-40 brass, as in Muntz metal, but with 0.3 percent manganese, 1 percent tin, and 1 percent iron added, the zinc being reduced to 39 percent and the copper to 59 percent. This alloy has good hot-working properties and high strength and abrasion resistance, frequently being used for small-craft propeller shafting. It has a yield strength ranging from 25 to 65 ksi (172 to 448 MPa), tensile strength ranging from 65 to 95 ksi (448 to 655 MPa), and elongation offrom 35 to 10 percent, depending upon the degree to which the material has been cold-worked and heat treated.

Silicon bronze is typically produced as high-silicon, containing 3 percent silicon and 1 percent manganese, or as low-silicon, containing 1.5 percent silicon and 0.25 percent manganese, the remainder being copper in both cases. High-silicon bronze has a high shear strength, ranging from 33 to 60 ksi (227 to 414 MPa), good corrosion resistance, excellent hot workability, and good weldability by all methods. Its hardness ranges

from 40 to 95 on the Rockwell B scale. Other bronzes used in marine work, particularly in ship's propeller manufacture, include nickel-manganese, nickel-aluminum, and manganese-nickel-aluminum compositions. Four representative approved compositions of these bronzes can be found in the ABS *Rules for Building and Classing Steel Vessels* (1998). Table 2-4 includes as well the ABS minimum strength and ductility requirements, which must be proved by test of a cast coupon.

TABLE 2-4

Propeller Bronze Compositions and Required Properties

	Type of Bronze			
	2	3	4	5
Composition	Mn	Ni-Mn	Ni-Al	Mn-Ni-Al
Copper %	55-60	53.5-57	78 min	71 min
Tin %	1.0 max	1.0 max	-	-
Lead %	0.4 max	0.2 max	0.03 max	0.03 max
Iron %	0.4-2.0	1.0-2.5	3.0-5.0	2.0--4.0
Manganese %	1.5 max	2.5-4.0	3.5 max	11.0-14.0
Aluminum %	0.5-1.5	2.0 max	8.5-11.0	7.0-8.5
Nickel %	0.5 max	2.5-4.0	3.0-5.5	1.5-3.0
Silicon %	-	-	-	0.10 max
Zinc %	remainder	remainder	-	-
Total Other %	-	-	0.50 max	0.05 max
Yield Point (kg/mm^2)	17.5	22.5	24.5	28.0
Yield Point (ksi)	25.0	32.0	35.0	40.0
Tensile Strength(kg/mm^2)	46.0	53.0	60.0	63.5
Tensile Str. (ksi)	65.0	75.0	85.0	90.0
Elongation (%)	20.0	18.0	15.0	20.0

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CAST IRON

Probably the material most used in the manufacture of machines, cast iron has many variations in its composition, production, and use. Basically a carbon-iron composition, certain other alloying elements may be added to improve its properties. The carbon content of cast irons ranges from about 1.7 to 5 percent, with melting points ranging from 2,100°F (1,150°C) to 2,460°F (1,350°C). Upon cooling the cast iron, the carbon tends to solidify as iron carbide (Fe_3C) or in the form of graphite flakes. This gives rise to several distinct kinds of commercially produced cast irons: gray, malleable, nodular, and white, depending upon the carbon content, the form the carbon takes, and the heat treatments applied. It is readily cast into intricate shapes or large and massive components, is very machinable, and is inexpensive. As a cast material, it resists warping and cracking at elevated temperatures, can be produced to have high hardness, and is resistant to abrasion and corrosion. It exhibits good vibration damping characteristics when used for machinery components. Although its tensile strength and elastic modulus are relatively low, its compressive strength is fairly high, typically 50 ksi (345 Mpa) or better, depending on the composition. Thus, it is most frequently used where tensile loads are low or not expected.

Gray cast iron is most important to machinery manufacturers as it is easily cast to produce smooth products having accurate dimensions which exhibit consistently fine grain structures upon microscopic examination. It is important that gray iron castings be of uniform grain structure and free from such defects as porosity or blowholes. The ASTM A48 specification provides a recognized standard. The name "gray" derives from the color of the metal surface upon it!:!exposure in a fracture. Most commercial gray cast irons have 2.5 to 3.75 percent total carbon content, the carbon being in the form of iron carbide to the eutectoid extent, the remainder being precipitated as graphite flakes upon solidification. Because of the high carbon content the melting temperature is lowered, making it not only easier but less costly to cast. The flakes form weakening planes for structural deformation, forming fracture surfaces, but they also tend to act as planes for chip formation when the material is being machined.

An ordinary gray cast iron could have 3 percent total carbon, 2 percent silicon, 0.6 percent manganese, 0.2 percent phosphorus, and 0.12 percent sulfur. Such a material would have a Brinell hardness of about 185 and a tensile strength ranging from 30 to 45 ksi (207 to 310 MPa). If small additions of chromium were made, cast irons of similar composition could be found in engine cylinder blocks. Adding 1 percent nickel and 1.25 percent molybdenum, increasing manganese to 1 percent and silicon to 2.5 percent, and adding 0.2 percent chromium, while reducing total carbon to 2.8 percent, would raise the tensile strength to 60 to 80 ksi (414 to 552 MPa), making it suitable for use in producing crankshafts. Several compositions suitable for piston rings are also recognized.

Malleable cast iron is made by melting pig iron, steel scrap, and cast iron scrap in proportions to yield a composition within a more limited range, generally 2.25 to 3.0 percent carbon, 0.3 to 0.5 percent manganese, 0.05 to 0.08 percent phosphorus, 0.06 to 0.11 percent sulfur, and 0.6 to 1.15 percent silicon. After being cast, the castings are packed in annealing boxes and gradually heated in an annealing furnace to about 1,600°F (870°C), held for three to five days, and gradually cooled. During this time, oxygen diffuses inward and iron carbide migrates to the surface, decomposing into carbon monoxide, carbon dioxide, and ferrite (alpha phase, BCC iron). The resulting metal is relatively soft and can be bent without breaking, though it is not truly malleable. Standard specifications for this iron often require elongation of 10 to 18 percent in a 2-inch specimen, as well as a yield strength of 32 ksi (220 MPa) and a tensile strength of 50 ksi (354 MPa). The elastic modulus is typically about 25 Mpsi (172 GPa). Because the migration process proceeds slowly, only thin-shelled castings can be produced in this manner. Thus, the malleable iron castings are usually small- to medium-sized fittings, valves, and mounting brackets.

Nodular cast iron, sometimes called ductile iron, is used to produce castings of a wide range of thicknesses. It is made by adding small amounts of cerium or magnesium to liquid gray cast iron just before it is to be poured. This results in carbon forming into spherical globs rather than graphite flakes. The result is greater ductility, allowing more mechanical working than for gray cast iron. The melting temperature for this material is also lower, making casting easier.

White cast iron has its carbon content in the iron-carbide form (Fe_3C), called cementite. The carbon will tend to dissociate from the iron in the liquid state if given enough time, so the metal must be cooled rather rapidly in order to maintain the carbide form. Silicon content is varied with carbon content to control this process. It has a white color when fractured, as there is an absence of graphite, and it is rather brittle, hard, and wear resistant. Due to its wear resistance, it is used in making brake shoes and similar components needing resistance to abrasive wear.

CONSTANTAN

A 45 percent nickel, 55 percent copper alloy, sometimes referred to by trade names such as Advance, Copel, Cupron, Excelsior, or Ferry, constantan is used as pyrometer wire to make thermocouples with copper, iron, or nichrome. These thermocouples operate on the Seebeck effect, wherein the joining of two dissimilar metal wires, joined at each end in a closed circuit, will generate an electromotive force (emf) which causes a current flow in the wires when one of the junctions is subjected to a temperature change, while the other junction's temperature is being held constant. The current flow is nearly linear with temperature change over a wide range and thus the current flow can be calibrated in terms of temperature difference.

Copper-constantan thermocouples can be used in temperature ranges up to about 680°F (360°C) with high precision, and up to about 900°F (480°C), with an accuracy of about 10°F (5.5°C). The iron-constantan and nichrome-constantan thermocouples can be used at temperatures up to about 2,000°F (1,100°C). Beyond these temperatures, thermocouples of other materials are available, and above 2,700°F (l,480°C), optical or radiation pyrometers would be used. It should be noted that at lower temperatures, the iron tends to oxidize in moist environments and the performance of the iron-constantan thermocouple will most likely deteriorate over time as a result.

INYAR

An iron alloy with 35 percent nickel, 0.5 percent manganese, and 0.5 percent carbon, invar's primary attribute is its very low coefficient of thermal expansion (0.000000374 to 0.000000440 per degree C). Steel expands about 26 to 31 times as much. It is therefore most useful in making precision measuring tapes as the tape dimensions are very stable through significant temperature changes.

MANGANIN

An 84, 12, and 4 percent alloy of copper, manganese, and nickel that has very stable electrical resistance when subjected to temperature changes and a relatively high melting point (1,868°F or 1,020°C), manganin has a resistance temperature coefficient of about 0.00000833 per degree Fahrenheit (0.000015 per degree Celsius) of temperature change. This makes it very useful in the development of electrical instruments and instrumentation systems. It is used to make precision resistors for Wheatstone bridges that precisely measure electrical resistance, for shunts in ammeters and voltmeters to measure line current and voltage, and in ampere-hour and watt-hour meters for measurement of power.

MONEL

A copper-nickel alloy, of about 65 to 70 percent nickel, 0 to 3 percent iron, 0 to 1.5 percent manganese, 0 to 0.25 percent silicon and carbon, and the remainder copper, monel is perhaps the most useful copper alloy. It is produced in cast, drawn, hot-rolled, and cold-rolled forms in billets and bars of various cross-section shapes, wire, plate, sheet and strip, and tubing. Easily machined, it resists corrosion in the marine atmosphere and is therefore favored for applications in such exposed locations as the weather deck of ships. Quite ductile (35 to 50 percent annealed condition elongation) and malleable, it retains its strength at elevated temperatures better than most brass and bronze alloys, and even some steels. As a metal that is useful for making special fittings, its primary drawback is added cost. Its elastic modulus is about 26 Mpsi (179 GPa) and its ultimate strength varies with the method of production, from 65 ksi (448 Mpa) when cast to 175 ksi (1.206 GPa) when cold-drawn into spring wire. Hot-rolled plate, as an example, has a 25 to 30 ksi (172 to 207 MPa) yield stress, an ultimate stress of 60 to 77 ksi (414 to 530 MPa) and a 25 to 35 percent elongation in a 2-inch sample.

SOLDER

Typically produced in bar and wire, solder is most commonly composed of 50 percent lead and 50 percent tin, the most popular mix for general purpose uses. The ASTM defines a range of solder grades that are suitable for various applications and that vary from 95 percent lead and 5 percent tin to 30 percent lead and 70 percent tin. The higher tin-content solders are useful for coating metals, while the higher lead-content solders are considered suitable for coating and joining materials. It should be noted that lead

has the higher melting temperature; therefore the higher the lead content, the higher the solder melting point. The 30 percent tin-70 percent lead solder has a melting temperature 113°F higher than the 70 percent tin-30 percent lead solder; heat for a high lead solder would likely be applied by torch. Consideration of lead leaching into potable water from high lead-content soldered joints has led to the suggestion that tin-antimony solder be used for copper joints in tubing used for such services. This solder would have a 95 percent tin and 5 percent antimony mix and no lead. Tin-lead-silver solder may be used in soldering brass, copper, and related metals where heat is applied with a torch; a suitable composition would be 1 percent tin, 97.5 percent lead, and 1.5 percent silver, which has a 588°F melting temperature. A suitable flux is needed to ensure wetting.

STEEL

As noted above, pure iron is relatively soft, ductile, and low in tensile strength. It is also expensive to produce, as a number of other elements included in the as-produced metal have to be removed. Carbon from the smelting process coke is a most typical inclusion. The simplest steel is defined as a pure iron alloy containing less than 2 percent carbon. It can be hardened by quenching it (quickly cooling as by immersion in a liquid) from a high temperature. This hardened metal is somewhat brittle, but by reheating it to a low temperature the brittleness can be reduced without significantly reducing the hardness. This reheating process is called tempering. Thus steels can be quenched and tempered to obtain the desired hardness and toughness. These processes have been known from ancient times but only in modern times has there been sufficient understanding to allow consistent processing to obtain particular properties. In addition to carbon, other alloying elements must be added to obtain a broader range of desired properties.

Steel is used in a wide variety of applications requiring a range of characteristics and is produced using several methods to develop products of various forms, sizes, and shapes. The processes used include the basic oxygen, basic open hearth, acid open hearth, and electric furnace methods, from which ingots are produced that may then be rolled into slabs and blooms; the blooms may be further reduced to billets. From these, a wide variety of commercial products are rolled, drawn, forged, and extruded. In order to improve the efficiency and quality of the steel, some producers pour the liquid steel from the ladle into continuous casting machines that convert the steel into the semifinished slabs and blooms, avoiding the initial rolling step.

Hydrogen has a deleterious effect on s';;eel;low-hydrogen steel can be produced by pouring the liquid in a vacuum chamber. The evolution of gases during the steel solidification process can cause defects in the ingot interior and various means are employed to reduce or cc,ntrol their generation.

When the steel is fully deoxidized and no gas is formed on its solidification, the result is "killed steel," such as used in ABS hull steel grades D, E, DS, CS, DH32, EH32, DH36, and EH36. If the steel is not fully deoxidized, and any small interior blowholes that form while the steel solidifies are welded closed during the hot-rolling process, the steel is called "semi-killed" such as used in ABS hull steel grades A, B, AH32, and AH36. With even less deoxidation, oxygen and carbon in the liquid steel can form carbon monoxide, which tends to evolve freely from the cast ingot. Ifthis process is allowed to continue to its completion, the steel is termed "rimmed steel"; if this process is prevented from continuing to its completion by use of some mechanical means, the result is. "capped steel." Each of these steel types has attributes making it suitable for particular uses.

As noted above, iron at ambient temperatures has a BCC crystal structure and is referred to as ferrite, or alpha *(a)* iron. As it is heated, it changes its crystalline structure into FCC as gamma *(y)* iron at a temperature of about 1,660°F (905°C). This is a stable crystalline structure up to a temperature of about 2,550°F (1,400°C). In the BCC structure there is inadequate space to fit the carbon atom, while iron in the austenite, FCC structure has more space to accommodate the carbon atom, but not without making distortions in the lattice structure, thus the 2 percent limit as to how much carbon can be dissolved in the iron lattice. Carbon then, while too small to replace an iron atom but too large to fit into the interstitial space ofthe iron BCC crystal lattice, does fit into the interstitial space of the iron FCC crystal lattice, but with a little lattice distortion. It can be appreciated then that most forging and hot-rolling operations are conducted while the steel is in this FCC, austenitic state (about 2,000°F, 1,100°C), with the metal still quite soft and ductile.

The alloying process can proceed by having the alloying elements fill in the spaces between the iron atoms ofthe lattice (interstitial) or by replacing one of the atoms (substitution) in the lattice. This depends upon the space available between the atoms in the lattice as well as the size of the alloying element atoms. Alloying elements include aluminum, boron, calcium, chromium, cobalt, columbium, copper, lead, manganese, molybdenum, nickel, nitrogen, vanadium, phosphorus, selenium, sulfur, silicon, tantalum, titanium, tungsten, and zirconium. Manganese at about 0.5 percent is included in all steels. For ship steel, this content is increased to 0.70 to 1.35 percent for most grades, but not less than 2.5 times the carbon content; phosphorus and sulfur are generally restricted to 0.04 percent maximum. Except in the silicon steels, silicon is restricted to a maximum of about 0.35 percent in ship steel. Other alloying elements are carefully added to control processes and obtain desired properties.

Many alloying elements are added in the form offerroalloys, such as ferrochromium, ferromanganese, ferromolybdenum, ferrotitanium, and ferrovanadium. These ferroalloys are high in alloy element content and are called

addition agents. They may be added with the charge or directly into the ladle to control the process by, for example, promoting exothermic reaction or deoxidizing or increasing the hardenability of the steel. For instance, aluminum aids deoxidation, while lead is added to improve the free-machining characteristics of steel; columbium and tantalum improve the mechanical properties of fine-grained carbon and low-alloy steels; selenium is added to stainless steels to improve machinability; tungsten is used to raise toughness in high-speed-tool steels; cobalt also helps make hard-tool steels for high-speed machining, permanent magnets, or special tough steels.

As can be appreciated from the above, the complexity of steel alloying makes it necessary to take a systematic approach in defining the resulting steels. The American Iron and Steel Institute (AIS!) and the Society of Automotive Engineers (SAE) have agreed upon nomenclature for various steels depending upon the main alloying elements. Table 2-5 sets down the main elements of the number system; the specifics are subject to change from time to time. As for the aluminum alloy numbering, a four-digit number is used, while prefix letters B, C, and E denote the Bessemer, openhearth, and electric furnace methods typically used for their production. A TS prefix indicates a tentative standard for the steel. The "xx" indicates the hundredth-percent carbon content range. The letter B between the second and third digit indicates boron content at a minimum of 0.0005 percent. The ASTM has developed a counterpart identification system for the alloys, cited in ABS *Rules for Building and Classing Steel Vessels (1998).*

TABLE 2-5

AlSI and SAE Nomenclature for Steels

AISI No.	*Description*
10xx	Low-carbon steel, 0.25-0.9% manganese, 0.04% phosphorus, 0.05% sulfur
llxx	Resulfurized (free-cutting) steels, sulfur 0.4-0.33%, phosphorus 0.04% max
12xx	Rephosphorized and resulfurized (free-cutting) steels, P 0.04-.12%, S 0.10-.35%
13xx	Manganese 1.6-1.9%
15xx	High-manganese carbon steels
23xx	Nickel steel, Ni 3.25-3.75%
25xx	Nickel steel, Ni 4.75-5.25%
31xx	Nickel-chrome steel, Ni 1.1-1.4%, Cr 0.55-0.9%
33xx	Nickel-chrome steel, Ni 3.25-3.75%, Cr 1.4-1.75%
40xx	Molybdenum steel, Mo 0.15-0.3%
41xx	Chrome-moly steel, Cr 0.4-1.2%, Mo 0.08-0.25%
, 43xx	Nickel-chrome-moly steel, Ni 1.65-2.0%, Cr 0.4-0.9%, Mo 0.2-0.3%
46xx	Nickel-molysteel, Ni 1.4-2.0%, Mo 0.15-0.3%

Continued on next page

TABLE *2·5-Continued*

AISI No.	*Description*
47xx	Nickel-chrome-moly steel, Ni 0.9-1.2%, Cr 0.35-0.55%, Mo 0.15-0.4%
48xx	Nickel-moly steel, Ni 3.25-3.75%, Mo 0.2--0.3%
50xx	Chromium steel, Cr 0.2-0.6%
51xx	Chromium steel, Cr 0.7-1.15%
521xx	High-chromium steel, Cr 1.3-1.6% (may have a B, C, or E prefix)
61xx	Chrome-vanadium steel, Cr 0.5-1.1%, V 0.1-0.15%
81xx	Nickel-chromium-moly steel, Ni 0.2-0.4%, Cr 0.3-0.55%, Mo 0.08-0.15%
86xx	Nickel-chromium-moly steel, Ni 0.3-0.7%, Cr 0.4-0.85%, Mo 0.8--0.25%
87xx	Nickel-chromium-moly steel, Ni 0.4--0.7%, Cr 0.4--0.6%, Mo 0.2-0.3%
88xx	Nickel-chromium-moly steel, Ni 0.4-0.7%, Cr 0.4-0.6%, Mo 0.3-0.4%
92xx	Silicon steel, silicon 1.8-2.2%
94xx	Nickel-chrome-moly-silicon steel, Ni 0.3-0.6%, Cr 0.3-0.5%, Mo 0.08-0.15%, Si 0.2-0.35%

Except for the 11xx and 12xx steels, phosphorus is typically 0.04 percent maximum with sulfur 0.05 percent maximum, while silicon is generally kept in the 0.2 to 0.35 percent range in all steels but the 92xx.

STELLITE

Stellite is an alloy of cobalt with metals such as 30 to 35 percent chromium, 12 to 17 percent tungsten, smaller amounts of other metals such as molybdenum and nickel, and 2.25 to 2.75 percent carbon, the remainder being cobalt. It is a very hard material (stellite J has a Brinell hardness of 600 at room temperature and it only reduces to about 320 at 1,470°F, 800°C) that resists erosion or abrasive wear, such as that caused by throttled steam or machining operations. It is not malleable or machinable and must be shaped by grinding and polishing. It is often used in steam valve seats, as it is highly resistant to erosion by high-velocity steam. It is used for edges of cutting tools since the heat of the cutting operation doesn't damage its cutting edge.

Nonmetals

Metallic materials are used for most engineering applications, and this will likely remain so into the foreseeable future, but some uses for wood remain, and ceramics, plastics, and composite materials are being increasingly used when their material properties meet environmental and technical requirements.

WOOD

Wood is primarily used for dry cargo dunnage, temporary shoring, and such decorative purposes as teak rail caps and laminated ash, mahogany, teak, or walnut bulkhead surfaces. When used for interior decoration, woods are typically impregnated with fire retardant chemicals to meet safety standards, as for a half-hour, two-hour, or six-hour fire exposure resistance. Although wood is still allowed for hatch boards of cargo hatches, this is now rare, as steel hatch covers of various designs have long been dominant. Lignum vitae, a very hard, dense, resinous wood of soapy feel long used to line propeller shaft stern-tube bearings, has been replaced by metal-lined, oil-lubricated, mechanically sealed stern bearings. Balsa wood is used as lightweight (specific gravity of 0.11) composite lamination separation in the fiber-reinforced plastic (FRP) skins of small craft and as low-temperature insulation (e.g. for LNG tanks). Its strength increases as its temperature is reduced, if moisture content remains unchanged.

CERAMICS

A class of materials composed mostly of inorganic, nonmetallic elements, usually oxides, carbides, and nitrides, with metallic components for certain compounds, ceramics are generally hard, abrasion resistant, but brittle materials and typically have excellent heat and electrical resistance. They are suitable for use in high-temperature and harsh environments, but not where shock loadings are expected. Traditional ceramics include brick, china, glass, porcelain, tiles, high-temperature refractories, and dielectric insulators, among others. The materials used to develop ceramics include various clays, such as of aluminum, silicon and oxygen, feldspar, silica, and talc. Compounds such as magnesium oxide (MgO) make simple ceramics and are used in refractory materials because they can withstand very high temperatures. Titanium carbide, silicon carbide, boron nitride, and zirconium nitride are neither metals nor ceramics, but they lack free electrons and therefore are not good conductors of electric currents. Ceramic coating of internal combustion engine cylinders has reduced heat loss and wear and has increased efficiency.

Ceramics can become semiconductors if they contain iron oxide compounds such as Fe304 or magnetite, which has about the same resistivity as graphite. They can become magnetic ifthe compounds include iron, cobalt, or nickel. The resistivity of some ceramics decreases with temperature, allowing them to be used with metallic components having positive circuit resistivity changes.

PLASTICS

Plastics, of increasing interest in marine engineering, are organic compounds, primarily based on carbon and hydrogen and usually consisting of long-chain molecules of low specific gravity and elastic modulus. Either thermoplastic (melts when heated) or thermosetting (hardens when heated), they can be cast, extruded, and molded. Some particular materials and uses are shown in table 2-6.

TABLE 2-6

Plastic Materials and Uses

Thermoplastic Materials	*Uses*
Acrylic	Lucite, Plexiglas transparent sheets, as for drafting equipment
Nylon	Bearings, bushings, cams, gears, handles (cast or machined)
Polycarbonate	Lexan, used for glasses, light globes, safety helmets (molded)
Polyester	Dacron, Mylar, Kevlar, used for magnetic tape, sail cloth
Polyethylene	Battery parts, film, trays
Polypropylene	Bottles, packaging film, rope (braided and stranded)
Polystyrene	Appliances, battery cases, indoor lighting fixtures
Vinyl	Floor covering, hose, wire insulation, pipe
Thermosetting Materials	*Uses*
Epoxies	Adhesives, moldings, paints, resins for FRP laminates
Phenolics	Bakelite for motor housings, electrical fixtures
Polyester	Resin for FRP laminates

COMPOSITES

Composites are two or more materials of different structural properties bonded together so as to act as one combined substance. The strain materials are subjected to must be the same on either side of an interface in order that they act compatibly, without differential displacements. Composites of different metals, or of a metal and nonmetal, have long been used, for example, reinforced concrete and wooden beams with steel rider plates. Improved performance, lower weight and cost, enhanced characteristics such as resisting corrosion and fatigue, and ease of series production are important attributes ofthese materials. Fiber-reinforced plastics (FRPs) are being used increasingly as better understanding of construction geometries and their structural properties is developed. Fire resistance and smoke toxicity when subjected to flames are concerns important to shipboard applications. For FRP composites, fibers carry the loadings while the resin fixes fiber geometry and acts as the shear transmission medium for keeping the fiber strains consistent. (For more on marine FRP applications, refer to U.S. Coast Guard Report SSC-360, "Use of Fiber Reinforced Plastics in the Marine Industry.")

INSULATION MATERIALS

Electrical elements must be isolated to prevent power loss and system short circuits with resulting component damage, while thermal system components must be isolated to prevent heat loss (with associated thermal inefficiencies). To these ends, insulation materials are used to coat, surround, and generally isolate components, and to protect personnel.

ELECTRICAL INSULATION

Electric circuits need high-resistance insulation to minimize current leakage and good dielectric strength to resist impressed voltages. Wire coatings should be mechanically strong, flexible, and chemically stable in the environment and should resist cracking and heat breakdown. Material resistivity is expressed in megaohms (MQ, or millions of ohms), for a given material thickness, say 1 centimeter. A I-millimeter thick insulation material would have a resistance of 0.1 times the megaohm . cm resistivity value given in tables. Dielectric strength is expressed in terms of volts per millimeter or volts per mil (1/1,000 inch). Thus, 400 V/mil converts to about 16,000 VImm, as 1 mm is about 40 mils. Good insulators include Bakelite epoxy, mica, nylon, treated paper, phenolic resin, polyethylene, polyvinyi chloride (PVC), rubber (natural or butyl) and Teflon. Mica has the best dielectric strength, ranging as high as 4kV/mil. Thermosetting Bakelite, epoxy, phenolics, and Teflon are moldable; thermoplastic nylon, polyethylene, PVC, rubber and Teflon (with difficulty) are extrudable. The melting temperature of Teflon is 750°F (499°C) and nylon melts at 260°F (126°C), while silicone rubber can be used in 300°F (149°C) environments for extended periods; thus, ambient conditions must be considered when selecting materials so as to prevent thermal breakdown under service conditions.

THERMAL INSULATION

Thermal insulation serves to restrict the transfer of thermal energy between two bodies of differing temperature. Heat transfer can take place by any combination of radiation, conduction, and convection, thus the purpose of insulation is to interrupt the paths that would allow one or more ofthese heat transfer mechanisms to function. This can be done by (1) reducing surface emissivities, (2) physically separating conducting bodies, and (3) stopping convective fluid flows over conducting surfaces. One or more of these means is used in each insulation system. Effectiveness of insulating materials depends upon the surface emissivities, surface bonding, material densities, the characteristics of contained gases and their humidity, as well as how resistant the materials are to compaction over time, such as by shock loading, vibration, or settlement. The vacuum bottle provides a good illustration of insulating principles because it has silvered interior surfaces of very low emissivity, and the conducting and convecting medium (namely air between the inner and outer bottles) has been evacuated, the inner bottle being connected to the outer bottle only at the neck. Aluminum LNG tanks have a similar system in that the emissivity of aluminum is quite low, the tanks are supported by thermally insulating structure elements, and the tank surfaces are covered with low conductivity materials. Whether the isolated body's temperature is cryogenic, ambient, or superheated, the principle is the same; the differences arise because of the materials used in the various temperature ranges and the means used to apply and protect them. To reduce conduction, low conductivity materials are used; convective transfer is reduced by fluid removal or separation into multiple minuscule spaces. Heat loss varies inversely with insulation thickness.

THERMALCONDUCTnnTY

Heat flow, q, through a material is dependent upon the temperature difference across the material thickness, !:J.T, the cross-sectional area of the material perpendicular to the thermal gradient, A, and a coefficient of thermal conductivity for the substance, k. In any heat transfer circumstance, the fluids on either side of the substance have complex flow regimes that affect the surface temperatures and the final heat flux. However, the conductivity of the materials can be expressed in the form $q = k \times A \times$!:J.T,k having dimensions of heat units per unit of time per unit of surface area per degree oftemperature difference, all in consistent units. Thermal conductivity coefficient k can also be defined as $k = L / (R \times A)$, where L is the insulation blanket thickness (inches), A is the unit area, andR is the material's thermal resistance. The coefficient could be, in U.S. customary units, expressed as Btu-in/sq ftlhr/F, or, in metric units it could be expressed in Watts-m/sq m/C. Coefficients vary with temperature, so mean values are typically given for a temperature range. For particular interests, careful attention to coefficient units is essential. For example, a ten-inch-thick fiberglass blanket with an R value of 30 would have a k value of 10/30 or 0.333. Approximate coefficients for some typical insulation materials at ambient temperatures are given in table 2-7 (in U.s. customary units).

TABLE 2-7

Coefficients of Thermal Conductivity, k

Material	*Coefficient of Thermal Conductivity (Plane Surface)*
Balsa wood (end grain)	0.3125 to 0.3458 (depending on density)
Cork	0.304 to 0.412 (depending on density)
Diatomaceous silica brick	2.44 (2,000°F mean temperature)
Fiberglass	0.333 (blanket or roll)
Magnesia (85 percent or similar)	0.507 (steam pipes with 300°F temperature difference)
Mineral wool	0.276
Rock wool	0.36 (100° F mean temperature) to 0.68 (600°F mean temperature)
Vermiculite	0.68 to 0.72

Balsa wood, a lightweight (specific gravity = 0.11) material, has been used to insulate LNG tanks. Its thermal conductivity increases with density. Balsa wood strength parallel to its grain increases as temperature decreases; the cellulose fibers have low conductivity and, being hollow, are gas filled in minuscule volumes. Cork board, with a density of about 10 Ibs per cubic ft, is frequently used to insulate refrigerated spaces as it holds shape well, resists vibration, and can be molded to specific shapes up to about 8-in thick. Closed-cell organic foams of polyurethane or polystyrene are also used, being sealed to keep moisture absorption, and consequently conductivity, down. Multiple layers of aluminized foil, separated by spacers and contained within a high-vacuum chamber, are also effective, though more expensive.

For near-ambient and higher temperatures, a variety of materials in the form of blocks, shaped sections, and blankets are available. For instance, fire-clay brick is used for furnace settings with diatomaceous earth blocks used for lining boiler walls behind fire bricks where the temperature drops below 1,800°F (980°C), and mineral wool in block or blanket form is used to line casings. For temperatures below 1,200°F (650°C), calcium silicate blocks are frequently used on piping and boiler enclosures. In lower temperature services, as for bulkheads and auxiliary services, glass fiber and rock wool blankets and shaped insulation sections are typically applied. Polyurethane and polystyrene are prepared in foam blocks or sheets and formed sections; they are also foamed in place when the expanding agents are acceptable to the space uses. Thermal conductivity, as represented by its coefficient k (Btu . in/sq ft/Flhr), of insulating materials generally increases with temperature in a nonlinear manner. However, the need for precise definition of conductivity as a function of temperature and material density is generally lacking and only linear or average values are typically available in handbooks.

Asbestos, a material extensively used in older insulating systems, has essentially vanished from the marine field because of the recognized maritime personnel health hazard of asbestosis. For further data on insulation and insulation systems, refer to ANSI or ASTM standards. The National Institutes of Standards and Technology, formerly the National Bureau of Standards, has conducted systematic tests of many materials and should be referenced. For more detailed information on metal and insulation conductivities, see the engineering handbooks referenced at the end ofthe chapter.

ENGINEERING MATERIAL MANUFACTURING PROCESSES

Metals, produced as pure or alloy ingot, must be worked into useful forms. A variety of processes are available for this work, and typically several processes are used in any production scheme. Though most modern ships have limited equipment for onboard materials processing, such as a lathe, drill press, power grinder, and electric arc welder, it is important to understand the many processes that may be used by shore-based organizations in preparing materials for use in shipboard applications. Some of these processes are briefly discussed below. Although metals are the focus, some of the processes are used for nonmetals.

Casting

Casting is a process in which a molten metal is poured or forced into a mold whose interior shape is the desired exterior form of the object to be produced. Casting processes include gravity, slush, centrifugal, die, and several precision methods. Molds may be temporary, used to produce but one object, or permanent, used in series production. Molds are usually created by using patterns of the shapes to be produced. They can be made of sand, plaster of paris, metal, or other moldable materials that can withstand the thermal and mechanical stresses ofthe casting process. Permanent molds for smaller objects are usually made of metal able to withstand the high temperatures of the metals to be cast. For finer surface finishes, finer grained materials must be used in making molds. To produce hollow castings, such as for a pump casing, a core with the desired interior shape is either initially fitted into the mold or, in some cases, forced into the mold after the molten metal has been poured. For large objects, sand is generally used to produce a one-off mold with appropriate core element(s). To keep the sand molds from failure prior to or during the casting process, binders are used such as clay, linseed oil, graphite, molasses, starch, silica flour, or water that will dry to give a hardened surface of adequate strength.

Cutting Off

In cutting off, a piece of material is separated into two or more pieces by sawing, shearing, oxyacetylene torch cutting, plasma-arc cutting, or other means. A number of power machines are available for sawing, including hack, band, and circular saws. The hacksaw, useful for virtually all materials of modest size, has a reciprocating motion; the cutting blades, with teeth set alternately to either side ofthe blade, are typically of high-speed alloys with various tooth sizes selected for the material to be cut and the intended cutting speed. The material is clamped in place during the cutting process. Band saws have a continuous blade of high-speed steel, with teeth set similarly to the hacksaw blade. The blade speed is usually adjustable in, a range from about 0.5 feet per second for cutting high-temperature alloys to 20 to 25 feet per second for softer materials like aluminum or plastics. The tooth size and spacing are increased for softer materials, so that cuttings may more readily fall clear of the teeth upon emergence from the cut, and blade speed is reduced for thicker materials. While the hacksaw is usually

vertically oriented, band saws may be vertical, horizontal, or universal (the latter using a tilting table). Circular saws generally have higher tooth speeds and come in a wide variety of sizes, with fixed or tilting tables. A variety of blade sizes and tooth shapes allows a broad range of materials to be cut; however, because of the higher tooth velocities, they are most suitable for softer materials. Using a "parting" tool, a lathe can also be used for cutting through materials of a size appropriate to the lathe's chuck and swing dimensions.

Metal sheets can be cut with a shearing machine with a wide sharpened and hardened blade fitted at an incline to the horizontal so the cut can advance progressively across the metal sheet as the blade travels vertically down. Fitted with hydraulically operated clamps, the machine holds the metal sheet firmly in place during operation. Shears are made sufficiently wide and powerful that steel plate greater than 0.4 in (1 cm) thick and 10 ft (3 m) wide can be cut.

Oxyacetylene cutting uses a torch to mix acetylene and oxygen, producing a flame of about 6,000°F (3,315°C) that heats the metal. The torch also controls release of oxygen, which promotes rapid oxidation of the metal and blows away the melted/oxidized material. Plasma-arc cutting uses a tungsten electrode to establish an arc in the center of a very small diameter plasma jet of argon or nitrogen and hydrogen, said to be heated to about 50,000°F (27,700°C). The heated gas jet quickly expands into a high velocity stream which rapidly melts and flushes away a narrow strip of metal (a kerf) forming the cut. This process can be used for cutting stainless steels, which cannot be cut with an oxyacetylene torch, and nonferrous metals.

Extruding

Extruding is a severe squeezing process in which a material is forced through a die and emerges in a desired shape. It is carried out cold or with heat, depending upon the properties ofthe extruding material. A wide variety of metals and plastics are extruded to produce products for engineering application. Mundane items, such as toothpaste, macaroni, and terra cotta soil pipe, are familiar as well. Insulation is typically applied to wire in an extrusion process wherein the wire passes through a die block and the insulation is forced around the wire as it emerges from the die center. Early used to create thin-walled tin and lead tubes, its use was broadened by application to materials like aluminum, brass, copper, magnesium, warm zinc (300°F, or 150°C) and many plastic items, producing tubes, rods, architectural shapes, and more. While steel shapes are hot-rolled, aluminum structural shapes are usually extruded. Impact extrusion presses generating pressures of up to 330 ksi are used to extrude mild steel.

Finishing

Materials can be hot-finished or cold-finished, as in hot- or cold-rolling of products to commercial stock dimensions, or finished to close tolerances or roughnesses by any of several cold processes such as wheel or belt grinding, polishing, honing, lapping, electroplating, painting, and enameling. Hot-rolling of steel typically results in formation of iron oxide "mill scale" that must be removed by shot blasting, followed by an antioxidation prime coating. While grinding, honing, lapping, and polishing remove progressively smaller scratches, electroplating, painting, and enameling add to dimenSIOns.

Forging

Forging is a hot-working process in which the metal is heated to a favorable temperature for its composition (ranging from 2,400°F for 0.1 percent carbon steel to 1,900°F for 1.5 percent carbon steeD, then hammered or squeezed. The process causes the metal to have its grain size refined and mechanical properties improved through the actions that take place. Forging is carried out by drop hammering, hydraulic pressing, successive rolling, upsetting, and so on, using various machines.

Joining

Engineering systems are composed of many elements that must be assembled by joining in one or more ways. Mechanical fastening is accomplished using bolts, rivets, and screws, but for larger assemblies of a more permanent nature, assembly is frequently carried out by welding, brazing, or soldering. Welding is a process in which two compatible metals are raised to a high enough temperature that an atomic-level fusing can take place. The process can be carried out with or without use of filler metals, but in either case, the surfaces to be joined must be clean and free of any foreign substance that could contaminate the union. Of particular importance to marine systems is arc welding in which a compatible filler material in the form of an electrode or filler wire melts to fill the gap between the two pieces, an electric arc providing the heat needed to melt the parent and filler metals, a powdered flux covering the weld to shield the joint as it cools, or a shielded electrode providing protection from the air. Gas welding is also available using an oxyacetylene torch with suitable filler rod, though this has been largely supplanted aboard ship in recent years by arc welding. Some metals such as aluminum that oxidize readily when heated to melting temperatures are welded using an inert gas (such as argon or helium) shielding of the filler and joint called MIG, metal-inert-gas, or TIG, tungsten-inert-gas, where the electrode is of tungsten. Extensive welding research has been conducted in recent years and reference should be made to the American Welding Society or the James F. Lincoln Arc Welding Foundation for current process data and electrode specifications. A wide variety of welding rods are available for use in particular applications. Classification society rules specify material grades, requirements, and tests the filler materials must satisfy to be suitable for important shipboard applications. Weld quality of important weldments (products of welding) must be sufficient to pass radiographic or ultrasonic examination.

When the materials to be joined are not compatible, oxyacetylene brazing can be used, bronze rods serving as filler metal and a suitable flux dissolving surface oxides to provide clean bonding surfaces. Brazing is carried out at lower temperatures, above 840°F (450°C), at which the base metals do not melt but are heated to a red heat (1,150° to 1,350°F, or 620° to 730°C). The brazing rod, a copper-zinc alloy, typically has small additions oftin, manganese, and other elements to improve the mechanical properties of the joint; the rod melts first, effectively "wetting" the base metal and then providing the filler metal. Because of its lower heat, brazing distorts the joint less than welding. Silver solder, for higher-temperature joining, has a silver-copper-zinc composition (several are available) that results in metal flow temperatures in the range of 1,325° to 1,600°F (720° to 870°C). This is typically applied with an oxyacetylene flame, as in brazing; borax is a frequently used flux.

For lower-temperature work, metals can be joined by soft soldering, a process in which the base metal surfaces to be joined are cleaned and wetted with melted solder filling the gap between the surfaces, the solder then cooling to seal the joint. Joint surfaces should be free of oxides, dirt, oils, and other foreign matter and a flux should be used to chemically clean the mating surfaces to assist the solder wetting. Rosin is a neutral flux for soldering that is mild in its action, while zinc or ammonium chloride fluxes are more active. Corroded surfaces are difficult to solder.

Heat Treating

Heat treating uses heating and cooling to develop desired metal structures and mechanical properties. Upon heating above a critical recrystallization temperature, grain growth and crystal structure changes can take place. Cooling rates determine whether harder or softer structures will result. For steel, quick cooling by quenching will fix harder structure; slow cooling (annealing) fixes softer structure. Quenching copper fixes annealed, soft structure.

Machining

Any of several processes for controlled removal of metal to change the shape of a body to its final dimensions is called machining. They include turning, planing, shaping, milling, hobbing, boring, broaching, reaming, and drilling, each being performed using a machine of similar name, as planer, shaper, miller, hobber, broach, and drill press, an exception being the lathe for turning. Reference should be made to a machinist's handbook for detailed discussion ofthese operations and descriptions ofthe particular machines used. The modern ship typically carries a lathe and drill press.

Rolling

In rolling, a metal ingot, billet, bloom, sheet, or bar is shaped by being passed sequentially between roller pairs, each set of rolls squeezing the metal to lengthen or widen the material or otherwise reach the final section shape. The process can be carried out "hot" or "cold," depending upon the metal properties, the extent of shaping to be accomplished, and the desired finish. The process changes metal grain size and mechanical properties. In hot-rolling structural shapes, each roll-pass makes a small change; up to ten passes are needed for some shapes.

Sintering

Sintering is a powder metallurgy process in which powdered material is compacted in a metal mold at pressures up to 200 ksi, depending upon the material composition. The process is carried out cold or with heat applied. If heat is applied, the temperature is kept below the melting point of the material, and it is sometimes carried out in a reducing atmosphere to minimize oxidation. The size of the particles and pressure promote atomic bonding. Depending upon the powders used, a wide variety of material characteristics can be obtained through sintering: for example, graphite can be added to bearing powders to improve lubrication, and cobalt can be bonded to tungsten carbide particles.

MATERIAL PROPERTIES AND PERFORMANCE

Properties

As most materials of interest to marine engineering are metal alloys, it is important to note that the alloying process usually involves mixing the alloy elements at temperatures high enough to allow them to disperse in the melted base metal. Thus the melting and boiling points ofthe elements are of much significance (see table 2-1).

MELTING POINT

Melting point is the transition temperature at which a metal can exist as either a solid or a liquid. The difference between the two states is that the liquid has absorbed the latent heat of fusion.

BOILING POINT

Boiling point is the transition temperature at which a metal can exist in either liquid or saturated vapor phases. The difference of the states results from the absorption of latent heat of vaporization by the vapor phase.

MATERIAL TESTING

Once engineering materials are produced, it is necessary to know their mechanical properties to design and produce efficient components and machines for marine engineering systems. To determine a material's mechanical properties, it is helpful to compare them to those of another material. A specimen of the test material is prepared in a standardized way and is subjected to loadings while its dimensional changes and energy absorption are measured, typically in accordance with ASTM specifications. Specimens are usually subjected to tension, compression, torsion, impact, and, in some cases, fatigue loadings. Specialized machines are used for conducting these tests. The figures that follow show examples of these machines and their uses.

Figure 2-2 shows a 120,000-pound-capacity universal test machine with tension specimen and extensometer (a gauge which measures specimen extension as the test proceeds) in place. In this application, the middle crosshead is fixed in place and the lower crosshead is attached to and driven by a hydraulically loaded piston mounted into the machine's base, which drives the upper crosshead through the four side columns. The hydraulic loading and the extensometer measurement of specimen elongation are recorded on the chart behind the operator.

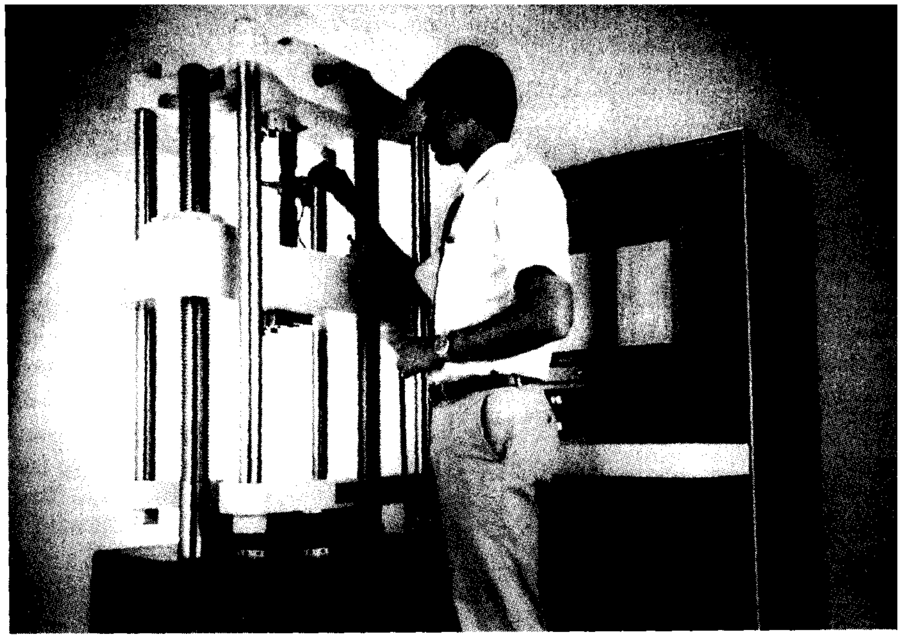

Figure 2-2. A 120,000-pound capacity universal test machine with tension specimen. Courtesy Tinius Olsen Testing Machine Co., Inc.

Figure 2-3 shows a 135,000-pound force test machine loading a compression specimen. The force magnitude is measured with a load cell. As the load head driving the lower head displaces, the upper crosshead remains fixed in the side support structure. Force and displacements can be recorded on appropriate instruments calibrated in U.S. customary, metric, or SI units, as desired.

Figure 2-4 shows a 10,000-inch-pound torsion testing machine (bench model) with the torsion specimen and torsion pick-up mounted and data recording components alongside. Considering the size of some engineering

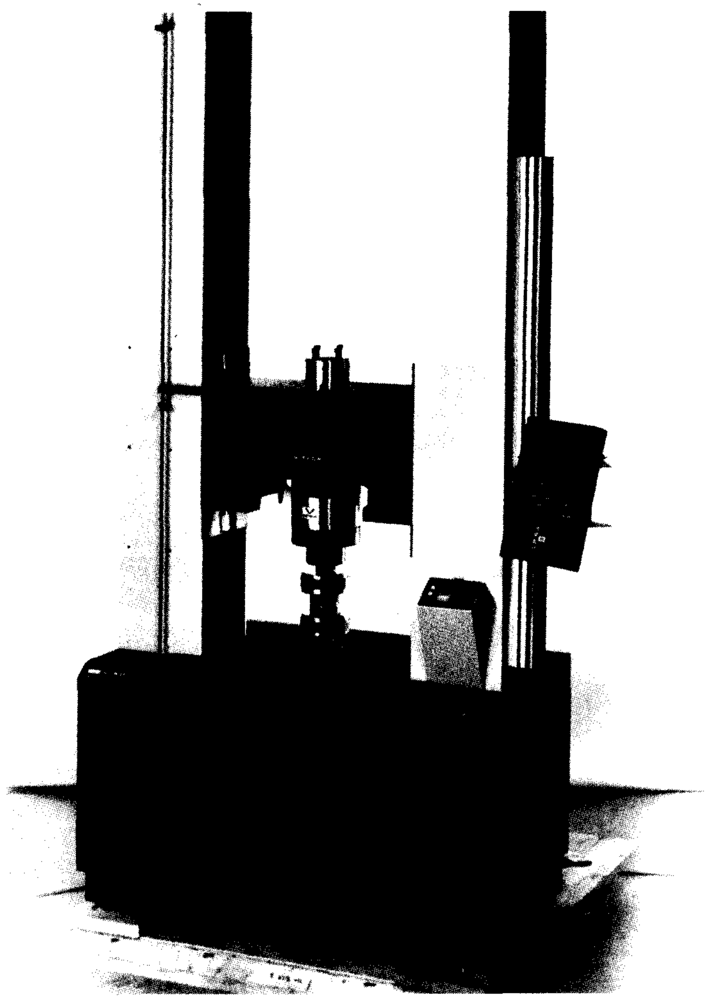

Figure 2-3. A 135,000-pound test machine with compression specimen. Courtesy Instron Corporation.

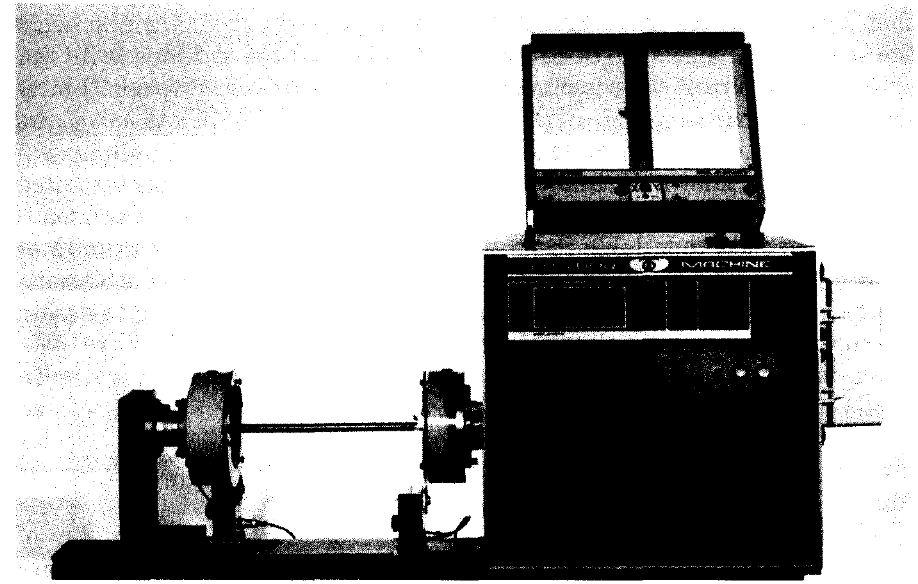

Figure 2-4. A 10,000-inch-pound torsion test machine with torsion specimen. Courtesy Tinius Olsen Testing Machine Co., Inc.

elements that transmit torque, particularly for power transmission shafts, it is readily appreciated that larger, floor-mounted torsion testing machines with 10 to 30 times this capacity are also available.

The energy absorption of BCC as well as some other types of metals when subjected to a high strain rate loading is important to satisfactory performance of many machines. Energy absorption is temperature sensitive; testing these metals by subjecting notched specimens to impact loading at various temperatures can provide valuable information about how they will respond under high strain rate loading (i.e., how much energy is absorbed in a specimen's rupture). In the U.S., two types of machines are used for gaining this information. One is the Charpy impact machine and the other is the Izod impact machine. The machines consist of a specimen vise and a pendulum with mounted striker. A test specimen with a v-notch cut into it (in accordance with ASTM dimensional specifications) is placed in the vise and the pendulum, raised to a preset position, is released to strike the test specimen in such a way as to place the v-notch root in tension. The difference in potential energy of the pendulum before and after striking the specimen is the energy absorbed during the specimen's rupture. Primary differences between the Charpy and Izod machines lie in the striking velocities (about 17 to 18 fps for the Charpy and 11 to 12 fps for the Izod machines), the type of vise used to hold the specimen, and the striker. The Charpy vise holds the specimen horizontally (over a 40 mm span) so that it can be struck in the back ofthe notch, while the Izod machine holds the specimen in a vertical cantilever position so that the striker can hit the extended portion of the specimen at a prescribed distance (about 22 mm) above and on the same side of the specimen as the notch. In both cases, the root of the notch is put into tension loading.

Figure 2-5 shows an impact tester of 300-foot-pound capacity fitted with a Charpy v-notch specimen vise (the Izod machine is similar). Calibrations are also available in metric (cm-Kg.) and SI (Joule) units. Machines of this type are used to subject the v-notched specimens at various temperatures within the range of interest to an impact loading to assess the material's notch toughness (energy absorption to cause rupture) as a function of temperature. As the notch toughness of steels varies significantly over the ranges oftemperature important to marine systems, classification societies specify minimum acceptable energy absorption for materials in temperature critical applications (See the ABS *Rules for Building and Classing Steel Vessels).*

Cyclic loading devices are used to determine fatigue characteristics of materials. Computer-controlled cyclictesting applications with appropriate data analysis capabilities are available from test machine manufacturers.

SYSTEMS OF UNITS FOR LENGTH, FORCE, AND TIME

To quantifY material properties and use the data in marine engineering design, a system of measurement units must be used. Three systems in common use in the United States are U.S. customary (English), metric, and the International System of Units (Sn. Table 2-8 indicates their unit names.

TABLE 2-8

Systems of Units and Dimensions

System of Units /Dimensions	*Length*	*Force*	*Time*
U.S. customary	foot	pound	second
Metric	meter	kilogram	second
SI	meter	newton	second

The standard meter and the standard kilogram are maintained at the International Bureau of Weights and Measures in Paris, France. A U.S. customary yard is defined, by act of Congress, in terms of the meter by the relationship: 1 yard = 3,600/3,937 meters. The foot and inch are determined therefrom. The U.S. customary pound is, for practical purposes, the same as the imperial pound of Great Britain, which is defined as 0.45359243 kilograms. The unit of time, the second, is internationally agreed to and currently defined in terms of fundamental vibrations of cesium 133. The National Institute of Standards and Technology (NIST) is charged with

maintaining U.S. prototypes of the meter, the standard kilogram, and time. For a more detailed discussion of these relationships, reference should be made to standard engineering handbooks such as Marks' or Eshbach's, or the ASTM *Standards for Metric Practice,* referenced at the end of this chapter.

COORDINATE SYSTEM, FORCE, AND EQUIVALENT FORCE SYSTEM

In order to relate objects, forces, and their senses in space and carry out systematic analyses, space coordinate systems are defined such that the

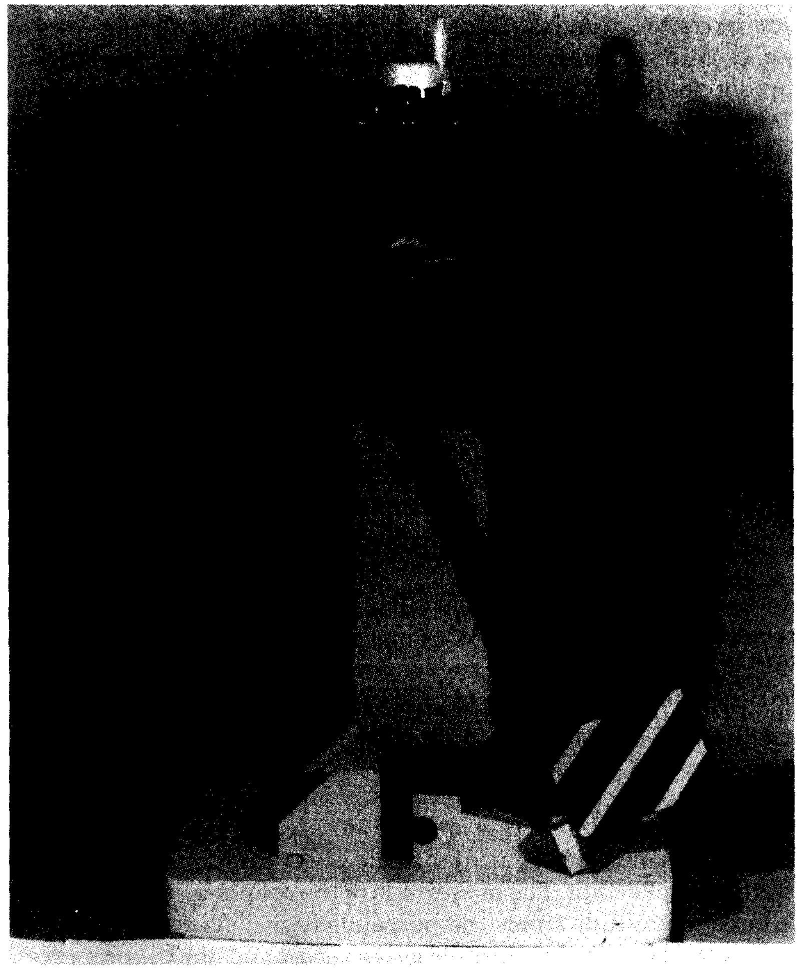

Figure 2-5. Impact tester with Charpy impact vise installed. Courtesy Instron Satec Systems.

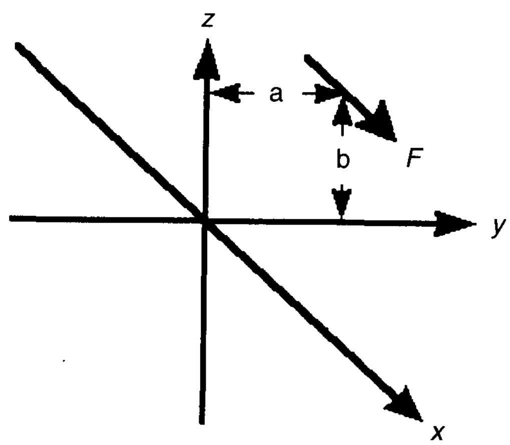

Figure 2-6. Space coordinate system

coordinate axes are mutually perpendicular, as illustrated by the right-hand space coordinate system of figure 2-6. The coordinate system may be fixed in space generally or in direct relation to body axes, whichever is most convenient for an analysis.

Force is an action of one body on another. A simple example is a weight (a mass in a gravitational field) hanging on a rod (a body whose lateral dimensions are small in comparison to its length). A force, F, has magnitude, a line of action, and direction, or sense, along that line of action, as illustrated in figure 2-6. These qualities of magnitude, sense, and line of action have a direct counterpart in a mathematical vector. Accordingly, vector mathematics provides tools for analytically treating forces and their actions on materials. Force is a constrained vector having transmissibility, that is, having the same force and effect on a rigid body if it is applied anywhere along its line of action. As a vector, a force can be resolved into components that are oriented in convenient coordinate directions and have vector component magnitudes consistent with those coordinate directions. Cartesian coordinates are most frequently easiest to use.

In engineering mechanics, it is demonstrated that a force acting on its line of action can be resolved into an equivalent force system composed of a force of equal magnitude acting on an alternate, parallel line of action and a couple about an axis perpendicular to the plane containing the original and alternate lines of action. The magnitude of the couple is equal to the product of the force magnitude times the perpendicular distance between the two lines of action. The couple is termed a free vector and has the property that it has the same rotational effect when applied anywhere on the rigid body. This concept is important, as forces seldom act on bodies in ways easy to analyze. Using this concept, forces are resolved into equivalent force-couple

systems acting at centroids of area or mass in respect to axes oriented in ways that allow simpler understanding of their effects.

LOAD: STATIC AND DYNAMIC

If a force is steadily applied to a body for an extended period of time, the force is referred to as a static load. When the force's magnitude varies in time, it is referred to as a dynamic load. Ifthe magnitude variation is relatively small in any given time period, and a dynamic response ofthe loaded material is not excited, the force is termed a quasistatic load, meaning that at any instant oftime the force and body on which it acts can be treated as statically related. A force that varies in a steady manner, oscillating between two magnitudes in a uniform, cyclic period, is referred to as a steady-state dynamic load. Applying and then removing a force in a short time is called transient loading, and if it is applied in varying magnitude over time but with no discernible application pattern, the force is called a random loading. A suddenly applied force is termed an impact load or impulsive force and it can excite large dynamic responses of materials so loaded (two or more times the static response), and the object loaded can develop multiple responses at various frequencies characteristic ofthe response modes. Design approaches can treat each ofthese force application types and material responses.

MOMENT

A moment is essentially the same as a couple, but may be of more complex origin than presented above. Consider a space having three mutually perpendicular axes, x, y, and z, which define coordinates of a reference system (see figure 2-6). If a force has a line of action that is parallel to one of the axes, say the x-axis, and perpendicular to the plane common to the other two axes, the y-axis and the z-axis, the force generates a moment about each ofthey andz axes, so long as the force's line of action doesn't intersect either of those axes. The moment about each axis will have a magnitude equal to the product of the force magnitude times the perpendicular distance from the force's line of action to the axis about which the moment is being generated. An equivalent force-couple system can thus be made of the force, F, acting parallel to the x-axis, and couples or moments about the other two axes equal to F times (b) about the y-axis and F times (a) about the z-axis. When discussing the responses of materials to loadings, flexural and torsional responses are two important modes of interest; moments relate to flexural responses while torques relate to twisting responses.

TORQUE

If a body having length, width, and depth dimensions, as in the case of a shaft, is subjected to a torque, T, acting about an axis oriented in the direction ofthe body's length dimension, the torque will cause the body to deform, rotating through an angle oftwist, e, about that axis (see fig. 2-7).

TWIST

A prismatic body oflengthL and circular section of radius p, subjected to a moment, couple, or torque, *T*, acting about its length axis will undergo a deformation in which a plane section transverse to that axis at one location will rotate relative to a plane section transverse to that axis at another location along that axis. This relative rotation is referred to as the twist of the body about the axis of moment, or torque action. The twist response, resulting from this torsion loading, is expressed in terms of an angle, e, in degrees or radians. (See fig. 2-7 for an illustration of twist.) The internal shearing stresses developed vary linearly outward from the center of a circular shaft and the shear strain, y, at the shaft surface, multiplied by the shaft material's shear modulus, G, yields the shear stress, 1, or *Ss*, at the shaft surface. Irregular section shapes are more complicated in their stress distributions; refer to strength of materials or mechanics of materials texts for further discussion.

STRESS

If the externally applied force, *P* (see fig. 2-8), is being transmitted internally through a body to its equilibrating force, *P*, acting on the same line of action but in the opposite sense, at the other end of the body, it is called a stress, that is, internally, the body is stressed. If the force magnitude, *P*, is divided by the body's cross-sectional area, *A* (refer to the shaded area of figure 2-8), perpendicular to the force's line of action, the quotient, *P / A = S =* cr, is

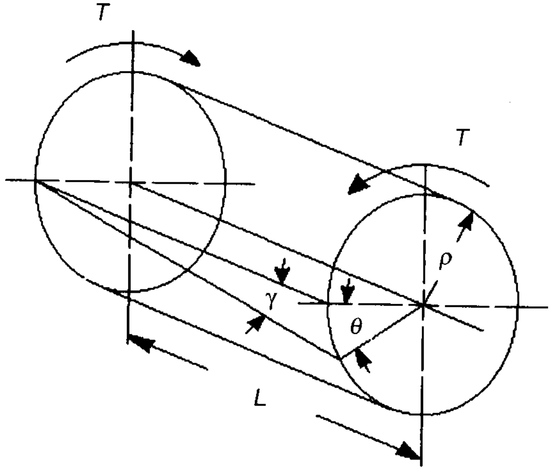

Figure 2-7. Shaft under torque

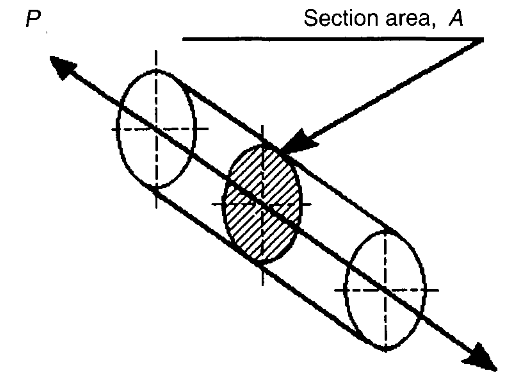

Figure 2-8. Axially loaded rod

termed a unit normal stress and has dimensions offorce per unit area (that is, the stress is acting normal, or perpendicular to the surface, *A*). For example, in the U.S. customary system of units, if the force *P* = 10 pounds and the cross-sectional area of the rod is 0.5 square inches, then the unit stress is *1010.5* = 20 pounds per square inch. In the metric system, if the force were 10 kilograms and the cross-sectional area were 5 square centimeters, the unit stress would be *10/5* = 2 kilograms per square centimeter. In the SI system, ifthe force were 500 newtons and the cross-sectional area were 10 square millimeters, the unit stress would be *500/10* = 50 newtons per square millimeter, or 50 megapascals, since 1 pascal equals 1 newton per square meter and there are one million (mega-) square millimeters in one square meter. It should be noted that civil engineers use *S* to define the unit normal stress, while mechanical engineers use cr. These symbols represent the same unit stress.

When a body in the shape of a rod, bar, shaft, or plate has varying cross-sectional areas along its length and is axially loaded, the unit stresses at various sections will change accordingly. At the location of a shape change, there will be a local stress concentration (unit stresses higher than the average unit stresses) as the unit stress in one section transitions to that of the next section. A stress concentration factor, *k*, which depends on the shape and shape changes of the body, can be determined that will relate the maximum unit stress to the average unit stress. This is an important factor in the design of machines (and ship structure); the interested reader should refer to a text on strength of materials or machine design for more information.

STRAIN

If a body, initially oflength *Lo*, is loaded by a tensile force, *P*, acting in the direction of the body's axis, as indicated in the figure 2-9a, it will be stretched to a longer length, L. Conversely, if a body, initially of length L_o is similarly loaded by a compressive force, P, as indicated in the figure 2-9b, it will be compressed to a shortened length, L. The difference between the body's final and initial lengths is called the axial deformation, 0, that is, the deformation of the body along the axis of the applied force. If the deformation is divided by the original length, the quotient is called the strain or, more precisely, the engineering strain. That is, engineering strain $= (L - L_o)/L_o = o/L_o = E$. It is seen that the strain can be either positive or negative, according to whether the load and thus the stress is positive (tensile) or negative, (compressive).

POISSON'S RATIO

Referring to figure 2-9, it should be noted that when the body is axially loaded, it not only changes dimensions axially, but laterally as well. Having an initial lateral dimension of D_o, after being loaded the lateral dimension becomes D. The difference between the final and the initial dimensions, $D - D_o$, is the lateral deformation, Olat. The ratio of the lateral deformation to the initial lateral dimension, $(D - D_o)/D_o = E_{lat}$, is the lateral strain.

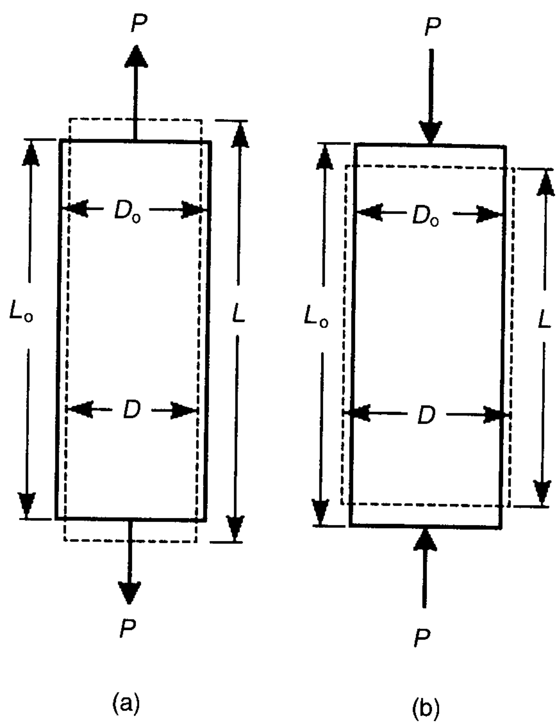

Figure 2-9. Strain

Noting that the sign of the axial and lateral strains are always opposite, a ratio can be written as $-E_{lat}/E_{axial} = $ ~ (sometimes v is used to represent this ratio). This ratio is called Poisson's ratio and it is an important characteristic of solid materials such as metals and plastics. This ratio ranges from about 0.1 for concrete, to about 0.5 for rubber, being about 0.29 for steel and 0.33 for aluminum.

Two other important characteristics of materials, discussed below, are the elastic modulus, E, and the shear modulus, G. It can be shown that these three values are directly linked: $G = E/2(1 +)$. Thus, when material properties are being measured, as in a laboratory test machine, only two of these three can be independently determined. It can be readily appreciated that if the material lateral dimensions only decrease 29 percent as much as the axial elongation, as in the case of steel, the body being stressed is actually getting larger. That is, it is dilating. It can also be appreciated that pure rubber, with a Poisson's ratio of about 0.5, pretty much retains its volume under load.

FLEXURE

When a prismatic body has one dimension that is much larger than its other two, and it is acted upon by two moments about axes perpendicular to its length, as indicated in figure 2-10, where the wall support at left provides the restraining moment, the body will be caused to deform in a circular arc. For a body to be in equilibrium, the sum of forces along and moments about any axis must be equal to zero. Therefore, if a moment is applied to the beam, the resisting moment must also be applied, in this case by the support at left. The body is said to be flexed and the mode of response to such a loading is called flexure. The body is called a beam; the bending stresses within the beam vary linearly from the beam's cross-section centroid, as indicated at right in the figure. The upper fibers of the beam are in compression while the lower fibers are in tension.

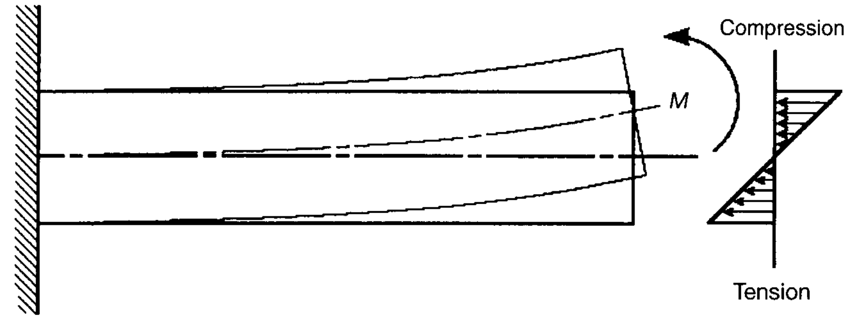

Figure 2-10. Flexure

THE TENSILE TEST

The most typical test a material is subjected to is a tensile test (see fig. 2-2). In this test, the specimen is prepared in accordance with the ASTM specifications and, after insertion in the test machine, it is loaded through a range, the elongation ofthe gauge length is noted, and the stress/strain relationship is determined. In figure 2-11, three types of stress/strain curves are depicted. Curve A represents a brittle material showing the elastic response up to a fracture without significant inelastic deformation at point (d). The indication is also that the elastic modulus of the material is quite high, the stress/strain line being quite steep. Curve B displays a linear elastic range up to (a) and a nonlinear elastic-plastic range from (a) to (b), with plastic deformation taking place after (b). The stress continues to rise as strain is increased beyond (b), caused by strain-hardening of the specimen material. At point (c), the maximum stress is reached, after which the stress falls off as strain continues. In this region, (c) to (d), the specimen cross-section is necking down as the plastic deformation continues up to rupture at (d). If this specimen were a steel, it would likely be a cold-rolled steel that no longer has a sharply defined yield point, as would be the case ifit were hot-rolled steel. In the case of a poorly defined yield point, the offset method of defining yield is used, as, say, a 0.2-percent offset. Here the line parallel to the linear-elastic portion of the curve is extended to

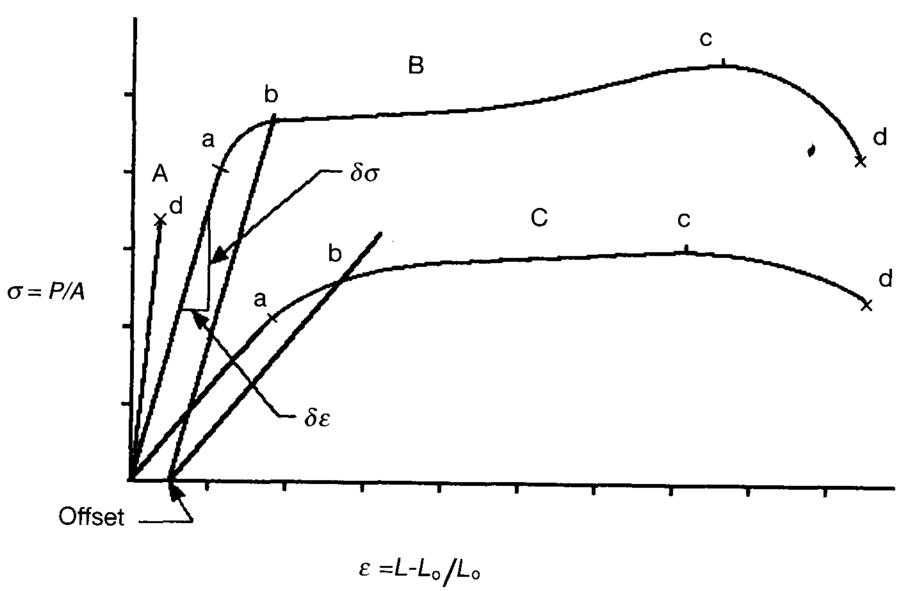

Figure 2-11. Stress versus strain diagram

intersect with the stress/strain curve. The stress at the intersection is then denoted as the yield stress.

Curve C represents a material of lesser elastic modulus and strain hardening, but exhibiting similar properties to that of curve B. Note that the representation in the figure is for illustration purposes only and is not to scale. Actually, the elastic response portion of a steel would be a much smaller portion of the diagram than depicted and certainly the plastic response portion of a ductile material would be much larger than could be put into the figure without a major change of strain scale.

STRENGTH

Tensile strength is the maximum load a test specimen can sustain, divided by the original cross-section area of the specimen, expressed in the appropriate system of units as force per unit area.

To understand ultimate strength, refer to figure 2-11, curves Band C, where points (c) indicate the maximum stress of the test specimens. Mter reaching this stress level, the stress falls off as the strain continues to increase and approach rupture at points (d). (Commonly, the term *tensile strength* is used to mean ultimate strength.)

Yield strength is the force first causing inelastic behavior in a test specimen divided by the initial cross-section area of the specimen, expressed as force per unit area. When the commencement ofinelastic behavior (between points [a] and [b] offig. 2-11) is poorly defined, yield stress can be defined by determining the inelastic load and deformation relationship causing a specified permanent deformation within the measured gauge length. This is done by plotting the stress-strain curve for a material and extending a line parallel to the elastic stress-strain line, but starting at a specified initial strain, such as 0.2 percent, up to the defined curve. The intersection determines the offset yield stress (see fig. 2-11).

DUCTILITY

Ductility is a material's ability to undergo inelastic deformation prior to rupture, assessed as percent elongation, percent reduction of area, or specific specimen deformation, when subjected to a specific test. Material ductility is often specified in important applications. Percent total elongation of a test specimen of specified gauge length at specimen rupture is one important measure. A test specimen, having an initial 2-inch gauge length, and a 0.5-inch elongation at rupture, the gauge length now being 2.5 inches, shows a 100 (0.5/2.0) = 25 percent elongation at rupture. The specimen's ductility is twice that of a similar specimen that experienced o,nlya 0.25-inch elongation at rupture, i.e., a 12.5 percent elongation. Another measure of ductility is percent reduction in specimen cross-section area at rupture. As a material deforms inelastically, test specimen gauge length increasing with loading in the post-yield range, lateral dimension(s) of the test specimen decrease(s) as does cross-section area. The greater the percent reduction in the specimen's cross-section area at rupture, the greater the material's ductility. A cylindrical test specimen with an initial diameter of 0.505 inch and ruptured section diameter of 0.350 inch has an initial cross-section area of 0.20029 square inches that is reduced to 0.09621 square inches, an area reduction of 0.10408 square inches. The percent reduction is thus 100 (0.10408/0.20029) = 51.965 percent.

A bend test is also used to assess satisfactory ductility. A plate specimen bent through 180 degrees, with an inner bend radius three times the plate thickness, without initiating rupture, could be considered to have adequate ductility. Ship classification societies use this test also.

RESILIENCE

The amount of elastic energy that can be stored and retrieved from a metal specimen under loading is a measure of the metal's resilience. A modulus of resilience is formed by noting that in the linear elastic range of response, the material's internal unit stress, cr, is proportional to the unit strain, E, where the elastic modulus, E, is the proportionality constant relating the stress to the strain, E = Ocr/OE of figure 2-11. Thus, in the linear elastic range, cr = E . E, and the area under the stress-strain curve up to the limit of proportionality, a triangular area, is cryp2/2E in consistent units.

TOUGHNESS

Toughness is the ability of a material to undergo high unit stress, or elasto-plastic deformation, before rupturing. This is defined in terms of total work to produce elongation over a specified gauge length of a test specimen that has been loaded and extended up to the point of rupture. Referring to the stress-deformation curve of figure 2-11, for the standard test specimen and gauge length, the relative toughness of two specimens compare as the area under their stress deformation curves, from the origin of the curves out to rupture at points (d).

BRITTLENESS

The inability of a material to absorb significant work prior to rupture is its brittleness. Glass is an example of a brittle material as it typically ruptures without significant inelastic deformation (curve A of fig. 2-11).

RUPTURE

Rupture is the point on the stress-strain curve at which a material separates into two pieces (points [d] of fig. 2-11).

ELASTICITY

Elasticity is the degree to which a material returns to its original shape after a loading is removed.

MODULUS OF ELASTICITY

Modulus of elasticity refers to the proportionality constant, E, relating axial unit stress, cr, to strain, E, in the expression cr = E . E. The value of E is determined from the slope, Ocr/OE, of the stress-strain curve in the linear elastic range, obtained from a tensile test specimen or a compression test. For materials such as aluminum, concrete, or copper having a limited linear elastic range of response, the lower range is used for this determination (see fig. 2-11).

SHEAR MODULUS

The shear modulus is the proportionality constant, G, relating unit shear stress, 't, to shear strain, y: 't = G . y.

CREEP

When a solid material such as concrete, lead, or steel experiences plastic deformation while subjected to modest loadings, i.e., below the material's yield strength, it is said to have undergone creep: the unyielding plastic deformation of a stressed material. Creep is a function of stress magnitude, temperature, and time. Materials stressed for an extended period of time tend to undergo this inelastic (plastic) deformation, with resultant dimensional changes. For instance, an alloy of steel, stressed to 4,000 psi and heated to 900°F could creep by 0.2 percent over a ten-year period. The same steel, stressed to 8,000 psi and heated to the same temperature, could stretch 1 percent in ten years. Although the temperatures of civil structures are much lower, this fact of creep is of consequence to their design. Foundations, meant to last for many decades, and high temperature components of engineering structures subjected to high operating stresses must be designed with this in consideration. Because the yield strength of steel decreases at elevated temperatures, creep is of importance in the design of steam drums, high-pressure turbines, main steam piping, and other high-temperature, high-pressure components of engines or continuously operating systems. Engine pistons and cylinder parts, for example, being heated and under high pressure while in operation, can grow due to creep.

COMPRESSIVE STRENGTH

For ductile materials, compressive strength is taken to be the same as tensile strength since lateral strain increases specimen cross-section area, the maximum load not reaching a determinable limit as for the case of tension. For brittle materials, where failure may occur by the material crumbling or fracturing along a plane of high shear stress, compressive strength is defined as the maximum load borne prior to failure divided by the original cross-section area of the specimen.

BUCKLING STRENGTH

When structural elements with lateral dimensions significantly narrower than their length are subjected to compressive loading, they are susceptible to buckling, i.e., undergoing large lateral displacements with consequential axial shortening, thereby shirking their axial compressive loading. These slender elements may be columns or thin plates, important for compressive load transmission; knowing the critical loading above which buckling or yielding may occur is crucial to design of structures. The material's yield stress, σ_{yp}, Euler load ($P_{cr} = \pi^2 EI/L^2$), and Secant column formula ($\sigma_{max} = (P/A)\{1 + (ec/r^2)\sec[(L/2r)(P/AE)^{1/2}]\}$) provide tools for predicting critical loads on columns. The Secant formula allows consideration of load application eccentricity, e, where c is the distance from the section neutral axis to the extreme fiber, and $r = (I/A)^{1/2}$, I being the least transverse moment of inertia of the column cross-section, A being the column cross-section area, and L being the column length. The Bryan formula, $N_{cr} = k\pi^2 D/b^2$, provides a theoretical counterpart to the Euler formula when considering plates. In this formula, N_{cr} is the critical axial loading of the plate edge being loaded, k is a factor depending upon the buckling mode and the aspect ratio of the plate element, b is the plate width over which the axial loading is applied, and D is the unit stiffness of the plate, $Eh^3/12$, with E being the material elastic modulus and h being the plate thickness. For more detailed information on buckling, the reader should refer to standard texts dealing with strength of materials and the theory of plates and shells.

SHEAR STRENGTH

Shear strength is the maximum transverse shear force a material can withstand divided by its cross-section area. This is important for such fastening devices as bolts and rivets that transfer forces between members by shear at connection interfaces, or punching through metals. It refers primarily to ductile fasteners that undergo large shear deformation prior to failure, in which the ductile material is essentially uniformly stressed at the yield stress level in shear. The value is determined by testing a material specimen in direct single or double shear in a test machine. The importance of fasteners has resulted in establishment of standards by the International Standards Organization (ISO) as well as SAE and ASTM, which should be referred to for design application.

HARDNESS

Materials are subject to damages by surface scratching and penetration. This has implications as to wear resistance, machinability, and material yielding under load. Though there are probably several factors of importance to these material properties, assessment of a material's resistance to such effects is carried out by hardness or scratch testing. Several scales and related instruments have been devised as means for determining

relative values. The Mohs scale is used to make comparisons of mineral scratch resistance, while for metals the Brinell and Rockwell hardness methods are employed. In the latter two methods, the material surface is penetrated by an indenter applied under a loading. The size of the penetration, depth, or surface diameter determines the scale value of material

Figure 2-12. Wilson 2000 Rockwell hardness tester. Courtesy Instron Corporation.

hardness. The size of the indenter and loading determines the scale used for interpreting hardness. Testers are calibrated to read hardness directly, and must be periodically calibrated using standard specimens (see fig. 2-12 showing a Rockwell hardness tester). The hardness can also be approximately related to other material properties (see ASTM specifications).

MALLEABILITY

The degree to which a material can be bent, rolled, hammered, or otherwise deformed without development of cracks or other failure is a measure of malleability. This is similar to ductility, where the percent elongation and reduction of section area of a test specimen are measured.

WELDABILITY

Weldabilityis the degree to which a metal will develop 100 percent joint efficiency upon being welded.

FATIGUE STRENGTH

The repeated or cyclic unit stress level a material can sustain without failure is its fatigue strength. This is an important consideration in marine structures and machine components, as the loadings on engines, shafts, propellers, and the ship's hull structure are generally oscillating. Not only the stress oscillations, but the average stress as well, have importance to material fatigue life.

COEFFICIENT OF THERMAL EXPANSION

Macroscopically, metals are considered to be homogeneous. They therefore undergo homogeneous dimensional change when subjected to a uniform temperature variation. The amount of dimensional change, while generally a function of the material's temperature as well as the temperature change, will be nearly linearly related to the temperature change over limited ranges. Thus, it is estimated that the change of dimension can be expressed as $\alpha = (L - L_o)/(\Delta T \cdot L_o)$ where L is the final dimension, L_o is the initial dimension, α is the coefficient of thermal expansion for the material, and ΔT is the temperature change that has taken place. The coefficient is expressed in terms of change in length per unit of length per degree temperature change and is generally a rather small amount. However, engineering structures are often quite large and are composed of various materials, many of which may have significantly different thermal expansion coefficients. This causes significant differential expansion or contraction of the structure's materials and must be carefully considered. Table 2-9 gives approximate thermal expansion coefficients for several important metals.

TABLE 2·9

Coefficients of Thermal Expansion

Metal	*Coefficient* $\times 10^6$ *per degree F (C)*
Aluminum	12.5 (22.5)
Copper	9.3 (16.7)
Brass (70-30)	11.0 (19.8)
Bronze (95-5)	10.0 (18.0)
Gold	8.2 (14.8)
Lead	15.2 (27.4)
Magnesium	14.5 (26.1)
Nickel	5.7 (10.3)
Silver	10.7 (19.3)
Steel (1020)	6.5 (11.7)
Tin	12.7 (22.9)
Zinc	16.5 (29.7)

Performance

SAFETY FACTOR

All load-response relationships involve uncertainty. Expected loadings may not in actual fact be realized, and the realization may be greater or less than anticipated in any time span. Material properties prescribed for a machine may not be realized either, as the possibility of greater strength or a material defect is always present. Machine geometry may be somewhat different than specified due to production being carried out within manufacturing tolerances at each level of material processing. Further, the importance of a machine to the success of a venture requires that it not fail to perform its intended function under any reasonable set of circumstances. If safety of human life is at issue, it is even more important that failure be avoided. Put another way, given that risk of failure cannot be totally avoided, it is important that risk of failure be kept within an acceptable range of possibility. To meet this need, a factor of safety can be prescribed to effectively separate the likelihood of loading from the likelihood of failure by a prescribed amount. For example, if a cable loading slightly exceeds the cable's breaking strength, the probability offailure is unity. If the loading is but one-quarter the breaking strength, there would be a factor of safety of four between the load, or demand, and the breaking strength, or capability. That is, the factor of safety is the ratio of capability to demand. The capability may be expressed in terms of anyone of a variety of aspects, such as rupture, fatigue, maximum carrying capacity, or maximum elastic capacity (before permanent deformation). The more that is known about a system's loading and capability, the lower the factor of safety can be, as the risk of exceeding capability is reduced.

RELIABILITY

All mechanical systems are subject to failure. Failures causing system shutdown reduce availability and, accordingly, reliability. When system faults are corrected and service resumes, availability is restored, but not reliability, as the fault may recur. Historical records of system or component failures and repairs can be used to determine the mean time between failure (MTBF) and th~ mean time to repair (MTTR) for a system or its components. Then, system availability is determined as availability = operating time/(operating + repair time), or availability = MTBF/(MTBF + MTTR). For multiple system components, independently subject to failure, net system availability is the product of individual availabilities. To determine system reliability, all operational scenarios and failure modes must be evaluated and appropriately weighted. It is obvious then that availability, and thus reliability, can be increased by providing system or component redundancy. In so doing, MTTR will be reduced by having a redundant system or component to bring into use upon a failure incident. System MTTR is then replaced by mean time to restore system function, by bringing a redundant system or component online.

GALVANIC PROPERTIES

Metals are able to easily give up electrons in their outer electron shell when it is not filled; this determines a metal's relative potential when part of a galvanic cell. This property allows metals to be ranked in a galvanic series from ignoble to noble, magnesium to gold. Any two metals connected by a conducting path and immersed in a conducting medium like seawater constitute a galvanic cell. In order to stop galvanic cell operation, the potential of the cell must be overcome, as by inserting anodic potential into the cell. The anode to be inserted can be a less noble metal or have electrical potential sufficient to turn anodes cathodic.

CORROSION

Corrosion of metals in the marine environment is an important problem requiring careful attention. Corrosive deterioration of metal results from chemical or electrochemical attack. Direct chemical attack by strong acids or alkalies is controlled by selection of materials and treatment of the fluids. Boiler and feedwater as well as engine cooling water are examples in which treatment is carefully exercised. Control of free oxygen, which would react with iron to form iron oxide, and hydrogen, which is an indicator of the rate of corrosion, reduces the potential for attack on boiler heating surfaces. Oxygen causes direct pitting attack on heating surfaces and other dissolved gases accelerate the attack.

Chemical attack of structural materials is mostly due to rusting and pitting corrosion of steel, the presence of mill scale accelerating the corrosion processes. Electrochemical attack by galvanic action is particularly threatening to the ship structure and related systems, as seawater provides the electrolyte for the galvanic action processes. The anode is the less noble metal and the metal that could dissolve and the cathode is the more noble metal and the metal on the surface of which hydrogen ions plate-out. In the galvanic series, magnesium, aluminum, zinc, iron, brass, copper, bronze, stainless steel, silver, titanium, platinum, and gold range from most anodic to most cathodic, so the steel elements are anodic to copper-based elements. It is important then to provide potential to the surfaces to be protected so they become cathodic and the galvanic action ceases. To this end, zinc, aluminum, or magnesium anodes are attached to shield steel surfaces from galvanic attack. An alternate method is to impress a low voltage on the galvanic cell, which makes the steel cathodic and stops the process. Both of these methods are used aboard ship.

COATINGS

A variety of coatings are available to protect finished surfaces from deterioration in the operating environment. Of particular interest to marine engineers are paints that may be applied to seal the surface, providing protection from oxidation and galvanic action as well as an attractive appearance. Coating systems for the marine environment typically consist of one or more coats of anticorrosive primer, usually rich in zinc, and one or more base and finish coats, depending upon the surface being protected and expected surface wear conditions. Paints are composed of pigment to give surface protection with extenders to give the paint film required properties, binder to make the paint adhere to the surface and add film resistance to environmental damage, and solvent to control paint fluidity and ease of application. The solvent is first to evaporate, forming a surface film.

To apply an effective coating system, the material to be coated must be free of oil, dirt, rust, scale, and other contaminants. This may be accomplished by sand blasting, solvent cleaning, hand or power tool cleaning, power brushing, or grinding. The Steel Structures Painting Council (SSPC) has developed a set of standards that, if applied, ensures good quality work.

REVIEW

1. What metal element is most abundant in the earth's crust?
2. What metal element is most commercially important?
3. Why are metals used primarily in the form of alloys?
4. What alloying element is in all steels?

5. If a bronze alloy has 0.25 percent phosphorus content, how will this affect alloy properties?
6. What metal is most malleable? What metal is the second most malleable?
7. What is a measure of ductility? What is a measure of toughness?
8. What is the characteristic of a brittle material?
9. What crystal structure is typically found in ductile metals?
10. What is the difference in composition of brass and bronze?
11. What are the main elements of babbitt? What element can be added to harden babbitt?
12. What are the main elements of solder? What solder composition gives the strongest joint?
13. Which metal is the best electrical conductor? What metal is better, on a weight basis?
14. Name three types of corrosion, how they take place, and how to protect metals against them.
15. What is meant by elongation? How is it evaluated?
16. What is meant by hardness? How is it measured?
17. Name three methods for joining materials. Which method is most commonly used?
18. How would you determine the strength of a material? What equipment would you use?
19. What insulating material would be suitable for use on a main steam pipe?
20. Name three processes of heat transfer that must be protected by insulation.

REFERENCES

American Bureau of Shipping (ABS). 1994. *Rules for Building and Classing Steel Vessels.* New York: ABS.

American Society for Testing Materials (ASTM). 1979. *Standard for Metric Practice,* ASTM Designation E 380-76. Philadelphia: ASTM.

Avallone, E. A., and T. Baumeister, eds. 1987. *Marks' Standard Handbook for Mechanical Engineers,* 9th ed. New York: McGraw-Hill.

Callister, W. D., Jr. 1994. *Material Science and Engineering: An Introduction,* 3rd ed. New York: John Wiley & Sons.

Clauser, H. R., ed. 1963. *The Encyclopedia of Engineering Materials and Processes.* New York: Reinhold.

Hampel, Clifford A., ed. 1968. *The Encyclopedia of the Chemical Elements.* New York: Reinhold.

Heiserman, David L. 1992. *Exploring Chemical Elements and their Compounds.* Blue Ridge Summit, Pa: TAB Books, McGraw-Hill.

Latimer, W. M., and J. H. Hildebrand. 1951. *Reference Book of Inorganic Chemistry,* 6th ed. New York: Macmillan.

McGannon, H., ed. 1964. *The Making, Shaping and Treating of Steel,* 8th ed. Pittsburgh: U.S. Steel Corp.

Tapley, B. D., ed. 1990. *Eshbach's Handbook of Engineering Fundamentals,* 4th ed. New York: John Wiley & Sons.

U.S. Coast Guard. 1990. "Use of Fiber Reinforced Plastics in the Marine Industry." Ship Structure Committee Report SSC-360. Washington: U.S. Coast Guard.

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CHAPTER 3

Steam Power Plants

JAMES A. HARBACH

The steam power plant is one of the major systems used to power ships and generate electricity. It was the steam plant that replaced sail for powering ships in the nineteenth century. While the diesel engine and the gas turbine are now being selected over the steam plant in many applications, it still remains a viable choice in certain situations. In this chapter we will look at the steam cycle from theoretical and practical perspectives and will examine some associated systems and operating procedures.

IDEAL VAPOR CYCLES

The Vapor Carnot Cycle

In chapter 1, the Carnot cycle was introduced. A Carnot cycle is the most efficient cycle operating between two temperature limits. The cycle consists of four processes: constant-temperature heat addition and removal, and isentropic expansion and compression. If a Carnot cycle is based on boiling a liquid and then condensing the vapor, however, some problems occur (see fig. 3-1). Since the steam entering the turbine is saturated, the moisture content of the steam at the turbine is above the 10 to 15 percent considered acceptable by most turbine manufacturers. Also, the compressor would have to be designed to handle a liquid-vapor mixture, and it is difficult to control the partial condensation process.

Ideal Rankine Cycle

The Rankine cycle overcomes many of the problems of the Carnot cycle. The heating and cooling cycles are constant-pressure rather than constant-temperature (see fig. 3-2a and 3-2b). Feedwater enters the boiler as a subcooled liquid at high pressure at point 1. Heat is added to the water at

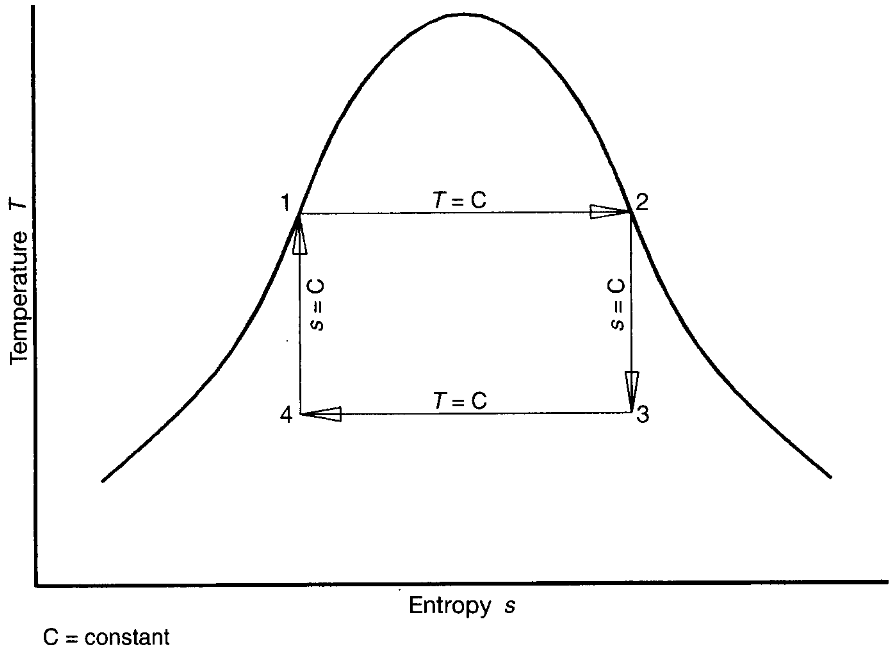

Figure 3-1. Temperature-entropy diagram for a Carnot cycle

constant pressure, leaving at point 2 as either a saturated or superheated vapor. The high-pressure steam is then expanded through the turbine, exiting at point 3 as a low-pressure mixture. The mixture is condensed at constant pressure, leaving the condenser at point 4 as saturated liquid. A pump is used to increase the liquid pressure to point 1 and the cycle begins again.

Note how the superheating of the steam in the boiler reduces the moisture content ofthe steam leaving the turbine exhaust. Condensing the turbine exhaust to a saturated liquid solves the partial condensing problem and also allows the use of a pump rather than a compressor.

Improving Ideal Rankine Cycle Efficiency

Engineers are continually seeking ways to improve the efficiency of power plants. Since a typical ship's propulsion plant will burn millions of dollars worth of fuel in a year, even a small increase in efficiency can result in significant dollar savings. The thermal efficiency of a power cycle was defined in chapter 1 as follows:

$$E_{th} = \frac{W_{net}}{Q_{in}} = \frac{Q_{in} - Q_{out}}{Q_{in}}$$

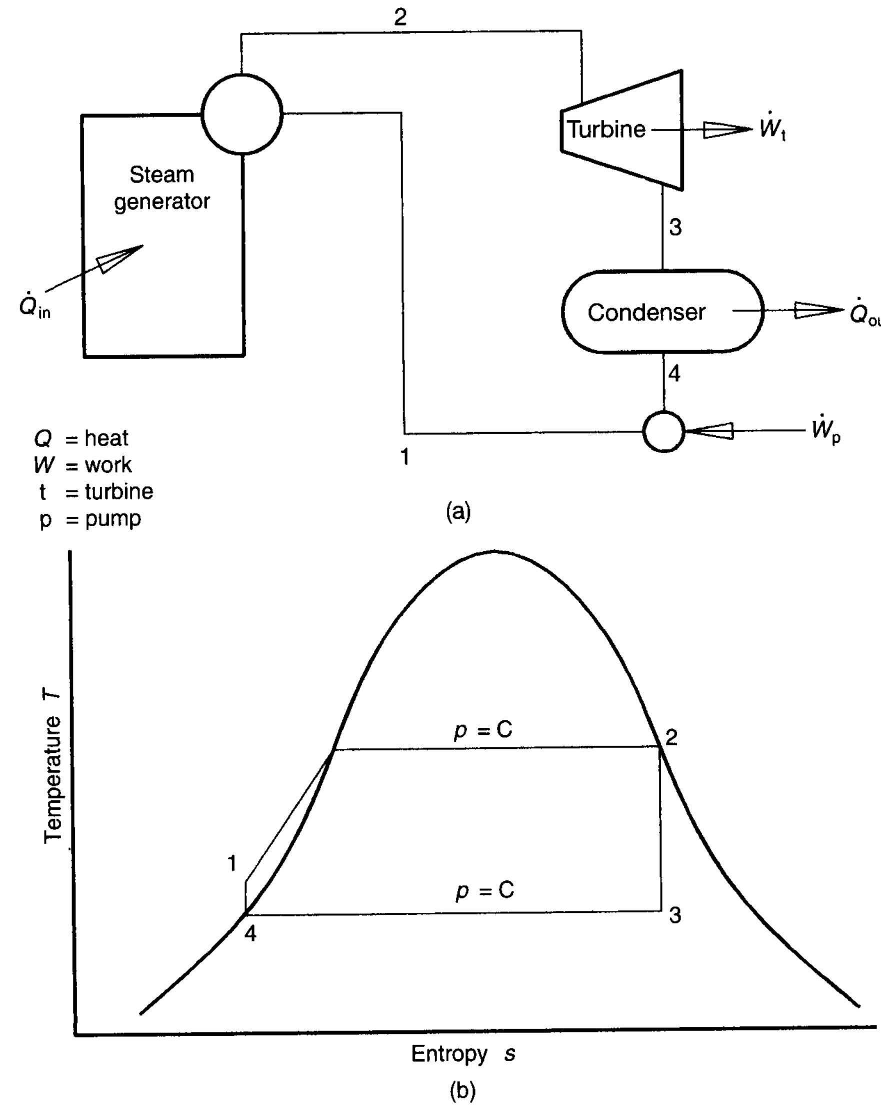

Figure 3-2. Ideal Rankine cycle

Figure 3-3 illustrates the relationship between the net work and the heat added and removed for a Rankine cycle. A little study of the figure will reveal that a couple of things could be done to improve the cycle efficiency. Lowering the condenser temperature by improving the vacuum would reduce the amount of heat rejected. Each reduction in heat rejected results in an increase in the net work. Obviously, there is a limitation on how much the vacuum can be improved-the condensing temperature can't be below the seawater temperature.

Another thing that would increase the net work would be to raise the heat addition process line (point 1 to point 2). This can be accomplished by either raising the steam generator pressure or raising the superheat temperature. Since raising the pressure alone increases the moisture content of the steam turbine exhaust, it makes sense to raise the superheat temperature at the same time. The increase in superheat temperature reduces the exit moisture content, compensating for the increase caused by the pressure increase, and both contribute to an increase in efficiency.

The cost and availability of materials that withstand high temperature and pressure puts practical limits on their use. In general, the higher the horsepower of the plant, the more the use of higher pressure and temperature is economically justified.

Ideal Regenerative Rankine Cycle

There is a limit to how much the efficiency of the simple Rankine cycle can be improved. The next step to increasing efficiency, then, is to consider modifications to the basic cycle. If there were a way to preheat the feedwater prior to it entering the boiler, less heat would need to be added to produce the high-pressure, high-temperature steam.

Let's consider extracting some steam from the turbine after it has been partially expanded and using it to preheat the feedwater (see fig. 3-4a and 3-4b). The preheating would occur with no increase in energy input in the boiler. It would also reduce the quantity of steam exhausted from the turbine and would thus reduce the heat rejected to the sea. A steam cycle using this type of heating is called a regenerative cycle.

The heaters used in the regenerative cycle are of two basic types-open and closed. In the open or direct-contact type, the steam and feedwater mix together within the shell of the heater. A second benefit of an open heater is that any air or noncondensable gases come out of solution, allowing the heater to also act as a deaerator. In a closed heater, the condensing steam and the feedwater are kept separated. Most closed heaters are shell-and-

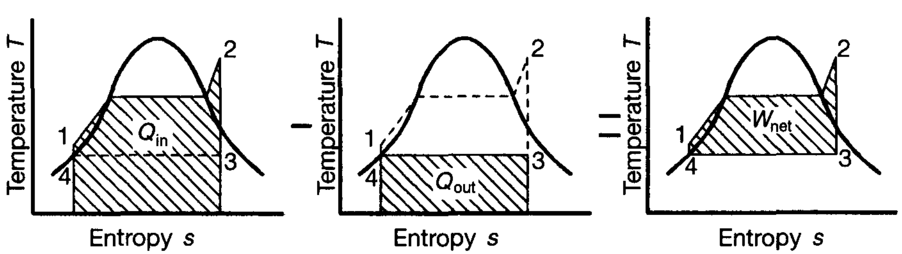

Figure 3-3. Net work, heat added and heat rejected for a Rankine cycle

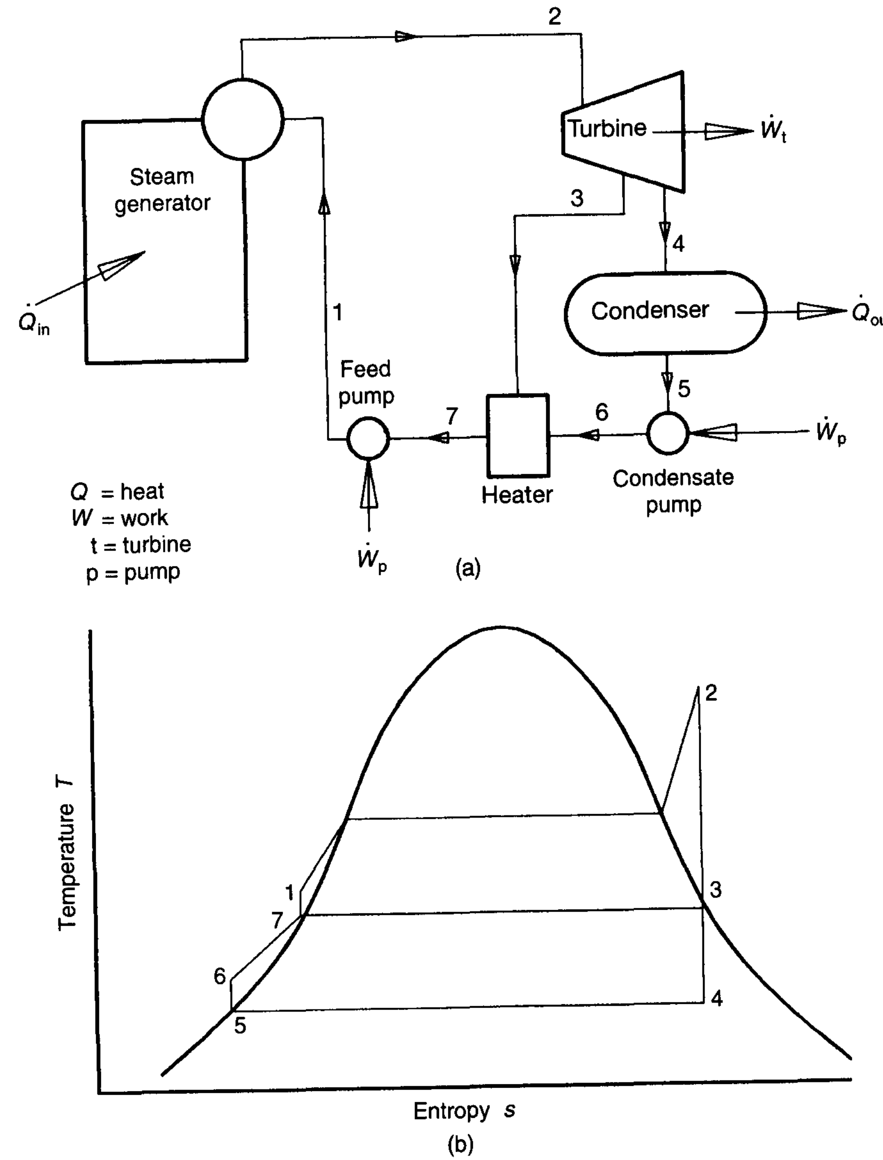

Figure 3-4. Ideal regenerative Rankine cycle diagram

tube heat exchangers, with the feedwater flowing through the tubes and the condensing steam in the shell. The drains from the shell can either be pumped into the discharge line of the heater or be returned to the shell of a lower pressure heater. While there is a slight efficiency benefit to pumping the drains forward, the cost of purchasing and maintaining the pump is not usually justified.

An important consideration is where to extract the steam from the turbine for maximum benefit. If steam directly from the boiler is used, it is no different than if the water were heated directly in the boiler, Le., a simple Rankine cycle. If exhaust steam from the turbine is used, no preheating of the feedwater occurs. Obviously, the optimum location lies somewhere between these two extremes. For a single heater system, the optimum point lies where the heater outlet temperature is halfway between the saturated steam temperature in the boiler drum and the condenser temperature. This results in half of the heating of the feedwater to the boiling point occurring in the heater and half in the boiler. If two heaters are used, the first heater accounts for one-third of the feedwater temperature rise, the second heater accounts for the second third, and the last third occurs in the boiler. This "equal temperature rise" principle can be extended to any number of heaters using the following equation to calculate the optimum temperature rise:

$$\Delta T_{opt} = \frac{T_{blr} - T_{con}}{n+1}$$

Feedwater heaters are also classified as high-pressure and low-pressure. Most marine cycles are fitted with an open deaerating heater supplied by bleed steam from the crossover between the high-pressure (HP) and low-pressure (LP) turbines. All heaters supplied by bleed steam from the HP turbine are classified as high-pressure. All heaters supplied by bleed steam from the LP turbine are classified as low-pressure.

Ideal Reheat Rankine Cycle

Another modification to the simple Rankine cycle that will improve efficiency is to reheat the steam after it has been partially expanded in the turbine. Steam expands in the high-pressure turbine, is returned to the boiler, where it picks up heat from the gases of combustion, and then is returned to the turbine to complete expansion down to the condenser pressure (see fig. 3-5a and 3-5b). Reheating increases the average temperature of the addition of heat to the cycle, thus increasing the cycle efficiency.

Obviously, the pressure at which the steam is extracted for reheating will affect the improvement in efficiency achieved. For a single stage of reheat, the reheat pressure should be about 15 to 20 percent of the superheater outlet pressure for maximum improvement. A second benefit of reheating is that it lowers the moisture content of the steam in the last stages of the low-pressure turbine (compare point 5' for the nonreheat cycle with point 5 for the reheat cycle on figure 3-5b).

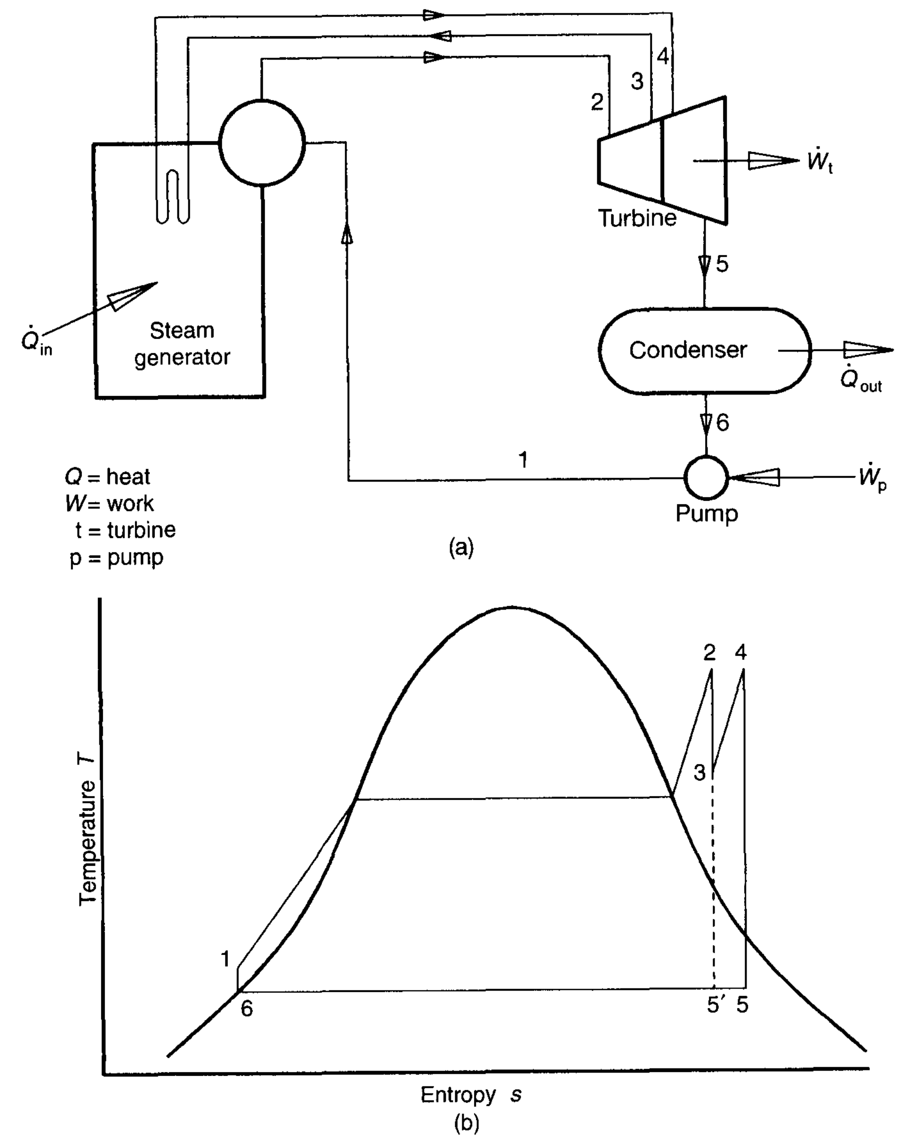

Figure 3-5. Ideal reheat Rankine cycle

ACTUAL MARINE STEAM POWER PLANTS

As powers have increased and fuel costs have escalated, the typical marine steam power plant has evolved over the last fifty years from a simple cycle with two feedheaters and steam conditions of 450 psig and 750°F, to modern cycles with four to six feedheaters and steam conditions of 850 to 1,450 psig and 950° to 1,000°F. Reheat is found on the highest performance cycles.

In addition to raising boiler pressure and temperature, adding feedwater heaters, and using reheat, there are other things design engineers can do to improve plant efficiency. A discussion of some of these techniques follows.

Improved Boiler Efficiency

The typical propulsion boiler found on ships built in the 1950s and early 1960s was a two-drum D-type fitted with an economizer. With a stack temperature of about 325°F and excess air levels of 15 percent, this results in a boiler efficiency of about 88.5 percent.

Two major things can be done to increase boiler efficiency: lowering the stack temperature and lowering the excess air level. The limitation on lowering the stack temperature is that sulfuric acid from the combustion process condenses and results in corrosion. Since most modern steam plants have high-pressure heaters fitted with feed temperatures of 400° to 450°F, air heaters rather than economizers are used to recover the heat from the stack gases. Air heaters, particularly the rotary regenerative type, can operate at stack temperatures of 240° to 260°F without significant corrosion problems.

Lower excess air levels are achieved by installing specially designed burners. These burners improve the atomization of the fuel and the mixing of fuel and air. The best designs can achieve excess air levels as low as 3 to 5 percent. The combination of a stack temperature of 240° to 260°F and excess air of 3 to 5 percent results in boiler efficiencies of 90 to 90.5 percent-a 1 1/2 to 2 percent improvement over older designs.

Attached Generators and Feed Pumps

The turbines that are used to drive the electric generators and feed pumps are less efficient than the main propulsion turbine. Smaller turbines are inherently less efficient than larger turbines. By use of power takeoffs from the reduction gear, during steady-state steaming the required electric power can be generated and the feed pump driven by the more efficient propulsion turbine. During maneuvering or in port, the attached units cannot be used and units driven by auxiliary turbines must be used.

Legend for figure 3-6, facing page

Shaft horsepower 17,500	Uptake temperature, deg. F 300
Fuel consumption, lb per hr 9,100	Sea water temperature, deg F 75
Fuel rate, lb per shp-hr 0.520	Generator load, kw 425
Main turbine water rate, lb per shp-hr 6.10	Distilling plant load, gpd 6,800
Turbogenerator water rate, lb per kwhr 11.0	Quarters heating system . . . . . not in use
Heating value of fuel, Btu per lb 18,500	Cargo refrigeration system . . . . . not in use

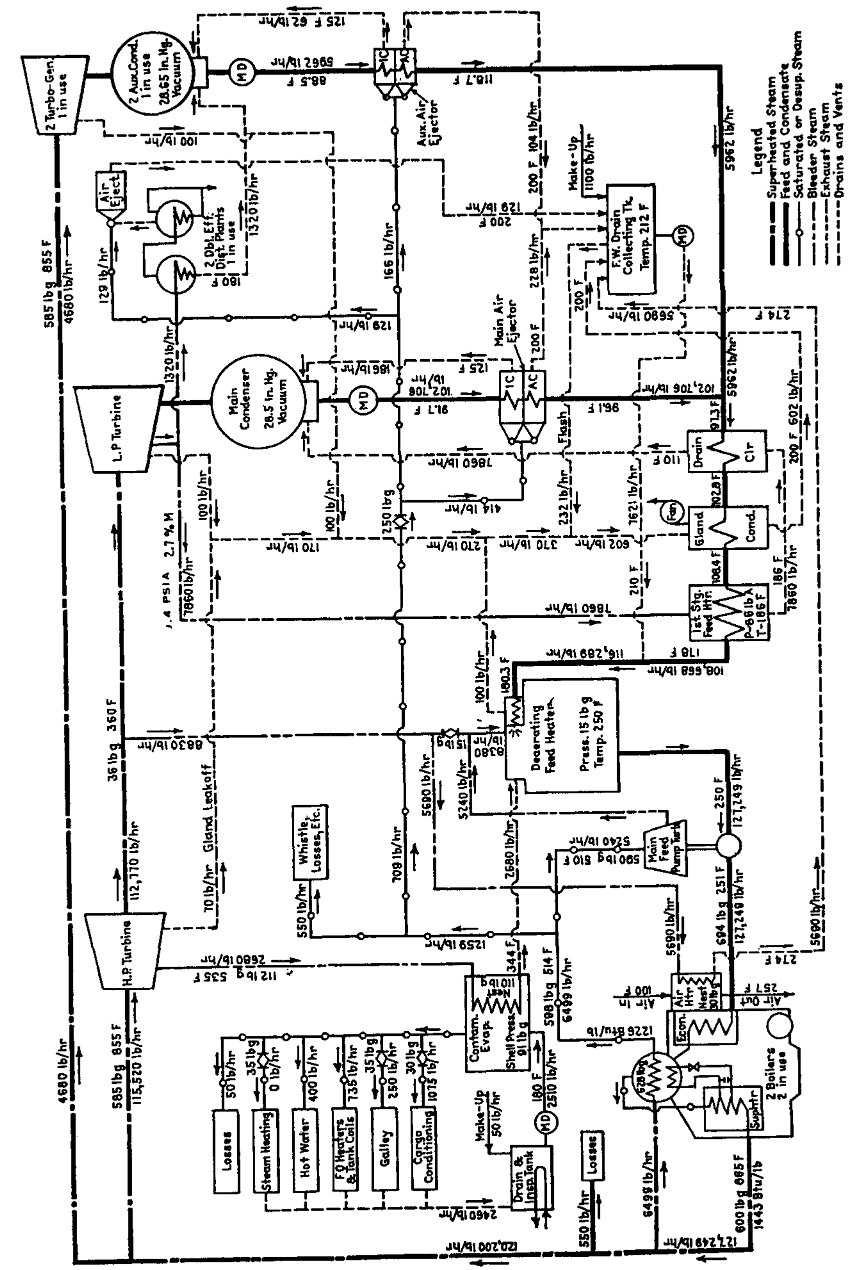

Figure 3-6. Heat balance diagram for Mariner Class C-4

Condensate-Cooled Evaporators and Lube Oil Coolers

The evaporators and lube oil coolers on steamships are typically cooled by seawater. This represents a rejection of potentially useful heat lost to the sea. If condensate from the steam cycle is used as the coolant, this heat would be recovered and used to help preheat the feedwater. This results in less bleed steam being required, and more power being developed for the same HP turbine inlet steam flow. The danger, of course, is possible contamination of the condensate with salt water or lube oil should a leak occur in the heat exchanger. Proper mechanical design and maintenance of the heat exchanger is vital.

Sample Steam Cycle Heat Balances

In this section, we will look at a sampling of heat balances for some typical marine steam power plant cycles. Heat balance diagrams (figs. 3-6 to 3-8) present the results of a mass and energy balance of a plant operating at a particular power level (most commonly full power) with the pressures, temperatures, flows, and enthalpies shown.

TWO-HEATER NONREHEAT CYCLE

Figure 3-6 is a heat balance diagram for a C-4 Mariner class vessel. This very successful class of dry cargo vessels built in the 1950s became the standard for most of the steamships built in the United States for years to come. The cycle is a simple cycle with two regenerative feedwater heaters (one LP heater and a deaerating heater). The turbine inlet steam conditions are 585 psig and 855°F. The boiler is a two-drum D-type boiler fitted with an economizer and a steam air heater. At the design power of 17,500 shp, the fuel rate is 0.52 lbmlhp-hr.

FIVE-HEATER NONREHEAT CYCLE

The cycle shown in figure 3-7 is typical of the plants installed in steamships in the late 1960s and early 1970s. The plant has five feedwater heaters: two LP heaters, a deaerating heater, and two HP heaters. Turbine inlet

Legend for figure 3-7, facing page

Superheated steam
Desuperheated steam
Low pressure and bleed steam
Feed and condensate
Drain
Gland leakoff and vent

Flow in lb/hr
P–pressure psia
F–temperature Fahrenheit
H–enthalpy steam Btu/lb
H–enthalpy water Btu/lb

Shaft output	20,000 shp
Fuel consumption	9,042 lb/hr
Fuel rate	0.4521 lb/shp-hr
Higher heating valve	18,500 Btu/lb
Boiler efficiency	90.2%
Temperature	75°F
Generator load	650 kW
Evaporator load	8080 gpd
Scoop injection	in service
Air conditioning	in service

Prepared in accordance with SNAME bulletin no. 3-11
Recommended Practices for Preparing Marine Steam Power Plant Heat Balances

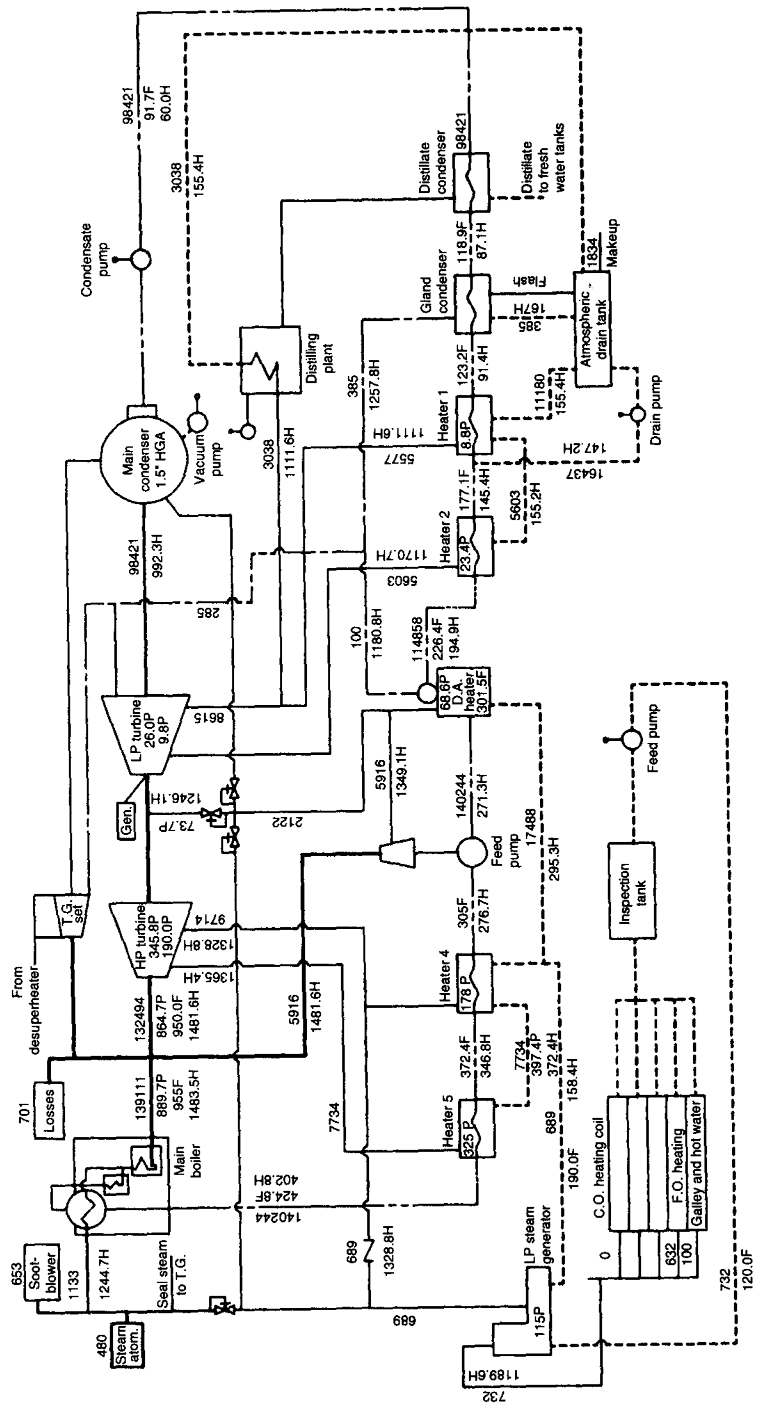

Figure 3-7. Heat balance diagram for five heater cycle. Courtesy Newport News Shipbuilding.

conditions are 850 psig and 950°F and boiler efficiency is 90.2 percent. A vacuum pump rather than an air ejector is used to remove noncondensables from the main condenser. An attached generator is used to produce electric power at sea and the distilling plant is cooled by condensate. At the design power of 20,000 shp, the fuel rate is 0.4521lbm/hp-hr. This represents a 13 percent improvement over the simple two-heater cycle.

REHEAT CYCLE

Figure 3-8 is a reheat cycle fitted with five feedwater heaters. Turbine inlet conditions are 1,500 psig and 1,000°F with reheat at 300 psia to 1,000°F. Like the five-heater cycle above, there is an attached generator and a condensate-cooled distilling plant. The feed pump is driven by an electric motor rather than an auxiliary steam turbine, using the electric power produced by the efficient main propulsion turbine. Boiler efficiency is 90.3 percent. At the rated power of 20,000 shp, the fuel rate is 0.411lIbm/hp-hr. This represents a 9 percent improvement over the five-heater cycle (fig. 3-7) and a 21 percent improvement over the two-heater cycle (fig. 3-6).

FLUID REGENERATIVE AIR HEATER CYCLE

It was mentioned above that plants with high-pressure feedwater heaters almost always employ either tubular or rotary regenerative air heaters to retain boiler efficiency. One disadvantage of these air heaters is that they are large devices and typically will not fit into an engine room not originally designed to accommodate one. One alternative for the operator wishing to retrofit high-pressure heaters to an existing steam plant is the use of a fluid regenerative air heater (FRAH) cycle.

The FRAH cycle (fig. 3-9) allows the economizer to serve as the final heat recovery device for the boiler. Feedwater from the deareator is sent to the economizer as in a cycle without high-pressure heaters. The feedwater from the economizer is now used to preheat the combustion air in the fluid regenerative air heater. This air heater replaces the steam air heater in the original boiler. The feedwater is now sent on to the high-pressure heaters for further regenerative heating.

Legend for figure 3-8, facing page

Superheated steam	———	Shaft output .... 20,000 shp
Desuperheated steam	———	Fuel consumption .... 8,222 lb/hr
Low pressure and bleed steam	———	Fuel rate .... 0.4111 lb/shp-hr
Feed and condensate	— — —	Higher heating valve .... 18,500 Btu/lb
Drain	- - - - - -	
Gland leakoff and vent	—— — ——	Boiler efficiency 90.3%
		Temperature .... 75°F
Flow in lb/hr		Generator load .... 950 kW
P–pressure psia		Evaporator load .... 6380 gpd
F–temperature Fahrenheit		Scoop injection .... in service
H–enthalpy steam Btu/lb		Air conditioning .... in service
H–enthalpy water Btu/lb		

Prepared in accordance with SNAME bulletin no. 3-11 (reheat supplement)
Recommended Practices for Preparing Marine Steam Power Plant Heat Balances

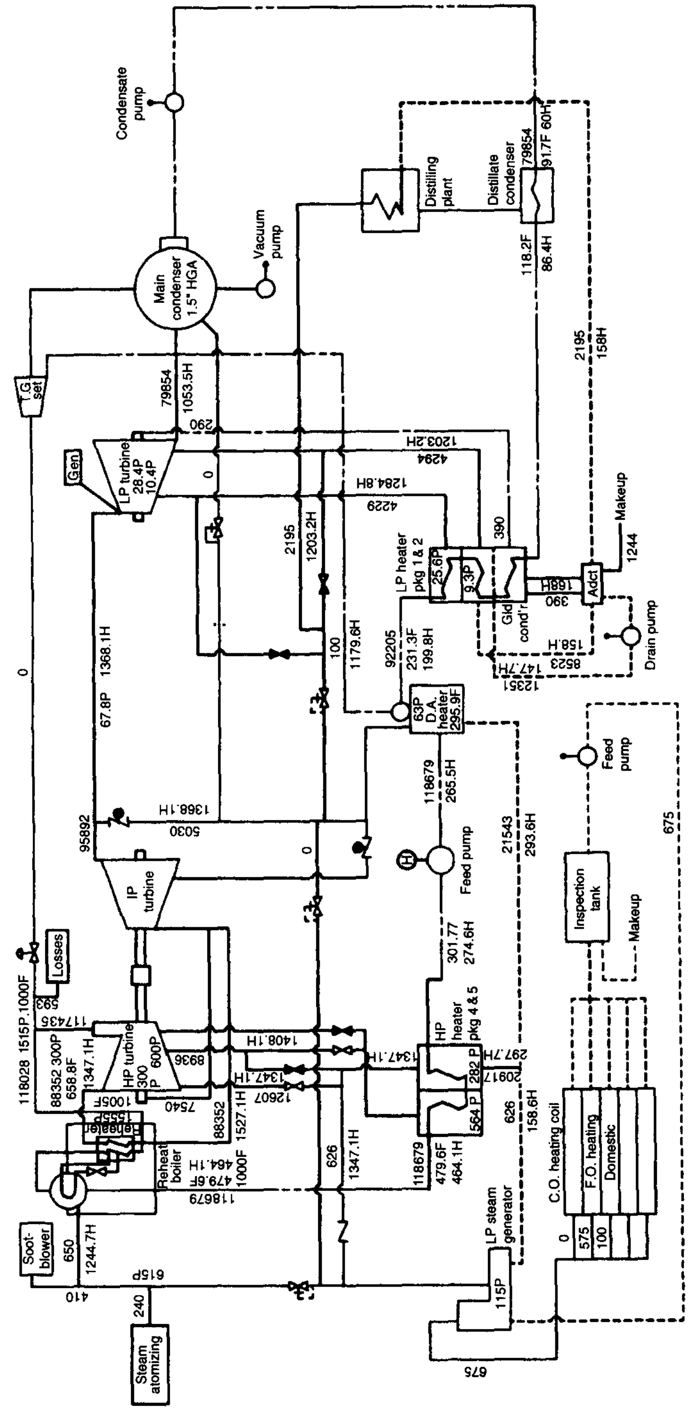

Figure 3-8. Heat balance diagram for reheat cycle. Courtesy Newport News Shipbuilding.

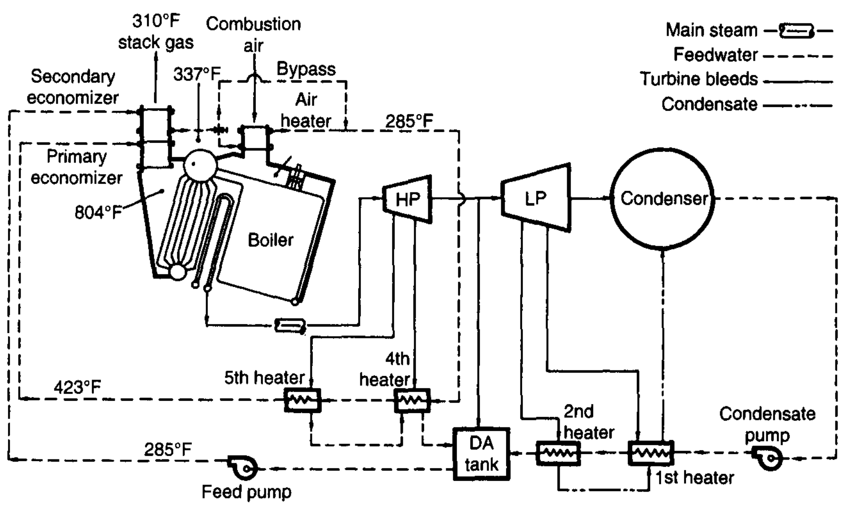

Figure 3-9. Fluid regenerative air heater cycle.
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CASCADED BLEEDS

One of the disadvantages of the steam power plant is that its efficiency falls off at part loads. There are three main reasons for this: (1) the reduction in boiler efficiency; (2) the reduction in steam turbine efficiency; and (3) the reduction in the temperatures of the feedwater leaving the heaters. The third results from the decrease in bleed pressures as the turbine power is reduced. As the bleed pressures fall off, the heater shell pressures are lowered, and lower heater outlet temperatures result.

If the power plant is to operate at part loads for any significant periods of time, consideration should be given to a cascaded bleed system. Figure 3-10 shows a cascaded bleed system for a two-heater cycle with an LP feedwater heater and a deaerating feedwater heater. As plant power is reduced and the bleed pressures fall off, the cascade control valves will open and supply the LP feedwater heater with steam from the crossover and supply the deaerator with steam from a high-pressure bleed. Cascade extractions may be used for any number or combination of high- and low-pressure heaters to retain their effectiveness over a wide range of power levels. A cascaded bleed system can be easily retrofitted to an existing plant.

Heat Balance Calculations

The preparation of a heat balance diagram is one of the first steps in the design of a steam power plant. The calculated operating characteristics and values of pressure, temperature, and flow rate become the basis of many of

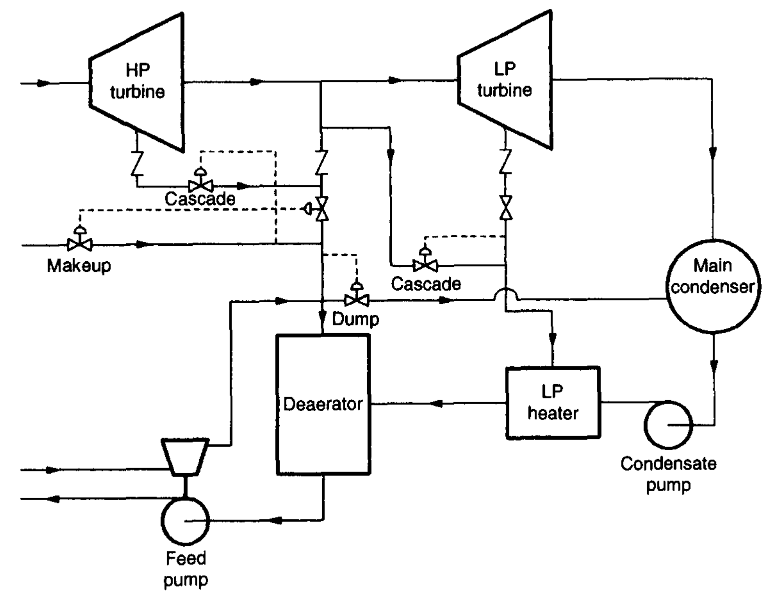

Figure 3-10. Cascade extraction system

the design tasks that follow, such as equipment selection and pipe sizing. The heat balance diagram also becomes a useful tool for the plant operating engineers, permitting them to compare observed conditions with those recommended by the plant designers.

The preparation of a heat balance diagram is not a trivial task. Most texts on thermodynamics contain a chapter on steam cycle analysis that demonstrates the basics of the process. The simplified idealized cycles presented in such introductory texts can be analyzed without too much trouble. However, once all the required pumps, fans, coolers, turbogenerators, evaporators, and other auxiliary equipment are included, even a relatively simple but realistic cycle becomes a challenge to analyze. T&R Publication 3-11, by the Society of Naval Architects and Marine Engineers, contains procedures and techniques for estimating such items as turbine efficiencies, electric load, freshwater usage, sootblower losses, atomizing steam flow, pressure losses, and air ejector flow.

A common technique for completing the heat balance is to use a trial-and-error approach. The flow leaving the condenser is assumed, and the bleed and other unknown flows are calculated using the first law of thermodynamics, i.e., conservation of mass and energy. After all flows are calculated, the power that would result from the assumed condenser flow is calculated and compared to the desired power. Further guesses of the condensate flow are made and the calculations repeated until the desired plant power is achieved. The last step is to calculate the fuel flow and the plant's all-purpose fuel rate.

A very important aspect of the steam power plant design process is the selection of the cycle configuration, including superheater pressure and temperature, number offeedwater heaters, reheat or nonreheat, and other items. The selection of the cycle for a particular application is based on both engineering and economic considerations. The economic considerations are a balance between the increased cost of a complex cycle versus the lqwer fuel consumption over the life of the plant. Figure 3-11 shows typical fuel rates for several common steam cycles over a range of rated powers. In general, the higher the ra.ted power of the plant, the more likely a complex cycle will be a logical selection.

Maintaining Steam Plant Efficiency

Once the steam power plant has been designed and built, it is up to the ship's engineer officers to operate the plant as efficiently as possible. A steam plant of 15,000 to 30,000 shp will burn several million dollars worth offuel per year. Savings of even a fraction of 1 percent can translate

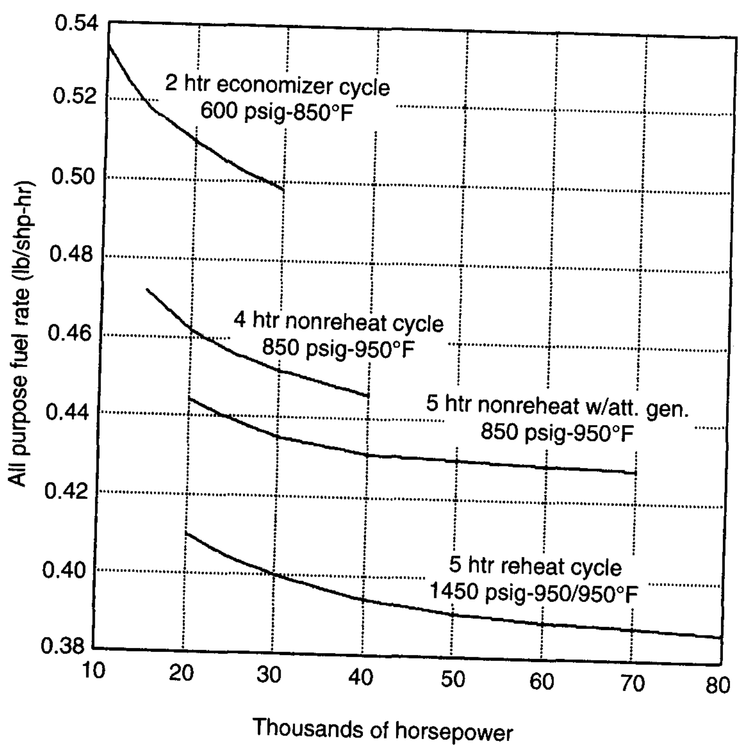

Figure 3-11. Typical steam power plant fuel rates

to significant annual dollar savings. The challenge facing the engineer officers is to get every bit of efficiency out ofthe plant that the plant designers put in. This task is sometimes referred to as "tuning," as it implies a certain amount of adjustment of plant operating parameters to improve plant performance.

The first step in any program of restoring and maintaining plant efficiency is ensuring that the plant gauges and instrumentation are accurate and reliable. Important decisions are going to be made based on the information provided by the plant's gauges and instruments. If the instrumentation is giving inaccurate data, the operator will not be able to make good plant tuning decisions, and reductions in fuel consumption and extended equipment life cannot be achieved. The accuracy of the key plant instrumentation should be checked every six to twelve months. A deadweight tester or a set of test gauges can be used for calibration of the pressure gauges in the plant. A precision digital temperature readout with a thermocouple or RTD sensor can be used for checking the thermometers. A high-quality digital multimeter (DMM) and a precision decade resistance box will aid in the calibration and troubleshooting of the remote electronic signal conditioning equipment.

Any investigation of steam power plant efficiency must begin at the boiler. This is where all the fuel is consumed and where the greatest potential for losses occurs. There are two main areas where losses occur due to improper boiler operation and maintenance: (1) failure to maintain design superheater outlet temperature and pressure; and (2) reduction in boiler efficiency due to high stack temperature and/or high levels of excess air.

Failure to maintain superheater pressure or temperature affects the efficiency ofthe main propulsion turbines. The lower energy steam admitted to the first stage of the turbine results in more steam flow required to maintain the same power and rpm. More steam flow from the boiler means more fuel consumed. Figure 3-12 shows the effect on plant efficiency from variation in superheater pressure. Figure 3-13 shows the effect of variation in superheater temperature. Note that a superheater temperature 25°F below design results in about a 0.5 percent decrease in efficiency, while a superheater pressure 25 psig below design results in a bit less than a 0.3 percent decrease.

It is important to note that these variations can occur from either operator inattention or instrument error. It is also important to note that while pressures and temperatures higher than design will result in increased efficiency, it comes at the expense of reduced superheater tube and turbine life. These parameters should be maintained at the design values, not above or below them, by adjusting the combustion control system and superheat temperature controller.

The reduction in boiler efficiency caused by high stack temperature (see fig. 3-14) is typically caused by fouled boiler tube surfaces. Sootblowers must be properly maintained and the boilers washed regularly. High level of excess air is the most common cause of reduced boiler efficiency. The common practice of clearing up the stack by merely increasing the combustion air flow results in high levels of excess air. An excess air level of only 15 percent above design, e.g., 30 percent versus design of 15 percent, results in a decrease in boiler efficiency of about 1 percent (see fig. 3-15). The only way of knowing the excess air level is to measure it using an oxygen and/or carbon dioxide stack gas analyzer. If the excess air must be maintained above design to avoid excessive smoking, the causes must be determined. Common causes are improper oil viscosity, worn atomizer tips, incorrect atomizer withdrawal depth, damaged or improperly repaired burner refractory throat tile, and low combustion air temperature. (See chapter 5 for more information on measuring boiler efficiency and oil burner adjustments.)

Mter the boiler, the next place to look for potential fuel savings is the plant's major heat exchangers-the main condenser and the feedwater heaters. Common causes ofloss of efficiency are condenser fouling, excessive condenser air leakage, and malfunctioning feedwater heater drain regulators. The main condenser is typically designed to maintain a 28.5-in Hg vacuum at full power with 75°F seawater. The most important parameter to watch on the feedwater heaters is the temperature of the feedwater leaving the heater. Any decrease in this temperature at full power compared to the value shown on the cycle heat balance diagram should be in-

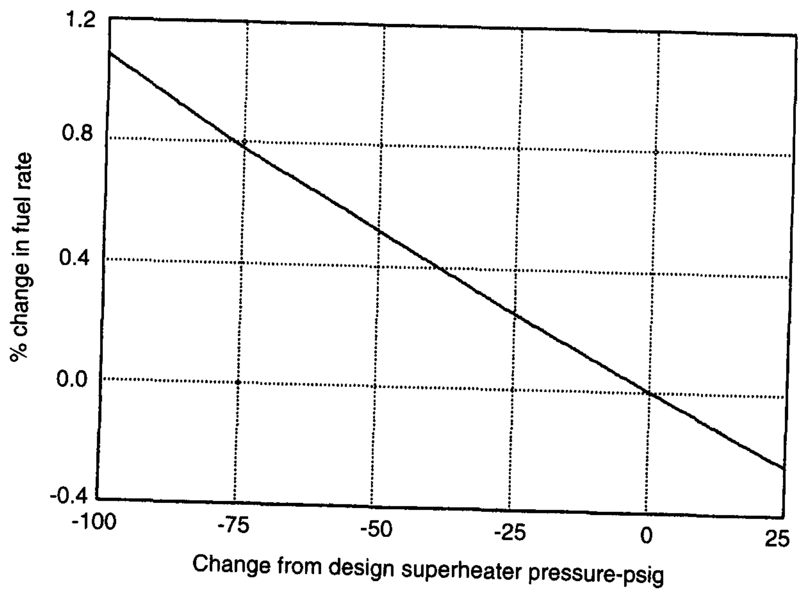

Figure 3-12. Effect of variation of superheater outlet pressure on fuel rate

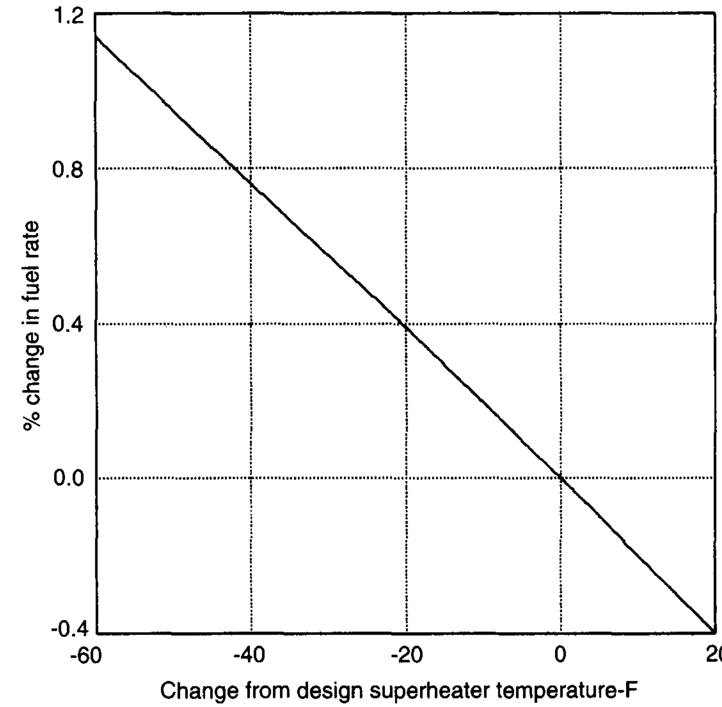

Figure 3-13. Effect of variation of superheater outlet temperature on fuel rate

vestigated. (See chapter 10 for further information on how to monitor the performance of these heat exchangers.)

Operating power plants at part-load presents some special problems. In the past, steam plants were optimized for peak efficiency at full power, with little thought given to maintaining efficiency at reduced powers. A few basic principles for improving part-load efficiency apply:

- Avoid shifting from bleed steam to live steam for as long as possible.
- Operate steam turbines on as few nozzles as possible.
- Operate pumps and fans on the slowest speed possible.

With regard to the first principle, the major use of bleed steam is for feedwater heating. As the plant power comes down, the bleed pressures drop and along with them the temperatures of the feedwater leaving the heaters. One technique is to lower the setting on the auxiliary exhaust live steam makeup valve, allowing the deaerator pressure to avoid use of "live" steam and "float" on the lower crossover pressure. If a cascaded bleed system is fitted (fig. 3-10), the heater will be shifted to higher-pressure bleeds, maintaining shell pressures and feed outlet temperatures at reduced power. With regard to the second and third principles, close hand nozzle

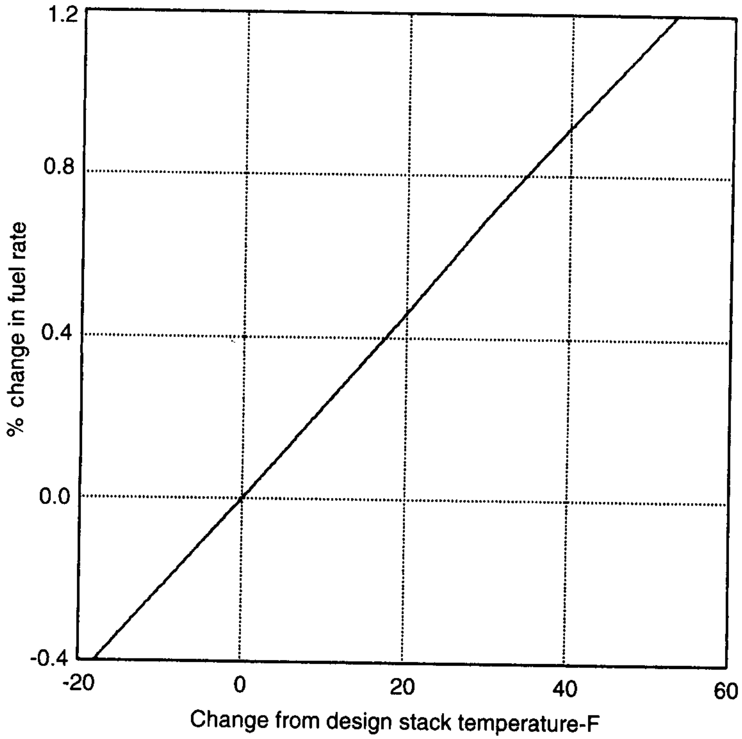

Figure 3-14. Effect of variation of boiler stack temperature on fuel rate

valves on all turbines fitted with them as soon as possible. Large consumers of electric power like forced draft fans should always be operated on the lowest speed possible.

Most marine boilers are not able to maintain design superheater temperature below about 60 to 70 percent of rated load. Figure 3-16 illustrates a typical attemperator control curve for a marine boiler. On ships fitted with two boilers, one thing to consider is securing one boiler at low powers. This will increase the load back into the region where design superheater temperature can be maintained. Obviously such considerations as plant reliability with only one boiler operating come into play, but it must be remembered that a number of one-boiler ships have been built and successfully operated for many years.

When operating in port with the main engines secured, it is sometimes necessary to dump auxiliary exhaust steam to the main condenser. The exhaust from the main feed pump is greater than that required by the deaerator and the boiler steam air heaters. An electrically driven port feed pump is an excellent solution to this problem and can result in significant reductions in port fuel consumption.

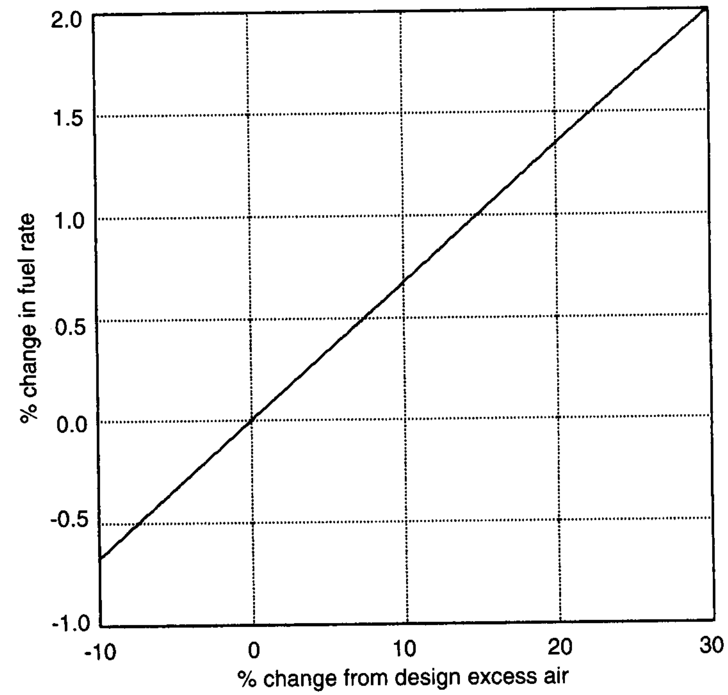

Figure 3-15. Effect of variation of boiler excess air on fuel rate

STEAM POWER PLANT SYSTEMS

The steam power plant systems described in the following sections are based on those found on a class of 30,000-shp roll-on/roll-off (Ro/Ro) vessels built by Sun Shipbuilding. The plant is a fairly simple two-heater cycle with steam conditions of845 psig and 905°F. The systems described should be similar, but certainly not identical, to those found on most contemporary steam vessels. Figure 3-17 shows the symbols that are used in the system diagrams.

Main Steam System

The main steam system shown in figure 3-18 delivers superheated steam produced in the two main boilers to the main propulsion turbines, the two turbogenerators, the two main feed pumps, and the auxiliary steam system via the desuperheaters. The system is arranged so either or both boilers can supply steam to the system. The outlet from each boiler is fitted with a stop-check and a stop valve, each having integral bypass valves for warm-up purposes. The stop-check valve prevents steam from backing up into a nonoperating boiler.

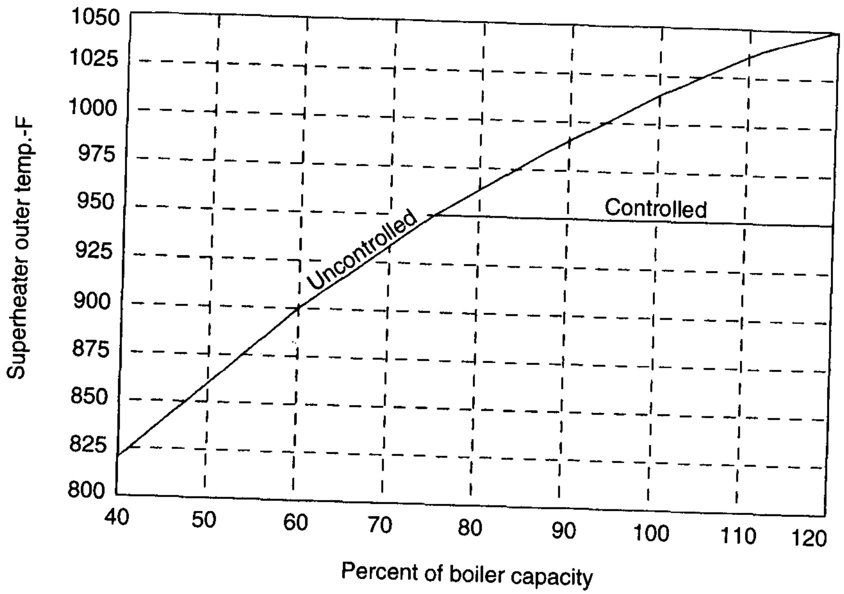

Figure 3-16. Control desuperheater temperature control curve

Lines from the boiler cross-connection supply steam to the propulsion turbines, the turbogenerators, and the feed pumps. Stop valves for isolation purposes are fitted. Pneumatically operated valves are installed on the feed pumps to permit remote startup at the central operating console (CaS). An extensive array of local and remote pressure and temperature instruments are installed, permitting monitoring of conditions of the steam out of each boiler and in the supply branches at the actual locations and at the cas. Additional safety devices include a high superheater temperature alarm for each boiler and safety valves for overpressure protection-two direct-acting valves on each boiler drum and a pilot-operated valve on each superheater.

Auxiliary Steam System

This system shown in figure 3-19 supplies desuperheated steam at a variety of reduced pressures to various services requiring auxiliary steam for operation. These services include the following:

- sootblowers at 545 psig (see detail B)
- fuel oil burner atomizing steam at 150 psig (see detail A)
- steam to contaminated evaporator at 225 psig
- emergency supply to contaminated steam system at 120 psig
- makeup to auxiliary exhaust steam system at 33 psig

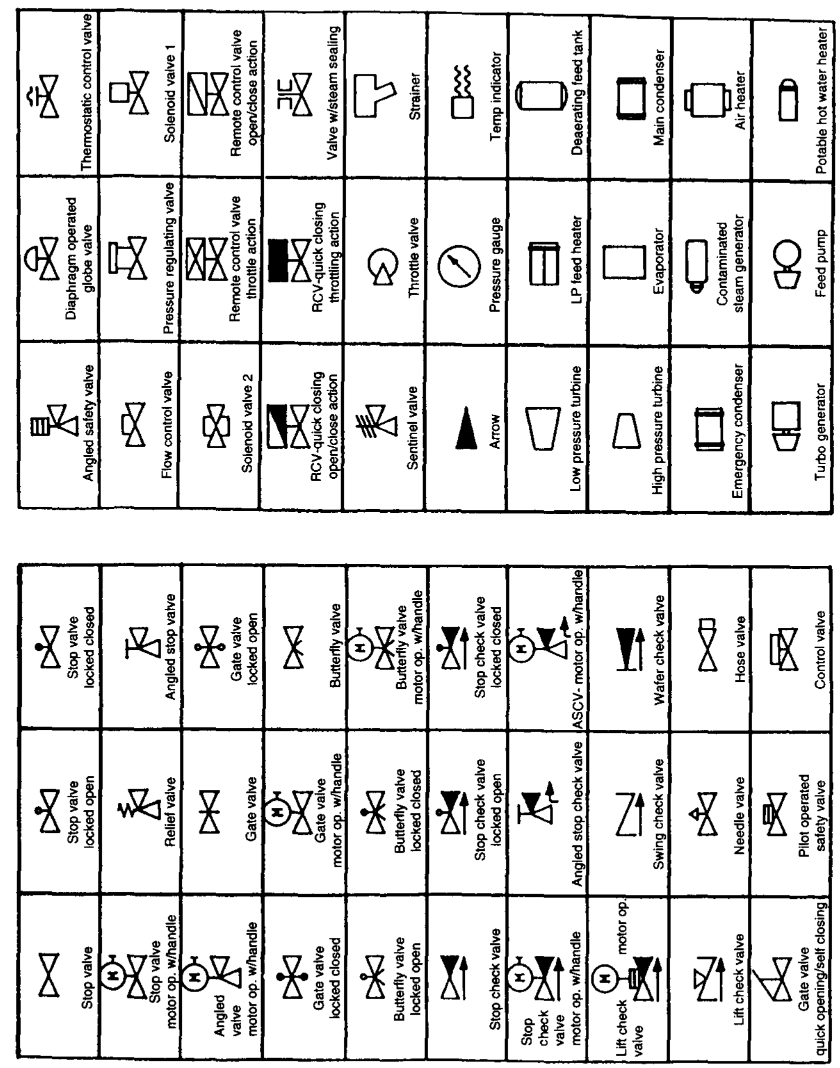

Figure 3-17. Symbol glossary. Courtesy Seaworthy Systems, Inc.

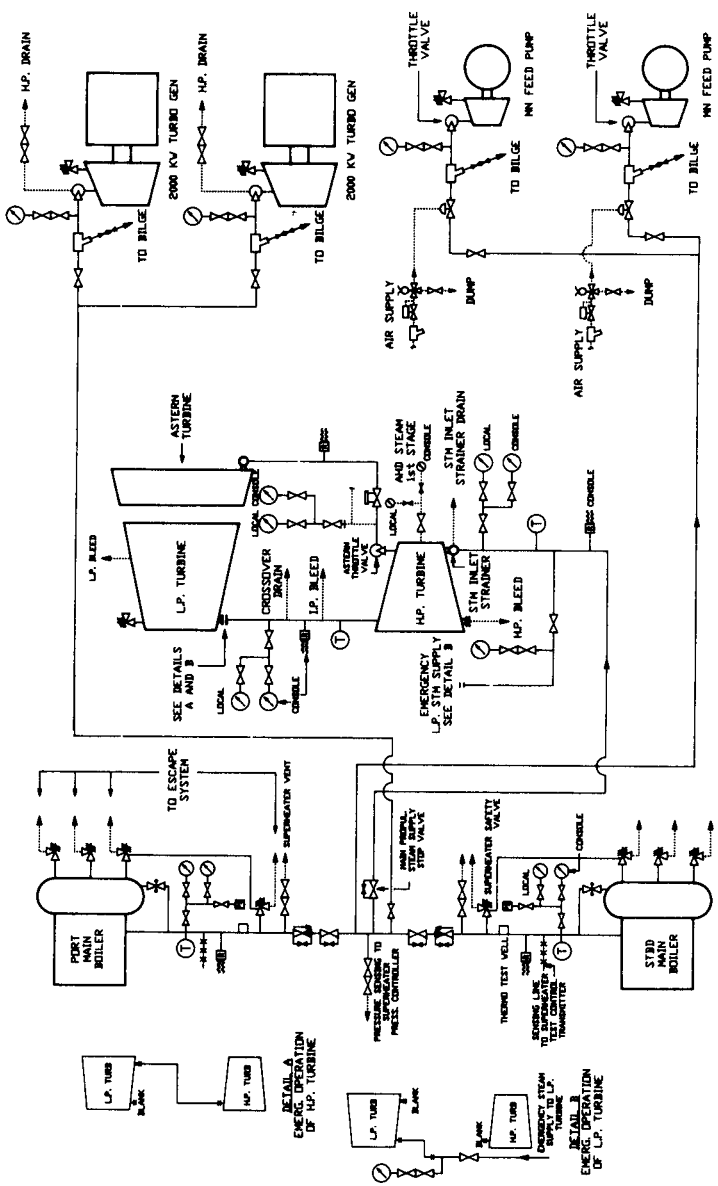

Figure 3-18. Main steam system.
Courtesy Seaworthy Systems, Inc.

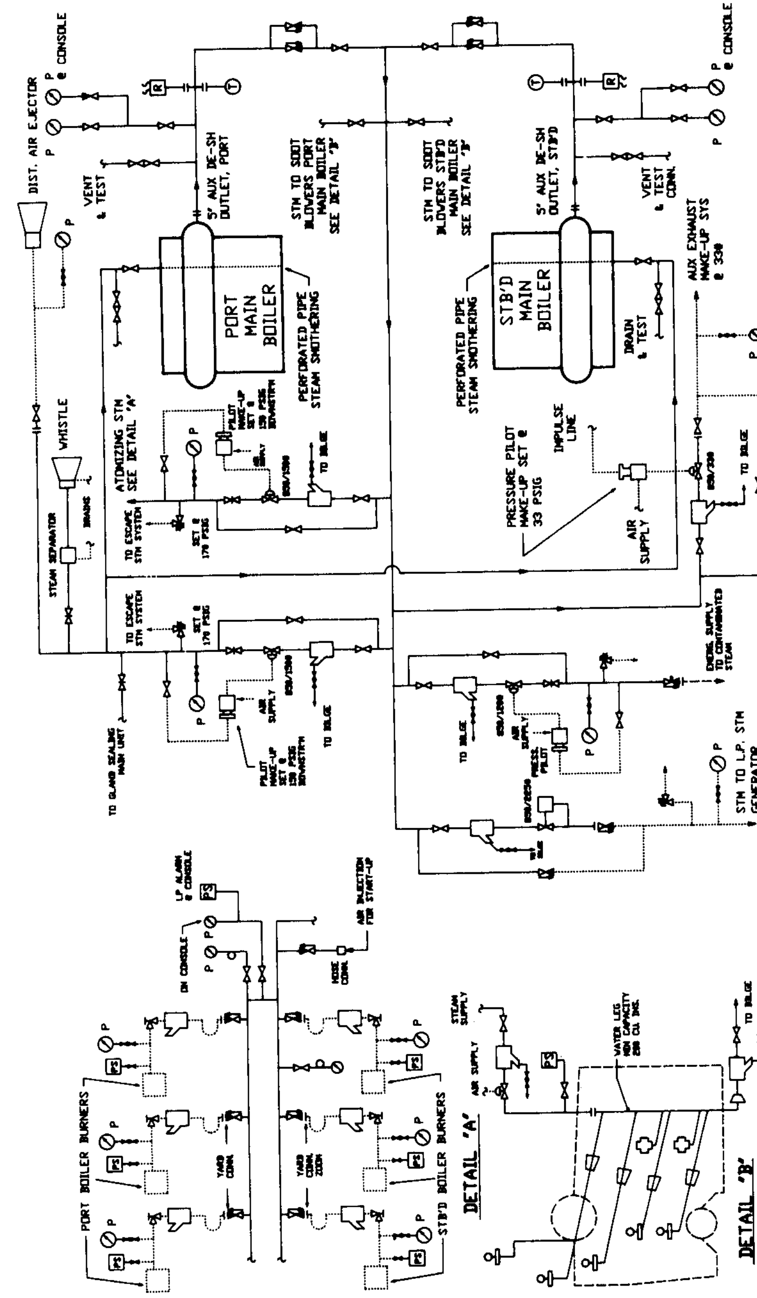

Figure 3-19. Auxiliary steam system.
Courtesy Seaworthy Systems, Inc.

- 150 psig steam line-service to main engine gland sealing, turbo-generator gland sealing, distilling plant air ejector, boiler steam smothering, and steam whistle supply

Desuperheated steam is supplied to the system from either or both boilers from an internal desuperheater located in each boiler steam drum. Double-valve protection against steam leakage into a secured boiler is provided by a stop-check valve and a stop valve in each boiler supply line. The boiler desuperheated steam pressure of 845 psig is reduced to that required for each service by a directly actuated pressure reducing valve or by a diaphragm control valve and pressure pilot system. Each reducing station is provided with a stop valve on the inlet and outlet, an inlet strainer, and a bypass valve. Reliefvalves are also fitted downstream of each reducing station to protect the system from overpressure.

Feed and Condensate Systems

The condensate system (fig. 3-20) serves to transfer condensate from the main condenser hotwell to the deaerating feedwater heater via the gland leak-off condenser and first-stage feed heater. Two main condensate pumps are installed. Each is sized to handle the flow under all normal operating conditions. Condenser hotwelllevel is maintained automatically by a diaphragm control valve and pneumatic level pilot which recirculates condensate back to the hotwell. The addition of makeup feed to the system and the spilling of excess feed from the system is controlled by automatically monitoring the deaerator level. A pair of diaphragm control valves and pneumatic level pilots add water to the system from the distilled water tank if the deaerator level is low and spill condensate to the tank if the level is high. An emergency condenser and condensate pump are provided to permit operation of the turbogenerators whenever the main condenser is inoperative.

The feedwater system (fig. 3-21) transfers heated and deaerated feedwater from the deaerating feed heater to the boiler steam drums via the boiler economizers. Two turbine-driven main feed pumps are fitted, each with sufficient capacity to handle the flow under all normal conditions. The pumps can be started and stopped remotely from the COSoA constant pressure governor varies the turbine speed to maintain the feed pump discharge pressure constant. A recirculation control valve is installed to avoid overheating when the feed pump is operating at low load. A boiler test pump is also provided for hydrostatically testing the boilers.

Auxiliary Exhaust and Extraction Systems

The auxiliary exhaust and extraction systems (fig. 3-22) deliver steam bled from the turbine extractions to the feedwater heaters and other heat exchangers. There are three turbine extractions: a high-pressure extraction from the HP turbine, an intermediate-pressure (IP) extraction from the

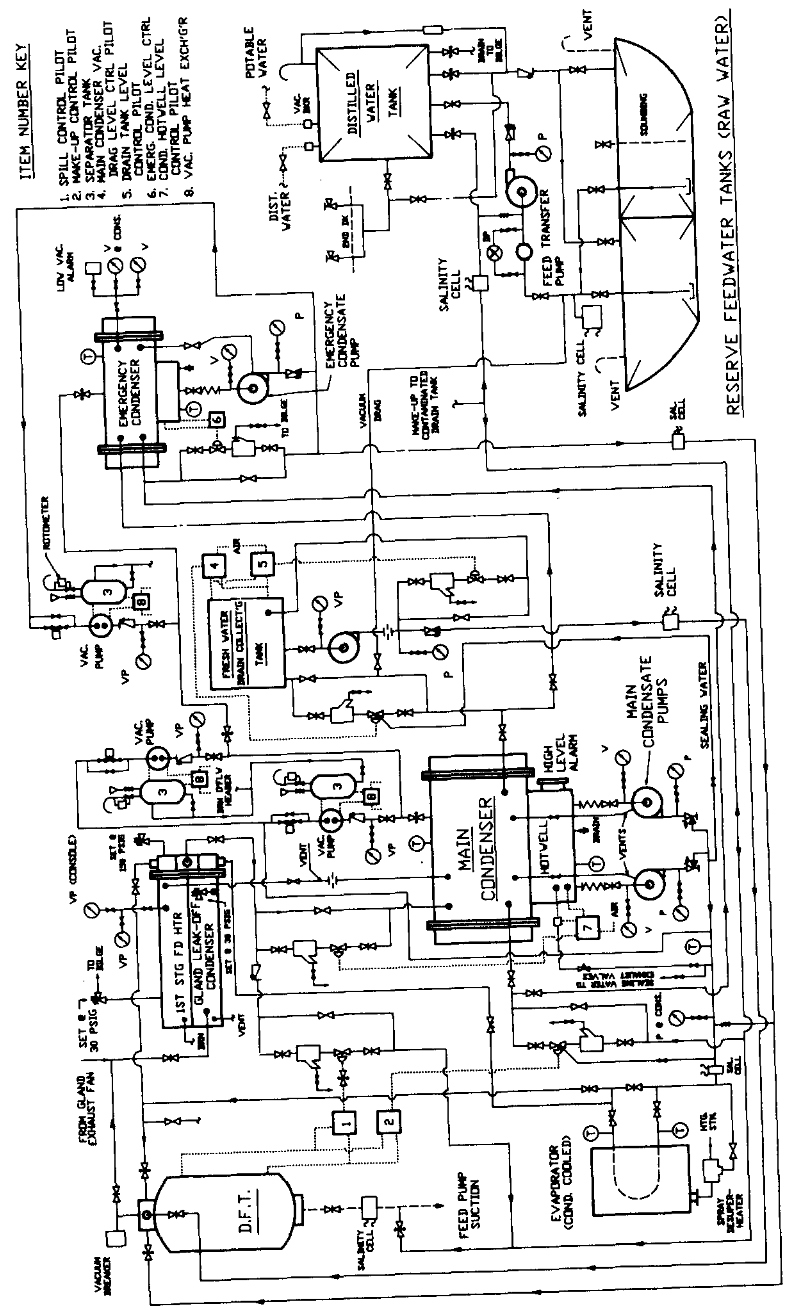

Figure 3-20. Condensate system.
Courtesy Seaworthy Systems, Inc.

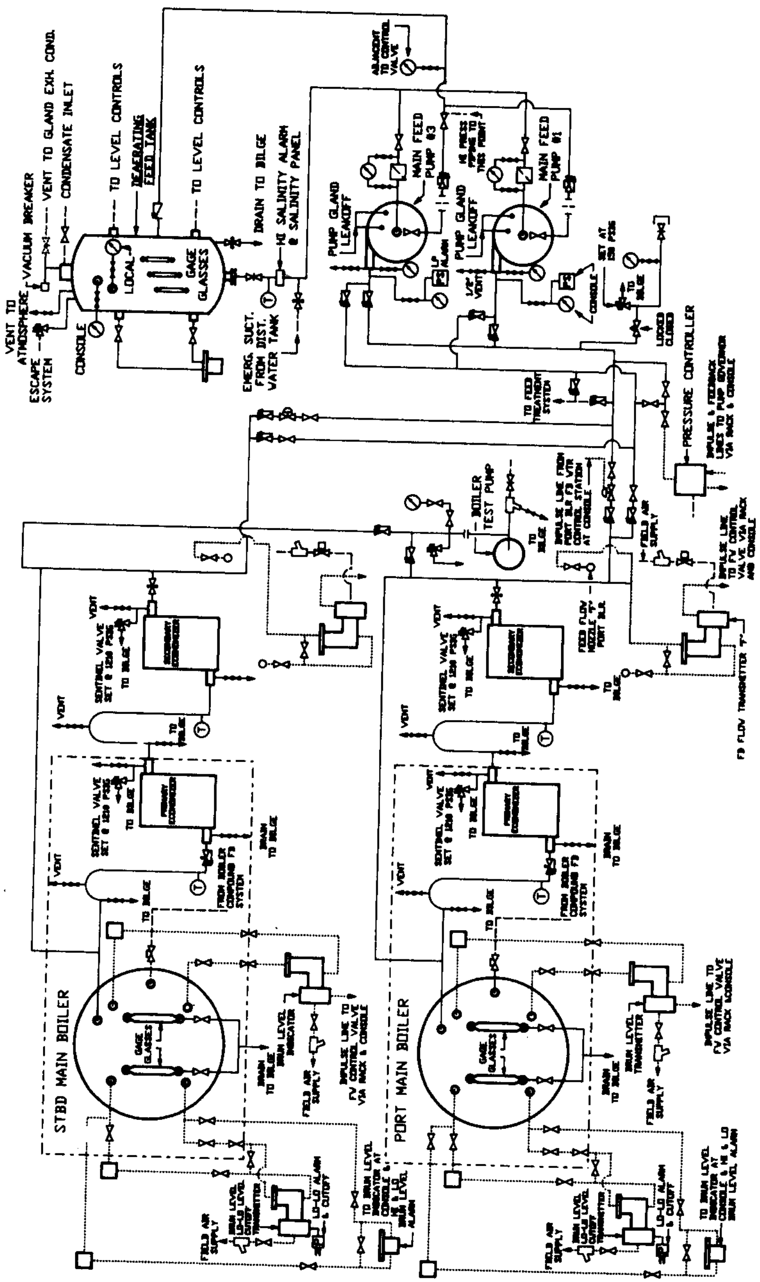

Figure 3-21. Feedwater system.
Courtesy Seaworthy Systems, Inc.

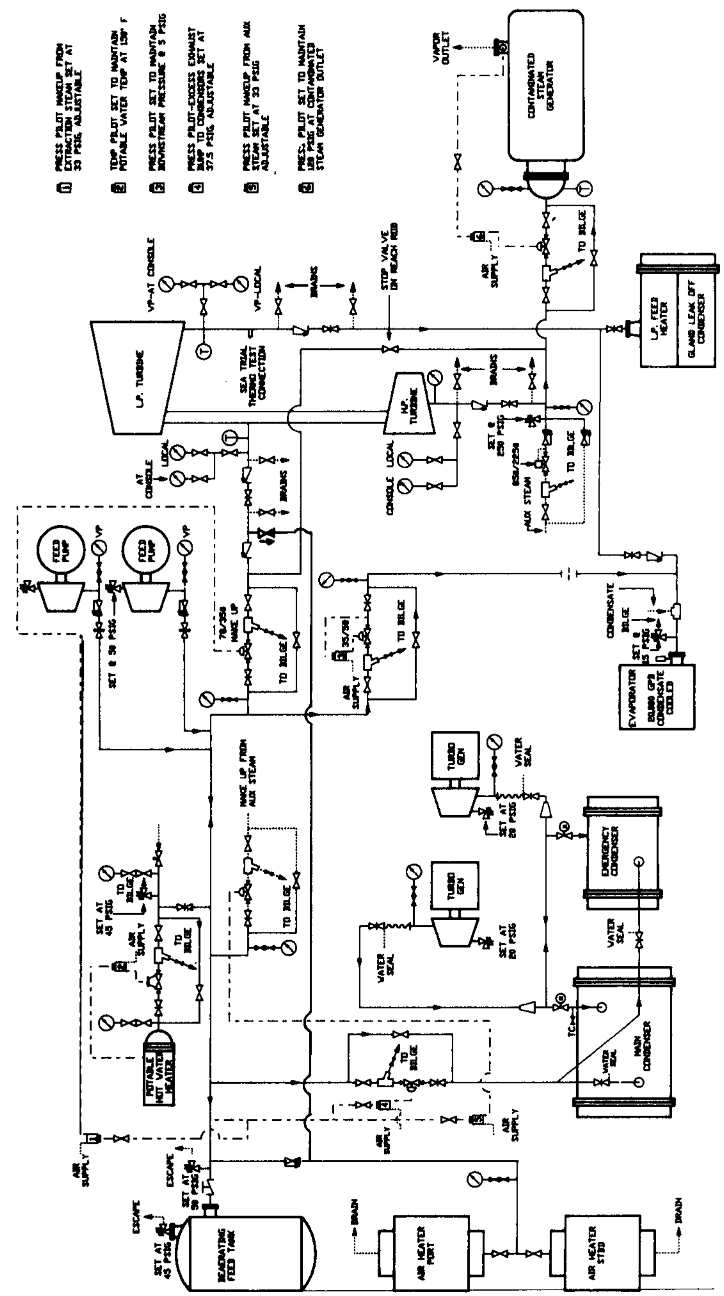

Figure 3-22. Auxiliary exhaust and extraction system.
Courtesy Seaworthy Systems, Inc.

crossover between the HP and LP turbines, and a low-pressure extraction from the LP turbine. A check valve is installed at each extraction to prevent steam from backing up into the turbine.

The main function of the auxiliary exhaust system is to supply steam to the deaerating feed heater. The system also supplies the boiler steam air heaters and the potable hot water heater and serves as a backup source for the evaporator. The source of the steam to the system varies depending on the operating condition of the plant. The feed pump turbines exhaust to the system. Under most conditions steam in addition to the feed pump turbine exhaust is needed to maintain system pressure. This additional steam can come from the IP extraction, the HP extraction, or the auxiliary steam system. Under some very low load or in-port conditions, the steam demands may be less than the steam being exhausted from the feed pump turbine and the excess steam must be dumped to the main condenser.

Pneumatic pilots and diaphragm control valves are installed on the IP bleed supply, the auxiliary steam supply, and the excess exhaust dump lines. It is essential that these valves be adjusted properly to ensure efficient system operation. Two basic principles apply: only one of these three control valves should be open at any one time; and extraction steam should be used to supply the system as much as possible. This is arranged by proper adjustment of the control valve pilot set points. The control pilot on the IP bleed supply is set for 35 psig. The pilot on the auxiliary steam supply is set at 33 psig. As the IP extraction pressure falls below 35 psig, the system supply will automatically shift from bleed steam to live steam. The IP extraction control valve will be wide open, but the extraction check valve prevents steam from backing from the system into the turbine. The extraction dump valve pilot is set at 37.5 psig. If the system pressure rises to that value, the dump valve will open, preventing further pressure rise. At this pressure, the pilots will close the IP extraction and auxiliary steam supply valves. The crossover between the HP extraction and the auxiliary exhaust system is a manual stop valve with a reach rod. It is opened when the crossover pressure is below 35 psig but the HP extraction is above, avoiding the use of live steam to maintain system pressure.

The HP extraction supplies steam to the contaminated evaporator. A diaphragm control valve and pneumatic pilot regulate the extraction steam flow to maintain 120 psig in the contaminated steam system. The HP extraction can also supply steam to the auxiliary exhaust system at low power when the IP extraction pressure is insufficient, e.g., a cascaded bleed.

The LP extraction supplies steam to the LP feed heater and to the distilling plant during normal operation at sea. If the LP extraction steam pressure is too low to operate the distilling plant, auxiliary exhaust steam via a 5-psig reducing station can be used.

Contaminated Steam and Drains System

The contaminated steam and drains system (fig. 3-23) supplies 120-psig steam to the fuel oil heaters, for fuel oil tank heating, for lube oil heating, and for other miscellaneous services. The drains from these services are collected in the contaminated drain inspection tank and then returned to the contaminated steam generator. This creates a closed system isolated from the steam cycle, protecting the main cycle from possible contamination.

The contaminated evaporator is essentially a small boiler, heated by high-pressure steam rather than the combustion of fuel. Under normal at-sea conditions, the heating steam for the contaminated evaporator is from the HP extraction. Under low power and in-port conditions, heating steam is supplied from the auxiliary steam system via a 850/225-psig reducing station.

Figures 3-24 through 3-27 show the arrangement of the major machinery in the engine room of the 30,000-shp Ro/Ro vessel whose systems are described above.

STEAM POWER PLANT OPERATING PROCEDURES

The following procedures are based on the 30,000-shp Ro/Ro vessels described in the previous section. They are representative of those that would be followed on a typical marine steam power plant. Different equipment, controls, automation, and systems on an individual vessel will result in modifications and changes from the procedures presented below. Consult the ship's manuals for further information and specific procedures to be followed on a particular vessel.

Preparing Plant for Departure

It is assumed that the plant is operating with both boilers in operation and vacuum on the main engine.

1. Insert standby burners in boilers, open steam atomization and fuel oil valves, and test burners.
2. Put forced draft fans on high speed.
3. One hour prior to sailing, test engine order telegraph and phones between engine room and bridge, and synchronize engine room and bridge clocks.
4. Test steering gear.
5. Disengage engine turning gear.
6. Open bypass around main steam supply valves. Notify bridge that engines will be rolled. Gradually open throttles, rolling engine slowly. DO NOT exceed 5 rpm on the shaft.
7. At "Stand By Engines," open main steam supply valves. As required, light off a second burner in each boiler, increase main feed pump pressure, start second fuel oil and main circulating pumps, and open lube oil cooler seawater supply valve.

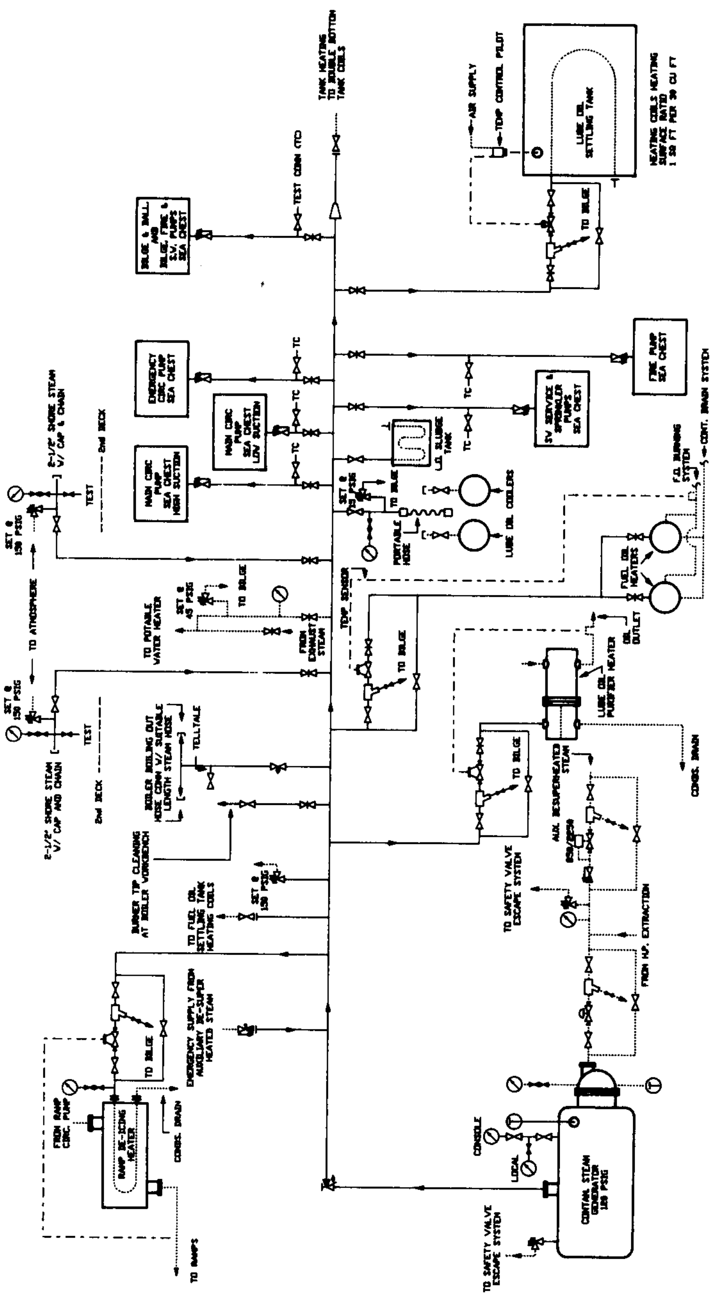

Figure 3-23. Contaminated steam and drains system.
Courtesy Seaworthy Systems, Inc.

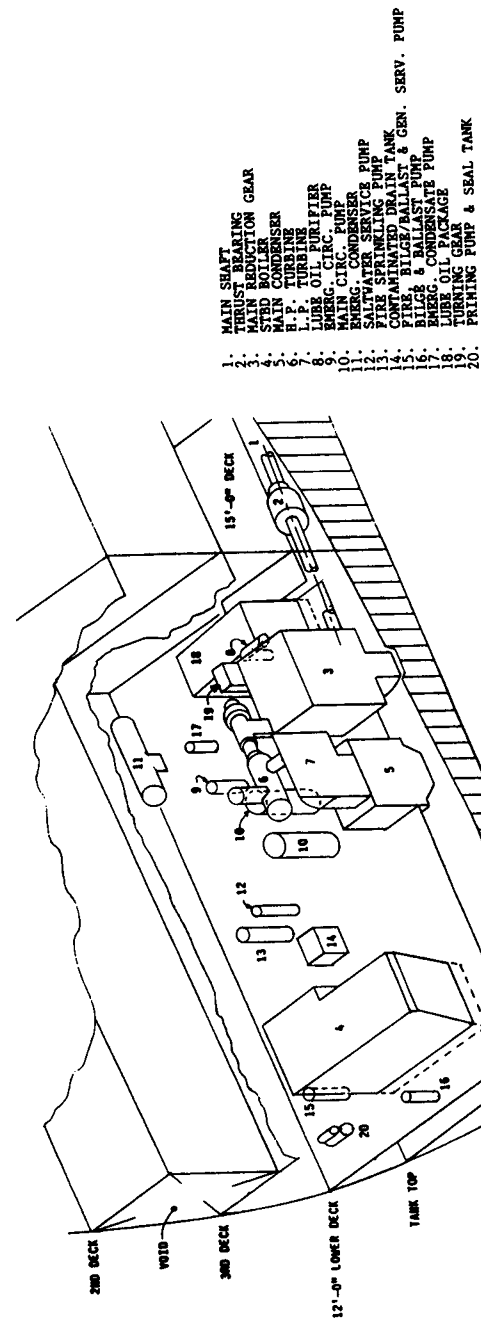

Figure 3-24. Starboard, lower deck, engine room.
Courtesy Seaworthy Systems, Inc.

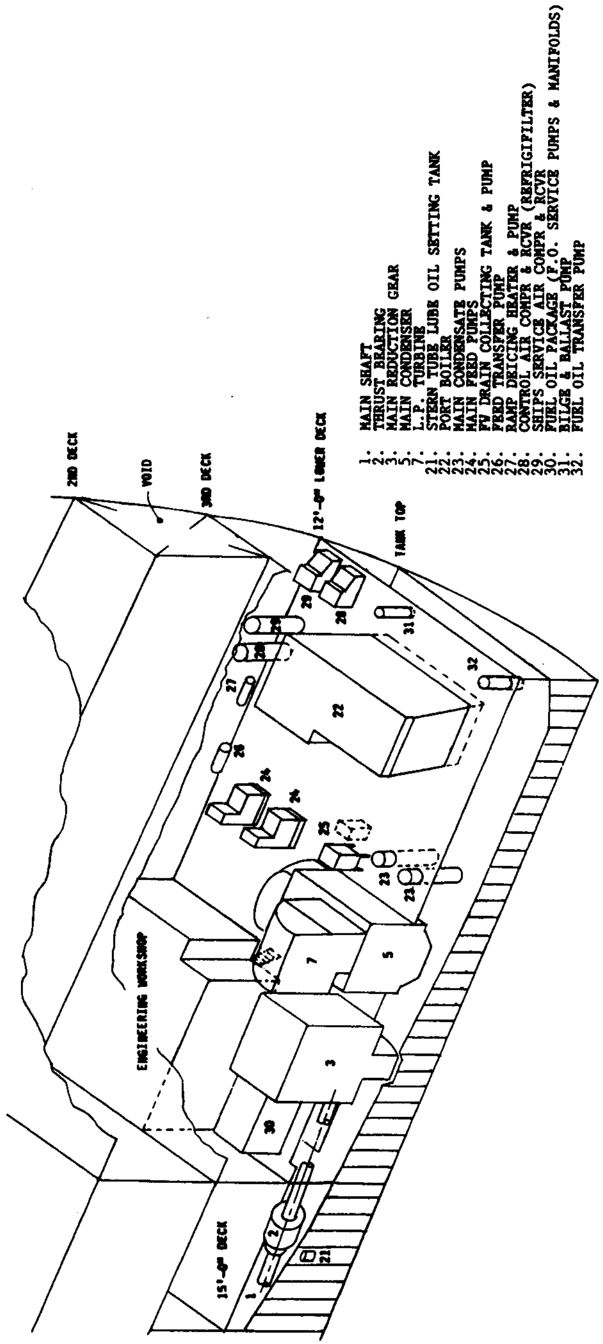

Figure 3-25. Port, lower deck, engine room. Courtesy Seaworthy Systems, Inc.

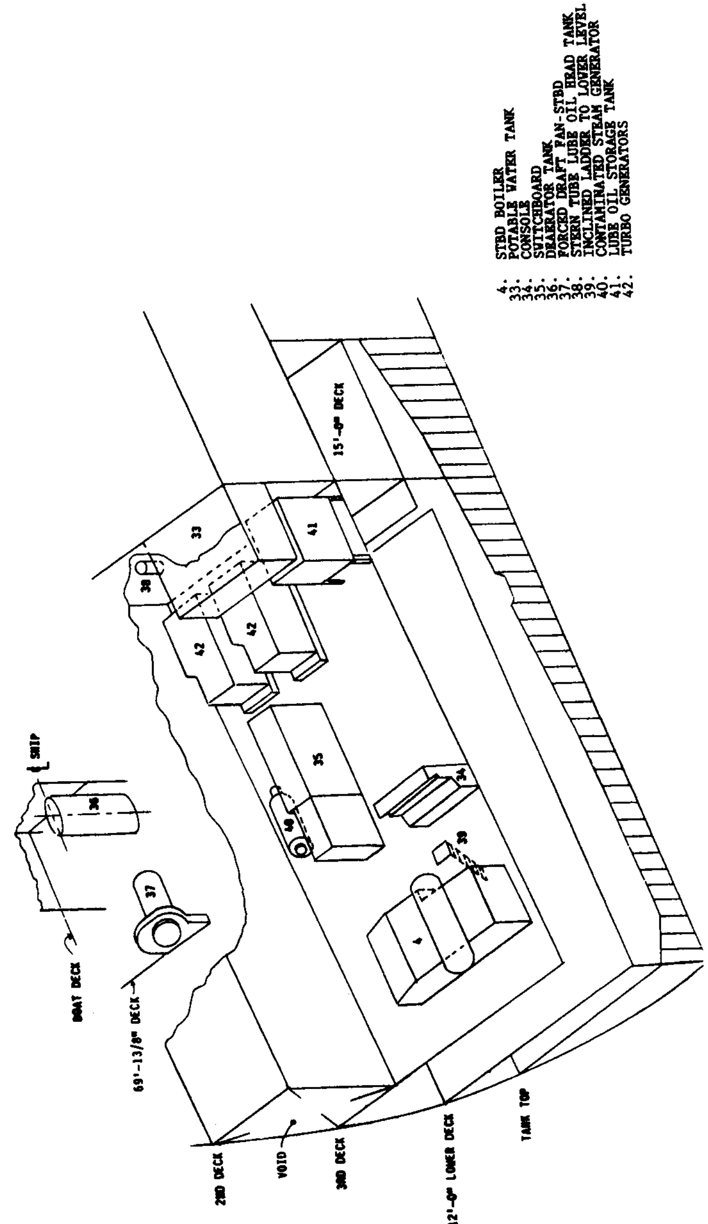

Figure 3-26. Starboard, third deck, engine room. Courtesy Seaworthy Systems, Inc.

Departing from Port

It is assumed that the plant has been set up as described above, "Preparing Plant for Departure." After maneuvering, when the departure order and sea-speed rpm are received from the bridge, the following procedures should be followed:

1. Gradually begin opening the ahead throttle, slowly increasing rpm as plant is set up for sea-steaming as described below.
2. Light off third burner in each boiler (as required).
3. Adjust main feed pump discharge pressure and close feed recirculating valve. Note: Carefillly monitor boiler drum levels during this period. Proper feed pump pressure can be set by checking feedwater regulator stroke.
4. Open turbine extraction valves and shift over appropriate equipment from live steam to bleed steam.
5. Blow tubes in boiler after plant stabilizes at sea-speed.
6. Start distilling plant.
7. Pump cargo hold and engine room bilges, using oily-water separator as appropriate.

At-Sea Steaming

During routine at-sea steaming, each watch engineer is responsible for maintaining an alert watch. The watch should be relieved properly-at the operating platform after all operating conditions have been determined. It is good practice to complete a round *prior* to relieving the watch. Rounds should be made regularly during the watch (usually at least hourly) and minor plant deficiencies such as small leaks corrected as time permits. The engineer and QMED (qualified member of the engine department) should alternate duties so that the operating console is attended at all times. The logbook should record actual observed conditions, not conditions as they should be or were on the previous watch.

Each watch engineer is typically assigned some additional duties to be completed during the watch. The second assistant is commonly responsible for blowing tubes, testing the boiler water, and transferring fuel oil during the watch. The other watch engineers are assigned duties such as starting and securing the lube oil purifier, shifting over turbine-generators, and operating the distilling plant.

Preparing Plant for Arrival

Within twelve hours of scheduled arrival, conduct and log all required U.S. Coast Guard tests, such as the following:

- steering gear unit changeover, alarms
- rudder indication and full movement

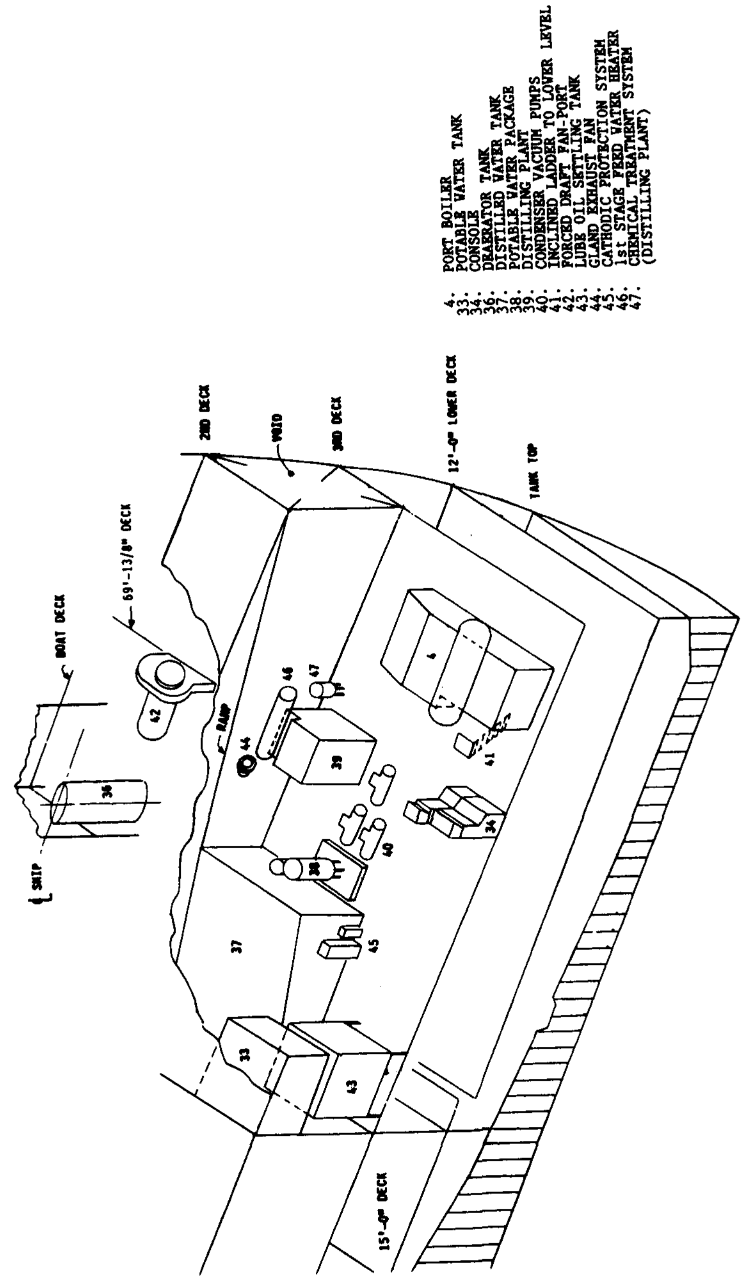

Figure 3-27. Port, third deck, engine room. Courtesy Seaworthy Systems, Inc.

- main engine ahead and astern operation
- emergency generator operation
- storage batteries
- internal communication systems
- vessel control systems

One hour prior to arrival, the chief engineer and first assistant engineer should be notified. The following actions need to be carried out:

1. Start standby turbine-generator (ifrequired).
2. Secure distilling plant.
3. Change over equipment currently operating on bleed steam to live steam.
4. At arrival, slowly reduce speed and prepare plant for maneuvering as outlined below.
5. Secure a burner in each boiler.
6. Open main feed pump and adjust feed pump discharge pressure to maintain proper feedwater regulating valve stroke.
7. Close LP and IP turbine extraction valves. Monitor auxiliary exhaust system pressure to ensure live steam makeup control valve opens.

"Finished With Engines" and Port Operation

Mter "Finished With Engines" is rung down from the bridge, the following operations should be carried out:

1. Place forced draft blowers on slow speed.
2. Secure a second burner in each boiler leaving one in operation in each boiler.
3. If two fuel oil service pumps or two main circulating pumps are in operation, one of each may be secured at this time.
4. Close main steam supply valve and bypass to main engine. Bleed steam off the line by rolling turbines slowly. Engage and start engine turning gear.
5. If port electric load can be carried by a single turbine-generator, secure one turbine-generator.
6. Iflube oil temperature drops below lOO°F, close seawater supply valve to lube oil cooler.
7. Secure atomizing S'team and fuel oil to secured burners on each boiler. Remove atomizers for cleaning.

EMERGENCY OPERATIONS

The following recommended emergency operating procedures are typical for most marine steam power plants and, with some minor variations, can be applied to most modern steam vessels. Consult the ship's manuals for specific procedures that apply to a particular vessel.

In all emergency situations, the first responsibility of the engineer is avoiding unnecessary damage to the ship and its machinery. It is better to act decisively than to take a chance and hope the situation can be resolved before damage occurs. The engineer's alarm should be sounded immediately and the bridge notified of the situation as soon as possible.

Loss or Securing of Fires in Boilers

There are a variety of reasons for the sudden loss of fires in the boilers or the need to quickly secure the fires while steaming. Some ofthe important reasons include the following:

- high or low level in boiler steam drum
- loss of fuel oil pressure
- water in fuel
- failure offorced draft fan
- loss of feedwater supply
- fire in boiler casing, economizer, or air heater

While the actions that must be taken with regard to the overall plant are similar regardless of the reason for the loss of the boilers, the actions that must be taken with regard to the boilers are different. This section will focus on the overall plant procedures. Chapter 5 ("Steam Generation") contains more detailed information on the procedures to be followed for the operation of marine boilers in such emergency situations.

In the event of the sudden loss of fires in one or both of the boilers, the following actions should be taken:

1. Reduce load on boilers by closing ahead throttle on main turbines. Do not use astern turbine to stop shaft, as this will unnecessarily use steam. Reduce other unnecessary steam usage such as cargo pumps, deck machinery, tank heating, and evaporators. Reduce unnecessary electrical loads. The idea is to maintain steam pressure in boilers as long as possible by reducing steam consumption. *Note:* Iffires were lost in only one boiler, it will only be necessary to slow main turbines enough so remaining boiler can maintain main steam pressure.
2. Attempt to get one boiler back on-line as quickly as possible. Cause for loss of fires must be determined and problem resolved. This may involve starting a standby pump, shifting over to other fuel oil settler, etc.
3. When problem is resolved, relight fiFes in boiler, following procedures and safety precautions in chapter 5. When main steam pressure is restored, main turbine throttles can be opened slowly and steam restored to services secured in step 1.
4. If problem cannot be quickly resolved and boilers restarted, steam pressure will continue to drop. Eventually boiler pressure will be insufficient

to power turbine-generators and electric power will be lost. Start emergency generator early so it will be available to take over if necessary.

Loss of Electric Power

1. Reduce load on boilers by closing ahead throttles and securing unnecessary services.
2. Loss of electric power should have started emergency generator, maintaining lights and limited power for vital auxiliaries. If emergency generator did not start automatically, start it manually and restore emergency electric power.
3. Check that main breaker of failed turbine-generator is open. Start standby turbine-generator. Close generator breaker and restore main electric power.
4. Relight fires in boilers and restore main steam pressure. This will require restarting forced draft fans and fuel oil service pump. Check and start air compressor if necessary.
5. Restore main condenser vacuum. This will require restarting main circulating pump and main condensate pump.
6. Start remaining auxiliaries as time and conditions permit.
7. Verify that main lube oil service pump is running and gravity tank level and lube oil pressure and temperature are normal. Open main turbine throttles slowly and bring plant back to desired rpm.

Low Water Level in Deaerator

1. Most important is to avoid loss of feed pump suction. If deaerator level is dangerously low, shift feed pump suction to the distilled water (makeup feed) tank. Consider reducing main turbine power to reduce feedwater flow requirements.
2. Determine cause oflow deaerator level. Possible causes are:
   - condensate pump failure. Check condensate pump discharge pressure and hotwelllevel.
   - empty distilled water tank. Shift makeup feed to spare tank.
   - makeup feed control valve or dump control valve not functioning. Check control air pressure, controller set points, and position of manual isolation and bypass valves.
   - malfunction of condensate recirculation (hotwelllevel) control valve.
3. If necessary to operate directly from distilled water tank for any period of time, divert condensate discharge and feed pump recirculation to distilled water tank. It will be necessary to carefully monitor boiler water oxygen scavenger (sodium sulfite or hydrazine) chemical levels.

Loss of Main Condenser Vacuum at Sea

1. Close in on main throttles and slow main engine. If some vacuum cannot be maintained it will be necessary to stop main engine.

2. Raise vacuum on the auxiliary condenser. Shift turbine-generator exhaust from main to auxiliary condenser.
3. Investigate cause of loss of vacuum. Some possible causes are:
   - loss of main circulating water supply
   - malfunction of air ejector or vacuum pump
   - excessive air leakage into condenser
4. When problem is resolved and main vacuum is restored, open main throttles and bring main engines back up to operating rpm. Shift over turbine-generator exhaust to main condenser and secure auxiliary condenser.

Condenser Leak

1. Control salinity of boiler water by use of surface blow or continuous bottom blow. *Caution:* Heavy bottom blow can interrupt boiler circulation and cause tube failure. Test boiler water hourly until leak is repaired.
2. Inject sawdust into condenser inlet circulating water line using firehose. This should temporarily slow or stop condenser leak.
3. If boiler salinity cannot be controlled by continuous blow, and conditions do not permit securing main condenser for repair, it will be necessary alternately to secure, blow down, refill with treated distilled water, and relight boilers.
4. At earliest opportunity, secure main condenser, locate leak, and plug leaking tube(s). Make note oflocation for replacement at next shipyard repair period. A common technique for locating leak is to drain cooling water side, open inspection plates, and fill steam side with distilled water treated with a fluorescent dye. An ultraviolet light usually will quickly locate leak. Note that shoring of condenser may be necessary due to added weight of water.

Loss of Control Air Pressure

The effects ofthis casualty will depend on the level and type of automation in the plant and fail-safe characteristics of the various control valves. The most critical service on steam ships is usually the combustion control system. The typical steps to take are as follows:

1. Operate boilers on manual control. This includes manual regulation of forced draft damper, fuel oil control valve, and boiler drum level using feed check valves.
2. Services with air-operated control valves that open wide can be controlled manually by throttling outlet isolation valves. Services with air-operated control valves that close can be controlled manually by throttling bypass valves.
3. Start backup compressor or open cross-connection from service air system and restore control air pressure.

4. Shift boiler combustion control system and other services back to automatic control.

Loss of Lube Oil Pressure

1. Most vessels are equipped with two electric-motor driven lube oil pumps, with one in standby status. In case of low lube oil pressure, standby pump should start automatically.
2. If standby pump fails to start automatically, start it manually.
3. If standby pump fails to start and restore lube oil pressure and level in gravity tank, close main throttles and stop main engine. Use astern throttles to stop rotation of propeller shaft.
4. Determine cause of lube oil failure. Possible causes include the following:
   - failure of electrical power supply to lube oil pumps
   - plugged suction or discharge strainer
   - low sump level due to leak in service line, strainer, or cooler. Check also for loss of water seal in lube oil purifier.
5. If lube oil supply cannot readily be restored, start auxiliary condenser and shift turbine-generator exhaust from main condenser. Properly secure main turbine. Use jacking gear to reposition turbine rotors occasionally until completely cooled.

REVIEW

1. What are the difficulties that prevent an operating Carnot cycle from being built?
2. What are the four processes that make up the Rankine cycle?
3. What are the benefits of adding reheat to a Rankine cycle?
4. What determines the best pressures for bleeding steam from a turbine for feedwater heating?
5. List five features that a designer can include in a steam cycle to improve its efficiency.
6. The boilers on plants with high-pressure feedwater heaters are almost always fitted with air heaters rather than economizers. Why?
7. List the things that can be done to reduce fuel consumption by proper boiler operation.
8. Explain why cascaded bleeds will improve the efficiency of a steam plant operating at part loads. What other things can be done to reduce fuel consumption at part loads?
9. Explain how the three control valves (bleed, live steam makeup, and dump) in the auxiliary exhaust system should be set up to minimize losses.
10. List the possible causes of the sudden loss of the fires in both boilers.

11. What is the first action that should be taken in the event of the loss of the fires in the boiler?

REFERENCES

Burghardt, M. D., and J. A. Harbach. 1993. *Engineering Thermodynamics,* 4th ed. New York: HarperCollins.

EI-Wakil, M. M. 1984. *Powerplant Technology.* New York: McGraw-Hill.

Seaworthy Systems, Inc. 1978. *A Practical Operating Guide for Tuning Steam Turbine Propulsion Systems.* NTIS Report MA-RD-920-78053. Essex. Conn.: Seaworthy Systems, Inc.

Seaworthy Systems, Inc. *"S.S.Caguas" Machinery Operating Manual.* Essex, Conn.: Seaworthy Systems, Inc.

Society of Naval Architects and Marine Engineers (SNAME). 1992. *Marine Engineering,* 3rd ed. R. L. Harrington, ed. Jersey City, N.J.: SNAME.

___ . 1973. *Marine Power Plant Heat Balance Practices.* T&R Publication 3-11. Jersey City, N.J.: SNAME.

___ . 1975. *Recommended Practices for Correcting Steam Power Plant Trial Performance.* T&R Publication 3-17. Jersey City, N.J.: SNAME.

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CHAPTER 4

Bearing Application and Lubrication

WILLIAM D. MARSCHER

INTRODUCTION

This chapter will discuss the selection, application, and maintenance of bearings and their lubrication systems as used in rotating machinery in marine applications.

Purpose of Bearings

Bearings are placed in machinery to allow reciprocating or rotating components to resist static and dynamic forces, thereby maintaining shaft position and alignment sufficiently to allow the machine to reliably perform the function for which it was designed. Therefore, bearings act as load carrying components, which are able to support the load in spite of the fact that the two components that they bridge are in relative motion. They must perform this function in a manner that requires minimum maintenance, which implies that their rate of wear is low and they must entail a power loss and associated heat loss that is also manageably low.

A secondary but often important purpose of bearings is to playa key role in setting the frequency and minimizing the strength of rotor critical speeds. This avoids the damaging vibration of "resonance" by keeping easily excited critical speeds away from the frequency of potentially large forces imposed on the rotor, such as the residual imbalance force occurring at one times the rotor speed.

Purpose of Lubrication

Lubrication or lack thereof, associated friction and wear, and the design of components to minimize friction and wear is known as *tribology.* Bearing design is one aspect oftribology. Another is the selection of an appropriate means and medium for bearing lubrication.

The purpose of lubrication is to allow parts in relative motion and loaded against each other to accomplish their tasks with a minimal friction and wear rate. Lubrication reduces the heat generation, noise, and power loss associated with friction and prevents wear-causing contact between the moving and the opposing stationary parts of the bearing. It helps to convey away from the load zone any heat and wear particles that are produced. It also protects the metal parts of the bearing from corrosion, by forming a protective coating on all surfaces exposed to the lubricant. This is particularly important in antifriction (also called rolling element) bearing applications, as described below.

BEARING SELECTION

The key factors in bearing selection and application are load carrying capacity, degree of maintenance required, and a given bearing's ability to avoid rotor dynamic difficulties through its adequate stiffness and damping characteristics. In addition, the initial cost and replacement cost of certain bearings can be substantial. Sometimes this additional cost is related to the need for a more sophisticated lubrication system, e.g., in the case of oil mist versus sealed grease lubricated ball bearings.

Bearing Types

Engineers often break bearing types into four categories:

1. Fluid film (or hydrodynamic) bearings, including product-lubricated, and air and gas bearings
2. Rolling element (or antifriction) bearings, such as ball or roller bearings
3. Sleeve bearings lubricated by a boundary film or solid particles like graphite
4. Externally energized bearings, such as hydrostatic bearings or magnetic bearings

Fluid film bearings are generally one of the most expensive types of bearing and are usually the most difficult to maintain. However, in difficult environments or under high load and/or high speed, fluid film bearings are easier to incorporate into a design. Rolling element bearings are generally less expensive, are available off the shelf in various sizes and tolerances, and have well characterized lubrication needs and life expectancies. However, at higher speeds and/or temperatures, they need sophisticated lubrication systems and eventually reach speeds at which their use entails too much risk. Sleeve bearings are the least expensive bearings of all and need little maintenance, but are restricted to relatively low speeds and loads. Externally energized bearings require special and expensive application-

specific engineering, and so are used today only where no other bearing will do the job. Hydrostatic bearings also involve too much indirect power loss for realistic clearances. Existing magnetic bearings include a bulky control box that needs frequent retuning. However, advanced forms of hydrostatic and magnetic bearings are under development that may make these bearings more suitable for many applications in the near future.

Marine Bearing Applications

Every type of machine on board a ship requires bearings of some sort. The most important applications of bearings that require occasional maintenance or replacement are fire pumps, ballast pumps, service pumps, cargo pumps (particularly vertical turbine pumps), boiler feed pumps, stern tubes, diesel components (such as crankshafts, connecting rods, crossheads, and piston pins), turbochargers, blowers, compressors, fans, gas turbines, steam turbines, gears, and electric generators and motors.

Bearing Details and Design Principles

The following section explains how the various bearing types operate, and describes some of the selection and application trade-offs.

FLUID FILM BEARINGS

By *fluid film bearings,* engineers mean bearings that are designed to operate under the principle of building up a hydrodynamic (moving liquid) wedge between parts in relative sliding motion. These include plain journal bearings, fixed pad journal bearings, and tilting pad bearings.

Journal bearings function through the occurrence of hydrodynamic lubrication. In this form of lubrication, when two bodies slide over each other in the presence of a liquid or gas, a wedge of fluid builds up between the two bodies because of the dynamics of the fluid, as shown for the cross section of an operating journal bearing (with greatly exaggerated clearances) in figure 4-1. The fluid generally adheres to each body and must shear in order to accommodate the relative motion between the two bodies. The basic equation that governs this process, assuming that the sliding velocity is not too high relative to the fluid's viscosity (viscosity is the fluid's resistance to shear, as explained below), is the so-called Reynolds equation, which in simplified form (incompressible fluid, no flow perpendicular to the sliding direction, steady load, and lubricant temperature at steady-state) is

$$d\,[h^3\ (dP/dx)]/dx = 6\ Ull\ (dh/dx)$$

This equation establishes how fluid film pressure P depends upon film thickness h, sliding velocity U, position in the contact zone x, and viscosity 11.

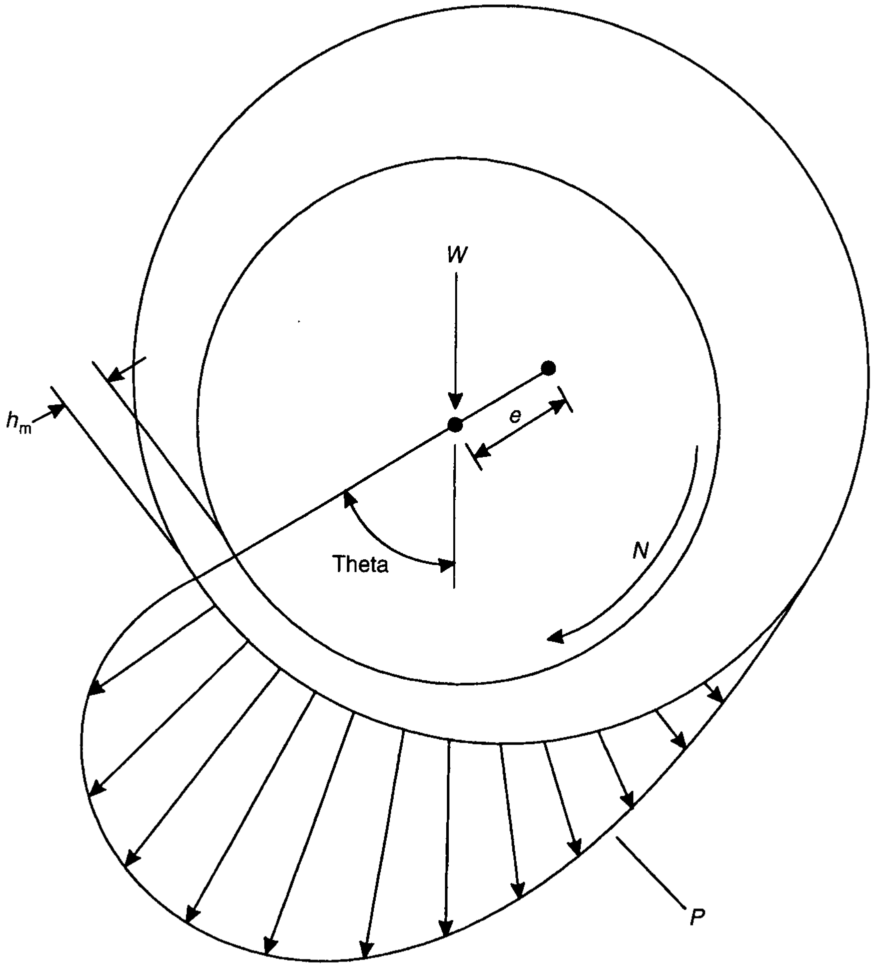


Figure 4-1. Functioning of hydrodynamic lubrication in a journal bearing

The Reynolds equation as simplified above is a good approximation for the actual behavior of a short, relatively wide plate (like a crosshead in its ways), or of a bearing that is much longer than its diameter.

Journal Bearing Theory

Most journal bearings have a length L equal to or less than their diameter D, and for this situation the Reynolds equation takes on a slightly different form, since flow in the direction of sliding, i.e., direction x, is much less than flow out the ends of the bearing, i.e., direction y. The revised form is

$$h^3\left(d^2P/dy^2\right) = 6U\mu(dh/dx)$$

This equation can be integrated twice in y, which is easy since for the short bearing situation none of the other parameters beside P depends on y. When this is done, it is seen that the pressure under the bearing is simply a parabola, and several useful approximate relationships can be derived, as follows, where the eccentricity ratio ε is introduced as the movement of the shaft centerline e divided by the centered radial clearance c. First, for load capacity:

$$\text{Load capacity } W = U\mu\, L^3\pi\,\varepsilon\big/\left(4c^2\right) \text{ for light loads}$$

and for heavy loads multiply the right-hand side by the additional factor K, where

$$K = \left[\left(0.62\,\varepsilon^2 + 1\right)^{1/2}\right]\Big/\left(1-\varepsilon^2\right)^2$$

Another useful parameter is the leakage out the ends of the bearing, which becomes the lubricant flow rate required:

$$\text{Flow rate } Q = U\,c\,L\,\varepsilon$$

A third parameter is the frictional power loss estimate, the so-called Petroff equation:

$$\text{Power loss hp}_f = 2\pi\mu\,U^2\,D\,L/(2c)$$

This frictional power loss can be translated into an average temperature rise in the lubricant. This is done by dividing the power loss by the volume flow rate calculated above, and then further dividing by the lubricant density and the lubricant specific heat, typically about 0.029 lb/in^3 and 0.5 Btu/lb-F, respectively.

Finally, minimum film thickness h_m can be defined in terms of eccentricity as

$$h_m = (1-\varepsilon)\,c$$

This still leaves the problem of determining ε. Eccentricity at low loads can be estimated from the load capacity equation by setting the load capacity equal to the actual imposed load. A more general method is to use the graphical procedure discussed next.

Lubrication engineers prefer to use a dimensionless number derived from the Reynolds equation, known as the Sommerfeld number S, to correlate the other parameters that are of interest in bearing behavior, such as the bearing eccentricity. Essentially, the Sommerfeld number is the sliding speed per unit load, and its mathematical definition is

$$S = (R/c)^2 \mu\, N/P$$

where

N = rotational speed of the journal in rpm (revolutions per minute)
R = bearing radius, i.e., $D/2$

and, as a reminder: $P = W/L\,D$ is mean pressure due to a load W on a bearing L long and diameter D; c is the bearing radial clearance, i.e., half the difference of the journal and bore diameter; and μ is the lubricant viscosity, adjusted to the local temperature.

S is the main parameter against which parameters are plotted of key importance to the designer and to the user. Examples of such parameters are eccentricity (fig. 4-2), bearing stiffness (fig. 4-3), bearing damping (fig. 4-4), and bearing instability threshold (fig. 4-5). Figures 4-2 through 4-5 are taken from the *Handbook of Lubrication and Tribology,* edited by Booser (see references at the end of chapter), which also has a great deal of additional detail on this subject.

Some additional comments are in order concerning the plots of the various parameters versus the Sommerfeld number. Each of the plots assumes a so-called plain journal bearing with constant radial clearance when the eccentricity is zero, a length that is shorter than its diameter, and theoretically no grooves. There are other types of journal bearings, the chief types of which are shown in figure 4-6. These journal bearings have the same kind of parametric dependencies versus Sommerfeld number, but the quantitative values of the plots, and even the shapes of the curves to some extent, will vary. In addition, the load capacity, lubricant flow requirement, and frictional power loss will not follow the simple relationships given above, although they can be used as rough approximations. A more generalized, but necessarily more empirical, form of the above relationships was given, for example, by Dudley Fuller in his chapter on fluid film bearings in *Marks' Standard Handbook for Mechanical Engineers,* listed in the references. The various bearing manufacturers also provide plots and relationships for the specific bearings that they manufacture, based on proprietary tests and calculations, and will provide them upon request.

Figure 4-2 shows different curves for ruptured versus unruptured films. This refers to whether or not the lubricant vaporizes, or cavitates, away from the portion of the lubricant that supports the externally im-

posed load. Although in a centrifugal pump such cavitation can cause damage to its vanes as the vapor bubbles form and then explosively collapse, this is usually not a problem in an oil-lubricated bearing, because the bubbles collapse too slowly for damage to occur. There are exceptions to this, however, as will be illustrated later. One exception is in reciprocating

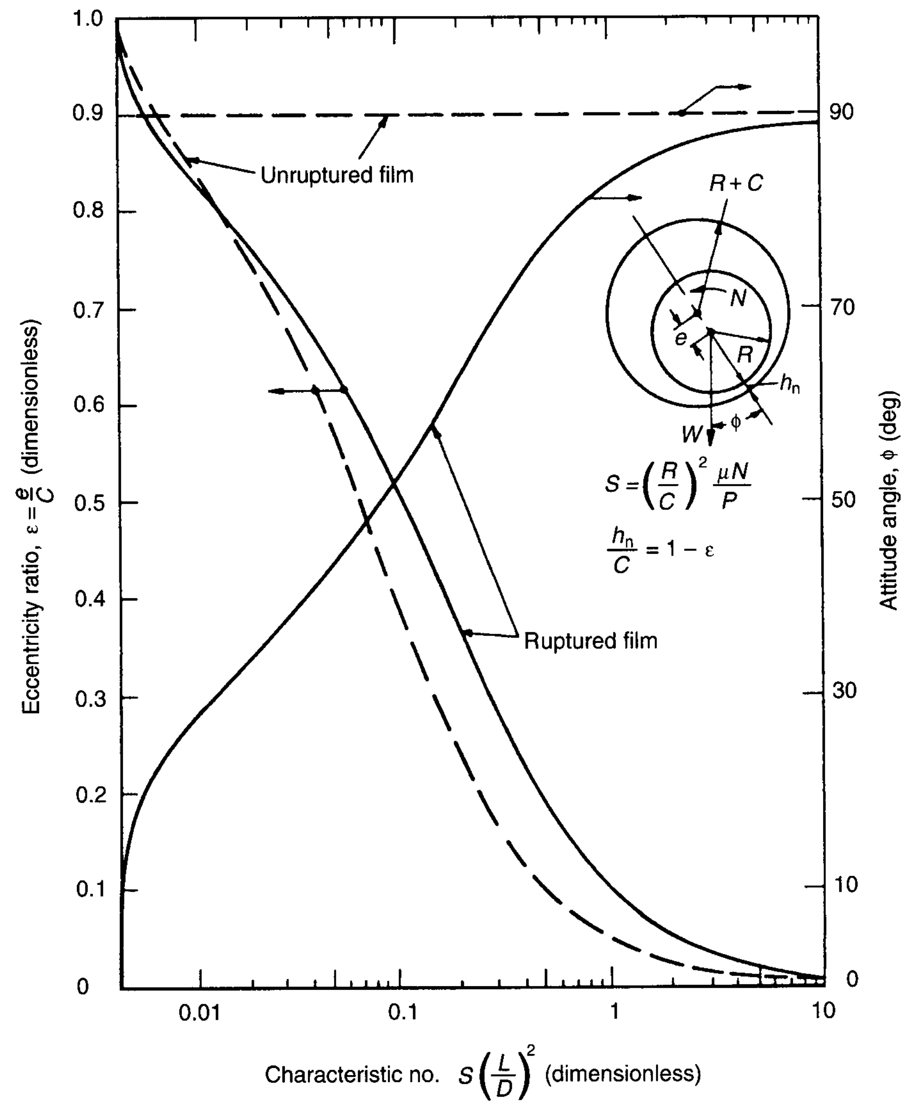


Figure 4-2. Dependency of eccentricity on Sommerfeld number. Courtesy Society of Tribologists and Lubrication Engineers and E. R. Booser, editor, *Handbook of Lubrication and Tribology,* CRC Press.

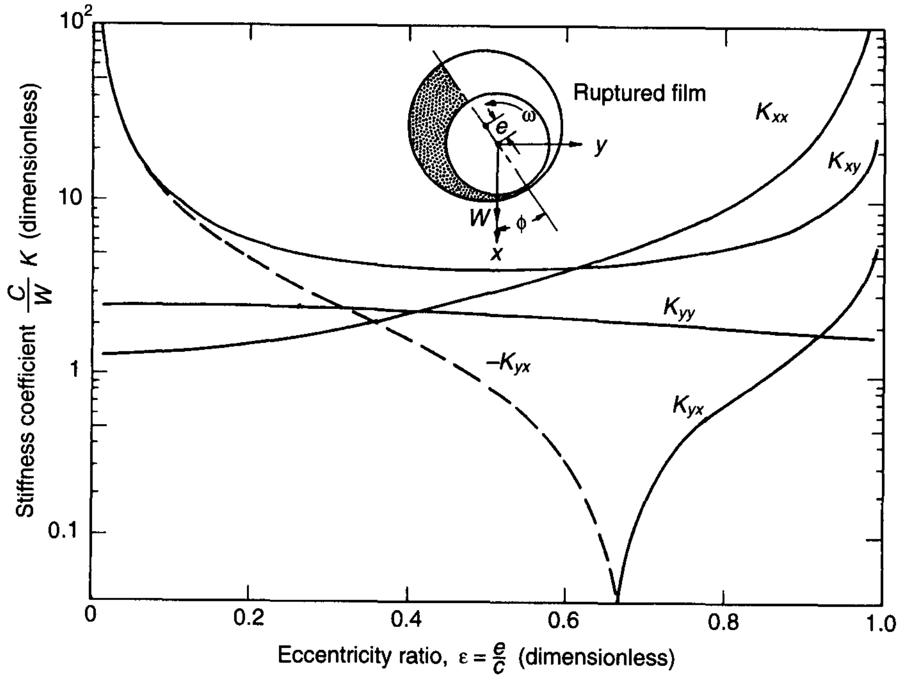


Figure 4-3. Dependency of bearing stiffness on Sommerfield number. Courtesy Society of Tribologists and Lubrication Engineers and E. R. Booser, editor, *Handbook of Lubrication and Tribology,* CRC Press.

machinery like diesels, where some loads are impulsive and are quickly applied and removed. If cavitation damage occurs, a different viscosity oil or a less fatigue-prone bearing liner should be sought from the manufacturer. In normal circumstances, however, cavitation is typical in machines operating at 1,800 rpm or faster and causes no harm. It does change some of the parameter curves as shown, and this should be taken into account when performing calculations. If a guess must be made, the best guess is usually that cavitation is taking place, and therefore the film is being ruptured.

The stiffness and damping coefficients given in the figures are necessary to determine the critical speeds of the shaftl bearing system and to evaluate its rotordynamic behavior, as discussed in detail later in this chapter. By coefficients, we mean the factors that the vibration displacement and velocity, respectively, must be multiplied by to determine the forces that created them. These coefficients are directional and are defined on the basis of the direction of the load, versus the direction of the displacement in response to the load. Although bearings are often conceptualized as ifthey are simple springs supporting the shaft, the situation is more complex than this.

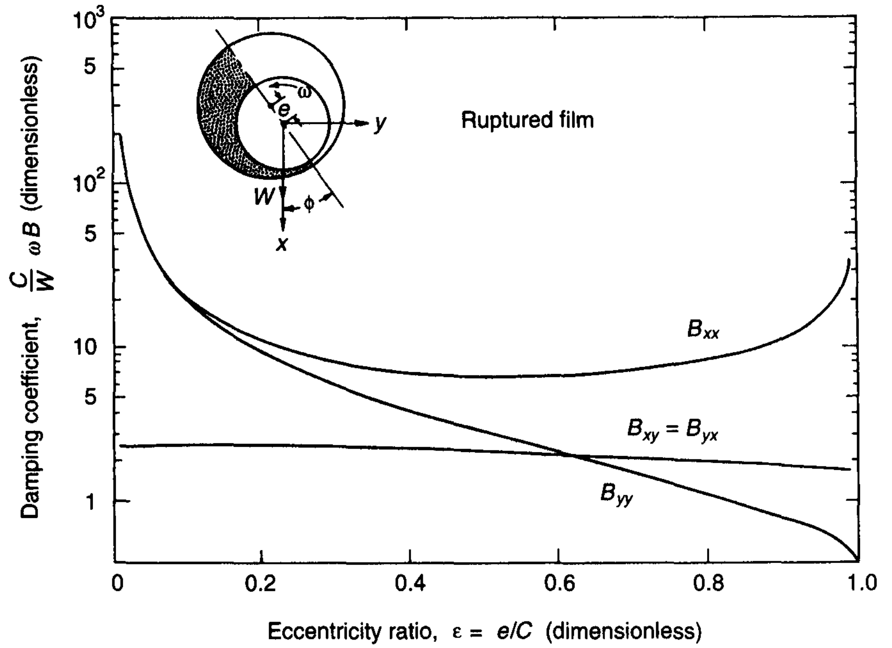


Figure 4-4. Dependency of bearing damping on Sommerfield number. Courtesy Society of Tribologists and Lubrication Engineers and E. R. Booser, editor, *Handbook of Lubrication and Tribology,* CRC Press.

Figure 4-1 shows how a static pressure profile skewed off to one side forms in the bearing lubricant in response to an externally imposed load on the shaft. The vector summation ofthis pressure load is an equal and opposite load vector to the originally imposed load. The amount that the shaft displaces in the direction of the externally imposed load as the next increment of load is applied can be divided into the load increment, to give a bearing "direct" stiffness, in pounds per inch or newtons per millimeter. This is normally written as K_{xx}. The plot in figure 4-3 plots K_{xx}, which is the same stiffness, but nondimensionalized by multiplying it by the radial clearance c, and dividing it by the total imposed load W. The plot also gives K_{yy}, K_{xy}, and K_{yx}, which are similarly nondimensionalized by c and W. What the first subscript in each of these represents is the load direction, and the second subscript is the displacement direction. Therefore, K_{yy} is the direct stiffness in the y direction (the direction perpendicular to the externally applied load), i.e., the amount of y direction load it would take to cause a given amount of y displacement, if at the same time the original load is kept in the x direction.

The most foreign concept to those exposed to it for the first time is represented by the so-called cross-coupled stiffnesses, K_{xy}, and K_{yx}. These are caused by the skewness in the static pressure in the lubricant film under the shaft, as shown in figure 4-1. This skewness causes the journal shaft to deflect not just downward when it is pushed downward, but also sideways, tending to make the shaft move in the same direction as the lubricant film that is whirling within the bearing clearance space. This factor is very important in setting the stage for the bearing to become potentially unstable, for reasons that are discussed later in this chapter. For the time being, suffice it to say that high cross-coupled stiffness is to be avoided unless very high energy-absorbing damping occurs simultaneously with it. This is the reason why a plot showing this factor is important. Figure 4-5 gives a map

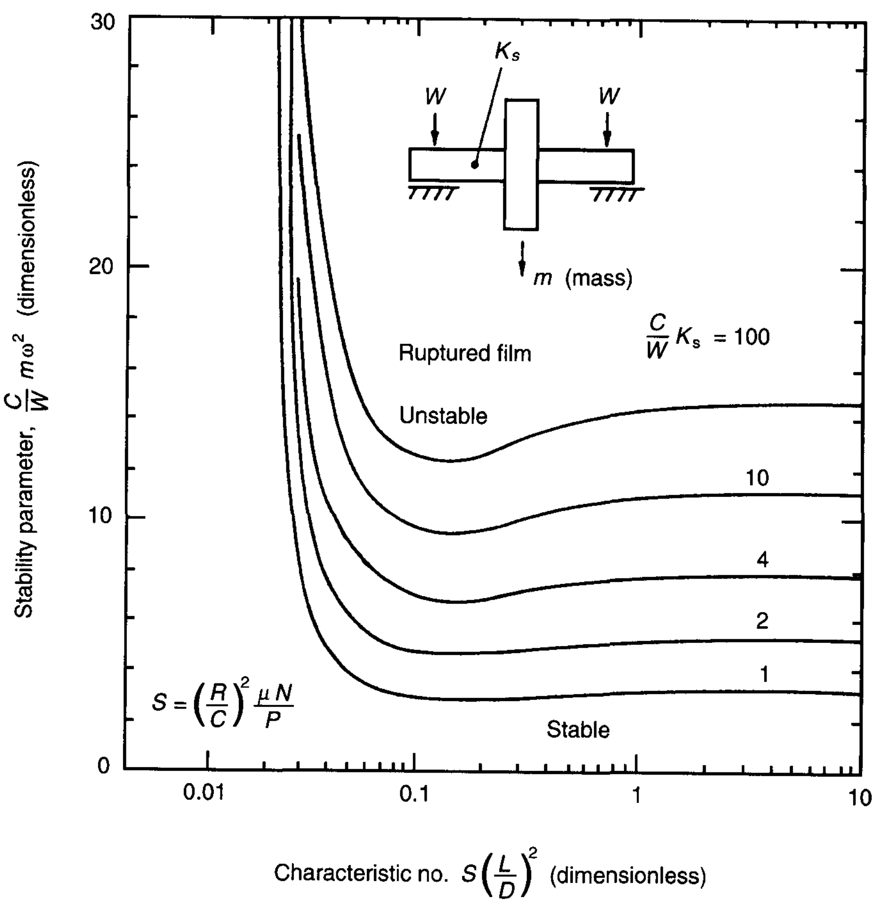


Figure 4-5. Dependency of bearing oil film instability (oil or shaft whip) on Sommerfield number. Courtesy Society of Tribologists and Lubrication Engineers and E. R. Booser, editor, *Handbook of Lubrication and Tribology,* CRC Press.

of potential for bearing-induced rotor instability for a plain journal bearing. The new parameters in this plot are *Ks*, which is the shaft beam stiffness as measured at the center of mass relative to the bearing supports, and the mass *m* of the rotor system. It is important to stay away from the upper right-hand portion of this plot, or high-vibration rotordynamic instability becomes likely.

The other forms of journal bearing, shown in figure 4-6, were originally developed primarily to combat the rotor instability problem, and all are generally superior to the plain journal bearing in this regard. Stability maps for these bearings are roughly of the same form as figure 4-5, but the boundary line of instability moves up and to the right, to open up a broader safe operating range. In the case of tilting pad bearings, the boundary essentially no longer exists, and they are not prone to bearing-caused rotor instabilities. They also have less damping than most of the journal bearing types, however, and damping is the main factor (along with minimized oscillating load level) limiting vibration when the machine operates at or

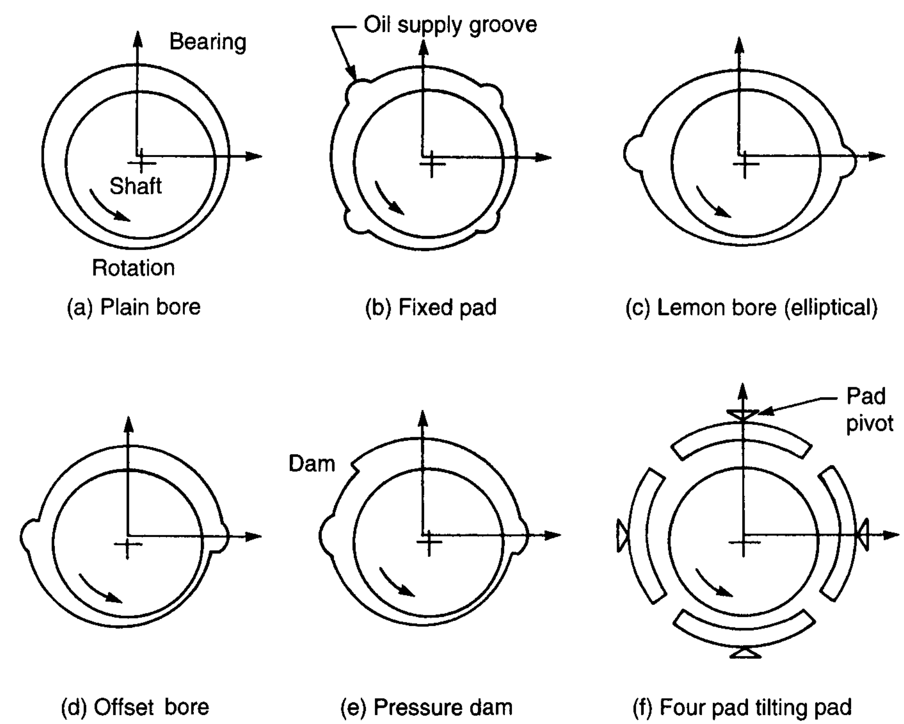


Figure 4-6. Different styles of journal bearings. From *Fundamentals of Design Fluid Film Bearings.* S. M. Rhode, C. J. Maday, and P. E. Allaire, eds. (1979). Courtesy American Society of Mechanical Engineers.

near a critical speed. Tilting pad bearings also are considerably more expensive, require greater maintenance, and have significantly greater power loss than other journal bearings. Therefore, tilting pad bearings, in spite of their significantly greater expense, are not automatically the best bearing for a rotating machine. If bearing-induced rotordynamic stability is an issue, however, they are superior to all other bearing types.

The damping factors B_{xx}, B_{yy}, B_{xy}, and B_{yx}, shown in figure 4-4, have a similar directional meaning and are nondimensionalized in a similar fashion to the stiffness factors, except to fully nondimensionalize them they must be multiplied by the additional factor ω, which is the rotational speed put in terms of radians per second, by dividing rpm by 60 and multiplying it by 2π.

Typical Journal Bearing Configurations

Journal bearings can be broken down into plain bore, lemon bore, offset bore, pressure dam, and fixed pad. A variation of the fixed geometry journal bearing is the tilting pad bearing, covered separately below. These various types of bearings are shown for reference in figure 4-6. There are advantages and disadvantages to each type of journal bearing.

Plain bore journal bearings have a circular cross section and generally only a single lubricant feed groove, either running along the length of the bearing, or running around the bearing circumferentially, splitting the bearing close clearance into two equal parts. A smooth, ground shaft surface rotates within a slightly larger-diameter smooth-surfaced circular cylinder. The load bearing effect is provided by a hydrodynamic wedge that builds between the rotating and stationary parts as rotating fluid flows through the narrow part of the eccentric gap between the shaft journal and the cylindrical bearing insert, as described above. The eccentricity of the shaft within the journal is caused by the net radial load on the rotor, forcing it to displace within the fluid gap. The hydrodynamic wedge provides a reaction force that gets larger as the eccentricity of the shaft journal increases, similar to the buildup of force in a spring as it is compressed.

This type of bearing is also sometimes called a cylindrical or sleeve bearing. It is simple in design and inexpensive to manufacture. The plain journal bearing has good load carrying capability and excellent damping. However, it is prone to a type of rotor instability called *shaft whip*, and surprisingly is most susceptible when it is under light rather than heavy loads, as shown in figure 4-5 (remember that Sommerfeld number goes up as load goes down). The source of its stability problems can be traced to its characteristically high cross-coupled stiffness, together with the whirling of the oil film lubricant at about ½ running speed (so-called half-speed whirl), as discussed later in this chapter. Figure 4-7 shows a typical steam turbine plain journal bearing, with a spherical seat feature to aid in bearing alignment.

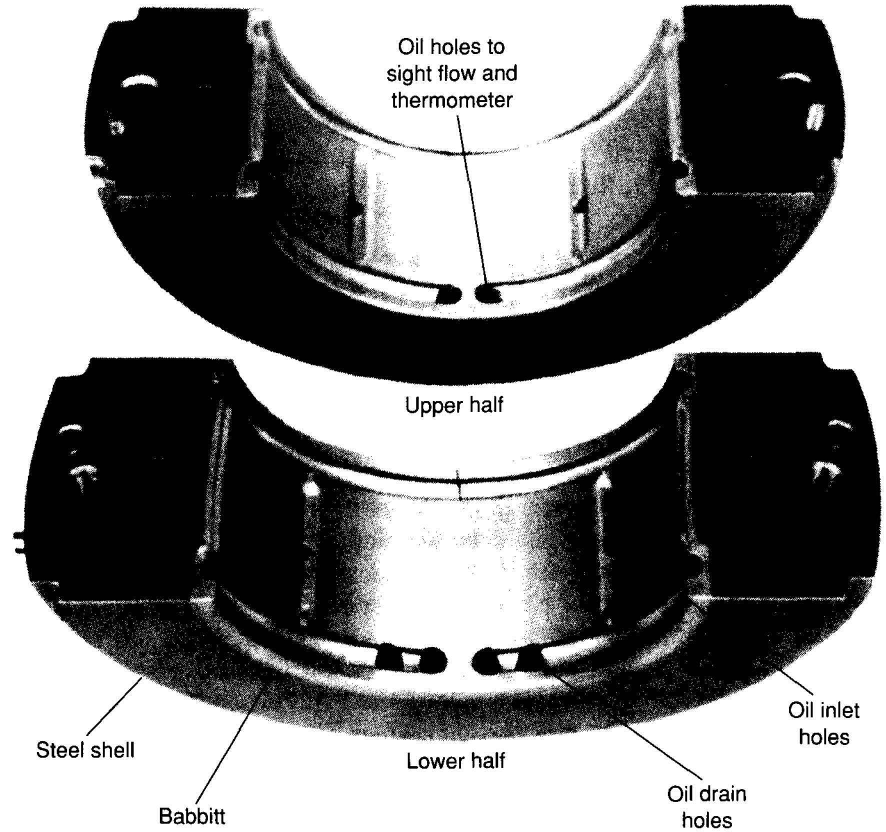


Figure 4-7. Typical marine steam turbine journal bearing with spherical seat for alignment. Courtesy General Electric Company.

In turbochargers, a common variation of the plain journal bearing is the floating ring bearing. Essentially, this is a bearing within a bearing, in that the shaft rides within a floating cylinder, the inner diameter of which provides the bearing surface for the shaft, and the outer diameter of which provides the bearing surface for the bearing housing. The purpose of the floating ring, which generally rotates at some speed below the shaft speed, is to limit the shearing in either of the two oil films to some fraction of what it would be if the high-speed shaft were running directly versus the stationary housing bore. It also tends to discourage large cross-coupling force buildup, since the ring is free to deflect to open up more clearance as this force develops. The dynamics of this bearing are not well understood, and it is very sensitive to small clearance changes due to less than outstanding machining of the bearing parts, unexpected rapid temperature variations, or relaxation of residual stresses locked into the ring during its fabrication.

The best policy is to only buy OEM replacement parts for these bearings and strictly follow OEM recommendations concerning bearing oil type, inlet temperature, and maintenance intervals.

Lemon bore (elliptical) or offset bore journal bearings, as shown in figure 4-6, have shells that were originally circular but through intentionally large crush, shimmed offset, or offset machining have clearance that varies evenly around the circumference each half-turn of the shaft. This is done to decrease the cross-coupled force the lubricant wedge creates, to break up consistent whirling ofthe lubricant around the bearing, and to discourage shaft whip at the same ftequency as the whirling, with its attendant destructive vibration amplitudes. However, these bearings have limited effectiveness in accomplishing this; they are usually effective only in eliminating shaft whip in marginal cases. Also, keep in mind that shaft whip is likely only for light loads and when the running speed of the machine is greater than twice the first critical speed of the shaft. Therefore, in highly loaded or stiff shaft machines, the ovalized or offset bore configuration is generally an unnecessary complication. Unless the person responsible for the repair has access to a rotordynamics expert, however, the best approach is to *never* replace a manufacturer's bearing of given complexity with one that is simpler without manufacturer concurrence.

Elliptical bore bearings have good load carrying capability and are often used in large steam turbines where the load direction is well known (generally, due to gravity), so that the load vector can be kept in line roughly with the minor axis of the bearing, which is its stiffest direction.

Fixed pad journal bearings have three or more grooves running along the length of the bearing, which are generally equally spaced circumferentially. The close clearance pads between the grooves can have constant radius like a plain journal bearing. At considerable extra expense, they also can have some circumferential variation in radius to aid wedging action in a given direction of rotation (the latter is sometimes called a multilobe bearing). The multiple pads further improve on the lubricant whirl discouragement principle of the lemon bore and offset bore bearings, and properly designed fixed pad bearings are often quite effective in suppressing whirl and shaft whip.

Fixed pad bearings are not particularly sensitive to the load direction. In addition, the extra grooves help to better distribute lubricant within the bearing. Negative issues are the increased bearing cost and (in some bearings with profiled pads) preferential rotational direction. In addition, vibration-absorbing damping and static-load carrying capacity may be somewhat reduced relative to a plain journal.

A variation ofthe fixed pad bearing is the pressure dam bearing, which has a single wide circumferential groove of gradually increasing depth over about 135 degrees of the upper bearing shell circumference. This pocket causes a static pressure to build up to keep the bearing somewhat off-

center, which stiffens it, avoids significant compromises in load capacity for downward loads, and replaces damping lost initially by the grooving. This is a good selection in machines where the machine's natural load on the shaft is always less than the pressure dam load.

A type of fixed pad journal bearing commonly used in stern tubes and vertical turbine cargo pumps is the Cutless rubber bearing. These bearings have a rubber or plastic composite insert, which is either octagonal in shape at the ID, or is round with one or more deep axial grooves. The axial grooves may be straight or spiral. The purpose of the grooves is to convey plenty oflubricant (usually water, or the pumped cargo) along the bearing to the loaded zone, and to allow foreign material and wear swarf to be carried away from the loaded zone. The principle of the effectiveness of these grooves to allow rubber to be a good bearing material, particularly for shafts running with water as a lubricant, was discovered by accident by a mining engineer. A failed bearing in a pump was allowing his mine to fill up with water, and in desperation he slit a piece of rubber hose longitudinally, slip-fit it over the shaft, and jammed it into the bearing housing. Not only was this a satisfactory fix, but it actually operated longer than the original metal bearing, and with less vibration.

Cutless bearing clearances are typically two to four times that of plain journal bearings. However, these bearings typically have lengths that are several times the shaft diameter, to help keep local bearing pressures low, to restrict the degree of shaft tilt prior to interference with the shaft, and to limit dynamic imbalance induced by the shaft whirling like a "jumprope." These bearings are very tolerant of bumps and rubs and resist oil film instability (oil whip) at low loads, so they make very good product lubricated bearings and are satisfactory for vertical shafts with uncertainty in load amount or direction. The worst case bearing load in lineshafts operating within this style bearing can be determined by calculating the amount of imbalance force that would be created by the shaft deforming as it whirls like ajumprope. This can be done by evaluating the shaft as a bowed beam simply supported at each end, deflecting into a roughly parabolic shape to the fullest extent allowed by the bearing until the shaft angle within each bearing becomes enough to take up the full clearance in the bearings. Typically, such loads are not large, which is one of the reasons why these bearings have been so successful.

Rubber bearings of the type described can be effective when used to support high-speed shafts because they allow the shaft to turn easily about its center of rotation even when this does not correspond to the planned geometric center. This is important, for example, when the propeller shaft is bent, or if one blade is broken off, as often happens in an icebreaker.

Other material besides rubber can be used for Cutless style bearings, as well as for plain journal bearings when they are lubricated by the liquid being pumped. Recently, there have been successful applications of silicon

Figure 4-8. Application of ceramics to journal bearings: *above,* typical pump set of ceramic bearings; *below,* ceramic tilting pad radial bearing and associated shaft sleeves. Courtesy Federal-Mogul Bearings Group.

carbide and carbon/carbon composite bearings, as well as use of reinforced plastics such as PEEK. Some examples of ceramic pump journal bearings are given in figure 4-8. However, none ofthese materials, particularly rubber, survive very long if the bearing is allowed to run dry, i.e., without the constant presence of the lubricating liquid.

Tilting pad bearings, in which essentially the fixed pads discussed above are cut free from the bearing backing and are put on pivots or "rockers" which mayor may not be centered under the pads, are shown in figure 4-6. Four to six pads are common, and the concept can be used for either radial or thrust bearings. The tilt of the pads allows the pad to cock against the lubricant flow, increasing the hydrodynamic wedging action. The tilt also does not allow a moment to develop across the pad circumferentially, so that large cross-coupled forces (forces offset relative to the minimum clearance "pinch" in the bearing) cannot develop in the lubricant wedge; the pad simply adjusts its tilt until these forces equalize on each side ofthe pinch. By eliminating this cross-coupling, the tilt pad bearing can minimize the opportunity for shaft whip or rotordynamic instability to occur. However, this is only a benefit if the rotor runs at speeds more than twice its first critical speed. Also, the tilt pad bearing decreases vibration-absorbing damping significantly; therefore, for normal vibration causes such as imbalance, it allows the vibration to become greater than other journal bearings. Other negatives are high relative cost and increased power loss associated with the tilt pad bearing, and its increased proneness to reliability problems because of its moving parts. It is also more difficult to check shaft clearances, and easier to damage the bearing during disassembly. Positives can be increased load capacity and alignment tolerance.

Figure 4-9 shows a typical tilting pad bearing, in this case with four movable pads or shoes. These particular pads pivot about small journal pins on each end ofthe pad. Oil is fed from behind the pads and is entrained into the pad-shaft interface by the draft of the shaft motion. What is shown is probably the simplest form of tilting pad bearing, with smooth pads of simple design and minimal parts for support and lubricant distribution, although there is at least one company (KMC, Inc.) that makes small scale tilting pad bearings where the pads are machined into the backing, supported by thin flexible pedestals of material left purposely by the machining. A more complex bearing of the tilting pad type is shown in figure 4-10, which includes a so-called leading edge groove (LEG) at the edge that the shaft approaches first during rotation. This groove is reported to do an excellentjob of uniform lubricant distribution to the pad surface, minimizing peak temperatures and maximizing load carrying capability.

Particularly for thrust bearing applications, it has been found useful to increase the design complexity somewhat to ensure the proper alignment of the pads to the rotor thrust disk. Figure 4-11 shows a different tilting pad concept, where pad pivot capability and support is provided by leveling

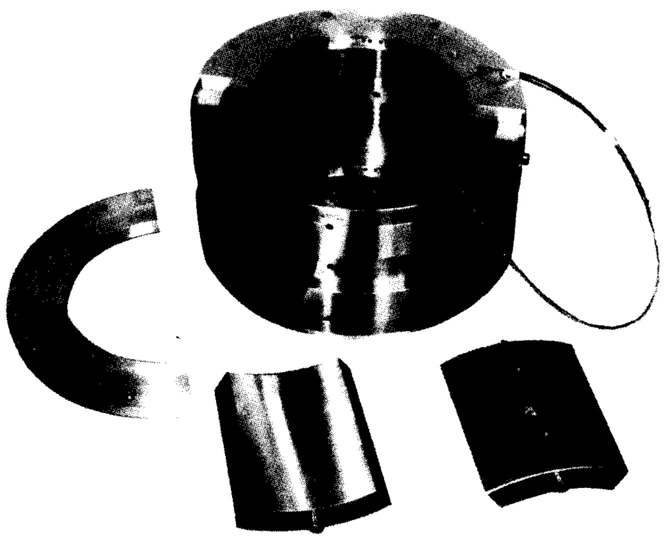


Figure 4-9. Four-shoe standard tilting pad bearing. Courtesy Kingsbury, Inc.

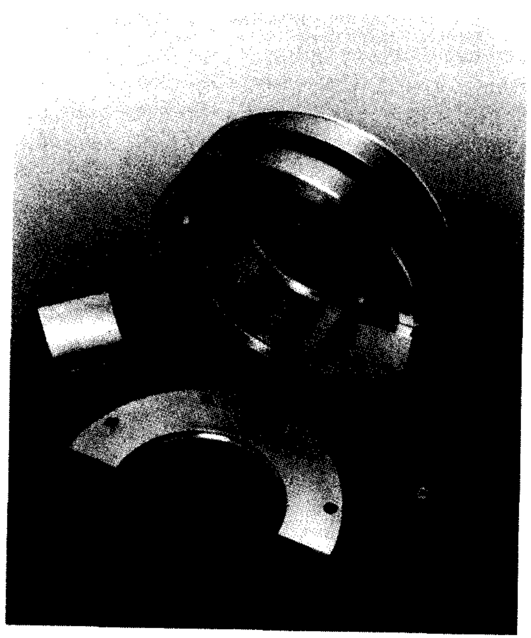


Figure 4-10. LEG tilting pad five-shoe journal bearing. Courtesy Kingsbury, Inc.

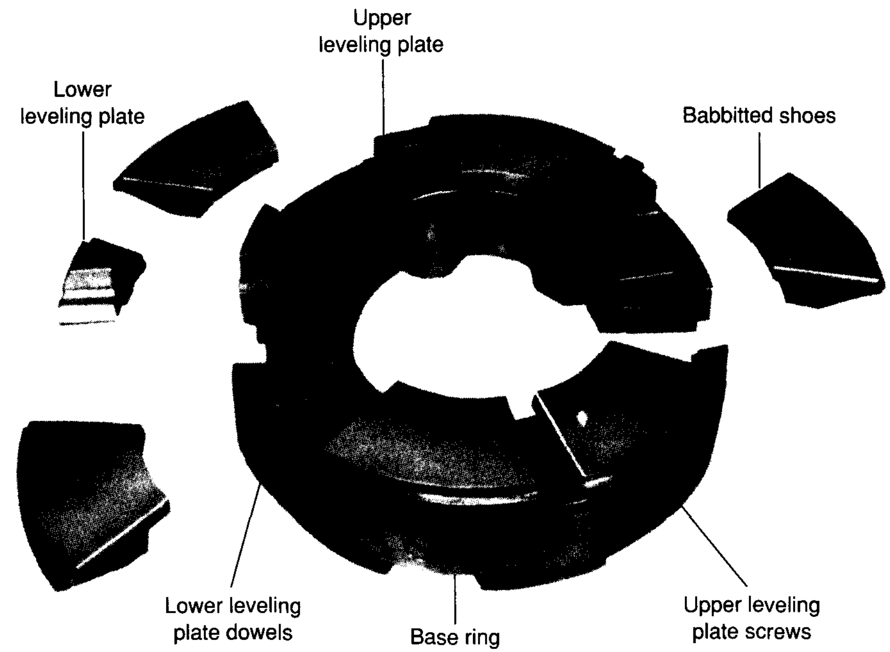


Figure 4-11. Self-aligning thrust bearing shown with component parts. Courtesy General Electric Company.

plates that in turn rest on *buttons* or dowels. The pad shapes of such bearings are necessarily also more complicated. Not only is alignment between the thrust pads and rotor a key factor in maintaining adequate minimum oil film thickness across the load zone, but stiffness and preciseness of the surface contour must also be adequate. This means that replacement parts should be obtained only from highly qualified sources, such as the original bearing manufacturer. Also, babbitt replacement and recontouring should be performed only by the manufacturer or a shop with proven success in refurbishing the given bearing. Even a very small change in pad surface contour or, depending upon design philosophy, flatness, can have dramatic effects on lubricant peak temperatures and long-term load carrying capacity of tilting pad thrust bearings.

Analysis ofthe load-carrying capability and other important characteristics oftilting pad bearings are best carried out with empirical curves available from catalogs or special publications available from the manufacturer. Although fundamentally based computer programs and even manual analysis techniques are sometimes touted as accurate, be skeptical in applying them. At this writing, there are few that do an adequate job of predicting load capacity or peak temperature. On the other hand, the manufacturer charts and curves for various bearing types and sizes are derived directly from laboratory testing, without reliance on assumptions or, necessarily, full understanding ofthe complicated physics of the bearing pad dynamics or film development. In general, such information is slightly conservative and very reliable.

Figure 4-12 shows a very large tilting pad thrust bearing with a leading edge groove and a built-in hydrostatic effect or high-pressure lift capability. The hydrostatic effect is useful for avoiding bearing pad wipes during start-ups, before a full hydrodynamic film has had the opportunity to develop. Basically, the effect is a very simple one: oil at a pressure sufficiently above atmospheric is fed into a pocket or cavity that has enough area exposed to the shaft or thrust collar surface that the static gauge pressure times the sum of all pocket areas is greater than the weight of the rotor system. Therefore, when the oil supply is put under pressure, the rotor lifts off ofthe stator, and an initial oil film is developed which prevents rubbing contact until a full hydrodynamic film develops. During operation, this lift (or some fraction of it if oil supply pressure is reduced) then adds margin to the load capacity of the bearing. For comparison, figure 4-13 shows a bearing also with leading edge grooves, but without hydrostatic lift, and figure 4-14 shows a plain shoe without either leading edge grooves or hydrostatic lift.

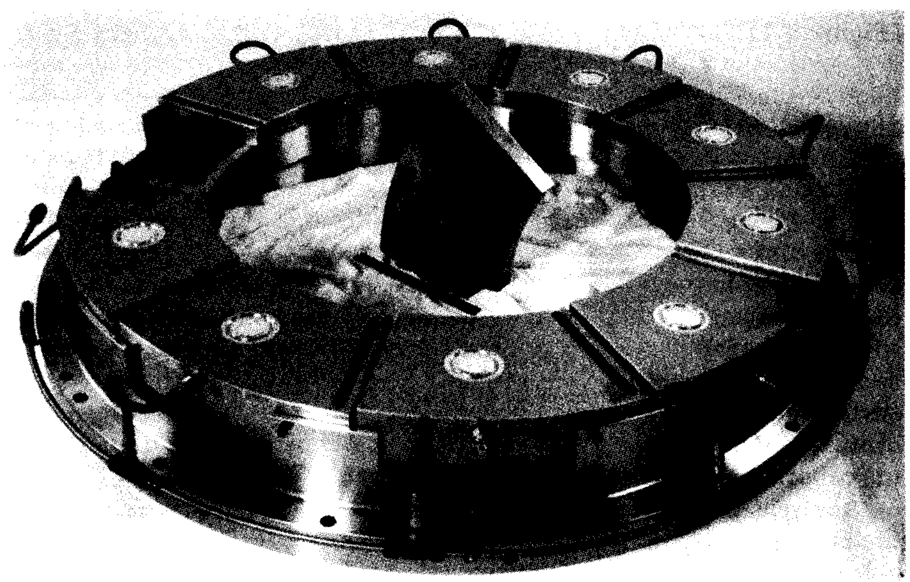


Figure 4-12. A 55-inch self-equalizing thrust bearing with leading edge groove and high-pressure lift shown with support pivot exposed. Courtesy Kingsbury, Inc.

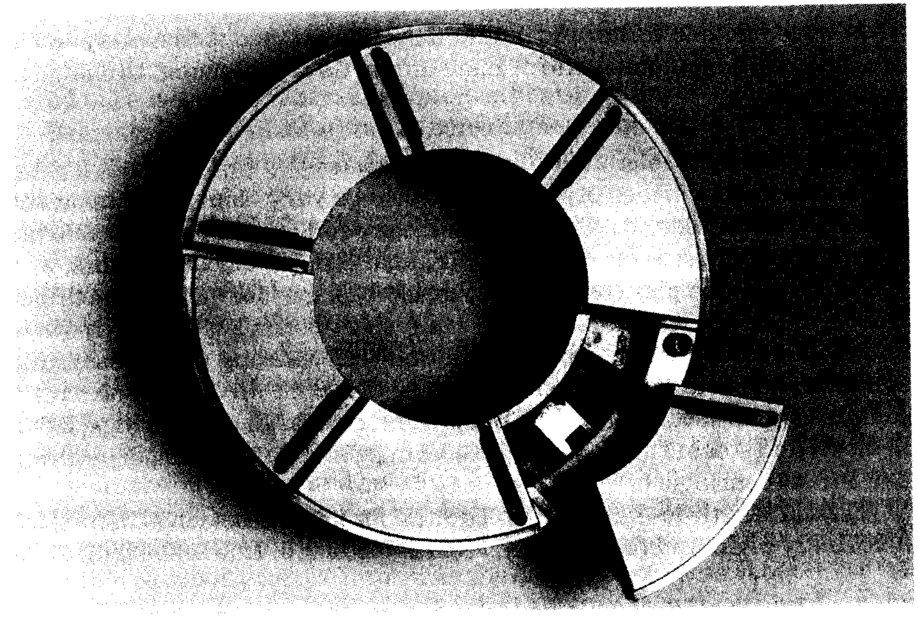


Figure 4-13. Six-shoe thrust bearing with LEG but without lift. Courtesy Kingsbury, Inc.

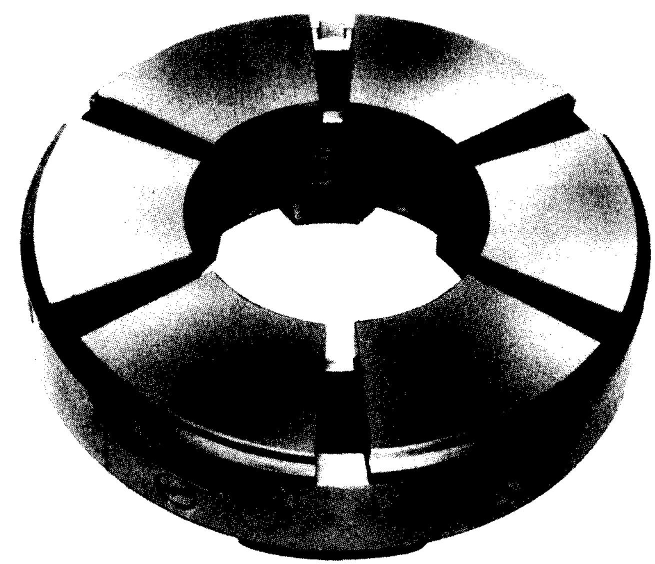


Figure 4-14. Six-shoe thrust bearing without LEG or lift. Courtesy Kingsbury, Inc.

For ease ofinstallation and maintenance, tilting pad thrust andjournal bearings can be combined into a single unit in its own housing, as shown in figure 4-15.

Journal Bearing Design and Application Issues

Allowable mean surface pressures in bearings vary, depending upon the bearing type and service. These pressures are roughly, in psi, 35 in marine lineshaft bearings; about 200 in centrifugal pumps, electric motors, and generators; 200 in plain or fixed pad thrust bearings for various machines and services; 250 in steatp. turbine main shaft bearings; 250 in flywheel and dynamometer bearings; 600 in diesel, reciprocating compressor, and steam engine crankshaft main bearings; 1,000 in diesel, reciprocating compressor, and steam engine crank pin and crosshead pin bearings; 2,000 in diesel and steam engine wrist pin bearings; and up to 2,200 psi in tilting pad thrust bearings.

Typicallength-to-diameter (L/D) ratio for journal bearings in modern machinery plain and fixed pad systems is 0.5 to 1.0. In lineshafting, L/D

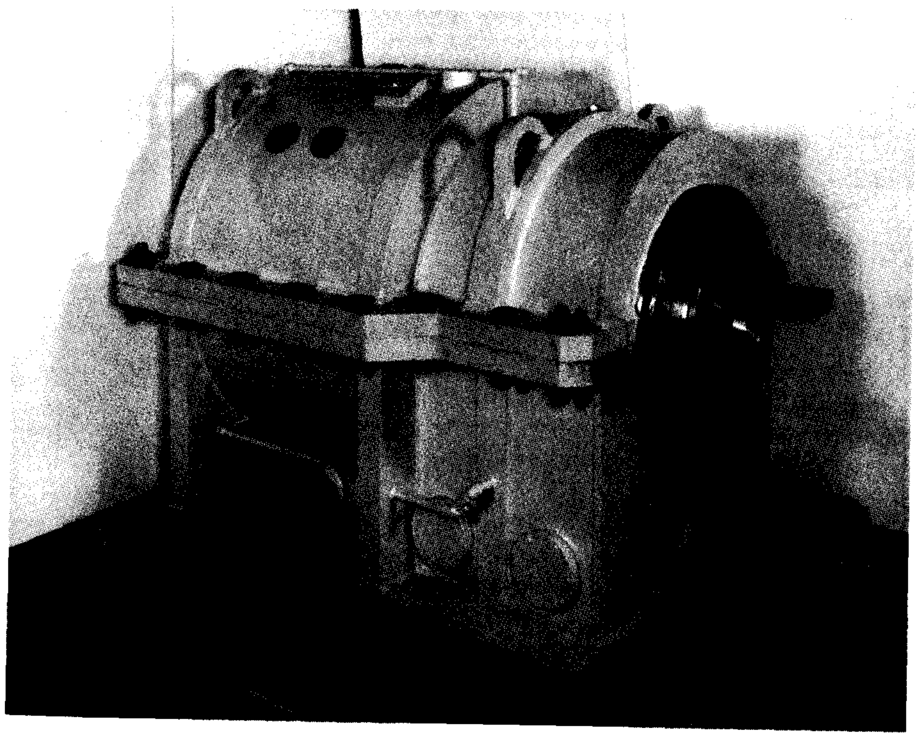


Figure 4-15. Combined 43.5″ thrust and 20″ journal bearing for DDG. Courtesy Kingsbury, Inc.

increases to 2 to 3.5, in order to better suppress vibration by restricting rotor angular tilt, and in order to discourage end leakage of the lubricating liquid once it is fed or entrained into the bearing film.

Bearing diametral clearance c (bore diameter minus journal diameter) is suggested by most manufacturers to be either 0.0015 inch per inch of diameter, or 0.002 inches plus 0.001 inches per inch of diameter D. For lineshaft bearings, these clearances often are approximately doubled.

Typical bearing shell materials, roughly in order of increasing life, are plastics, composites, steel, copper-lead, aluminum, babbitt overlays, and tin bronzes. Babbitt, or whitemetal, is a tin and lead base compound that has enough strength to support local bearing film pressures, but is also able to embed dirt particles that penetrate into the bearing film zone, to prevent them from scoring the shaft. This material is also tolerant of temporary rubs, in that its rate of increase in temperature during a rub is modest. When bronze or steel backed bearings are coated with babbitt, they are said to be babbitted. Such bearings are common for diameters of two inches or more (machined bronze is most common for smaller bearings). As discussed by Fuller, common practice is to make the backing thickness about 0.1 to 0.2 times the bore diameter, and to make the babbitt Ys inch thick plus X2 ofthe bore diameter. In the case of highly loaded bearings, however, the babbitt will tend to flow, and much thinner babbitt overlays are necessary.

During start-up and shutdown, and for very slow running speeds (such as when boiler feed pumps are put on turning gear), the journal film may break down. This is avoided in some machinery by static lift, where there is relatively high static pressure feed to the bottom of the journal prior to start-up, or during the slow speed event. This is done particularly in steam turbines. Be careful that such bottom feed holes do not become clogged or rotated, because if they do, a bearing rub during start-up is likely. The amount of oil pressure required to achieve static lift is roughly half the weight of the rotor divided by the feed pocket area under the shaft, i.e, pocket diameter squared times *pi* divided by 4, or pocket length times width for rectangular pockets. This concept is similar to the primary shaft support mechanism in hydrostatic bearings, as discussed later in this chapter.

ROLLING ELEMENT BEARINGS

Rolling element bearings consist of a set of spherical balls or cylindrical rollers (sometimes so thin that they are called *needles*) sandwiched between outer and inner metal *doughnuts* or *races*. Various examples ofthis style of bearing are shown in figure 4-16. Usually, the inner race rotates and the outer race is fixed to a grounding point, which typically is the machinery casing or housing. In thrust bearings, the races may be annular plates with the same inner and outer diameters, but more often thrust is carried by placing the rollers or balls in angular contact, so that there is still a definite inner and outer race. Often a *retainer* or *cage,* made of steel or an abrasion-resistant metal or plastic, is used to prevent roller-to-roller contact, to help keep worn rollers inside the races, and to aid in appropriate feeding oflubrication to the rolling element contact interfaces. Ifthe lubricant is too old or otherwise improper (including too viscous), retainer cages have been observed to develop a form of instability that leads to rapid deterioration of the bearing. Therefore, proper lubrication of rolling element bearings, particularly in higher-speed applications, should not be taken for granted.

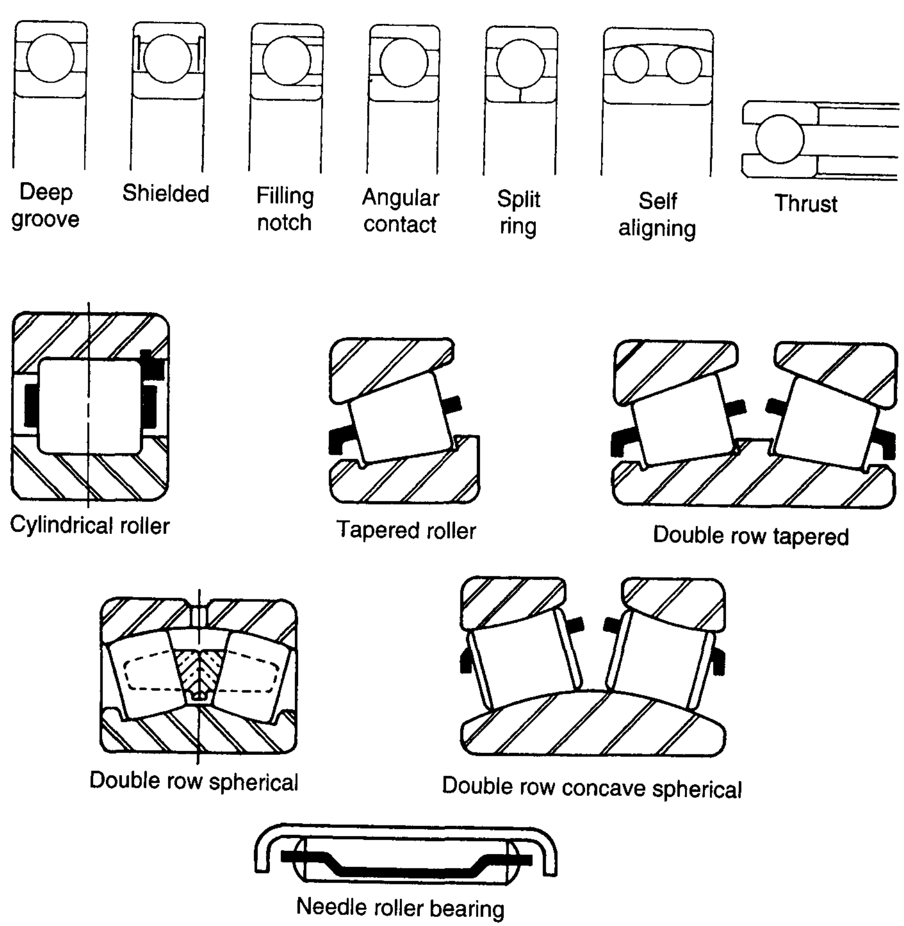


Figure 4-16. Rolling element bearings. Courtesy Society of Tribologists and Lubrication Engineers and E. R. Booser, editor, *Handbook of Lubrication and Tribology,* CRC Press.

To function properly for long periods of time, rolling element loaded contact materials must be very hard, to make them resistant to scratching and scoring that could induce fatigue crack growth. The races are commonly made of SAE 52100 steel, or some other low-carbon, highly hardenable steel like AISI M-l, M-I0, M-42, or M-50 tool steel, AISI 8720 or AISI 9310 high-strength, low-alloy steel, or AISI 440C stainless steel, hardened to about 60 Rockwell C. Ceramic balls and/or races have been experimented with, but as of this writing still seem to cause more problems than they cure, and are not in production machinery.

Ball bearings have theoretical point contact at each race, while cylindrical rollers maintain line contact. This means, for a given geometrical envelope, that roller bearings can support more load. However, the point contact of ball bearings allows them to accommodate considerable misalignment.

The basic load rating C of rolling element bearings is the constant radial load at which the average bearing of that type has a high probability of surviving for one million revolutions. Specifically, C is the load that the bearing can carry for 1 million inner race revolutions with 90 percent probability of no detectable surface failure. C is given in the bearing manufacturer catalogs for various bearing types and sizes. Generally, the load capacity C goes up with the number of balls and the bearing diameter, and goes down as contact angle (with zero being pure radial) goes up. The basic static load rating Co is similar but applies to the ability of the bearing at rest to resist permanent indentation of the races, or flattening of the balls or rollers. Co varies directly with the total number of rolling elements and with the square ofthe bearing diameter, and per ball and per inch squared diameter is about 1,000 for ball bearings and about 3,000 for roller bearings.

The rating life (BlO) of rolling element bearings is the number of hours that 90 percent of the bearings of a given type will run without failure (defined as absence of any detectable damage) at some given speed and given load. For reference, the BI0 life is generally about y.; the "50 percent probability offailure" life. If the life is quoted as LI0, it represents bearing life in millions of revolutions rather than hours. Life is equal to approximately the cube of the quantity C/P, where P is the combined steady plus dynamic (oscillating) load on the bearing. If the outer race rotates, or there is combined radial and significant axial thrust load, an equivalent load must be derived, which is bearing-specific and the derivation of which is explained in most bearing catalogs. Roller bearing life is more sensitive to C/P ratio than ball bearings are, as indicated by the following life rating equation:

$$L_{10} = (C/P)^p$$

where "p" is a bearing design-dependent exponent, equal to about 3 for ball bearings, and equal to about 3.3 for cylindrical roller bearings.

If free water is present in the oil, this can degrade the bearing life. According to Zaretsky, studies have shown that the effect of water roughly follows the equation:

$$\text{BID with water/BID without water} = (25/\text{ppm})^{0.6}$$

where ppm is the parts per million of the free water in the oil. Below a ppm of 25, no significant degradation is predicted in rolling element bearing life.

The basic load rating C is a function of bearing design and lubrication adequacy. Calculating C accurately from scratch is beyond the scope of this chapter, but for ball bearings it is proportional to the number of balls to the % power, and to the outer race diameter to the 1.8 power. For roller bearings, C is proportional to the number of rollers to the % power, and to the outer race diameter to the 1.07 power. Because typically much more than 1 million revolutions are required of a bearing in marine machinery, P is usually set significantly less than C. Loads are considered light when P is 7 percent or less of C, and are considered heavy when P is 15 percent or more of C.

A commonly used criterion that limits antifriction bearing use is the "DN" (bearing bore diameter in mm times the rotational speed in rpm) rating. For DN values less than 100,000, antifriction bearings work well with oil splash or grease lubrication. For DN values of 100,000 to 300,000, antifriction bearings can be used with directed oil lubrication or ring oil lubrication. For DN values of 300,000 to 1,500,000, rolling element bearings require caution in design and application, and preferably should be used with well filtered and dehydrated oil mist lubrication or directed jet lubrication, as well as more expensive, higher-tolerance bearing components (in some cases, lubricant is circulated through "tunnels" in the bearing race). If directed jets are used, at least two evenly spaced jets per bearing should be used, and the jet stream should be aimed at the clearance between the inner race and the cage. Exceeding a DN of 1,500,000 in antifriction bearings in marine machinery is not recommended, although special antifriction bearings have been used in experimental machinery at DN values up to 3,000,000, generally using directed jets and under-race forced convection passages. Such high-speed bearings may soon be used in marine gas turbines. In such bearings, it will be important that the design and upkeep of the oil scavenge system is excellent, so that no portion of the rotating elements is allowed to ever run submerged in oil, since this might lead to excessive oil heating and roller thermal shock.

In addition to DN requirements, *API Standard 610*, eighth edition, recommends that, at least for pumps, the multiplication of rated horsepower times rated speed in rpm should not exceed 2,000,000 in antifriction bearing applications.

In general, rolling element bearings can be lubricated with oil or grease, with the allowable bearing size or speed generally increased at least 50 percent with oil lubrication. Contrary to intuition, more rather than less grease is not necessarily better for the bearing. Once sufficient grease is in the bearing cavity, lubrication is taken care of and more grease simply results in shear heat dissipation, and may lead to overheating and lubricant breakdown in heavily loaded bearings.

Extreme care should be taken to prevent dirt and grit from entering into the bearing's lubricant. When installing or removing bearings to be reinstalled, never lay the bearing on the floor; place it on clean paper. Also, rolling element bearings are very susceptible to impact damage, so never drop them (replace them if this happens accidentally), and when driving them into their seat use a soft material like wood or bronze to interface directly with the bearing race. Also, when installing rolling element bearings, it is important not to place the balls or rollers in shear by forcing one race in a given direction while the other is locked or is driven in the opposite direction, or by jamming the retainer against the balls such that the races provide a counterforce. For example, if the outer race is to be driven into a housing bore with a slight press-fit, drive it in by placing load on the outer race only, not by placing pressure on the inner race or retainer.

Rolling element bearings are generally pressed on either the shaft or in the housing bore, and slip-fit on the opposite race. Alternatives to press-fitting are a c-ring or a nut clamp (often combined with a locking washer).

Ball Bearings

Ball bearings can generally operate at higher speeds and with more misalignment but with less load than cylindrical roller bearings. They are constructed in both radial and angular contact types. Radial ball bearings may support considerable lateral loading, but generally are intended to support only nominal axial thrust loading. Angular contact bearings are used when both substantial radial and axial loads are expected. Generally, however, angular contact bearings can take axial thrust in only one direction. If thrust can occur in either direction, then opposite sense angular contact bearings are mounted together, and in this case are called a duplex or back-to-back configuration.

Radial ball bearings. The most common type of radial ball bearing is the "Conrad" type, where typically 5 to 9 balls are used. For higher load capacities, "filler" races or rings have notches on one shoulder to allow the insertion of more balls. Sometimes, the inner or outer ring is split, so that an even greater number of balls can be inserted and fully trapped upon final bearing assembly. In such a design, a proper means of holding the halves together must be included in the design.

Radial ball bearings can tolerate greater misalignment than radial roller bearings, especially if they have a deep groove in the races, and in this case can take significant thrust as well as radial loads. Sometimes double rows of radially loaded balls can be used to increase radial load carrying capability further. Self-aligning ball bearing designs have an outer race radius perpendicular to the rolling direction that is larger than the radius ofthe balls, and an inner radius with two separate grooves for a double ball row.

Ball thrust bearings. Ball thrust bearings are the axial thrust counterpart of radial ball bearings. They can be either angular contact or, for greater loads, pure thrust carrying. They generally have 90-degree contact angles between the ball and the race, and can be mutiple direction, and multiple row.

Angular contact ball bearings, and duplex bearings. Angular contact ball bearings can take on greater axial thrust, misalignment, and shaft speed than straight radial or thrust bearings, but do so at some loss ofradial or thrust load support capability, respectively. However, the thrust capability of a single angular contact ball bearing is unidirectional.

Duplex or back-to-back bearings are essentially two angular contact bearings facing opposite directions, with their races loaded against each other, so that thrust can be taken in either direction, and so that radial load capability is at least doubled.

Cylindrical and Tapered Conical Roller Bearings

Roller bearings can take much greater load than similarly sized ball bearings, because they support the load with line rather than point contact. However, the amount of axial thrust that they can handle safely is generally low unless they are of the tapered variety, and they cannot tolerate large amounts of misalignment. Retainers or cages are invariably used to space the rollers. Sliding occurs between the retainers and the rollers, and this eventually leads to wear particle formation. The manufacturer-specified lubricant should generally be used, because it was selected with the minimization of this wear taken into account.

Straight roller bearings. Straight roller bearings consist of rolling elements that are cylinders (solid or, in high performance applications, possibly hollow) with rounded edges at each end. The rounded edges relieve the concentration ofload that would otherwise occur at the roller ends and the corresponding location on the races, resulting in premature fatigue. Any cylindrical roller bearings that develop sharp corners at the ends of the rollers have been operating with too much thrust or insufficient lubrication. In such cases failure is likely imminent; the bearings should be replacedimmediately and the lubricant flushed and replaced.

Tapered roller bearings. Tapered or conical roller bearings consist of clipped cones that usually have inner and outer races that are both conical on the side adjacent to the rollers. Tapered conical roller bearings have tapered rollers or cones trapped between the angled walls ofthe two races and can support relatively large combinations of radial as well as thrust load, separately or simultaneously. The ends of the rollers tend to rub against the race, and it is important to grease this zone well when replacing this style bearing, or wear flakes might be created quickly prior to the flow of grease into all the crevices ofthe bearing during operation. The inner and outer diameter of the bearing assembly is generally machined to be a cylinder to interface with the housing bore and shaft diameter.

Spherical rollers. Spherical rollers are a variation of the cylindrical roller and have roller surfaces that are not straight like the cylindrical or conical rollers, but rather follow a contour that is roughly parabolic. These bearings can tolerate significant axial thrust as well as radial load.

Bearing Mechanical and Support Components

Rolling element bearings are sometimes fit into a cylindrical cavity machined into a bearing housing. Another common method of securing them is to have a self-contained small housing, or "pillow block" support, that is designed to attach firmly to some structure that is intended to be fixed relative to the shaft supported by the bearing. Such pillow blocks are normally lubricated with grease.

It is important in many applications to install and maintain a certain amount of preload in ball bearings. This helps to prevent the rolling elements from sliding or skidding in preference to rolling during start-ups or operational points where the bearing load is light. Skidding causes extra noise, which is a problem in military marine applications, and can cause wear "flats" on the balls and races, significantly reducing bearing life. Preload can be achieved by maintenance of very tight manufacturing tolerances, press-fits, and operating temperatures. In such cases, extra care must be taken during installation to ensure that the bearing is fully seated, and that no dirt is trapped between the race face and its housing seat. In installations where preload is particularly critical, belleville (thin cone) or wave washers can be used between an opposing piece and whichever race is slip-fit onto its attached component. In such cases, the belleville should never be assembled so that it is squashed flat, so that it is not under compression, or so that it pushes both members that it is loaded between against their seats, or the benefit ofthe self-adjusting load provided by the belleville will be lost. The only counterforce to the belleville washer should be the bearing ball, sandwiched between the races by the limited belleville force.

Rolling element bearings typically have higher support stiffness than journal bearings, but have much less (in fact, generally negligible) damping. They also have minimal cross-coupled stiffness, as discussed for journal bearings above, thus discouraging large unstable motion of the rotor in certain circumstances, as shown later in this chapter. In rotor systems where too much vibration is occurring, some damping can be added to the zone of the bearing by providing an oil film support around the bearing outer race. Typically, this is done by some form of "squeeze film damper," where the bearing outer race, or some cylinder within which the race is installed, has a very small clearance gap machined between it and the housing bore.

This small cavity of the squeeze film damper is flooded with oil, and o-rings or c-rings are installed on each end of the cavity to keep the oil in place, or at least to prevent it from being squeezed out quickly. Then as the shaft dynamically loads the bearing, the load is transferred to the outer race and forces a squeezing action between the race and the housing, through the oil film. The trapped oil flows around the cavity circumference to allow the race to move in response to the dynamic force. In order for the oil to do this, of course, some of the oil in the closing gap must move out of the gap. In order for it and the neighboring oil film to move, it is necessary for the oil along the sides away from the squeezed gap to move much faster than the gap is closing, because of leverage caused by the projected area of the shaft sweeping the oil out of the way, versus the very thin side clearance through which this oil must move. This is known as the *Stokes effect,* and because of the large accelerations it causes in the gap lubricant, it can result in large dissipative (i.e., energy absorbing) forces resisting the shaft motion, which is how the squeeze film damper works. If the good seal is lost from the cavity axial ends, however, the squeezed fluid merely shoots out these ends, and little squeeze film effect occurs. Therefore, in turbines and compressors which employ squeeze film damper effect, make certain that the damper end seals are functioning properly and are well-maintained, or high vibration levels and damaged bearings may result.

Depending upon bearing type and service, different shaft attachment fits and tolerances will be applied. These are listed in the bearing catalog, with the fits reiterated (and possibly modified slightly) by the equipment manufacturer in the installation and maintenance manual. Standard internal bearing tolerances are given in tables for ABEC ball bearings and RBEC roller bearings, which, for those interested, are reprinted in various bearing catalogs and bearing handbooks (such as CRC's *Handbook of Lubrication and Tribology* listed in the references).

IMPREGNATED BRONZE AND NONMETALLIC SLEEVE BEARINGS

Porous bronze cylinders can be *sintered* (a process by which powder particles are cemented together at high temperature to form a bulk component) from bronze powder. If oil or some solid lubricant is forced into the pores under high pressure, this small amount of oil is able to provide excellent, replenished lubrication of bronze/steel shaft interfaces if the speed is low to moderate and if the load across the interface is relatively low. This form of bearing is common in smaller machine slider bearings, and in small electric motor bearings. It is an economical bearing to install and operate in situations benign enough to tolerate them. Once such bearings are worn to the point that they become noisy or overheated, or do not provide adequate shaft centering, they must be replaced as soon as possible, since they cannot be effectively externally relubricated. Also, once their clearances begin to open significantly, the whirling of the machine rotor that is allowed increases the load well beyond the tolerable levels for such bearings, so the situation degrades rapidly.

Particularly in recent years, nonmetallic materials have been used for sleeve bearings in certain applications. Such materials are typically rated in terms of their maximum operating temperature, maximum load capacity (in terms of load per unit area), and so-called PV rating, which is operating load per unit bearing area, times the surface speed of the bearing. Table 4-1 lists these parameters for common nonmetallic and porous/lubricant-impregnated metals used in unlubricated sleeve bearings.

TABLE 4-1

Limiting Factors for Sleeve Bearing Materials

Material	*Max Temp. (F)*	*Load Capacity (psi)*	*PV Rating (psi-in /s)*
Polyimides	500	2,500	22,000
Bronze	260	2,500	10,000
Iron	260	3,700	6,000
PTFE Fabric	480	58,000	5,000
Filled PTFE	480	2,500	3,000
Carbon-graphite	750	600	3,000
Wood	160	2,500	2,400
Phenolics	250	5,900	1,000
Acetal	210	2,500	600
Nylon	190	2,500	600
Unfilled PTFE	480	400	200

Along with the above constraints, the PTFE materials are also limited to surface sliding speeds of less than 1 meter/second, while the carbon, wood, and phenolics are limited to no more than about 10 *mis,* with the other materials averaging 3 to 6 *mls* maximum surface sliding speed. Determine surface sliding speed by multiplying the shaft diameter in inches by *pi,* and multiplying by shaft rpm divided by 60.

EXTERNALLY PRESSURIZED BEARINGS (INCLUDING MAGNETIC BEARINGS)

Externally energized bearings do not derive their reactive force from internal bearing fluid dynamic action, but instead operate through forces provided by a pressure or electrical source outside of the bearing shell. This includes hydrostatic bearings, in which cavities surrounding the shaft are pressurized by a line running to the pump discharge or to an independent pump. The pressure in the cavities is in general lower than the original supply pressure because of a valve or orifice restriction between the pressure inlet line and the cavity, together with continuous leakage over the cavity walls to the bearing environment. In hydrostatic bearings, as the shaft moves off center, the clearance between the shaft surface and the cavity walls closes in the direction of shaft motion and opens up on the other side. The external pressure-fed cavities on the closing clearance side increase in pressure due to decreased leakage from the cavity through the clearance, and the opposite happens on the other side. This leads to a reaction force that tends to keep the shaft centered, and that provides significant energy absorbing damping without potentially harmful cross-coupling, thus discouraging rotor instability, as discussed above. Unfortunately, the hydrostatic bearing's continuous leakage rate, needed for the bearing to function, is generally large and difficult to control as the bearing wears, and this bearing therefore consumes too much power to be practical in most machines.

Hitachi and Demag Delaval were among the first manufacturers to produce large turbomachines that have shafts supported entirely by a magnetic field. Such "active" magnetic bearings are actually feedback control devices, in which the shaft position is continuously monitored, and more or less current is fed into opposing magnets to cause an off-center shaft to shift back toward its centered position. In principle, such bearings can compensate for relatively large amounts of imbalance and shaft misalignment, and allow safe operation of rotors with significantly bowed shafts. Surprisingly, they also take up little more space than tilting pad bearings, and consume less total operational power than fluid film bearings do. They are not yet a bearing of choice, however, because of lack of user experience with such devices and the relatively high cost and complexity (eventually cost and complexity should be similar to that of variable speed electric motors comparable in size to the bearing coils). In addition, the existing backup bearing designs (needed to provide emergency rotor support if the magnetic controller fails or electrical power is lost) have difficulty reliably preventing rotor damage during rotor overloads (e.g., from rapid hull movements during storms or warfare conditions) or emergency coast-downs. Advancements in bearing control methods and in solid lubricated short-life bearing design are expected to resolve this problem, however.

Magnetic bearings are unlikely to be found in marine applications, other than experimental ones, at the time of publication of this edition of the manual. This situation may rapidly change, however, in the next few years. The British, French, and United States navies have been experimenting with magnetic support of noncritical machinery for about fifteen years, with increasing success. There are many advantages to magnetic bearings:

- elimination of the lubrication system, including its associated oil purification and reclamation systems
- allowance of (theoretically) very high operating speeds, with negligible bearing power loss and avoidance of critical speed resonances and rotordynamic instabilities
- reduction of noise and vibration; in advanced systems, noise and vibration may even be partially canceled by proper phasing of feedback response
- tolerance of, and compensation for, significant imbalance and misalignment, and even rapid seismic events such as in storms or in military activity
- built-in condition monitoring and predictive maintenance capability

Reasons magnetic bearings are not yet used include the following:

- lack of reliability, as guaranteed by other similar successful services, or by a redundant and fail-safe control system design
- a poor record for the ability of backup bearings to allow the rotor to coast safely to a stop in case of loss of magnetic support
- less load carrying capability for a bearing of given diameter and length, versus fluid film and rolling element bearings
- practical speed limitations due to the time constant of the control system and/or the large rotor-supporting magnets
- uncertain maintenance of magnetic support in case of shock loads in military service or in storms
- inability to withstand unexpectedly high fluid forces due to surge or stall, particularly at part load operation
- the existing control and current amplification systems are large, bulky, and expensive to build and maintain

Although the problems yet to be overcome by magnetic bearings are formidable, recent improvements have been dramatic, and it is expected that the first system commercially practical for marine usage is not far from being introduced. Digital control electronics and algorithms are making controls and amplifiers more compact, less expensive, faster acting, and more reliable. Solid lubricated or impregnated product lubricated bearings are being improved so that they can be counted on for back-up when needed in case of temporarily high loads, shocks, or electrical outages. However, successful operation of magnetic bearings in laboratory equipment, and in power plant and refinery applications, is required before widespread marine application would be sensible.

PROCESS FLUID- ("PRODUCT-") LUBRICATED BEARINGS

Because of the savings in cost, complexity, and number of components needing maintenance, wherepossible, manufacturers use bearings that do not need lubricating oil and the accompanying filtration, storage, water separation, and recirculation systems. Such bearings can be either without obvious means oflubrication, such as the air bearings shown in figure 4-17, or impregnated bronze slider bearings, as discussed in the sleeve bearing section above. They also can be designed to survive on lubrication by the pumped product, such as water, seawater, or compressed gas. In none of these cases can a bearing of given size support a significant percentage of the load that an oil or grease lubricated bearing can, but when loads and/or surface speeds are low, the reduced first cost and operating cost of these bearings make them an attractive selection.

Process bearings are used for certain pumps, particularly for vertical pumps where the bearing loads do not need to contend with gravity loads

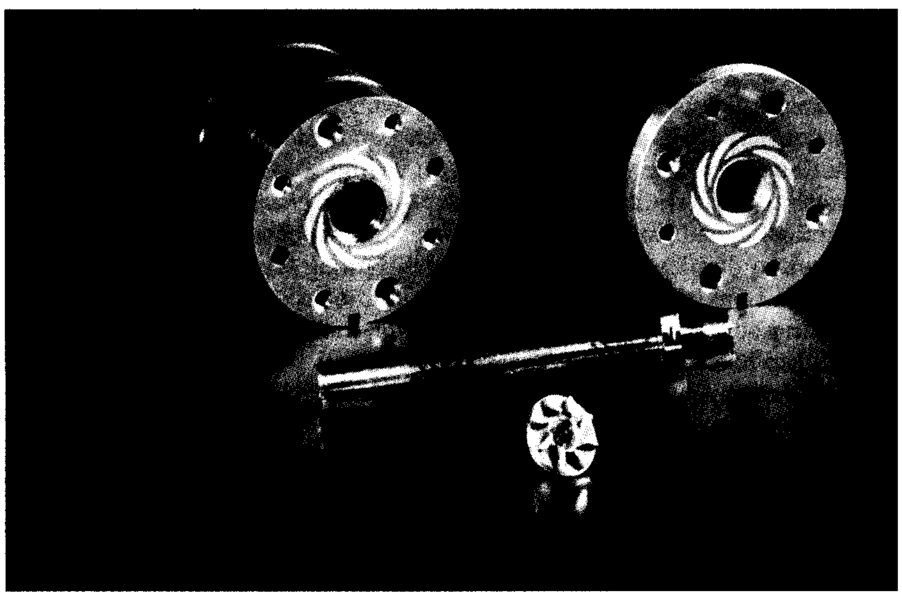


Figure 4-17. Air bearings. Courtesy Society of Tribologists and Lubrication Engineers.

on the rotor, as discussed in the journal bearing section above, relative to Cutless rubber bearings for lineshaft applications. Both deepwell-type and submersible cargo pumps commonly use water or cargo lubrication for bearings ofthis or similar types. This keeps the pumps less expensive and easier to maintain in principle, but problems can occur if bearing insert materials are used which are incompatible with the pumped product, or if the product becomes contaminated. One problem seen in marine service includes bearing materials that swell when exposed to certain cargos (or even water), causing them to tighten bearing clearances when the inserts react against their outer walls.

Another marine cargo service problem affecting process-lubricated bearings is insufficient cleaning of blasting grit or other abrasives from the cargo bay floor or walls after blast cleaning ofthe cargo bay, which allows such abrasives to collect around or embed within bearing walls. Product-lubricated bearings walk a thin line in both these regards. Some reactivity ofthe bearing material with the product allows the product to better lubricate the bearing (i.e., adhesion and *lubricity* is increased). Also, embedability is generally a desirable product-lubricated bearing characteristic, since it allows foreign bodies to be "captured" and "buried" before they can score the shaft. However, if such foreign bodies continue to spawn fresh cutting surfaces (such as some blasting grit is designed to do), the bearing inner surface can be gradually turned into a grinding wheel. If such problems occur, obtain advice from the manufacturer concerning a change of materials, or (in the case of chronically embedded grit and resulting shaft scoring) improve the cargo bay cleaning procedures.

Ceramic shells or pads made out of materials such as silicon carbide (SiC) are not unusual in product lubricated bearings for newer large equipment. An SiC bore or pads run against an SiC rotor journal, for example as shown in figure 4-8, is able to run with filtered water or seawater as the lubricant. Alternatives to ceramic are certain plastics (e.g., glass or bronze-filled teflon), rubber, or carbon composites, in situations requiring product lubrication where continuous loads are not high (under bearing pressures of 100 pounds per square inch of bearing diameter times length). The bearing materials discussed above under dry-lubricated sleeve bearings generally can also be applied as product-lubricated bearings, and generally with substantially increased PV rating and maximum surface speeds because of the cooling and at least partial lubrication provided by the product. Use of product-lubricated bearings saves the bulk, expense, and upkeep of a lubrication system and generally avoids bearing seals, degradation of which has been noted in most services as the most common cause of bearing failure. In addition, the elimination of bearing oil means that there is little issue of contamination, and no requirement for disposal of used oil.

A drawback to product-lubricated bearings is that they cannot run dry and unlubricated for more than several seconds to perhaps one minute, without serious degradation of the bearing running surfaces and clearances. At low loads, carbon composite and some plastic composite bearings are an exception to this, in which a PV rating (bearing pressure in psi times rotor journal surface speeds in inches per second) of up to about 25,000 has been reported, although 3,000 to 15,000 is more typical, as shown in table 4-1 on page 4-31. Although lack of dry running capability is a serious liability for ceramics in some applications, they do have the advantage of not swelling when exposed to water for long periods, or being susceptible to biological attack, each of which can be serious problems for the other product-lubricated beadng material types. Because of hydroscopic and other swelling issues (which cause the bearing bore to shrink since the outer diameter is usually constrained from growing outwards), diametral clearances must usually be set larger on rubber and some carbon and plastic bearings, such that 5 to 10 mils per inch of diameter might be required. Ceramics can also withstand higher temperatures (to 1,800°F), while carbon is limited to about 750°F, and plastic composites and rubbers are limited to 250° to 450°F, depending upon material specifics.

Some product-lubricated bearings rely on product flow into the bearing central region through grooves on their inner surface. Sometimes these grooves are spiral to convert fluid rotation near the shaft into static pressure at the groove inlet, causing a pumping action that propels lubricating liquid deep into the bearing. If the grooves fill in with deposits, this lubricant pumping and flow path is reduced, and the load carrying capabilities of the bearings can diminish substantially. This may lead to significant heating of the lubricant that migrates to the bearing center, and eventually to a large temperature rise in the bearing material. The thermal expansion coefficient ofthe bearing material then causes it to try to swell, but being constrained on the outer diameter, the increase in material volume results in a closure ofthe bearing bore. This reduced clearance encourages the process oflubricant flow decrease and temperature increase, and these effects therefore can spiral into rapid bearing wear or shaft seizure .

PUMP ANNULAR SEAL BEARING EFFECTS

Annular seals in pumps and hydraulic turbines, such as wear rings, balance drums, and interstage seals, can provide extra bearing support. Besides the extra load capacity that this provides, it can dramatically affect dynamics by changing the rotor support stiffness and therefore the rotor natural frequencies. This can either avoid or induce possible resonance between strong forcing frequencies at one and two times the running speed and one of the lower natural frequencies, as discussed later.

The stiffness and damping of an annular seal is provided in small part by the squeeze-film and hydrodynamic wedge effects discussed above. However, because of the high ratio of axial to circumferential flow rates in annular liquid seals relative to bearings, large forces can develop in the annular clearance space due to the circumferentially varying Bernoulli pressure drop induced by changes in axial leakage as rotor eccentricity develops. This is known as *Lomakin effect,* and is the largest stiffness and damping force generating mechanism within pump annular seals.

Lomakin effect is roughly proportional to the reciprocal ofthe square of the wear ring clearance. The physical reason for the strong influence of clearance is that it gives the opportunity for the circumferential pressure distribution, which is behind the Lomakin effect, to diminish through circumferential flow. Any annular seal circumferential cavity, which includes grooving, has the same effect as increased clearance, at least to some degree. Deep grooves are worse than shallow ones in this regard, and they remove most of the Lomakin effect.

AIR LUBRICATION

Air or other gases can be used as a lubricant in high-speed machinery, such as turbochargers or cryogenic compressors and expanders (see fig. 4-17). The advantages are elimination of the lubrication system and its maintenance and improved machine efficiency due to lower frictional losses in the bearings. The disadvantages are the following:

- because of the relatively high ratio of cross-coupled stiffness to damping in this style bearing, the susceptibility ofthe rotor systems operating in such bearings to rotordynamic instability (unless clearances and machinery loads, e.g., around impellers, are kept within narrow bounds) makes the assembly difficult to troubleshoot and maintain by anyone but the manufacturer
- the very close clearances required between the bearings and the shaft and housing, with tolerances often in the tenths of thousandths of an inch
- the low load capacity of this style of bearing

The use of air bearings has been progressing steadily for the last forty years, but remains a technology that has a dubious place in marine applications where access to the manufacturer's field service personnel may be practically limited. The use of machinery employing other bearing types is currently recommended.

BEARING LUBRICANTS AND LUBRICATION SYSTEMS

Lubrication is a process by which a film oflubricant is placed between bearing surfaces for the purpose of controlling friction and wear. Such films are designed to minimize contact between the surfaces by adhering tenaciously to them and having sufficient viscosity to resist being squeezed out of potential contact zones (viscosity can be thought of as resistance to flow), so as to prevent wear. However, the viscosity must not be so great as to prevent easy shearing of the lubricant, since this allows the frictional force opposing the movement of one surface across the other to be low.

Lubrication is part of the science of tribology, whose root comes from the Greek word tribos, meaning "to rub." The lubrication system, the lubricant, and all the bearings, seals, and related moving machinery components are considered a tribological system. All of the elements of the tribological system are selected by the equipment builder to suit the particular machine application and operating conditions. For best service, therefore, the equipment builder's lubricant and lubrication recommendations should be followed, unless there are very good reasons to do otherwise, such as chronic failures, or a condition of service not foreseen by the manufacturer.

Lubrication systems include the lubricant, its reservoir, and (depending on the system) pumps, piping, and other means of delivering lubricant to the bearing, and devices such as filters, coolers, centrifuges, and air and water separators that maintain the quality of the lubricant. These systems can range from very simple, consisting of only a grease gun fitting on the bearing housing and the grease gun holding the grease, to the complex circulating and control systems found on steam and gas turbines and boiler feed pumps.

Bearing lubricants can be liquid, such as oil, or solid, such as graphite, moly disulfide, or PTFE, or a grease with both solid and liquid characteristics. Fluids that are used as lubricants must have the best range of adhesive and viscous characteristics to satisfy the compromise of the need for viscous separation of rotating parts versus low enough viscosity to prevent excess frictional heat generation. In most applications, the best lubricants are naturally occurring mineral oils, usually hydrocarbon-based, or man-made synthetic oils, often silicon-based. Synthetic oils are often engineered to have superior oxidation, water resistance, or thermal properties, although at substantially higher cost than their natural counterparts. They may be superior in all three respects, but often some compromise has been made in one quality to the benefit of the others, and it should not be assumed that a synthetic oil is automatically better than a mineral oil for a given marine application.

Liquid lubricants can be specially designed for the sole purpose of lubrication, such as mineral and synthetic oils, or can be whatever is conveniently available, such as the product being pumped (often water or seawater). Obviously, in the latter case, compromises must be made by the designer, since the lubricating qualities of whatever is being pumped are unlikely to be ideal, although there is the benefit that the use of product lubrication avoids the need for many components that are costly and require maintenance, such as filters, reservoirs, oil coolers, and oil/product seals.

Modern lubricants are complex, sophisticated elements of tribological systems that must be cared for as carefully as the bearings themselves. They must be kept clean, cool, and contaminant free. A lubricant has many simultaneous jobs to perform, in that it controls friction and wear in the system and, depending upon the application, often is a heat transfer medium, a protection against rust and corrosion, a sealing medium, and a scavenger for rubbing interface contaminants.

Key Lubricant Properties

The main advantage that lubrication by oil and grease has over liquid product or gaseous product lubrication is high viscosity, even after the bearing temperature reaches steady-state. This allows a thicker hydrodynamic wedge to separate the moving parts, producing a thick film that is less likely to allow surface-to-surface contact, or damage by contaminating particles that are below the filtration limit. However, too high a viscosity can cause more heat to be generated in a bearing than can be conducted away by the lubricant flow or by heat transfer. To some extent, this is self-limiting, because lubricant viscosity is, in general, highly temperature dependent and decreases very rapidly as temperature increases. In addition, heat transfer out of the bearing occurs more quickly as temperature rises. Nevertheless, choosing a lubricant with too high a viscosity for a given application will lead to unnecessary additional power consumption, and, due to the resulting temperature rise, will cause more rapid deterioration of the lubricant by oxidation and other forms of temperature-dependent chemical breakdown. Like viscosity, oxidation resistance also decreases much more rapidly than temperature increases. Extra thermal growth from higher temperatures than the manufacturer anticipated can also lead to critical increases in bearing loads by changing clearances, or, in the extreme, even to the elimination of all clearance, causing the shaft to seize.

Oils and greases also possess lubricity, which essentially represents the adhesion of the oil to the bearing surfaces, keeping friction and scuffing low during start-up or under heavy loads, during which some metal-to-metal contact occurs across rolling and sliding interfaces, such as in gear teeth and heavily loaded ball and roller bearings.

In order to preserve their beneficial viscosity and lubricity characteristics, oil and grease lubricants should be resistant to forming new compounds or emulsions, which can also clog feed holes and grooves with sludge deposits. This implies that when water enters the lubrication system through a leaky seal or condensation on the reservoir tank walls, for example, that it separates or *demulsifies* from the oil reasonably quickly. Appropriate oils for most marine applications have good demulsification properties.

Another important factor in preserving lubricant properties is oxidation resistance. If bearing lubricant temperature is kept below 160°F exit temperature, this generally is not a major problem. However, in highly loaded or high-speed machinery, or machinery that gets very hot due to the operational processes of the machine, local oil temperatures within the bearings can increase to beyond 200°F. For many oils and greases, for each 18°F (lOOC) rise beyond this temperature, the oxidation life of the oil is cut consecutively in half. Fortunately, when this cannot be avoided synthetic oils are designed to resist oxidation better, so that they can operate at temperatures to over 100 degrees hotter than this before they reach comparable rates of degradation.

An often forgotten purpose of all forms ofliquid lubrication is to remove heat from the loaded zone of relative motion. Shear heating in this zone needs to be continuously removed, and higher loads or speeds require faster heat removal. Roughly speaking, in highly loaded bearings the heat removal is proportional to the lubricant flow rate. In forced-oil recirculation lubrication systems, this typically leads to about a 30° to 40°F rise in oil temperature as it passes through the bearing. In solid-lubricated or unlubricated contacts, frictional heat is removed almost entirely by metal conduction, and this does not allow very high combined loads and speeds. Magnetic bearings avoid the need for frictional heat removal by eliminating shear at the running clearances, but the electrical heat dissipation of the coils must still be removed from the bearing housings in some manner; in fact, sometimes liquid cooling has been used for this. Liquid cooling or forced convection cooling (using fans) is sometimes used to aid the lubricant flow in removing heat from the bearing zone. In the latter case, the bearing housings should be kept clean of significant dirt and grease buildup, since this can dramatically lower the rate of heat transfer from the bearing housing to the environment from that counted on by the designer.

Journal and other forms of fluid film bearings generally need total or near-total immersion in the lubricant. However, a common mistake made in providing lubrication in rolling element bearing systems is to provide too much lubricant. This provides more lubricant than is necessary to prevent metal-to-metal contact between the bearing parts in relative motion, and has the liability of the wasted lubricant being churned needlessly, creating significantly higher heat generation, perhaps enough to reduce the life of the lubricant, and possibly enough to damage the bearing directly. Grease-packed systems should not have the cavity around the bearing packed more than roughly one-third full. Oil-lubricated rolling element bearings should run with only the lower quarter of the bearing immersed in oil (i.e., up to the middle of the bottom ball or roller), or (in systems so designed) free of immersion altogether as oil is dripped, misted, or sprayed onto the bearing rollers instead, particularly in very-high-speed applications.

A property of oil sometimes not considered until it becomes a maintenance headache is the *pour point.* This is the temperature below which the

oil becomes so viscous that it can be considered semi-solid. If equipment is exposed to low temperatures, particularly below freezing, this can make it difficult to refill the lubrication system. More importantly, however, unless the equipment is in continuous service, when it is started cold, the start-up torque can be excessive, and components might become overloaded. In a circulating oil system, the highly viscous oil can also effectively block fresh oil from replenishing oil at the bearing at a proper rate.

In summary, the key issues in maintaining appropriate lubricant properties are to keep the lubricant clean, dry, and cool. In addition, the load must be kept at a reasonable level, such that *lambda,* the film thickness at the minimum film location divided by the peak-to-peak roughness of the opposing surfaces, is equal to 2.5 or higher. Lower values of *lambda,* caused by insufficient lubricant viscosity or excessive load, can lead to surface scuffing and undue heating of the lubricant. For each 18°F (lO°C) temperature rise, typical lubrication oils double their oxidation rate and reduce viscosity by a factor of about two. Thus above-normal temperature excessive-load problems are generally cumulative.

Engineered Lubricant Types and Properties

GREASE

The National Lubricating Grease Institute (NLGI) defines grease as "a solid to semi-fluid product of dispersion of a thickening agent in a liquid lubricant." The primary advantage to grease versus oil is that it "stays put" (more or less). Grease is generally used to lubricate bearings where it would be difficult to keep oil in place or where oil might too easily admit foreign particles. In general, if it is possible to use oil to lubricate a bearing, this is to be preferred over use of grease because of the ability of oil to replenish the close clearance area with fresh lubricant and to carry away excess heat, contaminants, and wear debris. However, recent strides in the design oflong-Iasting greases with temperature and contamination tolerance have made it possible to use greases at speeds of up to 80 percent, the maximum speeds able to be handled by circulated oil lubrication systems.

Grease is most commonly formed by combining a soap base made from fat with a mineral oil, agitated into a homogeneous mixture. Various soap bases have advantages and disadvantages. Calcium or lime-base grease will not absorb or emulsify water, but has a relatively low melting point. Sodium or soda-base grease becomes easily contaminated by water, but will offer good rust protection until it is washed away, and has a higher melting point than calcium-base grease. Aluminum grease has good water resistance, moderate temperature capability, and capacity for being manufactured with very-high-temperature capability. Lithium-base greases, which are currently the most commonly used (about 70 percent of the market), are adequate in water resistance, and have a high-temperature capability. There are also clay and polyurea greases with excellent high-temperature capabilities, but these tend to be expensive and have characteristics that are highly variable depending upon their additives.

Each of the above bases or thickeners shares one characteristic that makes it able to form grease, however, and that is that oil is soluble in it. Sometimes stored grease separates out some of its oil, in a process known as *bleeding*. In moderate cases the oil may simply be stirred back into the grease, where it will be reabsorbed by the base stock. Certain greases bleed at much lower temperatures than others and may even become chemically unstable, degrading their ability to lubricate. Calcium greases are generally suitable up to about 160°F, and sodium greases up to about 250°F. Lithium and aluminum greases are able to function well up to somewhat over 300°F, while certain synthetic greases lubricate and survive satisfactorily long term to over 400°F.

In addition to the oil and the thickener, most modern greases include important additives. The most common additives include:

- oxidation, corrosion, and rust inhibitors
- metal passivators
- dyes and pigments
- fillers and tackiness agents
- EP (extreme pressure) and anti wear agents

A measurable property of a grease's stiffness is its so-called *consistency*. The NLGI lists ten fluid and semi-fluid consistency grades, based on the "worked penetration" into the grease of a standard cone dropped from a fixed height in an ASTM test. The higher the grade (listed from 000, 00, and 0 to 6), the lower the penetration, and therefore the greater the stiffness or consistency. Typical greases specified for marine application are of NLGI No.2 consistency, with a rust inhibitor as a minimum recommended additive. Also, for a reasonable compromise of water and temperature resistance, lithium greases are typically used.

Although greases are only semi-fluid and hence do not have viscosity, or flow resistance, in the Newtonian sense, they can be considered to have an apparent viscosity at a given rate of shear. This results in a given frictional force and frictional heating rate, for a given relative velocity divided by spacing of two opposing surfaces. The apparent viscosity of a grease is much higher than that of a liquid oil, but drops to that of an oil at a temperature known as the *dropping point*, due to the grease's oil coming out of solution or its thickener soap becoming liquified. Greases have uncertain properties near or above the dropping point and are difficult to seal in this regime. Therefore, a grease should be chosen with a dropping point significantly higher than its expected temperature during operation.

Also important to greases is their oxidation resistance. At the typical usage temperature of a grease in marine machinery application, this resistance decreases roughly by a factor of 2 for each 18°F (lOOC) temperature rise, similar to what occurs for an oil. Also like an oil, oxidation and chemical resistance can be improved by placing certain additives in a grease. Antigalling additives, known as EP (extreme pressure) additives, are also useful in applications where local contact pressures are so high that microwelding across the contact surfaces has the potential of occurring, such as in highly loaded ball bearings. Interestingly, although EP additives reduce galling under high-contact pressure, in general they do not decrease scuffing or abrasive wear, and may even help to increase it. When wear rate is found to be excessive in rolling element bearings, therefore, other wear-limiting compounds should be added, and this is generally achieved with molybdenum- or zinc-based compounds.

When water is accidentally mixed with or condensed in the grease, in addition to possible problems with viscosity changes, there is a significant issue of rusting of bearing ferrous metal parts and of corrosion in cuprous parts. Water-soluble greases are better than water-insoluble greases, in general, in resisting these effects. After a certain maximum quantity of water is dissolved, however, the grease begins to lose its adhesive properties. In some greases, water at several times the weight of the grease can be absorbed before this happens. When the amount of free water in the grease exceeds about 25 parts per million, serious degradation of rolling element bearing life can be expected.

MINERAL OILS

Oils can be petroleum (mineral) or biologically (fixed) based, or can be chemically created (synthetic). Each form has advantages and drawbacks.

The primary factor involved in selecting an oil for bearing lubrication is the oil's inherent flow resistance, or viscosity. Viscosity in an oil is similar to shear strength in a solid. Since higher-viscosity oils resist motion, they tend to generate higher energy losses in bearings, and therefore the bearing runs hotter. However, they also resist being driven out of the thin film area across which the rotor load is being transmitted to the bearing shell, and thereby protect against scuffing and (in the worst case) bearing seizure. In general, when in doubt, a higher-viscosity oil should be chosen over a lower-viscosity one, so long as bearing temperature stays within manufacturer recommendations (typically about 140° to 180°F as read on the bearing outer shell, or as read in the bearing oil at or close to the bearing oil exit). However, a high viscosity oil will have difficulty being entrained into extremely tight clearances, and might allow seizure before it is entrained into the close clearance area. It also requires more force from either hydrostatic or force-feed lubricators, perhaps more than is available. On the other hand a low-viscosity oil can increase oil flow to the point that it outstrips the lube pump capacity, and adequate bearing pressure may not be maintained. Note that the viscosity for various oil grades is a strong function of temperature, as shown in the plot in figure 4-18.

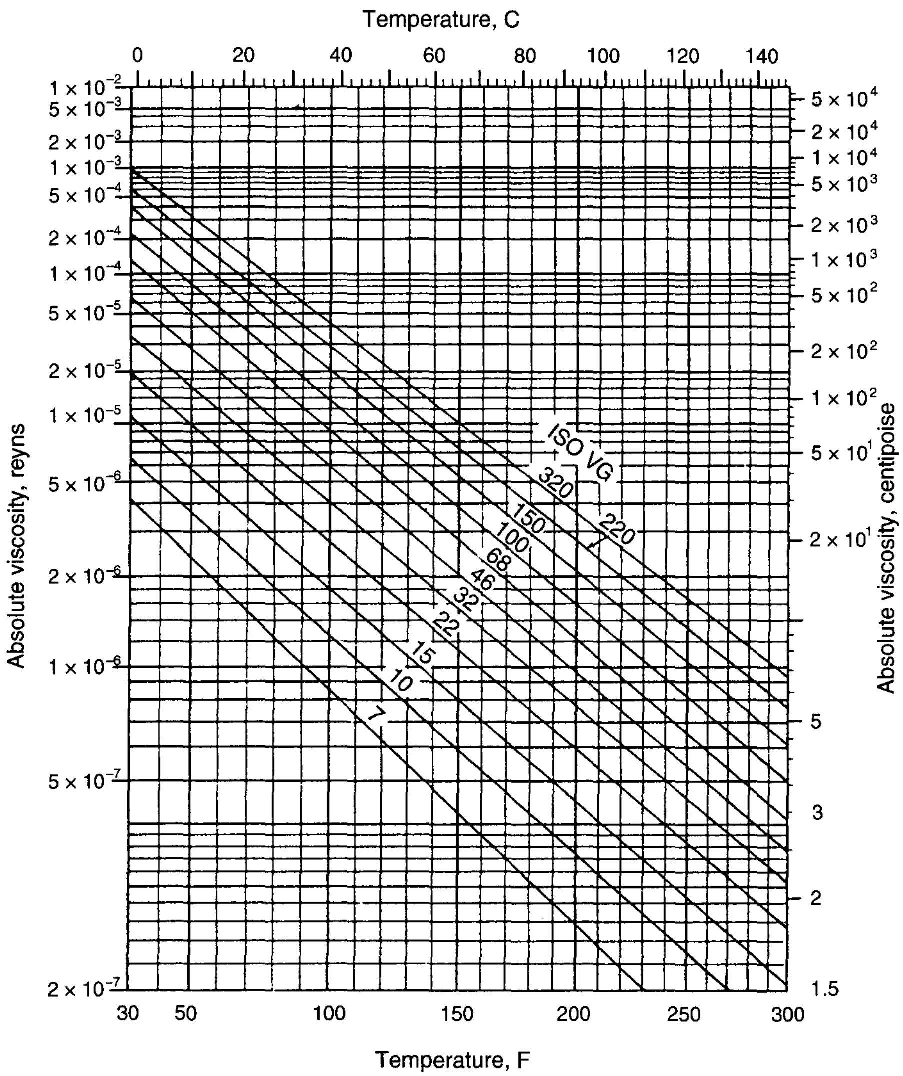


Figure 4-18. ISO viscosity grade, viscosity versus temperature. Courtesy Society of Tribologists and Lubrication Engineers and E. R. Booser, editor, *Handbook of Lubrication and Tribology,* CRC Press.

The operating point and the overall system capabilities and requirements must be taken into account when a change in oil viscosity is contemplated. The best advice is to seek out and follow machinery manufacturer recommendations, and, if not available, seek out the expertise of a bearing supplier or a qualified lubrication consultant. When seeking such advice, make sure to state machinery type and service and the range of environmental temperature to which the machine is exposed and at which the oil is likely to be in the sump and in the circulating lines to and from the sump. The latter is important because viscosity is a very strong function of temperature, and an oil that flows freely at the normal bearing temperature, for example, may become too viscous at some point in the circulating system under arctic conditions. In part, this is determined by the oil's pour point, the test temperature at which it no longer flows like a liquid.

Other important oil lubricating properties are its oxidation resistance and its resistance to degradation in the presence of water. The higher the maximum temperature of the oil, the faster it oxidizes, forming microscopic but abrasive particles and degrading the properties of its additives. Some oils, particularly synthetic oils, are superior in their ability to maintain their chemistry and lubricating properties in spite of very high temperatures. However, some of these same oils are not very tolerant of water, either forming sludge or no longer preserving a corrosion-resistant film on bearing parts. As little as twenty parts per million water in some lubricants can lead to 80 percent decrease in rolling element bearing life (although most lubricants only begin to show degradation at 25 ppm). Therefore, it is important to give special attention to the lubrication system before and after washing equipment with water or steam. The integrity of seals should be inspected before cleaning machinery with steam or water, and pools of water must not be left near the seals or the water may work its way into the bearing cavity. Although somewhat inconvenient and often contrary to practice, drums or grease and oil should be stored horizontally, not vertically. Otherwise, water can accumulate at the top of the drum, and work its way down the bung threads into the lubricant.

Some lubricants finely mix or *emulsify* with water. An oil which has this characteristic should not be used in a forced lubrication system. If water does contaminate the oil (inevitable in some services) and becomes mechanically mixed by turbulence in the circulation system or at the tribological contact, it must be separated out at the first opportunity. *Demulsibility* is the capability of the oil to reseparate, so that the water can be drained off and disposed. Some oils do not demulsify at all. This is often not a major problem for the oil itself, which might actually have its viscosity increase, although its ability to stick to parts and lower their friction coefficient (lubricity) might degrade, and over time it might have an increased tendency to form mudlike sludge (especially with the aid of biological organisms) that can clog oil passages. However, the main problem with water in the oil is that it exposes bearing parts, the circulation system, and sump tank walls to rust from water-assisted oxidation, and corrosion from acids and the biological microorganisms that the oil system is prone to in the presence of water. The resulting corrosion can directly damage the components involved, and can also create abrasive particles of corrosion product that can lead to third body damage of close running clearances or tribological contacts in bearings.

For reciprocating engines, an oil must emulsifY in the presence of water and yet continue to stick to the surfaces to be lubricated. In addition, reciprocating engine and compressor oils contain additives that delay breakdown of the oil by oxidation, and that act as detergents to hold carbonaceous oxidation materials in suspension, keeping them from coating close clearance or tribological contact areas. Extra additives are necessary if high-sulfur fuel oil is used. In general, reciprocating engine oils are more viscous than turbine or other turbomachinery oils, because they do not need to operate at high speed where heat generation is a problem, and must resist being squeezed out of close clearances during each engine cycle.

In summary, oil should be selected subject to manufacturer recommendations and must have sufficient viscosity to maintain separation of bearing potential contact surfaces, while maintaining bearing temperature within acceptable limits from the standpoint of what both the machine and the lubricating oil can tolerate. Also, it must be compatible with the realities of the operating environment. Most mistakes are made by not properly accounting for the actual operating temperature or its effects on viscosity and oxidation. Remember that bearing temperature will generally increase under any of the following conditions: more viscous oil, tighter bearing clearance, higher shaft speed, higher environmental temperature, higher vibration, larger rotor imbalance, larger misalignment, and lower bearing pressure.

SYNTHETIC OILS

Common synthetic lubricants include polyalphaolefins, esters, silicones, ethers, glycols, alkylated aromatics, and polyisobutenes. Each has its strong and its weak points. In general, when compared to mineral oils, all synthetics have improved oxidation and thermal stability, and decreased volatility. They also generally (but not in all cases) have improved viscosity/temperature tolerance (close to double that of mineral oils) and low-temperature pour point characteristics. Aside from cost (often five times that of mineral oil, and up to five hundred times for some blends), downsides include greatly diminished seal material, paint, or coating compatibility, and sometimes toxicity and poor biodegradability. Issues to consider in choosing a synthetic versus a conventional oil are the significantly higher cost of synthetics versus their increased load capacity, fire safety, and possibly improved environmental/disposal considerations. Cost factors include the price of synthetic lubricants versus manpower/oil usage credit for extended drain cycles and reduced annual downtime, reduced inventory and disposal cost, and energy credit for reduced friction.

If a switch is made to a synthetic oil from a mineral oil (or perhaps the other way around, due to cost or availability), never mix the two unless first obtaining assurance of compatibility (generally unlikely) from the manufacturer. Make sure that all of the old oil is fully drained and purged from the entire system, and change the filter element. If possible, run a smaller-than-normal amount of the new oil through the system for a short time to entrain all remaining old oil; then do a second complete drain before the final complete filling with the new oil.

Lubrication Systems

It is important for supervisors of equipment maintenance and operation in marine applications to ensure that those who maintain the equipment have been given proper education in lubricants and lubrication systems. The most important factors in lubrication system operation and maintenance are appropriate lubrication selection, timely replacement intervals, avoidance of significant lubricant contamination, and maintenance of appropriate temperature.

Generally, centrifugal pumps and compressors, and gas and steam turbines should maintain their bearing shell and/or oil exit temperatures in the range of 120° to 180°F, depending upon service and equipment type and manufacture. Higher temperatures, 180° to 200°F, are preferred for reciprocating compressors and engines with forced lubrication, while for tribological locations in areas of reciprocating machines that do not have forced lubrication (e.g., crosshead ways) 110° to 120°F is preferred. Long-term temperature limits for mineral (i.e., petroleum) oils vary from 200° to 250°F, although generally they can be run to 300°F for short periods of time without permanent harm other than more rapidly accumulated oxidation. Synthetic oils can typically run much hotter steady-state, up to 350° to 500°F, depending on the particular synthetic, with polyphenyl ethers able to run up to 700°F long term. Typically, synthetics can tolerate about lOO°F over-temperature for brief periods of time without noticeable permanent damage to the lubricant.

However, just as the oil should not be allowed to run too hot because of loss of viscosity and increased oxidation, it should also not be allowed to run too cool. Generally, lubricating oil should be at no less than 110°F, or viscosity will be so high that machine frictional losses will be excessive. This is seldom a problem for an operating machine at steady-state, but may present difficulty during start-up. Therefore, it is best not to cool the bearing oil until a system bearing exit temperature of 1lOoF has been realized, in order to speed up the warm-up process.

STATIC OIL FEED

A common method of static (i.e., nonforced, nonpressurized) oil feed is to provide an oil bath by the operation of chains, flingers, or *floating rings* which dip into a reservoir or pool of oil below or beside the bearing and then splash the bearing with oil that has temporarily adhered to the device. Thus, oil is replenished continuously in tribological zones as long as the machine is operating. In such systems, some heat transfer also takes place due to the gradual circulation of the dripped oil, and the hope is that wear particles, grit, and water will get carried away by the dripping oil and will sink to the bottom of the oil sump, where they can eventually be removed by draining. It is good practice to drain and replace the oil at the bottom of the sump in such systems on a bimonthly basis. As a minimum, the reservoir oil in bath systems should be completely replaced annually, either all at once, or on a rolling basis. If the oil temperature is relatively high (e.g., higher than 180°F at the bearing exit) in a particular service, oil replacement intervals must be more frequent. In any event, always follow or exceed manufacturer's recommendations.

Static oil distribution also can be provided by the dripping of an oil wick, fed from a small nearby elevated reservoir, onto a bearing surface likely to distribute the lubricant to all surfaces requiring it. Wicks must be lifted occasionally to prevent buildup of dirt obstructing oil flow. They should be removed from oil tubes when the machine is idle to prevent draining of the oil reservoir. When machines are restarted after being shut down for several days or more, tribological areas should be hand-oiled, and the wicks should be dipped in oil before being reinstalled in their reservoir to ensure the immediate initiation of siphoning action.

If oil supply is obtained from a wick, typically the wick reservoir is above the shaft and feed is by siphoning action. Care should be taken that the wick vertical length is at least 6 inches, which will ensure a delivery rate close to the maximum. Six-inch and over standard siphon wicks supply about 1 drop of oil per minute per strand, or about 3 drops per minute for a typical wick feeding oil at about 92 SSU. For other viscosities, delivery drops are inversely proportional to viscosity. On the other hand, bottom wicks feeding by capillary action should not be longer than 6 inches, and preferably should be less than 4 to the top of the oil in the reservoir. For lengths of less than 4 inches, delivery is roughly inversely proportional to wick length. Again, delivery is also roughly inversely proportional to viscosity. A typical reference delivery, as in a siphoning top-mounted wick, is about 3 drops per minute for a length of 4 inches and a viscosity of 92 SSU.

RECIRCULATING PRESSURE FEED

In a circulating oil system, oil is pumped from a collection and storage reservoir through a filtration system, and then either to gravity feed tanks mounted above the machines (fig. 4-19) or directly into the bearing, after

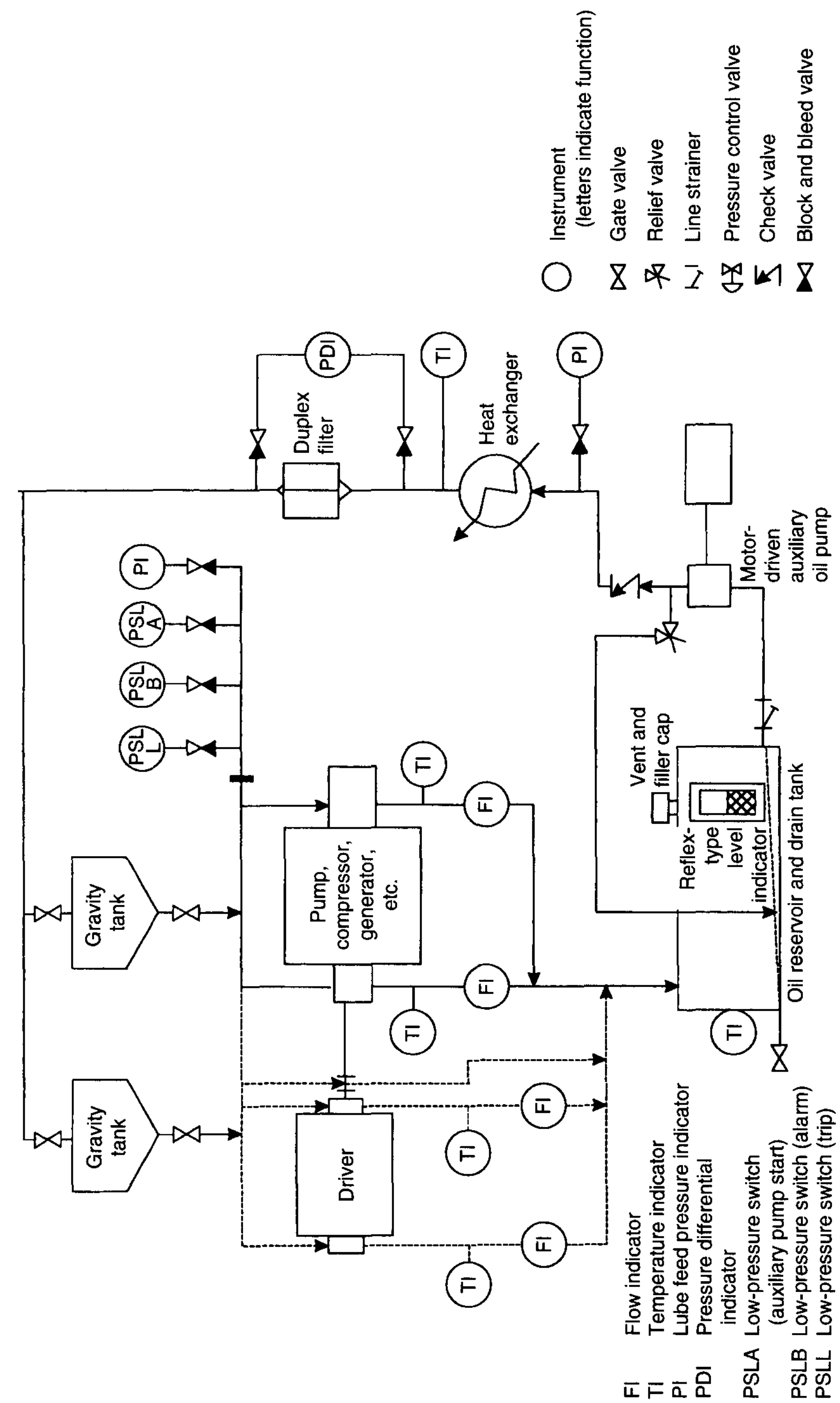


Figure 4-19. Gravity feed oil circulation lubrication system

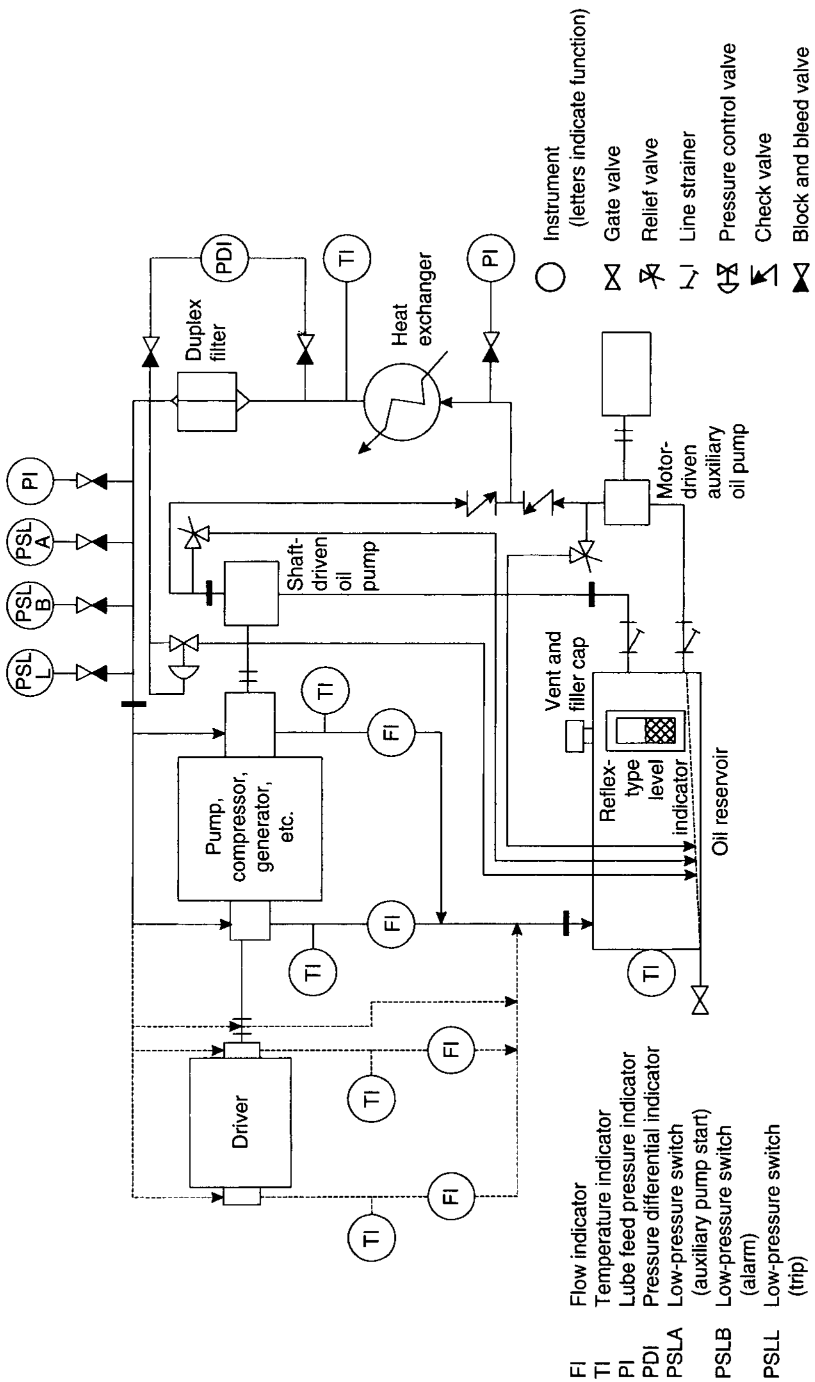


Figure 4-20. Forced feed lubrication system

which it drains from the bottom ofthe bearing cavity and is returned to the reservoir tank (fig. 20). Where the situation does not allow sufficient residence time in the reservoir, or sufficient unassisted heat transfer from the reservoir to the environment, an oil cooler may be included in the system. For operation in cold environments, an oil heater can be used in place ofthe oil cooler.

A forced lubrication system generally consists of the following:

- the bearing to be lubricated
- local oil distribution in and around the bearing contact zones
- the oil supply and return lines
- a lube supply pump (and emergency backup), normally providing 10 to 20 psig
- a reservoir and! or settling tank, including an oil level indicator (as in figure 4-21)
- oil strainers, filters, separators, and purifiers
- oil coolers (optional in some systems) set to operate only once bearing oil is 100°F.
- oil heaters (optional in some systems)
- system condition monitoring, particularly oil temperature and pressure

Figure 4-22 shows a typical marine fire pump and associated lubrication piping. Lubrication piping should be designed to be corrosion resistant and to have sufficient support spacing and piping diameter to resist permanent deformation under a mechanic's weight.

Oil Quantity and Distribution:
In forced lubrication systems, local oil distribution in the vicinity of the bearing is achieved by partial or full (generally undesirable for rolling element bearings) flooding ofthe bearing compartment by a sprayed jet or by an oil mist. For low-speed ball and roller bearings, the precise amount of oil is not very important, so long as the oil is able to be distributed in such a manner as to cover all potentially contacting surfaces. However, manufacturer recommendations should be followed concerning the minimum oil flow required. Supply and return lines should be kept unblocked by sludge or foreign objects, and the rings or other distribution devices should be checked for functionality. This means that flingers must not be loose on the shaft, chains must not be loose enough to come off oftheir pulleys or to drag on the sump bottom, and rings must be free from significant burrs, wear crescents, or deposits.

In higher-speed systems (3,600 rpm and above), care must be taken that the system does not become underlubricated. If there is a proper flow of oil, the oil usually picks up about 40°F temperature rise as it passes

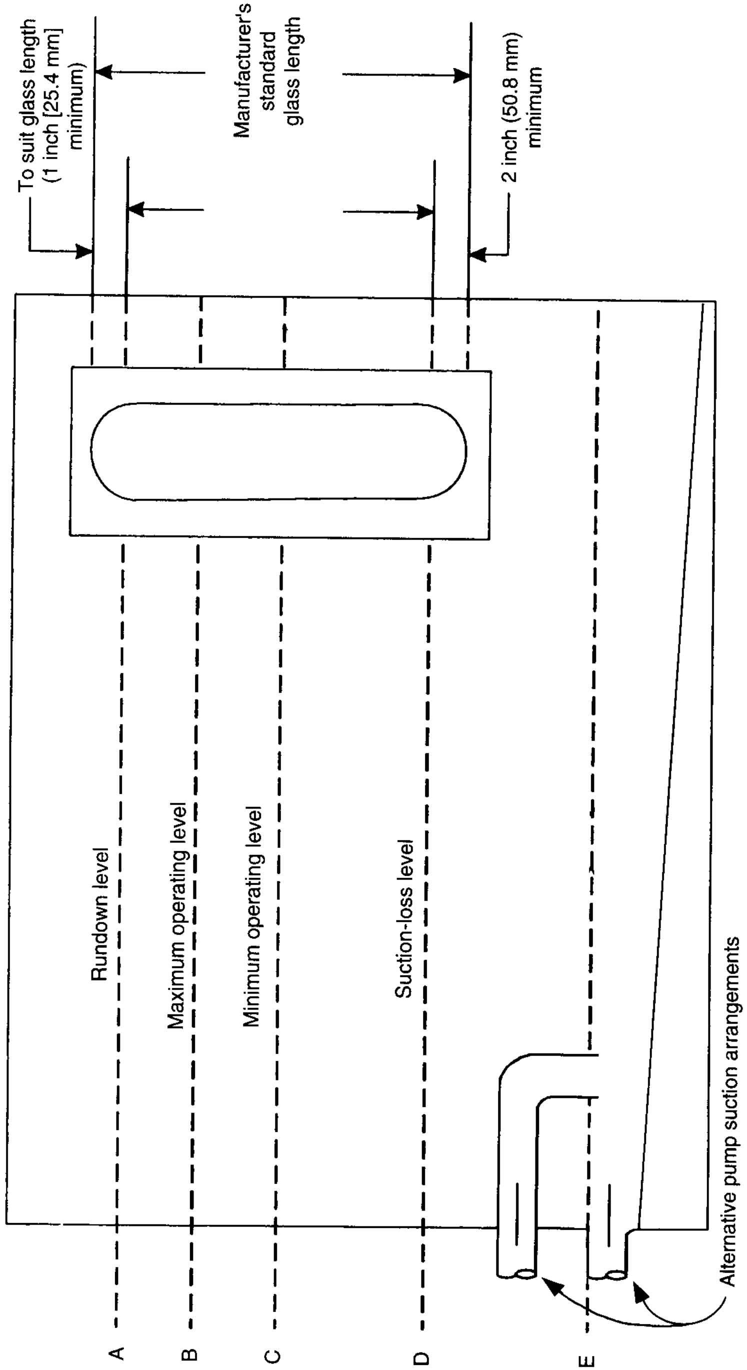


Figure 4-21. Oil level indicator as specified in API Standard 614, "Lubrication, Shaft-Sealing, and Control-Oil Systems and Auxiliaries for Petroleum, Chemical and Gas Industry Services," 4th ed, April 1999. Reprinted courtesy of the American Petroleum Institute.

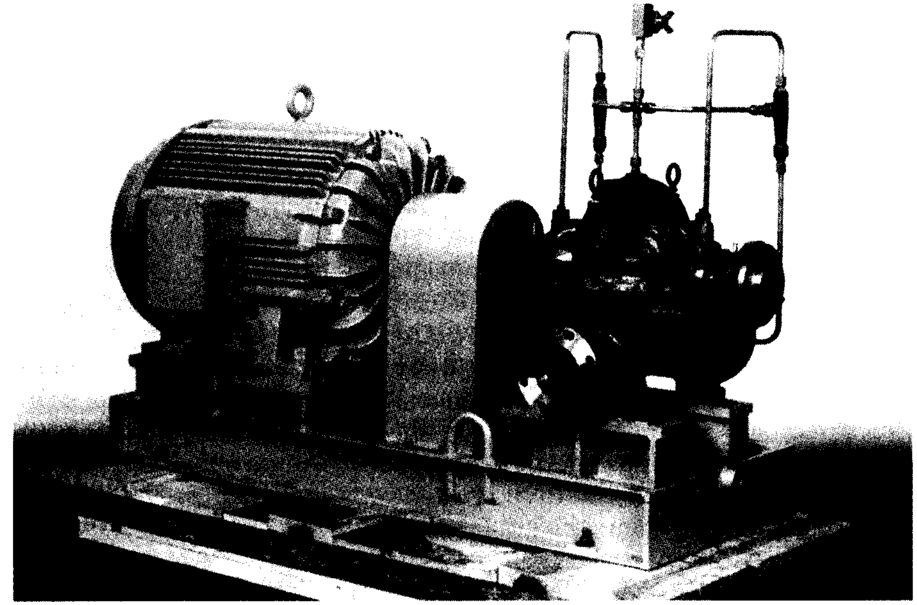


Figure 4-22. Navy fire pump and lubrication piping. Courtesy Ingersoll-Dresser Pump Company.

through the bearing, and this increases in inverse proportion to the oil flow rate. Even a brief period of time in a significantly reduced flow condition can lead to bearing failure and shaft seizure, possibly followed by massive rotor damage. In grease and nonrecirculating oil lubricated bearing systems, it is possible that enough lubricant can evaporate that this situation occurs, especially in applications where the bearing housing is covered with insulating dirt or is exposed to conduction or radiation from a hot nearby machine component. In recirculating oil systems, insufficient lubrication can be caused by some blockage in the system due to dirt, a failed check valve, or an accidentally crimped lubrication tube. Other common causes include degraded lubrication pump performance (e.g., from a failed pump seal or broken pump shaft), oil foaming (e.g., from initial overlubrication causing churning and air entrainment in the bearing housing, or because of insufficient residence time in the reservoir because it was not kept sufficiently filled with lubricant), and clogged oil filters. Avoid these situations by mounting at least two independent filters and pumps in parallel with each other, with their discharge lines either independent, or cross-connected with check valves in a manner that ensures that oil can never back-flow through a weak pump.

On the other hand, excess oil tends to produce extra drag, and therefore generates waste heat. This can be a significant problem in high-speed applications. Therefore, in such applications an oil mist is often used. Special care must be taken, however, in using such systems in a humid environment. If oil mist is present, humidity tends to condense along with the oil mist on the bearing, and particularly on cold sump wall surfaces, forming oil/water emulsions and resulting sludge and corrosion.

If excess oil is used, it can also lead to oil foaming, in which the excess oil is whipped together with air to form an air/oil mixture that provides much less lubricating and heat conveying capability per unit volume. Once in the sump, foam usually allows the oil to separate and return to fully liquid form, if the sump is well vented. In the typical system that was designed under the assumption that only liquid oil would be recirculating, a significant volume fraction of foam in the feed or return lines can lead to loss of lubrication at contacting surfaces, but more often bearing overheating is the first symptom. To discourage foaming, air leaks to the lubrication system should be located and sealed, the bearing cavity should not be overfilled or underfilled with oil (e.g., due to obstructed entry or exit lines), and oils with antifoaming agents should be used.

Contamination Control

Most bearing failures do not occur because of an improper amount of lubrication, but rather from a degraded lubricant. This degradation may be due to oxidation or chemical depletion of the lubricant, because of the oil being allowed to run for extended periods at a temperature in excess of manufacturer recommendations, or because of too much passage of time between oil change intervals. However, while such events are commonplace in consumer products, it is unusual for this kind of neglect to take place in the typical marine application. On board ship, lubricants become degraded for more subtle reasons, such as a failed seal in a pump, a crack in the water containment walls of an oil cooler, condensation on reservoir walls in humid environments (particularly if vent holes are not kept unclogged), or accidental introduction of dirt during bearing installation or lubricant addition. Dirt and corrosion flakes can damage bearing rollers and races, since such flakes are generally harder than the metal parts, and dent and score them when trapped between them during subsequent operation of the bearing. Although bearings can operate for some time in spite of a surprising amount of surface damage, there is no guarantee of this, and such situations are to be avoided at all cost. In fact, if it becomes known that a bearing has become scratched, it should be replaced immediately.

Concerning the latter, it is important not to let the bearing come in contact with dirt or grit of any kind from the time it is removed from its packaging until it is placed in the housing. For example, do not place a rolling element bearing or journal bearing pad set or shell on an apparently clean shelf or surface for even a moment. In fact, it is best to remove the bearing packaging just at the moment prior to its installation in the housing. Also,

clean all loose dirt, paint, and corrosion particles from around the housing exterior and gasket edges prior to mounting the new bearing, and make a best effort to block or mask the interior of the bearing cavity from stray dirt falling into it during this process. When installing a new bearing, always repack the bearing with entirely new grease, or fully drain and replace the oil in static or recirculating oil systems. During this process, keep the amount of time that the bearing housing is open to the air at a minimum, to keep dust, dirt, and Murphy's Law at bay.

Strainers are typically used to aid in control of particulate contamination. These can be of the sieve or magnetic kind. Install them in pairs so that one can be cleaned while the other continues to operate, without need to shut down the system. Have an emergency bypass line, normally kept closed, in case of blockage or component failure in the main line. In particular, always beware of clogging of the strainers causing a decrease in system oil flow. Examine strainers, preferably daily, for metallic particles as a sign of increased wear, particularly babbitt metal indicating rapid bearing wear. A popular type of oil strainer filtration device is shown in figure 4-23.

Filtration can also be implemented with cyclone or centrifugal separators. These are effective in removing any material that has a higher density than the lube oil, which includes water. Cyclone or centrifugal separators are essentially hollow cylinders rotated about their axis. Any impurities heavier than the oil are spun out to the outer wall. This includes water, metal particles, and most dirt particles. Such filtration devices work well on most impurities, but have difficulty with sludge and various oil emulsions, some of which are lighter than the oil itself. It is also possible for excess air to build up in the centrifuge center, forcing oil into the apparent impurity layer and encouraging oil foaming and oxidation. Therefore, centrifugal separators should be cleaned regularly and checked for clogged venting.

Water separators focus specifically on detection and removal of free water. Water can enter the oil from several locations:

- leaky joints or seams in oil coolers
- cracks in bearing water jackets
- the atmosphere, through vents, subsequently condensing on sump walls
- steam spray from steam engine or turbine stuffing boxes, jetting past bearing lip seals

Water in the oil is particularly a problem for steam turbines and for pumps with faulty seals. It may be removed by draining it from oils with high demulsibility after they have resided sufficient time (on the order of 30 minutes) in a settling tank, or by centrifuging as was discussed above, relying on the spinning or the oil/water mixture forcing the water, which

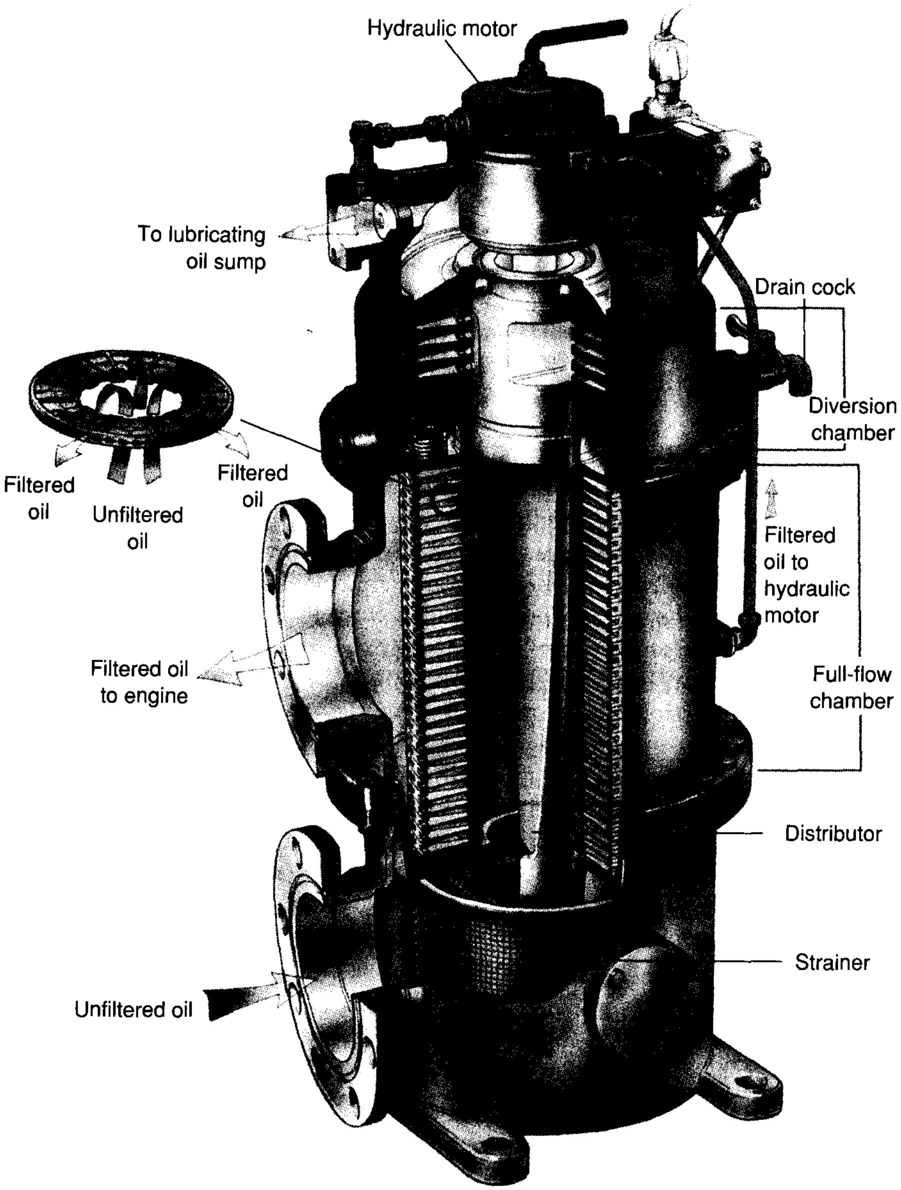


Figure 4-23. Marine equipment filtration device. Courtesy Alfa Laval Separation Inc.

has higher density, against the centrifuge wall, where it can be drained off. One type of popular cyclone separator, which removes water but also removes other heavier-than-oil contaminants, is the purifier device shown in figure 4-24.

Bearing Seals

The most common source of contamination in lube oil is a failed bearing seal, allowing ingress of water or dirt. The best practice is to replace the seal whenever the bearing requires replacement, particularly since bearing wear particles may have already begun the seal failure process on a microscopic level, in spite of the visually apparent good condition of the seal. If there is a choice, always replace the seal with an identical seal, unless prior approval of an alternate seal is received from the equipment manufacturer. Do not assume that a more expensive seal is superior for a given application. Each type of seal has its advantages and its drawbacks. For example, although mechanical seals are superior to packing in many respects, packing has a much higher degree of energy absorbing damping, and in some applications can also contribute to shaft support stiffness.

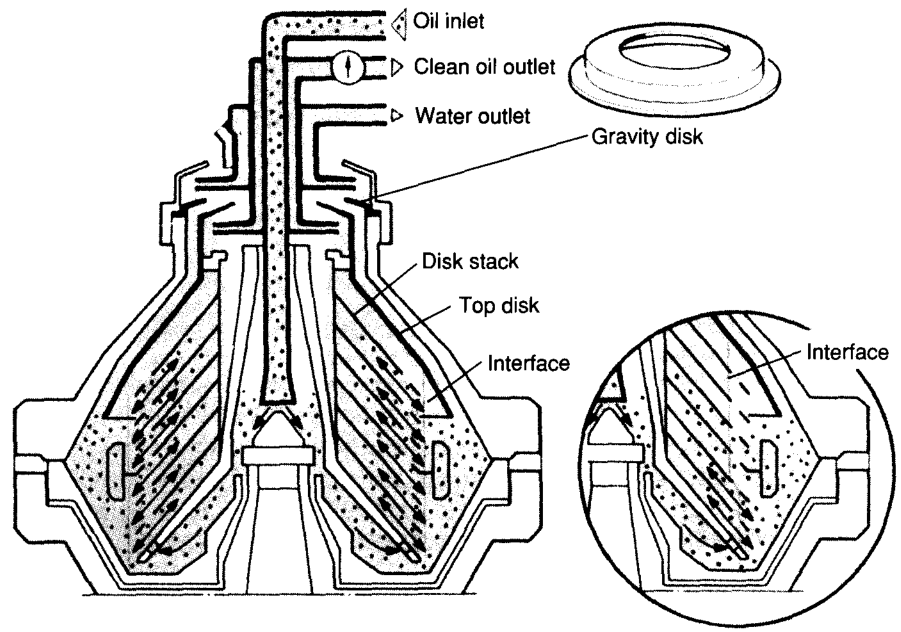


Figure 4-24. Water and contaminant separation device showing gravity disk and the effect on separated water interface of installing too small a gravity disk. Courtesy Alfa Laval Separation Inc.

There have been cases in which pumps, for example, ran perfectly acceptably for many years using packing, but then experienced chronic bearing and seal failures after the packing was replaced with a mechanical seal. In most cases, problems that occur with the replacement of packing with lip seals or mechanical seals are due to the sudden emergence of a critical speed that had been previously rendered harmless by the packing's damping.

Thermal Control by System Components and Design: Sumps, Coolers, and Heaters

A schematic of a typical system involving both a filter and a centrifugal separator or purifier, as well as a sump tank and a cooler and heater, is shown in figure 4-25.

In some installations, the purifiers are large enough that separate sumps and settling tanks are not required. In general, settling time for the oil in between times when it is lubricating the bearings should be about one hour. At least enough residence time must be provided that water, entrained particles, and air foam can separate from the oil, and the heat picked up by the oil as it performs its function (and possibly picked up by heat transfer from the process fluid, intentionally or not) can be released.

The minimum capacity of an oil sump is equal to the volume of the circulating lines and filters and heat exchangers, as well as the oil cavities surrounding the bearings and (if seal oil is used) the seals. The depth of the oil in the sump must be sufficient to provide adequate suction pressure to the oil pump at its inlet (the so-called net positive suction head requirement, NPSHR, of the pump, usually specified in feet of water, absolute rather

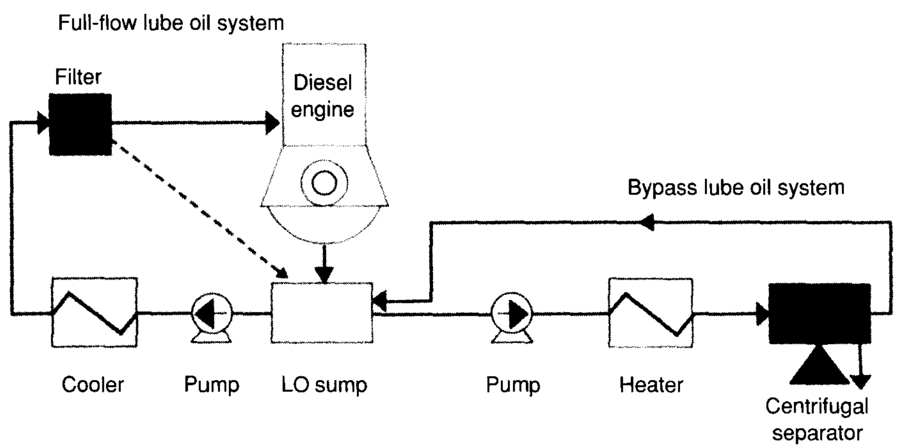


Figure 4-25. Filtration/separation device. Courtesy Alfa Laval Separation Inc.

than gauge pressure). Also, the sump should have capacity to allow sufficient residence time of the oil in the tank to allow for settling of solids, separation of water, and out-gasing of entrained air. Generally, the sump and/or the separator/purifier is also designed to have enough capacity to provide sufficient residence time to cool the lubricating oil adequately. In cases where needed, additional cooling can be performed by small water-cooled heat exchangers, or "coolers," or by a water jacket or, in modern systems, only in emergencies, by water-conducting copper tubing wrapped around the bearing housing. Such systems must be inspected often to ensure that water (especially seawater, if used as the cooling medium) is not leaking through to mix with the oil, and any water that does get in the tank needs to be drained regularly (it generally separates and settles at the tank bottom). Another key maintenance issue involving tanks is ensuring that they stay properly vented.

BEARING INSTALLATION AND MAINTENANCE

Refurbishment of Bearing Surfaces

Although recent NASA studies indicate that rolling element bearings can be successfully refurbished by grinding, this practice is generally more trouble and costly than it is worth in marine applications. Rebabbitting of journal shells and pads and thrust pads is more straightforward and economical, however. In the case of shells, they can be rolled out of the housing by unbolting and removing the housing upper half. The next step is to drive the bearing shell out with a wooden or plastic mallet or driver, hitting one side of the bearing shell at its split cross section to force the bearing shell to rotate enough to expose a lip of the shell on the other side. This lip can then be gripped to remove the bearing, or roll-out can continue until the shell breaks away.

Following removal, visual inspection can take place for wipes or other forms of bearing surface damage. If the bearing surface coating (e.g., babbitt white metal) needs to be replaced, the old babbitt can be scraped or melted off, and then new babbitt can be applied in the molten state, then formed and smoothed. After the babbitt has been smoothly contoured to the desired shape, good practice includes some *scraping* of the babbitt surface, to roughen the surface with a finish resembling fish scales, as shown in figure 4-26. Very little material is removed in this process, but the resulting mottled surface exhibits much better oil retention, particularly at high temperature.

When refurbishing a bearing, the bearing cavity and nearby components should be thoroughly cleaned. Never use solvents that could subsequently contaminate the oil, and never use rags to wipe the inside of bearing cavities, or bearing or lubrication system internal components,

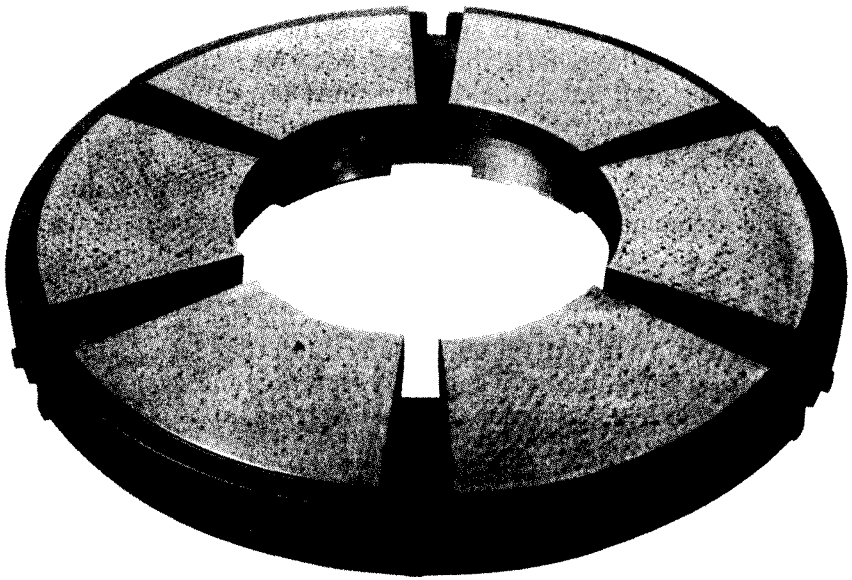


Figure 4-26. Thrust bearing showing results of hand scraping on the babbitt surface. Courtesy Kingsbury, Inc.

because of the stray fibers this might leave. Instead, use cotton swabs or paper towels.

Oil Change Frequency

Because modern oils contain detergent and other additives that are meant to scavenge dirt and soot from reciprocating cylinder walls and rotating machinery component passages, they lose their amber color quickly. This does not mean necessarily that the oil has degraded or is unsuitable for use. Time at a given temperature or some form of chemical and viscosity oil analysis should be used rather than color to determine when it is appropriate to change the lubricating oil.

Repacking with Grease

When a new bearing is installed, it is good practice to force grease over the rollers as well as in and around the retainer, by rotating the races against each other while working in grease from the sides. In addition, all surfaces of the bearing plus gland walls should be coated with at least a thin film of grease, to help protect against corrosion.

For the same reasons that excess oil should not be used to lubricate bearings, i.e., the additional heat generation due to shear of the extra fluid, grease should also not be used to excess. Various manufacturers give rules of thumb for repacking bearing glands. In general, these result in the gland

becoming about one-quarter to one-half full, never completely full. Some mechanics prefer to regrease machinery while it is running, and they use hot grease. This helps to purge the old grease from the contact zones and from around the ball retainer. When using this method, pack the new grease from an upper fitting, while the lower drain plug is open. When new grease begins to exit the drain, continue filling until little evidence of the old (generally oxidized or discolored) grease remains. At this point, leave the plugs open for five to ten minutes so that centrifugal force, heat, and circulation will empty some of the new grease from the cavity, leaving a roughly proper amount. The amount of retained grease should be checked at the next opportunity, however, to ensure that the gland cavity is about one-third full.

A common error, sometimes made out of expediency and other times out of ignorance or inexperience, is to mix greases which have different thickeners. This is done at the time of relubrication, sometimes because a guess is being made as to which lubricant was originally installed, due to poor recordkeeping. If the two greases or their additives are incompatible, there will be a serious degradation in the grease's lubricating ability. Alkaloids, acids, coagulation into forms of sludge, and significantly altered viscosity are some of the potential harmful side effects. If there is some apparently good reason to switch from one type of grease to another, it is important to get all the old grease out of the bearing housing cavity and to purge old grease from the bearing crevices, by the working back and forth of the balls or rollers and the retainer. In such situations, work the new grease into the bearing crevices by hand and continue until the color of the grease matches that of the new grease, without any sign of contamination.

Setting Clearance

Before installation, inspect the bearing shell OD versus the ID of the bore into which it will fit. Make sure that the proper interference is present, per manufacturer specifications. Also inspect versus specifications for the amount of bearing shell crush, in bearings requiring this. Measurement of a protruding shell lip to ensure a given amount of crush is illustrated in figure 4-27.

Mter installation of the shells, where possible check for assembled bearing clearance by placing plasti-gauge or leads on the shell, perpendicular to the axis, with several spread along the bearing length. Then temporarily install the shaft and fully tighten the upper bearing housing to crush the pliable gauge strips. Remove the installed shaft and check the crushed final thickness of the pliable gauges versus clearance specifications. Mter final assembly, it is good practice, where possible, to perform a "lift check" on the shaft/bearing assembly. This is accomplished by lifting the shaft ends gently until the shaft "tops out" on the upper bearing shells, while total shaft vertical movement is monitored, for example, with dial

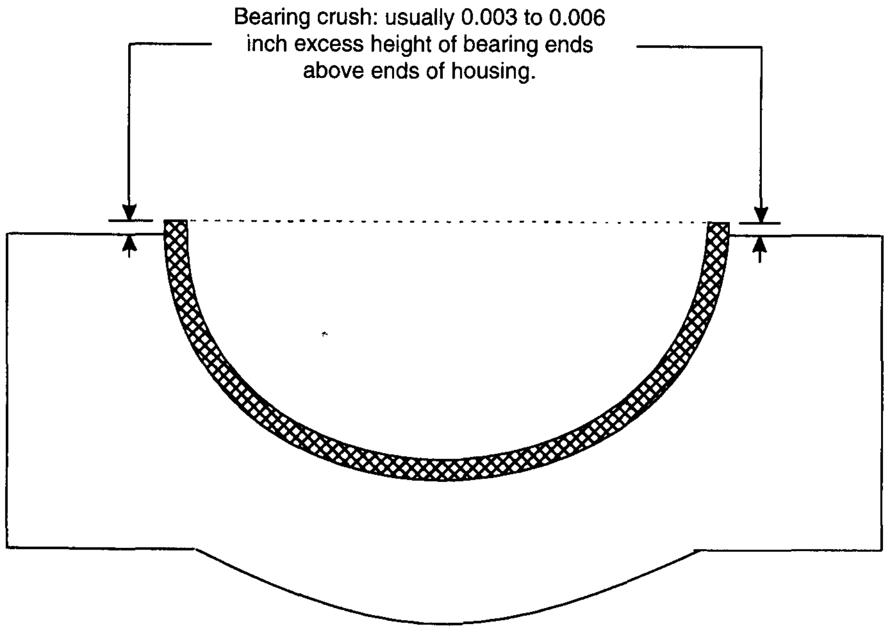


Figure 4-27. Bearing shell lip measurement to check crush

indicators fastened to the bearing housing. The vertical lift measured in this way represents the final diametral clearance of the shaft in the bearings and simultaneously checks for unforeseen binding of some other component, such as a seal or balance device.

Bearing Alignment

Most machines can tolerate an axial misalignment, due to combined coupling misalignment and a bearing race or shell cocked in the bore, of up to 2 mils (although no more than 1 mil is recommended) of centerline deviation per inch of length between coupling faces for flexibly coupled machines. For hard coupled machinery, a total of 2 mils of combined bearing cock and coupling misalignment should not be exceeded, regardless of coupling or bearing length, except for bearings of less than 2 inches length, for which no more than 1 mil of bearing cocking should be tolerated, and no more than half this value is recommended. The bottom-line rule for bearing axial misalignment with the shaft is that it should never be great enough to allow rubbing contact. Where this is not practical, bearing length should be decreased, or a spherical seat for the bearing insert outer diameter should be used.

Inside of a machine, the shaft cannot be aligned, but the bearings can, and mutual alignment of the two is critical to good bearing life. Also, in the

case of flexible shaft machinery, beware of the effect of *shaft sag*, particularly in many multistage compressors and pumps. When not operating and static, these shafts develop a V-shaped bend with maximum slope at the bearings, and this can lead to significant apparent angular misalignment that the mechanic is tempted to compensate for. However, in general this sag will disappear in service due to gyroscopic effects on the impellers or disks, and wear ring and balance device bearing-like hydrostatic stiffness (the so-called Lomakin effect discussed earlier), which develops when there is a pressure drop across them. Thus, realigning to make the bearings angularly align with a sagging shaft will actually lead to a substantial angular misalignment in service. This can be avoided by ensuring that, in most rotating machines, the inner race or bearing shell ID of one bearing is aligned angularly with the other bearing(s). This can be accomplished by bore sighting. In some machinery, however, such as some large steam turbines and ship drive shaftJstern tube applications, some shaft sag does continue during operation. The amount of this sag can be identified by a simple beam calculation, where the shaft weight and the impeller, disk, etc., weights are the primary load. In the case of stern tubes, bearing anchor points might shift with age or with hull deformations in service. This is best identified and compensated for on the basis of manufacturer recommendations, and with the engineer's experience with the equipment in question. If in doubt about whether a given centrifugal pump or compressor will have its shaft straighten out shaft sag during operation, contact the manufacturer.

JOURNAL BEARING SHELLS

Ensure that the bearing halves are indexed properly, where required, and in particular make absolutely certain that no oil feed holes or groove ends have been accidentally blocked because of, for example, reversed assembly. Typically, assembly alignment tolerances for bearing shells should be kept to 0.0005 inch per inch of bore diameter or length, whichever is smaller.

ROLLING ELEMENT RACES

Before installation of new bearings, measure the bore ID and bearing outer race OD to ensure a proper fit, per manufacturer specifications. Repeat the process for the inner race ID and the shaft OD. The maximum misalignment that can be tolerated per inch of inner race bore diameter is about 0.0005 to 0.001 inch per inch for cylindrical and tapered conical roller bearings, 0.008 to 0.025 inch per inch for spherical roller bearings, and 0.003 to 0.005 inch per inch for typical ball bearings.

Overall Shaft Alignment

A detailed discussion of this is beyond the scope of this chapter. However, it is generally recognized that the best method is the reverse dial indicator method, in which two dial indicators (or their optical equivalents) are mounted on the driver and driven shafts, respectively. Then the gauge rider is set against the opposite shaft versus the shaft that the particular gauge is mounted on. The shaft is then turned, and the total indicated run-out of each gauge is noted in no more than 90-degree increments. A simple calculation then allows determination of parallel offset and angular misalignment between the two shafts.

In order to temporarily relieve misalignment and check for new alignment adequacy, many machines include jacking bolts that can be tightened or loosened to raise or lower a given corner of the machine. Use of jacking bolts should be only temporary, however, and final alignment should be accomplished by placing shims between the two surfaces controlling the alignment and then tightening down on attachment bolts. Sometimes this procedure masks a lack offlatness between the machine feet and the mounting base, however, yielding a condition known as a *soft foot.* A soft foot makes it possible for serious misalignment to reappear after the machine is operating and can also shift the location of critical speeds by changing the effective rotor support. Soft foot determination requires placing a feeler gauge between the baseplate and each foot as the base bolts are progressively tightened, shimming as necessary under at least two of the four feet to ensure good contact with the base at all four feet once the assembly is tightened down.

A final important issue is the need for so-called hot alignment as well as cold alignment in most machinery. Cold alignment can be done repeatedly and is the easiest to perform, since it is done with the machine cold and not operating, with none of the piping pressures charged. Unfortunately, under speed and load and fully warmed up, the machine can change its alignment dramatically from what it was in the cold, static condition. Therefore, although initial alignment must be done cold, experience should be gained and logged concerning hot alignment shifts, so that cold offsets (i.e., intentional misalignment cold) can be used to compensate for hot alignment shifts, so that the machine is best aligned, driver-to-driven machine, at its typical operating conditions.

Hot alignment can be checked with lasers, or with the use of Dodd bars during operation. Dodd bars are cantilever supports that attach to a bearing housing and have 90-degree opposed shaft displacement proximity probes mounted at several locations along the bar length, so that parallel offset and angular "misalignment can be observed directly thoughout the warm-up and at steady-state. Typically, one bar is temporarily mounted on each bearing housing, so that casing/housing motions versus shaft eccentricity changes can be separately accounted for in deciding upon the best compromise alignment when the machine must operate over a significant range of conditions, and therefore varying hot alignment.

For adequate alignment, a good rule of thumb is less than 2 mils between coupling hubs, best determined by the reverse dial indicator method or its modern optical equivalent, as described in Karassik (see references), for rigid couplings, loosening this criterion by up to an additional! mil per foot of length between coupling hubs for flexible couplings.

Because of the difficulties that it causes for casing distortions and alignment, the use of unrestrained pressure-bearing expansion or flexible joints at machine nozzles must be avoided unless the contained pressure times the nozzle cross-sectional area is within the manufacturer's nozzle load limits. Although such joints relieve any piping thermal expansion or Bourdon effects from loading up the equipment nozzles, they do not allow the piping to absorb the cross-sectional hydraulic or aerodynamic load, which can become quite large and impose large forces on the machine's casing.

Rotor Balancing

It is important to maintain sufficiently low levels of excitation force by appropriate mechanical balancing. Experience has shown that it is adequate in marine applications to balance each disk or impeller and other rotating masses of significant diameter (such as flywheels, motor rotors, and possibly coupling hubs) to e = 4W/*N*, where e is the imbalance in ounce-inches, W is the weight ofthe balanced component in Ibm, andN is the peak operational speed, in rpm. Do not accept or impose any loose rotating component fits where the weight ofthe component, in ounces, times half the diametral clearance, approaches the value of e.

The reason for the constraints on balance and alignment is based on two principles. The first is that a finite amount of clearance exists between the rotor and the stator components within which it runs, particularly within the bearings themselves. Typical clearances in the bearing gaps were discussed above. In labyrinth and wear ring or balance devices, clearances tend to be about three to ten times the bearing clearance, but these areas can still rub first because of the U-shape and/or the S-shape that the rotor takes on when running at speed, due to excitation of its first bending or second bending critical speed, as discussed below. Imbalance is particularly a problem for the first bending U-shape mode. The deflection of the rotor at midspan roughly follows this equation, based on the development of centrifugal forces due to imbalance:

$$8 = [A\ (W/G_c)\ r\ (\mathrm{rcN}/30)^2]/k_{xx}$$

where $r = (DCG^2 + u^2)'1z$ on average, but $= DCG + u$ in the worst case.

A is the amplification factor, depending on how close the rotor speed W is to the first critical speed of the rotor weight (for speeds below 85 percent of the first critical speed this is roughly 1.0 to 1.5).

W is the rotor weight.

G_c is the gravitational units constant, in U.S. units 386 (lbm/JbD-in/s^2.

DCG is the amount that imbalance displaces the rotor true centerline from the apparent centerline, based on rotational center of gravity due to the fact that the rotor is not balanced (called "DCG" for "displacement of center of gravity" on some balance machines).

u is the amount that the rotor apparent centerline, which is based on the machining center for the ODs ofthe journal, is displaced from the journal bore centerline due to the size ofthe rotor motion or *orbit* in the bearing journal bore. (Note that *u* and e are vectors in that they have both magnitude and direction. In the worst case they add, and in the best case they subtract.)

N is the running speed in rpm.

kxx is the combined shaft and bearing stiffness in the direction ofthe imbalance load.

Although the balance criterion was originally established by experience, there is a theoretical justification for it, based on the above imbalance displacement equation. The amount of imbalance that could be removed by appropriate balancing is e = *W DCG*. Rearranging the above equation to solve for this shows that imbalance is equal to some constant times the acceptable level of vibration displacement, times the bearing stiffness divided by the square of the speed. However, the acceptable deflection in most machinery has been found to be inversely proportional to running speed. This is because as speed gets higher, the machine must on average get smaller because of structural and fluid dynamic specific speed constraints, and as the machine gets smaller so must the clearances to prevent unacceptable efficiency decline. Also, the shaft/bearing stiffness on average is proportional to the machine's rotor weight times running speed squared. This is because most rotors are designed to have minimal weight subject to acceptable shaft sag and avoidance of critical speed problems, and this leads to the first bending critical speed being designed to be in the vicinity of running speed (but outside the range of +/- 15 percent of running speed, to avoid resonance). Since the square of this critical speed equals the ratio of shaftlbearing stiffness to rotor weight, and the rotorlbearing system is designed so that this critical speed is running speed times a constant, then the relationship described above becomes true on average.

As can be seen, the classical balance criterion of 4*W/N*, attributed to standards first set by the U.S. Navy based on shipboard experience and still a part of MIL-STD-167 (SHIPS), are appropriate on average. However, for a specific machine the standard may be much more severe than necessary, typically up to about a factor of 4, while for a much smaller number of machines (because the standard was intended to be conservative), it may be too lenient. The important factors that form the basis for the criterion are no rubbing at bearings or internal clearances (a function of peak displacement), no bearing overloads (in most machines also a function of peak displacement, at least for vibration frequencies near running speed), and no shaft fatigue. Because of close clearances limiting shaft displacement, bending fatigue is generally a potential problem only on external portions of the shaft, due to coupling imbalance or misalignment. Therefore, for a specific machine, the key is what displacement it can tolerate, versus what displacement a given amount of rotor imbalance will produce in the machine in question. If for some reason, standard balance specifications do not prevent rubs, or it is necessary to run in an imbalanced condition for some period of time before overhaul is convenient or even possible, keep the above in mind and use the balance relationship to estimate percent of utilization of the machine's clearances, as given in the installation and maintenance manual. A good rule of thumb is to utilize no more than one-half ofthe theoretically available clearance. Ifthe relationship shows that more than half the clearances are being utilized, the machine is operating at great risk.

Therefore internal displacement is key to the acceptability of imbalance level and other vibration-causing forces (discussed further below), and clearance and therefore allowable displacement are roughly inversely proportional to speed. About forty years ago some engineers (particularly British marine engineers) noticed this meant that if measured vibrations displacements, which reflected clearance utilization, were put in terms of vibration peak velocity rather than vibration displacement amplitude, the allowable velocity level was constant regardless of running speed. This is because vibration velocity (which is literally what you would read on a radar gun if pointed at the shaft while it shakes) is merely displacement times a constant times frequency (which is speed, if the vibration occurs primarily at running speed, as is the case with imbalance). Since allowable displacement varies inversely with speed, this means that allowable velocity does not vary with speed at all.

Therefore, most allowable vibration limits today are in terms ofvelocity, and the vibration velocity level directly reflects clearance utilization percent, based on the assumption that the vibration frequency is primarily at running speed. In most cases this is true, because statistically, imbalance has been found to be the primary factor in most vibration of most machinery. Beware that if this is not true, however, the logic of using velocity as a universal single criterion value reflecting clearance utilization breaks down. For example, a velocity level that implies probable rubbing of the rotor if the vibration frequency is running speed translates into only millionths of an inch if the vibration frequency is instead at several times the blade passing frequency (i.e., number of impeller blades times running speed) due to fluid dynamic pressure oscillations that may not be doing any harm. Conversely, a velocity that is well within specs, because it implies very acceptable imbalance levels and low clearance utilization if its frequency is running speed, actually translates into displacements that are on the order of the clearances if the frequency is deeply subsynchronous, i.e., well below running speed. In pumps and compressors, for example, rotating stall in the diffuser or volute passages is not uncommon at flows well below their best efficiency point (BEP) flow, and the frequencies associated with the strong vibrations that this can produce are about one-quarter to one-tenth of running speed, causing up to ten times the clearance utilization as the observed velocity level would imply if occurring at running speed.

A general discussion of methods of rotor balancing is presented in volume 2 of the *Modern Marine Engineer's Manual.*

BEARING SYSTEM CONDITION MONITORING AND TROUBLESHOOTING

The majority of rotating equipment failures have been found to be caused by bearings, or their inadequate lubrication. It is important to pick up such problems in their very early stages, or permanent bearing damage and even machinery failure is likely to result. The planned observation of bearings and their lubrication systems in a manner likely to detect problems is known as condition monitoring, while the ad hoc application of similar procedures when a problem is suspected (e.g., because of a strange noise or excessively hot bearing housing) is known as troubleshooting.

Instrumentation for Condition Monitoring

It is good practice to monitor for lubrication oil temperature, pressure, and lubricant level, as shown in figure 4-21, and to keep an eye on the sight glass and sump surface for signs of lubricant foaming. Portable systems are available from various instrumentation manufacturers for oil monitoring of various sorts as well, and this is strongly recommended. By tracking oil color, opacity, chemistry (e.g., from an FTIR measurement), TBN (total base number), and TAN (total acid number), as well as observation of wear particle types, materials, sizes, and shapes, a great deal can be determined concerning the health of the bearings and their lubricating system. Oil pressure and temperature gauges, alarms, and trip switches should also be checked periodically for proper operation.

In addition to probes monitoring the lubricant, it is a good idea to monitor directly the bearing shell or race temperatures as well, since oil exit temperatures reflect more the average rather than the peak bearing surface temperture. Figure 4-28 is an illustration of a marine reduction gear bearing with a thin shell insert, with an embedded resistance temperature detector (RTD) to continuously monitor bearing temperature, as a rapid warning of lubricant breakdown or bearing overloads.

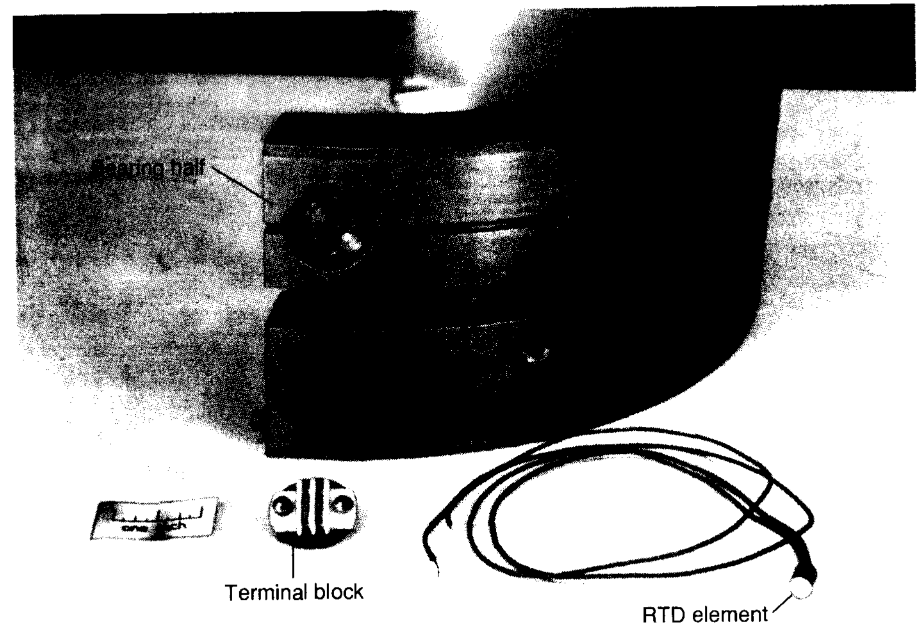


Figure 4-28. Marine reduction gear thin shell bearing with embedded RTD. Courtesy General Electric Company.

Indications of Bearing and Lubrication System Failures

Figures 4-29 through 4-43 illustrate the types of bearing failures that can occur in various shipboard machines and bearing types. The following types of failures, most of which have been discussed above, are included in the illustrations:

- ball bearing failure due to water contamination of lubricant (fig. 4-29)
- bronze bushing failure due to eccentric misalignment (fig. 4-30)
- babbitt degradation due to thrust collar wobble (fig. 4-31)
- thrust shoe damage due to electrical discharge from shaft (fig. 4-32)
- cavitation damage to thrust shoe and collar (fig. 4-33)
- cavitation and fatigue damage to journal bearing (fig. 4-34)
- babbitt fatigue in journal bearing (fig. 4-35)
- babbitt fatigue due to journal vibration (fig. 4-36)
- babbitt failure due to lack of lubrication (fig. 4-37)
- babbitt failure due to unfiltered dirt particles in the oil, with subsequent lubrication breakdown (fig. 4-38)
- large thrust shoe bearing babbitt wipe (fig. 4-39)
- oil additive deposit on thrust shoe due to overheating (fig. 4-40)

- babbitt degradation due to thermal cycling (fig. 4-41)
- thermal cracking due to overheating (fig. 4-42)
- thrust collar damage due to overheating (fig. 4-43)

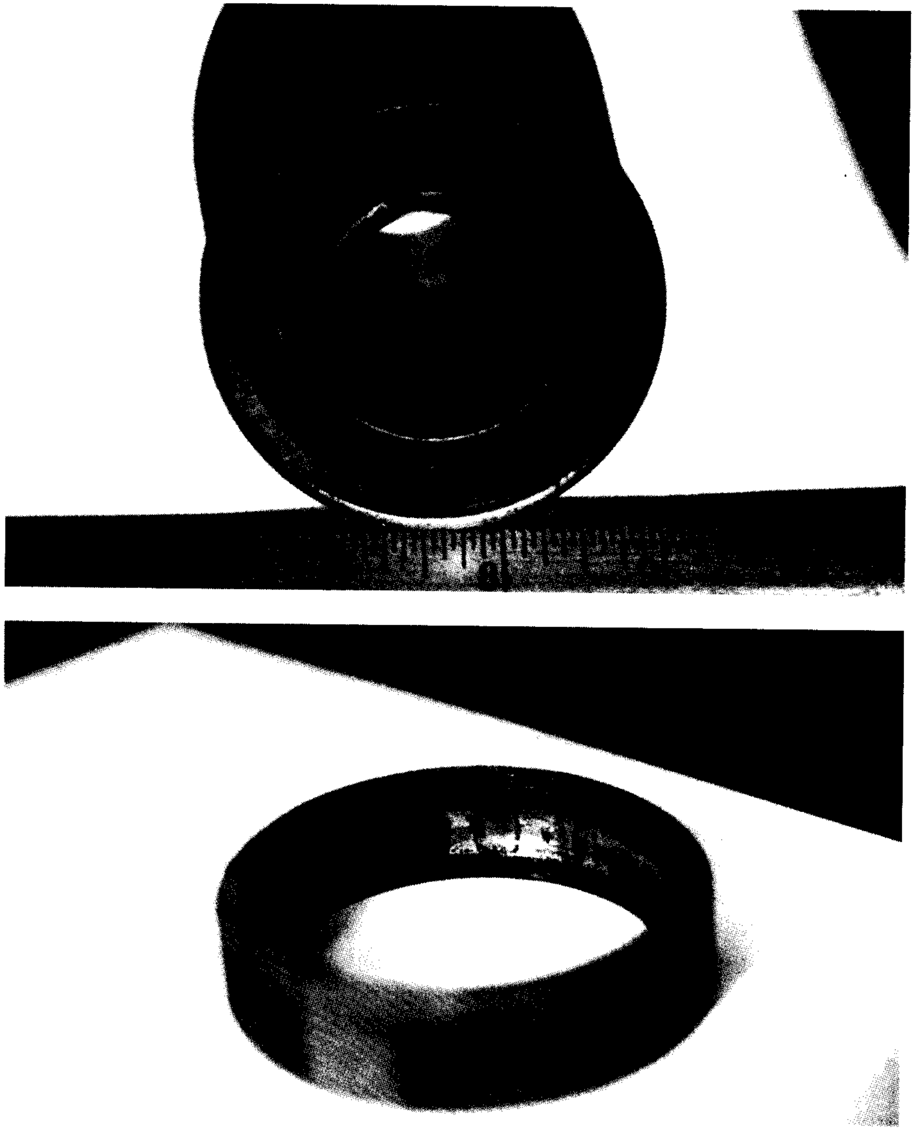


Figure 4-29. Chilled water double suction pump ball bearing failure due to lubricant contamination with water. Courtesy Ingersoll-Rand Company

Figure 4-30. Marine vertical turbine pump impeller bronze bushing failure due to severe eccentric misalignment. Courtesy Ingersoll-Dresser Pump Company.

Figure 4-31. Babbitt degradation due to thrust collar wobble. Courtesy Kingsbury, Inc.

Figure 4-32. Thrust shoe damage due to electrical discharge from shaft. Courtesy Kingsbury, Inc.

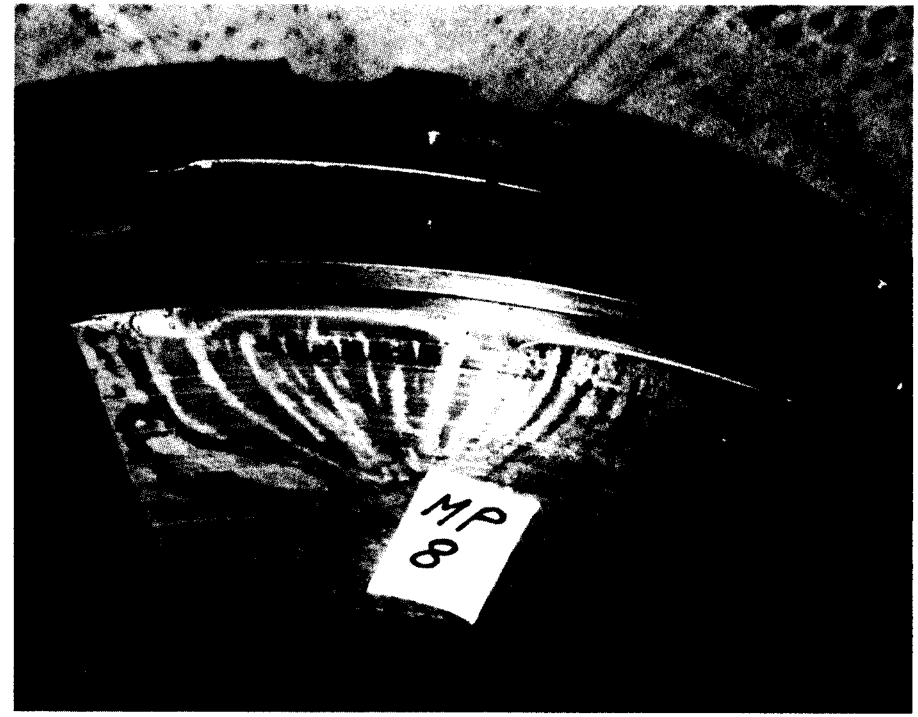


Figure 4-33. Cavitation damage at thrust shoe and collar OD. Courtesy Kingsbury, Inc.

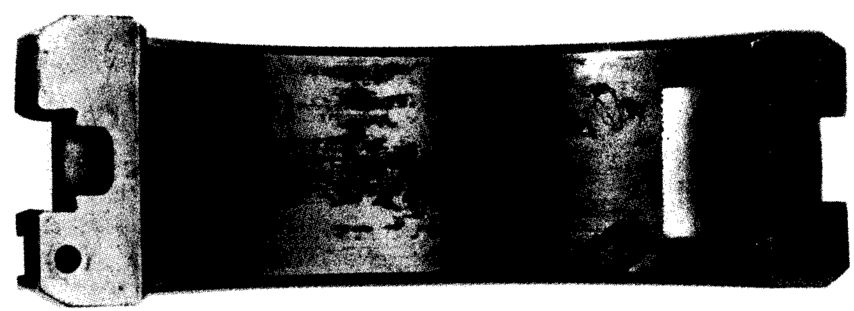


Figure 4-34. Combined cavitation and fatigue damage of journal bearing. Courtesy Kingsbury, Inc.

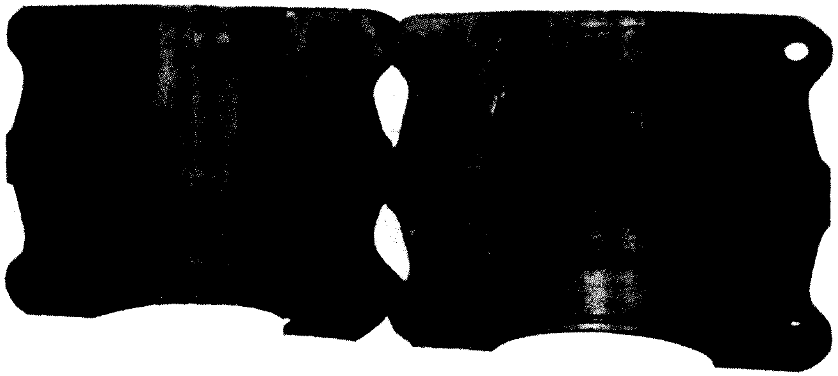


Figure 4-35. Babbitt fatigue in plain journal bearing. Courtesy Kingsbury, Inc.

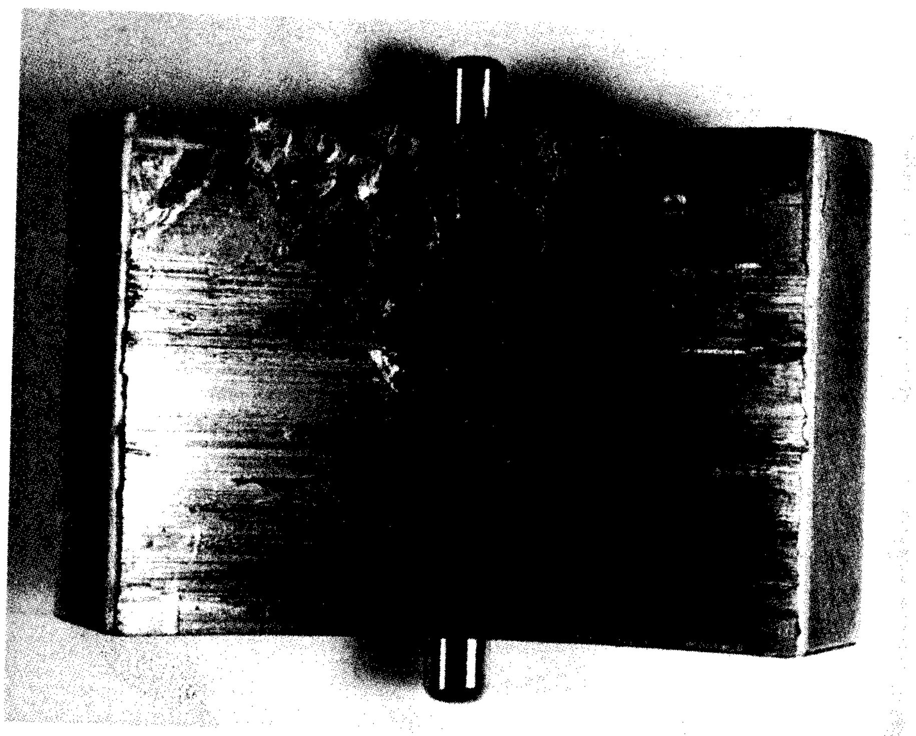
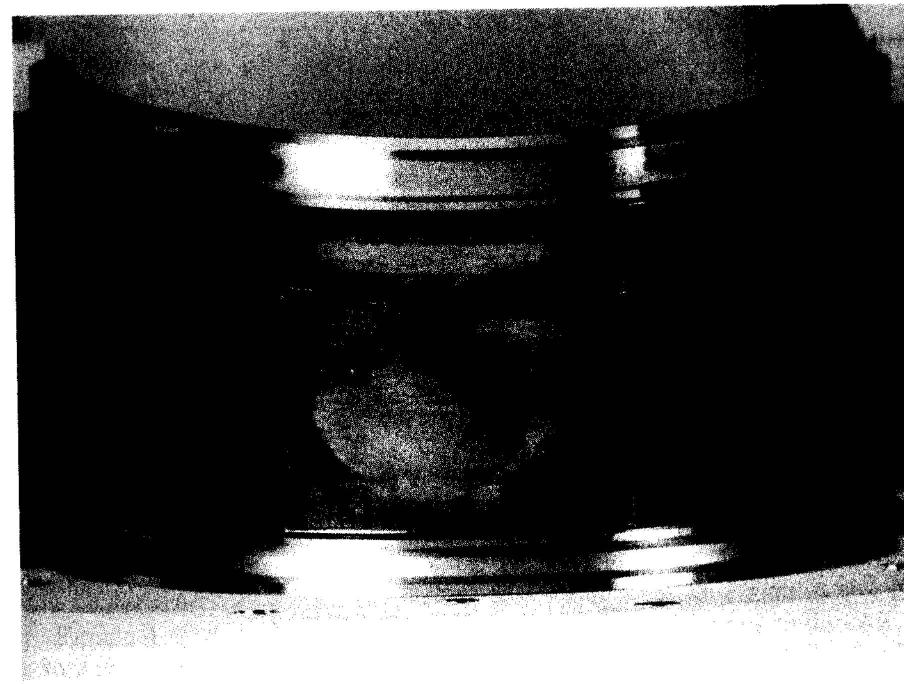


Figure 4-36. Babbitt fatigue from vibration of journal close to bearing surface. Courtesy Kingsbury, Inc.



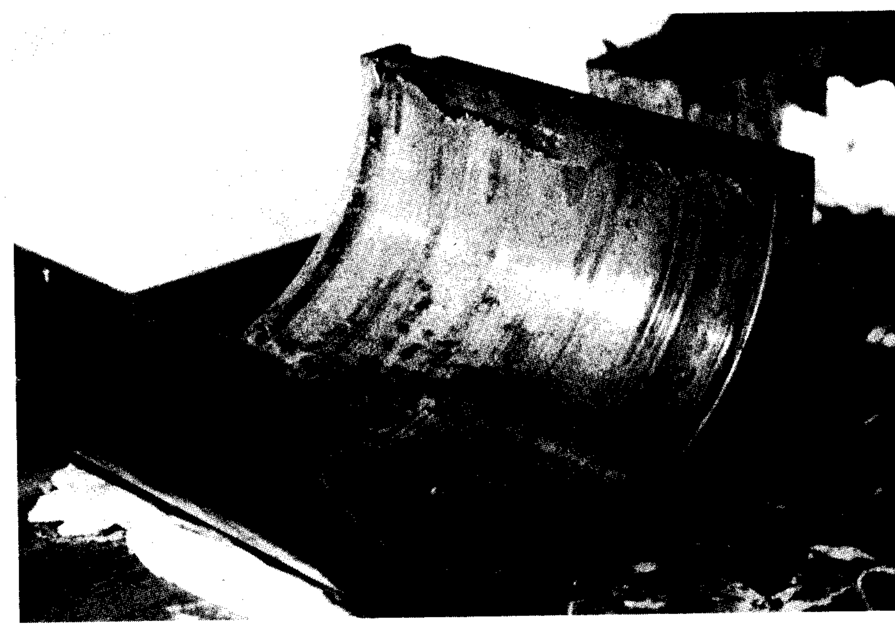


Figure 4-37. Babbitt wipes due to lack of lubrication. Courtesy Kingsbury, Inc.

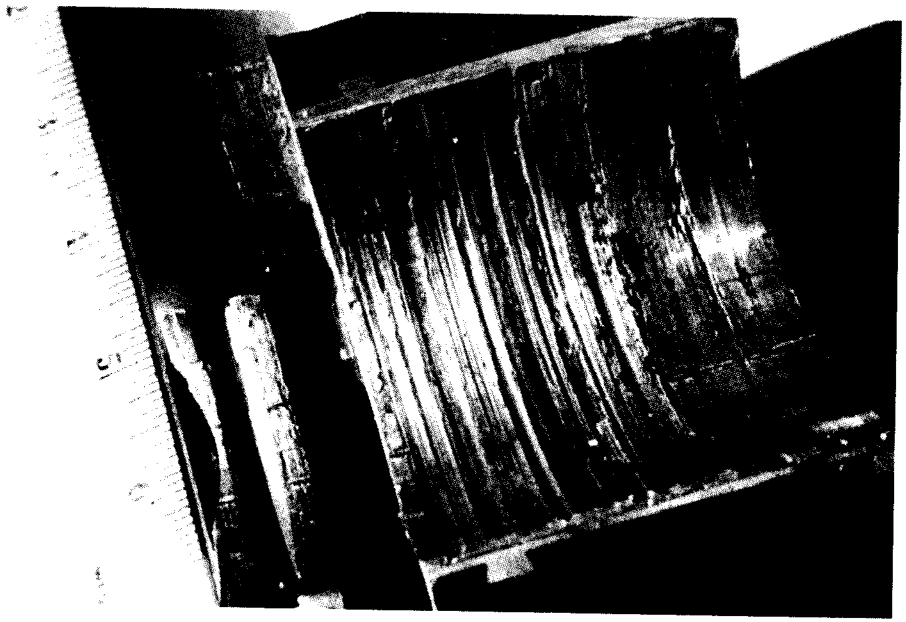


Figure 4-38. Marine boiler feed pump bearing failure caused by lack of sufficient lubrication due to presence of dirt particles. Courtesy Ingersoll-Dresser Pump Company.

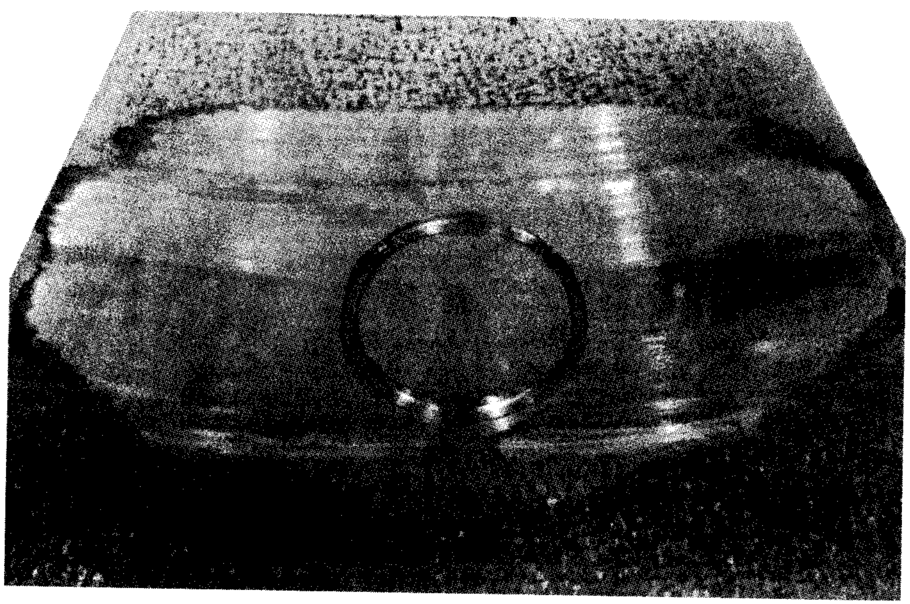


Figure 4-39. Wipe on large thrust bearing shoe. Courtesy Kingsbury, Inc.

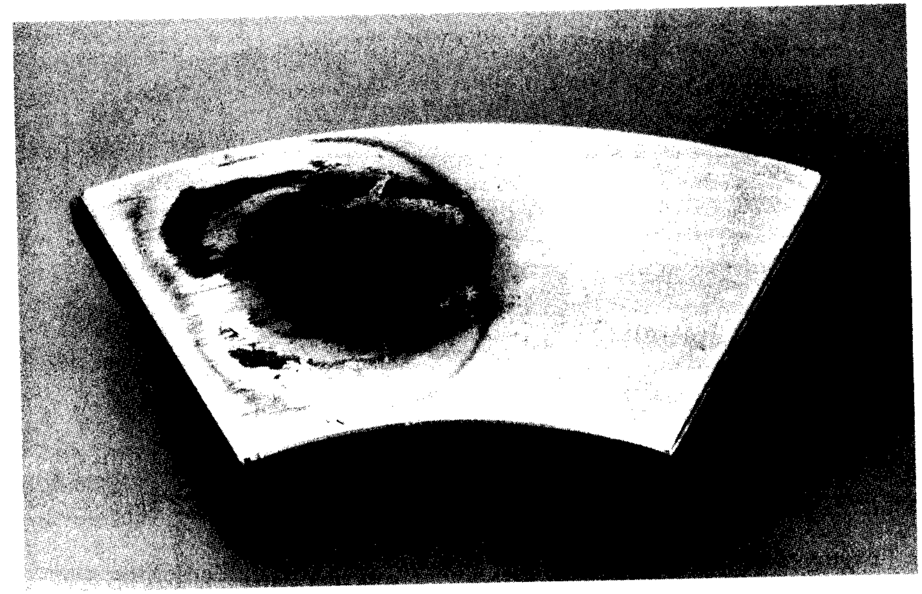


Figure 4-40. Overheated thrust shoe causing oil additives to deposit. Courtesy Kingsbury, Inc.

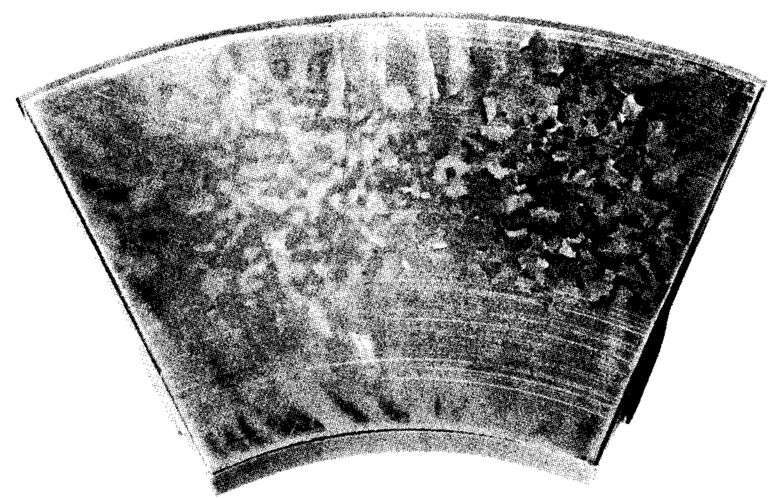


Figure 4-41. Babbitt degradation due to thermal cycling. Courtesy Kingsbury, Inc.

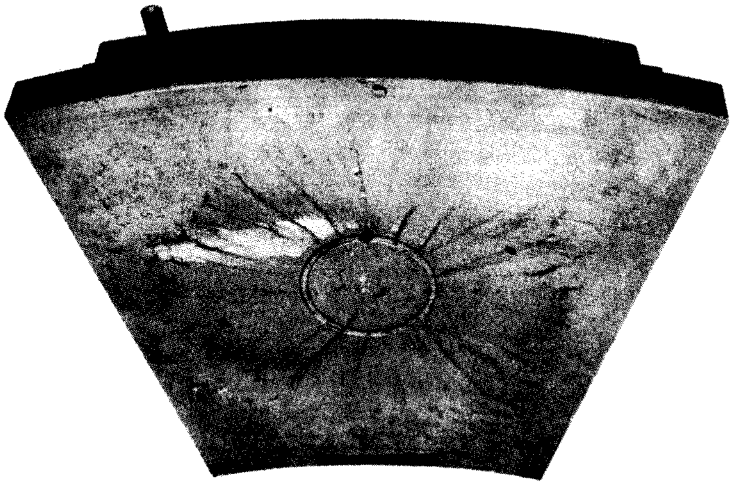


Figure 4-42. Overheated thrust shoe causing thermal cracking and spallation. Courtesy Kingsbury, Inc.

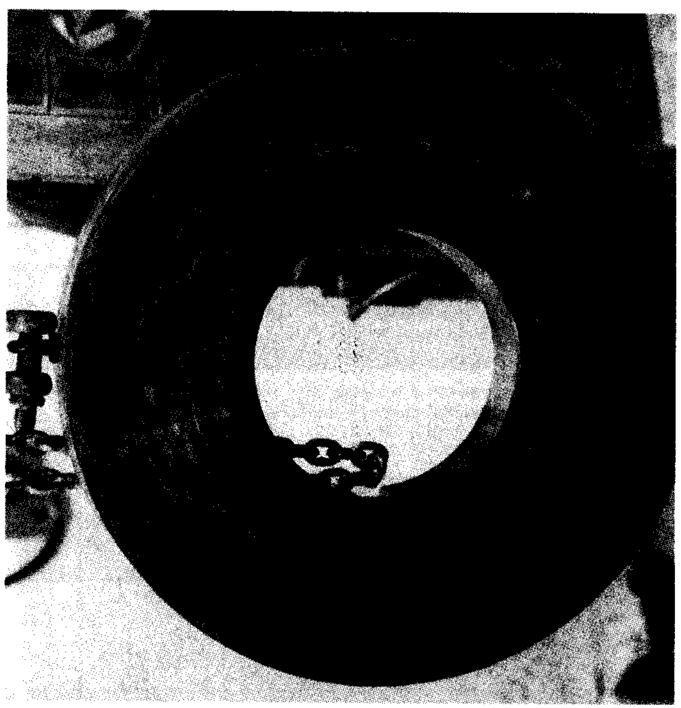


Figure 4-43. Thrust collar damage due to overheating during bearing wipe. Courtesy Kingsbury, Inc.

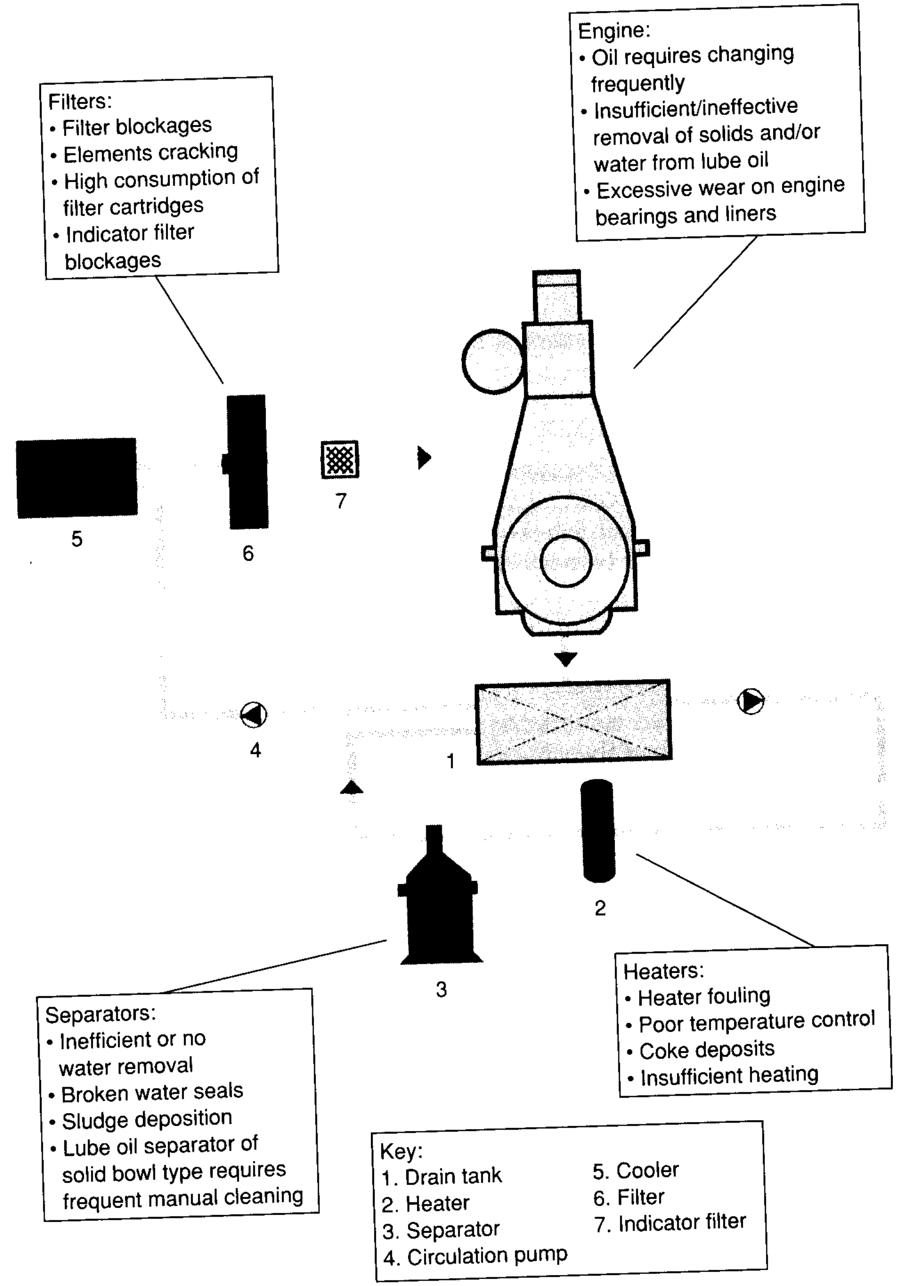


Figure 4-44. Potential lube oil problems in a diesel engine system. Courtesy Alfa Laval Separation Inc.

Figure 4-44 on the preceding page provides an excellent summary of the various problems that can occur in a marine lube oil system. Although an example of a diesel engine is shown, all the issues listed can occur in any type of rotating machine.

In addition to the problem examples above, and the issues mentioned earlier in this chapter that relate to troubles that can occur in the bearings or their lube oil system, special note should be made concerning the maintenance of adequate flow of the lubricant. This is absolutely essential to ensure against catastrophic shaft seizures and expensive machinery failures. If the oil pump pressure increases significantly, it suggests a clogged lubrication line or filter. The source should be investigated and removed immediately. Likewise, if the engineer becomes aware that oil supply has been interrupted or is likely to be interrupted, trip the affected machines immediately. Bearings in forced lubrication systems can function only for a short time without a constant supply of lubricant, because of overheating and inability to replace lubricant squeezed out of tribological contact areas. In this condition, the machine must be shut down without delay, and all troubleshooting must then be performed with the machine at rest.

Vibration and Rotordynamic Problems

Roughly 90 percent of all machinery vibration problems can be solved by careful balancing of the rotor assembly, alignment of the coupling when the system is hot at its rated conditions, and running of the machine close to its design point as much of the time as possible. Remaining vibration problems are generally due to either excessive fluid dynamic forces or to a resonance of a system natural frequency with one of the excitation forces common to machines, such as residual unbalance. During resonance, the rotor vibrations can exceed internal clearances, or excessive bearing loads can occur, even if unbalance, misalignment, and fluid dynamic loads are within normally acceptable limits. Solving a resonance requires one of the following approaches:

1. Hit-or-miss stiffening (or destiffening) of system components in a "blind" but hopefully experienced attempt to shift the offending natural frequency. Try stiffening first. Attach stiffeners at or near points of maximum vibration, and anchor them to a rigid part of the machine or foundation that is exhibiting little vibration itself.
2. Careful vibration analysis of the rotor (a *rotordynamics* analysis) and/or bearing housings and casing/base/foundation (and maybe piping) as an assembly, where as many stationary parts are included as seem to be participating in the vibrational motion.
3. Vibration testing, preferably with an FFT (Fast Fourier Transform, also called real time) analyzer, described in volume 2, which breaks down vibration versus time signals into the various frequencies at which they occur.

4. Modal testing, which involves using some impacting device (preferably an instrumented hammer) to artificially excite the rotor and/or stationary structural parts. Details of this method are beyond the scope of this chapter but are available in the references.

Obviously the least time-consuming and least expensive approach is the hit-or-miss, which is what most engineers generally apply first. This approach is successful more often than not, particularly when applied by an experienced engineer or mechanic. However, many resonance problems (especially rotordynamic ones) are unlikely to be resolved by this approach, so if it does not succeed quickly, try one of the more detailed approaches, as discussed briefly below.

ROTORDYNAMIC ANALYSIS

Rotordynamic analysis can be approximately performed on simple rotors for their first critical speed using the formulas below. For rotors with bearings on each side of the set of impellers or disks (straddle-mounted):

$$N_{c1} = (30/\pi)\left\{(6\,E\,I\,G_c)\Big/\left[L^3\,(W_i + 0.49\,W_s)\right]\right\}^{1/2}$$

For rotors that have their impellers and disks cantilevered from a shaft support at only one end by a double set of bearings (overhung):

$$N_{c1} = (30/\pi)\left\{(3\,E\,I\,G_c)\Big/\left[L^3\,(W_i + 0.24\,W_s)\right]\right\}^{1/2}$$

In both of these equations, E is Young's modulus of elasticity, I is the moment of inertia ($= \pi D^4/64$), G_c is the gravitational constant ($= 386$ [lbm/lbf-in/s^2]), L is the length from the center of the impeller and disk set to the nearest bearing, D is the average shaft diameter ($= 1/[(1/D_{max})^4 + (1/D_{min})^4]^{1/4}$ approximately, where D_{max} and D_{min} are the maximum and minimum shaft diameters in the span between the bearings), W_i is the total weight of all impellers, and W_s is the total shaft weight.

Many rotors are too complex to allow these simplified equations to be very accurate in their predictions, and in other cases one of the higher natural frequencies is involved (not the first one, which is the only one predicted by the equations above). To determine the critical speeds in such situations requires a specialized computer program which includes effects such as:

- three-dimensional stiffness and damping at bearings, impellers, and seals as a function of speed and load, including cross-coupled stiffness which acts perpendicular to journal motion

- impeller and thrust balance device fluid excitation forces
- gyroscopic effects.

Complex computer programs are needed to deal properly with these issues. Such programs typically calculate bearing and seal rotordynamic coefficients, and then use these with imbalance estimates and detailed rotor geometry information to construct either a *transfer matrix* or *finite element* computer model, as explained by Bielk and by Gunter (see references). Typical types of analyses that are run once the model is set up are critical speed prediction (together with the vibration pattern mode shape associated with each critical speed), forced response (how much motion and bearing force is caused by running at a certain speed with a certain imbalance, or a certain misalignment or fluid force), and rotordynamic stability.

These analyses reflect the three types of vibration problems that can occur in rotating machinery rotors: rotor critical speed resonance, due to a coincidence of a rotor natural frequency with an important driving force, as just discussed; rotordynamic instability; and excessive force levels (discussed for imbalance and misalignment in some detail in the section above). Rotordynamic instability refers to phenomena whereby the rotor and its system of reactive support forces are able to become self-excited, leading to potentially catastrophic vibration levels even if the active, stable excitation forces are quite low. Cross-coupling plays a significant role in this phenomenon, as discussed earlier. Figure 4-1 illustrates the concept of cross-coupled stiffness, which originates due to the way fluid films build up hydrodynamically in bearings and other close running clearances, such that it dams up on one side ofthe bearing due to the pinch point caused by deflection of the shaft in response to the externally applied force that the bearing is supporting. Therefore, the cross-coupling force is a reaction force which occurs perpendicular to the vibration, in a direction opposite to the damping force. This force can destabilize the rotor if the frequency at which the rotor is being excited is close to one of the rotor system's natural frequencies, as will be discussed next.

It is a well-known phenomenon in vibration theory that at a natural frequency there is a phase lag of about 90 degrees of rotation. This means that there is a delay relative to when a force is applied versus when the shaft responds to the force with motion in the direction that the force was applied. Therefore, when the frequency of oil whirling in the bearing clearances (as well as the other close clearances in the machine's fluid passages) becomes equal to a shaft bending natural frequency, the phase angle between the cross-coupling force and the vibration response motion due to it, previously at 90 degrees as discussed in the hydrodynamic bearing theory section above, shifts another 90 degrees. Therefore, the effects of the cross-coupling force are no longer perpendicular to the vibration, but occur in line with the vibration, such that it is not opposing the vibration but is in

the same direction as the vibration, encouraging the shaft to move even more. In other words, the cross-coupled force tends to act in a direction to close further the minimum gap. As the gap closes in response, the cross-coupled force which is inversely proportional to this gap increases further. The cycle continues until all gap is used up, and the rotor is severely rubbing. This latter process, called shaft whip, is a dynamic instability in the sense that the process is self-excited once it initiates.

The bearing and close clearance fluid rotation is typically about 45 percent of running speed because the fluid is stationary at the stator wall and rotating at the rotor velocity at the rotor surface, such that a roughly half-speed Couette flow distribution is established in the running clearance. Therefore, the situation discussed here becomes a potential problem when shaft speed exceeds twice the first critical speed of the shaft; below this speed, rotordynamic instability is very unlikely. The characteristics of shaft whip are that once it starts, all self-excitation occurs at the bending natural frequency ofthe shaft, so the vibration response frequency locks on to the natural frequency. Since whip begins when whirl is close to halfthe running speed and is equal to the shaft natural frequency, the normal one times the running speed frequency spectrum and roughly circular shaft orbit now show a strong component at about 45 percent of running speed, which in the orbit shows up as a loop reflecting orbit pulsation every other revolution. At higher speeds there is some variance to this, since the vibration, being excited by the natural frequency, tends to lock on to the natural frequency, causing vibration at speeds above whip initiation to deviate from the whirl's constant percentage of running speed.

One method of overcoming shaft whip, or more generically rotor dynamic instability, is to reduce the cross-coupling force which drives it by opening up close running clearances or by choosing a different type of bearing with less cross-coupling. A complementary solution is to increase system damping to the point that the damping vector, which acts exactly opposite to the direction ofthe cross-coupling vector, overcomes the cross-coupling. Typical design modifications that reduce the tendency to rotordynamic instability involve bearing changes to reduce cross-coupling and hopefully simultaneously increase damping. The top left corner of figure 4-6 shows the worst type of bearing in this regard, the plain journal bearing, which has very high cross-coupling. The other bearing concepts shown in figure 4-6 tend to reduce cross-coupling, dramatically so in terms of the axially grooved and tilt pad style bearings. Other bearings effective in reducing cross-coupling (but not shown) are floating ring journals, with the center bushing pinned to restrain rotation, and certain styles of pressure dam bearings.

If high vibration is not caused by resonance, by instability, or by excessive imbalance or misalignment, fluid forces should be suspect. Excessive fluid forces are usually the result of poor flow distribution at the inlet, or high blade/vane excitation pulse which take place each time a rotor blade passes by a stator vane.

In order to minimize hydraulic forces in general in centrifugal pumps (and avoid impeller damage), it is important to operate the pump with sufficient net positive suction head (NPSH), which is a measure of absolute suction pressure at the pump inlet, as discussed earlier. If this pressure, compensated for fluid velocity, drops below the fluid vapor pressure, then vaporization or cavitation will occur. The imploding bubbles from cavitation can cause serious erosion damage to the impeller and can also produce broadband vibration analogous to AM radio static. This can make the pump shake, at both low and very high frequencies. The cure is either to increase the suction pressure to beyond the NPSH required by the manufacturer or obtain a different impeller with lower NPSH requirements.

When excessive hydraulic forces occur at a frequency equal to the number of rotor blades or stator vanes or tongues, this is the so-called "blade pass" or "vane pass" frequency. Generally, it is possible to lower vane pass forces at some loss of efficiency by opening up the clearance between the impeller blades and the volute tongue or diffuser vanes. Sometimes these clearances are as close as 111z percent of the rotor diameter, but a minimum of roughly 6 to 10 percent has been found to work reasonably well in most cases, with significant reduction in vane pass forces. However, modifications like this should be done only with the guidance of the manufacturer or a qualified consultant.

Another common cause of excessive fluid dynamic forces in pumps, compressors, fans, and turbines is running them at flows too far from their design point. Generally, vibration, shaft deflection, and bearing load *increase* as a pump, fan, compressor, or turbine is run at lower flows, rather than decrease as is commonly assumed. This results in decreased life and reliability. The reason for this is that at off-design conditions, there is imperfect match between the angles of incoming flow and stage-to-stage flow versus the angles of the rotor blades, which were optimized for operation at or near the best efficiency point or design point. This should be factored into purchasing and operating decisions, in terms of lifetime cost· and maintenance headaches. Try to run machines near their design point most of the time.

EMA MODAL IMPACT TESTING

Natural frequencies are those frequencies at which mechanical systems prefer to "ring down" after an excitation force is applied and removed, such as after impacting, like a tuning fork. Marine machinery and the attached piping systems and deck supports have an infinite number of natural frequencies, but fortunately only those that are close in frequency to strong excitation forces are of practical significance. These can be found by using a shaker or an impact hammer to excite the system in a controlled manner,

and then observing the ringdown with an FFT analyzer to separate out the various natural frequency responses that make up the ringdown.

Resonance occurs when there is a matching of an excitation force's frequency to a natural frequency. This leads to storage of the vibration energy from cycle to cycle, similar to the dribbling of a basketball. In a resonant condition, very large vibration can build up even if the excitation forces are well within reasonable specifications. Generally, in order to fix such problems permanently, the offending natural frequency must be found and moved away from the resonant excitation frequency. *Experimental modal analysis* has proven its ability to solve vibration problems quickly by separately determining the natural frequencies and, by deduction, the excitation forces of the ship's machinery and system components. Using modern modal analysis software, a computer can create a no-assumptions model of the vibrating system based on input of a large number of test measurement points stored during an impact test.

CLOSURE

The biggest problem with bearings in marine applications is breakdown of the bearing lubrication. Usually, this occurs because of a seal failure or sump contamination with dirt or water. Another common problem is overheating of the bearings, due to overloading from shaft hot misalignment or from insufficient lubricant flow.

Sometimes bearings become overloaded due to vibration. Often vibration is a symptom of some other root cause, such as imbalance, misalignment, excessive fluid forces, or a natural frequency resonance or instability.

Occasionally bearing overheating or failure can be due to the selection of a bearing by the manufacturer or repair shop that has inadequate load capability; or it can be due to the improper installation of the bearings. The best way to avoid design and installation problems is to deal with reputable suppliers and consultants, and to spend the time and money to train engineering officers in machinery characteristics and bearing and lube system selection, installation, and maintenance.

ADDITIONAL INFORMATION

The Society of Tribologists and Lubrication Engineers (STLE) was established in 1945 (at that time called the American Society of Lubrication Engineers or ASLE). Its main purpose is the gathering and dissemination of information concerning lubrication science and engineering, as well as other aspects of tribology, such as bearing and seal design, application, and maintenance. Most bearing and seal manufacturers and suppliers belong to this society. Therefore, STLE is an excellent place to find additional information about any of the topics discussed in this chapter, specifics concerning your particular bearings or lube system, and up-to-date manufacturer contact information. STLE can be contacted at:

Society of Tribologists and Lubrication Engineers
840 Busse Highway
Park Ridge, Illinois 60068
Telephone: 847-825-5536
Fax: 847-825-1456
Web site: www.stle.org

REVIEW

1. What is the primary operating principle of a fluid film or journal bearing? How is this different from the hydrostatic principle? Can the two effects work together? If so, give an example of where this is put to a useful purpose in modern marine machinery.
2. What is the difference between a fixed pad and a tilting pad bearing? Why were tilting pad bearings invented, i.e., what is their main advantage over other types of fluid film bearings? What are several important disadvantages of tilting pad bearings?
3. What is the primary nondimensional number that is used to predict various fluid film bearing characteristics? Physically, what does this number represent? What kind of serious problems can you run into if it is too high? What are some steps you could take to reduce it if you encounter this problem?
4. Can cavitation occur in a bearing? If it can, is it good or bad? If either or both, give some specific examples to back up your answer.
5. What kind of unit load can a typical plain journal bearing accommodate in a rotating machine? Does this increase or decrease in a reciprocating machine, and how much? What kind of unit load capacity is available in a tilting pad thrust bearing? What sorts of manufacturing and installation issues can compromise this potentially high unit load capacity?
6. Name the four main solid components of a rolling element bearing. Why would someone use spherical rollers rather than cylindrical rollers in such a bearing? Which has a greater load capacity for the same size bearing: a roller bearing or a ball bearing? What is the primary reason for this?
7. What is the B10 life of a bearing? How does it depend on load? What ratio of water in the oil will begin to lower this? What lubricant property would resist water causing damage?

8. What is the basic load capacity of a rolling element bearing? What role does this play in B10 life?
9. What is the DN rating of a bearing, with units? What is a safe range of DN for standard forms of lubrication? What value of DN suggests the need for the best available technology and outstanding maintenance practices in order to be successful?
10. Why would duplex ball bearings be used on a shaft? What might be a good method of providing a sure preload between them, over a broad bearing load and temperature range?
11. Do pump wear rings and balance drums have any significant effect on the support of the rotor? Do they affect the rotor's critical speeds? What effect does lost clearance or deep grooving have on the wear ring support stiffness, generally and in the worst case? Can you think of a substantial unexpected problem that this could cause in a pump that performed well on the test stand as well as on ship when brand new and unmodified?
12. Finish the sentence: It is important to keep lubricants clean, cool, and
13. Name some of the important functions of a lubricant, other than maintaining shaft clearance separation by virtue of high viscosity.
14. Why would someone use grease rather than an oil? Why would someone select an oil over a grease?
15. What are the typical key benefits of a synthetic lubricant? What is the primary drawback?
16. In general, is it acceptable to mix a petroleum/mineral oil lubricant with a synthetic lubricant? What would be good practice in terms of system flushing if a switch from synthetic to petroleum-based oil had to be performed at sea?
17. Why is demulsification of an oil important? Under what conditions would pour point be important? Under what conditions would the dropping point of a grease be important?
18. Why would an engineer choose a recirculating oil feed system over a static feed system? What are the key components of a typical static feed system? What are the typical components of a recirculating oil system?
19. What structural characteristic should a lube oil line and its supports have that is sometimes overlooked, resulting in a broken line when maintenance or inspection is being carried out? Why is it a good idea to have two parallel filters? Why is an auxiliary lube oil pump needed?
20. How full should the bearing cavity be repacked with grease? What happens if you overfill? If you underfill?
21. What values of misalignment typically can be tolerated by the three different types of rolling element bearings (excluding needle bearings)?

22. *Mter* machine overhaul has been completed, if alignment is perfect while the machine is cold and not yet operating, is the alignment job over? What do you do if hot alignment is substantially different from cold alignment? What if alignment changes over the normal operating speed range?
23. What is the typical marine balance criterion, in ounce-inches, based on machine maximum operating speed and rotor weight?
24. What are some kinds of condition monitoring that should be done, and what are some of the typical instruments involved?
25. Name four common maintenance or failure problems in marine machinery bearings 0t their lube oil system.
26. What is rotordynamic instability? What causes it, and what are its symptoms?
27. What are the typical kinds of vibration tests and analyses that might be performed to identify a vibration problem? What are these tests looking for, specifically?
28. What vibration measurement is relatively independent of speed, in general? Is this a good parameter to use mostly near running speed, or at all frequencies? What is the most important issue in a machine, relative to whether a given vibration level will cause a problem for it?

REFERENCES

Agostinelli, A., D. Nobles, and C. R Mockridge. 1960. "An Experimental Investigation of Radial Thrust in Centrifugal Pumps." *Transactions,* v. 82. New York: American Society of Mechanical Engineers (ASME).

Allaire, P.E., et al. 1978. "Dynamics of Short Eccentric Plain Seals with High Axial Reynolds Number." *Journal of Spacecraft AIAA,* 15, no. 6.

American Petroleum Institute (API). 1995. *API 610,* 8th ed. Washington, D.C.: API.

--1992. *API* 614, 3rd ed. Washington, D.C.: API.

Bielk, J., and L. Barrett. June 1984. "Rotor Stability Code ROTSTB." UVA report no. UVA/643092/MAE84/308. Charlottesville, Va.: University of Virginia.

Black, H. F. 1979. "Effects of Fluid-Filled Clearance Spaces on Centrifugal Pump Vibrations." 8th turbomachinery symposium. Houston, Tex.: Texas A&M University.

Blake, M. 1960. *Shaft Couplings in Vibration and Acoustic Measurement Handbook.* New York: Spartan Books.

Blevins, R D. 1984. *Formulas for Natural Frequency and Mode Shape.* Malabar, Fla.: Robert Krieger Publishing Co.

Bolleter, U. September 1988. "Blade Passage Tones of Centrifugal Pumps," *Vibrations,* 4, no. 3.

Booser, E. R, ed. 1983, 1993, 1996. *Handbook of Lubrication and Tribology,* volumes 1 through 4. Boca Raton, Fla.: CRC Press (cosponsored by Society of Tribologists and Lubrication Engineers, Park Ridge, Ill.).

Busse, W. F., and W. H. Denton. November 1931. "Water-lubricated Soft Rubber Bearings." Proceedings of annual meeting of ASME. New York: ASME.

Childs, D.W. 1982. "Finite Length Solutions for Rotordynamic Coefficients of Turbulent Annular Seals." ASME paper 82-LUB-42. New York: ASME.

Downham, E. November 1957. "Theory of Shaft Whirling, No.5-The Influence of Plain Bearings on Shaft Whirling." *The Engineer.*

Ellison, L. F., and J. M. Partridge. 1976. "Vane Vibration in Radial Flow Turbochargers." Proceedings of Institute of Mechanical Engineers (IMechE) "Vibrations in Rotating Machinery."

Ewins, D. J. 1984. *Modal Testing: Theory and Practice.* Wiley, N.Y.: Research Studies Press.

Fraser, W. H. 1984. "Centrifugal Pump Hydraulic Performance and Diagnostics." *Pump Handbook.* New York: McGraw-Hill.

Fuller, D. D. 1967. "Fluid Film Bearings." Chapter 8, *Marks' Standard Handbook for Mechanical Engineers,* 7th ed. New York: McGraw-Hill.

Gunter, E. J., and D. F. Li. March 1978. "Unbalance Response Analysis of Dual Rotor Systems." Report No. UV*A/6430921MAE81/138.* Charlottesville, Va.: University of Virginia.

Hergt, P., and P. Krieger. 1970. "Radial Forces in Centrifugal Pumps with Guide Vanes." Proceedings ofIMechE, v.184.

Hydraulic Institute. 1994. *American National Standard for Centrifugal Pumps.* ANSI/HI 1.1-1.5-1994. Parsippany, N.J.: The Hydraulic Institute.

Jery, B., A. Acosta, C. Brennan, and T. Caughey. 1984. "Hydrodynamic Impeller Stiffness, Damping, and Inertia in the Rotordynamics ofCentrifugal Flow Pumps." Proceedings of "Rotordynamic Instability Problems in High Speed Turbomachinery." Houston, Tex.: Texas A&M University.

Karassik, I. J., W. Krutzsch, W. Fraser, and J. Messina, eds. 1984. *Pump Handbook.* New York: McGraw-Hill.

Karelitz, G. B. December 1929. "Performance of Oil-Ring Bearings." ASME Paper APM-52-5. Proceedings of 1929 ASME annual meeting. New York: ASME.

Kirk, R G. 1985. "Evaluation of Aerodynamic Instability Mechanisms for Centrifugal Compressors." ASME paper 85-DET-147. New York: ASME.

Marscher, W. D. December 1986. "Determination of Pump Rotor Critical Speeds During Operation through Use of Modal Analysis." Proceedings of ASME 1986 WAM symposium on troubleshooting methods and technology, Anaheim, Calif.

CHAPTER 5

Steam Generation

JAMES A. HARBACH

Boilers have been installed on ships since the nineteenth century. The early designs were fire-tube boilers generating low-pressure saturated steam to operate a reciprocating steam engine. Over the years, as pressures increased and the steam turbine replaced the reciprocating steam engine, the water-tube boiler replaced the fire-tube boiler. Today marine boilers are available in a wide variety of designs and sizes, from small auxiliary boilers producing a few thousand pounds per hour of low-pressure saturated steam, to large propulsion boilers producing hundreds of thousands of pounds per hour of high-pressure superheated steam.

Boilers of some shape or size are found on just about every oceangoing vessel, regardless of the type of propulsion plant. While this chapter will concentrate on the larger and more complex propulsion boilers, much of the information will also apply to the smaller auxiliary and waste-heat boilers found on diesel vessels.

BOILER TYPES

Boilers can be classified in a number of ways. Some of these include the following:

- fire-tube or water-tube
- straight-tube (sectional header) or bent-tube CD-type)
- reheat or nonreheat
- natural-circulation or forced-circulation
- fuel to be burned-oil, coal, natural gas

The following sections will describe some typical marine boiler designs, from those of historical interest to current applications. Boilers and combustion systems for coal fuel are described in volume 2 of *Modern Marine Engineer's Manual.*

Scotch Marine Boiler

In the early twentieth century, the most popular type of boiler in marine use was the Scotch boiler. It overcame many of the structural weaknesses of earlier fire-tube designs, using an easily fabricated cylindrical shell. Scotch boilers were manufactured either single- or double-ended, with from one to four furnaces on the flat ends. Figure 5-1 shows a single-ended Scotch boiler, with two furnaces fitted with spreader stokers for coal firing.

The design was noted for ruggedness, reliability, ease of maintenance, and the ability to withstand abuse. Its major limitation was that its operating pressure was limited to 300 psig. The development of steam turbine propulsion and its ability to use higher pressures spelled the end of the use of the Scotch boiler in main propulsion applications. The design is still produced for low-pressure heating applications.

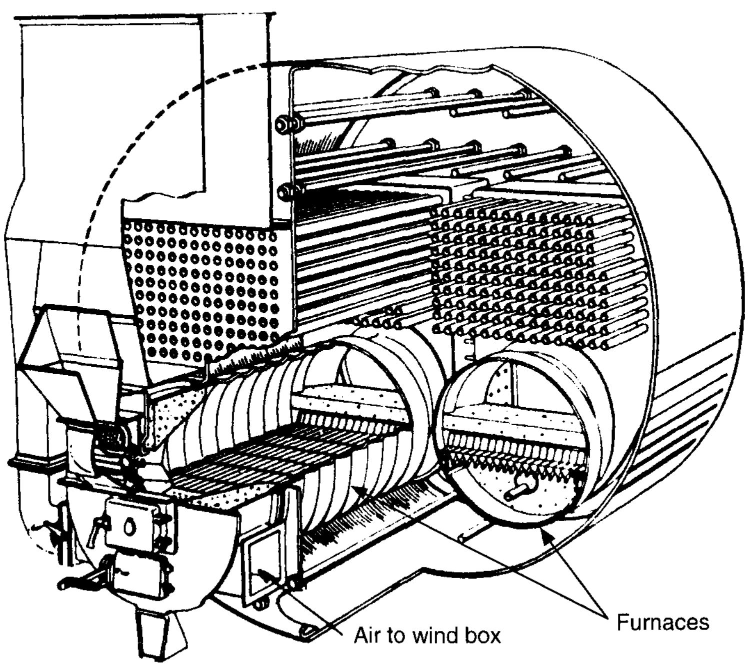


Figure 5-1. Scotch marine boiler with spreader stoker.
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Sectional Header Boilers

One of the popular early water-tube boiler designs was the sectional header boiler. The straight tubes and easy access to each end of the tubes by handholes in the headers makes for easy tube maintenance. There is no need to open and enter the steam drum. Figure 5-2 shows a 250 psig sectional header boiler installed on many of the over 2,500 Liberty ships built during World War II. It is a three-gas pass design with an overdeck superheater. No waterwalls are installed in the furnace.

The sectional header design was developed over the years for higher pressures and higher capacities. Figure 5-3 shows a sectional header boiler suitable for pressures up to 850 psig and 150,000 lbm/hr. It has a single-gas pass, an interdeck horizontal superheater, front and rear furnace waterwalls, and a tubular air heater to improve boiler efficiency.

Bent-Tube Boilers

In the years following World War **II,** the smaller and lighter bent-tube boiler replaced the sectional header boiler for propulsion applications. This design, best characterized by the two-drum D-type boiler, has fewer headers, smaller diameter tubes, and extensive waterwall surface. A bent-tube boiler is smaller and lighter than its sectional header equivalent and has the ability to raise steam pressure more quickly. The greater difference in elevation between the steam drum and the water drums and waterwall headers results in increased natural circulation.

A Combustion Engineering V2M-5 boiler is shown in figure 5-4. This design is typical of many built in the 1950s and 1960s. Major features of this design are the vertical superheater with maintenance access space, and the distributing header serving the screen and floor tubes. Tangent tube waterwalls cover the sides, front, and roof. The only significant refractory is on the furnace floor. An economizer is fitted as the final heat recovery element.

Figure 5-5 shows a CE V2M-8 boiler. This is a contemporary design with a number of changes and improvements over the more conventional design discussed above. The membrane waterwalls are airtight, permitting a single casing and minimizing refractory. The membrane waterwall panels are rectangular, facilitating automated assembly and equalizing thermal expansion stresses. The vertical "walk- in" superheater has widely spaced tubes, improving cleaning by the retractable mass-action sootblowers and providing space for slag removal. The furnace and boiler surfaces are arranged to give a gas path free from pockets for effective purging. The boiler can be fired from the roof, front, or side, depending on the particular machinery arrangement.

For higher-power installations, the greater steam production leads to the selection of a dropped furnace design, as shown in figure 5-6. The boiler is supported at the midpoint rather than the bottom. This minimizes the

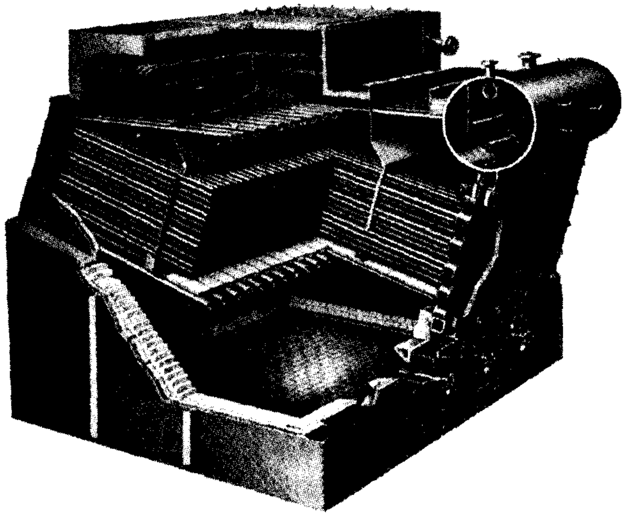

Figure 5-2. Three-pass sectional header boiler.
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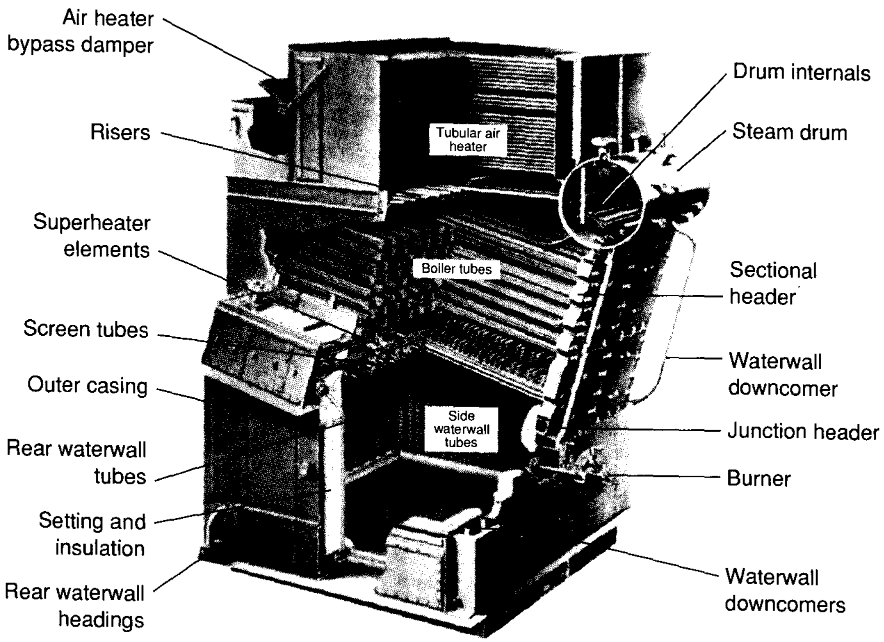

Figure 5-3. Single-pass sectional header boiler.
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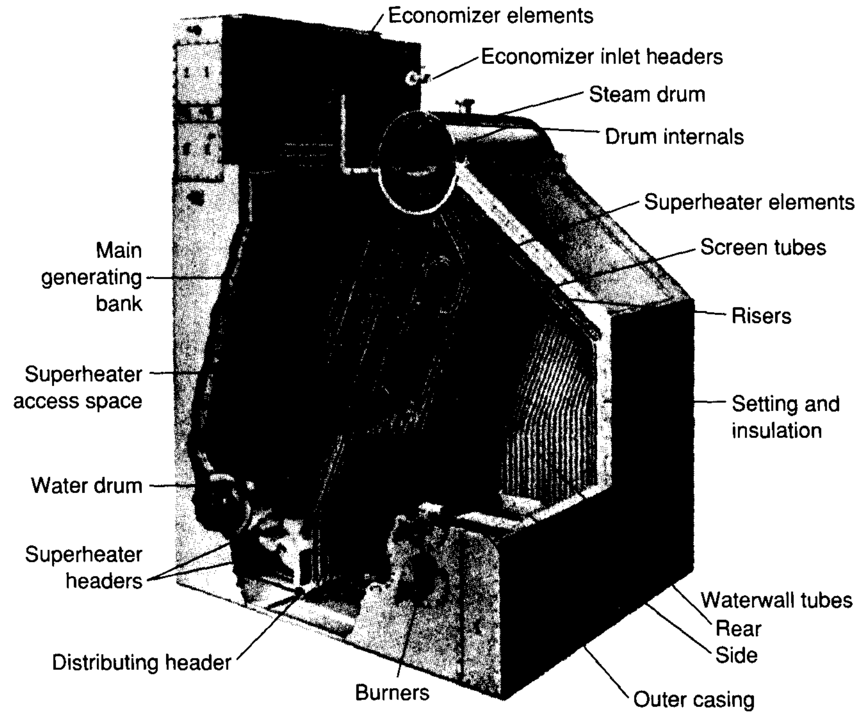

Figure 5-4. CE V2M-5 two-drum boiler. Reprinted with the permission of Combustion Engineering, Inc.

vibration problems than can occur in large bottom-supported units. The dropped furnace also permits the use of tangential firing. Burners are located in each of the four corners of the furnace, providing excellent mixing ofthe fuel and air and allowing operation at excess air levels of only 3 to 5 percent. This improves efficiency and reduces cold-end corrosion of the air heater and economizer.

Reheat Boilers

In the reheat cycle, steam that has been partially expanded in the high-pressure turbine is returned to the boiler to increase its temperature, typically to the same temperature level as the steam leaving the superheater. A boiler designed for use in a marine reheat cycle has a problem that its shoreside equivalent does not. A marine power plant must occasionally go astern and when it does so, there is no steam flow through the reheater. A similar situation exists during port operation. Means must be provided to protect the reheater to avoid overheating of the reheater tubes.

Figures 5-7 and 5-8 show the Foster Wheeler ESRD reheat boiler. The approach to protecting the reheater taken in this design is called gas

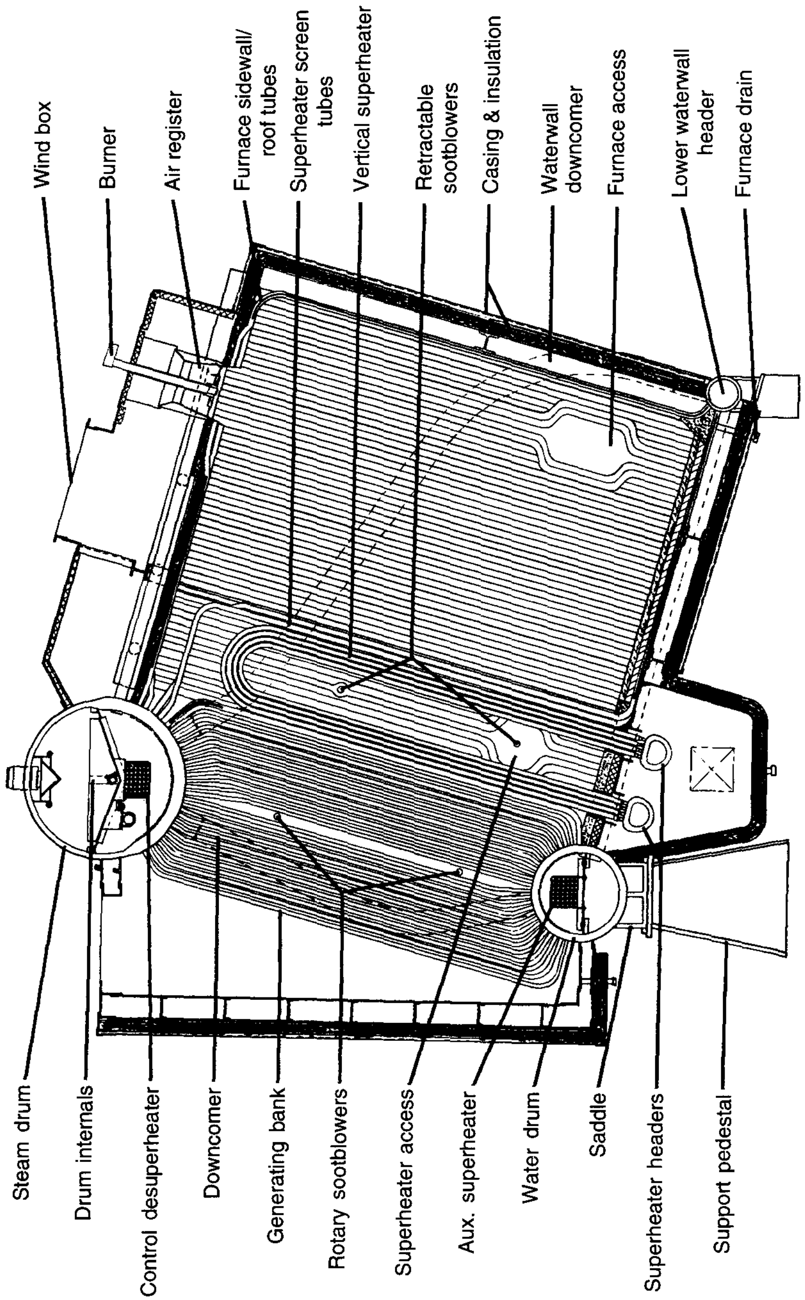

Figure 5-5. CE V2M-8 boiler. Reprinted with the permission of Combustion Engineering, Inc.

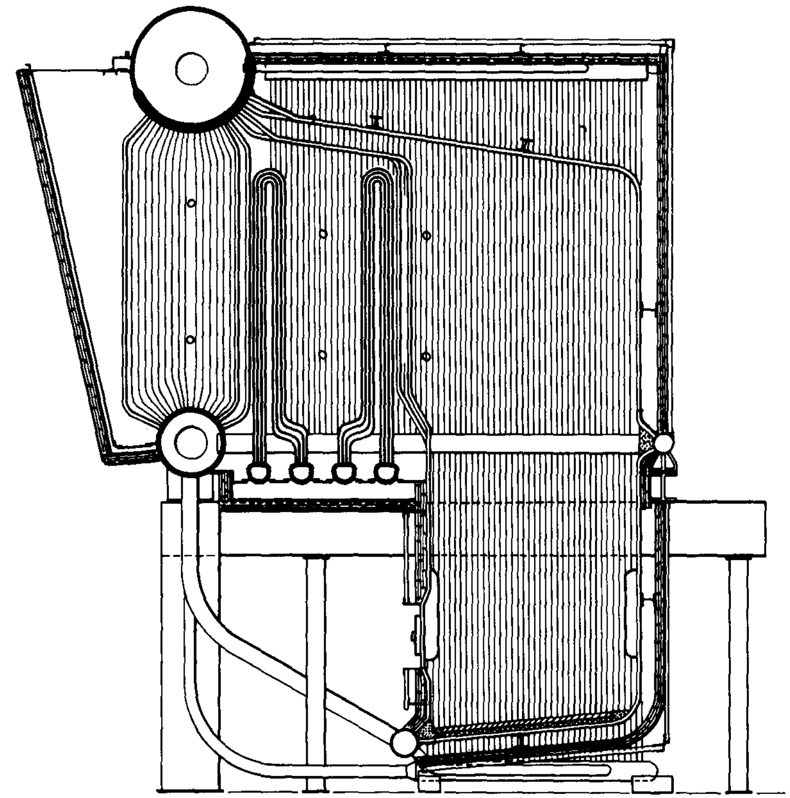

Figure 5-6. CE V2M-9 boiler. Reprinted with the permission of Combustion Engineering, Inc.

bypass. The gases leaving the furnace can take two paths. One path contains the superheater and reheater surfaces; the other contains superheater and bypass economizer surfaces. Dampers are fitted to control the gas flow through each path. During astern or port operation, the reheat shutoff damper is closed, stopping flow across the reheater. In addition, relatively cool air from the air heater is admitted to further protect the reheater. Hot gases flowing across the superheater pass through an opening in the bypass wall, maintaining superheat during astern operation. During normal ahead operation, the superheat and reheat control dampers vary the balance of gas flow through the two paths, thus providing a means of controlling the reheat temperature.

Combustion Engineering took a different approach for reheater control and protection. Figure 5-9 shows CE V2M-8 LTG and V2M-9 LTG boilers. A separately fired, water-cooled reheat furnace has been added after the main generating bank. During ahead operation, fuel flow is divided between

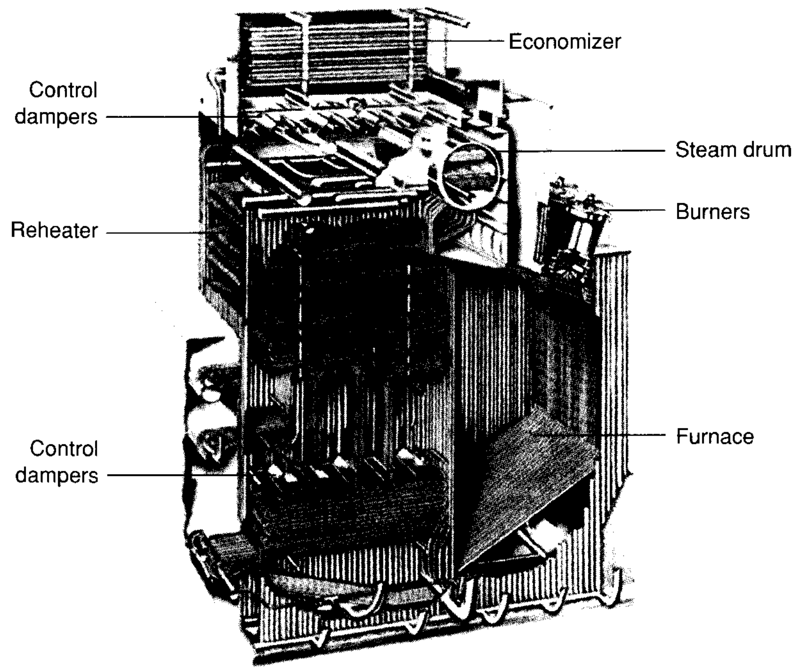

Figure 5-7. Foster Wheeler ESRD reheat boiler. Illustration courtesy of Foster Wheeler Corp.

the main furnace and the reheater furnace. The reheat outlet temperature can be controlled by varying the fuel flow to reheat furnace. During astern operation or in port, the fuel flow to the reheat furnace is secured. The reheater is thus subjected to the relatively low-temperature gases leaving the generating bank. No cooling steam or other reheater protection is necessary.

Coal-Fired Boilers

Boilers using coal as a fuel differ somewhat from the oil-fired boilers described above. Volume 2 of *Modern Marine Engineer's Manual* describes typical coal-fired marine propulsion boilers from Combustion Engineering and Foster Wheeler.

Auxiliary Boilers

Auxiliary boilers are found on a variety of vessels to supply lower-pressure steam for heating and other auxiliary services. Fire-tube, water-tube, and forced-circulation boilers may be selected depending on the required pressure and capacity. Lower-capacity boilers are commonly supplied as packaged units, complete with controls, combustion air fan, fuel oil pump, fuel oil heater, and sootblowers.

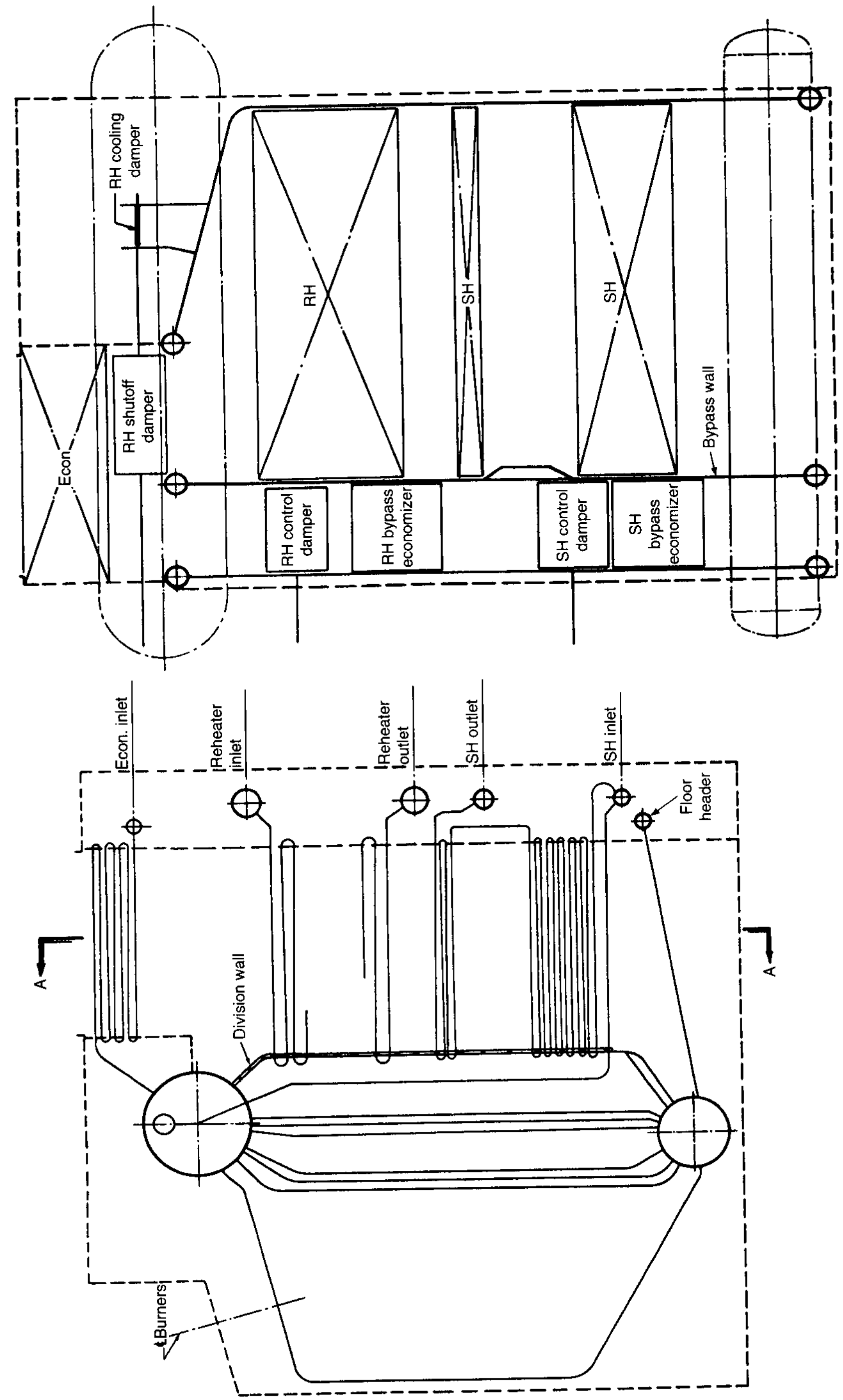

Figure 5-8. ESRD reheat boiler sectional elevations. Illustration courtesy of Foster Wheeler Corp.

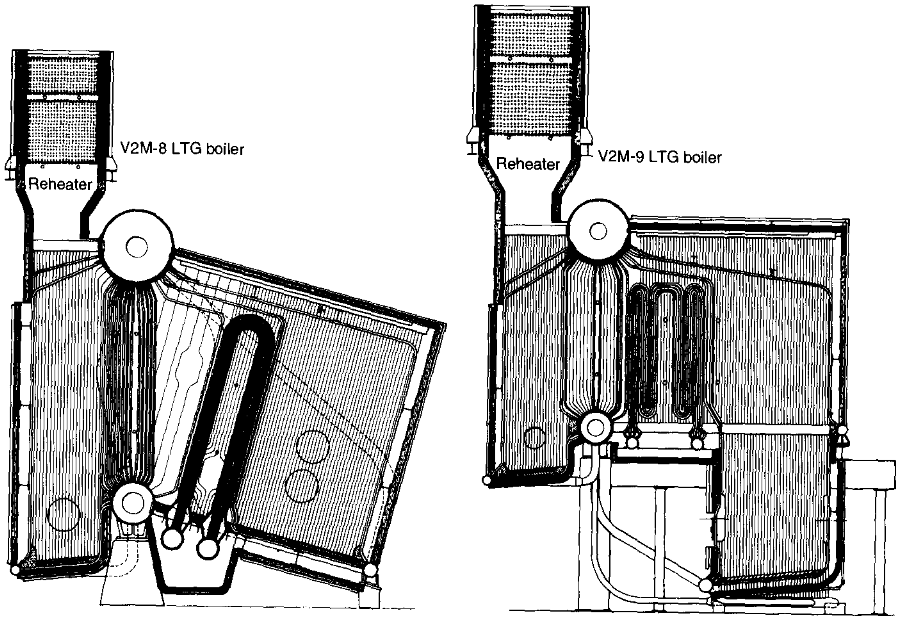

Figure 5-9. CE LTG reheat boilers.
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Figure 5-10 shows a typical water-tube auxiliary boiler, a Combustion Engineering model 2VPM. The boiler is two-drum, D-type construction with natural circulation and partial waterwalls in the furnace. A comparison with a bent-tube propulsion boiler will reveal many similarities.

Figure 5-11 shows a Clayton forced-circulation steam generator. A pump circulates water at high velocity through a series of coiled small diameter steel tubes. The gases of combustion pass across the coils, turning some of the water to steam. The steam-water mixture exits into the steam separator. The steam exits at the top, and the water drops to the bottom and is recirculated back to the coiled tubes. The level in the steam separator is maintained constant by a feed pump and an automatic level control system. With their small amount of water in circulation and their high circulation rates, boilers of this type permit steam to be raised rapidly. This makes the unit ideal for dealing with intermittent steam demands. The disadvantage is that even a small interruption of the feed flow can be disastrous. The makeup feedwater supply must be reliable and it is essential that the sophisticated automatic controls be properly maintained.

Waste-Heat Boilers

The installation of a waste-heat boiler in the exhaust of the main propulsion diesel engine is a common practice on motor ships. Waste-heat boilers

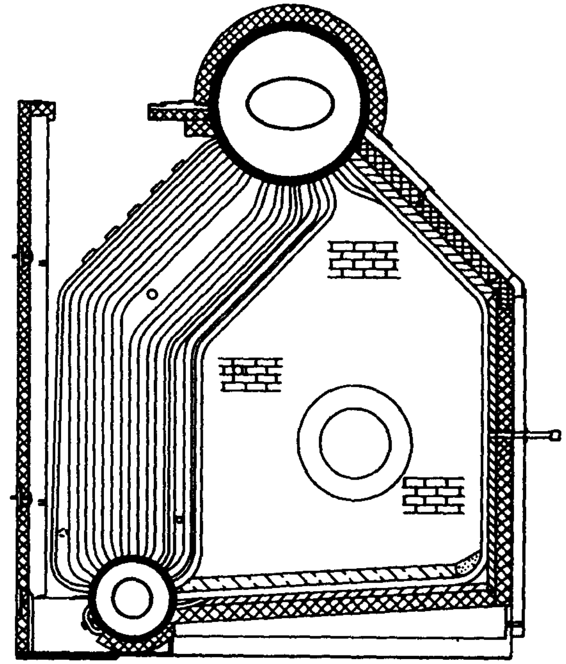

Figure 5-10. Auxiliary two-drum boiler.
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can also make a significant contribution to the performance of gas turbine main propulsion systems. In most merchant ship applications, the exhaust gas boiler can produce all required steam during at-sea operation, improving the overall plant efficiency. An oil-fired auxiliary boiler is still required for low-power and port operation.

Figure 5-12 shows a typical forced circulation water-tube waste-heat boiler. The heat exchanger is of the multiloop type. Some exhaust heat exchangers are fitted with extended surfaces, similar to those used in economizers. Mild steel fins may be used in the high-temperature regions, while cast-iron gills are commonly fitted in the lower temperature regions to minimize corrosion.

Waste-heat boilers require controls to maintain steam pressure under varying steam demands and main engine power. If the waste-heat boiler is generating more steam than is required, excess steam can be dumped to a condenser, or a gas bypass damper can be installed to reduce the flow of exhaust gases across the heat exchanger. If the steam demand is greater than the production of the waste-heat boiler, additional steam must be produced from another source. The simplest system consists of separate waste-heat and auxiliary boilers, each feeding the common auxiliary

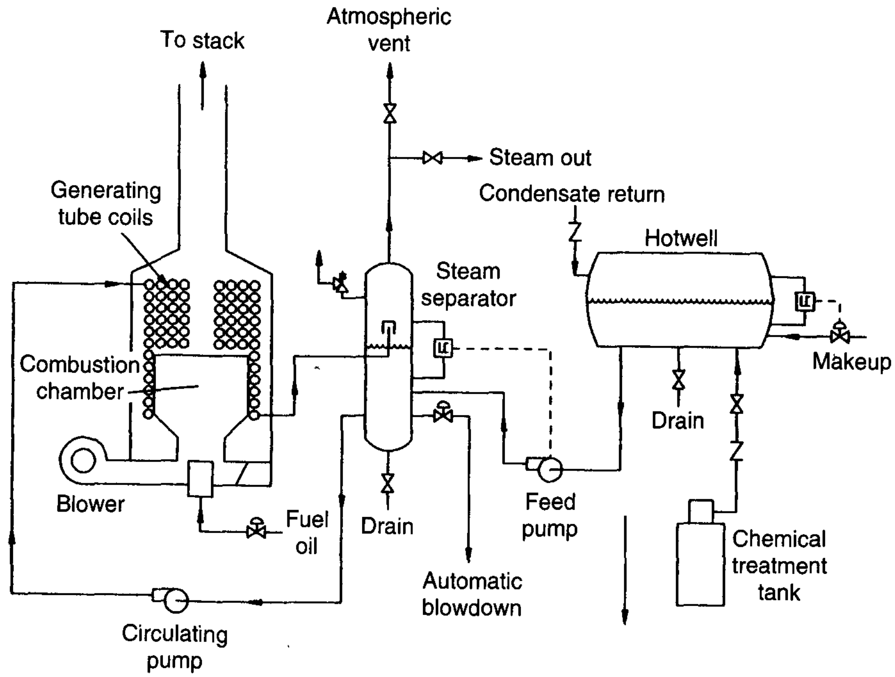

Figure 5-11. Clayton steam generator

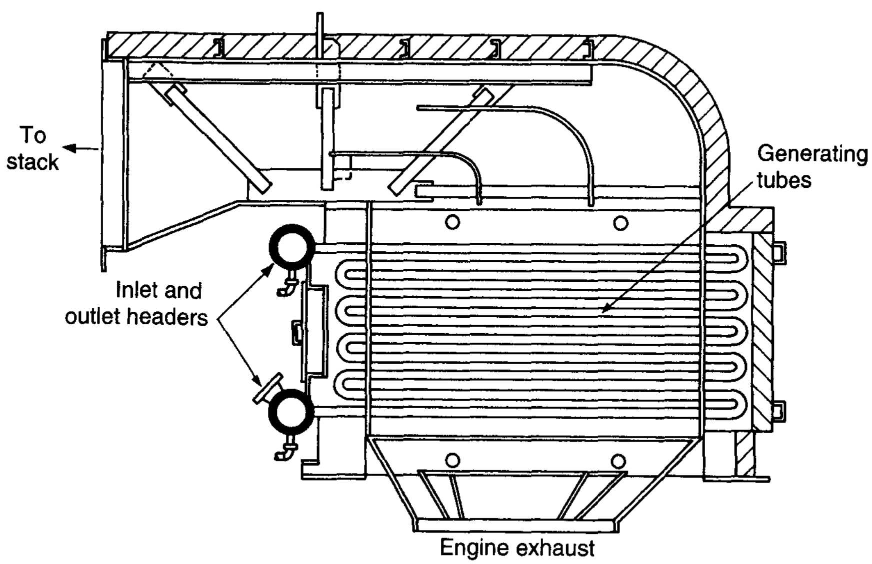

Figure 5-12. Waste-heat boiler. Reprinted with the permission of Combustion Engineering, Inc.

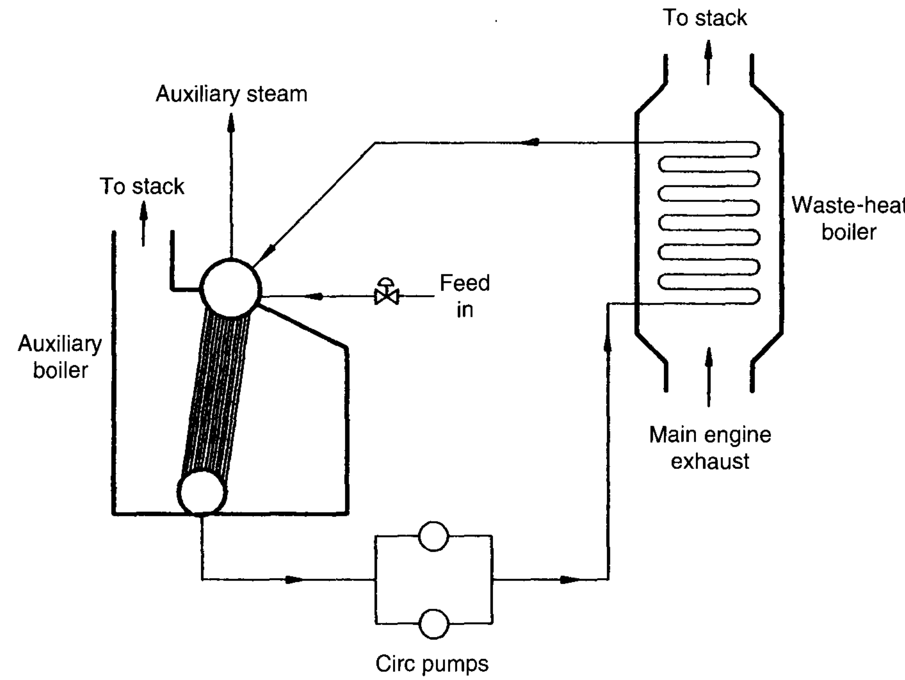

Figure 5-13. Combined auxiliary and waste-heat boiler installation

steam line. The auxiliary boiler will be automatically started ifthe steam line pressure drops too low. Another common arrangement is to use the steam drum of the auxiliary boiler as a steam receiver for the waste-heat boiler (see fig. 5-13). One advantage of this approach is that the oil-fired boiler is kept in a warm standby condition ready for immediate use. Another approach is to install an oil burner in the main engine stack at the boiler inlet and "fire" the exhaust gas boiler.

Some motor ships are fitted with steam turbine generators to supply electric power at or near normal service speed. The exhaust gas boilers on such an installation are typically fitted with a superheater to improve the turbine efficiency. An economizer section can also be installed. To further increase the efficiency, a dual pressure boiler can be used. The two sets of generating tubes produce higher-pressure steam for the steam turbine and lower-pressure steam for auxiliary uses. Figure 5-14 shows a diagram of such an arrangement.

COMBUSTION OF FUEL OIL

Combustion in a boiler is the rapid chemical reaction of the fuel with the oxygen in the air, accompanied by the release of large quantities of heat.

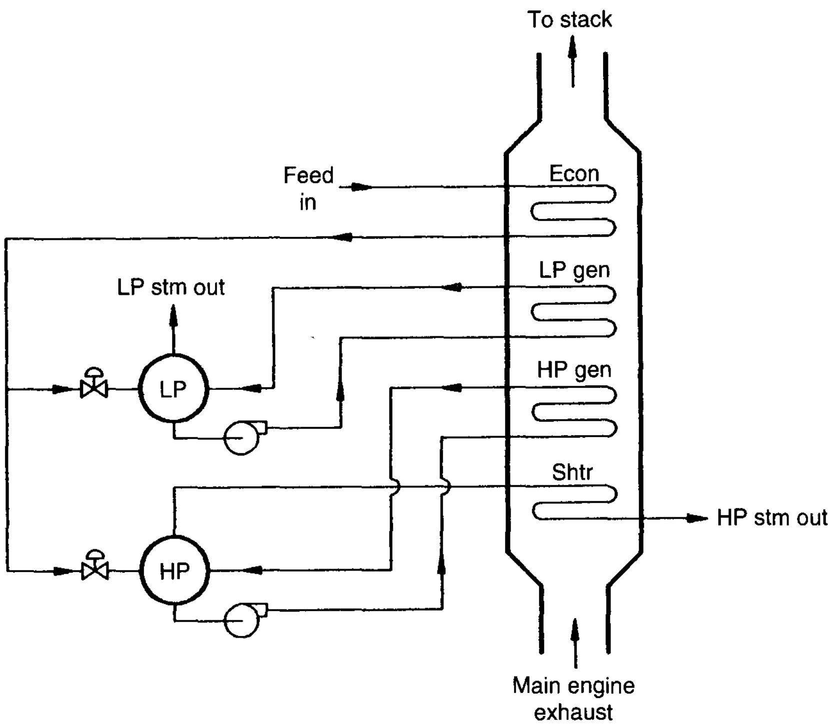

Figure 5-14. Dual-pressure waste-heat boiler system

Fuel oil is burned in a boiler by igniting a mixture of an atomized spray of oil with a high-velocity stream of air. The important elements for good combustion of fuel oil are proper atomization of the oil and the complete mixing ofthe correct quantity of air with the atomized fuel. For the combustion of fuel to be complete, "excess air" (more air than the amount theoretically needed) must be admitted. Too little excess air results in incomplete combustion and less heat released from the fuel. Too much excess air increases the flow of gases out of the stack and reduces the efficiency of the boiler.

In the following sections, the combustion reactions will be analyzed, the procedures for analyzing stack gases and determining boiler efficiency will be presented, and the various types of oil burners will be discussed.

Combustion Analysis

Carbon, hydrogen, and sulfur are the three primary combustible elements in fuel oil that release heat during combustion. Air is a mixture of oxygen and nitrogen. The oxygen in the air reacts with the carbon, hydrogen, and sulfur in the fuel as follows:

C + Oz -7 CO_Z + 14,100 Btu/lbm of carbon

2Hz + 0 z -7 $2H_ZO$ + 61,100 Btu/lbm of hydrogen

8 + Oz -7 80_Z + 4,000 Btu/lbm of sulfur

If sufficient air for complete combustion is not supplied, some ofthe carbon will react to form carbon monoxide as follows:

2C + Oz -7 2CO + 3,960 Btu/lbm of carbon

By comparing the heat released per Ibm of carbon in the fuel for producing carbon dioxide versus carbon monoxide, it is easy to see why sufficient excess air is essential for efficient boiler operation.

By using the above balanced equations, it is possible to determine the mass of air necessary for theoretical combustion of a particular fuel. This is referred to as the stoichiometric air/fuel ratio. Converting the above equations to a mass basis using the atomic masses for carbon (12), hydrogen (1), sulfur (32), and oxygen (16):

12 Ibm C + 32 Ibm Oz -7 44 Ibm CO_Z

4 Ibm Hz + 32 Ibm Oz -7 36 Ibm HzO

32 Ibm 8 + 32 Ibm Oz -7 64 Ibm 80_Z

This shows that for complete combustion of one pound of carbon, 2.667 pounds of oxygen are required. Similarly, 8 pounds of oxygen are required per pound of hydrogen, and 1 pound of oxygen required per pound of sulfur. Since air is 23.15 percent oxygen by mass, dividing the pounds of oxygen required for each reactant by 0.2315 shows 11.52 pounds of air are required per pound of carbon, 34.56 pounds of air are required per pound of hydrogen, and 4.32 pounds of air are required per pound of sulfur. The pounds of air required for theoretical combustion of one pound of fuel can thus be estimated using the following equation:

Ibm air/Ibm fuel = 11.52C + 34.56H + 4.328

where C, H, and S are the pounds of each element in one pound ofthe fuel. The ultimate analysis of a particular fuel will contain this information, along with the mass percentages of other constituents such as oxygen and nitrogen.

Ifit is assumed that any oxygen in the fuel is combined with some of the hydrogen as water, the equation above is modified as follows:

Ibm air/Ibm fuel = 11.52C + 34.56(H -0/8) + 4.328

The following example illustrates the use of this equation.

EXAMPLE: A fuel has the following ultimate analysis.

Carbon (84%)
Hydrogen (11%)
Sulfur (3%)
Oxygen (0.5%)
Nitrogen and inert material (remainder)

Determine the stoichiometric air/fuel ratio and the air/fuel ratio for 15 percent excess air.

Solution:

$$\text{Ibm air/Ibm fuel} = (11.52)(.84) + (34.56)(.11-0.005/8) + (4.32)(.03)$$
$$= 13.59 \text{ Ibm air/Ibm fuel}$$

For 15 percent excess air, the air/fuel ratio is

$$(13.59)(1.15) = 15.63 \text{ Ibm air/Ibm fuel}$$

Note that this is based on dry air, i.e., zero humidity. The actual pounds of moist air delivered to the burner will be slightly higher, typically about 1 to 2 percent.

Stack Gas Analysis

The importance of operating with the proper amount of excess air was discussed above. The usual technique for adjusting the air flow is to observe the stack opacity by using the smoke indicator (periscope). Too little air will result in black smoke. Too much air will result in a clear stack or, in certain extreme cases, white smoke. The proper condition is a light brown "efficiency" haze. This appears in the periscope as a flickering of the light. While an effective first step, this approach gives no indication at what level of excess air incomplete combustion begins. A variety of reasons can cause this to occur at a level above design, such as improper oil viscosity, a worn burner tip, or an improperly adjusted burner. The only way to determine if the burner is operating at design efficiency is to measure the excess air. This is done by performing a stack gas analysis.

An examination of the equations in the above section will reveal that the products of combustion are carbon dioxide, water vapor, and sulfur dioxide. Nitrogen and oxygen will be found also, since air is used to react with the fuel. If the combustion is incomplete, carbon monoxide will also be found. It is common practice to measure the oxygen and/or the carbon dioxide to determine the excess air. Figure 5-15 shows the relationship between oxygen and carbon dioxide in stack gas for a typical residual fuel oil.

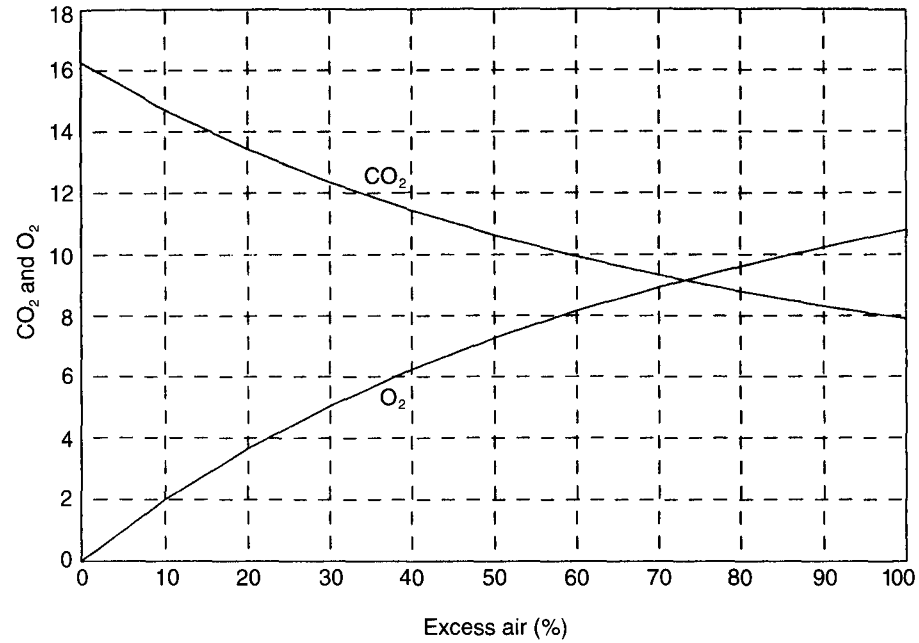

Figure 5-15. Oxygen and carbon dioxide in boiler stack gas

Note how the oxygen percentage increases with excess air but the carbon dioxide percentage decreases. The reason for the increasing oxygen is obvi-0us-more excess air means more oxygen in the stack. The carbon dioxide percentage decreases because the fixed amount produced by the combustion of a given amount of carbon is diluted by more nitrogen and oxygen as the excess air increases.

The device traditionally used to measure the percentages of oxygen, carbon dioxide, and carbon monoxide is the Orsat stack gas analyzer (fig. 5-16). The apparatus consists of a graduated measuring burette and three absorption pipettes. Each absorption pipette contains a chemical reagent to absorb one ofthe gases. Briefly, the operation of the analyzer consists of passing a 100-milliliter sample of stack gas successively through the three absorption pipettes, and measuring the reduction in volume of the gas sample at each step.

Another available gas analyzer is the Bacharach Fyrite unit shown in figure 5-17. It operates on the same chemical absorption principle as the Orsat analyzer, but there are separate units for measuring the carbon dioxide and the oxygen. A gas sample is pumped from the stack into the analyzer, the unit is inverted several times, and the volume reduction read directly as percent C02 or 02. It should be noted that for both the chemical analyzers discussed, any water vapor in the stack gas will condense out in

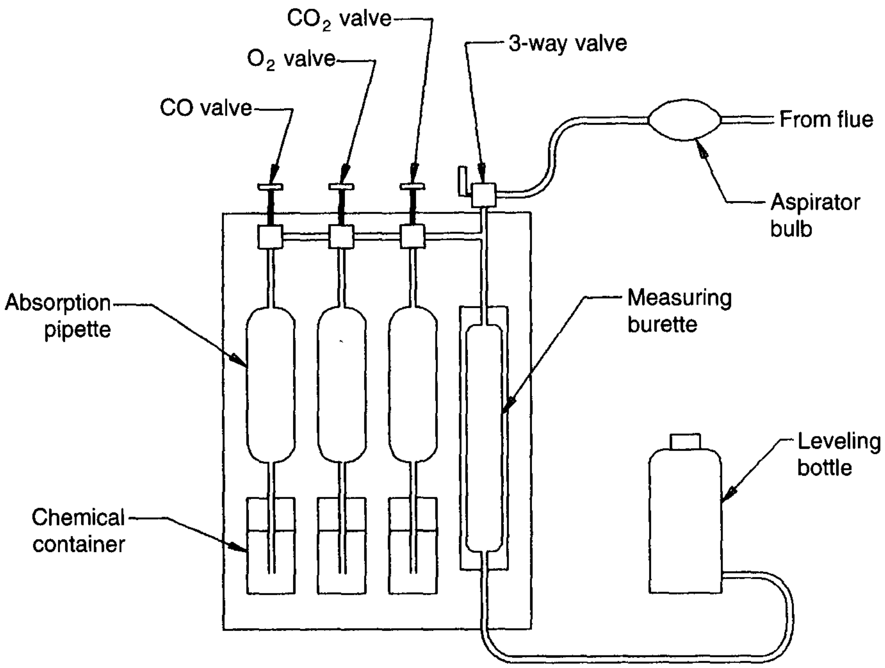

Figure 5-16. Orsat stack gas analyzer

the sampling system or into the chemicals. The volume percentages measured are thus referred to as "dry basis" since only the noncondensable gases are considered.

In recent years, electronic analyzers have become popular for measuring the oxygen content of stack gas based on a zirconium oxide cell. The "in-situ" type is mounted directly in the stack and provides a continuous readout of the oxygen content. Figure 5-18 shows a typical in-situ type sensor. In addition to providing a measurement of the stack gas oxygen content, the sensor can be interfaced to the combustion control system, providing automatic control of the excess air.

Boiler Efficiency

The boiler efficiency is a measure of the percentage of heat supplied to the boiler, primarily by the combustion of fuel (heat-in) that gets transferred to the feedwater and becomes steam (heat-out). Referring to figure 5-19, the boiler efficiency can be defined as follows:

$$\text{Boiler efficiency} = \frac{\text{Heat-out}}{\text{Heat-in}}$$

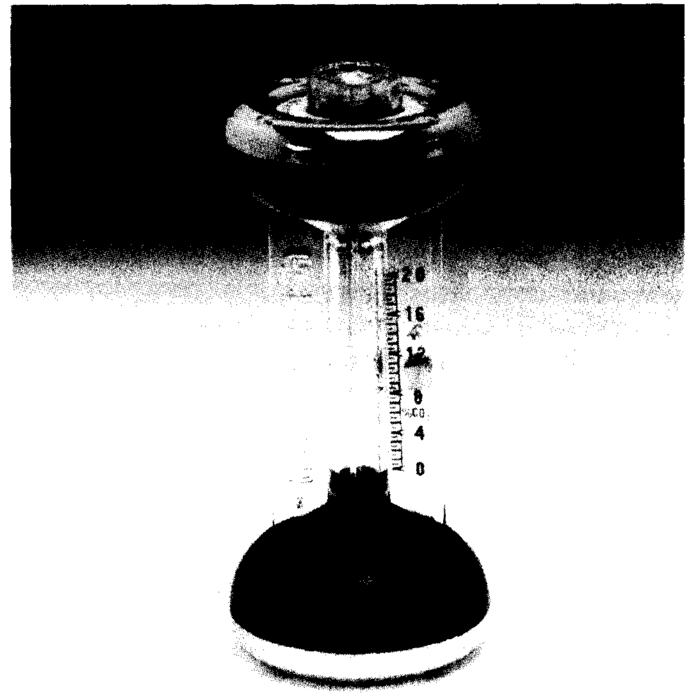

Figure 5-17. Fyrite gas analyzer. Courtesy Bacharach, Inc.

Referring again to figure 5-19, it can be seen that the heat-out is equal to the heat-in minus the losses. The boiler efficiency can thus also be defined as follows:

$$\text{Boiler efficiency} = \frac{\text{Heat-in} - \text{losses}}{\text{Heat-in}}$$

Since the boiler losses can be determined more easily and accurately than the heat transferred to the feedwater, the second definition is more useful in measuring the efficiency of an operating boiler.

The major loss from a boiler is the stack loss. It typically amounts to about 8 to 10 percent of the heat input in a propulsion boiler. All the other losses typically account for only another 1½ percent. This translates to a boiler efficiency of about 88 percent for older boilers, and slightly over 90 percent for the highest efficiency modern boilers.

The stack loss is basically a function of two things-the stack temperature and the excess air. The higher the stack temperature, the more heat is lost up the stack. An increase in excess air will increase the mass flow of

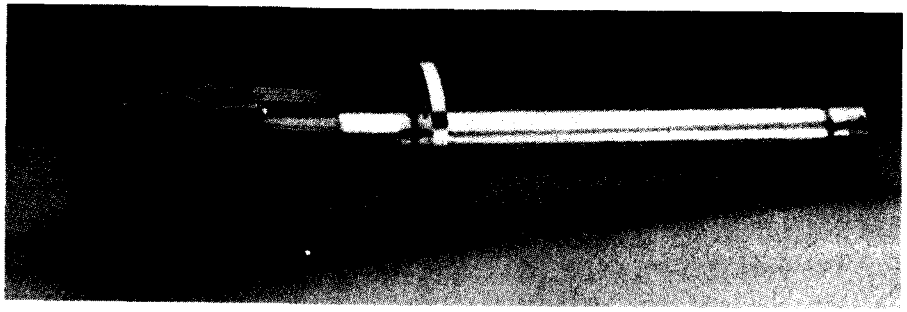

Figure 5-18. In-situ zirconium oxide oxygen sensor. Courtesy Rosemount Analytical, Inc.

stack gases, thus also increasing the stack loss. Figure 5-20 shows the relationship between stack temperature, excess air, and boiler efficiency for a propulsion boiler burning residual fuel oil. It assumes that the radiation and unaccounted-for losses are 1 1/2 percent. Note that a boiler with a stack temperature of 325°F and 15 percent excess air will have an efficiency of 88.5 percent, typical of an older propulsion boiler. A modern boiler operating with a stack temperature of 250°F and 5 percent excess air will have an efficiency of 90.3 percent, an almost 2 percent improvement. Figures 5-15 and 5-20 can be used to calculate the efficiency of an operating propulsion boiler. First, the stack temperature is measured. Second, the excess air percentage is determined by performing a stack gas analysis. Figure 5-15 is then used to determine the excess air based on the measurements of O_2 and/or CO_2. Entering figure 5-20 with the stack temperature and excess air percentage, the boiler efficiency can be determined.

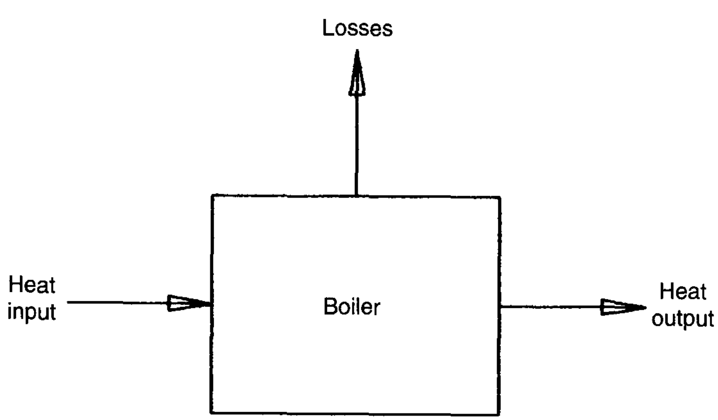

Figure 5-19. Definition of boiler efficiency

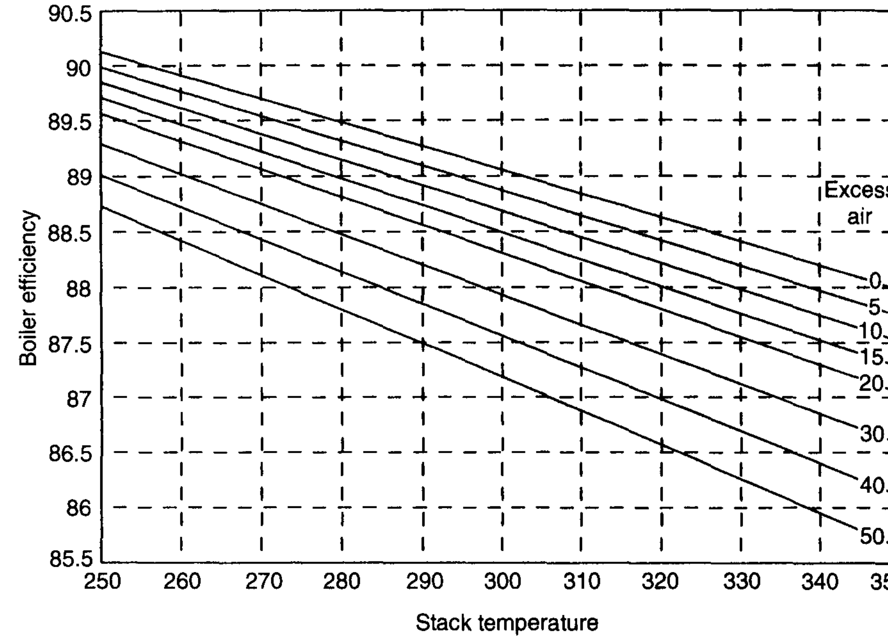

Figure 5-20. Boiler efficiency versus excess air and stack gas temperature

Fuel Oil Burners

The fuel oil burner has the important task of bringing the fuel and combustion air together and mixing them as completely and efficiently as possible. The burner must be capable of producing efficient firing over a wide range of oil flow rates and be able to change load rapidly to meet maneuvering requirements. The burner consists of two major components-the air register and the oil atomizer.

The register consists of a series of steel doors which are open when the burner is on (admitting air to the burner) and closed when the burner is off (stopping the airflow). The doors can be arranged radially (fig. 5-21) or circumferentially (fig. 5-22). A lever on the burner front opens and closes the register doors. The lever is moved either manually or by a remotely controlled actuator. The register doors should never be throttled to vary the air flow to the burner. Air flow control is accomplished at the forced draft fan, either by opening and closing the fan outlet damper or the fan inlet vanes, by varying the fan speed, or by some combination of these means.

The fuel oil atomizer has the task of breaking the oil flow into a fine spray of tiny oil droplets. The atomization of the oil greatly increases the surface area and allows the combustion process to proceed rapidly to completion. Four types of atomizers are in common use-straight mechanical, return-flow, steam, and rotary cup.

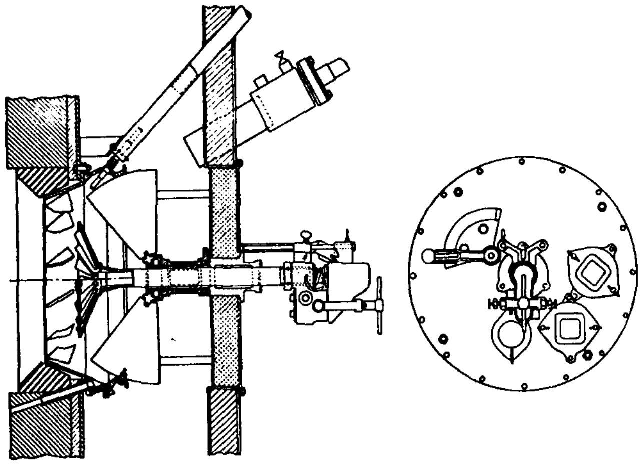

Figure 5-21. Radial door register. Courtesy Babcock & Wilcox.

A straight mechanical atomizer (fig. 5-23) consists of a burner barrel to which a burner tip is attached by a cap nut. The burner tip has a small exit hole in the center with a swirl chamber and several tangential grooves. The oil is atomized as it is forced under high pressure out ofthe small exit hole in the tip. The swirl chamber and tangential grooves cause the oil to exit into the furnace in a hollow rotating cone of oil droplets. The size of the tangential grooves varies the swirl and thus the angle of the oil spray. The size of the exit hole in the tip is expressed in drill numbers. Thus a 27-tip is larger than a 49-tip and will atomize more fuel with the same supply pressure.

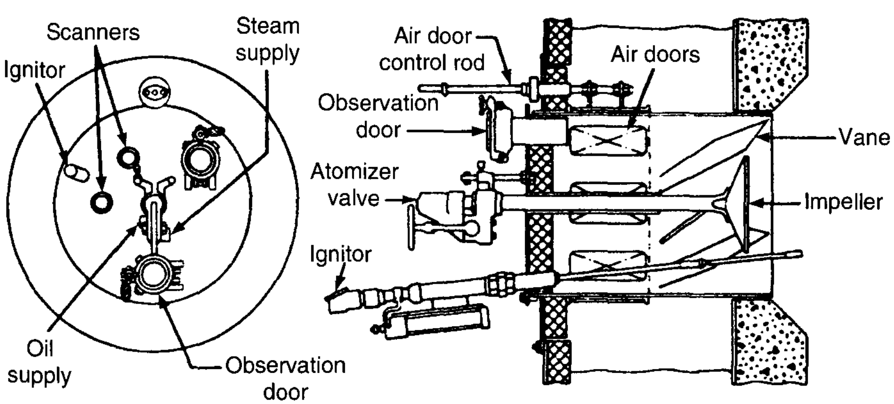

Figure 5-22. Circumferential door register.
Courtesy Babcock & Wilcox.

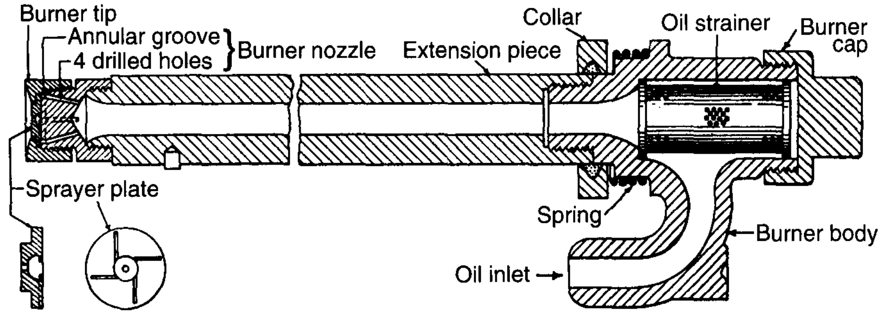

Figure 5-23. Straight mechanical atomizer.
Courtesy Babcock & Wilcox.

The quantity of oil atomized in a mechanical atomizer is controlled by varying the oil supply pressure. In a typical marine fuel oil service system, the service pump pressure will be maintained constant at 300 psig. The fuel oil regulating valve, controlled automatically by the combustion control system, will reduce the oil pressure at the burners. Since the atomizer tip is basically a simple orifice, the following relationship applies:

$$\text{Flow} = C\sqrt{\Delta p}$$

where the Δp is the pressure difference across tip.

Since the energy for atomizing the oil comes from the oil pressure, as pressure is decreased to reduce oil flow, the oil atomization will deteriorate. Typically, the minimum oil pressure needed to maintain adequate atomization is about 100 psig. Thus the "turndown ratio," defined as the maximum oil flow divided by the minimum oil flow, is as follows:

$$\text{Turndown ratio} = \frac{\text{Maximum oil flow}}{\text{Minimum oil flow}} = \sqrt{300/100} = 1.732$$

Thus, if the oil flow must be doubled or cut in half, this cannot be accomplished merely by changing the oil pressure with the fuel oil regulating valve. Burners must be cut in or out to maintain the boiler steam pressure. Changing the tip size will also vary the oil flow, but this requires removing the atomizer and replacing the tip. A common practice on propulsion boilers fitted with mechanical atomizers is to install several different size tips during maneuvering. This provides greater flexibility during rapid load changes. During steady-state operation after departure, all the tips are changed to a single size to maintain the same excess air at each burner.

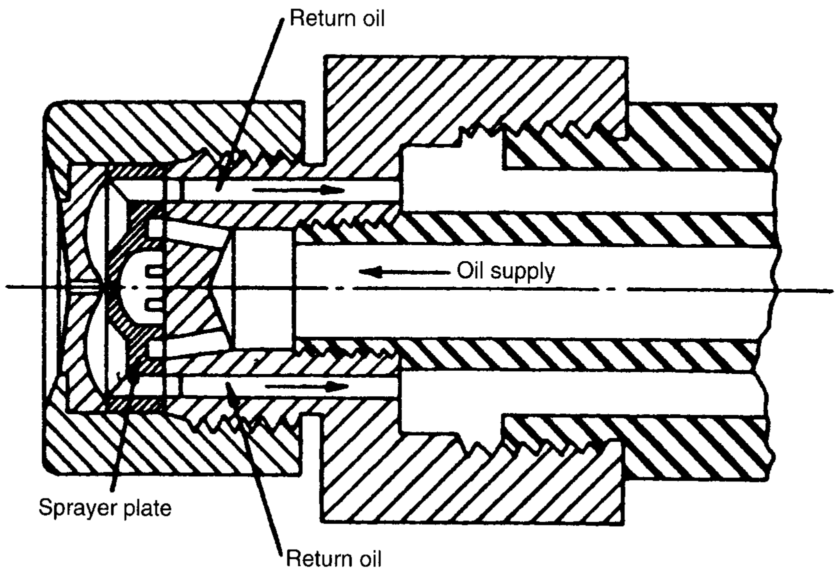

Figure 5-24. Return-flow atomizer.
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The return-flow atomizer was developed to provide a higher turndown ratio than the mechanical atomizer. A typical return-flow atomizer is shown in figure 5-24. Note that there are two oil passages in the atomizer barrel, one for supply oil and one for return oil. Oil is delivered at full supply pressure to the atomizer tip. Some ofthe oil is atomized and some is returned to the service pump. The fuel oil regulating valve is installed in the return-flow line, and the quantity burned in the furnace is controlled by varying the return flow. The atomization remains good as the return flow is increased (and the atomization flow is decreased) because the supply

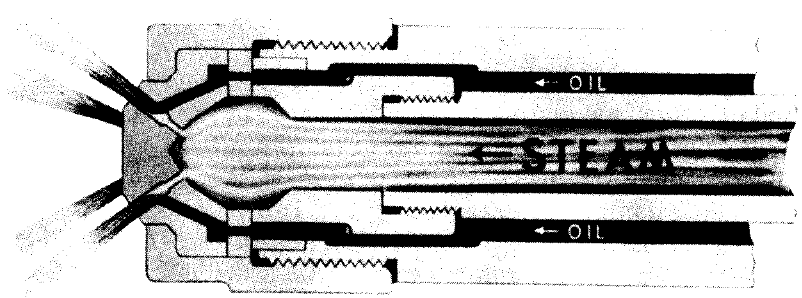

Figure 5-25. Racer steam atomizer. Courtesy Babcock & Wilcox.

pressure remains constant at full service pump pressure. Because of its high turndown ratio, typically 8 to 10, this atomizer design is sometimes referred to as a "wide range" atomizer.

The steam-flow atomizer uses auxiliary steam, typically at 135 to 150 psig, to provide the energy to atomize the oil. Like the return-flow atomizer, there are two passages in the atomizer barrel, one for supply oil and one for atomizing steam. In the Racer-type atomizer shown in figure 5-25, the steam and oil are mixed internally. As the oil-steam mixture exits the tip, the steam expands suddenly, atomizing the oil.

A steam atomizer typically has a turndown ratio of 15 to 20, even higher than a return-flow atomizer. Another advantage ofthe steam atomizer is that the atomizing steam keeps the tip clean. Steam atomizers do not have to be removed and cleaned daily like mechanical or return-flow atomizers. The steam flow also cools the atomizer, avoiding carbonizing of the oil in a secured burner. A typical Racer atomizer tip model number is 6Y-45-55-49-80. These numbers are defined as follows:

6Y (6 exit holes)
45 (drill size of exit holes)
55 (drill size of steam holes)
49 (drill size of oil holes)
80 (included angle of exit holes)

The rotary-cup burner shown in figure 5-26 atomizes the oil by throwing it off the edge of a tapered cup being rotated at 2,000 to 7,000 rpm by an

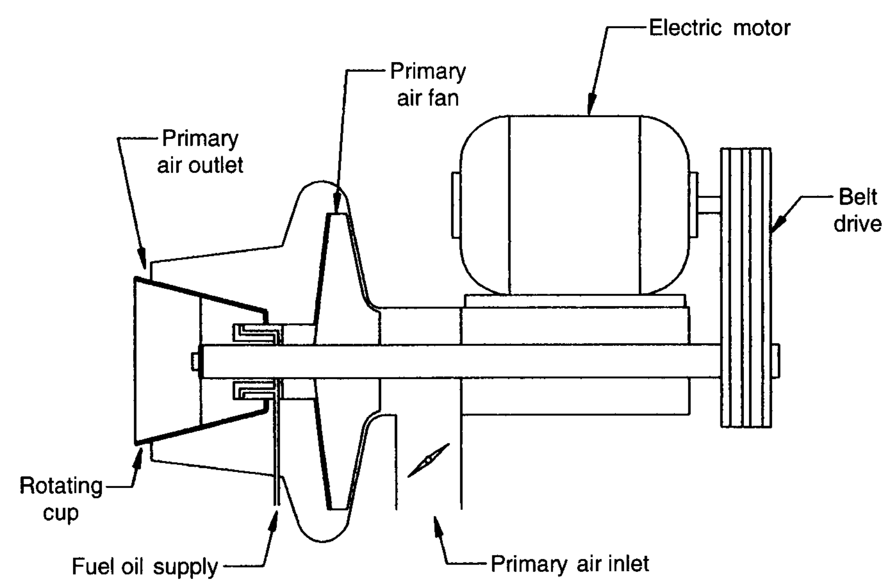

Figure 5-26. Rotary-cup atomizer

electric motor. The motor drives the center spindle via belts or gears. The fuel oil is supplied to the inner surface of the spinning cup. Centrifugal force causes the oil to spread evenly into a thin film which then moves along the taper until it reaches the lip ofthe cup. The radial component of the velocity breaks up the oil into a hollow cone offine particles as it leaves the cup. Also mounted on the center spindle is a fan which provides a small quantity of high-velocity primary air, which mixes with the oil being thrown off the spinning cup. The main air flow enters around the burner housing, mixing with the atomized oil and primary air. Rotary-cup burners typically have turndown ratios of about 10 to 1. Rotary-cup burners are most commonly found on a.uxiliary boilers of European manufacture.

BOILER COMPONENTS AND CONSTRUCTION

Refractory and Insulation

The high temperatures in the furnace of a boiler require special materials called refractories. Some ofthe materials available are firebrick, moldable refractory, castable refractory, plastic refractory, and mortars. Firebricks are manufactured in a variety of different materials for different applications and temperatures, and are fired at high temperatures in special kilns. The firebricks in a furnace wall must be firmly attached to the boiler casing to prevent wall collapse. Figure 5-27 shows a typical metal clip used periodically in the wall to provide support.

The moldable, castable, and plastic refractories are supplied in an unfired state. They are installed in the boiler and fired in-situ when the boiler is put into service. Moldable refractory is used where direct exposure to radiant heat takes place. It must be pounded into place during installation. Castable refractory is normally placed behind waterwalls or to form burner cones. It is installed in a manner similar to forming concrete. Plastic refractory is commonly used in the construction of studded waterwalls. The material is pounded into place onto the steel studs welded to the tubes.

Adequate expansion spaces must be provided to prevent undue thermal stresses as the refractory heats up and expands during boiler operation. Also, vent passages must be made in large castable sections to permit the escape of steam from the material during the initial firing. For maximum refractory life, the boiler should be brought up to pressure slowly when raising steam. If cold air from a secured burner is allowed to impinge on hot refractory, surface flaking known as spalling can occur. Slag formed by vanadium and sodium salts in the fuel can also cause damage to refractory material.

Waterwalls

Many early boilers such as the Liberty ship boiler in figure 5-2 had no waterwall surface. The furnace sides, front, and rear are entirely refractory.

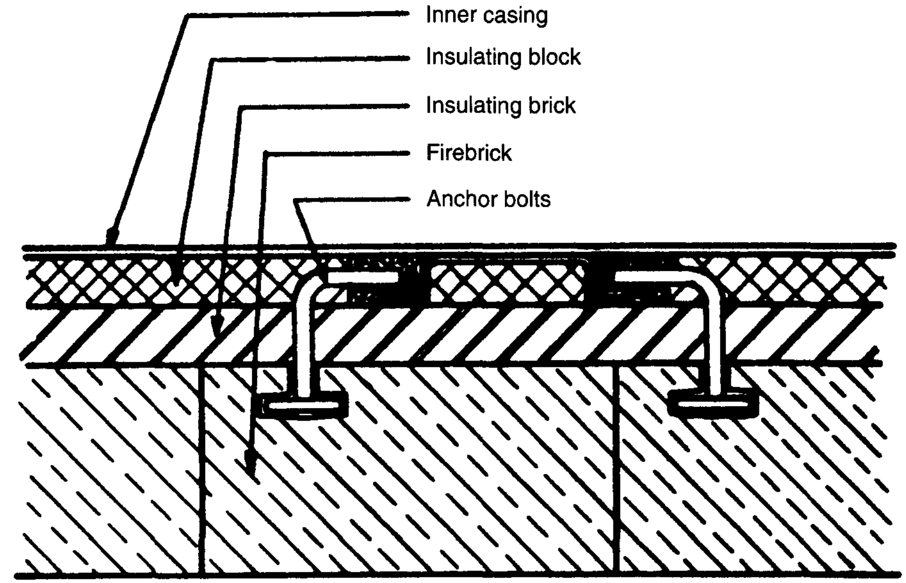

Figure 5-27. Firebrick anchor bolts. Courtesy Babcock & Wilcox.

While such an arrangement simplifies construction, the large quantity of refractory is a high maintenance item. The addition ofwaterwalls to a furnace accomplishes several things-the furnace temperature is lowered, brickwork maintenance is reduced, and the thermal loading on the critical screen tubes is reduced.

Early waterwall designs consisted of widely spaced tubes with studs welded to them and covered with refractory, as shown in figure 5-28. These waterwalls were installed mainly to reduce refractory maintenance and contributed little to steam generation. In later designs, shown in figure 5-29, more closely spaced plain tubes are exposed directly to the heat of the furnace, contributing significantly to the steam production. In current furnace designs, either tangent tube or membrane welded waterwalls are used. Tangent tube construction (fig. 5-30) uses traditional separate tubes spaced so there is no gap between them. In a membrane waterwall (fig. 5-31), a strip of steel is welded between the tubes, producing a strong airtight panel. Welded waterwall construction also reduces boiler weight and facilitates prefabrication, reducing field assembly time and expense. Figures 5-3 to 5-6 show typical arrangements of the waterwall tubes, waterwall headers, and downcomers in some typical marine boilers.

Superheaters and Reheaters

The superheater consists of a series of tubes connected to headers (fig. 5-32). Saturated steam produced in the generating tubes and waterwall

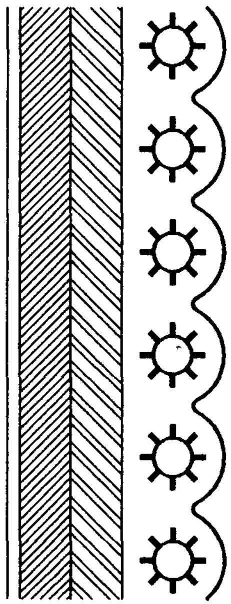

Figure 5-28. Studded waterwall

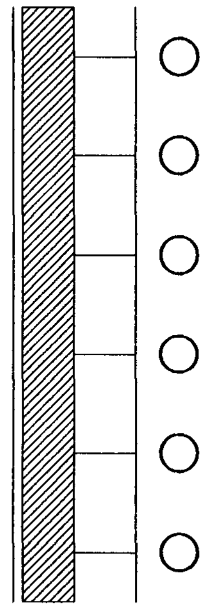

Figure 5-29. Plain tube waterwall

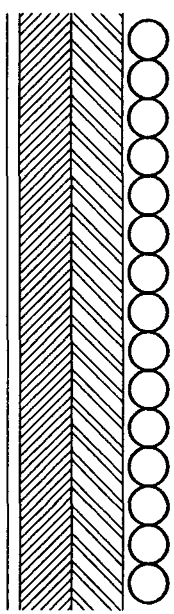

Figure 5-30. Tangent tube waterwall

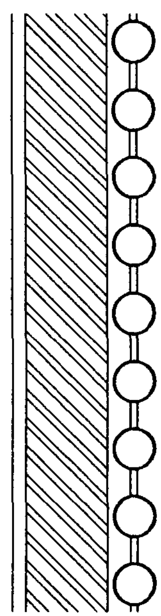

Figure 5-31. Membrane waterwall

tubes exits the steam drum and enters the superheater inlet header. In a single-pass superheater, the steam flows from the inlet header through the superheater tubes into the superheater outlet header. In a multipass superheater, diaphragms are installed at several locations in the headers, forcing the steam to make a number of passes back and forth between the headers before exiting (see fig. 5-33).

Superheaters are available in horizontal and vertical configurations. The sectional header boilers in figures 5-2 and 5-3 have horizontal superheaters. One advantage of a horizontal superheater is that the headers are readily accessible for maintenance. However, they require bulky supports and are prone to slag accumulations. The two-drum boilers in figures 5-4, 5-5, and 5-6 are fitted with vertical superheaters. This eliminates the bulky supports; it is more difficult for slag to accumulate, and since the elements are arranged parallel to the boiler tubes, clear lanes are present for effective action of the sootblowers.

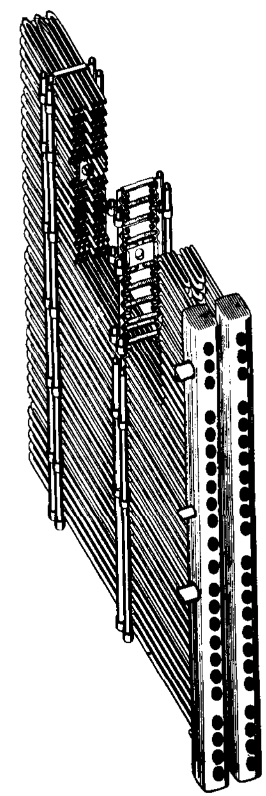

Figure 5-32. Horizontal superheater arrangement.
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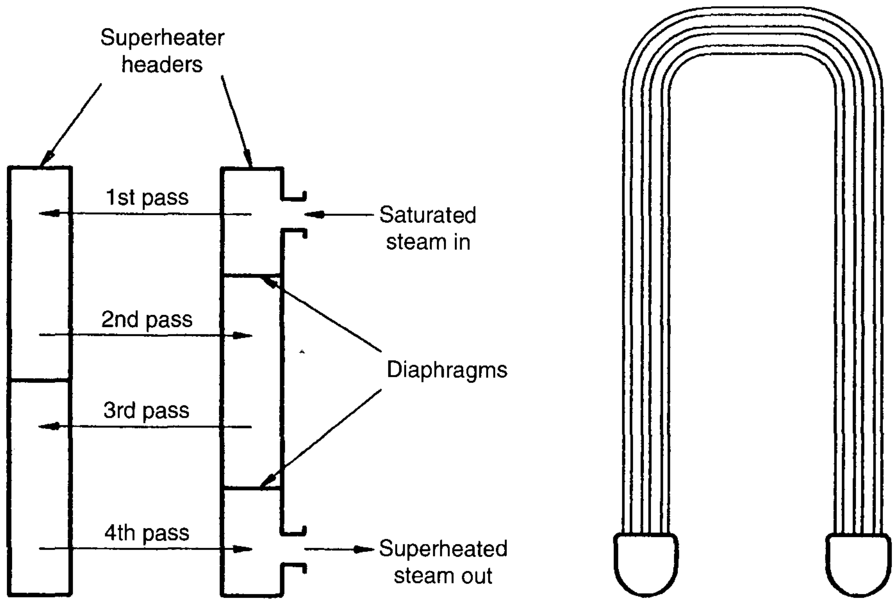

Figure 5-33. Multiple-pass superheater

Superheaters can be classified as either convection or radiant, depending on the primary mode of heat transfer from the combustion gases to the steam. The outlet temperature of a convection superheater tends to increase with boiler load. This is due to the increase in gas temperature and the increases in the gas and steam velocities, which increase the convective heat transfer. The outlet temperature of a radiant superheater tends to decrease with boiler load. This is due to the fact that the furnace temperature is relatively unchanged over a fairly wide range of boiler load. Ideally, a superheater would have a relatively flat temperature versus load characteristic. One way to accomplish this is to combine convection superheater surface with radiant superheater surface. Figure 5-34 shows the temperature characteristics of convective and radiant superheaters.

Marine superheaters are typically located behind only one or two rows of screen tubes. The high gas temperatures at this location produce a large temperature difference between the combustion gases and the steam, resulting in a smaller superheater. Also, locating the superheater closer to the furnace increases the percentage of heat transfer by radiation, producing a flatter steam outlet temperature versus load curve. Figure 5-35 shows the effect on the superheater temperature curve as the number of rows of screen tubes is decreased.

As the superheater outlet temperature has been pushed higher in order to increase cycle efficiency, it has also increased the superheater metal

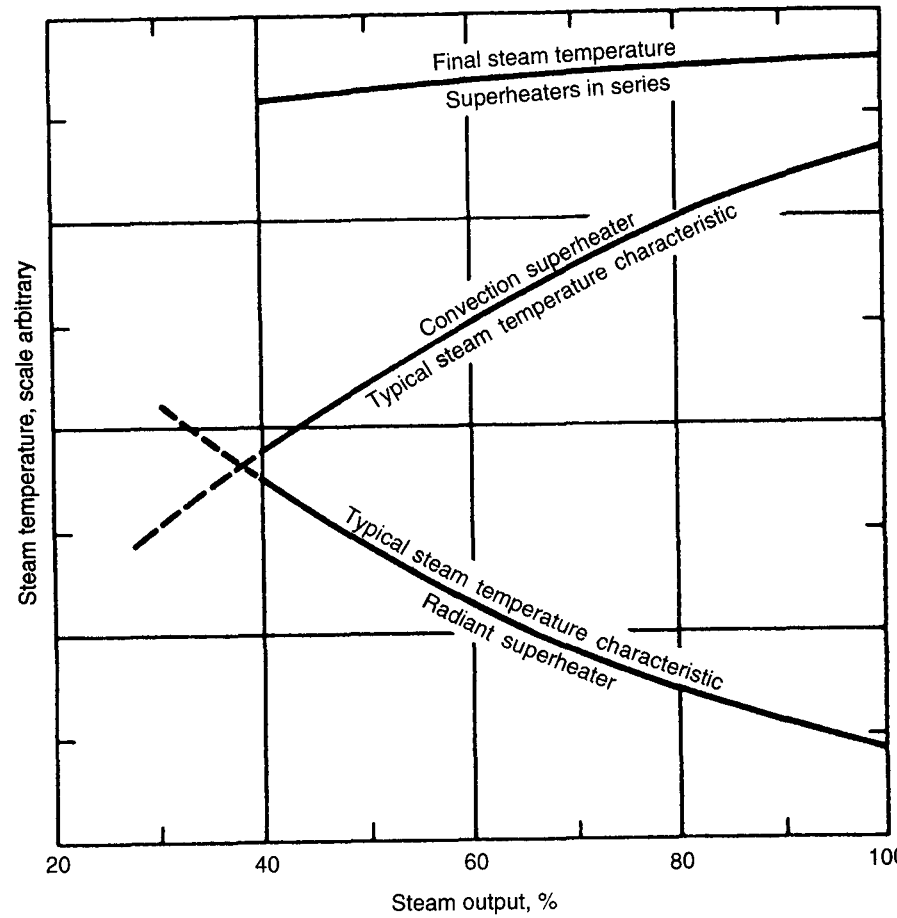

Figure 5-34. Radiant and convective superheater characteristics. Courtesy Babcock & Wilcox.

temperatures. With a superheater temperature of 950°F, tube metal temperatures can exceed 1,000°F. Under normal conditions, and the possibility of exceeding allowable temperature limits is significant. Multipass superheaters are used because the higher steam velocities bring the metal temperature closer to the lower steam temperature. Another potential problem is unequal flow distribution between the tubes in a superheater pass, resulting in overheating of the tubes with low flow. The boiler designer must pay careful attention to the header and tube arrangements to ensure even flow distribution.

At superheat temperatures of 850°F and above, some form of superheat temperature control is required. The use of an attemperator (control

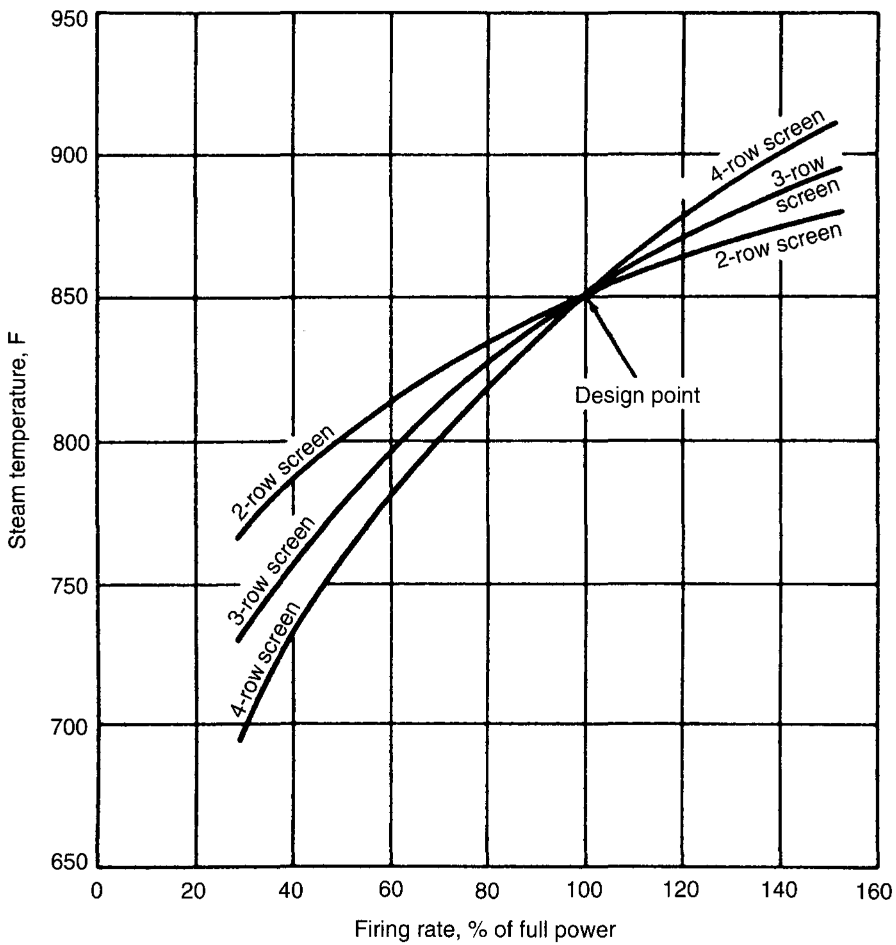

Figure 5-35. Effect of screen tubes on superheater characteristics. Courtesy Babcock & Wilcox.

desuperheater) is most common on marine boilers. Steam is removed from a pass within the superheater, passed through a coil of tubes located in the water drum or the lower half of the steam drum, and returned to a later pass of superheater. Figure 5-36 shows two common attemperator arrangements. In the first, the pressure drop across the third pass is used to cause steam to flow through the attemperator as the control valve is opened. In the second, a three-way control valve is used to proportion the steam flow between the attemperator and the next superheater pass.

Reheaters are similar in design and construction to superheaters. Since reheater steam temperatures are similar to those in superheaters, many of the same design problems and their solutions apply to reheaters.

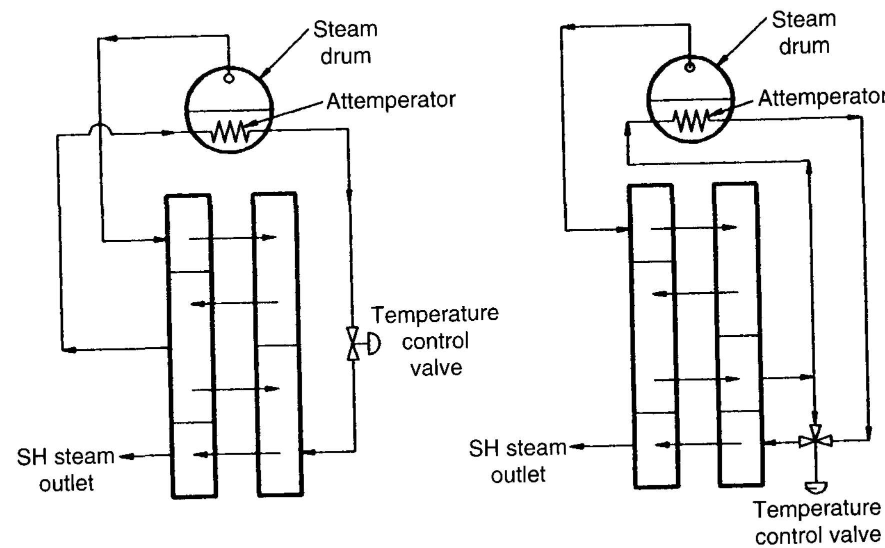

Figure 5-36. Superheater attemperator arrangements

Desuperheaters

Since all the steam generated in a modern boiler must pass through the superheater to avoid overheating it, a desuperheater is fitted to supply the auxiliary steam. As was discussed above, desuperheaters are also used as attemperators for superheat temperature control. Desuperheaters can be of the internal or the external type. The internal type consists of a series of tubular elements installed within the water drum or water space of the steam drum. Figure 5-37 shows a typical arrangement of an internal type desuperheater. Steam is cooled to within about 50°F of the saturation temperature corresponding to the drum pressure.

External desuperheaters are typically of the spray type with feedwater used for cooling. Figure 5-38 shows a common spray-type desuperheater. It consists of a mixing tube and a thermal sleeve, a spray nozzle, and a temperature control valve for regulation of feedwater flow. As the feedwater is sprayed into the superheated steam flow, the temperature of the steam is reduced and the feedwater receiving this heat is vaporized. To avoid overfeeding of feedwater, the temperature control valve must be adjusted to a temperature somewhat above the corresponding saturation temperature.

Economizers

The typical marine economizer consists of a series of horizontal tubes which recover heat from the combustion gases leaving the generating tubes. The heat recovered is used to heat the feedwater prior to admission to the steam drum. Most economizers use some form of extended surface

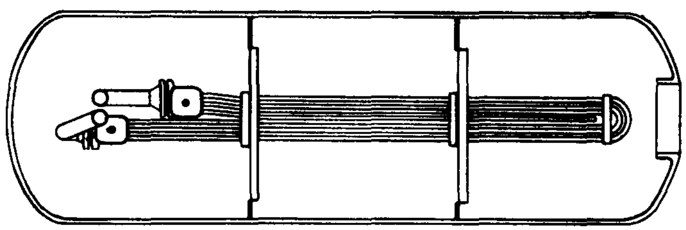

Figure 5-37. Internal (coil) desuperheater.
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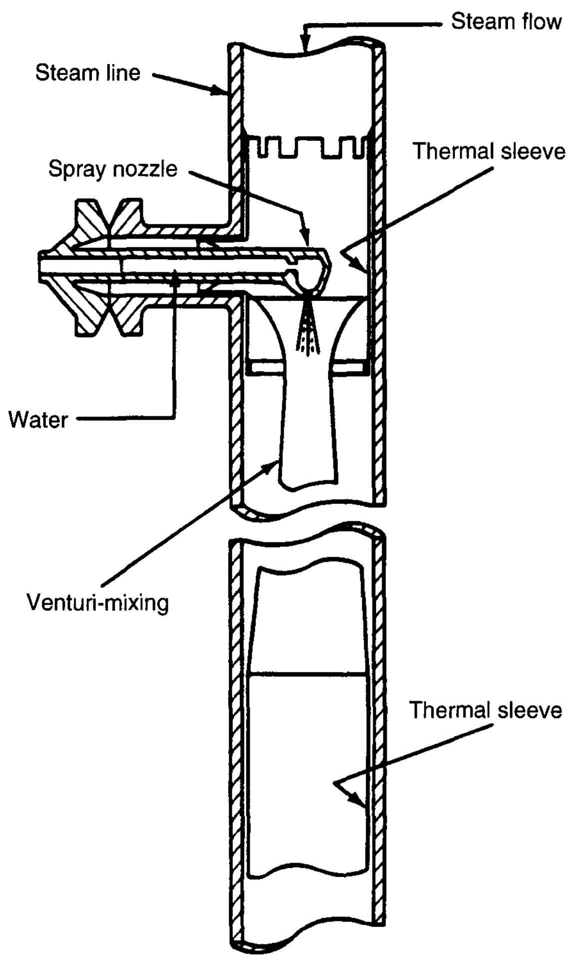

Figure 5-38. External (spray) desuperheater.
Courtesy Babcock & Wilcox.

due to the relatively low heat transfer coefficient on the gas side as compared to that on the waterside. Increasing the outside heating surface increases the heat transferred per linear foot of economizer surface. Usually, this extended surface consists of either steel fins welded to the outside of the tubes or cast-iron gill sections installed over the tubes. Figure 5-39 shows the extended surface of economizer tubes.

It is important to keep the economizer surface clean to maintain its thermal performance, to avoid corrosion, and to avoid the possibility of a stack fire. Proper use and maintenance ofthe sootblowers is the most important step that can be taken to ensure adequate cleanliness.

Economizers are used as the final recovery heat exchanger on boilers in plants without high-pressure feedwater heaters. If high-pressure feedwater heaters are installed, the higher feedwater temperature would result in an unacceptably high stack temperature and poor boiler efficiency. Boilers of plants with high-pressure feedwater heaters are typically fitted with an air heater as the final recovery heat exchanger.

Air Heaters

The preheating of the combustion air prior to mixing with the fuel at the burner improves the combustion process. Preheated combustion air provides the ability to operate with lower levels of excess air, thus improving boiler efficiency. It is not uncommon to be able to maintain complete combustion with excess air 5 to 10 percent lower if the air is preheated to 250° to 350°F.

As stated above, steam power plants without high-pressure feedwater heaters are usually fitted with an economizer. It is common in these plants also to find a steam air heater fitted. In most steam air heater installations, low-pressure bleed steam or steam from the auxiliary exhaust system is used. As the steam inside the tube condenses, heat is transferred to the air passing over the outside ofthe tube. Because the condensing steam

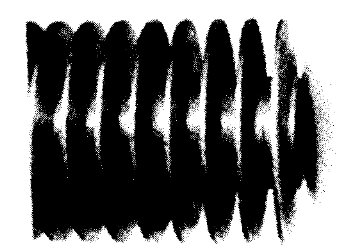

Figure 5-39. Extended economizer surface.
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film coefficient is much higher than the air film coefficient, extended heat transfer surface can be used to advantage. The finned tubes can be closely spaced since both fluids are clean, and fouling is not a significant problem.

An alternative to using an economizer as the final heat recovery device in a boiler is to use an air heater that uses the stack gases to preheat the combustion air. On plants without high-pressure feedwater heaters, the economizer is generally a better choice because water-gas heat transfer is more effective than the gas-gas heat transfer ofthe air heater. However, on plants with high-pressure feedwater heaters and therefore high feed temperatures, an economizr installation would result in poor boiler efficiency due to the resulting high stack temperatures. On these plants, either a tubular or rotary regenerative air heater is usually fitted. The stack temperature is limited by the dew point of the sulfuric acid in the combustion gases. Too Iowa temperature will result in condensation of the sulfuric acid, causing corrosion of the air heater. A tubular air heater is basically a shell-and-tube heat exchanger with the stack gases leaving the boiler on one side and the combustion air from the forced draft fan on the other. Most tubular air heaters are arranged in counterflow configuration. With an inlet air temperature of lOO°F and a stack temperature of 300°F, metal temperatures as low as 200°F will result. Sulfuric acid corrosion is thus a concern, especially with the lower stack temperatures at low boiler loads. Most air heaters are fitted with dampers which permit bypassing the combustion air at low loads.

Tubular air heaters are available in either horizontal or vertical configurations as shown in figure 5-40. In a horizontal air heater, the combustion air flows inside the tubes and the gases flow on the outside. In a vertical air heater, the stack gases flow inside the tubes and the combustion air is on the outside. Vertical air heaters are less prone to fouling initially, but sootblowers are not very effective due to the tube arrangement; manual cleaning by brushes is necessary to avoid eventual tube plugging.

The tubes in a horizontal air heater are easier to clean by sootblowers and are less susceptible to choking. However, once fouled, horizontal air heaters are more prone to stack fires which can result in serious damage to the heater and the uptake. Bypassing at low loads is important to avoid rapid fouling. If the heater is already fouled, the higher metal temperatures while bypassed increase the risk of a stack fire.

A rotary regenerative air heater as shown in figure 5-41 consists of a slowly rotating drum containing a series of corrugated steel plates, treated or coated to reduce corrosion. The rotor is enclosed in a gastight housing with connection for the gas and air inlets and outlets. Mechanical seals are fitted to minimize the carryover of gas or air between the ducts. Stack gases pass across one section of the rotor, heating the rotor metal. Combustion air from the forced draft fan passes across another section and is heated as the metal warmed by the stack gas is rotated into the combustion air flow.

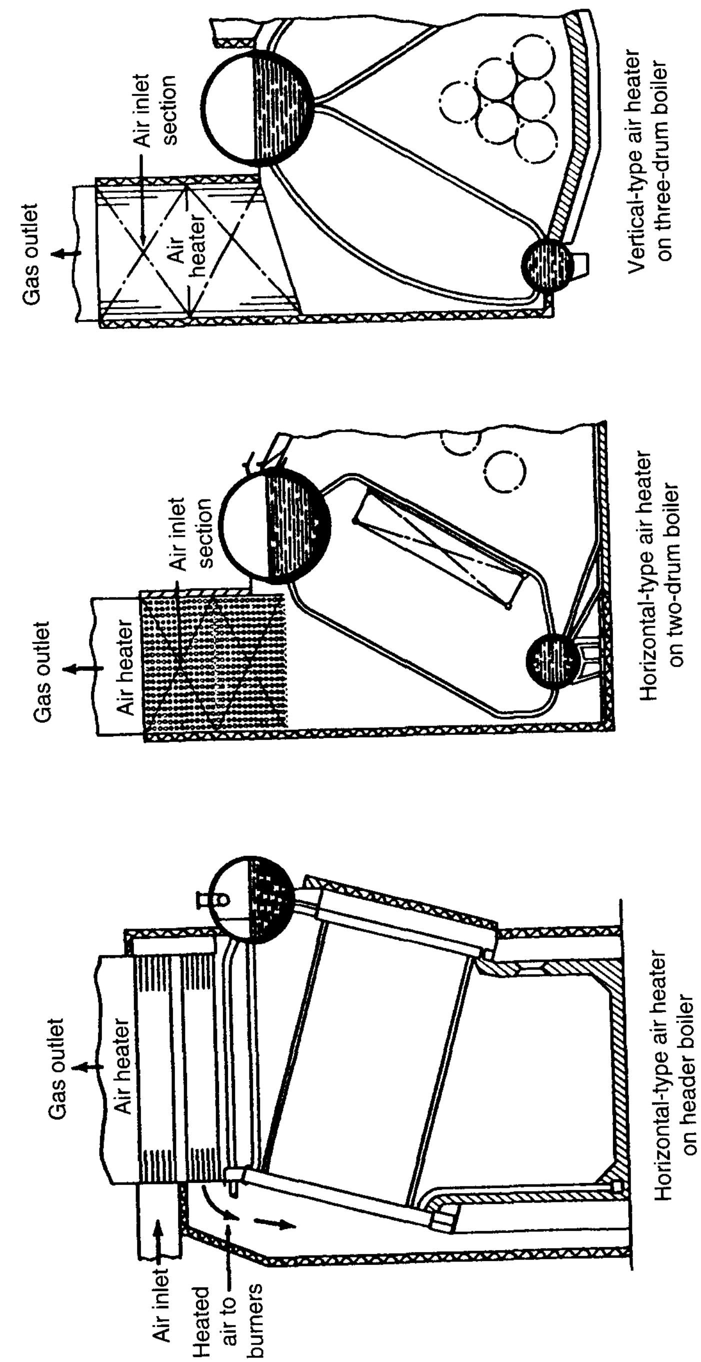

Figure 5-40. Tubular air heaters. Courtesy Babcock & Wilcox.

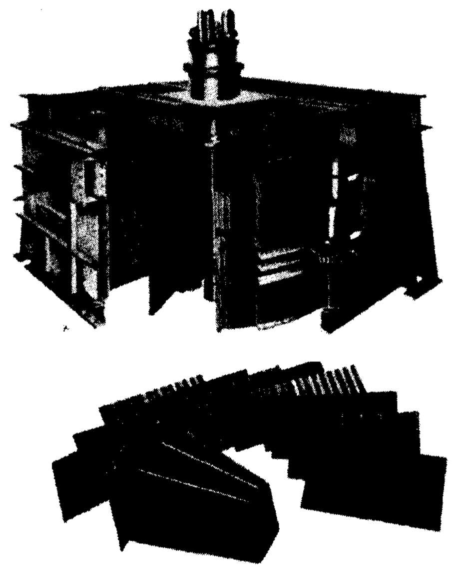

Figure 5-41. Rotary regenerative air heater.
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As with tubular air heaters, dampers are fitted to permit bypassing the rotary regenerative heater at low boiler loads. Because of the alternate heating and cooling of the rotor, average metal temperatures in a rotary regenerative air heater are higher than for a tubular unit. This permits operating with lower stack temperatures and results in higher boiler efficiency.

Sootblowers are fitted on both the air and gas sides ofthe rotary regenerative air heater. The gas-side sootblower is used at sea for normal cleaning of the heat exchange surfaces. The air-side sootblower is used only in port. The removed soot is blown into the boiler furnace, minimizing any releases in violation of harbor regulations.

Wear in the heater air seals can result in excessive carryover of combustion air into the stack gas flow and stack gas into the air flow. Normal leakage is 10 to 15 percent and should not exceed 18 percent. Leakage can be checked by measuring the oxygen percentage with a gas analyzer in the stack gas flow at the inlet and the outlet of the heater. With the two readings, the leakage percentage can be calculated using the following equation:

$$\text{Leakage \%} = \left(\frac{O_2\,\%\,\text{Out} - O_2\,\%\,\text{In}}{21 - O_2\,\%\,\text{Out}}\right) \times 100$$

If the leakage is high, the heater seals must be adjusted or replaced.

Boiler Mountings

Boiler mountings include various valves, fittings, and accessories mounted on the boiler to permit safe and efficient operation. Included are such devices as safety valves, sootblowers, gauge glasses, remote drum level indicators, steam stop valves, feedwater stop and check valves, blowdown valves, and vent valves.

SAFETY VALVES

Safety valves are installed on the boiler to protect it from overpressure. On a propulsion boiler, three valves are typically fitted-two on the steam drum and one on the superheater. The superheater safety valve must be set to lift either before, or simultaneously with, the drum valves to ensure a flow of steam through the superheater to protect it from overheating. Safety valves differ from relief valves in that they are designed to open completely (pop) when the set pressure is reached, and to stay open until the boiler pressure is reduced below the set pressure. The difference between the pressure at which the valve opens and the pressure at which it closes is called blowdown. Blowdown for a propulsion boiler safety valve is typically 2 to 4 percent ofthe set pressure. A safety valve includes adjustments for both the set or popping pressure and the blowdown. The two most common types of safety valves used are the huddling chamber type shown in figure 5-42 and the nozzle reaction type shown in figure 5-43. With either type of valve, the set pressure is adjusted by changing the compression in the upper spring. Increasing compression raises the set pressure.

The blowdown in the huddling chamber safety valve is adjusted by raising or lowering the adjusting ring. As the set pressure is reached and the valve begins to open, greater disk area is exposed, causing the valve to pop. With the valve wide open, the pressure acting on the valve disk is somewhere between the boiler pressure and atmospheric pressure. Raising the adjusting ring closes offthe exit area and increases the pressure acting on the bottom ofthe disk. The boiler pressure must decrease further before reseating, thus increasing the blowdown.

A nozzle reaction safety valve has two blowdown adjustments-an adjusting ring and a nozzle ring. During initial lift of the valve, increased area is exposed and the valve operates like a huddling chamber valve. As the stroke increases, however, the steam exiting the valve is turned through an angle, adding a "reaction" force to the pressure times area force. Lowering the adjusting ring increases the angle through which the steam is turned, increasing the reaction force, and thus increasing the blowdown. The nozzle ring controls the initial lift of the valve and should be adjusted only when the valve simmers or doesn't pop smartly.

Another type of safety valve sometimes found on a propulsion boiler is the pilot valve type shown in figure 5-44. This type of valve solves two problems: It ensures that the drum valves do not open before the superheater

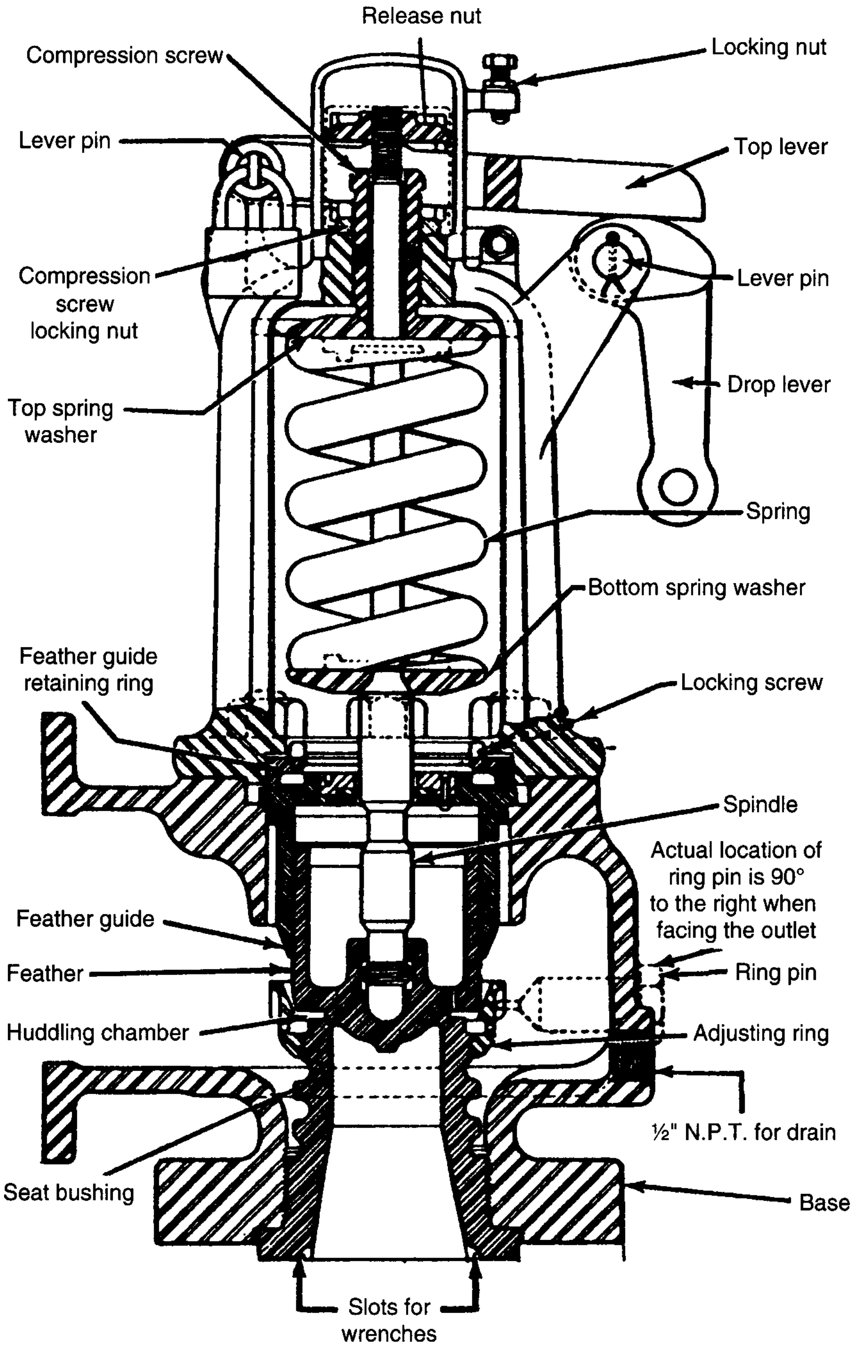


Figure 5-42. Huddling chamber safety valve

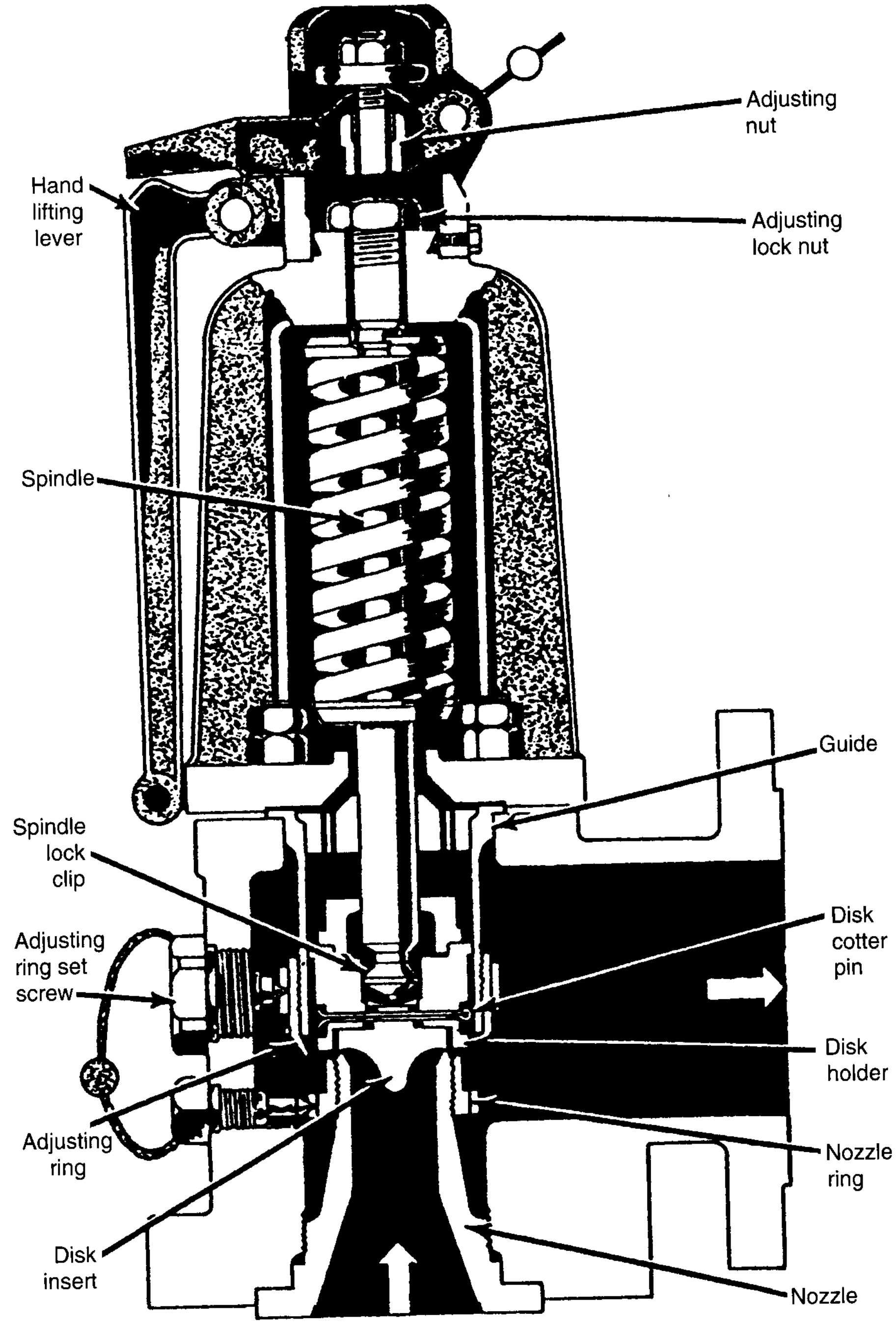


Figure 5-43. Nozzle reaction safety valve

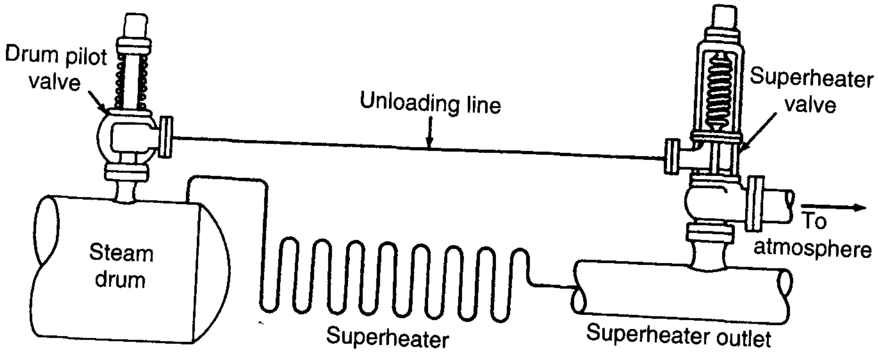


Figure 5-44. Pilot safety valve

valve, and it avoids problems such as distortion caused by the high-temperature superheated steam. The pilot is mounted on the drum and is actuated by the lower temperature saturated steam. The drum and the superheater valves lift together.

Safety valves must be tested periodically under the supervision of a flag authority and/or classification society surveyor. Both the popping pressure and the blowdown are tested. First, the seals are cut and the cap and easing gear removed from all the safety valves. Next, all the safety valves on the boiler are gagged except the one to be tested (see fig. 5-45). Then one burner is lit off and the boiler pressure is slowly increased. When the valve pops, the pressure is noted and the burner is secured. The pressure at

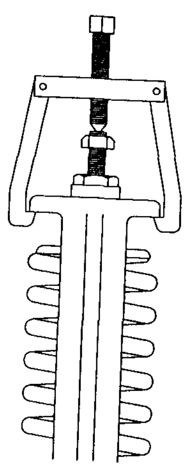


Figure 5-45. Safety valve gag

which the valve reseats is noted. If the set pressure or blowdown is unsatisfactory, the adjustments are changed and the valve is retested. This procedure is repeated for each valve. After testing is completed, the valves are reassembled and resealed by the surveyor.

SOOTBLOWERS

Sootblowers are used to keep the heat transfer surfaces in the boiler clean during normal boiler operation. Most sootblowers use jets of auxiliary steam to blow soot and other deposits off the outside of the generating tubes, superheater tubes, and economizer and air heater surfaces. Sootblowers are of two basic types-rotary and retractable.

Figure 5-46 shows a typical rotary sootblower. The blower element consists of a section of pipe with a series of exit nozzles along its length. The element is rotated either manually by chain or crank, or remotely by an air

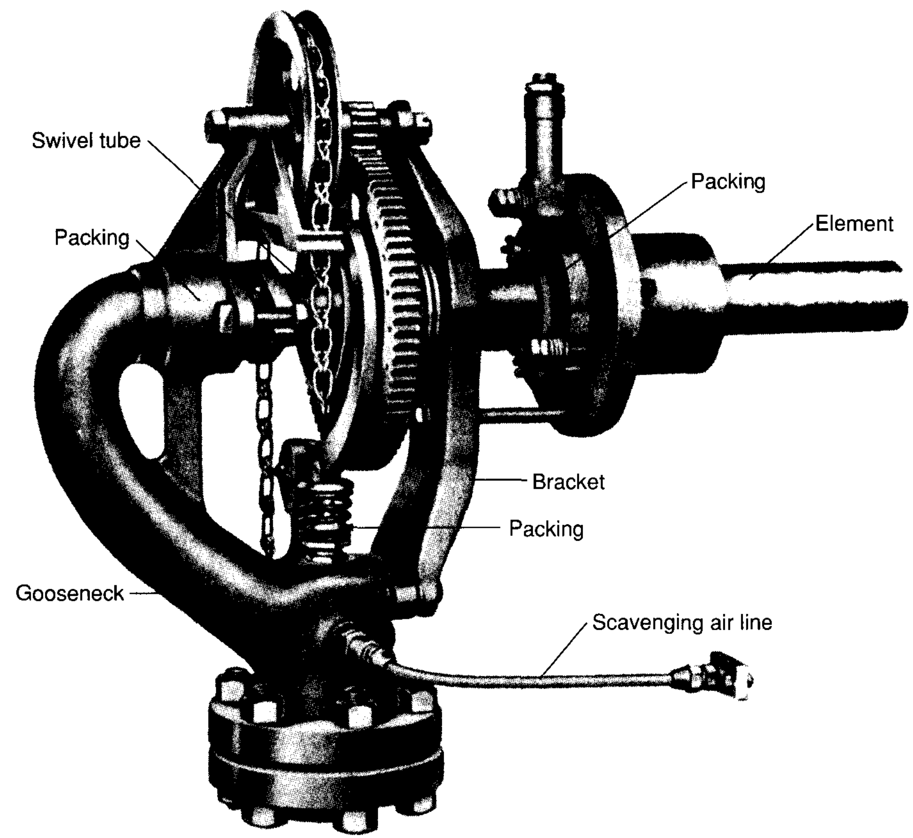


Figure 5-46. Rotary sootblower

motor. As the element is rotated, a cam opens and closes a valve admitting steam to the element. Most rotary sootblowers are full admission, meaning that the steam blows for almost the complete 360 degrees of rotation. In some installations, the blower is set up for partial admission, meaning it blows only for a portion ofthe full rotation. This is accomplished by changing the cam that actuates the steam admission valve. Some rotary sootblowers are fitted with scavenging air connections. Low-pressure air is admitted into the blower head to flush the unit of corrosive stack gases. A check valve prevents steam from backing up into the air line during blower operation.

Retractable sootblowers were developed to overcome the short life of even alloy rotary eleroents in high-temperature areas of the boiler like the superheater. The element is inserted into the boiler during operation and withdrawn when the blowing is complete. (Fig. 5-47 shows a typical retractable sootblower.) The element has only two steam exit nozzles located in the tip. Because of the high-volume and high-velocity steam jets from this type of element, it is referred to as a "mass action" blower. An air motor is used to extend and retract the element and to rotate the element. The steam admission valve is opened mechanically as the element is inserted into the boiler and closed as it is removed. The two steam jets trace a spiral path through the tubes as the element is inserted or removed and rotated at the same time.

GAUGE GLASSES

All boilers require at least two independent means of indicating the water level in the steam drum. On propulsion boilers, double-sided, plate glass-

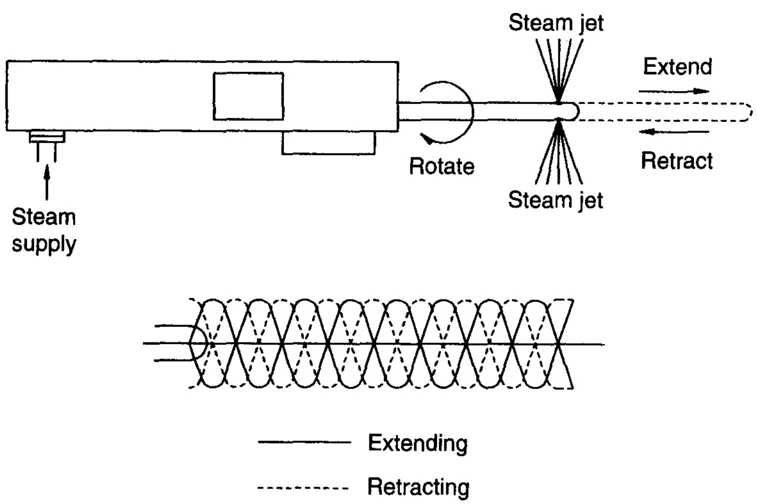


Figure 5-47. Retractable sootblower

type gauge glasses as shown in figure 5-48 will typically be fitted. The two flat glasses are clamped between the centerplate and the front and back covers. A sheet of mica is placed between the glass and the steam and water to prevent glass etching. At higher pressure, the glass would burst within hours if the mica were not installed. When a gauge glass is rebuilt, care must be taken to avoid undue stresses in the glass plates. Mter disassembly of the unit, the steel plate joint faces must be carefully cleaned. The gauge glass must be built up using new glasses, mica, and gaskets in the correct sequence. The nuts should be tightened gradually, working from the center toward the ends in an alternating pattern. When mounted back on the boiler, the gauge glass should be warmed as gradually as possible. The steam valve and water valve shutoff chains must be reconnected so that a single pull will close both valves.

DRUM LEVEL INDICATORS

Because it is difficult to observe the boiler gauge glasses in many installations, some type of remote drum level indicator is usually fitted (see fig. 5-49). The readout unit consists of a manometer or differential pressure gauge. If a manometer is fitted, a special gauge-indicating fluid is used, that is insoluble in water and has a density greater than that of water.

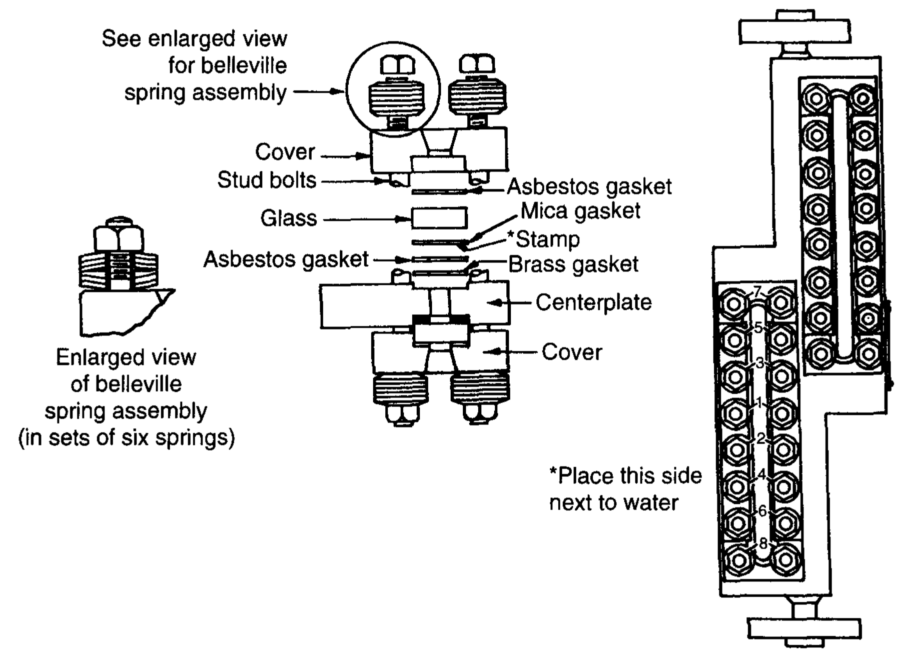


Figure 5-48. Flat plate gauge glass

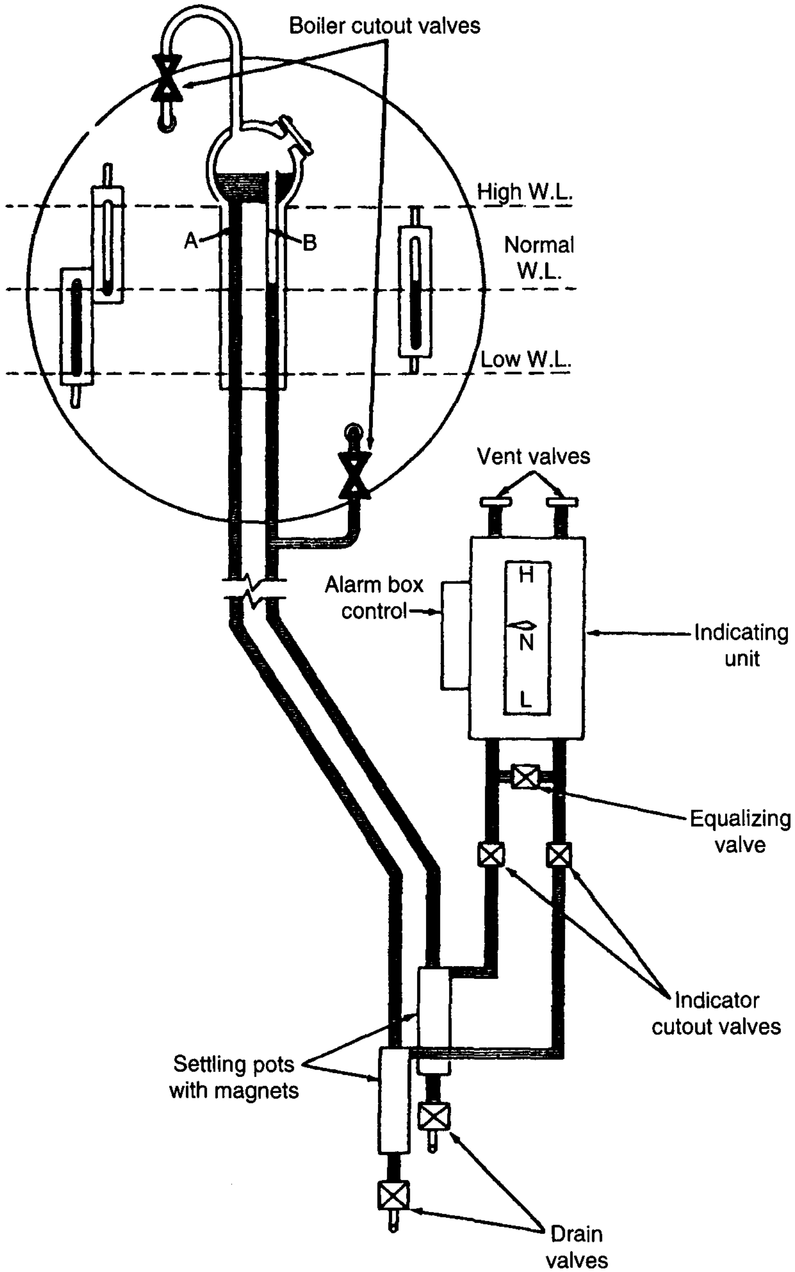


Figure 5-49. Remote drum level indicator

Above the readout are two legs filled with water. The leg connected to the upper space of the steam drum (A) is filled to the top. This leg will be maintained full as steam condenses. The level in the other leg (B) corresponds to the level in the steam drum. As the drum level changes, the differential pressure between the two legs changes. The manometer or gauge displays this differential pressure which is directly related to the drum level. Although subjected to full boiler pressure, the water in the unit is stagnant and cools to approximately engine room temperature. The devices are relatively trouble free, requiring only occasional cleaning.

Drum Internals

The steam drum of a marine boiler contains a number of components necessary for proper operation. Since these components are internal to the boiler drum, they are not classified as boiler mountings. Among the items included as drum internals are the dry pipe or dry box, baffles, desuperheater, surface blowdown, centrifugal steam separators, the chemical feed pipe, and the internal feed pipe (see fig. 5-50).

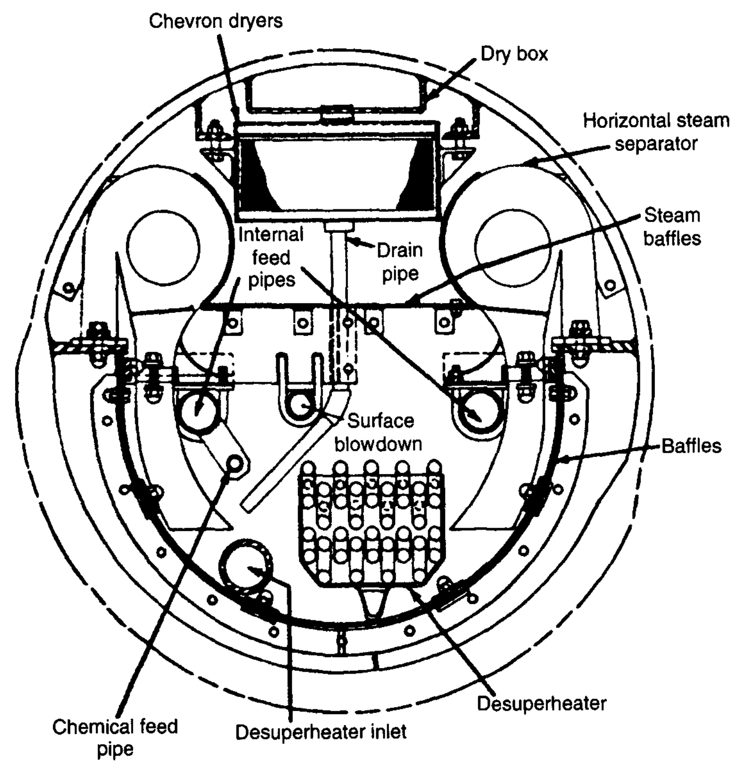


Figure 5-50. Steam drum internals

The dry pipe, dry box, and centrifugal steam separators have the same task-removing drops of liquid water from the steam being removed from the drum. A dry pipe is simply a pipe closed on the ends with holes or slots cut into the top. The steam being removed must make a 180° turn. At high velocity, the denser liquid is moved by centrifugal force to the outside ofthe turn, causing it to separate from the steam. Drain holes are located at each end to permit any collected liquid to return to the drum. A dry box works in the same fashion but has a rectangular configuration. In high-capacity boilers, centrifugal separators are commonly fitted to improve the effectiveness of separation. 'rhe steam-water mixture enters the barrel of the separator and is caused to rotate by baffles. The more dense liquid is thrown to the outside and the less dense vapor collects in the middle. The

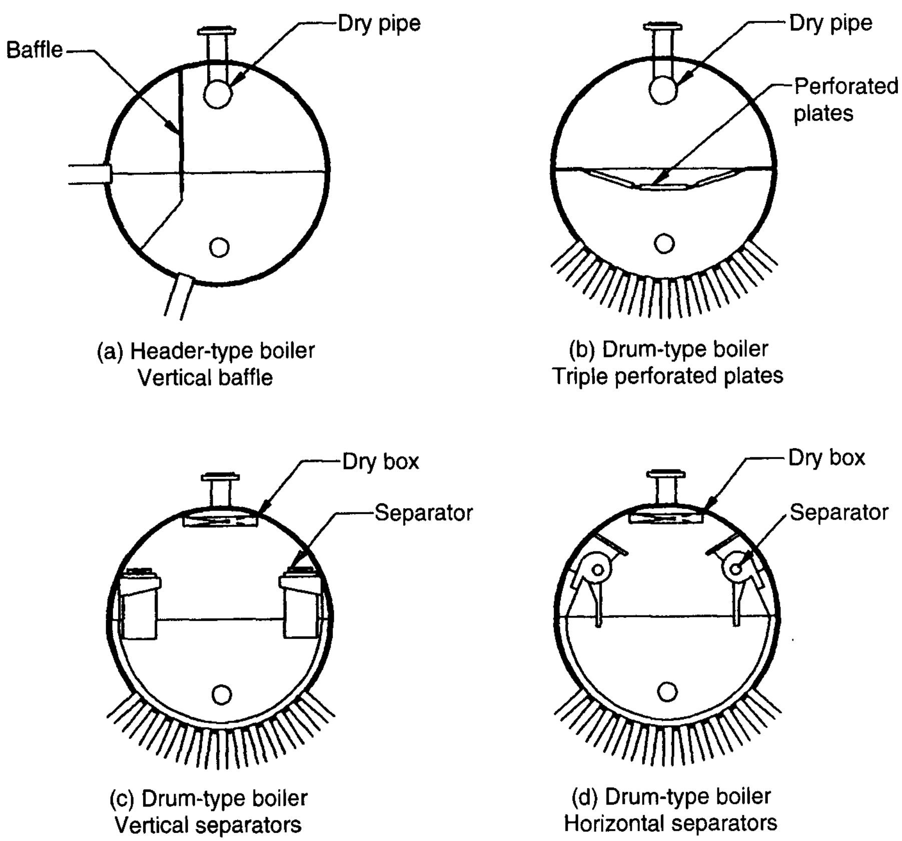


Figure 5-51. Steam drum baffles

liquid is returned to the lower portion of the steam drum and the vapor exits into the upper portion.

Baffles are installed to stabilize the steam-liquid interface in the steam drum. (Fig. 5-51 shows several typical steam drum baffle configurations.) Longitudinal baffles, sometimes called swash plates, are installed to reduce the movement ofwater back and forth in the drum due to pitching and rolling. In a boiler with centrifugal steam separators, baffles are installed to direct the steam-water mixture exiting the risers into the inlet of the separators.

The internal feed pipe located in the lower half of the steam drum serves to distribute the incoming feedwater along the length of the drum. This avoids localized temperature gradients and associated thermal stresses. The chemical feed pipe performs a similar function for the chemicals being added for treatment of the boiler water.

BOILER AUTOMATION SYSTEMS

Automatic control systems are an essential part of a modern marine power plant. They serve to assure safe, efficient, and reliable operation of the various systems and equipment by smaller crews. A variety of control systems are commonly installed on boilers to perform such functions as controlling the boiler pressure (combustion control system), controlling the steam drum level (feedwater regulating system), starting and securing burners (burner management system), and controlling the superheater outlet temperature. Control systems may be pneumatic, analog electronic, or digital electronic. While pneumatic control systems were common for many years, the trend today is to use digital electronic control systems based on microprocessors.

Control systems can be classified as open-loop (feedforward) or closed-loop (feedback). In closed-loop control, the actual output of the system is measured and compared with the desired value. If the difference between the actual and desired values (the error) is not zero, the system adjusts the process until it is. Because of its ability to adjust to variations in the process, closed-loop control is extensively used in combustion control and feedwater regulating systems. Open-loop control is sometimes combined with closed-loop control to improve system response.

The simplest form of closed-loop or feedback control system is on-off control. With this type of control, the input to the process being controlled is either fully on or fully off. Most home heating systems employ this type of control. When the temperature in the house drops below the desired value, the thermostat turns on the furnace. Heat is delivered and the house temperature begins to rise. When the house temperature reaches the desired value, the thermostat shuts off the furnace. The furnace will cycle on and off as necessary to maintain the temperature within a band about the desired temperature.

On-off control is obviously unsuitable for the control of the combustion process in a marine propulsion boiler. Rather than turning the burners completely on or off, it is preferable to continuously increase or decrease the firing rate as necessary to maintain the desired boiler pressure. The simplest type of such a system is proportional control. (Fig. 5-52 illustrates a simplified closed-loop proportional control system for automatically controlling the pressure of a boiler.) The input to the controller is the steam pressure. The output fr.om the controller which strokes the fuel oil valve is proportional to the deviation of the actual steam pressure from the desired value.

One characteristic of a simple proportional controller that should be understood is gain. Gain is defined as the change in controller output for a given change in controller input:

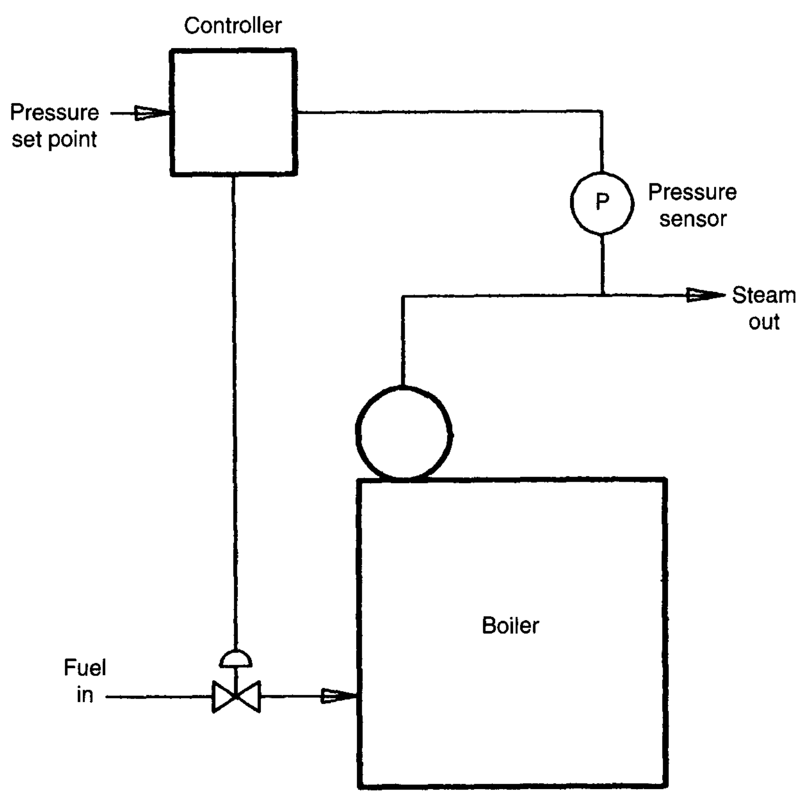


Figure 5-52. Simplified boiler pressure control system

$$\text{Gain} = \frac{\Delta\ \text{Output}}{\Delta\ \text{Input}}$$

A related term is proportional band defined as the range of the input over which the output will vary from minimum to maximum. The relationship between proportional band and gain is as follows.

$$\text{Proportional band} = \frac{100\%}{\text{Gain}}$$

Note that proportional band is essentially the inverse of gain. Doubling the gain will cut the proportional band in half.

As the controller gain is increased, a particular change in input will result in a greater change in controller output. (Fig. 5-53 illustrates the effect of different values of gain for a pneumatic proportional controller with an operating range of 3 to 15 psig.) With a gain of **1,** if the input changes 1 psig, the output will change 1 psig. If the gain is increased to 2, a 1 psig change in the input will result in a 2 psig change in the output. With a gain of 4, the output will change 4 psig for a 1 psig change in the input.

Let's follow the operation of the system in figure 5-52 through a load change. If the boiler load (steam flow) increases, the boiler pressure will drop. This change in the input to the controller will cause the output to change, opening the fuel oil valve and increasing the firing rate. Note that for the controller output to change and open the fuel valve, there must be a change in the input-the steam pressure. This change in the controlled variable as the system load changes is called "offset." Since the purpose of the system is to keep the steam pressure constant, offset is undesirable. Note that if the gain is increased, the firing rate will increase for a smaller change in the input steam pressure. This means that high gain will reduce offset. It would seem that the solution to this problem is to use very high gain. The problem with this approach is that, at some point, the high gain will cause the system to become unstable and begin to oscillate or "hunt." The hunting may begin at a value of gain that results in still unacceptably high offset.

The solution to this problem is to add integral or reset action to the controller. This feature will reduce the controller gain to a low value during load changes, thus avoiding hunting. During steady-state operation, high gain is restored, thus minimizing offset (see fig. 5-54). Mathematically, the controller output is based on the integration ofthe error signal over time in addition to the magnitude ofthe error, hence the term integral. This action is also referred to as reset since the band of proportional action is shifted or reset so that the controlled variable operates about a new base point.

One disadvantage of integral or reset in a controller is that the controller will respond more slowly to rapid load changes. The solution is to add

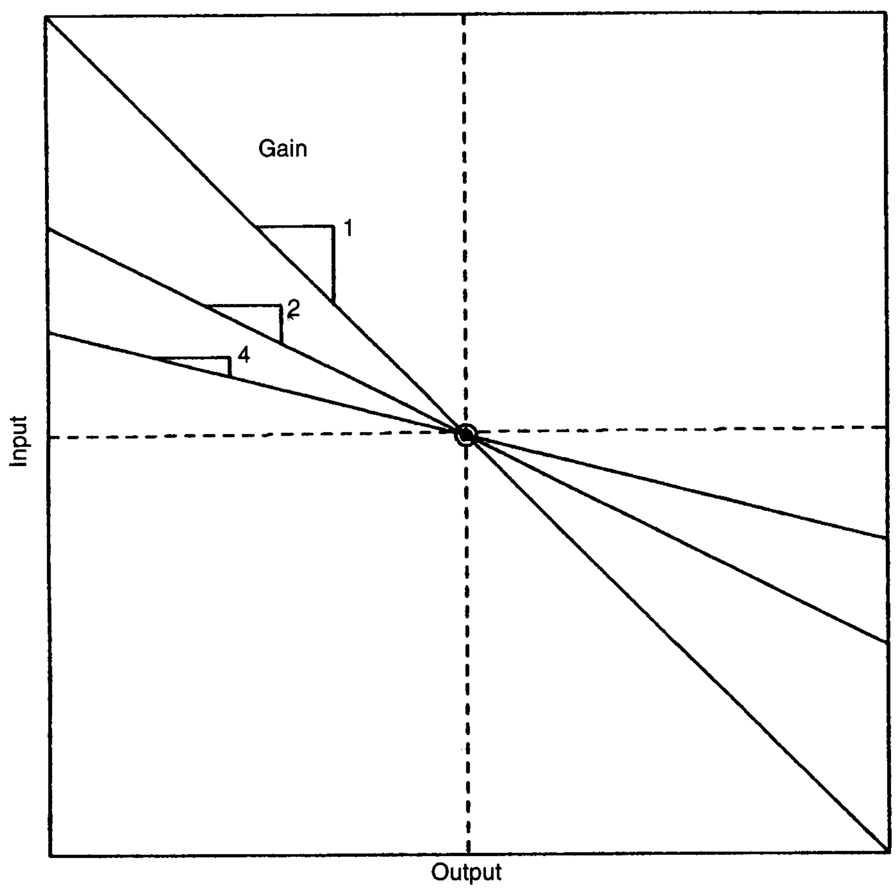


Figure 5-53. Proportional controller output versus gain

derivative or rate action to the controller. During the initial stages of a load change, the gain is retained at a high value to provide rapid adjustment. Before hunting can occur, the gain is reduced by the integral controller. Finally, the gain is restored to a high value, to reduce offset. Such a controller with all three control modes is referred to as a PID (proportional plus integral plus derivative), or three-mode controller. In most combustion control systems, derivative action is not required and proportional or proportional plus integral controllers are used.

In the following sections, some of the automatic control systems commonly found on marine boiler installations will be discussed in more detail.

Combustion Control Systems

The function of the boiler combustion control system is twofold: (1) to keep the boiler pressure constant by varying the fuel firing rate and (2) to con-

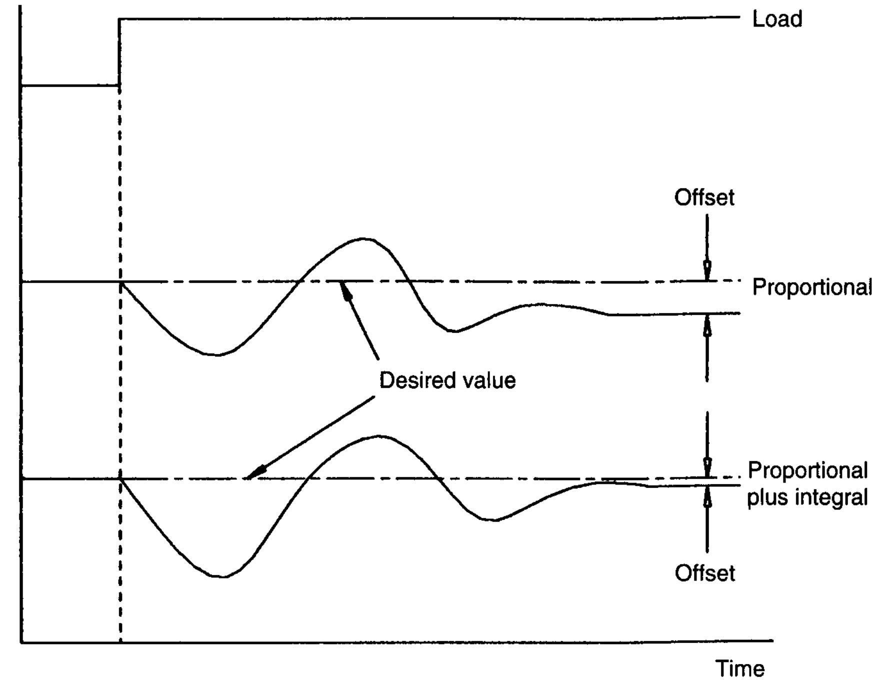


Figure 5-54. Proportional + integral controller response

trol the air/fuel ratio to ensure complete combustion with minimum excess air. It must accomplish these two somewhat unrelated tasks over a wide range of loads and during both steady-state operation and load changes. Many systems have been developed over the years to satisfy the needs of particular applications.

Combustion control systems can be classified in a number of ways. First, the system can be classified as series or parallel. (Fig. 5-55 shows block diagrams of series and parallel systems.) A series system can be classified as either fuel-follows-air or air-follows-fuel. In an air-follows-fuel series system, the boiler pressure is used to control the fuel, and the change in fuel flow is used to trigger the change in air flow. With a fuel-follows-air system, the boiler pressure will control the air, and the change in air flow will change the fuel flow. Series systems have problems during certain load changes. With a fuel-follows-air system, during a load increase, the fuel will increase before the air does, resulting in smoking. A similar thing will happen with an air-follows-fuel system during a load reduction as the air decreases before the fuel. For this reason, series systems are no longer used on propulsion boiler applications. Because of the lower operating pressures and relatively steady operation, series systems are still common on low-pressure auxiliary boilers.

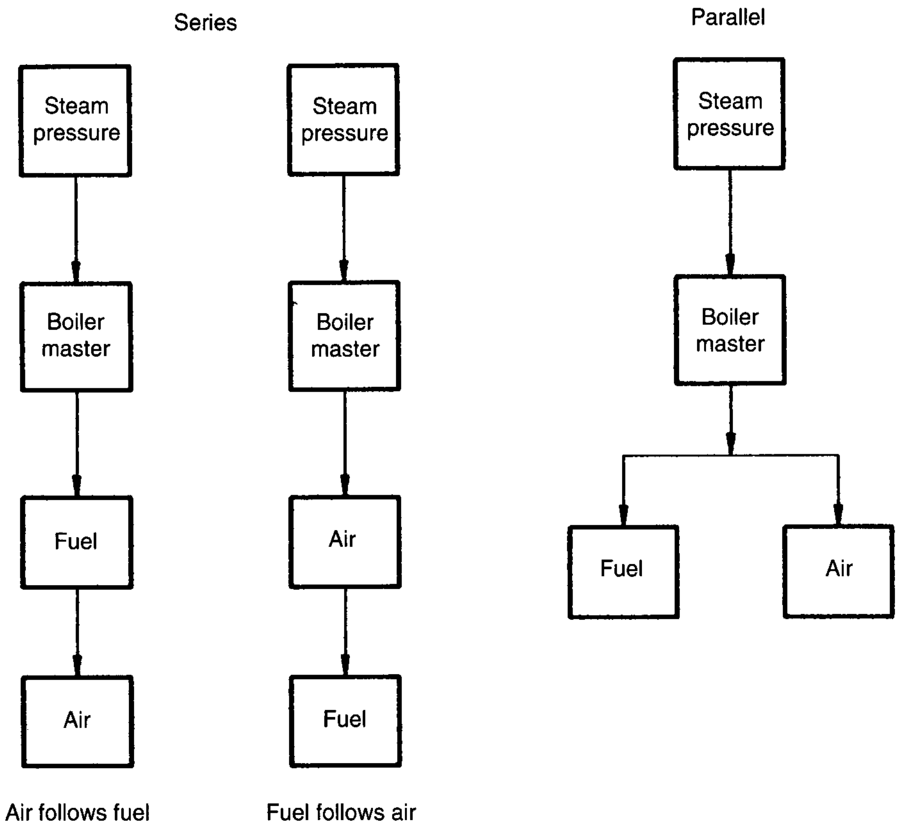


Figure 5-55. Series and parallel combustion control systems

By sending the load demand simultaneously to the air and fuel controller-a parallel system-the problem of smoking during load changes with series systems is solved. What is lost is the linkage between fuel and air and therefore the ability to easily control the air/fuel ratio over a wide load range. To improve the performance of the parallel system, it will be necessary to add some additional features to it, increasing its complexity.

The first step is to convert the system from a positioning system to a metering system. In a positioning system, the air and fuel controllers respond to the load demand from the boiler master controller by changing the *position* ofthe fuel oil valve and forced draft damper. This change in position of the two actuators is designed to cause the required changes in the air and fuel. However, the air and fuel flows will not increase or decrease by the same amount during a load change due to difficulties in accurately predicting the characteristics of the fuel and air systems, changes in operating conditions, or changes in the calibration ofthe components. (Fig. 5-56 is a

logic diagram for a parallel positioning combustion control system. The standard symbols used in the diagram are summarized in fig. 5-57.)

If we add sensors for measuring the air flow and the fuelflow to the parallel positioning system and feed these signals into their respective controllers, we convert the system to a metering system. (Fig. 5-58 is a logic diagram for a parallel metering combustion control system.) An important change is that the boiler master load demand is now simultaneously controlling changes in the air and fuel flow, not changes in the position of the fuel oil valve or forced draft damper. A couple of additional features can be added to the system to further improve its performance. The use of metering has improved the control of air and fuel flow, but it is still possible for either the air or the fuel to change faster than the other, causing smoking. To

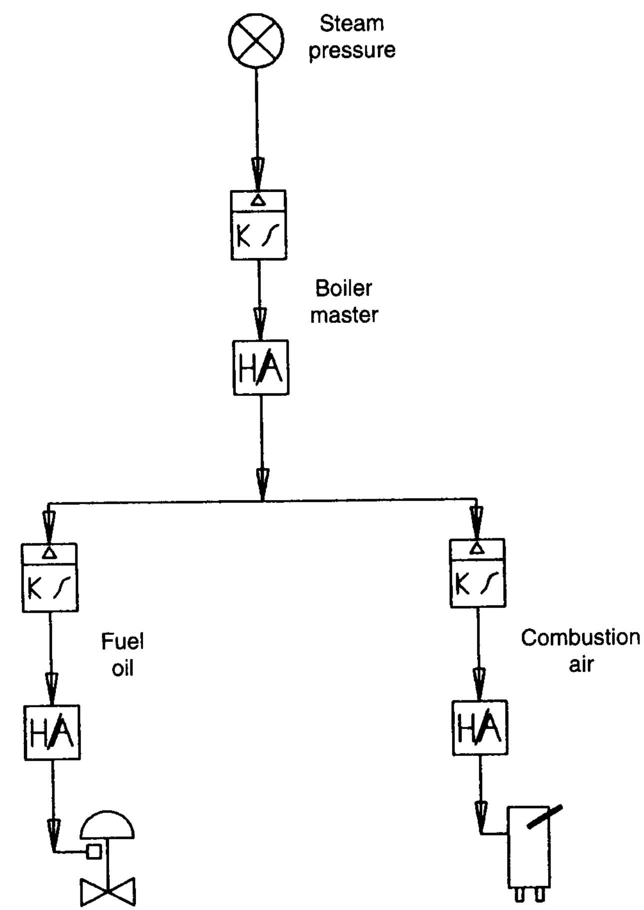


Figure 5-56. Parallel positioning system

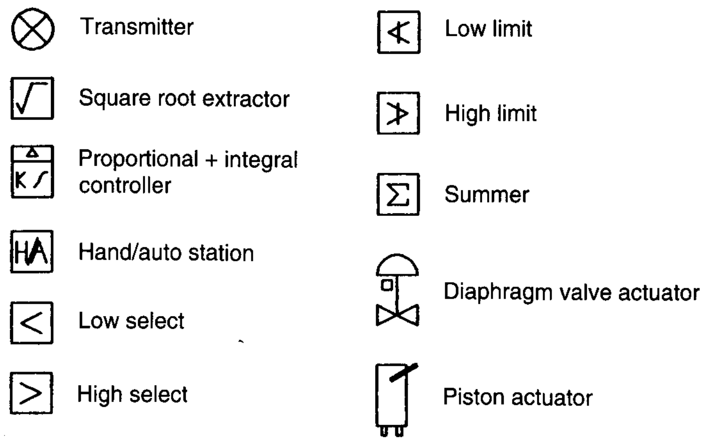


Figure 5-57. Control logic symbols

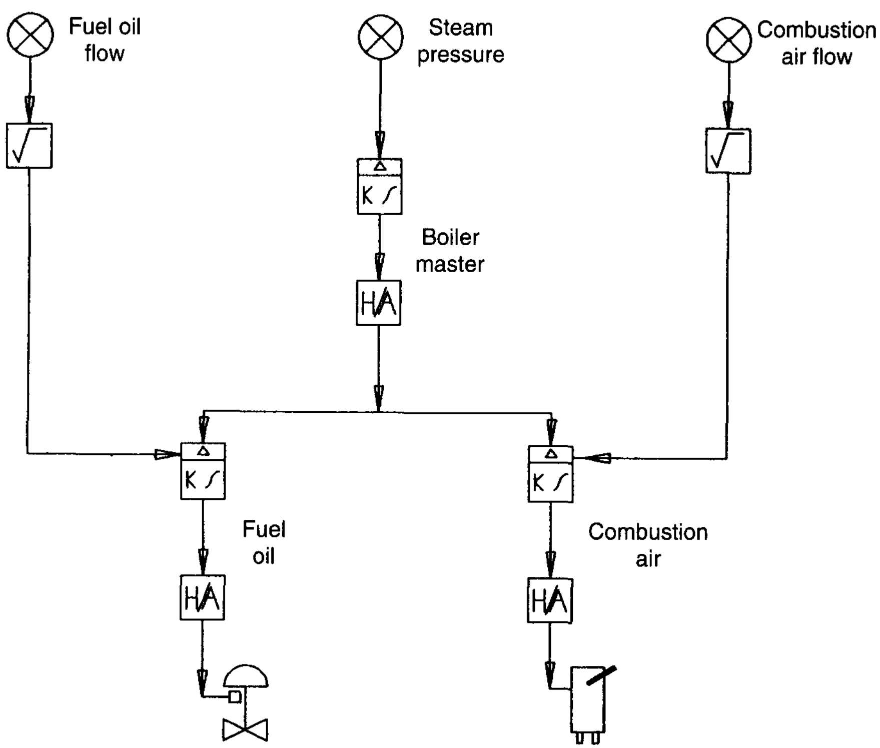


Figure 5-58. Parallel metering system

ensure that the fuel and air increase and decrease together, "cross limiting" can be used to tie the fuel and air together. This is done by providing information about the fuel flow to the air controller (using a high select) and information about the air flow to the fuel controller (using a low select). A select has two inputs and one output. The output from a high select is the higher of the two inputs. The output from a low select is the lower of the two. Referring to figure 5-59, note that the low select will send on the lower of the boiler master load demand and the air flow to the fuel controller, thus preventing the fuel flow from increasing ahead ofthe air flow. Similarly, the high select prevents the air flow from decreasing until the fuel flow has decreased.

Another feature included in the system in figure 5-59 is "steam flow anticipation." Steam flow is used as an input to the boiler master controller along with the steam pressure. The steam flow is a feedforward signal, used to sense load changes before the boiler pressure can change in response to the change in steam flow. In essence, the change in steam flow is used to anticipate a change in steam pressure, improving the ability of the system to maintain constant steam pressure during rapid load changes.

A combustion control system can be enhanced further with the addition of an oxygen trim system (see fig. 5-60). A zirconium oxide sensor (see fig. 5-18) is used to measure the oxygen content continuously in the stack gas.

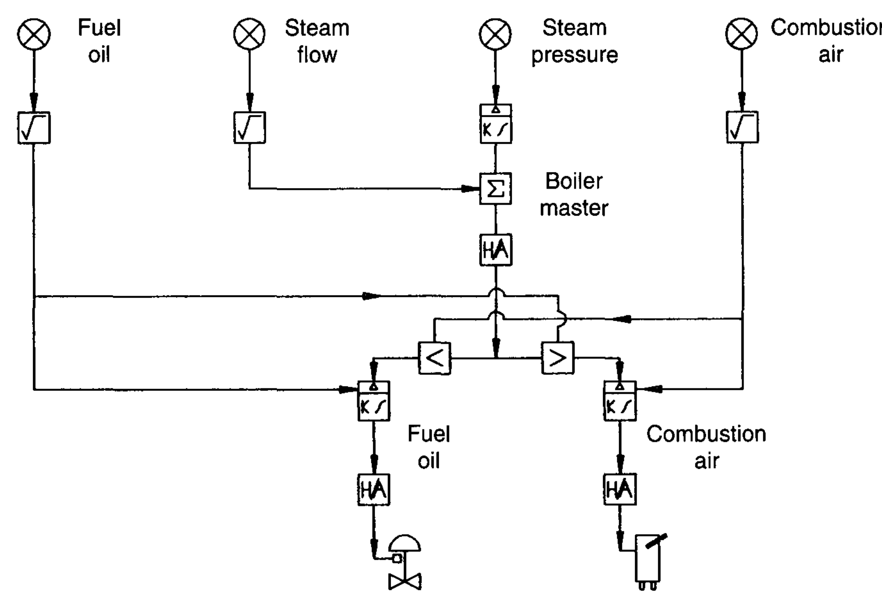


Figure 5-59. Parallel metering system
with steam flow anticipation and cross limiting

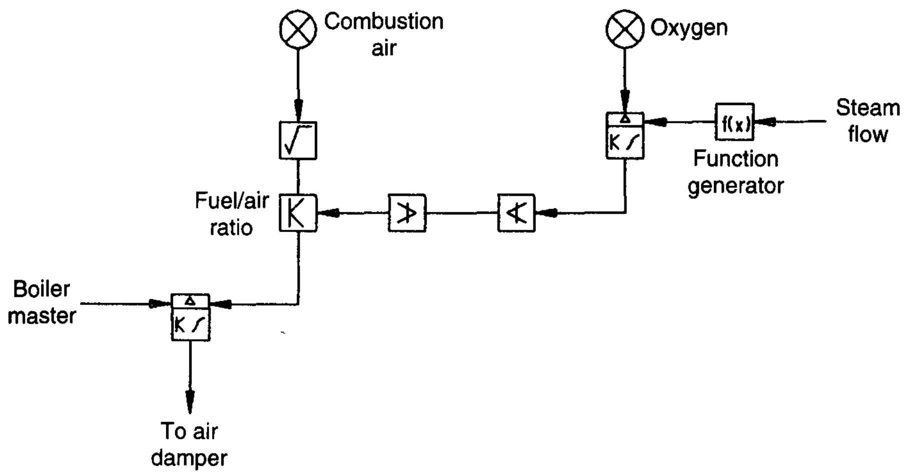


Figure 5-60. Oxygen trim system

The sensor output is fed into the combustion air control loop and trims the air automatically to maintain the desired oxygen (excess air) level. However, since more excess air is needed at low loads than at high loads, it is necessary to adjust the oxygen set point. This is accomplished by the steam flow input into the oxygen controller.

The combustion control system for a boiler using coal as fuel requires some changes from the systems described above. An important difference is that the firing rate of a boiler burning coal with a spreader stoker is varied by changing the undergrate air flow. Typically, the coal feed flow is controlled by the undergrate air flow, making the system series with fuel following air. Since most coal-fired boilers also can use oil as a fuel, the combustion control system can become quite complicated. Volume 2 of *Modern Marine Engineer's Manual* includes a description of a control system for a boiler that can burn coal only, oil only, or both fuels simultaneously.

PNEUMATIC SYSTEM COMPONENTS

The above discussion of combustion control systems is general and applies to pneumatic systems as well as analog electronic and digital electronic systems. Since pneumatic systems have been the most popular over the years, this section will discuss a few of the common components found in pneumatic combustion control systems.

A basic element used in many pneumatic control components is the nozzle-flapper amplifier (shown in figure 5-61). Control air at constant pressure is fed to the nozzle chamber through a fixed restriction. A nozzle opening, which is larger than the fixed orifice, allows air to escape from the nozzle chamber. As the flapper is moved to close off the exit, the pressure in

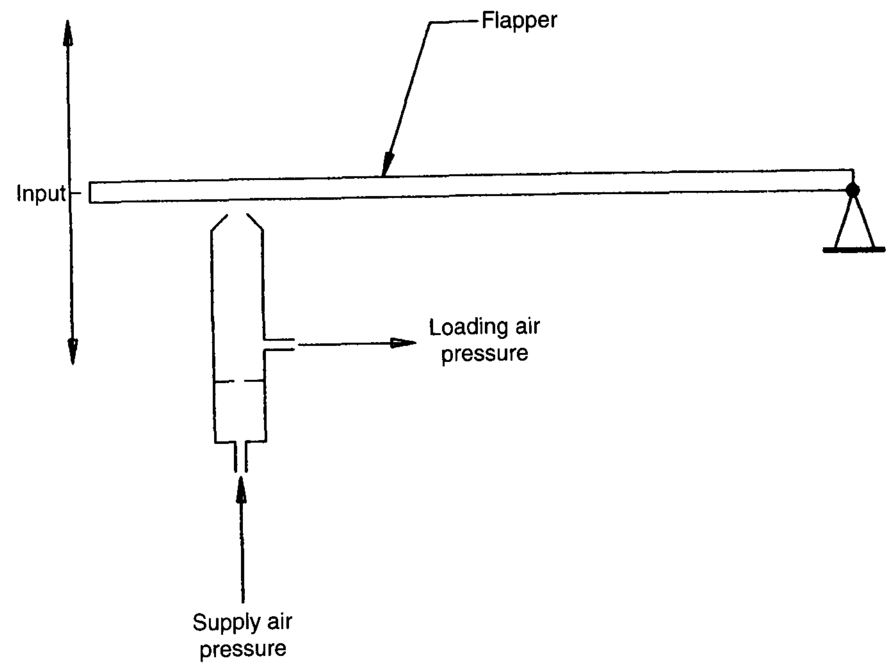


Figure 5-61. Nozzle-flapper amplifier

the nozzle chamber will rise. If the flapper moves away from the exit, the pressure in the chamber will drop.

Figure 5-62 is a simplified proportional controller based on a nozzle-flapper amplifier. An increase in the input pressure will cause the left end ofthe flapper to move up, closing off the nozzle exit and raising the output pressure. The output pressure is also fed to the restoring bellows. Note that the restoring bellows acts in opposition to the input, i.e., negative feedback. The more negative feedback, the lower the controller gain. Moving the pivot will increase or decrease the effect of the restoring bellows, thus changing the gain.

The proportional controller can be converted to a proportional plus integral (reset) controller by the addition of a third bellows and a throttling valve (fig. 5-63). Note that the reset bellows is positive feedback. During steady-state operation, the reset bellows will cancel out the effect ofthe restoring bellows, resulting in high gain. However, during a load change, the reset throttling valve will delay the effect of reset bellows, and the restoring bellows will reduce the controller gain. The gain will gradually be returned to the high value as the pressure equalizes in the reset bellows. Opening the valve will reduce the delay in restoring the high gain. Closing the valve will increase the delay.

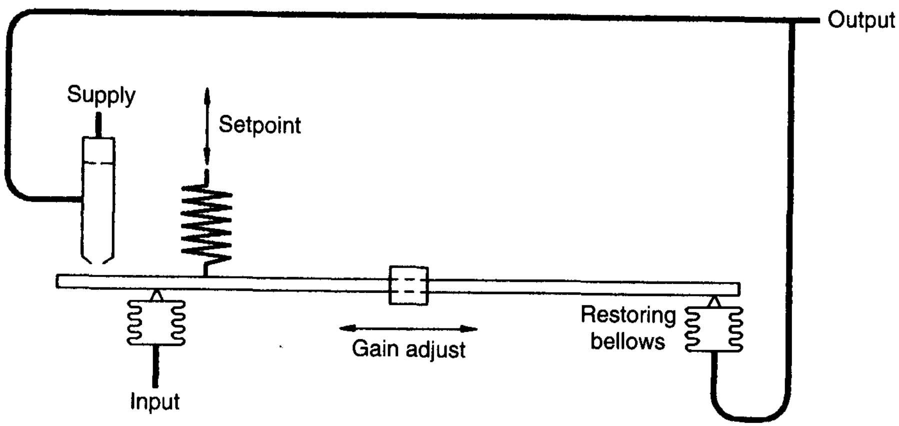


Figure 5-62. Proportional controller

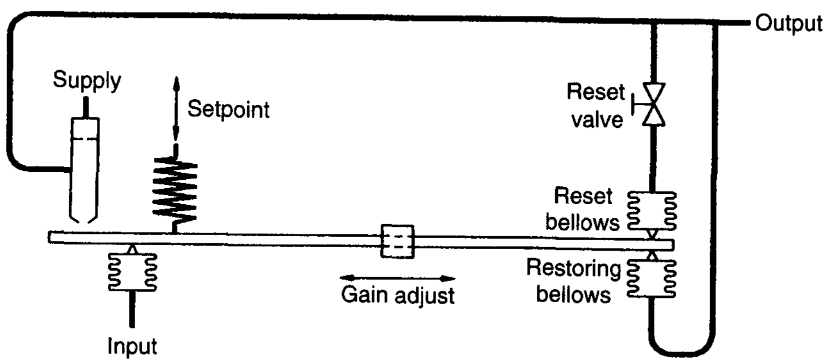


Figure 5-63. Proportional + integral controller

Transmitters deliver the control inputs to the system. They have the task of sensing a temperature, pressure, level, or other parameter in the plant and converting it to a varying pneumatic pressure. A typical pneumatic transmitter shares many design features with the proportional controller described earlier. The input sensing device moves a link, causing a corresponding movement of the vane (flapper). The vane movement changes the pressure in the nozzle chamber and therefore changes the output pneumatic pressure.

The pressure drop across an orifice or similar flow restriction is commonly used to measure flow. Unfortunately the pressure difference does not vary directly with the flow but varies with the square of the flow. In other words if the flow doubles, the differential pressure across the orifice will increase by four times. Square root extractors are used to produce a

linear flow signal from such a differential pressure signal. Figure 5-64 shows the use of a square root extractor to linearize the signal from a differential pressure transmitter measuring the pressure drop across an orifice. Figure 5-65 is a simplified pneumatic square root extractor based on use of a nozzle-flapper amplifier. The pivot of the horizontal beam is moved by the input from the differential pressure transmitter. The outlet pressure from the nozzle-flapper amplifier pressure is supplied to the restoring bellows. The restoring bellows moves the vertical beam, changing the nozzle-flapper relationship. The relationship between the two beams is such that the vertical beam must move a distance proportional to the square root of the movement of the horizontal beam. The outlet pressure is thus proportional to the square root of the inlet pressure.

Another device commonly used for measuring the fuel oil flow in combustion control systems is the area meter shown in figure 5-66. Oil from the service pump enters the bottom ofthe meter, forcing the metering plug up. The plug is restrained by a calibrated spring which exerts a constant force on the plug as it rises. The rising plug uncovers ports in the cage, allowing the oil to flow out the exit ports. The plug rises until the pressure-times-area force on the plug equals the spring force. The ports are shaped so that the movement ofthe metering plug is directly related to the oil flow. A pneumatic transmitter converts the movement ofthe plug spindle into a pneumatic pressure signal. This outlet pressure is therefore directly related to the oil flow and no square root extractor is required.

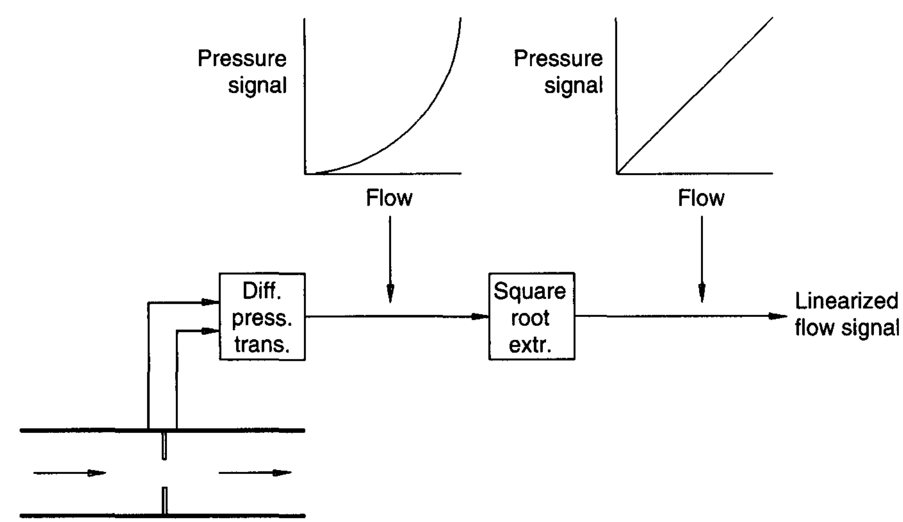


Figure 5-64. Flow measurement using a square root extractor

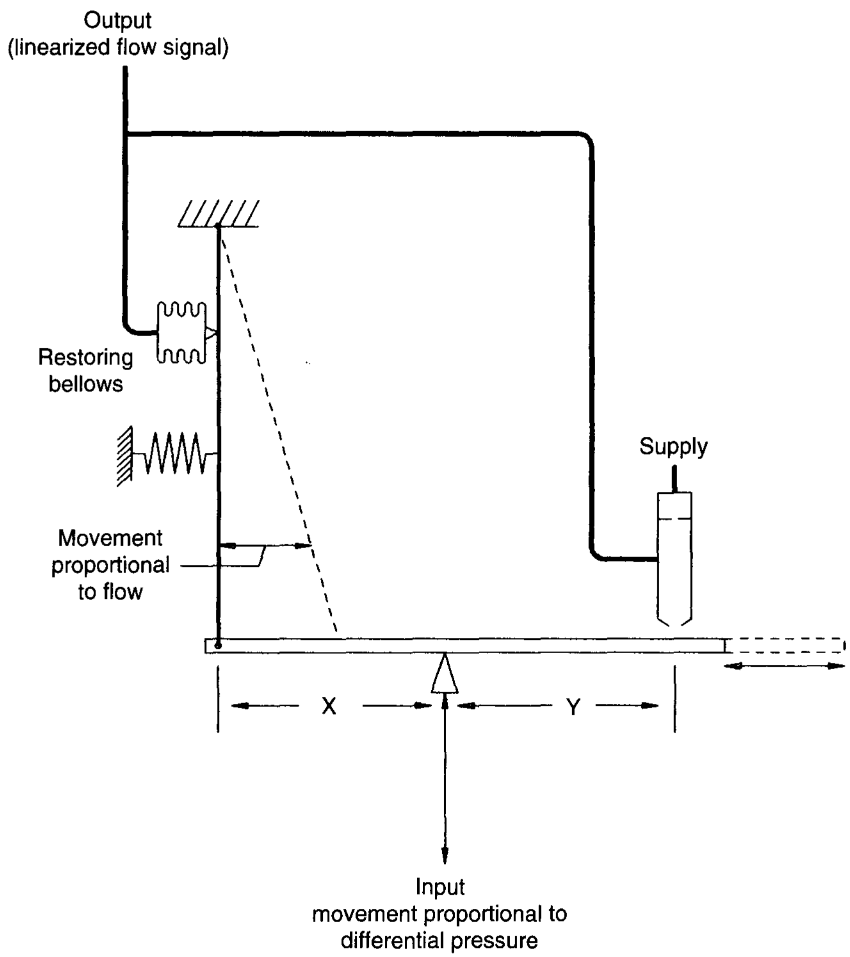


Figure 5-65. Simplified pneumatic square root extractor

Figure 5-67 shows a typical hand/auto station used in an Elsag Bailey pneumatic control system. It permits the operator to conveniently shift from automatic operation to manual and back again. The two operator controls are the auto/hand selector valve and the hand pressure regulator. There are three indicators on the panel, basically 3 to 15 pressure gauges which display pneumatic signals between the minimum (0 percent or 3 psig) and the maximum (100 percent or 15 psig). The far left indicator (A) shows the pressure coming into the station from the upstream controller. The far right indicator (D) shows the pressure being delivered to the downstream component. The transfer pressure is displayed on the indicator (C) just to left of the outlet indicator (D). When the selector is in "auto," this

gauge displays the outlet pressure from the hand pressure regulator. If the selector is switched to "hand," this will become the pressure delivered to the downstream component. When in "hand," the transfer indicator CC) displays the auto loading pressure. Again, ifthe selector is switched, it indicates what the output pressure CD) will become. Some hand/auto stations have a third knob and a fourth indicator. This third knob is a bias relay and

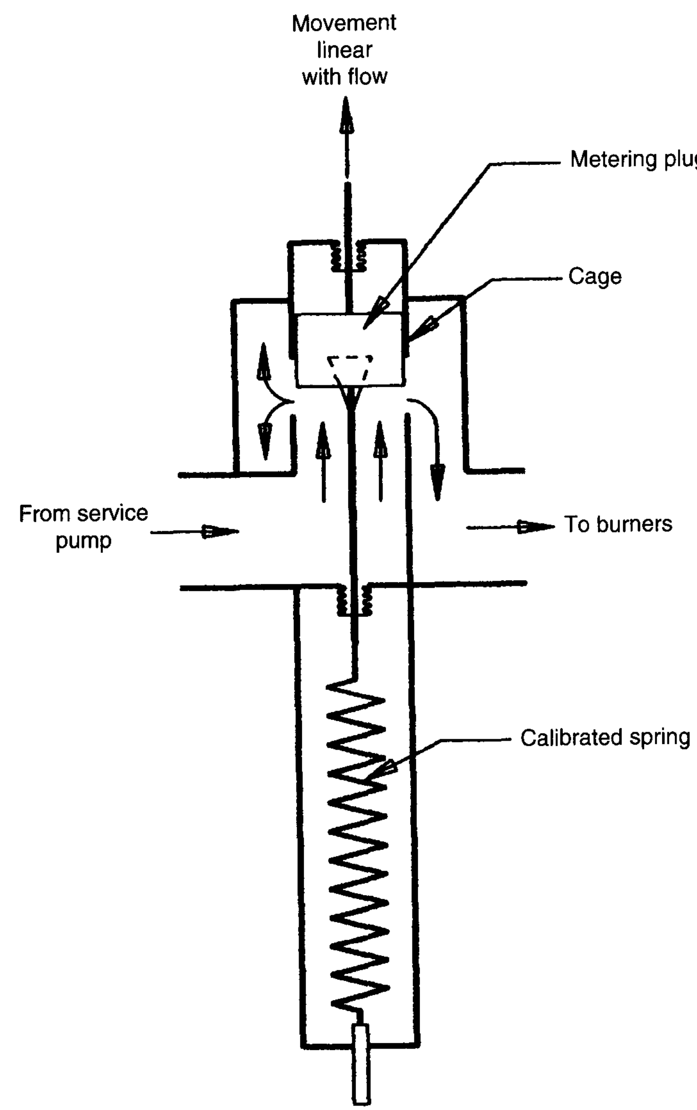


Figure 5-66. Fuel oil area meter

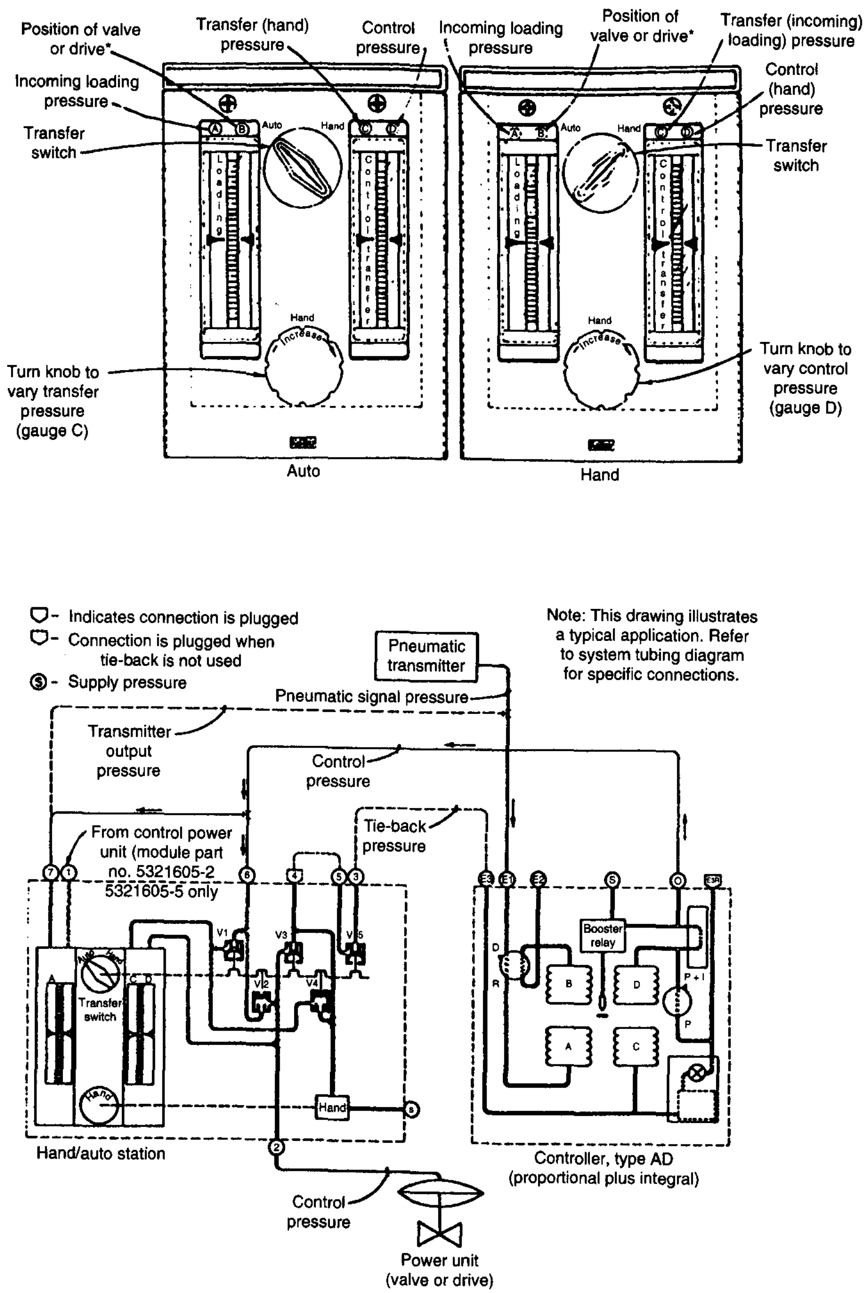


Fig. 5-67. Hand/auto station. Courtesy Elsag Bailey, Inc.

permits trimming the outlet pressure when in automatic mode. The fourth indicator (B) is a display of the bias relay setting. An indication of 50 percent means the bias relay is neutral and is neither adding to nor subtracting anything from the signal. Readings above 50 percent indicate positive bias and below 50 percent indicate negative bias.

The objective when switching modes is to achieve a "bumpless" transfer. This means that the outlet pressure should not suddenly increase or decrease, upsetting the process being controlled. To achieve bumpless transfer when going from "auto" to "hand," the hand regulator knob is turned until the transfer indicator (C) matches the outlet indicator CD), then the transfer valve is shifted to "auto." Achieving bumpless transfer when going from "hand" to "auto" is a little more complicated. Since the transfer indicator is basically displaying the incoming pressure (A), it is necessary to change the incoming pressure until it matches the outlet pressure (D) currently being controlled by the hand knob. To do this, the set point and/or bias adjustments ofthe upstream controllers must be changed to match the two gauges. At this point the selector can be switched to "auto" without affecting the controlled process. It is not good practice to use the hand knob to match the outlet pressure (D) to the transfer pressure (C). This just disturbs the controlled process prior to the transfer.

The final task of the combustion control system is to physically stroke the fuel oil control valve and move the air damper, changing the fuel and air flow. Pneumatically operated diaphragm and piston actuators are used to accomplish this task based on the pneumatic signals supplied. (Fig. 5-68 shows a typical diaphragm actuator used with a control valve.) As air pressure supplied to the top of the diaphragm increases, the diaphragm will move down until the pressure-times-area force equals the spring force. Full travel of the valve will normally occur as the loading air pressure changes from 3 psig to 15 psig.

Figure 5-69 shows a typical piston actuator used to open and close a forced draft fan air damper. By using higher air pressure (typically 100 psig), the piston actuator can generate higher forces than a diaphragm actuator. Note that the actuator spring of the diaphragm actuator in figure 5-68 has been eliminated. Air pressure is used to drive the piston in both directions. A positioner is used to control the main air flow to the piston actuator. When the control air input signal changes, the pilot valve is moved off center, sending air to the piston. The actuator movement is fed back to the positioner through the positioning beam and spring, eventually returning the pilot valve back to the center neutral position.

Other devices found in combustion control systems are selects, biases, limits, and summers. A select, also called an auctioneer, has two inputs and one output. The output becomes the higher ofthe two inputs in a high select, and the lower of the two inputs in a low select. A bias relay adds or subtracts a constant value to the input signal. A limit will prevent a signal from going higher than a particular value (high limit) or lower than a particular value (low limit). A summer takes two inputs and adds them together.

ELECTRONIC SYSTEMS

Combustion control systems using electrical components have been designed and built for many years. An electromechanical system popular in the 1950s and 1960s is described in *Modern Marine Engineer's Manual,* volume 2. In this system, reversible electric motors open and close the fuel oil valves and forced dra(t dampers.

Early electronic systems were based on the use of analog components. Analog electronic amplifiers built with discrete components replace the nozzle-flapper used in pneumatic control components. Voltage signals varying from 0 to 10 volts dc replace the 3 to 15 psig pneumatic signals. Later analog systems are based on integrated circuit operational amplifiers (Op Amps).

As in many other areas, digital systems rather than analog systems are being selected for new applications. (See volume 2 for a detailed description of digital control systems and components.) Many of the newer systems are based on the use of a microprocessor. Figure 5-70 shows a complete microprocessor-based combustion control system, contained in a box less than 1 cubic foot in volume. Essentially a dedicated special-purpose computer system, the various control components of the pneumatic

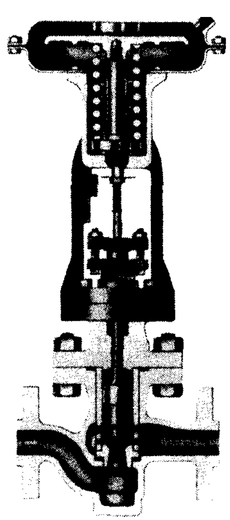


Figure 5-68. Diaphragm actuator. Courtesy Leslie Controls, Inc.

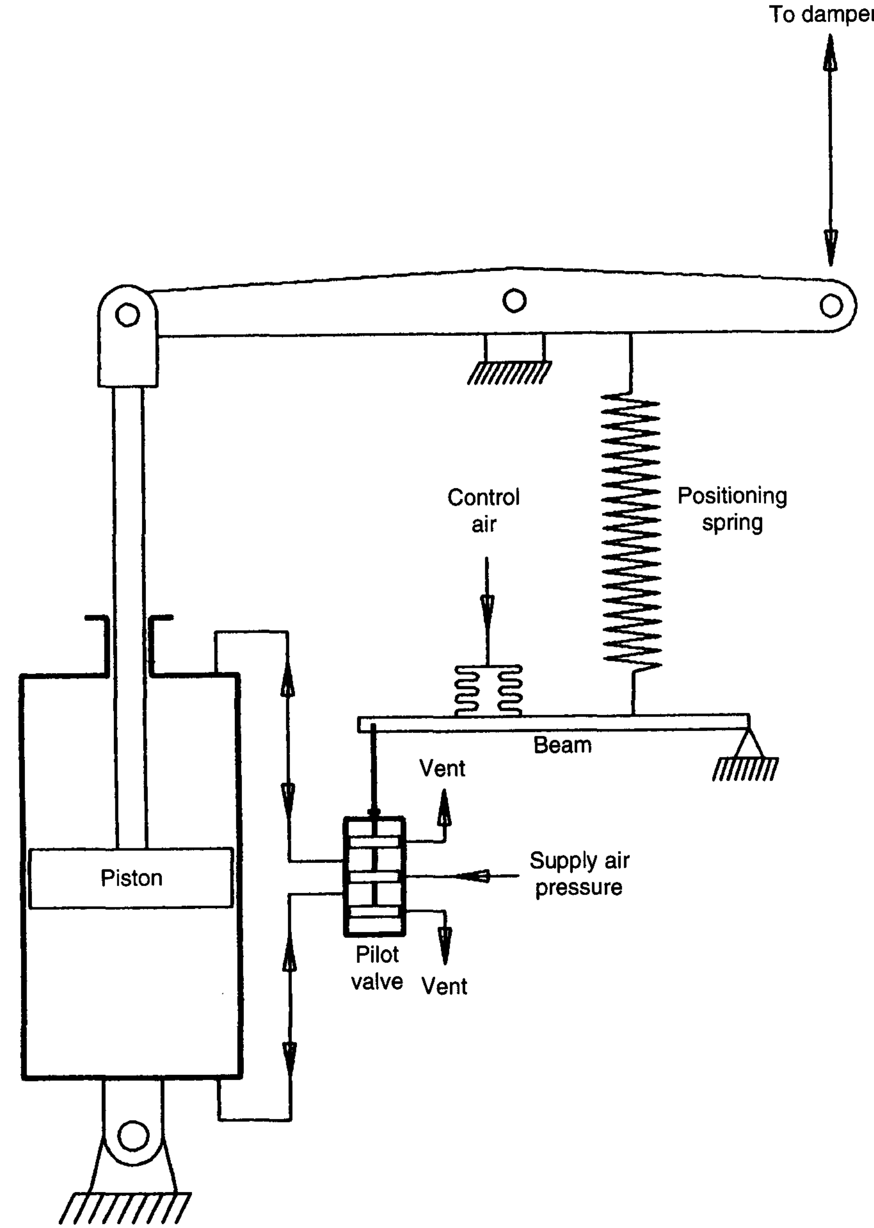
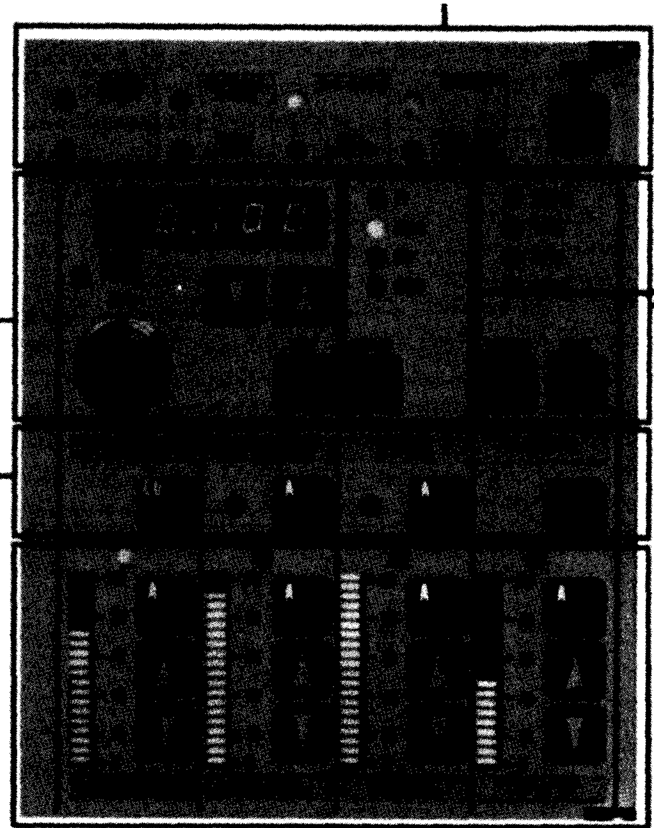


Figure 5-69. Piston actuator

The INCR-DECR pushbuttons under the 4½ digit LED display will change the displayed variable. The key switch prevents unauthorized personnel from changing configuration and tuning data. Examples of variables that may be changed are setpoints and loop outputs. The loop pushbutton cycles through the loop LEDs to identify the loop being displayed. The SERVICE MANUAL LED is lit when the analog tracking circuitry is controlling the outputs.

An eight alarm first out annunciator panel with customized legends and an alarm acknowledge pushbutton.

The DISPLAY pushbutton is used to select the process variable, local setpoint or bias, remote setpoint, or percent output of the loop identified by the LOOP LEDs.



The four auxiliary backlit pushbuttons may be configured to select percent or engineering units, perform automatic oxygen calibration, or serve as remote/local pushbuttons. The two pushbuttons with adjacent loop LEDs may also be configured to be Auto/Manual pushbuttons, and serve as auxiliary loops internal to a control system.

The four controlled outputs are indicated on separate bargraph indicators. Each primary loop has its own backlit Auto/Manual pushbutton and separate increase/decrease pushbuttons. The loop LEDs above the meters are used by the LOOP pushbutton to select the loop being displayed on the digital readout. The customized loop legends are used to specify the process variable and its engineering units.

The NUMBER AND VALUE pushbuttons identify configuration information and are used along with the shared INCR-DECR pushbuttons to enter configuration and tuning data.

Figure 5-70. Model 1500A combustion controller. Courtesy Rosemount Analytical, Inc.

or analog electronic control system are simulated with lines of computer code stored in read-only memory (ROM) as subroutines. The programming required for a particular application is thus greatly simplified, based on a block diagram similar to the control system logic diagrams discussed above.

While the controller is digital, the inputs and outputs are still analog, typically 4 to 20 mA dc current signals. This allows the use of standard electronic transmitters to measure the various plant parameters. However, since the actuators used are still typically pneumatic diaphragm or

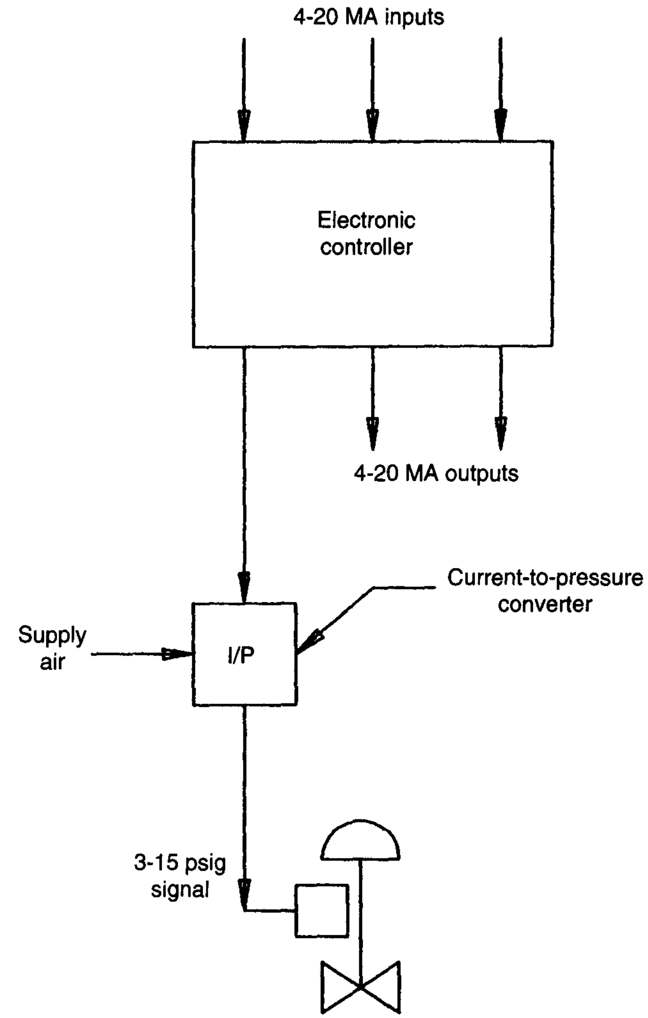


Figure 5-71. Current-to-pressure converter

piston units, the 4 to 20 current outputs from the controller must be converted to standard 3 to 15 psig pneumatic signals. This is performed by a current-to-pressure *(I/P)* converter, as shown in figure 5-71 (preceding page).

Feedwater Regulator Systems

The function of the feedwater regulating system is to regulate the flow of feedwater into the steam drum, maintaining the water level in the drum

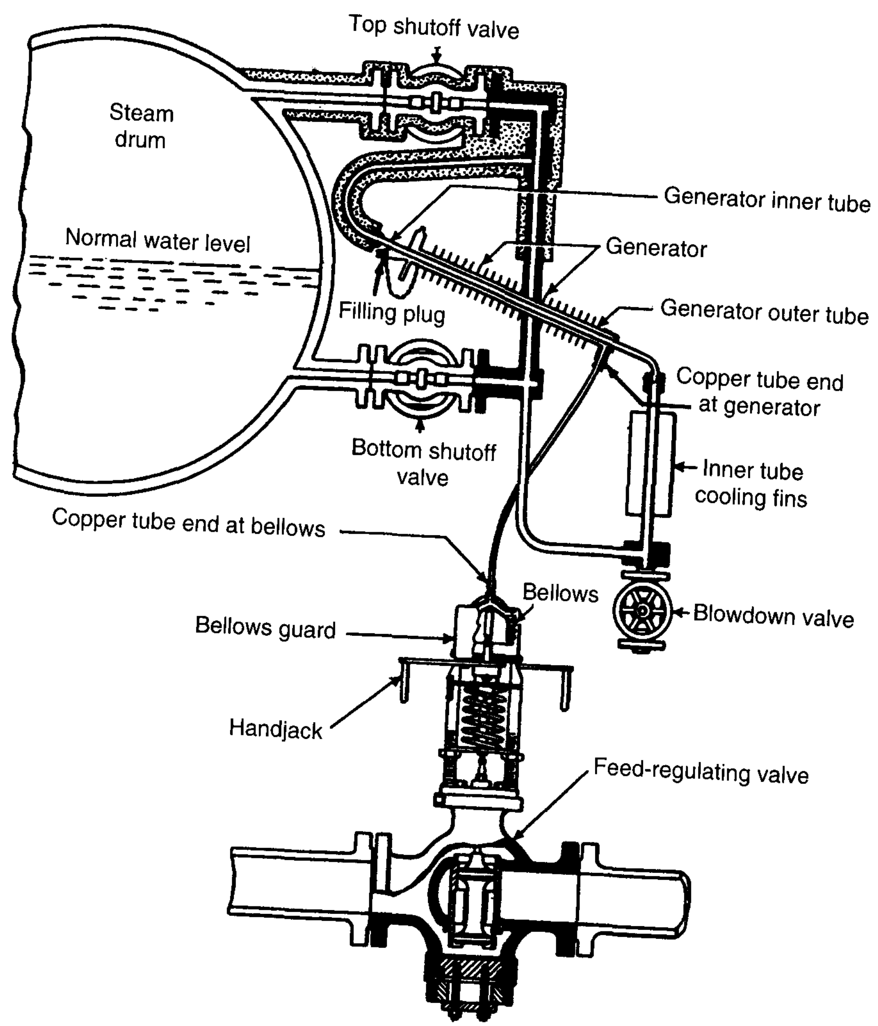


Figure 5-72. Thermohydraulic feedwater regulator.
Courtesy Babcock & Wilcox.

between the desired limits. The complexity of the control system will depend on the type and capacity of the boiler as well as characteristics of the load. High-pressure and high-capacity propulsion boilers that undergo rapid load changes require more sophisticated systems than do low-pressure auxiliary boilers that essentially operate at constant load.

Many package and auxiliary boilers in the lower capacity and temperature-pressure range are fitted with self-contained, single-element feedwater control systems. The thermohydraulic regulator shown in figure 5-72 is typical of such systems. The inner tube of the generator is connected to the steam drum so that the water level inside the tube tracks the level in the drum. Because the water in the inner tube is essentially stagnant, it will cool to a temperature below that of water and steam in the drum. The chamber between the inner and outer tube is partially charged with water. If the level in the drum drops, more of the water charge in the generator is exposed to the higher-temperature steam. This increases the pressure in the generator, opening the feedwater regulating valve and bringing the drum level up. If the drum level increases, more of the generator charge will be exposed to cooler water, lowering the generator pressure, closing the valve, and bringing the drum level down.

In a single-element feedwater regulator like the thermohydraulic one above, the only input to the system is the drum level. Figure 5-73 is the logic diagram for a typical single-element system. This is satisfactory for automatically controlling drum level during slow load changes, but will not perform well during rapid load changes due to "swell" and "shrink." Swell

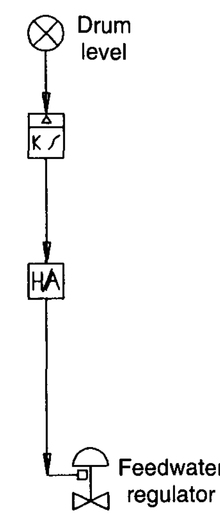


Figure 5-73. Single-element feedwater regulator

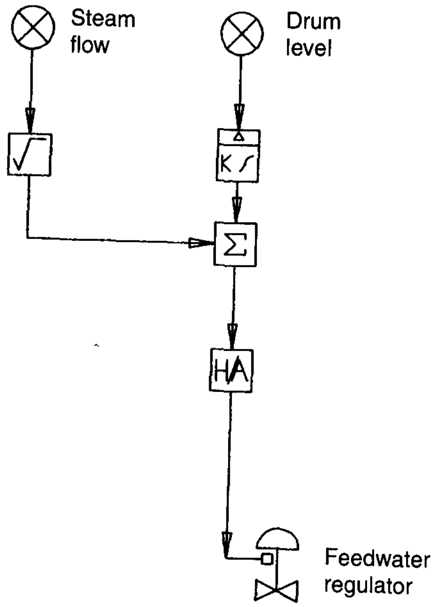


Figure 5-74. Two-element feedwater regulator

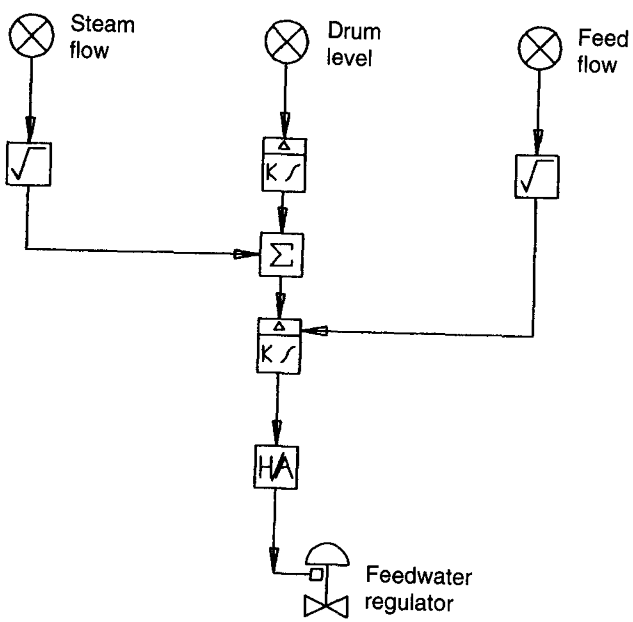


Figure 5-75. Three-element feedwater regulator

occurs during an increase in load. The load increase causes a temporary drop in pressure, and some of the water in circulation flashes to vapor. Since the vapor has a greater volume, the drum level increases (swells), and the feedwater regulator closes somewhat. Since the feedwater flow decreased when the steam production increased, once the steam pressure is restored to normal by the combustion control system, the drum level will be low. During a load decrease, the boiler pressure increases, steam bubbles collapse which lowers the drum level (shrink), and the feedwater regulator opens. When the boiler pressure is restored, the drum level will be high. In each case the system was fooled by the swell or shrink, stroking the valve in the wrong direction and causing wide variation in drum level.

The solution to this problem is to use a two-element or a three-element feedwater regulating system. A two-element system as shown in figure 5-74 has two inputs-drum level and steam flow. The steam flow input is used as a feedforward control signal, initially setting the valve position during a load change. During steady-state operation, the system responds to the drum level input. The steam flow feedforward input provides the necessary adjustments to cope with swell and shrink during rapid load changes.

A two-element system is essentially a positioning system as defined above for combustion control systems. By the addition of an input for feedwater flow, the system becomes a three-element feedwater regulating system. The system is now metering, because the parameter being controlled (the feedwater flow) is now being measured (see fig. 5-75). As long as the system maintains the feedwater flow equal to the steam flow, the drum level will basically take care of itself. In most marine propulsion boiler installations, a two-element system provides satisfactory control.

Superheat Temperature Control Systems

For superheaters fitted with an internal control desuperheater (attemperator) for superheat temperature control (fig. 5-36), a simple temperature controller will provide satisfactory operation. The controller responds to the superheater outlet temperature and varies the flow through the desuperheater to maintain the outlet temperature constant. Figure 5-76 shows a typical attemperator control curve for a marine boiler. For boiler loads below the control point, the control valve will be closed and the superheater outlet temperature will be determined by the uncontrolled characteristic. Above the control point, the control valve will open as necessary to bring the temperature down to the design temperature. The control point can vary from 60 to 80 percent depending on the boiler design.

For spray desuperheater installations, a single-element, two-element, or three-element control system can be used. (Fig. 5-77 shows logic diagrams for these systems.) The controller strokes the valve, varying the water flow to the desuperheater. With a single-element system, outlet temperature is the only input to the controller. With a two-element system, outlet temperature and outlet steam flow are the two inputs to the controller. With a three-element system, measurement of the inlet water flow is added. As with feedwater regulating systems, two- and three-element systems provide better control during rapid load changes.

Burner Management Systems

A burner management system assists the operator in the monitoring and operation of the burners in a boiler. A typical burner management system can include subsystems 'for flame detection, furnace purging, burner light-off, burner shutdown, and safety limits and alarms. In its simplest form, the system monitors the flame, shutting off the fuel in the event of flame failure. In its most complete form, the system completely automates the boiler light-off and burner sequencing and provides complete boiler safety monitoring, permitting unattended boiler operation.

Flame detection is an essential component of any burner management system. The flame detector must be reliable, it must be sensitive enough to detect the flame during low firing rates, and it must not be fooled by such things as hot refractory or an adjacent flame. Two sensors that have been used successfully are the ultraviolet detector and the "flicker" detector. Ultraviolet light, unlike visible or infrared light, is abundant in fossil flames

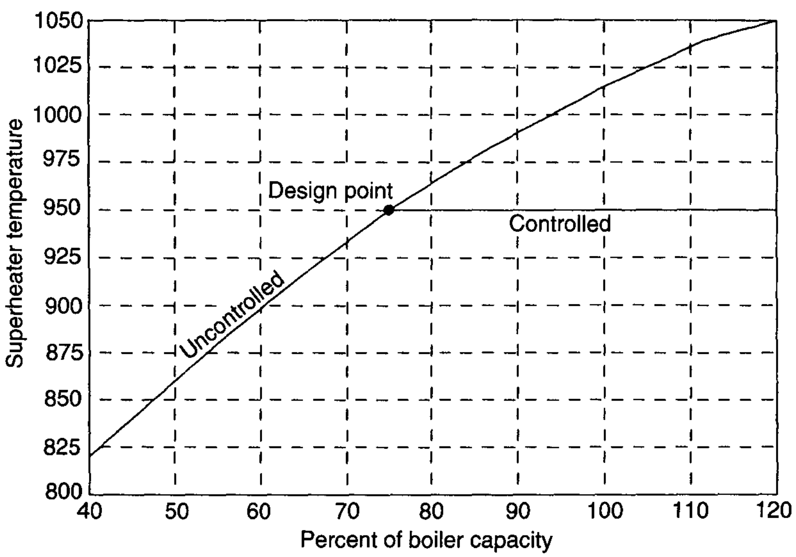


Figure 5-76. Superheat attemperator control curve

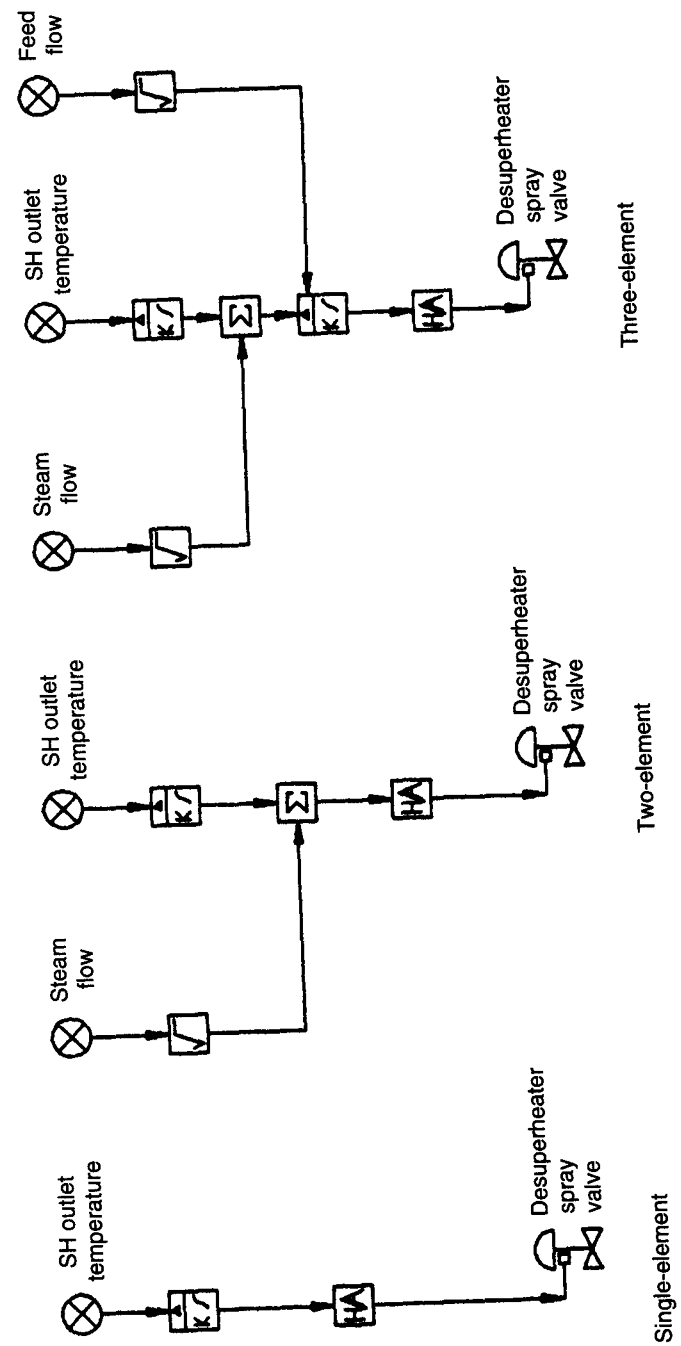


Figure 5-77. Spray-type desuperheater control

but is not emitted in significant quantities from hot bodies at the temperatures encountered in boiler furnaces. Figure 5-78 shows an ultraviolet detecting tube. The tube is filled with helium at low pressure and the electrodes are pure tungsten. Photons of ultraviolet light cause the release of electrons from the tungsten. Visible and infrared light do not possess enough energy. The light from a flame is not steady, but varies continuously in intensity. The emissions from hot refractory or boiler tubes do not vary. The frequency of this variation in light intensity is between 2 and 600 cycles per second. When a frequency detection technique is used, the sensor is termed a flicker detector.

The two major sequences of operations that can be easily automated are boiler purge and burner-light off. In most burner management systems, these sequences are initiated by the pressing of a single button. At that point, the automatic sequencer takes over, initiating the necessary operations in the proper sequence and with proper timing and delays between steps. The sequence is monitored and the process safely stopped if events do not OCcur when they are supposed to. While these sequences will vary

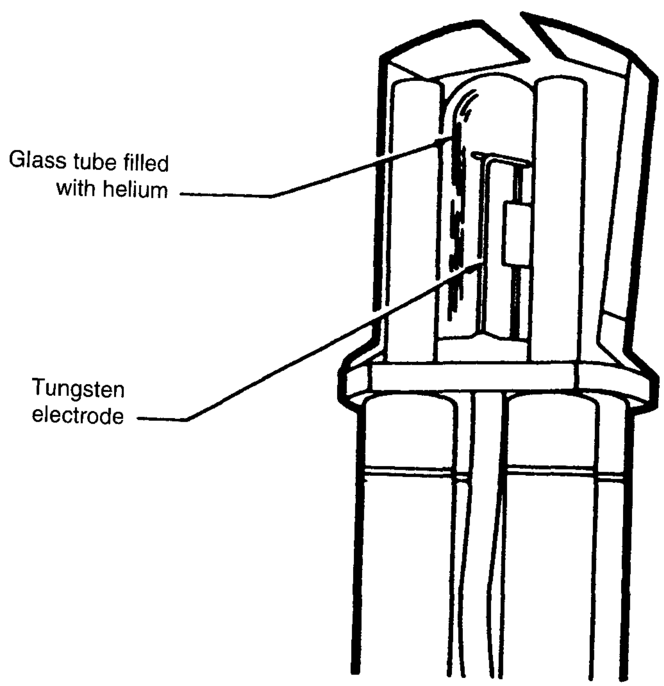


Figure 5-78. Ultraviolet flame detector.
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based on the installation and the equipment manufacturer, listed below are some typical sequences for boiler purging and burner light-off.

PURGE SEQUENCE

1. Check permissives:
   - wind box pressure above minimum (fan running)
   - main fuel oil valve closed
   - burner fuel oil valves closed
   - registers closed
   - no burner trip conditions exist
2. Set wind box pressure at purge pressure (5 inches).
3. Open all registers.
4. Verify wind box pressure and registers open, wait 20 seconds, then close registers.
5. Reduce wind box pressure.
6. Send "purge complete" permissive to burner sequencer.

BURNER LIGHT-OFF SEQUENCE

1. Check permissives:
   - wind box pressure above minimum (fan running)
   - main fuel oil valve closed
   - burner fuel oil valves closed
   - registers closed
   - no burner trip conditions exist
   - no flame detected
   - atomizing steam pressure above minimum
   - fuel pressure above minimum
   - fuel temperature above minimum
   - purge complete
2. Open main fuel oil valve.
3. Set "low fire" air pressure (0.5 to 0.75 inches).
4. Set light-off fuel pressure (70 psig).
5. Insert ignitor.
6. Open fuel oil burner valve.
7. Ifflame not detected within 10 seconds, send "secure burner" signal. Ifno other burners are on, send "boiler trip" signal.
8. After flame detection, open register.
9. Retract ignitor.
10. Set fuel pressure at "low fire" (50 psig max.).

The shutdowns included in the safety portion of the burner management system fall into two basic categories: burner trips and boiler shutdowns. A burner trip would result in the burner valve and register of a single burner being closed. A boiler shutdown would result in all fuel oil valves and register in the boiler being closed, and a purge initiated. The following conditions would typically initiate a burner trip:

1. Flame not detected within 10 seconds during burner light-off
2. Loss of flame
3. Burner valve not fully open
4. Register not fully open
5. Loss of atomizing steam

The following conditions would typically initiate a boiler shutdown:

1. Low-low water in boiler
2. Loss of all flames in boiler
3. Loss of wind box pressure
4. Low fuel pressure and burner valve(s) open
5. Low fuel temperature and burner valve(s) open
6. Unsuccessful light-off of first burner
7. All burner valves closed

The other main element of the safety system is the alarm system. Almost all the permissives listed above have alarms associated with them. Any of the conditions that initiate a burner trip or boiler shutdown would also set off an alarm. The alarms provide both a visual (light) and audible (horn) indication that a fault has occurred. The typical alarm logic sequence is as follows:

	Light	*Horn*
Normal condition	off	off
Fault condition	flashing	on
Acknowledge alarm	steady	off
Fault cleared	off	off

The burner management system, unlike the combustion control system, operates with discrete (on-off) signals and commands. Valves are either open or closed. A pressure is either acceptable or not. Switches (pressure, temperature, and limits) provide the major inputs to the system. Solenoid valves are the most common device actuated by the system. Traditionally, a burner management system would be designed and built using such components as relays and timers. Today, many such systems are being built using an electronic device called a programmable logic controller (PLC). A PLC is a microprocessor-based controller with a built-in application program which enables control system designers to easily develop control systems. The unit can be programmed with the aid of a personal computer (PC). On completion of the program development on the PC, the program is downloaded to a programmable read-only memory (PROM) on the controller. While originally developed for discrete relay control systems, PLCs now can also be programmed for analog systems with proportional, PI, and PID functions built in.

Programmable logic controller are available from dozens of manufacturers in a range of sizes from micro PLCs that handle a few dozen digital inputs and outputs to units capable of controlling large complex processes with thousands of digital and analog inputs and outputs. Figure 5-79 is a GE Fanuc Series 90-30 PLC suitable for small to midrange control applications. A variety of input/output (I/O) modules can be added to the basic unit to interface the required digital and analog inputs and outputs. The least powerful unit in this series can handle 160 inputs or outputs while the most powerful model can handle up to 2,048 inputs and 2,048 outputs.

A digital control system is normally developed using a ladder logic diagram. A ladder logic diagram looks very much like the wiring diagram for a relay system. The PC program will convert the ladder diagram into the program code used by the PLC. Figure 5-80 is an example of a simple ladder diagram and the PLC program code equivalent to the diagram.

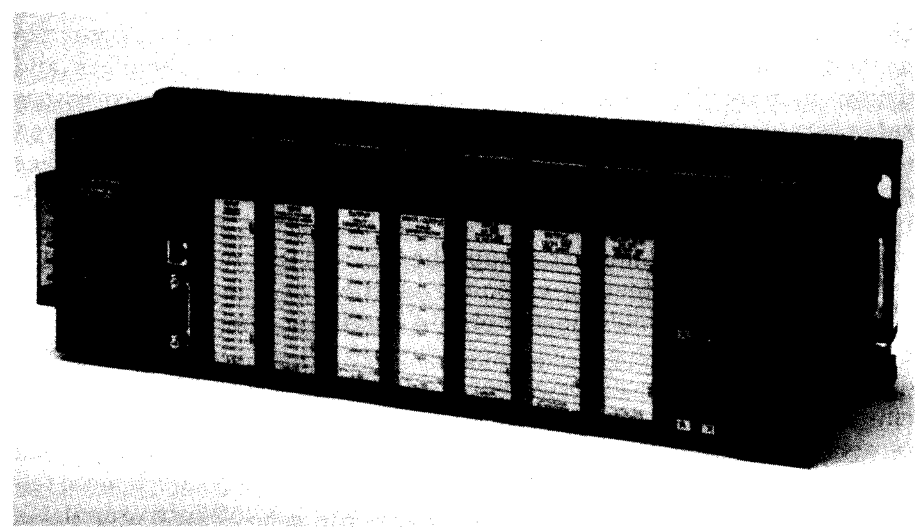
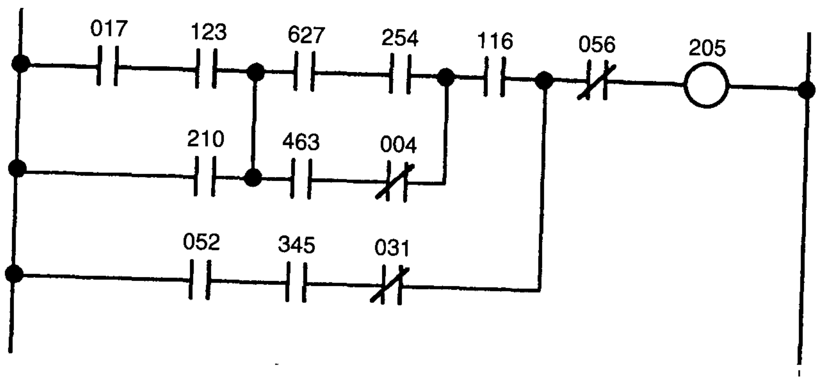


Figure 5-79. Programmable logic controller. Courtesy GE Fanuc Automation.




```
STR      017     AND NOT  004     AND NOT  031
AND      123     OR STR           OR STR
OR       210     AND STR          AND NOT  056
STR      627     AND      116     OUT      205
AND      254     STR      052
STR      463     AND      345
```

Figure 5-80. Ladder logic diagram and PLC program

BOILER OPERATION AND MAINTENANCE

Boiler Water Treatment and Control

Modern high-pressure boilers require careful monitoring and control of the type and quantity of impurities found in the boiler water. Incorrect or inadequate treatment can result in the failure of tubes, headers, and drums and result in costly unscheduled shutdowns. High corrosion rates can result from improper water pH or failure to remove dissolved oxygen from the water. Scale formation on the inside of the boiler tubes due to impurities such as calcium sulfate can result in tubes overheating. High levels of dissolved and suspended solids can also result in carryover, causing problems in the superheater and turbines. A program of regular boiler water testing, chemical addition, and blowdown is necessary to avoid such problems and obtain maximum service life from the boiler.

Table 5-1 lists typical recommended treatment control limits for marine boilers of various operating pressures. One thing to note is that the recommended limits for the various parameters decrease as the operating pressure increases. Also, a 125 psig auxiliary boiler will be much more tolerant of a temporary control deviation than will a 1,450 psig propulsion boiler. Note that the addition of treatment chemicals controls some parameters (pH, alkalinity, phosphates, sulfite, hydrazine), while others are controlled by blowdown (total solids, conductivity, chlorides). The impor-

tance of the various parameters and the techniques commonly used for their control will be discussed in the following sections.

TABLE 5-1
Recommended Treatment Control Limits

	100-200 psig	*450-650 psig*	*850-950 psig*	*1500 psig*
pH	10-11	10.5-11	10-10.3	9.7-10.2
Alkalinity (ppm as CaC03)	250-350	100-200	0	0
Phosphate (ppm as P04)	40-80	20-40	10-20	5-15
Sulfite (ppm as S03)	20-40	10-20	none	none
Hydrazine (ppm in feed)	.03-.10	.02-.04	.01-.03	.01-.03
Total Solids (ppm, max.)	3,500	2,500	1,500	750
Conductivity (micromhos)	500-1,000	100-300	75-200	50-100
Chloride (ppm)	20-40	15-30	10-20	5-10

Notes: 1. Assumes phosphate-hydroxide treatment for 100-200 and 450-650 psig ranges and coordinated phosphate for 850 psig and above.
2. Based on use of hydrazine for pressures of 850 psig and above.

CONTROL OF PH AND ALKALINITY

The pH scale is used to express the degree of acidity or basicity of a solution. Although a solution may contain many ions, the balance of hydrogen (H+) and hydroxyl (OH-) ions determines if a solution is acidic or basic. When the hydrogen ions exceed the hydroxyl ions, the solution is acidic. If the hydroxyl ions exceed the hydrogen, the solution is basic. Pure distilled water is neutral and has an equal balance of hydrogen and hydroxyl ions, about 10^{-7} gram equivalents per liter of each at 25°C. The pH is expressed mathematically as the negative logarithm to the base 10 of the hydrogen ion concentration. Neutral water thus has a pH of 7. An acid would have a pH less than 7 and a base would have a pH greater than 7. Table 5-2 shows the relationship between hydrogen and hydroxyl ion concentration and pH.

Many salts (most phosphates, carbonates, and bicarbonates for example) produce a basic solution when dissolved in water. This is the basis for the use of alkalinity to express the pH of basic solutions. Boiler water alkalinity is normally expressed in units of ppm (parts per million) of CaC03. An alkalinity of 200 ppm means the solution has the same pH as water with 200 ppm of CaC03 dissolved in it.

At the high temperatures found in boilers, the iron in the steel tubes, drums, and headers reacts with water. The end product of this reaction is magnetite (Fe304). The only reason that boiler steel can survive normal boiler operating conditions is that under certain conditions, the magnetite forms a protective layer, limiting further corrosion. One of the roles of water treatment is to maintain the pH of the water in the range where the magnetite film can stabilize and protect the steel from accelerated reaction with the water and other impurities introduced with the feedwater. Research has shown that the protective layer is stabilized if the pH is maintained between 5 and 13, and corrosion is a minimum when the pH is between 9 and 11. For pH levels below 5 or above 13, corrosion increases significantly. As the next section will show, proper pH control is also important for proper functioning of the scale control additive.

TABLE 5-2
Hydrogen and Hydroxyl,Ion Concentrations Versus pH Values at 250C

	pH	H + Concentration (gm equiv. / liter)	OH - Concentration (gm equiv. / liter)
	0	100	10-14
	1	10^{-1}	10-13
Acids	2	10^{-2}	10-12
	3	10^{-3}	10-11
	4	10-4	10-10
	5	1~	1~
Neutral	6	1~	1~
	7	1~	1~
	8	10-8	10-6
	9	10~	1~
Bases	10	10^{-10}	10-4
	11	10-11	10-3
	12	10^{-12}	10-2
	13	10^{-13}	10-1
	14	10^{-14}	100

CONTROL OF PHOSPHATES

Various impurities found in feedwater, primarily the sulfates, carbonates, and bicarbonates of magnesium and calcium, have the tendency to form a dense adherent scale on the inside of boiler tubes. The primary mechanism of scale formation is that compounds such as calcium sulfate ($CaSO_4$) have decreased solubility at higher temperatures. At the higher temperatures next to boiler tube surface, the compound comes out of solution and forms scale. The scale has a high resistance to heat flow, effectively insulating the boiler metal from the boiler water and causing the tube metal temperature to rise. The overheated tube metal loses strength, and the internal pressure causes a blister to form and the tube to burst.

The technique for the control of scale is to add phosphates to the boiler water. Compounds such as trisodium phosphate (Na_3PO_4) and disodium phosphate (Na_2HPO_4) will react with the scale forming magnesium and calcium salts, converting them to their respective phosphates, which are readily dispersed and removed by blowdown. For these reactions to go to

completion, the pH of the boiler water must be alkaline. By properly controlling pH and phosphate residual, the boiler is protected from both excessive corrosion and scaling.

The test most commonly used to measure the phosphate residual in the boiler water is based on color comparison. A molybdate reagent is added to a filtered sample of water. Any phosphate present reacts to form phosphomolybdic acid. The phosphomolybdic acid is then reduced by stannous chloride, producing a blue color. The intensity ofthe blue color in the sample is related to the phosphate residual in the water. The color in the sample is then compared with the standard colors in a comparator block to determine the ppm of phosphate. Concentrations of 5 to 40 ppm are commonly maintained in the boiler water.

CONTROL OF OXYGEN

The presence of oxygen accelerates the reaction of iron and water. This accelerated corrosion is typically localized, forming pits in the boiler metal. Pitting is commonly most prevalent in stressed areas of tubes such as at welds or cold-worked sections, at surface discontinuities in the metal, or at the locations of small deposits.

The approach used to avoiding pitting of boiler metal is to exclude all oxygen from the boiler water. This is accomplished primarily in two ways: (1) the proper operation ofthe deaerating feed heater and (2) the use of an oxygen scavenging chemical to react with any small residual not removed by the deaerator.

A properly operated and maintained deaerator will reduce the oxygen content of the feedwater to 0.007 ppm or less. The deaerator operates by raising the temperature ofthe feedwater to saturation, causing the oxygen to come out of solution and allowing it to be vented from the heater. If the heater is operating properly, the feedwater leaving the heater will be at the saturation temperature corresponding to the heater pressure. An exit temperature lower than saturation (subcooling) indicates a problem and should be investigated immediately. It is important to remember that the main condenser also contributes to oxygen removal. Minimizing air leakage into the condenser and eliminating subcooling of the condensate will contribute to the removal of air from the feedwater.

Two chemicals are commonly used for the scavenging of dissolved oxygen from feedwater: sodium sulfite (Na_2SO_3) and hydrazine (N_2H_4). Sodium sulfite reacts with oxygen to form sodium sulfate (Na_2SO_4), thus removing the free oxygen from solution and preventing accelerated corrosion. Sodium sulfite is an effective oxygen scavenging chemical. Its main disadvantage is that it increases the dissolved solids content of the boiler water and decomposes into acidic gases at high temperatures. Its use has been generally limited in recent years to boiler pressures of 600 psig and below. Concentrations of 5 to 40 ppm are typically maintained. The test for excess sulfite consists of titration of an acidified water sample with standard potassium iodide-potassium iodate solution. At the end point, free iodine is released, turning the sample a faint blue.

Hydrazine is used as the oxygen scavenger in most high-pressure boilers. Hydrazine reacts with oxygen to form nitrogen and water. Hydrazine has two important advantages over sodium sulfite: (1) the products of the reaction do not raise the dissolved content of the water, and (2) hydrazine also increases the pH of the condensate and feed, reducing corrosion in these systems and eliminating the need for pH treatment of the condensate. Because hydrazine decomposes rapidly at temperatures above 450°F, it is not added periodically to the boiler as are most other treatment chemicals, but is fed continuously into the feed system. The feed rate is controlled by monitoring the residual in the feed, not in the boiler water. The main disadvantage ofhydrazine is that the small residual ofhydrazine (0.02 to 0.06 ppm) that is maintained in the feedwater is not sufficient to handle large quantities of oxygen and thus requires an airtight feed system and a properly functioning deaerator. To test for the hydrazine residual, a reagent is added to the sample, causing a color change. The color intensity is compared to standard samples in a color comparator block.

CONTROL OF TOTAL SOLIDS AND CHLORIDES

Seawater contains about 35,000 ppm oftotal dissolved solids (TDS). These impurities are primarily sodium chloride and magnesium chloride. Seawater impurities can enter the boiler from two main sources: (1) the makeup feedwater and (2) leaks in the main and/or auxiliary condensers. The distilled water produced by the evaporators will typically have about 2 to 4 ppm total dissolved solids. This is well below the boiler water limits, so as long as the evaporators are operating properly and the makeup feedwater tanks are not being contaminated, this source is not a significant problem. However, because of the high dissolved solids content of seawater, even a minor condenser leak can be a serious problem for a high-pressure boiler. (See chapter 3 for a suggested procedure for operating a steam plant with a condenser leak.)

The total dissolved solids content of the boiler water is typically monitored by measuring the conductivity of the water with a meter. Pure distilled water has a very high resistance (low conductivity). Any dissolved solids added to the water will provide ions to allow the flow of electric current and the measured conductivity will increase. Conductivity cells are also used in the feed and condensate systems to warn the operator of condenser leaks or other similar problems.

Since the chemicals added to raise pH and phosphate levels in the boiler water also increase the dissolved solids level, the chloride test provides ad- :litional information regarding contamination of the boiler water. A couple)f different tests are available to check the chloride level. In one common test, potassium chromate indicator is added to the water sample and silver nitrate is added until the color turns a light reddish brown.

In normal boiler operation, the TDS and chloride levels will slowly rise over a period of days or weeks. Should either of the allowable limits be exceeded, the boiler must be blown down. After blowdown, the boiler must be tested and treatment chemicals added as required.

BOILER WATER TREATMENT SYSTEMS

Any complete system for the internal treatment of a boiler must accomplish a number of objectives. These include: (1) maintaining the pH alkaline, (2) preventing scale formation, (3) eliminating free oxygen, and (4) keeping the total solids below the allowable maximum for the operating pressure. The two systems in common use for marine natural-circulation boilers are the phosphate-hydroxide (conventional) treatment system and the coordinated phosphate treatment system.

The phosphate-hydroxide (conventional) treatment system is the most commonly used system on boilers below 800 psig. Caustic such as sodium hydroxide (NaOH) is added to raise the pH to 10 to 11 and phosphates are added to control scale. Either sodium sulfite or hydrazine can be used as an oxygen scavenger. Blowdown is used to limit total solids and chlorides.

In the coordinated phosphate system, no free caustic is maintained in the boiler water. At higher pressures and temperatures, sodium hydroxide can destroy the protective magnetite film. By relying on the tendency for phosphate compounds to form alkaline solutions, a combination of phosphates is used to precipitate scale forming salts and also to control pH. Trisodium phosphate has a much greater tendency to raise pH than does a similar quantity of disodium phosphate. By using these compounds in combination, it is possible to maintain both the pH and phosphate levels within desired limits. Figure 5-81 shows the relationship between pH and phosphate levels for the coordinated phosphate system. Hydrazine is typically used as the oxygen scavenger and blowdown is used to limit total solids and chlorides.

SAMPLING BOILER WATER AND ADDING CHEMICALS

The proper extraction of a representative sample ofthe boiler water is essential for accurate test results. The sample is commonly drawn from the steam drum, from the bottom blow connection. Because of the high temperature and pressure, the sample must be run through a cooler to avoid flashing (see fig. 5-82). The water should run for several minutes before the sample is taken. The sample container must be rinsed thoroughly with cooled boiled water. The container should be filled and capped immediately to avoid contamination.

If the testing of the boiler water sample indicates treatment is necessary, the appropriate chemicals must be injected into the steam drum

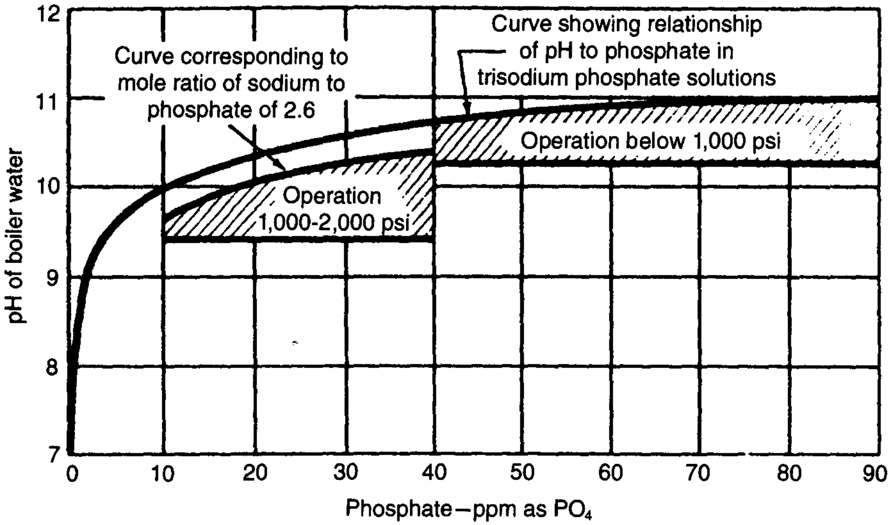


Figure 5-81. Coordinated phosphate treatment

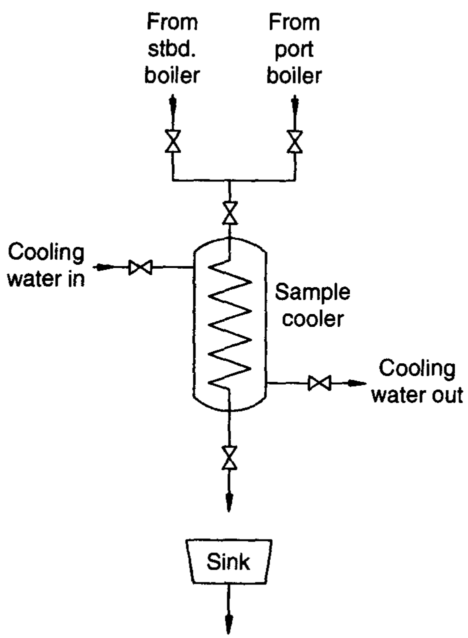


Figure 5-82. Boiler sample cooler system

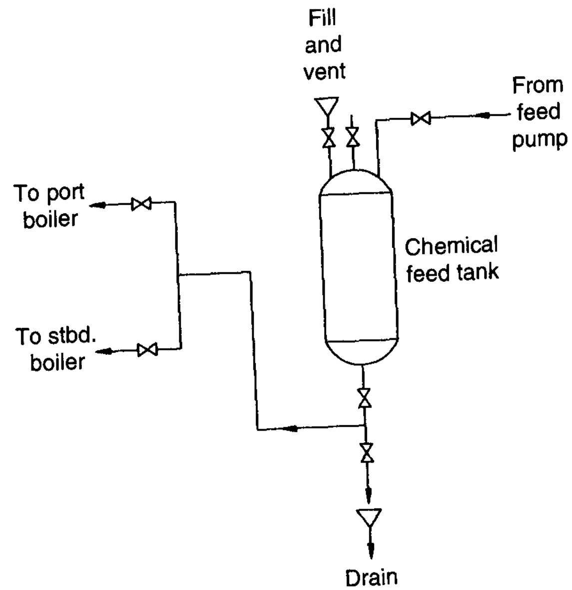


Figure 5-83. Boiler chemical addition system

through the chemical feed line. (Fig. 5-83 shows the typical piping arrangement for the chemical feed tank.) The chemicals are mixed and poured into the chemical feed tank and the tank is topped off with distilled water. The tank vent and fill are closed, and valves to the feed pump and the boiler to be treated are opened. The feed pump will thus inject the chemicals into the boiler steam drum. Precautions must be taken when opening up a closed tank; the boiler and feed pump valves must be closed and holding. This can be checked by carefully cracking the bottom drain.

Boiler Operation and Maintenance Procedures

The following operation and maintenance procedures are typical of those followed on most marine water-tube propulsion boilers. A particular installation may have unique characteristics that will require changes and variations from these procedures. Many can be adopted for use with auxiliary and waste-heat boilers. When in doubt, consult the manuals provided by the manufacturer of your equipment.

PREPARING A BOILER FOR SERVICE

The following procedure should be followed to prepare a boiler for service when the boiler has been secured for maintenance or has been out of service for some period of time.

1. Check that all maintenance has been completed. This includes checking for removal of all tools and installation of all handhole plates, manhole covers, access doors. If safety valve work was performed, check that gags are removed, lifting levers replaced, and easing gear functioning.
2. Be certain there is no accumulation of oil in the furnace or air casing. Wipe up any oil spills. If fireside washing was done, check that furnace drains are properly capped and loose fire bricks replaced.
3. Close the following valves: economizer drain, gauge glass drains, bottom blow, surface blow, header drains, chemical feed, auxiliary and main feedwater checks, auxiliary and main steam stops, and sootblower steam stop.
4. Open the following valves: steam drum vent, superheater header drains, superheater header vents, gauge glass shutoffs, remote level indicator shutoffs, pressure gauge shutoffs, feedwater regulator shutoffs, and feedwater stops.
5. Vent the economizer using the economizer vent valve.
6. Bring the boiler steam drum level to one inch above the bottom of the glass. If the boiler is full of water, drain the boiler until the water is at the bottom of the glass, then bring it up to check the feed supply. If the boiler is empty, fill to one inch above the bottom of the glass.
7. Prepare burners for operation. Inspect and clean all fuel oil strainers. Have atomizers made up with clean tips. Test registers for free operation.

NORMAL LIGHT-OFF

This procedure is followed when cutting in a second boiler. Since atomizing steam is available and the fuel oil system is in full operation, the regular steam atomizing tips can be used and the boiler lit off on residual oil. The procedure above (steps 1 through 7 for "Preparing a Boiler for Service") should be followed prior to beginning this procedure.

Several important precautions must be observed during the light-off and initial raising of steam pressure. Failure to follow proper procedures and observe precautions can result in serious damage to the boiler. Proper furnace purging is essential. Failure to purge properly can result in a furnace flare back or explosion. Do not attempt to light off with residual oil unless proper atomizing steam pressure is available and the oil is at the proper temperature at the burners.

During the initial firing, there is the danger of damage to the superheater. Until steam is being produced, there is no steam flow through the superheater. Once steam is being produced, only limited steam being vented is available for cooling. It is essential to fire only one burner at the minimum fuel oil pressure (50 psig or less) until the boiler is on-line and there is full flow of steam through the superheater.

1. Mter preparing the boiler for service, check the water level in the steam drum.
2. Purge the furnace. Line up the forced draft fan system. Check that fuel oil root valves are closed. Open all registers. Wind box pressure should be about 5 in. H_2O and furnace air should be changed at least 4 times.
3. Open fuel oil recirculating valve and recirculate oil from the burner header back to the service pumps until the proper operating temperature is reached at the header.
4. Insert an atomizer in the No.1 burner. Close all air registers. Open the atomizing steam and fuel oil root valves.
5. Adjust the wind box pressure for low fire-typically about 0.50 to 0.75 in H20. Set the fuel oil pressure at the burners to 70 psig.
6. Open the burner steam atomizing valve and allow steam to flow for at least 1 minute to clear line of condensate. Atomizing pressure should be 135 to 150 psig.
7. Close fuel oil recirculating valve. Open air register. Readjust fuel oil pressure to 70 psig.
8. Proceed with burner light-off. Open the burner fuel oil valve and, using a torch or ignitor (if fitted), ignite the atomized oil spray.
   Note: Ifignition does not occur within five seconds, secure the oil. Another purge ofthe furnace is essential before another light-off is attempted.
9. Mter ignition, reduce the oil pressure to not more than 50 psig. Reduce the forced draft pressure to establish a steady flame and avoid smoking.
10. Once steam is being produced and the superheater drains are blowing clear of condensate, close them. The drains should be opened periodically while raising steam to clear any condensate that may form.
    Note: Carefully observe manufacturer's recommended rate of raising pressure. (Fig. 5-84 is typical of these guidelines.)
11. When the steam pressure starts to rise, the steam drum vent may be throttled. Close the drum vent at a drum pressure of 50 psig. The superheater vent may be throttled as the boiler pressure increases but must remain at least partially open until the boiler is on the line.
12. As the steam pressure slowly increases, continuously monitor the steam drum level and maintain the water level with the auxiliary feed check.
13. As the boiler pressure approaches the line pressure, insert additional atomizers in the idle burners and turn on the atomizing steam. The atomizing steam will cool them until needed.
14. As the boiler pressure and line pressure equalize, open the main and auxiliary steam stops, bringing the boiler on-line. Close the superheater vent valve.
15. Switch the combustion control system and feedwater regulating system to automatic. Adjust the firing rate and number of burners in service to balance the load in each boiler.

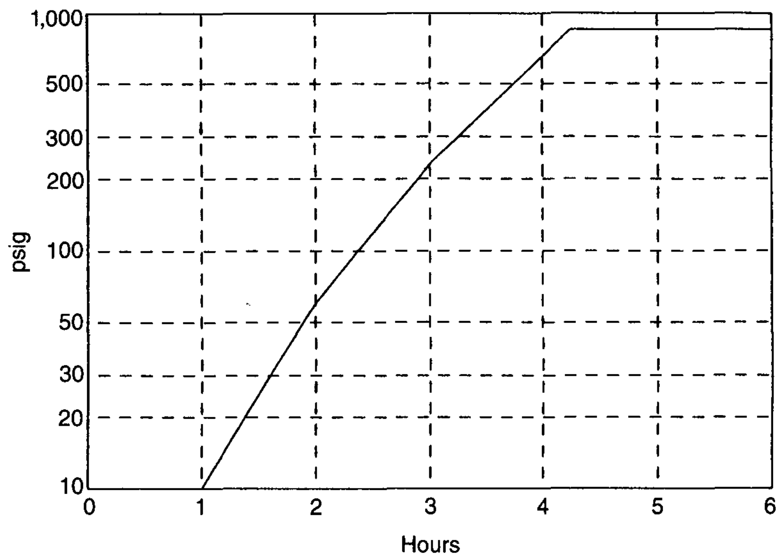


Figure 5-84. Guideline for raising boiler pressure

LIGHTING OFF-COLD PLANT

The following procedure should be followed when lighting off the first boiler from a cold plant condition. The procedure to be followed is similar to that for lighting off with one boiler already on-line. Some modifications are necessary since steam is not available for atomizing or for heating fuel oil. Diesel (distillate) oil will be used and the straight mechanical tips will be used in the atomizers. Electric power will be supplied from the emergency diesel generator or shore power until the boiler is on-line and supplying sufficient steam to run the main turbine generators.

Steps 1 through 7 for "Preparing a Boiler for Service" must be followed prior to beginning this procedure.

1. Follow the procedure described in steps 1 through 12 for "Normal Light-off." Since distillate fuel and mechanical atomizer ti ps will be used, modify those sections ofthe procedure as necessary. A feed pump will have to be prepared for service.
2. Prepare an atomizer with a steam atomizing tip. When sufficient steam pressure is available, supply steam to the fuel oil heater, fuel oil settler heating coils, and atomizing steam line. Start the feed pump.
3. Recirculate fuel oil to obtain the desired fuel temperature at the burner header. Install the atomizer in a secured burner and turn on the atomizing steam.

4. Secure the burner operating on diesel oil. Purge the furnace. Light offthe other burner on residual oil followingthe procedures outlined for norrnallight-off.
5. Continue to raise pressure and bring the boiler on-line following steps 12 through 15 of the procedure for normal light-off.

SECURING A BOILER

The following procedure should be followed when a boiler is to be taken off-line for maintenance, or for short- or long-term layup.

1. Operate the sootblowers when conditions permit.
2. Place the combustion control and feedwater regulator systems on manual. Secure the burners one at a time. Maintain atomizing steam flow until just prior to removing the atomizers.
3. Close the main and auxiliary steam stop valves. Crack open the superheater vent valve. Open it gradually as the boiler pressure drops.
4. Close the fuel oil root valve. With the forced draft blower still running, open all the registers to purge the furnace of any combustible gases. After a few minutes, close the registers and secure the forced draft fan.
5. Using the feed checks, raise the steam drum level to 3 to 5 inches above normal. As the boiler cools, add water as necessary to keep the level at least 2 inches above normal.
6. When the boiler pressure drops to 50 psig, open the steam drum vent.
7. When the boiler pressure drops to 15 psig, open the superheater header drains.
8. After the boiler has cooled, proceed with a short-term or a long-term layup, as described in the following sections.

BOILER LAYUP-SHORT- AND LONG-TERM

A boiler to be held out of service must be laid up properly. Failure to do so can result in accelerated corrosion ofthe boiler parts. While the protection of the watersides is of primary concern, the firesides of any boiler to be taken out of service should be thoroughly cleaned and dried. Soot on the boiler surfaces can absorb moisture and cause external corrosion. If a boiler is to be held out of service for more than 24 hours but fewer than 4 days, the steam blanket method or wet layup (short-term) method ,may be used. For periods longer than 4 days, the wet layup (long-term), dry layup with desiccant, or dry layup with nitrogen gas methods may be used. A description of each layup method follows.

Steam blanket layup method

1. Connect a temporary steam line from the auxiliary steam system to the steam drum vent or other suitable connection. When the boiler pressure drops below 150 psig, open the steam supply.

2. Keep all superheater vents and drains cracked. The superheater outlet header drain should be wide open. The drains and vents may be lined up to the drain system to minimize loss of water from the plant.
3. If the steam drum water level becomes too high due to condensing steam, open a waterwall header drain to lower the level back to normal.

Wet layup method: short-term

1. When the boiler pressure drops to 50 psig, open the steam drum vent. Allow the boiler to cool and pressure to continue to drop off.
2. Begin filling the b9iler slowly, using the auxiliary feed check. While filling, carefully monitor the feedwater supply to the operating boiler and the level in the deaerating feedwater heater.
3. When water flows from the steam drum vent, close the auxiliary feed check. Close the steam drum vent and the main and auxiliary feed stops. *Note:* The above protects drums, generating tubes, and waterwall tubes. The superheater and control desuperheater, however, are not protected. Wet layup of these is not justified for a short-term layup. The remaining steps outline the steps in protecting the superheater components with a nitrogen blanket.
4. Check that the main and auxiliary steam stops and bypasses are securely closed. Open the superheater temperature control valve wide.
5. Rig a nitrogen bottle and pressure regulator to the highest vent on the superheater. Check that the superheater drains are open.
6. Admit nitrogen at 2 to 5 psig into the superheater. The lighter nitrogen will displace the heavier air, forcing the air out of the superheater drains.
7. Once nitrogen begins flowing from the drains, close them. *Note:* Since nitrogen will not support combustion, a small candle can be used to check for the flow of nitrogen out of the vents. Exercise caution in the use of any open flame. A gas analyzer can also be used. Be alert to nitrogen collecting in enclosed spaces.
8. Maintain the nitrogen pressure of 2 to 5 psig on the superheater during the layup.

Wet layup method: long-term

1. In this procedure, the entire boiler, including the superheater, will be filled with condensate treated with hydrazine and ammonia to absorb dissolved oxygen and raise the pH of the water to 10. The superheater presents a special problem since the loops are essentially nonventing. Steam will be used after flooding to flush the elements and displace the air.
2. When the boiler pressure has fallen to zero, drain the boiler.
3. Connect a steam hose from a 10 to 15 psig source to the drain connection on the intermediate superheater header. Leave the steam secured for now.

4. Close all valves to gauge glasses, remote water level indicators, and other similar auxiliary equipment. Close the main and auxiliary stop valves and bypasses as securely as possible.
5. Prepare enough hydrazine and ammonia solution to permit bringing the concentration in the filled boiler to 10 ppm of ammonia and 200 ppm ofhydrazine. These additives will be introduced through the chemical feed line.
6. Open the vents on the superheater inlet line and the auxiliary desuperheater outlet. Leave the steam drum vent open. Close the superheater vent and the boiler drain valves.
7. Start filling the boiler slowly using the auxiliary feed check. Begin adding the hydrazine and ammonia through the chemical feed line. *Note:* Water added should be warmer than ambient to avoid condensation on tube surfaces.
8. When water flows from the steam drum vent, close the vent valve. When water appears at the auxiliary desuperheater vent, close the vent valve. When water appears at the superheater inlet line vent, secure the feed check and chemical feed line. Leave the superheater inlet line vent open.
9. Partially open the steam supply to the superheater header. Mter steam has been on for 15 minutes, gradually open the control desuperheater control valve to aid in venting any trapped air.
10. When steam is detected at the superheater inlet line vent, secure the steam. Let the boiler sit for 30 minutes, then open the chemical feed line. When water flows from the vent, close the vent valve.
11. If both boilers are under wet layup, a static pressure can be maintained on th.e boiler using the deaerating feedwater heater. If only one boiler is under layup, static pressure can be maintained using compressed nitrogen. (See the short-term wet layup procedure for connection method.) Maintain 5 to 10 psig on the unit.
12. During the wet layup, the boiler must be protected from freezing. Iffreezing is possible, use another method. The concentration of ammonia and hydrazine must be checked periodically to be sure the solution strength is being maintained.

Dry layup with desiccant

1. Drain the boiler completely while still slightly warm.
2. Open the manholes in the steam drum and water drum, and handhole plates in the waterwall and superheater headers, checking for any remaining water. All circuits where water can accumulate should be blown out with compressed air.
3. Wipe dry all accessible parts of the drums and headers. Install new gaskets on manholes, handholes, and other openings to be blanked.
4. Distribute five-pound bags of silica gel throughout the drums and headers. Suspend the bags from drum internals in order to expose as much surface as possible. Approximately 10 pounds of silica gel are required for

each 10,000 lb/hr of boiler generating capacity. This should be sufficient to maintain the relative humidity below 30 percent.

5. Tightly secure all manholes, handholes, drains, vents, and other openings to the boiler water spaces. Close all air inlets to the furnace and install the stack cover.
6. Check a few bags of silica gel for saturation after two weeks. When the silica gel becomes saturated, dry it for reuse by baking at 300° to 350°F.

Dry layup with nitrogen gas

1. Drain and dry the boiler as described above for the "Dry Layup with Desiccant" method. Close all vents, drains, and other openings to the boiler water space.
2. Connect compressed nitrogen bottles to a vent at the top of the boiler, and vent air from the bottom. (See "Wet Layup: Short-term" for more information on connecting the nitrogen bottle.) *Note:* A standard nitrogen bottle will supply approximately 145 cubic feet of gas at 10 psig.
3. Once all nitrogen in detected at the bottom vents, secure the vents. Charge the boiler to 10 psig. Secure the nitrogen gas supply.
4. Check the gas pressure periodically. To compensate for leakage, makeup with additional nitrogen gas.

BLOWING TUBES

Sootblowers are installed to conveniently remove soot which has deposited on boiler generating tubes, superheater tubes, economizers, and air heaters. Soot deposits must be removed to maintain boiler efficiency and to avoid external corrosion of the boiler surfaces. If not removed promptly, the soot deposits can pack into a solid mass requiring tedious hand cle~ing. Regular use of properly maintained sootblowers is mandatory. Unless experience with a particular installation indicates otherwise, tubes sh'ould be blown twice daily when burning residual oil and once weekly when burning distillate oil. In addition, tubes should be blown in the following situations:

- after departure from port
- prior to entering port
- immediately before securing a boiler
- after lighting off
- after loss of fires
- after any period of heavy smoking

The following is the recommended procedure for blowing tubes:

1. Blow tubes at half boiler load or above. When blowing tubes at sea-speed, it is normally necessary to slow down somewhat to permit maintaining the boiler pressure under the increased steam demands of sootblowing.
2. Open the sootblower piping drain valve.
3. Crack the sootblower steam supply valve. Allow steam to blowout the drain valve until all condensate has been drained and lines are warm.
4. Increase the forced draft pressure. The increased gas velocity aids in carrying away soot and dirt dislodged by the sootblowers.
5. Close the sootblower piping drain valve.
6. Operate the sootblowers one at a time. Blow the economizer or air heater first, then proceed in the direction of gas flow-superheater, generating tubes, and finally repeating the economizer or air heater. Latch each blower after use, then proceed to the next.
7. Observe the smoke indicator (periscope) during each blower operation. If one operation of a blower does not produce a clear stack, repeat until a clear stack is obtained. Never allow steam to blow if the element is not rotating.
8. After sootblowing is complete, secure the sootblower steam supply and open the drain valve. Reduce the forced draft air pressure to restore minimum excess air. If the plant power/revolutions were reduced, restore to normal.
9. If the sootblowers are operated by air motors, refer to the manufacturer's technical manual for detailed operating instructions.

Inspections and Surveys

Boiler surveys are required at prescribed intervals by the flag authority and/or the classification society. The ship's crew will have the responsibility of preparing the boiler for the various inspections and tests that will be performed by the surveyor.

Inspections should be performed by the ship's crew whenever the opportunity presents itself. During normal boiler operation, the engineers should be alert to the proper functioning of the various components and to developing problems. Whenever a boiler is shut down, a fireside inspection should be performed. Whenever a drum or header is opened, a waterside inspection will permit an analysis of the effectiveness of the water treatment and control.

Prior to an inspection or survey, the boiler should be cleaned sufficiently to permit complete examination. Observe proper "lock outJtag out" procedures to prevent the entrance of steam, hot water, or exhaust gases from another operating boiler into the unit under inspection. An assistant should be posted outside the boiler to help in case of an emergency.

The following sections discuss the inspection of the fireside and waterside inspections of a water-tube propulsion boiler. Similar procedures would be followed in the inspection of a water-tube auxiliary or a waste-heat boiler. The inspection of fire-tube boilers will require some modification of these procedures due to their construction.

FIRESIDE INSPECTION

Inspect the fireside at every shutdown. Be sure the tube bank is clean and free of any accumulation of soot or slag. Pay particular attention to the tubes behind the superheater and be sure there is no heavy accumulation of soot on the water drum. Surfaces of the water drum exposed to combustion gases, including tube ligaments and areas along sides of the drum underneath the last row of tubing between drum and casing, are particularly troublesome areas. These are not readily cleaned by sootblowing and are particularly vulnerable to the effects of hard-packed soot, moisture, and corrosion. These areas should be carefully inspected at every opportunity and thoroughly cleaned at every water washing.

See that there is no heavy accumulation of slag or crusted ash on the superheater or on the screen tubes between the furnace and superheater. Ordinarily the screen and waterwall tubes can be cleaned as necessary by rattling or bumping the tubes with a hammer handle or by use of a wire brush. Excessive slagging of the superheater tubes will require removal by a scraper or other similar means. Inspect the waterwall tubes and front-row boiler tubes for any indication of cracking, blistering, or overheating. Any boiler tube or waterwall tube that may be found faulty will require plugging or replacement. If a horizontal superheater is fitted, inspect the supports. Severely corroded supports can permit tube sagging, preventing proper drainage.

Inspect the furnace refractories for spalling, slagging, flame impingement, or inadequate provision for expansion. It is important that any refractory covering over the waterwall headers be maintained in good condition. Particular attention must be given to the castable refractory around the fuel oil burner openings. Failure of refractory around the spool-piece on the inner casing that surrounds the burner venturi can cause a burnout of the spool-piece and/or the venturi end-piece. The refractory is to be repaired in accordance with the instructions in the manufacturer's technical manual.

WATERSIDE INSPECTION

Make an inspection of the waterside whenever a boiler is opened for any reason, to be certain that the methods of analysis and treatment of the feedwater and boiler water are satisfactory. Before sending anyone into the boiler for inspection and cleaning subsequent to boiling out, ventilate it thoroughly. Because of the possible presence of inflammable or noxious vapors or lack of oxygen, it is especially important to ventilate the boiler thoroughly. Mter a boiler has been under nitrogen layup, exercise extraor-

dinary caution before entering the drums. Open all manhole covers and ventilate the drums thoroughly with fans before entering.

Tag, close, and secure by locking or wiring all valves that might permit the accidental entrance of steam or water into the boiler. While crewmembers are engaged in working on or cleaning the interior of boilers, one person should be stationed outside and in communication with the maintenance crew to render assistance as necessary. No boilers leading to the same stack pipe as the steaming boilers should have work done upon them that involves opening the uptake doors or furnace doors, unless division plates are installed to provide an individual gas passage for each boiler to the top of the smoke pipe.

The use of naked lights in an open boiler is prohibited. Portable electric lights may be used, but the use of hand electric flashlights is preferred. The electrical leads of portable lights must be in good condition. The portable lighting fixture should be of the watertight type with a substantial glass globe around the light bulb and a rubber-insulated guard around the globe. As a safety precaution, the entire electric installation should be tested for grounds, and defects remedied before entering the vicinity. This test for grounds should be made from the switchboard outside the fouled spaces and repairs made with the circuit dead.

To make a thorough examination of the upper tube circuits, open the steam drum and remove the internals as shown in the boiler technical manual. Open the water drum to examine the lower tube circuits. If an internal desuperheater is installed, its removal will be necessary to perform a complete inspection. Check the appearance of the drum and tubes carefully, looking for the presence of corrosion, scale, and other deposits. Open the waterwall handholes nearest the ends of the waterwall headers and see that waterwall supply tubes and headers are free of sludge or other foreign material which may restrict the circulation.

Boiler and waterwall tubes must be kept free of scale deposits, oil, and corrosion. Waterwall tubes and the boiler screen tubes between the furnace and superheater which are exposed to the radiant heat of the furnace wall will overheat with only a very thin layer of scale deposit. Scale deposits are caused by failure to maintain correct chemical conditioning of the boiler water, particularly the phosphate levels. Corrosion in the boiler is usually due to low alkalinity of the boiler water, failure to vent the boiler when starting to raise pressure, or improper layup and care of the boiler when out of service. If pitting or localized corrosion is present, the cause is most likely oxygen in the feedwater.

If scale is found, it must be removed before returning the boiler to service. Corrosion or pitting damage will require repairs. These repairs may have to be performed under the supervision of a classification society surveyor. Mter completing inspection or any repair work in the boiler, be sure that all tools, bolts, and other items have been removed. Brush out all dirt, welding spatter and similar material. Wipe up any oil and wash the drums with warm fresh water using a high-pressure water hose.

SUPERHEATER INSPECTION

Inspect the fireside of the superheater at every cleaning period. Remove any accumulation of soot or slag. When the gas spaces are partially blocked, the gas velocities through the remaining open spaces increase, giving rise to localized high tube metal temperatures. Look at the full width ofthe superheater from top to bottom. Do not assume the entire superheater to be clean if the first area inspected is clean.

Look for warped or bu'tned elements. Warping of superheater elements indicates overheating due to the raising of boiler pressure too quickly by overfiring, a failure to maintain sufficient steam flow through the superheater while raising pressure or securing the boiler, or the presence of scale in the superheater tubes. Slight warping is not serious if steps are taken at once to determine and eliminate the causes. If scale is the suspected cause of element warpage, remove a sufficient number of hand hole plates on each pass to examine the various circuits. If scale is present in every circuit, the steam drum is to be opened and the steam separation devices, baffle plates, and dry box examined to be sure they are properly installed and not adrift.

Ifthere is no scale present in the first two passes, but there is scale present in varying degrees in the third to last passes, the leakage of boiler water into the control desuperheater is to be suspected. The desuperheater must be subjected to a hydrostatic test in order to determine the point of leakage.

Note

Initial leakage of boiler water into the control desuperheater will usually become apparent by a low superheated steam temperature reading. As the leakage increases, the temperature control valves will "hunt" continuously with little or no control of the superheated steam temperature at the superheater outlet. The effect of water being incorporated into the superheated steam flow is the same as that of a separate spray-type desuperheater. At the first sign of a low or fluctuating steam temperature, the problem must be investigated as soon as possible.

Scaling of superheater elements on the interior surface can also be caused by priming and carryover of water from the steam drum. This condition will also cause rapid fluctuation of the superheated steam temperature and erratic operation of the control valves. However, evidence of scaling will show in all passes of the superheater with the heaviest deposits in the first two passes.

ECONOMIZER INSPECTION

The flue gas side of the economizer should be inspected on a regular basis. The surfaces of the elements are to be inspected for corrosion, thinning, and accumulations of packed soot. Remove access doors in the stack duct and in the uptake of the boiler in order to conduct the inspection.

Operate the sootblowers through a full cycle just prior to shutting down for the fireside inspection. In this way, the effectiveness of the sootblowing can be evaluated. If during the inspection it is seen that the sootblowing is inadequate (unusually heavy soot accumulation), the sweep of the sootblower incorrectly set (the bottom rows of elements only partially clean), or the blowing pressure is too high (fins and elements eroded), then the sootblowers must be readjusted, as necessary. When packed soot is found, the economizer is to be washed down using warm fresh water. For this procedure the drains on the floor of the uptake must be opened to carry off the water. Water washing of the economizer should be done, when practical, just prior to closing up the furnace and lighting off. This will permit a thorough drying ofthe element surfaces and fins while minimizing rusting.

Economizer return bends are usually located in end-boxes isolated from the flue gases. Inspection of the return bend weldments can be performed via view ports and inspection lamps on the end-boxes or by removal of access doors. Distorted tube plates can permit gas leakage and soot accumulation, causing corrosion of the bends. Seepage of water from water washing or condensed steam from sootblowing can cause acid attack in the presence of soot.

The waterside of an economizer should be inspected annually for oxygen corrosion or pitting. For this inspection, handhole plates are removed from the inlet and outlet headers after the unit is drained. The tube metal is examined visually as far as is practical on the inlet and outlet run of each circuit. Ifthere is evidence of oxygen pitting, the feedwater system and deaerator tank are to be checked out for intrusion of air. Steps are to be taken immediately to correct any influx of oxygen.

The handhole plates are replaced after the gasket surfaces on the plate and header are thoroughly cleaned. Compressed asbestos gaskets are typically used to seal the plate to the header. Flexitallic-type gaskets, made of spiral wound stainless steel and asbestos, are not used on economizer handhole plates because they result in premature leakage.

Emergency Procedures

An emergency situation in a steam plant never follows a planned script. The procedures below are typical of those that would be followed in common situations that may be encountered. It is up to the operator to apply judgment and experience to each unique situation to protect the machinery and bring the plant back to normal operation as quickly as possible.

WATER OUT OF SIGHT-HIGH OR LOW

Follow this procedure whenever the water disappears from sight in the steam drum gauge glasses, either due to high or low water.

1. Secure the burners *immediately.* If operating at sea-speed, it will be necessary to slow down to maintain pressure on the other boiler.
2. Close the main and auxiliary feed checks.
3. Open the superheater vent.
4. Close the main and auxiliary steam stops.
5. If there is any question as to whether the water is high or low, blow down the gauge glasses to determine the condition. Close the steam and water cocks and open the drain. Blow down by opening and closing first the steam cock, then the water cock. Line the glass up again to be sure if high or low water exists.
6. Now that high or low water has been determined, immediately take the appropriate action outlined below.

HIGH WATER IN BOILER

The following actions should be taken after the boiler has been secured as described above in "Water Out of Sight."

1. Blow down the boiler, bringing the water level to approximately design level. Use the surface blow, if fitted.
2. Drain all superheater headers. Drain desuperheater outlets.
3. Relight the burners and cut the boiler back on-line following the normal procedure.

LOW WATER IN BOILER

The following actions should be taken after the boiler has been secured as described above in "Water Out of Sight."

1. Close the burner registers and secure the forced draft fan.
2. Proceed to secure the boiler as described above for normal operation except as noted below.
3. *Warning.* Do not attempt to add water until the boiler has cooled sufficiently. Damage may result if cool water comes in contact with overheated tubes, headers, or drums.
4. If inspection indicates no damage has occurred, boiler can be restarted normally.

LOSS OF FIRES

The following procedure should be followed if the fires are lost in one boiler. If fires are lost in both boilers, the same procedure should be followed for each boiler, but the potential effects on the overall plant are more significant. See chapter 3 (Steam Power Plants) for more information on overall plant operation.

1. Secure the fuel oil to all burners. *Note:* If a burner management system is installed, the fuel oil will have been automatically secured on detection of flame failure.
2. If operating at sea-speed, it will be necessary to slow down to maintain pressure on the other boiler.
3. Reduce the forced draft air pressure. Purge the furnace.
4. Determine and correct the cause of the loss of fires. If water is in the oil, switch settlers or change to high suction. If atomizing steam pressure was lost, do not attempt to relight the burners until the steam pressure is restored. *Note:* If problem cannot be corrected in a brief period, proceed with securing of the boiler as outlined above for normal operation.
5. If any unburned oil has accumulated, wipe up completely and purge furnace for at least 10 minutes before attempting to relight the burners.
6. Proceed with normal boiler light-off as outlined above.

TUBE FAILURE

This procedure should be used following a rupture of a tube or other failure causing significant loss of water from the boiler.

1. Secure the burners immediately. Slow down the main turbines to permit the remaining boiler to maintain steam pressure. Proceed with securing the boiler as modified by the steps below.
2. Ifthe failure resulted from low water in the boiler, *do not* add water to the boiler. Close the feed checks, feed stops, and the main and auxiliary steam stop valves. Open the superheater vent. If the failure was *NOT* caused by low water, attempt to maintain a normal water level until the boiler has cooled. Secure the steam stop valves. Open the superheater vent. *Note:* It is essential to maintain the water level in the operating boiler. Transfer water from potable or reserve feed tanks to maintain supply of makeup feed, if necessary. *Do not* attempt to maintain level in the boiler with the tube failure at the expense of starving the operating boiler.
3. Use the safety valve easing gear to open the safety valves and drop the boiler pressure as quickly as possible.
4. Keep the forced draft blower operating. Adjust air flow to carry escaping steam out the stack.
5. Do not blow down the boiler unless escaping steam is endangering personnel. If necessary, use the bottom blow and dump water overboard.
6. Mter the boiler pressure has been reduced, secure the forced draft fan and close the registers. Allow the boiler to cool slowly.

FAILURE OF FORCED DRAFr

This procedure should be followed in the event of failure of the forced draft fan or other loss of combustion air to the boiler.

1. Secure the burners immediately. Slow down the main turbines to permit the remaining boiler to maintain steam pressure. *Note:* If a burner management system is installed, the burners may have been automatically secured.
2. Restart the blower orppen crossover from operating forced draft fan. *Note:* If it is evident there will be a delay in restoring combustion air, open the superheater vent as soon as boiler pressure drops below the line pressure. Maintain boiler drum level using the auxiliary feed check.
3. Purge the furnace. Light off the burners one at a time. If the superheater vent was opened, close it when the boiler is back on-line.

LOSSOFFEEDWATERSUPPLY

The loss offeedwater supply to the boiler should be treated as an extreme emergency. Failure to deal properly with the casualty can result in serious damage to all operating boilers. It is essential to determine quickly what the problem is and ifit can be resolved immediately. If water levels cannot be maintained above minimum by slowing (or stopping) the main turbines, the boilers will have to be secured.

The first items to check are the feed pump discharge pressure and deaerating feedwater heater level. A slightly low feed pump pressure may merely be due to low setting on the pump governor. Low deaerating feedwater heater level may be due to a problem with the makeup feedwater supply, such as an empty makeup feed tank. If both the feed pump pressure and deaerating feedwater heater level are satisfactory, the automatic feedwater regulators are suspect.

In the event of complete failure of the feed pump while under way, proceed as follows:

1. Slow down the main propulsion turbines as much as possible to conserve water in the boilers.
2. Secure burners as necessary in line with reduction in steam production to prevent popping the safety valves.
3. Bring a standby feed pump on-line as soon as possible.
4. Ifthe standby feed pump is brought on-line before the water level is lost from sight, raise the boiler levels equally in both boilers.
5. As the drum levels are restored, the turbine speed can be slowly increased, the boiler firing rate increased, and the plant operation restored to normal.

If a standby feed pump cannot be brought on-line before the boiler drum levels drop from sight, proceed as follows:

1. Secure the fires in both boilers when the drum level approaches the bottom of the gauge glasses.
2. Close the main engine throttles to conserve the water that remains and the boiler pressure. Secure steam to all equipment and systems not necessary to put the backup feed pump on-line.
3. Open the superheater vent to protect the superheater. Residual heat in the boiler will result in steam production for a short period of time.
4. If the emergency diesel generator does not start automatically, start the generator manually and supply electrical power to the emergency circuits.
5. Take appropriate action to avoid damage to the main turbines and turbine-generators. This includes checking lube oil pumps, opening drains, and starting the main engine jacking gear after the engines coast to a stop.
6. Make up two atomizers with straight mechanical tips. Line up the diesel oil system. Install one atomizer in each boiler.
7. Line up the backup feed pump for startup.
8. Start the forced draft fans and purge the furnaces. Light off one burner in each boiler on diesel oil, following the procedures in "Lighting Off-Cold Plant." Be sure water still shows in the gauge glasses.
9. Once steam is being produced, reopen the auxiliary steam stop valve and start the backup feed pump. Close the superheater vents.
10. Restore the steam supply to the fuel oil heaters.
11. Restart the main fuel oil pumps. Recirculate fuel oil to bring the oil temperature up to operating temperature at the boiler front.
12. Shift over to firing on residual oil. Follow the procedures in "Lighting Off-Cold Plant."
13. Progressively bring the entire plant back to normal operation. Carefully monitor the feed system and boiler water levels as systems are brought back on-line.

REVIEW

1. Why have water-tube boilers replaced fire-tube boilers for main propulsion use?
2. What advantages does a bent-tube D-type boiler have versus a sectional header boiler? Are there any disadvantages?
3. What problem does a marine reheat boiler have that its shoreside counterpart does not? Describe one solution.
4. Describe a dual-pressure, forced-circulation waste-heat boiler installation.
5. Calculate the theoretical air/fuel ratio for methane (CH_4).
6. The stack gases of a boiler burning residual oil are 300°F and contain 2 percent O_2. Use figures 5-15 and 5-20 to determine the boiler efficiency.

7. Why is the turndown ratio of a fuel oil atomizer important?
8. Why are most marine superheaters located close to the furnace, behind only a row or two of screen tubes?
9. Why is superheat temperature control required on some marine boilers?
10. Why is extended surface (in the form of fins) used on many economizers? Why isn't it used on superheaters?
11. Why are air heaters rather than economizers almost always used when the plant has high-pressure feedwater heaters?
12. How do you check the air leakage in a rotary regenerative air heater?
13. What is blowdown'in a safety valve? How is it adjusted in a huddling chamber safety valve? In a nozzle reaction safety valve?
14. What is scavenging air in a sootblower?
15. What is the function ofthe drypipe in a steam drum?
16. Compare open-loop and closed-loop control.
17. What is offset in a proportional control system? How can offset be reduced?
18. Why is reset (integral control) used in many controllers?
19. Define the following combustion control system terms: series, parallel, positioning, metering.
20. What is steam flow anticipation in a combustion control system? Why is it used?
21. What is cross limiting in a combustion control system?
22. Why is a square root extractor used after many flow transmitters?
23. Describe the procedure for bumpless transfer when shifting a controller from "hand" to "auto." From "auto" to "hand."
24. What are the input(s) to a single-element, a two-element, and a three-element feedwater regulator?
25. What are swell and shrink as related to the operation of a feedwater regulator?
26. Why are ultraviolet detectors, not visible light or infrared detectors, used in flame scanners? What is a "flicker" detector?
27. What is a programmable logic controller (PLC)?
28. Why are phosphates added to boiler water?
29. Why is sodium sulfite added to boiler water?
30. Describe the relationship between pH and alkalinity.
31. Describe how pH and scale are controlled when using the coordinated phosphate water treatment system.
32. Outline the procedure for adding chemicals to a boiler.
33. What precautions would you take prior to lighting off a boiler?
34. When raising steam in a boiler, why is it important to fire a boiler initially at the lowest rate possible?
35. Which boiler layup procedure would you recommend for a 48-hour layup? Which for a 60-day layup?

36. What is the first step to take if the water level cannot be seen in gauge glass?
37. The burner management system suddenly trips all the burners in one boiler. What is the first thing you would do?

REFERENCES

Babcock & Wilcox. 1978. *Steam/Its Generation and Use,* 39th ed. New York: Babcock & Wilcox.

Combustion Engineering, Inc. 1981. *Combustion /Fossil Power Systems,* 3rd ed. Windsor, Conn.: Combustion Engineering, Inc.

Flanagan, G.T.H. 1990. *Marine Boilers,* 3rd ed. Oxford: Heineman Newnes.

Combustion Engineering, Inc. 1962. *Marine Boilers.* Windsor, Conn.: Combustion Engineering, Inc.

Institute of Marine Engineering. 1992. *The Running and Maintenance of Marine Machinery,* 6th ed., ed. J. Crowley. London: Institute of Marine Engineering.

Milton, J. H., and R. M. Leach. 1980. *Marine Steam Boilers,* 4th ed. London: Butterworths.

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Combustion-Fossil Power Systems, 3rd ed., 1981: Figure 5-78.
Marine Boilers, 1-62-4M, 1962: Figures 5-1, 5-2, 5-3, 5-4, 5-24, 5-32, 5-37, 5-39, and 5-41.
C-E Marine Steam Generating Equipment, PSG-7188: Figures 5-6, 5-9, 5-10, and 5-12.

CHAPTER 6

Marine Steam Turbines

EVERETT C. HUNT

HISTORY OF STEAM TURBINES

The principle of a turbine was understood and applied by the ancient Greeks. However, the commercial development of steam turbines did not begin until the late nineteenth century. Engineers will recognize several early leaders in the field of turbine development whose names have become a part of turbine nomenclature: De Laval of Sweden, who was granted a patent in 1883; Parsons of England; Rateau of France; and Curtis of the United States.

Steam turbine ship propulsion first gained acceptance with the dramatic 30-knot performance of Sir Charles Parsons's *Turbinia* at the Spithead Naval Review in 1897. Parsons's understanding of the potential of steam turbines for high-power ship propulsion led to the *Mauretania* in 1906, for which he furnished main and auxiliary turbines with a total power capability of 73,000 shp. The early Parsons propulsion turbine systems were direct drive, which limited the system efficiency since the propeller speed was too high and the turbine speed was too low for optimum performance.

In 1915 the *Pacific* was launched with the first steam turbine-gear propulsion plant furnished by the General Electric Company. This started a long period of worldwide development of steam turbines and gears for naval and merchant ship propulsion. Initially the lack of gear manufacturing technology caused some problems that made turbine-electric propulsion plants popular for a period of time. The impulse-type steam turbine which General Electric Company designed and manufactured experienced a difficult period before 1920 because of turbine rotor failures due to a lack of understanding of the complex vibratory modes of turbine buckets and disks. These problems were resolved by the work of Wilfred Campbell, who reported on his analysis and solution in two classic papers presented before the American Society of Mechanical Engineers. By the early 1930s, steam turbine and gear technology had developed to the point that the turbine-gear unit had become the standard for efficient ship propulsion.

Following the lead of electric utility and naval turbine development, propulsion turbine-gear systems for merchant ships were continuously improved for higher powers, better performance, greater reliability, lower maintenance, and easier operation from the 1930s to the 1980s. This culminated in reheat regenerative steam turbine propulsion systems with direct connected auxiliaries and inlet steam conditions of 1,450 psig, 950°F, with reheat to 950°F. These systems have all-purpose fuel rates in the range of .390 pounds fuel per shp-hr. The popular electric utility steam conditions of 2,400 psig, 1,050°F, and reheat to 1,050°F, with a fuel rate of .371 pounds per shp-hr were proposed for marine applications in the late 1970s, but no plants were manufactured.

The advent in the 1980s of the very large, slow-speed, long-stroke diesel propulsion system with highly competitive fuel rates (described in volume 2 of *Modern Marine Engineer's Manual*) has greatly reduced the selection of steam turbine propulsion for new construction. Steam turbine gears continue to have some advantage in new construction for special applications such as LNG carriers, coal-fired vessels, large ice-breaking ships, and a variety of warships.

STEAM TURBINE CLASSIFICATIONS

Condensing and Noncondensing

The condensing steam turbine, which may be extracting or nonextracting, is the most common aboard ship. Condensing main propulsion turbines normally have steam extracted for feedwater heating, feed pump turbine steam supply, freshwater evaporator steam supply, turbine-generator steam supply, or some combination of these, depending on the power plant cycle design. The inlet steam conditions are determined by economic considerations. The exhaust condition is influenced by both economics and the temperature of the seawater. Condensing steam turbines that are nonextracting are employed in shipboard systems for electric power generation, cargo pump drives, and refrigeration compressor drives.

Noncondensing turbines usually exhaust at pressures in the range of 10 to 20 psig. The inlet steam conditions are determined by economics. The outlet steam conditions are established by the need for auxiliary steam supply for heating combustion air, feedwater, accommodations, or numerous other purposes. The noncondensing turbine is used to drive fire, flushing, condensate, feed, feed booster pumps, and forced draft, ventilation, and exhaust fans. On most modem steamships, electric motor drives have

replaced noncondensing turbines with the exception of the main feed pump, which is typically driven by a noncondensing turbine.

Inlet Valves

Turbines may be classified by the type or arrangement of valves used to control the inlet steam. A single-inlet throttle valve to control the turbine steam flow is the most common for propulsion turbine. Such turbines are fitted with hand control valves that control flow to individual groups of first-stage nozzles so that the throttle valve can be fully open during at-sea conditions and the ship speed can be economically determined by selection of the number of first-stage nozzles in use. The throttle valve is used for control only during the maneuvering condition.

On some modern propulsion turbines, turbine-electric propulsion, and turbine-electric generators, a series of control valves that are hydraulically opened in a sequence to admit inlet steam to groups of first-stage nozzles are used to control the steam flow to the turbine. This arrangement avoids much of the part load throttling loss inherent with a single inlet valve.

Bypass Valves

Bypass valves in various arrangements are employed to admit high-pressure inlet steam to downstream stages of a turbine when the turbine is designed for a high-overload condition such as flank speed of a warship or a defense capability speed of a superliner like the *United States,* at one time the fastest passenger vessel in the world. Since the maximum flow of a turbine is determined by the first-stage nozzle area, a significantly higher flow and therefore higher power can be obtained when steam is admitted to downstream stages. This higher load is achieved at a loss in efficiency since the bypassed stages contribute little to the power production.

Impulse and Reaction

An impulse turbine stage is defined by a nozzle in which a portion of the energy in steam is converted to velocity and moving buckets on which the high-velocity steam leaving the nozzle impinges to cause the movement of the buckets. Theoretically, there is a pressure drop as the steam moves through the nozzle, but no pressure drop across the bucket.

A reaction turbine has a pressure drop and a steam velocity increase across both the stationary blades (nozzle) and the moving blades. The moving blade receives both an impulse force from the steam entering and a reaction force from the steam leaving the blade.

Reheat and Nonreheat

A reheat turbine arrangement exhausts steam from a high-pressure turbine to a heat exchanger, called a reheater, located in the steam generator where energy is added to the steam before entering a reheat turbine. A nonreheat turbine does not include the steam reheat feature. In a properly designed system, reheat turbines have better all-purpose fuel rates.

Tandem Compound and Cross Compound

Compounding occurs when the exhaust of a high-pressure turbine enters a lower-pressure turbine. If the turbines are coupled on a single shaft, they are tandem compounded. If the turbines are not coupled on a single shaft so they rotate at the same speed, they are cross compounded. In a modern propulsion system, a high-pressure turbine and a reheat turbine might be tandem compounded and,then cross compounded to the low-pressure turbine. The most common marine arrangement is a nonreheat, high-pressure turbine cross compounded to a low-pressure turbine.

STEAM TURBINE PRINCIPLES

Energy conversion in a steam turbine occurs in two steps: in the nozzles or stationary blades where thermal energy is converted to velocity head and in the moving buckets or blades where the velocity head is converted to mechanical work. Impulse turbines usually develop some reaction force, especially in the low-pressure stages of a multistage turbine. When reaction represents 50 percent of the force on the moving buckets, the bucket is called a blade and the turbine is designated a reaction turbine.

Nozzles

A nozzle is a passageway designed to increase steam velocity during the passage of steam. A nozzle may be formed by a drilled and reamed hole in a steel block or by the space between two adjacent airfoils which have been shaped to provide the passage flow-area shape ..

To understand the performance of a nozzle, we must refer to the first and second laws of thermodynamics and to the flow continuity relationship, which were introduced in chapter 1.

The first law may be stated as follows:

$$q + h_2 + \frac{V_2^{\,2}}{2g_cJ} + \frac{Z_2g}{g_cJ} = h_1 + \frac{V_1^{\,2}}{2g_cJ} + \frac{Z_1g}{g_cJ} + u$$

where

q = heat transfer, Btu/lbm
h_1, h_2 = inlet and outlet enthalpy, Btu/lbm
V_1, V_2 = inlet and outlet velocity, ft/sec
g_c = dimensional constant, lbm-ft/lbf-sec^2
J = mechanical equivalent of heat, 778 ft-lbf/Btu

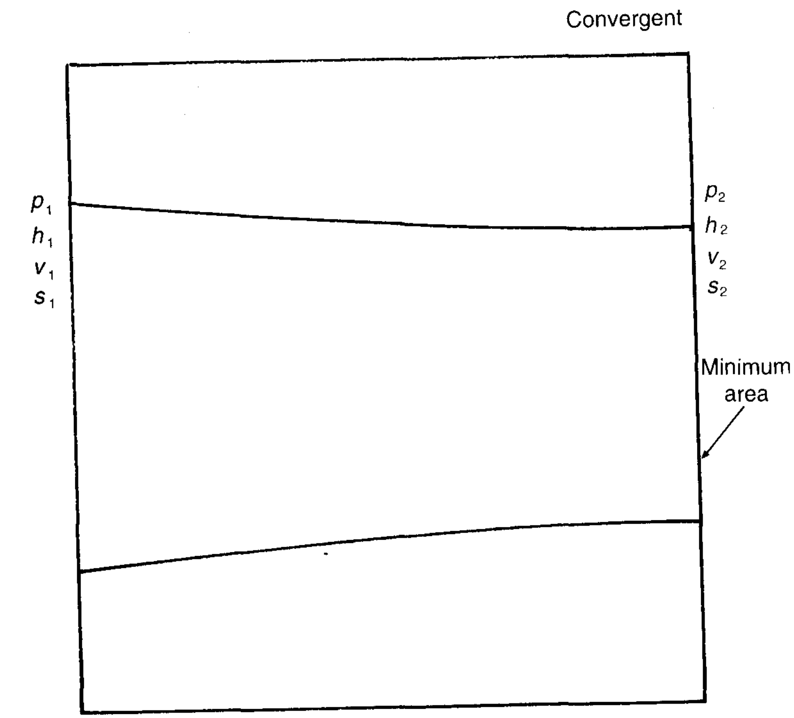


Figure 6-1. Convergent steam nozzle

Z_1, Z_2 = inlet and outlet distance above datum, ft.
g = acceleration of gravity, ft/sec^2
W = work
u = internal energy, Btu/lbm

For the nozzle shown in figure 6-1,

$$q = 0,\ W = 0,\ Z_1 = Z_2,\ V_1 = 0$$

which permits us to rewrite the first law for an ideal or isentropic nozzle as follows:

$$V_2^{\,2} = 2g_cJ\,(h_1 - h_2)$$
$$V = 223.8\sqrt{h_1 - h_2}\ \text{ft/sec}$$

Since there is some friction loss in the nozzle, the actual nozzle will have an efficiency

$$\eta_N = \frac{h_1 - h_2}{h_1 - h_2'}$$

and

$$V_2' = 223.8\sqrt{h_1 - h_2'} \text{ ft/sec}$$

For an isentropic expansion, the second law requires $S_1 = S_2$ and $S_1 \neq S_2'$ as shown in figure 6-2.

For a given mass flow, the continuity equation permits us to calculate the flow area of the nozzle if we know the ideal or actual velocity and use

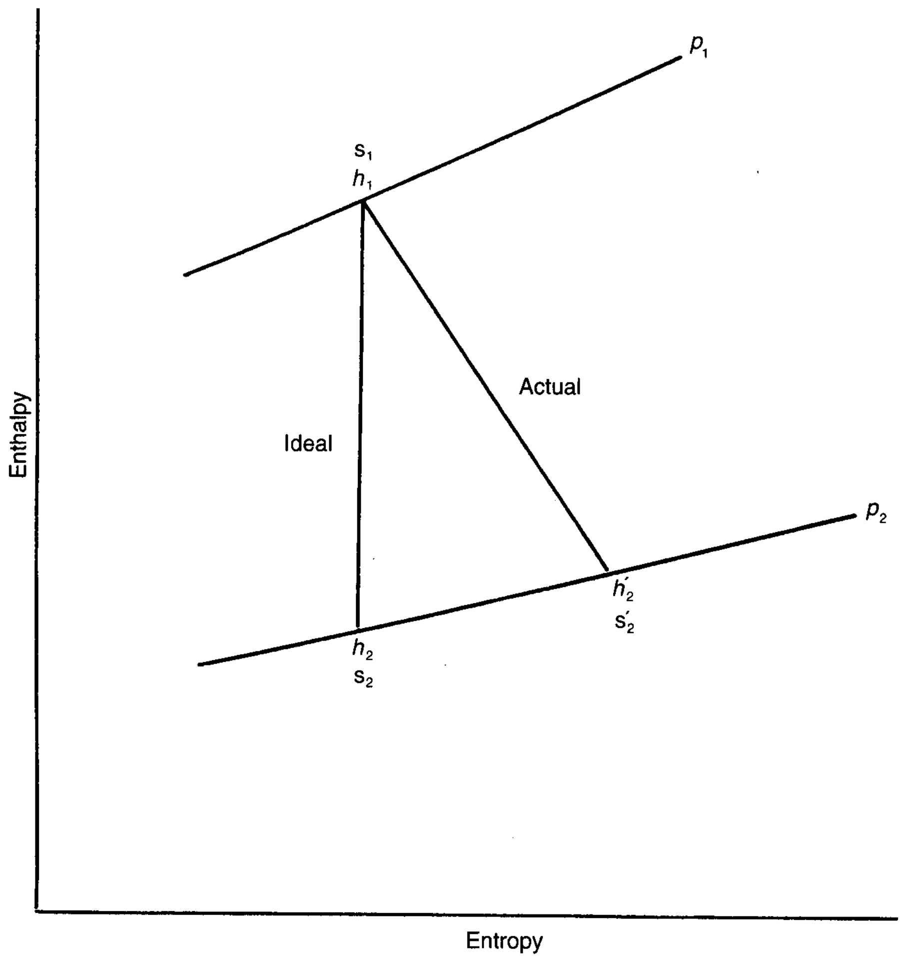


Figure 6-2. Isentropic and actual expansion in a nozzle

the second law and a Mollier chart to determine the steam conditions at the exit or any point between the inlet and exit:

$$\dot{m} = \frac{A_i V_i}{v_i}$$

where

$\dot{m}$ = mass flow rate, lb /sec
A_i = flow area, in
V_i = velocity, ft /sec
v_i = specific volume, ft^3/lb

EXAMPLE 6-1: For converging nozzles expanding 12 lb/sec of steam from 375 psia, 350°F to 225 psia, what nozzle area is required for an isentropic expansion and what is the velocity?

From a steam table or a Mollier chart, found in chapter 1,

p_1 = 375 psia	p_2 = 225 psia
t_1 = 640°F	t_2 = 525°F
h_1 = 1,332 Btu/lbm	h_2 = 1,279 Btu/lbm
s_1 = 1.62 Btu/ F/lbm	s_2 = 1.62 Btu/F/lbm
v_1 = 1.6575 ft^3/lbm	v_2 = 2.495 ft^3/lbm

$$V = 223.8\sqrt{1{,}332 - 1{,}279}$$
$$V = 1{,}629.28 \text{ ft/sec}$$

Using the continuity relationship, the nozzle discharge area may be calculated as follows:

$$A_2 = \frac{\dot{m} \quad v_2}{V_2} = \frac{(12)(2.495)}{1{,}629} = .01838 \text{ ft}^2 = 2.65 \text{ in}^2$$

If this is not an ideal nozzle and the nozzle efficiency is .9, then the exit velocity would be

$$\begin{aligned} V_2' &= 223.8\sqrt{\eta_N(h_1 - h_2)} \\ &= 223.8\sqrt{(.9)(53)} \\ &= 1{,}545.7 \text{ ft/sec} \end{aligned}$$

and the new state line end point would be

$$p'_2 = 225 \text{ psia}$$
$$t'_2 = 722°\text{F}$$
$$h'_2 = 1{,}284.3 \text{ Btu/lbm}$$
$$v'_2 = 3.06 \text{ ft}^3/\text{lbm}$$

to provide a nozzle area of

$$A' = \frac{(12)(3.06)}{1{,}545.7} = .0237 \text{ ft}^2 = 3.42 \text{ in}^2$$

For this nonideal nozzle, the losses due to friction, shock, and eddying increase the energy level (enthalpy) of the exit steam above the isentropic expansion as illustrated in figure 6-2.

The maximum flow that can be passed through a nozzle is determined by the sonic velocity at the minimum flow area, i.e., at the outlet of a convergent nozzle or the throat of a convergent-divergent nozzle, illustrated in figure 6-3. The convergent nozzle maximum velocity at the exit is limited to the speed of sound in the fluid. To achieve supersonic velocity, a divergent section must be added to the nozzle. The maximum flow and sonic velocity at the minimum area will occur in either nozzle when the ratio of inlet pressure to minimum flow area pressure is .545 for superheated steam and .577 for saturated steam. This ratio is called the critical pressure ratio. At lower pressure ratios, choked flow occurs and there is no increase in mass

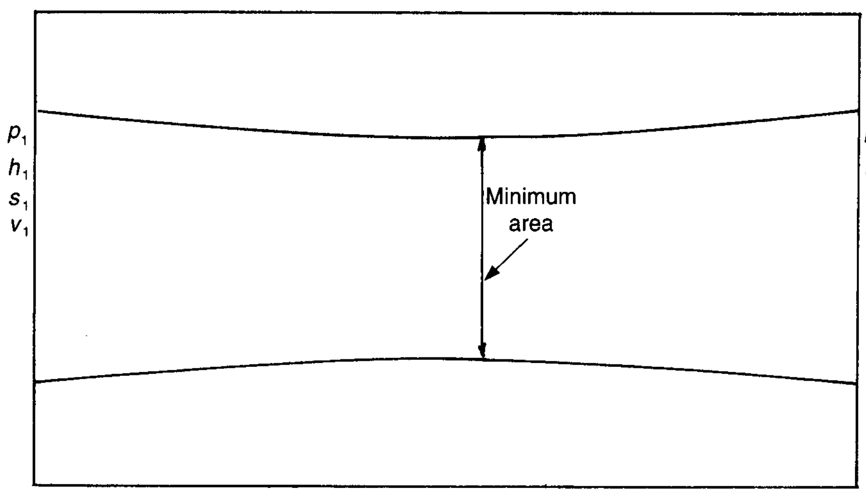


Figure 6-3. Convergent-divergent steam nozzle

flow even as the velocity of the convergent-divergent nozzle increases for lower-pressure ratios.

EXAMPLE 6-2: For a series of ideal round nozzles designed to pass 1 lbm/sec from a pressure of 100 psia saturated at decreasing pressure ratios, calculate the velocity, specific volume, minimum flow area, and nozzle diameter.

Pressure ratio	h_1–h_2 *Btu/lb*	*Velocity ft/sec*	*Specific Vol. ft³/lbm*	*Flow area in²*	*Diameter in.*
1.0	0	0	4.44	inf.	inf.
.9	9.5	689	4.87	1.018	1.138
.8	17.	922	5.46	.852	1.042
.7	27.	1,162	6.06	.751	.976
.6	40.	1,414	6.95	.708	.949
.577	44.	1,485	7.23	.701	.946
.5	54.	1,645	8.15	.714	.953
.4	69.	1,860	9.93	.769	.989
.3	90.	2,122	12.77	.865	1.050
.2	120.	2,450	18.28	1.074	1.168

Figure 6-4 is a plot of the velocity, specific volume, minimum flow area, and minimum nozzle diameter for the data above. Note that initially as the velocity increases rapidly for decreasing pressure ratios, the specific volume increases slowly. At the critical pressure ratio, the specific volume increases rapidly for further pressure ratio reductions while the velocity has a decreasing slope. The minimum point in the nozzle area indicates the critical pressure ratio below which choked flow exists.

EXAMPLE 6-3: Design a nozzle to expand 5,000 lbm of steam per hour from an initial pressure of 190 psia and 1 percent moisture to a final pressure of 16 psia. The nozzle efficiency is 87 percent.

$$\dot{m} = \frac{5{,}000}{(60)(60)} = 1.39 \text{ lbm/sec}$$
$$p_{\text{throat}} = (.577)(190) = 109.6 \text{ psia}$$

Use a Mollier chart and a steam table to determine quality, enthalpy, entropy, and specific volume for an isentropic expansion. From the Mollier chart, s = 1.54 Btu/deg/lbm.

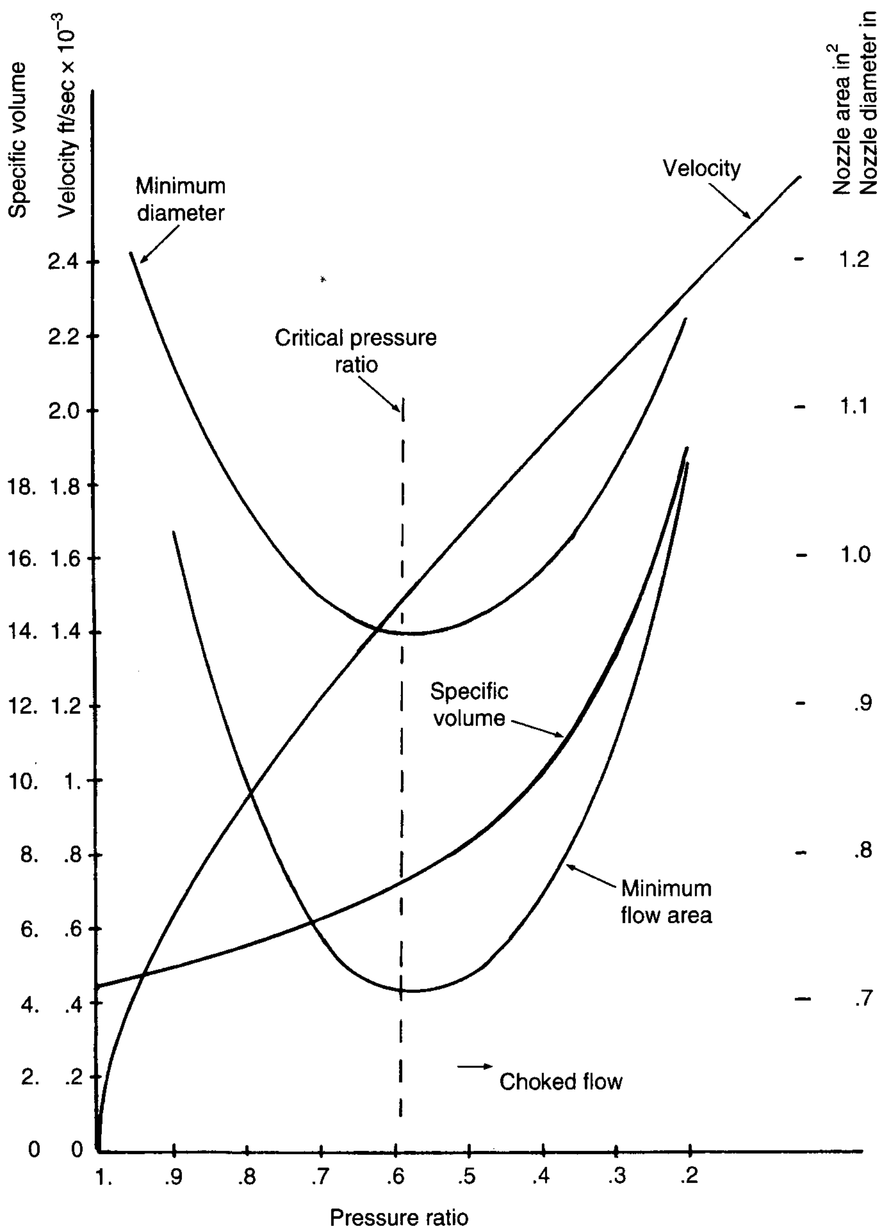


Figure 6-4. Nozzle characteristics

	p	x	h	v	h'	A'
entrance	190	.990	1,190	2.38	1,190	
throat	109.6	.949	1,145	3.86	1,053	.54
exit	16	.854	1,010	21.14	1,033.4	1.55

$$V_{\text{throat}} = 223.8\sqrt{1{,}190 - 1{,}145}$$
$$= 1{,}501.3 \text{ ft/sec}$$
$$V_2 = 223.8\sqrt{1{,}190 - 1{,}010}$$
$$= 3{,}002.59 \text{ ft/sec}$$
$$\eta_N = \left(\frac{V_2'}{V_2}\right)^2 = .87$$
$$V_2' = (3{,}002.59)\sqrt{.87}$$
$$V_2' = 2{,}800.63 \text{ ft/sec}$$
$$2{,}800.63 = 223.8\sqrt{1{,}190 - h_2'}$$
$$h_2' = 1{,}190 - 156.6$$
$$= 1{,}033.4 \text{ Btu/lbm}$$

From the Mollier chart

$$s_2' = 1.574$$
$$x_2' = .878$$
$$A_2' = \frac{\dot{m}v_2'}{V_2} = \frac{(1.39)(21.73)(144)}{2{,}800.63} = 1.55 \text{ sq. in.}$$
$$A_{\text{throat}}' = \frac{(1.39)(3.78)(144)}{1{,}400.3} = .54 \text{ sq. in.}$$

Impulse Stage

Figure 6-5 is a diagram of an impulse turbine stage consisting of a nozzle and moving buckets. The absolute velocities of the buckets and the steam entrance and exit are shown on the diagram.

Figure 6-6 is a velocity vector diagram for the steam and buckets of the impulse stage. The change in velocity of the steam in the bucket is equal to $w_1 - w_2$. The useful change in velocity is in the direction of bucket motion. These tangential velocities are components of the relative velocities, w_u, w_a. The change in velocity in the tangential direction is

$$w_{1u} - (-w_{2u})$$

or

$$w_{1u} + w_{2u} \text{ ft/sec}$$

The sum of these two vector components is called the velocity of whirl, v_w.

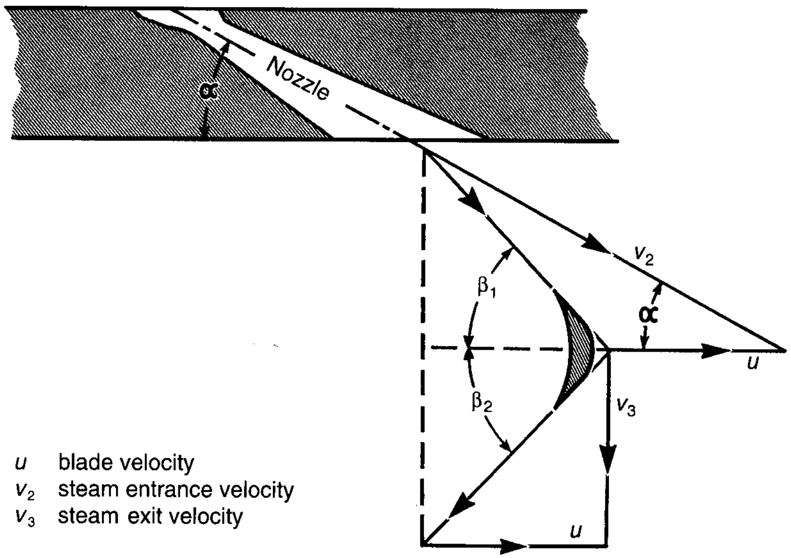


Figure 6-5. Turbine stage

According to Newton's second law

$$F = ma$$

$$F = \frac{W}{g_c}\frac{dv}{dt} \quad \text{lbf/sec}$$

and work is a force moving through a distance

$$\text{Work} = \frac{W}{g_c}\frac{dv_w u}{dt} \quad \text{dt}$$

$$= \frac{W\, v_w\, u}{g_c} \quad \text{ft-lbf/sec}$$

$$\text{hp} = \frac{W\, v_w\, u}{g_c\, 550}$$

The efficiency of the bucket is the ratio of the output energy to the energy available in the nozzle exit jet $\frac{Wv^2}{2g}$.

$$\eta_b = \frac{\frac{W}{g_c} v_w u}{\frac{W v_2^{\,2}}{2g_c}} = \frac{2 v_w u}{v_2^{\,2}}$$

In figure 6-6, we can see that

$$v_w = 2(v_2 \cos\alpha - u)$$

therefore the bucket efficiency is

$$\eta_b = \frac{2[2(v_2\cos\alpha - u)]u}{v_2^{\,2}}$$

If $\psi = u/v_2$, then

$$\eta_b = \frac{4uv_2\cos\alpha - 4u^2}{v_2^{\,2}}$$

$$\eta_b = 4\psi(\cos\alpha - 4)$$

$$\frac{d\eta_b}{d\psi} = 4\cos\alpha - 8\psi = 0$$

$$\psi = .5\cos\alpha$$

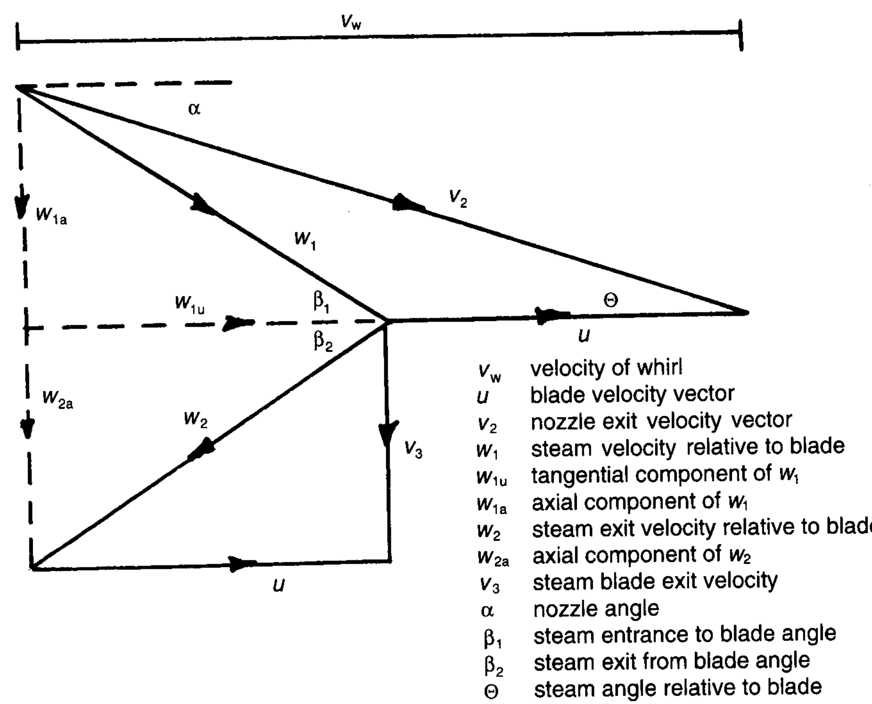


Figure 6-6. Velocity vector diagram for ideal impulse stage

then

$$\begin{aligned} \eta_{b\max} &= 4(.5\cos\alpha)(\cos\alpha - .5\cos\alpha) \\ &= (\cos\alpha)^2 \end{aligned}$$

If we select a constant value of v_2 and vary u so that the ratio $\psi = u/v_2$ increases from 0 to 1, the maximum value for the bucket efficiency occurs at $u/v_2 = .5$ as shown below:

$$\begin{aligned} \alpha &= 12 \text{ degrees} \\ v_2 &= 2{,}000 \text{ ft/sec} \\ \cos\alpha &= .978 \end{aligned}$$

	0	.2	.4	.5	.6	.8	1.0
u	0	400	800	1,000	1,200	1,600	2,000
	0	.6224	.9250	.9560	.9070	.550	0

EXAMPLE 6-4: An impulse turbine driving a centrifugal pump is rotating at 12,500 rpm. Steam flows from the nozzle at 2,900 ft/sec at an angle of 14 degrees. If the flow is 2 lb/sec and the velocity ratio is .5, what is the horsepower delivered to the buckets, the bucket efficiency, rotor diameter, and the inlet bucket angle, assuming no loss in the bucket?

$$\begin{aligned} \alpha &= 14 \text{ degrees} \\ \psi &= .5 \\ \eta_b &= .942 \\ v_2 &= 2{,}900 \text{ ft/sec} \\ u &= (.5)(2{,}900)(.942) = 1{,}366 \text{ ft/sec} \\ v_w &= (2{,}900)(.942) = 2{,}732 \text{ ft/sec} \\ \text{Power} &= \frac{(2)(2{,}732)(1{,}366)}{(32.2)(550)} = 421 \text{ hp} \\ \eta_b &= \frac{(2)(2{,}732)(1{,}366)}{(2{,}900)^2} = .887 \\ u &= \frac{Dn}{60} = \frac{D(12{,}500)}{60} \\ D &= \frac{(1{,}366)(60)}{\pi(12{,}500)} = 2.09 \text{ ft} = 25.08 \text{ in} \end{aligned}$$

Inlet bucket angle using the Law of Cosine

$$\begin{aligned} w^2 &= u^2 + v_2^2 - 2uv_2\cos\alpha = 1{,}609 \\ \sin\beta_1 &= \frac{702}{1{,}609} \\ \beta_1 &= 25.9 \text{ degrees} \end{aligned}$$

Stage Efficiency

Stage efficiency is a combination of the nozzle, bucket or blade, and mechanical efficiencies. It is expressed as follows:

118 = 11N 11B 11M

The nozzle and bucket efficiencies are discussed above. The mechanical efficiency is a result of bucket windage losses due to friction and fanning by the rotating parts, steam leakage through seals at the top of buckets and seals between the nozzle diaphragms and the turbine rotor, and losses due to entrained moisture striking the backs of blades.

Stage efficiency decreases over time for many reasons. Labyrinth packing clearance increases due to mechanical wear, permitting increased leakage. Buckets become rough because of erosion caused by water and solid particles such as boiler compound carried over from the boiler. Nozzles become rough due to erosion. Spill strips which control flow over the top of buckets wear and permit increased leakage. The serious decreases in stage efficiency occur when the radial sealing devices are damaged by operation with a bowed turbine rotor.

Pressure Compounding

Pressure compounding is the arrangement of a group of impulse stages in series. This is the arrangement used for all modern propulsion and electric generation steam turbines. The series of impulse stages are designed so the enthalpy drop across each ofthe nozzles will result in the optimum velocity ratio.

EXAMPLE 6-5: Two pounds of steam per second at 240 psi and .99 quality are supplied to an ideal, i.e., isentropic expansion, six-stage pressure-compounded turbine, and exhausted at 20 psia. The turbine operates at 4,000 rpm, the buckets have the same mean diameter, and the nozzle angle is 14 degrees for each stage.

Determine the power, mean bucket diameter, and the pressure in each stage.

$$s = 1.52 \text{ Btu/F/lbm}$$
$$\text{Available energy} = 1{,}192 - 1{,}011 = 181 \text{ Btu/lbm}$$
$$\text{Energy per stage} = 181/6 = 30.16 \text{ Btu/lbm}$$
$$v_2 = 223.8\sqrt{30.16} = 1{,}229.2 \text{ ft/sec}$$
$$v_w = 1{,}229.2 \cos 14 = 1{,}192.7 \text{ ft/sec}$$
$$u = (.5)(1{,}229.2)(.97) = 596.2 \text{ ft/sec}$$
$$\text{Power per stage} = \frac{(2)(1{,}192.7)(596.2)}{(32.2)(550)} = 80.3 \text{ hp}$$
$$\text{Total power} = (6)(80.3) = 281.9 \text{ hp}$$
$$\text{Mean diameter} = \frac{(60)(596.2)}{(3.1416)(4{,}000)} = 2.85 \text{ ft} = 34.2 \text{ in}$$

Use a Mollier chart to prepare the following table:

Stage	*Pressure psia*	*Enthalpy Btu / lb*
0	240	1,192
1	165	1,161.84
2	113	1,131.68
3	75	1,101.52
4	50	1,071.36
5	32.5	1,041.20
6	20	1,011

Velocity **Compounding**

A velocity compounded stage consists of a nozzle and two or three rows of moving buckets which are separated by fixed blades that redirect the steam into the following row of moving buckets. The fixed blades do not function as a nozzle. Therefore, the nozzle expands the steam, increasing the velocity. The velocity decreases in each of the rows of moving buckets. Analysis of the stage work would show that the optimum velocity ratio for a two-row stage is about .25 and .166 for a three-row stage, compared to .5 for a simple impulse stage. Since the two-row velocity compounded stage, called a Curtis stage, has an optimum velocity ratio equal to one-half that of the simple impulse stage, it can utilize a nozzle exit velocity, v_2, twice the magnitude of the single stage velocity for the same bucket velocity. The whirl velocity for the first row is six times the bucket velocity and the whirl for the second row is two times the bucket velocity which may be verified by drawing a velocity vector diagram to scale.

Curtis stages are commonly used in auxiliary turbines such as boiler feed pumps and the astern turbine of a propulsion turbine.

EXAMPLE 6-6: Steam is flowing at 3,000 ftlsec at a nozzle angle of 14 degrees into a Curtis stage with a mean bucket diameter of 3.6 ft. Assuming an ideal stage, calculate the force on each row of the moving buckets and the power developed. What is the turbine rpm?

$$u = (.25)(3{,}000)(\cos 14) = 727.7 \text{ ft/sec}$$
$$F_1 = \frac{2.0}{32.2}(6)(727.7) = 271 \text{ lb}$$
$$F_2 = \frac{2.0}{32.2}(2)(727.7) = 90 \text{ lb}$$
$$F_t = 271 + 90 = 361 \text{ lb}$$
$$\text{total power} = \frac{(361)(727.7)}{550} = 477.6 \text{ hp}$$
$$n = \frac{(60)(727.7)}{(3.6)\pi} = 3{,}861 \text{ rpm}$$

Reaction Stage

In a reaction stage, steam expands in both the stationary and moving blading. Blading is a term used for reaction stages compared to the nozzle and bucket terms used for impulse turbines. The steam entering the moving blading provides an impulse force on the blades. The steam exiting the moving blading provides a reaction force on the blading. In a typical reaction stage, the total force acting on the moving blades is 50 percent impulse and 50 percent reaction. It should be noted that in large multistage impulse turbines, the final few stages are designed with a significant amount of reaction force.

A reaction stage has an optimum efficiency when the velocity ratio, u/v, equals the cos a. The impulse stage with an optimum velocity ratio equal to one-half the optimum velocity ratio of the reaction stage has the same theoretical efficiency as the reaction stage, i.e., $\cos^2$a.

The fixed and moving blades of a reaction stage have equal discharge angles. One-half of the isentropic energy conversion occurs in the fixed blading, the other half in the moving blades or moving nozzles. For equal blade velocity, u, the reaction stage isentropic heat drop is one-half that of the impulse stage. Therefore, for given inlet and exhaust steam conditions, a reaction propulsion turbine would have twice the number of stages as a pressure compounded impulse turbine. This characteristic suggests higher manufacturing costs and turbine weight which may be a partial explanation of the fact that reaction turbines are uncommon for ship propulsion.

Figure 6-7 is a velocity diagram for a reaction stage. The relative velocity vector, w_2, leaving the moving blade is two-thirds the result of expansion of steam in the moving blade and one-third the result of the entering relative velocity, w_1.

The pressure drop across a row of moving blades results in an axial thrust on the turbine rotor. In a multistage reaction turbine, the total thrust is the sum of the pressure drops times the annulus areas of each

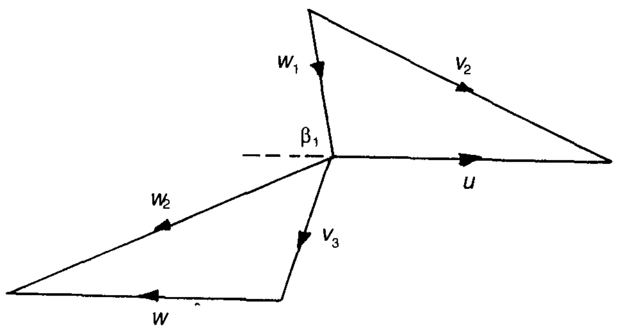


Figure 6-7. Velocity vector diagram for ideal reaction stage

stage. This large axial thrust requires a dummy piston on the rotor to apply a balancing force.

EXAMPLE 6-7: Three lbm of steam per second is passing through a four-stage reaction turbine. The nozzle efficiency is 90 percent. The blade velocity is 475 ft/sec and equal to 75 percent of the inlet velocity. The exit angle for all blading is 14 degrees. Calculate the horsepower.

$$475 = .75\,v_i$$

$$v_i = 633 \text{ ft/sec}$$

$$\eta_N = \left(\frac{v_i'}{v_i}\right)^2$$

$$.9 = \left(\frac{v_i'}{633}\right)^2$$

$$v_i' = 600.5 \text{ ft/sec}$$

Draw a scaled velocity diagram to determine the whirl velocity

$$v_w = 650 \text{ ft/sec}$$

$$\text{Work} = (4 \text{ stages})(\text{force/stage})(\text{blade velocity})$$

$$\text{Power} = \text{work}/550$$

$$\text{Total power} = \frac{(4)(3)(650)(475)}{(32.2)(550)} = 209.2 \text{ hp}$$

STEAM TURBINE PERFORMANCE

Design engineers determine steam turbine efficiency by evaluating the individual turbine stage efficiencies and combining them to obtain the over-

all efficiency for a multistage propulsion or auxiliary turbine. Shipbuilders and operators use manufacturers data to determine the expected turbine performance for a shipboard application.

Ship Propulsion Turbines

Richardson (see references at the end of this chapter) has documented a reliable method for determining the performance of a properly designed main propulsion impulse turbine given a maximum rating, steam conditions, and the last stage annulus area. The method is based on past design experience and considers the following factors: expansion line efficiency, exhaust loss of the last stage, steam packing losses to atmosphere, mechanicallosses, and the astern turbine windage loss.

The expansion line represents the internal efficiency of a group of stages such as a high-pressure turbine, a reheat turbine, or a low-pressure turbine. The overall expansion line efficiency, when plotted against volume flow, has a hyperbolic shape. The hyperbolic curve is modified to account for pressure ratio which changes internal efficiency by changing the average volume flow and operation in the moisture region, which decreases stage efficiency.

Efficiency ofthe last stage is about 86 percent. The pressure drop across the exhaust hood and the energy in the axial component of the steam leaving the last stage bucket make up the exhaust losses. These are charged to the last stage as exhaust loss.

End packing losses are the leakage of steam from the shaft sealing glands. They are usually provided by the manufacturer as a steam flow and enthalpy for incorporation in a heat balance calculation. Mechanical losses ofthe turbine and gear of a propulsion unit are charged to the turbine performance. The principal mechanical losses are bearing losses. The astern turbine windage loss is the final significant factor in determining the propulsion performance.

The most useful way to present the performance of a propulsion turbine is the overall plant heat balance which results in an all-purpose fuel rate for different operating modes. (This is covered in chapter 3.) The performance of an individual auxiliary or main propulsion turbine may be expressed as an internal efficiency in percent, utilizing the factors described above, or as a nonextracting operation steam rate. Steam rate is expressed in pounds of steam per unit output, i.e., Ibs/shp-hr or Ibs/kW-hr.

Propulsion steam turbine power is controlled with one of two methods: by throttling the flow to the first-stage nozzles, which controls the pressure ahead ofthe nozzles; or by controlling the number of first-stage nozzles in use, which controls the flow area at constant pressure. The latter method is also used to control the power developed by the shipboard turbine-generator sets.

Figure 6-8 is an *h-s* diagram which illustrates power control by throttling, a constant enthalpy process. Throttling control of propulsion turbines was most common until the late 1970s, when higher powers made the control ofthe number of first-stage nozzles economically attractive. Throttling control is inefficient when used for part-load operation since the throttling process increases the amount of unavailable energy, i.e., entropy. This is not a problem for commercial vessels designed for steady full-load operation. The performance of throttled propulsion turbines may be represented by a Willans line shown in figure 6-9 along with a steam rate curve for the same turbine. The Willans line is a linear function of steam flow, lbs/hour, and-the power output, horsepower or kilowatts.

Control of power by controlling the number of nozzles in use for the first stage is more efficient since the throttling is avoided for the first stage, which provides an increasing percentage of the total power as the flow through the turbine is decreased. Illustrations of turbine control arrangements are included later in this chapter.

TURBINE BUCKET CONSIDERATIONS

Perhaps the greatest challenge to the turbine designer is the design ofthe impulse turbine bucket, which is a compromise among the competing needs for high aerodynamic performance, strength to resist high stresses in an environment ofhigh temperature and erosive water droplets, and the ability to avoid or endure an infinite number of destructive vibratory modes and frequencies. The vibratory mode problems are aggravated in variable speed machinery such as in a main propulsion turbine. The ship's engineer officer has a role to play in maintaining a properly designed turbine in a condition to avoid problems by proper operation and planned maintenance during the machinery life.

Bucket Stress

The first consideration is to design a bucket so that it resists the centrifugal stress imposed at full speed rotation with a suitable factor of safety. In the high-temperature stages, the selected material must have properties that resist creep over time to avoid failures late in the life of the turbine. Even short periods of extreme overspeeding can result in broken tenons, bent bucket covers, and distortion of dovetails. An overspeeded turbine should be inspected promptly before resuming full-load operation.

Bucket Dovetails

A dovetail is the shape ofthe cutout located at the bottom of a bucket that is used to connect the bucket to the turbine wheel or disk. There are numerous dovetail designs but the most common used for marine turbines are the

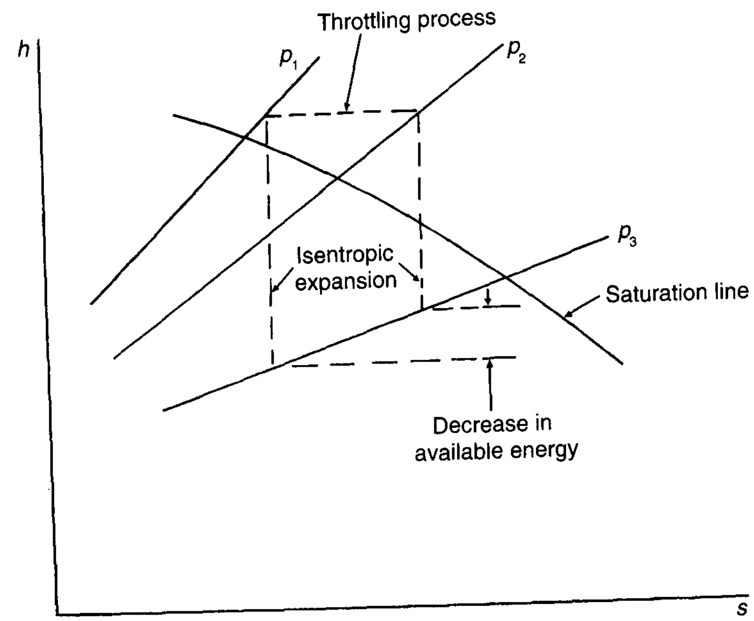


Figure 6-8. Throttling control of steam turbine

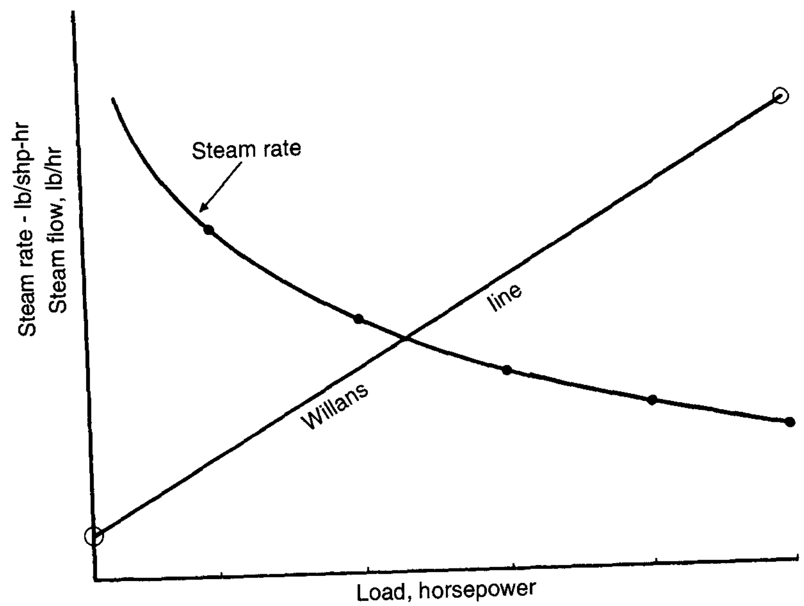


Figure 6-9. Willans line and steam rate

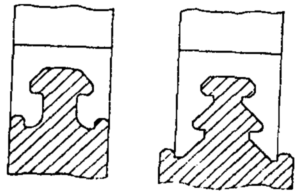


Figure 6-10. Typical impulse turbine dovetails

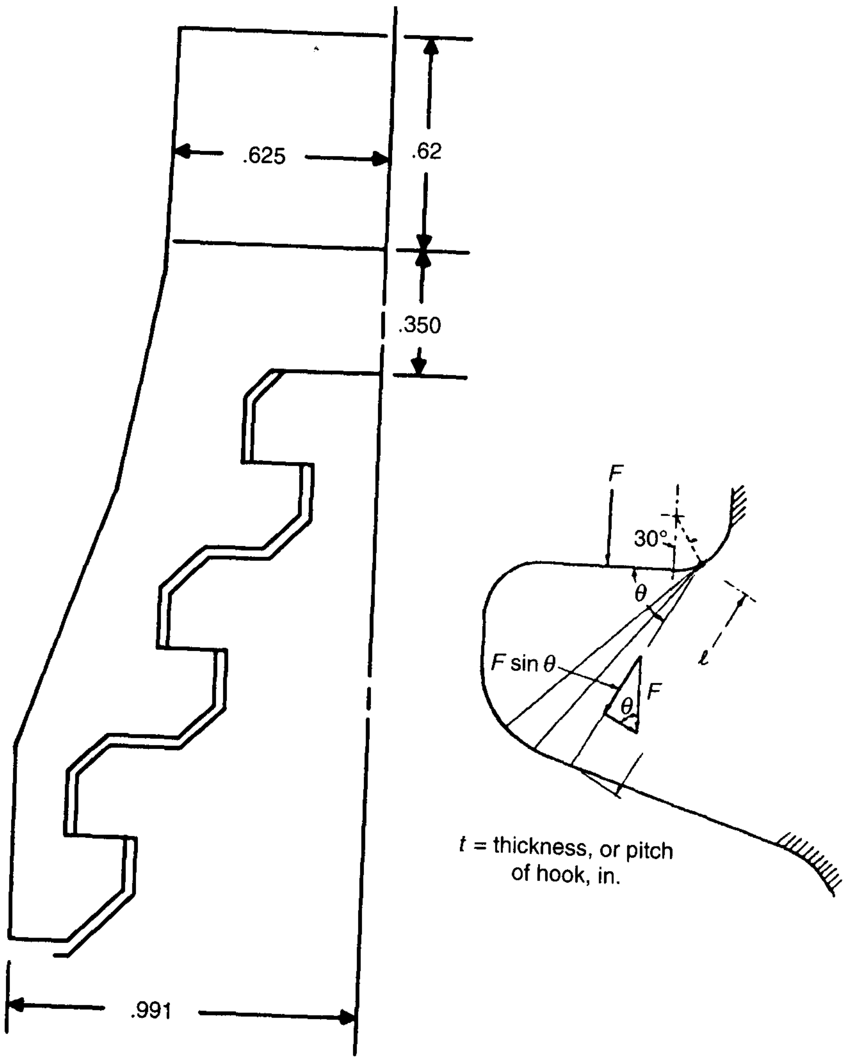


Figure 6-11. Dovetail hook shear stress

single- or multiple-hook designs shown in figure 6-10. The design assumes that the hooks will be equally loaded in shear under the centrifugal forces due to rotation. Figure 6-11 shows a typical single hook of a bucket and the factors required to calculate the shear stress including *F*, the force acting on the hook, *t*, the thickness of the hook in the plane of the wheel, and *l*, the length of the shear plane. The shear stress is

$$T = \frac{F}{t}\left(\frac{\sin\theta}{l}\right)$$

This stress varies with the length of the shear plane and the angle θ. It is these shear planes that must be examined to uncover damage due to overspeed of a turbine rotor.

Vibration Principles

The engineering model for a mechanical vibratory system is shown in figure 6-12. It consists of an oscillating mass connected to a fixed point by a spring and a dashpot. The spring tends to reinforce the vibration amplitude and the dashpot tends to reduce the vibration amplitude. The system may be described by the following differential equation, where the first term is force equal to mass times acceleration; the second term is force equal to the dashpot strength times velocity; and the third term is force equal to the spring constant times the extension of the spring, all of which sum to equal the term on the right side of the equation which is the exciting force that varies with the point in the sinusoidal vibration.

$$\frac{w}{g_c}\frac{d^2y}{dx^2} + \frac{c\,dy}{dx} + kx = F\sin 2\pi f\,t$$

where

F = exciting force, lb
f = frequency of exciting force, cycles/sec
k = spring constant, lb/in
c = capacitance of dashpot, lb sec/in
w = weight of mass, lb
x = displacement from reference point at any time, in.
g_c = conversion constant, 32.2 lbm-ft/lbf-sec
t = time, seconds

The solution of this equation for the vibration amplitude at any time may be expressed and plotted as the frequency ratio, f/f_n, versus the magnification factor, $A/(F/k)$. Figure 6-13 shows that the magnification factor may increase without bound to destructive amplitudes when the actual

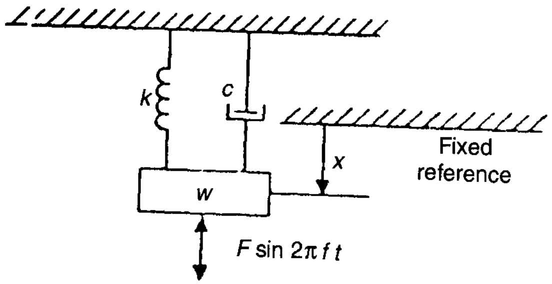


Figure 6-12. Mechanical vibratory system

frequency is equal to the natural frequency of the system if the constants w, c, and k have the correct values. This is a serious problem for the designer of a variable speed impulse turbine who must apply mathematical models, practical experience, and full-size physical tests to find a suitable combination of factors that will satisfy the infinite number of frequencies and modes in which the different buckets of each stage and their disks may vibrate in response to exciting force frequencies while in operation. The

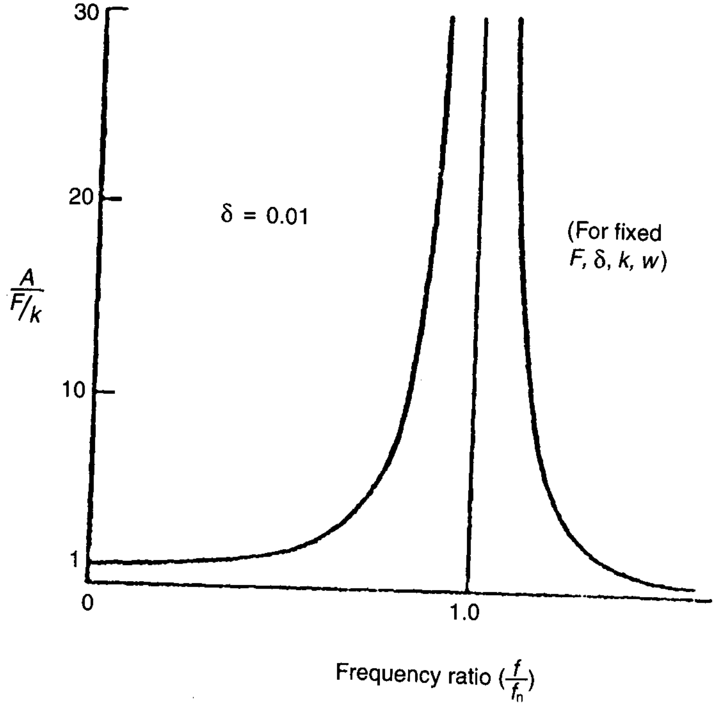


Figure 6-13. Vibration amplitude versus excitation frequency

amount of dampening provided and the fatigue strength are sufficient to prevent the vibration amplitude from reaching destructive levels; however, the engineer officer should avoid continuous operation at any point in the operating range where vibration is obvious.

STEAM CONDITIONS

Most commercial propulsion turbines in service or proposed have been designed for power ratings in the range 16,000 to 60,000 shp. Inlet steam conditions range from 1,465 psig and 950°F down to 600 psig and 850°F. When reheat is specified it is usually at 950°F with the same inlet temperature. Exhaust pressure of1.5 inches mercury absolute seems to be the most common. Depending on the power rating, there may be one to five regenerative heating extraction points. Steam may also be extracted for process use such as freshwater evaporators, contaminated evaporators, electric power generation, boiler feed pump operation, and combustion air heaters. These subjects are covered in detail in chapter 3, "Steam Power Plants."

STEAM TURBINE CONSTRUCTION

Since there are few, if any, reaction turbines in service for main propulsion or electric generation, the turbine construction portion of this chapter will be limited to impulse type turbines. Propulsion turbines designed in the United States are generally a two-turbine cross-compound arrangement. If the turbine is a reheat type, the high-pressure and reheat turbines are tandem compounded in a single casing within a single bearing span. Propulsion turbines built in Europe are sometimes a three-turbine cross-compounded arrangement, which for a given steam condition provides slightly better performance and much higher turbine and gear manufacturing costs.

Turbine Rotors

High-pressure, high-pressure-reheat, and low-pressure rotors are machined from solid steel forgings. In some designs the astern turbine wheels are separate forgings which are shrunk and pinned to the low-pressure rotor. It is the usual practice to locate the astern element on the LP turbine rotor. Some turbines used for Great Lakes propulsion units incorporate the astern turbine on the HP turbine rotor. Most commercial marine turbines are low-pressure, single-flow designs. A number of double-flow LP turbines have been placed in commercial service using surplus naval machinery.

The material selection for the forgings is based on the operating environment, including considerations of temperature, creep stress, centrifugal stress, fatigue stress, corrosion, erosion, and oxidation. These considerations require steel alloy forgings with small percentages of molybdenum, vanadium, chromium, and nickel.

Impulse turbines have balance holes drilled in each disk to ensure that pressure drop cannot develop across the disk and cause an axial thrust which is very significant in reaction turbine designs.

The rim of the integral disks or wheels have a machined shape for connection to the bucket dovetail. Buckets are assembled to the disk by inserting them into a radial slot, then moving the bucket in the circumferential slot to the desired position on the wheel.

Figure 6-14 is an arrangement of a typical high-pressure rotor drawing showing the disks, labyrinth packing grooves, grooves for insertion ofbalancing weights, thrust collar and bearing, output flange, and journal bearings. Figure 6-15 shows an eight-stage, high-pressure turbine rotor during factory assembly. The buckets are assembled, and the bucket covers are held in place by the peened tenons.

Figure 6-16 shows a seven-stage, low-pressure turbine. In five of the stages the block that fills the space used for assembly of the buckets can be seen. The balance holes in the first stage of the turbine are also visible. The astern turbine, separated from the last stage of the ahead turbine by a deflector, is a two-stage turbine with a Curtis first stage and a Rateau second stage.

Turbine Nozzles and Diaphragms

The first-stage nozzle plate is bolted to the high-pressure casing. The astern turbine nozzle plate is bolted to an astern steam ring which is located in the low-pressure casing but is free to expand in the radial direction without subjecting the LP casing to any forces.

With the exception ofthe first stage, each turbine stage has a weld fabricated diaphragm that includes the aerodynamically shaped blade sections that form the nozzle walls. The diaphragm, shown in figure 6-17, fits into grooves machined in the casing. The diaphragm is divided into halves at the center line. The horizontal joint between the halves contains a key to ensure alignment of the halves and to stop steam leakage across the joint. The diaphragm steam seal is a metal-to-metal surface between the casing and the low-pressure side of the diaphragm. The high-pressure side of the diaphragm has crush pins which fill the space between the casing groove and the diaphragm to ensure contact of the steam seal surface on the low-pressure side. The diaphragm is supported at the centerline by the turbine casing so that it is free to expand radially with temperature changes. The bore ofthe diaphragm has spring-packed labyrinth packing to control leakage along the shaft.

The lower-half diaphragm has a slot on the vertical centerline to accept a pin in the bottom ofthe casing groove. The upper half of the diaphragm is

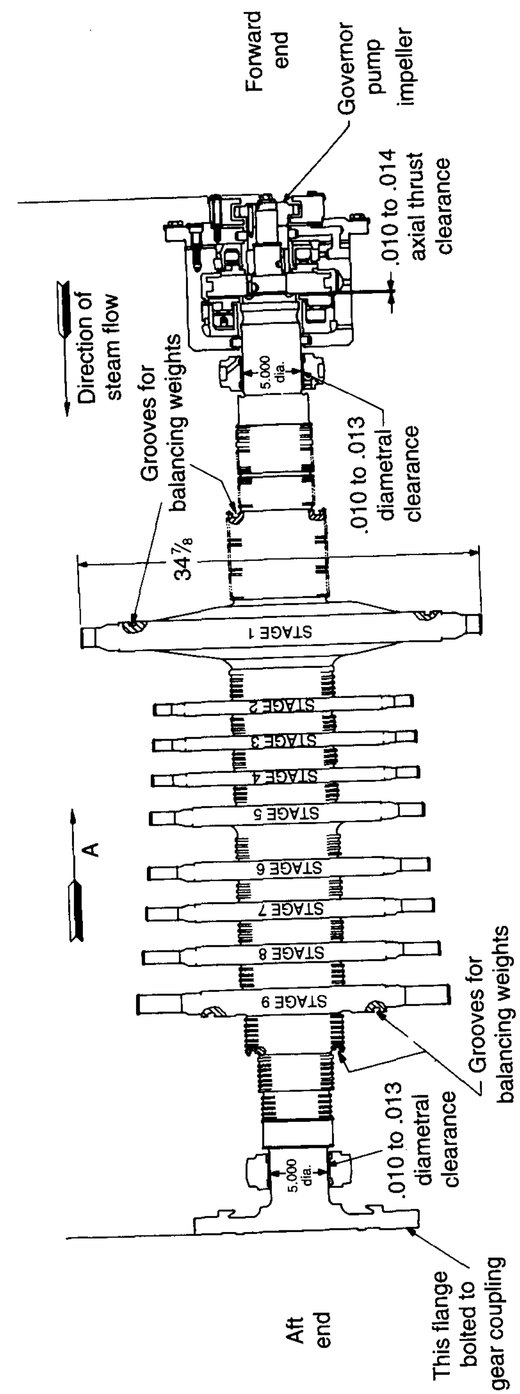


Figure 6-14. Typical nine-stage high-pressure turbine rotor. Courtesy General Electic Company.

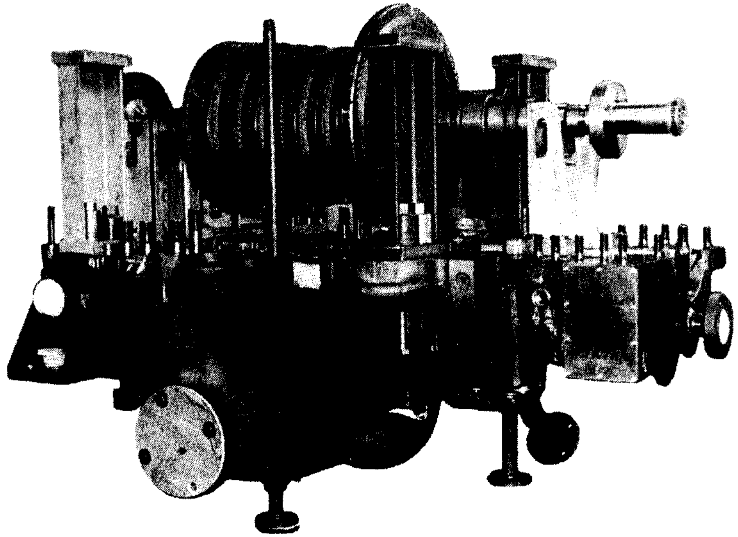


Figure 6-15. High-pressure turbine rotor during factor assembly. Courtesy General Electric Company.

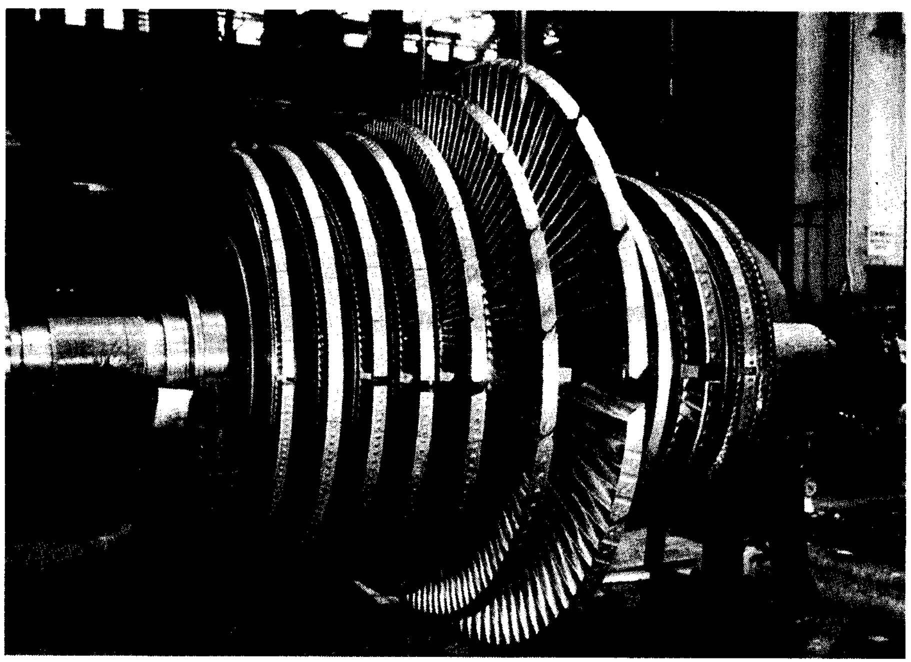


Figure 6-16. Low-pressure turbine rotor with astern element. Courtesy General Electric Company.

held in the upper casing by support bars and holding screws so that the diaphragm halves can be raised with the casing. Support bars support and locate the lower-half diaphragms at the centerline of the casing.

Turbine Casings

High-pressure and high-pressure-reheat turbine casings are made of cast steel using a material suitable for the temperature. These casings consist of upper and lower halves bolted together at the horizontal joint which has a metal-to-metal steam joint the entire length. The steam chest, which includes hand-operated nozzle control valves or hydraulically-operated sequential control valves, is integral with the HP turbine casing. Depending on the design, one or more high-pressure steam connections are cast onto the steam chest.

The upper halves of both forward and aft bearing brackets and packing boxes can be removed for access to these areas for inspection and repair. Figure 6-18 is a view of a lower half high-pressure-reheat turbine casing with diaphragms and shaft packing assembled. The horizontal steam joint, bolt holes, forward bearing bracket, flexible legs under the bearing

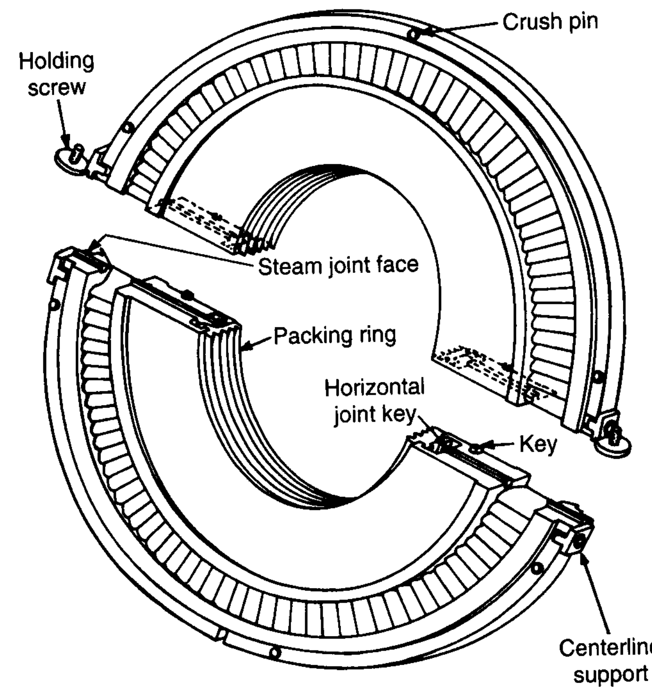


Figure 6-17. Typical turbine diaphragm. Courtesy General Electric Company.

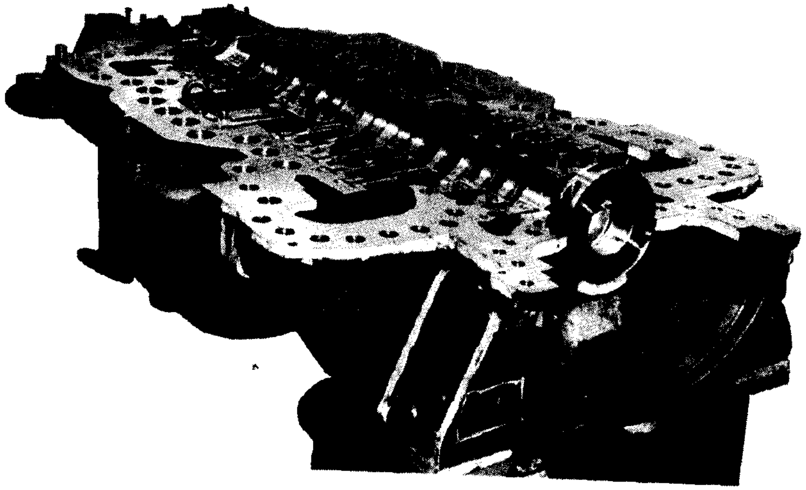


Figure 6-18. High-pressure-reheat turbine lower half casing. Courtesy General Electric Company.

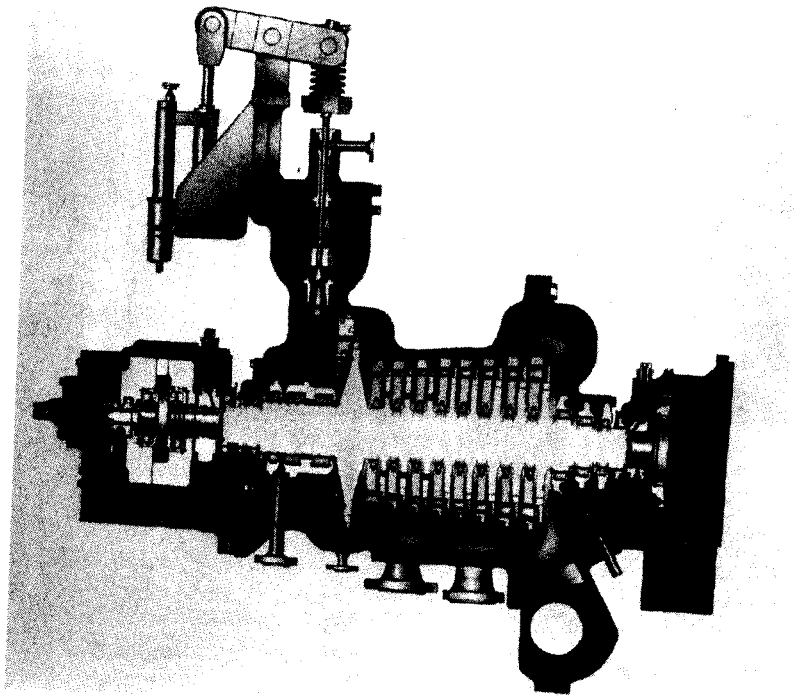


Figure 6-19. High-pressure cross section. Courtesy General Electric Company.

bracket, and forward journal bearing are all visible. Figure 6-19 is a cross section of a modern HP turbine in which the relationship among the casing and the turbine parts is evident.

The low-pressure turbine usually has a mild steel fabricated casing. Depending on the overall system design, the low-pressure casing may be arranged for axial flow exhaust or downward flow exhaust. The downward flow exhaust has design variations dependent on the support of the main condenser. Some condensers are supported by the turbine, which requires the incorporation of a large steel fabrication into the casing to support the weight. In this case, the condenser expansion is downward. If the condenser is supported on a foundation below, the condenser will expand upward, requiring an expansion joint between the turbine and the condenser.

The low-pressure turbine casing has a metal-to-metal steam joint for the entire length along the horizontal centerline. Access to end packing boxes and the bearings is always provided by removable upper covers in these areas. The crossover pipe to the low-pressure turbine is normally connected to the lower half of the LP casing. An inlet for the astern turbine is connected through an expansion joint to the astern turbine steam ring, usually in the forward end of the LP turbine casing for single-flow machines. The steam ring is a centerline supported steel casting which is divided into upper and lower halves which are bolted together. Figure 6-20 is a cross-section drawing of an eight-stage, low-pressure turbine showing the inlet steam crossover connection, two extraction steam belts, and the astern turbine steam ring. Note the provisions for moisture in the three stages before the last stage. Figure 6-21 shows the lower-half casing of a typical low-pressure turbine.

Sentinel Valve

To avoid the potential damage to a low-pressure turbine due to overpressure, a sentinel valve is located on the exhaust casing. This is not a relief valve. It is set to provide an audible alarm when the pressure in the LP turbine reaches about 5 psig. The watch officer must then take needed actions.

Labyrinth Packing and Steam Seal System

Although some auxiliary turbines and early propulsion turbines used carbon rings for sealing the shaft at casing penetrations, modern turbines employ labyrinth-type packing to control leakage from casings and between stages. Labyrinth packing consists of spring packed sections of nickel-lead-bronze sleeves with machined internal fins registering with ring projections machined on the shaft. The resulting close clearances between the stationary fins and the shaft rings progressively throttle the steam leakage out ofthe casing and the air leakage into the casing to levels where they can be handled by the steam seal system. In the case of the diaphragms, the labyrinth packing throttles the steam leakage to an acceptable quantity

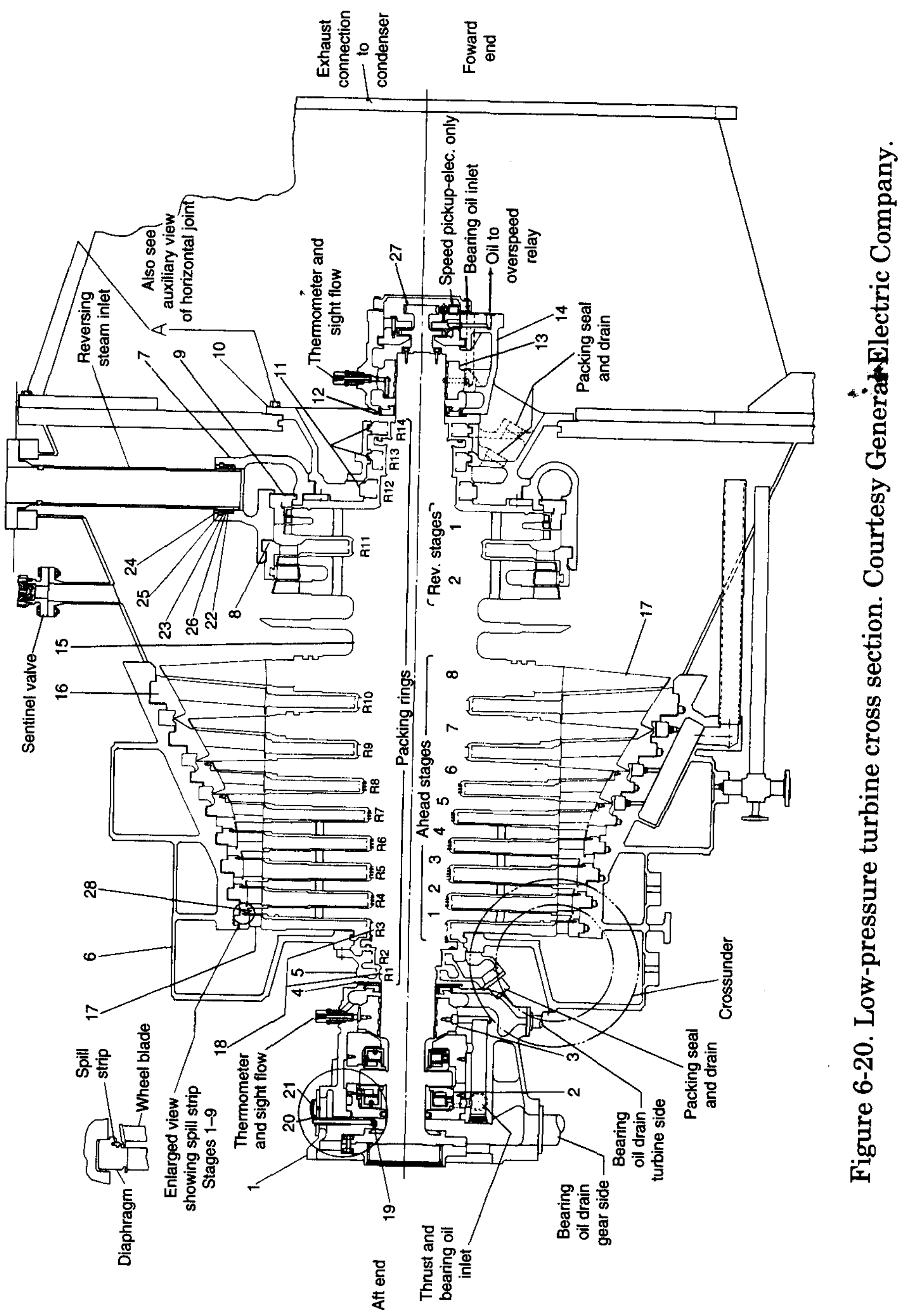


Figure 6-20. Low-pressure turbine cross section. Courtesy General Electric Company.

flowing to the following stage. Figure 6-22 shows a complicated modern labyrinth packing system between a high-pressure turbine and low-pressure turbine, which are located in one turbine casing. In this case, the higher-pressure steam ofthe HP turbine is throttled to the lower-pressure of the reheat turbine.

Since the casing packing glands must respond to a wide range of pressure differences between the internal steam pressure and the external engine room pressure, a steam sealing system is employed on propulsion turbines and turbine generators to control the steam and air flow. (Fig.

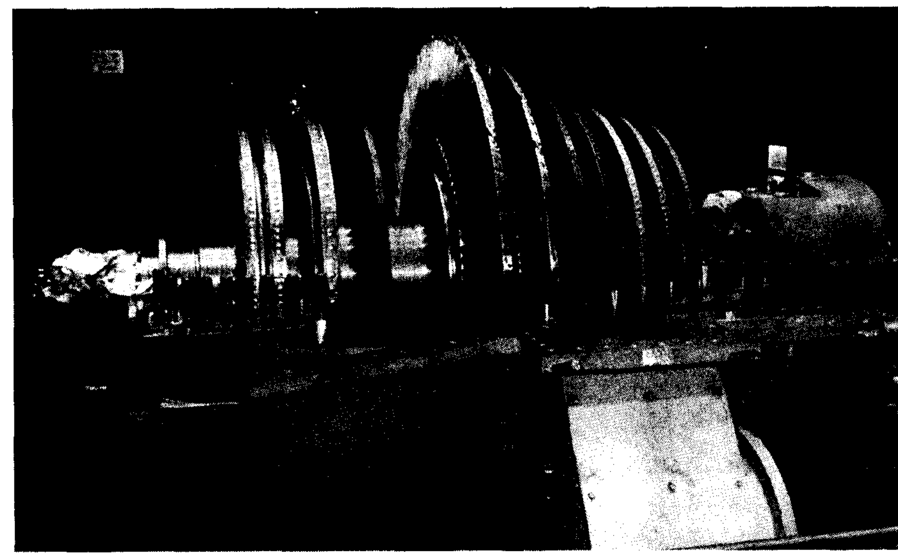


Figure 6-21. Low-pressure turbine casing lower half. Courtesy General Electric Company.

Legend for figure 6-20, facing page

1 Bearing bracket, aft
2 Thrust bearing
3 Journal bearing, aft
4 Oil deflector, aft
5 Packing box, aft
6 Exhaust casing
7 Steam ring for reversing element
8 Diaphragm, stage 2 rev.
9 Nozzle plate and intermediate
10 Reversing head
11 Packing casings, forward
12 Oil deflector, foward
13 Journal bearing, foward
14 Bearing bracket, foward
15 Rotor
16 Diaphragms, ahead stages
17 Nozzle blades for stages 1 to 8, R1 and R2
18 Packing rings, R1 to R14
19 Ball bearing for rotor position indicator
20 Spring for rotor position indicator
21 Rotor position indicator
22 Inner ring
23 Spacer
24 Locking ring
25 Locking ring
26 Outer ring
27 Overspeed impeller
28 Buckets, stages 1 through R1

6-23 shows a typical steam seal and exhaust system.) End packings in a turbine casing are usually divided into three sections of labyrinth seals with an intervening pocket between each section. The innermost pocket is connected to the steam seal regulator, which maintains a pressure of2 to 4 psig in this space. When the pressure inside the turbine casing is high, steam is throttled through the innermost set of labyrinth rings to the innermost pocket, where most of it is dumped to the main condenser by the regulator. If pressure is low inside the turbine casing, i.e., a vacuum, the regulator supplies steam to the innermost pocket to maintain pressure and to supply sealing steam which is throttled by the labyrinth rings to the turbine casing. The outermost pocket is connected to the gland exhauster which is a fan and heat exchanger. Both the air entering along the shaft through the outermost packing rings and the small quantity of steam leaking through the middle set of packing rings from the pressurized innermost pocket are exhausted from the outermost pocket by the gland exhaust system.

Figure 6-24 is a cross section of a steam seal regulator. The valves are hydraulically operated by lube oil pressure. A sensing line connected to the innermost pocket actuates a bellows (54), which moves a pilot valve (62),

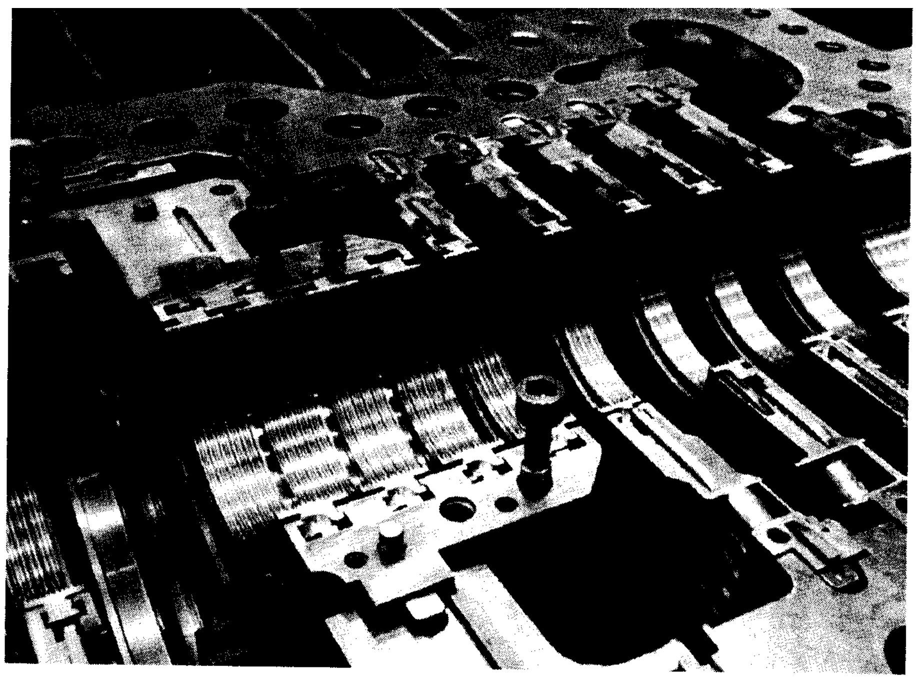


Figure 6-22. Modern labyrinth packing between HP turbine and reheat turbine in a common casing. Courtesy General Electric Company.

which controls the hydraulic piston (58), which operates the steam supply valve (30), to raise the pressure in the innermost pocket or the dump valve (60), which dumps excess steam to the condenser to lower pressure in the innermost pocket.

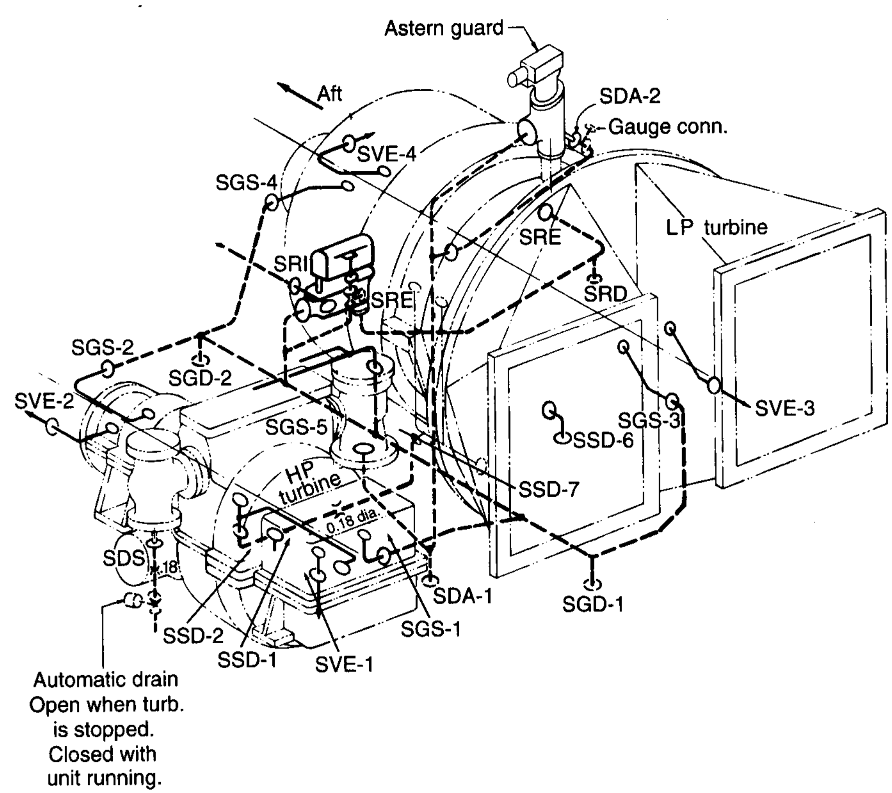
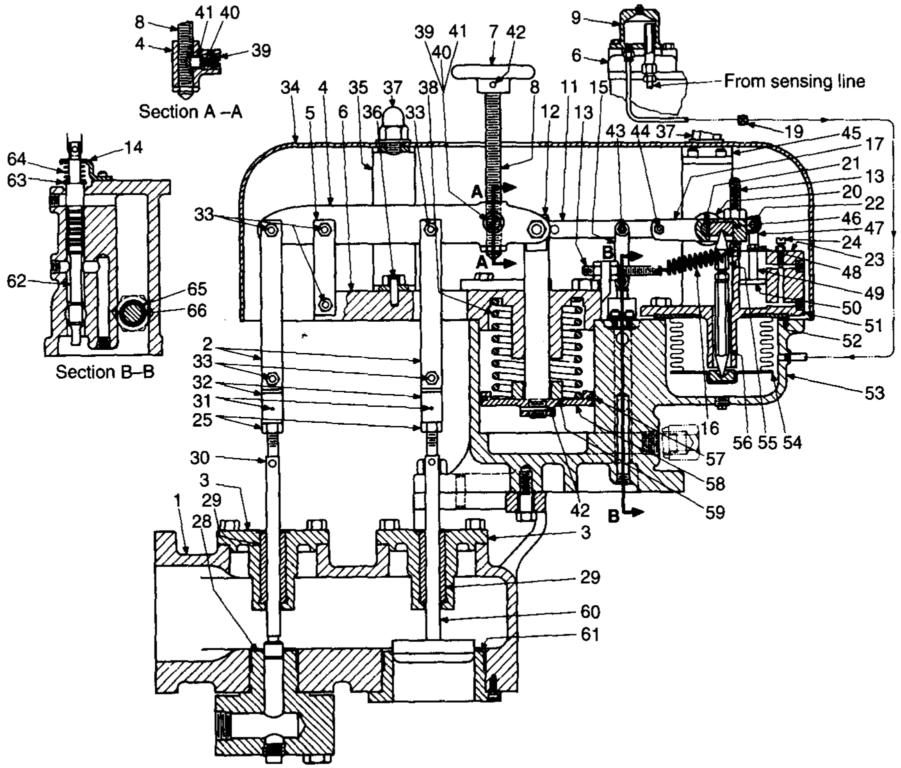


Table of Connections

SGS-1	Glandseal-H P turb.-fwd	SRI	Inlet-steam seal regulator
SGS-2	Glandseal-H P turbo-aft	SRE	Dump-steam seal regulator
SGS-3	Glandseal-LP turbo-fwd	SRD	Drain-steam seal regulatordump
SGS-4	Glandseal-LP turb.-aft	SDS	Drain-steam strainer
SGS-5	Valve stem leak-offs- ahead and stern	SDA-1	Drain-astern elementsteam line
		SSD-1	Drain-HP turbo-firststage
SVE-1	Glandvent-HP turbo-fwd	SSD-2	Drain-HP turbocrossunder and HP packing leak-off
SVE-2	Glandvent-H P turbo-aft		
SVE-3	Glandvent-LP turbo-fwd	SSD-6	Drain-LP turboexhaust
SVE-4	Glandvent-LP turbo-aft	SSD-7	Crossunderdrain conno from HP turb
SGD-1	Drain-steam seal manifold-fwd		
SGD-2	Drain-steam seal manifold-aft	SDA-2	Drain-astern guard valve

Figure 6-23. Typical steam seal and exhaust system. Courtesy General Electric Company.



1	Manifold	23	Cover	45	Bracket
2	Link	24	Adjusting screw	46	Pivot block
3	Cover	25	Nut	47	Helm joint
4	Lever	26	Taper pin	48	Lock nut
5	Link	27	Dowel	49	Piston rod
6	Cover	28	Valve seat	50	Piston
7	Handwheel	29	Bushing	51	Cover
8	Screw assembly	30	Valve	52	Gasket
9	Expansion tank	31	Taper pin	53	Primary relay housing
10	Pin	32	Knuckle	54	Bellows assembly
11	Lever	33	Pin	55	Pivot rod
12	Piston rod	34	Cover	56	Pivot rod
13	Adjusting screw	35	Bracket	57	Piston ring
14	Spring clip	36	Plug dowel	58	Piston
15	Link	37	Acorn nut	59	Spacer
16	Spring	38	Spring	60	Valve
17	Lever	39	Set screw	61	Valve seat
18	Bearing retainer	40	Spring	62	Pilot valve
19	Needle valve	41	Plug	63	Spring plate
20	Bearing	42	Taper pin	64	Spring
21	Shaft	43	Pin	65	Plug
22	Pin	44	Pin	66	Plug

Figure 6-24. Steam seal regulator. Courtesy General Electric Company.

A similar steam seal system with regulator is used with turbine-generator sets.

STEAM TURBINE CONTROLS

There are two basic approaches to the control of steam flow and consequently load for a propulsion steam turbine. First is a simple throttle valve for the ahead and astern turbines. Second is an electrohydraulic throttle control system which includes many other aspects of turbine control beyond steam flow control.

Main Throttle Valve System

Figure 6-25 illustrates a typical propulsion turbine throttle control system. All valves in the system are manually operated. During maneuvering operations, the nozzle control valves located in the steam chest are all open and the power is controlled by throttling the steam with the ahead valve. Since throttling causes a loss, the nozzle control valves are set to give the desired speed, i.e., power, and the ahead valve is set fully open during underway steady-state conditions.

The astern valve is used only during maneuvering for astern operation. The astern guarding valve, which is in series with the astern valve, is open during maneuvering and closed during underway conditions to preclude the chance of possible losses and damage due to steam leakage into the astern turbine while the ahead turbine is in operation. A steam strainer is located ahead of the throttle valves to prevent damage to the valves by foreign material. This strainer should be inspected periodically.

The ahead valve hydraulic operating cylinder is designed to close the throttle valve in the event of loss of lube oil pressure, LP turbine overspeed, or HP turbine overspeed. Each turbine has a centrifugal pump impeller located on the forward end of the rotor. The pump supplied with lube oil has a discharge pressure which is a function of the rotor speed. The pump discharge pressure actuates an overspeed relay which moves a pilot valve to close the throttle valve for the set overspeed. The system includes test valves to verify that the throttle functions properly for overspeed and low oil-pressure conditions. The system is designed to limit the speed to 110 percent of rated full-power speed. The low lube oil shutdown occurs when the lube oil pressure to the bearing drops below 6 psig.

Electrohydraulic Throttle Control System

The electrohydraulic throttle control system is particularly compatible with central control systems and bridge control systems described in volume 2 of *Modern Marine Engineer's Manual.* In this system, electronic signals generated on the bridge or at the main engine control station control

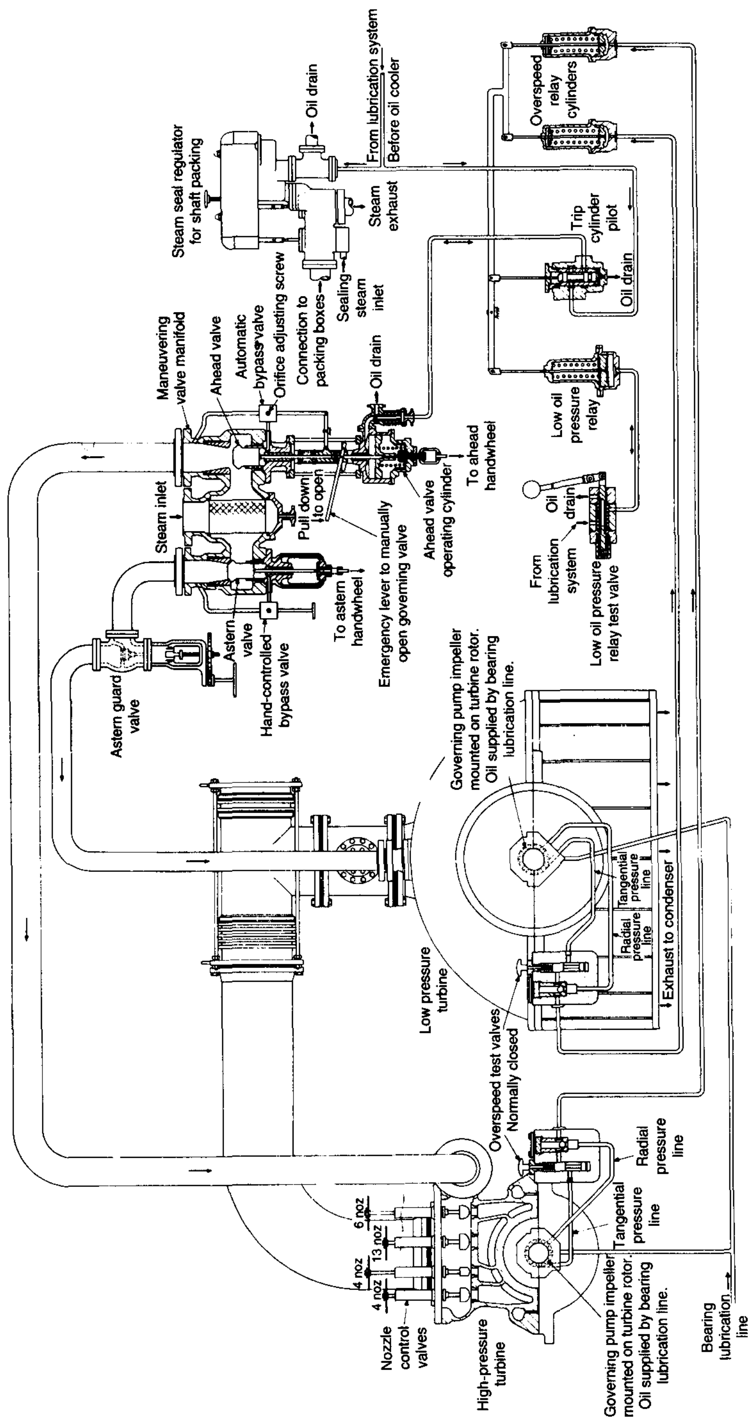


Figure 6-25. Propulsion turbine control system. Courtesy General Electric Company.

ahead and astern steam flow to provide the required speed and direction of shaft rotation. The system also receives and acts appropriately on signals from transmitters for turbine overspeed, low steam-generator pressure, boiler steam drum water level (high or low), excessive turbine vibration levels, turbine rotor axial position indicator, and automatic turbine rollover during standby conditions. Most turbine trip devices, loss of lube oil pressure, and excessive overspeed are independent ofthe electrohydraulic control system.

Figure 6-26 is a schematic ofthe control system. Referring to the drawing, it may be seen that the throttle lever synchro on the bridge or at the main engine control station is used to establish the shaft speed and direction. The synchro signal for desired speed is converted by the function generator into a signal for the selection and positioning of the steam control valve or valves. The rate limiter opens or closes valves at a rate suitable for proper operation of the overall power plant. A shaft speed transmitter provides a feedback signal to modulate the function generator for the exact speed required.

The rate limiter signal is transmitted to a servo system with valve position feedback which opens or closes the required throttle control valves. The servo system pump operates hydraulic rams that move the ahead or astern valves, as shown in figure 6-27.

Electrohydraulic turbine control systems are usually applied to high-horsepower turbines. Therefore, because of the improved underway turbine performance, it is common with these systems to sequentially control the nozzle valves instead of using a single throttle valve. Figure 6-28 shows the assembly of the bar lift for typical sequential opening of turbine nozzle valves for speed control of a propulsion turbine. A hand pump is included in the system for manual control of the valves in the event of a malfunction of the controls or the system's hydraulic pump. Direct mechanical operators are also included for emergency control of the valves.

TESTS AND INSPECTIONS

In addition to periodic tests and inspections required by regulatory agencies and described in chapter 15, it is desirable for the engineer officers and port engineers to take advantage of opportunities to ensure the satisfactory operating condition of steam turbines by performing confirming tests and inspections. Some of the potential problems that may be uncovered by routine test and inspections are described in the following sections.

Steam Path Deposits

Even when the boiler water treatment and boiler operation is scrupulously attended, solids carryover from the boiler may cause deposits of silica,

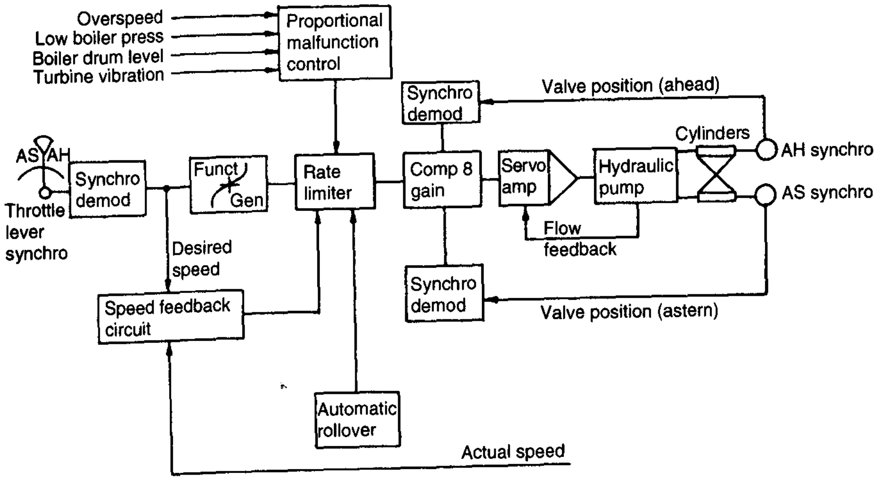


Figure 6-26. Electrohydraulic turbine control system. Courtesy General Electric Company.

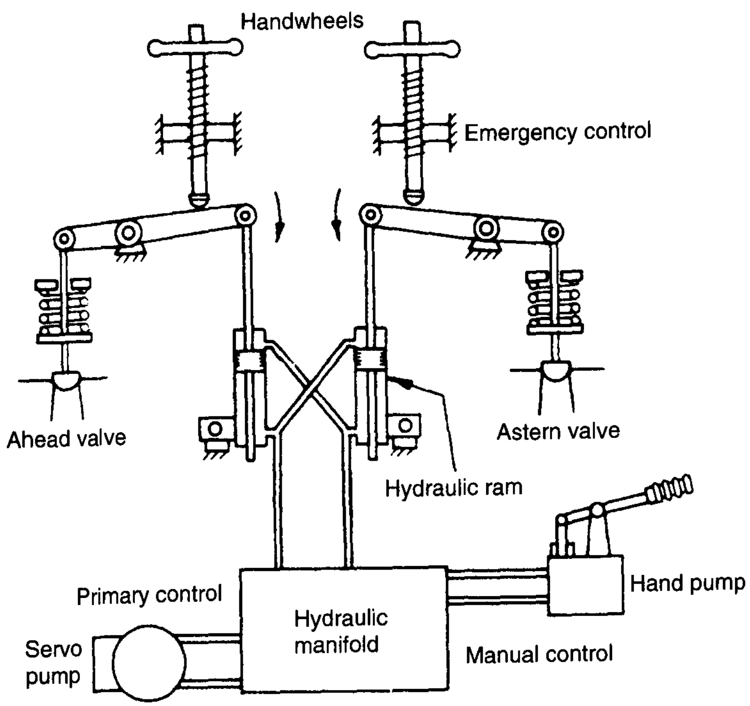


Figure 6-27. Hydraulic ram operators for control valves. Courtesy General Electric Company.

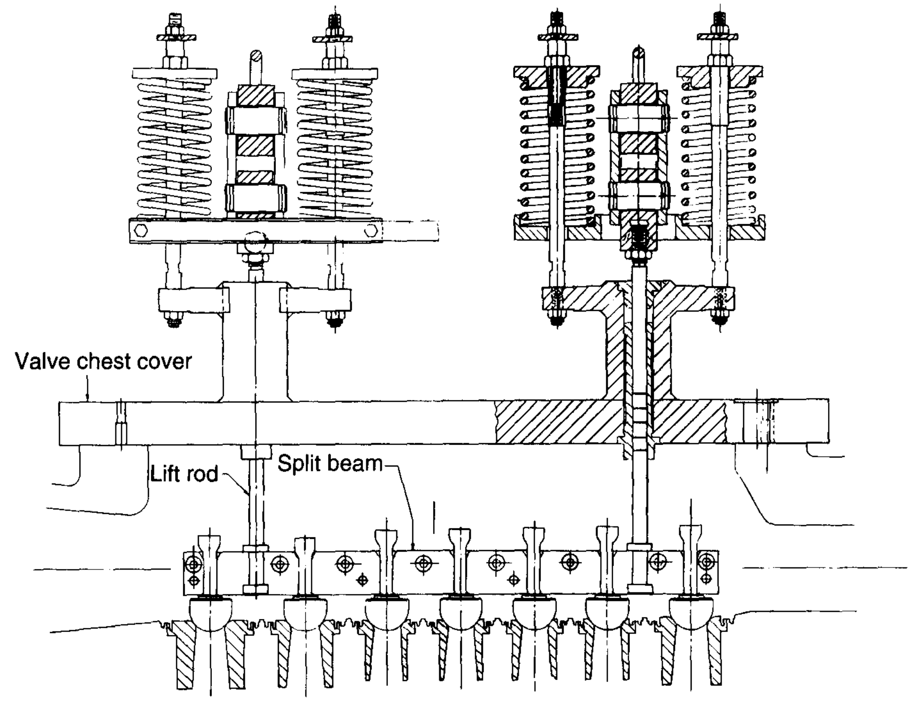


Figure 6-28. Bar lift arrangement for movement of ahead nozzle control valves. Courtesy General Electric Company.

salts, oxides, etc., in the turbine steam path. Such deposits cause a deterioration in the turbine efficiency which can be substantial, as much as 10 percent in extreme cases. A crude and not always satisfactory test is to record first-stage bowl pressure and extraction-stage pressures using a calibrated pressure gauge for a specific high-load setting of the first-stage nozzle valves. These pressures can then be compared to subsequent pressures recorded periodically at the same turbine load condition. Any significant change in pressure must be investigated to determine the presence of deposits and a need for cleaning.

Deposits in turbines may cause increased and unexpected loads on diaphragms, buckets, and thrust bearings as well as unbalance of the rotor. Deposits may also interfere with the proper operation of the first-stage nozzle control valves.

Enthalpy Drop Test

When a turbine (such as a modern high-pressure turbine or a reheat turbine) is operating entirely in the superheat region on the Mollier chart, it is possible to conduct a simple but accurate test to determine the efficiency, for comparison to original start-up efficiency or to a test conducted after the most recent overhaul. This test is called the enthalpy drop test.

At a specific full-load setting of the first-stage nozzle control valves, use calibrated instruments to measure the pressure and temperature of the steam in the first-stage bowl and the turbine last stage. Determine the absolute pressure by correcting the calibrated gauge pressures measured by the barometric pressure. Enter the Mollier chart and determine the isentropic enthalpy drop from the initial condition to the exhaust pressure. Then determine the enthalpy drop from the initial condition to the actual exhaust condition. The ratio of the actual enthalpy drop to the isentropic enthalpy drop is the efficiency of the turbine. If the instruments are accurate and calibrated, this is a reliable estimate of the efficiency which can be used for comparisons to past efficiency measurements at the same load condition. Unfortunately, the enthalpy cannot be used for a low-pressure turbine or any turbine operating in the moisture region of the Mollier chart since the quality of the steam is unknown.

A record of enthalpy drop tests over time will be a useful indicator of deterioration of turbine performance due to a variety of reasons.

Steam Leakage

Losses occur when steam leaks through shaft packing, bucket-to-nozzle clearances, and diaphragm joints. There are also numerous drains on turbines that cause losses if the valves are fully or partially open when the turbine is in the normal operating mode.

Modern propulsion turbines have spring-backed labyrinth packing which minimizes damage to the packing when there are momentary shaft rubs. If boiler carryover or some corrosive agent is introduced into the turbine, the freedom of the packing segments to move radially may be restricted, leading to damage and consequent leakage when a rub occurs due to a thermal distortion. Damage to packing on a propulsion turbine could cause a 1 to 2 percent decrease in turbine efficiency. An enthalpy drop test would detect such deterioration in a high-pressure turbine. The low-pressure packing damage is usually detected during physical examination of the packing. The best way to avoid damage to packing and spill strips is to operate the turbine so that excessive thermal transients and rotor bowing are avoided.

Steam Path Corrosion

Normally the interior of a steam turbine does not have an environment that is conducive to corrosion. However, the presence of salts from boiler carryover can lead to corrosion when the turbine is operating in a moisture region or when moisture is present during an extended shutdown. If salts are detected during an inspection, the turbine should be washed with fresh water.

Steam Path Erosion

Modern turbine steam path components are made of materials selected to resist erosion, especially in the low-pressure stages. Also the last few stages frequently have moisture removal devices built into the stages. As a result, erosion is not common. If erosion occurs, the aerodynamic surfaces of the buckets and nozzles lose their fine finish resulting in a decrease in stage efficiency. It may be necessary to replace buckets and nozzles if serious deterioration occurs.

Mechanical Damage

Mechanical damage to nozzles or buckets occurs when small metallic particles are able to pass through the steam strainer. Mechanical damage can also occur through handling of parts of a disassembled turbine.

Low-Pressure Turbine Inspection

If the engineer officers are making regular enthalpy drop tests, this will provide valuable information about the high-pressure turbine stages but will reveal nothing about the condition of the low-pressure turbine. Regular physical examination of the last stage buckets via the exhaust hood inspection openings will provide much information about the last-stage turbine. Last-stage bucket examination will reveal any erosion of last-stage buckets, deposits on nozzles or buckets, and any damage due to foreign material.

Wheel and Diaphragm Inspection

Whenever a turbine is open for periodic inspection, the engineer officers should use the opportunity to examine the turbine rotors with great care, irrespective of other inspections which may be undertaken by manufacturer representatives, repair workers, classification society engineers, or other regulatory agency representatives.

Any cracks or porous areas that extend into the wheel are an indication that the strength of the material has been reduced. The wheel disks, buckets, and bucket covers must be examined for defects, corrosion, erosion, and mechanical damage. Since the long low-pressure region buckets operating in wet steam are the most highly stressed in both centrifugal and vibratory modes, they must be more critically examined than the short buckets.

Bucket covers should be examined for looseness. This is accomplished by tapping the covers lightly with a hammer and noting if there is any change in the relative position between the covers and the buckets. A dead metallic sound when tapping a cover may indicate a broken tenon or a loose cover. Covers should also be inspected for fatigue cracks which usually radiate from the corners of the tenon holes.

The vane section of the buckets must be inspected for mechanical damage, fatigue cracks, and erosion. Fatigue cracks start at either side of the bucket and progress across the vane. Failures are more likely to occur near the root of the vane.

Inspection of dovetails of assembled buckets is difficult. The following checks will be helpful: sighting along the covers while looking for abrupt changes in the cover arc, observing the relative height of buckets by scanning along their bottom overhanging edges, and looking for excessive clearance between the wheel rim and the bucket. If a banded group of buckets leans in the axial direction, this may be the result of a cracked dovetail. If the bucket has a dovetail outside the rim of the wheel, cracks usually start on the inside of the bucket hook (fig. 6-11), so when they show up on the outside, it should be assumed that a complete failure has occurred across the hook shear area.

Buckets suspected of being loose may be inspected by lightly tapping the bucket with a hammer. A high metallic ring indicates a tight bucket and a dead sound indicates a loose bucket.

Ifthe bucket has tie wires, they should be inspected for cracks and the tie-wire holes must be inspected for the beginning of cracks radiating form the hole.

Overspeeded Rotors

If a turbine is overspeeded beyond the overspeed trip point, it is essential the rotor be carefully examined for stress beyond the elastic limit. Since all impulse turbine wheels have balance holes, they are a useful place to start an examination. Carefully clean the balance holes of rust and deposits with fine emery cloth. Then make accurate micrometer measurements of the balance hole diameters in the radial and tangential directions relative to the shaft. If these measurements show a greater dimension in the radial direction than in the tangential direction by .015 inches or more, this suggests that the wheel has been stressed beyond the elastic limit. Check all the balance holes for a verification of the elliptical shape of the holes. The manufacturer must be consulted to determine the need for a replacement rotor if there is evidence of an overstressed rotor.

ROTOR VIBRATION MODES AND AMPLITUDE

The vibration analysis of rotating machinery is covered in detail in volume 2. However, it is desirable to present the general characteristics ofpropulsion turbine rotor vibration in this chapter.

Vibration Limits

An acceptable level of vibration for propulsion turbines and major auxiliary turbines at full power is three mils displacement measured on the shaft and one mil displacement measured on the bearing cap.

Causes of Rotor Vibration

Rotor vibration is caused by a periodic driving stimulus and the response of the rotor structure. Vibration may be reduced by a reduced stimulus or by a reduced response. Unbalance is the most common vibration stimulus. Rotors are unbalanced when the center of rotation does not coincide with the center of mass. A very good balance might be interpreted as achievement of an eccentricity of 20 microinches or better.

Temporary unbalance due to rotor bowing by thermal distortion is gross compared to a well-balanced rotor to the extent that it is destructive to the turbine to operate with a bowed rotor. There are small changes in turbine rotor balance due to changes in load and temperature. These changes are readily detectable with modern instrumentation available for shipboard use.

Rubbing of shaft packing or deflectors will result in local heating. The local expansion due to the heating causes rotor unbalance. It is impossible to balance a rotor in place when rubbing is occurring.

If present, misalignment may change the vibrational response characteristics to the existing unbalance stimulus.

Critical Speed of Rotor

The critical speed of a rotor is the speed that corresponds to the natural frequency of vibration. There are several critical speeds which are multiples ofthe first critical. Ship propulsion turbines are designed for full-power operation between the first and second critical. Therefore, the turbines must pass through the first critical as propeller speed is increased. Although propulsion turbines are designed with sufficient damping to prevent destructive vibration amplitudes at the first critical, engineer officers should avoid operation of turbines at the first critical by quickly passing through the critical speed.

PROPULSION TURBINE OPERATION

Preparations for Getting Under Way

1. Start up lubricating oil system.
   a. Check that steam valves to turbine are closed.
   b. Open any valves in lines to bearings.
   c. Open any valves in sensing devices.
   d. Ensure that cooling water is available to lube oil coolers.
   e. Start lubricating oil pump.
   f. Check lube oil pressure at most remote bearing in the system.
   g. Check all lube oil sight flows in the system. The temperature of the lube oil should be about 90°F before getting under way.
2. Engage and start turning gear.

a. Check all line shaft bearings, seals, etc., to ensure that they are ready for operation.
b. Start the turning-gear motor.
c. Check the operation of the rotating system, listening for unusual noises. Secure and investigate any problems observed.

3. Start up the vacuum system.
a. Open shutoff valves for gauges and alarms.
b. Open all turbine orifice and trap-type drains and secure atmosphere drains.
c. Secure shutoff valve ahead of steam seal regulator.
d. Pressurize supply line to steam seal regulator.
e. Start up the main condenser.
f. Start up air removal system (air ejector or vacuum pump).

4. Start the gland exhaust system as soon as vacuum begins to rise.
a. Open the shutoff valve ahead of the steam seal regulator and check that the header pressure is between 2 and 4 psig.
b. Check the packing boxes on turbine shaft to ensure that steam is not leaking out.
c. Check that vacuum continues to rise.

5. Make final checks of overall system.
a. Recheck lubricating oil system, bearing oil pressure, control oil pressure, and sight flows. Ensure that oil temperature is above 90°F.
b. Recheck the rotating elements, including line shaft bearings, stern tube bearings, reduction gear, turbines, turning gear, etc.
c. Adjust the lube oil cooling system to set the bearing oil supply temperature to 110° to 120°F.

6. Check ahead and astern steam control valves or throttle valves for proper operation.
a. Ensure that the main steam supply or stop valve to the turbine is closed and that there is no steam pressure ahead of the turbine control valves.
b. Operate and observe the ahead and astern steam control valves through full stroke.
c. Close the ahead and astern steam control valves and open main steam supply valves to pressurize the line ahead of the control valves.
d. Use drains to remove water from main steam line and blow down the main steam strainer.

7. Warm up the propulsion turbines.
a. Ensure that the propeller is free to rotate and notify the bridge that the propeller will be turned.
b. Open astern guard valve.
c. Stop the turning-gear motor and disconnect the turning gear.
d. Immediately when off turning-gear, open the ahead throttle to bring speed to 5 percent oHull speed rpm, then secure the ahead throttle and open the astern throttle to bring the revolutions to 5 percent of speed. Close the astern throttle and repeat this cycle of operations for about forty minutes. During this period observe oil pressures, temperatures, and gland seal operation and listen for unusual noises. If there are no indications of improper operation, the turbine is now ready to get under way. Do not allow the turbine to remain stationary for more than one minute while in this standby mode. Some turbines are fitted with control systems that provide automatic standby rollover of the turbines.

Normal Underway Operation

When the maneuvering condition is ended and the steady-speed sea conditions have been established, a final check of all operating parameters should be made to ensure that the turbine is operating correctly, including thermal performance, mechanical behavior, gland seal system, and lubricating oil system. Test the overspeed relays; check that turbine monitoring systems are operating; check bearing temperatures; and ensure that control systems are functioning. Secure the astern steam guarding valve. Set nozzle hand control valves, if any, to provide the desired speed with the ahead throttle full open. Specific items to be regularly checked during underway operation include the following:

- oil pressure at most remote bearing
- oil level in lube oil tank
- main steam temperature and pressure
- vibration levels
- turbine exhaust pressure
- condenser hot-well level
- oil flow to bearings
- bearing discharge oil temperature
- rotor position indicators
- extraction stage temperatures and pressures
- first-stage bowl temperature and pressure
- HP turbine exhaust temperature and pressure
- reheat turbine exhaust temperature and pressure
- steam seal pressure
- condition oflube oil, chemical and physical
- operation of turbine drains

Securing the Propulsion Turbines

1. Close the ahead and astern control valves and the astern guarding valves. Close the main steam supply valves to the turbine.
2. Engage and start the turning gear. Check the oil flow to all bearings.
3. Mter thirty minutes on turning gear, secure the vacuum system and the steam seal system.

4. When vacuum has decreased to five inches Hg or less, secure the gland exhaust system.
5. When lube oil temperature difference between the oil entering and leaving all bearings has decreased to about 15°F, stop the turning gear and secure the lube oil system.

PROPULSION STEAM TURBINE OPERATING PROBLEMS

The engineer officer is faced with many possible problems which may arise during propulsion turbine operation. The following is a list of typical problems with suggestions for analysis and correction of the problems.

Low Bearing Oil Pressure

Symptoms: Low oil pressure (between 4 and 8 psig) at the remote bearing; operation of low pressure trip switch; loss of discharge pressure from main lube oil pump.

Action: Prepare to close steam throttle or control valve(s). Restore oil pressure if possible. Check that oil is available in sight glass. If unable to restore pressure, stop the shaft using astern turbine. Do not permit the shaft to turn until pressure is restored.

Probable Cause	*Remedy*
Plugged strainer	Switch to clean strainer.
Failure of oil pump	Restore pumps to service.
Oil header reducing valve stuck	Regulate by hand until valve is repaired.
Low oil level	Refill tank and check for leaks.
Defective monitor	Check pressure switches.
Broken oil line resulting in leakage	Secure pumps and stop shaft, then repair leaks while preventing oil from contacting hot surfaces where a fire might start.

Low Control Oil Pressure When Supplied by Lube Oil

Symptoms: Low oil pressure on gauge; steam control valves move toward closed position; steam seal pressure increases causing steam leakage at turbine shaft seals.

Action: Check bearing pressure and, iflow, follow low lube oil pressure actions. If bearing pressure normal, operate turbine steam control valves and steam seal regulator manually. Reduce the ship speed until the problem is resolved.

Probable Cause	*Remedy*
Failure of pump	Check pump is free to turn, check power to pump.
Bearing header regulating valve stuck in open or closed position	Start standby pump and repair valve.
Excessive air in oil	Check oil for water that may have removed foam inhibitor.
Defective gauge	Replace gauge.
'Broken oil line	Repair oil line.
Low oil level	Fill oil sump.

High Oil Temperature

Symptoms: Temperature alarm indication; local bearing temperature high; lube oil temperature high.

Action: Slow turbine to one-half of full speed and adjust cooler valves, check oil flow to bearings, check oil pressure at bearings, check for vibration, check oil cooler discharge temperature, check thermometers, check oil strainer for babbit, and check rotor position indicator.

Probable Cause	*Remedy*
Wiped bearing	Shut down turbine and inspect bearings.
Improper cooler outlet temperature	Correct oil inlet temperature.
Reduced oil pressure from pump	Adjust the oil feed pressure.
Dirt in the oil	Operate the oil purifier.
Plugged oil feed line	Locate and remove plug.
Improper bearing clearance	Check and correct bearing clearances.
Improper adjustment of cooling water or lube oil valves	Correct adjustment.
Interruption of water flow to cooler	Find cause and correct.
Water or cooler side of heat exchanger is air bound	Vent both sides of cooler.
Change of seawater temperature	Increase flow of cooling water.

No Oil in Sight Flow

Symptoms: Oil is not visible in sight flow glass.

Action: Slow turbine to 50 percent speed and stop shaft as soon as practical, then check the following: bearing oil pressure, vibration, rotor position, oil strainer.

Probable Cause	*Remedy*
Reduced oil pressure	Correct oil pressure reduction cause.
Wiped bearing	Stop shaft, inspect bearings, and replace if necessary.

Probable Cause	Remedy
Dirt in oil	Operate separator to clean system.
Plugged oil feed line	Locate and correct condition.
Improper bearing clearance	Locate and correct improper clearance by scraping or replacing bearing.

Lube Oil Leakage

Symptoms: Visible oil on machinery, smoke from hot machinery surfaces; lowering oflevel in oil supply tank; oil in gland evacuator drain; oil running out of oil deflectors; oil on top of supply tanks.

Action: Prepare to extinguish fire. Remove oil from hot surfaces and remove oil-soaked insulation. Determine reason for losses of oil. Reduce gland evacuator suction. Check that gear casing vent lines are free from plugging. Check for excessively high level in supply tank.

Probable Cause	*Remedy*
Leak in system	Check for and correct the following conditions: leaking cooler tubes, open drains on cooler or strainer, leaking oil deflector, leaking sample connection and excessive losses from purifier.
Gland evacuator suction too high	Reduce gland evacuator suction.
Plugged casing vents	Clean gear casing vents.
High supply tank level	Correct level.

High or Low Level in Oil Supply Tank

Symptoms: Supply tank indicator high or low.

Action: Check for excessive water in lube oil and stop the shaft if present. If below minimum, stop the shaft and restore level.

Probable Cause	*Remedy*
Water in oil	Find cause and correct, remove water.
Improper operation of purifying system	Correct problem.
Oil level gauge defective	Repair gauge.
Oil leaking from system	Find and correct cause.
Leaking oil cooler	Shift to standby cooler, find and repair cause.
Improper trim of ship	Correct trim.

Lubricating Oil Contamination

Symptoms: Foaming; emulsion with water; scoring of bearing; plugged strainers; excessive rust or dirt in system; sludging; chemical analysis reveals a problem.

Action: Check for high bearing temperature and low pump discharge pressure and stop the shaft if conditions exist. Check for excess water in oil; if present drain excess water, operate the purifiers, and increase the oil temperature to maximum allowable level. Replace scored bearing(s) and clean system. Clean the strainers. Correct oil condition following recommendations of oil supplier.

Probable Cause	*Remedy*
Water removal of inhibitors	In all cases identify and correct the cause.
Leak in cooler	
Dirt or chlorine additives in oil	
Wiped bearing residue, lint or sludge	
Improper oil temperature	
Improper oil chemistry	

Excessive Oil Strainer Differential Pressure

Symptoms: Oil strainer differential pressure indicates 2 psig above clean condition.

Action: Switch strainers and inspect strainer basket.

Probable Cause	*Remedy*
Plugged strainer	Identify source of foreign material and take appropriate action to correct situation.

Lubricating Oil Pump Shutdown Indication

Symptoms: Pump alarm and motor indicator lights.

Action: If pressure drops, start standby pump and stop shaft if pressure continues to fall. If pressure does not fall, determine cause for pump stopping.

Probable Cause	*Remedy*
Electrical power failure	In all cases check the electrical system for cause and correct.
Motor starter failure	
Indicator failure	
Motor overload	

Material in Lubricating Oil Strainer

Symptoms: High strainer differential pressure; material in strainer; babbit in strainer.

Action: Determine the type of material. If it is babbit, stop the shaft and check bearings. Ifit is not babbit, slow shaft and check pump strainer, check for correct lube oil pressure and temperature, check for vibration.

Probable Cause	*Remedy*
Dirt in system	Identify source of dirt and correct.
Bearing failure	Locate and replace bearing and clean the system.

Drop in Vacuum

Symptoms: Vacuum dropping; exhaust temperature rising; reduction in shaft speed; sentinel valve on LP turbine operates.

Action: Reduce speed to 60 percent for vacuum decrease to 20 inches or higher, reduce speed to 40 percent for vacuum decrease to between 10 and 20 inches, and secure the turbine when vacuum is less than 10 inches. Start the standby air removal equipment. Attempt to restore vacuum to within 1.5 inches of rated vacuum before resuming speed.

Probable Cause	*Remedy*
Air leakage	Look for leaks and correct.
Failure of air removal system	Start standby system, correct cause of failure.
Loss or reduction of cooling water flow	Start standby circulator, check condenser head for plugging, check sea chest for plugging, and use steam or compressed air connection to free chest.
Ice in sea chest or condenser head	Use heat to remove ice.
High condensate level in condenser	Check condensate pump and level regulator system, start standby system, find cause, and correct.
Loss of turbine seals	Shut down and replace the seals, check bearings and replace if necessary, check for rotor bowing.

Loss of Turbine Steam Seals

Symptoms: Steam seal regulator gauge indicates less than 1/2 psig; loss of condenser vacuum.

Action: Check to ensure that there is steam to the regulator. Use manual steam seal regulator valve if necessary to restore pressure to system.

Probable Cause	*Remedy*
Loss of steam supply pressure	Check the steam supply system, check valve lineup, check steam strainer in supply line.
Sticking regulator	Check sensing line, check supply steam valve, check the steam regulator.
Damage to turbine seals	Increase pressure to seals and repair seals at first opportunity.

Gland Seal Regulation

Symptoms: Steam blowing from shaft packing; hunting of regulator; low pressure in manifold of steam regulator system.

Action: Try to stop blowing by adjusting steam regulator pressure. Try to restore correct pressure.

Probable Cause	*Remedy*
Worn shaft packing	Replace packing.
Gland exhauster system not working properly	Check the steam seal regulator system for correct operation.
Hunting regulator caused by loose linkage or foreign material in regulator	Open regulator and check for loose parts, check adjustments and locknuts, clean the regulator, and lubricate parts.
Low pressure in manifold caused by pressure in gland exhauster system too low or incorrect operation of steam seal regulator	Increase pressure in gland exhauster system and check regulator completely by disassembly, cleaning, adjusting, and lubricating.

High Condenser Hot-Well Level

Symptoms: Condensate level alarm or gauge glass indication; reduction of condenser vacuum.

Action: Lower condensate level. Secure the turbine if unsuccessful in lowering level.

Probable Cause	*Remedy*
Failure of condensate pump	Start standby pump and determine cause of failure.
Condensate strainer plugged	Switch strainers and clean plugged strainer.
Level control failure	Adopt hand control and repair cause of regulator failure.
Condenser tubes leaking	Shut down and repair leaking tubes (modern boiler cannot operate on saltwater).
Condensate makeup valve needs adjustment	Adjust.
Air leak at condensate pump section	Repair leak.
High condensate temperature	Check vacuum, check circulating water.

High or Low Extraction Stage Pressure

Symptoms: A change in pressure from expected pressure at a given steady-state operating point.

Action: If gradual change, monitor pressure while looking for cause. If change is sudden and large, secure the turbine.

Probable Cause	*Remedy*
Buildup of deposits in turbine	Try operating at reduced speed with saturated steam, open turbine and wash out deposits if necessary.
Internal damage to turbine	Open and inspect turbine, repair damage if possible; if not single up turbine in emergency operating mode to reach port.
Off-standard operating conditions of power plant, such as heater out of service, damaged propeller, fouled bottom, instrument error, rough seas, and change in draft.	Correct off-standard condition.

Incorrect Steam Inlet Conditions

Symptoms: Instruments indicating pressure or temperature of inlet or reheat steam which is significantly different than rated conditions.

Action: Reduce speed and secure turbine. Restore the correct inlet conditions.

Probable Cause	*Remedy*
Malfunction of combustion control system	Operate manually until problem is corrected.
Instrument error	Check calibration of instruments, replace instruments if necessary.

Leaking Steam Valve

Symptoms: For steam inlet valves, high astern temperature when running ahead; turbine turning with steam valves closed; excessive corrosion inside turbine.

Action: Decrease turbine speed to below 60 percent and search for cause.

Probable Cause	*Remedy*
Valves not closing	Close the valves and reset the handwheel stops.
Valves not adjusted correctly	Readjust the settings of valves.
Steam cutting of valve seat	Repair or replace the valve seats and/or disks.
Foreign material on valve seat	Clean valves and repair seats.
Steam leaking back through drains	Repair drain valves.

Ahead and Astern Valves Malfunctioning

Symptoms: Valves do not respond correctly to control system; valve stems binding in bushings; operating cylinders not responding properly; control input devices not functioning properly.

Action: Control steam flow with main steam line valve if condition warrants. Check the valve operation at local control station to determine if problem is in an external system or in the local turbine control system. Attempt to free valves by slowly working valves open and closed.

Probable Cause	*Remedy*
Deposit of foreign material between stems and bushings, misalignment of lift rods in bar lift system	Disassemble and remove deposits, realign the lift rods.
Worn parts causing misalignment and binding	Replace and realign parts. Ifmotor drive, check electrical system for malfunction.

Drains Not Functioning Properly

Symptoms: Excessive vibration when increasing speed; deterioration of turbine performance; abnormal temperatures in drain lines; overheating of drain tanks.

Action: Slow down until vibration is at acceptable levels. Check to ensure that water is drained from turbine. Attempt to avoid large rapid changes in load until problem is corrected.

Probable Cause	*Remedy*
Improper lineup of drains, clogged drains or orifices, open traps, leaking check valves or shutoff valves	Identify and correct the drain system problem.

Unexpected Change in Propulsion Power

Symptoms: Speed change without control position change; change in engine sound; extraction and crossover pressure are unexpected for current steady-state operation.

Action: Check the boiler pressure, condenser vacuum, extraction valve lineup, tachometer, reduction gears and thrust bearing, relationship between the control setting and valve position, extraction and crossover pressures. Check if ship operating conditions have changed, i.e., sea conditions, shallow water, wind, propeller damage, stern tube bearing.

Probable Cause	*Remedy*
Cause will be identified during the above actions	Select remedy for the determined cause.

Turbine Vibration

Symptoms: Vibration above three mils double amplitude; a 25 percent increase in vibration above level normally expected at operating condition.

Action: Slow down to reduce vibration below 3 mils.

Probable Cause	*Remedy*
Bowed rotor	Operate at reduced speed until the shaft straightens.
Rotor unbalance	Run at reduced speed until the turbine can be inspected and repaired.
Lack of pinch fit in the baIrseated journal bearings	Install a temporary .002-inch shim between ball and seat at centerline until bearing can be corrected by machining bearing cap flange.
Loss of flexibility in coupling teeth due to damage or foreign material	Inspect and repair coupling.
Damaged bearing	Inspect and replace.
Water in turbine	Check and correct drain system.
Water carryover for boiler	Correct boiler operation.
Improper alignment of turbine and gear	Check and correct alignment.
Steam leaking into astern element during ahead operation	Correct leaking valve.
Turbine vibrating due to external exciting force	Locate and correct external vibration source.

Unusual Noise

Symptoms: Unusual turbine noise.

Action: Slow down and try to determine the noise source. Check the following: vibration of rotating or stationary parts, structural vibration, oil pressures, oil temperatures, rotor position indicator and sight flows.

Probable Cause	*Remedy*
Rubbing of rotating parts	Determine cause and make corrections as needed.
Vibration of structural nature	
Loose parts	
Bowed rotor	

Babbit in Strainer

Symptoms: Babbit found in strainer during inspection.

Action: Reduce speed to one-half or less. Stop shaft if oil flow or temperature is abnormal, if vibration is abnormal, if the rotor position indicator shows abnormal position.

Probable Cause	*Remedy*
High temperature in bearing	Take appropriate action based on investigation.
Heavy vibration	
Loss of oil	
Bearing misalignment	

Loss of Electric Power

Symptoms: Loss of electric power.

Action: Slow turbine. Check oil pressure and monitoring devices, and restore power.

Abnormal Turbine Rotor Position Indication

Symptoms: Increase of turbine rotor position reading overtime .010 for HP rotor and .020 for LP rotor.

Action: Check temperatures, vibration levels, and noise for unusual conditions. Slow down turbine if unusual condition exists.

Probable Cause	*Remedy*
Loss of thrust bearing	Inspect and replace bearing.
Damaged position indicator	Repair or replace the indicator.

TURBINE-GENERATOR SETS

Although some steam plants have been designed with an electric power generator driven by a takeoff from the main propulsion turbines with a diesel-generator set for port operation, it is most common to have two or more steam turbine-generator sets installed in steam power plants. In some cases steam turbine-generator sets are installed on ships with diesel main propulsion where they efficiently utilize steam generated from the energy of high temperature engine exhaust gases.

Typical turbine-generator sets are manufactured in ratings of 500 to 2,000 kW. The steam conditions are usually the same as the main propulsion steam conditions although a variety of steam power plant cycles have been designed to use extraction steam from the main turbine. Turbine speed for a turbine-generator set is typically in the range of 10,000 rpm driving a generator at 1,800 or 3,600 rpm through a single reduction gear.

Figure 6-29 shows a cross section of a typical seven-stage impulse turbine for electric generator drive. Flow to the first stage is controlled by a bar lift nozzle control arrangement moved by a hydraulic operator. The hydraulic operator pilot valve is controlled by a Woodward governor.

The second and third stages have partial arc admission while the following stages have full arc admission. The drive coupling is on the high pressure end of the turbine. The exhaust is downward. Operation and maintenance concerns are similar to a propulsion steam turbine.

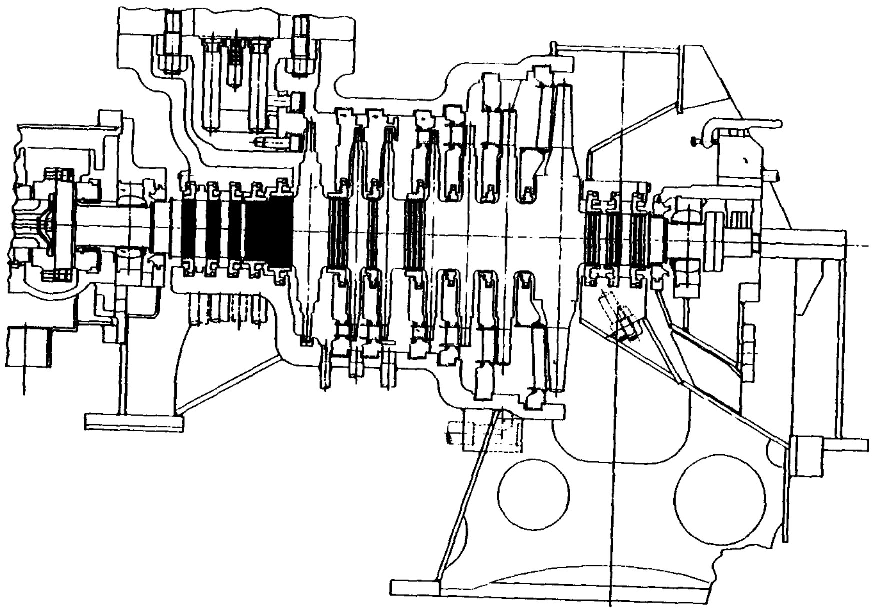


Figure 6-29. Seven-stage impulse turbine for electic power generation. Courtesy General Electric Company.

REVIEW

1. What are the common ways of classifying propulsion steam turbines?
2. What is a reaction stage?
3. What is an impulse stage?
4. What is the advantage of a reheat turbine?
5. What is the ideal process that Occurs in a nozzle?
6. Why do we need the continuity concept to find nozzle areas?
7. What is sonic velocity? What happens at sonic velocity in a nozzle?
8. At what pressure ratio does sonic flow occur for superheated steam?
9. What happens in an impulse turbine stage?
10. What is a Curtis stage?
11. Draw a graph showing the variation of steam velocity and steam pressure in a four-stage impulse turbine.
12. What is whirl velocity?
13. What are the three efficiencies that determine stage efficiency?

14. What is pressure compounding?
15. Why does a reaction turbine have an axial thrust on the rotor? Does an impulse turbine have an axial thrust? Why?
16. What is a Willans line?
17. What is isentropic expansion? What do you need to calculate an isentropic enthalpy drop?
18. What are the common units for turbine steam rate?
19. What are the two methods of controlling steam flow to a propulsion turbine?
20. What are the major considerations in the design of a turbine bucket?
21. What are the three forces included in the mathematical model of a mechanical vibration system?
22. What is the typical range of steam conditions for modern propulsion turbines?
23. Where are astern turbines commonly located? What are the other possible locations?
24. What material is used in the manufacture of turbine rotors?
25. What is the function of the holes in impulse turbine disks?
26. How are buckets assembled to impulse turbine rotor disks?
27. Why are Curtis stages used commonly in astern turbines?
28. How are diaphragms supported in an impulse turbine? Why?
29. What are crush pins?
30. Describe the operation of a steam seal regulator.
31. What control functions are usually incorporated into an electrohydraulic propulsion control system?
32. Describe the procedure to prepare a propulsion turbine for getting under way. Describe tloleprocedure for securing a propulsion turbine.
33. What action should the watch stander take when vibration is detected in the HP turbine?
34. What action should be taken to correct a bowed turbine rotor?
35. What causes babbit in a lube oil strainer?
36. What is the cause and correction of abnormally high crossover pressure?
37. Why is high condensate level in the condenser of an axial exhaust LP turbine a serious problem requiring prompt action?

REFERENCES

Budd, William I. H. 1971. *Marine Engineering,* chapter 5, "Steam Turbines." Jersey City, N.J.: Society of Naval Architects and Marine Engineers (SNAME).

Fox, W. J., and S. C. Birnie. 1975. *Marine Steam Engines and Turbines,* 3rd ed. London: Butterworth & Company.

General Electric Company. 1975. *Trouble Analysis Guide.* GEK-19371. Fitchburg, Mass.: General Electric Company.

Richardson, P. W. December 1976. "Method for Predicting the Performance of Marine Propulsion Turbine-gear Sets." Lynn, Mass.: General Electric Company.

Salisbury, J. Kenneth. 1950. *Steam Turbines and Their Cycles.* New York: John Wiley & Sons.

Spears, H. C. K., and E. C. Rohde. 1968. *Steam Propulsion Systems for Modern Ships.* STAR Symposium, Jersey City, N.J.: SNAME.

United States Government. January 1976. "High Performance Reheat Marine Steam Propulsion Plant." Maritime Administration Report No. MA-RD-920-76033..

United States Navy. 1970. *Principles of Naval Engineering.* Washington, D.C.: Bureau of Naval Personnel, U.S. Navy.

ACKNOWLEDGMENTS

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CHAPTER 7

Gas Turbines

PAUL A. DUPUY

INTRODUCTION

A turbine is a bladed wheel that absorbs kinetic energy from a fluid stream. A gas turbine is a device that expands a compressed gas through nozzles (usually in the form of vanes) changing pressure to velocity and directing the gas into the turbine where, through impulse and reaction, the turbine blades convert the energy to useful rotational work. So far, the gas turbine and the steam turbine are quite alike and their outputs similarly are at high rotational speeds (rpm). As discussed later, gas turbine cycles could use an external heat source, as the steam turbine uses a boiler, or they could use an internal combustion device, in which case, the gas turbine engine is a compact unit with compressor, combustor, and turbine. It has become common for gas turbine engines to be referred to simply as gas turbines and this terminology will be used in this chapter.

Marine Historical Background

The first vessel to be propelled at sea by a gas turbine was a British motor gunboat in 1947. Following this installation, the British Navy continued with several other patrol boat gas turbine installations during the 1950s. The 1950s also saw two commercial gas turbine installations on the British tanker, *Auris,* and the U.S. Liberty ship, *John Sergeant.* The *Auris* AEI 5,500 hp gas turbine plant operated satisfactorily for over four years and was withdrawn from service because 12,000-ton tankers had become uneconomical. The *John Sergeant* was converted to gas turbine power under a U.S. Maritime Administration (MARAD) development program. It was powered by a 6,000 hp General Electric (GE) industrial-type recuperated gas turbine. The gas turbine powered *John Sergeant* operated satisfactorily at sea for 9,270 hours before being removed from service in 1959 due to termination of the MARAD development program.

During the 1960s, the navies of the United States, the United Kingdom, and Canada, as well as the U.s. Coast Guard, utilized aeroderivative gas turbines for high-speed (boost power) operation in patrol boats, frigates, and destroyers. Compared to previous systems, the gas turbine required fewer and simpler support systems, adapted to propulsion system automation, and afforded reduction in manning. In the 1960s, several classes of gas turbine powered commercial ferries were built, one of which, the British Hovercraft Corp SR.N4, carried 282 passengers and 37 cars at cruising speeds of 40 to 60 knots. Five of the SR.N4s were still in service in 1993.

In the 1970s, there were fourteen GE industrial gas turbine powered vessels, including tankers, bulk carriers, Ro/Ro ships, and LNG carriers. Of these, five Chevron Oil Company product tankers with electric drive were still operational in 1994. In the 1970s, aeroderivative gas turbines were used to power Ro/Ro ships, containerships, and high-speed passenger-car ferries. Notable among these were the *Adm. Wm. M. Callaghan,* a 25,000-tonne displacement Ro/Ro ship used by the U.S. Navy to evaluate gas turbine models for naval ships, and the four Seatrain Lines Euroliner class containerships designed for 26.5-knot Atlantic transits. The Euroliner used two 30,000 hp class Turbo Power and Marine FT4 engines, which powered the propellers, and a 1,000 kW ship service generator driven off each main reduction gear. Since power system frequency would vary with shaft speed, this was compensated for by series connected thyristor-type converters. These ships pioneered the use of blended fuel in aeroderivative engines. In 1979, after nearly ten years' service, it was decided to reduce ship speed to a nominal 19 knots due to fuel costs, and diesel engines were retrofitted. The Finnjet, powered by two FT4 gas turbines, began service in the 1970s and is operational today. This 32-knot high-speed ferry epitomizes the value oflightweight gas turbines for maintaining reliable sailing schedules. For periodic shoreside engine maintenance, these gas turbines can be changed out and replaced in as little as two or three hours. A significant number of commuter hydrofoils and hovercraft were built with typical per-unit power from 1,000 to 8,000 hp. Some of these are still in operation. By today's standards, these turbines had low thermal efficiency (about 25 percent) and low specific power, requiring large inlet and exhaust ducts.

During the 1970s, using the advances of aircraft gas turbine technology combined with the lessons learned from over a decade of shipboard operation in a range of applications, the second generation aeroderivative gas turbine was introduced into naval service. Using internally air-cooled turbine components and higher gas temperatures combined with higher pressures and continuous upgrading, the following general improvements were achieved for engines in the 15,000 to 30,000 range:

Year	*1960*	*1990*
Efficiency (max rating)	26	37
Combustor discharge temperature (F)	1,600	2,375
Compressor pressure ratio	12	18-28
Compressor stages	17	10-16
Specific power (hp/lb/sec)	100	200
Bare engine specific weight (lbs/hp)	0.5	0.36

The U.S. Navy's commitment in 1970 to a thirty-ship destroyer class using all gas turbine power (four propulsion and three electric generator drive gas turbines per ship) was a major factor in the widespread usage of marine gas turbines. Similarly, military services' use of gas turbines for special purpose craft, such as landing craft, have placed many hundreds of 3,000 to 8,000 hp class gas turbines at sea.

From 1975 to 1994, there have been twenty-three world navies utilizing about 750 second-generation gas turbines in over 300 ships from below 300-ton patrol craft to over 50,000-ton displacement supply ships ranging in power from 5,000 hp to 100,000 hp. During that time period, there have been several thousand gas turbines from 3,000 to 30,000 hp utilized on offshore platforms and special marine craft where weight savings are paramount.

An explanation is in order as to why gas turbines have been so ubiquitous in naval ships and so little used in commercial ships. The largest factor is fuel cost. While steam plants and low and medium rpm diesel engines and some industrial gas turbines operate using residual fuel or a blend thereof, the higher efficiency second-generation gas turbines require a distillate fuel. (See discussion in chapter 8.) The two-year (1992-1993) world bunker fuel price for intermediate fuel oil (lFO) 180 fuel was $76 per tonne; for marine diesel oil (MDO), it was $190 per tonne or about $115 per tonne premium for MDO. This differential is sensitive to ports of call. Since the U.S. Navy and many other navies have adopted a single grade offuel for all their ships' engines and boilers, in essence a mil-spec marine gas oil (MGO), this price differential is not a consideration for navy applications. However, during the lives of their ships, navies average perhaps 90 percent oftheir operations at below 50 percent oftheir flank speed or at powers less than 20 percent of ship's power. Gas turbine efficiency reduces as power is reduced. Diesel engines, by comparison, maintain a relatively constant efficiency across their range. Thus, for ships like patrol craft, corvettes, and frigates, the combination of high-speed diesel engines for cruise and loiter speeds and gas turbines for higher speed operation, is popular. The high-speed diesel fuel specifications for grade and constituents are similar to the current gas turbines. This arrangement is called CODOG, combined diesel or *gas* turbine. For larger ships (destroyers, cruisers, many with four gas turbines) 20 percent ship power (58 percent speed) is available at an 80 percent power rating on one gas turbine, thus maintaining efficiency across the ship's speed range.

It is recognized that since the world oil price crisis of 1973, the trend in many forms of shipping has been to reduce speed and power to save fuel costs. However, a growing number of specialized markets of marine transport are emerging where high speed and compact, lightweight power plants are required. Two examples of the high-speed ferry market are the Aquastrada and the Stena HSS (high-speed sea service) classes of ferries.

The Aquastrada 100-meter monohull class ships are built by Rodriquez Cantieri Navali in Italy. The machinery layout is shown in figure 7-1. One gas turbine and two diesls each drive separate water jets. Because they can all be operated independently or simultaneously, the propulsion system is CODAG, combined diesel *and gas* turbine. The vessels, operating at 40 knots between the west coast ofltaly and Sardinia, reduce passage time to nearly half that of conventional ferries. Typically, diesel propulsion is used for in-harbor maneuvering and for any operations where speeds up to 18 knots are required. The 25,000 hp gas turbine is used for open water transit which is typically over 90 percent ofthe time. All three engines can be used for unusual conditions of heavy loading and sea state. The Aquastrada class ships displace 930 tons, carry 400 to 500 passengers and 90 to 120 automobiles for a range of 400 to 700 nautical miles.

The Stena HSS is a large car-passenger, twin-hull catamaran ferry. It has a total of 80,000 horsepower using one nominal 25,000 hp and one 15,000 hp gas turbine in each hull. The gas turbine power plant packages and the propulsion components integration are supplied by Kvaerner

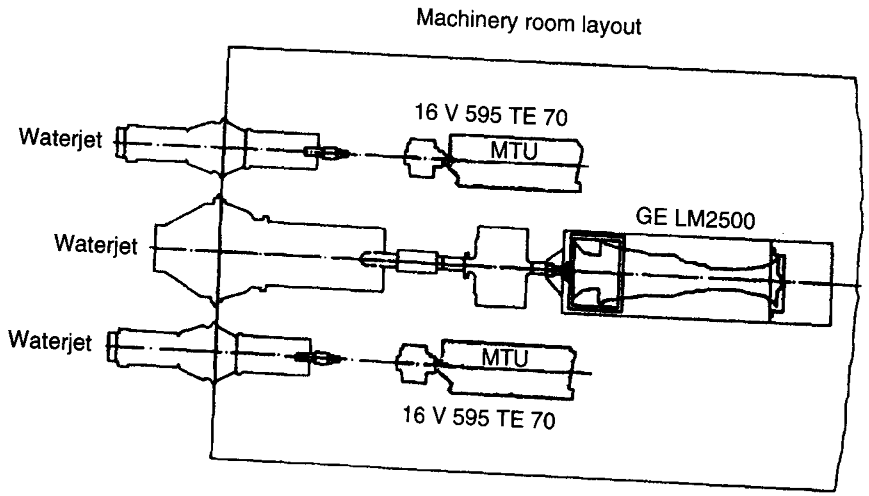


Figure 7-1. Aquastrada car-passenger ferry propulsion arrangement. Courtesy General Electric Company.

Energy of Norway for ships being built in Finland for Stena, a Swedish-based ferry operator, for potential routes worldwide. These ships will have 1,500 tons loading capacity and will carry 1,570 passengers and 375 cars, or a combination of 100 cars and fifty trucks or equivalent, at a service speed of 40 knots in five-meter wave heights.

Figure 7-2 shows the machinery layout for one hull. This arrangement was chosen to fit the width constraint of the hull. Since any of the four gas turbines can operate independently or in combination, this machinery arrangement is COGAG, combined *gas* turbine *and gas* turbine. This ship is designed for a diversity of routes, locations, seasons, and sea states. Typically, 20,000 hp would provide 25 knots, 50,000 hp would provide 32 knots, and 80,000 hp would provide over 40 knots with the gas turbines operating at efficient power levels.

An example of 1990s gas turbine application for cargo operations is the Sealift class ships. The propulsion transmission system of these ships is shown in chapter 9. These 62,400 full load ton displacement ships provide roll-onlroll-off transportation support for military overseas operation by the U.S. Military Sealift Command. They have typical Panamax dimensions with at least 380,000 square feet of cargo-carrying deck space and they can maintain at least 24 knots for a 12,000-nautical-mile mission. For the gas turbine to be selected, a life cycle cost analysis was required and showed that the sum of consumables costs, manning requirements, maintenance, and investment costs for commercial ship missions were competitive with other types of propulsion prime mover systems.

An application of gas turbines being studied for commercial ships is cruise liners, where the characteristics oflow installed weight and volume, high availability, low first cost, low manning and maintenance requirements, and low exhaust emissions can be attractive. By utilizing gas turbine exhaust heat recovery boilers, steam can be generated to provide for the large hotel requirements of a passenger ship such as electric service, heating, cooling, laundry, water heating, and evaporators for freshwater replenishment. This type system is COGAS (combined *gas* turbine *and* steam) and is discussed in the section on gas turbine principles. The COGAS cycle gives the aeroderivative gas turbine plant a ship's fuel consumption rate quite comparable with current day medium-speed diesel systems. Since these ships tend to be volume limited, the higher power density of a COGAS system makes enough revenue enhancing space available to more than offset its higher fuel costs.

It appears the unit fuel cost disparity between the gas turbine and diesel plants could narrow significantly due to the effect of environmental regulations concerning sulfur limits. Current IFO fuels may have up to 5 percent sulfur by specification and average about 3 percent. Distillate fuel may have 1.5 percent by specification and average less than 0.5 percent. Estimates of the increase in unit cost of heavy blended fuel provided to the

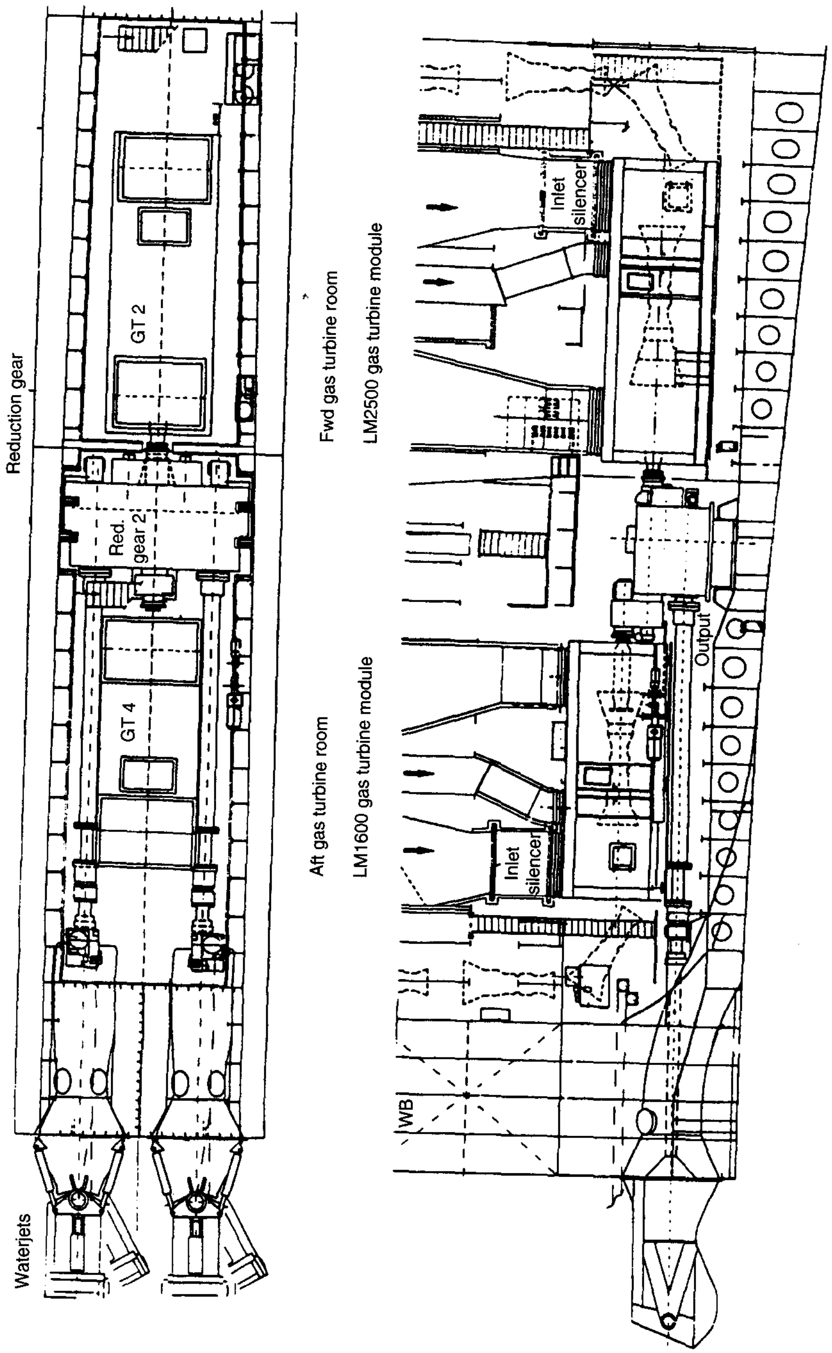


Figure 7-2. Stena HSS machinery layout (one of two hulls). Courtesy General Electric Company.

International Maritime Organization (IMO) by the Oil Companies International Marine Forum (OCIMF) in 1992 indicate that if the sulfur limit is lowered to a 1.5 percent range, blended residual fuel cost would increase by 53 to 76 U.S. dollars per tonne, and by 67 to 95 dollars per tonne if it is lowered to 1.0 percent. In 1994, the IMO and several localized governmental bodies are currently considering such limits. This magnitude of increase in unit cost would drastically reduce the cost difference between heavy and distillate fuels in most areas of the world. Similarly, emerging environmental regulations on the limits of oxides of nitrogen, NO_x, may also militate toward use of distillate fuel in controlled areas since blended heavy fuels can be high in fuel bound nitrogen (FBN) which produces organic NO_x. Heavy fuel organic NO_x is additional to the thermal NO_x, the reaction of nitrogen and oxygen in the combustion air at high temperature.

From the mid-1950s to the mid-1970s, limited numbers of gas turbines were tried in a broad range of marine services. From the mid-1970s to the mid-1990s, a new order of higher technology engines was introduced, offering better efficiency and increasing operating time between repairs; thousands of gas turbines have accumulated hundreds of millions of operating hours during a score of years in marine applications. Improvements based upon service experience and constantly evolving technology have made it possible to increase power capability as much as 50 percent for some models since their inception while retaining or improving maintenance intervals. Because marine gas turbines have industrial counterparts, many of the issues evolving for the marine industry such as reduction of pollutants are already being answered by these engines in their shoreside applications. As the changes have been rapid to date, there will be many further advances to what is described at the time of this writing.

In the brief history of gas turbine propulsion of ships previously mentioned, early ship applications were divided between the industrial gas turbines used in large commercial oceangoing ships and the compact and lightweight aeroderivative gas turbines used for naval craft, high-speed transport ships, and a few oceangoing freighters. In recent years, the technology of the two classes of gas turbines has become more similar. In fact, where there is consideration of variations to the simple cycle as discussed in this chapter under "Principles of Gas Turbines," the aeroderivative becomes a mix of aero and lightweight designs of industrial enhancement components such as recuperators, intercoolers, and heat recovery boilers.

While both classes of the two basic gas turbine approaches have prospered in ocean offshore plant usage, the aeroderivative, for the time, has dominated shipboard usage. Therefore, the text will be based predominantly upon large aeroderivative engines, where weight and volume differences are substantial, and an intermix of aero and industrial small gas turbines, where the differences are less dramatic. Explanations will be made where the industrial machines have different characteristics of concern to the ship operator.

PRINCIPLES OF GAS TURBINES

Gas turbine engines have been designed for closed and open cycles. Closed cycles may operate using any of various gases, continuously circulating the gas through the engine and then ducting the exhaust through a cooler and back to the compressor inlet. The following discussions apply to the open cycle, which draws air from the atmosphere and discharges back into the atmosphere. Where a combustor is mentioned in the open cycle, this would be replaced by a heat exchanger as the closed cycle, by definition, would not replace oxygen required for combustion. A closed cycle heat exchanger might use a nuclear or an oxygen-hydrogen catalytic reaction chamber heat source.

Simple Cycle

Figure 7-3 shows a basic gas turbine jet engine and the cycle events of the Brayton cycle on pressure-volume and pressure-temperature diagrams. The air is compressed isentropically (at constant entropy) and is heated by burning fuel at constant pressure. The gas is expanded isentropically through the turbine. At this point in the cycle, residual gas energy can provide jet thrust through the exhaust nozzle or additional stages of turbine could be used and a shaft extended from the turbine shaft to drive a load.

Figure 7-4 shows the interrelationships, for typical large (over 10,000 hp) simple-cycle gas turbines, of efficiency and specific power (brake horsepower per unit of mass flow) as a function of gas generator turbine entry temperature and compressor ratio. Such a plot is sensitive to the component efficiencies throughout the engine and typical current-day efficiencies were assumed.

In a ship, both efficiency and specific power are important. The greater the specific power (the power per weight of airflow per unit of time), the smaller the intake, exhaust, and cooling-air ducting that traverse from topside through all decks to the engine compartment. As will be seen, since the simple cycle offers the most autonomous and compact power unit (essential for aircraft), technology has striven to develop materials and hot section cooling methods to accommodate higher firing temperatures while matching appropriately higher compressor ratios. Note that at 2,000°F firing temperature and a compressor pressure ratio of 15, thermal efficiency is over 34 percent. If compressor ratio were increased to 26, efficiency would increase to 36 percent, but specific power would reduce from 170 to 135 bhp/lb-sec., a drop of about 20 percent, which would require a 26

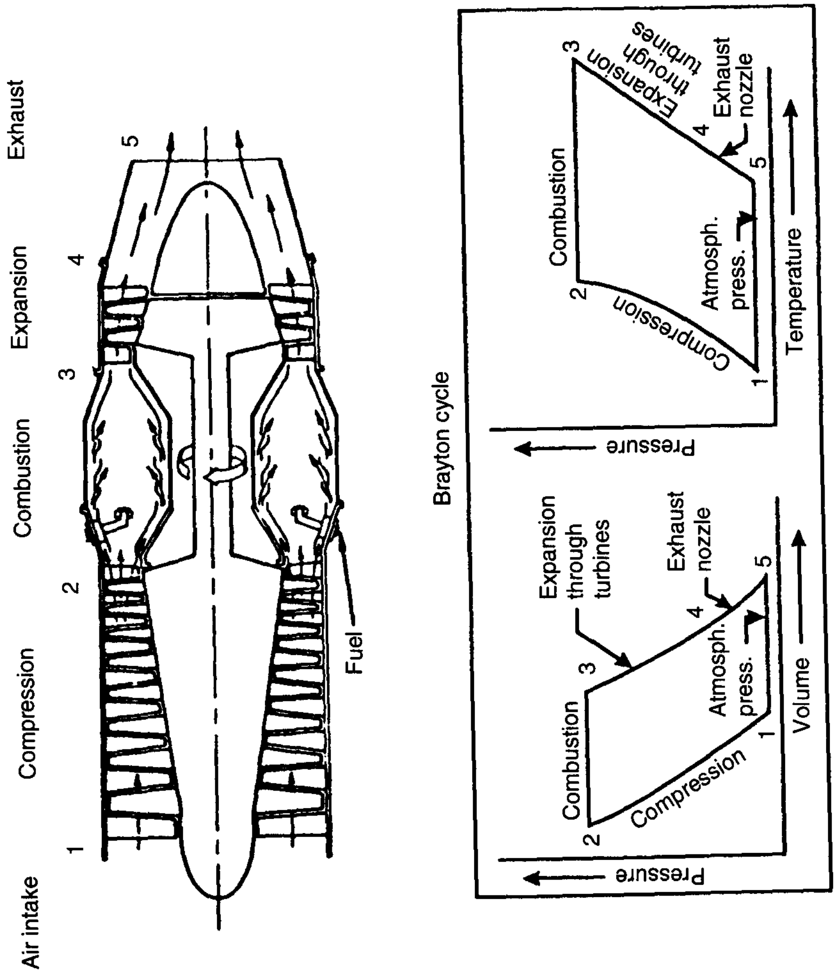


Figure 7-3. Cycle events of the gas turbine engine. Courtesy General Electric Company.

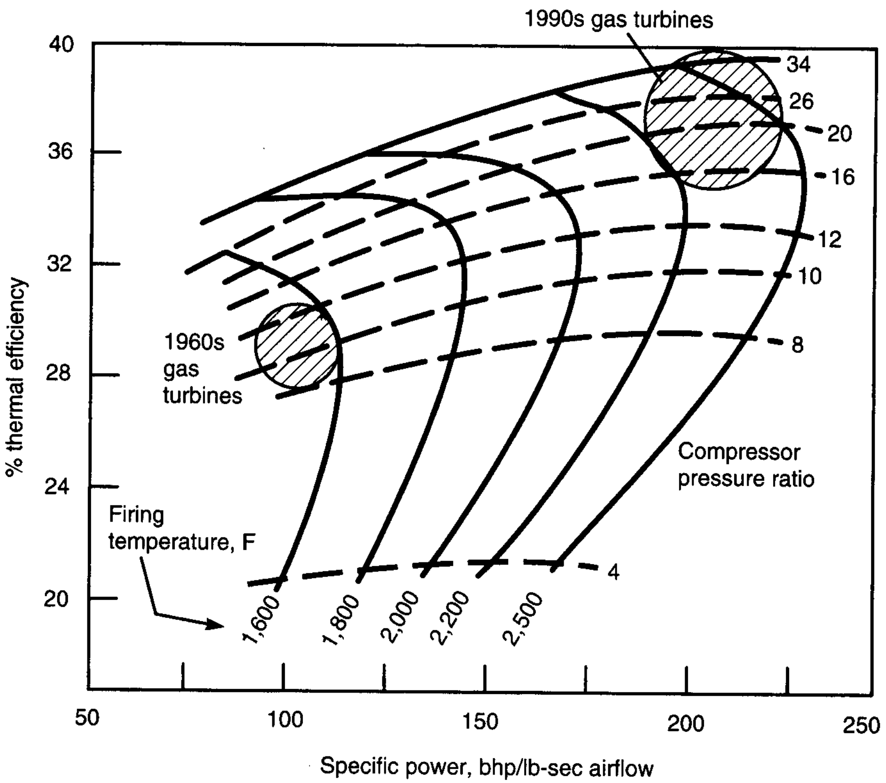


Figure 7-4. Characteristic simple-cycle specific power and efficiency. Courtesy General Electric Company.

percent increase in ducting cross-sectional area for comparable ducting velocities and losses.

Figure 7-5 shows the simple cycle for a single-shaft and a two-shaft, sometimes called split-shaft, gas turbine. The simple-cycle, single-shaft engine has a power shaft common to the compressor and turbine. This arrangement is sometimes used for an electric generator drive where the output speed is constant at any load and constant frequency control must be very precise during large load changes. With the single shaft, the compressor is at constant speed, regardless of load. This means the gas producer does not have to overcome the inertia to change rpm for a sudden load change, but it is less efficient than a two-shaft gas turbine at partial loads where compressor flow is in excess of cycle needs. Where generator frequency control requirements are more conventional, a two-shaft turbine (as is used for mechanical or electrical propulsion drives) would be used.

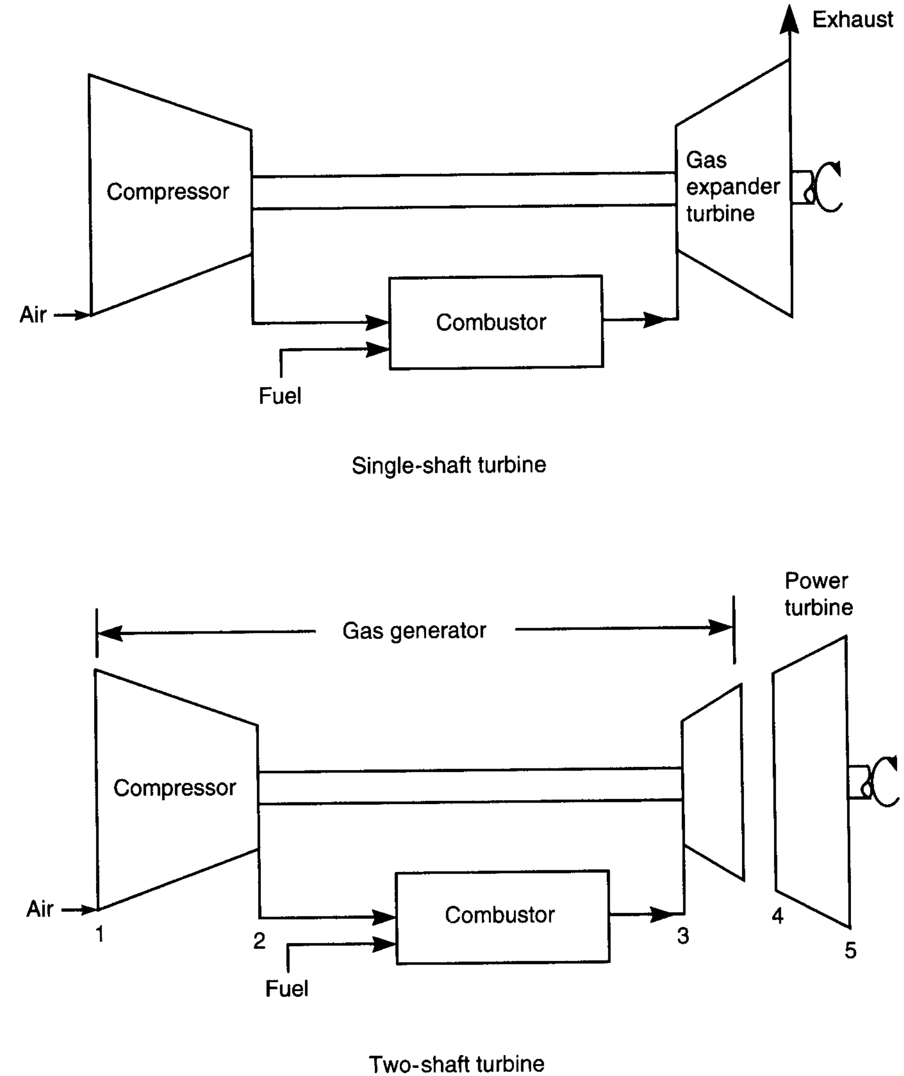


Figure 7-5. Simple cycle. Courtesy General Electric Company.

Comparative single-shaft and two-shaft performance is shown in the performance section.

With the two-shaft turbine, the turbine that drives the load is independent of the turbine that drives the compressor. This is sometimes referred to as aerodynamically coupled. The compressor-combustor-turbine assembly is usually referred to as the gas generator and the turbine that drives the load is commonly referred to variously as load turbine, free turbine, or power turbine. The gas generator may operate down to idle speed when low power is required at low output speed.

The idle gas-generator energy is usually low enough to windmill the power turbine at partial rpm when the power turbine is declutched from the load, and the power turbine may be braked to a stop while the gas generator idles. Similarly, the gas generator can be accelerated to a maximum power during acceleration to provide the drive train with maximum power turbine breakaway torque as might be required during ice-ramming and backing of an icebreaker. The two-shaft turbine offers the opportunity to design the power turbine tp operate at a lower rpm than the gas generator, which may reduce or eliminate the number of stages, the losses, volume, and weight of the speed-reducing gearing compared to a single-shaft machine.

While the Brayton cycle assumes ideal reversible adiabatic (isentropic) compression and expansion, there are actually losses through the cycle (component efficiencies) which do change the entropy. Figure 7-6 shows a typical enthalpy versus entropy curve for the two-shaft turbine with the

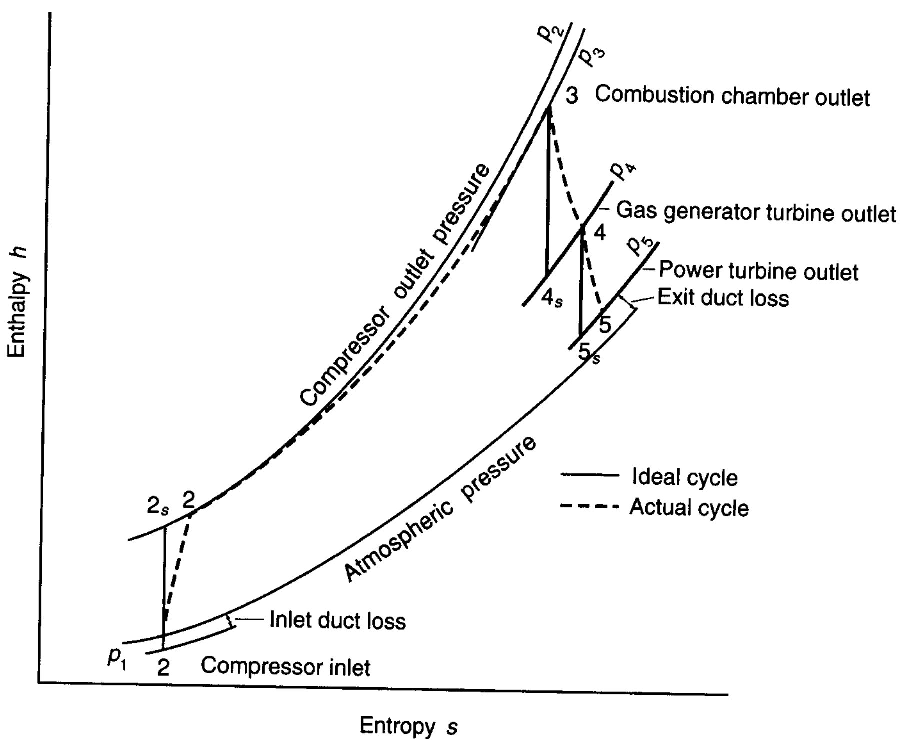


Figure 7-6. Two-shaft gas turbine cycle. Courtesy Generai Electric Company.

solid lines representing the ideal cycle and the dashed lines showing actual cycle for the cycle stations as numbered in figure 7-5.

Gas Turbine Cycle Variations

Figure 7-7 shows a regenerative cycle combined with a reheat cycle and a regenerative cycle using intercooling. Any ofthese cycle additions could be used independently or in combination.

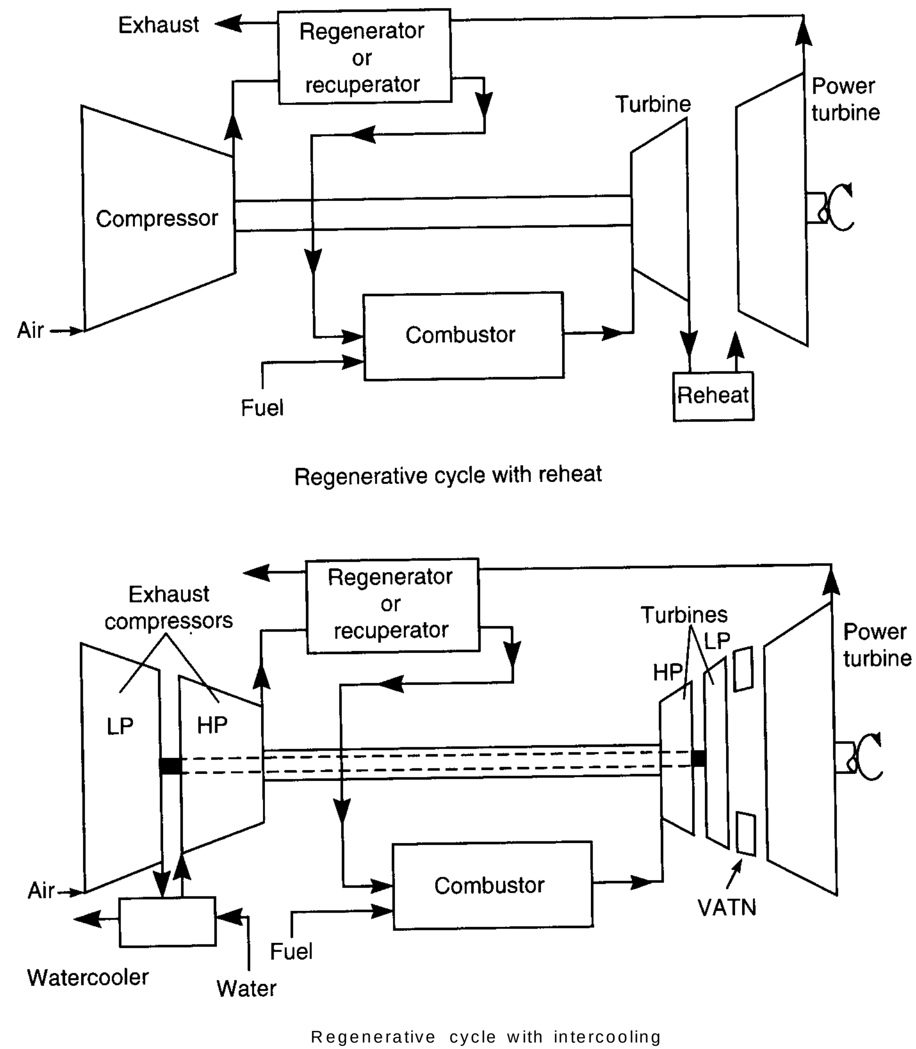


Figure 7-7. Gas turbine cycle variations. Courtesy General Electric Company.

REGENERATIVE OR RECUPERATIVE CYCLE

In a regenerative or recuperative cycle, the heat of the exhaust gas is used to heat the compressor discharge air before entering the combustor. Given a specific turbine entry temperature, every degree which the compressor discharge air can be raised is a degree which does not have to be added by the combustor fuel. A regenerator is a heat exchanger. More precisely, a regenerator either has a matrix rotating between the hot exhaust and the cooler compressor discharge air or it uses two fixed matrices where the gases are alternately switched between the matrices.

A recuperator is a heat exchanger with static surfaces, with the power turbine exhaust on one side and compressor discharge air on the other side of the surfaces. As gas turbine technology advanced and compressor discharge pressures advanced from 4 or 5 atmospheres in the 1940s to 15 to 35 atmospheres today, sealing problems with the rotary regenerators became more difficult and recuperators became more typical for shipboard size prime movers. Therefore, the regenerative cycle today is frequently called the recuperative cycle. As compressor discharge pressure increased, so did the compressor discharge air temperature. Coupled with greater temperature drop across modern turbines, the difference between power turbine exhaust gas temperature and compressor discharge temperature has decreased, which increases the size and reduces the effectiveness of regeneration/recuperation.

Recuperation can be combined with variable area power turbine nozzles (VATN) to improve modern turbine part-load performance. For naval ships' predominantly low-power duty cycle, this is attractive. The VATN keeps the power turbine inlet and exhaust temperatures higher at part load when compressor pressure and flow are lower making a recuperator very effective at low power requirements.

REHEAT CYCLE

Reheat is the addition of heat at constant pressure by means of a combustor between the gas generator turbine and the power turbine. By boosting the gas temperature, more output power can be obtained, but with a decrease in cycle efficiency. There is sufficient oxygen to accommodate the reheat burner because the primary compressor flow is three or more times that required for primary combustion. With reheat, a recuperator would be more effective. Reheat is not currently being considered for ship propulsion.

INTERCOOLING

Gas turbine compressors are typically constant volume air pumps for a given rpm. Compressor work is proportional to the absolute temperature of the air. As air is compressed adiabatically over a large pressure ratio, there is a large temperature rise. For example, for an 18:1 pressure rise, the air temperature increases about 800°F. By cooling the air at some intermediate point in the compression sequence, air volume and compressor work is decreased after the intercooler for the same final pressure ratio. This means the compression downstream of the intercooler can be done by a smaller compressor requiring less turbine work, or the same size compressor downstream of the intercooler can accept more air flow from a larger compressor upstream ofthe intercooler, which would increase the turbine output power capability. By itself, intercooling does not improve cycle efficiency because the compressor discharge air, being cooler than the simple cycle, requires more fuel for the same turbine entry temperature. However, with the lower compressor discharge temperature, there is a greater temperature difference between turbine exhaust and compressor discharge temperature, so that recuperation with intercooling effectively increases cycle efficiency throughout the power spectrum. When intercoolers are used, recuperation is usually included. When an intercooled-recuperated OCR) engine must operate with the intercooler out of commission, the engine power capability may drop to as low as 30 percent of its normal rated power.

The ICR sketch (fig. 7-7) also introduces a variation in the mechanical arrangement ofthe gas turbine components, that is, the two-spool or dual-spool gas generator. This arrangement is widely used with simple-cycle engines and is nearly essential when using intercooling. The gas generator is composed of two separate rotors, each driven by its own turbine. The low-pressure (LP) compressor rotor is driven through the high-pressure (HP) compressor and turbine rotors with a concentric shaft from the LP turbine rotor. The two-compressor turbine rotor shafts may be designed for different relative speeds; each operates at its optimum speed at different steady-state and transient speed conditions and each increases compressor stall margins.

While the line sketches in figure 7-7 might convey that various performance characteristics can be changed just by adding components to a given simple-cycle engine, incorporating any changes to an engine's cycle actually requires considerable redesign of the rest of the engine. For example, collecting gases and diverting them from the compressor into the heat exchangers and then out and back into the engines requires new or significant redesign of many of the engine components.

ICR CYCLE

The United States Navy has been involved in the development of an ICR marine gas turbine. In 1991, Westinghouse Electric Corporation was awarded the contract for the advanced development of the engine designated the WR-21. Westinghouse Marine Division is the primary contractor and system integrator, Rolls Royce is the RB211-derived engine developer, Allied Signal is the recuperator and intercooler developer, and CAE is the electronics-control developer. References noted at the end of this chapter explain this program in detail. Figure 7-8a shows the WR-21 ICR gas turbine engine; figure 7-8b shows the schematic diagram of the system.

The WR-21 ICR engine is a two-spool gas generator with free power turbine. It is based on the Rolls-Royce RB211 three-spool commercial aircraft engine family. It utilizes a dual-loop freshwater/seawater intercooler system, a plate fin recuperator, and a digital control system. The engine modules consist of the low-pressure compressor (LPC), intercooler, high-pressure compressor (HPC), combustor assembly, high-pressure turbine (HPT), low-pressure turbine (LPT), power turbine (PT), and recuperator. Air entering through the-ship's intake is directed through the LP compressor into the intercooler, where heat is extracted by way of the intercooler system. The high-density, low-temperature air is then more efficiently compressed by the HP compressor and in normal operation passes through the recuperator, where it is heated by the exhaust gas. The warmer air leaving the recuperator is then mixed with fuel in the combustor, where the mixture is burned. From the combustor, hot gas enters the HP and LP turbines, which extract work to drive the HP and LP compressors respectively. Gas leaving the LP turbine enters the variable area nozzle (VAN) and then the power turbine, which extracts work to drive the ship. After exiting the power turbine, exhaust gas enters the recuperator, which returns heat back into the cycle to improve specific fuel consumption (SFC) characteristics. The function of the VAN is to maintain the gas at a high temperature particularly at part power, thereby transferring more heat to the exiting HP compressor air resulting in improved SFC in the part power range.

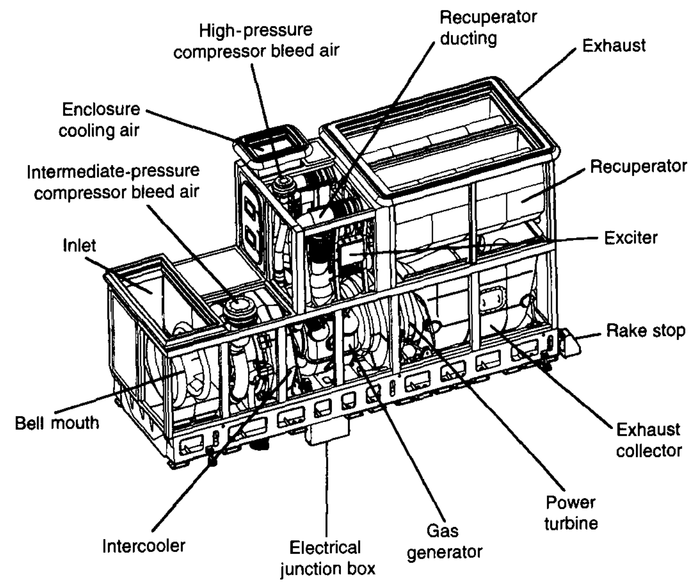
The U.S. Navy specifications for this program stipulate that the WR-21 enclosure base dimensions (L x W x H) be 315 x 104 x 190 inches, that the ICR system weight be limited to 120,000 pounds, and the maximum specific fuel consumption versus power, for U.S. Navy rating conditions defined subsequently, be as follows:

Power (hp)	880	1,320	2,640	5,280	7,920	13,200	19,800	26,400
Max SFC (lbs/hp-hr)	0.865	0.664	0.530	0.425	0.378	0.342	0.339	0.360

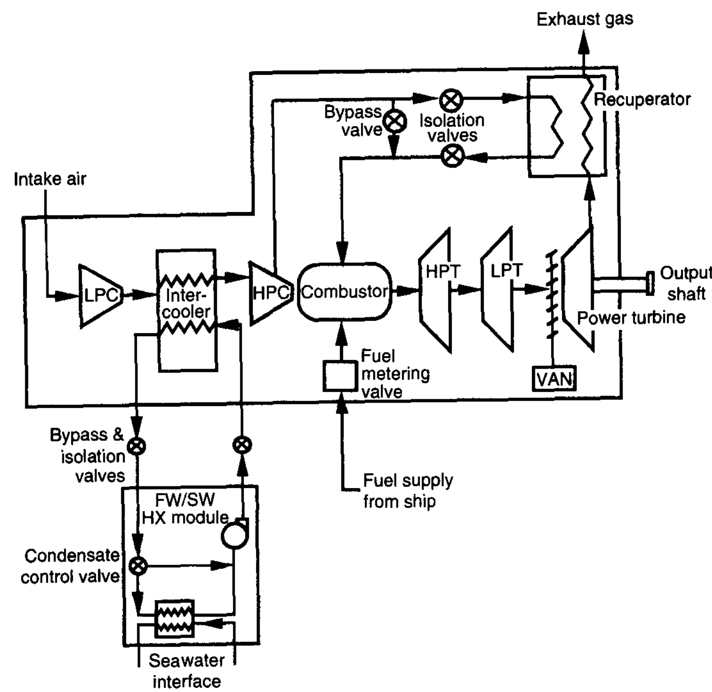
COMBINED GAS STEAM CYCLE

Figure 7-9 shows a diagram of a combined *gas* turbine *and* steam (COGAS) system. At full power, today's simple-cycle marine gas turbines dissipate their energy or heat intake about 35 percent to the output shaft and about 55 percent in their exhaust gases. The rest goes as losses to lube oil and cooling air. At 20 percent power, the output split is about 22 percent to the output shaft and 68 percent to the exhaust.

The uniform exhaust gas temperature and clean exhaust products that result from using distillate fuel are ideal for heat recover boilers. In figure 7-9, the boiler steam is applied to a propulsion steam turbine coupled to the



(a)



(b) WR-21 schematic diagram

Figure 7-8. Westinghouse WR-21 ICR gas turbine engine. Courtesy Westinghouse Electric Corporation.

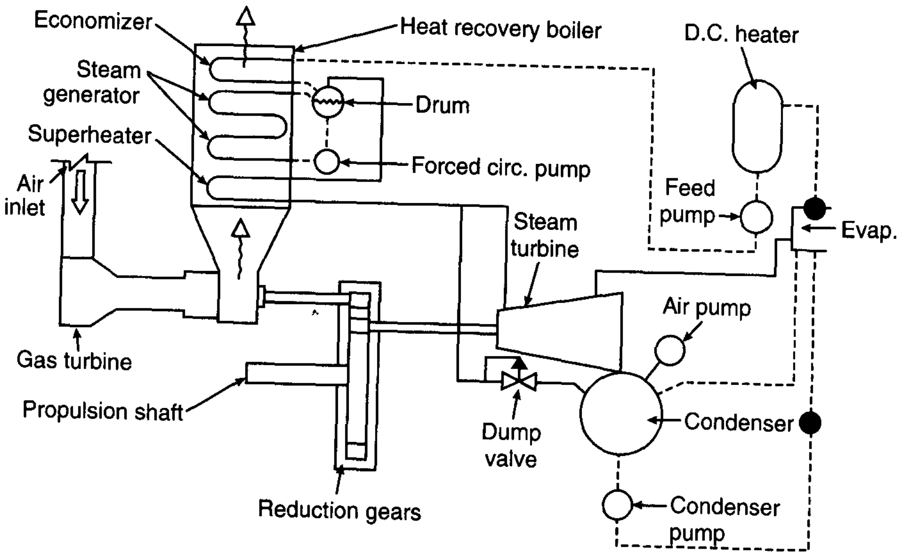


Figure 7-9. COGAS power plant cycle

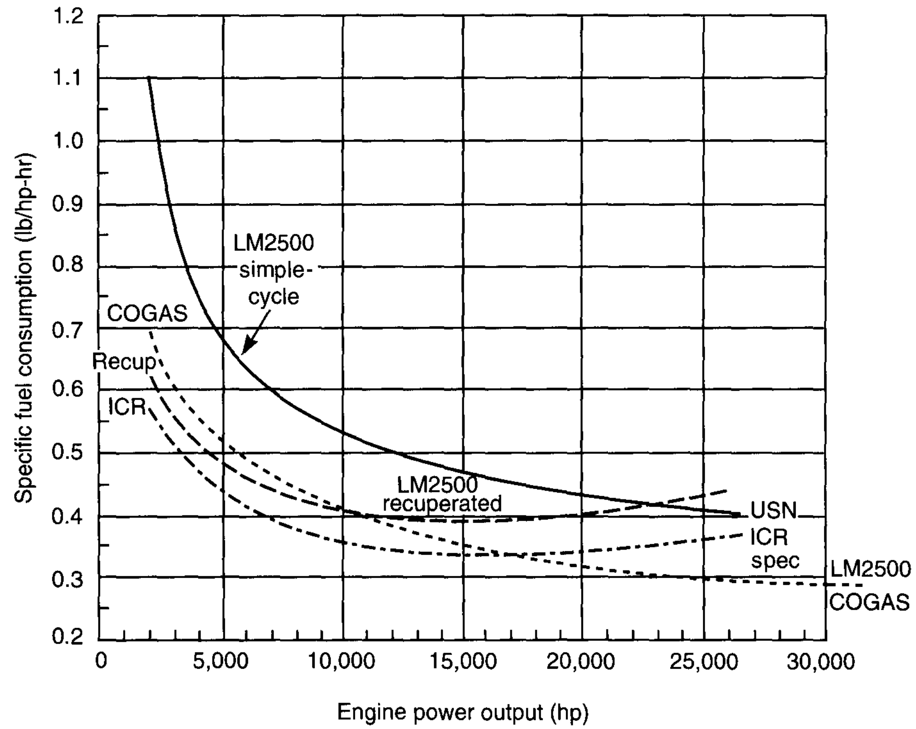
ship's main reduction gearing. The COGAS efficiency curve in figure 7-10 is based upon the figure 7-9 system. Obviously, the steam can be used jointly for steam generator sets and for luxury-liner hotel services. Thousands of COGAS systems commonly called *cogeneration* (COGEN) are used in industry at power stations and in process plants where the steam's heat of condensation may be applied to the process and even higher efficiencies can be obtained. The Russian Ro/Ro ship, *Kapitan Smirnov,* operated a COGAS plant for many years.

The U.s. Navy DD963 destroyers and CG 47 cruisers (over 80 ships) have exhaust heat recovery boilers on their three gas turbine generator sets to provide ship's service steam.

CYCLE COMPARISONS

Figure 7-10 shows a comparison of the estimated SFC versus power and system weight for a simple cycle, recuperated cycle, and a COGAS cycle using an LM2500 gas turbine as the base; and an ICR gas turbine as specified in the U.S. Navy ICRgas turbine development specification. The performance data is normalized to U.S. Navy rating conditions of lOO°F compressor inlet air; 14.696 psia ambient air; 4 inches and 6 inches H_2O inlet and exhaust duct losses; 40 percent relative humidity (RH); 85°F seawater; and 18,300 Btu/lb. fuel lower heating value (LHV).

While the SFC is carried out to 26,400 bhp, the capability of each engine is at least 29,000 bhp, with the COGAS capable of over 40,000 bhp.



Prime mover system	Weight, tonnes
One LM2500 simple cycle	22
One LM2500 recuperated	35
One LM2500 COGAS	114
One ICR unit (USN spec)	55

Figure 7-10. Cycle efficiencies and weights (author's comparative estimate)

It should be noted that in operating any exhaust heat recovery cycle, such as ICR or COGAS, the heat exchanger exhaust temperature should be kept above the temperature at which sulfuric acid could form in the exhaust, usually specified to be 300° to 350°F.

PERFORMANCE

From a ship operator's standpoint, there are three cardinal concerns regarding gas turbine performance: (1) available power, (2) installed fuel rate, and (3) engine maintenance interval. Gas turbine performance is sensitive to the following conditions:

- compressor inlet air temperature
- ambient (barometric air pressure)
- intake air system and exhaust gas system pressure losses
- relative humidity
- compressor bleed (air extracted from the compressor for ship service purposes)
- cleanliness of the engine
- power turbine speed
- fuel heating value

For each application, a gas turbine manufacturer will normally provide a guaranteed power and specific fuel consumption for a stated compressor inlet temperature, ambient pressure, power turbine speed, and the engine in a new and clean condition, and will stipulate the other conditions listed above. Where the installation and operating conditions differ from the rating conditions, performance needs to be corrected. It is quite common for manufacturers information data and gas turbine textbooks to provide general correction factors to show the decrease in rated engine power capability and change in fuel flow for inlet and exhaust duct losses, atmospheric pressure variations, and other operating conditions which would increase engine operating temperature if power were held constant. This is critical information to ensure proper service life and safety of operation. In essence, these are corrections for holding a constant firing temperature and specified output rpm. Expected maintenance interval of the engine hot section (combustor and turbine components) may be decreased if there is extensive operating time at maximum gas temperature rating and if concentrations of the potential impurities that may be in the fuel and intake air are not effectively removed. At high power, the life of hot section parts is inversely related to the cycle firing temperature. For two-shaft engines, power turbine entry temperature is usually measured With thermocouples and directly correlates to firing temperature. For single-shaft turbines, turbine discharge temperature or combustor exit temperature might be measured.

Sample Performance Corrections

For conditions of operation below the engine's rating point, as frequently experienced aboard ship, where power typically varies as the cube of the speed, it is desirable to correct engine operating parameters to compare them to the ship's instrumentation and determine if the engine is performing well. The correction factors, formulas, or curves for constant power and rpm differ from those used to correct data for constant turbine temperature and rpm.

Each model of gas turbine may vary in its rating conditions and the degree of sensitivity to given operating conditions and, therefore, precise

data and correction factors should be obtained for each specific engine model. Also, the turbine manufacturers have computer programs for their specific engine models which can precisely correct engine performance data for their engine, considering all of the variations that may be imposed by ambient conditions, installation characteristics, fuel, and support systems. This is particularly desirable for data comparisons at power and rpm operating conditions substantially reduced from rated power because gas cycle conditions have large changes. The power that a turbine develops is a function of the gas flow times the c_p of the gas times the temperature drop across the turbine. If, at a set power, a duct loss changes, then pressure in the cycle changes, flow changes, and gas temperature changes to compensate for the flow change. With ambient pressure and temperature changes, engine and duct flow functions change, and the method of corrections is complex and iterative and best performed by a computer program.

However, by using general curves and rather typical correction factors, an example of gas turbine performance characteristics and data correction can be appreciated.

Figure 7-11 shows a plot of brake horsepower versus power turbine rpm with lines of constant SFC and constant power turbine inlet temperature. Superimposed is a power versus rpm curve, often typical of a conventional hull, which assumes ship's power ratio varies as the cube of the rpm ratio. Notice that the performance data is based on 59°F, 14.696 psia ambient conditions, no ducting losses or relative humidity, and based on fuel lower heating value of 18,400 Btu/lb.

Figure 7-12 shows a plot of engine horsepower versus ambient air temperature for a 100 percent (3,600 rpm) power turbine speed, with lines of constant exhaust flow (lbs/sec), and power turbine inlet temperature (F). Notice that power above 60°F ambient air temperature is limited by the maximum power turbine inlet temperature. Below 60°F, power is limited by the compressor corrected speed line. Because the mass flow increases as inlet air temperature decreases, the gas turbine compressor becomes limited by mass flow capability or increased pressure ratio that may reach the compressor casing pressure limit. The compressor inlet temperature at which gas temperature and compressor limits intersect, and the slope of these lines with temperature, will vary for different engine models. Manufacturers data for many marine engines simply choose a rating below the gas temperature limit that may be maintained constant as ambient temperature drops. For example, this engine might have been given a constant maximum rating of 32,000 hp below 90°F compressor inlet temperature and data cutoff at minus 40°F, as data below that temperature is not typical for at-sea conditions.

In the following examples, algorithms are stated as a means of correcting certain engine parameters for changes in the operating conditions. In addition, for various calculations, it is necessary to change physical air

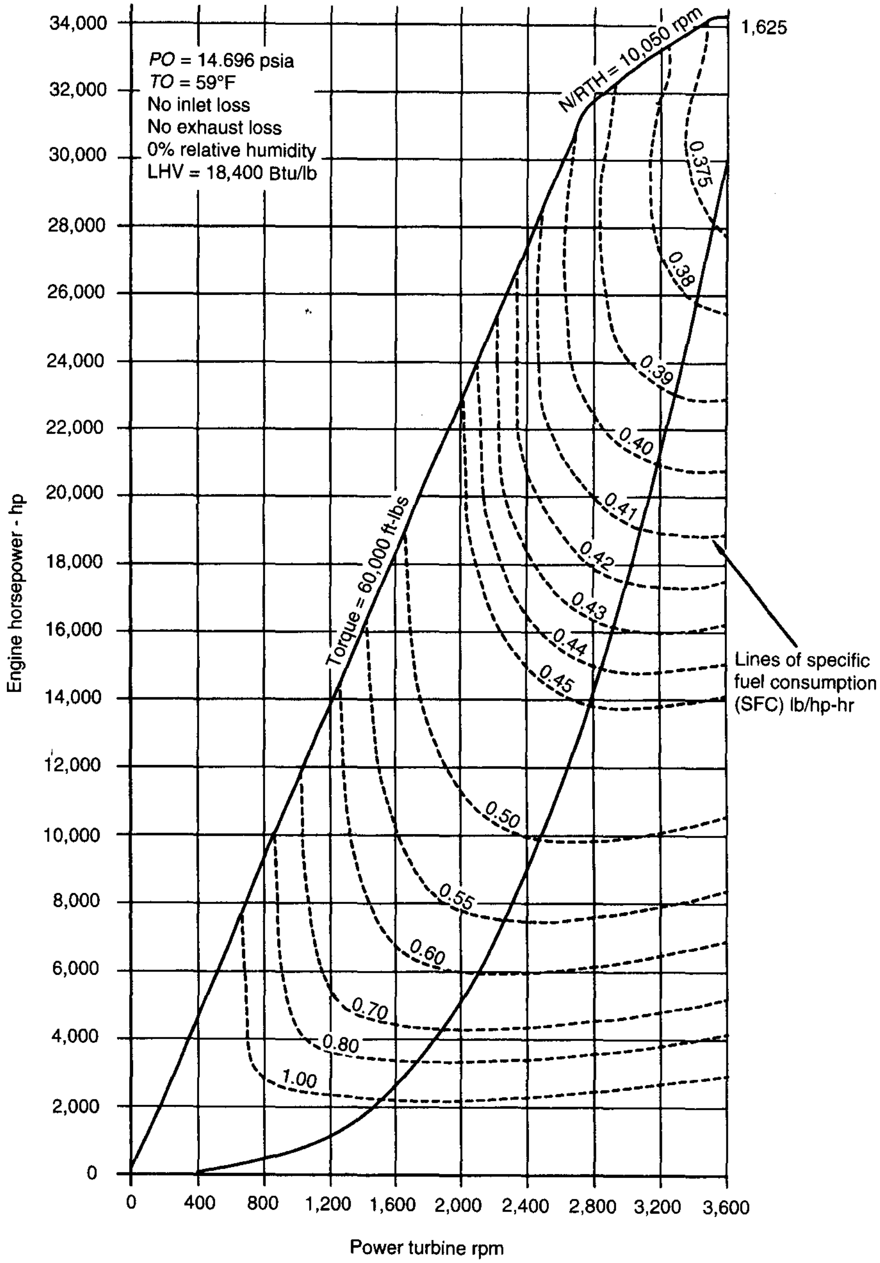


Figure 7-11. Specific fuel consumption variations with output power and rpm. Courtesy General Electric Company.

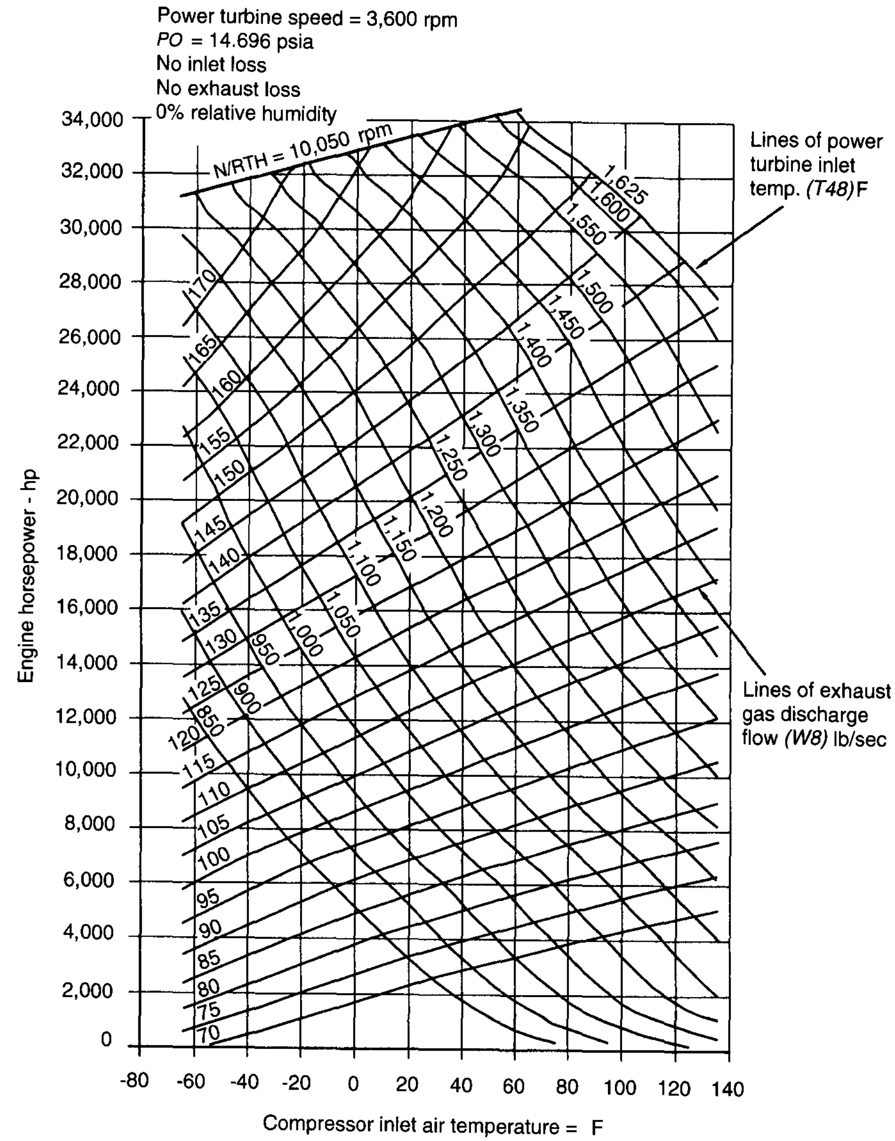


Figure 7-12. Cycle flow and temperature variations with power and inlet air temperature. Courtesy General Electric Company.

flow to corrected flow to adjust for changes from base line pressures and temperatures. The standard correction used for engine flow, in lbs/second, is:

$$\text{Corrected flow} = \text{Physical flow}\left(\frac{\text{Inlet air °R}}{519\text{°R}}\right)^{1/2} \div \frac{\text{Ambient psia}}{14.696\text{ psia}}$$

EXAMPLE 7-1: In order to work with this engine's performance characteristics, assume that during shipbuilder's acceptance trials, the ship demonstrated 27 knots speed with the rating specified at 30,000 hp input to the gearbox at 3,600 rpm power turbine speed. Further, assume trials were run at a 59°F and 14.696 psia ambient pressure and intake and exhaust ducting losses were measured to be 6 inches water column (W.C.) and 6 inches W.C. respectively. Assume trials data also confirmed ship power varied as the cube of ship's speed ratio, i.e.:

$$\frac{\text{Horsepower 1}}{\text{Horsepower 2}} = \left(\frac{\text{Ship speed 1}}{\text{Ship speed 2}}\right)^3$$

The task is to determine the engine fuel flow for the following conditions:

Ship speed	24 knots
Power turbine rpm	(TBD)
Engine hp	(TBD)
Inlet air temperature	80°F
Ambient pressure	14.5 psia
Intake system losses	(TBD)
Exhaust system losses	(TBD)
Relative humidity	60 percent
Compressor bleed	0 percent
Cleanliness of engine	Clean
Fuel LHV	18,200 Btu/lb
Engine fuel flow	(TBD)
Power turbine inlet temperature (T48)	(TBD)

1. Propeller rpm (power turbine rpm) is proportional to ship speed.

$$\frac{\text{Power turbine speed @ 24 knots}}{\text{Power turbine speed @ 27 knots}} = \frac{24}{27} \times 3{,}600\text{ rpm} = 3{,}200\text{ rpm}$$

2. $$\text{Engine hp} = \left(\frac{24\text{ knots}}{27\text{ knots}}\right)^3 \times 30{,}000\text{ hp} = 21{,}000\text{ hp}$$

3. To determine fuel flow, it is necessary to convert hp to that which would occur at standard atmosphere.

Correction to standard atmospheric pressure =

$$\frac{\text{Standard psia}}{\text{Actual psia}} = \frac{14.696\text{ psia}}{14.500\text{ psia}} = 1.0135$$
$$21{,}000\text{ hp} \times 1.0135 = 21{,}284\text{ hp}$$

Enter figure 7-11 at 3,200 rpm and 21,284 hp
Read SFC: .399 lbs/hp-hr × 21,284 hp = 8,492 lbs/hr at 59°F
Fuel flow, *WF* = 8,492 lbs/hr
Enter figure 7-12 at 80°F, 21,284 hp
Read exhaust flow, *W8* = 128 lbs/sec
Observe also that turbine inlet temperature

$$T48 = 1{,}360\text{°F} + 460° = 1{,}840\text{°R}$$

Note: The actual computer data at 3,200 rpm (rather than 3,600 rpm data shown) is *W8* = 130 lbs/sec and *T48* = 1,825°R

It is now necessary to adjust fuel flow and *W8* for the influence of ambient pressure

$$\frac{14.500\text{ psia}}{14.696\text{ psia}} = .987$$

$$WF\ 8{,}492\text{ lbs/hr} \times .987 = 8{,}378\text{ lbs/hr @ 59°F}$$
$$W8\ 128\text{ lbs/sec} \times .987 = 126.3\text{ lbs/sec}$$

$$\text{Corrected engine flow} = 126.3\text{ lbs/sec} \times \left(\frac{540\text{°R}}{519\text{°R}}\right)^{1/2} \div \frac{14.5}{14.696} = 130.6\text{ lbs/sec}$$

4. Fuel flow *WF* varies with ambient temperature. Since our data point is at 59°F, it must be corrected to 80°F. For this engine, variation is 0.04 percent per degree F, increasing or decreasing as compressor inlet air temperature increases or decreases.

$$WF\text{ at 80°F} = 8{,}378\left(.0004 \times 21\text{°F} + 1\right) = 8{,}448\text{ lbs/hr}$$

5. To correct fuel flow for duct losses, first correct duct losses. Duct losses vary as the square of the duct corrected gas flow. The corrected gas flow for this example is shown in step 3 as 130.6 lbs/sec. From figure 7-12, exhaust gas flow at 30,000 hp at 59°F, 14.696 psia was 156 lbs/sec during shipbuilders' trials (no correction needed).

$$\text{Thus actual duct losses} = 6'' \text{ W.C.} \times \left(\frac{130.6}{156.0}\right)^2 = 4.2'' \text{ W.C.}$$

Note: Had the actual flow of 130 lbs/sec *W8,* resulting from operating at 3,200 rpm instead of 3,600 rpm, been corrected for inlet temperature and pressure plus the gas temperature drop due to a lower power setting, the results would be less than 0.2″ W.C. difference in duct loss, which would be indiscernable as a correction factor for fuel flow.

6. The correction factors for this engine for inlet duct loss, exhaust duct loss, and humidity are shown in figure 7-13.

   The correction factor for 4.2″ W.C. inlet loss is:

$$WF\text{: } 8{,}448 \times 1.005 = 8{,}490 \text{ lbs/hr}$$

The correction factor for 4.2 W.C. exhaust loss is:

$$WF\text{: } 8{,}490 \times 1.0054 = 8{,}536 \text{ lbs/hr}$$

The humidity for 60 percent RH at 80°F from a psychometric chart is 92 grains.

$$WF\text{: } 8{,}536 \times 1.0035 = 8{,}566 \text{ lbs/hr}$$

7. Fuel flow varies as the baseline LHV (18,400 Btu/lb in this case) divided by the actual fuel LHV (18,200 Btu/lb)

$$8{,}566 \text{ lbs/hr} \times \frac{18{,}400}{18{,}200} = 8{,}660 \text{ lbs/hr}$$

$$\text{This is an SFC of 0.412 lbs/hp-hr, } \frac{8{,}660 \text{ lbs/hr}}{21{,}000 \text{ hp}}$$

In order to further assess engine health at 21,000 hp, 3,200 rpm, power turbine inlet gas temperature *(T48),* which greatly impacts maintenance interval, should be estimated. In step 3, this was noted from figure 7-12 to be 1,840°R while computer data was given to be 1,825°R. This 15° difference results because, when operating at 3,200 rpm (rather than 3,600 rpm), the gas temperature dropped while the gas flow increased to produce the same 21,000 hp. While the gas flow change had little effect on duct loss corrections of fuel flow, the 15° difference would be a significant discrepancy in evaluating engine cleanliness or wear. Therefore, specific data on variation of gas temperature with power turbine speed would have to be sought.

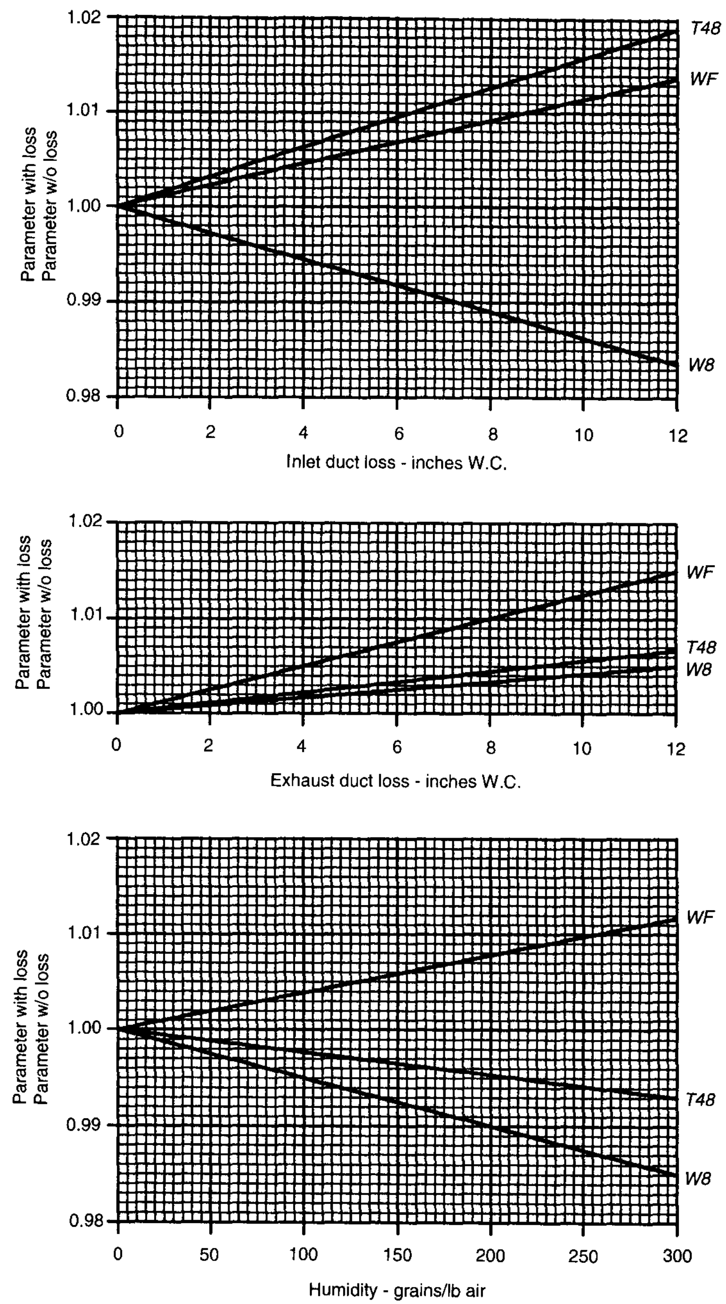


Figure 7-13. Nominal performance correction factors for constant hp and rpm. Courtesy General Electric Company.

Using the 1,825°R data point, correction factors for inlet loss (4.2" W.C.), exhaust loss (4.2" W.C.), and relative humidity (92 grains) obtained from figure 7-13 would be respectively.

$$1{,}825°R \times 1.006 \times 1.002 \times .998 = 1{,}836°R - 460° = 1{,}376°F$$

The basis for these calculations is average engine data. From the sea trials performance data taken, compared to average data (both corrected by computer program) a percentage better or poorer bias might be established for the particular' engine in its new and clean condition.

The ship's engine control station would have a fuel flow meter and a power turbine inlet temperature gauge. Suppose the fuel flow meter and/or the *T48* gauge read greater than that calculated. Possible causes could be instrumentation calibration, the ship hull is fouled (requiring more power for a specified speed), the ship is encountering a head wind, inlet air demisters and/or the gas turbine needs cleaning. Sea air contains entrained salt. While intake demisters remove most of this salt, over a period oftime, salt accretion on demisters can cause increased inlet loss and decreased efficiency, and accretion on the gas turbine surfaces reduces compressor efficiency and airflow thus resulting in higher fuel flow and gas turbine temperatures for a specified horsepower. If fuel flow or power turbine entry temperatures read substantively higher than calculated, determine the cause. It may be as simple as checking demisters and engine for cleanliness and proper operation.

EXAMPLE 7-2: This second example is to determine maximum power conditions without exceeding the *T48* temperature of 1,625°F. Assume all the above conditions are the same except the ambient temperature is lOO°F (unusual but possible, for example, in the Red Sea) and the ship's master wants to increase to maximum possible speed (27 knots) in order to reach port at the proper tide or perhaps avoid a storm. There are two questions: Can 27 knots (30,000 hp at 3,600 rpm) be reached and, if not, what maximum ship speed (engine power) does the engineering officer recommend?

1. First, convert 30,000 hp to that which would occur at standard atmosphere as done in example 7-1, step 1, above.

$$\frac{14.696}{14.500} = 1.0135$$

$$30{,}000 \text{ hp} \times 1.0135 = 30{,}405 \text{ hp}$$

2. Locate 30,405 hp at 100°F on figure 7-12. Uncorrected turbine temperature is 1,611°F and exhaust flow *W8* is 146 lbs/sec. To correct temperatures, they must be converted to absolute temperatures. Thus, 1,611°F + 460° = 2,071°R.

It is necessary, reference example 7-1, to uncorrect the flow for the corrected horsepower, or a double correction for altitude would occur:

$$146 \text{ lbs/sec} \times \frac{14.5}{14.696} = 144 \text{ lbs/sec}$$

$$144 \text{ lbs/sec} \times \left(\frac{560}{519}\right)^{1/2} \div \frac{14.5}{14.696} = 151.6 \text{ lbs/sec}$$

$$\text{Each duct loss} = \left(\frac{151.6}{156.0}\right)^2 \times 6'' \text{ W.C.} = 5.7'' \text{ W.C.}$$

From figure 7-13, the *T48* correction factors for inlet loss, exhaust loss, and humidity, which is 177 grains at 60 percent RH and lOO°Fare:

Inlet factor at 5.7" = 1.009
Exhaust factor at 5.7" = 1.003
Humidity at 177 grains = .996

Corrected *T48* at 30,000 hp, lOO°F is

$$2{,}071°R \times 1.009 \times 1.003 \times .996 = 2{,}088°R$$
$$2{,}088°R - 460° = 1{,}628°F$$

3. The calculated *T48* temperature is above the 1,625°F limit. From figure 7-12, it may be seen that about 4 hp is lost for each degree F drop in *T48*. The calculations are based on average engine data in a clean and unworn state. However, dropping ship speed from 27.0 to 26.9 knots would reduce required horsepower about 330 hp, which would drop calculated *T48* about 75°F below limit to afford margin to accommodate engine wear or other exigencies as discussed at the end of example 7-1. Whatever the circumstances, the engine should be operated below the operating limit to prolong engine life.

It is stressed that dififerent engine models may have performance stated in different ways and model sensitivity to ambient and installation conditions may vary so that the manufacturer's performance and correction factors should be used.

Typical Performance Variation with Configuration

In the "Principles of Gas Turbines" section, generalities in the differences in single-shaft versus split-shaft gas turbines and simple-cycle versus recuperated and intercooled recuperated (ICR) cycle performance were discussed. Figure 7-14 compares load versus efficiency and specific fuel consumption (SFC) for various models of Solar gas turbines. These include existing simple-cycle, single-shaft shipboard generator sets; simple-cycle, two-shaft main propulsion engines like the Solar Taurus 60M; the integrally recuperated with VATN Solar 5650 model; and the intercooled, recuperated, with VATN version (ICR). An advanced turbine system (ATS) currently under development by Solar to be available in the 50 to 60 percent thermal efficiency range, is shown for comparison. This engine is to be an ICR with VATN and low emissions combustion (LEC) using advanced cooling/manufacturing methods with ceramic components, and enhancements in component efficiencies.

PROPULSION GAS TURBINES

Types

When seeking to categorize marine gas turbine engine designs into types, there may be three general categories under which authors may suggest distinctions.

- the number of separate rotating shafts along the engine centerline
- the historic genesis for the design, be it the industrial or heavy-duty gas turbines designed originally for stationary sites or the aeroderivative designs derived by modifying aircraft gas turbines for shipboard and industrial services
- different approaches to major component designs

NUMBER OF SHAFTS

Basically, there is the single-shaft gas turbine where the gas generator and the load driving power turbine are on one common shaft, and the two-shaft gas turbine where the load driving power turbine is on a separate shaft from the gas generator turbine. The gas generator in the two-shaft gas turbine also may have multiple compressor-turbine rotors with concentric shafts, and is usually referred to as a two-spool or dual-spool gas generator. Line sketches of these distinctions are shown in the "Principles of Gas Turbine" section, figures 7-5 and 7-7. Single-shaft turbines are not generally used for propulsion. They will be discussed in the "Auxiliary Gas Turbine" section, starting on page 7-94.

HISTORIC GENESIS

Historically, in the 1700s and 1800s, large industrial power needs and ship propulsion needs were met by reciprocating steam engines. In the 1880s, steam turbines and internal combustion engines were introduced and, in the 1890s, diesel engines were introduced. In the 1920s, the gas turbine be-

came practical as a turbo supercharger drive using internal combustion engine exhaust to drive a supercharging intake air compressor. In parallel with the turbo supercharger, industrial gas turbine engines began service. The rudiments of their design and materials temperature capabilities were taken from steam turbine technology. Because of the limits of compressor design technology and materials technology, cycle enhancement methods (for example, recuperation) were prevalent. In the 1940s (by which time several governments had funded the technological development of gas

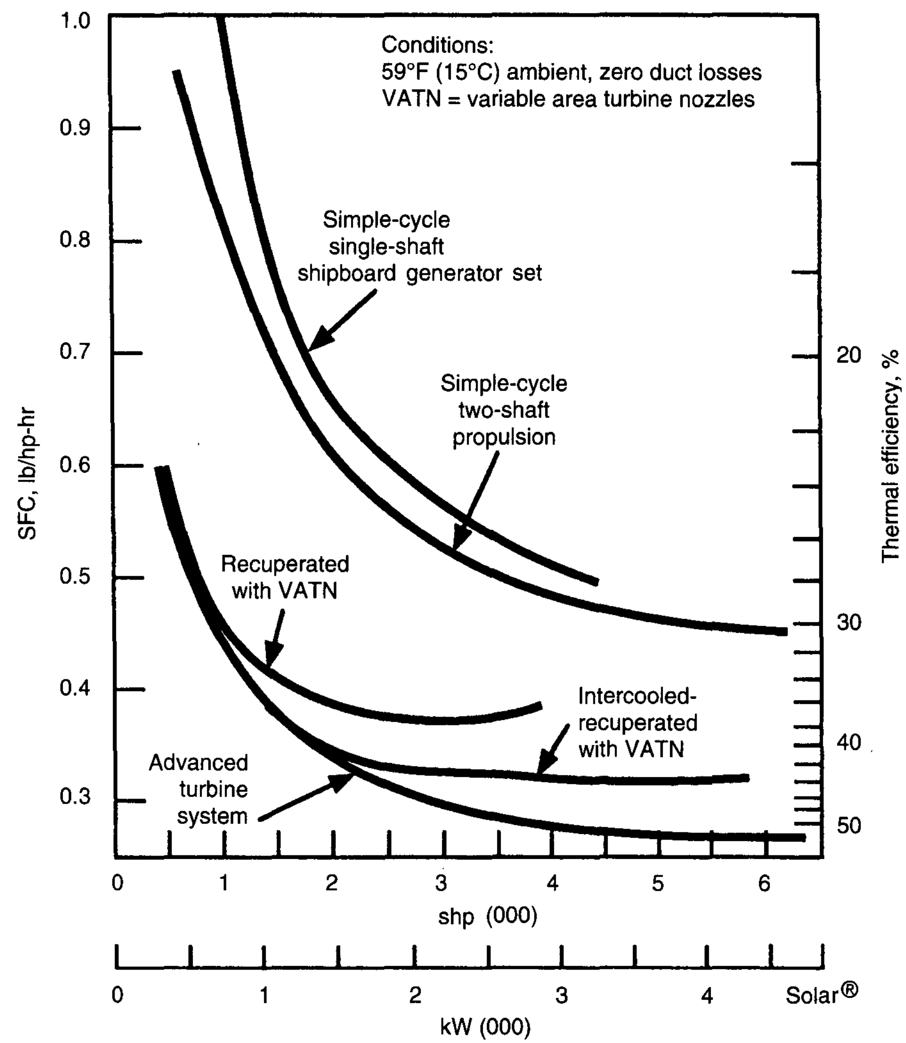


Figure 7-14. Marine gas turbines. Courtesy Solar Turbines Incorporated.

turbines for aircraft propulsion), this technology began gradual assimilation into industrial designs, and the use of gas turbines in the utility market began to flourish. By the 1980s, both types enjoyed high efficiency simple-cycle designs.

Deriving a marine gas turbine from an aero gas turbine starts with obtaining a suitable power turbine. In the case ofturboprop aircraft engines, turboshaft helicopter drive engines, and aircraft fanjet engines, in many instances, the turbine used to power the aero propeller, rotor, or fan may be directly adapted as a marine drive turbine. If the power turbine drive shaft emerges through the compressor, it is classed as a front-end or cold-end drive. Ifit emerges out the exhaust end of the engine, it may be referred to as a hot-end drive. In the case of adapting an aircraft jet engine, either a new turbine is designed or possibly a turbine from a different aero engine model may be adapted. Following the selection of a drive turbine, design change considerations may include reconfiguration ofthe compressor inlet and front frames to match the airflow needs ofthe compressor, particularly where a fan is removed that provided boost pressure to the compressor; special selection of materials and coatings for blades, vanes, and combustors to minimize the corrosive effects of a salt spray atmosphere and marine fuel corrosive elements; balancing bearing thrust loads for continuous operation at sea level air densities; the possible off-engine mounting of some pumps and fuel control components; and mounting and enclosure systems befitting ship propulsion spaces. Because aero gas turbines usually have requirements for greater degrees of roll, pitch, yaw, and slamming than do ships, their systems usually adapt readily to ship service. Industrial gas turbines, often designed for stationary mounting, may require some modification for these aspects of ship motion as well as selection of marine materials and coatings for the marine environment.

COMPONENf DESIGNS

Some texts differentiate gas turbines into types by the design variations in major component parts. Some examples might be axial versus radial or centrifugal rotor gas flow paths, impulse and reaction blading, static and variable stator geometry, et al. In this chapter, these are considered component variations and are discussed as construction details of the components.

Construction

ENGINE CONFIGURATIONS

Figures 7-15 through 7-18 are cutaway drawings of four different manufacturers' two-shaft gas turbine engine models offered for shipboard and other applications. Among these four engines are illustrated many of the distinctive variations in component designs that are subsequently discussed.

Figure 7-15 shows a partial cutaway drawing of an aeroderived 30,000 horsepower class simple-cycle, GE LM2500 marine gas turbine derived from a turbofan engine. The gas generator is composed of a sixteen-stage axial flow compressor with horizontally split stator casings and seven stages of variable angle stator vanes, an annular combustor with 30 fuel nozzles, and a two-stage, air-cooled, high-pressure turbine. The gas generator maximum rotor speed is about 10,000 rpm. The power turbine is a six-stage low-pressure turbine with horizontally split stator casings which is aerodynamically coupled to the gas generator. The power turbine maximum steady-state speed is 3,600 rpm. The exhaust gas is expanded through a diffuser cone into a 90 degree exhaust elbow or collector. The flexible coupling shaft joins the power turbine to the ship's speed reducing gearbox.

The engine is held in alignment by frames and the outer casings between the frames. The forward end of the compressor is supported by the front frame assembly roller bearing. This frame contains a right angle bevel gear and radial shaft that powers the accessory drive gear box that mounts under the front frame. This frame also supports the inlet bell mouth and bullet nose. The compressor stator casing rigidly joins to the compressor rear frame which contains a ball bearing to accommodate gas generator rotor thrust and a roller bearing for radial stability. The compressor rear frame also forms the outer casing for the combustor and high-pressure turbine and rigidly attaches to the turbine midframe. The midframe has a roller bearing to support the rear of the high-pressure turbine and a roller bearing to support the forward end of the power turbine. The power turbine stator casing rigidly attaches to the turbine rear frame, which supports the rear ofthe power turbine with a ball thrust bearing and a roller bearing for radial stability. The exhaust duct is mounted separately from the gas turbine and has a piston ring slip joint seal that rides on a land on the diffuser cone to accommodate axial and radial motion between the two components.

Figure 7-16 is a cross section of the Turbo Power and Marine FT8 ® marine gas turbine in the 25 MW (33,500 hp) class. Notations show borescope access ports which will be discussed under "Maintenance" in this section. The FT8 gas turbine is composed of two main sections: the gas generator and an aerodynamically coupled power turbine that provides power to the output shaft and driven load. They both employ antifriction bearings and share a common lube oil system. The FT8 gas generator is a two-shaft design. It consists of an eight-stage, low-pressure compressor driven by a two-stage, active clearance control, low-pressure turbine plus a seven-stage, high-pressure compressor driven by a single-stage air-cooled, high-pressure turbine. The compressor inlet guide vanes feature variable position aerodynamic flaps and the first two compressor stages have variable geometry stator vanes. The can-annular combustion section consists of nine combustor cans, each with a fuel nozzle. The power turbine is a four-

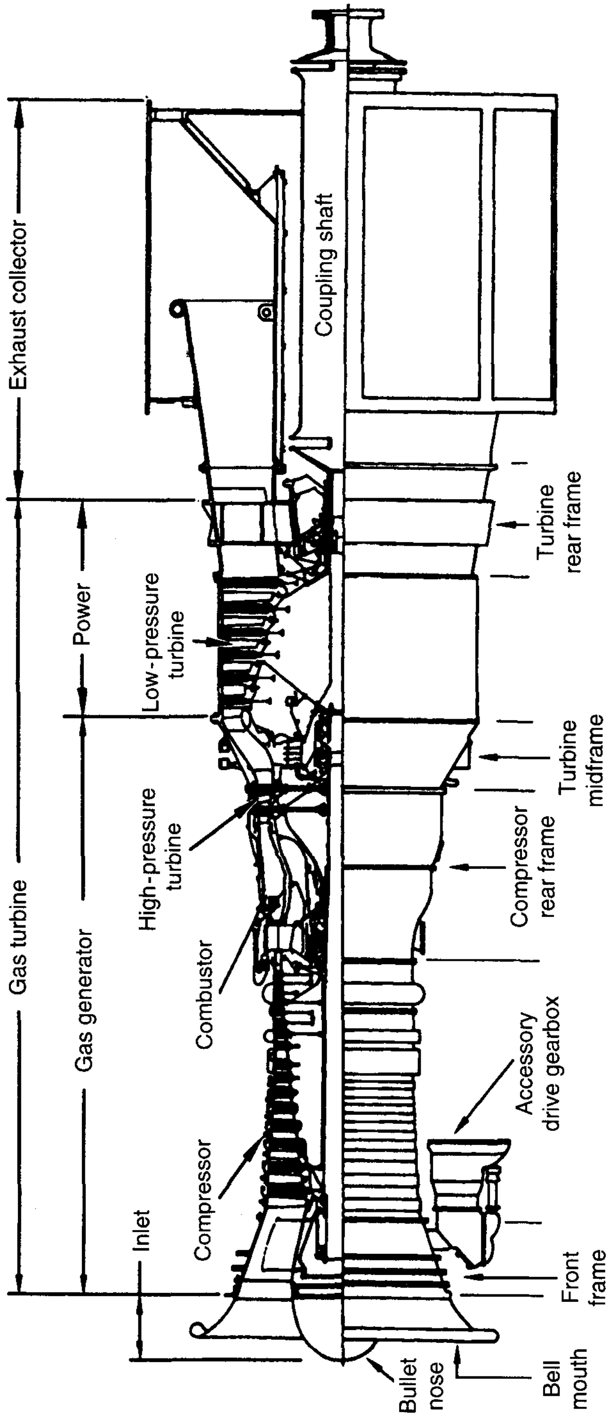


Figure 7-15. GE LM2500 propulsion gas turbine. Courtesy General Electric Company.

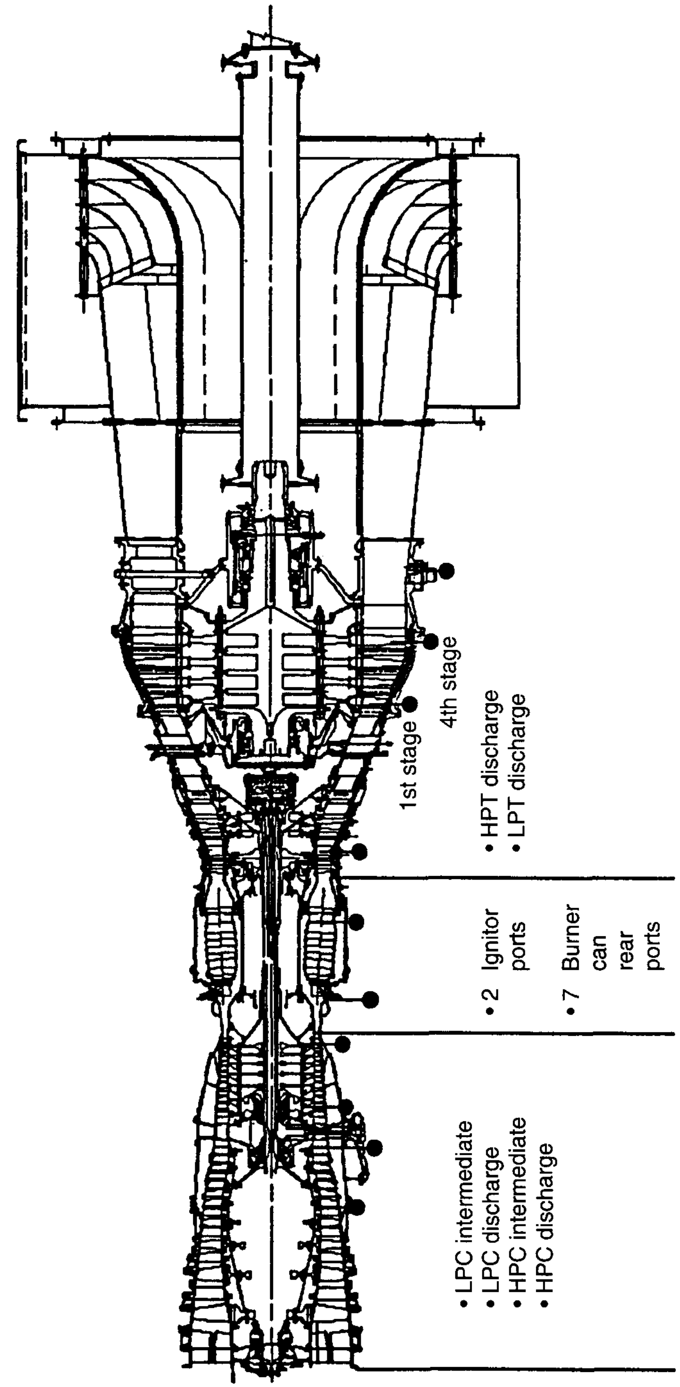
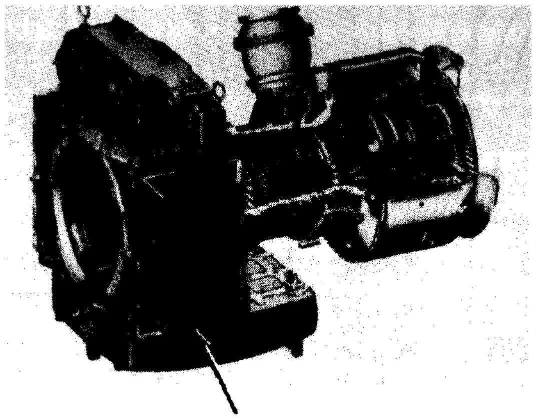


Figure 7-16. Turbo Power and Marine FT8® engine showing borescope access. Courtesy Turbo Power and Marine Systems, Inc.

stage industrial design mounted between antifriction bearings. Power turbine airfoils can be provided for clockwise or counterclockwise rotation.

Figure 7-17 shows an isometric cutaway and a flow diagram of the Textron Lycoming 4,500 hp class TF40 gas turbine. This engine is a cold-end drive and the air intake path is taken from both sides of the inlet housing.



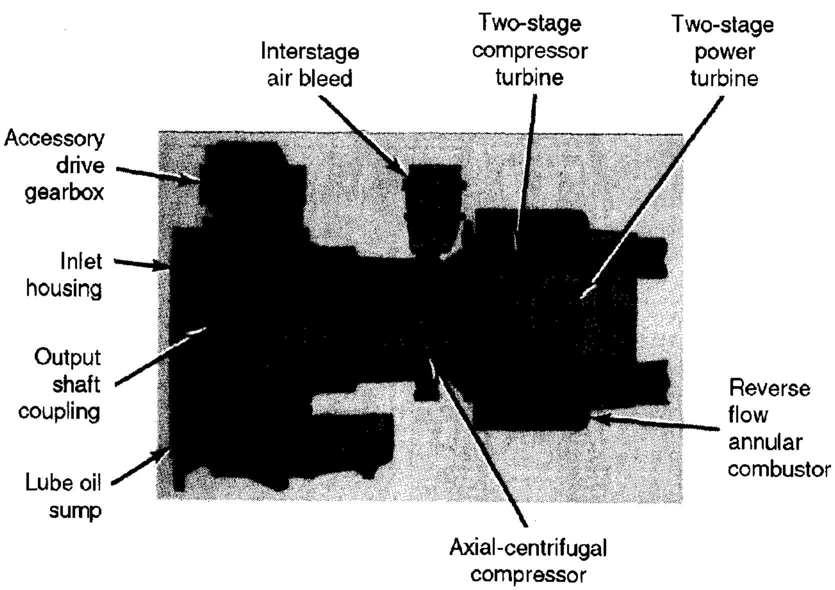


Figure 7-17. Textron Lycoming TF40 marine gas turbine. Courtesy Textron Lycoming Turbine Engine Division.

The compressor has six axial stages and a centrifugal stage on the same shaft driven by a two-stage high-pressure turbine. Air leaving the centrifugal compressor radially is channeled through an annular passage surrounding the reverse flow combustion chamber and reverses direction. The combustor fuel nozzles are at the rear of the combustor. The gas, heated by the combustor fuel upon leaving the combustor, again reverses direction and passes through the high-pressure nozzles and turbine and then through the power turbine assembly.

Figure 7-18 shows the Solar Centaur Taurus 5MW (6,700 hp) class gas turbine. The words Solar and Centaur Taurus are trademarks of Solar Turbines, Incorporated. The Taurus is a hot-end drive engine but drives the accessory gearbox drive assembly concentrically forward of the compressor. The other three engine examples utilize radial drives to the outside diameter of the engine for the accessory gearbox drive. The air inlet assembly draws the air radially inward and directs it axially into the compressor. The twelve-stage axial flow compressor has four variable geometry stator vane stages. The combustor is annular with twelve fuel injectors using compressed-air-assisted atomization during starting. There is a two-stage gas generator turbine and a two-stage power turbine that is cantilever supported by two tilting pad journal bearings located aft of the rotor. This engine is not aeroderivative, having been designed expressly for industrial and surface transport services.

ENGINE INLETS

For hot-end drive gas turbines, it is common to have a bell mouth and bullet nose, as shown in figure 7-15, to deliver a smooth and nonturbulent airflow into the compressor inlet. For cold-end drive turbines, as shown in figure 7-17, the inlet housing is designed to direct the air outside the space occupied by the power output shaft and coupling. Somewhere in front of each inlet orifice, a screen should be mounted. The screens are usually about one-fourth inch mesh and shaped to have an effective area at least as large as the inlet opening(s). Effective area is open area, total area minus the screen structure, wire, and their aerodynamic blockage effect. The screens catch moderate and large size objects to prevent compressor damage. They also tend to reduce inlet air pressure distortion by straightening and breaking up small air vortices. The screens preceding a bell mouth are sometimes mounted so they are not supported by the bell mouth as will be shown later. It is common for bell mouths and inlet housing to have liquid manifolds incorporated to spray liquid cleaning solution into the compressor for removal of salt, oil and dirt buildup from the blades and vanes of the engines (see "Installation, Engine Cleaning," this section). Some inlet systems may have heating arrangements to preclude ice formation in cold weather.

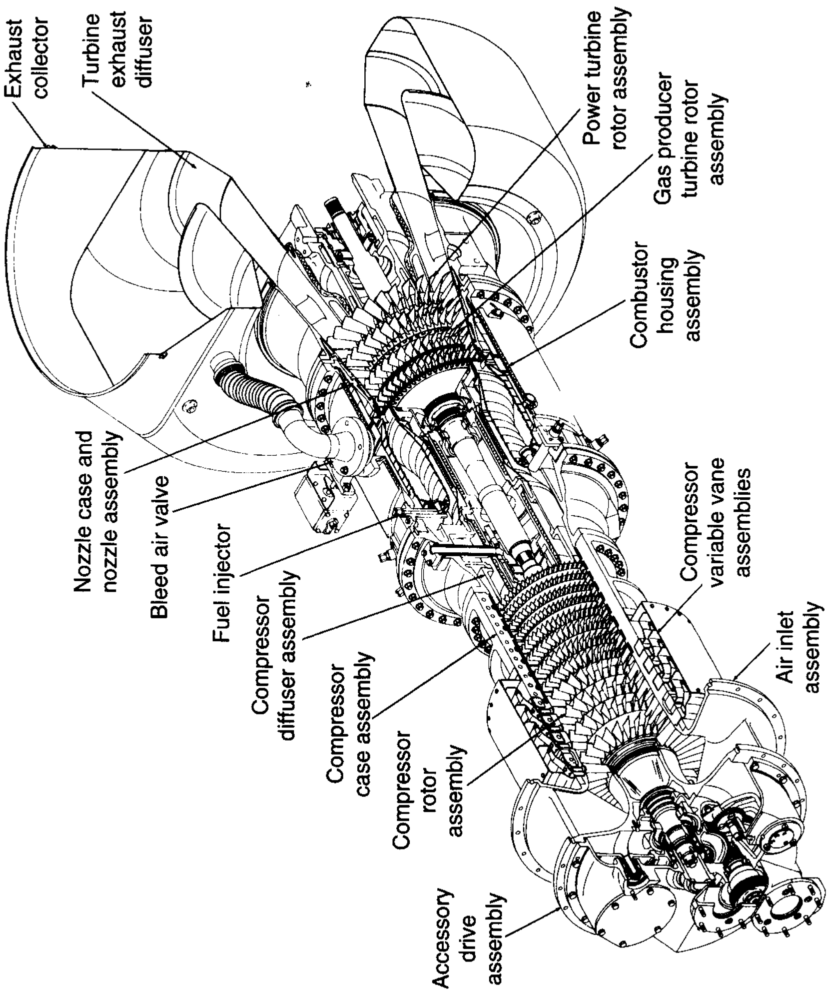


Figure 7-18. Solar® Taurus™ marine gas turbine. Courtesy Solar Turbines Incorporated.

FRAMES

In addition to supporting the rotating elements, gas turbine frames are designed to accomplish various other functions. Figure 7-19 shows the four frames of the engine in figure 7-15. The following is a listing offunctions performed by each of these frames:

Compressor front frame

1. provides a smooth flow path of the gases
2. supports bearings and accommodates bearing housing and seals
3. struts house oil, scavenge, and vent passages
4. supports inlet gearbox and radial drive shaft for accessory gearbox and mounts and supports accessory gearbox
5. has mounts for mounting engine
6. has handling mounts for lifting the engine
7. accommodates instrumentation sensors

Compressor rear frame performs functions 1,2, and 3, plus

8. mounts fuel nozzles and ignitor plugs
9. provides ports for compressor air bleed
10. accommodates borescope inspection ports
11. provides outer casing for combustor
12. supports turbine nozzles

Turbine midframe performs functions 1,2,3,6,7,10, and 12.
Turbine rear frame performs functions 1, 2, 3, 5, 6, and 7 plus

13. supports exhaust diffuser cone and shaft tunnel

Frame functions vary for different engine models, but most of the frame functions listed above are common among engines. Frames may be cast, fabricated, or made of a combination of castings and fabrications.

The TF40 engine (shown in fig. 7-17) compressor front frame cantilever mounts the engine to the inlet housing by a circumferential flange and the whole assembly as shown can be cantilever mounted to the ship's speed reduction gearbox.

BEARINGS AND SUMPS

There are two categories of bearings used in gas turbines: antifriction or rolling-contact bearing, and sleeve or journal bearings. Aeroderivative gas turbines generally use antifriction bearings because the bearings and lubrication systems are lighter and more compact. The journal bearings systems used with the industrial turbines and sometimes with power turbines added to aeroderived gas generators are similar to those described in the steam turbine chapter.

Antifriction bearing systems use a ball bearing on each rotor shaft system to fix the axial position and absorb the axial thrust of the rotor, and one or more locations of cylindrical roller bearings to provide radial support

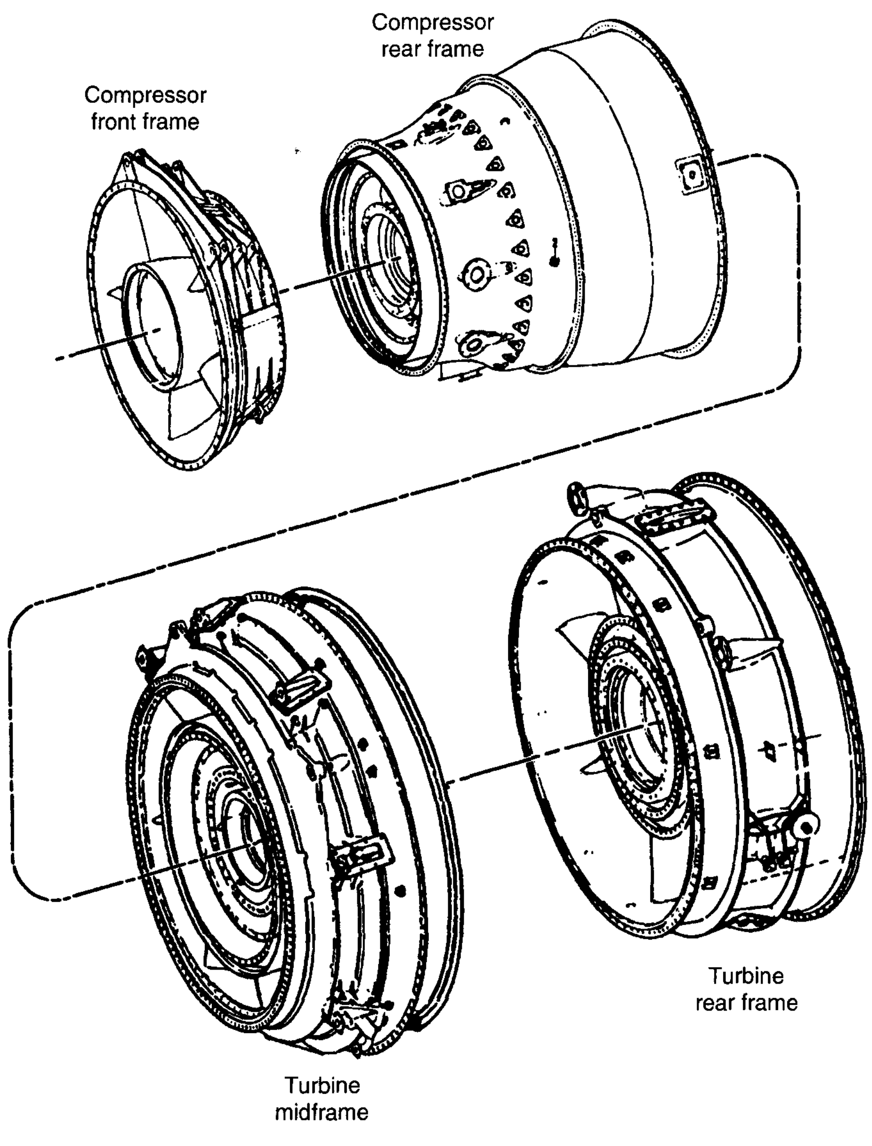


Figure 7-19. LM2500 gas turbine frames.
Courtesy General Electric Company.

while accommodating axial movement of the rotor. Antifriction bearing locations are discussed in the descriptive text for figure 7-15. Antifriction bearing designs are selected to have B-10 life many times that of the anticipated overhaul interval and are replaced at appropriate intervals of engine overhaul.

Figure 7-20 shows typical antifriction bearing sump philosophy. Each sump is encased in a sump pressurizing chamber maintained by air seals that prevents excessive heat from reaching the oil wetted walls of the inner oil seals, thereby preventing coking and thermal deterioration of the oil. To prevent oil leakage, rotating devices on the shaft sling oil away from the seals, and bearing oil seals use pressurization air to flow across the oil seals into the sumps. Seal pressurizing air is extracted from intermediate stages of the compressor and distributed to the oil seals. To remove the air that enters the sumps through the oil seals and to maintain a pressure drop across the oil seals, the sump air is vented. Each sump area is connected to a sump vent manifold through frame struts. The manifold connects to an air-oil

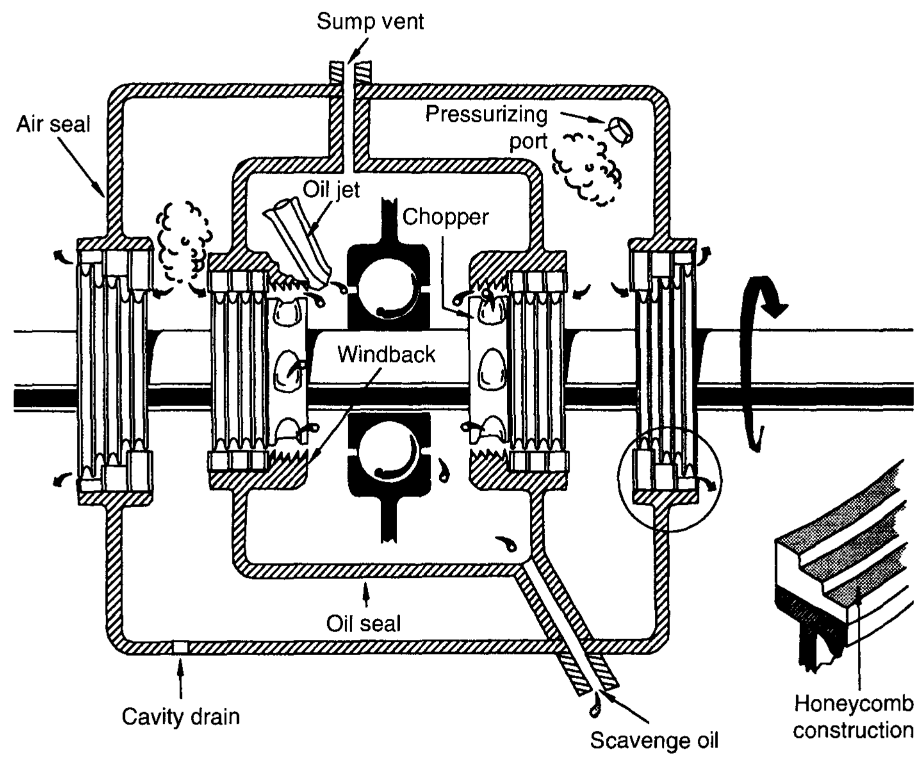


Figure 7-20. Typical antifriction bearing sump philosophy.
Courtesy General Electric Company.

separator that extracts oil from the air before venting the air into the exhaust duct. Extracted oil is returned to the scavenge oil pumping system. Because antifriction bearings require very little oil flow for lubrication, supply oil is jetted directly at the rolling elements sufficient to provide bearing cooling without flooding the space between rolling elements and races. The shaft seals are typically labyrinth, honeycomb, or carbon or graphite contact, or some combination of these designs.

Typically, gas turbines with antifriction bearings do not require pre- or post-operation lube flow to the bearings, and use lube supply and scavenge pumps which are driven off their accessory drive gearboxes.

COMPRESSORS

Compressors raise the pressure of the working fluid, in this case air, to afford a flow to the combustor, where heat is added at nearly constant pressure slightly less than compressor discharge pressure to provide the pressure drop across the turbine(s) required for temperature extraction to produce work to drive the compressor and the driven load. The pressure rise across the compressor is termed pressure ratio-the ratio of absolute discharge total pressure to absolute inlet total pressure-and should not be confused with the term compression ratio as used with reciprocating engines, which refers to the volume ratio.

There are two basic forms of compressor design: centrifugal and axial-flow compressors. Figure 7-21 shows typical centrifugal compressor impellers and typical axial-flow compressor rotor and stator assembly. The centrifugal impeller has an axial inlet and a radial discharge and typically is limited to a pressure ratio below 5:1 per stage. The impeller imparts momentum or kinetic energy to the air. The air is discharged into a diffuser and scroll or manifold, which are stator devices that transform the momentum into static pressure rise. Advantages of centrifugal compressors are simplicity and low cost. They may be mounted in tandem but multiple stages become heavy, with complicated flow paths, and are inefficient compared to axial-flow compressors. Peak efficiency is typically in the order of 80 percent. They are used on small, low-power turbines or in combination with axial-flow compressors. Figure 7-17 shows a centrifugal impeller as the last stage of compression of the TF40 engine. This is an ideal application in combination with reverse flow annular combustor which, by having the combustor outside the turbine section, makes a very short compact engine package.

Axial-flow compressors are a series of airfoil stages of alternate stator vanes and rotor blades ending with exit vanes. The blades are mounted into the rotor. Stator vanes are mounted into the stator casing. The vanes direct the angle of the flow into the blades and out of the compressor. Axial-flow compressors are used in virtually all high-power gas turbines albeit some may be supplemented with a centrifugal stage. Axial-flow

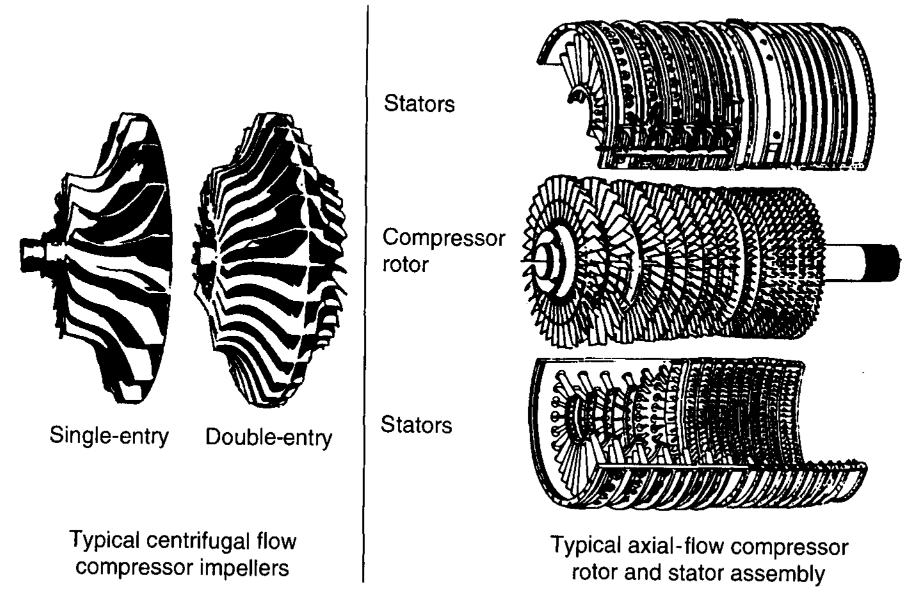


Figure 7-21. Types of compressors. Courtesy General Electric Company.

compressor peak efficiency can approach 90 percent. For compressor assemblies over about 5:1 pressure ratio, compressor rotors must operate. over a large speed range, which makes them sensitive to instability, particularly during transitions, for example, during starting mode and during power accelerations and decelerations. The area of instability is called the compressor stall line or surge line. The term stall is probably taken from the phenomena associated with the airplane wing. Each blade is an airfoil and as with the wing, if the angle of attack to the incoming air is excessive, each blade or row of blades suffers rapidly increasing losses and air pumping diminishes. Carried to its ultimate, if the pressure and flow drops off rapidly enough, the pressure in the combustor is no longer lower than the compressor discharge pressure and there is momentary reverse flow or stalls, with possible fire emerging out the compressor inlet with accompanying explosive noise. A surge is a condition where compressor flow from the high-pressure stages in stall reverses momentarily and then forward flow is reestablished and a sequence of stall and forward flow continues until the stall condition is relieved. There are four methods of design used to prevent these phenomena under normal operating conditions: (1) the use of compressor interstage or discharge stage bleed or bypasses; (2) variable geometry compressor stator vanes; (3) twin-spool compressors; and (4) combinations of the above.

The engines in figures 7-15, 7-16, and 7-18 all have some stages ofvariable stator vanes. Figures 7-17 and 7-18 show the use of a compressor bleed manifold and valve. The figure 7-16 engine shows the twin-spool compressor configuration. The principle of all these systems is to either throttle the inlet air or adjust the back pressure and flow of the higher pressure stages of the compressor during starting and throttle (power) transients.

The compressor bleed and bypass systems reduce the flow and pressure in the high-pressure stages. Regulation of the valves that meter the bleed air flow is done in tandem with the control offuel flow as a function of compressor rotor rpm and inlet air temperature.

The variable stator vane (VSV) mechanism changes the position of the variable vanes to reduce the airflow angle of attack to the rotor blades, and has the apparent effect of reducing the area of the inlet orifice, as by a shutter action, as a function of compressor inlet air temperature and gas generator rpm. As the speed increases, the VSV mechanism progressively opens the vanes until the vanes are fully open.

The twin-spool compressor shown in figure 7-16 (and in schematic form in fig. 7-7) has the low-pressure compressor and turbine matched to operate at this rotor's optimum speed and flow, and the high-pressure compressor and turbine matched to operate at this rotor's optimum speed and flow during acceleration and deceleration.

Compressor air is used throughout the engine for many purposes including cooling and pressurization of oil sumps and seals; thrust balancing of rotors; bleeds (via both the casing and the rotor passing through the shaft) to cool turbine vanes, blades, disks, and supporting structures, in addition to available bleed for uses outside the engine.

Blades have high centrifugal loads and high bending and vibratory loads although centrifugal force helps ameliorate this through centrifugal stiffening, as compared to stator vane bending stress. Bending and vibratory loads require adequate cord section to preclude high-cycle fatigue. Blade and vane materials and shape must be able to withstand minor foreign object damage (FaD) of nicks, dents, and tears with minimum crack propagation to remain intact lest they break and severely damage many stages downstream. Such damage caused by engine-generated debris is referred to as domestic object damage (DaD). FaD can result from objects small enough to pass through the compressor inlet screen. On large engines, long first-stage blades may have interlocking midspan shrouds for damping. Blades are typically retained in the rotor using fir tree roots, axial dovetail slots, and circumferential slots with a locking lug at the entry space for inserting blade bases in the slots. Blade dovetail slots and midspan shroud interlock surfaces often have hard coat spray or hard alloy strips, and dovetails may be lubricated with graphite to reduce wear. Blade mounting is designed to allow motion (tip shake) to reduce stresses. Typically, compressor rotors are composed of hollow bore disks or spools accommodating three or more stages, and in large engines, more typically, combinations of disks and spools held together by bolted joints. Some

spools may be comprised of partial spools, machined and then inertia-welded together. The interstice of disks among spool sections reinforces the spools to prevent distortion. Different rotor blade and vane materials are chosen as temperatures and pressures increase through the compressor. At the low-pressure end, materials such as aluminum and titanium alloys may be used while the higher pressure and temperature areas use nickel-chrome-steel alloys.

A device currently beginning use in some gas turbine compressors is the compressor blisk for low-pressure stages. The word blisk is the combination of the words blade and disk, and describes the device produced when the blades and the disk are manufactured as one integral piece. Potential advantages to selected applications ofblisks, compared to separate blades and disks, are weight savings, no dovetail problems such as fretting, potential cost savings considering parts plus assembly, and increased blend limits for damaged blisk airfoils.

Compressor stator casings for some gas turbines are one piece, with rotor and stator stages assembled and then loaded from the end into the stator casing. More frequently, stator casings are designed with horizontal split lines, which ease maintenance inspection and reparability by removing half a casing at a time.

Typically, variable angle stator vanes, as shown in figure 7-21, are installed in the casing through holes in the stator casing. The top shoulder of each vane rotates in a flanged bushing inside the casing. Outside there is a threaded stud, spacer or washer, and lever arm. A locknut and sleeve hold this together in compression. All the levers per stage are attached to a circumferential ring that is positioned by actuator arms powered by hydraulic actuators. The longer low-pressure stage vanes may pivot in a shroud at the inner diameter, which supports the vanes axially and circumferentially. Shorter variable and fixed vanes are cantilevered from the casing. Fixed stator vanes are typically retained in circumferential slots in the casing.

COMBUSTORS

The gas turbine combustion chamber must convert the chemical energy of the fuel to thermal energy to provide a uniform gas temperature into the turbine, ideally using a small combustor volume with low-pressure loss and high efficiency. Gas turbine combustion is a continuous process. Once the combustion flame is lit and the turbine is at a self-sustaining idle speed, an ignition source is not required.

Figure 7-22 shows the three basic types of combustor systems. The can or cellular system is composed of a number of burner cans, each with its own outer casing. The can-annular or cannular system has a number of burner cans with a common annular inner and outer casing. In both systems, each burner can has its own fuel injector nozzle(s) and each system has cross-fire tubes (seen in the can-type sketch) which join the flame zone

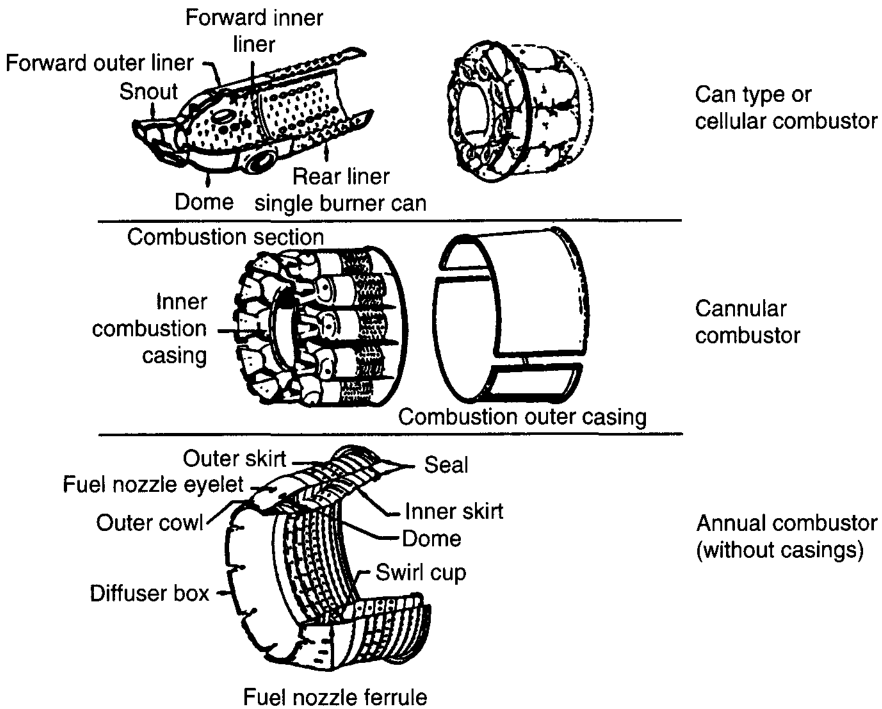


Figure 7-22. Types of combustors. Courtesy General Electric Company.

of each can to the next so that a flame initiated in one or two of the cans propagates and lights the other burner cans during starting. These two burner systems are typically mounted around the axis of the engine between the compressor and the high-pressure turbine; the number of cans may vary among engines from six to twelve. In some large industrial gas turbines burning very heavy fuels, one or two very large combustion cans are used outside the axis of the engine. The third type is the annular combustor shown here without the inner and outer casing. The annular combustor system has several advantages. It tends to be lighter and more compact for similar power class engines and because the number of fuel nozzle locations is not limited to the number of cans, there are two to four times as many smaller fuel nozzles used which results in more uniform temperatures in the combustor and of the gas going into the turbine.

A variation to the mounting of any of these systems in the axis of the flow of the engine is the reverse flow combustor as shown in figure 7-17. Combustors can also be mounted to flow radially inward, perpendicular or at an angle to the engine axis, and either of these variations can be useful in constraining the length of an engine when a recuperator is used.

The combustor design must achieve the following:

- stable operation over a wide range offlows, pressures, and temperatures with a short flame length and minimum pulsation and noise
- low pressure losses and high efficiency
- even, uniform gas temperature into the turbine
- high durability and life, minimum maintenance or repair
- clean combustion essentially without carbon deposits, without smoke, and within the ever-increasing restrictions on allowable oxides of nitrogen (NO_x)

Combustion intensities of current day marine gas turbines can be over ten times that of high output steam boiler furnaces. Figure 7-23 shows the typical distribution and use of the air in the combustor. The process of combustion entails the atomization, vaporization, mixing of the fuel to a combustible fuel/air ratio, and ignition of the mixture. In the primary combustion zone, the gas flow velocity must be partially reversed so that the flame is self-piloting, negating the need for an ignitor after engine idle is established.

Atomization is performed by the fuel nozzles. Because fuel flow at maximum power may be in the order of fifteen times that at idle condition, fuel nozzle systems may use multiple orifices and fuel flow dividers. Compressed air may also be used in the fuel nozzles to assist atomization. A small portion of the compressor air is directed around each fuel nozzle through an annulus often configured to produce vortex-induced swirl to enhance fuel vaporization and mixing ofthe fuel and air to a combustible mixture. Ignition is usually accomplished by electrical discharge ignitors. The

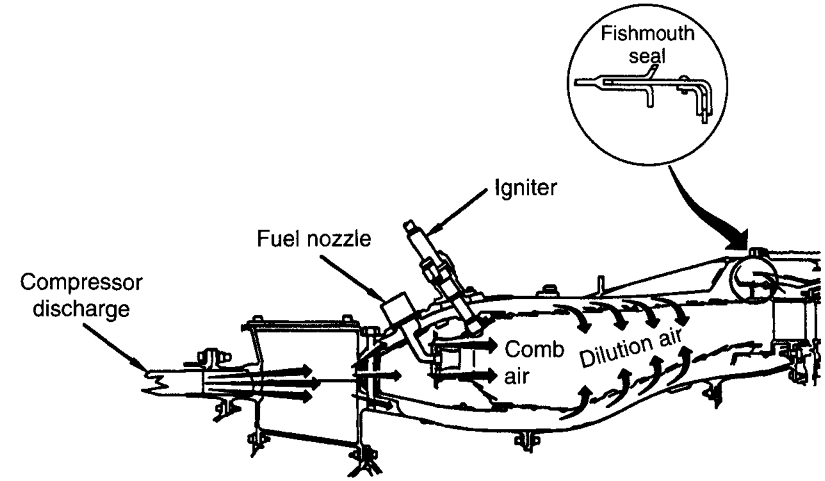


Figure 7-23. Combustor air distribution. Courtesy General Electric Company.

core ofthis flame is typically near stoichiometric, in the order of 3,600°F, varying with the fuel used. The remaining air cools the combustion liner and casings and is introduced downstream of the flame as a dilution air to produce a uniform radial and circumferential temperature profile into the turbine within the allowable turbine gas entry temperature. The total air-to-fuel ratio may be 50-to-60:1 at full power and 100:1 or greater at idle. Typical current day combustor efficiencies are 97-to 98-plus percent. Liner materials are typically high nickel-chromium and high cobalt-nickel-chromium with iron alloys. Ceramic thermal barrier coatings are used on some combustors.

Typically, aeroderivatite engines are designed to be capable of starting from cold condition and brought to full power in the order of 1.5 to 2 minutes from initiation of start. This means the entire engine can be subjected to tremendous thermal transients. In the combustor, change in metal temperature may be over 2,000°F in the period of one minute. A major combustor liner design consideration is thermal growth. In the combustor shown in figure 7-23, the liner is mounted in the compressor rear frame by ten equally spaced radial mounting pins through the frame casing and liner cowl forward ofthe fuel nozzles plane, and growth is accommodated by the fish mouth seals at the rear of the combustor.

HIGH-PRESSURE TURBINES

The high-pressure turbine(s) drives the compressor(s). Typically, the compressor drive turbines must extract 1.5 to 2 times the gas kinetic energy and power compared to the power extracted by the load driving power turbine. While turbines may be of axial or radial flow design (see APU description in "Auxiliary Gas Turbine" section on page 7-94), for large shipboard engines, they are axial flow. Turbine sections are composed of vanes which establish the flow angle and velocity ofthe gas into each stage ofturbine blades (also referred to as buckets) mounted on the rotor disks. The major high-pressure turbine design concern is cooling the materials well below the gas temperatures which may have peaks above 2,500°F. Figure 7-24 shows two basic methods of blade and vane cooling. With film cooling, compressor discharge bleed air is routed through the core passages of the airfoil and metered out the leading edge and airfoil surfaces to form a boundary layer of cool air, which insulates the surface from the gas temperature. With impingement cooling, air is selectively directed tQand concentrated in areas where cooling is most needed, using inserts with small metering holes inside the airfoil.

Figure 7-25 shows how the LM2500 high-pressure turbine rotor, disks, blades, and vanes are provided cooling air. Air cools the inner and outer surface of the rotor shaft connecting the turbine to the compressor. It sweeps the front and rear faces of the stage one disk and the front face of the stage two disk. Through holes in the disk rims, air enters the shanks of the blades for each stage.

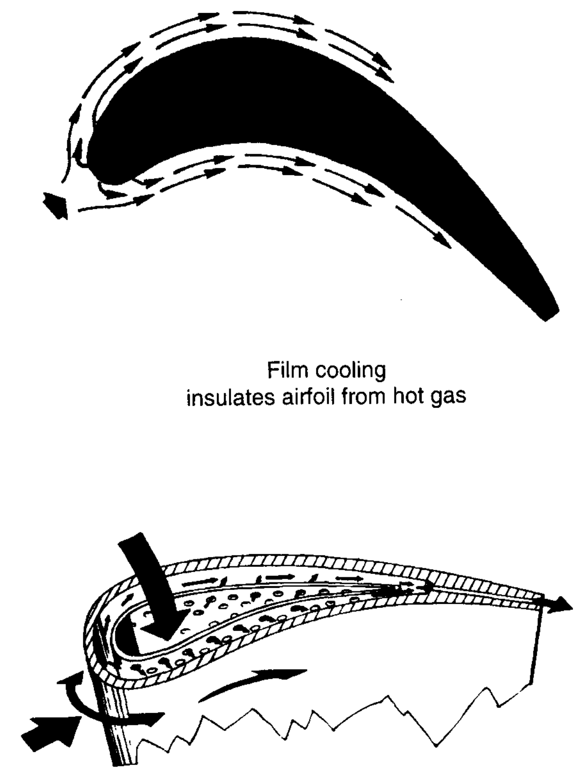


Figure 7-24. Turbine blade and vane cooling techniques

Turbine blades in both stages are long-shanked and internally air-cooled. Use of long-shank blades provides thermal isolation of dovetails, cooling air flow paths, high damping action for low vibration, and low disk rim temperature. The turbine blades are coated (typically using a plasma spray) to improve resistance to erosion, corrosion, and oxidation.

The high-pressure turbine rotor is cooled by a continuous flow of compressor discharge air that passes through vanes in the compressor rear frame pressure balance seal and turbine shaft. This air cools the inside of the rotor and both disks before passing into the turbine blades. The blades of both stages of the high-pressure turbine are cooled by compressor discharge air that flows through the dovetail and through the blade shanks into the blades. First-stage blades are cooled by internal convection and impingement and external film cooling. The convection cooling of the center area is accomplished through a labyrinth, within the blade. The leading edge circuit provides internal convection cooling by airflow through the

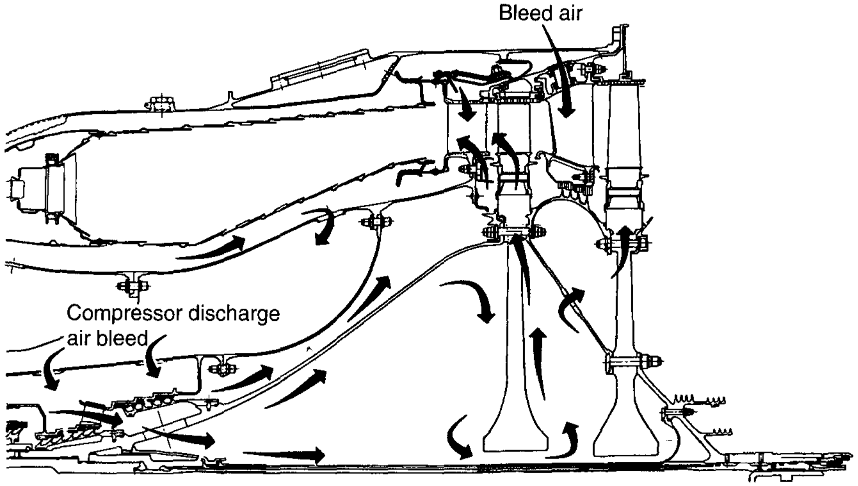


Figure 7-25. High-pressure turbine rotor cooling.
Courtesy General Electric Company.

labyrinth, then out through the leading edge, tip, and gill holes. Convection cooling of the trailing edges is provided by air flowing through turbulence promoting pins and the trailing edge exit holes. Stage two blades are cooled by convection, with all of the cooling air discharged at the blade tips.

Both stages of the high-pressure turbine nozzle assemblies are convection and impingement air cooled and are coated to improve resistance to erosion, corrosion, and oxidation. The first-stage nozzle is also film cooled.

Figure 7-26 shows typical characteristics of cooled turbine blades. These high-pressure turbine blades are made of vacuum-melt cast nickel-chrome-cobalt alloys to obtain the strongest high temperature crystal or grain structure. Small grain size is better at cold temperatures (as in compressor air foils) and large grain size is better for high temperatures. Modern blades are gradually evolving from conventionally cast to directionally solidified columnar grain and directionally solidified single grain or crystal. Early blade cooling passages were typically drilled (often chemically) leaving round, smooth passages. With modern casting methods, intricate shaped cores (leached out after casting) are used, allowing specifically shaped passages with passage bumps (turbulence promoters) to increase cooling effectiveness and blade structural strength. Air entering the blades is channeled up two central passages and makes two 180-degree turns into serpentine passages moving toward the leading and trailing edges. Air at the leading edge is impinged into the forward passage and out the nose and gill holes, providing film cooling. By having the cooling air pass through the center passages of the blade first, the center is kept cooler than the leading and trailing edges. This keeps leading and trailing edges in com-

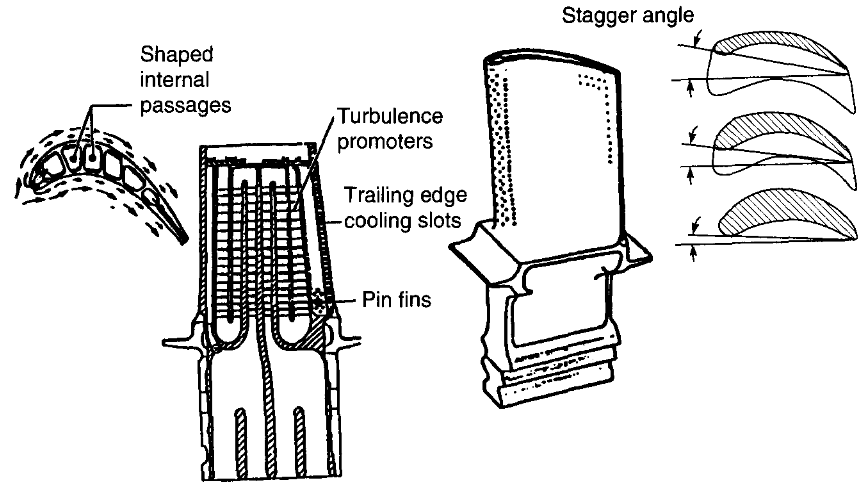


Figure 7-26. High-pressure turbine blade characteristics.
Courtesy General Electric Company.

pression, which reduces the formation and propagation of cracks due to wear, erosion, fatigue stresses, or impact damage. Blading can be impulse and reaction design. In figure 7-26, it may be seen that at the airfoil base, the blade has a small percentage of reaction which increases toward the tip as seen by the increase in the stagger angle. This method of reaction control more evenly distributes the blade flow along its length. Note also that the blade has a thin edge tip. This is a rather typical practice for blades without tip shrouds in both compressors and turbines and is called a squealer tip, undoubtedly from a steam turbine heritage. With gas turbines, the purpose of the squealer is not to make noise to alert the operator. Because the rate of heating and cooling of the thin lightweight casings may differ from the larger mass rotors and because minimum tip clearance for all blading is essential for low losses and efficiency, the squealer tip allows low stress abrading of the blade tip at the casing seal, when extreme cooling differences occur. Blades in large turbines are typically attached to the rotor disks using a fir tree design.

POWER TURBINES

As seen in figures 7-15 through 7-18, propulsion power turbines may have from two to six stages, the number usually being three or more for engines above 20,000 hp. The simplest method of changing rotation direction has been to use reversing reduction gearing or reversing hydraulic couplings. The use of controllable and reversible pitch (CRP) propellers negates the need for reversing turbine direction to reverse ship direction. Power turbines are usually unidirectional but some manufacturers offer left- and right-handed sets of blades and vanes for twin-shaft ships. This is more often handled with an idler element or more preferably a rotation changing element in the reduction gearing. Power turbine blades and vanes have not required air cooling as gas entry temperatures are usually below 1,6000F. Because power turbines are typically larger diameter than gas generator turbines, which allows lower output rpm, the blades are longer and blades are often fabricated with interlocking blade tip shrouds with a circumferential race for establishing a seal into the casing shroud. These tip shrouds dampen blade vibration and reduce tip leakage.

The power turbine for aeroderived engines may be adapted from an aero fan drive turbine, or may be a lighter weight industrial design derived expressly for marine and industrial service.

Power turbine horsepower to torque relationship is

$$\text{hp} = \frac{2\pi \times \text{rpm}\,(N) \times \text{torque}\,(T\ \text{lb}-\text{ft})}{33{,}000\ \text{lb}-\text{ft/min}}$$

or torque equals

$$T = \frac{\text{hp} \times 5{,}252}{N}$$

From a gas turbine performance curve and a plot of ship's power requirements versus rpm, similar to figure 7-11, torque for any given operating condition may be calculated.

As typical of steam turbines, when the gas generator energy input (or steam input) is held constant for rated rpm, if the rpm is decreased (by increasing torque) torque increases linearly to where, at zero rpm, the torque is twice that at rated rpm; put another way, turbine stall torque is twice engine rated torque.

EXAMPLE 7-3: From the performance curve (fig. 7-11), determine the engine torque (T_1) at 30,000 hp and N_1 at 3,600 rpm. Determine engine torque if load were applied sufficient to reduce rpm to N_2 = 1,200 rpm.

Solution 1

$$\text{Torque at 30,000 hp} = \frac{30{,}000 \times 5{,}252}{3{,}600\ \text{rpm}}$$

$$T_1 = 43{,}767\ \text{lb-ft}$$

$$\text{Torque at 1,200 rpm} = T_2$$

$$\frac{T_2}{T_1} = 2 - \frac{N_2}{N_1}$$

$$T_2 = \left(2 - \frac{1{,}200}{3{,}600}\right) \times 43{,}767 = 72{,}945\ \text{lb-ft}$$

The steady-state overtorque design margin of a ship's reduction gearing may well be less than 50 percent of rated torque. As such, for conditions where high torque at low rpm is desired (for example, trying to free an ice-bound or partially grounded propeller), care should be given that the gas turbine does not overtorque other components in the transmission system. Frequently, there is a torque limiting function in the gas turbine control system for this purpose.

Engine manufacturers may have their own reasons for limiting the amount of time and torque applied to break loose a power turbine stalled by excessive load on the output shaft or operating the gas generator in a standby mode while the propulsion shaft is braked to zero rpm. For example, while the turbine is stalled, flow in the power turbine is very turbulent and there is no work produced so there is no temperature drop across the turbine. As such, an exhaust system designed for 1,000° to 1,lOO°F gases could receive gas temperatures in excess of the 1,500°F at gas generator higher speed settings.

EXHAUST COLLECTORS

A variety of exhaust collector designs may be seen with the engines shown in figures 7-15 through 7-18. The purpose of the collector is to direct the engine exhaust gas into the ship's exhaust ducting as efficiently as practical. Collectors consist of a diffuser to recover gas velocity energy and a collector chamber which, for shipboard, is usually a 90-degree elbow. For hot-end drives, the collector system must have a coupling shaft tunnel which is usually insulated and air cooled to protect the coupling shaft. The collector design is usually a compromise among efficiency, length, weight, and shape, and is sometimes customized to fit the constraints of the application. In addition to exiting exhaust gas, the collector may also accommodate air discharge from engine air-oil separators, and/or the discharge from compressor bleed systems as for engines, as shown in figures 7-17 and 7-18. Figure 7-16 shows a collector using turning vanes in transitioning the gas from the diffuser to the collector. Figure 7-18 shows the use of a large diameter collector, shaped for good efficiency in a short length.

Exhaust collectors are usually separately mounted from the engine. The diffuser is often engine-mounted and the two interface with a flexible sealing device. In figure 7-17, the TF40 engine is a cold-end drive and the manufacturer offers a wide variety of exhaust collector designs which the installer may use for particular application arrangements.

POWER TURBINE COUPLING SHAFTS

Power turbine coupling shafts rotate at high speeds and may be subject to wide ranges of angular, offset, and thermal changes to their desired alignment. The two typical styles of design are crowned splines and flexible diaphragms. For hot-end drives, where the coupling shaft may traverse several meters of the hot environment of an exhaust elbow tunnel, flexible diaphragms may be advantageous in that they require no lubrication. The types of coupling misalignment are shown in figure 7-27. Flexible diaphragm coupling shafts are shown in figures 7-15 and 7-16.

In all new application designs, it is essential that the complete power train has a torsional, lateral, and usually an axial vibration analysis.

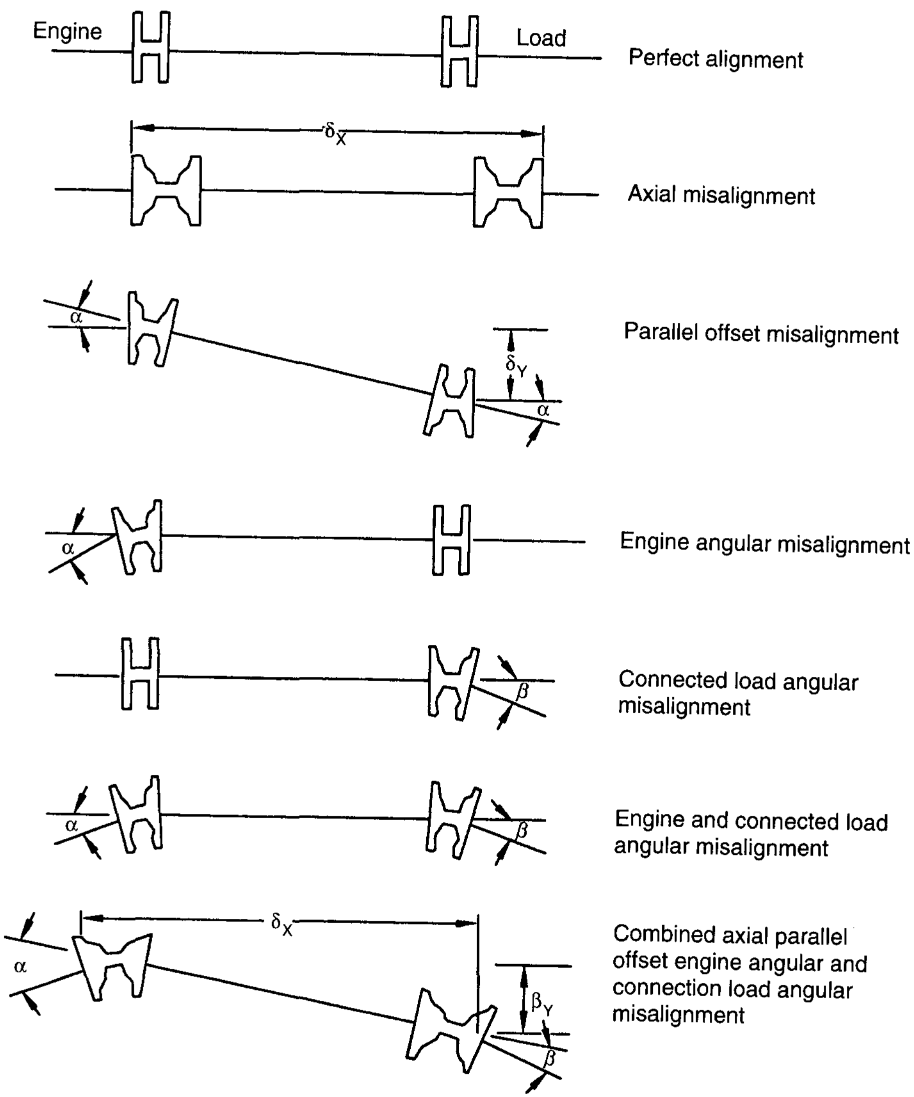


Figure 7-27. Types of flexible coupling misalignment. Courtesy General Electric Company.

These analyses are approved by the gas turbine manufacturer, and may be performed by, or in conjunction with, the turbine and the gearing manufacturers, including data from the propeller drive system.

Gas turbines are frequently mounted to a base that is mounted on elastic mounts to the ship's foundations. This serves to attenuate the transmission of vibration into the hull and, in the case of naval ships qualified for high shock, significantly reduces transmission of shock into the gas turbine from underwater explosions. With soft mounting, the flexible coupling shaft must accommodate larger changes in position between the engine and the gearbox. For example, with changes in power, the engine's countertorque moves the engine centerline in an arc about the gear input centerline.

The ideal situation is to have the coupling shaft operate with no axial tension or compression load, or angular misalignment. In the practical case, this is impossible because of changes to engine-to-gearing centerline due to varying operating temperatures and horsepower, alignment tolerance, and environmental conditions such as hull flexing due to ship motion. The effect of some of these factors can be minimized by calculating the expected misalignment for the range of operating conditions and then offsetting the gas turbine at the cold static condition to achieve an optimum alignment over the operating range of power which covers compensatable variables. These types of calculations are usually given consideration when the system designer and engine manufacturer develop the engine alignment procedure for a new application.

For mechanical drive applications, generally, the flexible coupling shaft has as much, or more, transient torsional margin than the gearing system while being able to accommodate the axial and lateral displacements anticipated with soft mounts. However, for electric generator drive, a short circuit generator condition could possibly develop two to three times rated load torque, depending upon the turbine rotors mass moment of inertia. For this type application, by hard-mounting the engine and generator on a common stiffbase, a coupling shaft with less angular deflection capability, but greater torsional capability, would be used.

To assure maintaining the flexible coupling shaft critical speed above the maximum allowable overspeed, the driven machinery bearing stiffness and shaft overhang length from this bearing is taken into consideration by the system designer in the system vibration analysis.

For hot-end drive turbines, care must be taken to assure that insulation and cooling air provisions ofthe exhaust duct shaft tunnel are not inhibited so that coupling shafts operate within their design temperature limits.

ENGINE MOUNTED ACCESSORY SYSTEMS

Marine gas turbines are frequently supplied with the components required for fuel control, lubrication, ignition, and starting mounted on, or adjacent to, the engine (a holdover or emulation of aeroderivative gas turbines). While the installation section of this chapter will discuss these systems with respect to their associated ship service systems, the following discusses components which may be included in the gas turbine engine scope of supply. All of the gas turbines shown in figures 7-15 through 7-18 have accessory gearboxes which connect to the high-pressure gas generator shaft to drive accessories and accept drive input from a starter motor. Not shown in figure 7-16, the FT8 accessory gearbox mounts below the engine casing between the low-pressure and high-pressure compressors.

Fuel Components

The accessory gearbox drives the engine's main fuel pump. This is a high-pressure pump which must increase fuel pressure from the ship's service supply system to pressures sufficiently above compressor discharge pressure to atomize the fuel across the fuel nozzle system pressure drop.

Fuel pump discharge may also be used as an energy source for hydraulic actuators, such as variable stator vane systems, although the engine lube oil system often provides these functions. Other engine-provided fuel components may include last chance filters; hydromechanical fuel control or main fuel control valve for scheduling fuel flow; engine fuel shutoff valve(s); bypass, fuel purge, and pressurizing valves; and, in a few cases, heat exchangers, where engine fuel is used to cool the lube oil.

Dual Fuel Components

Many gas turbine models used for marine propulsion have industrial applications using natural gas fuel and gas/liquid dual fuel systems. These engines are available for LNG carrier propulsion. Differences in the fuel system would be the replacement of piping, control and shutoff valving, fuel manifolds, and fuel nozzles to accommodate dual gas and liquid fuel components. The engine-driven liquid fuel pump would be replaced by an electric motor-driven pump and the engine may add a hydraulic pump using lube oil to position variable stators and the fuel metering valves. Discussion of the ship support equipment for delivery of LNG boil-off gas and liquid fuel to the gas turbine is given in "Installation, Fuel Supply System" on page 7-76.

Dual fuel systems typically can be operated on 100 percent gas fuel, 100 percent liquid fuel or a combination of gas/liquid ratios of *10/90* to *90/10*. Starting is done using 100 percent of either fuel. Operating procedures would be subject to the regulatory body requirements at the time of installation.

Lube Oil Components

The accessory gearbox drives the engine main lube oil pump. This pump usually provides the lubrication and cooling of engine bearings and accessory gearbox and may be a hydraulic pressure supply source to actuators for such functions as compressor variable vane positioning, bleed system actuation, and modulation. The main lube oil pump may contain scavenge pump elements for pumping the oil away from antifriction bearing oil sumps. For journal bearing equipped machines, there is usually a separate, electric motor-driven supply pump for prelube and postlube service. Where an aeroderivative gas generator is used with an industrial style power turbine with journal bearings, there may be separate lube systems for these. Other engine provided lube oil system components may include supply and scavenge oil filters, often with delta pressure sensors and metallic chip detectors; oil coolers with temperature regulating bypass valves; lube pressure control valves; air/oil separation system (some mechanically driven by the accessory gearbox) with venting; and lube oil tank(s).

Starter

For shipboard gas turbines, the starter motor is traditionally attached to the engine's accessory drive gearbox and drives into the high-pressure compressor rotor. Starters frequently have two requirements: (1) the high torque short duration of starting the engine and (2) the lower torque longer duration motoring of the gas generator during fluid washing of the engine to remove salt and dirt deposits from the gas turbine blades and vanes, or to uniformly cool an engine that has had an emergency shutdown from high power operation.

There are three common types of starter motors: electric motor, hydraulic motor, and pneumatic motor. The electric motor is most common on smaller size engines. Engine manufacturers usually offer an option oftwo or three different start systems. For example, the LM2500 and FT8 engines (figs. 7-15 and 7-16) offer options of hydraulic and pneumatic, the Solar Taurus (fig. 7-18) offers electric and hydraulic, and the TF40 (fig. 7-17) offers all three. The gas turbine start time varies somewhat with the size and type ofthe gas turbine and type of starter, but generally gas generators can be brought to self-sustaining idle speed within one minute of starter engagement and from idle to full gas generator power within a half minute.

The electric starter for the TF40 is a 24-volt dc automotive-type starter driving through a sliding spline engagement mechanism and draws a maximum of about 27 kW at 1,000 amps dropping to about 10 kWat 1,550 amps if a 26 Vdc maximum terminal voltage is provided. The ship service supply may be battery or ac-powered motor generators or transformer rectifiers. The electric starter for the Solar Taurus is an ac system using a conventional squirrel cage induction motor running through an overrunning clutch powered by a sine-coded, pulse-width modulated power supply that generates an adjustable voltage/frequency three-phase output. The maximum load imposed by this system is approximately 75 kW.

Hydraulic starters typically consist of a variable displacement or radial piston hydraulic motor with an overrunning, self-synchronizing clutch attached to the accessory gearbox. Power supply is a hydraulic pump which may be powered by an electric motor or from a power takeofffrom an auxiliary diesel or ship service diesel-generator set. The system typically includes pump and control, a makeup fluid tank or reservoir, filters, and a hydraulic fluid cooler. Depending upon the size ofthe gas turbine and the system's characteristics, maximum flows may be from 20 to 60 gpm and system operating pressures may require a pressure drop across the starter motor from 2,000 to 4,500 psig. The system operating fluid may vary among systems from the engine lube oil to special hydraulic fluids. Hydraulic systems may have the advantage of providing virtually unlimited time in the motoring mode whereas electric and some pneumatic systems may require intermittent motoring operations to preclude overheating the starter motor.

Pneumatic starters are by a large margin the least weight of engine mounted equipment. The starter is a low-pressure, high-volume air turbine equipped with an overrunning clutch and frequently a controlled opening rate supply on/off valve and an integral speed switch used to activate shutoff of the starter air supply when engine self-sustaining speed is reached.

Aboard ship, the motivating fluid is air which usually exhausts directly into the gas turbine machinery enclosure. Pneumatic starters are popular in gas pumping industrial service as there is an abundant supply of pressurized gas. When gas is used, the starter turbine exhaust is properly vented to meet safety and environmental regulations. The supply pressure for these starters may range from 20 to 40 psig and flow rates, depending upon the gas turbine inertia, may be from less than 1/2 pound per second to over 4 pounds per second, the flow also being dependent upon the temperature ofthe air to the starter. Inlet air must be filtered to be clean and dry and at temperatures usually below 350°F, if the starter has aluminum components. The source of ship provided air supply may be several options: high-pressure stored air, a centrifugal load compressor driven off a diesel-generator set, an auxiliary power unit APU (described in the auxiliary gas turbine section), and the use of compressor bleed air from other gas turbines operating in the installation.

Ignition System

Typically, as the starter accelerates the gas generator high-pressure rotor, fuel and ignition are initiated at about 25 percent of gas generator idle rpm and ignition is cut off as idle rpm is approached and the engine combustion is self-sustaining.

Systems are typically electric, ranging from an air-assisted torch ignitor, ignited by a conventional spark plug, to high-energy systems using capacitor discharge ignition exciters and surface gap-type spark ignitors. The ignitors are often designed with internal air passages and vents for cooling and prevention of carbon accumulation. A typical high-energy system may use two ignitors and each may have a 120 Vac 60 Hz, 1.3 amp input to the ignition exciter. Power is transformed, rectified, and discharged as dc energy pulses through coaxial shielded leads to the spark ignitors with an output of 15/20 kV, 2 joules, and 100,000 watts. This energy level is lethal, and output from the spark exciter, leads, or ignitor should not be contacted by personnel.

ENCLOSURES AND BASES

Large gas turbines are usually enclosed in their own compartment. This may be built around the engine as part of the ship's compartmentation, or the gas turbine may be provided with a mounting sub-base and enclosure as a self-contained complete assembly frequently called a module.

There are five major functions of gas turbine compartments or enclosures.

1. They are designed to direct the cooling air path and distribute cooling flow where required.
2. They attenuate noise transmission.
3. They protect personnel from hot surfaces.
4. They contain fire and concentrate extinguishing agent effectiveness.
5. They are a backup containment shield for flying objects, for example, a fragment of a ruptured rotor disk not fully restrained by the engine case.

Figure 7-28 shows the features ofthe base/enclosure available with the LM2500 gas turbine which is shown in figure 7-15. While content and location of functions may vary among different gas turbines, the features shown are typical of those desirable for a self-contained gas turbine.

This base, enclosure, and lube storage module group provides for the following general functions:

- enclosure inlet plenum designed for optimum flow characteristics into the compressor
- cooling flow passage with a cooling air inlet shutoff damper, distributing and maintaining air velocity around the gas turbine, into exhaust
- flexible joint assemblies for inlet air, secondary cooling air, and exhaust
- thermal isolation
- mounting support system of the gas turbine to the base assembly
- airborne noise attenuation
- shock and structureborne noise attenuation (for military applications)

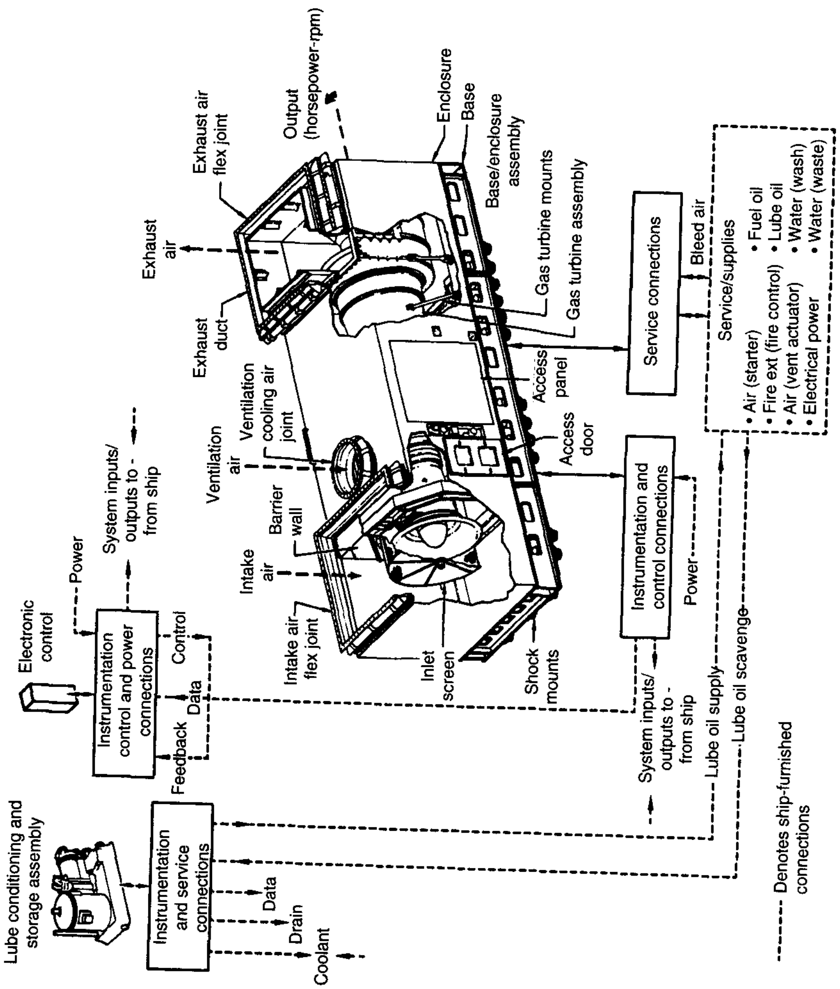


Figure 7-28. Base enclosure features. Courtesy General Electric Company.

- controlled drainage for waste water, fuel, oil
- fire detection (both temperature and flame) and extinguishing piping and nozzles
- icing condition detection
- off-engine mounting of fuel purge, starter supply, and bleed air valving and duplex oil filter
- enclosure heating
- electrical receptacles and lighting system
- instrumented, piped, and tested gas turbine package
- fixtures for mounting removable support tooling for onboard repair and engine removal/replacement
- leak-tight base penetrations for all piping and wiring into and from the module

The inlet air plenum is separated from the rest of the enclosure by a barrier wall. An inlet air screen is mounted in front ofthe bell mouth supported by the barrier wall. This screen prevents entry of large foreign objects (minimum dimension over 1;4 inch).

The fire detection system consists of three ultraviolet flame detectors with signal conditioner and two temperature switches. The fire extinguishing system is composed of piping and nozzle(s) for release of C02 or Halon, mounted within the base enclosure assembly. A manual inhibit switch precludes release when personnel are in the module.

Flex joints are provided at the air inlet, exhaust gas discharge, and cooling air inlet. These provide airtight and noise attenuating seals between the enclosure and the ship's ducting. At the cooling air inlet, a set of quick-action louvers is provided to shut off cooling air in the event of fire or when gas turbine is secured. Cooling air may be pumped either by a fan in the ship's cooling duct or by an eductor nozzle on the gas turbine exhaust elbow. The cooling air is usually combined with the exhaust gas which negates the need for a separate cooling air discharge duct. If a recuperator were to be used, cooling air would be routed to bypass the recuperator. All three flex joints are capable of accommodating the relative motion between the soft mounted (shock mounted) module and the ship's ducting. The shock and structureborne noise attenuating system consists of 32 shock mounts capable of accommodating shock inputs of over 230 Gs.

An enclosure heater may be supplied to maintain a comfort level during maintenance work and can serve to ensure fuel lines are kept above fuel waxing temperatures.

An ice detection system detects potential inlet air icing conditions when the relative humidity is greater than 70 percent and inlet temperature is less than 4.5°C (40°F). Then by adding enough heat to the inlet air to keep humidity below 70 percent, the formation of icing is prevented.

A bleed air system ducts compressor discharge air through an electromechanical shutoff valve to the base interface for shipboard uses.

Fuel is piped from the base to the engine mounted fuel pump. Instrumentation sensors and transducers are wired and piped into base connections. A fuel purge valve and piping allows bypass of cold fuel to the fuel drain return line if fuel viscosity is too high for starting.

Starter shutoff and pressure regulator valve, piping, and wiring are provided from base connections to the gas turbine mounted starter. The starter operates with ship provided fluid supply. The enclosure is provided with eight overhead lights plus one light on the base. A lube supply duplex filter, antistatic leak check valve, pressure transducers and associated wiring, and piping are provided.

For this particular enclosure assembly, storage and conditioning of the lube oil is performed by a separate assembly, but many gas turbine manufacturers include these provisions within the base-enclosure. Their scope and functions in either case are similar, that is, to provide lube oil cooling, filtering, deaeration, and storage. See "Installation, Lube Oil System" section on page 7-78 for further details.

The electronic control system may be mounted as part of the base-enclosure, but frequently it is provided as a free standing console in the ship's control station, which may be supplemented by bridge and remote station subconsoles. Controls are discussed in detail in the next section.

The outer walls of the enclosure have personnel access doors. Maintenance and engine removal panels may be included in the compartment walls or overhead. Noise outside this module, and typical of module design objectives, is less than 88 dBa at full power, well below OSHA limits for exposure eight hours per day. The heat output of this module into the engine room is less than 19,000 Btu/hour, or the equivalent of about 2 kW. The outside wall temperature in a properly ventilated engine room is less than 125°F.

Controls

GENERAL DESCRIPTION

The basic control of a gas turbine may be only a small part of the considerations regarding the engine integration into a ship's propulsion control system. It is generally impractical for operating personnel to be stationed inside the engine enclosure, except for pre- or postmaintenance checkout of the engine operation at low power by personnel using proper protective gear and precautions due to noise and engine surface temperatures inside the enclosure.

Control stations for ship propulsion may include a machinery control room with subordinate control stations at the bridge and possibly a local control station next to the engine for emergency operation.

Control and monitoring of a propulsion system gas turbine involves integration with the operation of ship-provided services and delivery systems (for example, fuel supply, cooling air, electric power) and other transmission components. As regards the gas turbine, control is the modulation of fuel flow to meet the desired power and output shaft rpm with the possible comodulation of engine airflow and/or power turbine entry temperature by means of variable compressor stator vanes, compressor bleed or bypass, or variable power turbine nozzle vanes. The modulation of these functions is performed by actuators which respond to a throttle command signal biased by monitored engine parameter sensors to ensure the safety of the engine, other equipment, and personnel. For a propulsion system, monitoring of other propulsion component parameters may be employed to ensure mutual safe interaction among the gas turbine and the rest of the system.

Safety monitoring incorporates the use of proper system interlocks and the display of critical engine operating parameters with provisions for automatic alarm and, in some instances, automatic shutdown, when safe limits or conditions are exceeded.

The motivating power to the actuators could be mechanical, pneumatic, hydraulic, electric, or electronic. For small, simple systems, the response to the sensor data and sequencing of events may be done manually by trained personnel providing the sequencing logic. However, with large, complex and often multiengine, multipropulsor systems, electronic logic systems are virtually necessitated for safe and proper system control. Such engine controls are frequently referred to as FADEC (full authority digital electronic control).

The gas turbine control system must set and maintain the engine output as commanded by the operator and it must manage transient conditions. Typical transient conditions include the proper sequencing of procedures for starting the engine and bringing it to idle, sequencing of shutdown procedures, proper rates of acceleration and deceleration, reversal of thrust of the propulsor (propeller or waterjet), and the switching of engines (the addition or removal of an engine when two or more engines drive into a common gearbox).

Safety protection must be provided to preclude damage and injury caused by propulsion component interaction. For example, the inboard propeller of a multipropeller ship in a tight, high-speed turn could overtorque the gearing, or an engine or system component or auxiliary system malfunction such as a clutch failure might drop load instantaneously.

Basic control of a free turbine system is the control of the rpm of the gas generator, which equates to the gas energy supplied to meet the output shaft requirements for power. For ship propulsors, there is a predictable steady-state load as a function of transmission system input rpm.

Figure 7-29 shows a functional diagram of typical engine mounted fuel components and gas generator control. Upon applying energy to the starter motor, the gas generator driven fuel pump provides fuel flow to the

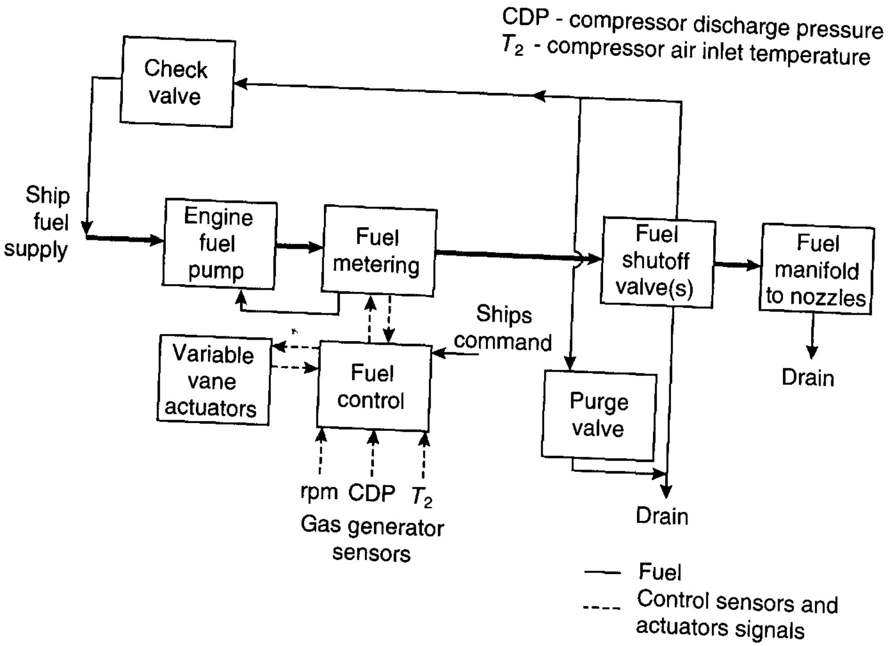


Figure 7-29. Functional diagram, engine fuel and control components

fuel flow regulating device. The fuel control typically senses gas generator rpm, compressor inlet temperature *(T2),* and compressor discharge pressure (CDP) and establishes suitable fuel flow schedule with synchronized scheduling of any variable stator vane and compressor bleedlbypass systems for starting, accelerations and decelerations, and steady-state operation without encountering compressor stall or combustor flameout.

Because the fuel pump always pumps more fuel than the engine requires, in order to have reserve capacity to accommodate acceleration and rapid changes in load, the fuel metering device may bypass excess fuel back to the pump. In this diagram, the fuel pump has a centrifugal boost pressure element which can accommodate a slight negative head from the ship supply system for short periods, and a high-pressure gear element to develop the high pressure for the fuel system. Here, the fuel metering device bypasses fuel between the pump stages. The fuel pump output pressure must accommodate the pressure drop across all system components including the fuel nozzles and atomize into the combustor pressure which may be 20 to 30 atmospheres. As such, peak fuel pressure in a fuel pump may well exceed 1,000 psig and pumps may incorporate internal high-pressure relief valves.

The control contains a governor to maintain a constant gas generator speed for the power commanded, and a gas generator overspeed governor

function. The engine fuel system will have a quick closing fuel shutoff valve near the fuel manifold. This valve (or possibly two arranged mechanically in series but electrically in parallel for redundancy) is typically a spring-opposed electric solenoid which is open when energized. Certain of the safety monitoring sensors are wired to interrupt the current to the solenoid(s) and the spring closes the valve(s) causing immediate engine shutdown in milliseconds. The most critical need for this valve is for the power turbine overspeed sensor. With an instantaneous complete loss ofload at high power, such as a gear input shaft or a clutch failure, a power turbine can accelerate at many thousand rpm/sec. Thus, overspeed shutdown must be nearly instantaneous. The shutoff valves bypass fuel to the pump inlet when they are closed and cease bypass when they are open.

There may be a fuel purge valve which can drain the fuel shutoff valve bypass line. This may be opened prior to engine start to drain low-temperature fuel, which is below the fuel temperature required for proper fuel viscosity for starting the engine. Purge fuel quantity is typically no more than several gallons and drains to a waste drain holding tank.

START AND STOP SEQUENCING

The start sequencing function ofthe control sequences energy to the starter. At about 25 to 30 percent of gas generator idle rpm, fuel and ignition are applied. At about 50 percent idle rpm, ignition occurs and combustion gases help supplement starter torque in accelerating the gas generator. At about 90 percent idle rpm, ignition and starter supply are cut off and the engine self-accelerates and stabilizes at idle. At this condition, the unloaded power turbine windmills below its maximum speed. The start-stop sequencers may activate fuel drain valves to open during engine coast down after shutdown and to close before initiation of a start. Automatic start sequencing may also sense such functions as flame detection, turbine gas temperature, and fuel manifold pressure. With an elapsed time function, the control will abort a delayed or "hung" start and a start where gas temperatures are excessive or if engine flameout (or blowout) occurs.

For a normal engine shutdown, the engine is brought to idle power by the throttle command. It is normal procedure to have the engine operate several minutes at idle power before activating the stop button, which simply shuts off electric power to the fuel shutoffvalve(s). The reason for the time at idle is to reduce the chance ofrotor-to-stator rubs. Aeroderived engines, particularly, have light mass engine casings, cooled by engine compartment fan flow. These typically shrink in diameter faster than the higher mass rotor assemblies after rotation has cased. Idle operation reduces the relative temperature difference between rotor and stator parts. In case of a manual emergency shutdown or an automatic shutdown from power, it is typical to motor the gas generator on the starter, after it stops rotation, to cool the rotor components.

POWER CONTROL

Thus far, the discussion has been about regulating fuel to establish a desired gas generator rpm. For a given gas generator rpm, there is a resultant gas flow and temperature or gas horsepower. Ship speed is essentially proportional to propulsor rpm and propeller power is predictable for propeller rpm. Therefore, by setting a gas generator rpm, which gives a desired propeller rpm, and resultant ship speed, the output speed would not change unless load changed. By contrast, in the case of a generator drive where load may vary widely but rpm must be held constant, it may be necessary for the power turbine rpm sensor to bias the throttle input signal so that fuel flow is adjusted fo govern a constant power turbine output shaft rpm. This is termed "speed governing." With load sensing, speed control can be virtually isochronous or regulated to a specified droop.

For ship propulsion, there can be continual small variations and possible sporadic larger variations in power. In a seaway, as the ship enters a wave, power demand increases, and in the wave trough, the power demand decreases. Should the seaway be sufficient to partially broach the propeller, a more significant series of power changes would occur. With speed control, for each power perturbation, there is modulation offuel flow with resulting changes in firing temperature and modulation of stator systems control devices. This continuous short-cycling accentuates engine wear and reduces the time between maintenance. By setting gas generator rpm, which is essentially a constant power input to the power turbine, there would be small changes in propulsor rpm, but the ship's average speed would be virtually unaffected. This type of control is termed "power governing" and is normally used for ship propulsion. The United States Navy has conducted extensive study and has opted to use power governing routinely. With an optional output speed signal loop biasing the power control system, speed governing can be selected for special situations.

When power governing is used, it is usual for the power turbine to have a topping governing function. The purpose of the topping governor is to prevent a large drop in propeller load from causing sufficient power turbine overspeed to result in an overspeed trip (shutdown) of the gas turbine. For example, if the ship were cruising at nearly maximum speed and power, say 98 percent propulsor and power turbine rpm, and the load suddenly dropped, the power turbine rpm would increase to over 100 percent. The topping governor would cut in at about 102 percent, and bias the power control command to ramp down fuel flow until power turbine speed is below 100 percent and prevent the power turbine from reaching trip speed which is nominally in the order of 110 percent rated rpm. Should a ship system be operating at a speed in a seaway where the topping governor fuel cutback is frequent, depending upon the exigencies ofthe situation, selection of a lower propulsor/power turbine power governed rpm would be in order.

Figure 7-30 represents typical propulsion system control components for a gas turbine powered merchant ship. The figure indicates alternative systems for integrating gas turbine control with either a fixed pitch propeller (FP) system with reversing reduction gearing or a CRP propeller system. With the FP system, engine power/speed is sequenced in conjunction with sequencing and actuation of gearbox clutching offorward and reverse elements and possible use of a shaft brake. With the CRP system, engine power/speed is sequenced with modulation ofthe propeller oil distribution (O.D.) box which actuates and controls the angle ofthe propeller blades. It is common for a gas turbine control system to limit or modulate the engine power to avoid exceeding limits of other propulsion components. Figure 7-31 shows the electronic control of the throttle command signal so that engine response is compatible with rates and limits of other propulsion system components. In this diagram, there is provision for applying variable limit inputs to power turbine acceleration limiting, power turbine speed (rpm) limiting, and power turbine torque limiting.

For examples of how these limits may be required, assume that after dropping offthe pilot, the ship had been cruising slowly at about 15 percent ship speed (4 percent power). The ship reached open water and the bridge command was made to go to 90 percent speed (73 percent power). If the propeller were a CRP, it may be at a very low forward pitch. It might take the propeller 20 to 30 seconds to go to full pitch whereas the gas generator may be able to accelerate to 73 percent power in about 5 seconds. For this

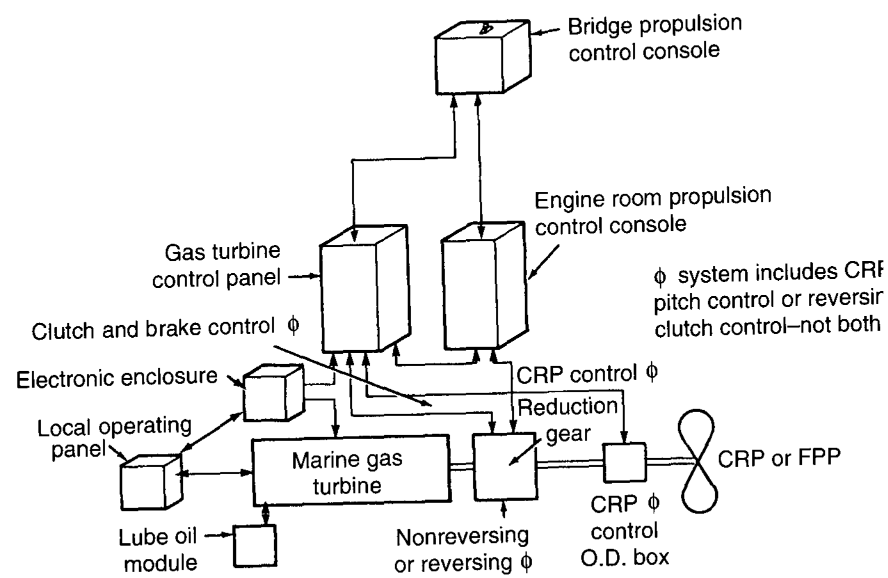


Figure 7-30. Typical ship propulsion system control

condition, the propeller control input might limit the engine acceleration rate to about 30 seconds. If, for example, the ship were at 50 percent speed and the CRP were at full pitch, then the propeller may be able to absorb the increase in power at the rate the engine can produce and the acceleration limit set point would be automatically changed to that best befitting engine life.

A condition where power turbine speed limiting may be varied would be where two gas turbines drive into one gearbox with one output shaft to the propulsor. In the single engine mode, a full-pitch propeller is limited to 50 percent power (the maximum available power from one of two engines) at 80 percent rpm. At 80 percent rpm and full power, the gas turbine would be producing 125 percent rated torque and the turbine operating temperature would be higher than at rated rpm. Thus, for one engine operation, the allowable maximum turbine output rpm might be limited to below 80 percent for either or both protection of gearing torque or turbine temperature to maximize engine mean time between repair (MTBR).

Whereas the speed and acceleration rate limit circuits need only sense power turbine rpm to function in conjunction with limit set point logic from other components, torque limiting requires either the external input from other components of torque or horsepower (for example, a torque meter on the propulsor line shaft) or an estimate of torque from engine parameter sensors. For example, the computer can be programmed with algorithms of the resulting gas temperature and flow into the power turbine by sensing the gas generator total pressures ratio obtained from compressor inlet pressure and power turbine inlet pressure, shown as P_{t2} and $P_{t5.4}$ on figure 7-31 and the inlet air density obtained from inlet total air pressure and total temperature, shown as P_{t2} and T_{t2}. Calculating gas flow rate and gas temperature data, power turbine inlet gas horsepower is established. With an algorithm for power turbine efficiency versus rpm, power turbine output horsepower and torque are calculated.

If the one-out-of-two engine per shaft mode turbine output speed limit were set to limit gearbox torque to 120 percent of full power torque, and a two-propeller ship at this condition went into a tight turn, some of the ship's total load and torque would transfer from the outboard to the inboard propeller and torque limiting would be required and provided by the torque limit loop. It is easily envisioned how an output speed control loop could be added to this diagram which would bias throttle command to maintain a constant shaft speed with varying load if a speed governing option were desired.

For an integrated propulsion control system, the control system designer creates a computer simulation model developed from a ship's performance requirements (for example, crash astern time or distance) and propulsion components limitations and ship hull performance. From this simulation model, a control system dynamic analysis is conducted to obtain the optimum system within the given constraints.

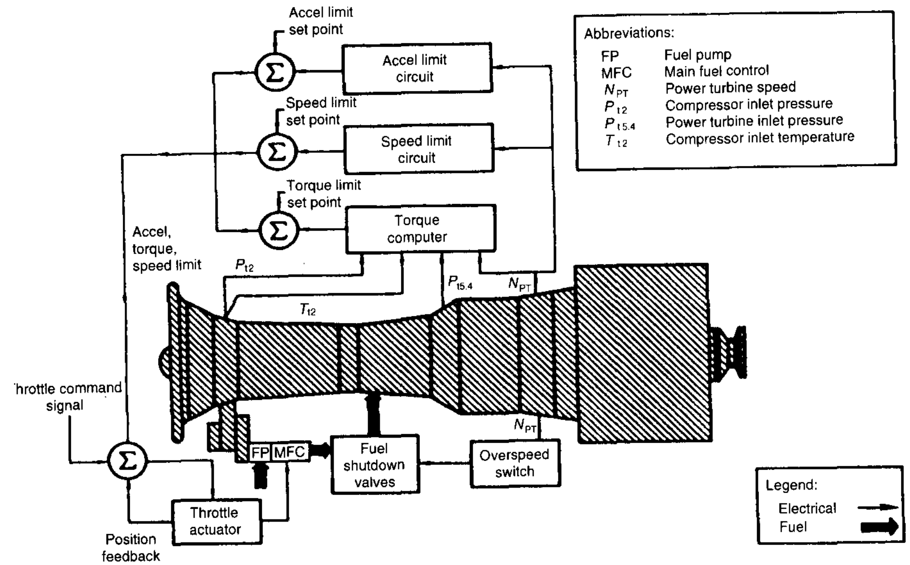


Figure 7-31. Electronic power control system.
Courtesy General Electric Company.

CRITICAL ENGINE OPERATING PARAMETERS

Gas turbine control systems ensure safe operating conditions by sensing many of the engine's operating parameters. When these are out of operating limits they will close an alarm circuit, which indicates the parameter out of limits and, for critical operating parameters, the control system will initiate a throttle cutback or an engine trip (instantaneous shutdown). Where there are both alarm and trip modes, the alarm setting actuates at a level where short-term operation is acceptable while the engine control operator takes corrective action and trip will occur at a more critical setting where engine operation could be immediately dangerous.

When a ship's gas turbine is certified by a regulatory agency (ABS, D.N.V., Lloyd's, G.L., NKK, and others), there is usually an engine model type certification requirement for the design and manufacture of the engine and an installation certification where assurance is made that the engine control monitoring of critical parameters is compatible with the ship type and the other propulsion components. The regulatory agencies establish a list of alarm and shutdown requirements for each ship class. These requirements must be encompassed in the scope of supply of the system, along with other parameters specified by the particular engine manufacturer, which are not necessarily mandated by the regulatory agency.

Table 7-1 shows typical propulsion gas turbine alarm and shutdown parameters and is representative for a gas turbine and enclosure like that shown in figure 7-28. Instrumentation and sensors will vary among gas turbines, influenced by their component designs and their particular operating characteristics.

TABLE 7-1
Typical Propulsion Gas Turbine Alarm and Shutdown Parameters

Type	*Limit Parameter*	*Alarm*	*Trip*
Start	Time to rpm		X
Sequencing	Time to lightoff		X
Power turbine	Gas over temperature	X	X
	Overspeed	Topping governor	X
	Vibration	X	X (to idle)
	Loss of flame	X	X
Gas generator	Overspeed	X	
	Underspeed	X	
	Vibration	X	X (to idle)
Gas turbine	Low oil supply	X	X
	Low oil level	X	
	High oil supply temperature	X	
	High oil scavenge temperature	X	X
	High oil filter Ll pressure	X	
	Low fuel supply pressure	X	
	Low fuel supply temperature	X	
	Low fuel manifold pressure		X
	Fuel valve faults		X
Enclosure	High air temperature	X	
	Fire detection	X	X
	Inlet air ice detection	X	

Where a trip is shown without a previous alarm, a console fault light would indicate the cause of the trip.

The monitoring of gas temperature is typically accomplished by multiple thermocouples usually located between the gas generator and power turbine. They are located at a number of radial locations and may have individual elements at two or more immersion depths. Gas flow has a swirl that varies with power setting such that gas temperature pattern shifts. To realize accurate indication of gas temperature for all operating conditions, the signals from multiple thermocouples at various circumferential and radial locations are usually averaged together into one readout. Selective readout capability of individual thermocouples may be provided in some systems. Vibration shutdown in some cases, as approved by the manufacturer and regulatory agency, may be replaced by an automatic throttle command adjustment to engine idle until acceptable vibration levels are reached.

A document written by manufacturers for each of their engine models is an "Electrical Interface Control Specification." While this document is useful initially for the engine system installer, it is of value aboard ship during any repair or modification to the propulsion/engine control system as it typically contains data for the engine control requirements on

- input requirements
- output requirements
- interlocks
- start/stop sequencing
- alarm of faults
- fault initiated shutdown
- control modes
- power requirements (electric)
- data display

Installation

The installation of a gas turbine into a ship must be planned to allow proper interface with all of the connecting points with the ship and the equipment required to support the operation of the gas turbine. Different gas turbine models will have differing specific interface requirements and different ships will have various constraints placed upon the installation and the support systems that can be made available for the gas turbine.

A ship installation is particularly demanding compared to typical shoreside installations because of a ship's environment. There are significant variations in engine position caused by heave, pitch, rake, trim, roll, list, and yaw which affect mounting loads and fluid control such as sloshing in tanks and pump suction heads. The environment is hostile in that salt and water, both deleterious to gas turbines, can permeate the air and fuel systems. Airborne salt fouls the compressor and causes combustor and turbine section corrosion; water in fuel carries salt and impurities and promotes growth of microbial slimes in fuel tanks (see chapter 8).

Frequently, engine location within the ship as well as availability of space complicate engine installation and removal, accessibility for onboard maintenance and repair, and routing of air ducting from the engine to suitable intake and discharge points above deck. Ship hulls are comparatively limber, and for hogging, sagging, and racking conditions, there may be large relative motions possible at connecting points.

The basic system interface considerations include

- mounting and enclosure
- inlet air
- cooling air
- exhaust

- fuel supply
- lube oil system
- fire detection and extinguishing
- engine cleaning
- engine bleed
- starting system
- vents and drains
- control and electric supply

MOUNTING AND ENCLOSURES

Mounting systems are de'signed to accomplish the following:

1. Accommodate the thermal growth of the engine and the bending of the ship structure without transmitting torsional or bending moments to the engine.
2. Minimize the effect of ship deflection on engine alignment.
3. Provide for adequate chocking and shimming clearances for initial installation and alignment, and realignment after each engine exchange.
4. Avoid resonances between engine and structure.
5. Provide adequate stiffness to minimize engine motion due to ship motions.
6. Maintain forces on mounts, piping to the ship, and duct connections below interface limits for all operating conditions.
7. Minimize structureborne noise transmission.
8. Permit convenient removal for maintenance and repair.
9. Protect engine bearings of a nonoperating gas turbine from damage by the input of vibratory and impact loads.
10. Provide ship-to-gas turbine, or enclosure, connecting piping flexibility to meet the gas turbine manufacturer's interface control drawing load limits for compression, tension, bending, and shear. While this is provided in the initial ship's installation design, care should be exercised that any ship alterations or modifications conform with these requirements.

Gas turbines are usually placed in a form of compartment which may be provided as part of an engine enclosure assembly with the gas turbine or built as ship's structure. In either case, the enclosure usually provides for a controlled flow path of engine cooling air, isolation by insulation of engine heat and engine casing noise from the engine room, fire protection through detection systems and extinguishing agent distribution systems, and other possible features described in the enclosures and bases subsection of "Construction" on page 7-59. Larger engines typically are mounted using yokes, clevises, and links, directly or through semielastic spacers, to the ship's foundation. Alternatively, to attenuate structureborne noise, vibration, or shock, a base-mounted engine is elastically mounted to the ship's founda-

tions. Smaller engines may offer unique opportunities such as the TF40, shown in figure 7-17, where the engine inlet housing module may be flanged directly to the driven equipment.

INLET AIR

Figure 7-32 is a sketch of some of the rudiments of an inlet duct system, a cooling air duct system, and an exhaust duct system.

The basic design of the duct itself should be as straight as possible and made of noncorrosive materials. Where there is no access to remove the gas turbine from the module via the engine room, or no access ways to the outside of the ship to accommodate the gas generator assembly and power turbine assembly, as is frequently the case with naval ships, the area and dimensions of the inlet duct may be dictated by dimensions needed to accommodate the exchange of gas turbines. In any case, ducting dimensions should be large enough to maintain air velocities low enough to avoid flow distortion and large pressure losses. While maximum velocity limits vary, a demonstrated nominal value for large gas turbines has been 40 feet/sec. Duct walls are usually lined with silencing material. Based upon the noise requirements of ship's spaces traversed by inlet ducting, the addition of

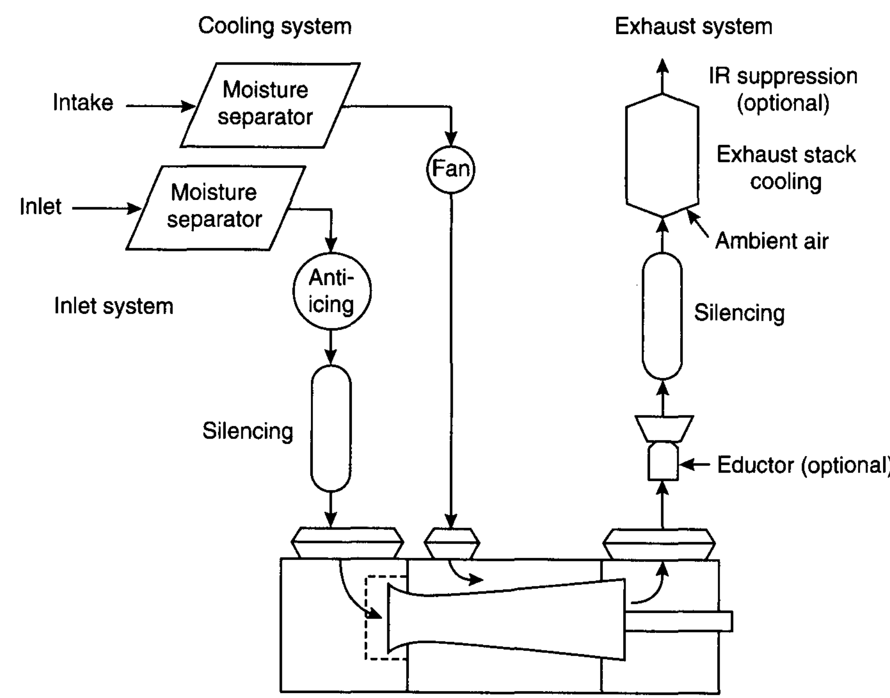


Figure 7-32. Air ducting systems. Courtesy General Electric Company.

sound suppressing barnes may be required. Where the duct is used for engine removal, barnes must also be easily removable. Construction of inlet ducting should be capable of accommodating gas turbine-induced pressure surges of 3 psi for 5 milliseconds. Gas turbine manufacturers will usually request to review the ducting design. For especially contorted routing requirements, they may ask to see model or actual duct flow tests, to ensure proper dimensions, no protuberances likely to create distortion, and fabrication specs which preclude generation of small foreign objects such as weld beads or rivets that could break and enter the compressor. An operator should scrutinize any repair to uphold the same standards.

Common ducts are not recommended. Good general design practice calls for separate ducts for each engine. A possible exception is discussed on page 7-102 under "Auxiliary Gas Turbine," APU installation.

The intake duct should have its upper deck suction sides oriented so that they face away from ship-generated sea spray and away from the place where fire-hose deck wash-down by the deck crew is likely to occur; they must be positioned where there is no likelihood of ingesting exhaust from galley vents or any engine or auxiliary exhaust, and no possibility of deck gear or cargo disturbing flow into the inlets.

At the entrance ofthe inlet duct are moisture or mist separators or filters. Their design may be inertial separators, coalescing separators, and frequently a combination ofboth. The filter house must have adequate flow deck drains with traps to carry away the filtered water. The operator should check these to ensure no blockage is present, especially whenever engines seem to require unusually frequent water washing. The filter elements themselves require periodic cleaning. Separator filters are subject to icing blockage or partial blockage due to maintenance neglect. As such, filter systems usually include delta pressure released blow-in doors which bypass outside inlet air directly into the duct to preclude gas turbine suction from collapsing the duct. Typical gas turbine allowable levels of salt ingestion are in the order of .0015 ppm average and .01 ppm maximum.

As with the filter elements, all parts ofthe engine intake system, especially areas where a change in air velocity occurs, are candidates for icing. While actual icing may occur infrequently, when it does occur, icing can cause flow blockage and distortion and particles breaking off surfaces, any of these being detrimental to the compressor. Typically, ice detection systems located in the inlet system actuate at the combination of temperatures below 40°F and relative humidity greater than 70 percent. The temperature 40°F is chosen to accommodate temperature drops in the system where air velocity increases and static pressure and temperature decrease, such as in the inlet bell mouth. Anti-icing systems may use electrically heated surfaces or turbine compressor discharge air bleed mixed into the intake air, well upstream ofthe engine inlet. Bleed systems may be installed upstream or downstream of filter elements. At any temperature below 40°F, it is necessary only to heat the air a maximum of about 8°F to reduce relative humidity to below 70 percent. Blow-in doors usually have their seals electrically heated to ensure they will open when activated by delta pressure caused by demister blockage. Some natural gas fueled gas turbines in industrial service have used the ingestion of the engine exhaust into the inlet for anti-icing and also to preclude power drop-off at extremely cold temperatures. This method would not be practical for even vaporized LNG burning ship engines because the water generated during combustion, when mixed with cold inlet air, results in an increase in relative humidity to super saturation, mandating the inlet air must always be heated to 40°F to avoid icing conditions.

Where a particular ship route includes frequent traversing close to coastlines where there is a high sand content, such as the Red Sea, special filter elements capable of sand and dust removal may be required in addition to moisture separators.

COOLING AIR

The supply of cooling air may be provided by either a fan, as shown in figure 7-32, or an exhaust eductor, which uses exhaust gas velocity to pump cooling air, shown as an option on the sketch. Required cooling air flow will vary with engine models and the engine compartment design. For the LM2500 engine enclosure, flow rate is sized for about 15 percent of primary air flow at full power. The benefits ofthe eductor are that it is lightweight, simple, and autonomous. In the instance of an electric power outage (assuming redundant power supply for the small current for engine controls), cooling air continues and the engine can continue to operate using fuel from a head tank. By comparison, the fan for a 25,000 hp turbine requires an 80 to 100 horsepower electric motor. The disadvantage of the eductor is that it tolerates less suction pressure loss and exhaust system back pressure, and the increase in engine fuel consumption is somewhat greater than the fuel required to generate the electric power for the fan. The eductor's restriction on losses is generally not a problem. Cooling air demisters need not be as efficient as intake demisters as they reduce the salt formation on the engine outer casing, which has less impact on engine operation. The exhaust system loss limit might only be a problem for a military ship where exhaust system infrared OR) suppression devices could increase exhaust back pressure beyond the engine eductor limit.

When an air eductor is used, the cooling air inlet system is usually bifurcated with suitable dampers so that a much smaller fan may provide cooling air flow for a short period after the engine is shut down. Cooling air inlet ducting walls are usually lined with sound-absorbing treatment. A damper at the entrance of the cooling air to the engine enclosure is required to shut off air flow when the fire extinguishing system is discharged and may be used at other times to shut off the natural draft effect when the engine is shut down.

EXHAUST SYSTEM

The exhaust ducting is usually sized for the combined flow of the engine and the cooling air. The system, depending on the engine, may occasion temperatures of 1,000°F. The ducting is lined with sound-absorbing construction but the use of noise attenuating baffies is seldom required. Exhaust ducting velocities up to 200 ft/sec are usually acceptable.

, FUEL SUPPLY SYSTEM

Figure 7-33 is a diagram of typical basic fuel system components. The purpose of the system is to thoroughly clean the fuel of particulate matter and water and sediment, and to deliver fuel at an adequate flow, pressure, and temperature. The flow is a function of the maximum required by the engine. When the engine drives a main fuel pump, delivery to the engine is a regulated constant positive pressure, usually in the order of 5-10 psig at the engine interface. Proper viscosity is that required by the particular engine manufacturer and usually during operation from idle and above can be in the order of 10 cSt. The figure shows a biocide additive system as part of the fuel storage tank's loading system. The purpose of this system is discussed in chapter 8 of this text. The fuel heater shown between the transfer pump and the purifier is required only if the purifying equipment has a maximum viscosity requirement below that which might be expected at double bottoms temperatures (normally not below 30°F). As chapter 8 discusses, gas turbine fuels are typically below 6.0 cSt at 40°C (104°F). Generically, centrifuge fuel oil purifiers are highly efficacious in the fuel transfer system. They excel at removing gross amounts of water and sediment without needing frequent filter element replacement. Where fuel tanks may be used for seawater ballasting (compensating), good rugged fuel cleansing is particularly important. The coalescer and separator filters provide the final cleansing to meet the gas turbines system requirements. The need for a fuel heater and its capacity may vary with different engines and locales of operation. For example, the LM2500 gas turbine requires 6 cSt maximum viscosity only during the starting cycle up to idle speed. At idle speed and above, 12 cSt maximum viscosity is suitable. For a ship where fuel is routinely about 4 cSt maximum at 40°C upon delivery, and service or head tank fuel temperatures are always above 35°F, the heater capacity may only be required for the several gallons used during the start cycle and up to idle. Where installation space can accommodate them, head tanks afford the opportunity of continued engine operation if there is an electrical supply interruption or outage to the ship's fuel service boost pump.

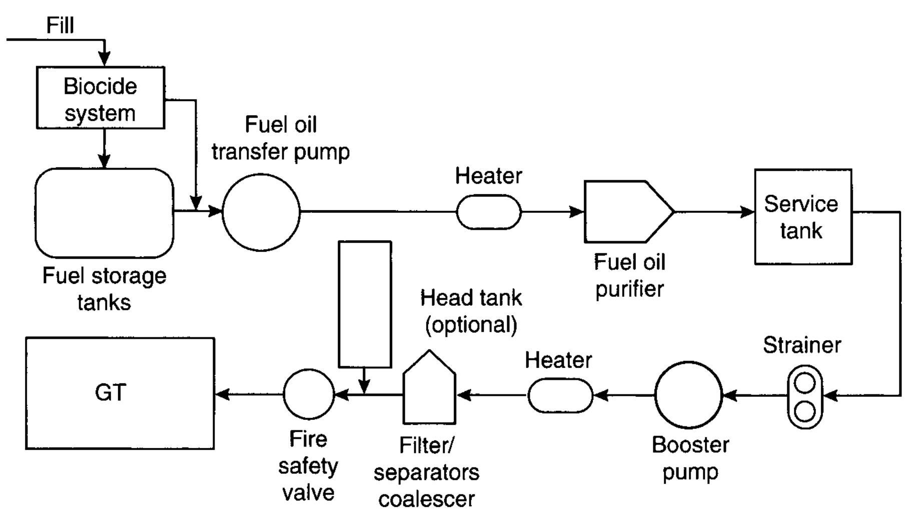


Figure 7-33. Basic ship fuel components

The figure shows a ship system provided fuel fire safety valve. This is a quick acting, remotely activated fuel shutoff valve, upstream of the engine enclosure in case of a fire in the engine compartment or if a fuel line is damaged. Closure of this valve should simultaneously close the engine mounted fuel shutoff valve as operation with the fire safety valve closed causes engine fuel pump cavitation, fuel pressure fluctuations, vibratory stress, and dislodging of debris in engine fuel filters.

DUAL FUEL SYSTEMS

The potential use of dual gaseous and liquid fuel for LNG carrier applications is discussed in "Construction, Engine Mounted Accessory Systems" (fuel components section, page 7-56). The ship's fuel supply system for vaporized LNG and liquid distillate fuel operation must provide both fuels at the required engine fuel manifold pressures. The engine-driven liquid fuel pump is removed as its flow is no longer a function of gas generator speed. An electric-driven pump would take its place. For LNG natural boil-off, it is necessary to provide very low temperature gas compressors. For high pressure ratio gas turbines, the gas must be compressed to 24 to 28 atmospheres or greater, which requires gas compressor drive power equivalent to 3 percent to 5 percent of the power of the prime mover output. There may be a need for gas coolers to reduce the heat of compressors to a satisfactory inlet temperature to the engine's fuel control system. Because of the special regulations for gas burning aboard ship, there are many other differences in the fuel systems, but the fuel delivery equipment is mentioned as the most prominent of additional machinery requirements.

LUBE OIL SYSTEM

Marine gas turbines derived from aero engines typically use antifriction (rolling-contact) bearings, which require synthetic lube oil. Industrial designed turbines typically use journal (fluid film) type bearings and use mineral oil. In the case where an aero engine is used as a gas producer (gas generator) into an industrial power turbine, the gas generator usually retains the synthetic oil system while the power turbine may share a mineral oil system common to driven load members such as gearing and/or generator. In some cases, industrial gas turbines may have been designed to use either one or the other. This is a special case as synthetic oil can deteriorate seals and gaskets traditionally used with mineral oil.

Where journal-type bearings are used, there may be a need for a lube oil pump to lubricate the bearings before start-up and after shutdown. Gas turbines have very low lube oil consumption rates usually measured in tenths of a pound of oil per hour. The consumption rate is nearly constant irrespective of power setting.

Most shipboard gas turbines drive their own lube supply and scavenge pumps, and many provide the complete lube system as part of their scope of supply. Beside pumps, other required lube oil system components are an oil tank, oil cooler, oil filters, air/oil separation devices, venting, piping, and valves. Because lube consumption is small (usually less than a gallon per day per engine) lube oil tanks are small, ranging from 5 to 50 gallons. The tank size and other component capacities may be dictated by the use of engine lube oil for other functions such as hydraulic actuation of variable stator vane systems. Tank location requirements may vary. For gravity drain, they are below the engine. For dry sump aero design with scavenge pumps, they may require mounting above the lube supply pump to assure a positive head. Lube tanks usually include large cover port(s) for cleaning, a filling port, drain connection, oil level sensor, and possibly sight glasses to determine lube level. Tanks usually have baffles to reduce sloshing and a deaerating provision. Aeroderived engines using antifriction bearings and air pressurizing between shaft oil seals (see fig. 7-20 and associated discussion) may have total scavenge oil pump capacity 3 to 4 times the volume capacity of the supply pump to accommodate oil aeration. These engines may have an engine-driven air/oil separator to significantly reduce the deaeration required in the lube tank.

Oil coolers will vary depending upon the cooling medium. The medium could be air, fuel, gearing lube oil, freshwater, or saltwater. Oil-to-air radiators are not generally optimum for ship applications. Aero engines may have the capability of using their fuel flow to cool the lube oil, as is frequently done on aircraft. If this alternative is used, pay particular attention to the manufacturer's fuel specifications as some marine fuels could possibly break down at peak allowable oil scavenge temperatures causing coolant side deposits. Freshwater and gear system lube oil are the most

widely chosen coolant mediums for synthetic oil systems. Gear lube oil has two advantages. First, turbine damage due to contamination of engine oil with gearing oil is significantly less than from salt or freshwater contamination. Second, the heat release of the antifriction bearing engine is small compared to that of the gearing system and only requires a nominal increase in the capacity of the already required gear lube oil cooler. Maximum temperatures for antifriction bearing scavenge oil are frequently in the order of 100°F hotter than for journal-type bearings. Where the turbine manufacturer allows seawater cooling, the coolant allowable discharge temperature is less than other systems and frequently two coolers in parallel are provided so that one can be cleaned while the other is in operation. Coolant flow should be regulated to provide a constant oil discharge temperature as specified by the engine manufacturer (usually within 130°-200°F). Supply and scavenge filters are normally duplex, nonbypassing filters, with delta pressure sensors across each element. Piping size and materials, check valves and maintenance (normally open) shutoff valving, connectors, and filtration micron specifications are specified by turbine manufacturers for their system requirements.

FIRE DETECTION AND EXTINGUISHING

The previous descriptions of gas turbine enclosures (modules, compartments) listed purposes of such structures as fire containment, parts rupture containment, and directing of cooling flow with features such as air dampers, floor drain systems, and fire detection and extinguishing systems. Whether the enclosure is premanufactured or built into the ship as a compartment, these features are one major factor in fire protection. Additional factors are the design features of gas turbines that minimize the incidence and risk of fire. Such features may include shields between the combustor and the downstream hot surface areas to preclude the spray of fuel or lube oil on hot surfaces to avoid auto ignition. Engine piping may include spray shields at piping couplings and elbows and use of double wall construction on fuel lines and manifolds so a leak in the inner tube is captured in the outer tube and directed to a properly vented drain system. Piping and tubing clamping and support locations and methods are engineered to preclude vibratory damage. Any maintenance and repair to engines should keep original piping retention methods and locations.

The fire extinguishing agents, over time, may change as a result of ecological considerations. To date, while the use of Freon is diminished, the use of CO_2 and Halon ($CBrF_3$) are common. In using these agents, the objective is to clear the enclosure of personnel and then quickly establish a given concentration to extinguish the fire and to retain concentration with follow-on discharge to prevent relight. Size, number, and location of discharge nozzles in the enclosure are selected to give optimum extinguishing agent distribution. It would be unusual that fire would occur concurrent with personnel being inside the enclosure, but it is critical that personnel be evacuated from the enclosure before an extinguishing agent is discharged. Two methods of achieving this are (1) to locate a release inhibit switch outside the entry door to the enclosure (usually in conjunction with enclosure lights and fire alarm switches) and (2) to use a time delay during which an alarm (such as blinking ofthe module lights) is shown inside the module before agent discharge. The enclosure may have a small window through which fire may be verified from outside, as well as the need for secondary discharge of agent in case of relight.

Typical features and sequence of a fire stop sequencing system follow.

1. Enclosure fire/flame detectors transmit fire signal to the machinery control station. Detectors might typically be temperature switches at the enclosure overhead above the engine hot section and ultraviolet flame detectors located forward in enclosure focused at lower levels of enclosure.
2. Actuation of the fire stop sequencing system may be automatically or manually initiated at the machinery control station.
3. Sequencing of the fire stop sequencer would simultaneously initiate appropriate signals to accomplish the following:
   - De-energize (close) engine fuel shutoffvalve(s).
   - Close ship fuel supply valve located just outside engine enclosure. This valve must be kept closed until engine coast-down and fire extinguishing are complete.
   - Close engine bleed air valve(s) as appropriate.
   - Close cooling air dampers and shut down supply fan and exhaust doors, if appropriate.
   - Sound fire alarm and display specific enclosure "fire" light at machinery control station.
   - Cause flashing oflights inside enclosure to warn personnel inside.
   - After a time delay, sequencer activates discharge of primary battery of extinguishing agent bottles. Time delay allows time to activate extinguishing inhibit switch at the enclosure for reasons such as a detectable false alarm or evacuation of personnel from enclosure. Upon discharge, the extinguishing and distribution system is designed to automatically control rate for proper fire extinguishing duration.
4. It is usual that the primary system may also be activated manually outside the enclosure, and a secondary system released manually, as required.

Mter a fire stop sequence has occurred (using Halon) and postfire safety requirements have been followed (for example, cooling-ventilating for personnel safety, inspections) some manufacturers may recommend special procedures, such as water washing engines externally before resuming engine operation.

ENGINE CLEANING

Fouling of gas turbine blades and vanes by impurities in the inlet air which adhere to these surfaces can drastically impair engine performance and, without cleaning, ultimately lead to compressor surge or stall, excessive hot parts temperature, and accelerated hot section corrosion.

Historically, throughout the development of gas turbine cycles and their operation in a wide range of environments, a wide range of specialized cleaning methods have been developed, modified, abandoned (and new method research continued) to be compatible with particular engine cycle developments and to cope with particular contaminants oflocal environments ashore. Examples might include proximity to concrete factories or sulfur and other chemical processing plants, and to agricultural areas where wind carries fertilizer and other crop treatments.

This discussion will deal with cleaning of impurities routinely experienced at sea, but operators should keep in mind that operation of gas turbines in port or coastwise, where a high content of shoreside impurities may be ingested, may call for consultation with manufacturers regarding use of special intake filtering measures (for example, filtration of sand, which is abrasive to compressor surfaces, for extensive operation in the Red Sea) and specialized cleaning agents and procedures for both gas turbines and inlet filtering systems. Major shipboard inlet air contaminants are salt, oil, and soot which pass through or bypass the air intake filtering systems. Oil and soot sources may come from tank vents, galley exhausts, and other machinery exhaust or vent systems. If there is a persisting increase in required engine cleaning frequency, try to determine and correct the source of contamination. Sources might include plugged filter house drains, intake system leaks downstream of the filters, filter element deterioration or need of cleaning, and more. Some symptoms portending the need for engine cleaning include increased turbine gas temperature for a given power and ambient temperature (T2), reduced power for a set gas temperature and given T2, and visual detection of contaminates during inspection of shutdown engine compressor surfaces through the engine inlet and compressor borescope inspection ports. Conditions of loss of power or high gas temperature are determined by comparison of corrected performance with the engine performance during the engine's first voyage.

Typically, the cleaning agent is demineralized or distilled freshwater. Engine manufacturers provide their water quality standard of impurity limits. The use of hard water can leave deposits in the engine which must be removed manually. A solvent is usually added to the water to break surface tension and dissolve oily or waxy constituents ofthe impurities during the wash and soak portion of the cleaning cycle. Approved solvents usually include detergents, much like household cleaners but controlled by specification. Some manufacturers also permit an emulsified kerosene/water solution, again the emulsifier controlled by specification.

There may be two modes of cleaning: (1) by motoring the cooled engine with the starter and (2) on-line cleaning during operation, if allowed by manufacturer. On-line cleaning is usually less effective and is required more frequently, if allowed at all. When good intake filtering systems are used, gas turbines can operate many hundreds of hours between cleaning, affording opportunities to' wash the engine when shut down.

For air temperatures approaching and below freezing, addition of specified antifreeze agents (for example, isopropyl alcohol) may be permitted.

All wash solutions must be homogeneous. Wash cycles are usually comprised of a wash solution spray during motoring, a static soak period, and several freshwater rinse cycles, followed by a briefrunniIig of the engine at idle to dry the engine. The wash solution is injected into the engine inlet via a spray manifold, often built into the engine bell mouth or inlet housing. Some engines may have hardware kits whereby water nozzles can be inserted in the turbine system borescope or instrumentation apertures in the engine casing to permit a pressure rinse of the turbine directly following the compressor washing and before engine drying.

Some manufacturers require external washdown of their engines after discharge of Halon for fire extinguishing. Similarly, good housekeeping suggests occasional external washing to clean off salt carried by the cooling air. This can usually be performed (on cold shutdown engines) using a garden hose brought through the engine enclosure access door.

Each engine manufacturer will have its own engine cleaning procedures, cleaning agent specifications including amounts and concentrations, and methods of application.

ENGINE BLEED AIR SYSTEMS

Many gas turbines have the provision for bleeding air from the compressor discharge (and a few from intermediate stages) for ship service uses. Maximum bleed capability varies but frequently is in the order of 10 percent of the engine air flow. Depending upon the engine and its percent power output and ambient temperature, compressor discharge air may be from 4 to over 20 atmospheres and temperature from 400° to over 900°F. If maximum flow were bled while the engine was at its maximum gas temperature limit, power capability might be reduced to below 60 percent of no-bleed power with specific fuel consumption increasing over 30 percent. On the other hand, used prudently for selected purposes, bleeding can be practical and economical. One example is for anti-icing. At any temperature below 40°F, it is only necessary to raise inlet air by less than 9°F to reduce 100 percent relative humidity to 70 percent relative humidity. This would typically require about 1 percent compressor bleed flow and a 2 percent increase in fuel flow with no reduction in output power capability because inlet air is well below the temperature at which engine is rated. Even with the fuel penalty, total fuel flow at a given power is less than the fuel flow required at 75°F inlet air temperature with no bleed.

The ship's system that receives and distributes engine compressor bleed air usually must provide the required bleed flow regulating and shutoffvalving. For some ship service purposes, the compressor bleed air must be cooled as well as pressure and flow regulated. On ships having multiple gas turbines, all using pneumatic starters, the bleed from one operating gas turbine may be used to sequentially start the other gas turbines, but the supply air temperature must be cooled below the starter inlet air temperature limit. For naval ships having Prairie and Masker systems that emit air bubbles through small apertures at select locations on propeller(s) and hull surfaces, in order to reduce propeller, hull, and machinery generated waterborne noise, bleed air temperatures must be reduced to approach sea temperatures.

STARTING SYSTEMS

The discussion ofthe choice of engine starting motors and the various duty cycles they must perform is given in "Construction" on page 7-57.

Alternating-current electric starter needs are typically met by the ship's 230/460 volt 60 Hz, 3-phase ship service electrical system and a 24 Vde supply for motor starter coils. Direct current electric starters typically require a 24 to 28 Vde supply source.

Hydraulic starter motor supply needs are met by a hydraulic supply pump driven by electric motor or a power takeoff drive from ship service generator drive engine. The hydraulic motor supplier will usually offer a hydraulic supply system that matches the motor requirements. Factors such as specific fluid used, reservoir capacity, filtration, and cooler capacity would be determined according to the number of engine starters to be supplied and their duty cycle requirements. The hydraulic motor drive power for a large gas turbine may be in the order of 100 to 125 kW.

Pneumatic starters require flows of several pounds per second in the order of2 to 3 atmospheres gauge at the starter. Flow rate will vary inversely about 10 percent for each lOO°Fchange in air temperature. Supply sources include stored high-pressure air, auxiliary engine-driven load compressor, gas turbine auxiliary power unit (see "Auxiliary Gas Turbines" section), and compressor bleed air from other operating gas turbines. All of these sources, except stored air, can provide a continuous prolonged air flow. When choosing stored air capacity, ifthere is no source of compressor bleed air from a second gas turbine, then the capacity of the bottle storage and recharge time should be able to accommodate at least the capacity to motor an engine for cooling after an emergency shutdown and restarts, or engine motoring plus starts during and after a water wash procedure, whichever is greater. This could be equivalent to ten or more times the air required for a simple start.

VENTS AND DRAINS

A gas turbine installation must accommodate the venting and draining of air, fuel, lube oil, water, and mixes of these.

Vented air falls into two categories-clean and with oil content. Clean air vents are usually discharged into the engine enclosure and carried into the exhaust with the enclosure cooling air. Oily air is passed through oil separators, which may be mechanically driven on the engine or static in the lube oil reservoir tank. Engine-mounted oil separator air discharge may be vented into the exhaust elbow with a liquid drain at the low point in the piping. Lube storage tank separator air discharge may be vented into a ship's atmospheric drain tank.

Engine drains are often piped in conjunction with enclosure floor (deck) drains. The design of floor drains must prevent siphoning back into engine compartments by negative pressure and prevent backflow from drain system malfunctions. An engine compartment or enclosure is essentially two chambers-the air inlet chamber, which is always at a negative pressure during operation, and the engine compartment, which has a positive pressure iffan cooled but a negative pressure if cooling air is pumped by an exhaust eductor. Drainage consists of lube oil and fuel from engine pump seals and manifold drains (collected by engine piping to be routed to the ship's drain tanks) and enclosure floor drains and exhaust elbow drains which may contain water and fuel or lube oil. Sources of water are from engine wash water and rain entering air ducts or exhaust duct while shut down.

The ship's vented drain tank should have a means to monitor liquid level. The normal collected fuel and lube oil drain flows from seals and combustor system and fuel manifold drains is small. However, fuel manifold shrouds, while having no flow unless there is a manifold failure, could have a momentary 100 percent of engine maximum fuel flow at time of manifold failure. Therefore, this drain line should have a leakage detector. Similarly, where fuel shutoff valves bypass fuel flow, if shutdown occurs at high power and the bypass line should rupture, the flow could be a momentary 100 percent fuel flow. In either case, fuel flow would diminish to about 7 percent in approximately 5 seconds as engine drops to idle and thence to zero as gas generator speed decays to zero if not first stopped by closing the ship's fuel supply valve. Fuel purge systems for diverting cold fuel before starts have a brief and low flow (perhaps several gallons) that drains to the ship's drain tank.

The exhaust collector drain is an example of a mixed drain and vent and may contain water from water wash and rain, and/or fuel from a false start, followed by a very small flow of hot exhaust gas when engine is running. To avoid mixing hot exhaust with fuel, these drains must be arranged using an open cup or funnel or tee-fitting for liquid drainage downward; back pressure causes the hot gases to flow upward into the compartment and exhaust with cooling air.

ELECTRIC SUPPLY

Besides the ship's support systems (fuel supply, starter, cooling air, and oil coolant), the actual power requirements to sustain gas turbine operation are small. The ac requirements are usually met with nominal ship service 230 volt, 3-phase, 60 Hz and 115 single-phase 60 Hz, and most engines can accept 50 Hz equivalents. The dc requirements are for a nominal24 Vdc supply system. Generally, the nominal 24 Vdc requires 21 to 24 volts at the engine interface, and since 24 Vdc is prone to high-line losses in transmission, the supply source may be specified from 24 Vdc to 28 V dc.

The manufacturers specifications regarding electric supply power may require regulation or filtering to limit transient voltages, voltage spikes, dc pulsing, and similar problems.

When the engine control is a dc system with an uninterrupted power supply of batteries charged by a suitable ac power system charging device, the engine may be able to sustain a short ac-power outage without shutting down the gas turbine. This assumes the ship's fuel system shutoff valve to engine does not shut on ac outage, and fuel head or service tank can provide fuel at close to atmospheric pressure without a boost pump running. For an electric fan cooled enclosure, operation might be possible for a minute or so before enclosure temperature exceeds limit. Where a cooling air eductor is used, then operation might continue until head tank fuel supply was exhausted or lube oil temperature exceeded limits from lack of coolant flow.

Important considerations in the electric supply and wiring to a gas turbine system for initial installation as well as any changes made after installation are the manufacturers specifications regarding

- power quality (previously mentioned)
- sealing of arcing devices
- connector types and temperature limits
- bonding locations and cables
- shielded cables and grounding and floating locations
- isolation of instrumentation leads from power leads
- electrical interface control specifications

Operation

There are five basic steps to operation of a propulsion gas turbine: prestart checks, starting, operation, shutdown, and postshutdown.

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Operation

There are five basic steps to operation of a propulsion gas turbine: prestart checks, starting, operation, shutdown, and postshutdown.

PRESTART CHECKS

The following are generally representative of checks and procedures for a propulsion engine within an enclosure:

- enclosure doors closed, fire system activated
- cooling damper open, fan on if used
- waterwash valve closed
- oil tank level suitable, oil coolant flow ready
- electric power to engine on
- fuel supply pressure/temperature proper
- compressor bleed air v.alve(s) properly positioned
- inlet and exhaust clear
- reduction gear/propeller systems ready
- starter supply available
- starter, fuel, ignition, auto/manual start switches positioned
- any prestart procedures such as bearing prelube for engines with journal bearings

STARTING

Control systems may have automatic as well as manual modes. The functions performed are similar except the monitoring of time versus speed of gas generator and turbine temperature in the manual mode is performed by the operator.

A typical start sequence consists of three basic steps:

1. Activate starter.
2. Within specific time, at about 25 percent gas generator idle rpm, activate fuel and ignition on.
3. Within specified time, when gas generator is about 90 percent idle rpm, turn off ignition and starter. The engine continues to accelerate to the idle rpm.

During the start sequence, as a function of time, control systems normally monitor and provide fault signal or start abort for

- failure to reach fuel/ignition on speed versus time, failure to light or flame out, and failure to reach idle versus time
- low lube pressure versus time
- overtemperature

IDLE

At idle, operator or control system typically activates start counter, running time meter, and appropriate sensor systems (for example, icing detection), and checks power turbine rpm (clutch and brake disengaged) to ensure it is spinning freely. At idle and at other power settings, the operator monitors engine instrumentation for the functions listed in the next paragraph.

OPERATION

Mter engaging reduction gearing clutch and disengaging any shaft brake, set throttle command to desired power setting and monitor turbine gas temperature during transient operations. Mter stabilizing at new power setting, observe the following typical parameters:

- power turbine inlet gas temperature
- gas generator rpm
- power turbine rpm
- lube oil supply temperatures, pressures, tank level
- gas generator and power turbine vibration
- fuel inlet pressure, temperature
- enclosure air exit temperature
- other performance sensors per manufacturer

SHUTDOWN

Shutdown falls into two categories: normal and emergency.

Normal shutdown procedure is typically as follows:

- Set throttle to idle power.
- Disengage clutch as appropriate.
- Operate at idle for specified time.
- Mter about three minutes at idle, observe parameters listed above under "Operation." In addition, observe lube and fuel filters differential pressure.
- At five minutes or per engine manufacturer, close (de-energize) engine fuel shutoff valves.
- Check that gas temperature and fuel manifold pressure decay normally within specified time (1 to 2 minutes).
- Initiate postlube of bearings, if required.

Emergency shutdown or trip can be activated automatically and instantly by the control system sensing an operating parameter reaching the trip limit, or by the operator due to an alarm condition that cannot be rectified, or by emergencies in other machinery or ship's operation. Shutdown of a gas turbine prior to normal shutdown idle operation to reduce and stabilize engine temperatures may cause uneven cooling of rotors and stators leading to adverse effects such as rubs. The following is a typical emergency shutdown procedure:

- If condition permits, decelerate to idle.

- Initiate emergency stop. De-energize (shut) engine fuel shutoff valves and close ship's fuel supply valve.
- Initiate postlube procedures if appropriate.

POSTSHUTDOWN

Postshutdown procedures may vary depending upon whether shutdown was normal or emergency. The variations typically concern control and timing of the enclosure cooling air supply fan and cooling air damper as a function of outside air temperature and enclosure air temperature. In the case of emergency shutdown, there may be special restart procedures or motoring procedures, as conditions permit, to avert engine rotor to stator rubs. Postshutdown procedures and securing of the engine and ship's support systems should be followed as stipulated for the engine and propulsion system.

Maintenance and Repair

REPAIR INTERVALS

Early gas turbine applications required periodic engine disassembly and inspections to determine the need for internal repair. For aircraft gas turbines, disassembly on the aircraft was seldom practical and a "fixed time" internal maintenance interval was established based upon experience. Each engine model demonstrated, through test cell endurance qualification testing and teardown inspection, an initial repair interval at which time engines were removed for teardown inspection and repair. As service experience was acquired on fleets of this model of engine, this "fixed time" interval was extended by the regulatory agency as experience showed a longer interval might be granted.

Today, in most cases, internal maintenance intervals for both aeroderivative and industrial derived marine gas turbines are established on the basis of "on condition" by using condition monitoring, which may be comprised of such elements as monitoring and trending of engine sensors of critical operating parameters, visual inspection of the engine externally, borescope internal inspection with photo records of progression of deterioration, and trend analysis of engine performance.

The maintenance objective of gas turbine manufacturers is to maximize engine availability and minimize cost. Because different applications place a wide variety of operating stress levels upon an engine, manufacturers usually will recommend a complete comprehensive maintenance program for each operator. For example, gas turbines operating as gas pumpers with clear air, a constant continuous steady-state power rating, and burning natural gas, may achieve overhaul intervals well over 50,000 hours with an intermediate hot-section repair. Because of the marine atmosphere, liquid fuel, and more rigorous operating conditions, marine gas turbines repair intervals are shorter. By comparison, ship propulsion engines with good quality, well-purified fuel, efficient air filtering systems, and moderate duty cycle often attain 20,000 to 30,000 hour overhaul intervals with a midpoint hot section repair.

The types of maintenance usually fall within three categories: onboard preventative maintenance, onboard corrective maintenance, and shore-based corrective maintenance.

FACTORS AFFECTING TIME BETWEEN OVERHAULS

Typical factors that affect the time between overhauls (TBO) for a shipboard gas turbine include air quality, fuel quality, maintenance, and duty cycle.

Inlet Air Quality

Air filtration systems, discussed under "Installation, Inlet Air" on page 7-73, should meet manufacturer's criteria for salt and solids entrained with the inlet air. Salt limits may be in the order of .0015 ppm average and .01 ppm maximum. Solids, such as sand near coastlines or chemicals in port, may be limited to one or several grains per 10,000 ft^3 of air. Levels higher than manufacturer's stipulations reduce TBO time.

Fuel Type and Quality

Chapter 8 of this volume discusses typical allowable levels of fuel impurities (metals from residuum, sulfur content, salt, water, solids) for gas turbines. Excessive levels of any of these will reduce TBO.

Prescribed Maintenance

Strict adherence to prescribed preventative maintenance will enhance TBO. A major element is periodic water washing.

Engine Rating and Duty Cycle

A power rating at particular ambient conditions determines the internal engine temperatures and the resulting expected service life of engine components exposed to these temperatures. These "hot section" components include the combustor and the high-pressure turbine blades and vanes. The life of the hot section components determines the interval between major maintenance actions. Figure 7-34 indicates the typical relationship between internal parts life and a representative gas temperature, in this case, as reflected by power turbine inlet temperature. At higher temperatures, the component life is determined by creep-rupture and decreases rapidly with increased temperature. At lower temperatures, component life is determined by corrosion and does not vary as rapidly with temperature. Operation for large percentages of time at power levels in the creep-rupture regime will severely limit component life in the gas turbine.

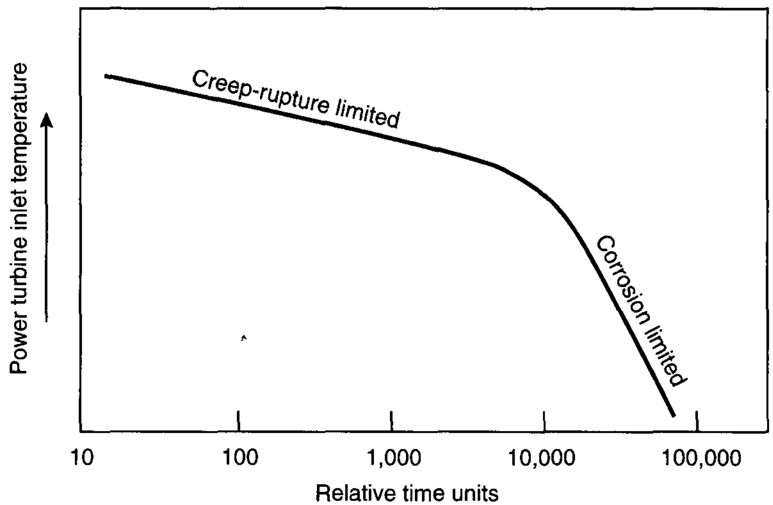

Figure 7-34. Hot section refurbishment interval.
Courtesy General Electric Company.

Conversely, the best advantage of gas turbine capability can be taken with respect to maintenance below the top of the corrosion controlled regime.

Another factor that affects maintenance interval is the number of cycles of operation per unit time. Since some internal components may be limited in life to a maximum number of cycles, operation with very short mission lengths may result in higher maintenance costs due to thermal or low-cycle fatigue limitations causing more parts changes at overhaul. The cyclic limitations typically might be a factor in ferryboat operations where mission lengths might require a start-up to shutdown cycle of 1 to 2 hours of operation or less.

ONBOARD PREVENTATIVE MAINTENANCE

Most preventative maintenance is scheduled on the basis of operating hours, calendar time, or a combination of both. Typical preventative measures include the following:

- Condition monitoring of critical parameters such as gas generator inlet and exit gas temperatures and pressures, vibration, oil pressures, and temperatures.
- Visual inspection of inlet and external conditions such as leaks; security of engine mounts and engine piping and wiring mounting devices; condition and calibration of sensors and variable vane systems; filter inspections and changes; inspection and cleaning of fuel nozzles and ignitors. Task frequency may vary from weekly to annually.

- Water washing. The frequency will vary with the amount of spray going into the inlet filters, filter efficiency, and engine sensitivity. The prime purpose of washing is to recover performance lost by fouling. Timing may run from 100 hours for severe conditions to over 500 hours during nominal weather. System and procedure is described under "Installation, Engine Cleaning" on page 7-81.
- Borescope inspection of interval parts. Borescope equipment is a derivative of the medical endescope. The device is a fiber-optic probe with a light source conducted down the probe and the image conducted up through the eyepiece. Adapters are selected for levels of magnification of the image and for camera and low intensity light video recording. Figure 7-16 shows the axial location of bores cope ports on the FT8 engine. Typically, at each of several stages, one port is located along the compressor and the turbines to observe blades as the rotors are incrementally rotated. There are usually several circumferential locations for viewing hot section static parts such as combustor, transition liners, fuel nozzles. Often apertures in the casings for fuel nozzles, and temperature and pressure probes may be used as additional accesses for borescope inspections. Borescope photography can provide an essential history of gas turbine condition, such as propagation rates of cracks and corrosion versus operating time, allowing the operator to predict and schedule future timing for maintenance action at the operator's convenience.
- Trend analysis of performance is best done using a computer program. Many manufacturers can provide a performance trending program to monitor gas turbine health. A computerized trending of power output, in conjunction with selected gas path parameters, allows evaluation of a unit's performance over a period of time and aids in identifying the section ofthe gas turbine that is causing degradation of power.

ONBOARD CORRECTIVE MAINTENANCE

The basic, or in military parlance, "Levell," onboard maintenance includes preventative maintenance as previously discussed, and corrective maintenance involving troubleshooting or fault isolation; adjustments; replacement of external control components (engine-driven accessories such as pumps, starters, piping, wiring, sensors, and related hardware); and onboard operational checkout of replaced hardware. In addition, Levell encompasses the removal, replacement, and operational checkout of the gas generator and often the power turbine.

To accommodate engine removal, engine enclosures are designed with removal hatches, either a top hatch where the engine may be lifted to the deck above, or a side hatch where the engine can be removed laterally into the engine room. With military ships, where there are no convenient

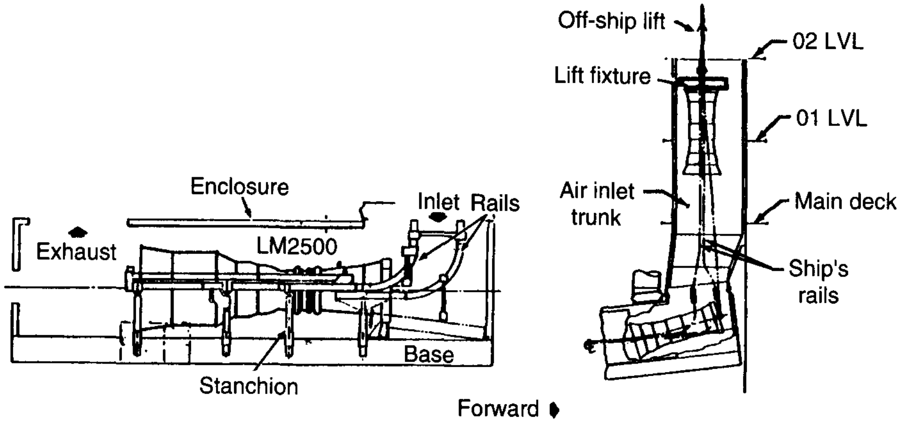

Figure 7-35. Gas generator/power turbine replacement system via inlet duct. Courtesy General Electric Company.

passage routes to remove the engine through the ship, the gas generator and power turbine are removed separately through the inlet ducting system. Where the access route is through the ducting, the enclosure is designed to mount a removal rail system for transitioning the gas generator and power turbine horizontally forward and through a 90-degree turn for vertical removal through the inlet ducting. Figure 7-35 shows the rail removal system used in the enclosure shown in figure 7-28 to remove the LM2500 engine via the inlet duct. By using a horizontal rail system that extends to the enclosure forward wall, in lieu of the transition rails to vertical, this engine and many others may be split into assemblies to conduct Level 2 onboard maintenance as discussed in the following paragraph.

Level 2 maintenance expands the scope of on-site repair. This ability is a trade-off of convenience versus engine downtime. The exchange of a gas generator, which requires removal more frequently than a power turbine by a factor of two or more, can often be exchanged and running in three or four hours when side or overhead hatches are available and a spare is onboard. For a duct removal system, a gas generator exchange may take up to 20 hours. This is generally less elapsed time than some of the onboard repair procedures may require. Depending upon the engine model, various of the following internal repairs may be possible onboard, often within the enclosure or with small engines, by removing selected enclosure panels. Often shore-based personnel come onboard to assist in these repairs.

- compressor blade and vane blend repair
- compressor blade and vane replacement

- compressor trim balance
- power turbine blade and vane blend repair
- power turbine selected blade replacement
- replacement of high-pressure turbine stator segments
- replacement of selected bearings and seals
- replacement of engine component modules

SHORE-BASED DEPOT REPAIR

Manufacturers have shore-based or depot repair facilities for their engines. Some of them may also license the repairs of their engines to independent shops so that there may be available facilities at various geographicallocations offering options as to available work scheduling. Aeroderived engines or gas generators are typically air transportable.

Depot levels of repair range from Level 2 as described previously to modular maintenance, light repair, or complete engine overhauls. Modular maintenance typically is the replacement exchange of engine modular subassemblies such as combustors and high-pressure turbine assemblies, typically needing refurbishment at the midpoint repair between overhauls.

Frequently, engine manufacturers and sometimes their licensed repair representatives offer accessory components exchange, engine modular component exchange, and gas generator and power turbine lease programs, to shorten accessory and depot repair turnaround time and to provide a lease engine for use while an engine is in depot repair.

MANUFACTURERS PRODUCT SERVICES

Other services are often offered by manufacturers, including but not limited to the following:

- technical manuals, updates, bulletins
- training courses
- technical field service assistance
- spare parts inventories
- maintenance video tapes
- user conferences and symposiums
- spares, tools provisioning planning
- engine-to-ship interface control drawings and specifications (both mechanical and electrical)

Gas turbine manufacturers appreciate that the reputation of, and further applications potential for, their engines is allied to their users' service experience and satisfaction. Consider contacting their product service people early on for their suggestions and potential services.

AUXILIARY GAS TURBINES

Auxiliary uses of marine gas turbines are usually electric generating sets, auxiliary power units (APUs) and, in ocean technology, closed cycle units used under the ocean, usually for electric generation using an external oxygen-hydrogen catalytic reaction chamber to heat the working fluid, a helium-xenon mixture.

In the sizes used for ship service electric generation, the actual gas turbine used often may be a slight model variation from that used for propulsion purposes. Whereas for propulsion, the turbines are typically two-shaft machines, for electric generation, applications may use either two-shaft or single-shaft (gas generator and power turbine are on a common shaft) gas turbines. The choice of the single-shaft unit is usually based upon the need for very tight frequency control during large changes in load.

The "Propulsion Gas Turbine" section of this chapter, starting on page 7-30, discusses various gas turbines which among them incorporate typical variations in component designs and the ship support systems they require. Since these discussions are equally applicable for auxiliary uses of gas turbines, this section will discuss specific examples of a ship's electric generator and APU unit, denoting only special features and support requirements beyond those typical for propulsion units.

Ship Service Generator Sets

To illustrate a typical ship's service electric generator set and the features of a single-shaft gas turbine, the Allison Engine Company Model AG 9140 ship service gas turbine generator was selected. The generator set is the latest model of units to use the single-shaft 501-K series gas turbines which have been used in the U.s. Navy destroyers and cruisers since the mid-1970s.

The model AG 9140 generator set is shown in figure 7-36. The set is rated for simultaneous electric power output and compressor bleed for ship services, such as Prairie and Masker systems for ship radiated underwater noise suppression, at 2500 kW, 3-phase, 450 Vac, 60 Hz, at 0.8 p.,f.,at operating conditions of 14.696 psia ambient, lOO°F engine inlet, and 6 and 10 inches of water inlet and exhaust losses respectively. The bleed flow stipulated at the USN 2500 kW point is 2.37 lbs/sec which equates to a diminution of657 kW (881 HP) power output for the same turbine inlet temperature with zero bleed.

MODEL 501-K34 GAS TURBINE

The generator set is powered by the 501-K34 single-shaft gas turbine as shown in figure 7-37. The inlet screen is in the ship's ducting.

The compressor is fourteen-stage, axial-flow, directly coupled to a four-stage turbine, with a rotor speed of 14,340 rpm reduced by a gearset to the 1,800 rpm generator speed. During starting, bleed valves on stages

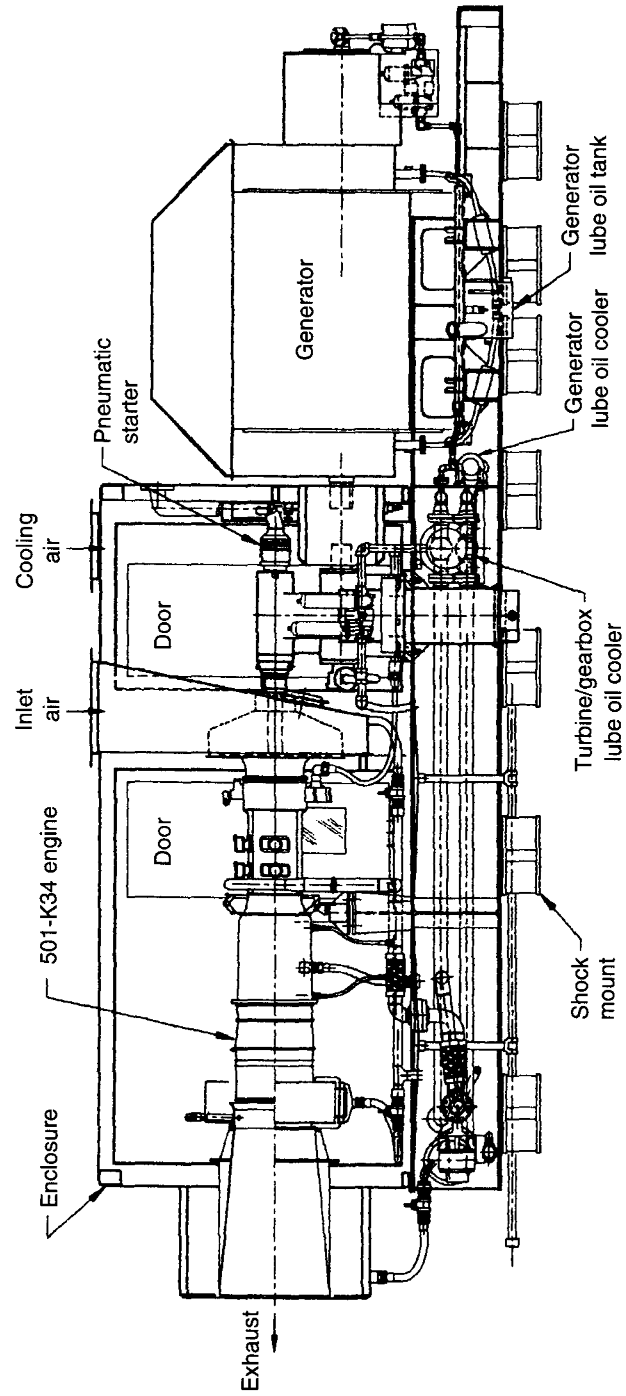

Figure 7-36. Allison model AG 9140 ship service gas turbine generator set. Courtesy Allison Engine Company.

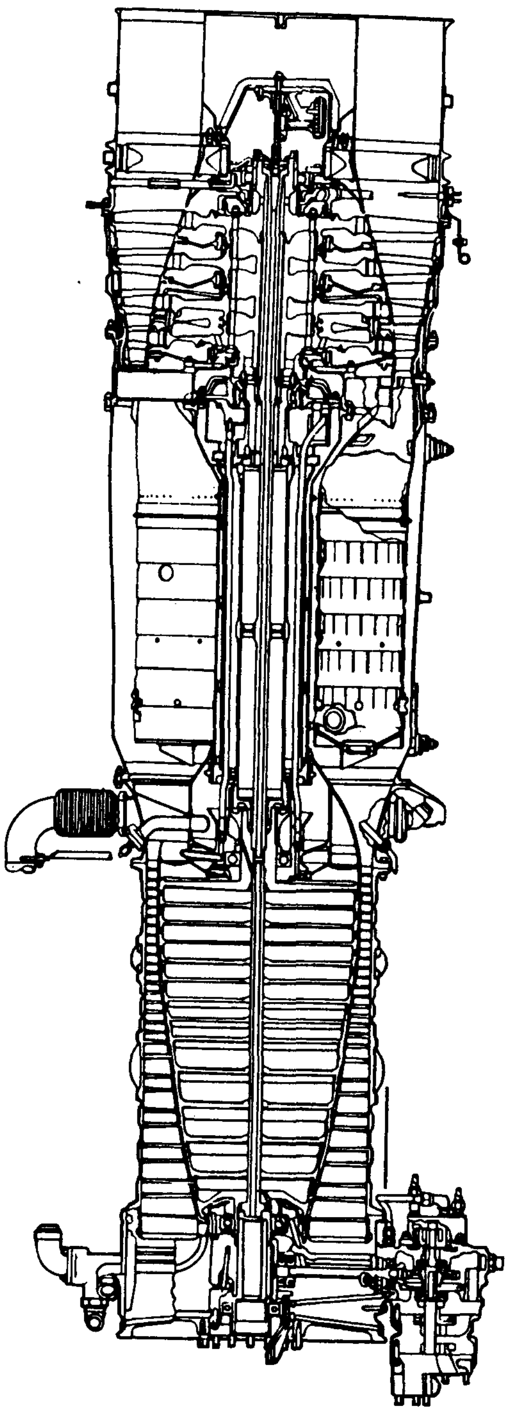

Figure 7-37. Allison model 501-K34 single-shaft marine gas turbine. Courtesy Allison Engine Company.

five, ten, and fourteen are opened to unload the compressor. During engine operation, 10 to 12 percent oftotal airflow can be bled from compressor discharge for ship service usage. During substantive load transients, sensed by the engine control system, bleed is momentarily interrupted to ensure optimum system frequency response.

The combustion system is can-annular with six combustion liners ofthe through-flow type using convection cooling, equispaced circumferentially in the single annular chamber. The outer casing is split axially. By removal in halves, easy access is gained for combustion liner changeout at hot section repair intervals.

The four-stage turbine uses air cooling of the first stage blades and stage one and two vanes and utilizes corrosion protective coatings on the first three stages of vanes and blades. Gas temperature is monitored by thermocouples at the turbine exit plane and used both for operation condition sensing and for fuel governing in conjunction with a closed-loop temperature control system.

The engine has five pressure lubricated antifriction bearings. An accessory drive housing, driven by a radial drive shaft from the compressor, powers gear-driven fuel and lubrication system accessories.

The gas turbine uses synthetic oil, cooled directly by seawater in a shell-and-tube heat exchanger. Vents from engine sumps and engine oil reservoir are vented to the engine exhaust eductor.

Following are sample engine specific fuel consumption points at the USN conditions:

Shaft Power	*Zero Bleed Flow*	*2.37 lb. / see Bleed Flow*
3,327 kW (4,461 hp)	304 g/kWh (0.500 Ibs/shp-hr)	
2,670 kW (3,580 hp)	320 g/kWh (0.5261bs/shp-hr)	358 g/kWh (0.588 Ibs/shp-hr)
2,125 kW (2,850 hp)	344 g/kWh (0.566 Ibs/shp-hr)	388 g/kWh (0.638 Ibs/shp-hr)

MODEL AG 9140 SHIP SERVICE GENERATOR SET

The generator set is supported and enclosed as seen in figure 7-36. The base supports all the major drive train components. The enclosure surrounds the gas turbine and reduction gearbox. The functions and features of the enclosure are similar in purpose to those discussed in the propulsion gas turbine section with figure 7-28.

The base is a single-piece weldment comprised of two main lengthwise beam rails with transverse members located to support the major components while providing structural rigidity. The unit is attached to the ship's foundation through fourteen, 5,000-pound capacity isolation mounts located to equally distribute the unit weight. In addition to attenuating structureborne noise into the ship, these mounts provide for naval service protection ofthe set from external high shock input as qualified to Grade A, Class I, Type A shock test per MIL-S-901C.

The gas turbine is directly coupled to the high-speed shaft of the vertical offset reduction gear. The gearing is double helical, parallel shaft, single-stage, utilizing sleeve bearings. It reduces the rated engine 14,340 rpm to the generator speed of 1,800 rpm, nearly an eight-to-one ratio. The air turbine starter is mounted on the gearbox at the opposite end of the high-speed shaft and provides the motive force to rotate the entire drive train from rest to the 9,100 rpm starter cutout speed, over 63 percent of generator rated speed.

Start time of the generator set to applied load is within one minute. Generator set start time, using an APU air supply, is shown in the APU discussion following this generator set discussion.

The generator is a salient pole, two-bearing unit driven by the gearbox low-speed shaft through a flexible coupling. An exciter and a permanent magnet alternator are integral with the generator. A separate excitation control panel receives voltage and current signals from the generator output and ac power &om the permanent magnet alternator and controls the generator output voltage. The &equency response of this unit, for full load pickup or dump, is plus or minus 1 percent of rated speed for no more than 1.5 seconds.

Generator rotor fans circulate generator air, cooled by an air-to-seawater heat exchanger. The system contains an auxiliary pump to pre lube the bearings prior to start-up or during motoring. Oil temperature is controlled by a seawater heat exchanger system during operation.

The generator sets are typically interconnected by means of a ship's ring bus which, with appropriate breaker placement, allows each set to run individually or in combination for parallel, load-sharing, and standby service and may be paralleled with shore power when docked. Load management provided by a ship system may also maintain prioritized control for load shedding in the event of an emergency.

The enclosure cooling air entry (shown in fig. 7-36) enters in the gearbox space, sweeps through the gas turbine space, and is educed into the exhaust stream. A damper closes off air when engine is shut down and in the event of fire.

Other module services, which are explained more completely under "Propulsion Gas Turbines" (starting on page 7-30), include

- ultraviolet flame sensors and a Halon distribution system
- spray nozzles in the inlet plenum for water or chemical cleaning during motoring
- icing conditions sensors to alert crew to provide compressor bleed air into intake system for anti-icing
- ship's supply of 440 Vac, 120 Vac, and 28 Vdc

The enclosure walls are removable panels of thermal and acoustic insulating material, allowing removal of the reduction gearbox and the engine.

Each panel contains a personnel door for access into the module. The enclosure contains overhead rails for attaching a rolling hoist to aid in the maintenance tasks for removal of engine and reduction gear when necessary. The hoist is normally externally stowed when not in use.

The philosophy of engine maintenance, which includes onboard preventive, onboard corrective, and depot repair and overhaul, is similar to the overview presented for "Propulsion Gas Turbines."

These type units have been used in conjunction with exhaust heat recovery steam generators for ship service steam and studies show the potential of adapting intercooling and recuperation OCR) for future fuel-efficient generator sets.

Auxiliary Power Units

BACKGROUND

Turbine powered auxiliary power units (APUs) were originally developed as a lightweight source of power for airborne applications. Installed in aircraft, these units provide power for electrical generators and for starting the main engines, and they provide compressed air for cooling the cabin. Some of the same engines used in aircraft applications have been adapted for shipboard use. Therefore, the transition of the APU to the shipboard uses for starting gas turbine and diesel propulsion engines and, collaterally, as compact emergency electric generator sets is most natural.

The following discussion is based upon AlliedSignal units that are in shipboard service. AlliedSignal APUs are currently employed on gas turbine powered ships including Indonesian Navy PSK patrol gunboats, Israeli SAAR 5 class Corvettes, Korean PCC Corvettes, and the follow-on flight of the U.S. Navy DDG-51 Arleigh Burke class destroyers.

For the DDG-51, the U.S. Navy and AlliedSignal Engines developed the concept of applying APU technology to provide a source of low-pressure bleed air that is expanded through an air turbine starter to start the ship's gas turbine generator sets. This system replaces the high pressure air system (HPAS) which consists of banks of 3,000 psi air flasks and pressure-reducing valves, large recharge compressors, coolers, air-water separators, high-pressure piping, and numerous isolation valves. The benefits of the APU system include an installed weight for three units of about 4,500 pounds as compared to about 24,000 pounds for the HPAS system plus the availability of three 75 kW emergency generators driven by the same APUs.

These same style APUs may be applied to ships using diesel engine ship service generator sets and/or diesel propulsion engines by using the APU bleed to power air turbine starter motors geared to the diesel engine crank shaft.

CONSTRUCTION

AlliedSignal has a wide selection of models of APUs available for shipboard applications. A unit representative of this class of APUs is shown in figure 7-38. This is a single-shaft gas turbine. The compressor section consists of a double inlet low-pressure and a single-entry high-pressure radial outward flow centrifugal impeller. The compressor discharge air is diffused into a manifold and directed into the single, reverse flow, annular combustor. From the air manifold, high-pressure air may be extracted between the compressor discharge and the combustor through a bleed control valve as shown in the figure 7-38. Compressor discharge conditions are in the order of 75 psig and 500°F witfi flow capabilities for starting both generator set and propulsion gas turbines. Some other APU models may have multiple tubular combustors and may offer the ability to extract interstage bleed between the two impellers as well as compressor discharge bleed. Hot gas from the combustor transitions through a torus scroll nozzle diaphragm and flows radially inward into the single-stage turbine and exits axially out the rear of the engine. Model variations may also employ axial flow turbine stages and a mix of turbine stages. The weights of the various marine APU models range between 350 and 500 pounds. These APUs have long been used aboard U.S. Navy patrol planes and aboard aircraft carriers to start aircraft and are constructed using materials and coating selected for the marine atmosphere.

Most of the APU operation power is provided in the form of compressed air extracted during the short periods required to start other engines. The second power output is from a gearbox pad which provides shaft power, usually to a generator. This gearbox also provides for the mounting of accessories necessary for APU operation including oil pump assembly with a pressure pump and dual scavenge pump, oil filter, and pressure relief valve; a fuel accessory assembly which includes fuel pump, fuel filter, relief and bypass valves, governor, and fuel shutoff solenoid; electric start motor; and a cooling fan to cool the generator and the engine oil. This oil cooler air is discharged into the APU enclosure cooling air which is educed into the APU exhaust. The generator, usually in the range of 60 to 75 kW, is not shown in figure 7-38.

CONTROLS

The current models are equipped with a fully automatic hydromechanical control system. This system provides all of the control functions necessary for self-starting, monitoring, and shutdown. During operation, the APU runs at a constant speed regardless of load so that a generator (if mounted) provides constant frequency output.

Future APUs will incorporate full authority digital electronic controls (FADEC). The FADEC functions as a full authority controller from start initiation to governed speed, providing all temperature, speed, and fuel

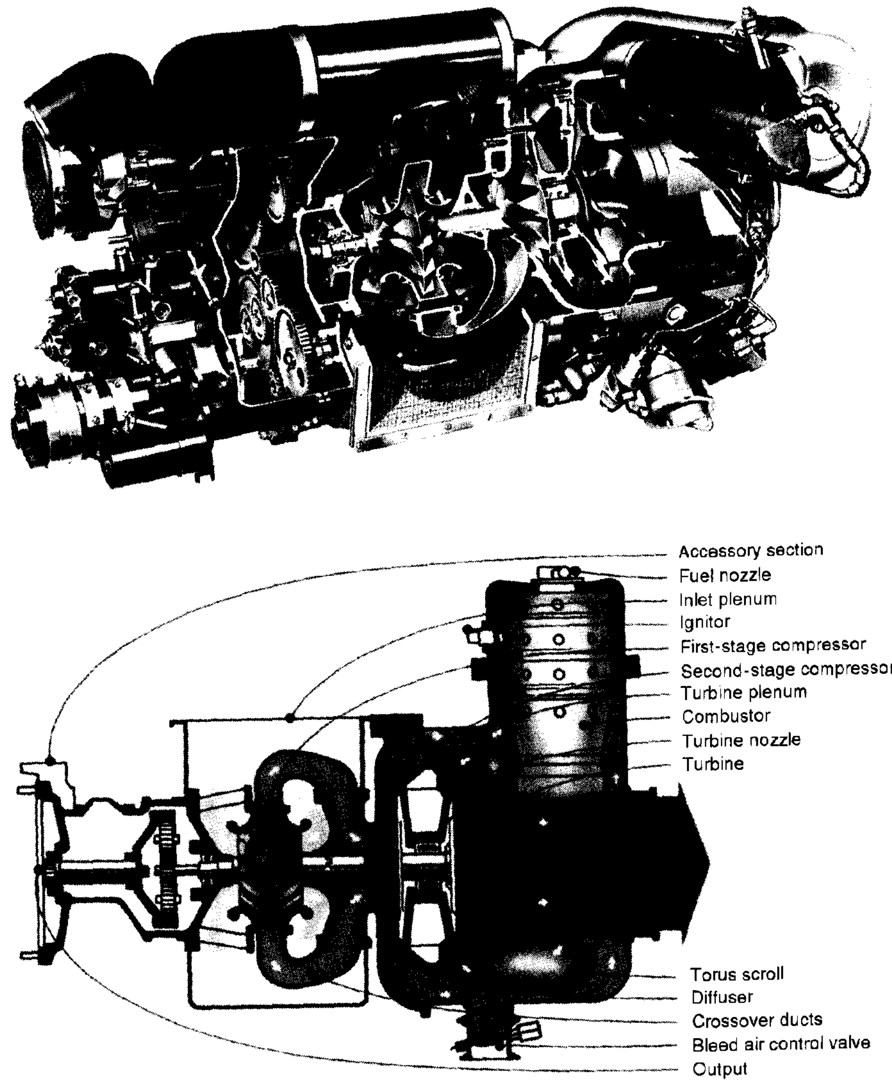

Figure 7-38. AlliedSignal marine APU gas turbine.
Courtesy AlliedSignal Inc.

scheduling, and all emergency shutdown functions. The controller is also capable of self-diagnosis, and displays current or past faults to aid in troubleshooting.

APU starting is accomplished electrically from a dedicated 24 Vdc battery which is located as near as possible to the unit. Recharging ofthe battery is accomplished by an inverter powered from the ship's main service bus. The short APU cycle, typically three to four minutes, is usually not sufficient to allow self-charging.

INSTALLATION

As with all gas turbines, iulet and exhaust ducting must be properly sized and configured as described in the "Propulsion Gas Turbine" section. However, since APU airflow is very small compared with engines to which they provide starter energy, APU ducting can usually be fitted within inlet air and exhaust trunks used for the ship's propulsion or generator drive engines while simultaneously placing them close to these engines to minimize APU bleed air pressure and temperature losses.

The APU operational cycle begins before the engine is running and continues only shortly after engine start while the engine is still at idle. Therefore, the cycles overlap only briefly and at conditions when back pressure is not a problem. Provisions must be made in both the intake and exhaust of the APU for dampers to prevent recirculation of the engine exhaust gases when the APU is not operating.

The APU requires an enclosure for mounting the unit while also providing the necessary interfaces to the ship. The enclosure provides inlet and exhaust duct mounts, cooling air, fuel and electrical interfaces. In addition, the enclosure provides noise suppression, shock isolation, fire detection, and suppression.

Figure 7-39 shows the arrangements of an APU with relation to an Allison AG 9130 service gas turbine generator set as planned for the U.S. Navy DDG 51 class. The APU enclosure is suspended from the overhead above the generator. The APU shares intake and exhaust with the generator set gas turbine, with appropriate dampers to preclude gas recirculation as discussed in the paragraph above. This arrangement minimizes the need for additional ducting and deck space while providing adequate clearance for APU and auxiliary turbine generator set maintenance.

Figure 7-40 is a chart of the typical APU bleed pressure, generator set gas turbine starter air pressure, and generator set gas turbine rpm versus time from initiation of start of the APU to stabilized operating speed of the generator set.

MAINTENANCE

Maintenance of the APU engine is basically as described for propulsion gas turbines, that is, onboard preventative and corrective maintenance, and

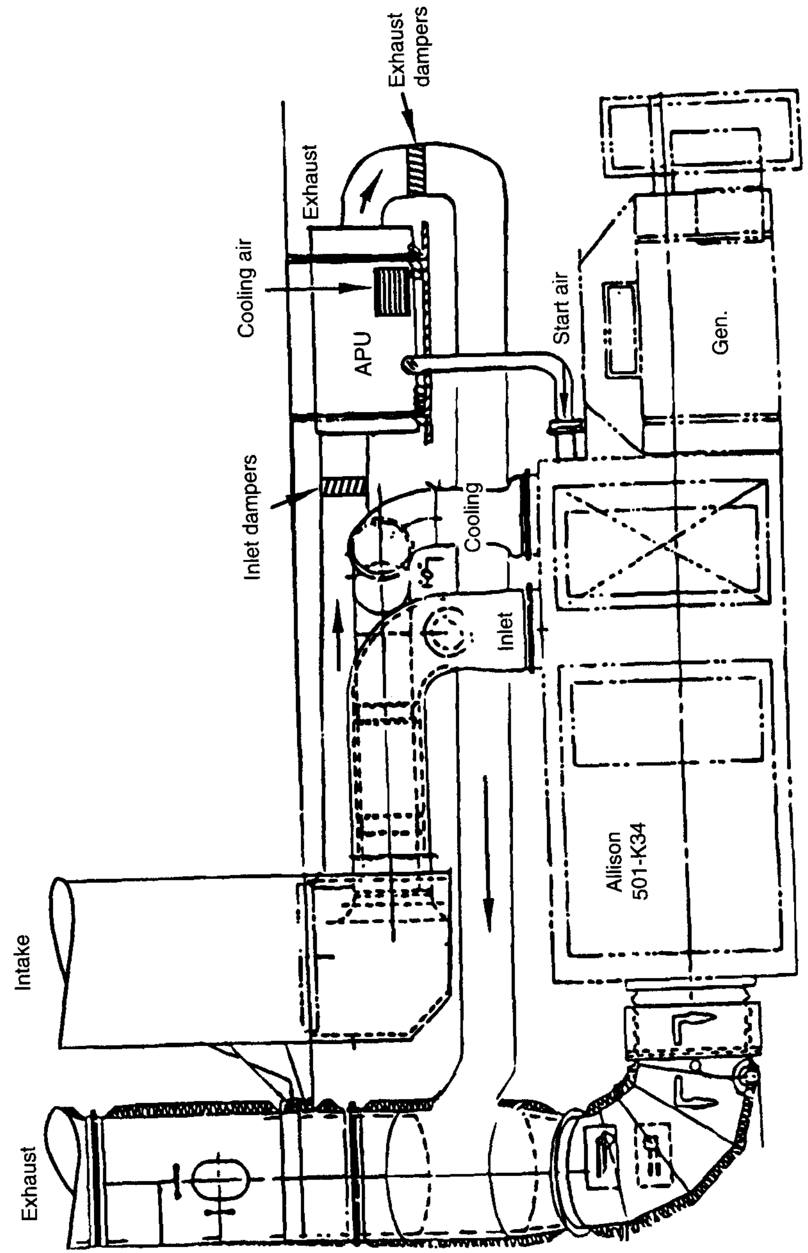

Figure 7-39. Arrangement of APU sharing ducting with generator set. Courtesy AlliedSignal Inc.

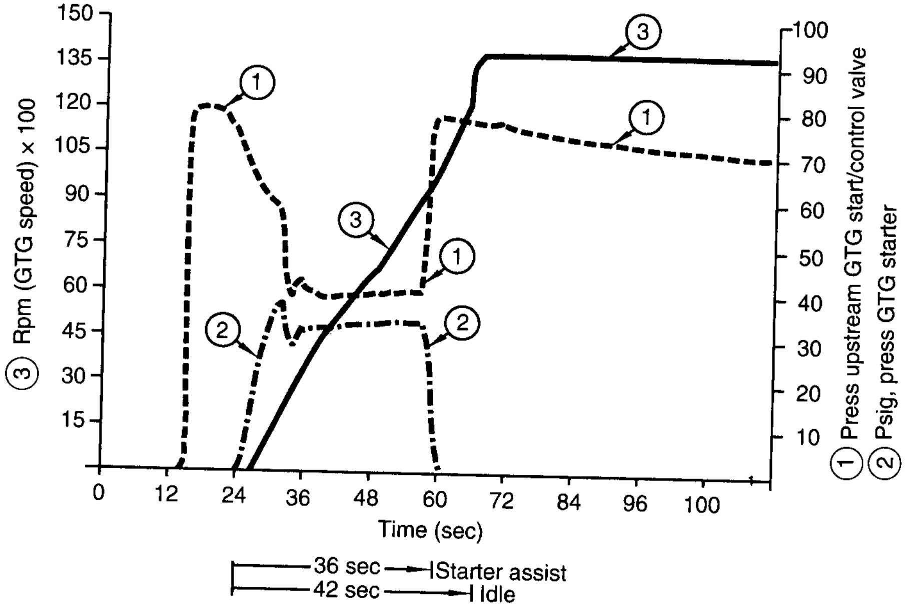

Figure 7-40. Start sequence of AlliedSignal APU and Allison generator set. Courtesy AlliedSignal Inc.

depot repair for hot section repair and overhauls. Preventative maintenance includes oil level checks, filter monitoring, periodic visual mechanical checks of the exterior structures, mountings, piping and wiring, and engine by bores cope examination of internal conditions. These can all be performed with the engine in the enclosure. On-condition monitoring establishes the need for repair or depot level maintenance. Depot level repair interval for shipboard operation may range from 2,000 to 4,000 hours as a function of the ratio of steady-state and short cycles operation.

REVIEW

1. Why have gas turbine ship applications been more numerous in naval ships than commercial ships through 1994?
2. For a simple-cycle gas turbine, if compressor pressure ratio is increased at a constant firing temperature to increase efficiency, what performance penalty may occur ifratio is raised above optimum?
3. Describe two alternatives for utilizing exhaust heat to improve system efficiency.

4. Why would a single-shaft gas turbine have poorer part load efficiency than a free turbine version of the same basic engine?
5. What is the distinction usually made between a regenerator and a recuperator?
6. What type of gas generator configuration might you expect on an ICR engine?
7. Would you select a COGAS or ICR cycle for a naval ship? For a transoceanic cargo ship? Why?
8. For a gas turbine operating at a fixed output rpm and horsepower, what change would occur to turbine gas temperatures and airflow if inlet air relative humidity were to increase? Why?
9. What is the difference between the terms compressor pressure ratio and compression ratio?
10. What three gas conditions determine the amount of power a turbine will produce? What additional gas condition must be able to change sufficiently for a turbine to extract power?
11. What other function may a compressor inlet screen provide beside precluding ingestion of debris into the compressor?
12. What is the distinction between the terms blades and vanes? Which are found in a nozzle diaphragm?
13. Cite five functions engine frames may serve beside supporting the engine rotating elements.
14. Name three uses of compressor bleed air inside the engine.
15. What is a blisk?
16. How do one or two ignitor plugs light the flame in each ofthe cans in an eight-can cannular combustor? Does a gas turbine require continuous ignitor operation?
17. Why are high-pressure turbine blades cast? What type metal crystal or grain structure is desired for high-pressure turbine blades?
18. What design technique is used to minimize cooled turbine blade leading and trailing edge cracking?
19. For a given gas flow condition, how does free power turbine torque vary with rpm?
20. Name three types of motors used to start gas turbines.
21. Why might improperly servicing an engine ignition system be potentially dangerous? How dangerous?
22. Name five major functions provided by a gas turbine enclosure (module).
23. Why is power governing preferable to speed (rpm) governing for a propulsion gas turbine?
24. What limits might be integrated into the propulsion control to bias the gas turbine throttle command to protect other propulsion system components?
25. Name five gas turbine operating parameter limits that could cause the control system to automatically trip (shut down) the engine.

26. Name three conditions that might abort an automatically sequenced engine start.
27. What are nominal limits to gas velocities in inlet and exhaust ducting?
28. Why could outside air of 35°F and 82 percent relative humidity generate intake system icing? Why would outside air of 29°F and 69 percent relative humidity not generate icing conditions?
29. What type oflube oil is used with aeroderived gas turbines? How does lube oil consumption vary with power output?
30. How are salt deposits typically removed from the gas path of a gas turbine engine?
31. Name three different types and sources of gas turbine efflux that would be drained to a ship's atmospheric drain tank.
32. What single action will shut down a gas turbine engine?
33. What method is most effective to monitor engine internal parts condition and trend progression of gradual deterioration?
34. Name three operating factors that can be detrimental to hot section parts' life.
35. Name an operating factor that might limit mechanical life of major engine components.
36. Name four types ofinternal parts' repair that might be performed on a gas turbine aboard ship.
37. Name two routes for removal and replacement of an engine when shoreside depot repair or overhaul is required. What is the typical range of time required to remove and replace a gas turbine?
38. What considerations would you make in deciding whether to select a single-shaft or a free turbine gas turbine for a ship service generator set?
39. Why might an air starter supply APU utilize common ship's intake and exhaust ducting with a gas turbine it is servicing? What special devices would be mounted in the ducts and why?

REFERENCES

3rady, C. O., and D. L. Luck. 1993." The Increase in Use of Gas Turbines as Commercial Marine Engines." ASME paper 93-GT-142. New York: American Society of Mechanical Engineers (ASME).

:::hrisalli, A. J., and M. J. Parker. 1993. "Overview of the WR-21 Intercooled Recuperated Gas Turbine Engine System, A Modern Engine for a Modern Fleet." ASME paper 93-GT-231. New York: ASME.

rancis, George F. 1991. "Testing an APU for Potential Service Aboard a U.S. Navy Destroyer." ASME paper 91-GT-46. New York: ASME.

teneral Electric Company. March 1991. LM2500 Marine Gas Turbine performance data. Cincinnati: General Electric Co.

Giblon, R. P., and 1.H. Rolih. February 1979. "COGAS Marine Power Plant for Energy Saving." Jersey City, N.J.: Society of Naval Architects and Marine Engineers (SNAME).

Halsey, J. E., and D. Russom. 1991. "Design, Development, and Operation Experience of the Allison Model AG 9130 Ship Service Gas Turbine Generator Set." ASME paper 91-GT-46. New York: ASME.

Harrington, R. L., ed. 1992. *Marine Engineering.* Jersey City, N.J.: SNAME.

Higson, A. J., J. A. Ridgway, J. Zakary, and B. B. Tedford. July 1984. "Combined Cycle Propulsion System for LNG Carriers." TP5586E. Montreal, Quebec: Transportation Development.

International Maritime Organization (IMO). July 1992. "An Assessment of the Environmental Contribution and Cost Implications of Reducing Emissions of Sulfur Dioxide from Ships in Europe." Submitted by Oil Companies International Marine Forum (OCIMF). London: IMO.

Jennings, B. H., and W. L. Rogers. 1953. *Gas Turbine Analysis and Practice.* New York: Dover Publications Inc.

Luck, D. L. 1994. "Recent Gas Turbine Applications in Large Commercial Vessels." ASME paper 94-GT-120. New York: ASME.

Shepard, S. B., T. L. Bowen, andJ. L. Chiprich. 1994. "Design and Development ofthe WR-21 Intercooled Recuperated OCR) Gas Turbine." ASME paper 94-GT-79. New York: ASME.

Shieda, T. Kusano. April 1986. "The Application and Fuel Economy of Gas Turbine Combined Cycle for LNG Carriers." Jesmond, New Castle Upon Tyne, England: Northeast Coast Institution of Engineers & Shipbuilders.

Technical Marine Planning Ltd. May 1989. "Comparative Study of Diesel Electric & Gas Turbine Electric Power Plants for Cruise Ships." London: Technical Marine Planning Ltd.

Thompson, Bruce D. 1993. "Recuperating the LM2500 Gas Turbine." ASME paper 93-GT-372. New York: ASME.

United States Navy. December 1984. "Survey and Characterization of Commercial Marine Fuels." U.S. Navy Report DTNSRD-PASD-CR-16-84. Washington: U.S. Navy.

Wilson, David Gordon. 1984. *The Design of High Efficiency Turbomachinery and Gas Turbines.* Cambridge, Mass.: The MIT Press.

Woodward, John B. 1975. *Marine Gas Turbines.* New York: John Wiley & Sons.

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CHAPTER 8

Petroleum Fuels

EVERETT C. HUNT AND RONALD A. lEVA

INTRODUCTION

Since the demise of sailing vessels, fuels have played a major role in the economics and technology of ship operations. Depending on the type and rating of the propulsion plant, fuel represents up to 50 percent ofthe total operating costs of a modern vessel. Fuel can become a significant contributor to the cost of boiler or engine maintenance. The technology of selecting, handling, treating, and burning fuels has become complex in response to internal combustion engine developments, increasing boiler superheat temperatures, and developments in the refining process. This chapter deals with petroleum fuels in the forms used for ship propulsion. Coal fuels are covered in volume 2 of *Modern Marine Engineer's Manual.*

Marine Fuel Oil

Marine fuel has undergone many changes in the past thirty years. Prior to 1970, most diesel vessels burned distillate or blended distillate fuel, and steam ships enjoyed good quality straight run residual fuels from single source crude oils. Traditionally, a residual oil was the remaining result of a refinery extracting distillates from the crude oil of a specific oil field. Quality was somewhat predictable based on the port of delivery, as crude oil sourcing and distribution patterns were relatively stable.

Subsequent to the early 1970s, refinery improvements, political and national upheaval, and simple economics caused dramatic changes in marine fuel quality and availability. Large main propulsion diesel engines began to burn heavier residual fuels, and a design revolution occurred in an effort to cope with the deteriorating quality of marine fuel.

Today, although still called residual, almost all marine fuels are blends of refinery waste residual and various cutter stocks or distillates. It is not uncommon to find that the residual components are derived from multiple crude oil sources. Cutter stock is always derived from multiple crude oil sources. This process, which solves a waste disposal problem for the refinery, makes the residual usable by ships as it is distributed throughout the bunker supply system.

Crude Petroleum Sources

There are hundreds of global crude petroleum sources, each having different physical and chemical 'properties. Even within the same geographical area, each is a unique blend of chemical compounds that are all in general combinations of hydrogen and carbon, hence the term hydrocarbon fuel.

Crude petroleum compounds can be divided into four general categories: *paraffinic, aromatic, naphthenic and olefinic.* These can be further subdivided into specific molecules based on the number of hydrogen and carbon atoms in their structure, the resulting compounds being determined by the number of each atom and how they are arranged. This can become quite complicated and is beyond the scope of this text. For the purpose of this text, it is enough to recognize that the nature and structure ofthe molecules will determine the properties of the mixture of compounds called crude oil.

Refining Process

Crude petroleum compounds differ in their chemical and physical properties, and it is these differences that ultimately determine the nature and quality of marine fuels. One property, the boiling temperature of each of the many different fractions, is the basis for refining crude petroleum into some well known *light ends* such as butane, gasoline, naphtha, kerosene, and gas oil.

The quest for greater refinery efficiency to meet increasing demands for hydrocarbon fuels and petrochemical feedstocks has resulted in refining methods that produce an ever deteriorating quality of residual product.

In the distillation process, both atmospheric and vacuum, crude petroleum is heated and enters distillation towers where vapor components are fractionated into several light ends. Residual from this process is good quality.

The *cracking* process uses the residual from distillation units as a feedstock. Higher temperatures are used, resulting in the cracking of molecules into lower boiling point components. Catalytic cracking is a development that has been used universally for many years to further improve the production yield of a variety of products from crude. A catalyst material is used to promote or accelerate the cracking process.

Visbreaking or thermal cracking is a cracking process that produces a lower viscosity residual to help meet heavy fuel viscosity specifications. Visbroken products are reactive and can result in precipitated sediments days after they are produced.

Coking is another thermal process for converting low grade feedstocks into light ends. This process is characterized by longer reaction times. The principal refining processes for crude oil are illustrated in figure 8-l.

It is evident that improvements in these and other methods will continue to result in greater light end production at the expense of marine fuel in the form of blended residual. In fact, several papers have suggested that as refineries become increasingly efficient at meeting the demand for light petroleum, it is quite possible that the unlimited availability of heavy marine fuels could become uncertain.

MARINE FUEL PROPERTIES

There are many chemical and physical properties of fuel oil about which the marine engineer must be knowledgeable. These properties determine the handling, storage, combustion, and postcombustion characteristics of the fuel.

Marine fuels are classified as *gas oils, marine diesel oils, intermediate fuel oils,* and *heavy fuel oils.* Fuels within these four broad categories share the same properties but have significant differences in magnitude. In addition to the four categories, different grades have been identified in specifications now in general use in the marine bunkering industry. These specifications have been established by ISO (International Organization for Standardization) and CIMAC (International Counc~ on Combustion Engines). There are also BSMA (British Standard) and A:STM(American Society of Testing Materials) standards for marine fuels, as well as numerous specifications by individual engine builders and shipowners.

Some specifications for key physical and chemical properties evolved as a result of serious declines in fuel quality. Dramatic increases in maintenance and engine damage, as well as an inability to properly store and treat the fuel, produced specifications that addressed important purchasing and operational issues. The marine engineer should be familiar with the following important characteristics.

Density

Density is the mass of a substance divided by its volume. Marine fuels are purchased with density measured at 15°C (59°F) expressed in kg/m^3 using ISO procedure 3675. Density changes with temperature; the rate of change is dependent upon the coefficient of expansion of the fuel.

The density of a fuel is a measure of its economic value, and an accurate measurement is necessary to calculate the actual quantity offuel delivered to a vessel. Knowing a fuel's density is also necessary for the efficient

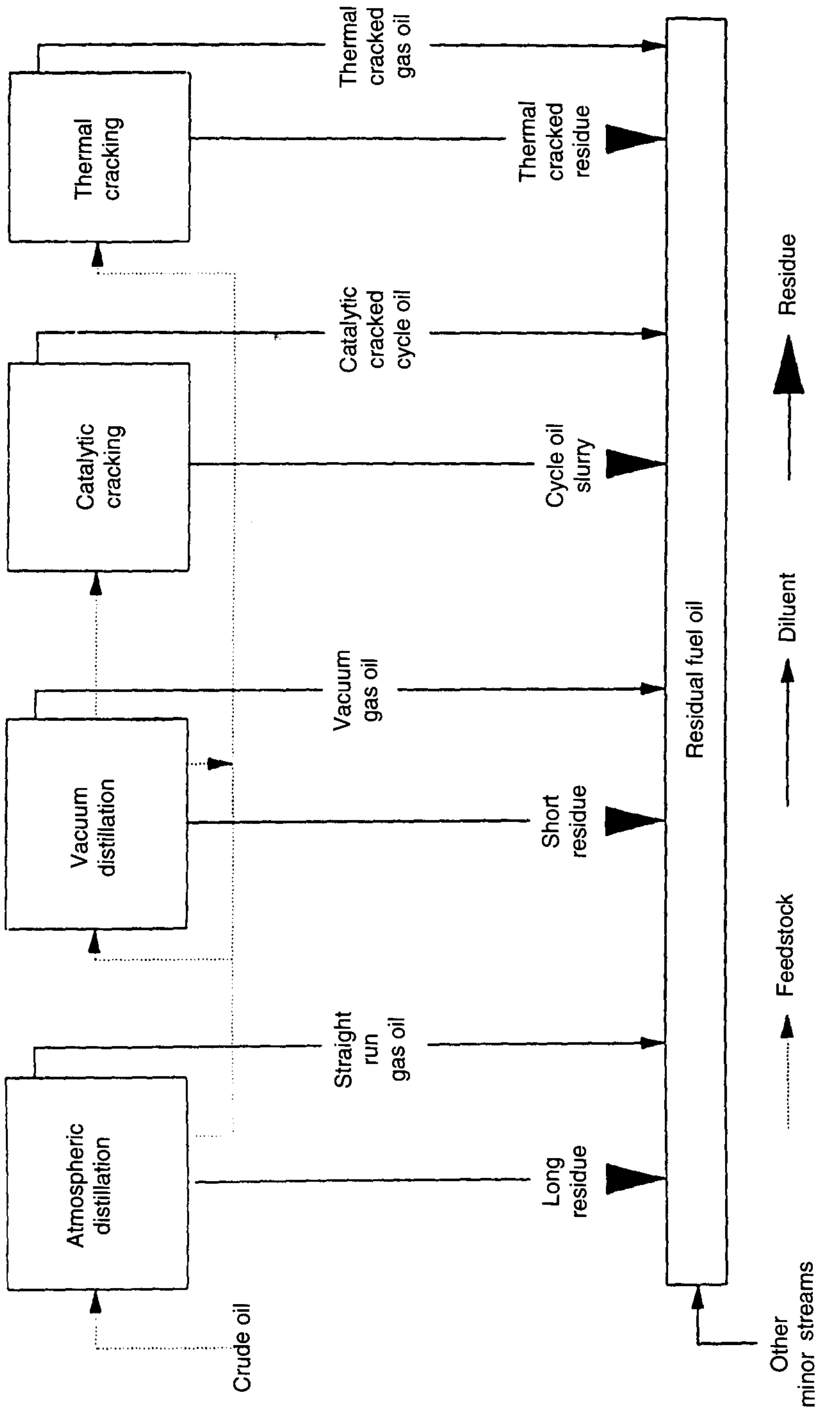


Figure 8-1. Residual fuel manufacture process

operation ofthe fuel separator (purifier), specifically in the selection of the correct "gravity disk". For a conventional centrifuge, the density ofthe fuel should not exceed 991 kg/m3 (61.87 pounds/cu.ft.) at 15°C. Many modern vessels are equipped with separators that can deal with densities up to 1,010 kg/m3 (63.05 pounds/cu.ft.) at 15°C.

There is a correlation of density and hydrogen content of the fuel, and this factor is important with regard to the energy content of the fuel. Also, if this factor is used in conjunction with viscosity, the ignition quality ofthe fuel can be calculated.

Specific Gravity

Specific gravity is the ratio ofthe density of oil to the density of water at the same temperature. There are no specifications for specific gravity of marine fuel.

Viscosity

Viscosity is the measure of a fluid's resistance to flow. Marine fuel specifications use kinematic viscosity at lOO°C (212°F) with units of measurement in centistokes. A centistoke equals one mm squared per second. The procedure for measuring viscosity is described in standard IP 3104. Since samples offuel are usually at lower temperatures, a conversion table (table 8-1) is used to convert to the 100°C viscosity value.

TABLE 8-1
Viscosity Estimated for Different Temperatures

Measured at	Kinematic viscosity, mm^2/sec* Approximate estimations at:			
IOOoe	40°e	50°e	BOoe	130°e
10.0	80	50	17	5.5
15.0	170	100	28	7.5
25.0	425	225	50	11.0
35.0	780	390	75	14.5
45.0	1,240	585	105	17.5
55.0	1,790	810	130	20.5

* 1 mm^2/sec: 1 cSt.

Although it is the measure by which fuels are traded and priced, viscosity is by no means a measure of quality. Knowledge of accurate viscosity is required to determine ifthe fuel delivered to the vessel meets the viscosity specification. Knowledge of viscosity is also needed to determine correct pumping temperature, since the maximum pumping viscosity is approximately 600 cSt at 50°C. This is especially important in cold climates, where fuel tanks and transfer lines may be subject to external cooling effects.

An accurate viscosity measurement is also needed to determine optimum preheating temperatures for effective separator operation and to achieve the correct injection viscosity for each specific engine. This is typically in the range of 10 to 15 cSt at 100°C.

Flash Point

Flash point of a fuel is that temperature at which the vapors given off by the fuel can be ignited by an external ignition source. The minimum permissible flash point of marine fuels is 60°C as regulated by SOLAS (International Convention for the Safety of Life at Sea) using ISO Standard 2719, Pensky-Martens cloed-cup method.

Pour Point

Pour point is the lowest temperature at which a fuel will continue to flow when cooled in accordance with the test method. ISO Standard 3016 procedure is used for heavy fuel. Pour point is expressed at 3°C above the highest temperature at which the fuel will not flow. Pour point is important in the determination of minimum storage temperatures. If the fuel temperature is allowed to fall below its pour point, wax will precipitate. This phenomena can result in filter blockage and fouling of heating coils and heat exchangers.

Chemical Properties

ASPHALTENES

Asphaltenes are large hydrocarbon molecules having a high carbon to hydrogen ratio and covering a wide range of aromatic compounds with high molecular weights. High asphaltene levels generally result in the formation of precipitated particles during storage, handling, and combustion. Asphaltenes are a principal factor in the formation of sludge and sediment in fuel storage tanks. There is no ISO specification for asphaltenes

TOTAL SEDIMENT

Total sediment is the sum ofthe insoluble organic and inorganic materials that can be separated from the fuel by a filtration process after aging. Sediment results from instability of the fuel, causing increased sediment levels with the passage of time and the application of heat. Incompatibility among the blended residual fuel components is another contributor to the total sediment content. A blended residual fuel can be incompatible due to the nature of the blend components, and/or it can be incompatible with fuels already present in fuel tanks at the time of bunkering. ISO and CIMAC heavy fuel specification for total sediment after aging is 0.10 percent.

WATER

Residual fuel leaving the refinery contains 0 percent water. Water enters the fuel through contamination offuel storage and delivery facilities. Once in the fuel, water can be responsible for sludge formation, corrosion, microbial fouling, overloading of purifiers, poor engine performance, and even stopping the engine. Water in the fuel can also cause loss of fires in boilers. Furthermore, any water present in fuel is purchased at the same price as the fuel resulting in an unnecessary operating cost for the vessel. ISO/CIMAC specification for water is 1 percent.

SULFUR

Sulfur is a natural soluble contaminant offuel. Typical sulfur levels are in the range of 2 to 3 percent; however it is not uncommon to find sulfur well in excess of 4 to 5 percent. Sulfur has a low calorific value and can cause reductions in power output or heat release. The principal effects, however, are to cause low temperature corrosion and SOx emissions in engines, and acid attack to the heat transfer surfaces and structural components in the low temperature sections of a boiler. ISO specification for sulfur is 3.5 percent for grades RMAI0, RMBI0 and RMCI0; 4 percent for grade RMDI5, and 5 percent for all other heavy fuel grades.

VANADIUM

Vanadium is present in marine fuel as a soluble contaminant and cannot be removed by mechanical means. The principal effect of vanadium is the formation oflow melting point compounds, which can be very corrosive to high temperature exhaust components in engines and superheaters in boilers. These compounds are formed when vanadium combines with sodium, which is also soluble and primarily present in the fuel as a result of seawater contamination. ISO/CIMAC limits for vanadium range from 150 ppm to 600 ppm.

ALUMINUM AND SILICON

The presence of aluminum and silicon (catalytic fines) in marine fuels is a result of catalyst contamination or carryover from the catalytic cracking process at the refinery. These are extremely hard contaminants and can cause excessive and rapid wear of engine and fuel system components. These particles also settle out of the fuel during storage, possibly resulting in high concentrations in tank bottoms. Aluminum and silicon can be greatly reduced by effective mechanical pretreatment by centrifugal separators. ISO/CIMAC limit for combined aluminum and silicon is 80 ppm.

Marine Fuel Residue

CARBON RESIDUE

Carbon residue is a measure of the residue of carbon after the volatile components of the fuel have been burned in the absence of oxygen. Several methods are employed to determine carbon residue including ISO Standard 4262, Ramsbottom for distillate grades offuel. The CCR (Conradson carbon residue) test is commonly used for heavy fuels in addition to the MCR method defined by ASTM D 4530 and ISO Standard 10370. Carbon residue values are expre!'!sedin percent by weight with an ISO specification range from 12 percent for a 10 cSt to 22 percent for a 55 cSt fuel.

ASH

Ash content is a measure ofthe noncombustible material in the fuel. These materials can be abrasive, contributing to excessive wear in fuel combustion and exhaust system components of an engine.

Ash content also causes boiler problems such as slagging. The ISO and CIMAC specifications for ash range from 0.10 to 0.15 and are expressed in percent by weight using ISO Standard 6245, "Determination of Ash."

Ignition Quality

Ignition quality is a characteristic that governs the combustion behavior of blended residual fuels. Fuels having a high aromaticity generally have poor ignition quality characteristics. These fuels are double bonded and have high carbon to hydrogen ratios. As such, they are harder to ignite because of the double bonding and the lower hydrogen levels. This phenomena manifests itself as ignition delay, i.e., the time between injection and ignition of the fuel in the combustion space of an engine. If the fuel quality is poor, ignition delay will be excessive, the result being an explosion of the fuel (called engine knocking) rather than a slower controlled burn. These explosions produce very rapid and incredibly high pressure increases in the combustion space, frequently resulting in fatigue and shock failure of engine components.

Although there is no laboratory method for determination of ignition quality of residual fuels at this time, several empirical methods have been developed based on the relationship between density and viscosity. These fuel characteristics have a correlation with the aromaticity level of residual fuels and are as follows:

SHELL METHOD - CALCULATED CARBON AROMATICITY INDEX (CCAI)

$$CCAI = D - 81 - 141 \log\log(V50 + 0.85)$$

$$CCAI = D - 111 - 141 \log\log(V100 + 0.85)$$

BP METHOD - CETANE IGNITION INDEX (CII)

$$CII = (270.795 + 0.1038T) - 0.254565D + 23.708 \log\log(Vk + 0.7)$$

where

D = Density as kg/m^3 at 15°C
$V50$ = Kinematic viscosity in cSt at 50°C
$V100$ = Kinematic viscosity in cSt at 100°C
Vk = Kinematic viscosity in cSt at T C
T = Temperature C

Most diesel engine builders have acceptable limits for CCAI and in some cases CII. Observation and testing indicate that fuels with CCAI values in the range of850 to 890 may begin to result in excessive ignition delay, with a higher probability as the value increases. Figure 8-2 is a nomogram for deriving CCAI and CII values. In this nomogram, the extension of a line connecting the viscosity and the density will yield the CCAI or CII values.

Table 8-2 contains the requirements for marine distillate fuels as established by the International Organization for Standardization in ISO Standard 8217. Table 8-3 shows the ISO Standard 8217 requirements for marine residual fuels.

MARINE BUNKER INDUSTRY

Marketing and purchasing of marine bunkers is a worldwide activity requiring a complicated relationship among suppliers, buyers, authorities, and middlemen. Bunkers may be purchased from major oil companies, independent suppliers, traders, and agents. Brokers also have a role to act for buyers or sellers, and provide an interface between buyers and the numerous suppliers.

Since most marine fuels are residuals, a leftover product of refinery operation, marine bunker prices are influenced by availability rather than demand. Prices are determined by such factors in a given region as price of crude, type of crude, number and type of refineries, regional industrialization, and the availability of oil. Prices vary significantly among regions, providing an opportunity for ship operators to carefully schedule bunkering to take advantage of regional costs.

Fuel Quality and Price

All ship operators will wish to purchase fuel at the best possible price while getting appropriate value for the expenditure. Generally, a ship manager will purchase the highest viscosity fuel compatible with a ship's propulsion

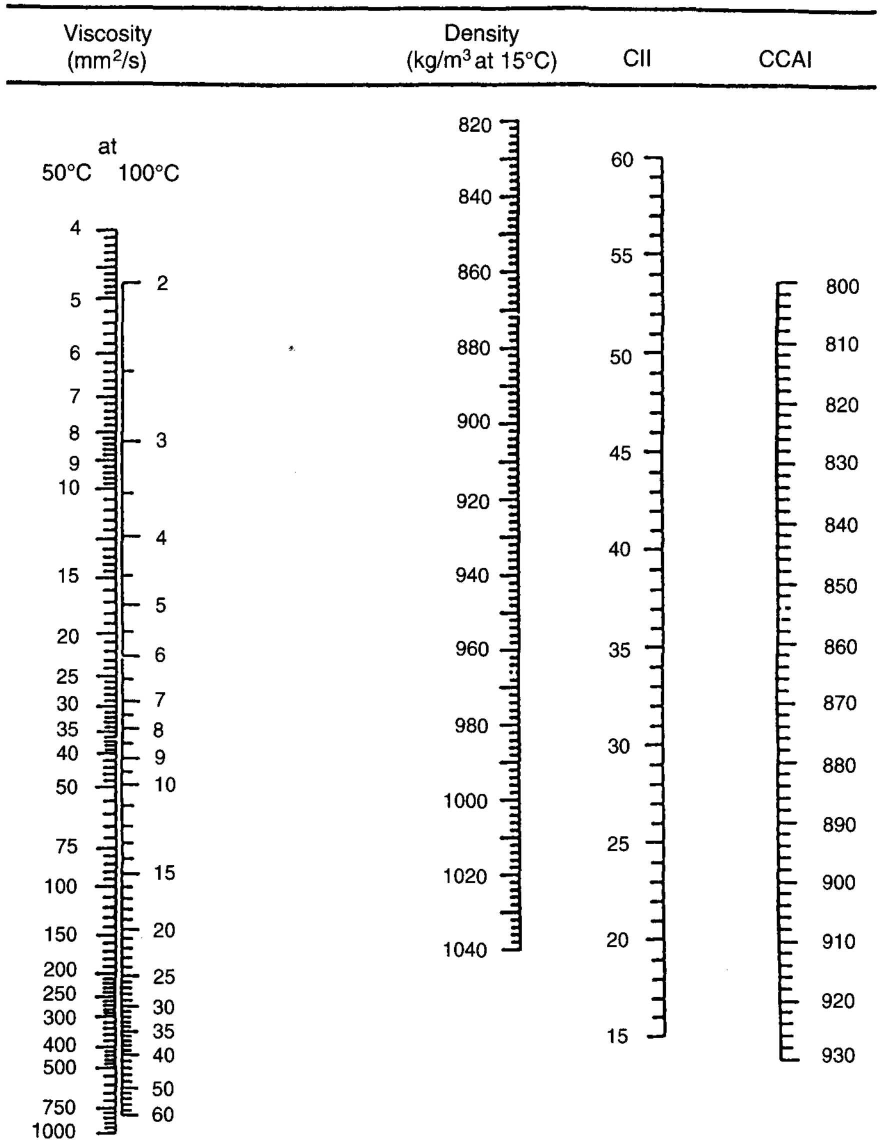


Figure 8-2. Nomogram for deriving CCAI and CII

plant, since this is the least expensive. However, there are other considerations evaluated by the experienced buyer.

In some ports, the highest viscosity fuel available may be less than the maximum usable by the vessel. In such cases, there is no advantage in specifying the unavailable higher viscosity fuel. In a few locations where

TABLE 8·2
Requirements for Distillate Fuels

Characteristic		*Designation, ISO-F*			
		DMX	*DMA*	*DMB*	*DMC*
Appearance		*Visual*		-	-
Density at 15°C, *kg/m³*	max	*	890.0	900.0	920.0
Viscosity at 40°C, *mm²/s* t	min	1.40	1.50	-	—
	max	5.50	6.00	11.0	14.0
Flash point, °C	min	43	60	60	60
Pour point (upper), oct					
Winter quality	max	-	-6	0	0
Summer quality	max	-	0	6	6
Cloud point, °C	max	-16§	-	-	-
Sulfur, % (m/m)	max	1.0	1.5	2.0	2.0
Cetane number	min	45	40	35	-
Carbon residue (micro), (10%b), % (m/m)	max	0.30	0.30	-	-
Carbon residue (micro), % (m/m)	max	-	—	0.30	2.50
Ash, % (m/m)	max	0.01	0.01	0.01	0.05
Sediment, % (m/m)	max	-	—	0.07	-
Total existent sediment, % (m/m)	max	-	-	-	0.10
Water, % (VN)	max	-	-	0.3	0.30
Vanadium, mg/kg	max	-	-	—	100
Aluminium plus silicon, mg/kg	max	-	-	-	25

* In some areas there may be a maximum limit.

t 1 mm²/s = 1 cSt

t Purchasers should ensure that this pour point is suitable for the equipment on board, especially if the vessel is operating in both the Northern and Southern hemispheres.

§ This fuel is suitable for use at ambient temperatures down to -15°C without heating the fuel.

the fuel dealers may not check all the characteristics of bunker fuel, there is an advantage to the ship operator to specify the heaviest fuel compatible with the engine, since the supplier will have standardized the viscosity and checked other characteristics to ensure that the international standard has been met. The alert buyer will approach all reliable fuel suppliers in a given port, since there is sure to be a range of prices. Suppliers purchase different residuals and blending components, which contribute to the different prices. To ensure that the ship has a reasonable expectation of obtaining a desired fuel quality level, the buyer should always purchase fuel rated to an international standard grade. To effectively compare prices offered by the numerous suppliers, the buyer should include density or specific gravity as one ofthe fuel purchase properties. Since fuel is

TABLE 8-3
Requirements for Marine Residual Fuels

Characteristic		*Designation ISO-F-*														
		RMA 10	*RMB 10*	*RMC 10*	*RMD 15*	*RME 25*	*RMF 25*	*RMG 35*	*RMH 35*	*RMK 35*	*RMH 45*	*RMK 45*	*RML 45*	*RMH 55*	*RMK 55*	*RML 55*
Density at 15°C, kg/m^3	max	975.0	981.0		985.0	991.0		991.0		1,010.0	991.0	1,010.0	—	991.0	1,010.0	—
Kinematic viscosity at 100°C, mm^2/s	max	10.0			15.0	25.0		35.0			45.0			55.0		
Flash point, °C	min	60			60	60		60			60			60		
Pour point (upper), °C																
Winter quality	max	0	24		30	30		30			30			30		
Summer quality	max	6	24		30	30		30			30			30		
Carbon residue, % (m/m)	max	10		14	14	15	20	18	22		22		—	22		—
Ash, % (m/m)	max	0.10			0.10	0.10	0.15	0.15	0.20		0.20			0.20		
Water, % (V/V)		0.5			0.8	1.0		1.0			1.0			1.0		
Sulfur, % (m/m)		3.5			4.0	5.0		5.0			5.0			5.0		
Vanadium, mg/kg	max	150		300	350	200	500	300	600		600			600		
Aluminium plus silicon, mg/kg	max	80			80	80		80			80			80		
Total sediment, potential, % (m/m)	max	0.10			0.10	0.10		0.10			0.10			0.10		

Purchasers should ensure that the pour point is suitable for the equipment on board, especially if the vessel is operating in both Northern and Southern hemispheres.

purchased by the tonne but is measured at delivery by volume, there is a potential difference of up to 2 percent in weight of fuel purchased.

Not all fuel suppliers are reputable businessmen. Buyers must be alert to dishonest behavior such as pumping less fuel than specified on the invoice, or failure of the delivered product to meet the specified properties. For example, in a July 14, 1995, issue of *Trade Winds* it was reported that two suppliers in the Port of Rotterdam fraudulently received 3.3 million U.S. dollars over a two-year period by delivering less fuel than invoiced. A survey of tramp operators during a two-year period indicated that the average operator had 3.4 complaints for fuel deliveries that failed to meet specifications.

Bunkering Process

Bunkers are most commonly loaded from barges. However, tanker and passenger ship terminals often have bunkering pipeline service on the pier. An uncommon form of bunkering is the tanker truck. The typical bunkering barge is self-propelled, with a cargo capacity of about 2,000 GRT and a pumping capacity of 200 cubic meters per hour (880 gpm).

Numerous complaints by shipowners concerning fuel quality, fuel quantity, delivery procedures, onerous contract language, and the responsibilities and liabilities of the contractual parties have fostered standardized contracts, such as the Baltic International Maritime Council Standard Marine Fuels Purchasing Contract issued in 1995, and formal port bunkering procedures, such as Singapore Bunkering Procedure, first issued in 1992.

Standard contracts deal with difficult subjects such as fuel grades. Methods of fuel measurements and the witnessing thereof are specified. A usually highly controversial subject leading to much distrust is the method of sampling the delivered fuel, the subsequent handling and testing of the samples, and the processing of disputes arising from the sampling. Notification of delivery times and the customs of the port are handled. Contracts specify the documentation that must be presented with the fuel delivery, including the details of the method for acknowledging the quantity delivered. The method of processing claims by either party is stated in the contract. Definition of title and the passing of title are also covered.

Responsibility for safety and spillage is defined. The arbitration method is outlined. Such standard contracts are designed for fairness to both parties and are very useful in avoiding mistrust and disputes in the bunkering process.

Some major ports, such as the Port of Singapore, have published comprehensive booklets describing the bunkering process and associated regulations. A booklet does not replace the contract between buyer and seller, but it does set forth a procedure to ensure that the buyer is satisfied with the process. The Port of Singapore bunkering procedure covers predelivery, delivery, and postdelivery checks and documentation including bunker requisition form, supplier's noncargo tank declaration and inspection form, tank measurement and calculation form, and the bunker delivery receipt. The process includes provision, if required, for the buyer to retain the services of a surveyor to monitor and verify the quality and quantity of the fuel delivered. The attention to process details by the Port of Singapore is a response to a history of complaints by ship operators bunkering in the port.

Typical forms used in the bunkering process are included in figures 8-3 through 8-5.

Figure 8-3 shows a noncargo tank declaration/inspection form, used to identify the contents of the noncargo tanks of the delivery vessel before and after delivery.

Figure 8-4 shows a tank measurement/calculation form, used to identifY the contents of the delivery vessel cargo tanks before and after delivery.

Figure 8-5 shows a bunker delivery receipt. This form acknowledges the quantity and properties of the fuel delivered.

The shipowner/operator usually provides the bunker supplier with the following information prior to delivery:

- estimated time of arrival
- mooring location
- bunkering time anticipated, earliest and latest
- any restrictions on bunkering time
- location of bunker point and access to vessel
- size of bunker connections
- confirmation of grades and qualities
- independent bunker surveyor, if any

SAFETY CONSIDERATIONS DURING BUNKERING

Specific details of precautions to ensure safety during bunkering will be found in any seagoing vessel's safety manual, which is required by the IMO mandated safety management system. However, it will be useful to summarize the considerations here.

Fire

Both the ship's personnel and the bunkering barge crew must take precautions during bunkering, including no smoking on deck, no unguarded electric lights on deck, no unapproved battery operated equipment on deck.

Mooring

Mooring of vessels alongside must be adequate to ensure safe delivery operation. The receiving vessel should assist the barge crew in mooring activities. The mooring should be regularly checked by both crews.

NON -CARGO TANK DECLARATION / INSPECTION FORM

PRE-DELIVERY DECLARATION (PART I)

To : The Chief Engineer

Date : ____________

MV/SS : ____________

Location : ____________

Name of Barge : ____________

Dear Sir

SB No : ____________

We hereby declare the pre-bunkering measurements and contents of the non-cargo tanks or our tanker as stated below. You are invited to verify the declared measurements and contents.

Tank No/ Compartment	Declared Before Bunkering			Remarks
	Contents	Gauge ( )	Volume ( )	
Fore Peak Tank				
Aft Peak Tank				
Cofferdam				
Eng. Fuel Tank				
Ballast Tank				
D/Bottom Tank				

Declaration Made By :

____
Cargo Officer of Bunker Tanker

____
Name in full (Block Letters)

____
Bunker Tanker's Stamp

____
Date/Time

Acknowledged By:

Chief Engineer of Vessel

TANK INSPECTION/GAUGING (PART II)

(To be completed & signed after bunkering process if required by the Chief Engineer)

The following tanks/compartments were inspected:

Tank No/ Compartment	After Bunkering			Remarks
	Contents	Gauge ( )	Volume ( )	
Fore Peak Tank				
Aft Peak Tank				
Cofferdam				
Eng. Fuel Tank				
Ballast Tank				
D/Bottom Tank				

Witnessed By :

____
Chief Engineer of Vessel

____
Name in full (Block Letters)

____
Vessel's Stamp

____
Date/Time

Comments : ____________

Figure 8-3. Noncargo tank declaration/inspection form

Safe Access

Safe ladder access should be established between the vessels prior to the hose connection to permit safe transfer of personnel. This area must be well lighted during night operations.

Static Electricity

Metal-to-metal contact between the hulls of the receiving and supplying vessels must be avoided to prevent the risk of a static electricity discharge. The oil hose strings must have insulating flanges.

TANK MEASUREMENT / CALCULATION FORM

Note : All tanks must be gauged and witnessed regardless of whether or not they are delivery tanks.

Name of vessel receiving Bunkers:____________________ Name of Bunker Tanker : ____________________ Licence No. SB :__________

The following Bunker Tanker's tanks are gauged and witnessed:-

(I) **TANK READING** :

Opening Gauges & Temperatures						Closing Gauges & Temperatures					
Draft: For'd _____ m Aft ______m. Trim _______m List _______ ° P/S						Draft: For'd _____ m Aft ______m. Trim______m List _______ ° P/S					
Tk No/ Product	Reference Height	Temp °C	Observed Gauge	Corrected Gauge	Gross Vol (Litre)	Tk No/ Product	Temp °C	Observed Gauge	Corrected Gauge	Gross Vol (Litre)	Gross volume Delivered
TOTAL GROSS VOLUME DELIVERED											

(II) **METER READING**: Opening:____________ Closing :__________________

Remarks :____

Opening gauges & Product Temperatures witnessed by Chief Engineer in the presence of Cargo Officer	Closing gauges & Product Temperatures witnessed by Chief Engineer in the presence of Cargo Officer
Signature of Chief Engineer : ______________	Signature of Chief Engineer : ______________
Name in full : ______________ *(Block Letters)*	Name in full : ______________ *(Block Letters)*
Date/Time signed: ______________	Date/Time signed: ______________
Signature of Cargo Officer: ______________	Signature of Cargo Officer: ______________
Name of Cargo Officer: ______________ *(Block letters)*	Name of Cargo Officer: ______________ *(Block letters)*

Figure 8-4. Tank measurement/calculation form

Emergency Communications

The chief engineer and the cargo officer of the supply vessel must establish a communication system between the crews to facilitate emergency shut-down of the bunkering process. Throughout the process, there must be sufficient personnel on the decks of both vessels to take needed emergency actions and to control the process. Regular checks must be made by both vessels to ensure that no leaks exist.

Hoses and Connections

All hoses and connections that are not in use must be blanked off and secured. Hoses should not be subjected to pressure surges during the bunkering

(BUNKER SUPPLIER'S NAME)
(LICENCE NO : __________)

BUNKER DELIVERY RECEIPT BDR NO.__________

DELIVERED AT : ____________________ DATE : ____________________
DELIVERED BY : ____________________ (Name of Bunker Tanker) VESSEL'S NAME : ____________________
SB No. : ____________________ NRT : ____________________
ALONGSIDE VSL : ____________________ OWNER/OPERATOR : ____________________
PUMPING STARTED: ____________________ (Date / Time) NEXT PORT : ____________________
FINISHED : ____________________ (Date / Time) ETD : ____________________

PRODUCT SUPP LIED :

FUEL CHARACTERISTICS		QUANTITY	
Visc. cSt @40°C/50°C (ASTM D445 / ISO 3104)	__________	Gross Vol Litres	__________
Density @ 15 °C (ASTM D1298 / ISO 3675) (ASTM D4052 / ISO 12185)	__________	Net Vol Litres	__________
Water Content % Vol (ASTM D95 / ISO 3733)	__________	Net Metric Tons	__________
Flashpoint °C (ASTM D93 / ISO 2719)	__________	Net Barrels at 60°F	__________ (if applicable)
Observed Tank Temp °C	__________	Volume Correction Factor (ASTM Table 54B)	__________
		Weight Correction Factor (ASTM Table 56)	__________

SUPPLIER'S CONFIRMATION	MASTER'S /CHIEF ENGINEER'S ACKNOWLEDGEMENT
We confirm that the above product was delivered and that the quantities were correct.	We acknowledge receipt of the above product and confirm that samples were taken, sealed and numbered as follows:
	Vessel : __________
	Bunker Tanker : __________
	Surveyor : __________
For __________ Company's Name	Others : __________
	ACKNOWLEDGED BY :
__________ Signature of Cargo Officer	__________ Signature of Master/Chief Engineer
__________ Full Name in block letters	__________ Full Name in block letters
__________ Bunker Tanker's Stamp	__________ Ship's Stamp

REMARKS ____

Was a Note of Protest issued ? Yes / No

Figure 8-5. Bunker delivery receipt

operation. Hoses should conform to a recognized international industry standard. Hoses must be maintained in good condition and regularly pressure tested with documentation ofthe testing available. All flanged connections used in the process must be fully bolted and fitted with proper gaskets.

Fuel Oil Flammability

Since residual oils in storage are capable of evaporating light hydrocarbons into the tank empty spaces, the gases in these spaces are usually in the flammability range. All personnel responsible for handling and storage must be aware of this potential danger and take precautions to (1) avoid excessive local temperatures in the fuel system, (2) keep sources of ignition away from venting systems, (3) be sure that tank heating systems are secure during tank filling or emptying operations, (4) monitor the flammability of residual oil tank void spaces, and (5) follow all local regulations to protect the environment during filling operations.

FUEL ANALYSIS

Vessel economics, operational efficiency, maintenance, performance, and, in some cases, security depend on owners and operators understanding and managing fuel quality. International fuel oil quality standards, laboratory fuel analysis services, and shipboard testing are available to assist owners and operators in this important activity. The first step in fuel analysis is the selection of the fuel sample, which leads to contentious issues such as the point of sampling, acquisition method, sample handling, sample size, and the ratio of sample size to the volume of the delivery sampled.

Sampling

Sampling may be the weakest link in a long and complex process of converting marine fuels to energy. During this process the fuel passes through many hands and is subject to treatments and handling that can significantly alter its physical and chemical properties. Sampling and testing can provide the information to monitor quality and make decisions at any stage in the process. For the shipowner, the samples taken at delivery are the most important and provide the information that will permit appropriate decisions to be made.

Samples collected during delivery to a ship are taken for operational analysis and decision making and to provide information ifthere is a quality dispute with the fuel supplier. The challenge in sampling at delivery is to take a sample, perhaps as little as one liter, that correctly represents the properties of the many tons of fuel being delivered.

Since there are some unscrupulous fuel suppliers in the world, it is unwise for the ship operator to relinquish control of the fuel sampling process.

The receiving ship must specify the sampling method, witness the procedure, and retain control of the samples that are sent to an independent laboratory and/or are tested aboard ship. There are several approaches to sampling.

GRAB SAMPLES

A grab sample is taken at one moment in time. It will be a representative sample only if the fuel is truly homogeneous. With the wide variances in modern fuel quality, this method is least reliable. A mixture of several grab samples is only marginally better.

DRIP METHOD

A number of industry standards recommend this approach to obtain a representative sample. The method requires that a constant drip be taken from the fueling line during delivery and gathered in a suitable bottle to be sealed when full. Experience has shown that this method fails to provide the desired degree of accuracy because the drip is not proportional to the flow of oil. Climatic variations, differences in oil and equipment, and the need for constant supervision are other factors which contribute to inaccuracy in this method. Despite the disadvantages ofthe method, it is in common use aboard ship. Figure 8-6a illustrates the general arrangement of this sampling method.

PROPORTIONAL EXTRACTION

Proportional extraction is a sampling method that uses the kinetic energy of the fuel flow to extract a sample in direct proportion to the flow rate of the delivered fuel. The fuel flow drives an impeller that mechanically operates a positive displacement pump to extract the precise amount needed over time to obtain the correct proportion. The fuel sample is delivered to three sample bottles as shown in figure 8-6b. This device can be adjusted to perform over a wide range of operating conditions. It is a reliable and cost-effective option for shipboard use.

AUTOMATIC PROCESS SAMPLING

This fuel sampling method uses a computer managed needle valve to precisely extract microsamples over the entire fuel delivery period. Shown in figure 8-7, the system is programmable to accept a wide range of operating conditions; it automatically adjusts for variations in temperature, viscosity, flow rate, and interruptions in the flow. The resulting sample is very accurate. The system has been adopted by a number of bunker supply companies to eliminate errors in their operations and provide quality service to their customers.

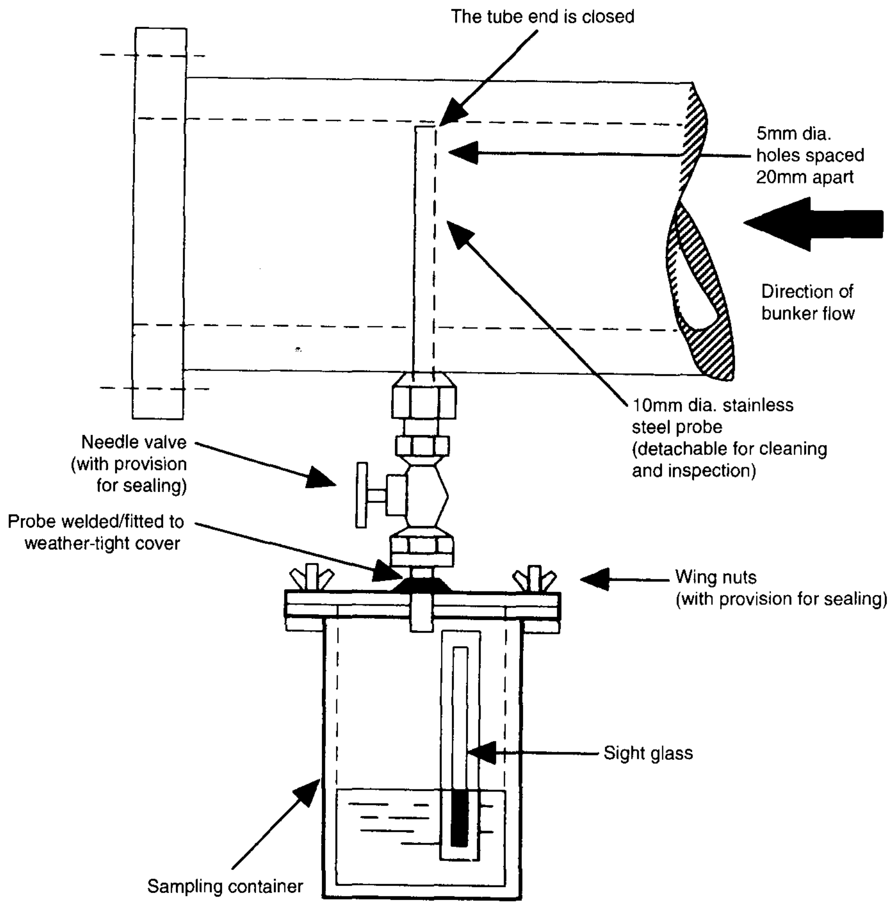


Figure 8-6a. Drip method of shipboard fuel sampling

POINT OF SAMPLING

Legal and technical considerations are involved in the selection ofthe sampling connection. Currently the sampling location is specified as the delivery manifold on board the ship. Since this is the point were title to the fuel changes from the supplier to the shipowner/operator, it is the logical sample point for legal purposes. However, since the homogeneity problems with fuels require equipment that for accurate results may be expensive, shipowners are sometimes reluctant to make the investment. It has been suggested that it would be more logical to take the sample from the supply barge end of the delivery hose using the automatic process sampling. This would provide the best sample at the lowest cost since the frequent use of the sampling device by a delivery barge would result in the lowest cost per sample.

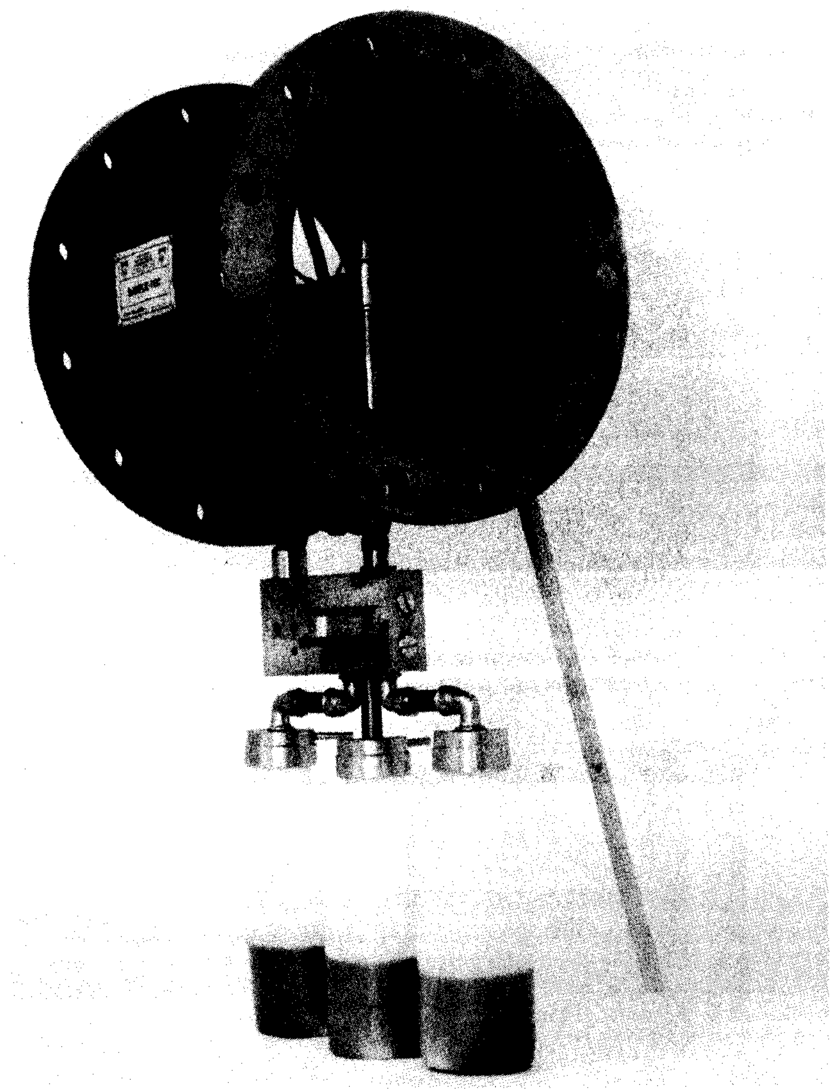


Figure 8-6b. Proportional extraction fuel sampling. Courtesy Ashland Chemical Company.

Shipboard Testing

Most prudent ship operators employ the services of a laboratory, such as ABS Oil Testing Services, to analyze their fuel samples and provide valuable reports. However, there are some basic tests which can be conducted immediately aboard ship to provide useful information prior to the laboratory report.

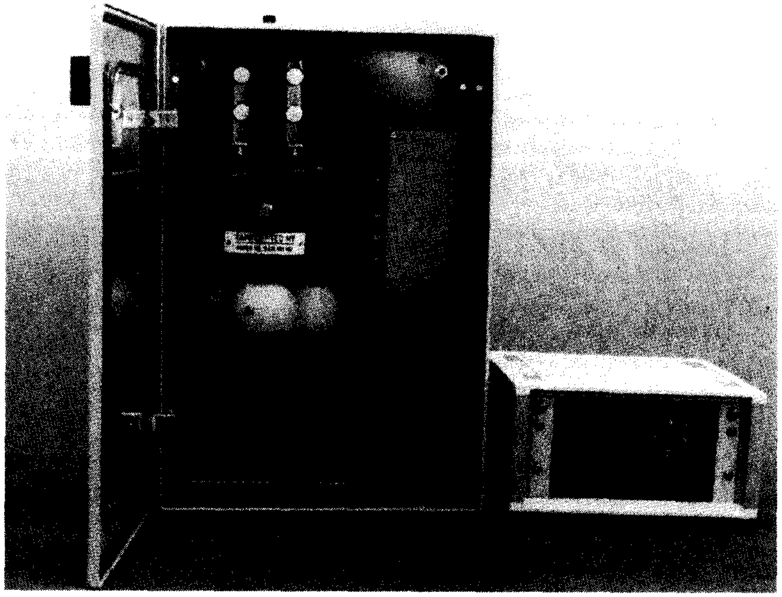


Figure 8-7. Computer managed fuel sampling system. Courtesy Ashland Chemical Company.

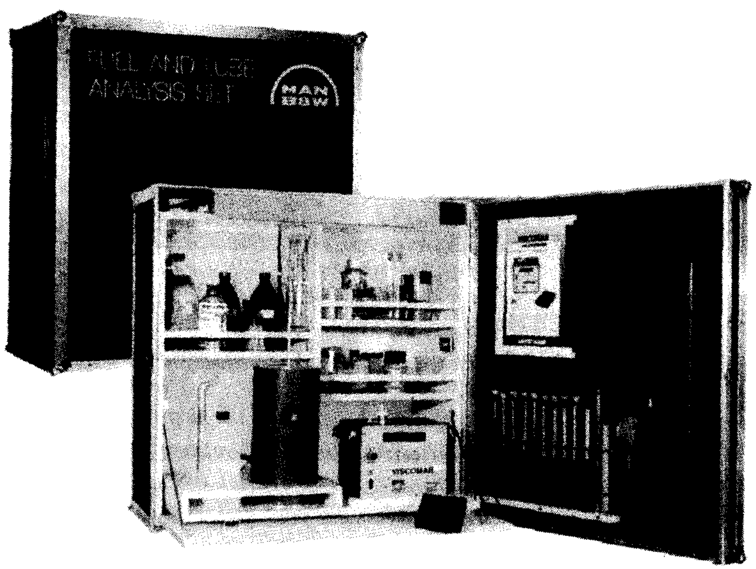


Figure 8-8. Typical shipboard fuel test equipment. Courtesy Ashland Chemical Company.

Water content, viscosity, density, and compatibility are four parameters that can be easily and quickly checked aboard ship to alert the ship's personnel to the need for preventive action. Tests for pour point and fuel ignition quality also provide valuable information to the engineer officers. Osugi reported a survey of 600 bunker operations on 121 vessels from 26 companies in 1981. The survey shows that 23 percent of all bunker fuels led to fuel related engine or system problems. Of these, 58 percent were precombustion problems involving fuel storage, transfer, heating, purification, filtration, and the fuel injection system, and 34 percent were post-combustion problems involving combustion, late burning, ring wear, and liner wear. Such data suggests the merits of shipboard fuel testing to permit remedial action while the fuel is in use. A typical shipboard fuel test package is shown in figure 8-8.

USES OF SHIPBOARD TESTING DATA

Viscosity is the most commonly specified characteristic when purchasing fuel and is the basis for the pricing structure. A knowledge of the actual viscosity is essential for many engine room activities and decisions. The ability to pump fuel from storage tanks depends on setting a tank temperature that will provide a viscosity of 800 to 1000 cSt. Centrifugal separator capacity is a function of the viscosity of the fuel in process. Knowledge of fuel viscosity permits verification of the proper operation of the engine preheaters.

Knowledge of fuel density is needed to calculate tank contents and estimate the fuel calorific values. The following formula for estimation of calorific values of fuels has been developed by the British Standards Institute.

$$CV_{net} = (46.423 - 8.792d + 3.17d)[1 - (x + 8)] + 9.428 - 2.449 \text{ MJ/kg}$$

where

d = density corrected to 15°C (Kg/l)
x = proportion of water, percentage divided by 100
8 = proportion of sulfur in fuel estimated as fuel oil, 2.5 percent; diesel, 1.0 percent; gas oils, 0.0 percent.

Density is the key factor in the selection of the correct purifier gravity disk for fuel treatment to remove water and solids from the fuel oil. The temperature correction factors must be applied to correct to 15°C to avoid acceptance of a fuel that has a density higher than the normally accepted maximum of .991 kg/l for effective treatment. (See fig. 8-9.)

Pour point, the temperature at which wax crystallization occurs to inhibit fuel flow, occurs commonly in the range of 10° to 45°C. The knowledge of pour point permits storage and handling temperature to be maintained 3 to 5 degrees higher to avoid formation of a jelly-like substance.

Most fuel delivered to a vessel contains 1.0 percent or less of water. Testing for water content offuel can alert the engineer officers to excessive water content in delivered fuel. If the water is saline, it may be the result of a leaking tank. Ifit is freshwater, the source may be leaking heating coils. Saline water can cause serious problems with sodium and vanadium, high temperature corrosion, and engine turbocharger fouling. Purifier operation effectiveness can be verified by testing for water content before and after the process.

Onboard test kits have the ability to test for instability (the tendency of a residual fuel to produce sludge) and incompatibility (the tendency to produce sludge when blended with other fuels). The reduction in effectiveness of the fuel treatment system and possible engine damage that may occur because the asphaltenes are no longer in solution are indicated by these important fuel characteristics. Sludge deposits block tank suctions, filters, and pipes and quickly clog separators. In the engine, the sludge blockage of injector nozzles, uneven heat release, late burning, and coking can result in damage to pistons, rings, and liners. The fuel supplier has the responsibility to blend bunkers so that a reserve of asphaltene absorbency is sufficient to ensure stability. If this is not the case, there is no option but to off-load the unstable fuel.

The ship's officers must be alert to the possibility of incompatibility resulting from shipboard mixing of fuels. The only sure method of avoiding incompatibility is to prevent mixing of fuels aboard ship.

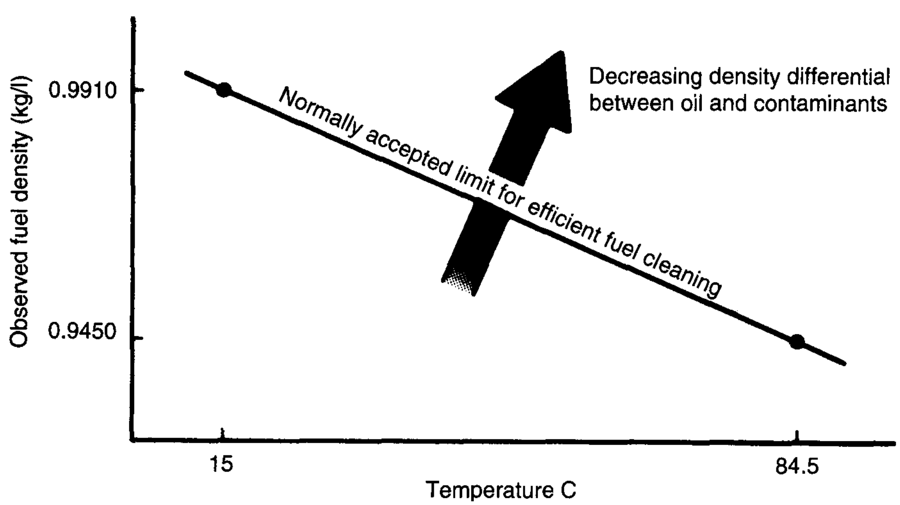


Figure 8-9. Temperature correction for fuel oil density

In general, engineer officers aboard motor vessels should be alert to any ofthe following fuel characteristics that can lead to serious problems:

- fuel from catalytic cracking refineries
- Conrad son carbon residue greater than 10 percent
- density greater than 991 kg/m^3 at 15°C for heavy fuels
- water content greater than 1 percent
- asphaltenes greater than 6 percent

Finally, both laboratory tests and onboard tests are useful only if the engineer officers have the proper knowledge and training for a comprehensive understanding of the significance of the test results.

Laboratory Fuel Testing

There are many laboratory testing services worldwide which provide prompt analysis of fuel samples collected during bunkering and expressed from the vessel to the laboratory. Typical of these laboratories in the United States is the oil testing service of ABS Oil Testing Service. These services provide operators with accurate information regarding the quality of fuel they use and assist them in optimizing the fuel pretreatment operation. ABS Oil Testing Service manages contract laboratories worldwide, with locations in the Americas, the United Kingdom, Rotterdam, Shmjah, and Singapore.

Results of testing are sent to the shore staff and ship within twenty-four hours after the receipt of a sample from the ship. Figures 8-10 through 8-12 show the tests offered by a laboratory service for fuel oils, marine diesel and gas oils, and optional tests. The ASTM and equivalent ISO test methods are also indicated in these figures.

Accurate fuel test results from a laboratory have many valuable uses in addition to the obvious ones concerning information needed for handling, storage, and combustion. Reliable third-party test results are useful in disputes with fuel suppliers and claims against engine manufacturers. A long term record of purchased fuel properties is useful in scheduling ship operations and bunkering to avoid ports and suppliers where only low quality fuels are available. A vessel's fuel properties history can be correlated with engine maintenance and performance problems by multivariant regression analysis to develop mathematical relationships that can be used to predict engine problems before they occur. Table 8-4 shows possible correlations between engine problems and fuel properties.

EMISSIONS TESTING AND CONTROL

The main pollutants designated for control by international, national, and local authorities are oxides of nitrogen (NO_x), which promote photochemical

ABS Oil Testing Services
Fuel Oils (Standard Tests)

Test	Method	Corresponding ISO Test Method
Density at 15°C (hydrometer method)	ASTM D1298	ISO 3675
Density at 15°C (digital meter)	ASTM D4052	
API gravity at 60°F	ASTM D1250	ISO 91/1
Flash point °C	ASTM D93	ISO 2719
Viscosity kin. at 50°C cSt	ASTM D445	ISO 3104
Upper pour point °C	ASTM D97	ISO 3016
Carbon residue % by mass	ASTM D4530	ISO 10370
Ash % by mass	ASTM D482	ISO 6245
Water % by volume	ASTM D95	ISO 3733
Sulfur % by mass	ASTM D4294	ISO 8754
Metals analysis (ppm)		
Vanadium Sodium Aluminum Silicon	ICP spectrometer or atomic absorption IP 377	ISO 10478 (draft status)
Total sediment % by mass	ASTM D4870/IP 375 & IP 390	ISO 10307-2 (draft status)
Net calorific value	By calculation (ISO 2817-Annex A)	ISO 2817
CCAI (Calculated Carbon Aromaticity Index)	By calculation	

Figure 8-10. Typical standard laboratory fuel tests. Courtesy ABS Oil Testing Service.

smog over cities, and sulfur oxides (SOx), which contribute to acid rain. Ship operators can expect to see increasing demands from authorities for ever greater reductions in these emissions.

Improper engine operation and maintenance contribute to emissions problems. However, this chapter deals only with the fuel contribution to the problem. The sulfur oxides emission problem is most economically addressed by burning low sulfur fuel. The current International Maritime Organization standard can be met with a fuel sulfur maximum content of 1.5 percent. Some ports (such as those in California) are far more stringent, requiring sulfur content of .05 percent to meet their standard.

Fuel related methods of reducing oxides of nitrogen emissions include emulsified fuel oil and the use of alternative fuels. The latter approach is usually not economic.

ABS Oil Testing Services
Marine Diesel Fuel, Marine Gas Oil (Standard Tests)

Test	Method	Corresponding ISO Test Method
Density at 15°C (hydrometer method)	ASTM D1298	ISO 3675
Density at 15°C (digital meter)	ASTM D4052	
API gravity at 60°F	ASTM D1250	ISO 91/1
Viscosity kin. at 50°C cSt	ASTM D445	ISO 3104
Appearance	Visual	
Pour point °C (note 1)	ASTM D97	ISO 3016
Carbon reside % by mass	ASTM D4530	ISO 10370
Ash % by mass	ASTM D482	ISO 6245
Water % by volume	ASTM D95	ISO 3733
Flash point °C	ASTM D93	ISO 2719
Existent sediment % by mass	ASTM D4870/IP 375	ISO 10307-1
Sulfur % by mass	ASTM D4294	ISO 8754

Note 1: Cloud point °C to be run in place of pour point for emergency diesel fuel (Marine Gas Oil).

Figure 8-11. Typical laboratory tests for marine diesel fuel. Courtesy ABS Oil Testing Service.

ABS Oil Testing Services
Optional Tests

Test	Method
Additional metals ppm	I.C.P. spectrometer or atomic absorption
Asphaltenes % by mass	IP143 or asphaltenes analyzer
Bottom sediment and water % by vol	ASTM D1796
Compatibility rating	ASTM D2781
Compatibility rating	ASTM D4740
Microbial infection	Dip slides
Total acid number mg KOH/gram	ASTM D664

Figure 8-12. Typical laboratory optional fuel tests. Courtesy ABS Oil Testing Service.

TABLE 8-4
Possible Correlations Between Engine Problems and Fuel Properties

Engine problem /fuel properties	*May be correlated to*
Damage to exhaust valves	CCAI
	CCR
	Water in fuel
	Vanadium
	Sodium
Carbon trumpet formation	CCR
	Asphaltene
Clogging in heaters	Density
	Sulfur
	Ash
	Sediment
Blowby	Sulfur
	Ash
Piston ring wear and breakage	Sulfur
	CCAI
	Water and sediment in fuel
Cylinder liner wear	Ash
	Vanadium
	Silicon
	Aluminum
Engine surging	Flash point
	CCR
	Water and sediment
	Asphaltene
	Sodium
	CCAI
Combustion residue deposits	CCR
	Sulfur
	Sodium
	Asphaltene
Vapor locking	CCR
	Water
	Asphaltene
Fuel pump wear	Sulfur
	Ash
	Sodium
	Aluminum
	Silicon
Inability to transfer fuel	Pour point
	Viscosity
Strainer clogging	Density
	Water and sediment
	Silicon
	Aluminum
Difficult purification	Density
	Flash point
	Water and sediment
	Ash
	Silicon
Sludge	Water and sediment
	Asphaltene
	Biological contamination
Fuel system corrosion	Biological contamination
	Sulfur
	Water
	Sodium

Fuel emissions are expressed in parts per million (ppm), percent Oz and grams per kWh. For example, in 1995, the California emission standards for main internal combustion engines were as follows:

For new ships-130 ppm, 15 percent Oz, which is approximately 1.2 g/kWh.

For in-use ships-600 ppm, 15 percent Oz, which is approximately 10 g/kWh.

A typical four-stroke main propulsion engine operating under full load with residual fuel will have the following composition of exhaust gases:

Nitrogen	Nz	75 percent by volume
Oxygen	Oz	12.3 percent by volume
Inert gasses	Ar	0.9 percent by volume
Carbon dioxide	COz	5.6 percent by volume
Steam	HzO	6.0 percent by volume
Sulfur dioxide	SOz	12 g/kWh
Nitrogen oxide	NOz	16 g/kWh
Carbon monoxide	CO	0.6 g/kWh
Hydrocarbons	HC	OAg/kWh
Soot particles		0.05 g/kWh

In some U.S. ports, the states have established exhaust gas visibility standards. The measure used for the opacity is the Ringelmann test. The Ringelmann test is at best a crude and inaccurate test that uses a standard chart for visual comparison to the stack emission. The prescribed limit is 20 percent for ships.

Emulsified Fuel Effect on Emissions

Investigations and operating experience have shown the fuel-water emulsion has a very beneficial effect in the reduction of NO_x emissions. The stability and water droplet size ofthe emulsion has a significant effect on the emission reduction. Generally, a mixing of 10 percent water to the residual fuel will result in a 6 to 12 percent reduction in NO_x depending on the engine type. For engines with high thermodynamic efficiency, there is little or no change in fuel consumption resulting from the addition of 10 percent water to the fuel.

In addition to the beneficial effect on NO_x emission, the fuel-water emulsion improves the opacity of the smoke. This effect is very noticeable at low loads on supercharged engines. There is no increase in stack particle emission, CO, or HC as a result of a fuel-water emulsion. In 1991, Harbach reported on an extensive laboratory investigation of the effects of emulsified diesel fuel in a four-stroke engine.

MECHANICAL FUEL TREATMENT

Fuel oil systems for diesel engine, steam plants, or gas turbine plants are specified by the ship designers in the shipbuilding contract. As a result there are many variations in the type and quantity of mechanical fuel treatments found aboard vessels. The most complicated systems are required by the modern slow and medium-speed diesel engines burning residual fuels. Despite the variations in fuel oil systems, there are many common functions found in all systems including settling, filtering, heating, purification, clarification, venting, pumping, and monitoring.

Operational and Design Considerations

Figure 8-13 illustrates a typical modern diesel engine fuel oil cleaning system which must be designed and operated properly to achieve the desired result. In such a system, different storage tanks (1) must be provided and used for oil of different origin unless they are proved to be compatible. Oil from the bunker tanks must be delivered to the top of the settling tanks (2) to avoid the possibility of a low temperature at the oil transfer pump suction. Level switches (3) installed and operating in the settling tank provide for topping off the settling tank to avoid temperature fluctuations at the suction point of the transfer pump. To maintain a constant separation temperature in the settling tank, a temperature control (4) must be installed. The temperature in the tank should be constant at a value not below 45°C. Temperature in the settling tank should not be higher than lOoCbelow the oil flash point. The main cleaning purposes ofthe settling tank (5) are to act as a buffer tank, to provide a constant temperature, and to settle and drain gross water contamination. A drain valve (6) must be located at the bottom of the settling tank and used regularly to remove water and sludge from the tank. Both regular and standby positive displacement pumps (7) are installed with one pump operating at constant flow rate. Coarse particles are removed from the oil flow by strainers (8) located in the suction of each transfer pump. Strainers must be monitored for pressure drop and cleaned regularly. A constant flow regulator system (9) provides the desired constant flow rate to the separator and enables the flow to be distributed to the separators when they are operated in parallel.

A control circuit including a sensor (10) is provided with an accuracy of plus or minus 2°C to control temperature entering the separators. If a steam preheater is used, a properly sized steam valve is essential along with a float type steam trap. The main and standby separation systems (11) are installed for single operation with the possibility of parallel operation (12) in the event of excessive water or sludge in the fuel. The overflow pipe (13) from the bottom of the service tank to the settling tank permits return of overflow and permits return of water which may have entered the fuel due to condensation and coil leakage. The service tank bottom (14) is sloped to collect water and sludge. The recirculation line (15) from the three-way valve returns oil to the settling tank. A line (16) for cleaning the service tank is located slightly above the drain line. A suitable sludge tank (17) is used to collect sludge from the separators.

Limitations of Conventional Purifier

In treating heavy oils for use in an engine, the problem is generally the separator gravity disk shown in figure 8-14, a cross section of an Alfa Laval separator. In this purifier, cleaned oil and separated water are continuously discharged during operation. The location of the interface between water and oil is affected by density, viscosity, temperature, and flow rate. To achieve the best separation, the interface must be located between the outer edge ofthe top disk and the disk stack as shown in the figure. For effective operation, it is important that the water seal or separated water never enter the disk stack. This is true for both purifier and clarifier type of separators. The interface position of the purifier is adjusted by means of gravity disks. To get the correct interface position, the gravity disk must be changed, which presents serious operating difficulties when oil properties are changing.

A conventional purifier separator is practical for cleaning fuels with densities up to 960 kg/m^3 at temperatures equal to and above 15°C. When a

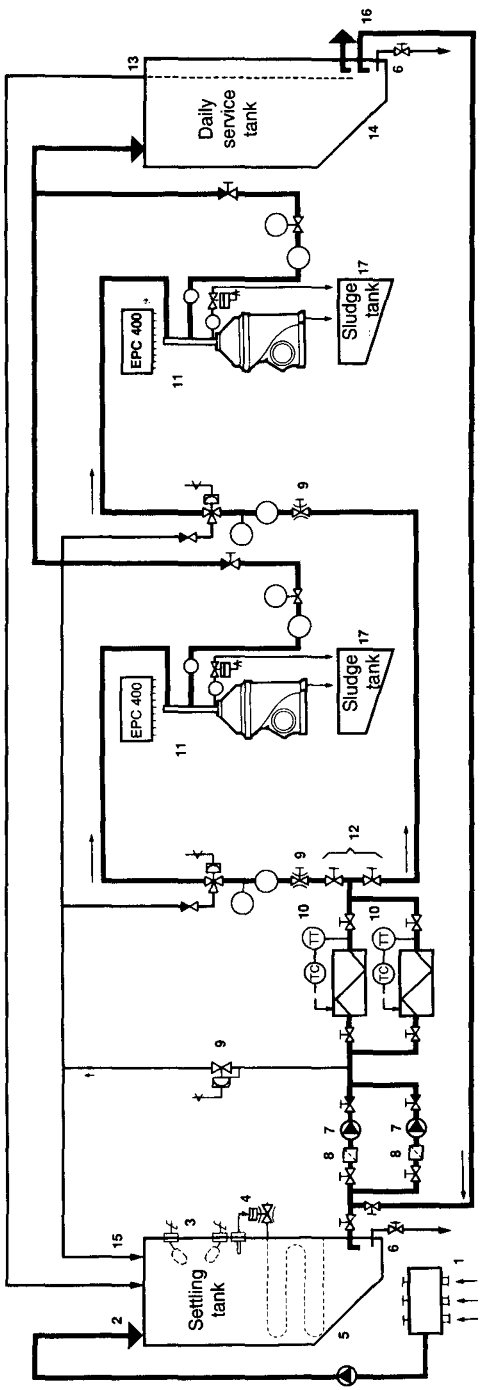


Figure 8-13. Typical oil cleaning system. Courtesy Alfa Laval Separation Inc.

second clarifier separator is added in series after the first separator as a safety backup, fuel oil up to 991 kg/m^3 at 15°C can be cleaned effectively.

ALCAP FOPX Fuel Oil System

Alfa Laval has developed a fuel oil cleaning system which is essentially a clarifier separator with sensors and controls to accurately monitor the water and sludge discharge to ensure that the water does not approach the disk stack as shown in figure 8-15. This system is effective in cleaning fuel oil with density up to 1,010 kg/cu. m. at 15°C and viscosity up to 700 cSt at 50°C. The ability to treat fuels with varying density and viscosity eliminates many operating problems.

Separator Alarm Functions

Typical alarm functions in a mechanical fuel treatment system include emergency stop due to vibrations, low oil flow, high or low oil temperature, no sludge discharge, preheater fault, water transducer fault, insufficient capacity of water drain valve, power failure, microprocessor errors, communication errors, calibration errors, and actual temperature displays.

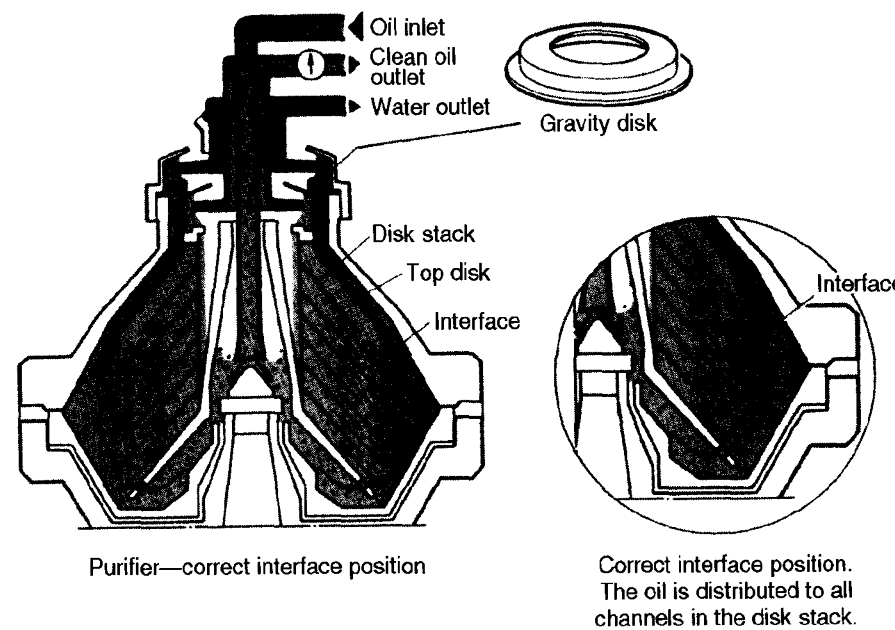


Figure 8-14. Fuel oil purifier.
Courtesy Alfa Laval Separation Inc.

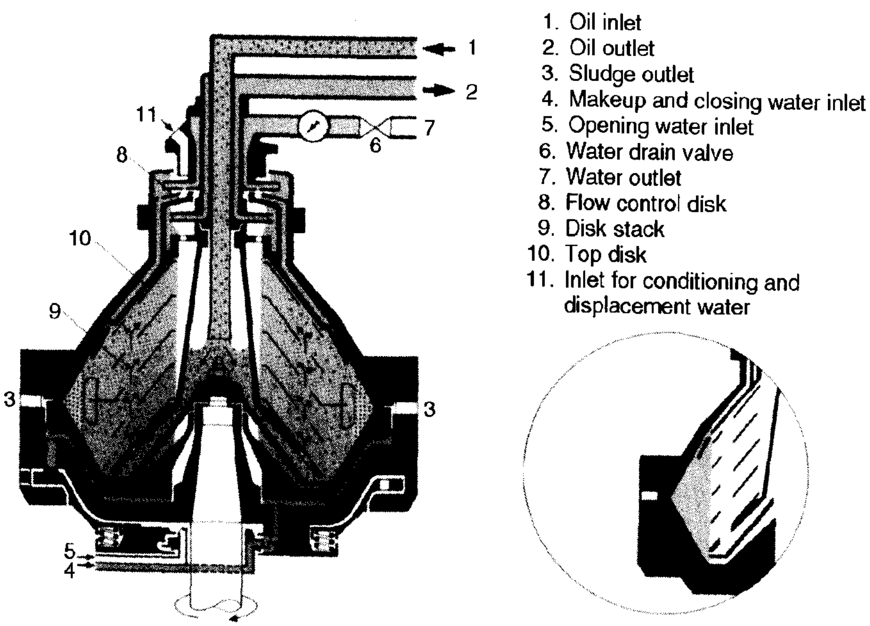
CHEMICAL FUEL TREATMENT

Chemical additives have gained a role in fuel treatment as the quality of residual oil continues to decrease with changes in the refining process. The use of such additives can improve the overall performance of marine boilers and diesel engines.

Marine Boilers

For steam generators, chemical additives containing magnesium are frequently used to ameliorate problems associated with high vanadium, sodium, and sulfur in the fuL Magnesium combines with the vanadium to form high-melting noncatalytic compounds that result in minimal slag formation and the consequential hot and cold end corrosion.

The magnesium may be added in the form of an oxide slurry. This approach has the disadvantage of the need to feed the slurry at a stoichiomet-



FOPX/MFPX separator. Separated sludge and water accumulate at the periphery of the bowl.

FOPX/MFPX—Water approaches the disk stack.

Figure 8-15. ALCAP controlled discharge clarifer/separator. Courtesy Alfa Laval Separation Inc.

ric ratio, which requires that large quantities must be used. The use of magnesium in a soluble organic form minimizes this requirement because of the higher activity. Furthermore, the use of soluble magnesium eliminates the abrasive process which would occur at the burner tips with a magnesium slurry.

A chemical combustion improver is useful for reducing carbon deposits and soot buildup. The combustion improver lowers the energy of activation required for the combustion reaction to occur. Metal catalysts in soluble organic form are most commonly used for this purpose. Magnesium and iron have been found to be most effective as catalysts.

Fuel Handling

Specific handling problems are minimized by different chemical treatments. For organic sludge and water, a solvent/surfactant product is normally used to penetrate the sludge, disperse the water, and keep the filters clean.

For heavy oil burning diesel engines, chemical additives added to bunkers to reduce sludge problems associated with the use of blended fuels have wide acceptance.

Diesel Engines

For medium-speed engines, i.e., engines with exhaust valves, organometallic additives are used successfully to raise the melting points of vanadium, sodium, and sulfur oxide deposits to make them powdery and thereby extend the life of the exhaust valves.

Demulsifiers for free water removal, pour point depression, viscosity improver, and separator aids offer opportunities to improve engine performance.

For combustion related problems, such as carbon and soot deposits, low cetane numbers, ignition delay, and uneven burning, a combination of organometallic combustion improvers are most commonly used with successful results.

Summary

Chemical additives are useful when properly applied but they are not a substitute for careful specification and sampling of fuel purchases and shipboard mechanical treatment offueL Used properly with the advice of an expert, they provide cost-effective improvements in the handling and combustion processes. Critical to the success of chemical additives are (1) correct identification of the problem, with expert advice; (2) selection of the correct additive, with expert help; (3) proper use aboard ship, i.e., dosing and control; and (4) close follow-up ofthe chemical program by the engineer officers.

GAS TURBINE FUELS

Gas turbines will operate on a very wide range of fuels. However, the presence of even minute quantities of trace metals is detrimental to engine service life. During the 1960s, industrial gas turbines and aeroderivative turbines operated aboard Euroliner ships with IFO fuels. These fuels were treated by washing to remove soluble contaminants such as sodium. Magnesium compounds were also added to these fuels to reduce the corrosive effect of vanadium compounds. However, the magnesium treatment produces four parts of contaminant ash for each part of vanadium in the fuel. The ash accumulates in the engine hot gas passages. For engines with low firing temperature, the ash was removed by frequent engine cleaning but if recuperators were used in the system, the ash quickly plugged and fouled the heat transfer surfaces. Today, both the industrial and the aeroderivative gas turbines operate at 2,300°F firing temperature and 14 atmospheres compressor discharge temperature, making the use of distillate fuels essential to reliable and economical operation. Gas turbine manufacturers, like diesel manufacturers, publish fuel specifications for marine applications. The following discussion provides fuel properties and contamination characteristics of distillate fuels that are generally accepted for aeroderivative marine gas turbine use.

The following are specifications for fuels widely accepted for aeroderivative gas turbines. Included is a notation of the minimum flash point (FP) for each specification. While it is not a limit required for the gas turbine, the U.S. Coast Guard and other regulatory agencies have established 140°F (60°C) as the minimum flash point for shipboard fuels for safety purposes. Gas turbine fuels are commonly defined by military specifications.

MIL-T-5624, JP5 (F-44) - FP 140°F. This is an aviation fuel specification. JP5 is used for aircraft aboard ship.

MIL-F-16884 (F-76) (F-75) - FP 140°F. These are two grades of navy marine diesel fuel, MDF. The U.S. Navy has standardized fuels for all types of vessels (steam, diesel, gas turbine) to F-76. The F-75 fuel has slightly higher viscosity and lower pour point.

VV-F-800 (F-54) - FP 125°F. This is a diesel fuel available in a range of grades. Grades DF-a, DF-1 and DF-2 are usually acceptable for aeroderivative engines.

ISO 8217:1987 Grade ISO-F-DMA, FP 140°F - FP 140°F. This is a marine gas oil.

ASTM D2880 '92 No. O-GT, 1-GT, 2-GT. - FP lOO°F or lower. This standard specification for gas turbine fuels now covers fuels ASTM D396 and ASTM D975 grades 1 and 2.

Some general properties of the above fuels include:

Specific gravity	0.9 to 0.8
Lower heating value	18,200 to 18,300 Btu/lbm
Kinematic viscosity, min	1.5 to 2.1 cSt at 40°C
Kinematic viscosity, max.	3.6 to 8.5 cSt at 40°C
Carbon residue	.35 percent by weight or lower
Sulfur	1.0 percent by weight
Ash	.01 percent by weight

Fuel Contaminants

Fuel contaminants are not found in distillate fuels as refined and are therefore not mentioned in the specifications. However, contaminants such as water, dirt particles, trace metals, and microorganisms that are harmful to gas turbines find their way into fuel during handling and storage.

Water may be entrained, dissolved in the fuel, or free in the bunker. Freshwater is usually the result of condensation. Freshwater has an affinity for salt compounds, i.e., sodium, chlorine, and sulfates, which are present in the marine environment. Seawater may enter from many sources including tank vents, during tank cleaning or ballasting operations, and from supplier oil barges.

Dirt particles enter the fuel during handling and storage and from piping systems; rust and tank coating material are common.

Trace metals and metallic compounds are present in crude oils. In the refining process, these contaminants are concentrated in the residual oils. When a distillate is transported in a line or tanks previously used for blended IFO or gasoline fuels, metallic compounds of vanadium, sodium, potassium, calcium, and lead can be picked up.

Microorganisms, which are primarily bacteria and fungus, live and grow at the interface of water and fuel feeding on the fuel hydrocarbons. This usually occurs in storage tanks and results in microbial slimes.

Coping with the Contaminants in Gas Turbine Fuel

Dirt particles cause wear and salt causes rapid corrosion in the hot sections of an engine. The solution is an effective fuel filtering and coalescing system. Filtering removes particles and a centrifuge removes free water. A coalescent filter removes fuel dissolved water by coalescing water into particles large enough to be filtered from the fuel.

Vanadium and sodium plus potassium can interact independently and together with sulfur in the fuel and with sodium and sulfate compounds remaining in the intake air after filtration to produce vanadium eutectics and sodium sulfates that melt and cause severe corrosion of hot sections of a turbine.

Calcium can form deposits which will accrete on hot turbine components, distorting their shapes and clogging the cooling holes. Normal engine-washing methods do not remove the calcium deposits. These compounds can reduce normal intervals for maintenance procedures from years to months. ASTM D2280 1992 specifications call for limits for trace metals in fuel not to exceed 0.5 ppm for each category of fuel-bound vanadium, sodium, calcium, and lead.

There is no effective method to treat these trace metal compounds aboard ship. The control depends on the contractual relationship with the fuel supplier who has the capability of meeting the trace metal limits ofthe gas turbine manufacturer. This is accomplished by ensuring that the gas turbine fuel is never stored in tanks or transported in pipes that have been used for other fuels.

Contaminants not found in distillates at the refinery can be picked up during transport and from the ship's fuel tanks. Marine gas oil (MGO) and marine diesel oil (MDO) both leave the refinery as true distillates free of contamination by residuals. To remain free of contamination, the distillate must be transported, handled, and stored in separate facilities. Such facilities are available throughout the world for MGO but not for MDO. In a 1984 U.S. Navy study, samples ofMGO from sixteen major ports throughout the world were analyzed and were found to be free of residual contamination. This is not the case for MDO, which is often contaminated.

MGO procured to specification ISO 8217, ISO-F-DMA requirements and processed onboard in a filter-coalescer treatment system would be the logical choice for acceptable commercial marine gas turbine fuel.

Microbial contamination produces a number of adverse conditions in gas turbine operation including corrosion, generation of hydrogen sulfide, and formation of organic growth such as slime that plugs filters. Filters are not a solution to slime in the fuel. Elimination of water in the fuel tanks is a useful precaution. Biocides may be added to the fuel to control microorganisms. Biocides meeting MIL-S-53021 specifications are widely acceptable for gas turbine fuel. Biocides may be added during bunkering operations using a metered pump or may be added to the full tank in concentrations of less than 200 ppm.

Gas turbines can be provided to burn gaseous fuel such as the boil-off of an LNG carrier. In this case, the fuel system would handle dual fuel (gas and oil), an arrangement commonly found in steam plants of LNG carriers.

FUEL OIL SYSTEMS

In general, regardless ofthe type power plant, fuel oil service systems include day tanks, pumps, filters, heaters, controls, meters, sensors, and separators.

Figure 8-16 shows a modern slow-speed diesel fuel oil service system with an economical sludge recovery system for both the lubricating oil and the fuel oil.

Figure 8-17 illustrates a modern steam ship fuel oil fill, transfer, and purification system designed for fuels with a viscosity of 600 cSt at 50°C. The storage tank bottom drains are arranged to facilitate the draining of the tank bottoms directly to the sludge tank, and the stripping of the tank by use of the fuel oil transfer pump.

Figure 8-18 shows a fuel oil service system for a steam plant designed for 600 cSt at 50°C fuel. The system is arranged to provide a positive suction head to the fuel oil service pumps. The pump discharges to the fuel heaters, where the temperature is raised to a level that provides the proper viscosity for atomization at the burners. Leaving the heaters, the fuel passes through a second duplex strainer, a viscometer, and a fuel totalizing meter. This meter provides information to the burner management system. This piping system must be well insulated with trace heating downstream of the heaters.

OPERATIONAL PROBLEMS IN A DIESEL ENGINE FUEL OIL SYSTEM

Figure 8-19 is a summary of operational problems that may be encountered in a diesel engine fuel oil system. The figure shows the major system components and the potential problems associated with the components.

SUMMARY APPROACH TO SHIPBOARD FUEL PROBLEMS

The following steps represent a summary and practical approach to shipboard fuel problems.

- all fuel incidents investigated and documented
- total segregation offuel during handling and storage
- representative fuel sampling during bunkering
- onboard testing of fuel
- laboratory fuel analysis
- controlled fuel handling, separation, and preheating
- engine performance monitored and correlated to fuel properties
- maintenance correlated to fuel properties
- systematic analysis of fuel problems and suppliers
- effective maintenance of separators
- upgrading of fuel handling equipment

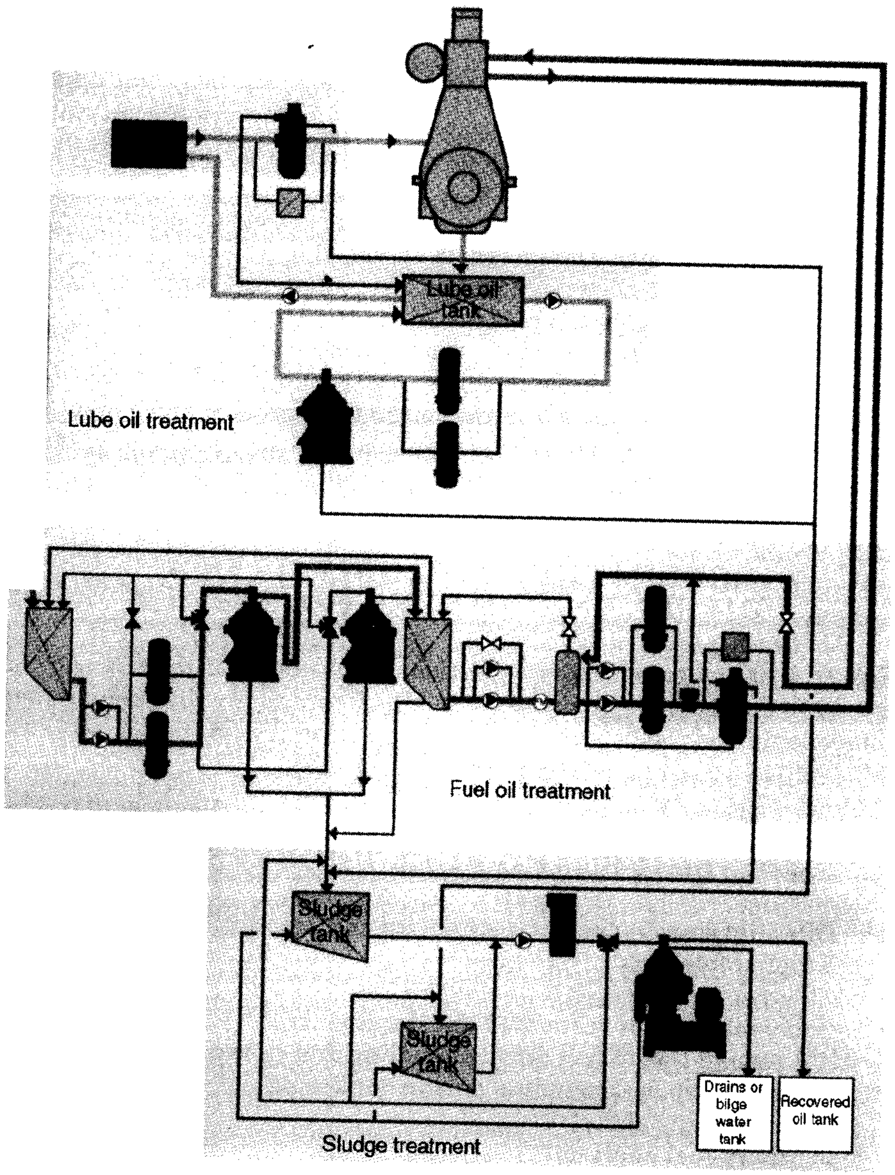


Figure 8-16. Comprehensive engine fuel oil system with economical sludge treatment and recovery system. Courtesy Alfa Laval Separation Inc.

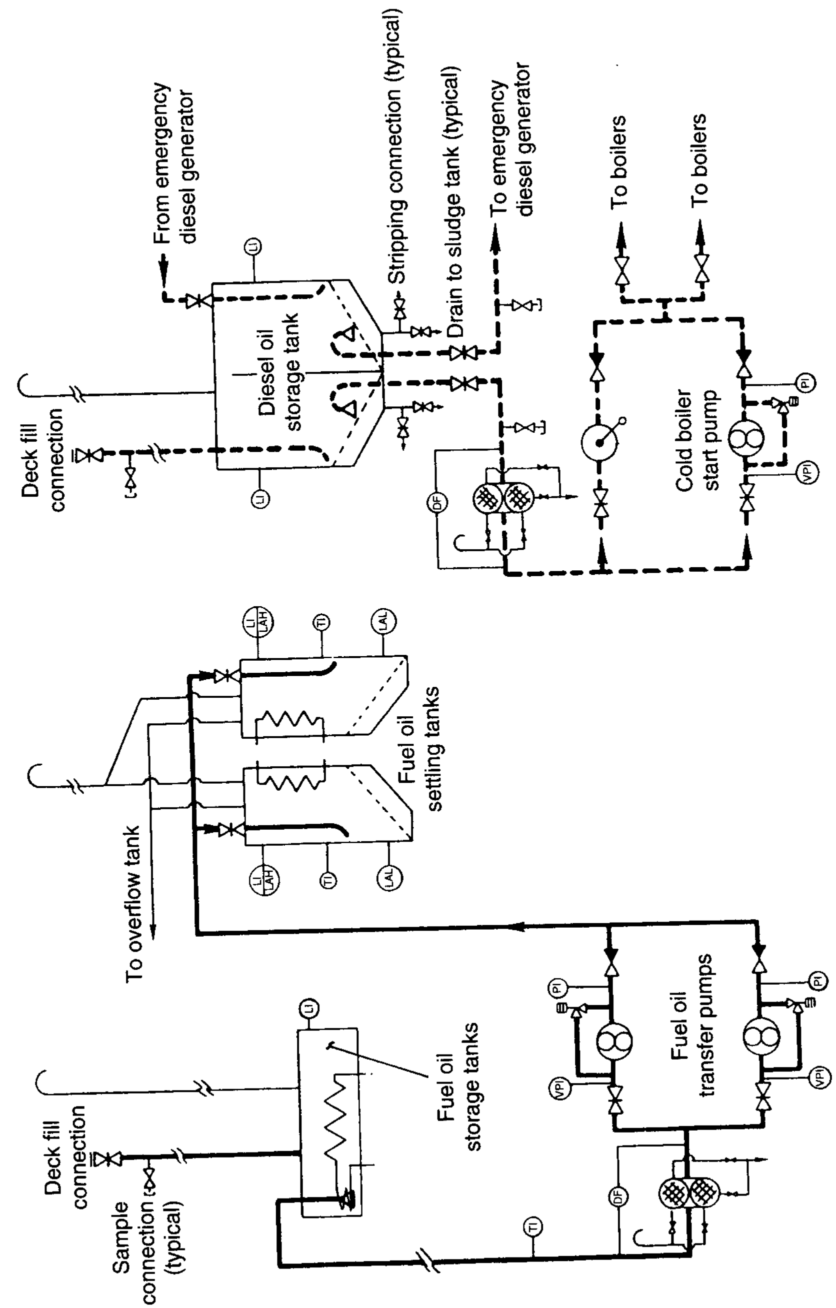


Figure 8-17. Steam propulsion plant fuel oil and diesel oil fill, transfer, storage, and purification system for 600 cSt @50°C fuel. Courtesy Seaworthy Systems, Inc.

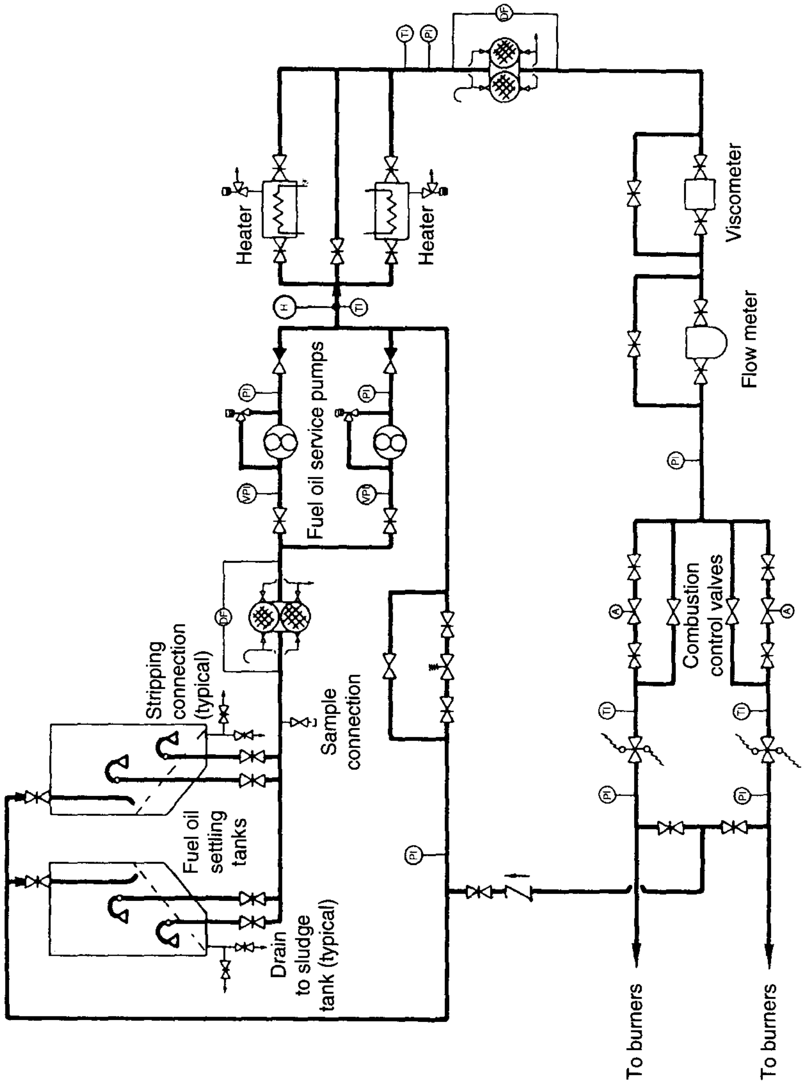


Figure 8-18. Steam propulsion plant fuel oil service system for 600 cSt @ 50°C fuel. Courtesy Seaworthy Systems, Inc.

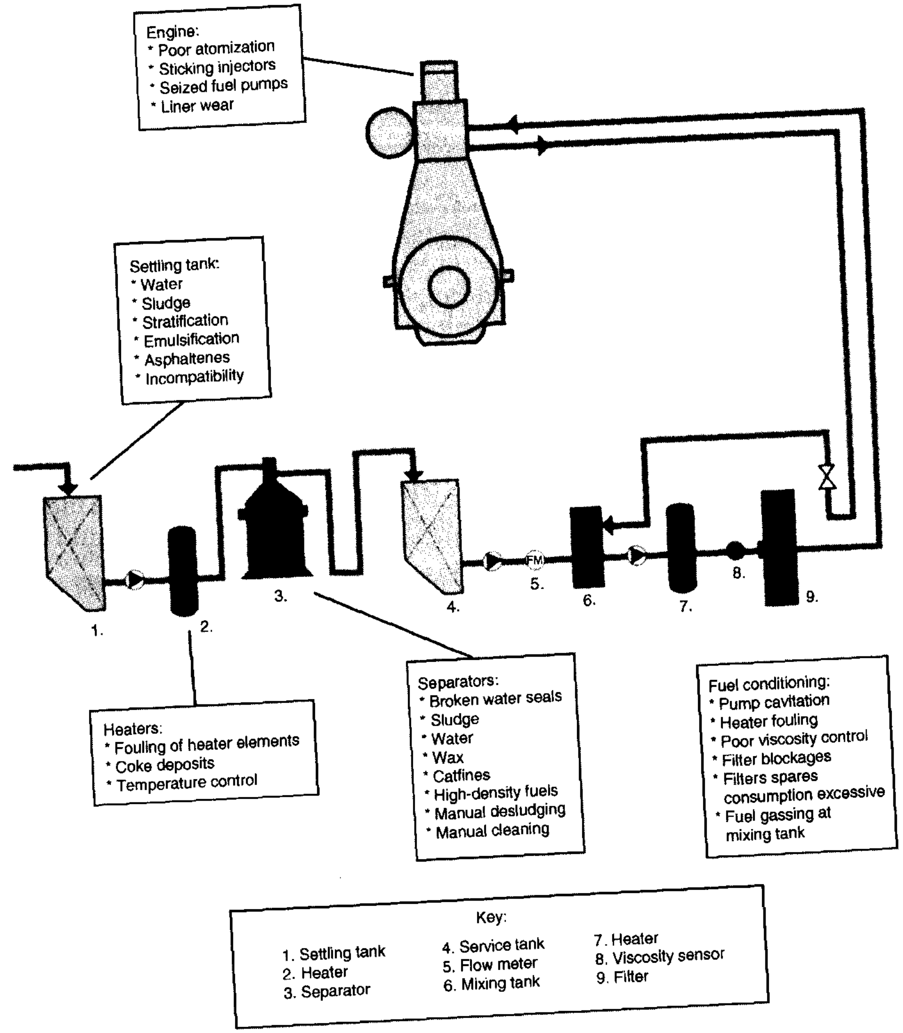


Figure 8-19. Potential fuel oil related problems in diesel engine system. Courtesy Alfa Laval Separation Inc.

REVIEW

1. What are the four general categories of crude oil compounds?
2. What do we mean by residual oil? Why do technical improvements in the refining process degrade the quality of residual oils?
3. What is the catalytic cracking process?
4. What are the general classifications of marine fuels?
5. Name the major international organizations that issue fuel specifications.
6. What is viscosity? How is it measured? What are the units?
7. Define pour point ..
8. What is CCR?
9. What is the CIMAC specified range for vanadium in fuel?
10. Suggest some steps a ship operator should take to protect against unethical fuel suppliers.
11. Describe two shipboard fuel sampling techniques.
12. What are the major environmental pollutants found in stack emissions?
13. What steps may be taken to avoid trace elements in gas turbine fuel?
14. What type of system is effective in removing dissolved water and dirt particles from gas turbine fuel?
15. What is the purpose of a biocide in fuel treatment?
16. What fuel properties are possibly associated with diesel engine surging?
17. What fuel properties are possibly associated with cylinder liner wear?
18. What fuel properties are associated with the deterioration of high-temperature sections of a boiler?
19. When is chemical treatment offuel useful?
20. What steps should be taken to have success with chemical treatment offuel?
21. What are the eleven activities to resolve shipboard fuel problems?

REFERENCES

Alfa Laval. February 1996. *ALCAP Cleaning System for Fuel Oil,* 9th ed. Tumba, Sweden: Alfa Laval.

Drewry Shipping Consultants. 1994. *Marine Bunker Fuels: Markets, Economics and Technical Developments.* London: Drewry Shipping Consultants.

Ewart, W. D. 1982. *Bunkers.* London: Fairplay Publications.

Harbach, James A., and Vito Agosta. December 1991. "Effects of Emulsified Fuel on Combustion in a Four-Stroke Diesel Engine." *Journal of Ship Research,* vol. 35, no. 4.

Heron, B., and J. S. Mills. 1990. "Development Trends in Marine Bunker Fuel Properties and Their Impact on Marine Diesel Engine Performance." *Transactions.* London: Institute of Marine Engineers.

Jones, Cletus H. 1985. *Marine Fuels.* Philadelphia: American Society of Testing Materials (ASTM) Publication 878.

Jones, Maurice V. January 1984. "Fuel System Design Features and Operating Practice." Fuel seminar. London: Institute of Marine Engineers.

Newbery, P. J. February 1996. "Ongoing Development of the International Marine Fuel Standards." *Transactions.* London: Institute of Marine Engineers.

Osugi, James T. October 1985. "An Analysis of Shipboard Fuel Quality Testing Results." Singapore Branch. Institute of Marine Engineers.

Schmidt, Paul F. 1969. *Fuel Oil Manual,* 3rd ed. New York: Industrial Press.

Winkler, Matthew F., and Thomas J. Pakula. December 1982. "Systems Design For Future Fuels, Steam and Diesel." New York: Institute of Marine Engineers.

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CHAPTER 9

Mechanical Transmission Systems

EVERETT C. HUNT

INTRODUCTION

Mechanical transmission systems have many applications on board a ship. These systems consist of devices to transmit torque at fixed or variable speed between prime movers and the energy absorbers. The devices include shafts, couplings, reduction gears, and clutches, for which rotating elements are supported by antifriction or hydrodynamic bearings. For example, pumps are connected to driving motors by a shaft including a flexible coupling; a propulsion gas turbine drives a propeller through a complex system of couplings, double reduction gears, and shafting; and a medium-speed diesel engine is connected to a propeller by a single reduction gear, shafting, and numerous solid and flexible couplings. Most of these systems also include one or more thrust bearings to position the rotating elements axially and to absorb axial thrust. This chapter will deal primarily with the main propulsion transmission system.

System Components

1. Propulsion shafting transmits the torque of the propulsion machinery to the propeller. A section ofthe shafting contains a thrust bearing, which transmits the propeller thrust to the hull.

The section of shafting that penetrates the hull is fitted with sealing devices. The final shaft section or tailshaft supports the propeller, which is fitted to the tapered end and secured by a propeller nut. The line shaft is supported by oil lubricated spring bearings designed to accommodate misalignment due to the flexing of the hull in a seaway.

2. Gears are toothed cylinders employed to transmit torque from one shaft to another. An assembly of two or more gears is gearing. Main propulsion reduction gearing is required for prime movers that operate most

efficiently at speeds higher than the efficient operating speeds of propellers. In the case of steam and gas turbines, the speed reduction ratio is 60:1 or more, requiring double reduction gears. For medium-speed diesel engines, single reduction gears with a ratio in the range of10:l are used. The higher ratio gears are designed with double helical teeth while the lower ratio gears are usually single helical teeth. Power takeoffs to drive auxiliary machinery such as electric power generators and feed pumps are sometimes fitted to the reduction gears.

3. Prime movers, gears, and shafting are connected by couplings. In applications where the components are required to operate with temporary or permanent misalignment due to temperature differences and foundation flexing, flexible couplings are employed to accommodate the misalignment. Solid flange bolted couplings are used to connect line shafting that deflects to accommodate design misalignment.

4. Clutches are required in some systems to disconnect multiple prime movers such as medium-speed diesel engines. Power takeoffs are also connected to main reduction gears by clutches.

Air operated, hydraulic, or friction-type clutches are employed for these services. Mechanical-type overriding clutches actuated by a torque reversal are also used for some applications.

Applications

The location and arrangement of the propulsion system is determined by the ship designer to accommodate many requirements of the ship and the machinery system including economics, size, and stability. For example, general cargo vessels usually have the machinery located midships with a propulsion line shaft extending one-half the vessel length. Bulk carriers have machinery aft with relatively short line shafts to the stern tube seal. Passenger vessels usually have multiple machinery plants located fore and aft of each other.

Gas turbine plants may have one or two turbines driving a single propeller through a double reduction gear and line shafting. A two gas turbine arrangement would have clutches between theturbines and gear to permit two or one engine operation. A typical engine room arrangement of a twin-screw gas turbine propulsion system is shown in figure 9-1.

Steam turbine propulsion units are normally cross-compounded which requires two input pinions for the double reduction gear that has a single output connected to the line shaft by a solid coupling. The turbines are connected to the input pinions by two flexible couplings separated by a hollow shaft. Low power steam turbine gear installations sometimes have the main thrust bearing located in the forward end of the gear casing so that the entire gear casing structure transmits the propeller thrust to the hull. Larger power installations always have the thrust bearing located in the line shaft aft of the gear. A typical modern steam turbine propulsion transmission system arrangement with direct corrected auxiliaries is shown in figure 9-2.

Medium-speed diesel engines are sometimes arranged with one to four engines driving input pinions of a single reduction gear. Multiple engine arrangements are connected to the input pinions by clutches to permit changes of power delivered to the propeller by changing the number of engines. In a multiple engine arrangement, clutches are also employed to reverse the propeller direction by continuously operating one of the engines in the reverse direction when maneuvering the vessel. Some medium-speed engines are not reversible, which requires a controllable reversible-pitch propeller for maneuvering the vessel. A typical modern medium-speed diesel propulsion transmission system is shown in figure 9-3.

The slow-speed diesel engine operates at propeller speeds that permit a simple mechanical transmission system consisting of an engine-to-shaft coupling and a line shaft with thrust bearing.

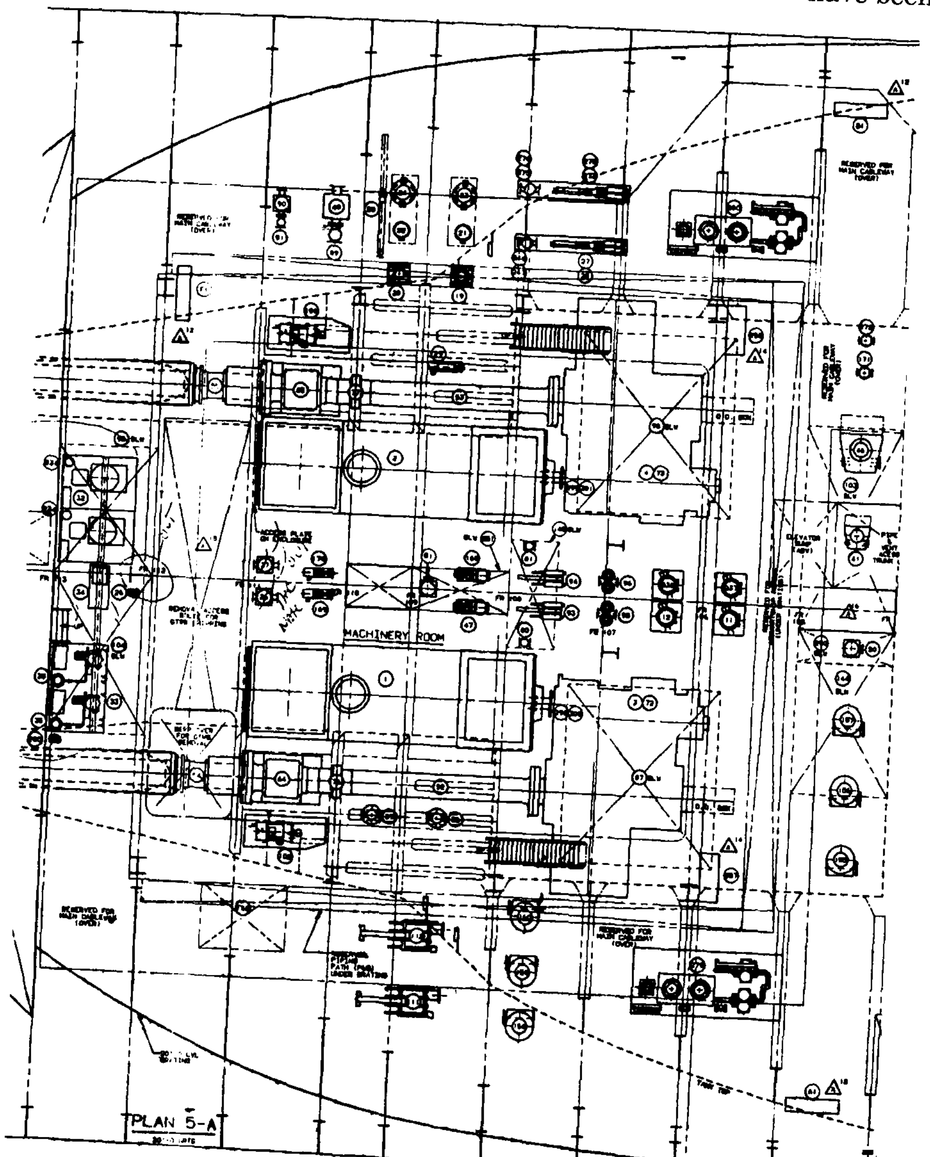
REDUCTION GEARS

Gearing may be classified according to shaft orientation, number of inputs, number of reductions, number of torque paths, system of measurement, pitch, and precision.

Gear Classifications

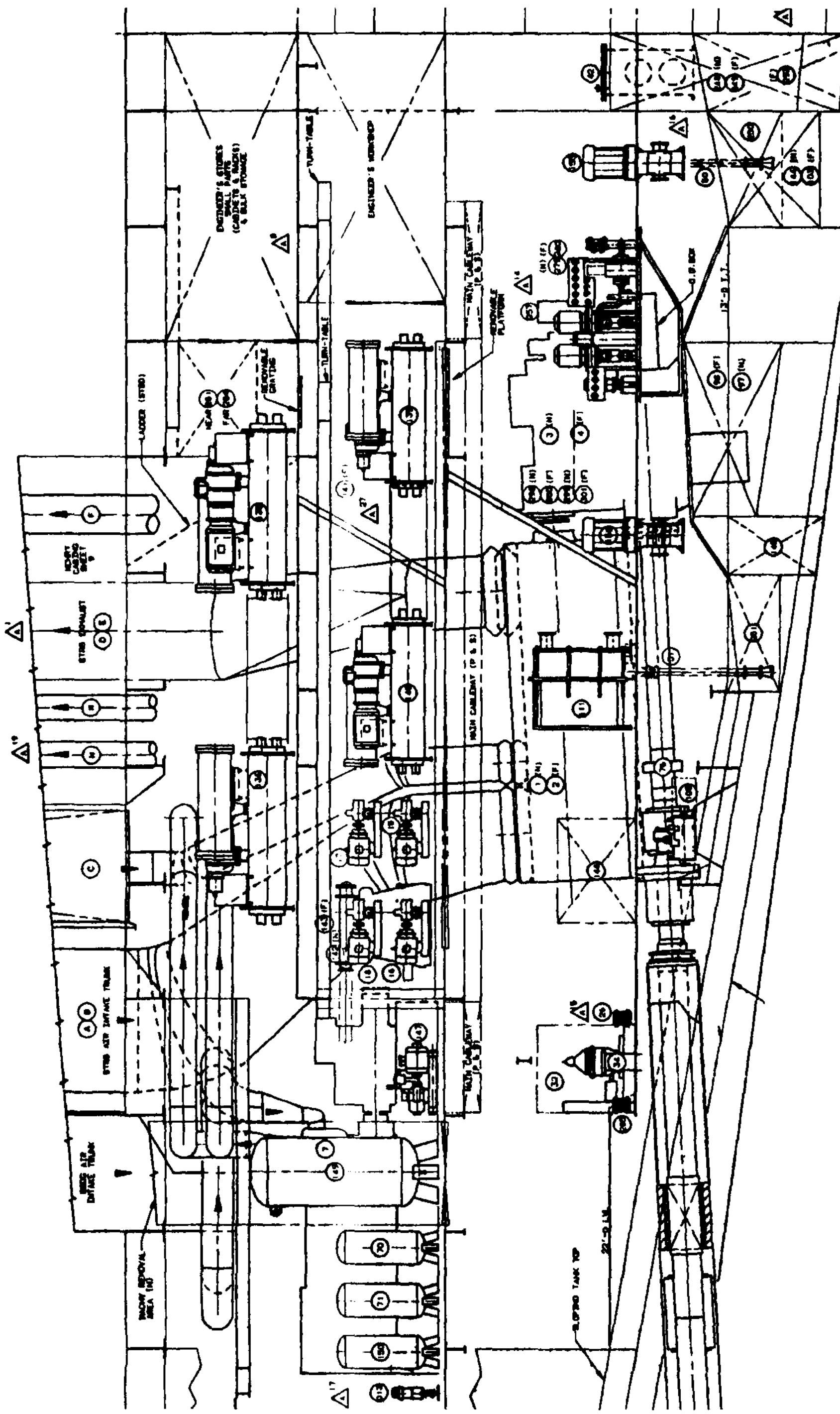
Shaft orientation classifications include parallel axes, bevel, worm, and epicyclic. For parallel axis gearing, the shaft ofthe pinion is parallel to the shaft of the driven gear. Typical propulsion gearing has parallel axes. Bevel gearing has axes which typically intersect at 90 degrees. Such gearing has been applied to some hydrofoil vessels propelled by a gas turbine through two right-angle drive transmission systems and to portable diesel drives for barges. Worm gearing has nonintersecting axes which cross at 90 degrees. A shipboard application of worm gearing is the turning gear designed to rotate the main propulsion system at very slow speed during plant start-up, shutdown, and maintenance procedures. Epicyclic gearing consists of center sun gear input, surrounded by three planetary gears and an outer ring gear with internal teeth. Epicyclic gears have the advantage of in-line input and output shafts. They have been employed as reduction gears for turbine generators, and first reduction gearing in propulsion systems. Epicyclic gearing has been frequently proposed as second reduction gearing for a propulsion system with counter-rotating propellers. In such a system, the ring gear would drive the outer line shaft, the planetary gear cage would drive the inner line shaft, and sun gear would be connected to the input shaft.

The number of inputs to propulsion gearing is determined by the number of prime movers to which the gearing is connected. Steam and gas turbine plants usually have two inputs. A few steam turbines have been



(a) Plan view

Figure 9-1. Gas turbine propulsion system transmission arrangement. Courtesy National Steel and Shipbuilding Company.



(b) Elevation

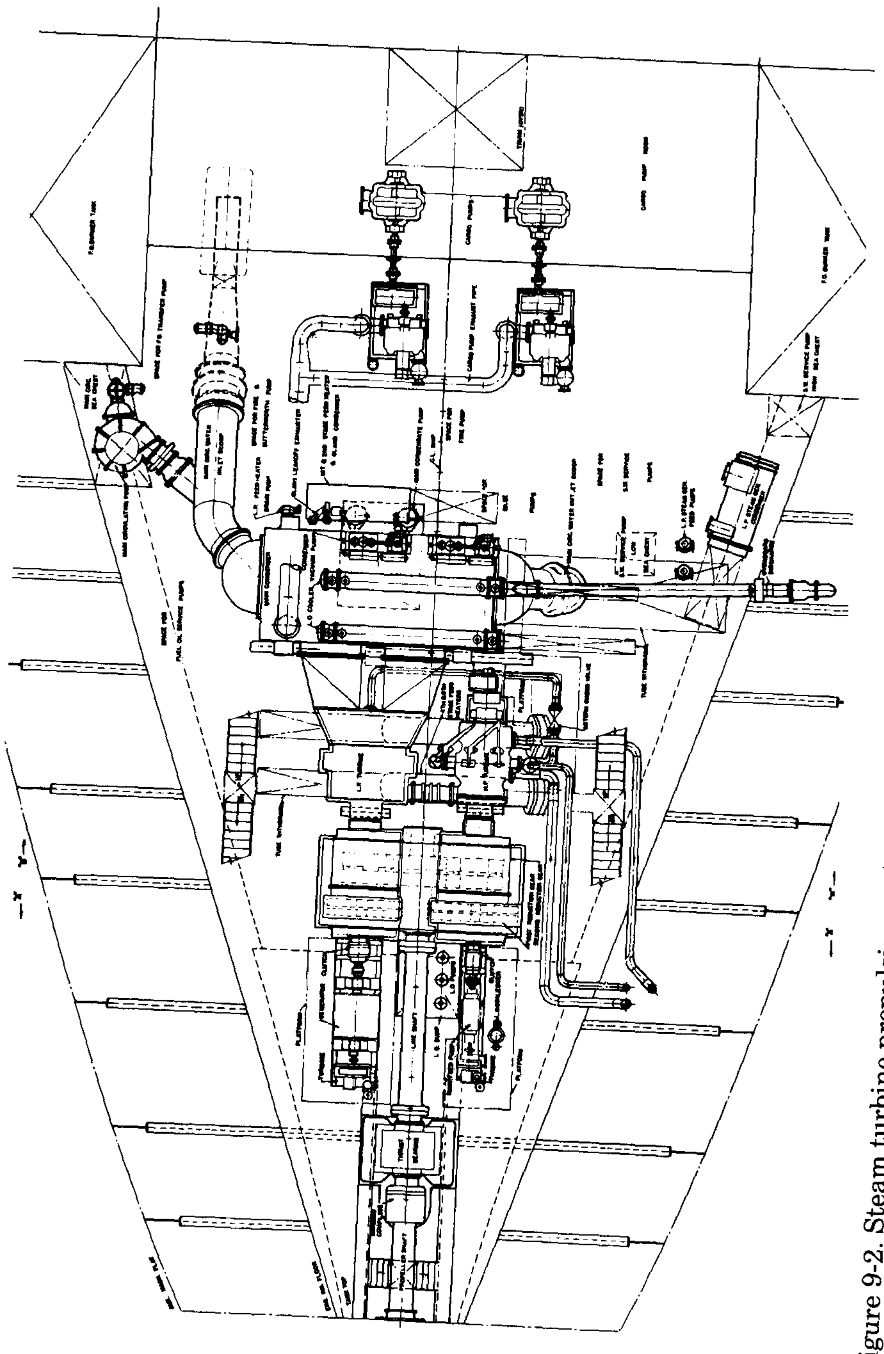


Figure 9-2. Steam turbine propulsion system transmission arrangement. Courtesy General Electric Company.

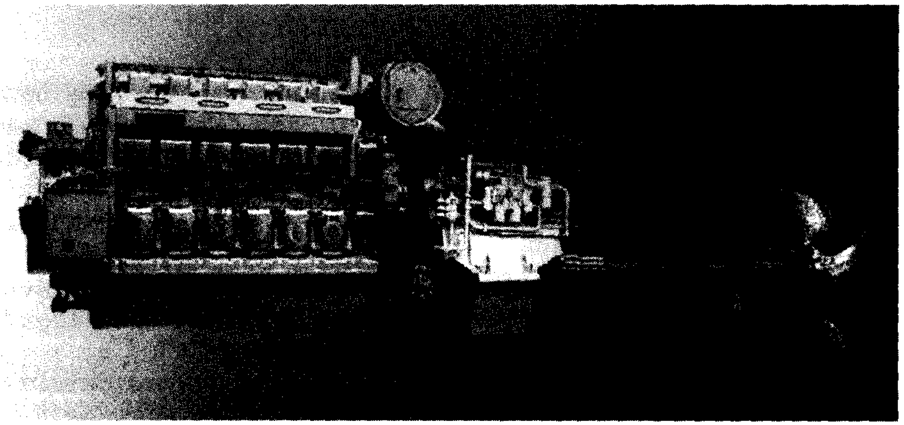


Figure 9-3. Medium-speed diesel propulsion system transmission arrangement. Courtesy Wartsila Diesel, Inc.

designed with three inputs to the gearing. Medium-speed diesel engines may have one to four inputs to a single reduction gear.

The number of speed reductions is determined by practical limitations on the maximum size of the gear and the minimum size of the pinion. To achieve the high-speed ratios required for turbine plants driving a propeller, two reductions are normally required. Triple reductions have been infrequently employed for main propulsion. Turbine generator sets usually have single reduction gears.

As the torque transmitting capacity of double reduction propulsion gearing is increased to accommodate higher horsepower ratings and slower propeller speeds, a limit is reached on the ability of a pinion to transfer the torque. The torque that the teeth must transmit can be reduced by one-half if the pinion engages two gears, each of which then would drive a pinion of the second reduction. This arrangement is sometimes called locked train because the relationship between gears and pinions must be timed to ensure an even split of the torque between the two paths.

The system of measurement refers to the use of metric units or conventional U.S. customary units. The United States is the only country in the world that does not use the metric system for gear design and measurement.

Pitch refers to the size of teeth or the number of teeth per gear diameter. Commercial ship propulsion gears have relatively large teeth.

Precision refers to the tolerances to which gearing is manufactured and installed. High precision permits gearing to operate at high torques and speed with minimal wear and noise.

Precision also contributes to the reliability of the gear since it ensures that the gearing is not subjected to loading that exceeds the design limits.

GEAR NOMENCLATURE

The following gear symbols, definitions, and nomenclature have been quoted or extracted from AGMA Standard 1012-F90, *Gear Nomenclature, Definitions of Terms with Symbols* and AGMA Standard 2000-A88, *Gear Classification and Inspection Handbook,* with permission of the publisher, American Gear Manufacturers Association, 1500 King Street, Alexandria, Virginia 22314.

Addendum, *a,* is the height by which a tooth projects beyond the standard pitch circle or the radial distance between the pitch circle and the addendum circle (fig. 9-4). '

Dedendum, *b,* is the depth of a tooth space below the standard pitch circle or the radial distance between the pitch circle and the root circle (fig. 9-4).

Backlash, *B,* is the amount by which the width of a tooth space exceeds the thickness of the engaging tooth on the operating pitch circle (fig. 9-5).

Center distance, C, is the shortest distance between nonintersecting axes. It is measured along the mutual perpendicular to the axes called the line of centers.

Clearance, c, is the distance between the root circle of a gear and the addendum circle of its mate (fig. 9-4).

Pitch diameter, *D* for gear, and *d* for pinion, is the diameter of the pitch circle (fig. 9-4).

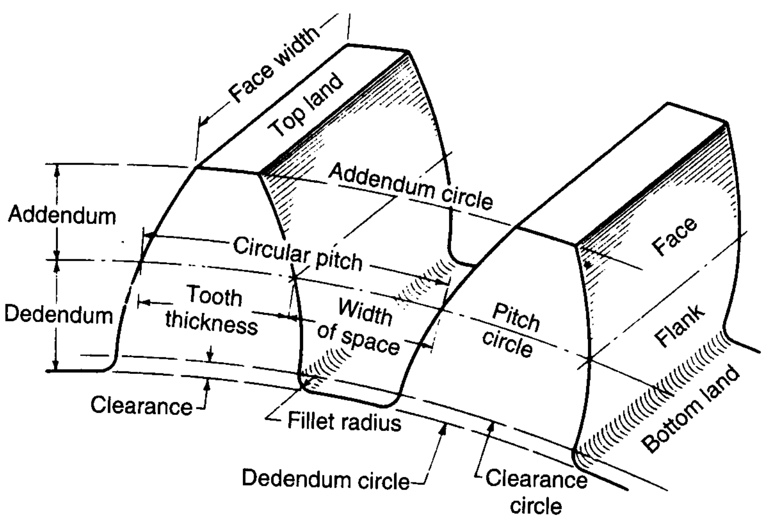


Figure 9-4. Principal gear tooth dimensions and nomenclature from *Mechanical Engineering Design,* 5th ed. (1977), by Joseph E. Shigley and Charles R. Mischke. Published by and reproduced with permission of The McGraw-Hill Companies.

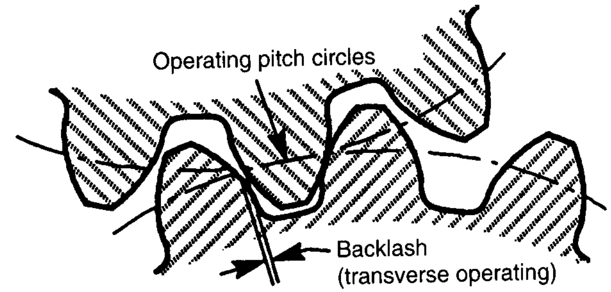


Figure 9-5. Backlash

Base diameter, D_{bg} for gear, D_{bp} for pinion, is the diameter of the base circle of an involute gear.

Outside diameter, D_{og} for gear, D_{op} for pinion, is the diameter of the addendum circle of an external gear (fig. 9-4).

Root diameter, D_{rg} for gear, D_{rp} for pinion, is the diameter of the root circle.

Face width, *F,* is the length of teeth in the axial plane (fig. 9-4).

Face width effective, F_e, is the portion that may actually come into contact with the mating teeth.

Face width total, F_t, is the actual dimension of a gear blank including the portion that exceeds the effective face width.

Whole depth of tooth space, h_t, is the total depth of a tooth space, the sum of the addendum and dedendum (fig. 9-4).

Working depth, h_k, is the depth of engagement of two gears, the sum of their operating addendums (fig. 9-4).

Lead, L, is the axial advance of the helix for one complete turn. $L = p_x N$ (use subscript "p" and "g" with L to designate pinion or gear).

Transverse module, m, is the ratio of the pitch diameter in millimeters to the number of teeth.

$$m = D/N$$
$$m = 25.4/P_d$$

Normal module, m_n, is the value of the module in the normal plane of a helical gear.

$$m_n = m \cos \psi$$

Face contact ratio, m_F, is the contact ratio in the axial plane, or the ratio of the face width to the axial pitch.

Gear ratio, m_g, is the ratio of the larger to the smaller number of teeth in a pair of gears.

$$m_g = N_g/N_p$$

Contact ratio, m_c, is the number of angular pitches through which a tooth surface rotates from beginning to end of contact.

Total contact ratio, m_t, is the sum of the transverse contact ratio and the face contact ratio.

$$m_t = m_p + m_F$$

The number of gear teeth is N_g; the number of pinion teeth is N_p.

The speed, in rpm, is n_g, for the gear and n_p for the pinion.

Axial pitch, p_x, is the linear distance between a point on one tooth and the corresponding point on an adjacent tooth in the axial direction (fig. 9-6).

Base pitch, P_b, is the distance on the base circle, i.e., an arc, between a point on one tooth and the corresponding point on an adjacent tooth.

Circular pitch, p, is the distance on the pitch circle, an arc, between a point on one tooth and the corresponding point on an adjacent tooth (fig.9-4).

Transverse diametral pitch, P_d, is the ratio ofthe number ofteeth to the standard pitch diameter in inches.

$$P_d = N/D$$

Normal diametral pitch, P_{nd}, is the value of diametral pitch in the normal plane of a helical gear.

$$P_{nd} = \frac{P_d}{\cos\psi}$$

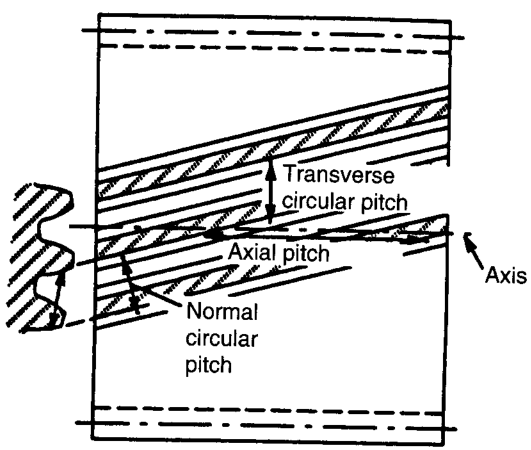


Figure 9-6. Tooth pitch

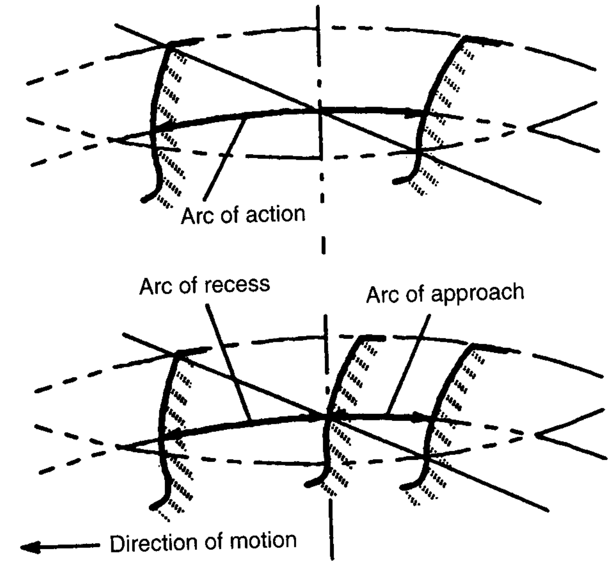


Figure 9-7. Arc of action

Arc of action, Q_t, is the arc ofthe pitch circle through which a tooth profile moves from the beginning to the end of contact with a mating tooth profile. It is subdivided into an arc of approach, Q_a, the distance from start of contact to the pitch point and an arc of recess, Q_r, the distance from the pitch point to the end on contact (fig. 9-7).

Top land is the surface at the top of a tooth (fig. 9-4).

Bottom land is the surface at the bottom of a tooth space adjoining the fillet (fig. 9-4).

Pitch plane of a pair of gears is the plane perpendicular to the axial plane and tangent to the pitch surfaces (fig. 9-8).

Transverse plane is perpendicular to the axial plane and to the pitch plane. In parallel axis helical gears, the plane of rotation is the pitch plane (fig. 9-8).

Normal plane is normal to a tooth surface at a pitch point and perpendicular to the pitch plane (fig. 9-8).

Involute teeth have a profile that is the involute of a circle in a transverse plane.

Root circle coincides with the bottom of the tooth space (fig. 9-4).

Line of action is the path of action for involute gears. It is a straight line passing through the pitch point and tangent to both base circles (fig. 9-9).

Line of contact is the line or curve along which two tooth surfaces are tangent to each other (fig. 9-10).

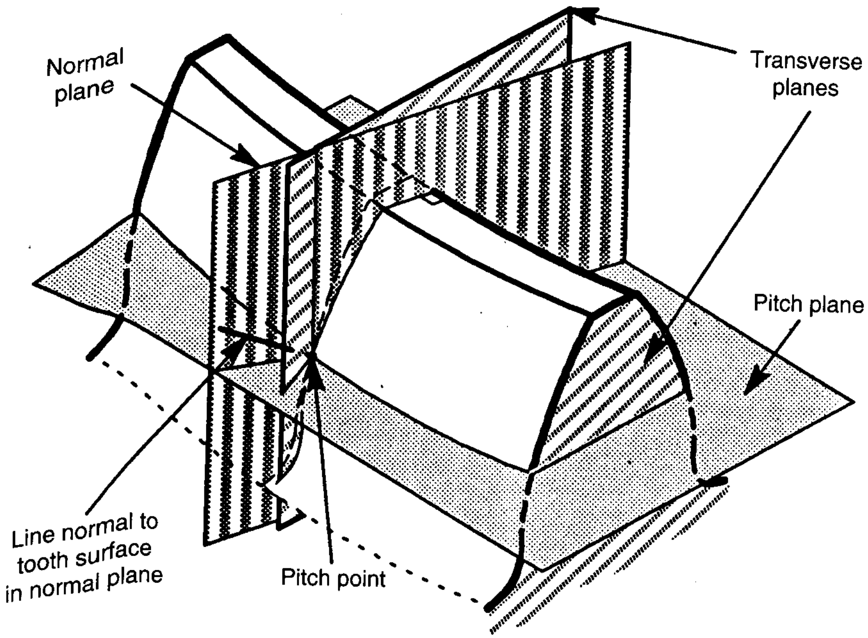


Figure 9-8. Reference planes for helical tooth

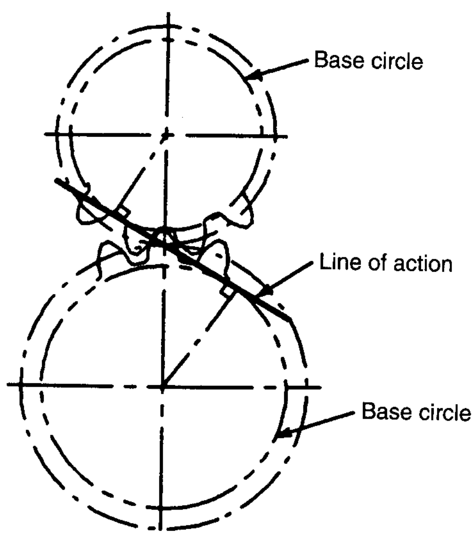


Figure 9-9. Line of action

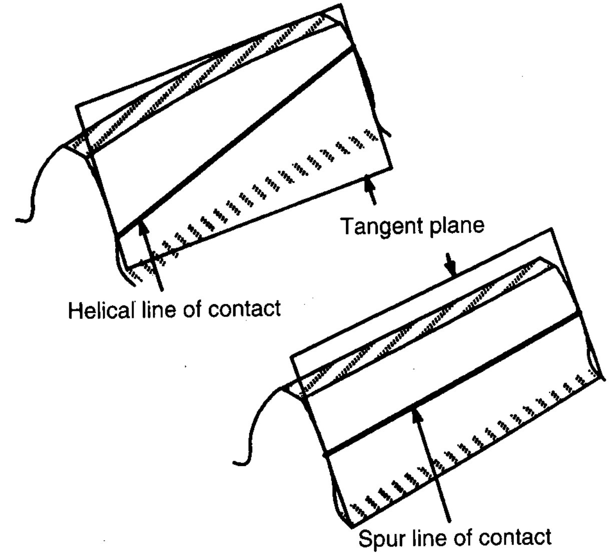


Figure 9-10. Line of contact

Pressure angle, $, is the angle at a pitch point between the line of pressure which is normal to the tooth surface and the plane tangent to the tooth surface. The pressure angle is equal to the profile angle at the standard pitch circle (fig. 9-11).

REDUCTION GEAR PRINCIPLES

Conjugate Action

The teeth of gears must be shaped so that contact between mating gear teeth occurs on the pitch circles and the angular velocity of the gears is constant. This is the motion that would occur if two smooth cylinders with parallel axes had one of the cylinders driving the other without slipping. Mating tooth profiles that yield constant angular displacement are called conjugate. There are many such profiles including the cycloid and the involute.

The involute is the highly standardized profile that is used for parallel axes spur and helical gears. It is the only profile this text will describe.

The basic law of gearing states that normals to the profiles of mating gears must, at all points of contact, pass through a fixed point located on the line of centers. In the spur and helical gears, the fixed point is the pitch point.

Involute Profile

An involute profile may be generated by a point at the beginning of a taut cord that is unwound from a cylinder with the diameter of the base circle.

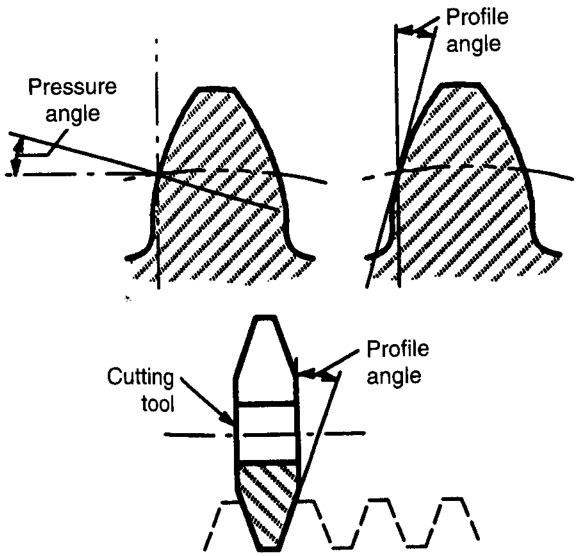


Figure 9-11. Pressure angle

The cord is a tangent to the base circle and is always normal to the involute. The length of the cord is the radius of curvature of the involute at every point. A given involute has a unique base circle. A base circle may have any number of involutes generated from it. Involutes to the same base circle are congruent. Involutes to different base circles are geometrically similar, which permits the teeth of a small gear to mesh with the teeth of a large gear.

As the geometrically similar profiles pass over each other during meshing operation, the pinion must travel through greater angular displacement than the gear. This means that the profile surfaces in contact on the pitch circle must slide relative to one another as well as roll.

Spur Gear

Spur gear teeth have an involute profile and a helix angle equal to zero. Since there is usually only one tooth in contact at any time, the operation of the gear is rough and noisy as the teeth engage and disengage. As a result, spur gears are not used for main propulsion but are commonly applied to deck machinery such as winches and windlasses.

To transmit torque, a spur gear exerts a force at the point of contact of the involute surfaces. This force may be resolved into two components: a force tangential to the pitch circle and a radial force that tends to separate the gears. These forces are opposite and equal to the bearing reaction forces ofthe pinion or gear. The forces are related to one another by functions of the pressure angle.

$$W_r = W_t \tan \phi$$

$$W_t = W \cos \phi$$

The horsepower transmitted is

$$\text{hp} = \frac{W_t V}{33{,}000}$$

where V is the pitch line velocity of a gear element in feet per minute.

EXAMPLE 9-1: A 10-horsepower machine has a spur gear train with the input pinion driven at 1,500 rpm. The pinion has 18 teeth, the idler or intermediate gear has 40 teeth, and the driven gear has 80 teeth. The diametrical pitch is 4 and the pressure angle is 20 degrees.

To determine the shaft speeds N, tooth loads W, and bearing reactions, proceed as follows for the arrangement shown in figure 9-12.

Shaft speeds:

Pinion speed = 1,500 rpm
Idler speed = 1,500 (18/40) = 675 rpm
Gear speed = 675 (40/80) = 337.5 rpm

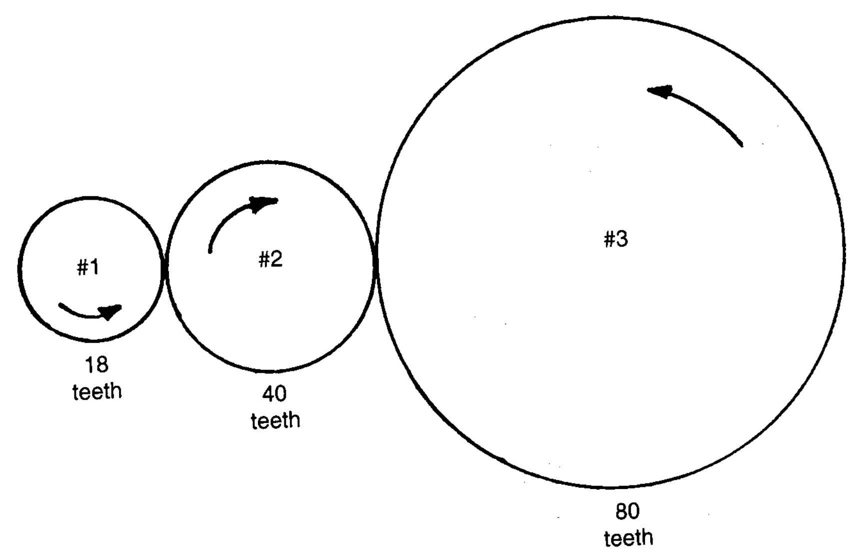


Figure 9-12. Double reduction gear arrangement

The pitch line velocity of the gear element is

$$V = rcDN$$

where N is the rpm.

The diameter on the driven gear or gear 3 is

$$D = N_{g/}\ P_{nd} = 80/4 = 20 \text{ inches} = 1.66 \text{ feet}$$

Therefore

$$V = \text{'rc}\ (1.66)(337.5) = 1{,}760 \text{ feet/minute}$$

and for the gear

$$W_t = 33{,}000/1{,}760 = 18.75 \text{ pounds force}$$
$$W = W_t/\cos 20° = 19.9 \text{ pounds force}$$
$$W_r = 18.75 \tan 20° = 6.8 \text{ pounds force}$$

If we neglect the weight of the gear elements, the forces acting on the teeth and the reaction forces acting on the bearings are shown in figure 9-13.

Helical Gears

Helical gears, like spur gears, have teeth with an involute profile, but the teeth are cut on the gear wheel or cylinder at an angle to the shaft. The angle is called the helix angle. Helical gears can be designed for nonparallel shaft applications, but such arrangements are uncommon on shipboard machinery. The common shipboard uses of helical gears are the single and double helix gears used for main propulsion. Typical helix angles are in the range of 20 to 40 degrees.

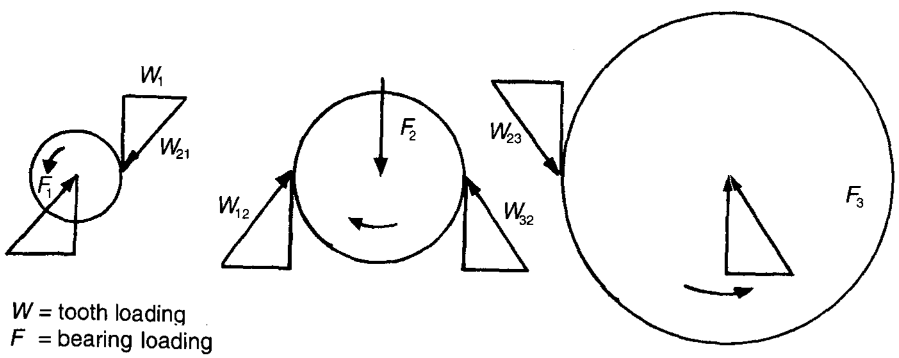


Figure 9-13. Gear and bearing forces

The advantage of a helical gear is that many teeth are in contact at the same time to provide a smooth and quiet transmission of power from a pinion to a gear. A typical propulsion gear may have 8 to 10 teeth engaged at the same time. If a high diametral pitch is selected to provide many small teeth, a helical gear will have a very low vibration level, providing it is manufactured with high accuracy. Such gears are used for submarines. Most commercial turbine propulsion units have double helical gears with low diametral pitches or large teeth. In this case acceptable noise levels are achieved by high manufacturing and installation accuracy.

Main propulsion single reduction helical gears are often single helix. Such gears are typically used with medium-speed diesel engines. Single helix gears develop significant axial thrust, which requires a thrust bearing to maintain axial position.

Double helical gears, sometime called herringbone gears, do not have an external axial thrust since the two helixes have equal and opposite thrusts that cancel one another.

Mating helical gear teeth have opposite hands for the matching helical angles, i.e., if the pinion of a single helical gear is left-hand, the gear must be right-hand.

The pressure angle of a helical gear may be measured in a plane normal to the teeth or in a plane normal to the axis ofthe gear. The forces on a tooth may be expressed as functions of either pressure angle and the helix angle. For example:

	Pressure angle in transverse plane	*Pressure angle in normal plane*
Force	Perpendicular to axes	Perpendicular to tooth
Tangential	$W_t = T/R$	$W_t = T/R$
Radial	$W_r = W_t \tan$ <1>	$W_r = W_t \tan$ <1>/cos\jf
Thrust	$W_a = W_t \tan$ \jf	$W_a = W_t \tan$ \jf

where

T = torque at pitch circle, pound feet
R = pitch radius, feet
<1> = pressure angle, degree
\jf = helix angle, degree

EXAMPLE 9-2: Given a single helical pinion and gear on parallel shafts with a helix angle of 24 degrees relative to the axes, a pressure angle of 20 degrees measured in the plane normal to the tooth, and a tangential force of 2,800 pounds, the radial and thrust forces may be determined as follows:

$$W_t = 2{,}800 \text{ pounds force}$$

$$W_r = \frac{W_t \tan\phi}{\cos\psi} = \frac{(2{,}800)(\tan 20°)}{\cos 24°}$$

$$W_r = 1{,}115.6 \text{ pounds force}$$

$$W_a = 2{,}800\,(\tan 20°) = 1{,}019.1 \text{ pounds force}$$

Typical helical reduction gear teeth proportions for a marine reduction gear are shown in table 9-1.

TABLE 9-1
Typical Helical Gear Tooth Proportions for a Marine Reduction Gear

Pressure Angle, ϕ	20 degrees
Addendum, a	$1.000/P_d$
Dedendum, b	$1.250/P_d$
Working depth, h	$2.000/P_d$
Whole depth, h_t	$2.250/P_d$
Circular tooth thickness, t	$\pi/2P_d$
Clearance, c	$.350/P_d$
Minimum number of pinion teeth	18
Minimum number of teeth per pair	36
Minimum width of top land	$.25/P_d$
Helix angle, ψ	35 degrees
Face width	$2px$

where

P_d = diametral pitch
p_x = axial pitch

GEAR TOOTH LOADING AND STRESSES

Gear designers take into consideration three types of stresses when specifying the size and shape of a gear tooth for strength purposes. These are (1) bending stress, (2) surface or Hertzian stress, and (3) fatigue stress due to repeated reverse bending of the tooth going into and out of mesh.

Tooth Bending Stress

Tooth bending stress is based on a cantilevered beam calculation first formulated by Lewis in the late nineteenth century. The involute tooth shown in figure 9-14 as a cantilevered beam has a maximum stress at the center of the root radius *(r)*. The distance x is

$$x = t^2/4l$$

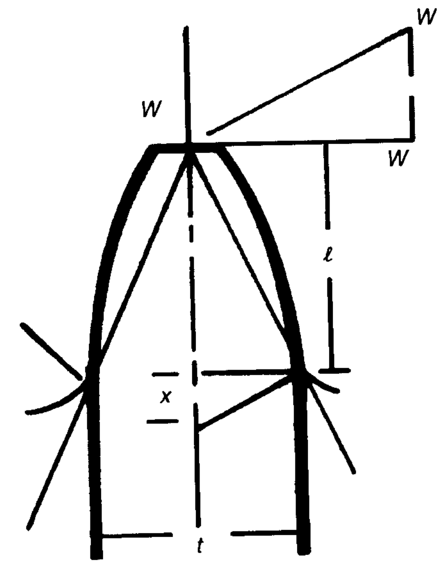


Figure 9-14. Gear tooth loading for bending stress calculation

Since the bending stress is

$$\sigma = \frac{M}{I/c} = \frac{6W_t l}{Ft^2}$$

where F is the tooth face width. If x is substituted into the above relationship and the result is multiplied by p/p the stress is

$$\sigma = \frac{W_t p}{F(2/3)xp}$$

where p is the circular pitch. If $y = 2x/3p$ the result is the Lewis Equation for bending stress

$$\sigma = \frac{W_t}{K_v F p y}$$

where y is the Lewis Form Factor which is tabulated in most machinery or gear handbooks. If diametral pitch is used, the Lewis Equation is

$$\sigma = \frac{W_t P_{nd}}{K_V F y}$$

EXAMPLE 9-3: The tangential load on a spur gear tooth is 3,000 lbs, the diametral pitch is 3 teeth per inch, the form factor is .310 and the face width is 5.0 inches, then

$$\sigma = (3{,}000)(3)/(5)(.310) = 5{,}806.45 \text{ psia}$$

To correct this stress for impact loading of tooth contact, it is necessary to divide the stress by the velocity factor, $K_v = 600 / 600 + V$ where V is the pitch line velocity in feet per minute. If the pitch line velocity is 1,200 feet per minute, then $K = 600 / 600 + 1{,}200 = .333$ and

$$\sigma = 5{,}806.45/.333 = 17{,}437 \text{ psia}$$

Surface Stress

When two curved surfaces, such as gear teeth, are pressed together, the line contact results in a three-dimensional stress. The stress is called Hertz contact stress. The Hertz stress for a gear tooth is

$$\sigma_n \approx C_p \sqrt{\frac{W_t}{C_v F d_p I}}$$

where

W_t = tangential tooth load, lbs.
C_p = elastic coefficient, for steel 2,300
F = gear face, inches
d_p = pitch circle diameter, inches
I = geometry factor = $\dfrac{\cos\phi \sin\phi}{2} \cdot \dfrac{m_g}{m_g - 1}$
ϕ = pressure angle
m_g = gear ratio
C_V = velocity factor = k_v

Gears that operate at excessive contact surface stress usually develop pitting, a surface fatigue failure caused by the repetitions of the contact stress. Excessive contact stresses can result from misalignment of gear elements.

Gear Load Indexes

In specifying or describing the load capability of marine reduction gears, it is common practice to use two indexes called "K factor" and "unit load." K factor is an index of the Hertzian stress on the gear tooth contact surfaces. Unit load is an index of the bending stress at the base of the tooth. The indexes are based on the respective stress formulations cited above.

A commercial marine double reduction gear with a normal pressure angle of 14.5 degrees and a standard root fillet would have typical K factors and unit loads for through-hardened materials as shown in table 9-2.

TABLE 9-2
Typical K Factors and Unit Load Indexes

Pinion material	Gear material	1st reduction K factor	2nd reduction K factor	Unit load psi
Alloy steel 200-240 BRN	Carbon steel 160-190 BRN	90	75	5,000
Alloy steel 300-350 BRN	Carbon steel 160-190 BRN	120	95	5,000
Alloy steel 300-350 BRN	Alloy steel 223-262 BRN	150	125	6,000
Alloy steel 350-400 BRN	Alloy steel 300-350 BRN	200	150	7,500

PROPULSION REDUCTION GEAR CONSTRUCTION

Single Reduction Single Input Gear

The simplest propulsion reduction gear is designed for application to a medium-speed diesel engine with one directional shaft rotation driving a controllable pitch propeller. A section of such a gear is shown in figure 9-15. This is a single reduction gear with a power takeoff to drive an alternator. The input shaft is clutched, permitting the alternator to be used as an emergency propulsion motor. The clutch is a multiple-plate hydraulically operated type. The main thrust bearing is built into the forward end of the gear casing. Typically, gears of this type are single helix. The manufactured finish of the teeth is usually provided by a precision grinding process. The material is alloyed gas-carburized steel. Since the gear rotates in one direction, it is necessary to precision-finish only one flank of the teeth. The gear casing is made of cast iron. The main bearings supporting the gear elements are thin shell hydrodynamic type. Other bearings in the gear casing are antifriction type.

The gear shown in figure 9-15 is manufactured in ratings from 2,682 hp to 16,092 hp (2,000 kW to 12,000 kW) for engine speeds of 750 rpm to 1,050 rpm. Depending on the propeller speed, typical reduction ratios are from 3 to 6. Gear weights are in the range of 10 to 26 tons. Heat dissipation by the lube oil cooler varies with ratings from 75 kW to 160 kW. Lube oil pumps are direct driven with an electric motor driven standby.

Single Reduction Multiple Input Gear

To achieve higher power ratings with medium-speed diesel engines, it is necessary to have multiple engine inputs to a single reduction gear. Such

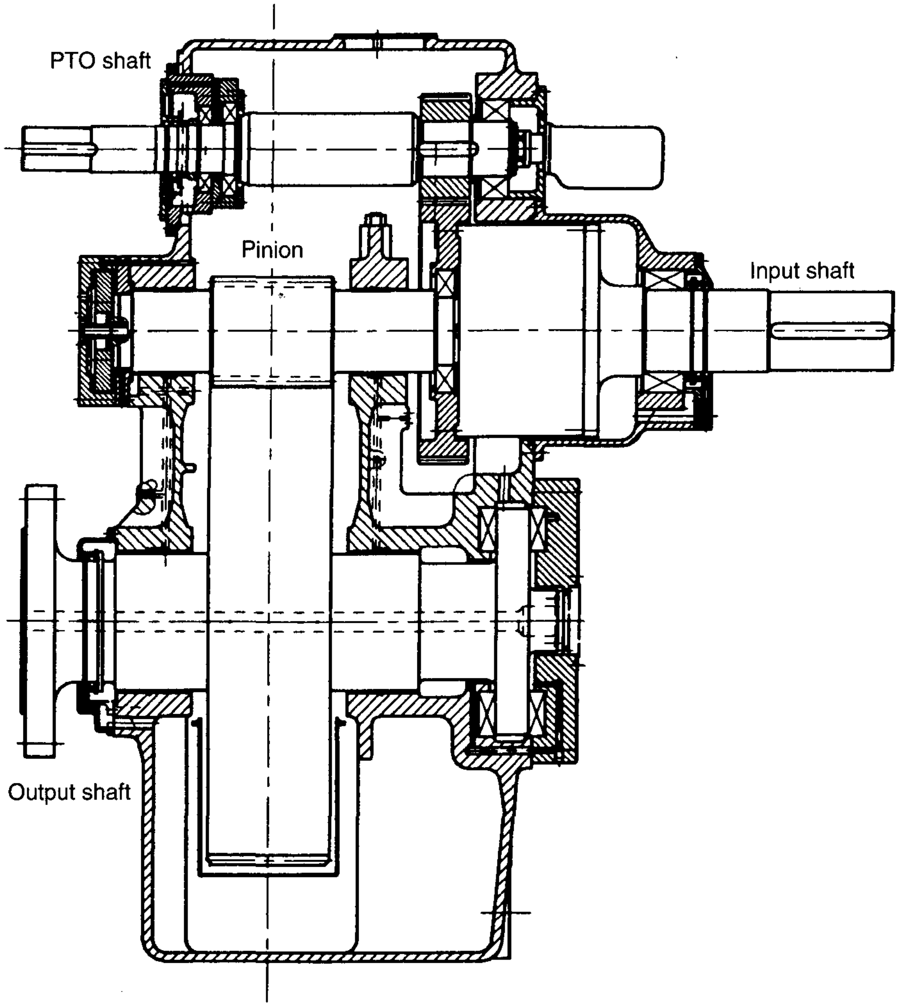


Figure 9-15. Single reduction gear. Courtesy Wartsila Diesel, Inc.

installations may have two to four input shafts and one or more power takeoff shafts. All engines are connected to the pinions by multidisk or plate clutches. If the engines are single rotation, a controllable reversible-pitch propeller must be fitted. Ifthe engines are reversible, maneuvering of a multiple engine propulsion system is accomplished with the clutches during the maneuvering mode. In 1994, the maximum available rating of a medium-speed engine was approximately 21,450 hp (16,000 kW, 21,760 metric hp). Therefore, multiple inputs permit the full range of possible power requirements for commercial vessels.

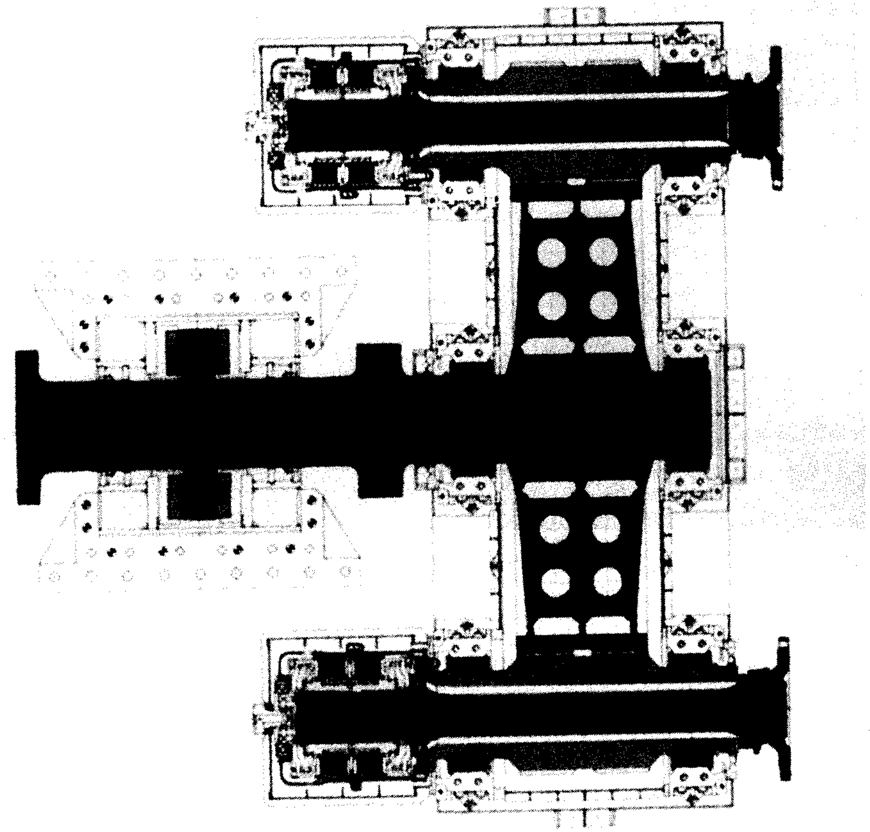


Figure 9-16. Dual input single reduction gear section. Courtesy RENK Tacke.

Figure 9-16 is a cross section of a single reduction, dual input, double helical teeth gear with a separate main thrust bearing and a multidisk clutch on each input shaft. A typical 19,713 shp (14,700 kW) vessel would have normal engine torque of 1.442 million pound force-inches (163 kNm) and a tangential tooth force of 76,844 pounds force (342 kN). The center distance between the pinion and gear is 83.661 inches (2,125 mm) for a reduction ratio of 3.44. Figure 9-17 is a photograph of a dual input single reduction gear with the casing removed to expose the gear elements and the multi disk clutches.

Double Reduction Multiple Input Gear

The exploded view of figure 9-18 illustrates a double reduction gear with dual inputs which has been regularly specified for main propulsion

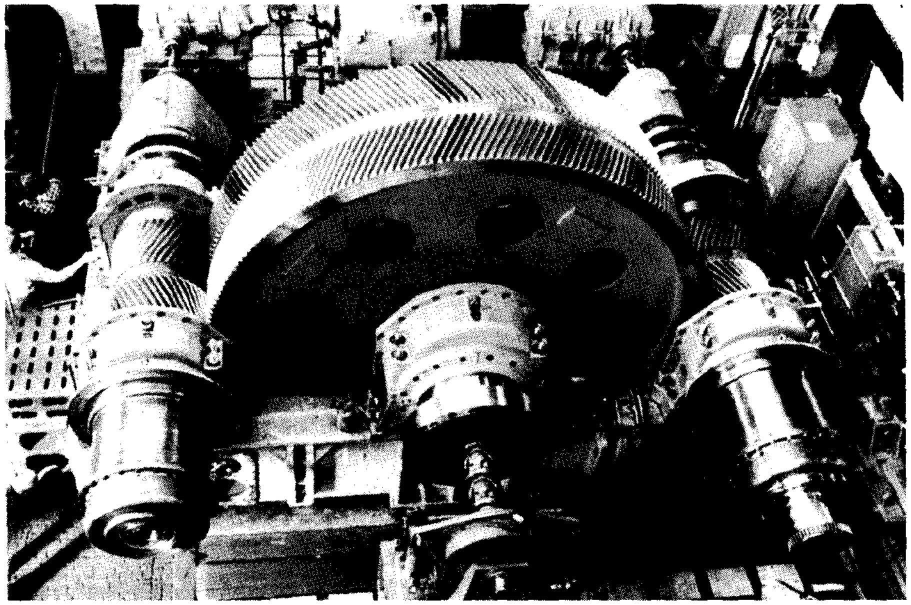


Figure 9-17. Dual input single reduction gear assembly. Courtesy RENK Tacke.

Legend for figure 9-18, facing page

1. Lower casing
2. Oil deflector halves, main gear shaft
3. Bearing halves, main gear, aft
4. Bearing halves, main gear, fwd.
5. Main gear
6. Main thrust bearing assembly
7. Bearing cap, main gear, fwd.
8. Main thrust housing, upper half
9. Main thrust bubbler
10. Bubbler
11. Bubbler assembly
12. Bearing cap, main gear, aft
13. Coupling sleeve, second-reduction pinion, HP
14. Coupling hub, second-reduction pinion, HP
15. Middle casing
16. Bearing halves, second-reduction pinion, HP, aft
17. Bearing halves, second-reduction pinion, HP, fwd.
18. Oil nozzles, second-reduction pinion meshes, HP
19. Bearing halves, first-reduction gear, HP, aft
20. First-reduction gear, HP
21. Bearing halves, first-reduction gear, HP, fwd.
22. Bearing cap, first-reduction gear, HP, fwd.
23. Middle casing, first-reduction, HP
24. Turning gear worm and worm wheel
25. Turning gear engaging spline
26. Turning gear engaging handle
27. Bearing halves, first-reduction pinion, HP, aft
28. Bearing halves, first-reduction pinion, HP, fwd.
29. First-reduction pinion, HP
30. First-reduction coupling, HP
31. Bearing cap, first-reduction pinion, HP, fwd.
32. Bearing cap, first-reduction pinion, HP, aft
33. First-reduction pinion cover, HP
34. Oil nozzles, first-reduction pinion meshes, HP
35. Turning gear motor
36. First reduction coupling, LP
37. Bearing cap, first-reduction gear, LP, fwd.
38. Bearing halves, first-reduction pinion, LP, aft
39. Bearing halves, first-reduction pinion, LP, fwd.
40. Bearing cap, first-reduction pinion, LP, fwd.
41. Bearing cap, first-reduction pinion, LP, aft
42. First-reduction pinion, LP
43. Bearing halves, first-reduction gear, LP, fwd.
44. First-reduction gear, LP
45. Bearing halves, first-reduction gear, LP, aft
46. Bearing cap, first-reduction gear, LP, aft
47. Bearing cap, second-reduction pinion, LP, fwd.
48. Bearing cap, second-reduction pinion, LP, aft
49. Bearing halves, second-reduction pinion, LP, fwd.
50. Bearing halves, second-reduction pinion, LP, aft
51. Second-reduction pinion, LP
52. Coupling sleeve, second-reduction pinion, LP
53. Coupling hub, second-reduction pinion, LP
54. Bearing cap, first-reduction gear, HP, aft
55. Bearing cap, second-reduction pinion, HP, fwd.
56. Bearing cap, second-reduction pinion, HP, aft
57. Second-reduction pinion, HP
58. First-reduction gear shaft, HP
59. First-reduction gear shaft, LP

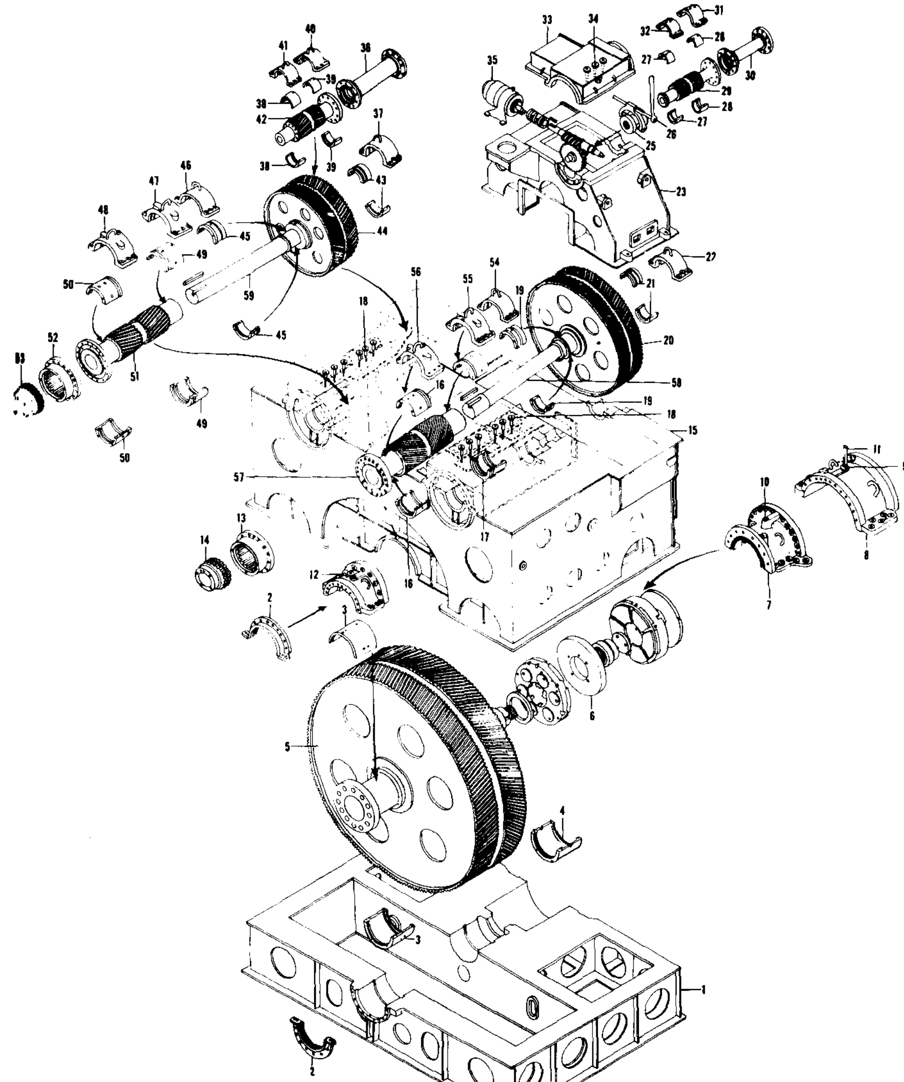


Figure 9-18. Exploded view dual input double reduction gear. Coutesy General Electric Company.

cross-compound steam turbine applications up to approximately 20,000 shp (14,914 kW). A box-like lower casing is continuously supported by the ship's gear foundation. The main thrust bearing is located in the forward end of the lower casing for lower power ratings and a separate thrust bearing is used for higher ratings. The main thrust bearing establishes the axial position of all rotating elements. This is facilitated by limitation of the axial movement of flexible couplings connecting the first reduction and second reduction trains. A solid coupling connects the second reduction output shaft to the line shaft, thus subjecting the second reduction gear to lateral forces of the line shaft. These forces can become an opera,ting and maintenance problem'depending on the length and flexibility of the line shaft. The middle casing is flange-connected to the lower casing. The middle casing supports the second reduction pinions and the first reduction gears. Proper operation and life ofthe gear requires that the design parallelism between pinions and gears be maintained with high precision. This will be discussed later in the section on alignment.

The turbines are connected to the first reduction pinions by flexible couplings that accommodate misalignment due to temperature changes in the turbines. When the turbines are at operating temperature, there is no misalignment. The transmission oflateral forces from the first reduction gear to the second reduction pinion is prevented by the use of a quill shaft and a flexible coupling between these elements. This feature is called an articulated gear. Upper casings flanged to the middle casing provide support for the first reduction pinion bearings and the turning gear assembly. Thin-shell hydrodynamic pressure-lubricated bearings are used for all journal bearings. Oil deflectors or seals are fitted to all shaft penetrations of the gear casing. Typical reduction ratios for this type of gear are 9.04 and 6.02 for the first reductions and 6.74 for the second reduction.

In power ranges above 25,000 shp (18,642 kW), it is common to use A-frame type construction and a dual torque path arrangement for propulsion reduction gears. Figure 9-19 contains section views of an A-frame reduction gear. In this arrangement, each side ofthe A-frame supports two first reduction gears that engage a single first reduction pinion.

The torque and power is divided at this point to permit approximately one-half reduction of gear face width. As a result, the second reduction gear has four pinions. This arrangement is sometimes called locked-train since the gear elements must be timed, i.e., adjusted angularly, to ensure that torque is shared equally in the two paths. Quill shafts and flexible couplings connect the first and second reduction trains. Figure 9-20 shows typical intermediate speed elements of a dual torque path reduction gear. The main thrust bearing is located in the line shaft aft of the gear casing. Figure 9-21 is a photograph of one side of an A-frame gear arrangement with the upper casing covers removed.

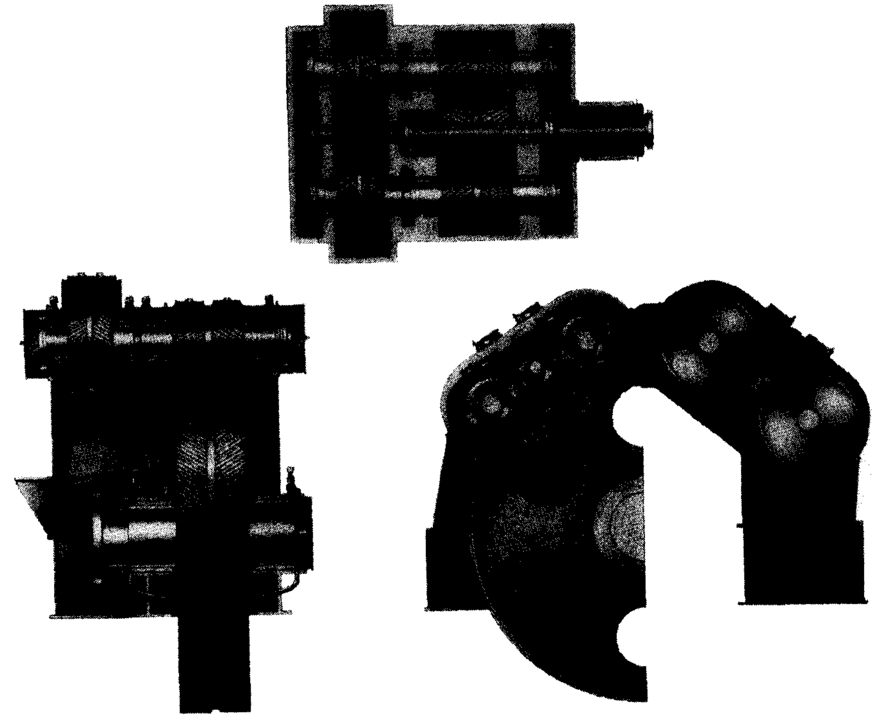


Figure 9-19. Arrangement of dual torque path double reduction gear. Courtesy General Electric Company.

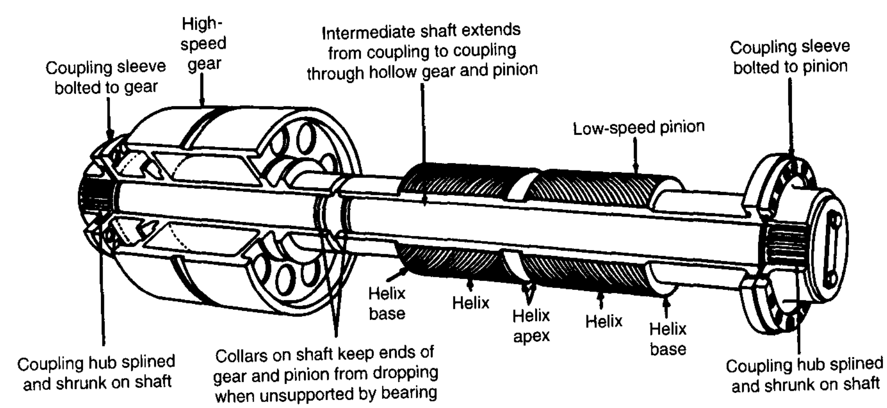


Figure 9-20. Intermediate span double reduction gear. Courtesy General Electric Company.

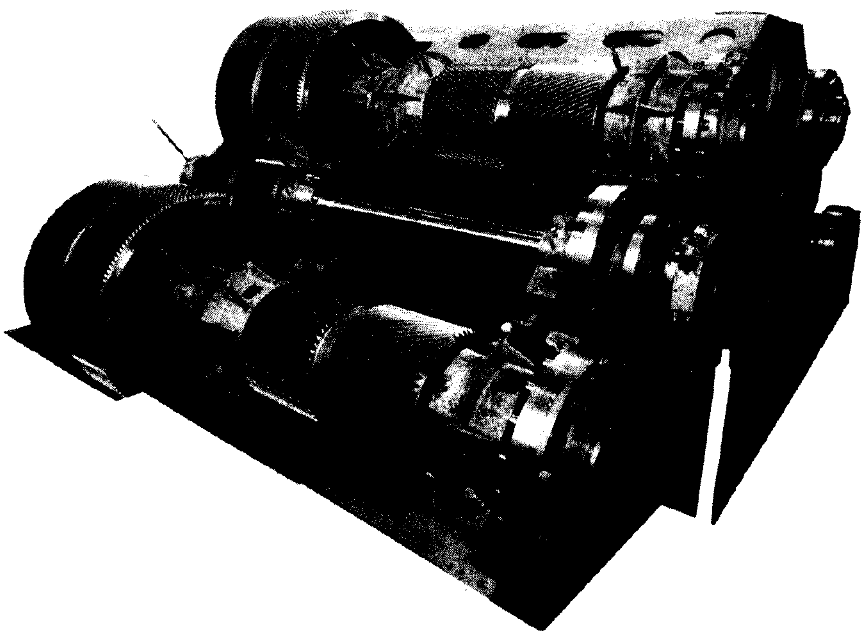


Figure 9-21. Dual torque path gear assembly.
Courtesy General Electric Company.

Single Input Dual Torque Path Gear

An interesting variation of the A-frame propulsion gear construction is the 32,000 shp (23,862 kW) single input gear designed for the twin-screw aeroderivative gas turbine Sealift ship. Since gas turbines are usually manufactured for rotation in a single direction, it is necessary to provide a rotation changing element in the starboard gear unit to permit opposite hand rotation of the controllable reversible-pitch propellers. The gear elements for the starboard unit are shown in figure 9-22.

The Sealift gear is really a triple reduction gear since the rotation changer pinion has one less tooth than the gear. In a true idler gear, i.e., equal number of teeth in pinion and gear, the same teeth engage in each revolution, providing an opportunity for undesirable wear patterns. The extra tooth in the gear, called a hunting tooth, requires all the teeth to engage in a finite number of rotations.

The first reduction pinion engages two first reduction gears dividing the torque equally. The two second reduction gears are connected to the second reduction pinions by quill shafts and dental-tooth type flexible couplings. Table 9-3 includes the data for this single input propulsion gear.

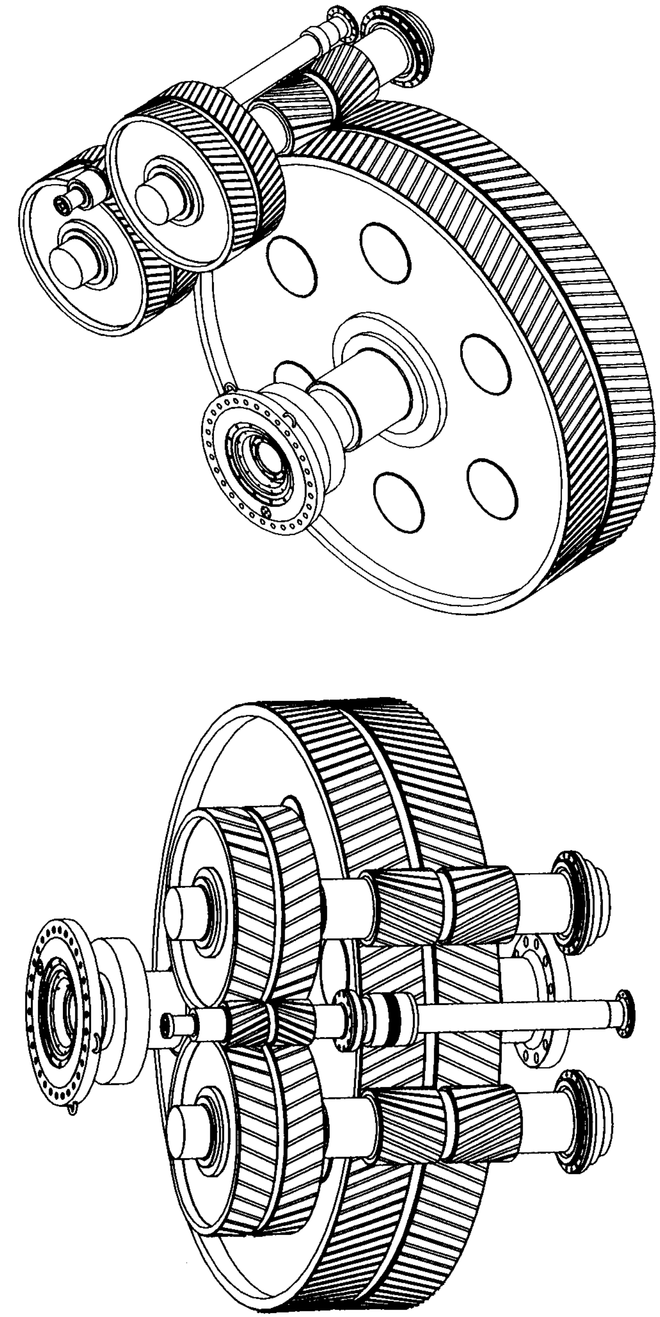


Figure 9-22. Single input dual torque path gear elements.
Courtesy General Electric Company.

TABLE 9-3
Sealift Propulsion Gear Data

	Changer		First Reduction		Second Reduction	
	Pinion	Gear	Pinion	Gear	Pinion	Gear
Center distance	18.0		37.5		102.0	
Horsepower	32,000	32,000	32,000	16,000	16,000	32,000
Speed rpm	3,665	3,608	3,608	775	775	95
Ratio	1.0159		4.6351		8.1633	
No. of teeth	63	64	49	228	49	400
Pitch diameter	17.858	,18.142	13.267	61.773	22.263	181.737
Effective face	18.75		24.5		41.0	
Tooth load lb/in	3,287		1,719		2,849	
K factor	365		157		144	
Unit load	13,311		7,347		7,237	
Circular pitch	.89052	.89054	.85060	.85116	.51867	1.42735
Diametral pitch	3.53		3.69		2.20	
Helix angle	30.		30.1857		30.9424	
Pressure angle, normal			17.5		20.	

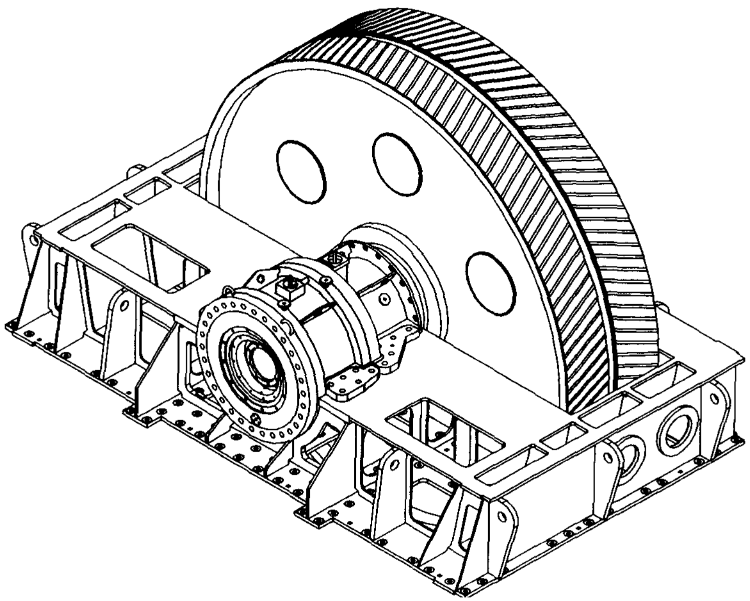


Figure 9-23. Second reduction gear assembled in the lower gear casing. Courtesy General Electric Company.

Figure 9-23 illustrates the second reduction gear assembled in the lower gear casing with the second reduction gear bearing caps in place. The lower gear casing is supported continuously on a stiff foundation which prevents bending and racking ofthe casing that would cause misalignment ofthe gear elements. Figure 9-24 shows the upper casing assembled to the lower casing. Removable covers for access to all the gear elements are assembled to the upper or A-frame casing.

The total weight ofthe port unit is 196,600 lbs and the starboard unit is 205,400 lbs. The second reduction gear weighs 89,000 lbs.

The efficiency of the port unit is 98.7 percent. The starboard unit has approximately 1 percent lower efficiency due to the direction changer gear element. Total oil flow including bearings and the gear meshes for the port unit is 245 gpm and the starboard unit is 389 gpm.

Manufacturing Tolerances (AGMA)

Typical manufacturing tolerances at AGMA Quality Level 12 for the 64-tooth changer gear are as follows:

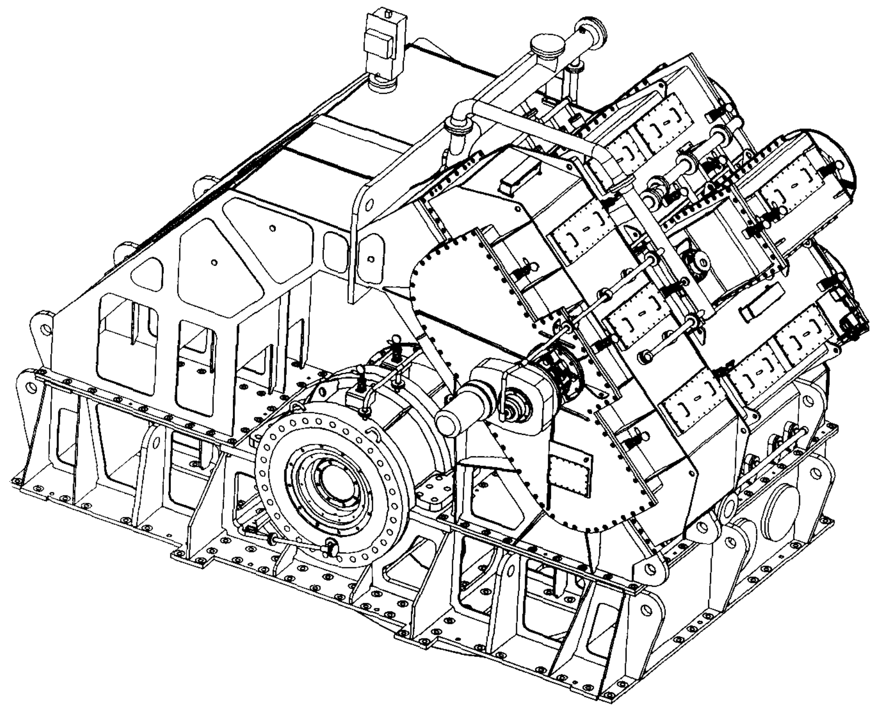


Figure 9-24. Single input dual torque path gear assembly. Courtesy General Electric Company.

Radial runout tolerance	.00176 inches
Allowable pitch variation + or -	.00035 inches
Profile variation	.00053 inches

Most manufacturers achieve better accuracy compared to the specified levels.

Turning Gear

The main engine turning gear permits the entire propulsion system to be rotated slowly for inspection, maintenance, and start-up and shutdown procedures. The turning gear is most commonly a part of the propulsion reduction gear. Figure 9-25 illustrates a turning gear system that drives the main reduction gear's first reduction high-pressure side pinion by an electric motor via a double reduction worm gear assembly and a clutch. The worm gear drive provides the high torque increase needed to break away the entire system and to push the gear elements into their bearing reaction positions for inspection checks. Turning gears are also driven by air motors and hydraulic motors. Safety devices are fitted to prevent the inadvertent operation of the main propulsion prime mover when the turning gear is engaged to the propulsion system. Such operation will result in serious damage to the machinery.

Reduction Gear Materials and Processes

Gear casings for main propulsion gears are made of fabricated mild steel.

Some small gear casings are cast from nodular iron or steel. The larger gear elements are usually fabricated with mild steel hubs, shafts, and webs with forged alloy steel tires. Smaller pinions and gears including their shafts are made of forged alloy steels. Table 9-4 lists typical through-hardened alloy materials used for the manufacture of propulsion gears.

TABLE 9·4
Typical Gear Material Properties

	Gear	*Pinion*	*Function of*
Specification	AISI4140	AISI4340	Alloy element
Carbon	.35-.45	.35-.45	hardness, strength
Silicon	.15-.35	.15-.35	strength, hardenability
Sulfur	.015 max	.015 max	low as possible
Phosphorus	.015 max	.015 max	low as possible
Manganese	.75-1.0	.60-.90	strength, hardenability
Nickel	-	1.65-2.0	strength, toughness, grain
Chromium	.80-1.0	.70-1.0	hardenability, strength
Molybdenum	.15-.25	.40-.60	strength, temper
Vanadium	.03 max	.05-.10	grain refinement

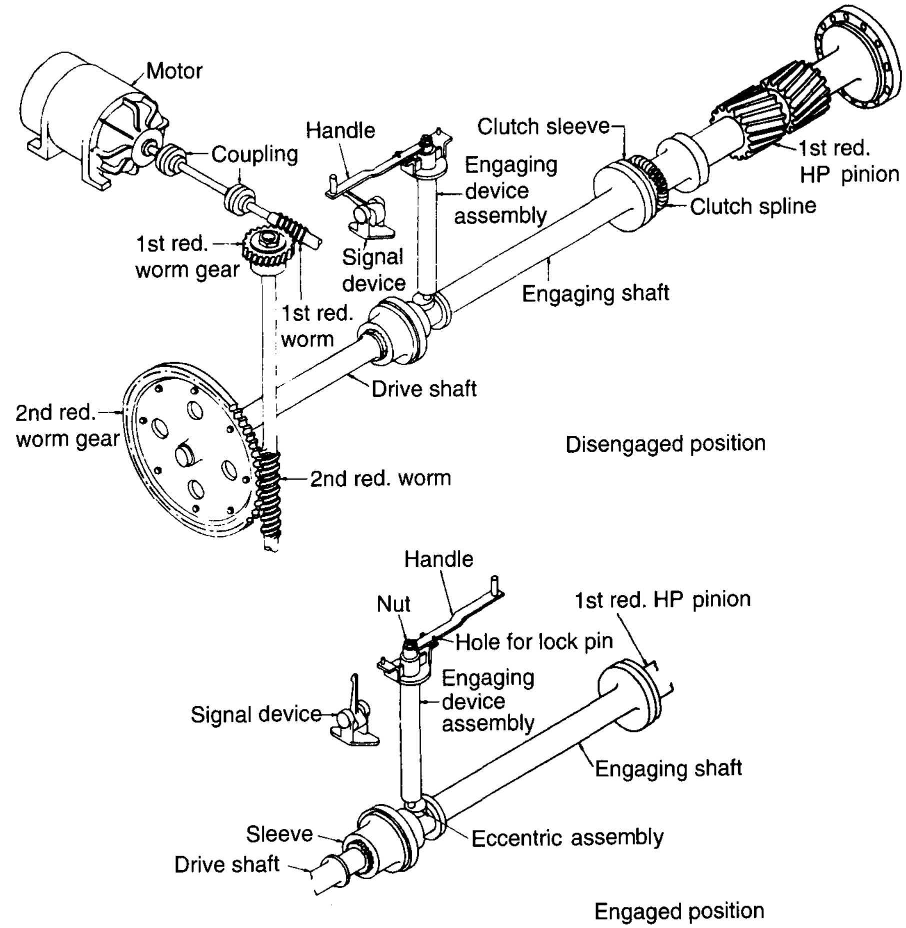


Figure 9-25. Turning gear arrangement.
Courtesy General Electric Company.

If the teeth are cut on through-hardened materials, the propulsion gears are manufactured by a hobbing process and finished by a shaving process.

Cutting of involute teeth with a hob having straight sides for a given pressure angle is the most common process of gear manufacture. Robbing is the generation of gear teeth in a continuous indexing process in which the cutting tool and the workpiece rotate in a constant relationship while the hob is fed into the work. When the hob is fed across the face of the gear blank, all the teeth are generated with the involute profile. Gear shaving is a free cutting process, i.e., the shaving cutter is engaged with and driven by the gear element being shaved. Very small amounts of metal are removed by the shaving process. The purpose of shaving is to correct errors in index, helical angle, tooth profiles, and eccentricity. Improvements in gear accuracy contribute to lower operating noise and fewer maintenance problems.

If gear teeth are hobbed and then heat-treated to a high hardness level, case hardening, case-carburizing, or nitrided grinding is normally used to provide the finish and accuracy required.

For the Sealift gear unit described above, the forged through-hardened rims or tires are made of ASTM 290 class G with a Brinell hardness of 321-375 and a minimum tensile strength of 125,000 psi. The pinions are through-hardened alloy steel forgings made of ASTM 291 class 7 material with a Brinell hardnes& of 352-401 and a minimum tensile strength of 160,000 psi. The rotation changer pinion and gear are made to ASTM 837 grade 9310. The material is carburized with a hardness of 58/62 Rc.

LUBRICATING OIL SYSTEM

A propulsion gear lubricating system provides lubrication to the journal bearings, the tooth-type flexible couplings, main thrust bearing, and the friction surfaces of the engaged gear teeth. In addition, the lubrication system removes heat from all gear, coupling, and bearing surfaces to maintain the gear at a uniform operating temperature. A uniform temperature is essential to the maintenance of correct alignment of the gear elements. The gear mesh lube oil supply nozzles are sized and located so that they will provide the proper cooling when the system is operating with the design lube oil feed pressure, temperature, and viscosity. Figure 9-26 is a typical oil flow diagram for a single torque path double reduction gear of a 25,000 shp cross-compounded steam turbine installation. The typical controlled inlet temperature is 115°F with a temperature rise of 50°F for an outlet temperature of 165°F. Typical oil pressure to the bearings and the gear mesh is 8 psig.

Lubricating Oil System Operation

The lubricating oil should be tested periodically to determine water content, viscosity, neutralization number, and the presence of foreign material, which would indicate unusual wear of bearings or gear surfaces.

Laboratory services are available in major ports for such testing of oils. Lube oil suppliers provide technical services and consultation of lube oil problems.

Lube oil systems are fitted with purifiers that must be operated regularly to remove moisture and impurities. The formation of sludge in the system indicates a breakdown of the oil. Regular analysis of the oil will avoid problems with sludge. Microbiological contamination of lubricating oil which occurs from time to time is detected by observation of slimy

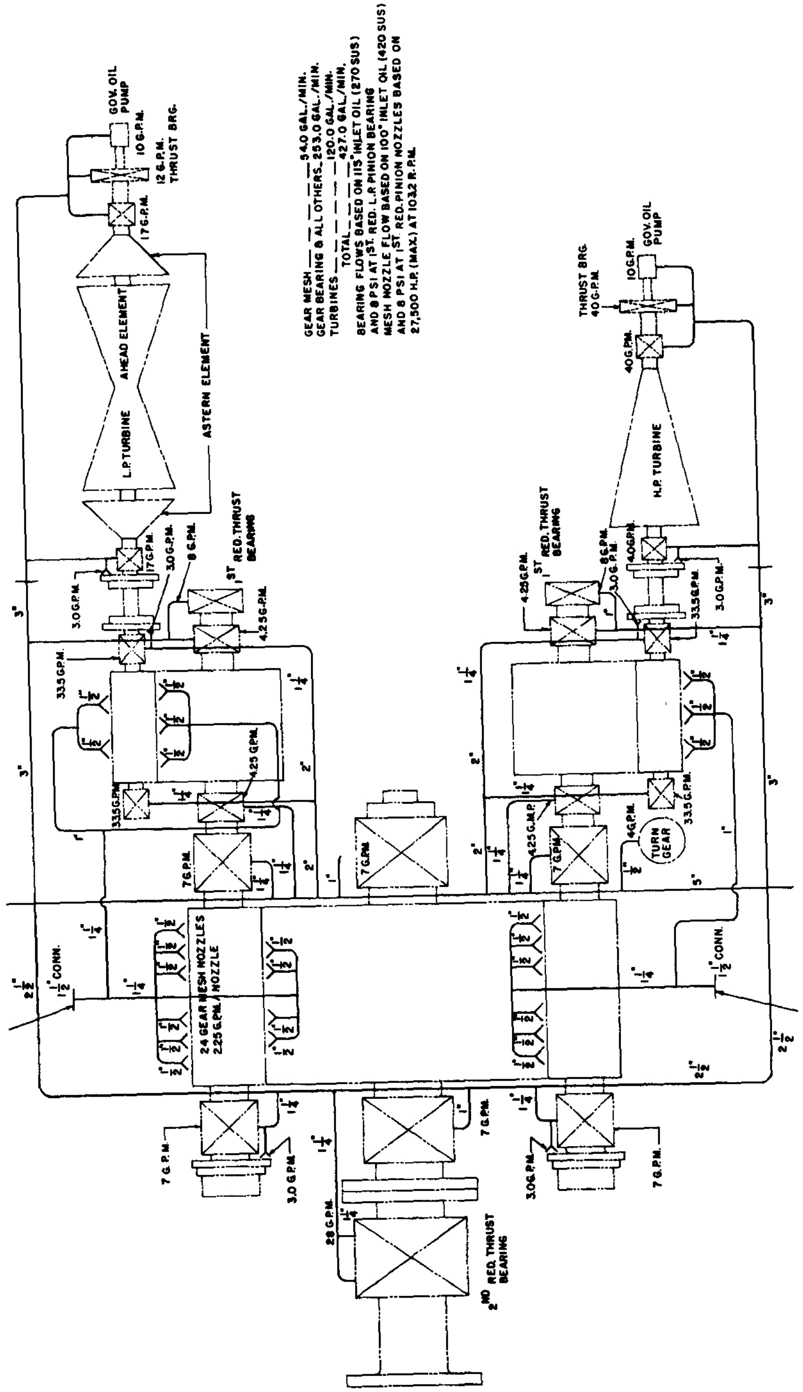


Figure 9-26. Lube oil system diagram. Courtesy General Electric Company.

deposits in oil tanks, deposits on lube oil cooler surfaces, unsuccessful operation of purifiers to remove moisture, and unpleasant ordures of lube oil. Expert help is usually required to eliminate such contamination.

During port visits, the lube oil system should be operated every other day to coat all surfaces with oil. During this operation the second reduction gear should be turned 114 turns with the jacking gear to ensure that all surfaces are coated with moisture-free oil.

To prevent the possibility of explosions, open flames and electrical equipment must be kept away from gear openings when the oil is hot and vaporous.

Carefully monitor the inlet and outlet oil temperatures and pressure to the lube oil system. Monitor individual bearing oil outlet temperatures.

TRANSMISSION SYSTEM MONITORING

There are numerous monitoring devices that can be applied to the main propulsion transmission system to provide early warning of potential problems and to improve the reliability of the system. Most commercial vessel monitoring is limited to display and recording of bearing shell temperatures at the reaction points by imbedded RTDs (resistance temperature detectors). A less effective alternative is to place RTDs in the lubricating oil flow out of each bearing. Other monitoring systems include the following devices.

Gear casing vibrations may be measured in three directions by a multi-direction velocity pickup. This will permit the display of the three vibration signatures for vertical, athwartship, and fore and aft directions using a computer based Fast Fourier Transform, which converts the amplitude versus time signal to an amplitude versus frequency signature. The value of recording such signatures is to compare them over time to detect changes that will indicate the onset of problems in the gearing. Volume 2 discusses changes in machinery vibration levels that point to developing problems.

When combined with a speed indicator, measurement of main shaft torque with a torque strain gauge will permit the display and recording of torque, propeller speed, and horsepower. A single instrument called a shaft horsepower meter is usually installed for this purpose. If a thrust strain gauge is also installed in the line shaft, the additional information can be combined with the torque and power recordings to provide an analysis of changes in the main propulsion system.

Main thrust bearing pads may also be fitted with RTDs to display, alarm, and/or record the thrust pad temperatures. Early warning of a developing problem in the main thrust bearing will contribute significantly to the overall reliability of the propulsion system. Monitoring and record-

ing the axial position of the main thrust shaft or the second reduction gear will also provide useful information concerning the condition of the thrust bearing.

All gear manufacturers install strain gauges on gear teeth to measure root bending stress during development tests and factory demonstration tests. In some cases, upon the owner's request, the manufacturers provide root bending stress monitoring with shipboard units. An example is the RENK-Checker, which permits the engineer officer to continously monitor the output gear wheel tooth root stresses and their distribution over the face width. As shown in figure 9-27, four strain gauges are arranged on the output wheel, i.e., bull gear, with which the root stresses can be measured at four points on the face width. Since this propulsion system has three medium-speed engine driven pinions, there are root stress indications at four points on the gear face width for each of the pinions, for a total of twelve stress indications for each revolution. For ideal operation, the twelve stress indications would be equal. If a bearing fails or there is a significant distortion in the hull due to a seaway or grounding, an increase in the gear root stresses will cause an alarm to sound. Periodic recordings and comparisons of the root stresses over time will provide additional useful information on the condition of the gear foundation, bearings, and tooth profile.

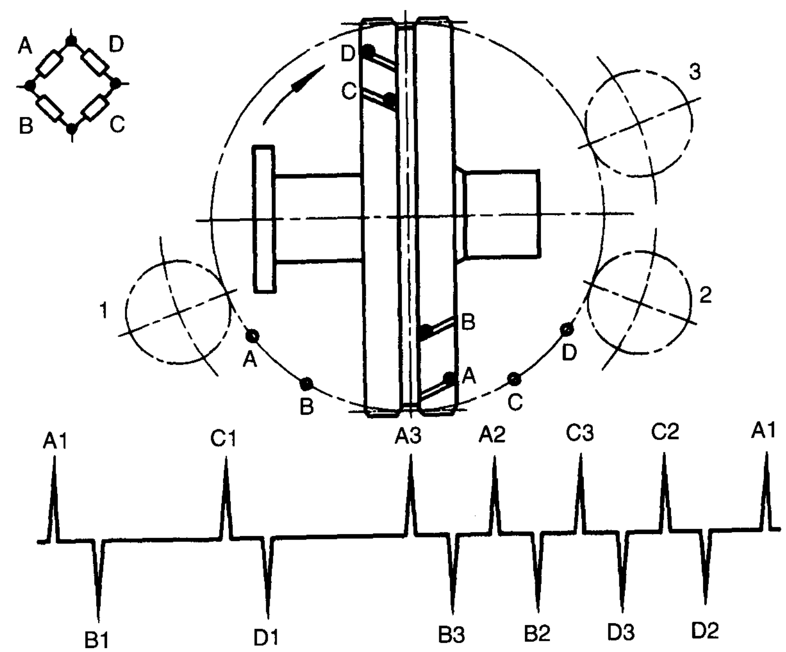


Figure 9-27. RENK-Checker for root stress monitoring. Courtesy RENK Tacke.

Figure 9-28 shows a RENK-Checker output display for the three medium-speed installations. Below the display is a diagram showing how the stress peaks are distributed over the pinions 1 to 3 and the strain gauges A to D. In this case, the load is well distributed on the three pinions. It is obvious that the input load of the three engines and the torsional vibrations of the system will greatly influence the displayed stresses. Careful observation of the stresses over time will permit the engineer officer to correct for these influences.

Although seldom installed, main shaft torsional and axial vibration pickups are useful in determining the extent of damage to a propeller or vane wheel as well as the accumulation of marine growth on a propeller.

GEAR ALIGNMENT

With few exceptions, operating problems with properly designed and installed main propulsion gears can be attributed to improper lubrication or incorrect alignment of the gear elements. Correct alignment requires that the mating gear elements be in the same plane and that the element centerline distances be maintained at the design values. Many factors in the

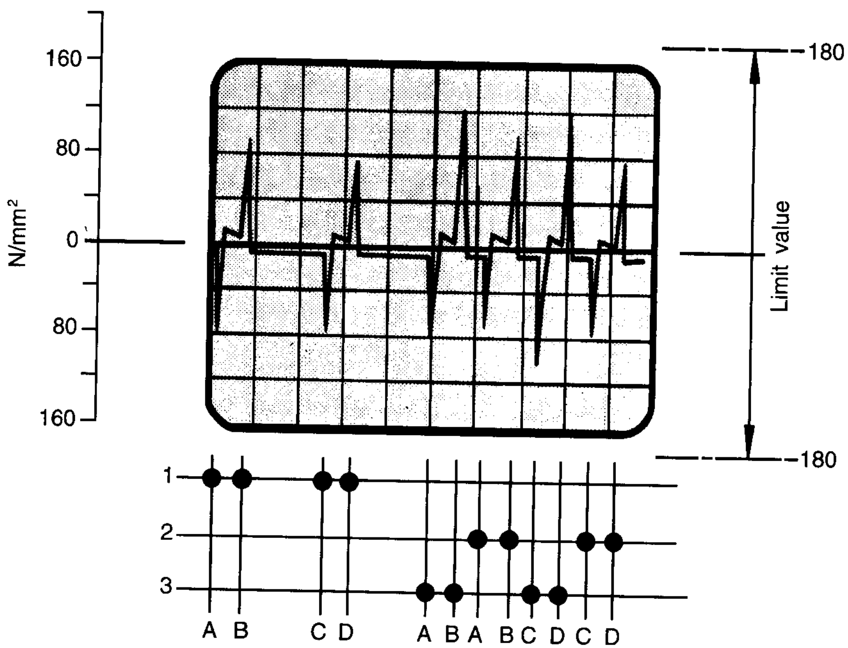


Figure 9-28. RENK-Checker display. Courtesy RENK Tacke.

shipboard environment influence the alignment of the propulsion gearing: hogging, sagging, and racking of the hull in a seaway; cargo storage methods; gear bearing wear or failure; temperature differentials; line shaft misalignment due to many causes, including stern tube bearing wear, grounding of the vessel, collision of the vessel, improper lube-oil distribution or temperature, and other causes. The vertical distortion of a gear foundation by a few mils can affect the alignment and consequently the tooth loading of a propulsion gear.

The high accuracy of manufacture ensures that propulsion gears are sensitive to very small dimensional changes. The writer recalls an event where ship's engineer officers placed a piece of canvas over an open inspection port to prevent the entry of foreign material. During subsequent turning gear operation, the canvas entered the second reduction gear mesh, requiring the turning gear to be reversed to extract the canvas. There was no visible damage as a result of the canvas entering two teeth of the mesh. However, when the gear was placed in service, a noise was heard twice per revolution of the second reduction gear. It was necessary to cast a hone on the second reduction gear and use it to locate and remove the distortion of the teeth.

Contact Patterns

Given a theoretical gear and pinion with proper center distances and parallel axes, where both elements have a perfect match of helix angles and perfect involute profiles, and there is no lateral or torsional bending of either gear element, the instantaneous contact will appear as a straight line on each tooth of the forward and aft helixes. Forward and aft refer to the orientation of the gear in the ship. Figure 9-29 shows the instantaneous

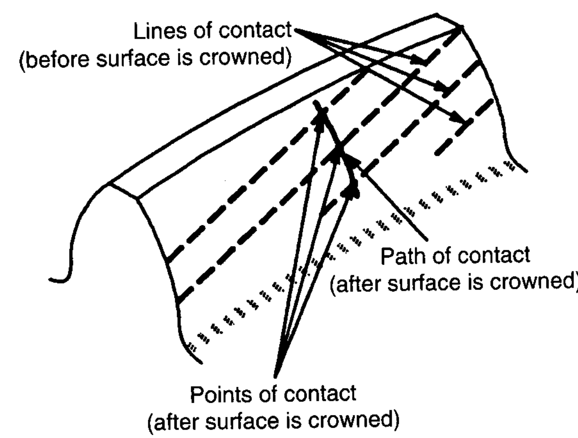


Figure 9-29. Instantaneous line of tooth contact

line of contact for a helical gear tooth. The top land and the ahead face of the teeth are shown with the instantaneous contact lines on the face. Due to their appearance, the instantaneous contact points of engaged helical gear elements looking into the mesh are sometimes called crossovers.

EXAMPLE 9-4: The length of helical teeth equals the face width divided by the cosine of the helix angle. If the theoretical gear has a pitch line helix angle equal to 30 degrees, a base helix angle of 28.49 degrees, an axial pitch of 1.78 inches, a face width of 11.3 inches, and a contact ratio of 1.6, then the total contact and contact per crossover are as follows:

Length of tooth = 11.3/cos 30 + 13.05 inches

Total contact = 11.3/cos 28.49 = 20.57 inches

Number of crossovers = 11.3/1.78 = 6.35

Length of contact per crossover = 20.57/6.35 = 3.24

As the gear and pinion go through mesh, i.e., follow the arc of action, the locus of the points of contact lies on the straight line called the line of action. The instantaneous contact line sweeps out an area simultaneously across the working depth and the length of the tooth. Except at the pitch circle where rolling contact occurs, the pinion teeth slide relative to the gear teeth to sweep out the contact area on the gear teeth. On the theoretical gear, the sweep-out area would cover the entire working depth of the

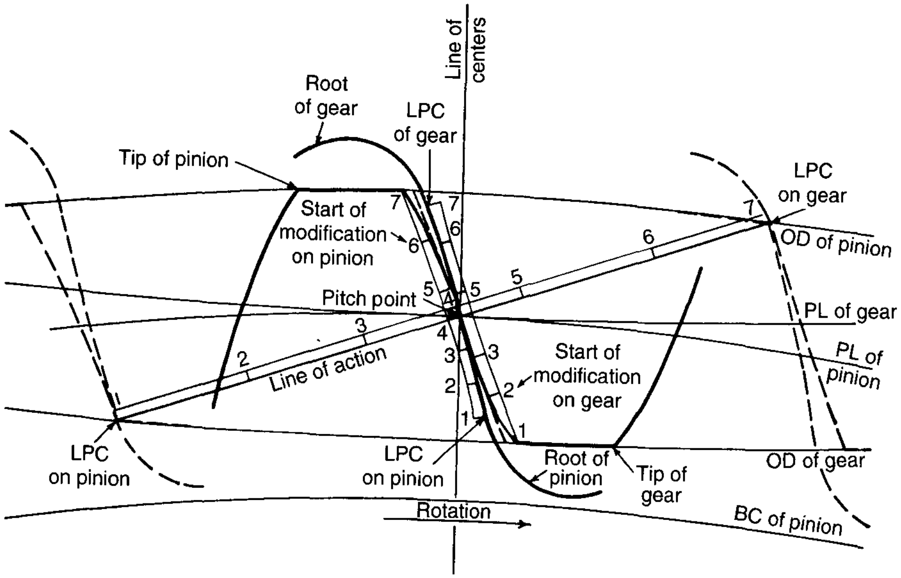


Figure 9-30. Tooth action in the transverse plane

gear teeth for the entire length of the teeth. Figure 9-30 illustrates the contact of a pinion and gear in the transverse plane during movement through one circular pitch.

Red and Blue Check

To view the contact of a pinion and gear that have gone through mesh, a red and blue check is very useful. For the number of adjacent teeth on each helix required to include a circular pitch, paint a very light coat (i.e., .00005 to .0001 inch thick) of red lead on the gear teeth and Prussian blue on the pinion teeth. As the teeth contact each other in mesh, the Prussian blue will be deposited on the red lead, leaving a pattern that shows the swept-out area of the instantaneous contact lines. For the theoretical gear, a broad band of blue will be deposited the length of the gear. If the theoretical gear has the involute relieved at the top and bottom as shown in figure 9-30, the red and blue check would indicate a narrower uniform band the full length of the teeth when compared to the unrelieved gear. If the teeth are relieved on their ends, the blue band will be shorter than the tooth length. Red and blue checks are very useful for the investigation of gear problems and verification of acceptable alignment of installed gears.

Tooth Bending

In an actual gear, the design engineer must accommodate lateral and torsional bending of the gear elements. Lateral bending occurs in single torque path gears due to the tangential forces tending to push the gear elements apart. In a dual torque path gear, the pinion drives two gears that are normally located in the same plane so that the tangential forces are equal and in opposite directions and there is no lateral bending. Torsional bending is due to the torque transmitted by the gear elements, with the bending greatest at the coupling end of the pinion. The amount of bending is a function of the torque transmitted and the stiffness of the element. Design engineers specify a stiffness that maintains all bending at minimal values. The bending of gears compared to pinions is insignificant due to the much greater stiffness of the gear. In single torque path double reduction gears, the first reduction has significant torsional bending because of the small diameter, while the second reduction pinion has significant lateral bending due to the high torque and consequent higher tangential forces.

Since the maximum load and bending occurs at full power ratings, the design engineer specifies a manufacturing adjustment of the helix angle and the shaft parallelism so that full tooth contact occurs at full power. Less than full tooth contact at part loads is acceptable because of the greatly reduced loads on the teeth. Helix angle modification or tapered teeth (which is the only way to accommodate lateral bending) must be accomplished during the manufacturing process. Both helix angle modification and adjustment of centerline parallelism may be employed to correct for torsional bending. Adjustment of gear centerline distance and parallelism may be accomplished aboard ship by carefully scraping material from the journal bearing at the reaction point.

Figure 9-31a illustrates the tooth contact patterns that would be expected at increasing torque on gear teeth that have been adjusted by helix angle modification or alignment to compensate primarily for torsional bending. At zero torque, the tooth contact exists only on the ends of the two helixes that are furthest from the coupling. As torque increases, the contact extends across the tooth surface to a uniform band at full torque. If the band is not uniform at full load, the bearing at the coupling end may be scraped to increase contaGt at the coupling end. Scraping the journal bearing away from the coupling end will increase contact band width at that end.

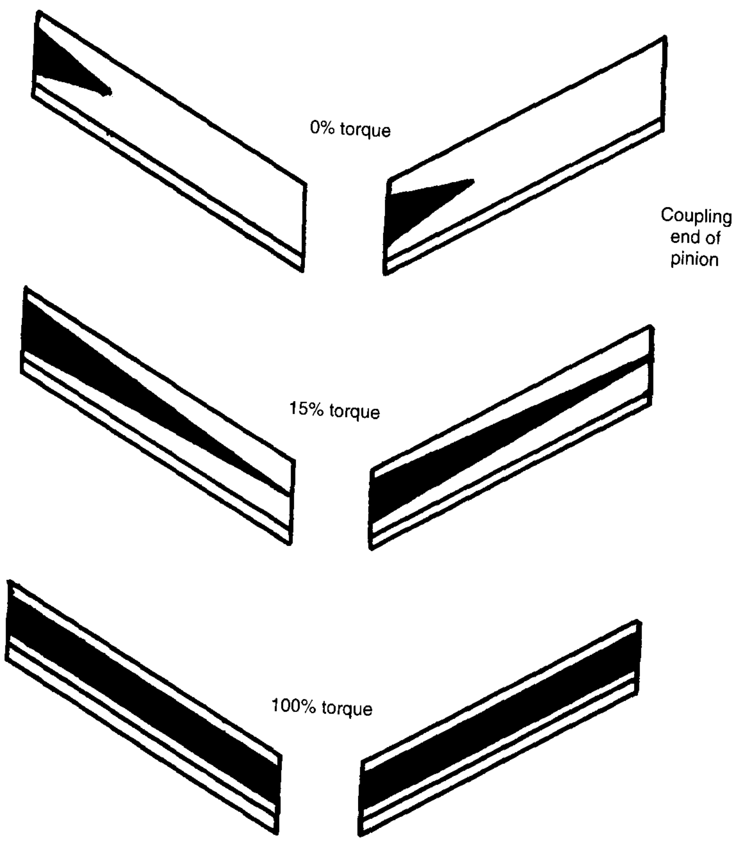


Figure 9-31a. Contact pattern at increasing load on tooth corrected for torsion bending.

Figure 9-31b illustrates tooth contact patterns at increasing load for a gear that has modified helix angle to compensate primarily for lateral bending. Zero torque contact occurs at the apex of the gear element helixes. As the pinion deflects laterally, the contact spreads along the teeth, reaching full uniform contact at maximum design torque.

The no-load opening at the contact crossovers for a typical double helical gear is in the range of 0 to .0005 inch. With the gear and pinion in their no-load bearing reaction positions, the opening may be measured with a feeler gauge at each ofthe crossovers for comparison to the manufacturer's factory assembly data. Use of feeler gauges with thickness variations in the range of .0001 and .0002 inch require skill and experience to obtain consistent and satisfactory results.

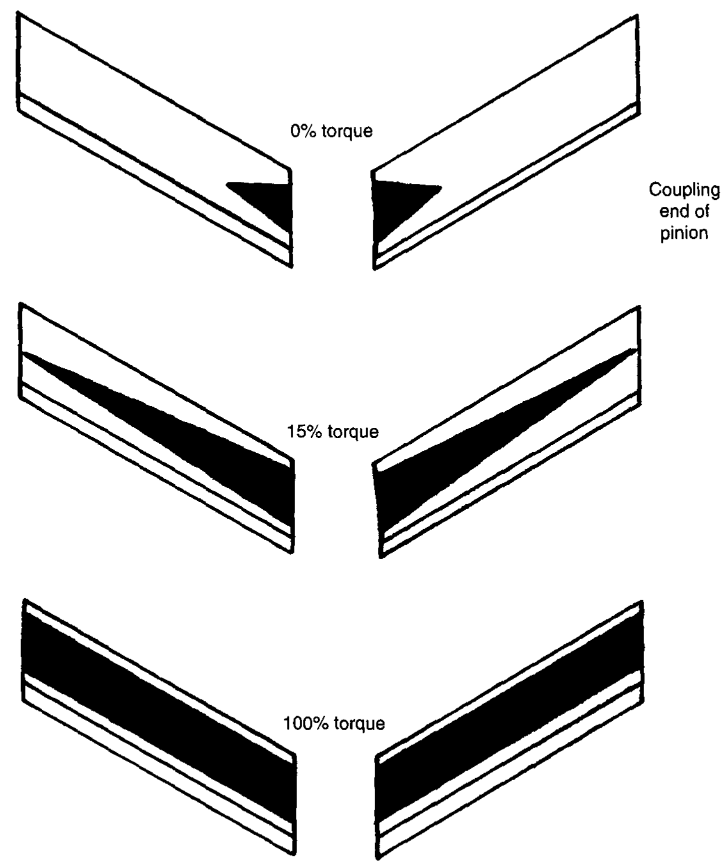


Figure 9-31b. Contact pattern at increasing load on tooth corrected for lateral bending.

Bearing Reactions

Reduction gears with antifriction bearings have no bearing clearance and the pinion and gear are always in the same position relative to each other regardless of the torque transmitted. However, antifriction bearings are an uncommon application for main reduction gears. Hydrodynamic bearings, described in chapter 4, with typical clearance of .001 inches per inch of journal diameter are the normal case. As a result, for a gear with bearing clearance greater than .015 inch, it is not unusual to have a full load gear center distance tolerance of .0001 inch or smaller when the elements are in the reaction position since the calculated deflections of the pinion under load are in the order of .0002 to .0003 inch.

The ahead and astern bearing reaction points are the positions that the pinion and gear journals assume in their bearings when torque is applied. These positions change slightly from low to full load but generally are close enough so that the no-load reaction position may be used in the field as the bearing reaction position. The gear bearing reaction position is the resultant of forces acting on the gear and causing the journal to assume different locations in the bearing clearance. These forces include the weight of the gear element, a force acting downward, and the force(s) acting normal to the tooth surface, a force(s) equal and opposite to the tangential tooth force times the cosine of the pressure angle in the plane of rotation. The resultant force may be determined graphically by drawing a force polygon for each element in the ahead and astern directions as shown in figure 9-32. Except in unusual cases, it is assumed that the bearing reaction forces are divided equally between the two journal bearings of a gear element. The exception might be a case where a lightweight first reduction pinion is connected to a heavy coupling and distance piece, causing a significant difference in bearing reaction loads. In figure 9-32, it is interesting to note that the high-pressure first reduction gear and the high-pressure second reduction pinion that are on the same axial centerline have bearing reactions that are 33 plus 55 (or 88) degrees out of phase. Since there is ample bearing clearance for the elements to move significant radial distances from one another, a large bending moment is established in the shaft connecting the first reduction gear to the second reduction pinion. The moment is accommodated by one or two flexible couplings and a quill shaft as shown in figure 9-33. Bending moments are also established at the solid coupling between the second reduction gear and the main line shaft. Alignment of these components seeks to minimize this bending moment when the second reduction gear is in its full load bearing position. This journal position is off-center due to the movement of the journal to the reaction position, and above-center due to the combination of gear case expansion at operating temperature and the movement to the reaction position.

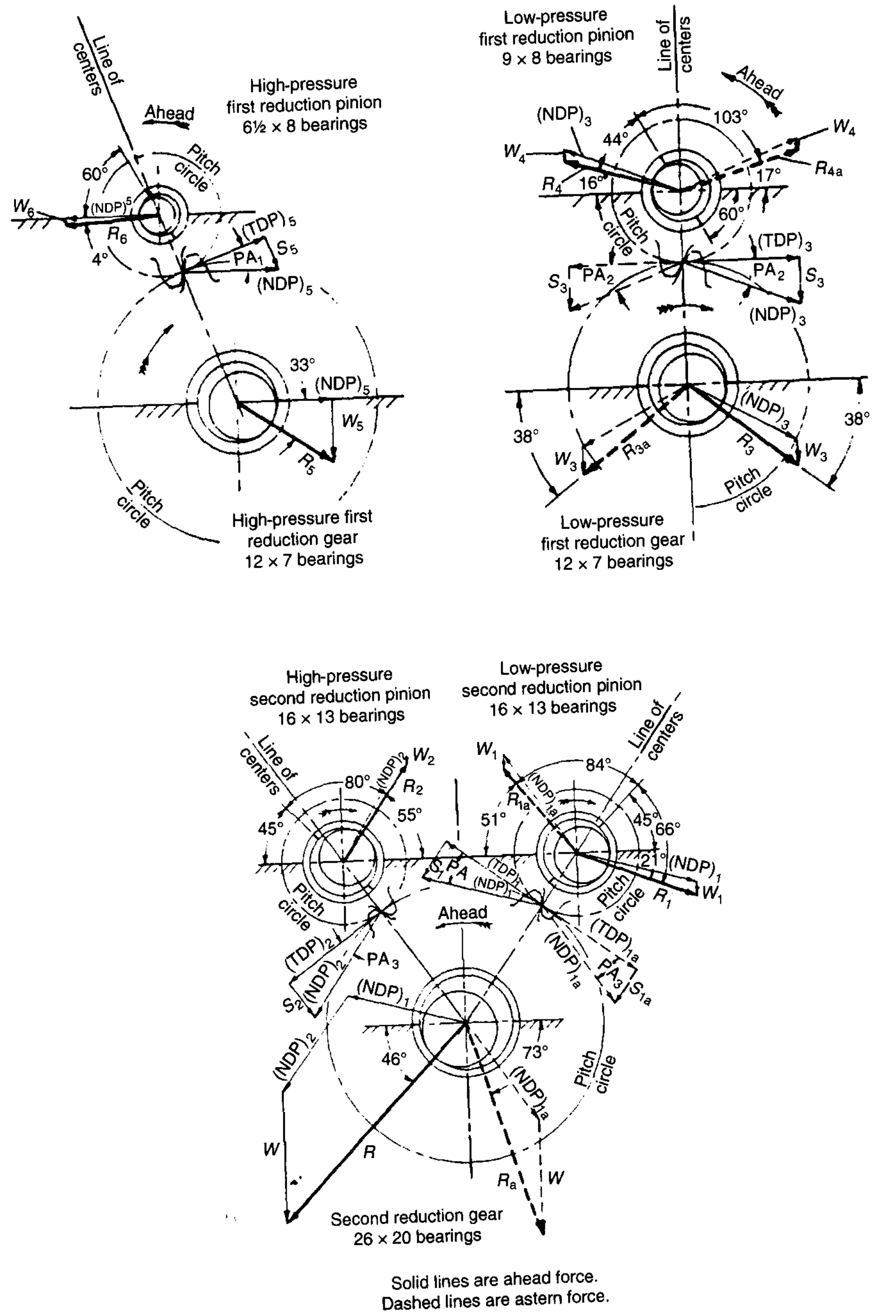


Figure 9-32. Ahead rotation looking aft. Typical bearing reaction diagram. Courtesy General Electric Company.

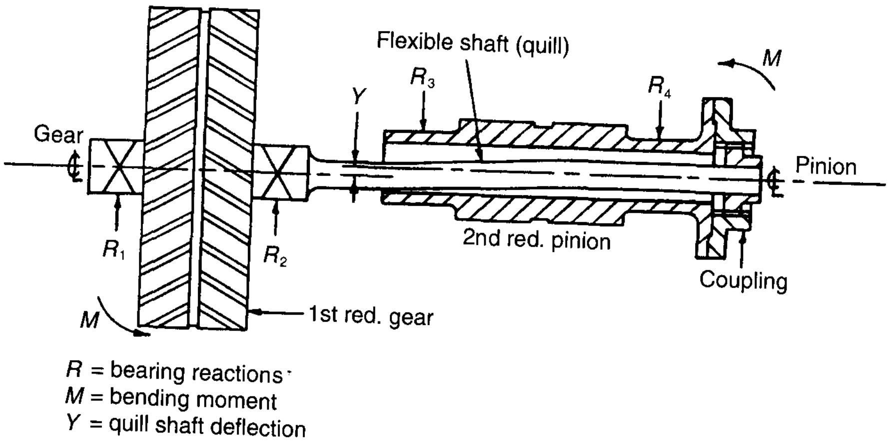


Figure 9-33. Quill shaft bending moments.
Courtesy General Electric Company.

Tooth Contacting Procedures

During manufacture, pinion and gear contacts are checked at various stages using several methods. At factory assembly, rotors in their bearings are placed in the bearing reaction positions so that red and blue checks may be made and recorded. A common method of placing the rotors in the reaction position is to install a brake on the low-pressure first reduction input flange and to engage and operate the turning gear at full amperage to apply sufficient torque to lift the elements into the reaction positions. With red and blue on the gear teeth and the brake providing about 10 percent torque, the unit can be rotated through the number of teeth necessary to make a near no-load contact for one full tooth. These contacts are usually recorded so that they can be compared to subsequent shipboard contacts after installation, or to a periodic examination of the gear if the brake is available as a shipboard tool.

Ifthe gear is tested in the factory, gear tooth contacts are observed and recorded at 50, 75, and 100 percent ofload. Without factory full torque tests for comparison, the shipboard sea trial full load checks, the dock trial lower torque contacts checks, and subsequent periodic load contact checks are the most useful and informative checks of the condition of a propulsion gear (except the permanent installation of tooth bending strain gauges that provide a real time display of the gear alignment condition).

To make shipboard tooth contact checks under load, use the following procedure for the elements to be checked: (1) Clean the teeth carefully for a band of about 30 degrees of arc using chlorothen. Only one of the two meshing elements will be coated to provide the better accuracy of a single coating of the dye medium. Use Dykem Layout red dye and apply it to the

cleaned area using a piece of felt. This lacquer-based dye is resistant to lube oil. It will not be removed by oil. The dye must be applied thinly so that it has a thickness of only .0001 to .0002 inch. (2) Operate the gear for about one hour at the torque for which the contact pattern is desired. The gear casing is then opened and the contact pattern observed. Full scale sketches ofthe pattern should be made for comparison to the patterns developed at higher torques. (3) Compare the contents with factory data and prior shipboard contacts. (The tooth contact pattern is indicated by the areas where the dye has been removed.) It is desirable to make full torque contact checks anytime the ship is subjected to unusual hull loading such as very heavy seas, grounding, or other mishap.

GEAR PROBLEMS

Except for the small area at the pitch line where rolling contact occurs, helical gear tooth contact is a sliding action as the instantaneous contact lines are swept across the involute surface and along the tooth length. Despite the high accuracy of gear manufacture and the quality tooth finishes provided by shaving or grinding operations, small imperfections exist in the profile shape, helix angle, and surface finish. Correct lubrication and initial operation under load usually smooths out the imperfections. Gear units that are lubricated with the proper oil at the design flows and temperatures and maintained free of water and foreign material through the use oflube oil purifiers can be expected to have a long trouble-free life. The following are descriptions of gear failure conditions that require immediate correction of causes.

Tooth Breakage

A cracked or broken tooth is the result of severe overstress of a portion of the teeth due to misalignment or other form of overload. When this is detected, the gear must be operated at greatly reduced torque until the source of the overload can be detected and corrected. For example, if a crack appears in one or more teeth at some distance from the end of the tooth, an appropriate emergency measure would be to use a portable milling machine to remove the portion of the tooth between the crack and the end of the tooth and to make a proportional reduction in the maximum operating torque of the gear until the gear elements can be replaced. The replacement gear elements must be a matched pair.

Pitting

If a gear tooth contact surface has local high spot or variations in material hardness, localized high Hertzian stress occurs. Since these local stresses occur each time the gear passes through mesh, metal fatigue eventually occurs, causing minute particles to break away, leaving a pattern of small pits. Pitting usually Occurs near the pitch line, where rolling contact is the predominant movement between the two surfaces. The sliding action in other tooth areas tends to wear away minute imperfections before pitting occurs. If pitting becomes severe, it may be necessary to remove and rehob the gear element.

Abrasion

Abrasion is usually identified by fine tracking marks or scratching on all the teeth in a transverse plane; it is caused by foreign material in the mesh. The scratching marks usually extend in the direction of the sliding action on the teeth. Sources of abrasion are dirt, rust from the casing, or material particles larger than the oil film on the tooth surface, which are caused by pitting of the gear. Periodic use of the oil purifier system and care when opening a gear casing are the best protection against abrasion. When abrasion upsets the surface of gear teeth, the upset can be removed by using a hone, which will be described later.

Scratching

Scratching Occurs when a tooth has a small upset area that gouges short furrows in the teeth of the mating gear element. Eventually, both mating elements have scratching in the transverse plane. If the damaged area continues to grow, it is necessary to relieve the high spots with a hone or small file to stop the growth of the scratched area.

Spalling

Spalling OCcurs when the repeated Hertzian stress exceeds the design endurance limit for the gear material due to misalignment or other overload. In time, subsurface fatigue cracks develop, causing metal pieces to break away from the surface, leaving large pits. Spalling is most likely to occur on the area of the tooth flank of a pinion below the pitch line. The cause of the overload must be identified and corrected to halt the spread of spalling.

Galling

In propulsion gear units, thin-film lubrication exists due to the high load transmitted by the teeth. When the combined loading and sliding velocity exceeds the limit of a thin-film boundary, the high spots of the surfaces weld together and then break apart as the teeth go through mesh. This is the mechanism called galling. Galling tends to occur near the tips of teeth since the sliding action is greatest in this area leading to boundary-film failure. Galling is a lubrication problem that must be solved with the technical assistance of lubrication experts.

GEAR INSPECTION AND REPAIR USING A PLASTIC HONE

Visible Observations

Visual observation of gear teeth in proper light and angle will frequently reveal damaged areas. Quantification of the damage is impossible and only judgment and experience will permit the determination that the element will operate satisfactorily. Upset in the gear surface is the result of mishandling or the introduction of foreign material. Magnaflux is useful to determine if the damage observed has caused a crack in one or more teeth.

In an emergency, the upset area can be handworked with a file or stone to relieve the upset area.

Noise and Vibration Checks

Noise and vibration are important indicators of a gear problem. At slow speed, a once-per-revolution noise, such as a knock, points to an upset area on one or more teeth. Noise and vibration are very important when the upset area is not visible.

Operation of the gear for ten minutes at a light load will sometimes polish the area of an upset that was not originally visible. The use of a Fast Fourier Transform vibration analyzer described in volume 2 is an accurate method of identifying the magnitude and location of a damaged gear tooth.

The application of a dye to the suspected area of tooth damage, followed by a short low load run of the machinery, will identify the specific upset area.

Plastic Hones

An abrasive plastic hone may be used to detect and correct minor gear tooth surface upset areas. The hone is prepared by casting a material directly on an undamaged portion of the teeth. The hone will detect minute tooth irregularities as small as .00002 inch when passed back and forth over the tooth surface a few times until the damaged area becomes visible due to the polishing action of the hone. Minor surface imperfections are not only made visible, they may be removed by a few strokes of a hone.

Large imperfections on the tooth surface will cause a hone to bind. The binding can be relieved by raising the hone slightly and continuing the strokes until the upset surface is polished and visible. The high spot can then be handworked with an India stone until it is possible to stroke the hone through the teeth; then the surface of the tooth profile can be relieved and restored. Forcing a hone into a binding area will distort the hone shape by excess wear and bring poor results.

A hone that has been cast on a given gear may be used on similar gear teeth that have the same diametral pitch, helix angle, pressure angle, and number of teeth. A separate hone must be cast for the forward and aft helixes.

If a hone is prepared in advance, it is possible to use the hone through a gear inspection port for both inspection and minor corrections. If hones must be cast, or substantial work undertaken, it is usually necessary to remove the gear covers. Proper lighting, staging, and precautions against introduction of foreign material are essential when the covers are removed for this purpose.

Ifan upset area on a tooth is small, i.e.,,Ysinch in diameter, and the upset is about .001 inch high, it will require five to ten passes of the hone to remove the bump. Ifthe upet area is relatively large, i.e., ,Ysinch wide by 1 inch long by .00001 inch high, it will require fifty or more passes ofthe hone to remove the bump. Experience suggests that it is possible to make 100 strokes or more with a hone using 10 to 15 pounds of hand pressure without damaging the tooth profile, the tooth spacing, or the helix angle. It is estimated that .0001 inch of material is removed by 100 strokes of a hone using kerosene as a lubricant.

For a significant number of strokes, it is convenient to attach the hone to a portable vibrating-type sanding machine to improve the effectiveness ofthe procedure.

Preparing an Abrasive Hone

The following steps are required to prepare an abrasive hone.

1. Assemble the following materials: epoxy resin, Epon type; catalyst, Epon curing agent U; abrasive, #60 grit alundum; mold release, Thalco #225; modeling clay; concentrated sulfuric acid; neutralizing solution, sodium bicarbonate; kerosene; glass trays; rubber gloves; putty knife; wire brush, emery cloth, set of files; wooden plugs to fit root width; soft brush; metal identification strips.
2. Remove gear covers from the elements of interest and provide lighting and staging for area. Clean a band ofteeth located on top of gear. Cover all surfaces to prevent intrusion of foreign material.
3. Figure 9-34 is a drawing of a mold prepared for multiple tooth hone. Prepare the mold using the following steps.
   a. Fill the root area with molding clay up to the working depth of the teeth and scrape out excess clay above the root area with a shaped wooden tool. Knead the clay until it is workable.
   b. Apply clay to top lands of teeth and trim clay even with the teeth.
   c. Form sides of mold using blocks of clay.
   d. Form ends of mold using blocks of clay and insert looped wire.
   e. Apply mold release to exposed tooth surfaces inside the mold.
4. Prepare the casting.
   a. Thoroughly stir the resin-grit mixture.

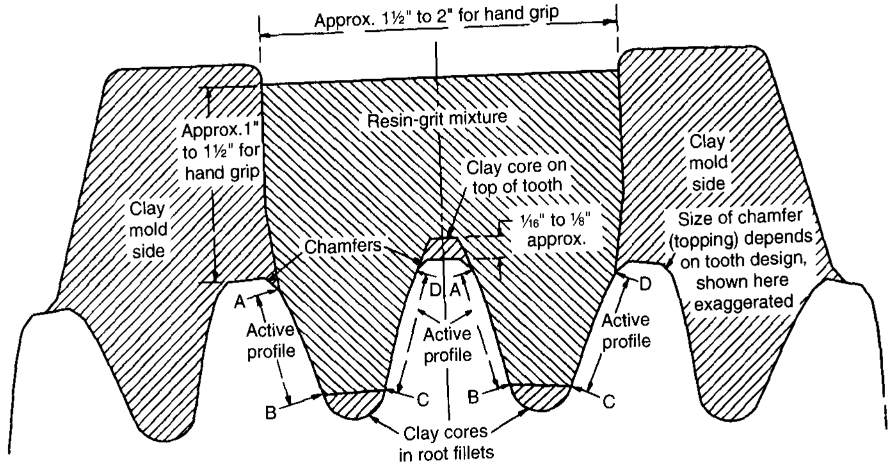


Figure 9-34. Hone mold. Courtesy General Electric Company.

   b. Add catalyst to resin-grit mixture and stir mixture for about 7 minutes.
   c. Pour mixture into mold to a level 1 inch over the tops of teeth.
   d. Allow the casting to set half an hour, then place identification plates into the top surface.
   e. Allow casting to cure for 8 hours minimum.
5. Remove and finish the cast hone.
   a. Remove clay and tap hone to remove from gear teeth.
   b. Clean hone with wire brush and water washing.
   c. Remove any sharp edges from hone by filing.
   d. Etch tooth surfaces of hone for 12 minutes in sulfuric acid. Use caution handling acid.
   e. Rinse in water.
   £. Rinse in neutralizing solution.
   g. Wash in water and dry.
   h. Remove grit from surfaces of hone with emery paper.
   i. Mark identification plate with the gear teeth data and helix location.

PROPULSION LINE SHAFT

A ship propulsion system line shaft has the following functions which it must provide with high reliability since for single-screw vessels there is no redundancy in this part of the system.

1. Transmit torque, Q, from prime mover to propeller.
2. Transmit the propeller thrust, T, ahead and astern, to the main thrust bearing and hence to the hull.
3. Support the overhanging weight ofthe propeller, which is significant.
4. Withstand the alternating bending forces and the consequent fatigue stress on the tailshaft due to propeller forces.

These functions must be provided within acceptable vibration limits. Depending on the type of ship, type of line shaft arrangement, and the forces involved, the designer must take into consideration lateral, axial, and torsional vibrations. With the exception of passenger ships, in most commercial vessels torsional vibrations are the major concern. Torsional vibrations that are stimulated by the propeller blade passing frequency are accommodated by tuning the system so that the natural frequencies occur outside the normal operating speed range of the system. Turbine gear shaft systems normally pass through a resonance frequency or first critical vibration mode to reach operating speed. In multi-input turbine gear systems, the system tuning is accomplished by selecting a torsional stiffness ofthe gear quill shafts that moves natural system frequency to the desired value.

The line shaft loading is primarily a torsional stress which determines the shaft diameter. Thrust and lateral stresses are also present due to the propeller thrust and shaft weight respectively, but they are not significant compared to the torque. The thrust shaft or portion of the line shaft that includes the thrust collar is larger than the line shaft to accommodate the ahead and astern thrust loads in combination with bending forces tending to upset the thrust bearing and the longitudinal vibration forces from the propeller. The tailshaft, which supports the overhanging weight of the propeller (in the range of 10 to 20 tons), is subject to a complex combination of alternating torsion, bending, compression, and shear. Modern ships have oil-lubricated stern tube bearings which are not subject to excessive wear. However, in ships with water-lubricated bearings, the bearings wear, allowing the tailshaft to move about in the excessive clearance, thus increasing the shaft bending to levels not anticipated in the design. The World War II Liberty ship design experienced many tailshaft failures before the design was modified.

Propeller Shaft Stress

Since the tailshaft or propeller shaft is the most highly stressed member on the propulsion shafting system, it will be instructive to review the complex tailshaft stresses of a typical 31,000 DWT product carrier.

EXAMPLE 9-5: The following information applies to the solid shaft of the product carrier.

Shaft horsepower	12,000
Propeller rpm	92
Vessel speed, knots	16
Propeller diameter, ft.	21.9
Number of propeller blades	4
Propeller weight, lbs	26,000
WR^2, in air, lb-in^2	68,280,000
Shaft diameter, inches	16.5
Moment arm of the propeller, L_p, in	50

Torque:

$$Q = \frac{63{,}025\,(\text{shp})}{\text{prpm}} = \frac{(63{,}025)(12{,}000)}{92} = 8{,}220{,}652 \text{ inch-lbs}$$

Shear stress due to torque:

$$S_s = \frac{5.1Q}{D^3} = \frac{5.1(8{,}220{,}652)}{(16.5)^3} = 9{,}333 \text{ psi}$$

Thrust:

$$T = 326\,\frac{\text{Ehp}}{V}\cdot\frac{1}{1-t}$$

where

T = thrust, lbs
Ehp = effective horsepower
V = ship speed, knots
t = thrust deduction

$$T = \frac{326\,(12{,}000)}{(16)(.8)} = 305{,}625 \text{ lbs force}$$

Compressive stress in tailshaft:

$$S_c = \frac{T}{A} = \frac{4T}{\pi D^2}$$

where

S_c = compressive stress, psi
T = thrust, lbs, force
A = area of tailshaft, sq. inches
D = diameter of tailshaft, inches

$$S_c = \frac{4(305{,}625)}{\pi(16.5)^2} = 1{,}429.3 \text{ psi}$$

Resultant steady-state stress:

$$S_{sr} = \sqrt{(S_c)^2 + (2S_s)^2}$$
$$S_{sr} = \sqrt{(1{,}429.3)^2 + [2(9{,}333)]^2} = 18{,}720 \text{ psi}$$

where

S_{sr} = resultant steady stress, psi
S_c = compressive stress, psi
S_s = shear stress, psi

Moment due to propeller overhang, lbs:

$$M_g = W_p L_p$$

where

W_p = propeller weight, lbs
L_p = distance of overhang, inches
M_g = (26,000) (50) = 1,300,000 in-lbs

Moment due to off-center thrust:

$$M_{oc} = 2\,M_g = 2{,}600{,}000 \text{ in-lbs}$$

Combined propeller moment:

$$M_p = M_g\,M_{oc} = 3{,}900{,}000 \text{ in-lbs}$$

Alternating bending stress at propeller:

$$S_b = \frac{(10.2)M}{D^3} = \frac{(10.2)(3{,}900{,}000)}{(16.5)^3} = 8{,}855 \text{ psi}$$

Torsional vibratory stress:

$$S_s = .05(9{,}333) = 466.5 \text{ psi}$$

The torsional vibratory stress is estimated to be 5 percent of the steady-state stress for a steam turbine propulsion system.

Alternating resultant stress:

From Maleev, the torsional stress concentration factor, k, is 1.9 for the vibratory shear stress. Then the combined alternating stress is

$$S_{sr} = \sqrt{(S_b)^2 + (2k_t S_s)^2}$$
$$= \sqrt{(8{,}855)^2 + [2(1.9)(466.5)]^2}$$
$$= 9{,}030.7 \text{ psi}$$

Factor of safety:

$$\frac{S_{\text{steady resultant}}}{YP} + \frac{S_{\text{alternating resultant}}}{FL} = \frac{1}{FS}$$

where

YP = yield point, psi
FL = fatigue limit, psi
FS = factor of safety

Select a material with the following properties:

Tensile strength = 95,000 psi
Yield strength = 75,000 psi
Fatigue limit = 46,500 psi

$$\frac{18{,}720}{75{,}000} + \frac{9{,}030.7}{47{,}500} = \frac{1}{FS}$$
$$FS = 2.27$$

Line Shaft Bearings

The bearings that support the line shaft between the thrust bearing and the stern tube bearing are called line shaft bearings, spring bearings, or steady bearings. Since these bearings are normally self-lubricated and must operate successfully from full speed (60–100 rpm) down to turning gear speed (one revolution in ten minutes) they are conservatively designed.

Line shaft bearings include cast or fabricated steel housings that are split horizontally at the shaft centerline for access to the bearing shell. This housing supports a heavy bearing shell that is fully lined with babbitt in the lower half and only partially lined in the upper half. The principal load on the line shaft bearing is the weight of the shaft and there is no operating condition where bearing reactions are transferred to the upper half of the shell. The bearing shell is sometimes made to be self-aligning by including a crowned seat at the interface between the shell and the bearing housing so that the shell can follow the incline of the shaft.

To achieve adequate lubrication, the length-to-diameter ratio of the line shaft bearings is in the range of 1 to 2. The bearings have an oil reservoir in the lower portion of the housing. The oil is delivered to the top of the bearing by an oil ring or a disk mounted on the shaft. A cooling coil, which is furnished with circulating seawater, may be located in the oil reservoir for use when required.

An estimate of the number ofline shaft bearings required can be made by dividing the load carrying capacity of one of the bearings into the total shaft weight less the weight supported by the stern tube bearing and the gear or engine bearing at the forward end of the shaft.

Shaft Alignment

Before the introduction of the high-speed electronic computer as an engineering tool in the 1950s, the alignment of the main reduction gear, propulsion generator, or engine to the line shaft, and the alignment of the line shaft to the tailshaft was determined by a few simplistic rules. The rules frequently failed to deal with the complex relationships, leading to large moments that caused pitting damage to second reduction gears, high fatigue stresses in the tailshaft and engine cranks, and bearing failures due to overloads.

Today all shipbuilders and propulsion machinery manufacturers have sophisticated computer programs with which to analyze the line shaft and generate the following information:

1. Shaft diameters that provide the desired factor of safety for the shaft material.
2. Bearing reactions or loads for the initial installed alignment and a number of operating conditions.
3. Shafting slope at discrete locations.
4. Shafting deflections at discrete locations.
5. Moments in the shafting at discrete locations.
6. Shear stress at discrete locations.
7. Bearing reaction influence numbers.
8. Vibration modes and frequencies of the system.

The influence numbers are a matrix arrangement of numbers that indicate the change in bearing reaction for every bearing in the system when a selected bearing is raised .001 inch. This information is used to adjust bearing height to achieve the design bearing reactions for a normal operating condition, i.e., the propulsion machinery at operating temperature. Figure 9-35 shows a typical bearing reaction influence number matrix. In the figure, if the first line shaft bearing, i.e., bearing number 3, is raised

Brg. no.	1	2	3	4	5	6	7	8
	Fwd gear	Mt gear	1st line sh	2nd line sh	3rd line sh	4th line sh	Fwd st tube	Mt st tube
	Reaction loads with all bearings in straight line							
	31,953.8	29,914.7	15,415.4	16,218.0	13,950.3	24,624.7	-11,907.3	91,541.0
	Bearing reaction influence numbers (lb per 0.001 in. bearing rise)							
Brg. no.								
1	2,664.2	-5,012.4	2,552.3	-291.9	100.9	-18.5	6.6	-1.2
2	-5,012.4	9,934.5	-5,625.3	1,005.3	-347.5	63.9	-22.7	4.0
3	2,552.3	-5,625.3	3,885.9	-1,293.2	552.3	-101.6	36.0	-6.4
4	-291.9	1,005.3	-1,293.2	1,419.8	-1,131.9	412.0	-146.1	26.1
5	100.9	-347.5	552.3	-1,131.9	1,358.0	-875.9	418.7	-74.7
6	-18.6	63.9	-101.6	412.0	-875.9	1,359.4	-1,218.5	379.2
7	6.6	-22.7	36.0	-146.1	418.7	-1,218.5	1,503.7	-577.8
8	-1.2	4.0	-6.4	26.1	-74.7	379.2	-577.8	250.8

Figure 9-35. Bearing reaction influence number

.001 inch, then the change in bearing reactions for all the bearings in the system is as follows:

1.	fwd gear	2,552.3
2.	aft gear	-5,625.3
3.	1st line	3,885.9
4.	2nd line	-1,293.2
5.	3rd line	552.3
6.	4th line	-101.6
7.	fwd stern	36.0
8.	aft stern	-6.4

As expected, the influence diminishes as the position of a given bearing increases in distance from the raised bearing.

Hydraulic Jack Measurement Method

To verify line shaft alignment and make necessary corrections, the hydraulicjack method is the best choice. This method compares the computer calculated bearing reactions with the influence to the bearing reactions measured on the shipboard shafting system. A calibrated hydraulic jack is placed under the shaft in the position nearest to each bearing. A dial indicator is located above the shaft directly over the jack. The bearing foundation sometimes includes a welded pedestal to support the jack. In new ship design, the owner may wish to specify that such a pedestal be furnished.

The support of the dial indicator should be independent of the bearing housing. Before recording data on the jack pressure versus mil rise of the shaft, it is desirable to raise and lower the shaft about .025 inch to ensure the shaft is free and does not engage the housing or bearing liner. This will also reduce the hysteresis in the mechanical elastic system, which consists of the bearing housing and the bearing foundation.

When measuring equipment is properly installed, the jack is used to raise the shaft in small increments, i.e., .0005 inch, and the pressure and lift are recorded at each increment up to a total of about .025 inch. The same data is recorded as the jack is lowered to the original position. It is desirable to repeat this cycle of measurements and record the data. Figure 9-36 is a plot ofthe two sets of data for a bearing. It should be noted that due to hysteresis, the data recorded during the shaft raising is different than the data for the shaft lowering. A line drawn parallel to and centered between the two data lines is a good estimate of the actual lift force versus shaft lift. The intersection of this median line with the horizontal coordinate provides the jack pressure needed to support the shaft in the current bearing position. This pressure times the jack calibration factor, which may be a curve, yields the bearing reaction. On the plotted curves, the portion with the lesser slope represents the deflection of the elastic foundation, bearing housing, and bearing shell.

The bearing reactions measured by the hydraulic jack method are sufficiently accurate for alignment use. The method does not yield accurate influence numbers. The calculated influence numbers must be used with the measured bearing reactions when making decisions concerning the vertical movement of bearing to correct the operating bearing reactions.

PROPULSION CLUTCHES

Clutches are required in some propulsion system where two or more prime movers are attached to a reduction gear or there is takeoff power from a reduction gear to an auxiliary such as a generator. Gas turbines and medium-speed diesels are examples of the former and in the latter case, the electric generator may also be used as an emergency propulsion motor.

SSS Clutch

In multiple engine gas turbine drives, the engine changeovers are accomplished automatically at any shaft speed within the operating ranges ofthe incoming and outgoing engines. The SSS-type clutch shown in figure 9-37 is suitable for this service.

The clutch functions as follows. Referring to figure 9-37, when the input half ofthe clutch is running slower than the output, the pawls ratchet over the ratchet teeth. However, when the input and output speeds are synchronized, the pawls engage and drive a sliding element. The sliding element carries the inner main clutch teeth along helical splines into engagement with the outer main clutch teeth. Phasing between the ratchet teeth and main clutch teeth ensures that the latter engage precisely. When the two halves ofthe main clutch are engaged, the pawls disengage from the end of the ratchet ring so that the pawls never transmit any portion of the main drive torque force. When the clutch output rotates faster than the input, disengagement occurs immediately and the helical splines return the sliding member back to the ratchetting position.

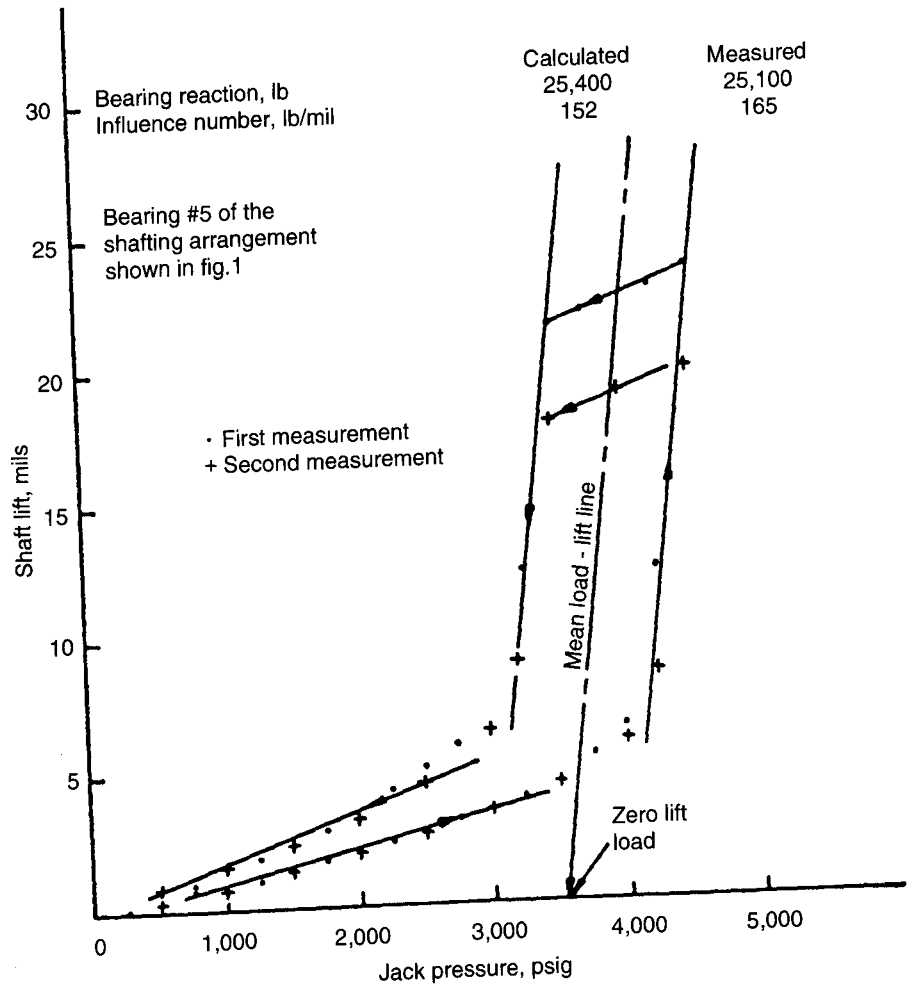


Figure 9-36. Hydraulic jack measurement of bearing reaction in line shaft

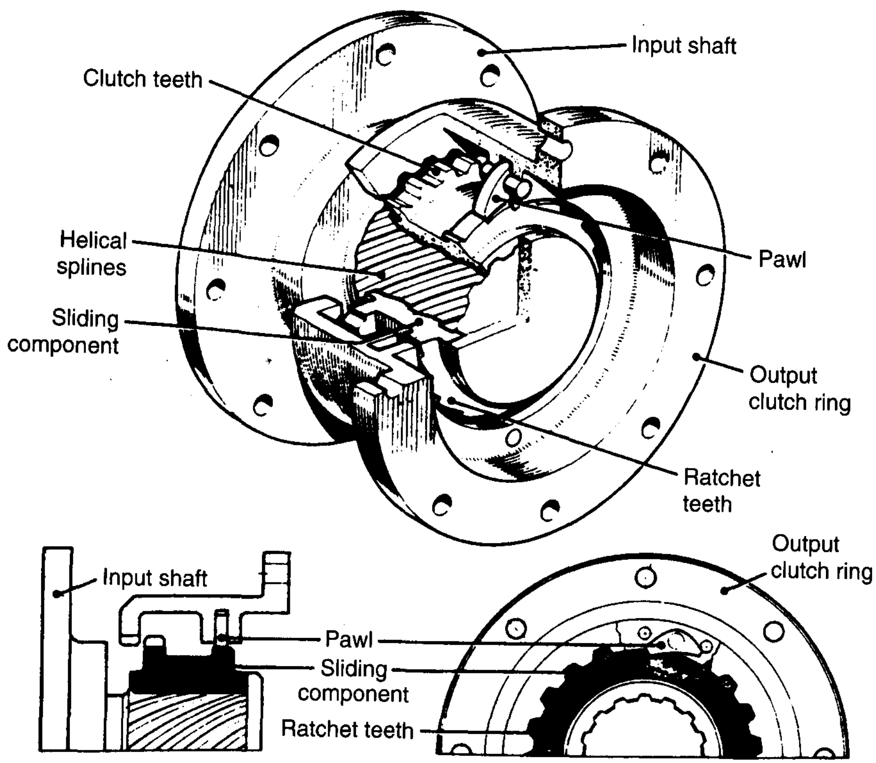


Figure 9-37. SSS clutch

When the main clutch is fully engaged, a lock-in feature operates automatically to prevent the clutch from shuttling between open and closed if torque reversals occur during maneuvering. A manually operated lock-out device may be employed to disengage the pawls to permit operation of the engine independent of the gear for test purposes.

Another feature is a device that prevents the pawls being reengaged with the ratchet ring when the clutch input is rotating faster than the output.

The SSS clutch is a reliable device. Problems or failures of the clutch can usually be attributed to misalignment between clutch input and output.

Friction Clutch

Figure 9-38 illustrates two multiple disk friction clutches employed to drive an electric generator from a takeoff from a main propulsion single reduction gear. The use of two clutches permits the selection of one of two speed reductions so that the propeller may be operated near an optimum speed. The generator may also be operated as an emergency propulsion motor.

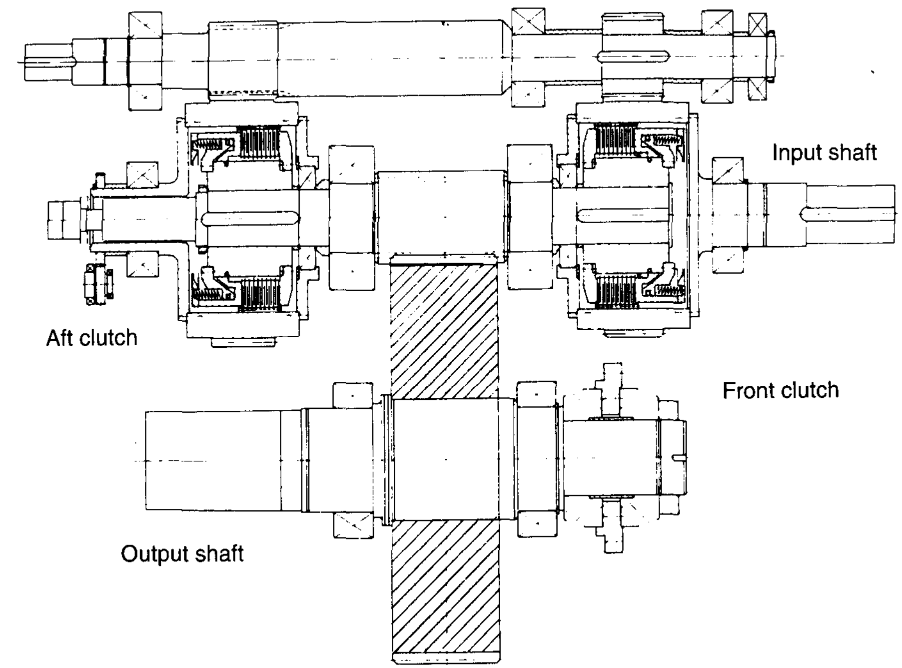


Figure 9-38. Transmission system with disk clutches. Courtesy Wartsila Diesel, Inc.

This high capacity multiple-plate clutch is hydraulically operated and cooled by the hydraulic fluid. Operating oil pressure control ensures shock-free engagement. The oil pressure is about 420 psi.

BOOSTER MOTOR AND TUNNEL GEAR

In 1964, the need for Sea-Land Services Inc. of New Jersey to convert three of their slow-speed diesel-propelled container vessels to higher-speed reduced-capacity vessels resulted in a unique transmission system. The conversion required hull modifications and a power increase of 4,000 kW to provide a speed increase of 3 knots. Since it was impossible to increase the power of the slow-speed diesel, a new diesel generator was installed to run an electric motor which transmitted power to the line shaft by a gear. Figure 9-39 shows the steps followed by Blohm+Voss AG to accomplish the conversion.

Figure 9-40 shows the power take-in solution for the upgraded transmission system. A Lohmann and Stolterfoth tunnel gearbox is flange-mounted by a Vulkan coupling to the main engine flywheel.

The 6,600 volt, 4,000 kW motor power is fed into the gearbox and thus to the shafting that passes through the gear box tunnel.

(a)

(b)

(c)

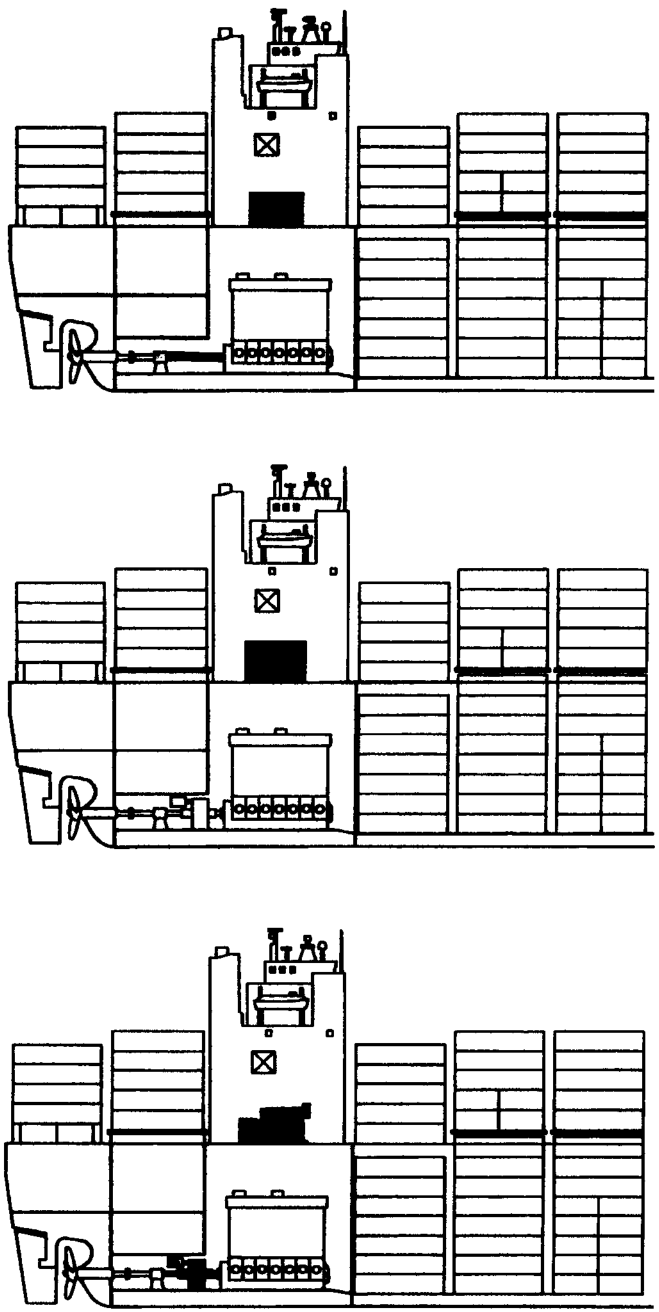


Figure 9-39. Sea-Land conversion employing booster motor and tunnel gear. Courtesy Blohm+Voss AG.

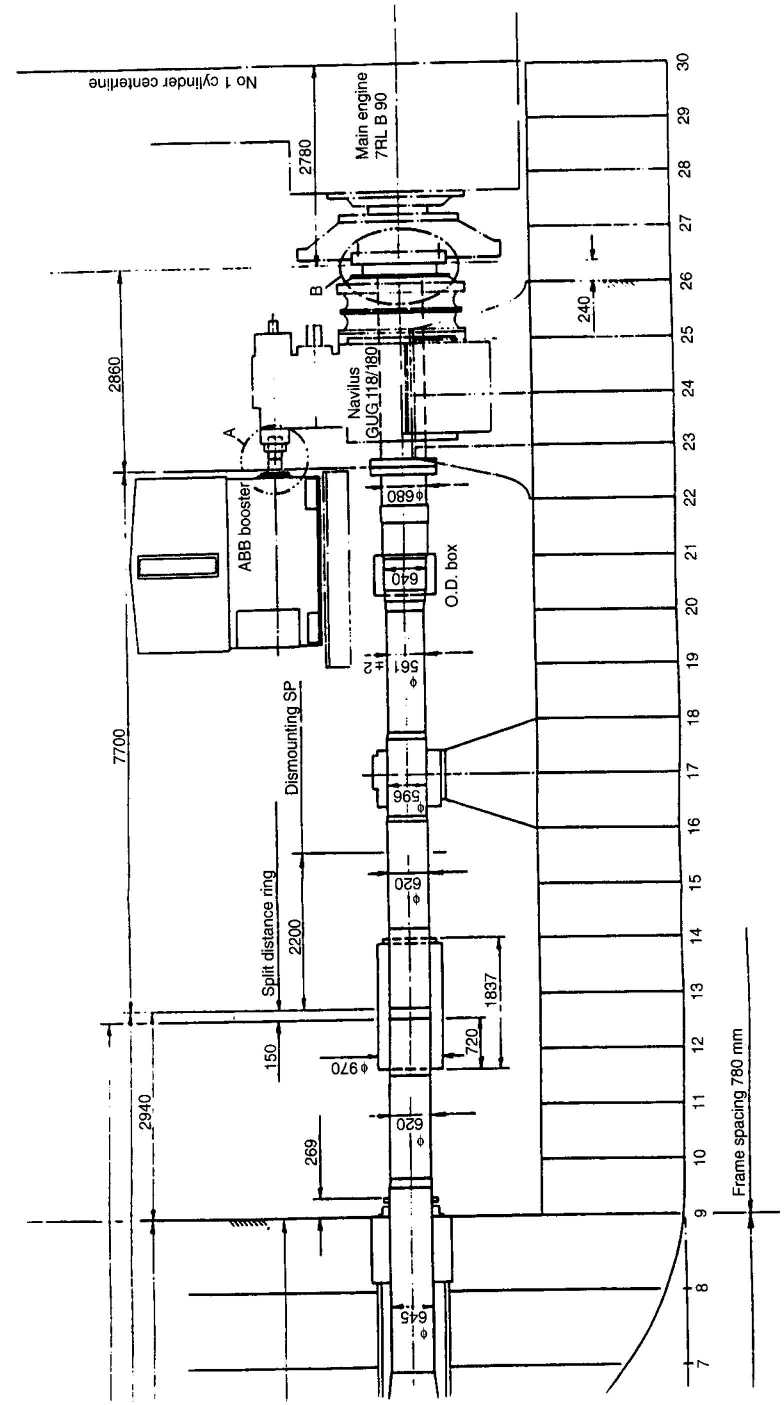


Figure 9-40. Shaft arrangement for booster motor and tunnel gear. Courtesy Blohm+Voss AG.

REVIEW

1. What components would you expect to be included in a main propulsion transmission system for a steam turbine? Gas turbine? Slow-speed diesel?
2. Name several of the ways that main propulsion gears may be classified.
3. Does a small diametral pitch indicate large or small teeth for a gear?
4. What is backlash in a reduction gear? How would you measure backlash?
5. What is the pitch diameter of a gear?
6. What is the difference between the face width and the effective face width of a gear?
7. What is the gear ratio? How do you determine a gear ratio?
8. What is the difference between an axial pitch and a base pitch of a gear?
9. Given a normal diametral pitch, how do you calculate the transverse diametral pitch of a gear?
10. What is conjugate action?
11. What is a pressure angle? How many pressure angles exist?
12. Define a helix angle. Where is it measured?
13. What are the three principal stresses considered in the design of a gear?
14. What are the loading indexes used to specify gears and to describe their loading?
15. How do you calculate K factor for a gear?
16. What is the difference between a dual torque path gear and a single torque path gear? If they transmit the same torque, what is the approximate relationship of their face widths?
17. Is a profile variation tolerance of .0004 inch reasonable for a quality gear used in ship propulsion?
18. What processes are used to form the teeth of high accuracy gears used for ship propulsion?
19. What are the functions of a gear lubricating oil system?
20. Describe a spring bearing.
21. Describe the red and blue contact pattern of a first reduction gear that has had helix angle modifications to accommodate primarily torsional bending.
22. What is a contact crossover? What may be measured at the contact crossovers?
23. What forces must be resolved to determine the bearing reaction of a single torque path second reduction gear? Draw a sketch of a typical force polygon.
24. Describe four typical gear problems and the causes.

25. What stresses are considered in a line shaft design? In a tailshaft design?
26. What are line shaft bearing influence numbers? How would you modify the bearing reaction of a line shaft bearing? How would you measure line shaft bearing reactions aboard ship?
27. What are the advantages of using a booster motor and gear to upgrade the power of a slow-speed diesel propulsion system?

REFERENCES

American Gear Manufacturers Association (AGMA). 1988. *Gear Classification and Inspection Handbook.* ANSI/AGMA 2000-A88. Alexandria, Va.: AGMA.

---. April 1990. *Gear Nomenclature, Definitions of Terms With Symbols.* ANSI/AGMA 1012/F90. Alexandria, Va.: AGMA.

Dudley, Darle W. 1962. *Gear Handbook.* New York: McGraw-Hill.

Levinson, Irving J. 1978. *Machine Design.* Reston, Virgina: Reston Publishing Company.

Shigley, Joseph Edward, and Charles R. Mischke. 1977. *Mechanical Engineering Design,* 5th ed. New York: McGraw-Hill.

United States Navy. *Reduction Gears.* Washington, D.C.: U.S. Navy.

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CHAPTER 10

Heat Exchangers and Desalination

EVERETT C. HUNT

Since the first steamships were designed and constructed, there has always been a large quantity and variety of heat exchangers aboard ships irrespective of the type of main propulsion system. For example, the more familiar shipboard heat exchangers include main and auxiliary condensers, refrigeration condensers and evaporators (vol. 2 of *Modern Marine Engineer's Manual),* waste-heat boilers (vol. 2), HVAC heat exchangers (vol. 2), steam generators, superheaters and reheaters (chap. 5 of this volume), cargo oil heaters, fuel oil heaters, lube oil coolers, feedwater heaters, combustion air heaters, freshwater evaporators and condensers, engine jacket coolers, regenerators and recuperators, intercoolers, and tank heating coils.

This chapter will deal primarily with shell-and-tube heat exchangers, plate-type heat exchangers, and desalination equipment.

HEAT EXCHANGER PERFORMANCE

Evaluation for the selection of a heat exchanger design for shipboard application is based on four considerations: (1) heat transfer requirements, (2) pressure drop, (3) physical size, and (4) economics. The heat transfer requirements include temperature changes, quantity of heat transferred, and the specific heat of the fluids. The pressure drops of the fluids passing through the exchanger are a significant operating cost due to pumping power cost and maintenance cost for high velocity fluids. Higher velocity fluids increase the overall heat transfer coefficient, U, but also increase pumping and maintenance costs. The physical size is a cost in ship construction and an operating cost. Increased size of surface increases overall heat transfer rate and reduces pressure drop. The installed cost must be combined with operating cost to permit the selection of an optimum heat exchanger. Economics is a major consideration in both the design and proper operation of heat exchangers. Basic heat transfer relationships and the definition of overall heat transfer rate, U, are included in chapter 1 of this volume. Table 10-1 includes typical overall heat transfer rates for shipboard shell-and-tube heat exchanger and plate-type applications.

TABLE 10·1
Typical Overall Heat Transfer Rates

Heat exclianger application	*U, Btu / hr-ft³-F*
Shell-and-tube	
Feedwater heater	200 to 1,500
Refrigerant condenser	50 to 150
Feedwater economizer	5 to 10
Steam condenser	200 to 1,000
Lube oil cooler	20 to 60
Drain cooler	150 to 300
Plate-type	
Water heater	800 to 850
Oil cooler	100 to 110

Effectiveness

In making comparisons among heat exchangers for the purpose of selecting one, effectiveness is a useful approach. Effectiveness is defined as the actual heat transferred by a heat exchanger divided by the maximum possible heat transfer. The maximum possible heat transfer is determined by the side of the heat exchanger with the minimum thermal capacity, i.e., mass flow times the specific heat at constant pressure. Calculate the maximum possible heat transfer as follows:

$$q = (mc_{p\ \min})(t_{h1} - t_{c1})$$

For parallel-flow heat exchangers (defined in chapter 1) with minimum capacitance in the hot or cold fluids, effectiveness is as follows:

$$E_h = \frac{t_{h1} - t_{h2}}{t_{h1} - t_{c2}}$$

$$E_c = \frac{t_{c2} - t_{c1}}{t_{h1} - t_{c2}}$$

For counter-flow heat exchangers with minimum capacitance in the hot or cold fluids, effectiveness is as follows:

$$E_h = \frac{t_{h1} - t_{h2}}{t_{h1} - t_{c2}}$$

$$E_c = \frac{t_{c1} - t_{c2}}{t_{h1} - t_{c2}}$$

EXAMPLE 10-1: A lube oil cooler counter-flow heat exchanger cools oil from 150° to 110°F. The water is heated from 70° to 100°F. The minimum capacitance is on the hot fluid side. What is the effectiveness of the exchanger?

$$E_h = \frac{150 - 110}{150 - 100} = .80$$

Fouling Factors

After heat exchangers have been in service for a period of time, various coatings form on the heat transfer surface. The material of the coating depends on the exchanger application.

The coating adds resistance to the conduction heat transfer rate. Manufacturers recognize this decrease in the overall heat transfer rate, U, by including a fouling factor or resistance in the heat exchanger design calculations. The fouling factor, R, is based on the difference between the clean and dirty overall heat transfer rates. The factor is based on experience with particular exchanger applications. The factor is defined as

$$R = \frac{1}{U_{\text{dirty}}} - \frac{1}{U_{\text{clean}}}$$

Higher fouling factors suggest higher margins for in-service performance. Typical fouling factors are listed in table 10-2.

TABLE 10-2
Typical Shell-and-Tube Heat Exchanger Fouling Factors

Fluid	*R, hr-ft²-F / Btu*
Fuel oil	.005
Combustion air	.002
Lube oil	.001
Refrigerant liquid	.001
Seawater less than 125°F	.0005
Seawater greater than 125°F	.001
Boiler feedwater	.001
Steam	.0005
Engine jacket	.001
Engine exhaust gas	.010

A cleanliness factor is sometimes applied to the overall heat transfer coefficient instead of the fouling factor. For example, a shell-and-tube heat exchanger with freshwater on both sides of the tubes has a calculated U equal to 544 Btu/hr- ft^2- F. A cleanliness factor of .9 would reduce the overall U to 490 Btu/hr-ft^2-F. A cleanliness factor of .9 in this case is equivalent to a fouling factor of .0002. A typical cleanliness factor for shipboard main condensers is .85.

Figure 10-1 is a comparison of the absolute pressure versus seawater temperature for the design values and the clean condenser measured values for a cleanliness factor of .85. A clean condenser provides better than design performance since resistance to heat transfer due to fouling is not present.

In a steam power plant, maintenance of the design temperatures leaving all feedwater heaters is a significant contribution to the maintenance of design all-purpose fuel rate. For example, heat balance calculations for a typical four heater cycle steam power plant reveal the following losses in all-purpose fuel rate per 2 degrees Farenheit depression in output temperature:

Low-pressure heater	.04 to .08 percent
Direct contact heater	.09 percent
Third-stage heater	.05 to .06 percent
Fourth-stage heater	.02 to .03 percent

Fouling is the usual cause of reduced outlet temperature in shell-and-tube heaters. The subcooling of a direct contact heater is caused by the presence of air due to inadequate venting.

HEAT EXCHANGERS

Heat exchangers are characterized by different construction methods. The major types include plate-type, fin-fan, and shell-and-tube.

Plate Heat Exchangers

Plate heat exchangers are frequently selected for jacket water and lube oil cooling service on motor ships. They are also used for many other purposes, including saltwater evaporators. Most plate heat exchangers in marine service are designed for use at inlet pressures under 290 psig (2,000 kN/sq.m.) and at temperatures under 300°F (150°C).

Plate Heat Exchanger Construction

The arrangement of a plate heat exchanger consists of a series or stack of identical metal plates that are clamped together in a frame with heavy

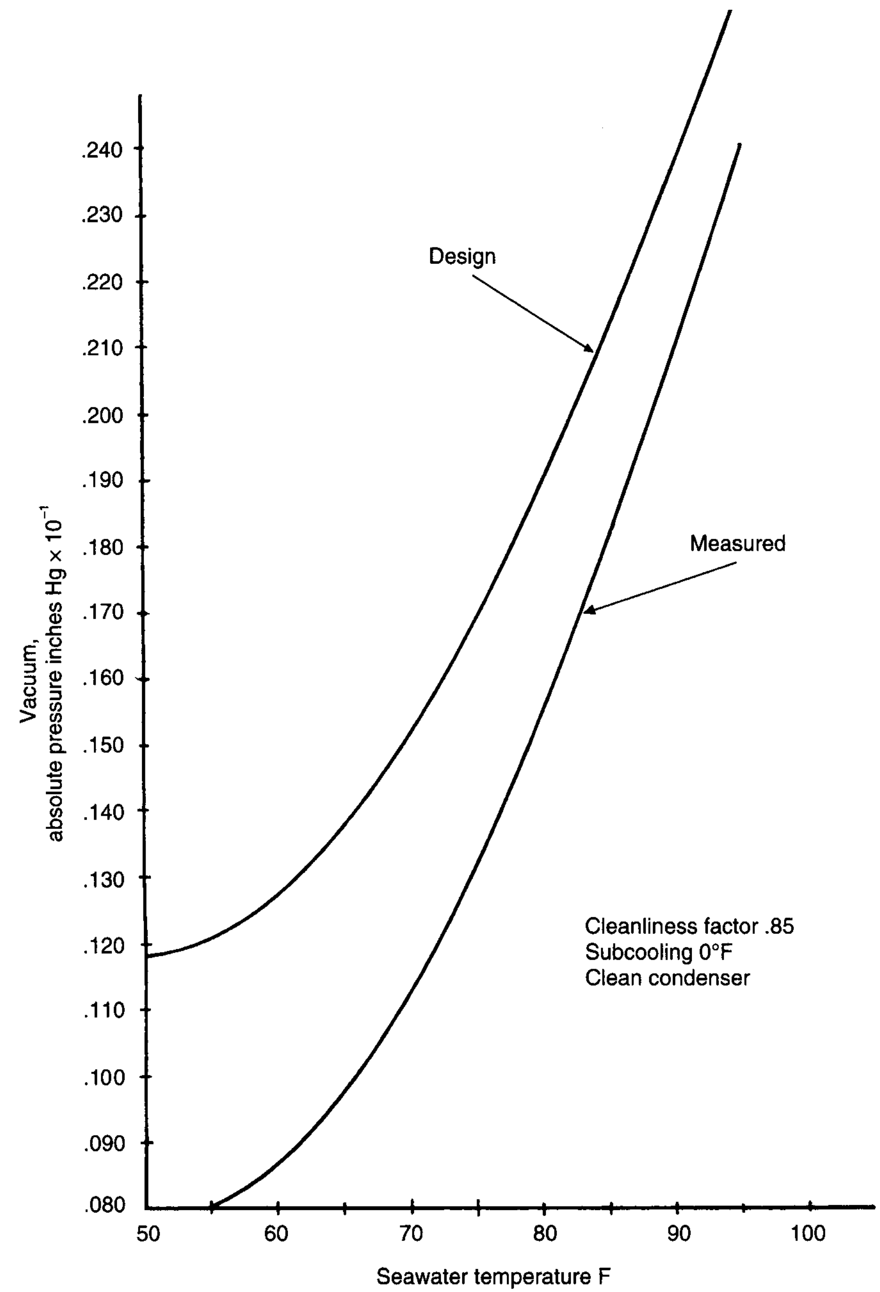

Figure 10-1. Typical main condenser: vacuum versus seawater temperature

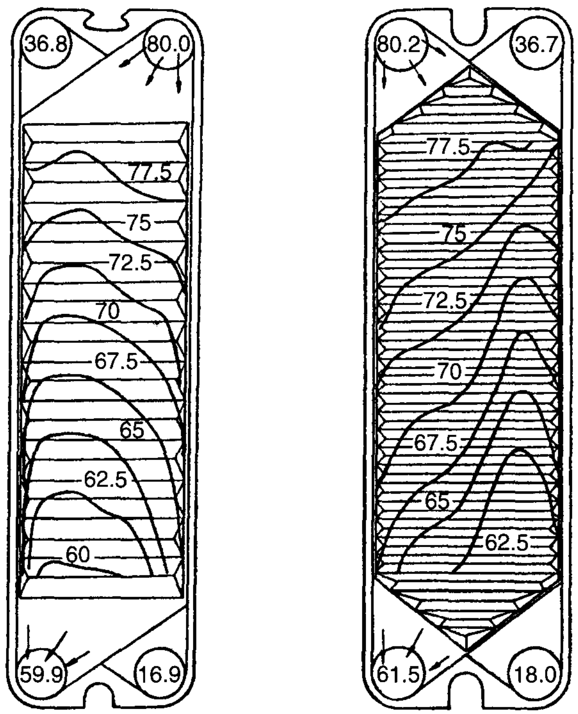

Figure 10-2. Temperature distribution on vertical-flow and diagonal-flow plates

plates at each end. The individual plates with thickness in the range of 0.6 to 1.0 mm (.0236 to .0394 inches) are corrugated to provide turbulence in the fluid passing over the surface and to enhance the mechanical strength of the plates. There are about sixty different designs of plate corrugations in use by various manufacturers who employ standardized sizes and corrugations to meet the needs of a particular application. An elastomer seal is fitted around the periphery ofthe plate to prevent external leakage and to direct the two fluids into the narrow passages between the alternate pairs of heat transfer plates. A hole is located in each of the four corners of the plates. When the plates are assembled with the gaskets, the four holes line up to provide the inlet and outlet headers which, when connected to external piping at the end clamping plate, direct the fluids to the interplate spaces. The plate corner holes have a gasket completely around the hole when the header is not connected to an interplate flow passage.

Plate material for the service can be selected from any needed alloy such as Monel, stainless steel, nickel, etc. The gasket materials may be selected from a wide range of compounded rubber materials.

The gaskets on the headers may be arranged for parallel or counter flow. Counter-flow is the common arrangement since it leads to higher effectiveness for the exchanger. Flow may be arranged for fluid entry and exit on the same side ofthe plate, or the ports may be selected for flow entry and exit on opposite sides ofthe plates. Figure 10-2 compares the temperature profiles on the plates for vertical and diagonal flow arrangements of identical fluids operating at the same Reynolds number. Gaskets may be arranged for any number of passes over the plates, for example, single-pass, multipass with equal passes on each side, and multipass with unequal passes on each side. A two-pass/two-pass exchanger is a common arrangement. Multiple passes improve thermal performance but increase the pressure drop, causing a rise in operating costs. Figure 10-3 shows the plate arrangements for three different pass configurations.

Plate Heat Exchanger Fouling and Cleaning

The high induced turbulence of a plate heat exchanger minimizes the buildup of fouling and improves the effect of in-place cleaning methods when employed. If mechanical cleaning is required, the heat exchanger may be easily disassembled for mechanical cleaning ofthe plate surfaces.

Cooper et. al. ran a series of tests to compare plate-type and shell-and-tube heat exchanger fouling in a cooling tower water, i.e., freshwater,

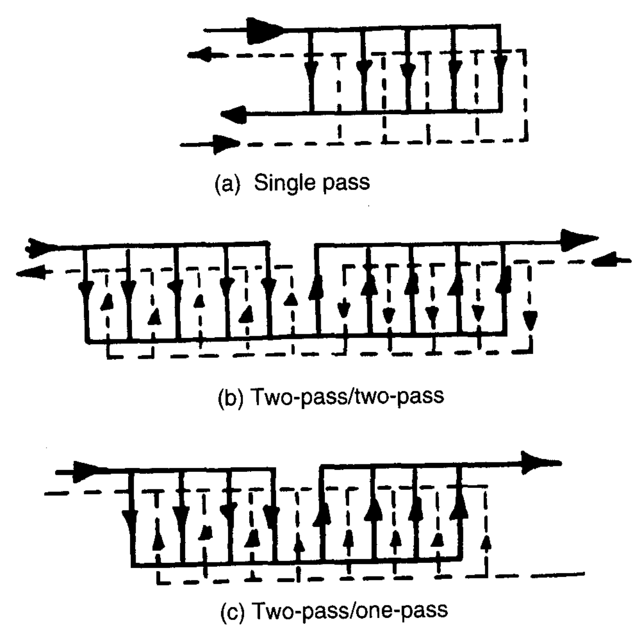

Figure 10-3. Typical plate heat exchanger flow arrangements

application. The tests showed that the resistance due to fouling in a plate exchanger was less than half the amount of a shell-and-tube exchanger. The tests were performed on a chevron trough surface which provides a higher turbulence than the intermating plates. For plate-type heat exchangers, the design fouling resistance is usually one-fifth of the values given in table 10-2 for shell-and-tube heat exchangers. A regular method of monitoring heat exchanger performance is desirable for all shipboard heat exchangers. One such method is discussed later in this chapter.

Although there are numerous types of corrugation designs, each with different thermal performance and pressure drop characteristics, they usually fall into two general categories: the intermating plate and chevron troughs. The intermating plate, a two-dimensional trough, has corrugations that are pressed to a depth greater than the compressed gasket depth. When the exchanger is assembled and clamped, the corrugations fit into one another to provide flow passages that are constantly changing in direction and area to induce turbulence. Liquid velocities vary from 0.2 to 3.0 m/sec (0.65 to 9.84 ftlsec) depending on the pressure drop acceptable. Chevron troughs, a three-dimensional design, have parallel corrugations that are pressed in the plate to the compressed gasket depth to provide a constant flow area with changing directions. Turbulence is provided by the changing direction of flow. Typical velocities are about 0.1 to 1.0 m/sec (0.328 to 3.281 ftlsec). Figure 10-4 shows sectional views perpendicular to the flow of the intermating and diagonal troughs.

Air to Liquid Heat Exchangers

Because of the low heat transfer properties of air, air-to-water coolers usually have extended heat transfer surfaces on the air side of the exchanger. The extended surface may be spiral fins brazed or shrunk-fit to the tubes, or plate-type fins through which a large number of tubes pass. The air side is typically single-pass while the tube side may be single or multipass.

The charge air cooler which permits significant increases in the specific output of a diesel engine is normally a rectangular heat exchanger located in the air intake trunk. The tubes have spiral fins on the outside to increase the surface. The tubes are grouped into headers on the water side so that they may be removed from the rectangular array for periodic cleaning of the air side. The water side is cleaned by removing the header cover plate for access to the tubes.

Large shipboard electric generators and motors are usually designed for closed circuit air cooling. A rectangular finned tube heat exchanger is installed in the air duct to remove the heat from the air. Since seawater is circulated through the tubes, it is necessary to design an exchanger that cannot leak saltwater into the electrical machinery. This is accomplished by employing a double tube, i.e., one tube inside the other, so that any leakage is drained away to an external telltale to alert the operators to the

problem. Such an arrangement has a poor heat transfer rate. However, a metal bridge between the inner and outer tube improves the transfer rate to an acceptable level. Figure 10-5 is a photograph of such a heat exchanger. Figure 10-6 illustrates the double tube arrangement showing the separate headers for inner and outer tubes and the mechanical heat conduction bridge between the inner and outer tube.

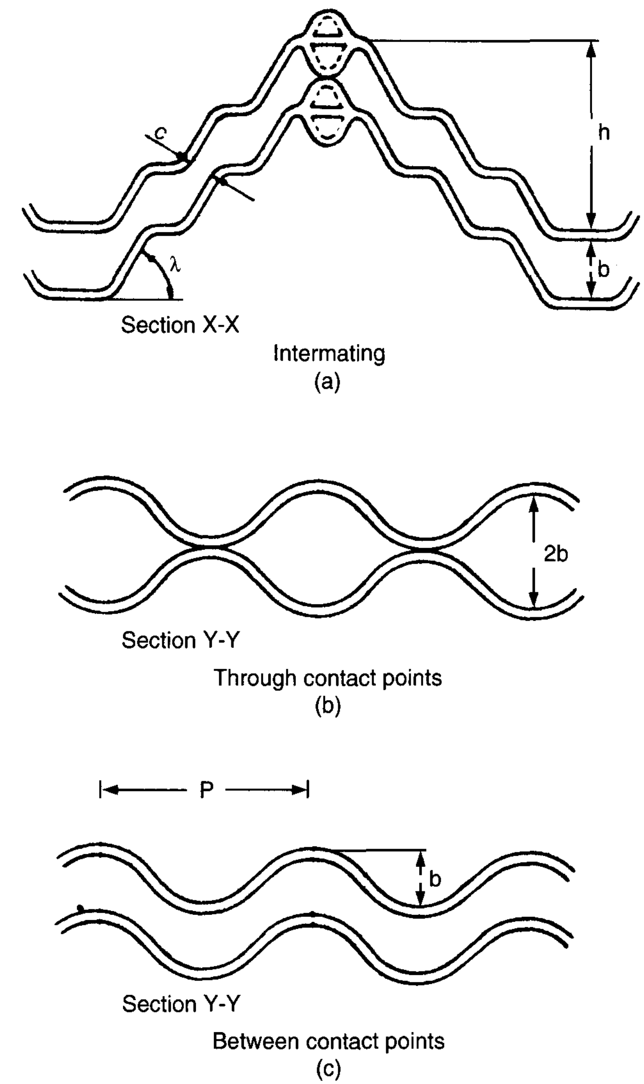

Figure 10-4. Plate heat exchanger sections through typical flow passages

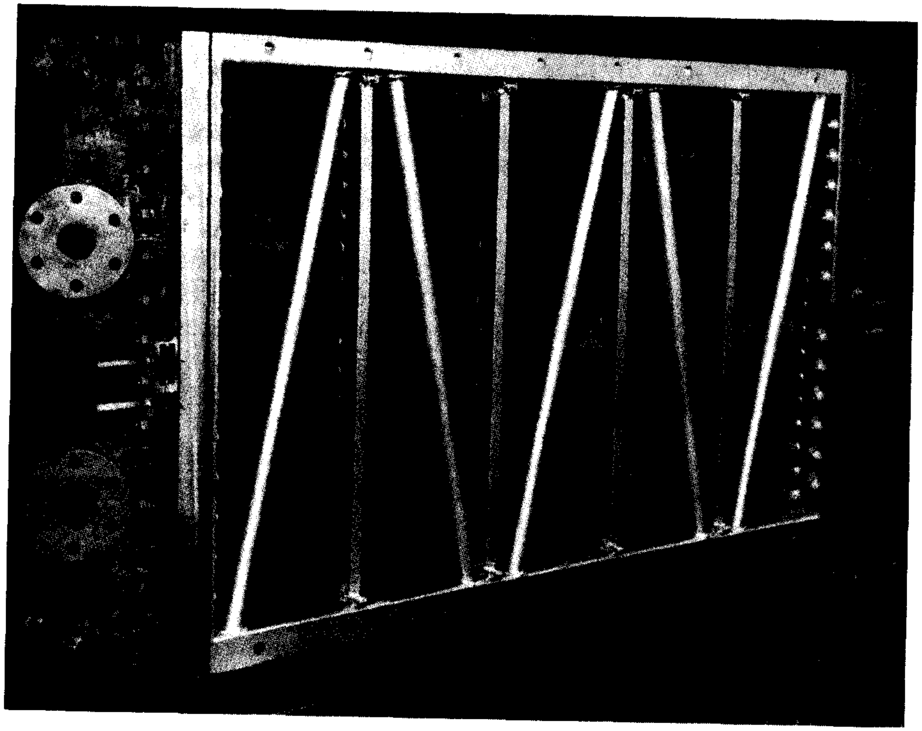

Figure 10-5. Double-tube heat exchanger for electric generator cooling

Shell-and-Tube Heat Exchangers

The shell-and-tube heat exchanger is the most common aboard ship. Shell-and-tube applications range from the very large main condenser on a steamship to a very small heat exchanger used to coolboiler water samples for testing. In general, the shell-and-tube heat exchanger consists of a bundle oftubes, each of which is expanded into a drilled tube sheet on each end ofthe tube bundle. The tube bundle is parallel to the axis ofthe shell. When pressure and temperature require it, the tubes may be brazed or welded to the tube sheets. The tube bundle is surrounded by a cylindrical or rectangular shell that has removable heads on each end for access to the tube sheets.

The tube-side flow may be single or multiple pass. The shell-side flow is usually single-pass but can be two-pass for special applications. For single-pass shell-side flow, baffies are provided frequently to distribute flow effectively over the heat transfer surface and to promote turbulence in the fluid for improved heat transfer rates.

Since the shell side and tube sides of an exchanger are at different temperatures, provision must be made in the exchanger design to accommodate the difference in thermal expansion between the tubes and the shell.

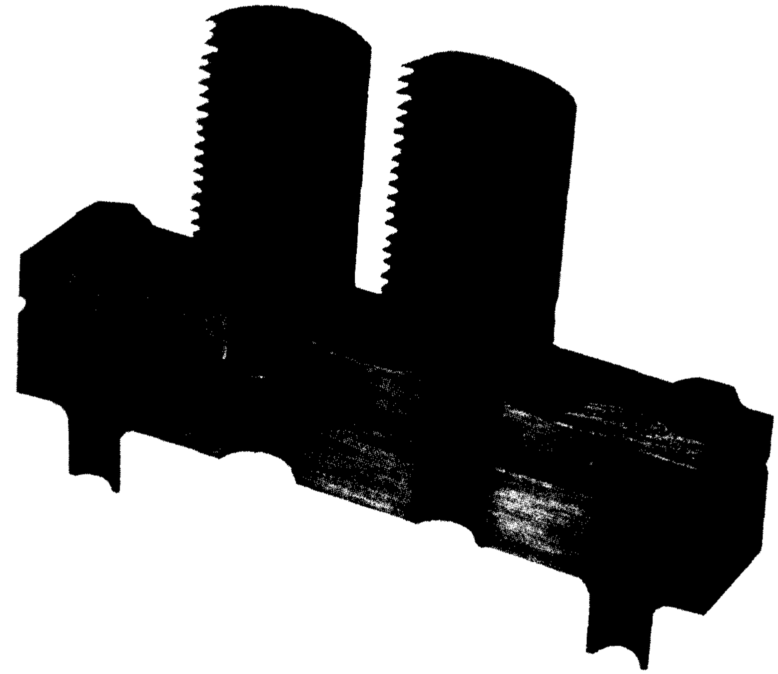

Figure 10-6. Header arrangement of double-tube heat exchanger for generator application

Depending on the amount of temperature difference, there are a variety of design approaches to accommodate the expansion difference.

The approaches include U-tube, a variety of floating heads, and an expansionjoint in the shell. Some shell-and-tube heat exchangers are especially arranged to permit effective cleaning ofthe tube side and/or the shell side.

Materials for shell-and-tube heat exchangers are selected for the service intended. Service conditions including temperature, pressure, corrosion, erosion, and material properties including galvanic action, heat conduction, welding, and machinability are considered in the selection of materials. When heat transfer effectiveness will be improved, tubes are sometime fitted with fins for extended surface inside or outside the tube.

Main Condensers

A modern steamship main condenser consists of a welded steel shell with a large number of copper-nickel tubes between tube sheets ofthe same material. Water boxes located at each end are usually cast iron or nodular iron. Zinc plates are fitted inside the water boxes to deter galvanic action.

The tubes are roller-expanded into both tube sheets. Differential expansion between the tubes and shell is accommodated by an expansion

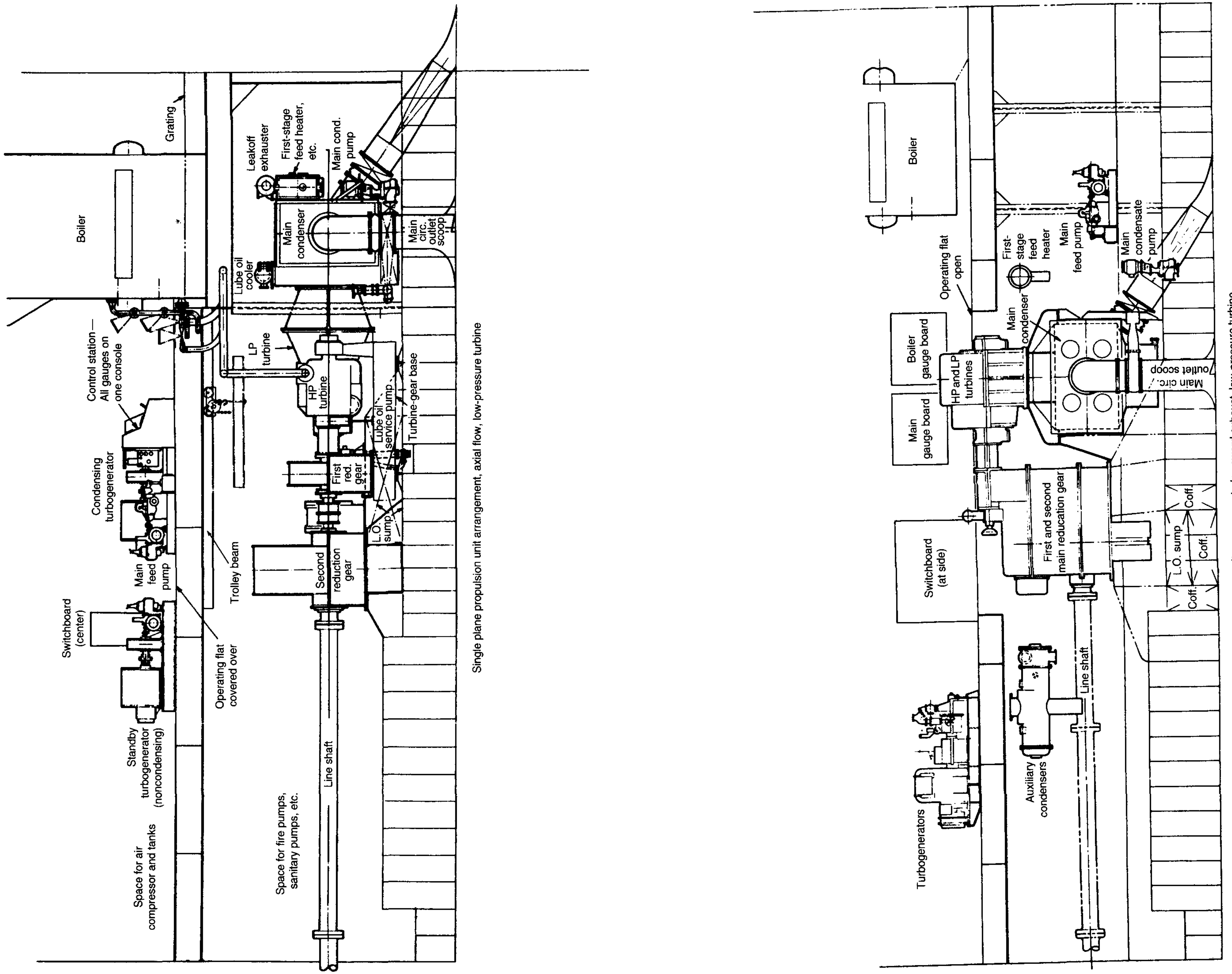

Figure 10-7. Comparison of engine arrangements for axial flow and downward exhaust to main condenser. Both condensers are one-pass scoop injection seawater cooling.

joint in the shell or by installing the tubes in a bowed configuration so that the tubes flex as differential expansion occurs. The tube matrix is arranged to provide lanes of decreasing flow area as the incoming steam is progressively condensed. A bypass steam lane is provided to deliver some steam to the condensate to avoid subcooling and consequent thermal losses. Tube support plates with holes drilled to accommodate the tubes are located at intervals along the axis of the tube. These support plates also determine the unsupported tube length for resonant vibration of the tube caused by the velocity force of the steam engaging the tube.

The water side is arranged for single-pass when scoop injection is used to provide the cooling water head. Most ships that rely on a circulating pump for seawater head are arranged for two passes on the water side. Three and four passes on the seawater side are possible but they are usually not economical.

Steamship propulsion turbines may be arranged for axial flow or downward flow of exhaust into the main condenser. Figure 10-7 (preceding pages) shows the comparison of two engine room elevation arrangements. The plants are designed for the same power and for scoop injection of cooling water. One of the plants has an axial exhaust from the low-pressure turbine to the condenser while the other has a downward exhaust.

Figure 10-8 shows the support arrangement used for a main condenser in a downward exhaust arrangement. Note that the main condenser is supported on the ship's foundation by a flange located near the top of the condenser. The hold-down bolts securing the condenser to the foundation are in oversize holes to permit condenser shell growth relative to the foundation. The condenser is held in position by fitted bolts located at the center point between the tube sheets.

Figure 10-9 shows the seawater inlet to a large single-flow main condenser for a modern LNG carrier. Table 10-3 contains the characteristics of a main propulsion steam condenser for a 26,000 shp normal propulsion system with initial turbine steam conditions of 850 psig, 950°F.

TABLE 10-3
Typical Main Condenser Characteristics for a
26,000 shp Steam Turbine Propulsion System

Surface	21,000 sq. ft.
Tube length	20 ft.
Number of passes	1
Water velocity	6 ft. /sec.
Condensing pressure @ 75°F water	1.5 inches Hg abs.
Cooling water flow	34,000 gpm
Main and auxiliary steam flow	143,000 lbs/hr
Heat rejection rate	1,985 Btu/lb of steam
Total heat rejection	135,000,000 Btu/hr

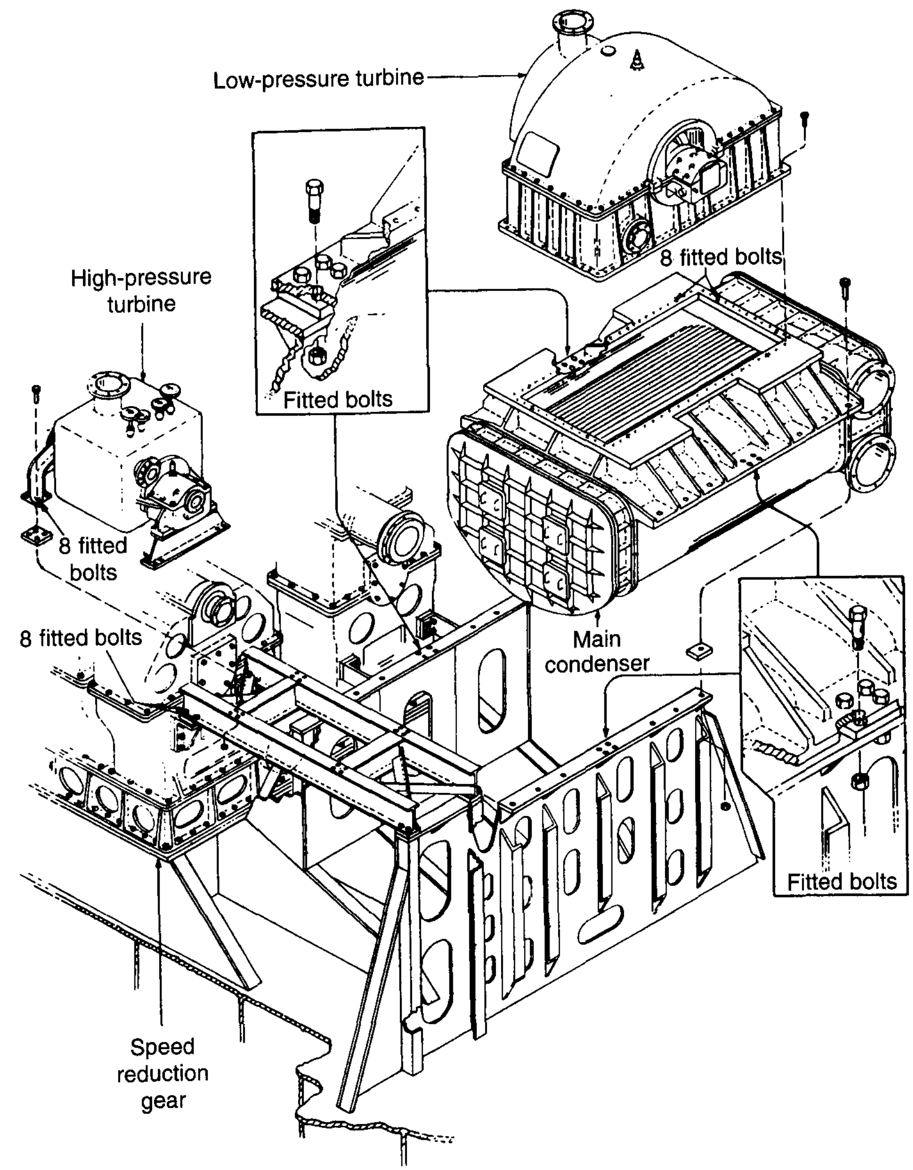

Figure 10-8. Support arrangement for main condenser in downward exhaust arrangement

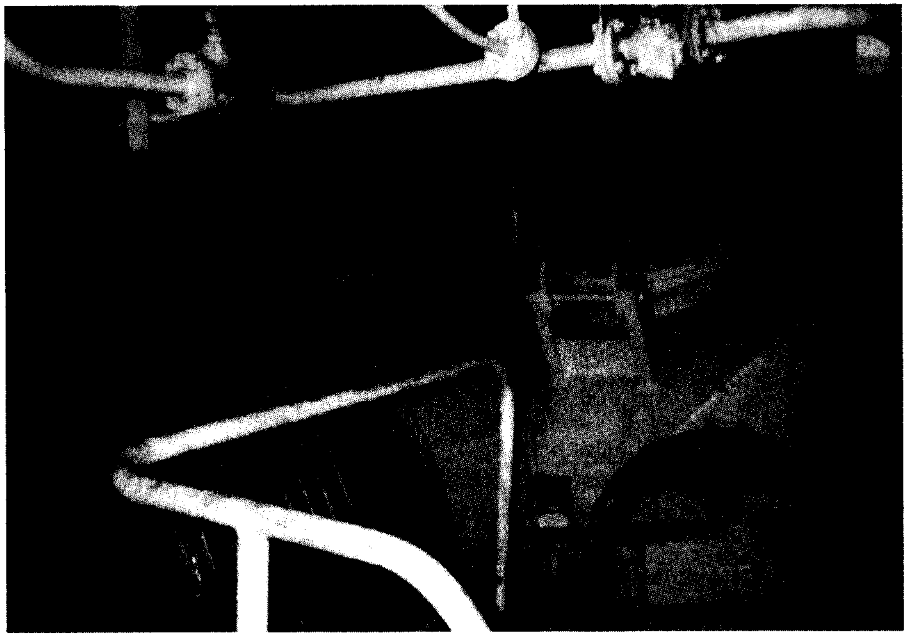

Figure 10-9. Single-flow main condenser seawater inlet. Condenser manufactured by Mitsubishi Heavy Industries, Ltd. Photo courtesy Hyundai Heavy Industries Co., Ltd., builder of LNG carrier in which condenser is installed.

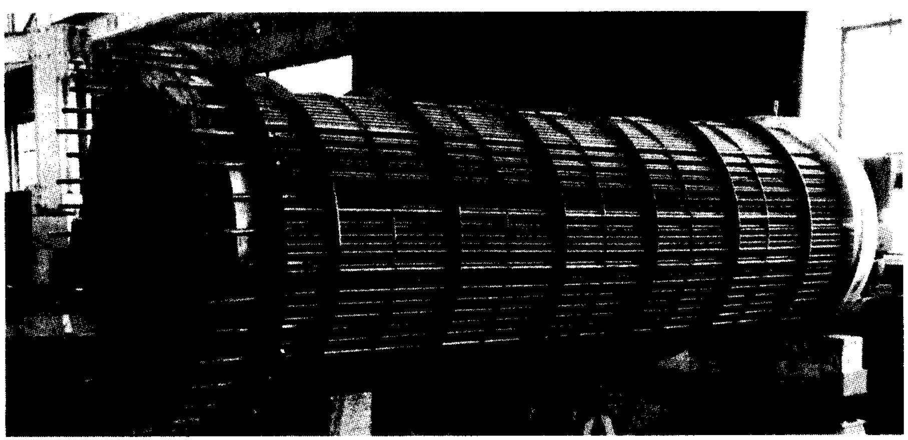

Figure 10-10. Lube oil cooler tube bundle showing disk and donut baffle arrangement

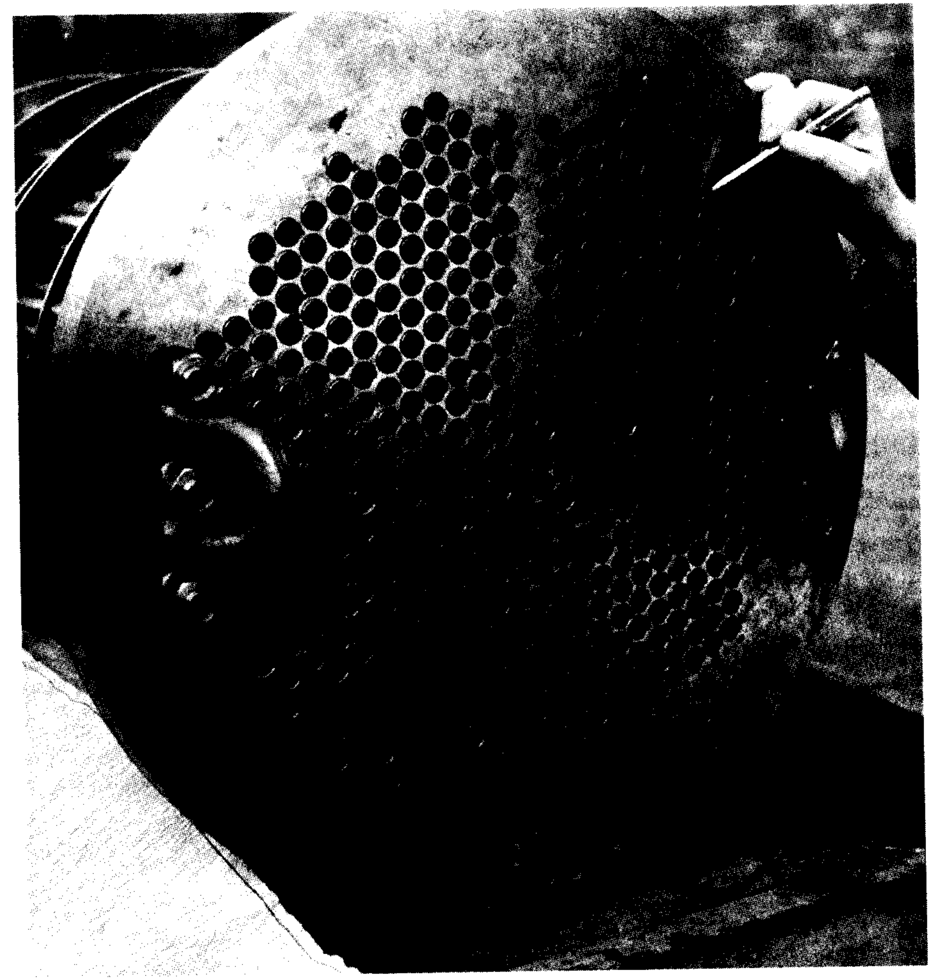

Figure 10-11. Tube sheet of lube oil cooler showing tubes expanded into drilled tube sheet

Lube Oil Coolers

Lube oil coolers are shell-and-tube coolers usually designed as one pass on the shell side and two or more passes on the tube side. The shell side is baffled, most likely with the alternating disk and donut baffle arrangement shown in figure 10-10. Figure 10-11 is a photograph ofthe tube sheet of a lube oil cooler showing the tubes roller expanded into the tube sheet. The number of passes is established by the arrangement of partitions in the heads at the end of each shell.

A typical lube oil cooler for a 26,000 shp steam propulsion plant has a seawater flow of about 500 gpm, an oil flow of 400 gpm, and a total heat transfer rate of 2,400,000 Btu/hr.

Feedwater Heaters

Shell-and-tube heat exchangers are used for regenerative heating in steam power plants. Since high-pressure feedwater heaters operate on superheated steam extracted from the high-pressure turbine, the shell side is usually divided into three sections: a desuperheating section, a baffied condensing section, and a drain cooling section. The tube side is a two-pass U-tube arrangement. The tube and shell materials are selected for the operating temperature and pressure of the heater.

Low-pressure feedwater heaters are also shell-and-tube type. Figure 10-12 is a photograph of a low-pressure feedwater heater installed in a modern steamship. Low-pressure feedwater heaters are usually designed for operation with steam extracted from the low-pressure turbine, or, when the ship is in the maneuvering mode, the heater may be supplied with auxiliary steam.

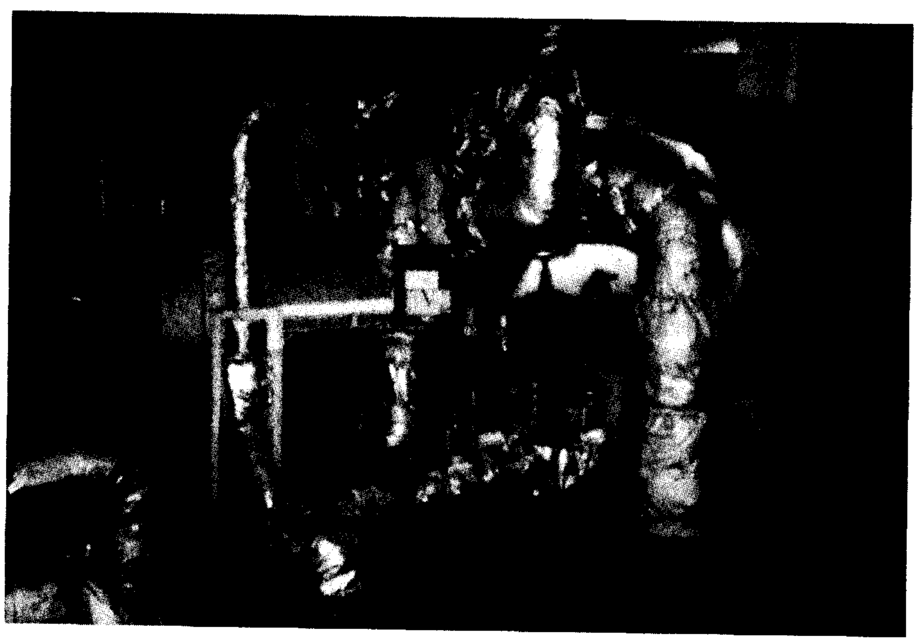

Figure 10-12. Low-pressure feedwater heater manufactured by Mitsubishi Heavy Industries, Ltd. Photo courtesy Hyundai Heavy Industries Co., Ltd., builder of LNG carrier in which feedwater heater is installed.

HEAT EXCHANGER MONITORING

Stoecker's work in the modeling of thermal systems for design and analysis suggests a tool for the continuous monitoring of heat exchanger performance that may be useful to the engineer officer.

Condenser or Evaporator

The relationships among cooling water inlet and outlet temperature and condensing temperature and inlet and outlet heating fluid temperature and evaporating temperature are shown in figure 10-13.

It can be shown that the outlet temperature in both cases is equal to

$$t_o = t_i + (t_c - t_i)\left(1 - e^{\frac{-UA}{wc_p}}\right)$$

where

t_o = temperature out, F
t_i = temperature in, F
t_c = temperature condensing or evaporating, F
U = overall heat transfer coefficient, Btu/ft^2-F-hr
A = heat transfer area, ft^2
w = fluid flow, lbm/hr
c_p = specific heat, Btu/lbm F
W = Btu/hr F

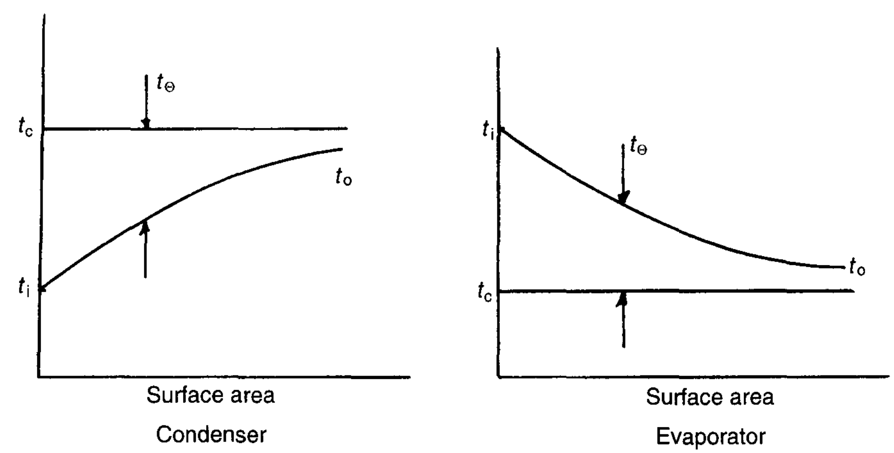

Figure 10-13. Evaporator and condenser temperature relationships

Rearranging the above equation so that all the constant terms for steady-state operation of a given condenser or evaporator are on the right side will yield

$$\frac{UA}{wc_p} = \frac{UA}{W} = \ln \frac{t_c - t_i}{t_c - t_o}$$

For a typical steam turbine condenser, the following temperatures might be observed

t_i = = 70°F
t_o = = 85°F
t_c = = 92°F

then

$$\frac{UA}{mc_p} = \ln \frac{92-70}{92-85} = \ln 3.1428 = 1.145$$

If condenser operation is considered a steady-state process at a specific normal full power operation, then it is possible to apply statistical process control to monitoring the condenser UA/mc_p. This value should remain constant for steady-state operation, and significant deviation from the original sea trial value (or a value determined after an overhaul period that included condenser cleaning) will indicate a deterioration in the overall heat transfer rate or a change in mc_p. Calibrated thermometers would be the best choice for collecting the data; however, there will still be errors due to data collecting anomalies, changes in c_p, variations in water flow, etc. A statistical approach will permit dealing with the errors while providing the criteria to determine that a change in UA/mc_p is significant, indicating a change in the process which should be investigated.

The procedure that will apply to any heat exchanger for which UA/mc_p has been defined in terms of the inlet and outlet temperatures is as follows:

1. For the new or recently cleaned heat exchanger, take twenty sequentially numbered samples of three observations per sample on the inlet and outlet temperatures and the condensing temperatures over a period of days at the same full power operating point. The twenty samples should be distributed randomly among the watches during the collection period. The three observations of a sample should be taken at random times during a single watch. Calculate UA/mc_p for each of the sixty observations.
2. For each of the twenty samples, calculate the mean and the standard deviation of the three observations as follows:

$$\bar{X}_i = \frac{\sum_{i=1}^{3} X_i}{3}$$

$$\sigma = \sqrt{\frac{X_1^2 + X_2^2 + X_2^3 - 3\bar{X}^2}{3}}$$

3. Plot the twenty means on a sequential chart shown in figure 10-14. This is called a control chart.
4. Calculate the expected value of the twenty means and standard deviations of the samples.

$$\bar{X}' = \sum_{i=1}^{20} X_i/20$$

$$\bar{\sigma} = \sigma_1 + \sigma_2 + \sigma_3 \ldots \sigma_{20}/20$$

5. Plot $\bar{X}'$ as the centerline on the control chart as shown in figure 10-14.
6. Plot the upper and lower control limits determined as follows on the control chart.

$$\text{UCL} = \bar{X}' + 2.394\,\bar{\sigma}$$
$$\text{LCL} = \bar{X}' - 2.394\,\bar{\sigma}$$
$$\text{or} \quad \bar{X}' \pm 2.394\,\bar{\sigma}$$

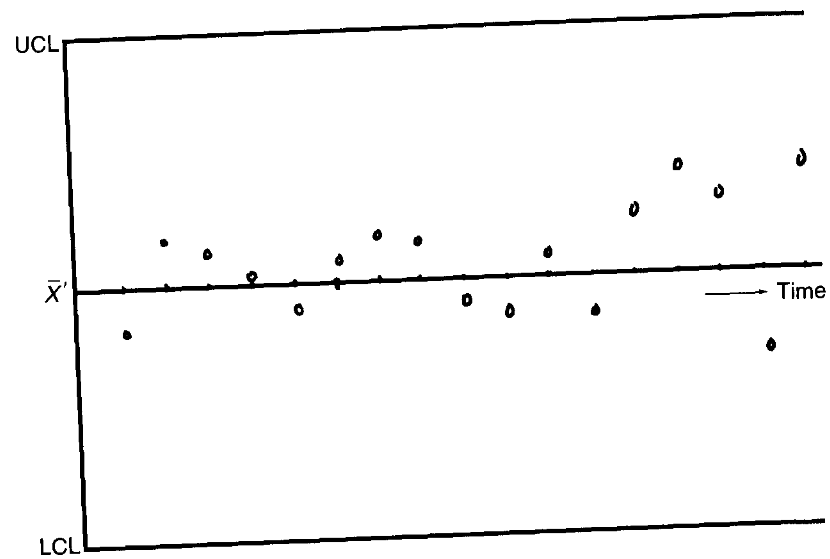

Figure 10-14. Statistical control chart

7. During subsequent operation at the same power level, periodically make three randomly spaced observations of temperatures and calculate Xi from the data. Continue to calculate and plot new points on the control chart using the periodic samples collected during randomly selected watches.

If the plotted points are randomly distributed about the line Xi and are within the control limit lines, then it should be assumed that the condenser operation has not changed. However, if several points fall outside the control limits or a series of points shows an increasing or decreasing pattern, the engineer officer should look for an assignable cause which may include the following:

- flow obstruction in tubes
- fouling of heat transfer surface inside or outside tubes
- changes in scoop or circulating pump operation
- air leakage

Counter-Flow Heat Exchangers

A similar approach may be taken to monitor a counter-flow heat exchanger such as a lube oil cooler for which the temperature characteristics of the fluids are shown in figure 10-15.

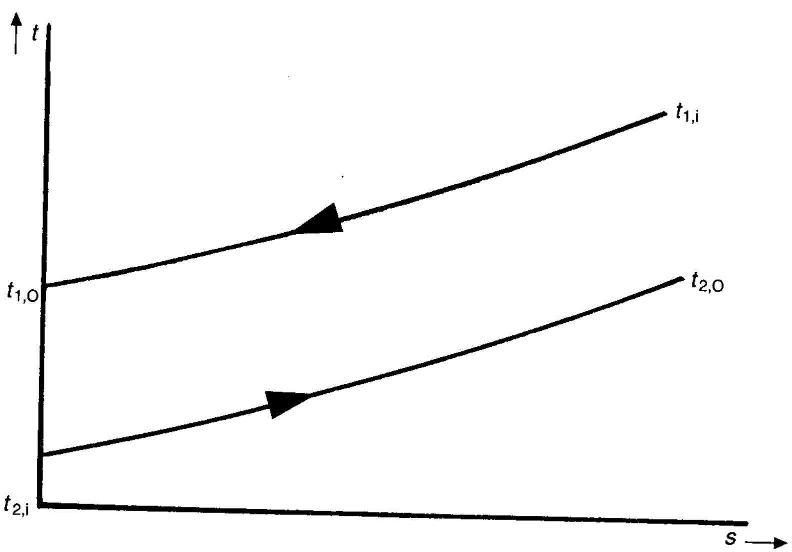

Figure 10-15. Counter-flow heat exchanger, one pass each side

For a counter-flow heat exchanger where $mc_{p_1} = mc_{p_2}$, use the following relationship to monitor performance:

$$\frac{UA}{mc_p} = \frac{t_{1,c} - t_{1,o}}{t_{1,o} - t_{2,i}}$$

If $mc_{p_1} = mc_{p_2}$, use the following relationship:

$$\frac{UA}{mc_{p_1} - mc_{p_2}} = \ln\left[\frac{t_{1,i} - t_{2,i} + \left(\frac{mc_{p_1}}{mc_{p_2}}\right)\left(t_{1,i} - t_{1,o}\right)}{t_{1,o} - t_{2,o}}\right]$$

EXAMPLE 10-2: A lube oil cooler has the following characteristics and operating temperatures for three observations during a watch when the main propulsion unit is operating at the normal full power condition adopted for monitoring purposes.

	tube side	*shell side*
flow lbm/hr	7,200	144,000
c_p Btu/lbm F	1.0	0.5

Therefore, $mc_{p1} = mc_{p2}$

1st observation

tin F	90	150
tout F	100	140

2nd observation

tin F	89	152.5
tout F	102	142

3rd observation

tin F	90.5	149
tout F	99	138.5

Since $W_1 = W_2$ then

1st $$\frac{UA}{mc_p} = \frac{150-140}{140-90} = .2$$

2nd $$\frac{UA}{mc_p} = \frac{152.5-142}{142-89} = .1981$$

3rd $$\frac{UA}{mc_p} = \frac{149-138.5}{138.5-90.5} = .2188$$

$$\overline{X}_i = \frac{.2+.1981+.2188}{3} = .2056$$

Values of Xi are entered on the control chart.

Air Leakage

Steam condensers and feed heaters operating below atmospheric pressure are subject to deterioration in performance when air leaks into the equipment. It may be difficult to determine if the poor performance is due to fouling or the air leakage.

If air leakage is suspected in a main condenser, a simple test may be run. With the condenser at full vacuum, secure the main air ejector or vacuum pump and observe the rate at which the vacuum decreases. A normal rate of decrease is one percent per minute or about three inches of Hg in ten minutes. If the rate is significantly higher, there is an air leak in the system. Check the turbine shaft sealing system, the flanges between condenser and turbine, the condensate pump seals, and the air ejector suction piping systems for leaks. If air leaks are suspected in the low-pressure heaters, check to ensure that the vents are not plugged.

HEAT EXCHANGER OPERATION

Regardless of the specific type of construction, all heat exchangers will provide the most satisfactory service when operated within the range of design specifications for temperature, pressure, and flow.

Shell-and-Tube Heat Exchangers

A key to successful operation of shell-and-tube heat exchangers is the avoidance of thermal shock, overpressure, and water hammer, all of which cause mechanical stresses which may exceed the design strength. If the heat exchanger has removable tube bundles usually found on U-tube or floating head or floating tube sheet construction, the heat exchanger

should be started by introducing the cold fluid first, then gradually introducing the hot fluid until full flow is established. The reverse procedure should be followed when shutting down the exchanger. If the heat exchanger has a nonremovable bundle, i.e., a fixed tube sheet, mechanical stresses can be reduced by gradually and simultaneously introducing the hot and cold fluids.

Vent connections should be opened before starting up a heat exchanger to remove air or gases. When both sides of the exchanger are filled, the manual vent valves should be closed. In a steam heated exchanger, the condensate should be drained during start-up and shutdown to avoid water hammer.

The bolting on gasketed or packed joints should be inspected when the exchanger is at operating temperature to ensure that leaks are not present. Bolting must be retightened if leaks are observed.

Heat exchangers are designed for specific pressures and temperatures on the tube and shell sides. Do not operate at temperature and pressure conditions in excess of the rated condition usually specified on the nameplate and in manufacturers instructions.

Corrosion of the shell and heads can be controlled if the heat exchanger is drained of all fluids when a prolonged shutdown is anticipated.

Tube bundles are subject to damage due to vibration caused by fluid flow pulsation and operation above design flow rates. Such operation should be avoided.

Plate Heat Exchangers

Plate heat exchangers are subject to damage and gasket failure when operated above design temperatures or pressures.

Avoid such circumstances by proper operation with calibrated gauges and thermometers.

HEAT EXCHANGER MAINTENANCE

Regular cleaning for heat exchangers subject to fouling is essential to continued design performance. Fuel oil heaters, lube oil coolers, cargo oil heaters, and heat exchangers with seawater cooling are examples of heaters subject to fouling.

Performance may be substantially reduced by a light sludge or scale coating on either side of the tubes. Monitoring of exchanger performance supplemented by periodic physical inspection is a useful approach to detecting a reduction in the effectiveness. In the extreme case, random plugging of a few tubes will cause temperature differentials among the tubes that may lead to mechanical damage. If zinc plates are installed in water headers, they should be regularly inspected and replaced as necessary.

Inspection and cleaning of the shell side on the tubes usually requires removal of the tube bundle, which, depending on the size of the heat exchanger, may be a major task requiring rigging and care not to support the bundle weight on individual tubes.

Removal of floating head tube bundles is accomplished by passing threaded rods through two of the tubes. A steel bearing plate separated from the tube bundle by a piece of soft board is installed at each end with the rods passing through. Forged steel eyebolts must be screwed into each bearing plate and used for pulling and handling the tube bundle.

Cleaning oftube side or shell side of a heat exchanger will depend on the fluids and the resulting deposits. Circulating a hot distillate oil will usually clean a fuel oil heater or lube oil cooler. Salt deposits that are not hard may be cleaned out by circulating hot freshwater. Commercial cleaning compounds such as Oakite, used in accordance with the manufacturers instruction, are useful for removing some stubborn deposits. Hard scales such as those found in saltwater evaporators are removed with a carefully controlled circulation of hydrochloric acid in a freshwater solution, i.e., one percent acid or .275 normal HCL followed by a flush with a mild alkaline solution. This activity should be accomplished in a shipyard to avoid damage to the material or injury to personnel.

To inspect the tube sheet for leaks, clean and dry the tube sheet and then pressure the shell side with freshwater. Careful observation of the tube sheet(s) will reveal leaking tubes that must be repaired using a suitable parallel roller tube expander. The usual rolling depth is the tube sheet thickness less,Ysinch. When reassembling a heat exchanger, use new gaskets on clean joints.

Saltwater Cooled Heat Exchangers

There are usually a number of heat exchangers aboard ship that are saltwater cooled, typically with the saltwater inside the tubes, for example, main condenser, turbine generator set condensers, and lube oil cooler. When a structure composed of dissimilar metals is immersed in saltwater, a galvanic cell is formed and an electric current flows through the electrolyte and the two metals. Depending on the materials used in the manufacture of the shells, tube sheets, and tubes, various levels of electrolytic activity are established that lead to rapid corrosion of the anode or positive terminal ofthe cell leading to deterioration of the heat exchanger parts. Since zinc is one of the most active metals in the galvanic action table, zinc plates installed in the saltwater inlet and outlet headers will have a flow of current from the plates to the adjacent metal parts. If the plates are maintained clean by regular inspection and cleaning, the zinc will be corroded by the current flow, thereby protecting the heat exchanger parts. A practical rule is to install a square foot of zinc surface for every 1,000 square feet of heat exchanger surface. Plates should be cleaned every eight weeks. The plates

should be securely bolted to the headers in a manner that does not increase the turbulence and therefore the pressure drop of the cooling water flow.

DESALINATION SYSTEMS

Diesel, gas turbine, and steam turbine propelled vessels all require substantial amounts offresh and distilled water for boiler makeup, freshwater cooling system makeup, potable water uses, flushing of onboard sewage systems, washing, etc.

The quantities vary from modest amounts satisfied by a single 8,000 gallons-per-day evaporator to the huge quantities required by a large passenger or cruise vessel with accommodations for thousands of passengers and crew.

Low-Pressure Evaporator Plants

During the period 1940 to 1960, the low-pressure steam distilling plant was installed on many commercial and naval vessels. These plants were supplied with auxiliary exhaust steam at 5 psig, 227°F. Three basic types of low-pressure distilling units were manufactured during that period: submerged tube units, flash-type units, and vertical basket units. This chapter will describe the most common, the submerged tube unit.

Submerged tube distilling units were designed and built in capacity ranges of 4,000 to 50,000 gallons per day. The designs were single, double, or triple effect. An effect is a stage ofthe distillation process. A single effect uses the low-pressure steam supply in the submerged tube bundle to evaporate seawater. In a double-effect unit, the steam generated in the first effect is used in the second effect to generate lower pressure steam from sea water. If a third effect is added, the steam from the second effect is used to evaporate seawater in the third effect. The multieffect system makes the most economical use of the energy in the supply steam.

Figure 10-16 is a schematic of a double-effect seawater distilling plant manufactured by Griscom-Russell. This system consists of a single horizontal shell with a vertical longitudinal partition dividing the shell into the first-effect and second-effect evaporators. The first-effect shell contains a submerged evaporator tube bundle, a vapor separator, and a vapor feed heater. The second-effect shell contains the second-effect submerged tube bundle, a vapor separator, and the distilling condenser. A heat exchanger to cool the first-effect condensate is mounted remote from the main shell. An air ejector condenser is mounted on the top of the shell. The air ejector maintains a pressure of about 26 inches Hg vacuum in the second-effect shell from which it takes suction. A pressure of about 16 inches Hg vacuum is maintained in the first-effect shell via an orifice plate throttle in a connection line between the two shells.

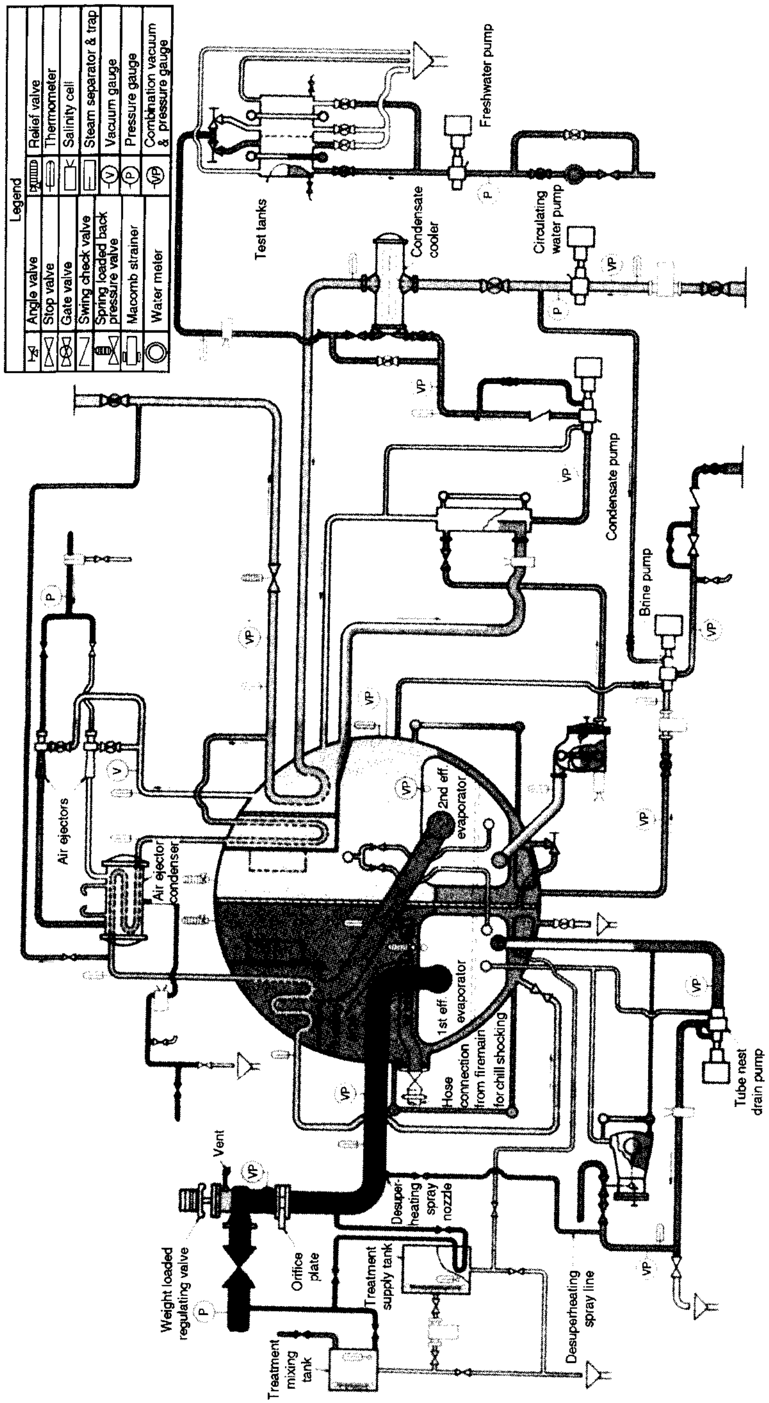

Figure 10-16. Double-effect low-pressure distilling plant. Courtesy Ecolaire Corp.

Steam is supplied to the first-effect tube bundle from auxiliary steam at a pressure less than 5 psig. Steam flow is controlled by a fixed orifice located in the inlet line. A desuperheating spray connects to the inlet steam line to the first effect to reduce the superheat that the supply steam gains during the inlet throttling process. The desuperheating spray water is taken from the discharge of the first-effect drain pump. Desuperheating is needed when the steam temperature exceeds a maximum of 240°F.

The desuperheated supply steam enters the first-effect tube bundle which is submerged in seawater. The seawater evaporates and the resulting vapor passes through a series ofbaffies and a vapor separator that removes any particles of seawater that are carried along with the vapor. The condensate that results from the steam giving up energy in the first-effect tube bundle is discharged to the low-pressure drain system by the first-effect drain pump. The first-effect vapor heats the incoming seawater supply to the first effect in a vapor heater located in the upper portion of the first-effect shell. The vapor then enters the second-effect tube bundle where it condenses, providing the latent heat energy to evaporate the seawater surrounding the tube bundle.

The vapor generated in the second effect passes through a baffie and separator system and into the distilling condenser, where it adds some heat to the incoming seawater in a feed heating section and rejects the remaining heat overboard with the circulating water in the condensing section. The condensate is pumped from the second effect to freshwater tanks through salinity test equipment by a condensate pump and followed by a freshwater pump. A brine pump removes high salinity seawater from the second-effect shell for discharge overboard. A circulating water pump provides cooling water for the distiller condenser and the first-effect seawater feed.

A triple-effect distilling system is similar to the double-effect system described above except there are three shells and three pressure levels for evaporation.

Shell Side Cleaning of Submerged-Tube Evaporators

In time, submerged-tube multieffect evaporator tube bundles become coated with scale. The development ofscale may be delayed by proper feed treatment with boiler compound in the first stage. When a scale forms on the tube bundles, it can be partially removed by chill shocking the tube bundle. Following manufacturers instructions, it is possible to heat the tube bundle without seawater in the shell and then to introduce cold seawater by spray or flooding so that the tubes receive a thermal shock that causes the hard scale to be shed from the tubes. Chemical cleaning by circulating a 1 percent solution ofHCL in distilled water is the most effective approach for periodic cleaning.

Evaporator Air Leakage Problems

Since low-pressure evaporators operate with most of the system at less than atmospheric pressure, it is important to maintain the system leak-free to obtain satisfactory performance. When air leaks are suspected, the entire system should be hydrostatically tested while ensuring that the test pressures stamped on the name plate are not exceeded. A hydrostatic test of the entire system including the first-effect steam chest may be made safely using the distilling condenser circulating pump.

Submerged-Tube Bundle Double-Effect Evaporator Operation

To start up and operate the submerged-tube bundle double-effect evaporator, use the following procedure, which is typical for low-pressure evaporators.

1. Familiarize yourself with the piping system, valves, instruments, and controls.
2. Open valves in drain lines to bilge from first-effect tube bundle and air ejector condenser.
3. Open valves in circulating water suction and discharge lines and start circulating pump.
4. Ensure that the spring-loaded back pressure valve in the circulating water line is providing a 5 psig pressure at the circulating water outlet connection on the distiller condenser.
5. Open all valves in the seawater feed system and fill evaporator shells to the top of the tubes. Open air vents to bleed air from feed system.
6. Secure feed valves to first effect and between effects.
7. Open valve in the overboard line at the feed outlet from the air ejector condenser.
8. Close vent line valves from first-effect evaporator head to the shell. Open steam chest vent valve on second effect.
9. Check that valves in condensate pump and brine pump discharge lines are closed.
10. Open necessary valves and start steam air ejector. Ensure that rated pressure is available to air ejector.
11. Check condensate drains from air ejector and close drain to bilge if water is pure. The evaporator shell should have a vacuum of 26 inches Hg. If vacuum is lower, look for leaks or air ejector problem.
12. Open the first-effect steam supply valve. Adjust steam control valve to provide 5 psig pressure ahead of the control orifice plate.
13. Start the first-effect tube nest drain pump.
14. When condensate appears in the second-effect drain regulator gauge, open the discharge valve to the flash chamber.
15. Start the condensate pump after opening suction, vent, and gland seal valves. Then open the pump discharge valve.

16. Open test tank valve to bilge.
17. Open valves in brine overboard line except pump discharge valve. Open vent valve in line from brine pump suction to second-effect shell. Start brine pump and open bypass valve in discharge line two turns.
18. Adjust seawater feed valve to maintain level in shells. Secure valve in emergency overboard line. Adjust valves in steam chest vent lines to one turn open.
19. Check salinity of water in test tanks and secure drain valves to bilge. Start freshwater pump.
20. Check salinity of brine in shells and adjust the brine pump to maintain design levels.
21. Normally the time to start up the evaporator unit is twenty minutes.

A typical shutdown procedure for a low-pressure evaporator is as follows:

1. Secure the steam supply to first effect.
2. Secure the first-effect drain pump.
3. Secure air ejector and close suction valve on air ejector.
4. Secure condensate pump.
5. Secure freshwater pump.
6. Open all vents on shell, steam chests, etc.
7. Continue to operate circulating pump and brine pump until the unit has cooled down.
8. Secure brine pump and brine overboard valve.
9. Secure circulating pump and seawater injection valve.
10. Close feed valve(s) to evaporator. Secure drains to the drain collecting system and open drains to bilge.
11. Secure the valves in vent lines between steam heads and shell.

Thin-Film Vertical-Tube Low-Pressure Evaporator

In the mid-1960s, a compact double-effect evaporator based on a thin-film heat transfer surface developed at General Electric Company Engineering Laboratory was made available for shipboard installation. This surface employs double-flute vertically oriented tubes to establish and maintain a thin liquid film on both the evaporating and condensing sides of the tubes. Figure 10-17 shows a section of the fluted tube and figure 10-18 illustrates the formation ofthe thin film on the heat transfer surface. On the outside of the tube, condensate is held in the grooves by surface tension. The liquid thin films reduce the conduction heat transfer resistance which permits a design with less surface leading to a compact design that requires less space, weight, steam consumption, and electric power compared to submerged-tube low-pressure evaporators. Figure 10-19 is a photograph of an assembled thin-film evaporator. These units were manufactured in capacity ranges of 12,000 to 20,000 gallons per day.

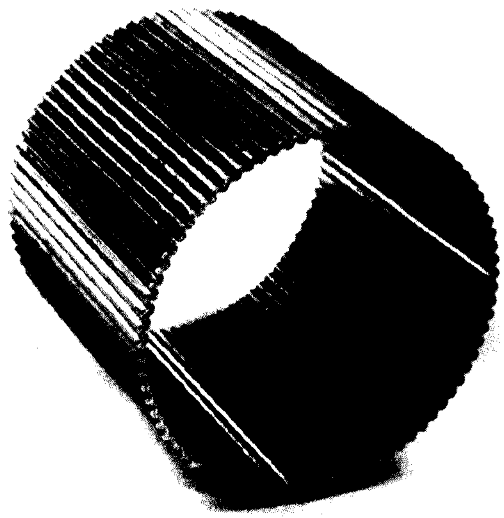

Figure 10-17. Heat transfer surface for thin-film heat transfer

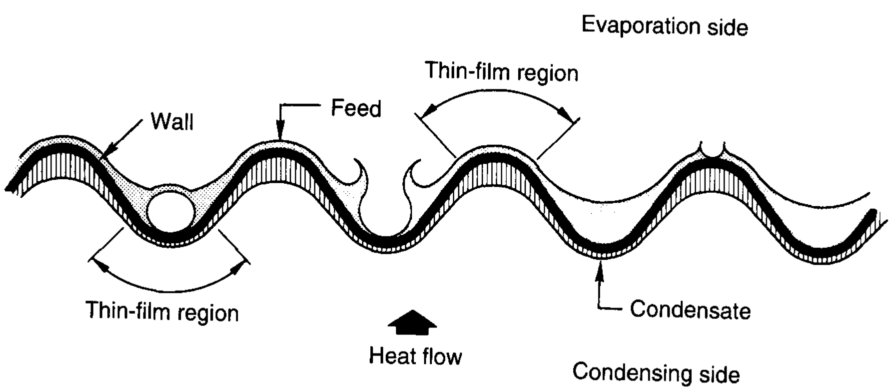

Figure 10-18. Thin-film formation on vertical double-flute tube

In the first effect, heated seawater is sprayed from nozzles into the vertical fluted tube as shown in figure 10-20. Gravitational forces distribute the seawater evenly over the inside surface, where a portion is evaporated and then passes upward through a steam separator and to the outside of the vertical tubes in the second effect. The unevaporated brine is collected at the bottom ofthe shell and discharged overboard by the brine overboard pump. The bundle of vertical tubes in the second effect is heated by the vapor from the first effect. The vapor condenses on the outside of the tubes and flows by gravity from the shell to the condensate well of the distiller

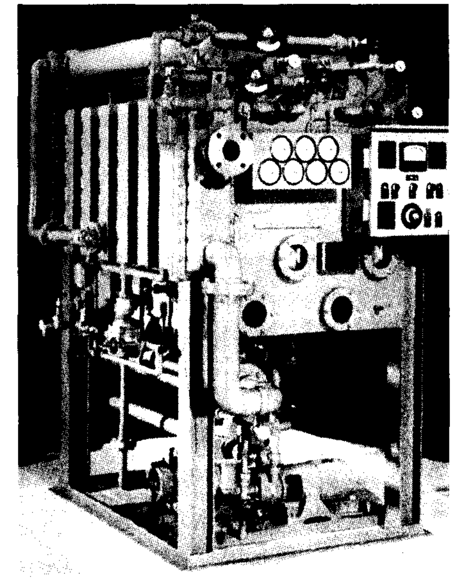

Figure 10-19. Thin-film evaporator

condenser, where it mixes with the condensed vapor from the second effect and is pumped by the distillate pump to the distillate cooler, salinity cell, and freshwater storage tanks. The condensate from the first effect is returned to the ship's low-pressure drain system. Incoming feed is heated in the distillate cooler, distiller condenser, and air ejector condenser. Figure 10-21 is a schematic of the thin-film evaporator system.

Successful operation of this system requires chemical treatment of the feed seawater. The starting and securing procedures are similar to a submerged tube low-pressure evaporator. These systems produce about 1.84 pounds of distillate per 1,000 Btu of energy at purity of JIs grains of salt per gallon.

Flash Evaporators

Like the submerged-tube and thin-film evaporators, the flash evaporator depends on a supply of low-pressure steam for the energy source. Such steam may be extracted from the propulsion steam turbine or taken from the auxiliary steam system and reduced to 5 psig and desuperheated to saturation temperature. The principle of a flash evaporator is the introduction of seawater heated to 170°F into a chamber that is at a reduced pressure of about 6.5 inches Hg absolute. Since the seawater is then at a temperature that is significantly above the saturation point, a portion of the seawater flashes into steam to restore equilibrium to the mixture in the chamber. Since most flash evaporators are two-stage, the remaining

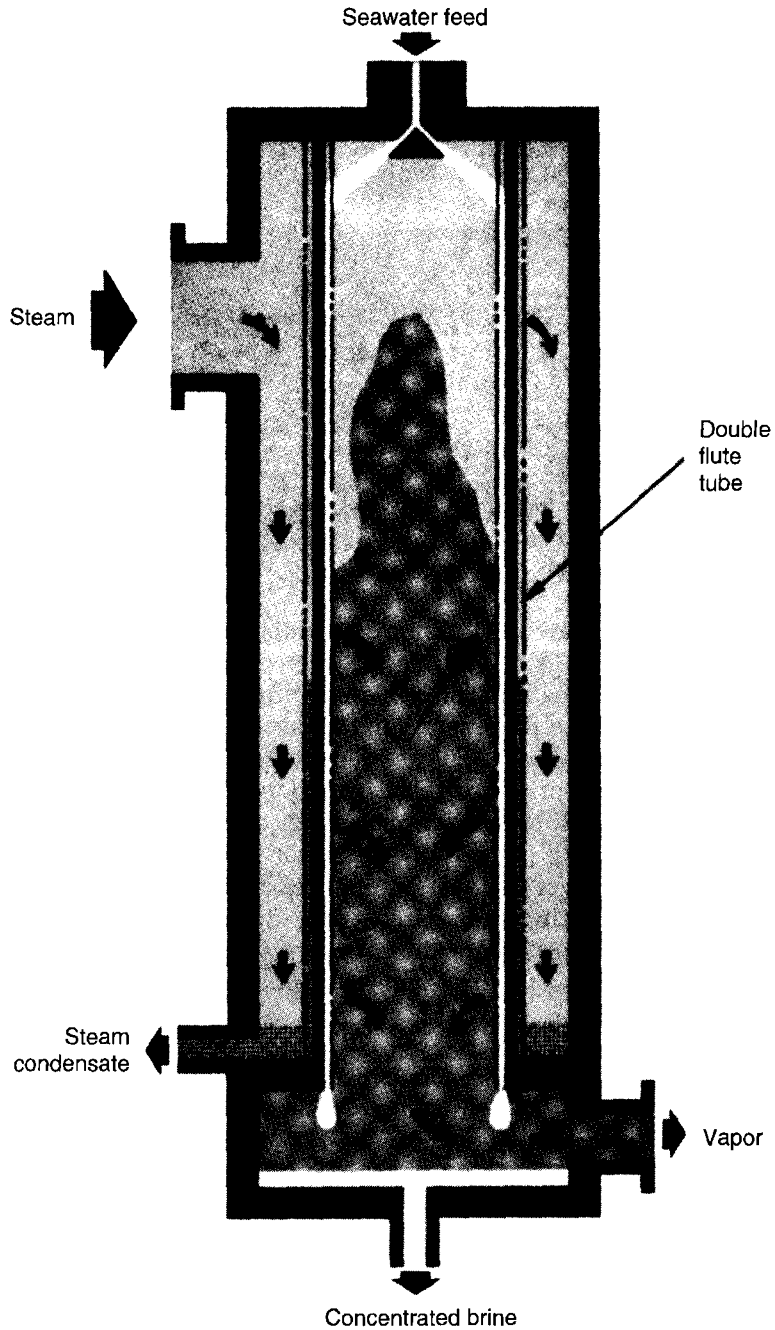

Figure 10-20. Vertical tube of thin-film evaporator

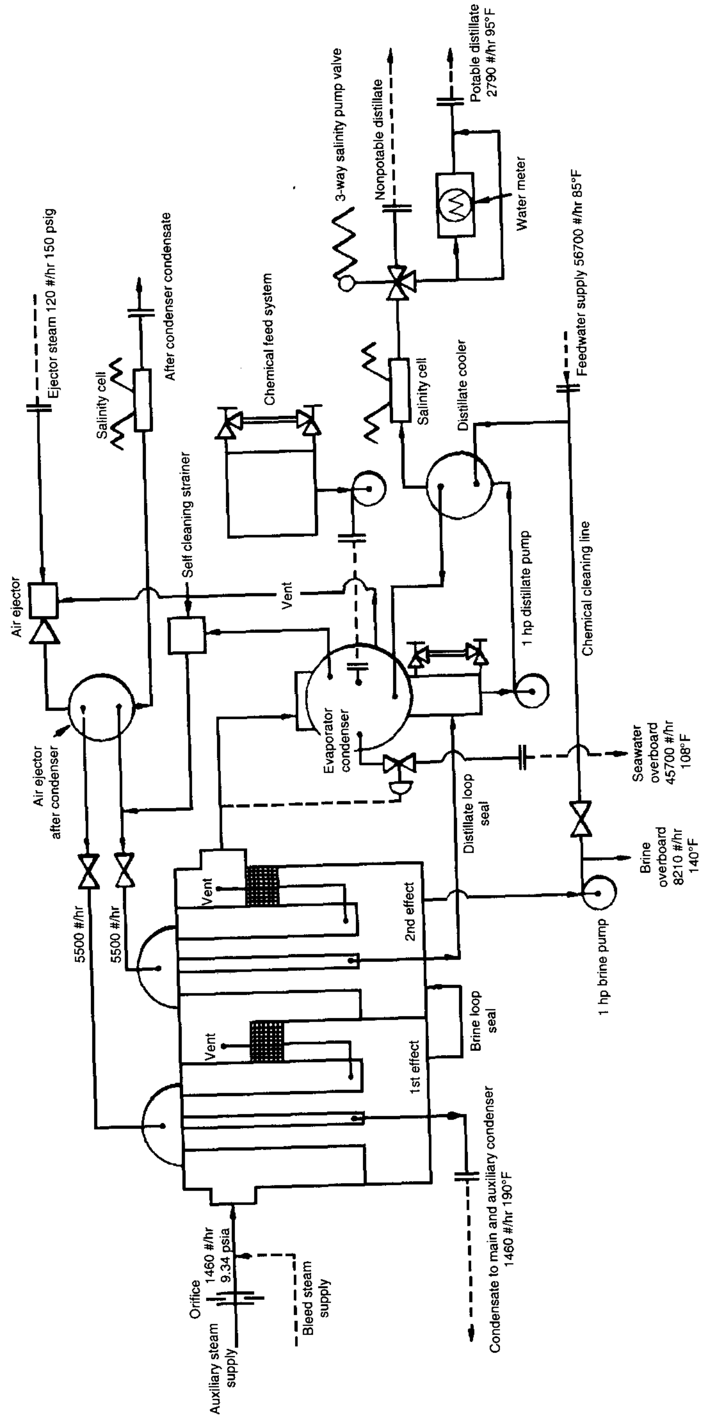

Figure 10-21. Schematic of thin-film evaporator system

seawater is introduced into a second chamber where the pressure is about 3 inches Hg lower than the first-stage chamber. Steam flashes from the brine in the second chamber to restore equilibrium.

The flash evaporator has the advantage that evaporation occurs without a heat transfer surface, which reduces the maintenance problems associated with hard scale formation that can occur when seawater is evaporated from a hot surface. The heat exchangers used to raise the level of the feed seawater temperature must have their tube sides cleaned periodically to maintain performance.

A typical flash evaporator has a saltwater feed to distillate ratio of 15 to 1. The energy consumption is .6781bs of steam at 5 psig for every lb of distillate produced, plus a 135 psig steam air ejector flow of 150 lbs/hr. The steam air ejector could be replaced by a vacuum pump.

Figure 10-22 is a schematic of a two-stage flash evaporator. The incoming feed is heated first in the distillate cooler (86°F), followed by the second-stage condenser (111°F), first-stage condenser (137°F), air ejector after condenser (138°F), and the saltwater heater (170°F). Brine from the first stage flows to the second stage through a loop seal. A brine overboard pump takes suction from the second stage. The distillate from the first stage flows to the second and the combined distillate is pumped to storage tanks via the distillate cooler and metering pump. The condensate from the seawater heater is returned to the ship's condensate system by a pump.

The start-up procedure for a flash evaporator is as follows:

1. Open valves in brine and feedwater lines.
2. Ensure that all distillate valves and steam supply valves are closed.
3. Start feedwater and brine pumps.
4. Open the saltwater heater vent valve one tur~ and open the after condenser vent valve fully.
5. Open stage one vent valve one turn.
6. Start the steam air ejector.
7. When the evaporator pressure reaches 27 inches Hg vacuum, admit steam to the seawater heater.
8. Adjust the steam inlet control valve to raise inlet seawater temperature to 170°F.
9. When condensate forms in glass on heater hot well, open valves and start condensate pump.
10. Control desuperheater spray in steam supply line to keep temperature 5°F above heater shell temperature.
11. When distillate shows in the glass, start the distillate pump, check salinity of output, and open discharge valve to storage tank.

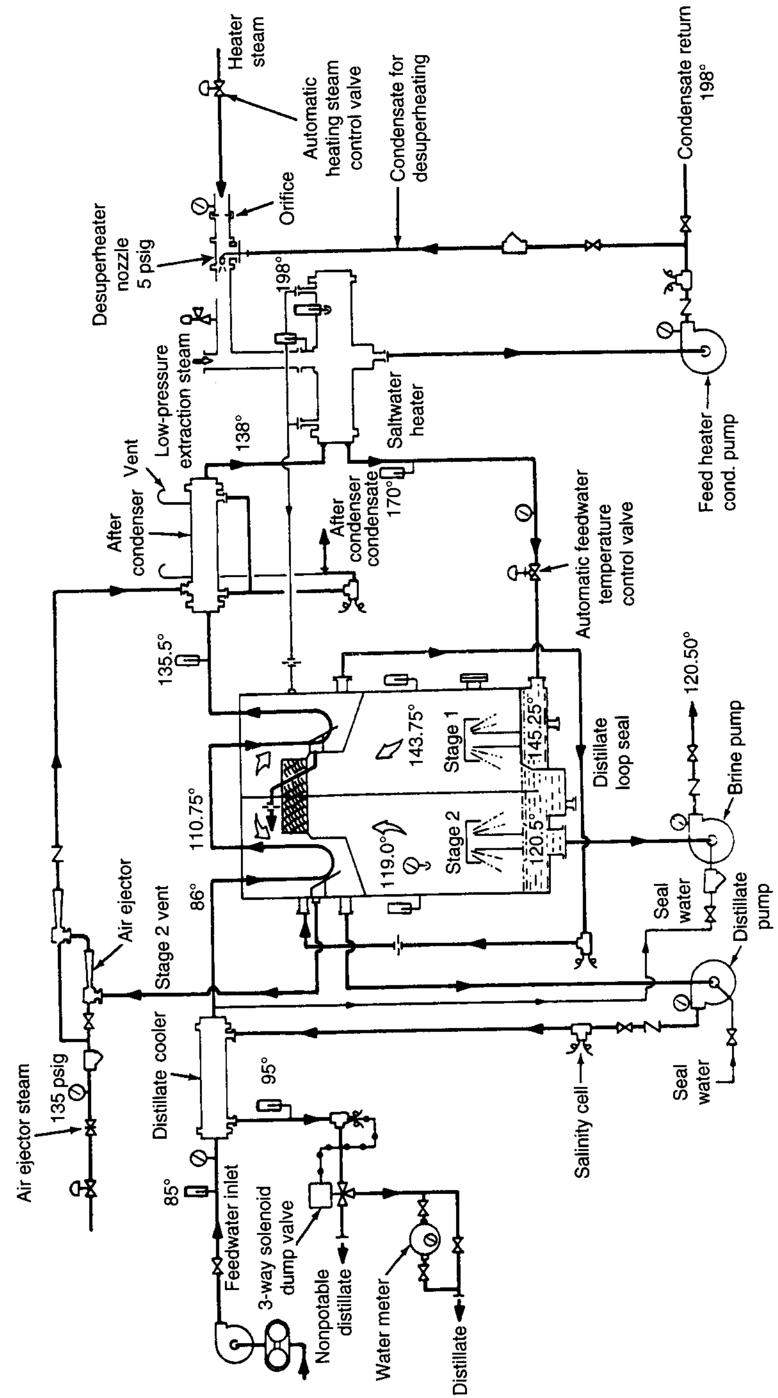

Figure 10-22. Schematic of two-stage flash evaporator

To secure the flash evaporator:

1. Secure steam supply line to saltwater heater.
2. Secure the steam ejector system.
3. Secure condensate pump, close pump discharge valve.
4. Secure the distillate pump and close discharge valve.
5. Secure feed and brine pumps and close discharge valves.
6. Drain system.

Plate-Type Evaporators

Figure 10-23 is the diagrammatic arrangement of a two-stage plate-type freshwater distiller. The plates are made of titanium and the other parts are made of materials that resist saltwater corrosion. The energy source is usually the main propulsion diesel engine jacket cooling water.

Referring to figure 10-23, the incoming seawater is preheated in the condenser (13) and then flows to the first stage (11), where a portion is evaporated at 158°F (70°C) or 9.2 inches Hg absolute. The vacuum is maintained in the first stage by a combined brine pump and ejector. The steam generated in the first stage passes through a demister (10) to the second-stage evaporator (12), when it gives up the latent heat to the feedwater from the first stage (2).

The distilled water condenses on one side of the plate and the brine evaporates on the other at 113°F (45°C) and 2.8 inches Hg absolute, which is maintained in the second stage by a combined brine pump/ejector.

The steam generated by boiling a portion of the brine in the second stage passes through a demister (10) to the condenser from which it flows to join the freshwater output from the second stage (7). The brine from the second stage (14) is pumped overboard.

Figure 10-24 is a photograph of a single-stage plate-type evaporator with the cover removed to show the two-plate heat exchangers, i.e., evaporator and condenser.

Plate-type evaporators with titanium plates that are in continuous operation must be disassembled twice per year for cleaning of the plate surfaces. Cleaning can be done by shipboard personnel in about three hours.

REVERSE OSMOSIS DESALINATION

Shipboard desalination plants usually depend on a large source of low-level energy such as extraction steam from a turbine or jacket cooling water from a diesel. When such energy is not readily available, as is the case with a gas turbine that is not fitted with a heat recovery steam generator, the reverse osmosis desalination plant is a useful solution to the freshwater supply problem.

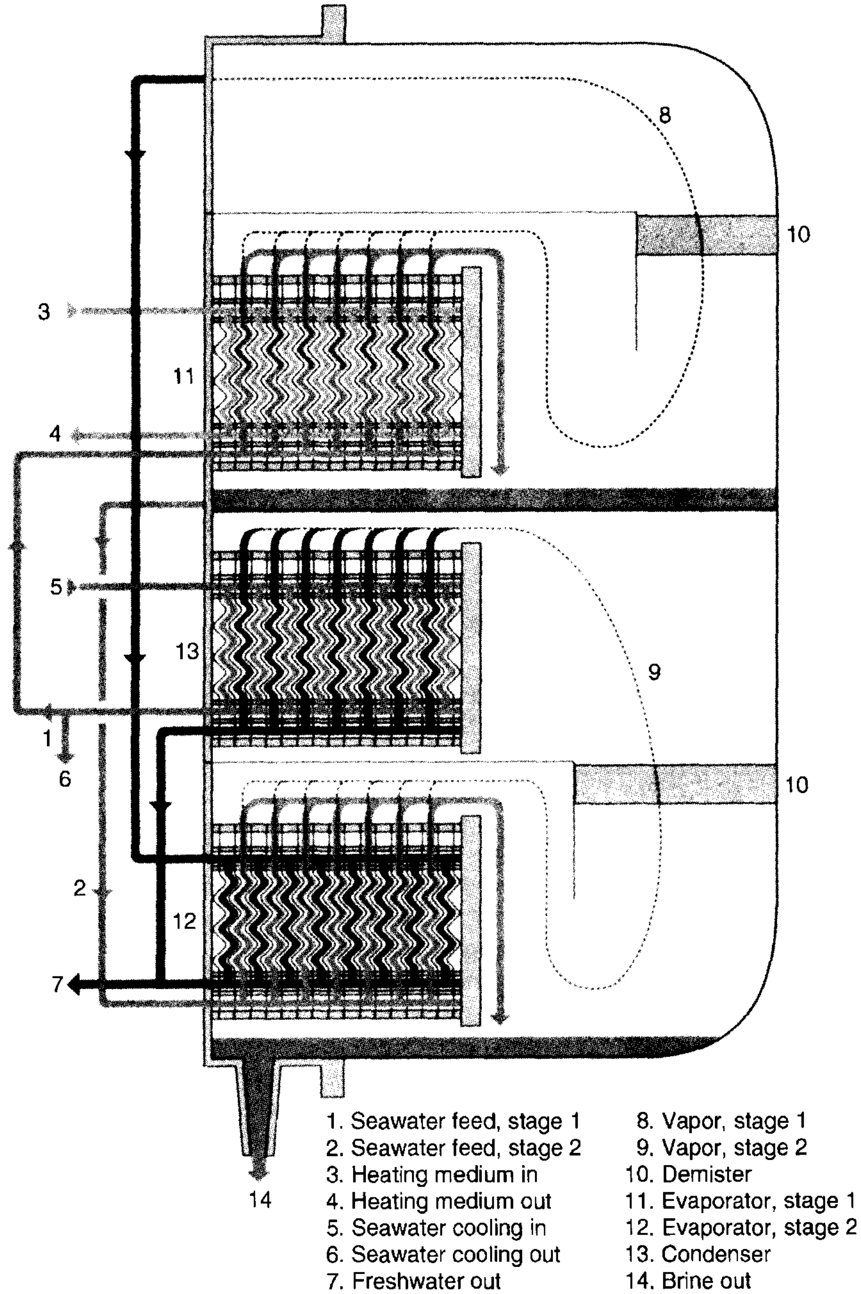

Figure 10-23. Two-stage plate-type freshwater distiller.
Courtesy Alfa Laval Separation Inc.

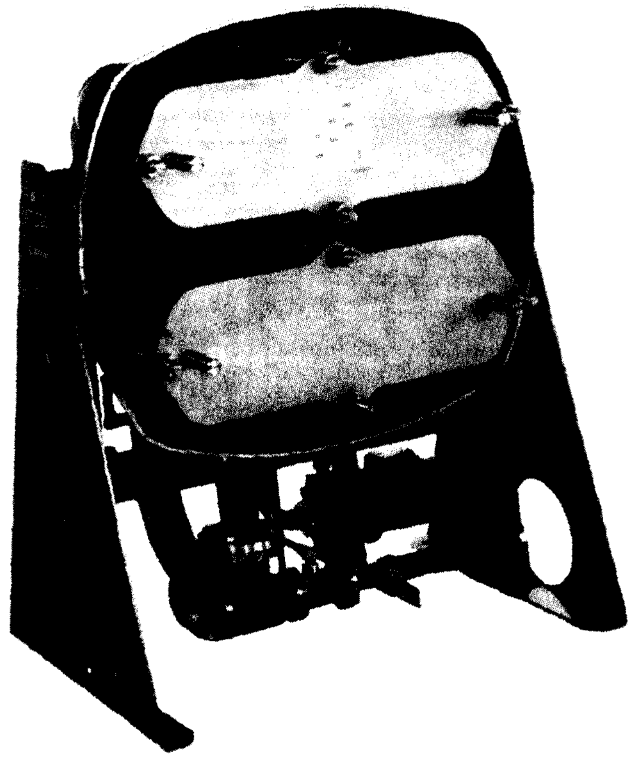

Figure 10-24. Single-stage plate-type evaporator.
Courtesy Alfa Laval Separation Inc.

Operating Principle

Osmosis is a naturally occurring phenomenon in which the less concentrated solution of two solvents separated by a semipermeable membrane will pass through the membrane to dilute the more concentrated solution. When the process reaches equilibrium, the concentrated solution side of the membrane will have a higher pressure called osmotic pressure. If this pressure, i.e., osmotic pressure, is applied to the concentrated solution side of the membrane, the process will reverse so the concentrated solution flows to the dilute side of the membrane. Reverse osmosis will cause the solvent, freshwater, to flow through the membrane, leaving a concentrated saltwater or brine on the other side. Figure 10-25 shows schematic diagrams of the osmosis and reverse osmosis processes. Osmosis occurs naturally in plant life.

In reverse osmosis desalination plants, the typical operating temperature of the seawater ranges from 60° to 80°F (*15.5° to 26.6°* C). The osmotic

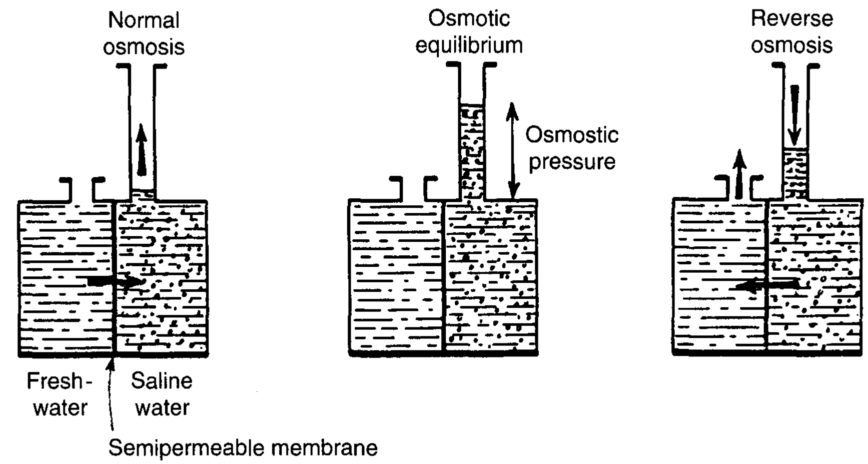

Figure 10-25. Osmosis process

pressure for a seawater solution is about 400 psi (2,750 kPa) but the actual operating pressure is in the range of 400 to 1,000 psi.

At constant pressure, the production rate for the reverse osmosis process varies linearly with the temperature of the concentrated solution. Typically, the production rate increases 100 percent from 40° to 80°F.

Reverse Osmosis Plant

There are several approaches to semipermeable membrane configuration for reverse osmosis including hollow fibers, tubular, and spiral-wound. This discussion will be limited to the spiral-wound arrangement that is the most popular for shipboard application.

Generally, reverse osmosis desalination plants consist of a seawater inlet, a feed pump, a series of two or more filters that remove suspended particles down to 3-micron size, and a high-pressure pump that delivers the filtered seawater to the reverse osmosis modules at a typical pressure of 1,500 psi. Brine is discharged overboard and freshwater is pumped to potable water tanks. Chemical treatment ofthe freshwater is usually provided to ensure that it is potable.

The reverse osmosis membrane, a semipermeable, polymeric material, is the main component ofthe system. This material is usually cellulose acetate or polyamide. The spiral-wound module shown in figure 10-26 consists of a permeate spacer placed between two sealed membrane sheets. This arrangement allows the permeate to flow to the permeate tube located at the center of the wound module. The other faces of the sealed membrane sheets are placed against a screen-like spacer which, when rolled up in the spiral-wound module, provides the path for the feedwater to flow to the membrane surface.

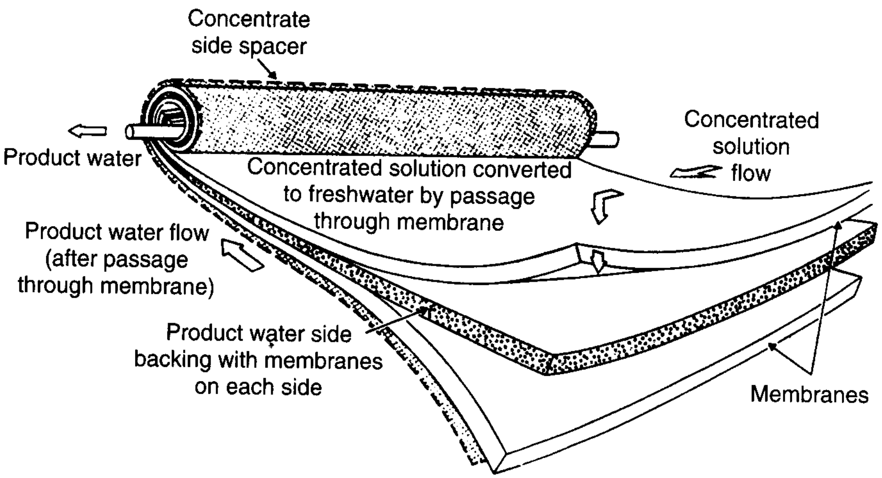

Figure 10-26. Spiral-wound membrane module

The high-pressure pump is typically a multicylinder plunger-type pump with a discharge pressure dampener, but multistage centrifugal pumps may be employed. The spiral-wound modules shown in figure 10-27 are usually about 6 inches in diameter and 40 inches long. The modules are arranged in parallel so that the desalination unit may continue in operation at reduced capacity if a module becomes inoperative due to fouling. To achieve high purity levels of the permeate, modules may be arranged in series. Salinity of the freshwater output is continuously monitored by a meter that controls a dump valve that operates if salinity exceeds 500 ppm.

Table 10-4 is a comparison of operating parameters between a typical 3,000 gpd evaporator distiller and a reverse osmosis plant.

TABLE 10-4
Comparison of Typical Evaporative Distiller and Reverse Osmosis for 3,000 Gallons-Per-Day

	Evaporative Distiller	*Reverse Osmosis*
Plant weight (lb)	950	610
Plant volume (cu ft)	35	40
Power (kW)	366	15
Water quality (ppm)	< 4	< 400
Availability	.98	.99
Start-up time (minutes)	30	5
Cleaning frequency (months)	20	none
Membrane replacement (months)	none	18

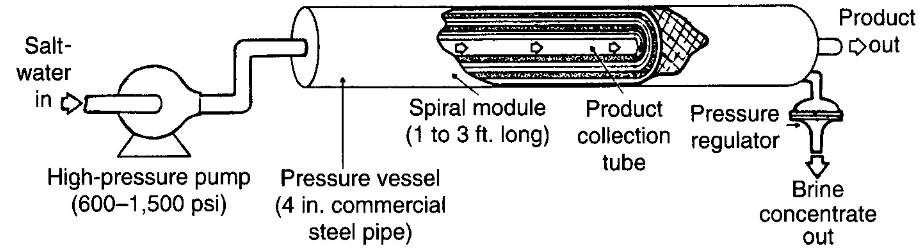

Figure 10-27. Schematic of spiral-wound module

Reverse Osmosis Plant Maintenance

When the plant is shut down, it must be flushed with freshwater and antibacterial material must be added to the system. This treatment is desirable even for relatively short shutdown periods. The recommended bacterial growth inhibitor is formaldehyde or ammonia blend solutions. Sodium bisulfate is sometimes recommended for this purpose. The manufacturer's instructions should be followed.

Manual cleaning and/or replacement ofthe cartridge filters must be accomplished at about 200 hours of operation. A pressure differential measurement across the filters will provide the indication of cleaning need.

Indications offouled membrane modules are salinity alarm, low freshwater production rate, increasing feed pressure, and high differential pressure across the modules. The modules may be isolated and replaced while the unit remains in service. The change time is about fifteen minutes. Modules may be cleaned using the detergents and procedure specified by the manufacturer.

REVIEW

1. List ten shipboard applications of heat exchangers.
2. What are the four considerations for the design of heat exchangers?
3. How does pressure drop through a high-pressure feed heater contribute to annual operating cost?
4. Define effectiveness of a heat exchanger.
5. Define fouling factors.
6. What is a typical fouling factor for a shell-and-tube fuel oil heater?
7. How does fouling of heat exchangers contribute to annual operating costs?
8. Describe a plate-type heat exchanger.
9. What are the typical upper pressure and temperature limits for a plate-type heat exchanger application?
10. Describe the typical surface on the air side of a charge air cooler.

11. What type of coolers are used for generator air cooling?
12. What is a multipass heat exchanger? What is the advantage and disadvantage of multipass arrangements?
13. What is the advantage of a scoop injection condenser?
14. Describe the control chart approach to heat exchanger monitoring. How would you incorporate this approach into a shipboard computer system?
15. If assigned to assess the condition of a shipboard shell-and-tube heat exchanger, what would you inspect? For a plate-type exchanger?
16. What is galvanic a£tion?
17. What is a multieffect evaporator? Describe the operation.
18. What is the start-up procedure of a submerged-tube multieffect evaporator?
19. Describe the thin-film vertical-tube evaporator. What is the advantage over the submerged-tube evaporator?
20. What is a flash evaporator?
21. Describe the operation of an evaporator with plate-type heat exchangers.
22. What is the total heat transferred in a typical 26,000 shp power plant main condenser?
23. What is the operating principle of a reverse osmosis plant?
24. What are the disadvantages and advantages of a reverse osmosis plant compared to a flash evaporator?

REFERENCES

Adamson, Wayne L., et. al. May 1984. "Experiences With Reverse Osmosis Desalination Aboard *USSFletcher." ASNE Journal.* Washington, D.C.: American Society of Naval Engineers (ASNE).

Bowker, Albert H., and Gerald J. Lieberman. 1959. *Engineering Statistics.* Englewood Cliffs, N.J.: Prentice Hall Inc.

Cooper, A., et al. 1978. "Cooling Water Fouling in Plate Heat Exchangers." Sixth international heat exchanger conference, Toronto.

Fraas, Arthur P., and M. Necati Ozisik. 1965. *Heat Exchanger Design.* New York: John Wiley & Sons.

Harrington, Roy L., ed. 1971. *Marine Engineering.* New York: Society of Naval Architects and Marine Engineers (SNAME).

Holman, J. P. 1972. *Heat Transfer,* 3rd ed. New York: McGraw-Hill.

Hunt, E. C., and T. N. Kelly. November 1952. "Evaporator Cleaning By Ship's Personnel." *ASNE Journal.* Washington, D.C.: ASNE.

Kern, D. Q. 1950. *Process Heat Transfer.* New York: McGraw Hill.

Okada, K., et aI., 1970. "Temperature Distribution of Fluids in Plate Type Heat Exchangers." *Kagaku Kogaku.*

Pizzino, Joseph F., et. al. May 1991. "Reverse Osmosis for Surface Ship Desalination-An Overview." *ASNE Journal.* Washington, D.C.: ASNE.

Smith, D. W. 1983. *Marine Auxiliary Machinery,* 6th ed. London: Butterworth & Co.

Stoecker, W. F. 1980. *Design of Thermal Systems,* 2nd ed. New York: McGraw-Hill.

United States Navy. 1966. *Principles of Naval Engineering.* Washington, D.C.: United States Navy.

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CHAPTER 11

Piping Components and Systems

WILLIAM J. SEMBLER

The term *piping* refers to any assembly of pipes, tubes, valves, and fittings that forms a whole or part of a system used for the conveyance of one or more fluids (liquids, vapors, or gases). Piping systems should meet applicable regulations, which often include specifications or restrictions that address various issues such as wall thickness, materials of construction, and method of fabrication, including joint design.

PIPE

Pipe size is generally specified using designators that are related to diameter and wall thickness. For example, when uses units are used, pipe is frequently identified by its nominal pipe size (NPS), which is related to diameter, and its schedule, which is related to nominal wall thickness. Dimensions and weights for selected sizes of steel pipe are shown in table 11-1, which also includes dimensions for the older commercial designations of standard pipe (STD), extra-strong pipe (XS), and double-extra-strong pipe (XXS). As shown in this table, regardless of schedule, all pipe sections with a given NPS have the same outside diameter. Also, when the schedule is increased for a given NPS, the wall thickness is increased and, therefore, the inside diameter is reduced. For pipe with an NPS from .Ys to 12, the nominal pipe size is approximately equal to the inside diameter in inches of schedule 40 pipe; consequently, the actual outside diameter of this pipe is greater than the nominal size. However, when the NPS is 14 to 80, the nominal pipe size is approximately equal to the outside diameter of the pipe in inches.

In the metric system, pipe diameter is designated by a nominal diameter, which is referred to as DN and, as shown in table 11-1, is approximately equal to a rounded-off value of the outside diameter in mm.

Dimensions for regular and extra-strong copper and brass pipe are shown in table 11-2a. The use of copper and brass pipe is generally limited by temperature and pressure. For example, some specifications permit copper or brass pipe to be used in steam or water service only when the design

TABLE 11-1
Nominal Dimensions and Weights of Selected Sizes of Welded and Seamless Wrought Steel Pipe

USCS Units				Identification		SI (Metric) Units			
NPS[A]	Outside Diameter (Inches)	Wall Thickness (Inches)	Plain-End Weight (lbf/ft)	Commercial Designation[B]	Schedule Number	DN[C]	Outside Diameter (mm)	Wall Thickness (mm)	Plain-End Mass (kg/m)
1/8	0.405	0.068	0.24	STD	40		10.3	1.73	0.37
		0.095	0.31	XS	80			2.41	0.47
1/4	0.540	0.088	0.42	STD	40		13.7	2.24	0.63
		0.119	0.54	XS	80			3.02	0.80
3/8	0.675	0.091	0.57	STD	40	10	17.1	2.31	0.84
		0.126	0.74	XS	80			3.20	1.10
1/2	0.840	0.109	0.85	STD	40	15	21.3	2.77	1.27
		0.147	1.09	XS	80			3.73	1.62
		0.188	1.31		160			4.78	1.95
		0.294	1.71	XXS				7.47	2.55
3/4	1.050	0.113	1.13	STD	40	20	26.7	2.87	1.69
		0.154	1.47	XS	80			3.91	2.20
		0.219	1.94		160			5.56	2.90
		0.308	2.44	XXS				7.82	3.64
1	1.315	0.133	1.68	STD	40	25	33.4	3.38	2.50
		0.179	2.17	XS	80			4.55	3.24
		0.250	2.84		160			6.35	4.24
		0.358	3.66	XXS				9.09	5.45
1-1/4	1.660	0.140	2.27	STD	40	32	42.2	3.56	3.39
		0.191	3.00	XS	80			4.85	4.47
		0.250	3.76		160			6.35	5.61
		0.382	5.21	XXS				9.70	7.77
1-1/2	1.900	0.145	2.72	STD	40	40	48.3	3.68	4.05
		0.200	3.63	XS	80			5.08	5.41
		0.281	4.86		160			7.14	7.25
		0.400	6.41	XXS				10.15	9.56
2	2.375	0.154	3.65	STD	40	50	60.3	3.91	5.44
		0.218	5.02	XS	80			5.54	7.48
		0.344	7.46		160			8.74	11.11
		0.436	9.03	XXS				11.07	13.44
2-1/2	2.875	0.203	5.79	STD	40	65	73.0	5.16	8.63
		0.276	7.66	XS	80			7.01	11.41
		0.375	10.01		160			9.53	14.92
		0.552	13.69	XXS				14.02	20.39
3	3.500	0.216	7.58	STD	40	80	88.9	5.49	11.29
		0.300	10.25	XS	80			7.62	15.27
		0.438	14.32		160			11.13	21.35
		0.600	18.58	XXS				15.24	27.68
3-1/2	4.000	0.226	9.11	STD	40		101.6	5.74	13.57
		0.318	12.50	XS	80			8.08	18.63
4	4.500	0.237	10.79	STD	40	100	114.3	6.02	16.07
		0.337	14.98	XS	80			8.56	22.32
		0.531	22.51		160			13.49	33.54
		0.674	27.54	XXS				17.12	41.03
5	5.563	0.258	14.62	STD	40	125	141.3	6.55	21.77
		0.375	20.78	XS	80			9.53	30.97
		0.625	32.96		160			15.88	49.11
		0.750	38.55	XXS				19.05	57.43

Continued on next page

TABLE 11-1—*Continued*

USCS Units				Identification		SI (Metric) Units			
NPS[A]	Outside Diameter (Inches)	Wall Thickness (Inches)	Plain-End Weight (lbf/ft)	Commercial Designation[B]	Schedule Number	DN[C]	Outside Diameter (mm)	Wall Thickness (mm)	Plain-End Mass (kg/m)
6	6.625	0.280	18.97	STD	40	150	168.3	7.11	28.26
		0.432	28.57	XS	80			10.97	42.56
		0.719	45.35		160			18.26	67.56
		0.864	53.16	XXS				21.95	79.22
8	8.625	0.322	28.55	STD	40	200	219.1	8.18	42.55
		0.500	43.39	XS	80			12.70	64.64
		0.875	72.42	XXS				22.23	107.92
		0.906	74.69		160			23.01	111.27
10	10.750	0.365	40.48	STD	40	250	273.0	9.27	60.31
		0.500	54.74	XS	60			12.70	81.55
		0.594	64.43		80			15.09	96.01
		1.000	104.13	XXS	140			25.40	155.15
		1.125	115.64		160			28.58	172.33
12	12.750	0.375	49.56	STD		300	323.8	9.53	73.88
		0.406	53.52		40			10.31	79.73
		0.500	65.42	XS				12.70	97.46
		0.688	88.63		80			17.48	132.08
		1.000	125.49	XXS	120			25.40	186.97
		1.312	160.27		160			33.32	238.76
14	14.000	0.375	54.57	STD	30	350	355.6	9.53	81.33
		0.438	63.44		40			11.13	94.55
		0.500	72.09	XS				12.70	107.39
		0.750	106.13		80			19.05	158.10
		1.406	189.11		160			35.71	281.70
16	16.000	0.375	62.58	STD	30	400	406.4	9.53	93.27
		0.500	82.77	XS	40			12.70	123.30
		0.844	136.61		80			21.44	203.53
		1.594	245.25		160			40.49	365.35
18	18.000	0.375	70.59	STD		450	457	9.53	105.16
		0.500	93.45	XS				12.70	139.15
		0.562	104.67		40			14.27	155.80
		0.938	170.92		80			23.83	254.55
		1.781	308.50		160			45.24	459.37
20	20.000	0.375	78.60	STD	20	500	508	9.53	117.15
		0.500	104.13	XS	30			12.70	155.12
		0.594	123.11		40			15.09	183.42
		1.031	208.87		80			26.19	311.17
		1.969	379.17		160			50.01	564.81
40	40.000	0.375	158.70	STD		1000	1016	9.53	236.53
		0.500	210.93	XS				12.70	314.22
		1.250	517.31					31.75	770.62
60	60.000	0.375	238.80				1524	9.53	355.92
		0.500	317.73					12.70	473.31
		1.250	784.31					31.75	1168.36
80	80.000	0.562	476.80			2000	2032	14.27	710.04
		1.000	843.72					25.40	1256.86
		1.250	1051.31					31.75	1566.11

A. NPS = Nominal Pipe Size
B. Sometimes referred to as Iron Pipe Size or IPS
C. DN = Nominal Diameter
STD = Standard pipe
XS = Extra-strong pipe
XXS = Double-extra-strong pipe

The data listed above is based on information included in ASME B36.10M-1996. Variations in pipe dimensions from the nominal values shown herein differ based on the method of manufacture. Individual specifications should be consulted for permissible variations. Weights and masses listed are theoretical values.

TABLE 11-2A
Standard Sizes of Seamless Copper and Red Brass Pipe

Nominal or Standard Pipe Size (inches)	Outside Diameter (inches)	Regular Wall Thickness (inch)	Regular 99.9Cu Weight (lbm/ft)	Regular 85Cu Weight (lbm/ft)	Extra Strong Wall Thickness (inch)	Extra Strong 99.9Cu Weight (lbm/ft)	Extra Strong 85Cu Weight (lbm/ft)
1/8	0.405	0.062	0.259	0.253	0.100	0.371	0.363
1/4	0.540	0.082	0.457	0.447	0.123	0.625	0.611
3/8	0.675	0.090	0.641	0.627	0.127	0.847	0.829
1/2	0.840	0.107	0.955	0.934	0.149	1.25	1.23
5/8	—	—	—	—	—	—	—
3/4	1.050	0.114	1.30	1.27	0.157	1.71	1.67
1	1.315	0.126	1.82	1.78	0.182	2.51	2.46
1-1/4	1.660	0.146	2.69	2.63	0.194	3.46	3.39
1-1/2	1.900	0.150	3.20	3.13	0.203	4.19	4.10
2	2.375	0.156	4.22	4.12	0.221	5.80	5.67
2-1/2	2.875	0.187	6.12	5.99	0.280	8.85	8.66
3	3.500	0.219	8.76	8.56	0.304	11.8	11.6
3-1/2	4.000	0.250	11.4	11.2	0.321	14.4	14.1
4	4.500	0.250	12.9	12.7	0.341	17.3	16.9
5	5.562	0.250	16.2	15.8	0.375	23.7	23.2
6	6.625	0.250	19.4	19.0	0.437	32.9	32.2
8	8.625	0.312	31.6	30.9	0.500	49.5	48.4
10	10.750	0.365	46.2	45.2	0.500	62.4	61.1
12	12.750	0.375	56.5	55.3	—	—	—

Notes:

1. The data given above for seamless copper pipe (99.9Cu) is based on information included in ASTM B 42-93. Copper pipe included in this specification can contain 99.9% to 99.95% copper (including silver). The remainder of the material is primarily phosphorus.
2. The data given above for seamless red brass pipe (85Cu) is based on information included in ASTM B 43-94. Copper pipe included in this specification can contain from 84% to 86% copper. The remainder of the material is primarily zinc.
3. Tolerances on the dimensions listed are included in the referenced specifications.
4. Weights and masses listed are theoretical values.

pressure does not exceed 250 psig 0,715 kPa) and the design temperature does not exceed 406°F (208°C).

Seamless metal pipe can be made by piercing a heated billet with a mandrel and squeezing its outer surface by forcing it between rollers or through dies, by machining (turning and boring) a cylindrically shaped forging, or by pouring molten metal into a stationary or rotating mold. Seamed pipe is fabricated from rolled sheet or plate with either a butt or an overlapping scarfed seam that is welded or brazed. The outside weld bead is typically ground down until it is flush with the outer wall of the pipe so that it will not interfere with fittings that may be attached to the ends of the pipe.

Plastic and composite pipe is typically manufactured from thermoplastics or glass-reinforced thermosetting resins. Although pipe in this latter group can be centrifugally cast, it is more common for it to be manufactured by winding continuous fiberglass filaments that are saturated with liquid resins around a mandrel.

TUBING

Tubing, which can usually be bent more easily than pipe, is often identified by outside diameter and by wall thickness, which may be specified as a

TABLE 11-2B
Selected Birmingham Wire Gauge (BWG) Values in Decimals of an Inch

BWG	Decimal Value (inch)	BWG	Decimal Value (inch)	BWG	Decimal Value (inch)
5/0	0.5000	6	0.2030	16	0.0650
4/0	0.4540	7	0.1800	17	0.0580
3/0	0.4250	8	0.1650	18	0.0490
2/0	0.3800	9	0.1480	19	0.0420
1/0	0.3400	10	0.1340	20	0.0350
1	0.3000	11	0.1200	21	0.0320
2	0.2840	12	0.1090	22	0.0280
3	0.2590	13	0.0950	23	0.0250
4	0.2380	14	0.0830	24	0.0220
5	0.2200	15	0.0720	25	0.0200

decimal value or in terms of a Birmingham wire gauge (BWG) number (table 11-2b). In addition, tube size is sometimes designated by a number that is equal to the number of sixteenths of an inch in the tube's outside diameter. For example, a #16 tube has an outside diameter of one inch. Also, copper tubing is sometimes identified by a nominal or standard size that is 0.125 in. less than the actual outside diameter. Furthermore, the wall thickness of copper tubing may be specified by a letter designation: type K, L, or M, where the letter establishes the wall thickness in inches oftubing with a given outside diameter (table 11-2c).

Both type K and type L tubing are available in soft and hard tempers. In addition, type K tubing is ordinarily suitable for more severe duty than type L tubing. Type M tubing, which is available only in a hard temper, is usually used for low-pressure service. Similar designations that are referred to as types A, B, and C are used for copper tube when the wall thickness is given in millimeters.

Tubing can be seamless or seam welded. Postweld finishing processes include drawing the tube through a die to cold finish the exterior surface (sink drawn) or drawing the tube both through a die and over a mandrel (DOM) to finish both the interior and exterior surfaces. Some tubing materials are available in hard grades, which are generally used in straight lengths, and soft grades, which can be bent without the use of special bending equipment.

HOSE

Hose may be used in applications where flexibility is required, minor misalignment exists, or line connections are temporary. The lining or inner tube of a hose is frequently constructed from an elastomer and should be compatible with the fluids that will pass through the hose at the system operating temperatures. A reinforcement consisting of one or more layers of fiber or fabric made from natural or synthetic materials or metal wire is

TABLE 11-2C
Standard Sizes of Seamless Copper Water Tube

USCS Units							
		Type K		Type L		Type M	
Nominal or Standard Size (inches)	Outside Diameter (inches)	Wall Thickness (inch)	Weight (lbm/ft)	Wall Thickness (inch)	Weight (lbm/ft)	Wall Thickness (inch)	Weight (lbm/ft)
1/8	*0.250	*0.032	*0.085	*0.025	*0.068	*0.025	*0.068
1/4	0.375	0.035	0.145	0.030	0.126	*0.025	*0.106
3/8	0.500	0.049	0.269	0.035	0.198	0.025	0.145
1/2	0.625	0.049	0.344	0.040	0.285	0.028	0.204
5/8	0.750	0.049	0.418	0.042	0.362	*0.030	*0.263
3/4	0.875	0.065	0.641	0.045	0.455	0.032	0.328
1	1.125	0.065	0.839	0.050	0.655	0.035	0.465
1-1/4	1.375	0.065	1.04	0.055	0.884	0.042	0.682
1-1/2	1.625	0.072	1.36	0.060	1.14	0.049	0.940
2	2.125	0.083	2.06	0.070	1.75	0.058	1.46
2-1/2	2.625	0.095	2.93	0.080	2.48	0.065	2.03
3	3.125	0.109	4.00	0.090	3.33	0.072	2.68
3-1/2	3.625	0.120	5.12	0.100	4.29	0.083	3.58
4	4.125	0.134	6.51	0.110	5.38	0.095	4.66
5	5.125	0.160	9.67	0.125	7.61	0.109	6.66
6	6.125	0.192	13.9	0.140	10.2	0.122	8.92
8	8.125	0.271	25.9	0.200	19.3	0.170	16.5
10	10.125	0.338	40.3	0.250	30.1	0.212	25.6
12	12.125	0.405	57.8	0.280	40.4	0.254	36.7

SI (Metric) Units							
		Type A		Type B		Type C	
Nominal or Standard Size (mm)	Outside Diameter (mm)	Wall Thickness (mm)	Mass (kg/m)	Wall Thickness (mm)	Mass (kg/m)	Wall Thickness (mm)	Mass (kg/m)
6	6.0	0.80	0.117	0.70	0.104	0.60	0.091
8	8.0	0.90	0.179	0.80	0.162	0.60	0.125
10	10.0	0.90	0.230	0.80	0.207	0.60	0.158
12	12.0	1.2	0.364	0.90	0.280	0.60	0.192
15	15.0	1.2	0.465	1.0	0.393	0.70	0.281
18	18.0	1.2	0.566	1.0	0.477	0.70	0.340
22	22.0	1.6	0.917	1.1	0.646	0.80	0.476
28	28.0	1.6	1.19	1.2	0.903	0.90	0.685
35	35.0	1.6	1.50	1.4	1.32	1.1	1.05
42	42.0	1.8	2.03	1.5	1.71	1.2	1.37
54	54.0	2.1	3.06	1.7	2.50	1.5	2.21
67	67.0	2.4	4.35	2.0	3.65	1.6	2.94
79	79.0	2.8	5.99	2.3	4.95	1.8	3.90
105	105.0	3.4	9.70	2.8	8.04	2.4	6.92
130	130.0	4.0	14.2	3.1	11.0	2.7	9.65
156	156.0	4.8	20.3	3.5	15.0	3.1	13.3
206	206.0	6.8	38.0	5.0	28.2	4.3	24.4
257	257.0	8.5	59.3	6.3	44.4	5.4	38.2
308	308.0	10.3	86.1	7.1	60.0	6.4	54.2

Notes:

1. The data given above for seamless copper water tube with standard sizes in inches is based on information included in ASTM B 88-95a. Sizes marked with an * are not included in this specification and may not be generally available.
2. The data given above for seamless copper water tube with standard sizes in millimeters is based on information included in ASTM B 88M-95.
3. Tolerances on the dimensions listed are included in the referenced specifications.
4. Weights and masses listed are theoretical values.

typically braided or spirally wound around the lining to give the hose the ability to withstand both internal pressures and external forces. In some cases, alternating layers of fiber, fabric, and metal wire are used. In addition, an outer cover is sometimes provided to shield the reinforcement from sharp objects, high or low ambient temperatures, external substances

(such as solvents), and other environmental conditions that could damage the hose; synthetic fabrics and elastomeric materials are commonly used. Wire in the form of a spiral helix (i.e., resembling a spring) may be installed inside a hose or wrapped around its cover to increase the hose's rigidity, which can prevent collapse when there is a vacuum inside the hose, and to prevent the hose from becoming kinked at bends.

Hydraulic-system hose is often rated based on three different pressures: (1) minimum burst pressure, which is normally equal to approximately 90 percent of the actual burst pressure determined by performing a series of tests on identical hose; (2) proof pressure, which is usually equal to approximately 50 percent of the minimum burst-pressure rating and can be applied to a hose assembly for a specified period of time; and (3) working pressure, which is the maximum allowable operating pressure for a hose and is usually 20 to 33 percent of the minimum burst-pressure rating.

Low-pressure hose is typically suitable for working pressures up to 600 psig (4,140 kPa) or less; medium-pressure hose for working pressures up to 1,000 to 3,000 psig (6,870 to 20,610 kPa); high-pressure hose for working pressures up to 3,000 to 4,000 psig (20,610 to 27,480 kPa); and extra-high-pressure hose for working pressures that exceed 3,000 psig (20,610 kPa) and that are, in some cases, as high as 6,000 to 10,000 psig (41,220 to 68,700 kPa). However, in each category, the maximum allowable working pressure is typically reduced as the hose size increases. A hose should also be rated for the range of ambient temperatures (that of the surroundings) and operating temperatures (that of the fluids being conveyed through the hose) to which it will be exposed.

The size of hydraulic hose is sometimes designated by a dash followed by a number that is marked on the hose or on a metal tag attached to the hose and is equal to the number of sixteenths of an inch in the hose's nominal size. For example, a -16 hose has nominal size of one inch. A hose's nominal size is usually equal to or slightly larger (e.g., 0.0625 to 0.125 in. larger) than the hose's actual inside diameter.

Flexible metal hose constructed from a series of corrugations or interlocking continuous strips of metal is often suitable for use in harsh environments or high-temperature applications, such as the exhaust lines of internal combustion engines. A metal hose may be provided with an outer cover to protect it and to increase its axial strength. In addition, the space between the inner hose and the outer cover may contain insulation to reduce heat transfer to the engine space.

When a hose is assembled and connected to a system, it should not be bent with a radius less than the minimum value specified for it. In addition, high-pressure hose should never be bent in more than one plane. When bending in more than one plane is required, multiple sections of hose should be used. Also, a hose should never be twisted. After assembly, the free or excess length of the hose should be sufficient to allow for a reduction in length as the diameter expands during pressurization. However, high-pressure hose should be adequately secured so that it will not jump around when pressurized. Furthermore, a hose should always be protected from sharp, abrasive, or heated objects that could damage it.

CONNECTIONS

Pipe Connections

The following paragraphs describe common joints that can be used to connect sections of pipe or to connect fittings to pipe.

BUTT-WELDED JOINT

Butt welding is one of the most common methods used to join two lengths of pipe or to join a pipe to a fitting. The ends of the two pieces being joined are frequently beveled before being aligned and welded (fig. 11-1a). In addition, when a circumferential butt joint is only being welded from the outside, which is common in piping systems, a backing ring or a consumable insert ring is sometimes placed inside the joint during alignment to form a base for the weld's root pass.

SOCKET JOINT

The end connection of a socket-welding fitting has a socket into which the end of a pipe is inserted (fig. 11-1b). A circumferential fillet weld is then applied around the joint between the pipe and the fitting. Generally, the pipe should not be bottomed in the socket prior to welding. Instead, a small gap should remain between the end of the pipe and the bottom of the socket,

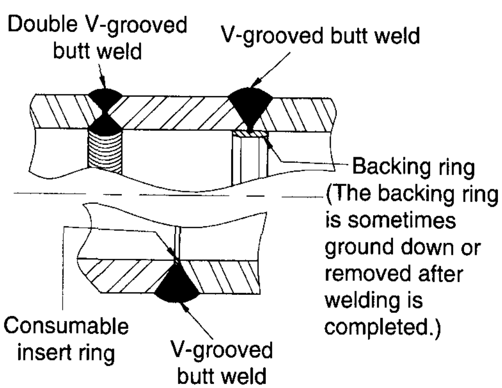


Figure 11-1a. Butt-welded pipe joints

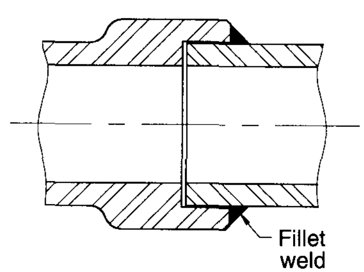


Figure 11-1b. Socket-welding joint

e.g., 0.06 in. or 2.0 mm, to allow for thermal expansion of the pipe. Because the pipe end is supported by the socket, this type of a fitting is easier to align than a fitting with a butt-welded joint. In addition, when a socket-welding fitting is used, the pipe end does not have to be beveled and the need for a backing ring or insert is eliminated. However, although socket-welding fittings are often available in sizes up to NPS 4 (DN 150), the use of socket-welded connections is frequently limited to pipe sizes of NPS 2 1/2 to 3 (DN 65 to 80) and smaller.

Similar brazed-socket fittings are often used with copper and copper-alloy pipe. During brazing, it is important that the brazing material, which is generally a silver- or copper-base alloy, fills the annular space between the pipe and the socket. A commonly used type of brazed-joint fitting has a groove within the socket that contains a preinserted ring of silver brazing alloy (fig 11-2). End- or face-fed brazed fittings are rarely used. Brazed-socket joints are generally not suitable for high-temperature (e.g., over 425°F or 218°C) applications.

Socket fittings used with plastic and composite pipe typically have cemented or heat-fused joints.

INTERSECTION-TYPE WELD JOINT

An intersection joint is formed when two sections of pipe are connected with an angle between their longitudinal axes. Examples include 90° branch or nozzle (tee) and wye intersections. When a branch has a set-on joint, the opening in the main pipe, or header, matches the inside diameter of the pipe that forms the branch, and the end of the branch pipe is beveled and is contoured to match the outside surface of the header (fig. 11-3a). When a set-through nozzle joint is used, the hole in the header matches the outside diameter of the branch, which is contoured to match the inner surface of the header (fig. 11-3b). With either arrangement, during assembly, the two sections of pipe are first joined with a through weld that is then covered with a fillet weld. A pad is sometimes welded around the outside of the branch to reinforce the joint.

BOLTED-FLANGED JOINT

A bolted-flanged joint is frequently used when a piping section, fitting, or component installed within a system (e.g., pumps, heat exchangers, valves, etc.) may have to be removed periodically for inspection, maintenance, or replacement. A flange may be cast or forged integrally with a pipe or component. Alternatively, a flange may be manufactured separately and then mounted on the end of a piece of pipe. A welding-neck flange (fig. 11-4a) has an extended neck that can be butt-welded to the mating section of pipe. A socket-welding flange has a socket into which the end of the mating pipe is inserted. The joint is then sealed with a circumferential fillet weld (fig.

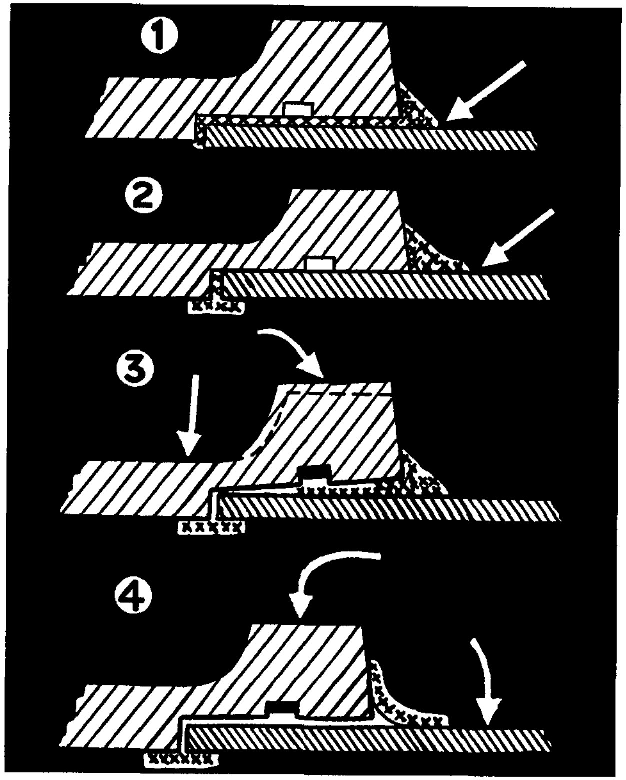


Figure 11-2. Steps to braze a socket joint: (1) At the start, the clearance area is filled with flux. (2) Tube is heated and flux flows out; clearance area is closed and the braze alloy contacts the heated tube surface. (3) Fitting is heated and clearance area opens. Flux and alloy flow out. (4) Both tube and fitting are then heated by wiping motion. As alloy flows out it penetrates the surfaces and forms a fillet around the edge of the fitting.

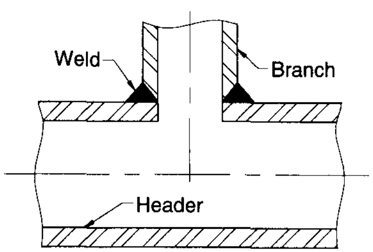


Figure 11-3a. Set-on welded intersection joint

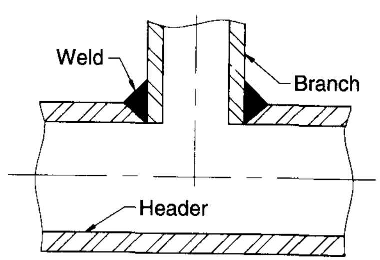


Figure 11-3b. Set-through welded intersection joint

11-4b). A similar brazed socket flange, usually with a high hub that is bored to a depth at least equal to the length of a comparably-sized threaded connection, is sometimes used with copper or copper-alloy pipe. In addition, socket flanges with cemented or heat-fused joints are frequently used with plastic and composite pipe. The opening in the face of a welding-neck or socket flange is bored to match the inside diameter of the pipe on which it will be mounted.

A slip-on flange has a bore slightly larger than the pipe's outside diameter. During fabrication, the flange is slipped over the end of the mating pipe, and both the inside and outside edges of the joint are sealed with fillet welds (fig. 11-4b). In an alternate configuration, the end of the pipe is deformed (flanged) to form a lip that fits into a recess machined in the face of the flange (fig. 11-4c). With this latter arrangement, which is typically used only with nonferrous pipe, the flange may be brazed to the pipe. A lap-joint (Van Stone) flange also has a bore that is slightly larger than the mating pipe's outside diameter. However, during fabrication, the lap-joint flange is slipped over the pipe or over one end of a special lap-joint stub piece that is then butt welded to the end of the pipe. The end of the pipe, or the opposite end of the stub piece when used, is lapped over to form a lip that is larger in diameter than the bore of the flange (fig. 11-4d). After the pipe-and-stub-piece assembly is installed in the system, the lip is sandwiched between the face of the lap-joint flange, which generally has a curved radius along the opening in its face that fits against a fillet on the back of the lip, and the mating flange to which the lap-joint flange is bolted. With this arrangement, the face of the lip and not the flange forms the bearing surface of the joint. The width of the lip formed for both the flanged-pipe arrangement (fig. 11-4c) and the lap-joint flange (fig. 11-4d) should generally be at least three times the wall thickness of the pipe.

A threaded flange, which has a bore that is threaded, is screwed onto the threaded end of the mating piece of pipe during assembly (fig. 11-4a). In some cases (especially with larger sizes), the threaded connection between the pipe and the flange is then seal welded.

Adjacent flange faces may be ground and lapped to obtain a gasketless metal-to-metal fluid-tight joint. However, to avoid the need for this and to enable a tight seal to be achieved when a flange face has minor imperfections, a gasket is often installed between the contacting faces of two mating flanges to help seal the joint. A flat full-face gasket extends to a flange's outside diameter; consequently, it must have holes for the flange bolts. A flat ring gasket, however, extends only to the inner edges of the flange bolts and, therefore, has no bolt holes. To improve the seal that is formed between a gasket and a flange and to reduce the potential for the gasket to be extruded or blown out of the joint, concentric or spiral serrations (the latter referred to as a phonographic finish) are sometimes machined into a flange's face. The serrations act like teeth that dig into the gasket when the joint is tightened.

A bolted-flanged joint may also be sealed with an elastomeric O-ring or a metal seal ring. When an O-ring is used, it is typically inserted into a special circular groove machined in the face of one of the mating flanges (fig. 11-4e). Because the depth ofthe groove is slightly less than the O-ring's diameter, the ring is compressed and forms a seal when the two flanges are bolted together. When a metal seal ring is used, a circular groove is ordinarily machined in the face of each of the mating flanges (fig. 11-4a). The metal ring, which often has an oval or octagonal cross section, is inserted into the grooves during assembly. A gap, referred to as standoff, should frequently remain between the faces of the two mating flanges after a metal-ring-and-groove joint is assembled.

The entire face of a flat-face flange is often in contact with a gasket or the mating flange's face (fig. 11-4e). However, the contact surface of a raised-face flange is reduced to only a portion of the flange's face that resembles a step near the flange's opening (fig. 11-4d). Because of this reduced contact area, the clamping force necessary to produce a given pressure across the joint is less than the force needed with a comparably sized flat-faced flange. Raised-face flanges are frequently used in higher-pressure applications. When two raised-face flanges are bolted together, they form what is commonly referred to as a male-male joint. Alternatively, tbe raised face on one flange can fit into a circular recess in the face of the mating flange and form a male-female joint (fig. 11-4f). Also, some flanges have a tongue-groove arrangement in which a protrusion on the face of one flange fits into a groove machined in the face ofthe mating flange (fig. 11-4g). When either a male-female or tongue-groove arrangement is used, a ring gasket is often inserted into the groove or recess to help seal the joint. With a male-male joint, a ring gasket is frequently inserted between the raised faces of the mating flanges.

When piping, fittings, or components in a system have flanged joints, mating flanges should be of the same size. In addition, the orientation of the bolt-hole drilling pattern in mating flanges must match. Flange bolt holes are generally equally spaced around the circumference ofthe bolt circle. Also, bolt holes in standard commercial flanges, which are ordinarily in multiples of four (i.e., four holes, eight holes, twelve holes, etc.), are frequently arranged in horizontal piping connections so that the bolt holes straddle both the horizontal and vertical centerlines ofthe mating flanges. The back or nonmating flange faces are typically spot-faced around each bolt hole so that bolt heads and nuts will seat squarely during assembly. Bolts used to join mating flanges are normally 0.125 in. (3.2 mm) smaller in diameter than the diameter of the flange bolt holes.

A blind or blank flange is not used to join piping but is used to seal the open end of a flanged section of piping. A spectacle flange has a center

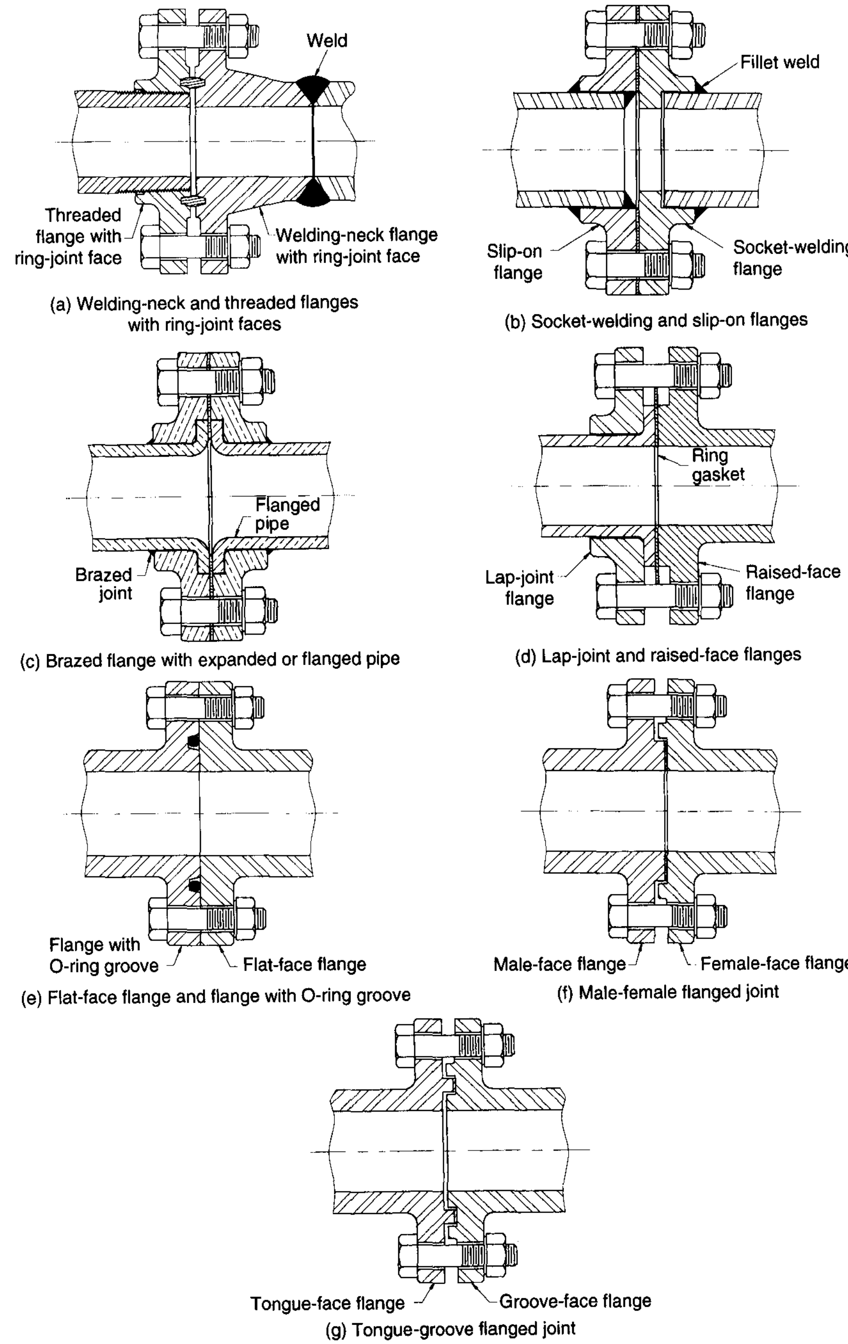


Figure 11-4. Flanges

plate, referred to as the spectacle plate, resembling a figure eight that is installed between two companion flanges. One portion of the spectacle plate has a hole through it and the other portion is blind. When the open portion is positioned between the flanges, fluid can flow through the joint. However, when the plate is rotated 180°, the blind portion moves between the flanges, and flow through the joint is prevented. The joint should not be pressurized while the spectacle plate is being repositioned .

• THREADED JOINT

A threaded or screwed joint is sometimes used to connect a pipe and fitting in a smaller-sized system, e.g., not larger than NPS 2 (DN 50). However, some types of fittings are available with screwed connections in sizes up to or exceeding NPS 12 (DN 300).

Threads used for piping connections are often tapered, such as the American National Standard taper pipe thread (NPT), which has a taper of 0.75 in. in diameter per foot oflength. Because of the taper, the interference between mating threads increases as the joint is tightened, which helps to prevent leakage. Dimensions for NPT threads are shown in table 11-3. NPT tap drill sizes are shown in table 11-4. To increase the effectiveness of the seal created between the mating threads in a screwed joint, the male threads are frequently coated with a sealing compound, called dope, or wrapped with special tape, such as polytetrafluoroethylene (PTFE) tape, prior to assembly. When using tape, several threads at the open end of the pipe or fitting should remain uncovered so that they will engage more easily with the female threads in the mating component during assembly.

EXPANSION JOINT

An expansion joint or coupling provides flexibility in a piping system and can reduce stresses resulting from thermal expansion or contraction, misalignment, and piping deflection. In some designs, one section of piping is attached to a sleeve that is free to slide back and forth within the expansion joint's body, which is attached to the other section of piping (fig. 11-5a). This slip-joint arrangement permits lateral movement between the two sections of piping. A packed stuffing box is often provided to control leakage between the expansion joint's sleeve and body. In addition, bolts or stops are required to prevent excessive piping movement from pulling the sleeve out ofthe joint's body.

In another expansion-joint design, the two pieces of pipe being joined are inserted into opposite ends of a coupling that resembles a sleeve. Each end of the sleeve is sealed by an elastomeric ring that fits between the inner surface of the sleeve and the outer surface of the adjacent section of pipe and is compressed by a flange. The coupling assembly is held together with

TABLE 11-3
Basic Dimensions for American National Standard Taper Pipe Threads (NPT)

Nominal Pipe Size (NPS)	Number of Threads per Inch	Thread Height	Pipe Outside Diameter D	Pitch Diameter at End of Pipe E0[A]	Pitch Diameter at Gauging Notch E1	Normal Hand-Tight Engagement L1[B]	Length of Effective External Thread L2[C]	Wrench Makeup Length of Internal Thread L3[D]	Overall Length of External Thread L4	Nominal Length of Perfect External Threads L5	Pitch of Thread P	Length of Imperfect Threads Due to Chamfer on Die V	Approximate Wrench-Tight Thread Engagement[B]
1/16	27	0.02963	0.3125	0.27118	0.28118	0.160	0.2611	0.1111	0.3896	0.1870	0.03704	0.1285	1/4
1/8	27	0.02963	0.405	0.36351	0.37360	0.1615	0.2639	0.1111	0.3924	0.1898	0.03704	0.1285	1/4
1/4	18	0.04444	0.540	0.47739	0.49163	0.2278	0.4018	0.1667	0.5946	0.2907	0.05556	0.1928	5/16 to 3/8
3/8	18	0.04444	0.675	0.61201	0.62701	0.240	0.4078	0.1667	0.6006	0.2967	0.05556	0.1928	3/8
1/2	14	0.05714	0.840	0.75843	0.77843	0.320	0.5337	0.2143	0.7815	0.3909	0.07143	0.2478	7/16 to 1/2
3/4	14	0.05714	1.050	0.96768	0.98887	0.339	0.5457	0.2143	0.7935	0.4029	0.07143	0.2478	1/2 to 9/16
1	11-1/2	0.06957	1.315	1.21363	1.23863	0.400	0.6828	0.2609	0.9845	0.5089	0.08696	0.3017	9/16 to 11/16
1-1/4	11-1/2	0.06957	1.660	1.55713	1.58338	0.420	0.7068	0.2609	1.0085	0.5329	0.08696	0.3017	9/16 to 11/16
1-1/2	11-1/2	0.06957	1.900	1.79609	1.82234	0.420	0.7235	0.2609	1.0252	0.5496	0.08696	0.3017	9/16 to 11/16
2	11-1/2	0.06957	2.375	2.26902	2.29627	0.436	0.7565	0.2609	1.0582	0.5826	0.08696	0.3017	5/8 to 3/4
2-1/2	8	0.100000	2.875	2.71953	2.76216	0.682	1.1375	0.2500	1.5712	0.8875	0.12500	0.4337	7/8 to 15/16
3	8	0.100000	3.500	3.34062	3.38850	0.766	1.2000	0.2500	1.6337	0.9500	0.12500	0.4337	1
3-1/2	8	0.100000	4.000	3.83750	3.88881	0.821	1.2500	0.2500	1.6837	1.0000	0.12500	0.4337	1-1/16
4	8	0.100000	4.500	4.33438	4.38712	0.844	1.3000	0.2500	1.7337	1.0500	0.12500	0.4337	1-1/16 to 1-1/8
5	8	0.100000	5.563	5.39073	5.44929	0.937	1.4063	0.2500	1.8400	1.1563	0.12500	0.4337	1-3/16 to 1-1/4
6	8	0.100000	6.625	6.44609	6.50597	0.958	1.5125	0.2500	1.9462	1.2625	0.12500	0.4337	1-3/16 to 1-5/16
8	8	0.100000	8.625	8.43359	8.50003	1.063	1.7125	0.2500	2.1462	1.4625	0.12500	0.4337	1-5/16 to 1-7/16
10	8	0.100000	10.750	10.54531	10.62094	1.210	1.9250	0.2500	2.3587	1.6750	0.12500	0.4337	1-7/16 to 1-5/8
12	8	0.100000	12.750	12.53281	12.61781	1.360	2.1250	0.2500	2.5587	1.8750	0.12500	0.4337	1-9/16 to 1-7/8
14	8	0.100000	14.000	13.77500	13.87262	1.562	2.2500	0.2500	2.6837	2.0000	0.12500	0.4337	
16	8	0.100000	16.000	15.76250	15.87575	1.812	2.4500	0.2500	2.8837	2.2000	0.12500	0.4337	
18	8	0.100000	18.000	17.75000	17.87500	2.000	2.6500	0.2500	3.0837	2.4000	0.12500	0.4337	
20	8	0.100000	20.000	19.73750	19.87031	2.125	2.8500	0.2500	3.2837	2.6000	0.12500	0.4337	
24	8	0.100000	24.000	23.71250	23.86094	2.375	3.2500	0.2500	3.6837	3.0000	0.12500	0.4337	

A. In some special high-pressure applications, dimension E0 may be reduced to increase thread engagement.
B. Values given are approximate only. Actual thread engagement may vary from the values given above due to the effect of manufacturing tolerances.
C. The last two effective threads (identified as 2P) are perfect at the root but imperfect at the crest.
D. Dimension L3 is equal to three threads for NPS 2 and smaller and two threads for larger sizes.

Notes:

1. Thread dimensions given above are based on information included in ASME B1.20.1-1983 (R1992). Allowable tolerances appear in this specification. Wrench-tight thread engagement values are based, in part, on information included in the *National Standard Plumbing Code*, the *Worthington Pump Corp. Drafting Manual*, and *Marks' Standard Handbook for Mechanical Engineers*.
2. All dimensions are in inches.
3. Thread taper on the diameter is 3/4 inch per foot of length. The angle of the taper is 1.783° with the axial centerline of the pipe or fitting.
4. Although basic dimensions are given to four or five decimal places, this is a greater degree of precision than is ordinarily attained.

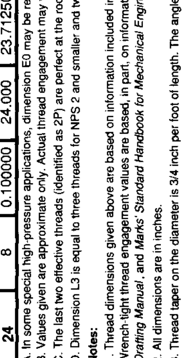


TABLE 11-4
Typical Tap Drill Sizes for American National Standard Taper Pipe Threads (NPT)

Nominal Pipe Size (NPS)	Number of Threads per Inch	Root Diameter at Small End of Pipe	Tap Drill Size	Decimal Equivalent
1/8	27	0.3339	R	0.3390
1/4	18	0.4329	7/16	0.4375
3/8	18	0.5676	37/64	0.5781
1/2	14	0.7013	23/32	0.7188
3/4	14	0.9105	59/64	0.9219
1	11-1/2	1.1441	1-5/32	1.1563
1-1/4	11-1/2	1.4876	1-1/2	1.5000
1-1/2	11-1/2	1.7265	1-47/64	1.7344
2	11-1/2	2.1995	2-7/32	2.2188
2-1/2	8	2.6195	2-5/8	2.6250
3	8	3.2406	3-1/4	3.2500
3-1/2	8	3.7375	3-3/4	3.7500
4	8	4.2344	4-1/4	4.2500
5	8	5.2907	5-5/16	5.3125

All dimensions are in inches.
Prior to tapping the drilled hole, it should be reamed with a reamer having a taper of 3/4-inch per foot.

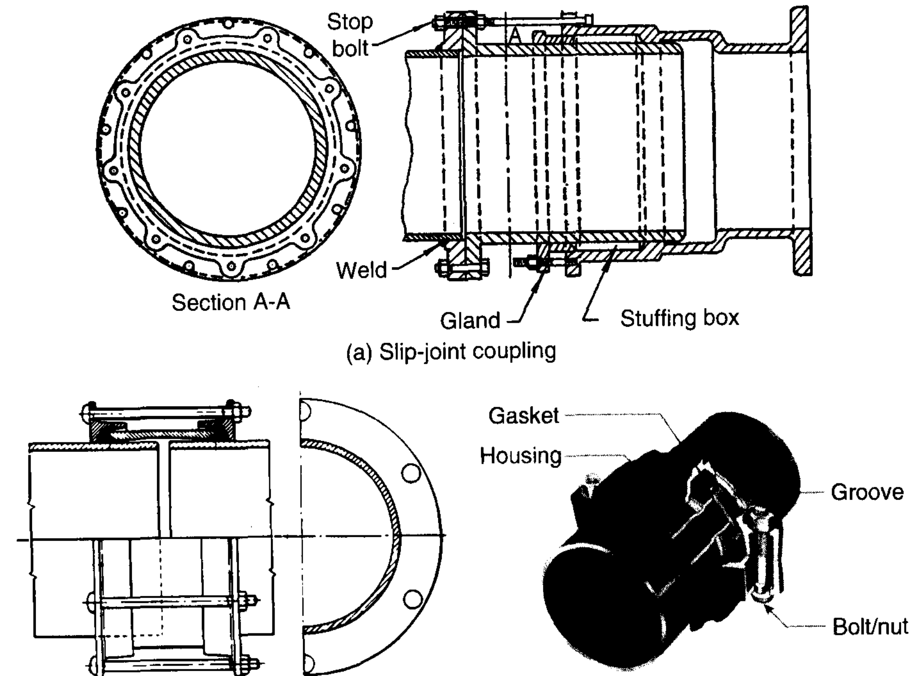
bolts that pass through holes in the two end flanges (fig. 11-5b). The opposite ends of the two piping sections joined should be anchored to prevent either pipe from being pulled out of the coupling. Other types of packless expansion-joints have flexible metallic or elastomeric bellows that permit movement between the sections of attached piping.

A segmented-grooved coupling can be used to join sections of pipe that have a circumferential groove near each mating end (fig. 11-5c). During installation, the adjacent ends of the two pipes being joined are inserted into a single ring-shaped elastomeric gasket. The gasket is positioned so that it bridges and seals the piping joint. A multisegment housing or collar is then installed around the gasket. When the bolts that connect the housing segments are tightened, the housing grips the groove in the mating end of each pipe and prevents the joint from separating. Additionally, the housing reinforces and partially compresses the gasket. This type of coupling can also be used to connect a section of grooved pipe to a similarly grooved fitting. Both rigid and flexible grooved coupling designs are available.

Tubing Connection

Fittings are often used to join two or more sections of tubing. In addition, a fitting may be used to connect the end of a tube to another component in a system (e.g., a pump, hydraulic motor, or pressure gauge).

The size of a tube connector or fitting is sometimes designated by a number that is equal to the number of sixteenths of an inch in the outside diameter of the tube used with the fitting. For example, a #16 fitting is used with tubing having an outside diameter of one inch. The end of a connector that is attached to a component (the end not attached to the tube) often has



(a) Slip-joint coupling

(b) Sleeve coupling. Courtesy Viking Johnson.

(c) Segmented-grooved coupling. Courtesy Victaulic Company.

Figure 11-5. Pipe couplings

male tapered NPT or *dryseal* American National Standard taper (NPTF) pipe threads. Unlike a standard NPT threaded connection, the roots and crests of mating NPTF threads are in contact, which increases the effectiveness of the seal. This arrangement works best with ductile materials. Tubing connectors are also frequently available with ends that have straight threads. With this latter arrangement, the joint is sealed by an O-ring that fits over the threaded end of the connector and is compressed against a groove in the hole of the mating component when the connector is tightened (fig. 11-6).

Common types of fittings are listed below. (Some of the fittings described are also used with pipe.)

SOCKET FITTING

A socket fitting has a socket at its outlet into which the end of a tube is inserted. The joint between the tubing and the fitting is then welded, brazed, or soldered. When brazing or soldering, which are most often performed with copper and copper-alloy tubing, it is important that the solder or brazing material be drawn into the annular space between the tube and the socket.

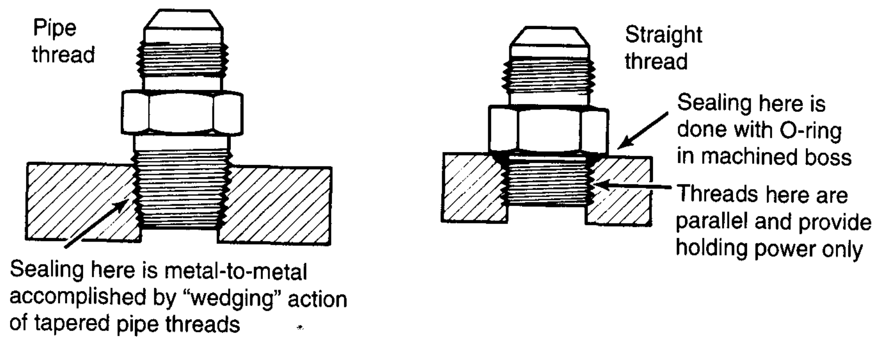


Figure 11-6. Tapered-pipe-thread versus straight-thread fittings. Courtesy Parker-Hannifin Corporation.

The socket fitting that is attached to the end of a tube is sometimes the tailpiece of a union. A nut that fits over the tubing is used to secure the tailpiece to the union's body, which may be threaded and screwed into a system component or joined to another tube. With some designs, the joint between the union's tailpiece and body is sealed with an O-ring that is compressed against the face of the tailpiece when the union nut is tightened (fig 11-7a). In another arrangement, the tailpiece that is attached to the end of a tube is secured to a system component with a split flange that fits around the tube. An O-ring inserted into a groove in the face of the tailpiece seals the assembled joint.

FLARED FIITING

During the installation of a flared tube fitting, the fitting's nut and, in some cases, a small sleeve are passed over the end of the tube. Typically, the end of the tube is then expanded or flared with a special tool, often at an angle of 37° for hydraulic systems (fig 11-7b) or 45° for other applications (fig 11-7c). As the threaded fitting nut is screwed onto the fitting's body and is tightened, the nut draws the tapered end of the body against the inner surface of the tube's flared end. In some designs, a sleeve is located between the nut and the flared end of the tube to reduce the likelihood that the tube will be twisted when the nut is tightened (shown in fig. 11-7b). Typical values of the minimum and maximum tubing wall thicknesses suitable for use with flared fittings are shown in table 11-5.

INVERTED FLARE FIITING

This fitting is similar to a flared fitting; however, the piece that slides over the end of the tube has external threads and the mating portion of the fitting's body has internal threads. In addition, during installation, the tube end is typically flared at an angle of 42° (fig. 11-7d).

SELF-FLARING FIITING

Designed to eliminate the need for the flaring operation required when flared tube fittings are used, a self-flaring fitting has a special wedge-shaped sleeve that flares the tube end as the fitting is being tightened. This type of fitting works best with thin-walled tubing.

BITE-TYPE FLARELESS FIITING

A bite-type flareless fitting is frequently used to connect plain-ended tubing to a system component. A typical bite-type flareless fitting consists of a special sleeve called a ferrule, a nut, and a body that is threaded at both ends (fig 11-7e). During assembly, the nut and ferrule are passed over the end ofthe tubing. Frequently, the end ofthe fitting's body has a socket into which the end of the tubing is then inserted, and the nut is screwed onto the body's threads. As the nut is tightened, it pushes the sleeve portion of the ferrule into the annular space between the tubing's outer surface and the fitting's body. The fitting's body forces the end of the ferrule to deform inward and bite into the tube's outer surface, which results in the formation of a small ridge of tube mr..Lerialin front of the ferrule. The ridge forms a seal between the ferrule and tubing and increases the holding power ofthe fitting. Typical values of the minimum and maximum tubing wall thicknesses suitable for use with bite-type flareless fittings are shown in table 11-5.

COMPRESSION FIITING

Many compression fittings are similar in appearance to bite-type flareless fittings. However, the ferrule on this type of fitting does not bite into the

TABLE 11-5
Typical Tube Thickness Values Used with Various Types of Fittings

Fitting Type	Three-Piece 37 Degree Flared	Bite-Type Flareless	
Tube Materials	Steel, Stainless Steel, Copper, Aluminum	Stainless Steel, Nickel-Copper Alloy	Copper, Aluminum, Plastic
Tube Outside Diameter (Inches)	Tube Thickness (Inch)	Tube Thickness (Inch)	Tube Thickness (Inch)
1/8	0.010 to 0.035	0.010 to 0.035	0.012 to 0.028
3/16	0.010 to 0.035	0.020 to 0.049	0.012 to 0.035
1/4	0.020 to 0.065	0.028 to 0.065	0.020 to 0.049
5/16	0.020 to 0.065	0.028 to 0.065	0.020 to 0.065
3/8	0.020 to 0.065	0.035 to 0.095	0.028 to 0.065
1/2	0.028 to 0.083	0.049 to 0.120	0.035 to 0.083
5/8	0.035 to 0.095	0.058 to 0.120	0.035 to 0.083
3/4	0.035 to 0.109	0.065 to 0.120	0.035 to 0.095
7/8	0.035 to 0.109	0.072 to 0.120	0.049 to 0.095
1	0.035 to 0.109	0.083 to 0.148	0.049 to 0.120
1-1/4	0.049 to 0.120	0.095 to 0.188	—
1-1/2	0.049 to 0.120	0.095 to 0.203	—
2	0.058 to 0.134	0.095 to 0.220	—

Dimensions given are based on information included in Parker-Hannifin Corporation's *Design Engineer's Handbook* and are representative only. Fitting manufacturer's recommendations and specification requirements should always be considered for specific applications.

tubing. Instead, the ferrule compresses the end of the tubing as the fitting's nut is tightened.

PERMANENTLY SWAGED FITTING

A permanently swaged fitting is another type of compression fitting. During assembly, a straight piece of tubing is inserted into one end of the fitting. A special tool is then used to apply a compressive force to the outside of the fitting. As the fitting is compressed, or swaged, its inner surface contacts the outer surface of the tubing and forces the tubing to deform to match the fitting's shape.

HEAT-RECOVERABLE COUPLING

A heat-recoverable coupling (HRC) is manufactured from a material possessing "shape memory," such as nitonol, a nickel-titanium alloy. At ambient temperature, the coupling's inside diameter is slightly smaller than the outside diameter of the tube sections it will join. Prior to assembly, the coupling is cooled, typically to a temperature -150°F (-100°C), and is mechanically expanded until its inside diameter is larger than the tubing's outside diameter. The two tube ends being connected are then inserted into the coupling, and the coupling is allowed to warm to ambient temperature. Because of the shape memory, the coupling tries to return to its original shape; consequently, the coupling shrinks and grips the tubing forming a seal. In addition, circumferential ribs on the inside of the coupling form grooves in, and bite, the tubing (fig 11-7D. Some HRCs include a separate liner that is placed between the tubing and the outer coupling.

Hose Connections

A connector or fitting on the end of a hose can be permanently attached to the hose, or it can be reusable, i.e., it can be removed from a worn section of hose that is being discarded and then reassembled onto a new section of hose.

One end of a typical one-piece permanently attached hose connector (fig. 11-8a) has a nipple that is surrounded by an outer socket. When the connector is attached to a section of hose, the nipple is inserted into the end of the hose and the socket fits over the hose. The socket is then crimped or swaged so that it grips the hose. Barbs on the outer surface of the nipple dig into the hose's lining and increase the holding power of the connector.

During the assembly of a clamp-type reusable connector (fig. 11-8b), a barbed nipple on one end of the fitting is inserted into the end of a hose. Two halves of an external clamp are then bolted together around the outside of the hose and compress the hose against the outer surface of the nipple. Barbs on the inside of the clamp dig into the hose's outer cover as the clamp's bolts are tightened and increase the holding power of the connector. In some low-pressure applications, a circular clamp, referred to as a hose clamp, is used with a reusable plug instead of a two-piece bolted clamp.

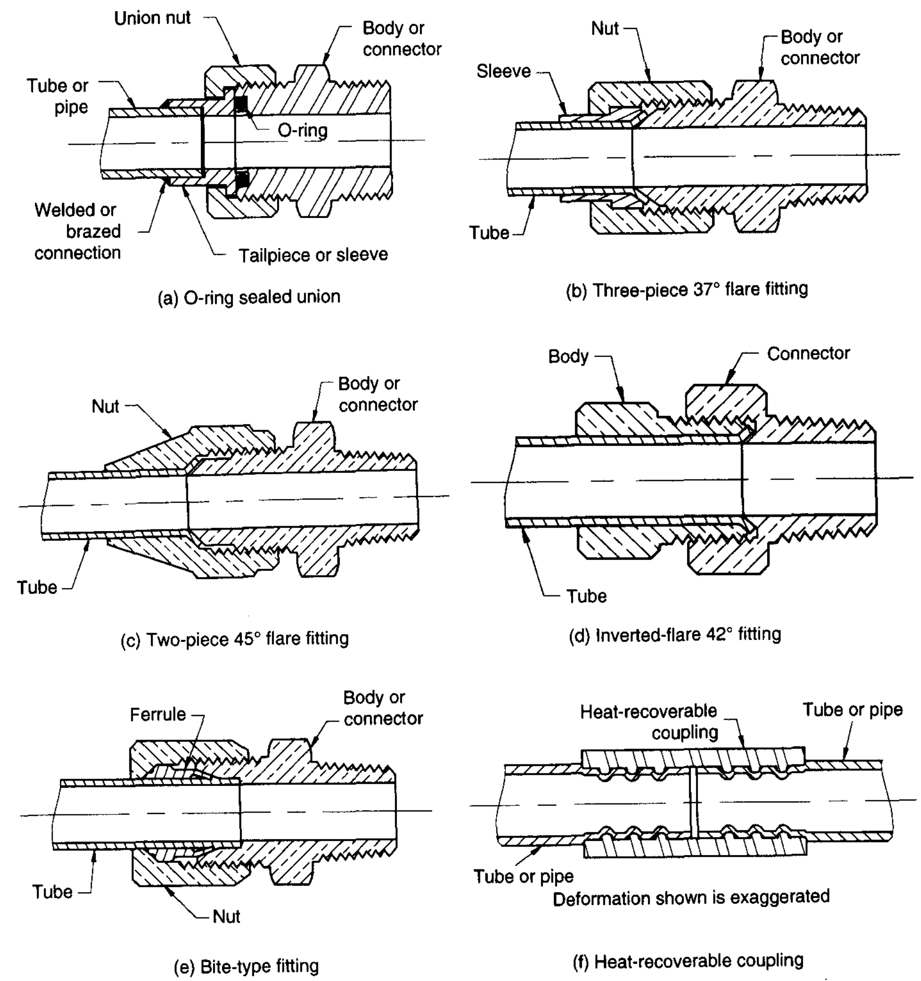


Figure 11-7. Tubing connectors

Prior to assembling a skive-type reusable connector (fig. 11-8c), which is typically used with hose having a thick outer cover, the cover must be removed from the end of the hose. The connector's outer socket is then screwed onto the uncovered end of the hose. Finally, the connector's nipple is inserted into the end of the hose and external threads at the base of the nipple are screwed into internal threads on the portion of the socket that hangs over the end of the hose. Although similar to a skive-type reusable connector, a no-skive reusable connector (fig. 11-8d) is used with hose having a thinner outer cover that does not have to be removed when the connector is attached to the hose.

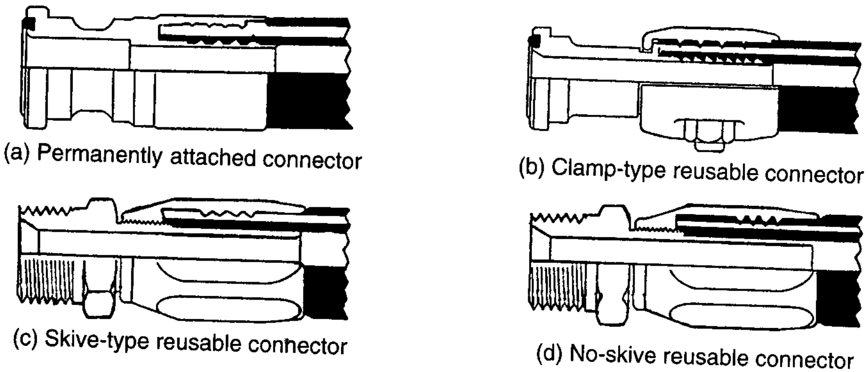


Figure 11-8. Hose connectors. Courtesy Parker-Hannifin Corporation.

The end of a hose fitting that mates with other components in a system (i.e., the end not connected to the hose) frequently has tapered external threads, straight external threads with an O-ring, external threads with an inverted flare at the end, a swivel nut having internal threads and a 37° or a 45° seat, a two-bolt swivel flange, a circular flange, or a split flange. Alternatively, when a hose will be repetitively connected and disconnected to one or more components, the fitting at the end of the hose may be one half of a quick-disconnect coupling.

Each half of one type of a quick-disconnect coupling has two claws that are 180° apart. When the two coupling halves are pressed together and one half is rotated, each claw engages a tapered lip on the back of the mating half and the two coupling halves are drawn together.

In another type of quick-disconnect coupling, the male half of the coupling, which is often referred to as the plug or nipple body, is inserted into the coupling's mating female half, which is sometimes referred to as the socket or coupler body. Although the two halves may be locked together by pins, pawls, cams, or dogs, the locking device used in many quick-disconnect couplings consists of a ring of small balls (similar to the balls in a ball bearing) located around the opening in the female half of the coupling that fit into a circumferential groove in the nipple portion of the coupling's male half. As the two halves of the coupling are pushed together during assembly, the balls are forced down into the nipple's groove by a spring-loaded sleeve that is mounted on the female half of the coupling. However, if the operator pushes the sleeve back against its spring and away from the male half of the coupling, the balls will no longer be held tightly in the groove. Consequently, if the male end of the coupling is then pulled away from the female half, the balls will be able to roll out of the groove in the nipple and the coupling halves will be disconnected.

The female half of a "single shutoff' quick-disconnect coupling (fig. 11-9a) is fitted with a spring-loaded valve. When the two halves of the coupling are joined, the end of the nipple on the male half of the coupling holds this valve off its seat so that fluid can flow through the coupling. When the coupling is disconnected, however, the valve closes and flow through the female half of the coupling, which is typically installed on the upstream or supply side of the system, is prevented. This type of a coupling is often used in pneumatic systems. When flow through both halves of a disconnected coupling must be prevented, a "double shutoff' quick-disconnect coupling (fig. 11-9b) may be used. This latter type of a coupling, which is frequently installed in hydraulic systems, has a spring-loaded valve in both its male and female halves. A "straight-through" quick-disconnect coupling does not have a valve in either of its two halves (fig 11-9c). Because of this,

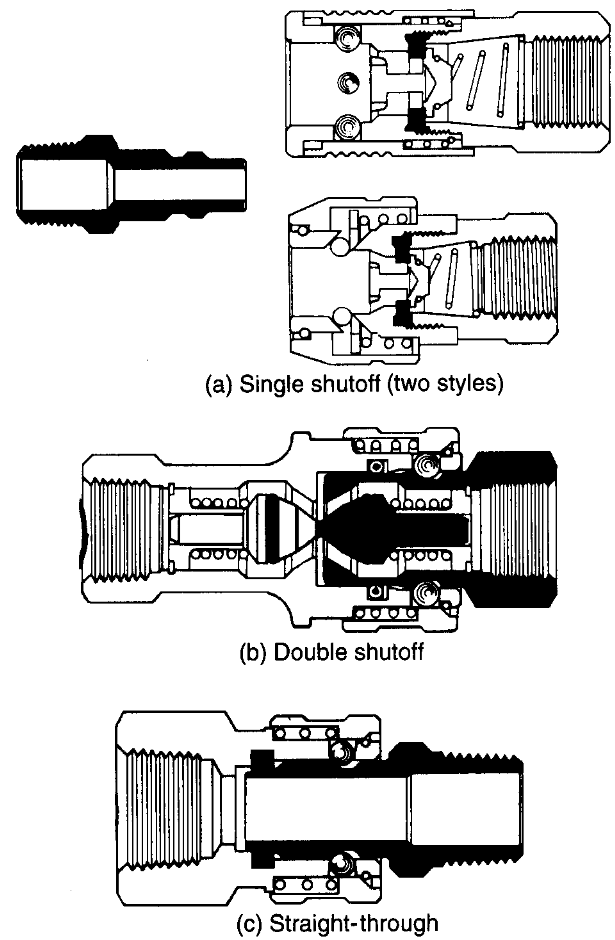


Figure 11-9. Quick-disconnect couplings.
Courtesy Parker-Hannifin Corporation.

ordinarily, there is a low pressure drop through a straight-through coupling; however, when flow must be prevented through the ends of a disconnected straight-through coupling, separate shutoff valves should be installed in the system.

Dimensions for American National fire hose connection screw threads (NH) are shown in table 11-6.

FITTINGS

Fittings are often employed to join sections of pipe, tubing, or hose that compose a system. Fittings used with pipe may have welded, brazed, flanged, or threaded end connections. Banded fittings have narrow bands or collars around their internally threaded outlets to increase strength. Typical fittings include those described below (see also figure 11-10).

Elbow

Also referred to as an ell, an elbow can be used to connect pipe sections at locations where there is a bend in the system. Standard elbows are available with bend angles of 22½°, 45°, 60°, or 90°. Both end connections have the same size and, when threaded, are typically both female. A short-radius elbow has a bend radius that is typically equal to the elbow's NPS, and a long-radius elbow has a bend radius that is ordinarily equal to 1½ times the elbow's NPS.

Reducing Elbow

A reducing elbow is similar to an ordinary elbow, except that one of a reducing elbow's end connections is smaller than the other. Reducing elbows are available with the same bend angles as ordinary elbows.

Street Elbow

A street elbow, or street ell, has one end connection with male threads and one with female threads. Street ells are available with the same angles as ordinary elbows. Additionally, reducing street ells are typically available with a male end connection that is smaller than the female connection.

Side-Outlet Elbow

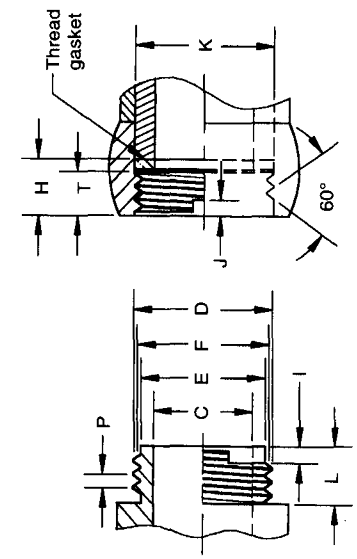
A side-outlet or three-way elbow, which resembles a 90° elbow with a third connection in the side of the bend, can be used to join three sections of a piping system where each piece of pipe is perpendicular to the other two pipes.

Side-Outlet-Reducing Elbow

A side-outlet-reducing elbow is similar to a side-outlet elbow, except that the side connection and one of the end connections are smaller than the opposite end connection. When identifying a side-outlet-reducing elbow, the size of the larger elbow end connection is given first, the size of the reduced

TABLE 11-6
Dimensions for American National Fire Hose Connection Screw Threads (NH)

Nominal Size of Hose	Number of Threads per Inch	Basic Thread Height	Inside Diameter of Nipple C	Approximate Outside Diameter of External Thread D	Maximum Minor Diameter of External Thread E	Maximum Pitch Diameter of External Thread F	Depth of Internal Connector H	Length of Pilot for Start of Second External Thread I[A]	Length of Pilot to Start of Second Internal Thread J[A]	Outside Diameter of Gasket Seat in Coupling K	Minimum Length of External Threads L	Pitch P	Depth of Internal Threads T
3/4	8	0.08119	3/4	1-3/8	1.2126	1.2938	19/32	5/32	5/32	1-7/16	5/8	0.12500	13/32
1	8	0.08119	1	1-3/8	1.2126	1.2938	19/32	5/32	5/32	1-7/16	5/8	0.12500	13/32
1-1/2	9	0.07217	1-1/2	2	1.8457	1.9178	19/32	5/32	5/32	2-1/16	5/8	0.11111	13/32
2-1/2	7-1/2	0.08660	2-1/2	3-1/16	2.8954	2.9820	15/16	1/4	3/16	3-3/16	1	0.13333	11/16
3	6	0.10825	3	3-5/8	3.4073	3.5156	1-1/16	5/16	1/4	3-3/4	1-1/8	0.16667	3/4
3-1/2	6	0.10825	3-1/2	4-1/4	4.0273	4.1356	1-1/16	5/16	1/4	4-3/8	1-1/8	0.16667	3/4
4	4	0.16238	4	5	4.6861	4.8485	1-3/16	7/16	3/8	5-1/8	1-1/4	0.25000	7/8
4-1/2	4	0.16238	4-1/2	5-3/4	5.4361	5.5985	1-3/16	7/16	3/8	5-7/8	1-1/4	0.25000	7/8
5	4	0.16238	5	6-1/4	5.9352	6.0976	1-5/16	7/16	3/8	6-3/8	1-3/8	0.25000	1
6	4	0.16238	6	7-1/32	6.7002	6.8626	1-5/16	7/16	3/8	7-1/8	1-3/8	0.25000	1

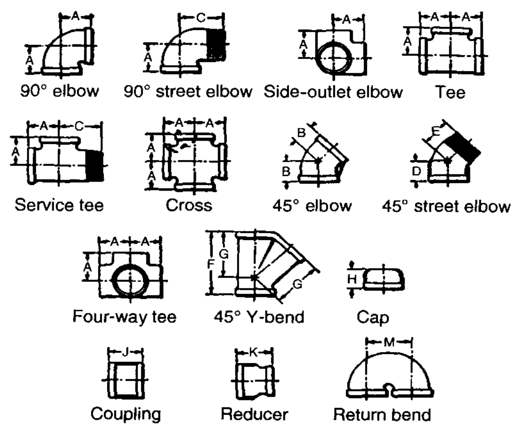


A. The pilot for both the internal and the external threads is referred to as a blunt start or a Higbee cut and is used to help prevent the crossing and mutilation of threads.

Notes:
1. All dimensions are in inches.
2. The width of the flat at the root and crest of each NH thread is equal to 0.125 × the Pitch (P).
3. The externally-threaded half of the connector is often referred to as the nipple, and the internally-threaded half is referred to as the coupling.
4. The above is based on information included in NFPA 1963 - 1993 Edition. Allowable tolerances are included in this specification.

Nominal Pipe Size (NPS)	Nominal Diameter (DN)	A			B			C		D		E		F		G		H (min.)		J^J		$K^{J,K}$		M Closed Pattern		M Medium Pattern	M Open Pattern	
		I	II	III	I	II	III	I	III^G	I	III	I	III	I	III	I	III	I^H	III^I	I^H	III^I	I	III^L	I	III	I	I	III
1/8	6	0.69	0.81	0.54	—	0.69	0.42	1.00^G	0.92	—	0.42	—	0.78	—	—	—	—	0.53	0.49	0.96	0.80	—	—	—	—	—	—	—
1/4	8	0.81	0.81	0.71	0.73	0.69	0.56	1.19	1.11	0.73	0.56	0.94	0.88	—	—	—	—	0.63	0.59	1.06	0.97	1.00	0.88	—	—	—	—	—
3/8	10	0.95	0.97	0.82	0.80	0.75	0.63	1.44	1.24	0.80	0.63	1.03	0.92	1.93	1.78	1.43	1.28	0.74	0.64	1.16	1.05	1.13	1.01	—	—	—	—	—
1/2	15	1.12	1.12	1.01	0.88	0.88	0.78	1.63	1.48	0.88	0.78	1.15	1.06	2.32	2.19	1.71	1.58	0.87	0.76	1.34	1.29	1.25	1.17	1.00	1.00	1.25	1.50	1.50
3/4	20	1.31	1.31	1.18	0.98	1.00	0.89	1.89	1.65	0.98	0.89	1.29	1.23	2.77	2.62	2.05	1.90	0.97	0.84	1.52	1.43	1.44	1.36	1.25	1.25	1.50	2.00	2.00
1	25	1.50	1.50	1.43	1.12	1.12	1.06	2.14	1.98	1.12	1.06	1.47	1.40	3.28	3.18	2.43	2.33	1.16	0.99	1.67	1.68	1.69	1.56	1.50	1.50	1.87	2.50	2.50
1-1/4	32	1.75	1.75	1.69	1.29	1.31	1.22	2.45	2.24	1.29	1.22	1.71	1.64	3.94	3.85	2.92	2.83	1.28	1.10	1.93	1.86	2.06	1.77	1.75	—	2.25	3.00	3.00
1-1/2	40	1.94	2.00	1.84	1.43	1.38	1.30	2.69	2.46	1.43	1.30	1.88	1.80	4.38	4.24	3.28	3.14	1.33	1.15	2.15	1.92	2.31	1.89	2.19	—	2.50	3.50	3.50
2	50	2.25	2.38	2.12	1.68	1.69	1.45	3.26	2.88	1.68	1.45	2.22	2.14	5.17	5.00	3.93	3.76	1.45	1.32	2.53	2.20	2.81	2.06	2.62	—	3.00	4.00	4.00
2-1/2	65	2.70	3.00	2.70	1.95	2.06	1.95	3.86	—	1.95	—	2.57	—	6.25	—	4.73	—	1.70	1.70	2.88	2.88	3.25	3.25	—	—	—	4.50	—
3	80	3.08	3.38	3.08	2.17	2.50	2.17	4.51^G	—	2.17	—	3.00	—	7.26	—	5.55	—	1.80	1.80	3.18	3.18	3.69	3.69	—	—	—	5.00	—
3-1/2	—	3.42	—	—	2.39	—	—	—	—	—	—	—	—	—	—	—	—	1.90	—	—	—	—	—	—	—	—	—	—
4	100	3.79	4.19	3.79	2.61	3.12	2.61	5.69	—	2.61	—	3.70	—	8.98	—	6.97	—	2.08	2.08	3.69	3.69	4.38	4.38	—	—	—	6.00	—
5	125	4.50	—	—	3.05	—	—	6.86^G	—	—	—	—	—	—	—	—	—	2.32	—	—	—	—	—	—	—	—	—	—
6	150	5.13	—	—	3.46	—	—	8.03^G	—	—	—	—	—	—	—	—	—	2.55	—	—	—	—	—	—	—	—	—	—

Nominal Pipe Size (NPS)	Nominal Diameter (DN)	Wall Thickness			Minimum Inside Diameter of $Fitting^D$		Minimum Band Outside Diameter			Minimum Band Length		Minimum Internal Thread Length		Minimum External Thread $Length^F$		Maximum Port $Opening^F$	
		$I^{A,B}$	II^C	III^A	I	III	I	II	III	I	III	I & III	II^E	I	III	I	III
1/8	6	0.09	0.125	0.08	0.40	0.41	0.69	0.88	0.67	0.20	0.14	0.25	0.25	0.26	0.27	0.20	0.22
1/4	8	0.09	0.125	0.08	0.54	0.54	0.84	0.88	0.81	0.21	0.16	0.32	0.32	0.40	0.41	0.26	0.28
3/8	10	0.10	0.125	0.09	0.67	0.68	1.01	1.00	1.00	0.23	0.17	0.36	0.36	0.41	0.41	0.37	0.40
1/2	15	0.10	0.125	0.09	0.84	0.84	1.20	1.31	1.17	0.25	0.19	0.43	0.43	0.53	0.54	0.51	0.53
3/4	20	0.12	0.125	0.10	1.05	1.05	1.46	1.50	1.42	0.27	0.23	0.50	0.50	0.55	0.55	0.69	0.72
1	25	0.13	0.145	0.11	1.31	1.32	1.77	1.81	1.72	0.30	0.27	0.58	0.58	0.68	0.69	0.91	0.93
1-1/4	32	0.14	0.153	0.12	1.66	1.66	2.15	2.19	2.10	0.34	0.31	0.67	0.67	0.71	0.71	1.19	1.25
1-1/2	40	0.15	0.158	0.13	1.90	1.90	2.43	2.44	2.38	0.37	0.34	0.70	0.70	0.72	0.73	1.39	1.47
2	50	0.17	0.168	0.15	2.37	2.38	2.96	2.97	2.92	0.42	0.41	0.75	0.75	0.76	0.76	1.79	1.91
2-1/2	65	0.21	0.221	0.17	2.87	2.88	3.59	3.62	3.49	0.48	0.48	0.92	0.93	1.14	—	2.20	—
3	80	0.23	0.236	0.19	3.50	3.50	4.28	4.31	4.20	0.55	0.55	0.98	1.02	1.20	—	2.78	—
3-1/2	—	0.25	—	—	4.00	—	4.84	—	—	0.60	—	1.03	—	—	—	—	—
4	100	0.26	0.258	0.22	4.50	4.50	5.40	5.75	5.31	0.66	0.66	1.08	1.09	1.30	—	3.70	—
5	125	0.30	—	—	5.56	—	6.58	—	—	0.78	—	1.18	—	1.41	—	4.69	—
6	150	0.34	—	—	6.62	—	7.77	—	—	0.90	—	1.28	—	1.51	—	5.67	—



A. The actual metal thickness must typically be at least 90% of the value listed above. The minimum wall thickness at the top of a cap may exceed the values given.
B. Does not apply to return bends.
C. Minimum wall thickness.
D. Does not apply to male end of street elbows and service (street) tees.
E. The values given for forged-steel fittings are the minimum lengths of perfect threads.
F. Applies to male end of street elbows and service (street) tees.
G. Applies to street elbows only. Service (street) tees are not typically made in these sizes.
H. NPS 3/8 and smaller Class 150 caps and straight couplings may be manufactured from steel rod having a minimum yield strength of 30,000 psi. Steel fittings have dimensions that are different from those shown above.
I. NPS 3/4 and smaller Class 125 straight couplings and caps may be manufactured from brass rod. These fittings have dimensions that are different from those shown above.
J. Right-hand couplings typically have no more than two ribs. Right-and left-hand couplings typically have four or more ribs unless the left-hand opening is clearly marked with an "L" in which case the fitting may or may not have ribs.
K. The reducing-coupling (reducer) length is shown in the row corresponding to the size of the coupling's larger end. The minimum band outside diameter and thread length in this row apply only to the large end of the coupling. The band outside diameter and thread length for the coupling's smaller end, which may be up to three sizes smaller than the large end, can be found in the row corresponding to the smaller end's size.
L. Dimensions shown only apply to reducing couplings (reducers) that reduce one size. Reducing couplings that reduce two or three sizes have lengths that are different from those shown above.

Notes:

1. Except for the column labeled Nominal Diameter (DN), all dimensions are in inches. DN sizes are in millimeters and are given for reference only.
2. Dimensions in columns marked with an "I" are for Class 150 Malleable-Iron Threaded Banded Fittings and are based on information included in ANSI/ASME B16.3-1992.
3. Dimensions in columns marked with an "II" are for Class 2000 Forged-Steel Threaded Fittings and are based on information included in ANSI/ASME B16.11-1996.
4. Dimensions in columns marked with an "III" are for Class 125 Cast-Bronze Threaded Fittings and are based on information included in ANSI/ASME B16.15-1985 (R-1994).
5. Tolerances on the dimensions listed above are included in the referenced specifications.
6. Typical pressure and temperature limits for these fittings are as follows:

Temperature (F)	Maximum Allowable Pressure (psig) for Metal Temperatures Not Exceeding the Values Shown -20°F to 150°F	200°F	250°F	300°F	350°F	400°F
Class 150 Malleable-Iron	300 psig	265 psig	225 psig	185 psig	150 psig*	—
Class 125 Cast-Bronze or Brass	200 psig	190 psig	180 psig	165 psig	150 psig	125 psig

* Class 150 Malleable-Iron Fittings are typically suitable for use with pressures up to 150 psig at temperatures up to 366°F.
The maximum allowable pressure for Class 2000 Forged-Steel Threaded Fittings, which can be manufactured in various grades of carbon or alloy steel, should be based on the limits for straight seamless threaded Schedule 80 pipe with the same NPS as the fitting and manufactured from an equivalent material. The maximum allowable temperature should be based on applicable specification limits for the fitting's material.

7. Malleable-iron fittings are often available either black or galvanized.
8. Dimensions for side-outlet and four-way elbows are not included in the referenced specifications.
9. Internal and external threads are typically American National Standard Taper Pipe Threads (NPT). (An exception to this can apply to Class 150 caps and straight couplings manufactured from steel bar and Class 125 caps and straight couplings manufactured from brass bar, which sometimes have straight threads.)
10. Additional classes of threaded fittings that are covered in the referenced specifications include Class 250 Malleable-Iron, Class 3000 and 6000 Forged-Steel, and Class 250 Cast-Bronze Threaded Fittings. Dimensions for these fittings are typically different from those shown above.
11. The information given above is representative only. Applicable specifications should always be consulted for dimensions and limitations for actual fittings.

Figure 11-10. Representative dimensions for selected pipe fittings elbow end connection is given second, and the size of the side connection is given last. In addition, when a side-outlet-reducing elbow is oriented with its side connection on the front and the larger elbow end connection directed downward, if the smaller elbow end connection points to the right, the fitting can be classified as a right-hand side-outlet-reducing elbow. When a left-hand side-outlet-reducing elbow is oriented in the same fashion, its smaller elbow end connection will point to the left.

Return Bend

A return bend is used to make a 180° turn in piping. Threaded return bends typically have female connections. A back-outlet return bend has a third connection in the center of the bend and can be used to connect two pipes to a third pipe.

Tee

A tee is used to add a right-angle branch to a straight section or run of piping. Tees are typically identified by giving the sizes of the two end connections in the straight run of the fitting followed by the size of the branch or side connection. When the tee is threaded, all three openings typically have female connections.

SERVICE TEE

A service or street tee is similar to a regular tee. However, a service tee has two female connections, and a third male connection that is usually in the run of the fitting.

REDUCING TEE

A reducing tee can have a reduced size connection at one end of the run, in the branch, at both of these locations, or at both ends of the run. When identifying a tee with a reduced run connection, the nominal size of the larger run connection is given first, the size of the reduced run connection is given second, and the size of the branch connection is given last.

SIDE-OUTLET TEE

A side-outlet tee resembles a standard tee with an additional fourth connection in the side of the tee that is perpendicular to the other three connections. The use of a side-outlet tee permits two separate branches that are 90° apart to be connected to a run of piping.

SIDE-OUTLET-REDUCING TEE

A side-outlet-reducing tee is similar to a side-outlet tee, except that one or more of the connections are reduced in size. When identifying a side-outlet-reducing tee, the size of the larger run connection is given first, the size of the second run connection is given second, the size of the larger

branch connection is given third, and the size of the remaining branch connection is given last. In addition, when a side-outlet-reducing tee is oriented with its smaller branch connection on the front and its larger run connection directed downward, if the reduced branch connection points to the right, the fitting can be classified as a right-hand side-outlet-reducing tee. If a left-hand side-outlet-reducing tee is oriented in the same fashion, its smaller branch connection will point to the left.

Y-BEND

A Y-bend, or Y-branch, which may also be referred to as a lateral or angle tee, has a branch connection that is at an acute angle (usually either 45° or 60°) with the run connection. In a reducing Y-bend, the size of any of the three end connections may be reduced. In addition, a threaded Y-bend typically has only female connections. The sequence for giving the size of each of a Y-bend's connections is identical to that used for a tee. In addition, the angle between the run and the branch is specified.

Cross

A cross is used to connect two branches that are 180° apart to a straight run of piping. Additionally, the two branches are typically perpendicular to the run. All four connections on the cross may be the same size, or some may be reduced. A threaded cross typically has only female connections. When identifying the size of a cross, the size of one run connection (the larger run connection when the two run connection sizes are different) is given first, the size of the opposite run connection second, the size of one branch connection (the larger branch when the two branch sizes are different) third, and the size of the remaining branch connection last.

Nipple

A nipple is a short length of pipe (e.g. ⅛ in. to 12 in. or 3 mm to 305 mm long) that is used to join fittings or components. A close nipple is the shortest type and has external threads along its entire length. A short nipple is slightly longer than a close nipple; in addition, the threads on each end of a short nipple are separated by a short unthreaded shoulder in the center of the nipple. With the exception of having a longer unthreaded shoulder between its threaded ends, a long nipple is similar to a short nipple. Long nipples are supplied in lengths up to 12 inches (305 mm). A tank nipple, which can be used to connect a section of pipe to a tank, has a short external thread on one end and a long external thread on the opposite end that, when assembled, passes through the tank wall. The two sets of threads are separated by an unthreaded shoulder.

Most nipples have right-hand threads at both ends. However, when neither of the two fittings or components being joined by a nipple can be turned, a right-and-left-hand nipple with external right-hand threads on one end and external left-hand threads on the opposite end may be used. These two sets of threads are often separated by a hexagonally-shaped shoulder so that the nipple, referred to as a hexagon-center nipple, can be gripped more easily with a wrench. Although it is less common, one or both ends of a nipple may remain unthreaded to facilitate use with socket-welding fittings.

Coupling

A coupling is a sleeve with internal threads at each end that is used to join two pieces of pipe. Although many couplings have right-hand threads at both ends, a right-and-Ieft-hand coupling has right-hand threads at one end and left-hand threads at the other end. This latter type of coupling can be used to join two sections of pipe that can not be turned during assembly.

Reducing Couplings

The size of the connection at one end of a reducing coupling is smaller than that at the other end. A reducing coupling can, therefore, be used to connect two pieces of pipe that are not the same size. Both outlets in a standard reducer are concentric. An eccentric or offset reducer is arranged so that the centerline of the smaller opening is offset from the centerline of the larger opening.

Bushing

A bushing, which has both an external thread and an internal thread, is used to reduce the size of a fitting's internally-threaded outlet so that the fitting can be connected to a smaller externally-threaded fitting or pipe. The external surface at one end of a bushing is typically shaped like a hexagon so that the bushing can be gripped with a wrench during assembly or disassembly.

Union

A typical union consists of two end pieces or outlets that are each connected to the end of a pipe or fitting. Although the two end pieces are frequently joined by a special nut, in a flange union, bolts are used to join the two end pieces together. The use of a union permits the connection between two sections of a piping system to be assembled or dissembled without turning either section of the system. Although some unions have a gasket or O-ring to prevent leakage at the joint formed between the two end pieces, a union may rely on the fit between the mating surfaces of its end pieces to form a fluid-tight seal. Various types of unions are shown in figure 11-11.

Plug and Cap

A plug is an externally threaded fitting and a cap is an internally threaded fitting; both are frequently used to close the open ends of pipes or fittings.

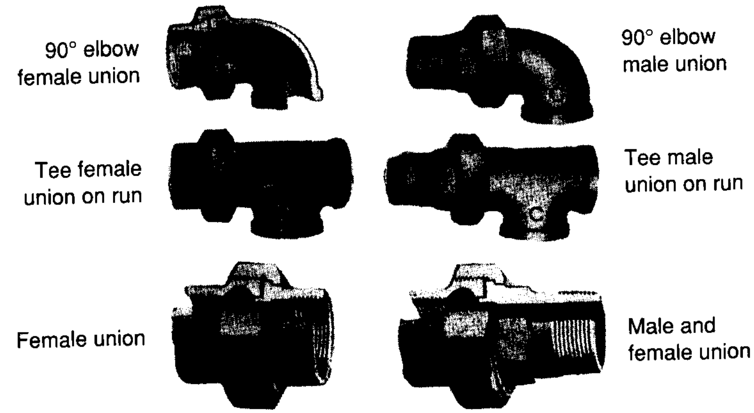


Figure 11-11. Unions

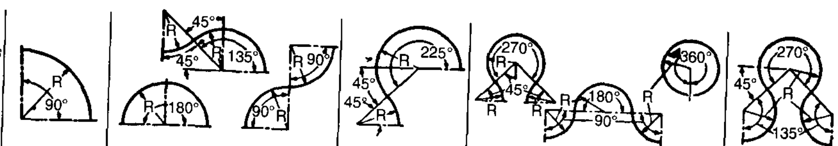
Plugs can be supplied with square or hexagonal heads or they can be countersunk.

PIPE BENDS

Piping is often bent to reduce the number of required elbows and joints in a system. In addition, bends and loops may be added to a piping system to increase flexibility, avoid obstructions, and reduce stress from expansion, contraction, or misalignment. When a pipe or tube is bent, the outer portion of the bend stretches, which results in a reduction in wall thickness. In addition, the inner portion of the bend is compressed and thickens. Furthermore, the cross section of the pipe or tube in the area of the bend becomes elliptically shaped. Pipe or tube that is bent should be long enough to permit straight runs, referred to as tangents, to precede and follow the bend so that the ends of the pipe or tube, which must typically be joined to other portions of the system, will not be distorted by the bend. The wall thickness of the thinnest part of a bend should never be less than the minimum wall thickness required in straight runs of the piping before and after the bend. Also, out-of-roundness should be kept within specified criteria. A typical bend radius, measured to the piping centerline, is usually from 2.5 to 6 times a pipe's or a tube's nominal size (referred to as 2.5D to 6D). However, using special booster-bending machines, bend radii as small as 1.5D to 2D are sometimes achievable (with regulatory-body approval, when applicable). Because the amount of thinning caused by bending increases as the bending radius is reduced, the minimum acceptable bending radius for piping of a given diameter can typically be reduced if the initial wall thickness of the piping is increased.

The lengths of various types of pipe bends are shown in table 11-7. The bend radii given in this table are measured to the piping centerline. In addition, the lengths given do not include the straight sections or tangents that should be included at either end of the bend. With a suitable bending machine, pipe sizes up to NPS 12 (DN 305) often can be bent at ambient temperature. Typically, force is applied to the pipe by bending forms or dies. To reduce the potential for flattening and buckling, the inside of the pipe may be supported with mandrels or it may be packed with sand. When a suitable cold-bending machine is not available or when bending larger-sized pipe, hot bending may be employed. Steel pipe is typically heated in the range of 1,900° to 2,05(l°F 0,038° to 1,121°C) prior to hot bending. The pipe may be packed with sand to increase its rigidity; however, the effectiveness of sand to prevent buckling is reduced as the ratio of pipe diameter

TABLE 11-7
Length of Pipe in Various Types of Bends



Radius in Inches					
5″	7¾″	15¾″	23½″	31½″	47¼″
6	9½	18¾	28¼	37¾	56½
7	11	22	33	44	66
8	12½	25¼	37¾	50¼	75½
9	14¼	28¼	42½	56½	84¾
10	15¾	31½	47¼	62¾	94¼
11	17¼	34½	51¾	69¼	103¾
12	18¾	37¾	56½	75½	113
13	20½	40¾	61¼	81¾	122½
14	22	44	66	88	132
15	23½	47¼	70¾	94¼	141½
20	31½	62¾	94¼	125¾	188½
25	39¼	78½	117¾	157	235½
30	47¼	94¼	141½	188½	282¾
35	55	110	165	220	330
40	62¾	125¾	188½	251¼	377
45	70¾	141½	212	282¾	424
50	78½	157	235½	314¼	471¼
60	94¼	188½	282¾	377	565½
70	110	220	330	439¾	659¾
80	125¾	251¼	377	502¾	754
90	141½	282¾	424	565½	848¼
100	157	314¼	471¼	628¼	942½
110	172¾	345½	518½	691¼	1,036¾
120	188½	377	565½	754	1,131
130	204¼	408½	612½	816¾	1,225¼
140	220	439¾	659¾	879¾	1,319½
150	235½	471¼	706¾	942½	1,413¾

divided by initial wall thickness increases. After being heated, force is applied (often by a winch) to one end of the pipe while the opposite end is restrained. When multiple bends are required, the pipe should usually be reheated for each additional bend.

VALVES

Valves are used to stop, divert, or regulate the flow rate of fluids through piping systems. Various types of valves are described below.

Linear-Shaft Valve

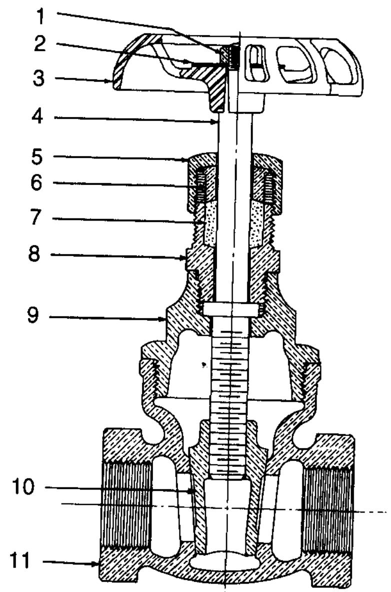
A valve is classified as a linear-shaft or reciprocating valve when the closure element must be lowered into the valve body to restrict or stop flow. Common types of linear-shaft valves include the following.

GATE VALVE

As shown in figures 11-12a through 11-12f, a typical gate valve consists of a body having in-line inlet and outlet connections, a bonnet, and a gatelike disk that is attached to a threaded stem. The bonnet may be screwed directly into or onto the valve body (fig. 11-12a), secured to the valve body with a threaded union ring (fig. 11-12b), held in place by a V-shaped bolt or clamp that fits around the bottom of the valve body, which is usually limited to moderate-pressure applications (fig. 11-12c), or flanged and attached to the valve body with bolts or studs and nuts (fig. 11-12d). In addition, in some high-pressure and high-temperature applications, the bonnet is welded to the valve body (fig. 11-12e). Alternatively, when a pressure-seal bonnet is used, the bonnet is forced against a wedge-shaped metal gasket by pressurized fluid within the valve (fig. 11-12D.

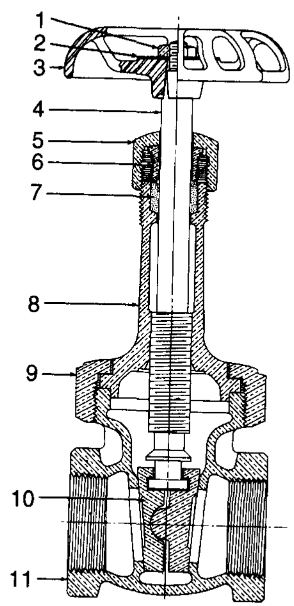
A bonnet that is flanged or secured with a union ring is often fitted with a gasket to prevent leakage. Leakage through the opening provided for the valve stem is ordinarily controlled by packing that is inserted into a stuffing box in the bonnet and is compressed by a nut or a gland. A lantern ring may be installed between two of the packing rings to permit fluid from an external source to be injected into the stuffing box (usually for cooling or flushing).

When fully open, the resistance to flow through a gate valve is generally relatively low. However, it is difficult to accurately control flow rates through a gate valve. In addition, fluid flow through a partially opened gate valve can result in vibration and chattering of the gate. This type of operation can also lead to erosion and cutting of the gate and the mating seats in the valve body. Consequently, a gate valve should ordinarily not be used to throttle flow but should be used only in applications where it will be either fully open or fully closed, i.e., as a stop valve.



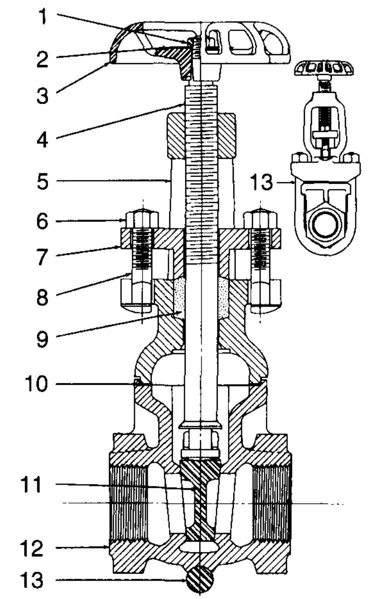
1 Handwheel locknut
2 Indentification plate
3 Handwheel
4 Stem
5 Packing nut
6 Gland
7 Packing
8 Packing box
9 Bonnet
10 Wedge
11 Body

Figure 11-12a. Nonrising-stem, screw-in bonnet, one-piece (hollow) wedge disk, inside screw gate valve. Courtesy Crane Company.



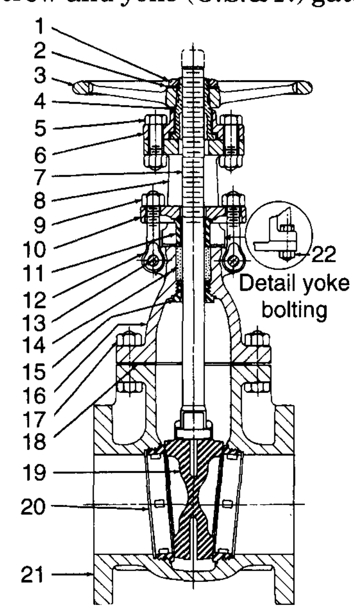
1 Handwheel locknut
2 Identification plate
3 Handwheel
4 Stem
5 Packing nut
6 Gland
7 Packing
8 Bonnet
9 Union bonnet ring
10 Wedge
11 Body

Figure 11-12b. Rising-stem, union bonnet, split-wedge disk, inside screw gate valve. Courtesy Crane Company.



1 Handwheel locknut
2 Identification plate
3 Handwheel
4 Stem
5 Yoke bonnet
6 Gland bolt nut
7 Gland
8 Gland bolt
9 Packing
10 Bonnet gasket
11 Wedge
12 Body
13 U-bolt and nut

Figure 11-12c. Rising-stem, U-bolt bonnet, one-piece (solid) wedge disk, outside screw and yoke (O.S.&Y.) gate valve. Courtesy Crane Company.



1 Yoke bushing nut
2 Identification plate
3 Handwheel
4 Yoke bushing
5 Yoke cap bolt and nut
6 Yoke cap
7 Stem
8 Yoke
9 Gland eye bolt nut
10 Gland flange
11 Gland
12 Gland eye bolt
13 Gland lug bolt and nut
14 Packing
15 Bonnet bushing
16 Bonnet
17 Bonnet bolt and nut
18 Bonnet gasket
19 Wedge
20 Seat ring
21 Body
22 Yoke bolt and nut

Figure 11-12d. Rising-stem, flanged bonnet, flexible-wedge disk, O.S.&Y. gate valve. Courtesy Crane Company.

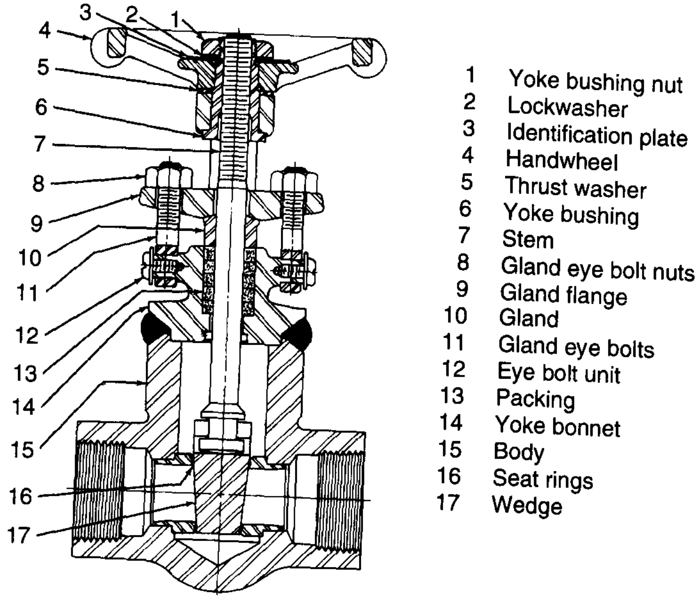


Figure 11-12e. Rising-stem, welded bonnet, one-piece (solid) wedge disk, O.S.&Y. gate valve. Courtesy Crane Company.

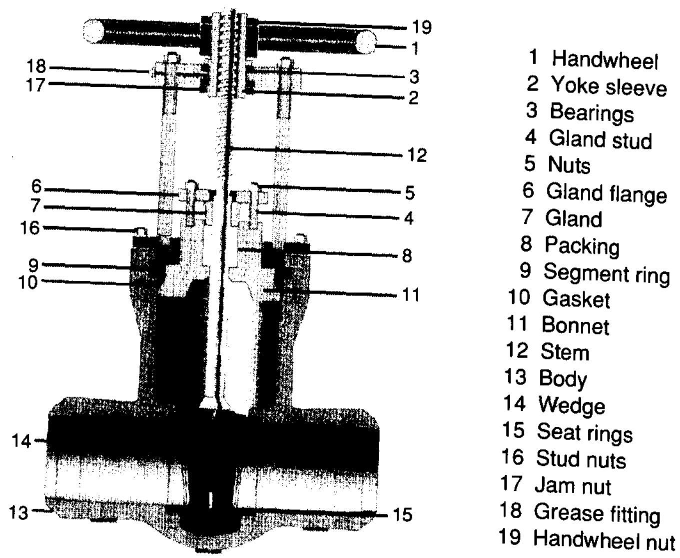


Figure 11-12f. Rising-stem, pressure-seal bonnet, flexible-wedge disk, O.S.&Y. gate valve. Courtesy Crane Company.

Many gate valves have a one-piece wedge-shaped disk (figs. 11-12a, c, and e). When this type of a gate valve is closed, the disk fits between two tapered seats in the valve body. However, if a valve is closed when hot fluid is passing through it, the downstream side of the body will frequently cool and contract while the upstream side of the disk is still being heated. As a result of this uneven temperature distribution, the disk can become jammed between its seats and a great deal of force may be required to reopen the valve. In addition, the valve's sealing surfaces can be damaged by galling when the valve is being reopened. A similar situation can occur if the upstream side of the valve is exposed to hot fluid while the valve is tightly closed. Also, the force exerted on the upstream face of a closed gate valve's disk by fluid at the supply pressure can make it difficult to open the valve. To reduce the potential for sticking, a portion of a one-piece tapered disk is sometimes split (referred to as a flexible-wedge or flex-wedge disk, figs. 11-12d and D to provide some flexibility between the disk's two tapered sealing faces. The required disk thickness increases with a gate valve's pressure rating; consequently, the actual flexibility of a single flexible-wedge disk is limited in high-pressure valves. Alternatively, some gate valves have a two-piece disk with either tapered sides (referred to as a split-wedge disk [fig. 11-12b]) or parallel sides (referred to as a double disk). As a gate valve with a two-piece disk is being closed, an internal spreader often separates the two disks during the final turn of the handwheel and forces each disk's sealing surface against the adjacent seat in the valve body. 80 that the disks will properly hang on the end of the valve stem, double-disk gate valves must usually be installed with the external end of the stem pointing up. To permit stationary sealing surfaces in a gate valve's body to be periodically renewed, some gate valves are fitted with replaceable seat rings (figs. 11-12d, e, and D.

In a rising-stem gate valve with an outside screw and yoke (abbreviated 0.8.& Y.), the upper end ofthe valve stem is threaded and passes through a nut that is held in the yoke located above the valve's stuffing box. The handwheel is often attached to the yoke nut (figs. 11-12d, e, and D. With this arrangement, as the handwheel-and-yoke-nut assembly is turned to open the valve, the stem, which does not rotate, rises through the center of the yoke and the handwheel. A grease fitting is sometimes provided to permit the yoke-nut assembly to be lubricated. Although it is less common, the handwheel in a rising-stem outside-screw gate valve may be fixed to the external end ofthe valve stem (fig. 11-12c). With this latter arrangement, as the handwheel is turned to open the valve, it rotates and rises with the valve stem. With an outside-screw design, the threads on the valve stem are isolated from fluid flowing through the valve. In a rising-stem gate valve with an inside screw, the threaded portion of the valve stem engages internal threads in the valve's bonnet (fig. 11-12b). As the handwheel, which is attached to the top of the valve stem, is turned to open the valve, the unthreaded portion of the stem, together with the attached handwheel, rises through the top of the bonnet. The connection between the disk in a rising-stem gate valve and the valve's stem generally has a relatively loose fit to prevent side loads applied to the disk by fluid in the valve from being transmitted to the stem. The use of a stem with a T-shaped head that fits into a mating slot in the disk is common.

In a nonrising-stem gate valve, the handwheel is attached to the external end of a valve stem that is threaded to the valve's gate (fig. 11-12a). As the handwheel is turned to open the valve, the gate is drawn up onto the internal portion of the stem.

An advantage of a rising-stem valve is that an operator can determine whether the valve is open or closed based on the length of the portion of the valve stem that is protruding from the yoke or bonnet. However, a position indicator (fig. 11-13) can be mounted on a nonrising-stem gate valve to enable the operator to easily determine how far the valve has been opened. In addition, a nonrising-stem valve typically requires less headroom than a valve with a rising stem. Also, because a rising stem slides against the valve-stem packing as the valve is being opened or closed, packing in a rising-stem valve can wear more rapidly than packing in a comparably-sized nonrising-stem valve.

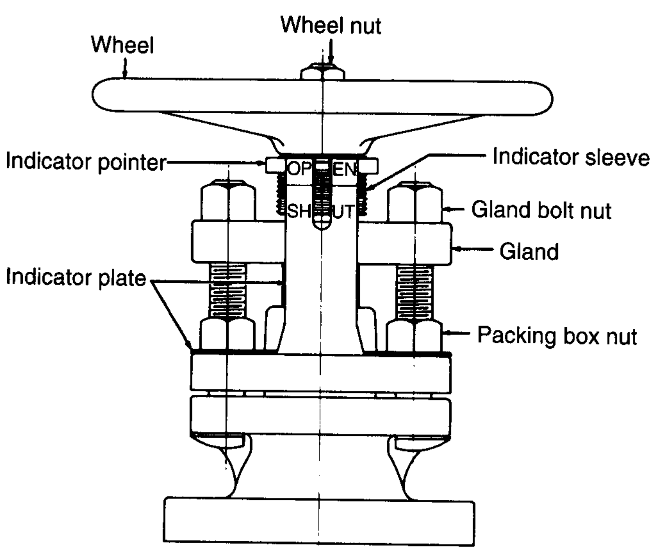


Figure 11-13. Valve-position indicator. Courtesy Crane Company.

GLOBE VALVE

As shown in figures 11-14a through 11-14d, a globe valve is fitted with a disk that is mounted on the end of a threaded stem. To close the valve, the stem is screwed into the bonnet, which forces the sealing surface on the disk against a mating seat in the valve's globular-shaped body. With the exception of the clamp-type bonnet arrangement, which is not used with globe valves, methods utilized for bonnet attachment and stem sealing are similar to those previously described for gate valves.

Although globe valves can be used as stop valves, they can also be used to throttle fluid flow through a piping system. To enable the valve to be quickly opened or closed, a globe valve, which ordinarily will be fully opened when the amount that its disk is moved away from the valve's seat is equal to the diameter of the port in the valve's seat divided by four, can generally be fully opened or closed with fewer turns of its handwheel than a comparably sized gate valve. However, because fluid flowing through a typical globe valve must change direction as it passes through the valve body, the pressure drop within a globe valve is ordinarily greater than the pressure drop in a comparably-sized gate valve.

Most globe valves are constructed with a handwheel attached to the external end of the valve stem; consequently, the stem rises as the valve is opened. Threads on the valve stem, however, may be outside of the valve and engage threads in a yoke nut (fig. 11-14a), or they may be inside of the valve and engage threads in the bonnet (figs. 11-14b-d). The rotation of the disk against the valve seat when a globe valve is being opened or closed can damage the valve's sealing surfaces; consequently, th~ disk-to-stem connection should permit the disk to stop turning when it comes in contact with the valve seat. However, the fit between the disk and the stem should be close enough for the stem to help guide the disk squarely to the seat while the valve is being closed. A guide pin that protrudes from the bottom of the disk and passes through a hole in the center of a rib that bridges the seat ring is sometimes added in larger globe valves to provide additional guidance for the disk (fig. 11-14a).

When a globe valve fitted with a typical beveled disk is closed, there is a narrow line of contact between the tapered face of the disk and the mating surface of the seat (fig. 11-14a). This thin line of contact can help to break up deposits that may form on the seat. Alternatively, some globe valves have a plug-type disk with a long taper that, when closed, creates a wide area of contact with the mating seat (fig. 11-14c). Because of this large sealing area, minor damage caused by erosion (especially during throttling) or contaminants in the fluid passing through the valve is less likely to result in leakage across the seat when the valve is closed. Additionally, a globe valve may be fitted with a composition disk, which has a replaceable insert ring with a flat face that covers the opening in the seat when the valve is closed (fig. 11-14b). The disk's insert is often made from an elastomeric material. To enable the stationary sealing surface in a globe valve to be renewed periodically, the seat is frequently part of a replaceable ring that is screwed into the valve's body (figs. 11-14a-c).

It is common for a globe valve to be installed so that fluid enters the body below the seat. With this arrangement, the valve's stuffing box is isolated from fluid at the supply pressure when the valve is closed. In addition, the inlet pressure pushing up on the disk can reduce the force required to open the valve. However, when being used in a high-pressure system, a globe valve installed in this fashion may be difficult to close. Consequently, in a high-pressure application, a globe valve may be installed so that fluid enters the body above the seat.

Globe valves are typically constructed with in-line inlet and outlet connections. In addition, the axis of the valve stem in a standard globe valve is perpendicular to the common centerline of the valve's connections. The valve stem in a Y-pattern globe valve, however, is at an angle of 45° with respect to the centerline of the body's inlet and outlet connections (fig. 11-14d).

LEVER-OPERATED GLOBE VALVE

A lever-operated globe valve, which is typically designed to be quick opening and self-closing, has a lever-actuated valve stem that slides back-and-forth without rotating (fig. 11-15). When the lever is forced against the external end of the stem, the stem is moved into the valve body and pushes the disk away from the seat, which opens the valve. When the lever is released, a spring pushes the disk back against the seat and closes the valve.

ANGLE VALVE

An angle valve is similar in design and method of operation to a globe valve. However, unlike a standard globe valve with in-line inlet and outlet connections, the centerline of the inlet connection in an angle valve is perpendicular to the centerline of the outlet connection (fig. 11-16). Consequently, an angle valve can eliminate the need for a separate elbow when there is a 90° bend in a piping system. The pressure drop through an angle valve is typically less than that in a comparably-sized standard globe valve.

NEEDLE VALVE

A needle valve has a cone-shaped or needle-shaped disk, which is usually an integral part of the valve stem, that fits into the seat in the valve body when the valve is closed (fig. 11-17). Needle valves, which are generally

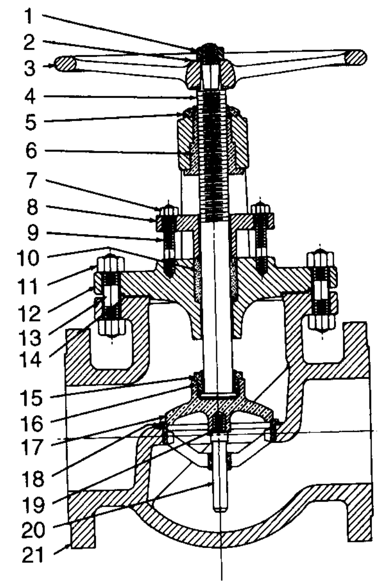


Figure 11-14a. Outside screw and yoke (O.S.&Y.) globe valve. Courtesy Crane Company.

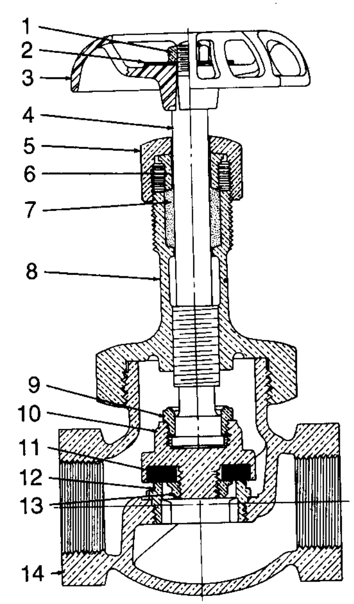
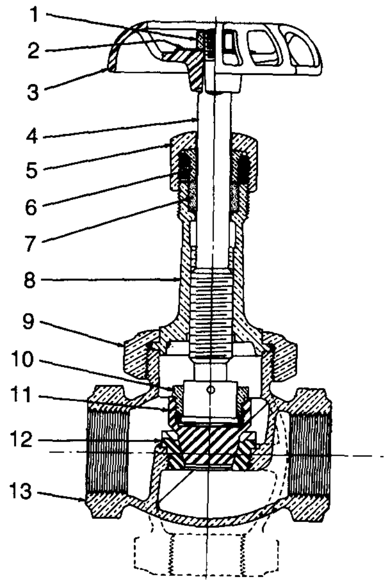
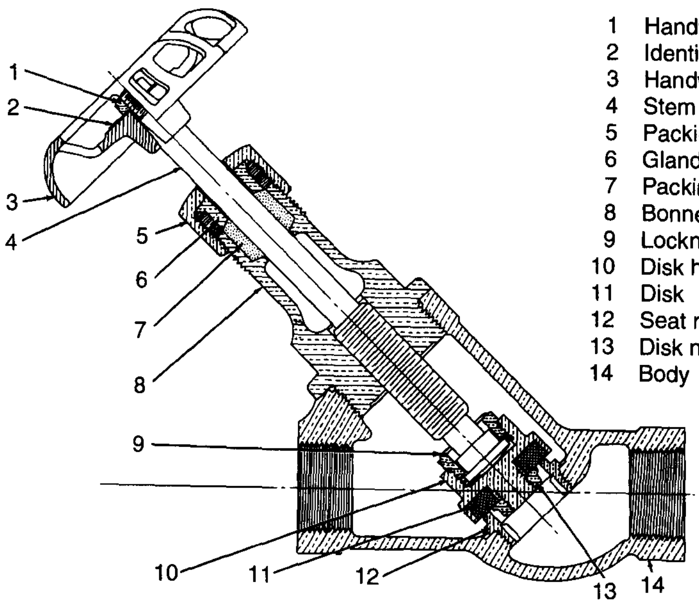


Figure 11-14b. Inside screw, composition disk globe valve. Courtesy Crane Company.



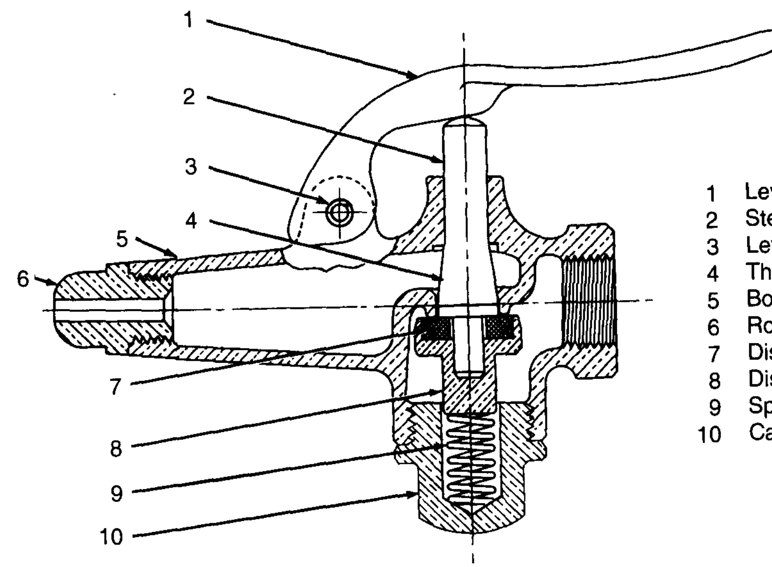
1 Handwheel locknut
2 Identification plate
3 Handwheel
4 Stem
5 Packing nut
6 Gland
7 Packing
8 Bonnet
9 Union bonnet ring
10 Locknut
11 Plug disk
12 Seat ring
13 Body

Figure 11-14c. Plug disk globe valve. Courtesy Crane Company.



1 Handwheel locknut
2 Identification plate
3 Handwheel
4 Stem
5 Packing nut
6 Gland
7 Packing
8 Bonnet
9 Locknut
10 Disk holder
11 Disk
12 Seat ring
13 Disk nut
14 Body

Figure 11-14d. Y-pattern globe valve. Courtesy Crane Company.



1 Lever handle
2 Stem
3 Lever pin
4 Throttling plug
5 Body
6 Round tip
7 Disk
8 Disk holder
9 Spring
10 Cap

Figure 11-15. Lever-operated globe valve. Courtesy Crane Company.

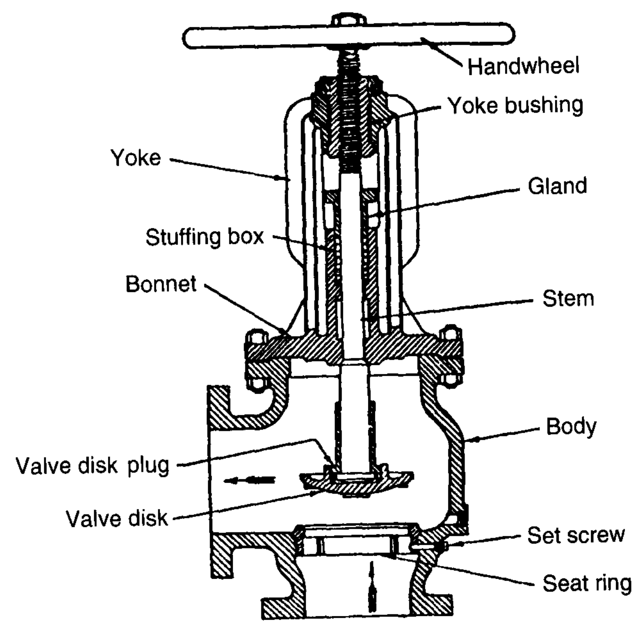


Figure 11-16. Angle valve

available only in small sizes, are used primarily in applications where accurate flow control is required.

DIAPHRAGM VALVE

A diaphragm valve has a flexible diaphragm that forms the upper pressure boundary of the valve's body. The diaphragm is forced to deflect by the movement ofa stem-mounted piece called the compressor. When the stem is screwed into the valve, the diaphragm is pressed against the seat in the bottom of the valve's body and flow through the valve is prevented. When the stem is screwed in the opposite direction, the diaphragm is pulled away from the seat and flow through the valve can resume. The seat in many diaphragm valve bodies is formed by the top edge of an internal weir, or dam, that partially obstructs fluid flow when the valve is open. Alternatively, a straight-through diaphragm valve has a body with no weir. To increase corrosion resistance, a diaphragm valve's body is often constructed from a nonmetallic material or is lined with an elastomer. Because the stem in a diaphragm valve is completely isolated from the fluid passing through the valve, there is no leakage path around the stem; consequently, stem packing is not required.

SPOOL VALVE

The movable element in a spool valve resembles one or more pistons that are connected by reduced-diameter sections. As the spool is moved within

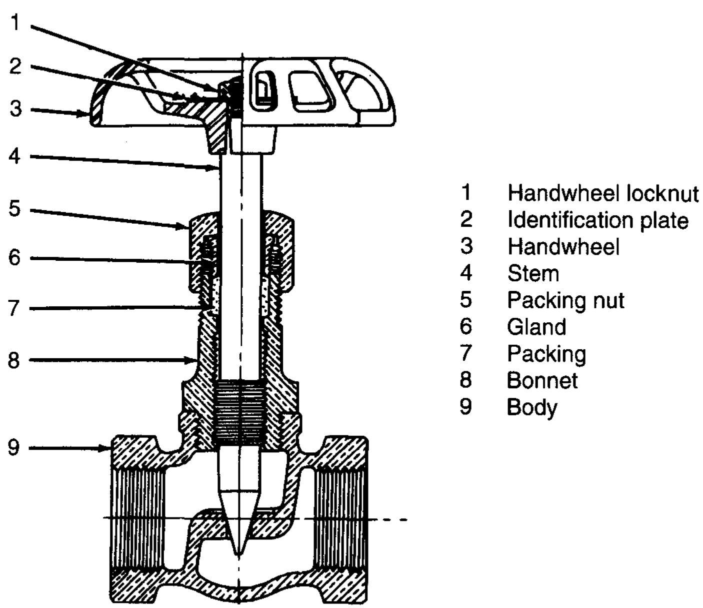


Figure 11-17. Needle valve. Courtesy Crane Company.

the valve's body, it can block or uncover ports in the body to direct flow along the desired path. The spool can be actuated by a lever, pressurized fluid, or a magnetic field.

Rotary-Shaft Valves

Rotary-shaft valves can be fully opened or closed by rotating the closure element through only a portion of a complete turn, usually 90°. Types of rotary-shaft valves are described below.

BUTTERFLY VALVE

As shown in figure 11-18, a butterfly valve has an internal disk (with a diameter approximately equal to the inside piping diameter) that can be rotated from a fully closed position (perpendicular to the flow) to a fully opened position (parallel to the flow) with a quarter turn of the attached stem. The disk can also be rotated to an intermediate position to throttle the flow through the valve.

In a "general purpose" symmetric butterfly valve, the valve stem passes through the centerline of the disk. When the valve is closed, a seal is typically created by the tight fit between the periphery of the disk and a resilient liner installed in the valve body. The single-piece liner, which may also be used to increase the corrosion and erosion resistance of a butterfly valve's body, ordinarily wraps over the valve body's end flanges and eliminates the need for separate gaskets between the valve and the attached piping. In addition, a butterfly valve's disk may be coated to increase its resiliency, corrosion resistance, or erosion resistance.

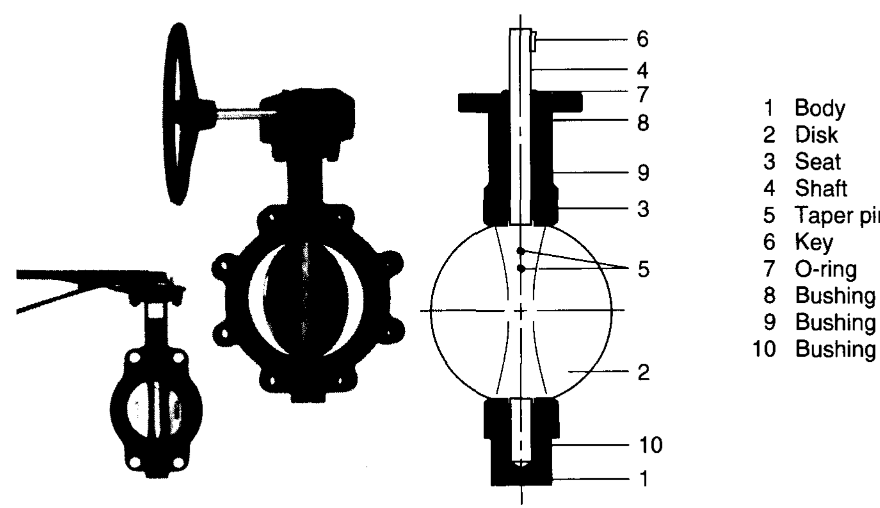


Figure 11-18. Butterfly valve. Courtesy Crane Company.

In a "high-performance" eccentric butterfly valve (HPBV), the stem is offset from the centerline of the disk's sealing surface. With this arrangement, the seating surface in the valve body is not interrupted by a penetration for the stem. In addition, when the valve is opened, the sealing surface of the disk, which is usually tapered or rounded and may have a replaceable seal ring, is drawn away from the seat rather than being dragged over it. Also, to improve shutoff capability, an HPBV's body is often fitted with a pressure-assisted seat ring that is forced against the disk by fluid in the supply side of the system when the valve is closed.

The opening betweep a butterfly valve's stem and the valve's body is frequently sealed with packing that is inserted into a stuffing box. Alternatively, when a resilient body liner is used, the primary seal for the stem may be produced by the close fit between the liner and the top of the disk. Additionally, an O-ring may be installed between the valve stem and the body to form a secondary seal. A bearing is often fitted at each end of the stem.

A butterfly valve may be furnished with a wafer body, which is designed to be sandwiched between two flanges in a piping system while being centered inside the flange-bolt circle. When a wafer valve body is fitted with a resilient liner, gaskets between the valve and the piping flanges are ordinarily not required. A lug-wafer body, which is essentially a wafer body with lugs that are drilled and, in some cases, tapped to permit the valve to be mounted on one ofthe piping flanges, permits the valve to be used to seal the end ofthe system when the piping on one side ofthe valve is disassembled for maintenance. A two-flanged butterfly-valve body is flanged at each end and can be bolted to the piping flanges on both sides of the valve. Gaskets should generally be installed between the body of this latter type of butterfly valve and the mating pieces of pipe.

Forces created by fluid flowing around a conventional butterfly-valve disk can increasingly act to close the valve when it is 60 to 70 percent open. These forces can then rapidly diminish when the valve is 75 percent open. The resulting variation in the torque necessary to position the disk can make precise valve control difficult. To reduce the effect that valve position has on closing force, some butterfly valves have a dynamically balanced disk with a small protrusion on its downstream edge. The flow rate, velocity, and pressure on each side of this type of a disk are roughly balanced, which reduces the fluid force acting to close the valve.

PLUG VALVE

A plug valve has a stem-mounted plug that resembles either a truncated cone (referred to as a tapered plug) or a cylinder. The plug in a two-way valve (fig. 11-19) has a single hole, which is usually oblong-shaped, passing through it. When this hole is in line with the flow path, fluid can pass through the valve. If, however, the valve stem is rotated 90°, the hole in

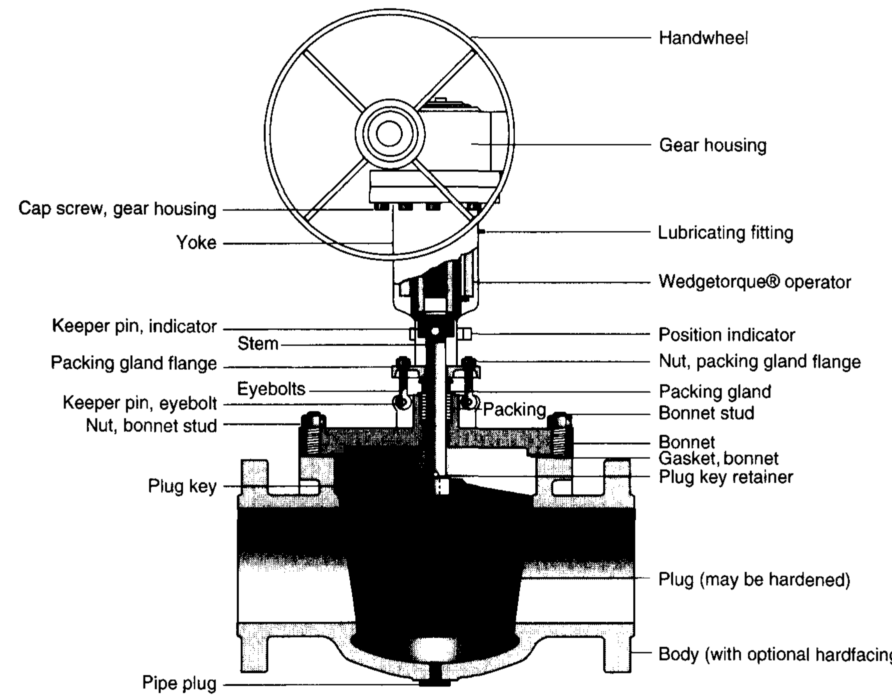


Figure 11-19. Two-way plug valve. Courtesy Crane Company.

the plug will be perpendicular to the flow path and the valve will be closed. Plug valves are also available in multi port configurations that can be used to direct flow in more than one direction. A three-way plug valve has a body with three piping connections and a plug having either an L-shaped or a T-shaped passage; a four-way plug valve has a body with four piping connections and a plug having two unconnected L-shaped passages. Although most plug valves have poor throttling characteristics, special plug valves have been designed that are suitable for use in throttling applications.

A plug valve is ordinarily furnished with a top-entry body having a bonnet that can be removed to install or remove internal parts. Packing is frequently used to seal the opening in the bonnet for the valve stem.

In a lubricated plug valve, which is usually of all-metal construction, grease is injected through a fitting in the valve stem, passes through a nonreturn check valve, is distributed through internal ports in the plug, and enters grooves machined into the plug's outer surface. The grease, which must be compatible with the fluid in the system, helps to both lubricate the plug and seal the clearances between the plug and the valve body. Grease must often be added to a lubricated plug valve every time the valve is opened or closed.

A nonlubricated plug valve may have a body that is fitted with a plastic sleeve (usually PTFE) against which the plug seats, a fully plastic-lined body with a plastic-coated plug, or an all-plastic plug. The use ofthe plastic materials typically reduces friction between the plug and the seat. Alternatively, a nonlubricated plug that is tapered may be designed to be lifted slightly and held away from the valve body's seating surfaces while being rotated, which eliminates rubbing contact between the plug and the seat. The plug is then set back down against the seat after the valve stem has been turned. Rubbing contact between the plug and seat is also eliminated in an eccentric-plug valve. In this type of valve, a partial plug, which is generally made from a resilient material, is mounted eccentrically on the valve stem and is drawn away from the seat while the valve is being opened.

BALL VALVE

A typical two-way ball valve has a spherically shaped plug, or ball, with a round hole passing through it that can be moved from a fully opened position (with the hole in line with the flow) to a fully closed position (with the hole perpendicular to the flow) by rotating the valve's stem 90° (fig. 11-20). The ball can also be rotated to an intermediate position to throttle the flow through the valve; however, the impingement of the resulting high-velocity flow against the valve's partially exposed seat can lead to erosion ofthe sealing surfaces. Consequently, ball valves are generally not satisfactory for use in throttling applications. The body of a two-way ball valve

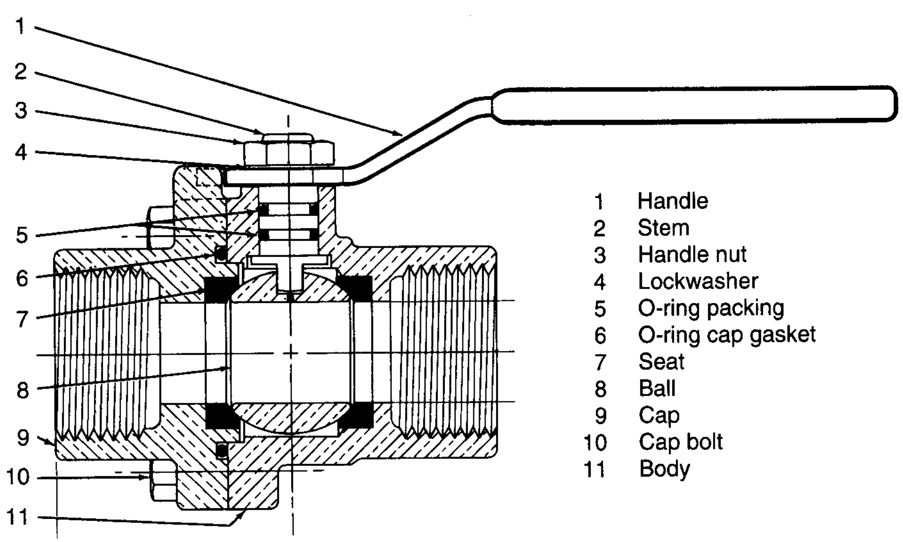


Figure 11-20. Ball valve. Courtesy Crane Company.

ordinarily has in-line inlet and outlet ports. A three-way ball valve has a third port located in the side or bottom of its body.

A ball valve's stem may be fitted with a thrust washer to reduce wear and the force necessary to operate the valve. In addition, when a ball valve has a blowout-proof stem, which is required by many specifications, a shoulder or pin is provided on the internal portion of the stem to prevent the stem from being blown out of the valve body. Packing rings or O-rings are frequently used to seal the opening for the stem in the valve body.

A typical ball-valve body is fitted with seat rings against which the ball seals. In a floating-ball valve, the ball is supported by and rotates between the two seat rings. When the valve is closed, fluid in the supply side of the system forces the ball against the downstream seat and increases the valve's shutoff capability. In some cases, slots are cut into the periphery of the seat rings to permit fluid from the inlet side of the valve to leak into the valve body when the valve is closed. The resulting equalization of the pressure acting on both sides of the supply-side seat ring reduces the torque necessary to open the valve. The internal end ofthe valve stem in a floating-ball valve frequently has a tongue that engages a mating slot or groove in the ball. In a trunnion-ball valve, the top and bottom of the ball are supported by trunnion stubs that rotate within bushings. With this latter arrangement, which must generally be used with larger-sized balls, the seat rings in the valve body may be spring loaded to increase the contact pressure between these rings and the ball. Fluid pressure may also increase the sealing force when a trunnion ball valve is closed. In another ball-valve design, a cam mechanism is used to force the ball into its seat when the valve is closed and to lift the ball away from its seat when the valve is being opened.

The ball furnished with a ball valve may have (listed in order of decreasing size) a full port, which is equal to the inside diameter of the piping to which the valve is connected, a regular port, or a reduced port. Many ball valves have a one-piece end-entry body into which all of the internal parts are installed through one ofthe two end ports. Alternatively, however, a ball valve may be furnished with a top-entry body and a bonnet that can be removed to install or remove internal parts. This latter arrangement eliminates the need to remove the valve body from a piping system when maintenance must be performed on the valve's internal parts. The seat rings in a top-entry ball valve are often installed at angles so that there is a slight taper betw,een their seating surfaces. This taper provides guidance to the ball during valve assembly. A spring is sometimes mounted on the valve stem to firmly push the ball into the wedge-shaped space between the tapered seat rings and increase the effectiveness of the seal created when the valve is closed. In addition to end-entry and top-entry bodies, radially split two-piece and three-piece ball-valve bodies are sometimes used.

LEVER-GATE VALVE

A lever-gate valve has a gate that can be rotated into a position across or away from the flow opening by moving an attached lever (fig. 11-21). This type of valve, which is designed for quick opening and closing, is often used at the top of a sounding tube. The lever is sometimes spring loaded so that the valve will be self-closing when the lever is released.

Linear- and Rotary-Shaft Valve Operators

It is typical and often required by regulatory bodies that a valve's stem must be rotated in a clockwist! direction (when facing the external end of the stem) to close the valve and rotated in a counterclockwise direction to open the valve. The external end of a manually-operated linear-shaft valve's stem is often fitted with a handwheel. However, if additional force is necessary for tight seating or to initially open a larger valve or a valve installed in a high-pressure system, a hammer-blow wheel (fig. 11-22a) may be used. When opening a valve fitted with a hammer-blow wheel, the wheel is rotated quickly so that one of its lugs strikes a lug on the valve stem with a sharp impact. This process can be repeated until the valve stem turns freely. The reverse procedure can be followed when closing the valve. (Tightening a valve excessively can damage its sealing surfaces and should be avoided.) In some cases, the movement of a handwheel may be transmitted to the valve stem through a gearbox (fig. 11-22b). By using a step-down gear arrangement between the handwheel and the valve stem, the torque that must be provided by the operator to open or close the valve is reduced. The use of a gearbox can also enable the handwheel to be oriented at an angle (e.g., 90°) with respect to the valve stem. Mter fully opening a linear-shaft valve, the stem should gen-

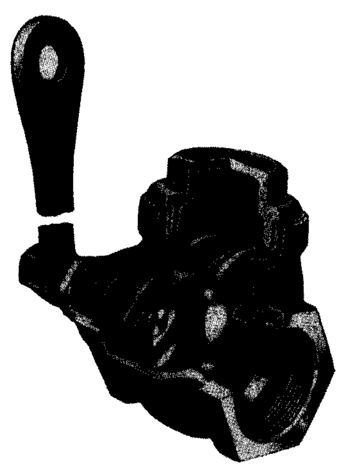


Figure 11-21. Lever-gate valve. Courtesy The Lunkenheimer Company.

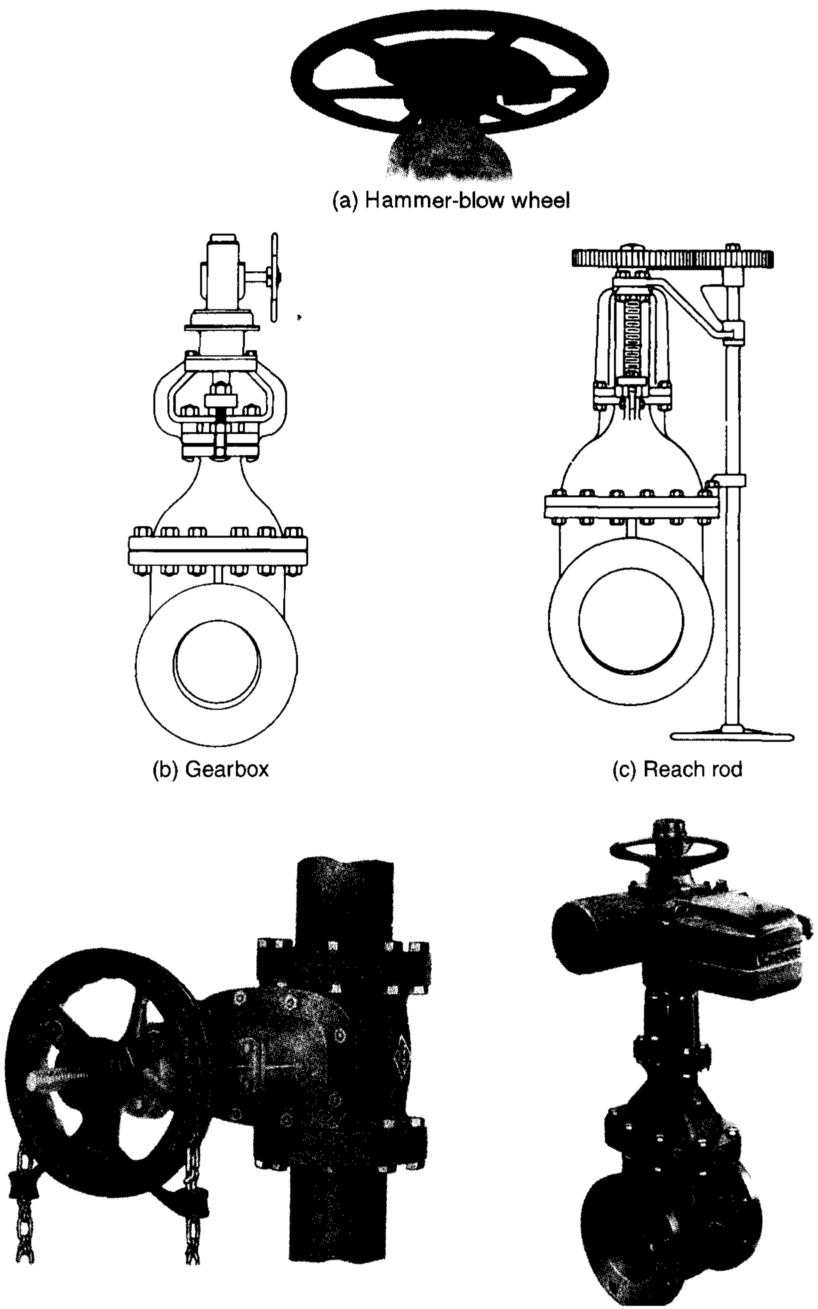
erally be rotated approximately one-quarter turn in the clockwise direction so that the back of the valve's seat will not be forced against the inside of the bonnet. This does not apply to valves that are designed to be back seated when opened to prevent leakage around the valve stem, such as the valves used in many refrigeration systems.

The external end of a manually-operated rotary-shaft valve's stem is often fitted with a lever or a tee that can be used to rotate the stem. Additionally, a lever will often have a spring-loaded pin or trigger so that it can be locked in place. It is typical and often required by regulatory bodies that the handle on a two-way valve be in line with the piping when the valve is open and perpendicular to the piping when the valve is closed. A multi port valve may have a groove cut in the top of the valve stem or some other indicator to show the flow path through the valve when the stem is rotated to different positions. A manually-operated rotary-shaft valve may also be opened and closed by rotating a handwheel that is perpendicular to the piping and connected to the valve stem through a gearbox.

When a valve is located in a hard-to-reach location, an extended reach rod may be mounted on the handwheel to enable the operator to adjust the valve more easily (fig. 11-22c). Alternatively, when a valve with a horizontal stem is located too far above the deck to be reached, a chain wheel may be mounted on the stem (fig. 11-22d). By pulling on one side of a chain that is suspended from the stem-mounted wheel, the operator can adjust the valve.

When a valve is used to control the flow of a flammable fluid, such as fuel oil, a linkage or flexible cable that extends to a location outside of the machinery space may be attached to the valve's handwheel so that the valve can be closed remotely in the event of a fire or emergency. Also, a large valve or a valve that must routinely be operated remotely is often fitted with an electric, pneumatic, or hydraulic motor that is geared or coupled to the stem (fig. 11-22e). When used with a rising-stem linear valve, a remote valve operator frequently is stopped by a motion limit switch when opening the valve and by a torque limit switch when closing the valve, which protects the valve from being damaged if a foreign object prevents complete closure. When used with a 90° rotary valve, however, a motion limit switch generally stops the operator both when opening and when closing the valve. Remote valve operators are frequently fitted with a handwheel for local operation if the motor fails. A lever must ordinarily be moved to declutch the motor before the handwheel can be used.

To equalize the pressure and temperature on both sides of a closed valve's disk, a small bypass line (e.g., NPS % to IX or DN 18 to 40) with a globe valve in it is frequently fitted around a larger gate, globe, or angle valve installed in a high-pressure steam system. In some cases, the bypass may be an integral part of the larger valve. When a bypass valve is provided, it should be slowly opened prior to opening the larger valve. In addition, the bypass valve should be closed after the larger valve has been fully opened.



(a) Hammer-blow wheel

(b) Gearbox

(c) Reach rod

(d) Chain wheel

(e) Motor

Figure 11-22. Valve operators. Courtesy Crane Company.

Check Valves

A check valve is used to prevent reverse flow through a line. Although there are many different types, the typical check valve is opened by the pressure offluid entering the inlet side ofthe valve's body and can be closed by fluid entering the valve's outlet port, gravity, and, in some cases, a spring. However, when the velocity or pressure of fluid entering a check valve is too low, the valve may only partially open or it may not open at all. An arrow indicating the direction offlow is often marked on a check valve's body.

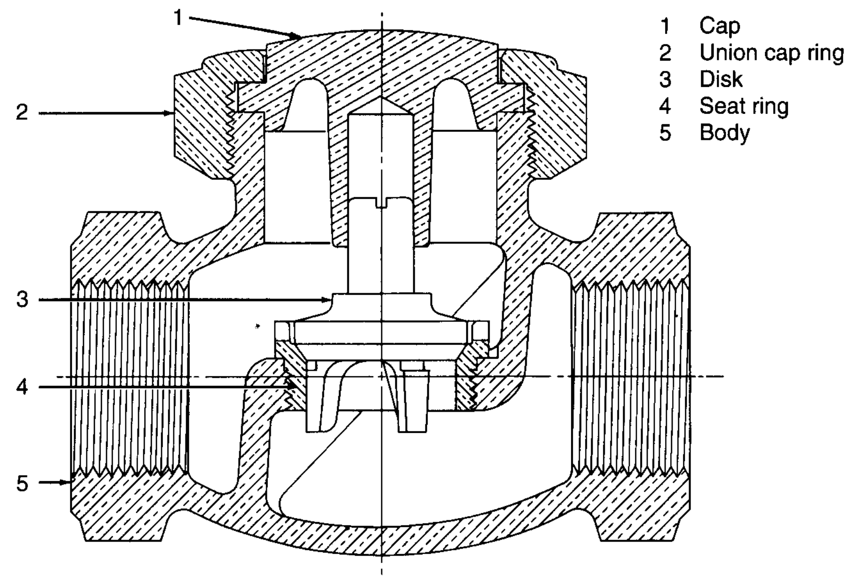
LIFT-CHECK VALVE

A typical lift-check valve has an internal piston or disk similar to the type used in a globe valve that rests against a flat or tapered seat in the lower part of the valve's body (fig. 11-23). Fluid that enters the inlet side of the valve is directed to the underside ofthe disk. When this fluid has sufficient pressure, it lifts the disk offthe seat and flows through the body to the outlet port on the opposite side of the valve. If the flow of fluid into the valve stops, gravity (fig. 11-23a) and, in some cases, a spring (fig. 11-23b) will force the disk to drop down onto the seat and close the valve. In addition, if fluid enters the outlet port and attempts to flow through the valve in the reverse direction, it will force the disk down against the seat.

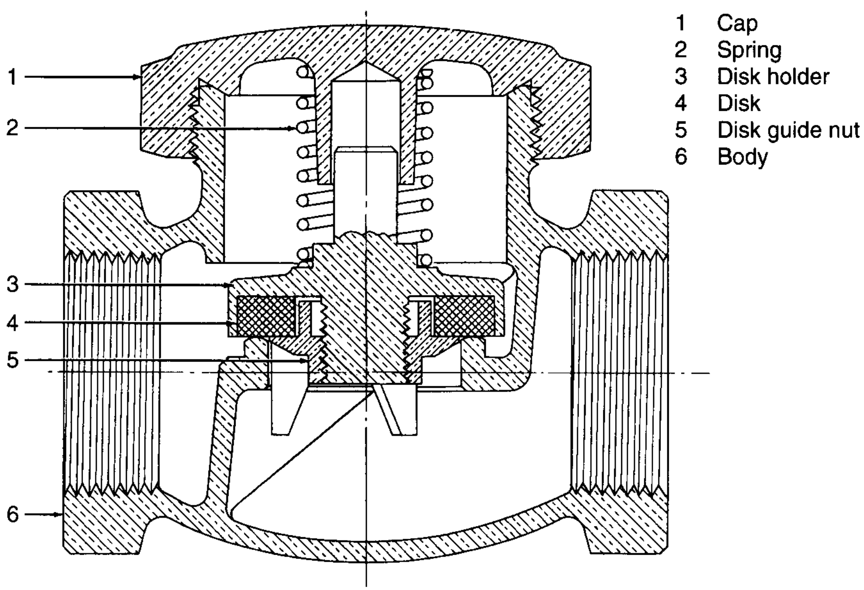
Guide pins are often provided to prevent the disk from becoming misaligned as it moves up and down. In addition, stops may be provided to prevent the disk from being rotated by fluid flowing through the valve. Many lift-check valve bodies are fitted with replaceable seat rings (fig. 11-23a). In addition, a renewable composition disk may be used (figs. 11-23b and c). Access to the disk and seat can generally be achieved by removing a bonnet that is either screwed on or bolted to the top ofthe body. A small orifice may be added to the disk in a larger-sized lift-check valve to reduce the pressure difference acting on the disk and the speed at which the disk seats. This can enable the valve to operate satisfactorily with pulsating flows. In addition, although many lift-check valves are only suitable for use in horizontal pipe lines, lift-check valves with a bottom inlet and top outlet (fig. 11-23c) or with a Y-pattern body and a spring-loaded disk are ordinarily suitable for use in vertical lines with upward flow. Lift-check valves may also be furnished with an angle body having the inlet connection in the bottom of the body and the outlet on the side.

BALL-CHECK VALVE

A ball-check valve is essentially a lift-check valve with a ball in place of a disk (fig. 11-24). When a ball valve with a bottom inlet and a top outlet (fig. 11-24a) is mounted in a vertical line (with upward flow), the ball can be lifted off the valve's seat by fluid entering the bottom of the valve body. Once the valve opens, the fluid flows past the ball and exits the valve

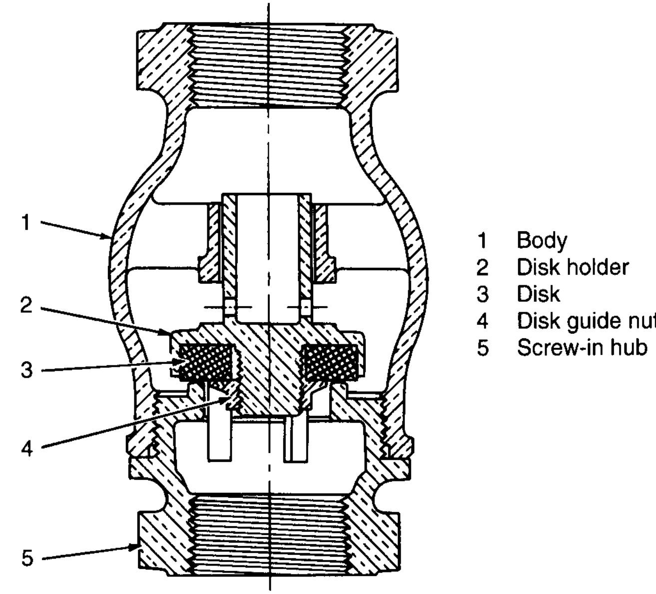


(a) Side connections without a spring



(b) Side connections with a spring

Figure 11-23. Lift-check valves.
Courtesy Crane Company. *Continued on next page.*



(c) Body with top and bottom connections

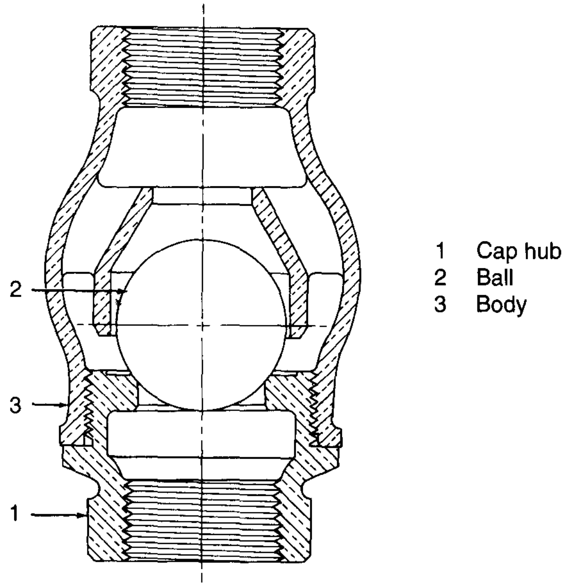
Figure *11-23-Continued*

through the outlet port at the top of the body. Reverse flow will act on the top of the ball, forcing it against the valve's seat. In addition, when there is no flow through the valve, the ball will be seated by gravity. A spring may be installed to increase the closing force applied to the ball and to enable the valve to be used in a horizontal line. Alternatively, a ball valve may be furnished with side connections for use in a horizontal line (fig. 11-24b). Ball-check valves are often used with viscous fluids.

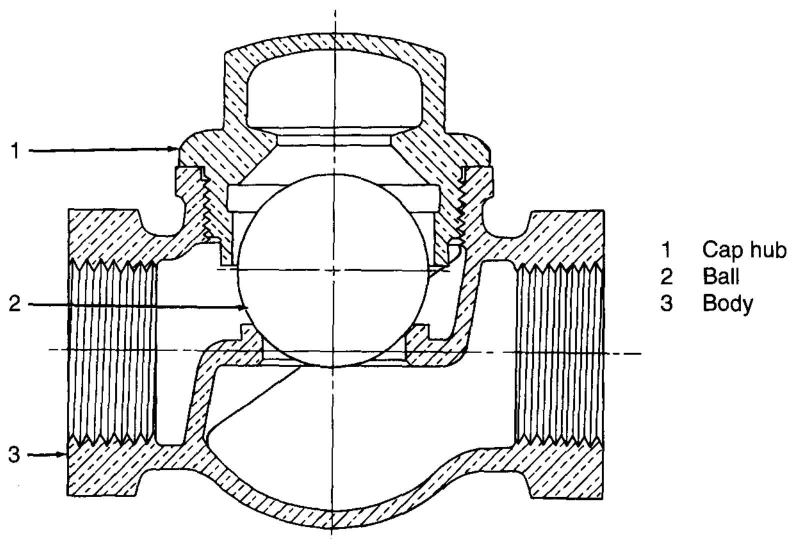
SWING-CHECK VALVE

The disk in a swing-check is mounted on a hinged joint within the valve body (fig. 11-25). When fluid having sufficient pressure enters the valve's inlet port, it forces the disk to pivot on the hinge and move away from the seat. Fluid can then flow through the open valve. If the pressure of the fluid entering the valve is too low, if the flow entering the valve's inlet port stops, or if fluid attempts to flow through the valve in reverse, the disk will swing back against the seat and the valve will close. In some cases, an external weighted lever is linked to a swing-check valve's disk to increase or reduce (depending on how the lever is connected to the valve) the fluid pressure necessary to open the valve.

Access to a swing-check valve's disk and seat can generally be achieved by removing the bonnet from the valve body. Composition disks and replaceable



(a) Vertical



(b) Horizontal

Figure 11-24. Ball-check valves. Courtesy Crane Company.

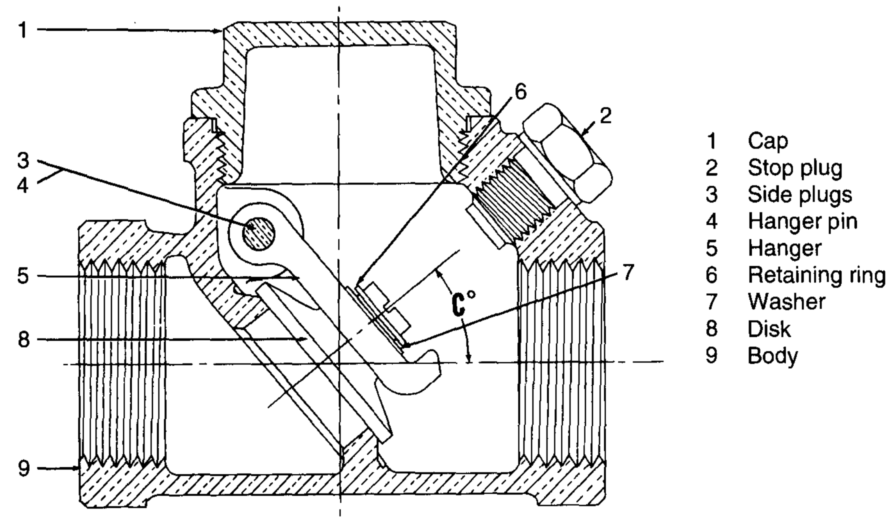


Figure 11-25. Swing-check valve. Courtesy Crane Company.

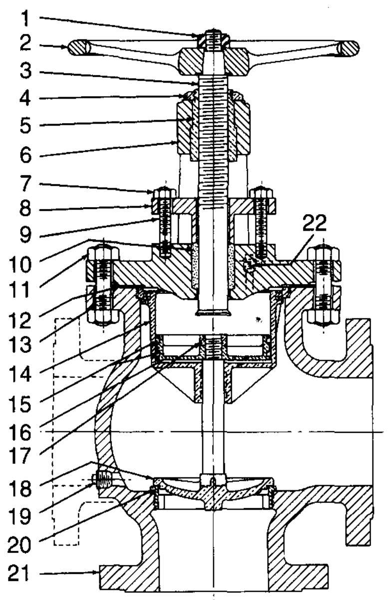
seat rings are sometimes used. Many swing-check valves are suitable for use in horizontal and vertical (when the flow is upward) lines; however, the pressure required to open a swing-check will typically be greater in a verticalline (because ofthe weight ofthe fluid resting against the closed valve's disk). Also, to prevent pounding, swing-check valves are generally not recommended for use in systems with pulsating flow.

STOP-CHECK VALVE

A typical stop-check valve is similar to a globe valve, except that the stop-check valve's disk is not attached to the valve stem (fig. 11-26). When the stem is screwed tightly into the valve body, it holds the disk against the valve's seat and prevents the valve from opening. When the stem is raised, however, it does not open the valve. Instead, it moves away from the disk and permits the valve to function like a lift-check valve. The amount that incoming flow can lift the disk off its seat can be limited by the position of the stem. Consequently, a stop-check valve can be used to throttle flow. Stop-check valves are available with in-line, angle, and Y-pattern body configurations.

Control Valves

Control valves are used to automatically regulate flow through a system in order to maintain a required pressure, temperature, flow rate, level, or some other parameter.



1 Handwheel nut
2 Handwheel
3 Stem
4 Yoke bushing nut
5 Yoke bushing
6 Yoke bonnet
7 Gland stud nut
8 Gland
9 Gland stud
10 Packing
11 Bonnet bolt nut
12 Bonnet gasket
13 Bonnet bolt
14 Dashpot
15 Piston rings
16 Piston
17 Locknut
18 Disk
19 Drain plug
20 Seat ring
21 Body
22 Lubrication plug

Figure 11-26. Stop-check valve. Courtesy Crane Company.

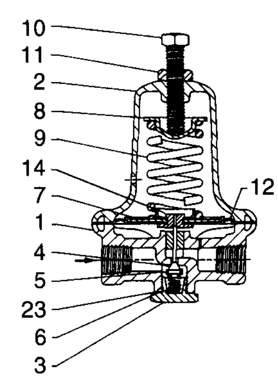
PRESSURE-REDUCING VALVE

A pressure-reducing valve is a device used to reduce the pressure of steam, a compressed gas, or a liquid supplied from a higher-pressure source. The amount that a reducing valve opens or closes will generally be adjusted automatically to maintain a relatively constant outlet pressure with limited fluctuations in the pressure of the fluid entering the valve or in the demand downstream from the valve. Methods used to raise the set point for a reducing valve's outlet pressure include increasing the compression of an adjusting spring or increasing the pressure of a compressed gas, such as air, being supplied to the valve's actuator. The force exerted by the spring or the gas, which is typically applied to one side of a flexible diaphragm, is opposed by a force resulting from the pressure of the fluid leaving the valve. When these two forces are out of balance, that is, when the pressure ofthe fluid leaving the valve is above or below the set point, the movement ofthe diaphragm may be directly transmitted to the reducing valve's stem, or the diaphragm may act on a smaller pilot valve used to control the flow of a higher-pressure fluid that actuates the main reducing valve.

A typical direct spring-operated reducing valve is shown in figure 11-27. This valve has an adjusting spring (9) that applies a downward force to the upper surface of a diaphragm (12). Pressure from the outlet side ofthe valve is transmitted through an internal port in the valve body (1) to the underside of the diaphragm (12). When the force on the bottom of the diaphragm (12) is less than the force exerted by the spring (9), which can result from an increase in the set-point adjustment, a drop in the supply pressure, or an increase in demand downstream from the valve, the diaphragm (12) deflects downward and pushes the nozzle (14) located in the diaphragm (12) against the upper end of the main valve's stem (4). As a result of this motion, the nozzle (14) will be closed, and the plug on the main valve (4) will be moved away from its seat in the valve body (1). The flow rate through the main valve and the valve's outlet pressure will, therefore, increase.

If the pressure of the fluid leaving the valve body (1) exceeds the set point, the diaphragm (12) will deflect upward, and the main-valve spring (6) will push the main valve's plug (4) closer to its seat, which will increase the resistance to flow through the main valve (4). This may cause the outlet pressure to drop sufficiently for the forces acting on the diaphragm (12) to be balanced. If, however, the valve's outlet pressure is still too high, the diaphragm (12) will continue to deflect upward and permit the main valve (4) to close completely. If, after the main valve (4) has closed, the pressure acting on the underside of the diaphragm (12) is still above the set point, the continued upward deflection ofthe diaphragm (12) will push the seat in the nozzle (14) away from the upper end of the main valve's stem (4). Once the nozzle (14) opens, fluid from the outlet side ofthe valve body (1) will be vented through the nozzle (14) and an opening in the adjusting-spring case (2) to the atmosphere, which should permit the excess pressure to be relieved. Because the fluid passing through the valve is released to the atmosphere, caution should be exercised whenever operating with cold, hot, flammable, or toxic fluids.

The set point of this valve is adjusted by changing the compression of the adjusting spring (9) with the adjusting screw (10) that protrudes from



1 Body
2 Adjusting-spring case
3 Body plug
4 Main valve
5 Main-valve-spring seat
6 Main-valve spring
7 Diaphragm disk
8 Top spring seat
9 Adjusting spring
10 Adjusting screw
11 Adjusting-screw locknut
12 Diaphragm
14 Nozzle
23 Body-plug gasket

Figure 11-27. Direct spring-operated reducing valve. Courtesy Fisher Controls International, Inc.

the top of the valve. To facilitate set-point adjustments, the external end of the adjusting screw (10) may have a hexagonally shaped head, or it may be fitted with a handwheel. Also, a locknut (11) is often provided to prevent the adjusting screw from being turned unintentionally (e.g., by vibration). Turning the screw (10) clockwise (when facing the external end of the screw) increases the compression of the adjusting spring (9) and the set pressure for fluid leaving the valve. Turning the screw (10) counterclockwise reduces the spring compression and the set pressure. However, with the adjusting screw (10) in a given position, the compression of the adjusting spring (9) will be reduced as the main valve (4) opens. Consequently, when operating with a fixed supply pressure, the outlet pressure maintained by the valve will drop slightly as the main valve (4) opens and the flow rate through the valve increases. The percentage of a reducing valve's outlet pressure setting at the minimum flow rate that can be maintained at the rated capacity is sometimes referred to as the valve's accuracy of regulation.

In a typical direct diaphragm-operated pressure-reducing valve (fig. 11-28), the set pressure is adjusted by varying the pressure of air (or another suitable gas) acting on the upper surface of a diaphragm. In addition,

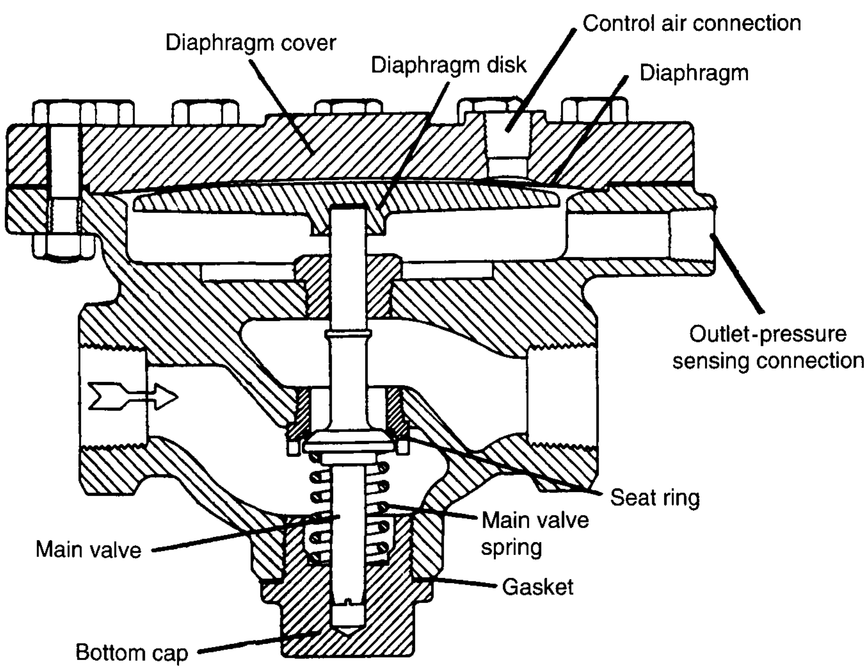
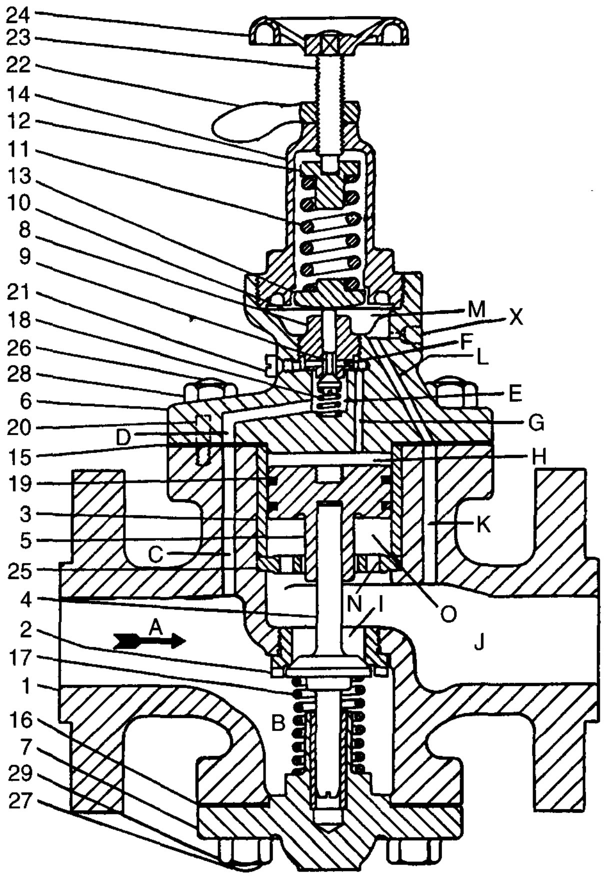


Figure 11-28. Direct diaphragm-operated pressure-reducing valve. Courtesy Leslie Controls, Inc.

fluid from the outlet side of the valve is directed to the underside of the diaphragm through an internal port (not shown in fig. 11-28) or an external line. Ordinarily, when the valve is in equilibrium, the upward force exerted by the fluid acting on the underside of the diaphragm (i.e., fluid at the outlet pressure), together with the upward force exerted by the valve spring, will equal the downward force exerted on the diaphragm by the loading air signal. If, however, the combined force resulting from the outlet pressure and the valve spring is too low to balance the air pressure, the diaphragm will deflect downward and push the valve disk away from its seat, which will reduce the throttling action of the valve. Conversely, if the combined force resulting from the outlet pressure and valve spring exceed the force exerted by the loading air signal, the diaphragm will deflect upward and allow the valve spring to move the disk closer to its seat.

A typical internal-pilot piston-operated pressure-reducing valve is shown in figure 11-29a. The set point for the pressure of fluid leaving the valve is adjusted by changing the compression of an adjusting spring (11) that applies a downward force through a bottom seat (13) to the upper surface of a flexible metal diaphragm (10). Fluid from the valve's outlet (J) is directed through internal ports (K) and (L) to chamber (M) where it acts on the diaphragm's lower surface. Turning the adjusting screw (23) clockwise (while facing the external end of the screw) increases the compression of the adjusting spring (11) and the set point for the pressure of fluid leaving the valve. Turning the screw (23) in the opposite direction reduces the spring compression and the outlet-pressure set point. Although most of the fluid entering the valve is directed to chamber (B) at the underside of the main valve (4), a portion of the fluid that enters the valve passes through ports (C) and (D) to chamber (E) at the underside of a small pilot or controlling valve (9). When the force exerted by fluid at the valve's outlet pressure in chamber (M) is less than the force exerted by the adjusting spring (11) (i.e., the pressure of fluid leaving the valve is below the set point, which can result from a drop in the supply pressure, an increase in the demand downstream from the valve, or an increase in the valve's set-point adjustment), the diaphragm (10) deflects downward and pushes the controlling valve (9) away from its seat (8). Fluid in chamber (E), which is at the supply pressure, then passes through the open controlling valve and is directed through ports (F) and (G) to chamber (H), where it applies a force to the top of the main valve's piston (5). The area of this piston is larger than the area of the underside of the main valve (4) by an amount that is sufficient to overcome the upward force exerted by the main-valve spring (17), together with the upward force of fluid from chamber (J) that passes through ports (N) into chamber (O) and acts against the lower surface of the main-valve piston (5). Consequently, the piston (5) is moved downward by the supply fluid. This motion pushes the main valve (4) away from its seat (2) and reduces the throttling action of the valve. The



1 Main body	11 Adjusting spring	21 Top-cap screw
2 Seat ring	12 Top spring seat	22 Adjusting-screw lock nut
3 Cylinder liner	13 Bottom spring seat	23 Adjusting screw
4 Main valve	14 Spring case	24 Handwheel
5 Piston	15 Top-cap gasket	25 Cylinder bottom
6 Top cap	16 Bottom-cap gasket	26 Top-cap stud
7 Bottom cap	17 Main-valve spring	27 Bottom-cap stud
8 Controlling-valve seat	18 Controlling-valve spring	28 Top-cap nut
9 Controlling valve	19 Piston ring	29 Bottom-cap nut
10 Diaphragm	20 Dowel pin	

Figure 11-29a. Internal-pilot, piston-operated pressure-reducing valve. Courtesy Leslie Controls, Inc.

pressure of fluid leaving the valve will, therefore, increase. In addition, this increased pressure will be transmitted to chamber (M) where it will act to restore balance to the valve.

When the force exerted on the underside ofthe diaphragm (10) by the reduced-pressure fluid leaving the valve exceeds the force of the adjusting spring (11) (i.e., the outlet pressure is above the set point, which can result from a reduction of the set-point adjustment, an increase in the supply pressure, or a reduction in the demand downstream from the valve), the diaphragm (10) will deflect upward, and the controlling valve (9) will be closed by the controlling-valve spring (18). With the controlling valve (9) closed, the supply fluid acting on top ofthe piston (5) will slowly leak past the piston rings (19). In addition, when the fluid above the piston (5) is steam, it will gradually condense. The pressure pushing down on the piston (5) will, therefore, drop, and the main-valve spring (17), together with the upward force of fluid from the valve's outlet (J) that passes through ports (N) into chamber (O) and acts against the lower surface of the main piston (5), will eventually be able to push the main valve (4) towards its seat (2). This motion will increase the throttling action of the main valve (4) and result in a reduction in the pressure offluid leaving the valve.

Instead of being furnished with internal ports (ports [K] and [L] in fig. 11-29a) to direct fluid at the outlet pressure to the chamber below the valve's diaphragm (chamber [M] below 10), some pilot-type reducing valves have an external sensing line that connects this chamber via port (X) to the valve's outlet line at some point (usually approximately three feet) downstream from the valve. With this alternate arrangement, turbulence in the outlet portion ofthe valve body, which is likely to occur when there is a high pressure drop through the valve, will not affect valve operation. To prevent liquid pockets from accumulating within the external sensing line, the line should be pitched towards the reducing-valve outlet line. In another pilot-operated pressure-reducing valve design, which is used in some low-flow applications, a lower diaphragm is utilized instead of a piston to open the main valve (fig. 11-29b). Typical installation arrangements for pressure-reducing valves with internal sensing ports and for pressure-reducing valves with external sensing lines are shown in figure 11-30.

CONSTANT-PRESSURE GOVERNOR

A constant-pressure (CP) governor is sometimes used to control the flow of steam supplied to a turbine driving a pump or to the drive-end cylinders of a direct-acting reciprocating pump. The steam flow is automatically regulated to maintain a constant discharge pressure from the pump. A typical CP governor is shown in figure 11-31. This governor is similar to the internal-pilot pressure-reducing valve described previously and shown in figure 11-29a. The CP governor, however, has an additional diaphragm assembly and a port (X)that is connected to the discharge line ofthe pump being controlled. The discharge pressure set point is adjusted with the

1 Adjusting screw with handwheelandlocknut
2 Top spring seat
3 Adjusting spring
4 Adjusting-spring case
5 Bottom spring seat
6 Upper diaphragm
7 Top cap
8 Controlling-valve seat
9 Controlling valve
10 Controlling-valve spring
11 Lower diaphragm
12 Diaphragm disk
13 Bolt and nut
14 Main valve
15 Seat ring
16 Main body
17 Main-valve spring
18 Bottom-cap gasket
19 Bottom cap
20 Dowel pin

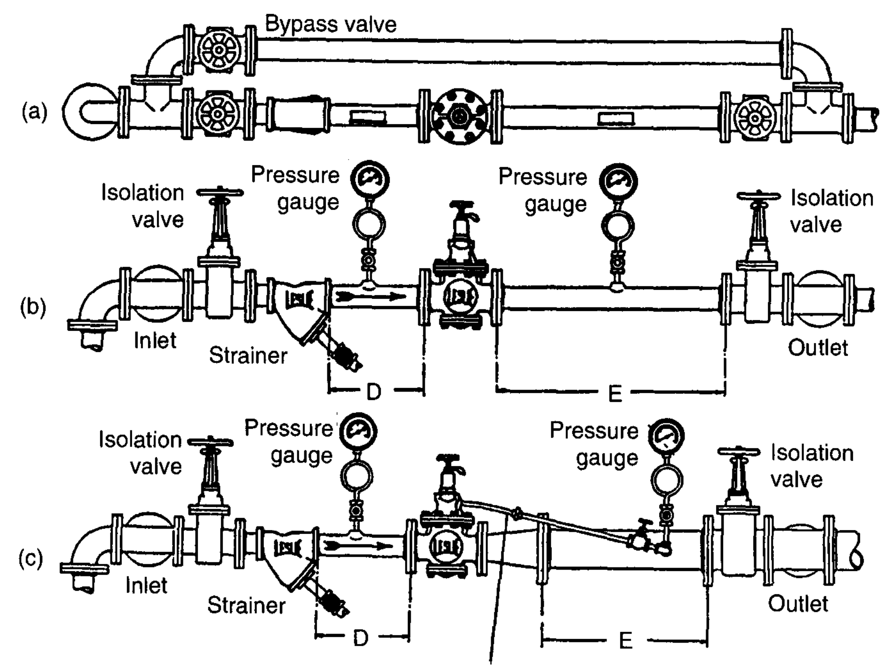
Note "A"

For low-outlet-pressure applications, port U is sometimes eliminated and an external sensing line is used to connect the threaded boss in the top cap to the main valve's reduced-pressure outlet line, usually at a location approximately three feet downstream from the reducing-valve outlet. To avoid the accumulation of liquid pockets within the external sensing line, the line should be pitched toward the outlet line.

Figure 11-29b. Internal-pilot, diaphragm-operated pressure-reducing valve. Courtesy Leslie Controls, Inc.

handwheel (53) at the top of the governor. When this handwheel is turned clockwise (when facing the handwheel), the adjusting screw (30) increases the compression of the adjusting spring (33), and the pressure setting of the fluid being discharged from the pump is raised. Screwing the handwheel (53) in the opposite direction reduces the spring compression and the pump's discharge-pressure set point.

If the pump's discharge pressure, which is applied to the underside of the governor's upper diaphragm (38), drops below the set point, (i.e., the force exerted on the underside of the upper diaphragm [38] by the fluid entering port [X] is less than the downward force exerted by the adjusting spring [33]), the adjusting spring (33), acting through the bottom spring seat (34), will push the upper crosshead (25) downward. This downward motion will be transmitted through the connecting rods (36), lower crosshead (25A), and diaphragm stem (26) to the lower diaphragm (23A). As the lower diaphragm (23A) deflects downward, it will push the pilot or controlling valve (49) away from the controlling-valve seat (48). Steam from the inlet side of the governor will then pass through the open controlling



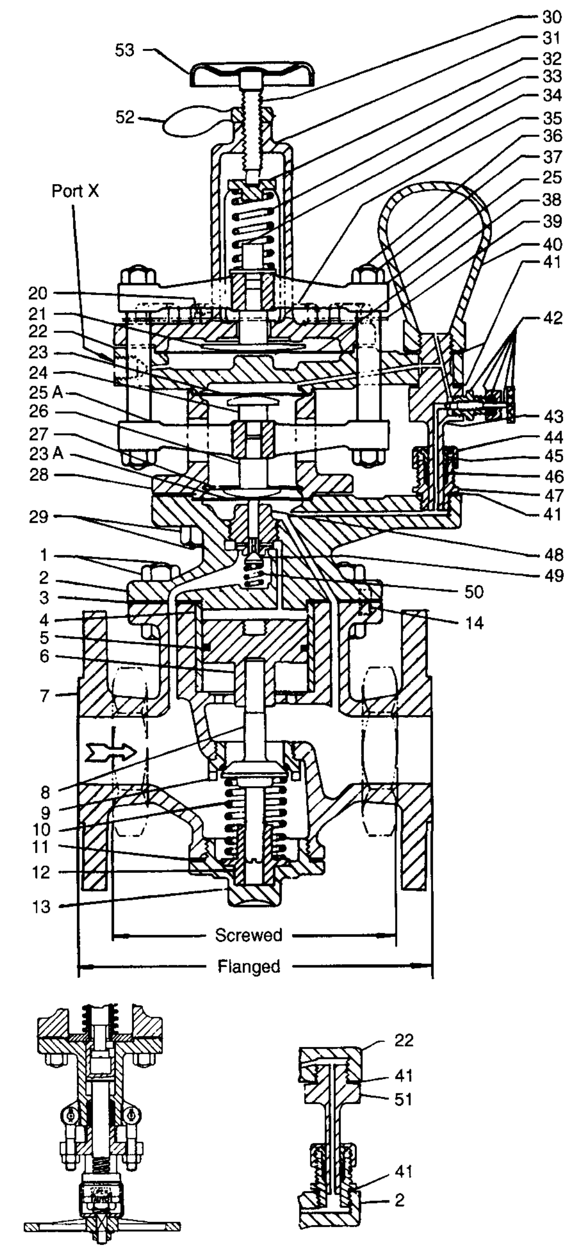
Outlet-pressure sensing line with shutoff valve

Reducing valve size (size of end connections)	Minimum length of straight pipe D	E
Y2" to 1Y2"	1' 6"	3'0"
2" to 4"	3'0"	5'0"
5" to 6"	4'0"	8'0"

Figure 11-30. Typical pressure-reducing valve installation arrangements: (a) plan view, (b) elevation, reducing valve with internal sensing port, (c) elevation, reducing valve with external sensing line. Courtesy Leslie Controls, Inc.

valve (49), apply a force to the top of the main valve's piston (6), and force the main valve (8) to move away from its seat ring (9). The resulting increase in the flow of steam through the governor will cause the pump's speed and discharge pressure to increase.

As the pump's discharge pressure acting on the underside of the upper diaphragm (38) reaches the set point, this diaphragm will deflect upward and force the diaphragm disk (21), crossheads (25 and 25A), connecting rods (36), and diaphragm stem (26) all to move in an upward direction. The resulting reduction in the downward force applied by the diaphragm stem (26) to the upper surface of the lower diaphragm (23A) will enable the controlling-valve spring (50) to push the controlling valve (49) towards its seat (48), which will reduce or stop the flow of steam acting on the top of the



Optional manual pull-open device available with some governors

Alternate unrestricted connector arrangement

1 Bolt and nut
2 Top cap
3 Top-cap gasket
4 Cylinder liner
5 Piston ring
6 Piston
7 Main body
8 Main valve
9 Main-valve seat ring
10 Main-valve spring
11 Bottom-cap gasket
12 Bottom-cap bushing
13 Bottom cap
14 Dowel pin
20 Connecting-rod-guide screw
21 Diaphragm disk
22 Diaphragm chamber
23 Middle diaphragm
23A Lower diaphragm
24 Diaphragm-stem cap
25 Upper crosshead
25A Lower crosshead
26 Diaphragm stem
27 Diaphragm-stem guide
28 Diaphragm spacer
29 Stud and nut
30 Adjusting screw
31 Adjusting-spring case
32 Top spring seat
33 Adjusting spring
34 Bottom spring seat
35 Spring-case cap screw
36 Connecting rod
37 Connecting-rod nut
38 Upper diaphragm
39 Connecting-rod guide
40* Steam volume chamber
41 Gasket
42* Needle valve
43* Needle-valve body
44 Connector nut
45 Connector gland
46 Connector packing
47 Connector union
48 Controlling-valve seat
49 Controlling valve
50 Controlling-valve spring
51** Connector nipple
52 Adjusting-screw lock nut
53 Adjusting-screw handwheel
Port X Pump-discharge-pressure sensing port

* Normally furnished with a governor for a turbine-driven pump.
** Normally furnished with a governor for a reciprocating pump.

Figure 11-31. Constant-pressure (CP) pump governor. Courtesy Leslie Controls, Inc.

main valve's piston (6) and limit the amount that the main valve (8) opens. However, because ofthe delay in the response of the pump to a change in the steam flow supplied to its driver, by the time the pump's discharge pressure does reach the set point, too much steam could be going to the pump's driver. With this condition, the pump's discharge pressure would eventually exceed the set point and the governor would then have to reduce the steam flow to the driver. This overcorrection by the governor can lead to unstable operation (sometimes referred to as hunting). For pump operation to be stable and the potential for overcorrection by the governor to be reduced, the controlling valve (49) must reduce the flow of steam to the main valve's piston (6) before the pump's discharge pressure actually reaches the set point. To achieve this characteristic, steam leaving the governor is directed to the underside ofthe governor's lower diaphragm (23A). Consequently, as the governor's main valve (8) opens and allows more steam to be supplied to the pump's driver, the increased outlet pressure from the governor forces the lower diaphragm (23A) to deflect upward and permits the controlling-valve spring (50) to push the controlling valve (49) towards its seat (48). The flow of steam acting on the top of the main valve's piston (6), the amount that the main valve (8) opens, and the amount that the pump's discharge pressure increases are, therefore, all limited by this feedback.

Although the intent ofthis feedback is to prevent overcorrection by the governor, the upward force applied by steam leaving the governor to the underside of the lower diaphragm (23A) also reduces the pump discharge pressure necessary to balance the downward force of the adjusting spring (33), that is, it reduces the effective set point of the governor. So that the original set point can be restored, an equalization port is provided that connects the chamber below the governor's lower diaphragm (23A) to the chamber above a middle diaphragm (23). The downward force that results from the application ofthe governor's outlet pressure to the top ofthis middle diaphragm (23) balances the upward force applied to the lower diaphragm (23A) and, therefore, eliminates the effect that the governor's outlet pressure has on the position of the controlling valve (49).

When a CP governor is used with a reciprocating pump, an unrestricted nipple (51) is generally used to connect the chamber below the lower diaphragm (23A) to the chamber above the middle diaphragm (23). However, when a CP governor is used with a turbine-driven pump, which will generally operate with a wider range of steam-chest pressures and react more slowly to changes in steam flow than a reciprocating pump, a needle valve (42) is often installed in the equalization port connecting these two chambers. When this valve is properly adjusted, the time that it takes for a change in the governor's outlet pressure to be transmitted to the middle diaphragm (23), which represents the time delay in the cancellation of the effect that the governor's outlet pressure has on the position of the controlling valve (49), should match the time it takes for the pump to respond to a change in the steam flow to its driver. To achieve this, the needle valve should ordinarily be throttled just enough to stop unstable governor and pump operation (usually, the valve will be approximately one-quarter turn open after it is adjusted). So that the equalization port remains effective, the needle valve should never be completely closed. A volume chamber (40) may be connected to the equalization port to help attenuate pressure pulsations transmitted through the needle valve.

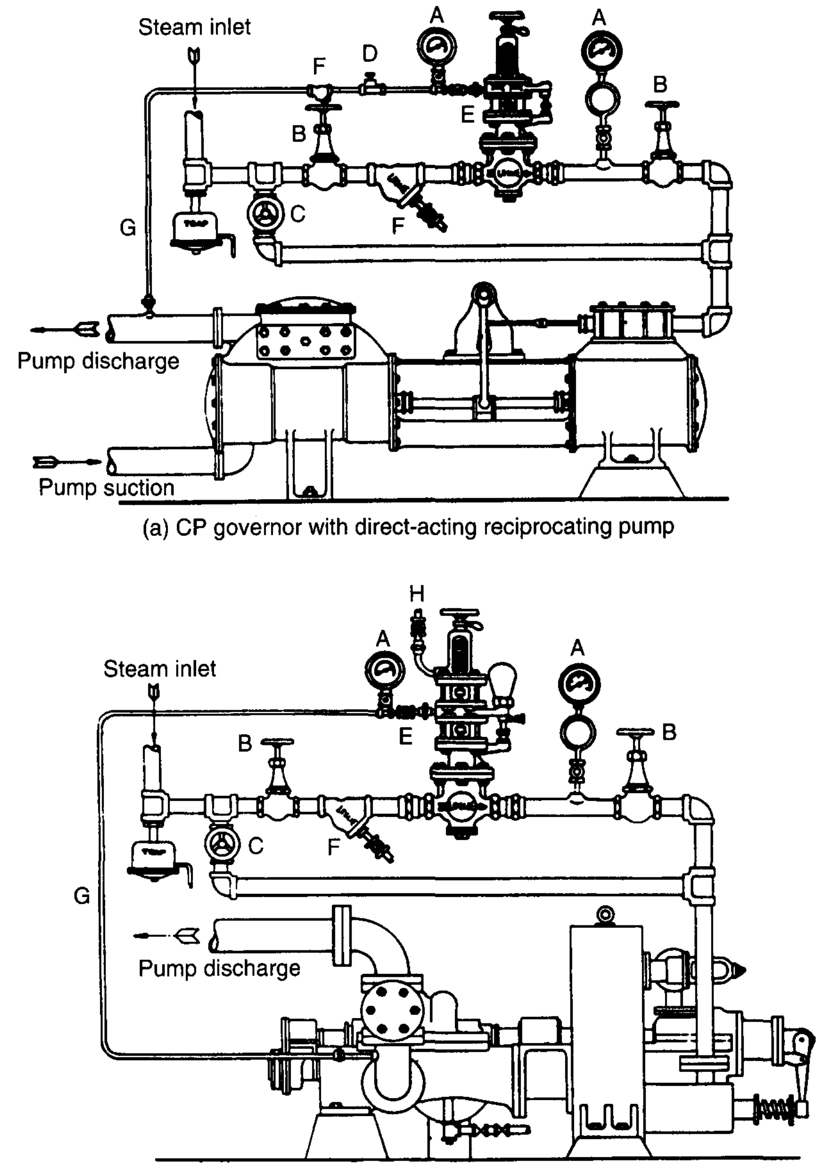
An increase in the pump's discharge pressure above the set point has an effect opposite to that just described and results in reductions in the steam flow through the governor: the pump's operating speed, and the pump's discharge pressure.

A typical installation arrangement for a CP governor is shown in figure 11-32a. In an installation where a bypass line (shown in fig. 11-32) is not provided, a manual pull-open device with a handwheel is usually mounted on the lower end of the governor (see fig. 11-31). When the handwheel is turned in a counterclockwise direction (while facing the handwheel), the stem pulls the main valve (8) away from its seat ring (9). The handwheel, therefore, enables an operator to manually increase the flow of steam to the driver of the pump being controlled during start-up, emergency operation, or an overload condition.

Legend for figure 11-32, facing page

Typical guidelines

- Install governor upright in a horizontal line as close to pump as possible, with arrow on governor body in direction of steam flow.
- Ordinarily, connect pump end of discharge-pressure sensing line 1 to 3 feet from pump discharge between pump and discharge stop valve. Sensing-line size should typically be ¼″ for low-viscosity liquids, such as water, and ⅜″ for high-viscosity liquids. (When a governor will be used to automatically bring a standby pump up to speed if an on-line pump fails, the sensing connection must be made downstream from the standby pump's discharge check valve, if used.
- Install needle valve in a CP or XP governor's pump-discharge-pressure sensing line to suppress pressure pulsations produced by a reciprocating pump. (This valve is not normally required in the sensing line from a turbine-driven centrifugal pump.) The needle valve should be throttled until sensing-line pressure-gauge needle movement is smooth and even, with a small movement at the end of each pump stroke.
(Usually, the valve will be approximately ¼ turn open.) The needle valve should never be completely closed.
- Install a strainer in the pump-discharge-pressure sensing line when necessary to protect the governor from contaminants in the pumped liquid.
- Connect XP governor's boiler-pressure sensing line to a location where full boiler pressure exists. Typical sensing-line size is ¼″.
- Prior to pump start-up, the governor isolation valves and the stop valves in the pump-discharge-pressure and boiler-pressure (for an XP governor) sensing lines should be closed. In addition, the governor adjusting-spring compression should be relieved (i.e., the governor's set point should be reduced).
- When starting pump, drain condensate from lines, and slowly open bypass valve. After pump and piping are warm, slowly open valves in pump-discharge-pressure and boiler-pressure (for an XP governor) sensing lines, and slowly open governor inlet and outlet isolation valves wide. Then, slowly screw down on governor adjusting screw until pump pressure just begins to rise, indicating that the governor is taking control of the pump. The bypass valve should then be closed tightly, and the governor adjusting screw should be turned (clockwise from top to increase pressure, counterclockwise from top to decrease pressure) until the desired pump discharge pressure or differential pressure is achieved. The adjusting screw should then be locked in place with the lock nut.
- These guidelines are general in nature and do not include requirements for the pump or its driver. Actual pump and governor technical manuals should always be consulted to obtain installation and operational guidelines applicable to specific installations.



(a) CP governor with direct-acting reciprocating pump

(b) XP governor with turbine-driven pump

A	Pressure gauge (with stop valve)	E	Stop valve
B	Governor isolation valve	F	Strainer (with blowoff connection)
C	Governor bypass valve	G	Pump-discharge-pressure sensing line
D	Pulsation-dampening needle valve	H	Boiler-pressure sensing line (with stop valve)

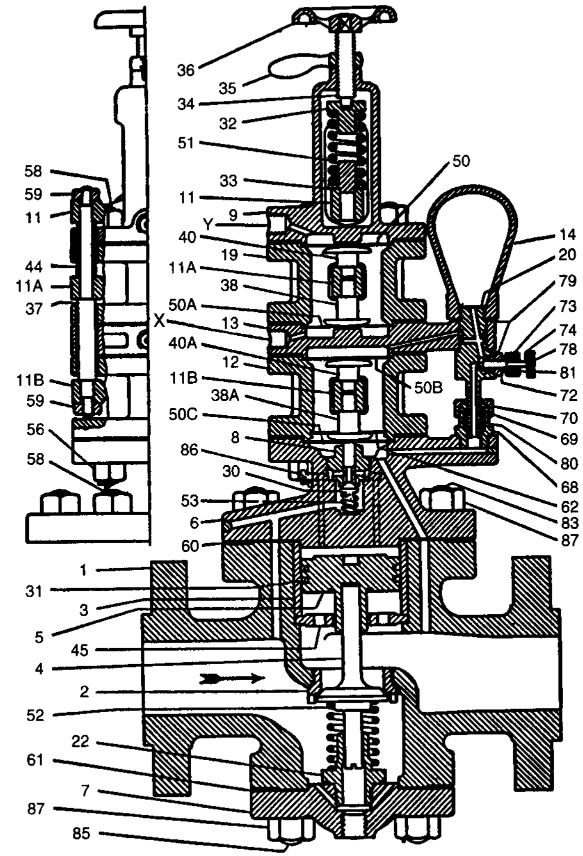
Figure 11-32. Typical pump governor installation arrangements. Courtesy Leslie Controls, Inc.

EXCESS-PRESSURE GOVERNOR

An excess-pressure (XP) governor can be used to control the flow of steam supplied to a turbine that drives a centrifugal main feed pump or to the drive-end cylinders of a direct-acting reciprocating feed pump. The steam flow is automatically regulated to maintain a pump discharge pressure that exceeds the boiler pressure by a fIxed amount. A typical XP governor is shown in fIgure 11-33. This governor is similar to the CP governor described previously, except that the XP governor has an additional upper diaphragm assembly. Pressure from the boiler is transmitted through a sensing line connected to port (Y) in thi,s assembly and acts on the upper surface of the upper diaphragm (50). The difference desired between the feed pump discharge pressure and the boiler pressure is adjusted with the handwheel (36) at the top of the governor. When this handwheel is turned clockwise (when facing the handwheel), the adjusting screw (34) increases the compression ofthe adjusting spring (51) and the amount that the pressure ofthe feedwater discharged from the feed pump must exceed the boiler pressure to balance the governor. Screwing the handwheel in the opposite direction reduces the spring compression and the amount that the pump discharge pressure must exceed the boiler pressure to achieve balance.

If the combined downward force exerted by the boiler pressure acting on the top ofthe upper diaphragm (50) and the adjusting spring (51) acting against the bottom spring seat (33) exceeds the upward force exerted on the bottom of the upper intermediate diaphragm (50A) by fluid at the pump's discharge pressure, which is directed through port (X), the upper diaphragm (50) and the upper intermediate diaphragm (50A) will both deflect downward. In addition, the bottom spring seat (33) and upper -crosshead (11) and the upper diaphragm-stem cap (40) and intermediate crosshead (l1A) will all be forced to move downward. This downward motion will be transmitted through the connecting rods (37), lower crosshead (l1B), and lower diaphragm stem (38A) to the lower diaphragm (50C). As the lower diaphragm (50C) deflects downward, it will push the pilot or controlling valve (30) away from its seat (8). Steam from the inlet side of the governor will then pass through the open controlling valve (30), apply a force to the top of the main valve's piston (5), and force the main valve (4) to move away from its seat ring (2). The resulting increase in the flow of steam through the governor will cause the pump's speed and discharge pressure to increase. The method offeedback through the equalization port connecting the chamber below the lower diaphragm (50C) to the chamber above the lower intermediate diaphragm (50B), together with the adjustment of the needle valve (78) in the equalization port, is identical to that previously described for the CP governor.

If the amount that the pump's discharge pressure exceeds the boiler pressure rises above the governor's set point, the action of the governor will be the opposite of that described above and will result in reductions in the



1 Main body
2 Seat ring
3 Cylinder liner
4 Main valve
5 Piston
6 Top cap
7 Bottom cap
8 Controlling-valve seat
9 Adjusting-spring case
11 Upper crosshead
11A Intermediate crosshead
11B Lower crosshead
12 Diaphragm spacer
13 Diaphragm chamber
14 Steam chamber
19 Upper diaphragm spacer
20 Needle-valve body
22 Main valve guide
30 Controlling valve
31 Piston ring
32 Top spring seat
33 Bottom spring seat
34 Adjusting screw
35 Adjusting-screw lock nut
36 Handwheel
37 Connecting rod
38 Upper diaphragm stem
38A Lower diaphragm stem
40 Upper diaphragm-stem cap
40A Lower diaphragm-stem cap
44 Connecting-rod bushing
45 Cylinder bottom
50 Upper diaphragm
50A Upper intermediate diaphragm
50B Lower intermediate diaphragm
50C Lower diaphragm
51 Adjusting spring
52 Main-valve spring
53 Controlling-valve spring
56 Nut
58 Stud
59 Connecting-rod nut
60 Top-cap gasket
61 Bottom-cap gasket
62 Diaphragm-stem guide
68 Connector union
69 Connector gland
70 Connector nut
72 Needle-valve bonnet
73 Needle-valve bonnet nut
74 Needle-valve stem
78 Needle-valve-stem head
79 Needle-valve-body gasket
80 Connector packing
81 Needle-valve packing
83 Top-cap stud
85 Bottom-cap stud
86 Top-cap screw
87 Nut
X Feed-pump-discharge-pressure sensing port
Y Boiler-pressure sensing port

Figure 11-33. Excess-pressure (XP) pump governor. Courtesy Leslie Controls, Inc.

steam flow to the pump's driver, the pump's speed, and the pump's discharge pressure.

A typical installation arrangement for an XP governor is shown in figure 11-32b.

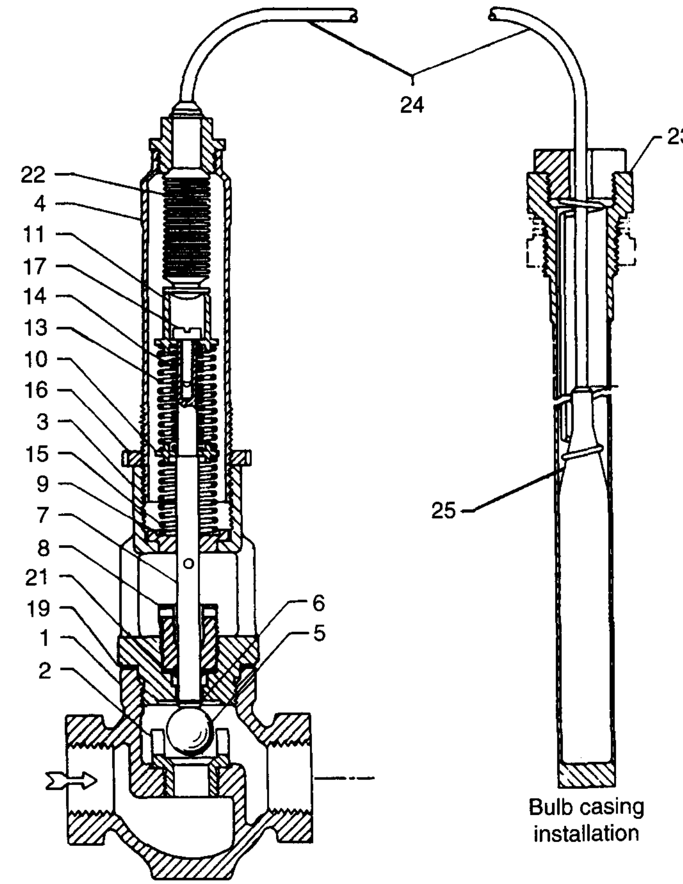
TEMPERATURE-REGULATING VALVE

A temperature-regulating valve can be used to automatically control the flow of a fluid passing through the valve in order to maintain the temperature either of that fluid or of a second fluid at a desired value.

A three-way blending or mixing valve with either a built-in or an external thermostatic element 'is sometimes used to control the amount of a fluid that passes through a heat exchanger. The valve, which would typically be located downstream from the heat exchanger, has two inlet ports: one for fluid that has passed through the heat exchanger and one for fluid that has bypassed the heat exchanger. The two flows, which are at different temperatures, mix within the valve and leave through a single outlet port. As the temperature of the mixture leaving the valve, which is sensed by the valve's thermostatic element, changes, the valve's internal element shifts to allow more or less of the fluid to pass through the heat exchanger.

A three-way diverting valve, which has one inlet port and two outlet ports, is sometimes located upstream from a heat exchanger. As the temperature being sensed downstream from the heat exchanger changes, more or less of the fluid entering the valve is directed through the outlet port that leads to the heat exchanger. The fluid leaving the valve's other outlet port flows through a line that bypasses the heat exchanger. The two flows are recombined downstream from the heat exchanger.

A typical direct-acting temperature-regulating valve has a temperature sensing bulb, which can be either submerged directly in the fluid that has its temperature being controlled or installed within an outer casing or well that shields the bulb from direct contact with the fluid. This bulb, which is filled with a fluid, is connected via a flexible capillary tube to a bellows. The temperature set point is raised by screwing down on an adjusting nut or sleeve, which increases the compression of one or more springs that oppose the expansion of the bellows. A typical valve designed to control the flow of a heating fluid is shown in figure 11-34. When the temperature of the fluid surrounding the sensing bulb (25) or its casing (23) exceeds the set point, the force exerted on the bellows (22) due to the expansion of the fluid within the bulb (25) and flexible tube (24) will be greater than the force exerted by the spring (13), and the valve's stem-mounted ball (5) will be moved closer to the valve's seat ring (2). Alternatively, when a valve ofthis type is designed to control the flow of a cooling fluid, it will have a stem-mounted ball or disk located below the valve's seat that will be pushed farther away from the valve's seat as the temperature being sensed rises above the set point.



1	Main body	13	Outer yielding spring
2	Seat ring	14	Inner yielding spring
3	Bonnet	15	Valve spring
4	Adjusting sleeve	16	Locknut
5	Ball	17	Valve-stem screw
6	Valve-stem clip	19	Bonnet gasket
7	Valve stem	21	Stem packing
8	Packing gland	22	Thermostatic-element bellows
9	Bottom spring seat	23	Bulb casing
10	Intermediate spring seat	24	Thermostatic-element flexible tubing
11	Top spring seat	25	Thermostatic-element sensing bulb

Figure 11-34. Direct-acting temperature control valve for heating service. Courtesy Leslie Controls, Inc.

A typical piston-operated temperature-regulating valve (fig. 11-35a) can be used to automatically control the flow of steam being supplied to a heater so that the temperature of the fluid being heated by the steam remains constant. This valve is similar to the piston-operated pressure-reducing valve shown in figure 11-29a. However, the temperature-regulating valve has additional components, including a temperature sensing bulb, which can be submerged in the fluid that has its temperature being controlled. This bulb, which contains a volatile fluid with a boiling point that should be slightly less than the minimum temperature the control valve is suitable to maintain, is connected via a flexible capillary tube to a thermostatic diaphragm assembly located in the element above the pressure plate. The temperature set point is raised by screwing the temperature-adjusting nut counterclockwise (when viewed from the top), which increases the compression of the temperature-adjusting spring that opposes the downward deflection of the diaphragm. As the sensing bulb's temperature increases, the pressure of the fluid within the bulb rises, increasing the downward force applied by the thermostatic diaphragm to the pressure plate. When this downward force exceeds the upward force exerted by the temperature-adjusting spring, the pressure plate will be pushed downward, and the resulting downward movement of the rod will force the lever to pivot in a clockwise direction and push up on the limit spring's bottom seat. The reduction in the downward force applied to the diaphragm located above the pilot or controlling valve resulting from the upward movement of the limit spring's bottom seat will permit the controlling valve to be moved towards its seat by the controlling-valve spring, and the steam pressure acting on the top of the main valve's piston will be reduced. Consequently, the main valve will be pushed closer to its seat ring by the main-valve spring, the steam flow to the heater will be reduced, and the temperature of the fluid being heated will drop.

If the temperature of the heated fluid is below the valve's set point, the downward force exerted by the fluid in the sensing bulb and flexible capillary tube against the thermostatic diaphragm and pressure plate will be less than the force imposed by the temperature-adjusting spring. Consequently, this spring will force the pressure plate to move upward, and the lever will pivot in a counterclockwise direction and permit the limit spring to expand. This downward motion is transmitted through the limit spring's bottom seat and causes the diaphragm located above the controlling valve to deflect downward and push the controlling valve open. Steam from the inlet side of the main valve will then pass through an internal port and the open controlling valve, and it will force the main-valve piston to move down. This motion will push the main valve away from its seat, and the flow of steam to the heater, together with the temperature of the fluid being heated, will increase. The expansion of the limit spring that results from the counterclockwise rotation of the lever is resisted by the upward

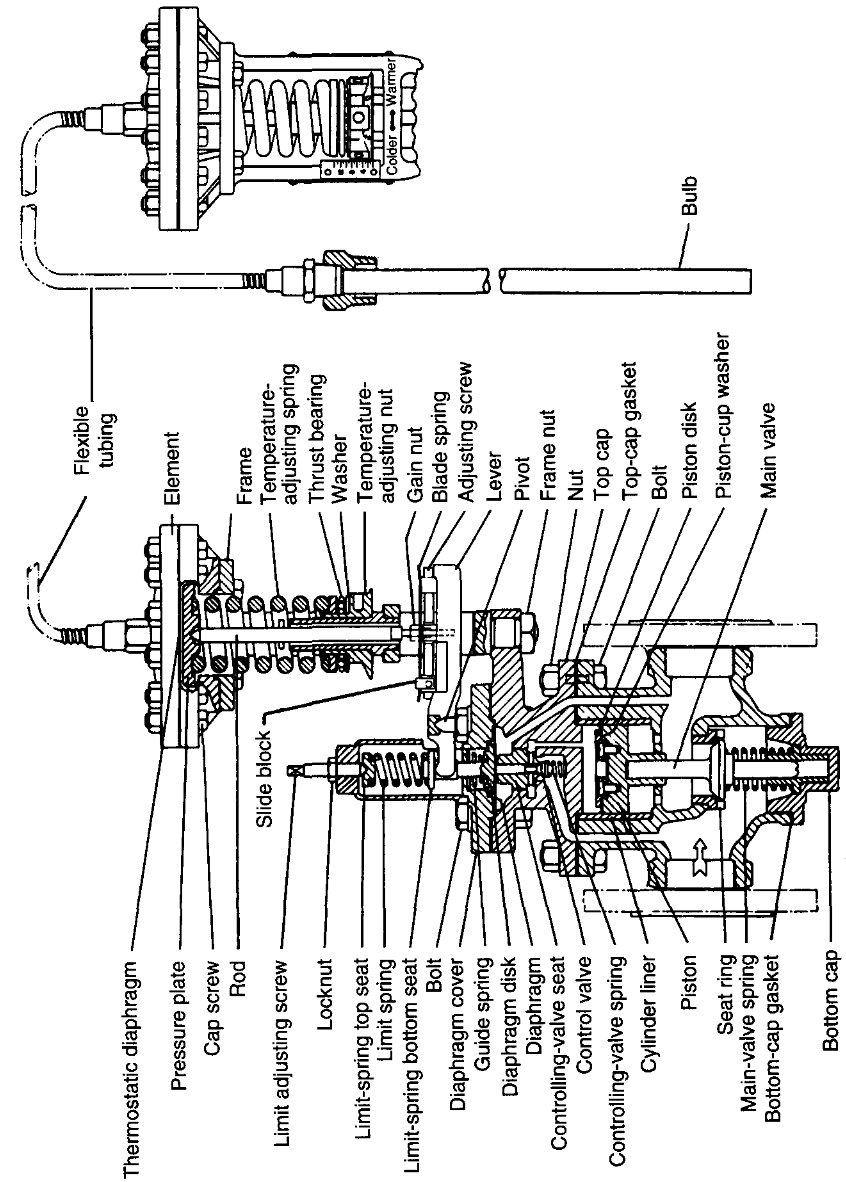


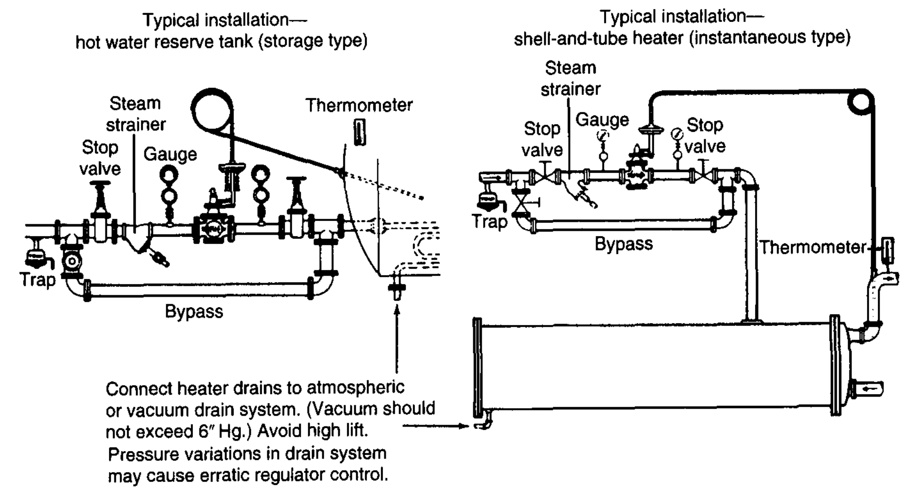
Figure 11-35a. Piston-operated temperature-regulating valve. Courtesy Leslie Controls, Inc.

force exerted by both the controlling-valve spring and steam from the main valve's outlet that passes through an internal port and acts against the underside of the diaphragm located above the controlling valve. Therefore, as the main valve opens, the increased outlet pressure being transmitted through the port and pushing up on the diaphragm will act to restore balance to the controlling valve.

In addition, once the combined upward force of the controlling-valve spring and steam under the diaphragm located above the controlling valve is equal to the combined downward force exerted by the limit spring and the guide spring, the limit spring will be unable to expand any further. If the lever continues to pivot in a counterclockwise direction (e.g., because of a further drop in the sensing-bulb temperature), it will no longer be in contact with the limit spring's bottom seat. Consequently, the controlling valve and, therefore, the main valve will no longer be affected by the position of the lever. Instead, the main valve will function as a standard pressure-reducing valve and will limit the maximum steam pressure that can be supplied to the heater. The valve's maximum outlet pressure set point can be raised by turning the limit-adjusting screw clockwise (when viewed from the top), which compresses the limit spring and increases the steam pressure necessary to balance it. Conversely, the set point for the valve's maximum outlet pressure can be lowered by turning the limit-adjusting screw counterclockwise.

Because the pressure of the steam being supplied to the heater is transmitted to the underside of the controlling-valve diaphragm, if the condensing temperature and pressure of this steam change due to a change in the temperature of the fluid being heated (e.g., if the heated fluid's temperature drops, the condensing temperature and pressure of the steam passing through the heater will drop), the controlling valve will be repositioned to compensate for the change in advance of any action caused by the fluid within the sensing bulb. The overall response time of the temperature-regulating valve is, therefore, reduced. Typical installation arrangements for this type of a valve are shown in figure 11-35b.

In some temperature-regulating valves, a blade spring is installed between the rod located below the pressure plate and the lever. By moving the slide blocks that support the blade spring closer together, the spring's stiffness, the amount that the lever rotates and, therefore, the change in the steam pressure supplied to the heater for a given change in the sensing-bulb temperature will increase. The gain (i.e., the corrective action divided by measured error) of the temperature-regulating valve is, therefore, increased. Reducing the spring stiffness by moving the slide blocks farther apart reduces the effect that a given temperature change has on the steam pressure to the heater and, therefore, reduces the temperature-regulating valve's gain. Adjustments to the slide block position are made by turning the adjusting screw.



Install regulator upright in a horizontal line within 3 to 5 feet from heater with arrow on regulator body in direction of steam flow.

Install gauges, stop valves, and thermometers as shown so that proper adjustment and check of regulator control may be made.

Avoid sharp bends in capillary tubing when installing thermal element bulb in heater. Bulb should be installed vertically or sloped downward with end of bulb facing flow. Capillary should be protected from damage.

Thermal bulbs should never be installed closer than 6 inches to heat source.

Storage type heaters: locate thermal bulb approximately at horizontal center line of tank and to one side.

Instantaneous type heaters: locate thermal bulb in heater fluid discharge pipe as close to heater as possible.

Injection heaters: regulator should be installed above level of fluid in tank with check valve to prevent backflow of fluid into regulator.

Piping: blow out piping, making sure it is clean before installing regulator.

Provide proper drainage of steam piping to prevent water hammer and possible erosion of regulator parts.

Figure 11-35b. Typical installation arrangements for temperature-regulating valve.

BACK-PRESSURE-REGULATING VALVE

A back-pressure-regulating valve is used to maintain a set pressure on the inlet side of the valve. If this pressure rises above the set point, the amount that the valve is open increases. Because of the resulting increase in the flow rate passing through the valve, the pressure at the valve inlet drops. If the inlet pressure drops below the set point, the back-pressure-regulating valve will close more, which reduces the flow rate through the valve and, provided that the pressure source is sufficient, causes the valve's inlet pressure to rise.

RELIEF VALVE

A relief valve is used to prevent overpressurization in a system or component. The set point of a typical spring-loaded relief valve (fig. 11-36) is raised by increasing the compression of a spring that holds the valve's disk against its seat. Fluid in the system being protected by the valve acts against the opposite side of the disk. When the force exerted by this fluid exceeds the spring force, the relief valve's disk will be pushed away from the seat. The pressure at which a relief valve just starts to open is sometimes referred to as the set or cracking pressure. However, as the valve's disk is forced off its seat, the compression of the valve's spring and the pressure required to continue to open the valve increase.

The pressure at which a relief valve is fully open and has the full capacity passing through it, which is sometimes referred to as the full-flow bypass pressure, generally exceeds the cracking pressure by 10 to 20 percent. This full-flow bypass pressure should always be less than the maximum allowable working pressure of the protected system and its components. In addition, a relief valve should be large enough to properly relieve the pressure in the system, i.e., the valve's capacity should not be less than the capacity of the pressure source. Also, shutoff and check valves should not be installed between a relief valve and the pressure source or in a reliefvalve's outlet line. It is common for a relief valve to have a lever (not shown in figure 11-36) that can be used to manually open the valve.

SENTINEL VALVE

Like a relief valve, a sentinel valve generally opens when the pressure in the system or component in which the valve is installed exceeds a preset value. However, a sentinel valve is generally not large enough to relieve

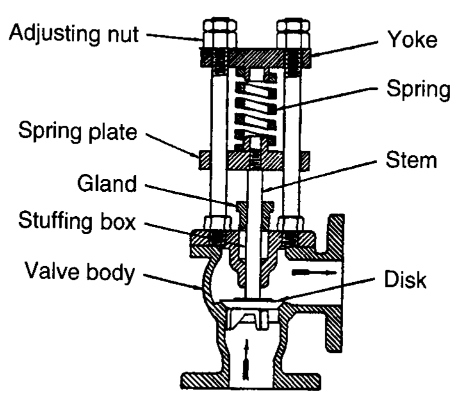


Figure 11-36. Spring-loaded relief valve

the pressure in the system or component. Instead, its purpose is to make a noise that will warn the operator of an abnormally high pressure.

CONTROL VALVE

A control valve is fitted with an actuator that controls the position of the valve's flow-restricting element in response to a pneumatic, hydraulic, or electric signal. A typical pneumatically-operated diaphragm control valve is controlled with compressed air that applies a force to a spring-opposed diaphragm enclosed within a chamber located at the top of the valve's actuator. The operating air pressure sent to the diaphragm actuator can be based on a measured variable, such as pressure, temperature, flow rate, or level, in the same system in which the control valve is installed or in a different system. Automatic actuators are most frequently used with globe-style (fig. 11-37a), ball, and butterfly control valves.

When a pneumatically-operated diaphragm actuator is direct acting, the control air supplied to the actuator acts on the upper surface of the actuator's diaphragm and moves the actuator stem downward. If there is a reduction in the air pressure, the actuator's spring moves the stem up. In a typical reverse-acting diaphragm actuator, the control air is supplied to the underside of the diaphragm and moves the stem up. With a reduction of control air pressure, the reverse-acting actuator's spring moves the stem downward. When either a direct- or a reverse-acting actuator is used with a globe-style control valve, the movement of the actuator's stem is directly transmitted through the control valve's stem to each of the valve's flow-restricting

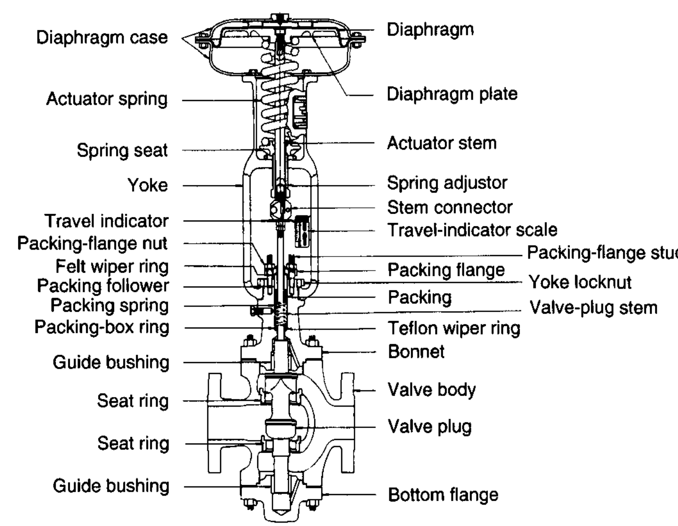


Figure 11-37a. Globe-style diaphragm control valve.
Courtesy Fisher Controls International, Inc.

plugs, which may be upward seating or downward seating. A control valve is considered to be normally closed, or to fail closed, if it closes with a loss of control air and normally open, or to fail open, if it opens with a loss of control air (e.g., an upward seating valve furnished with a direct-acting actuator will be normally closed).

Although a globe-style control valve may have a single plug, two plugs are sometimes used for hydraulic balance. The two plugs are arranged so that the higher-pressure fluid entering the valve acts against the upper surface of one plug and the lower surface of the second plug. Consequently, the net force that must be overcome when the valve is being opened or closed is greatly reduced. tn general, a double-port valve will also be suitable for use with higher flow rates than a comparably-sized single-port valve. However, because of various factors, such as differences between the thermal expansion of a valve body and its stem, double-port valves fitted with two plugs often do not seal tightly when they are fully closed.

The internal alignment ofa globe-style control valve's stem and plugs may be maintained by an upper guide bushing located in the valve's bonnet and a lower guide bushing in the bottom ofthe valve body (fig 11-37b). With this top-and-bottom guided arrangement, the valve's plugs are located between the two bushings. In a stem-guided valve (fig. 11-37c) or a top-guided valve (fig. 11-37d), the plugs maybe mounted on the lower end of a stem that passes through only an upper guide bushing. Some valve stems fitted with only an upper bushing, however, receive additional guidance from extensions on each plug that remain engaged with the mating port in the valve body even when the valve is fully open (fig. 11-37e). Plug extensions frequently resemble wings attached to the sides or bottom of a plug, or they can form V-ports in a plug. Port-guided plugs with extensions are also frequently utilized to maintain the alignment of valve stems that have no guide bushings (fig. 11-37£).Alternatively, guidance for a plug may be provided by a body-mounted cage that surrounds it (fig. 11-37g).

Some automatic control valves are fitted with a handwheel that can be used to limit how far the valve can be opened or closed by the control device or to permit the valve to be operated manually in the event of a control-system or valve-actuator failure.

The capacity passing through a valve can often be determined from the valve's flow coefficient, C_u. With liquids, this coefficient is equal to the capacity in U.S. gpm offreshwater having a specific gravity of 1.0 that will pass through a valve when the differential pressure across the valve is 1.0 psi. Using this coefficient, the rate at which liquid with a viscosity close to that of water will flow through a valve, based on fully turbulent, noncavitating flow and the use of piping equal in size to the valve (a valve's size is equal to the nominal size ofthe valve's inlet and outlet connections), can be predicted as follows:

$$Q = k_1 C_v \sqrt{\frac{p_1 - p_2}{sg}} \tag{11.1}$$

Q = volumetric flow rate through the valve, U.S. gpm (m^3/hr)
k_1 = 1.0 when using USCS units of measurement (0.0865 when using metric units)
C_v = flow coefficient
p_1 = valve inlet pressure, psia (kPa abs)
p_2 = valve outlet pressure, psia (kPa abs)
sg = specific gravity of the fluid passing through the valve (1.00 for freshwater at 60°F or 15.6°C)

A correction must be added to equation 11.1 if the valve size is less than the size of the piping in which the valve is installed. (Control valves are typically not installed in piping that is smaller than the valve size.)

A similar flow coefficient, K_v, which can be used with metric units of measurement, is equal to the capacity in m^3/hr of freshwater at a temperature from 5° to 40°C that will flow through a valve when the differential pressure across the valve is 1 bar. The relationship between K_v and C_v is shown below:

$$K_v = 0.865(C_v) \tag{11.2}$$

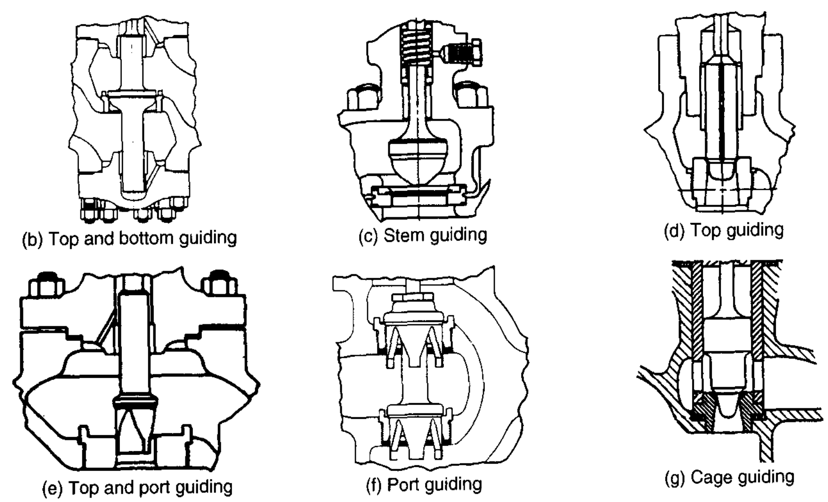


Figure 11-37b–g. Control-valve stem arrangements.
Figures 37b through 37f courtesy Fisher Controls International, Inc.
Figure 37g courtesy Leslie Controls, Inc.

The value of C_v (or K_v) for a given valve is reduced with the amount that the valve is closed. Additionally, the value of C_v (or K_v) can sometimes be reduced by using restricted trim, i.e., less than full-size ports and plugs, with a given size valve body, which reduces the flow area and increases the resistance to flow within the valve. (Trim refers to the internal parts of a valve, such as the stem, plugs or disks, and seat rings, that are exposed to the fluid passing through the valve.)

EXAMPLE 11-1: What flow rate of freshwater at a temperature of 180°F (sg = 0.972) will pass through a valve having a flow coefficient (C_v) of 15 if the pressure at the inlet to the valve is 100 psig and the valve's outlet pressure is 20 psig?

Solution: Using equation 11.1

$$Q = 15\sqrt{\frac{(100 \text{ psig} + 14.7 \text{ psia}) - (20 \text{ psig} + 14.7 \text{ psia})}{0.972}}$$
$$= 15\sqrt{\frac{(114.7 \text{ psia} - 34.7 \text{ psia})}{0.972}} = 136.1 \text{ U.S. gpm}$$

As liquid passes through the location within a valve where the cross-sectional area of the flow stream is at its minimum value, which is referred to as the vena contracta and is ordinarily located just downstream from the smallest opening within the valve, the liquid's velocity reaches the highest value and the local pressure of the liquid is reduced because of a conversion of pressure head to velocity head. If this pressure drops below the liquid's vapor pressure, a portion of the liquid will be permitted to flash into vapor. After passing through the vena contracta, the liquid decelerates and some of its velocity head is converted back into pressure head, resulting in an increase in pressure. If after the pressure increases it continues to be below the liquid's vapor pressure, the vapor bubbles created in the vena contracta will remain in the fluid stream. However, if the pressure rises above the liquid's vapor pressure, the vapor bubbles will implode. This two-step process (flashing followed by bubble collapse) is referred to as cavitation and can result in the generation of very high local pressures, noise, vibration, and mechanical damage to the valve. The potential for flashing or cavitation to occur within a valve can often be predicted based on a valve's cavitation index, also referred to as the index of incipient cavitation. The cavitation index based on the flow conditions can be calculated using the following:

$$K_c = \frac{p_1 - p_2}{p_1 - p_v} \tag{11.3}$$

where

K_c = cavitation index (calculated based on flow conditions)
p_v = true vapor pressure of the fluid passing through the valve (at the valve inlet temperature), psia (kPa abs)

When the value of K_c calculated using equation 11.3 exceeds the actual value of K_c for the valve being evaluated but is less than one, cavitation can occur within the valve. If, however, the value of K_c calculated using equation 11.3 exceeds one, only flashing can occur. It should be noted that the exact point of incipient cavitation or flashing (i.e., the point at which cavitation or flashing just begins) is affected by many variables, such as the amount of gas or impurities entrained in the liquid passing through a valve. This should be considered when selecting a valve and evaluating calculated results.

Typical values of K_c for fully-open single-port control valves are in the range of 0.65 to 0.75. Using a valve's cavitation index, the maximum allowable pressure differential across a valve that will not result in flashing or cavitation can be determined as follows:

$$\Delta p_{\text{max,cav}} = K_{c,\text{valve}}(p_1 - p_v) \tag{11.4a}$$

where

$\Delta p_{\text{max,cav}}$ = maximum allowable pressure differential across valve to prevent flashing and cavitation, psi (kPa)
$K_{c,\text{ valve}}$ = valve's cavitation index

By substitution, the minimum pressure at the outlet from a valve that is necessary to prevent cavitation and flashing can be found using the following:

$$p_{2,\text{min,cav}} = p_1 - K_{c,\text{valve}}(p_1 - p_v) \tag{11.4b}$$

where

$p_{2,\text{min,cav}}$ = minimum allowable pressure of liquid leaving the valve necessary to prevent flashing and cavitation, psia (kPa abs)

EXAMPLE 11-2: The cavitation index (K_c) for the valve used in example 11-1 is equal to 0.72. (The vapor pressure of freshwater at a temperature of 180°F is 7.506 psia.)

(a) Will cavitation occur within the valve?

(b) If the answer to part (a) is yes, how can cavitation within this valve be prevented?

Solution:

(a) Using equation 11.3

$$K_c = \frac{114.7 \text{ psia} - 34.7 \text{ psia}}{114.7 \text{ psia} - 7.506 \text{ psia}} = 0.75$$

Because K_c (0.75) exceeds $K_{c,\text{valve}}$ (0.72) but is less than 1.0, cavitation will probably occur within the valve.

(b) To prevent cavitation, the differential pressure across this valve would have to be reduced until the valve's outlet pressure (p_2) is not less than the following minimum value, which is found using equation 11.4b:

$$p_{2,\text{min, cav}} = 114.7 \text{ psia} - 0.72(114.7 \text{ psia} - 7.506 \text{ psia}) = 37.5 \text{ psia} = 22.8 \text{ psig}$$

Note: As an alternative to reducing the pressure drop through the valve considered in this example, cavitation could be prevented with the original differential pressure by selecting a different valve with a cavitation index ($K_{c,\text{valve}}$) not less than 0.746.

The flashing of liquid within a valve results in an increase in the mean specific volume and local velocity of the fluid passing through the valve. When the amount of vapor that forms is sufficient to result in choked flow (i.e., when the speed of sound is reached at the vena contracta), the flow through the valve will no longer increase in response to reductions in outlet pressure.

The shape of a control valve's plugs and ports affects the relationship between the position of the valve stem, or valve travel, and flow rate through the valve. Three common control-valve-flow characteristics are shown in figure 11-38. With a fixed inlet pressure, the flow rate passing through a linear valve is directly proportional to the amount that the valve is opened. A linear valve may be used in an application where the desired flow rate is directly proportional to valve travel or to the control signal supplied to the valve's actuator (e.g., level control). The flow rate passing through an equal-percentage valve varies in accordance with an exponential function of valve travel. With this relationship, if an equal-percentage valve is opened in a series of small but equal increments, the change in the flow rate passing through the valve after

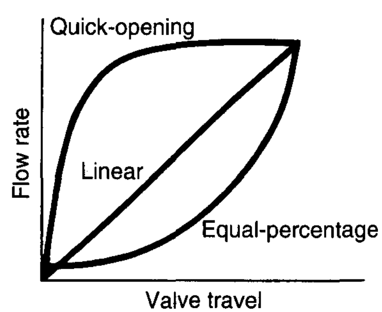


Figure 11-38. Control-valve-flow characteristics. Courtesy Fisher Controls International, Inc.

each adjustment will be equal to a constant percentage of the flow rate that was passing through the valve just before the adjustment was made. Consequently, the change in flow rate with valve travel will be relatively small when the valve plug is near its seat but will increase rapidly when the valve is nearly fully opened. Equal-percentage valves are used in some throttling applications. When a quick-opening valve is initially opened, the flow rate passing through the valve increases rapidly in a linear fashion. After the valve is opened a certain amount, however, the rate at which the flow rate increases with valve travel drops and approaches zero as the valve reaches its fully-opened position. Quick-opening valves are frequently used when a valve is required to be in only two positions (i.e., open or closed) with no throttling.

CONTROL PILOTS

The pressure sent to an automatic control valve's actuator may be regulated by a control pilot. A description of some commonly used types of pilots follows.

Direct-Acting Constant-Pressure Control Pilot

A constant-pressure control pilot typically regulates the pressure of the output signal that it sends to the control valve's actuator based on a measured pressure, which is frequently the fluid pressure being regulated by the control valve. When the pilot is direct-acting, the pilot's output signal increases as the controlled pressure rises above the set point. In a common application for a direct-acting constant-pressure pilot, the pilot is used to control an upward-seating valve fitted with a direct-acting actuator and installed in a piping system downstream from the pilot's sensing line (fig. 11-39a). To help ensure that accurate pressure signals are sent to a control pilot, a pilot's sensing lines should never include loops. As the pressure on the inlet side of the valve rises, the output pressure from the pilot also rises, and the amount that the control valve is open increases. A reduction in the pressure sensed by the pilot will have the opposite effect and cause the control valve to close more.

A typical direct-acting constant-pressure control pilot is shown in figure 11-40a. The operating fluid, which is often compressed air, is directed at a constant pressure (often, approximately 20 to 30 psig or 140 to 210 kPa) to the pilot's operating-supply port. In addition, fluid at the controlled pressure is directed to the chamber above the pilot's upper diaphragm. Whenever the downward force exerted by the fluid acting on top of the upper diaphragm is equal to the combined upward force exerted by the pilot's adjusting spring and by the operating fluid acting on the lower surface of the nozzle diaphragm, both the bleed nozzle and the pilot valve will be closed. While the pilot is in this balanced condition (the condition shown in fig. 11-40a), the output pressure sent to the control valve's actuator remains

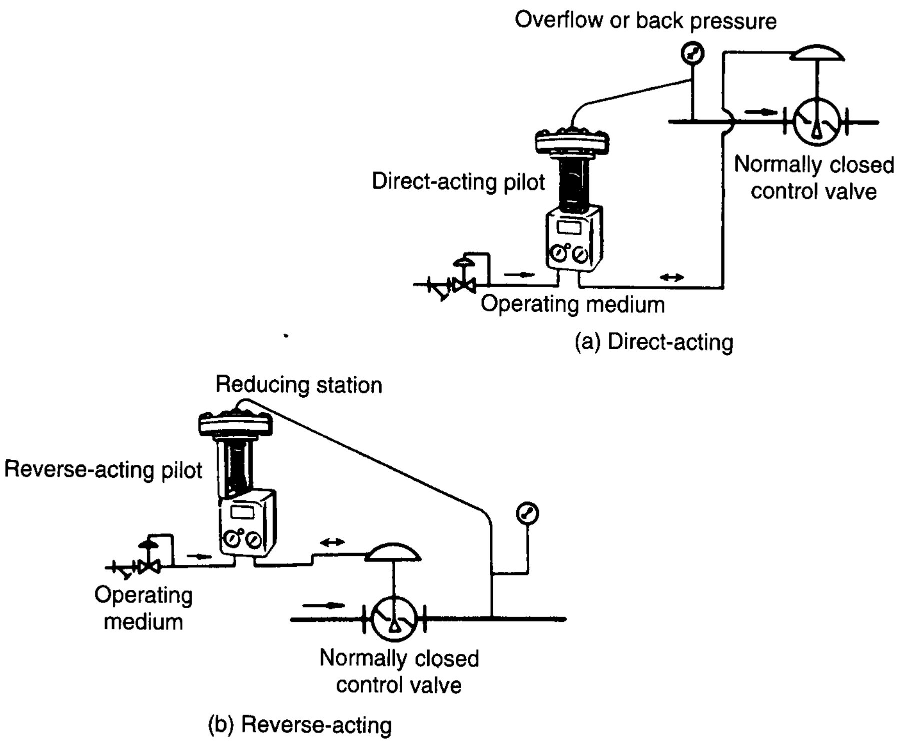


Figure 11-39. Typical constant-pressure control pilot applications. Courtesy Leslie Controls, Inc.

constant. If, however, the controlled pressure acting on the top of the pilot's upper diaphragm increases sufficiently to overcome the upward force exerted by the pilot's adjusting spring, nozzle diaphragm, and pilot-valve spring, the pilot's stem will be pushed downward, and the pilot valve will open. Operating fluid will then pass through the open pilot valve and the pilot's output pressure will increase.

Alternatively, if the controlled pressure of the fluid acting on the top of the pilot's upper diaphragm drops when the pilot is in the previously described balanced condition shown in figure 11-40a, the pilot's stem will be pushed upward. Initially, the nozzle diaphragm will deflect upward and the nozzle will remain seated against the nozzle disk. During this period, the output pressure sent to the control-valve actuator will not change. Because of this characteristic, small disturbances in a system that result in minor reductions in the controlled pressure are ignored by the pilot. However, if the controlled pressure continues to drop, there will be continued upward motion of the pilot's stem, the nozzle will eventually contact the nozzle stop, and the nozzle disk will be lifted away from the nozzle. Operat-

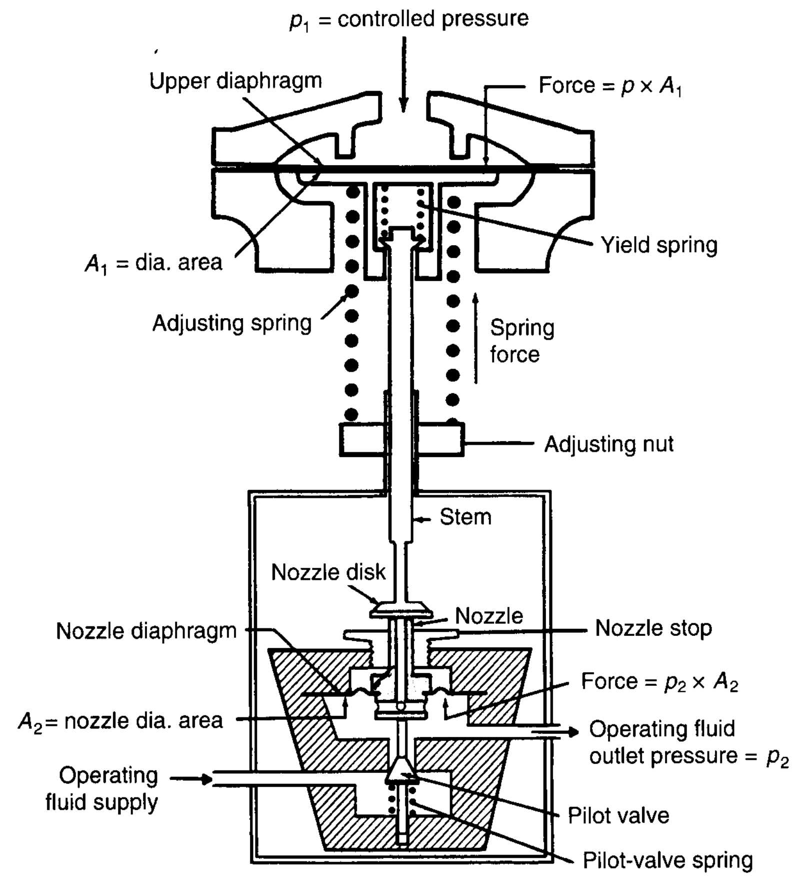


Figure 11-40a. Direct-acting constant-pressure control pilot. Courtesy Leslie Controls, Inc.

ing fluid will then bleed through the open nozzle, and the pilot's output pressure will drop.

The set point for the pilot shown in figure 11-40a is adjusted by turning the adjusting nut. Screwing the nut upward increases the compression of the adjusting spring and, therefore, the pressure that must act on the pilot's upper diaphragm to increase the output pressure (i.e., the controlled-pressure set point is raised). Screwing the adjusting nut downward reduces the set point of the controlled pressure.

In some control pilots, the pilot valve shown in figure 11-40a is replaced with a fixed metering orifice that is used in series with the variable-bleed nozzle (fig. 11-40b). With this arrangement, the output pressure sent by the pilot to the control-valve actuator, which drops as the bleed rate through the nozzle increases, varies almost linearly with the position of

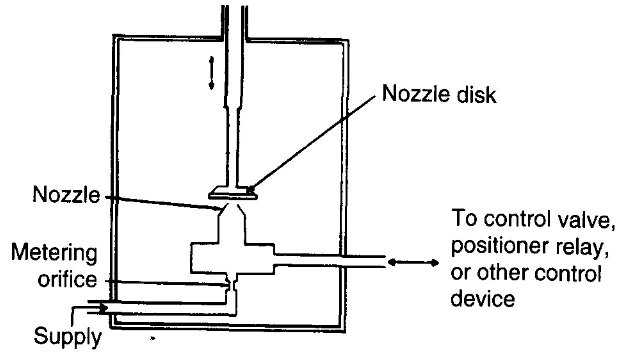


Figure 11-40b. Control-valve-pilot metering orifice. Courtesy Leslie Controls, Inc.

the nozzle disk. Therefore, because the nozzle disk is moved by the pilot's stem in response to changes in the controlled pressure acting on the upper diaphragm, the output pressure sent by the pilot to the control valve's actuator is nearly proportional to the controlled pressure. However, because the metering orifice does not close, operating fluid must be bled continuously through the pilot's nozzle whenever the required output pressure is less than the pressure at which the operating fluid is supplied to the pilot.

The ratio of the change in the controlled pressure acting on the pilot's upper diaphragm divided by the corresponding change in the pilot's output pressure is referred to as the pilot's proportional band width. Therefore, a pilot with a narrow proportional band width will produce a relatively large change in its output pressure in response to a small change in the controlled pressure. Although this will reduce the magnitude of variations of the controlled pressure necessary to initiate changes in the pilot's output pressure, because of time delays in the response ofthe controlled pressure to changes in the pilot's output pressure, operation with too narrow a proportional band width can lead to overcorrection and instability.

To enable the proportional band width of a control pilot to be adjusted based on the response time of the system in which the pilot is installed, some control pilots are fitted with a blade spring that acts in series with the adjusting spring (fig. 11-40c). The blade spring, which is installed on the pilot's stem above the nozzle disk, is supported by two movable slide blocks. The position of the blocks can be changed by turning an adjusting screw. As the slide blocks are moved closer together, the blade spring's stiffness increases. Because of this increased stiffness, a given change in the controlled pressure results in less movement of the nozzle disk and a smaller change in the output pressure. The pilot's proportional band width is, therefore, increased. Moving the slide blocks farther apart reduces the blade-spring stiffness and the pilot's proportional band width. In most

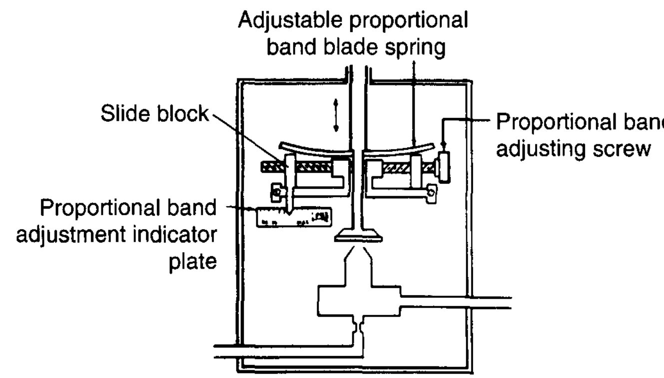


Figure 11-40c. Control-pilot adjustable proportional band width. Courtesy Leslie Controls, Inc.

cases, this type of a control pilot is adjusted to operate with the narrowest proportional band width that does not result in unstable operation.

Reverse-Acting Constant-Pressure Control Pilot

In a reverse-acting constant-pressure control pilot, the output pressure sent by the pilot to the control valve increases as the controlled pressure that acts on the pilot's upper diaphragm drops. This can be accomplished by placing a lever between the pilot's stem and the disk of a variable-bleed nozzle (fig. 11-41a). With this arrangement, upward motion of the pilot's stem resulting from a reduction in the controlled pressure closes the bleed nozzle and opens the pilot valve. The pilot's output pressure, therefore, increases. An increase in the controlled pressure has the opposite effect and causes the pilot's output pressure to drop. Alternatively, the pilot may be supplied with an upward-seating variable-bleed nozzle valve that is in series with a fixed metering orifice and opens to reduce the output pressure as the pilot's stem is pushed downward by an increasing controlled pressure (fig. 11-41b). The reverse-acting constant-pressure pilot's controlled-pressure set point can be raised by screwing the adjusting nut upward and reduced by screwing the adjusting nut downward. In addition, the method used to adjust the proportional band width, when it is adjustable, is similar to that described previously for the direct-acting constant-pressure pilot.

A typical application for a reverse-acting constant-pressure pilot is to control an upward-seating reducing valve fitted with a direct-acting actuator installed in a piping system upstream from the pilot's sensing line (fig. 11-39b). As the pressure in the sensing line rises, the operating-air signal to the control valve will drop and allow the control valve to close more. A reduction in the pressure sensed by the pilot will have the opposite effect and cause the control valve to open more.

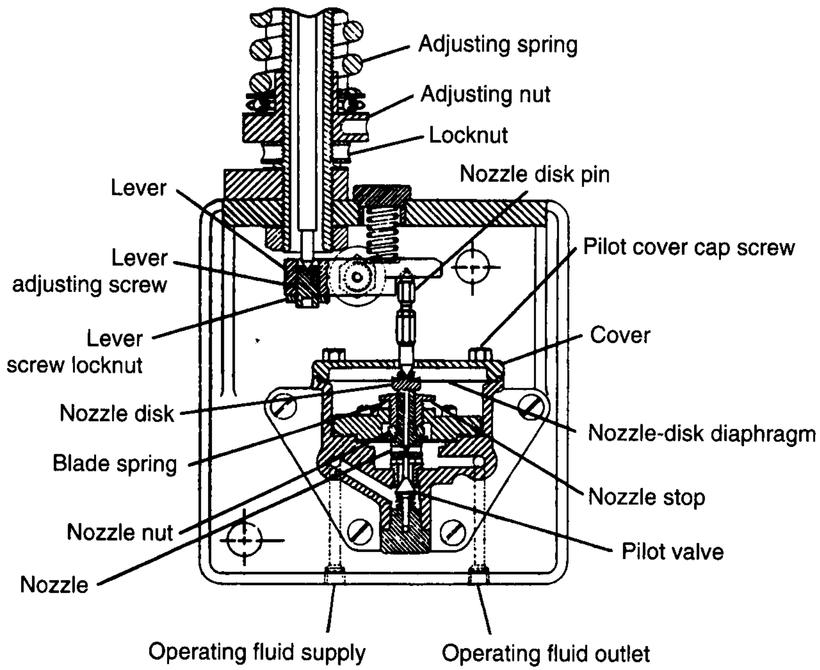


Figure 11-41a. Reverse-acting constant-pressure control pilot with lever
Courtesy Leslie Controls, Inc.

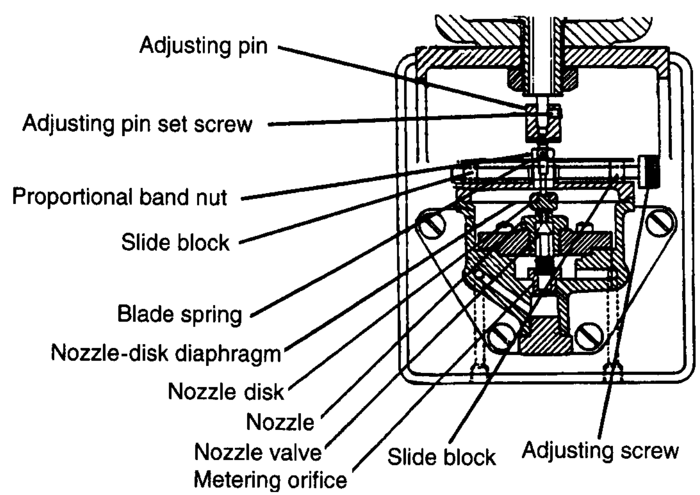


Figure 11-41b. Reverse-acting constant-pressure control pilot with upward-seating nozzle valve. Courtesy Leslie Controls, Inc.

Differential-Pressure Control Pilot

A differential-pressure control pilot regulates the pressure of its output signal based on the difference between two pressures that it senses. The two pressurized fluids are directed through connections to the upper and lower chambers in the top of the pilot (fig. 11-42). When the pilot is installed so that the higher-pressure fluid acts on the top surface of the upper diaphragm and the lower-pressure fluid acts on the bottom surface of the lower diaphragm, screwing the adjusting nut upward increases the differential pressure maintained by the pilot. However, when the higher pressure

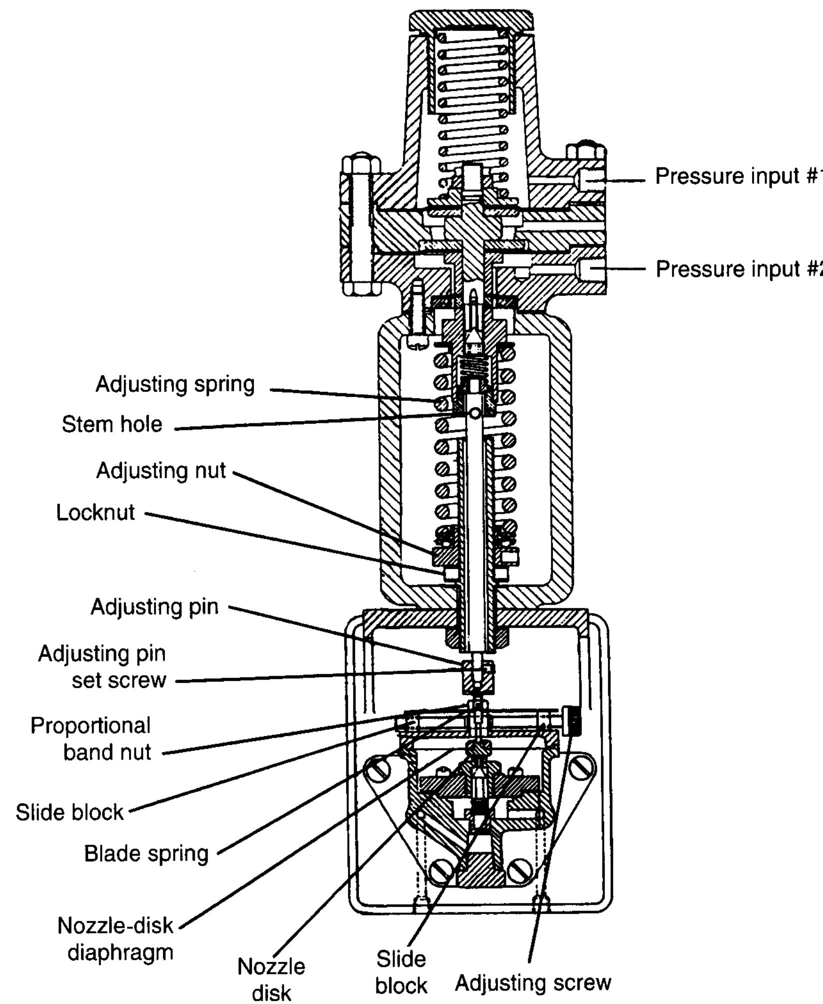


Figure 11-42. Differential-pressure control pilot.
Courtesy Leslie Controls, Inc.

is applied to the bottom surface of the lower diaphragm and the lower pressure is applied to the top surface of the upper diaphragm, screwing the adjusting nut upward reduces the differential pressure maintained by the pilot. The output pressure from a direct-acting differential-pressure pilot increases as the pilot's stem moves down, while the output pressure from a reverse-acting differential-pressure pilot increases as the stem moves up. (The differential-pressure pilot depicted in figure 11-42 is reverse-acting.) Some differential-pressure pilots are furnished with an adjustable proportional band width that is similar to the adjustable band width previously described for the constant-pressure pilot. Typical applications for differential-pressure pilots include maintaining a controlled pressure at a constant differential above or below a second pressure (fig. 11-43a) and maintaining a constant pressure drop across a control valve (fig. 11-43b).

Liquid-Level Control Pilot

A liquid-level control pilot is a differential-pressure pilot that regulates the output pressure sent to a control valve based on the liquid level in a tank or chamber (fig. 11-44). The valve receiving the output signal from the pilot is frequently used to control the flow of liquid either into or out of the chamber (fig. 11-45).

Typically, the pressure acting on one side of the pilot's upper diaphragm varies with the level being regulated, and the pressure on the opposite side of the diaphragm, which may be generated by a static column of liquid from the tank, is fixed. When the variable pressure sensed by the level pilot shown in figure 11-44 is applied to the top of the upper diaphragm and the fixed pressure is applied to the bottom of the diaphragm, the pilot will be direct-acting, and its output pressure will increase as the

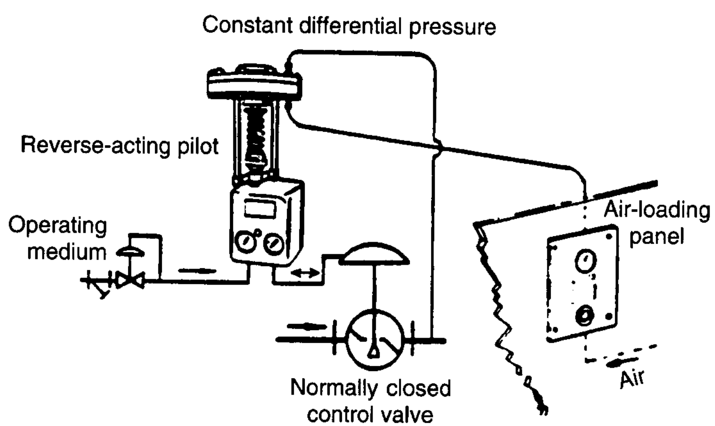


Figure 11-43a. Typical reverse-acting differential-pressure control-pilot application. Courtesy Leslie Controls, Inc.

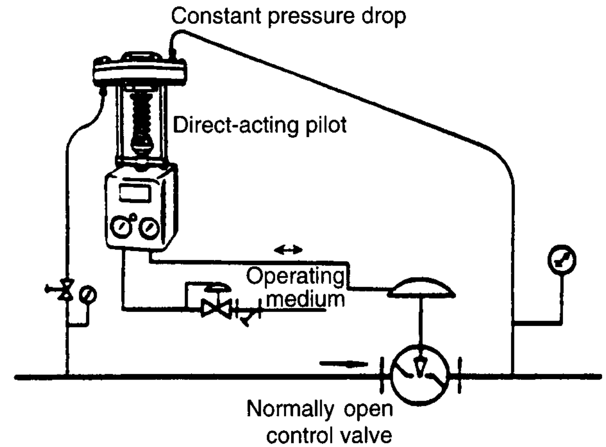


Figure 11-43b. Typical direct-acting differential-pressure control-pilot application. Courtesy Leslie Controls, Inc.

level being measured rises. The level set point can be raised by screwing the adjusting nut upward. Alternatively, when the variable pressure is applied to the bottom of the diaphragm and the fixed pressure is applied to the top of the diaphragm, the pilot will be reverse-acting, and its output pressure will increase as the level being measured drops. In this latter case, the level set point can be raised by screwing the adjusting nut downward. Some level pilots also have an adjustable proportional band width similar to the adjustable band width previously described for the constant-pressure pilot.

VALVE POSITIONER

In some installations, the output signal from a control pilot or other control device is sent to a valve positioner that then controls the pneumatic signal actually sent to an automatic control valve's actuator. A typical valve positioner is shown in figure 11-46a. Air at a constant pressure, which is typically between 20 psig and 35 psig (140 to 245 kPa), is supplied to the positioner's relay. Some of this air passes through an internal valve within the relay and is directed to the control valve's actuator, and some of the air passes through an internal fixed restriction and bleeds through the relay's nozzle (the internal arrangement of the relay is not shown in fig 11-46a). The bleed rate is controlled by the clearance between the nozzle opening and the end of the flapper.

Referring to figure 11-46a, an increase in the pressure of the pneumatic instrument input signal received from a control device causes the bellows to expand. As a result of this expansion, the D-shaped beam pivots about the input axis in a clockwise direction (when looking down at the top of the

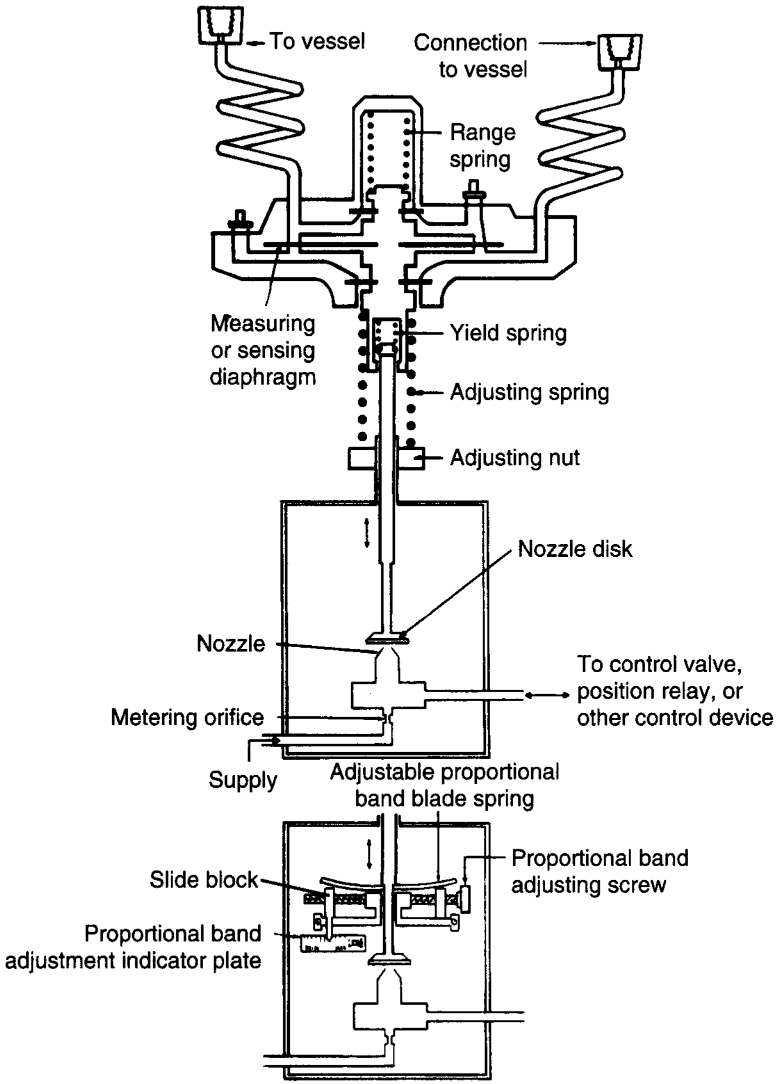


Figure 11-44. Liquid-level control pilot. Courtesy Leslie Controls, Inc.

valve), and the clearance between the end of the flapper and the nozzle is reduced. The bleed rate through the nozzle is, therefore, reduced, and the nozzle pressure rises. The increased nozzle pressure opens the relay's internal valve, and the output pressure sent to the control-valve actuator also increases. The downward motion of the control-valve stem resulting from the increased pressure sent to the actuator is transmitted through the linkage shown to the cam, and, as the cam rotates, it forces the beam to

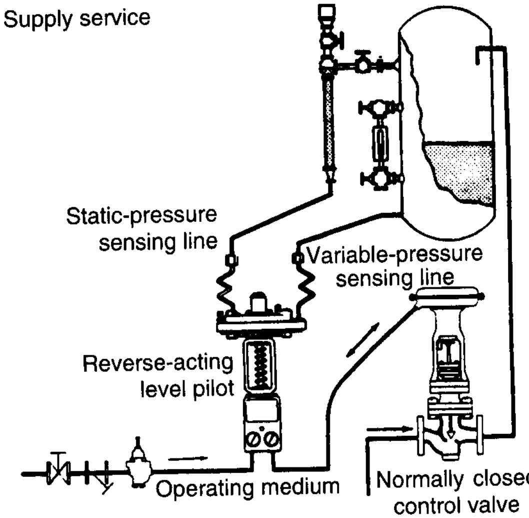


Figure 11-45a. Typical reverse-acting liquid-level control-pilot application. Courtesy Leslie Controls, Inc.

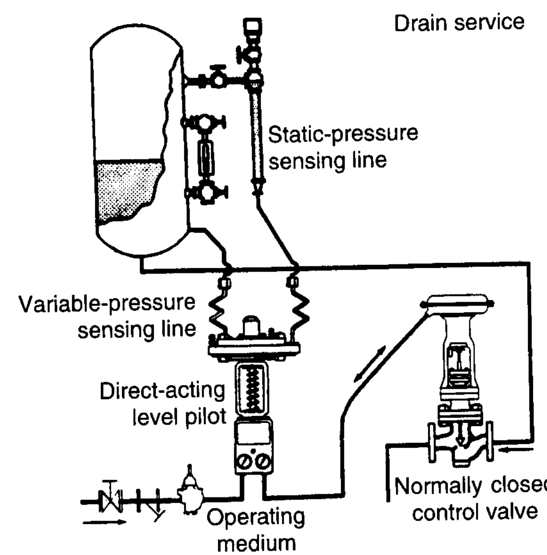


Figure 11-45b. Typical direct-acting liquid-level control-pilot application. Courtesy Leslie Controls, Inc.

rotate in a counterclockwise direction (when facing the left side of the valve) about the feedback axis. This rotation of the beam moves the end of the flapper away from the nozzle, reducing both the nozzle and the relay

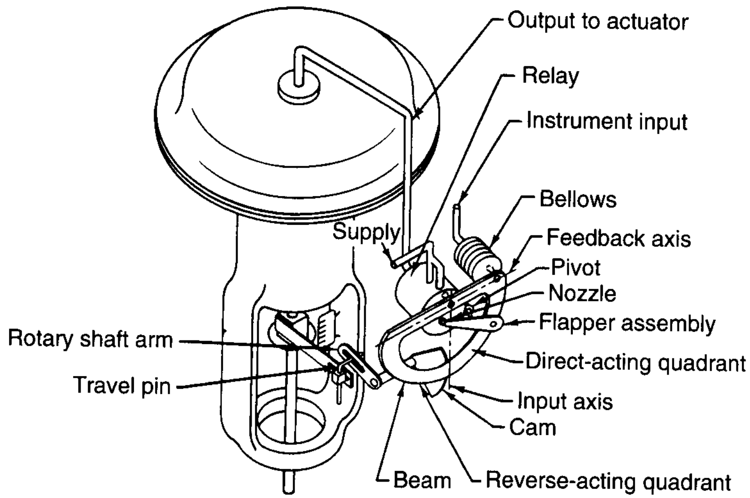


Figure 11-46a. Direct-acting control-valve positioner. Courtesy Fisher Controls International, Inc.

output pressures. When the relay's output pressure drops sufficiently for equilibrium to be restored in the control-valve's actuator (i.e., when the downward force exerted by the air pressure acting on top of the actuator's diaphragm equals the upward force exerted by the actuator's spring), movement of the valve stem stops. A reduction in the input signal sent to the positioner's bellows would have an effect that is opposite to that described. In some cases, the input to the positioner may be an electrical signal, which is converted to a pneumatic signal that is supplied to the positioner's bellows (fig. 11-46b).

The effect that a change in the input pressure to the positioner's bellows and the resulting rotation of the D-shaped beam have on the flapper-to-nozzle clearance (and, therefore, on the positioner's output pressure and the movement of the control-valve stem) can be adjusted by changing the position of the flapper assembly with respect to the beam. When the flapper is in the six o'clock position, input pressure has no effect on the flapper-to-nozzle clearance or on output pressure. As the flapper is rotated from the six o'clock position to either the three o'clock or the nine o'clock position on the beam, the amount that the output pressure will change for a given change in input pressure increases. However, when the flapper is between the six o'clock and the three o'clock positions, the positioner is direct-acting and an increase in input pressure reduces the flapper-to-nozzle clearance and results in an increased output pressure (fig. 11-46a). When the flapper is between the six o'clock and the nine o'clock positions, the positioner is reverse-acting and an increase in input pressure increases the flapper-to-nozzle clearance and results in a reduced output pressure (fig. 11-46b).

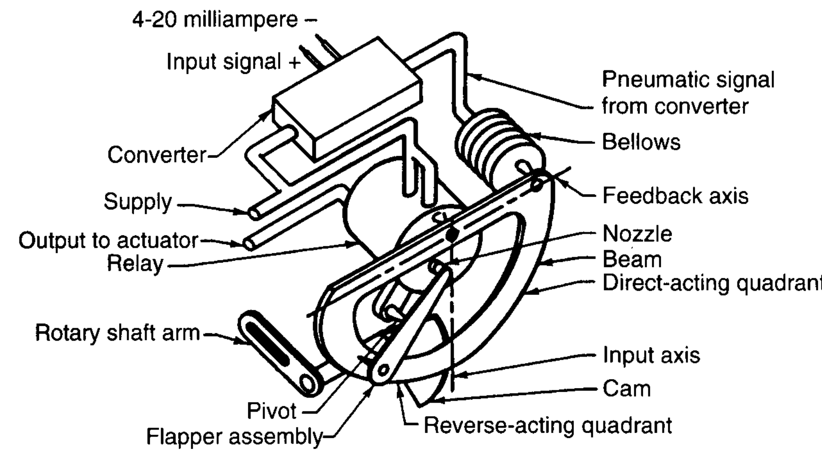


Figure 11-46b. Reverse-acting control-valve positioner with I/P converter. Courtesy Fisher Controls International, Inc.

Because the restoration of equilibrium (after the input pressure to the positioner's bellows has changed) is established by the rotation of the cam, the amount that the control-valve stem will move for a given change in the input pressure to the positioner's bellows can also be altered by changing the shape of the cam. This feature permits a valve with a particular flow characteristic to behave like a valve with a different characteristic. For example, the response of a positioner that can be fitted with one of three different cams is shown in figure 11-47a, and the flow characteristics of both an equal-percentage valve and a linear valve controlled by this positioner are shown in figures 11-47b and 11-47c, respectively.

ELECTRONIC CONTROLLER

The pneumatic signal to a control valve's actuator or positioner may be sent by an electronic controller that receives an electrical signal from a transmitter (e.g., thermocouple, pressure transducer, etc.), compares the value of the measured variable to a set point, and sends out an electrical output signal that varies with the error. A standard output is from 4 to 20 mA. This electrical output can then be converted to a pneumatic signal by a current-to-pressure (I/P) converter and sent to the valve (fig. 11-48a). The I/P converter is sometimes an integral part of the electronic controller (fig. 11-48b).

ORIFICES

An orifice is a restriction in a pipeline that can be used to limit flow or to produce a pressure drop. In some cases, the pressure drop across an orifice

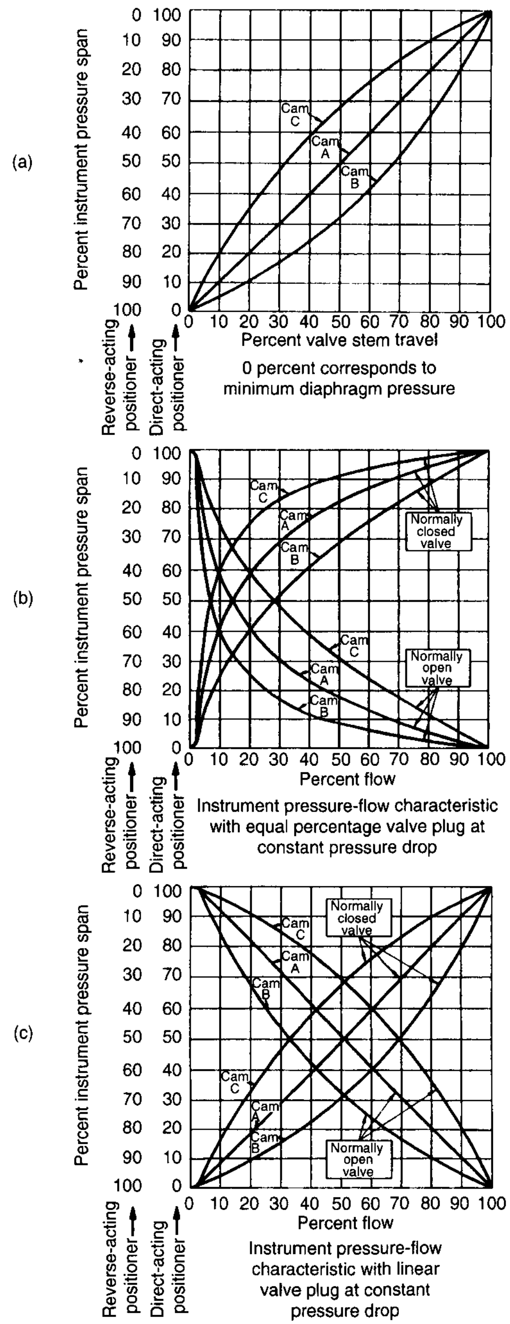


Figure 11-47. Control-valve characteristics with different positioner cams. Courtesy Fisher Controls International, Inc.

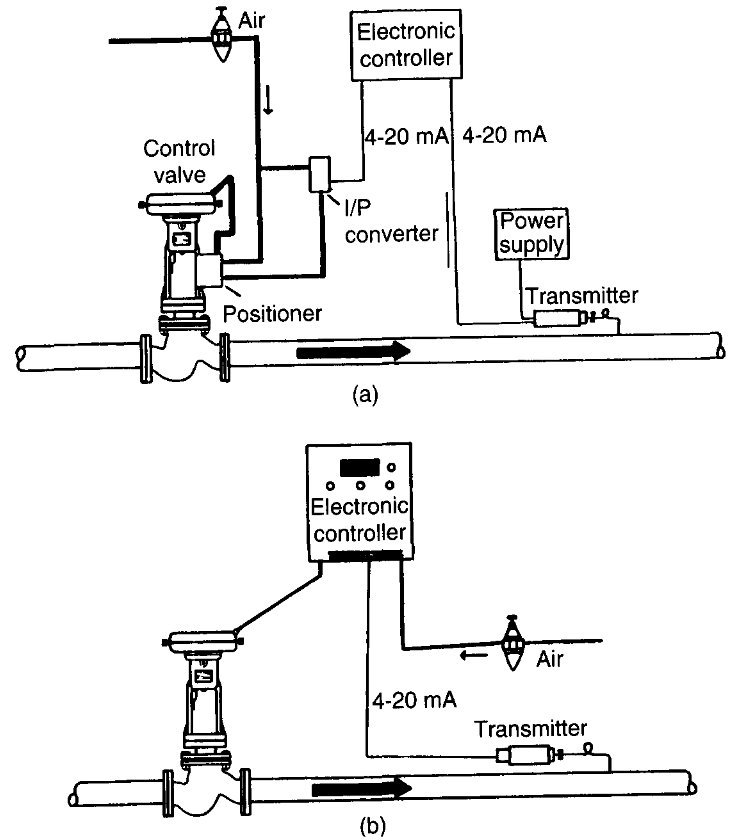


Figure 11-48. Control-valve electronic controllers. Courtesy Leslie Controls, Inc.

may be measured and used to determine the flow rate through a system. As fluid passes through an orifice, its velocity increases because ofthe reduction in flow area. The velocity reaches the highest value at the vena contracta, which is located downstream from the orifice. As the fluid's velocity increases, there is a conversion of potential energy to kinetic energy that results in a reduction of the fluid's local pressure. As with a valve, if the velocity in the vena contracta following an orifice reaches the speed of sound, flow will be choked. With choked flow, reductions in the downstream pressure have no effect on the mass flow rate passing through the orifice. In addition, if the pressure in the vena contracta drops below the fluid's vapor pressure, flashing and possibly even cavitation can occur. The flow rate of a noncavitating liquid through an orifice can be predicted from the following:

$$Q = k_2 C_{d,or} \frac{\pi (d_{or})^2}{4} \sqrt{\frac{2g\frac{\Delta p}{\gamma}}{1-\left(\frac{d_{or}}{d_i}\right)^4}} = k_2 C_{or} \frac{\pi (d_{or})^2}{4} \sqrt{2g\frac{\Delta p}{\gamma}} \tag{11.5}$$

where

- Q = volumetric flow rate through orifice, U.S. gpm (m^3/hr)
- k_2 = 37.4 when using USCS units of measurement (3.6E–3 when using metric units)
- $C_{d,or}$ = orifice discharge coefficient
- π = 3.1416
- d_{or} = diameter of hole in orifice, in. (mm)
- g = acceleration from gravity, 32.2 ft/s^2 (9.81 m/s^2)
- Δp = pressure drop through the orifice, psi (kPa)
- γ = liquid specific weight, lbf/ft^3 (kN/m^3)
- d_i = pipe inside diameter, in. (mm)
- C_{or} = orifice flow coefficient $\dfrac{C_{d,or}}{\sqrt{1-\left(\frac{d_{or}}{d_i}\right)^4}}$

Typical values of $C_{d,or}$ with liquids having viscosities close to that of freshwater include 0.61 for square-edged orifices and 0.98 for well-rounded orifices. When installing a square-edged orifice with a tapered opening, the taper should face downstream.

EXAMPLE 11-3: What is the flow rate of freshwater with a specific weight (γ) equal to 9.8 kN/m^3 that will pass through an orifice with a 50 mm diameter opening (d_{or}) and a discharge coefficient ($C_{d,or}$) of 0.70. The orifice is installed in a pipeline with a 150 mm inside diameter, and the pressure drop measured across the orifice (Δp) is 200 kPa.

Solution: Using equation 11.5

$$Q = 3.6\mathrm{E}{-3}(0.70)\frac{\pi(50\mathrm{mm})^2}{4}\sqrt{\frac{2\left(9.81\frac{\mathrm{m}}{\mathrm{s}^2}\right)\frac{200\mathrm{kPa}}{9.8\frac{\mathrm{kN}}{\mathrm{m}^3}}}{1-\left(\frac{50\mathrm{mm}}{150\mathrm{mm}}\right)^4}} = 99.6\frac{\mathrm{m}^3}{\mathrm{hr}}$$

STRAINERS

Strainers are often used to remove solid contaminants from fluid passing through a piping system. A typical basket strainer resembles a perforated cylindrically shaped basket that is open at one end. A simplex basket

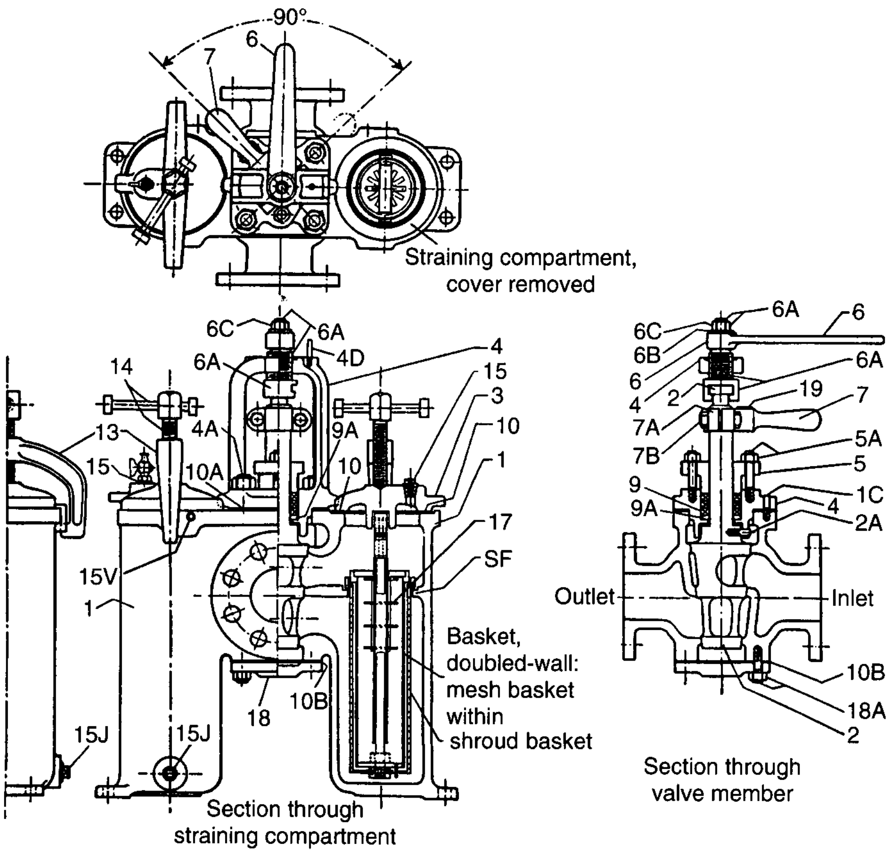
strainer has one basket that is installed in an enclosed cylindrical housing with in-line inlet and outlet ports. The centerline ofthe basket is often perpendicular to the piping's longitudinal axis, i.e., the strainer is mounted vertically in a horizontal run of piping. A duplex basket strainer has two baskets that are mounted side by side in a common housing (fig. 11-49). The two baskets in a duplex strainer are typically separated by a ported plug valve that can be rotated with a lever to direct the incoming flow through either basket. In addition, a jackscrew may be provided to lock the flow-diverting plug valve in place during operation. With this duplex arrangement, system operation can continue while the basket being bypassed is removed and cleaned. Duplex strainers are generally used in fuel-oil and lubricating-oil systems. They are also used in many seawater systems.

A Y-type strainer has a cylindrical screen that is open at both ends and is oriented at an angle with respect to the piping (fig. 11-50). The extended end ofthe strainer's housing typically protrudes from the bottom ofthe piping when used in a horizontal system and points in the direction of flow. After being installed in the housing, the screen is held in place and seats against a cover that is located at the end ofthe housing and can be removed to clean or replace the screen. In addition, a plugged or valved blowoff connection is often added to the housing's cover to permit debris to be blown out of the strainer while it is in service. Y-type strainers are generally used with relatively clean fluids where frequent cleaning is not required, e.g., steam systems.

A cone-type strainer consists of a conically shaped screen that is mounted between two mating flanges in a piping system. Oriented with the closed end of the cone pointing downstream, a cone-type strainer is generally used only temporarily during system startup and is then removed.

A strainer's basket or screen is often fabricated from perforated sheet or wire mesh. In addition, some perforated sheet baskets or screens have a fine wire mesh liner. Regardless of the configuration used, incoming fluid is directed into the center portion of a strainer's basket or screen and then passes through the perforated walls. With this flow path, foreign material that is too large to pass through the perforations is trapped within the strainer. Some basket strainers, such as those used in lubricating-oil systems, are fitted with magnetic inserts so that fine ferrous metal particles that could pass through the basket's perforations will be removed from the fluid stream.

The total flow area through the openings in a strainer's basket or screen should be at least three to four times the cross-sectional area of the outlet line. As this area ratio is reduced, a strainer will ordinarily have to be cleaned more frequently. The opening size in a basket or screen should be selected based on the size ofthe contaminants that could damage components



1	Body	7B	Machine screw
1C	Dowel pin	9	Packing
2	3-way duplex plug valve	9A	Packing-support ring
2A	Stop pin	10	Straining-compartment-cover gasket
3	Straining-compartment cover	10A	Bonnet gasket
4	Valve bonnet	10B	Bottom-plug-cover gasket
4A	Stud and nut	13	Straining-compartment-cover clamp
4D	Stop pin for jack lever	14	Clamp set screw with sliding tee-handle
5	Gland	15	Straining-compartment-cover vent
SA	Stud and nut	15V	Straining-compartment vent
6	Jack lever	15J	Straining-compartment drain
6A	Jack screw	17	Magnet assembly
6B	Washer	18	Bottom plug cover
6C	Nut	18A	Stud and nut
7	Plug-turning lever	19	Indicator plate
7A	Turning-lever cap	SF	Support flange

Figure 11-49. Duplex basket strainer. Courtesy Tate-Andale.

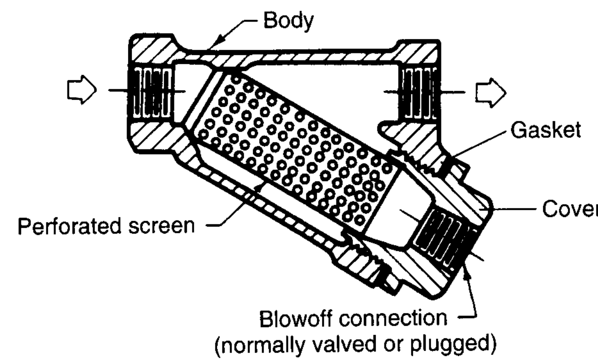


Figure 11-50. Y-type strainer. Courtesy Spirax Sarco, Inc.

downstream from the strainer, the viscosity of the fluid passing through the strainer, and the desired pressure drop across the strainer. Perforations in a sheet or basket are often specified in terms of hole size. Typical hole sizes range in diameter from 0.02 to 0.25 in. (0.5 to 6.4 mm). Wire screen size is often specified in terms of the diameter of the wire used and the number of strands of wire per unit length, e.g., a wire mesh basket liner may be constructed from 0.023-in. diameter wire with 20 mesh per linear inch. Commonly used mesh sizes range from 20 mesh, which will usually stop particles of approximately 840 microns (!lm) in diameter, to 200 mesh, which will usually stop particles of approximately 80 microns in diameter. (Wire screens are available with greater than 200 mesh; however, screens with mesh sizes in excess of 325 are usually classified as filters.)

When a differential pressure gauge is connected to piping on the inlet and outlet sides of a strainer, an increase in the pressure drop across the strainer is an indication that foreign material is accumulating within the basket or screen. Prior to cleaning a simplex strainer, valves at the inlet and outlet to the strainer should be closed. Alternatively, when cleaning a basket in a duplex strainer installed in a continuously operating system, inlet and outlet valves should remain open, and the incoming flow should be directed through the opposite basket. Typically, this can be accomplished by loosening the strainer's jackscrew, when one is provided to lock the flow-diverting plug in place, and then slowly moving the flow-diverting lever to rotate the plug valve and redirect incoming flow through the opposite basket. As the plug valve is turned, the amount of fluid passing through the side of the strainer being taken off-line will gradually be reduced while the amount passing through the strainer being placed on-line will gradually increase. (Switching over to the opposite basket of a duplex strainer should generally be done only if the basket being put on-line is clean, and its housing has been properly filled with the incoming fluid and vented of air.) Mter the strainer to be cleaned has been taken off-line, any pressure within its housing should be relieved before removing the cover. This can often be accomplished by cracking open a vent valve when one is provided. Also fluid should typically be drained from the housing to a level that will prevent unstrained fluid from passing into the outlet portion of the housing after the basket or screen has been removed (e.g., below support flange SF in fig. 11-49). Once the housing cover has been removed, the basket or screen can be extracted from the housing.

Mter being cleaned, the basket or screen can be reinserted into the housing. Mating surfaces on the housing and cover should be cleaned. In addition, when a gasket is used between the housing and cover, it should be inspected and either ;eused or, if necessary, replaced. The cover can then be replaced. Many strainer covers are threaded or are secured with a clamp. However, when bolts are used to hold the cover in place, they should be tightened evenly in an alternating pattern so that the cover will be drawn down squarely.

Prior to putting a simplex strainer back in service, the inlet valve should be cracked open to allow the housing to slowly fill with fluid. Mter the fluid has filled the housing, a vent valve, when one is provided in the housing cover, can be cracked open to allow the incoming fluid to force air out of the strainer housing. Venting with incoming fluid can only be accomplished, however, when this fluid has sufficient pressure to flow into the strainer. In cases where the incoming fluid is under a vacuum, it may be possible to fill the housing before the inlet valve has been opened with fluid from a pressurized source or by pouring fluid into the housing. (In cases where the presence of air will not adversely affect system performance, venting a strainer after it has been cleaned may not be necessary.) The strainer inlet valve can now be fully opened and the outlet valve can be slowly opened so that the strainer can be placed back into service.

In the case of a duplex strainer, the housing for the off-line basket can often be filled with incoming fluid by loosening the jackscrew that locks the flow-diverting plug valve in place, when a jackscrew is used, and by shifting the flow-diverting valve slightly in the direction of the off-line strainer. (In some cases, loosening the jackscrew is sufficient to allow incoming fluid to enter an off-line strainer and the shifting of the flow-diverting valve is not necessary.)

When a vent valve is provided and the incoming fluid is not under a vacuum, the off-line basket can then be vented. Following filling and venting, the flow-diverting lever can be slowly moved to reposition the flow-diverting plug valve and redirect the incoming flow through the clean basket. Alternatively, the flow-diverting valve can be placed in a position to completely bypass the basket that has just been cleaned, the locking jackscrew, when provided, can be retightened, and the clean basket can remain off-line until a changeover is desired. It is advisable, however, to re-vent the off-line strainer prior to placing it back into service to remove any air

that over time may have separated from the liquid within its housing. Extreme caution should always be exercised when draining, opening, or venting any strainer to prevent pressurized liquid from leaking at too high a rate from the housing, which can lead to a hazardous condition ifthe fluid is heated, flammable, or toxic.

STEAM TRAPS

A steam trap is a device that closes automatically to prevent or hinder the flow of steam through a line but then automatically reopens to allow water (condensed steam) to flow through the line. When steam is used as the heating medium in a heat exchanger, a trap is frequently installed in the drain line connected to the steam side of the heater to ensure that the heating steam condenses and releases its latent heat before leaving the heater. If the steam passing through a heater does not condense, the mass flow of steam required to raise the temperature of the fluid being heated will increase significantly, which will reduce plant efficiency. In addition, steam leaving a heater can pressurize a drain tank. A steam trap may also be installed in the drain for a steam line to permit condensed steam to be removed from the line, e.g., during warm-up. The trap, however, prevents steam from passing through the drain. To increase its effectiveness, a steam trap should normally be installed in the lowest horizontal run of pipe located below the line or component being drained. In addition, so that steam within a trap can condense rapidly, steam traps should never be painted or covered with insulation. A description of some commonly used types of steam traps follows.

Mechanical Steam Traps

FLOAT-AND-LEVER STEAM TRAP

A float-and-Iever steam trap has a float that is mounted on the end of a lever (fig. 11-51). When condensate enters the trap's body, the float rises with the water level within the trap, and the trap's outlet-valve disk, which is attached to the lever, is moved away from its seat. The condensate can then pass through the trap's open outlet valve. The float will continue to rise until it reaches its uppermost position or until the rate at which the condensate is entering the trap matches the outlet flow rate, whichever occurs first. If the condensate flow rate and the level of the condensate within the trap begin to drop, the float and lever will move downward and close in on the outlet valve until the outlet flow rate again matches the inlet flow rate. If the flow of condensate into the trap stops or if steam enters the trap,

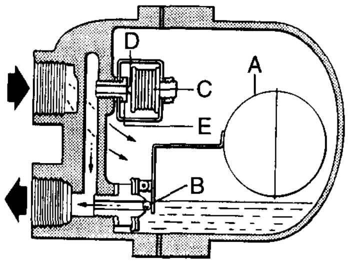


Figure 11-51. Float-and-lever steam trap. Courtesy Spirax Sarco, Inc.

the float and lever will drop to their lowest position and close the outlet valve. The flow of steam through the trap will, therefore, be prevented.

Because air cannot raise the float and open the outlet valve, air in a drain line that enters a float trap during start-up can become trapped within the body. The air can also prevent condensate from entering the trap. To permit air that may enter a float trap to be removed during start-up, a manual or thermostatically operated vent valve is sometimes provided near the top of the trap's body (a thermostatic-type vent valve, which is filled with a volatile fluid, opens when cold and closes when hot). In addition, a float trap often has a jacking screw that can be used to manually open the trap's outlet valve.

INVERTED-BUCKET STEAM TRAP

An inverted-bucket steam trap has a bucket mounted with its open end at the bottom (fig. 11-52). Fluid enters the trap through an opening located at the top of a vertical tube that extends up into the inverted bucket. When this fluid is water, it flows under the lower edge of the bucket and fills the trap's body, submerging the bucket. With the inverted bucket in this position, the trap's outlet valve is held open, and water is permitted to leave the trap.

If steam now enters the trap, it will displace the water inside the upper portion of the bucket. The buoyancy of the steam, because of its relatively low density compared to that of the water that surrounds the bucket, forces the bucket to float within the trap's body. As the bucket moves upward, the trap's outlet valve, which is attached to a hinged lever that is connected to the top of the inverted bucket, is pushed towards the valve's seat in the trap's body. The valve eventually closes, and the flow of steam through the trap is prevented.

Steam in the bucket, together with air or carbon dioxide that may have entered the trap, will slowly pass through a small vent hole in the top of the bucket and will fill the top of the trap's body. This will permit additional fluid to enter the trap. If this fluid is steam, the trap will remain closed. However, if water enters the trap, or if the steam within the bucket begins to condense because of heat lost due to radiation, the upward force applied

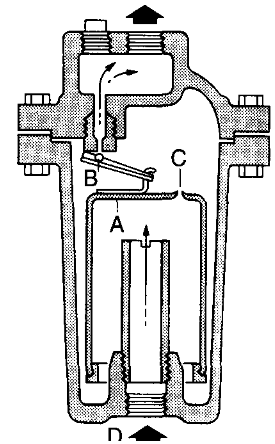


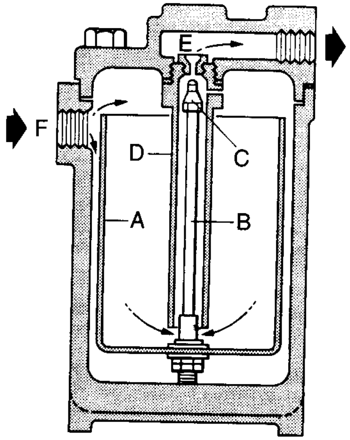
Figure 11-52. Inverted-bucket steam trap. Courtesy Spirax Sarco, Inc.

to the bucket will be reduced. When the combined effect of this force and the upward force resulting from the differential pressure across the trap's outlet valve is less than the effect of downward force exerted by gravity, the bucket will sink and open the trap's outlet valve. The flow through the trap will continue until steam again enters the trap and the above cycle is repeated. An inverted-bucket steam trap will often discharge continuously when operating with a low inlet pressure or a high back pressure. In addition, if the water seal around the open end of the bucket is lost, steam can blow through the trap. To help prevent this seal from being lost because of reverse flow, a check valve is sometimes incorporated into the inlet of an inverted-bucket trap.

OPEN-TOP-BUCKET STEAM TRAP

The bucket in an open-top-bucket steam trap is positioned with its open end on top (fig. 11-53). A vertical tube that is open at the bottom extends into the bucket. The seat for the trap's outlet valve is located at the top of this tube. The trap's valve is on the upper end of a vertical rod or spindle that is secured to the bottom of the bucket and fits up into the vertical tube. When condensate enters the trap, the bucket initially floats on the water and rises. However, once the condensate level within the trap reaches the top of the bucket, the water spills over into the bucket and forces it to sink. As the bucket moves downward, the outlet valve is pulled away from its seat. The pressure at the inlet to the trap can then force condensate that is inside the bucket to flow up through the vertical tube and out through the open outlet valve.

If steam with a sufficient pressure enters the trap, it will force the condensate out of the bucket. The bucket will then be able to float on the water



A Bucket
B Spindle
C Outlet valve
D Tube
E Outlet-valve seat
F Inlet connection

Figure 11-53. Open-top-bucket steam trap. Courtesy Spirax Sarco, Inc.

remaining in the trap's body. As the float rises, the outlet valve will be closed, and the flow of steam out of the trap will be prevented.

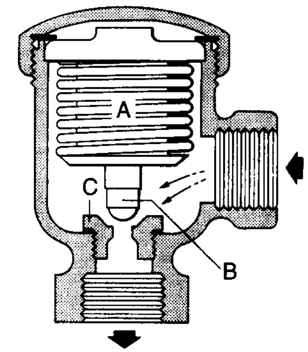
Thermostatic Steam Traps

BELLOWS-TYPE STEAM TRAP

A typical bellows-type thermostatic steam trap is fitted with a flexible hermetically sealed bellows that contains a volatile fluid (fig. 11-54). One end of the bellows is secured to the trap's body. The trap's outlet valve is secured to the opposite or free end of the bellows. When cool condensate enters the trap, the bellows contracts and the outlet valve's disk is pulled away from its seat. However, when hot condensate or steam enters the trap, the volatile fluid within the bellows is heated. As this fluid's volume increases, the bellows expands, and the outlet-valve disk is pushed against the valve's seat. Consequently, the flow of steam through the trap is prevented.

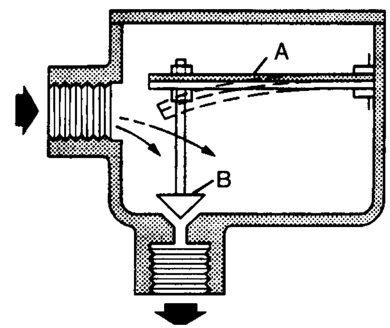
BIMETALLIC STEAM TRAP

The outlet-valve disk in a typical bimetallic steam trap is attached to an element consisting of one or more bimetallic strips (fig. 11-55). Each strip is constructed by bonding together two metals with different coefficients of thermal expansion. When the bimetallic element, which is secured at one end to the trap's body, is cool, the strips are flat, and the outlet-valve disk is held away from the valve's seat. However, when hot condensate or steam enters the trap and heats the element, the metal with the higher coefficient of thermal expansion in each bimetallic strip expands more than the other metal and forces the strip to become curved. The deflection of the bimetal-



A Thermostatic element
B Outlet valve
C Outlet-valve seat

Figure 11-54. Bellows-type steam trap. Courtesy Spirax Sarco, Inc.



A Bimetallic strip
B Outlet valve

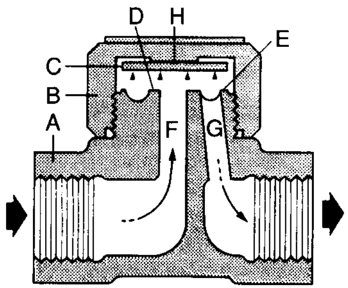
Figure 11-55. Bimetallic steam trap. Courtesy Spirax Sarco, Inc.

lic element moves the outlet-valve disk towards the seat and forces the valve to close. The valve will remain closed until steam in the trap condenses and cools sufficiently for the bimetallic element to straighten.

Thermodynamic Steam Traps

DISK-TYPE STEAM TRAP

A typical disk-type steam trap is fitted with a free-floating disk that rests against the seat in the trap's body (fig. 11-56). When cool condensate or a mixture of cool condensate and air enters the trap, it lifts the disk off the seat and flows through the outlet passage. As this condensate passes through the inlet orifice and under the disk, its velocity increases and, because of the resulting conversion of pressure head to dynamic head (i.e., potential to kinetic energy), its static pressure drops. Typically, as condensate continues to drain through the trap, the temperature ofthe incoming condensate gradually increases. Eventually, some ofthe reduced-pressure condensate begins to flash into steam. Because of the steam's relatively



A Body
B Top cap
C Disk
D Inner-seat ring
E Outlet-seat ring
F Inlet orifice
G Outlet passage
H Boss in top cap

Figure 11-56. Disk-type steam trap. Courtesy Spirax Sarco, Inc.

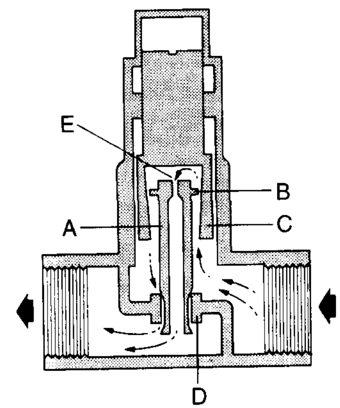
high specific volume, the average velocity of the steam and condensate mixture passing under the disk rises, the static pressure below the disk drops, and the disk is drawn down towards its seat. In addition, flash steam that passes into the space above the disk applies a downward force to the disk, which helps to close the trap. Because the area on top ofthe disk is greater than the area exposed to the inlet pressure, fluid at the entrance to the trap is initially unable to lift the disk off the seat. However, as the flash steam trapped above the disk gradually cools and condenses, the pressure above the disk drops. Eventually, the fluid at the trap's inlet can lift the disk off its seat, and the cycle is repeated.

A typical disk-type trap's disk has one smooth face and one face with one or more concentric grooves. When the disk is installed with the grooved face against the trap's seat rings, i.e., with the smooth face on top, the grooves break up the flow pattern and delay the reduction of the static pressure below the disk. This raises the minimum temperature at which hot condensate that enters the trap will begin to flash into steam and permits more condensate to be drained from the system. If, however, the disk is installed with its smooth face against the seat rings, condensate passing through the trap will flash and close the trap at a lower temperature. Consequently, more condensate will remain in the system.

When a disk-type trap is used with a low inlet pressure or a high back pressure, the resulting low flow rate and velocity through the trap can prevent the pressure below the disk from being reduced sufficiently to properly close the trap. In addition, if air enters a disk trap and fills the space above the disk, it can prevent the disk from being lifted off its seat rings by incoming fluid.

PISTON-VALVE-TYPE IMPULSE STEAM TRAP

The only movable piece in a piston-valve-type impulse steam trap is a hollow, piston-type valve with a thin disk near its upper end and a tapered lower end that passes through a seat in the trap's body (fig. 11-57a). When cool condensate, together with any air that may be within the inlet line,



A Outlet valve
B Disk
C Control cylinder
D Outlet-valve seat
E Control orifice

Figure 11-57a. Piston-valve-type impulse steam trap. Courtesy Spirax Sarco, Inc.

initially enters the trap during start-up, it exerts a force against the underside of the valve's disk and lifts the valve offits seat. Once the valve opens, most of the condensate entering the trap drains through the annulus formed between the lower end ofthe valve and the opening in the center of the valve's seat. The remaining portion of the incoming flow, which is referred to as the control flow, passes through the annulus formed between the periphery of the valve disk and the inner wall of the control cylinder that fits over the valve. The pressure ofthe control flow drops as it passes through this close-clearance space. Mter entering the chamber above the valve disk, referred to as the control chamber, the control flow passes through an orifice in the top of the hollow valve. As the control flow passes through this orifice, which is referred to as the control orifice, its pressure is reduced even further. The control flow then continues on through a port in the center ofthe hollow valve, mixes with the condensate that has flowed through the open valve's seat, and drains through the outlet port in the trap's body.

The surface area above the trap's valve disk is greater than the area below the disk. However, because the pressure of the control flow drops as it enters the control chamber above the disk, as long as the condensate remains cool, the force acting on the lower surface of the valve disk will be sufficient for the valve to stay open. As cool condensate gradually drains from the line in which the trap is installed, the temperature of the incoming condensate will typically increase. When this temperature reaches the saturation temperature of the reduced-pressure control flow, condensate passing through the control orifice will begin to flash into vapor. Condensate in the control chamber above the valve disk will also typically start to flash. Because ofthe resulting increase in the volume ofthis fluid, the flow rate through the control orifice will drop, and the pressure in the control chamber will increase. When the force acting on the upper surface of the valve disk exceeds the force applied to the disk's lower surface, the valve will be pushed downward against its seat, and flow between the valve and its seat will stop. However, a small flow rate can continue to pass through the control orifice.

As the condensate within the trap cools, flashing in the control orifice and the control chamber will be reduced or stop. The flow rate through the control orifice will, therefore, increase, the pressure acting on top of the valve disk will drop, the .valve will eventually rise and reopen, and the cycle will be repeated. An impulse trap can repeatedly open and close when operating with low condensate flow rates. In addition, its performance can be adversely affected by a moderate to high back pressure.

When the position of the control cylinder is adjustable, it is typically preset by the manufacturer. However, as a result of a reverse taper in the control cylinder's inner wall, if the control cylinder is screwed farther into the trap's body, the clearance between the control cylinder and the periphery of the valve disk when the valve is closed will increase. As a result of this increased clearance, there will be less of a pressure drop as any given amount of the control flow entering the chamber above the valve disk passes through the annulus formed between the disk and the cylinder, and there will be a greater force holding the piston valve closed. The flow. rate through the control orifice that is necessary for the control-chamber pressure to drop sufficiently for the valve to reopen will, therefore, increase. Consequently, it will take longer for the trap to reopen. Because of this, more condensate will collect in the drain line upstream from the trap. Raising the control cylinder has the opposite effect. (It should be noted that if the control cylinder is screwed too far into the trap's body, it will hold the valve closed and prevent the trap from operating properly. Conversely, if the control cylinder is raised too far, it will lift the trap's valve off of its seat and prevent this valve from ever closing.)

LEVER-VALVE-TYPE IMPULSE STEAM TRAP

The lever-valve-type impulse steam trap, which is designed to handle higher condensate flow rates than the piston-valve impulse trap described previously, has a lever-valve assembly that pivots on a fulcrum (fig. 11-57b). Air that may initially enter the trap during system startup can pass through the gap between the closed inlet valve and its seat and then through the control orifice. However, when cool condensate or a mixture of cool condensate and air enters the trap, the lever-valve assembly is forced to pivot in a clockwise direction and the inlet valve is lifted off its seat. Condensate can then pass through the open inlet valve and the outlet orifice. As the flow of condensate continues, cool condensate gradually drains from the line in which the trap is installed, and the temperature ofthe incoming

Figure 11-57b. Lever-valve-type impulse steam trap.
Courtesy Yarway Corporation.

condensate typically increases. Once this temperature increases sufficiently for flashing to begin in the trap, the specific volume and velocity of the fluid passing through the trap will rise and, due to the resulting conversion of potential to kinetic energy, the pressure below the lever will drop. Additionally, the velocity of any flash steam that enters the upper chamber above the lever will be reduced, which results in an increase in the pressure acting on top of the lever. The increased pressure above the lever accompanied by the reduced pressure below the lever forces the lever-valve assembly to pivot in a counterclockwise direction and closes the trap's inlet valve. Because a small control flow continues to pass through the control orifice, the trap can respond quickly to a change in the inlet conditions. As the condensate cools, flashing will be reduced or stop, flow through the control orifice will increase, the pressure in the upper chamber will be reduced, and the trap will reopen. The cycle will then be repeated.

INSULATION

Insulation reduces the rate at which heat is transferred through the walls of piping, valves, and fittings. It may be applied to a piping system to reduce the heat either lost from or added to fluid flowing through the system. It may also be used to reduce the heat gain in an enclosed space, to shield operating personnel from hot pipes, and to reduce condensation on the outer surfaces of cold pipes. The outer temperature of hot pipes, including the insulation, should generally not exceed 125°F to 140°F (52°C to 60°C). Also, the outer temperature of cold piping, including the insulation, should not drop below the ambient dew point.

Materials used for piping insulation include calcium silicate, cellular glass, elastomeric foam, fiberglass, mineral wool, and plastic foam such as polyurethane. The use of asbestos, which was a common insulation material in the past, has been virtually discontinued because of health and environmental concerns. Insulation may be applied as a cement, blanket, sheet, block, or premolded form that matches the shape of the component being covered. In addition, insulation is frequently protected with lagging consisting of sheets of polyester fabric, fiberglass, aluminum, galvanized steel, or stainless steel. A vapor barrier may also be required on a low-temperature system to prevent water vapor in the air from condensing on the surface of the cold piping. Insulation should be removable around take-down joints in a system.

JACKETED PIPE

A jacketed pipe is formed by a single pipe that is surrounded by a second pipe having a larger diameter. Jacketed pipe may be used when the temperature of the fluid passing through the inner pipe must be accurately controlled. For example, when molten sulfur is transported through the inner or core section of jacketed pipe, steam or another heated fluid is typically circulated through the annulus formed between the inner and outer pipes to keep the molten sulfur at the proper temperature. Jacketed pipe is also sometimes used to provide a buffer zone around fluid passing through the core pipe. With this arrangement, fluid that may leak from the core pipe will be retained within the annular space between the two pipes and prevented from entering the environment surrounding the outer or jacket pipe. In addition, an inert fluid can be circulated through the annular space to keep it purged of contaminants and, when the inert buffer fluid is inspected, to enable leaks in the core pipe to be discovered quickly.

Heat Tracing

The temperature of fluid flowing within a piping system is sometimes maintained by heat tracing the outside ofthe pipe. This may be required to prevent fluid in a piping system from cooling and becoming too viscous (i.e., too resistant to flow) or to prevent freezing.

Electric heat tracing is accomplished by wrapping an insulated electric cable around the outside of the piping system being heated in a spiral or helical pattern. Because of resistance, the cable heats up as electric current is passed through it.

Fluid heat tracing is accomplished by attaching a small-diameter pipe or tube through which steam or a hot liquid is circulated to the outside of the piping system being heated. The piping system, together with the attached heat tracing line, is then frequently covered with insulation.

Liners and Coatings

The inside surfaces of piping, valves, and fittings are sometimes lined or coated (e.g., with elastomeric materials, epoxies, etc.) to improve corrosion or erosion resistance, reduce contamination of the fluids passing through the system (e.g., from pipe scale), or improve the pipe's surface finish and reduce losses from friction. Outside surfaces may also be coated to improve corrosion resistance. In addition, outer surfaces of a pipe may be color-coded so that the contents of the piping system can be easily identified. A typical color-coding scheme for shipboard piping systems is shown in table 11-8.

TABLE 11-8
Typical Identification Colors for the Contents of Piping Systems

Main Color	*Medium within System*
Black	Waste media (e.g., waste water, black water, gray water, waste oil, exhaust gas)
Blue	Freshwater
Brown	Fuel oil
Green	Seawater
Gray	Nonflammable gases
Maroon	Dry or wet masses not used for fire extinguishing
Orange	Oils other than fuel oil
Silver	Steam
Red	Fire fighting and fire protection
Violet	Acids or alkalis
White	Air in ventilation systems
Yellow-ocher	Flammable gases

Notes:
1. Pipes should be marked at least once in each space, at each bulkhead or deck penetration, and close to each valve. In addition, markings should be placed every 10 to 15 feet (3 to 5 meters) along the length of a pipe, or at shorter intervals when warranted by bends or the close proximity of other pipes.
2. Hoses should be marked close to fittings.
3. The main color may be
   a. Painted on the pipe in transverse stripes
   b. Painted on the total length of pipe
   c. Applied to the pipe as an adhesive tape or a sign. Tape applied to pipes up to NPS 8 (DN 200) should be wrapped around the entire circumference of the pipe. Tape applied to pipes larger than NPS 8 (DN 200) may be wrapped around half the pipe circumference.
4. Additional secondary colors may be used to distinguish between several systems having the same main color. Additional-color markings should be smaller than main-color markings, and may be
   a. Incorporated into pipe markings, such as flow arrows
   b. Applied as an adhesive tape between two taped stripes of the main color
   c. Painted on the pipe in transverse stripes
   d. Applied as a sign
5. The above is based, in part, on information included in ISO/DIS 14726-1.

INSTALLATION

Piping Installation

When installing or assembling a piping system, all piping sections should be properly supported, for example with anchors, hangers, clamps, or swivels, and aligned to prevent excessive deflection, stress, and vibration. The use of bolts, pry bars, slings, hoists, and the like to force mating ends of a piping system together should ordinarily be avoided. Components in the system, such as valves and pumps, should also be properly supported. In addition, piping should have sufficient flexibility to limit stresses resulting from expansion or contraction to acceptable values. Methods used to incorporate flexibility into a piping system include the installation ofloops, expansionjoints, and sections of hose in a system. The use of spring-loaded or resilient pipe hangers and supports can also increase a piping system's flexibility.

When installing tubing, bends should also be provided to allow for a reduction in length that can occur when the tubing is pressurized. Correct and incorrect methods of installing tubing are shown in figure 11-58. For the same reason, a sufficient excess must be added to the length of an assembled hose to allow for a reduction in length as the hose's diameter expands during pressurization. Additionally, tubing and hose should never be twisted during assembly.

PRESSURE TEST

Mter installation, the leak tightness of a piping system is often tested. To perform a typical hydrostatic test, the system is initially filled with water. The pressure ofthe water is then slowly increased in small increments until it is equal to a specified value (e.g., 1.5 times the system's maximum allowable working pressure). However, the system hydrostatic-test pressure should not result in excessive stresses in any part of the piping system and should not exceed the test pressures of any of the components installed in the system (e.g., valves, pumps, etc.). *Mter* the test pressure is maintained

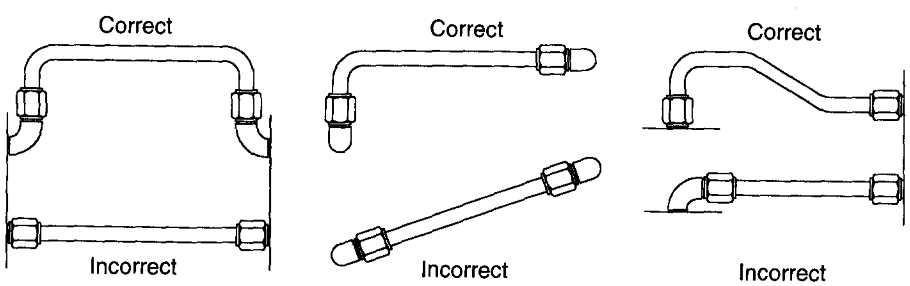


Figure 11-58. Correct versus incorrect tubing installation

for a specified period oftime (e.g., ten minutes), the system is inspected for leaks. The pressure is then relieved.

In cases where it is impracticable to perform a hydrostatic test using water or when water could contaminate the system being tested, compressed air or another nonflammable gas is sometimes used to perform a pneumatic pressure test. (Pneumatic pressure tests are often performed to 1.25 times a system's maximum allowable working pressure,) However, because a compressed gas can store a considerable amount of potential energy that is released if the system being tested leaks, extreme caution should be exercised whenever pneumatic pressure tests are performed.

GROUNDING

Piping systems that contain flammable fluids or are routed through hazardous spaces should be properly grounded to the vessel to control static electricity. A shipboard piping system can be grounded by bolting or welding a portion of the system directly to the vessel's hull or by connecting the piping system to the hull with conductive bonding straps. Bonding straps, when used, should be installed in locations where they will be protected from corrosion and mechanical damage. The resistance across a ground from the piping system to the hull should generally not exceed one megohm.

Gasketed-Flanged Joints

Prior to assembling a flanged joint, mating flanges, together with required bolting, should be thoroughly inspected. When necessary, any pieces of the old gasket or other foreign material should be removed from the sealing surfaces without damaging the flanges. A sealing surface finish of from 125 to 200 /lin. (3.2 to 5.1 /lm) rms is generally satisfactory. Because the friction between a gasket and the mating flange surfaces must be sufficient to prevent the gasket from being extruded or blown out of the joint, very fine flange finishes, less than 32 /lin. (0.8 /lm) rms, should ordinarily be avoided. Spiral or concentric serrations are sometimes machined into a flange's face to increase the flange's ability to retain a gasket. Flange sealing surfaces should be free oflarge nicks, dents, and gouges. It is especially important that radial defects (i.e., scratches and grooves that cut across the sealing surface), which could become leak paths after assembly, are not present. In addition, warpage, which can be caused by overheating, high internal pressures, or high bolt loads (the latter is usually referred to as bowing) should not be excessive. In some cases, flanges may have to be remachined or replaced. When the original flange must be replaced because it was overloaded, a redesigned flange that is thicker or has more bolts may be required.

The material used for a gasket must have sufficient resiliency and be compatible with both the fluids and the flanges it will contact. Some gasket materials, for example, can lead to corrosion ofthe mating flange surfaces.

A gasket must also be suitable for the system pressures and temperatures. In addition to having a maximum temperature limit, many gasket materials have a maximum $P \times T$ factor, which is equal to the product of system pressure times temperature and should not be exceeded.

Materials used to make nonmetallic gaskets include paper, cork, cellulose, vegetable fiber, flexible graphite, rubber, and various synthetic elastomers and plastics, such as polytetrafluoroethylene (PTFE). Compressed asbestos, which was frequently used in the past, has been banned for most applications because of health and environmental concerns. Some nonmetallic gaskets are reinforced with metal wire. Metals, such as copper and soft iron, also are used sdmetimes to make ring gaskets. In addition, some ring gaskets have an outer metal jacket that surrounds a nonmetallic core. A common type of gasket used in many high-pressure steam systems consists of a thin V-shaped metal strip that is wound in a spiral with a nonmetallic filler and is inserted into an outer metal retaining ring. The retaining ring reinforces the spiral-wound strip so that it is not blown out of the joint. It also facilitates the centering of the spiral-wound gasket, which is used only with raised-face flanges, within the flange bolt circle during assembly.

Although a nonmetallic ring gasket should generally be used only once, when a suitable replacement is not available, it is sometimes possible to reuse an old gasket, provided that the gasket was not permanently compressed while in service and was not damaged during disassembly. A copper ring gasket may also be reusable, provided that the gasket is annealed prior to reassembly. The center insert in a spiral-wound metal gasket, however, is typically crushed during assembly and should never be reused. In some cases, the outer retaining ring of a spiral-wound metal gasket can be reused with a new insert.

To reduce the force or bolt load necessary to adequately compress a gasket, it is generally preferable to use as thin a gasket as practicable. However, it may be necessary to use a thicker gasket to compensate for flange-face irregularities, such as when flange faces are warped, bowed, or pitted. Although the force required to properly compress a gasket, which increases with gasket width, can also generally be reduced by using a flat ring gasket that extends only to the inner edges of the flange bolts, a flat full-face gasket with bolt holes is often installed between flat-faced flanges. This arrangement is necessary when the gasket is so flexible that it requires support from the bolts for proper alignment during installation.

A coating is sometimes applied to a gasket prior to installation to improve sealing, reduce fretting (i.e., reduce damage to the gasket due to flange shearing motions), and reduce the potential for sticking when the gasket is later replaced. It is always important to ensure that any coating used is compatible with the gasket and mating flange materials. Typical gasket coating compounds include PTFE, molybdenum disulfide, and silicon.

A nonmetallic gasket is often cut from a sheet of gasket material. The gasket's inside and outside diameters should be cut to the proper size with a sharp blade or gasket cutter on a hard, flat surface. The inside diameter of the gasket should generally be slightly larger (e.g., 0.25 in. or 6 mm) than the diameter of the opening in the flange so that the gasket does not extend into the flow path after assembly. Bolt holes in a flat full-face gasket, which can be made with an inside-bevel punch approximately $\frac{1}{16}$ in. (1.6 mm) larger in diameter than the bolt size, should be added before the gasket's outside diameter is cut. The practice of placing the gasket over the flange and pounding it out of the sheet with a hammer can damage many gasket materials and should be avoided. To facilitate centering, a flat ring gasket often has an outside diameter that extends beyond the raised faces of the mating flanges and fits against the inner edges of the bolts used to secure the joint. This is also typically true of the outside diameter of the outer metal reinforcing ring used with a spiral-wound metal gasket.

Chemical gaskets, such as room-temperature-vulcanizing (RTV) silicons and anaerobic compounds, are sometimes used to seal joints when a precut gasket is not available. RTV silicons cure by absorbing moisture in the air. However, acetic acid or amines may be emitted during the curing process. Anaerobics cure in the absence of oxygen when in contact with an active metal. A chemical gasket fills gaps between mating flange faces; consequently, flange-to-flange contact may occur during assembly.

When an O-ring is used to seal a flanged joint, it may be purchased in the required size. Alternatively, O-rings are sometimes fabricated from a kit consisting of rolls of O-ring material in various diameters. A piece of the required O-ring material having the proper diameter is typically cut in the required length from a roll, and its ends are then glued together using a special adhesive. When installing an O-ring into a groove, it should not be twisted. In addition, the groove should be free of all foreign objects. Installation can often be facilitated by lubricating the O-ring with a compatible oil or grease prior to inserting it into its groove. Special lubricants are often available from O-ring manufacturers for this purpose. Additionally, the O-ring should not be pinched between the mating flanges when they are bolted together. Whenever possible, an O-ring should be used only once and replaced if the joint in which it is installed is disassembled.

When making up a flanged joint, it is important to verify that the fasteners are in good condition and are manufactured from the proper materials. In addition, mating flange faces should be parallel and the proper distance apart, and adjacent bolt holes should line up. Any discrepancies should be corrected by realigning the piping rather than by forcing the flanges together. When using a flat full-face gasket with bolt holes or an O-ring, the gasket or O-ring must be inserted between the mating flanges before the bolts are installed. When using a ring gasket, however, the lower bolts may be slipped through the flange holes before the gasket is installed to prevent the gasket from falling through the joint while it is being lowered into place.

After installing the gasket, two bolts installed in diametrically opposite holes should initially be tightened evenly by hand. Two additional bolts installed in holes that are approximately 90° from the first two bolts should then be tightened evenly until they are as tight as the first two bolts. This process should be repeated until all of the bolts are hand tight. Bolting should then be tightened evenly in small increments following an alternating crossover pattern (fig. 11-59) until the final torque value is achieved. While tightening the joint, it is important to ensure that the flange faces are drawn together squar·ely. The final torque value should be sufficient to adequately preload the bolting and compress the flange sealing surfaces and gasket, when used. (Insufficient joint compression can result in flange face separation and leakage when the piping system is pressurized.) However, joint compression should not result in overstressing of the bolts or flanges. When possible, maximum torque values should be obtained from applicable drawings or manuals. To minimize the potential for damage, it is best to use a calibrated torque wrench to tighten flange bolting. In addition, bolting should generally be lubricated with a suitable thread lubricant or antiseize compound prior to assembly.

Antiseize lubricants, which help to prevent galling and binding, are typically composed of small flakes or particles of solid fillers, such as PTFE, graphite, molydenum disulfide, copper, or nickel, that are suspended in a heavy oil base. Because the lubricant applied to a threaded fastener affects the coefficient of friction between meshing threads and the percentage of an applied torque that actually loads the fastener, the lubricant used should, when possible, be the same as the type on which the

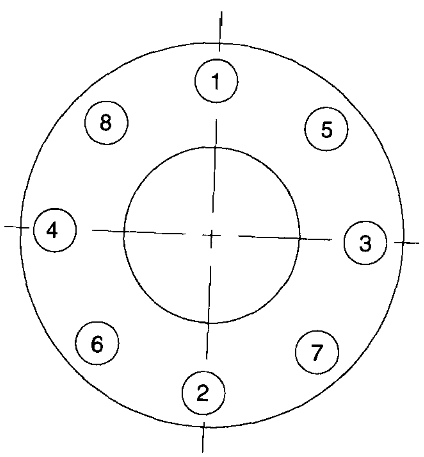


Figure 11-59. Torque pattern for a flanged joint

specified required torque values are based. To compensate for thermal expansion and creep relaxation, bolting should be checked, following the same alternating pattern used previously, after the system has been in operation for twelve to twenty-four hours and retightened if necessary to restore the specified torque.

In some cases, disk springs, which are similar in appearance to conical washers, are installed on the flange bolts between the back face of the flange and the nuts to help maintain the initial bolt load and compensate for thermal expansion and pressure surges within the system.

Prior to disassembling, or breaking, any piping joint, piping should be adequately supported so that the ends of the pipe or components in the system will not drop after the joint has been disconnected. In addition, the system in the area of the joint should always be isolated, depressurized, drained, and, when applicable, allowed to cool prior to disassembly. After shutoff valves on either side of the joint have been closed, they should be wired shut and warning tags should be affixed to their handles to prevent the valves from being accidentally opened. Also, any electrical equipment in the vicinity of the joint should be properly shielded and, when possible, de-energized. When disassembling a flanged joint, it is recommended that bolting on the side of the joint facing away from service personnel (i.e., the bolting on the far side of the flange) be loosened first. By doing this, any residual fluid in the piping that sprays out of the joint will be less likely to injure personnel. After making certain that no pressurized fluid is contained within the piping, remaining flange bolts should be loosened evenly in an alternating crossover pattern (fig. 11-59). This will prevent the flange faces from springing apart unevenly due to piping strain.

Screwed Joints

To increase the effectiveness of the seal created between the mating threads in a screwed joint, the male threads are often coated with a sealing compound (dope) or wrapped with special tape (e.g., PTFE tape) prior to assembly. When using tape, several threads at the open end ofthe pipe or fitting should remain uncovered so that they will engage more easily with the female threads in the mating component during assembly. When assembling a threaded piping and components, it is important not to overtighten the joint, which can result in damage to the threads. Typical values for the thread engagement of assembled screwed joints are shown in table 11-3.

Valve Installation

In addition to being properly supported, when practicable, valves should be installed in locations that are readily accessible and with their stems in an upright position, which reduces the potential for sediment to collect in the bonnet around the stem and stuffing box. In addition, when a valve is installed, there should be sufficient room to enable the valve's handle or handwheel to be moved from a fully closed to a fully open position. When the valve is designed for flow in only one direction, it should be mounted with the proper connection on the supply side of the system. Many types of valves, such as check valves and reducing valves, are marked with an arrow that should point in the direction of flow after installation.

When it is necessary for a pump to take suction from and discharge to multiple locations, such as fuel-oil or ballast tanks, multiple suction and discharge valves are frequently installed in a common manifold. Stop-check valves are often used for the suction valves to prevent fluid from draining back through an open valve when the pump is stopped.

VALVE MAINTENANCE

Valve stems should always be kept clean and free of rust or paint. When grease fittings are provided for the lubrication of a valve's operating mechanism or components, as in a yoke-nut assembly, grease should be replenished periodically. Leakage through a closed valve typically results from the inability of the disk to seal properly against the valve's seat. In some cases, this is caused by a buildup offoreign material, such as scale or dirt, on the sealing surfaces. This material can sometimes be flushed away by opening the valve. The valve can then be reclosed. If leakage persists, however, trying to force the valve to seal by jamming it shut (e.g., by using a wrench on the handwheel) should generally be avoided, as it can lead to increased disk and seat damage and a bent or broken valve stem. (A valve that has been jammed closed will also be very difficult to open, especially if the valve was initially hot when it was closed and it then cools or if the valve was initially cold when it was closed and it is then heated.) The leaking valve should instead be disassembled, inspected, and cleaned or repaired as needed. Piping adjacent to the valve should always be isolated, drained, and, when applicable, allowed to cool prior to valve removal.

Foreign material trapped between a valve disk and seat can SCorethe sealing surfaces as the valve is being opened and closed. In addition, a valve's sealing surfaces can be worn due to erosion and corrosion caused by . the fluid passing through the valve. Consequently, cleaning is not always sufficient to stop leakage, and the repair or replacement of the disk and seat are sometimes required.

Problems can also occur if a valve is jammed open, especially if the valve was initially hot when it was open and it then cools or if the valve was initially cold when it was open and it is then heated. Not only will the valve be difficult to close, but someone mistakenly thinking that the valve is already closed (because the stem is tight) may try to open it. Since the valve is actually already open, the stem can be bent or broken. To prevent this,

whenever a linear valve is fully opened, the stem should be rotated one-quarter to one-half turn in the closed direction. (This does not apply to valves that are designed to be backseated when open.)

Globe and Angle Valves

To replace the disk in a globe or angle valve with a composition disk, the bonnet assembly should be removed from the valve body. The disk holder should then be slipped off the end ofthe valve stem, and the disk retaining nut should be removed from the underside ofthe disk holder. Mter the nut has been removed, the old disk can be removed from the disk holder and replaced with a new disk. The disk holder and valve can then be reassembled. When possible, a valve's bonnet gasket should be replaced prior to reassembly. The new bonnet gasket should always be the proper type and size, and made from the proper material. When a flanged bonnet is reassembled, a suitable lubricant or antiseize compound should be applied to the threads on the bolts or studs, and bolting should be tightened evenly in an alternating pattern (similar to the pattern used on a flanged pipe joint in fig. 11-59) until the specified torque is achieved. The bonnet's bolting should be checked and retightened, if necessary, after the valve has been put back into service and again after two days of operation.

The amount of contact between the disk and seat in a globe or angle valve not fitted with a composition disk can be determined by spotting-in the seat and the disk. To spot-in the seat, the valve's disk, together with the stem and bonnet, should be removed from the valve body and a thin coat of nondrying blue dye or equivalent should be applied evenly over the face of the disk. Then, using the bonnet for guidance, but with the packing removed, the disk should be placed squarely against the seat and, while a light pressure is being applied to the disk, rotated against the seat for a few degrees. The dye transferred to a seat in good condition should form an unbroken ring located slightly above the opening in the base of the seat and with a width equal to approximately one-third of the seat's width. In addition, an unbroken ring of dye having a uniform width will be removed from the sealing surface of the disk that is in satisfactory condition.

Small irregularities that prevent a globe or angle valve's seat and a one-piece disk (i.e., not a composition disk) from fitting together properly can sometimes be removed by lapping or grinding-in the valve seat and disk. Prior to performing this procedure, the disk must be secured on the valve stem so that it will rotate as the stem is turned. This can often be done by inserting a washer between the disk and the bottom of the valve stem that is thick enough to take up the clearance between these two components. To install this washer, the disk nut or stem ring should be removed from the disk. The disk can then be removed from the end of the valve stem, and the washer can be inserted into the recess in the top ofthe disk. The end ofthe valve stem can then be reinserted into the disk's recess on top of the washer, and the disk nut can be replaced. Alternatively, some valves have a hole through which a split pin can be inserted to lock the disk in place on the stem.

After securing the disk to the stem, a small amount of lapping compound should be applied evenly to the disk's sealing surface. Lapping compound typically consists of an abrasive grit, such as aluminum oxide or silicon carbide, that is mixed with a heavy oil, grease, or, in some cases, water. The grit size and concentration used varies depending on the grade of the compound (i.e., as the coarseness oflapping compound increases, the size of the grit used is increased but the number of pieces of grit contained in a given volume ofthe compound is reduced). When deep to moderate cuts and scratches are found on the valve's seat or disk, a coarse grade of lapping compound may be used at the beginning of the lapping operation. When a valve's sealing surfaces are not too badly scored, however, it may be suitable to begin lapping with a medium grade of compound. The disk, stem, and bonnet should now be replaced on the valve's body. Packing should be removed from the valve's stuffing box. In addition, if the bonnet is flanged, the nuts that secure it should not be installed at this time. A threaded bonnet or bonnet union ring should be screwed into or onto the valve body two to three turns, but should not be tightened. Then, with the disk pressed against the seat, the stem should be rotated back and forth in 45° to 90° increments. Periodically, the angular relationship between the disk and the seat should be changed (i.e., the disk should be backed away from the seat, turned slightly, and replaced against the seat) so that the disk will gradually be moved through several complete rotations. In addition, approximately once per minute, the seat and disk should be cleaned, and the lapping compound should be replenished. As the irregularities get smaller, progressively finer grades of lapping compound should be used, ending with an extremely fine grade of compound. The final lapping operation may be performed with clean oil. When the condition of the disk and seat are considered to be acceptable, the spotting-in procedure should be repeated. Excessive lapping or the application of excessive pressure to the disk can result in a grooved or curved disk and should be avoided.

Irregularities in a globe or angle valve's seat slightly larger than the ones that can be removed by lapping-in with the valve's disk can sometimes be removed by lapping the valve with a special lapping tool that is the same size and shape as the valve's disk. The procedure used is identical to that previously described for lapping-in using the valve disk. It is important that excessive force not be applied to the lapping tool, which should be held straight and not tilted to the side. After using the lapping tool, the valve seat should be lapped with the valve disk. The spotting-in procedure should then be repeated.

When a valve disk cannot be repaired by being lapped, it should be remachined on a lathe, ground, or replaced. In addition, a badly damaged seat that screws into the valve body should generally be replaced. (The screw-in seats in larger valves may be locked in place with a small weld. This weld must be removed before the seat can be unscrewed from the valve body.)

When the integral seat in a globe valve body is too badly scored to be renewed by lapping, it can sometimes be repaired by being refaced in a lathe with a power grinder or with a hand-operated valve-reseating machine. When using a valve-reseating machine, it should be inserted into the valve body and secured with a chuck that grips the valve body around the opening provided for the bonnet. (If a valve body's bonnet opening is not bored true with respect to the seat, a valve-reseating machine should not be used to reface the seat.) With the proper cutter attached to the end of the machine's spindle, the hand crank should be rotated slowly. As very thin shavings are removed from the valve seat, the cutter should slowly be fed deeper into the valve body. Periodically, the tool should be removed and the seat should be inspected. When the imperfections in the seat have been removed, it may be necessary to use a flat cutter to dress the top of the seat and reduce the seat's width to the proper value. Refacing should be followed by lapping with the valve disk and spotting-in the valve.

Extremely worn sealing surfaces can sometimes be renewed by being faced with metal alloys, such as those containing cobalt, chromium, and nickel, and then ground. These alloys, which can often be applied utilizing various welding or spraying processes, may also be used to face sealing surfaces on new components to increase resistance to corrosion or erosion, especially when the surfaces will be exposed to high-temperature, high-pressure, or abrasive fluids.

Gate Valves

Contact between a gate valve's disk and seats can be checked by disassembling the valve, applying nondrying blue dye to the sealing surfaces of the disk, and then setting the disk in place between the seats. Minor scratches in the disk or seats can sometimes be removed with emery cloth. Because most of the wear on a gate valve's disk usually occurs on its downstream side, if leakage persists, it may be possible to make a temporary repair by reversing the disk. If sealing surfaces cannot be repaired, a gate valve's seat rings, when they are removable, and disk should be replaced at the same time.

Stuffing Boxes

Leakage through the hole in a valve's bonnet provided for the valve stem is often controlled by packing that is inserted into a stuffing box. In most cases, no leakage should be visible from a properly adjusted stuffing box. Once packing leakage starts, however, it will gradually get worse if no action is taken to stop it. In addition, as packing leakage is allowed to persist, the force of the fluid blowing out of the stuffing box can result in damage to the packing, valve stem, and stuffing box. In the case of a high-temperature or high-pressure fluid, stuffing-box leakage can also a create a hazard for operating personnel.

When the leakage through a stuffing box becomes excessive, it is frequently possible to stop or reduce it by increasing the compression of the packing. Depending on the construction of the valve, this may be accomplished by tightening a packing nut or the nuts that hold a gland or gland flange in place. When packing is compressed by a gland, it is important that the nuts on both sides of the gland be tightened equally, often by being turned no more than one flat (one-sixth of a turn) at a time (especially for minor adjustments), so that the gland does not become cocked with respect to the valve stem.

When the stuffing box leakage can no longer be controlled by increasing packing compression, the stuffing box should be repacked. Certain types of valves can be repacked without depressurizing the supply line, provided that the valve is closed and does not leak. However, if the valve does leak while closed, fluid may blow through the stuffing box as the packing is being removed. Not only can this result in injury to the service personnel, but it can make installation of the new packing rings extremely difficult. Consequently, whenever possible, a valve's supply and outlet lines should be depressurized and drained prior to removing packing from the stuffing box.

To replace packing, a valve's gland nuts and gland or packing nut should be removed from the bonnet and the old packing should be removed from the stuffing box. A wire packing puller can often be used to facilitate packing removal. It is important that all of the old packing be removed. After this has been completed, the stuffing box and valve stem should be cleaned and inspected. Minor defects on the stem should be removed with emery cloth. If the stem is badly scored or bent, however, it should be repaired or replaced. A scored or bent valve stem can frequently result in excessive stuffing box leakage that will persist even after the packing has been replaced. When a throat bushing is installed at the base of the stuffing box, it too should be inspected.

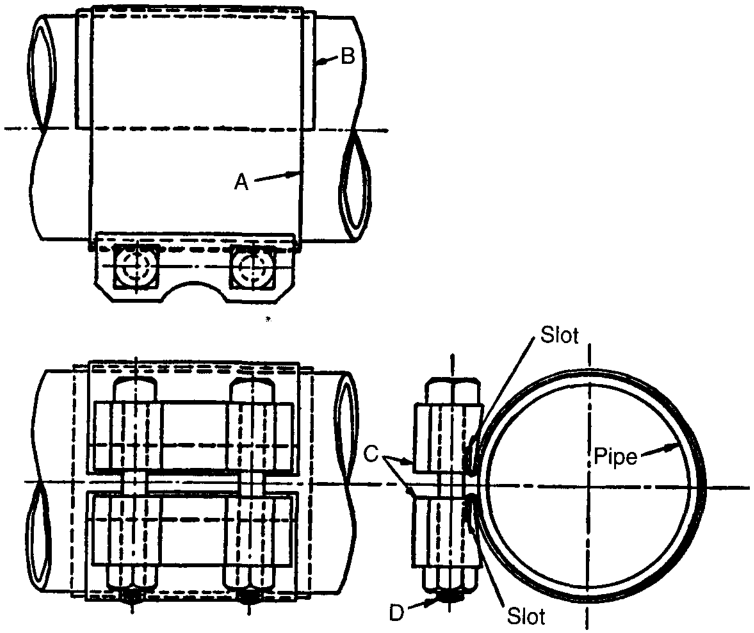
Some small valves may be repacked by coiling string packing around the valve stem. The ends of the string should be beveled to form smooth surfaces in the bottom of the stuffing box and against the gland or packing nut. When it is secured with a packing nut, string packing should be wound in the direction that the nut will be turned while being tightened. This will prevent the packing from being folded back as the packing nut is screwed onto the stuffing box.

Alternatively, when split packing rings are used, they should be of the proper size (based on the clearance between the valve stem and the inside of the stuffing box) and material, and can be either precut and preformed or cut from a coil that is wrapped around a mandrel having an outside diameter equal to that of the valve stem. Individual packing rings are often cut with beveled (45°) joints or scarfs that overlap after installation. The material used for packing varies with the application. Although asbestos was often used in the past, due to its widespread prohibition, many packings now used are composed of graphite and polytetrafluoroethylene (PTFE) compounds. The packing may also be reinforced with metal foil or wire. Rings should be installed one at a time and should be pushed down into the stuffing box. It is important that each packing ring be seated squarely. The gland can often be used to push or tamp each packing ring down into the stuffing box. The splits of successive rings should be placed 180° apart when two rings of packing are used, 120° apart when three rings of packing are used, and 90° apart when four or more rings of packing are used. If a lantern ring is used, it should be installed in the proper location: adjacent to the injection-port opening inside the stuffing box. After the proper number of rings of packing have been installed, which can be determined by measuring the depth of the stuffing box, the gland or packing nut should be replaced and tightened until resistance is felt. If necessary, additional packing adjustments to reduce stuffing box leakage should be made after the valve has been put back into service.

During packing installation and adjustment, packing should not be pinched between the gland and the stuffing box. In addition, if, after adjustment, the bottom of the gland does not extend into the stuffing box, one ring of packing should be removed. This is especially important when the valve is in a high-pressure system to prevent packing from being extruded through a gap between the bottom of the gland and the top of the stuffing box. Conversely, if, after adjusting the packing on a repacked valve that has just been placed back into service, the gland is fully seated in the stuffing box (i.e., no further compression with the gland is possible), it is sometimes prudent to secure the valve and install an additional ring of packing into the stuffing box. Excessive tightening of packing can bind a valve's stem and should be avoided.

Leaks

Leaks in a piping system should be corrected as soon as possible after they are discovered. This is especially important when pressurized fluid is passing across a sealing surface, such as the face of a flange. Because the fluid will erode the surfaces over which it passes, the damage will become progressively more severe as the leak continues. Small leaks in piping can sometimes be repaired temporarily with a "soft patch," which can be made from sheet metal and sheet rubber and then clamped around the pipe over the hole (fig. 11-60). In addition, a hole or rupture in a pipe can sometimes be patched using a moldable epoxy compound or by welding.



A Sheet metal (e.g., 20- or 22-gauge) cut to suit piping
B Sheet rubber (e.g., 1/8" thick) cut to suit piping
C Steel clamps-one set adapted to piping size (slotted for item A).
D Clamping bolts

Figure 11-60. Soft patch for emergency repair of piping

PIPING DESIGN

Classification

Regulatory bodies often classify piping based on service, pressure, and temperature. Following is a typical classification scheme for Group I piping as shown in the American Bureau of Shipping (ABS) *Rules for Building and Classing Steel Vessels.*

Service	*Working Pressure*	*or*	*Working Temperature*
Vapor and gas	over 150 psig (1,030 kPa)		over 650°F (343°C)
Water	over 225 psig (1,550 kPa)		over 350°F (177°C)
Lubricating oil	over 225 psig (1,550 kPa)		over 400°F (204°C)
Fuel oil	over 150 psig (1,030 kPa)		over 150°F (66°C)
Hydraulic fluid	over 225 psig (1,550 kPa)		over 400°F (204°C)

Group II piping, which generally has to meet requirements that are less stringent than Group I requirements, would include piping with working pressures and temperatures below those specified above. The information given above is for illustrative purposes only. Applicable regulations and specifications should always be consulted for requirements and limitations that pertain to actual systems. (The designations of Classes I and II in the *Code of Federal Regulations* are similar to Groups I and II in the ABS *Rules for Building and Classing Steel Vessels.)*

Materials

In addition to having adequate strength and ductility over the range of operating temperatures, materials used in the construction of a piping system must be resistant to corrosion, including pitting and crevice corrosion, from the fluids that will flow through the system. Materials used for components exposed to flow should also have adequate erosion resistance. Furthermore, whenever possible, metals that are used should be compatible galvanically. When dissimilar materials are coupled, it is generally preferable that the more noble (cathodic) material be used for the smaller or more critical component involved.

Common materials used in shipboard piping systems are shown in table 11-9. Chromium and molybdenum are often added to steel used for piping in high-temperature steam systems to increase the piping's ability to withstand high temperatures. In addition, polyvinyl chloride (PVC), a thermoplastic, and glass-reinforced thermosetting resin pipe (RTRP or GRP) are used in some nonvital seawater and freshwater systems. Some typical maximum temperature limits for the use of certain materials are included in table 11-10. This information is general in nature; applicable regulations should be always be consulted for requirements and limitations in specific applications.

Sizing Piping

The pipe or tubing size selected for a specific application is often based on limiting fluid velocities to a maximum value. Maximum allowable velocities are frequently determined based on the desire to limit pressure losses, turbulence, noise, and erosion within a pipe or tube and, in the case of liquids, to prevent flashing. Consequently, velocity limits are typically reduced as liquid temperatures and viscosities increase and as absolute pressures are reduced. Typical maximum velocity limits for fluids in piping systems, which range from 1 to 25 ft/s (0.3 to 7.6 m/s) for liquids and 200 to 330 ft/s (60 to 100 m/s) for steam, are shown in table 11-11. In addition, in a seawater piping system, a minimum fluid velocity of 3 ft/s (0.9 m/s) is often recommended to reduce fouling.

When sizing potable water, sanitary, waste water, and soil drainage systems, all of the various fixtures installed in a given system will not be

TABLE 11-9
Representative Pipe Materials Used in Selected Shipboard Fluid Systems

Systems	Typical Pipe Materials
Steam, Steam Drains, Boiler Blowdown, Safety Valve Escape	Seamless 1-1/4Cr-1/2Mo-Si Alloy Steel (Up to 1,100°F)
	Seamless Grade B Carbon Steel (Up to 775°F)
	Electric-Resistance-Welded Grade B Carbon Steel (Up to 650°F and 350 psig)
Feed, Condensate	Seamless Grade B Carbon Steel
	Electric-Resistance-Welded Grade B Carbon Steel (Up to 350 psig)
Freshwater for Auxiliary Machinery and Engine Cooling	Seamless or Electric-Resistance-Welded Grade B Carbon Steel
	Filament-Wound or Centrifugally-Cast Grade 1 GRP
Cold and Hot Potable Water, Air-Conditioning Chilled Water, Hot-Water Heating, Freshwater Sanitary	Seamless Annealed Type K or L Copper
	Seamless Drawn Type K or L Copper (Up to 225 psig)
	Filament-Wound or Centrifugally-Cast Grade 1 GRP
Seawater Circulating, Seawater Cooling, Distilling Plant Feed and Brine	Seamless or Welded 90Cu-10Ni Alloy
	Filament-Wound or Centrifugally-Cast Grade 1 GRP
Seawater or Freshwater Tank Washing	Seamless or Electric-Resistance-Welded Grade B Carbon Steel[A]
	Filament-Wound or Centrifugally-Cast Grade 1 GRP (Must be independent of firemain)
Wet Firemain	Seamless or Welded 90Cu-10Ni Alloy
Dry Firemain, Foam, Sprinkling, Deckwash	Seamless or Electric-Resistance-Welded Grade B Carbon Steel[A]
Clean Ballast, Pump Priming	Seamless or Electric-Resistance-Welded Grade B Carbon Steel[A]
	Filament-Wound or Centrifugally-Cast Grade 1 GRP
Bilge	Seamless or Electric-Resistance-Welded Grade B Carbon Steel[A]
	Filament-Wound or Centrifugally-Cast Grade 1 GRP (Suction only)
Diesel Oil, Lubricating Oil	Seamless or Electric-Resistance-Welded Grade B Carbon Steel
Fuel-Oil Filling and Transfer, Fuel-Oil Service Suction	Seamless or Electric-Resistance-Welded Grade B Carbon Steel
	Filament-Wound or Centrifugally-Cast Grade 1 GRP (Within tanks only)
Fuel-Oil Service Discharge	Seamless or Electric-Resistance-Welded Grade B Carbon Steel (SCH 80 minimum)[B]
Cargo Oil, Crude-Oil Washing	Seamless or Electric-Resistance-Welded Carbon Steel[C]
	Filament-Wound or Centrifugally-Cast Grade 1 GRP (Not over 225 psig and within tanks only)
Steering-Gear Fill and Drain	Seamless Annealed Type K or L Copper
	Seamless Drawn Type K or L Copper (Up to 225 psig)
Hydraulic Systems	Seamless or Electric-Resistance-Welded Carbon Steel
	Seamless or Welded Austenitic Stainless Steel
Compressed Air	Seamless Grade B Carbon Steel
	Electric-Resistance-Welded Grade B Carbon Steel (Up to 350 psig)
	Seamless Type K Copper (Up to 150 psig)
Refrigeration	Seamless Annealed Type K or L Copper
	Seamless Drawn Type K or L Copper (Up to 150 psig)
CO_2, Smoke Detection	Seamless or Electric-Resistance-Welded Grade B Carbon Steel
Sounding Tubes, Vents[D], and Overflows for Freshwater, Seawater, Oil, and Plumbing	Seamless or Electric-Resistance-Welded Grade B Carbon Steel
	Filament-Wound or Centrifugally-Cast Grade 1 GRP (Should not be used for weather-deck vents; should not pass through tanks containing hazardous materials or flammable or combustible liquids.)
	PVC (Plumbing vents)
Waste (Gray Water), Soil (Black Water), and Interior Deck Drains	Seamless or Electric-Resistance-Welded Grade B Carbon Steel
	Filament-Wound or Centrifugally-Cast Grade 1 GRP (Not to be used outboard of overboard skin valves)
	PVC (Plumbing drains, not to be used outboard of overboard skin valves.)
Weather-Deck Drains	Seamless or Electric-Resistance-Welded Grade B Carbon Steel
Liquified Natural Gas (LNG)	Seamless or Welded Austenitic (type 304L or 316L) Stainless Steel

A. Steel pipe used in seawater applications often must have a minimum wall thickness equal to that of extra strong (XS) pipe or be galvanized.
B. Short lengths of steel, annealed copper nickel, nickel copper, or copper pipe and tubing is often used to provide flexible connections at burners. The pipe or tubing wall thickness should satisfy maximum allowable stress limitations and must frequently be no less than 0.35 inch (0.9 mm).
C. The inside surfaces of steel cargo piping is sometimes epoxy coated.
D. Tank vents extending above the freeboard or superstructure deck must usually have a minimum wall thickness equivalent to schedule 40 pipe.

Notes:
1. The above is based, in part, on ASTM F 1155-98 and is provided for general information only. Applicable specifications and regulations should always be reviewed to determine requirements for specific installations.
2. Cr = chromium, Mo = molybdenum, Si = silicon, Cu = copper, Ni = nickel. Number preceding the element abbreviation represents the percentage of the element in the material.
3. Representative specifications for many of the materials shown above are listed in table 11-10.
4. GRP = glass-fiber-reinforced plastic (also referred to as FGP for fiberglass pipe and RTRP for glass-fiber-reinforced thermosetting-resin pipe). The use of GRP is typically subject to limitations on application, pressure, temperature, location, etc., based on various factors such as requirements established by regulatory bodies, the method used to manufacture the pipe, and the type of resin used. Typical resins include glass-fiber-reinforced epoxy resin (Grade 1), glass-fiber reinforced polyester resin (Grade 2), and glass-fiber-reinforced phenolic resin (Grade 3). Additionally, filament-wound GRP is classified as Type I and centrifugally-cast GRP is Type II. (Classifications given are in accordance with ASTM D 2310-97.) GRP pipe is commercially available both with and without a liner. GRP pipe may also be manufactured with conductive materials for use in applications where it is necessary to reduce the potential for the accumulation of static-electric charges. Representative limits on maximum allowable temperature and pressure include values that can range from 140°F to 240°F and 150 psig to 225 psig, respectively. The strength of GRP and the allowable pressure typically drop as the operating temperature increases.
5. PVC = polyvinyl chloride thermoplastic. The use of PVC is often limited to nonvital (not essential to the safety of the vessel, its crew, or passengers) freshwater and seawater applications (including plumbing vents and drains not exposed to treatment chemicals that are incompatible with the pipe or flammable vapors) with temperatures and pressures not exceeding 140°F and 150 psig, respectively.

used simultaneously. Consequently, the design flow rate through each branch of a system is based not on the total maximum flow rate, but rather on the total number of supply or drainage fixture units in the branch, which is determined by summing the individual fixture unit values assigned to each fixture installed in the branch (sinks, shower heads, and the like). Typical fixture unit values and supply flow rates corresponding to total fixture unit values are shown in tables 11-12 and 11-13, respectively. In the case of drainage systems, which are generally not filled with fluid, the downward slope, or pitch, ofthe piping also affects the piping size. In general, the number of fixture units that can be used with a given line size in a drain system increases with the downward pitch ofthe line (see table 11-14).

Strength

The diameter and wall thickness of piping selected for a specific application must be sufficient to limit piping stress resulting from internal and external pressures and applied loads to an acceptable value. Minimum piping wall thickness requirements are often specified by regulatory agencies. A typical equation used to calculate the minimum piping wall thickness required based on hoop or tangential stress resulting from the hydrostatic pressure of fluid contained within a straight length of circular pipe follows.

$$t_{\min} = \frac{p_{\max} d_o}{2(k_3 S + p_{\max} y)} + A \tag{11.6}$$

where

$t_{\min}$ = minimum wall thickness of pipe, in. (mm)
$p_{\max}$ = maximum allowable system working pressure, psig (kPa)
d_o = actual outside diameter of pipe, in. (mm)
k_3 = 1 when using the USCS units of measurement shown (1,000 for the metric units shown)
S = maximum allowable stress in material due to internal pressure (this value is typically reduced as the working temperature increases), psi (N/mm^2)
y = factor based on the piping material and operating temperature; typical values for ferritic steels vary from 0.4 for temperatures up to 900°F (482°C) to 0.7 for temperatures above 1,000°F (538°C)
A = allowance added for threading, grooving, or mechanical strength

Values for S used in equation 11.6 should include allowances for joint efficiency, in the case of welded pipe, or casting quality when cast pipe is used. Typical values of allowable stress S and temperature factor y for representative piping materials are shown in table 11-10.

TABLE 11-10
Representative Values of Allowable Stress for Selected Pipe Materials

	Material	Representative ASTM[G] Material Specification	Minimum Tensile Strength (psi)
(1)	Class 35 Gray Cast Iron[A]	A 278, Class 35	35,000
(2)	Ductile Iron[B]	A 395	
(3)	Furnace-Butt-Welded Carbon Steel[C]	A 53, Type F	45,000
(4)	Seamless Grade A Carbon Steel	A 53, Grade A Type S	48,000
(5)	Seamless Grade A Carbon (C-Si) Steel	A 106, Grade A	48,000
(6)	Electric-Resistance-Welded Grade A Carbon Steel[D]	A 53, Grade A, Type E	48,000
(7)	Seamless Grade B Carbon (C-Mn) Steel	A 53, Grade B Type S	60,000
(8)	Seamless Grade B Carbon (C-Si) Steel	A 106, Grade B	60,000
(9)	Electric-Resistance-Welded Grade B Carbon (C-Mn) Steel[D]	A 53, Grade B, Type E	60,000
(10)	Seamless Alloy (C-1/2Mo) Steel	A 335, Grade P1	55,000
(11)	Seamless Alloy (1/2Cr-1/2Mo) Steel	A 335, Grade P2	55,000
(12)	Seamless Alloy (1-1/4Cr-1/2Mo-Si) Steel	A 335, Grade P11	60,000
(13)	Seamless Alloy (1Cr-1/2Mo) Steel	A 335, Grade P12	60,000
(14)	Seamless Alloy (2-1/4Cr-1Mo) Steel	A 335, Grade P22	60,000
(15)	Seamless Austenitic (18Cr-8Ni, Type 304) Stainless Steel[E]	A 312, Type TP304	75,000
(16)	Seamless Austenitic Stainless (18Cr-8Ni, Type 304L) Steel	A 312, Type TP304L	70,000
(17)	Seamless Martensitic Stainless (13Cr, Type 410) Steel	A 268, Type TP410	60,000
(18)	Seamless Austenitic Stainless (16Cr-12Ni-2Mo, Type 316H) Steel	A 312, Type TP316H	75,000
(19)	Seamless Austenitic Stainless (16Cr-12Ni-2Mo, Type 316L) Steel	A 312, Type TP316L	70,000
(20)	Seamless Annealed Copper[F]	B 88, Type K	30,000
(21)	Seamless Drawn Copper[F]	B 88, Type K	36,000
(22)	Seamless Annealed 90Cu-10Ni Alloy	B 466, Alloy C70600	38,000
(23)	Seamless Annealed Grade 3 Titanium	B 337, Grade 3	65,000

A. Some grades of cast iron are only suitable for use at temperatures up to 400°F (204°C). In addition, the use of cast iron is often limited to fittings and valves with pressures not exceeding 250 psig (1,724 kPa). Also, the use of cast iron with flammable or combustible fluids when near either an open flame or objects at temperatures above 500°F (260°C) or in systems that convey lethal fluids is usually prohibited. Before using cast iron, consideration should be given to its low ductility, which can result in sudden failure if shock loading occurs.

B. The use of ductile iron is sometimes limited to fittings and valves with operating pressures not exceeding 350 psig (2,413 kPa) and temperatures not exceeding 450°F (204°C).

C. The use of furnace-welded steel is often limited to Class II or Group II systems with temperatures not exceeding 450°F (232°C). In addition, its use with combustible or flammable fluids within machinery spaces is frequently prohibited.

D. The use of electric-resistance-welded steel is often limited to pressures not exceeding 350 psig (2413 kPa) and temperatures not exceeding 650°F (344°C).

E. Allowable stress values at temperatures above 1,000°F (538°C) apply only when carbon content is 0.04% or higher.

F. The use of copper in steam or water service is often limited to pressures not exceeding 250 psig (1724 kPa) and temperatures not exceeding 406°F (208°C). In dead-end instrument service, however, copper tubing can frequently be used with pressures up to 1,000 psig (6,895 kPa). In addition, copper installed in Class I or Group I systems must typically be annealed. Also, except for short flexible connections at burners, the use of copper with flammable fluids or hot oil is frequently prohibited.

G. ASTM = American Society for Testing and Materials

Notes:

1. C = carbon; Cr = chromium; Cu = copper; Mn = manganese; Mo = molybdenum; Ni = nickel; Si = silicon. Number preceding the element abbreviation represents the nominal percentage of the element in the metal.
2. Interpolation may be used to determine allowable stress values between the temperatures shown.

Continued on next page

Although the value used for A shown in equation 11.6 is often the depth of a groove or threads included on the end of a pipe, in some cases an allowance may be added even for pipe with plain ends. For example, A is usually set equal to 0.065 in. (1.65 mm) for plain-ended steel pipe with an outside diameter of 4.5 in. (115 mm) or less. In addition, when applicable, a corrosion and erosion allowance should be added to A. A nominal thickness value obtained from a piping table should be reduced to account for mill tolerances before being compared to the minimum required thickness calculated using equation 11.6.

EXAMPLE 11-4: Is a plain-end (i.e., not threaded or grooved) NPS 6 schedule 40 pipe suitable, based on hydrostatic stress, for use in a system with a

TABLE 11-10—*Continued*

	Maximum Allowable Stress in Tension (psi) for Metals at Temperatures Not Exceeding the Values Shown													
	-20° TO 100°F (-29° TO 38°C)	200°F (93°C)	400°F (204°C)	450°F (232°C)	600°F (316°C)	650°F (343°C)	700°F (371°C)	800°F (427°C)	850°F (455°C)	900°F (482°C)	1,000°F (538°C)	1,100°F (593°C)	1,350°F (732°C)	1,500°F (815°C)
(1)	3,500	3,500	3,500	3,500										
(2)	9,600	9,600	9,600	9,600	9,600	9,600								
(3)	6,800	6,800	6,800	6,800	6,800	6,800	6,500							
(4)	12,000	12,000	12,000	12,000	12,000	12,000	11,700	9,000						
(5)	12,000	12,000	12,000	12,000	12,000	12,000	11,700	9,000						
(6)	10,200	10,200	10,200	10,200	10,200	10,200	9,900	7,700						
(7)	15,000	15,000	15,000	15,000	15,000	15,000	14,400	10,800						
(8)	15,000	15,000	15,000	15,000	15,000	15,000	14,400	10,800						
(9)	12,800	12,800	12,800	12,800	12,800	12,800	12,200	9,200						
(10)	13,800	13,800	13,800	13,800	13,800	13,800	13,800	13,500	13,100					
(11)	13,800	13,800	13,800	13,800	13,800	13,800	13,800	13,500	13,100	12,800	5,900			
(12)	15,000	15,000	15,000	15,000	15,000	15,000	15,000	14,400	14,000	13,100	6,300	2,800		
(13)	15,000	15,000	15,000	15,000	15,000	15,000	15,000	14,750	14,200	13,100	6,600	2,800		
(14)	15,000	15,000	15,000	15,000	15,000	15,000	15,000	15,000	14,400	13,100	7,800	4,200		
(15)	18,800	15,700	13,000	12,600	11,400	11,300	11,100	10,600	10,400	10,200	9,800	8,900		
(16)	15,700	13,400	11,000	10,650	9,700	9,500	9,400	9,100						
(17)	15,000	14,300	13,300	13,100	12,400	12,300	12,100							
(18)	18,800	16,200	13,400	12,950	11,800	11,600	11,300	11,000	10,900	10,800	10,600	10,300	3,100	1,300
(19)	15,700	13,300	10,800	10,400	9,400	9,200	9,000	8,600	8,400				3,100	
(20)	6,000	4,800	3,000											
(21)	9,000	9,000	8,200											
(22)	8,700	8,300	7,600	7,500	6,000									
(23)	16,300	14,300	9,300	8,300	6,000									

Notes—*continued:*

3. The typical value of temperature factor y used in equation 11.6 for cast iron, ductile iron, and nonferrous materials (e.g., copper, copper-nickel, titanium) is 0.4. Values of y for ferritic and austenitic steels are shown below.

Material Type	Temperature (Up to 900°F) (Up to 482°C)	950°F (510°C)	1,000°F (538°C)	1,100°F (593°C)	1,200°F (649°C)	1,250°F and Above (677°C and Above)
Ferritic Steel (includes the carbon steels, alloy steels, and martensitic stainless steel shown above)	0.4	0.5	0.7	0.7	0.7	0.7
Austenitic Steel (includes the austenitic stainless steels shown above)	0.4	0.4	0.4	0.5	0.7	0.7

Intermediate values of y may be determined by interpolation.

4. Unless otherwise noted above, materials should generally not be used above temperatures for which stress values are given. In addition, prolonged exposure of carbon steels to temperatures above 775°F (413°C), carbon-molybdenum steels to temperatures above 875°F (468°C), or chromium-molybdenum steels with less than 0.6% chromium to temperatures above 975°F (524°C) can result in the conversion of the carbide phase of the material to graphite.
5. Allowable stress values and temperature limits given above are based, in part, on information included in ASME B31.1-1995 and the ABS Rules and are representative only. Actual values can vary considerably based on chemistry and manufacturing processes, such as the method of heat treatment. Applicable regulations and specifications should always be consulted for requirements that pertain to actual materials and installations.
6. For Class I or Group I systems or when temperatures will be below 0°F (-18°C), the maximum stress used in equation 11.6 must often be reduced to 80% of the piping material's tabulated allowable stress value to account for dynamic loads resulting from vessel motion.

maximum allowable working pressure of 600 psig at a temperature of 280°F? The pipe material is seamless grade B carbon steel. Assume that erosion and corrosion will be negligible.

Solution: From table 11-1, the outside diameter (d_o) of NPS 6 pipe = 6.625 in. In addition, from table 11-10, the maximum allowable stress, S, for either material 7 or 8 at a temperature of 280°F is 15,000 psi and the value for temperature factor y is 0.4. Using these values and equation 11.6

$$t_{\min} = \frac{600 \text{ psig}(6.625 \text{ in.})}{2\left[1(15{,}000 \text{ psi}) + 600 \text{ psig}(0.4)\right]} = 0.130 \text{ in.}$$

TABLE 11-11
Maximum Recommended Piping System Velocities

Service	*Nominal Velocity Limit (ft/s)*	*Maximum Velocity Limit (ft/s)*
Condensate pump suction	$\sqrt{d_i}$	3
Condensate pump discharge	$3\sqrt{d_i}$	8
Condensate drains	$0.3\sqrt{d_i}$	1
Hot water suction	$\sqrt{d_i}$	3
Hot water discharge	$3\sqrt{d_i}$	8
Feedwater suction	$1.3\sqrt{d_i}$	4
Feedwater discharge	$4\sqrt{d_i}$	10
Cold freshwater suction	$3\sqrt{d_i}$	15
Cold freshwater discharge	$5\sqrt{d_i}$	15
Lube-oil service pump suction	$\sqrt{d_i}$	4
Lube-oil discharge	$2\sqrt{d_i}$	6
Heavy fuel-oil service suction	$\sqrt{d_i}$	4
Heavy fuel-oil service discharge	$1.5\sqrt{d_i}$	6
Heavy fuel-oil transfer suction	$\sqrt{d_i}$	6
Heavy fuel-oil transfer discharge	$2\sqrt{d_i}$	15
Distillate fuel-oil suction	$2\sqrt{d_i}$	7
Distillate fuel-oil discharge	$5\sqrt{d_i}$	12
Distillate fuel-oil loading or unloading		25
Hydraulic-oil suction	$1.5\sqrt{d_i}$	8
Hydraulic-oil discharge	$8\sqrt{d_i}$	20
Seawater suction	$3\sqrt{d_i}$	12*
Seawater discharge	$5\sqrt{d_i}$	12*
Steam, high-pressure	$50\sqrt{d_i}$	200
Steam, exhaust, 215 psig	$75\sqrt{d_i}$	250
Steam, exhaust, high-vacuum	$75\sqrt{d_i}$	330

*9 ft/s maximum for galvanized steel pipe; 6 ft/s maximum at inlet nozzles of and within tubular heat exchangers; 3 ft/s minimum to discourage fouling by marine organisms.

Notes:

1. Nominal fluid velocity limits vary with piping inside diameter, where d_i = the piping inside diameter in inches. Maximum fluid velocity limits are independent of pipe size. Both limitations should typically be satisfied.
2. All fluid velocity values are in ft/s.
3. Fluid velocity limits are given for guidance only. Applicable regulations and specifications should always be consulted for requirements that pertain to specific installations.

TABLE 11-12
Representative Fixture Unit Values with Minimum Supply and Drain Sizes

Fixture	Type of Water Supply Flow Control	Number of Water Supply Fixture Units (WSFUs)			Minimum Size of Fixture Supply Branch[E]	Number of Drainage Fixture Units (DFUs)	Minimum Size of Fixture Drain and Trap[F]
		Cold Water	Hot Water	Total			
Lavatory (Private)	Faucet	0.75	0.75	1	3/8	1	1-1/4 to 1-1/2
Shower	Mixing Valve	1.5	1.5	2	1/2	2	2
Urinal (Public)[A,D]	Flushometer Valve	4 to 5	—	4 to 5	3/4	4 to 5	1-1/2 to 2
Water Closet (Private)[B]	Flushometer Valve	5	—	5	1	3	3
Water Closet (Private)[C]	Flushometer Valve	7	—	7	1	4	3
Water Closet (Public)[B,D]	Flushometer Valve	5 to 8	—	5 to 8	1	4 to 6	3
Water Closet (Public)[C,D]	Flushometer Valve	8 to 10	—	8 to 10	1	6 to 8	3
Service Sink	Faucet	2.25	2.25	3	1/2	2 to 3	2 to 3
Scullery Sink	Faucet	3	3	4	1/2	4	1-1/2 to 2
Clothes Washer	Automatic	3	3	4	1/2	3	2
Drinking Fountain[C]	3/8" Valve	0.5 to 0.75	—	0.5 to 0.75	3/8 to 1/2	0.5 to 0.75	1-1/4

A. 1.0 U.S. gallon per flush, integral trap
B. 1.6 U.S. gallons per flush, integral trap
C. 3.5 U.S. gallons per flush, integral trap
D. Larger values given apply to heavy-use areas.
E. Size refers to nominal inside diameter in inches. A fixture supply branch is located between the water distribution pipe and the fixture supply connection.
F. Size refers to nominal inside diameter in inches. A trap should generally not be larger than the pipe into which it drains.
The above values are based, in part, on information included in the *National Standard Plumbing Code* and are representative only. Actual values may vary based on the plumbing fixtures and drainage arrangement used and on applicable specifications (e.g., values for water closets on vessels fitted with a vacuum sewage system will be less than those shown above).

TABLE 11-13
Representative Supply System Flow Rates

Number of Water Supply Fixture Units (WSFUs)	Supply-System Flow Rate (U.S. gpm)
10	27
15	31
20	35
30	41
40	47
50	51
100	68
200	91
300	110
400	125
500	140
1,000	218
2,500	380
5,000	600

The above values are based on information included in the *National Standard Plumbing Code* and on the use of flushometer valves on urinals and water closets. They are representative of data that can be used to estimate sizes for potable water and sanitary system distribution piping. This information is typical only.

As shown in table 11-1, the wall thickness of NPS 6 schedule 40 pipe is 0.280 in. When a 12.5 percent mill tolerance is subtracted from this value, the minimum wall thickness for NPS 6 schedule 40 pipe is reduced to 0.245 in., which exceeds the 0.130 in. calculated above. Consequently, NPS 6 schedule 40 pipe is suitable based on the consideration of hydrostatic stress.

Equation 11.6 does not include the effects of longitudinal stresses, externally applied loads (including loads due to gravity), expansion or contraction,

TABLE 11-14
Representative Maximum Number of Connected Drainage Fixture Units (DFUs) for Gravity Drains

	Horizontal Fixture Branch	Vertical Stack			Common Drain Pipe	
			Stack of More than Three Branch Intervals			
Nominal Size of Pipe (Inches)	Minimum Slope of 1/4-Inch per Foot of Horizontal Distance	Total for Stack of Three or Fewer Branch Intervals	Total per Branch Interval	Total for Stack	Slope per Foot of Horizontal Distance 1/4-Inch	1/2-Inch
1-1/4[A]	1	2	1	2		
1-1/2[A]	3	4	2	8		
2[A]	6	10	6	24	21	26
2-1/2[A]	12	20	9	42	24	31
3[B]	20	30 to 48	16 to 20	60 to 72	27 to 42	36 to 50
4	160	240	90	500	216	250
5	360	540	200	1,100	480	575
6	620	960	350	1,900	840	1,000
8	1,400	2,200	600	3,600	1,920	2,300
10	2,500	3,800	1,000	5,600	3,500	4,200
12	3,900	6,000	1,500	8,400	5,600	6,700

A. No water closets

B. Not more than two water closets per fixture branch, branch interval, or drain pipe; not more than six water closets per vertical stack.

Notes:

1. The above values, which are based on information included in the *National Standard Plumbing Code* and in the *National Plumbing Codes Handbook*, are representative only. They do not apply to vacuum sewage systems. Applicable specifications should always be consulted when designing an actual system.
2. Individual fixture drains are connected to fixture branches. Multiple fixture branches are then frequently connected, either directly or through vertical stacks, to a common drain pipe.
3. The NPS of a fixture branch should not be less than the size of any fixture drain or trap connected to it.
4. The NPS of a vertical stack should not be less than the size of any of the horizontal fixture branches connected to it.
5. A branch interval is a distance along a vertical stack approximately equal to the height between two adjacent decks within which the horizontal branches from one deck are connected to the stack.
6. When a vertical stack is sized based on the total accumulated load at each branch interval, the upper portion of the stack may be smaller than the lower portion of the stack. However, the upper portion of a stack should be no less than one-half of the size required for the lower portion of the stack.
7. The NPS of a common drain pipe should not be less than the size of any of the vertical stacks or horizontal fixture branches connected to it.
8. Adequate consideration should be given to the effects of list, trim, pitching, and rolling when designing a shipboard drainage system.
9. Drains and drain piping should be properly vented.
10. Soil or black water drains, which contain human excrement, are frequently segregated from waste or gray water drains, which convey liquid waste that does not contain human excrement.
11. Drainage piping selected should be large enough to permit adequate drainage. However, if too large a pipe size is used, velocities may not be sufficient to properly scour the inner pipe surface.

and cyclic loading, which would all have to be considered in a thorough stress analysis.

The effects of longitudinal stress due to the pressure of fluid within a pipe, piping weight, and sustained applied mechanical loads should frequently meet the following:

$$S_L = \frac{p\,d_o}{4t_n} + \frac{k_3(0.75i)M_A}{Z} \leq S_h \tag{11.7}$$

where

S_L = longitudinal stress from pressure, weight, and other sustained loads, psi (kPa)
t_n = nominal wall thickness of pipe, in. (mm)
k_3 = 1 when using the USCS units of measurement shown (1,000 for the metric units shown)

i = stress-intensification factor to account for bends, etc. (0.75i should never be less than 1.0)
M_A = resultant moment due to weight and other sustained loads, in-lbf (mm-N)
Z = section modulus of pipe, in.^{3} (mm^3)
S_h = basic material allowable stress at maximum working temperature, psi (kPa)

The section modulus for a section of straight circular pipe can be determined as follows:

$$Z = \frac{2I}{d_o} = \frac{\pi}{32d_o}\left(d_o^4 - d_i^4\right) \tag{11.8}$$

where

I = area moment of inertia, in.^{4} (mm^4)
π = 3.1416
d_i = inside diameter of pipe, in. (mm)

A modification to equation 11.7 is sometimes made to add the effects of occasional loads to the effects of longitudinal stress due to the pressure of fluid within the pipe, piping weight, and sustained applied mechanical loads, as is shown below:

$$\frac{pd_o}{4t_n} + \frac{k_3(0.75i)M_A}{Z} + \frac{k_3(0.75i)M_B}{Z} \leq kS_h \tag{11.9}$$

where

M_B = resultant moment due to occasional loads, in-lbf (mm-N)
k = 1.15 for occasional loads acting less than 10 percent in any 24-hour operating period, 1.2 for occasional loads acting less than 1 percent in any 24-hour operating period

Piping systems that undergo temperature changes can be subjected to forces and moments resulting from thermal expansion or contraction. Typical values of linear thermal expansion for pipe are shown in table 11-15. The acceptability of stresses due to thermal expansion can sometimes be determined using the following:

$$S_E = \frac{k_3 i M_C}{Z} \leq S_{A'} = F\left[1.25\,S_c + 0.25\,S_h + (S_h - S_L)\right] \tag{11.10}$$

TABLE 11-15
Linear Thermal Expansion of Pipe

Material	Linear Thermal Expansion in Inches per 100 Feet of Length over Temperature Range from 70°F to Indicated Temperature						
	70°F	200°F	400°F	600°F	800°F	1,000°F	1,200°F
Carbon Steel, C-Mo Steel, Low-Chrome (up to 3Cr) Steel	0.00	0.99	2.70	4.60	6.70	8.89	11.10
12Cr Stainless Steel	0.00	0.86	2.30	3.90	5.60	7.40	9.20
Austenitic Stainless Steel	0.00	1.46	3.80	6.24	8.80	11.48	14.20
Brass	0.00	1.52	4.05	6.80	9.78	12.98	16.39
Wrought Iron	0.00	1.14	3.01	5.01	7.12	9.36	
70Cu-30Ni Alloy	0.00	1.33	3.52				

Notes:

1. To determine the linear thermal expansion of any length of pipe:
   (a) Obtain the value from the above table corresponding to the minimum temperature and subtract it from the table value corresponding to the maximum temperature. The result will be the thermal expansion in inches per 100 ft of length for the actual temperature range.
   (b) Divide the result from step (a) by 100. This yields the thermal expansion in inches per foot of length.
   (c) Multiply the result from step (b) by the actual length of the pipe in feet. The result will be the linear thermal expansion of the pipe in inches.
2. The above data is based in part on information included in ASME B31.1-1995 and given to provide typical information regarding thermal expansion. It should not be inferred from this table that any of the materials listed are suitable for all of the temperatures shown.
3. C = carbon; Cr = chromium; Mo = molybdenum; Cu = copper; Ni = nickel. The number preceding the element abbreviation represents the percentage of the element in the metal.

where

S_E = expansion stress, psi (kPa)
M_C = resultant moment due to thermal expansion, in-lbf (mm-N)
$S_{A'}$ = allowable stress range, psi (kPa)
F = stress range reduction factor for cyclic loading
S_c = basic material allowable stress at minimum working temperature, psi (kPa)

Typical values of F used in equation 11.10 can vary from 1.0 for 7,000 or less temperature cycles over the life of the piping system to 0.5 for over 100,000 cycles.

Resultant moments used in equations 11.7, 11.9, and 11.10 can be determined based on the square root of the sum of the squared values of the individual moments acting about three mutually perpendicular axes. This is shown below for a straight length of pipe.

$$M_j = \sqrt{\left(M_{bp,j}\right)^2 + \left(M_{bt,j}\right)^2 + \left(M_{t,j}\right)^2} \quad (11.11)$$

where

M = resultant moment, in-lbf (mm-N)
M_{bp} = bending moment in plane of the pipe, in-lbf (mm-N)
M_{bt} = bending moment transverse to the plane of the pipe, in-lbf (mm-N)
M_t = torsional moment, in-lbf (mm-N)

Subscript) = A when calculating the resultant moment due weight and other sustained mechanical loads for use in equation 11.7, B when calculating the resultant moment due to occasional loads for use in equation 11.9, and C when calculating the resultant moment due to thermal expansion for use in equation 11.10.

Moments and forces resulting from thermal expansion can sometimes be estimated by performing a simplified analysis. In addition, more complex stress analyses can often be performed using a general-purpose finite-element-method (FEM) program or a specialized computer program written specifically for piping system design. When expansion stress is excessive, it can often be reduced by using expansion joints and by including U_and L-shaped loops and expansion joints in a piping system. In addition, transient thermal stresses can often be reduced by heating and cooling piping slowly.

EXAMPLE 11-5: An NPS 4 schedule 80 pipe is used for a pipeline filled with steam at a working pressure of 800 psig and a temperature of 900°F. The pipe material is 11/2 percent chrome, lIz percent molybdenum (liz Cr-¥2 Mo) steel. The resultant moment due to the weight of the pipe is 500 in-lbf. The applied mechanical loads are negligible, and the resultant bending moment from thermal expansion is 1,500 in-lbf. The stress-intensity factor (i) is equal to 1.5. In addition, the number of temperature cycles over the life of the system will be 5,000 (F = 1.0).

Are the longitudinal stress resulting from internal pressure and weight and the expansion stress below the allowable values?

Solution:

(a) From table 11-1, the outside diameter (d_o) and nominal wall thickness (t_n) of NPS 4 schedule 80 pipe = 4.500 in. and 0.337 in. respectively. Also, from table 11-10, the material being used has a maximum allowable stress of 13,800 psi at ambient temperature and 12,800 psi at 900°F.

Based on the above pipe dimensions, the pipe's inside diameter, d_i, equals

$$d_i = d_o - 2(t_n) = 4.500 \text{ in.} - 2(0.337 \text{ in.}) = 3.826 \text{ in.}$$

(b) Using equation 11.8

$$Z = \frac{\pi}{32(4.500 \text{ in.})}\left[(4.500 \text{ in.})^4 - (3.826 \text{ in.})^4\right] = 4.271 \text{ in.}^3$$

(c) Using equation 11.7

$$S_L = \frac{800 \text{ psig}(4.500 \text{ in.})}{4(0.337 \text{ in.})} + \frac{1.0\left[0.75(1.5)\right]500 \text{ in-lbf}}{4.271 \text{ in.}^3} = 2{,}802 \text{ psi} \le 12{,}800 \text{ psi}$$

(d) Using equation 11.10

$$S_E = \frac{1.0(1.5)1{,}500 \text{ in-lbf}}{4.271 \text{ in.}^3} = 527 \text{ psi}$$

and the allowable tress equals

$$S_{A'} = 1.0[1.25(13{,}800 \text{ psi}) + 0.25(12{,}800 \text{ psi}) + (12{,}800 \text{ psi} - 2{,}802 \text{ psi})] = 30{,}448 \text{ psi}$$

(e) Referring to step (c), the longitudinal stress is less than the allowable value (2,802 psi < 12,800 psi). In addition, referring to step (d), the expansion stress is less than the allowable value (527 psi < 30,448 psi).

Water Hammer

When a steam valve is opened quickly and steam begins to flow into a line that is filled with condensate, the steam forces the condensate to accelerate rapidly. If the fast moving condensate hits a bend, elbow, closed valve, or a similar obstruction, the sudden change in the momentum of the condensate as its path is altered or it is stopped will result in a pressure surge and in the transmission of a mechanical force to the piping system. In extreme cases, this force, which is often accompanied by a loud banging noise referred to as water hammer, and vibration can result in a failure of the piping system. Water hammer in steam lines can be minimized by draining lines of condensate prior to the admission of steam and by opening steam valves slowly. Large steam valves are frequently fitted with small warm-up bypass valves that can be opened first to allow steam to slowly fill and warm the line before the larger valve is opened. Opening the warm-up valve first also equalizes the pressure and temperature on both sides of the larger valve, which can reduce the force needed to open the larger valve.

Water hammer can also occur when the flow ofliquid through a piping system is abruptly stopped by rapidly closing a valve. The conversion ofkinetic energy to potential energy as the liquid's velocity is suddenly reduced to zero results in a rapid increase in the liquid's pressure. The pressure wave is initially transmitted backward through the pipe. In addition, the force necessary to stop the flow is absorbed by the piping system. This force can be reduced by closing valves slowly.

The increase in fluid pressure if flow is stopped instantaneously can be estimated as follows:

$$p_{\text{rise}} = \frac{V_0}{k_4}\left\{\frac{g}{\gamma}\left[\frac{1}{K_b} + \frac{d_i}{E(t)}\right]\right\}^{-1/2} \qquad (11.12)$$

where

p_{rise} = fluid pressure rise, psi (kPa)
V_0 = average fluid velocity within the pipe in the direction of flow prior to valve closure, ft/s (m/s)
k_4 = 12 when using the USCS units of measurement shown (1 for the metric units shown)
g = acceleration from gravity, 32.2 ft/s^2 (9.81 m/s^2)
γ = fluid specific weight, lbf/ft^3 (kN/m^3)
K_b = bulk modulus of elasticity of the fluid, psi (kPa)
d_i = pipe inside diameter, in. (mm)
E = Young's modulus of elasticity for the piping material, psi (kPa)
t = piping wall thickness, in. (mm)

Typical values for K_b include 294,000 psi for cold freshwater at pressures up to 500 psig and 150,000 psi for gasoline.

EXAMPLE 11-6: Calculate the pressure rise resulting from an instantaneous valve closure in an NPS 4 schedule 80 pipe through which cold freshwater with a specific weight of 62.4 16 f/ft^3 is flowing at a velocity of 10 ft/s. The piping is steel with a modulus of elasticity of 30E+6 psi.

Solution:

(a) Ignoring mill tolerances, piping wall thickness *(t)* will equal the specified nominal thickness (t_n) (e.g., in table 11-1). Therefore, from example 11-5, the wall thickness *(t)* and inside diameter (d_i) of NPS 4 schedule 80 pipe are 0.337 in. and 3.826 in., respectively.

(b) Using equation 11.12

$$p_{\text{rise}} = \frac{10\,\frac{\text{ft}}{\text{s}}}{12}\left\{\frac{32.2\,\frac{\text{ft}}{\text{s}^2}}{62.4\,\frac{\text{lbf}}{\text{ft}^3}}\left[\frac{1}{294{,}000 \text{ psi}} + \frac{3.826 \text{ in.}}{30\text{E}{+}6 \text{ psi}(0.337 \text{ in.})}\right]\right\}^{-1/2} = 597 \text{ psi}$$

Flanges, Valves, and Fittings

Butt-welded fittings are generally classified by nominal pipe size (NPS or DN), where the fitting's outside diameter is equal to that of pipe having the same nominal size. The wall thickness of the fitting can be determined based on a designation, such as standard, extra strong, double extra strong, or a schedule number, which also corresponds to the designations used with pipe. Flanges, valves, and flanged, socket-welded, and threaded fittings are typically rated or classified by the maximum allowable working pressure, usually in psi or kPa, at a specific working temperature. As the actual working temperature is reduced, however, the allowable working pressure often increases above the rated value or class. Some sample pressure and temperature limits for carbon-steel standard class valves are included in table 11-16. As shown, a carbon-steel standard class 150 valve can be used in a system with a maximum working pressure not exceeding 140 psig (965 kPa) when the working temperature is 600°F (315°C). In addition, this same valve is also suitable for use with system pressures up to 285 psig (1,966 kPa) ifthe working temperature is reduced to lOO°F(38°C). Values shown in table 11-16 are typical only; applicable specifications should always be referred to for·the pressure-temperature limits of actual components being used in a system.

Common class ratings for flanges and flanged fittings constructed from various materials are shown below.

Cast iron	Class 125 and 250
Steel	Class 150,300,400,600,900, 1500, and 2500
Bronze	Class 150 and 300

Steel flanges are often provided with raised faces. To reduce the possibility of failure due to bending stress created when a raised-face flanged joint is bolted together, Class 125 cast-iron flanges typically have flat faces. In a case where a Class 125 cast-iron flange is being bolted to a raised-face flange constructed from another material, it is generally recommended that the raised face be machined off of the mating flange prior to assembly. This results in a flat-faced flangedjoint. Threaded or screwed joints should ordinarily not be used in locations where severe erosion, corrosion, shock, or vibration is expected to occur. In addition, threads should generally not be cut into pipe having a wall thickness less than that corresponding to schedule 40 steel pipe. Furthermore, when used in some applications, such as with steam above 250 psig (1,750 kPa) or water above both 100 psig (700 kPa) and 220°F (l05°C), threaded steel pipe should be seam-

TABLE 11-16
Representative Allowable Working Pressures at Various Temperatures for Carbon-Steel Standard-Class Valves

	Maximum Allowable Nonshock Working Pressure in psig by Valve Class					
Temperature (F)	**Class 150**	**Class 300**	**Class 600**	**Class 900**	**Class 1,500**	**Class 2,500**
–20 to 100	285	740	1,480	2,220	3,705	6,170
200	260	675	1,350	2,025	3,375	5,625
400	200	635	1,270	1,900	3,170	5,280
600	140	550	1,095	1,640	2,735	4,560
800	80	410	825	1,235	2,060	3,430

Values given above are based on information included in ASME B16.34-1996 for standard-class valves with ASTM A 216 Grade WCB carbon-steel bodies and bonnets and are representative only. Applicable specifications and regulations should always be consulted for requirements pertinent to specific installations.

less, have a minimum tensile strength of 48,000 psi (330,000 kPa), and have a thickness at least equal to schedule 80.

Threaded or screwed fittings are usually not permitted to be used in Group I piping systems when temperatures exceed 925°F (496°C). Typical pressure limitations on the use of threaded or screwed fittings in Group I piping systems are shown below.

Pipe Size	*Maximum Pressure*
Above NPS 2 (DN 50)	Not permited in Group I systems
Above NPS 1 (DN 25) to NPS 2 (DN 50)	600 psig (4,140 kPa)
Above NPS % (DN 20) to NPS 1 (DN 25)	1,200 psig (8,280 kPa)
NPS % (DN 20) and below	1,500 psig (10,300 kPa)

An exception to the above is sometimes made to allow screwed connections with tapered threads (e.g., NPT) in sizes up to NPS 3 (DN 80) to be used for making connections to equipment, such as valves, pumps, hoses, gauges, accumulators, etc., in hydraulic systems that are not related to steering or propulsion.

Flared, flareless, and compression fittings may typically be used in Group I systems only with tubing having an outside diameter not exceeding 2 in. (50 mm). In addition, in flammable-fluid systems, only flared and nonbite-type flareless fittings may be used with steel, nickel-copper alloy (monel), or copper-nickel alloy tubing, and only flared fittings may be used with copper or copper-zinc alloy tubing. Typical values of the minimum and maximum tubing wall thicknesses suitable for use with flared and bite-type flareless fittings are shown in table 11-5.

The above information is general in nature and all limitations and exceptions are not listed; applicable regulations and specifications should always be consulted for requirements that pertain in actual applications.

Markings that are normally stamped on a flange or fitting can include the manufacturer's trade name, nominal size, wall thickness, pressure rating or class, and material designation. Many dimensions for fittings and flanges are specified in standards issued by organizations, such as the American National Standards Institute (ANSI). Consequently, flanges or fittings of a given size, type, and rating manufactured to the same specification by different vendors should be interchangeable. Valves are often marked with their manufacturer's name, material designation, and pressure rating. Some valves are also marked with an arrow indicating the required direction offlow through the valve. In addition, some smaller valves are marked with a WOG rating, which refers to the cold water-oil-gas nonshock rating; this is the valve's maximum allowable working pressure in psi at ambient temperature (32° to lOO°For 0° to 38°C).

Valves can also have an SP (sometimes WSP) rating, which stands for steam pressure and, unless otherwise specified, is the pressure limitation that applies to saturated steam.

Bolting

The bolting used to secure a flanged joint or to hold a valve bonnet or gland in place must have adequate strength and be suitable for the operating temperature range. Heat-treated, high-strength bolts are typically required in high-pressure and high-temperature applications, such as steam systems. In some cases, special markings are applied to a fastener so that its material can be identified. When the material or adequacy of a fastener is in question, the fasteer should not be used.

System Head

The energy per unit mass that fluid must have to flow through a system at a given flow rate can be represented by the total system head. Total system head is the sum oftwo components:

1. A static component due to differences in the fluid's static pressure and elevation at the beginning and at the end of the system (this component does not vary with flow rate).
2. A dynamic component due to both differences in the kinetic energy of the fluid at the beginning and at the end of the system and losses from friction and turbulence in the system's piping, valves, and fittings (this component, which also typically includes system entrance and exit losses and, when applicable, losses in bends, enlargements, and contractions in a system, increases with flow rate).

The total system head for a piping system can be calculated using the following:

$$SH = \frac{k_5(p_2 - p_1)}{\gamma} + Z_2 - Z_1 + \frac{V_2^2 - V_1^2}{2g} + h_f \qquad (11.13)$$

where

SH = total system head, ft (m)
k_5 = 144 when using the USCS units of measurement shown (1 for the metric units shown)
p = absolute static fluid pressure, psia (kPa abs)
γ = fluid specific weight, lbf/ft^3 (kN/m^3)
Z = elevation above (+) or below (–) a specified datum, ft (m)
V = average fluid velocity within the pipe in the direction of flow, ft/s (m/s)

g = acceleration from gravity, 32.2 ft/s^2 (9.81 m/s^2)
h_f = head loss from friction in piping, valves, and fittings (including system entrance and exit losses and, when applicable, losses in bends, enlargements, and contractions), ft (m)

Subscripts 1 and 2 in equation 11.13 refer to the system inlet and the outlet, respectively.

The average fluid velocity within a pipe in the direction of flow can be found as follows:

$$V = \frac{k_6 4Q}{\pi d_i^2} \qquad (11.14)$$

where

k_6 = 0.321 when using the USCS units of measurement shown (278 for the metric units shown)
Q = volumetric flow rate through pipe, U.S. gpm (m^3/hr)
d_i = pipe inside diameter, in. (mm)

Equations 11.13 and 11.14 can be used for liquids; they can also be used for gases when the change in fluid density throughout the pipe is small (i.e., when the change in static pressure is less than 2.25 percent and the Mach number is less than 0.3). Values of total system head can be calculated at various flow rates and used to plot a curve of system head versus flow rate (see fig. 11-61).

When a system has multiple parallel and series branches (fig.11-62a), the system head curve can first be determined for each branch individually. The combined system head curve for each set of parallel branches that begin or end at a common junction can then be developed by adding the

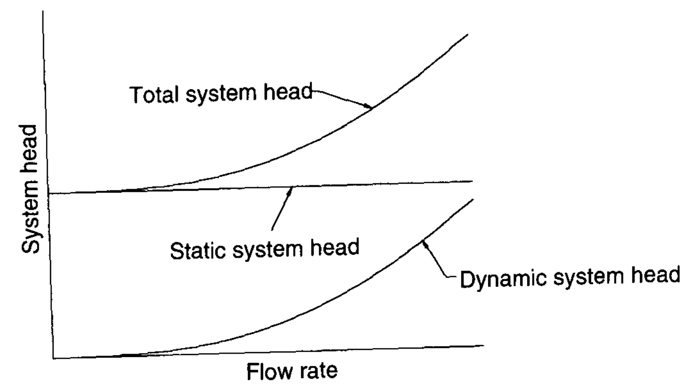


Figure 11-61. System-head curve

capacity through each branch at equal values of system head. The overall total system head curve can be determined by adding the system head of all series branches (including the combined system head of any parallel branches that are in series with other sections of the system) at values of equal capacity (fig. 11-62b). When a pump is used to deliver liquid through a multiple-branch system, such as a fire pump or seawater service pump, the pump will deliver a capacity corresponding to the point at which its

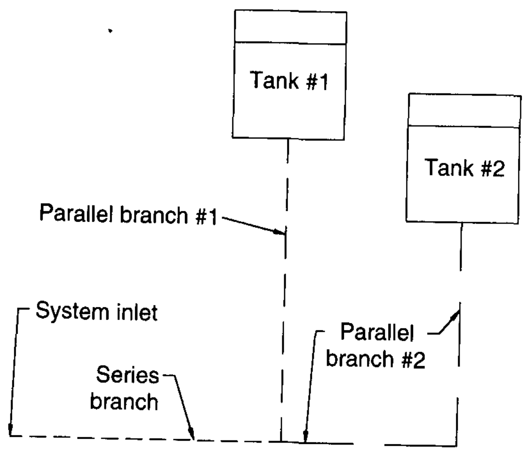


Figure 11-62a. Multiple-branch system

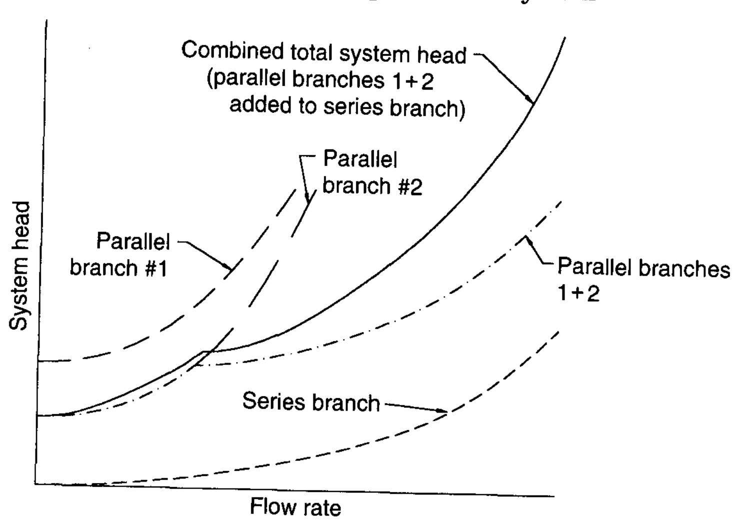


Figure 11-62b. System-head curve for multiple-branch system

performance curve of total head versus capacity intersects the overall total system head curve.

Losses due to friction in piping are sometimes referred to as major losses. The head loss in a straight length of piping can be calculated using the Darcy-Weisbach equation:

$$h_{f,\,\text{pipe}} = k_7 f\left(\frac{V^2}{2g}\right)\frac{L}{d_i} \tag{11.15}$$

where

$h_{f,\text{pipe}}$ = head loss in piping, ft (m)
k_7 = 12 when using the USCS units of measurement shown (1,000 for the metric units shown)
f = Darcy-Weisbach friction factor
L = pipe length, ft (m)
d_{i} = pipe inside diameter, in. (mm)

Equation 11.15 can be used for liquids and for gases when the change in fluid density throughout the pipe is small. Values of the Darcy-Weisbach friction factor used in equation 11.15 can be determined graphically from the Moody diagram (fig. 11-63) using the Reynolds number, which is a dimensionless ratio of inertial forces divided by viscous or friction forces, and the relative pipe roughness. The Reynolds number and the relative pipe roughness can be calculated using the following:

$$Re = \frac{Vd_{\text{i}}}{k_7 \nu} \tag{11.16}$$

where

Re = Reynolds number
ν = kinematic viscosity of fluid, ft^2/s (m^2/s)

and

$$\varepsilon/D = k_7 \frac{\varepsilon}{d_{\text{i}}} \tag{11.17}$$

where

ε/D = relative pipe roughness (where D represents the pipe inside diameter in ft or m)
ε = absolute pipe roughness, ft (m)

Values of absolute and relative pipe roughness are shown in fig. 11-64.

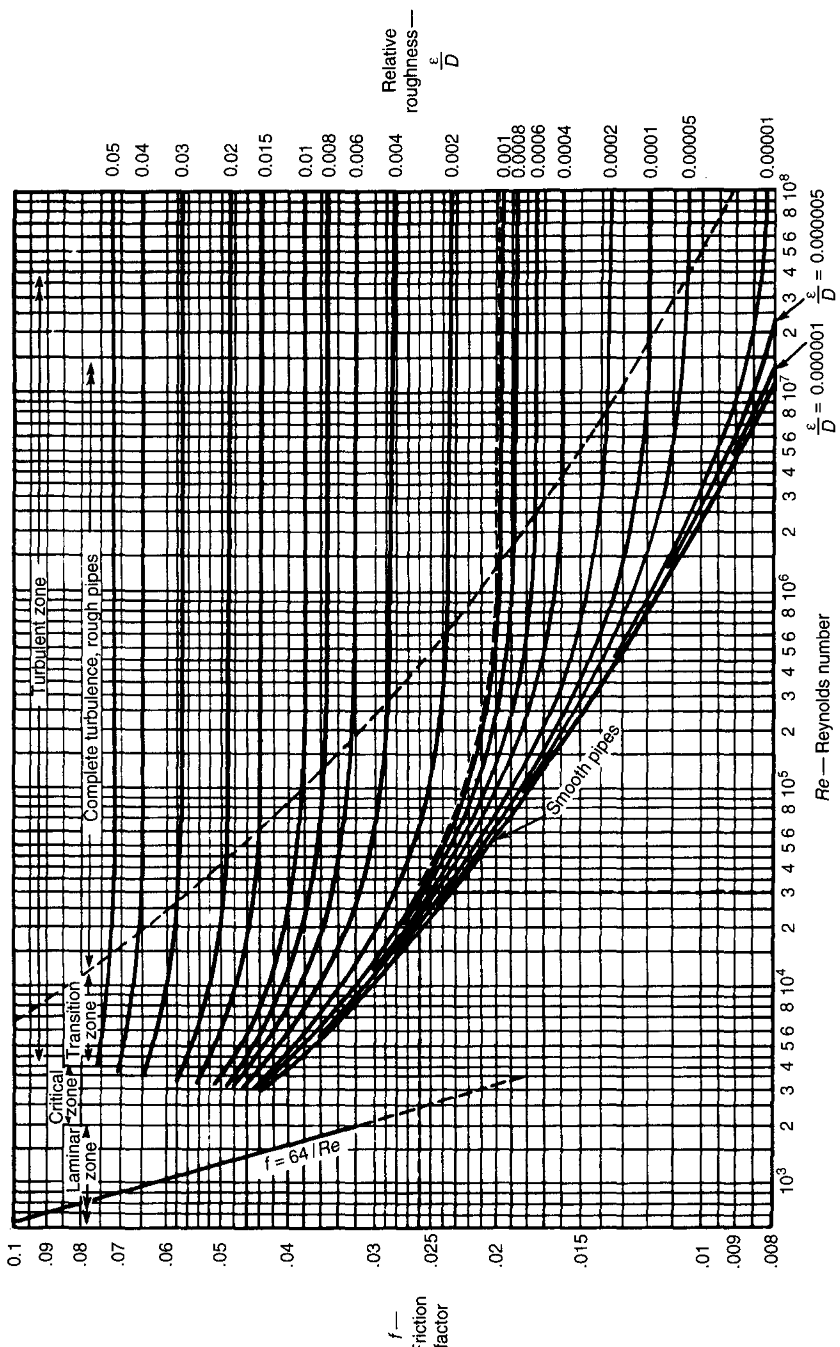


Figure 11-63. Moody diagram. From L.F. Moody, "Friction Factors for Piping Flow," *Transactions* (1944). Courtesy American Society of Mechanical Engineers.

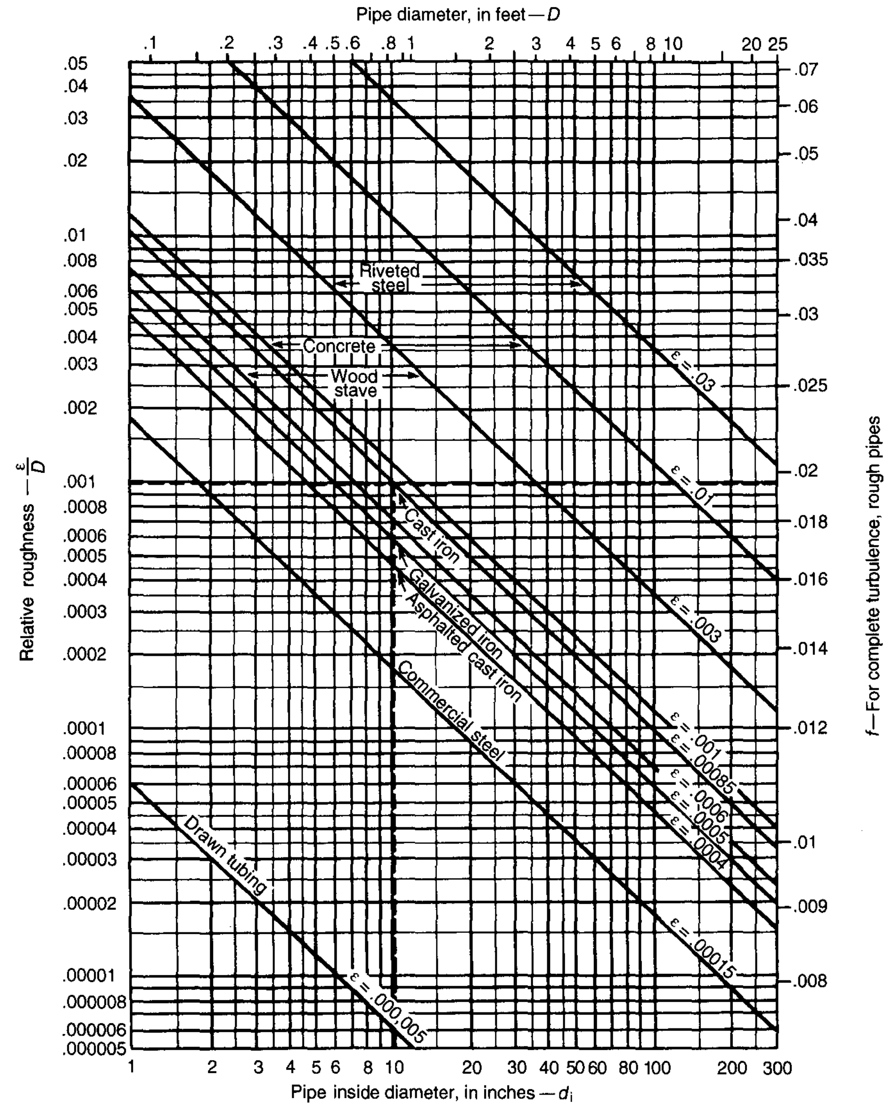


Figure 11-64. Pipe roughness. From L.F. Moody, "Friction Factors for Piping Flow," *Transactions* (1944).
Courtesy American Society of Mechanical Engineers.

When the Reynolds number is less than approximately 2,100 to 2,300, flow is generally laminar and follows well-defined streamlines. As shown in figure 11-63, in the laminar zone

$$f = \frac{64}{Re} \tag{11.18}$$

The velocity profile corresponding to laminar flow through a straight section of circular pipe resembles a paraboloid of revolution, with zero velocity directly adjacent to the inner wall of the pipe and the maximum velocity, which is equal to approximately twice the average velocity, in the center of the pipe. As shown in equation 11-18, the value of f in the laminar zone is not affected by pipe roughness. In addition, because values of f in the laminar flow region are inversely proportional to fluid velocity, values of $h_{f,Pipe}$ in this region increase linearly with average velocity.

When the Reynolds number is between 2,000 and 4,000, flow is in the critical zone and can be either laminar or turbulent and the value for f is indeterminate.

When Re is approximately 4,000, flow enters the turbulent-flow transition zone and is usually turbulent in the center of a pipe. With turbulent flow, both longitudinal (in the direction of flow) and transverse velocity components exist in the fluid. In addition, although the longitudinal velocity of fluid directly adjacent to the pipe's inner wall is zero, this velocity increases more rapidly at short distances away from the wall than in laminar flow. This results in a much flatter longitudinal-velocity distribution with turbulent flow than that occurring during laminar flow. The ratio of the average velocity divided by the maximum velocity in the direction of flow is approximately 0.80 when Re equals 10,000.

As the Reynolds number increases within the turbulent-flow transition zone, the width of the turbulent core increases, and the thickness of the laminar or viscous sublayer of fluid adjacent to the inner wall of a pipe, i.e., the layer in which turbulent fluctuations are suppressed, is reduced. Because protuberances on a pipe's inner wall increase the amount of turbulence, the Darcy-Weisbach friction factor in the transition zone is affected by both the Reynolds number and the roughness of the pipe wall. When Re increases to approximately 11,000 in a pipe with a relative roughness ratio of 0.05, the viscous sublayer becomes so thin that its effect on flow is negligible, and turbulent flow becomes fully developed. In this zone, referred to as the zone of complete turbulence, the Darcy-Weisbach friction factor becomes a function of only the relative pipe roughness and is no longer affected by the Reynolds number. Because of this, within the zone of complete turbulence, h_f varies approximately with the flow rate squared. The Reynolds number necessary for complete turbulence, which cannot be achieved in smooth pipe, increases as the relative roughness of a pipe's inner wall is reduced (see fig. 11-63).

As an alternative to using the Moody diagram, values of f for turbulent flow can be found from the Colebrook equation.

$$\frac{1}{\sqrt{f}} = -2\log_{10}\left(\frac{k_7\varepsilon}{3.7d_i} + \frac{2.51}{Re\sqrt{f}}\right) \tag{11.19}$$

Additional equations that can be used to find approximations of f in the turbulent zone are found in Hodge's *Analysis and Design of Energy Systems*.

Head losses occurring in valves, fittings, contractions, enlargements, and bends, together with entrance and exit losses, are sometimes referred to as minor losses. In some cases, a valve or fitting can be represented by an equivalent length of straight pipe at the diameter of the piping in the system through which there would be a friction loss equal to that in the valve or fitting. If the total equivalent length for valves and fittings in a system is then added to the piping length, L, in equation 11.15, the result of this equation will include the head loss both in the piping and in the valves and fittings. Typical equivalent lengths for representative valves and fittings can be found in figure 11-65. Alternatively, the head loss in valves, fittings, enlargements, contractions, and bends, together with entrance and exit losses, can be determined by using resistance coefficients, or K factors, and the following equation:

$$h_{f,\,\mathrm{minor}} = \left(\frac{V^2}{2g}\right)\sum_{i=1}^{n} K_i \tag{11.20}$$

where

$h_{f,\,\mathrm{minor}}$ = head loss in valves, fittings, etc., ft (m)
n = number of valves, fittings, etc. being considered
K_i = the resistance coefficient or K factor for valve, fitting, enlargement, contraction, bend, entrance, or exit i

Typical K factors for various types of valves, fittings, enlargements, contractions, bends, entrances, and exits are shown in table 11-17. Additional information can be found in the *Hydraulic Institute Engineering Data Book*, Crane Company's *Flow of Fluids Through Valves, Fittings, and Pipe*, and I. E. Ildechek's *Handbook of Hydraulic Resistance*. Values given often vary from source to source. However, in general, the K factor for a particular type of valve or fitting increases as the size of the valve or fitting is reduced. In addition, a valve or fitting with threaded connections normally has a greater resistance coefficient than the same size valve or fitting with flanges. Most K factors given for valves are for valves that are fully open. An automatic control or regulating valve, however, typically operates in a partially open condition. The pressure drop through an

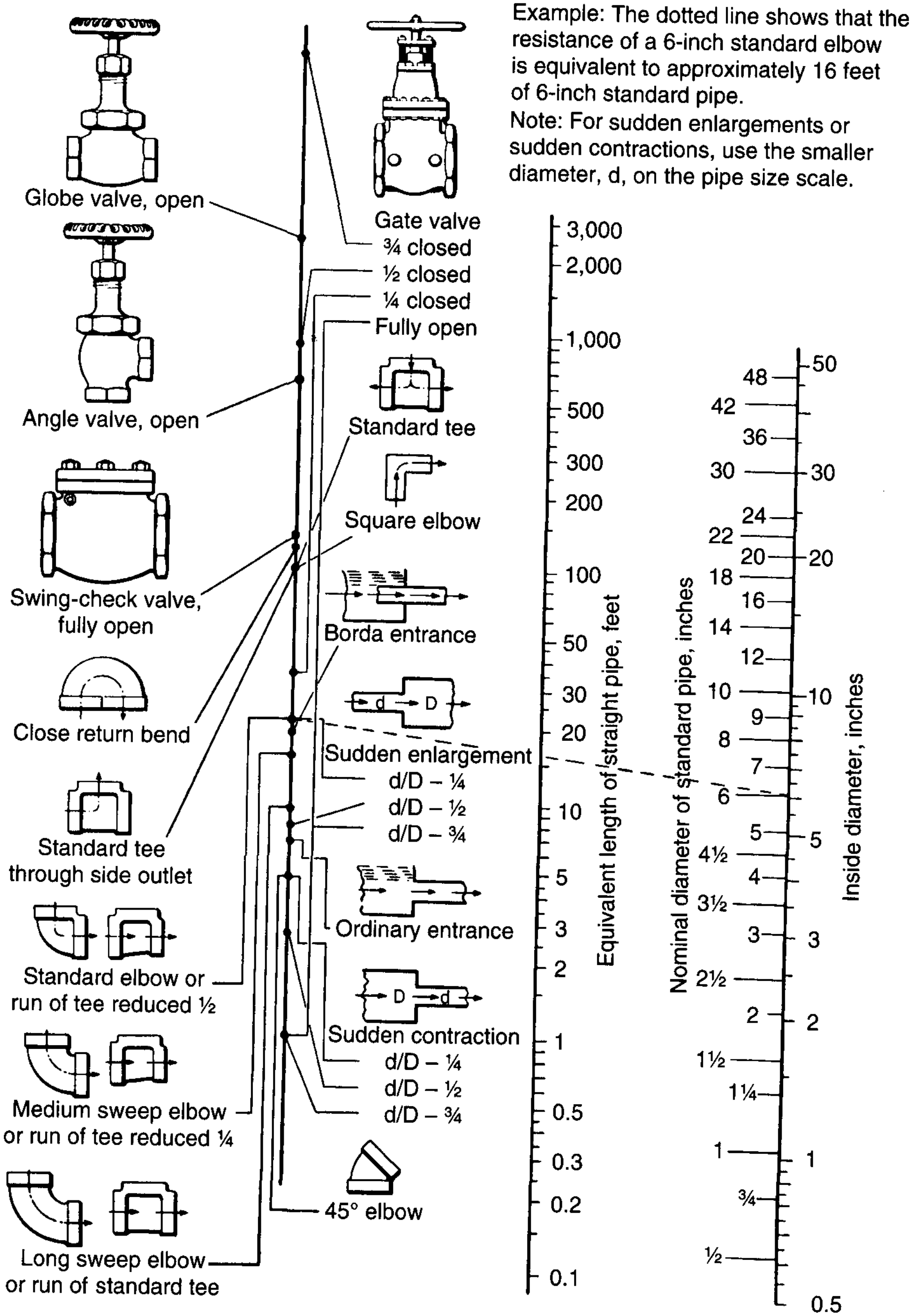


Figure 11-65. Resistance in valves. Courtesy Crane Company.

TABLE 11-17
Representative Resistance Coefficients (*K* Factors) for Various Types of Pipe Fittings and Valves

Fitting or Valve Type	Nominal Size (Inches) 1/2	1	2	4	6	10	20	Approximate Range of Variation (Percent)
Bell-Mouth Inlet or Reducer	0.05	0.05	0.05	0.05	0.05	0.05	0.05	
Square-Edged Inlet	0.50	0.50	0.50	0.50	0.50	0.50	0.50	
Square-Edged or Rounded Outlet	1.00	1.00	1.00	1.00	1.00	1.00	1.00	
Regular Screwed 90 Degree Elbow	2.00	1.50	1.00	0.67				20 over NPS 2 40 below NPS 2
Long-Radius Screwed 90 Degree Elbow	1.20	0.78	0.43	0.23				+/- 25
Regular Flanged 90 Degree Elbow		0.44	0.38	0.33	0.29	0.25	0.21	+/- 35
Long-Radius Flanged 90 Degree Elbow		0.40	0.31	0.22	0.18	0.14	0.09	+/- 30
Regular Screwed 45 Degree Elbow	0.38	0.33	0.30	0.28				+/- 10
Long-Radius Flanged 45 Degree Elbow		0.21	0.17	0.14	0.13	0.11	0.05	+/- 10
Screwed Return Bend	2.10	1.45	0.95	0.65				
Regular Flanged Return Bend		0.43	0.36	0.30	0.27	0.24	0.20	
Long-Radius Flanged Return Bend		0.43	0.31	0.22	0.18	0.14	0.01	
Screwed Tee, Line or Run Flow	0.90	0.90	0.90	0.90				+/- 25
Screwed Tee, Branch Flow	2.40	1.90	1.40	1.10				+/- 25
Flanged Tee, Line or Run Flow		0.27	0.20	0.15	0.12	0.09	0.07	+/- 35
Flanged Tee, Branch Flow		1.00	0.84	0.70	0.61	0.54	0.42	+/- 35
Screwed Globe Valve	14.00	8.90	6.90	5.80				+/- 25
Flanged Globe Valve		14.00	8.50	6.50	6.00	5.80	5.80	+/- 25
Screwed Gate Valve	0.33	0.24	0.18	0.13				+/- 25
Flanged Gate Valve		0.31	0.25	0.16	0.11	0.06	0.03	+/- 50
Screwed Angle Valve	10.00	4.80	2.20	1.00				+/- 20
Flanged Angle Valve		4.80	2.50	2.10	2.10	2.10	2.10	+/- 50
Screwed Swing-Check Valve	5.70	3.00	2.30	2.10				+/- 30
Flanged Swing-Check Valve		2.00	2.00	2.00	2.00	2.00	2.00	+/- 200/-80
Basket Strainer		2.50	1.50	1.10	0.89	0.69	0.44	+/- 50
Ball Valve, Full Port	0.11	0.10	0.08	0.06	0.05	0.04	0.03	
Butterfly Valve				1.60	0.90	0.68	0.55	
Union or Coupling	0.13	0.10	0.06	0.04				+/- 50

Notes:

1. Data above is based in part on information included in the *Hydraulic Institute Engineering Data Book* and is representative only. Published values can vary based on the source of the data. Approximate ranges of variation for K factors are shown above.
2. K factors for flanged fittings and valves can be used for fittings and valves with welded connections.
3. Valves are assumed to be fully open. In cases where a stop valve is throttled or the system velocity is too low to keep a check valve fully open, the resistance coefficient will be greater than the value shown above.
4. Flow through fittings and valves is generally assumed to be in the zone of complete turbulence.
5. No allowance is made for wear or fouling.

automatic control valve can usually be calculated using the valve's flow coefficient, C_v, and the following modified version of equation 11.1:

$$h_{f,\text{minor}\,(Cv)} = \frac{k_5}{\gamma_{fw}}\left(\frac{Q}{k_1 C_v}\right)^2 \tag{11.21}$$

where

$h_{f,\text{minor}\,(Cv)}$ = head loss in control valve, ft (m)
k_5 = 144 when using the USCS units of measurement shown (1 for the metric units shown)
γ_{fw} = specific weight of freshwater at 60°F, 62.34 lbf/ft^3 (9.8 kN/m^3)

Limitations on the use of equation 11.1 also apply to equation 11.21. The total head loss in a system, h_f, can be found by adding the results of equations 11.15, 11.20, and 11.21.

As a system ages, the piping material may deteriorate due to corrosion or erosion, which can increase the roughness of the internal surface. In addition, fouling may occur, which will not only affect the surface roughness but will reduce the effective inside diameter. Allowance factors are sometimes added to the head loss calculated for a new system to enable the effect that age will have on head loss to be predicted.

EXAMPLE 11-7: Calculate the total system head for the section of the cargo unloading system shown in figure 11-66 when the flow rate being discharged is 2,000 U.S. gpm. The pressure at the abovedeck discharge connection is 125 psig (139.7 psia). Because the cargo tank is inerted, the pressure within the cargo tank is 1.5 psig (16.2 psia). The specific weight (γ) and kinematic viscosity (ν) of the cargo being pumped are 55

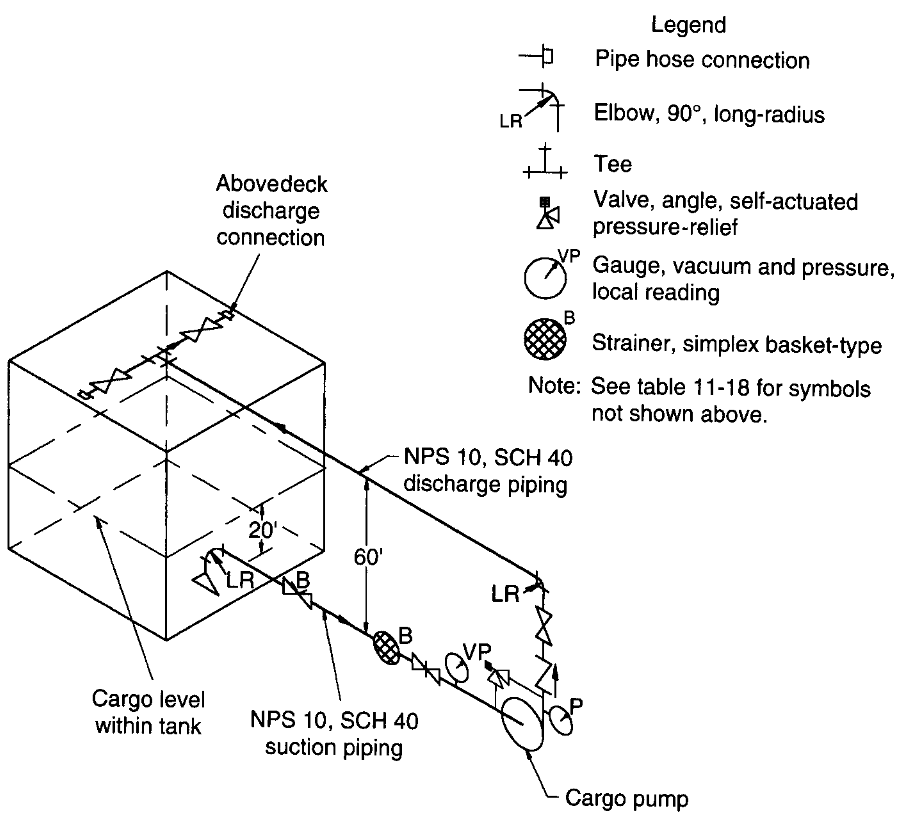


Figure 11-66. System for example 11-7

lbf/ft^3 and 9.63E–4 ft^2/s, respectively. In addition, the total length of the NPS 10 schedule 40 suction piping is 90 feet, and the total length of the NPS 10 schedule 40 discharge piping is 200 feet. The piping material is steel. Ignore any losses that may occur as cargo being discharged passes through the abovedeck connection and enters a shoreside arm or discharge hose. Also, ignore losses entering, within, or leaving the pump (these would be accounted for in the pump efficiency), and effects due to age, corrosion, or erosion of the piping.

Solution:

1. Suction side of cargo unloading system (i.e., the portion of the system on the suction side of the cargo pump)—

(a) From table 11-1, the outside diameter (d_o) and nominal wall thickness (t_n) of NPS 10 schedule 40 piping are 10.750 in. and 0.365 in., respectively. Based on these dimensions, the inside diameter (d_i) can be found as follows:

$$d_i = d_o - 2(t_n) = 10.75 \text{ in.} - 2(0.365 \text{ in.}) = 10.02 \text{ in.}$$

(b) Using equation 11.14

$$V = \frac{0.321(4)2{,}000 \text{ gpm}}{\pi(10.02 \text{ in.})^2} = 8.1\,\frac{\text{ft}}{\text{s}}$$

(c) Using equation 11.16

$$Re = \frac{8.1\,\frac{\text{ft}}{\text{s}}(10.02 \text{ in.})}{12\left(9.63\text{E}{-4}\,\frac{\text{ft}^2}{\text{s}}\right)} = 7{,}060$$

(d) From figure 11-64, the relative roughness ratio (ε/D) for commercial steel pipe with an inside diameter of 10.02 in. equals 1.8E–4.

(e) Using the results of steps (c) and (d), together with figure 11-63, the Moody friction factor (f) is found to equal 0.034.

(f) Using equation 11.15:

$$h_{f,\text{pipe}} = 12(0.034)\left[\frac{\left(8.1\,\frac{\text{ft}}{\text{s}}\right)^2}{2\left(32.2\,\frac{\text{ft}}{\text{s}^2}\right)}\right]\frac{90 \text{ ft}}{10.02 \text{ in.}} = 3.8 \text{ ft}$$

(g) Based on table 11-17, the resistance coefficients (K) for an NPS 10 90-degree long-radius flanged elbow, an NPS 10 fully-open butterfly valve, an NPS 10 fully-open flanged gate valve, and an NPS 10 basket strainer are 0.14, 0.68, 0.06, and 0.69, respectively. In addition, the entrance loss for a bell-mouth inlet is 0.05. Using equation 11.20

$$h_{f,\text{minor}} = \left[\frac{\left(8.1\,\frac{\text{ft}}{\text{s}}\right)^2}{2\left(32.2\,\frac{\text{ft}}{\text{s}^2}\right)}\right]\{0.14+0.68+0.06+0.69+0.05\} = 1.7\text{ ft}$$

(h) Adding the results of steps (f) and (g), the total head loss in the suction side of the cargo unloading system is

$$h_{f,\text{total suction}} = 3.8\text{ ft} + 1.7\text{ ft} = 5.5\text{ ft}$$

2. Discharge side of cargo unloading system (i.e., the portion of the system on the discharge side of the cargo pump)—

(a) Repeating steps (a) to (h) shown above, results for the discharge side of the system are d_o = 10.75 in., t_n = 0.365 in., d_i = 10.02 in., V = 8.1 ft/s, Re = 7.060, ε / D = 1.8E–4, f = 0.034, $h_{f,\text{pipe}}$ = 8.4 ft, K (NPS 10) flanged check valve = 2.0, K (NPS 10) flanged globe valve = 5.8, K (NPS 10) 90-degree long-radius flanged elbow = 0.14, K (NPS 10) flanged tee (branch) = 0.54, $h_{f,\text{minor}}$ = 10.2 ft, and $h_{f,\text{ total, discharge}}$ = 18.6 ft.

3. Total cargo unloading system—

(a) Referring to figure 11-66 and taking the centerline of the cargo pump as the reference datum, the elevation of the cargo at the system inlet (i.e., the cargo within the tank) is 20 ft. In addition, the velocity at the surface of the cargo within the tank is assumed to be negligible. The elevation and velocity at the system outlet (i.e., the abovedeck discharge connection) are 60 ft and 8.1 ft/s, respectively.

(b) The total head loss in the system equals 5.5 ft from the suction side of the system added to the 18.6 ft from the discharge side of the system, or 24.1 ft.

(c) Using equation 11.13, the total system head for the portion of the cargo unloading system being considered is calculated below.

$$SH = \frac{144\,(139.7\text{ psia} - 16.2\text{ psia})}{55\,\frac{\text{lbm}}{\text{ft}^3}} + 60\text{ ft} - 20\text{ ft} + \frac{\left(8.1\,\frac{\text{ft}}{\text{s}}\right)^2 - 0^2}{2\left(32.2\,\frac{\text{ft}}{\text{s}^2}\right)} + 24.1\text{ ft} = 388.5\text{ ft}$$

Vibration

When a natural frequency of a piping system is too close to the frequency of an applied force, such as the frequency at which pressure

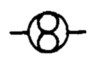
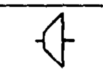
pulsations are generated in the fluid passing through the system, excessive vibration can occur. In some cases, this vibration can be lessened by reducing the amplitude of the force or by increasing the gap between the offending forcing frequency and the system natural frequency being excited. Typically, increasing the stiffness or reducing the mass of a piping system will raise its natural frequencies. However, because multiple natural frequencies will be affected by changes made to a system, a change to the number, type, or locations of supports, that moves one natural frequency away from a known forcing frequency can often move another system natural frequency closer to a different forcing frequency. To avoid a trial-and-error approach, a vibration analysis can frequently be effective in predicting the effect that system changes will have on vibration characteristics. Although specialized computer programs are available for performing vibration analyses, many general-purpose finite-element-method programs can also be used.

Piping-System Diagrams

A piping-system diagram is a schematic drawing of a piping system showing the functional relationship of piping sections that compose a system in a simplified, single-line format with flow arrows, interconnections, and system components such as valves, heat exchangers, pumps, gauges, thermometers, filters, and strainers, which are depicted using symbols. Primary sections of the system being depicted are typically drawn with solid lines that are heavier than the lines used for any secondary portions ofthe system, such as drain lines, gauge lines, and control lines. Boundaries located in close proximity to the system, such as the vessel's hull, bulkheads, and the sides oftanks, may also be shown. However, bends, elbows, and fittings included on arrangement drawings that show how piping is actually routed through a vessel are generally not included on piping-system diagrams.

Notes regarding applicable specifications and regulations; fabrication, cleaning, coating, and testing procedures; special construction requirements; references to interfacing drawings; and similar details are also ordinarily included on a piping-system diagram. Additional information provided, often in tabular form, can include piping nominal sizes and schedules; materials of construction for piping, fittings, valves, bolting, and gaskets; fluids, flow velocities (during normal operation), and friction pressure drops in each section of the system; operating, design, and test pressures; design temperatures; an equipment list showing information regarding pumps and other machinery installed in the system; and a legend identifying the symbols used to depict system components. Table 11-18 includes examples of some typical piping-diagram symbols.

TABLE 11-18
Typical Piping Diagram Symbols

Title	Symbol	Title	Symbol
Strainer, duplex basket-type		Valve, globe, self-actuated pressure-relief	
Strainer, Y-type basket		Valve, boiler safety	
Filter		Manifold, single row	
Centrifugal purifier		Manifold, double row, stop-check	
Valve, globe		Valve, pressure-reducing (increase in downstream pressure shuts valve)	
Valve, globe with flow-control device		Valve, control diaphragm (increased actuating pressure closes valve)	
Valve, globe, stop-check		Valve, thermostatic expansion	
Valve, globe, Y-pattern		Gauge, pressure, local reading	
Valve, angle		Transducer, pressure	
Valve, angle, stop-check		Thermometer, remote reading	
Valve, angle, ball		Thermocouple	
Valve, swing-check		Pump, centrifugal	
Valve, lift-check		Pump, rotary, positive-displacement	
Valve, check, in-line ball or popper		Pump, reciprocating, positive-displacement	
Valve, ball		Compressor	
Valve, ball, three port		Turbine, steam	
Valve, butterfly		Turbine, gas	
Valve, gate		Tailpipe, bell-mouth suction	

The above is based on information included in ASTMF 1000-92.

REVIEW

1. Define NPS and DN.
2. Describe four different ways to join two sections of pipe.
3. Describe six different types of pipe fittings.
4. Describe four different types of fittings that can be used with tubing.
5. Describe four different types of hose connectors.
6. Identify three different types of linear-shaft valves and list the characteristics of each type.
7. Identify three different types of rotary-shaft valves and list the characteristics of each type.
8. Explain the difference between a direct-acting and a reverse-acting actuator for a control valve.
9. Explain the difference between a normally open and a normally closed control valve.
10. What flow rate of freshwater at a temperature of 120°F will pass through a valve having a flow coefficient (C_v) of 35 if the pressure at the inlet to the valve is 200 psig and the valve's outlet pressure is 10 psig?
11. The cavitation index (K_c) for the valve in question 10 is equal to 0.65. Will cavitation occur within the valve?
12. What is a constant-pressure control pilot and how does it work?
13. What is a differential-pressure control pilot and how does it work?
14. What is a level control pilot and how does it work?
15. Describe five different types of steam traps and explain how they work.
16. Prepare a procedure to repack a main-steam stop valve.
17. Prepare a procedure to inspect and lap a main-steam stop valve.
18. What is the flow rate of freshwater with a specific weight equal to 9.8 kN/m^3 that will pass through an orifice with a 75 mm diameter opening and a discharge coefficient of 0.65? The orifice is installed in a pipeline with a 200 mm inside diameter, and the pressure drop measured across the orifice is 300 kPa.
19. Based on hydrostatic stress, is a plain-end NPS 12 schedule 80 pipe suitable for use in a system with a maximum allowable working pressure of 750 psig at a temperature of 180°F? The pipe material is seamless carbon steel.
20. A DN 250 schedule 40 pipe is used for a pipeline filled with steam at a working pressure of 4,500 kPa and a temperature of 350°C. The pipe material is carbon steel. The resultant bending moment due to the weight of the pipe is 170,000 mm-N. Assume that applied mechanical loads are negligible, and that the resultant bending moment from thermal expansion is 250,000 mm-N. The stress-intensity factor is equal to 1.5. In addition, the number of temperature cycles over the life of the system will be 7,000.

21. Calculate the pressure rise resulting from an instantaneous valve closure in a DN 300 schedule 40 pipe through which cold freshwater is flowing at a velocity of 4.5 m/s. The piping is steel with a modulus of elasticity of 200 GPa.

REFERENCES

American Bureau of Shipping (ABS). 1997. *Rules for Building and Classing Steel Vessels.* New York: ABS.

American Society of Mechanical Engineers (ASME). 1986. *Cast Bronze Threaded Fittings.* ASME B16.15-1985 (R1994). New York: ASME.

———. 1997. *Forged Fittings, Socket-Welding and Threaded.* ASME B16.11-1996. New York: ASME.

———. 1993. *Malleable Iron Threaded Fittings.* ASME B16.3-1992. New York: ASME.

———. 1992. *Pipe Threads, General Purpose (Inch).* ASME B1.20.1-1983 (R1992). New York: ASME.

———. 1995. *Power Piping.* ASME B31.1-1995. New York: ASME.

———. 1997. *Valves—Flanged, Threaded, and Welding End.* ASME B16.34-1996. New York: ASME.

———. 1997. *Welded and Seamless Wrought Steel Pipe.* ASME B36.10M-1996. New York: ASME.

American Society for Testing and Materials (ASTM). 1997. *Classification for Machine Made "Fiberglass" (Glass-Fiber-Reinforced Thermosetting-Resin) Pipe.* ASTM F 2310-97. West Conshohocken, Pa.: ASTM.

———. 1993. *Seamless Copper Pipe, Standard Sizes.* ASTM B 42-93, West Conshohocken, Pa.: ASTM.

———. 1995. *Seamless Copper Water Tube.* ASTM B 88-95a. West Conshohocken, Pa.: ASTM.

———. 1995. *Seamless Copper Water Tube, [Metric].* ASTM B 88M-95. West Conshohocken, Pa.: ASTM.

———. 1994. *Seamless Red Brass Pipe, Standard Sizes.* ASTM B 43-94. West Conshohocken, Pa.: ASTM.

———. 1992. *Standard Practice for Piping System Drawing Symbols.* ASTM F 1000-92. West Conshohocken, Pa.: ASTM.

———. 1998. *Standard Practice for Selection and Application of Piping System Materials.* ASTM F 1155-98. West Conshohocken, Pa.: ASTM.

———. 1992. *Standard Specification for Thermosetting Resin Fiberglass Pipe and Fittings to be Used for Marine Applications.* ASTM F 1173-95. West Conshohocken, Pa.: ASTM.

Avallone, E. A. and T. B. Baumeister, eds. 1996. *Marks' Standard Handbook for Mechanical Engineers,* 10th ed. New York: McGraw-Hill.

Crane Company. 1982. "Flow of Fluids Through Valves, Fittings, and Pipe." Crane technical paper no. 410. New York: Crane Co.

Czernik, D. E. 1996. *Gaskets, Design, Selection, and Testing.* New York: McGraw-Hill.

Fisher Controls Company. 1965. *Control Valve Handbook.* Marshalltown, Iowa: Fisher Controls Co.

Frankel, M. 1996. *Facility Piping Systems Handbook.* New York: McGraw-Hill.

Harrington, R. L., ed. 1992. *Marine Engineering.* Jersey City, N.J.: Society of Naval Architects and Marine Engineers.

Heald, C. C., ed. 1994. *Cameron Hydraulic Data,* 18th ed. Liberty Corner, N.J.: Ingersoll-Dresser Pumps.

Hodge, B. K. 1990. *Analysis and Design of Energy Systems,* 2nd. ed. Englewood Cliffs, N.J.: Prentice-Hall.

Honeywell, W. "Enhanced Piping System Reliability with Heat Recoverable Couplings." *Marine Technology,* vol. 24, no. 3.

Hydraulic Institute. 1990. *Hydraulic Institute Engineering Data Book,* 2nd ed., Parsippany, N.J.: Hydraulic Institute.

Ildechek, I. E. 1986. *Handbook of Hydraulic Resistance,* 2nd ed. New York: Hemisphere Publishing.

International Standards Organization (ISO). 1996. "Ships and Marine Technology - Identification Colours for the Contents of Piping Systems." Draft International Standard ISOIDIS 14726-1. Geneva: ISO.

Leslie Controls, Inc. 1986. *Control Valve Sizing Manual.* Tampa, Fla.: Leslie Controls, Inc.

King, Horace W., et. al. 1980. *Hydraulics,* 5th ed. Malabar, Fla.: Krieger.

Merrick, Ronald C. 1991. *Valve Selection and Specification Guide.* New York: Van Nostrand Reinhold.

National Association of Plumbing-Heating-Cooling Contractors. 1993. *1993 National Standard Plumbing Code.* Falls Church, Va.: National Association of Plumbing-Heating-Cooling Contractors.

___ . 1995. *1995 Supplement to the 1993 National Standard Plumbing Code.* Falls Church, Va.

National Fire Protection Agency (NFPA). 1993. *Fire Hose Connections.* NFPA 1963-1993 Edition. Quincy, Mass.: NFPA.

Nayyar, Mohhinder L., ed. 1992. *Piping Handbook,* 6th ed. New York: McGraw-Hill.

Parker-Hannifin Corporation. 1979. *Design Engineer's Handbook.* Bulletin 0224-B1. Cleveland, Ohio: Parker-Hannifin.

Smith, Paul R., and Thomas J. Van Lann. 1987. *Piping and Pipe Support Systems.* New York: McGraw-Hill.

Stepanoff, A. J. 1957. *Centrifugal and Axial Flow Pumps,* 2nd. ed., New York: John Wiley & Sons.

Ulanski, W. 1991. *Valve & Actuator Technology.* New York: McGraw-Hill.

United States Government. 1997. *Code of Federal Regulations.* 46CFR, Subchapter F-Marine Engineering. Washington, D.C.: U.S. Government Printing Office.

___ . 1980. *Standard Specifications for Slow Speed Diesel Merchant Ship Construction.* Washington, D.C.: U.S. Maritime Administration.

White, Frank M. 1986. *Fluid Mechanics,* 2nd. ed. New York: McGraw-Hill.

Woodson, R. Dodge. 1998. *National Plumbing Codes Handbook,* 2nd. ed., New York: McGraw-Hill .

Zappe, R. W. 1991. *Valve Selection Handbook*, 3rd ed. Houston: Gulf Publishing.

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CHAPTER 12

Fluid Transfer Devices

WILLIAM J. SEMBLER

PUMPS

Pumps are used to add energy to liquids to produce flow or increase pressure. The two basic types of pumps are the kinetic pump, in which a spinning rotor increases the velocity and pressure of the pumped liquid, and the positive-displacement pump, in which a movable element forces liquid through the pump's casing. The energy added by a pump to the liquid passing through it is often represented by the total pump head, which is equal to the total discharge head of the liquid at the pump outlet minus the liquid's total suction head at the pump inlet. Using subscripts *s* and *d* to refer to conditions at the points of measurement in the suction and discharge lines, respectively (points s and d in figure 12-1), the total head developed by an operating pump equals

$$H = \left(\frac{C_1 p_d}{\gamma} + \frac{V_d^2}{2g} + Z_d + h_{fd}\right) - \left(\frac{C_1 p_s}{\gamma} + \frac{V_s^2}{2g} + Z_s - h_{fs}\right) \tag{12.1}$$

where*

H = total (pump) head, ft (m)
C_1 = 144 when using the USCS units shown (1 for the metric units)
p = absolute pressure at the point of measurement, psia (kPa abs)
γ = specific weight of the pumped liquid at the pumping temperature, $\mathrm{lbf/ft^3}$ ($\mathrm{kN/m^3}$)

*When units of measurement and constants for both the USCS and metric systems are given, the metric values are shown in parentheses.

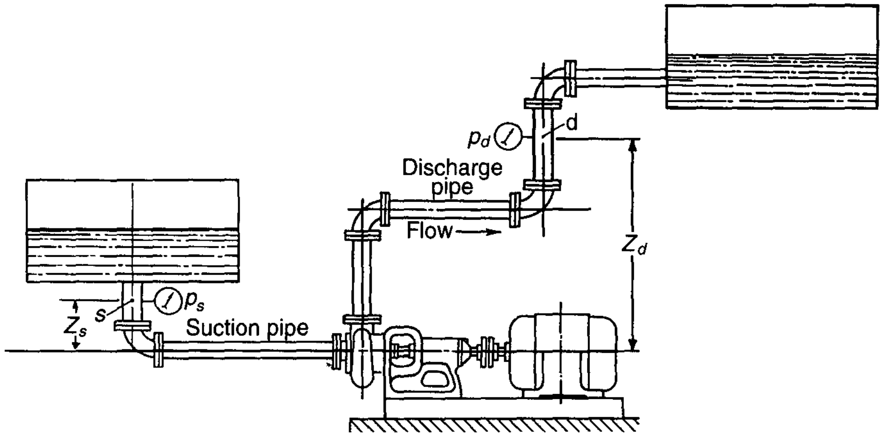


Figure 12-1. Typical pump and system

V = average liquid velocity at the point of pressure measurement, ft/s (m/s)
g = acceleration from gravity, 32.2 ft/s^2 (9.81 m/s^2)
Z = elevation at the point of pressure measurement above (+) or below (–) the standard datum, ft (m)
h_f = head loss from friction and turbulence in piping, valves, and fittings from the pump to the point of pressure measurement, ft (m)

For a horizontal centrifugal pump or a vertical centrifugal pump with a double-suction first-stage impeller, the standard datum is established by a horizontal plane that passes through the centerline of the first-stage impeller. For a vertical centrifugal pump with a single-suction first-stage impeller, the standard datum is at the elevation of the first-stage impeller's entrance eye. In the case of a rotary or reciprocating pump, the standard datum is established by the horizontal plane passing through the centerline of the pump's inlet port.

A modification to equation 12.1 that enables total pump head to be calculated using the specific gravity of the pumped liquid is shown below:

$$H = \frac{p_d - p_s}{C_2 sg} + \frac{V_d^2 - V_s^2}{2g} + Z_d - Z_s + h_{fd} + h_{fs} \qquad (12.2)$$

where

C_2 = 0.433 when using the USCS units shown (9.789 for the metric units)

sg = liquid specific gravity at pumping temperature based on 1.00 for freshwater at 68°F (20°C)

A value of total head in ft (m) can be thought of as the height of a column of the pumped liquid that could be supported by the total differential pressure developed by a pump. In addition, if when using USCS units of measurement the right-hand sides of equations 12.1 and 12.2 are multiplied by g/g_c, where g_c = 32.2 ft-lbm/lbf-s^2, values of total pump head will be in ft-lbf/lbm. Because values of head in ft and in ft-lbf/lbm are numerically equal at sea level, specifying total head in feet of liquid can be thought of as an abbreviation for ft-lbf/lbm, which is the net energy added by a pump per unit mass of the liquid pumped.

Example 12-1: For the system shown in figure 12-1, Z_s = 3 ft, p_s = 20 psia, Z_d = 10 ft, and p_d = 35 psia. In addition, the velocity within the suction pipe at point s is 5 ft/s, and the velocity within the discharge pipe at point d is 10 ft/s. Also, the head loss in the suction line from point s to the pump's suction flange equals 1 ft, and the head loss in the discharge line from the pump's discharge flange to point d equals 2 ft. The liquid being pumped is seawater with a specific gravity of 1.03. Determine the total head developed by the pump.

Solution: Using equation 12.2

$$H = \frac{35\text{ psia} - 20\text{ psia}}{0.433(1.03)} + \frac{\left(10\frac{\text{ft}}{s}\right)^2 - \left(5\frac{\text{ft}}{s}\right)^2}{2\left(32.2\frac{\text{ft}}{s^2}\right)} + 10\text{ ft} - 3\text{ ft} + 2\text{ ft} + 1\text{ ft} = 44.8\text{ ft} \cong 45\text{ ft}$$

Note: Provided that the values used for pressure, velocity, elevation, and head loss are correct at the points of measurement, the total pump head calculated will be the same regardless of the location of point s in the suction side of the system or the location of point d in the discharge side of the system.

Kinetic Pumps

CENTRIFUGAL PUMPS

Centrifugal Pump Design and Configuration

Impeller. A typical centrifugal pump has one or more shaft-mounted vaned impellers that rotate within a stationary casing and transfer energy to the fluid flowing between their vanes. Liquid entering a centrifugal pump impeller generally flows primarily in the axial direction. Any tangential component to the inlet flow is often referred to as prewhirl or prerotation. Liquid enters a double-suction impeller through two inlets, or entrance eyes. A single-suction impeller, however, has only one entrance eye.

A centrifugal pump impeller can be designed for radial-flow, mixed- (radial and axial) flow, or axial-flow operation, where these classifications refer to the primary orientation with respect to the shaft axis of flow at the impeller's discharge. The impeller-flow orientation required or used in a centrifugal pump can often be determined from the specific speed, which is a characteristic number that can be calculated using the following equation:

$$N_S = \frac{N(Q_{BEP})^{0.5}}{(H_{BEP})^{0.75}} \tag{12.3}$$

where

N_S = specific speed
N = pump operating speed, rpm
Q_{BEP} = total capacity delivered by the pump during operation at its best efficiency point (BEP), U.S. gpm (m^3/hr)
H_{BEP} = total pump head developed by the pump (total head per stage for a multistage pump) during operation at its BEP, ft (m)

With the units of measurement shown above, N_S can be thought of as the speed at which a geometrically similar impeller would have to be operated if it were sized to deliver a capacity of one U.S. gpm (m^3/hr) and develop a total head of one ft (m).

Table 12-1 shows an approximate correlation between flow orientation and values of specific speed (N_S) calculated using the units of measurement given with equation 12.3.

TABLE 12-1
Flow Orientation Versus Specific Speed

Flow Orientation	*Specific Speed, USCS*	*Specific Speed, Metric*
Radial (single suction)	< 4,200	< 4,880
(double suction)	< 6,000	< 6,970
Mixed (single suction)	4,200 to 9,000	4,880 to 10,450
Axial (single suction)	> 9,000	> 10,450

A low specific speed represents a relatively low capacity and a relatively high total head; consequently, low-specific-speed impellers tend to have large outside diameters with respect to their waterway widths. Conversely, a high-specific-speed impeller often has relatively wide waterways with respect to its outside diameter.

Centrifugal-pump impellers can also be classified based on shroud configuration: a closed impeller's vanes are located between both a front and a rear shroud; a semiopen impeller's vanes are attached to a single rear shroud; and an open impeller consists of vanes that are attached to the periphery of a hub and, in some cases, a partial rear shroud. The impellers in high specific-speed axial-flow pumps have no shrouds and frequently are referred to as propellers.

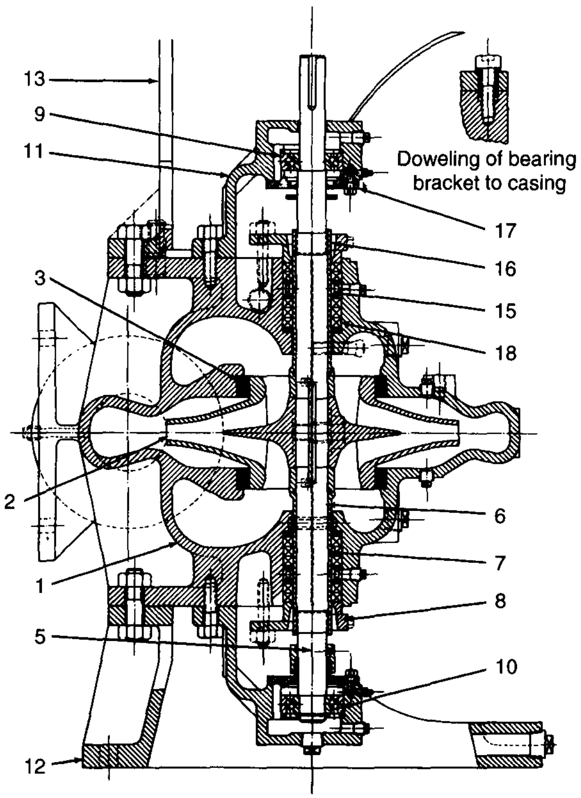
EXAMPLE 12-2: A single-stage centrifugal pump delivers a capacity of 2,000 U.S. gpm and develops a total head of 100 ft when operating at its best efficiency point. The pump's shaft is driven at a speed of 1,750 rpm. What is the pump's specific speed?

Solution: Using equation 12.3

$$N_S = \frac{1{,}750 \text{ rpm}(2{,}000 \text{ U.S. gpm})^{0.5}}{(100 \text{ ft})^{0.75}} = 2{,}475$$

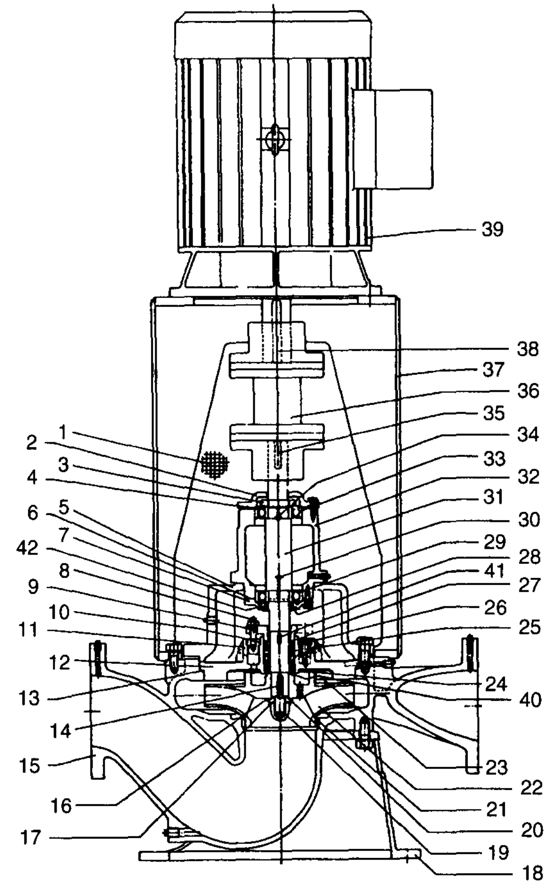
Casing. A centrifugal pump casing can be arranged so that the pump's shaft is oriented either horizontally or vertically. In addition, as shown in figure 12-2, the casing can be split in a plane that is parallel to the shaft's axis (referred to as an axially split casing), or, as shown in figures 12-3 and 12-4, the casing split can be in a plane that is perpendicular to the shaft (referred to as a radially split casing). With the exception of smaller pumps, which may have threaded connections, suction and discharge connections in a pump's casing are often flanged. When an axially split design is used, the pump's suction and discharge connections are both generally in the stationary half of the casing. This enables the casing to be opened for inspection or maintenance without disconnecting the suction and discharge piping. In the case of a radially split casing, this same feature can be incorporated into the pump design by utilizing a back-pull-out configuration. With this arrangement, both the suction and discharge connections are in a common portion of the casing (fig. 12-3). The name back-pull-out is derived from the ability to back the pump's rotating assembly out of the casing without disconnecting the suction and discharge piping. Because the removable section of the casing typically contains the pump's shaft seal, this configuration is also sometimes referred to as a seal-head design. Alternatively, some radially split casings have a suction head that is separate from the discharge portion of the casing (fig. 12-4). With this latter configuration, either the suction or the discharge piping must be disconnected from the pump before the casing can be disassembled. Radially split casings can also be classified based on the relative orientation of the pump's suction and discharge connections. For example, a vertical in-line (VIL) pump has suction and discharge connections that are 180° apart (fig. 12-3). In an end-suction pump, however, the casing's suction port, which leads directly into the eye of the first-stage impeller (the only impeller in a single-stage pump), is usually perpendicular to the discharge connection (fig. 12-4).

The suction portion of a casing guides fluid entering the pump from the suction connection to the eye of the first-stage impeller. Stationary guide vanes are sometimes added to the inner wall of a casing's suction nozzle to straighten the flow path ,at the entrance to the impeller and to break up vortices that may form if a portion of the fluid that enters the impeller is recirculated back out of the impeller's eye. The discharge portion of a casing includes a collector to catch the fluid discharged from the impeller and a channel to guide this fluid either to the pump's discharge connection or, in the case of a multistage pump, to the inlet of the next stage. Additionally,



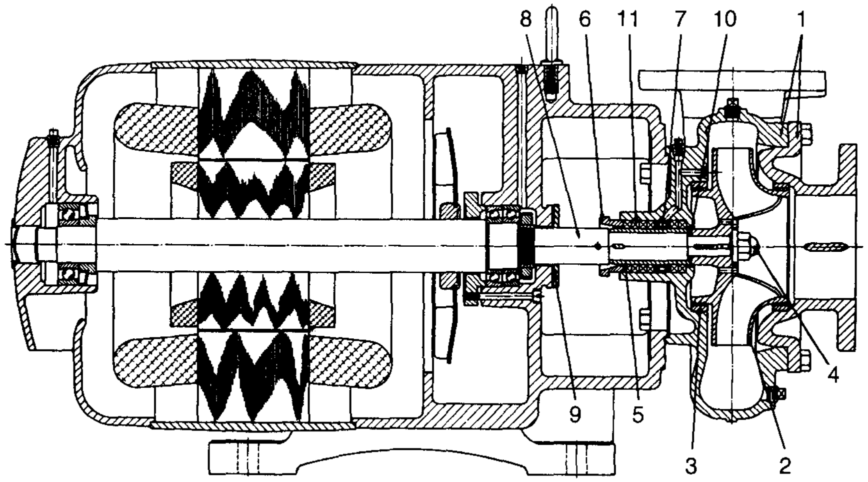
1 Casing
2 Impeller
3 Casing wearing ring
4 Impeller wearing ring
5 Shaft
6 Shaft sleeve
7 Stuffing box packing
8 Glands
9 Thrust bearing
10 Line bearing
11 Bearing housing
12 Pump base
13 Motor bracket
14 Coupling
15 Seal cage (lantern ring)
16 Shaft nut
17 Bearing cover
18 Packing

Figure 12-2. Vertical axially split casing centrifugal pump. Courtesy Ingersoll-Dresser Pump Company.



1 Coupling guard
2 Bearing cover, drive side
3 Bearing locknut
4 Lockwasher
5 Bearing bracket lantern
6 Gasket, bearing cover
7 Bearing cover, pump side
8 Gland stud
9 Gland
10 O-ring, inner
11 Cooling cap
12 Stuffing box cover
13 Gasket, casing
14 Impeller key
15 Pump casing
16 Gasket, lockwasher
17 Gasket, impeller nut
18 Pump foot
19 Impeller nut
20 Lockwasher
21 Wearing ring, casing
22 Impeller
23 Wearing ring, stuffing box cover
24 Throat bushing
25 Lantern ring
26 O-ring, outer
27 Gland packing
28 Slinger
29 Bearing, line
30 Grease nipple
31 Pump shaft
32 Bearing bracket housing
33 Bearing, thrust
34 Grease nipple
35 Coupling key, pump
36 Flexible spacer coupling
37 Mounting frame
38 Coupling key, motor
39 Motor, flange-mounted
40 Shaft sleeve (when fitted)
41 Shaft sleeve key (when fitted)
42 Radial lip seal, pump side

Figure 12-3. Vertical radially split casing (back-pull-out) centrifugal pump. Courtesy Ingersoll-Dresser Pump Company.



1 Casing and suction head
2 Impeller
3 Casing wearing ring
4 Impeller nut
5 Shaft sleeve
6 Gland
7 Lantern ring
8 Shaft
9 Slinger
10 Shaft sleeve seal ring
11 Packing

Figure 12-4. Horizontal close-coupled radially split casing (suction head) centrifugal pump. Courtesy Ingersoll-Dresser Pump Company.

the casing ordinarily has some type of diffuser in which a portion of the velocity head of the fluid being discharged from the impeller is converted to static pressure head. Referred to as pressure recovery, this conversion is necessary due to the relatively high absolute velocity of fluid leaving the typical centrifugal pump impeller.

One of the most common types of casing collectors used in single-stage radial- and mixed-flow centrifugal pumps is the volute, which is a scroll-shaped channel with a gradually increasing radius and cross-sectional area that surrounds the periphery of the impeller (fig. 12-5). To increase pressure recovery, the fluid leaving the volute is generally decelerated in the casing's discharge nozzle, which forms the transition from the volute's throat to the pump's discharge connection. As an alternative to a spiral volute, some centrifugal pumps are furnished with a circular or concentric collector having a constant radius and cross-sectional area. In addition, a modified or semiconcentric casing design, in which the radius and cross-sectional area of the collector remain constant over only a portion of the circumference, is sometimes used.

Volutes are also used in some multistage centrifugal pumps (fig. 12-6). When a multistage-volute pump has an axially split casing, the flow passages that connect successive stages, which are referred to as crossovers,

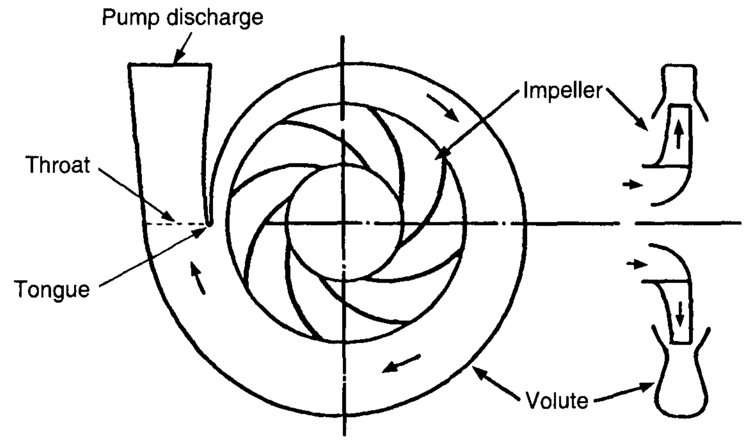
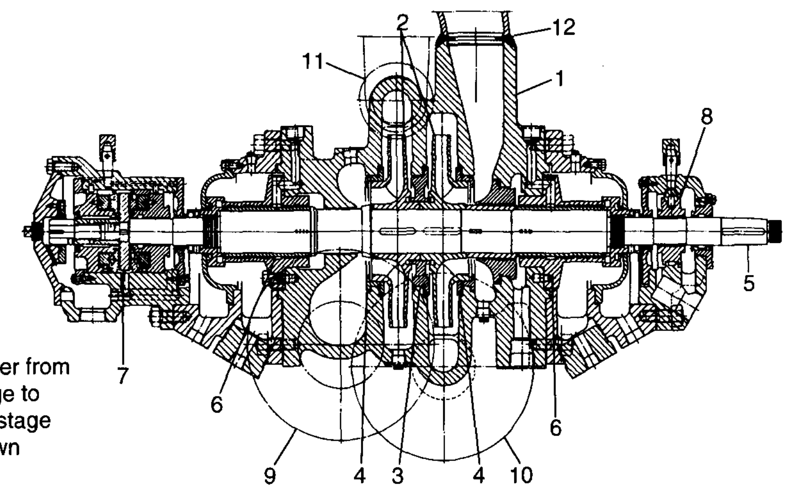


Figure 12-5. Centrifugal-pump volute. Courtesy Ingersoll-Dresser Pump Company.



1 Casing
2 Impeller
3 Stage piece
4 Casing wearing rings
5 Shaft
6 Sleeves
7 Thrust bearing
8 Line bearing
9 Suction flange
10 Discharge flange
11 Crossover inlet
12 Crossover outlet

Figure 12-6. Multistage axially split volute-type centrifugal pump. Courtesy Ingersoll-Dresser Pump Company.

may be integrally cast with or welded onto the casing. Alternatively, a multistage pump may be fitted with multivaned diffusers instead of volutes. As shown in figure 12-7, a multivaned diffuser, which is also used in some

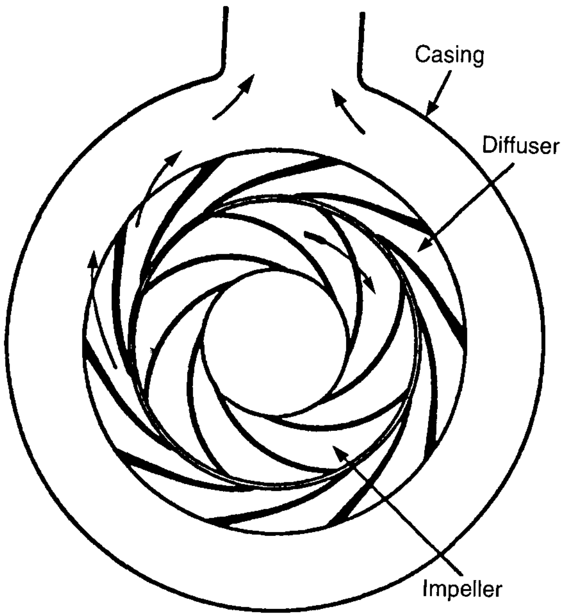
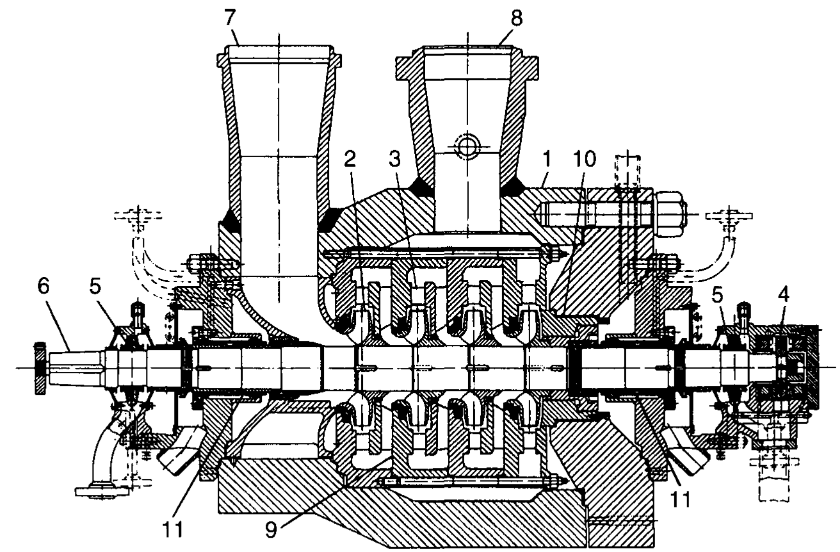


Figure 12-7. Centrifugal-pump diffuser.
Courtesy Ingersoll-Dresser Pump Company.

single-stage radial-flow pumps, contains a number of diverging vanes mounted in a ring that surrounds the periphery of the impeller. Diffuser-type multistage pumps frequently have radially split casings. With this configuration, the flow channels that join adjacent stages are generally formed by vanes that either are on the back side ofthe diffusers or are part of separate diaphragms, or stage pieces, used to separate adjoining stages. In addition, the pump's rotor, together with the stationary diffusers and stage pieces, can often be inserted into the casing as an assembled cartridge. Due to the casing's cylindrical shape, units of this type are frequently referred to as barrel pumps (fig. 12-8).

Many axial-flow pumps are fitted with an axial diffuser that is located downstream from the discharge of the propeller (fig. 12-9). In addition to converting a portion of the velocity head of the fluid leaving the propeller into pressure head, the axial diffuser straightens the flow path of the pumped liquid.

The casing configurations described above are typical only and are not all-inclusive. For example, some radially split casing barrel pumps have removable volutes in place of diffusers. In addition, the axially split casings used with some multistage pumps are fitted with removable diffusers in lieu of integral volutes. However, regardless of the configuration used, because the casing is a pressure-containing boundary, its minimum thick-



1 Casing
2 Impeller
3 Diffuser
4 Thrust bearing
5 Line bearings
6 Shaft
7 Suction port
8 Discharge port
9 Diaphragm with diffuser
10 Combined balancing drum and disk
11 Condensate injection shaft seals

Figure 12-8. Barrel-type centrifugal pump.
Courtesy Ingersoll-Dresser Pump Company.

ness should be sufficient to withstand the maximum working pressure for the application (normally, the maximum total suction pressure added to maximum total head that the pump can develop when operating at its maximum speed). The casing must also be suitable to withstand stresses due to hydraulic forces in its waterways, stresses that result from vessel motion, and stresses from loads imposed by the ship's suction and discharge piping. (Ideally, piping should be supported and aligned so that the loads it imposes on a pump are reduced to the minimum amount practicable.) In some cases, external ribs may be added to a casing to increase its strength and rigidity. Marine machinery must be suitable for operation over an extended period of time (intervals between scheduled overhauls can frequently exceed five years); consequently, an allowance is often added to the casing wall thickness to account for the effects of corrosion and erosion that occur while the pump is in service.

To avoid problems resulting from the corrosion of a threaded fastener in a tapped hole, casing joints often have through-holes and are secured with bolts and nuts. Alternatively, externally threaded center bolts with nuts at each end may be used. If a casing joint does have tapped holes, the use of studs and nuts to secure the joint is usually preferable to using cap screws. One reason for this is that a stud does not need to be removed from a tapped hole during disassembly, which reduces the risk of damage to the hole's internal threads. To prevent studs from backing out oftapped holes due to vibration or during disassembly, they are often held in place with either an interference fit between the mating threads of the stud and the hole or an anaerobic adhesive compound that is applied during assembly. To reduce the potential for overtightening, galling, leaks, and fatigue failure, threaded fasteners should always be lubricated with an antiseize thread compound in accordance with the equipment manufacturer's recommendations and torqued to specified values. Dowel pins or registered fits are frequently used to maintain proper alignment between mating sections of a casing.

To make casing joints fluid-tight, they are generally sealed with gaskets or, in the case of some continuous radial joints, a-rings. Flanged suction and discharge piping connections are also often sealed with gaskets or a-rings. Additionally, external auxiliary connections, such as those for vents, drains, and pressure gauges, must be fluid-tight. To verify that a casing is leakproof, a hydrostatic test is performed during which the casing is fIlled with water that is pressurized to a specified value (e.g., 150 percent

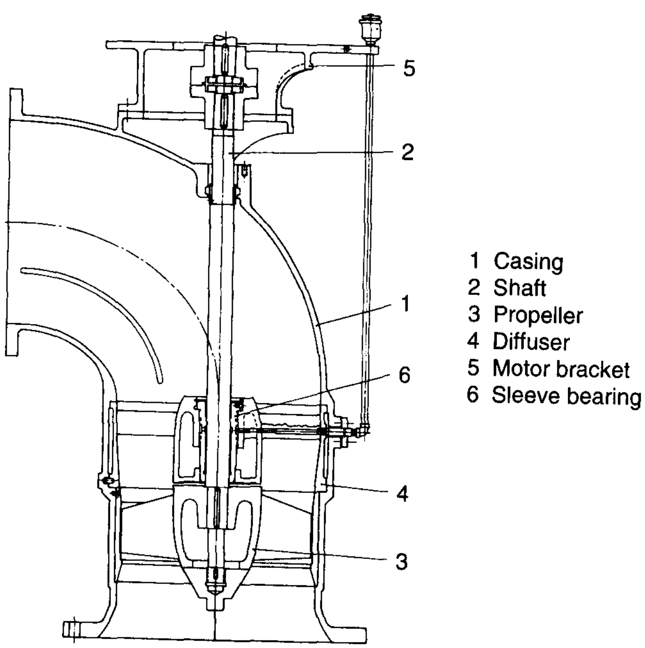


Figure 12-9. Axial-flow pump.
Courtesy Ingersoll-Dresser Pump Company.

of the pump's maximum working pressure). Because of the energy stored in compressed gases, it is important that air (or any other gas) be completely vented from any component being hydrostatically tested before the water within the component is pressurized.

Rotating assembly. In a single-stage pump with a between-bearings design, a single- or double-suction impeller is mounted on a shaft that is supported at each end by one or more bearings (fig. 12-2). When external bearings are mounted at both shaft ends, two shaft seals are required. If the outboard (opposite to the drive end) bearing is located within the pump's casing, however, the outboard shaft seal can be eliminated. Alternatively, an overhung arrangement is sometimes used in which the bearings are located behind an impeller that is mounted on the unsupported end of a cantilevered shaft (fig. 12-3). With this latter configuration, the pump's outboard bearing and shaft seal are both eliminated. When a pump with an overhung impeller is furnished in a close-coupled configuration, the pump's rotating parts are mounted directly onto the extended end of the driver's shaft (fig. 12-4). A close-coupled pump's rotor receives support from the bearings installed in the driver; however, ifthe shaft extension is long or radial loads are high, additional bearings may be installed in the pump.

Except for the first stage in some applications involving operation with limited values of net positive suction head, the typical multistage centrifugal pump is fitted with single-suction impellers. In addition, a multistage pump may be fitted with internal bearings between some or all of its stages to increase the support provided by the external bearings that are typically located either at one or both ends of its shaft.

Each impeller in a centrifugal pump is often keyed to the pump's shaft (a key should always fit snugly in its keyways). However, when a pump has an overhung rotor, an impeller is sometimes threaded onto the end ofthe shaft, which eliminates the need for a key. The need for an impeller key can also be eliminated when the pump shaft is splined or shaped like a polygon. With these alternate configurations, which are used most often in high-speed applications, the shaft engages a similarly shaped hole machined into each impeller.

Wearing rings. To reduce the leakage of high-pressure liquid from the discharge side ofa centrifugal pump back to the suction area of the casing, a close-clearance noncontacting seal is typically maintained between the casing and the impeller. When a single-suction closed impeller is used, this seal is formed between the periphery ofthe impeller's outer hub (the outer portion ofthe shroud adjacent to the impeller's entrance eye) and the inner wall of the pump's casing. With a double-suction closed impeller, a close clearance is maintained at both outer hubs. To enable this clearance to be periodically renewed, adjacent casing surfaces are typically fitted with replaceable stationary wearing rings. Rotating wearing rings may also be installed over the impeller's outer hubs. The use of both impeller and casing wearing rings is often referred to as double-ring construction. Two common types of casing wearing rings used are the cylindrical or flat-type ring and the L-type casing ring, which extends over the face of the impeller's outer hub and directs the leakage flow back into the impeller's eye. Alternatively, when reduced leakage flow is required, intermeshing labyrinth-type wearing rings may be used. Wearing-ring leakage is also sometimes reduced by adding a series of serrations or circular grooves to the inside surface of the casing ring or to the outeJ;surface of the adjacent impeller hub or ring. In addition, serrations and grooves can be added to wearing rings to reduce the potential for damage from occasional inadvertent contact between the close-clearance rotating and stationary surfaces or from foreign material in the pumped fluid that enters the gap between these surfaces.

Although satisfactory wearing-ring clearance values vary with pump design, typical diametral clearances for flat cylindrical rings, which increase with ring diameter, are in the range of 0.01 to 0.04 in. (0.25 to 1.00 mm). (A standard wearing ring's diametral clearance will often be approximately equal to 0.0015 times the ring's diameter.) Ordinarily, wearing rmgs should be replaced when the original design clearance doubles. Replacement casing rings are frequently furnished with a reduced or undersized inside diameter that can be machined at the time of installation based on the diameter of the adjacent impeller hub or ring. This enables the casing ring to be used with an impeller that has a remachined reduced-diameter wearing ring or outer hub. Similarly, a replacement impeller ring may be furnished with an oversized outside diameter that permits it to be used with a remachined casing ring having an increased inside diameter. Wearing rings installed in an axially split casing are often held in place with a tongue-and-groove arrangement. Wearing rings installed in radially split casings or on impeller hubs will typically have an interference fit and are frequently locked in place with set screws; the sealing surfaces of these rings should always be trued-up after installation.

When a centrifugal pump is fitted with either an open or a semiopen single-suction impeller, wearing rings are not required on the suction side ofthe impeller. However, to reduce the leakage, or slippage, of fluid across the impeller's vanes, a close axial clearance must be maintained between the unshrouded edges of the impeller's rotating vanes and the inner wall of the casing. So that this clearance can be periodically renewed, a replaceable wear plate is often fastened to the casing's inner wall adjacent to the suction side of a semiopen impeller and on both sides of an open impeller. The axial clearance between the impeller vanes and the wear plates can sometimes be adjusted by varying the thickness of shims installed on the pump shaft.

Interstage seals. To control interstage leakage in a multistage pump, close-clearance noncontacting seals are typically maintained between the pump's rotor and portions ofthe casing, stage pieces, or diffusers that separate adjacent stages. Replaceable bushings are frequently installed at these locations to enable the close running seal clearances to be periodically renewed.

Bearings. The net axial and radial loads applied to a pump's rotating assembly are transmitted to and absorbed by the bearings that support the shaft. Typically, one of these bearings, referred to as the thrust bearing, absorbs axial loads. To permit the shaft to expand and contract freely in response to changes in loading and temperature, the remaining bearings, sometimes called line bearings, are ordinarily configured to absorb only radial loads.

Single-row deep-groove-type ball bearings, which are able to absorb radial loads combined with moderate axial thrust in both directions, are used as line bearings and combined line-and-thrust bearings in many smaller-sized centrifugal pumps. When increased radial load carrying capability is required, however, double-row deep-groove ball bearings, double-row self-aligning ball bearings, and various types of roller bearings are sometimes used. In addition, when axial loads are high, an angular-contact ball-type thrust bearing may be used. Because a single-row angular-contact bearing can absorb axial loads applied in only one direction, a minimum oftwo rows mounted back-to-back (DB) or face-to-face (DF) must be used when this type ofbearing is required to absorb axial thrust applied in both directions. DB mounting provides a greater resistance to angular deflection caused by bending moments that may be applied to a pump's shaft; consequently it is sometimes preferred over DF mounting.

Ball bearings in marine centrifugal pumps are frequently lubricated with grease, which is generally easier to contain than oil, is superior to oil at keeping contaminants away from a bearing, and is relatively unaffected by vessel motion. The typical housing for a grease-lubricated bearing is fitted with inlet and outlet ports that are used for the addition and the removal of grease during relubrication. In addition, to help prevent grease from leaking out ofthe housing and to shield the bearing from contaminants, stationary lip or labyrinth seals are generally installed adjacent to openings provided for the pump's shaft. Also, one or both ends of a ball bearing can be fitted with shields or seals that help retain grease in the bearing and keep contaminants out. Slingers or flingers are frequently mounted on a pump's shaft external to bearing housings to help prevent dirt and moisture that may be traveling along the shaft from reaching the bearings. When pumping high-temperature liquids, it is sometimes necessary to use special bearings with increased running clearances or to circulate cooling liquid through internal passages within a jacketed bearing housing.

When relubricating a grease-lubricated bearing, the proper grade of grease should always be used. Although this grade varies with the application, a moisture-resistant grease with a National Lubrication Grease Institute (NLGI) No.2 consistency that contains a rust inhibitor is often suitable. Typical grease bases used include lime soap for bearing temperatures up to 150°F (66°C) and lithium soap for higher temperatures. Before adding grease, the bearing housing's relief plug should be removed and the opening should be cleaned. Then, with the pump running, if possible, grease should be added through a hydraulic grease fitting (e.g., with a hand-operated grease gun) or by tightening the cap of a grease cup. If the bearing housing's grease connection is initially fitted with a plug, a grease fitting or grease cup should be installed in place of the plug prior to relubrication. The relief plug should not be replaced until after the pump has been operated for four to eight hours following relubrication to allow excess grease to escape from the housing. Immediately after relubrication, bearing temperatures will normally rise, but should return to normal after this four-to-eight-hour period. Overlubrication will often lead to bearing failure and should be avoided; ordinarily, grease should occupy only one-third to one-half of the volume in a bearing housing. A typical bearing in normal duty should be properly lubricated prior to the initial start-up of the pump and driver, and then relubricated after every 1,000 to 2,000 hours of operation. (This is representative only; in all cases, the manufacturer's recommendations should be followed.)

When an antifriction bearing operates at a high speed or has a large-diameter bore, oil lubrication is frequently required. In a horizontal pump, the lower portion of the bearing (e.g., up to the center ofthe lowest ball in a ball bearing) may be submerged in lubricating oil, or oil may be distributed by a shaft-mounted rotating ring that dips into oil in the bearing housing. With either arrangement, the top ofthe bearing housing should be vented. In a vertically mounted pump with oil-lubricated bearings, a forced-feed lubrication system must often be provided.

A centrifugal pump shaft may also be supported by fluid-film bearings, such as a pivoting-shoe thrust bearing and cylindrical-journal or sleeve-type line bearings. These bearings are sometimes used because of operating conditions, including high temperatures, operating speeds, or loads, for which antifriction bearings are not suitable. Externally mounted fluid-film bearings are often lubricated and cooled with pressurized oil that is circulated through a forced-feed system by a small rotary pump. This rotary pump, which may be attached to and driven by the centrifugal pump's driver, frequently takes suction from a sump or reservoir that is part of the centrifugal pump's base. Oil discharged by the rotary pump generally passes through a cooler and filter before reaching the bearings. After passingthrough the bearings, the oil drains back to the lubricating-oil (LO) reservoir. Sight glasses may be installed in the LO-system piping to permit oil flow to or from the bearings to be observed. During operation, the proper level should always be maintained in the LO reservoir. The lubricating oil used should normally contain rust, oxidation, and foam inhibitors and should have the proper viscosity in the range of expected operating temperatures. Internal fluid-film bearings are typically submerged in and lubricated by the liquid being pumped.

In addition to the support provided by bearings, the shaft in a centrifugal pump often receives some secondary support because of the stiffness and damping generated in internal close-clearance noncontacting annular seals, such as interstage bushings, wearing rings, and balancing drums, whenever the rotor moves off-center. This is referred to as the Lomakin effect.

Sources ofloads applied to a centrifugal pump's rotating assembly include the following:

- Hydraulic axial unbalance. Hydraulic axial loads acting on a double-suction impeller are theoretically balanced. Although a net unbalance often exists because of slight differences in the flow rate entering each side of the impeller, it is usually negligible. This is not the case, however, with a single-suction impeller. When a closed single-suction impeller is used, the fluid acting on the inside surface of the back shroud adjacent to the impeller's entrance eye is essentially at the suction pressure. The outside surface of the shroud, however, may be exposed to fluid that, having already been discharged from the impeller, is at an elevated pressure. The difference in these fluid pressures results in the generation of an axial force directed towards the suction side of the impeller (fig. 12-10). An even greater unbalanced force can exist when a semiopen impeller is used because of the absence of a front shroud. The pressure on the outside of the entire back shroud is, therefore, only partially balanced by the pressure inside the impeller. As shown in figure 12-10, the pressure acting on the outside of an impeller's shrouds decreases as the radius is reduced. This reduction in pressure is caused by the rotation of the fluid located within the axial gap between the casing side walls and the rotating impeller's shrouds, which results in the conversion of a portion of the fluid's static pressure head to velocity head. The axial thrust applied to an axial-flow propeller or to an open radial- or mixed-flow impeller that does not have shrouds results from the differential pressures acting on opposite ends of the propeller's or impeller's inner hub, together with the axial component of pressure differences acting on the front and back faces of its vanes.

  The magnitude of the hydraulic axial thrust acting on a closed or semiopen single-suction impeller can be changed by varying the velocity at which the fluid behind the impeller's back shroud rotates. Radial pump-out vanes or ribs are sometimes added to the external

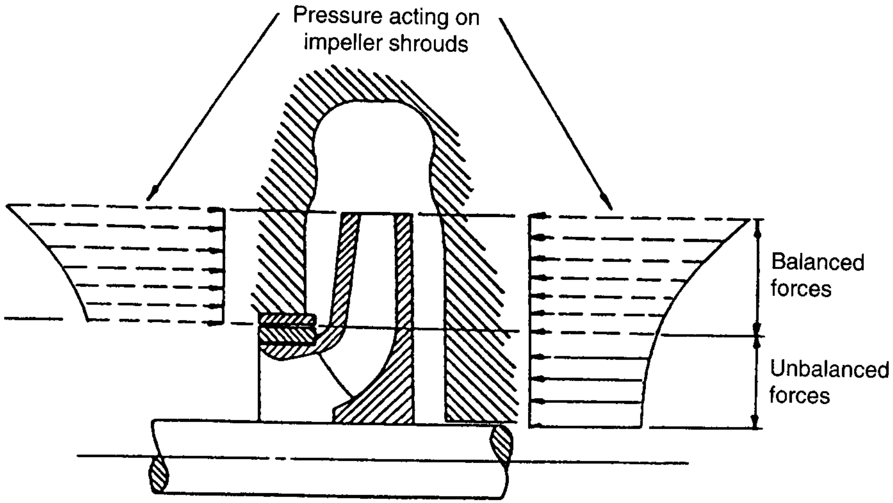


Figure 12-10. Hydraulic axial unbalance on a single-suction centrifugal-pump impeller. Courtesy Ingersoll-Dresser Pump Company.

portion of the shroud to increase the velocity at which the fluid behind the impeller rotates. Consequently, the fluid's static pressure and, therefore, the axial thrust acting towards suction are reduced. Alternatively, stationary radial vanes can be added to the casing adjacent to the impeller's back shroud to reduce the rotational velocity ofthe fluid behind the impeller. This latter arrangement, which increases the axial thrust directed towards the suction side of the impeller, is sometimes used when it is necessary to balance the axial thrust created by a high suction pressure.

The addition of multiple axial balancing holes in the impeller's back shroud, together with the installation of a back wearing ring in the casing, is another method that can be used to reduce the axial thrust acting on a closed or semiopen single-suction impeller (fig. 12-11). With this arrangement, pressure acting on the portion ofthe outer surface of the impeller's rear shroud opposite to the impeller's eye is reduced, which results in a lower differential pressure over the unbalanced area.

The net hydraulic axial thrust in a multistage pump with an even number of single-suction impellers can be reduced by orienting the impellers so that an equal number face in opposite directions (fig. 12-6). In lieu of this opposed-impeller arrangement, however, the single-suction impellers in a multistage pump may all be mounted facing in the same direction (fig. 12-8) and the axial thrust may be reduced in each stage individually with a back wearing ring and axial balancing holes. Alternatively, the hydraulic axial thrust acting on the entire rotor may be reduced with a balancing drum, a balancing disk, or a combination of these two devices.

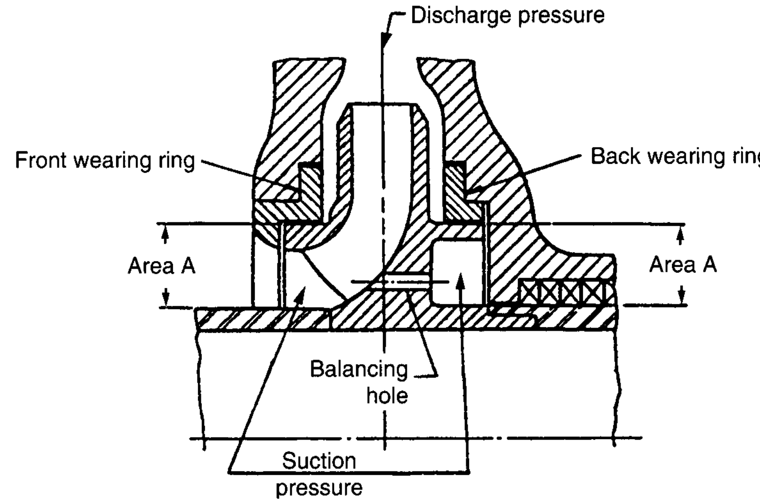


Figure 12-11. Centrifugal-pump back wearing ring and impeller with balancing holes. Courtesy Ingersoll-Dresser Pump Company.

As shown in figure 12-12a, a balancing drum consists of a drum that rotates within a close-clearance stationary sleeve. By locating the drum behind the last-stage impeller, its inboard face is exposed to fluid that is at the pump's full discharge pressure, less the pressure recovery in the last stage. As fluid passes through the close-clearance annulus formed between the drum and sleeve, its pressure drops. Consequently, because of the reduced pressure of the fluid acting on the outboard face of the drum, the net axial thrust applied to the drum is opposite in direction to the hydraulic thrust acting on the impellers. To prevent pressure from building up in the balancing chamber on the outboard side of the drum, which would reduce the flow rate through the device and, therefore, its effectiveness, this chamber is generally connected through a port to the suction area of the pump's casing.

In a balancing disk, a pressure breakdown occurs as fluid passes through a small axial gap formed between a stationary disk head and a rotating disk (fig. 12-12b). The difference in the pressures acting on the faces ofthe disk results in the generation of an axial force that is opposite in direction to the hydraulic thrust acting on the impellers. An orifice is generally installed in the recirculation port connecting the balancing chamber on the outboard side of the disk to

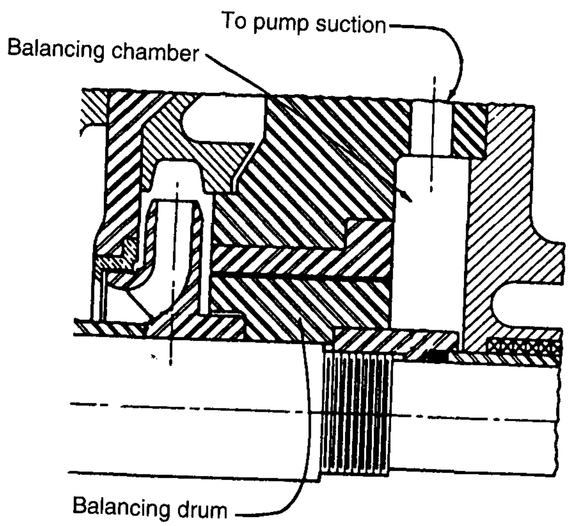


Figure 12-12a. Balancing drum.
Courtesy Ingersoll-Dresser Pump Company.

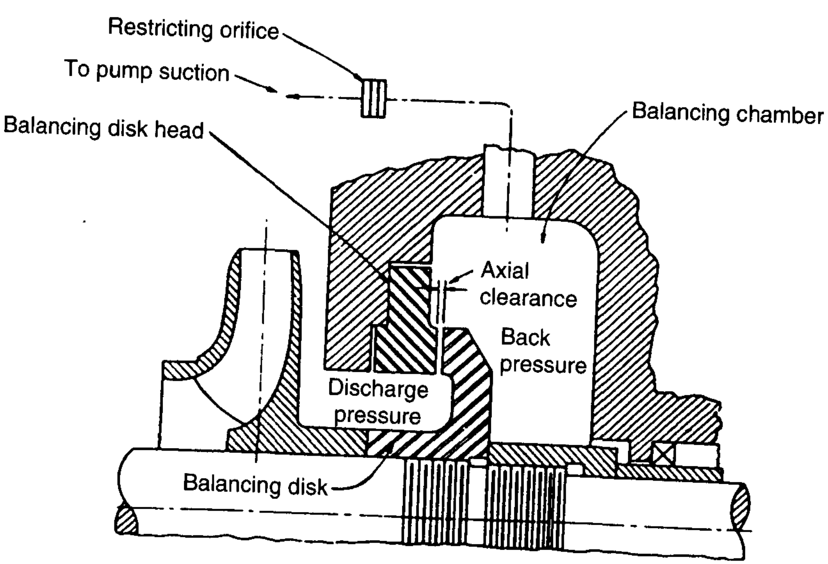


Figure 12-12b. Balancing disk.
Courtesy Ingersoll-Dresser Pump Company.

the pump's suction so that the back pressure in this chamber will be sufficient to restore the rotor to its proper position if the axial gap between the disk and the head increases. This enables the typical balancing disk to be self-adjusting in response to minor wear and to small changes in the pump's axial thrust.

A combination balancing drum and disk is a single balancing device that contains both a close-radial-clearance annulus and a close-axial-clearance gap between its rotating and stationary parts.

- Change in momentum. As fluid being pumped passes through a radial-flow impeller, its path is redirected from an axial to a radial direction. In the case of a single-suction impeller, this change in the fluid's axial velocity and, therefore, in the fluid's momentum results in the generation of an axial thrust directed towards the impeller's rear shroud. A similar axial thrust that increases with the magnitude of the radial flow component in the fluid leaving the impeller will also be generated in a mixed-flow pump. The axial thrust resulting from the change in the fluid's momentum in radial- and mixed-flow impellers is generally greater in high-capacity pumps.
- Shaft axial unbalance. In a pump with an overhung impeller, fluid entering the impeller acts against the exposed end of the pump's shaft. When a different pressure acts on the opposite end of the shaft, a piston effect is created that results in an axial force that is applied to the pump's rotor. The magnitude ofthis axial force can be determined by multiplying the differential pressure acting on the shaft (this is usually equal to the suction pressure) times the shaft's unbalanced cross-sectional area.
- Radial reaction. A typical centrifugal-pump volute is ordinarily designed so that fluid pressures and velocities around the periphery of the impeller are nearly uniform during operation at the pump's best efficiency point (BEP). However, if the flow rate being pumped is increased above the BEP capacity, the nonuniformity of the pressures and velocities within the volute results in a radial load that acts on the impeller. This radial load also increases as the flow rate being pumped is reduced below the pump's BEP capacity and rises to a maximum value during operation at shutoff(i.e., with the discharge valve closed and no flow through the pump). The changes in the magnitude of this radial force with flow rate are accompanied by changes in the direction in which it is applied. This load, which is often called radial reaction, typically results in increased shaft deflections and bearing loads and, if it is not properly accounted for in the design of a pump, can potentially result in contact between parts with close running clearances, increased bearing wear, and an eventual shaft failure. The radial reaction acting on an impeller usually increases with the total head developed by an impeller, the impeller's

outside diameter and width, and the specific gravity ofthe pumped fluid.

A double- or twin-volute casing is sometimes used to reduce radial reaction and its effects. As shown in figure 12-13, this type of casing has two volutes with cutwaters that are located approximately 180° apart. Fluid discharged from the impeller is divided between the two volutes. Because this results in the generation of two radial forces that are very close in magnitude but opposite in direction, the net radial reaction during operation off the BEP is reduced. Volute-type pumps that develop high total heads or have impellers with relatively large outside diameters are often fitted with double-volute casings. In addition, because the greatest benefit from a double volute occurs during off-BEP operation, this type of configuration is frequently used when a volute-type pump will deliver a wide range of capacities.

Radial reaction generated in concentric and modified concentric casings and in multivaned diffusers during off-BEP operation is also typically less than the radial loads generated in a single-volute casing; however, in some cases, radial reaction with these alternate configurations during operation at or near the BEP may exceed the loads developed in a comparably-sized single-volute pump. In multistage-volute casings, the angular orientation of the volutes in adjacent stages is often staggered so that the net radial load applied to the shaft is minimized.

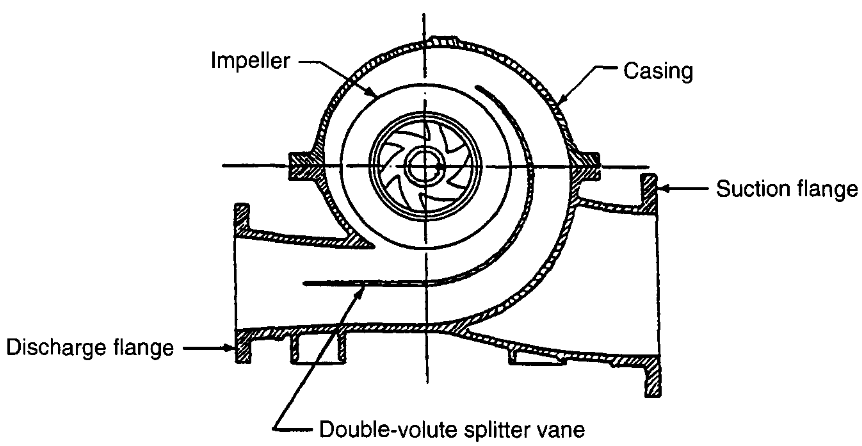
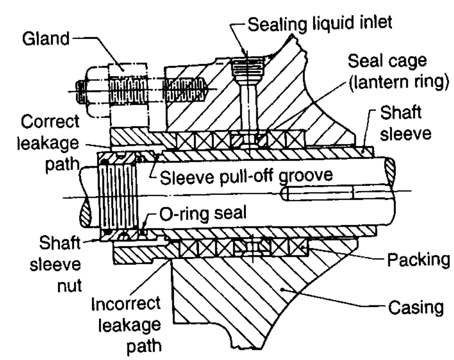


Figure 12-13. Twin-volute centrifugal-pump casing. Courtesy Ingersoll-Dresser Pump Company.

- Gravity loads due to the weight of the rotating assembly. In some cases, the orientation ofthese loads can change because ofthe list or trim of the vessel.
- Dynamic loads resulting from the motion of the vessel (including pitch and roll).
- Loads resulting from any unbalance in the pump's rotating assembly. These loads and the vibration that they frequently create can often be reduced by properly dynamically balancing the pump's rotating assembly.

Shaft sealing. Openings provided for a pump's shaft to pass through the casing must be sealed to control the leakage of the pumped fluid. When a pump has an external bearing at each end of its shaft, two shaft seals are usually required. With an overhung impeller arrangement, however, only one shaft seal is needed.

In some pumps, shaft sealing is accomplished by inserting multiple rings of flexible packing into a stuffing box (fig. 12-14). The portion of the shaft passing through the stuffing box is often protected by an outer sleeve. To increase its useful life, the sleeve may be hardened or faced with a wear-resistant coating. A leakage flow must ordinarily be maintained between the rotating sleeve and the stationary packing for cooling and lubrication. In addition to resulting in packing failure, operation with insufficient packing leakage can, in some cases, lead to thermal expansion and cracking ofthe sleeve and even to the eventual seizure of the pump's shaft. The minimum leakage rate required for any specific installation should be based on the liquid being pumped and its temperature, the peripheral



When using packing rings with a square cross section—

$$\text{cross section} \approx \frac{(\text{stuffing-box bore} - \text{shaft-sleeve O.D.})}{2}$$

$$\text{no. of rings} \approx \frac{(\text{stuffing-box depth} - \text{seal-cage width})}{\text{cross section}}$$

Prior to start-up, end of gland should typically extend into stuffing box at least 0.125 in. (3.2mm).

Figure 12-14. Packed stuffing box

speed ofthe pump shaft sleeve, and the materials used for both the sleeve and the packing. Thirty to sixty drops per minute, however, is a typical range of values.

It is important that the leakage-flow path through a stuffing box be between the packing and shaft sleeve. As shown in figure 12-14, with this flow path, fluid leaking from the stuffing box will pass between the opening in the gland and the outer surface of the shaft or sleeve. If, however, fluid is observed leaking between the gland and the stuffing box, the flow path may be between the outside of the packing and the inner wall of the stuffing box. Fluid flowing a.round the outside of the packing does not provide lubrication and can push the packing rings against the sleeve, increasing the amount of heat generated from friction; consequently, operation with this latter stuffing-box flow path should be avoided. (Leakage flow around the outside of packing rings often results from using rings that are too short in length. Replacing the packing with rings having the proper length will frequently eliminate the problem.)

The leakage rate through a packed stuffing box is controlled by adjusting the axial position of the gland plate that holds the packing in place. Tightening the gland compresses the packing rings and reduces the amount of leakage, while loosening the gland enables the leakage rate to increase. When the pressure of fluid at the base of the stuffing box is too low to result in an adequate packing leakage flow rate, high-pressure liquid is usually injected into the stuffing box through a lantern ring or seal cage that is sandwiched between two of the intermediate rings of packing. This sealing liquid, which can be supplied from the discharge area of the casing through either an internal port or an external line or, when necessary, from a separate external source, helps to lubricate the packing and to prevent air from being drawn into the pump through the stuffing box. A pump furnished for an application involving dirty fluids may also be fitted with a lantern ring to permit clean liquid to be injected into the stuffing box and flush contaminants away from the packing. Alternatively, when it is impractical to provide clean high-pressure liquid to the stuffing box, grease is sometimes injected through the lantern ring connection.

The individual rings included in a set of packing often have a square or rectangular cross section and can be cut with diagonal (scarf-cut) joints (usually at 45° angles with respect to the shaft's axis) that overlap after installation or with square (butt-cut) joints. The material used for packing varies with the application. Although asbestos was often used in the past, due to its widespread prohibition because of health and environmental concerns, many packings now used are composed of graphite and polytetrafluoroethylene (PTFE) compounds. In addition, the end rings in a set of packing sometimes include metallic compounds to make them harder and more resistant to extrusion than the inner packing rings. The potential for extrusion is also sometimes reduced by providing a close radial clearance

between the rotating shaft sleeve and a replaceable throat bushing that is installed at the base of the stuffing box.

When packing leakage rates can no longer be kept within acceptable limits by tightening the gland, the entire set of packing should be replaced. Mter the gland has been removed from the pump, the old packing rings should be removed from the stuffing box with a suitable packing puller. If the shaft sleeve is badly worn, it should be replaced. (If the shaft sleeve has a gasket or a-ring, which is frequently provided to prevent leakage between the sleeve and the shaft, it should be replaced along with the sleeve.) Replacement packing rings should be of the proper size and material and can be either precut and preformed or cut from a coil that is wrapped around a mandrel having an outside diameter equal to that of the shaft sleeve. One ring should be installed at a time, and each can be pushed down into the stuffing box with the gland. When the ends of a preformed packing ring are being separated to fit around the shaft, they should be twisted sideways (i.e., in the form of a twisted "s") rather than spread straight apart. It is important that each packing ring be seated squarely. The splits of successive rings should be placed 90° apart. If a lantern ring is used, it should be installed in the proper location: adjacent to the injection-port opening inside the stuffing box. Mter the last ring of packing has been installed, the gland should be replaced. (In some cases, the last ring of packing may not fit in the stuffing box until after the pump has been run and the packing has been seated.) When nuts are used to hold the gland in place, they should be replaced and tightened evenly until they are just finger tight, and then they should be loosened slightly. When starting the pump for the first time with the new packing, no gland adjustments should be made for the first twenty to thirty minutes of operation. If after that period the leakage is excessive, the gland nuts should be tightened evenly one flat (¥6 of a turn) at a time, waiting at least ten to fifteen minutes between adjustments, until the desired leakage rate is achieved. If leakage stops or the packing overheats, the pump should be stopped and the gland should be loosened. Once the packing overheats, it may be necessary to repack the pump with a new set of packing.

As an alternative to a packed stuffing box, many pumps are fitted with mechanical axial-face shaft seals (fig. 12-15). The primary sealing surfaces in a mechanical seal are the highly polished faces of a rotating ring and a nonrotating ring that are separated by a thin film offluid. Frequently, one ofthe mating primary seal rings is made from a soft material with some inherent self-lubricity, such as carbon, while the second ring is made from a hard material, such as aluminum oxide, tungsten carbide,or silicon carbide. The typical mechanical seal also includes secondary seals that controlleakage between the inside diameter of the rotating ring and the shaft and between the nonrotating ring and the gland. To permit the faces of the rotating and nonrotating seal rings to remain parallel with limited radial

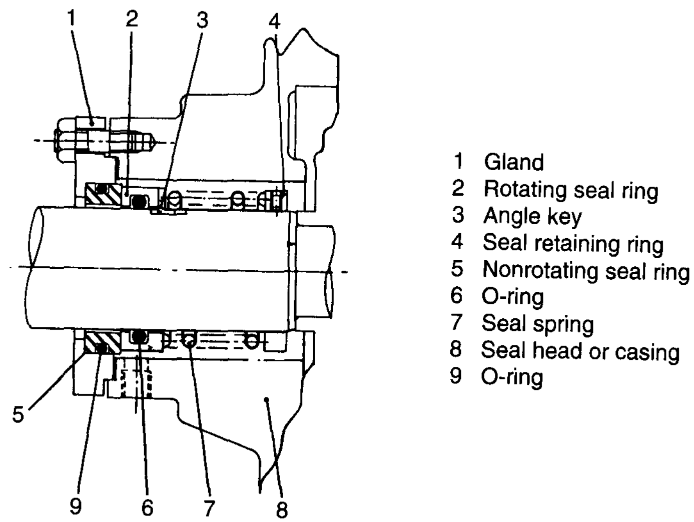


Figure 12-15. Mechanical seal

or axial shaft movement, the rotating ring is often flexibly mounted on the pump's shaft with a single coil spring that fits over the shaft, multiple smaller coil springs positioned around the circumference ofthe seal's rotor, or either a metallic or an elastomeric bellows. Alternatively, in some high-speed applications, it is the mechanical seal's nonrotating seal ring that is spring loaded.

Flushing liquid to cool and lubricate a pump's mechanical seal is often provided through a recirculation line that is connected to the discharge side of the casing. High-pressure liquid passes through this line and enters the seal area through a drilled port in the pump casing or gland. When the liquid being pumped contains abrasive particles that could damage the faces of the mechanical seal, a cyclone or abrasive separator is sometimes installed in the recirculation line. The effectiveness of a cyclone separator ordinarily increases with the differential pressure across it and the specific weights of the various solids entrained in the pumped fluid. In addition, for solids within the pump's casing to be flushed away from the seal, the pressure of clean liquid injected into the gland must be greater than the pressure of the liquid that surrounds the seal. Consequently, if the total head developed by a pump is low or the mechanical seal is surrounded by liquid that is at or near the pump's discharge pressure, providing clean high-pressure flushing liquid from an external source or the use ofa double hard-face mechanical seal in which both the rotating and nonrotating seal faces are made from abrasion-resistant materials, such as tungsten or silicon carbide, may be considered. If solid particles will be present in the pumped liq-

uid, the use of a mechanical seal design that is resistant to clogging is also recommended.

To enable pump operation to continue in the event ofa mechanical-seal failure, the glands used with some seals have a built-in auxiliary or emergency stuffing box. If a mechanical-seal failure occurs, two or more rings of packing can be installed in the auxiliary stuffing box to control the leakage around the shaft until the faulty seal is replaced. However, because the emergency packing will operate dry ifused with a properly functioning mechanical seal, it should be installed only after a seal failure has occurred.

To simplify seal replacement by eliminating the need to pass seal parts over the end of a pump's shaft, split mechanical seals have been developed. The parts used with this type of seal are split along axial joints; therefore, they can be replaced without disassembling or removing other parts of the pump.

When high-temperature fluids are pumped, cool liquid supplied from an external source is often circulated through a jacket that surrounds the pump's sealing area. Alternatively, by installing a heat exchanger in the seal-flush line, high-temperature liquid being recirculated from the pump's casing can be cooled prior to being injected into the stuffing box or gland.

In special applications, twin mechanical seals mounted face-to-face (opposed seals), back-to-back (double seals), or in tandem are sometimes used. With these arrangements, a barrier fluid is often circulated between the two seals.

When zero leakage is required, the use of even a special mechanical shaft seal may not be suitable. For these applications, two alternate types of sealless pumps are available: magnetically coupled pumps and canned-motor pumps. In a magnetically coupled pump, torque developed by the driver is transmitted to the pump's shaft, which is completely enclosed within the casing, through a magnetic coupling. The two halves ofthis coupling are not in physical contact; therefore, the need to provide an opening for the shaft in the pump's casing is eliminated. The pump and electric motor that compose a canned-motor pump form a single sealed assembly. The typical canned-motor pump is fitted with an end-suction impeller that is mounted directly on the shaft of the driving motor. Frequently, a portion of the fluid discharged from the impeller is circulated through the motor for cooling and to provide lubrication for the motor's bearings. The motor's windings are usually isolated from this liquid by a corrosion-resistant liner.

Couplings. In all but close-coupled pumps, torque developed by a pump's driver is transmitted to the pump's shaft through a coupling. A flexible coupling, which can tolerate some relative motion between its driving and driven halves, is frequently used whenever the pump and driver shafts are fitted with separate thrust bearings.

A gear-type flexible coupling has external gear teeth either on one or both ofits hubs that mesh with internal gear teeth on an external sleeve. A flex-flex gear coupling has teeth on both of its hubs, while a flex-rigid gear coupling has gear teeth on only one hub (its opposite hub is rigidly attached to the outer sleeve through a flanged joint). A spring-grid or steelflex coupling has two metal hubs that are joined by a flexible steel grid inserted into slots located around the periphery of each hub. Gear and spring-grid couplings must be lubricated for proper operation.

Two commonly used types of nonlubricated flexible couplings are the disk coupling and the diaphragm coupling. With a disk coupling, torque is transmitted through a stack of thin metallic rings, or disks, that are alternately attached around a common bolt-circle located near their periphery to the driving and the driven hubs. A typical diaphragm coupling has one or more metallic membranes, or diaphragms, that are bolted near their inside diameter to the driven hub and near their outside diameter to the driving hub.

Flexible couplings that have elastomeric elements include the pin-and-bushing coupling, which consists of one flange fitted with pins that slide into holes lined with flexible bushings in the opposite flange; the elastomericjaw-type coupling, which consists of two metal hubs having meshing jaw-like teeth that are separated and cushioned by an elastomeric center member, referred to as a spider; and the elastomeric sleeve-type coupling, which has two metal flanges that are joined by a common removable flexible sleeve.

When a pump's shaft is supported axially by the thrust bearing in the driver, the pump and driver shafts must typically be connected through a rigid-type coupling. With a flanged-rigid coupling, torque is transmitted from a hub mounted on the driver's shaft to a mating hub on the pump's shaft through the bolts that join the two hubs together.

A spacer coupling has a removable spacer located between its driving and its driven halves. The insert permits the gap separating the adjacent ends of the pump shaft and the driver shaft to be increased, which can eliminate the need to disturb the driver when pump parts, such as inboard bearings and mechanical seals, are replaced. Additionally, the use of a spacer coupling with a radially split casing pump can enable the pump's rotating assembly to be backed out of the casing with the driver in place.

Centrifugal Pump Installation

Foundation. A pump's foundation should be sufficiently rigid to prevent excessive relative movement between the pump and its driver. In addition, the foundation must be strong enough to support static loads resulting from the wet weight of the pump (i.e., the weight of the pump when it is filled with the pumped liquid), together with the weight of the

dJ'iver and any attachments, and loads resulting from the motion of the vessel. Soft feet, or mounting feet that are not supporting their proper share of the load, should be avoided. To check for a soft foot, after all of the mounting bolts have been tightened, the mounting bolt of the foot in question should be loosened and, with a dial indicator initially set to zero, the amount that the foot rises should be measured. If the foot moves more than 0.002 in. (0.051 mm), a shim with a thickness equal to the indicator reading should be inserted under the foot. The mounting bolt should then be retightened, and the foot should be rechecked. This procedure can then be repeated for each mounting foot.

Piping. Suction piping should be as short and direct as practicable with a minimum of valves and fittings, and it should be air- and liquid-tight. It is usually desirable that straight pipe having a length equal to at least five to ten pipe diameters be located directly upstream ofthe pump's suction connection. Furthermore, high spots where air pockets can form should be eliminated. If the pumped liquid will contain foreign objects that could clog or damage the pump, a strainer with a net flow area equal to three to five times the suction pipe area should be installed in the suction line and cleaned regularly. When practicable, both suction and discharge piping should be one size larger than the corresponding pump connections. (The use of piping that is smaller than the pump connections should ordinarily be avoided.) If a reducer is used in the suction piping of a pump that operates with a suction lift (i.e., with the liquid source below the inlet to the pump), it should be ofthe eccentric type installed with the straight side on top. In addition, suction or discharge reducers that are used should always be located on the pump side of any elbows installed in the piping. Also, any elbows that are used should be of the long-radius type. When used with a pump that has a double-suction impeller, a suction elbow should be oriented with its bend in a plane that is perpendicular to the axis of the pump's shaft. Stop valves are frequently installed in both the suction and discharge lines to permit the pump to be isolated from the system for inspection and maintenance. Whenever possible, a check valve should be installed between the pump discharge and the discharge stop valve to prevent the back flow of fluid through the pump. To permit pump pressures to be measured, suction and discharge pressure gauges should be installed on the pump side of any valves. All piping should be independently supported and properly aligned so that loads imposed on the pump's casing will be minimized and will be within the pump manufacturer's allowable values. Additionally, piping should always be thoroughly flushed and cleaned prior to pump installation.

Coupling alignment. Although it is actually the shafts of the pump and driver that must be aligned, this process is generally referred to as coupling alignment. When a rigid coupling is used to connect the shaft of a pump to the shaft of its driver, precision alignment between the two shafts is critical. A flexible coupling can often compensate for slight changes in the pump-to-driver alignment that may occur during normal operation. However, a flexible coupling is not designed or intended to correct for excessive amounts of continuous misalignment, which can result in the transmission of excessive loads to the pump's shafting and bearings and can increase vibration. Consequently, even when a flexible coupling is used, both angular and parallel misalignment between the centerlines of the pump and driver shafts (fig. 12-16a) should be kept within limits allowed by the pump manufacturer. (These limits will normally be less than misalignment limits allowed by the coupling manufacturer.) In addition, the proper gap should be maintained between adjacent ends of the pump and driver shafts. Prior to aligning a pump and driver, foundation fasteners should be properly tightened and all required piping should be connected to the pump (and to the driver, when applicable). Also, ordinarily, the pump and driver coupling halves should not be joined. Misalignment can be measured with a straightedge and feeler gauges, micrometers, electronic gauges, optical devices, lasers, or dial indicators.

Excessive pump-to-driver shaft misalignment can often be reduced to acceptable values by placing nonferrous shims having the proper thicknesses under the appropriate driver feet to reduce misalignment in a vertical plane, or by moving the driver horizontally on its base to reduce misalignment in a horizontal plane. Jacking bolts are sometimes provided to help move the driver during alignment. (In the case of a turbine-driven pump, it is often easier to move the pump than the driver during alignment. In addition, if a gearbox is located between the pump and driver, both the pump and the driver may be moved while being aligned to the

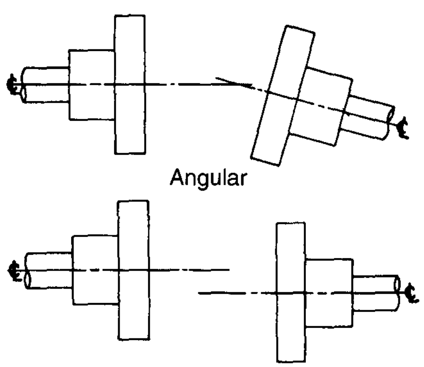


Figure 12-16a. Angular and parallel misalignment

gearbox.) When the temperature of the pump or driver changes during operation, effects from thermal expansion or contraction should be compensated for during the initial alignment. After the unit is operating and temperatures have stabilized, the driver should be secured and the alignment should be rechecked while the pump and driver are still at their normal operating temperatures. Any necessary corrections should then be made. Following the alignment, the coupling-end pump (in the case of a horizontal unit) and driver feet are frequently doweled to the base.

One of the most commonly used methods to measure pump-to-driver shaft misalignment is the face-and-rim method. To measure angular misalignment using the face-and-rim method, a dial indicator is typically mounted on either the pump shaft or the pump half ofthe coupling and positioned with its stem on the front or back face of the driver's coupling hub (fig. 12-16b). Then, when possible, both shafts and the dial indicator should be slowly rotated together in the direction of normal shaft rotation, and four indicator readings should be taken at 90° intervals. (By rotating both the pump and driver shafts together, shaft runout and coupling irregularities will not affect the indicator readings. If only one shaft can be rotated, the dial indicator should be mounted on the rotated shaft or coupling hub.) It is important that neither shaft move axially (float) while being turned.

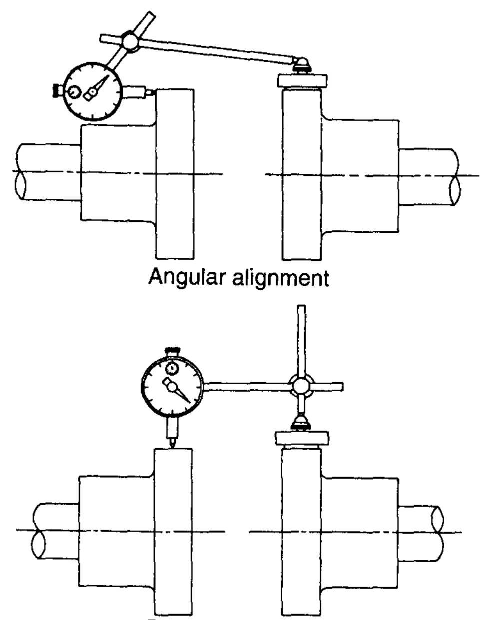


Figure 12-16b. Coupling alignment, face-and-rim method with a dial indicator. Courtesy Ingersoll-Dresser Pump Company.

(To ensure that shaft float does not adversely affect face readings, prior to taking each reading, the shafts in some pumps and drivers are both pushed as far apart as the bearings in each machine will allow.) For a horizontal shaft, the four readings are ordinarily taken with the dial indicator positioned at 0° (top), 90° (side), 180° (bottom), and 270° (opposite side). When measurements are made at these locations, the algebraic difference between the indicator readings at 0° and 180° is the total indicator reading (TIR) in a vertical plane, and the algebraic difference between the indicator readings at 90° and 270° is the TIR in a horizontal plane. (In the case of a vertically mounted pump, the two TIRs represent misalignment in two perpendicular vertical planes.) To simplify the calculation ofTIR, the indicator's dial is generally set to read zero in the initial or 0° position. Mter any excessive angular misalignment is reduced to an acceptable value, parallel misalignment can be determined by positioning the dial indicator's stem on the rim of the driver's coupling hub and repeating the aforementioned procedure. When taking rim readings, it is important that the centerline ofthe indicator stem be perpendicular to the surface of the coupling hub's rim (i.e., that an extension of the stem's centerline passes through the centerline of the shaft). The rim TIR in each plane can be divided by two to determine the actual pump-to-driver shaft-centerline offset in that plane. Mter correcting any excessive parallel misalignment, the angular alignment should be rechecked and corrected if necessary. In some cases, both face and rim readings may be taken simultaneously by using two dial indicators.

If a dial indicator is not available, alignment following the face-and -rim method may be performed with feeler gauges and a straightedge (fig. 12-16c). Mter the initial measurements are taken adjacent to a reference point on the coupling, both shafts should be rotated together in quarter-turn increments. While in each position, additional sets of measurements should be taken adjacent to the same coupling reference mark.

When the reverse-indicator method, which is another way to measure misalignment, is used, one dial indicator is generally mounted on the pump shaft or coupling hub and a second indicator is mounted on the driver shaft or coupling hub. The two indicator stems are positioned against the rims ofthe opposite coupling hubs 180° apart. Both shafts are then rotated 360° while readings are taken from both dial indicators at 90° intervals. This data can then be used to graphically determine both parallel and angular misalignment. Because only rim readings are taken when using the reverse-indicator method, it is preferred over the face-and-rim method ifthe float or end play of either the pump shaft or the driver shaft prevents accurate face readings from being taken.

In the case of a horizontal pump, face and rim TIRs measured in vertical planes (i.e., the TIRs based on 0° and 180° readings) should be corrected to account for any sagging of the indicator-support bracket. (Bracket sag does

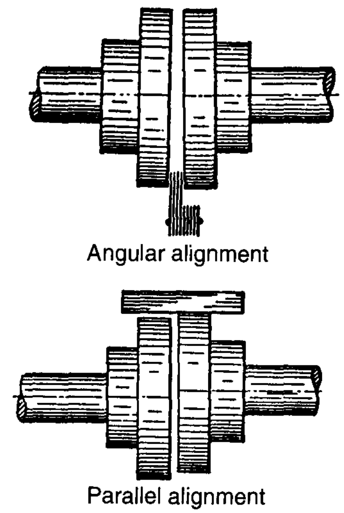


Figure 12-16c. Coupling alignment, face-and-rim method with feeler gauges and a straightedge.
Courtesy Ingersoll-Dresser Pump Company.

not affect indicator readings taken while aligning a vertical pump.) Bracket sag when taking rim readings can generally be determined by mounting the dial indicator and its bracket on the upper surface of a horizontal piece of rigid pipe having a diameter equal to that of the shaft or coupling hub on which the indicator will be mounted during alignment and, with the indicator stem resting against the top ofthe pipe, setting the indicator dial to read zero. The pipe should then be rotated 180°. The dial indicator reading in this position should be subtracted algebraically (i.e., a negative value should be added) from the rim reading taken in the bottom or 180° position during coupling alignment. Bracket sag when taking face readings can be determined by positioning the dial indicator stem against the face of a flange on the same piece of pipe at a radius equal to that at which the coupling face readings will be taken during alignment. (The face ofthe flange should be perpendicular to the pipe's centerline and should be oriented in the same direction with respect to the bracket support as the coupling face that will be used during alignment.) The indicator dial should again be set to read zero in the 0° or top position, and the pipe should be rotated 180°. The indicator reading in the 180° position should be subtracted algebraically from the face reading taken in the bottom or 180° position during coupling alignment.

Permissible values of misalignment vary with the point and method of measurement, operating speed, and the design of the pump, driver, and coupling. However, when specified in terms of TIR, parallel and angular alignment tolerances with a flexible coupling frequently do not exceed 3 mils (0.075 mm) TIR, where 1 mil = 0.001 in. = 0.0254 mm. Permissible misalignment with rigid couplings is often in the range of1 to 2 mils (0.025 to 0.050 mm) TIR. Alternatively, parallel pump-to-driver alignment tolerances may be specified in terms of allowable shaft-centerline offset, which would be equal to one half ofthe corresponding allowable TIR. Also, the angular alignment tolerances may be specified in terms of allowable shaft-centerline offset per unit of coupling hub separation in mils (mm) of offset per in. (mm) of separation, where 1 mil/in. (0.001 mm/mm) = 0.0573° of angular misalignment. A typical value for acceptable angular misalignment in terms of offset per unit of separation when a flexible coupling is used is 0.5 mils/in. (0.0005 mm/mm).

The correct direction' of driver rotation should be verified prior to connecting the pump and driver coupling halves. (When the driver is a three-phase ac electric motor, the direction of rotation can be changed by reversing any two of the three electrical cable connections for each set of the motor's windings.) In addition, regardless ofthe coupling type used, to protect personnel, a stationary guard should be installed around the coupling after the alignment is completed. The coupling guard should be in place whenever the pump is operating.

Centrifugal Pump Performance and Operation

Pump head. The maximum theoretical total head that could be developed by an ideal centrifugal pump, referred to as the Euler head, can be determined from fluid velocity vectors at the inlet and outlet of an impeller. Using subscripts 1 and 2 to refer to conditions at the impeller inlet and outlet, respectively, the following is one form of the Euler equation:

$$H_e = \frac{1}{C_3(2g_c)}\left[\left(c_2^2 - c_1^2\right) + \left(u_2^2 - u_1^2\right) + \left(w_1^2 - w_2^2\right)\right] \qquad (12.4)$$

where, referring to figure 12-17

- He = Euler head, ft-lbfflbm (kJ/kg)
- C3 = 1 when using the USeS units shown (1,000 for the metric units)
- gc = gravitational constant, 32.2 ft-lbm/lbf-s^2 (gc is not required in the metric system and should be set equal to 1 when using metric units of measurement)
- c = the pumped liquid's absolute velocity with respect to the pump casing, ftls (m/s)
- u = the peripheral or tangential speed of the impeller at a given radius, ft/s (m/s)
- w = the pumped liquid's relative velocity with respect to the rotating impeller, ftls (m/s)

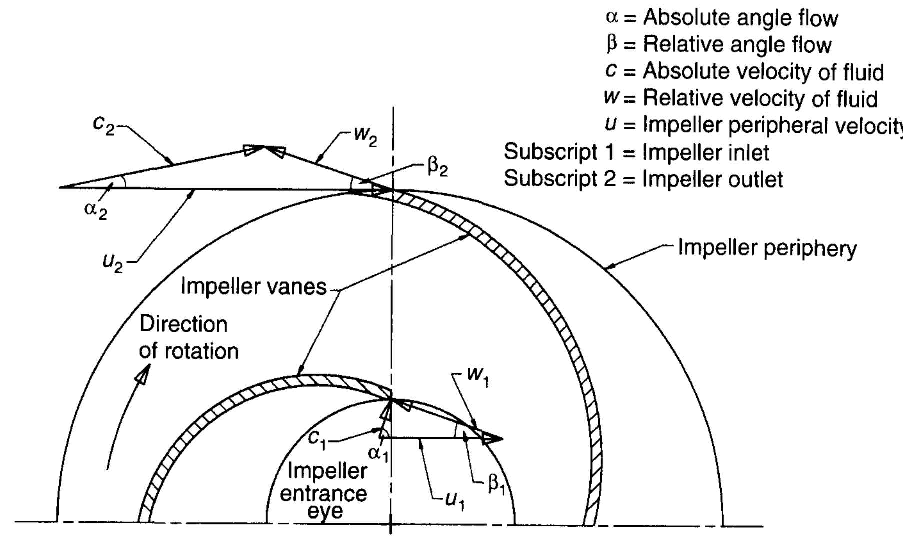


Figure 12-17. Radial-flow centrifugal pump flow-velocity vectors

The term within the first set of parentheses in the above equation represents the increase in the kinetic energy or dynamic head of the liquid being pumped because of its acceleration from the inlet to the outlet of the impeller. The term within the second set of parentheses in equation 12.4 can be thought of as representing an increase in static pressure due to the centrifugal effect that results from the increase in the pumped liquid's radius of rotation and peripheral speed as it passes through the impeller. The third term can be thought of as representing an additional increase in static pressure due to any diffusion (reduction in relative velocity) of the liquid as it passes through the impeller's flow channels. If when using USeS units of measurement, g (32.2 ft/s^2) is substituted for gc in equation 12.4, values of the Euler head will be in ft, which is the same unit of measurement used for total pump head in equations 12.1 and 12.2.

The plot of Euler head (He) versus capacity is a straight line. In addition, with no pre rotation in the inlet flow, He remains constant as the capacity increases when an impeller has radial vanes (angle $_2$ in figure 12-17 = 90°), and He drops as the capacity increases when an impeller has backward-curved vanes ($_2$ < 90°). Most centrifugal pump impellers have either radial or backward-curved vanes. Because of velocity gradients across an impeller's flow channels, together with friction, turbulence, and hydraulic shock losses within the impeller and casing, the total head actually developed by a centrifugal pump is always less than the Euler head (fig. 12-18).

Power. The power required to drive a pump shaft can be calculated as follows:

$$P_P = \frac{HQsg}{C_4 \eta_P} \tag{12.5}$$

where

P_P = power required to drive the pump, hp (kW)
H = total head developed by the pump, ft (m)
Q = total capacity delivered by the pump, U.S. gpm (m^3/hr)
C_4 = 3,960 when using the USCS units shown (367.6 for the metric units)
η_P = pump efficiency, %/100

The pump efficiency, η_P, accounts for the effects of hydraulic flow losses in a pump's casing and impellers; mechanical losses resulting from friction in bearings, seals, and packing; disk friction between external surfaces of the impeller shrouds and the fluid within the pump's casing; and internal leakage or recirculation flow through clearances adjacent to wearing rings, interstage bushings, and internal balancing devices. Because efficiencies of the transmission train (i.e., reduction gears external to the pump, fluid couplings, etc.) and driver ordinarily are not included in the pump efficiency, the

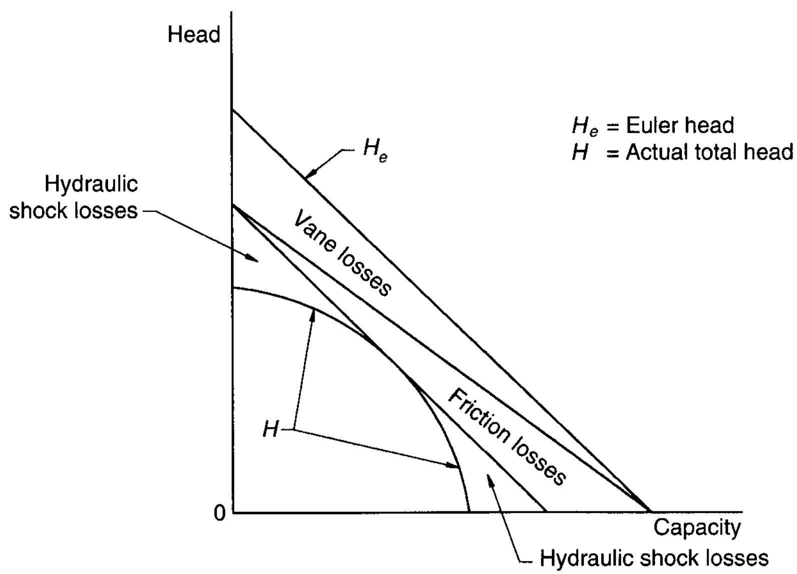


Figure 12-18. Theoretical and actual pump head developed by a centrifugal pump fitted with an impeller having backward-curved vanes

total power that must be supplied to a pump's driver exceeds the power calculated using equation 12.5. Typical values of pump efficiency plotted versus specific speed are shown in figure 12-19.

EXAMPLE 12-3: A centrifugal pump delivers a capacity of 450 m^3/hr and develops a total head of 30 m while pumping seawater with a specific gravity of 1.03. The pump's efficiency at this operating point is 85 percent. What is the power required to drive the pump?

Solution: Using equation 12.5

$$P_P = \frac{\left(450 \frac{m^3}{hr}\right)(30\ m)1.03}{367.6(.85)} = 44.5\ kW$$

Single-pump performance. The capacity delivered by a centrifugal pump operating in a system is determined by the point at which the curve showing the total head developed by the pump versus the capacity of fluid delivered, referred to as the pump's head-capacity or H-Q curve, intersects

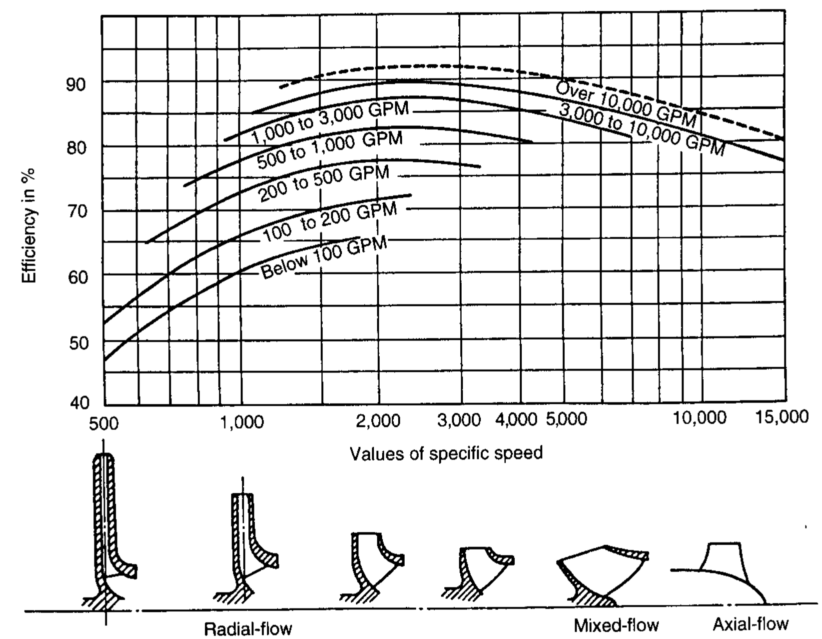


Figure 12-19. Specific speed (calculated using the USCS units of measurement given with equation 12.3) versus efficiency. Courtesy Ingersoll-Dresser Pump Company.

the curve representing total system head versus capacity (point A on fig. 12-20). A pump furnished for a new application is often sized to deliver the specified capacity and develop the specified total head at a point on its H-Q curve that is at or slightly to the left of the pump's best efficiency point (BEP), which is the point on the H-Q curve corresponding to the pump's maximum efficiency (point D during full-speed operation on fig. 12-20). However, operation at the rated conditions of service will only occur when the curve of system head versus capacity passes through the same point. In addition, the head curves for many systems change during operation. For example, throttling a valve in the discharge line of a system will increase friction losses, cause the system-head curve to shift to the left, and change the point of intersection between the system-head curve and the pump's H-Q curve (point C on fig. 12-20). A change in the liquid level within a tank at either end of a system will also alter the system-head curve. Consequently, throughout its life, the typical centrifugal pump actually delivers capacities at various points along its H-Q curve. A pump furnished for any application should, therefore, be suitable for operation over the entire range of expected performance.

Typical performance curves showing normalized values of total head developed, pump efficiency, and power required that are plotted versus ca-

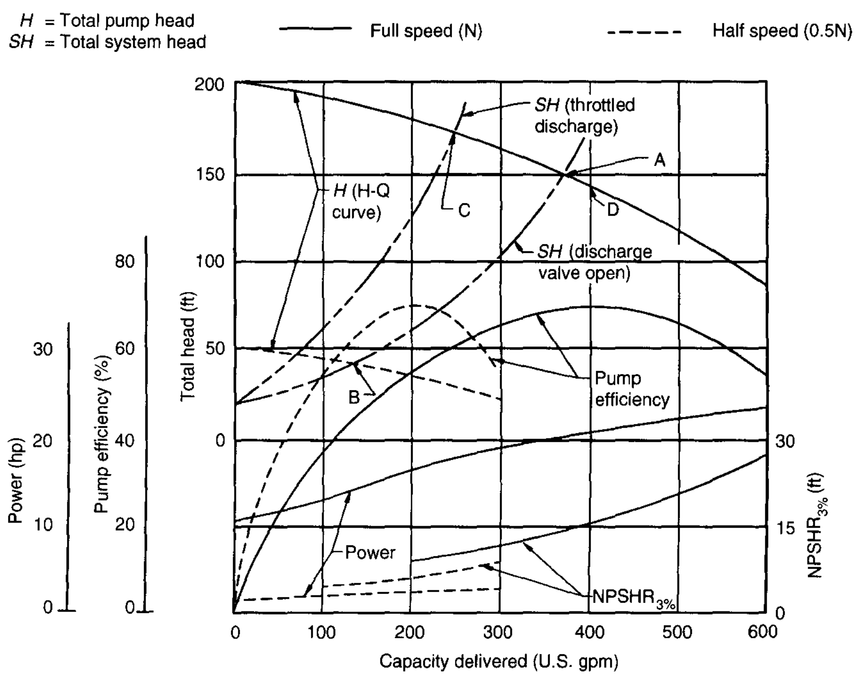


Figure 12-20. Typical centrifugal-pump performance curves

pacity delivered by centrifugal pumps with various values of specific speed are illustrated in figure 12-21. As shown, the H-Q curve typically becomes steeper as the specific speed increases. (The H-Q curve also tends to become steeper as the impeller-vane discharge angle, $_2$ in figure 12-17, is reduced.) In addition, although the power required by a low-specific-speed pump often increases with the capacity delivered, the power required by a pump with a high specific speed usually increases as the capacity delivered by the pump is reduced. An unexpected overload during a change in the point of operation along a centrifugal pump's H-Q curve can be avoided by using a driver that is rated to deliver the maximum power required anywhere within the expected range of performance.

Affinity laws. The performance of a pump can be altered by changing the pump's operating speed and, in the case of a radial- or mixed-flow pump, the outside diameter of any of the pump's impellers. If effects on pump efficiency, which are generally small, are ignored, the capacity delivered (Q), total head developed (H), and power required (P_P) by a single-stage centrifugal pump being operated at a new speed or with an impeller having a new outside diameter can be predicted using values from the pump's original performance curves and the following affinity laws (also referred to as the laws of similitude):

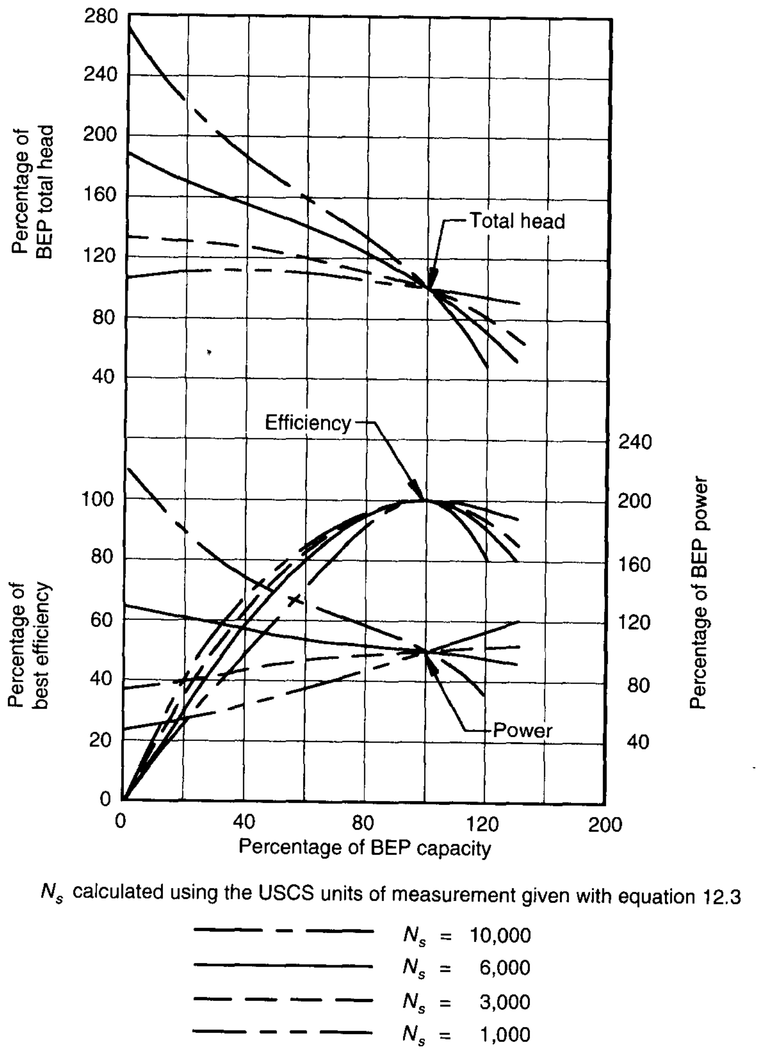
$$Q_2 = Q_1\left(\frac{N_2}{N_1}\right)\left(\frac{d_2}{d_1}\right) \tag{12.6a}$$

$$H_2 = H_1\left(\frac{N_2}{N_1}\right)^2\left(\frac{d_2}{d_1}\right)^2 \tag{12.6b}$$

$$P_{P,2} = P_{P,1}\left(\frac{N_2}{N_1}\right)^3\left(\frac{d_2}{d_1}\right)^3 \tag{12.6c}$$

where

N = pump operating speed, rpm
d = outside diameter of the impeller (an average value should be used when the outside diameter varies across the width of the impeller's waterways), in. (mm)
subscript 1 refers to conditions with the original speed and impeller diameter
subscript 2 refers to corresponding conditions with the new speed and impeller diameter



BEP = best efficiency point

Slopes of performance curves for an actual pump can vary from those shown based on the design of the pump (e.g., the number of impeller vanes, vane angles, etc.) and of the system and on the characteristics of the fluid being pumped.

Figure 12-21. Representative constant-speed performance characteristics for centrifugal pumps with different specific speeds (N_s)

The pump's new operating point is established by the intersection of the new H-Q curve, plotted using the results of equations 12.6a and 12.6b, with the system curve (point B on figure 12-20).

Equations 12.6a, 12.6b, and 12.6c can also be used to predict the effect that a speed change will have on a multistage centrifugal pump. However, they can only be used to predict the effect of an impeller diameter change on a multistage pump when all of the pump's impellers have the same original diameter and the same new diameter. In addition, these equations should only be used when both the original and the new performance data for a centrifugal pump apply to operation with low-viscosity liquids (i.e., liquids having a viscosity of approximately 32 SSU or 1 cSt at the pumping temperature). Furthermore, if the specific gravity of the liquid being pumped at the new speed or with the new impeller diameter is different from the specific gravity used to develop the pump's original performance data, the right-hand side of equation 12.6c must be multiplied by a correction factor equal to the specific gravity of the new liquid divided by the specific gravity of the original liquid (sg_2/sg_1) to account for the effect that the change in the pumped liquid's specific gravity will have on the pump's power requirement. Effects from the changes in an impeller's vane angles, vane lengths, and channel areas that occur when the impeller's outside diameter is reduced are not reflected in equations 12.6a, 12.6b, and 12.6c; consequently, these equations should only be used to predict radial- or mixed-flow pump performance with a new impeller diameter when the change in d is small (e.g., 10 percent or less). Also, the change in the diameter of a mixed-flow impeller will generally have a greater effect on pump performance than the effect predicted by the results of equations 12.6a, 12.6b, and 12.6c.

In addition to the effect that changes in impeller diameter have on pump performance, the performance of a centrifugal pump can also be modified by altering the shape of the vanes in the pump's impellers. For example, removing material from the undersides of impeller vanes at the impeller's outlet, referred to as underfiling, often results in an increase in the total head developed and, in some cases, the power required by a pump. Although the effect that reshaping an impeller's vanes will have on pump performance is often difficult to predict quantitatively, this technique is sometimes effective when an increase in the total head developed by an existing pump is required.

EXAMPLE 12-4: The solid curves shown in figure 12-20 are for the performance of a centrifugal pump being operated at 3,550 rpm. Determine the total head developed, capacity delivered, pump efficiency, and power required by this pump while being operated at a speed of 1,775 rpm. Assume that, in both cases, the pump will handle freshwater at a temperature of 68°F.

Solution: Using equations 12.6a, 12.6b, and 12.6c, the following results were calculated:

3,550 rpm				1,775 rpm			
Qi (U.S. gpm)	Hi (ft)	llPi (%/100)	PPi (hp)	Q2 (U.S. gpm)	H2 (ft)	llP2 (%/100)	PP2 (hp)
0	200	0	10.5	0	50	0	1.3
200	180	0.55	16.5	100	45	0.55	2.1
400	145	0.70	20.9	200	36.3	0.70	2.6
600	88	0.55	24.2	300	22	0.55	3.0

Pump performance at 1,775 rpm is plotted using dashed curves on figure 12-20.

Net positive suction head. The suction condition for a centrifugal pump is often expressed in terms of net positive suction head (NPSH), which is the amount by which the total absolute suction pressure (including atmospheric pressure) of the fluid entering the pump exceeds the fluid's true vapor pressure (also expressed in terms of absolute pressure) at the pumping temperature. When referring to NPSH, it is necessary to differentiate between the amount available to a pump (NPSHA) and the amount that the pump requires (NPSHR). If the net positive suction head available to a pump is less than the amount required (NPSHA is less than NPSHR), the pressure ofthe fluid entering the impeller will drop below the fluid's vapor pressure and boiling will occur. The vapor bubbles that form interfere with the normal flow ofliquid through the pump and cause the total head developed by the pump and the pump efficiency to drop. In addition, as these bubbles travel into higher pressure regions of the impeller, they collapse. This formation and subsequent collapse of vapor bubbles is called cavitation. The resulting uneven flow through the pump, combined with the impingement against the impeller's waterway boundaries of the liquid that fills the voids created when the vapor bubbles collapse, can increase noise and vibration and produce severe pitting of the impeller. In some cases, cavitation can even lead to the eventual failure of the pump. NPSHA, which is affected by both the system design on the suction side of the pump and the fluid being pumped, can be calculated using the following:

$$\text{NPSHA} = \frac{p_s - p_{\text{vapor}}}{C_2(sg)} + \frac{V_s^2}{2g} + Z_s - h_{fs} \tag{12.7}$$

where

NPSHA = net positive suction head available at the inlet to the pump, ft (m)
p_s = absolute suction pressure at point of measurement, psia (kPa abs)

p_{vapor} = true vapor pressure of fluid entering the pump, psia (kPa absolute)
C_2 = 0.433 when using the USCS units shown (9.789 for the metric units)
sg = liquid specific gravity at pumping temperature based on 1.00 for freshwater at 68°F (20°C)
V_s = fluid velocity at point of suction pressure measurement, ft/s (m/s)
Z_s = elevation at point of suction pressure measurement above (+) or below (–) the standard datum, ft (m)
h_{fs} = head loss due to friction and turbulence from the point of suction pressure measurement to the pump inlet, ft (m/s)

Values of NPSHR are generally determined by performing a test. During a typical NPSH test, the pump is operated at a constant capacity and speed while its suction pressure is gradually reduced until the total pump head drops. The value of NPSHA corresponding to a specified (frequently 3 percent) reduction in total head is considered to be the pump's net positive suction head requirement at the test capacity. This test is repeated at various capacities until a curve of NPSHR versus capacity can be drawn. As shown in figure 12-20, values of net positive suction head required based on a 3 percent drop in total pump head, identified as $\text{NPSHR}_{3\%}$, usually increase with capacity. It should be noted that cavitation bubbles can first begin to form with values of NPSHA that are significantly greater than those that result in a 3 percent drop in total pump head.

The change in $\text{NPSHR}_{3\%}$ with operating speed N for a given impeller of constant diameter when operating at the BEP can be estimated using the following:

$$(\text{NPSHR}_{3\%,\text{BEP}})_2 = (\text{NPSHR}_{3\%,\text{BEP}})_1 \left(\frac{N_2}{N_1}\right)^2 \tag{12.8}$$

where

$\text{NPSHR}_{3\%,\text{BEP}}$ = net positive suction head required at the BEP based on a 3 percent drop in total pump head, ft
subscripts 1 and 2 refer to conditions at the original and at the new operating speeds, respectively

It should be noted that the value of $(\text{NPSHR}_{3\%,\text{BEP}})_2$ found using equation 12.8 will not occur at the original best efficiency point flow rate, but rather at the flow rate corresponding to the BEP at the new operating speed, which can be found using equation 12.6a. In addition, because of various factors, such as thermodynamic effects on the fluid's vapor pressure, gas content, and changes in surface tension, the relationship given in equation 12.8 is only approximate. Therefore, it is sometimes recommended that equation

12.8 be used only when increasing speed, and that NPSHR3%,BEP be reduced by the ratio of the speed change when the operating speed is being reduced, which results in a higher, more conservative estimate for the NPSHR3%,BEP at the new lower speed.

Reductions in a pump's NPSH requirements may be achieved by installing an inducer in the inlet of the first stage impeller (fig. 12-22). An inducer is an axial-flow impeller containing a limited number of helically shaped vanes (typically from two to four) with diameters that increase as the axial distance to the impeller's eye is reduced. As a result of the larger flow area and due to the low number of vanes, blockage from cavitation bubbles that may form wit:bin an inducer is less than in a conventional impeller. In addition, an inducer acts as a booster stage and raises the pressure of the liquid being pumped, which reduces the potential for cavitation in the main impeller. During operation at the BEP, an inducer can sometimes reduce values ofNPSHR3% by up to 50 perecnt. However, an inducer can also result in an increase in NPSHR3% at capacities outside of the region of improvement. Consequently, prior to being used, the effect of an inducer on the performance of a pump that must operate at various capacities should be carefully analyzed.

Centrifugal pumps pumping hot water or certain hydrocarbon liquids can sometimes operate satisfactorily with less NPSH than the amount found to be required during a test performed using cold water. A chart for estimating the reduction in NPSHR3%when pumping some specific hydrocarbon liquids and high-temperature water is included in the Hydraulic Institute's *Pump Standards* (1994). However, this information is based on test data from pumps handling pure liquids; therefore, it is not recommended for use with pumps that will handle liquids containing entrained or dissolved noncondensable gases.

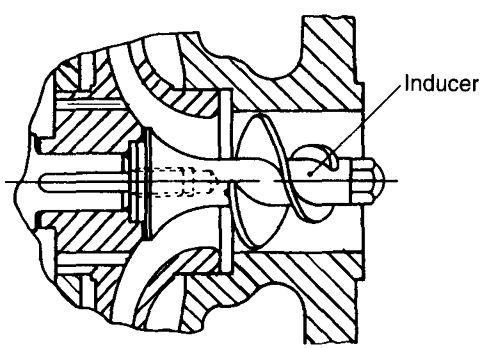


Figure 12-22. Centrifugal-pump inducer.
Courtesy Ingersoll-Dresser Pump Company.

EXAMPLE 12-5: For the system shown in figure 12-1, $Z_s = 3$ ft and $p_s = 20$ psia. In addition, the velocity within the suction pipe at point s is 5 ft/s. Also, the head loss in the suction line from point s to the pump's suction flange equals 1 ft. The liquid being pumped is freshwater at a temperature of 68°F with a specific gravity of 1.0 and a vapor pressure of 0.339 psia. Determine the net positive suction head available to the pump.

Solution: Using equation 12.7

$$\text{NPSHA} = \frac{20\text{ psia} - 0.339\text{ psia}}{0.433(1.0)} + \frac{(5\text{ ft/s})^2}{2\left(32.2\ \frac{\text{ft}}{\text{s}^2}\right)} + 3\text{ ft} - 1\text{ ft} = 47.8\text{ ft}$$

Note: Provided that the values used for pressure, velocity, elevation, and head loss are correct at the point of measurement, the NPSHA calculated will be the same regardless of the location of point s in the suction side of the system.

Suction and discharge recirculation. As shown in figure 12-23, suction recirculation refers to flow reversals that occur at the entrance to a centrifugal pump's impeller during operation below a certain capacity. Discharge recirculation, which is also depicted in figure 12-23, is a similar phenomenon occurring at the impeller outlet during low-flow operation. In

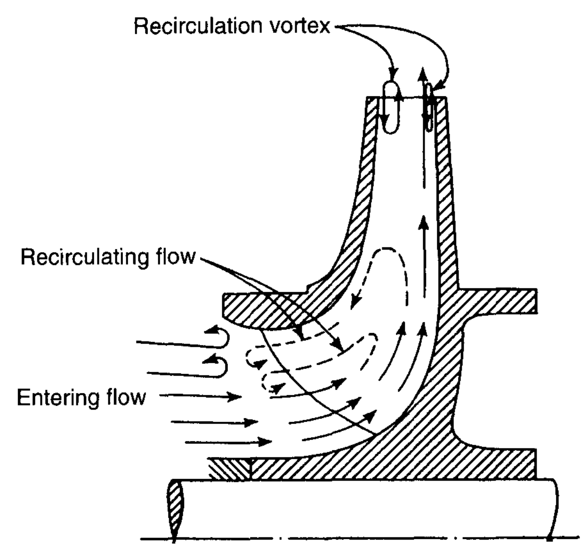


Figure 12-23. Centrifugal-pump suction and discharge recirculation.
Courtesy Ingersoll-Dresser Pump Company.

both cases, the interaction of the recirculated fluid with the fluid flowing in the proper direction results in the formation of high-velocity vortices. Because of the drop in local pressure at the cores of the recirculation vortices, cavitation often occurs. Operation with recirculation can result in increased noise and vibration, flow surges, erosion of the impeller and casing and, in the case of discharge recirculation, axial thrust reversals on the shaft. Although many of the effects of recirculation can appear to be similar to effects resulting from operation with insufficient NPSHA, cavitation damage at the inlet end of an impeller vane due to insufficient NPSHA normally begins on the vane's low-pressure or visible side; however, when caused by suction recirc,ulation, erosion begins on the pressure or hidden side of a vane. In addition, while recirculation typically occurs during low-flow operation and can frequently be suppressed if the capacity is increased sufficiently, cavitation from insufficient NPSHA can sometimes be suppressed by reducing the pumped capacity.

Recirculation should be considered when a centrifugal pump's minimum recommended capacity is established. The capacities at which suction and discharge recirculation will begin in a pump, which are frequently different, can be predicted using equations included in "Flow Recirculation in Centrifugal Pumps," by W. H. Fraser (1981). In addition, A. R. Budris presents empirically derived criteria for predicting the severity of the effects due to recirculation in specific designs in "Sorting Out Flow Recirculation Problems" (1989).

The capacity at which suction recirculation begins generally gets closer to QBEP as the suction specific speed, which is a characteristic number based on a centrifugal pump's suction capability, is increased. Suction specific speed can be calculated using the following:

$$S = \frac{N(Q_{\mathrm{BEP}})^{1/2}}{\left(\mathrm{NPSHR}_{3\%,\mathrm{BEP}}\right)^{3/4}} \tag{12.9}$$

where

S = suction specific speed
N = operating speed, rpm
Q_{BEP} = capacity pumped at the BEP, U.S. gpm (m^3/hr)
$\mathrm{NPSHR}_{3\%,\mathrm{BEP}}$ = net positive suction head required at the BEP based on a 3 percent drop in total pump head, ft (m)

Values of Q_{BEP} and $\mathrm{NPSHR}_{3\%,\mathrm{BEP}}$ used to calculate S are generally based on the capacity delivered and the NPSH required with a full-diameter impeller. In addition, because S is an index of the impeller's suction performance, when a pump has a double-suction first-stage impeller, the value of Q_{BEP} used in equation 12.9 should be only one-half of the total capacity pumped at the BEP.

To reduce the capacity at which suction recirculation begins, it is sometimes recommended that the maximum operating speed for standard pump designs be based on limiting S to 8,500 using the USCS units of measurement (9,870 with metric units) given with equation 12.9. Nevertheless, in applications for which standard impeller designs may not be suitable, pumps with special low-NPSHR high-S impellers are sometimes used. However, a pump that is fitted with a high-suction specific-speed impeller is often not suitable for low-flow operation. Because the additional stages in a multistage pump are supplied with fluid at an elevated pressure, when used, a special low-NPSHR high-S impeller is generally installed only in the first stage of a pump.

EXAMPLE 12-6: A single-stage centrifugal pump with a double-suction impeller delivers a capacity of 2,000 U.S. gpm and requires a net positive suction head of 10 ft (based on a 3 percent drop in total pump head) when operating at its best efficiency point. The pump's shaft is driven at a speed of 1,750 rpm. What is the pump's suction specific speed?

Solution: Using equation 12.9

$$S = \frac{1{,}750\ \mathrm{rpm}\left(\dfrac{2{,}000\ U.S.\ \mathrm{gpm}}{2}\right)^{0.5}}{(15\ \mathrm{ft})^{0.75}} = 7{,}261$$

Priming and gas entrainment. As liquid is discharged from a centrifugal pump impeller, a low-pressure region is created within the impeller's eye. Provided that liquid at a higher pressure is in the suction area of the casing, flow into the impeller will continue. However, because a centrifugal pump cannot pump a gas, the differential pressure necessary for flow will not be created if the impeller eye is full of air or another gas. Prior to start-up, a pump's casing should, therefore, be primed, or filled with liquid and vented of all gases. When a centrifugal pump must operate with a suction lift, a vacuum must initially be created to draw liquid from its source into the suction side of the pump casing and impeller. Arrangements used to prime a centrifugal pump being started with a suction lift include the following:

- The pump can be connected at its highest venting point to a priming system containing either vacuum pumps or air ejectors that draw gas out of the pump's casing and the suction line. This type of system may be used to prime multiple pumps on a vessel.
- A centrifugal pump can be furnished with a dedicated vacuum-priming pump. In some cases, the vacuum pump is driven through a belt by the centrifugal pump's motor.

- A special self-priming centrifugal pump can be used (fig. 12-24). This type of pump, which is often fitted with an open or a semi open impeller, typically has a casing with an enlarged suction chamber that retains liquid when the unit is not operating. Once the pump is started, this liquid, together with gas that may be mixed with it, is pumped through the impeller and enters a second chamber installed at the casing's discharge. The gas entrained in the liquid is vented through the top of the discharge chamber. The liquid, however, is returned to the casing through either an internal or an external port. As this liquid reenters the impeller, it mixes with gas that has been drawn into the evacuated suction chamber, and the priming cycle is repeated. Each time the stored liquid is recirculated through the pump, an additional amount of gas is removed from the suction line. Because of the limited air-handling capability of many self-priming centrifugal pumps, they should be carefully applied when large volumes of air or other gases must be removed from the suction piping.
- A foot or check valve can be used to retain liquid within a pump's casing and suction line when the pump is not operating. However,

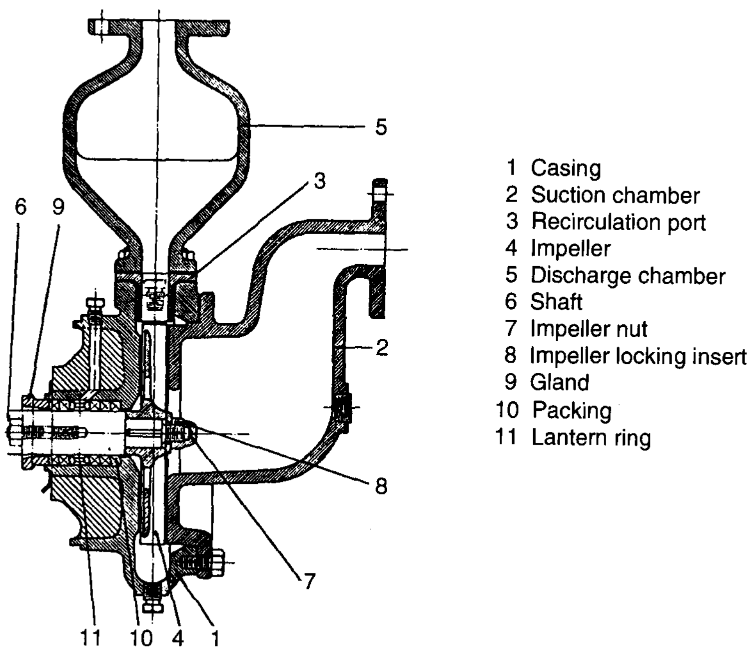


Figure 12-24. Self-priming centrifugal pump. Courtesy Ingersoll-Dresser Pump Company.

liquid can sometimes leak past the valve during periods when the pump is not in use. In addition, the installation of a suction foot valve increases the suction-line friction losses and, therefore, reduces NPSHA.
- Liquid supplied to a pump's inlet from a pressurized source can be used to prime the pump. The pressure drop resulting from the increase in the velocity of this liquid as it flows through the pump's suction connection will often create a vacuum that is sufficient to evacuate gases from a suction line connected to a second liquid source located below the pump.

Even after a centrifugal pump is primed, air and other noncondensable gases may be entrained in the liquid being pumped. Gas bubbles that accumulate in the eye of an impeller interfere with the flow of liquid through the pump and cause the discharge pressure and pump efficiency to drop. In addition, as the gas content increases, there is a reduction in the operating range of the pump due to choking at capacities above the BEP and surging caused by instability at flow rates below the BEP. Furthermore, handling a fluid with an entrained gas increases pump vibration and noise due to the unevenness of the flow through the impeller. With high flow rates and low gas volumes, some of the bubbles entering a centrifugal pump are flushed through the impeller by the liquid being pumped. However, as the capacity being delivered is reduced or the percentage of gas by volume in the pumped fluid is increased, the flow of liquid through the pump will eventually stop (fig. 12-25). Standard centrifugal pumps typically become gas-bound when handling fluids with a gas content of 5 to 8 percent by volume.

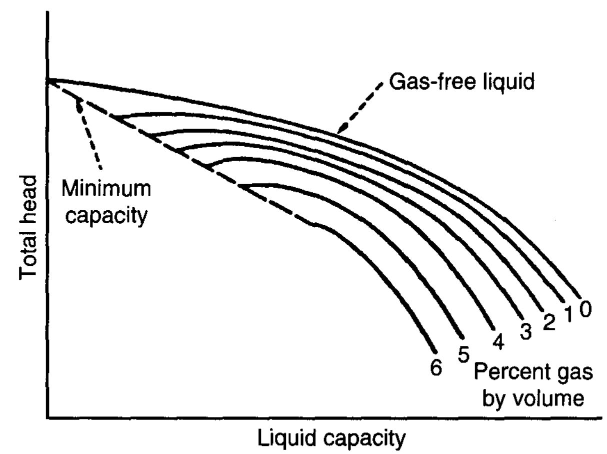


Figure 12-25. Effect of entrained gas on centrifugal-pump performance. Courtesy Ingersoll-Dresser Pump Company.

Viscosity. When the viscosity, or resistance to flow, of a pumped fluid is greater than the viscosity offreshwater (32 SSU or leSt), the total head developed and capacity delivered by a centrifugal pump will be reduced. As shown in figure 12-26, the pump's efficiency also will be reduced. In addition, because the effect that this deterioration in efficiency has on the pump's power requirement is often greater than the effect from the reductions in head and capacity, there is usually an increase in the power required to drive the pump. Performance corrections for centrifugal pumps with radial-flow impellers that deliver viscous Newtonian liquids at capacities not exceeding 10,000 U.S. gpm (2,270 m^3/hr) can be estimated using charts included in th~ Hydraulic Institute's *Pump Standards (1994).* Also, an extended chart that can be used to estimate viscous performance corrections for centrifugal pumps delivering capacities up to 100,000 U.S. gpm (22,700 m^3/hr) is included in Feck and Sommerhalder's "Cargo Pumping in Modern Tankers and Bulk Carriers" (1967).

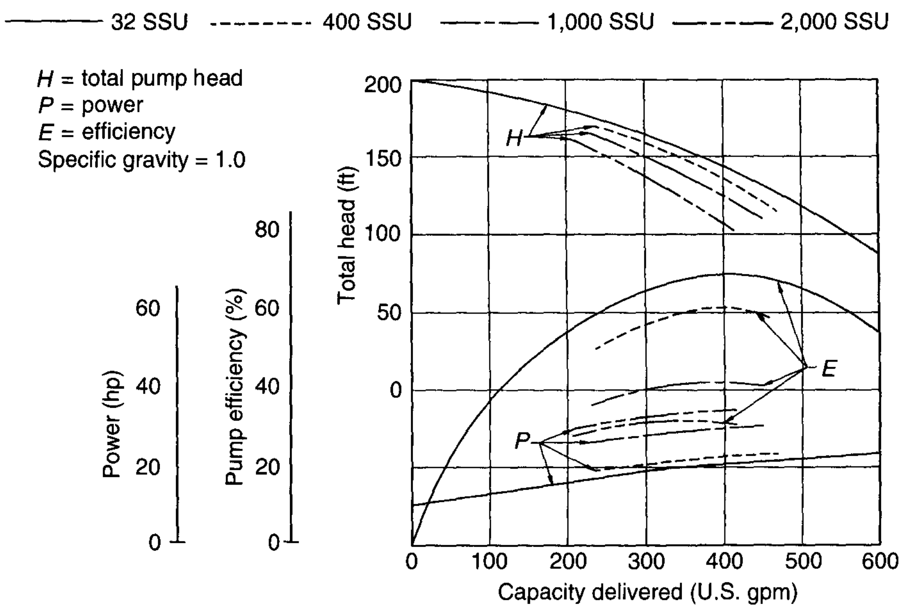


Figure 12-26. Centrifugal pump performance with viscous liquids

Overheating. As the capacity delivered by a centrifugal pump is reduced below the BEP, the pump efficiency and, therefore, the percentage of the power required by the pump that is converted into useful work drop. The increased losses result in an increase in the temperature of the pump and the liquid passing through it. The temperature rise of the pumped liquid can be estimated using the following:

$$\Delta T = \frac{H}{C_5\, c_p}\left(\frac{1}{\eta_P} - 1\right) \tag{12.10}$$

where

ΔT = temperature rise of liquid passing through the pump, F (C)
H = total head developed by the pump, ft (m)
C_5 = 778 when using the USCS units shown (102 for the metric units)
c_p = specific heat of the pumped liquid, Btu/lbm-F (kJ/kg-C)
η_P = pump efficiency, $\%/_{100}$

A rough approximation of the minimum flow rate required to limit the temperature rise across a centrifugal pump to a specified value can be estimated as follows:

$$Q_{\min} = \frac{C_6\, P_{P,SO}}{\Delta T_{\max}\, c_p\, sg} \tag{12.11}$$

where

$Q_{\min}$ = minimum capacity, U.S. gpm (m^3/hr)
C_6 = 6 when using the USCS units shown (4.25 for the metric units)
$P_{P,SO}$ = power required by the pump during operation at shutoff (i.e., operation with the proper suction conditions, a closed discharge valve, and no flow through the pump), hp (kW)
$\Delta T_{\max}$ = maximum allowable temperature rise across pump, F (C)

For general service, the maximum allowable temperature rise is often limited to 15°F (8.3°C). However, when handling hydrocarbons with relatively high vapor pressures or when there is very little margin between values of NPSHA and NPSHR, it is often desirable to limit the temperature rise to a lower value (e.g., 5° to 10°F or 2.8° to 5.6°C).

When operating at shutoff, there is no flow through a pump; consequently, no useful work is being done. If heat transferred to the surrounding atmosphere, which is initially negligible, and the heat absorbed by the pump are ignored, the rate of the temperature rise of the fluid that is trapped within the pump's casing will be approximately equal to the following:

$$\Delta T_{SO} = \frac{C_7 P_{P,SO}}{m_F\, c_p} \tag{12.12}$$

where

ΔT_{SO} = rate of temperature rise within the pump, F/min (C/min)

C_7 = 42.4 when using the USCS units shown (60 for the metric units)
m_F = mass of the fluid contained within the pump, lbm (kg)

As the temperature of the fluid within the pump increases, the rate of heat transfer through the pump casing to the surrounding atmosphere will increase and the rate of temperature rise (ΔT_{SO}) will gradually be reduced until an equilibrium condition is achieved and the fluid temperature stabilizes.

An increase in the temperature within a pump can result in the reduction of internal running clearances, overheating and possible failure of the bearings and seals, and, in extreme cases, seizure of the rotor. This increase in temperature can also lead to a hazardous situation when liquids with low flash and fire points are being pumped. Because thermal stresses vary with the temperature gradient across a pump's components, during start-up and shutdown, the temperature of a pump handling heated or cooled liquids should be changed slowly.

EXAMPLE 12-7: A centrifugal cargo pump develops 500 ft of total head and has an efficiency of 70 percent when operating at its rating point. It pumps oil with a specific heat of 0.5 Btu/lbm-F and a specific gravity of 0.9. During operation at shutoff, the pump requires 500 hp. Estimate the following:

1. The temperature rise of oil passing through the pump during operation at the rating point.

2. The minimum capacity required to limit this temperature rise to 150F.

3. The rate of the temperature rise of oil within the pump's casing during operation at shutoff. The mass of the liquid cargo contained within the pump casing is 300 lbm.

Solution: Using equations 12.10, 12.11, and 12.12 to answer questions 1, 2, and 3, respectively

$$\Delta T = \frac{500 \text{ ft}}{778\left(0.5 \dfrac{\text{Btu}}{\text{lbm}-\text{F}}\right)}\left(\frac{1}{0.70}-1\right) = 0.6°\text{F}$$

$$Q_{\min} = \frac{6(500 \text{ hp})}{15°\text{F}\left(0.5 \dfrac{\text{Btu}}{\text{lbm}-\text{F}}\right)0.9} = 444 \text{ U.S. gpm}$$

$$\Delta T_{SO} = \frac{42.4(500 \text{ hp})}{300 \text{ lbm}\left(0.5 \dfrac{\text{Btu}}{\text{lbm}-\text{F}}\right)} = 141.3 \frac{\text{F}}{\text{min}}$$

Multiple-pump performance. During series operation, the fluid discharged from one pump is directed to the inlet of a second pump. The combined performance of two or more pumps operating in series can be determined by adding the total head developed individually by each pump when all of the pumps involved are delivering the same capacity (fig. 12-27a). This arrangement is sometimes used when the speed required to prevent cavitation is too low for one pump to develop sufficient head to overcome the system resistance or back pressure. In this situation, the first pump can be operated at a low speed and utilized as a booster or inducer pump with its discharge directed into the suction of the second pump. Because of the increased NPSH available to the second pump, it can be operated at a higher speed and can, therefore, develop a greater total head than the first unit. When necessary, the discharge from the second pump can be directed to the inlet of an additional pump.

During parallel operation of two or more pumps, each pump delivers only a portion of the total flow. The combined performance of pumps operating in parallel in a common system may be found by adding the capacity delivered individually by each pump when all of the pumps involved are developing the same total head (fig. 12-27b). This arrangement can be used when the capacity that must be delivered exceeds the capability of a single pump in the system. If pumps operating in parallel have similar performance characteristics, have approximately the same suction pressure, have similar suction and discharge piping and valve configurations, and are operating at the same speed, they generally will each deliver approximately the same capacity. However, if one or more of the pumps develops a higher head at a given capacity, the other units may be prevented from delivering adequate flow rates. Consequently, if pumps operating in parallel are not similar, or their suction conditions or piping arrangements are not similar, pump operating speeds should be adjusted so that an adequate flow is delivered by each unit.

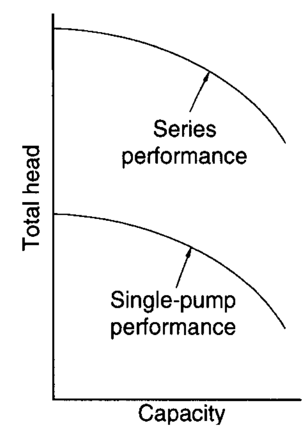


Figure 12-27a. Series performance of two identical centrifugal pumps

The operating point during multiple-pump operation is determined by the intersection of the curve showing combined pump head versus capacity with the curve of system head versus capacity.

Droop at shutoff and instability. The total head developed by a centrifugal pump generally increases as the capacity delivered is reduced. However, the head developed by some pumps drops as the capacity is reduced during operation near shutoff. This is commonly referred to as a drooping head-capacity characteristic. Droop can lead to difficulties during single-pump operation whenever the total head developed at shutoff is insufficient to overcome the corresponding resistance in the system, which can oc-Cur if the majority of the system head requirement is composed of static head that exceeds the pump's shutoff head. Difficulties can also occur if the system head curve crosses the drooping pump's H-Q curve at two locations. In addition, if operation is along the drooping portion of a pump's H-Q curve (i.e., at a capacity to the left of the flow rate at which the maximum total head is developed), a sudden increase in the system back pressure will result in reductions in both the capacity being delivered and the total head developed by the drooping pump. The reduced pump head may be momentarily less than the system's back pressure, which, in some cases, can lead to back flow through the system and possibly even to flow swings.

When two drooping pumps operate in parallel, there can be an unequal sharing of the load, resulting in the delivery of a higher capacity by one of the pumps while both are developing the same total head. In addition, over a portion of the performance range, it may be possible for the system head curve to cross two different combined pump H-Q curves. The number of possible combined pump H-Q curves increases with the number of drooping pumps operating in parallel.

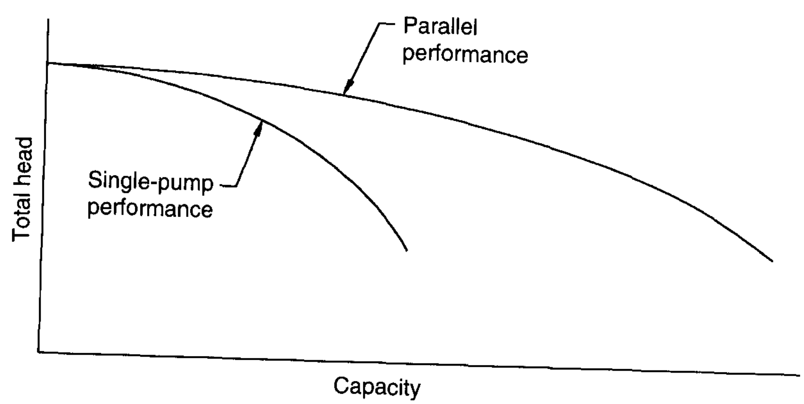


Figure 12-27b. Parallel performance of two identical centrifugal pumps

In addition to droop, the H-Q curves for some pumps also have dips, or saddles, at low flow rates. Therefore, even when the maximum total pump head is developed at shutoff, performance in the low-flow region can still be unstable.

Although droop and saddles in a pump's H-Q curve are undesirable, in some pumps they are difficult to eliminate without adversely affecting other design criteria. In addition, many drooping pumps operate successfully in critical applications. Therefore, a head-capacity characteristic that does not rise continuously to shutoff may be suitable for some applications, provided that effects resulting from the shape ofthe pump's H-Q curve are carefully analyzed and will not have a detrimental impact on the pump's performance in the system.

Vibration and critical speed. If a rotor is excited at a frequency that coincides with one of its natural frequencies, (i.e., one ofthe frequencies at which it would vibrate if rapped like a tuning fork), the resulting vibration will be limited only by the damping of the materials used for the rotating components and the liquid within the casing. It is therefore important that the rotor's natural frequencies be a sufficient distance away from applied forcing frequencies, such as the pump's running speed and its impeller vane passing frequency. In general, the lowest frequency of interest is the rotor's first bending natural frequency, which is also often referred to as its first critical speed. A pump that operates below its first critical speed is frequently referred to as a stiff-shaft unit, while a pump that operates above the first critical speed is sometimes called a flexible-shaft unit. Although it is desirable to use a stiff-shaft pump, in some cases (e.g., for some high-speed multistage pumps), it may be impractical to design a pump's rotating assembly to meet this requirement. Therefore, flexible-shaft designs are sometimes used. However, all of a pump's operating speeds should be at least 25 to 30 percent above or below the nearest critical speed and the pump should pass through any critical speeds below its operating speed quickly during start-up or shutdown. A pump's critical speed can generally be increased by reducing the bearing span and by increasing the shaft diameter. To a lesser extent, it can also be increased by reducing the weight of the rotating assembly and by using a shaft material with a higher modulus of elasticity.

In addition to operation too close to a critical speed, common causes of excessive pump vibration include operation with rotor unbalance, coupling or shaft misalignment, a bent shaft, loose or broken rotating parts, rubbing contact between rotating and stationary parts, too small a gap between the impeller vanes and a volute or diffuser, bad bearings, cavitation, and internal recirculation.

Operation. Prior to starting a typical centrifugal pump, the unit should be thoroughly inspected, and any abnormalities that are found should be corrected. When practicable, the pump shaft should be turned by hand to ensure that it rotates freely. (In some cases, it may not be possible or advisable to turn the shaft by hand because of the size, configuration, or design of the pump.) When bearings are oil-lubricated, the LO level in the sump should be checked, and oil should be added ifneeded. Pumps with a forced-feed lubrication system may have a hand- or motor-driven lubricating pump that should be operated to pre lubricate bearings. Valves in sealing lines, gland leak-off lines, gauge lines, bearing lubrication supply and return lines, and cooling-jacket supply and return lines should ordinarily be opened. In addition, the suction valve should be fully opened, and the pump casing should be filled with liquid and vented of all gases (i.e., the pump should be primed).

When the pressure ofliquid at the pump inlet is positive (i.e., above atmospheric pressure) and the discharge connection is in the top of the casing, the pump may be self-venting. When the discharge is in a different location, however, a valved vent opening is frequently located in the highest point of the pump's casing. Opening the valve often enables liquid in the suction line to force any gas that may be present out of the pump. *M*-ter all of the gas has been expelled from the pump, the vent valve should be closed. Caution should always be exercised when opening a vent valve that discharges to atmosphere to prevent injury to personnel and damage to equipment. If the pump suction is under a vacuum, opening a vent valve that discharges to atmosphere will allow air to be drawn into the pump. To prevent this, it may be necessary to connect the vent to a priming pump or system or to pipe it to a low-pressure chamber. The vent valve is sometimes left open during operation so that any air that enters the pump will be drawn out of the casing.

When starting a typical radial-flow centrifugal pump, the discharge stop valve should be almost completely closed, which will generally reduce the load on the pump's driver and the NPSH required by the pump. However, to reduce the potential for overheating due to insufficient flow ;hrough the pump, the duration of operation with the discharge valve in ;his choked-in position should be kept to a minimum. With many mixed-low and almost all axial-flow pumps, throttling the discharge valve in-Teases the driver load (fig. 12-21); consequently, when starting a pump laving this latter type of power characteristic, the discharge stop valve hould ordinarily be fully open. (A pump should generally not be started if ts shaft is being rotated in the reverse direction by fluid flowing back from he discharge line. In systems where this can occur, a check valve should be tlstalled in the discharge line to prevent reverse flow through the pump.)

After starting a centrifugal pump's driver, a check should be made to erify that the pump's discharge pressure rises to the proper value and, rhen possible, that liquid is being discharged from the pump. Once a mstant-speed pump has accelerated to its rated speed, the discharge alve position should be slowly adjusted until the desired capacity is being delivered. When a variable-speed driver is used, however, it is often possible to fully open the discharge valve and to regulate the capacity being pumped by adjusting the driver's speed. Typically, a pump's suction valve should never be throttled.

While a pump is operating, its suction and discharge pressures should be monitored and its casing, bearings, and seals should be checked frequently for any signs of overheating or leaks. When oil is used for bearing lubrication, the level in the LO reservoir should be checked periodically and maintained at the proper value. In addition, when a forced-feed lubrication system is used, the LO pressure should be monitored, and, when sight glasses are provided, oil flow through the bearings should be observed. If an LO cooler is provided, the flow rate of the cooling water medium passing through the cooler must often be regulated to maintain the proper oil outlet temperature. A pump should generally be stopped ifthere are any abnormalities, including excessive or unusual noise or vibration.

Prior to shutting down a radial-flow centrifugal pump that does not have a check valve in its piping, the pump's discharge stop valve should ordinarily be closed. This will prevent reverse flow through and possible reverse rotation ofthe pump when it stops. The driver should then be secured following the manufacturer's instructions. In most cases, other stop valves associated with the pump can then be closed. However, when a check valve is installed in the discharge line, stop valves (including the discharge stop valve) are sometimes left open when a centrifugal pump is shut down so that the pump can be restarted quickly, automatically, or remotely. The procedure for stopping a mixed- or axial-flow pump is similar to that just described. However, because closing the discharge stop valve of either a mixed- or axial-flow centrifugal pump generally results in an increase in the horsepower required by the pump, the discharge valve should normally be closed immediately after the pump is stopped.

Troubleshooting. Reasons why a centrifugal pump may fail to deliver the proper capacity or develop the proper discharge pressure can include operation at the wrong speed or with the wrong direction of shaft rotation, the use of an impeller that has the wrong outside diameter or is mounted on the pump's shaft in the reverse direction, external leakage through casingjoints or piping connections, excessive leakage through shaft seals, excessive internal recirculation through enlarged wearing-ring or interstage-bushing clearances, the presence of air or other gases in the pump or suction line (in some cases air may be drawn into a pump through its shaft seal or through the stuffing box of its suction valve), insufficient submergence of the suction pipe, insufficient NPSHA, foreign material in the pump or piping, a clogged suction strainer, mechanical defects (including a suction or discharge valve that has failed closed, and an impeller key, coupling, pump shaft, or driver failure), high system back pressure, and excessive viscosity of the pumped liquid. In addition, because the capacity delivered by a centrifugal pump is determined by the point at which the pump's H-Q curve intersects the system-head curve, for a centrifugal pump to deliver the desired capacity, the pump curve must generally intersect the system-head curve at the point corresponding to the same flow rate.

Among the possible causes of an excessively high power requirement by a centrifugal pump are rubbing contact between rotating and stationary parts, which can result from thermal expansion, a bent shaft, a distorted casing (this may be the result of excessive piping loads), misalignment of the casing or rotor, worn bearings, or excessive shaft deflection; operation at too high a speed; the use of an impeller with too large an outside diameter; a high specific gravity or viscosity ofthe pumped liquid; improperly lubricated or overloaded bearings; overly tightened or incorrect packing; mechanical defects; and foreign matter in the impeller. In addition, when a centrifugal pump's driver is not sized to deliver the maximum power required by the pump at any operating point along the pump's entire H-Q curve, the driver can be overloaded ifthe system-head curve intersects the pump's curve at too high a capacity (for most radial-flow pumps) or at too Iowa capacity (for most axial-flow and many mixed-flow pumps).

A casing that is unusually warm may be the result of operation with no flow or with too Iowa flow rate, internal contact between rotating and stationary parts, and foreign material or a mechanical failure within the pump. An overheated bearing can be caused by excessive bearing loads, which may be the result of improper installation, misalignment, a bent shaft, or an unbalanced rotor; bearing wear; improper bearing lubrication; inadequate cooling of the bearing housing or the lubricant; and contamination of the lubricant. Excessive vibration can be caused by misalignment, radial unbalance, a bent shaft, worn bearings, rubbing contact between the rotor and stator, pipe strain, mechanical defects, a flexible or uneven foundation, cavitation, suction or discharge recirculation, air or other gases in the pumped fluid, and foreign material within the pump. Excessive packing leakage or premature packing or mechanical-seal failure can be caused by misalignment; worn bearings; a bent shaft; a shaft or shaft sleeve that is scored in the area of the packing or seal; improper installation; the use of the incorrect packing or seal size or type; operation with insufficient cooling flow through the packing, to the seal, or to a cooling jacket surrounding the seal area; the presence of abrasive particles in the pumped fluid; and excessive shaft vibration.

REGENERATIVE TURBINE PUMPS

A regenerative turbine pump has a solid disk-shaped impeller with multiple equally spaced radial vanes that rotates within a cylindrically shaped casing (fig. 12-28). The typical regenerative-turbine-pump impeller is hydraulically balanced axially and can either be mounted on a shaft that is supported at each end by a bearing or overhung on the end of a cantilevered shaft that is supported by bearings located behind the impeller. Although the pump's shaft is often flexibly coupled to its driver, pumps with an overhung impeller can be furnished in a close-coupled configuration. Shaft penetrations in the casing are sealed with packing or mechanical seals. To permit internal running clearances to be periodically renewed, the casing is often fitted with a replaceable radially split liner that surrounds the impeller.

Fluid entering a typical regenerative turbine pump is admitted to both sides of the impeller and is forced by the impeller's vanes to rotate. As its velocity increases, the fluid is thrown radially outward due to centrifugal force. Mter the fluid leaves the impeller, its velocity is gradually reduced in the casing and there is a conversion of kinetic energy to potential energy that results in an increase in the fluid's pressure. Because of the shape of the casing's passages, the fluid is then directed back into the rotating impeller and the cycle is repeated. The fluid travels around the circular casing in a spiral path and continues to be recirculated into and out of the impeller until it reaches the discharge port. Because the fluid pressure is increased during each of the multiple recirculation cycles, regenerative turbine pumps have the ability to develop relatively high values of total head while delivering relatively low capacities. The total head developed and the power required by a regenerative turbine pump typically peak at shutoff and are both reduced as the capacity delivered increases.

As fluid reaches the casing's discharge port, a close-radial-clearance stripper impedes its recirculation back to suction. Because of this stripper, a regenerative turbine pump can typically operate with relatively large

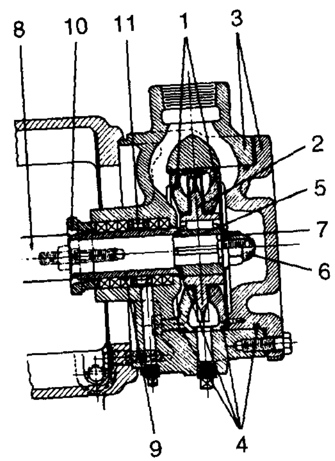


Figure 12-28. Regenerative turbine pump.
Courtesy Ingersoll-Dresser Pump Company.

amounts of gas or vapor entrained in the pumped fluid, provided that the casing is initially filled with liquid prior to start-up. Because ofthe small flow passages and close operating clearances, regenerative turbine pumps are not recommended for applications involving liquids with high viscosities or liquids that contain solids.

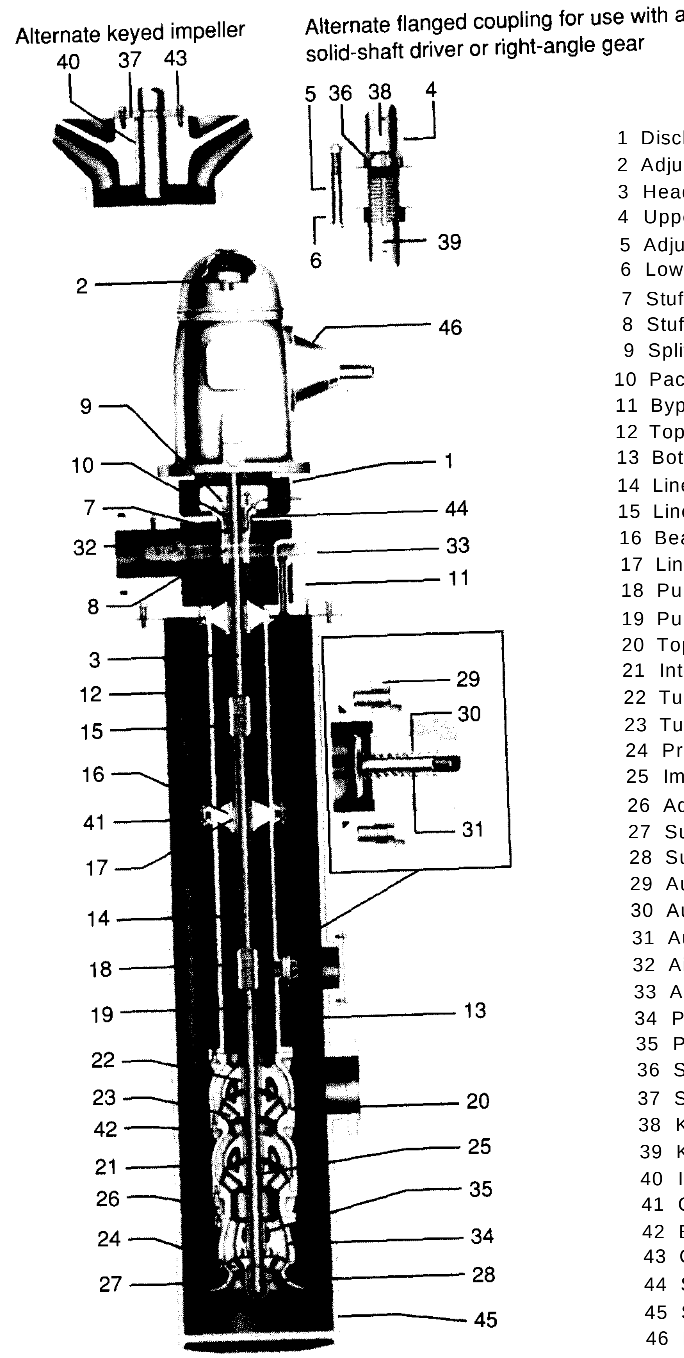
VERTICAL TURBINE PUMPS

A vertical turbine pump (VTP) is a centrifugal pump with a suction inlet that is submerged within the liquid being pumped (fig. 12-29). Torque developed by the abovedeck driver is transmitted to one or more impellers, which are located at the lower end of the pump, through a vertical line shaft that extends throughout the length ofthe VTP. This line shaft is surrounded by a column pipe that also serves as the conduit through which fluid discharged from the impellers is delivered to the above deck discharge head. When furnished with long lengths of column pipe and line shafting, a VTP is also commonly referred to as a deepwell pump.

Vertical Turbine Pump Components

Bowl assembly. The bowl assembly is located at the lower end of a VTP and contains one or more casing sections, referred to as bowls, that can be stacked vertically to form multiple stages. Each stage also includes a single-suction impeller and either a multi vaned diffuser or a volute. The number of stages used is based on the pump's total head requirement and the total head developed per stage. Joints between adjacent bowls can be flanged or threaded, or the entire bowl assembly may be held together with external tie-bolts. Each bowl is generally fitted with a sleeve-type journal bearing that supports the impeller shaft. Impellers can be secured to the shaft with keyless tapered lock collets, or they can be keyed to the shaft and held in place axially with snap rings, split thrust rings, or locknuts. So that internal running clearances can be periodically renewed, bowls and, in some cases, impellers are often fitted with replaceable wearing rings. The first-stage impeller is sometimes overhung on the end ofthe impeller shaft. Alternatively, the lower end ofthe shaft may extend through the eye ofthe first-stage impeller and be supported by a bearing located in a suction bell that is installed below the first-stage bowl. Also, when NPSHA is limited, an inducer may be mounted on the lower end of the impeller shaft. If the fluid being pumped contains foreign particles, a strainer is sometimes mounted on the open end of the suction bell.

Column and line shaft assembly. Fluid discharged from the upper bowl enters and flows through a vertical column pipe. This pipe also encloses the line shaft that transmits torque from the driver to the impeller shaft. Ordinarily, mating sections of column pipe either have flanges that are bolted



1 Discharge head
2 Adjusting nut
3 Headshaft
4 Upper half coupling
5 Adjusting plate
6 Lower half coupling (center)
7 Stuffing box
8 Stuffing box bushing
9 Split stuffing box gland
10 Packing
11 Bypass pipe
12 Top column pipe
13 Bottom column pipe
14 Line shaft
15 Line-shaft coupling
16 Bearing retainer
17 Line-shaft bearing
18 Pump-shaft coupling
19 Pump shaft
20 Top turbine bowl
21 Intermediate turbine bowl
22 Turbine bowl bearing
23 Turbine impeller
24 Priming stage impeller
25 Impeller tapered lock collet
26 Adapter case
27 Suction bell
28 Suction bell bearing
29 Autoprime valve housing
30 Autoprime valve spring
31 Autoprime valve
32 Air release valve
33 Air release line
34 Priming stage bowl
35 Priming stage bearing
36 Split ring, upper half coupling
37 Split ring, impeller
38 Key, motor shaft
39 Key, headshaft
40 Impeller key
41 Column flange bolt
42 Bowl cap screw
43 Cap screw, split ring collar
44 Stuffing box gasket
45 Suction can
46 Right-angle gear

Figure 12-29. Vertical turbine pump. Courtesy Goulds Pumps, ITT Industries.

together or are joined with threaded sleeves. Sleeve bearings that support the line shaft are typically installed in retainers, or spiders, that are sandwiched between the mating ends of adjacent column sections. Although the bearings are often lubricated by the pumped fluid, the line shafting in some vertical turbine pumps is enclosed within a tube and its bearings are lubricated by oil supplied from an external source. Adjacent sections ofthe line shaft are typically joined by keyed or threaded-type rigid couplings. (When threaded line-shaft couplings are used, it is important that adjacent ends of mating shaft sections be in contact after assembly to prevent the couplings from tightening and pulling up the impeller shaft during operation.) A tapered transition piece may be installed between the upper bowl and the lower column pipe. However, as an alternative to this arrangement, the bowl and column assemblies are sometimes connected through an axially split spool piece that permits the bowl assembly to be removed while the column assembly is left in place.

Discharge head. The discharge head supports the column-and-bowl assembly, which is hung from the bottom of the head, and the pump's driver or right-angle gear that is mounted on top of the head. It also serves as a 90° elbow and directs fluid leaving the vertical column pipe into the discharge piping. The penetration in the discharge head for the pump's top shaft is sealed with packing or with one or more mechanical seals. Various arrangements can be used that enable the shaft seal to be cooled and lubricated either by the pumped fluid or by liquid supplied from an external source.

Thrust bearings. It is common for the rotating assembly in a VTP to be supported axially by the thrust bearing in the pump's vertical driver or right-angle gear. Alternatively, however, a VTP may be fitted with an independent thrust bearing that is mounted in its discharge head. Because ofthe hydraulic unbalance resulting from the single-suction design of the typical VTP impeller, together with the weight ofthe pump's rotating assembly, axial thrust acting on a vertical turbine pump's shaft is usually directed downward. However, when the suction pressure or capacity delivered is high or the total head developed is low (e.g., during start-up), the net axial thrust can, at times, be directed upward. Therefore, in addition to being rated for the maximum downthrust, which may be reduced in each stage individually with a back wearing ring and axial balancing holes or for the entire pump with a balancing drum, the thrust bearing that supports the pump's shaft axially should also be rated for any expected intermittent upthrust. Oil-lubricated multiple-row ball bearings are frequently used.

Vertical driver or right-angle gear. A VTP may be driven by a vertical driver mounted directly on top of its discharge head or by a horizontal

driver through a right-angle gear. When the vertical driver or right-angle gear that is used has a hollow shaft, the pump's top shaft extends up through the center of the hollow shaft and is secured at the top of the driver or gear with an adjusting nut. Alternatively, the pump's top shaft may be connected to a solid-shaft vertical driver or right-angle gear through an abovedeck coupling. When the pump's shaft is supported axially by the thrust bearing located in the solid-shaft driver or gear, a rigid-type abovedeck coupling must be used. If the pump has a separate thrust bearing, however, a flexible coupling should be used. If the pump is fitted with a mechanical seal, a spacer is often provided between the two coupling halves so that the seal can be replaced without disturbing the driver or right-angle gear.

A VTP behaves like a hydraulic turbine when driven by the liquid that drains from its column and flows backward through its impellers after the driver is stopped. Reverse rotation ofthe pump shaft resulting from this can sometimes cause mechanical damage, especially if the unit is restarted before the shaft stops turning. To prevent reverse rotation, a driver or right-angle gear that will be used with a VTP is sometimes fitted with a nonreverse ratchet that permits the pump shaft to rotate in only one direction.

Vertical Turbine Pump Installation

When a VTP is installed directly into the tank or sump from which it will remove liquid, no suction piping is required or used. It is important that the suction area ofthe sump be sized so that the velocity of incoming fluid is not excessive (e.g., less than 3 ftls or 0.9 m/s). Also, an adequate axial clearance must be maintained between the bottom ofthe pump and the sump. So that relative movement of a vessel's structural members will not result in pump misalignment, rigid attachments below the pump's foundation at the base of the discharge head should be avoided. However, vanes are often installed in close proximity to a VTP's suction inlet to break up vortices that may form within the fluid entering the pump and to limit the radial movement ofthe bowl assembly due to vessel motion. In addition, for a long pump, one or more belowdeck support rings may be installed along the column pipe to prevent excessive radial movement of the column and bowl assemblies as a result of vessel motion. To provide flexibility, a spring-loaded or resilient stabilizer should usually be mounted on the column pipe adjacent to each support ring. Proper alignment ofthe pump's belowdeck supports is critical.

If a VTP is to receive liquid from multiple sources, it will generally be installed within a separate suction tank or can. The suction can is then connected through piping to the various locations from which the pump will take suction. To reduce relative movement between the suction can and the VTP, whenever possible, the can should be suspended from the same deck that supports the pump's discharge head. When a VTP installed in a suction can must be capable of evacuating the suction piping of gas and vapor, it is typically fitted with one or more automatic self-priming valves that open when the pump loses suction and allow liquid contained within the column pipe to be recirculated to the can. The gases in the bottom of the can are displaced by this liquid and pass through a vent line into the discharge head and column pipe. When the liquid level in the can is sufficient te ump regains suction, the priming valves close, and the gases contained wIthm the column n~ discharge head are forced out through a discharge check valve. The pnmmg cycles ordinarily continue until enough gas has been emoved from the suction piping for the pump to operate normally.

AlIgnment of a vertical driver or right-angle gear to a VTP should be performed only after the pump's discharge head is aligned with and bolted to its fudation and th~ dischaJ;ge piping has been connected to the pump. The pIpm~ should be mdependently supported and properly aligned so that loads Imposed on the pump's discharge head will be minimized and will be within the pump manufacturer's allowable values. Additionally, piping should always be thoroughly flushed and cleaned prior to pump installation. To prevent backflow through the discharge head when a VTP is shut down a check valve is frequently installed in the pump's discharge piping. '

. Comments presented previously regarding coupling alignment for centnugal pumps,.for th~ most part, also apply to a VTP that is driven by a sohd-shaft vertIcal dnver or through a right-angle gear. However, the shat~ of a '7~ that does not have its own thrust bearing cannot be rotated ntl~ after It IS coupled to the driver or right-angle gear; consequently, dial mdIcators that may be used during the initial alignment should be attache~ to the driver or gear half of the coupling. In addition, a bushing or cenenng. gauge may be required to properly center the pump's top shaft durmg ahgnment. Because a rigid coupling is used between a VTP not fitted with its own thrust bearing and a solid-shaft vertical driver or right-angle gear, accurate alignment is critical. A hollow-shaft vertical driver or right-angl~ gar should be aligned so that the pump's top shaft is properly entered wIhm the hollow driver or gear shaft. It is always important to verI£?that a drIver has the correct direction of rotation before coupling it (either lrectly or trough a right-angle gear) to a VTP. This is especially important Ithe pump s rotating assembly has threaded components, such as threaded lme-shaft couplings, which can unscrew if the pump is driven by its driver in the wrong direction.

. Mter the driver or gear has been aligned and coupled to the pump, it is Important that he rotaing assembly in a VTP not fitted with an independent thrust bearmg be raIsed to a height that is sufficient to allow the impellers.to turn freey uring operation. Failure to properly adjust the height of the Impellers wItm the bowls can result in severe damage to the pump afer start-up. The Impellers can typically be raised by tightening an adjustmg nut o? the .upper end of the pump's top shaft. This nut will ordinarily bar agamst eIther a clutch at the top of a hollow-shaft vertical driver or rIght-angle gear or, when a solid-shaft driver or gear is used, the pump half

of a rigid above deck coupling. The amount that the pump's rotor is raised, including an allowance for the amount that the line shaft will stretch during operation, should, whenever possible, be based on the pump manufacturer's recommendation. If the manufacturer's recommended value is not available, starting with the pump's impellers resting on the bottom oftheir bowls, the pump's rotor can be raised until the impellers are bearing against the tops of the bowls. The shaft should then be lowered by an amount equal to one half ofthe amount that it was raised, less the amount that the line shaft will stretch during operation. The impellers will then be slightly above the bowl centerlines when the pump is not running, and they will be centered axially within the bowls during operation.

Vertical Turbine Pump Performance Characteristics

Because a VTP is a centrifugal pump, its performance characteristics are similar to those previously presented. In a multistage pump, the amount of cavitation in the first stage necessary to produce a 3 percent reduction in total head can be significant. Therefore, because of a VTP bowl assembly's modular construction, NPSH testing is often performed using only the pump's first stage. Even when adequate NPSH is available to a VTP, suction can be lost if the submergence ofthe pump's suction bell is not sufficient to suppress vortex formation on the surface of the liquid. As a vortex forms and its size and intensity increase, air can be drawn through its core and into the pump. The minimum submergence required to suppress vortex formation can be reduced by properly designing the sump and reducing the velocity of the fluid entering the pump.

ADDITIONAL TYPES OF WET·PIT PUMPS

Additional types of vertically mounted wet-pit pumps that are submerged within the pumped liquid include the following:

- Propeller-type pumps that deliver high capacities at relatively low heads
- Single-stage double-suction impeller pumps that, because of their double-suction design, typically require less NPSH than comparably sized VTPs
- Single-stage, end-suction volute-type pumps furnished with short settings or lengths that are often referred to as sump pumps. The typical sump pump's volute casing has a side discharge connection. Because of this configuration, the vertical pipe through which fluid travels to the abovedeck discharge flange is often mounted adjacent to the line shaft rather than concentric with it.
- Single-stage, end-suction, submersible centrifugal pumps that are submerged, together with their hydraulic or electric motors, in the pumped liquid. This arrangement eliminates the need for both an abovedeck driver and long lengths of line shafting.

VACUUM PUMPS

A typical liquid-ring wet-type vacuum pump, which can be used to remove air and other gases from piping and components, has a multivaned rotor that rotates either within an eccentric-cylindrical or a concentric-elliptical casing (fig. 12-30). Separate pumping chambers are formed between adjacent pairs of the rotor's rigid vanes, which are generally oriented radially with tips that may be curved slightly forward. Leakage between the individual chambers is prevented by shrouds that are attached to both ends of the rotor. Inlet and discharge ports leading to the rotating pumping chambers are in a stationary cone located in the center of the casing. For proper operation, the casing must be partially filled with water or some other suitable liquid. As the rotor turns, it forces this liquid to rotate in a ring that, because of centrifugal force, follows the contour of the casing's inner wall. Consequently, when each of the rotor's chambers sweeps past a portion of the casing with an increased rotor-to-casing clearance, the liquid contained between the vanes is thrown outward. This evacuation of the inner portion of the chamber creates a vacuum that draws air or other gases through the adjacent inlet port and into the space between the vanes. As the rotor continues to turn and the chamber passes through a close-

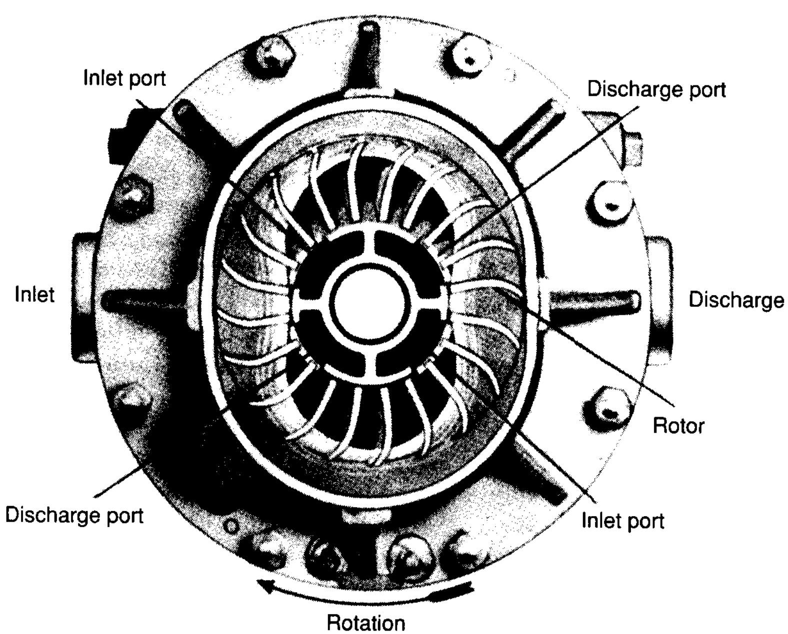


Figure 12-30. Liquid-ring vacuum pump.
Courtesy The Nash Engineering Company.

clearance portion ofthe casing, the liquid is forced to reenter the space between the rotor's vanes and it forces the gases out through the pump's discharge port. To reduce radial unbalance, pumps with an elliptically shaped casing typically have two inlet ports and two discharge ports that are alternately spaced at 90° intervals. With this arrangement, there are two pumping cycles per chamber for every revolution ofthe rotor. Because the liquid acts like a piston as it flows into and out of each pumping chamber, these units are sometimes referred to as liquid-piston pumps.

Over much of the normal operating range, the capacity delivered by a typical liquid-ring vacuum pump remains relatively constant. However, the capacity delivered and vacuum developed will generally both be reduced as the temperature of the pump's sealing liquid increases. During operation, a small amount ofliquid should be continuously supplied to the pump's casing to replace any of the sealing liquid that may be discharged during each pumping cycle and to keep the pump cool.

A liquid-ring vacuum pump's rotor may be mounted between bearings or overhung. When this latter arrangement is used, the rotor is sometimes mounted directly on the driver's shaft in a close-coupled configuration. Openings where the shaft penetrates the casing are often sealed with packing. Two-stage vacuum pumps, which have two rotors that are mounted on a common shaft and operate in series, are frequently used when high vacuums must be developed.

Positive-Displacement Pumps

ROTARY PUMPS

A rotary pump is a positive-displacement pump with one or more pumping elements that rotate within a stationary casing. Fluid entering a rotary pump is trapped within cavities in the rotating elements and is forced through the casing. The return of fluid to suction is impeded by close internal running clearances and, in the case of multiple-rotor units, by the meshing ofthe pumping elements. Consequently, during the revolution of each rotor, a nearly fixed amount of fluid is discharged from the casing. In addition, as the vacant cavities within each rotor pass the casing's suction port, a partial vacuum is created that draws additional fluid into the pump. Most types ofrotary pumps can be furnished in both horizontally and vertically mounted configurations.

Types of Rotary Pumps

Screw pumps. Fluid entering a screw pump is trapped within a series of helical cavities that are formed between the threads of one or more screws and are sealed by a close-clearance casing. As each screw turns, the fluid trapped within the sealed cavities is forced to advance axially through the pump's casing from the inlet to the discharge port.

Single-screw pumps, which are often referred to as progressing-cavity pumps, have one rotor with an external helical thread (fig. 12-31). As it turns, the rotor also orbits around the centerline of the driveshaft, and threads on the rotor mesh with the stator's internal threads. Although many progressing-cavity pumps are fitted with a single-helix rotor that turns within a double-helix stator, designs with other configurations, such as a double-helix rotor that turns within a triple-helix stator, are also available. The casing is frequently fitted with an elastomeric liner to allow particles entrained in the pumped fluid to pass through the pump without causing serious damage to the unit. In a typical progressing-cavity pump having a single-end design, fluid flows in only one direction along the rotor. With this arrangement, the pump's single shaft seal, which is often a packed stuffing box, and the bearings that support the driveshaft are located behind the rotor. Due to the orbital motion of the rotor and the friction resulting from contact between the rotor and the stator, the speed at which progressing-cavity pumps can be operated is usually limited.

A twin-screw pump contains a pair of rotating shafts. To minimize axial thrust resulting from hydraulic unbalance, a double-end arrangement in which each shaft is fitted with a pair of opposed helical screws is often utilized. Fluid entering the suction nozzle of a double-end screw pump is frequently divided and admitted to the rotors from both ends of the casing

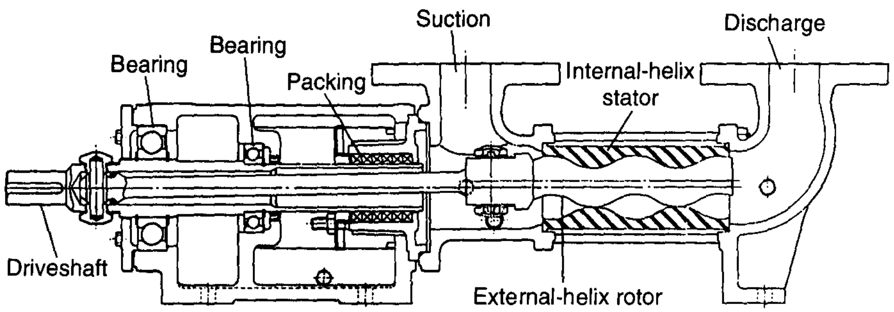


Figure 12-31. Single-screw progressing-cavity pump. Courtesy Ingersoll-Dresser Pump Company.

(figs. 12-32a and 12-32b). The fluid is then forced to advance axially from each end of the pump, is recombined in the center of the casing, and is discharged through the outlet port. With this flow path, the pump's shaft seals must contain fluid that is only at the suction pressure. Alternatively, to reduce the path length of the fluid entering the screws, in applications with poor suction conditions, the casing's suction and discharge connections are sometimes reversed and fluid enters the center of the pump's rotating assembly and is discharged from both ends of the meshing rotors. However, with this latter arrangement, the shaft seals must contain fluid at the pump's discharge pressure.

Although each pumping screw may be pressed onto and pinned to its shaft, a one-piece rotor design in which the screw is an integral part of the

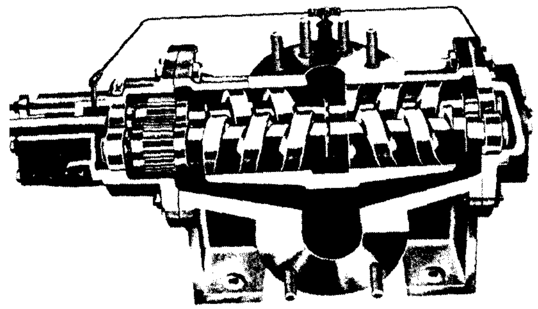


Figure 12-32a. Twin-screw rotary pump with internal bearings and timing gears. Courtesy Ingersoll-Dresser Pump Company.

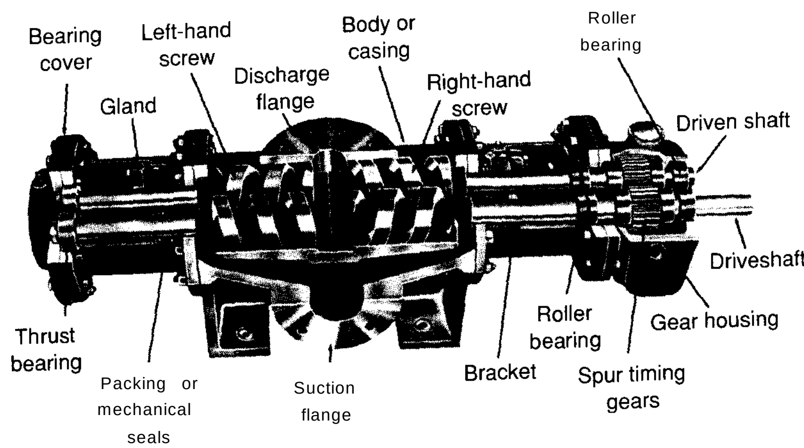
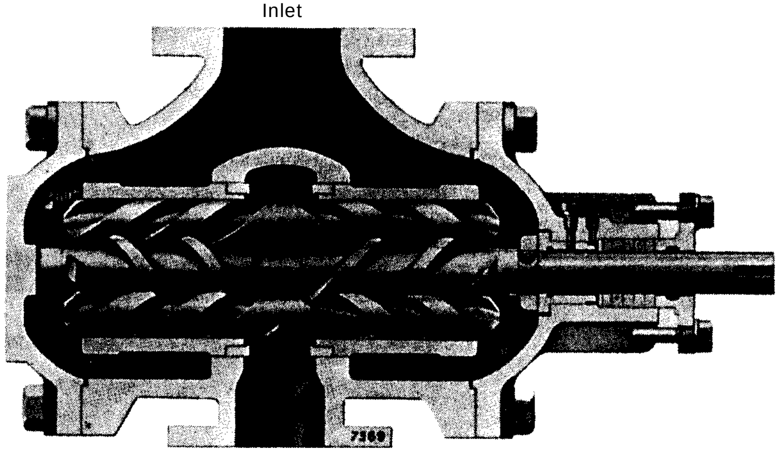


Figure 12-32b. Twin-screw rotary pump with external bearings and timing gears. Courtesy Ingersoll-Dresser Pump Company.

shaft (i.e., the screw threads are machined into the shaft) is frequently used in high-pressure or high-horsepower applications. Loads applied to the pump's rotors are generally absorbed by bearings located at both ends of each shaft. Torque from the pump shaft that is coupled to the driver is often transmitted to the idler shaft through timing gears. The pump's bearings and timing gears are sometimes located within the casing where they are lubricated by the pumped fluid (fig. 12-32a). With this design, only one shaft seal is required to prevent leakage at the point where the driven shaft penetrates the casing. If the pumped fluid is not a suitable lubricant, however, bearings and timing gears must usually be installed in separate housings located outside the pumping chamber and lubricated with oil supplied from an independent source (fig. 12-32b). Four seals are required in a twin-screw pump with external bearings and timing gears to seal the openings where the shafting penetrates the casing. Because contact between the screws and the casing cannot always be avoided, the tips of the threads and the bore of the casing are sometimes coated with various wear-resistant materials.

Pumps containing three screws are typically furnished without timing gears (fig. 12-33). Instead, the helix angle of the pumping threads is large enough to permit torque to be transmitted directly from the driven screw on the center shaft to the idler screws that surround it. Axial thrust is frequently reduced by using either a double-end screw configuration or an internal hydraulic-balancing device when the pump has single-end screws. Each idler screw is typically supported radially by the inner wall of the casing and, therefore, may not be fitted with bearings. An internal or an external bearing, however, is often provided to support the drive end of the

Inlet



Discharge

Figure 12-33. Three-screw rotary pump. Courtesy Imo Pump.

center shaft. Only one shaft sal is required at the location where the center shaft penetrates the housmg.

Gear pumps. Fluid enterig a ge:e: is trapped within the cavities rotating gears and is forced formed between the teeth o. two an external-gear pump, the two pump- through a close-clearance C;Sg. In helical or herringbone design, are ing gears, which can be o t e sur, h fts Torque from the driven shaft mounted on parallel couterrotatmg s te eshing teeth of the two pump- to the idler can be transmItted trough t timing gears (fig 12-34b). In ad- ing gears (fig. 12-34a) or throug .seara e nted at both ends of each shaft, dition, the bearing~ tat are tYPIca e ::;~ can be submerged in and lubri- together with the tImm~ rs, ;;34a) or ounted outside of the pumping cated by the pumpd flm ~ r d from an independent source (fig. chamber and lubncated by o1 supp Ie

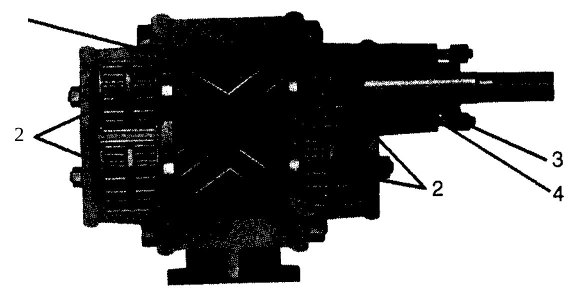
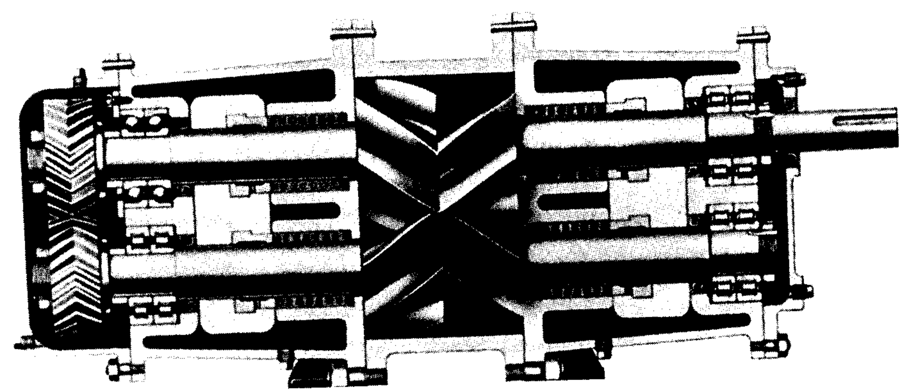


Figure 12-34a. External gear pump with internal bearings. Courtesy Ingersoll-Dresser Pump Compa y.



Fi re 12-34b. External gear pump Wth extern al bearings and timing gu gears. Courtesy Imo Pump-Warren.

12-34b). To enable the required close clearance between the ends of the casing and the faces of the pumping gears to be periodically renewed, external-gear pump casings often have replaceable wear plates.

An internal-gear pump contains both an internal ring gear that rotates against the inner wall of the pump's casing and a second smaller external gear that meshes with and rotates inside the internal gear (fig. 12-35). A stationary crescent-shaped piece is sometimes inserted between the two gears to help seal the fluid-carrying cavities. A bearing is often installed either internally or externally to the pumping chamber to support the coupling end of the driven shaft. Because of the internal-gear pump's single-shaft design, only one casing penetration and shaft seal is required. Torque from the driven rotor, which can be either the external or the internal gear, is transmitted to the mating idler through the meshing gear teeth.

Lobe pumps. Each of the two counterrotating rotors in the typical lobe pump has from one to three large rounded lobes (fig. 12-36). Fluid entering the pump is trapped within the cavities formed between the lobe surfaces and the inner wall of the casing. Rotors can be overhung or mounted between bearings. Because of the shape of the lobes, one rotor cannot drive the other; consequently, torque from the driven shaft to the idler must be transmitted through separate timing gears. Lobe pumps are frequently used in high-capacity, low-pressure applications. In addition, because of the relatively large cavities formed between the rotating lobes, many of these pumps are capable of handling fluids that contain some entrained solids.

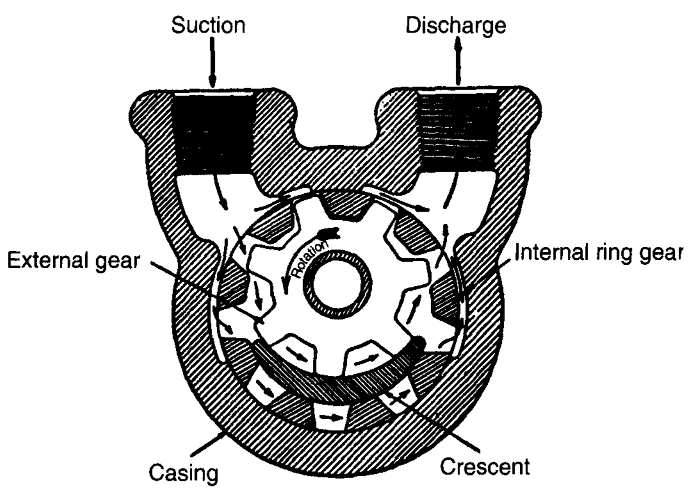


Figure 12-35. Internal gear pump. Courtesy Tuthill Pump Group.

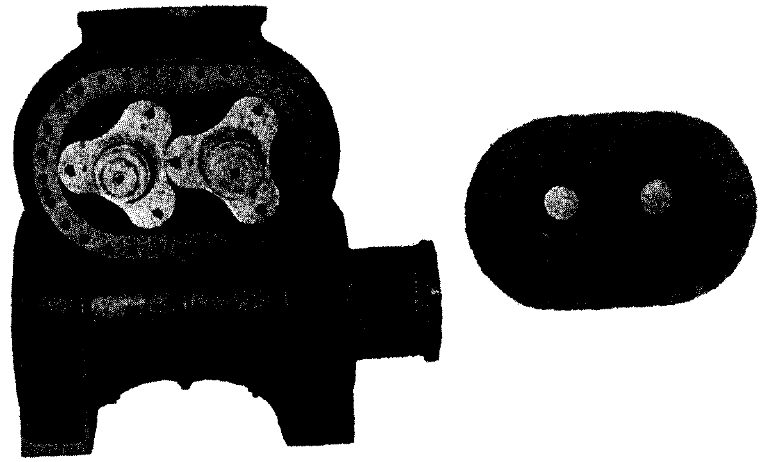


Figure 12-36. Lobe pump. Courtesy Waterous Company.

Sliding-vane pumps. A typical sliding-vane pump contains from four to eight rigid vanes that slide back and forth within radial slots located around the circumference of a cylindrical rotor (fig. 12-37). The rotor is frequently mounted eccentrically within a cylindrical or earn-shaped casing. As the rotor turns, centrifugal force, together, in some cases, with springs, push rods installed between opposing vanes, or the admission of pressurized fluid behind the vanes, keeps the outer tip of each vane in contact with the inner wall of the casing. As each vane sweeps past the casing's inlet port, it follows the contour of the casing and slides radially outward from the rotor. Fluid entering the pump is drawn into the cavity formed between

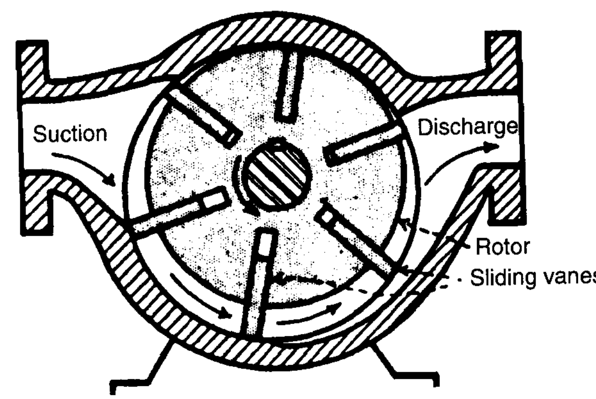


Figure 12-37. Sliding-vane pump.
Courtesy Ingersoll-Dresser Pump Company.

the rotor and the casing's inner wall. When the next vane passes the inlet port, the cavity is sealed and the trapped fluid is forced through the casing. As the cavity approaches the discharge port, the vanes are pushed back into the rotor by the inner wall ofthe casing, and the fluid is expelled from the pump. The rotor can be overhung or mounted between bearings. To eliminate metal-to-metal contact, vanes are often made from plastic or composite materials.

To enable the capacity delivered by a sliding-vane pump to be adjusted during constant-speed operation, in some units, the rotor is mounted within an inner cylindrically shaped ring that can be moved radially with respect to the axis of the pump's shaft and the pump's outer casing. With this arrangement, when the ring is centered around the rotor, the radial clearance between the rotor and the ring's inner wall is constant and the pump's vanes do not slide into or out ofthe rotor as the shaft turns; consequently, no fluid is pumped. However, as the ring is moved off-center and the eccentricity between the inner wall of this ring and the pump's rotor increases, the capacity delivered by the pump will also increase.

Flexible-vane pumps. The rotor in a flexible-vane pump has a series of flexible vanes and rotates within a cam-shaped casing. The typical flexible-vane pump operates in a fashion similar to that of a sliding-vane pump. However, unlike rigid vanes that slide into and out of the rotor, the flexible vanes are forced to deflect by the casing's inner wall. Due to the flexibility of the vanes, flexible-vane pumps are only suitable in low-pressure applications.

Rotary Pump Installation

Alignment. Information given previously regarding the foundation and coupling alignment for centrifugal pumps can also be applied to rotary pumps. When a pump is driven through a multi-V-belt drive (MVD), the pump and its driver should be positioned so that their shafts are parallel. In addition, the grooves in the sheave on the pump's shaft should be aligned with the grooves in the driver's sheave. The distance between the shafts of the pump and driver should produce the proper belt tension. (Typically, belts should be just tight enough to prevent slippage.) Belts should be checked periodically and readjusted or replaced as needed. The driver used with an MVD is sometimes mounted on slide rails and fitted with jacking screws that facilitate driver movement when the tension of the drive belts is adjusted. Because belts stretch after being used, whenever any of the belts in an MVD must be changed, the entire set of belts should be replaced. When installing new belts, the driver should be moved close enough to the pump so that the belts do not have to be forced onto the pump or driver sheaves. After the new belts are in place, the distance between the pump and driver should be adjusted until the proper belt tension is obtained. The belt tension should then be rechecked during the first few days of operation and readjusted, if necessary, to compensate for any initial stretching of the new belts. A belt guard should always be in place whenever a belt-driven machine is operating.

Piping. Suction piping for a rotary pump should be as short and direct as practicable with a minimum of valves and fittings, and it should be airtight. It is usually desirable that straight pipe having a length equal to at least two pipe diameters be installed between the pump's suction connection and any bends or elbows in the suction line. In addition, high spots where air pockets can form should be eliminated. The elevation of horizontal suction piping should preferably be below the liquid source. If the pumped liquid will contain foreign objects that could clog or damage the pump, a strainer with a net flow area equal to three to five times the suction pipe area should be installed in the suction line and cleaned regularly. Both suction and discharge piping diameters should be equal to or one size larger than the corresponding pump connections. Any elbows that are used should be of the long-radius type. If a reducer is used in the suction piping of a pump that operates with a suction lift, it should be of the eccentric type installed with the straight side on top. When possible, the discharge piping should have a vertical rise equal in length to five pipe diameters to prevent air or gas pockets from forming in the pump's casing. Stop valves are frequently installed in both the suction and discharge lines to permit the pump to be isolated from the system for inspection and maintenance. If the backflow of fluid through the pump can occur when the driver is secured, a check valve should be installed in the discharge line between the pump and the stop valve. To permit pump pressures to be measured, suction and discharge pressure gauges should be installed on the pump side of any valves. All piping should be independently supported and properly aligned so that loads imposed on the pump's casing will be minimized and within the pump manufacturer's allowable values. Additionally, piping should always be thoroughly flushed and cleaned prior to pump installation.

Foot valve. When a rotary pump handling low-viscosity fluids operates with a suction lift, a foot valve is sometimes installed near the base of the suction line so that liquid will be retained within the pump's casing when the unit is not operating. However, it is important that the net flow area within the foot valve exceeds the cross-sectional area within the suction line and that the valve's friction losses not result in operation with insufficient NPSHA.

Relief valve. As fluid displaced by a rotary pump's rotor is forced to flow through the casing, the pressure at the pump's discharge port increases until it is sufficient to overcome the system back pressure. Therefore, during normal operation, the discharge pressure developed within a rotary pump matches the requirement of the system. Consequently, if a rotary pump is operated against a closed discharge valve, the pressure ofthe fluid trapped within the pump's casing will continue to rise until the driver is overloaded, the pump or valve fails, or the pump casing or discharge piping bursts. To protect the pump and piping from overpressurization, a relief valve sized to handle the pump's maximum rated capacity should be installed on the discharge side of a rotary pump. Frequently, the pressure at which a discharge reliefv,alve starts to open, which is sometimes referred to as the set or cracking pressure, is adjusted to be equal to approximately 110 percent of the pump's maximum rated discharge pressure. However, the pressure within the pump when the relief valve is fully open and has the full pump capacity passing through it, which is sometimes referred to as the full-flow bypass pressure and is generally greater than the cracking pressure, should always be less than the maximum allowable working pressure of the pump and piping. So that it cannot be isolated from the pump, the relief valve should be located upstream (i.e., on the pump side) of any other valves in the discharge line.

An integral relief valve is sometimes mounted directly on the discharge side of a rotary pump's casing. When the valve opens, it recirculates fluid back to the pump's inlet. However, because the outlet side of an integral relief valve is exposed to fluid at the suction pressure, this type of valve generally opens based on differential pressure and not on discharge pressure, which may not be suitable in a high-suction pressure system. In addition, extended operation with a closed discharge valve can result in the overheating of the pump when fluid passing through the relief valve is being returned directly to the pump's inlet. Consequently, even when a rotary pump is fitted with an integral relief valve, it may be necessary for a separate relief valve with an outlet that is piped back to an unpressurized area ofthe liquid source to be installed in the discharge line.

Rotary Pump Performance Characteristics

Single-pump performance. The total head developed within a rotary pump is often expressed in terms of total differential pressure, which is equal to the total discharge pressure minus the total suction pressure. During constant-speed operation, the volume displaced by the rotors in a rotary pump is constant with respect to time. The capacity actually discharged from a rotary pump operating at a constant speed, however, is reduced by the internal recirculation of fluid back to suction, referred to as slip, which increases as the total differential pressure developed increases and as the viscosity of the pumped fluid is reduced (fig. 12-38).

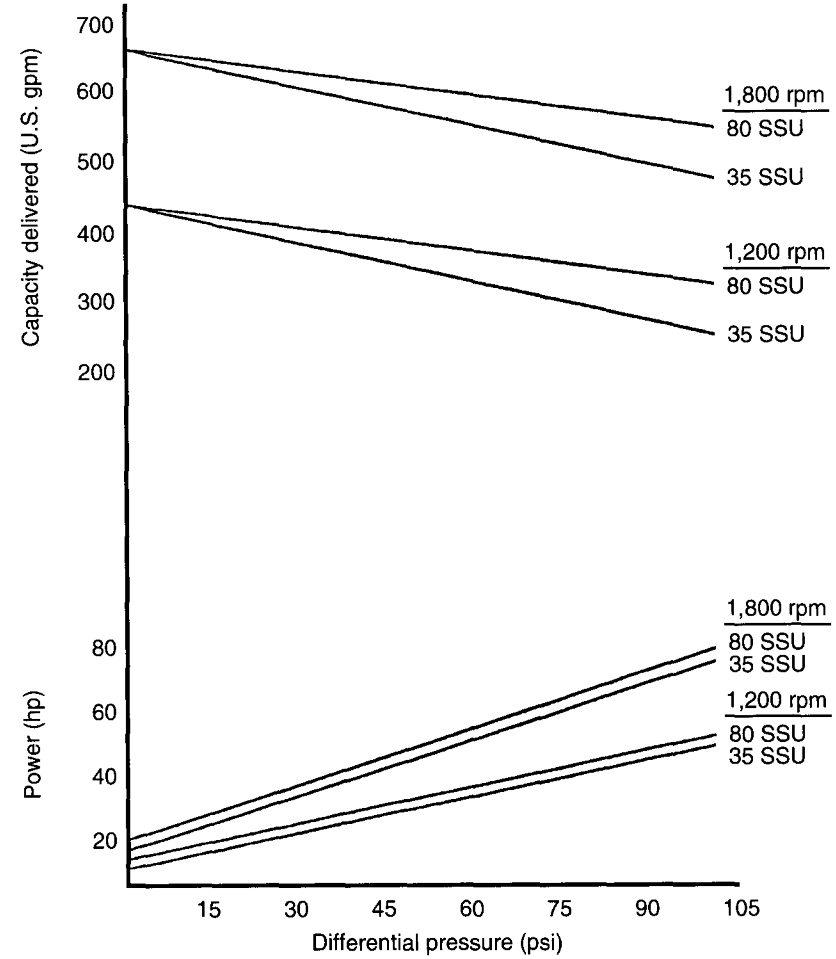


Figure 12-38. Rotary-pump performance curves. Courtesy Ingersoll-Dresser Pump Company.

Power. The power required to drive a rotary pump, which increases with operating speed, the viscosity of the pumped fluid, and the total differential pressure developed by the pump, can be determined as follows:

$$P_P = \frac{p_{td} Q}{C_8 \eta_P} \tag{12.13}$$

where

P_P = power required to drive the pump, hp (kW)
p_{td} = total differential pressure, psi (kPa)

Q = capacity delivered, U.S. gpm (m^3/hr)
C_8 = 1,714 when using the USCS units shown (3,600 for the metric units)
η_p = overall pump efficiency, %/100

To reduce the likelihood of an unexpected overload, a rotary pump's driver is sometimes sized based on the power required by the pump when operating at the maximum rated running speed, with the highest viscosity liquid that will be handled, and with the maximum differential pressure permitted by the discharge relief valve when it is fully open and is handling the full pump capacity ..

EXAMPLE 12-8: A rotary pump delivers a capacity of 200 m^3/hr and develops a total differential pressure of 700 kPa. The pump's efficiency at this operating point is 75 percent. What is the power required to drive the pump?

Solution: Using equation 12.13

$$P_P = \frac{\left(200 \frac{m^3}{hr}\right)(700\ kPa)}{3{,}600(0.75)} = 51.9\ kW$$

Multiple-pump performance. When two or more rotary pumps are operated in parallel, combined pump performance can be determined by adding the capacity delivered individually by each pump when all of the pumps involved are developing the same total differential pressure. Combined series performance can be determined by adding the total differential pressure developed individually by each pump when all of the pumps involved are delivering the same capacity. Some types of rotary pumps, such as internal gear and vane pumps, are available in multistage configurations containing two or more sets of casings and rotors. The output from a multistage rotary pump is equivalent to the output from multiple individual pumps operating in series.

Net positive inlet pressure. The suction condition at the inlet to a rotary pump is often expressed in terms of the net positive inlet pressure available (NPIPA), which is equal to the total absolute suction pressure (i.e., including atmospheric pressure) of the fluid entering the pump minus the fluid's true vapor pressure (also expressed in terms of absolute pressure) at the pumping temperature. The net positive inlet pressure required (NPIPR) by a rotary pump is determined by gradually reducing the inlet pressure to the pump during operation with a constant differential pressure and speed until the pumped capacity or power consumed by the pump's driver is reduced by 5 percent, until the differential pressure developed by the pump or the operating speed become unstable, or until loud or erratic noise is audible.

The value of NPIPA at which any of these conditions first occurs is considered to be the pump's NPIP requirement. The reductions in capacity and power, the performance instability, and the increase in noise during this test (when it is performed with gas-free liquid) are caused by cavitation, which occurs when a portion of the liquid entering the pump flashes and forms vapor bubbles that subsequently collapse when they reach higher-pressure regions within the casing. When it occurs in a rotary pump, cavitation can result in increased vibration, discharge pressure pulsations, and pitting of the rotors and casing. The net positive inlet pressure required by a rotary pump generally drops with reductions in rotor-cavity size, operating speed, and the viscosity of the pumped fluid.

Priming and entrained gases. Although a rotary pump is a positive-displacement pump, for it to be self-priming, a liquid film is generally required to seal internal running clearances. If a rotary pump is started dry, air and other gases within the casing can be recirculated through these clearances, which can extend the priming time indefinitely. In addition, dry operation can often damage a rotary pump's internal components. Therefore, before a rotary pump is started, its casing should be filled with liquid and vented of all gases. This is especially important when the pump operates with a suction lift or a negative inlet pressure. Once the pump is primed, any gas entrained in the pumped fluid will reduce the amount of liquid delivered. The detrimental effect that entrained gas has on a rotary pump's performance increases as the inlet pressure is reduced. In addition, when gas is entrained or dissolved in the fluid entering a rotary pump, cavitation-like symptoms can sometimes be caused by the expansion of the gas in the low-pressure region at the inlet to each of the pump's rotor cavities. Consequently, the NPIP required for the proper performance of a rotary pump can be increased by the presence of gas in the fluid being pumped.

Operation. Prior to starting a rotary pump, the unit should be thoroughly inspected and any abnormalities that are found should be corrected. When bearings or timing gears are oil-lubricated, the LO level in the sump should be checked and oil added if needed. When practicable, the pump shaft should be turned by hand. (In some cases, it may not be possible or advisable to turn the shaft by hand because of the size, configuration, or design of the pump.) Valves in sealing lines, lubrication lines, bearing cooling lines, gauge lines, and similar lines should be opened. In addition, both the suction and the discharge valves should be fully opened, and the pump's casing should be filled with liquid and vented of all gases.

Mter starting a rotary pump's driver, a check should be made to verify that liquid is being discharged from the pump. Capacity adjustments, if required, should be made by varying the pump's operating speed. Typically, the suction and the discharge valves should not be throttled. During operation, the pump should be checked periodically for signs of overheating, leaks, excessive or unusual noise or vibration, loss of suction, and overpressurization. When oil is used for bearing or timing-gear lubrication, the LO temperature and the level in the LO reservoirs should also be checked. In addition, when a forced-feed lubrication system is used, the LO pressure should be monitored, and, when sight glasses are provided, oil flow through the bearings should be observed. When an LO cooler is used, the flow rate of the cooling medium must often be regulated to maintain the proper oil outlet temperature. The pump should be stopped if problems occur.

When shutting down a rotary pump, the driver should be secured following the manufacturer's instructions. The pump's suction and discharge valves, together with other valves associated with the pump, can then be closed. However, if these valves are to be left open so that the pump can be restarted quickly, remotely, or automatically (or if reverse flow can occur when these valves are opened prior to start-up), a check valve should be installed in the discharge line to prevent reverse flow through, and reverse rotation of, the pump.

Troubleshooting. Common reasons for a rotary pump to deliver too low a capacity include operation at too low a speed or with the wrong direction of shaft rotation (when a rotary pump is driven backwards, the flow path through the casing is in the reverse direction), external leakage through casing joints or piping connections, excessive shaft-seal leakage, excessive internal running clearances, the presence of foreign material or air in the pump or suction line, insufficient submergence of the suction pipe, insufficient NPIP available to the pump, a clogged suction strainer, mechanical defects, pumped-liquid viscosity that is too high (which results in an increased resistance to flow) or too low (which results in increased slip), an excessivly high system back pressure, and a leaking relief valve. Reasons for a rotary pump requiring excessive power include rubbing contact between rotating and stationary parts; operation at too high a speed; too high a viscosity of the pumped liquid; excessive shaft misalignment; a bent shaft; improperly supported or misaligned piping; worn, overloaded, or improperly lubricated bearings; overly tightened packing; mechanical defects; foreign matter in the rotors; and an excessively high differential pressure. (Possible causes of a high rotary-pump differential pressure include a partially or fully closed discharge valve and an obstruction in the pump's discharge line.) Reasons for an unusually warm rotary-pump casing or bearing, excessive vibration, and premature mechanical seal or packing failure are similar to those previously given for centrifugal pumps.

ROTARY PISTON PUMPS

Rotary piston pumps have multiple single-acting reciprocating pistons that operate within close-clearance cylinders. In addition, the pistons, together

with their cylinders, also revolve about the pump's axis. Consequently, these pumps are generally classified as rotary pumps.

The axes of the pistons (often seven) in a radial-piston pump are oriented radially around the driveshaft (fig. 12-39a). The outboard end of each piston is pinned to a roller or slipper that is in contact with the inside wall of a nonrotating floating ring. The inboard end of each piston is inserted into a close-clearance cylinder in the liquid cylinder block, which rotates with the driveshaft. When the floating ring is concentric with the shaft, the amount that each piston protrudes into its cylinder remains constant throughout each revolution of the cylinder block. Consequently, the effective stroke of the pistons is zero, and no fluid is pumped. However, when the floating ring is moved off-center, each piston is forced to slide back and forth within its cylinder as it revolves around the shaft axis with the rotating cylinder block. Because of this reciprocating motion, fluid is alternately drawn into and discharged from each cylinder through suction and discharge ports in the stationary housing. The capacity delivered during constant-speed operation increases with the eccentricity of the floating ring with respect to the shaft. Also, the direction of flow through the pump can be reversed by reversing the direction of the eccentricity. Typically, the eccentricity of the floating ring with respect to the pump's shaft and liquid-cylinder block is controlled by the position of a guide yoke that can be adjusted manually or with an automated actuator.

In an axial-piston pump, the single-acting pistons (usually seven or nine), which are equally spaced around the pump's shaft with axes that are parallel to the shaft's axis, are inserted into a multicylinder barrel. The nonpumping end of each piston may be connected through a short connecting rod and a ball-and-socket joint to a socket ring. Alternatively, the connecting rods are sometimes eliminated, and the nonpumping end of each piston is fitted with a slipper pad that bears against a nonrotating cam plate (fig. 12-39b). As the pump's driveshaft turns, the multicylinder barrel and the pistons revolve around the shaft's axis. When the socket ring or cam plate is inclined with respect to the shaft, the pistons on one side of the

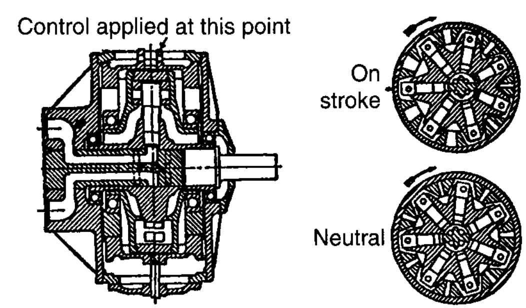


Figure 12-39a. Radial-piston pump. Courtesy Hele-Shaw.

pump are pushed further into the rotating multicylinder barrel than those on the opposite side; consequently, each piston slides back and forth within its cylinder. As a result of this reciprocating motion, fluid is alternately drawn into and discharged from each cylinder through suction and discharge ports in a stationary plate. Each piston's stroke and, therefore, the capacity delivered during constant-speed operation can be increased by increasing the tilt angle of the socket ring or cam plate. In addition, tilting the socket ring or cam plate in the opposite direction reverses the direction offlow through the pump. When the socket ring or cam plate is perpendicular to the shaft, the effective piston stroke length is zero and no fluid is pumped. In some axial-piston pumps, the tilt angle of the socket ring or cam plate is adjustable and can be changed during operation.

RECIPROCATING PUMPS

Liquid End

The liquid end of a reciprocating pump has one or more displacement elements that are moved back and forth within enclosed cylinders. In addition to the cylinders and displacement elements, the pump's liquid end also includes suction and discharge nonreturn valves, which are necessary for proper operation. During each suction stroke, the volume within the liquid cylinder is gradually increased due to the retraction of the displacement element, which results in a reduction in the cylinder's pressure. When this pressure is sufficiently below the pressure at the pump's inlet port, the suction valve opens and fluid in the suction line is drawn into the cylinder. The low pressure created in the cylinder during the suction stroke also acts to keep the discharge valve closed. When the displacement element reaches the end ofthe stroke, its direction oftravel is reversed and a force is applied

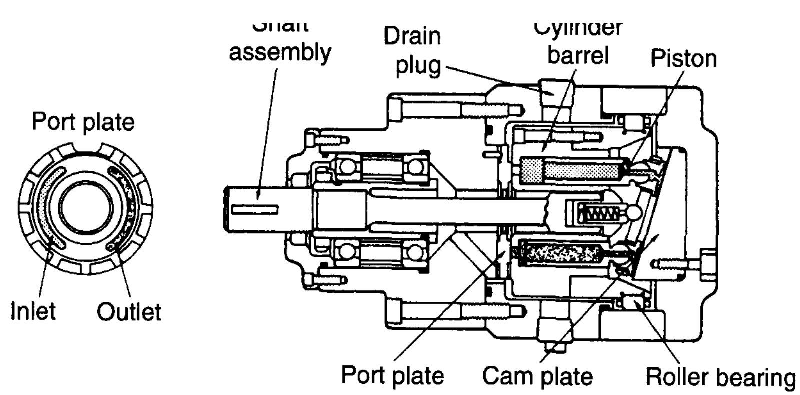


Figure 12-39b. Axial-piston pump. Courtesy Denison Hydraulics.

to the fluid that has filled the cylinder. The pressure within the cylinder increases until it exceeds the pressure at the pump's outlet port by an amount that is sufficient to open the discharge valve. The fluid is then forced through this valve and out ofthe pump. To prevent fluid from being recirculated back to suction, the pressurized fluid within the liquid cylinder acts to hold the suction valve closed. At the end of each stroke of the displacement element, the flow of fluid through the cylinder stops. To increase the pumping capacity, reciprocating pumps are often supplied with multiple cylinders arranged so that their discharge strokes overlap. The moving displacement elements installed in the liquid ends of reciprocating pumps can be either pistons, which typically resemble flat disks, or plungers, which often look like smooth rods. Reciprocating pumps are furnished in both horizontal and vertical configurations, where the direction refers to the orientation of the displacement-element axis.

To maintain the required seal between a piston and the cylinder's inner wall, soft-fibrous or hard-composition packing rings, metallic snap rings, or molded-elastomeric cups are installed in circumferential grooves in the piston's outer surface. So that seal rings can be replaced without removing the piston from the cylinder, they are often held in place with a removable follower that fits over one end ofthe piston's body. In addition, to enable the cylinder's mating surface to be periodically renewed, the liquid cylinder can be fitted with a replaceable liner. To increase the capacity delivered, double-acting pistons that pump fluid on both sides are frequently used. With this arrangement, while one side ofthe piston is drawing fluid into its end of the liquid cylinder, the other side of the piston is discharging fluid from the opposite end ofthe cylinder. This sequence is reversed at the end of the stroke. Consequently, there are two suction strokes and two discharge strokes during every complete reciprocating cycle of the piston. A packed stuffing box is generally provided to reduce fluid leakage at the location where the rod that connects the liquid piston to the pump's drive end penetrates the inboard head of the liquid cylinder.

In a plunger pump, a stationary seal is typically formed around each plunger by square or chevron (V-shaped) packing rings installed within a stuffing box located at the base of the cylinder. To enable these packing rings to be self-adjusting, they are sometimes spring loaded. Secondary backup packing may also used. Although a conventional plunger is a single-acting displacement element, the plungers in some pumps are mounted in pairs that operate within opposed cylinders to create a double-acting effect.

Various types of nonreturn valves are used in reciprocating pumps, such as stem-guided disk valves with flat seats for general service, wing-guided valves with tapered seats for higher-pressure applications, and ball valves and semispherical valves for applications involving viscous or abrasive fluids. With the exception of ball valves, the valve types listed above are typically fitted with spring-loaded retainers. In addition, in some applications, the valves are fitted with elastomeric sealing elements. Reciprocating-pump suction and discharge valves are generally installed within chambers located adjacent to the liquid cylinder and are often arranged so that the suction valves are at a lower elevation than the discharge valves.

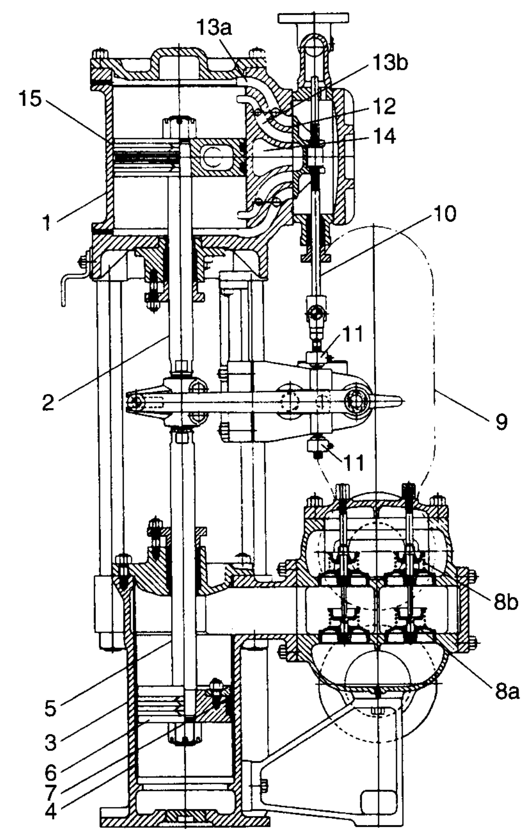
Drive End

Direct-acting pumps. There are two basic types of drive ends used with reciprocating pumps: direct-acting and power. A direct-acting pump's drive end typically has either one (simplex) or two (duplex) cylinders that are each fitted with a close-clearance double-acting piston. Although direct-acting pumps can be driven by a variety of compressed gases, because they are frequently driven by steam, they are commonly referred to as steam pumps. In addition, components in the drive end of a direct-acting pump are often identified as steam parts (e.g., the drive-end valves may be called steam valves).

During normal operation, when steam (or a suitable compressed gas) is admitted into one end of a drive cylinder, it forces the piston to move to the opposite end of the cylinder. Because of this motion, steam on the other side of the drive piston is expelled from the cylinder. At the end of the stroke, steam is admitted to the end of the drive cylinder from which steam was just expelled, and the piston's path is reversed. As this cycle is repeated and steam is alternately admitted to and exhausted from each end of the drive cylinder, the piston within the cylinder continues to move back and forth. The drive piston's reciprocating motion is transmitted, often through piston rods that are joined by a spool-shaped connector, to a piston or plunger that pumps fluid in the adjacent liquid cylinder. The flow of steam into and out of a direct-acting pump's drive cylinder is regulated by a valve that is installed within a chamber, referred to as the steam chest, located on the side of the cylinder. To limit leakage, stuffing boxes with replaceable packing rings are typically provided at locations where drive piston rods enter the cylinders and where valve rods penetrate the steam chest. When steam is used as the driving medium, steam inlet and exhaust piping must be designed to minimize loads imposed on the pump due to thermal expansion.

In a duplex direct-acting pump, each drive cylinder's steam valve is actuated mechanically through a lever or rocker arm, crank, and valve rod by the movement of the opposite cylinder's piston rod (fig. 12-40a). As the valve is moved back and forth, it alternately covers and uncovers the inlet ports through which steam enters each end of the cylinder. It also alternately connects each of the cylinder's two exhaust ports, which are located inboard of the inlet ports, to the steam chest's outlet port located adjacent to the middle position of the valve. Although flat-faced unbalanced slide

valves are often used, larger pumps and pumps driven by high-pressure or high-temperature steam may be fitted with balanced piston-type valves. The steam valves and linkages are generally designed and adjusted so that the strokes of the two drive pistons overlap. With this arrangement, the inlet ports for both cylinders are never covered simultaneously; consequently, during normal operation, steam is always being admitted into at least one of the drive cylinders. In addition, because at least one of the two drive pistons is always in motion, fluid is always being discharged from at least one of the pump's liquid cylinders. Lost motion, or play, is typically provided in the drive-end valve linkages to allow each valve to remain stationary during a portion, often approximately one-quarter to one-half, of the stroke of the piston that actuates it. This permits the admission of steam into each drive cylinder to continue throughout the entire stroke of



1 Steam cylinder
2 Steam rod
3 Liquid cylinder
4 Liquid cylinder liner
5 Liquid piston rod
6 Liquid piston
7 Liquid piston packing
8a Liquid suction valve
8b Liquid discharge valve
9 Pulsation-dampening air chamber
10 Valve rod
11 Lost-motion adjusting nuts
12 Steam valve
13a Steam-cylinder inlet port
13b Steam-cylinder exhaust port
14 Steam chest exhaust port
15 Steam piston

Figure 12-40a. Duplex direct-acting (steam) pump. Courtesy Ingersoll-Dresser Pump Company.

the piston within the cylinder. Furthermore, it gives the valves in the pump's liquid end time to seat quietly at the end of each stroke.

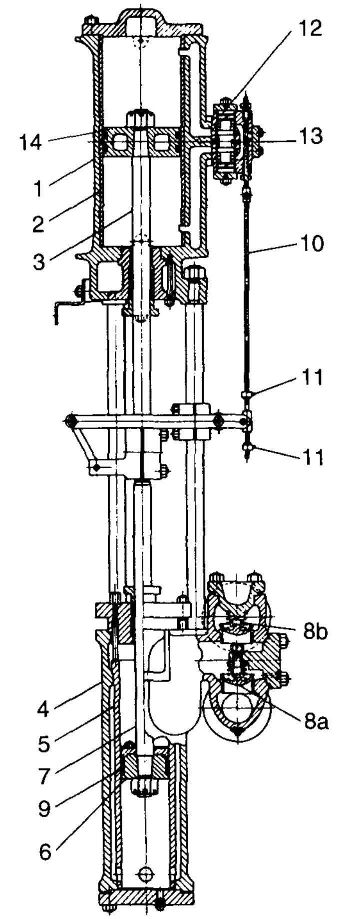
To set, or adjust, the drive-end or steam valves in a typical duplex direct-acting pump, the steam-chest cover must be removed and, with each drive piston at the midstroke position within its respective cylinder, the drive-end valve rods should be adjusted until each steam valve just covers both of its cylinder's inlet ports. Ordinarily, ifthe pump has a fixed amount oflost motion, during the steam-valve adjustment, a tappet or nut on each drive cylinder's valve rod should be centered between lugs on the corresponding steam valve. LQst motion in this type of pump can be changed only by changing the thickness of the valve-rod tappet or nut (that is, reducing the nut thickness increases lost motion). Alternatively, when a pump has inside-adjustable lost motion, lost motion can be adjusted by changing the clearances between nuts on each drive-end valve rod and lugs on the corresponding steam valve. Moving the two nuts on each valve rod farther apart increases the valve-to-nut clearances and the pump's lost motion. For an initial adjustment, the nut clearance on each side of the steam valve can be set equal to one half the width of the cylinder's inlet port. A third option, which is incorporated in the design of many larger steam pumps, is outside-adjustable lost motion. With this arrangement, lost motion can be increased by increasing the clearances between adjusting nuts located on the external portion of each drive-end valve-rod link and a tappet connected to the valve-rod link's actuating crank. Initially, the nut clearance on each side ofthe tappet, which can be adjusted with the steam-chest cover in place, is often set equal to one-half the width of the cylinder's inlet port.

Mter a duplex direct-acting pump's steam valves have been adjusted, one valve must be pushed off-center before the steam-chest cover is replaced so that the pump can be started. The pump should then be operated to verify that the steam valves are operating properly and to determine, when applicable, if additional lost-motion adjustments are necessary. An increase in the lost motion of a duplex direct-acting pump increases the duration ofthe pause at the end of each piston stroke and reduces the amount that the strokes of the two drive pistons overlap.

Typically, the exhaust port at each end of a duplex-pump drive cylinder is located inboard ofthe adjacent inlet port and is covered by the piston operating within the cylinder before the piston's stroke has been completed. Steam that remains within the cylinder acts as a cushion between the piston and the nearby cylinder head and smoothly decelerates the pump's moving parts. To enable the cushioning steam to be exhausted through the uncovered inlet port, a bypass that connects adjacent inlet and exhaust ports is sometimes provided at each end ofthe drive cylinder. In addition, a valve may be installed in each bypass to permit the rate at which the cushioning steam escapes from the drive cylinder to be adjusted. Opening

a duplex direct-acting pump's cushion valves reduces the cushion and lengthens the piston stroke. Conversely, closing the valves increases the cushion and shortens the stroke. To maximize a duplex direct-acting pump's capacity, cushion valves, when provided, should generally be adjusted to give the longest stroke that does not result in contact between the drive pistons and the cylinder heads.

In contrast to a duplex pump, the main valve in the drive end of a simplex direct-acting pump is a piston-type valve that is moved by the steam used to drive the pump. The admission of steam to either end ofthis valve is controlled by a separate pilot valve that is linked to and moved by the drive-end piston rod (fig. 12-40b). Before the drive piston reaches the end of



1 Steam cylinder
2 Steam cylinder liner
3 Steam rod
4 Liquid cylinder
5 Liquid cylinder liner
6 Liquid piston
7 Liquid piston rod
8a Liquid suction valve
8b Liquid discharge valve
9 Liquid piston package
10 Valve rod
11 Lost-motion adjusting nuts
12 Steam piston valve
13 Steam pilot valve
14 Steam piston

Figure 12-40b. Simplex direct-acting (steam) pump.
Courtesy Ingersoll-Dresser Pump Company.

each stroke, the pilot valve is repositioned, and steam is directed to one end of the main piston valve. The steam moves the main piston valve to the opposite end of the steam chest, which changes the end of the drive cylinder into which steam is admitted. Although flat slide-type pilot valves are often used, if the valve is large or the steam pressure and temperature are high, the pump may be fitted with a balanced piston-type pilot valve. Lost motion in a simplex direct-acting pump's pilot-valve linkage generally results from the clearances between adjusting nuts on the external portion of the valve rod and a tappet on the lever that moves the rod; it is frequently increased by moving these nuts farther apart until the drive piston has the longest stroke that does not result in its contact with the cylinder heads.

Each end of a simplex direct-acting pump's drive cylinder often has one main port and a smaller starting port. At the beginning of each stroke, the main port is covered by the drive piston, and steam is admitted into the cylinder at a reduced rate through the adjacent starting port. The pump's moving parts, therefore, accelerate slowly. Prior to the completion of the stroke, the larger main port at the opposite end ofthe drive cylinder is covered by the piston, and steam that remains within the cylinder is exhausted at a reduced rate through the smaller port. This steam acts as a cushion between the piston and the cylinder head and slowly decelerates the pump's moving parts. At each end of the drive cylinder, a valve, referred to as a cushion valve, is sometimes installed in a bypass connecting the smaller starting port with the adjacent larger main port. As a simplex direct-acting pump's cushion valves are closed, the acceleration and deceleration rates of the pump's moving parts are both reduced.

Power pumps. In a power pump, a crankshaft and connecting rods convert the rotary motion of the driver's shaft to reciprocating motion that is transmitted to the liquid end's plungers or pistons (fig. 12-41). The crankshaft is typically supported at each end by either a tapered roller bearing or a sleeve-type journal bearing. Additional intermediate bearings may also be installed along the length of the crankshaft. Replaceable sleeve bearings are generally fitted into both ends of each connecting rod. In addition, a crosshead that absorbs side loads is frequently installed between each connecting rod and the corresponding plunger or piston rod.

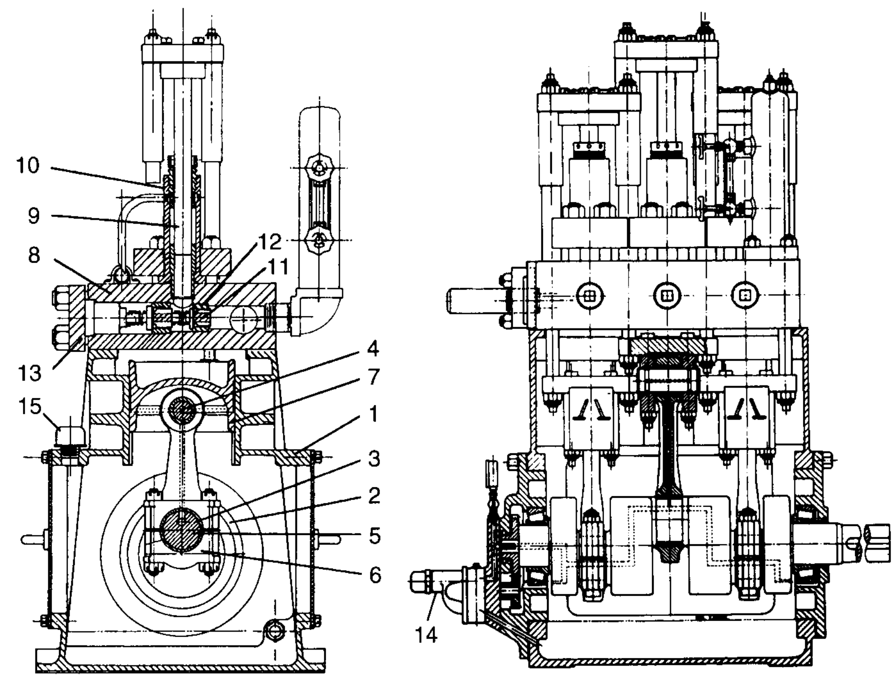
A multi-V-belt drive, fluid couplings, or a set of reduction gears is often used to enable a power pump to operate at a speed below that of its driver. Large power pumps are frequently furnished with integral reduction gears. These gears, together with the pump's bearings and crossheads, are generally lubricated by oil stored within the power end's crankcase. A rotary pump that is geared to the crankshaft may be provided to deliver pressurized lubricating oil to the power end's running gear. Some pumps are also fitted with an integral LO cooler. Power-

pump liquid ends ordinarily have two double-acting pistons or three (triplex), five (quintuplex), seven (septuplex), or nine (nonuplex) single-acting plungers.

Reciprocating Pump Installation

Alignment. Information given previously regarding pump foundations and coupling and multi-V-belt alignment for centrifugal and rotary pumps can also be applied to reciprocating pumps.

Piping. All piping should be independently supported and properly aligned so that loads imposed on the pump will be minimized and will be



1 Frame	6 Connecting rod	11 Valve
2 Crankshaft bearing	7 Crosshead	12 Valve seat
3 Crankpin bearing	8 Cylinder block	13 Valve cover
4 Crosshead pin	9 Plunger	14 LO pump
5 Crankshaft	10 Stuffing box	15 Vent cap

Figure 12-41. Reciprocating power pump.
Courtesy Ingersoll-Dresser Pump Company.

within the pump manufacturer's allowable values. This is especially important when steam piping is connected to the drive end of a direct-acting pump. Additionally, piping should always be thoroughly flushed and cleaned prior to pump installation. Suction piping should be as short and direct as practicable with a minimum of valves and fittings, and it should be airtight. In addition, high spots where air pockets can form should be eliminated. The elevation of horizontal suction piping should preferably be below the liquid source. Operation with a plugged strainer can damage a reciprocating pump; consequently, the use of a suction strainer is not recommended unless the pumped liquid will contain foreign objects that could clog or damage the pump. A strainer, if used, should have a net flow area equal to four to five times the suction pipe area and should be cleaned regularly.

Suction and discharge piping diameters should be at least equal to and preferably larger than the sizes of the corresponding pump connections. Any elbows that are used should be of the long-radius type. If a reducer is used in the suction piping of a pump that operates with a suction lift, it should be of the eccentric type installed with the straight side on top. Because fluid is not discharged from a reciprocating pump's liquid cylinders at a steady rate, there are pulsations in the flow rate and pressure within the pump's suction and discharge lines. These pulsations are sometimes reduced by installing air-charged chambers, referred to as pulsation dampeners, adjacent to the pump's suction and discharge ports. Stop valves are often installed in both suction and discharge lines to permit the pump to be isolated from the system for inspection and maintenance. To prevent overpressurization, a properly sized relief valve should be installed in a reciprocating pump's discharge piping, upstream (Le., on the pump side) of any stop valves. Also, to prevent liquid in the discharge line from resting on the pump's liquid-end discharge valves, a check valve is frequently installed in the discharge line between the relief valve and the discharge stop valve. Furthermore, a bypass line with a valve that can be opened to return discharged fluid back to suction and reduce the driver load during start-up may be connected to a power pump's discharge line between the relief valve and the check valve. To permit pump pressures to be measured, suction and discharge pressure gauges are ordinarily installed on the pump side of any valves.

Reciprocating Pump Performance Characteristics

Single-pump performance. The volume displaced per cycle in the liquid end of a reciprocating pump, which is constant, increases with the outside diameter and stroke length of the liquid piston or plunger. These dimensions can often be determined from a reciprocating pump's nomenclature. A direct-acting pump is typically identified by three numbers: the first equals the drive-end piston's outside diameter, the second equals the outside diameter of the liquid piston or plunger, and the third equals the length ofthe stroke. Because it does not have drive-end cylinders, a power pump is identified by only the last two of these numbers. The dimensions included in the nomenclature are typically in inches or mm.

As a result of slip (leakage and backflow through the liquid end's suction and discharge valves, together with leakage across the internal seals in a piston pump), leakage at the liquid end's external stuffing boxes, and the compressibility of the pumped fluid, the actual average capacity delivered by most reciprocating pumps is less than the theoretical value based on the displaced volume. If the volume within liquid cylinders displaced by piston rods is ignored, the average capacity delivered by a reciprocating pump can be estimated from the following:

$$\overline{Q} = \frac{f}{C_9}\left[\frac{\pi (d_L)^2}{4}\right] S N n \eta_V \tag{12.14}$$

where

- $\overline{Q}$ = average capacity delivered, U.S. gpm (m^3/hr)
- f = 1 for plungers or single-acting pistons, 2 for double-acting pistons
- C_9 = 231 when using the USCS units shown (1.67E+7 for the metric units)
- π = 3.1416
- d_L = piston or plunger outside diameter, in. (mm)
- S = stroke length, in. (mm)
- N = cycles per cylinder per min (equal to strokes per min/2), cpm
- n = number of liquid cylinders
- η_V = volumetric efficiency = actual capacity/theoretical capacity, %/100

During normal operation, the pressure of fluid within a reciprocating pump's liquid cylinders is increased until it is sufficiently above the pressure in the pump's discharge line to open the liquid-end discharge valves. Therefore, the pump's discharge pressure is approximately equal to and changes with the system back pressure. If the reduction of piston area because of piston rods is ignored, the maximum differential pressure that can ordinarily be developed in the liquid end of a direct-acting pump fitted with either double-acting pistons or opposed plungers is approximately equal to the following:

$$p_{td,\max} = \Delta p_D \eta_m \left(\frac{d_D}{d_L}\right)^2 \tag{12.15}$$

where

$p_{td,\max}$ = maximum liquid-end total differential pressure, psi (kPa)
Δp_D = drive-end net or differential steam or gas pressure, psi (kPa)
η_m = pump mechanical efficiency, %/100
d_D = drive-end piston outside diameter, in. (mm)

To equalize the effective area on both sides of a double-acting liquid piston, a tail rod equal in diameter to the piston rod is sometimes added to the outboard side of the piston. The opening for the tail rod in the liquid cylinder's outboard head is typically sealed with a packed stuffing box. When there is excessive leakage past the rings on a double-acting liquid piston not fitted with a tail rod, the pressure on both sides of the piston can equalize and the effective piston diameter can be reduced to that of the piston rod, which significantly increases the maximum pressure that can be developed within the liquid cylinder. This maximum pressure, referred to as the ram pressure, can be estimated with equation 12.15 by substituting the piston-rod diameter for d_L.

EXAMPLE 12-9: A 14 in. × 10 in. × 12 in. duplex direct-acting reciprocating pump fitted with double-acting liquid pistons is operated at a speed of 30 cpm while being driven by steam at a pressure of 100 psig. The steam exhaust pressure is 15 psig. The pump's volumetric efficiency is 90 percent, and its mechanical efficiency is 70 percent. The liquid-piston rod diameter is 1.5 in. The pump's liquid pistons do not have tail rods. Estimate the following:

1. The average capacity delivered by the pump.
2. The maximum total differential pressure that can be developed by the pump's liquid end.
3. The ram pressure that could be developed if the liquid-piston rings fail.

Solution: Using equation 12.14 to answer question 1 and equation 12.15 to answer questions 2 and 3

$$\bar{Q} = \frac{2}{231}\left[\frac{\pi(10\text{ in.})^2}{4}\right]12\text{ in.}(30\text{ cpm})\,2(0.90) = 441\text{ U.S. gpm}$$

$$p_{td,\max} = (100\text{ psig} - 15\text{ psig})\,0.70\left(\frac{14\text{ in.}}{10\text{ in.}}\right)^2 = 117\text{ psi}$$

$$p_{\text{ram}} = (100\text{ psig} - 15\text{ psig})\,0.70\left(\frac{14\text{ in.}}{1.5\text{ in.}}\right)^2 = 5{,}183\text{ psi}$$

Moisture in saturated steam that is used to drive a direct-acting pump helps to lubricate the drive-end pistons and valves. If superheated steam or a compressed gas is used as the driving medium, a small amount of oil is often mixed with it to lubricate the drive end's internal components. In some cases, this oil is injected directly into the steam chest by a mechanicallubricator that is mounted on the pump and actuated by the movement of the piston rods.

The power that must be applied to a power pump'~ shaft can be calculated using equation 12.13, except that for a reciprocating pump, the value used for Q in this equation is the volumetric capacity measured at the pump's suction conditions. The power required by a power pump can fluctuate in a cyclicfashion during operation because of pulsations in the pressure developed and capacity delivered by the pump. Unloaders that hold a liquid cylinder's suction valves open or bypass valves that recirculate discharged fluid back to suction are sometimes used to reduce the discharge pressure that must be developed during start-up, which reduces the torque required to accelerate a power pump's drive end.

Net positive suction head. The NPSH required by a reciprocating pump is determined by gradually reducing the NPSH available to the pump during operation with a constant discharge pressure and at a constant speed. The value of NPSHA at which the capacity delivered by the pump is reduced by 3 percent is considered to be the pump's NPSH requirement at the test speed. The capacity delivered during this test (when it is performed with gas-free liquid) is reduced by cavitation, which occurs when a portion of the liquid entering each cylinder flashes and forms vapor bubbles that subsequently collapse when the pressure within the cylinder increases. The NPSH required by a reciprocating pump normally drops with reductions in operating speed. In addition, values of NPSHR can sometimes be reduced by using lighter suction-valve springs in the pump's liquid end. However, when lighter springs are used, the speed at which the suction valves close at the end of each suction stroke can be reduced, which can lead to an increase in the backflow through these valves and a reduction of the pump's volumetric efficiency. Cavitation occurring during operation with insufficient NPSHA can lead to the pitting of pistons or plungers and to mechanical damage from increased vibration.

Pressure pulsations and velocity changes that occur within the suction piping as the liquid-end suction valves repeatedly open and close have a detrimental effect on the net positive suction head available to a reciprocating pump. To account for this effect, the acceleration head, which represents the energy required to accelerate the fluid in the suction line, can be subtracted from values of NPSHA calculated using equation 12.7. The maximum acceleration head, which occurs at the beginning of a suction stroke, can be estimated using the following:

$$H_{ac} = \frac{NC}{Kg} \sum_{i=1}^{m} \left(L_{s,i} V_{s,i} \right) \tag{12.16}$$

where

H_{ac} = acceleration head, ft (m)
N = pump operating speed, rpm (for power pumps) or cpm (for direct-acting pumps)
C = constant based on the pump's liquid-end configuration (e.g., 0.200 for simplex and 0.060 for duplex direct-acting pumps with double-acting pistons; 0.066 for triplex, 0.040 for quintuplex, 0.028 for septuplex, and 0.022 for nonuplex power pumps with plungers and standard ratios of connecting-rod length over crank radius)
K = factor representing the relative compressibility of the pumped fluid (e.g., 1.4 for hot water, 2.5 for hot oil)
g = acceleration from gravity, 32.2 ft/s^2 (9.81 m/s^2)
m = number of suction-line sections with different diameters
$L_{s,i}$ = length of suction piping section i, ft (m)
$V_{s,i}$ = average velocity of fluid within the suction piping section i, ft/s (m/s)

The accuracy of equation 12.16 is reduced as the length of the suction pipe increases. In addition, acceleration head can frequently be reduced by installing a properly sized pulsation dampener in the suction line close to the pump's inlet.

EXAMPLE 12-10: A triplex plunger power pump operating at 360 rpm handles hot water. The suction line has a 4-in-diameter, 3-ft-long section and a 5-in-diameter, 5-ft-long section. The velocity in the 4-in-diameter section is 2.5 ft/s and the velocity in the 5-in-diameter section is 1.6 ft/s. Determine the acceleration head.

Solution: Using equation 12.16

$$H_{ac} = \frac{360 \text{ rpm } (0.066)}{1.4\left(32.2 \dfrac{\text{ft}}{s^2}\right)} \left[3 \text{ ft } (2.5 \text{ ft/s}) + 5 \text{ ft } (1.6 \text{ ft/s})\right] = 8.2 \text{ ft}$$

Priming capability. Because reciprocating pumps are positive-displacement pumps, they are typically capable of delivering liquids that contain entrained gases. However, if a reciprocating pump that operates with a suction lift is started with gas or vapor in its liquid cylinders and with liquid resting on its discharge valves, the pressure developed during each discharge stroke may not be sufficient to open these valves and expel the gas from the pump. In addition, as the gas in the cylinders reexpands during each suction stroke, the cylinder pressure may not be reduced sufficiently to permit the suction valves to open. If this occurs, fluid in the suction line will not be drawn into the pump, and the pump will be unable to prime itself. When there is a large clearance volume between the inner walls of a reciprocating-pump liquid cylinder and the fully extended piston or plunger, more gas or vapor can remain in the cylinder at the end of a discharge stroke and the ability of the pump to prime itself is reduced. Consequently, reciprocating pumps with liquid pistons that operate in close-clearance cylinders typically have self-priming characteristics that are superior to those of plunger pumps, which are often not self-priming. In some cases, a foot valve may be installed in the suction line of a reciprocating pump that operates with a suction lift so that liquid will be retained in the line when the pump is shut down.

Operation. Prior to starting a reciprocating pump, the unit should be thoroughly inspected and any abnormalities found should be corrected. Valves in the suction and discharge piping should be fully opened. The liquid-end suction-valve chamber should be filled with liquid and vented of all gases. To enable the pump to start with a low discharge pressure, if possible, liquid that may be in the discharge line should be prevented from resting on the pump's liquid-end discharge valves. (A check valve is frequently installed in the discharge line for this purpose.) Allowing the pump to start without pressure on its discharge valves is especially important when operating with a suction lift. Any air in the pump's cylinders can often be vented through a connection in the discharge-valve chamber immediately after the pump is started.

Before starting a direct-acting pump, valves in the drive-end exhaust piping should also be opened and joints in the steam-valve gear should be lubricated. In addition, when steam is the driving medium, the steam supply and exhaust lines should be drained of condensate, and drains in the pump's drive cylinders should be opened. At start-up, the steam supply valve should initially be only cracked open to permit the pump to warm up slowly. Drive-cylinder drain valves should be closed when condensate is no longer draining from the pump.

Before starting a power pump, the LO level should be checked, and oil should be added if needed. Valves in sealing lines, bearing cooling lines, and similar lines should be opened. If a bypass valve is installed in the pump's discharge line, it should also be opened to reduce the initial load on the driver. The bypass valve should normally be closed after the pump is running at its normal operating speed. (Suction-valve unloaders may also be provided to reduce the starting load on a power pump's driver.)

Mter starting a reciprocating pump, a check should be made to verify that liquid is being discharged from the pump. In general, the capacity delivered by a reciprocating pump is adjusted by varying the pump's operating speed. This can be done with a direct-acting pump by adjusting the rate at which the driving medium is admitted into the drive-end cylinders. To enable a power pump's operating speed to be changed, a variable- or multispeed driver may be used, or a fluid coupling may be installed between the pump and driver. Alternatively, the capacity delivered by some power pumps can be adjusted with an internal linkage that enables the plungr~ stroke. to be changed. In addition, some power pumps have synchromzed suctIOn-valve unloaders that hold the suction valves open in selected liquid cylinders w:Q.en the capacity delivered is to be reduced. While it is running, a reciprocating pump should be checked periodically for signs of overheating, leaks, excessive or unusual noise or vibration loss of suction, and overpressurization. When oil is used for lubrication, the LO temperature and the oil level in the LO reservoir should be checked. In addition, when a forced-feed lubrication system is used, the LO pressure should be monitored, and, when sight glasses are provided, oil flow trough the bearings should be observed. Also, when an LO cooler is provIded, the flow rate of the cooling medium passing through the cooler must often be regulated to maintain the proper oil outlet temperature. The pump should be stopped if problems occur.

Troubleshooting. Common reasons for a reciprocating pump to deliver too low a capacity include worn piston rings or packing, excessive internal running clearances, leaking liquid-end valves, operation at too low a speed, an excessively high liquid-end differential pressure, leaking or improperly adjusted drive-end valves (for direct-acting pumps), a low driving-medium supply pressure or a high driving-medium exhaust back pressure (for direct.-acting pumps), air or foreign matter in the pump or suction line, insufficIent submergence ofthe suction pipe, insufficient NPSHA, mechanical defects, a clogged suction strainer, and a leaking relief valve. Reasons for a reciprocating pump to require excessive power include rubbing contet and binding between moving and stationary parts; operation at too hIgh a speed; too high a viscosity ofthe pumped liquid; excessive misalignment between the liquid and drive ends; a bent shaft or piston rod; improperly supported or misaligned piping; worn, overloaded, or improperly lubricated bearings or reduction gears; overly tightened packing; mechanical defects; foreign matter in the cylinders; and operation with too high a liquid-end differential pressure.

DIAPHRAGM PUMPS

A diaphragm pump is a positive-displacement pump in which fluid is pumped by the reciprocating motion of a flexible membrane. In a typical

air-operated double-diaphragm pump, two diaphragms wi:hin enclose~ chambers are mounted side-by-side (fig. 12-42). Each pumpmg chamber s inlet and outlet ports are sealed during the discharge and suction strokes, respectively, with nonreturn check valves. Flap, ball, or poppe-type valves are frequently used. During normal operation, as comprssed aIr (or anoter suitable gas) is admitted into the inner chamber behmd one of the diaphragms, the diaphragm is forced to deflect away fr.om the center ?f the pump. Because of this motion, the volume of the pumpm~ chamb.er adJcet~ to the outboard side of the diaphragm is reduced, and flUid contamed Wlthm the chamber is discharged through the internal discharge valve and the pump's outlet port. At the same time, a shaft connecting the two diapragms pulls the second diaphragm toward the center of the pum~ epadmg the volume of the opposite pumping chamber. The pressure withm thIS pumping chamber drops, and fluid is drawn through the internal section valve and into the chamber. In addition, air in the inner chamber behmd the second diaphragm is exhausted from the pump. When the diaphrgms reach the end of the stroke, air is readmitted to the inner chamber behmd the se~ ond diaphragm, and the motion of both diaphrgms is eversed. The flUI~ that had entered the second pumping chamber IS now discharged. In addItion, fluid is drawn into the first pumping chamber, and the air initially admitted behind the first diaphragm is exhausted from the pump.

In some pumps, the alternating admission and exhaust of the driving air to the two inner diaphragm chambers, which is necessary for pump operation to continue is controlled by an air-actuated piston-type distribution valve. In additin, the flow ofthe compressed air that moves this valve is controlled by the position of two notches that are machined into the shat~ connecting the two diaphragms. At the end of a stroke, one ofthe notches IS

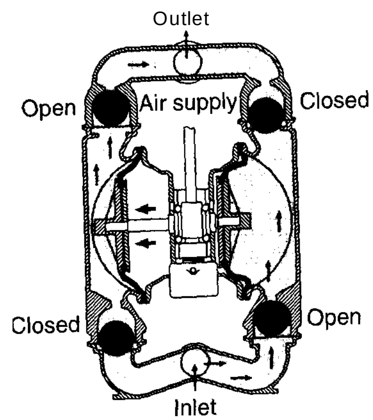


Figure 12-42. Air-operated diaphragm pump.
Courtesy Wilden Pump & Engineering Company.

located in a position where it allows some of the compressed air being supplied as the driving medium to be directed to one end of the air-actuated distribution valve. Mter the compressed air shifts this distribution valve, the inner diaphragm chambers that air is admitted to and exhausted from are reversed, which reverses the direction of pumping diaphragm deflection. At the end ofthe next stroke, the opposite notch on the shaft directs air to the other end of the air-actuated distribution valve, the air shifts the valve, and the diaphragm motion is again reversed. At the end of the second stroke, the cycle is repeated.

During normal operation, an air-operated diaphragm pump's discharge pressure equals the system back pressure. In addition, although the maximum discharge pressure that can be developed in many diaphragm pumps is limited to the pressure of the compressed air used to drive the pump, some designs are available that enable the pump to develop a discharge pressure that is greater than the driving air pressure. A typical driving-air pressure is 125 psig (862 kPa gauge).

With a constant driving air pressure, the capacity delivered by a typical pump increases as the pump's discharge pressure is reduced. Also, the pumped capacity increases with the rate at which the compressed air is admitted into the pump. However, if a diaphragm pump is operated at too high a speed, it may have difficulty drawing fluid into the pumping chambers. For this reason, a diaphragm pump operating with a suction lift should generally be started and operated at a low speed until the pump is completely primed. In addition, although it may be possible to increase the pump speed after the pump is primed, any increase should be made slowly and should be limited to a speed that does not result in cavitation or in a partial or complete loss of suction.

The air line connected to an air-operated diaphragm pump should be large enough to supply the volumetric flow rate necessary for the pump to deliver the desired flow rate (i.e., to operate at the desired speed). A filter should usually be installed in the air supply line upstream of the pump to protect the pump from contaminants in the air. Also, a lubricator that adds a small amount of oil to the incoming supply air should be installed after the filter to reduce air-distribution-valve friction and wear. The grade of lubricating oil used, which should always meet the pump manufacturer's recommendations, should generally be suitable for low-temperature applications. As driving air passes through the air distribution valve, its pressure and temperature drop. Moisture present in the air can form ice and lead to sticking of the air valve. To prevent this, a drier should also be installed in the air supply line if wet air will be used to drive the pump.

In a power diaphragm pump, each flexible pumping diaphragm is forced to move back and forth by hydraulic fluid that is alternately pumped into and out of the chamber behind the diaphragm. The flow of the hydraulic fluid results from the reciprocating motion of pistons or plungers that

are connected to a rotating crankshaft. In most cases, the crankshaft, which is frequently an integral part of the power-diaphragm-pump assembly, is driven by an electric motor. The effective stroke length of each piston or plunger is sometimes adjustable so that the capacity delivered by the diaphragm end of the pump can be regulated without changing the speed at which the crankshaft is driven. To minimize flow pulsations, power diaphragm pumps are generally furnished with two or more pumping diaphragms. Also, power diaphragm pumps are typically capable of developing discharge pressures that are much higher than the pressures developed by air-operated diaphragm pumps.

Pump Materials of Construction

Although the materials used in the construction of any marine pump should be selected based on the specific application, there are some general guidelines:

- Ductile materials are generally preferred.
- Wetted components (i.e., components exposed to the pumped fluid) should be resistant to corrosion from the liquids that will be pumped. In addition, if seawater or another liquid that can conduct an electrical current will be pumped, dissimilar materials that are used should be compatible galvanically. When a galvanic couple does exist, it is generally desirable for the smaller parts to be made from the more noble material; with this arrangement the smaller parts will act as cathodes, and the components with the larger mass will become the anodes. In some cases, corrosion resistance is increased with protective coatings.
- Wetted components should be resistant to erosion. This is especially important if the liquids that will be pumped may contain abrasives, (e.g., seawater, which can often contain sand and grit). Erosion resistance is also an important factor in the selection of materials for components that will be exposed to high-velocity or cavitating fluids.
- Material combinations used for close-clearance rotating and stationary components should be resistant to galling.
- Materials selected for major components should be repairable by welding.

Typical materials used for components in various shipboard pumps are listed in table 12-2 and include the following metals and composites.

SEAWATER PUMPS

Various grades of bronze, copper-nickel alloys, high-alloy austenitic chromium-nickel stainless steels, nickel-copper and nickel-capper-aluminum

TABLE 12-2
Typical Materials of Construction for Centrifugal Pumps Used in Shipboard Systems

Service	Boiler Feed	Condensate	Other Freshwater	High-Pressure Seawater	Low-Pressure Seawater	Cargo (Hydrocarbon)	Cargo (Chemical)
Casing	12Cr Stainless Steel Carbon Steel[A]	Bronze 70Cu-30Ni Alloy Ni-Al Bronze 12Cr Stainless Steel	Bronze 70Cu-30Ni Alloy Ni-Al Bronze Type 316 Stainless Steel Ductile Iron Close-Grained Cast Iron	Ni-Al Bronze 70Cu-30Ni Alloy Titanium Alloy 20	Ni-Al Bronze 70Cu-30Ni Alloy Bronze	Ni-Al Bronze Bronze 70Cu-30Ni Alloy	316 or 317 Stainless Steel
Impeller	12Cr Stainless Steel	17-4 PH Steel Ni-Al Bronze Ni-Cu Alloy 12Cr Stainless Steel Bronze[B]	Bronze Ni-Al Bronze Type 316 Stainless Steel	Alloy 20 Titanium Ni-Cu Alloy Ni-Al Bronze 70Cu-30Ni Alloy	Bronze Ni-Al Bronze 70Cu-30Ni Alloy Ni-Cu Alloy	Bronze Ni-Al Bronze 70Cu-30Ni Alloy	316 or 317 Stainless Steel
Wearing Rings	12Cr Stainless Steel	Bronze	Bronze	Bronze Titanium Ni-Cu Alloy	Bronze	Bronze	Bronze PTFE
Pump Shaft	Type 410 or 416 Stainless Steel Alloy Steel	Type 410 or 416 Stainless Steel Ni-Cu Alloy Ni-Cu-Al Alloy Alloy Steel	Type 304 or 316 Stainless Steel Type 410 or 416 Stainless Steel Carbon Steel[C]	Ni-Cu-Al Alloy Ni-Cr-Mo-Cb Alloy Alloy 20 Nickel-Copper Alloy[C]	Ni-Cu-Al Alloy Nickel-Copper Alloy[C]	Ni-Cu-Al Alloy Type 410 Stainless Steel Bronze	316 or 317 Stainless Steel
Shaft Sleeve	Type 304, 316, or hardened 410 Stainless Steel	Type 304, 316, or hardened 410 Stainless Steel	Type 304, 316, or hardened 410 Stainless Steel	Ni-Cu-Al Alloy Titanium Alloy-20	Ni-Cu-Al Alloy	Bronze Ni-Cu-Al Alloy	316 or 317 Stainless Steel
Bearing Brackets	Carbon Steel Ductile Iron	Carbon Steel Ductile Iron	Carbon Steel Ductile Iron	Bronze	Bronze	Bronze Carbon Steel Ductile Iron	Carbon Steel Ductile Iron

A. When water velocity is reduced sufficiently in a diffuser, or in a barrel pump where the casing is exposed primarily only to suction pressure.
B. Limited to stages not being subjected to cavitation.
C. Should be fitted with a shaft sleeve.

Notes:
1. Al = aluminum; C = carbon; Cb = columbium; Cr = chromium; Cu = copper; Mo = molybdenum; Ni = nickel. Number preceding the element abbreviation represents the nominal percentage of the element in the metal.
2. Type 304 (18Cr-8Ni), type 316 (16Cr-12Ni-2Mo), type 317 (18Cr-13Ni-3Mo), and Alloy 20 (20Cr-Ni-Mo-Cu) are austenitic stainless steels; types 410 and 416 are 13Cr martensitic stainless steels; and 17-4 PH is 17Cr-4Ni-3Cu precipitation-hardening stainless steel.
3. PTFE = polytetrafluoroethylene.
4. The above is based, in part, on information included in ASTM F998-86 (R 1993). It is representative only and is provided for general information. Applicable specifications and regulations should always be reviewed to determine requirements for specific installations. In addition, materials used in the construction of any component should be compatible with all fluids that they will contact and with other materials being used.

alloys, and titanium are often used in seawater pumps. In addition, fiber-reinforced composite materials are used in some applications.

FRESHWATER PUMPS

Cast and ductile irons, 'as well as carbon and alloy steels, are often used in freshwater pumps. In addition, in specialized applications (e.g., feed and condensate service), various grades of stainless steel may be used.

OIL PUMPS

Cast and ductile irons, carbon and alloy steels, and various grades of bronze are used in oil pumps. In addition, when contaminated fluids are to be pumped, such as dirty oil or sludge, stainless steels may be used.

Marine Pump Applications

A description of the features typically incorporated into the designs of pumps used in selected shipboard services follows. This information, however, is general in nature and may not apply in all cases based on the requirements for specific installations or the preferences of vessel owners and designers.

MAIN AND AUXILIARY STEAM SYSTEM PUMPS

Feed Pumps

On a steam-powered ship with fossil-fueled boilers, a main feed pump typically receives feedwater from a deaerating feed tank (DFT) or, in some cases, a low-pressure booster pump. When a group feed system is utilized, water discharged from the feed pump is directed to the steam drum in each of the vessel's boilers. Before entering the steam drum, the feedwater is usually preheated as it passes through heaters installed in the feed piping or an economizer in the boiler's uptake. Both a main feed and a separate auxiliary feed line are often attached to the discharge side of a main feed pump so that there are two separate lines for delivering water to the boilers. In addition to a stop valve in the feed pump's suction line, a stop-check valve is usually installed near the discharge side ofthe pump in both the main and auxiliary feed lines. Also, a stop valve and a stop-check valve are ordinarily installed in both the main and auxiliary feed lines near the inlet to each steam drum or economizer. An automatic feedwater regulating valve may also be installed between each boiler's main-feed stop and stop-check valves. Ifthe feed-pump discharge piping and valves are not designed for the pump's maximum discharge 'pressure, they should be protected by a relief valve installed between the pump discharge and any stop valves.

In addition to overcoming resistance to flow in the feed system and its components, the total head developed by a main feed pump must be sufficient to force water into the boilers. Because boiler operating pressures are frequently relatively high, centrifugal pumps used in this application are often driven at high speeds and may have multiple stages. Typical feed pump configurations include single- and two-stage centrifugal pumps that are close-coupled to steam turbines (fig. 12-43). In addition, multistage flexibly coupled centrifugal pumps (fig. 12-6) are sometimes used. Although pumps in this latter category are also frequently driven by steam turbines, some are electric-motor driven. However, because electric motor speeds are generally limited to 3,600 rpm when 60 Hz alternating current is used (3,000 rpm with 50 Hz current), motor-driven feed pumps typically have more stages than compal"ably rated turbine-driven units. While it is not common, one main feed pump on a vessel may be driven by the main propulsion machinery. Also, on some older vessels, steam-driven simplex direct-acting reciprocating pumps are used for main feed service (fig. 12-40b).

When a vessel has two independently driven main feed pumps, each pump should typically be capable of supplying the vessel's boilers with 100 percent of the normal feedwater requirement. When more than two independently driven pumps are used, the total feed-pump capacity should generally be not less than 200 percent of the vessel's normal requirement. With this latter arrangement, it may be necessary to operate two or more pumps in parallel to deliver the full-load capacity. If a feed pump that is driven by the main-propulsion machinery is used, it should usually be capable of supplying the vessel's boilers with 100 percent of the normal feedwater requirement. In addition, a second feed pump that is also rated for the normal full-load requirement but is independently driven should ordinarily be provided. Furthermore, a third independently driven pump rated for at least 75 percent ofthe normal requirement is typically required for emergency use. A partial-capacity emergency feed pump can sometimes be used for other purposes; regulations, however, often prohibit this for main feed pumps.

Although turbine-driven centrifugal feed pumps are generally mounted horizontally, motor-driven centrifugal feed pumps are furnished in both horizontally and vertically mounted configurations. When a feed pump is motor driven, external support for its shaft is often provided by grease-lubricated ball bearings. External shaft support in a turbine-driven feed pump, however, is generally provided by either a ball or tilting-pad thrust bearing and ball, roller, or journal-type radial bearings that are lubricated with oil supplied by the same system that is used to lubricate the bearings in the turbine driving the feed pump. After leaving the feed-pump and turbine bearings, the lubricating oil drains to a reservoir tank or sump that may be built into the common pump-and-turbine base. During normal operation, oil is typically removed from the sump and circulated through the bearings by a rotary pump that is geared to and driven by the turbine's shaft. A strainer, filter, and seawater-cooled heat exchanger are often installed in the LO

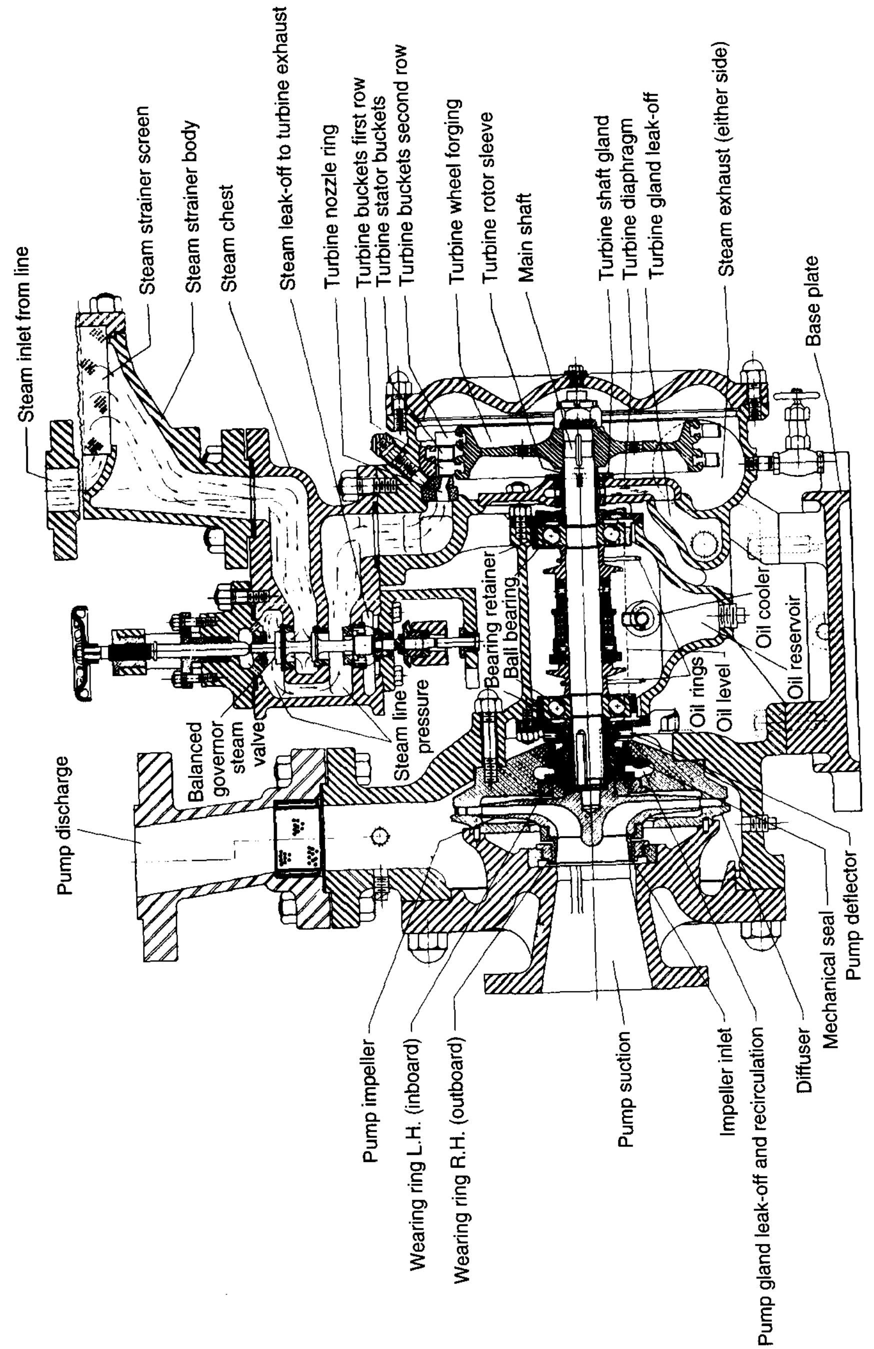


Figure 12-43. Close-coupled steam-turbine-driven main feed pump. Courtesy Coffin Turbo Pump, Inc.

piping between the rotary pump discharge and the bearings. In some installations, the flow of seawater through the cooler must be regulated to maintain the proper LO outlet temperature. Alternatively, oil temperature may be controlled with a thermostatic valve that permits a portion of the LO flow to bypass the cooler. A separate hand-operated or electric-motor-driven LO pump is sometimes provided to enable the feed-pump and turbine bearings to be prelubricated prior to start-up. Sight glasses are usually installed in the LO-system piping to permit the flow of oil to or from the bearings to be observed. In addition, a dipstick is ordinarily provided so that the level in the LO sump can be checked. The proper oil level should be maintained in this sump 'whenever the feed pump is operating. Furthermore, because leaking shaft seals can permit steam or feedwater to enter the feed-pump or turbine bearing housings, samples of the lubricating oil should be checked periodically for water contamination. The lubricating oil added to the turbine-driven feed pumps on many steam-turbine-propelled vessels is the same grade of oil used in the main propulsion LO system.

Packed stuffing boxes are often used to seal shaft penetrations in a centrifugal feed pump's casing. To reduce the pressure being contained by the packing, the base of each stuffing box is frequently fitted with a multiple labyrinth-type breakdown bushing. In addition, a leak-off connection piped back to suction may be located between the bushing and the packing to further reduce the pressure of the feedwater in the stuffing box. To reduce the packing temperature, feed-pump stuffing boxes may include external jackets through which cooling water is circulated.

To eliminate the need for packing, some centrifugal feed pumps have condensate-injection shaft seals. These packless stuffing boxes are fitted either with stationary labyrinth-type fixed breakdown bushings or a series of spring-loaded floating rings that are stacked axially. With either arrangement, a close radial clearance is maintained between the nonrotating sealing elements and the rotating shaft. Cool water diverted from the discharge of the condensate pump is introduced into the middle ofthe seal. A small portion of this water typically flows into the pump. The remainder, however, flows outward into a collection chamber that is usually piped back to the vessel's gland-exhaust condenser. The use of mechanical seals for shaft sealing is an additional option utilized in some feed pumps.

During constant-speed operation, a main feed pump's capacity is ordinarily controlled by the throttling action of a feedwater regulating valve. When multiple partial-capacity feed pumps are used, the amount of valve throttling can be reduced by turning units on or off as needed. In addition, when steam is used to drive a feed pump, the throttling of the regulating valve is frequently reduced by controlling the pump's operating speed with either a constant-pressure or a constant-differential-pressure governor.

A constant-pressure governor automatically regulates a steam-driven feed pump's operating speed to maintain a constant pressure at the pump discharge. When the feedwater regulating valve begins to close because of a reduction in plant load, the feed pump initially responds by delivering less feedwater at a higher discharge pressure. The constant-pressure governor, however, reduces the pump's speed until the discharge pressure returns to the set point. As a result ofthe speed reduction, the amount that the regulating valve must be throttled to obtain the necessary feedwater flow rate is reduced. An increase in plant load has the opposite effect. The set point of a typical constant-pressure governor is often adjusted so that the feed pump's discharge pressure is sufficient to permit the proper steam-drum water level to be maintained with the feedwater regulating valve in a half-open position. To satisfy this requirement, a constant-pressure governor frequently must be set at a pressure that is 50 to 100 psi (345 to 690 kPa) above the steam-drum pressure. When a boiler's superheater outlet pressure is held constant, the steam-drum pressure will increase with boiler load. Consequently, a constant-pressure governor's setting must often be readjusted with changes in boiler load (e.g., during maneuvering) to maintain the necessary difference between the feed-pump discharge pressure and the steam-drum pressure.

A constant-differential-pressure governor regulates a steam-driven feed pump's speed to obtain a pump-discharge pressure that exceeds the boiler pressure by a set amount. The boiler-pressure sensing line for a constant-differential-pressure governor is frequently connected to the economizer inlet or to the steam drum. With either arrangement, the governor will automatically vary the feed-pump speed in response to changes in boiler load to maintain the set difference between the pump discharge pressure and the boiler pressure. Because changes in feed flow when a constant-differential-pressure governor is used result, primarily, from variations in pump speed, the throttling action ofthe feedwater regulating valve is greatly reduced.

Steam turbines that are used to drive centrifugal feed pumps are generally fitted with low-LO-pressure, overspeed, and high-exhaust back pressure trips. In addition, a low-suction-pressure trip is sometimes provided to prevent a feed pump from operating with too Iowa suction pressure, which can result in excessive cavitation, operation at too Iowa capacity, and a loss ofload on the driver.

Feedwater in a DFT is at its vapor pressure. Consequently, when no feed booster pump is used, the NPSH available to a main feed pump is equal to the elevation of the water level within the DFT measured above the pump's first-stage impeller, less frictional losses within the pump's suction line. Because a centrifugal pump's NPSH requirements increase with operating speed, limiting the pump's operating speed is sometimes necessary to suppress cavitation. In addition, some feed pumps are fitted with a double-suction first-stage impeller, which can reduce NPSH requirements.

The NPSH available to a main feed pump that operates without a booster pump can drop if there is a sudden reduction in the pressure and temperature within the DFT because of a change in plant load. Although this reduction in pressure is transmitted directly to the feed-pump suction, the temperature ofthe feedwater already in the pump's suction line is still at the original value. Therefore, until the cooler water in the DFT reaches the pump, there is a reduction in NPSHA. The safety margin between the steady-state NPSHA and the feed pump's NPSH requirement that is necessary to suppress cavitation during operation with transient conditions increases with the rate of pressure decay within the DFT and the volume within the feed pump's suction piping. If a sufficient margin cannot be achieved, provisions are sometimes made to inject cool condensate into the feed pump's suction during transient operation, which lowers the vapor pressure of the feedwater entering the pump.

Operation of the typical feed pump at shutoff or with very low flow rates for even a short period of time can result in a rapid increase in the temperature of the fluid within the pump. Because feedwater is already at an elevated temperature, additional heating can cause it to vaporize. To prevent a feed pump from operating at too Iowa capacity, a recirculation line is generally provided from the feed pump's discharge back to the DFT. On some vessels, this line is fitted with an orifice that permits feedwater to be recirculated through it whenever the pump is operating. Alternatively, a valve that can be closed during full-flow operation is sometimes installed in a feed pump's recirculation line.

In addition to the main feed pumps, a smaller capacity feed pump is installed on some steam-powered vessels for use in port or during emergencies. Motor- or steam-driven reciprocating pumps are often used for in-port feed service. Also, electric-motor-driven centrifugal, regenerative turbine, and power diaphragm pumps are typically used in auxiliary steam systems on motor- and gas-turbine-driven vessels to deliver feedwater to exhaust-gas or auxiliary oil-fired boilers.

Feed Booster Pumps

When the elevation ofthe DFT is not sufficient to provide adequate NPSH to the main feed pump, a booster pump is usually installed in the main feed pump's suction line. Because the total head developed by the booster pump is only a fraction of the head developed by the main feed pump, the booster pump can be operated at a lower speed and, therefore, with less NPSHA. In addition, because the booster pump increases the pressure of the feedwater entering the main feed pump during normal operation, cavitation in the main pump is ordinarily suppressed. Typical configurations used for feed-booster service include single-and two-stage vertically mounted centrifugal pumps with operating speeds that are generally limited to 1,780 rpm. Like a main feed pump, the discharge line of a feed booster pump is often fitted with a recirculation line back to the DFT to prevent operation at too Iowa capacity.

Main Condensate Pumps

On a steam-powered vessel, a main condensate pump removes condensate from the hotwell in a main condenser and discharges it, through various heat exchangers, to a DFT. The total head that must be developed by the pump is, therefore, based on the difference in elevation between the water level in the hotwell and the level in the DFT, the difference in the pressures within these two chambers, losses due to friction and turbulence in the condensate system, and the pressure drop through the water-spray nozzles at the entrance to the DFT. Vertically mounted centrifugal pumps (fig. 12-44) are frequently used in this application. Two condensate pumps are generally provided for each condenser, with each pump sized to handle full-load requirements; however, if necessary, the pumps may be operated together in parallel. Although many of these pumps are driven by electric motors, some main condensate pumps are driven through reduction gears by steam turbines. In addition, although it is not common, one main condensate pump on a vessel may be driven by the main propulsion machinery.

Because a condensate pump's inlet is under a vacuum, all suction piping should be airtight. To aid in the removal of air that may enter the pump, a vent line is typically installed from a connection in the suction area of a condensate pump's casing back to the main condenser.

A grease-lubricated ball bearing is often provided at the upper end of a condensate pump's shaft to absorb both axial and radial loads. Radial loads are also frequently absorbed by an internal water-lubricated sleeve bearing installed above the first-stage impeller. A single shaft seal, which can consist of a packed stuffing box or a mechanical seal, is generally located at the top ofthe casing. With this arrangement, the base ofthe seal is exposed to condensate that has been discharged through one or more of the pump's stages and is, therefore, at an elevated pressure. This reduces the potential for air to be drawn through the seal and into the pump. When it is possible for the base ofthe seal to be under a vacuum, pressurized condensate recirculated from the pump's discharge is often injected into the seal.

Because the condensate within a condenser's hotwell is normally at or close to its vapor pressure, the NPSH available to a condensate pump is essentially equal to the height of the water level in the hotwell above the pump's first-stage impeller, less frictional losses in the suction piping. To maximize NPSHA, condensate pumps are generally installed as far below the condenser as possible with suction lines that are relatively large and free of unnecessary bends. In addition, to achieve low net positive suction head requirements, condensate pump operating speeds are generally limited to 1,780 rpm. Consequently, to enable the necessary total head to

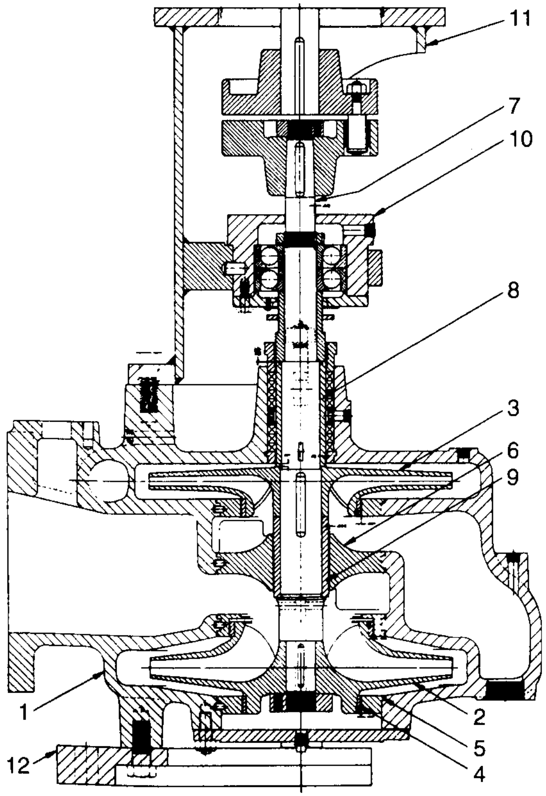


Figure 12-44. Two-stage centrifugal main condensate pump. Courtesy Ingersoll-Dresser Pump Company.

be developed, pumps with two or three stages are often used. To increase the submergence of the first-stage impeller, in a typical vertically mounted condensate pump, this impeller is mounted on the lower end ofthe pump's shaft. Furthermore, this impeller is usually oriented with its suction eye on top so that it will be self-venting. (The second impeller in a typical two-stage vertical condensate pump is generally mounted with its suction eye directed downward, which reduces the net hydraulic axial thrust that must be absorbed by the pump's thrust bearing and increases the pressure of water at the base of the shaft seal.)

If a condensate pump is driven by a variable-speed turbine or motor, its speed can be adjusted with plant load so that the capacity removed from the hotwell matches the rate at which condensate enters the hotwell. The capacity delivered by a constant-speed condensate pump can be controlled using one or a combination of the following methods:

1

- Submergence control. During operation with sufficient NPSH available, a condensate pump delivers the capacity corresponding to the operating point on its head-capacity (H-Q) performance curve at which the total pump head developed matches the condensate-system-head requirement. However, when the mass flow rate ofthe condensate being removed from the hotwell exceeds the rate at which steam (and, in some cases, water) enters the condenser, the hotwelllevel drops and the NPSH available to the condensate pump is reduced. Eventually, cavitation begins and reduces the capacity pumped. The hotwelllevel, the NPSH available to the pump, and the capacity pumped will all continue to drop until the flow rate being removed from the hotwell matches the flow rate entering the condenser. When this occurs, the hotwelllevel will stabilize at the elevation that produces an NPSHA approximately equal to the pump's NPSH requirement at the new reduced capacity. Therefore, provided that the NPSH available is too low to permit the pump to operate along its normal H-Q curve, the flow rate removed from the hotwell is regulated by cavitation to equal the rate at which steam enters the condenser. Although submergence control eliminates the need for control valves, the condensate pump's first-stage impeller can be subjected to almost continuous cavitation. In some cases, the region of cavitation can even extend into the second-stage impeller and result in a loss of lubrication to the internal journal bearing, typically located between the pump's first and second stages. When there is a potential for this to occur, which increases with the number of pump stages, pressurized condensate from the pump's discharge is often directed to a drilled port in the casing adjacent to the internal bearing .
- Throttle control. As the hotwelllevel drops, a level-sensing device automatically throttles a valve in the condensate pump's discharge line. Conversely, this valve is opened in response to an increase in the hotwelllevel. The throttling action of this valve is, therefore, used to adjust the condensate-system-head curve so that it crosses the condensate pump's H-Q curve at the capacity required to keep the hotwelllevel at a desired preset value. Under steady-state conditions, the capacity delivered by the condensate pump will also equal the rate at which steam enters the condenser. With this type of control system, cavitation can often be suppressed. However, when steam-jet air ejectors are used to deaerate the main condenser, condensate pump operation at too Iowa capacity can result in insufficient condensate flow through the ejector inter- and after-condensers. In addition, low-capacity operation can lead to surging caused by suction and discharge recirculation within the condensate pump. The potential for suction recirculation in a condensate

pump can increase when the pump has a high-suction specific-speed low-NPSHR first-stage impeller.

- Recirculation control. A level-control valve installed in a recirculation line connected to the condensate system (downstream from any air-ejector nd glad-exhaust condensers) opens as the hotwelllevel drops to permIt a portion ofthe condensate discharged by the main condensate pump to be returned to the main condenser. The hotwell level can threfore, be mtained at a value that is sufficient to suppress cavi~ tatIOn. In additIOn, because the main condensate pump can always operate at or near its rated capacity, low-flowoperation is avoided, Furthermore, adequate cOhdensate flow through air-ejector condensers, when used, and gland-exhaust condensers can be maintained during plant start-up and low-load operation. (In some cases the condensate-recirculation line may be fitted with a thermostaticall; controlled valve that opens and permits the condensate flowrate through the air-ejector condensers to increase when the temperature ofthe condensate leaving these heat exchangers exceeds a preset value.)

Auxiliary Condensate Pumps

When ~urbogenerator on a steamship exhausts into an auxiliary condenser, an auxIhary condensate pump is typically used to remove condensate from te otell ad return it to the DFT. Vertical two-stage centrifugal pumps sImlr III desIgn o. the main condensate pumps are often used in this ap-ph,catIOn; the auxIhary pumps are, however, smaller in size and may be dnven at seeds up t~ ;580 rpm. On a motor- or gas-turbine-propelled vessel fitted wIth an auxIhary steam system, centrifugal-type pumps may also be used to transfer condensate from the hotwell of an auxiliary condenser to a drain tank.

Freshwater-Drain-Collecting_Tank Pumps

Condensate fro~ uncontaminated drains that are above atmospheric pressure ae often dIrected to a freshwater drain collecting tank. In addition, a pump IS frequently used to transfer the condensate collected in this tank to he DFT. A typical freshwater-drain-collecting_tank (FWDCT) pump, hIch may also be referred to as an atmospheric-drain-tank (ADT) pump, IS a close-coupled elecric-motor-driven single-stage centrifugal pump that may be mounted honzontally or vertically. Two full-capacity pumps are usually pr0.vided. With this arrangement, one pump can be operated while the second IS used as a standby unit or is maintained. The on-line pump is often started and stopped automatically by a float control in the drain tank. Because the temperature of the condensate in the drain tank is usually close to its boiling point of 212°F (100°C), the NPSH available to a FWDCT pump is limited. To help suppress cavitation, the pump should be

installed as low in the vessel as practicable. In addition, the use of a pump with a low NPSH requirement is recommended.

Condenser Exhausting Pumps

Electric-motor-driven liquid-ring vacuum pumps are sometimes used to deaerate main or auxiliary condensers. A typical vacuum pump removes air and vapor from a condenser at essentially a constant volumetric flow rate. Consequently, because the density of a gas increases with pressure, the mass flow rate through the pump increases as the vacuum within the condenser is reduced. To enable the high condenser vacuums necessary for efficient steam-plant operation to be developed and maintained, two-stage vacuum pumps are often used in this application. In addition, because the vacuum created by a liquid-ring vacuum pump is limited by the vapor pressure of the sealing-liquid within the pump's casing, which increases with temperature, water separated from the gases discharged by the pump is generally cooled in a heat exchanger before being returned to the unit. In some installations, only a portion of the cooled water is returned to the pump casing, with the remainder being injected directly into the pump's suction line. This cools the air/water-vapor mixture removed from the condenser, which reduces the mixture's vapor content and the load on the pump. Two pumps that are each capable of individually maintaining the required vacuum are frequently provided for each condenser.

COOLING WATER PUMPS

Circulating Pumps

On a steam-powered vessel, a main circulating pump takes suction from one of the vessel's sea chests and circulates seawater through the tubes in a main condenser. A portion of the seawater discharged by this pump may also be diverted to other seawater-cooled heat exchangers, such as the main LO coolers. Only one circulating pump is often furnished for each main condenser on a vessel. Typically, a main circulating pump must deliver a relatively high capacity at a relatively low pressure. In addition to the main suction flange, which is connected to the sea chest, a main circulating pump may also have an auxiliary side-suction connection that can be used for emrgency dewatering ofthe machinery space bilges. This emergency bilge connection should normally be fitted with a stop-check valve.

Single-stage axial-flow propeller pumps are often used in this application (fig. 12-9). Radial shaft loads are generally absorbed by a journal bearing located above the overhung propeller. Many ofthese pumps, however, do not have a thrust bearing. Instead, the pump shaft is rigidly coupled to the driver's shaft, and axial loads are absorbed by the driver's thrust bearing. With this arrangement, the thrust bearing in the driver must be sized to absorb the additional axial loads applied to the pump's rotor.

Single-stage mixed-flow pumps with overhung end-suction impellers and single-stage radial-flow pumps with double-suction impellers mounted between bearings are also sometimes used as main circulating pumps.

If a steam-powered vessel is fitted with a scoop-injection system, the main circulating pump (or pumps) may be stopped when the vessel is moving at a speed that is high enough for an adequate amount of seawater to be forced into the scoop and through each main condenser. Because much of the pump operation, therefore, occurs while the vessel is in shallower water, the pumped water often contains silt, sand, and other abrasives. Consequently, internal journal bearings that are lubricated by water discharged from the propeller or impeller are sometimes furnished in abrasion-resistant grades of rubber or composite materials. Alternatively, some internal bearings are lubricated by either grease or clean water supplied through an external connection in the pump's casing.

Main circulating pumps are generally driven at speeds not exceeding 880 rpm by steam turbines with reduction gears or two-speed electric motors. This permits the capacity of seawater discharged through a main condenser to be reduced (e.g., during cooldown periods or when raising condenser vacuum) by reducing the speed of the main circulating pump's driver. Because the vapor pressure of seawater at ambient temperature is negligible, the NPSH available to a main circulating pump is approximately equal to the submergence of its propeller or impeller below the vessel's waterline added to atmospheric pressure, less losses due to friction and turbulence in the sea chest and suction line. When an axial- or mixed-flow pump having power requirements that rise towards shutoff is used, operation at flow rates below the point at which the driver is rated should be avoided.

Auxiliary circulating pumps supply seawater to smaller condensers that are sometimes used to receive steam exhausted from a vessel's turbogenerators. Seawater discharged by these pumps may also be directed to LO and air coolers used with the turbogenerators. Additionally, a crossover line is often provided so that, in the event of a main circulating pump failure, seawater discharged from the auxiliary circulating pumps can be directed to the main condenser. Single-stage, electric-motor-driven, radial-flow centrifugal pumps that have either a single- or a double-suction impeller are generally used in this application.

Jacket-Cooling-Water Pumps

Single-stage vertically or horizontally mounted centrifugal pumps are ordinarily provided to circulate freshwater through jackets in a propulsion-diesel engine's cylinders and cylinder heads, the engine's turbocharger, and, on some vessels, a freshwater generator (where the jacket water provides heat to produce distilled water). The jacket water may also pass through the engine's LO and charge-air coolers and ajacket-cooling-water cooler. The cooling medium in the jacket-cooling-water cooler can be either seawater or freshwater. A three-way automatic thermostatic valve is usually provided to control the amount ofjacket water that passes through the cooler and the amount that bypasses it. (The jacket-cooling-water cooler is eliminated on vessels fitted with a central freshwater-cooling system in which cooling of the jacket water is accomplished by admitting some of the freshwater from the low-temperature loop into the high-temperature jacket-water loop.)

Although the pumps that circulate freshwater through a diesel engine's coolingjackets are often referred to as jacket-cooling-water pumps, on vessels with a central freshwater cooling system, they may be called high-temperature cooling-water pumps. Two pumps are frequently provided, with each pump sized to meet the vessel's normal full-load propulsion-engine requirements. Both pumps may be driven by electric motors; however, when used with a high- or medium-speed propulsion engine, one of the pumps is often mounted on and driven off the engine. Diesel engines that drive generators also typically have an attached jacket-water circulating pump. An elevated expansion tank typically maintains a positive pressure at the pump suction.

The jacket-cooling-water pump or a separate smaller-capacity motor-driven centrifugal pump may be used to circulate jacket water through a preheater and the engine prior to start-up. Jacket water may also be circulated through the preheater during low-load engine operation to raise the temperature of the jacket water entering the freshwater generator.

Separate single-stage centrifugal pumps may also be used to circulate freshwater through a propulsion engine's pistons when they are water-cooled. Alternatively, this water is sometimes circulated by sump or vertical turbine pumps that are submerged in the piston-cooling-water drain tank. With either arrangement, two piston-cooling-water pumps that are both electric-motor-driven are ordinarily furnished. On some diesel-propelled vessels, an additional pair of electric-motor-driven centrifugal pumps is provided to circulate freshwater through an independent loop that cools the main engine's fuel valves or injectors. A head tank that maintains a positive suction pressure to the pumps is typically installed in this loop.

Seawater Cooling Pumps

A seawater cooling pump (sometimes referred to as a seawater service or auxiliary seawater pump) takes suction from a sea chest and supplies seawater to heat exchangers that utilize this water as a cooling medium. On many vessels, this includes refrigeration and air-conditioning condensers, various LO coolers, and air-compressor coolers. Two or more horizontally or vertically mounted, electric-motor-driven, single-stage centrifugal pumps are used in this application on a typical vessel. Separate centrifugal pumps may be used to supply seawater to diesel-engine charge-air, LO, and jacket-water coolers. Although these latter pumps may also be electric-motor-driven, medium- and high-speed diesel engines are sometimes fitted with an attached seawater-cooling pump. In lieu of the aforementioned arrangement, on vessels that have a central freshwater cooling system, the seawater cooling pumps supply seawater only to large freshwater coolers. After leaving the heat exchangers that it passes through, the seawater is ordinarily directed overboard.

A seawater cooling pump should be located sufficiently below the vessel's waterline so that its impeller has adequate submergence with design list and trim conditions. In many cases, a seawater pump's suction line is connected to both an upper and a lower sea chest. The lower sea chest should generally be used when the vessel is at sea, and the upper sea chest should be used when the vessel is operating in shallow water or in port. On a diesel-propelled vessel, the main-engine seawater cooling pumps are also often connected to an emergency bilge suction line. Strainers are usually installed in the seawater pump suction lines and should be kept clean.

Freshwater Cooling Pumps

On diesel-powered vessels that have a central freshwater-cooling system, single-stage centrifugal pumps are generally provided to circulate freshwater through two separate cooling loops. A high-temperature (HT) cooling-water pump, which serves as the jacket-cooling-water pump, typically circulates freshwater through diesel-engine jackets, turbochargers, and a freshwater generator (evaporator). A low-temperature (LT) cooling-water pump circulates freshwater through various condensers, and through oil and air coolers. Freshwater in the LT loop also passes through a central seawater-cooled heat exchanger. In addition, on some vessels, freshwater in the LT loop also passes through a heat exchanger in which it absorbs heat from the freshwater in the HT loop. Alternatively, however, this HT freshwater cooler is eliminated in some central freshwater cooling systems, and cooling is accomplished by allowing some of the water in the LT loop to pass through a thermostatically controlled valve located near the inlet to the HT cooling-water pumps and mix with the water in the HT loop. After the HT freshwater passes through the engine jackets and other heat exchangers, an equal amount of water from the HT loop enters the LT loop. The mixture of high- and low-temperature freshwater then passes through the central seawater cooler. Two full-capacity cooling pumps are frequently installed in each freshwater loop. In addition, smaller partial-capacity pumps for use when the vessel is in port may also be provided.

Freshwater in cooling loops should generally be treated chemically to reduce scale formation, corrosion, and the growth of bacteria. In addition, if freezing can occur anywhere within the system, antifreeze should be added.

OIL PUMPS

Fuel-Oil Service Pumps

On a steam-powered vessel with one or more oil-fired boilers, a fuel-oil (FO) service pump typically takes suction through either high- or low-suction ports in the FO service or settling tanks and supplies fuel oil to the burners in the boilers. With this arrangement, if excessive water or sludge enters the pump through a settling tank's low-suction port, operation can be switched over to the high-suction port. At least two service pumps that are each capable of delivering the vessel's full-power fuel-oil requirement must generally be provided. Horizontally and vertically mounted rotary screw or gear pumps that are driven by either steam turbines or two-speed electric motors are often used in this application. In addition, some older ships have steam-driven reciprocating FO service pumps. A separate electric-motor-driven rotary-type pump is usually provided to supply light fuel oil (usually diesel oil) to the boilers during plant start-up until the heavier fuel oil that is normally burned has been heated to its proper temperature. Motor-driven rotary pumps are also used on nonsteam-powered vessels to supply fuel oil to auxiliary oil-fired boilers.

On a diesel-propelled vessel, the FO service pump, which is usually referred to as a supply or booster pump, typically takes suction either directly from the daily service tanks or, in some cases, from a separate mixing or buffer tank and supplies fuel oil to the main-propulsion-engine injector pumps. Two multiscrew or gear pumps that are each capable of meeting the vessel's full-power requirements are usually provided for this application. Although both pumps may be electric-motor driven, one pump is sometimes mounted on and driven by the diesel engine that it serves. In high-temperature systems, separate low-pressure electric-motor-driven rotary pumps may be used to increase the fuel-oil pressure and prevent vaporization on the suction side of the pumps that discharge to the engine. In systems of this latter design, the low-pressure FO pumps are often referred to as the supply or feed pumps and the high-pressure FO pumps are called booster or circulating pumps. All of the FO pumps should be suitable to handle any of the various grades of fuel oil that may be burned in the engine, which can often include both light distillates and heavy oils.

A propulsion gas turbine is typically fitted with an attached gear pump that delivers fuel oil to the combustion-chamber nozzles. Twin two-speed electric-motor-driven rotary pumps that operate upstream of and in series with the attached pump are also frequently used. A diesel engine or gas turbine that drives a generator is also commonly furnished with an attached rotary-type FO pump.

A duplex strainer is generally installed in an FO service pump's suction line to protect the pump. In addition, a duplex strainer, a duplex filter, or a simplex filter with standby filter in a bypass is ordinarily installed in the discharge line. With this arrangement, strainers or filters can be cleaned or replaced without interrupting system operation. Flow meters and heaters :may also be installed in the discharge side of the FO service system. When fuel oil must be heated, at least two heaters are typically required so that anyone of the heaters can be overhauled while system operation continues.

Changes in the flow rate passing through a rotary or reciprocating FO service pump are made by varying operating speed. During constant-speed operation, an FO service pump typically delivers a relatively constant capacity of fuel oil that exceeds the amount required for combustion. Excess fuel oil supplied to a diesel engine is typically returned through a recirculation line to the mixing or service tank. Excess fuel oil discharged by a boiler's FO service pump is typically returned through a recirculation line connecting the pump discharge line to the suction side of the pump. The flow through this recirculation line is typically regulated by a control valve that opens and closes automatically to maintain a constant fuel-oil pressure at the outlet side of the strainer in the service pump discharge piping. A similar arrangement that permits excess fuel oil to be recirculated back to the service tank is frequently incorporated into a propulsion gas turbine's FO service system. To prevent FO-service-system overpressurization, a relief valve that dumps back to the suction line should be installed on the discharge side of each pump. (If the setting of the constant pressure recirculation valve is too close to that of the relief valve, the two valves can alternately open and close creating flow oscillations in the FO service system.) Unexpected overloads can be prevented by sizing an FO service pump's driver based on the maximum differential pressure that the relief valve will allow the pump to develop during operation with fuel oil at the highest viscosity at which it will be removed from the settling or service tanks. This viscosity and the FO service pump's power requirement can often be reduced by heating fuel oil in the settling tanks. Heating of the fuel oil will also enable the FO service pump to remove viscous fuel oil from the settling tanks more easily; however, an FO service pump that is suitable to operate at the elevated temperatures should be used and overheating of the fuel oil should be avoided.

Remote controls should be provided that enable FO pumps to be stopped in an emergency from outside of the machinery spaces. It is recommended and sometimes required by regulations that flanged FO pump discharge connections be fitted with wraparound shields to deflect spray in the event of a leak and that coamings or drip pans be installed under FO pumps to prevent leaking oil from draining into the bilge.

Fuel-Oil Transfer Pumps

A fuel-oil (FO) transfer pump typically takes suction from a vessel's FO storage tanks and discharges fuel oil to the settling tanks or to other storage tanks on the vessel. This pump can also usually be used to remove fuel

from the storage and service tanks and direct it to abovedeck connections for off-loading. Two pumps that can be used to transfer fuel oil are frequently provided on a vessel. Horizontally and vertically mounted rotary gear and multiple-screw pumps are often used in this application. Although these pumps are frequently located in the machinery spaces, vertical units are sometimes submerged directly within the FO storage tanks and are driven through line-shafting by above-tank motors. In addition, on some steam-powered vessels, steam-driven reciprocating-piston pumps are used to transfer fuel oil.

The rate at which settling tanks are refilled by a reciprocating or rotary FO transfer pump can be adjusted by varying the pump's operating speed. To permit this speed to be changed, rotary transfer pumps are sometimes driven by two-speed electric motors or, in the case of some steamships, steam turbines. Similarly, the speed of a steam-driven direct-acting reciprocating FO transfer pump can be adjusted by regulating the steam flow to the unit. Alternatively, to permit the flow rate delivered to settling tanks by a constant-speed FO transfer pump to be adjusted, a valved bypass line through which oil is recirculated back to suction may be connnected to the pump's discharge line. Speed reduction or the opening ofthe recirculation valve will also reduce an FO transfer pump's discharge pressure and, therefore, the load on the pump's driver. In addition, it may increase the amount of fuel oil that the pump can remove from a tank being stripped. (The ability of an FO transfer pump to remove viscous fuel oil from a tank can often be improved by heating the fuel prior to transfer. This will also reduce the load on the FO transfer pump's driver. However, a transfer pump that is suitable to operate at the elevated temperatures should be used. In addition, overheating ofthe fuel oil should be avoided.)

A suction strainer should generally be installed at the inlet to each FO transfer pump, and the discharge side of each pump should typically be protected with a relief valve. Additionally, remote controls should be provided that enable FO transfer pumps to be stopped in an emergency from outside of the machinery spaces.

On diesel-propelled vessels, fuel oil is ordinarily directed through one or more centrifugal purifiers when it is transferred from the settlers to the daily-service tanks. Although the rotary pumps used to transfer this oil may be independent motor-driven units, they are often mounted on and driven by the purifiers. Similar pumps are used to circulate fuel oil through purifiers on gas-turbine-propelled vessels.

Lubricating-Oil Pumps

On a typical steam-turbine-propelled vessel, a lubricating-oil service (LOS) pump removes lubricating oil from the main reduction-gear sump and discharges it into piping that distributes the oil to the propulsion turbine bearings, reduction gears, and thrust bearings. Also, some ofthis oil may be directed to an overhead gravity tank and to the propulsion-turbine throttle valv~ controls and speed-limiting governor pumps. (The gravity tank IS provIded to supply oil to the propulsion bearings and reduction gears for several minutes following an interruption in the LOS pump discharge.)

Most vessels are fitted with at least two LOS pumps. One of these pump~ may be designated as the main LOS pump and a second pump may be desIgnated as the standby unit. Alternatively, however, when two identic.a~full-capacity LOS pumps are provided, either unit can typically be utilIzed for mam or standby duty. In addition, when three LOS are installed on a vessel, one may be only for emergency use. Although horizontally and vertIcally mounted multiple screw pumps are frequently used for La service, steam-driven duplex direct-acting reciprocating-piston LOS pumps are installed on some older vessels.

Typically, at least one of the rotary LOS pumps on a vessel is driven by an electric motor: Although the remaining rotary LOS pumps may also be elect.nc-motor dnven, some are driven through reduction gears by steam turbmes or are attached to and driven by the main propulsion machinery. (When an attached LOS pump is used, it must typically be operated in parallel with one ofthe vessel's other LOS pumps at low-ahead vessel speeds. Also, an unloading valve is ordinarily provided to recirculate excess oil discharged by the attached LOS pump during high-ahead-speed operation back to the La sump, and a bypass must generally be provided to permit oil to pass around the pump during astern operation.) In addition, emergency rotary LOS pumps are frequently driven by battery-powered direct-current electric motors. (A vessel fitted with a battery-operated emergency LOS pump frequently does not have a gravity tank in the LOS system.)

VertIcally mounted rotary LOS pumps are sometimes submerged directly within the La sump. An externally mounted LOS pump should be mounted as close to the La sump from which it takes suction and as low in the vessel as is practicable. In addition, its suction piping should be free of unnecessary elbows and reducers and should begin with a properly designed bell mouth. TypIcally, a control valve or a variable orifice installed in the LOS pump's discharge line establishes the pump's discharge pressure. Coolers are also installed in the LOS pump discharge piping. The cooling medIUm IS frequently seawater supplied from the main circulating system. In most cases, the pressure of the cooling seawater is less than the oil pressure. With this arrangement, a cooler leak will not result in contamination of the LOS system. During normal operation, La temperature is usually regulated by controlling the flow rate of seawater leaving the cooler or by allowing some oil to bypass the cooler. Two separate means should typIcally be provided for circulating seawater through the La coolers.

The main engine La pump on a diesel-propelled vessel typically removes lubncatmg 011 from a sump or drain tank located below the engine

and discharges the oil into piping that conveys it to the engine's bearings, governor controls, turbochargers, and, in some cases, pistons. Vertically or horizontally mounted multiple-screw and gear pumps are frequently used in this application. In addition, vertical turbine pumps that are submerged within the La sump have been used. Most vessels have at least two main La pumps. Although these pumps may be driven by electric motors, one pump is sometimes attached to and driven off a medium- or high-speed engine. A diesel engine that drives a generator is also ordinarily fitted with an attached rotary-type La pump. Coolers installed in a diesel engine's main La system may utilize either seawater or freshwater as the cooling medium. The lubricating oil temperature is typically regulated by a three-way valve that allows some of the oil to bypass the cooler.

The use of additional smaller electric-motor-driven pumps is often necessary because of special La requirements for different parts of a diesel engine. For example, with a crosshead-type engine, some ofthe oil discharged by the main La pumps is directed to the suction side oftwo lower-capacity, rotary-type booster pumps. The high-pressure oil discharged by these booster pumps lubricates the engine's crosshead bearings. A separate low-capacity rotary pump is also used with crosshead engines to fill a head tank that supplies oil to engine-mounted cylinder lubricators. Additionally, a separate pair of rotary pumps may be furnished to supply lubricating oil to the engine's camshaft bearings. Furthermore, on vessels with medium- or high-speed propulsion engines, separate rotary pumps are generally used to supply lubricating oil to the engine's reduction gears.

La pumps used with a propulsion gas turbine take suction from the La reservoir and discharge synthetic oil to the turbine's bearings and control devices. A gear-type rotary pump or a centrifugal pump that is driven off the gas turbine is often used. In addition, an electric-motor-driven centrifugal, gear, or vane pump is generally provided as a backup to the attached pump and for use during start-up or cooldown. An air-turbine-driven emergency LOS pump may also be provided. Furthermore, separate scavenge pumps may be used to transfer oil that drains from the bearings to the reservoir tank. Excess oil delivered by a gas turbine's La pump is returned, through a pressure regulating valve, to the reservoir. Separate multiple-screw pumps are frequently used to circulate mineral oil through an independent reduction-gear lubrication system. Although one of these screw pumps is ordinarily driven off the reduction gears, remaining reduction-gear La pumps are usually electric-motor driven. A portion of the mineral oil in the reduction-gear La system typically passes through a heat exchanger where it cools the synthetic oil that lubricates the gas turbine's bearings.

Although La sumps are ordinarily vented, oil being pumped can frequently have small percentages (e.g., 2 to 5 percent by volume) of entrained air. La system piping should be independent of other systems. A duplex strainer is often installed in an LO pump's suction line, when used, to protect the pump. In addition, a duplex strainer, a duplex filter, or a simplex filter (with a standby filter in a bypass when a vessel has only one propulsion engine) is ordinarily installed in the discharge line. Strainers installed in LO systems frequently have magnetic inserts to remove ferrous particles from the oil. The discharge side of each rotary or reciprocating LO pump should be protected with a relief valve. In many cases, this valve relieves back to the LO sump. Unexpected overloads can be prevented by sizing an LO pump's driver based on the maximum differential pressure that the relief valve will allow the pump to develop during operation with tpe lubricating oil at the lowest temperature and highest viscosity at which it will be pumped.

On most vessels, a pressure-controlled switch or valve is provided that automatically starts a standby pump if the LO pressure at the discharge of the operating pump drops below a preset value. To prevent reverse flow through standby LO pumps when they are not operating, a check valve should be installed in each pump's suction or discharge line. (This check valve is sometimes installed in the suction line below the oil level within the sump so that the standby pump's suction line will remain filled with oil when the pump is stopped.) Controls are also ordinarily provided to sound an alarm and automatically stop the main propulsion machinery in the event of a loss of pressure in the LO system.

Rotary pumps are typically provided to circulate lubricating oil through centrifugal purifiers. These pumps, which are often attached to and driven by the purifiers, or separate motor-driven rotary pumps may also be used to transfer lubricating oil from storage tanks to service tanks and to tanks in various locations throughout the vessel where oil is stored for use in auxiliary machinery. Furthermore, the pumps may be used to add or remove and replace lubricating oil in turbine, engine, and reduction-gear sumps. Additional motor-driven rotary gear, screw, or vane pumps are ordinarily used to circulate lubricating oil through the stern tube when a vessel is fitted with oil-lubricated tailshaft seals and bearings.

Hydraulic-Fluid Pumps

Positive-displacement pumps are used to pressurize and circulate the liquids that drive hydraulically powered equipment. These pumps are usually driven by electric motors or diesel engines and can be installed either in a self-contained system that is an integral part of the hydraulically powered component or in a larger central hydraulic system that provides power to all of the vessel's hydraulic equipment, such as anchor windlasses, winches, and hatch covers.

Gear, vane, multiple-screw, and fixed-displacement rotating-piston pumps are frequently used in constant-flow hydraulic systems. These pumps, which often operate continuously, take suction from a sump or tank and deliver pressurized hydraulic fluid to the components powered by the system. If the pumped capacity exceeds the system requirement, excess oil is recirculated through an unloading valve back to the sump.

In a constant-pressure hydraulic system, one or more variable-displacement rotating-piston pumps are often used to supply hydraulic fluid to the driven components at essentially a constant pressure. Ifthe capacity delivered by a pump exceeds the system's demand, the system supply pressure will increase and a pressure compensator will reduce the pump's stroke and, therefore, the capacity pumped. Conversely, if the load increases and the system pressure drops, the pump's stroke and the capacity delivered will be increased. The pumped capacity is, therefore, automatically adjusted to match the demand on the hydraulic system.

SHIP'S SERVICE APPLICATIONS

Fire Pumps

A fire pump takes suction from a vessel's sea chest and discharges seawater through the fire main to hydrants located throughout the vessel. Each fire pump must generally be capable of delivering seawater simultaneously from a specified number (usually 2 or 3) ofthe highest hydrants at a specified total pressure. Typical pressure requirements include 75 psig (517 kPa gauge) for many tankers and 50 psig (345 kPa gauge) for many nontankers. In addition, each fire pump must ordinarily deliver a minimum capacity of seawater while developing this required pressure. A minimum fire-pump capacity equal to % ofthat required from an independent bilge pump but not less than 110 U.S. gpm or 25 m^3lhr is a typical requirement. Most vessels have at least two independently power-driven fire pumps (one may be designated as an emergency fire pump) that are installed in separate spaces. These pumps, together with their sea connections and sources of power, are ordinarily arranged so that a fire in anyone location cannot prevent all of a vessel's fire pumps from being operated. (An exception to this can apply on some smaller vessels that may have only one fire pump. In addition, in some cases, this pump may be hand operated.)

Single-stage centrifugal pumps are often used in fire service. Larger centrifugal fire pumps typically have an axially split casing and a double-suction impeller that is mounted between bearings (fig. 12-2). Vertically mounted pumps are generally driven by electric motors; horizontal pumps, however, can be driven by electric motors, steam turbines, or diesel engines. Smaller centrifugal fire pumps can have a radially split casing and a single-suction impeller. Radially split casing fire pumps are frequently close-coupled to and driven by electric motors (fig. 12-4). Alternatively, vertical turbine pumps may be used for fire service when the pump's impellers are to be submerged within a tank. The abovedeck driver used with a vertical turbine fire pump can be a vertical electric motor, or a horizontal motor, steam-turbine, or diesel engine that is coupled to the pump's shaft through a right-angle gear.

Strainers installed in fire-pump suction lines should always be kept clean. A fixed fire pump should be installed low enough in a vessel so that it operates with a flooded suction. With this arrangement, the NPSH available to the fire pump is approximately equal to the submergence of its impeller below the vessel's waterline added to atmospheric pressure, less losses due to friction and turbulence within the sea chest and the pump's suction line. The impeller's submergence can vary with the vessel's list and trim conditions; consequently, pumps used for fire service should have NPSH requirements that are less than the NPSH that can be available during emergency conditions.

Fire pumps should ordinarily never be connected to oil piping. However, they can generally be used for other seawater applications, provided that at least one pump is available for use in the fire-main system at all times. A pressure gauge should be installed in the fire pump's discharge line. In addition, a relief valve that is set for 25 psi (172 kPa) above the fire-pump discharge pressure necessary to provide the required pressure at the highest hydrants or 125 psig(862 kPagauge), whichever is greater, is often required to be installed at the fire pump discharge. (If the maximum discharge pressure that a fire pump is capable of developing is less than these values, the relief valve may be omitted.) Fire-pump suction and discharge valves are sometimes motor operated and controls are provided so that the valves can be opened and the pump can be started remotely.

When an emergency fire pump is electric-motor driven, the motor is usually connected to the vessel's emergency switchboard. When an emergency fire pump is diesel driven, the starting system must typically have the capability for six starts within thirty minutes, including at least two starts within the first ten minutes. In addition, the fuel tank for the pump should ordinarily contain sufficient fuel for at least three hours offull-load operation, and sufficient fuel reserves should be available outside the machinery space for an additional fifteen hours of operation.

In addition to fixed fire pumps, some vessels also carry portable fire pumps that can be moved from place to place by the crew. These single-stage end-suction pumps are often close-coupled to gasoline engines. To enable it to be used in areas ofthe vessel that are above the waterline, a portable fire pump is often fitted with an integral vacuum priming pump.

Bilge Pumps

Bilge pumps are used to drain liquid from machinery-space bilges, tank tops, the shaft alley, and watertight compartments located throughout a vessel. Fluid discharged from a bilge pump is typically directed, depending on its composition and applicable regulations, overboard or to an oily-

waste holding tank. The minimum number of bilge pumps that must be installed on a vessel and their minimum capacity are usually specified by regulatory bodies based on the vessel size and type. Most vessels have multiple bilge pumps. Although some or all of a vessel's bilge pumps are often connected to a common suction main, one or more independent bilge-pump suction lines must also generally be provided in machinery spaces. In addition, the bilge pumps, together with their sources of power, may be installed in various watertight compartments throughout a vessel so that the flooding of one compartment will not prevent all ofthe pumps from being operated. Alternatively, some vessels have an emergency bilge pump that is suitable for use even when the space in which the pump is located is flooded. Controls that enable this submersible bilge pump, which is normally powered through the vessel's emergency switchboard, to be operated remotely from outside of the machinery spaces should be provided. Although a vessel's bilge pumps may also be used for other purposes, at least one pump should be available at all times for bilge dewatering. Ordinarily, each bilge well is fitted with a strainer, and an additional strainer is installed in each bilge pump's suction line. So that they do not restrict flow into the bilge pump, these strainers should be kept clean.

Single-stage centrifugal pumps mounted either horizontally or vertically are frequently used for bilge service. Although most of these pumps are driven by electric motors, in some cases, a bilge pump may be driven off a vessel's propulsion machinery. Because bilge pumps often operate with a suction lift, a non-self-priming centrifugal bilge pump must usually be connected to a vacuum priming pump. The vacuum pump may be part of a central system used to prime multiple pumps on a vessel, or it may be provided only for use with the bilge pump. With this latter option, an electric-motor-driven vacuum pump is sometimes mounted on the same baseplate as the bilge pump that it primes. Alternatively, a dedicated vacuum pump is occasionally mounted on and driven directly off the bilge pump. (When a centrifugal bilge pump's suction manifold has a connection to the sea chest, the bilge pump can sometimes be primed without the aid of a vacuum pump by partially opening the sea-suction valve in the suction manifold and allowing the incoming stream of seawater to flood the bilge pump's impeller and draw air and vapor out of the bilge suction line.) If a centrifugal bilge pump loses suction prematurely, the pump should be reprimed and then operated at a reduced speed or with a throttled discharge. The reduction in the flow rate entering the pump will usually help to reduce the ingress of air into the bilge suction tailpipe. If air is being drawn into the system at some other location, the point of entry should be found and sealed.

To eliminate the need for a separate vacuum priming pump, self-priming centrifugal pumps are sometimes used in bilge service. However, self-priming centrifugal pumps can have extended priming times when operating in systems with long lengths of suction piping; consequently, their performance should be carefully analyzed prior to being used in a bilge system. Additional types of pumps used in bilge service without vacuum priming pumps include motor-driven sump pumps that are submerged within the bilge drain wells (a float switch is sometimes provided for automatic operation); motor-, steam-, or air-driven piston-type reciprocating pumps; rotary-type vane pumps; and air-operated double-diaphragm pumps.

An oily-water separator (OWS) that removes oil from bilge water will often have one or more dedicated pumps that circulate oily water through the separator, discharge clean water overboard, and transfer separated oil to a waste-oil tank. Separate pumps may be used to transfer waste oil collected from an oily-water separator and sludge from Fa and La purifiers to an onboard sludge-treatment plant, to an incinerator where they are burned onboard the vessel, or ashore. Electric-motor-driven progressing-cavity and air-operated double-diaphragm pumps are often used in these applications.

Ballast Pumps

A ballast pump is used to transfer seawater into and out of a vessel's ballast tanks to adjust list, trim, and draft. The ballast pump may also be used during a voyage to exchange water contained within ballast tanks to prevent the introduction of nonindigenous aquatic species into coastal and inland waterways. A ballast pump, therefore, can ordinarily take suction either from a sea chest when adding water to ballast tanks or from ballast tanks when the tanks are being emptied. In addition, seawater discharged by a ballast pump can typically be directed either to ballast tanks being filled or overboard. The capacity rating of each ballast pump is based on the size of the vessel's ballast tanks, the number of ballast pumps on the vessel, and the time allotted for ballasting and deballasting operations. On some vessels, pumps used for other applications, such as bilge, fire, or seawater service, may also be connected to and used in the ballast system. On tankers, however, segregated ballast pumps are generally required.

Horizontally and vertically mounted single-stage centrifugal pumps are often used for ballast service. A centrifugal ballast pump is normally located low in the vessel and, therefore, operates with a flooded suction when it is lined up to the sea chest or begins to empty a ballast tank that is full. However, as the water level in a ballast tank being emptied drops, the submergence of the ballast pump's impeller and, therefore, the NPSH available to the pump are both continuously reduced. Consequently, ballast pumps should have relatively low NPSH requirements. So that a low NPSH requirement can be achieved with a suction specific speed that is not excessive, ballast pumps are frequently driven at speeds not exceeding 1,780 rpm. Drivers used with centrifugal ballast pumps include electric motors, hydraulic motors, and steam turbines. To maximize the amount of water removed from ballast tanks, the flow rate through a ballast pump shOi,lldbe reduced during the final stage of deballasting by speed reduction, when possible, and discharge throttling. So that centrifugal ballast pumps can be reprimed if suction is lost prematurely during operation with a suction lift, they are often connected to vacuum priming pumps.

On some vessels, vertical turbine pumps are used for ballast service. To permit the pump to take suction from multiple locations, a vertical turbine ballast pump is frequently installed in a suction can that is connected to suction piping. In addition, so that air and vapor can be removed from the can and suction piping, the pump is often fitted with self-priming valves (fig. 12-29).

HOTEL SERVICES

Flash Distilling Plant

A multistage flash distilling plant's feed pump takes suction from a sea chest and supplies seawater to the first-stage flash chamber. The pump must develop sufficient head to overcome the friction drop in the various heat exchangers, such as the distillate cooler, distillate condensers, air-ejector condenser, and seawater heater, that the feedwater typically passes through before it enters the distiller. Flow through the feed pump is generally regulated by a valve at the inlet to the evaporator that is throttled either manually or automatically to maintain a set seawater temperature at the feed heater outlet (usually approximately 170°F or nOC).

Brine remaining in the distiller's last stage is removed by a brine pump that discharges it overboard. The flow rate through the brine pump is often regulated with a discharge valve that is throttled to maintain a constant level in the evaporator's last-stage flash chamber. Packed stuffing boxes are frequently used for brine-pump shaft sealing. Additionally, a line is often provided to permit the packing to be flushed with seawater discharged from the distiller feed pump.

A distillate pump removes freshwater produced by the distilling plant from the last-stage distillate condenser and discharges it through a cooler to the distilled-water, potable-water, or reserve-feedwater tanks, or to the bilge if the water's salinity is excessive. A valve at the distillate-pump discharge is frequently throttled to regulate flow through the distillate pump and maintain a constant level in the distillate condenser.

An additional pump is provided on some vessels to transfer condensate from the seawater heater's hotwell to a freshwater drain tank. A valve in the pump's discharge line can normally be throttled to maintain a constant hotwellievei. (Ifthis condensate is returned directly to the main condenser or an auxiliary condenser through a vacuum-drag line, the feed-heater condensate pump may not be required.)

Electric-motor-driven centrifugal pumps are often used in the aforementioned applications associated with a flash distilling plant. The distiller-feed pump should be located sufficiently below the vessel's waterline so that the NPSHA is adequate. Because the distillate, brine, and feed-heater condensate pumps each take suction from a chamber in which the pumped liquid is at or near its vapor pressure, the NPSH available to these pumps is limited. Consequently, pumps used should have low NPSH requirements. In addition, to increase NPSHA and submergence, it is beneficial to locate these pumps as far below the evaporator as possible. Vent lines may be connected to brine and distillate pumps so that air or vapor in their suction lines will be drawn back into the distilling plant.

Plate-Type Distilling Plant

A feed pump used with a distilling plant having a plate-type evaporator and condenser typically takes suction from a sea chest and delivers seawater to the distiller. A portion of this water enters the evaporator section of the shell as feed. The remainder, however, serves as the motive fluid for eductors that remove air and brine from the evaporator. Mter leaving the eductors, this seawater is discharged overboard with the brine and air. A second pump is used to remove distillate from the condenser and discharge it to either freshwater tanks or the bilge. Both the eductor-feed pump and the distillate pump are generally motor-driven single-stage centrifugal-type units.

Potable-Water Pumps

A potable-water pump takes suction from a vessel's potable-water tanks and discharges potable water either to an air-charged pressure tank or directly to sinks, showers, and other potable-water fIxtures located throughout the vessel. Horizontally and vertically mounted single-stage centrifugal pumps are often used in this application. In addition, when higher pressures must be developed, two-stage centrifugal pumps or regenerative turbine pumps may be used. At least two pumps are normally provided.

A potable-water pump is often cycled on and off automatically by a pressure switch in the discharge line. Alternatively, on a vessel with high potable water consumption, a potable-water pump may be operated continuously. With this latter arrangement, however, a recirculation line must be provided to prevent the pump from overheating during periods of low demand.

A portion of the water discharged from a vessel's potable-water pumps is circulated through hot-water heaters. Mter being heated, the water is directed to sinks, showers, and other fIxtures that require hot water. Separate pumps are provided to recirculate unused water in hot-water distribution piping through the heaters so that the water remains hot. Motor-driven, single-stage centrifugal pumps are typically used in this application. In addition, many hot-water circulating pumps, which typically operate continuously, are furnished in a close-coupled configuration.

Sanitary Pumps

When seawater is utilized as a flushing medium, a vessel may have dedicated single-stage centrifugal sanitary pumps that take suction from a sea chest and discharge seawater through an air-charged pressure tank to the urinals and water closets on the vessel. (Alternatively, the flushing water is sometimes supplied through a pressure-reducing valve from another seawater system on the vessel.) Two electric-motor-driven sanitary pumps are generally provided, with each sized to meet peak-demand requirements. With this arrangement, one pump is ordinarily cycled on and off automatically by a pressure switch in its discharge line, and the second pump is a standby unit. A sanitary pump should be located sufficiently be-Iowa vessel's waterline to provide adequate NPSHA to the pump.

Sewage Pumps

On a vessel that does not have a sewage-treatment plant or marine sanitation device (MSD), a sewage pump usually takes suction from the vessel's sewage-holding tanks and discharges the tank contents to an abovedeck shore connection or overboard (subject to the limitations set forth in applicable regulations). Horizontally and vertically mounted, motor-driven, single-stage centrifugal pumps with overhung single-suction impellers are sometimes used in this application. These pumps are typically fitted with a special large-waterway casing and a "nonclog" impeller containing only two or three vanes that can pass moderately sized solids. Large covered hand holes are generally provided in the walls of the pump casings to permit internal passages to be cleaned as needed. Many ofthese pumps are fitted with a packed stuffing box for shaft sealing. To reduce leakage, grease is sometimes injected into the packing through a lantern ring connection located in the side of the stuffing box. When a mechanical seal is utilized for shaft sealing, a restriction bushing must generally be installed at the base of the seal cavity to isolate the seal from the sewage being pumped. In addition, a freshwater flush must also typically be supplied from an external source to cool and lubricate the mechanical seal's faces.

Macerator pumps are frequently used to transfer the contents of lift stations where sewage and waste (black) water may initially be collected to a larger holding tank or to an MSD. The macerator grinds the sewage and breaks up solids into small particles that are more easily treated. These pumps, which are usually provided in duplicate, may be started and stopped automatically by a float switch in the lift station. The treated liquid effiuent produced by an MSD is often pumped overboard by a float-actuated motor-driven centrifugal pump.

Chilled-Water Pumps

A chilled-water pump circulates freshwater through a chiller where the water is cooled and then through cooling coils where the water absorbs heat from air being directed to temperature-controlled spaces located throughout a vessel. The chilled water may also be used to cool electronic components. Horizontally and vertically mounted, single-stage, electric-motor-driven centrifugal pumps are generally used in this application. A pressurized or elevated expansion tank typically maintains a minimum pressure at the pump suction.

CARGO PUMPS

Cargo pumps are used to discharge liquid cargo from a vessel's tanks. In many cases, a cargo pump must be suitable to transfer a wide range ofliquids having different specific gravities, vapor pressures, viscosities, and temperatures. The capacity rating used to size a cargo pump is typically based on the number of tanks to be discharged at anyone terminal, the number of pumps available to empty the tanks, and the amount oftime allowed for the pump-out. The system head that a cargo pump must overcome is approximately equal to the pressure drop in the cargo-system piping, valves, and fittings (both on the vessel and in the terminal) added to the elevation of the liquid level in the terminal's tank above the level in the vessel's tank that is being emptied. Because this system head is not constant but changes with both the point of delivery and the cargo tank level, total head values used to size cargo pumps are often rated based on developing a desired pressure at the vessel's discharge manifold, referred to as the rail pressure, while taking suction from a tank that is half full.

The NPSH available to a cargo pump is essentially equal to the elevation of the liquid level within the tank being emptied above the pump's impeller added to the pressure of the atmosphere within the tank, less the cargo's vapor pressure and losses within the suction line. Many cargoes have relatively high vapor pressures. In addition, the liquid level within a tank being emptied and, therefore, the NPSH available to a cargo pump are continuously being reduced during the pump-out cycle. Consequently, cargo pumps must often operate with low values of NPSHA. To enable a low NPSH requirement to be achieved with a suction specific speed that is not excessive, large centrifugal cargo pumps are frequently driven at speeds not exceeding 1,780 rpm.

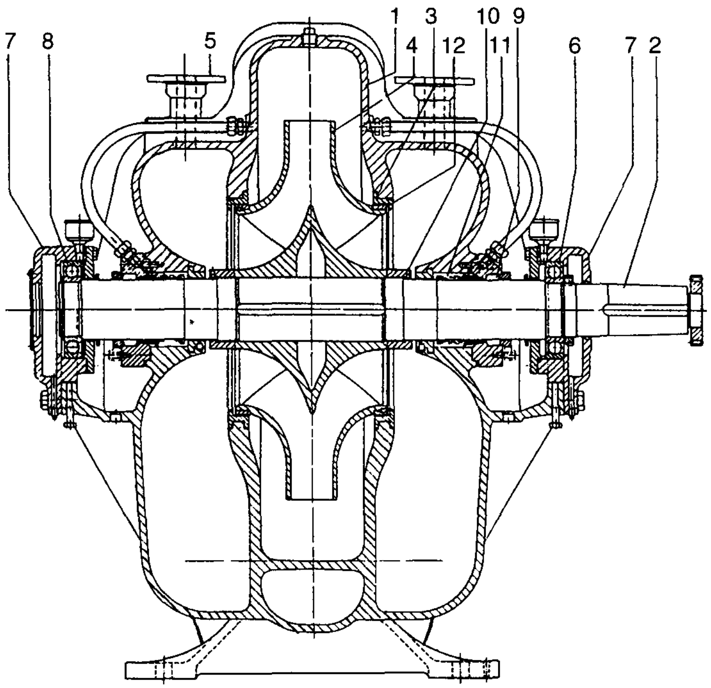
As the liquid level in a tank being emptied approaches the inlet to the suction tailpipe or, in the case of a deepwell or submerged pump, the inlet to the cargo pump, air or inert gas from the tank's atmosphere can be drawn into the pump through vortices that form on the surface of the liquid. The presence of this gas in the pumped liquid can cause a centrifugal cargo pump to lose suction prematurely. Although a positive-displacement cargo pump is typically more tolerant of entrained gas, its operation with large amounts of gas will often become erratic. Consequently, it is desirable to delay vortex formation and the ingress of gas into the suction inlet by reducing the flow rate through a cargo pump during the later stages of the pump-out. Operation at a reduced capacity will also typically reduce a pump's NPSH requirement, which can suppress cavitation. When a centrifugal cargo pump is used, the capacity being pumped can be reduced by throttling the pump's discharge valve and, when possible, reducing operating speed. With a positive-displacement pump, however, only speed reduction should generally be used to reduce flow rate. (Throttling a cargo pump's suction valve will reduce NPSHA and possibly induce cavitation; it can also lead to the starvation of the pump, overheating, and, in an extreme case, dry operation. In general, therefore, throttling of the pump's suction valve should be avoided.)

Cargo-pump materials of construction must be compatible with all of the fluids that will be discharged. In addition to the cargoes that the vessel will transport, this can include seawater if the cargo pumps will be used to remove slops during tank washing. If fluids that are flammable or explosive will be pumped, material combinations used for components with contacting surfaces should also be nonsparking.

A description of the types of pumps typically utilized in marine cargo service follows.

Centrifugal Cargo Pumps

Single-stage centrifugal pumps are often used to unload crude-oil carriers or product carriers that transport a limited number of different liquid cargoes. Typically, three or four centrifugal cargo pumps are installed in one or more pumprooms that are located in the lower part of the vessel. Through interconnected suction piping, each pump can generally be used to remove cargo from multiple tanks. These pumps are often furnished with an axially split casing and a double-suction impeller that is mounted on a shaft between two external bearings (fig. 12-45a). Pumps ofthis type, which are installed in both horizontal and vertical configurations, require two shaft seals. Alternatively, vertically mounted pumps with an impeller overhung on the end of a cantilevered shaft are also used in this application. With this latter arrangement, both of the pump's external bearings, together with the single shaft seal, are located above the impeller. Explosion-proof, intrinsically-safe resistance temperature detectors (RTDs) that activate an alarm at the cargo-control station if bearing temperatures become excessive are frequently mounted in a cargo pump's bearing housings. Openings provided for the shaft in a cargo pump's casing are usually sealed with packed stuffing boxes or mechanical seals. Glands used with cargo-pump mechanical seals often have an auxiliary stuffing box that can be fitted with packing in the event of a seal failure to enable a pump-out to continue. In addition to suction and discharge connections, one or more vents are generally provided in the suction area of a centrifugal cargo pump's casing so that gas (e.g., air or inert gas from the cargo tank) and cargo vapor can be removed from the pump during repriming or stripping operations.



1 Casing
2 Shaft
3 Casing wearing ring
4 Impeller
5 Stripping connection
6 Thrust bearing
7 Bearing housings
8 Line bearing
9 Auxiliary stuffing box
10 Shaft nut
11 Mechanical seal
12 Impeller wearing ring

Figure 12-45a. Centrifugal cargo pump.
Courtesy Ingersoll-Dresser Pump Company.

A centrifugal cargo pump can be driven by a steam turbine, diesel engine, or electric motor. The driver is usually installed in a separate machinery space adjacent to the pumproom and is flexibly coupled to the cargo pump through an intermediate jackshaft. The opening in the bulkhead (horizontal units) or overhead (vertical units) of the pumproom that the jackshaft passes through should be sealed with a gastight stuffing box to prevent explosive vapor in the pumproom from entering the machinery space. The seal's design should permit lubrication and adjustment to be

performed from outside the pumproom. In addition, seal parts should be constructed of nonsparking materials. Also, a temperature-sensing device should be used to monitor the temperature of the seal's gland from outside the pumproom. This device should be connected to audible and visual alarms located at the cargo-control station. Jackshafts used with vertically mounted cargo pumps are generally supported by a thrust bearing mounted at the deck penetration.

If a centrifugal cargo pump loses suction before a cargo tank is empty, cargo remaining in the tank can often be discharged by a separate positive-displacement stripping pump. However, the use oflow-capacity stripping pumps extends the time of the pump-out. Therefore, self-priming/stripping systems are often employed to increase the amount of cargo that can be discharged by a centrifugal-type main cargo pump and to reduce or, in some cases, eliminate the need for a separate stripping pump. Two types of systems are frequently used: the recirculation priming system and the automated cargo stripping system. They are described in "The Design and Operation of Pumps Furnished for Marine Cargo Service" (Sembler, 1988).

Vertical Turbine Cargo Pumps

Vertical turbine or deepwell pumps are often used to unload vessels that transport a variety ofliquid cargoes. When the maximum degree of cargo segregation is required, a separate vertical turbine pump (VTP) is installed in each cargo tank, which eliminates the need for suction valves and suction piping. On a double-bottom vessel, the pump's suction opening is usually submerged in a suction well located in the bottom of the cargo tank. Each VTP is generally driven by a vertical electric or hydraulic motor mounted abovedeck on top of the pump's discharge head (with some cargoes, the driver should be explosion-prooD.

Alternatively, on vessels that carry either a limited number of different cargoes or cargoes that are less sensitive to contamination, each vertical turbine pump may be used to discharge cargo from several ofthe vessel's tanks. With this latter arrangement, each VTP is generally mounted in a suction tank or can that is connected to multiple cargo tanks via suction piping (fig. 12-29). Automatic self-priming valves are often used to enable the pump to remove gas and vapor from the suction can and suction piping. Although many of these pumps have vertical drivers, some are driven through right-angle gears by horizontal motors, steam turbines, or diesel engines. On some vessels, the discharge heads of vertical turbine cargo pumps are installed belowdeck in a common pumproom. By locating horizontal drivers in an adjacent space and coupling them to the pumps through intermediate jackshafts, they can be isolated from explosive vapor in the pumproom.

Although the use of a single shaft seal is sometimes suitable, a multiple-sealing arrangement may be required when a VTP will be pumping volatile petroleum products or chemicals. On some pumps, a reservoir tank that stores liquid for seal lubrication during priming cycles or operation with loss of suction is mounted on the side of the discharge head.

Vertical turbine cargo pumps must often be capable of discharging cargoes with a wide range of temperatures. For example, lubricating oils, waxes, and other viscous cargoes can be heated to temperatures exceeding 160°F (71°C). In some cases, a VTP may even have jackets surrounding its discharge head, column pipe, and bowls through which steam or a heated liquid is circulated to keep the temperature of the cargo being pumped within a desired range. Cryogenic cargoes, such as liquified petroleum gas (LPG), which is generally cooled to a temperature of -60°F (-50°C) may also be pumped by a \ZTP. Cargo-lubricated pump bearings should be compatible with all of the cargoes that will be discharged .

. To prevent fluid in the discharge head, column pipe, and bowl assembly from draining back into the cargo tank when the pump is stopped, a vertical turbine cargo pump's suction opening is sometimes fitted with a nonreturn valve such as a flap valve (fig. 12-45b). During normal operation, the valve is open. However, after the pump-out cycle is completed, the valve is closed (manually or automatically, depending on valve design). The pump is then stopped, its abovedeck discharge valve is closed, and compressed air or inert gas is injected into the pump through a connection in the discharge head. The gas forces the cargo contained within the VTP through a bypass line connected to the bowl assembly and into the ship's piping.

Hydraulically Driven Submersible Cargo Pumps

Hydraulically driven submersible pumps are used to discharge cargo on many petroleum-product and chemical carriers and on some crude carriers. Each unit consists of a vertical single-stage end-suction centrifugal pump that is connected through a short shaft to a hydraulic motor (fig. 12-45c). The pump-and-motor assembly is mounted on the lower end of a vertical pipe stack that includes the hydraulic-oil supply and return lines and is suspended from the main deck. Typically, a separate pump is installed in each cargo tank. On a double-bottom vessel, a sump is generally provided at the bottom of the tank for the pump's suction opening. Cargo discharged from each pump's volute-type casing typically enters and flows to the main deck through a separate vertical pipe that is adjacent to the hydraulic pipe stack. Proper alignment must be maintained between the pump assembly and in-tank support rings. A control valve mounted on top of the abovedeck cover plate is used to vary the flow of hydraulic oil to the motor and, therefore, the pump's speed. The hydraulic oil required by all of the cargo pumps on a vessel is frequently supplied by a central hydraulic system; however, on some vessels, each cargo pump is powered by its own self-contained hydraulic power pack. Portable hydraulically driven submersible pumps are ordinarily carried to enable cargo to be discharged from a tank in which the main pump has failed.

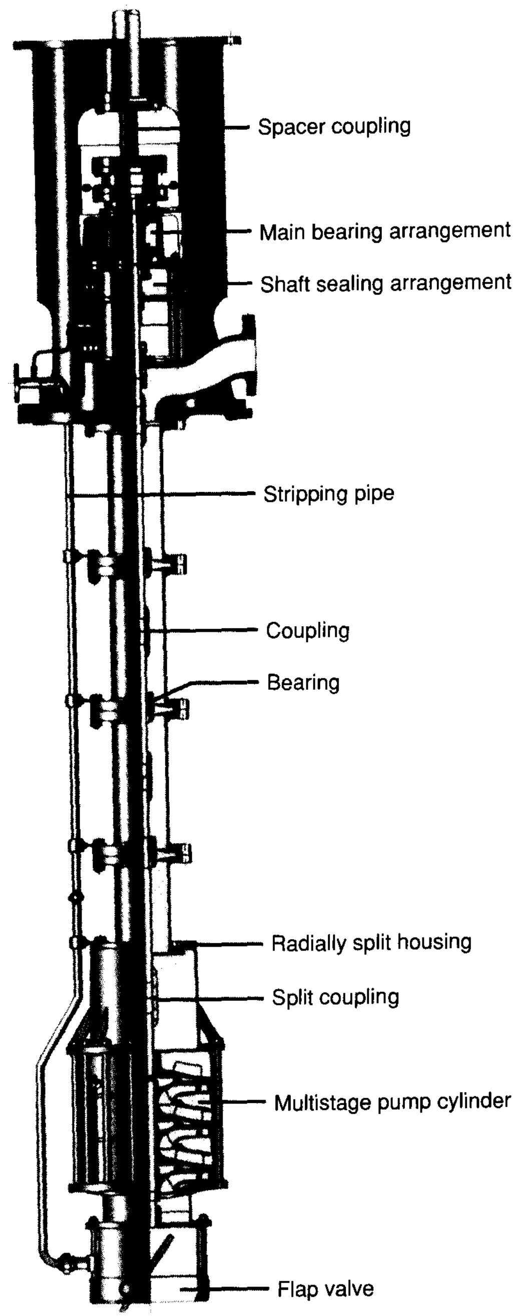


Figure 12-45b. Deepwell cargo pump. Courtesy Svanehoj International AlS, a member of Hamworthy Marine Limited.

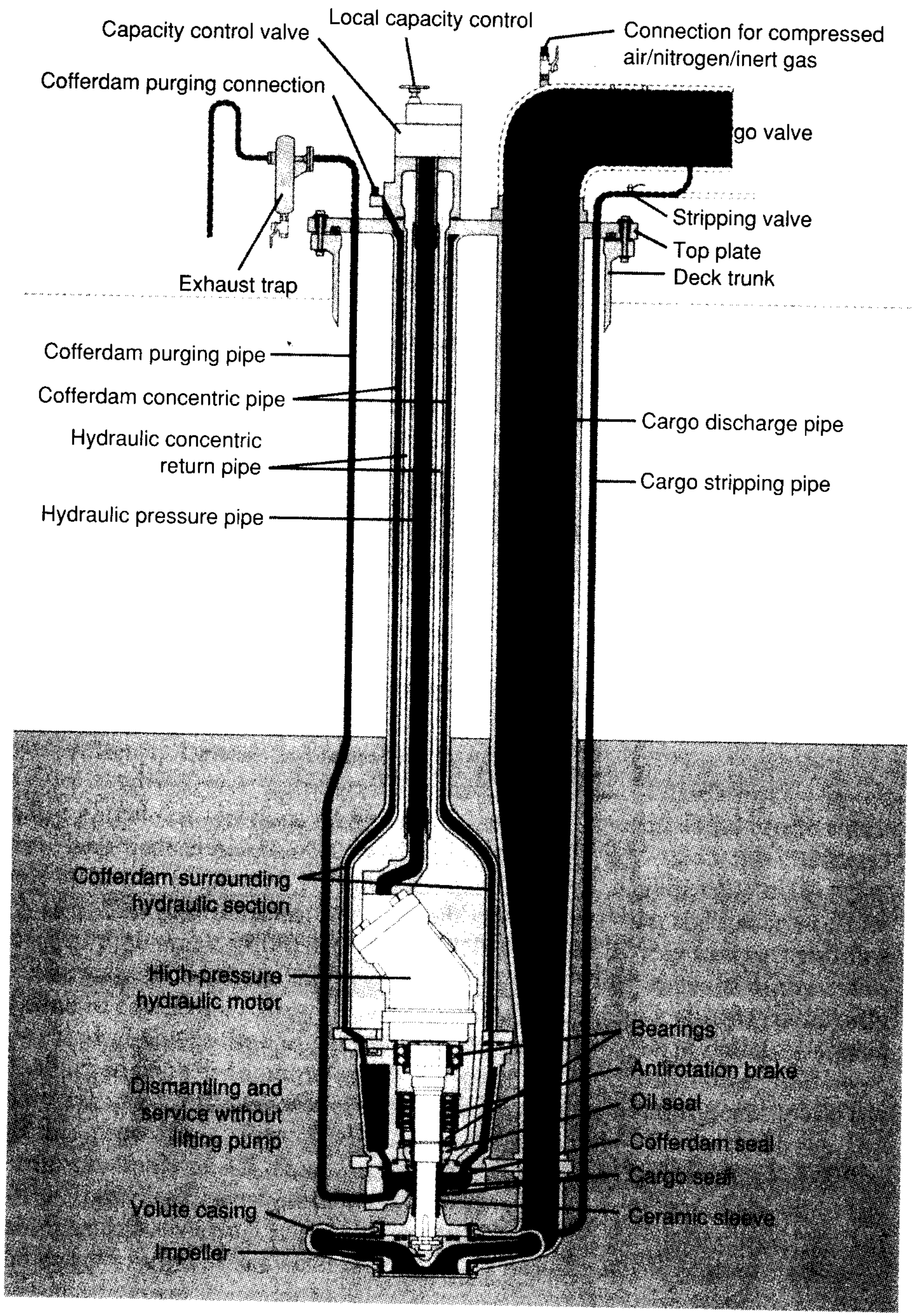


Figure 12-45c. Submersible hydraulically driven cargo pump. Courtesy Frank Mohn A/S.

Support for the pump's shaft is provided by antifriction ball or roller bearings, which are submerged in and lubricated by the hydraulic oil that drains from the motor. To simplify maintenance, the pump can typically be removed from the cargo tank without disturbing the pipe stack. Twin mechanical or lip-type seals are generally used to prevent cargo from mixing with hydraulic oil at the shaft penetration in the pump's casing. Air or inert gas can be circulated through a void space or cofferdam located within the hydraulic pipe stack and between the two seals and inspected abovedeck to permit a leak to be detected. Ordinarily, compressed air or inert gas is also used to purge cargo from the pump's vertical discharge pipe at the end of the pump-out cycle. Because the pump's inlet is generally not fitted with a nonreturn valve, purging must be performed before the unit is stopped.

On some vessels, the cargo pump vertical discharge pipes also serve as drop lines through which cargo is loaded into the cargo tanks. When this is the case, an antirotation brake or nonreverse ratchet should be mounted on each pump's shaft to prevent the pump from being driven in the reverse direction during loading by cargo flowing backwards through its impeller.

Electric-Motor-Driven Submersible Cargo Pumps

Electric-motor-driven submersible pumps are often used to discharge cargo from liquefied natural gas (LNG) and liquefied petroleum gas (LPG) carriers. Each unit consists of a vertical single-stage end-suction centrifugal pump that is mounted on the lower end of a submersible electric motor. An inducer is often installed below the inlet to the impeller to reduce the pump's NPSH requirements. A portion of the cargo discharged from the impeller passes through and cools the motor; it also lubricates the ball bearings that support the common pump and motor shaft. The unit's discharge flange at the top of the motor is generally connected to a portion of the vessel's discharge pipe that extends from the bottom of the cargo tank to the main deck. Cables that carry electrical current to the motor are enclosed in special insulation to protect them from the cargo and are connected to the ship's wiring in an explosion-proof junction box installed abovedeck. (See volume 2 of the *Modern Marine Engineer's Manual* for additional information regarding cryogenic cargo systems.)

Rotary Cargo Pumps

Rotary multiple-screw and gear pumps are often used to discharge high-viscosity cargoes. In addition, some vessels with centrifugal-type main cargo pumps have lower-capacity screw, gear, lobe, or sliding-vane pumps that are used to strip cargo tanks. The fluids discharged by rotary cargo and stripping pumps often do not have good lubricity. Also, during stripping, a pump can ingest large slugs of gas and vapor. Consequently, rotary pumps used in cargo service may have external bearings that are mounted outside the pumping chamber and are lubricated by grease or oil supplied from an independent source. Rotary cargo pumps must often be suitable to handle heated cargos, such as asphalt, which is commonly pumped at a temperature of approximately 280°F (138°C). Bearing housings used with these pumps are sometimes fitted with cooling coils.

Rotary cargo and stripping pumps may be installed in pumprooms located in the lower part of a ship. With this arrangement, the pumps are connected via suction piping to the vessel's cargo tanks. By installing drivers in a separate compartment, they can be isolated from explosive vapors in the pumproom. Alternatively, rotary cargo pumps may be furnished in a deepwell configuration, which can eliminate much of the suction piping. Although vertical drivers can be used with these units, deepwell rotary pumps are often driven by horizontal motors or diesel engines through right-angle gears. Torque from the abovedeck vertical driver or right-angle gear is transmitted to the pumping rotors through a line shaft that is enclosed within a vertical column pipe. A pressurized forced-feed system is often provided to circulate lubricating oil through the pump's timing gears, when used, and the pump and line-shaft bearings.

Reciprocating Cargo Pumps

Duplex direct-acting reciprocating-piston pumps are used to strip cargo tanks on some vessels. When installed on a vessel that carries crude oil or a limited number of different cargoes, reciprocating stripping pumps are usually mounted in a pumproom and are connected to the vessel's cargo tanks through suction piping. This enables one pump to be used to strip multiple tanks. Pumps installed in this fashion frequently are driven by steam. Alternatively, when used on multiproduct carriers, a separate reciprocating stripping pump may be installed in the bottom of each cargo tank. Typically, when this latter arrangement is used, pumps are driven by compressed air or inert gas.

Inert-Gas Related Pumps

A tankship that uses flue gas (either from a fossil-fueled steam boiler or a dedicated oil-fired inert-gas generator) to inert cargo tanks typically has a pump that delivers seawater to a scrubber where the gas is cooled, cleaned, and desulfurized. A separate pump is generally used to supply seawater to a deck seal that prevents vapor in the vessel's cargo tanks from flowing backwards through the inert-gas supply piping into the machinery spaces. Single-stage electric-motor-driven centrifugal pumps are usually used in both of these applications.

Tank-Cleaning Pump

On vessels that use seawater to clean cargo tanks, a tank-cleaning or tank-washing pump takes suction from a sea chest and discharges seawater through a heater where it is often heated to a temperature of approximately 200°F (93°C). The hot seawater is then directed to nozzles in tank-washing machines and is sprayed onto the sides of the cargo tanks being cleaned. Single- and two-stage centrifugal pumps that are driven by electric motors or steam turbines are generally used in this application.

COMPRESSORS

A compressor is used to increase the pressure of a gas. There are two basic types of compressors: the kinetic type, which includes centrifugal and axial compressors, and the positive-displacement type, which includes reciprocating and rotary compressors. Compressor capacity ratings are generally based on the volumetric flow rate of gas entering a compressor.

As discussed in chapter 1, ideal compression is a reversible adiabatic process. Based on this, and using subscripts 1 and 2 to refer to the compressor suction and discharge, respectively

$$H_{ad} = C_{10}(h_2 - h_1) + \frac{V_2^{\,2} - V_1^{\,2}}{C_3 2g_c} \tag{12.17}$$

where

H_{ad} = adiabatic head developed by compressor, ft-lbf/lbm (kJ/kg)
C_{10} = 778 when using the USCS units shown (1 for the metric units)
h = enthalpy per unit mass, Btu/lbm (kJ/kg)
V = velocity, ft/s (m/s)
C_3 = 1 when using the USCS units shown (1,000 for the metric units)
g_c = 32.2 ft-lbm/lbf-s^2 (1)

In addition

$$P_{ad} = \frac{\dot{m}H_{ad}}{C_{11}} \tag{12.18}$$

where

P_{ad} = power required for reversible adiabatic compression, hp (kW)
$\dot{m}$ = mass flow rate, lbm/s (kg/s)
C_{11} = 550 when using the USCS units shown (1 for the metric units)

If the change in kinetic energy across the compressor, which is generally small, is ignored, and substitutions are made based on the ideal gas laws, the power required to drive a single-stage compressor can be determined as follows:

$$P_C = \frac{P_{ad}}{\eta_{ad}\,\eta_m} = \frac{1}{C_{12}}\left(\frac{p_1 q_1}{\eta_{ad}\eta_m}\right)\frac{k}{k-1}\left(r_p^{\frac{k-1}{k}} - 1\right) \tag{12.19}$$

where

P_C = power required to drive compressor, hp (kW)
η_{ad} = isentropic adiabatic compression efficiency, %/100
η_m = mechanical efficiency of compressor, %/100
C_{12} = 229.2 when using the USCS units shown (3,600 for the metric units)
p = absolute pressure, psia (kPa abs)
q = volumetric flow rate, cfm (m^3/hr)
k = gas specific-heat ratio (1.395 for air)
r_p = compression ratio = p_2/p_1

The terms η_{ad} and η_m are sometimes combined into a single compressor efficiency:

$$\eta_C = \eta_{ad}\eta_m \tag{12.20}$$

where

η_C = compressor efficiency based on isentropic adiabatic compression, %/100

Equation 12.19, which is for a single-stage compressor, can also be used to calculate the power required to drive a multistage compressor when there is no cooling within or between the stages. However, multistage compressors are frequently fitted with intercoolers that reduce the temperature of the gas being compressed as it travels from one stage to the next. This reduces the volumetric flow rate and, therefore, the power required for compression in additional stages. In addition, because of the reduction in temperature, a portion of any water vapor that may be mixed with the gas normally condenses and can be drained from the cooler. The reduction in gas temperature also simplifies lubrication requirements and reduces the likelihood of overheating the compressor. When intercoolers are used, the entire compression process is no longer adiabatic. Still, the compression occurring within each stage prior to intercooling can be compared to an adiabatic process. With perfect or ideal intercooling, the interstage temperature would be reduced to the stage inlet temperature. Based on this condition and an equal compression ratio in each stage, which results in

equal stage loading and minimizes the total work, the power required for reversible adiabatic compression in a multistage compressor fitted with ideal intercoolers can be determined using the following:

$$P_{ad} = \frac{1}{C_{12}}(n\,p_1 q_1)\frac{k}{k-1}\left(r_p^{\frac{k-1}{nk}} - 1\right) \tag{12.21}$$

where

n = number of compressor stages

Additionally, the power required to drive an actual multistage compressor with intercoolers can be determined using the adiabatic efficiency and the following:

$$P_C = \frac{P_{ad}}{\eta_{ad}\eta_m} = \frac{P_{ad}}{\eta_C} = \frac{1}{C_{12}}\left(\frac{n\,p_1 q_1}{\eta_C}\right)\frac{k}{k-1}\left(r_p^{\frac{k-1}{nk}} - 1\right) \tag{12.22}$$

Alternatively, the compression within each stage of a compressor is sometimes compared to a nonadiabatic polytropic process, which results in the following:

$$P_C = \frac{P_{\text{poly}}}{\eta_{\text{poly}}\eta_m} = \frac{1}{C_{12}}\left(\frac{p_1 q_1}{\eta_{\text{poly}}\eta_m}\right)\frac{\gamma}{\gamma-1}\left(r_p^{\frac{\gamma-1}{\gamma}} - 1\right) \tag{12.23}$$

where

P_{poly} = power required for ideal polytropic compression, hp (kW)
η_{poly} = polytropic compression efficiency, %/100
γ = polytropic exponent

and

$$\eta_{\text{poly}} = \left(\frac{\gamma}{\gamma-1}\right)\left(\frac{k-1}{k}\right) \tag{12.24}$$

When $\gamma = 1$, which represents isothermal compression, the power required to drive a compressor can be found as follows:

$$P_C = \frac{P_{\text{iso}}}{\eta_{\text{iso}}\eta_m} = \frac{1}{C_{12}}\left(\frac{p_1 q_1}{\eta_{\text{iso}}\eta_m}\right)\ln\left(r_p\right) \tag{12.25}$$

where

P_{iso} = power required for ideal isothermal compression, hp (kW)
η_{iso} = isothermal compression efficiency, %/100
ln = natural logarithm

The preceding compressor power equations are based on the assumption that the gases being compressed follow the ideal or perfect gas laws. Although this generally results in sufficient accuracy for lower pressure applications, at high pressures, compressibility factors should be incorporated into these equations to account for the behavior of real gases.

EXAMPLE 12-11: A two-stage air compressor with an intercooler and an efficiency of 75 percent (based on isentropic adiabatic compression) has a rated inlet capacity of 180 m^3/hr. Air enters the compressor at atmospheric pressure and is discharged at a pressure of 700 kPa gauge (801.4 kPa abs). Determine the power required to drive the compressor.

Solution: Using equation 12.22

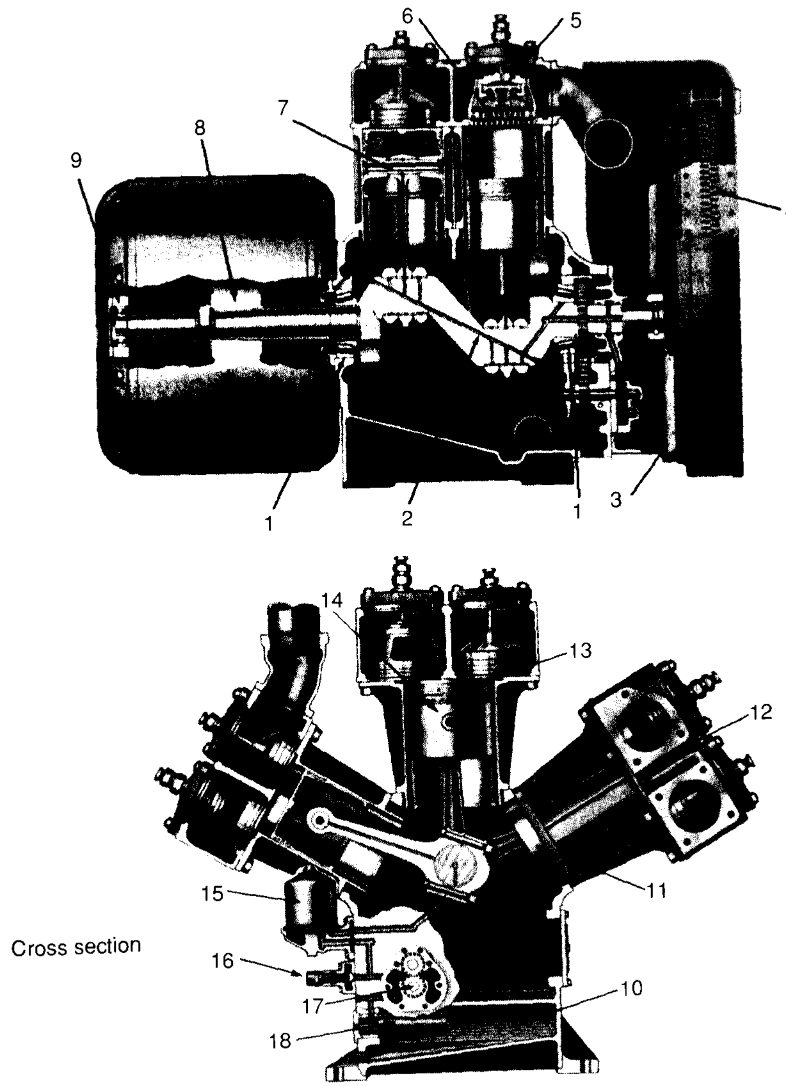
$$P_C = \frac{1}{3{,}600}\left[\frac{2(101.4\,\text{kPa})180\,\frac{\text{m}^3}{\text{hr}}}{0.75}\right]\left[\frac{1.395}{1.395-1}\right]\left[\left(\frac{801.4\,\text{kPa}}{101.4\,\text{kPa}}\right)^{\frac{1.395-1}{2(1.395)}}-1\right] = 16.2\,\text{kW}$$

In addition to intercoolers that cool gas traveling between successive stages in a multistage compressor, aftercoolers are often used to reduce the temperature, the volume, and, even more importantly, the moisture content of gas discharged from a compressor.

Positive-Displacement Compressors

RECIPROCATING COMPRESSORS

A reciprocating compressor is a positive-displacement machine in which gas is drawn into, compressed in, and discharged from one or more enclosed cylinders by close-clearance pistons that move back and forth within the cylinders (fig. 12-46). Each piston is connected, either directly or through an intermediate piston rod and crosshead, to a connecting rod that is driven by the compressor's rotating crankshaft. Reciprocating compressors are classified as either horizontal or vertical based on the orientation of the reciprocating motion within the cylinders. Horizontal compressors are sometimes supplied with multiple opposed cylinders located on opposite sides of a horizontal crankshaft. In some vertical compressors, the cylinders are located directly above the axis of a horizontal crankshaft in an in-line configuration. Alternatively, a vertical compressor's cylinders may



1 Roller bearing
2 Lubricating-oil passages
3 Fan
4 Intercooler (air cooled)
5 Suction-valve unloader
6 Cylinder valve (suction side shown)
7 Piston pin
8 Motor rotor
9 Bearing
10 Lubricating oil
11 Cylinder
12 Cylinder head—separate intake and discharge air passages
13 Piston rings
14 Piston
15 LO filter
16 LO relief valve
17 LO pump
18 LO-pump suction screen

Figure 12-46. Two-stage six-cylinder W-type reciprocating compressor. Courtesy Dresser-Rand.

be arranged in groups that are oriented radially about the crankshaft, which reduces the compressor's length. A V- or V-type compressor's cylinders are grouped into pairs and arranged so that the two cylinders in each pair are located on opposite sides of the crankshaft's axis. A W-type compressor's cylinders are divided into groups that each contain two outer cylinders located on opposite sides of a third vertical cylinder. All of the pistons that operate within a single radial group of cylinders are frequently linked through an articulated connecting rod to a common crankthrow on the crankshaft.

A reciprocating compressor with multiple cylinders that operate in series is referred to as a multistage compressor. A multicylinder reciprocating compressor may also have some cylinders that operate in parallel as part of a common stage, and others that operate in series as parts of separate stages. For example, a three-cylinder reciprocating compressor can be supplied in a two-stage configuration with two first-stage cylinders that operate in parallel and discharge jointly into the remaining third cylinder, which operates in series with the two first-stage cylinders and forms the compressor's second stage. The bore sizes of a compressor's cylinders are often included in the nomenclature. A two-stage, three-cylinder compressor with a nomenclature of 6 in./6 in./5 in. x 5 in.052 mm/152 mm/127 mm x 127 mm) would have two first-stage cylinders with 6-in. 052-mm) bores followed by a single second-stage cylinder with a bore of 5 in. 027 mm). The finaI5-in. 027 -mm) dimension following the "x" identifies the length of the compressor's stroke. Because the volume of a fixed quantity of gas is reduced as the gas is compressed, the bores of cylinders and the corresponding outside diameters of pistons used in successive stages of a multistage compressor are usually progressively reduced.

Pistons used in reciprocating compressors may be classified as single-acting when only one face compresses gas or double-acting when gas is compressed by both of the piston's faces. A piston rod and crosshead are generally installed between a double-acting piston and its connecting rod. A trunk-type single-acting piston, however, is generally driven directly by its connecting rod and often has an extended sidewall, or skirt, that absorbs side loads transmitted from the connecting rod.

Because of the reduction in piston diameter in the latter stages of a multistage compressor, a single piston with multiple outside diameters, referred to as a differential-multistage piston, is sometimes used to simultaneously compress gas in two or more separate cylinders that have a common centerline. The lower portion (drive end) of a typical vertical, two-diameter, differential-multistage piston has a larger diameter than the upper portion and is used in one ofthe compressor's lower-pressure stages. With this arrangement, gas in the larger-diameter lower portion ofthe cylinder is compressed within the annulus formed between the cylinder's bore

and the outside diameter of the piston's smaller-diameter upper section. Gas is also compressed by the upper face of the piston in a smaller-diameter cylinder that forms one of the compressor's higher-pressure stages. Stepped double-acting pistons with three different diameters that operate simultaneously in three separate stages have also been used in some reciprocating compressors.

To reduce weight, reciprocating-compressor pistons may be hollow. In addition, larger-diameter pistons installed in low-pressure stages are sometimes furnished in aluminum. Pistons in high-pressure stages, however, are frequentIymade from cast or ductile iron, which may also be used for cylinders. To permit the necessary close-clearance seal between a moving piston and the inside wall of its cylinder to be renewed periodically, pistons are typically fitted with replaceable compression rings. In addition, cylinders are sometimes fitted with replaceable liners.

Each cylinder in a reciprocating compressor must be fitted with valves to control the admission and the exhaust of the gas being compressed. A strip-type valve used in some low-pressure compressor stages contains multiple flat strips with fixed ends that flex in the middle to uncover ports in the valve seat. Alternatively, a strip-type valve may be fitted with rigid channels that are held in place by leaf springs and lift without flexing to uncover the valve's ports. A ring-plate valve, which contains several concentric ring-shaped plates that are held in place over ports in the valve's seat by multiple coil springs, and a dished-disk valve, which includes a dish-shaped disk with a spherical seating surface that is held against the valve's port by a single coilspring, are among the valve types used in intermediate- and high-pressure stages, respectively. Each of the aforementioned types of valves is actuated by the pressure created within a reciprocating compressor's cylinder. As the compressor's piston begins a suction stroke, the pressure within the cylinder is reduced until it is less than the pressure at the compressor's inlet. At this point, the suction valve opens and gas enters the cylinder. The low pressure within the cylinder keeps the discharge valve closed and prevents high-pressure gas in the discharge line from re-entering the cylinder. Once the piston begins the compression stroke, the pressure within the cylinder rises and forces the suction valve to close. When the pressure within the cylinder exceeds the pressure in the discharge line, the discharge valve is forced open and the compressed gas is discharged from the cylinder. Although they are less common, cam-operated, spring-loaded poppet valves are used in some reciprocating compressors.

Compressed gas remaining in a reciprocating compressor's cylinder after the piston completes its compression stroke reexpands and reduces the amount of new gas that can enter the cylinder during the next suction stroke. Leakage past piston rings, packing, and valves also reduces the capacity delivered by a reciprocating compressor. Furthermore, this capacity is reduced by the inability of a cylinder to fill completely during each suction stroke and by the expansion of gas entering a cylinder because of the pressure drop across the suction valve, the heat absorbed from the hot cylinder walls, and heating due to friction. A reciprocating compressor's volumetric efficiency, which is equal to the actual volumetric capacity entering a compressor divided by theoretical capacity based on the volume displaced by the pistons, generally drops as the compression ratio increases. If the volume within cylinders displaced by piston rods is ignored, the average capacity entering a reciprocating compressor fitted with either single-acting or double-acting flat-faced (i.e., not the differential-multistage type) pistons in the first stage can be estimated from the following:

$$\bar{q}_1 = \frac{f}{C_{13}}\left[\frac{\pi\left(d_{p1}\right)^2}{4}\right] S N n_1 \eta_V \tag{12.26}$$

where

$\bar{q}_1$ = average capacity entering the compressor, cfm (m^3/hr)
f = 1 for single-acting pistons, 2 for double-acting pistons
C_{13} = 1,728 when using the USCS units shown (1.67E+7 for the metric units)
π = 3.1416
d_{p1} = first-stage piston outside diameter, in. (mm)
S = stroke length, in. (mm)
N = operating speed (equal to strokes per min/2), rpm
n_1 = number of cylinders in first stage
η_V = volumetric efficiency = actual inlet capacity/theoretical inlet capacity, %/100

EXAMPLE 12-12: A 6 in./5 in. × 5 in., two-stage, two-cylinder reciprocating compressor fitted with single-acting pistons is operated at a speed of 900 rpm. The compressor's volumetric efficiency is 90 percent. Estimate the average capacity entering the compressor.

Solution: Using equation 12.26

$$\bar{q}_1 = \frac{1}{1{,}728}\left[\frac{\pi(6\text{ in.})^2}{4}\right] 5\text{ in.}(900\text{ rpm})\,1\,(0.90) = 66.3\text{ cfm}$$

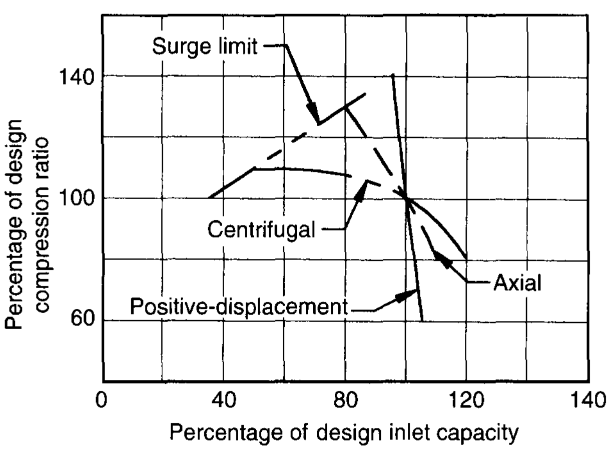
Gas that blows past a compressor's piston rings during compression strokes often enters the crankcase. When the leakage of this gas into the atmosphere or of atmospheric air into the crankcase must be prevented, a shaft seal, which is often a spring-loaded mechanical seal, is typically installed at the location where the drive-end of the crankshaft passes through the compressor's casing.

Crankshafts are sometimes fitted with balance weights to offset the unbalance created by the eccentric crank throws. Although crankshafts in reciprocating compressors are frequently supported by antifriction-type roller bearings, sleeve-type journal bearings are sometimes used. Sleeve-type bearings are also generally provided to absorb radial loads at both ends of each connecting rod. The crank-pin bearing is located at the crankshaft end of the connecting rod. The bearing at the opposite end of the connecting rod is called a piston- or wrist-pin bearing when no crosshead is used and is referred to as a crosshead-pin bearing in compressors that have crossheads. In small compressors, lubricating oil stored in the crankcase may be splashed onto the bearings by the rotating crankshaft's eccentric crank throws. In larger compressors, however, oil is frequently delivered to the bearings through a forced-feed lubrication system by a rotary pump that is driven off the crankshaft. The pump typically takes suction from the crankcase and discharges oil through a filter to the main bearings on the crankshaft. The pressurized oil also flows through drilled ports in the crankshaft and connecting rods to the connecting-rod bearings. Mter passing through the compressor's bearings, the oil drains back into the crankcase. Crossheads, when used, are typically lubricated by oil draining from the crosshead-pin bearings.

Oil may also be used to lubricate a reciprocating compressor's pistons and cylinders. In smaller compressors with oil-lubricated cylinders, dippers are often fitted on the end of each connecting rod to splash oil onto the cylinder walls. In larger compressors, a multiplunger pump that supplies oil through individual lines to each cylinder may be provided. A check valve is ordinarily installed in each oil-supply line to prevent the gas that is being compressed from entering the lubrication system. A piston used in a compressor with oil-lubricated cylinders may be fitted with special oil-control rings that scrape oil off the cylinder wall; however, residual oil remaining in the cylinder will mix with the gas being compressed. Consequently, when oil-free compressed gas is required, a reciprocating compressor with cylinders that are not lubricated, referred to as a nonlubricated compressor, is sometimes used. Each piston in a nonlubricated compressor is separated from the oil-lubricated bearings and running gear by a distance piece located between the cylinder and the frame that houses the crankshaft and connecting rod. A crosshead and piston rod must generally be used to bridge the gap between the connecting rod and the piston. Stationary wiper and packing rings are fitted around the piston rod to prevent oil from entering the base of the cylinder. Because the piston and cylinder receives no lubrication, materials having adequate self-lubricity, such as PTFE, are typically used for piston rings and cylinder packing.

Reciprocating compressors are generally either air- or water-cooled. An air-cooled compressor's cylinders often have large external fins that increase the surface area for heat transfer. In a water-cooled compressor, freshwater or seawater is circulated throughjackets that are built into the cylinder walls, the cylinder heads, and sometimes the crankcase (to cool the compressor's lubricating oil). When freshwater is used as the cooling medium, a centrifugal pump may be provided with the compressor to circulate the water through a closed jacket-water system. A heat exchanger to cool the freshwater and a surge tank may also be included in the system. Separate air- or water-cooled intercoolers are often provided with multi-stage compressors to cool the gas travelling between successive stages. In addition, a water-or air-cooled aftercooler may be used to cool gas discharged from the compressor's final stage. Air-cooled compressors and compressors with air-cooled intercoolers or aftercoolers are frequently fitted with a belt- or crankshaft-driven fan. In some cases, the fan is an integral part of a sheave or flywheel mounted on the end of the crankshaft.

A typical performance curve for a reciprocating compressor operating at constant speed is shown in figure 12-47. Changing the compressor's operating speed will change this curve and the capacity of gas delivered by the compressor. Although steam or gas turbines and diesel engines can be used to drive reciprocating compressors, shipboard units are generally driven by electric motors. The driver can be coupled directly or through a speed reducer to the extended end of the compressor's crankshaft, torque can be transmitted to the compressor's crankshaft through a multi-V-belt drive, or, in the case ofa smaller unit, the driver's rotor may be mounted directly on the extended end ofthe compressor's crankshaft in a close-coupled configuration. Close-coupled reciprocating compressors used in some applications, such as refrigeration service, may also be hermetically sealed.



The slope of the performance curves for an actual compressor can vary from those shown based on the design of the compressor, the compressed-gas system arrangement, and the characteristics of the gas being compressed.

Figure 12-47. Representative constant-speed performance characteristics for centrifugal, axial, and positive-displacement (reciprocating and rotary) compressors

A reciprocating compressor may be fitted with unloaders that relieve pressure within the cylinders, often by venting the cylinders or holding the cylinder-suction valves open, when the compressor is stopped. A typical unloader, which can be actuated with compressed gas, pressurized lubricating oil, an electromagnetic solenoid, or crankshaft-driven weights, remains open during start-up to reduce the load on the compressor's driver until normal operating speed has been achieved. In addition, suction-valve unloaders are sometimes used to hold the suction valves open in selected cylinders as a way of regulating the capacity delivered by a reciprocating compressor during constant-speed operation. A clearance pocket, which is a chamber located at the end of a cylinder that can be opened to increase the cylinder's clearance volume (the volume of gas remaining in the cylinder at the end of the compression stroke) and decrease the capacity delivered, may also be used for constant-speed capacity control.

To reduce condensation on surfaces adjacent to cylinder jackets and water-cooled heat exchangers (e.g., oil coolers, intercoolers, and aftercoolers), automatic valves that close when the compressor is stopped or unloaded may be installed in water supply lines. Additionally, many reciprocating compressors are fitted with integral relief or safety valves to prevent cylinders and coolers from being overpressurized.

ROTARY COMPRESSORS

A rotary compressor is a positive-displacement machine in which the gas being compressed is forced through the casing by one or more rotating displacement elements. The flow discharged from a rotary compressor generally has fewer pulsations than that from a reciprocating compressor. A description of some common types of rotary compressors follows.

Single-Screw Compressors

The main rotor in a single-screw compressor consists of a screw with multiple helical grooves that is centered in a close-clearance casing and is coupled, either directly or indirectly, to a driver. A typical single-screw compressor also has a flat gate rotor positioned on each side of the screw. The two gate rotors, which resemble gears with straight radial teeth, mesh with and are driven by the screw. Gas entering the casing fills cavities that are formed between the grooves ofthe rotating screw and are sealed by the casing. As the main rotor turns, the gas is compressed against the teeth of the gate rotors.

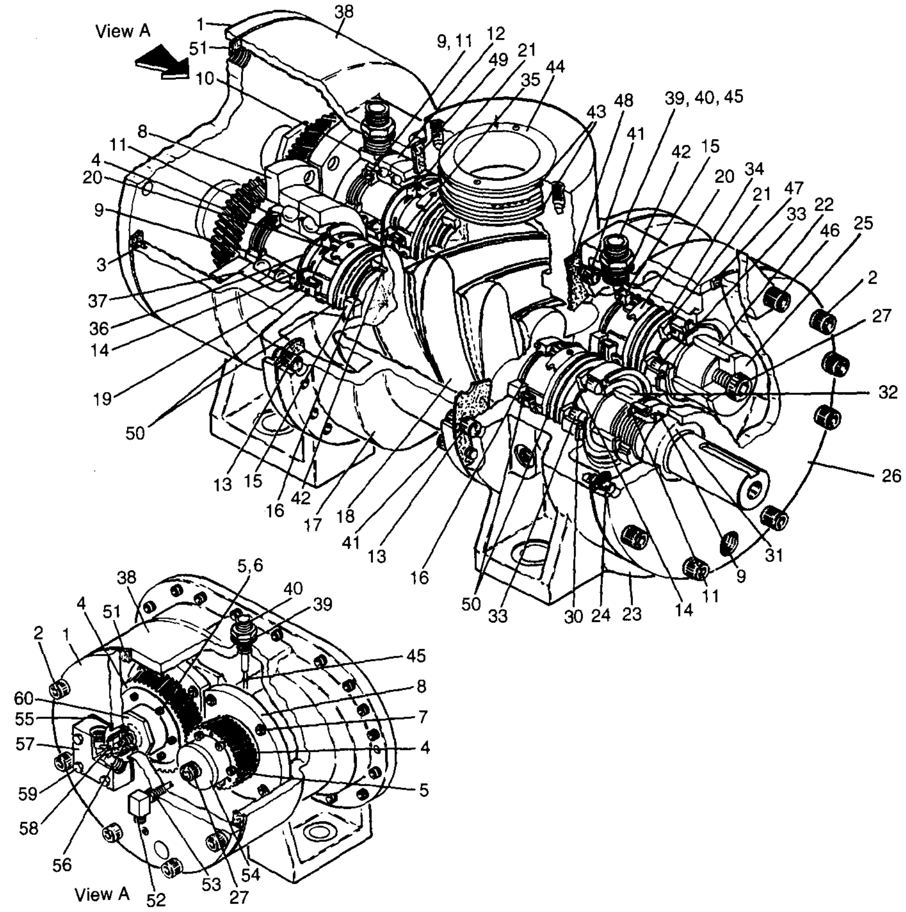
Oil is often injected directly into the casing to lubricate the compressor's components and cool the gas being compressed. Alternatively, water-flooded compressors may be used when oil-free gas is required. Water-lubricated graphite or silicon-carbide sleeve bearings are often used to support a water-flooded compressor's gate rotors and the outboard or suction end ofthe main rotor. The drive end ofthe main rotor, however, is generally supported by grease-lubricated antifriction bearings that are isolated from the gas being compressed by a mechanical or labyrinth-type seal. Additional materials used in the construction of water-flooded compressors include bronze for the main rotor and casing, stainless steel for shafts, and glass-reinforced composites for the gate rotors.

Twin-Helical-Screw Compressors

A single-stage, twin-helical-screw compressor has a male or main rotor with helical lobes that mesh with helical grooves on a female or gate rotor. The two meshing rotors, which are mounted on parallel counterrotating shafts, are enclosed withip. a close-clearance dual-bore casing (fig. 12-48). As the rotating rotors unmesh adjacent to the inlet port at one end ofthe casing, gas is drawn into the cavities formed both between the lobes in the male rotor and within the grooves in the female rotor. With the continued rotation of the rotors, the cavities move past the inlet port and the gas is forced to move axially towards the opposite end ofthe casing. The meshing of the lobes on the male rotor with the grooves in the female rotor reduces the volume of each cavity and increases the pressure of the trapped gas. Compression continues until each pocket of gas reaches the outlet port and, in the case of a single-stage compressor, is discharged from the casing. Alternatively, in the case of a two-stage compressor, the gas is directed to a second set of smaller rotors in the compressor's second stage. The rotors in both stages are typically mounted on common shafts; however, the two sets of rotors are separated by a dividing plate that reduces interstage leakage.

Many twin-screw compressors are fitted with ball and roller bearings; sleeve and tilting-pad bearings are, however, used in some larger units. Bearings are typically mounted at both ends of each shaft and are normally lubricated with oil. In an oil-flooded compressor, oil is also injected directly into the casing to lubricate and cool the rotors, reduce the discharge temperature ofthe gas being compressed, and help to seal internal clearances between the rotors and the casing. During compressor operation, the oil in the lubrication-cooling system, which is pressurized by the force of the gas that has been compressed and, in some larger compressors, by a shaft-driven rotary pump, leaves a sump or reservoir, passes through a cooler and a filter, and is directed to the bearings and to injection ports in the casing. The compressed gas and oil mixture discharged from an oil-flooded compressor generally passes through a separation device in which a large portion of the entrained oil is removed from the gas and returned to the oil reservoir. The oil reservoir is sometimes in the lower portion of the separator. In many oil-flooded twin-screw compressors, torque is transmitted directly from the lobes on the male rotor, which is on the driven or power shaft, to the grooves in the female rotor on the idler shaft.

When it is undesirable to mix oil with the gas being compressed, a nonlubricated, dry-type twin-screw compressor may be used. The casing



1 Inlet end plate
2 Capscrew
3 Rollpin
4 Timing gears
5 Capscrew
6 Washer
7 Capscrew
8 Clamping plate
9 Locknut
10 Distance sleeve
11 Lockwasher
12 Inlet end bearing set (female rotor)
13 Dowel pin
14 O-ring
15 Seal mating ring (air side)
16 O-ring
17 Rotor case
18 Rotor set
19 Seal
20 Distance sleeve
21 Seal mating ring (oil side)
22 Distance sleeve
23 Outlet case
24 Dowel pin
25 Retaining washer
26 Outlet end plate
27 Capscrew
30 Internal retaining ring
31 Seal
32 Distance sleeve
33 Outlet end rotor shaft bearing
34 Retaining ring
35 Distance sleeve
36 Adjusting washer
37 Inlet end bearing set (male rotor)
38 Inlet case
39 O-ring
40 Nozzle
41 Capscrew
42 Pin
43 O-ring
44 Water injection ring
45 O-ring
46 Capscrew
47 Gasket
48 Shim-gasket
49 Shim-gasket
50 O-ring
51 Gasket
52 Gear oil nozzle
53 O-ring
54 Gear washer
55 Gasket
56 Spring
57 Oil pump support
58 Oil pump
59 Capscrew
60 Adapter

Figure 12-48. Twin-screw compressor. Courtesy Dresser-Rand.

furnished with a dry compressor is often fitted with external jackets through which cooling water is circulated. Alternatively, in some nonlubricated compressors, freshwater is injected directly into the compressor's casing, which cools the gas being compressed and helps to seal internal clearances. The compressed-gas-and-water mixture discharged from the compressor generally enters a separator-holding tank in which a portion of the entrained water is removed from the gas and collected for reinjection. The compressed gas within this tank provides the force necessary to circulate the water through a cooler and a filter and return it to the compressor's casing. Connections and valves for adding make-up water to the freshwater injection system are typically provided.

To maintain a clearance between the nonlubricated rotors, torque from the driven shaft to the idler shaft in a nonlubricated twin-screw compressor is generally transmitted through timing gears that are fitted on the end of each shaft. The timing gears must be sized so that the female rotor rotates at the proper speed, which is generally different from the speed of the male rotor. For example, some twin-screw compressors have a four-lobe male rotor that meshes with a female rotor containing six grooves. With this arrangement, the female rotor must rotate at a speed equal to two-thirds of the male rotor's speed. Although smaller compressors may rely on splash lubrication, larger twin-screw compressors with nonlubricated rotors are usually fitted with a shaft-driven rotary pump that delivers oil to bearings and timing gears through a forced-feed lubrication system. This lubrication system also ordinarily includes an air- or water-cooled heat exchanger and a filter. When the LO cooler is air-cooled, the compressor may be fitted with a belt- or shaft-driven fan.

Twin-screw compressors are frequently provided with cast or ductile iron casings and steel, stainless-steel, or nickel-alloy rotors. A seal must be installed at the opening in the casing for the compressor's drive shaft. In a nonlubricated compressor, additional seals are required to prevent the oil that lubricates the bearings from mixing with the gas being compressed. Various types of labyrinth, carbon ring, or mechanical seals may be used for shaft sealing. To increase their effectiveness, the shaft seals are often cooled and lubricated by oil or water.

Many twin-screw compressors are driven at speeds from 800 to 20,000 rpm by electric motors, turbines, or diesel engines. To maximize efficiency, a twin-screw compressor should be sized so that the compression ratio corresponding to the reduction in the volume of gas passing through its rotors matches the requirements of the system. The efficiency of a twin-screw compressor typically drops, however, with higher compression ratios and when heavier gases are compressed. Some twin-screw compressors are fitted with an internal slide-type bypass valve that can be opened to reduce the region of compression along the length of the rotors. This permits the compressor's capacity to be controlled during constant-speed operation by

allowing a portion of the gas entering the casing to return to suction before it is compressed.

Sliding-Vane Compressors

A typical sliding-vane compressor contains a single rotor that is mounted eccentrically within a cylindrical or a cam-shaped casing (fig. 12-49). Radial vanes fit into slots that are equally spaced around the circumference of the rotor. Centrifugal force resulting from the rotation of the rotor and, in some cases, springs on a pin connected to the inboard end of the opposing vane keep the tip of each vane in contact with the inner wall of the casing. As each vane sweeps past the casing's inlet port, it slides radially outward from the rotor, and gas is drawn into the cavity formed between the rotor and the inner wall of the casing. When the next vane passes the inlet port, it seals the cavity and forces the trapped gas through the casing. As the compressor's shaft continues to rotate, the vanes are gradually pushed back into the rotor by the eccentric casing's inner wall, which reduces the volume and increases the pressure of the trapped gas. Compression continues until the leading vane reaches the casing's outlet port. At that time the gas is either discharged from the compressor or, in the case of a two-stage compressor, directed to the second stage. Some two-stage sliding-vane compressors are furnished in an in-line configuration in which both rotors are mounted on a common shaft. Alternatively, an over-under configuration is sometimes used in which the second-stage rotor is mounted directly beneath the compressor's first stage and is driven by the first-stage rotor through gears or a timing chain. Sliding-vane compressors may also be furnished in three-stage configurations.

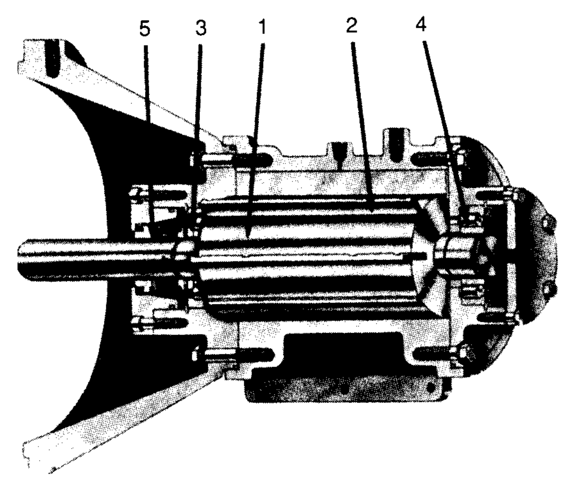


Figure 12-49. Sliding-vane compressor. Courtesy Dresser-Rand

Each end ofthe rotor in a sliding-vane compressor is often supported by a roller bearing. Mechanical seals or packing rings are generally utilized for shaft sealing. The bearings and shaft seals are frequently lubricated with oil that is pressurized by a small belt-driven multiplunger pump and distributed through individual supply lines to each point of lubrication. In some compressors, oil discharged from the multiplunger pump is also injected through ports in the cylinder to lubricate the vanes. A check valve should be installed in each oil-supply line to prevent compressed gas from blowing back through the lubrication system. A separator is frequently installed in the compressor's discharge line to remove some of the injected oil that becomes entrained in the gas being compressed. As an alternative to the individual forced-feed arrangement, in a flooded sliding-vane compressor, oil is continuously injected into the cylinder at a high flow rate both for lubrication and to cool the gas being compressed. Mter the oil-gas mixture is discharged from a flooded compressor, much of the oil is typically separated from the gas that has been compressed. This oil is then cooled, filtered, and returned to the compressor. Oil circulation is generally maintained by the pressure of the compressed gas or by a shaft-driven gear pump.

Many sliding-vane compressors have a cast-iron casing and a steel shaft that has either an integral rotor or a separate cast-iron rotor. Typical vane materials include aluminum, laminated cloth impregnated with phenolic resins, and carbon. To reduce the temperature of the gas being compressed, freshwater is often circulated through external jackets that surround the casings used with many forced-feed lubricated sliding-vane compressors. Mtercoolers and, in the case of two-stage compressors, intercoolers are also frequently provided. External cooling jackets, intercoolers, and aftercoolers are generally not required, however, for flooded sliding-vane compressors. Sliding-vane compressors are generally driven through couplings, reduction gears, or belts by electric motors. Engines and turbines have also been used as sliding-vane-compressor drivers.

Rotary Compressor Performance

A typical performance curve for a rotary compressor operating at constant speed is shown in figure 12-47. As shown, inlet capacity remains relatively constant at a given speed. Consequently, the capacity delivered by a rotary compressor can be adjusted by varying operating speed. In addition, an unloader is sometimes used to control capacity during constant-speed operation. When the pressure in the discharge line reaches a preset value, a typical unloader device simultaneously closes a valve at the inlet to the compressor and opens a bypass valve that allows the compressor to discharge to suction. This relieves the pressure upstream (i.e., on the compressor side) of a discharge check valve. If the pressure in the discharge line downstream of the check valve drops below a specified value, the unloader allows the bypass valve to close and the inlet valve to reopen so that

the compressor can again deliver pressurized gas to the system in which it operates. An unloader is sometimes used in conjunction with pressure switches to unload and shut down a compressor when the discharge pressure rises above a high-pressure set point and then to restart and reload the compressor when the discharge pressure drops below the low-pressure set point. To reduce the number of compressor starts and stops, a delay may be incorporated intO'the control system so that the compressor is stopped only after it has operated unloaded for a preset period of time.

LIQUID-RING COMPRESSORS

Liquid-ring compressors, also referred to as liquid-piston compressors, are identical in design to the wet-type, liquid-ring vacuum pumps described previously (fig. 12-30). Liquid-ring compressors are used primarily in applications requiring oil-free compressed gas. Due to the use of a liquid compressant, they are also suitable for compressing gases that are already mixed with a liquid prior to entering the compressor. It is the liquid compressant (generally water) and not the compressor's rotor that actually compresses the gas being handled; consequently, the wear rate of internal components in liquid-ring compressors is typically low. Also, because the liquid compressant absorbs much of the heat generated during compression, the need for intercoolers or aftercoolers is eliminated. A separator is generally installed at the compressor's outlet to remove the liquid compressant that is mixed with the gas being discharged. Mter being separated from the compressed gas, this liquid is typically circulated through a cooler and returned to the compressor.

Dynamic Compressors

The velocity and pressure of a gas passing through a dynamic compressor are increased by a spinning rotor. The theory of performance for dynamic compressors is similar to that presented previously for centrifugal pumps, except that compressibility must be considered in the case of a compressor. Dynamic compressors with radial flow at the rotor discharge are usually referred to as centrifugal compressors, and dynamic compressors with axial flow at the rotor discharge are ordinarily called axial compressors.

CENTRIFUGAL COMPRESSORS

Centrifugal compressors, which are often similar in appearance to radial-flow centrifugal pumps, are furnished in both single-stage and multistage configurations (fig. 12-50). The impeller in a single-stage centrifugal compressor is often overhung on the end of a cantilevered shaft. An overhung arrangement is also utilized in some multistage compressors to support multiple in-line impellers mounted on a common shaft or impellers mounted on the ends of multiple shafts that are located around the periphery of and driven by a central gear. Alternatively, the impellers in single-stage and

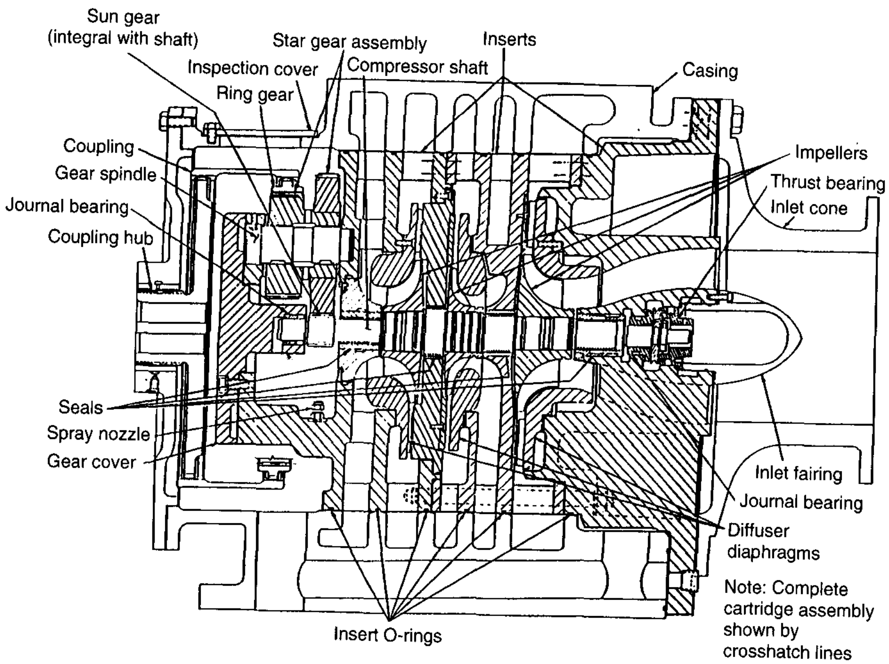


Figure 12-50. Multistage centrifugal compressor.
Courtesy Dresser-Rand.

multistage centrifugal compressors are sometimes mounted on a shaft that is supported by bearings at each end. Because the volume of a gas passing through a compressor is reduced as the pressure of the gas is increased, the widths of impellers used in multistage centrifugal compressors are often progressively reduced in each successive stage. In addition, to maintain geometric proportionality, impeller diameters are often reduced with the widths.

The high velocity of the gas being discharged from a centrifugal compressor impeller is generally reduced in a scroll-shaped volute, multivaned diffuser, or vaneless diffuser. A combination of these devices is sometimes used, such as a diffuser followed by a volute. As the velocity is reduced, a portion of the gas's kinetic energy is converted to potential energy, which results in an increase in pressure. Diffusers used in multistage compressors may be part of the stationary diaphragms that separate adjacent stages. One side of each diaphragm may also have channels that direct the gas being compressed to the eye of the impeller in the next stage. Intercoolers are often used to reduce the temperature of gas traveling between successive stages in a multistage compressor. Mtercoolers are also frequently provided to reduce the temperature of gas being discharged from the compressor. Typical materials used in the construction of centrifugal compressors include stainless steel, aluminum, and titanium for impellers and cast iron, ductile iron, steel, and stainless steel for casings.

Many centrifugal compressors have single-suction open, semiopen, or closed impellers. Because pressure differences on each side of a single-suction impeller result in axial thrust, when all ofthe single-suction impellers in a multistage compressor are mounted in the same direction, the net axial load acting on the compressor's rotor is often reduced with an internal balancing piston or drum. Alternatively, to minimize the net axial thrust applied to a compressor's shaft, single-suction impellers in a multistage compressor may be installed with a near-equal number ofthe impellers facing opposite ends of the casing. In some of these compressors, a double-flow configuration is used in which gas is simultaneously admitted to both ends of the rotor, flows inward, and is discharged from the center of the casing. The two single-suction impellers that can be mounted back to back in the center of a multistage double-flow compressor's casing are frequently replaced by a single double-suction impeller. A double-suction impeller is also installed in many single-stage double-flow compressors.

A centrifugal compressor's shaft is typically supported radially by cylindrical, multilobe, or tilting-pad journal bearings. Net axial loads acting on a centrifugal compressor's shaft are often absorbed by a thrust bearing consisting of a rotating shaft-mounted collar that is positioned axially between two stationary flat lands or between multiple pivoting shoes. These bearings are frequently lubricated with oil that is pressurized and circulated by a shaft-driven rotary pump. An oil filter and a cooler may also be included in a compressor's lubrication system.

To reduce internal leakage and improve volumetric efficiency, replaceable close-clearance stationary seals are often fitted adjacent to the outer hubs of closed impellers. In addition, close-clearance labyrinth-type seals are typically installed between adjacent stages in a multistage compressor. Seals are also required at locations where the compressor's shaft penetrates the casing. Various types of labyrinth, carbon ring, or mechanical seals may be used for shaft sealing. To increase their effectiveness, in some cases, the shaft seals are pressurized with either liquid or air supplied from an external source.

AXIAL COMPRESSORS

Axial compressors, which are typically used to deliver relatively high capacities, generally have multiple stages that each consist of a row of rotating vanes followed by a row of nonrotating diffuser vanes. A row of nonrotating inlet guide vanes is frequently located on the inlet side ofthe initial row of rotating vanes. Accurate vane-to-vane spacing and orientation are critical for proper performance.

Vane profiles used in an axial compressor are frequently based on airfoil shapes. It is common for an enlarged section at the root of each rotating vane to be machined in the form of a dovetail, a fir tree, or some similar shape and inserted into a mating circumferential groove in the rotor. A typical rotor consists of a hollow drum, a solid drum that is integral with the shaft, or individual disks that are either installed on a common shaft or stacked and held together with through-bolts. When either the stacked-disks or hollow-drum configuration is used, a stub shaft is generally pressed into each end of the rotor.

The nonrotating stator vanes may be mounted in slots machined directly into the casing or installed in a separate inner carrier. In some compressors, the stagger angle of the stator vanes can be adjusted during operation to control the capacity of gas delivered by the compressor. The inner ends of both movabl~ and nonmovable stator vanes are sometimes fitted with thin cylindrical shrouds that join groups of vanes in a single stage. To reduce interstage leakage, a close clearance is ordinarily maintained between the inner ends of the stator vanes and the rotor.

The volume of a gas passing through a multistage compressor is reduced as the gas travels from stage to stage. To permit the axial velocity of gas passing through an axial compressor to remain relatively constant, the lengths of the rotor and stator vanes used in each successive stage frequently are progressively reduced. Several of the compressor's final stages are, however, sometimes fitted with vanes having an equal length. This results in a reduction in the velocity of the gas leaving the compressor and, therefore, in exit losses.

An axial compressor's shaft is typically supported axially by a pivoting-shoe thrust bearing. To reduce the loads that the thrust bearing must absorb, the discharge end of the rotor is often fitted with an internal balancing piston or drum that is vented to suction. Radial support of the rotor is often provided by cylindrical or tilting-pad journal bearings mounted at each end of the shaft. These bearings are lubricated with oil that is generally pressurized and circulated by a shaft-driven rotary pump. An oil filter and cooler are usually included in the compressor's lubrication system. Labyrinth seals are used in many axial compressors for shaft sealing. Materials of construction used for axial compressors are similar to those used for centrifugal compressors.

DYNAMIC COMPRESSOR PERFORMANCE

As shown in figure 12-47, during constant-speed operation, the curve of compression ratio versus inlet capacity for an axial compressor is generally steeper than the corresponding curve for a centrifugal compressor. However, the steepness of a centrifugal compressor's performance curve will ordinarily increase if the impeller-vane discharge angle (β_2 in figure 12-17) is reduced. In addition, the compression ratio developed by a dynamic compressor when operating at a constant speed and with a given inlet capacity will usually increase with the density of the gas entering the compressor.

Although centrifugal compressors, which frequently have impellers with radial- or backward-curved vanes, can typically develop higher compression ratios per stage than axial compressors, most axial compressors have higher efficiencies than comparably rated centrifugal compressors.

Variations in a dynamic compressor's inlet capacity, head, and required power with changes in operating speed and, in the case of a centrifugal compressor, impeller outside diameter can be estimated using the relationships given previously in equations 12.6a, 12.6b, and 12.6c. However, because the higher compression ratio during operation at an increased operating speed or with a larger impeller diameter increases the density of gas being discharged from a dynamic compressor, the head actually developed at the new inlet capacity given by equation 12.6a will often exceed the head found using equation 12.6b. Reductions in speed or impeller diameter can have the opposite effect. The difference between calculated and actual values ordinarily increases with higher heads, heavier gases, reduced impeller-vane discharge angles, and the magnitude of the speed or impeller-diameter change.

Typically, the range of satisfactory performance for a centrifugal or axial compressor does not extend back to shutoff but stops at a minimum flow rate. Operation below this minimum capacity, which is sometimes referred to as the surge limit, can result in extreme instability with brief periods of forward flow followed by intermittent back flow. These oscillations in flow generally result in pressure pulsations, vibration, noise, and an increase in temperature. Continued operation with these conditions can, in some cases, result in severe damage to a compressor. The capacity at which surging begins generally increases with the density of the gas entering a dynamic compressor and with operating speed. Because of this latter relationship, if surging begins in a centrifugal compressor, it can often be suppressed by reducing the compressor's speed. Once surging begins in a multistage axial compressor, however, the surge limit can shift to a higher capacity; consequently, surging will frequently continue even after the operating speed has been reduced.

The maximum capacity that can be delivered by a centrifugal or an axial compressor is frequently limited by choking, which occurs when the relative velocity of the gas passing through the rotor or the gas velocity in a high-velocity region of the casing (e.g., at the inlet to a diffuser) reaches the local acoustic velocity (i.e., the speed of sound within the gas). A reduction in a compressor's discharge pressure during operation with choked flow has no effect on the mass flow rate passing through the compressor. (During normal operation, a reduction in discharge pressure generally results in an increase in the mass flow passing through a dynamic compressor.) Gas velocities within a dynamic compressor and the potential for choking increase with operating speed. Because of this, together with the increase in the surge limit with operating speed, the curve of compression ratio versus inlet capacity for a dynamic compressor often becomes steeper as the compressor's speed is increased. In addition, as the speed of a dynamic compressor is increased, the region ofthe compressor's performance curve between the surge limit and the point at which choking begins (i.e., the useful portion ofthe curve) sometimes gets progressively smaller.

Typical operating speeds for centrifugal and axial compressors can range from 5,000 to 40,000 rpm. The maximum speed for any specific machine is often established based on stresses acting on the compressor's rotor or on limiting choked flow. Because of their high operating speeds, centrifugal and axial compressors are frequently directly coupled to and driven by steam or gas turbines. In addition, they may also be driven through speed increasers by electric motors or diesel engines. Speed increasing gears are sometimes furnished as an integral part of a motor-driven compressor.

Methods of flow control, listed in order based on relative efficiency, include varying the operating speed, adjusting the angle of inlet (prerotation) guide vanes, throttling an inlet valve, and throttling a discharge valve. In addition, the performance of some axial compressors can be adjusted during operation by varying the angular orientation of interstage stator vanes. A blowo£r(recirculation) valve is sometimes used in conjunction with the aforementioned methods of flow control to prevent the flow rate from dropping below the surge limit.

Compressor Installation and Operation

COMPRESSOR INSTALLATION

Information regarding foundations and alignment presented previously for pumps can, for the most part, also be applied to compressors. Air-cooled compressors should be installed only in locations where there is sufficient ventilation to keep the unit cool.

Compressor suction piping, when used, and discharge piping should be as short and direct as practicable and should be no smaller in diameter than the corresponding connections on the compressor. In addition, when a centrifugal compressor has suction piping, straight pipe having a length equal to at least four pipe diameters should be located directly upstream of the compressor's inlet connection. Elbows that are used in compressor piping should be of the long-radius type. If possible, piping should not have any low spots or pockets where moisture can collect. When pockets cannot be eliminated, low points should be fitted with drains. If a vertical run of piping is installed immediately before or after a compressor, any liquid (e.g., water or oil) that separates from the suction or discharge stream, respectively, can accumulate at the base of the line; consequently, this arrangement should be avoided. A separator or similar device is sometimes installed in a compressor's inlet or discharge piping to remove liquid entrained in the gas stream.

A check valve should ordinarily be installed in the discharge line of a rotary, centrifugal, or axial compressor to prevent backflow through the compressor when the unit is not running. Check valves are also frequently installed in the discharge lines of reciprocating compressors. In addition, a reciprocating or rotary compressor's discharge line should be fitted with a properly sized relief valve (upstream of any stop valves) to protect the compressor and system from overpressurization.

All piping should be independently supported and properly aligned so that loads imposed on the compressor will be minimized and within the compressor manufacturer's allowable values. Additionally, when necessary, provisions should be made to allow for thermal expansion of discharge piping. All piping should be thoroughly cleaned prior to compressor installation.

COMPRESSOR OPERATION

Prior to starting a compressor, the unit should be thoroughly inspected, and any abnormalities that are found should be corrected. When practicable, the compressor's shaft should be turned by hand to ensure that it rotates freely. (In some cases, it may not be possible or advisable to turn the shaft by hand because of the size, configuration, or design of the compressor.) When bearings or timing gears are oil-lubricated, the LO level in the reservoir should be checked, and oil should be added if needed. The lubricating oil used should be suitable for the operating temperatures and compatible with the gas being compressed, and it should be in accordance with the compressor manufacturer's recommendations. (In some oil-flooded compressors, only certain synthetic lubricants should be used.) When installed, valves in the compressor's inlet and discharge lines should be opened. Also, when applicable, valves in sealing lines, gauge lines, bearing-lubrication supply and return lines, and intercooler, aftercooler, and cooling-jacket water-supply and -return lines should ordinarily be opened.

Many compressors are designed to be started only in an unloaded condition. In addition, after start-up, some compressors should be allowed to run in an unloaded condition for several minutes before they are loaded. During this time delay, which may be controlled manually or with an automatic device, the compressor warms up and oil in the compressor's lubrication system is distributed to bearings and running gear before these components are fully loaded.

During operation, frequent checks should be made of a compressor's casing, bearings, and seals for signs of overheating or leaks. Compressed gas pressures and temperatures should be monitored. When oil is injected into the compressor or used for bearing lubrication, the level in the LO reservoir should be checked periodically and maintained at the proper value. (If a dipstick that is used to check the LO level in a reciprocating compressor is removed from the crankcase while the compressor is running, compressed gas that leaks past the compressor's piston rings can, in some cases, blow hot lubricating oil out ofthe crankcase through the hole for the dipstick. To prevent this, the dipstick should generally not be removed from the crankcase of a reciprocating compressor while the compressor is running.) When a forced-feed lubrication system is used, the LO pressure should also be monitored, and, when sight glasses are provided, oil flow through the bearings should be observed. If an LO cooler is provided, the flow rate of the cooling water passing through the cooler must often be regulated to maintain the proper oil outlet temperature. Alternatively, a thermostatically controlled valve may be provided to allow some oil to bypass the cooler. In addition, when a compressor is water-cooled or has a water-cooled intercooler or aftercooler, automatic or manual valves are typically provided to regulate the cooling water flow to maintain a set water outlet temperature. Cooling-water supply pressure and temperature and outlet temperature should be monitored while the compressor is operating. A compressor should ordinarily be stopped ifthere are any abnormal conditions (e.g., pressures, temperatures) or if there is any unusual noise or vibration. Controls are sometimes provided to stop a compressor automatically for a high discharge pressure or temperature, a low suction pressure, and a high temperature, low pressure, or low flow rate of either lubricating oil or cooling water.

TROUBLESHOOTING

Common reasons for a compressor to fail to deliver the proper capacity or develop the proper discharge pressure include excessive demand; operation at less than the design speed; dirty suction filters (filters should be replaced periodically); operation with too low a suction pressure; foreign matter in the compressor or piping; improperly adjusted or defective pressure switches or unloaders; leakage through casing joints, piping, shaft seals, or relief valves; excessive internal running clearances; mechanical defects; loose drive belts; and insufficient lubrication or cooling. In the case of reciprocating compressors, problems can also be caused by cylinder valves that fail to open or seat properly and worn piston rings or seals. In addition, problems can occur if centrifugal or axial compressors operate too close to the surge limit or to the point at which choking begins. A compressor may also be unable to deliver the proper capacity if it is operating with too high a compression rate.

Reasons for a compressor to require excessive power include rubbing contact and binding between moving and stationary parts, operation at too high a speed, excessive misalignment between the compressor and its driver, a bent shaft or piston rod, improperly supported or misaligned piping, worn or improperly lubricated bearings, overly tightened packing or drive belts, mechanical defects, foreign matter in a compressor or piping, and operation with too high a compression ratio.

Marine Compressor Applications

See volume 2 ofthe *Modern Marine Engineer's Manual* for a description of compressors used in refrigeration and air-conditioning systems.

SHIP'S SERVICE AIR COMPRESSORS

Ship's service air compressors supply compressed air, through receivers or accumulator tanks, to locations throughout a vessel for various uses, such as the operation of pneumatic tools and air-operated pumps. Discharge pressures developed by these compressors are generally in the range of 100 to 150 psig (690 to 1,035 kPa gauge); consequently, they are sometimes referred to as low-pressure air compressors. At least two units that can operate in parallel are typically installed on a vessel. An air compressor should be installed in an area of the machinery space where the incoming air will be as cool, clean, and dryas practicable. Impurities mixed with air entering a compressor can increase wear. In addition, dust and dirt can combine with lubricating oil and form gumlike carbonaceous corrosive deposits that adhere to hot surfaces, such as a reciprocating compressor's discharge valves. (Because the temperature of leaking reciprocating-compressor discharge valves will often increase, which increases the potential for carbon buildup and can eventually lead to a hazardous situation, valves should be inspected regularly and cleaned or replaced as needed.) Consequently, inlet filters should always be used with air compressors to remove contaminants mixed with the incoming air. A silencer or muffler is often incorporated into the inlet-filter housing. (Filters should never be cleaned with flammable substances.) Also, a large chamber may be installed in the inlet line just upstream from the compressor to act as a pulsation dampener and reduce any pressure fluctuations in the inlet flow. A drain or trap should be provided for the removal of any moisture that may condense in this chamber.

A section of flexible hose that is rated for the proper pressure is often installed between an air compressor's outlet and the vessel's piping. In addition, a discharge receiver, which acts as a storage chamber and a pulsation dampener, is often installed in an air compressor's discharge line. The receiver piping should be arranged to promote flow through the tank and to avoid areas of stagnation. Also, the volume of an air receiver should be large enough to enable velocities to drop sufficiently for condensed moisture and oil present in the air to separate and settle to the bottom of the tank. (Stagnant air in a receiver can accumulate oil vapor and lead to a hazardous situation.) A drain or automatic trap should be installed in the bottom of a receiver to permit water and oil to be drained from the tank. To further reduce the compressed air's water or oil content, a refrigerated drier and a filter may be installed in the receiver outlet line.

Two-stage reciprocating compressors are frequently used in this application. They can be furnished in air-cooled or water-cooled configurations and with lubricated or nonlubricated cylinders. Intercoolers and aftercoolers that are used may be fitted with traps or moisture separators to permit water vapor that condenses when the air temperature is reduced to be drained from the cooler. A breather tube is often connected from the frame to the inlet housing to prevent the frame or crankcase from being pressurized by any air that blows past the piston rings. Screw or centrifugal compressors may also be used to supply ship's service air (especially when high capacities are required).

Ship's service air compressors, which are usually driven by electric motors, are often automatically cycled on and off as necessary to keep the pressure in the air system's receivers between two preset values. Systems with multiple compressbrs may be fitted with pressure switches adjusted for sequential operation. With this arrangement, whenever the air-receiver pressure drops below a preset cut-in value, the first compressor is automatically started. If the demand exceeds the first compressor's capacity and the receiver pressure continues to drop, a second compressor starts when a second lower preset pressure is reached. As the receiver pressure increases, the compressors are automatically stopped in the reverse order from that in which they were started. Alternatively, to eliminate frequent starts and stops, unloaders that are activated when the receiver pressure reaches a preset cut-out value may be used to reduce the capacity delivered by a compressor during constant-speed operation. In addition, a control system that includes both unloaders and a time-delay relay may be used to stop a compressor after it has operated unloaded for a preset period of time. If, however, the pressure drops below the cut-in pressure, the compressor is automatically restarted and reloaded. Ship's service compressors can be protected by automatic safety devices, such as switches that stop the driver when the temperature of the air being discharged, lubricating oil, or cooling water (when used) is too high or when the pressure in the lubrication system or the cooling-water system is too low.

HIGH-PRESSURE AIR COMPRESSORS

High-pressure air compressors supply air at discharge pressures from 1,000 to 5,000 psig(6,895 to 34,475 kPagauge) for uses that include the operation of pneumatic machinery and gas-turbine starting. This air may also be supplied through a reducing valve to the ship's service air system. Four-, five-, and six-stage reciprocating compressors are often used in this application. Control systems used with high-pressure air compressors are similar to those described previously for ship's service air compressors.

DIESEL-STARTING AIR COMPRESSORS

Diesel-starting compressors charge receivers that store air used to start diesel engines. Multistage reciprocating compressors that discharge air at pressures from 250 to 580 psig (1,725 to 4,050 kPa gauge) are frequently used in this application. At least two compressors of approximately equal size that are capable of charging the starting air receivers within one hour are usually provided. A smaller topping air compressor may also be used. The total capacity of the starting air receivers must generally be sufficient for twelve consecutive starts of a reversible main engine or six consecutive starts of a nonreversible engine.

CONTROL-AIR COMPRESSORS

Control-air compressors are used to deliver compressed air to pneumatically operated instruments, valves, and control systems. Reciprocating compressors are sometimes used. In addition, because of their ability to deliver clean oil-free air, liquid-ring compressors are also used.

BLOWERS AND FANS

A blower is typically used to supply a relatively large quantity of a gas at a pressure that is generally less than 40 psig (275 kPa gauge). The term *fan* is applied to centrifugal and axial-flow blowers that increase the density of the gas passing through them by a maximum value of 5 percent. This corresponds to a discharge pressure of approximately 1 psig (6.9 kPa gauge) when gas enters the fan at atmospheric pressure. The total differential pressure developed by a blower or fan is often expressed in inches of water gauge (mm wg), which can be related to total pressure or head expressed using other units of measurement as follows:

$$h_w = C_{14}\frac{p_{td}}{\gamma_w} = C_{15}\frac{\gamma_g}{\gamma_w}H_g \tag{12.27}$$

where

h_w = total (differential) pressure, in wg (mm wg)
C_{14} = 1,728 when using the USCS units shown (1,000 for the metric units)
p_{td} = total differential pressure, psi (kPa)
γ_w = specific weight of freshwater at 68°F (20°C), 62.32 lbf/ft^3 (9.789 kN/m^3)
C_{15} = 12 when using the USCS units shown (1,000 for the metric units)
γ_g = specific weight of gas entering the blower, lbf/ft^3 (kN/m^3)
H_g = total head developed by blower, ft of gas (m of gas)

The power required to drive a blower can be determined as shown below:

$$P_B = \frac{h_w\bar{q}}{C_{16}\,\eta_B} = \frac{p_{td}\bar{q}}{C_{12}\,\eta_B} = \frac{\gamma_g H_g\bar{q}}{C_{17}\,\eta_B} \tag{12.28}$$

where

P_B = power required to drive blower, hp (kW)
$\bar{q}$ = mean volumetric flow rate or capacity, ft^3/min (m^3/hr)
C_{16} = 6,362 when using the USCS units shown (3.678E+4 for the metric units)
η_B = blower total efficiency, %/100
C_{12} = 229.2 when using the USCS units shown (3,600 for the metric units)
C_{17} = 33,000 when using the USCS units shown (3,600 for the metric units)

Because the differential pressure developed by a blower is typically low, changes in the density of the gas passing through it are often ignored, and the volumetric flow rate is assumed to remain constant from the blower's inlet to its outlet. In addition, although the total head developed by a blower is composed of both static and dynamic components, the increase in the velocity head of gas that has passed through a blower is often negligible when compared to the increase in static pressure head. Consequently, blower performance is frequently shown by plotting static pressure versus capacity. Blower efficiency that is based on static pressure is referred to as static efficiency. It can be used to calculate a blower's power requirement as follows:

$$P_B = \frac{h_{w,st}\bar{q}}{C_{16}\,\eta_{B,st}} \qquad (12.29)$$

where

$h_{w,st}$ = static pressure, in. wg (mm wg)
$\eta_{B,st}$ = blower static efficiency, %/100

EXAMPLE 12-13: A blower delivers 10,000 cfm of air while developing a static pressure of 12 in. wg. The blower has a static efficiency of 70 percent. Determine the power required to drive the blower.

Solution: Using equation 12.29

$$P_B = \frac{12 \text{ in. wg}(10{,}000 \text{ cfm})}{6{,}362(0.70)} = 27 \text{ hp}$$

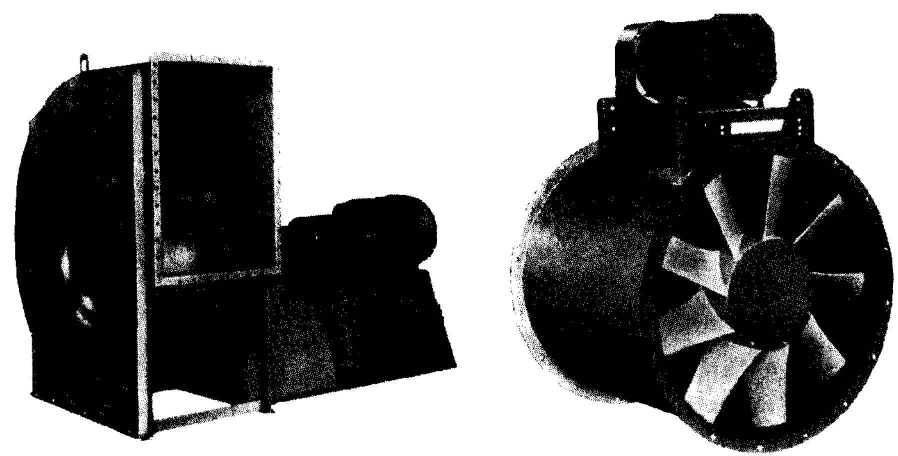
Dynamic Blowers

CENTRIFUGAL BLOWERS

Gas entering a centrifugal blower is directed into the eye of a rotating radial-flow open, semiopen, or closed impeller that increases both its velocity and static pressure (fig. 12-51a). A single-inlet centrifugal blower has a single-suction impeller that is either mounted between bearings or overhung on the end of the unit's shaft. A double-inlet centrifugal blower has an inlet opening at each end of the casing and is typically fitted with a double-suction impeller that is mounted between bearings. Gas leaving a centrifugal blower's impeller passes through a stationary scroll-shaped volute or diffuser in which a portion of gas's velocity head is converted into static pressure head. The gas is then discharged from the blower through a tangential outlet port in the casing. Blower impellers can have forward-curved blades (i.e., curved in the direction of rotation), straight-radial blades, or backward-curved blades (i.e., curved opposite to the direction of rotation). Blades having a double curvature are also used. With this latter configuration, each blade generally has a backward curvature that is either increased or reduced at the tip.

AXIAL-FLOW BLOWERS

Gas passing through an axial-flow blower travels through the blower's casing in an in-line or axial direction. The shaft-mounted rotor in a vane-axial blower consists of a hub with airfoil-shaped blades extending from its periphery (fig. 12-51b). To increase rigidity, the tips of the rotor blades in some vane-axial blowers are joined by an outer ring. Gas enters one end of the stationary cylindrically shaped housing that surrounds the rotor, passes through the rotor, and is discharged from the opposite end of the housing. Stationary guide vanes are provided at the outlet of a vane-axial blower's housing to straighten the discharge gas flow. A tube-axial blower is similar



(a) Centrifugal blower (b) Vane-axial blower

Figure 12-51. Blowers. Used with permission of The New York Blower Company.

to a vane-axial blower; however, a tube-axial blower's housing is not fitted with stationary guide vanes. In a propeller blower, which is also similar to a vane-axial blower, a simpler rotor design is utilized consisting oftwo or more relatively long constant-thickness or airfoil-shaped blades mounted around the circumference of a relatively small hub. Axial-flow blowers can be furnished in both single-stage and multistage configurations.

DYNAMIC BLOWER PERFORMANCE CHARACTERISTICS

The theory of performance for dynamic blowers is similar to that for centrifugal pumps. In addition, if the change in the density of a gas as it passes through a blower is ignored, the laws of similitude given previously for centrifugal pumps (equations 12.6a, 12.6b, and 12.6c) can be applied to centrifugal and axial-flow blowers. Typical performance curves for centrifugal and axial-flow blowers are shown in figure 12-52. Although the head (or pressure) developed by a centrifugal blower often remains relatively constant with flow rate during operation below the best efficiency point (BEP), the head developed by some axial-flow blowers can rise sharply with reductions in flow rate.

Centrifugal blowers with impellers that have backward-curved blades typically operate at higher speeds and are generally more efficient than centrifugal blowers having impellers with radial or forward-curved blades. Additionally, when fitted with an impeller having forward-curved blades, a centrifugal blower's power requirement often increases with flow rate and reaches a maximum value at or near the point offree delivery (the flow rate corresponding to operation with a negligible differential pressure). The power required to drive a centrifugal blower fitted with an impeller having backward-curved blades, however, frequently increases with flow rate during operation below the BEP, but then reaches a maximum value and begins to drop as the capacity delivered increases beyond the BEP. Consequently, although a driver used with a centrifugal blower that has an impeller with forward-curved blades must generally be sized based on the free-delivery capacity, the driver of a blower fitted with an impeller having backward -curved blades can usually be sized for the power requirement near the BEP. Ordinarily, the power required by an axial-flow blower also drops during operation at capacities beyond the BEP, but it can rise as the flow rate is reduced to shutoff. As a result of this characteristic, the rating for an axial-flow blower's driver must often be based on the minimum flow rate that the blower will deliver.

During operation at low flow rates, stalling (a loss oflift) of gas at the inlet to a blower's rotor or stator can result in a drop in the head or pressure developed and in surging. In the case of an axial-flow blower, if surging begins as the blower's capacity is being reduced, the capacity necessary to suppress the surging can shift to a higher value than the flow rate at which the surging started. An axial-flow blower may, therefore, have two differ-

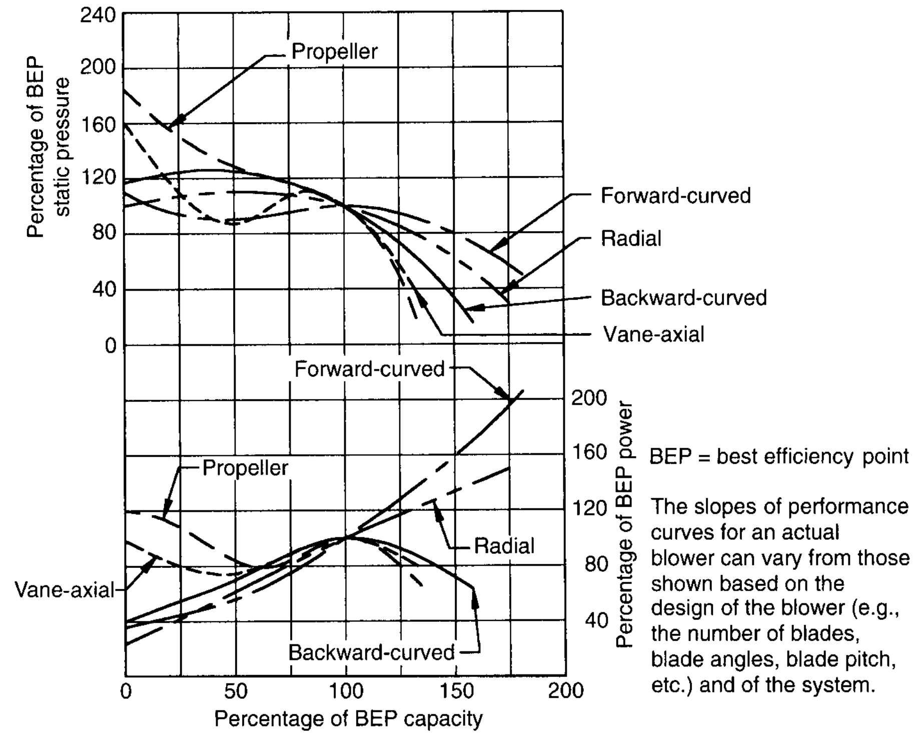


Figure 12-52. Representative constant-speed performance characteristics of centrifugal blowers with forward-curved, radial, and backward-curved impeller blades and of axial-flow propeller and vane-axial blowers.

ent head-capacity curves in the low-flow region, with the curve representing blower performance when capacity is being increased located slightly to the right of the curve representing performance when the flow rate through the blower is being reduced.

Positive-Displacement Blowers

Positive-displacement helical-lobe- and sliding-vane-type blowers are similar in design to the corresponding types of rotary compressors described previously. Although less common, in some applications, reciprocating blowers also have been used.

Straight-Lobe Blower

A straight-lobe rotary blower has two meshing rotors, which are referred to as impellers, that are mounted on parallel counterrotating shafts (fig. 12-53). Each impeller typically has two or three involute- or cycloidal-shaped lobes. The casing in which the shaft-mounted impellers turn is typically referred to as a cylinder. As each lobe rotates past the blower's inlet port, a constant volume of gas (usually air) is trapped in the cavity formed between the rotor and the inner wall of the cylinder. With the continued rotation of the rotor, the trapped gas is forced towards the cylinder's discharge port. An increase in the pressure of the gas results from the resistance created by the back pressure in the blower's discharge line. The close clearances between the meshing rotors and between the lobes and the cylinder's inner wall limits the amount of gas that slips back to the blower's inlet. Straight-lobe compressors are typically driven through reduction gears, multi- V-belt drives, or couplings by electric motors at speeds not exceeding 1,800 rpm. In addition, although it is not common, a lobe-type blower may be geared to and driven by a diesel engine when the blower is being used as a supercharger.

Straight-lobe blowers are often furnished with cast-iron or aluminum casings, ductile-iron or aluminum impellers, and steel shafts. Each shaft is generally supported at both ends by either ball, roller, or cylindrical journal bearings. Because of the shape of the lobes on the rotors, torque must be transmitted from the driven shaft to the idler shaft through timing gears. These gears, together with the pump's outboard bearings, are generally lubricated with oil that is either distributed by splashing or is pressurized and circulated by a shaft-driven gear pump. The inboard or driver-end bearings may also be lubricated with oil if the unit is fitted with a pressurized forced-feed lubrication system or, for less severe duty, the bearings may be grease lubricated. Lip, labyrinth-type, or mechanical seals are generally installed adjacent to bearings to prevent the lubricant from mixing with the gas passing through the blower.

Blower Installation

For the most part, comments made previously regarding air compressor installation also apply to the installation of blowers.

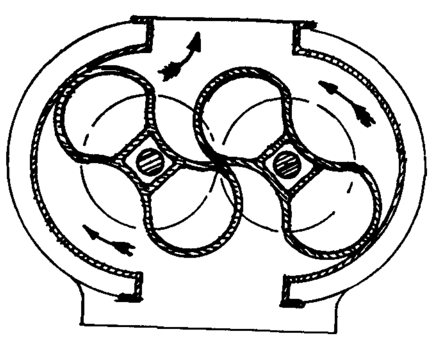


Figure 12-53. Straight-lobe blower. Courtesy Roots/Dresser.

OPERATION

Prior to starting a blower, the unit should be thoroughly inspected, and any abnormalities that are found should be corrected. When practicable, the blower's shaft should be turned by hand to ensure that it rotates freely. (In some cases, it may not be possible or advisable to turn the shaft by hand because of the size, configuration, or design of the blower.) When bearings are oil-lubricated, the LO level in the sump should be checked, and oil should be added if needed. When applicable, valves in gauge lines and bearing-lubrication supply and return lines should ordinarily be opened.

While the blower is operating, its casing, bearings, and seals should be checked frequently for any signs of overheating or leaks. In addition, when possible, its discharge pressure should be monitored. When oil is used for bearing lubrication, the level in the LO reservoir should be checked periodically and maintained at the proper value. In addition, when a forced-feed lubrication system is used, the LO pressure should be monitored, and, when sight glasses are provided, oil flow through the bearings should be observed. If an LO cooler is provided, the flow rate ofthe cooling-water medium passing through the cooler must often be regulated to maintain the proper oil outlet temperature. A blower should generally be stopped if there are any abnormalities, including excessive or unusual noise or vibration.

TROUBLESHOOTING

Reasons why a blower pump may fail to deliver the proper capacity or develop the proper discharge pressure can include operation at the wrong speed or with the wrong direction of shaft rotation, the use of an impeller that has the wrong outside diameter or is mounted on the pump's shaft in the reverse direction, external leakage through casing joints or ducts, excessive leakage through shaft seals, excessive internal leakage through enlarged wearing-ring or interstage-bushing clearances, foreign material in the blower or ducts, a clogged suction filter, mechanical defects (including a suction or discharge damper that has failed closed, and an impeller key, coupling, pump shaft or driver failure), a high system back pressure, and operation too close to the low-flow surge limit.

Among the possible causes of an excessively high power requirement by a blower are rubbing contact between rotating and stationary parts, which can result from thermal expansion, a bent shaft, a distorted casing, misalignment of the casing or rotor, worn bearings, or excessive shaft deflection; operation at too high a speed; the use of an impeller with too large an outside diameter; improperly lubricated or overloaded bearings; mechanical defects; and foreign matter in the impeller. In addition, when a blower's driver is not sized to deliver the maximum power required by the blower at any operating point along the entire range of performance, the driver of a centrifugal blower with radial or forward-curved vanes can be overloaded if the blower is delivering a capacity that exceeds the rated flow. Similarly, the driver of an axial-flow blower can sometimes be overloaded if the blower is delivering a capacity that is less than the rated flow.

Marine Blower Applications

See volume 2 of the *Modern Marine Engineer's Manual* for a description of blowers used for turbocharging diesel engines and blowers used in inert-gas systems.

FORCED-DRAFT BLOWERS

Forced-draft blowers are used to supply combustion air to fossil-fueled boilers. Centrifugal and axial-flow propeller blowers are typically used in this application. Although an inlet trunk is sometimes used to direct the flow of incoming air to the blower, many forced-draft blowers take suction directly from a vessel's machinery spaces. Exposed openings should be screened to prevent foreign objects from entering a blower. To prevent an idle blower from being driven in reverse by pressurized air in the discharge duct, nonreturn shutters are often installed at the inlet or outlet of each unit.

A typical centrifugal forced-draft blower is a horizontally mounted single-stage unit fitted with either a single- or double-suction impeller that is driven by a steam turbine or an electric motor. The shaft in a centrifugal forced-draft blower can be supported by ball and roller bearings that are either grease- or oil-lubricated. Alternatively, journal bearings that are lubricated with oil distributed by partially submerged shaft-mounted rings or through a pressurized forced-feed lubrication system may be used.

A propeller forced-draft blower generally has two or three stages that each consist of a rotating multibladed propeller followed by a row of stationary guide vanes. In addition, a set of stationary guide vanes is generally installed in the blower's inlet, and a diffuser generally follows the blower's final stage. Propeller forced-draft blowers are frequently driven by steam turbines. In some units, the turbine's rotor and the blower's propellers are mounted on a common shaft. Bearings that support this shaft are generally lubricated with oil that is pressurized and circulated by a shaft-driven pump. Propeller forced-draft blowers are supplied in both horizontal and vertical configurations. When mounted vertically, the turbine is typically located below the blower.

The discharge pressure that must be developed by a forced-draft blower is affected by the pressure drop in the discharge duct, combustion-air heaters (if used), and wind box; the number of burners in operation; the amount that the burner air-register doors are opened; the pressure drop on the gas side of the boiler's tube banks and economizer; and stack losses. The capacity of air delivered by a forced-draft blower is generally regulated by the vessel's combustion control system based on the boiler requirements. Typical methods used for forced-draft blower flow control, listed in order based

on relative efficiency, include varying operating speed (through the use of a variable- or multiple-speed driver or a constant-speed driver with a fluid coupling), adjusting the angle of inlet (pre rotation) guide vanes, throttling an inlet damper, and throttling a discharge damper. In addition, the capacity delivered by some propeller blowers can be adjusted by changing the pitch or stagger angle of the rotor's blades. Because blower performance can be unstable at low flow rates, the capacity delivered by a forced-draft blower should never be reduced below the surge limit. In addition, when two or more similar forced-draft blowers are operated in parallel, both blowers should be operated at approximately the same speed so that each delivers approximately the same capacity.

VENTILATION FANS

Centrifugal and axial-flow fans are used to deliver air for ventilation and air-conditioning to spaces throughout a vessel. The centrifugal fans used are typically of the single-inlet design and are driven through couplings or V-belts by electric motors. Vane-axial, tube-axial, and propeller axial-flow fans are also used in many ventilation applications. Because of an axial-flow fan's in-line configuration, it can be installed directly in the duct work. An axial-flow fan's rotor is sometimes driven through a V-belt by an electric motor mounted outside the housing. Alternatively, the blower's rotor may be mounted directly on the extended shaft of the drive motor. When furnished in this close-coupled configuration, the motor is located within the blower housing and is, therefore, in the air stream. Vane-axial fans are generally selected to operate to the right of any dip in the head-capacity curve. However, motors are typically sized with a margin to prevent an overload if the fan is operated too close to shutoff. Capacity control of a ventilation fan can be achieved by opening and closing flow dampers, regulating blower speed, or adjusting the angle of stationary guide vanes in the fan's housing. In addition, the capacity delivered by some fans can be adjusted by changing the pitch or stagger angle of the blades in the rotor. Portable vane-axial blowers that may be driven by internal combustion engines, by electric or hydraulic motors, or by air, steam, or water turbines are frequently used to ventilate tanks and other confined spaces. Flexible removable ducts are often used with a portable blower. (See volume 2 of the *Modern Marine Engineer's Manual* for additional information regarding blowers used in heating, ventilation, and air-conditioning [HVAC] systems.)

SEWAGE TREATMENT BLOWERS

In activated-sludge marine sanitation devices, straight-lobe blowers are frequently used to supply high capacities of low-pressure air that is required to aerate and agitate the sludge collected in the vessel's sewage-treatment tanks. This air promotes the growth of aerobic bacteria that decompose and digest the organic matter being treated. Typical discharge pressures for these blowers are below 10 psig (69 kPa gauge).

EJECTORS

Ejectors, which are also referred to as jet pumps, can be used to remove gases or liquids from various locations on a vessel. An ejector has no moving parts or mechanical driver but, instead, receives energy from a pressurized motive or operating fluid that creates a pumping action as it passes through the ejector. A typical ejector consists of a body with a suction connection, a converging-diverging diffuser with a discharge connection,

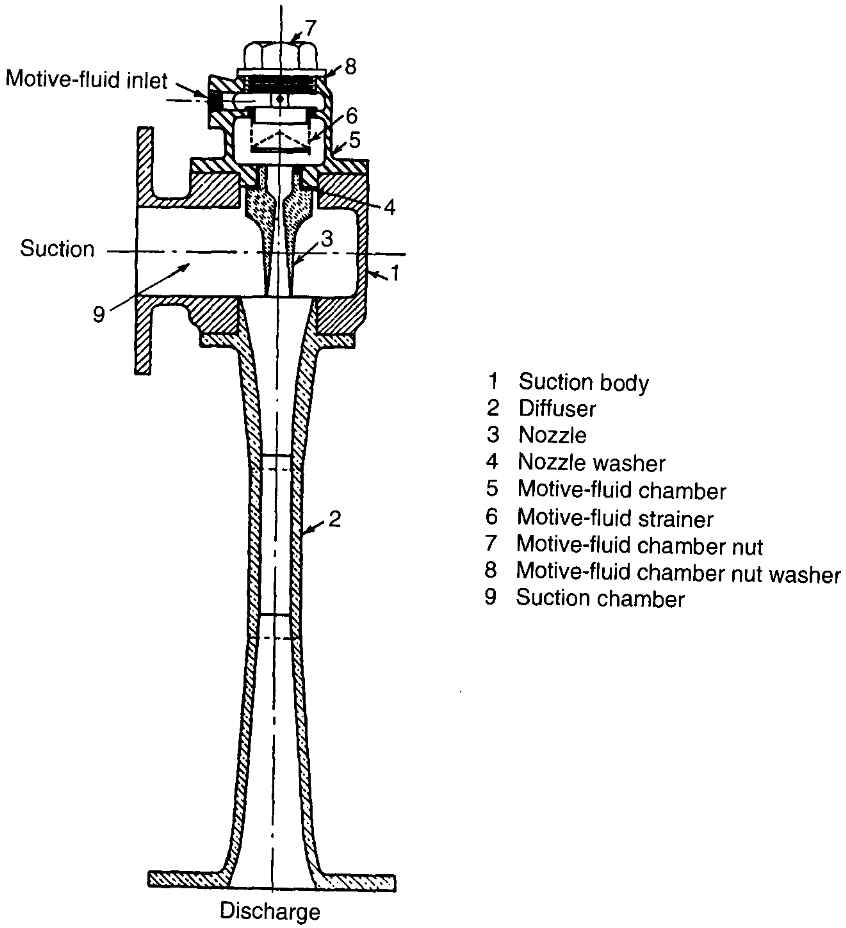


Figure 12-54. Ejector. Courtesy Foster Wheeler Corporation.

and a nozzle (fig. 12-54). As the pressurized motive fluid that is supplied to the ejector passes through the nozzle, some of its potential energy is converted to kinetic energy, and it leaves the nozzle at a reduced pressure but with an increased velocity. Fluid present in the suction chamber becomes entrained within the high-velocity motive-fluid jet exiting the nozzle and is carried into the converging portion of the diffuser. The evacuation of the suction chamber creates a partial vacuum that draws additional fluid present at the suction connection, referred to as the suction fluid, through the suction connection and into the ejector. The suction fluid and the motive fluid mix as they pass through the parallel section of the diffuser. In the diffuser's diverging section that follows, the velocity of the mixture drops. The increase in pressure resulting from a conversion of kinetic energy to potential energy enables the mixture to be discharged from the ejector. Although the discharge pressure from an ejector is greater than the pressure of the suction fluid, the discharge pressure is less than the pressure of the motive fluid supplied to the nozzle.

A loss of motive-fluid pressure can permit reverse flow to occur through an ejector's body and result in a loss of vacuum in the suction chamber. Consequently, the motive fluid should be supplied to an ejector at a steady flow rate and at a pressure that does not drop below a specified minimum value. Although an ejector can sometimes operate satisfactorily with a motive-fluid supply pressure that is above the specified value, this condition will generally result in an increase in the motive-fluid consumption rate.

Ejectors can operate individually as single-stage components, or they can be furnished in a multistage configuration consisting of several ejectors arranged in series (i.e., the discharge from one ejector is directed into the suction of the next ejector). In addition, an ejector can be operated as part of a multiple-element assembly composed of two or more single-stage or multistage ejectors arranged for parallel operation.

Steam-Jet Ejector

In a steam-jet ejector, steam is used as the motive fluid. The steam pressure required by an ejector that must maintain a constant vacuum increases with the back pressure at the ejector's discharge. Also, during operation with motive steam at a constant pressure, the vacuum in an ejector's suction chamber will drop (i.e., the absolute suction pressure will rise) if the back pressure at the ejector's discharge increases. Typical steam supply pressures range from 135 to 150 psig (930 to 1,035 kPa gauge) for small-capacity ejectors and can be as high as 300 psig (2,069 kPa gauge) for larger units. The steam used may be superheated to reduce nozzle erosion and the detrimental impact that moisture can have on ejector performance. Most steam-jet ejectors are fitted with a converging-diverging nozzle, which results in supersonic nozzle-exit velocities whenever the suction pressure is less than approximately one-half of the steam supply pressure.

Nozzle-exit velocities in some ejectors can be as high as 3,000 to 5,000 *ftls* (915 to 1,525 m/s).

A single-stage steam-jet ejector can typically produce vacuums up to 26.5 to 27 in. Hg (675 to 685 mm Hg). To produce vacuums that are up to 29 in. Hg (735 mm Hg), a two-stage ejector is frequently required. The use of three-stage ejectors is limited because of their increased size, weight, and complexity. A reduction in vacuum will occur if an ejector has a clogged or worn nozzle or is supplied with steam at too Iowa pressure.

The steam-and-suction-fluid mixture discharged from a steam-jet ejector is often directed to a heat exchanger in which the steam, together with condensable gases in the suction fluid, condenses. This permits latent heat in the steam-and-suction-fluid mixture to be recovered and used to heat another fluid. In addition, by draining offthe condensate, the mass flow rate enteringthe next stage ofa multistage ejector is reduced. The heat exchanger installed between two ejector stages is referred to as an intercondenser, while the heat exchanger that follows a single-stage ejector or the final stage of a multistage ejector is called an aftercondenser (fig. 12-55).

Eductors

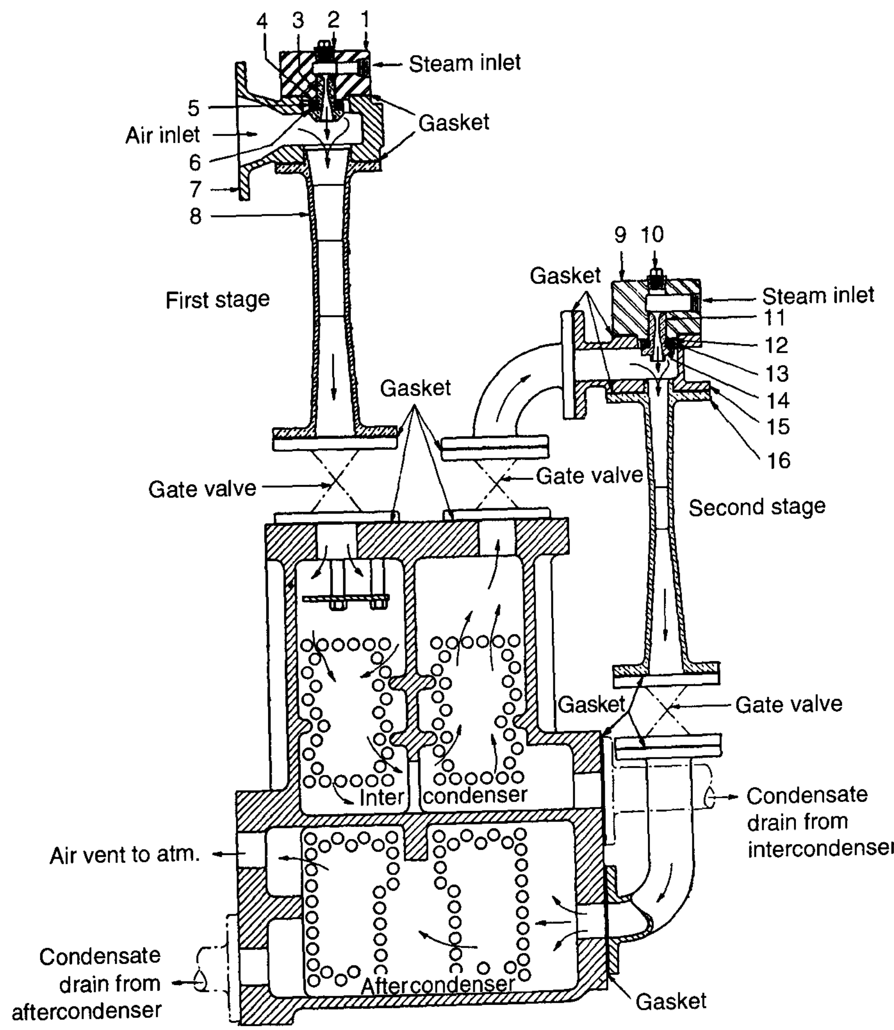
When a liquid is used as the motive fluid, an ejector is often referred to as an eductor. The suction fluid pumped by an eductor is also frequently a liquid. Intercondensers and aftercondensers are, therefore, not used. In addition, most eductors are fitted with converging nozzles. Some eductors have a single nozzle that extends into the center ofthe suction chamber and is surrounded by the suction fluid. Alternatively, a peripheral-jet eductor has a ring of nozzles located around the periphery of the suction-fluid inlet port. Because this ring creates less of an obstruction to the suction flow than a centered single nozzle, peripheral-jet eductors are often used in applications where solid particles may be present in the suction fluid.

Eductors can often operate with suction lifts as high as 25 ft (7.6 m) and values ofNPSH available as low as 2 ft (0.6 m). Typical values for the ratio of discharge pressure at the eductor's outlet divided by the motive-liquid supply pressure can be as high as 0.4. In addition, the ratio of the suction capacity being pumped divided by the motive-liquid inlet capacity can exceed 1.0. This latter ratio generally declines, however, as the suction lift and the ratio of the discharge pressure divided by motive-liquid supply pressure increase.

Marine Ejector Applications

CONDENSER DEAERATION

Steam-jet ejectors are used on many steam-powered vessels to remove air and other noncondensable gases from the main condenser. A typical main



1 Steam chest (first stage)	9 Steam chest (second stage)
2 Pipe plug (first stage)	10 Pipe plug (second stage)
3 Nozzle (first stage)	11 Nozzle (second stage)
4 Nozzle gasket (first stage)	12 Nozzle gasket (second stage)
5 Nozzle ring (first stage)	13 Nozzle ring (second stage)
6 Nozzle gasket (first stage)	14 Nozzle gasket (second stage)
7 Suction chamber (first stage)	15 Suction chamber (second stage)
8 Diffuser (first stage)	16 Diffuser (second stage)

Figure 12-55. Two-stage ejector with inter- and aftercondensers. Courtesy Westinghouse Electric Company.

air-ejector assembly has a pair of two-stage air-ejector elements that are each sized to handle 100 percent of the condenser's normal gas-removal requirements, including air that may leak into the condenser; noncondensable gases, such as carbon dioxide and ammonia, that may be present in steam exhausted into the condenser; and air that may be present in makeup feedwater added to the condenser. With this arrangement, either one of the two-stage ejector elements can be used alone during normal plant operation. However, if necessary, both two-stage ejector elements can be operated in parallel to handle unusually high air-removal requirements that may arise. The installation of a twin-element air ejector also enables maintenance to be performed on one ejector element without interfering with normal plant operation. In addition, in the event of a problem, it is often possible to operate the first stage of one ejector element in series with the second stage of the other ejector element.

During normal operation, saturated noncondensable gases removed from a main condenser are initially drawn into the first-stage suction chamber of one of the two-stage air-ejector elements. These gases become entrained in the high-velocity jet of motive steam discharged from the ejector's nozzle, pass through the ejector's diffuser, and are generally discharged into the shell of a surface intercondenser. Condensate discharged from the main condensate pump passes through the intercondenser's tubes and absorbs heat from the fluid within the shell. Water formed from the steam and water vapor condensing in the shell of the intercondenser drains to the main condenser. The intercondenser's drain line has a water-sealed V-shaped loop that prevents air and other gases within the intercondenser from being drawn back into the main condenser. Typical intercondenser-shell pressure and loop-seal height values are 4 to 6.5 in. (102 to 165 mm) Hg abs and 7 to 8 ft (2.1 to 2.4 m), respectively. A small branch line from the condensate system is frequently connected to the top ofthe intercondenser's drain line to permit water to be added to the drain line's loop seal when needed (e.g., during start-up, or if the water seal in the loop is lost as a result of a temporary increase in the absolute pressure within the intercondenser's shell). The required shutoff valve in the condensate branch line should be closed during normal operation. If a condensate branch line is not provided, the water seal in the intercondenser's drain loop may be restored by operating the air ejector for a short period of time with a valve in the drain line (located downstream from the loop seal) closed. However, to avoid flooding the intercondenser's shell, the intercondenser drain valve should be reopened as soon as the loop seal is reestablished.

The gases remaining in the intercondenser's shell are drawn into the suction chamber of the air-ejector element's second stage, entrained in a second high-velocity jet of steam, and discharged through the second-stage ejector's diffuser into the shell of a surface aftercondenser. Condensate leaving the tubes of the intercondenser passes through the aftercondenser's

tubes and absorbs additional heat. Water formed from the condensing steam and water vapor in the shell ofthe aftercondenser typically is under a slight positive pressure (e.g., 30 to 32 in. [762 to 813 mm] Hg abs) and often drains by gravity to a freshwater-drain-collecting tank. The gases remaining in the shell ofthe aftercondenser can be vented to the atmosphere or to the vessel's gland-exhaust condenser in which steam that leaks past propulsion-turbine glands is condensed. (On some vessels, the gland-exhaust condenser is an integral part of the main air-ejector assembly.)

Steam should never be supplied to a main air ejector's nozzles until the flow of condensate has first been established through the ejector's intercondenser and aftercondenser. If the flow of condensate through the ejector's condensers is too low to properly condense the steam and water vapor in these heat exchangers, which can occur during start-up or while operating at reduced power, the temperature within the air-ejector assembly will rise and the ability of the ejector to maintain the required vacuum in the main condenser will be reduced. To enable condensate flow through the air-ejector condensers to be increased when necessary, a recirculation line is normally provided through which a portion of the condensate that leaves the vessel's gland-exhaust condenser (which is downstream from the air ejector's aftercondenser) can be returned to the main condenser. Typically, a manually operated valve is provided that can be used to control the flow rate through the recirculation line. In addition, the recirculation line may also be fitted with a separate thermostatically controlled recirculation valve that opens automatically whenever the temperature of the condensate leaving the air ejector's intercondenser or, in some cases, the aftercondenser reaches a preset value. The opening of either a manual or an automatic recirculation valve results in greater condensate flow through the air ejector's condensers and, in most cases, can permit the condensation of steam and water vapor in these heat exchangers to continue at an adequate rate.

For design purposes, the mixture of air (and other noncondensable gases) and water vapor removed from a main condenser is usually assumed to enter an air ejector's first stage at a pressure of 1 in. (25.4 mm) Hg abs and a temperature of 71.5°F (22°C), which represents 7.5°F (4°C) of subcooling. With these operating conditions, the air and vapor mixture contains about 70 percent water vapor and 30 percent air, and a typical two-stage air ejector requires approximately 4.5 Ibm (2 kg) of motive steam per Ibm (kg) of air and vapor removed from the main condenser. Operation of an ejector with motive steam at a pressure below the rated value will often result in a reduction in the vacuum within the main condenser. Consequently, ejectors are sometimes operated with steam pressures that are up to 10 psi (70 kPa) above rated nameplate values to compensate for slight pressure fluctuations in the steam supply line. Small safety valves are frequently provided to protect the ejector assembly from overpressurization.

The suction line from the main condenser to the main air-ejector assembly should be as short and direct as possible. Low points in the line where water may accumulate should be avoided. (If low points do exist, they should be drained with a loop seal back to the main condenser.) A shutoff valve should be installed at the inlet to each first-stage ejector. This is necessary to permit one ejector element in a twin-element unit to be used while the second element is secured. Shutoff (usually gate) valves in the discharge of each ejector element's first stage and the suction of each element's second stage, which are needed to isolate an off-line ejector element from the intercondenser, and a valve in the discharge from each second-stage ejector are norma1ly provided as part of the air-ejector assembly.

When raising main-condenser vacuum using a twin-element two-stage air ejector, typically, the main circulating pump should be started (usually on low speed) to pump water through the main condenser's tubes, and the main condensate pump should be started (usually with the recirculation valve open) to circulate condensate from the main-condenser hotwell through the air-ejector intercondenser and aftercondenser. In addition, in most cases, the propulsion-turbine lubricating-oil-service system, jacking gear, and gland-sealing systems should all be in operation. Isolation valves at the suction and discharge of all air-ejector stages should then be opened. Any valves in the intercondenser_ or after-condenser-drain lines or in the aftercondenser's vent should also be opened. The steam-supply valves to both second-stage ejectors can then be opened slowly. (Using both elements in a twin-element-ejector assembly when raising vacuum reduces start-up time.) After the condenser vacuum has increased to approximately 20 in. (508 mm) Hg, steam-supply valves for both first-stage ejectors can be slowly opened. (Operation ofthe first-stage ejectors is sometimes delayed until after the propulsion tUrbines have been warmed up.) After the normal vacuum (e.g., 28.5 in. or 724 mm Hg) has been achieved, the suction valve to the first stage of one of the ejector elements can usually be closed. The steam-supply valve to this first-stage ejector's nozzle should then be closed, followed by the ejector's discharge valve. The suction, steam-supply, and discharge valves for the second stage of the same ejector element should then be closed in the order listed. With normal operating conditions, the two-stage-ejector element that remains on-line should ordinarily be able to maintain the required main-condenser vacuum.

To switch ejector elements while underway, the discharge, steam-supply, and suction valves for the off-line ejector element's second stage should be opened in the order listed. Then, the discharge, steam-supply, and suction valves for the off-line element's first stage should be opened, also in the order listed. (When both ejector elements are operated simultaneously, the main-condenser vacuum may drop slightly due to the increased heating load in the ejector intercondenser and aftercondenser.)

After proper operation ofthe ejector element that has just been started has been verified, the opposite element can be secured by closing its valves in the reverse order to that followed for starting up the off-line element.

The procedures given above are representative only. The engineer's operating manuals and manufacturer's technical manuals should always be consulted for the start-up procedures applicable to a specific vessel.

Causes of improper ejector performance include operation with motive steam that is wet, has excessiye superheat, or is at too Iowa pressure or flow rate; clogged or eroded ejector nozzles; clogged or leaking intercondenser or aftercondenser tubes or drain lines; loss of a water seal in the intercondenser drain line; insufficient condensate flow through the air-ejector condensers; excessive back pressure at the aftercondenser's vent; or excessive air leakage into the main condenser or the ejector suction line.

In addition to a steam-powered vessel's main air ejectors, twin- or single-element two-stage air ejectors fitted with intercondensers and aftercondensers may be used to deaerate auxiliary condensers that receive steam from nonpropulsion-related components, such as turbogenerators. The air-ejector suction lines from the main and auxiliary condensers are sometimes cross-connected so that either the main or the auxiliary air ejectors can be used to deaerate the auxiliary condensers.

DEWATERING

Eductors are often used for bilge drainage and compartment dewatering. The motivating seawater supplied to the nozzles in dewatering eductors is typically discharged from a fire, bilge, ballast or seawater-service pump.

DISTILLING PLANT EJECTORS

Two-stage steam-jet ejectors are often used to remove air and other non-condensable gases from flash distilling plants. In a typical installation, the first-stage ejector takes suction from the shell of the distiller (from the second-stage distillate condenser in a two-stage flash distilling plant) and discharges directly into the suction chamber of the second-stage ejector. Discharge from the second-stage ejector is usually directed to the shell of an aftercondenser in which the motive steam supplied to both ejector stages, together with water vapor mixed with the air removed from the distiller, condenses. Heat released by the condensing steam and vapor is absorbed by seawater flowing through the aftercondenser's tubes. After leaving the aftercondenser, this seawater normally passes through an additional heater and is then admitted to the distiller shell as feedwater. The aftercondenser, therefore, serves as an evaporator-feedwater heater. The condensate formed in the aftercondenser can be contaminated with salt if seawater is entrained in the air and vapor removed from the distiller; consequently, a salinity cell is typically installed in the drain line, and this condensate can generally be directed either to a freshwater-drain-collecting tank or to the bilge based on its salinity. Noncondensable gases remaining in the aftercondenser's shell are vented to the atmosphere.

Seawater-motivated eductors are frequently used both to deaerate and to remove brine from plate-type distillers. Seawater is usually supplied to the nozzles in these eductors by the same pump that supplies the distiller with feedwater. In some units, the eductor that deaerates the distiller takes suction from the upper portion of the shell, while a separate brine eductor takes suction from the bottom of the distiller. Alternatively, one eductor may be used for both deaeration and brine removal. With either arrangement, the discharge from the eductors is typically directed Overboard. Also, each edactor's suction line is ordinarily fitted with a check valve to prevent reverse flow into the distiller shell.

STRIPPING

Eductors are sometimes used to strip liquid-cargo tanks (fig. 12-56). During stripping, a portion of the cargo being discharged by the main cargo

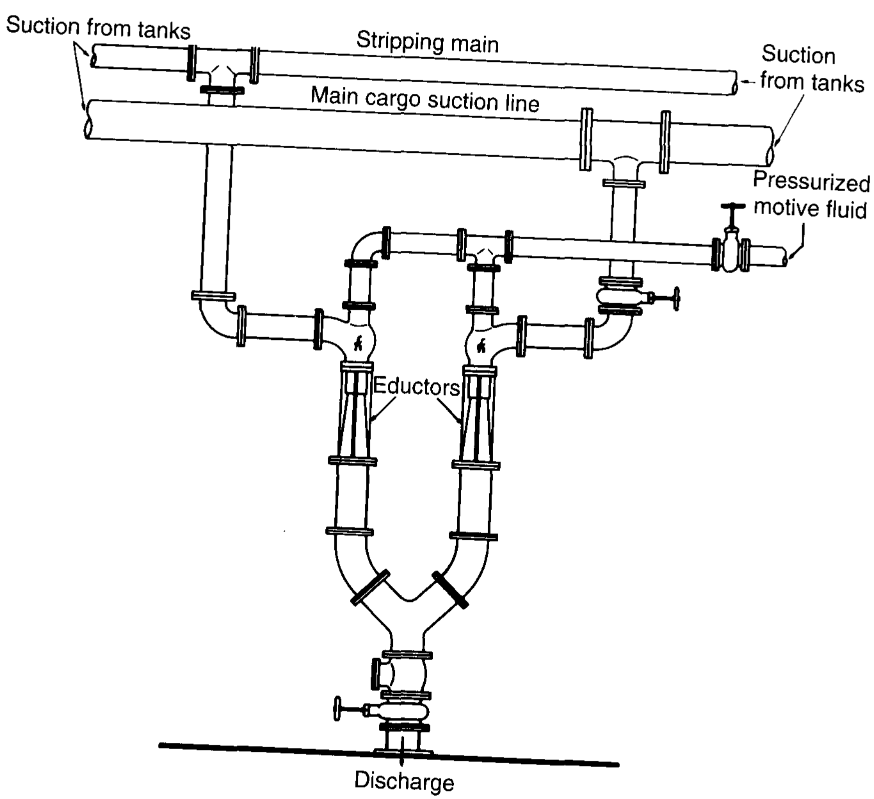


Figure 12-56. Stripping eductor. Courtesy Schutte-Koerting.

pumps is diverted through a bypass line and is used as the motive fluid for one or more eductors that take suction through the main or the stripping suction lines and remove cargo remaining in tanks that the main pumps have already partially emptied. This arrangement is only suitable when the pressurized cargo used as the motive fluid is the same or is compatible with the cargo being stripped. Liquid viscosities in excess of 100 cp can adversely affect eductor performance; consequently, the viscosity of cargoes that a vessel will carry must be considered when stripping eductors are sized. Stripping eductors may also be motivated by seawater and used to remove slops from a vessel's cargo tanks during tank washing. In addition, on some vessels, seawater is supplied as the motive fluid to eductors that are used to strip ballast tanks. To reduce the potential for clogging, stripping eductors often have large flow passages.

REVIEW

1. What procedure should be followed to replace the packing in a pump's stuffing box?
2. What procedure should be used to start up a radial-flow centrifugal pump?
3. How would the procedure developed in answer to question 2 be changed to start a rotary pump?
4. A pump with a 40 mm suction and a 25 mm discharge delivers a capacity of 4,400 m^3lhr of freshwater at a temperature of 20°C. Gauges mounted at the pump's suction and discharge connections measure pressures of 35 kPa gauge and 700 kPa gauge, respectively. Assume that both gauges are at the same elevation. The pump efficiency at the operating point is 65 percent. Calculate the total head developed by the pump and the power required to drive it. (Hint: Because pressure measurements are being taken directly at the pump's suction and discharge connections, friction losses can be ignored.)
5. Calculate the NPSHA to the pump described in question 4.
6. A single-stage centrifugal pump with a single-suction impeller delivers a capacity of 10,000 U.S. gpm and develops a total head of 400 ft while operating at 3,550 rpm and its best efficiency point. What is the pump's specific speed and what is the flow orientation at the impeller discharge?
7. What is the purpose of a balancing drum and how does it function?
8. What is a close-coupled pump?
9. What can happen if the NPSH available to a pump is less than the pump's NPSH requirement?

10. How do capacity delivered, total head developed, and power required by a centrifugal pump change with operating speed and impeller outside diameter?
11. What is slip in a rotary pump and how does it affect performance?
12. What is lost motion in a direct-acting reciprocating pump and how can it be adjusted?
13. What are the primary causes of axial thrust in a centrifugal pump? How can this thrust be reduced?
14. What should be checked during the inspection of an operating pump or compressor?
15. What is the purpose of an unloader in a reciprocating compressor?
16. What is the purpose of a mechanical seal?
17. What methods are suitable to adjust the flow rate delivered by a centrifugal pump, a rotary pump, and a reciprocating pump?
18. What factors should be considered when determining the minimum acceptable capacity for a centrifugal pump?
19. A 6 in. x 6 in. x 4 in. duplex steam pump with double-acting pistons operates with 60 strokes per min. The pump's volumetric efficiency is 85 percent. What capacity is delivered by the pump?
20. What methods can be used to control the flow rate delivered by a centrifugal blower?
21. Describe the face-and-rim method of measuring coupling misalignment.
22. Explain how a vacuum is developed by a steam-jet ejector.
23. A two-stage air compressor with an intercooler and an efficiency of 75 percent (based on isentropic adiabatic compression) has a rated inlet capacity of 100 cfm. Air enters the compressor at atmospheric pressure and is discharged at a pressure of 100 psig. Determine the power required to drive the compressor.
24. Explain how a vacuum is developed by a liquid-ring vacuum pump.
25. What is the function of the valves installed in a reciprocating compressor and how are these valves typically actuated?
26. Why is oil injected into some twin-screw compressors?
27. How does the impeller-blade curvature affect the performance of a centrifugal blower?
28. How is a pumped liquid's pressure increased in a regenerative turbine pump?
29. Describe a procedure for relubricating a grease-lubricated ball bearing in a centrifugal pump.
30. Why must the shaft of a VTP that is not fitted with an independent thrust bearing be raised after the shaft is coupled to the pump's vertical driver?

31. A blower delivers 300 m^3/min of air while developing a static pressure of 300 mm wg. The blower has a static efficiency of 75 percent. What is the power required to drive the blower?
32. What types of compressors can be used to supply oil-free air?

REFERENCES

American Society for Testing and Materials (ASTM). 1993. *Standard Specification for Centrifugal Pump, Shipboard Use.* ASTM F 998-86 (Reapproved 1993). West Conshohocken, Pa.: ASTM.

—. 1994. *Standard Specification for Rotary Positive Displacement Pumps, Commercial Ships Use.* ASTM F 1510-94. West Conshohocken, Pa.: ASTM.

Brown, R. N. 1986. *Compressors, Selection and Sizing.* Houston: Gulf Pub. Co.

Budris, A. R. August 1989. "Sorting Out Flow Recirculation Problems." *Machine Design.*

Compressed Air and Gas Institute. 1989. *Compressed Air and Gas Handbook,* 5th ed. Cleveland, Ohio: Compressed Air and Gas Institute.

Dufour, J., and W. Nelson. 1993. *Centrifugal Pump Sourcebook.* New York: McGraw-Hill.

Feck, A. W., and J. O. Sommerhalder. July 1967. "Cargo Pumping in Modern Tankers and Bulk Carriers." *Marine Technology* 4, no. 3.

Fraser, W. H. November 1981. "Flow Recirculation in Centrifugal Pumps." Paper presented at the winter annual meeting of the American Society of Mechanical Engineers, Washington, D.C.

Heat Exchanger Institute. 1970. *Standards for Steam Surface Condensers,* 6th ed. New York: Heat Exchanger Institute.

Hydraulic Institute. 1994. *Hydraulic Institute Pump Standards.* Parsippany, N.J.: Hydraulic Institute.

Jackson, C. April 1985. "Reverse Indicator Cold Alignment with Graphical Plotting." Proceedings of the Second International Pump Symposium, Texas A&M University.

Karassik, I. J. 1989. *Centrifugal Pump Clinic.* New York: Marcel Dekker.

Karassik, I. J. 1973. *Engineer's Guide to Centrifugal Pumps.* Milan (Italy): Ulrico Hoepli.

Karassik, I. J., and T. W. Edwards. June 1955. "How to Calculate Temperature Rise in Centrifugal Pumps." *Industry & Power.*

Karassik, I. J., et al., eds. 1986. *Pump Handbook.* New York: McGraw-Hill.

Krinsky, J. L., et al. May 1985. "Navy Air Compressors-Past, Present and Future." American Society of Naval Engineers. *ASNE Journal* 97, no. 4.

Marscher, W. D. October 1989. "Analysis and Test of Multistage Pump Wet Critical Speeds." Paper presented at the thirty-fifth STLE/ASME Tribology Conference, Fort Lauderdale, Fla.

Paashaus, R. F. December 1964. "An Analysis of Cavitation Damage in Commercial Marine Condensate Pumps." Paper presented at the Society of Naval Architects and Marine Engineers/American Society of Mechanical Engineers meeting, New York City.
Schiavello, B. November 1986. "Visual Study of Cavitation-An Engineering Tool to Pump Reliability." Paper presented at the International Conference ofImproved Coal-Fired Power Plant, PaloAIto, Calif.: Electric Power Research Institute.
Sembler, W. J. January and April 1988. "The Design and Operation of Pumps Furnished for Marine Cargo Service." *Marine Technology 25,* nos. 1 and 2.
Shepard, D. G. 1957. *Principles of Turbo machinery.* New York: Macmillan.
Stepanoff, A. J. 1957. *Centrifugal and Axial Flow Pumps,* 2nd ed. New York: John Wiley & Sons.
Wislicenus, G. F. 1947. *Fluid Mechanics of Turbomachinery.* New York: McGraw-Hill.
Yedidiah, S. 1980. *Centrifugal Pump Problems, Causes and Cures.* Tulsa, Okla.: The Petroleum Publishing Co.

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Imo Pump-Warren, Warren, Massachusetts
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CHAPTER 13

Management and Safety of Marine Engineering Operations

BORIS s. BUTMAN AND ROGER BUTTURINI

GENERAL PRINCIPLES

The principal goal of proper management of shipboard marine engineering operations is to have the safest, most reliable, and most efficient ship. When applied to a ship, the term *marine engineering operations* denotes the activities related, on one side, to running or operating the ship's machinery and equipment, and, on the other side, to keeping the entire ship-including hull structures, machinery, and equipment-in normal operational condition by conducting the required maintenance and repairs. While the thrust of other chapters of this book is on design and operation of onboard equipment, this chapter deals with the second component of shipboard marine engineering operations, i.e., maintaining a reliable and safe ship.

Management of shipboard marine engineering operations, or simply of ship engineering operations, is a two-tier organization. Figure 13-1 presents an example of the organizational structure of a shipping company. The upper tier belongs to the shore management, which is headed by the chief executive officer (CEO). The following functional subdivisions of the company are involved with the management of engineering operations.

1. Administrating and personnel: takes charge of the day-to-day activities of the headquarters, including those related to ship engineering operations.
2. Operations and marketing: takes care of finding cargoes, participates in planning and arranging voyages, leasing and chartering, port operations, and also plans shipyard repairs based on the schedule and areas of operation of the ship.

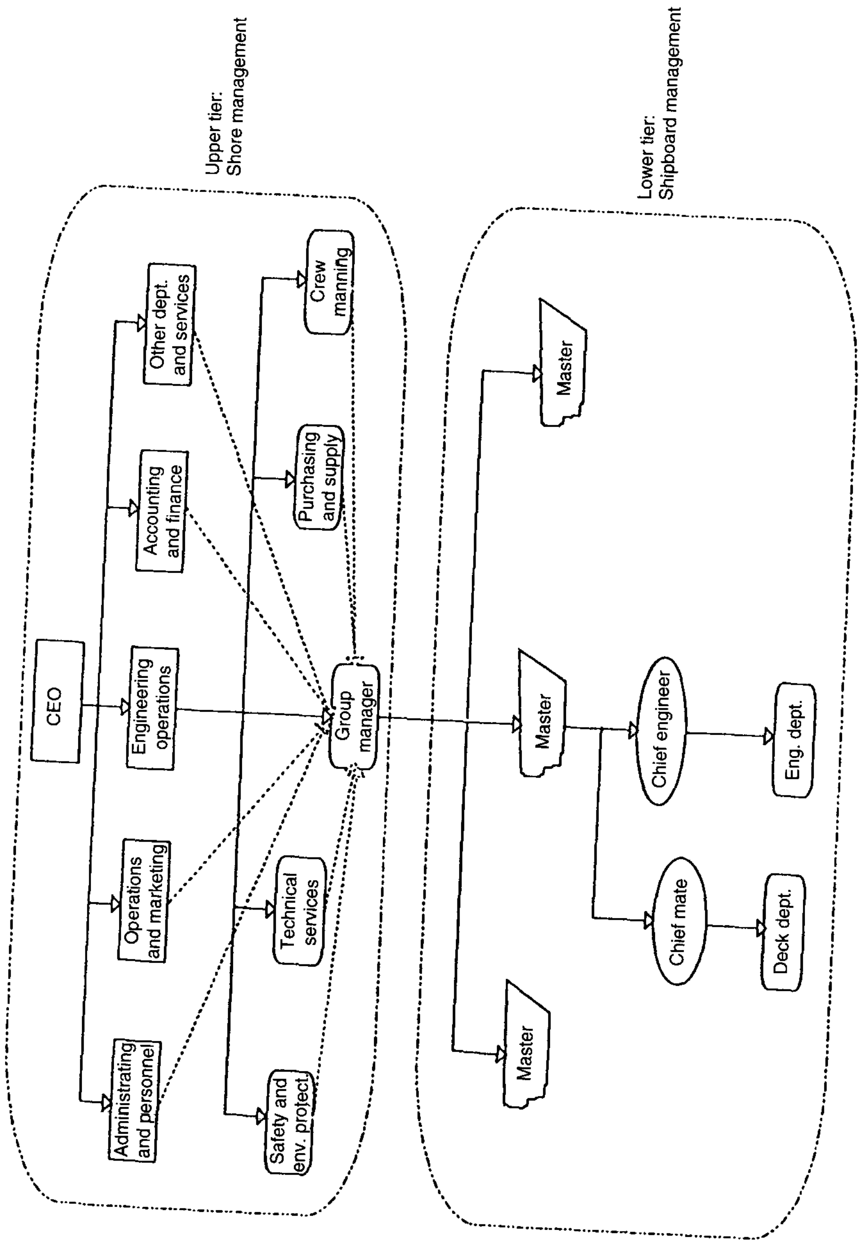


Figure 13-1. Organizational structure of engineering operations management

3. Engineering operations: direct supervision is done by the engineering manager or vice president of engineering, whatever title might be used by a company.
4. Accounting and finance: handles allocation of funds, budget, payments, and financial control including maintaining separate accounts for each ship.
5. Other departments and services: deal with training, insurance and legal matters, long- and short-term planning of the company activities, managing the agents, and so on.

The shore-based management organization in a large shipping company includes immediate engineering supervision of a group of ships by a group manager; the members of the group manager's team are called either marine, ship, or operation superintendents, or port engineers. This variety oftitles used by different shipping companies describes practically the same line of responsibilities:

- ensuring efficient and safe operation of the entire ship and all its systems, machinery, and equipment
- safeguarding normal operational conditions of every ship component and system by providing required maintenance at a minimal cost and with the lowest possible loss of ship's operating time
- coordinating the preparation and supervising the performance of all planned and emergency repairs to the ships under command
- taking care of the ship's compliance with all requirements of the classification agency, U.S. Coast Guard, and other regulatory bodies, and arranging for timely inspections and surveys

Functional departments under the engineering manager assist the engineering operations personnel. For instance, the technical services (R&D, designers, system engineers, and others) work in close cooperation with the engineering operations team. They provide vital engineering support including troubleshooting, devising of modernization projects, analysis of shipboard technical data, development of manufacturing procedures and requirements for maintenance and repairs, and more. The purchasing and supply department arranges for ordering and procurement of materials, fuel, lubricants, and spare parts. The safety and environmental protection department is responsible for implementing and monitoring safety procedures and requirements in ship operation, maintenance, and repairs in order to prevent pollution and other hazardous accidents. Crew manning takes care of selecting, training, medical control, bringing the crew on and offthe ships, and also monitoring licensing and crew documents.

The lower tier of the organization belongs to the shipboard management team headed by the ship's master. Most commercial ships have a traditional organization with deck and engine departments. On full-crew ships, a steward department may be added. The master is in command of the entire ship and is fully responsible for effective operation and safety of the ship, crew, and cargo. The engineering department, headed by the chief engineer, is responsible for the operation, maintenance, and repair of the engineering plant, and also for repairs of all other systems and equipment on board. The organization and management of the engineering department varies from one shipowner to another, depending on a number offactors, which include basic,management policies of the company, specifics of ships, type oftrade, crew size, level of automation of the engineering plant, etc. Naval, cruise, and some other types of ships have large engineering departments with several engineering officers, and also electricians, qualified members of engineering departments (QMEDs), wipers, and oilers. Some ships, such as offshore supply boats and ferries, sail either without engineering officers, or with a limited one- or two-member engineering crew, as on a ship with engine room automation.

In job descriptions, most shipowners formally assign certain maintenance and repair functions to specific engineering positions. However, with a reduced engineering department, many of these assignments are shifted to the shore engineering staff, creating additional shores ide positions. It is obvious that in order to keep the fleet in normal operational condition, the entire scope of maintenance and repair activities should be performed, and only the distribution between the engineering department on board and the engineering services of the shore office varies.

SHIPBOARD ENGINEERING OPERATIONS

There are two principal components of the day-to-day engineering operations on board ships: engineering watchstanding, and maintenance and repair activities.

Engineering Watchstanding

The *Engineering Operations Manual* of the Military Sealift Command identifies the content of engineering watchstanding as inspecting, operating, and testing machinery and equipment according to set requirements during the course of the watch. Another goal of the watch officer is the safe and efficient operation of machinery and equipment, which affects the safety of the ship and the crew on board.

Before assuming watch, the engineering officer makes a thorough inspection of all engineering spaces. The primary purpose of this inspection is to determine the condition and operation of main and auxiliary systems,

special modes of operation due to equipment failure and malfunction, any changes of operation conditions due to bad weather, changed properties of water, etc. To become familiar with the events of the previous watch, the engineering officer reviews information recorded in the engine room log and other documents, such as the standing orders and special instructions ofthe chief engineer, a list of maintenance and repair activities carried out, etc.

At the commencement of his watch, the engineering officer checks the levels of fuel, lubricating oil, water, and other liquids in machinery and tanks; the condition and water level in bilges; and the availability and condition of fire fighting and other safety equipment. During the watch period, the officer assumes full responsibility over the operation of the engine room machinery and equipment. This includes supervision ofthe activities ofthe engineering crew when they manually operate machinery and equipment.

At regular intervals, the watch officer examines the condition of operating machinery, levels of liquids, temperatures, and pressures in machinery. He or she immediately initiates required corrective action whenever any irregular or unsafe condition occurs. The watch officer is also required to make notes in the logs and other relevant documents regarding corrective actions and maintenance and repair activities, as well as the responses to the bridge orders for direction, speed, and power changes. At the completion of the watch, the watch officer reviews the events ofthe watch with the relieving officer, especially any variations from normal plant operations and ongoing maintenance and repair work.

In order to reduce the engineering crew, many companies operate ships that are equipped with engine room automation. Depending on the scope, the ABS classes engine room automation as "Automatic Control System for Unattended Engine Room" (ACCU) and "Automatic Bridge Control System for Unattended Engine Room" (ABCU). In both cases the reduced engineering crew performs maintenance and repair work during normal working hours instead of standing a regular watch. When necessary, even after normal working hours, the engineering officers must respond to any alarm conditions. Inspection and certification procedures for ships equipped with unattended engine rooms are set in the ABS *Rules for Building and Classing Steel Vessels.*

Management of Routine Service and Maintenance Procedures

The following are the major components of a management system for shipboard maintenance and repairs related to ship's structures, pipelines, machinery, and equipment:

- routine service and maintenance procedures
- inspections and surveys
- inventory of materials and spare parts

Maintenance embraces routine services such as adding or changing lubricants and coolants, cleaning, etc., performed at frequent intervals (every watch, daily, or weekly), and planned maintenance procedures including inspections, tests, adjustments, and replacement of nonbasic parts like filters, packing, belts, some bearings, and valves. Planned maintenance procedures are normally based on the manufacturer's recommendations regarding required services, time intervals (running hours, elapsed time, etc.) and certain operational parameters like pressure, temperature, vibration, material loss, etc.

Shipboard maintenance is traditionally divided between deck and engine departments according to function. The following is the scope of involvement of the ship's administration in maintenance activities.

MASTER

In coordination with the chief engineer, the master ensures that the ship's planned maintenance program is carried out properly and sustains a safe and efficient ship operation with minimum cost and minimum downtime.

CHIEF ENGINEER

The chief engineer coordinates and supervises the performance of the maintenance program for all machinery and equipment assigned to the engineering department. The engineer also provides assistance to the deck department team when certain maintenance actions and repairs are beyond their capability.

FIRST ASSISTANT ENGINEER

The first assistant engineer carries out the maintenance work according to the schedule, reports the completion and the related data to the chief engineer, and holds responsibility for the spare parts inventory management.

CHIEF MATE

The chief mate coordinates and supervises the performance of the maintenance program for all machinery and equipment assigned to the deck department except major actions that are beyond the capability of the department.

The scope of the shipboard maintenance activities performed by the engineering crew includes the following:

- creating a list of servicing routines related to ship's structures, machinery, and equipment for every watch, day, and week, in compliance with planned maintenance procedures set by the company and the manufacturer's recommendations
- planning and scheduling maintenance actions and assigning responsibilities
- monitoring performance of the maintenance procedures

- keeping logs and records of maintenance actions
- reporting maintenance scheduling data and accomplished maintenance actions to the shore management
- keeping a machinery history

A preventive maintenance and repair system is the prevailing approach in modern ship operations. It means that maintenance and repairs are carried out before failures occur, so that running costs do not become excessive. There are two preventive maintenance methods: periodic and conditional, or predictive in naval terminology. The periodic method requires that maintenance and repair procedures are performed at fixed calendar intervals or upon achieving certain accumulated running hours. Conditional or predictive maintenance is based on the actual condition of the ship components; maintenance and repairs are carried out when the condition reaches a certain level. Most shipowners employ a combination of both methods. The periodic method is applied to ship components where actual operational conditions cannot be determined at any given moment by visual inspection or by other nondestructive methods. It is also used when machinery or equipment must be opened and inspected on a periodic basis according to classification requirements.

Where the condition of other ship components might be determined by employing the known nondestructive diagnostic procedures, these components are routinely maintained on a periodic basis, but special maintenance and repair actions are performed depending on the results of the inspections.

Shipping companies use a wide variety of nondestructive methods and instruments to evaluate the condition of ship components. These methods and instruments fall into two groups: direct measurement and indirect diagnostic. For instance, evaluation of the condition of a cylinder liner might be performed in accordance with the diagram shown in figure 13-2. In addition

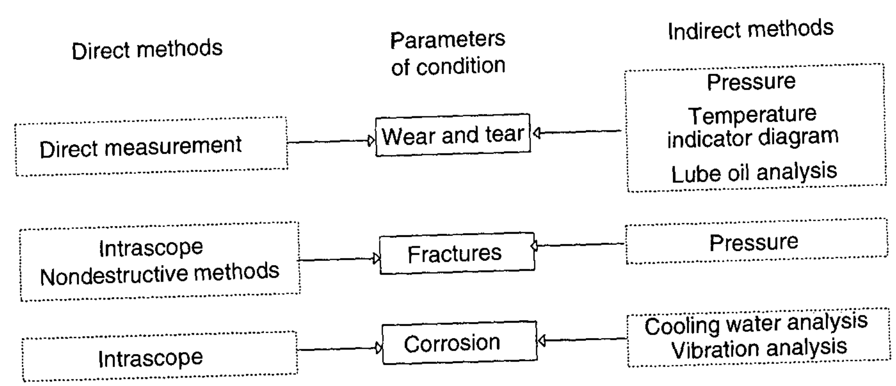


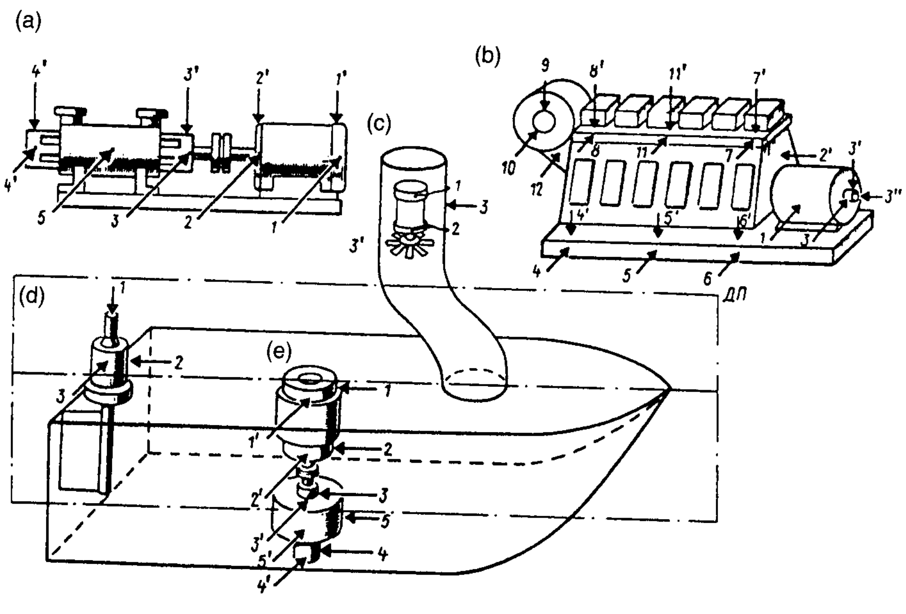
Figure 13-2. Methods of evaluating a cylinder liner condition

to fixed or mounted thermometers, different types of portable gauges have been developed, both contact and noncontact, primarily of infrared type.

Vibration tests might be used to evaluate the status of practically any rotational mechanism on board a ship. The results of the test analyses provide valuable data on the operational condition of the rotating and load bearing elements. This data is used to determine the required maintenance and repairs without relying on the time-based maintenance schedule. Figure 13-3 presents diagrams of vibration measurements applied to several types of machines. The points and directions of measurements shown on the diagrams might be standardized based on the recommendations onso as well as the allowed rates of vibration. A detailed description of vibration analysis methods may be found in of volume 2 of this manual.

Other widely used methods of indirect monitoring of the condition of the ship's machinery and equipment include the following:

- analysis and monitoring of lubricating and hydraulic oil
- measurement of the resistance of electric machinery insulation



(a) Rotating machinery units with horizontal shafts
(b) Diesel generators and turbochargers
(c) Ventilation fans
d) Steering gear and rudder stock
e) Rotating machinery units with vertical shafts

Figure 13-3. Diagrams of vibration measurement

- thermographic analysis of machinery performance
- testing, analysis, and treatment of boiler and feedwater, and also diesel engine cooling water

Oil monitoring and analysis include taking samples of oil from different units of equipment and performing onboard tests. More comprehensive oil analyses may be performed by a shipping company's laboratory or by specialized engineering companies. Test results are evaluated and necessary corrective actions are undertaken. It may be found necessary to change the oil or to use a better brand. In the case of excessive contamination, the storage and sump tanks and piping should be flushed, cleaned, and dried before refilling is done.

Shipboard electrical machinery is subjected to heat, humidity, corrosive effects of seawater, and possible physical damages. The combined impact of these factors might compromise the integrity of the insulation and lead to a failure of the equipment. Therefore, the engineering crew should determine the necessary preventive actions by conducting regular testing of the insulation. The testing is normally performed in accordance with the manufacturer's recommendations and the approved company procedures. It is a responsibility of the shore management to ensure that all licensed engineering officers are familiar with the methods and procedures of insulation testing, and that the proper equipment for taking readings is available on board.

Testing and treating boiler and feedwater is essential for the safe and efficient operation of boilers and heat exchangers on board. Chemical testing of water provides the data for evaluation of the conditions inside boilers, other steam system equipment, and pipelines. Chemical treatment of water decreases the concentration of dissolved solids and oxygen. Combined with periodic bottom blowdown, the chemical treatment prevents sludge accumulation and equipment failures. The engineering department should test water daily, periodically check system equipment for the absence of water contamination, and inspect boilers to evaluate the effectiveness of the water treatment program.

Modern diesel engines operate at high temperatures and heat transfer rates, causing increased accumulation of mineral deposits on the water cooled surfaces. Distilled water used for cooling contains dissolved oxygen, which makes it even more corrosive than regular water, especially at higher temperatures. Therefore, it is necessary to chemically treat the cooling water by adding corrosion inhibitors to provide a thin protective film on metal surfaces.

Both periodic and conditional methods allow for planning and scheduling of maintenance and repair activities, so most shipowners employ a planned preventive maintenance and repair system. The difference between the periodic and conditional methods is in the actions that are planned. In the first case, the scope of the planned maintenance and repairs is identified in advance, while in the case of the conditional method, the nondestructive inspection is planned, but the maintenance actions are based on the inspection results.

The principal responsibilities of the port engineer (operation superintendent, etc.), who plays the key role in managing ship maintenance, include the following:

- maintaining a current database for each ship under command
- developing maintenance requirements and monitoring ship crew compliance
- planning and reviewing maintenance plans developed by crews, and monitoring their accomplishment
- assisting the ship's crew in carrying out condition monitoring activities including vibration measurement, liquid analysis, and also machinery performance monitoring, and arranging for outside companies' participation

A planned preventive maintenance and repair system provides definite economic benefits when applied to those ship components that, if failed, would affect the safety of the ship or that might cause delays, damage to the cargo, and other serious losses such as fines for pollution, legal costs, etc. If these losses when weighted by the estimated probability of their occurrence exceed the average cost of the required maintenance and repair actions, the preventive maintenance system is justified. On the other side, the most efficient way of performing maintenance and repairs of some internal structures and nonvital auxiliary machinery is to wait until the excessive wear or failure occurs.

Management of Inspections and Surveys

These include planned statutory and classification ship surveys and government inspections by the country of registry. The principal management functions consist of keeping track of the status of inspections and surveys, scheduling and arranging them, and preparing the ship for the inspection and survey visits. Chapter 15 gives more details on the functions and operations of a classification agent.

Management of Inventory of Materials and Spare Parts

The following functions are included in inventory management:

- creating an indexing system by assigning fixed codes to every equipment and machinery unit and also to their components and parts
- analyzing and evaluating a required stock of spare parts
- ordering spare parts

- receiving, recording and storing spare parts
- recording use of spare parts and change in onboard stock, and reporting it to shore management
- updating records

Two considerations affect the volume of inventory on board. First, it must be assured that whenever there is a need for material or a spare part, it will be found in stock. Absence of some spare parts or material might cause substantial economic loss due to ship's delays and out-of-service time, and also the high cost of delivery on short notice. Second, there is an opposing economic consideration, i.e., too many spares on board means an unnecessary investment.

Therefore, the fundamental goal of the inventory management system is to maintain the optimal stock of spares and material on board. The initial amount of spares is provided by the shipbuilder upon ship's delivery. This original stock is constantly replenished, changed, and modified while the crew and the owner carry out maintenance and repair activities, and also replace, alter, and modernize some machinery and equipment. Specific inventory quantities are determined by the statutory and classification requirements, manufacturer's recommendations, owner's own experience, and also the type of trade in which the ship is involved.

The principal functions and components of the shipboard maintenance and repair organization are easily computerized; therefore, most shipowners use some type of computer-based system. Chapter 14 describes the fundamental principles of such a system.

SHIPYARD REPAIRS AND OVERHAULS

Planning of Shipyard Repairs

Shipyard repairs include repair work that is normally beyond the crew's capability. While there are a variety of classifications of shipyard repairs, the most general one recognizes two principal subdivisions: scheduled or planned repairs and unscheduled or emergency repairs.

Scheduled shipyard repairs, or overhauls, are performed on a periodic basis, every 24-30 months, in order to accomplish general repair work, drydocking, and alterations.

Unscheduled repairs, sometimes also called emergency repairs, are carried out to accomplish repair work of an emergency or unanticipated nature that cannot be deferred until the next overhaul.

As it relates to planning of ship repairs, an efficient and economical engineering approach should be based on the following fundamental principles:

- accurate and current assessment of condition of the ship's equipment and components

- strict compliance with statutory and classification requirements and maximum use of allowed grace periods
- maximum use of ship's crew for maintenance and repairs
- strict adherence to all safety requirements of the relevant owner's and manufacturer's instructions and procedures
- survey and selection of qualified repair facilities, and designation of a shipyard using competitive bidding

Long range planning of shipyard repairs is based primarily on the classification and regulatory requirements. When the ship enters operation, a schedule of surveys and repairs is set for the entire life cycle. This subject is discussed in detail in chapter 15. An example of a typical schedule of surveys is given in figure 13-4. As a normal practice, the shipyard repairs are scheduled at the time when a special or an intermediate survey should be done. Every shipping company tends to use the allowed grace period and to set the time for repairs at its end. However, the operational schedule of the ship might force an adjustment to the planned date of repairs.

The necessity to comply with various domestic and international regulations might also affect the scheduled repair time. If, for instance, the effective date of a certain regulation requiring a specific modernization or alteration project appears to be before the scheduled repairs, the planned arrival to the shipyard is moved ahead.

Obviously, the condition of the ship is another determining factor. Results of onboard maintenance inspections and/or emergency situations like

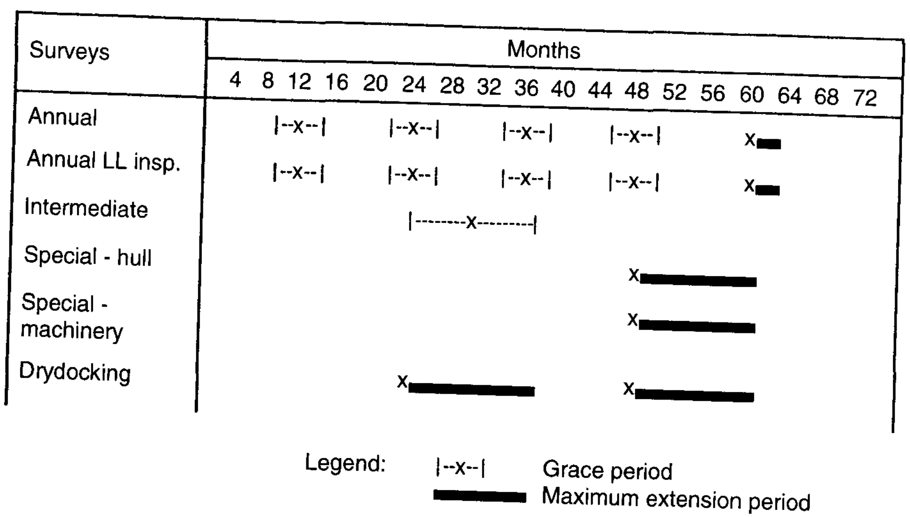


Figure 13-4. Typical five-year cycle of classification surveys

accelerated wear, poor paint condition, or failures of ship's machinery and equipment might advance the date of repairs.

Another aspect of planning for repairs is estimating their duration and allocating an appropriate budget. They are interrelated, i.e., longer repairs normally cost more. Furthermore, longer repair periods mean increased time out of operations, and corresponding loss of revenue. Therefore, shipowners employ various methods intended to reduce repair time such as in-port or voyage repairs, riding repair teams, etc. The time and projected costs of these activities must be planned in advance and reflected in a general plan of shipyard repairs and in the allocated budget.

Planning of an overhaul starts during the previous shipyard repairs, when certain items are excluded from the repair specification and scheduled to be performed at the next overhaul. In addition, some survey reports might allow for deferral of rectification actions, or some results of operational tests and ship trials might reveal certain deficiencies to be dealt with during the next visit to the shipyard. During the inspections carried out when the machinery and equipment units are opened or hull structures are exposed, many repair items to be dealt with in the next repair period are identified. At the same time, drawings are prepared and spare parts are ordered or fabricated. Actually, this is the time when the preparation of the repair documentation commences.

Ship Repair Facilities

The industrial base for repairs of ships includes a large variety of enterprises:

1. Specialized ship repair yards capable of carrying out full-scale repairs of large ships including drydocking.
2. Topside repair facilities for any type of ship overhauls except repairs of the underbody. Most of these yards generally have extensive berth/pier space for various repairs from simple jobs to topside overhaul.
3. Ship repair shops specializing in repairs of certain ship components such as machinery, equipment, accommodations, etc. These enterprises include specialized, independently owned facilities located at major shipyards (repairs of electrical machinery, electronic and radio equipment, furniture, automation, etc.) and facilities in ports that specialize in emergency and voyage repairs.
4. Most shipbuilding yards combine ship construction activities with repairs, overhaul, and conversion of ships.
5. Another type of ship repair organization that is becoming very popular with shipowners is a mobile repair gang. With the constantly growing cost of construction and operation of ships, taking a ship out of operation to visit a shipyard is getting more costly every year. The shipowners are looking for alternative methods of repair by bringing ship

repair personnel on board rather than sending ships to shipyards. Major ship repair yards as well as small repair shops, and also ship equipment manufacturers, create mobile teams of repair workers equipped with necessary portable tools. These teams board ships while they are in port or underway to carry out emergency and voyage repairs and also to do regular maintenance of equipment and machinery. The main goal-the ships are not taken out of operation.

Based on the survey of the United States shipbuilding and repair facilities conducted annually by the Maritime Administration (MARAD), in 1997, over two hundred private companies were involved in ship repairs in the U.S. Under the MARAD classification, the term "major shipyard" applies to a yard capable of drydocking ships 400 feet (122 meters) long and larger. There are available 43 floating drydocks, 31 graving docks, and 2 marine railways capable of handling ships of this size. In addition to drydocks, these major shipyards have extensive berthing capacities. Figure 13-5 presents the number of available floating drydocks for ships of various length. Figure 13-6 illustrates the principal features of shipyard drydocks and marine railways.

An average full-scale repair shipyard has shiplifting facilities and piers with heavy-lift cranes, and a set of repair departments (shops), including the following:

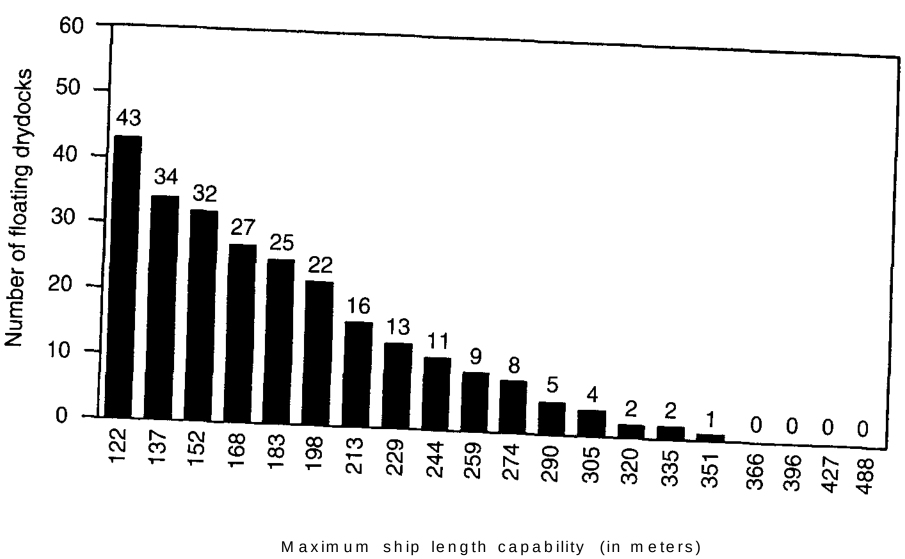


Figure 13-5. Number of floating drydocks by maximum length capability in major U.S. shipbuilding and repair yards (as of October 1, 1997)

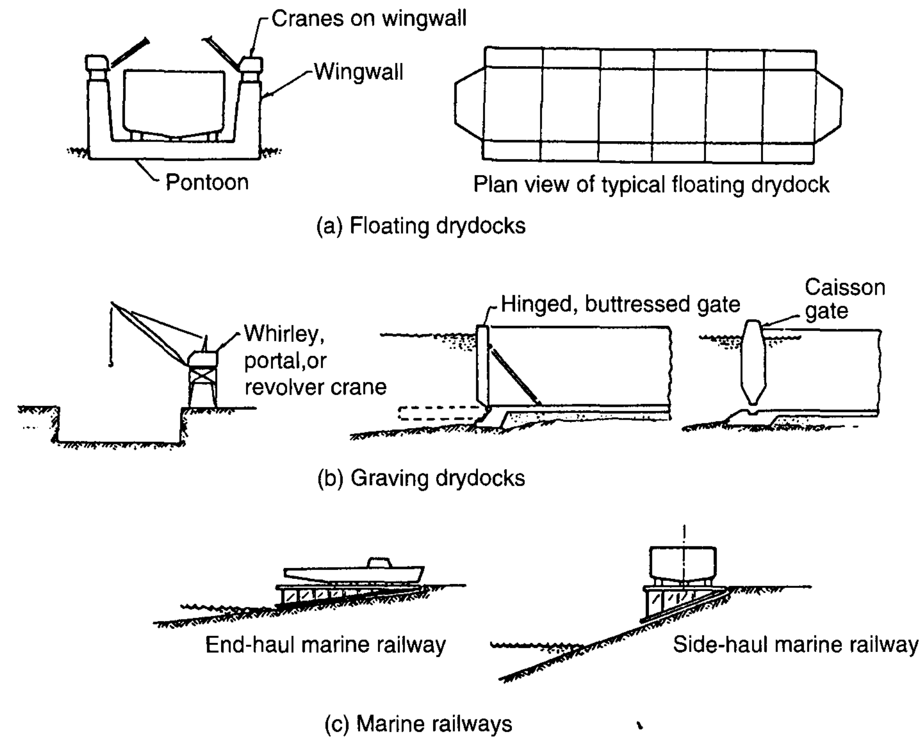


Figure 13-6. Principal features of shipyard drydocks and marine railways

- shipfitting: hull repairs
- pipefitting: repairs of pipelines, valves and fittings
- electrical: repairs of electric lines, switchboards and terminals (electrical machinery is repaired in this shop or, more often, in an on-site specialized shop of a large company)
- machine shop: repairs and fabrication of shafts, pump and valve components, etc. using various machine tools
- sheet metal: fabrication of ducts, conduits, boxes, control panels, etc.
- painting: cleaning and painting of ship's hull, equipment and intemals
- woodworking: repairs and fabrication of wooden elements of systems

Some major yards have small foundries and even equipment manufacturing facilities. Figure 13-7 presents a layout of a major ship repair yard.

Preparation for Shipyard Repairs

Preparation for shipyard repairs is a long and tedious process, lasting the entire period from the time of previous repairs until the ship enters a shipyard again. The principal management and organizational actions

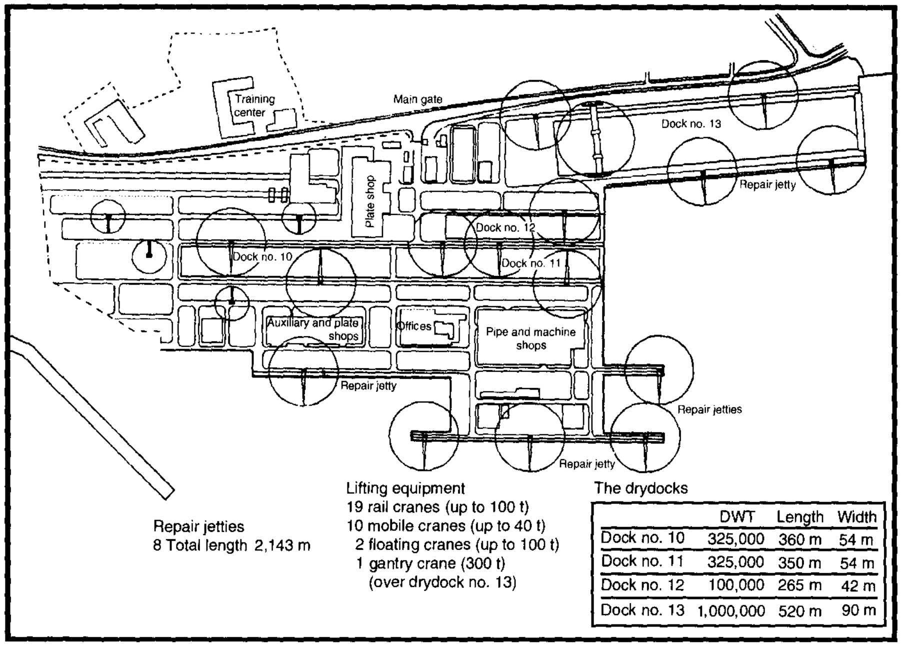


Figure 13-7. Layout of a major ship repair yard.
Courtesy of the Lisnave Shipyard.

performed during this period are outlined in figure 13-8. Besides the operations department, the process involves the crew of the ship, other departments of headquarters, like technical services (R&D, engineering, design, etc.), purchasing, accounting, and others. The process also involves marine manufacturers and vendors, engineering companies, classification agency, and shipyards.

The shipowner assigns the supervision of the entire process to a port engineer who acts as the owner's representative. This person might have another title such as repair engineer, operations superintendent, marine superintendent, repair superintendent, etc.; however, the term "port engineer" will be used in this chapter to identify the shipowner's representative. The primary responsibilities of the port engineer, in preparation for an upcoming shipyard repair and before awarding the contract to a particular shipyard, include the following steps:

- development of a schedule of all activities required to prepare for the repairs
- preparation of the repair documentation package
- preliminary estimation of the expected repair cost based on the repair documentation package

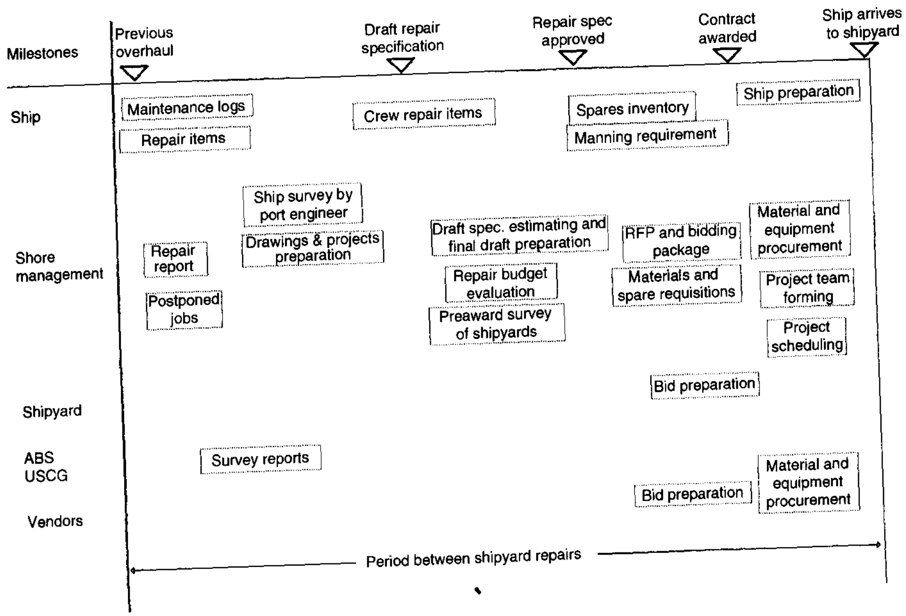


Figure 13-8. Time scaled sequence of actions in preparation for shipyard repairs

- pre award survey of the prospective shipyards
- issuance of the request for proposal and forwarding the repair documentation package to the prospective shipyards
- arrangement of pre bid conferences, receipt of the bids and their evaluation

SCHEDULE OF ACTIVITIES

The port engineer coordinates preparation for shipyard repairs and supervises their performance. In keeping with the company's policies and procedures, the first action is a development of a schedule of activities to identify all company actions required in preparation for shipyard repairs and to assign responsibilities and completion dates. The list should include the following principal activities:

1. Developing the repair documentation package to be offered for bids by shipyards.
2. Compiling a listing of the owner furnished materials, equipment, and replacement parts, and arranging for their procurement.
3. Reviewing the need for crew members at a shipyard with an intention to minimize the crew size while ensuring that all needed activities and jobs are properly assigned.

4. IdentifYing all capital improvement and modernization projects to be carried out in conjunction with the shipyard repairs.
5. Arranging for classification and statutory surveys in accordance with the company policy to perform as many surveys as possible in ports or underway in order to minimize the out-of-service time.

There are several reasons to increase the scope of Owner furnished equipment and materials. First, the cost is lower because no markup fee is applied as is customary when the shipyard is responsible for the procurement. Second, higher quality and better performance might be achieved by properly choosing the best but not necessarily the least expensive vendor. And third, better warranty conditions might be negotiated by the owner's purchasing department. However, it is important to keep in mind that by supplying the materials and equipment, the owner releases the shipyard from their responsibility for timely delivery and high quality. This could lead to possible disputes and claims. Therefore, thorough analysis and justification are required when the owner decides to furnish certain materials, equipment, and replacement parts.

Every shipowner wants the ship's crew to do as much of the repair work as possible while the ship is at a shipyard. However, the shipyards resist because they wish to do more repairs using their personnel. Furthermore, the crew activities disrupt the production process by interfering with the shipyard personnel. And finally, the crew work at the shipyard creates certain insurance and legal problems. As a result, the scope of the crew work should be thoroughly analyzed and approved by the shipyard before the contract is signed.

Most capital improvement projects involve substantial design activities and equipment procurement. It might take time. Therefore, the port engineer should check that the preparations are complete, the project documentation is adequate, and the equipment is ordered and expected to be delivered before its planned installation. Quite often, a capital project causes delays and excessive expenses at shipyards because the project documentation was inadequate.

REPAIR DOCUMENTATION

A repair documentation package normally includes a list of repair items or a repair specification, relevant drawings, job description, and technical requirements for certain repair items, and also quality control procedures. A repair specification is not only a principal repair document, but also a foundation of the entire repair cycle. There are two types of repair specifications-individual and standard.

The *individual repair specification* is a simple list of repair jobs to be carried out, with as many details as possible regarding manufacturing procedures and requirements, needed materials, and spares. The individual

repair specifications are prepared either when the emergency repairs happen, or when there is not enough available information on previous repairs of this ship or of the same class of ships.

The *standard repair specification* is used when a company operates several ships of the same class or, if a single ship, when sufficient data on the wear and tear ofthe ship is obtainable. A repair specification ofthis type is used for any future repairs of a class of ships or even for a specific single ship. It contains repair items related to ship structures and to every system and piece of machinery and equipment. An excerpt from a standard specification is given in figure 13-9. In addition to the provided information, a more advanced standard specification might include needed materials, spare parts, and estimated man-hours per unit of each repair item.

Because a standard specification is prepared in advance, it includes comprehensive information on manufacturing procedures and requirements, and a detailed list of repair items. When a standard specification is used for a particular overhaul of a ship, the appropriate items of the

No.	Unit code	Repair item	Amount			Manufacturing procedures and requirements
			Unit	Plan	Actual	
. . .						
II. Hull systems and machinery						
33	415	Anchor windlass: Renew drum and wildcut brake lining	Each	2		Free up all linkages and grease, remove scale, free up and grease pins in locking bars.
. . .						
V. Auxiliary engine room systems						
87	740	Electric motors—fuel oil system Overhaul motors: a. FO service pump, 45 kW, 1,800 rpm b. FO transfer pump, 37 kW, 900 rpm	 2 2	 2 2		Disconnect and remove to shop, overhaul, thoroughly clean all wiring with approved solvent, bake dry, renew all ball bearings, reassemble and test in shop to owner/representative's satisfaction. Return on board and reinstall, check alignment and retest.

Figure 13-9. Excerpts from a typical standard repair specification

specification are selected, and those not planned for this repair cycle are simply disregarded.

Obviously, the development of a standard specification requires a substantial investment of engineering effort. However, this investment is justified by obvious improvement of repair organization when the standard specification is used. The following list shows the principal advantages of a standard repair specification compared to an individual one:

- saves time in preparing a specification for the given shipyard repair project
- ensures that technical adequacy and sufficiency are maintained
- ensures compliance with the company's policy regardless of who is preparing the final draft
- improves quality of the final draft by avoiding errors in job description and omissions of vital repair items
- provides cost savings

Similar to procedures followed in auto and air transport industries, some shipowners use *fixed work packages* for repairs of some equipment. These packages include a mandatory list of repair items to be carried out regardless of the subsequent inspection results. Normally a part of the standard specifications, fixed work packages are based on running hours or elapsed time in operation in accordance with the manufacturer's recommendations. The fixed packages have a limited use and are applied to the most vital machinery and equipment units.

Both individual and standard repair specifications have practically the same format. They are subdivided into major divisions, including the following:

1. Hull.
2. Hull system machinery and piping.
3. Accommodations, outfit and furnishing.
4. Propulsion plant.
5. Auxiliary ER systems, machinery, and piping.
6. Electrical (power plant, electrical machinery, and cables).
7. Communications and navigation systems.
8. Specialized systems (IGS and COW for tankers, lashing for container carriers, etc.).

In addition to the above divisions, a specification also contains a special "General Conditions" section that includes the shipyard and port services, tugboat assistance, tank cleaning and gas freeing, oily water removal, drydocking and pier services, classification surveys, agency services, and other support. Some companies choose to apply to the repair specifications

the format ofthe ship construction specification, which uses more detailed subdivision. In this case, a special column is inserted (see fig. 13-9) with a code for every piece of equipment or ship structure that is taken from the ship construction specification.

DEVELOPMENT OF A REPAIR SPECIFICATION

The preliminary draft of the repair specifications is prepared by the port engineer prior to a visit on board ship for survey and inspection. This preliminary draft is based on the following information:

- Master's and chief engineer's communications regarding maintenance and repairs, including delayed emergency repair items and required actions due to equipment malfunctioning.
- Information from shipboard documentation, both written and computerized, like logs, maintenance books, and machinery operational files transmitted by mail, radio, or computer communication.
- Deferred items and recommendations removed from the final specification of the previous overhaul, and also those items recommended by the port engineer in the last shipyard report.
- Outstanding corrective actions suggested by the surveyors in the current statutory and classification survey reports.
- Historical data concerning the ship, its equipment, and machinery.
- Recommendations of the company's technical staff regarding improvements, modernizations, and alterations.

During a working visit on board while the ship is in a port or underway, the port engineer adds new repair items as a result of his or her survey and inspection. This extended draft of the repair specification should be compiled a few months prior to a major overhaul. The draft might contain two or three times as many repair items as will be in the final draft. With assistance from technical services and other departments of headquarters, the port engineer evaluates and analyzes every repair item and makes a recommendation regarding inclusion in the final draft.

In order to improve the decision making process and to assist in the analysis of the repair specification, it is useful to note a practice of assigning priorities to the repair items. A tentative distribution of repair jobs based on priority might be as follows:

1. The highest priority would be assigned to *mandatory* jobs, which are required to maintain seaworthiness, structural integrity, and reliability of the ship's hull, propulsion and power plant, navigation and communication equipment, etc. These items include also U.S. Coast Guard and classification requirements and corrective actions related to machinery and equipment failures and excessive wear.

2. The next priority level might be assigned to *conditional* jobs, or those whose final Scope cannot be correctly identified until additional survey is performed during the actual repairs. Examples of these jobs are most of the repairs on underwater hull, structures, and system components (propeller, rudder, tailshaft, sea valves, bow thruster, etc.), and also items of the "open, inspect and repair if necessary" type related to machinery, heat exchangers, tanks, etc.
3. Another level of priority might be allocated to *urgent* jobs in connection with auxiliary machinery and equipment, accommodations, deckhouses and superstructures.
4. *Desirable* jobs are those nonvital repair items that might be performed if the funds are available, or postponed until the next repairs, or carried out by the crew or by a riding repair team at much less cost.
5. A special category is assigned to *optional* items that are dependent on the budget status and require further analysis. Some capital improvement and modernization projects, if not mandated by a regulatory body, might be included in this group.

Out of all the above priority groups, only optional repair items should be formally identified as such in the final repair specification, and a separate cost proposal is normally requested from the shipyard.

The tedious work of finalizing the specification starts with distributing the work items into priority groups. As a rule, all mandatory, conditional, and a large portion of the urgent jobs remain in the draft. The desirable items are analyzed and some of them are removed and either deferred until the next repairs, dropped completely, or reassigned for completion by the crew or a riding team. A similar procedure is applied to the optional jobs.

The next stage of refining the draft specification is estimating the approximate cost of every item. The port engineer and his assistants establish the estimates based on the unit costs collected from various shipyards and on the final costing data from the repairs performed on this and other ships of the company. An estimate is an approximation of the cost of every repair item in terms of required labor, material and equipment, energy, and services. Obviously, this is not an exact process, but rather a reflection of the port engineer's judgment regarding fair and reasonable compensation for the work. It is assumed that the job is performed by an average shipyard, with average labor force, and with average work performance.

The results of cost estimation are used for analyzing and assigning a proper repair budget and funding, for evaluation of the bids submitted from shipyards, and also for finalizing the development of a repair specification that includes all the priority items. The sum of cost estimates for all remaining repair items in the draft specification is measured against the available budget, which starts another round of cost adjustment and refining the specification. Based on the assigned priorities, more jobs are removed from the specification, some others are modified to reduce scope and cost. For instance, originally planned blasting and recoating of a ballast tank due to insufficient funding might be replaced by mechanical scraping and touch-up coating.

PREAWARD SURVEY OF SHIPYARDS

Before requesting a bid from a shipyard, the shipowner needs to know if the yard is capable of performing the proposed repairs and overhauls. A pre award survey serves this purpose. It is a vital component of the contract award process. Its main goal is to assemble adequate information on the shipyard in order to make an educated judgment on its capability to successfully perform repairs according to the owner's requirements relating to quality, quantity, schedule, and costs.

Two principal components of a pre award survey are an on-site inspection of the yard and the analysis of information related to the yard. This information normally includes data received from the yard as well as from other customers of the yard, from credit agencies, from corporate rating services, from industry publications, and from the shipowner's own reports on the yard's performance during previous repairs. The on-site surveys might be carried out by direct touring of the facilities, observing the work in progress, watching operational demonstrations and tests of systems, interviewing key personnel, analyzing procedures, records, and other pertinent data, and studying the production plans and schedules.

As per MARAD guidelines, a complete and comprehensive pre award survey should embrace the following areas.

- facilities: berths, docks, production shops and equipment, support and auxiliary services, and also quality assurance system
- production: labor resources, engineering support, material procurement and control
- finance: owned and borrowed financial resources, accounting system and procedures
- management: production planning, scheduling and control, organizational structure, responsibilities and authorities, information system
- safety and security: facilities and procedures

Obviously, the scope of the survey is highly dependent on the capabilities of the shipowner, expected frequency of repairs at the target shipyard, and the scope of the planned overhaul. Major shipowners who carry out quite a few repairs per year would benefit greatly from thorough and comprehensive preaward surveys.

It is important to keep in mind that the survey might involve sensitive proprietary data; therefore, special security arrangements should be made, and the shipyard should be assured by the shipowner that no unauthorized persons will be allowed to review the findings.

COMPETITIVE BIDDING AND CONTRACT AWARD

In order to choose a shipyard to carry out planned repairs, most shipowners use competitive bidding procedures. As Soon as the repair documentation has been finalized, a proposal request is prepared and sent to solicit quotations from as many shipyards as is feasible. The bidding package includes the repair specification, drawings and other documentation, and also the contractual terms, conditions, and requirements that the owner wants the shipyard to follow. The list of shipyards selected for bidding includes only those that have been previously formally surveyed or are known to the owner as reputable and qualified enterprises based on previous repairs performed.

The owner usually specifies the total maximum duration of the repairs and requests the yard to quote the total cost. It is a normal practice, however, to request the yard to submit a bid on the itemized basis. This way the owner can evaluate the proposals and discover possible pricing errors or underbidding, as well as determine when a yard anticipates additional jobs and change orders to compensate for a low bid if the contract is awarded. Itemized bids also allow the owner to collect valuable economic data for future reference.

These itemized prices are also used for pricing additional job items and changes, and to assist in negotiations. For that purpose, the shipyards are requested to provide certain unit rates as a part of the quotation package. These unit rates might include the cost of one ton of hull plating renewal, one running foot of piping replacement, one foot of welding seam, etc.

The shipowner evaluates the submitted prices and, as a general rule, awards the contract to the lowest bidder. Ifthe owner uses standard specifications, the bidders submit their prices for each item of the specification, and based on these prices the owner might negotiate a further discount in return for a better deal for the shipyard, i.e., awarding a group of ships on the scheduled basis. For a shipyard, this arrangement might be quite attractive; the yard might be willing to lower quoted prices because of the larger volume of work and scheduling advantages. Moreover, the labor requirements and duration of the subsequent repairs of the same class ships might be reduced because of identical tools or special equipment and technology employed.

REQUIRED CREW SIZE

Another important activity to be done prior to arrival at the shipyard is estimating the required manning level for the repairs. The primary component of this requirement is the crew needed to carry out the repairs as per the crew repair list. While the port engineer estimates the repair specification, the master, together with the chief engineer and the chief mate, evaluates the crew repair list for the required labor force, and also for materials and spare parts. The port engineer reviews the estimates and includes the material and spares requirements in the budget. Final estimate for the crew size to arrive at the shipyard should be approved by the group manager.

Management of Shipyard Repairs by a Shipowner

SHIPOWNER'S PROJECT TEAM

Overhaul of a ship at a shipyard represents a specific example of a large-scale project implemented under a contract. The content and the goal of shipyard repairs corresponds to the general definition of a project as a series of related activities or events required to be carried out according to a contract that has specific start and finish points. Like a regular industrial project, the shipyard overhaul is finite, cost-defined, complex, homogeneous, and nonrepetitive. Therefore the principal project management techniques apply to shipyard repair contracts.

Project management is a process of establishing and organizing objectives, and planning and employing resources to accomplish them. The basic objective of a project team at the stage of actual performance of repairs is to ensure that all the requirements of the contract are satisfied: the contract documentation, the contract delivery schedule, and the financial and legal requirements of the contract. Depending on the scope of repairs, the project team might consist ofthe port engineer alone, or include one or two assistants and possibly a specialist with practical expertise in the above major areas.

The shipowner's project team should be established and ready to start functioning immediately after the contract is awarded. First, the date of ship's arrival to the yard is set. The next step is development by the shipyard ofthe detailed production schedule for the entire scope of repair work. The making of a schedule is a necessary component of the contract, and should be formally included in the contractual requirements. The schedule will be used by the shipowner's project team for monitoring the progress of repairs. It is suggested that the project team maintain its own repair schedule, using either the one provided by the shipyard, or a recreated version. Maintaining a schedule includes periodic updating and analyses of bottlenecks and critical jobs.

Another area of the project team's activities is delivery of the owner supplied materials, equipment, and spare parts. Most of the material and supplies should reach the yard prior to the arrival of the ship. A substantial amount of equipment and spares might be brought on board the ship itself in order to save on transportation expenses.

The project team also makes sure that the classification surveyors are available for conducting the scheduled surveys and inspections. The team should verify the availability ofthe representatives of equipment and material manufacturers or of service companies who may assist in carrying out necessary supervision and tests.

Upon arrival of the ship at the shipyard, two principal commencement activities take place: a kickoff conference with the shipyard administration and a similar meeting with the ship's crew. During the discussions with the shipyard administration, the following subjects are revisited and emphasized: contract conditions, requirements and performance strategy, safety procedures, and the authority and process of managing additional work and changes in the work scope and content.

The primary subjects,ofthe ship's meeting are as follows:

- review and analysis ofthe production schedule with the ship's officers
- assignment of responsibilities in shipyard progress monitoring, test witnessing, safety procedures, providing spare parts and equipment to the shipyard personnel
- scope and procedures related to performing the repair items from the crew repair list
- procedures and authorities of the port engineer, his or her assistants, and the engineering officers in identifying additional work and changes and requesting their performance from the shipyard.

Based on the objectives discussed above, the major functions of the shipowner's project team during the performance of the shipyard repairs include engineering decision-making, cost management, contract legal subjects and negotiations, planning and scheduling, and control and monitoring. The members of the team conduct daily inspections of all work carried out both on board and in the shops. Any noted discrepancies, inefficiencies, and defective work are immediately identified to the shipyard production management team for correction. At the same time, all shipyard-caused delays, material waste, defective work, or inefficient labor utilization are recorded in order to be used in the final negotiations and possible disputes. In certain extreme situations, these data might assist in preventing or resolving legal actions. It is a good practice to record the number of workers and the duration of their work on each repair item.

The port engineer and his assistants meet on a daily basis with the shipyard production supervisor assigned to the ship (ship manager) in order to review the repair progress, to coordinate scheduled activities, to resolve production deficiencies and safety hazards, and also to discuss any changes in the scope and content ofrepair work. Periodically, the port engineer meets with the shipyard production management in order to settle production problems and discrepancies.

PROJECT PLANNING AND SCHEDULING

Modern production management theory treats planning and scheduling as two distinct, separate, and sequential operations. In the planning stage, the work units suitable for various levels of management are identified, their logical sequence is set, and the interrelations among the work units are determined. Also identified are all labor, material, and support resources, and engineering and management operations required to perform every work unit.

The project schedule should represent the overall logical sequence through which work on the project must proceed. The most widely used scheduling technique is a network diagram based on a thorough analysis and identification of the logic and relationship among all jobs describing the project. Development of a network diagram is a time consuming process requiring thorough knowledge ofthe production process and also basic familiarity with the scheduling methodology. A sequence of scheduling actions might be presented as follows:

- breakdown of a project into activities (a term used in project scheduling to identify single jobs)
- estimation of activity durations
- identification of interrelations of the activities
- preparation of a network diagram
- calculation of the schedule and identification of the critical path

The fundamentals of network scheduling are discussed in many publications. A basic familiarity is easily obtained by exercises within the development and calculation of simple network diagrams. Figure 13-10 presents a simplified arrow diagram of a ship drydocking project based on the list of activities given in table 13-1.

The principal purpose of a schedule development is the creation of a document for monitoring and evaluating the contract progress and also for analysis of actual delays and identification of the causes and consequences. The activities of a schedule might be classified as production and support activities. A schedule should reflect the inherent complexity of a project; separate activities should cover all support actions including managerial decisions and procedures, engineering, material and equipment procurement. Packaging of activities requires tying together by the same schedule date all support activities needed for implementation of a certain production activity.

COMPUTERIZED SCHEDULING

Multiple computer programs are used by shipyards and shipowners for project scheduling. Among them are systems based on Program Evaluation and Review Technique (PERT), and on Critical Path Method (CPM).

TABLE 13-1

List of Activities of a Ship Drydocking Project

Activity ID	*Activity Description*	*Original Duration*
1	Drydock ship	1
2	Washdown hull	1
3	Survey hull	1
4	Clean hull	2
5	,Repair hull	3
6	Paint hull	4
7	. Remove sea valves	2
8	Remove propeller	1
9	Polish propeller	2
10	Remove tailshaft	1
11	Replace anodes	1
12	Repair tailshaft	2
13	Reinstall tailshaft	1
14	Repair sea valves	4
15	Reinstall sea valves	2
16	Reinstall propeller	1
17	Repair bow thruster	8
18	Undock ship	1

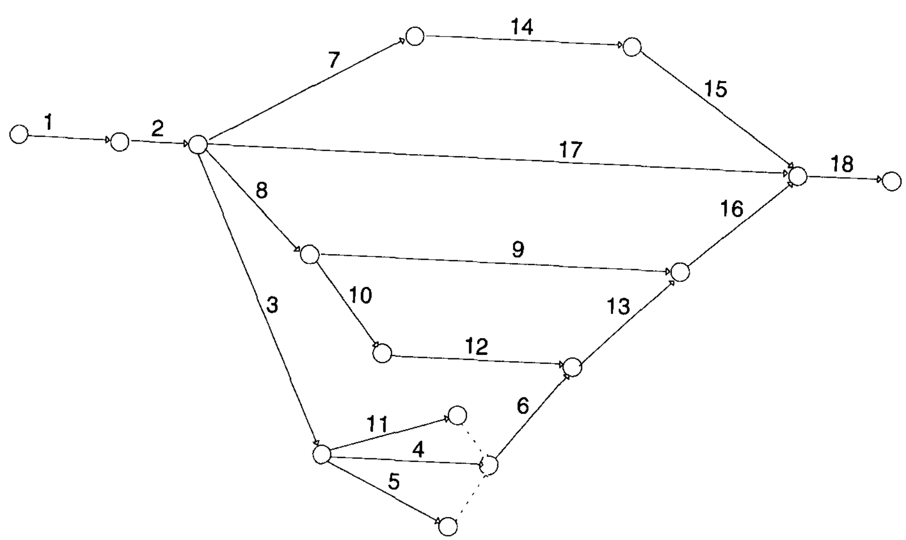


Figure 13-10. A simplified arrow diagram of a ship drydocking project

The PERT system allows users to deal with the uncertainty ofthe duration of activities and is applied mainly to major long-term projects with a comparatively high level of uncertainty. CPM methods are appropriate for comparatively small projects with a high level of certainty. A critical path is a sequence ofthe scheduled activities defined such that ifthe completion of anyone of these activities is delayed, the completion of the project will also be delayed.

A typical computerized scheduling software includes the following components:

- project organization and planning package including data collection, selection, and structuring
- schedule development package including the table and graphics tools
- schedule analysis facility including rescheduling, pop-up activity strings, constraints, etc.
- schedule optimization procedures based on resources allocation and reallocation
- schedule presentation, communication, and reporting facilities based on tabular and graphics reports
- project evaluation and control facilities based on schedule updates and rescheduling
- project monitoring facilities for upper management, as well as for the shipowner project team

A variety of available graphics tools might be illustrated by a bar chart and a time scaled logic diagram of the above simplified drydocking schedule (see figs. 13-11 and 13-12). Both graphics have been produced using a Finest Hour software package of Primavera Services, Inc. ofBala Cynwyd, Pennsylvania.

An important duty of the shipowner's project team is the evaluation of any impacts on the schedule caused by project changes, production plan modifications, late delivery of materials or equipment, increased or altered work scope, etc. As soon as the schedule is developed and the progress monitoring and reporting system is available, the rescheduling will follow automatically. Regular updates ofthe schedule should include revisions to the duration-to-completion and resource requirements for activities already underway, based on the owner's estimates and on those of the ship superintendents.

A proper scheduling and monitoring system should first of all provide the port engineer with the schedule information on a timely basis. The completion of an activity should be easily identified, and properly reported. It is very important to establish a system of activity completion verification by the crew members with a proper assignment of responsibilities for delegated authorities.

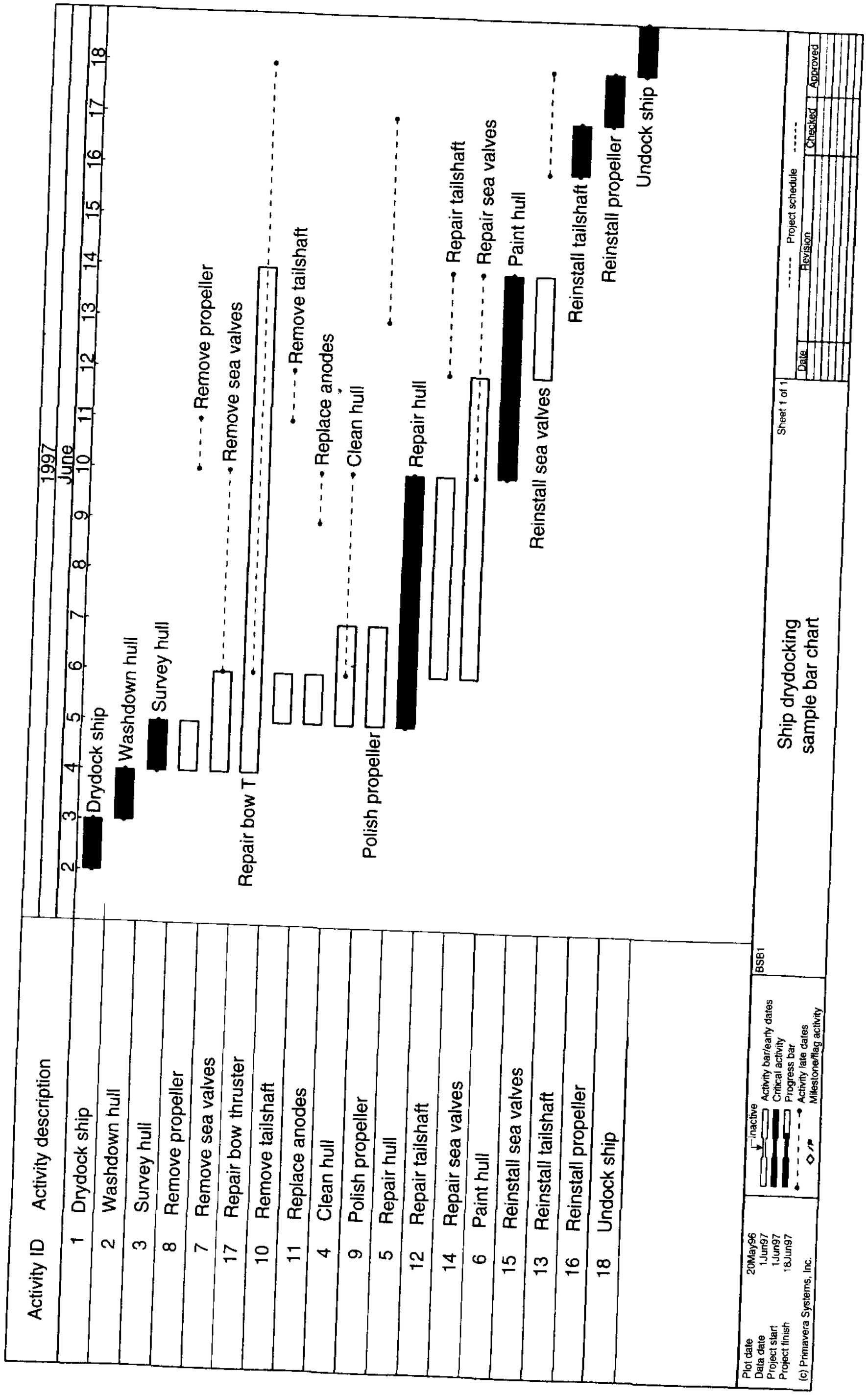


Figure 13-11. Bar chart of a ship drydocking operation

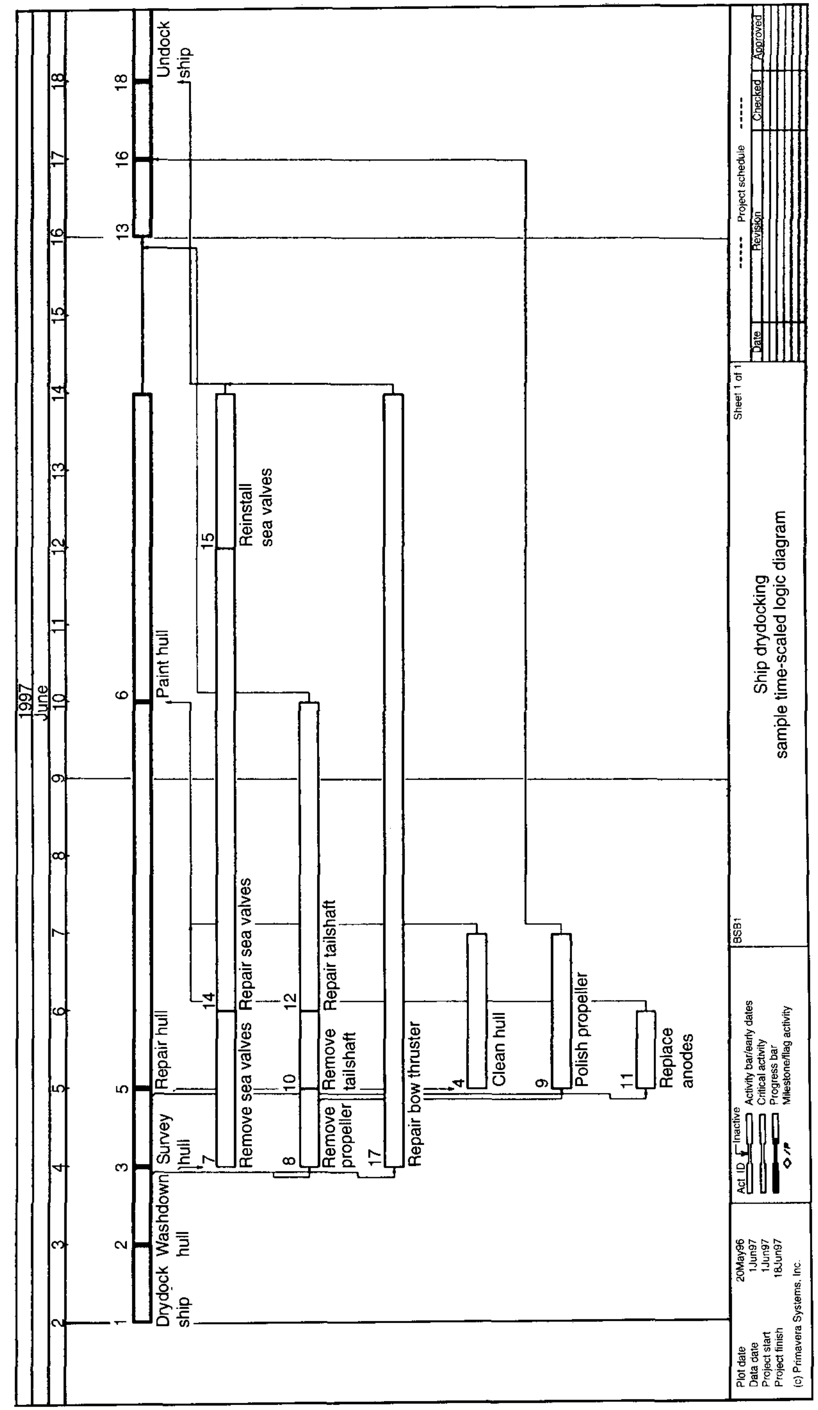


Figure 13-12. Time scaled logic diagram of a ship drydocking operation

COMPLETION OF REPAIRS

At the completion stage, the shipowner's project team is involved sequentially in two major activities: ship's trials and settling of the final bill. The agenda and duration of ship's trials depend upon the scope ofthe repairs. Normally, in order to evaluate the quality of repairs and to demonstrate it to the shipowner's representative, the shipyard conducts dock and sea trials. The dock trials are intended to check the auxiliary systems and machinery and all safety systems, and also to ensure that the ship is ready for sea trials.

The agenda ofthe sea trials is developed by either the shipowner's agent, or by the shipyard with an appropriate approval of the shipowner. The principal goal is to test the propulsion plant underway, and also all support systems including propulsion control, steering gear, and navigation and radio equipment. Both dock and sea trials are carried out by the shipyard crews; however, the ship's engineering team observes every test and reports to the port engineer any discrepancy and defective repair work.

Soon after the repairs are completed and the ship returns to normal operation, the port engineer prepares a report for the company management. The report contains the chronological data, detailed description ofthe procedures, and the repairs performed. The most valuable part of the report is the analytical portion, where the port engineer evaluates the shipyard's performance and analyzes the costing data. Finally, he or she makes recommendations regarding the future use of this shipyard and suggests those repair items that should be included in the next repair's specification.

Economics of Ship Repairs

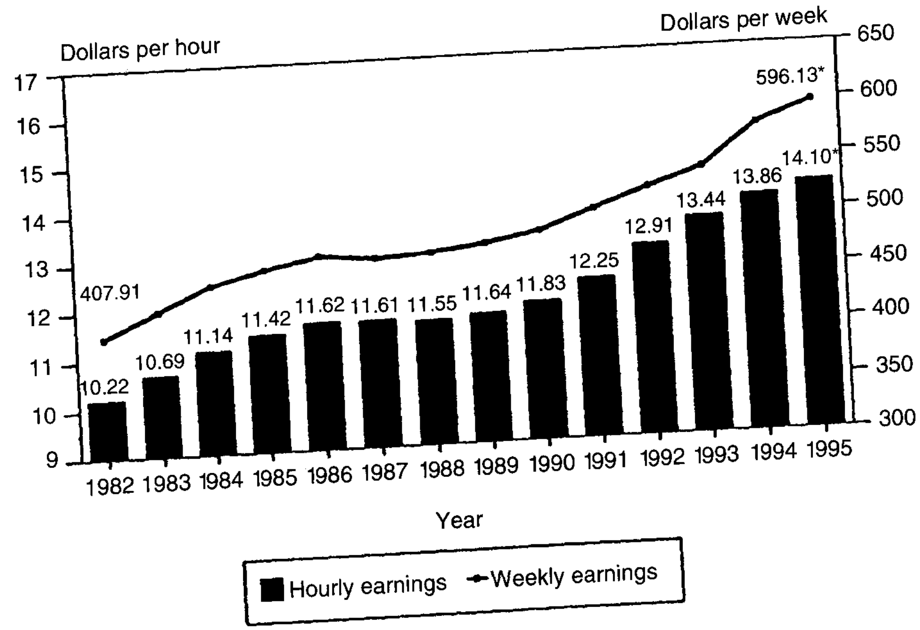
Economic analysis procedures should be applied at various stages of shipyard repairs. First of all, in the case of a relatively aged ship, a feasibility study is carried out in order to decide if it is worth repairing the ship or if it might be the right time to scrap it. Another possible consideration is to replace an expensive overhaul with low-scale repairs that will support operation of the ship for another two or three years until it is scrapped. The principal data considered in the analysis includes the initial and residual cost of the ship, annual cost in operation, cost of the proposed overhaul, and also the cost and annual operating expenses for another ship that might replace the one under the evaluation. The details of this type of economic analysis and examples are provided in the publication *Marine Engineering Economics and Cost Analysis* (see references at the end of this chapter).

Another economic analysis based on the cost estimates of the repair items should be performed when the final draft of the repair specification is prepared. The following list shows the major components of these estimates:

- cost of direct production labor, including all principal manufacturing trades (shipfitters, pipefitters, mechanics, painters, etc.), and

also direct engineering, testing, quality control, and production supervising
- cost of materials and equipment including those provided by the shipowner and by the shipyard
- overhead or indirect shipyard expenses (management, clerical, maintenance of the shipyard equipment and tools, certain energy costs, and also additional labor costs like benefits, insurance, etc.) that are allocated as a fixed percent to the direct labor cost
- direct charges including surveys, drydocking, use of a pier, firewatch, etc.
- shipyard profit that is included in the final cost as a fixed percent of all above expenses.

Production labor cost is calculated as a product of the required labor measured in man-hours and the average labor rate at the shipyard in dollars per one man-hour. Figure 13-13 shows the trend ofthe average labor rate at United States shipyards as per U.S. Bureau of Labor Statistics. It is important to keep in mind that shipyards often provide the shipowner with a labor rate that already includes the overhead. A rough estimate of the



Source: Bureau of Labor Statistics
* Average for 9 months

Figure 13-13. Average earnings in U.S. private shipyards required labor is based on total duration of a given repair job and the size of a team of workers. For instance, three workers will need five hours to install a pump. Thus, the total estimate of the required labor is 3 x 5 = 15 man-hours. Shipyard estimators use their own rates oflabor requirement for various standard jobs; these rates are normally based on statistical evaluation of the historic data. It is a good practice when a port engineer collects labor data from various shipyards for future use.

The fOllowingadditional factors should be considered in the proces~ of estimation of the labor cost:

- specific work conditions such as congestion on board, weather conditions, tight spaces. etc..
- travel time that depends on location of the work station on board, location of the shop, storages, availability of accesses, etc.
- safety requirements
- testing, certification, and survey requirements
- need for assistance of manufacturer's representative, or other trade workers like riggers, crane operators, etc.

Estimates of material cost are based on the drawings, material lists, and price quotes from the material manufacturers and vendors that are obtained by the purchasing department. A similar procedure is applied when the cost of equipment and machinery is evaluated. If the shipyard supplies certain material or equipment, a 10 to 15 percent markup is included to cover transportation and handling.

A port engineer applies the same procedures to evaluate the changes in the scope and content of repair items and to request additional work. However, certain data needed for estimation, like labor rates, markup and profit rates, direct charges, etc., will be provided by the shipyard to reflect the actual situation.

SHIPBOARD SAFETY

Safety Organization

The parties with the most direct impact on shipboard safety are the vessel owner and operator, the ship's crew, the classification society, government agencies, international safety organizations, and industry organizations.

CLAsSIFICATION SOCIETIES

Virtually all large, oceangoing ships receive seaworthiness certificates from a classification society. Classification societies such as the American Bureau of Shipping, Det Norske Veritas, Lloyd's of London and numerous others fill a role between the owner and the insurer. Through class

rules-standards for building and outfitting the ship-class societies provide the technical expertise the insurer lacks. Through inspections (or surveys) to ensure the vessel meets class rules, class societies provide the owner with third party certification of seaworthiness, enabling the owner to become eligible for reduced insurance rates.

Classification societies' efforts are very similar to those of government safety agencies, and, in many instances, governments accept work perfonned by class surveyors, adopt class rules, and even delegate safety responsibility to class societies. There is a philosophical difference, however, between the primary goals of government safety agencies and class societies. Government safety agencies have a responsibility to the public to ensure the vessel is not a threat to life, property, or the environment. Government agencies must consider their marine safety program's interaction with other transportation safety efforts and the overall national transportation needs. Classification societies are businesses and exist to help protect the investment of the owner and the'insurance company.

GOVERNMENT AGENCIES

Most countries with ships under their registry maintain a government agency responsible for maritime affairs. Although safety on the seas is a global concern, both from an economic and an environmental standpoint, the methods governments use to manage marine safety vary.

In the United States, the primary agency responsible for shipboard safety is the United States Coast Guard. The U.S. Coast Guard's Marine Safety and Environmental Protection Directorate manages all aspects of waterborne transportation safety through the marine safety regulations contained in Titles 33, 46, and 49 ofthe Code of Federal Regulations. These regulations address safety requirements on all types of commercial vessels-tugs, fishing vessels, tank vessels, cargo vessels, mobile offshore drilling units, passenger vessels, nautical school ships, oceanographic research vessels, barges, and miscellaneous vessels such as dredges. The regulations also establish minimum safety standards for offshore structures and pleasure craft. Marine safety regulations are increasingly developed in partnership with the maritime community. Industry advisory groups, trade organizations, and standards development organizations frequently playa direct role in establishing policy and providing feedback from their constituency to government agencies.

United States regulations are extensive, covering all aspects of ship construction, outfitting, stability, manning, crew qualifications, and the safe carriage of passengers and cargo. Compliance is ensured through a comprehensive inspection program conducted by the Coast Guard and recognized third parties. In addition, the Coast Guard conducts investigations of marine accidents to determine cause so that corrective action can be identified to prevent recurrence. The Coast Guard also administers the United States seamen's licensing and certification program.

All marine safety field activities are recorded by the Coast Guard on the Marine Safety Information System (MSIS). MSIS is a computer network that links all Coast Guard marine safety offices. Through data stored in MSIS, the Coast Guard tracks the safety history ofthe U.S. fleet and foreign flag ships calling in U.S. ports. MSIS enables the Coast Guard to identify trends and target resources for their most efficient use.

Several other United States government agencies have responsibilities that affect shipboard safety. The Maritime Administration (MARAD)promotes U.S. shipping and the shipbuilding industry. The Occupational Safety and Health Administration (OSHA) maintains regulations related to workplace safety that include ships in a shipyard and requirements for confined space entry. The Mineral Management Service (MMS) regulations address industrial equipment for th"eproduction of petroleum products on offshore platforms and mobile offshore drilling units.

Few other nations have as sophisticated a network of marine safety resources as the United States. In many other countries, classification societies function as quasi-governmental agencies, conducting inspections and issuing documents on behalf of the flag state.

INTERNATIONAL ORGANIZATIONS

The primary international organization responsible for shipboard safety is the International Maritime Organization (IMO). IMO is a specialized agency ofthe United Nations. Like similar U.N. bodies, IMO is a forum for the discussion and resolution ofinternational issues. IMO membership includes representatives from governments and the maritime industry to ensure both perspectives are heard. IMO's organizational structure is shown in figure 13-14.

The Maritime Safety Committee is the most senior of the committees that carry out IMO's technical work and is most directly involved with shipboard safety. The Marine Environmental Protection Committee is responsible for coordinating activities in the prevention and control of pollution from ships. The Legal Committee is responsible for considering any legal matters within the scope ofthe organization. The Technical Cooperation Committee is responsible for coordinating work in the provision of technical assistance in the maritime field, particularly to developing countries. The Facilitation Committee is a subsidiary ofthe Council and is responsible for activities and functions related to reducing the formalities and simplifying the documentation required of ships when entering or leaving ports or other terminals.

IMO agreements are implemented through binding international treaties and nonmandatory instruments. Nations ratifying an IMO treaty agree to enforce the treaty provisions on their own ships and rely on other

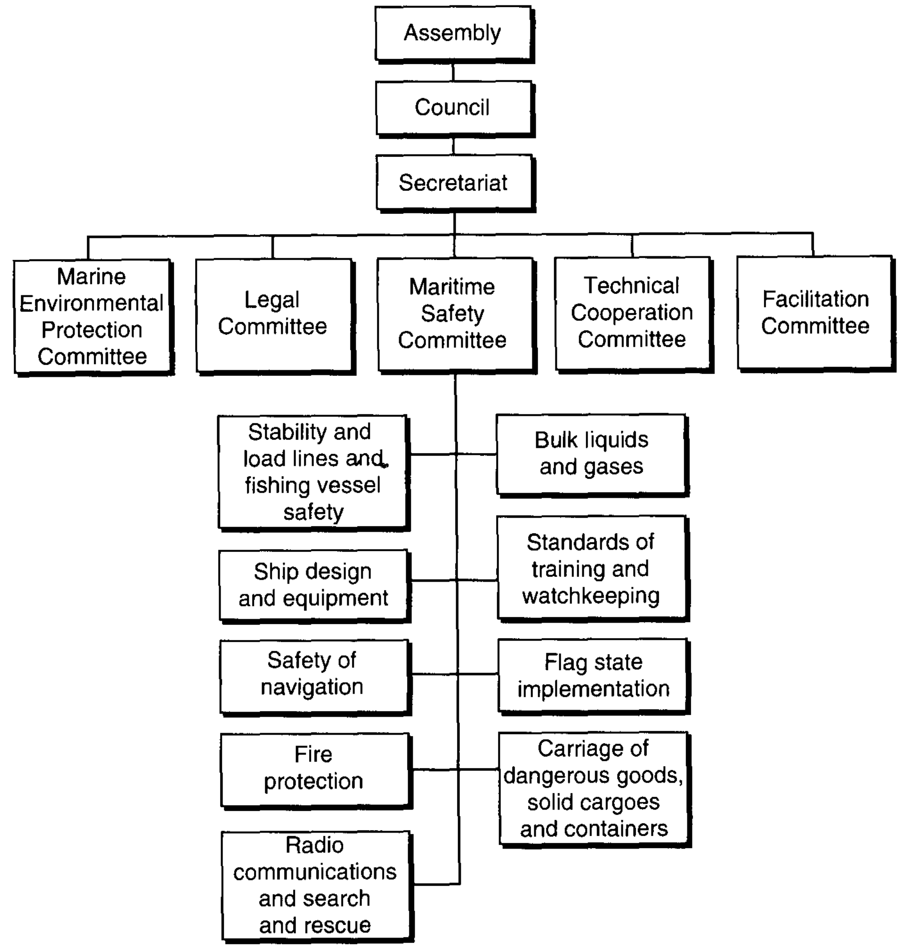


Figure 13-14. Structure ofInternational Maritime Organization (IMO)

nations to do likewise in the interest of consistent application of requirements and overall maritime safety. Although reciprocity is a crucial element of the IMO process, port states have the authority to intervene on foreign-flag ships in their waters to ensure compliance with treaty provisions when necessary to protect port state resources.

Nonmandatory instruments ofIMO include circulars, codes, and guidelines that are published to address particular safety issues or specific types of vessels. The application of these standards depends on the flag state administration. In some cases, a flag state, or its designated third party organization, may choose to incorporate a nonmandatory IMO policy into its national rules to be applied to domestic shipping.

The International Labor Organization (ILO) is another specialized agency of the United Nations with interest in marine safety, particularly as it relates to crew conditions. The ILO has actively drawn attention to the operation of substandard ships and the treatment of crewmembers. As early as the 1930s, ILO published recommendations for member states to discourage their seamen from serving on ships under substandard conditions with regard to repatriation, medical care, certificates of competency, and documentation of service.

In 1976, the ILO adopted the Merchant Marine (Minimum) Standards Convention, ILO 147, which references a number of other ILO conventions to address minimum age, medical examination, articles of agreement, officers competency certifictes, food and catering aboard ship, crew accommodations, prevention of occupational accidents, sickness or injury benefits, repatriation, protection of collective bargaining and the right to organize, hours of work and manning, and vocational training. Member nations signatory to ILO 147 are obligated to have laws or regulations established to implement the provisions of the treaty. The convention also contains provisions for investigating complaints against foreign ships calling at a signatory nation's ports and for reporting the results to the ship's flag administration.

TRADE AND STANDARDS ORGANIZATIONS

Trade and standards organizations contribute to safety in the marine industry by promoting consistency. Through trade and standards organizations, industry guidelines, practices, and expectations are often translated into consensus standards and procedures. These standards and procedures help establish the accepted norms, promote quality control among equipment manufacturers, and raise the level of confidence in the marine 'community. Noncompliance with broadly recognized standards and procedures can lead to loss of credibility, labor problems, and, in some cases, liability.

The trade or standards organizations in the U.S. are too numerous to list completely. However, some ofthe organizations most directly involved in shipboard safety in the U.S. are as follows:

- American Boat and Yacht Council
- American Institute of Merchant Shipping
- American National Standards Institute
- American Petroleum Institute
- American Society for Testing and Materials
- American Society of Mechanical Engineers
- American Waterways Operators
- American Welding Society
- Institute of Electrical and Electronics Engineers
- Lake Carriers' Association
- Marine Engineer's Benevolent Association
- Masters, Mates and Pilots
- National Fire Protection Association
- Passenger Vessel Association
- Society of Naval Architects and Marine Engineers

Similarly, a number of international trade and standards organizations have had a large influence on standardization in the maritime community. A few of these are the following:

- International Association of Classification Societies
- International Association of Drilling Contractors
- International Electrotechnical Commission
- International Organization for Standardization

OWNER AND OPERATOR

The owner and operator of a vessel mayor may not be the same party. In many cases, ships are owned by one company and operated, or chartered, by another. Safety regulations typically identify both the owner and operator as responsible parties. Responsibility for shipboard safety may depend on contractual agreements that define which party is to ensure compliance with applicable safety requirements, or the responsibility may be manifested by the party exercising the most direct control over shipboard operations and maintenance.

Governments and international agencies are expected to protect the public and the environment from unsafe ships. It is the business ofclassification societies to develop seaworthiness standards for ships and to enforce those standards in their clients' interests. The greatest burden of responsibility for safety falls upon the owner/operator. Shipowners and operators must bear the economic burden of creating and then maintaining a safe ship. The owner/operator must also bear the moral responsibility for safety.

Safety is not free. While operating a vessel safely will translate into long-term savings, it is not without an initial expenditure of resources. These initial costs are mainly involved with the construction and outfitting of the vessel to meet the minimum requirement of every organization having actual or constructive control over the vessel's eligibility to operate. For example, shipboard cable must not produce toxic smoke in a fire. Pressure retaining piping and pressure vessels must be designed with appropriate safety factors. Primary fire-fighting equipment must be redundant. An auxiliary means of steering must be provided. Essentially every component on a vessel must be chosen according to prescribed safety criteria. The owner/operator must pay for each required safety feature that could otherwise be replaced by a component with lower design standards or omitted altogether.

A vessel that undergoes an examination to confirm its initial or continued eligibility to operate is normally issued a document attesting to its seaworthiness. In actuality, the document is only a statement of the vessel's condition at the time of issuance. It is the owner/operator's responsibility to maintain the safety of the vessel through regular maintenance, replacement, and repair.

Also, authorities generally publish requirements to describe when accidents or equipment outages must be reported. When these criteria are met, a determination of the severity of the incident is usually required. Minor damage or equipment outages may result in no lost time. More severe accidents or breakdowns may lead to the vessel being delayed until either temporary or permanent repairs are completed.

The owner and/or operator has an obligation to communicate expectations with respect to safety to the crew. This may be done through such instruments as operating manuals, company publications, or instructions. It is through this type of communication that the shipboard safety organization is explained and responsibilities are established. The following items show the types of information necessary to structure the shipboard safety organization: (1) a description ofthe safety duties of each crewmember and the training or qualifications required for that duty, (2) a list and description of the safety equipment aboard the vessel, (3) an explanation of whether inspection and maintenance of safety equipment will be conducted by the crew or shoreside personnel, and (4) the identification of resources to manage the shipboard safety organization.

Once safety expectations have been communicated, it is the owner/operator's responsibility to ensure the crew has the appropriate training or qualifications to fulfill its role in shipboard safety. In some cases, this may be legally mandated. It is unreasonable to expect crewmembers to consistently achieve satisfactory results in duties for which they are not trained or qualified. In some cases it may be possible for the owner/operator to rely on the government's licensing and certification structure to ensure crew qualifications. In other instances, it will be necessary for the owner/operator to arrange for specialized training in order for the crew to fulfill its safety responsibilities.

The owner/operator's obligation to the customer is a simple matter of business-timely delivery ofthe product at the agreed upon rates. Over the short term, business is probably feasible without an expenditure on safety. However, in a time when public awareness is high and litigation is costly, business will not be consistently viable without the owner/operator's commitment to at least the minimum level of safety required by the cognizant jurisdiction.

In recent years, governments and classification societies have taken a more active role in assisting vessel owner/operators in managing ship-

board safety. The International Management Code for the Safe Operation of Ships and for Pollution Prevention (ISM Code) was developed by the International Maritime Organization to encourage continuous improvement of safety management skills in the maritime industry. The American Bureau of Shipping is considered qualified by the U.S. Coast Guard to certify companies' compliance with the ISM via a Document of Compliance. The ISM Code is described in chapter 15.

The moral responsibility for safety can be difficult to define. It may be thought of as the unwritten responsibility of a member of the maritime community. Governments and the public want the maritime industry to behave responsibly, to exceed the minimum requirements. It is an unreasonable expectation when weighed against the realities of economics and competition. However, over the long term, an owner/operator cannot ignore the value of a good reputation. A good reputation creates trust between the owner/operator and jurisdictional bodies, potential customers, crewmembers, and the public at large. An owner/operator with a reputation for safety will have greater opportunity for autonomy from government due to the creation of a less mutually-suspicious relationship. A reputation for safety relieves customers' anxiety over investments in cargo and helps attract conscientious crewmembers. Lastly, a reputation for safety builds goodwill with the public. All of these factors lead to increased business opportunities and a decreased burden from government.

THE CREW

Safety begins and ends with the efforts of the crew. No number of regulations, international treaties, or classification society surveys will succeed in making a vessel safe without the cooperation of the crew. It is the crew who must ultimately make the safety process work by reporting problems, performing required maintenance, keeping records, watching for occupational hazards, following written instructions, and providing feedback to the other parties responsible for ship safety.

The senior ship's officers play an important role in establishing the culture and personality of a vessel and, consequently, the importance given to safety. A conscientious regard for safety on the part of the senior officers will usually translate into a greater appreciation for safety by the other professional mariners aboard.

Training and qualifications enable the crew to function effectively and fulfill their safety responsibilities. The International Convention on Standards of Training, Certification, and Watchkeeping for Seafarers (STCW), adopted by IMO in 1978, establishes international standards for crew competency. The crew should also receive the necessary training for their specific duties under the owner/operator's safety management program.

Safety Management

Unfortunately, the word management can connote passivity. It is often associated with care, guidance, and administration. It is not usually a word that is taken to imply aggressive action. Where safety is concerned, management must translate to action because the consequences of inaction might be deadly or, at least, extremely costly.

PHILOSOPHY OF SAFETY MANAGEMENT

Shipboard safety management has four goals:

- to comply with regulatory requirements
- to minimize vessel downtime
- to protect the owner/operator from liability
- to be cost-effective

These four goals are interrelated. The vessel must comply with the regulations of the flag state or administration or it will not be eligible to engage in trade. Damage, outages, or unscheduled repair time take the vessel out of service and prevent it from accomplishing its primary task of making a profit. The owner/operator is liable for environmental damage, loss of cargo, and personnel injuries that result from inadequate safety precautions. Lastly, the safety program must be able to balance cost with payback by preventing a loss from failure to meet one of the first three goals.

The word "safe" can be defined in various ways: free from harm or risk; secure from the threat of danger, harm, or loss; or nonthreatening.

The word "safety" describes the condition of being safe. But being safe is a subjective state that depends on the perception of human beings. Therefore, a better definition for the word safety might be "the condition of feeling safe."

Every human activity entails some degree of risk. Even inactivity has its own risks. Human reaction to risk depends on the consequences of the activity's hazard, the likelihood of the hazard occurring, and awareness of the hazard scenario. For example, a hazard of airline travel is crashing, which typically leads to loss of life. However, accident statistics show that the likelihood of experiencing the consequences of crashing is very low. Human perception of the hazard scenario and the likelihood of occurrence cause some people to fly and others to find alternative transportation. Removing the awareness of either the hazard scenario or the likelihood of occurrence may result in different reactions. Given awareness of the scenario, more people might choose to fly if they were convinced that either the hazard consequence or the likelihood of occurrence was being controlled.

This example illustrates the keys to effective safety management:

- identification of the hazard
- computation of the consequences
- determination of the likelihood of occurrence
- implementation of controls
- communication of the results

Finally, to achieve the goal of cost-effectiveness, the cost of the controls must be calculated and compared with the cost of occurrence.

SAFETY MANAGEMENT SYSTEMS

There are numerous safety management systems. They define a spectrum from "taking no action" at one end to "controlling every hazard" at the other end. Neither of these extremes is likely to achieve all of the goals of safety management. Taking no action will not ensure vessel compliance with safety requirements, nor will it minimize downtime or protect the owner/operator from liability. While the initial investment in safety will be low, the long-term consequences make "doing nothing" inefficient. On the other hand, taking action to control every hazard will achieve the first three goals of safety management but costs will make this option inefficient. Somewhere along the axis, cost versus benefit will balance. Effective safety management seeks to find and maintain the balance.

Safety management systems can be divided into two categories-reactive and proactive. The difference between the categories lies in how each of the systems evolve. Both types of systems begin with a set of safety criteria that are based on known or anticipated hazards. The hazards may be identified by several means: experience, examining other safety management systems, or benchmarking similar operations.

Once a baseline safety level has been established, a reactive system changes only in response to the occurrence of a hazard. In other words, a reactive system evolves from what has happened. The advantage of this type of system is its simplicity. For example, once the baseline is established, it becomes clear whether or not regulatory requirements are being met. No new safety criteria are required unless a loss occurs that is outside the scope of existing safety criteria. An investment in developing new criteria is automatically justified by the lack of criteria that would have prevented the loss. The system quickly becomes institutionalized, facilitating development of instruction manuals and standard procedures. In some cases, this degree of stability may be desirable, particularly when the human element is considered. A model of a typical reactive system is presented in figure 13-15.

The main disadvantage of a reactive system is that it does not encourage steps to identify and prevent the occurrence of hazards that remained unforeseen after the baseline was established. This characteristic makes a reactive system weak in its ability to achieve the goals of minimizing vessel downtime and protecting the owner/operator from liability.

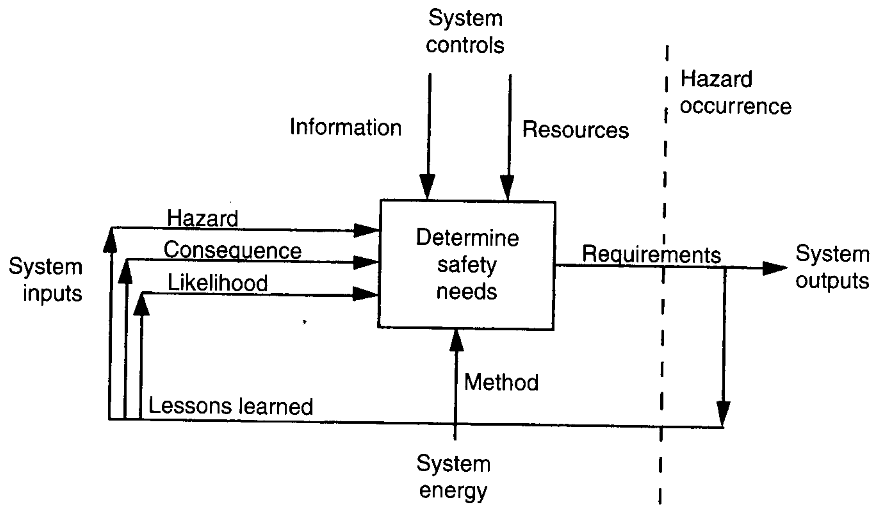


Figure 13-15. Reactive safety management system

A proactive system is shown in figure 13-16. The system is different from a reactive system in that the system evolves from what might happen. The actual occurrence of a hazard is not necessary to stimulate change. A proactive system model is characterized by an additional feedback loop.

This additional loop represents the process of anticipating and controlling hazards before they manifest. The advantage of a proactive system is its ability to prevent the occurrence ofloss. This makes a proactive system strong in its ability to minimize vessel downtime and protect the owner/operator from liability.

The main disadvantage of a proactive system is its complexity. The dynamics require a higher level of sophistication than the reactive system because of the greater need for coordination, foresight, and planning. The additional feedback loop constantly supplies new information on which to base decisions. The flow of information necessitates a long-term obligation of resources to maintain the system. These factors make the proactive system potentially less cost-effective.Also, a less obvious disadvantage is its effect on the humans who use the system. The possibility of constant change must be recognized and balanced with the human need for stability and experience.

The reactive and proactive safety management systems are not mutually exclusive. The establishment ofthe baseline should be done by the same process of initially identifying and implementing controls for as many hazards as possible. A well-managed reactive system can easily become a proactive system ifit functions to anticipate hazards. Likewise, a poorly managed proactive system can become a reactive system if the feedback loop is inefficient. Which ofthe two methods is employed to anchor a safety management

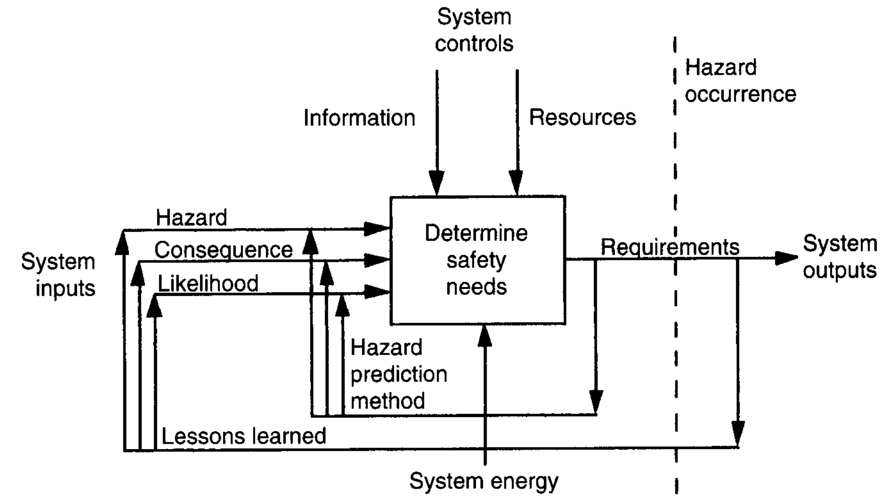


Figure 13-16. Proactive safety management system

system depends on the uncertainty in the accuracy of the baseline criteria, the resources available to administer the system, the level of sophistication of the users, and the complexity of the system being protected.

Effective safety management systems share similar characteristics. A description of these characteristics is presented below. The ISM Code discusses these concepts more fully.

Definitions of Terminology

It is critical that all participants in the safety management system understand the terms used to describe the system and how it functions. Commonly applied terminology facilitates every aspect of the safety management process. The terminology should include a description ofthe boundaries ofthe system being managed.

Recognition of Boundaries

The boundaries of the safety management system must be drawn to set the limits on responsibilities. One obvious boundary is the confines of the ship. But consideration must be given to how the "ship" boundary relates to shoreside boundaries, environmental boundaries, and other boundaries created by other safety sytems. Within the ship boundary, portions of the safety system will be operating within smaller boundaries.

Assignment of Responsibilities

The role and responsibility of each participant in the safety management system must be clearly explained. The system must include accountability and well-defined lines of authority. Responsibilities should only be assigned to individuals qualified to carry out those responsibilities. Most importantly, the system must have the owner/operator's dedication to safety management.

Identification of Actions

Managing safety means taking action. It is within this phase that risks will be evaluated and actions to implement controls will be identified. The actions must be translated into tasks and consequent resource requirements.

Identification of Priorities

The system must include a method for determining which actions to ensure safety are the most important. Priorities will be established by consequence, likelihood, and cost-effectiveness. Part of this process will entail comparing available resources against required resources.

Dedication of Resources

Once actions have been identified and prioritized, the appropriate resources must be allocated to carry out the actions. Schedules must be developed to define the timelines during which tasks must be completed. Dedication of resources includes arranging the training necessary for each responsible individual to perform the actions expected of them.

Measurements of Performance

Methods to measure whether the actions taken achieved the intended result must be integral to the system. These measurements form a large part ofthe feedback that drives system evolution.

Communications

The system must have efficient means of communication among the individuals assigned responsibilities in the system. Communications should include instruction manuals, directives, reports, and other instruments to describe the goals and expectations within the system and to promote rapid flow of information and feedback.

Documented Procedures

All procedures in the system must be described in detail to ensure continuity and consistency and to aid in the training of new personnel within the system. These include, but are not limited to, procedures for the methods of identifying and prioritizing hazards, assigning responsibilities, allocating resources, conducting and reporting periodic maintenance, reporting and investigating accidents, recordkeeping, providing feedback information, and measuring performance. The documents prepared for the system must

be available to each responsible individual commensurate with their duties within the system.

Rewards for Successes

Humans generally need recognition for their efforts. A well-run safety management system demands effort, foresight, and effective planning. Successes achieved in system management should be rewarded to provide a tangible sense of worth for the individuals assigned responsibilities within the system. A reward system can also stimulate esprit de corps and healthy competition that will add to effective management.

HUMAN FACTORS

It is a well-established fact that most accidents are caused by human error. Until recently, the response in the safety community was to admit human frailties and attempt to design humans out of the system being protected. Understanding human factors has become the next frontier in safety improvement.

Human factors and the term "ergonomics" are synonymous to an extent. They both concern recognition of the human involvement in a system rather than engineering humans out of a system. Regulations and industry standards are increasingly being written on how to construct a productive human-machine interface. This may entail planning for such factors as sizing and placing controls for ease of operation, determining the best direction of motion, or ensuring the environment is comfortable. In other cases, a keen insight into the interface is necessary. For example, the operator of a dynamic positioning system on a diving support vessel, on watch late at night, may typically take a cup of coffee to the system operating station. If liquid spilled into the controls is a hazard to system operation, a special receptacle for coffee cups may be provided to prevent occurrence. This is a type ofhuman factors engineering (HFE) that is beyond hardware considerations but is still a part ofthe human-machine interface.

Watertight Integrity

Ships float upright when the weight of the vessel is counteracted by its buoyancy in a stable condition. Ships will not float when the force of buoyancy cannot counteract the weight of the ship. Since mariners generally know the extent of cargo allowed and its proper stowage, the main concern with overloading becomes the loss of watertight integrity. The uncontrolled ingress of water into a vessel not only adds to the vessel's weight but may also induce instability.

Water can enter a vessel through a hull envelope rupture from the inside, through a penetration from the outside, as fire-fighting water, through piping failure, or from improper valve operation. The following list summarizes the possible hazards that may result in flooding:

1. Collision with another vessel, a fixed structure, or ice
2. Collision with an underwater object
3. Grounding
4. Military action
5. Explosion
6. Fire
7. Sabotage
8. Corroded piping components
9. Insufficient or defective closures
10. Insufficient training or instructions

FLOODING BOUNDARIES

Watertight integrity is maintained throughout the vessel by creating watertight boundaries. In addition to the hull structure, the weather boundaries consist of the scuttles, hatch covers, vent closures, doors, and portholes. Along the hull, the sea valves and side ports protect against water entry. Belowdecks, the boundaries are valves, piping components, bulkhead penetrations, and watertight doors. A failure of any of these boundaries has the potential to cause or contribute to flooding.

HATCH COVERS

Hatch covers are the primary boundary against flooding from the main deck. Most modern hatch covers are constructed of steel and may employ hydraulic opening and stowing devices. A seal between the hatch and hatch coaming is usually maintained by a resilient gasket and a locking mechanism.

Mechanical hatch covers, such as the folding or roll-stowing panel type, require more extensive maintenance than nonmechanical covers. The manufacturer should be able to furnish a manual that details the required maintenance items and maintenance intervals.

VENTS, VALVES, PUMPS, AND PIPING COMPONENTS

These items are not as likely to cause major flooding as leaking or failed hatch covers. However, due to the isolated nature of some of these components, small leakage could go undetected for long periods of time unless the components are periodically examined. Vents on deck can be checked to verify that the automatic or manual closure is functional. Pumps, sea valves, and piping are usually given a comprehensive survey at regular intervals and at drydocking.

BULKHEAD PENETRATIONS

Watertight bulkheads subdivide vessels into zones for controlling flooding. Bulkheads are frequently penetrated to allow the passage of piping or cabling. Bulkhead penetrations must be designed to maintain the watertight

integrity ofthe bulkhead. Piping is usually required to have metallic spool pieces with shutoff valves on both sides of the bulkheads. These valves should be periodically examined to ensure proper operation, particularly when corrosive fluids are carried such as seawater or sanitation waste.

Bulkhead penetrations for cable runs are usually made near the top of the bulkhead, where the water pressure from a flooded compartment will be lowest. Cables are routed through metallic openings in the bulkhead, and the annular space around the cable is sealed with a stuffing compound. The stuffing compound should be checked on both sides of the bulkhead to ensure it is intact.

WATERTIGHT DOORS

Watertight doors are divided into three classes. Class 1 doors are hinged, quick-acting doors, closed on both sides by means of a handle which brings the "dogs" tight against wedges. The seal is made by a resilient gasket seating against the door sill. Class 1 doors are used in high traffic areas, but are limited to bulkheads that do not form main flooding boundaries. The seal in a Class 1 door can be easily checked by a chalk test.

Class 2 and Class 3 sliding watertight doors are discussed extensively in *SaLAS* because they must maintain the integrity of main watertight bulkheads. Class 2 doors are hand-operated from either side ofthe bulkhead and must be capable of remote, manual closing from a station above the bulkhead deck. Hand operation is by a clockwise crank or a reciprocating crank. The seal will either be a soft metal, such as brass, or a resilient material.

Class 3 sliding watertight doors are power-operated from both sides of the bulkhead either by electric motor or hydraulic pump. A second remote, power-operated station capable of closing the door is required on the navigating bridge. Class 3 doors must also be capable of manual operation at the door location on either side ofthe bulkhead and from a remote station above the bulkhead deck. A mechanical indicator is installed between the door and the remote manual station. The mechanical indicator is usually a flexible cable that transmits the door position to a receiver indicator. The watertight seal is either soft metal or a resilient material.

Microlimit switches actuate indicator lights installed on the navigating bridge to show whether the doors are open or closed-red for "door open," green for "door closed," and flashing red for "intermediate position." The "close" limit switch is usually mounted on the leading edge at the upper corner of the door frame. It is actuated by closing of the door, which not only prevents the power or pump motor from continuously running after the door returns to the closed position, but also disconnects the red indicating light at the remote station. The "open" limit switch is usually mounted on the upper corner at the opposite side of the door frame. It is actuated when the door reaches the open position. It disconnects the motor and green indieating light.

The controls are so arranged that local operation cannot be overridden by remote operation. When the remote switch is left in the "close" position, the door can be opened locally, but will automatically close and the motor will stop when the local switch is released. Similarly, when the switch on one side of the door is left in the "close" position, the door may be opened from the other side, but will close automatically when the switch is released. Class 3 doors are also provided with an alarm that sounds at the door location when the door is opening or closing. Sound precedes door closing by five to ten seconds for personnel safety.

Hydraulic and electrohydraulic powered doors use a hydraulic ram to close and open the door. A valve block controls the flow of hydraulic oil into and out of a double-acting cylinder to move the shaft in the desired direction. The hydraulic oil supply is usually located at the remote manual station to make up the difference in demand of oil when opening or closing the door due to the difference in area of the two sides of the cylinder piston.

A means of operating the door through three open/close cycles is provided for emergency situations. Emergency power to the door may be by battery, centralized hydraulic accumulators, or an auxiliary oil supply positioned near the door to ensure operation of the door in the event the hydraulic fluid supply is damaged more than ten feet from the door. A typical watertight door is shown in figure 13-17.

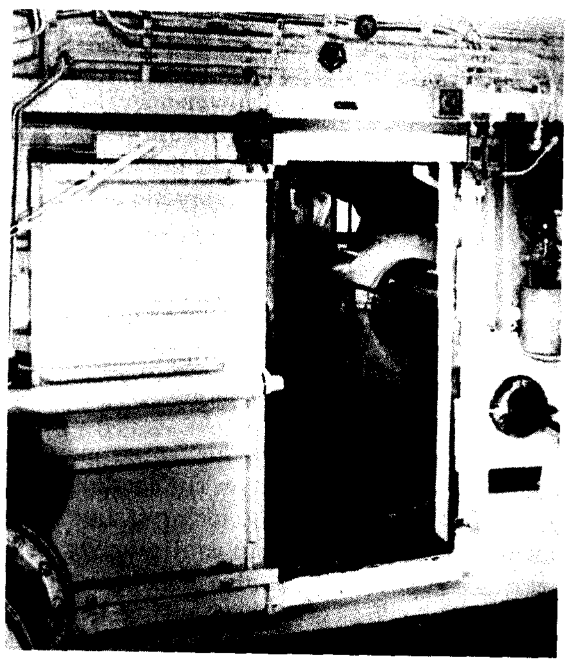


Figure 13-17. Watertight door. Courtesy Walz and Krenzer, Inc.

The door manufacturer should provide a maintenance schedule. The following information for an electrohydraulic door can serve as a guide in determining a maintenance schedule. Any maintenance or repair item not covered by the manufacturer's operations manual or maintenance schedule should be referred to the manufacturer or other qualified repair organization.

Weekly

1. Close doors individually and simultaneously while checking indicator lights at the navigating bridge station.
2. Mter operating the door, check the rod end of the cylinder for leakage.
3. Check hydraulic oil in the expansion tank while the door is open.
4. Check for hydraulic oil leakage while opening and closing the door using the local hand pump.
5. Check for hydraulic oil leakage while opening and closing the door using the remote manual pump.
6. Check for hydraulic oil leakage while opening and closing the door using the local motor pump.
7. Visually check for leakage between the valve blocks.

Monthly

1. Open the motor controller and check terminal connections, screws, and bolts and nuts for tightness.
2. Check the electric pump motor terminal connections, assembly screws, and bolts and nuts for tightness.
3. Check that the selector switch returns to neutral from the open position so the door cannot be locked open.
4. Open selector switch and check wire terminals for tightness.
5. Check limit switch actuator arms for freedom of movement.
6. Check the gasket for deterioration or damage. Conduct a chalk test of the sealing surfaces.
7. Check all rollers and pins for proper lubrication.

Semiannually

Check controller starter contacts visually for excessive pitting.

Annually

1. Check the clearance between the door and frame.
2. Check the condition of all electrical cable and hydraulic oil tubing for damage or deterioration.
3. Check the condition ofthe gasket between the frame and bulkhead, if not welded.

Fires, Explosions, and Fire Control Systems

Fires and explosions accounted for approximately 10 percent of all accidents reported for merchant vessels over 1,000 gross tons between 1978 and 1992. The same incidents caused approximately 24 percent (2,177) of the fatalities aboard merchant vessels over the same period, second only to foundering (2,805). Fires and explosions also resulted in the third highest number of vessels lost (600), after foundering (660) and wrecked (658).

GENERAL FIRE-FIGHTING RULES

In the United states, fires are classed into four types: A, B, C, and D. Class A fires occur in ordinary combustible material such as wood, paper, and cloth. Class B fires involve flammable liquids such as petroleum products, greases, and chemicals. Class C fires occur in electrical equipment from sparking or overheating. Less common are Class D fires, involving combustible metals such as potassium, sodium, and the magnesium found in high-performance aircraft. Class D fires are especially dangerous because, once ignited, they are very difficult to extinguish. Class D fires are a special danger to vessels with aluminum structures. European classification is different in that Class C fires involve gases from such sources as coal or fermenting sugar.

Obviously, a fire may begin as one class and quickly become another. An understanding of how fires begin and the type of fire being combated is essential to developing a fire-fighting strategy.

Fire Occurs whenever a given material is heated in the presence of oxygen to a temperature corresponding to the kindling point of the material. Three elements must be present iffire is to exist: material, oxygen, and sufficient heat. Education in fire fighting starts with this simple fact, because fires may be extinguished by removing anyone ofthese essential elements.

The above statement leads to three simple rules for fire fighting: (1) remove the material, (2) remove the oxygen, and (3) remove the heat.

Remove the material

Turning off the gas supply to a gas stove extinguishes the cooking fire. This is a simple illustration of how removing the material works. It is often the best solution to fires that emanate from sources under pressure. In some cases, a blanket of cooling water may be necessary to protect personnel attempting to reach a valve within the fire area.

Remove the oxygen

This technique can be demonstrated by covering a lit candle with a container. As soon as the candle consumes the oxygen in the container, the fire goes out.

Remove the heat

Removing the heat lowers the material's temperature to below its kindling point. In fact, many materials, such as shipboard cable and composite pipe, are designed to be self-extinguishing. That is, they will not burn unless directly impinged by fire.

Typical fire-fighting agents include liquid water, water fog, carbon-dioxide gas, foam, and powder. All fire-fighting agents act to remove one or more of the three factors of fires. For example, water is a remarkable cooling agent for removing the heat. Carbon-dioxide gas, being colder and denser than air, displaces the oxygen and starves the fire. Foam cuts off the material from oxygen with a blanketing effect. Powder functions in the same way and is the preferred agent for extinguishing fires in electrical equipment because powder is more easily removed than water or foam and does not cause electric shock.

Table 13-2 lists common sources of shipboard fires, their class, and recommended extinguishing agents.

PORTABLE EXTINGUISHERS

Portable fire-fighting extinguishers are often the first response to a fire in its initial stages. Portable extinguishers are available in a variety of sizes and fire-fighting agents. They should be arranged aboard ship according to the most likely class of fire for the area near the extinguisher. Table 13-3 lists the sizes and types of extinguishers according to the United States Coast Guard categorizations.

Maintenance on portable extinguishers can usually be performed locally. The manufacturer's recommendations should be followed closely. The U.S. Department of Transportation also requires portable extinguishers to periodically undergo hydrostatic pressure testing. Discharged extinguishers should be replaced or refilled without delay.

It is also important that portable extinguishers be properly secured to their mounting bracket. The larger classes on wheeled carts must be prevented from rolling. Lastly, a ship's fire control plan must accurately depict the location, size, and type of extinguishers distributed throughout the vessel.

Fire Main System

With the ongoing trend to reduce vessel manning, reliance on the fire main system has decreased. Fixed fire-extinguishing systems provide the first response in high risk areas such as machinery rooms, vehicle decks, galleys, and pumprooms. However, the fire main system remains the most reliable and versatile method of fighting fires. Water has the greatest heat absorbing capacity of any extinguishing medium. It can be sprayed as a straight stream for deep-seated fires or as a fog of small droplets when cooling and minimum agitation is desired. As it absorbs the heat of a fire, it may be vaporized to steam, which tends to smother the fire in an enclosed space. It can be used to cool firefighters approaching a fire or to beat back flames to gain entrance to a space. The quantity of water available for fire fighting is limited only by the capacity ofthe fire pumps. The disadvantage of water for combating fires aboard vessels is that excessive quantities may impair the vessel's stability.

TABLE 13-2
Common Shipboard Combustibles and Extinguishing Agents

Combustibles	*Class*	*Extent*	*Extinquishing Agents*
Woodwork, bedding, clothes, combustible stores	A	Small	1. Solid water stream 2. Water fog 3. C02 extinguisher 4. Foam 5. Powder extinguisher
		Large	1. Solid water stream 2. C02 fixed system 3. Water fog 4. Foam
Paints, greases, flammable stores, flammable fuels, and cargoes[1]	B	Small	1. C02 extinguisher 2. Water fog 3. Foam 4. Powder extinguisher
		Large	1. C02 fixed system 2.Water fog 3. Fixed sprinkler system 4. Water spray system
Electrical and communications equipment[2]	C	Small/Large	1. C02 extinguisher 2. Powder extinguisher
Combustible metals[3]	D	Small/Large	1. Solid water stream 2. Powder extinguisher

1 Closing the supply of combustible liquid either during or prior to fire-fighting efforts will aid in extinguishing the fire.

2 Disconnecting power to the affected equipment or area should be the first action taken.

3 Extreme caution should be exercised when using water on a combustible metal. A violent reaction or explosion can result. A solid stream of water is an effective cooling agent when cycled continuously onto and away from the metal. Powder extinguishers have also proven effective on some types of Class D fires. Special crew training or qualifications should be required whenever the vessel will be carrying cargoes with a potential for Class D fires.

SYSTEM PIPING, HYDRANTS, AND PUMPS

Fire main piping is designed to provide adequate fire-fighting water to all parts ofa ship. Numerous valves and cross-connections are included in the system to allow damaged sections to be isolated. Sufficient hydrants are installed to ensure each accessible space can be reached with two streams of water. Each hydrant station should have a hose and the tools, if any, re-

TABLE 13-3
Categories of Portable Extinguishers

Classification Type (Size)	*Soda-Acid and Water (Gallons)*	*Foam (Gallons)*	*Carbon Dioxide (Pounds)*	*Dry Chemical Powder (Pounds)*
A-II	2.5	2.5	-	
B-I	-	1.25	4	2
B-II	-	2.5	15	10
B-III	-	12	35	20
B-IV	-	20	50	30
B-V	-	40	100	50
C-I		-	4	2
C-II		-	15	10

quired to attach a fire hose. Hydrants have either 1Yz in. or 2112 in. discharge outlets.

Fire main piping should be protected from mechanical damage in areas where damage is possible. In cold climates, the fire main piping, valves, and hydrants should also be protected from freezing.

SOLAS requires each vessel to have at least one dedicated fire pump. Larger vessels are required to have two independently driven and physically separated fire pumps. Multiple-pump arrangements allow for separation of the pumps so that an accident in one space does not affect all pumps. Pumps and valve controls are arranged so that water can be made available as soon as possible after sounding an alarm. If the pumps are in a manned space, the controls will be located at the control station in that space. Ifthe pumps are located in a normally unmanned space, such as an automated engineroom, the controls will be located at the control station for a manned space.

Each pump should have a relief valve installed on its discharge side to prevent overpressure. The relief valve is usually set to open at a pressure 25 psi greater than the pressure needed to meet hose stream requirements. Besides protecting the fire main from excessive pressure, the relief valve setting prevents excessive reactions to personnel holding the nozzle.

Pumps may be used for other purposes that must be considered when the pumps and piping are sized. For example, fire pumps may be used in bilge/ballast service, foam systems, and sprinkler systems. Fire pumps must not have a connection to a system containing oil.

FIRE HOSES AND NOZZLES

Each fire main hydrant should have at least one length of hose adjacent to or attached to the station. It is critically important that the hose and hydrant are the same size and have the same threads. Fire hoses have a male and female fitting. These fittings are typically brass or aluminum. Aluminum fittings are not suitable for vessels in ocean service because of the tendency for aluminum to corrode in salt air. Mismatched hydrant and hose fitting materials may also lead to corrosion through galvanic action.

The female fitting attaches to the fire main hydrant. Prior to connecting a hose to a discharge outlet, the fitting should be checked to ensure its gasket is in place. Care should be taken not to cross thread the connection. The male fitting attaches to another length of hose or to a fire hose nozzle. Hoses should be uncoiled and pressure tested before being placed into service and at regular intervals thereafter.

Modern fire hose nozzles are designed with a variety of settings that allow the operator to control the flow of fire-fighting water. The adjustments are used to obtain the most effective spray pattern and the greatest degree of opera tor protection. All nozzles have controls to change the spray pattern from solid stream to high-velocity fog and low-velocity fog. Nozzles also incorporate a flush feature to remove debris in the flow passage. Some models allow the nozzle operator to adjust the flow rate, while others allow adjustments to the nozzle discharge pressure. Fire hose nozzles are manufactured in brass, aluminum, and plastic. The choice of material and nozzle size should be consistent with the fire hose fittings and the vessel's service. Most maintenance can be performed by the ship's crew. However, the manufacturer's recommendations should be checked before attempting to dismantle a nozzle. Figure 13-18 shows an Akron Brass nozzle.

FIXED CARBON DIOXIDE SYSTEMS

Carbon dioxide as an extinguishing agent has many desirable properties. It will not damage cargo or machinery and leaves no residue to be removed after a fire. Even if the ship is without power, a charged carbon dioxide system can be released. Since it is a gas, carbon dioxide will penetrate and spread to all parts of the protected space. It does not conduct electricity and therefore can be used on live electrical equipment. Carbon dioxide as an extinguishing agent has two disadvantages: (1) it has little cooling effect and (2) the quantity available is limited.

Carbon dioxide extinguishes fires by reducing the oxygen concentration to the point where the atmosphere will no longer support combustion. The gas concentration must be maintained for a sufficient period to allow the maximum temperature to be reduced below the kindling point. For this reason, all ventilation in the protected space must automatically stop when a carbon dioxide system is activated and all openings must be secured.

Carbon dioxide is most effective against flammable liquid fires. A reduction of the oxygen content of the air from the normal 21 percent by volume to 15 percent will extinguish most fires. In enclosed spaces, fires

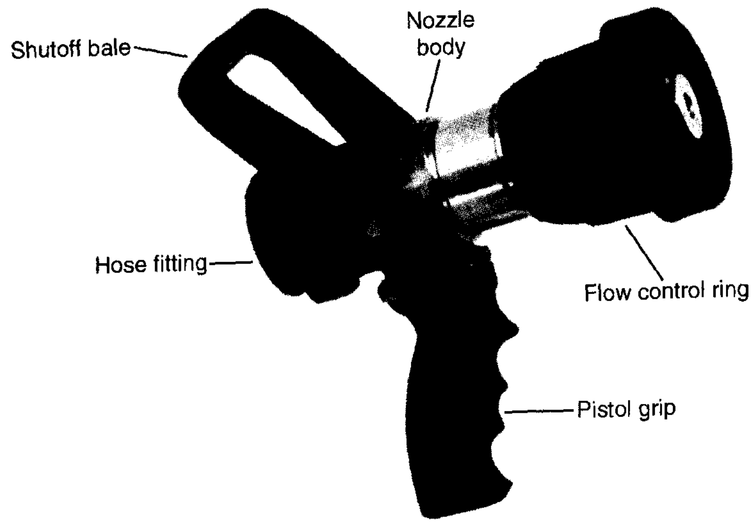


Figure 13-18. Akron style 3019 U.S. Coast Guard-approved nozzle. Courtesy Akron Brass Company.

.nvolving Class A combustibles may not be completely extinguished, but may be controlled for further fighting with water.

Manned spaces protected by a fixed carbon dioxide system have warning sirens and lights if the ambient noise level is high. The system will also contain a time delay device to allow personnel to escape the space before the gas is introduced. Personnel working in a space protected by a fixed carbon dioxide system must know the indications of the warning devices and must be aware of all exits.

Two types of fixed carbon dioxide systems are installed on ships-the "cargo system" and the "total flooding" system. The cargo system is intended for use in cargo holds where slow heat buildup is expected (Class A combustibles). Once openings to the hold are secured, gas is introduced into the hold until a sufficient concentration is obtained to bring the fire under control. The hold remains secured with additional gas being introduced to maintain the concentration until the vessel reaches a port where the hold can be opened, the cargo removed, and the fire completely extinguished.

Fires in machinery spaces are typically Class B fires. In this type of fire, the heat buildup is rapid. For this reason, it is important to introduce the extinguishing gas as quickly as possible. A quick response keeps structural members from reaching high temperatures, prevents heat updraft from carrying away the extinguishing gas, and limits damage to vital machinery. In a total flooding system, 85 percent of the gas will be released in two minutes.

A carbon dioxide system employs either high-pressure bottles or a large, low-pressure tank located outside the space being protected. The system will include remote controls that are usually cable operated. For this reason, attention should be paid to maintaining and lubricating the cable and pulleys to ensure the system will remain operable. Small diameter piping directs the gas to discharge nozzles in the protected space. The spacing of nozzles should be uniform, but it is not as important as in other fixed extinguishing systems because the gas will fill the space. However, it is important to consider that carbon dioxide is heavier than air and will tend to sink. Nozzle placement,Vato X the height ofthe space is usually adequate.

FOAM SYSTEMS

Mechanical foam-foam that is not a product of a chemical reaction-is produced by introducing a foam mixture in proper proportions into a flowing stream of water and aspirating with air. On ships, the foam is normally mixed by means of proportioning equipment near the foam storage container at some central location. The foam solution is pumped through fixed piping to foam nozzles or monitors. Air is introduced into the foam solution at the nozzle and the resulting foam is sprayed onto the area being protected.

Foam is ineffective on most Class A fires due to its inability to cover other than horizontal surfaces for long periods of time, its limited cooling properties, and its inability to penetrate to deep-seated fires.

Foam is most effective when sprayed on flammable liquid fires where a surface may be blanketed. Foam extinguishes a flammable liquid fire by forming a continuous layer over the burning liquid, separating the combustible vapors from the oxygen necessary for combustion. Foam also has cooling properties because it contains water dispersed in a very thin film. Foam may also be sprayed on flammable liquids that are not burning to prevent the escape of combustion gases and subsequent ignition.

Normal foam concentrate is either 3 percent or 6 percent liquid. The usual ratio offoam concentrate, water, and air to produce a foam with good fire-extinguishing properties is as follows:

3 percent-3 parts concentrate, 97 parts water, 900 parts air
6 percent-6 parts concentrate, 94 parts water, 900 parts air

Common foam is normally suitable for most flammable liquids. However, it is impossible for common foam to form a blanket on alcohols, esters, ketones, or ethers (called "water-soluble" or "polar" solvents). For these products, "polar solvent" foam concentrate is used. Polar solvent foams include a 3 percent catalyst that reacts with the polar solvents to strengthen

the foam where other foams are quickly broken down. The ratio of polar solvent foam is 20 percent concentrate-3 parts catalyst, 20 parts foam, 77 parts water, 900 parts air.

High-expansion foam, as its name implies, has a much higher expansion ratio than common foam. It is capable of producing vast volumes of foam with a small quantity of water. The expansion ratio for high-expansion foam is about one hundred times higher than for common foam. This characteristic makes high-expansion foam an option for large volume spaces where Class B fires are the most likely type and the quantity of carbon dioxide required would be excessive.

Aqueous film forming foam (AFFF), also known as "light water," is another type offoam agent. The chemical characteristics ofthis product enable water to float on top of petroleum products, thus producing a doubly effective blanket.

It is generally accepted that the effectiveness of a foam fire-extinguishing system in controlling a flammable liquid fire depends upon the rate at which foam is applied. For machinery spaces, where the foam will be contained, a value of .16 gpm/ft^2 of protected space is used. For deck areas where the foam will not be contained, the foam rate is the greater of .016 gpm/ft^2 of total tank area or .24 gpm/ft^2 of the deck area of the largest tank.

AUTOMATIC SPRINKLER SYSTEMS

Automatic sprinkler systems enjoy unprecedented support in the marine industry and IMO. As of October 1995, they are mandatory on all passenger vessels subject to SOLAS. A combination of characteristics makes sprinkler systems desirable over other types of fire-extinguishing systems in certain spaces: (1) water is an inexpensive and versatile extinguishing medium, (2) sprinkler systems cool the protected space and help control the fire, (3) sprinkler systems protect the overhead, and (4) sprinkler system heads act as both detectors and water distributors. Two of the disadvantages of sprinkler systems are that they are easily clogged by debris from water suctions and the cost of installation can be high. However, improved shipboard design standards and the expanding use of plastic pipe have helped decrease installation costs and made sprinkler systems more practical.

The sprinkler system concept is quite simple. When the sprinkler mechanism is activated, water is allowed to flow through the sprinkler nozzle. The solid stream of water is transformed into a spray pattern of droplets through a variety of methods. In the simplest method, water contacts a fixed deflector to form a spray. In a swirl-type nozzle, water is forced tangentially into the whirling chamber where the spinning motion causes it to exit in a horizontal spray. Vaned, spinning deflectors use centrifugal energy to create a spray parallel to the plane of the deflector. In all ofthese types of nozzles, the water is formed into ajet or sheet which is broken into small filaments by interaction with the air. Surface tension causes the filaments to contract and form droplets of various sizes.

Sprinklers suppress fire by using one or more ofthe following methods:

Surface cooling. Water spray applied to the burning surface cools the surface to a temperature below which it will not support combustion.

Smothering. The steam produced when water spray is applied to a high-intensity fire displaces the oxygen available to the fire.

Emulsification. Water spray agitates the surface of a nonmiscible flammable liquid causing the liquid to mix with the water. The liquid cools and releases fewer flammable vapors.

Dilution. Water spray dilutes water-soluble flammable liquids so that they become nonflammable.

Prewetting. Sprinkler water wets fuels in the protected area before they ignite, making subsequent ignition difficult.

Both wet and dry pipe sprinkler systems are commonly installed aboard ships. The wet pipe system has a quicker response and can be installed level because draining is not required. The dry pipe system is slower to respond and must be drained, but is protected from freezing. The normal components of a shipboard sprinkler system are a water pressure tank, piping, sprinkler heads, sprinkler system pump, pressure tank water pump, and pressure tank compressor.

The water pressure tank provides freshwater for system use under an air head. The pressure tank is supplemented by a pressure tank pump to maintain water in the tank and a compressor to maintain system air pressure. Both the pump and compressor are automatically activated during system use. The sprinkler pump also supplements the water pressure tank. It activates when system water usage cannot be compensated by the water pressure tank pump. Sprinkler pumps are treated much the same as fire main pumps. That is, when multiple pumps are required, they must be located in separate spaces.

The sprinkler system includes a number of sensors. A visual and audible alarm is located on the navigating bridge or other central safety station to indicate when the system is in operation and when a condition exists that would impair operation. Alarms are also installed to indicate problems with water tank levels, system air pressure, pump power supplies, control valve position, and water flow.

Sprinkler system piping has typically been metallic. With improved technology, plastic pipe is becoming more common. The advantages of)lastic pipe include weight savings, ease ofinstallation, and reduced main;enance and repair costs. However, plastic pipe itself is heat sensitive. Unler U.S. Coast Guard policy, plastic pipe must be installed behind a hermal barrier, and a smoke detection or other fire detection system must Ie installed in concealed spaces containing portions of plastic sprinkler ystem piping.

Sprinkler system nozzles are the key to system performance. The nozzles must sense heat, open the flow passage, and distribute fire-fighting water in a desirable pattern. Fusible element sprinklers and glass bulb sprinklers are the most common type of nozzle, although research continues on other designs.

Fusible element sprinklers use a metallic solder to actuate the sprinkler mechanism. The solder must be a carefully chosen material with certain characteristics. For example, the solder material must have a well-defined eutectic. Otherwise, it will melt slowly and only partially open the sprinkler, resulting in reduced system performance. Solder is usually an alloy consisting of antimony, bismuth, cadmium, indium, lead, silver, tin, or zinc. Since different combinations of metals can produce a range of results, fusible element sprinklers are classed using a range of melting points. A color code on the frame arm or deflector is also used to mark fusible element sprinkler nozzles for ease of identification (see table 13-4).

TABLE 13-4
Fusible Element Sprinkler Temperature Ratings

Temperature Rating (OF)	Classification	Color Code
135-170	Ordinary	Uncolored
175-225	Intermediate	White
250-300	High	Blue
325-375	Extra high	Red
400-475	Very extra high	Green
500-575	Ultra high	Orange
650	Ultra high	Orange

The solder melting point is not the only consideration. The salt air environment can cause surface oxidation on the solder, which can make the fusible element less sensitive to heat. The long-term strength of the solder must also be considered, because the fusible element must hold back relatively high water pressure. Lastly, some sprinklers maintained in high-temperature environments for long periods have experienced tin migration, that is, movement of tin atoms into the base metal. Some of the copper in the base metal has, in turn, been found to migrate into the solder alloy. The result is a new alloy with a higher melting temperature.

Glass bulb spinklers were developed as an answer to corrosive environment problems. A glass bulb containing liquid (usually glycerine) and air acts as the fusible element. The temperature rating is established by controlling the relative amounts of liquid and air in the bulb. The liquid and air are normally in equilibrium with a fixed vapor pressure. As the temperature increases, the liquid/air equilibrium is affected. The vapor pressure rises until the bulb shatters, actuating the sprinkler mechanism. The type of glass selected and the wall thickness also affect the temperature rating (see table 13-5). Glass bulb sprinklers are rated similarly to fusible element sprinklers; however, there are differences in the rating schemes because they were developed independently.

TABLE 13-5

Glass Bulb Sprinkler Temperature Ratings

Temperature Rating (°F)	*Color Code*
135-170	Orange or red
175-225	Yellow or green
250-300	Blue
325-375	Purple
400-475	Black
500-575	Black
650	Black

WATER MIST SYSTEMS

Water mist suppression systems are similar to sprinkler systems in that pressurized water is sprayed from overhead nozzles. Fire suppression is achieved through flame cooling, fuel cooling, prewetting, and oxygen depletion. Water mist systems, like sprinkler systems, are not effective on deep-seated Class A fires. However, unlike sprinkler systems, water mist is effective on Class B fires.

The most important difference between water mist systems and sprinkler systems is the size of the water droplets created. Water mist system droplets are much smaller than sprinkler system droplets. Descriptions of water mist systems have been standardized into three classes, based on the droplet size. The standard symbology is Dvf, where "Dv" is the droplet diameter and "f" is an established percentage. The three classes of systems are as follows:

Class 1, $DV_{0.90}$: 90 percent of the droplets produced have a diameter less than 200 microns.

Class 2, $DV_{0.90}$: 90 percent of the droplets produced have a diameter less than 400 microns.

Class 3, $DV_{0.99}$: 99 percent of the droplets produced have a diameter less than 1000 microns.

Water mist systems are also described by the pressure rating. Low-pressure systems operate up to 175 psi. Medium-pressure systems operate from 175 psi to 500 psi. High-pressure systems operate over 500 psi. Some system pressures are as high as 3,000 psi. A potential disadvantage of high-pressure systems is the need for large pumps or multiple pump configurations.

Small droplets are produced by a number of methods, depending on the system design. In very-high-pressure systems, water is forced through tiny nozzle openings where shear with the surrounding air creates small droplets. In other systems, an atomizing media, such as nitrogen, is injected into the water stream at the nozzles to shear the water into small droplets. Special deflector shapes are also used to create small droplets.

System design is similar to sprinkler system design in terms of the types of components used. A storage tank or a bank of storage cylinders supplies water to the system. The tank or cylinders operate under a pressure head of air or an inerting gas, such as nitrogen. The inerting gas may be used to supplement the water mist in enclosed spaces as a fire suppressant, or it may be used to help create the mist. A compressor and water supply pump keep the storage tank or cylinders pressurized. Water is distributed through nozzles installed in a piping network.

System performance depends on four factors: (1) droplet size, (2) spray momentum, (3) water distribution, and (4) flow rate. These factors are interdependent. That is, changing one factor affects the others. For example, spray momentum and water distribution are critical to ensuring droplets reach the fuel and are not carried away by hot swirling combustion gases. Since momentum is a function of mass and velocity, the droplet size and flow rate will dictate the spray momentum.

Water mist systems have a great weight savings advantage over common sprinkler systems. The higher surface-area-to-volume ratio means greater cooling capacity. Therefore, less water is required for the same degree of suppression. For example, a one-minute supply of water mist is the current standard for machinery and accommodation spaces. The flow rate is also lower, so smaller pipe is needed. Smaller pipe sizes lead to lower friction losses and reduced pump requirements.

SMOKE DETECTORS

Two types of smoke detectors are in common use aboard ships. In photoelectric detectors, air samples are passed between a photovoltaic cell and a light source. If the sample contains smoke, the photovoltaic cell will receive less light and will activate an audible alarm, indicator light, and/or an automatic extinguishing system. The detector may be equipped with a voltage regulator to compensate for variations in the ship's generator output, preventing false alarms due to changes in the intensity of the light source. A light intensity control may also be included to compensate for aging of the light source or dirt and dust that may reduce the amount of light reaching the photovoltaic cell.

In ionization smoke detectors, ionized air molecules are created and allowed to flow through the smoke chamber. The ionized molecules enable a drift current to be established between poles in the chamber. Smoke particles entering the chamber impede the drift of ionized air molecules, reducing the current and increasing the effective impedance of the chamber. When sufficient smoke is present, differences in the impedance of parts of the chamber cause voltage increases and thus increased current flow to the controlling elements of the detector. The current flow activates indicator lights and alarms.

FLAME DETECTORS

Most flames emit infrared radiation CIR). Also, any mass that contains stored heat emits IR. An infrared detector can give an almost instantaneous warning when fire develops. Also, the penetrating quality of the long IR wavelength allows flames to be detected through smoke.

IR detectors contain discriminating functions to prevent false alarms due to detection of non fire-related sources. Since a low-frequency flicker is associated with fire, a flicker detector circuit will make IR detection more reliable. However, shimmering reflections, flickering lights, or moving heat sources may still cause false alarms. The IR detector can be desensitized to this type of situation, but may then fail to detect small fires.

A significant characteristic of IR detectors is the ability to sense a specific and limited band oflR radiation in the wavelength range of 4.1 to 4.6 microns. This is an important advancement because all hydrocarbon fires produce radiation in this region. Although false alarms are still possible, they are far less extensive than in broadband IR detectors.

Ultraviolet (UV) light detectors are also used to sense flames. Since UV light is absorbed by most media, UV detectors are not as susceptible to false alarms as IR detectors. However, UV light may not be detected through smoke or the intensity may be reduced by a buildup of dirt or oil on the detector lens. Sources of false alarms in UV detectors include solar radiation, lightning, and arc welding.

Solar Radiation

The sun may produce background radiation that results in a detectable signal. Detectors can be desensitized to this "noise" by obtaining a high ratio between the threshold fire signal and the background radiation.

Lightning

Lightning is a remarkable source ofUV as the electrical arc ionizes a path to the ground or from cloud to cloud. Though powerful sources ofUV, lightning strikes are of short duration and can normally be compensated for by incorporating a time delay into the UV detector. However, this may limit the speed of detection of an actual fire.

Arc Welding

Arc welding is a rich source ofUV and is frequently present on ships, particularly in shipyards. Since welding flashes are longer in duration and

welding is a common source of shipboard fires, time delays may not be appropriate. The most common method used to prevent false alarms due to welding flashes is to disable the detection system and provide a fire watch.

FIRE ALARM SYSTEMS

Fire alarm systems have undergone a significant evolution since the early 1980s. The inevitable integration of microprocessors has made fire alarm systems simpler, more reliable, and more versatile. Where "sniffers" once drew air samples into a central detector, modern computer-controlled systems can fulfill a myriad of automated fire detection and control functions including the following:

- smoke detection
- flame detection
- heat detection
- fixed extinguishing system activation
- ventilation deactivation
- fire boundary closure
- activation oflow-Ievellighting
- addressable voice emergency instructions
- addressable alarm identification
- emergency alarm activation
- self diagnostics
- graphic location displays
- short circuit isolation
- paging of emergency personnel
- escape routing
- closed circuit television activation
- fault logging

OCCUPATIONAL HEALTH

Safe Space Entry

One of the most common shipboard hazards is entry into confined spaces. Spaces such as holds, tanks, voids, cofferdams, and pumprooms may contain dangerous accumulations of toxic or flammable gases. Spaces containing dangerous gases, spaces that have been closed for long periods, or those that contain certain types of cargoes may also be oxygen depleted.

MARINE CHEMISTS AND SPACE DESIGNATIONS

Individuals who possess a marine chemist certificate issued by the U.S. National Fire Protection Association (NFPA) are qualified to test shipboard spaces for entry or work. These individuals are able to qualify for certification only after completing rigorous on-the-job training and passing the certification exam. Marine chemists are employed by vessels and shipyards when the condition of a space must be determined before hot work, cold work, or personnel entry can take place.

Marine chemists test spaces for Oxygen content, flammable atmospheres, and toxic, corrosive, irritant, or fumigated atmospheres and residues. A space that is found to have less than 19.5 percent oxygen by volume will be labeled "not safe for workers." This means that the space may not be entered until it has been ventilated or the oxygen content has otherwise been raised to the minimum level. Conversely, a space that is found to have an oxygen content over 22 percent by volume-an oxygen enriched atmosphere-will be labeled' "not safe for workers-not safe for hot work." In this case, ventilation will be provided to lower the oxygen content to below 22 percent and above 19.5 percent. Workers can enter the space only for emergency reasons or for a short duration to install ventilation equipment provided the Oxygen content is continuously monitored and respiratory protection equipment is provided.

Spaces may also be tested for flammable gases. Spaces with an atmosphere containing over 10 percent of the lower explosive limit will be labeled "not safe for workers" and "not safe for hot work." Ventilation must be provided to lower the explosive limit to below 10 percent. The space may not be entered except for emergency reasons or to install ventilation provided that no ignition Sources are present, the atmosphere is continuously monitored, the atmosphere is maintained above the upper explosive limit, and respiratory protection equipment is provided.

Lastly, if a space contains an air concentration of toxins, corrosives, or irritants above the established permissible expOsure limit (PEL) or is immediately dangerous to life or health (IDLH), the space will be labeled "not safe for workers." Ventilation must be provided until the concentration is below the PEL or, in the case of contaminants for which there is no PEL, below the IDLH. The space may not be entered except for emergency reasons or to install ventilation provided that the atmosphere in the space is continuously monitored and respiratory protection and other appropriate protective equipment and clothing are provided.

TOXIC GASES

Modern ships may carry an enormous assortment of potentially toxic gases or liquids either in bulk or packaged form. Accumulation of toxic gases is normally controlled on shipboard by procedures for loading, unloading, and gauging, and with standards for cargo containment and requirements for ventilation. A failure in anyone of these systems could lead to formation of a toxic atmosphere, particularly in pumprooms and holds. Failure of pump seals or spillage in cargo holds, away from the natural ventilation of the weather decks, can cause a concentration of gases. Cargo fires may also

destroy a control system and cause toxic gases to be released or produced in the smoke.

The effects of toxic gases depend on the substance and the concentration. A permissible exposure limit (PEL) and threshold limit value (TLV) are determined for many toxic substances. The PEL and TLV refer to an airborne concentration of a product expressed in parts per million (ppm) by volume in air. These are the time weighted average concentrations believed to be safe for the average person during an eight-hour workday and forty-hour workweek for prolonged periods. For example, the PEL for chlorine is 0.5 ppm. Exposure at higher concentrations can cause coughing, choking, and burning eyes and throat. Exposure to vapor concentration of 1,000 ppm for ten minutes can cause death. Other substances, such as benzene, a known carcinogen, may have a higher PEL, but also serious long-term effects.

SaLAS requires ships carrying toxic cargoes to have protective equipment for each crewmember and toxic vapor detectors for the specific cargoes being shipped. In many cases, breathing masks with appropriate replaceable filters provide adequate protection from inhaling toxic vapors. With especially dangerous products, fully encapsulated suits may be necessary to enter an area of a toxic cargo release.

Cargo manifests, stowage plans, and cargo placards provide crucial information that should be consulted before entering any space that may contain or has contained toxic gases or liquids, or is adjacent to a space containing toxic gases or liquids. A number of publications, including the *Chemical Data Guide for Bulk Shipment by Water* published by the U.S. Coast Guard, are available to determine exposure hazards, exposure limits, and the type of protective equipment required. Several computer data bases, such as the Office of Hazardous Materials Technical Assistance Database (OHMTADS), supported by the Research and Special Projects Administration ofthe U.S. Department of Transportation, can also be accessed for chemical data.

FLAMMABLE GASES

The primary hazard of petroleum products is flammability. A number of tank vessel explosions and fires have been caused because a source ofignition was introduced into a space containing a high concentration of flammable gases. In many of the instances, welding or burning was being conducted. In other instances, a piece of equipment dropped into a tank caused a spark and subsequent explosion. Flammable gases have also been found to have a tendency to migrate through openings and pipelines, making them doubly susceptible to sources of ignition.

Inhalation of small concentrations of flammable gases has been known to cause mild euphoria. Larger concentrations or prolonged exposure can result in irritation to eyes and lungs, severe headaches and, eventually, unconsciousness and death. Skin exposure can cause symptoms ranging from rash and dryness to skin poisoning

LOW-OXYGEN ATMOSPHERES

Oxygen-deficient atmospheres are the most frequently encountered space entry hazard aboard ship. In a low-oxygen environment, the body's normal flow of oxygen is reversed so that the exhaled breath removes oxygen instead of carbon dioxide. The brain quickly starves and unconsciousness results. Death occurs rapidly through asphyxiation.

Oxygen is often removed from a space intentionally. Inert gases, such as nitrogen or combustion exhaust gas, are used to fill the ullage space in flammable cargo tanks during unloading when oxygen might otherwise be drawn into the tank, creating an explosion potential. If a leak develops in the inert gas system, a low-oxygen atmosphere can develop in any enclosed space that contains a portion of the system. It is vital that inert gas systems receive regular maintenance and operational tests to check for tightness, and tanks connected to the inert gas system must be checked for oxygen content before entry.

Perishable products are increasingly being preserved by the use of controlled atmosphere (CA) technology. In this process, the space containing the product is sealed and flooded with gas. The gas displaces oxygen and suspends respiration in the cells ofthe product, thereby delaying aging. Nitrogen and carbon dioxide are frequently used, although the gas may be customized to produce a desired characteristic in the product, such as coloring. The protected space may be as small as a single container or as large as an entire hold. Besides the danger of entry into the space, leakage during the introduction of gas or after sealing the space can result in an oxygen-deficient environment in adjacent spaces. There are currently no commonly accepted marking requirements for CA spaces or containers. Controls, such as locking entry points and posting warning signs, must be imposed on any space with a controlled atmosphere. Precautions must be taken when operating a CA system to ensure that workers in adjacent spaces are aware of the operation and that sealing procedures are adequate.

Low-oxygen environments can also result from oxygen depletion within the space. Coatings, paints, and steel (rust) can absorb enough oxygen to make the space hazardous when sealed for long periods of time. Organic bulk cargoes may absorb oxygen and give off carbon dioxide or carbon monoxide. Plant cargoes such as linseed cake, rosin, tobacco, and some fruits and vegetables have been found to have this characteristic. Coal also absorbs atmospheric oxygen and can produce carbon dioxide, methane, and carbon monoxide even at relatively low temperatures.

An oxygen level indicator with a remote probe should be used to test closed spaces and organic cargo spaces prior to entry. Ventilation should

also be provided even when the initial tests indicate an acceptable level of oxygen.

Breathing Equipment

SELF-CONTAINED BREATHING APPARATUS

Self-contained breathing apparatus (SCBA) are the most commonly used means of respiratory protection aboard modern ships. They are required by SOLAS for shipboard emergencies. SCBAs are portable, reliable, and provide for almost unrestricted movement. They have a long history of successful operation among shipboard, shoreside, and military firefighters and rescue personnel.

The SCBA shown in figure 13-19 consists of a facepiece, regulator, backpack with replaceable air supply, and a gauge and alarm assembly. The facepiece consists of a rubber headstrap harness and a wide-view, acrylic plastic lens. Straps allow the headstrap harness to be adjusted for a tight fit. The facepiece incorporates a flexible rubber seal for the face. However, facial hair can impair the seal and personnel expected to use an SCBA may need to be close shaven. The lens is treated with an abrasion-resistant coating on the outside and an antifog coating on the inside. A diaphragm is attached to the facepiece to allow communications.

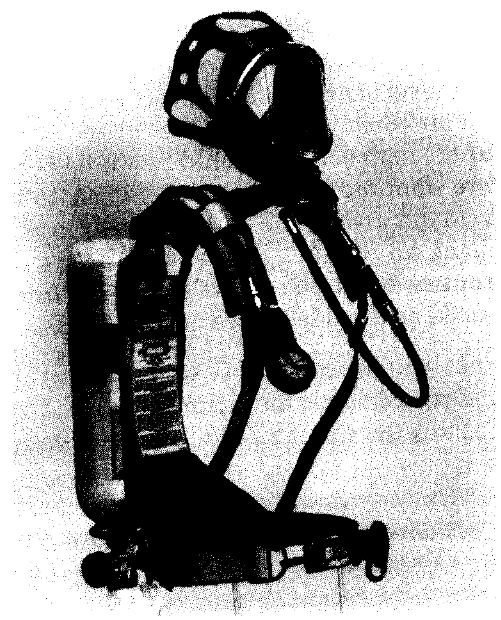


Figure 13-19. Self-contained breathing apparatus. Courtesy Survivair.

The regulator reduces the air cylinder supply of 2,000-4,500 psi to the system pressure of90-135 psi. The regulator also includes a relief valve to prevent overpressure. Two types of regulators are used on SCBAs: constant-pressure and pressure-demand. The constant-pressure regulator continually supplies air to the facepiece. The advantage of this type of regulator is its ability to prevent entry of harmful gases. The disadvantage is a decrease in the amount of operational time available. The pressure-demand regulator supplies air to the facepiece only when the wearer inhales. A second regulator may be provided for this purpose.

The backpack consists of the frame, harness, and air cylinder. Modern backpacks are designed to be lightweight and fully adjustable for the comfort of the wearer. The cylinder may be made of steel or it may have a lighter aluminum or composite liner with composite overwrap. Air cylinders are available in three standard classes of thirty, forty-five, and sixty minute durations.

The gauge/alarm is the wearer's method of determining remaining SCBA usage time. When the cylinder valve is opened, the gauge indicates cylinder pressure. The pressure will decrease as air in the cylinder is used. The alarm warns the wearer oflow air supply. The alarm may be activated by cylinder pressure to indicate a percentage of air supply remaining. Alternatively, the alarm may be a timer set by the wearer based on the rated duration ofthe cylinder. In either case, a safety margin must be considered to allow the wearer to exit the danger area before the cylinder empties completely. The safety margin is particularly important for timer alarms because different wearers consume air at different rates. The wearer's size, the temperature, the level of exertion required, and the level of excitement all affect the rate of air consumption.

The manufacturer's instructions should be studied and training should be conducted before donning or doffing an SCBA to ensure it is done properly and to avoid damage to the equipment. It is always best to have assistance when using an SCBA. The manufacturer will also provide instructions on the routine function tests of the SCBA components. The following list can be used as a guide for testing an SCBA:

Facepiece test:
1. Don and adjust the facepiece.
2. Block the air inlet with the palm of the hand.
3. Gently inhale. The facepiece should collapse and hold for a few seconds without leaking.
4. Exhale with the air inlet opening covered. The exhalation valve should not stick.

Leak test:
1. Close the regulator outlet valve.
2. Open the cylinder valve to fully pressurize the regulator.
3. Close the cylinder valve.
4. Observe the pressure gauge for fifteen seconds. Significant needle movement indicates a leak, and the SCBA should not be used.

Alarm test: (pressure type)
1. Open the cylinder valve to fully pressurize the SCBA.
2. Close the cylinder valve.
3. Close the regulator outlet valve.
4. Slightly open and close the bypass valve to incrementally move the needle toward the alarm point.
5. The alarm should sound when the needle reaches the set point on the gauge.

SCBA functions test:
1. Attach the regulator to a fully charged cylinder.
2. Close the regulator outlet valve.
3. Open the cylinder valve.
4. The cylinder valve gauge and SCBA gauge should read the same pressure.
5. Attach the regulator to the facepiece. A constant-pressure regulator will start air flow upon opening the regulator outlet valve. The pressure-demand regulator requires an inhalation to provide air flow. In either case, breathing should be normal, not be labored, and the air flow should be smooth.
6. The shutoff control, if any, should be operated to ensure it does not stick.
7. The bypass valve should be tested.

Most of the maintenance for an SCBA can be performed by the ship's crew. Classes are also commonly available from the manufacturer to teach the more complicated procedures for SCBA maintenance. Maintenance should never be performed by untrained personnel. A program for use, training, inspection, record keeping, and maintenance is described in the U.S. National Fire Protection Association Standard 1404, *Fire Department Self-Contained Breathing Apparatus Program*.

AIR_PURIFYING RESPIRATORS

Air-purifying respirators are used for work in environments that do not pose an immediate health hazard. They are also supplied aboard chemical carriers for escape purposes. Air-purifying respirators use filters or sorbents to remove harmful substances from the air. However, this type of breathing protection does not supply oxygen and may not be used in oxygen-deficient atmospheres or in environments that contain a concentration of toxins that is immediately dangerous to life and health.

Air-purifying respirators are comprised of a facepiece (with or without eye protection) and replaceable filter cartridges. A diaphragm in the face-piece facilitates communication. The facepiece includes a resilient seal to prevent leakage around the edges. Air-purifying respirators are generally easy to maintain by thorough washing, checking the condition of the seals, and ensuring the rest of the mask is not cracked or torn. Figure 13-20 shows a cartridge-mask-type air-purifying respirator.

Air-purifying cartridges are chosen for the particular atmosphere. Adherence to the limitations of the cartridge is crucial to wearer protection. Cartridge attachment differs among mask types. Therefore, it is also important to ensure the cartridge and mask are compatible. Mask and cartridge manufacturers can provide a list of the types of cartridges available. A list of commonly available cartridges is shown below.

Cartridge Types

(cartridges may also protect from combinations of hazards)

Organic vapors
Dusts and mists
Pesticides
Paint and lacquer mists
Chlorine, hydrogen chloride, hydrogen fluoride, and sulfur dioxide
Ammonia and methylamime
Formaldehyde
Radionuclides
Asbestos

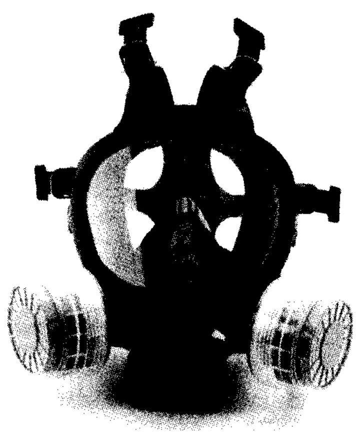


Figure 13-20. Cartridge-mask-type air-purifYing respirator. Courtesy 3M.

EMERGENCY ESCAPE BREATHING DEVICES (EEBD)

EEBDs are used expressly for escape from life threatening situations. They are commonly employed on chemical carriers and in large, high-risk shipboard spaces such as engine rooms. EEBDs are insufficient to provide protection for personnel rescue or work in hazardous areas.

Many types of EEBDs consist of a simple, transparent hood, pressure reducer, and an air cylinder. When needed, the EEBD is taken out of its case, the hood is placed over the head, the drawstring is made snug around the neck, and the regulator valve is opened. The small, high-pressure cylinder (around 3,000 psi) supplies air-usually a five-minute supply. Once an EEBD has been used, it should be checked promptly for damage, cleaned, and the cylinder recharged. Figure 13-21 shows an emergency escape breathing device.

Noise

Noise has been recognized as an important shipboard occupational health hazard. High noise levels interfere with speech intelligibility, internal shipboard communications, and the audibility of warning signals. Unlike workers in other industries who can retreat to a quiet environment after their shift, merchant seamen are part of a mobile "factory" and may not have the opportunity to retreat to a quieter, more relaxed atmosphere. As a

Figure 13-21. Emergency escape breathing device. Courtesy National Draeger, Inc.

result, in 1981, IMO adopted Resolution A.468, "Code on Noise Levels on Board Ships," as guidelines for shipboard noise control. The concepts of Resolution A.468 are contained the the U.S. Coast Guard Navigation and Vessel Inspection Circular (NVIC) 12-82, "Recommendations on the Control of Excessive Noise."

The effect of noise on hearing is a function of the actual noise level, its component frequencies, and the duration of exposure. An excessive combination of these elements results in a shift in a person's threshold of hearing, i.e., an elevation in the lowest level of sound detectable to the ear. A threshold shift may be recoverable to varying degrees, depending upon its magnitude, provided the person retreats to a quiet environment, generally accepted as below 75 dB(A), for a sufficient time. While small threshold shifts may be totally recovered, large shifts are only partially recoverable. Each occurrence leaves a small permanent threshold shift, known as hearing loss. The minimum goal of a shipboard noise control program should be to ensure that an exposure is not so great that the temporary threshold shift cannot be recovered during the following rest period. NVIC 12-82 establishes a $L_{eff}(24)$ as 82 dB(A). Noise exposures exceeding this limit should be controlled.

Following are some important definitions contained in NVIC 12-82 relating to noise control:

Twenty-four-hour effective exposure limit, $L_{eff}(24)$: The constant sound level that produces the same noise level as the actual time-varying noise over a twenty-four-hour period within the prescribed sound level limits. L_{eff} is based on a 5 dB exchange rate that assumes that personnel exposures to high noise levels are intermittent. In calculating this level, all noise less than 80 dB(A) may be disregarded. The approximate formula for L_{eff} is:

$$L_{\text{eff}} = 16.16 \log\left[\left(\frac{1}{T}\right)\sum_{i=1}^{n} 10^{L_{\text{ai}}/16.61}\Delta ti\right]$$

where

T = Total time interval, 24 hours
L_{ai} = A-weighted sound level during the ith time interval, Δti
Δti = ith time interval in hours

db(A): The sound pressure level in decibels weighted according to the A-weighting curve, as per ANSI S1.4-1971, "Specification for Sound Level Meters." The A-weighting values for octave bands 31.5 to 8,000 Hz are as follows:

Frequency (Hz)	31.5	64	125	250	500	1K	3K	4K	8K
A-weighting (dB)	−39	−26	−16	−8	−3	0	+1	+1	−1

Exchange rate: The amount of decrease in noise level that would allow doubling of the exposure time.

Impulse noise: Noise of less than one second's duration that occurs as an isolated event, or as one of a series of events with a repetition rate of less than fifteen times per minute.

Intermittent noise exposure: A daily personnel noise exposure during which the normally encountered noise exposure is interspersed with periods of low-level noise, i.e., below 80 dB(A), that are conducive to auditory rest. The recommended exposure duration limits for noise levels over 105 dB(A), when it is unavoidable, are:

Noise level [dB(A)]	106	107	108	109	110	111	112	113	114	115
Time (min.)	7.4	6.5	5.7	4.9	4.3	3.7	3.2	2.8	2.5	2.1

Sound pressure level: The level of sound pressure, L, measured on a logarithmic scale and given by the formula

$$L = 20 \log_{10}(p)/(p_0)\text{dB}$$

where

p = rms value of measured sound pressure

$p_0 = 2 \times 10^{-5} \frac{N}{m^2}$ (the reference level).

Steady noise: A sound where the level fluctuates through a total range of less than 5 db(A) as measured on the "slow" response of a sound level meter in one minute.

EXAMPLE 13-1: A noise exposure study is conducted on the first assistant on a VLCC. The study reveals the following exposure data during a routine 24-hour day:

1 hour at 95 dB(A)
5 hours at 93 dB(A)
2 hours at 88 dB(A)
2 hours at 85 dB(A)
12 hours at 77 dB(A)
Using the approximate formula for L_{eff}:

Interval	*dB(A)*	*Time (t)*	*10E(L_{ai}/16.61)*	*t*	*1/24*
1	95	1	5,241,238	5,241,238	21,839
2	93	5	397,225	1,986,125	82,755
3	88	2	198,615	397,231	16,551
4	85	2	131,038	262,077	10,920
				Total	132,065

Note: In accordance with the definition for $L_{eff}(24)$, noise levels below 80 dB(A) are disregarded.

$$L_{eff} = 16.61\log\ [132{,}065] = 85\ \text{db(A)}$$

The calculated noise exposure exceeds the limit of 82 dB(A) and noise controls should be imposed. The options (described thoroughly in NVIC 12-82 and Resolution A.468) are as follows:

Option 1. Require personnel to wear hearing protection in all machinery spaces where noise level exceeds 90 dB(A).

Option 2. Construct a soundproof booth around the operator's station in the engine room.

Option 3: Apply engineering controls at several key noise-emitting sources in the engine room.

ERM-ENGINE ROOM RESOURCE MANAGEMENT

The recent proliferation of international regulations, management codes and systems, and flag and port state controls for trading vessels described in this chapter, and in chapter 15, combined with awareness of potential environmental damage, have prompted a new and effective approach to the management of ship operations. For the navigation and deck operations of a vessel, this approach is termed Bridge Resource Management; for machinery spaces, it is called Engine Room Resource Management. The resources involved include personnel, equipment, spare parts, maintenance procedures, operational procedures, and communication equipment and methods.

The Need for ERM

There are numerous specific factors that have led to the development of ERM. The safety and environmental regulations promulgated by IMO and state or port authorities require a formal approach to achieve uniform compliance. Insurance companies concerned by claims are requiring better crew training and performance. Vessel accident and casualty histories are published widely and receive the attention of the shipping industry and government authorities. Training regulations such as STCW require a dedicated response. Complicated machinery systems require understanding and planning of operational, maintenance, and emergency procedures. There is a new awareness of the need to integrate shipboard personnel into effective teams for proper response to emergency conditions.

Statistics show that accidents are usually the result of a chain of errors including organizational, managerial, procedural, and individual or human factor. The chain of errors leading to an incident may include the following contributory factors: lack of knowledge, lack of training, absence of formal procedures, frequent interruptions, poor communications, lack of personnel assertiveness, need for integration and interaction of team activities, and poor awareness of situations. Experience has shown that these factors can be greatly reduced and the errors eliminated or ameliorated by the ERM approach.

Elements of ERM

ERM is a new way of thinking about the traditional tasks of the shipboard engineering personnel and their relationship with other personnel. Through better organization, it achieves more efficient use of the existing traditional assets. ERM involves human factors including behavior and the elements of reliable communication among all parties. ERM seeks to minimize risk of casualty through the reduction of human errors and the chain of such errors. The techniques for error control take into account situation awareness, planning, teamwork, human fatigue, and personnel competence. Where appropriate, ERM incorporates ISO 9000 quality management systems, state and international authorities such as IMO, and training codes such as STCW and the ISM Code.

The ERM approach is reinforced by recognition of the need for understanding and training, including the regular practice of the team approach, an attitude of mutual respect and loyalty among team members and shore management, and mutual vested interest in the safe, reliable, and efficient operation and maintenance of the vessel and the machinery systems.

ERM AND THE HUMAN FACTOR

Both seagoing and shoreside personnel of a shipping organization provide the resource and major means of implementing ERM. They are also the major source of error and consequent liability. The human factor is controlled through adequate education, continuous training, appropriate personnel attitudes fostered by good leadership, recognition of emotional needs, character building, and the physical and psychological well-being of personnel.

TRAINING

Continuous training is the most effective method of error prevention. ERM specifies training for normal plant operations; pollution prevention; emergency plant operations; use of hazardous materials; familiarization in mechanical, electrical, and control repairs; welding, cutting, and brazing operations; and safety in all aspects of ship operations.

Normal plant operations training includes considerations of manned and unmanned machinery spaces, underway operations, maneuvering operations, port operations, and the documented standing orders, policies, and procedures required to effectively carry out these operations. Regular drills in all aspects of normal plant operations support this training.

Pollution prevention training provides in-depth knowledge of safe bunkering operations, bilge pumping, sewage treatment, and the associated procedures and policies.

The documented procedures and training for emergency plant operations include the same considerations presented above for normal operations, with special emphasis on preparation for rapid response to the emergency through an effective drill program.

Hazardous material training requires careful assessment of the job requirements and comparisons with the technical capabilities of personnel, familiarization with aSsigned tasks, and attendance at specialized training facilities. Shipboard drills and procedures for the use of specialized equipment such as breathing apparatus, protective clothing, etc. are vital to effective training.

Training in repairs includes the proper use of tools, instruments, and apparatus, as well as an understanding of safety procedures for all electrical, mechanical, and welding operations.

Recognition and definition of training needs are obtained by observing the shipboard execution of drills and exercises, by reviewing reports and instructions for specific operations, by examining results of safety inspections, and by carefully evaluating accidents, incidents, and hazardous situations.

The ERM approach to training fulfills the requirements of management systems such as ISM and ISO 9002, i.e., training is accomplished with a written program and schedule, with a qualified instructor, and with proper means of training. Training may be done either on the job or through a distinct separate function.

TEAM MANAGEMENT

A team of shipboard engineering personnel is composed of a number of people working together in coordination to achieve a common goal. The goal may be defined as the safe, reliable, and efficient operation and maintenance of the ship's machinery systems.

Effective team management requires that the team be assembled with competent members who have received proper orientation. The team must have a leader. In the ERM case, this is the chief engineer or a task-assigned engineer officer. Each team member is an important asset and contributor to the overall effort. Each team member must understand the concept of the ERM and use a positive problematic approach to obtain solutions. Guidelines to problem solutions must be established in advance of the need. Management must encourage and reward personnel with positive reinforcement. In team work, errors are assessed promptly, root causes identified, and corrective action taken. The team performance is a basis for

the modification and update of policies and procedures found in the formal management system.

INFORMATION

ERM supports the critical flow of information. Adequate and current technical manuals must be available for all machinery and machinery systems. Good prints of systems and machinery should be available on the vessel in a well-organized setting. Clear and thorough standing orders issued by the master and chief engineer, as well as the management system policies and procedures, must be available and understood by personnel. Proper logbooks, operating records, and maintenance documents should be maintained and available.

SITUATION AWARENESS

Situation awareness in the ERM environment can be defined as an accurate perception of the factors affecting the engineering operating team during a specific period of time.

Situation awareness requires personnel to be proactive and to anticipate potential problems, acquire rapid and relevant information concerning a situation, develop quick recognition and evaluation of a problem situation, and interact effectively with the team. In particular, a team member must know the status of machinery operations and the system status regarding failures, the crew conditions during the given time, and the engineer's capabilities. A team member must understand the crew's familiarity with procedures and the machinery systems and have detailed knowledge of the hazards to the operating systems; he or she must understand and be able to communicate maneuvering requirements, and also anticipate crew fatigue.

Symptoms that indicate a degraded situational awareness among an engineering team and that require prompt action are ambiguity in monitoring equipment; distractions from the task at hand; feelings of uncertainty; disruptions in communications; abnormal departures from operating or maneuvering plans; violation of rules, orders, or operating procedures; improper monitoring rounds of spaces; and complacency or overconfidence.

Situational awareness provides the tool to break the error chain.

EXAMPLE SITUATIONS THAT ERM AVOIDS

In June of 1982, a steam-turbine-propelled tanker experienced a rupture of the nonmetallic expansion joint in the main low sea suction line. The engineer officer started the bilge pump and notified the chief engineer. Arriving in a rapidly flooding engine room, the chief engineer closed the main circulating pump's high sea suction valve and the main condenser overboard discharge valve. Nevertheless, flooding continued at an alarming rate. Both the chief engineer and the watch engineer officer failed to close the auxiliary condenser overboard discharge valve, allowing seawater to back-flow through the auxiliary cooling system to the ruptured expansion joint. The ship was near foundering.

Several of the problems in this case are addressed by ERM. An effective and reliable maintenance program probably would have prevented the expansionjoint failure. The lack of detailed knowledge ofthe seawater piping system was a significant part of the problem. Standing instructions for emergency relief of a watch could have reduced the confusion attending the situation after arrival of the chief engineer.

In December 1995, in a bay vessel with an unmanned engine room, a failed insulator separating a solenoid from its case caused a short circuit in the electrical starting system of the starboard engine. An electrical fire started, causing an alarm. The master failed to manually start the C02 system. Furthermore, there had been recurring problems with the starboard engine electrical starting system for a period of time.

ERM could have provided a solution to these problems. A recurring problem suggests lack of preventive action and inadequate maintenance. The lack of rapid response to the electrical fire is due to unstructured, inadequate, and ineffective training. Apparently the master and crew members were not familiar with the safety systems, in this case, the C02 extinguishing system.

In December 1996, an oceangoing freighter maneuvering near New Orleans experienced a loss of discharge pressure from the main engine lubricating oil pump. The backup pump failed to start automatically. The main engine control system shut down the main engine. The backup lube oil pump was started manually, but it was too late. The vessel struck a wharf, causing damage and injury to passengers during a subsequent evacuation.

From the ERM perspective, the ship was unprepared for maneuvering operations. The backup pump was unreliable, and the maintenance procedures were inadequate. The exchange of information among the engine room personnel and the bridge was lacking. A lack of training for emergency situations aggravated the problems.

JOB STRESS

The most common sources of job stress on an oceangoing vessel are boredom, frustration, and dissatisfaction. Unusual ship or environmental conditions and intense emergency conditions also cause stress.

Stress among personnel is an obstacle to good communication and safe operating practices. Not all personnel manage stress well. Stress contributes to the error chain.

In an individual, sources of stress that warrant management action include poor physical condition, lack of sleep, poor eating habits, alcohol, caffeine or drugs, and illness, as well as noise, heat, and vibration. Other causes of stress are personal difficulties, family distractions, and financial problems.

The alert manager or team leader will be aware of the signs of stress in individuals employed aboard ship. Difficulty in thinking, inattention to duties, slow reactions or poor coordination, tendency to skip proper procedures, chills, dull eyes, muscle aches, and slurred speech are all indications of stress.

Emergency Response

To respond to any engine room emergency, first it is necessary to identify the problem. This must be followed by a determination of the resources currently available to deal with the situation. Next, the team leader must plan and communicate the appropriate action. After the action is taken, it is monitored and modified if necessary.

In dealing with a crisis, all personnel must remain calm and controlled while taking prompt and correct team oriented actions that are fully communicated to the bridge.

Clear and effective communication among the engineering team, between the team and the bridge, and between the chief engineer and the master is an important aspect of ERM. Communication should be brief, clear, simple, and precise, providing only necessary information and avoiding overload.

EMERGENCY EVENTS

ERM training and planning is designed to respond to major emergency events including collision, grounding, main engine failure, electric power failure, rudder failure, hull damage, flooding, and fire.

In the event of a fire, ERM planning requires the following actions: notification through the alarm system; initial response in the vicinity of the fire; action to warn and save people; containing the fire initially using closures; action to fight the fire including activation of fire pumps, C02 systems, etc.; securing of ventilation systems; disconnection of electric power in the fire area; evacuating fire zones; and evaluating effectiveness of activities to control fire.

In the event of a main engine failure, ERM requires prompt notification of bridge and chief engineer. This is followed by planned use of standby equipment and proper use of electrical assets. Predetermined procedures for protecting main engines should be followed. The crew should identify and isolate the problems, commence actions to correct and repair problems, and evaluate procedures to determine if changes are required for long-term solution.

If a collision occurs, ERM response requires the watch officer to notify all engineer officers and take the following actions: employ watertight doors to isolate flooding; line up the pumps to bilge and cargo spaces suctions; take steps to isolate electric switchboards to ensure that power is available to operate dewatering pumps; prepare to take soundings oftanks and spaces to provide information on the extent of flooding.

In any emergency, ERM requires personnel to react promptly but to avoid overreaction: take the first measure to correct the situation but not to try to handle the situation alone; and inform the bridge and the chief engineer. At the conclusion of the emergency situation, the logs, records, and other documentation should be completed.

Personnel Interrelationship Responsibilities

The new STCW requirements promulgate ongoing training requirements among the engine room team. New crew members must be instructed in basics concerning communications, procedures, survival and operations. Pollution and safety guidelines must be understood and implemented.

Error Chain Review

In conclusion, Engine Room Resource Management is designed to use available resources effectively to break the error chain that may lead to ship emergency incidents or accidents. The factors in the error chain that ERM seeks to control include inexperience, personnel overload and fatigue, ambiguous or conflicting information, distractions or interruptions, poor or no communications, failure to follow procedures, loss of confidence, confusion, overconfidence, complacency, or loss of concentration. The use of ERM principles to monitor the situation, machinery, and personnel combined with overlapping duty assignments will help to control these factors.

SHIPBOARD COMPUTER APPLICATIONS

A combination of factors have significantly altered the way engineering officers handle shipboard operations: the accessibility of the modern personal computer, the ability of PCs to communicate with other computers via satellite links, and the availability of a profusion of marine-oriented software.

Chapter 14 contains a detailed description of an important computer-resident maintenance management system, including related software. In addition to maintenance records and projections, other computer applications-can be useful to the engineer officer, providing immediate access to survey records, ship's drawings, ERM files, or any of the following programs:

- cargo-loading programs
- management systems programs
- personnel and payroll software
- machinery operating information analysis programs

- training programs

Cargo-Loading Programs

As part of automating shipboard operation, a cargo-loading program can be used to perform all vessel loading calculations. Such programs are employed on tankers, containerships, bulkers, and multipurpose ships. A properly configured loading program will help to maximize the amount of cargo that can be safely carried; it will increase the cargo-loading efficiency with careful load sequence planning and improve the general productivity of shipping operations by quickly providing a multitude of error-free calculations.

A loading program is a tool that ship's personnel can use to confirm basic seaworthiness by rapidly calculating the following:

- draft, freeboard, trim, propeller immersion
- stability
- hull strength

The purpose and scope of a loading program is to perform basic loading-related computations such as ullage/volume conversions, API density calculations, and grain stability assessment. The program checks loading limits and ascertains that hull bending moments and shear forces do not exceed allowable values. The loading program must be consistent with specifications in the *Trim and Stability* booklet, *Loading Manual,* and *Grain Stability* booklet. In addition to performing the calculations, a loading program graphically illustrates the effect of a specific loading condition on the hull strength.

Classification societies made it a requirement for vessels of certain age, type, and condition categories to have an onboard cargo-loading computer program. Both IMO and flag state requirements specify their use. For example, the latest lACS *Bulk Carrier Rules* require all bulk carriers to have a loading program with special strength calculation options and load sequence planning. All new containerships and tankers are now required to have loading programs. In addition to meeting international regulations, a loading program provides economic payback to the ship venture.

Management Systems Programs

Management systems are the set of controlled and documented procedures that a company adopts for the rational operation of the business in conformance with standards such as ISO 9000 for quality, ISM for safety and pollution prevention, and ISO 14000 for environmental protection. IMO conventions require all shipping companies to meet the ISM code. In addition, some shipping companies are adopting the ISO 9000 and/or the ISO 14000 standard.

Many computer programs designed to support management systems are commercially available. These software systems are designed to assist the responsible person (referred to as the management representative, designated person, or safety officer) in developing, implementing, and operating a management system.

The programs provide model templates for the policy manual, procedures, and work instructions ofthe system as a starting point for the development of documentation specific to a given company, vessel type, and trade. Necessary training is prescribed and training records are maintained by the program. Control of the management system documentation is included. The scheduling, conducting, and reporting of internal audits and the management reviews of the system are provided by such programs. Processing and control of nonconformance and corrective-action reports are an important part of the system. As a payoff, an appropriate software package helps the company to create and maintain a computerized management system that is compliant with a specified standard.

Personnel and Payroll System

Computer-resident personnel and payroll systems allow vessels to efficiently access, track, and process personnel and payroll information. The objective of these systems is to streamline the processes by which crewmembers are signed on and off a vessel and subsequently paid.

A computer-based system typically contains a database of crewmember information that includes home address, medical history, licensing information, employment history, and training records. The database is used to electronically sign crewmembers on and off articles (either one at a time or through use of predefined joining lists), to reference crew information, and to generate crew lists and crew reports.

The medical and training records provide a convenient, easily updated source of required crew information. The ability to track crew credentials and training is essential to comply with the Standards of Training, Certification, and Watchkeeping for Seafarers (STCW) promulgated by IMO.

The payroll portion of such software systems allows the master to peri0dically record crew payroll information including time sheet entries and adjustments such as allotments, allowances, advances, reimbursements, and deductions.

The payroll information entered in the ship's computer may be processed on the ship or transferred to the company's shore-based computer. At the end of a voyage, the system prepares final pay vouchers, prints payroll or allotment checks, and produces a variety of payroll reports. Using an electronic format for personnel and payroll information provides an efficient data system that is easily updated and transferred ashore.

Machinery Operating Information Analysis

Computer software is available that permits periodic recording and analysis of main and auxiliary machinery vibration levels. Such recordings can be compared to previously recorded vibration levels to determine changes in the condition of machinery. It is also possible and desirable to compare actual machinery vibration levels to manufacturer-recommended levels for satisfactory operation.

Temperature recordings over time are also useful indicators ofthe condition of bearings and heat exchangers. In the latter case, it is possible to develop a program that continuously measures the effectiveness of shipboard heat exchangers and allows an economically justified time to be set for cleaning of the heat transfer surfaces. There are numerous opportunities to gather operating condition data such as temperatures, pressures, flows, etc. on main and auxiliary machinery. This data can be used for analysis to determine the condition of the machinery and the operating efficiency when compared to the manufacturers specifications.

On commercial vessels, it is possible and economically feasible to measure and record the tailshaft torque and thrust. This information can be analyzed and compared to past data to determine deterioration of propeller performance or increase in hull resistance due to fouling. Such information is very valuable for determining the most efficient schedule for the submerged hull cleaning afloat or in drydock.

Computer-Based Training Methods

Computer-based training schemes for all officer and rating categories have been developed for shipboard use. These programs provide instruction, examination, and record keeping for training as required by STCW.

Special Software

The opportunities for shipping companies to develop special and unique shipboard computer applications for their ships are limitless. For example, it is feasible to develop a special program that will determine a cost-optimized transit speed and course for a certain vessel in a specific trade given a large number of controlling variables such as wind conditions, wave conditions, loading condition, propeller condition, analysis of fuel, days out of drydock, sea temperature, air temperature, fuel rate versus power level, etc. The only limitation for the development of new computer software for shipboard application is the limitation of the ship managers and the software developers.

REVIEW

1. Describe the organizational structure of a typical shipping company.

2. What are the typical responsibilities of a port engineer?
3. What are the responsibilities of the engineer officer of the watch?
4. What are the chief engineer's responsibilities relative to maintenance?
5. What is preventive maintenance?
6. What methods may be used to monitor the operation and condition of machinery?
7. What is a statutory inspection or survey?
8. What functions are included in the management of spare parts?
9. What is the relationship between shipyard repair and classification and regulatory requirements?
10. How does a shipping company plan a budget for shipyard overhaul of a vessel?
11. What factors must be considered in selecting a shipyard for repair work?
12. Who prepares a ship repair specification? What are the sources of information?
13. What is a standard repair specification?
14. What are the major subdivisions of a ship repair specification?
15. What is determined during a preaward survey of a ship repair facility?
16. What are the relative advantages of a riding repair crew, repair by ship's personnel, and shipyard repair work?
17. What is a PERT schedule? A CPM schedule?
18. What factors are included in the estimation oflabor cost?
19. What organizations influence ship safety? How?
20. What information is needed to structure the shipboard safety organization?
21. What is the master's responsibility relative to ship safety?
22. What are the goals of safety management?
23. What is the difference between a reactive and a proactive safety management system?
24. What are the characteristics of an effective safety management system?
25. What are the major sources of ship flooding?
26. What are the classes of fires?
27. What are the three basic approaches to fire fighting?
28. Describe the fixed fire extinguishing systems found on board ships.
29. Foam fire systems are most effective on what type of fire?
30. What type of fire system is mandatory on all passenger vessels?
31. What is a smoke detector? Where are they used?
32. What is the function of a marine chemist?
33. How do you deal safely with a low-oxygen atmosphere?
34. What is an SCBA? When is it needed?
35. When is an air-purifying respirator used?

36. How are personnel protected from excessive noise levels aboard ship?
37. What is an emergency escape breathing device?
38. What is Engine Room Resource Management?
39. Why is ERM needed?
40. What do we mean by shipboard team management?
41. What are the major engineering department emergency events?
42. What is the error chain?
43. What are the error chain factors?
44. How is the error chain controlled?
45. What is a management system?
46. Name five computer software systems that are useful to the engineer officer.
47. What is the purpose of a cargo-loading system?
48. What main propulsion information would you collect for analysis on a diesel vessel? Steam vessel?
49. From your own experience suggest a potential shipboard computer application.

REFERENCES

Akron Brass Company. 1995. *Industrial Turbojet Product Catalog.* Wooster, Ohio: Akron Brass Company.

American Bureau of Shipping (ABS). 1995. *Rules for Building and Classing Steel Vessels.* New York: ABS.

Consilium Marine AB. 1993. *Salwico CS3000 Product Catalog.* Stockholm: Consilium Marine AB.

Hassani, J. August 1987. "Shipyard Facilities-New and Old Closures for Dry Docks." *Ship Production* 3, no. 3.

Hemond, Ron, and Roger Wendt. July 1983. "The Technology of Infrared and Ultraviolet Detection." *Fire Journal.*

Hersey, Pulli, and Taylor. 1993. "Fire Safety Analysis of Foreign and Domestic Cargo Ships." U.S. Coast Guard Academy report.

Hunt, Everett C., and Boris S. Butman. 1995. *Marine Engineering Economics and Cost Analysis.* Centreville, Md.: Cornell Maritime Press.

International Labour Organization (ILO). 1990. *Inspection of Labour Conditions On Board Ship: Guide-Lines for Procedure.* Geneva: ILO.

International Maritime Organization (IMO). 1992. *International Convention for the Safety of Life at Sea (SOLAS).* London: IMO.

---1993. *International Convention on Standards of Training, Certification and Watchkeeping for Seafarers (STCW).* London: IMO.

Lavery, H. 1. 1990. *Shipboard Operations.* Oxford: Heineman Newnes.

Mawhinney, J.R. May/June 1994. "Water Mist Suppression Systems May Solve an Array of Fire Protection Problems." *NFPA Journal.*

Meredith, Jack R., and Samuel J. Mantel Jr. 1995. *Project Management.* New York: John Wiley & Sons.

National Draeger, Inc. 1989. *National Draeger MAX Selfcontained Escape Breathing Apparatus Product Catalog.* Pittsburgh: National Draeger, Inc.

Survivair. 1992. *Self-Contained Breathing Apparatus (SCBA) Operation Manual.* Santa Ana, Calif.: Survivair.

United States Coast Guard. 1994. "Guidance Regarding Voluntary Compliance with the International Management Code for Safe Operation of Ships and for Pollution Control." Navigation and Vessel Inspection Circular 2-94. Washington: U.S. Coast Guard.

---1972. "Guide to Fixed Firefighting Equipment Aboard Merchant Vessels." Navigation and Vessel Inspection Circular 6-72. Washington: U.S. Coast Guard.

--- 1994. *Marine Safety Manual,* vol 2. (Commandant Instruction M16000.7) Washington: U. S. Coast Guard.

---1982. "Recommendation on Control of Excessive Noise." Navigation and Vessel Inspection Circular 12-82. Washington: U.S. Coast Guard.

United States Department of Transportation, Maritime Administration. *1994 Administrative Guide: Conducting Pre-Award Surveys.* Washington: U.s. Department of Transportation.

--- 1995. *Outlook for the U.S. Shipbuilding and Repair Industry.* Washington: U.S. Department of Transportation.

---*1995. Report on Survey of U.S. Shipbuilding and Repair Facilities.* Washington: U.s. Department of Transportation.

United States Government. 1994. Title 46 Code of Federal Regulations. Washington: Office of the Federal Register.

--1994. Title 29 Code of Federal Regulations, Part 1910. Washington: Office of the Federal Register.

United States Navy, Military Sealift Command. 1992. *Engineering Operations and Maintenance Manual.* Washington: U.S. Navy.

Walz and Krenzer, Inc. 1993. *Sliding Watertight Door Maintenance and Operations Manual.* Rochester, N.Y.: Walz and Krenzer, Inc.

Wilson, Lt. Cmdr. Marc B. 1992. "Human Factors: An Initiative in the United States Coast Guard." Paper presented at Ship Production symposium, New Orleans. Society of Architects and Marine Engineers (SNAME).

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CHAPTER 14

Computerized Maintenance Management

EUGENE D. STORY AND DONALD A. DAILEY

INTRODUCTION

The maritime industry is fast moving into an era when virtually all aspects of vessel operation, from inventory and maintenance management to personnel and payroll processing, will be organized electronically, using computer databases to facilitate data entry and retrieval. New operational requirements that have come into effect under agencies such as the IMO are furthering the need to move in this direction. Because of the importance of maintaining the vessel's operational status, a primary consideration is some form of equipment maintenance application.

An organized maintenance management system is now considered essential to ship management operation. The goal ofthis system is to organize, standardize, and simplify the access to all inventory and maintenance information, on both ship and shore, in order to make vessel maintenance more efficient and to document vessel maintenance. The process starts with identifying all the equipment of the ship, including the hull structure, so that a determination can be made as to what actions must be taken to maintain the ship in a safe and efficient manner. This will include determining the planned maintenance that must be performed on a periodic or running-hour basis, as well as maintaining the parts, materials, and tools necessary to perform the maintenance actions. Key to these systems is that they standardize information and procedures for a single vessel or an entire fleet.

The system should provide the means by which the engineers may analyze the performance of a vessel and identify problem areas, and it must facilitate the maintenance and repair of equipment that has failed. In the more advanced systems, the operation of the equipment is monitored and predictive techniques are used to identify potential trouble spots before they become serious problems.

While the first inventory and maintenance systems were implemented using manual methods such as scheduling boards and handwritten reporting, this is no longer a practical solution for multivessel fleets.

Although a few companies have attempted to develop their own computerized maintenance management system, the most common approach, even for the largest fleet operators, is to purchase existing software systems. These may be implemented by the company's Own engineering staff or by outside professionals.

The process of implementing a computerized inventory and maintenance system begins with the following steps:

1. Plan a system.
2. Review the system hardware issues.
3. Consider the system applications.
4. Plan the database development and implementation.
5. Establish a data communications link.

The modern computer-based maintenance management system, along with the communications vehicle to transmit data, is now standard on most modern ship operations. This chapter will review the elements of such a system, including the actions necessary to implement them.

PLANNING A SYSTEM

When planning to implement a computerized inventory and maintenance system, the first step is to analyze the company's needs. Consider the requirements of your operation and the goals for the system (for example, to reduce surplus inventory or to standardize maintenance procedures across a fleet), and then consider the elements required to meet those goals. Consider also the company's current procedures, such as the way inventory control and planned maintenance are currently handled, and how those procedures can be improved.

If a system can be adapted to the way a company operates, it will gain acceptance more easily. If it is a logical extension of the engineer's day-to-day operation and saves time by organizing shipboard routines, then it is more likely to be accepted by the engineers. On the other hand, transferring an inefficient system to a computer will not yield the desired results. For this reason, a thorough review and prioritization of the current needs will help create the best potential for success.

System planning should also identify what data will be collected and what will be done with that data, including such things as interfacing with shore-based systems (for example, a purchasing system for requisitions).

Strategic Information Technology (IT) Decisions

For computer applications in the maritime industry, the emphasis is shifting away from the consideration of the single system and moving toward consolidated systems capable of sharing information and providing a single repository of all vessel data that can be accessed by both ship and shore. There is an increasing need for this type of data consolidation, particularly in view of newer maritime regulations that necessitate an efficient means of documenting various aspects of vessel operation. Once this need is recognized, the challenge is to find a computer system capable of keeping pace with the constantly changing information technology.

For many, the process of selecting and installing a computer application is dwarfed by the need to understand the implications of the rapidly changing information technology industry on maritime organizations. Just as ship design changed to reflect current technology, so the computer industry continues to change dramatically. When implementing a new system such as a maintenance and inventory system, the challenge is to take advantage of what IT has to offer today without inhibiting our ability to capitalize practically on technologies in the future. The only way to avoid insupportable systems is to make tactical IT decisions in light of a larger strategic IT plan.

Certain technologies have emerged to ensure that systems built today will not become obsolete quickly. It is important that the individuals responsible for integrating new systems understand these concepts and that the systems implemented take advantage of these technologies, which include the following components.

ENTERPRISE-WIDE COMPUTING MODEL

An enterprise-wide computing model seeks to model the business for both data and process. It forces evaluation and definition of work flows and has the potential to eliminate redundancies and help ensure consistency of process. In the maritime application, among other things, this translates into standardized fleetwide coding structures and the development of a framework of historical data for management decision making.

LAYERED ARCHITECTURE

With layered architecture, separate components can be changed as necessary. In the past, each system was built with a number of components, including the user interface, the business logic, communication services, the database, and the operating system or IT platform services. When a new system was added, some of its components may have been redundant or its design inconsistent with the legacy system. By extracting the common components and grouping them into layers, systems are now built layer by layer. In this economic architecture, individual layer components can be upgraded or replaced as technology changes require.

CLIENT-SERVER ARCHITECTURE

In a client-server type architecture, systems are separated into multiple components, and computer services are provided by "servers" to multiple users, or "clients." In this environment, systems can be designed so that processing may be done at multiple locations, thereby optimizing hardware resources. By partitioning systems in a client-server environment, systems can be developed that allow changes with minimal impact on the overall system.

OPEN ARCHITECTURE

Systems designed using. an open architecture model include clearly defined interfaces to allow data to be shared across different systems. This technology is manifested in the use of APIs, or application programming interfaces. This architecture is of crucial importance when selecting service layers in the layered architecture. In maritime systems, it is this architecture that allows for the flow of data from, for example, a shipboard requisition system to a shore-based purchasing system.

DISTRIBUTED SYSTEMS

In a distributed system, components can reside in a variety of places; this is called "locational independence." This means that a given user-application does not care where individual components reside or are executed, and the system includes an infrastructure to take care of issues related to where things are, both logically and physically. The Internet is an example of a distributed system.

COMMON OPERATING ENVIRONMENTS

Ifwe describe these next-generation systems as a ring to which all new and legacy systems can be attached, and all systems interact with each other, then that describes a common operating environment (COE). In this architecture, systems are designed and built around a common technology system. The benefits provided by COE are the same as those for all technologies discussed here- increasing productivity and reducing redundancies in an effort to meet the increasingly complex demands of IT systems today and tomorrow.

THE SIGNIFICANCE OF IT

Using these technologies, maritime organizations can deploy IT resources in the manner most likely to evolve and grow with the organization, and most successfully to meet business requirements. Systems implemented without regard to these resources run the risk of becoming orphaned as a result of changing technologies.

Regulations for Systems

While some classification societies consider a planned maintenance system as a part of the criteria for granting a safety environmental protection certificate or preventative maintenance system certification, there are no other current class society requirements to have such a system on board the vessel. However, the installation of a shipboard maintenance system remains a recommended procedure by most regulatory agencies, and the impetus for implementation of these systems has increased significantly in recent years.

The growing emphasis on ship safety and protection for the environment, along with the broadly impacting regulations passed down from the International Maritime Organization, now make the implementation of a computerized shipboard maintenance system a practical necessity for vessel operations.

One of these regulations is the International Management Code for the Safe Operation of Ships and for Pollution Prevention (ISM). The ISM Code seeks to ensure ship and personnel safety, in part through improved management and documentation of vessel maintenance (see page 15-49). While there is no current requirement that the system be computerized, this is the only practical way to deal with the volume of data and to make that data economically available to the different regulatory agencies. The question of enforcement will likely move shipowners toward a computerized solution, since the documentation for class and regulatory authorities could be generated easily in an electronic format, thereby saving considerable labor and expense and ensuring complete data.

Maintenance Management Functional Elements

The requirements of a computerized inventory and maintenance system will vary depending upon the specific goals for the system. At a minimum, the system should include the following functional elements of maintenance management.

1. Equipment file: Includes technical specifications and other detailed information on all shipboard equipment to be maintained.
2. Spare parts file: Includes technical specifications and inventory status on all spare parts belonging to the shipboard equipment.
3. Maintenance procedures file: Lists maintenance procedures for all shipboard equipment based on manufacturers recommendations, equipment history, and company policy. These procedures are used to generate maintenance that is due based on calendar intervals or running hours.
4. Equipment history file: A record of all planned and corrective maintenance performed, including the action taken and the parts and manpower used.
5. Shore communications: Ability to communicate efficiently with the shore office to send and retrieve updated inventory and maintenance information.

Maintenance Management System Elements

System elements are composed of the following items and processes.

1. System hardware: The equipment necessary to perform the functions of a computerized maintenance management system. The vast majority of shipboard systems operate on personal computers (PCs).
2. Applications software: The software required to perform the various maintenance management functions. These are usually provided as modules that allow a phased implementation if necessary.
3. Databases: The maintenance systems are only as good as the data that is collected and entered before start-up, as well as the data maintained during the operation of the system. The thoughtful organization of the data at the time of implementation is critical to creating a system that is easy to use. This includes the methods to keep the shipboard and shoreside databases in synchronization. A critical element of database development is the shipboard validation of the data.
4. System installation and start-up: Shipboard systems have some unique installation issues compared to shore-based plants. While the most obvious time to set up a system is when a new ship is being built, many new installations today are installed on operating ships. This means the installation and start-up must take place primarily while the ship is at sea. Start-up is usually done with the assistance of the system professionals who must have a thorough knowledge of both the ship's equipment and the computer system's hardware and software.
5. System maintenance and operator training: One of the largest causes of system problems is the lack of proper system maintenance and operator training. The computer hardware and software requires the same level of maintenance as any other system on the ship. In addition, with crew rotations and turnover, it is essential to maintain an ongoing training program for new people who will operate the system.
6. Data communications: Modern maintenance management is a joint effort between the shipboard engineer officers and the shore support offices. Parts requisitioning and requests for technical assistance are normally channeled through the shore-based engineering support office. This requires that the shore-based engineers have the same systems and data available to them as the engineer officers. With modern data communications over satellite, it is possible to economically communicate the data so that both computer systems are kept in synchronization.

SYSTEM HARDWARE

The computer system hardware components selected for use in any marine application play an important role in the overall functionality and performance of the system(s). This includes not only the PCs themselves, but all related peripherals such as printers, modems, tape backups, line conditioner/battery backup units, and bar-code readers.

Shipboard computers present a unique maintenance problem-that of system support. Remote support of equipment is usually through marine system suppliers who provide worldwide exchange/replacement of equipment rather than on-site service. When planning the implementation of an inventory and maintenance system, it is imperative that the plan include provisions for remote system support.

Suggested Hardware Configuration

Most maintenance management systems should run adequately on a configuration similar to the one described on the following pages. With fast changing computer technology, these minimum requirements may become obsolete in a short period of time.

System unit: A Windows 95-compatible personal computer with a 1G hard disk drive, CD ROM, diskette drive, parallel port, and keyboard. Most systems today are installed with 3.1G hard drives, although the average user workstation will not require that capacity. Installing a high-speed computer is advantageous since most maintenance management programs will do a significant amount of sorting of large databases.

Monitor: Generally, any monochrome monitor will display the desired information; however, some maintenance systems come with graphic options to display equipment and spare part images, and therefore a high resolution Super VGA color monitor may be desired for the clearest part identification and easiest readability. This is also an office standard.

Printer: Any dot matrix printer with cable will print the available system reports; however, a laser printer will provide letter quality reports for office submission and is more of an office standard.

Backup unit: If used regularly, a reliable tape backup unit provides an indispensable asset in terms of protecting large databases from hard-drive failures or other unforeseen events.

Optional Hardware

In addition to the basic configuration, the following enhancements may be added to the system.

Local area network systems (LAN): A local area network allows multiple personal computers to be linked to a central file server and allows several users to access the systems simultaneously. A client-server architecture will provide the most efficient environment for database management. The LAN system should provide full "file and record-locking" control. Novell and NT are popular examples of network operating systems.

Modem: A high-speed modem (9600 baud) is required for efficient analog data communication between ship and shore. Higher baud rate modems are available, but 9600 is standard for current INMARSAT "A" satellite transmissions. Note that modems are not required when using digital satellite services such as INMARSAT "B."

Bar-code reader: Some maintenance management systems provide an option to link to a bar-code reader in order to make electronic inventory adjustments. Since the bar-code reader has to interface with the system, the choice of readers will be largely determined by the software that is being run. Ideally, the bar-code reader should be portable, light, and able to retain at least one thousand records.

Line conditioner and ,surge suppressor: This device protects the computer from electrical surges and fluctuations commonly found in a shipboard environment.

Uninterrupted power supply (UPS): An uninterrupted power supply will ensure a constant power source in the event of a power shortage. This unit is especially important in a shipboard environment where nO-volt power may be inconsistent or occasionally intermittent.

Computer System Location

Locating the computers is an important decision concerning system implementation. Modern maintenance management systems normally have networked computer terminals in the captain's office,the chief engineer's office, the engineering office,and the machinery control room. Providing adequate access to the computers is a key point in securing crew acceptance of the system. Avoid placing the computers near primary power transformers, and always ensure that the computer hardware is secured appropriately for the sea conditions in which the vessel will be operating. Vibration dampening pads will also prolong the life of shipboard computers.

SYSTEM APPLICATIONS

A comprehensive computerized maintenance system should contain modules for organizing equipment, inventory, requisitions, and maintenance information as illustrated here in figure 14-1.

This system allows for the exchange of data between various modules (equipment, inventory, maintenance, etc.), each one maintaining a database of information. In this illustration, the primary database is in the equipment module, which contains the technical equipment information; the other modules maintain a database of supporting information.

A well-designed maintenance system should be flexible, allowing the scope of the system to match user requirements. It should also support an interface to existing systems (such as a purchasing system).

Computer applications are primarily task oriented. When selecting a computerized inventory and maintenance program, ensure that the func-

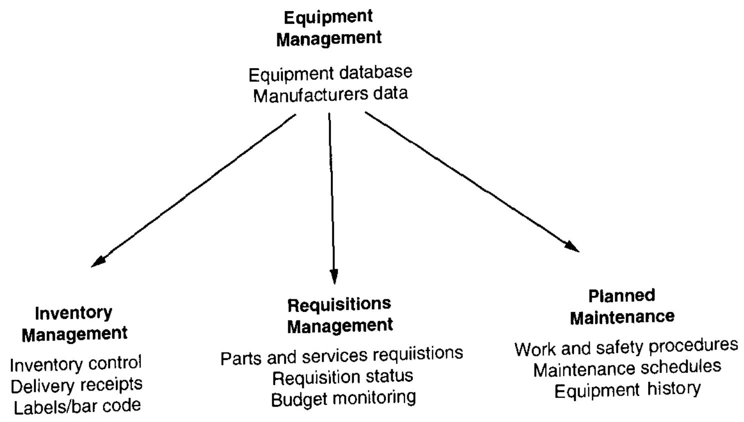


Figure 14-1. Comprehensive computerized maintenance system. Courtesy Marine Management Systems, Inc.

tionality of the program is consistent with the goals for the company's proposed maintenance system. Before any of these modules can be utilized on the vessel, there should be a planned strategy of database development. This process consists of taking information about the vessel equipment and maintenance in its current form and entering it into the system.

Database development is a crucial step in obtaining a successful maintenance management system. It begins with choosing the scope of information to be contained in the system and establishing a coding scheme so that the equipment information will be logically organized and therefore easy to find and maintain. These considerations, along with collecting and organizing the raw data (i.e., equipment, spare part, and maintenance information), are part of the preliminary engineering services. These steps are done one time for each class of vessels.

Next, the system is physically installed on the ships, the database is completed, and a shipboard inventory is conducted to verify the information in the system for each vessel. This is the shipboard engineering and validation step. The validation process is a time intensive one, and requires good organization and planning to ensure that it goes smoothly.

Further information on database concepts can be found later in this chapter in the section on database development.

Equipment Management

The starting point of any maintenance management system is the identification of all the equipment that must be maintained. The equipment database should catalog and maintain all information on each vessel's equipment in an efficient and organized manner.

This database should contain the complete information for all onboard machinery, including technical information, the manufacturers data, a description of the equipment, serial numbers, basic inventory information, warranty information, and the location of the equipment. It should also allow the inclusion of additional data that is specific to each company based on its unique operation.

Once complete, this database can be kept in one central location. It allows equipment information to be displayed efficiently and in a logical manner, thereby organizing data that previously may have been spread out among card catalogs, manufacturers manuals and drawings, and various binders. This equipment database may also serve as the primary source of equipment information for the vessel's crew and for fleet managers.

EQUIPMENT CODING

A simple coding scheme is required to uniquely identify each piece of equipment and each part. The coding scheme organizes the records into logical groupings, (e.g., it groups equipment into functional systems). It may then be used to access records within this system as well as through other integrated systems for reviewing, updating, and reporting.

One possible coding scheme divides the vessel's equipment into different systems or component groups. The individual pieces of equipment are then coded so that each piece is identified as part of a specific system.

For example, the fuel delivery system may include all equipment related to fuel delivery. This system may be identified numerically within the coding structure (e.g., as system 651). Subsequently, each piece of equipment related to the fuel delivery would be associated with that system. Port engine fuel pump 001 would appear in the equipment system as equipment 651001. Additional pieces of equipment in this system would be numbered 651002, 651003, etc.

The coding scheme that is used should meet the specific requirements of the company using the system. Additional information about the coding concepts can be found in the section on database development.

USER INTERFACE

A key component of any software is its user interface. This is the way information in the system is presented to the user, including procedures to access and update the information on the system. A comprehensive maintenance management system that does not have an easy-to-use, intuitive interface will quickly fall into disuse.

An intuitive interface should accomplish three things. First, it should provide a quick and easy way to access the desired information. Second, it should display the information clearly. Third, it should provide an easy way to update the information and generate reports based on the information.

The options for accessing data should work consistently for each module, and should include a means of selecting, viewing, adding, modifying, and deleting information.

To protect database integrity, the system should also provide some level of password security. Ideally, it should be user specific, functionally driven, and independent for each system. This means that some users will be able to modify or delete records in one system (such as maintenance), but they may only view records in another system (such as equipment). At the same time, another user may have unlimited access to all records.

The equipment record accesses a wide range of information. Often, all this information cannot be contained on one screen. To display additional information, some systems use pop-up windows that appear over the current screen. An example of a pop-up window is the spare parts list shown in figure 14-2.

This window provides a list of spare parts belonging to the equipment shown on the screen. Pop-up windows are also used to enter information into the screen and may be used for the description field to display additional data for each unit of equipment.

A well-constructed system should be able to access information in a number of ways, such as searching through system lists and equipment lists, sorting these lists alphabetically or by equipment number, or searching for information based on a specific user-defined search criteria. The ease with which the users will be able to find the desired information directly affects their acceptance of a computerized system.

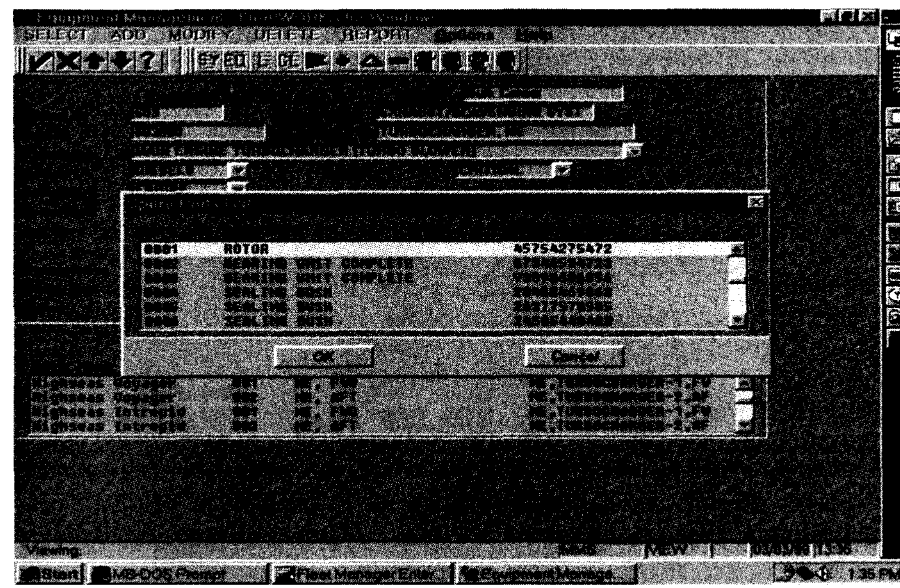


Figure 14-2. Spare parts list.
Courtesy Marine Management Systems, Inc.

Once the desired record is selected, the next consideration is the way information is displayed on the screen. Information may be displayed in different ways, but in any case, it should be clear and complete.

In the example shown in figure 14-3, the information contained in the displayed equipment record is separated into two parts. The top portion of the screen is information that is common to a particular make and model of machinery, such as equipment description, manufacturer, model, type, and rating. This information is common to each of the pieces of that equipment listed at the bottom of the screen.

The bottom portion of the screen provides information that is specific to each vessel. It lists one line item for each unit of equipment on board the vessel. In this case, it indicates that there are two turbochargers on board the *Voyager* and two on the *Intrepid*. The information contained in this screen may also be accessed by different components of the system, such as inventory or maintenance.

EQUIPMENT REPORTING

Any comprehensive equipment management system should provide ample reporting options to meet the user's reporting requirements. These reports provide a convenient source of documentation and reference. They also become key when addressing issues of compliance for the newer maritime regulations.

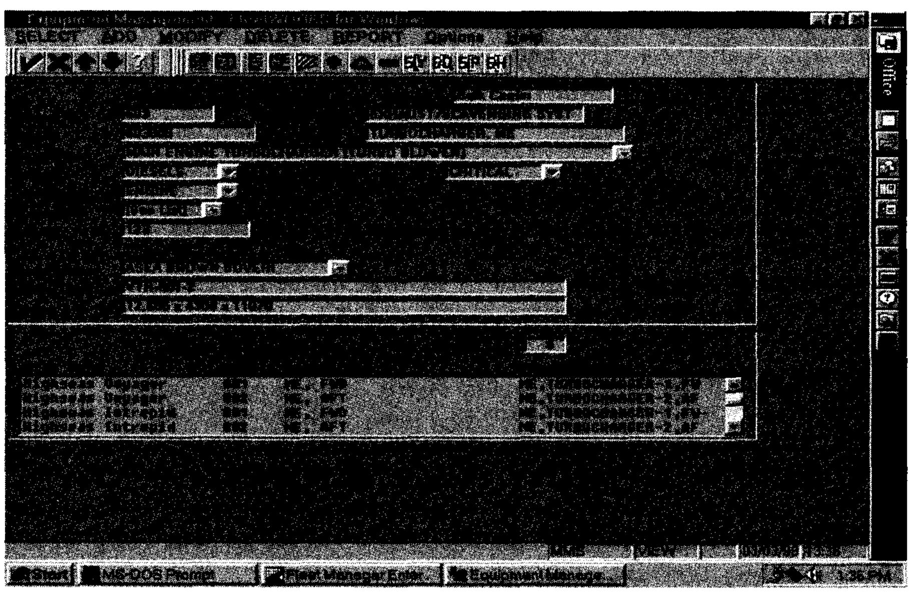
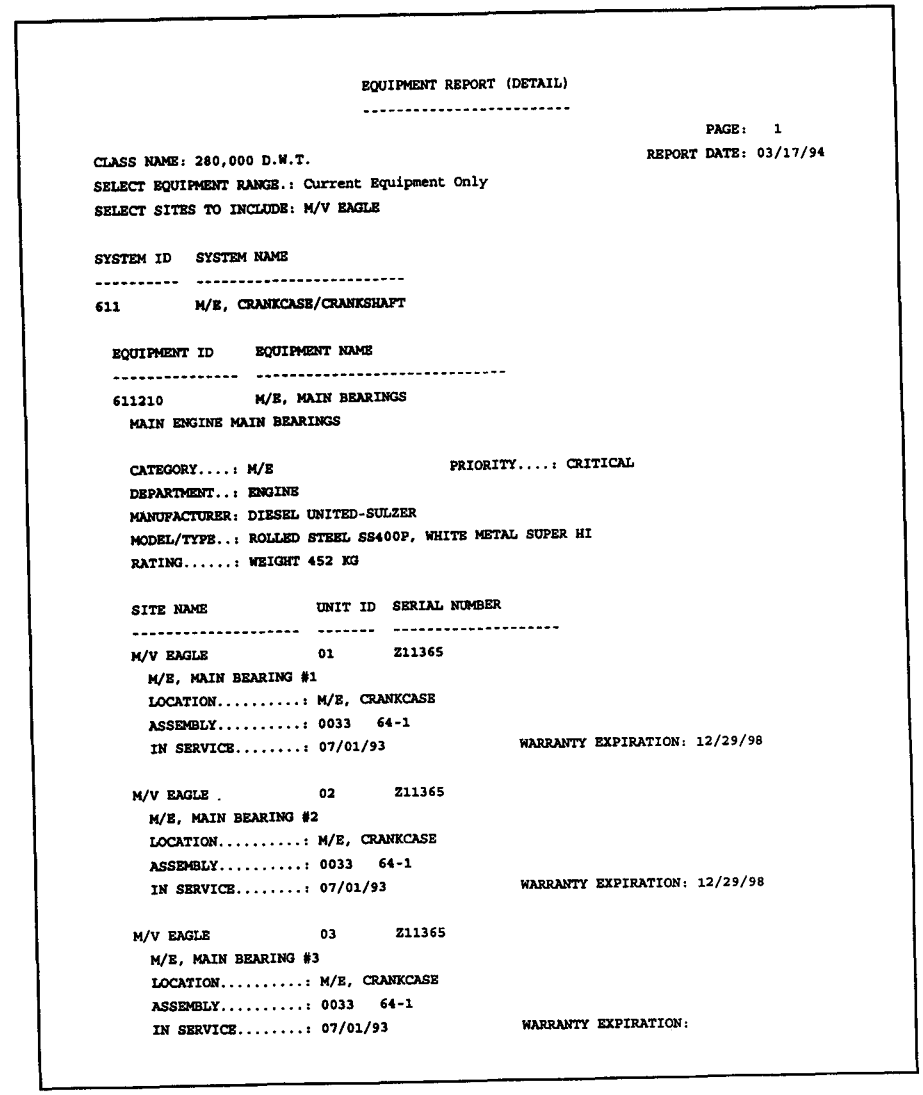


Figure 14-3. Equipment record.
Courtesy Marine Management Systems, Inc.

The reports included with the equipment component are a system report, which lists all the defined vessel systems in the database; an equipment report, which lists all pertinent information about a piece of equipment or a range of equipment; a spare parts report; and an equipment history report. Figure 14-4 shows an example of an equipment report.



```
                         EQUIPMENT REPORT (DETAIL)
                         -------------------------
                                                              PAGE:   1
CLASS NAME: 280,000 D.W.T.                              REPORT DATE: 03/17/94
SELECT EQUIPMENT RANGE.: Current Equipment Only
SELECT SITES TO INCLUDE: M/V EAGLE

SYSTEM ID   SYSTEM NAME
----------  -------------------------
611         M/E, CRANKCASE/CRANKSHAFT

  EQUIPMENT ID    EQUIPMENT NAME
  --------------  ------------------------------
  611210          M/E, MAIN BEARINGS
    MAIN ENGINE MAIN BEARINGS

    CATEGORY....: M/E                         PRIORITY....: CRITICAL
    DEPARTMENT..: ENGINE
    MANUFACTURER: DIESEL UNITED-SULZER
    MODEL/TYPE..: ROLLED STEEL SS400P, WHITE METAL SUPER HI
    RATING......: WEIGHT 452 KG

    SITE NAME            UNIT ID   SERIAL NUMBER
    -------------------  -------   --------------------
    M/V EAGLE            01        Z11365
      M/E, MAIN BEARING #1
      LOCATION..........: M/E, CRANKCASE
      ASSEMBLY..........: 0033   64-1
      IN SERVICE........: 07/01/93            WARRANTY EXPIRATION: 12/29/98

    M/V EAGLE .          02        Z11365
      M/E, MAIN BEARING #2
      LOCATION..........: M/E, CRANKCASE
      ASSEMBLY..........: 0033   64-1
      IN SERVICE........: 07/01/93            WARRANTY EXPIRATION: 12/29/98

    M/V EAGLE            03        Z11365
      M/E, MAIN BEARING #3
      LOCATION..........: M/E, CRANKCASE
      ASSEMBLY..........: 0033   64-1
      IN SERVICE........: 07/01/93            WARRANTY EXPIRATION:
```

Figure 14-4. Equipment report.
Courtesy Marine Management Systems, Inc.

Next there must be a mechanism for defining the scope of information to include in the report. This may take the form of an options window that allows the user to specify the following:

- what equipment, spare parts, or work procedures to include in the report
- what database sites to include in the report
- what level of detail to include in the report

There should be a sel;lrch mechanism that allows the user to further specify the equipment to include (equipment for a specific department, category, priority level, etc.). This same feature should be available for inventory reports and maintenance reports.

Since an infinite number of report queries may need to be prepared on both the ship and the shore, the system should be designed to create unique custom-tailored reports. The goal ofthis system is to provide complete and accurate inventory and maintenance information; the reporting capabilities of the system will provide a clear indication of how well it performs that task.

SETTING UP A DATABASE

Each component of a maintenance management system (equipment, spare parts, maintenance, etc.) maintains information in records. Initially, records are created for each piece of equipment on each vessel in the fleet.

Systems may be created when the equipment database is initially set up. The systems reflect the logical groupings ofthe equipment records. The window shown in figure 14-5 displays a list of systems.

Mter the systems are created, equipment nameplates (a short list ofthe major shipboard equipment) are entered next.

Once the coding scheme assigns an identifier to each system and to each piece of equipment in that system, then additional numerals may be added to an equipment number to uniquely identify each spare part and work procedure. For detailed information on setting up a database and coding scheme concepts, refer to the section on database development.

FLEET CONSIDERATIONS

Each fleet is unique in terms of size, the type of vessels in operation, and the mode of operation. Therefore, it is important to perform a needs analysis and identify the type of database structure and system capabilities that are appropriate for that fleet.

One consideration is the scope of fleet data to which each vessel will have access. When an equipment list is displayed on any vessel, that list may include only the equipment on board the single vessel, or it may include the equipment recorded for any vessel of the same class, or (as the

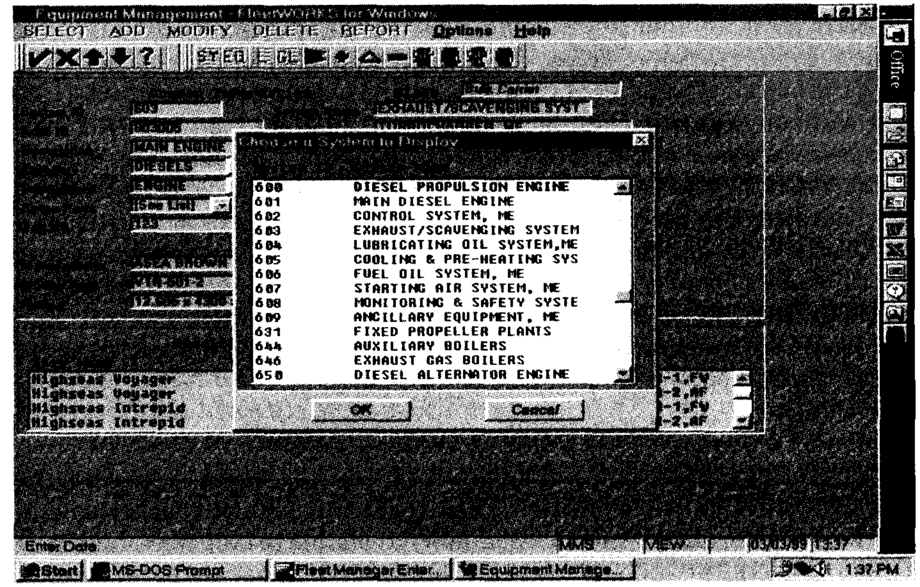


Figure 14-5. System list. Courtesy Marine Management Systems, Inc.

case may be for the shore office) it may include all equipment for all vessels in the fleet.

A maintenance management system that incorporates a record domain allows the shore to control the scope of information that is displayed at any site. The larger the domain, the more comprehensive the lists become (each vessel references a master database of equipment information throughout the fleet). The narrower the domain, the fewer the records to be searched to find the desired information.

Another consideration is the flexibility of design. A successful maintenance management system will have a modular structure by which there can be a phased approach to systems implementation. This building-block approach to systems allows each fleet to maintain the level of detail that is appropriate for their use (beginning with the basic entry-level systems) and allows the system to incorporate more advanced features or systems as needed.

Depending upon the number of users at any site, a local area network (LAN) can add greatly to the usability of the system. This option offers a "multiuser" system that allows multiple computers to access the software simultaneously. In this way, several users can share data at multiple computers at any site and obtain up-to-date reliable information. A LAN system should offer full "file and record-locking" control in place to prevent data use conflicts. The systems may also be configured to protect the files from unwanted access.

Security is another key element. The system design should allow for custom configuration of access for each crewmember of each vessel. This way, the broadest number of crewmen will be able to benefit from the system without jeopardizing data integrity.

A maintenance management system should be a configurable system capable of incorporating these features, and it should be sufficiently flexible in its design to allow the users to tailor the system to the specific needs.

Once a basic equipment database has been created that itemizes and identifies each piece of equipment on the vessel, a corresponding database needs to be constructed for the spare parts that are required to maintain the vessel equipment. Refer to the section on database development for more information.

Parts Inventory

The inventory component of a maintenance management system organizes an array of spare parts information, including quantities and location, and makes that information immediately available to ship and shore personnel. It should provide complete and accurate lists of spare parts for each piece of equipment, maintain inventory levels, record delivery receipts, and adjust inventory levels electronically via portable bar-code readers.

The goal of an inventory component is to simplify inventory control and reduce overall inventory investment by organizing the database of spare parts, streamlining inventory levels, and providing an easy way to reference and update inventory information. The components of a complete inventory system include the following:

- spare part records
- consumable records
- inventory adjustment records
- bar coding

These options work together to organize and maintain spare part information.

SPARE PART MASTER RECORDS

A master spare part database allows the user to add and maintain detailed spare part records and inventory information. Due to the volume of spare part records, the system should provide an efficient search engine for locating or reporting on parts via the part name or by any of the part numbers associated with the part (manufacturers part numbers, vendor part numbers, drawing numbers, catalog numbers, etc.).

At a minimum, the system should provide the following information: a complete part name and description, inventory storage location, the manufacturer's name, part number, part rating, additional part numbers (e.g., vendor part numbers), current and recommended inventory levels, an indication of any open orders for the part, pricing information, and miscellaneous notes.

By indicating minimum inventory levels, the vessel should be able to produce monthly shortage reports and to use that information for requisitioning replacement parts. Maintaining appropriate inventory levels helps to ensure that the required parts are on hand for both planned and unscheduled maintenance. This is a crucial element in reducing downtime on the vessel.

Some inventory systems provide access to digitized drawings, and digital cameras are becoming increasingly affordable. These images not only assist with part identification, but are of considerable help when reporting mechanical failures to the shore office.

The inventory component should provide various reporting options for extracting additional inventory information. These reports provide a convenient source of documentation and reference, particularly when performing a physical inventory or for inventory tracking and control. As with the equipment reports, there must be a mechanism for defining the scope of information to include in the report: what spare parts to include, what database sites to include, and what level of detail to include.

The reports associated with the inventory component may include reports for inventory status, physical inventory, inventory valuation, alternate part numbers, and parts usage.

An inventory status report provides the user with a printed record of spare part information for a selected range of equipment, including manufacturer information and status information such as the quantity on hand, the quantity on order, the recommended inventory levels, and purchase order and pricing information. It should also have an option to include only those spare parts that have fallen below the recommended inventory level or that exceed the recommended maximum inventory level (shortage and surplus reports).

A physical inventory report includes inventory information for spare parts in a selected range of storage locations. It should provide the part location, a manufacturer's part number, an inventory count, and a place for the user to record an adjusted inventory number.

An inventory valuation report includes inventory value information for a selected range of equipment. It calculates the total value for the selected inventory and provides updated replacement values. Inventory valuation should be calculated based on average prices, and replacement value based on last price paid. This capability will help identify inventory investment throughout the fleet.

An alternate part number report lists inventory parts based on an alternate part type, such as vendor part number. This is a convenient way to create a spare part report that cross-references different part numbers.

The system should support any number of alternate part numbers within its spare part master record, and these numbers should be user defined during the database development phase.

A parts usage summary report provides a list of all spare parts drawn from inventory between a selected range of dates. This report is convenient for monitoring the parts usage for a given period of time. The report identifies the parts that were used and the quantity used.

SPARE PARTS STATUS

A spare parts status module, shown in figure 14-6, should be provided to display a list of all spare parts belonging to a particular piece of equipment, along with the current inventory levels of those parts. This module may indicate the manufacturer part number, the number on board, location of the parts, and the number on order, thereby providing a convenient reference when doing maintenance on a piece of equipment.

CONSUMABLE INVENTORY

The inventory system should provide a separate database for consumable items. This will include items with high turnover rates (such as electronic parts and small hardware), steward's supply, sundry items, and perishable items (safety gear, pharmaceuticals, provisions, etc.).

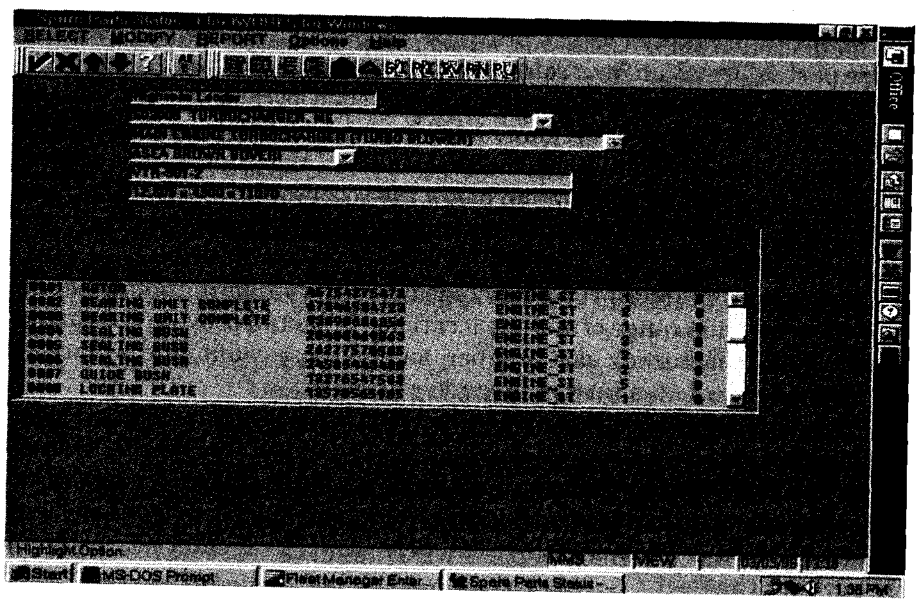


Figure 14-6. Spare parts status list.
Courtesy Marine Management Systems, Inc.

The purpose of maintaining separate databases is to provide a logical grouping of the inventory (rather than to combine engine parts with galley provisions) and to minimize the time required to locate any specific item. There is also additional information that is desirable in a consumables database including expiration dates, dates last ordered, and dates last received.

From the perspective of data integrity, separate databases allow the steward's department to access and modify consumable records without requiring access to the parts inventory database.

INVENTORY ADJUSTMENTS

Any computerized inventory system will require an organized means of entering adjustments to the inventory levels and tracking those adjustments (e.g., who made them, when, how many). This function should be used to record changes in inventory due to inventory received, used, lost, or damaged. The three areas of inventory adjustments that should be accommodated in the system include delivery receipts, parts usage, and ad hoc adjustments.

A delivery receipts adjustments option is used to record incoming delivery information for open requisitions. As inventory is received on board the vessel, it should be recorded, the spare parts master records updated, and the status of open requisitions adjusted. This option should provide a record of the person who took receipt of the parts, the date the parts arrived, an indication of whether this was a partial delivery, and any miscellaneous notes.

A parts usage adjustments option is used to record spare parts drawn from the ship's inventory, and an ad hoc adjustments option is used to record miscellaneous fluctuations in inventory levels (such as lost or damaged parts, and revised inventory levels recorded during a physical inventory). These options should also provide a record of the person who recorded the change in inventory, the date when it was recorded, and any miscellaneous notes.

BAR CODING

A bar-coding option significantly reduces the number of keystrokes required by a user to operate the system while doing data entry of inventory information. It increases the portability of a system through the use of a portable, handheld bar-code reader. Bar coding provides an expedient way to process inventory adjustments and to perform a parts inventory audit.

A quality bar-code reader can scan approximately one thousand bar-code records before it needs to be uploaded to the computer. It has an LCD display and a keypad for manual input of numeric data.

Some readers will automatically record the time and date of the readings. Once all desired readings have been taken, the user must upload the readings from the reader into the computer.

The parts inventory audit readings can be used for inventory validation. This is the process of comparing the recorded inventory (as reflected in the maintenance manager system) with the actual physical inventory, then updating the system inventory levels, if necessary, to reflect the actual inventory. This process is usually done periodically to confirm inventory levels. When this information is later processed, the values entered here should overwrite the previous inventory level and create an adjustment record reflecting the change in inventory level.

Requisitioning

A comprehensive inventory and maintenance system should be supplemented by a requisition option. This is a powerful companion to the inventory component, facilitating the creation and processing of inventory requisitions that are normally transmitted to the shore office, thereby helping to ensure the availability of spare parts for scheduled and unscheduled maintenance activities. This can help keep maintenance routines on schedule and prevent downtime for onboard equipment.

Through this option, part requisitions can be created. The system should keep track of the parts ordered, dates ordered, prices, and the status of pending orders; it should make that information accessible to the shipboard personnel.

An effective requisitions option will read the inventory database and use that information to display spare parts in a pop-up list from which users may select the desired parts and enter the quantity needed, thereby creating a requisition. The system should automatically use the information in the inventory database to fill in the remaining requisition details.

In order for the crew to accept it, the requisition system must be designed with ease of use in mind. The system must provide a faster and easier way to generate requisitions than the existing procedure.

Properly designed, a computerized system saves the time normally associated with creating handwritten requisitions. It also eliminates errors in the requisitions since all the information about the parts (manufacturers part numbers, serial numbers, model, rating) is automatically entered by the system, and arrives at the shore in a computer-ready format; it doesn't have to be rekeyed into a computer, but can simply be imported from a disk or the network. Lastly, when combined with a communications system, it means requisitions can be submitted on a daily basis from wherever the ship is operating (which significantly expedites the requisition process).

The financial savings that can be expected from this system stem from a number of factors. One is that the system should reflect the current quantity on hand for a given part as well as the number currently on order, when creating a new requisition. This helps minimize surplus inventory on the vessel. Another consideration is that this system can be combined with an efficient communications system, thereby significantly reducing the costs incurred when faxing requisition information between ship and shore.

The system should support three types of requisitions: part requisitions, consumable requisitions, and general (or service) requisitions. Other options are desirable to monitor the status of the requisitions and manage requisition budgets. The components of a complete requisition system include the following:

- requisition status
- part requisitions
- general (or service) requisitions
- consumable requisitions
- budget tracking and vendor information

A requisition should be easy to create and, once submitted, available for review and processing. Processing requisitions involves approving or denying the submitted requisition, creating vendor bids, and generating purchase orders. One question to consider is the level of approval required on the ship and/or on the shore. The system needs to accommodate that process.

The approval process helps streamline spare parts inventory. Once a requisition is approved, it is passed to the purchasing department, which performs the necessary purchasing functions. While some systems may incorporate a purchase order function as part ofthe requisition process, it is sometimes necessary to keep the purchase order function separate as it is often part of a corporate accounting system. In such cases, a requisition interface will be required to function with the existing purchasing system. It is imperative that the system support an open architecture to allow for this compatibility. When all the approved parts have been delivered to the vessel, the requisition is closed.

REQUISITION STATUS

A requisition status option provides quick access to the status of all open requisitions including when they were prepared, whether they were approved, the cost, whether a purchase order was prepared, and whether they have been filled. The purpose of this information screen is to provide the type of information that, in the past, could only be ascertained by ship's crew calling or faxing the purchasing department. By automating the flow of this information, the desired information is automatically provided to the vessel, thereby reducing the number of calls to the shore and hel ping to schedule upcoming maintenance more efficiently.

PARTS REQUISITIONS

The parts requisitions option draws on the information in the spare parts database and is used to review existing spare parts requisitions and prepare new ones. Specifically, it may requisition any spare part identified under the inventory option. Once completed, the parts requisition screen identifies the equipment for which the requisition was created (in the Eqpt ID/Name field) and lists at the bottom of the screen the spare parts that are being requisitioned for that equipment. The requisition keeps track of parts ordered, dates, and prices.

In figure 14-7, the part requisition screen is separated into two sections. The top portion of the screen displays the general information about the requisition (including the requisition status and ID), who prepared the requisition, when it was prepared, and the name of the equipment for which the parts are being requisitioned.

The bottom portion of the screen lists the requisition line items. On a shipboard system, this area will list the individual spare part requisitions associated with the equipment specified in the top half of the record. One line item will appear for each spare part currently being requisitioned.

Next to each part name will be the number of parts being ordered, the number currently approved, the purchase order quantity, and the total number of that part that has been received against this requisition.

GENERAL REQUISITIONS

A general requisition option is an open, text-based requisition used for service or other noninventory items. This record should display general in-

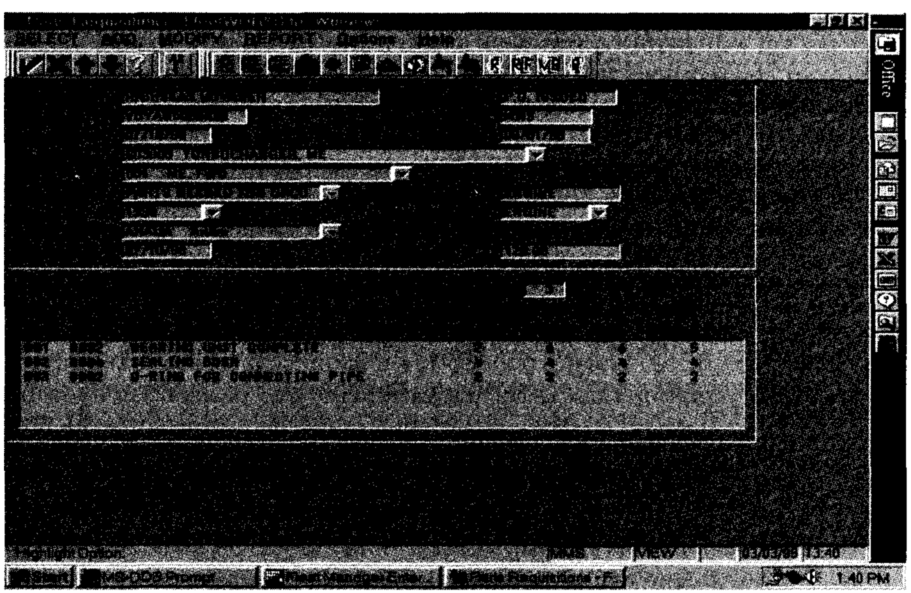


Figure 14-7. Part requisition screen.
Courtesy Marine Management Systems, Inc.

formation about the requisition, including the requisition status and ID, who made the requisition, when it was prepared, and a brief description of the services or items that are being requisitioned. The screen should also provide an extensive text area to describe fully the services or the noninventory items that are being requisitioned.

CONSUMABLE REQUISITIONS

A consumable requisition option allows the user to requisition consumable items, including the steward's supply, sundry items, and perishable items (safety gear, pharmaceuticals, provisions, etc.).

This option should provide a means of copying older requisitions in order to create new requisitions. This capability is important since some consumable requisitions may have in excess of one hundred line items.

BUDGET TRACKING

Budget tracking is a key element of a computerized requisition system. It helps the users to organize their expenses and to requisition more cost-effectively. This capability can be used to track budgets and provide budgetary information for parts and services requisitioned. It should allow users to create different budget categories and assign an annual budget to each category. Each time a requisition is created, the cost ofthe requisition should be assigned to one of these categories. The system will then maintain the year-to-date expenditures for each budget category.

Maintaining a requisitions budget on the vessel allows the individual vessels to more easily monitor their own operating costs for the parts and services they are requisitioning.

The budget category record contains information about the defined budget categories for that year. In the example in figure 14-8, the top portion of the screen identifies the site and fiscal year for that record.

The bottom portion of the window provides the budget amount allocated for each category, the amount currently committed to open requisitions, the amount spent so far in the current year, the budget remaining, and the percentage of the original budget remaining.

When the requisition is filled, the "year-to-date used" is updated with the actual cost ofthe requisition and the estimated cost is subtracted from the "year-to-date committed" column.

PLANNED MAINTENANCE

A comprehensive maintenance management system organizes maintenance information and makes it accessible to both shipboard and shoreside personnel. This information identifies what needs to be done, when it needs to be done, how long it will take, what resources are required, and what steps are to be perfOI:med.It also documents completed maintenance in the form of equipment histories.

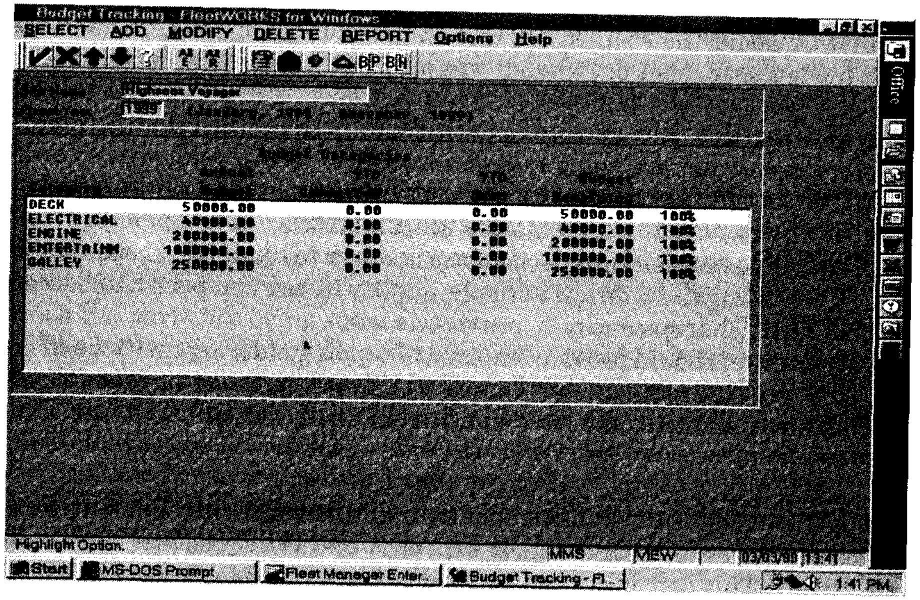


Figure 14-8. Budget category record.
Courtesy Marine Management Systems, Inc.

The goal of a planned maintenance system is to provide an efficient means of standardizing a maintenance routine for onboard equipment, streamlining the administration of vessel maintenance by generating the work schedules, and documenting completed maintenance for regulatory compliance.

The term maintenance may refer to planned maintenance, predictive maintenance, or unscheduled "corrective" maintenance. Planned maintenance refers to work that is done on a regular basis (a calendar or running-hours basis). Predictive maintenance involves identifying potential engine problems before they occur by monitoring specific aspects of equipment condition (such as vibration readings, lube oil analysis, pressure readings, etc.). Unscheduled maintenance refers to work that is performed after a failure has occurred.

Initially, the maintenance component is used to plan maintenance events and organize maintenance routines. Each work procedure should be detailed with a comprehensive list of steps to follow. The bulk of this work is done during the initial system installation and database development.

A primary consideration in the selection and implementation of a planned maintenance system is its ability to address issues of ISM compliance. Section 10 of the ISM Code stipulates that equipment inspections should be held at appropriate intervals and documented along with any nonconformity. A comprehensive maintenance system should allow the

user to identify ISM maintenance surveys, generate a list of upcoming surveys, and document their results. This can be done by including within the maintenance records a key field to indicate the nature of the maintenance procedure and also by including with the equipment history records occurrences of noncompliance and the corrective steps taken to address the issues. If these factors are taken into account, then the maintenance system should meet the requirements for ISM compliance.

Once these records are in the system, the maintenance component is used to generate scheduling information for upcoming equipment maintenance, as a reference source for procedural information, and as a record of equipment history. The system may also allow the user to schedule shipyard repair items and generate an initial shipyard repair specification.

The elements required in a comprehensive planned maintenance system include the following:

- work procedures
- running hours
- maintenance due
- equipment history

These elements, when coupled with timely completion of planned maintenance and attention to predictive indicators, are all key to establishing a standardized maintenance routine to prolong the life of the equipment, to improve safety, and to minimize breakdowns.

WORK PROCEDURES INFORMATION

A work procedure record is used to create and maintain the information about each individual maintenance job. These records contain the scheduling information for the equipment maintenance; they provide a section to record procedural information about the maintenance and a section to record related information such as the resources required to do the work.

Each work procedure record should identify a specific maintenance job, indicate the equipment for which the maintenance needs to be done, and show scheduling information for completing the work (i.e., how often it needs to be done, when it was last completed, when it is scheduled to be done next). A flexible maintenance system will allow maintenance to be scheduled on the basis of a calendar period, running hours, condition, or "as required." It may also differentiate between routine (or daily jobs) and larger maintenance projects.

Initially, the maintenance routines are usually taken from the manufacturers manuals, but they should also reflect company policy and maintenance information that has been compiled by the engineers on each vessel. This information provides a base of work procedure information that can be used for generating maintenance schedules and that can be updated to reflect changing conditions. Attention must be paid to the scheduling portion of the record. If the maintenance schedule generated by the system does not match the operational reality of the vessel, then the system will quickly fall into disuse.

Once created, these records are available as reference guides for how and when to perform any maintenance task. An example of a work procedure record is shown here in figure 14-9.

The top portion of the screen identifies the equipment for which this work procedure is defined. It also identifies the equipment location, specifies the work procedures to perform, and provides an area for notes that further describe the work procedure.

The work procedures should identify each step required to perform the maintenance. It is crucial that an appropriate scope is identified for the maintenance that will be tracked in the system. Making the work procedure database too broad will result in an overload of maintenance tracking. See the section on database development for more information.

The scheduling information provides the scheduling type and interval for the work procedure. This area lists how often the work procedure should be performed, when it was last completed, and its next scheduled date. It also identifies maintenance that has been deferred and provides a reschedule method for the maintenance (e.g., if the system automatically

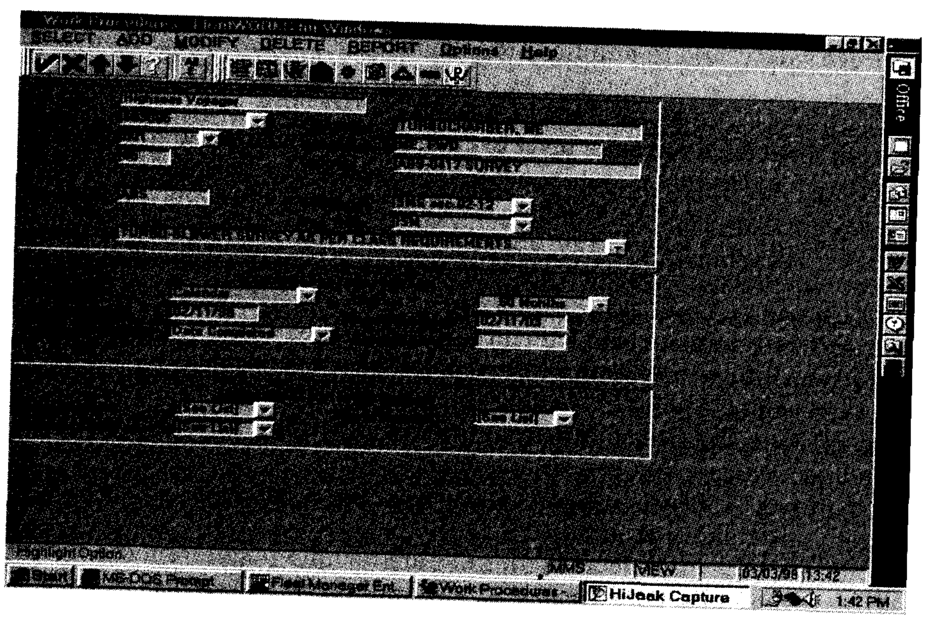


Figure 14-9. Work procedure record.
Courtesy Marine Management Systems, Inc.

reschedules maintenance, it may calculate the next scheduled date based on when the work was last completed or when it was scheduled to be completed).

The system should support some means by which the user is able to identify maintenance to be done in the shipyard, to itemize a requirements list for shipyard repairs, and to draft repair specifications.

The system should allow for a variety of interval types such as the following:

- calendar for maintenance that will be done on a monthly or yearly basis
- running hours for maintenance done on an hourly basis (i.e., every two thousand hours)
- "as required" for maintenance that is not regularly scheduled
- condition-based for predictive maintenance systems
- routine maintenance that will be done on a daily or weekly basis
- shipyard scheduled maintenance to be done in a shipyard or at a repair facility

In a well-designed maintenance system, the rescheduling of a work procedure will be done automatically after the maintenance is completed. Usually, the maintenance is rescheduled based on the date the work was done or based on the running-hour reading when the work was done.

The supplemental information section in this example provides an area to record the related tasks associated with the work procedure as well as the resources required to perform the procedure. Resources required may include spare parts and labor to perform the work.

The maintenance component should also provide ample reporting options to meet each user's maintenance reporting requirements. These reports provide a convenient source of documentation and reference.

The reports included with the maintenance component may include maintenance schedules, work orders, work procedure information, equipment history, etc., as shown in figure 14-10.

RUNNING HOURS

Much of the shipboard maintenance is performed on an hourly basis, so it is important that any maintenance system be able to schedule that maintenance accordingly. A running-hours option could identify the metered equipment and allow the user to enter meter readings for each piece of metered equipment. The system can then use the hours entered here to schedule work procedures for equipment that is maintained on an hourly basis (i.e., clean every seven hundred hours). For fully integrated systems, the running-hour meter readings may be fed automatically to the planned maintenance system.

If an item of equipment does not have its own meter, the system should still be able to maintain a running-hours maintenance schedule by referencing the meter readings of some other piece of related equipment that runs concurrently.

The system should provide a calendar override for maintenance performed on an hourly basis. This will ensure that equipment is not being run

```
                    WORK PROCEDURES INDEX REPORT (SUMMARY)
                    --------------------------------------
                                                              PAGE:   1
                                                       REPORT DATE: 03/15/94

SITE ID/NAME: 01   M/V EAGLE

EQUIPMENT RANGE.: FROM: 610000   PROPULSION DIESEL MAIN ENGINE
                  TO..: 611210   M/E, MAIN BEARINGS
INCLUDE WP NOTES: Yes

EQUIPMENT ID: 610000            NAME....: PROPULSION DIESEL MAIN ENGINE
UNIT ID.....: 01                LOCATION: ENGINE ROOM
                                                                  LAST       NEXT
WP ID  WP TITLE                     INTERVAL TYPE  INTERVAL      COMPLETED  SCHEDULED
-----  ---------------------------  -------------  ------------  ---------  ---------
001    ABS SPECIAL SURVEY           Calendar          60 Months  11/12/93    08/25/98

  WP NOTES:
    MAIN ENGINE SPECIAL SURVEY AS PER ABS REQUIREMENTS.


001    (ABS) CRANKSHAFT DEFLECTION  Running Hours  6000 Hours      032800    081780

  WP NOTES:
    CHECK CRANKSHAFT DEFLECTION. WHEN DOING SO THE SHIP MUST BE FLOATING
    FREELY. UNDER NORMAL CIRCUMSTANCES CRANKSHAFT DEFLECTION SHOULD BE
    TAKEN ANNUALLY, & MEASURED ASAP AFTER GROUNDING OR AFTER REPLACING THE
    MAIN BEARINGS. ADDITIONALLY IF YOU FIND DAMAGE IN THE MAIN BEARINGS THE
    CRANKSHAFT DEFLECTION SHOULD BE CHECKED.

    NOTE:
    PRIOR TO TURNING THE CRANKSHAFT WITH THE TURNING GEAR, MAKE SURE THAT
    NOBODY IS INSIDE THE ENGINE & THAT NO LOOSE PARTS, TOOLS, DEVICES CAN
    BE JAMMED. ALSO ENSURE THAT THE COUPLED PROPELLER TURNS TOO. (DANGER
    IN SURROUNDINGS).

)01    (ABS) CLEARANCE INSPECTION   Running Hours     500 Hours  83787   84287

  WP NOTES:
    CHECK AXIAL & RADIAL CLEARANCE. CHECK BOTTOM DRAIN FOR FREE PASSAGE.

)01    CHECK THRUST BOLT TENSION    Running Hours  18000 Hours       653      18653

  WP NOTES:
    CHECK MAIN BEARING THRUST BOLT TENSION. RE-TENSION AS NECESSARY.
    THREAD M125 X 6. HYDRAULIC PRETENSION 600 BAR.
```

Figure 14-10. Work procedure record.
Courtesy Marine Management Systems, Inc.

into the red due to incomplete or incorrect meter readings. It should also have a means of providing early notification for upcoming work.

MAINTENANCE DUE INFORMATION

Once work procedure records are entered, a scheduling option will help to plan maintenance activities for the day, the month, or the year. A key aspect of a maintenance system is its ability to generate maintenance schedules quickly for a variety of criteria (maintenance due this month, maintenance due for a specified interval, maintenance due for a specified regulatory agency, etc.). The more flexible the system is in providing accurate and complete maintenance schedules, the more effective it will be.

The following questions should be considered when reviewing maintenance systems:

1. How versatile is the schedule generating function (what criteria is available for generating the schedule)?
2. How much information is provided with the schedule? (An integrated system will allow the user to access detailed maintenance information about each job directly from the maintenance list.)
3. How flexible is the schedule (can a work order be generated from the list, can a scheduled procedure be easily deferred from the list)?

A versatile scheduling mechanism will provide some kind of criteria search list to specify the criteria for the schedule to be generated. The search criteria allows the maintenance due list to be custom-tailored to whatever specifications the user has (for example, all maintenance due that week or all maintenance due for a specific piece of equipment). The greater number of search fields available for generating the maintenance schedules, the more useful and practical the system becomes.

The comprehensive search criteria may be used to generate a custom list of upcoming maintenance for a specific department or category or for a specified priority. It can also show maintenance that is due for a regulatory surveyor maintenance that is past due or has been deferred.

A regulatory agency field, for instance, may be used to specify the work procedures associated with a specific regulatory agency (ABS, LRS, DNV, etc.). This option is key to preparing the vessel for routine survey inspections. By specifying an agency, the maintenance schedule that is generated will include all work procedures that are due for the agency and that match any other specified criteria included in this screen.

The scheduling capabilities of a computerized maintenance system are paramount. It is essential that the system is able to look in three directions-what is past due, what is currently due, and what is coming due-and to provide that information based on any criteria.

Other search criteria that should be available include overdue work procedures and deferred work procedures. The option to list all overdue work procedures becomes a convenient check to make sure that all engine maintenance is up-to-date.

A forward planning option is essential for projecting work schedules in the future and identifying those that must be done in a shipyard. It creates a maintenance schedule that lists each scheduled occurrence of the work procedure over a specified time span (three months, six months, one year, etc.).

For example, if a work procedure is done monthly, a forward planning maintenance schedule shQuld show each occurrence, specifying the scheduled date for each for the specified range (for example, the next six months). To generate this projected maintenance list, the system assumes the work procedure is performed as scheduled for each date.

Occasionally, scheduled maintenance will have to be deferred, either due to time constraints or to some other resource issue. The maintenance component needs to provide a means of deferring the scheduledjobs and rescheduling the work for a later date.

EQUIPMENT HISTORY

Tracking equipment performance is a significant tool for evaluation ofthe equipment and therefore is an important component of a maintenance management system. An equipment history module is used to record maintenance that has been completed for scheduled and unscheduled work, to document the parts and labor used for that maintenance, and to record relevant notes about the work that was completed. These records will then be available for reference and help in establishing patterns of performance. They help identify problems with the equipment and are a source of solutions previously used to correct the problem.

An equipment history record should be created each time maintenance is performed on a piece of equipment. Since many work procedures are performed on a calendar basis (as indicated in the work procedure record), the completion date is often used by the system to automatically reschedule that work procedure.

Unscheduled maintenance events, such as repairing a piece of equipment that breaks down underway, may also be recorded here, providing a record of equipment failure or damage.

Figure 14-11 shows an equipment history screen. In this example, the top portion of the screen identifies the particular piece of equipment, the work procedure performed, and the date the work was completed.

The section labeled "time and materials" displays the resources used the last time the work was done; this can be reviewed to determine maintenance costs for equipment over time. This information is important for making evaluations about the cost-effectiveness of different equipment.

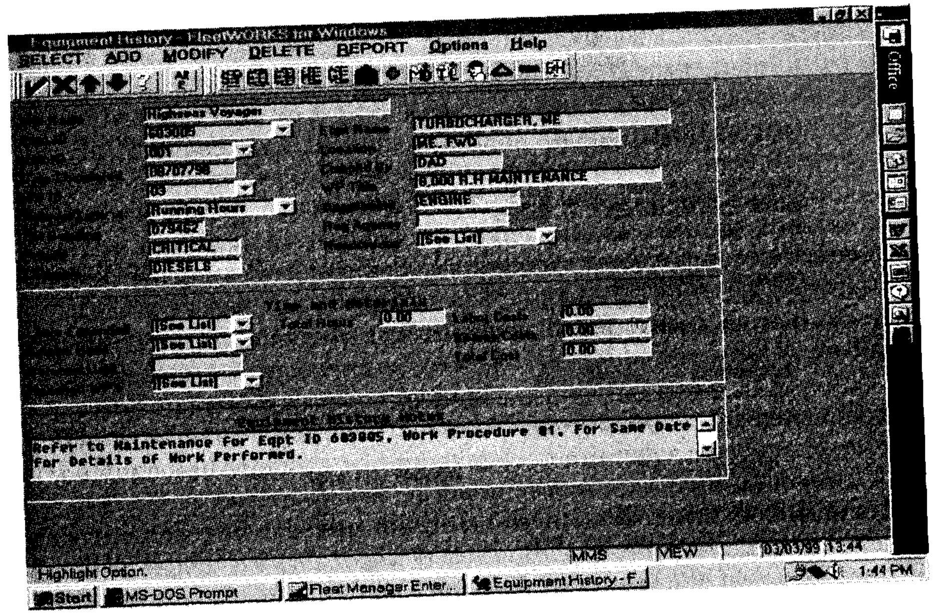


Figure 14-11. Equipment history screen.
Courtesy Marine Management Systems, Inc.

The bottom portion of the screen-equipment history notes-provides an area for the user to document any relevant observations about the equipment, the work performed, or any other related information. This section is important because the observations about the equipment that are recorded here can be reviewed subsequently to identify performance patterns.

It is important that the user not be too restricted in the amount oftextual information that can be entered into the notes section of an equipment history record, as this information will be used for equipment evaluation.

The equipment history reports (fig. 14-12) are key to documenting equipment maintenance for regulatory agencies. There should be a mechanism for defining the scope of information to include in the report: the equipment, the database sites, and the level of detail.

Predictive Maintenance

The benefits realized from utilizing condition-based predictive maintenance systems for cost-effective maintenance management are well recognized. Predictive maintenance is based on monitoring various types of measurements on the equipment, comparing the readings to the base reading of a healthy machine, and interpreting any changes in terms of possible faults. These changes may be small but, if consistent over time, they can indicate pending problems. If a monitored reading has reached a level where

```
                    EQUIPMENT HISTORY REPORT (DETAIL)
                    ---------------------------------
                                                          PAGE:  1
                                                   REPORT DATE: 03/01/94
CLASS NAME: Training (RELIANT)

SELECT EQUIPMENT RANGE......: Current Equipment Only
SELECT SITES TO INCLUDE.....: RELIANT
INCLUDE EQPT WITH NO HISTORY: No

SYSTEM ID   SYSTEM NAME
----------  -------------------------
51          AUXILIARY DIESELS

  EQUIPMENT ID     EQUIPMENT NAME
  ---------------  ------------------------------
  51620000         A/E, TURBOCHARGER AIR COOLER

    CATEGORY....: A/E                    PRIORITY....: HIGH
    DEPARTMENT..: MECHANICAL
    MANUFACTURER: GEAMAKE
    MODEL/TYPE..: 45/17/13/4  SR19
    RATING......: 20,300 KW @ 97 rpm

    SITE NAME            UNIT ID  SERIAL NUMBER
    -------------------- -------  --------------------
    HUSTON TRAINER # 1   01       145/10490/2

    AUXILIARY ENGINE NO.1 TURBOCHARGER AIR COOLER
     LOCATION..........: AUXILIARY ENGINE NO.1
     ASSEMBLY..........: 42344
     IN SERVICE........: 3/17/91            WARRANTY EXPIRATION: 12/30/96

      DATE      SUMMARY
      --------  -----------------------------
      06/14/93  Unscheduled Maintenance
        Cleaned and checked. Replaced filter.

      02/22/94  A/E, AIR COOLER CLEANING
        Cleaned Scav. Air Cooler and unit.
        Moderate accumulation, replaced oil scraper plate.
        Reduced cleaning intervals for this equipment.

Report Completed.
```

Figure 14-12. Equipment history report.
Courtesy Marine Management Systems, Inc.

the measurements exceed established alarm limits, it should generate an exception report that can be used to evaluate, schedule, and document equipment history. Various types of condition monitoring systems are in use, measuring specific data such as vibration or temperature. The systems may be interfaced directly with the ship's engine monitoring system to receive various readings.

The use of "expert systems" or "knowledge-based systems" can be very beneficial in interpreting the output of condition-based monitoring systems. These systems do a more in-depth analysis, utilizing the input of a number of measurements and conditions. They are tools available to the maintenance engineer to be used when making maintenance planning decisions or reporting what action to take and when to take it.

While a predictive maintenance system may stand alone, it is most effective when linked to a planned maintenance system that establishes the maintenance schedules, allows for scheduling the available resources, and records all the maintenance actions taken. To effectively use this information, a predetermined maintenance action should be triggered by an alarm indicating a possible specific problem.

The final determination ofthe maintenance action is the responsibility ofthe shipboard engineer, who must make the decision on where to commit the available resources based on the information at hand. The predictive maintenance system brings all the information from the condition monitoring or expert systems together with other historic and technical information so that the maintenance engineer can make the best decision.

PREDICTIVE OR CONDITION-BASED MAINTENANCE SYSTEMS

There is increased interest in condition-based maintenance in industries where proactive action is taken to divert machinery failures. The airline industry has been very successful in this endeavor and large industrial plants also apply the technology. The U.S. Navy has been promoting this activity for years and is installing a standard platform for diagnostic data on most surface ships.

Condition-monitoring systems have been used in commercial marine operations for years. Monitoring equipment takes readings that either trigger an alarm or provide output to be interpreted by the user.

A vibration analysis system is the most commonly used type of monitoring equipment, consisting of a portable unit from which the crew takes readings periodically. When vibration monitoring is implemented, classification societies have agreed not to open up certain equipment for inspection and also to allow continuous survey status, though this requires an independent annual audit of the readings by an outside consultant. A number of companies use the outside consultant to take all the readings, often on a semiannual basis. Other companies have crews that actively take readings on board on a monthly basis or more often. Latest systems incorporate some type of diagnostic software to determine equipment condition on a general, not equipment specific, basis. See volume 2 for a complete discussion on vibration analysis.

A second condition-monitoring method is lube oil analysis. Oil samples are sent to a lab periodically for various tests, including some that use diagnostic software. A current trend utilizes on-line lube oil monitoring, which primarily consists of doing particle counts on possible debris in the oil drawn from various pieces of equipment. (An effective method of monitoring equipment for both vibration and oil analysis is to plot trends over a period of time. This can identify a deteriorating condition even though the cause may not be immediately known.)

The third type of condition monitoring in use relates to diesel engine monitoring. Since diesel engines are now used as propulsion machinery for most commercial ships, the manufacturers of the engines have developed diagnostic systems to be sold with the engine. These normally require inputs from special sensors installed with the system, or they may use sensor readings from the machinery control system or a combination of the two. There may also be some manual input of data. A few generic-type diesel monitoring systems exist, but the engine manufacturer has the most information on its engines and is in the best position to develop diagnostic software.

While these described condition-monitoring systems can be effective in a limited way, some major deficiencies prevent them from being used as an effective condition-based maintenance (CBM) system:

- Limited direct access or interface to a common monitoring network with the necessary data input from sensors. (Custom sensor systems are too expensive and hard to support.)
- Limited means of combining the information from various systems to diagnose a particular piece of equipment. Data presented in non-compatible forms.
- Lack of diagnostic software for specific equipment that can make use of the information from multiple sensors and condition monitoring systems.
- No common way of interfacing the output of the diagnostic systems with maintenance management systems that are used to schedule all shipboard maintenance action, record history of actions taken, and record the conditions that triggered the action.

The basis for the solution relies on the premise that a maintenance management system with maintenance scheduling and equipment history is fundamental to the maintenance process. It is the minimum starting point. The CBM diagnostic systems are used to trigger the maintenance actions. A maintenance action should be associated with each diagnostic.

The major limitation in implementing a condition-based maintenance system is the lack of expert diagnostic knowledge on specific equipment. Until there is a common method of developing and interfacing this information it will be necessary to use some of the expert systems already developed: Therefore, it is recommended that equipment currently monitored by any of the three basic condition-monitoring systems-vibration monitoring, lube oil analysis, and diesel engine monitoring (by engine manufacturer)-be used where available. Each of these will interface with the maintenance management system to trigger specific maintenance actions.

In addition there is a need to monitor various other pieces of critical equipment by interpreting sensor readings from the general machinery control system. The assumption is that most new ships will have a centralized machinery control and monitoring system because of the economics of unmanned engine rooms requiring such systems.

Since there is little diagnostic software for the secondary machinery, it is recommended that a common diagnostic platform be interfaced with the machinery control system that can read any required sensors necessary for the diagnostic. While it would be a great benefit to have diagnostic software from the vendor for each item of critical equipment, very few manufacturers offer the data such software would require. Therefore, it is necessary to provide some minimum level of standard tools to develop the required diagnostic. This platform must send diagnostic information to the maintenance management systems in a manner similar to the other condition-monitoring systems described.

The maintenance management interface has the following requirements:

1. All condition-monitoring systems will have a common equipment numbering scheme conforming to the maintenance management system.
2. Each diagnostic identified by any of the diagnostic systems will have a specific work procedure description that describes the fault and the action that should be taken to correct it. These work procedures will also have a standard coding as a subset of the equipment number. They will be transmitted to the planned maintenance system on a periodic basis. It is assumed that the work procedures are developed with the equipment manufacturer or taken from the technical manuals.
3. The diagnostic system will generate a history record for the particular diagnostic that will be sent to the planned maintenance system equipment history file, again with a standard equipment coding. This record will describe the failure and the readings that triggered the diagnostic.

Maintenance Reporting

The maintenance component should have a reporting option that provides the user with a printed record of the maintenance information for documentation purposes and other general reporting or reference purposes. At a minimum, the system should be able to produce two reports-a maintenance due report and corresponding maintenance work orders for the scheduled work procedures.

The maintenance due report is a listing of upcoming maintenance. Ideally, it should provide a summary format for quick reference and a detailed report for further information.

The maintenance work order is a printed work order for the maintenance job. It should include basic work procedure information and provide space to record history notes, work performed, labor expended, time invested, parts used, and cost. There should be a mechanism for defining the scope of information to include in the report-the equipment, the database sites, and the level of detail.

The maintenance reporting capabilities should include the same search mechanism for culling through key fields of maintenance records to produce reports for class societies and shipyard repairs. This capability is key to assisting with meeting ISM documentation requirements.

DATABASE DEVELOPMENT

To this point, the chapter has discussed the elements that comprise an effective inventory and maintenance management system. This section looks at the steps involved in developing a database for such a system.

Basic Requirements

Implementation of any maintenance management system requires a significant effort to set up the equipment, inventory, and maintenance databases prior to actual implementation of a fully functioning system. Besides the computer software being easy to use and logical in its design, accuracy ofthe database is probably the most important factor in establishing shipboard confidence in a system. If the system is started with accurate data, there tends to be a stronger commitment by engineers to keep it accurate. Therefore, database development and validation are perhaps the most critical tasks for a successful system implementation.

Proper implementation of a maintenance management system begins by deciding on the scope of information to be contained in the system and establishing a coding scheme so that the system can associate spare parts and maintenance routines with specific equipment. These considerations, along with collecting and organizing the raw data, are part ofthe preliminary engineering services required for implementation.

Shipboard engineering services are required for installing the system as well as completing and validating the database for each vessel. The validation process is a time-intensive one, and requires good organization and planning to make it go smoothly.

Database Implementation Steps

The following is a summary of the activities involved in preliminary engineering services to establish the initial database. Note that these steps are typically required for only the first ship of each class.

1. Collect source materials for shipboard machinery, including spare parts and maintenance requirements in a form suitable for database development.
2. Establish an equipment coding scheme to be used throughout the system to identify equipment and spare parts, and to associate specific maintenance routines with each piece of equipment.
3. Create an equipment nameplate list utilizing the equipment coding scheme. This list will become the primary equipment database.
4. Create a spare parts list for spares maintained on board as well as any additional parts that may be requisitioned. Each spare part should be linked to a piece of equipment.
5. Create maintenance work procedures required to meet regulatory requirements and maintain shipboard machinery according to accepted company standards.
6. Organize source materials into a form suitable for data entry into the database.
7. Develop the preliminary database by entering the equipment, spare parts, and maintenance information in the computer. Get operator's comments on preliminary database and make necessary adjustments or modifications.

Once the above steps have been completed, the final step of the implementation process is performing the shipboard engineering services effort. The following tasks must be performed for each vessel:

1. Install the hardware system and ensure existing system meets new system requirements. Install and test communications link between ship and shore systems.
2. Install the system software, along with preliminary database of machinery, spares, and maintenance procedures. Validation personnel joins vessel at this time.
3. Validate the equipment database and complete entry of missing nameplate data, such as serial numbers, ratings, etc. Confirm equipment database matches shipboard equipment.
4. Perform a physical inventory validation of onboard spare parts. Confirm and enter part locations, quantity on hand, and "bag and tag" spare parts. All spare parts (or sets of spares) should have a printed label identifying the part. Once complete, the computer database should represent the actual physical inventory onboard the vessel.
5. Initialize the maintenance schedule by updating the schedule information for each calendar-based maintenance item, update running-hour information for each piece of equipment for which maintenance is based on running-hours, and enter current running-hour meter readings for all metered equipment. Once complete, a maintenance schedule of current and upcoming maintenance will be available to the crew.
6. Transfer updated data to shore system.

Assuming that the crew has been trained, the system is now ready for day-to-day use. Requisitions may be prepared and electronically transmitted ashore, parts usage and deliveries may be updated, machinery history may be recorded as maintenance activities are performed, and management reports may be accessed both onboard ship and ashore.

The rest of this section discusses each of the steps in more detail, and explores the various options available at each step along the way. Depending upon resources available, each step may require any number of options to be taken; however, none of the above steps should be omitted if a successful system is to be installed.

Preliminary Engineering Services

Some engineering services are required to establish the initial database of inventory and maintenance information. These steps are listed below.

COLLECTING SOURCE MATERIALS

The method of collecting source materials varies greatly depending upon several factors, such as the age of the vessel and the perceived reliability of the source materials. With a new vessel delivery, for example, the shipyard should provide a complete equipment list, an allowance list, and manufacturers manuals as source material.

If the vessel has a manual maintenance management system that is in use, the vessel should photocopy "card ex" materials and other documents from the manual system and send them ashore to be used for data entry activities.

If it is an older vessel, and the source materials are questionable, there is a need to perform a "ship check," where an engineer visits the ship for several days to collect and photocopy source materials to be used for data entry ashore. This requires a longer validation effort.

Clearly, performing as much work ashore as possible provides the most cost-effective solution. Requiring extra work of shipboard engineers should be minimized, as their current workload may not allow this significant task to be added to their job.

If a ship check is required, access to ship's drawings, manufacturers manuals, and a reliable onboard photocopier is essential. In addition, shipboard engineers should be available to answer questions concerning plant configuration. Also as part of this effort, the classification society's maintenance list and requirements should be obtained.

ESTABLISHING AN EQUIPMENT CODING SCHEME

Once source materials are collected, the machinery list needs to be consolidated and organized into an orderly presentation. All maintenance information entered into the system should be organized according to a coherent equipment coding scheme. The coding scheme is a very important element of the overall system as it determines how equipment will be grouped and displayed. Most modern coding schemes group equipment by systems.

In cases where a new maintenance program is being established or where an outdated manual system is being replaced, guidance may be required in developing a coding structure that will optimize the use of a computerized maintenance system. This is particularly true where the system is being implemented for an entire fleet. Ideally, this coding scheme should be based on a hierarchical structure that identifies and organizes shipboard machinery (i.e., the equipment database) according to its function (propulsion, deck machinery, cargo systems, etc.).

EQUIPMENT CODING CONCEPTS

Under one recommended coding concept, each piece of equipment within a class of vessels is assigned a unique common equipment ID and a specific equipment ID. The common equipment ID is shared by all the sites within a class while the specific equipment ID identifies a particular site (ship) within a class.

The common equipment ID may be structured as follows:

MABBB

where MA identifies the system and the function of the equipment
BBB identifies each specific equipment

An example of this structure would be

240110, steering gear pumps
where 2 identifies deck systems
40 identifies steering gear systems
110 identifies the steering gear pumps

In addition to the common equipment ID, a specific equipment ID is then assigned to each actual unit. The specific equipment ID consists of the common equipment ID and the unit ID.

An example of this application would be

24011002, steering gear pumps-M.V. *Highseas* port unit
where 2 represents deck machinery/systems
40 represents steering gear systems
110 represents steering gear pumps
02 identifies the port unit on the M.V. *Highseas*

The specific structure that is used is less important than the fact that it supports a logical identification for each piece of equipment, and that it accommodates the full scope of the database.

SPARE PARTS CODING

Mter the equipment nameplate database has been defined, the next step is to define the spare parts associated with the equipment. Spare parts records may be identified by combining the common equipment ID with a unique part ID. Similar to the equipment nameplate database, the spare parts database may be defined by both general and specific records, where general spare parts records would be common throughout a class while specific spare parts records define information unique to an individual site.

This scheme can be structured simply or with as much detail as the user requires. It should be noted, however, that the overall coding structure used must take into account any spares, planned maintenance, and condition-monitoring requirements, even if all these systems are not included with the initial system implementation. Limiting the focus of the coding scheme to accommodate only equipment and spares may result in some difficulty in using the scheme to identify work procedures if a planned maintenance system is added in the future.

The coding scheme that is selected must allow new systems, equipment, spare parts, etc. to be added, moved, or deleted from the system easily. Also note that the coding scheme is used primarily to group the equipment in a logical manner. Once in the database, equipment is normally selected by its functional name, not its number.

CREATE AN EQUIPMENT NAMEPLATE LIST

An equipment nameplate list (the list of all onboard machinery) is, in fact, the basis for the equipment database and should be developed along with the coding scheme.

The coding scheme will organize the list of equipment into a hierarchical structure. Several steps must be taken when compiling this list, and some decisions are required along the way.

- Create a list of shipboard systems that will serve as the top layer ofthe hierarchy ofthe equipment number assigned to each type ofmachine.
- Establish conventions for assigning codes to the subsystems, which are the specific pieces of equipment to be listed in the database.
- Establish the length ofthe equipment ID, which will be the combination of the system ID and the equipment identifier. The length of this code will vary depending upon the complexity of the plant.
- Establish conventions for assigning the equipment name. Since the machinery list may be sorted by equipment name, key words are used to create an alternative method of sorting the equipment list (i.e., motor, winch, pump, etc.).
- Establish a complete cross-reference table of integrated equipment for use in generating consolidated reports on machine assemblies.

- Once this step is complete, each piece of equipment on the vessel may be assigned a unique equipment ID. Additionally, spare parts and maintenance work procedures should be assigned an ID number that associates them with the equipment to which they belong.

CREATE A SPARE PARTS LIST

It is not practical to expect the database to contain every possible spare part for every machine onboard the vessel, as engineers would have to review large lists of parts, many of which will seldom be required. Therefore, a decision must be made as to what spare parts information should be entered and maintained in the computer database. Establishing an appropriate scope for the system is essential to the success ofthe system.

The following is a list of subjects that need to be addressed when creating the spare parts list for shipboard machinery:

- What are the minimum parts levels required by the classification society?
- For each machine, what spare parts are maintained onboard, and what additional spares may be required?
- For each spare part, what is the prescribed minimum and maximum stock level that should be used as a guideline for reordering by shipboard engineers?
- There should be a determination ofthe number of a specific part in use on the ship that will give a good indication ofthe required level of spares.
- In order to facilitate the labor-intensive shipboard validation process, is there a reasonably accurate list of part storage locations and quantity on hand values that may be entered into the preliminary database?
- Is information available to create the preliminary database for cross-referencing commonly shared spares?
- If an older vessel, or one of a similar class, is there information on the current usage level of particular spare parts?
- Which parts have the highest consumption level?

The naming conventions used for the spare parts are as important as those used for the equipment. The name should begin with a significant descriptor, followed by the modifiers used to specify the part. For example, the XXX butterfly valve for the scavenging air receiver should be listed as "valve, butterfly, XXX." This way the valve will be grouped with all other valves when the spare part list is displayed alphabetically, and the modifiers will identify the particular valve.

CREATE MAINTENANCE WORK PROCEDURES

Creating a list of maintenance work procedures requires judgment as to what should be included in the maintenance program. The scope of the maintenance routines reflected by the system must be feasible and realistic. The following suggests some considerations.

The minimum starting point of a planned maintenance database is a list ofthe regulatory survey requirements. Therefore, this is the first set of source materials required. The next set of maintenance procedures has to do with main propulsion and auxiliaries, which in the case of a diesel is usually running-hour-"Qased maintenance. Lists of these procedures from manufacturers technical manuals, as well as copies of any current logs maintained by the vessel, are valuable source materials.

Most engineers have some sort of system they follow regarding periodic scheduled maintenance. Ifthis is in a written format, copies ofthese procedures will provide another source of valuable material.

Manufacturers manuals are very important in developing the database of work procedures to be scheduled. However, as in developing the spare parts list, the list of manufacturers maintenance procedures cannot be entered into the computer database verbatim. Decisions must be made as to the scope of manual material to include. The maintenance scheduling must also be consistent with the operating reality of the vessel, or the system will fall into disuse. The specific pages of a manual could be referenced in the work procedure.

Additionally, some systems include a list of spares required to effect certain repairs, and an estimate ofthe time and labor required to complete a maintenance activity.

Each one ofthese advanced features has a cost associated with creating the database to support it. The following is a brief description of the effort required in each case.

Standard tasks: Creating textual descriptions of detailed procedures to follow in effecting repairs is a very time-consuming effort. A decision should be made initially as to how detailed these descriptions must be, as well as who should author them. Expending significant effort in this area does not always prove to be cost-effective, since the shipboard engineers are assumed to have the basic knowledge of work procedure descriptions as provided in the repair manuals.

Spares required: Some systems can reference (through each work procedure) the list of spares required to complete the job. Collecting this information is also a time-consuming task. Often the manufacturer will simply list all recommended spare parts. Since engineers can access the list of spares for a piece of equipment upon demand, it may not be necessary to list all these spares again under the work procedure.

Labor required: The ability to estimate how long it will take to complete each work procedure, and knowledge ofthe skill level required, can be used to analyze staffing requirements to complete shipboard maintenance routines.

Whether to utilize the three features above, each requiring additional engineering services effort, depends upon the needs of each vessel. In most cases, establishing the basic computerized maintenance program represents a significant effort, and implementing advanced features at the beginning not only drives up the cost, but can in fact complicate the system for new users.

With this in mind, many companies will defer implementation ofthese features to a second phase of the project, or allow shipboard engineers to enter this information at their own discretion.

ORGAmZESOURCEMATERS

In order to utilize nontechnical data entry clerks to load the preliminary database, the raw source materials described above must be placed in a format amenable to efficient data entry. There are generally three approaches to be taken in this regard:

1. Use data in an existing electronic format and convert it as needed to the new system.
2. Use photocopied materials as source documentation for data entry operators to use.
3. Use preprinted data entry forms to enter data into database.

The latter two options tend to be labor-intensive (especially for spare parts lists). The simplest solution is to convert from an existing database. Fortunately, this option is becoming increasingly available since more and more companies are recording inventory information electronically and new builds are customarily being delivered with electronic inventory and maintenance databases in place.

DEVELOP A PRELIMINARY DATABASE

Creating the preliminary database consists of entering data into the system. This is the final step prior to the shipboard validation phase of the project. The following is a list offactors that should be taken into account during this activity.

Equipment database: The equipment database must be created prior to entering in spares or maintenance information. Generally, this database is constructed early in the project, so that coding assignments can be reviewed and altered by project engineers.

System backups: Because of the volume of data being entered into the system, an effective routine of creating backups should be designed and implemented.

Quality assurance: Various reports are available within most maintenance management systems to print or display the database in a formatted fashion. These reports should be used to review all entered data to minimize corrections that must be made onboard the vessel.

Shipboard Engineering Services

Once the preliminary database is created or converted from an older maintenance system, shipboard installation and validation is required prior to using the system. This task is both the most expensive part of the project and the most critical. TQ.ereare several options available to vessels in the implementation of the shipboard validation.

Having an outside validation team (often supplied by the hardware/software vendor) perform the validation on each vessel is easiest for the company and provides the greatest consistency of work.

Having the fleet personnel perform the tasks will reduce external costs, but it places the greatest burden on fleet engineers. They may not have the necessary time to perform the tasks in an effective manner. In many cases the ship operator may use professional assistance to set up the first ship and train ship's personnel to implement the following ships.

In addition, the following considerations should be addressed when planning the implementation effort.

Validation supplies: A variety oflabeling supplies and related materials should be available to be used in the storage and labeling effort.

Shipboard storage areas: Examining the shipboard storage areas may reveal inadequate facilities requiring purchase of additional cabinets or other storage apparatus.

Preualidation preparation: The validation team should not be expected to clean up a ship's unorganized parts storage plan. Each ship should be instructed to anticipate the validation team's visit by organizing the storage areas.

Computer skills: Validation teams are often called upon to establish backup procedures, assist the crew in setting up new software, and help with other activities relating to the computer. Each validation team should have one member who is familiar with the computer environment and personal computers in general. If these skills are not present, a general training on personal computers is recommended. If a local area network is to be installed, this training is even more important.

The following sections describe the onboard activities that must take place.

HARDWARE INSTALLATION

If new computer hardware is to be delivered to the vessel and used for the validation effort, it should be checked out prior to delivery to the vessel. Optionally, the software may be installed and tested prior to shipment. If existing shipboard hardware is to be utilized, sufficient hard disk storage should be available, and crew should confirm that all components are compatible (printers, modems, etc.). In addition, the following should be considered.

Network cabling: If a shipboard network is to be installed, sufficient time should be budgeted for installing cables in the overheads and through bulkheads. Additionally, unique shipboard problems, such as chafing of cabling due to vibration, must be taken into account.

Workstation location: Location of the single-user computers or the network workstations is a critical consideration, depending upon the application being installed. There should be a plan as to the best location of workstation(s) aboard each vessel, and assurance that appropriate resources are provided for each location.

SOFTWARE INSTALLATION

Installing the maintenance management system should be accomplished quickly and easily. Today's computers, however, are often full of memory resident device drivers and other third-party software that can occasionally hamper the installation process. Once installed and checked out, the validation process can commence.

EQUIPMENT VALIDATION

The main purposes of equipment validation on board ship are the following:

1. To confirm machinery list is accurate and up-to-date, especially models and ratings, which often change as older machinery is replaced.
2. To update missing manufacturing information, where practical. Additional information is usually available aboard ship that may be missing from other source materials. Often, the nameplate on the machine itselfis the best source of accurate information (if it is still readable).
3. To review machinery list for equipment onboard not in the computer database.
4. To confirm equipment locations are correct.

Normally equipment validation will take anywhere from one to three days, depending upon the accuracy of the source materials used to build the preliminary database.

SPARE PARTS VALIDATION

Because ofthe volume of spare parts maintained on most vessels, this task is time-intensive. The time required to perform the spare parts validation will depend upon the condition ofthe parts storage on the vessel and the accuracy of the data in the preliminary database. A major factor is whether the current parts are identified in terms of onboard equipment. This task alone can take two people several weeks.

If the preliminary database contains mostly accurate storage location information, the validation team would simply print part labels and a physical inventory list for each inventory storage locker, bag and tag the spares in the locker, and update the system database with accurate inventory level.

Ifthe preliminary database contains incomplete storage location information, each locker would be inventoried and the inventory list updated with corrected counts and locations. When this is done, the validation team would print part labels, bag and tag the spares in the locker, and update the system database with accurate inventory level.

In most cases the primary database will be incomplete because of the unavailability of accurate storage locations prior to boarding the vessel. Additionally, each validation team establishes its own routines that fit the unique circumstances of each implementation.

INITIALIZE MAINTENANCE SCHEDULE

Implementing the maintenance schedule requires the validation team to work with the chief engineer to perform the following tasks:

- Review equipment history log to determine the last completed date or meter reading when each defined work procedure was performed.
- Review regulatory survey status, updating work procedure list with the latest information.
- Collect current meter readings for all metered equipment.
- Update computer with collected information that determines maintenance schedule.
- Print a maintenance schedule projected over the next twelve months, and review the list to determine if work load is realistic and evenly spread out over the year.
- Make adjustments to the scheduling information for work procedures to evenly distribute the work load. If necessary, adjust work procedure intervals to create a realistic schedule. It is important for the final maintenance schedule that is generated to be feasible, given the resources aboard the vessel.

There is also the need to train the end users ofthe system. This should be done as close to the time ofimplementation as possible. The use of Computer Based Training (CBT) programs is becoming increasingly popular; they allow for flexible training schedules and penalty-free training environments.

TRANSFER DATA

Once onboard databases are finalized, the completed databases should be copied to the shore system, either via diskette transmission or electronically via communications software. This is the step where the ship and shore databases are initially synchronized.

Data Communications

A key component of modern vessel management is the ability to efficiently manage information on a fleetwide basis, including the ability to access current inventory and maintenance information from both the ship and shore sites. This is possible today through contemporary data communication technology.

Central to managing fleetwide information is the data link between ship and shore. This link should be designed with the ability to properly and efficiently exchange data between remote sites, to forward information, and to maintain reliable error-checking procedures for data integrity.

Two aspects of communications need to be addressed. The first is data communications and the second is data management. The first case deals with the physical movement of data from one site to another. The second case considers how the data will be managed before, during, and after the communication session.

Consider a practical example: generating a shipboard requisition. In the past, requisitions created on vessels were handwritten paper documents that were dormant until the vessel reached a port from which the requisition could be forwarded to the shore office. The process of responding to the vessel was similarly slow, and the whole transaction could take weeks or more. Alternatively, the requisition could be faxed to the shore; however, without data compression capabilities this process is costly, especially when dealing with a significant volume of material. In either case, there is no channel for information to come back to the ship from the shore. Clearly, enhanced data communications will significantly reduce the time and cost required to conclude a requisition process, because the information can be passed more efficiently between ship and shore at any time. How this process will work is the realm of data management.

Establishing the correct link for data communications is a matter of selectingthe appropriate communication method (or carrier) for the applications. Managing the data efficiently requires the use of effective data management routines.

DATA COMMUNICATION METHODS

Data communication methods available today include satellite, cellular, phone line, fax line, and diskette. Satellite transmissions provide the most efficient communications, although they incur the highest costs. Cellular is a cost-effective alternative for vessels operating in coastal waters. The option to transmit files via modem and phone line is a convenient, low-cost alternative for vessels while in port. Transmitting across a fax line (or data line) is practical, although the fax hardware itself usually does not incorporate any data compression techniques, nor does it allow for a two-way exchange of data. Finally, the diskette option provides a failsafe should the other options become unavailable.

The ability to incorporate satellite transmissions as a standard vehicle for performing data communications is being enhanced by the increasing number of satellite communication platforms now being offered. The conventional INMARSAT "A" transmissions are now supplemented by additional, lower-cost platforms such as Standard "C," digital platforms such as M-SAT, and high-speed data platforms such as INMARSAT "B." In addition, the expanding use oflow earth orbiting satellites is expected to keep the cost of satellite services competitive.

DATA MANAGEMENT ROUTINES

Data management routines are the procedures used by the system that determine how information is processed. More specifically, if a user is sending updated records from the vessel to the shore, the data management routines will determine exactly what data will be included in the transmission, how the data will be prepared for the transmission, and how it will be received by the remote site. Lastly, the data management routine may specify what will be done with the data once it is received at the remote site.

The success of a computerized fleet management system is dependent on effective data management routines. If the data cannot be correctly and efficiently transferred to multiple remote sites on a timely basis, the sers will not achieve the full benefit of the technology.

For some time, the maritime industry has looked for a timely and affordable way for the shore office to maintain current information for each of its vessels. To address this issue, data management routines called "distributed databases" and "field level transactions" were developed.

Distributed Databases

Efficient fleet management requires that inventory and maintenance information be available to both the ship and the shore operating staffs on a timely basis. Unfortunately, real time sharing of files between ship and shore is not usually a practical alternative. One answer to this is a distributed database architecture with data replication.

Data replication simply means that the ship and shore systems are able to maintain duplicates ofthe vessel's database at each site. The distributed database architecture is the method by which these duplicate databases are maintained.

A distributed database architecture allows shipboard and shoreside sites to enter and update information independently, while providing the means for a synchronized data exchange between sites. This method of data management allows a shore system to access and process consolidated information (such as inventory and maintenance information) for each vessel in the fleet.

A distributed database architecture should offer

- integrity: audit and recovery functions to ensure the integrity of data
- automation: data transfer routines that ensure all data is properly transferred automatically
- efficiency: ship/shore data transmissions that reduce costs
- flexibility: configurations that support multiple site setups

As the database is updated at any site (for example, the inventory levels are adjusted on the vessel or equipment histories are updated), these updates, or "transactions," are recorded and automatically stored in a pending "transmission" file, along with all other transactions. Periodically, this transmission file is exchanged between sites, updating each database with these transactions according to its method of data replication while also accepting all transactions from the other site, and then automatically queuing information to be passed to any other sites that are affected by the update. Figure 14-13 shows a typical distributed database.

At this point, both sites' databases are again fully synchronized. The frequency ofthe data transmissions will vary depending upon the needs of

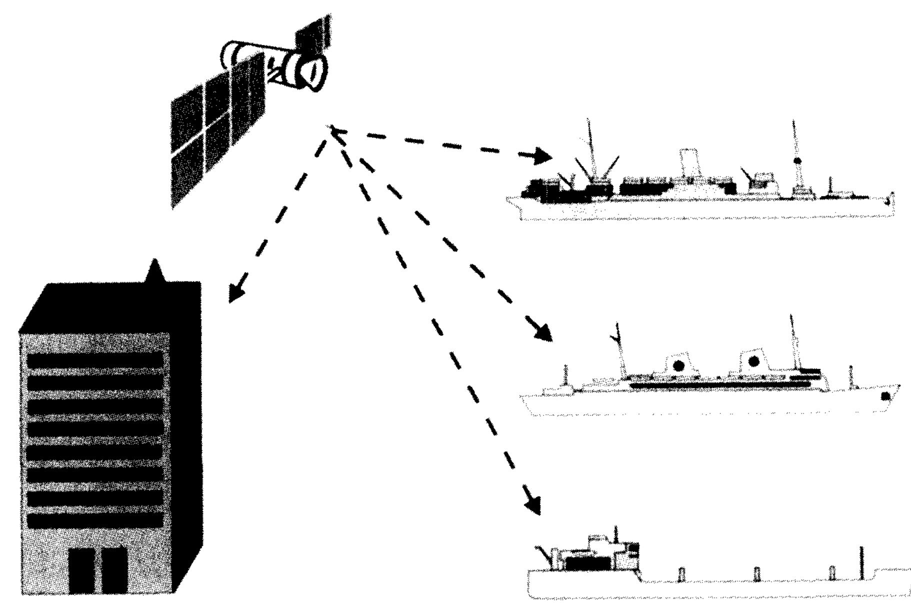


Figure 14-13. Distributed database.
Courtesy Marine Management Systems, Inc.

the users, although it is common when using satellite communications that transmissions are sent on a daily basis.

An efficient distributed database routine should operate independently of the system users, automatically accumulating transactions and building an outgoing transmission file. These transmission files then need to be exchanged between sites (using a two-way exchange of data) and decoded when they reach their destination. Data integrity checks must be present to protect against loading duplicate transmissions and to provide recovery in case of transmission problems.

Field Level Transactions

The traditional vehicle to support fleetwide data management was predicated on "record-based" data transfers to link ship and shore sites. This method requires that entire records, or even full databases, be imported and exported between sites. This approach results in large data transmission files and unnecessarily high communications costs. In addition, these systems have no way of ensuring the integrity of data, since no data checking was being performed on the files.

Field level transactions allow data communications to operate on a lower and more efficient level. "Field level" means that, as the user makes changes to data, only the changed "fields" are recorded as a transaction to be processed, not the entire data record. Field level transactions isolate the updated fields, and transmit only their contents to any affected remote site(s), updating those databases with the new information. The potential for data collisions is all but eliminated because the scope of the transmission is narrowed to include only updated data, and therefore data integrity is greatly enhanced. This is supplemented by additional data checking of the field information that is being passed between sites.

The field level transaction capability is the key to maintaining remote databases on a global basis. It also allows the ship and shore sites to modify records simultaneously. Transmitting and processing only field level transactions also dramatically reduces the size of data transmission files, ensuring much lower communications costs when compared with other systems' requirements to send full records or databases.

As a result of using distributed database technology and field level transactions, the ship and shore can each maintain a copy of the same database, they can each update information independently, and a single communications session will cause each site's database to be updated with information coming from the other site.

CONCLUSION

The level of sophistication on systems aboard ships today requires a carefully considered and well-organized approach to vessel maintenance. The

volume of information involved in maintaining today's equipment, the dictates currently emerging from regulatory agencies such as the ISM Code, and the need for individual operators to streamline vessel management all indicate that computerized maintenance systems will continue to be an essential element of any ship management operation.

For any application, the process of computerization requires careful consideration of the desired goals. Furthermore, the changing face of information technology requires that IT resource allocation decisions be made with regard to the larger strategic plan to continue to capitalize on IT industry changes.

The combination of an effective inventory and maintenance system together with an efficient data communications system has provided significant advances in the organization and accessibility of information used to maintain vessels on a fleetwide basis. On the individual vessel, this translates to less time devoted to administrative duties and more time to maintain the ship. On the shore, this means access to current vessel information that can be used to make cost-effective decisions regarding vessel maintenance, scheduling, and inventory purchasing on a fleetwide basis.

REVIEW

1. What are the steps involved in selecting a computerized inventory and maintenance system?
2. What new IT technologies are in place that ensure system compatibility in the future?
3. What are some considerations when developing a system coding scheme?
4. What are the steps involved in the shipboard installation of a computerized inventory and maintenance system?
5. What are the advantages of an automated requisitioning system?
6. List the different communication platforms currently available.
7. Describe the process of field level transactions.
8. What are some features to look for in a computerized maintenance system?

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CHAPTER 15

Classification and Regulatory Requirements

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SHIP CLASSIFICATION

Ship classification is a truly unique service, for it has no counterpart in any other field of industry or commerce. Briefly stated, classification societies establish and administer standards, known as rules, for the design, construction, and periodic survey of ships, offshore drilling units, and other marine structures. Classification certifies adherence to these rules, thus representing that a ship or structure possesses the structural and mechanical fitness required for its intended service. A classed vessel provides for the safety ofthe people and the cargo the vessel may carry as well as for the protection of the environment in which it may operate.

The Classification Society and the Marine Underwriter

As it was the eighteenth century underwriters who took the initiative in setting up a system of ship classification, it is necessary to review a bit of the history of classification in order to define the relationship between class and the underwriters.

In the eighteenth century, the City of London was already a major center of shipping and a great place for coffee drinking. As a point of interest, it has been suggested that the classification societies owe their existence to coffee, which was first introduced into England in 1652. It is significant perhaps that the habit of enjoying a cup of coffee started in the heyday of the Puritans. Gathering and drinking in pubs was out; gathering and drinking in a coffee house was in. An enterprising gentleman, one Edward Lloyd, discovered that the growing shipping industry also had other needs-and he set about to fulfill them. He opened a coffee shop in London that catered to the shipping community. Thus, he not only gave his name to the corporation of Lloyd's and to Lloyd's List, but also to Lloyd's Register of Shipping, the first ship classification society in the world.

Among the customers at Lloyd's was a group of marine underwriters. They were being asked to insure ships about which they knew little or nothing, so they decided to produce a register of shipping as a guide to the assessment of maritime risks. The underwriters employed surveyors, retired masters, and other knowledgeable people to inspect the ships. This was the starting point of ship classification as we know it today.

The earliest known register of ships, which is held in the British Museum, was published in 1764. As well as listing the ships and giving details such as tonnage, owners, builders and master (and the number and size of guns), the early registers also clearly indicated their general condition.

In the days of wooden sailing ships, the classification was assigned for a specific number of years; thus, a ship classed 8A1 remained in the highest class for 8 years nOOA1, the present day classification symbol, first appeared around 1870, when it was assigned to iron ships; it was thought that they would last at least a hundred years).

Although it was the underwriters who took the initiative in setting up a system of ship classification, shipowners and merchants soon realized the advantages of classification from their own point of view. Merchants wanted to know that ships carrying their cargoes were sound and fit for the intended voyage, while the owner who purchased a new or existing ship wanted assurance that he was investing in a sound, seaworthy vessel.

Despite the undoubted need for the register, it was to run into serious problems with shipowners and other parties who felt their needs were not being met. The only income came from subscriptions to the register book, and this shaky financial position was further threatened when a rival register was published by the shipowners. The ensuing competition for subscribers resulted in a "cut-price war" that brought both registers to the verge of bankruptcy. For a time, it seemed that the growing shipping industry in Britain would be left without a register of any kind. However, a committee of inquiry was set up to resolve the issue.

It took eleven years before the committee's recommendations were finally implemented, but they set the sound course Lloyd's Register was to follow. One of the most important recommendations was that the governing body, known today as the General Committee, should include members from all sectors of the shipping community: owners, shipbuilders, and underwriters. This helped the shipowners' demands for a better deal and has to this day guaranteed the impartiality of Lloyd's Register.

There were two other key points: that classification should be assigned in accordance with established rules, and that the register should have a permanent, qualified staff.

With no existing funds, the committee of inquiry felt that the sort of register they envisioned could only be financed with assistance from the government, but this was refused by the then British Board of Trade. At the time, it seemed that the whole project was doomed to failure. In hindsight,

it was the best thing that could have happened. This refusal meant that when the recognized society was finally set up, it was truly independent-free from political influence and expediency and governed solely by the representatives of the industries it served. The reconstitution, which laid the foundation for the subsequent expansion and influence of Lloyd's Register, is the most important milestone in its history; it took place in 1834.

The reason for the existence of classification societies is not always understood, probably because there is nothing quite like classification service in other fields of activity. From the beginning of maritime commerce, it has been in the interest of the shipowner and the shipper of goods, and later the marine underwriter, to ensure the soundness and fitness of ships. The forces to which the ship is subjected by the sea are not wholly understood; therefore, the only criterion by which a ship can be appraised reliably is by comparison with similar ships known to have been successful in service.

Forward strides have been made through research, both analytically and by instrumentation of ships in service, in learning the nature and magnitude of the force of the sea. Also, continual review has made the rules of the classification societies more precise in comparing one ship with another and in comparing individual components on the basis of recognized engineering principles.

Through application of the record of successful experience of ships in service and the theory of structures, standards for the construction of ships and their machinery have evolved that are acceptable to all parties interested in ships. These standards, or rules, have changed greatly and continually over the years, as experience was gained with new types of ships, new materials, and different services.

When a ship is built to the requirements of the rules of a classification society and under the survey of the society's surveyors, the hull material and other components are tested to specifications given in the rules. If all tests and trials prove satisfactory, the society grants "classification" to the ship by formal action of its committee in accepting the recommendation and reports of the surveyors. This fact is then published in the society's register book, where anyone may see that the ship in question conforms to recognized standards of sound construction.

In the event of a casualty, the classification of the ship helps the owner to establish that he has used the "due diligence" required of him; it informs the shipper that he is not taking a disproportionate risk by sending his goods aboard that particular ship; and it helps the underwriter decide the nature of the risk involved when he is asked to insure the ship, especially if it is a new type or an unusual ship. We use the term "risk" as a matter of simplification as the word risk in marine insurance is more often used to mean the insurance proposition as a whole rather than a special sense of peril or hazard.

International Association of Classification Societies (lACS)

The International Association of Classification Societies (lACS) can trace its origin back to the International Conference on Load Lines of 1930, which recommended that classification societies recognized by governments under Article 9 ofthe Load Line Convention of 1930 "should confer from time to time ... with a view to securing as much uniformity as possible in the application of the standards of strength on which freeboard is based .... "

HISTORY

In 1939, the first conference of international classification societies was hosted by Registro Italiano Navale in Rome and was attended by reJ1resentatives of the American Bureau of Shipping, Bureau Veritas, Det Norske Veritas, Germanischer Lloyd, Lloyd's Register of Shipping, and Nippon Kaiji Kyokai. During this conference it was agreed that cooperation between classification societies should be further developed and conferences should be convened as deemed desirable. There was no formal organization at that time.

The next conference was held in Paris in 1955 with Bureau Veritas as host, followed by meetings in London, 1959 (Lloyd's Register); New York, 1965 (American Bureau of Shipping); and Oslo, 1968 (Det Norske Veritas). It was during this Oslo conference that the establishment of an International Association of Classification Societies was agreed upon.

THE PURPOSE OF lACS

The International Association of Classification Societies was formally established in 1968 with three main purposes: (1) to promote improvement of standards of safety at sea, (2) to consult and cooperate with relevant international and marine organizations, and (3) to maintain close cooperation with the world's maritime industries.

lACS MEMBERS AND ASSOCIATES

Membership in lACS is held by ten leading classification societies:

American Bureau of Shipping	(ABS)
Bureau Veritas	(BV)
China Classification Society	(CCS)
Det Norske Veritas	(DNV)
Germanischer Lloyd	(GL)
Korean Register of Shipping	(KR)
Lloyd's Register of Shipping	(LR)
Maritime Register of Shipping	(RS)
Nippon Kaiji Kyokai	(NK)
Registro Italiano Navale	(RIN a)

In addition, the Polish Register of Shipping, the Croatian Register of Shipping (CRS), and the Indian Register of Shipping (IRS) are recognized as associate members.

lACS MANAGEMENT

The government body of lACS is the council, which consists of one senior executive from each member society. The council meets regularly once a year to conduct the activities of the association. Meetings to deal with matters of immediate concern may be held more frequently and at short notice. The principal objective of the council is to establish the general policy of the association, to solve any policy problems, and to plan for future activities.

The council also considers and adopts resolutions on technical issues within the classification societies' scope of work. Numerous "unified requirements" (URs), and "unified interpretations" (Uls) of international codes and conventions have been adopted by the council. Typical examples of lACS unified requirements are

- minimum longitudinal strength standard
- special hull surveys of oil tankers
- loading guidance information
- use of steel grades for various hull members
- hull and machinery steel castings
- cargo containment on gas tankers
- prototype testing and test measurement on tank containers
- inert-gas generating installations on vessels carrying oil in bulk
- fire protection of machinery spaces
- survey of hatch covers and coamings

Between the regular meetings of the council, the general policy group, a subsidiary body of the association, meets to deal with current affairs and progress of the lACS working groups.

lACS WORKING GROUPS

Working groups are established by the council in accordance with the charter of the association. They include both permanent working parties and ad hoc groups. Long before the formal foundation of lACS was established, a number of working parties existed to carry out studies of specific topics. The first of these was the working party on hull structural steel, established in 1957. It produced "Unified Requirement No. 1" for hull structural steels.

Following are the general responsibilities of the working groups:

- to draft unified rules and requirements between the member societies

- to draft responses to requests of the International Maritime Organization (IMO) and to prepare unified interpretations of conventions, resolutions, guides, and codes
- to identify problems related to the working group's area of activity and to propose IACS action
- to monitor the work of organizations related to the expertise of the working group and to report to the council

The following topics are the responsibility of individual working groups:

- containers
- drilling units
- electrical systems
- engines
- fire protection
- gas and chemical tankers
- hull damages
- inland waterway vessels
- marine pollution
- materials and welding
- mooring and anchoring
- pipes and pressure vessels
- strength of ships
- subdivision, stability, and load lines
- survey, reporting, and certification

IACS RELATIONSHIP WITH IMO

Since 1969, IACS has been granted consultative status with IMO. A representative of IMO has since then attended IACS Council meetings, and IACS representatives have regularly participated as observers at the meeting of the assembly, the Maritime Safety Committee, the Marine Environment Protection Committee, and different subcommittees and working groups of IMO. Recognizing the importance of a mutual relationship between IACS and the increasing contribution of IACS work to IMO activity, in 1976 the IACS Council appointed a permanent representative to IMO.

IACS is the only nongovernmental organization with observer status at IMO able to develop rules. These rules, implemented by its member societies, are accepted by the maritime community as technical standards. In areas where IMO intends to establish detailed technical or procedural requirements, IACS endeavors to ensure that these requirements are easily applicable and as clear and unambiguous as possible.

RELATIONSHIP WITH OTHER ORGANIZATIONS

IACS liaises with international organizations for exchange of views and information on matters of mutual interest. This ensures that the views of the industry are taken into consideration in the work of IACS. Examples of such international organizations are International Marine Insurers, International Chamber of Shipping, Oil Companies International Marine Forum, Society of International Gas Tanker and Terminal Operators Ltd., International Standardization Organization, and Economic Commission for Europe.

American Bureau of Shipping

The history of the American Bureau of Shipping followed much along the lines of that of the earlier classification societies, such as Lloyd's Register and Bureau Veritas, which started operations in the wood ship era, in that it was controlled by and for underwriters and rated ships by class symbols and terms of years. The members of the Board of Survey who approved these ratings were all inspectors employed by the various marine insurance companies. One must, however, acknowledge the fact that it was the support of the underwriters, especially of Atlantic Mutual, that kept the society in existence during many critical and hard times prior to 1916, when the society was completely reorganized with a properly representative board of managers and technical committees, all in conformity with the best modern practice.

The year was 1860 and it was a time of crisis for the marine industry. In the prior decade fires, groundings, collisions, and capsizings had destroyed an increasing number of oceangoing ships. The advent of steam propulsion contributed significantly to this situation since the installation of heavy coal burning steam engines, boilers, and attendant machinery into wooden hulls involved-risks which the traditional shipbuilder did not always fully consider and which the average ship captain was not always equipped to handle. About 14 percent of all steam vessels suffered boiler explosions, and between 1853 and 1855 alone, explosions destroyed nineteen such vessels.

Moreover, ship captains experienced other difficulties as what seemed an inordinate number of sailing ships were lost in that period. Concern grew when it became apparent that the United States Steamboat Inspection Service was inadequate to cope with these dangerous trends. Thus, John D. Jones, president of the Atlantic Mutual Insurance Company, led a group of New York City marine insurance companies in forming a not-for-profit, benevolent purpose society to promote the safety of life at sea. Chartered in 1862, its name was the American Ship Masters Association and its primary function initially was to conduct examinations and issue certificates of competency to masters and mates.

However, the need to rate ships as well as masters and mates soon became evident, and in 1867 the association took on this function as well.

The technical function of rating ships, known by the term classification, grew to such importance that just before the turn of the century it became the association's primary mission. To properly reflect its new emphasis, the association's name was changed in 1898 to the American Bureau of Shipping CABS);it ceased issuing certificates of competency two years later.

In 1869, the predecessor organization to ABS published its first ship register, a register that has been published yearly ever since. Titled the *Record,* this volume contained pertinent characteristics of all vessels in ABS classification.

The first ABS Rules, those for *Building and Classing Wooden Vessels,* were published in 1870. It was realized that to be truly effective, it would be necessary for these rules to keep pace with developments and properly reflect progress. When wood gave way to iron as a popular building material, ABS published the *Rules for Building and Classing Iron Vessels* in 1877; and when iron gave way to steel, ABS published the *Rules for Building and Classing Steel Vessels* in 1890. These latter standards have been published annually in updated editions ever since, thereby providing owners, builders, naval architects, marine engineers, and the marine industry in general with minimum standards for design, construction, and periodic survey of ships and other marine structures. Each yearly edition of these "steel rules" is an embodiment of service experience and technological advancement accumulated since the first edition was published over a centuryago.

In the early 1900s, ABS remained a regional class society, but then two significant developments led to a growth in its size and stature. First, in 1916 it bought the Great Lakes Register and incorporated it into a department of ABS. This organization had been founded at the turn of the century for the registry and classification of vessels in Great Lakes service. Second, in 1917 an agreement was made with the British Corporation for the Survey and Registry of Shipping that enabled reciprocal use of each organization's surveyors. The terms of the agreement also allowed ABS to adopt certain of the British Corporation's rules for the construction of steel ships, considered to reflect the best standards and practices of the time.

A substantial increase in activity was experienced by ABS during World War I when it classed the majority of ships built by the United States government to meet the demands ofthat conflict. During that time, ABS proved itself an irreplaceable institution on the marine scene. The American Bureau of Shipping was officially recognized by the United States government in the Merchant Marine act of 1920. The legislation required that, in work involving a classification organization, every United States government agency would turn to ABS. This was reaffirmed by the United States Congress in 1983.

During World War I, ABS started to become more widely known and more active on the international scene. Indicative of this was its participation in the first International Load Line Convention of 1930 (load line being the depth, or draft, to which a ship can be safely loaded). At that time, as part ofthe United States delegation, ABS presented technical aspects of load lines that remain the basis of those now in effect.

In the 1930s, the art of welding was being developed into a science, and it held great potential for shipbuilding applications. Mter carefully studying this process, in 1931 ABS accepted its use in boiler construction. A few years later, all boilers constructed for the Maritime Commission's World War II shipbuilding program were to be in accordance with the ABS requirements. The 1936, ABS rules were the first among classification societies to accept welding for application to all parts ofthe hull and to stipulate detailed requirements for electrode approval. In 1937, ABS approved the automatic welding process and began classing tankers in excess of 18,000 deadweight tons-supertankers of their day-with tank spaces of all-welded construction using this process.

Prior to the start ofWorld War II, the United States Maritime Commission's long-range program included the well-known C-1, C-2, and C-3 cargo ships. During the war years, 868 such vessels of 5,000,000 gross tons were built to ABS classification. At the same time ABS also participated in the famous Liberty ship program. In some cases these ships were built in as little as four and a half days; at such a rate ABS surveyors were kept quite busy in their survey efforts.

The demands on shipyards during the first half of the 1940s were as unrelenting as they were on ABS. It was a time when the activity and growth of ABS were unprecedented. A total of 2,710 Liberty ships of 19,500,000 gross tons were built toABS class, as were 531 Victory ships of over 4,000,000 gross tons and 525 T-2 Tankers of 5,400,000 gross tons. From the start of World War II through 1945, a total of 5,171 vessels of various types were built to ABS classification as part of the maritime commission's wartime shipbuilding program. This set the stage for the next era of ABS-its emergence into becoming a truly international organization.

In 1939 ABS employed 92 exclusive surveyors, but at the height of wartime shipbuilding activity in 1944 the survey staff size increased to some 480 in order to survey the ships building and those in class. This staff was stationed mostly in the United States although some surveyors served in Europe. Following the war, ABS representation expanded to all continents as the United States government sold hundreds of ABS class surplus vessels to allies who had lost a large proportion of their fleets.

However, there is more to being an international classification society than having exclusive surveyors on worldwide station. In keeping with its then newly-established international status, ABS realized it would be essential to enlarge its network to include national technical committees. Thus in the early 1950s, ABS commenced adding committees representing leading shipbuilding nations. This afforded ABS close contact with interests in various geographical regions and with various advancements in technological and scientific disciplines in those regions. Through this means, all segments were afforded a participating voice in the ABS rules ensuring that they would be authoritative, international, and impartial.

ABS rules are integral to the classification procedure. These rules are industry-derived standards and in order that they be meaningful and current, a committee networ~ of individuals eminent in various technical disciplines associated with the marine and allied industries meets on a regular basis (without compensation) to consider the merits of newly-proposed rules and modifications to existing ones.

The promotion ofthe safety oflife and property has remained the central focus of ABS. While this focus is unchanged, the knowledge, experience, and resources of ABS have continuously progressed. Today, ABS provides ship classification services on a worldwide basis, with nearly 850 exclusive surveyors and engineers stationed in 168 locations around the world and on call at any hour.

ABS RULES-ESTABLISHMENT AND ADMINISTRATION

Representation of fitness may be considered as the core of classification, and it follows that rules provide the foundation to which this core is anchored. An understanding of classification then is best approached through an appreciation of these rules that have two fundamental and distinct aspects-one is the process by which rules are established and updated and the other is the procedure by which rules are administered.

Establishment of Rules

ABS Rules are derived from principles of naval architecture, marine engineering, and other allied engineering and scientific disciplines that have proven satisfactory by service experience and systematic analysis. ABS develops and updates its rules through a committee structure comprised of fifteen special technical committees and ten national and regional technical committees. Each committee and panel is composed of eminent naval architects, marine engineers, underwriters, owners, builders, operators, material manufacturers, machinery fabricators, and individuals in other related fields.

Through these committees, ABS maintains direct contact with various scientific disciplines as well as technical activities and advancements in various geographical regions worldwide. Moreover, these committees generate feedback that is vital in making the rules effective. In this way, each committee represents a cross section of views without bias toward any special interest. All members of the ABS committees serve without compensation. Because of this committee system, the ABS rules are authoritative, impartial, and current.

Throughout the rule development process, the ABS staff plays a major contributing role rooted in its expertise acquired from a multitude of daily technical involvements with design review and analysis, detailed service histories of innumerable vessels classed over the years, and findings from many various ambitious ABS research projects.

Administration of Rules

ABS classification is a four-step procedure involving

- technical plan review
- survey during construction
- acceptance by the classification committee
- subsequent periodic surveys for maintenance of class

When an owner first requests that the vessel or structure be classed, the shipyard or design agent presents design drawings and calculations to ABS for a systematic detailed review for compliance with the rules. ABS engineers review the plans to verify that the structural and mechanical details conform to the rule requirements. Their review may also include sophisticated analytical procedures employing one of the many ABS computer programs. In this way, ABS is able to determine whether the design is adequate in its structural and mechanical concept and, therefore, suitable for production. During the entire review process, ABS is available for consultations with the owner and designer.

Mter a design has been reviewed by ABS engineers and found to be in conformance with the rules, ABS field surveyors "live with the vessel" at the shipyard from keel-laying to delivery to verify that the approved plans are followed, good workmanship practices are applied, and the rules are adhered to in all respects. During the construction of a vessel built to class, ABS surveyors witness, at the place of manufacture or fabrication, the tests of materials for hull and certain items of machinery and equipment, as required by the rules. They also survey the building, installation, and testing of the structure and principal mechanical and electrical systems. Throughout the time of construction, ABS maintains an ongoing dialogue with the owner and shipyard to make sure the rules are understood and adhered to and also to assist in resolving differences that may arise.

When completed, a vessel undergoes sea trials attended by an ABS field surveyor to verify that the vessel performs according to rule requirements. The vessel's "credentials" are then presented to the ABS Classification Committee (ABS members who are appointed from the maritime industry, U.S. Coast Guard, and ABS officers) which, based on collective experience and recommendations from the ABS staff, assesses the vessel's compliance with the rules. Provided all is in order, the vessel is accepted into class and formal certification is issued.

Periodic Surveys

Though a new vessel may be granted ABS classification and thereby be judged fit for its intended service, such status is not automatically retained throughout its service life. As the rigors of sea can be wearing on a vessel's hull and machinery, the society conducts periodic surveys to determine whether a vessel is being maintained in a condition worthy of retaining classification status.

As specified in the rules, in order to maintain class, the owner must present his vessel to ABS on a periodic basis for survey of hull and machinery items. Also, should there be any reason to believe that an ABS classed vessel has sustained damage that may affect classification status, it is incumbent upon the owner to so inform the society. ABS surveyors would then survey the vessel to determine what repairs, if any, would be necessary to retain classification.

Chronology of Major Periodical Class Surveys and Brief Descriptions of Each for Conventional Self-propelled Vessels

Note: Because of the evolving nature of the rules, the comments that follow are of necessity very general. For details of particular survey requirements, the latest edition of the ABS *Rules for Building and Classing Steel Vessels,* Part 1, "Classification, Testing and Surveys" must be consulted. A copy may be obtained from any local ABS port office.

ANNUAL CLASSIFICATION SURVEYS OF HULL AND MACHINERY

Annual class surveys of hull and machinery are to be carried out within three months either way of each annual anniversary date of the crediting of the previous special periodical survey of hull or original construction date.

The Annual Survey of Hull is a general examination of the vessel's hull and closing appliances. All accessible parts of the hull (especially those that are normally subject to rapid deterioration) should be included within the scope of the survey. The surveyor may broaden or restrict the scope based on his evaluation of the vessel, but the intent of the rules must be complied with.

The Annual Survey of Machinery is intended to establish by visual external examination that the machinery and machinery spaces are being maintained in satisfactory and safe operating condition. In particular, however, the steering arrangements should be carefully examined.

INTERMEDIATE SURVEY

The Intermediate Survey is to be carried out either at or between the second and third annual classification surveys.

SPECIAL SURVEYS OF HULL AND MACHINERY

A special periodical survey is to be completed within five years after the date of build or after the crediting date of the previous special periodical survey.

Special surveys may be carried out on a continuous basis over five years instead of at the end of the fifth year. If carried out on a continuous basis, approximately 20 percent of the required survey items should be carried out each year.

Special Survey-Hull

The purpose of the Special Survey-Hull (which includes the requirements of the annual survey and the drydocking survey) is to examine all parts of the hull structure including the hull, rudder, anchors and chains, decks, and watertight bulkheads. In addition, all tanks-double-bottoms, deep, ballast, peak, and cargo-and all holds, tween decks, pump rooms, pipe tunnels, duct keels, machinery spaces, dry spaces, cofferdams, and voids are to be examined internally. Thickness measurements are required, the extent of which will be based on the age of the vessel and the condition of the coatings. For tankers and bulk carriers, close-up survey (normally within hand's reach) will be required for ballast and cargo tanks and cargo hold structure. As part of the survey, all tanks are to be tested with a head of liquid depending on the use of the tank, i.e., water ballast or fuel oil.

Special Survey of Machinery and Electrical Equipment

The rules intentionally do not detail the extent of the survey of the various parts of the main vessel's machinery that are required to be reported on for the special survey, not only because of the difficulty in providing for all the many kinds and combinations in use, but so as to permit the surveyor sufficient latitude to apply good engineering judgment in confirming the proper ongoing maintenance and fitness for continued operation under the applicable special survey arrangement. The rules intend, however, that all items affecting the propulsion, steering, essential services, and safety of the vessel be covered. Equipment such as drinking water pumps, ship service air compressors, sanitary pumps, air conditioning apparatus, and similar items are normally not subject to anything more than a general examination except insofar as they might represent a hazard to the vessel; however, if a sanitary pump is also used as a fire pump, or a service air compressor is arranged also for starting diesel engines, for combustion controls, for automation controls, or for supplying the sootblowers, it is considered as vital equipment and must be so surveyed.

On certain vessels, where duplicate machinery in excess of rule requirements is carried, questions may arise as to just how much of the machinery is to be examined. In general, any machinery or equipment that is connected to any system that may be used for vital service is subject to survey. In addition, nonvital equipment that may affect the fitness of the vessel or its essential equipment, or that represents a possible hazard to the safe operation of the vessel may be subject to survey. As noted in the rules, safety and protective devices, such as overspeed trips and relief or safety valves, must also be checked and confirmed for special survey because failure or maloperation of these could result in serious damage to the ship.

The amount of opening up of machinery and equipment to be recommended by the surveyor is influenced by such things as the general quality of the vessel's maintenance (evidence of leaks, dirty bilges, smoke); the age or operating history of the equipment in respect to wear-and-tear or wastage; the severity of service in which the item is used; the frequency of use (whether intermittent, continuous, or emergency); for pumps, whether the pumped fluid is erosive, corrosive, abrasive, or a lubricant; results of spot checks or examinations of similar units; performance test or condition monitoring results (output, temperature, vibration, noise, input power); data from the engineers' maintenance records (crankshaft deflections, clearances, trends); and problem history of the same or similar items. A vessel that appeared to be managed well-with clean bilges, few leaks, smooth-running machinery, satisfactory readouts on condition monitors, and a good record system indicating that adequate preventive maintenance had been carried out-would obviously not require the same degree of opening up as a ship with poor management indications.

DRYDOCKING

A Drydocking Survey is to be carried out two times in any five-year period with an interval not exceeding three years between drydocking surveys.

The vessel is to be in drydock and the keel, stem, stern frame, rudder, propeller, and outside of side and bottom plating are to be cleaned as necessary and examined together with bilge keels, thrusters, exposed parts of the stern bearing and seal assembly, sea chests, rudder pintles and gudgeons and their respective securing arrangements. The stern bearing clearance or weardown and rudder bearing clearances are to be ascertained.

TAILSHAFT SURVEYS

1. Water-lubricated bearings
   - with continuous liners, survey intervals to be every five years if design incorporates stress reducing features
   - all other shafts, survey intervals of three years for single screw, four years for multiple screw
2. Oil-lubricated bearings
   - survey intervals of five years.

   Note: During the survey the propeller is to be removed and the shaft drawn in and examined in its entirety including an examination of the taper-end by a surface crack detection method.

BOILER SURVEYS

1. Water-tube for propulsion
   - more than one boiler, survey interval not to exceed two and a half years
   - single boiler, survey interval not to exceed two and a half years for the first seven and a half years, thereafter annually
2. Fire-tube for propulsion.
   - all at four and six years old, thereafter annually
3. Auxiliary
   - survey interval not to exceed two and a half years with a provision for extension up to six months

Note:

1. At each survey the boilers, superheaters, and economizers are to be examined internally (water-steam side) and externally (fire side).
2. Boiler mountings and safety valves are to be examined at each survey and opened as considered necessary by the surveyor.
3. The proper operation ofthe safety valves is to be confirmed at each survey.

DAMAGE SURVEYS

A damage survey must be done at any time vessel's hull or machinery sustains damage that affects or could affect vessel's classification. It is the owner's responsibility to call for survey.

MARITIME ADMINISTRATIONS-FLAG STATE

Maritime administrations represent the interests of a sovereign state for the purpose of regulating shipping and shipping-related activities. Nations that have a mechanism for registering tonnage, called a registry, are commonly referred to as flag states. Hence, the maritime administration of a nation is responsible for determining what regulations apply to vessels in its registry and for effecting inspection and certification of those vessels.

Internationally accepted regulations usually apply to oceangoing vessels, whereas vessels operating in national waters (e.g., inland rivers, lakes, etc.) are usually regulated by national regulations. National regulations may also be used to supplement areas of international regulation that are not felt to be sufficiently prescriptive or leave details of application to the discretion of the administration.

Role of Flag State

The flag state is responsible for the following:

1. Developing and determining maritime regulations: Maritime regulations can be of domestic origin (national laws) or international in nature, and

their applicability is usually based upon the vessel's size/tonnage or geographical trading areas.

2. Representing its nation at international maritime forums: In carrying out this role, the flag administration represents the maritime interests ofthe nation at such forums as the International Maritime Organization.
3. Maintaining its registry (e.g., registering vessels): Vessels that qualify for entry are duly registered and documented by the flag state.
4. Applying regulations to registered tonnage: After vessels are duly registered, the flag state administers applicable regulations to those vessels.
5. Providing inspection/certification service for registered vessels: The flag state must provide certification services directly for its vessels or delegate the authority for these services to a capable technical body, such as a classification society.
6. Acting in accordance with relevant international regulations: The flag state must abide by international agreements to which it is party or signatory. Such agreements may require the flag state to provide an auditing function over its inspection and certificate function as well as compile and share information related to fleet statistics, accident investigations, and interpretations of regulations.

Delegation of Authority

Most flag states delegate the authority to survey and certify tonnage to qualified technical bodies. Usually, classification societies are the recipients of such delegations, and this delegation of authority is then recorded in a formal document that spells out the specific responsibilities of both parties. The classification society, as delegated party under such an agreement, is expected to provide timely, professional service, using criteria determined and interpreted by the flag state. The degree oflatitude that can be used by a class society as well as reporting obligations are usually contained in the delegation of authority.

Flag State Relationship with IMO

Of particular interest is the flag state's relationship with the International Maritime Organization (IMO), the body of the United Nations charged with regulating oceangoing tonnage by consensus means. Member states are those flag states that are members ofthe IMO and hence subject to its binding agreements. Nonmember states are those flag states that do not hold membership at IMO but that may follow proceedings as observer states and may voluntarily adopt IMO criteria as part of their maritime regulations. Signatory states are those member states of IMO who have signed into force IMO conventions and are thus bound by the convention's provisions. Nonsignatory states are member or nonmember states ofIMO that have not signed IMO instruments placing regulations into force, but have often voluntarily adopted IMO regulations and standards as part of their maritime requirements.

United States Coast Guard as a Flag State

The U.S. Coast Guard is generally recognized as one of the premier maritime safety organizations in the world, not only because of its size, but also because of the breadth of its responsibilities and activities. It traces its origin back to the Revenue Cutter Service in 1790, and over the last two centuries has continually broadened its responsibilities as new laws have been created in response to developing national and world issues or as various other governmental entities have been merged with it. The official name of "Coast Guard" was created in 1915. Despite the Coast Guard's assimilation of a great many diverse responsibilities, several other governmental agencies are concerned with specific aspects ofthe maritime affairs ofthe United States, such as the Maritime Administration (MARAD) and the Federal Maritime Commission (FMC). However, these organizations are primarily concerned with various chartering, leasing, and subsidy issues, and overseeing various trade agreements. With approximately 40,000 military and civilian employees, the Coast Guard is the largest organization in the U.s. Department of Transportation and exercises the traditional flag and port state responsibilities of the United States.

MISSIONS AND PROGRAMS OF THE UNITED STATES COAST GUARD

The U.S. Coast Guard has four traditional mission areas: maritime safety, environmental protection, law enforcement, and national security. In accomplishing these missions, the Coast Guard conducts the following twelve operating programs: Aids to Navigation, Boating Safety, Defense Operations, Environmental Response, Ice Operations, Maritime Law Enforcement, Marine Inspection, Marine Licensing, Marine Science, Port Safety and Security, Search and Rescue, and Waterways Management.

ORGANIZATION OF THE UNITED STATES COAST GUARD

The U.S. Coast Guard is directed by its commandant and the supporting program staffs in Washington, D.C., who develop policy, guidance, and implementation strategy. The programs of Environmental Response, Marine Inspection, Marine Licensing, and Port Safety Security, which are commonly referred to as the "M" programs, are directed by a rear admiral in charge of the Office of Marine Safety, Security, and Environmental Protection and his or her staff.

The Marine Safety Center (MSC), with a centralized technical staff, is a separate "M" subordinate command that provides day-to-day plan review responsibilities.

Authority then passes through an overall Atlantic and Pacific area commander to a smaller geographic entity called a district, which is responsible for implementation of all the various Coast Guard programs in this geographic area.

The field level Marine Safety Office (MSO) is the primary entity charged with carrying out the "M" programs in the various ports around the country. A specific MSO is assigned geographic responsibility for each segment of the U.S. coastline and inland rivers system as well as for U.8. flag vessel inspection issues worldwide. The commanding officer of the MSO also has the important titles of Officer in Charge of Marine Inspection (OCM!) for flag state responsibilities and Captain of the Port (COTP) and On-Scene Federal Pollution Coordinator (OSC) fpr port state responsibilities.

These titles, which are specifically used in the various U.S. Code laws and the U.S. Code of Federal Regulations, provide the direct link and legal basis for conducting the "M" programs. Inspection questions on policy implementation, clarifications, and appeals of Coast Guard actions/decisions would proceed through the MSO office to the District Commander's "M" staff and then back to the commandant's "M" staff for final resolution. Technical questions would proceed through the MSC to the commandant's "M" staff for final resolution.

BASIS OF UNITED STATES COAST GUARD AUTHORITY AND IMPLEMENTATION

The U.S. Coast Guard derives its authority from the laws of the United States. The basic statutory authority for the Coast Guard itself is found in Title 14 of the U.S. Code. U.S. laws are prepared by the U.S. Congress and signed by the president. They generally have a generic title and are numbered by the congress according to the numerical sequence oflaws passed. These public laws are generally not stand-alone documents, but rather they amend existing laws of the United States. These public laws are then compiled into their appropriate sections of the U.S. law and published annually in the updated U.S. Code. It is the U.S. Code that then becomes the viable reference document when referring to U.S. laws. Various titles of the U.S. Code address specific areas of particular interest to the mariner: Title 33 USC-Navigation; Title 46-Shipping; and Title 49-Transportation. Generally, the laws are broad in nature, and it is then the responsibility of the U.S. Coast Guard to carry out the intent of the law by developing regulations.

The Code of Federal Regulations (CFR) is the document that actually provides the specific requirements in the form of regulations to implement the law. Changes to existing regulations or new regulations are proposed and published daily in a document called the *Federal Register* (FR). Each year, the CFR is republished incorporating all changes from the previous year.

The *Marine Safety Manual* (MSM) provides detailed guidance as to how the Coast Guard itself will achieve the requirements of the laws and regulations. It contains organizational arrangements and relationships and subject matter explanations and guidance.

Navigation and Vessel Inspection Circulars (NVICs), which currently go back to 1956, are continually published and updated and provide additional guidance on specific detailed inspection and technical issues.

Finally, the "M" policy letters/memos file contains additional guidance addressing decisions on very specific questions which are not adequately addressed in the above documents.

MARITIME ADMINISTRATIONS-PORT STATE

A maritime administration, while representing the interest of a sovereign state by regulating its shipping, is bound by those international agreements to which it is signatory as a flag state. When vessels of another flag state (state "B") enter the waters of a particular flag state (state "A"), that entity, having an interest in upholding safety and pollution standards for all vessels (both foreign and domestic) trading in its waters, assumes a role as a port state. Hence, a vessel flying the flag of state "B" that enters the territorial waters of state "A" may be subject to intervention (inspection, etc.) by the latter as a port state. Furthermore, flag states may, under the provisions ofIMO conventions (SOLAS, MARPOL) board a vessel and conduct such inspections.

An increasing number of flag states are entering into agreements outside IMO to cooperate regionally in matters of port state control. This essentially reaffirms the right of these flag states to board and inspect vessels (of member states) entering their ports. Such agreements also provide a mechanism to effectively communicate among themselves during port state interventions and pool and disseminate their findings to other member states participating in the regional agreements. IMO regulations common to all member states usually are used as the basis for port state inspections. Should deficiencies be found during a port state inspection, a port state may elect to detain a foreign vessel until such time as the deficiencies have been corrected.

Examples of such agreements are the "Paris Memorandum of Understanding on Port State Control" to which fourteen flag states are signatory, the "Asia-Pacific Memorandum of Understanding," and the "Vina Del Mar Memorandum of Understanding."

Role of Port State

The maritime administration, while acting in its capacity as a port state, may board and inspect foreign vessels to confirm their compliance with applicable regulations. Normally, such regulations are applied from the relevant IMO conventions such as SOLAS, MARPOL, load line, or tonnage conventions. In cases where a flag state has additionally enacted specific national regulations that it applies to all applicable foreign vessels visiting, the usual procedure becomes more complex. While any maritime authority has the right to regulate foreign vessels entering its jurisdiction, the practice of adopting and applying unique requirements (unilateral requirements) is discouraged by the IMO. Despite this general policy, there are some flag states that have passed national legislation requiring foreign vessel compliance.

IMO has formally encouraged the creation of "regional" agreements among neighboring states. In fact, IMO, recognizing that such information may well be used to judge the performance of flag states in fulfilling their responsibilities under the conventions, is amalgamating the requirements presently contained in numerous IMO Resolutions, the objective being to standardize the port state inspections. This work will allow for a common database of deficiencies or detentions reported.

While acting as a port state, the administration may inspect a foreign vessel using its own government inspectors or other inspectors (such as class society surveyors) to whom it has delegated authority to act on its behalf. Usually, port state inspections are conducted by government inspectors. The flag state where the vessel is registered may be notified of such inspections as may the class society with which the vessel is classed.

Port state control inspections are carried out under the authority of international conventions and are carried out to ensure that foreign flag ships have current documents and certificates showing a vessel's compliance with relevant conventions; that the hull and machinery are fit for their intended service; that the vessel does not pose a pollution risk; and that a healthy and safe working environment exists for the crew.

Initially, the PSC inspection generally consists of a visit on board to verify that necessary certificates and documents are valid. The initial visit also gives the inspector an opportunity to judge the general appearance and condition of the vessel. When certificates are overdue or expired, or where there appear to be reasons to suspect that the ship and/or its equipment may not be in compliance with the relevant convention standards, a more detailed inspection is undertaken to determine whether the ship is substandard and/or not fit for service.

Grounds for carrying out a detailed inspection may consist of any of the following: A report or notification from another authority; report or complaint from the master, a crew member (or any person or organization with a legitimate interest in the safe operation of the ship or in the prevention of pollution); or the finding of serious deficiencies during the inspection.

During any such inspection, differences of opinion as to the interpretation of international regulations may develop. In any such instances, the flag state where the vessel is registered should be consulted for their interpretation of any applicable requirement. Where national criteria are applied by a port state, the interpretation obviously rests with the issuing authority.

If deficiencies are found that might affect safety, health, or the environment, the port authorities will ensure that the deficiencies are rectified before the ship is allowed to proceed to sea. Ifnecessary, they will sometimes detain the ship for that purpose, notifying the flag state ofthe action taken.

Where deficiencies cannot be remedied in the port of inspection, the port authority may allow the ship to proceed to another port for repairs, if they are satisfied that the ship can proceed without endangering the safety of the crew, their health, or the environment. In such cases, the port authority will notify the competent authority ofthe next port of call and the flag state of the action taken.

In recent years, PSC has escalated tremendously, particularly in the major maritime countries that at one time flagged a majority of the commercial fleet. However, with the proliferation of open registries (i.e., countries with ports having no significant maritime activity that offer reduced registration rates), the traditional maritime powers have lost their fleets and, therefore, flag state ability to ensure ship safety. Thus, PSC has taken on new significance, since it can provide a means to evaluate the performance of flag states. Furthermore, certain PSC regimes also weigh the performance of classification societies, owners, and operators.

INTERNATIONAL MARITIME ORGANIZATION (IMO)

The convention that founded the International Maritime Organization was adopted on March 6, 1948, by the United Nations Maritime Conference. The convention was then known as the Convention on the Inter-Governmental Maritime Consultative Organization, and it entered into force on March 17, 1958, thus establishing the IMCO. This new organization was inaugurated on January 6,1959, when the assembly held its first session. The name of the organization was changed to the International Maritime Organization on May 22, 1982, in accordance with an amendment to the convention that entered into force on that date.

When the United Nations Maritime Conference first met, it recognized that the most effective means to improve the safety standards of the international shipping community would be through an international forum devoted exclusively to maritime matters. Hence, the purposes of the organization, as stated in Article l(a) of the convention, are "to provide machinery for cooperation among Governments in the field of governmental regulation and practices relating to technical matters of all kinds affecting shipping engaged in international trade; to encourage and facilitate the general adoption of the highest practicable standards in matters concerning maritime safety, efficiency of navigation and prevention and control ofmarine pollution from ships."

Because the IMO is an international forum and not an executive body, it has no powers of enforcement or initiative. Instead, its member states have the power to initiate proposals, to conduct or commission research, and to implement decisions made with regard to maritime standards. The IMO Secretariat is limited to encouraging member states to address issues raised within the IMO.

Organization

The organization consists o'fan assembly, a council, and four main committees: the Maritime Safety Committee (MSC), Marine Environment Protection Committee (MEPC), Legal Committee, and Technical Cooperation Committee. There are also a number of subcommittees of the main technical committees, as well as a facilitation committee.

Given this structure, the *assembly* is the highest governing body ofthe IMO. It consists of all member states meeting every two years in regular sessions. Extra sessions may be held outside ofthe regular sessions, ifnecessary. The assembly approves the work program, votes the budget, and determines the financial arrangements of the IMO. The assembly also elects the council.

The *council* exercises the function of the assembly during the period between the assembly sessions. The council consists of thirty-two member states elected for two-year terms by the assembly. The IMO Convention requires that when electing the members of the council, the assembly shall comply with the following criteria:

1. Eight shall be states with the largest interest in providing international shipping services.
2. Eight shall be other states with the largest interest in international shipping.
3. Sixteen shall be states not etected under (1) or (2) above that have special interests in maritime transport or navigation, and whose election to the council will ensure the representation of all major geographic areas of the world.

Based on an amendment to the IMO Convention adopted by the assembly in November 1993, it is possible that the membership of the council will increase in number to forty. Groups (1) and (2) will be increased to ten, and group (3) will be increased to twenty. However, the amendment will go into force only once two-thirds of the member states accept it. It will then go into force twelve months later.

The council is the executive organ ofIMO and is responsible, under the assembly, for supervising the work of the organization. The council, between sessions of the assembly, carries out all the duties of the assembly except for making recommendations to governments on maritime safety and pollution prevention. These functions are the sole responsibility ofthe assembly. The following are the council's other functions:

1. Coordinate the activities of the organs of the organization.
2. Consider the draft work program and budget estimates of the organization and submit them to the assembly.
3. Receive reports and proposals of the committees and other organs and submit them to the assembly and member states, with comments and recommendations.
4. Appoint the secretary-general, subject to the approval of the assembly.
5. Enter into agreements or arrangements concerning the relationship ofthe organization with other organizations, subject to approval by the assembly.

The *Maritime Safety Committee* (MSC) is the most senior technical body of the organization. All member states are part of the MSC, and the functions of the MSC are to "consider any matter within the scope ofnavigation, construction and equipment of vessels, manning from a safety standpoint, rules for the prevention of collisions, handling of dangerous cargoes, maritime safety procedures and requirements, hydrographic information, logbooks and navigational records, marine casualty investigation, salvage and rescue, and any other matters directly affecting maritime safety."

MSC also has the responsibility to provide a mechanism to perform any functions assigned to it by the IMO convention or any duty within its scope of work that may be assigned to it by or under any international instrument and accepted by the organization. It is also required to consider and submit recommendations and guidelines on safety for possible adoption by the assembly.

MSC also operates with several subcommittees appropriately titled with the subjects with which they deal: Safety of Navigation (NAV); Radiocommunications and Search and Rescue (COMSAR); Standards of Training and Watchkeeping (STW); Dangerous Goods, Solid Cargoes and Containers (DSC); Ship Design and Equipment, including lifesaving equipment (DE); Fire Protection (FP); Stability and Load Lines and Fishing Vessel Safety (SLF); and Bulk Liquids and Gases (BLG). In April of 1993, a new subcommittee was formed to deal with the numerous problems flag states, particularly those associated with third world nations, experience when implementing the regulations of the various conventions. This new subcommittee is called the Flag State Implementation Subcommittee (FSI).

Like the MSC, the *Marine Environment Protection Committee* (MEPC) is also composed of all member states. However, MEPC is required to consider any matter within the scope of the organization concerned with prevention and control of pollution from ships. These duties include the adoption and amendment of conventions and other regulations and measures to ensure their enforcement. The subcommittees reporting to MSC also report to MEPC when addressing pollution matters.

Because of the legal issues involved in the organization's activities and work, the *Committee on Technical Cooperation* directs and coordinates this activity with the *Legal Committee.* These two committees are composed of all member states. Simplification and minimization of documentation in international maritime traffic is the responsibility ofthe *Facilitation Committee,* a subsidiary of the council. Participation in this committee is open to all member states ofIMO.

As stated earlier, the IMO *Secretariat* is limited to encouraging member states to address issues raised within the IMO. The secretariat orIMO consists of the secretary-general and nearly three hundred personnel based at the headquarters in London, United Kingdom. Presently, the secretary-general is William A. O'Neil of Canada. The past secretaries general have been

Ove Nielsen, Denmark	1959-1961
William Graham (acting), United Kingdom	1961-1963
Jean Roullier, France	1964-1967
Colin Goad, United Kingdom	1968-1973
Chandrika Prasad Srivastava, India	1974-1989
William A. O'Neil, Canada	1990-

IMO Codes and Conventions

To achieve its purposes of "developing the highest practicable standards in matters concerning maritime slfety, efficiency of navigation, and prevention and control of marine pollution from ships," IMO has developed and adopted nearly forty conventions and protocols as well as hundreds of codes and recommendations. The initial work performed on a convention is normally done by a committee or a subcommittee. The committee's work, a draft instrument, is then submitted to a conference to which delegations from all states within the United Nations system (including states that may not be IMO member states) are invited. The conference adopts a "final text" by "general consensus" rather than by vote. The final text is then submitted to governments for ratification.

A convention enters into force after fulfilling certain requirements that usually include adoption of the text at a United Nations conference followed by ratification by a specified number of countries. Generally, the more important the convention, the more stringent are the requirements for entering into force. For example, some conventions stipulate that 50

percent of the world's shipping by a minimum number of countries must ratify the conventions before they enter into force. Amendments to conventions are usually ratified differently. They enter into force through a "tacit acceptance" process. Member states are assumed to accept the amendment unless a specific reservation to the contrary is filed with the IMO Secretary. If rejections have been received within a specific time period from member states representing a minimum amount of world tonnage, the amendment will not enter into force.

Observance of the convention's requirements are mandatory for the countries that are party to it. On the other hand, codes (e.g., gas or chemical codes) are resolutions (i.e., recommendations) adopted by the assembly and are not as binding. Resolutions normally "invite" or "urge" participating governments to enact the contents through their own national requirements, preferably in their entirety and not partially.

Of the forty conventions and protocols adopted by IMO, the four that have probably had the most profound effect on international shipping are the International Convention for Safety of Life at Sea (SOLAS), International Convention for the Prevention of Pollution from Ships (MARPOL), International Convention on Load Lines (ICLL), and the International Convention of Tonnage Measurement of Ships (Tonnage).

SOLAS CONVENTION

As discussed earlier, IMO is principally concerned with safety at sea and mitigating the possibilities of marine environmental pollution. Of all the international conventions addressing maritime safety, the most significant is the International Convention for the Safety of Life at Sea (SOLAS). As will be discussed, the SOLAS Convention has undergone and will continue to undergo numerous revisions. Generally, the SOLAS Convention provides requirements that address six main categories of vessel safety: navigation, design, communication, lifesaving appliances, fire protection, and safety management.

Although the first version ofthe SOLAS Convention was adopted at the 1914 International SOLAS Conference, it never entered into force. Yet, four other versions of SOLAS were developed, adopted, and eventually entered into force. The second version was adopted in 1929 and entered into force in 1933. The third version was adopted in 1948 and entered into force in 1952. The fourth version was adopted in 1960 and entered into force in 1965. The latest version was adopted in 1974 (SOLAS 1974) and entered into force in 1980.

Each version enhanced the previous version's safety requirements and was based on the latest technology or marine accident investigations. For instance, the 1912 sinking ofthe ocean liner *Titanic* led to the development of the 1914 SOLAS Convention, which was then amended in 1929. Moreover, significant improvements to subdivision and stability standards, emergency services, structural fire protection, and collision regulations were included in the 1948 SaLAS Convention.

The 1960 SaLAS Convention was the first SaLAS Convention developed under IMCO. Numerous technical improvements were made for cargo ships requirements, including emergency power and lighting and fire protection. Six sets of amendments to the 1960 SaLAS Convention were adopted during the eight years following the convention's entry into force. These amendments included safety measures specific to tankers, automatic pilot requirements, and shipborne navigational equipment requirements, among others:

SOLAS 1974

At the present time, the convention that is applied is the 1974 SaLAS Convention. The following discussion provided a brief summary of the 1974 SaLAS Convention:

Chapter I provides the format of the certificates that are issued to signify compliance with SaLAS as well as the minimum survey periods. This chapter also empowers the port state to carry out port state control, which ensures that ships calling at their ports possess valid certificates and are in compliance with the SaLAS requirements. If ships are not in compliance, this chapter allows the port state to take appropriate action to detain the ship and notify IMO.

Chapter II-I addresses minimum extents of watertight integrity and subdivision governed by a probability of collision criteria. Extensive requirements for electrical and machinery installations and control systems are also included. These requirPJilents ensure that services essential to the safety of the vessel and its crew and passengers are maintained under normal and emergency conditions.

Chapter II-2 contains detailed fire safety provisions for various types of vessels based on the following principles:

- maintenance of thermal and structural boundaries
- separation of accommodation spaces
- limited use of combustible material
- fire detection in zone of origin
- fire containment and extinction in zone of origin
- protection of means of escape or firefighting access
- availability of firefighting appliances
- minimizing the possibility of cargo vapor ignition

Chapter III provides requirements for the amount and location of lifesaving appliances specific to each type of vessel, as well as details concerning the capacity and construction of the different lifesaving appliances.

Chapter IV provides for radio equipment specifications and operating obligations of the crew.

Chapter V provides for navigational requirements directed at the coast state as well as requirements for shipborne navigational equipment and pilot ladders.

Chapter VI provides stowage provisions when loading grain. Stability criteria particular to each loading condition are included, taking into account potential shifting of cargo and heeling moments.

Chapter VII delegates to contracting states the mandatory responsibility to adopt procedures for handling dangerous goods. For this purpose, this chapter refers to the International Maritime Dangerous Goods Code (IMDG).

Chapter VIII gives very basic principles concerning atomic radiation safety on nuclear ships (except ships of war) and refers to the International Atomic Energy Association for special control in ports.

Chapter IX requires that specific vessels and their shore-based operating company meet the requirements of the International Safety Management Code (ISM Code), which is contained in an assembly resolution. The resolution, which is based on the appropriate sections of the ISO 9000 series, calls for periodic inspections and maintenance of conditions to provide for safety and environmental protection.

Chapter X requires that craft greater than 500 gross tons, built after January 1, 1996, and more than four hours from a harbor of safe refuge, meet the provisions of the High Speed Craft Code (HSC). Two principles of the code are used to categorize requirements for the type of craft as either category "A" craft (allowing a reduction in passive and active protection of passengers, limited to 450, based on the ready availability of rescue assistance) or category "B" craft (requiring additional active and passive safety precautions to accommodate the unlimited number of passengers allowed and the unavailability of rescue assistance).

Chapter XI includes special measures to ensure maritime safety and requires enhanced surveys on bulk carriers and oil tankers as well as maintenance of survey report files and supporting documents (i.e., main structural plans of cargo and ballast tanks, repair history, and cargo and ballast history) on board.

Amendments and Protocols to SOLAS 1974

The 1974 SaLAS Convention has been amended several times by protocols and amendments. The following paragraphs provide a brief, chronological summary of the significant changes:

1978 Protocol tightened the survey periods and strengthened port state control requirements. Inert gas systems were required for new and certain existing tankers. Additionally, two independently operated radar systems and two remote steering gear controls were required.

1981 Amendments required completely separate steering gear control systems. It also included provisions for halon extinguishing systems, collision bulkheads on cargo ships, and specific navigational equipment to be on the bridge.

1983 Amendments provided requirements for separation of accommodations from machinery and other high risk spaces. Significant changes were introduced concerning lifesaving appliances including their design, capacity, and the use and placement of partially and totally enclosed life boats. Requirements for immersion suits and improvements in locating ship's survivors (EPIRBs, additional requirements for lifebuoys and life-jackets) were introduced. The amendments also introduced into chapter VII a reference to two new codes (Gas Carrier Code and Bulk Chemical Carrier Codes)

April 1988 *Amendments* focused on maintaining and monitoring the watertight integrity of passenger-Ro/Ro vessels in light of the *Herald of Free Enterprise* sinking.

October 1988 *Amendments* furthered requirements for damage residual stability, expanded the stability information supplied to the master, and required periodic (five-year intervals) lightweight surveys, based on the *Herald of Free Enterprise* disaster.

1988 Protocol, not yet in force in 1998, will harmonize the system of survey intervals relative to SOLAS, MARPOL, and Load Line Conventions, with a maximum five-year duration of certificates for cargo ships (one-year for passenger ships). It also will discontinue unscheduled inspections.

Additionally, after almost twenty years of work, the 1988 amendments to SOLAS 1974 concerning radiocommunications for the Global Maritime Distress and Safety Systel\} (GMDSS) entered into force in February 1992. These amendments base communication capabilities on the vessel's area of operation (rather than the vessel's tonnage) and phase out Morse Code, utilizing more advanced technologies offered by satellite communications.

1989 Amendments reduced the amount of openings in watertight bulkheads and required that power-operated sliding doors be fitted in all new passenger ships. Safety improvements in fire extinguishing, smoke detection, and separation of spaces containing fuel were included.

1990 Amendments changed the philosophy of evaluating damage stability and subdivision for dry cargo ships from the "deterministic" to the "probabilistic" method. These amendments provide a more realistic damage scenario based on statistical evidence.

1991 Amendments extended chapter VI ("Carriage of Grain in Bulk") to include storing and securing other cargoes, such as timber. Fire safety provisions to accommodate new passenger ship designs were also included.

1992 Amendments were somewhat of a landmark for IMO since they required significant improvements to be made to existing passenger and

passenger-Ro/Ro ships. Existing ships are required to meet the new requirements at stages between 1994 and 2010. Notable among the new requirements is the need for sprinkler systems and smoke detection systems in all accommodation and service spaces, stairway enclosures, and corridors; requirements for additional fireman's outfits; requirements for portable foam applicators ofthe inductor type; and requirements for a fixed fire extinguishing system in compliance with Regulation II-2/7 in machinery spaces of category A.

1994 Amendments added three new chapters to the SOLAS 1974 Convention. Chapter IX requires vessels and their operators to meet the requirements of the International Safety Management Code; chapter X introduces the High Speed Craft Code; and chapter XI addresses special measures to enhance maritime safety, which include requirements for enhanced surveys on bulk carriers and oil tankers.

June 1996 *Amendments,* among other items, extensively modified chapter III of SOLAS 1974. Requirements for marine evacuation systems are included in the revision as are requirements for antiexposure suits. The requirements for free-fall lifeboats were also thoroughly revised. Additionally, the regulations in chapter III that dealt with design and approval oflifesaving appliances were removed from(.1apter III and put into a separate, mandatory code, the International Lifesaving Appliance Code.

The June 1996 Amendments also require all oil tankers and bulk carriers built on or after July 1, 1998, to have in place an efficient corrosion prevention system in all dedicated seawater ballast tanks.

December 1996 *Amendments* are notable for the requirement for every tanker to be provided with the means to enable the crew to gain safe access to the bow even in severe weather. The amendments contain extensive revisions to chapter II-2, "Construction-Fire Protection, Fire Detection, and Fire Extinction," and also the adoption of the International Code for Application of Fire Test Procedures (FTP Code).

1997 Amendments added a new chapter to SOLAS 1974, chapter XII, "Additional Safety Measures for Bulk Carriers." The effective date of this new chapter is July 1, 1999. Regulations 4 and 6 in this chapter require all new bulk carriers of 150 meters and above in length that are of single-side skin construction and that are designed to carry solid bulk cargoes having a density of 1,000 kg/m^3 and above to have sufficient stability and strength when loaded to the summer load line to withstand the flooding of anyone cargo hold in all loading conditions and to remain afloat in a satisfactory condition of equilibrium. The aforementioned regulations also require all existing bulk carriers of 150 meters in length and above that are of single-side skin construction and that are designed to carry solid bulk cargoes having a density of 1,780 kg/m^3 and above have sufficient stability and strength when loaded to the summer load line to withstand the flooding of the foremost cargo hold in all loading conditions and remain afloat in a satisfactory condition of equilibrium. Other highlights of the new chapter include regulation 3, which lists the implementation schedule of regulations 4 and 6 for existing bulk carriers (constructed before July 1,1999), as well as regulation 9, which contains requirements for existing bulk carriers not capable of complying with the damage stability requirements of regulation 4.2 due to the design configuration of their cargo holds.

MARPOL CONVENTION

During the early 1900s, various countries introduced measures to control and deter discharges of oil within their coastal waters. Attempts had been made in the mid-1900s for internationally accepted standards for controlling oil pollution, but the Second World War interrupted progress prior to an agreement being reached. Based on the growing concern about the amount of oil being transported by sea, the United Kingdom organized an international conference on the subject in 1954. The conference culminated in the adoption of the International Convention for the Prevention of Pollution from Ships, which was transferred to IMO in 1958. The 1954 Convention, with amendments in 1969 and 1971, prohibited deliberate discharge in "special areas" and within fifty miles from shore, limited operational discharge elsewhere for tankers (15 ppm and 60 liters per nm) and other ship types (l00 ppm and 60 liters per nm), and limited the size ofVLCC tanks to provide some oil outflow limits in the event of collision or grounding.

MARPOL 1973

Concerned over the enormous growth of maritime oil transport and the adequacy ofthe 1954 OILPOL Convention, IMO decided to convene an international conference in 1969. In 1973, an entirely new convention was adopted, which was to enter into force twelve months after receiving ratification from fifteen states constituting 50 percent of the world gross tonnage. The convention contained administrative articles and five technical annexes. Annexes I and II are mandatory, but the remaining three annexes are optional. The following paragraphs summarize each of the annexes:

Annex I reduced by 50 percent the operational oily discharges to Yao,ooo of the cargo. Similarly, it stated that bilges from machinery spaces had to contain less than 100 ppm of oil and could not be discharged within twelve miles from land, Discharge of "oil" was expanded to include sludge, refuse, and refinements, and discharge of oil was completely prohibited in ecologically sensitive "special areas." Furthermore, equipment requirements were placed on all ships of 400 gross tons and above such that they were required to have oily-water separating equipment. Constraints were also imposed on tankers and their arrangements, thus requiring onboard residue retention facilities, "load-on-top" operations, tank size limits, segregated ballast tank (SBT) arrangements for tankers of 70,000 tons deadweight and above, and compliance with side and bottom damage standards.

Annex II contained discharge criteria and measures for control of pollution by noxious liquid substances (NLS) carried in bulk. Substances were divided into four categories according to the hazard they presented to the marine environment, to human health, or amenities. Retention span and toxicity levels were used to categorize over 250 substances. Moreover, the regulations in this annex were weighted based on the substance's category, and they addressed onshore reception, onboard retention facilities, discharge limitations, and tank arrangements.

Annex III (optional annex) addressed ships carrying harmful substances in packaged form, such as containers, portable tanks, and rail tanks. This annex provided requirements for quantity limits, packaging, marking, stowage, and documentation of harmful substances categorized by the International Maritime Dangerous Goods Code (IMDG Code).

Annex N (optional annex) prohibited sewage discharge within four miles of land unless it was treated by an approved treatment plant. Furthermore, any sewage discharged between four and twelve miles from land must first be pulverized and treated prior to discharge.

Annex V (optional annex) provided minimum distances for the discharge of domestic and operational waste, other than those wastes previously addressed by any other annex ofthe MARPOL Convention. The discharging of plastics is completely prohibited under Annex V of MARPOL.

1978 MARPOL Protocol

The 1973 MARPOL Convention never entered into force due to technical difficulties associated with implementing Annexes I and II. Because amendments could not be made to a convention that had not entered into force, a protocol was developed. The 1978 MARPOL Protocol, which ,entered into force in October 1983, absorbed the 1973 MARPOL ConventIOn, while changing the requirements of Annex I and allowing a three-year implementation period for contracting states to solve the technical problems associated with Annex II. Because of this action, the 1978 Protocol and the 1973 MARPOL Convention are referred to as one treaty: MARPOL 73/78.

The changes made to Annex I by the 1978 Protocol included limits on hypothetical oil outflow requiring segregated ballast tank (SBT) arrangements to protect cargo tanks in the event of collision or grounding for all new tankers of 20,000 tons deadweight and above (previously 70,000 tons deadweight). Existing tankers allowed the use of crude oil ,:,ashing (CO,w) as an alternative to SBT, provided an inert gas system IS used dunng washing operations. A second interim alternative to SBT or COW allowed existing tankers to use dedicated clean ballast tanks (CBT) for two to our years (depending on the vessel's size) after MARPOL 73/78 entered mto force. CBT arrangements required the identification of dedicated tanks to sl~ly carry ballast, but transfer of ballast could be made through cargo plpmg systems.

Amendments to MARPOL 73/78

Like the 1974 SaLAS Convention, MARPOL 73/78 has been amended on sv~e.ral occasions. The following paragraphs chronologically highlight the sIm~cant amendments, some of which have had a far-reaching impact on shlppmg.

1984 Amem~ents af!ected Annex I of the convention only. The signifi-cat~ changs~ It Imposed mcluded providing oily-water discharge and moni-torng~ eqUIpment ~rovi"Sions to limit or restrict discharge; permitting carnage of ballast m cargo tanks under emergency conditions to ensure ~dequate.strength; r~ducing slop tank size from 3 percent of the oil carry-mg capacIty ~f the shIp to 2 percent under certain conditions; limiting dis-charge of. oIly waste from drilling operations to 100 ppm; and strengthenmg of damage stability requirements to enhance a tanker's sur-vivability.

!985 A~endment~ recognized that the end of the two-to-four-year grace penod for Implementmg Annex II was nearing and that changes would be neee~~ to facilitate practicable application. These changes included har-~olllzmg survey requirements with Annex ɪ, further restricting the car-nage of ~ategory ~ and C substances, requiring prewashing of cargo tanks, mandatmg complIance with the International Maritime Dangerous Goods Code (IMDG Code), and mandating compliance with the International Bulk Chemical Code (IBC Code).

1987 Amendments, October 1989 *Amendments, and* 1991 *Amendments* further defined ecologically sensitive "special areas" under Annexes I and V, respectively.

March 1989 *Amendments* mandated compliance with the Bulk Chemi-cal Code, which is applicable to existing ships, although it was not manda-tory under SaLAS 1974. Also, substances listed in Annex II were again updated.

. *1990 Amendments* harmonized survey requirements of MARPOL 73/78 wI~h the SaLAS and Load Line Conventions. These harmonized survey re-~UIrements are k~own as the Harmonized System of Survey and Certifica-tlO.n (HSSC). UnlIke the latter two conventions, which required a protocol to I~trod~~e this harmonization, an amendment under the "tacit" approval regIme WIll enter these MARPOL amendments into force six months after the similar amendments (protocols) to the SaLAS and Load Line Conven-tions enter into force.

1991 Amendments now require that in the event of failure of the oil discharge monitoring and control system, the defective unit shall be ~a~e operable as soon as possible. These amendments also prohibit any plpmg to and from the sludge tanks to have any direct connection over-

board other than the standard discharge connection. Finally, these amend-ments require ships (oil tankers of 150 gross tons and above and other ships of 400 gross tons and above) to have a Shipboard Oil Pollution Emer-gency Plan (SOPEP) on board, and they revised the format of the Oil Re-cord Book.

1992 Amendments added new regulations 13F and 13G to Annex 1. These regulations are perhaps the most significant changes to MARPOL 73/78 yet.

The first new regulation, 13F, applies to new tankers, as defined by these amendments. New tankers of 5,000 tons deadweight and above must be fitted with either a double-hull or a mid-deck design. Other methods of design and construction of oil tankers may also be accepted as alternatives to the aforementioned designs, provided that such methods ensure at least the same level of protection against oil pollution in the event of collision or stranding and are approved by the committee, MEPC.

Regulation 13F also sets minimum wing tank widths and minimum double-bottom heights that are dependent on the tanker's deadweight. With some minor exceptions for short lengths of piping, this regulation also prohibits ballast and other piping, such as sounding and vent piping to ballast tanks, from passing through cargo tanks and prohibits cargo piping and similar piping to cargo tanks from passing through ballast tanks.

On the other hand, the requirements of regulation 13G, effective July 6, 1995, apply to crude oil tankers of 20,000 tons deadweight and above and to product carriers of 30,000 tons deadweight and above. Nonsegre-gated ballast tankers must either comply with the requirements ofregu-lation 13F not later than twenty-five years after their delivery date or be phased out. An additional five years of operation may be gained if the ves-sel has SBT and COW or 30 percent of the cargo block is protected with wing tanks or double-bottom spaces that are not used for the carriage of oil.

Again, other structural or operational arrangements may be accepted as an alternative to the double-hull requirements, provided such arrange-ments ensure at least the same level of protection against oil pollution in the event of collision or stranding and are approved by the flag administra-tion.

Finally, regulation 13G requires an enhanced program of inspection during special, intermediate, and annual surveys to be implemented. An oil tanker over five years old to which this regulation applies shall have on board a complete file of survey reports, scantling gaugings, a statement of structural work carried out, and a structural condition evaluation report.

As can be seen from the above discussion, the 1992 Amendments will have a profound effect on tanker design and construction, and especially on existing tankers in the years to come.

1996 Amendments will form a new annex to the convention, Annex VI, "Air Pollution from Ships." The amendments will include requirements for fuel oil quality, use/discharge of ozone depleting substances, machinery discharges of nitrogen and sulfur oxides, incinerator discharges, and reception facilities.

INTERNATIONAL LOAD LINE CONVENTION

History

In 1875, English legislation passed a requirement that a mark be placed on the vessel's side to' prevent overloading. As accident investigations came under increased scrutiny and monitoring, underwriters and the Lloyd's Register of Shipping became concerned with issues such as reserve buoyancy, watertight integrity, hull strength, stability, and safe working conditions on deck for the crew. Subsequently, two governments (British and German) established rules embracing these principles. Other maritime nations soon adopted their own sets of similar standards. Britain, seeing the increase of international trade during the early 1900s, invited maritime governments to participate in a conference to develop international standards for all vessels operating internationally. However, due to World War I, the conference's objectives were not met until 1930, which saw the completion of the first International Convention of Load Lines, 1930 (ICLL). The concerns previously mentioned were covered by this convention and served the maritime industry for thirty-eight years. Taking advantage of IMO's wealth of international and technical expertise concerning marine safety, which was not available during development of the 1930 ICLL, maritime governments set goals to develop a new convention on load lines to consider the almost four decades of technological advances that had occurred in the marine industry. This culminated in the development of the 1966 ICLL under the management of IMO.

1966 International Load Line Convention

The principal provisions of the 1966 ICLL can be categorized by three areas: survey requirements, conditions of assignment, and minimum geometric freeboard.

The survey requirements included in the convention, which call for initial, annual, and renewal surveys, ensure that the vessel's structure, fittings, and appliances, as addressed by the convention, are maintained in an effective condition. Furthermore, the convention issued the conditions of assignment that must be met prior to the vessel being assigned a freeboard and issued a Load Line Certificate to embody the following areas: master's information, weathertight integrity, and protection of the crew.

Information to be supplied to the master consists of a loading manual to assess the vessel's stresses and longitudinal bending moments as well as a trim and stability booklet that assesses the stability ofthe vessel for various loading conditions. Weathertight integrity provisions address the closing arrangements, minimum sill heights, and structural integrity of the closure for ventilators, airpipes, companionways, hatches, scuppers, and other openings that penetrate the hull and provide possible sources ofwater ingress. Lastly, protection of crew addresses requirements necessary to ensure safe passage of the crew about the main deck. These requirements include location, spacing and height of guardrails, gangways, and lifelines. Requirements for sufficient accessibility to crew accommodations are also addressed.

A major part of the ICLL are the regulations to determine the minimum geometric freeboard for a vessel. The criteria is empirically based considering several geometric and hydrostatic parameters ofthe vessel relative to providing sufficient reserve buoyancy to resist capsizing and alleviating the buildup of water on deck to minimize the potential for water ingress. These requirements have remained intact since their inception. Yet, there is movement at IMO by some members to reconsider the requirements comprising the minimum geometric freeboard and perhaps use analytical simulations and model tests to determine the vessel's seaworthiness (in terms of water on deck for certain sea conditions). Given the other convention's (SOLAS 1974)requirements that address water ingress and sufficient amounts of reserve stability in terms of intact stability and subdivision requirements, the objectives ofthe minimum geometric freeboard may also be satisfied by a more realistic assessment ofthe vessel's stability characteristics. This is presently seen in IMO's development of dynamic-motion-response-based guidelines that assess the amount of water shipped for containerships without hatch covers.

1969 TONNAGE CONVENTION

Virtually all flag states require that before a ship is registered, it must be measured in accordance with its national tonnage regulations to ascertain gross and net tonnage. Determination of a vessel's tonnage is necessary since the figures are used to determine the applicability of international and national regulations, port fee charges, manning requirements, and ship's identification.

Existing national tonnage regulations were derived from the British tonnage measurement system dating back to the British Merchant Shipping Act of 1854. As many maritime states adopted this measurement system, conflicting interpretations and amendments unique to individual states led to considerable differences worldwide in its application. Reciprocal agreements among some maritime states alleviated some of the differences but not all. Consequently, various attempts were made to standardize a system of tonnage measurement that could be used by all maritime states. However, it was not until an investigation was carried out under the guidance of the IMO that any progress was achieved. The IMO's work resulted in the 1969 Tonnage Convention.

Interim Scheme

The 1969 Tonnage Convention entered into force on July 18, 1982, and it applies to all vessels whose load line length is 79 feet or greater. Vessels constructed after July 18, 1982, fell under an "interim scheme" developed by the IMO. Under the interim scheme, existing commercial vessels greater than 24 meters needed to be provided with tonnage values in accordance with the 1969 Tonnage Convention before July 18, 1994.

However, utilization and publication of the 1969 gross tonnage values falls into three categories, dependent on the date the vessel's keel was laid:

(a) Vessels built prior to December 31, 1985: The 1969 gross tonnage is indicated only on the 1969 International Tonnage Convention Certificate. The gross tonnage calculated under national laws existing at the time of the vessel's build will continue, for the life of the vessel, to be indicated on statutory certificates issued under the SaLAS and MARPOL Conventions and the International Convention of Standards of Training, Certification, and Watchkeeping, 1978 (STCW). Thus, the national tonnage is applied when determining compliance with the requirements of these conventions.

(b) Vessels built between December 31, 1985, and July 19, 1994: The policy indicated in (a) above was extended to apply to vessels that have been issued the SaLAS Radiotelephony Certificate for the life of the vessel and to vessels with the MARPOL 73/78 International Oil Pollution Prevention Certificate until July 18, 1994.

(c) Cargo vessels less than 1,600 gross tons (national tonnage figures) built after December 31, 1985: The policy indicated in (a) above applied until July 18, 1994.

Although the interim scheme allows certain vessels to be excluded from the additional requirements that would otherwise have been imposed due to an increased gross tonnage, the scheme does not address the setting of port fees.

Convention Regulations

The 1969 Tonnage Convention consists of only seven regulations. These regulations establish gross and net tonnage based on relatively simple formulas. The formulas use parameters that, unlike previous national requirements, are independent of each other. Moreover, since measurements are now taken from molded lines (versus previous measurements

that provided allowances for framing depth), the tonnage can be calculated from plans in the initial stages of construction.

The *gross tonnage* is dependent only on the volume (1 ton equals 100 ft^3) of the vessel's enclosed spaces (i.e., spaces bounded by the hull, fixed or portable bulkheads, partitions, awnings, etc.). Also, any space fitted with either shelves, a means of securing cargo, openings with a means of closure, or an arrangement that may allow closure shall also be included in the enclosed volume.

The *net tonnage* is a function of three vessel characteristics that are independent of the enclosed volume used for determining gross tonnage. These characteristics are cargo space molded volume, draft/depth ratio, and number of passengers. The draft is that determined under the 1966 ICLL or national requirements, the deepest subdivision draft, or 75 percent of the molded depth. Passengers exclude the master and crew and are discretely based on cabin capacity being greater than or less than eight berths. In no case shall the net tonnage be 30 percent of the gross tonnage.

This new system of measurement has, to some extent, specifically affected new ships, such as Ro/Ros or those vessels of the open shelter deck types, since their gross tonnage increased by as much as 200 percent. Especially affected were those vessels that had gross tonnage less than 1,600 tons. Many of the international conventions base the application of their requirements on gross tonnage. Hence, many owners found their vessels had to comply with additional regulations with the increase in tonnage.

To counteract this disadvantage, IMO provided the interim scheme, which allows a vessel built before December 31, 1985, to use her previously calculated tonnage when applying the SOLAS Convention Regulations.

Vessels can change their net tonnage (i.e., due to changes in draft or passenger carrying capacity), but only once per year. Additionally, any modifications in cargo capacity and principal dimensions necessitates new tonnage measurements. It should also be noted that for oil tankers (e.g., double-hull), the gross registered tonnage (GRT) as calculated under the 1969 Tonnage Convention can be calculated excluding the tonnage of tanks used exclusively to carry segregated ballast water. For open-top containerships, a formula to reduce the GRT has been provisionally accepted to lessen the economic disadvantages for these types of vessels.

QUALITY ASSURANCE FOR SHIP MANAGEMENT

ISO 9000 Series Standard

The International Organization for Standardization was founded in 1946 to develop a common set of international standards for manufacturing, trade, and communication, and is commonly referred to as ISO. According to ISO officials, the name ISO was taken from the Greek word, *isos,* meaning equal or uniform or standard.

ISO is based in Geneva, Switzerland, and is composed of ninety-two member countries, including the United States. The organization that represents the United States in ISO is the American National Standards Institute (ANSI). ANSI in turn appoints organizations to administer technical advisory groups (TAG) which represent ANSI in the various technical committees in ISO. The American Society for Quality (ASQ) administers the Technical Advisory Group (TAG) to ISO Technical Committee 176 for Quality Management and Quality Assurance Standards. The ISO Technical Committee 176 (TC 176) developed the ISO 9000 Series Standards on Quality Systems and is responsible for revisions and further development of the standards.

All standards developed by ISO are voluntary and there are no internationallegal requirements to force individual countries to adopt them. However, at present, eighty countries have adopted the ISO 9000 series standards and some countries, such as the thirteen countries that compose the European Union (EU), have attached legal requirements to these standards for some regulated products. The standards have been adopted in the United States as voluntary standards and are known as ANSI! ASQ Q 90 Series.

BACKGROUND FOR QUALITY SYSTEM STANDARDS

The Department of Defense (DOD) developed MILQ 9858, a quality management program that became a contract requirement for companies doing business with the Department of Defense (DOD).

Using the MILQ 9858 quality management program as a basis, the North Atlantic Treaty Organization (NATO) in 1968 developed the NATO AQUAP series of standards. In 1979, the United Kingdom's British Standards Institution (BS!) used these standards to develop commercial quality assurance system standards that are known as BS 5750 series.

The International Organization for Standardization (ISO) decided to develop international series standards for quality systems and formed the Technical Committee 176 (TC 176) in 1979 to accomplish this task. Using the above standards for guidance, the committee developed the ISO 9000 series standards that were published by ISO in 1987. It is interesting to note that with the adoption of the ISO 9000 series standards by the U.S. Department of Defense (DOD) and the North Atlantic Treaty Organization (NATO), the quality system standards have come full circle.

The basic ISO 9000 series standards published in 1987 consisted of five standards:

1. Three conformance standards: ISO 9001, 9002, 9003.
2. Two guidance standards: ISO 9000 and 9004.

ISO standards are reviewed for revision approximately every five years. Consequently, the ISO 9000 series standards were revised and republished in 1994.

The three conformance standards maintained the same numbering and are titled as follows:

ISO 9001	Quality system: model for quality assurance .in design, development, production, installation, and servicing.
ISO 9002	Quality system: model for quality assurance in production, installation, and servicing.
ISO 9003	Quality system: model for quality assurance in final inspection and test.

The two guidance standards numbering was changed and are titled as follows:

ISO 9000-1	Quality management and quality assurance standards: guidelines for selection and use.
ISO 9004-1	Quality management and quality system elements: guidelines.

The addition of part numbers permits their use for additional guidance standards and to date a number of additional guidance documents have been published or are being developed. The following is a list of the ISO 9000 series standards that have either been published or are being developed at the present time.

ISO Std. No.	*Description*
9004-2	Guide for services
9004-3	Guide for processed materials
9004-4	Quality improvement
9004-8	Quality principles
10005	Quality plans
10006	Project management
10007	Configuration management
10011-1	Auditing guidelines
10011-2	Auditor qualifications
10011-3	Audit program management
10012-1	Measurement equipment management
10012-2	Measurement assurance
10013	Quality manuals
10014	Economics of quality
10015	Education and training
10016	Inspection/test records

Since the standards are continually reviewed, revised, and supplemented, the guidance standard numbers may be changed in the future.

ANSI should be contacted for a current listing of the ISO 9000 series and related standards.

Headings ofthe requirements ofISO 9001-1994 Standard are as follows:

4.1	Management responsibility
4.2	Quality system
4.3	Contract review
4.4	Design control
4.5	Document and data control
4.6	Purchasing
4.7	Control of customer-supplied product
4.8	Product identification and traceability
4.9	Process control
4.10	Inspection and testing
4.11	Control of inspection, measuring, and test equipment
4.12	Inspection and test status
4.13	Control of nonconforming product
4.14	Corrective and preventive action
4.15	Handling, storage, packaging, preservation, and delivery
4.16	Control of quality records
4.17	Internal quality audits
4.18	Training
4.19	Servicing
4.20	Statistical techniques

The only difference between ISO 9002 and 9001 is that ISO 9002 does not contain a requirement for design control (4.4). The differences between ISO 9003 and 9001 are that ISO 9003 does not contain requirements for the following:

4.4	Design control
4.6	Purchasing
4.9	Process control
4.19	Servicing

In addition, the following requirements in ISO 9003 are less comprehensive than the requirements in ISO 9001:

4.1	Management responsibility
4.2	Quality system
4.8	Product identification and traceability
4.10	Inspection and testing
4.13	Control of nonconforming product
4.14	Corrective and preventive action
4.16	Control of quality records
4.17	Internal quality audits
4.18	Training
4.20	Statistical techniques

ACCREDITATION AND REGISTRATION

The following discussion applies to the accreditation and registration processes in the Unites States for ISO 9000 standards.

The American Society for Quality (ASQ) established the Registrar Accreditation Board (RAB) as an affiliate of ASQ in 1989 to develop a national accreditation program jointly with ANSI. The program is called the ANSIIRAB American National Accreditation Program for Registrars of Quality Systems. RAB accredits registrars that comply with the requirements of the accreditation program. To assure compliance, RAB performs initial audits and regular follow-up surveillance audits. Registrars who comply with the requirements are issued certificates of accreditation and are listed in a directory of accredited registrars which is maintained by RAB.

Registrars who are accredited may then evaluate the quality system of a company wishing to be registered to an ISO 9000 standard. If the registrar finds that the company's quality system complies with the standard, a registration certificate is issued by the registrar that is recognized by RAB. The registration certificate is good for a period ofthree years, during which the registrar conducts surveillance audits to determine that the company's quality system continues to comply with the ISO 9000 standards.

In addition to accreditation of registrars, RAB has established a program for certification of auditors that consists of training and audit experience. Auditors who comply with the requirements are issued a certificate as a provisional auditor, auditor, or lead auditor, depending on their level of training and experience.

The RAB accreditation and registration process is shown in figure 15-1.

Discussions are currently being held in ISO to develop an international registration system that would permit recognition of certificates around the world. In the meantime, accrediting bodies are attempting to develop systems that would permit recognition of each other's registration certificates.

SHIP MANAGEMENT AND THE ISO 9000 SERIES STANDARDS

The quality system standard that is most adaptable to ship management is ISO 9002. As noted above, ISO 9002 does not contain a requirement for design, which is included in ISO 9001. ISO 9002 is the standard to which ship management companies are being registered. Since ship management is engaged in providing a service, the guidance standard ISO 9004-2, "Guidelines for Services," should be referenced in conjunction with ISO 9002. ISO 9004-2 provides guidance for establishing and implementing a quality system within an organization specifically for services. The quality

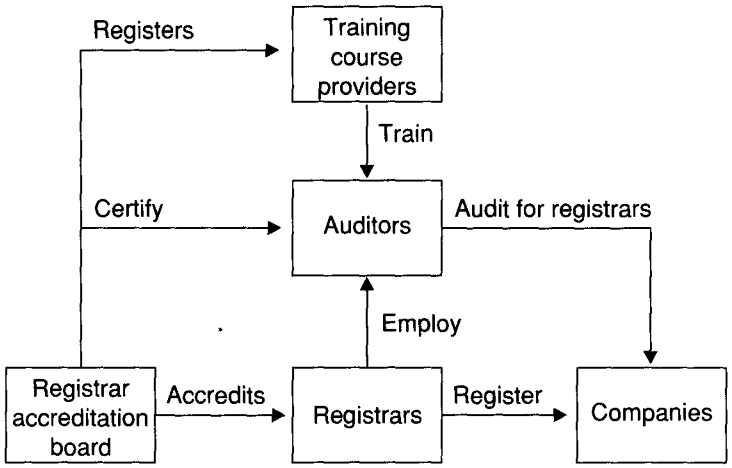

Figure 15-1. RAB accreditation/registration process

system elements are applicable to all kinds of service including the marine industry.

When developing an ISO 9002 quality system for registration, a company should review each element of the standard carefully to determine which elements are applicable to the operation of the company. In addition, compliance with the many requirements that are applicable to the marine industry are to be considered. Some of these are codes, standards, rules, and regulations that are developed by organizations such as the International Maritime Organization (IMO), the International Ship Managers Association (ISMA), the International Labor Organization (ILO), classification societies, flag states, and port states.

A list of the requirements of ISO 9002 indicating the responsibility of the shoreside management and the shipboard staff to develop and implement the requirements follows:

ISO 9002 Requirement		*Primary Responsibility*
4.1	Management responsibility	Shoreside management
4.2	Quality system	Shoreside management with assistance from shipboard staff in developing system levels procedures and work instructions
4.3	Contract review	Shoreside management
4.4	Design control	Not applicable to ISO 9002
4.5	Document and data control	Shoreside management with assistance in implementation from the shipboard staff
4.6	Purchasing	Shoreside management
4.7	Control of customer supplied product	Shoreside management with assistance from shipboard staff
4.8	Product identification and traceability	Shoreside management
4.9	Process control	Shoreside management with assistance in implementation from the shipboard staff
4.10	Inspection and testing	Shoreside management with assistance in implementation from the shipboard staff
4.11	Control of inspection, measuring and test equipment	Shipboard staff
4.12	Inspection and test status	Shipboard staff
4.13	Control of nonconforming product or service	Shipboard staff
4.14	Corrective and preventive action	Shoreside management and shipboard staff
4.15	Handling, storage, packaging, preservation, and delivery	Shipboard staff as appropriate
4.16	Control of quality records	Shoreside management and shipboard staff
4.17	Internal quality audits	Shoreside management and shipboard staff
4.18	Training	Shoreside management and shipboard staff
4.19	Servicing	Not applicable to a ship management system
4.20	Statistical techniques	Shoreside management and shipboard staff

An ISO 9002 quality system involves the development of the documentation necessary to describe the system and implementation of the system that is described in the documentation. The documentation used to describe the quality system can be found in a quality manual, system level procedures, work instructions, and quality records.

1. The quality manual should provide a description of the system as a permanent reference.
2. System level procedures usually describe activities across various departments and define how these activities are conducted, controlled, and recorded.
3. Work instructions are detailed instructions for an individual or group to describe how a specific task is to be accomplished.
4. Quality records provide information on the results of the quality systems for review and improvement of the system. Quality records are to be defined, readily retrievable, and retained for a designated period.

The International Management Code for the Safe Operation of Ships and for Pollution Prevention

This code is also known as the International Safety Management Code (ISM Code) and was adopted by the International Maritime Organization (lMO) on May 2, 1994. In addition, "Guidelines on Implementation of the ISM Code by Administrations" were adopted by IMO resolution A.788 (19) on November 23, 1995.

The purpose of the ISM Code is to provide an international standard for the safe management and operation of ships and for pollution prevention.

BACKGROUND

In the past, IMO has adopted a number of resolutions having objectives similar to the ISM Code. Some of these resolutions are as follows:

Resolution A.441 (xi) invited every state to take the necessary steps to ensure that the owner of a ship that flies the flag ofthat state provides such state with the current information necessary to enable it to identify and contact the person contracted or otherwise entrusted by the owner to discharge his responsibilities for that ship in regard to matters relating to maritime safety and the protection ofthe marine environment.

Resolution A.443 (xi) invited governments to take the necessary steps to safeguard the shipmaster in the proper discharge of his or her responsibilities in regard to maritime safety and the protection ofthe marine environment.

Resolution A.680(17) recognized the need for appropriate organization of management to enable it to respond to the need of those on board ships to achieve and maintain high standards of safety and environmental protection. This resolution was adopted on November 6, 1991.

Resolution A. 647(16) "Guidelines on Management for the Safe Operation of Ships and for Pollution Prevention" was adopted on October 19, 1989.

Resolution A. 741(18) is the present ISM Code and was adopted on November 4, 1993.

THE IMO/ISM CODE REQUIREMENTS

The ISM Code is mandatory for all passenger ships and the following vessels of 500 gross tonnage and over effective July 1, 1998:

- Oil tankers
- Chemical tankers
- Gas carriers
- Bulk carriers
- Cargo high speed craft

For the following vessels, the ISM Code becomes mandatory on July 1, 2002:

- All other cargo ships
- Self-propelled mobile offshore drilling units (MODUs)

To demonstrate compliance with the ISM Code, companies should develop a safety management system (SMS). When compliance is demonstrated, companies will be issued a document of compliance (DOC) for the company and a safety management certificate (SMC) for each ship. The DOC and the SMC certificates are valid for a period of five years. Administrations of flag states may recognize organizations to issue these certificates on their behalf when the dates for compliance are reached. At this time in the United States, the American Bureau of Shipping can evaluate organizations for compliance to the ISM Code and issue the appropriate documentation.

The headings of the requirements of the ISM Code are as follows:

1. General
   1.1 Definitions
   1.2 Objectives
   1.3 Application
   1.4 Functional requirements for a safety management system (SMS)
2. Safety and environmental protection policy
3. Company responsibilities and authority
4. Designated person(s)
5. Master's responsibility and authority
6. Resources and personnel
7. Development of plans for shipboard operations
8. Emergency preparedness
9. Reports and analysis of nonconformities, accidents, and hazardous occurrences
10. Maintenance of the ship and equipment
11. Documentation
12. Company verification, review, and evaluation
13. Certification, verification, and control

When a company is developing a quality system to comply with ISO 9002, the requirements of the ISM Code should be incorporated into the ISO 9002 Quality System.

Code of Ship Management Standards of the International Ship Managers Association (ISMA Code)

The International Ship Managers Association (ISMA) is a nonprofit body with its registered office in Cyprus and its secretariat in London.

BACKGROUND

A group of leading ship management companies met in 1988 to explore ways to correct the proliferation of comments in the press about poor standards of ship management worldwide. A committee was formed to develop a "Code of Ship Management Standards." Included in the code were the requirements in ISO 9002 and IMO Resolution A.680(17), among other requirements. The code was distributed to managers worldwide and a meeting was held in London on April 29 and 30, 1991, in which thirty-five companies became founding members of ISMA.

The code specifies requirements for quality assured ship management and operation. The requirements apply to both shore-based and shipboard management. Among the objectives of the code are "operating the ship and transporting cargo safely and efficiently" and "conserving and protecting the environment."

Verification of compliance with the code is carried out by classification societies that have entered into an agreement with ISMA. Companies that have successfully completed the assessment will be issued a certificate by ISMA.

THE ISMA CODE REQUIREMENTS

In addition to technical requirements, the ISMA Code includes such requirements as business ethics, insurance, and accounting.

The headings of the requirements of the ISMA Code are as follows:

1. General
2. Business ethics
3. Organization
4. Personnel
5. Safety
6. Environmental protection
7. Contingency planning
8. Operational capability
9. Cost efficiency/purchasing/contracting
10. Maintenance/maintenance standard
11. Technical support
12. Insurance
13. Accounting
14. Certification and compliance with rules and regulations
15. Cargo handling and cargo care
16. Communication procedures
17. Management agreement
18. Records
19. Auditing body
20. Quality system
21. Document control
22. Internal quality audits

With the advent of the above codes and standards, the internationa: shipping industry has joined the worldwide movement to quality assurance. As this movement gains momentum, shipping companies are finding that compliance and certification to these codes and standards are providing them with valuable management tools.

REVIEW

1. What is the purpose of a flag state, and what role does it play in its maritime community?
2. What type of certification services does the flag state ensure are provided for its vessels? (Hint: SOLAS, MARPOL, load line, tonnage, etc.)
3. Describe the flag state's relationship with IMO.
4. What is the mission of the U.S. Coast Guard? Ofthe twelve programs to accomplish its mission, which are most important to the mariner and why?
5. Which office in the U.S. Coast Guard is responsible for vessel inspection? What else is this office responsible for?
6. Describe the difference between the U.S. Code and the Code of Federal Regulations, and which titles of the U.S. Code are of particular interest to the mariner? List their titles.
7. What is the function of the port state, and why have numerous flag states been entering into agreements to cooperate regionally? Name some of these agreements.
8. What role is IMO playing in port state control?
9. What is the purpose of IMO? Describe the organization. Which committee considers the construction and equipment of vessels, and which committee considers the prevention and control of pollution from ships?
10. How do conventions enter into force, and how do amendments to conventions generally enter into force?
11. Of all the conventions, which has probably had the most significant effect on international shipping? Why?
12. Which chapter in SOLAS 1974, as amended, addresses requirements for fire safety provisions? Watertight integrity and subdivision? Lifesaving requirements?
13. Describe the different annexes ofMARPOL 73/78. Which are mandatory?
14. What is the significance of the 1992 Amendments to MARPOL 73/78, and how will these amendments affect the oil tanker industry in the future?
15. What are the weathertight integrity provisions addressed in the 1966 International Load Line Convention? What are the crew protection provisions addressed in this convention?

16. Describe the three categories, based on the vessel's keel date, for the utilization of the 1969 Tonnage Convention.
17. Desribe gross and net tonnage as calculated under the 1969 Tonnage Convention.
18. What organization represents the United States in the International Organization for Standardization?
19. What is MILQ 9858?
20. What is the difference between an ISO conformance standard and a guidance standard?
21. Why is the ISO 9002 standard usually applied to shipping companies?
22. How many elements are in the ISO 9001 and ISO 9002 standards as revised in 1994?
23. How does a shipping company achieve certification to the ISO 9002 standard?
24. What is the highest level of documentation in a management system that complies with the ISO 9002 standard?
25. What international organization adopted the ISM code?
26. What organization is responsible for certifying compliance to the ISM code?
27. What is the objective ofthe ISM code?
28. Are the ISO 9000 standards and the ISM code similar in structure? Is it feasible to integrate them into a single management system?
29. What is the purpose of the ISMA code?
30. What shore-based organization is needed within a shipping company to implement these codes? What organization is needed aboard ship?

REFERENCES

ABS Marine Services. 1994. "Marine Management and Ship Operation Guideline ISO 9002/ISM Code." Houston, Tex.: ABS Marine Services.

Duncan, William, and W. M. van Zharen. 1994. "Environmental Risk Assessment and Management in the Maritime Industry: The Interaction with ISO 9000, ISM, and ISM Management Systems." SNAME *Transactions,* vol. 102. Jersey City, N.J.: Society of Naval Architects and Marine Engineers (SNAME).

International Maritime Organization (IMO). 1994. "Focus on IMO." London: IMO.

International Organization for Standardization (ISO). 1994. *ISO Standards Compendium,* 5th ed. Geneva, Switzerland: ISO.

Tiratto, Joseph. April 1991. "Preparing for EC 1992 in the U.S. Through Quality System Registration." Institute of Electrical and Electronics Engineers (IEEE) *Computer Magazine.*

Tiratto, Joseph. 1995. "Registrar Accreditation." *ISO 9000 Handbook,* 2d ed. Fairfax, Va.: CEEM Information Services.

APPENDIX TO CHAPTER 15

ISM Code Interpretations

The material in this appendix is from "Management System Audits For Ship Operators-The Auditor's Experience" by J. R. Gray and M. D. Sims, published in the Institute of Marine Engineers *Transactions,* volume 109, part 3, and reproduced here with the permission of the Institute of Marine Engineers.

The ISM Code is planned to be non prescriptive. As a result, there are opportunities for inconsistencies among interpretations of the involved parties. The International Association of Classification Societies (lACS) and the International Chamber of Shipping (lCS) have provided their members with guidelines to the interpretation of the ISM Code. Following is a summary of the code, with a clause by clause commentary by lACS and ICS appearing on facing pages.

ISM Code

1 **General**

1.1 **Definitions**

1.1.1 'International Safety Management (ISM) Code' means the International Management Code for the Safe Operation of Ships and for Pollution Prevention as adopted by the Assembly, as may be amended by the Organization.

1.1.2 'Company' means the Owner of the ship or any other organization or person such as the Manager, or the Bareboat Charterer, who has assumed the responsibility for the operation of the ship from the Shipowner and who on assuming such responsibility has agreed to take overall the duties and responsibility imposed by the Code.

1.1.3 'Administration' means the Government of the State whose flag the ship is entitled to fly.

1.2 **Objectives**

1.2.1 The objectives of the Code are to ensure safety at sea, prevention of human injury or loss oflife, and avoidance of damage to the environment, in particular to the marine environment, and to property.

1.2.2 Safety management objectives of the company should, inter alia:

1.2.2.1 provide for safe practices in ship operation and a safe working environment;

1.2.2.2 establish safeguards against all identified risks; and

1.2.2.3 continuously improve safety management skills of personnel ashore and aboard ships, including preparing for emergencies related both to safety and environmental protection.

1.2.3 The safety management system should ensure:

1.2.3.1 compliance with mandatory rules and regulations; and

1.2.3.2 That applicable codes, guidelines and standards recommended by the Organization, Administrations, classification societies and maritime industry organizations are taken into account.

1.3 **Application**

The requirements of this Code may be applied to all ships.

1.4 **Functional requirements for a Safety Management System** (SMS)

Every Company should develop, implement and maintain a Safety Management System (SMS) which includes the following functional requirements:

1.4.1 a safety and environmental protection policy;

1.4.2 instruction and procedures to ensure safe operation of ships and protection of the environment in compliance with relevant international and flag state legislation;

1.4.3 defined levels of authority and lines of communication :between, and amongst, shore and shipboard personnel;

IACS

1.1 Definitions

Additional definitions covering:

a. Safety Management System
b. Safety Management Audit
c. Observation
d. Objective evidence
e. Nonconformity
f. Major nonconformity
g. Finding

1.2 Objectives

Nil differences in interpretation

1.3 Application

Nil differences in interpretation

1.4 Functional Requirements

1.4.1/2 All requirements ofthe Code are to facilitate that safe practices are taken into account through written procedures and work instructions.

ICS

Nil interpretations.

Nil interpretations.

ISM Code

1.4.4 procedures for reporting accidents and nonconformities with the provisions of this Code;

1.4.5 procedures to prepare for and respond to emergency situations; and

1.4.6 procedures for internal audits and management reviews.

2 **Safety and environmental protection policy**

2.1 The Company should establish a safety and environmental protection policy which describes how the objectives, given in paragraph 1.2, will be achieved.

2.2 The Company should ensure that the policy is implemented and maintained at all levels of the organization, both ship-based as well as shore-based.

3 **Company responsibilities and authority**

3.1 If the entity who is responsible for the operation of the ship is other than the owner, the owner must report the full name and details of such entity to the Administration.

3.2 The Company should define and document the responsibility, authority and interrelation of all personnel who manage, perform and verify work relating to and affecting safety and pollution prevention.

3.3 The Company is responsible for ensuring that adequate resources and shore-based support are provided to enable the designated person or persons to carry out their functions.

4 **Designated person(s)**

4.1 To ensure the safe operation of each ship and to provide a link between the Company and those onboard, every Company, as appropriate, should designate a person or persons ashore having direct access to the highest level of management. The responsibility and authority of the designated person or persons should include monitoring the safety and pollution prevention aspects of the operation of each ship and to ensure that adequate resources and shore-based support are applied, as required.

5 **Master's responsibility and authority**

5.1 The Company should clearly define and document the master's responsibility with regard to:

IACS	*ICS*
1.4.5 The auditor would expect to find risks identified, and contingency arrangements documented.	
2.0 Safety and Environmental Protection Policy Responsibility of the Company to define and document the safety management objectives which form an integral part of the SMS. Responsibility of the auditor to verify the completeness, adequacy and effective functioning of the SMS. Where a Company has chosen to set objectives and standards in excess of the Code, then any audited deficiencies against such Company requirements should not be raised as an NCN. Such deficiencies shall be logged as 'Finding.'	The policy should be concise and clear. Should describe the aim of SMS and outline plan of action to achieve that aim. A strategy for implementation should be considered at the time the policy is developed, including how best to ensure that all employees understand its content.
3.0 Company Responsibility and Authority 3.1 Holder ofthe DOC is the company who has responsibility for the operation of the Ship. If subcontractors provide a service to the company, eg manning, then the SMS must identify such subcontractors and their role with respect to defined elements ofthe Code. Any activity which is part of the SMS and which is subcontracted continues to remain the responsibility of the company.	Personnel concerned with the SMS, on shore and sea should be given clearly worded definitions of their responsibilities and authority. Senior management should ensure that all shore and sea personnel are qualified and experienced.
3.2 Organigrams, diagrammatic charts, job descriptions etc may be used for each functional area both ashore and afloat.	The use of diagrammatic charts of the organization should be considered to show how defined responsibilities of shore and sea personnel interrelate.
4.0 Designated Person(s) Nil differences in interpretation.	
5.0 Master's Responsibility and Authority 5.1 The methods by which the Master is expected to carry out his responsibilities should be defined and documented.	5.0 Clear guidance on the Master's responsibility on matters affecting safety of the crew, the environment, the ship and cargo should be given.

	IBM Code	lACS	ICS
5.1.1	implementing the safety and environmental protection policy of the Company;	5.1.1 Effective implementation ofthe safety and environmental protection policy should be checked under audit with the crew.	If senior management's commitment to the system is to be translated into effective action, masters should be given every assistance to implement the SMS. Any guidance given must be compatible with the Master's overriding authority and discretion to take whatever action he considers to be in the best interests of the vessel, crew and the marine environment.
5.1.2	motivating the crew in the observation ofthat policy;	5.1.2 Motivation could be achieved through regular meetings with the crew where members are encouraged to participate in the fulfilment of Company objectives.	
5.1.3	issuing appropriate orders and instructions in a clear and simple manner;	5.1.3 A demonstration of appropriate orders would be Master's Standing Orders, Night Order Book etc.	
5.1.4	verifying that specified requirements are observed; and	5.1.4 Master and officers should verifY that the specified requirements ie Procedures, Work Instructions etc are observed.	
5.1.5	reviewing the 8M8 and reporting its deficiencies to the shore-based management.	5.1.5 The Master should advise company of deficiencies in SMS and review same in accordance with company policy.	
5.2	The Company should ensure that the 8M8 operating onboard the ship contains a clear statement emphasizing the master's authority. The Company should establish in the 8M8 that the master has the overriding authority and the responsibility to make decisions with respect to safety and pollution prevention and to request the Company's assistance as may be necessary.	5.2 SMS should contain statement that the Master has overriding authority both in normal and extreme cases. See also IMO Resolution A.443 Om-Decision of the Shipmaster.	
6	**Resources and personnel**	*6.0 Resources and Personnel*	
6.1	The Company should ensure that the master is:		
6.1.1	properly qualified for command;	6.1.1 Certification must be valid and the Master experienced in the vessel type.	6.1 Same.
6.1.2	fully conversant with the Company's 8MB; and	6.1.2 The company should ensure that the master is informed of all requirements relating to safety and environmental protection.	6.1.2 Nil interpretations.
6.1.3	given the necessary support so that the master's duties can be safely performed.	6.1.3 Examples of support: a. Maintaining required complement. b. Responding to identified deficiencies.	6.1.3 Nil interpretations.
6.2	The Company should ensure that each ship is manned with qualified, certificated and medically fit seafarers in accordance with national and international requirements.	6.2 Company should have a system for selecting personnel especially if subcontracted.	6.2 Following should be considered when addressing manning: a. Vessel trade. b. Skills required by crew. c. Awareness of crew with respect to the SMS. d. Availability of records for qualifications and medical fitness.
6.3	The Company should establish procedures to ensure that new personnel and personnel transferred to new assignments related to safety and protection of the environment are given proper familiarization with their duties. Instructions which are essential to provide prior to sailing should be identified, documented and given.	6.3 'New Assignments' are -another ship -different job -promotion 'Essential Instructions' are those that define crew member's role within SMS, eg lifeboat station and responsibilities, fire station and responsibilities. 'Familiarization' practice would include: -onboard training -seminars ashore -overlap handovers -use of visual aids, videos etc.	6.3 The familiarization of the crew with SMS related duties is important from the point of view of maintaining continuity and effective performance levels.

ISM Code	IACS	ICS
6.4 The Company should ensure that all personnel involved in the Company's SMS have an adequate understanding of relevant rules, regulations, codes and guidelines.	6.4 Company should plan how to provide personnel involved in safety and pollution prevention with information on mandatory requirements.	6.4 ICS Guide lists major international conventions. Other relevant information and guidelines published by Class, industry organizations, etc should be made available to shore staff and crew.
6.5 The Company should establish and maintain procedures for identifying any training which may be required in support of the SMS and ensure that such training is provided for all personnel concerned.	6.5 Company should identify individual(s) having responsibility to define training needs for specific tasks.	6.5 Safety training drills should be carried out in accordance with the SMS. Results of drills, and analysis of accidents should be analyzed to assist in identifying any additional training or changes to the SMS. The company should consider the establishment of procedures for the conduct of refresher courses and on job training.
6.6 The Company should establish procedures by which the ship's personnel receive relevant information on the SMS in working language or languages understood by them.	6.6 The details and amount of documentation should be determined by what is necessary to ensure the crew can understand their respective roles.	6.6 Important that all procedures and instructions established in the SMS are written in a clear manner. Where contract crew agencies are used, copies of the relevant part ofthe SMS should be supplied.
6.7 The Company should ensure that the ship's personnel are able to communicate effectively in the execution of their duties related to the SMS.	6.7 Sufficient instructions in a suitable language need to be verified, as well as ensuring an understanding of them by the crew. This could be verified by witnessing an exercise.	6.7 The ability of the crew to communicate with each other should be reviewed at recruitment stage and during appraisals. Companies using crewing agencies should ensure that company requirements are understood and the agency should be monitored.
7 **Development of plans for shipboard operations** The Company should establish procedures for the preparation of plans and instructions for key shipboard operations concerning the safety of the ship and the prevention of pollution. The various tasks involved should be defined and assigned to qualified personnel.	*7.0 Development of Plans for Shipboard Operations* Procedures for key shipboard operations should have safety and pollution prevention as primary objectives, eg watchkeeping, loading, discharging, gas freeing, tank cleaning, confined waters navigation, passage planning, etc. Comprehensive list of applicable codes, guidelines and regulations. Examples of key Shipboard Operations: cargo shifting, collision, grounding, fire, flooding, heavy weather, pollution control, loss of propulsion.	Emphasis should be placed on preventative actions. Companies should identify key shipboard operations and issue instructions on the manner in which these are performed. Continuing supervision and verification of compliance with instructions is important. The ICS Guidelines append a list of typical key shipboard operations. Reference is also made to dividing key operations into 'Special Operations' and 'Critical Operations.'

	ISM Code	IACS	ICS
8	**Emergencies preparedness**	*8.0 Emergency Preparedness*	8.0 Shore based contingency plans may include: duties of personnel, procedures for mobilization of shore staff, communications, ship specific plans, checklists appropriate to the type of emergency, list of contact names and telecommunication details of all organizations that may be involved.
8.1	The Company should establish procedures to identify, describe and respond to potential emergency shipboard situations.	8.1 Examples of emergency situations: overloading, ballasting, deballasting, collision, abandon ship, man overboard etc. Duties and responsibilities of crew members in each emergency should be documented and methods of communication defined.	8.1 Shipboard contingency plans should take account of: Allocation of duties, action to regain control, procedures for requesting assistance, communication methods onboard, procedures for dealing with the media, maintaining communications with the shore.
8.2	The Company should establish programs for drills and exercises to prepare for emergency actions.	8.2 Drills should cover those required by statutory regulations and company defined emergency situations.	8.2 Contingency plans should be established on how to deal with emergencies associated with: ship damage, fire, pollution, injury to personnel, security, passengers, cargo etc. Emergency drills should be carried out for defined contingencies.
8.3	The SMS should provide for measures ensuring that the Company's organization can respond at any time to hazards, accidents, and emergency situations involving its ships.	8.3 The company should have available shoreside organizational structure, resources and equipment for responding to a shipboard emergency.	
9	**Reports and analysis of nonconformities, accidents and hazardous occurrences**	*9.0 Reports and Analysis of Nonconformities, Accidents, and Hazardous Occurrences*	9.0, 9.1 and 9.2 as per IACS. SMS should require the Master to report: accidents, hazardous occurrences, NCNs within the SMS, suggested modifications to the SMS. The company should have a system for recording, investigating, evaluating, and analyzing reports. Feedback to ships' crews of such analysis should be provided.
9.1	The SMS should include procedures ensuring that nonconformities, accidents and hazardous situations are reported to the Company, investigated and analyzed with the objective of improving safety and pollution prevention.	9.1 Records of non conformities, corrective actions, and internal audits should be provided by the company to the auditor(s).	
9.2	The Company should establish procedures for the implementation of corrective action.	9.2 The company should have procedures for responding to nonconformities.	
10	**Maintenance of the ship and equipment**	*10.0 Maintenance of the Ship and Equipment*	
10.1	The Company should establish procedures to ensure that the ship is maintained in conformity with the provisions of the relevant rules and regulations and with any additional requirements which may be established by the Company.	10.1 Maintenance of the ship and equipment should be in accordance with procedures based on conventions, flag, class and company policy. Objective evidence is required to demonstrate conformance with established maintenance requirements.	10.1 Procedures should be established which ensure maintenance, repairs and relevant surveys are carried out in a safe and timely manner. Procedures should include reference to the provision of tools, technical information, spare parts and supplies. These guidelines provide list of essential equipment, machinery and hull items that should be covered by maintenance procedures.
10.2	In meeting these requirements the Company should ensure that:		
10.2.1	inspections are held at appropriate intervals;	10.2.1 The Company should define the appropriate intervals and may be expected to justify them. There should be routine inspections of machinery, equipment, and structural integrity of the ship.	10.2.1 Company initiated inspections should be properly planned and carried out by competent and qualified personnel. The SMS should include ship safety inspection instructions.

ISM Code

10.2.2 any nonconformity is reported with its possible cause, if known;

10.2.3 appropriate corrective action is taken; and

10.2.4 records of these activities are maintained.

10.3 The Company should establish procedures in the SMS to identify equipment and technical systems, the sudden operational failure of which may result in hazardous situations. The SMS should provide for specific measures aimed at promoting the reliability of such equipment or systems. These measures should include the regular testing of standby arrangements and equipment or technical systems that are not in continuous use.

10.4 The inspection mentioned in 10.2 as well as the measures referred to in 10.3 should be integrated in the ship's operational maintenance routine.

11 **Documentation**

11.1 The Company should establish and maintain procedures to control all documents and data which are relevant to the SMS.

11.2 The Company should ensure that:

11.2.1 valid documents are available at all relevant locations;

11.2.2 changes to documents are reviewed and approved by authorized personnel; and

IACS

10.2.2 Shipboard personnel should be expected to correct NCNs whenever possible. NCNs can mean damage, malfunction, deficiencies concerning ship and equipment. NCNs should be documented, including a record of their correction. System should be in place to notify appropriate personnel both ashore and onboard of NCNs.

10.2.3 Company should have documented procedures for corrective action.

10.2.4 Records of inspections, maintenance, damages, defects and relevant corrective actions should be kept.

10.3 List of critical equipment and systems should be available, and periodicity off unction testing defined. Examples of equipmentJsystems: alarms and shutdowns, fuel oil systems, cargo systems, safety equipment, emergency equipment etc.

11.0 Documentation

Nil interpretation.

ICS

10.2.2 Maintenance nonconformities should be reported promptly and a finite time set for their rectification. The Master should be made aware of his responsibilities for reporting maintenance and repair requirements.

10.2.3 Procedures should ensure that reports are investigated and that corrective action is taken. The responsibility of persons dealing with these reports should be defined.

10.2.4 Signed originals of bona fide statutory certification and reports should be held onboard. Copies of certification, survey reports, and certificates for national requirements should be held ashore. Records should be retained onboard and ashore.

10.3 When critical equipment and systems onboard are identified, appropriate tests and other procedures should be developed to ensure functional reliability. The testing of standby equipment should assist in ensuring that a single failure does not cause the loss of a critical ship function.

11.1 The control of all documents and data relevant to the SMS is a vital element in the effectiveness of the system. A document control procedure should be established which allows personnel to identify the revision status and so preclude the use of out of date documentation. Care should be taken to limit the SMS documentation to that which adequately covers the application of the system.

11.2.1 The method of distrubuting documents and the place or person prescribed to keep them should be defined. The company should consider appointing a person ashore to be responsible for the control of documentation.

11.2.2 Procedures should be established to allow changes to be made to documentation in a controlled manner.

ISM Code

11.2.3 obsolete documents are promptly removed.

11.3 The documents used to describe and implement the SMS may be referred to as the 'Safety Management Manual.' Documentation should be kept in a form that the Company considers most effective. Each ship should carry onboard all documentation relevant to that ship.

12 Company verification, review and evaluation

12.1 The Company should carry out internal safety audits to verify whether safety and pollution prevention activities comply with the SMS.

12.2 The Company should periodically evaluate the efficiency and when needed review the SMS in accordance with procedures established by the Company.

12.3 The audits and possible corrective actions should be carried out in accordance with documented procedures.

12.4 personnel carrying out audits should be independent of the areas being audited unless this is impracticable due to the size and the nature of the Company.

12.5 The results of the audits and reviews should be brought to the attention of all personnel having responsibility in the area involved.

12.6 The management personnel responsible for the area involved should take timely corrective action on deficiencies found.

13 Certification, verification and control

13.1 The ship should be operated by a Company which is issued a document of compliance relevant to that ship.

13.2 A document of compliance should be issued for every Company complying with the requirements of the ISM Code by the Administration or by the Government ofthe country, acting on behalf of the Administration in which the Company has chosen to conduct its business. This document should be accepted as evidence that the Company is capable of complying with the requirements of the Code.

13.3 A copy of such a document should be placed onboard in order that the master, if so asked, may produce it for the verification of the Administration or organizations recognized by it.

13.4 A Certificate, called a Safety Management Certificate, should be issued to a ship by the Administration or organization recognized by the Administration. The Administration should, when issuing the Certificate, verify that the Company and its shipboard management operate in accordance with the approved SMS.

13.5 The Administration or an organization recognized by the Administration should periodically verify the proper functioning of the ship's SMS as approved.

lACS

12.0 Company Verification, Review and Evaluation

12.1 Internal audits should be carried out in the company and on each ship once per year. Records of Internal Audits and NCNs are to be made available to the auditor.

12.2 The management review of the 8MS should be initiated by: results of internal. audits, investigations as a result of accidents, results of implementation of the SMS, change of company policies/practice etc.

13.0 Certification, Verification and Control

13.1 Reference should be made to IMO Resolution A788(19)-Guidelines on the Implementation of the ISM Code by Administrations, and lACS 'Procedural Guidelines for ISM Code Certification.'

ICS

11.2.3 Procedures should be established to deal with the removal and destruction of obsolete documentation.

11.3 The company's 8MS should encompass all the requirements of the ISM Code and should consist of both office and shipboard manuals. The company should ensure that the relationship between the 8MS and other shore and shipboard systems is properly understood.

12.0 Audit plans should be established and should encompass all departments involved with the SMS, and the ships. Plans should cover: specific areas to be audited, qualifications of auditors, procedures for reporting findings. Evaluations of results from audits should review: organization structures, administrative procedures, personnel and their authority, adherence to the SMS policies, training, reports and record keeping. Management Reviews should be conducted into: analyses of accidents, hazardous occurrences, and nonconformities, audit findings, review of the SMS for updating etc. Minutes should be taken and retained.

13.1 Reference is made to IMO Resolution A788(19)-see lACS on left.

APPENDIX

Useful Conversion Factors

To convert from	*To*	*Multiply by*
atmospheres	feet of water @ 4°C	3.39E+1
atmospheres	inches Hg @ QOC	2.992E+1
atmospheres	meters Hg @ O°C	7.6E-1
atmospheres	Kg/sq em	1.0333E+OO
atmospheres	pounds f/sq in	1.47E+1
barrels (U.S. liquid)	gallons	3.15E+1
barrels (oil)	gallons (oil)	4.2E+1
Btu	foot-pounds	7.7816E+2
Btu	horsepower-hours	3.927E-4
Btu	joules	1.055E+3
Btu	kilowatt-hours	2.928E-4
Btu/hour	horsepower	3.929E-4
Btu/hour	watts	2.931E-1
Btu/minute	horsepower	2.356E-2
Celsius degree	Fahrenheit degree	C x 9/5 + 32
Celsius degree	Kelvin degree	C + 273.18
centimeters	feet	3.281E-2
centimeters	inches	3.937E-1
centimeters	mils	3.937E+2
centimeters/second	knots	1.943E-2
centimeters/second	miles/hour	2.237E-2
centimeters/see/see	feet/see/see	3.281E-2
centipoise	gram/em-see	1.0E-2

To convert from	To	Multiply by
centipoise	pound/ft-sec	6.72E--4
cubic centimeters	cubic feet	3.531E-5
cubic centimeters	cubic inches	6.102E-2
cubic centimeters	gallons U.S.	2.642E--4
cubic feet	cubic inches	1.728E+3
cubic feet	cubic meters	2.832E-2
cubic feet	cubic yards	3.704E-2
cubic feet	gallons U.S.	7.48052E+O
cubic feet	liters	2.832E+l
cubic inches	cubic feet	5.787E-4
cubic inches	gallons U.S.	4.329E-3
cubic meters	cubic feet	3.53IE+1
cubic meters	gallons U.S.	2.642E+2
days	seconds	8.64E+4
days	minutes	1.44E+3
degrees angle	radians	1.745E-2
dynes/square centimeter	inches of Hg	2.953E-5
dynes/square centimeter	inches of H20 @ 4°C	4.015E-4
dynes	poundals	7.233E-5
ergs	Btu	9.486E-ll
ergs	foot-pounds	7.376E-8
ergs	horsepower-hrs	3.725E-14
ergs/second	kilowatts	1.0E-IO
faradays	ampere-hour	2.68E+l
farsdays	coulombs	9.649E+4
fathoms	meters	1.8288E+O
fathoms	feet	6.0E+O
feet	kilometers	3.048E-4
feet	meters	3.048E-l
feet	nautical miles	1.645E--4
feet	statute miles	1.894E--4
feet of water	inches of Hg	8.826E-l
feet of water	pounds/sq inch	4.335E-l
feet/second	knots	5.921E-l
foot-candle	lumens/sq meters	1.0764E+l
foot-pounds	Btu	1.286E-3

To convert from	To	Multiply by
foot-pounds	horsepower-hours	5.05E-7
foot-pounds	joules	1.356E+O
foot-pounds	kilowatt-hours	3.766E-7
foot-pounds/minute	horsepower	3.03E-5
foot-pounds/minute	kilowatts	2.26E-5
gallons	cubic feet	1.337E-l
gallons	cubic meters	3.785E-3
gallons of water, fresh	pounds of water	8.337E+O
gallons/minute	cubic feet/second	2.228E-3
gausses	lines/sq inch	6.452E+O
gilberts	ampere-turns	7.958E-l
grams	avdp ounces	3.527E-2
grams	troy ounces	3.215E-2
horsepower	Btu/minute	4.244E+ 1
horsepower	Foot-pounds/minute	3.3E+4
horsepower metric	horsepower	9.863E-l
horsepower	kilowatts	7.457E-l
horsepower boiler	Btu/hour	3.352E+4
horsepower-hours	Btu	2.547E+3
horsepower-hours	Foot-pounds	1.98E+6
horsepower-hours	joules	2.684E+6
horsepower-hours	kilowatt-hours	7.457E-l
inches	centimeters	2.54E+O
inches of Hg	feet of water	1.133E+O
inches of Hg	pounds/sq inch	4.912E-l
inches of H20 a 4°C	pounds/sq inch	3.613E-2
joules	Btu	9.486E--4
joules	foot-pounds	7.736E-l
joules	watt-hours	2.778E-4
kilograms	pounds	2.2046E+O
kilograms	long tons	9.842E--4
kilograms	short tons	1.102E-3
kilograms	avdp ounces	3.5274E+l
kilograms-calories	Btu	3.968E+O
kilograms-calories	foot-pounds	3.086E+3
kilograms-calories	horsepower-hours	1.558E-3

To convert from	To	Multiply by
kilograms-calories	joules	4.183E+3
kilograms-calories	kilowatt-hours	1.163E-3
kilometers	feet	3.281E+3
kilometers	statute mile	6.214E-l
kilometers	nautical mile	5.396E-l
kilowatts	Btu/minute	5.692E+l
kilowatts	horsepower	1.341E+O
kilowatt-hours	Btu	3.413E+3
kilowatt-hours	foot-pounds	2.655E+6
kilowatt-hours	joules	3.6E+6
knots	feet/hour	6.076E+3
knots	kilometer/hour	1.852E+O
knots	statute miles/hour	1.151E+O
knots	feet/second	1.688E+O
liters	cubic feet	3.531E-2
meters	fathoms	5.468lE-l
meters	feet	3.281E+O
meters	inches	3.937E+l
meters	nautical miles	5.4E-4
miles nautical	feet	6.076E+3
miles nautical	meters	1.852E+3
miles/hour	feet/minute	8.8E+ 1
miles/hour	feet/second	1.467E+O
miles/hour	kilometers/hour	1.6093E+O
miles/hour	knots	8.684E-l
newtons	dynes	1.0E+5
poise	gram/centimeter-second	1.0E+O
poundals	dynes	1.3826E+4
poundals	joules/centimeter	1.383E-3
pounds	kilograms	4.536E-l
pounds troy	long tons	3.6735E-4
pounds troy	metric tons	3.7324E-4
pounds troy	short tons	4.1l43E-4
radians	degrees	5.7296E+l
slugs	kilograms	1.459E+l
slugs	pounds	3.217E+l

To convert from	To	Multiply by
square centimeter	square inch	1.55E-l
square feet	square meters	9.29E-2
square meters	square feet	1.076E+l
tons long	kilograms	1.016E+3
tons long	pounds	2.24E+3
tons long	tons short	1.12E+O
tons metric	kilograms	1.0E+3
tons metric	pounds	2.205E+3
tons short	kilograms	9.0718E+2
tons short	pounds	2.0E+3
tons short	tons long	8.9287E-l
tons short	tons metric	9.078E-l
watts	Btu/hour	3.4129E+O
watts	foot-pounds/minute	4.427E+ 1
watts	horsepower	1.34lE-3
watts	metric horsepower	1.36E-3

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