SECOND EDITION

HANDBOOK OF HYDRAULIC FLUID TECHNOLOGY

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Edited by
George E. Totten
Victor J. De Negri

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This book is dedicated to our families, without whose continued support the completion of this work would not have been possible:

For my wife Alice.

G.E.T.

For my wife Rosely and my daughter Fernanda.

V.J.D.N.

Contents

Preface to the Second Edition

This book is a significant revision of the first edition of the *Handbook of Hydraulic Fluid Technology*, which was edited by Dr. George E. Totten and published 10 years ago. Since the original publication of this text, no other similar book has been published that treats hydraulic fluids as a component of a hydraulic system and addresses all the major aspects of hydraulic fluid technology. In view of the unique position of the *Handbook of Hydraulic Fluid Technology*, a decision was made to significantly update this invaluable text.

The *Handbook of Hydraulic Fluid Technology—Second Edition* contains 21 chapters. Chapter 1: Fundamentals of Hydraulic Systems and Components, Chapter 5: Control and Management of Particle Contamination in Hydraulic Fluids, Chapter 11: Noise and Vibration of Fluid Power Systems, and Chapter 18: Biobased and Biodegradable Hydraulic Oils have been completely rewritten to more effectively address and expand coverage of critical new technology developments. Chapter 21: Food-Grade Hydraulic Fluids, is a newly added chapter to the book. The remaining chapters of the book have been revised and updated, and in many cases substantially. The updated and expanded coverage necessitated the elimination of three chapters from the first edition: Lubricant Additives for Mineral Oil–Based Hydraulic Fluids, Bearing Selection, and Lubrication and Electro-Rheological Fluids. With the exception of the chapter on electro-rheological fluids, the necessary content has been integrated into the remaining chapters of the book as appropriate. In general, the *Handbook of Hydraulic Fluid Technology—Second Edition* is a substantially new text on this very important critical hydraulics technology.

The editors of the *Handbook of Hydraulic Fluid Technology—Second Edition* are George E. Totten, PhD and Victor De Negri, D.Eng. Both editors are deeply indebted to the contributing authors for their vital assistance in completing this project. The editors also express appreciation to the staff of CRC Press for the opportunity to undertake this task and for their ongoing encouragement and vital support during all aspects of the book, from concept to production. Most importantly, the encouragement of our families is particularly appreciated.

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Preface to the First Edition

One of the most frustrating practices of my career has been the search for information on hydraulic fluids, which includes information on fluid chemistry; physical properties; maintenance practices; and fluid, system, and component design. Although some information on petroleum oil hydraulic fluids can be found, there is much less information on fire resistant, biodegradable, and other types of fluids. Unfortunately, with few exceptions, fluid coverage in hydraulic texts is typically limited to a single-chapter overview intended to cover all fluids. Therefore, it is often necessary to perform a literature search or a time-consuming manual search of my files. Some time ago, it occurred to me that others must be encountering the same problem. There seemed to be a vital need for an extensive reference text on hydraulic fluids that would provide information in sufficient depth and breadth to be of use to the fluid formulator, hydraulic system designer, plant maintenance engineer, and others who serve the industry.

Currently, there are no books dedicated to hydraulic fluid chemistry. Most hydraulic fluid treatment is found in handbooks, which primarily focus on hydraulic system hardware, installation, and troubleshooting. Most of these books fit into one of two categories. One type of book deals with hydraulic equipment, with a single, simplified overview chapter covering all hydraulic fluids, but with a focus on petroleum-derived fluids. The second type of book provides fluid coverage with minimal, if any, discussion of engineering properties of importance in a hydraulic system.

The purpose of the *Handbook of Hydraulic Fluid Technology* is to provide a comprehensive and rigorous overview of hydraulic fluid technology. The objective is not only to discuss fluid chemistry and physical properties in detail, but also to integrate both classic and current fundamental lubrication concepts with respect to various classes of hydraulic fluids. A further objective is to integrate fluid dynamics with respect to their operation in a hydraulic system in order to enable the reader to obtain a broader understanding of the total system. Hydraulic fluids are an important and vital component of the hydraulic system.

The 21 chapters of this book are grouped into three main parts: hardware, fluid properties and testing, and fluids.

HARDWARE

Chapter 1 provides the reader with an overview of basic hydraulic concepts, a description of the components, and an introduction to hydraulic system operation. In Chapter 2, the rolling element bearings and their lubrication are discussed. An extremely important facet of any well-designed hydraulic system is fluid filtration. Chapter 3 not only provides a detailed discussion of fluid filtration and particle contamination and quantification, but also discusses fluid filterability.

An understanding of the physical properties of a fluid is necessary to understand the performance of a hydraulic fluid as a fluid power medium. Chapter 4 features a thorough overview of the physical properties, and their evaluation and impact on hydraulic system operation, which includes: viscosity, viscosity-temperature and viscosity-pressure behavior, gas solubility, foaming, air entrainment, air release, and fluid compressibility and modulus.

FLUID PROPERTIES AND TESTING

Viscosity is the most important physical property exhibited by a hydraulic fluid. Chapter 5 presents an in-depth discussion of hydraulic fluid viscosity and classification. The hydraulic fluid must not only perform as a power transmission medium, but also lubricate the system. Chapter 6 provides a thorough review of the fundamental concepts involved in lubricating a hydraulic system. In many applications, fluid fire resistance is one of the primary selection criteria. An overview of historically important fire-resistance testing procedures is provided in Chapter 7, with a discussion of currently changing testing protocol required for industry, national, and insurance company approvals. Ecological compatibility properties exhibited by a hydraulic fluid is currently one of the most intensive research areas of hydraulic fluid technology. An overview of the current testing requirements and strategies is given in Chapter 8.

One of the most inexpensive but least understood components of the hydraulic system is hydraulic seals. Chapter 9 provides a review of mechanical and elastomeric seal technology and seal compatibility testing. An often overlooked but vitally important area is adequate testing and evaluation of hydraulic fluid performance in a hydraulic system. Currently, there is no consensus on the best tests to perform and what they reveal. Chapter 10 reviews the state-of-the-art of bench and pump testing of hydraulic fluids. Vibrational analysis is not only an important plant maintenance tool, but it is also one of the most important diagnostic techniques for evaluating and troubleshooting the operational characteristics of a hydraulic system. An introductory overview of the use of vibrational analysis in fluid maintenance is given in Chapter 11. No hydraulic system operates trouble-free forever. When problems occur, it is important to be able to identify both the problem and its cause. Chapter 12 provides a thorough discussion of hydraulic system failure analysis strategies.

FLUIDS

Although water hydraulics do not constitute a major fluid power application, they are coming under increasing scrutiny as ecocompatible alternatives to conventional hydraulic fluids. Chapter 13 offers an overview of this increasingly important technology.

The largest volume fluid power medium is petroleum oil. In Chapter 14, the reader is provided with a thorough overview of oil chemistry, properties, fluid maintenance, and change-out procedures. Chapter 15 reviews additive technology for petroleum oil hydraulic fluids. There are various types of synthetic hydraulic fluids. A description of the more important synthetic fluids, with a focus on aerospace applications, is given in Chapter 16.

Chapters 17 to 20 describe fire-resistant hydraulic fluids. Emulsions, water glycols, polyol esters, and phosphate esters are discussed individually and in depth in Chapters 17, 18, 19, and 20, respectively. This discussion includes fluid chemistry, physical properties, additive technology, maintenance, and hydraulic system conversion.

Vegetable oils are well-known lubricants that have been examined repeatedly over the years. Currently, there is an intensive effort to increase the utilization of various types of vegetable oils as an ecologically sound alternative to mineral oil hydraulic fluids. Chapter 21 provides a review of vegetable oil chemistry, recovery, and properties. The applicability of these fluids as hydraulic fluid basestocks is examined in detail.

Chapter 22 discusses electrorheological fluids, which are becoming increasingly interesting for use in specialized hydraulic applications. In Chapter 23, various standardized fluid maintenance procedures are discussed and a summary of equivalent international testing standards is provided.

The preparation of a text of this scope was a tremendous task. I am deeply indebted to many colleagues for their assistance, without whom this text would not have been possible. Special thanks go to Dr. Stephen Lainer (University of Aachen), Professor Atsushi Yamaguchi (Yokohama National University), Professor Toshi Kazama (Muroran Institute of Technology), K. Mizuno (Kayaba Industrial Ltd.), and Jürgen Reichel (formerly with DMT, Essen, Germany).

Special thanks also goes to my wife, Alice, for her unending patience, and to Susan Meeker, who assisted in organizing and editing much of this material; to Glenn Webster, Roland J. Bishop, Jr., and Yinghua Sun, without whose help this text would never have been completed; and to Union Carbide Corporation for its support.

George E. Totten

Editors

George E. Totten received his BS and MS degrees from Fairleigh Dickinson University in New Jersey and his PhD from New York University. Dr. Totten is past president of the International Federation for Heat Treating and Surface Engineering (IFHTSE) and a fellow of ASM International, SAE International, IFHTSE and ASTM International. Dr. Totten is an adjunct professor at Texas A&M University in College Station, TX and he is also president of G.E. Totten & Associates LLC, a research and consulting firm specializing in thermal processing and industrial lubrication problems.

Dr. Totten is the author or coauthor (editor) of over 500 publications including patents, technical papers, book chapters, and books, which include *Handbook of Hydraulic Fluid Technology; Handbook of Aluminum* Vol. 1 and Vol. 2; *Handbook of Lubrication and Tribology – Volume 1: Application and Maintenance; Handbook of Quenchants and Quenching Technology, Quenching Theory and Technology*, 2nd edition*; Steel Heat Treatment Handbook; Handbook of Residual Stress and Deformation of Steel; Handbook of Metallurgical Process Design*; and the *ASTM Fuels and Lubricants Handbook: Technology, Properties, Performance, and Testing (MNL 37).*

Victor Juliano De Negri, D.Eng. received his mechanical engineering degree in 1983, from UNISINOS, Brazil, a M.Eng. degree in 1987 and a D.Eng. degree in 1996, both from UFSC, Brazil. Since 1995, he has been associate professor in the mechanical engineering department at the Federal University of Santa Catarina (UFSC). He is currently the head of the Laboratory of Hydraulic and Pneumatic Systems (LASHIP). He is a member of the Brazilian Society of Mechanical Sciences and Engineering (ABCM) and LASHIP official company representative to the National Fluid Power Association (NFPA). His research areas include analysis and design of hydraulic and pneumatic systems and components and design methodologies for automation and control of equipment and processes. He has coordinated several research projects with industry and governmental agencies in the areas of hydraulic components, power-generating plants, mobile hydraulics, pneumatic systems, and positioning systems. He supervised 40 academic works including master's and doctorate theses and final term projects. He has 2 patents and written more than 90 journal and technical papers, conference papers, and magazine articles.

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1 Fundamentals of Hydraulic Systems and Components

*Irlan von Linsingen and Victor J. De Negri**

CONTENTS

* Some parts of this chapter are based on the chapter titled "Basic Hydraulic Pump and Circuit Design" by Richard K. Tessmann, Hans M. Melief, and Roland J. Bishop, Jr. from the *Handbook of Hydraulic Fluid Technology*, 1st Edition of this book.

1.1 INTRODUCTION

A hydraulic system, from a general perspective, is an arrangement of interconnected components that uses a liquid under pressure to provide energy transmission and control. It has an extremely broad range of applications covering basically all fields of production, manufacturing and service. Consequently, the energy transmission and control requirements are very diverse and thus the structure of each hydraulic system has its specificities.

However, on analyzing the current hydraulic systems, one can identify four main functions [1], as presented in Figure 1.1, which are: primary energy conversion, energy limitation and control, secondary energy conversion, and fluid storage and conditioning.

Furthermore, this figure shows the main resources that flow through a hydraulic system and which can be grouped into the classes: information, material, and energy [2].

The input of mechanical energy (M), which is a result of the external conversion of primary electrical or chemical (combustion) energy, is converted into hydraulic energy (H). Using signals or data (S, D) from an operator or from other equipment, the hydraulic energy (H) is limited and controlled such that it becomes appropriate for conversion into mechanical energy (M). This mechanical energy is the desired output of the hydraulic system and will be used to drive or move external devices.

The hydraulic energy is carried by the hydraulic fluid (F) and thus its storage and conditioning, including contamination and temperature control, are also essential functions.

As a consequence of the physical phenomena, construction characteristics, and circuit arrangement, part of the useful energy is dissipated in a hydraulic system. Therefore, all functions transfer thermal energy (T) to the fluid and to the environment.

Since this *Handbook* is concerned with fluid technology, the objective of this chapter is to characterize hydraulic systems, that is, applications in which hydraulic fluids are used.

The construction characteristics and the functioning principles of the main hydraulic components are presented, with the aim of providing an overview of the interaction between the fluid and the mechanical parts.

Moreover, the main equations that govern the component and circuit behavior are presented, where one can identify the influence of the fluid parameters, which, in turn, are a consequence of the physical-chemistry proprieties.

An important aspect of this chapter is the symbol notation that is used in the diagrams and equations. Both the hydraulic circuit diagrams and the component identification codes are in accordance

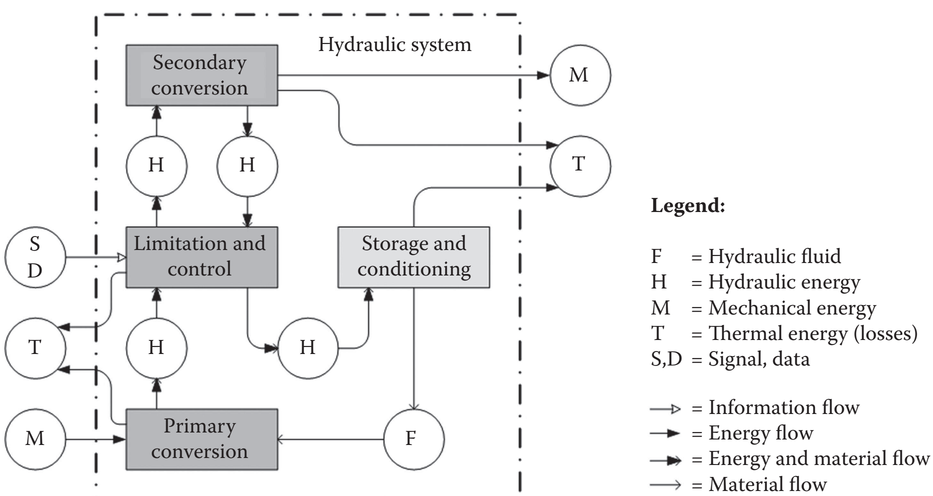


FIGURE 1.1 Generic hydraulic system: Functions and resource flows.

with ISO 1219-1 [3] and ISO 1219-2 [4]. The quantities (variables and parameters) used in the circuit diagrams, component illustrations, and equations are represented by letter symbols, including subscripts and superscripts, in compliance with ISO 4391 [5], IEC 27-1 [6], and ISO 1219-2 [4] standards.

1.2 HYDROMECHANICAL PRINCIPLES

Essentially, a hydraulic system consists of mechanical parts operating together with a hydraulic fluid. Hence, its behavior is described by the classic laws of both mechanics and fluid mechanics. Although it is not the focus of this text, it is important to remember that several hydraulic components comprise electromechanical converters, such as solenoids, linear motors and torque motors and/or electro-electronic systems like sensors, power amplifiers and controllers. Therefore, the principles of electricity, electronics and electro-magnetism are also required for their modeling.

1.2.1 Hydrostatics: Pascal's Principle

Fluids (gases or liquids) are compressible, which means that their mass density varies with the pressure to which they are submitted. Consequently, an abrupt local pressure variation will be propagated through the fluid with a velocity equal to the fluid sound velocity until the equilibrium has been re-established. This means that the fluid will have a dynamic behavior alternating between the two equilibrium states.

When a fluid is treated as incompressible it is assumed that a local pressure perturbation is instantaneously transmitted throughout the fluid. This means that considering a fluid as being compressible or incompressible is dependent on the observer's viewpoint and its validation depends on the use of the system and the particular design or analysis that is being carried out.

Pascal's principle states that "a change in the pressure of an enclosed incompressible fluid is conveyed undiminished to every part of the fluid and to the surfaces of its container" [1,7]. Hence, when a fluid is in a state of equilibrium, that is, in a steady state, the whole system is under the same internal pressure.

The practical use of Pascal's principle can be exemplified by the hydrostatic press principle whose objective is to amplify the force. As shown in Figure 1.2a [1], it consists of two cylinders (actuators) (A1 and A2) that are connected by a pipe.

In this press, the resistive force (F^{A2}* [N]) offered by the material to be pressed must be compensated by the input force (F^{A1} [N]) such that the equilibrium occurs. Since in a steady state the pressure (p [N/m^2] or [Pa]) is equal throughout the volume, one has

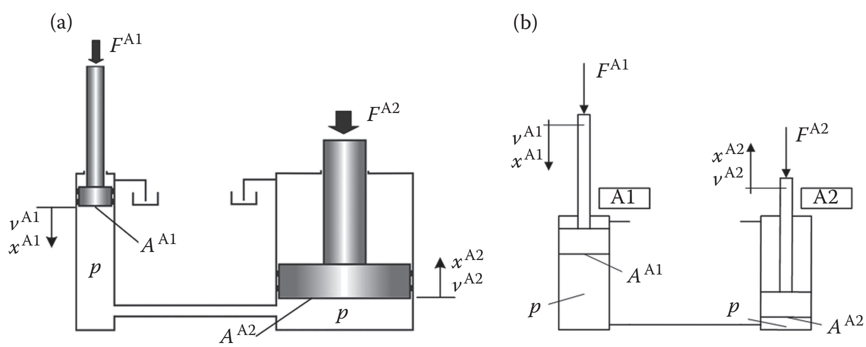


FIGURE 1.2 Hydrostatic press principle: (a) Illustration of the hydraulic circuit; (b) Hydraulic circuit diagram.

* The kernel (central part of the letter symbol) represents the generic quantity. The subscript indicates the quantity application and the superscript is used to indicate to which component or system the quantity is associated (ISO 4391, *ISO 1219-2 - Fluid Power Systems and Components – Graphic symbols and circuit diagrams – Part 2: Circuit diagrams,* Switzerland, 1991.

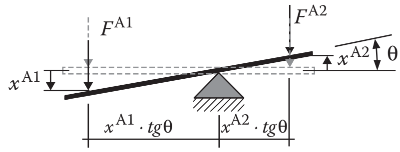


FIGURE 1.3 Mechanical system of force amplification.

$$p = \frac{F^{A1}}{A^{A1}} = \frac{F^{A2}}{A^{A2}} \Rightarrow \frac{F^{A2}}{F^{A1}} = \frac{A^{A2}}{A^{A1}} \text{ or } F^{A2} = \left(\frac{A^{A2}}{A^{A1}}\right) \cdot F^{A1}, \quad (1.1)$$

where A^{A1} [m^2] and A^{A2} [m^2] are the piston areas.

Equation 1.1 shows that, for $A^{A2}/A^{A1} >> 1$, a low force F^{A1} is sufficient to overcome a higher force like F^{A2}, which is the objective of most hydraulic systems.

Moreover, considering the incompressible fluid, the volume variations in the two cylinders (ΔV^{A1} [m^3] and ΔV^{A2} [m^3]) are equal. According to Equation 1.2, in this case the displacements x^{A1} [m] and x^{A2} [m] are different, their relationship being determined by the area ratio:

$$\Delta V^{A1} = \Delta V^{A2} \Rightarrow x^{A1} \cdot A^{A1} = x^{A2} \cdot A^{A2} \text{ or } x^{A2} = \left(\frac{A^{A1}}{A^{A2}}\right) \cdot x^{A1}. \quad (1.2)$$

Considering an efficiency of 100%, the work required of cylinder A1, determined by the product of the force and displacement, is equal to the work applied to cylinder A2. Hence, according to Equation 1.3, the correlation between F^{A1} and F^{A2} is given by the displacement ratio:

$$W = F^{A1} \cdot x^{A1} = F^{A2} \cdot x^{A2} \Rightarrow \frac{F^{A2}}{F^{A1}} = \frac{x^{A1}}{x^{A2}} \text{ or } F^{A2} = \left(\frac{x^{A1}}{x^{A2}}\right) \cdot F^{A1}. \quad (1.3)$$

Equation 1.3 is designed as the hydraulic lever equation [1], since the same force amplification could be obtained through a mechanical system—such as that shown in Figure 1.3.

These hydrostatic relationships allow the static behavior of a system to be determined—that is, the relationships between the forces and displacements in the equilibrium condition. The behavioral description with temporal variation is carried out using the laws of hydrodynamics [1].

1.2.2 Hydrodynamics: Conservation of Mass

The steady-state and transient behavior of hydraulic components and systems is described by the basic principles of hydrodynamics and thermodynamics [1,8]. In this chapter, two of these principles are studied; namely, the conservation of mass (continuity equation) and the conservation of energy (Bernoulli's equation), which are essential to the comprehension of the hydraulic component behavior.

From the conservation of mass principle, an important expression is obtained which describes the behavior of pressure in volumes. Consider the hydraulic device shown in Figure 1.4 [1], which has an inlet port (1), an outlet port (2) and a movable piston.

The mass density (ρ [kg/m^3]), the pressure (ρ [Pa]), and the temperature (T [K] or [°C]) of the fluid are considered constant in the space defined by the chamber, but they vary over time. The flow rate in the inlet port is considered positive when entering the chamber and the flow rate in the outlet port is positive when leaving the chamber. The chamber volume changes with the piston movement.

The result of the continuity equation [1,8,9] applied to this case is [5].

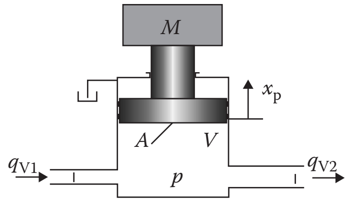


FIGURE 1.4 Chamber with variable volume.

$$q_{V1} - q_{V2} = \frac{dV}{dt} + \frac{V}{\beta} \cdot \frac{dp}{dt}, \tag{1.4}$$

where V [m^3] is the chamber volume and q_{V1} [m^3/s] and q_{V2} [m^3/s] are the volumetric flow rates (commonly referred to as the "flow rate") at the inlet and outlet ports, respectively. β [Pa] is the bulk modulus (inverse of compressibility), which characterizes the mass density variation with the fluid pressure.

In this equation, the terms on the right are related to the mass accumulation in the volume, where dV/dt represents the variation in the chamber volume over time and $(V/\beta)(dp/dt)$ the variation in the pressure over time associated with the fluid compressibility.

Therefore, Equation 1.4 describes the dynamic behavior of the pressure in the chamber as a consequence of the change in flow rate at port 1 and/or port 2. The pressure change will take the piston out of equilibrium, causing its movement. As a consequence, the first term on the right will be different from zero, in turn changing the pressure.

It is important to note that the continuity equation, as presented in Equation 1.4, is the basic form used in the hydraulics area to model the dynamic behavior of a fluid in cylinders, accumulators, motors, pipes and so forth.

Studying again the hydrostatic press (Figure 1.2), one can observe that the volume variation in cylinders A1 and A2 is dependent on the displacement direction of the pistons, which means that volume V^{A1} will be decreasing and volume V^{A2} increasing toward the positive directions indicated in this figure, that is

$$\frac{dV^{A1}}{dt} = -A^{A1} \cdot v^{A1} \text{ and } \frac{dV^{A2}}{dt} = A^{A2} \cdot v^{A2}, \tag{1.5}$$

where v^{A1} [m] and v^{A2} [m^2] are the piston velocities.

Appling Equations 1.4 and 1.5 to cylinders A1 and A2 for a constant pressure condition and taking into account that the flow rate that leaves cylinder A1 is the same as that entering cylinder A2, one can obtain

$$q_V = A^{A1} \cdot v^{A1} = A^{A2} \cdot v^{A2} \Rightarrow \frac{v^{A2}}{v^{A1}} = \frac{A^{A1}}{A^{A2}} \text{ or } v^{A2} = \left(\frac{A^{A1}}{A^{A2}}\right) \cdot v^{A1}. \tag{1.6}$$

Equation 1.6 describes the velocity relationship for the hydraulic press, completing the set of equations together with Equations 1.1 and 1.2.

1.2.3 Hydrostatic Press: Linear Motion

By means of the circuit in Figure 1.2, it is possible to have an upward moving cylinder A2 when cylinder A1 is moving downward. The displacement relationship (Equation 1.2) and velocity

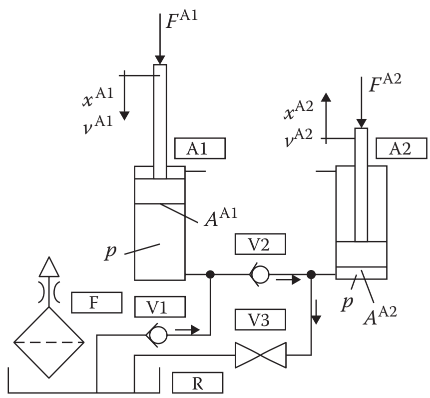


FIGURE 1.5 Hydraulic circuit diagram of a real hydrostatic press.

relationship (Equation 1.6) imply that a movement of cylinder A1 with displacement and velocity according to human capacity results in a press operation with both small displacement and velocity. Cylinder A1, having reached the required displacement, will reach its stroke end much earlier than cylinder A2.

Therefore, this basic circuit is not valuable for real uses. A typical circuit found in hydrostatic presses and hydraulic jacks is presented in Figure 1.5, where some components are added to the original circuit (Figure 1.2).

In this circuit an external reservoir (R), which compensates for the difference between the cylinder volumes, and two non-return valves (V1 and V2) are included. These valves allow fluid suction from the reservoir on the upward movement of cylinder A1 and fluid pumping to cylinder A2 on the downward movement. Valve (register) V3, when opened, allows the fluid in cylinder A2 to return to the reservoir as a consequence of the external force (F^{A2}) applied to the piston.

Correlating Figure 1.5 and 1.1, the arrangement constituted by A1, V1, and V2 performs the primary energy conversion function, V3 the energy control, and A2 the secondary energy conversion. The fluid storage and conditioning is performed by both the reservoir (R) and the air filter (F). The filter establishes the connection between the fluid and the external environment in order to keep the reservoir cleaned and at atmospheric pressure.

1.2.4 Hydrostatic Transmission: Rotary Motion

The principles presented previously for linear motion are now applied to rotary motion transmission using a pump and a motor (hydrostatic machines) as presented in Figure 1.6. According to ISO 1219-2 [4], the pump has its own symbol, P, while the hydraulic motor is an actuator and, for this reason, it is designed as A.

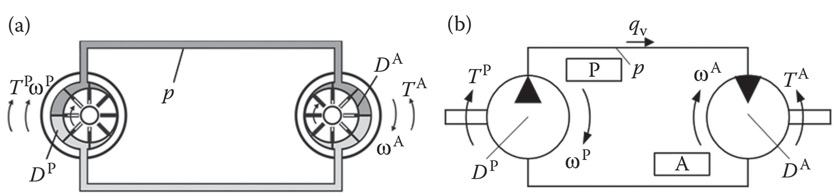


FIGURE 1.6 Hydrostatic transmission: (a) Illustration of the hydraulic circuit; (b) Hydraulic circuit diagram.

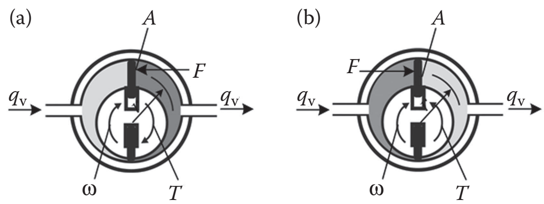


FIGURE 1.7 Principles of a hydrostatic machine: (a) Functioning as a pump; (b) Functioning as a motor.

The hydrostatic pump driven by an electric motor, for example, runs at an angular speed (ω^P [rad/s])* supplying a flow rate (q_V [m³/s]) to the hydraulic motor that causes an angular speed (ω^A [rad/s]) at the motor axis. At the same time, a loading applied to the axis causes a torque (T^A) in the opposite direction to the movement, inducing a pressure (p) increase. This pressure, which is transmitted to the whole system, acts on the pump increasing the mechanical torque T^P.

In fact, the pressure in the motor inlet is not the same as that in the pump outlet, as a consequence of the flow energy losses. However, as an ideal system is being considered, the load losses, leakages, and mechanical friction are neglected. In the same way as the hydrostatic press (Figure 1.2), both the pump suction port and motor discharge port are at atmospheric pressure, which means that the gauge pressure is equal to zero.

At each complete revolution of a hydrostatic machine rotor (Figure 1.7) (1 revolution = 2π rad) a certain fluid volume displacement (V [m³]) occurs. From this effect, the volumetric displacement (D [m³/rad]) is defined as

$$D = \frac{V}{2\pi}. \tag{1.7}$$

The volume displaced in one complete revolution is a function of the rotor geometry. For a rotor with vanes, as shown in Figure 1.7, this volume is the product of the vane area and the mean perimeter—that is, $V = A \cdot 2\pi \cdot r$. Hence, the volumetric displacement is $D = A \cdot r$.

Moreover, the torque on the pump or motor axis can be calculated by the product of the resulting force on the vanes and the mean radius, that is, $T = F \cdot r$. Thus, the pressure in a pump or motor chamber can be written as

$$p = \frac{F}{A} = \frac{T/r}{D/r} = \frac{T}{D}. \tag{1.8}$$

Equivalently to the hydrostatic press (Equation 1.1), the pump and motor torques can be related by

$$p = \frac{T^M}{D^M} = \frac{T^A}{D^A} \Rightarrow \frac{T^A}{T^M} = \frac{D^A}{D^M} \text{ or } T^A = \left(\frac{D^A}{D^M}\right) \cdot T^M. \tag{1.9}$$

Since the tangential velocity (v [m/s]) at a distance r [m] from the rotor axis is related to the angular velocity (ω [rad/s]) and to the rotational frequency (n [rps]) by $v = r \cdot \omega$ and $v = r \cdot 2\pi \cdot n$, respectively, Equation 1.6 can be modified to describe the relationship between the pump and motor velocities as

$$q_V = D^M \cdot \omega^M = D^A \cdot \omega^A \Rightarrow \omega^A = \left(\frac{D^M}{D^A}\right) \cdot \omega^M \Rightarrow n^A = \left(\frac{D^M}{D^A}\right) \cdot n^M. \tag{1.10}$$

* Observe that the quantity rotational frequency (or just rotation) (n [rps]) is commonly used instead of angular velocity (ω [rad]) and these are correlated by $\omega = 2\pi \cdot n$.

1.2.5 Hydrodynamics: Conservation of Energy

To understand the energy transmission and control in hydraulic systems it is fundamental to apply Bernoulli's equation [8,9]. According to this equation the sum of all forms of mechanical energy in a steady and unidimensional flow of an ideal and incompressible fluid is the same at all points in the stream line.

One fundamental use of Bernoulli's equation is to describe the flow behavior through a sharp-edge orifice in a pipe, which causes an abrupt reduction in the flow cross section, as shown in Figure 1.8 [1].

In this case, the stream lines converge to a point where the diameter of the stream is the smallest. This point is called *vena contracta* and corresponds to cross section 2 in the figure. By applying Bernoulli's equation to cross section 1 (orifice upstream) and cross section 2 (orifice downstream), one obtains

$$p_1 + \frac{1}{2} \cdot \rho \cdot v_1^2 + \rho \cdot g \cdot z_1 = p_2 + \frac{1}{2} \cdot \rho \cdot v_2^2 + \rho \cdot g \cdot z_2, \tag{1.11}$$

p [Pa] being the static pressure, $1/2 \cdot \rho \cdot v^2$ [Pa] the dynamic pressure and $\rho \cdot g \cdot z$ [Pa] the gravitational pressure.

Since Bernoulli's equation is valid for steady flow, the use of Equation 1.4 implies that the inlet and outlet flow rates are the same, that is, $q_V = A_1 \cdot v_1 = A_2 \cdot v_2$. Furthermore, since the orifice area (A_0) and, consequently, the *vena contracta* area (A_2), are much smaller than the inlet area (A_1), the velocity in the inlet cross section (v_1) is neglected.

Therefore, since the change in the $\rho \cdot g \cdot z$ term along the stream line is very small compared with the other terms it can be ignored and Equation 1.11 can be written as

$$q_V = A_2 \cdot \sqrt{\frac{2 \cdot (p_1 - p_2)}{\rho}}. \tag{1.12}$$

Aiming at its practical use, this equation must be corrected to include viscosity losses. Additionally, experimental data from the literature [9,10] correlate the *vena contracta* area (A_2) with the real orifice area (A_0) such that Equation 1.10 can be rewritten as

$$q_V = cd \cdot A_0 \cdot \sqrt{\frac{2 \cdot \Delta p}{\rho}}, \tag{1.13}$$

where *cd* is the discharge coefficient whose value is dependent on the orifice geometry and flow type.

Another important aspect is that the turbulence downstream of the orifice causes a significant energy loss such that the velocity reduction in cross section 3 (Figure 1.8), as a consequence of the cross-sectional area increase, does not cause a static pressure increase. Hence p_3 is very close to p_2.

Therefore, Equation 1.13, known as the orifice flow equation, is appropriate to calculate the flow rate through an orifice as a function of the cross-sectional area and the pressure drop between the cross sections of the inlet (1) and outlet (3).

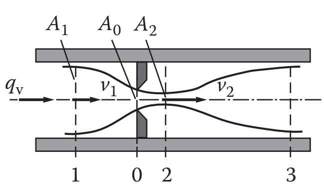


FIGURE 1.8 Flow through an orifice. (From Linsingen, I. von, *Fundamentos de Sistemas Hidráulicos*, 3rd ed., Florianópolis, Brazil: UFSC Ed., 2008. With permission.)

Finally, since the hydraulic power is defined as

$$P_h = p \cdot q_V, \tag{1.14}$$

the fact that the input pressure (p_1) is greater than the output pressure (p_3) implies that the hydraulic power is reduced with the fluid passing through an orifice. This hydraulic power difference is transformed into thermal energy, heating the fluid and the environment.

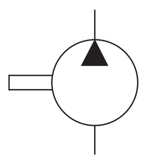
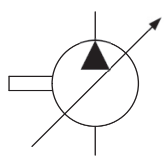
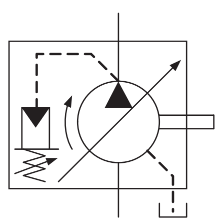
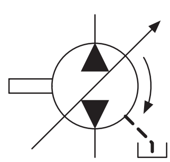
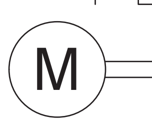
1.3 HYDRAULIC CIRCUITS

Hydraulic circuits are comprised of interconnected components so as to perform the four functions as identified in Figure 1.1. Typically, these circuits are represented by diagrams composed of graphical symbols that represent fluid power components and devices.

ISO 1219-1 [3] establishes basic elements for symbols and rules for devising fluid power symbols for use in components and circuit diagrams. ISO 1219-2 [4] establishes the rules for drawing fluid power diagrams using symbols from ISO 1219-1 [3], including rules for identification of equipment.

Table 1.1 presents the symbols according to ISO 1219-1 [3] for the hydraulic components used in this chapter. Furthermore, an identification code will be associated with these symbols following the rules shown in Figure 1.9.

TABLE 1.1
Some Symbols of Hydraulic Components

Primary Energy Conversion		
Hydraulic pumps	Fixed-displacement	
	Variable-displacement	
	Variable-displacement, with pressure compensation, external drain line, one direction of rotation	
	Variable-displacement, two directions of flow, external drain line, one direction of rotation	
Electric motor		

(*continued*)

TABLE 1.1 (Continued)
Some Symbols of Hydraulic Components

Energy Limitation and Control

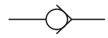
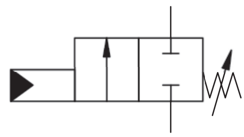
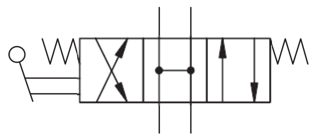
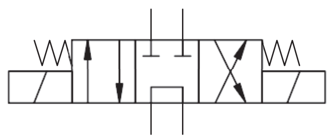
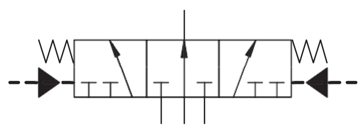
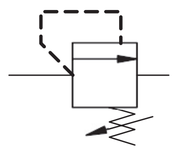
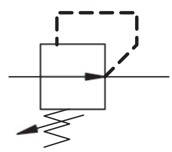
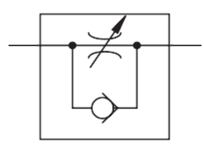
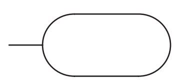
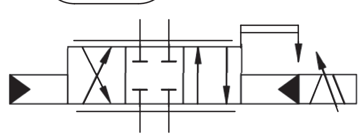
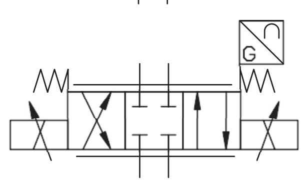
Directional control valves	Manual shut-off	
	Non-return (check)	
	2-port, 2-position, controlled by hydraulic pilot control, opening pressure adjusted by spring	
	4-way, 3-position, controlled by lever, with spring-centered central position	
	4-way, 3-position, directly controlled by two solenoids with spring-centered central position	
	5-way, 3-position, hydraulically controlled, with spring-centered central position	
Pressure control valves	Pressure relief, directly controlled, opening pressure adjusted by a spring (See Figure 1.45)	
	Pressure reducing valve, directly operated, closing pressure adjusted by a spring	
Flow control valves	Flow control adjustable, with reverse free flow	
Accumulators	(See Figure 1.55)	
Directional continuous control valves	Servo-valve, pilot-operated, pilot stage with electrical control mechanism with two coils, continuously controlled in both directions, with mechanical feedback of the main stage to the pilot stage	
	Proportional directional control valve, directly operated, with closed-loop position control of the main stage	

TABLE 1.1 (Continued)
Some Symbols of Hydraulic Components

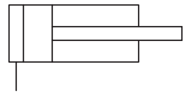
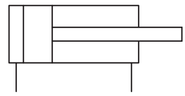
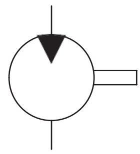
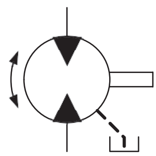
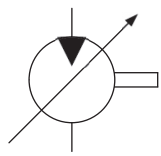
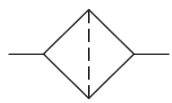
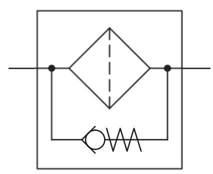
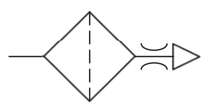
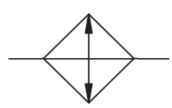
Secondary Energy Conversion		
Hydraulic cylinders	Single-acting (See Figure 1.32)	
	Double-acting (See Figure 1.33)	
Hydraulic motors	Fixed-displacement	
	Fixed-displacement, two directions of flow, two directions of rotation, with external drain	
	Variable-displacement	
Hydraulic filters	Filter	
	Filter with bypass valve	
	Filter with air exhausting	
Reservoir	Reservoir with return line / Reservoir with drain line	
Heat exchanger	Cooler	

Figure 1.10 shows a typical hydraulic circuit where the fixed-displacement pump (P) runs at a constant rotational frequency driven by the electric motor (M). Since the pump theoretically supplies a constant flow rate, it is necessary to direct part of the flow through the relief valve (V1) aiming to obtain velocity control in the cylinder (A). Therefore, the effect of the flow control valve (V3) is to cause a pressure loss such that the supply pressure (p_P) is above the setting pressure (p_{Pset}) at the relief valve (V1), and it opens. The directional control valve (V2) directs the fluid from the supply

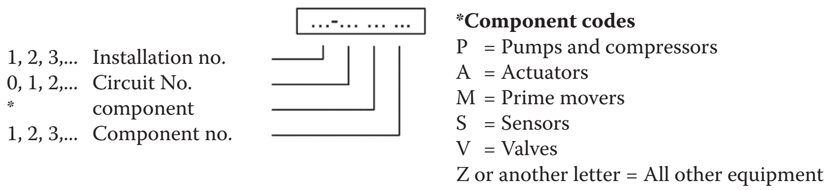


FIGURE 1.9 Identification code according to ISO 1219-2.

line (P) to cylinder chamber A or B and from cylinder chamber B or A to the reservoir line (This type of circuit is considered an open circuit since the fluid does not return directly to the pump suction port but to the reservoir, where it is stored before undergoing suction by the pump. The motion control of the actuators is fundamentally dissipative since it is carried out by directional, pressure, and flow control valves. The functioning principle of these valves is described by the orifice flow equation (Equation 1.13).

By comparing Figures 1.10 and 1.1, one can observe that the pump (P), together with the electric motor (M), performs the primary conversion function; the actuator (A) performs the secondary conversion; and the pressure relief valve (V1), directional control valve (V2) and flow control valve (V3) perform the energy limitation and control. The fluid storage and conditioning is performed by the reservoir (R) and filter (F1).

The open-loop circuit is by far the most popular design. The advantage of an open-loop design is that, if necessary, a single pump can be used to operate several different actuators simultaneously. The main disadvantage is its large reservoir size.

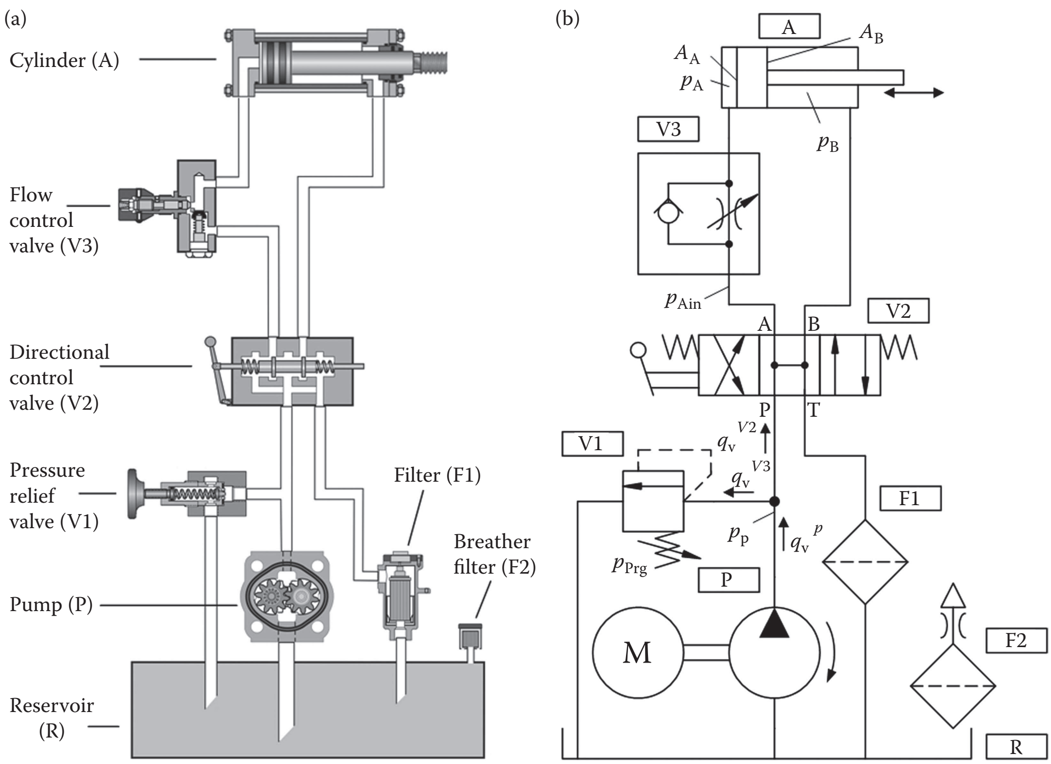


FIGURE 1.10 Open-loop hydraulic circuit: (a) Illustration; (b) Circuit diagram.

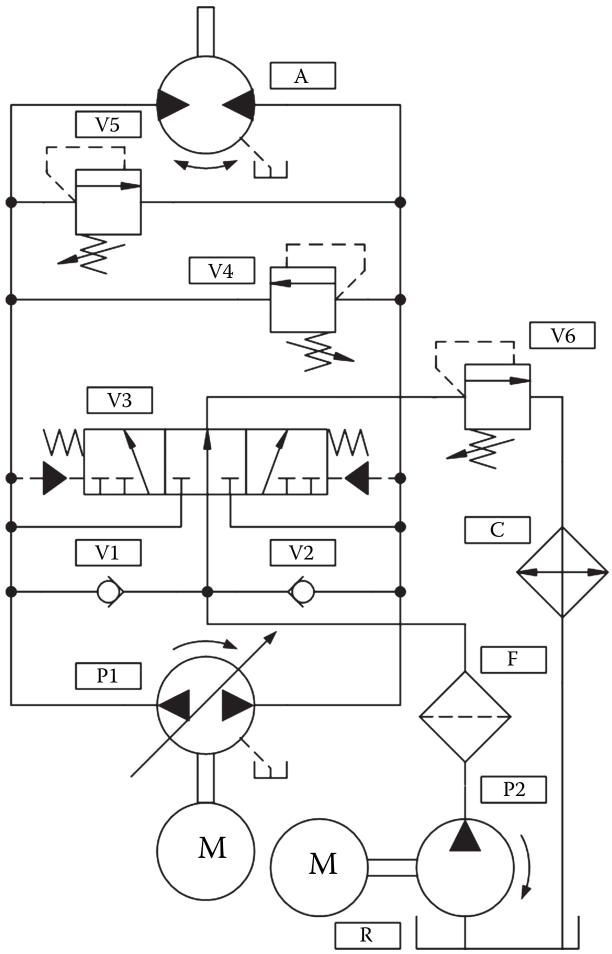


FIGURE 1.11 Closed-loop circuit diagram.

A second general type of hydraulic circuit is the closed-loop circuit [7], whose main operational difference relates to the means of hydraulic energy control. As can be observed in the example in Figure 1.11, it is not only the pump discharge but also the pump suction that is directly connected to the motor ports. Therefore, the motor rotational frequency will be modified if the volumetric displacement of either the motor or the pump is varied or the pump rotational frequency is changed. The relationship between the flow rate, volumetric displacement, and rotational frequency of a pump or motor is described by Equation 1.10.

In the circuit shown in Figure 1.11 [7], a variable-displacement pump (P1) is used to drive a fixed-displacement hydraulic motor (A). A closed-loop circuit is always used in conjunction with a smaller replenishing circuit. The replenishing circuit consists of a small fixed-displacement pump (P2) (usually about 15% of the displacement of the main pump), a small fluid reservoir (R), filter (F), and a heat exchanger (cooler) (C).

The replenishing circuit always works on the low-pressure side of the main loop. Its function is to pump freshly filtered fluid into the closed loop through non-return valves (V1 and V2) while bleeding-off a percentage of the hot fluid through a directional control valve (V3). This hot fluid is then cooled by a cooler (H) and stored in a small reservoir (R) before returning to the main system. The pressure in the replenishing circuit is limited to 10–20 bar (1–2 MPa) by the supercharge relief valve (V6). The pressure setting of this valve is determined by the requirements of the pump/motor combination and the operating conditions of the system. The cross-port relief valves (V4 and V5) on the motor are there only to protect the actuator from load-induced pressure spikes. They are not intended to function like those found in open-loop circuits, which would cause severe overheating of the circuit due to the diverting of the unnecessary flow through the relief valve.

The advantages of a closed-loop circuit are that high-power systems are compact and efficient and require less hydraulic fluid storage. The high efficiency of this circuit is the result of the pump control being designed to supply only the fluid flow required by the actuator to operate at the load-induced pressure. The pump is the heart of the system and controls the direction, acceleration, speed, and torque of the hydraulic motor, thus eliminating the need for pressure and flow control components.

In this type of circuit the energy control is transformative, instead of dissipative as in open-loop circuits, since it is the energy transformed in the pump or motor that is controlled. However, the secondary valves (pressure, directional and flow-control valves) impose energy losses—besides the internal mechanical and fluid flow losses—in pumps and motors, thereby reducing the overall efficiency.

A major disadvantage of a closed-loop circuit is that a single pump can only operate a single output function or actuator. In addition, this type of hydraulic circuit is generally used only with motor actuators.

The third general configuration is the half-closed-loop circuit as shown in Figure 1.12 [7]. This circuit is similar to the closed-loop circuit except that it can be used with cylinder actuators with different areas. As can be seen from the figure, during cylinder extension, the pump (P) must generate a higher flow rate from its left-hand port than that being returned to its right-hand port from the cylinder (A). The extra fluid needed by the pump (P) is supplied by its left-hand inlet non-return valve, which is an integral part of the pump. When the pump control moves the pump over

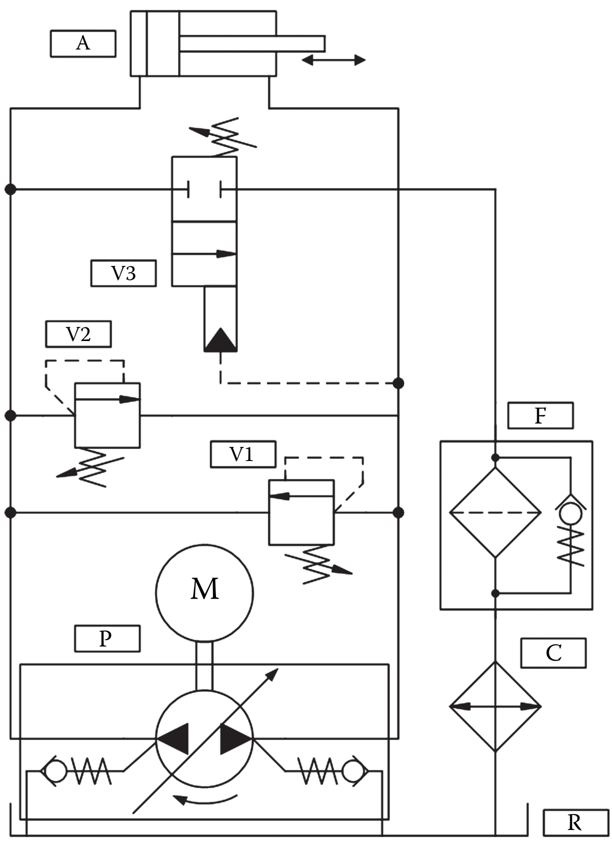


FIGURE 1.12 Half-closed-loop circuit diagram.

the center, the flow from the pump (P) is reversed and the cylinder (A) begins to retract. During retraction, the larger area of the cylinder piston causes a higher flow rate than needed at the inlet of the pump (P). This excess flow is directed to the reservoir (R) through the unloading valve (V3). The unloaded fluid is filtered and cooled prior to its return to the reservoir. In this way, a portion of the closed-loop fluid is filtered (by F) and cooled (by C) in an open-loop circuit each time the cylinder (A) is cycled.

In this case, the fluid volume and reservoir size reductions are not as significant as in the closed-loop scenario.

As can be seen in the above examples, each hydraulic component has a basic function, but it is the circuit itself that determines the hydraulic system behavior. Hence, for a designer to conceive a hydraulic system he/she needs to have an understanding of the functional and behavioral characteristics of the components which, in turn, are dependent on the fluid-mechanical interaction inside the component.

1.4 HYDRAULIC COMPONENTS

1.4.1 Hydrostatic Machines: Pumps and Motors

The energy conversion functions in a hydraulic system are performed by pumps and actuators (basically motors and cylinders). The pumps perform the primary conversion, transforming mechanical energy into hydraulic energy. The actuators retransform the hydraulic energy into mechanical energy to be used by the machine or the equipment.

There are two classes of hydraulic machines: hydrodynamic and hydrostatic machines. They differ in the way the internal energy is transformed and, consequently, in their form of construction [1].

In hydrodynamic machines (such as centrifugal pumps, turbines, and fans), the fluid energy involved on the transformation process is fundamentally kinetic, due to the variation in the fluid velocities of the impeller blades. In these machines there is a gap between the pump housing and the impeller (or rotor) leading to a high internal leakage even with low differential pressure.

In the centrifugal pumps, as shown in Figure 1.13a, when the output fluid flow resistance is increased (e.g., as a consequence of the load loss in the discharge line) the output flow rate is reduced until it drops to zero, as shown in the characteristic curve in Figure 1.14a.

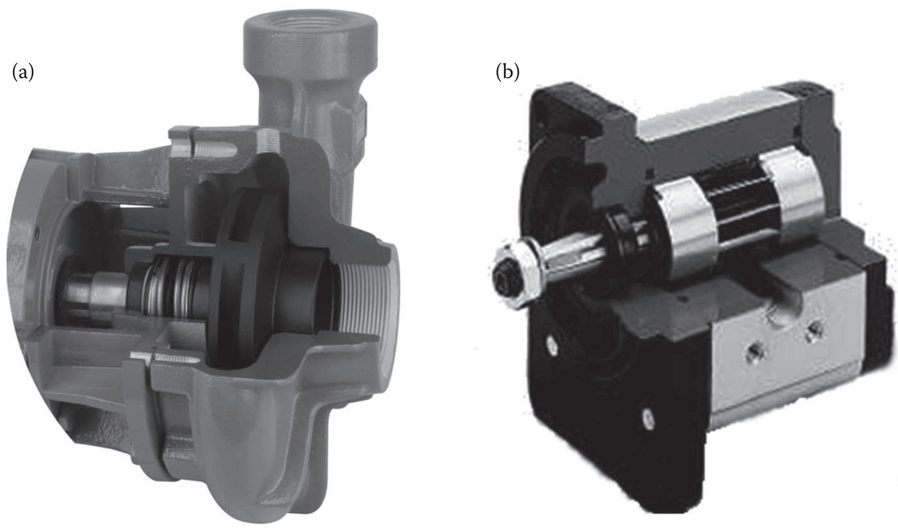


FIGURE 1.13 Classes of pumps: (a) Hydrodynamic pump (centrifugal pump) (Courtesy of Franklin Electric–Joinville–SC-Brazil); (b) Hydrostatic pump (gear pump). (Courtesy of Bosch Rexroth–Pomerode–SC-Brazil).

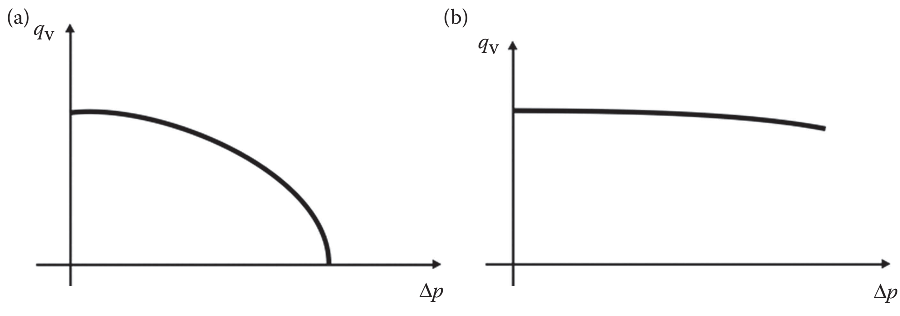


FIGURE 1.14 Characteristic curves of pumps: (a) Hydrodynamic pump; (b) Hydrostatic pump.

In hydrostatic machines, also referred to as "positive displacement machines," the fluid energy involved in the transformation process is mainly related to the variation between the inlet and outlet pressures through the rotor. Since the pressure in a system is caused by the fluid flow resistance, the effective pump outlet pressure increase is dependent on the valves and actuators downstream of the pump. In turn, the pressure in an actuator inlet is dependent on the rotor movement resistance caused by an external mechanical loading.

In hydrostatic pumps the clearance between the housing and the rotor is very small and thus the suction and discharge chambers are basically isolated. As a consequence, the pump flow rate is slightly influenced by the downstream pressure, as illustrated by the characteristic curve shown in Figure 1.14b.

Since the construction principle of hydrostatic (rotary) motors is the same as that of pumps, an increase in the mechanical axis loading leads to a small leakage increase. Hence, the motor rotational frequency can be considered constant in several applications [1].

The fact that the hydrostatic pumps are an almost ideal flow rate source and operate under high pressures makes this class of hydraulic machine basically the only one used in fluid power systems [1]. At same time, to attain the requirements of the several application fields, different construction principles of hydrostatic machines have been developed, as shown in Table 1.2.

In the right column of this table, an important feature of hydrostatic machines is indicated. According to Equation 1.10, the volumetric displacement establishes the proportionality between the flow rate and the rotational frequency. Machines whose construction characteristics do not allow changes in the volumetric displacement are named *fixed-displacement machines*, and those where is possible to obtain different flow rates at the same rotational frequency are named *variable-displacement machines*.

TABLE 1.2
Classification of Hydrostatic Machines According to Construction Principle and Volumetric Displacement

Constructive Principle			Volumetric Displacement
Gear	External		Fixed
	Internal	Crescent	Fixed
		Gerotor	Fixed
Screw			Fixed
Vane	Balanced		Fixed
	Unbalanced		Fixed or Variable
Piston	Radial		Fixed or Variable
	Axial	Swash Plate	Fixed or Variable
		Bent-Axis	Fixed or Variable

TABLE 1.3
Typical Pump Performance Parameters

Pump Type	Max. Working Pressure [MPa (bar)]	Flow Rate [dm³/s (Lpm)]	Rotational Frequency [rps (rpm)]	Global Efficiency [%]
External Gear	15–25 (150–250)	0.08–9.5 (5–570)	8.3–83.3 (500–5,000)	80–90
Internal gear	3.5–20 (35–200)	0.08–12.7 (5–760)	15–41.7 (900–2500)	70–90
Screw	0.4–40 (4–400)	0.017–350 (1–21,000)	16.7–58.3 (1,000–3,500)	80–85
Vane	7–21 (70–210)	0.08–10 (5–600)	10–45 (600–2,700)	80–95
Radial piston	7–815 (70–815)	0.08–12.7 (5–760)	16.7–56.7 (1,000–3,400)	85–95
Axial piston	14–81.5 (140–815)	0.08–12.7 (5–760)	8.33–71.7 (500–4,300)	90–95

In Table 1.3 some typical values of the operational characteristics of pumps are presented. Similar values are applicable to hydraulic motors.

In the next sections, the functional and construction principles of these hydrostatic machines are presented. Although pumps and motors are very similar, some specific construction aspects—such as internal channels for lubrication, external leakage drain, seals, and so forth—differ since motors do not have a port under low pressure all the time, as in the case of pumps.

Therefore, a pump cannot be used as a motor and vice-versa, unless the component has been designed to carry out both functions.

1.4.1.1 Gear Pump and Motors

External gear pumps and motors. This type of hydrostatic machine consists of a pair of equal gears assembled in housing with one inlet and one outlet, enclosed by two side plates. The drive gear is responsible for the external motion transmission and the driven gear runs free in its shaft.

According to Figure 1.15 (pump), fluid transport cells are formed between two consecutive teeth of each gear and the housing through the rotational movement. At the same time, the ungearing

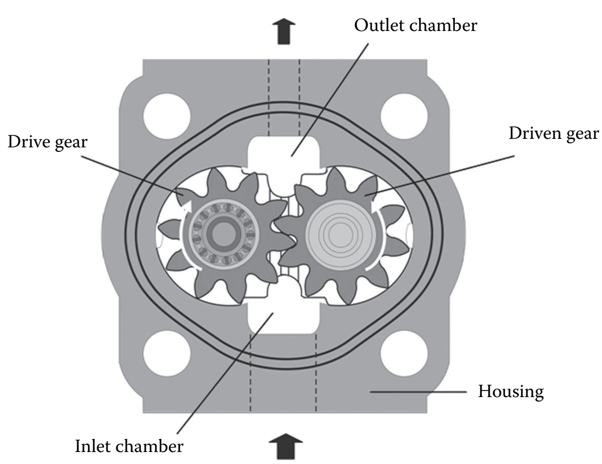


FIGURE 1.15 External gear pump (and motor).

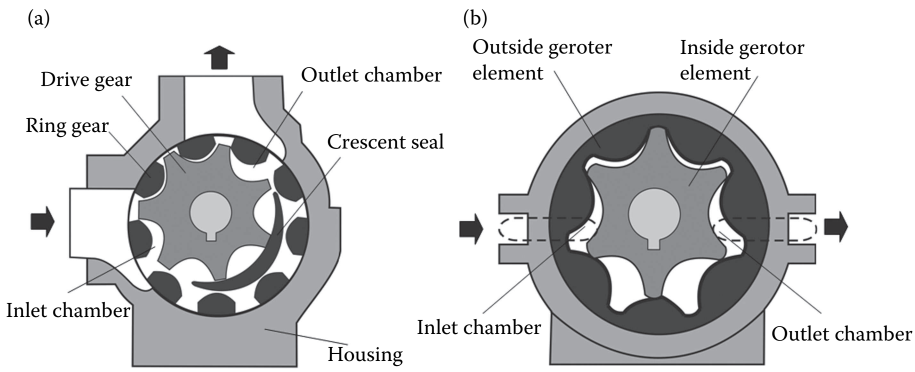


FIGURE 1.16 Internal gear pumps: (a) Crescent-seal type; (b) Gerotor type.

produces new cells to which the fluid is suctioned. In the outlet chamber the continuous gearing pushes the fluid out to the outlet port.

It is generally agreed that the gear pump is the most robust and rugged type of fluid power pump and thus its use is predominant in hydraulic services and also very intensive in industrial machines.

Gear pumps and motors are not very sensitive to fluid viscosity variations and to fluid contamination. However, since the outlet and inlet ports are opposite to one other, the forces over the gear axis are unbalanced. This limits the maximum values of pressure and flow rate.

As a consequence of the friction between the gears and the side plates, and the fluid leakage between the tips of the gears and across the side plates, the overall efficiency is lower than that of solutions based on the other construction principles.

Internal gear pumps. Given the possibility of operating under high pressures with low ripple pressures and low noise, these pumps are used in several systems such as injection machines, hydraulic presses, machine tools, and so forth. The operational principle is the same as that of external gear machines—that is, the continuous tooth ungearing and gearing of a gear pair.

The crescent seal internal gear pump consists of a small internal gear and a larger ring gear (see Figure 1.16a**)**. The small internal gear is driven by the prime mover. The internal gear meshes with the ring gear and turns it in the same direction. The sealing of the high-pressure chamber from the pump inlet is achieved by a crescent seal between the upper teeth of the internal small gear and the upper teeth of the ring gear. In the gerotor gear pump, the inner gerotor has one less tooth than the outer element (Figure 1.16b). The internal gear is driven by the prime mover and, in turn, drives the outer element in the same direction [7].

In the same way as in external gear pumps, internal gear pumps are unbalanced, limiting the maximum pressure and efficiency. Furthermore, the gear pump design does not allow the displacement to be varied.

1.4.1.2 Screw Pumps

Screw pumps for fluid power systems are composed of two or more helical screws assembled inside housing. The relative movement of the screws can be obtained driving one shaft where the movement is transmitted to the others by either their own gearing or by external gears mounted on the shafts. An illustration of this type of pump is shown in Figure 1.17.

Each screw thread is matched to carry a specific volume of fluid. Fluid is transferred through successive contact between the housing and the screw flights from one thread to the next. Its

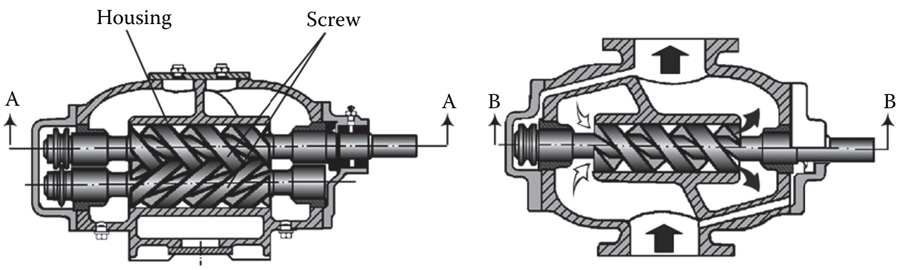


FIGURE 1.17 Screw pump. (From Linsingen, I. von, *Fundamentos de Sistemas Hidráulicos*, 3rd ed., Florianópolis, Brazil: UFSC Ed., 2008. With permission.)

operational characteristics imply that the flow does not present pulsation and the unbalanced forces are axial, being compensated for easily.

Screw pumps are generally used for hydraulic systems where high flow rates are necessary and they are also suitable for high pressures. The disadvantages are their low efficiency and high cost.

1.4.1.3 Vane Pumps and Motors

Fixed-displacement vane pumps and motors. Vane machines are comprised of a cylindrical rotor with vanes sliding in its grooves. This set runs inside a cam ring and the sides of the rotor and vanes are sealed by side bushings (port plates). Figure 1.18 presents an illustration of this type of machine.

The vanes are forced against the internal surface of the cam ring due to centrifugal force and either high pressure applied on the vane bottom or the force of the spring mounted on the vane bottom.

Between two consecutive vanes, rotor, cam ring and port plates, fluid transport cells are formed that increase in the inlet chamber and decrease in the outlet chamber. The port plates include apertures connecting these chambers to the external ports of the machine.

In the construction principle shown in Figure 1.18 the low and high pressures act appositively over the axis, causing unbalanced forces and limiting the maximum work pressure of the pump or motor. An alternative is the balanced vane pump shown in Figure 1.19 where there are two low-pressure chambers and two high-pressure chambers and thus the resultant radial forces tend to be null.

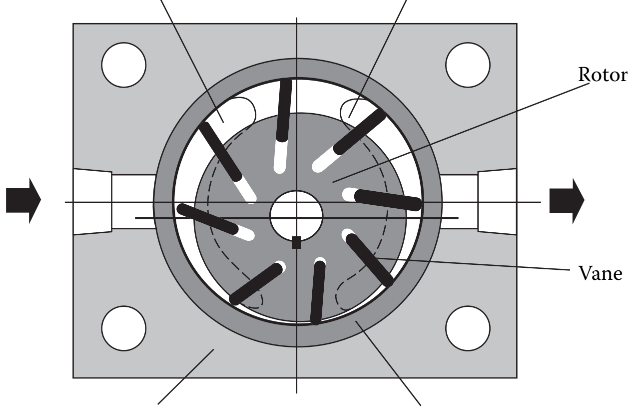


FIGURE 1.18 Vane pump. (From Linsingen, I. von, *Fundamentos de Sistemas Hidráulicos*, 3rd ed., Florianópolis, Brazil: UFSC Ed., 2008. With permission.)

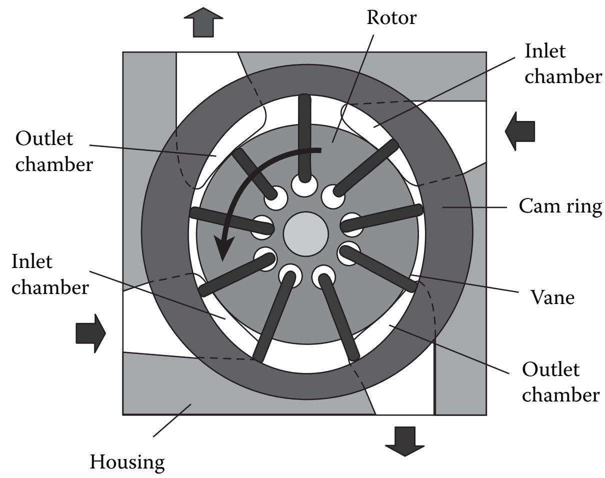


FIGURE 1.19 Balanced vane pump.

The total pumping flow results from the superposition of the flow rate from the two outlet chambers. The resulting amplitude and frequency at the outlet port is dependent on the number of vanes, where an odd number of vanes is advantageous, since the volumes discharged from each discharge chamber are not in phase.

Variable-displacement vane pumps. The variation in the volumetric displacement in vane pumps is achieved by moving the cam ring and, therefore, changing the eccentricity between it and the rotor. This can be seen in Figure 1.20, where the flow direction can also be inverted without changing the rotational frequency direction. The hydraulic circuit shown in Figure 1.11 is an example of the use of this type of pump.

Variable-displacement pumps can also include internal pressure compensation as shown in Figure 1.21. In this case, the maximum eccentricity is obtained while the internal pressure in the discharge chamber produces a force lower than the spring force. When the outlet pressure increases over the pre-load force of the spring, the cam ring moves against the spring, changing the flow rate delivered.

In general, fluid leakage in vane pumps occurs between the high- and low-pressure sides of the vanes and across the side bushings, which results in decreased volumetric efficiency and, hence, reduced flow output. The unbalanced design suffers from shortened bearing life because of the unbalanced thrust force within the pump.

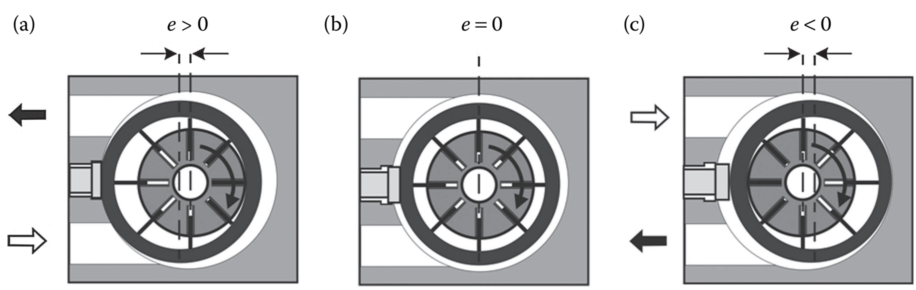


FIGURE 1.20 Illustration of the volumetric displacement variation: (a) Regular flow; (b) Null flow; (c) Reverse flow. (From Linsingen, I. von, *Fundamentos de Sistemas Hidráulicos*, 3rd ed., Florianópolis, Brazil: UFSC Ed., 2008. With permission.)

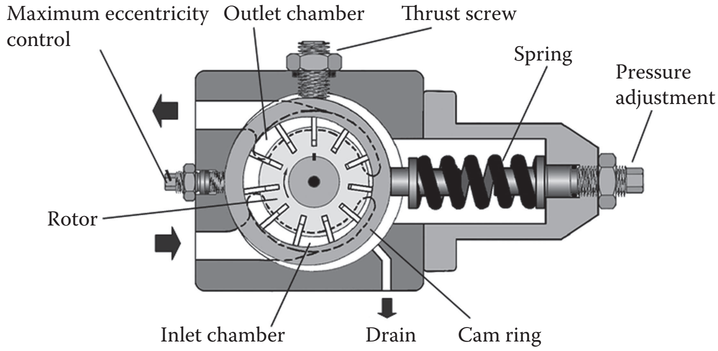


FIGURE 1.21 Variable-displacement vane pump with pressure compensation. (From Linsingen, I. von, *Fundamentos de Sistemas Hidráulicos*, 3rd ed., Florianópolis, Brazil: UFSC Ed., 2008. With permission.)

1.4.1.4 Piston Pumps and Motors

Piston machines have radial clearances in their main movable parts of between 2 and 5 mm. Consequently, they can operate under higher pressures and lower volumetric losses when compared with other hydrostatic machines.

According to the position of the pistons in relation to the shaft, these machines are classified as axial piston pumps (swash plate and bent-axis) and radial piston pumps.

Fixed-displacement axial piston machines. In this type of machine the pistons run in cylindrical holes machined in a cylinder block. The alternative movement of each piston is obtained by the rotary movement of the cylinder block or the swash plate.

a. Swash plate design

As shown in Figure 1.22, axial machines can be constructed with either rotary or stationary swash plates. In the motor shown in Figure 1.22a, the cylinder block is stationary and the swash plate is rigid with the shaft. In Figure 1.22b, the swash plate is stationary and the cylinder block rotates with the shaft. The swash plate angle defines the piston stroke and, hence, the volumetric displacement [1,11].

The valve plate identified in this figure consists of a plate with circumferential apertures and its function is to connect the inlet and outlet ports to the bottom of each piston.

In this type of machine there is a continuous leakage that is necessary for the lubrication of parts with relative mechanical movement such as that between the valve plate and the cylinder block, and that between the cylinder block and the swash plate. Therefore, a port for external drainage is required.

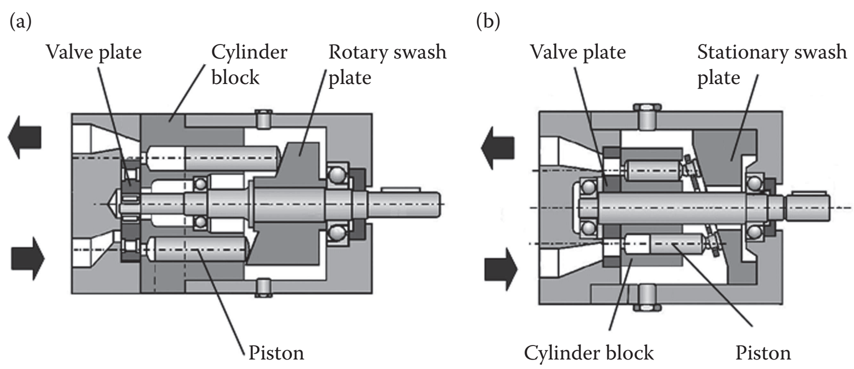


FIGURE 1.22 Swash plate design: (a) Motor with rotary swash plate; (b) Pump with stationary swash plate. (From Linsingen, I. von, *Fundamentos de Sistemas Hidráulicos*, 3rd ed., Florianópolis, Brazil: UFSC Ed., 2008. With permission.)

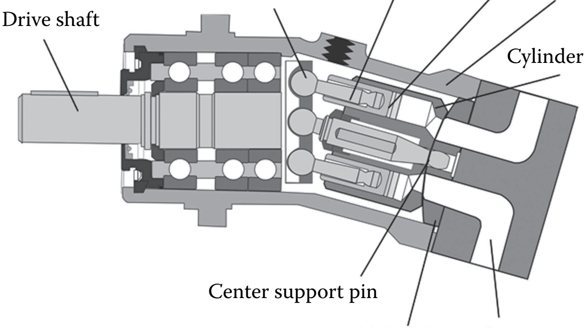


FIGURE 1.23 Illustration of axial piston machine bent-axis design. (From Frankenfield, T.C. *Using Industrial Hydraulics*, 2nd ed., Penton Publishing, 1985, ISBN-13: 9780932905017. With permission.)

b. Bent-axis design

In this design the cylinder block is mounted obliquely in relation to the driven shaft (Figure 1.23). The piston rods are coupled to the driven shaft by spherical articulations such that the rotary movement of the cylinder block produces the alternating piston movement. The connection between the pistons and the inlet and outlet ports is through the valve plate, as shown in this figure.

Since pistons have no lateral forces, angles of around 25°, and even 40°, are allowable. In relation to the swash plate, the bent-axis type has as disadvantages a greater occupied volume and higher moment of inertia. On the other hand, it has higher efficiency and less sensitivity to contaminants.

Variable-displacement axial piston machines. The swash plate machines can also have variable volumetric displacement by changing the swash plate angle. An angle equal to zero corresponds to null flow rate and the maximum positive angle produces the maximum volumetric displacement and, consequently, the maximum flow rate supplied by a pump or consumed by a motor. When a negative angle is allowed, the machine has two flow directions. In the same way, in the bent-axis type the angle between the cylinder block/valve plate axis and the shaft can also be controlled.

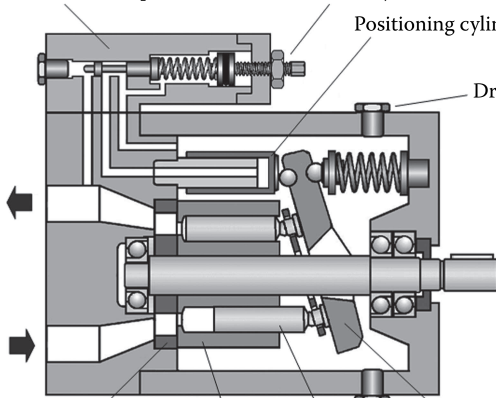


FIGURE 1.24 Variable-displacement axial piston pump, swash plate design, with pressure compensation. (From Linsingen, I. von, *Fundamentos de Sistemas Hidráulicos*, 3rd ed., Florianópolis, Brazil: UFSC Ed., 2008. With permission.)

Variable-displacement piston pumps lend themselves to the incorporation of various mechanisms that will alter their performance. One typical example is the pressure-compensated pump where the hydraulic mechanism will alter the pump displacement to limit the outlet pressure to some pre-adjusted value. Figure 1.24 presents a pressure-compensated axial piston pump (swash plate type).

Other commercial solutions allow the control of the hydraulically supplied power according to the system demand. Electro-hydraulic pumps using proportional valves are also available to design circuits with transformation control, which means directly through the primary conversion function. The circuits presented in Figure 1.11 and 1.12 are examples of the use of variable-displacement pumps.

Radial piston machines (fixed- and variable-displacement). In these machines, the piston axes are perpendicular to the driven shaft. Depending on the construction principle, the pistons can be mounted in a star format around the shaft or in line on a crankshaft.

Figure 1.25 shows the basic configuration of a three-piston pump. Each hollow piston consists of an inlet non-return valve, a spring, a piston barrel, a pumping chamber, an outlet non-return ball, and a support bearing. As the driven shaft is rotated, the spring holds the base of the piston in contact with the eccentric cam shaft. The downward motion of the piston causes the volume to increase in the pumping chamber. This creates a reduced pressure that enables the inlet check valve to open, thereby allowing oil to enter the pump chamber. The oil enters the chamber by way of a groove machined into the cam-shaft circumference. Further rotation of the cam shaft causes the piston to move back into the cylinder barrel. The rapid rise in chamber pressure closes the inlet check valve. When the rising pressure equals the system pressure, the outlet check valve opens, allowing flow to exit the piston and pass to the pressure port of the pump. The resulting flow is the sum of all the piston displacements. The number of pistons that a radial pump can have is only limited by the spatial restrictions imposed by the size of the pistons, housing, and cam shaft.

Figure 1.26 illustrates a variable-displacement pump with pressure compensation composed of a cam ring eccentrically mounted relative to a cylinder block. The alternative movement of the pistons is obtained by the rotary movement of the cylinder block reaming the pistons in contact with the cam ring through shoes. The shoes slide on a trail fixed on the cam ring. The fluid suction and discharge occurs via semicircular ports and pipes machined on a stationary piece inside the driven shaft.

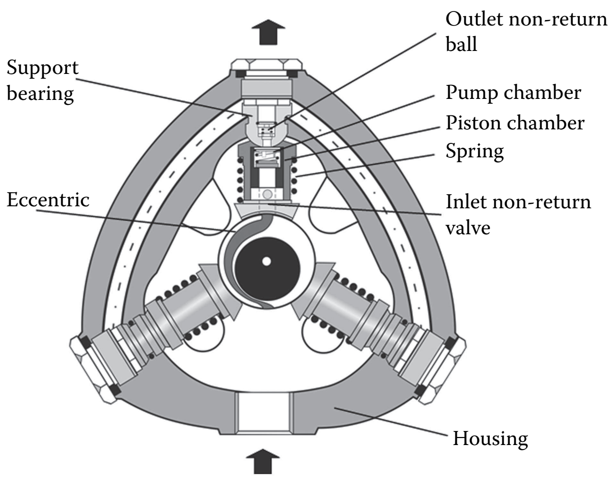


FIGURE 1.25 Radial piston pump. (From Frankenfield, T.C. *Using Industrial Hydraulics*, 2nd ed., Penton Publishing, 1985, ISBN-13: 9780932905017. With permission.)

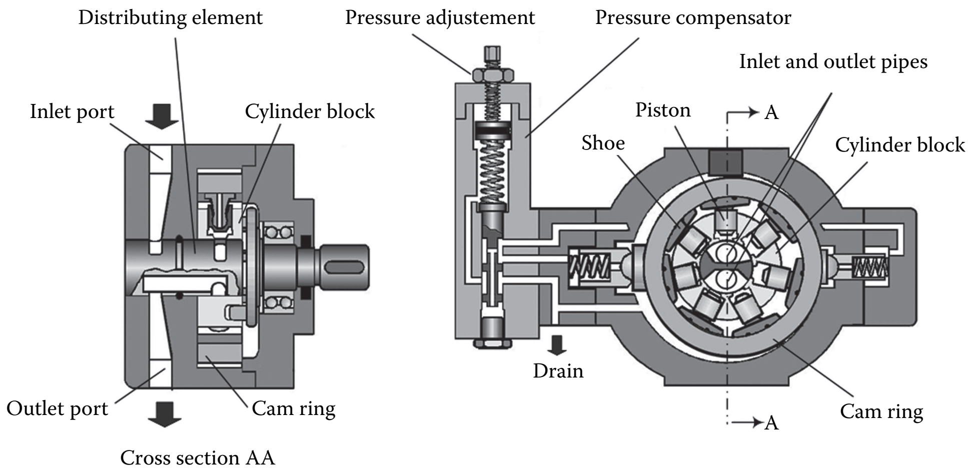


FIGURE 1.26 Radial piston pump with pressure compensation. (From Linsingen, I. von, *Fundamentos de Sistemas Hidráulicos*, 3rd ed., Florianópolis, Brazil: UFSC Ed., 2008. With permission.)

In general, the radial piston pump has a higher continuous-pressure capability than any other type of pump (Table 1.3). However, it should be noted that for extremely high-pressure applications, the volumetric displacements of radial pumps are usually not larger than 2.4×10^{-6} m^3/rad (0.015 dm^3/rev).

1.4.1.5 Pump and Motor Performance Characteristics

In Section 1.2.4 the flow rate and torque equations of pumps and motors was presented, where they were considered as ideal machines, without internal or external leakage and friction. However, these losses are present in real machines and they are identified in a general way by the volumetric, mechanical and overall efficiencies.

Consider Figure 1.27, where the main variables associated with pumps and motors are presented. Based on this figure, the equations given below describe the steady-state behavior and efficiency expressions valid for pumps and motors.

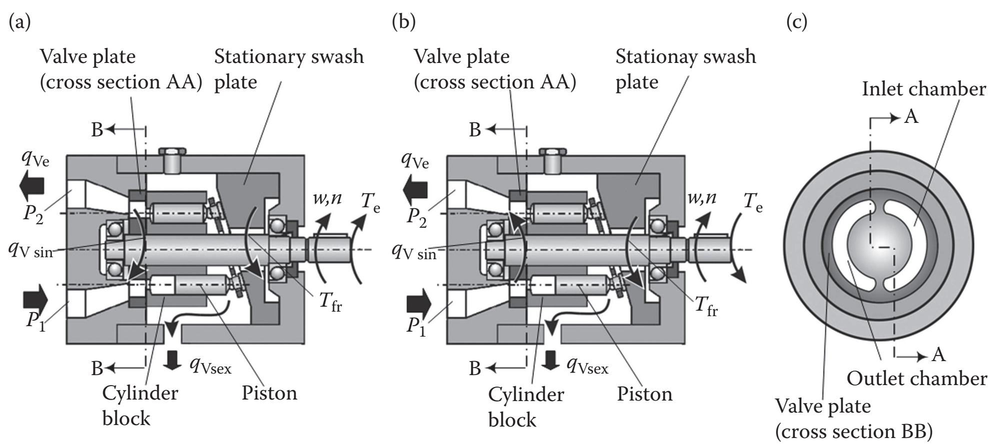


FIGURE 1.27 Main variables associated with: (a) Pumps; and (b) Motors (c) Cross section BB.

Flow rate and volumetric efficiency. The volumetric losses in hydrostatic machines occur as a consequence of the mechanical clearances, pressure drops and relative velocity between movable parts. Cavitation and fluid aeration also induce flow losses. However, since these phenomena should not occur under normal operational conditions, they are not considered in the mathematical description of volumetric efficiency [12].

The theoretical flow rate given by Equation 1.10 and rewritten in Equation 1.15 is dependent on the volumetric displacement (D). This parameter is calculated according to the geometric dimensions or by measuring the absorbed or discharged volume for a complete revolution with differential pressure close to zero.

$$q_{\mathrm{Vtc}} = D \cdot \omega = D \cdot 2\pi \cdot n. \tag{1.15}$$

The effective flow rate (discharged) (q_{Ve} [m^3/s] or [L/min]) in pumps is lower than the theoretical flow rate (q_{Vtc} [m^3/s] or [L/min]) and can be determined by

$$q_{\mathrm{Ve}}^{\mathrm{P}} = q_{\mathrm{Vtc}}^{\mathrm{P}} - q_{\mathrm{Vs}}^{\mathrm{P}}, \tag{1.16}$$

where q_{Vs} [m^3/s] or [L/min] is the flow rate loss that can be due to internal leakage ($q_{\mathrm{V\,sin}}$) (between the pump chambers), or external leakage ($q_{\mathrm{V\,sex}}$), as in vane and piston pumps that have a drain port.

In motors, the effective flow rate (inlet) (q_{Ve} [m^3/s] or [L/min]) is higher than the theoretical flow rate (q_{Vtc} [m^3/s] or [L/min]), since part of the fluid is lost through leakage (q_{Vs}). Therefore:

$$q_{\mathrm{Ve}}^{\mathrm{M}} = q_{\mathrm{Vtc}}^{\mathrm{M}} + q_{\mathrm{Vs}}^{\mathrm{M}}. \tag{1.17}$$

The volumetric efficiency is then calculated through the following expressions:

For pumps:

$$\eta_{\mathrm{V}}^{\mathrm{P}} = \frac{q_{\mathrm{Ve}}^{\mathrm{P}}}{q_{\mathrm{Vtc}}^{\mathrm{P}}}. \tag{1.18}$$

For motors:

$$\eta_{\mathrm{V}}^{\mathrm{M}} = \frac{q_{\mathrm{Vtc}}^{\mathrm{M}}}{q_{\mathrm{Ve}}^{\mathrm{M}}}. \tag{1.19}$$

The leakage in pumps and motors is approximately laminar and thus under operational conditions, with approximately constant temperature, the leakage is proportional to the pressure difference ($q_{\mathrm{V\,sin}} \propto \Delta p$). Hence, the volumetric efficiency also changes proportionally to the pressure difference.

Torque and mechanical efficiency. Based on Equation 1.8, the theoretical torque (T_{tc} [N·m]) can be expressed as

$$T_{\mathrm{tc}} = D \cdot \Delta p, \tag{1.20}$$

where Δp [Pa] is the pressure difference between the inlet and outlet ports of the pump or motor.

However, this is not the real torque in the machine shaft, since there are losses associated with mechanical friction and fluid viscous friction [12].

For pumps, the effective torque required in the driven shaft (T_e [N · m]) is higher than the theoretical torque (T_{tc} [N · m]), that is

$$T_e^P = T_{tc}^P + T_{fr}^P, \tag{1.21}$$

where T_{fr} [N · m] is the friction torque.

In the case of motors, the effective torque available in the shaft (T_e) is lower than the theoretical torque (T_{tc}), such that

$$T_e^A = T_{tc}^A - T_{fr}^A. \tag{1.22}$$

Consequently, the mechanical efficiencies are defined by the following expressions:

For pumps:

$$\eta_m^P = \frac{T_{tc}^P}{T_e^P}. \tag{1.23}$$

For motors:

$$\eta_m^A = \frac{T_e^A}{T_{tc}^A}. \tag{1.24}$$

Power and overall efficiency. The useful power of a pump is the hydraulic power at the outlet port and for a motor it is the mechanical power at the driven shaft. The useful power can be described by:

For pumps:

$$P_h = q_{Ve} \cdot \Delta p \cong q_{Ve} \cdot p_2 = q_{Vtc} \cdot p_2 \cdot \eta_V, \tag{1.25}$$

where $\Delta p = p_2 - p_1$, p_2 p_2 is the pressure in the outlet port (discharge) and p_1 is the pressure in the inlet port (suction). Since the pressure p_1 is close to atmospheric pressure ($p_1 \approx 0$ Pa [gauge pressure]), this expression can be written considering only the output pressure (p_2).

For motors:

$$P_m = T_e \cdot \omega = T_e \cdot 2\pi \cdot n = T_{tc} \cdot 2\pi \cdot n \cdot \eta_m. \tag{1.26}$$

Or applying Equation 1.20:

$$P_m = D \cdot (p_1 - p_2) \cdot 2\pi \cdot n \cdot \eta_m = q_{Vtc} \cdot (p_1 - p_2) \cdot \eta_m. \tag{1.27}$$

The drive power is the mechanical power at the shaft for a pump and the hydraulic power at the inlet port for a motor. Hence:

For pumps:

$$P_m = T_e \cdot \omega = T_e \cdot 2\pi \cdot n = \frac{T_{tc} \cdot 2\pi \cdot n}{\eta_m}. \tag{1.28}$$

Or applying Equation 1.20:

$$P_m = \frac{D \cdot (p_2 - p_1) \cdot 2\pi \cdot n}{\eta_m} = \frac{q_{Vtc} \cdot (p_2 - p_1)}{\eta_m}. \tag{1.29}$$

For motors:

$$P_h = q_{Ve} \cdot \Delta p = q_{Ve} \cdot (p_1 - p_2) = \frac{q_{Vtc} \cdot (p_1 - p_2)}{\eta_V}. \quad (1.30)$$

Consequently, the overall efficiency is defined as
For pumps:

$$\eta_t^P = \frac{P_h^P}{P_m^P} = \eta_V^P \cdot \eta_m^P. \quad (1.31)$$

For motors:

$$\eta_t^M = \frac{P_m^M}{P_h^M} = \eta_V^M \cdot \eta_m^M. \quad (1.32)$$

1.4.1.6 Characteristic Curves

The variables presented in the section above are frequently presented in graphs as a function of the pressure difference to which the hydrostatic machine will be submitted. Moreover, operational conditions like temperature and rotational frequency, and fluid specification, need to be pre-fixed when these operating curves are obtained experimentally.

Fixed-displacement pumps. A typical characteristic curve is shown in Figure 1.28 where the curve of the effective flow rate (q_{Ve}) represents the basic characteristic of a pump, where its scope shows the operating pressure influence on the leakage. From this curve the volumetric efficiency curve (η_V) is obtained using Equation 1.18.

The mechanical efficiency (η_m) increases with the fluid leakage, improving the lubrication and reducing the friction torque (Equations 1.22 and 1.23). The useful power (P_h) is a linear function of the effective flow rate and the output pressure (Equation 1.25) and, in turn, the drive power (P_m) is dependent on the mechanical losses (Equation 1.28). According to Equation 1.31, the curve of the overall efficiency (η_t) is determined by either the useful power to drive power ratio, or the volumetric and mechanical efficiency product.

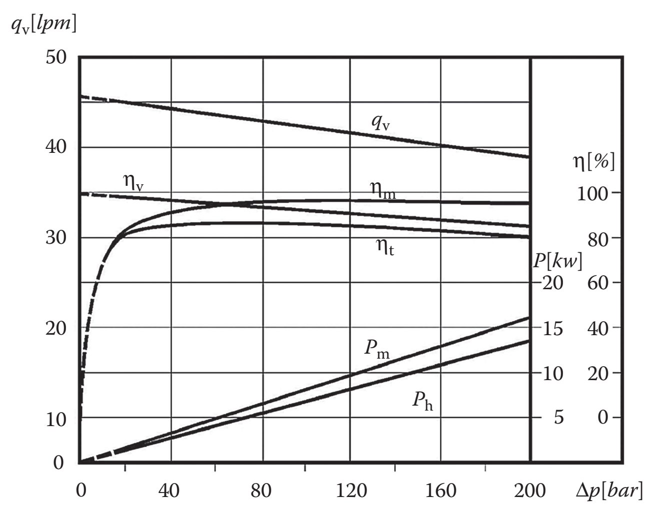


FIGURE 1.28 Operating curves of a fixed-displacement pump. (From Linsingen, I. von, *Fundamentos de Sistemas Hidráulicos*, 3rd ed., Florianópolis, Brazil: UFSC Ed., 2008. With permission.)

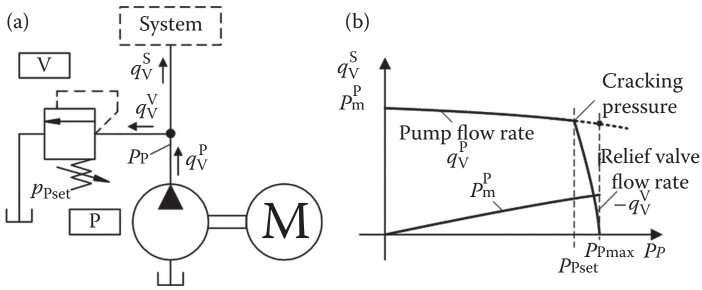


FIGURE 1.29 Fixed-displacement pump with relief valve: (a) Hydraulic circuit; (b) Characteristic curve.

The strong reduction in the overall efficiency at low pressures is a consequence of poor lubrication and high friction in this operational range. For this reason, the manufacturers recommend a minimal operation pressure, with the aim of not reducing the useful life of the pump. In the case of Figure 1.28, the pump must operate above 2 MPa (20 bar) [1].

Fixed-displacement pumps with relief valve. The fixed-displacement pump is frequently used together with a relief valve since the flow from the pump needs to be diverted to a reservoir when it is not being used by the system (Figure 1.29a).

The effective flow rate supplied to the system (q_V^S) is obtained by combining the characteristic curves of the two components, as can be seen in Figure 1.29b. The cracking pressure is the pressure adjusted at the relief valve (p_{Pset}) at which it opens. From this operational point onward, any increase in the system pressure (p_P) causes a significant decrease in the flow rate to the system (q_V^S).

For pressures lower than the cracking pressure the system flow rate is equal to the pump flow rate ($q_V^S = q_V^P$), and for higher pressures part of the flow is diverted to the relief valve ($q_V^S = q_V^P - q_V^V$). At the maximum supply pressure (p_{Pmax}) the relief valve flow rate is equal to the pump flow rate ($q_V^V = q_V^P$), which means that all the hydraulic power (P_h^P) is being dissipated at the relief valve, increasing the temperature of the fluid that returns to the reservoir. Consequently, the pump drive power (P_m^P) continues to increase after the cracking pressure has been reached.

Variable-displacement pumps with pressure compensation. In the case of variable-displacement pumps with pressure compensation, as shown in Figure 1.21, 1.24, and 1.26, the use of a relief valve is not required, although it can be installed in the hydraulic circuit for safety reasons.

As shown in Figure 1.30a, the system flow rate is always equal to the pump flow rate. When there is no demand from the system, the pressure increases above the set pump pressure, changing

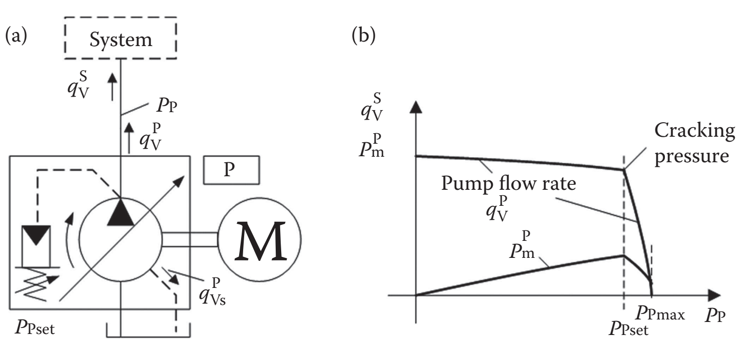


FIGURE 1.30 Variable-displacement pump with pressure compensation: (a) Hydraulic circuit; (b) Characteristic curve.

its volumetric displacement (*D*). Therefore, the power consumption is reduced when the cracking pressure is surpassed, as illustrated in Figure 1.30b. One can observe that this power is not null when $q_V^S = q_V^P = 0$ since there is always a small lubrication flow rate (q_{Vs}^P), which is drained to the reservoir [13].

1.4.2 Hydraulic Cylinders

Hydraulic systems are designed to provide controlled mechanical energy through linear or angular movement. The action over the external environment occurs on the last block of the functional chain shown in Figure 1.1, the secondary energy conversion, and it is performed by the hydraulic actuators, which in this case are the motors, oscillators and cylinders.

The basics of motors were described in the section above, since their construction principles are the same as those of pumps.

The hydraulic oscillators also produce angular movement but they do not provide continuous rotation and the angle is limited to a value below 360°. Their construction is derived from the hydraulic motor design (from the vane motor, for example) or from double-acting hydraulic cylinders with mechanical transmission converting linear into angular displacement.

In turn, cylinders are the hydraulic actuators most used in hydraulic systems. They are typically comprised of (1) a barrel, (2) piston assembly, (3) piston rod, (4) end caps, (5) ports, and (6) seals, as shown in Figure 1.31. The piston provides the effective area against which the fluid pressure is applied and supports the piston assembly and rod. The opposite end of the rod is attached to the load. The cylinder bore, end caps, ports, and seals maintain a fluid-tight chamber in which the fluid energy is contained. Whether the rod will extend or retract is dependent on the port to which the fluid is directed.

Hydraulic cylinders are classified according to different premises, with two of them being particularly important in terms of understanding the use and behavior of cylinders. Hence, in relation to the operating principle they are sub-divided into single- and double-acting (single- and double-effect) and, considering the area ratio, they are classified as either symmetrical or asymmetrical (non-differential or differential) cylinders.

In Figure 1.32, the several types of single-acting cylinders are symbolically represented. In this construction principle, the hydraulic power is available in only one direction of movement—that is, on either extension or retraction. In the opposite direction the movement results from an external force (including gravitational force) as shown in Figure 1.32a, b and e, or from an internal spring force as in Figure 1.32c and d.

Unlike the other types, telescopic cylinders have two or more stages which, when fully extended, can produce a stroke that exceeds the length of the cylinder when fully retracted. The symbol shown in Figure 1.32e presents a two-stage model.

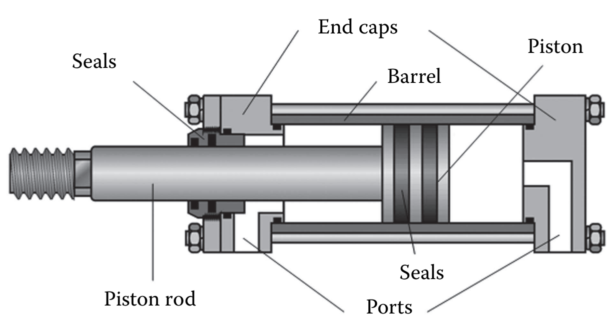


FIGURE 1.31 Main parts of a hydraulic cylinder. (From Linsingen, I. von, *Fundamentos de Sistemas Hidráulicos*, 3rd ed., Florianópolis, Brazil: UFSC Ed., 2008. With permission.)

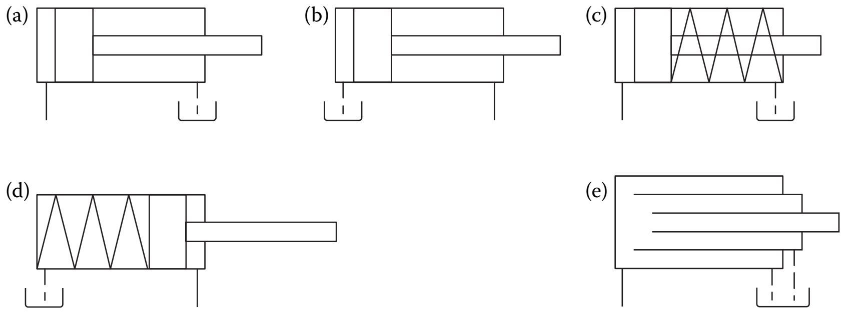


FIGURE 1.32 Single-acting cylinders: (a) Retraction by external force; (b) Extension by external force; (c) Retraction by spring; (d) Extension by spring; (e) Telescopic cylinder with retraction by external force.

As a consequence of the inevitable leakage between the piston and barrel, the non-active chambers must have an external drain avoiding counter-pressure and cylinder blocking.

Some examples of double-acting cylinders are shown in Figure 1.33. In this type of cylinder, the effective work is carried out in both directions of movement (extension and retraction).

The most common double-acting cylinder is the single-rod cylinder (Figure 1.33a), which is classified as an asymmetric (differential) cylinder since the piston areas on the bottom-side and the rod-side are different. As a consequence, the velocity and hydraulic force are generally different during the extension and retraction movements.

The double-rod cylinders (Figure 1.33b) can be designed with rods of the same diameter (symmetric [non-differential] cylinder) and with different diameters (asymmetric [differential] cylinder). In the case of symmetric cylinders the hydraulic force and velocity are the same, considering the same loading and supplied flow rate, during extension and retraction.

Tandem actuating cylinders (Figure 1.33c) consist of two or more cylinders arranged one behind the other but designed as a single unit. The main operational characteristic is the greater force when compared with a regular cylinder of the same diameter.

In the same way as in telescopic single-acting cylinders, the double-acting cylinders (Figure 1.33d) have the advantage of being compact. However, since their construction costs are higher than those of other designs, their use is somewhat limited.

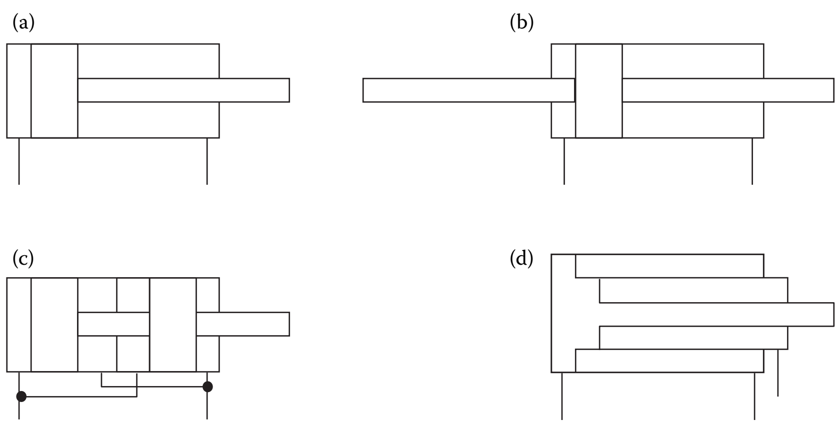


FIGURE 1.33 Double-acting cylinders: (a) Single rod; (b) Double rod; (c) Tandem; (d) Telescopic.

1.4.2.1 Hydraulic Cylinder Behavior

The hydraulic cylinders are intended for use under several operational conditions, including motion with a constant velocity, positioning control, force control, or just to provide a force to fix something.

In all these situations, the motion achieved is influenced by factors such as inertia, fluid compressibility and friction, and must be considered in the analysis and design of the hydraulic system [14,15].

By observing Figure 1.34 one can identify two main parts to be modeled: the movable piston and the fluid in the cylinder chambers.

The linear motion of the piston is described by Newton's second law, which establishes that the sum of the forces must be equal to the product of the mass and acceleration ($M_t \cdot a = M_t \cdot d_2x/dt_2$). Therefore, for an asymmetric double-acting cylinder, as shown in Figure 1.34, the motion equation is

$$(A_A \cdot p_A) - (A_B \cdot p_B) = M_t \cdot \frac{d^2 x_p}{dt^2} + F_{fr} + F_e, \tag{1.33}$$

$A_A \cdot p_A$ being the force in area A_A caused by the pressure in chamber A (p_A), $A_B \cdot p_B$ is the force in area A_B caused by the pressure in chamber B (p_B), and x_p is the piston displacement. F_{fr} is the friction force associated with the cylinder and external load and F_e is the effective force available at the rod piston to move the load. The total mass (M_t) includes the piston mass (M_p) and external mass (load) (M_{ex}).

Equation 1.33 demonstrates that a hydraulic force $(A_A \cdot p_A) - (A_B \cdot p_B)$ is necessary in order to overcome the external forces, friction force and inertia. Therefore, for the piston to achieve a new position or velocity the chamber pressures must change.

The dynamic behavior of the pressure in the chambers is determined by the conservation of mass principle as presented in Section 1.2.2. Hence, applying Equation 1.4 to chamber A (Figure 1.34) the following expression is obtained:

$$q_{VA} = A_A \cdot \frac{dx_p}{dt} + q_{V\sin} + \frac{V_A}{\beta} \cdot \frac{dp_A}{dt}. \tag{1.34}$$

For the cylinder extension, the input flow rate at port A (q_{VA}) leads to a pressure increase (dp_A/dt) caused by the fluid compression. With the pressure increase internal leakage ($q_{V\sin}$) can occur and the cylinder will start to move. The product of the area and velocity ($A_A \cdot dx_p/dt = A_A \cdot v_p$) establishes the chamber volume variation with the piston movement and this volume is occupied by the fluid.

For chamber B the fluid behavior is expressed by Equation 1.35, such that, on the cylinder extension q_{VB} is the flow rate induced by the piston motion at which the fluid exits the cylinder in the direction of a directional valve.

$$q_{VB} = A_B \cdot \frac{dx_p}{dt} + q_{V\sin} - \frac{V_B}{\beta} \cdot \frac{dp_B}{dt}. \tag{1.35}$$

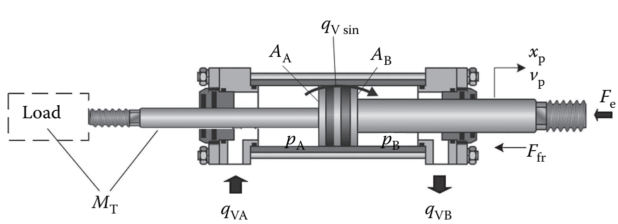


FIGURE 1.34 Parameters and variables associated with a hydraulic cylinder.

One can observe that Equations 1.33 through 1.35 are suitable for any type of cylinder, symmetrical or asymmetrical, single- or double-acting. In the case of symmetrical cylinders, the piston areas are equal ($A_A = A_B$). In the case of single-acting cylinders, the continuity equation is applied only to the controlled chamber. In the other chamber, the pressure is considered to be constant or the spring force is included in Equation 1.33.

1.4.2.2 Cylinder Performance Characteristics

Mechanical efficiency. The mechanical efficiency of the cylinder is the ratio between the theoretical force (hydraulic force) and the effective force available for the external system motion. Since the efficiency characterizes the steady-state performance of the cylinder, the cylinder is considered to have a constant velocity, equal to or differing from zero, and null acceleration. Therefore, the mechanical efficiency can be expressed by

$$\eta_m = \frac{F_e}{F_{tc}} = \frac{F_e}{F_h} = \frac{F_e}{(A_A \cdot p_A) - (A_B \cdot p_B)}. \tag{1.36}$$

Volumetric efficiency. Similarly to hydraulic motors, the volumetric efficiency is the ratio between the geometric (theoretical) flow rate and the effective flow rate through the cylinder ports, that is

$$\eta_V = \frac{q_{Vtc}}{q_{Ve}} = \frac{A_A v}{q_{VA}} = \frac{A_B v}{q_{VB}}. \tag{1.37}$$

However, cylinders remain stopped for some periods of time, as they can stay at the stroke end or in a controlled position when enclosed in a closed-loop system. Therefore, aiming to obtain representative values, this efficiency must be calculated beyond these specific operational conditions.

Power and overall efficiency. Considering the cylinder at constant velocity, the useful power (mechanical power) present at the piston rod is

$$P_m = F_e \cdot v. \tag{1.38}$$

Or applying Equation 1.36:

$$P_m = F_{tc} \cdot v \cdot \eta_m = (A_A p_A - A_B p_B) \cdot v \cdot \eta_m. \tag{1.39}$$

The drive power of a cylinder is the net hydraulic power at the cylinder ports, such that

$$P_h = q_{VA} \cdot p_A - q_{VB} \cdot p_B = \frac{A_A \cdot v}{\eta_V} \cdot p_A - \frac{A_B \cdot v}{\eta_V} \cdot p_B. \tag{1.40}$$

The overall efficiency of the cylinder is expressed by

$$\eta_t = \frac{P_m}{P_h} = \eta_V^M \cdot \eta_m^M. \tag{1.41}$$

Natural frequency and dynamic performance. The concept of efficiency is a direct way to evaluate the steady-state performance of a system. Thus, the dynamic performance can also be characterized through a simplified analysis as follows.

As seen above, the hydraulic cylinder behavior is described by differential equations. Dynamic systems like this do not respond instantaneously to an input and a behavior analysis must be carried out according to system control theory.

Most mathematical models of systems can be reduced to a second-order equation, such as that presented in Equation 1.42.

$$\frac{1}{\omega_n^2} \cdot \frac{d^2 y}{dt^2} + \frac{2 \cdot \zeta}{\omega_n} \cdot \frac{dy}{dt} + y = K_{ST} \cdot u, \tag{1.42}$$

where u is the input, y is the output, ω_n [rad/s] is the natural frequency, ζ [1 (non dimensional)] is the damping ratio and K_{ST} [output unit/input unit] is the steady-state gain of the system [16].

The response time of a second-order system to a step input is shown in Figure 1.35a. Since the abscissa is $\omega_n \cdot t$, these curves show how both the natural frequency and the damping ratio influence the dynamic response.

In Figure 1.35b the time-domain specifications used in hydraulic system design are shown. According to ISO 10770-1 [17] and ISO 10770-2 [18], the response time (t_{re}) is defined as the time required for the response to reach 90% of the final value. The settling time (t_s) is defined as the time required for the response to decrease to and remain at a specified percentage of its final value. The settling time definition is well known from control theory [16] and 5% is the percentage recommended by the standards mentioned above.

The natural frequency can be correlated with the settling time by [16]:

$$t_s = \frac{3}{\zeta \cdot \omega_n} \quad \text{for 5\% error.} \tag{1.43}$$

Since there is no algebraic correlation with the time response as defined by ISO 10770-1 [17], one can use the rise time, defined as the time required to change from 0% to 100% of the final value [16]. The expression associated with the rise time is presented in Equation 1.44 [16] and can be used to approximately calculate the natural frequency when the time response is known.

$$t_r = \frac{1}{\omega_n \cdot \sqrt{1-\zeta^2}} \cdot \arctan\left(\frac{\sqrt{1-\zeta^2}}{\zeta}\right). \tag{1.44}$$

On continuing the study of the hydraulic cylinder and its loading, Equations 1.33 through 1.35 can be combined such that, for ports A and B closed ($q_{VA} = q_{VB} = 0$), the system model is

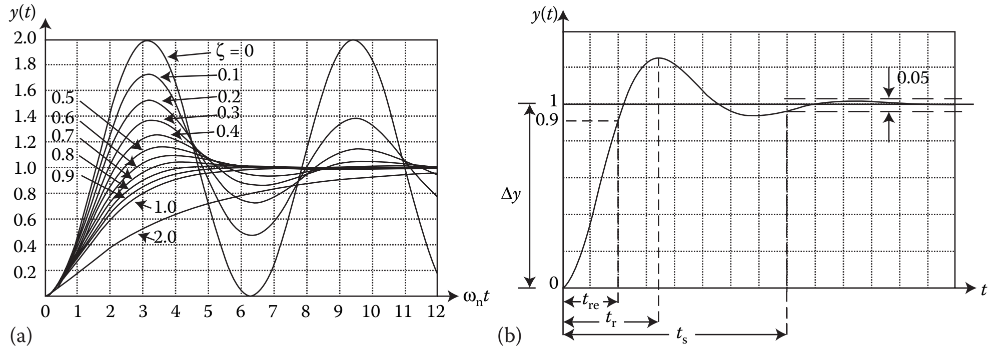


FIGURE 1.35 Response of a second-order system to a unit step input: (a) Influence of the natural frequency and damping ratio; (b) Time-domain specifications.

$$\frac{M_t}{\beta e \cdot \left(\frac{A_A{}^2}{V_A} + \frac{A_B{}^2}{V_B}\right)} \cdot \frac{d^2 x}{dt^2} + x = 0. \quad (1.45)$$

Through comparing this equation with Equation 1.42, it can be concluded that the natural frequency of the cylinder with loading is expressed by

$$\omega_n = \left[\frac{\beta_e}{M_t} \cdot \left(\frac{A_A{}^2}{V_A} + \frac{A_B{}^2}{V_B}\right)\right]^{1/2}. \quad (1.46)$$

Besides Equation 1.46 being valid for asymmetrical double-acting cylinders, it can also be applied to symmetrical double-acting cylinders considering $A_A = A_B$. For application to single-acting asymmetrical cylinders the term related to the non-controlled chamber needs to be excluded (A_A^2/V_A or A_B^2/V_B).

1.4.3 Directional Control Valves

One of the main functions of the directional control valves is the connection or isolation of one or more flow paths. These valves are identified according to their specific function, as will be presented below, but some characteristics are common to all of them, such as the number of ports, number of positions, and the type of control mechanism [19].

The port means the terminus of a flow path in a component, to which connections can be made. The number of ports refers only to those related to the power flow paths, thus excluding drain and pilot ports. For example, a valve with four ports [19] is commercially identified as a four-way valve.

The number of valve positions refers to the number of pre-defined states in which the valve can operate and it is related to the feasible stable positions of a movable valve element. Designations such as two-position valve or three-position valve are used in valve identifications.

Finally, control mechanisms are devices that provide an input signal to a component. Levers, solenoids, plungers, and pilots are examples of control mechanisms that are used in directional valves.

1.4.3.1 Non-return Valves (Check Valves)

The simplest type of directional control valve is a non-return valve or check valve. Its function is to permit free flow in one direction and prevent flow in the opposite direction. Figure 1.36a shows a simple non-return valve for line mounting which consists of a seat, a poppet, and a spring.

The valve remains closed to the flow until the pressure at its inlet port (A) creates sufficient force to overcome the spring force. Once the poppet leaves its seat, hydraulic fluid is permitted to flow

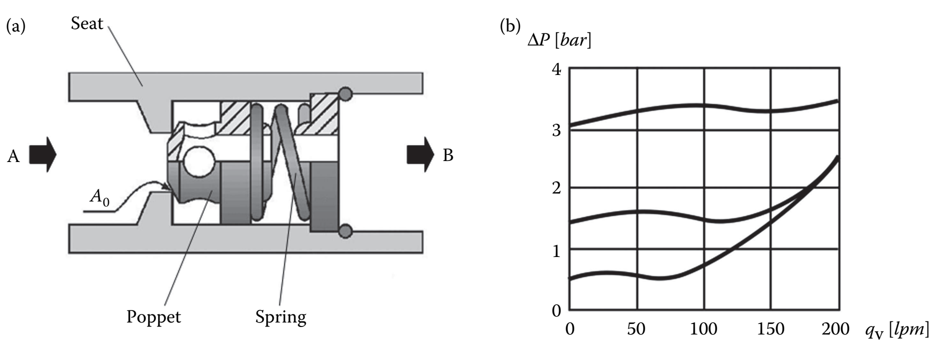


FIGURE 1.36 Single non-return valve: (a) Illustration; (b) Characteristic curve. (From Linsingen, I. von, *Fundamentos de Sistemas Hidráulicos*, 3rd ed., Florianópolis, Brazil: UFSC Ed., 2008. With permission.)

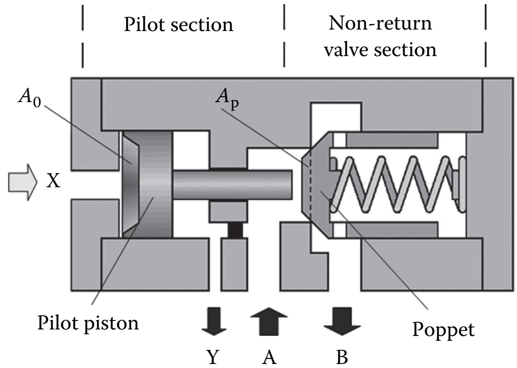


FIGURE 1.37 Pilot-operated non-return valve. (From Linsingen, I. von, *Fundamentos de Sistemas Hidráulicos*, 3rd ed., Florianópolis, Brazil: UFSC Ed., 2008. With permission.)

around and through the poppet to the valve outlet port (*B*). For this reason, a simple non-return valve can only allow flow in one direction. By changing the spring, cracking pressures between 0.05 MPa (0.5 bar) and 0.5 MPa (5 bar) can be obtained. For special applications, a no-spring version is also available.

In Figure 1.36b, characteristic curves for three different springs are presented. The cracking pressures are 0.05, 0.15, and 0.3 MPa (0.5, 1.5, and 3 bar). In each curve the pressure drop remains basically constant until a specified flow rate. Above this value the load loss in the valve increases and the valve behaves like a fixed orifice, as described by Equation 1.13.

Examples of circuits using non-return valves are shown in Figure 1.5 and 1.11. In Figure 1.12 the pump is designed with two internal non-return valves allowing the fluid suction through one port without fluid return through the other. This type of valve is also enclosed in filters, as shown in Figure 1.12, to prevent line blocking in the case of filter obstruction.

For load holding and in decompression-type hydraulic press circuits, a pilot-operated non-return valve is used. This performs the same function as the simple non-return valve described above. However, in contrast, a pilot-operated non-return valve can be piloted to remain opened when a reverse flow is required. Figure 1.37 illustrates the components of a pilot-operated non-return valve. The valve has two distinct sections—the non-return valve section and the pilot section. The non-return valve section allows free fluid flow from port *A* to port *B* while preventing reverse flow from *B* to *A* without leakage. However, if a pilot pressure signal is supplied to port *X***,** then a force is applied to the pilot piston, which forces the piston rod against the non-return valve poppet. This force then unseats the poppet, allowing free flow of fluid from port *B* to port A.

1.4.3.2 Spool-type Directional Control Valves

As presented in Sections 1.4.1 and 1.4.2, the actuators normally have two ports. If hydraulic fluid is pumped into one of the ports while the other is connected to the reservoir, the actuator will move in one direction. In order to reverse its direction of motion, the pump and reservoir connections must be reversed. The sliding spool-type directional control valve has been found to be the best way to achieve this change.

These valves have a cylindrical shaft called a "spool," which slides into a machined bore in the valve housing. The housing has ports to connect the valve to the hydraulic circuit.

The sliding spool-type directional control valves can be designed with different combinations of spool and housing. Therefore, two-way and two-position, either normally closed or normally open valves (2/2 NC or 2/2 NO), are available as well as three-way and four-way, or with more ports with three or more positions and different configurations of valve center positions.

Because of their construction characteristics, these valves present internal leakage, which can be a serious restriction in some applications. The use with pilot-operated non-return valves or counter-balanced valves is a common solution.

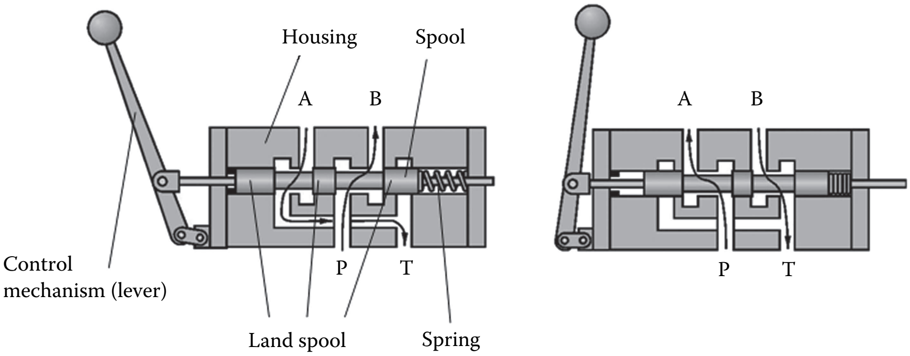


FIGURE 1.38 4/2 sliding spool-type directional control valve. (From Linsingen, I. von, *Fundamentos de Sistemas Hidráulicos*, 3rd ed., Florianópolis, Brazil: UFSC Ed., 2008. With permission.)

Two-position directional control valves. Figure 1.38 shows an illustration of a four-way, two-position, lever-controlled, spring return sliding spool-type directional control valve.

In the solution shown in this figure, the normal position (non-actuated position) establishes the flow paths *P-B* and *A-T*. While actuated by the lever the flow paths *P-A* and *B-T* are maintained.

A common use for such a valve is in a cylinder application which only requires the cylinder to extend or retract to its fullest positions. Another application would be in hydraulic motors, which only run in forward or reverse directions.

Three-position directional control valves. A three-position valve is similar in operation to a two-position valve except that it can be stopped in a third or centered position. While in the centered

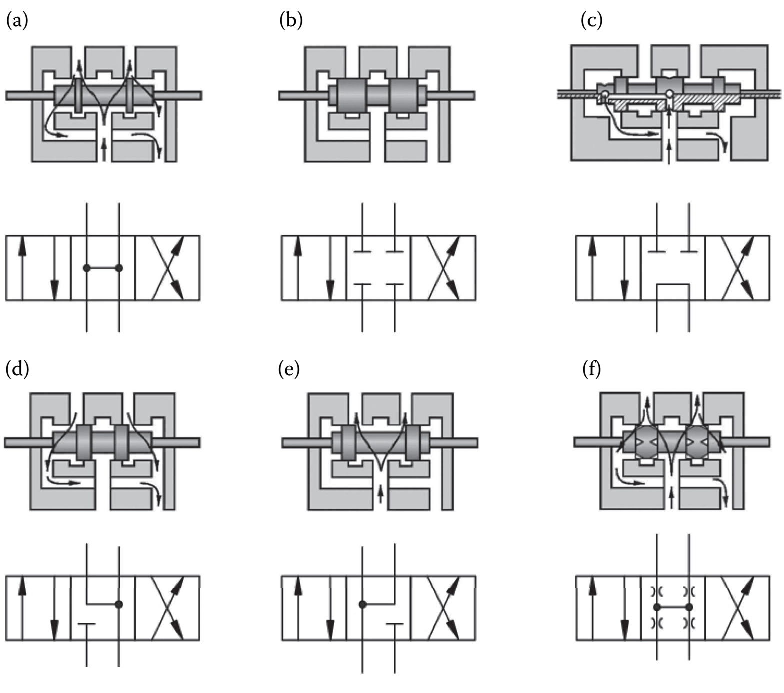


FIGURE 1.39 Typical center flow paths for four-way, three-position valves: (a) Open center; (b) Closed center; (c) Tandem center; (d) Pressure closed center; (e) Reservoir closed center; (f) Restricted open center. (From Linsingen, I. von, *Fundamentos de Sistemas Hidráulicos*, 3rd ed., Florianópolis, Brazil: UFSC Ed., 2008. With permission.)

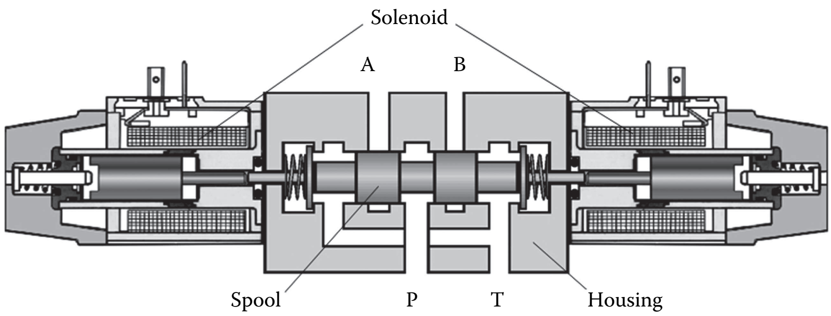


FIGURE 1.40 4/3 directional control valve, directly controlled by two solenoids with spring-centered central position. (From Linsingen, I. von, *Fundamentos de Sistemas Hidráulicos*, 3rd ed., Florianópolis, Brazil: UFSC Ed., 2008. With permission.)

or neutral position, flow may or may not be possible, depending on the spool design of the center position. Figure 1.39 shows some common three-position spool designs.

The open center valve (Figure 1.39a) and the tandem center valve (Figure 1.39c) divert the pump flow to the reservoir keeping the supply pressure low. In the closed center valve (Figure 1.39b), all ports are blocked in the centered position, preventing the actuator movement. At the same time, the pump flow can be used for other parts of the circuit. The restricted open center valve shown in Figure 1.39f avoids both the complete actuator relaxation and peak pressures during the valve commutation.

The pressure closed center design (Figure 1.39d) allows low pressure at ports *A* and *B* to be maintained while the reservoir closed center design (Figure 1.39e) means that the supply pressure is applied to both working ports (Figure 1.33a). This has a regenerative effect when an asymmetrical cylinder is used, causing the cylinder to extend rapidly due to the difference in the effective areas at opposite sides of the piston. The cylinder extension velocity is determined by the sum of the pump flow rate (q_V^P) and the flow rate at the rod end of the cylinder (q_{VB}^A), that is, $q_{VA}^A = q_V^P + q_{VB}^A$. When the cylinder chambers are interconnected, the pressure has the tendency to be the same but, as the areas are different, the hydraulic force (Equation 1.33) differs from zero, causing movement.

Control mechanisms, flow and pressure in directional valves. Besides the mechanical control mechanisms, as exemplified in Figure 1.38, hydraulically-controlled and solenoid-controlled valves are common. An example of a solenoid-controlled directional control valve is shown in Figure 1.40.

A typical characteristic curve of directional control valves is the graph of the pressure drop (Δp) versus the flow rate (q_V) through each flow path, as shown in Figure 1.41. This steady-state behavior is described by Equation 1.13 presented above, and shows that the load loss can be different for each valve position (*P–A*, *P–B*, *A–T*, *B–T*).

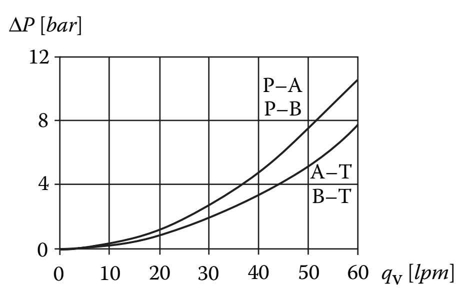


FIGURE 1.41 Characteristic curve of the steady-state behavior of directional control valves. (From Linsingen, I. von, *Fundamentos de Sistemas Hidráulicos*, 3rd ed., Florianópolis, Brazil: UFSC Ed., 2008. With permission.)

1.4.4 Pressure Control Valves

One of the most important characteristics of hydraulic systems is the possibility for pressure control. Besides providing security against overloading, the hydraulic system has the capability of limiting and/or controlling the force and torque of the actuator, thereby avoiding mechanical damage.

Basically, there are two groups of pressure control valves: the *normally closed* (NC) valves and the *normally open* (NO) valves. In the first group the pressure at the inlet port is controlled and in the second the outlet pressure is controlled. In both cases, the valve begins to control the pressure when the pressure set in the control mechanism is reached.

1.4.4.1 Normally Closed Pressure Control Valves

This group includes valves that have the same operational principle but with a few construction differences and which thus can perform different functions in the hydraulic circuit. These are the pressure relief valve, counterbalance valve, unloading valve and sequence valve [20].

The pressure relief valve is usually installed in parallel with the hydrostatic pump and remains closed until the system pressure surpasses the set pressure, when pump flow is partially or completely diverted to the reservoir. Figure 1.10 shows this situation where the fixed-displacement pump (P) runs at a constant rotational frequency, driven by the electrical motor (M), supplying a basically constant flow rate to the circuit. As discussed in Section 1.3, for the effective velocity control of the cylinder (A) the cracking pressure of the pressure relief valve (V1) must be reached and, in this way, the flow rate to the cylinder is reduced.

A typical design of a pressure relief valve is shown in Figure 1.42, which is composed of a poppet held in the valve seat by a spring force. In operation, the flow enters from the bottom of the valve (port *A*). When the inlet pressure (p_A) reaches the value such that the pressure times the exposed area of the poppet is greater than the spring setting ($F_{k0} = Kx_0$), the valve will begin to pass hydraulic fluid. Note that the spring must be compressed in order for the poppet to move and provide a greater flow area.

Characteristic curves. The steady-state characteristic curve of a pressure relief valve is given in Figure 1.42b, which shows that the inlet pressure increases as the flow rate through the valve increases. The pressure at which the valve first begins to open is called the "cracking pressure" and

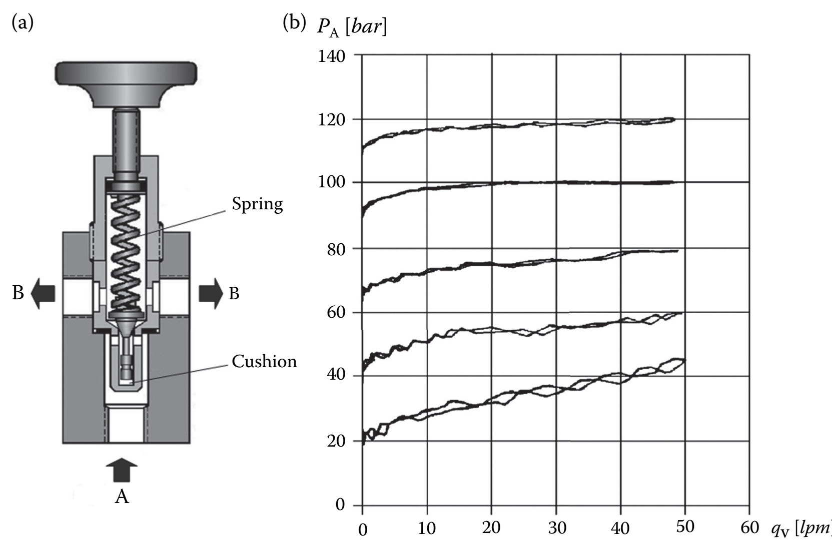


FIGURE 1.42 Directly-operated pressure relief valve: (a) Illustration; (b) Steady-state characteristic curve. (From Linsingen, I. von, *Fundamentos de Sistemas Hidráulicos*, 3rd ed., Florianópolis, Brazil: UFSC Ed., 2008. With permission.)

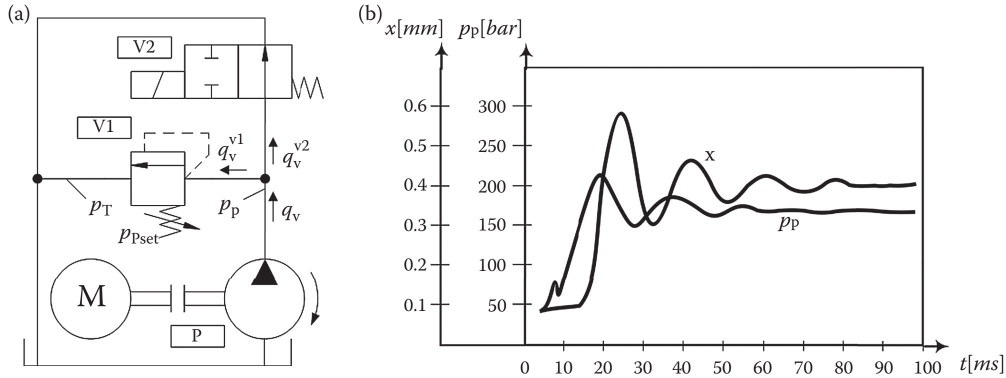


FIGURE 1.43 Dynamic behavior of a directly-operated pressure relief valve: (a) Test circuit; (b) Dynamic response.

it corresponds to the set pressure through the control mechanism (screw) ($p_A = P_{Aset}$). The override pressure is essentially a result of the spring force and flow force in the valve.

The dynamic behavior of a pressure relief valve has a strong influence on the system pressure behavior, as shown in Figure 1.43. Observing the circuit in Figure 1.43a, when the directional control valve (V2) is closed rapidly, the displacement of the valve element (poppet) and the system pressure oscillate as shown in Figure 1.43b. The cushion in the valve shown in Figure 1.42a must be designed to reduce the pressure spikes while at same time reaching the steady state as quickly as possible.

Since the pressure in a hydraulic system is described by the mass conservation principle (Equation 1.4) the pressure behavior is dependent on the circuit fluid volume and the fluid compressibility (bulk modulus) and not only on the valve behavior.

Pilot-operated valve. The pilot-operated pressure relief valve, as shown in Figure 1.44, increases pressure sensitivity and reduces the pressure override normally found in relief valves using only the direct-acting force of the system pressure against a spring element. In operation, the fluid pressure acts on both sides of the piston because of the small orifice through the piston, and the piston is held in the closed position by the light-bias piston spring. When the pressure increases sufficiently

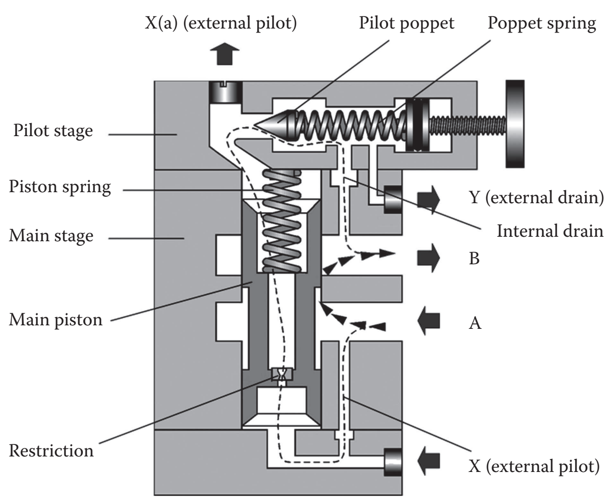


FIGURE 1.44 Pilot-operated pressure relief valve. (From Linsingen, I. von, *Fundamentos de Sistemas Hidráulicos*, 3rd ed., Florianópolis, Brazil: UFSC Ed., 2008. With permission.)

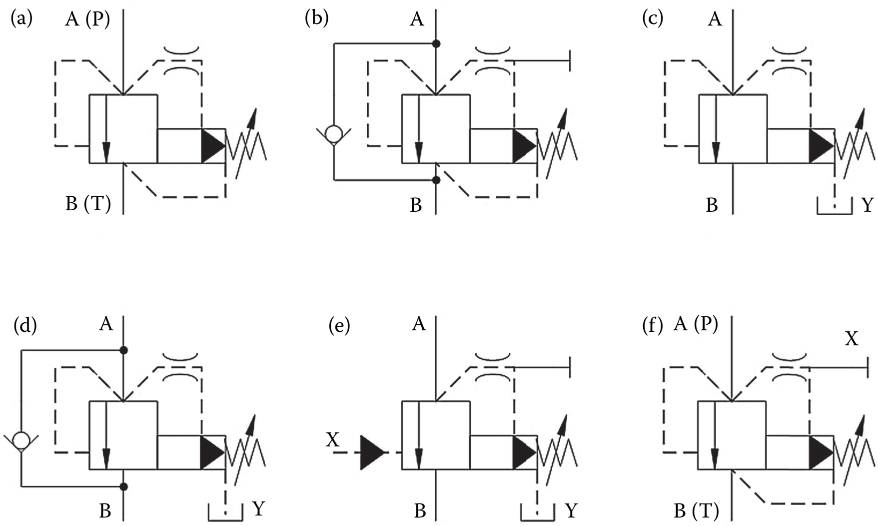


FIGURE 1.45 Pressure control valves according to ISO 5781. (a) Pressure relief valve; (b) Counterbalance valve; (c) Sequence valve; (d) Sequence valve with bypass non-return valve; (e) Unloading valve; (f) Remote-controlled pressure relief valve. (From ISO, ISO 5781 - *Hydraulic fluid power – Pressure-reducing valves, sequence valves, unloading valves, throttle valves and check valves – Mounting surfaces*, Switzerland, 2000. 20p. With permission.)

to move the pilot poppet from its seat, the fluid behind the piston will be directed to a low-pressure area, such as the return line. The resulting pressure imbalance in the piston will cause it to move in the direction of the lower-pressure area, compressing the piston spring and opening the discharge port. This action will effectively prevent any additional increase in pressure. The setting of the pilot-operated pressure relief valve is adjusted by the preload of the poppet spring.

The valve design shown in Figure 1.44 allows different operational configurations. In the configuration presented, the valve can be used as a pressure relief valve and when a non-return valve is incorporated it becomes a counterbalance valve. Closing the internal drain and using the external drain (Y) results in a sequence valve, with the incorporation of a bypass non-return valve being optional. When the internal pilot line is closed and an external pilot signal (X) is used, the valve is utilized as an unloading valve. It is also possible to open the valve at low pressure or promote a remote control using the another external pilot port (X(a) in Figure 1.44). The symbolic representation of these valves is shown in Figure 1.45.

1.4.4.2 Normally Open Pressure Control Valves (Pressure-Reducing Valves)

Pressure-reducing valves (directly- or pilot-operated) are used to supply fluid to branch circuits at a pressure lower than that of the main system. Their main purpose is to bring the pressure down to the requirements of the branch circuit by restricting the flow when the branch reaches some preset limit. One example of pressure-reducing valve is illustrated in Figure 1.46. In operation, a pressure-reducing valve permits fluid to pass freely from port A to port B until the pressure at port B becomes high enough to overcome the force of the spring. At this point, the spool will move, obstructing the flow to port B and thus regulating the downstream pressure. The direction of flow is irrelevant with a pressure-reducing valve, as the spool will close when the pressure at port B reaches the set value. If free reverse flow is required, a non-return valve must be used.

The reduced pressure (p_B) must be kept constant even though there is no flow downstream. Since the valve operational principle is based on the pressure drop control, an internal leakage (port Y) is required so that there is a continuous flow through the control orifice.

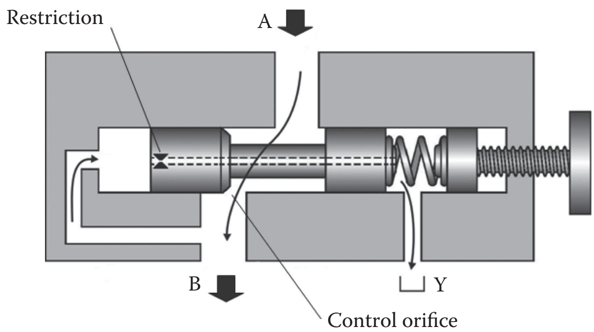


FIGURE 1.46 Directly-operated pressure-reducing valve. (From Linsingen, I. von, *Fundamentos de Sistemas Hidráulicos*, 3rd ed., Florianópolis, Brazil: UFSC Ed., 2008. With permission.)

1.4.5 Flow Control Valves

Flow rate control in a hydraulic system is commonly used to control the rod velocity of linear actuators or the shaft rotational frequency of hydraulic motors. There are three ways to carry out flow rate control. One is to vary the speed of a fixed-displacement pump; another is to regulate the volumetric displacement of a variable-displacement pump. The third way is with the use of flow control valves.

Flow control valves may vary from a simple orifice to restrict the flow to a complex pressure-compensated flow control valve or flow divider. In all designs the flow rate control is carried out according to Equation 1.13, which means that the hydraulic energy is dissipated through the valve.

Uncompensated flow control valves. The simplest uncompensated flow control is the fixed-area orifice. Normally, these orifices are used in conjunction with a non-return valve so that the fluid passes through the orifice in one direction, but in the reverse direction the fluid may pass through the non-return valve, thus bypassing the orifice. Another design incorporates a variable-area orifice so that the effective area of the orifice can be increased or decreased (usually manually). One example of a variable-area orifice with a reverse-flow non-return valve is shown in Figure 1.47. These uncompensated flow control valves are used where exact flow control is not critical.

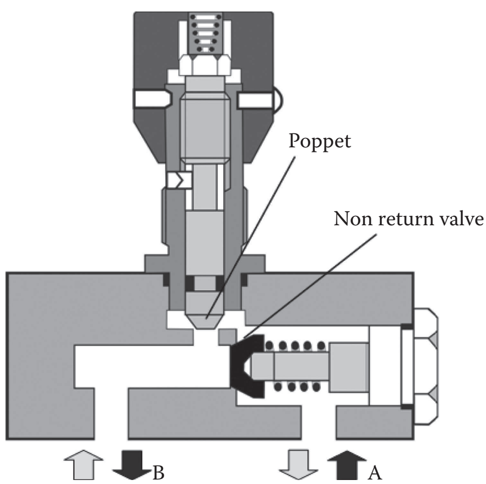


FIGURE 1.47 Uncompensated flow control valve.

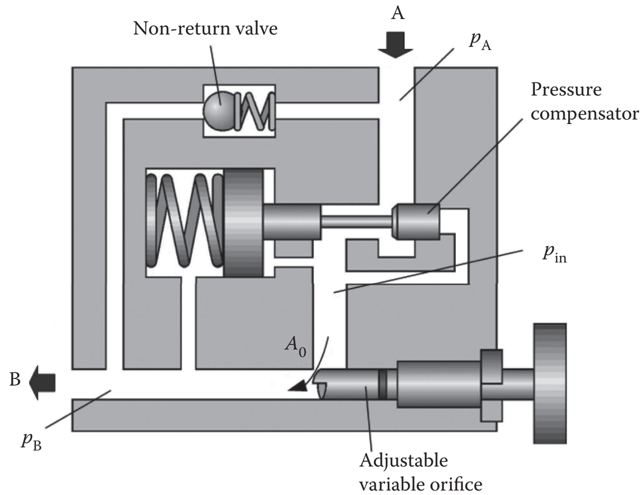


FIGURE 1.48 Example of pressure-compensated flow control valve. (From Linsingen, I. von, *Fundamentos de Sistemas Hidráulicos*, 3rd ed., Florianópolis, Brazil: UFSC Ed., 2008. With permission.)

Recalling Equation 1.13, the flow rate through an orifice is dependent on the pressure drop across the orifice. Therefore, if the pressure differential increases or decreases, the flow will also increase or decrease. To avoid this, a compensated flow control valve must be used.

Pressure-compensated flow control valves. A pressure-compensated flow control valve is shown in Figure 1.48. In this valve, as the pressure differential across the valve from the inlet to the outlet increases, the flow would also increase. However, any increase in flow will be accompanied by a resulting increase in the pressure drop across the control orifice (A_0) ($\Delta p = p_{in} - p_B$). When this pressure differential begins to produce a force larger than the spring preload, the valve spool will shift and the secondary orifice (A_1) will be restricted. These valves normally incorporate a non-return valve for a free inverse flow.

Flow dividers. Flow dividers are also a form of flow control valve. There are at least two types of flow dividers: One is called a "priority flow divider"; the other is a "proportional flow divider." The priority type of flow rate control provides flow to a critical circuit at the expense of other circuits in the system. Figure 1.49 [11] illustrates a priority flow divider. In operation, the flow will enter the priority flow divider from port *B*. When the flow reaches a value and the

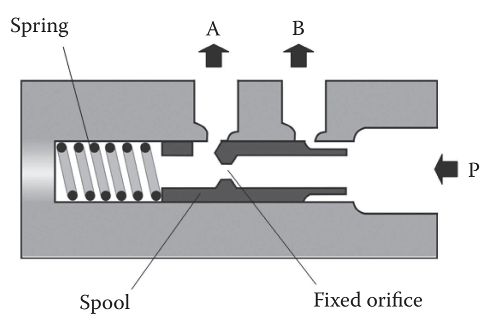


FIGURE 1.49 Priority flow divider. (From Sullivan, J.A. *Fluid Power: Theory and Application*, 2nd ed., USA: Prentice–Hall International, 1982, ISBN 013907668-9. With permission.)

pressure drop across the fixed orifice produces a force larger than that provided by the spring, the spool will move to the left. This action will begin to close the priority outlet port (*A*) and open the secondary outlet (*B*). When the flow rate is below the designed priority flow rate, the spool will be all the way to the right, the secondary outlet will be closed, and the priority outlet will be wide open. The proportional-type flow divider follows the same principle as the priority flow divider, except that two orifices are used and the spool is normally spring-loaded to a particular flow split ratio.

1.4.6 Directional Continuous Control Valves

As established by ISO 5598 [19], continuous control valves are valves "that control the flow of energy of a system in a continuous way in response to a continuous input signal.'

Moreover, according to the function performed by the valve in the system, these valves can be classified as directional continuous control valves, pressure continuous control valves and flow continuous control valves.

Observing the directional control valves described in Section 1.4.3, it can be seen that there is an intrinsic possibility for continuous movement of the valve element (typically the spool). However, several of the control mechanisms used for directional control valves, like a solenoid, detent lever, hydraulic pilot, and so forth, only allow the valve to move to specific positions.

With directional continuous control valves, continuous position changing is possible; for example, from the *P-A*/*B-T* position to the blocked port center position and then to the *P-B*/*A-T* position.

Directional continuous control valves with mechanical control are well known in mobile hydraulics where the position of the command lever is defined by a human operator based on his or her own observation of the position or velocity of the cylinder or motor.

Valve technology with continuous electrical input started with the servo-valves in the early 1940s [21]. Another notable event was the development of the proportional directional control valves in the late 1970s [22]. Encompassing technological principles from both these valve types, new products are being offered on the market, such as servo-proportional valves [23]. Regardless of their commercial identification or construction principle, according to ISO 10770-1 [17] and ISO 10770-2 [18] these are electrically-modulated hydraulic flow control valves, since they provide a degree of proportional flow control in response to a continuously variable electrical input signal.

1.4.6.1 Servo-valves

Since their beginning in the 1940s, different conceptions have been developed and the two-stage valve is a representative servo-valve concept. The first stage (pilot stage) is composed of either a jet pipe valve or flapper-nozzle valve driven by a torque motor (a permanent magnet, variable reluctance actuator). The second stage is a spool valve, its position being fed back in order to place the torque motor armature at the null position.

Figure 1.50 shows a typical servo-valve with mechanical feedback or force feedback. Other methods of position feedback are the spring-centered spool, direct position feedback or hydraulic follower, and electric feedback using a position transducer [24].

Frequently, the spool slides into a sleeve where the ports were machined. The relative position between the spool lands and sleeve ports then determines the flow control orifices. The same solution is adopted for directly operated valves with electrical feedback, driven by a linear force motor. This valve design is referred to as the "servo-proportional valve" [23,25].

Advances in the manufacturing process and changes in the user requirements have led to changes in the construction details. For example, pilot-operated servo-valves like that shown in Figure 1.50 but without a sleeve are also available.

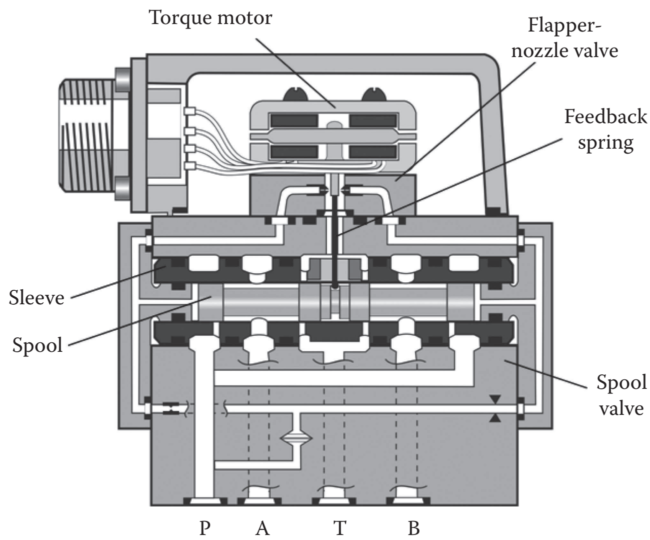


FIGURE 1.50 Pilot-operated servo-valve with mechanical feedback.

1.4.6.2 Proportional Directional Control Valves

The conception of proportional directional control valves comes from two distinct fields: mobile hydraulics and industrial hydraulics. In both cases, the objective was to obtain the same functional characteristics as servo-valves—that is, the continuous control of flow direction and rate, but with a distinct mechanical design.

The proportional valves are controlled by proportional solenoids, which, unlike the torque motor and linear force motor, do not comprise a permanent magnet and the force is provided in only one direction for any current polarity.

Figure 1.51 shows a proportional directional valve, directly controlled by two solenoids, with a spring-centered central position and a spool position transducer. The operation of this type of valve requires an electronic controller/amplifier that receives both the external reference signal and the feedback signal from the position transducer, processes them and sends electrical signals to the solenoids.

There is a significant diversity of proportional directional valves on the market, including valves without feedback position, valves with only one solenoid acting against a spring and valves with controller/amplifier assembled together in the valve (on-board electronics). The metering notches on the spool, as shown in Figure 1.51, can be of different types and are used to define the curve of the flow rate against the spool displacement. However, they are not machined on all valve designs.

Valve designs with spool-sleeve mounting are also available with both smaller machining tolerances and radial clearances. Usually these valves include position feedback optimizing their static

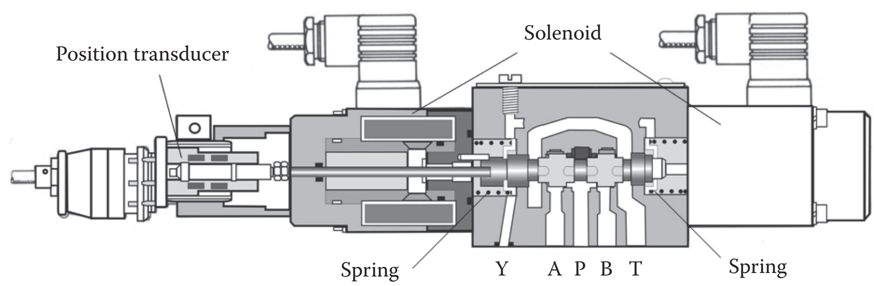


FIGURE 1.51 Proportional directional control valve.

and dynamic behavior. The servo-proportional valve designation has also been used by valve manufactures for these construction solutions [23,26,27].

1.4.6.3 Fundamental Model and Characteristic Curves

Considering a directional continuous control valve as being the valve itself with the controller/amplifier, on-board or not, its main function is to control the flow rate (output) in response to a input voltage (reference signal).

The valve behavior can be described through the composition of two parts—with feedback or without feedback. The first block corresponds to the transformation of the input voltage into spool displacement. The second one refers to the output flow rate as a consequence of the spool displacement and the pressures in the supply (P), return (T), and working (*A* and *B*) ports of the valve (Figure 1.52).

In essence, the valve amplifier controls the current applied to each proportional solenoid or to the pair of coils of a torque motor or linear motor. According to electromechanical principles, this current produces a force (or torque) that is transmitted to a valve element.

In the case of a pilot-operated servo-valve, as shown in Figure 1.50, the torque produces the pipe motion (on jet-pipe valves) or the flapper motion (on flapper-nozzle valves) which, in turn, changes the pressure on the spool sides. The pressure difference makes the resting spool change its position, which is fed back to the pilot valve. In directly-operated valves, as shown in Figure 1.51, the force produced by the electromagnetic actuator is applied directly on the spool.

Based on these principles, a dynamic relationship between the control voltage (U_c) and the spool displacement (x_s) can be expressed by

$$K_{RP} \cdot U_c = \frac{1}{\omega_n^2} \cdot \frac{d^2 x_s}{dt^2} + \frac{2 \cdot \zeta}{\omega_n} \cdot \frac{dx_s}{dt} + x_s, \tag{1.47}$$

where K_{RP} [m/V] is the steady-state gain (ratio between the spool displacement and control voltage in a steady state), ω_n [rad/s] is the natural frequency and ζ [1 (non-dimensional)] is the damping ratio.

The parameter values of Equation 1.47 can be obtained from valve data sheets; for example, from the response time curves shown in Figure 1.53 [28]. Comparing these curves with the general response time of a second-order system (Figure 1.35b), it can be concluded that this valve has a damping ratio (ζ) close to 0.8 and a settling time (t_s) of approximately 50 ms for an input of 50% of the maximum amplitude. Using Equation 1.43, the natural frequency is determined as 53.6 rad/s (8.5 Hz).

The valve catalogs also inform the response time defined according to ISO 10770-1 [17] and shown in Figure 1.35b. The approximate calculation of the natural frequency based on the response time is carried out using Equation 1.44, where $\zeta = 0.7$ can be used when the value is not given in the catalog.

Another way to present the valve dynamic response is through a frequency response diagram (Bode diagram), where it is possible to extract directly the values of the natural frequency and damping ratio [16].

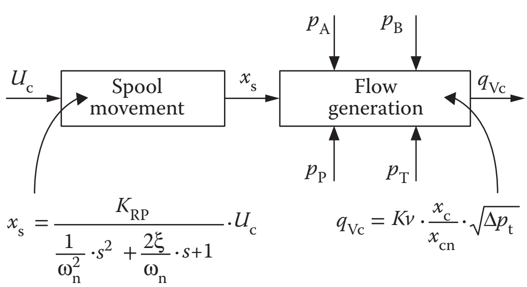


FIGURE 1.52 Block diagram of the directional continuous control valve.

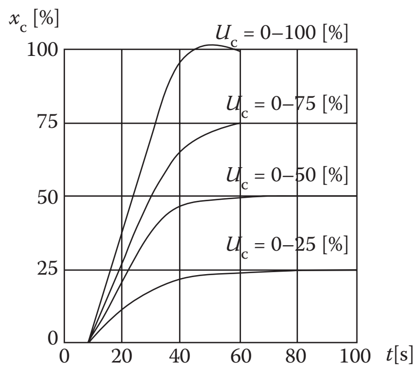


FIGURE 1.53 Response time of a directional proportional valve.

The second block in Figure 1.52 refers to the flow rate control as a function of the orifice opening and the pressures at the valve ports. By applying the concepts related to Equation 1.13, the following expression is valid for directional continuous control valves [14,29,30]:

$$q_{Vc} = Kv \cdot \frac{x_c}{x_{cn}} \cdot \sqrt{\Delta p_t}, \tag{1.48}$$

where q_{Vc} [m³/s] is the control flow rate, Kv [(m³/s)/(Pa)$^{1/2}$] is the flow coefficient, x_{cn} [m] is the nominal spool displacement and Δp_t [Pa] is the total pressure drop at the valve.

By combining Equations 1.47 and 1.48, one obtains the general expression for a directional continuous control valve—that is

$$q_{Vc} = Kv \cdot \left(\frac{1}{\frac{1}{\omega_n^2} \cdot D^2 + \frac{2\xi}{\omega_n} \cdot D + 1} \right) \cdot \frac{U}{U_n} \cdot \sqrt{\Delta p_t}, \tag{1.49}$$

where $D = d/t$ is the differential operator.

When the valve is under a steady-state condition, this equation takes the following form:

$$q_{Vc} = Kv \cdot \frac{U}{U_n} \cdot \sqrt{\Delta p_t}. \tag{1.50}$$

The total pressure drop at the valve (Δp_t) corresponds to the pressure drop between the supply port (P) and the return port (T), which, for the flow paths P–A/B–T, is expressed by

$$\Delta p_t = \Delta p_{P-A} + \Delta p_{B-T} = (p_P - p_A) + (p_B - p_T), \tag{1.51}$$

where $\Delta p_{P-A} = p_P - p_A$ is the pressure drop between ports P and A and $\Delta p_{B-T} = p_B - p_T$ is the pressure drop between ports B and T.

For the flow paths P–B/A–T, the total pressure drop is:

$$\Delta p_t = \Delta p_{P-B} + \Delta p_{A-T} = (p_S - p_B) + (p_A - p_T), \tag{1.52}$$

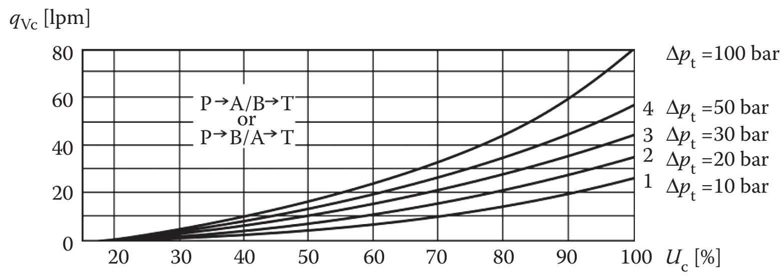


FIGURE 1.54 Flow rate versus input voltage of a directional proportional valve.

where $\Delta p_{P-B} = p_S - p_B$ is the pressure drop between ports *P* and *B* and $Dp_{A-T} = p_A - p_T$ is the pressure drop between ports *A* and *T*.

The valve catalogs inform the nominal flow rate (q_{Vcn}) at a determined pressure drop that can be either 1 MPa (10 bar), 7 MPa (70 bar), or 1/3 of the nominal supply pressure [17,18]. The nominal flow occurs when the valve is operating with nominal voltage, that is, with the nominal opening. The flow coefficient (*Kv* [(m^3/s)/(Pa)$^{1/2}$] or [(lpm/(bar)$^{1/2}$]) can be calculated as:

$$Kv = \frac{q_{Vcn}}{\sqrt{\Delta p_{tn}}}. \tag{1.53}$$

The data for the *Kv* calculation can also be obtained from curves, as shown in Figure 1.54 [28], at 100% of the input signal. In this case, the nominal flow rate presented on the data sheet is 25 lpm@10 bar (41×10^{-3} m^3/s@1 MPa) (which corresponds to curve 1).

It is important to observe that for some valves the nominal flow is specified at a partial pressure drop (Dp_{P-A}) and this must be multiplied by two to allow the flow coefficient calculation.

Constructive aspects of the directional control valves, like different center position arrangements (Figure 1.39) and the existence of symmetrical and asymmetrical designs, are also applicable to directional continuous control valves.

1.4.7 Hydraulic Accumulators

The purpose of a hydraulic accumulator is to store fluid or provide fluid at a certain pressure in order to minimize short-duration pressure spikes or to reach a short-duration high-flow demand. The accumulators used in hydraulic systems can be grouped into three categories: weight-loaded or gravity type, spring-loaded type, and gas-loaded type [31] (Figure 1.55). The weight-loaded type consists of a cylinder with a piston where a mass is attached to its top. The gravitational action on the mass creates a constant fluid pressure, irrespective of the flow rate and fluid volume in the cylinder chamber.

The spring-loaded accumulator simply uses the spring force to load the piston. When the fluid pressure increases to a point above the preload force of the spring, fluid will enter the accumulator to be stored until the pressure reduces. In this type of accumulator, the fluid pressure varies with the piston position and, consequently, with the fluid volume in the accumulator.

The gas-loaded accumulator can be either without separation between liquid and gas, a piston type or a bladder and diaphragm type, as shown in Figure 1.55. In the gas-loaded accumulator, an inert gas, such as dry nitrogen, is used as a pre-charge medium. In operation, this type of accumulator contains the relatively incompressible hydraulic fluid and the more readily compressible gas. When the hydraulic pressure exceeds the pre-charge pressure exerted by the gas, the gas will compress, allowing hydraulic fluid to enter the accumulator. The hydraulic pressure changes with the volume occupied by fluid as a consequence of the pressure gas variation caused by its compression/decompression.

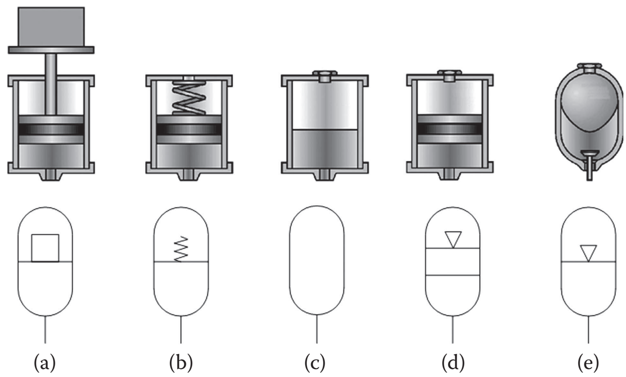


FIGURE 1.55 Basic types of accumulators. (From Linsingen, I. von, *Fundamentos de Sistemas Hidráulicos*, 3rd ed., Florianópolis, Brazil: UFSC Ed., 2008. With permission.)

1.4.8 Reservoir and Its Accessories

A typical design for an industrial reservoir is shown in Figure 1.56 where the main parts can be identified. The reservoir should be sized to both afford adequate fluid cooling and to enclose a sufficient volume of oil to permit air bubbles and foam to escape during the residence time of the fluid in the reservoir. Commonly, the reservoir is sized to hold at least three times the volume of fluid that can be supplied by the pump in one minute. Baffles are also provided to prevent channeling of the fluid from the return line to the inlet line and the bottom of the return line is usually cut at a 45° angle to assist in the redirection of the fluid away from the inlet.

The reservoir depth must be adequate in order to assure that during peak pump demands, the oil level will not drop below the pump inlet level. Moreover, the pump should be mounted below the reservoir so that a positive head pressure is available at all times. This is critical when water-based hydraulic fluids are used, as these fluids can have a higher mass density as well as a much higher vapor pressure than mineral-oil-based fluids.

Sight gauges are normally used to monitor the fluid level and a cleanout plate is provided to promote cleaning and inspection. A breather system with a filter is also provided to admit clean air and to maintain atmospheric pressure as fluid is pumped into and out of the reservoir. With water-based hydraulic fluids, a pressurized reservoir is recommended. Special breather caps can be installed to vent between 0.005 MPa (0.05 bar) and 0.1 MPa (1 bar). If one of these is used, it

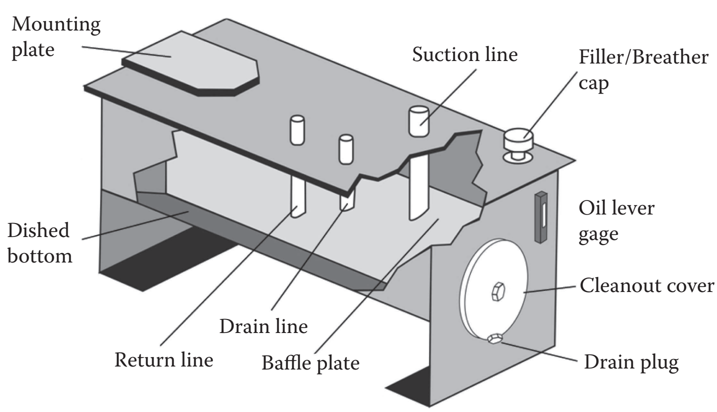


FIGURE 1.56 A typical design for an industrial reservoir. (From Norvelle, F.D. *Fluid Power Technology*, New York, NY, West Publishing Company, 1995. With permission.)

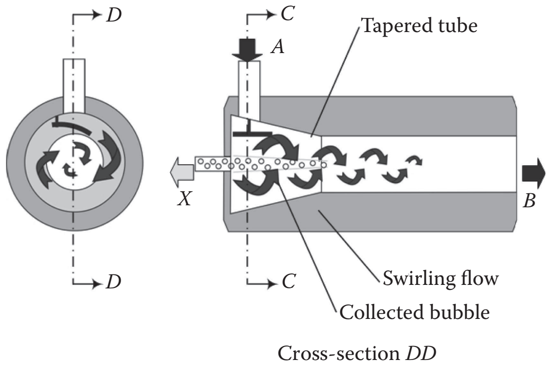


FIGURE 1.57 Bubble eliminator.

must have a vacuum brake to vent at approximately –0.003 MPa (–0.03 bar). This is an important feature to have so that when the reservoir is cooling down, no appreciable vacuum develops in the reservoir. This feature will minimize pump cavitation upon start-up and also prevent a possible reservoir implosion.

Recent trends in industrial manufacturing are to compact machines and equipment in order to economize materials, energy consumption, and required space. A reduction in the size of fluid power systems is encouraged in order to conserve energy and reserve oil. It is somewhat inevitable in designing these systems to minimize the size of the oil reservoir, meaning that the bubbles entrained in the oil may not be removed effectively during the fluid sojourn time in the reservoir. As mentioned above, in order to remove bubbles big vessels are generally used, but it takes a long time to eliminate minute bubbles from fluids by flotation alone.

Another solution is the device shown in Figure 1.57, which has the capacity to eliminate bubbles and decrease dissolved gases using a swirl flow [33,34]. This device, called a "bubble eliminator," consists of a tapered tube where the fluid containing bubbles flows tangentially from the inlet port (port *A*) and generates a swirling flow. Due to the difference in centrifugal forces created in the swirl flow, the bubbles tend to move toward the central axis (port *B*) where they are collected and ejected through the vent port (port *X*).

1.4.9 Filters

As discussed throughout this chapter, hydraulic components are composed of mechanical elements with relative movement and small clearances between them. The hydraulic fluid is expected to create a lubricating film, thereby keeping precision parts separated. Particulate contaminants can break this film, cause erosion on the surfaces or even block the relative movement. Consequently, the hydraulic component life expectancy is reduced, impairing its performance or even causing its complete failure.

The contaminants in hydraulic systems come from several sources, such as the degradation of the circuit components, the external environment, the circuit assembly, and from the new hydraulic fluid which can have a standard contamination level below the system requirements.

The removal of particulate matter and silt from a hydraulic fluid is performed by filters that can be installed at different locations in the hydraulic circuit, characterizing the following types of filtration: suction, pressure, return and off-line filtration [35,36].

Suction line filtration: Suction filters are located before the suction port of the pump and provide pump protection against fluid contamination (Figure 1.58a). Some may be inlet strainers, submersed in the fluid. Others may be externally mounted. In either case, they utilize relatively coarse elements

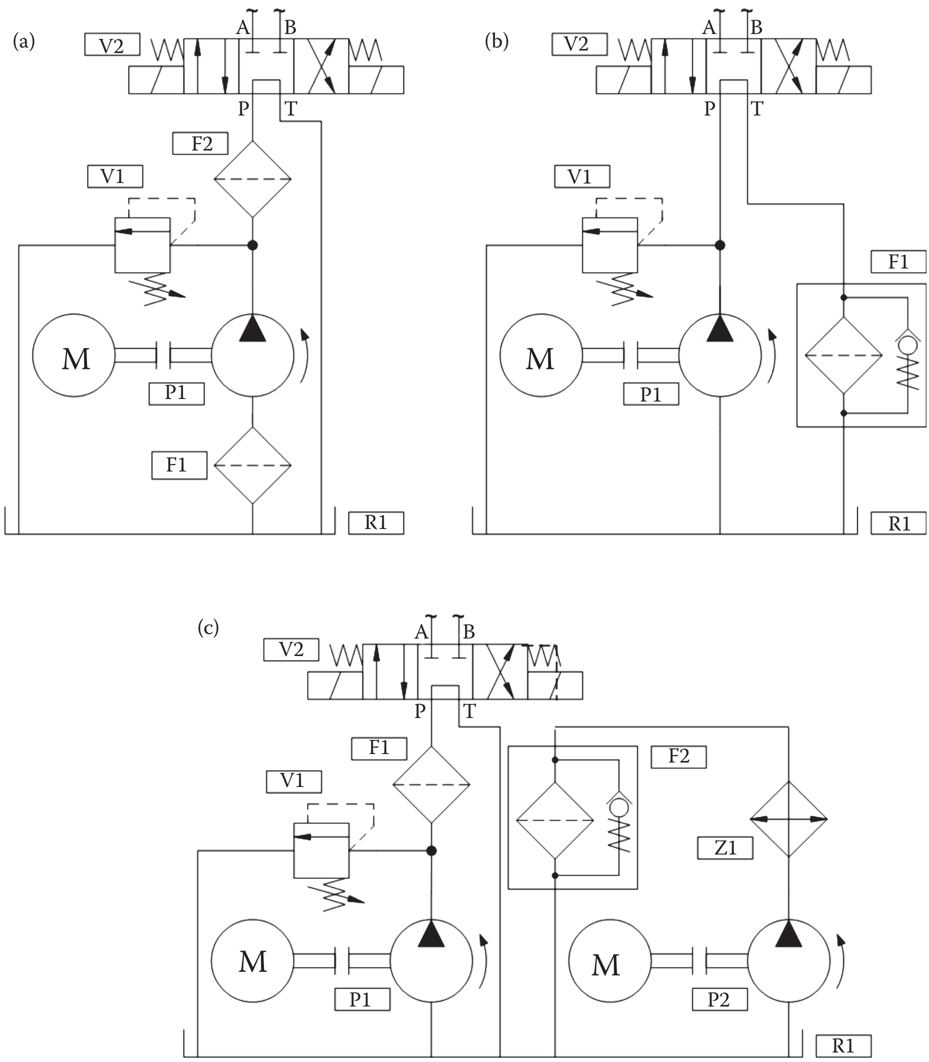


FIGURE 1.58 Types of filtration: (a) Suction filter (F1) and Pressure filter (F2); (b) Return filter (F1); (c) Pressure.

to avoid high pressure drops that can cause cavitation on the pump. Some pump manufacturers do not recommend the use of a suction filter.

Pressure line filtration: Pressure filters are located downstream of the pump (Figure 1.58a and c). They usually produce the lowest system contamination levels to assure clean fluid for sensitive high-pressure components and provide protection of downstream components from pump-generated contamination.

Return line filtration: In most systems, the return filter is the last component through which fluid passes before entering the reservoir (Figure 1.58b). Therefore, it captures wear debris from system working components and particles entering through worn cylinder rod seals before such contaminants can enter the reservoir. A special concern in applying return filters is sizing for a potential flow rate greater than the pump output, since large rod cylinders and other components can cause induced return line flows. Return lines can have substantial pressure surges, which need to be taken into consideration when selecting filters and their locations. The relatively low cost and the cleanliness of the fluid suctioned by the pump are factors that make the use of these filters attractive.

Re-circulating or off-line filtration: Off-line filtration consists of a hydraulic circuit with at least a pump and its prime mover and a filter. These components are installed off-line as a small subsystem separate from the working lines or can be included in a fluid-cooling loop (Figure 1.58c). As with a return line filter, this type of system is best suited to the maintenance of overall cleanliness, but does not provide specific component protection. An off-line filtration loop has the added advantage of being relatively easy to retrofit on an existing system that has inadequate filtration. Also, the filter can be serviced without shutting down the main system.

The circuits shown in Figure 1.10 through 1.12 also present some examples of filter installations. In general, the systems can incorporate multiple filtration techniques, using a combination of suction, pressure, return, and off-line filters.

1.4.10 Hydraulic Fluid

The main characteristic of hydraulic systems, as well as of pneumatic systems, is their requirement that matter flow in such a way as to promote the flow of energy. As discussed in Section 1.1, the hydraulic system must perform three fundamental functions in terms of the energy: primary conversion, limitation and control, and secondary conversion. A fourth function is related to fluid storage and conditioning. This function is required because the fluid must be available for the energy transmission, and since the fluid is continuously in contact with the hydraulic components its proprieties must be controlled.

Fluid proprieties such as viscosity, mass density, vapor pressure, contamination, gas solubility, and bulk modulus change the physical relations modeled by the continuity equation, and conservation of energy, among others. Therefore, besides causing component degradation, the modifying of physical proprieties also changes the hydraulic system behavior.

Throughout the chapters of this *Handbook* the proprieties of different fluids that are used in hydraulic systems are analyzed as well as their effect on the life and behavior of the components.

ACKNOWLEDGMENTS

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Appendix

Equipment Builder's Viscosity Guidelines for Hydraulic

Fluids

NFPA T2.13.13 2002 n Equipment Operating Startup (Under Load) Optimum Minimum mm 2 /s (cSt) Maximum mm 2 /s (cSt) Maximum mm 2 /s (cSt) mm 2 /s (cSt)

Bosch (see Rexroth

Corporation)

Commercial

Intertech (see Parker

Hanni�n)

Danfoss (see

Sauer-Danfoss, USA)

Denison Hydraulics

SPO-AM305 Piston Pumps Vane Pumps 10 10 162 107 1618 860 (low speed and pressure) 30 30

Dynex/Rivett

axial piston pumps PF4200 Series PF2006/8, PF/PV4000, and PF/PV6000 Series PF 1000, PF2000 and PF3000 Series 1.5 2.3 3.5 372 413 342 372 413 342 20-70 20-70 20-70

Eaton Heavy-Duty Piston Pumps and Motors, MediumDuty Piston Pumps and Motors Char ged Systems, Light-Duty Pumps Medium-Duty Piston Pumps and Motors – Non-charged Systems Gear Pumps, Motor, and Cylinders 6 6 6 – – – 2158 432 2158 10-39 10-39 10-43

Eaton - Vickers Mobile Piston Pumps Industrial Piston Pumps Mobile Vane Pumps Industrial Vane Pumps 10 13 9 13 200 54 54 54 860 220 860 860 16-40 16-40 16-40 16-40

Eaton - Char-Lynn J, R, and S Series Motors, and Disc Valve Motors A Series and H Series Motors 13 20 – – 2158 2158 20-43 20-43

Haldex Barnes W Series Gear Pumps 11 – 750 21

Kawasaki

P-969-0026

P-969-0190 Staffa Radial Piston Motors K3V/G Axial Piston Pumps 25 10 150 200 2000 (no load) 1000 50

Linde All 10 80 1000 15-30

Mannesmann Rexroth

(see Rexroth

Corporation) (continued)

NFPA T2.13.13 2002 (Continued) n Equipment Operating Startup (Under Load) Optimum Minimum mm 2 /s (cSt) Maximum mm 2 /s (cSt) Maximum mm 2 /s (cSt) mm 2 /s (cSt)

Parker Hanni�n Roller and Sleeve-Bearing Gear Pumps Gerotor Motors Gear Pumps PGH Series Gear Pumps D/H/M Series Hydraulic Steering PFVH / PFVI Vane Pumps Series T1 VCR2 Series Low-Speed High-Torque Motors Variable Vol Piston Pumps PVP and PVAC Axial-Fixed Piston Pumps Variable Vol Vane - PVV 10 8 – – 8 – 10 13 10 – – – – – – – – – – – – – – – – – – 1600 – 1000 1000 – 1000 1000 1000 – 1000 1000 850 440 20 12-60 17-180 17-180 12-60 17-180 10-400 – – 17-180 17-180 12-100 16-110

Poclain Hydraulics H and S Series Motors 9 – 1500 20-100

Rexroth Corporation

F orm No S/106 US FA, RA,; K Q, Q-6, SV-10, 15, 20, 25, VPV 16, 25, 32 SV-40, 80 and 100 VPV 45, 63, 80, 100, 130,164 Radial Piston (SECO) Axial and RKP Piston V3, V4, V5, V7 Pumps V2 Pumps R4 Radial Piston Pumps G2, G3, G4 Pumps and Motors G8, G9, G10 Pumps 15 21 32 10 14 25 16 10 10 216 216 216 65 450 – 160 200 300 864 864 864 162 647 800 800 – 1000 26-45 32-54 43-64 21-54 32-65 25-160 25-160 25-160 25-160

Rotary Power “SMA” Radial Piston Motor 15 – 1000 20-200

Sauer-Danfoss, USA Steering and Valves PVG Valves Gear Pumps and Motors Closed-Circuit Axial Piston Pumps and Motors Open-Circuit Axial Piston Pumps Bent Axis Motors LSHT Motors 10 4 10 7 6 7 10 – – – – – – – 1000 460 1600 1600 1000 1600 1000 12-60 12-75 20-40 12-60 9-110 12-60 20-75

Sauer-Danf oss,

GmbH Series 10 and 20, RMF(Hydrostatic Motor) Series 15 Open Circuit Series 40, 42, 51 and 90 CW S-8 Hydrostatic Motor Series 45 Series 60, LPM (Hydrostatic Motor) Gear Pumps plus Motors 7 12 7 9 9 10 - - - - - - 1000 860 1600 1000 1600 1000 12-60 12-60 12-60 12-60 12-60 12-60

Su ndstrand (see

Sauer-Danfoss, USA)

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FIGURE 7.3 Mechanical efficiency of hydraulic motors during

startability test.

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Classification Comparison of Fire-Resistant Hydraulic Fluid Types by FM Approval

Standard 6930 and ISO 15029-2

Fire-Resistant Hydraulic Fluid

Types	Typical FM Approvals Fire-Resistance Classification by CN 6930 (2009)	Typical RI Grade according to ISO 15029-2
HFA	FM Approved	A
HFB	FM Specification Tested *	E
HFC	FM Approved	B-C
HFDR	FM Approved or Specification Tested	D-E
HFDU	FM Approved or Specification Tested	G-H

* For typical invert emulsions containing 40% water. New formulations containing ≥ 45% water may be FM Approved.

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Biodegradability Requirements

Title Test Parameter Definition or Pass Criteria Ready Biodegradability

301A: DOC die-away % DOC removal ≥ 70% DOC removed within 28 days and within 10 days window after 10% CO_2 production is reached

301B: CO_2 evolution

(Modified Sturm Test) % CO_2 production ≥ 60% of theoretical CO_2 production within 28 days and within 10-days window after 10% CO_2 production is reached

301C: MITI (I) % BOD removal ≥ 60% theoretical BOD removal within 28 days

301D: Closed bottle % BOD removal ≥ 60% theoretical BOD removal within 28 days and within 10- or 14-days window after 10% BOD removal is reached

301E: Modified OECD

screening % DOC removal ≥70% DOC removal within 10% 10-days window after DOC removal is reached

301F: Manometric

respirometry % BOD removal ≥ 60% theoretical BOD removal within 28 days and within 10-days window after 10% BOD removal is reached Inher ent Biodegr adability

302A: Modified SCAS Test % DOC removal in daily cycles 20%–70% daily DOC removal during 12-week testing

302B: Zahn-Wellens/EMPA % DOC removal 20%–70% DOC removal within 28 days

302C: Modified MITI Test (I) % BOD removal and/or % loss of parent compound 20%–70% BOD or parent compound removal within 28 days

304A: Inherent

biodegradability in soil Production of $^{14}CO_2$ from radiolabeled substrate Not specified Simulation (confirmation)

303A: Aerobic sewage

treatment: coupled units test % DOC removal Degradation rate is calculated IL, 1985.

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GLOSSARIES

Acid Number: A measure of the amount of KOH needed to neutralize all or part of the acidity of a petroleum product.

Additive: Any material added to base stock to change its properties, characteristics, or performance.

Anhydrous: A lubricating grease without water (as determined by ASTM D 128).

Aniline Point: The lowest temperature at which equal volumes of aniline and hydrocarbon fuel or lubricant base stock are completely miscible. A measure of the aromatic content of a hydrocarbon blend, used to predict the solvency of a base stock or the cetane number of a distillate fuel.

Apparent Viscosity: A measure of the viscosity of a non-Newtonian fluid under specified temperature and shear rate conditions.

Bactericide: Additive to inhibit bacterial growth in the aqueous component of fluids, preventing foul odors.

Bases: Compounds that react with acids to form salts plus water. Alkalis are water-soluble bases, used in petroleum refining to remove acidic impurities. Oil-soluble bases are included in lubricating oil additives to neutralize acids formed during the combustion of fuel or oxidation of the lubricant.

Base Number: The amount of acid (perchloric or hydrochloric) needed to neutralize all or part of a lubricant's basicity, expressed as KOH equivalents.

Base Stock: The base fluid, usually a refined petroleum fraction or a selected synthetic material, into which additives are blended to produce finished lubricants.

Bleeding: Separation of liquid lubricant from grease. ing desired physical properties.

Boundary Lubrication: Lubrication between two rubbing surfaces without the development of a full fluid lubricating film. It occurs under high loads and requires the use of antiwear or extreme-pressure (EP) additives to prevent metal-to-metal contact.

Bright Stock: A heavy residual lubricant stock with low pour point, used in finished blends to provide good bearing lm strength, prevent scuffing, and reduce oil consumption. Usually identified by its viscosity, SUS at 210°F, or cSt at 100°C.

Brookfield Viscosity: Measure of apparent viscosity of a non-Newtonian fluid as determined by the Brookfield viscometer at a controlled temperature and shear rate.

Bulk Appearance: Appearance of an undisturbed grease surface. Bulk appearance is described by:

- Bleeding: Free oil on the surface (or in the cracks of a cracked grease.)
- Cracked: Surface cracks.
- Grainy: Composed of small granules or lumps of constituent thickener.
- Rough: Composed of small irregularities.
- Smooth: Relatively free of irregularities.

Cetane Number: A measure of the ignition quality of a diesel fuel, as determined in a standard single cylinder test engine, which measures ignition delay compared to primary reference fuels. The higher the cetane number, the easier a high-speed, direct-injection engine will start, and the less “white smoking” and “diesel knock” after startup.

Cloud Point: The temperature at which a cloud of wax crystals appears when a lubricant or distillate fuel is cooled under standard conditions. Indicates the tendency of the material to plug filters or small orifices under cold weather conditions.

Coefficient of Friction: Coefficient of static friction is the ratio of the tangential force initiating sliding motion to the load perpendicular to that motion. Coefficient of kinetic friction (usually called “coefficient of friction”) is the ratio of the tangential force sustaining sliding motion at constant velocity to the load perpendicular to that motion.

Cohesion: Molecular attraction between grease particles contributing to its resistance to flow.

Complex Soap: A soap crystal or fiber formed usually by co-crystallization of two or more compounds. Complex soaps can be a normal soap (such as metallic stearate or oleate), or incorporate a complexing agent which causes a change in grease characteristics—usually recognized by an increase in dropping point.

Consistency: The resistance of a lubricating grease to deformation under load. Usually indicated by ASTM Cone Penetration, ASTM D 217 (IP 50), or ASTM D 1403.

Copper Strip Corrosion: A qualitative measure of the tendency of a petroleum product to corrode pure copper.

Corrosion: The wearing away and/or pitting of a metal surface due to chemical attack.

Corrosion Inhibitor: An additive that protects lubricated metal surfaces from chemical attack by water or other contaminants.

Demulsibility: A measure of the fluid’s ability to separate from water.

Density: Mass per unit volume.

Dispersant: An additive that helps keep solid contaminants in a crankcase oil in colloidal suspension, preventing sludge and varnish deposits on engine parts. Usually nonmetallic (“ashless”), and used in combination with detergents.

Dropping Point: The temperature at which grease becomes soft enough to form a drop and fall from the orifice of the test apparatus of ASTM D 566 (IP 132) and ASTM D 2265.

Dry Film Lubricant: A low shear-strength lubricant that shears in one particular plane within its crystal structure (such as graphite, molybdenum disulfide and certain soaps). and high speeds in rolling elements where the mating parts deform elastically due to the incompressibility of the lubricant film under very high pressure.

Emulsifier: Additive that promotes the formation of a stable mixture, or emulsion, of oil and water.

Evaporation Loss: The loss of a portion of a lubricant due to volatization (evaporation). Test methods include ASTM D 972 and ASTM D 2595.

Extreme Pressure Property: That property of a grease that, under high applied loads, reduces scuffing, scoring, and seizure of contacting surfaces. Common laboratory tests are Timken OK Load (ASTM D 2509 and ASTM D 2782) and 4-Ball

Load Wear Index (ASTM D 2596 and ASTM D 2783).

Flash Point: Minimum temperature at which a fluid will support instantaneous combustion (a flash) but before it will burn continuously (fire point). Flash point is an important indicator of the fire and explosion hazards associated with a petroleum product.

Friction: Resistance to motion of one object over another. Friction depends on the smoothness of the contacting surfaces, as well as the force with which they are pressed together.

Fretting: Wear characterized by the removal of fine particles from mating surfaces. Fretting is caused by vibratory or oscillatory motion of limited amplitude between contacting surfaces.

Fuel Ethanol: Ethanol (ethyl alcohol, C_2H_5OH) with impurities, including water but excluding denaturants.

Homogenization: The intimate mixing of grease to produce a uniform dispersion of components.

Hydrolytic Stability: Ability of additives and certain synthetic lubricants to resist chemical decomposition (hydrolysis) in the presence of water.

Kinematic Viscosity: Measure of a fluid's resistance to flow under gravity at a specific temperature (usually 40˚C or 100˚C).

Lubricating Grease: A solid to semifluid dispersion of a thickening agent in liquid lubricant containing additives (if used) to impart special properties.

Naphthenic: A type of petroleum fluid derived from naphthenic crude oil, containing a high proportion of closed-ring methylene groups.

Neutralization Number: A measure of the acidity or alkalinity of an oil. The number is the mass in milligrams of the amount of acid (HC1) or base (KOH) required to neutralize one gram of oil.

Neutral Oil: The basis of most commonly used automotive and diesel lubricants; they are light overhead cuts from vacuum distillation.

Newtonian Behavior: A lubricant exhibits Newtonian behavior if its shear rate is directly proportional to the shear stress. This constant proportion is the viscosity of the liquid.

Newtonian Flow: Occurs in a liquid system where the rate of shear is directly proportional to the shearing force. When shear rate is not directly proportional to the shearing force, flow is non-Newtonian.

NLGI Number: A scale for comparing the consistency (hardness) range of greases (numbers are in order of increasing consistency). Based on the ASTM D 217 worked penetration at 25˚C (77˚F).

Non-Newtonian Behavior: The property of some fluids and many plastic solids (including grease), of exhibiting a variable relationship between shear stress and shear rate.

Non-Soap Thickener: Specially treated or synthetic materials (not including metallic soaps) dispersed in liquid lubricants to form greases. Sometimes called “synthetic thickener,” “inorganic thickener,” or “organic thickener.”

Oxidation: Occurs when oxygen attacks petroleum fluids. The process is accelerated by heat, light, metal catalysts and the presence of water, acids, or solid contaminants. It leads to increased viscosity and deposit formation.

Oxidation Inhibitor: Substance added in small quantities to a petroleum product to increase its oxidation resistance, thereby lengthening its service or storage life; also called “antioxidant.”

Paraffinic: A type of petroleum fluid derived from paraffinic crude oil and containing a high proportion of straight chain saturated by hydrocarbons; often susceptible to cold-flow problems.

Poise: Measurement unit of a fluid’s resistance to flow (i.e., viscosity), defined by the shear stress (in dynes per square centimeter) required to move one layer of fluid along another over a total layer thickness of one centimeter at a velocity of one centimeter per second. This viscosity is independent of fluid density and directly related to flow resistance. Viscosity shear stress shear rate dynes/cm cm/s/cm = dynes/cm 2 = = 2 s =1 poise

Pour Point: An indicator of the ability of an oil or distillate fuel to flow at cold operating temperatures. It is the lowest temperature at which the fluid will flow when cooled under prescribed conditions.

Pour Point Depressant: Additive used to lower the pour point or low-temperature fluidity of a petroleum product.

Pumpability: The low temperature, low shear stress-shear rate viscosity characteristics of an oil that permit satisfactory flow to and from the engine oil pump and subsequent lubrication of moving components.

Rheology: The deformation and/or flow characteristics of grease in terms of stress, strain, temperature, and time (commonly measured by penetration and apparent viscosity).

Rust Preventative: Compound for coating metal surfaces with a film that protects against rust. Commonly used to preserve equipment in storage.

Saponification: The formation of a metallic salt (soap) due to the interaction of fatty acids, fats, or esters generally with an alkali.

Sludge: A thick, dark residue, normally of mayonnaise consistency, that accumulates on nonmoving engine interior surfaces. Generally removable by wiping unless baked to a carbonaceous consistency. Its formation is associated with insolubles overloading of the lubricant.

Stoke (St): Kinematic measurement of a fluid's resistance to ow defined by the ratio of the fluid's dynamic viscosity to its density.

Synthetic Lubricant: Lubricating fluid made by chemically reacting materials of a specific chemical composition to produce a compound with planned and predictable properties.

Texture: The texture of a grease is observed when a small portion of it is pressed together and then slowly drawn apart. Texture can be described as:

• Brittle: ruptures or crumbles when compressed

• Buttery: separates in short peaks with no visible fibers

• Long fibers: stretches or strings out into a single bundle of fibers

• Resilient: withstands a moderate compression without permanent deformation or rupture

• Short fiber: short break-off with evidence of fibers

• Stringy: stretches or strings out into long fine threads, but with no evidence of fiber structure

Thickener: The structure within a grease of extremely small, uniformly dispersed particles in which the liquid is held by surface tension and/or other internal forces. study of lubrication, friction, and wear.

Viscosity: A measure of a fluid's resistance to flow.

Viscosity Index: Relationship of viscosity to temperature of a fluid. High-viscosity-index fluids tend to display less change in viscosity with temperature than low-viscosity-index fluids.

Viscosity Modifier: Lubricant additive, usually a high-molecular-weight polymer, that reduces the tendency of an oil's viscosity to change with temperature.

Water Resistance: The resistance of a lubricating grease to adverse effects due to the addition of water to the lubricant system. Water resistance is described in terms of resistance to washout due to submersion (see ASTM D1264) or spray (see ASTM D4049), absorption characteristics and corrosion resistance (see ASTM D1743).

White Oil: Highly refined lubricant stock used for specialty applications such as cosmetics and medicines.

Yield: The amount of grease (of a given consistency) that can be produced from a specific amount of thickening agent; as yield increases, percent thickener decreases.

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FIGURE 19.22 The fluid degradation cycle.

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APPENDIX 1

International Specifications and Use Guides for Fire-Resistant Hydraulic Fluids Including

Phosphate Esters

Organization Standard Number Title

ISO 6743-4 Lubricants, industrial oils and related products (class L). Classi�cation—Part 4: Family H (Hydraulic systems)

ISO 7745 Hydraulic Fluid Power—Fire-resistant (FR) �uids—Guidelines for use

ISO

ISO

CEN 10050 12922 TR14489 Lubricants, industrial oils and related products (class L)— Family T (Turbines) — Speci�cations of triaryl phosphate ester turbine control uids (category ISO-L-TCD) Lubricants, industrial oils and related products (class L)— Family H (hydraulic systems)—Speci�cations for categories HFAE, HFAS, HFB, HFC, HFDR and HFDU Fire-resistant hydraulic �uids—Classi�cation and speci�cation—Guidelines on selection for the protection of safety, health and the environment

ISO 11365 Maintenance and use guide for triaryl phosphate ester turbine control �uids

Additional National Specifications and Use Guides for Fire-Resistant Fluids Including

Phosphate Esters

Country Organization Standard number Title

Canada Canadian Standards CSA M423-M87 Fire-resistant hydraulic �uids

China Chinese National Standards DL/T 571-95 Guide for acceptance, in-service supervision, and maintenance of �re-resistant �uid used in power plant

Germany DIN 24320 Schwerentflammbare Flüssigkeiten—Flüssigkeiten der Kategorien HFAE and HFAS—Eigenschaften und Anforderungen

India Indian Bureau of Standards IS: 10531 Code of practice for the selection and use of fireresistant fluids

USA ANSI/(NFPA) T2.13.8 T2.13.1 T2.13.5 Hydraulic fluid power—Fire-resistant fluids— Definitions, classifications and testing Practice for the use of fire-resistant hydraulic uids for industrial fluid power systems Hydraulic fluid power—Industrial systems—Practice for the use of high water content fluids Key to Appendix 1 ISO International Standards Organization IEC International Electrotechnical Commission ANSI American National Standards Institute NFPA National Fluid Power Association (USA)

APPENDIX 2 Suitable Test Methods for Monitoring Phosphate Ester Quality Fluid property Test method Kinematic viscosity ISO 3104 Neutralization no. ISO 6618/6619 Pour point ISO 3016 Density ISO 3675 Foaming ISO 6247 Air release ISO 9120 Rust pre vention ISO 7120 Corrosion protection ISO 4404-2 Water content ISO 760 Flash/fire points ISO 2592 Spray ignition ISO 15029-2 Hot surface ignition ISO 20823 Wick flame persistence ISO 14935 Particulate levels ISO 11500/4406 Emulsion stability ISO 6614 Color ISO 2049 Volume resistivity IEC 60247 Chlorine content IP 510 Mineral oil Thin-layer chromatography Metal content ASTM D2788 (mod)

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21 Chapter 21 Food-Grade Hydraulic Fluids

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