Evaporators

On all vessels there is a need for drinking water and on many vessels there is also a need for distillate water. In most instances it is more economical not to load this water from the shore but to produce it from seawater. One of the options is to evaporate seawater and afterwards condense the vapour. The salt in the seawater does not evaporate and when no small droplets of water are going along with the vapour, the condensate will be pure distillate water.
Distillate water is used as:

1. feed water for boilers
2. jacket cooling water
3. batteries

To use distillate water as drinking water air must be blown trough it, which takes place in the fresh water hydrofoor and minerals must be added to give taste to the water.
In most cases chloride tablets are added in the drinking water tanks to kill bacteria which can be present in the small water droplets going along with the vapour.

On modern vessels evaporators use the waste heat as heating medium. On motor vessels this is heat from the jacket cooling water (80 to 90^0 C). By the low pressure in the evaporator (+/- 93 % vacuum) the sea water will boil at +/- 40^0 C.

Description of the ALFA LAVAL NIREX evaporator. (See figures 1 and 2))

The NIREX freshwater distiller is a vacuum evaporation distiller, normally using waste heat from the fresh cooling water of a diesel engine as heating medium.

The distiller consists of the following main components:

1. Separator vessel
   The separator vessel separates the produced steam from the brine.
2. Evaporator section
   The evaporator section consists of a plate heat exchanger, and is enclosed in the separator vessel.
3. Condenser section
   The condenser section like the evaporator section consists of a plate heat exchanger, and is enclosed in the separator vessel.
4. Combined brine/air ejector
   The combined brine/air ejector extracts air and brine from the separator vessel.

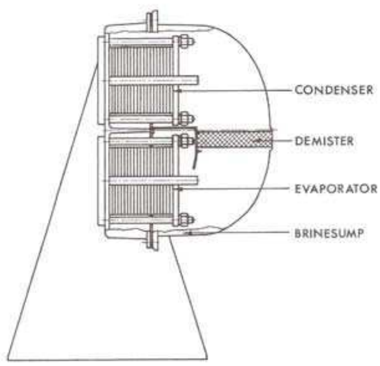


figure 1 Cross section evaporator

5. Combined ejector/cooling water pump
   The combined ejector/cooling water pump supplies sea water for the condenser, jet water for the combined brine/air ejector, and feed water for evaporation.

6. Fresh water extraction/transfer pump
   The fresh water extraction/transfer pump extracts the produced water from the condenser, and transfers same to the fresh water tank.
7. Salinometer
   The salinometer checks continuously the salinity of the produced water.
8. Electrical panel
   The panel contains starters for the electric motors and terminals for the salinometer.

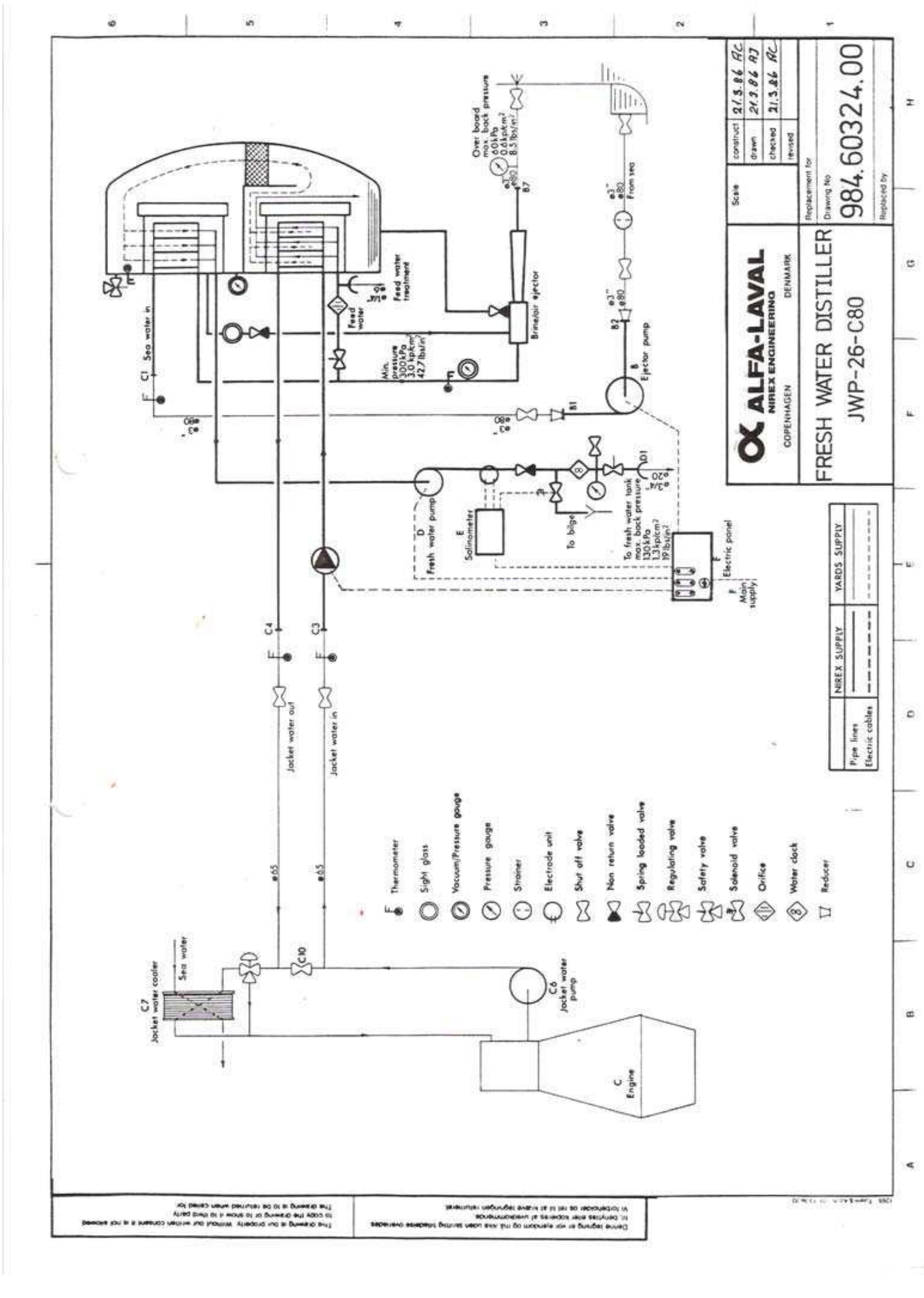
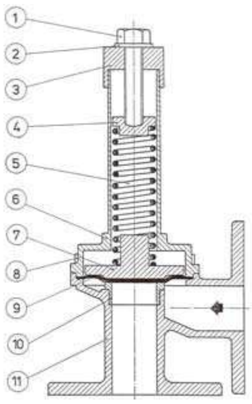


figure 2 ALFA LAVAL NIREX installation

The principle of operation of a vacuum evaporator

The vacuum necessary for the evaporation is established and maintained by the combined brine/air ejector. The ejector is driven by the combined ejector/cooling water pump.
The feed water enters the evaporator section through an orifice and distributes itself into every second plate channel (evaporation channels). The jacket cooling water distributes itself into the remaining channels thus transferring its heat to the feed water in the evaporation channels. Having reached the boiling temperature the feed water undergoes partial evaporation.
The mixture of generated steam and brine enters the separation vessel, where the brine is separated from the steam. When a part of the sea water evaporates the salt contents of the remaining water will increase. When the salt contents rises too much the density of the water increases resulting in the creation of viscous bubbles filling up the total space and penetrate through the demister destroying the fresh water production. To limit the salt contents a continuous amount of brine must be extracted by the combined brine/air ejector.
Having passed a demister (filter) the steam enters every second plate channel in the condenser section (condensation channels). The sea water supplied by the combined ejector/cooling water pump distributes itself into the remaining channels thus absorbing the heat transferred from the steam during the condensation. The produced fresh water is extracted and transferred to the tank by the fresh water extraction/transfer pump.

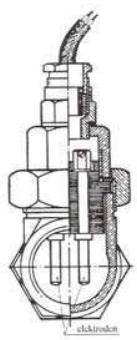


On the delivery side of the pump an electrode unit is fitted, which together with the salinometer continuously checks the salt content of the produced water. In case the salinity measured by the salinometer will exceed the set value a signal is given to the solenoid valve to open en the water is dumped in the bilges. Further on the delivery side a water meter and a back pressure valve (figure 3) are fitted. The function of the back pressure valve is to maintain a steady pressure in the delivery line and consequently in the suction line of the extraction/transfer pump. This will create a certain level of fresh water to remain in the condenser area avoiding that the suction line runs dry and air might enter the evaporator.

figure 3 Back pressure valve

Salinometer

figure 4 Electrodes salinometer



The salt content is measured by the electrical conductivity of the produced fresh water. Pure distillate water can not conduct an electric current. The more acids, base or salts are diluted in the water the better it will conduct electricity.

The salinometer indicates how much parts per million (ppm) salts are present in the fresh water. When the ppm level will exceed a preset value (mostly 15 ppm) an electrical signal will open the solenoid valve and the water is dumped. At the same time an acoustic alarm is activated.

Ejector (see figure 5)

The principle of an ejector is based on the fact that a jet of water with a high speed is capable to drag air or water surrounding the jet with it.

Water is supplied to the ejector and in the nozzle (1) the pressure is converted into a high speed. The surface of the jet leaving the nozzle is not smooth but frayed. The jet will drag air and brine from the separator body with it. The mixture will enter the ventury tube. The first part of the tube is called the mixing nozzle (2) were water, air and brine will mix and the speed of the mixture will increase. The second part of the ventury, called the diffuser (3) has an expanding shape transferring the speed of the mixture into pressure.

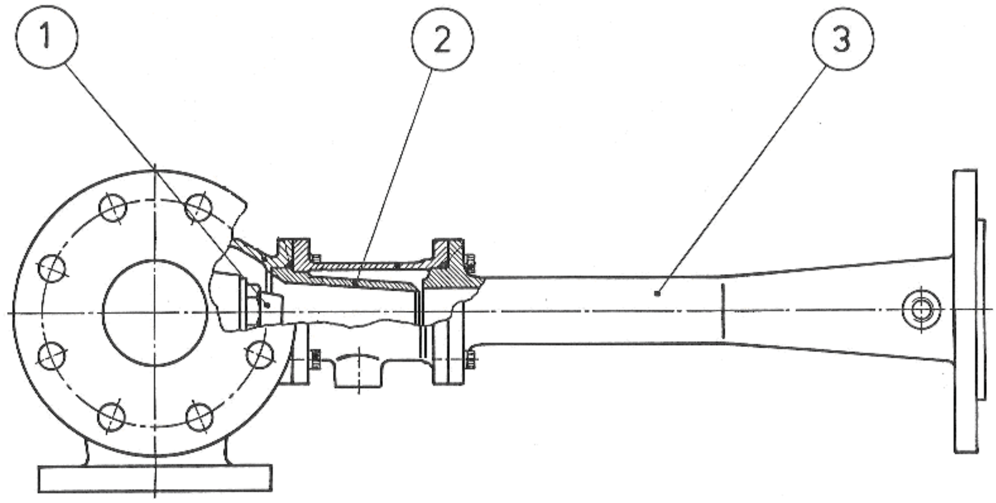


Figure 5 Combined Brine/Air ejector

Problems and maintenance

At places where sea water evaporates scaling will occur. Scales hamper transfer of heat. The lower the temperature at which the evaporation takes place, the smaller the possibility that scales will create a hard layer. This layer will cause the production of the evaporator to reduce. The scaling can mainly only be removed by using chemicals. The acid chemicals will react heavily on the scales. Caution must be taken handling these chemicals. Wearing of safety goggles and gloves is a must.

When after some time the scales have been removed by the chemicals the chemicals still are active. By adding soda the chemicals can be neutralized. By checking the pH of the liquid inside the evaporator it is possible to check if the chemicals are still active. As soon as the pH has become 7 the liquid has become neutral and is it allowed to be pumped overboard.

In every system working with vacuum there is the possibility that air might be sucked in. When this happens the combined brine/air ejector might not be capable to maintain the vacuum. The consequence is that the production of the evaporator will reduce.

When air is present in the evaporator is the total pressure inside equal to the vapour pressure and the air pressure. (Law of Dalton)

Salt balance

The salt contents z (mass salt) of sea water varies from place to place. It is difficult to measure direct and therefore it is chosen to use the density, which can be measured with an aerometer. Knowing the temperature during the measurement the salt contents can be easily calculated from the density using the table or the graph shown in figure 6 and table 1.

Salt contents Z (mass percentage sea salt)	Density at 20^0C 10^3 kg/m^3
0.005	1.0019
0.010	1.0057
0.015	1.0094
0.020	1.0132
0.025	1.0170
0.030	1.0207
0.035	1.0245
0.040	1.0283
0.045	1.0320
0.050	1.0358
0.055	1.0396
0.060	1.0434
0.065	1.0472
0.070	1.0509
0.075	1.0547
0.080	1.0585
0.085	1.0623
0.090	1.0662
0.095	1.0700
0.100	1.0738
0.110	1.0814
0.120	1.0891
0.130	1.0968
0.140	1.1045
0.150	1.1122

table 1

The relation between the salt contents and the amount of brine to be extracted can be found by the salt balance:

salt in = salt out

$(a + x)\, Z_{sw} = x\, Z_b$ in which:
- a = mass produced fresh water
- x = mass brine
- $a + x$ = mass supply feed water
- Z_{sw} = salt contents sea water
- Z_b = salt contents brine

In many evaporators the density of the brine is 1060 kg/m^3 or lower.

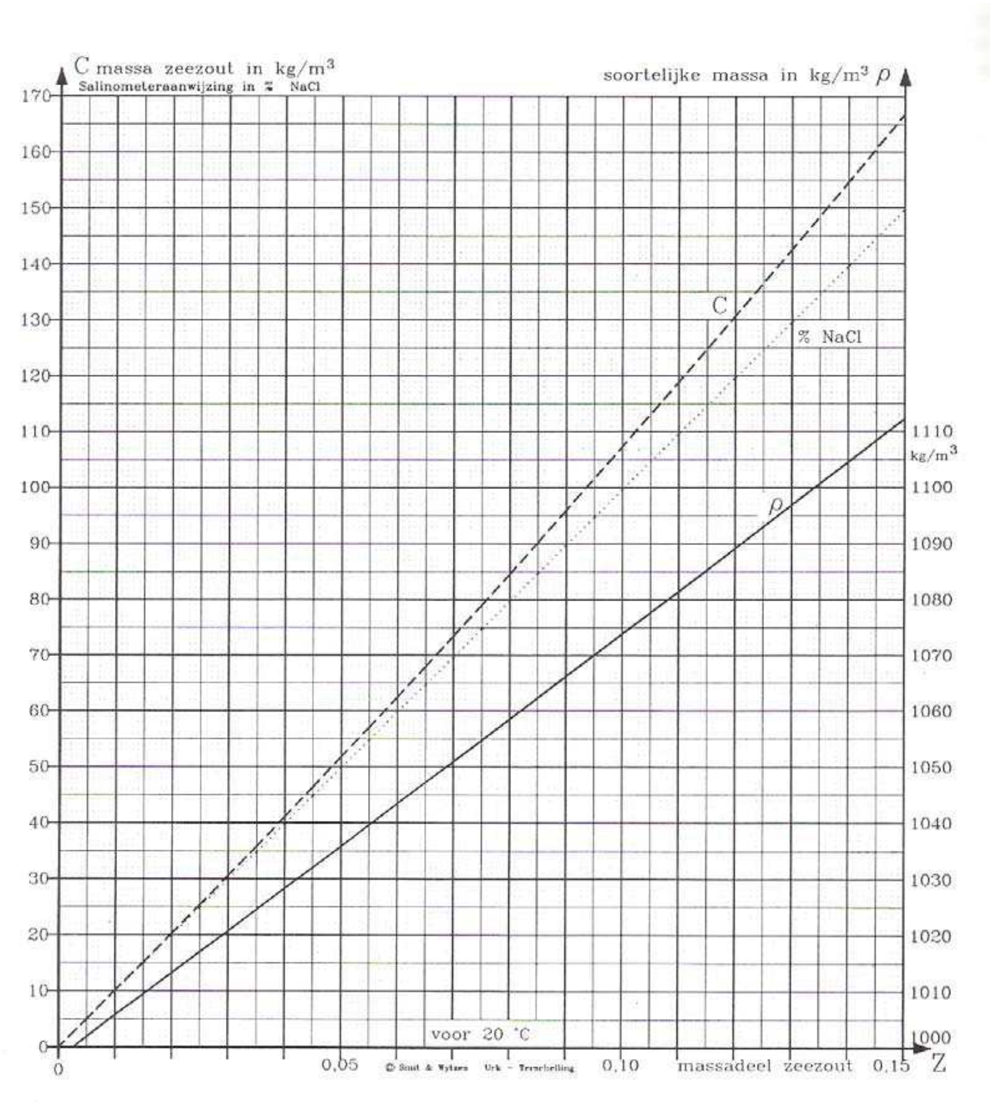


figure 6 Salt water graphs

Example

Evaporator production $\quad a = 24$ tonnes/24 hr
Density sea water $\quad \rho_z = 1024.5\ kg/m^3$

Determine the mass brine and feed water to maintain the maximum density of the brine $\rho_b = 1045\ kg/m^3$

Solution

From the table $\quad \rho = 1024.5 \rightarrow Z = 0.035$
$\quad \rho = 1045 \rightarrow Z = 0.0625$

Salt in = Salt out

$(24 + x) * 0.035 = x * 0.0625$
$24 * 0.035 + x * 0.035 = x * 0.0625$
$24 * 0.035 = x (0.0625 – 0.035)$
$x = 30.55$ tonnes/24 hr
$a + x = 54.55$ tonnes/24 hr

Example

Density sea water	$\rho_z = 1030\ kg/m^3$
Production fresh water	$a = 2000$ kg/hr
Volume feed water	$V_{fw} = 4000$ m3/hr

Determine:
a. Mass feed water
b. Mass brine
c. Density brine

Solution

Table:	$\rho_z = 1030\ kg/m^3$ -> $Z_{sw} = 0.0423$ (interpolation)
Mass feed water:	$m_{fw} = a + x = V_{fw} * \rho_z = 4000 * 1030 = 4120$ kg/hr
Mass brine:	$x = 4120 – 2000 = 2120$ kg/hr
Salt in = Salt out:	$(a + x) * Z_{sw} = x * Z_b$
	$4120 * 0.0423 = 2120 * Z_b$
	$Z_b = 0.0822$
	In table can be found: $\rho_b = 1060\ kg/m^3$

Heat balance

The extraction of brine keeps the evaporator clean but creates a heat loss. To calculate the production of an evaporator at a given heat supply you make use of the heat balance:

heat in = heat out

$$(a + x)h_{sw} + m_{cw}h_1 = ah_v + xh_b = m_{cw}h_2$$

Or in other words:

heat from the cooling water = heat in the vapour + heat in the brine

$$m_{cw}(h_1 – h_2) = a(h_v-h_{sw}) + x(h_b – h_{sw})$$

in which:
m_{cw} = mass flow cooling water
h_{sw} = enthalpy supply sea water
h_1 = enthalpy cooling water in
h_2 = enthalpy cooling water out
h_v = enthalpy vapour
h_b = enthalpy brine

h_b and h_v can be found in the steam table at the corresponding pressure in the evaporator. h_1, h_2 and h_{sw} can be found in the temperature table. The mistake, as we will find there the enthalpy of pure water instead of sea water is very small and can be disregarded.

h_1, h_2 and h_{sw} can also be calculated by using: $h = c * t$

in which: c = specific heat = 4.19 kJ/kg.^{0}C
t = temperature in ^{0}C

Fresh water hydrofoor

The water is sucked from one of the fresh water tanks by the pump and discharged to the pressure vessel. This tank is partly filled with air. When more water is pumped in the tank the volume of air becomes smaller and the pressure will rise. (Law of Boyle: $p * V$ = constant). When the required pressure is reached the pump will be switched of by a pressure switch.
The connection to the various consumers is at the bottom of the tank. When water is consumed, the air pressure will push the water out of the tank to the consumer. The air will expand and consequently the pressure will drop. When the pressure drop reaches the lower set point (called differential) of the pressure switch, a signal is given to the pump to start again.
By the pressure switch the pressure in the hydrofoor is maintained between the upper and lower set point.
Air slowly dissolves in water over a period of time, creating the water level in the tank to rise and the air cushion become too small. The consequence is that the on-off switching of the pump takes place at shorter intervals. When this happens additional air must be pressed inside the tank.

As described above air must be blown through the distillate water from the evaporator to give taste to the water. This can be done in the hydrofoor by spraying the water via nozzles in the air cushion.(see figure 7)

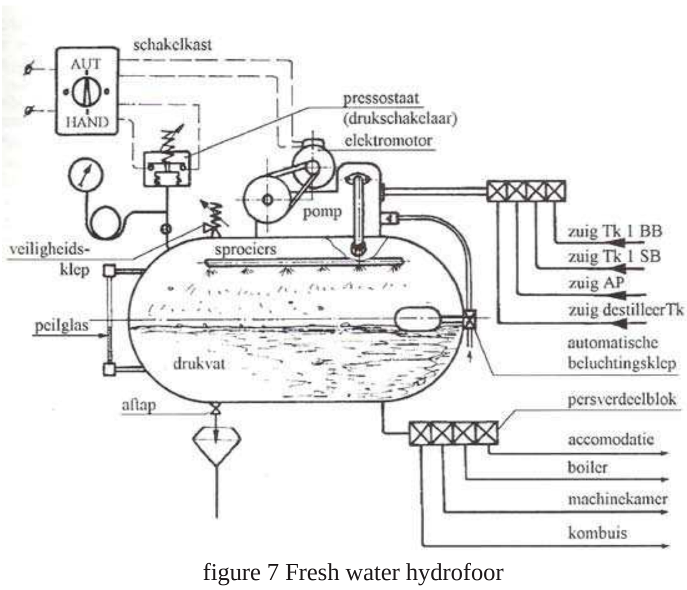


figure 7 Fresh water hydrofoor

Questions evaporators

1. A vacuum evaporator makes use of cooling water from the main engine. The condenser is cooled by sea water, after which a part is used as feed water to the evaporator. ¾ of the feed water is extracted as brine.

Production fresh water	a = 4000 kg/24 hr
Pressure in evaporator	$p_v = 0.1$ bar
Temperature cooling water in	$t_{cw1} = 80^0C$
Temperature cooling water out	$t_{cw2} = 70^0C$
Temperature sea water condenser in	$t_{sw1} = 8^0C$
Temperature sea water condenser out	$t_{sw2} = 10^0C$
Temperature condensate	$t_{cond} = 15^0C$

Determine:

a) Mass flows feed water and brine
b) Mass flows engine cooling and sea cooling water circulating in evaporator
c) The amount of heat produced by the engine cooling water to produce 1 kg of condensate

Answers:

a) 4 kg of feed water will give 3 kg of brine and 1 kg of condensate
$a = 4000/(24 * 3600) = 0.0.463$ kg/sec
$x = 3 * a = 3 * 0.0463 = 0.139$ kg/sec
$m_{fw} = x + a = 4 * 0.0463 = 0.185$ kg/sec
b) Heat balance evaporator: heat in = heat out
$m_{cw}(h_1 – h_2) + m_{fw} * h_{fw} = a * h_v + x * h_b$
$m_{cw}(334.72 – 292.78) + 0.185 * 42.03 = 0.0463 * 2583.9 + 0.139 * 191.7$
$m_{cw} * 41.9 + 7.78 = 120 + 26.6$
$m_{cw} = 3.3$ kg/sec

heat balance condenser
$a * h_v – a * h_{cond} = m_{sw}(h_{sw2} – h_{sw1})$
$120 – 0.0463 * 62.9 = m_{sw} (42.03 – 33.6)$
$m_{sw} = 10.8$ kg/sec
c) Per second $m_{cw}(h_1 – h_2) = 3.3(334.72 – 292.78) = 138$ kJ energy is transferred from the engine cooling water.
Per second 0.0463 kg of fresh water is produced
Energy used per kg produced fresh water = 138/0.0463 = 2990 kJ/kg

2. Following data are given of a vacuum evaporator

Production condensate	a	= 0.06 kg/sec
Enthalpy vapour	h_v	= 2500 kJ/kg
Enthalpy brine	h_b	= 160 kJ/kg
Enthalpy engine cooling water in	h_{cw1}	= 335 kJ/kg
Enthalpy engine cooling water out	h_{cw2}	= 290 kJ/kg
Enthalpy feed water	h_{fw}	= 80 kJ/kg

For each kg condensate 2 kg brine are extracted

Determine the mass flow of engine cooling water circulating through the evaporator.

3. Following data are given of a vacuum evaporator

Density seawater	ρ_{sw}	= 1028.3 kg/m^3
Density brine	ρ_b	= 1054.7 kg/m^3
Temperature feed water	t_{fw}	= 20^{0}C
Specific heat sea and fresh water	c	= 4.2 kJ/kg.K
Temperature brine = evaporation temp	t_b	= 40^{0}C
Temperature decrease engine cooling water	$t_1 - t_2$	= 8^{0}C
Production condensate	a	= 500 kg/hr

Determine the mass flow of engine cooling water circulating through the evaporator.

4. Following data are given of a vacuum evaporator

Temperature engine cooling water in	t_1	= 70^{0}C
Temperature engine cooling water out	t_2	= 60^{0}C
Temperature feed water	t_{fw}	= 17.51^{0}C
Temperature brine = evaporation temp	t_b	= 40^{0}C
Mass flow of engine cooling water	m_{cw}	= 40 kg/sec

For each kg condensate 2.5 kg brine are extracted

Determine the production of condensate.

5. Following data are given of a vacuum evaporator

Mass flow feed water	m_{fw}	= 0.5 kg/sec
Density feed water	ρ_{fw}	= 1024.5 kg/m^3
Density brine	ρ_b	= 1050.9 kg/m^3

Determine the production of the condensate and the mass flow brine.

6. Following data are given of a vacuum evaporator

Mass flow feed water	m_{fw}	= 0.36 kg/sec
Density feed water	ρ_{fw}	= 1032 kg/m^3
Production condensate	a	= 0.12 kg/sec

Determine the density and the mass flow of the brine

7. Following data are given of a vacuum evaporator

Salt contents of feed water	Z_{fw}	= 0.030
Production condensate	a	= 250 kg/hr
Maximum salt contents brine	Z_b	= 0.045

Determine how many kg brine must be extracted per hour and how many kg feed water must be supplied to the evaporator.

8. If the vapour in an evaporator condensate at 40^0C and the pressure is 0.09 bar, does this mean there is air present in the evaporator?

9. What is the function of the demister?

10. What is the reason brine must be extracted from the evaporator?

11. Make a schematic drawing of a vacuum evaporator.

12. Describe how scales can be removed from an evaporator.

13. What must be done to make distillate water suitable for drinking water?

14. Why must the fresh water extraction pump be located at a lower position as the condenser?

15. Describe the working principle of a vacuum evaporator.

Heat exchangers

Heat exchangers are used on board vessels to transfer heat from one medium to another. Examples of heat exchangers are:

1. Lubrication oil cooler
2. Jacket cooling water cooler
3. Steam condenser
4. Refrigeration condenser
5. Fuel oil heater
6. Air cooler

The heat transfer normally takes place via a metal partition. This can either be a tube or a plate.

Shell tube heat exchangers

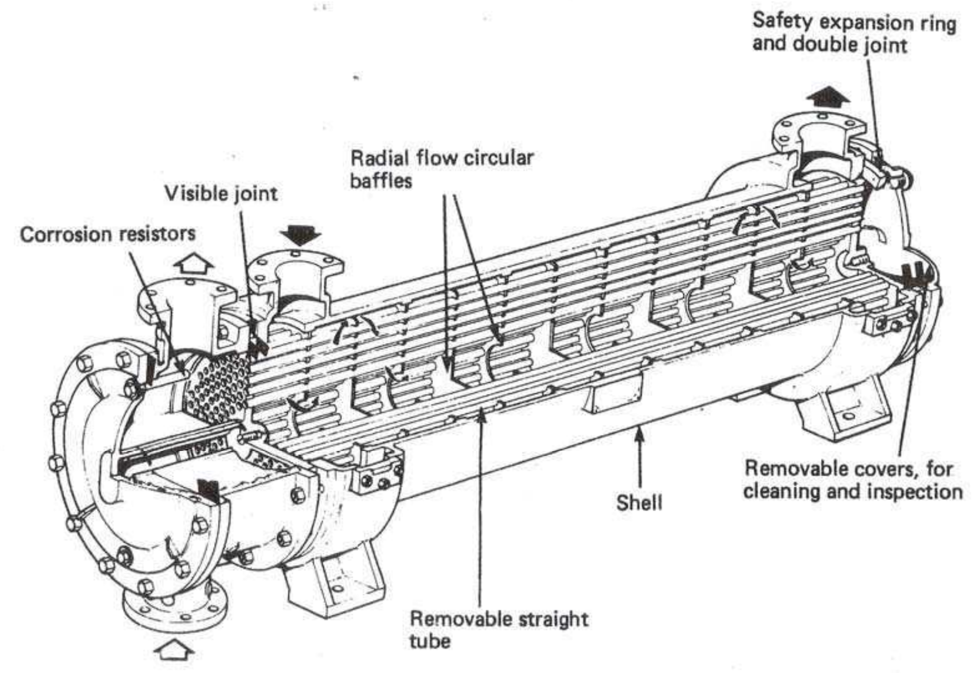


figure 1 Shell – tube type heat exchanger

Shell tube type heat exchangers consist of a pipe bundle placed in a cylindrical housing called the shell. When sea water is used to cool down the other liquid, i.e. lubricating oil or jacked cooling water, the sea water will flow through the pipes. The reason for this is that the pipes can easily be cleaned from inside. The other liquid flows around the pipes.

The pipe bundle is supported by alternating placed radial flow circular baffle plates. The second function of these plates is to force the liquid to flow zigzag through the heat exchanger to increase the heat exchange.

At the entrance and the exit openings of the heat exchanger thermometers are placed in order to check the performance of the heat exchanger.

The materials used in the heat exchanger depend on their function. The tubes and pipe plates are made of bronze, the housing and covers can be made of bronze, steel or cast iron.
To protect the heat exchanger against corrosion sacrificial anodes are placed inside the sea water side. In addition to this the covers can also be coated with a plastic layer.
When different types of material are used and the heat exchanger is put in operation, may cause the different metals to expand differently, depending on their composition. The tubes normally want to expand more as the housing. To allow the tubes to expand freely one of the pipe plates is not fixed between housing and cover but can move freely (See figure 1: Safety expansion ring and double joint). Around this pipe plate two rubber O rings are placed to guarantee no mixing of liquids. At the location of the O rings there is an inspection hole in the housing. In case one of the O rings might damage liquid will flow out of the inspection opening. As the inspection hole is situated between the 2 O rings it becomes directly clear which O ring is damaged as the seal to that liquid side is broken and that liquid will flow out.
Depending on the required cooling surface the cooling medium will enter the heat exchanger at one side and will leave at the other side. This arrangement is referred to as a single-pass heat exchanger.
In case more cooling surface is required it is normal practice to place a division plate in one of the covers. The inlet and outlet connections for the liquid passing through the tubes are fitted on this head. The division plate prevents the liquid bypassing the tubes and causes it to pass through half the tubes in the heat exchanger, which is referred to as the inlet bank.
After the liquid passes through the inlet bank, it enters the other header. The direction of the liquid is reversed in this header and it passes back through the outlet bank of tubes and leaves at the outlet branch.
The liquid has passed through the tubes in two separate paths from which it gets the name two-pass or double-pass type.

Plate type heat exchangers

The disadvantage of shell tube type heat exchangers is that when a large capacity is required they will require a large space. On top of that additional space is required for maintenance or repair. By using plates instead of tubes a much smaller space is required.

Plate type heat exchangers consist of parallel plates fitted with sealing material like rubber, which alternately has coolant, and the liquid to be cooled, flowing through them.
Every plate has in each corner a circular hole. These holes create channels when a number of plates are placed on top of each other. The rubber gaskets are not only placed at the outer circumference of the plate but at each plate two holes are enclosed by a gasket and the other two are open to the centre area of the plate, creating an entrance and an exit opening. Alternating the gaskets at each other plate around two holes will allow the two liquids to flow alternating in between two plates.

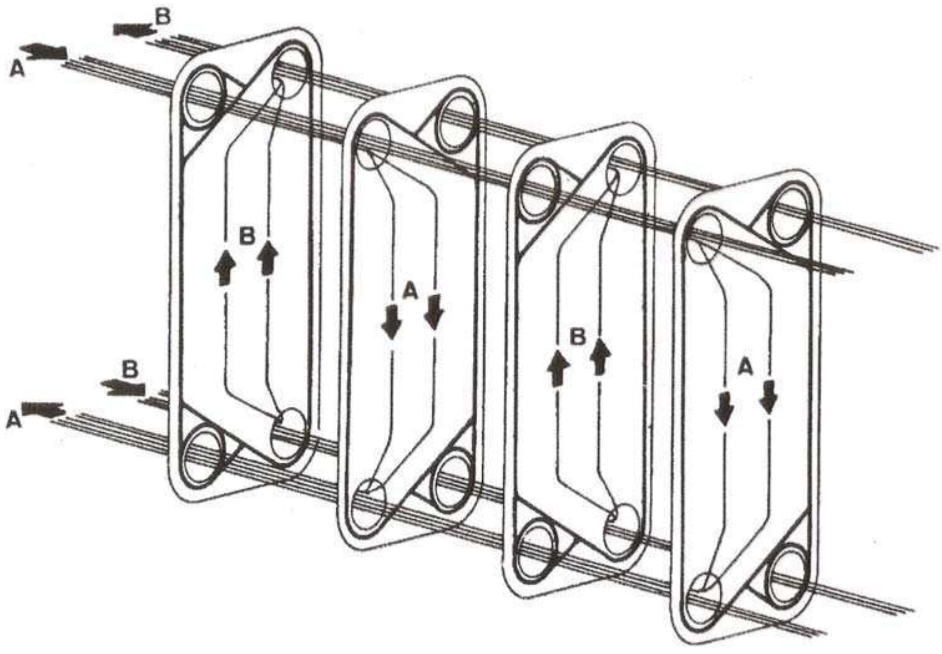


figure 3 Flow through plate type heat exchanger

The plates are made of titanium, which is highly resistant against corrosion. The plates are corrugated in order to:

1. increase the cooling surface
2. make them stronger
3. guide the liquid flow
4. guarantee a certain passage between two plates

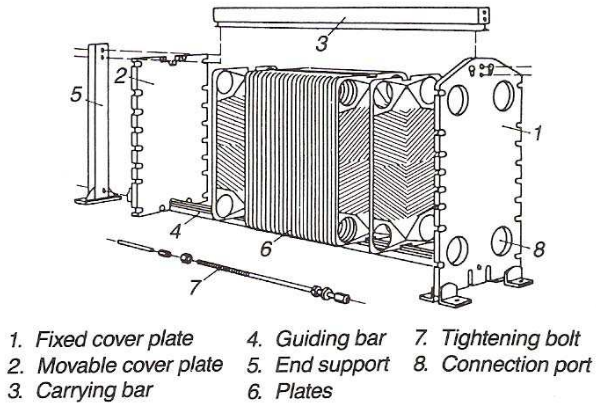


figure 4 Construction plate type heat exchanger

The whole set of plates is placed in between two thick strong cover plates (1 and 2). These plates are connected with each other by six or more tightening bolts (7). By tightening the bolts all the plates will be compressed and become one body. Great care has to be taken not to over tighten the bolts and to make sure that they are equally tight, otherwise leakages might occur. Measuring the distance between the end plates at the position of each bolt before removing the bolts will give an indication how far the bolts must be tightened after the repair or maintenance.

Maintenance

In case a plate type cooler needs maintenance the bolts can be released and the end plate can be slide aside like a curtain. There is now access to all the individual plates and they can easily be cleaned. Caution should be taken not to damage the gaskets. Normally the gaskets will not be removed during maintenance or inspection.

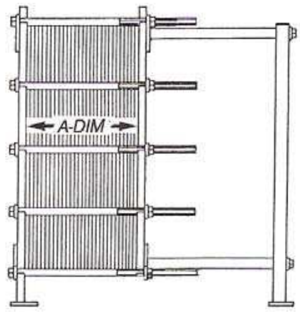


figure 5 Distance between end covers

Heat exchange

The transfer of heat in a heat exchanger from one liquid to the other can be calculated as follows:

$Q = k * A * (t_1 - t_2)_{average}$

$q = Q/A = k * (t_1 - t_2)_{average}$

in which:
- Q = heat in W
- A = cooling surface in m^2
- $(t_1 - t_2)_{average}$ = mean temperature difference between the liquids in ^0C or K
- k = thermal conductivity in $W/m^2.K$
- q = heat transfer concentration in W/m^2

In most instances the amount of heat to be exchanged is known and has the cooling surface of the heat exchanger to be calculated, depending on the temperature difference and the thermal conductivity k which depends on the type of heat exchanger. In order to compare two heat exchangers for a given situation the thermal conductivity must be compared or from both units the heat transfer concentration has to be calculated. For temperature differences ^0C and K are equal.

Thermal conductivity

The thermal conductivity indicates the heat transfer concentration per ^0C or K temperature difference in a heat exchanger. The thickness of the division wall and extend in which it conduct heat are minor factors. Most important are the non moving layer of liquid sticking to the wall and the heat resistance of that layer. By giving the liquids a high speed the thickness of this layer can be reduced, in order to increase the heat transfer. High liquid speeds might result in turbulence which will increase the heat transfer even more. The corrugations of the plates in a plate type heat exchanger are also creating turbulence.
In the table 1 below some k factors are given for different heat exchangers and different constructions.

	Plate type	Shell tube type
Steam condenser	3500 W/m^2.K	2000 W/m^2.K
Refrigerator condenser	2500	1500
Lubricating oil cooler	2000	1000
Fresh cooling water cooler	4000	2500
Boiler tubes		100
Car radiator		50
Refrigerator evaporator		10
Fin type refrigerator evaporator		15
Brine cooler		500

table 1

The values in the table are rough estimates. The differences between the different types of heat exchangers might be large depending on the types of liquid used and their respective speeds along the walls.

Mean temperature difference

Heat will only transfer from hot to cold and never back. A large temperature difference between the cold and hot liquid will create a stronger heat transfer as when the temperature difference is small. Especially in the case of uniflow this effect can be seen clearly.

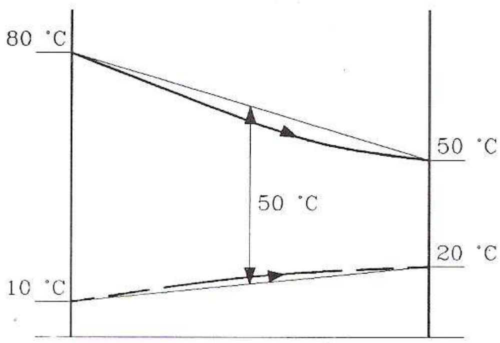


figure 6

In figure 6 oil is entering the heat exchanger with 80^{0}C and sea water with 10^{0}C: temperature difference 70^{0}C. On leaving the heat exchanger the oil is 50^{0}C and the sea water 20^{0}C; temperature difference 30^{0}C. When you calculate the mean temperature difference by mathematical approach you get:

$(t1 - t2)_{max}$ of 70^{0}C and $(t1 - t2)_{min}$ of 30^{0}C giving $(t1 - t2)_{mean}$ = 50^{0}C

In figure 6 it is easy to see that in reality the mean temperature difference is smaller as the calculated value.

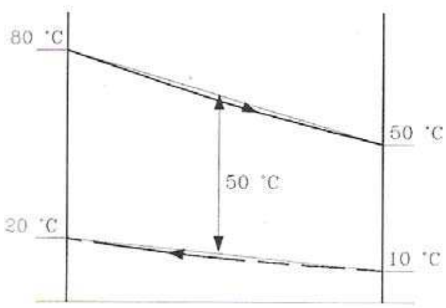


figure 7

In the case of counter flow the change of the temperatures is closer to straight lines. Especially when the rise in temperature of the coolant is equal to the temperature drop of the liquid cooled down.

In figure 7 oil is entering the heat exchanger with 80^{0}C and at that end sea water is leaving with a temperature of 20^{0}C: temperature difference 60^{0}C. The oil leaves the heat exchanger with 50^{0}C and sea water enters that side with 10^0C: temperature difference 40^0C. The mathematical mean temperature difference is (60 + 40)/2 = 50^0C. In the figure we can see that the calculated mean temperature difference is closer to the real mean temperature difference as in the uniflow heat exchanger.

In a shell-tube heat exchanger the flow is mainly cross flow. Cross flow is a mixture of uniflow and counter flow.

In case of condensation of steam or vapour the condensation temperature is constant from entry to exit. This is indicated by a straight line (see figure 8). The temperature development of the cooling water is according a curved line. Calculating the mean temperature difference will also here give an error compared to the real situation.

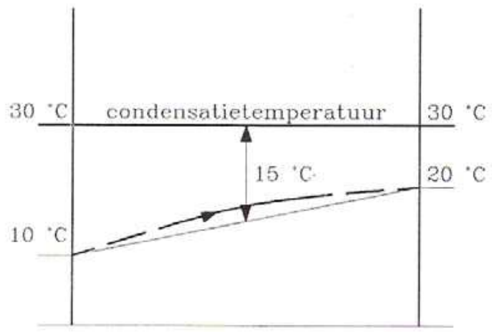


figure 8

Heat balance

For all heat exchangers the following applies:

$$\text{heat in} = \text{heat out}$$

$$m_1H_{11} + m_2h_{21} = m_1h_{12} + m_2h_{22}$$

Or in other words:

$$\text{heat from substance 1} = \text{heat from substance 2}$$

$$m_1(h_{11} - h_{12}) = m_2(h_{22} - h_{21})$$

in which:
- m_1 = massflow substance 1 in kg/sec
- h_{11} = enthalpy substance 1 at entrance in kJ/kg
- h_{12} = enthalpy substance 1 at outlet in kJ/kg
- m_2 = massflow substance 2 in kg/sec
- h_{21} = enthalpy substance 2 at entrance in kJ/kg
- h_{22} = enthalpy substance 2 at outlet in kJ/kg

For every liquid or gas which is not evaporating or condensing in the heat exchanger the enthalpy can be calculated by:

$h = c * t$ in which: c = specific heat in kJ/kg.K
t = temperature in ^0C

P.S. If during a calculation of a heat exchanger the massflow of cooling water has been chosen too small it might happen that the heat balance will reveal an outlet cooling water temperature being higher as the temperature of the liquid to be cooled down, which is impossible in reality. A correct heat balance is not a guarantee that a heat exchanger can work with the found values.

Examples:

In a heat exchanger lub oil is cooled down by sea water

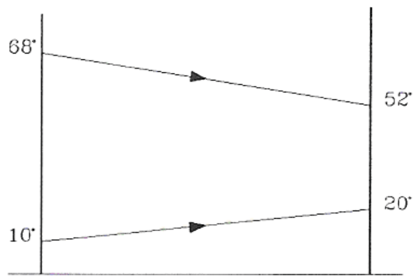
Massflow oil	m_o	= 2.5 kg/sec
Temperature oil at entry cooler	t_{o1}	= 68^0C
Temperature oil at outlet cooler	t_{o2}	= 52^0C
Specific heat oil	c_o	= 2.65 kJ/kg.K
Temperature water at entry cooler	t_{w1}	= 15^0C
Temperature water at outlet cooler	t_{w2}	= 20^0C
Specific heat water	c_w	= 4.19 kJ/kg.K

Determine:

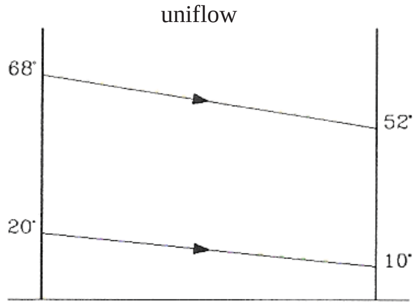
a) Amount of heat transferred by the lub oil.
b) Massflow cooling water.
c) Draw the temperature developments (in straight lines) for uniflow and counter flow.

Solution:

a) $Q = m_o * c_o * (t_{o1} - t_{o2}) = 2.5 * 2.65 * 10^3 * (68 - 52) = 106$ kW
b) $Q = m_w * c_w * (t_{w2} - t_{w1})$
   $106 * 10^3 = mw * 4.19 * 10^3 * (20 - 15)$
   $m_w = 5.06$ kg/sec
c)



uniflow



Counter flow

In an air heater ventilation air is heated by steam.

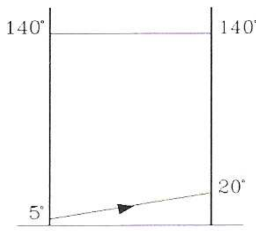
Heating surface	A	= 5 m^2
Thermal conductivity	k	= 150 $W/m^2.K$
Condensation temperature steam	t_{cond}	= 140^0C
Air temperature at entrance	t_{a1}	= 5^0C
Air temperature at outlet	t_{a2}	= 20^0C
Specific heat air	c_a	= 1.0 kJ/kg.K

Determine:

a) Draw the temperature developments and calculate the mean temperature difference.
b) The amount of heat transferred.
c) The massflow of air.

Solution:

a)



$(t_{steam} - t_{air})_{mean} = ((140 - 5) + (140 - 20))/2 = 127.5\ ^0C$

b) $Q = k * A * (t_s - t_a)_{mean} = 150 * 5 * 127.5 = 95.6 * 10^3\ W = 95.6\ kW$
c) $Q = 95.6 * 10^3 = m_a * c_a * (t_{a2} - t_{a1}) = m_a * 1 * 10^3 * 15$
$m_a = 6.37$ kg/sec

In a feedwater heater of an evaporator the feedwater is heated up by the brine taken out from the evaporator. The flow type is counter flow.

Massflow feedwater	m_{fw}	= 2 kg/sec
Massflow brine	m_b	= 1.5 kg/sec
Specific heat of feedwater and brine	c	= 4.19 kJ/kg.K
Temperature feedwater at entrance	t_{fw}	= 10^0C
Temperature brine at entrance	t_{b1}	= 46^0C
Temperature brine at outlet	t_{b2}	= 26^0C
Thermal conductivity	k	= 2500 $W/m^2.K$
Inside diameter pipe	d_i	= 20 mm
Outside diameter pipe	d_u	= 22 mm
Length pipe	l	= 1.03 m

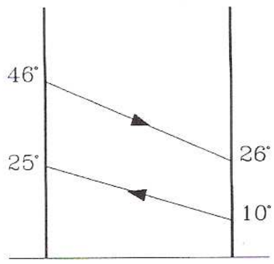
Determine:

a) Temperature of the feedwater at the outlet of the heat exchanger
b) Draw the temperature development for both liquids and determine the calculated mean temperature difference.

c) Calculate the required heated surface.
d) Calculate the number of pipes required.
e) Is it possible to place the heater in uniflow? Explain your answer.

Solution:

a) heat from brine = heat in feedwater
$m_b * c_b * (t_{b1} – t_{b2}) = m_{fw} * c_{fw} * (t_{fw2} – t_{fw1})$
$1.5 * 4.19 * 10^3 * (46 – 26) = 2 * 4.19 * 10^3 * (t_{fw2} – 10)$
$t_{fw2} = 25^0C$
b) mean temperature difference = $((46 – 25) + (26 – 10))/2 = 18.5^0C$



c) $Q = m_b * c_b * (t_{b1} – t_{b2}) = 1.5 * 4.19 * 10^3 * (46 – 26) = 126$ kW
Also:
$Q = k * A * (t_1 – t_2)_{mean}$
$126 * 10^3 = 2500 * A * 18.5$
$A = 2.72\ m^2$
d) Mean pipe diameter $d_{mean} = 21$ mm
Surface of 1 pipe: $A_{pipe} = \pi * d_{mean} * l$
$A_{pipe} = \pi * 0.021 * 1.03 = 0.068\ m^2$
Number of pipes: $z = A/A_{pipe} = 2.72/0.068 = 40$
The cooler must have 40 pipes.
e) When the cooler was placed in uniflow the outlet temperatures would be too close to each other so the required heat transfer will not be achieved.

Temperature control

When a heat exchanger is placed in a system i.e. in the jacket cooling water system of the main engine the amount of heat to be transferred will be depending on the engine load. Other variables are amount and temperature of the cooling medium i.e. sea water. For the engine is important that the jacket cooling water temperature will stay within a certain range in order to avoid stresses in the materials.

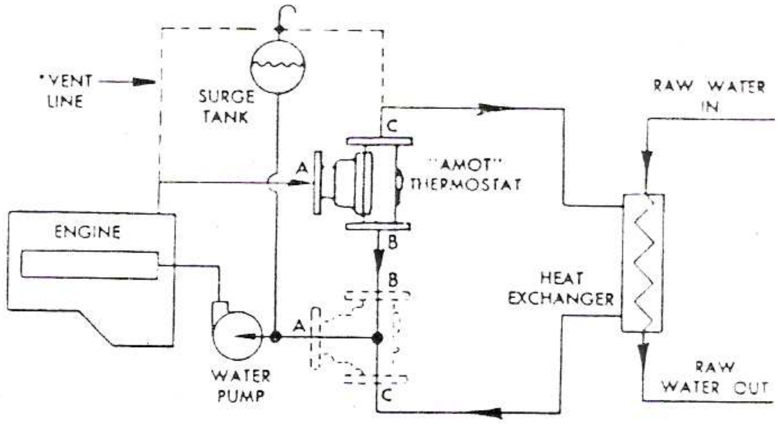


figure 9 Diesel engine cooling water system

In order to keep the temperature within the required range a thermostat is placed in the system. In figure 9 the thermostat is placed in the outlet cooling water of the engine. The thermostat maintains a constant outlet cooling water temperature by recirculation a part of the cooling water via outlet B to the by-pass of the cooler and the remaining water is going to the heat exchanger and will be cooled down. Before the water pump the two flows will mix again.
When the engine load increases the thermostat will react on that by sending more cooling water to the heat exchanger and reducing the flow through the by-pass. This will result in a stable temperature at the outlet of the engine.
When the thermostat is placed at the inlet of the cooling water system (dotted position in figure 9) it will maintain a constant inlet temperature.

The AMOT thermostatic valve (see figure 10)

The AMOT thermostatic valve is equipped with a number of thermostatic elements. The elements work in parallel operation. The liquid will enter the valve at opening A and flows along the sensing bulb which is filled with a hydrocarbon wax mixed with powdered copper, which remains in a semi-solid form and is highly sensitive to temperature changes.

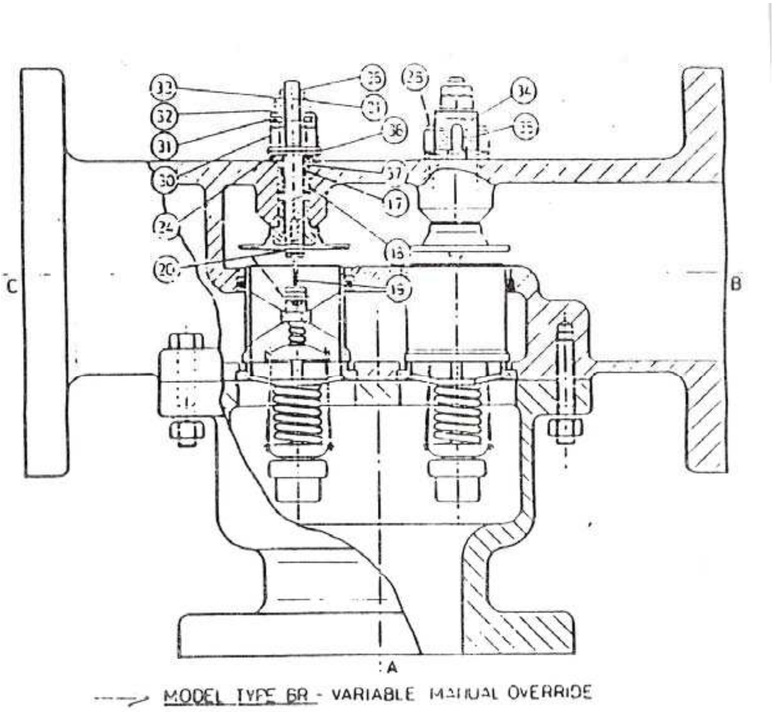


figure 10 AMOT thermostatic valve

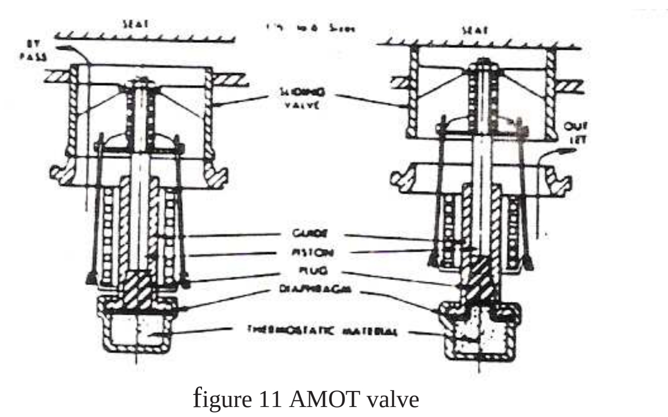


figure 11 AMOT valve

In the left drawing of figure 11 the sensing bulb is cold and the valve is not pushed up and the liquid can flow through the valve to the by-pass (connection B in drawing 10). In the right drawing the thermostatic material has expanded and it pushes the valve upwards opening the connection to the heat exchanger (connection C in drawing 10) and closing the opening to the by-pass.

In case a thermostatic valve is not functioning it is possible to manually override the valve. By lifting the lever (no 34) placed outside on the housing, the valve will be lifted and a permanent connection is made from the inlet (connection A) to the heat exchanger (connection C).

Questions heat exchangers

1. For what purpose heat exchangers are used?

2. What are the most common types of heat exchangers used on board?

3. Why is in a fresh water cooler the sea water flow through the pipes and not around them?

4. What are the functions of the baffle plates in a shell-tube heat exchanger?

5. What is the function of the cathodes placed in the sea water covers?

6. How is the difference in expansion between the tubes and the shell absorbed?

7. What is the function of the hole in between the 2 O rings at the sliding pipe plate of a shell-tube heat exchanger?

8. What are the two functions of the gaskets in a plate type heat exchanger?

9. Why are the plates corrugated in a plate type heat exchanger?

10. Which flow will give a better heat transfer: uniflow or counter flow? Explain your answer.

11. Following data is given of a cooler:

Mean temperature difference	$t_1 - t_2$	$= 28^0C$
Cooling surface	A	$= 20\ m^2$
Thermal conductivity	k	$= 1200\ W/m^2.K$

Determine the amount of heat transfer in the cooler.

12. Following data is given of a cooler:

Thermal conductivity	k	$= 1000\ W/m^2.K$
Mean temperature difference	$t_1 - t_2$	$= 35^0C$
Amount of heat to be transferred	Q	= 500 kW

Determine the cooling surface needed.

13. In a fuel pre-heater to the main engine heavy fuel oil is heated up from 50^0C to 70^0C by jacket cooling water of 80^0C, which will cool down to 60^0C.
Determine
a. Draw the temperatures development in the pre-heater
b. Must this heater be placed in uniflow or counter flow? Explain your answer.
c. Calculate the mean temperature difference

14. In a lubricating oil cooler 12 kg/sec oil is cooled down from 60^0C to 38^0C by sea water of 15^0C which will rise 5^0C in temperature. The specific heat of oil is 2.4 kJ/kg.K and of sea water is 4.3 kJ/kg.K
Determine the massflow sea water.

15. An air-conditioning system has a capacity of 24 kW. In the heat exchanger (evaporator) the refrigerant will cool down the air.

Evaporation temperature refrigerant	t_{evap}	= - 5^{0}C
Air temperature at entrance	t_{a1}	= 20^{0}C
Air temperature at outlet	t_{a2}	= 15^{0}C
Thermal conductivity	k	= 15 W/m^2.K

Determine
a. Massflow of air.
b. Required cooling surface. (calculation based on mean temperature difference)
c. Draw the temperature developments.

Steering Gears

The rudder

The rudder is the most common form of manoeuvring device fitted in ships. Modern rudders have the profile of a wing as shown in figure 1. A flow pattern will occur around the rudder when an angle φ is given to the rudder. On the side to which the rudder is turned the water flow is compressed creating a pressure on the rudder surface. Water flowing to the other side of the rudder can expand resulting in a reduction of pressure. The result is a pressure difference between the two sides of the rudder creating the rudder force F_r.
The rudder force can be divided in a component D, which is the force turning the vessel and a component W, which is the force working on the rudder as result of the water flow (resistance).
Z is the pressure point of the wet surface area in which all the forces working on the rudder are concentrated. The position of the pressure point Z is not fixed but depends on the rudder angle φ.

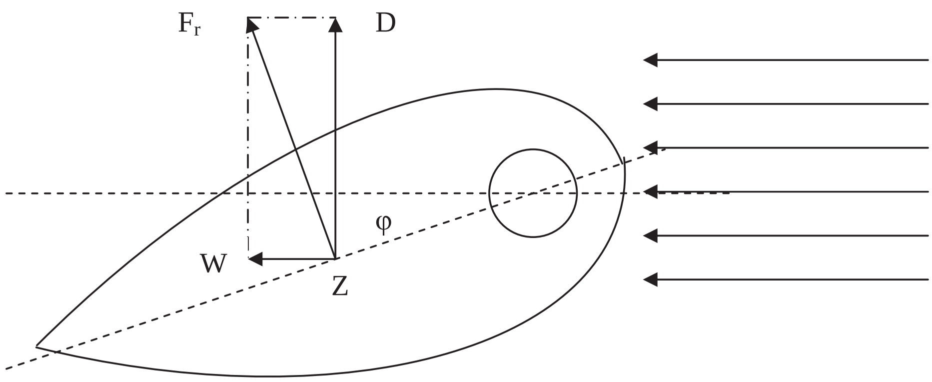


figure 1

The torque working on the rudder depends on various factors. The major factors are:

- Speed of the vessel
- Distance of the pressure point Z to the centre of the rudder stock (arm)
- Wet surface area of the rudder

When the rudder angle φ reaches up to 20 to 25 degrees the water on the low pressure side will no longer flow laminar but will become turbulent. This turbulence will increase the pressure resulting in a lower rudder force F_r and a lower rudder effect. It is therefore not advisable to give big rudder angles. This can be seen in figure 2.

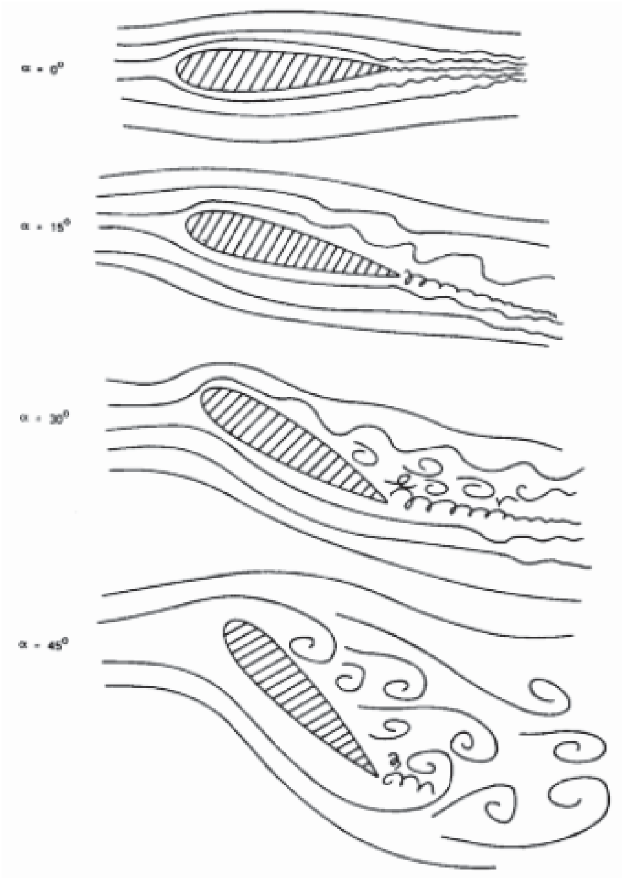


figure 2

Rudder types

Rudders can be categorized according to the degree of balance. Rudders are termed as:

1. Balanced
2. Semi-balanced
3. Unbalanced

Balanced rudders

Balanced rudders can be divided in:

1. Simplex
2. Spade

Figure 3 shows a Simplex rudder. From the picture it can be seen that the centre line of the rudderstock and the turning point (pintles) of the rudder are positioned in the same line. A part of the wet surface area of the rudder is positioned forward of the centre line. The position of the pressure point Z will therefore be close to the centre line resulting in a small arm and consequently a small torque. At small rudder angles it might even happen that the pressure point Z is positioned in front of the centre line. In that case the steering gear does not have to create a torque to turn the rudder but must prevent the rudder from turning more.

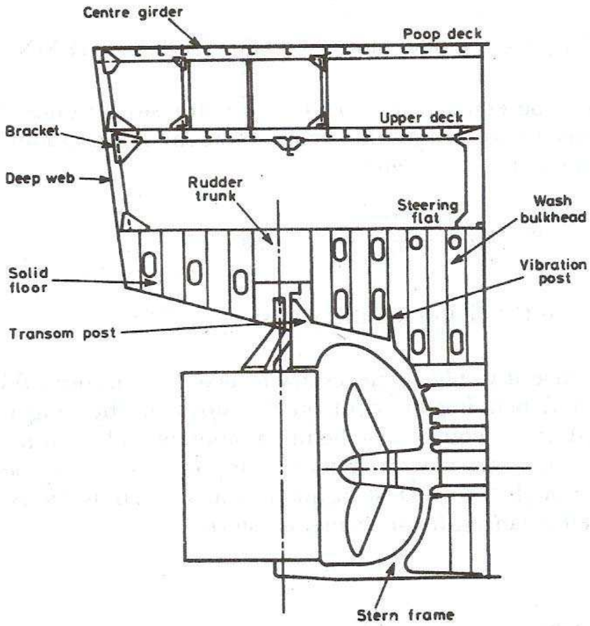
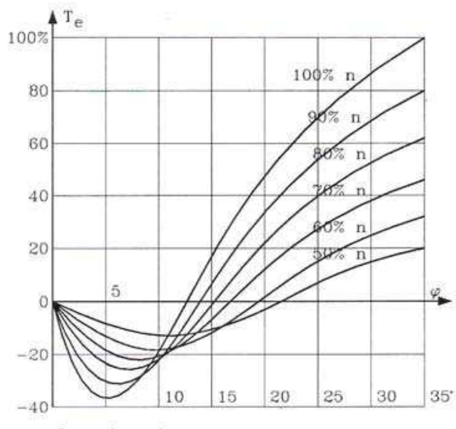


figure 3



In figure 4 graphs are drawn from a balanced rudder. The vertical axis is showing the torque (T_e) and the horizontal axis the rudder angle φ. The six graphs show the development of the torque at different revolutions (n) of the propeller shaft creating different ship speeds.

It can be seen that at smaller rudder angles the torque is negative and that the rudder angle at which the torque becomes positive is getting smaller when the speed increases. During normal manoeuvring the revolutions of the propeller shaft are reduced allowing the steering gear to operate at low torques.

figure 4

Figure 5 shows a spade rudder. This type of rudder is free hanging on the hull and has no stern frame to support it. This type of rudder can mostly be found on warships.

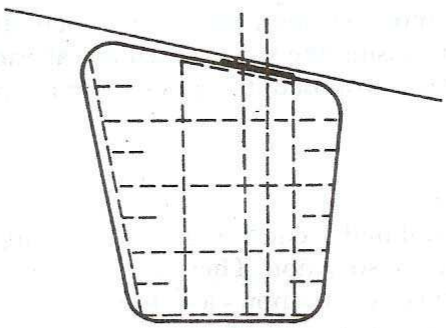


figure 5

Semi-balanced rudder

Figure 6 shows a semi balanced rudder. Another name for this rudder is Mariner rudder. The stern frame is integrated into the ship's construction and the rudder can rotate around the pintles in the frame. The bottom half of the rudder is partly extending forward of the centre line of the rudder stock. As with the balanced rudder this will help to reduce the torque needed to turn the rudder. However as this area is not extending over the full height of the rudder is this rudder type called semi-balanced.
The semi balanced type rudder is used on large ships like container ships, bulk carriers and tankers.

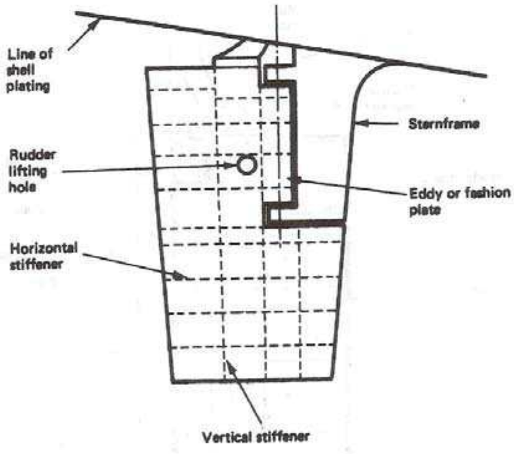


figure 6

Unbalanced rudder

Figure 7 shows an unbalanced rudder. The pintles are located at the forward side of the rudder and no wet surface area is found forward of the centre line of the rudder stock. The pressure point Z will always be located aft of the centre line resulting in higher torques to turn the rudder.

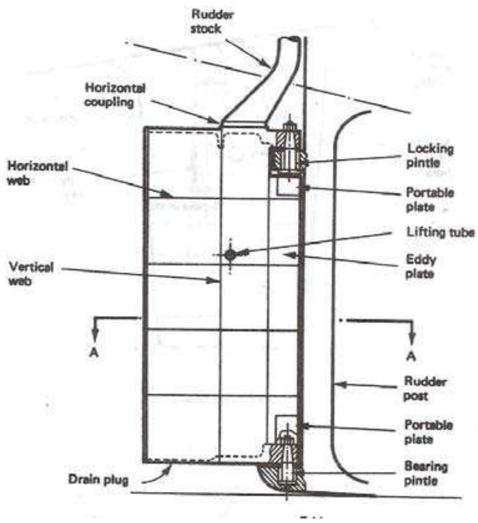


figure 7

Figure 8 shows the development of the torques of an unbalanced (a Oertz – roer) and a balanced (b balansroer). It shows that to reach the same rudder angle the unbalanced rudder must create a higher torque. The steering gear unit for semi-balanced and balanced rudders can therefore be smaller which is more economical.

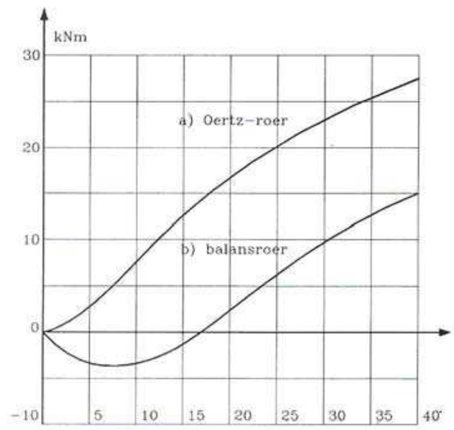


figure 8

Flap rudder

The flap rudder has a hinged flap at the back of the rudder blade. This flap is moved mechanically by the flap guide on top of the rudder in such a way that the flap's turning angle is twice as large as the turning angle of the main rudder blade. When the maximum rudder angle is 45^o the flap has a maximum angle of 90^o with respect to the ship. In this position it is possible that 40% of the ship's propulsion force is directed sideways.
Advantages of flap rudders are:

- Extra manoeuvrability
- Course corrections can be performed with smaller rudder angles. This means that the ship loses less speed and therefore consumes less fuel.

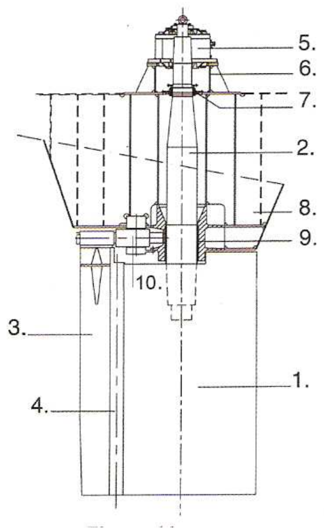


figure 9

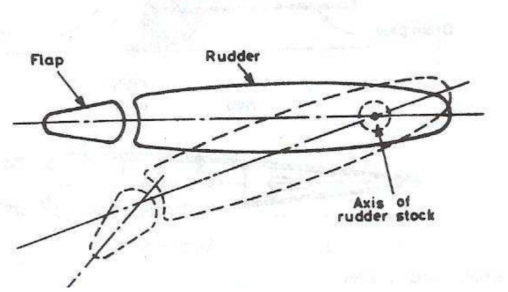


figure 10

1. Rudder blade
2. Rudder stock
3. Flap
4. Hinge line
5. Steering engine
6. Steering engine foundation
7. Gland and bearing
8. Rudder dome
9. Bearing
10. Flap actuator

Steering gears

The function of the steering gear is to turn the rudder as required by the officer on the bridge in order to alter course. The signal given from the bridge can come from the helm in case of hand steering or from the automatic pilot. When the helm is used the helmsman will turn the wheel to a certain required rudder angle. When the automatic pilot is used the officer of the watch has set a course and the auto pilot will compare this course with the course as indicated by the gyro compass. In case of offset the autopilot will send a signal to the steering gear to compensate the offset by turning the rudder to a certain angle. In both modes of steering it is necessary that there is a feedback signal, indicating the actual position of the rudder, coming from the steering gear. This feedback signal will be compared with the required (set point) rudder angle and in case both signals are identical the rudder will stop turning.

Telemotor

Figure 11 shows the telemotor system. The system comprises of a telemotor transmitter placed on the bridge and a telemotor receiver placed at the steering gear. The transmitter and the receiver are connected by two copper tubes. The oil used is either glycerine or oil which has a low viscosity at low temperatures.

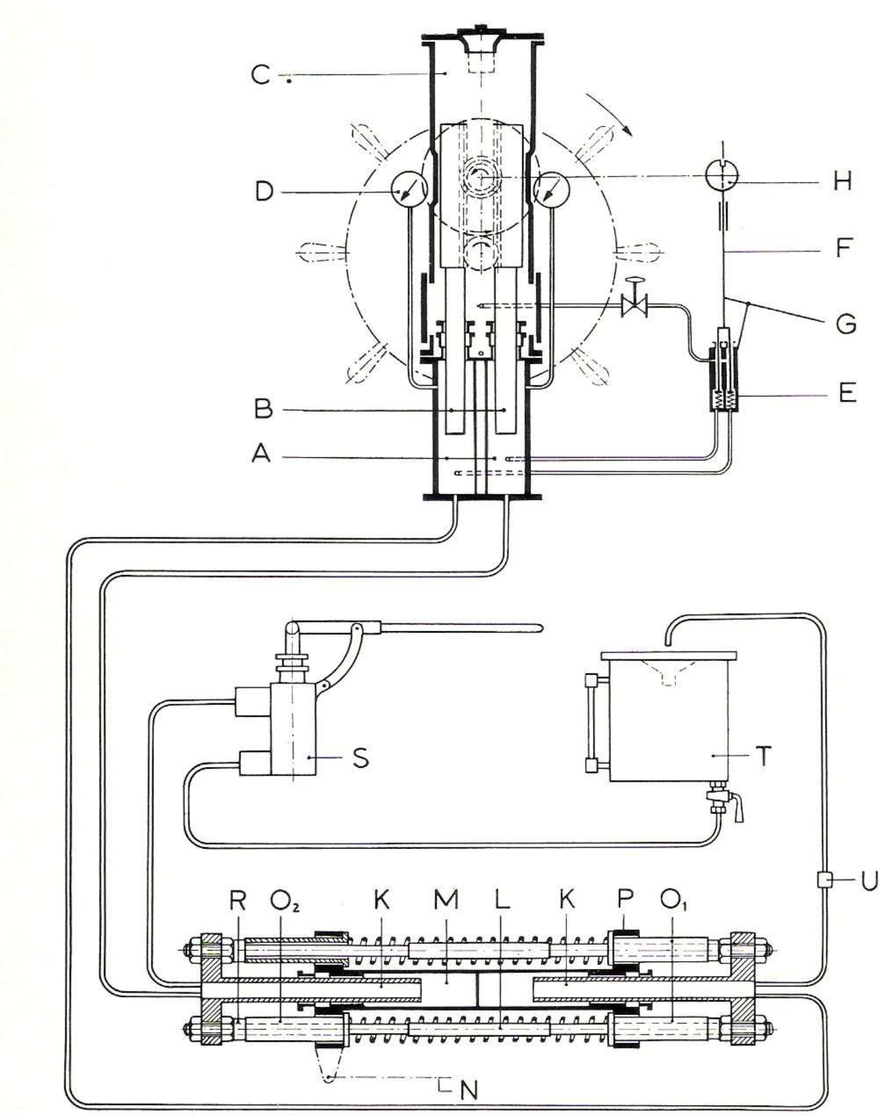


figure 11

The transmitter has two cylinders A in which two plungers B can move up and down. The upper part of the plungers are geared and in between a gearwheel is located which can be turned by the steering wheel. When the wheel is turned one plunger will move upwards and at the same time the other one will move down. The plungers are moving up and down inside the oil tank C. Tank C is equipped with a sight glass and a drain valve in order to clean the tank without shutting down the system. Each of the cylinders A is equipped with a pressure gauge in order to check the pressure in the lines. If the pressure difference will reach over 3.5 bar, when the telemotor is in middle position, (i.e. caused by change of temperature) this can be compensated by opening valve E. Valve E is a combination of two valves, for each cylinder one, which are connected to tank C. The valves can be opened simultaneously by rod F and lever G. Rod F is extended above disk H which is connected with the gear wheel. In disk H there is at one position a hole. When the rudder is placed in the centre position this hole will be positioned above the rod and that is the only time the valves E can be opened. By opening the valves E the difference between the real rudder position and the indicated rudder position will be equalized.

The telemotor receiver consists of two hollow plungers K. They have a fixed position and are connected to each other by the rods L. The plungers are moving inside cylinder M which is divided in two equal sections. Cylinder M can move and is via N connected to the control valve of the steering gear.

When the steering wheel is turned to the right the left plunger B will move down and the right plunger will move up. Oil is pressed to the right side of cylinder M and removed from the left side. Cylinder M will move to the left and will via N move the control valve of the steering gear. The bushings O_1, which can slide over rods L, are moved to the right by the ears P of cylinder M and compressing the springs, because on the other side the bushings O_2 cannot move as they are stopped by the nuts R. The steering will now start to operate and the rudder will be turned. The turning of the rudder will move the control valve back to its middle position and the steering gear will stop turning. Transmitter, receiver and rudder are now placed off centre and the control valve is in its middle position.
When the helmsman releases the steering wheel the springs will push back cylinder M in the middle position. The plungers in the transmitter will also move back to the middle position. The control valve on the steering gear has been moved in opposite direction. The rudder will be turned back into its middle position.

To charge the system with oil we use hand pump S. This pump suctions oil from tank T. During charging the vent cocks located at the gauges and the valves E must be opened. The system is full as soon as the oil level in tank C starts to rise. After that the vent cocks at the telemotor receiver must be opened and continue pumping. As long as oil is flowing out with a constant flow it means that there is still air in the system. (the air works like a cushion) When only oil flows out during a pump stroke it means that all the air is removed. As tank T is placed below the telemotor receiver a spring loaded valve U is placed in the bleed line to avoid the receiver will be drained.

Figure 12 shows the lay out of a two ram steering gear unit. At the top of the rudderstock the tiller F is fitted. The tiller can be moved sideways via the yoke C by the rams B which are moving inside the cylinders A. To absorb the large forces which are working on the cylinders, these are placed on a heavy foundation E. This is to avoid that misalignment might occur. The foundation also absorbs the side forces via slippers D.
The cylinders are connected via hydraulic piping with Hele-Shaw pump G which is driven by electric motor R.
The amount and direction of the oil flow depends on the direction in which control rod H is moved.

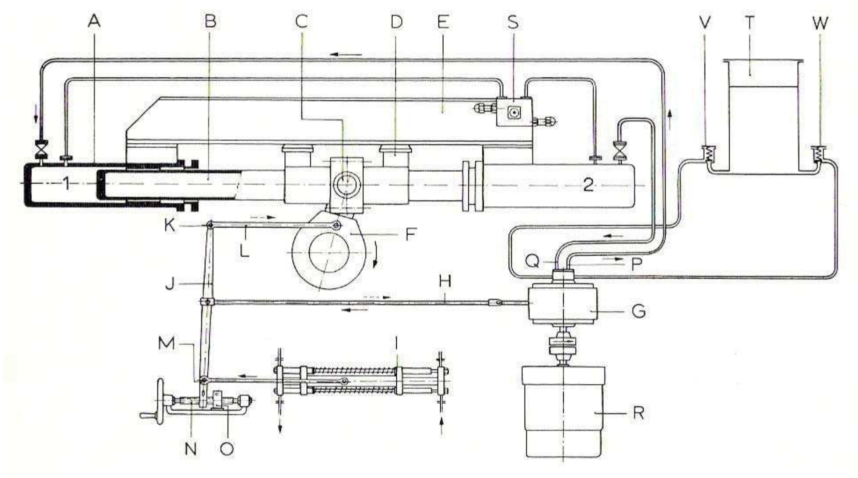


figure 12

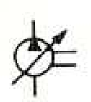
From the bridge a signal is given to the telemotor receiver resulting in for instance a movement of point M to the left. In first instance the floating lever J will turn around point K as the steering gear has not moved yet. As a result of this control rod H is also moved to the left changing the position of the cylinder body of the pump. The pump will now start to deliver oil to cylinder 1 and oil will flow from cylinder 2 back to the pump. The rams will move to the right and give a clockwise turn to the rudderstock. The floating lever J is connected to the tiller via rod L. As this rod is now moving to the right the floating lever J will start to turn using point M as the turning point resulting in moving control rod H to the right. At the time the cylinder body of the pump is pushed back into its middle position the pump will stop delivering oil and the rudder will stay in its new position.

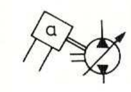
In case of failure of the telemotor system the steering gear can be operated manually using the hand wheel of device N. For manual control floating lever J must be disconnected from the telemotor receiver at point M and reconnected to nut O which can be moved along the threaded spindle of N by turning the hand wheel.
Electrical-Hydraulically 2 ram steering gear

On modern vessels the signals from the bridge to the steering gear are normally electrical signals. The signals are used to operate a hydraulically valve.
2 Ram steering gear machinery can be used on vessels up to 15,000 GT, 4 rams are used on higher tonnage.

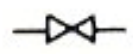
Symbols used in the diagrams:

 Pump unit with constant flow in one direction (direction of the arrow)

 Pump unit with variable flow in one direction

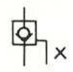


Pump unit with reversible variable flow direction. Reversing is done via pilot unit

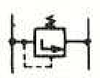


Valve

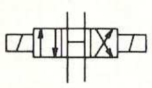
Non return valve



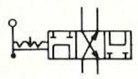
Controlled non return valve

Relieve  valve

Filter



Control valve with 4 connections and 3 possible positions. De position of the valve can be changed by the electrical coils which are placed on both sides of the valve.
The arrows show the flow directions inside the valve.
A common name for this valve is the 4/3 way valve (4 connections, 3 positions)



Manual controlled 4/3 way valve with locking device for each position.

Figure 13 shows the diagram of a two ram steering gear unit. The installation consists of two identical hydraulic systems. Pump 1 is an axial plunger pump with constant flow driven by electric motor 2. This kind of pumps will deliver a pressure up to 250 bars. With each pump there is valve block 3 containing several parts.

To explain the operation of the system we look at the right system. When the 4/3 way valve 7 is in centre position the oil will circulate from the pump through valve 7 and back to the pump. When an electrical signal is given from the bridge to the right coil of valve 7 the valve will move to the right. High pressure oil will start to flow to the left non return valve 4 and the oil will flow to the left ram. The controlled non return valve 4 on the right side is opened by the high pressure oil coming from the left side. The right ram will no be connected to the suction side of the pump. The rudder will start to turn counter clockwise as the rams are moving to the right. The movement will continue until no more signals are given from the bridge and valve 7 will return to its centre position. The rudder will remain in its position as both non return valves will close and the oil flow is stopped.

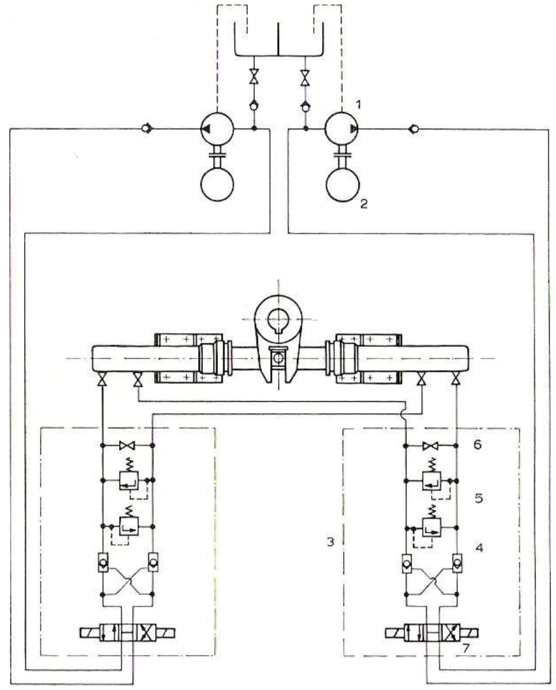


figure 13

In case a large outside force will suddenly work on the rudder, for instance if the vessel is heavily pitching and rolling in bad weather, the oil pressure in the system might reach high values. In order to protect the system against high pressures one of the relieve valves 5 (depending on which side the force will work) will open and oil will flow to the other side of the system.
The consequence is that the rudder will turn a little.

By opening valve 6 the two rams are directly connected with each other. This valve will be opened during dry-docking when repairs have to be carried out on the rudder. It is than possible to turn the rudder from outside without using the steering gear unit.

In case of manoeuvring during docking and undocking it is normal practise to run both pumps 1 simultaneously. The two systems are running parallel which means a double amount of oil will flow to and from the rams, giving a faster movement of the rudder.

Operating modes of steering

On the bridge there are three possible ways to operate the steering gear

- Emergency hand steering
- Auto pilot steering
- Hand steering

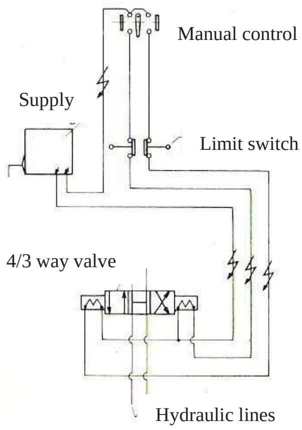


figure 14

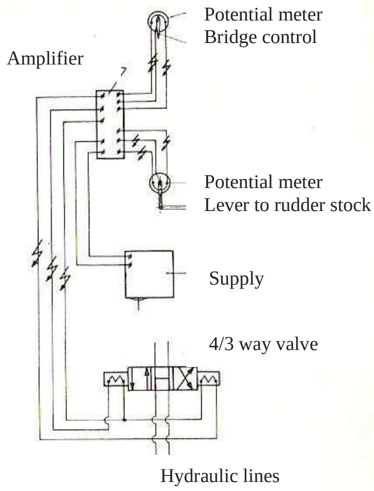


figure 15

Of the three steering modes the first one is a non follow-up system as shown in figure 14 and the last two are a follow-up system as shown in figure 15.

Non follow-up system

By moving the manual control on the bridge to left or right a direct signal is given to the 4/3 way valve of the steering gear unit. The rudder will move to either port or starboard as long as the manual control is pushed. The movement of the rudder can be followed on the bridge by looking to the rudder indicator. When the required rudder angle has been achieved the control is put in the middle position and the rudder will stop moving.
The limit switches are placed on the steering gear unit. In case the unit has been moved to one of its maximum rudder angles the switch will open and the signal to the 4/3 way valve is stopped.

Follow-up system

During hand steering the helmsman will move the steering wheel to a certain number of degree port or starboard. By turning the wheel the tiller of a variable resistor (potential meter) is moved creating an electrical signal. This signal is strengthen in the amplifier and will move the 4/3 way valve in the required direction. The steering gear will start to operate and the rudder starts turning. The rudderstock is connected via a lever to the tiller of an identical potential meter. As soon as the rudder has turned the required number of degree the position of the second potential meter is in balance with the one on the bridge and the total signal to the amplifier will be 0 causing the 4/3 way valve to come back to its middle position and the rudder will stop moving.

When the vessel is at sea normally the auto pilot is used. In that case the set course of the vessel is compared with the real course of the vessel as indicated by a gyro repeater. In case there is a difference the auto pilot will send a signal to the steering gear to compensate this difference.

On the auto pilot a number of adjustment buttons can be found which are used to optimize the rudder control in various situations.
The buttons are:

- R – Rudder: This is the amplifying factor of the controller. How many degrees rudder for each degree course error
- C – Counter rudder: This is the derivative action of the controller. When the rudder is turned to bring the vessel back on her course there is a possibility that she might cross the course line on the other side. To avoid this with counter rudder the controller can react on the rate of turn.
- Y – Yawing: This is the allowance set how many degree the vessel may go off before the controller will react. This can be called the dead-band.
- P – Permanent helm: This is the integrating factor of the controller. This can be used to give the rudder a preset degree of rudder angle to compensate wind from abeam.

Figure 16 shows the total principle diagram of a steering gear controlled by an auto pilot.

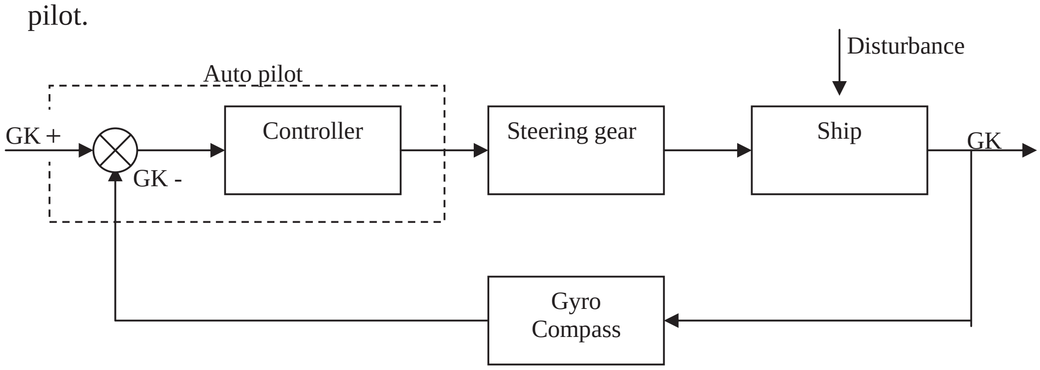


figure 16

The auto pilot consists of a PID controller. In the balance point of the controller the set point, which is the ground course (GK+) is compared with the measured value, which is the actual ground course (GK-). As soon as there is a difference between the two values the controller will send an output signal to the steering gear. The output signal acts as the input signal of the steering gear (compare with telemotor system) which will activate the steering gear. As the rudder will turn the ground course of the vessel will change. The gyro compass measures the ground course of the ship and this signal is used as the measured value in the controller. As soon as GK+ and GK- are equal the output signal of the controller will become 0.
The reasons why there might be an off-set between GK+ and GK- are:

- Change of course set by the officer of the watch
- Disturbance working on the ship, like waves, wind or current.

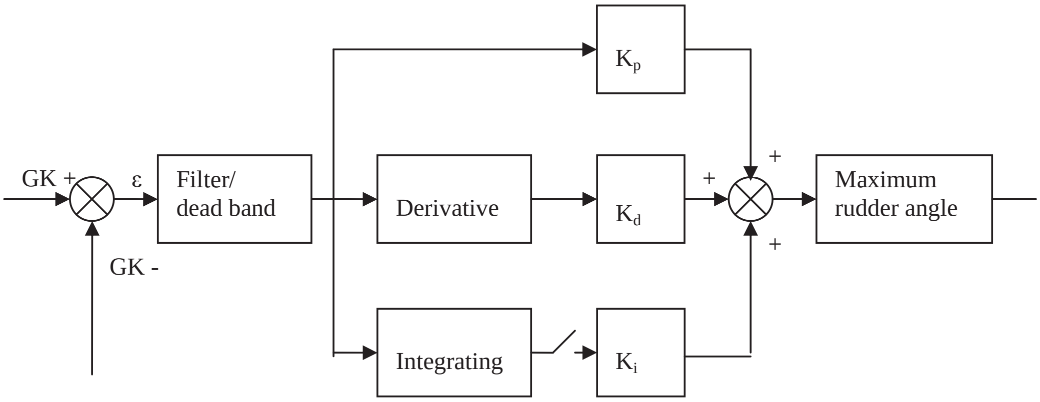


Figure 17

Figure 17 shows the inside of the controller of the auto pilot. The difference between the set point: ground course GK+ and the real ground course GK-, measured by the gyro compass is the input signal ε of the controller. This signal passes a filter or dead band to absorb the yawing of the vessel. In bad weather the dead band can be increased with the knob “Weather” on the control panel, in order to keep the number of signals to the steering gear low.

After passing the filter the signal is send to:

1. A proportional circuit of which the transfer ratio or amplifying factor K_p can be adjusted by a knob on the control panel, mostly indicated by “Rudder”.
2. A derivative circuit of which the transfer ratio K_d can be adjusted with the knob “Counter rudder”.
3. An integrating circuit which can be switched on if needed. The transfer ration K_i can be adjusted with the knob “Permanent helm”.

The three signals are combined and the resultant will pass the maximum rudder angle controller. The maximum rudder angle can be set by the knob “Rudder limit”. Sometimes it is necessary to limit the maximum rudder angle, i.e. if the initial static stability of a vessel is low. (The centre of gravity is high above the keel). When in such a situation the rudder would give a large moment over the longitudinal axis of the ship the high position of the centre of gravity will cause the vessel, to list. This list can cause shifting of cargo or create high tension in the lashings.

The use of the rudder must be limited till the minimum. Unnecessary rudder movements will have negative influence on the speed of the vessel, resulting in higher fuel consumption.

Emergency steering

In case the connection between the bridge and the steering gear room is broken the steering has to take place inside the steering gear room. By international regulations there must still be a possibility to operate the steering gear from the steering gear room. On vessels equipped with an electrically-hydraulically operated steering gear using 4/3 way valves there is a possibility to operate the valve manually. On both sides of the valve there is a small hole. By using a pin or handle the valve can be pushed manually to either left or right. Figure 18 shows this operation.

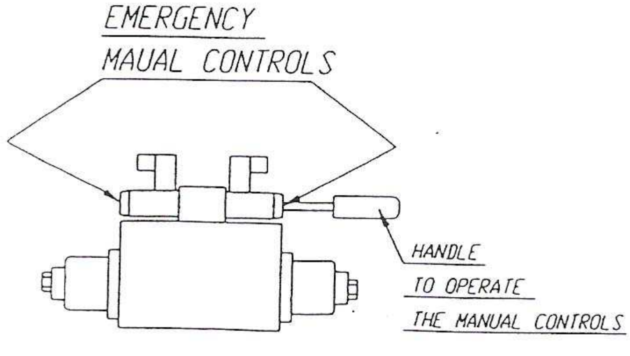


figure 18

By law there is a requirement for a direct telephone connection between the bridge and the steering gear room. The bridge can inform the person in the steering gear room to what side the steering gear has to be operated. On vessels of 1,600 GT and above it is even a requirement that there is a gyro compass repeater in the steering gear room. By looking at the repeater it is possible to keep the vessel on the required course.

Mandatory requirements steering gears

The IMO has issued rules and regulations for the lay-out and operation of steering gears. Some of them are listed below:

- Each vessel must be equipped with a main and an auxiliary steering gear or:
- A steering gear which has a complete redundant operating system. (There is a difference between this regulation for passenger vessels and other type of vessels).
- The main steering gear must be able to operate at the highest possible speed of the vessel.
- The correct position of the rudder must be indicated at each position the rudder can be operated.
- The electrical supply must be consist of two cables of which each cable must be able to take up the maximum load of the total steering gear unit.
- The electrical cables must be separated from each other as much as possible while running from the main switch board to the steering gear room.
- The steering gear must be capable to move the rudder from 35^o at one side to 30^o on the other side in maximum 28 seconds

For tankers of 10.000 Tons or more additional requirements are applicable.

Operational requirements:

- In case automatic steering has been used for a long period and the vessel has to manoeuvre the hand steering must be tested well in time.
- During manoeuvring more as one operating system must be switched on.
- Within 12 hours before departure the following has to be tested:
- Main and auxiliary steering gear system.
- Emergency supply.
- Position of the rudder indicator on the bridge with the real position of the rudder.
- All alarms of the steering gear system.
- The phone connection between bridge and steering gear room.
- During testing the rudder must be turned over its maximum angle and the total installation must be checked.
- Both on the bridge and the steering gear room diagrams and operational procedures must be posted.
- All officers must be knowledgeable in operating and changing over systems.
- The emergency steering must be tested at least every three months.
- All checks and tests must be recorded in the deck and engine logbooks.

Other types of steering gear

Figure 19 shows a diagram of a 4 ram steering gear unit. In this case the pumps are axial plunger pumps with variable flow direction. The flow rate and direction are controlled by servo cylinder 3. The servo cylinder is part of a separate hydraulic system, which contain 4/3 way valve 4, safety valve 5 and a small axial servo-plunger pump 2. This pump is fitted to the same shaft as pump 1.
The discharge side of the servo pump 2 can be connected to both sides of the servo cylinder 3 depending on the position of 4/3 way valve 4. When valve 4 is in its centre position the oil will circulate from tank 6 via line a, pump 2, safety valve 5 and line b back in to tank 6.

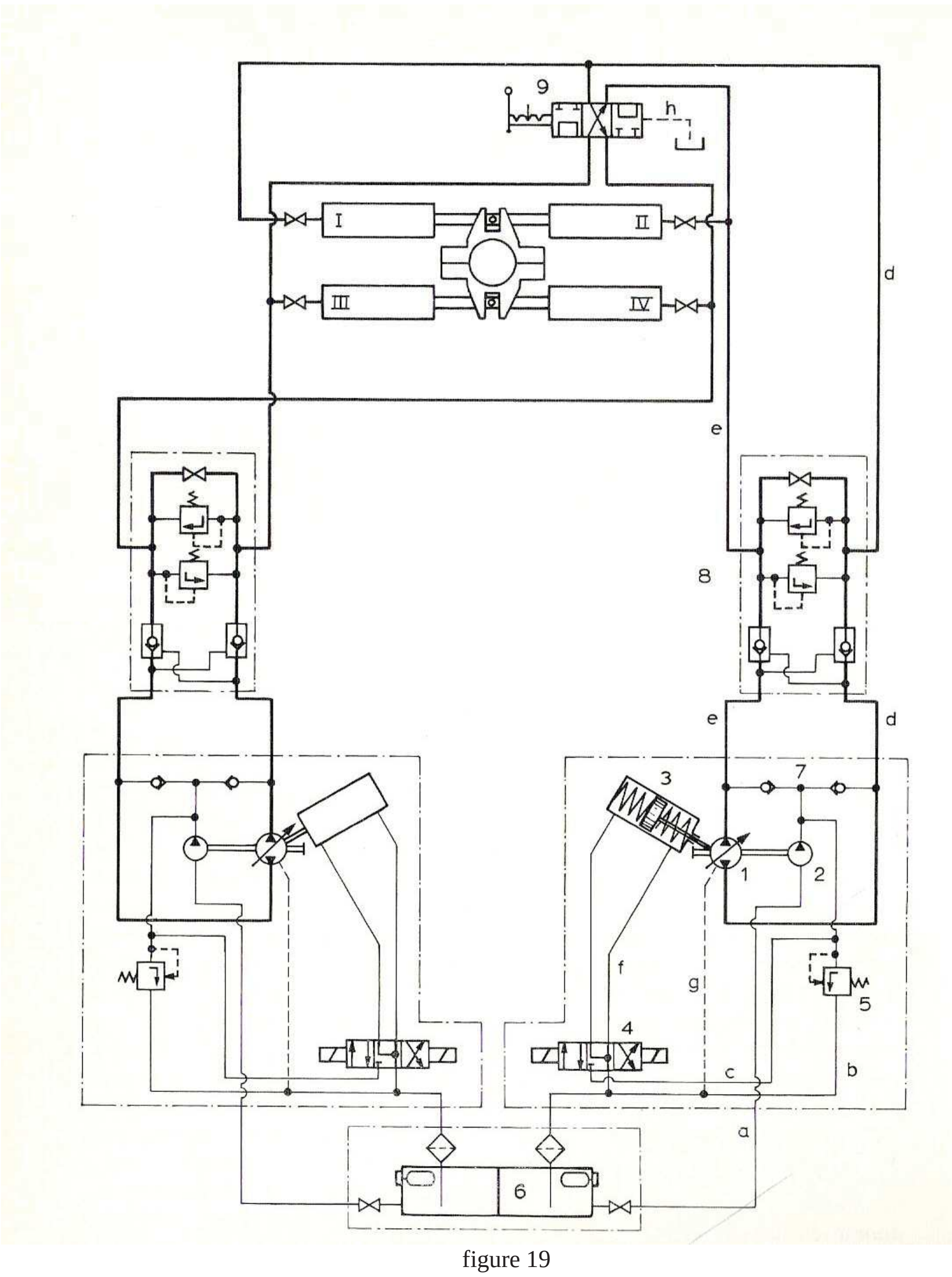


figure 19

When valve 4 is energized to one side oil will flow via line c to one of the sides of the piston of servo cylinder 3. This will cause pump 1 to deliver oil to the rams. When valve 9 is placed in its middle position all 4 rams are in operation.
Suppose the right coil of valve 4 is energized. This will cause the valve to move to the right. The delivery side of pump 2 is now in connection with the left side of servo cylinder 3. Valve 5 will close until the piston of the servo cylinder has reached its maximum position and the oil pressure is back to the set point of the safety valve. The oil on the right side of the servo cylinder 3 can flow back to the tank via line f.
Pump 2 will start to deliver oil to line d. Via the right controlled non return valve of unit 8 oil will flow to rams I and IV. The left controlled non return valve will be opened by the high pressure oil in line d allowing oil from rams II and III to flow back to the suction side of the pump via line e.
In case of damage to the hydraulic lines or one of the rams valve 9 can be placed in a position that either rams I and II or III and IV are connected with each other and will be out of service. The function of the other components is the same as in the 2 ram type steering gear.
Lines g and h are for collection of leakage oil.

Rotary vane steering gear

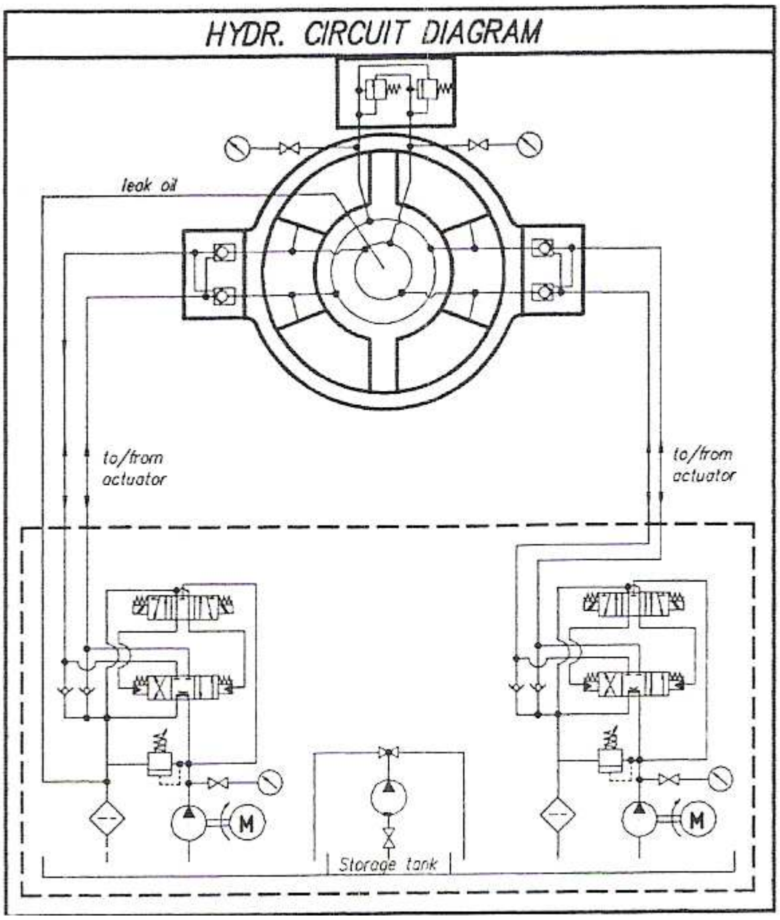


figure 20

Figure 20 shows the diagram of a rotary vane steering gear.
The system is composed of one hydraulic rotary vane actuator mounted directly on the rudder stock, served by two pump units delivering the necessary oil pressure for operating the rudder. The two pump units may be operated together or separately.

Operation of the unit.

Figure 21 shows the two 4/3 way valves of one unit. In this figure the pilot valve 1 is in its middle position. Oil is flowing from the pump via the control valve 3 back to the suction side of the pump. There is no oil flowing to the actuator.

1 Pilot valve
2 Safety relieve valve
3 Control valve
4 Non return valve

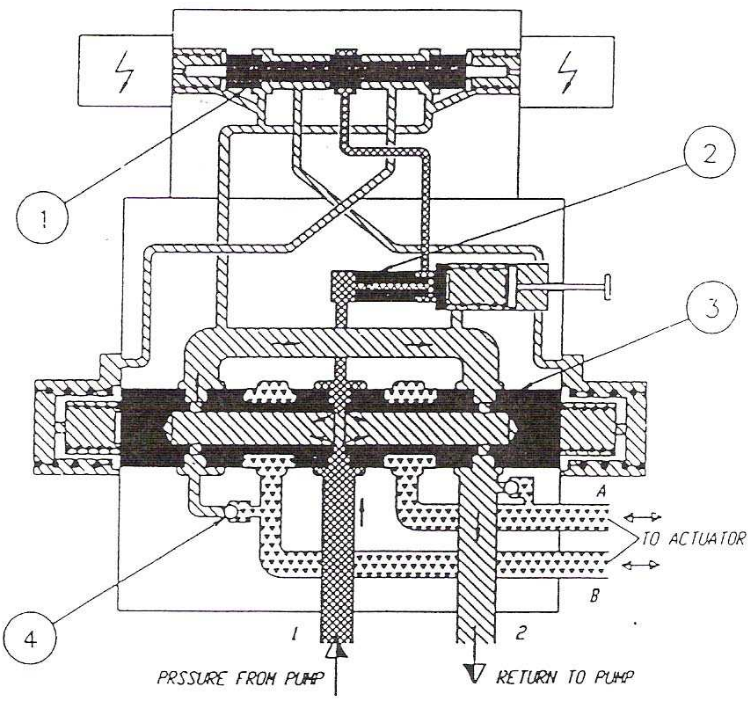


figure 21

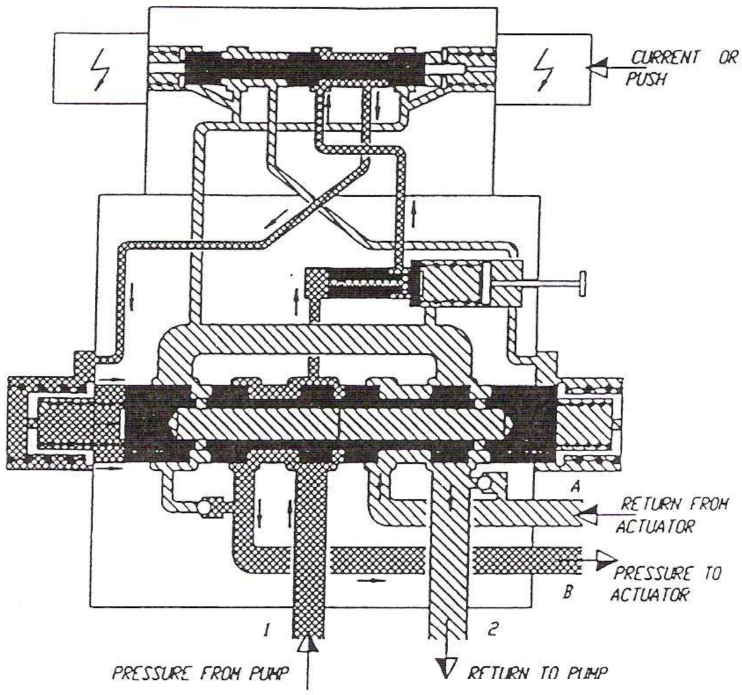


figure 22

Figure 22 shows the situation that the coil of pilot valve 1 gets an electrical signal from the bridge moving the valve to the left. The two control valve spindles 3 are now forced against each other and held in the right hand position by the oil pressure in the left chamber.

The oil flow from the pump has now free passage from channel (1) into pipe (B) leading to the actuator. The return oil from the actuator flows through pipe (A) and has free passage to channel (2) and back to the suction side of the pump.

When the oil pressure exceeds the preset value of the safety relieve valve 2 the valve will open allowing the high pressure oil to flow to the return line. The pressure in the control valve end chamber will drop and the spool will move to centre position.

The actuator consists of three main components: A cylindrical housing with stoppers, a rotor and a bolted-on cover.

The rotor, turning in bearings at top and bottom, is equipped with vanes upon which the oil pressure is acting and creating the turning torque. The turning movement is limited by stoppers fixed in the housing.

Figure 23 shows the top view of the actuator and figure 24 shows the cross section. The centre of the unit is the rudder stock on which the vanes are mounted. The vanes are almost able to make an 180^0 turn inside the housing.

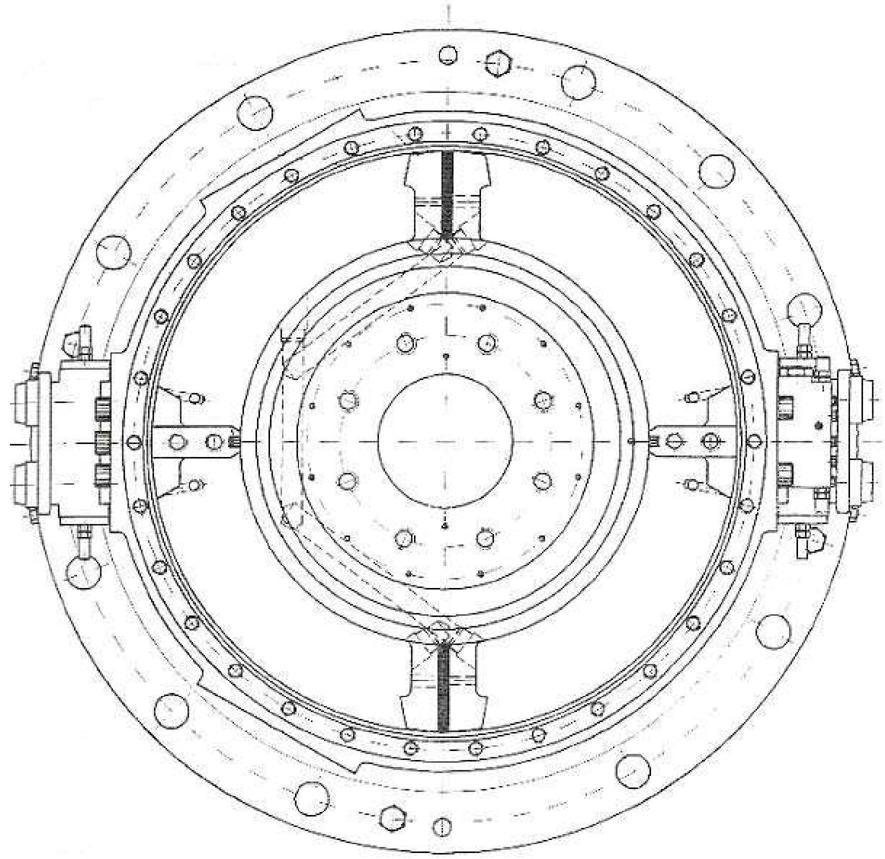


figure 23

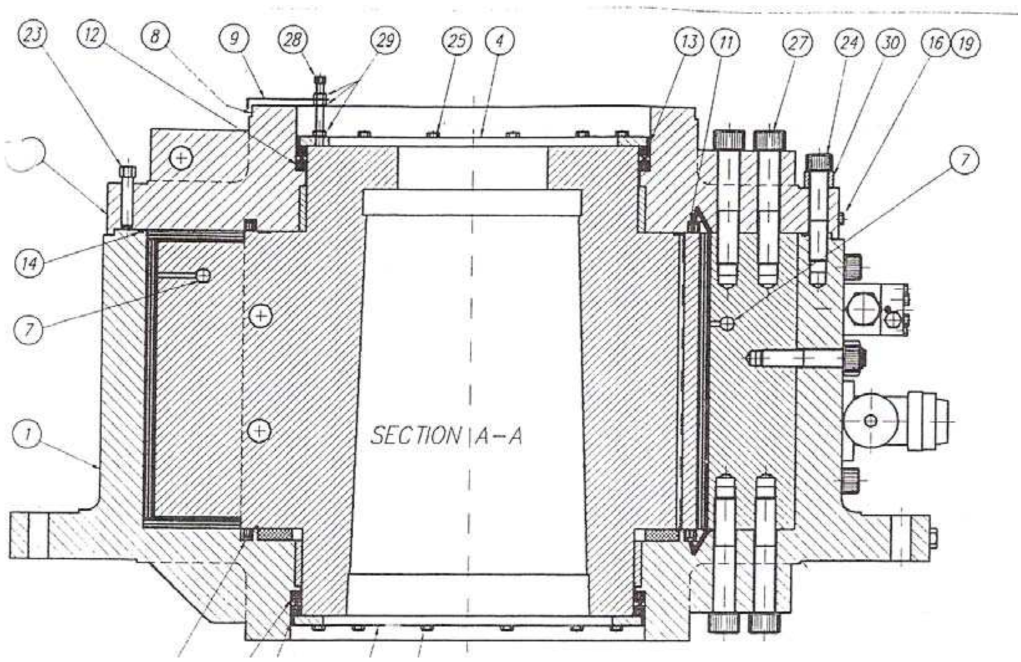


figure 24

In figure 19 can be seen that the actuator is equipped with two safety valves in case the oil pressure becomes too high. There also non return valves which will close when there is no oil flow to the actuator and keeping the actuator and the rudder in their position.

Two cylinder steering gear unit

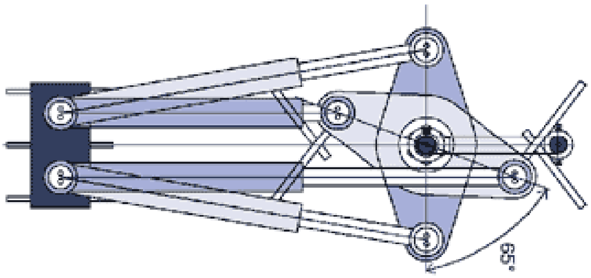


Figure 25

This type of steering gear unit is mostly used on smaller type of vessels. Around the rudder stock a tiller is fitted. At the outer ends of the tiller hydraulic cylinders are mounted with double acting pistons inside. The cylinders are mounted to the foundation using ball type joints at their outer end because the cylinders will move when the rudder is turning. Therefore the hydraulic connections to the cylinders consist of hydraulic hoses.

In normal circumstances both cylinders are in operation. In case of damage to one of the cylinders the steering gear will still be able to operate on one cylinder. The two sides of the damaged cylinder must be connected to each other using the by pass valve in order to avoid pressure building up in the cylinder.

Hydraulic pumps for steering gears

Hele-Shaw pump

Figure 26 shows a cross section of this pump. The pump consists of a number of plungers which are placed in a star configuration. The plungers are connected to the outer stationary ring via sliding shoes. The inner pump housing is driven by an electric motor and will rotate. The sliding shoe connection makes it possible that the plungers can rotate with the pump housing. The suction and discharge lines are connected to the stationary centre of the pump. This area is divided in two spaces.
The stationary ring can be moved to left or right.
As the plungers are facing the centre of the pump this pump is called a radial plunger pump.

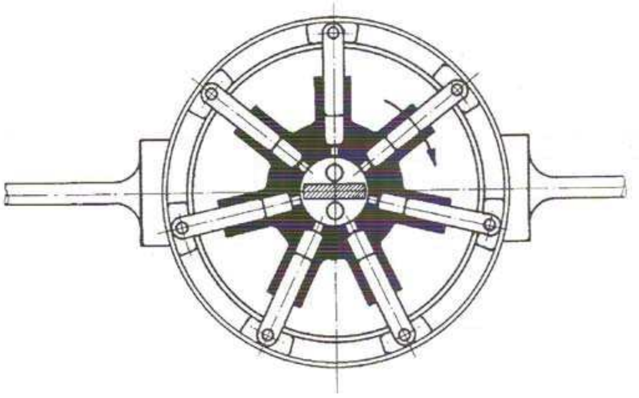


figure 26

In figure 27 the ring has moved to the right. The centre of the pump is now concentric with the ring. At the top side of the pump the plungers are moving outward creating the volume in the cylinders to increase which creates suction. The plungers at the bottom side move inward and are now delivering oil.

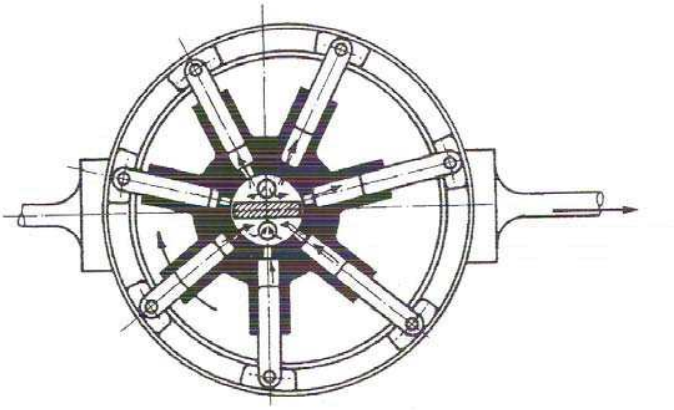


figure 27

When the ring is moved to the left the action of the plungers will be reversed (Figure 28). The top side becomes the discharge side and the bottom side becomes the suction side. This shows that the Hele-Shaw pump is a pump with variable flow control and variable flow direction.

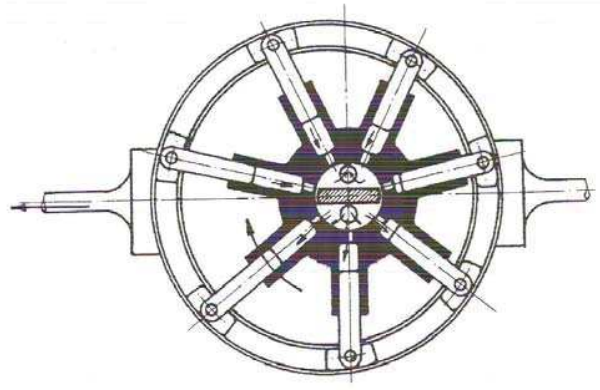


figure 28

Swash-Plate type pump

Figure 29 shows a Swash-Plate type pump. In this pump the plungers are connected to a swash plate which is driven by an electric motor. The plungers are placed under an angle with the swash plate. Both the plungers and the cylinder block rotate. A plunger which is rotating from the top side of the pump to the bottom side will move inward creating a discharge of oil. The plunger moving from the bottom to the top will move outward creating suction.
As the plungers are placed in (almost) the same direction as the driving shaft this pump is called an axial plunger pump.
The pump in figure 29 has a fixed capacity and flow direction.

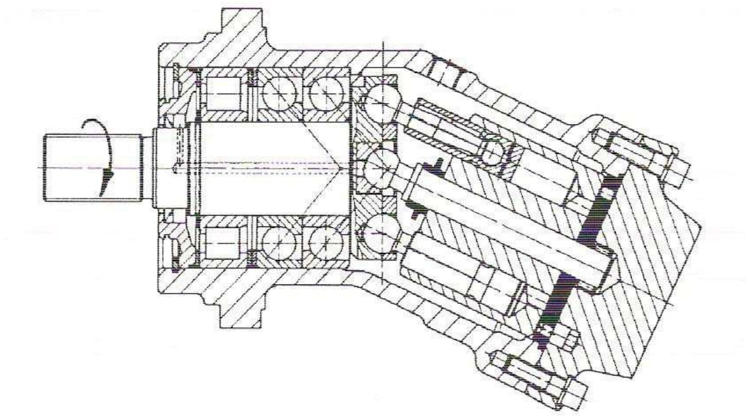


figure 29

Figure 30 shows a similar type of pump but in this pump the angle of the cylinder block can be varied creating the possibility for variable flow direction and control.

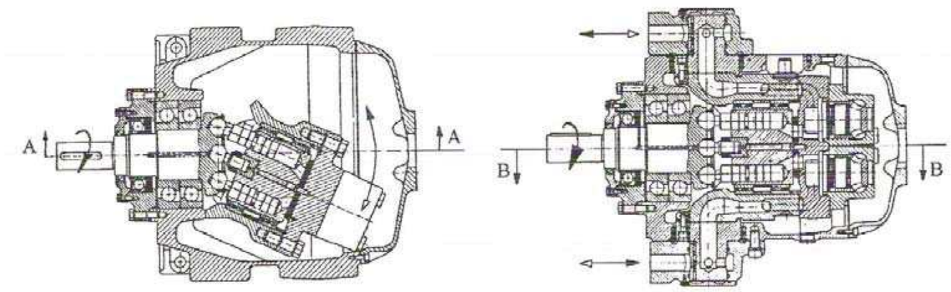


figure 30

Rudder angle and rudder torque

Depending on the rudder type, the steering gear must develop a corresponding torque at each rudder angle. For each of the steering gear types discussed before we will analyse the development of the torque developed at maximum oil pressure.

Ram type steering gear

The force developed in a ram type is: $F = p * \pi/4 * D^2$

In which: F = force on the ram in N
p = oil pressure in N/m^2
D = plunger diameter in m

In the yoke the force F is divided in a force F_h working perpendicular on the tiller and F_n working perpendicular on the slippers.

The rudder torque becomes: $T_r = F_h * l$

In which: T_r = rudder torque in Nm
l = effective length of tiller in m
F_h = force on the tiller in N
F_n = normal force in N

As both shaded triangles in figure 31 have the same shape we can state:

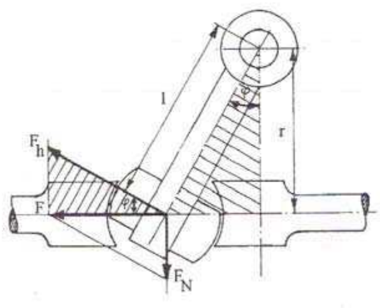


figure 31

$F_h = F / \cos \varphi$ and $l = r / \cos \varphi$ this gives : $T_r = (F * r)/ \cos^2 \varphi$

In which: φ = rudder angle in degree
r = distance centre of rudder stock to centre of rams in m

Conclusion is that to increase the rudder angle φ the rudder torque must be increased as $\cos \varphi$ is getting smaller at bigger angles and r is constant and the force F is only depending on the oil pressure.

Steering gear with two cylinders

In a double acting piston type with two cylinders the force delivered by the outgoing piston is bigger as the force delivered by the ingoing piston.

$F = p * \pi/4 * D^2$ $F' = p * \pi/4 * (D^2 - d^2)$

In which: F = force outgoing piston in N
F' = force ingoing piston in N
D = piston diameter in m
d = rod diameter in m

When the tiller turns α degree (see figure 32) the cylinders will also turn, only not with the same angle but with an angle β. The rudder torque is the moment caused by the forces working perpendicular on the tiller, this are the tangential forces F_t.
If the rudder is in middle position and at that time both centre lines of the cylinders are parallel to each other we can state the following:

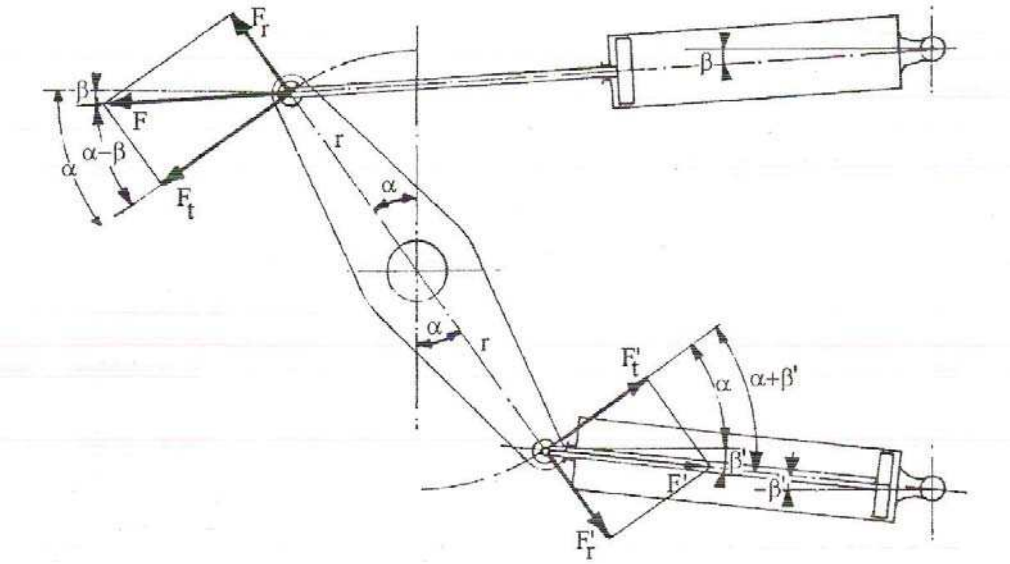


figure 32

The angles between the forces delivered by the rods and the tangential forces can be defined as:

Angle between F and F_t = $\alpha - \beta$
Angle between F' and F_t' = $\alpha + \beta'$

In which: α = rudder angle
β = angle between centreline of the outgoing piston and the middle position
β'= angle between centreline of the ingoing piston and the middle position

The rudder torque becomes: $T_r = r * F_t + r * F_t'$

In which: F_t = tangential force outgoing piston
F_t'= tangential force ingoing piston

At increasing rudder angle the required torque becomes smaller caused by the increasing influence of the angles β and β'. This results in the situation that if the unit must be able to give the required torque at maximum rudder angle the steering gear is actually to big for small rudder angles. By placing the cylinders towards each other the influence of angles β and β' is reduced.

Steering gear with constant torque

Rotary vane type steering gear units have the same effective working surface and arm at each rudder angle resulting in a constant rudder torque.

$T_r = (D + d)/4 * z(D – d)\ h * p$

In which: D = outer diameter vane in m
d = inner diameter vane in m
z = number of vanes
h = height of vane
p = pressure difference between both sides of the vanes in N/m^2

$(D + d)/2$ = mean diameter $(D + d)/4$ = mean radius

Example 1

A hydraulic steering gear consists of a hydraulic plunger pump and a steering gear engine with two hydraulic cylinders connected to a tiller and separated by controlled non return valves.

Number of plungers	z	= 5
Volume displacement per plunger	V_s	= 2 cm^3
Cylinder diameter	D	= 40 mm
Piston rod diameter	d	= 18 mm
Diameter steering wheel	D_{sw}	= 0.5 m
Distance tiller to centre of rudderstock	l	= 100 mm
Position of cylinders are perpendicular when rudder in midship position		
Pressure loss on controlled non return valves	Δ_p	= 0.3 bar
Hydro mechanical efficiency pump	η_{hmp}	= 0.95
Hydro mechanical efficiency motor	η_{hmm}	= 0.9

Determine:

a. Calculate the rudder torque delivered when the rudder is in midships and the helmsman maintains a force of 10 N with both hands on the circumference of the steering wheel.
b. Calculate the rudder torque when the helmsman is maintaining the same force on the wheel when the rudder is positioned at 35^o from the centreline.

   During steering, when the rudder is at midships, the rudder is hit by a wave resulting in the torque working on the rudder temporarily increases to 6 kNm.

c. What will be the highest pressure in the hydraulic lines during this situation?
d. Which force must the helmsman create in order to keep the rudder in position?
e. Which force must the helmsman develop in order to move the rudder in this sea state?
f. Which pressure would be appropriate as set point of the relieve valves if the above situation can be accepted as normal?

Solution:

a. The torque at the steering wheel is: $T_e = 2.F.r = F.D_{sw}$
   T_e = 10 N.0.5 m = 5 Nm
   The delivered pressure is: $T_e = (V_{s\ tot} \, . \, P_{man})/(2\pi \, . \, \eta_{hmp})$
   $P_{man} = (Te \, . \, 2\pi \, . \, \eta_{hmp})/V_{s\ tot}$ = (5 Nm . 2π . 0.95)/5 . 2 .10^{-6} m^3) =
   P_{man} = 2.98 . 10^{-6} N/m^2 = 29.8 bar

   The pressure after the controlled non return valves is: 29.8 – 0.3 = 29.5 bar

For the steering gear motor consisting of the two cylinders the torque will be:

$T_e = \eta_{hmm} (F . l + F' . l)$
$= \eta_{hmm} (p_{man} . \pi/4 . D^2 . l + p_{man} . \pi/4 . (D^2 – d^2) . l)$
$= \eta_{hmm} . p_{man} . \pi/4 . (D^2 + D^2 – d^2) . l$
$= 0.9 . 2.95 . 10^6 N/m^2 . \pi/4 (0.04^2 + 0.04^2 – 0.018^2) m^2 . 0.1 m$
$T_e = 601 Nm$

b. For $\varphi = 35^o$ $Te_{35}{}^o = Te_0{}^o . \cos \varphi = 601 . \cos 35^o \approx 492 Nm$

c. In this case the steering gear motor will act as a pump as the force on the rudder is coming from the outside.
$Te = (F . l + F' . l)/\eta_{hmm} = (p_{man} . \pi/4 . (D^2 + D^2 – d^2) . l)/ \eta_{hmm}$
$6000 Nm = p_{man} . \pi/4 (0.04^2 +0.04^2 – 0.018^2) m^2 . 0.1 m/0.9$
$p_{man} = 23.9.10^6 N/m^2 = 239 bar$

d. The helmsman does not have to give any force as the controlled non return valves keep the rudder in position.

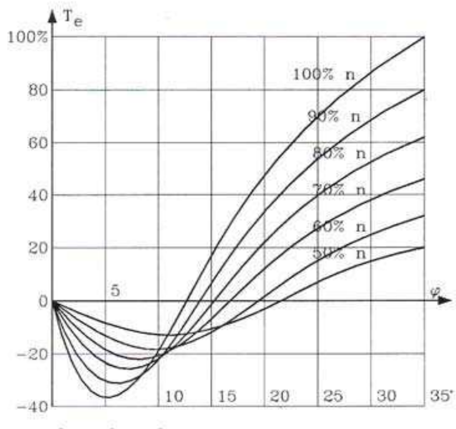
e. To move the rudder against the force of the wave the helmsman has to create a force which is equivalent to:
$T_e = \eta_{hmm} . p_{man} . \pi/4 . (D^2 + D^2 – d^2) . l$
$6000 Nm = 0.9 . p_{man} . \pi/4 (0.04^2 +0.04^2 – 0.018^2) m^2 . 0.1 m$
$p_{man} = 29.5.10^6 N/m^2 = 295 bar$
The pressure at the pump to be delivered by the helmsman is:
$p_{man} = 295 + 0.3 = 295.3 bar$
The torque to be delivered by the helmsman becomes:
$T_e = (V_{s\ total} . p_{man}) /(2\pi . \eta_{hmp})$
$= (5 . 2.10^{-6} m^3 . 29.53.10^6 N/m^2)/(2\pi . 0.95) = 49.47 Nm$
The force the helmsman has to deliver on the steering wheel becomes:
$T_e = 2. F . r = F . D_{sw} = 49.47 Nm = F . 0.5m$ $F = 98.94 N$

f. If the pressure found in answer e is just acceptable for the system the relieve valves should have a set point which is just above this pressure. A setting of 300 bar would be appropriate.

Example 2 Following data are from a four ram steering gear:

Diameter rams	D	= 160 mm
Distance centre of rudderstock to centre of rams	r	= 400 mm
Set point of relieve valves	p_{max}	= 150 bar
Revolutions of steering gear pump	n	= 25 Hz
Hydro-mechanical efficiency steering gear motor	η_{hm}	= 0.90
Volumetric efficiency steering gear motor	η_{vm}	= 1.00
Volumetric efficiency steering gear pump	η_{vp}	= 0.98
The steering gear is equipped with controlled non return valves		
The effective rudder torque T_e, when the rudder is at 35^o and the main engine is running full power (100 % n) is	T_e	= 216 kNm

The relation between the rudder torque, rudder angle and the main engine revolutions can be found in the graph.



Determine:

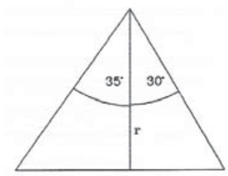
a. Between which rudder angles, when the main engine is running full rpm, must the controlled non-return valves avoid pressure shocks in the pipe lines?
b. Calculate the required oil pressure in order to move the rudder from 34^o to 35^o.
c. Until what value can the oil pressure drop before the rudder will start moving back from 35^o rudder angle?
d. Calculate the required displacement volume of the steering gear pumps, when using four rams and one pump, the steering gear has to meet the mandatory requirement for the movement of the rudder.

As a result of damage two rams are taken out of service and the vessel has to continue using one pump and two rams.

e. Is it still possible to turn the rudder to 35^o at full rpm? What would be the oil pressure? If this is not possible, with how many percent must the rpm be reduced in order to be able to turn the rudder to 35^o?

Solution:

a. Pressure shocks can occur if the rudder wants to move faster as the pumps can deliver oil to move the rudder. This phenomenon can take place when the rudder is moved from 0^o to 13^o and when is moved from 35^o to 13^o.
b. $T_e = (2 . F . r)/\cos^2 \varphi = (2 . \eta_{hm} . \pi/4 . D^2 . p_{man} . r)/(\cos^2 \varphi)$
   $216.10^3 = 2 . 0.9 . \pi . 0.16^2 . 0.4 . pman/(\cos^2 35^o)$
   $p_{man} = 10.10^6$ N/m^2 = 100 bar
c. When the rudder moves back the steering gear motor will act as a pump:
   $T_e = (2 . F . r)/\cos^2 \varphi = (2 . \pi/4 . D^2 . p_{man} . r)/(\eta_{hm} . \cos^2 \varphi)$
   $216.10^3 = 2 . \pi . 0.16^2 . 0.4 . p_{man}/(0.9 . \cos^2 35^o)$
   $p_{man} = 81.1.10^6$ N/m^2 = 81 bar
d. The steering gear must be able to turn the rudder from 35^o one side to 30^o on the other side in 28 seconds using one pump.
   Distance travelled by the rams:
   $s = r . \tan 35^o + r . \tan 30^o$
   $V_{vm} = s . A = s . \pi/4 . D^2$ $\eta_{vm} = 1.00$ so:
   $V_{vm} = r (\tan 35^o + \tan 30^o) . \pi/4 . D^2$
   $V_{vm} = r (\tan 35^o + \tan 30^o) . \pi/4 . 0.16^2 = 10.3.10^{-3}$ m^3 = 10 dm^3



This is the amount of oil required to move the rudder from 35° one side to 35° other side. This has to take place in 28 seconds so the displaced volume of the pump per stroke is:

$V_{vm}/t = 10.3 \cdot 10^{-3}\ m^3/28\ s = 0.367 \cdot 10^{-3}\ m^3/s$

$V_{vm} = 0.367\ dm^3/s = V_{dis} \cdot n \cdot \eta_{vp} = V_{dis} \cdot 25 \cdot 0.98 = 0.015\ dm^3$

e. T_{max} at 35° using two rams and one pump is:

$T_{max} = (1 \cdot \eta_{hm} \cdot \pi/4 \cdot D^2 \cdot p_{max} \cdot r)/\cos^2 \varphi$

$T_{max} = (0.9 \cdot \pi/4 \cdot 0.16^2 \cdot 150.10^5 \cdot 0.4)/\cos^2 35^o$

$T_{max} = 162.10^3$ Nm = 162 kNm

In order to move the rudder to 35° in normal condition a torque of 216 kNm was required. In this case there is only 162/216 ≈ 75% available. In order to move the rudder to 35° the rpm of the main engine has to be reduced to +/- 87 % n.

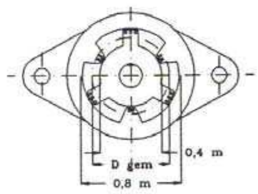
Example 3 Following data are from a rotary vane type steering gear:

Number of vanes	z	= 3
Outer diameter vanes	D	= 0.8 m
Inner diameter vanes	d	= 0.4 m
Height of the vanes	h	= 0.2 m
Maximum pressure delivered by the pump	p_{man}	= 100 bar
Hydro mechanical efficiency	η_{hm}	= 0.9
When the rudder is moved at maximum speed the volumetric efficiency is	η_v	= 0.88

Determine:

a. The maximum rudder torque.
b. The volume of oil needed theoretically to turn the rudder in 28 seconds from 35° one side to 30° other side.
c. The volume of leaking oil when the oil pressure is maximal.

Solution:



a. $b = (D – d)/2 = (0.8 – 0.4)/2 = 0.2$ m

$A = b \cdot h = 0.2 \cdot 0.2 = 0.04\ m^2$

$F = p \cdot A = 100.10^5 \cdot 0.04 = 400.10^3$ N = 400 kN

$D_{average} = (D + d)/2 = (0.8 + 0.4)/2 = 0.6$ m

arm $l = D_{average}/2 = 0.6/2 = 0.3$ m

$T = \eta_{hm} \cdot z \cdot F \cdot l = 0.9 \cdot 3 \cdot 400.10^3 \cdot 0.3 = 324.10^3$ Nm T = 324 kNm

b. The total volume of the vane area without vanes and partitions is:

$V_{360o} = \pi/4(D^2 – d^2)h = \pi/4(0.8^2 – 0.4^2)0.2 = 75.4.10^{-3}\ m^3 = 75.4\ dm^3$

To turn the rudder from 35° one side to 30° other side the vanes have to turn over an angle of 65°

$V_{65o} = 65^o/360^o \cdot V360^o = 65^o/360^o \cdot 75.4\ dm^3 = 13.6.10\text{-}3\ m^3 = 13.6\ dm^3$

$V_{th} = (3 \cdot V65^o)/t = (3 \cdot 13.6.10^{-3}/28) = 1.46.10^{-3}\ m^3/s = 1.46\ dm^3/s$

c. $\eta_{vol} = V_{th}/V_{eff} = 0.88 = 1.46.10^{-3}/V_e$

$V_{eff} = 1.66.10^{-3}\ m^3/s$

$V_{\ leakage} = (1.66 – 1.46).10^{-3} = 0.199.10^{-3} = 199\ cm^3/s$

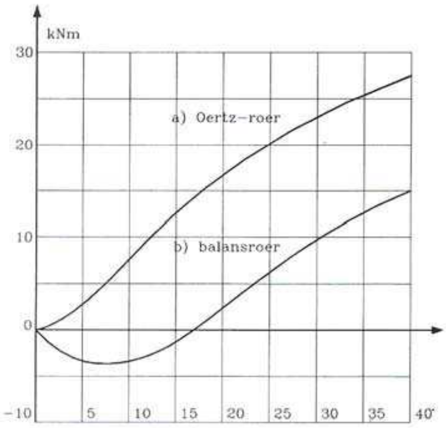
Questions steering gear

1. What is the function of the interconnection valve 6 in figure 13?
2. What is the function of the spring loaded valves 5 in figure 13?
3. What is a 4/3 way valve? Draw the symbol of this valve.
4. What is a telemotor?
5. Describe the operation of a controlled non-return valve. Draw the symbol of this valve.
6. Describe the function of a floating lever?
7. What is non-follow up steering mode?
8. What is follow-up steering mode?
9. By a big wave on the rudder the safety valve will open causing the rudder to change position. If the steering gear is operated in follow-up mode how will the rudder be moved back to its original position without turning the steering wheel?
10. What is the function of the limit switches in figure 14?
11. What is the reason why during manoeuvring two steering gear pumps are in use?
12. State three different types of steering gear units.
13. State two different type of pumps used in hydraulic steering gears.
14. Explain the operation of a radial plunger pump.
15. Explain the operation of an axial plunger pump.
16. State five mandatory constructional rules for steering gears.
17. State five mandatory operational rules for steering gears.
18. What solution can be found if in a four ram steering gear one ram becomes defective?
19. Which rudder type requires the lowest torque in order to turn it?
20. Explain what influence the position of the lateral point in relation of the turning axes has on the capacity of the steering gear.
21. When can it happen that a steering gear motor is acting as a pump?
22. Why is it inefficient to give big rudder angels?
23. Why are hydraulic hoses used to connect the hydraulic system to the steering gear of figure 25?
24. How can emergency steering take place on a steering gear using 4/3 way valves?
25. Make a drawing which shows that the required torque is increasing to increase the rudder angle for a ram type steering gear.
26. The following data are from a four ram steering gear:

Diameter rams	d	$= 0.15$ m
Distance centre ram to centre of rudderstock	r	$= 0.50$ m
Oil pressure	p_{man}	$= 120$ bar
Rudder angle	φ	$= 22^o$

    Determine:
    a. Calculate the delivered torque.
    b. Calculate the delivered torque if the hydro mechanical efficiency is 0.9.
    c. If the set point of the safety valves is 150 bar, at what torque will the valves open when the rudder is at 22^o.

27. If the above question was applying to a rotary vane type steering gear what would be the influence of the rudder angle on the answers?
28. The graph shows the development of the rudder torque in relation to the rudder angle for two types of rudder. a) the Oertz rudder and b) the Balance rudder.
    Which steering gear would be the best option to drive the Oertz rudder:
    a. A ram type steering gear
    b. A two cylinder type steering gear



29. If both rudders from question 28 are being moved from 0^o to 35^o when the ship is sailing full ahead, during which periods could there be a possibility that pressure shocks might occur?
30. Following data are from a rotary vane type steering gear:

Number of vanes	z	= 2
Outer diameter vanes	D	= 0.5 m
Inner diameter vanes	d	= 0.3 m
Height vanes	h	= 0.2 m
Maximum pump pressure	p_{man}	= 180 bar
Hydro-mechanical efficiency	η_{hm}	= 0.9
At 180 bar the volume of leaking oil	$V_{leakage}$	= 16.2 cm^3/sec

    Determine:
    a. The maximum torque this unit can produce.
    b. The theoretical oil flow in order to meet the mandatory requirements for the movement of rudders.
    c. The volumetric efficiency of the steering gear motor if it turned with maximum pressure and speed.

Air-conditioning

Air-conditioning is a field of engineering dealing with the design, construction, and operation of equipment used in establishing and maintaining desirable in door air conditions. These conditions vary according to the special requirements of the installation, which may be in a theater, factory, store, ship, or any other enclosure occupied by human beings.

In the modern engineering practice of air conditioning, two phases are involved:

- Actual conditioning of air, that is, the alteration under control of its temperature, humidity, purity, and oxygen content
- Ventilation, or replacement of stale air in an enclosure by conditioned air

Need for air

Human beings are air breathing animals. As such, man's lungs work unceasingly, awake or asleep. No one can hold his breath longer than a minute or two. The mechanism of the body demands that air, regardless of whether it is fresh or stale, be pumped in and out of the lungs continuously. The muscles that do the pumping operate automatically and no effort of will can stop them for longer than a couple of minutes at most. As a result, every person inhales and exhales a large number of cubic meters of air per day.

Purposes of air-conditioning

Apart from this merely mechanical reaction, considerations of health, efficiency, and morale require that the air should be fit to breathe. This necessitates, among other things, the removal of fumes from a ship's galley, engine room, battery room, and water closets. Stale air must then be replaced by fresh air. More over, the body gives off excess heat and moisture by means of the air that is breathed and the air in contact with the surface of the body. It becomes obvious, therefore, that proper air-conditioning within the enclosed quarters of a ship is important.

Air-conditioning is also needed for the protection of equipment, especially electrical apparatus. The large amount of moisture in the air given off daily from the bodies of the crew, from cooking, batteries, and bilges, would condense on any cool surface if it were not removed by air-conditioning. This moisture is extracted from the air by the air conditioning equipment.

Air as affected by human presence

Oxygen content of the air

What we call air is not a single substance, but is a physical mixture of various gases. About 1/5 of ordinary outdoor air is oxygen; a little less than 4/5 is the inert gas nitrogen; about 0.03 % is carbon dioxide; and the balance, less than 1 %, is composed of the gases argon, helium, krypton, neon, xenon, and hydrogen. These are the components of dry air. Usually some water vapor is present also, varying greatly from day to day according to the weather.

It is oxygen, of course, that is used by the body. Measurements have shown that for each lungful of air breathed in by a person, only 4 % of the oxygen in it is absorbed by the blood. It is therefore, evident that the air in a room can be circulated and breathed for a considerable period without ill effects.

Odors

Odors are always present, though the human sense of smell is not a keen one and usually is not aware of them. Almost everything gives off an odor, machinery, clothing, leather, books, food, the human body, even when clean and newly bathed, flowers, and perfume. The odor in an enclosed space is a combination of all such odors and is carried by the air. However, it is well known that even if an odor is noticed on entering a room, a person rapidly becomes unaware of it because the sense of smell is easily fatigued.

The heat and moisture contributed by human presence

An adult, when engaged in light work gives off on an average of about 145 W. In comparison, a 25-watt electric light bulb gives off 25 W. The heat from a bulb can be felt by the hand at a distance of several centimeters, because the heat-giving surface is concentrated in a small area. The heat from a human body cannot be felt because the surface of the body is large, measuring about 2.5 m^2 for the average adult male. The body also gives off considerable moisture, but the amount varies greatly according to the activity.

Under normal conditions a person takes in somewhat more than 1.5 liters of water per day, in beverages and food. Since the body is maintained at an average condition of equilibrium, this means that he gives off the same quantity per day, and much of this is evaporated directly into the air. If his activity or the air temperature is such as to cause a greater loss of water through perspiration, he feels thirsty and drinks more to maintain the balance. In a small enclosure, or room, the rise in temperature and moisture of the air caused by the presence of a number of people is considerable.

Theory of Air-conditioning

Psychrometry (sy-krometry) means literally, the measurement of cold, from the Greek psychros, cold. It is the special name that has been given to the modern science that deals with air and water vapor mixtures. The amount of water vapor in the air has a great influence on human comfort. Such atmospheric moisture is called humidity, and the common expression, "It isn't the heat, it's the humidity," is an indication of the popular recognition of the discomfort-producing effects of moisture-laden air in hot weather.

Air and humidity a physical mixture

The water vapor in the air is not absorbed or dissolved by the air. The mixture is a simple physical one, just as sand and water are mixed. The temperature of the water vapor is always the same as that of the air.

Saturated air

If a tin can is filled with sand to the top, there is still room into which water can flow between the sand grains. If the can of sand is then filled with water to the top, that sand is holding all the water it is able to hold. It is said that the sand is saturated with water.

In the same way, air can hold different amounts of water vapor, and when it is holding the entire vapor it is able to hold, it is called saturated air.

The amount of moisture at the saturation point varies with the temperature of the air; the higher the temperature, the more moisture the air can hold.

Dewpoint

The saturation point is more usually called the dewpoint, for if the temperature of the saturated air falls below its dewpoint, some of the water vapor in it must condense to liquid water, generally in drops. The dew that appears early in the morning on foliage when there is normally a drop in temperature, if the air is moist, is such a condensation, and is, as is readily recognized, the source of the term dewpoint. The sweating of cold water pipes, with which almost everyone is familiar, is also the condensation of dew from moist air on the cold surface of the pipes.

Condensation of saturated air

Condensation of water vapor from the air can take place at any air temperature, providing the temperature is below its dewpoint. In nature, moisture is condensed on foliage and other surfaces as dew if the air temperature is above $0^{o}C$. If the temperature of the surface is below freezing, the moisture condenses as frost. Above the earth's surface it is mist. When the mist is very thick, it is called a fog. If such condensation on dust particles is high in the air, the fog is then called a cloud. Under certain conditions of sudden cooling with much condensation, the droplets grow so large that they can no longer float in the air, and then they fall as rain. Sometimes a layer of air at a temperature below $0^{o}C$ exists high in a storm area; through this cold layer, raindrops may be carried down and up several times by air currents until they freeze and fall as hail. In cold weather when the temperature is below $0^{o}C$, condensation on the dust in the air forms snowflakes.

Difference between water vapor and water drops

The question is sometimes asked: If the air contains moisture, why does the moisture not freeze when the temperature is below $0^{o}C$? The answer is that only a liquid can freeze and a vapor is not a liquid. Drops of water, however small they may be, are merely small masses of liquid. In a mist or fog, the drops are so small that they float in the air, but they are nevertheless liquid. Air moisture does indeed freeze sometimes, if that moisture is in the state of liquid drops, and then it takes the form either of hail, or of sleet which is partially frozen moisture. Liquid moisture in the air (for example, mist) may exist in the form of drops subdivided so small as to be imperceptible to the human eye as individual drops; yet each single drop is formed of a great multitude of molecules. In a vapor or gas, the subdivision actually consists of single molecules.

Intermolecular distance determines state

The fundamental difference between the three states of matter-solid, liquid, and gaseous-is the distance between the molecules. In a solid, they are close and hold to one another, so that each has little or no freedom of motion. In a vapor or gas, the molecules are so far apart that all mutual attraction is lost, and each has complete freedom of motion, except as bounded by a container. Solids and liquids are visible to the human eye, but vapors and gases, with few exceptions, are invisible. Water vapor is invisible. The visible white cloud arising from a tea kettle or steam pipe is not really vapor or steam although it is usually called steam, but is formed of minute liquid droplets, that have condensed on striking the cooler air. They re-evaporate in a few minutes and are invisible again.

Formation of frost

When molecules of water vapor come sufficiently close together, they take on the liquid state and become visible. If this is in the air, the finely sub divided liquid appears as mist, fog, cloud, or rain. If the molecules come still closer, they finally get close enough to take on the solid state, that is, the liquid water freezes. It is apparent, therefore, that water vapor in the air cannot freeze, for as the molecules get closer, they first pass through the liquid state. Thus, only a liquid can freeze.

Water vapor can, however, condense directly to the solid state under certain conditions. This is not freezing, for the vapor does not become a liquid. The required condition is a surface at or below the freezing temperature, on which the water vapor arranges itself in solid geometrical forms or designs, called crystals. If the cold surfaces are extended, such as the ground, glass windows, and so forth, the crystals are called frost. Frosting always appears on the cooling coils (evaporators) of mechanical refrigerating systems. This frost must be removed periodically since it has some insulating quality and lessens the refrigerating capacity.

Heat of the air

Sensible heat of air

The heat of air is considered from three standpoints. First, sensible heat is that measured by household, or dry-bulb, thermometers. This is the temperature of the air itself, without regard to any humidity it may contain. It may be well to emphasize this by stating that sensible heat is the heat of dry air.

Latent heat in air

Second, air nearly always contains more or less moisture. Conditions of complete absence of moisture rarely occur, perhaps only in desert regions. Any water vapor present, of course, contains the latent heat which made it a vapor. Such latent heat of the moisture in the air may be spoken of as the latent heat in the air.

Total heat of air

Third, any mixture of dry air and water vapor, that is, air as we usually find it, does contain both sensible heat and latent heat. The sum of the sensible heat and latent heat in any sample of air is called the total heat of the air. It is usually measured from zero degrees as a convenient starting point.

The three air temperatures

Need for three air temperatures

Inasmuch as air-conditioning deals with these various heats of the air and the condensation of the moisture in it as well three -different temperatures are needed to understand and control the operations. These are the dry bulb, wet-bulb, and dewpoint temperatures.

Dry-bulb temperature

The dry-bulb temperature is the temperature of the sensible heat of the air, as measured by an ordinary thermometer. Such a thermometer is called in psychrometry, or air-conditioning engineering, a dry-bulb thermometer, because its bulb is dry, in contrast to the wet-bulb type next described.

Wet-bulb temperature

A wet-bulb thermometer is an ordinary thermometer with a cloth sleeve, of wool or flannel, placed around its bulb and then wet with water. The cloth sleeve should be clean and free from oil and thoroughly wet with clean fresh water. The water in the cloth sleeve is caused to evaporate by a current of air at high velocity, and the evaporation, withdrawing heat from the thermometer bulb, lowers the temperature, as then measured, a certain number of degrees. The difference between the dry-bulb and wet-bulb temperatures is called the wet-bulb depression. If the air is saturated, evaporation cannot take place, and the wet-bulb temperature is the same as the dry-bulb. Complete saturation, however, is not usual, and a wet bulb depression is normally to be expected.

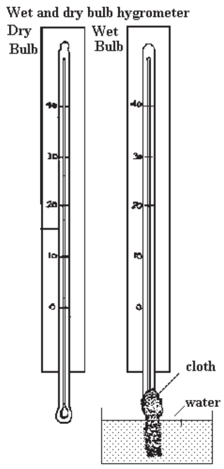


figure 1 dry & wet bulb thermometer

Wet-bulb temperature measures total heat

The wet-bulb thermometer indicates the total heat of the air being measured. If air at several different times or different places is measured and the wet-bulb temperatures found to be the same for all, the total heat would be the same in all, though their sensible heats and respective latent heats might vary considerably. Again, in any given sample of air, if the wet-bulb temperature does not change, the total heat present is the same, even though some of the sensible heat might be converted to latent heat, or vice versa.

Sling psychrometer

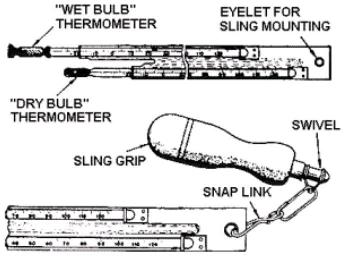


Figure 2 Sling psychrometer

In air-conditioning work, the two thermometers, wet-bulb and dry-bulb, are usually mounted side by side on a frame, to which a handle or short chain is attached so that the thermometers may be whirled in the air, thus providing the high velocity air current for evaporation. Such a device is called a sling psychrometer.

The psychrometer must be whirled around rapidly, at least four times per second. When the wet bulb thermometer is examined at intervals, its temperature reading will be found to be dropping; when no further drop is observed, that reading gives the correct wet-bulb temperature.

Dewpoint temperature

The dewpoint depends upon the amount of water vapor in the air. If air at a certain temperature is not saturated, that is, if it does not contain the full quantity of water vapor it can hold at that temperature, and the temperature of that air falls, a point is finally reached at which the air is saturated for the new, lower temperature and condensation of the moisture then begins. This point is the dewpoint temperature of the air for the quantity of water vapor present,

Relation of dry-bulb, wet-bulb, and dewpoint temperatures

The definite relationships between the three temperatures should be clearly understood. These relationships are:

When the air contains some moisture but is not saturated, the dewpoint temperature is lower than the dry-bulb temperature, and the wet-bulb temperature lies between them.
As the amount of moisture in the air increases, the differences between the temperatures grow less.
When the air is saturated, all three temperatures are the same.

Dewpoint Temperatures (°C)

Dry-Bulb Temperature (°C)	Difference Between Wet-Bulb and Dry-Bulb Temperatures (C°)															
	0	1	2	3	4	5	6	7	8	9	10	11	12	13	14	15
−20	−20	−33														
−18	−18	−28														
−16	−16	−24														
−14	−14	−21	−36													
−12	−12	−18	−28													
−10	−10	−14	−22													
−8	−8	−12	−18	−29												
−6	−6	−10	−14	−22												
−4	−4	−7	−12	−17	−29											
−2	−2	−5	−8	−13	−20											
0	0	−3	−6	−9	−15	−24										
2	2	−1	−3	−6	−11	−17										
4	4	1	−1	−4	−7	−11	−19									
6	6	4	1	−1	−4	−7	−13	−21								
8	8	6	3	1	−2	−5	−9	−14								
10	10	8	6	4	1	−2	−5	−9	−14	−28						
12	12	10	8	6	4	1	−2	−5	−9	−16						
14	14	12	11	9	6	4	1	−2	−5	−10	−17					
16	16	14	13	11	9	7	4	1	−1	−6	−10	−17				
18	18	16	15	13	11	9	7	4	2	−2	−5	−10	−19			
20	20	19	17	15	14	12	10	7	4	2	−2	−5	−10	−19		
22	22	21	19	17	16	14	12	10	8	5	3	−1	−5	−10	−19	
24	24	23	21	20	18	16	14	12	10	8	6	2	−1	−5	−10	−18
26	26	25	23	22	20	18	17	15	13	11	9	6	3	0	−4	−9
28	28	27	25	24	22	21	19	17	16	14	11	9	7	4	1	−3
30	30	29	27	26	24	23	21	19	18	16	14	12	10	8	5	1

figure 3 Table for dewpoint temeratures

Humidity

The word humidity is often used in speaking generally of the moisture, or water vapor, in the air. It has, besides, two technical meanings in the forms absolute humidity and relative humidity.

Absolute humidity

Humidity in air is expressed according to its weight. The weight of the moisture that air can contain depends upon the temperature of the air, and is independent of the pressure of the air. This weight is usually given in kg. Absolute humidity is the weight of water vapor in kg/kg or gr/kg of dry air. It should be understood that the weight of moisture in kg or grams refers only to moisture in the actual vapor state, and of in any way to any moisture that may be present in the liquid state, such as fog, rain, dew, or frost.

Relative humidity

Relative humidity is the ratio of the weight of water vapor in a sample of air to the weight of water vapor that same sample of air contains when saturated. This ratio is usually stated as a percentage. For example, if the air were fully saturated, its relative humidity would be 100 %. If the air contained no moisture at all, its relative humidity would be 0 %. If the air were half saturated, its relative humidity would be 50 %.

Importance of relative humidity

As far as comfort and discomfort resulting from humidity are concerned, it is the relative humidity and not the absolute or specific humidity that is the important factor. This can be most easily understood by an example.

It should be understood that moisture always travels from regions of greater wetness to regions of lesser wetness, just as heat travels from regions of higher temperature to regions of lower temperature. If the air above a liquid is saturated, the two are in equilibrium and no moisture can travel from the liquid to the air, that is, the liquid cannot evaporate. If the air is only partially saturated, some moisture can travel to the air, that is, some evaporation can take place.

Suppose the absolute humidity of the air to be 17 gr/kg of dry air. This is the actual weight of the water vapor in that air. If the dry-bulb temperature of the air is 24^{o}C, the relative humidity is nearly 90 %, that is, the air is nearly saturated. The body perspires but the perspiration does not evaporate quickly because the air already contains nearly all the moisture it can hold. The general feeling of discomfort is a warning that the environment under such conditions is not suitable for the best maintenance of health. Nature has, however, given the human body extraordinary powers of resistance, and the body can take a great deal of punishment without permanent harm, though its efficiency drops for the time being.

But if the dry-bulb temperature is 30^{o}C, the relative humidity is only 64 %. That is, although the absolute amount of moisture in the air is the same, the relative amount is less, because at 30^{o}C the air can hold more water vapor than it can at 24^{o}C. The body is now able to evaporate its excess moisture and the general feeling is much more agreeable, even though the air temperature is ten degrees hotter.

In both cases, the specific humidity is the same, but the ability of the air to evaporate liquid moisture is quite different at the two temperatures. This ability to evaporate moisture is indirectly indicated by the relative humidity. It is for this reason that extreme importance is placed upon control of relative humidity in air-conditioning.

Mollier's chart

There is a relationship between dry-bulb, wet-bulb, and dew point temperatures, and specific and relative humidity. Given any two, the others can be calculated. However, the relationship can be shown on a chart, and in air-conditioning it is customary to use the chart, since it is far easier than calculating. Such a chart is called a Mollier's chart, and it is given in figure 4

What is a Mollier's chart?

- the tool for determining of isobaric psychrometric processes of moist air
- suitable for steady conditions
- similar to psychrometric chart used mainly in Anglo-Saxon literature
- each chart is determined for specific pressure

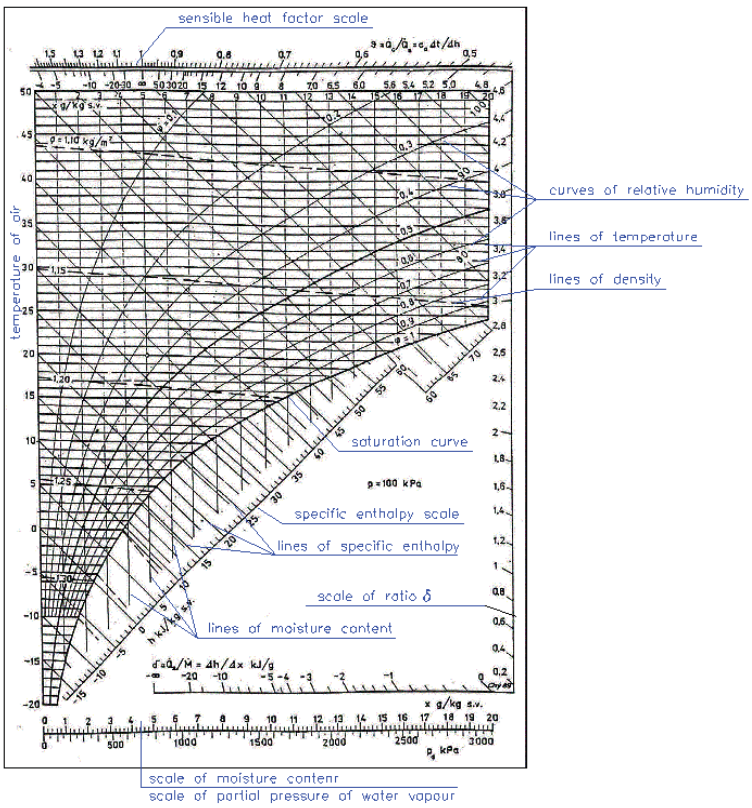
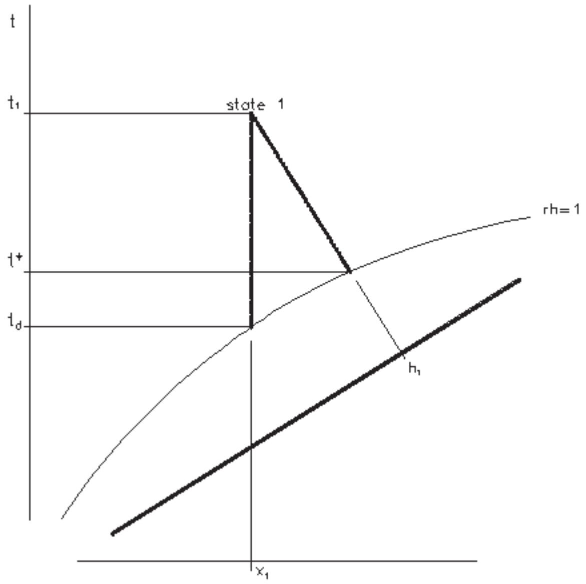


figure 4 Mollier's chart

Determination of the dew point temperature and the wet-bulb temperature

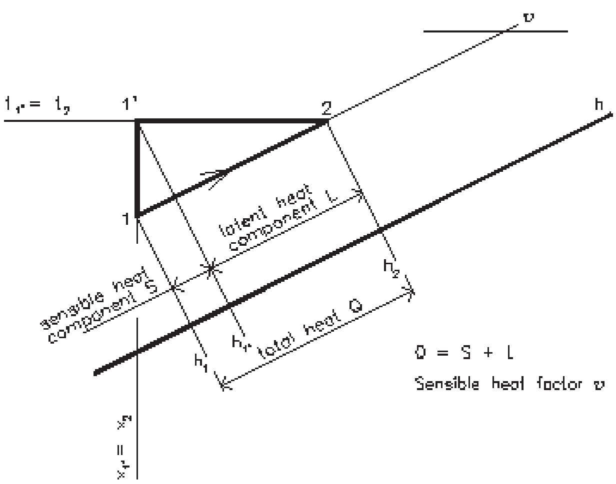
dew point temperature:	The temperature of moist air saturated at the same pressure *p* and with the same moisture content *x* as the given state of moist air.
wet-bulb temperature:	The temperature at which water evaporating into moist air, can bring air to the adiabatic saturation. The lowest temperature of adiabatic cooling of air.



Basic psychometric processes

Any process has two heat components:

sensible heat S	- moisture is constant
latent heat L	- temperature is constant



1 - 2 is any process in Mollier's chart

total heat Q = L + S

where: latent heat component $L = m_a.(h_2 - h_{1'})$
sensible heat component $S = m_a.(h_{1'} - h_1)$

sensible heat factor υ: $\vartheta = \frac{S}{Q} = \frac{c.\Delta t}{\Delta h}$ ratio of sensible heat component included in total heat

Sensible heating and cooling

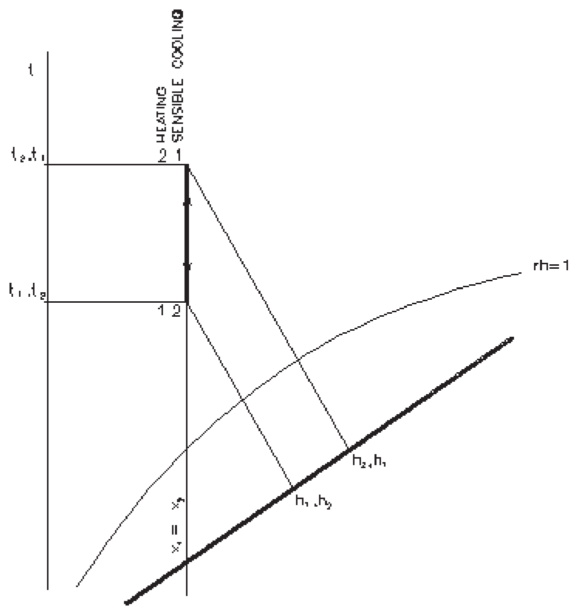
- the process with no changes of moisture content *x*
- the sensible cooling and the sensible heating are reversed processes
- in the case of sensible cooling the surface temperature of cooler have to be higher than the dew point temperature of moist air

Properties of states: $x_1 = x_2$

sensible cooling: $t_2 < t_1$, $rh_2 > rh_1$, $h_2 < h_1$
heating: $t_2 > t_1$, $rh_2 < rh_1$, $h_2 > h_1$

The heat required for the process:

sensible cooling: $Q = m_a.(h_1 - h_2) = m_a.c_d.(t_1 - t_2)$ [kJ/s] = [kW]
heating $Q = m_a.(h_2 - h_1) = m_a.c_d.(t_2 - t_1)$ [kW]



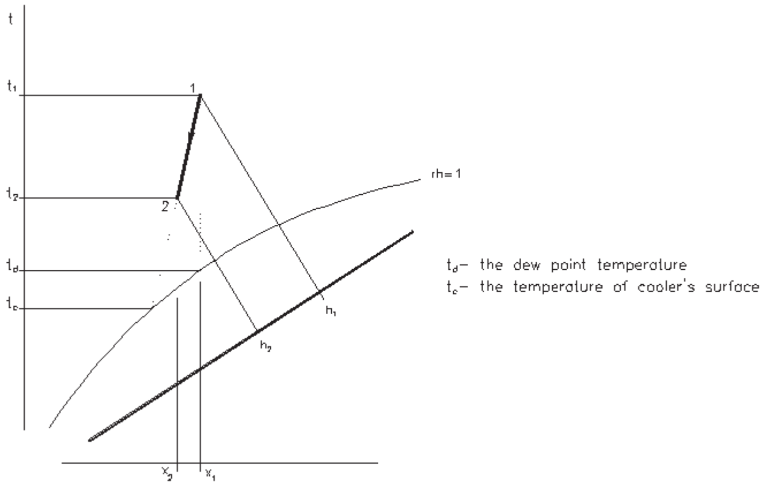
Cooling

- the process with change of moisture content *x*
- the surface temperature of cooler is lower than the dew point temperature of moist air -> the condensation occurs

Properties of states: $x_1 > x_2$
$t_2 < t_1$
$rh_2 > rh_1$
$h_2 < h_1$

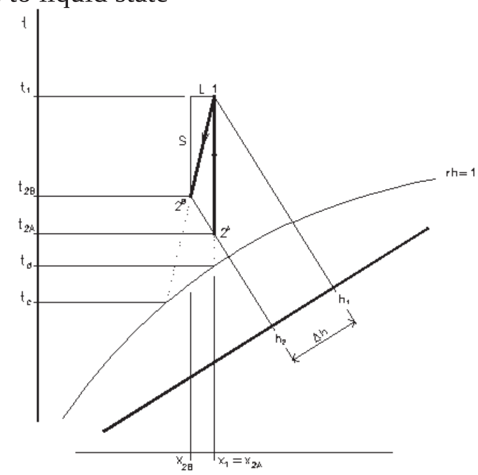
The heat required for the process: $Q = m_a.(h_1 - h_2)$ [kW]
The rate of condensing moisture: $m_w = m_a.(x_1 - x_2)$ [kg/s]

Mollier's chart:



The different between sensible cooling and cooling

While the cooler's surface temperature is lower than the dew point temperature condensation of water vapour occurs -> a part of total heat is used for change from gaseous state to liquid state



$$Q = m\Delta h;\ \Delta h = h_1 - h_2$$

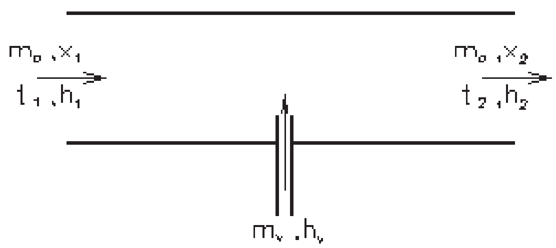
$$A)\quad Q = S + L;\quad L = 0$$

$$B)\quad Q = S + L;\quad L \neq 0 \ \Rightarrow\ x_2 < x_1 \rightarrow t_{2,B} > t_{2,A}$$

Humidification

The moisture content can be increased in two ways:

- the direct injection of steam into the air stream
- passing the air through a spray chamber containing small water droplets



a) Steam injection

the mass balance: $m_{a1} = m_{a2} = m_a$

steady-flow energy equations:

$m_a.x_1 + m_v = m_a.x_2$

$m_a.h_1 + m_v.h_v = m_a.h_2$

$\Rightarrow m_v.h_v = m_a.(h_2 - h_1)$

$m_v = m_a.(x_2 - x_1)$

$$\rightarrow \frac{h_2 - h_1}{x_2 - x_1} = h_v = q_{12} \rightarrow \textit{specific heat of the proces}$$

Properties of states:

$x_1 < x_2$

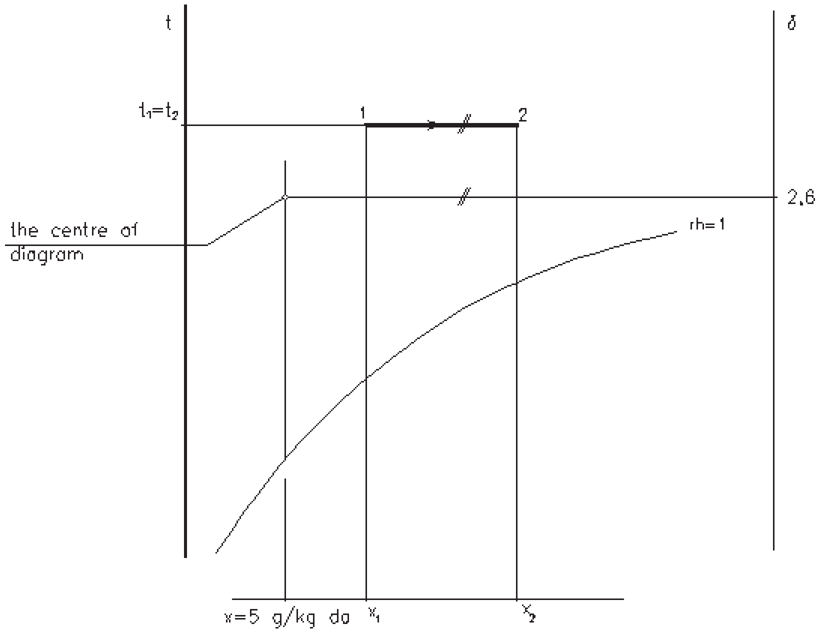
$t_2 = t_1$

$rh_2 > rh_1$

$h_2 > h_1$

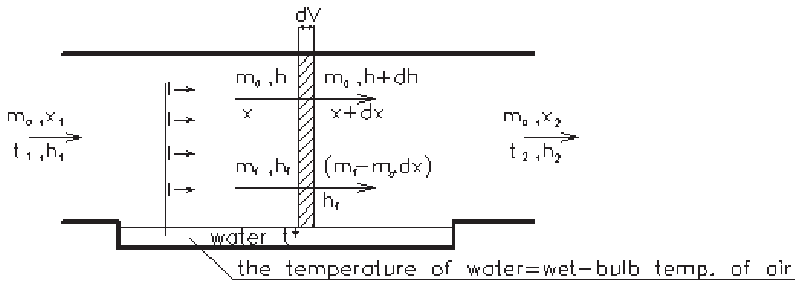
Mollier's chart: $\delta = \frac{\Delta h}{\Delta x} = h_v$ ratio δ where: h_v - specific enthalpy of injected steam

h_v varies about *2,6 kJ/kg*



The process of steam injection in the air stream is taken with reasonable accuracy as <u>isothermal</u>.

b) Humidification in spray chamber



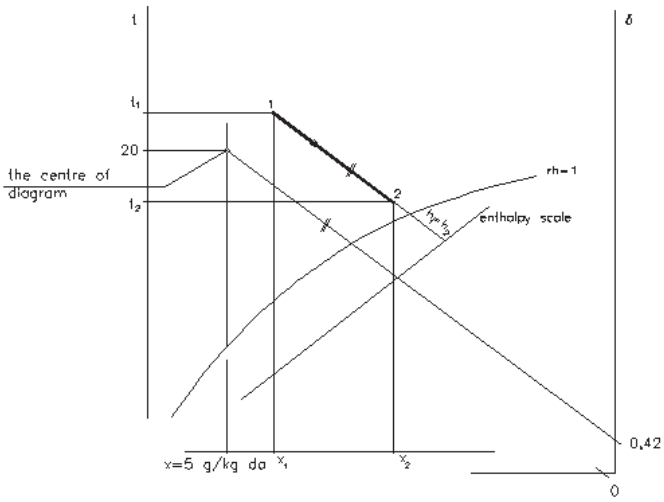
Energy balance: $m_a.dh = m_a.dx.h_f$

$m_a \cdot dh = m_a \cdot dx \cdot h_f \;\rightarrow\; \frac{dh}{dx} = h_f$; **the same equation as in the steam injection humidification**

h_f of water at temperature 10 to 15 °C varies between 0 to 0,42 kJ/kg

Properties of states: $x_1 < x_2$
$t_2 < t_1$
$rh_2 > rh_1$
$h_2 = h_1$

The rate of water transported into the air stream: $m_w = m_a.(x_2 - x_1)$ [kg/s]



Mollier's chart:

Humidification with water in spray chamber is taken with reasonable precision as <u>adiabatic</u> => specific enthalpy is constant

Mixing

Definition: mix of two air streams

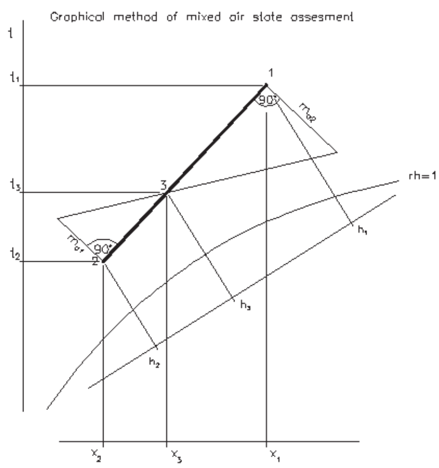
steady-flow energy equation: $m_{a1}.h_1 + m_{a2}.h_2 = m_{a3}.h_3$

mass balance equation: $m_{a1} + m_{a2} = m_{a3}$ for dry air
$m_{a1}.x_1 + m_{va2}.x_2 = m_{a3}.x_3$ for water vapour

from the equations ensues:

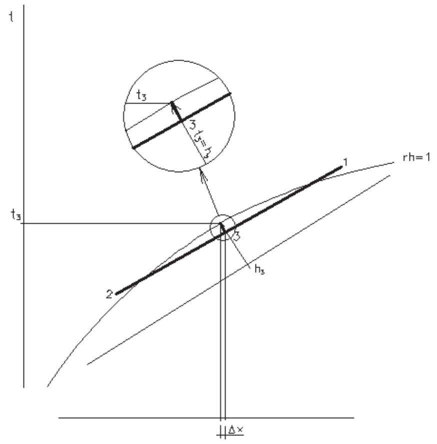
$$\frac{m_{a1}}{m_{a2}} = \frac{h_3 - h_2}{h_1 - h_3} = \frac{x_3 - x_2}{x_1 - x_3} \Rightarrow \frac{h_3 - h_2}{x_3 - x_2} = \frac{h_1 - h_3}{x_1 - x_3} = \delta_{23} = \delta_{13}$$

Mollier's chart:



In the case of result state is below saturation curve:

state 3 is oversaturated air => air immediately changes to the saturated air according to the line of constant temperature, *Dx* is amount of condensed water steam.



Examples using Mollier's chart

Example 1

Re-circulated air with a dry-bulb temperature of 20^{o}C and a wet-bulb temperature of 12.5^{o}C is mixed with half the amount fresh air of 36^{o}C with a relative humidity of 60%.

Determine the condition of the mixture.

Solution:

Point A is the condition of the re-circulated air.
Point B is the condition of the fresh air.
The mixture (M) is on the line AB
Length of line AB = 78 mm   AM : MB = 1 : 2
So AM = 1/(1+2) * 78 = 26 mm and MB = 2/(1+2) * 78 = 52 mm
With this point M can be found on the Mollier's chart

Example 2

Air of 29^{o}C with a dew-point of 19^{o}C is cooled down to 12^{o}C.

Determine how much heat and water will be extracted from the air, calculated per kg of dry air.

Solution:

The condition of the air (D) can be found on the intersection of the isotherm of 29^{o}C and the vertical line through point E, where the isotherm of 19oC intersects with the saturated line. The enthalpy in point D h = 65 kJ/kg. The moisture contents x = 0.014.
By cooling down the air to 12^{o}C the air condition will reach point F.
In point F the enthalpy h = 34 kJ/kg and the moisture contents x = 0.009.
Per kg dry air 0.014 – 0.009 = 0.005 kg water and 65 – 34 = 31 kJ are extracted.

Example 3

The outside air temperature is 20^{o}C with a vapour pressure of 1.2 kPa = 12 mbar

Determine the relative humidity.

Solution:

Follow the isotherm of 20^{o}C until the saturation line, this is point G. Go up vertical and intersect with the scale for partial pressure of water vapour p_d, this is point H. The maximum vapour pressure at 20^{o}C p_d = 0.233 bar = 23.3 mbar. The relative humidity can be found by p_d actual/p_d max = 12/23.3 = 0.0515

Example 4

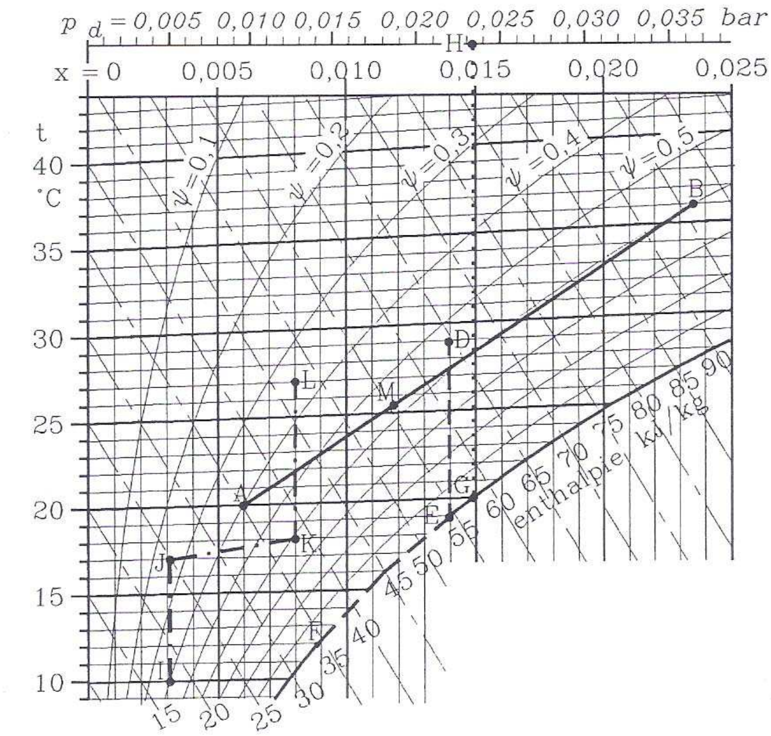
A pre-heating system first heats up 1.2 kg air/sec with a relative humidity of 0.4 and a temperature of 10^{o}C till 17^{o}C. After this steam with an enthalpy h = 2750 kJ/kg is injected in the air which raises the temperature of the air to 18^{o}C. In an after-heater the air is heated up to 27^{o}C.

Draw the total process in the Mollier's chart and determine mass flow of steam.

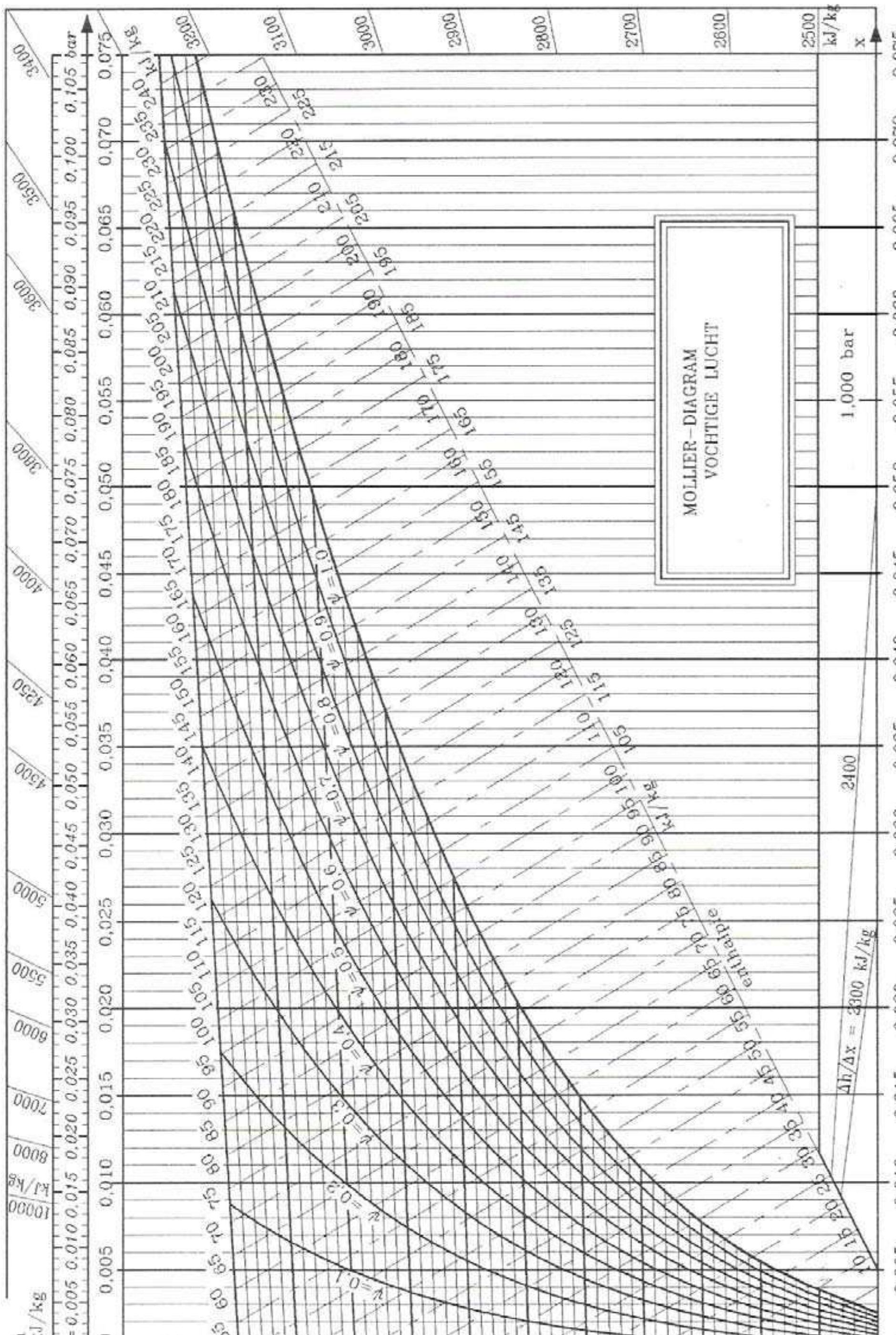
Solution

Line I-J is the heating from 10oC to 17oC.
Line J-K is perpendicular with the line through 0.0 and 2750. This is the portion where steam is added to the air.
Line K-L is the process in the after-heater.

The moisture content of the air before the injection of the steam x = 0.003. After the injection the moisture contents increases to x = 0.008.
This gives 0.008 – 0.003 = 0.005 kg steam/kg dry air.
Per second this becomes 1.2 * 0.005 = 0.006 kg



Graphical solutions examples 1 – 4


MOLLIER-DIAGRAM
VOCHTIGE LUCHT
1,000 bar
enthalpie
kJ/kg
Δh/Δx = 2300 kJ/kg

Psychometric chart

The Psychometric Chart is the same as the Mollier diagram, first reflected in a vertical mirror and then rotated through 90 degrees:

In this chart, note that the wet-bulb temperature scale and dew point temperature scale lie along the same line; which is, of course, the 100 percent relative humidity line. But note that the dewpoint temperature lines run horizontally. The wet bulb temperature lines run obliquely down to the right.

To use the chart, take the point of intersection of the lines of the two known factors, interpolating if necessary. From this intersection point, follow the lines of the unknown factors to their numbered scales and read the measurement.

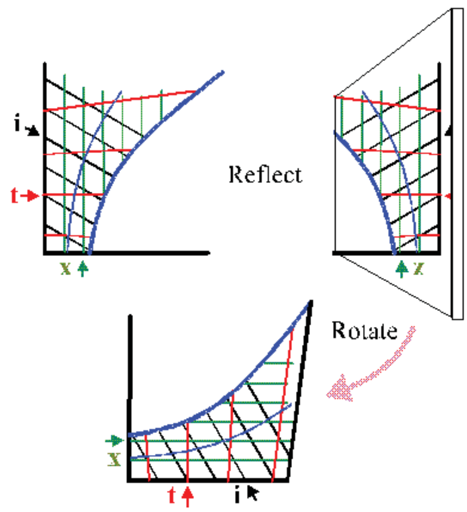


figure 5 from Mollier to Psychometric chart

Example 1.

Given a dry-bulb temperature of 30°C and a wet-bulb temperature of 20°C. What are the dewpoint temperature and the relative humidity? Note the intersection of the two given lines. From this intersection, follow horizontally along the dewpoint line (by interpolation) to the dewpoint scale. Answer. The dewpoint temperature is 15°C; the relative humidity is 40 %, read by interpolating the intersection point between curved relative humidity lines.

Example 2.

If the dewpoint remains at 15°C, what is the relative humidity if the air is then raised to the dry-bulb temperature of 35°C? Answer. Follow the dewpoint line horizontally to the 35°C dry-bulb temperature line, where interpolation reads 30%.

Example 3.

Given a dry-bulb temperature of 20°C and a dewpoint temperature of 15°C . What is the relative humidity if the dry-bulb temperature of the air is then raised to 25°C ? Answer. Note the intersection of the dewpoint 15°C line running horizontally from the dewpoint scale to the vertical 20°C line. Follow from the intersection horizontally to the 25°C dry-bulb line and the relative humidity is 52 %.

Example 4.

What is the actual weight of water vapor (absolute humidity) in air at 25°C dry-bulb and 20°C wet-bulb temperature? Answer. About 0.013 gr/gr dry air.

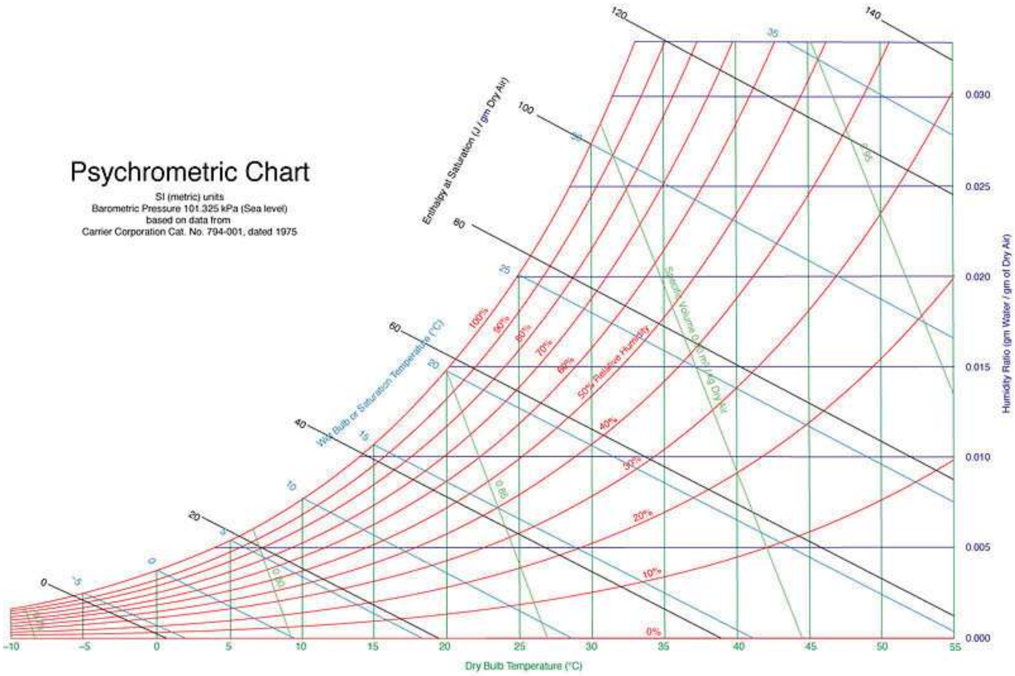


figure 6 Psychometric chart

Factors affecting human comfort and efficiency

Comfort

In air-conditioning practice, the term comfort is used to mean not comfort in the sense of mere pleasure, such as relaxing in a soft armchair, but rather comfort in the sense of physiological well-being and general efficiency of mind and body.

Humidity requirements for good health

If air is too dry, the mucous membranes of the mouth, nose, and lungs are adversely affected, and not only feel parched and uncomfortable, but are also more susceptible to germs. If air is too moist, the body is constantly in a state of perspiration, cannot maintain a proper rate of evaporation, and clothing stays damp. It has been found that for best health conditions, a relative humidity of from 40 to 60 % is desirable. Even within this range, a distinction can be made between winter and summer conditions, for the best possible results. In cold weather a range of 40 to 50 % of relative humidity, and in hot weather a range of 50 to 60 % is best. However, these optimum ranges cannot always be maintained in practical working, so that an overall range of 30 to 70 % relative humidity is acceptable, if not the best.

Temperature regulation of the human body

Ordinarily, the body is at a fairly constant temperature of 37°C. This, of course, refers to the interior of the body and not to the skin surfaces, which vary in temperature. Nature has so evolved the human body that any serious departure from this normal temperature of 37°C is dangerous to health. Even a change of one degree, up or down, is noticeable. But since the body is continually receiving a heat gain from surrounding and interior processes, there must also be a continuous outgo of heat to keep a balance. Fortunately, the body is equipped to maintain this balance automatically, and on the whole does, an extraordinarily good job.

Body heat gains

The body gains heat by:

Heat radiation gain

The heat radiation gain comes from our surroundings, but since heat always travels from regions of higher temperature to regions of lower temperature, such surroundings must have a temperature higher than 37°C for the body to receive heat from them. Indoor heat radiation is gained from heating devices, stoves, operating machinery, hot pipes, and electric light bulbs (this latter in small or negligible amount). The great source of heat radiation is the sun. The sun's radiation has healthful properties beyond mere heat.

Heat convection gain

The heat convection gain comes from currents of heated air only, and is usually found on shipboard only near a galley stove or engine.

Heat conduction gain

The heat conduction gain comes from objects with which the body may, from time to time, be in contact.

Body heat production

Most of the body's heat comes from within the body itself. Heat is being continuously produced inside the body by the oxidation of foodstuffs and other chemical processes, by friction and tension within the muscle tissues, and by other causes as yet not well known.

Body heat losses

The heat given off by the body is of two kinds, sensible and latent. Sensible heat is given off by the three methods: radiation, convection, and conduction. Latent heat is given off by evaporation.

Heat radiation loss

The body is usually at a higher temperature than that of its surroundings, and therefore radiates heat to walls, floors, ceilings, and other objects. The temperature of the air does not influence this radiation, except as it may alter the temperature of such surroundings.

Heat convection loss

Heat is carried away from the body by convection currents, both by the air coming out of the lungs, and by exterior air currents. These may exist in the air itself or be caused by a person's moving about.

Heat conduction loss

Since the body is usually at a higher temperature than that of its surroundings, it gives up heat by conduction through bodily contact with them.

Heat loss by evaporation

Under normal air conditions, the body gets rid of much excess heat by evaporation. When the body perspires, liquid water comes through the pores to the outer surface of the skin. There it immediately begins to evaporate, and it does so by withdrawing heat from the body. Inside the body the heat is sensible heat; in the process of evaporation, it becomes latent heat. The rate of evaporation, and hence of heat loss, depends upon the temperature, relative humidity, and motion of the air.

Ordinarily, that is, with air at not too high a temperature and relative humidity, and when not too active, the body gets rid of its excess heat by radiation, convection, and conduction. When engaged in work or exercise, the body develops much more internal heat, and perspiration begins. But perspiration rapidly evaporates if the relative humidity is not high. If, however, the relative humidity of the air is high, the moisture cannot evaporate, or does so only at a slow rate. In such cases, the excess heat cannot be removed by evaporation, and the body is dependent on radiation, convection, and conduction to eliminate its excess heat; this, of course, it cannot do and discomfort follows.

Amount of body heat loss

The amount of heat given off by the body varies according to its activity. When seated at rest, the aver age adult male gives off about 0.12 kJ/sec. When working at fullest exertion, he gives off 1.2 - 1.5 kJ/sec.

Research has shown that the total amount of heat loss is divided as follows for light work on a ship: about 45 % by radiation, 30 % by convection and conduction, and 25 % by evaporation. Research has shown further that for normal body comfort, it is important that the heat loss be in these proportions.

Thus, if a person loses the same total of heat in the proportions of 40 % by radiation, 50 % by convection and conduction, and 10 % by evaporation, he feels uncomfortable, damp, and chilly. This represents a condition of high relative humidity and too much air motion, as from a direct draft or fan breeze. On the other hand, if the total heat loss is the same, but divided in the proportions of 30 % by radiation, 25 % by convection and conduction, and 45 % by evaporation, he feels uncomfortable, hot, and parched. This represents a condition of low relative humidity and no air motion.

It is apparent that while the total heat loss may be a desirable amount in total, it may be so given off as to produce distinct discomfort. It is essential that the, air-conditioning be so controlled as to enable these heat losses to occur in the best proportions to produce comfort.

Comfort zones

Extensive research has shown that a normal feeling of comfort is experienced by most persons in air at different temperatures, relative humidities, and air motion, within not too great a range. The average temperature within a range in which the greatest percentage of persons feel comfortable has been given the name comfort line, and the range itself is called the comfort zone. Since summer and winter weather conditions are markedly different, the summer comfort zone varies from the winter comfort zone. But the human body is able to adapt itself automatically to summer and winter conditions. Indoor air conditions that are quite comfortable in summer are decidedly uncomfortable in winter, and vice versa.

All the information gathered in the tests has been assembled in a chart called the comfort chart (figure 7). This chart provides an authentic guide for air-conditioning, and if the air is maintained within the zones shown, it is found that general comfort is experienced.

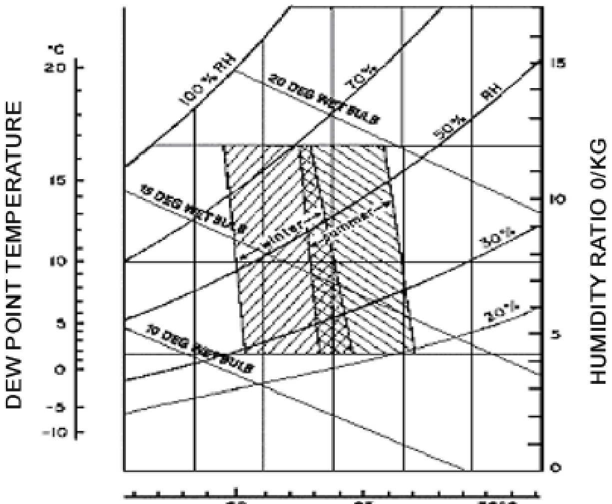


figure 7 Comfort zone

Heat and humidity as affected by air motion

In this chapter it has been necessary to explain individually the action of the various factors of heat and humidity. In reality, they act simultaneously and, moreover, the motion or lack of motion of the air itself influences their effects considerably.

Ventilation

Need for ventilation

The term ventilation is an old one, certainly long in use before air-conditioning was developed. It is derived from the Latin ventilare, meaning to whip up a breeze with wings or a fan. In common usage it means the supplying of fresh air to enclosed areas. In the old days, the only way of doing this was by opening doors and windows to permit the fresh outdoor air to blow in. That method is still in widespread use, of course. It is simple, costs nothing, and is reasonably efficient, provided that a sufficient cross current exists to siphon out the old air and bring in the new.

In modern air-conditioning, however, the term ventilation is restricted and means the motion of conditioned air inside an enclosure, fresh air being blown by electric fans through ducts or mains to the locations where it is needed, while the old air is removed by similar ducts and fans.

Importance of air motion for comfort

It is a well-known fact that when the air in a room is motionless, it soon feels stuffy to its occupants, even though the air may be quite fresh. On the other hand, air that is kept stirred, even if it is somewhat stale, at least does not feel stuffy, and though perhaps too warm, it is nevertheless bearable. It is chiefly to keep the air in motion that electric fans are used during hot weather in subways, street cars, offices, household rooms, and other in door quarters.

Effects of air motion

Three effects of air motion

When the air in a room is stirred, three effects on the human body result, all adding up to a feeling of greater comfort. One is a purely sensory effect, another affects humidity, and the third affects the room temperature. The three are closely interrelated and depend upon the velocity of the air motion.

Sensory effect of air motion

Air in motion has a definite action on the sensory organs in the skin. When the air has a gentle motion, a velocity of 0.10 – 0.25 m/s, the tactile sensory nerves in the skin are stimulated, and a feeling of greater comfort is experienced than when the air is completely still.

Effect of air motion on temperature

The body is always giving off heat to the air around it by conduction. If the air is still, the air close to the body gradually becomes more heated, and this heat is not carried away by convection currents in the air. Thus, although the average temperature of the air in a room may remain nearly constant, the body itself is in air of higher temperature. If the air is in motion, however, the heat coming from the body is carried away by convection and not permitted to build up.

Effect of air motion on humidity

The body is always evaporating moisture, even though the evaporation may be at such a slow rate that it is not perceptible as perspiration. If the air is still, this evaporated moisture stays close, forming with the heat also given off, a damp hot blanket around the body. Within such a blanket as the relative humidity rises, air is less able to absorb the evaporation from the body; hence a feeling of discomfort ensues. But if the air is stirred, the convection currents thus formed carry away the moisture as rapidly as it is given off, and a normal rate of evaporation is restored.

Interrelationship of the three effects

Up to an air motion of about 0.3 m/s, a person is conscious only of the stimulating effect, that is, the air feels alive. Above this velocity, an average person feels comfortable up to an air velocity of about 0.5 m/s. This is the limit of definite comfort. At air velocities above 0.6 m/s, the air motion does more than the mere removal of moist air from the vicinity of the body; it causes evaporation at a rate greater than normal. Such evaporation can take place only by using up additional body heat; as a result, the body feels cold and uncomfortable.

Limits of air motion

Thus, while the air needs to be kept in motion, there are necessary limits, from about 0.07 – 0.10 m/s to about 0.50 m/s. In general, if an air current is definitely perceptible, that is, if it attracts attention, then it is too much for comfort and may be a hazard to health.

In air-conditioning, air with a motion of 0.07 – 0.13 m/s is called relatively still air, because though it is not completely still, it does not readily attract attention.
The upper limit of 0.50 m/s is suitable for persons at rest or doing light work. When engaged in heavier work or exercise, a somewhat higher velocity of air motion may be accepted with comfort; but any substantial increase, while it may be momentarily cooling, is likely to be hazardous to health.

Direction of air current for comfort

When indoors, the body can stand a considerably greater air current from the front than from the rear or above. The higher air velocity limit should, for comfort, be avoided on the back of the body, on the head, and also on the feet, and still higher velocities indoors should be avoided for reasons of health.

Difference between indoor and outdoor air motion

It should be noted that the conditions described above apply only indoors. There is a surprising difference between the effects of air motion indoors and outdoors. A person can stand much greater air motion outdoors than indoors, without feeling discomfort. A strong draft indoors would be a mere pleasant breeze outside.

Location of air motion

It should be noted that the air motion or comfort lies within the layer occupied by a person, from the floor to a little over his head. Naturally the air coming out of the air-conditioning ducts into the rooms and quarters is at a considerably higher velocity, but the direction of these inlet currents should be and usually is so arranged that they do not strike directly on the occupants.

Air currents in ventilation

Natural convection

When air is warmed, it expands. Therefore, a unit volume of warm air becomes lighter and rises, or tends to rise. When air is cooled, it contracts, and a unit volume of it becomes heavier and sinks, or tends to sink. In an enclosed space, if air masses of different temperatures are present, and if no extraneous forces such as fans are present to move them about, the mass of warm air rises and the mass of cool air drops. A current thus created by masses of air moving in opposite directions because of differences in their relative weights is called a natural convection current.

If a heater is near the floor on one side of a room, warm air rising from it draws cooler air along the floor to take its place. A continuous circuit of air around the room, created by natural convection currents, is set up.

Forced convection

If, on the other hand, forces such as fans act on the air and actually move the air, currents are set up. Such currents are caused, not by the relative weights of the air, but by extraneous forces, and hence are called forced convection currents. Air-conditioning operates by forced convection.

Natural ventilation

When ventilation occurs by natural means, that is, by fresh air blowing in through open windows, doors, portholes, ordinary ventilators, or air ducts not containing a fan, and the used air finds an outlet, the method is called natural ventilation. Natural ventilation depends largely upon natural convection. Unfortunately, natural convection currents and drafts always take the most direct paths possible, and many places in a group of rooms such as in a building or in a ship, are bypassed and left as dead-air pockets. A ship is a difficult structure to ventilate by natural means.

Forced ventilation

Fortunately, it is no longer necessary to rely on natural ventilation. Forces can be brought to bear on the air to move it wherever desired. Fresh or newly conditioned air is pumped by fans through ducts to interior enclosed spaces, and used air and fumes are pumped out through separate ducts. Since such ducts can be led wherever necessary, thorough ventilation is thus assured. Such ventilation is known as forced ventilation.

Air-conditioning installation for ship's accommodation (figure 8)

The air is sucked in by the electrical driven ventilator A. The air will normally consist of a mixture of re-circulated air from the accommodation and fresh air from outside. The ratio between the airflows can be controlled by flaps. The reason that re-circulated air is used is to reduce energy consumption.

In case the ship is sailing in a cold climate the air has to be heated. If only fresh air would be used a higher level of energy would be required. By using re-circulation air which is already warmer the energy consumption can be reduced. The same principle applies when sailing in hot climates as then the re-circulation air is already colder.

The use of too much re-circulation air should be avoided as this air has a lower oxygen contents and it might contain unpleasant odors coming from the galley or other places. In no circumstance the re-circulation flap should be opened more than 70%.

In air filter B the dust and other particles in the air are removed. This filter needs regular cleaning or replacement.

In case the vessel is sailing in hot climate the air can be cooled down by the evaporator C. This evaporator is part of a Freon refrigeration system. The system consists of:

- Compressor F
- Condenser G
- Filter/dryer H
- Side glass J
- Thermostatic expansion valve K
- Pressure switch L for automatic control of the compressor

When air is cooled down the air can contain fewer vapors. Part of the vapor will condense and is removed from the unit.

When the vessel is sailing in cold climate the air can be heated up by the air heater D. In the example shown low pressure steam is used. Heating can also be achieved by using hot water. The amount of steam to the heater is controlled by thermostat M.

In case the air is heated up using heater D the relative humidity will drop and the air will become dry. To increase the relative humidity low pressure steam is blown into the air through a pipe E which has contains a series of nozzles. Instead of steam injection of water can be used to increase the humidity of the air.

Finally the conditioned air will leave the air conditioning unit and will be distributed around the accommodation.

Sanitary spaces

On board there is a separate system for ventilating the sanitary spaces like comfort rooms and showers. To avoid odors and vapors to mix with the air inside the accommodation the air from the sanitary spaces is sucked out separately by ventilators. In the lower portion of the doors of the sanitary spaces are grills which allow air to flow in from the corridors. In the ceilings of the sanitary spaces are the openings to the separate ventilation system. By this construction no air and vapors from the sanitary spaces can enter the accommodation.

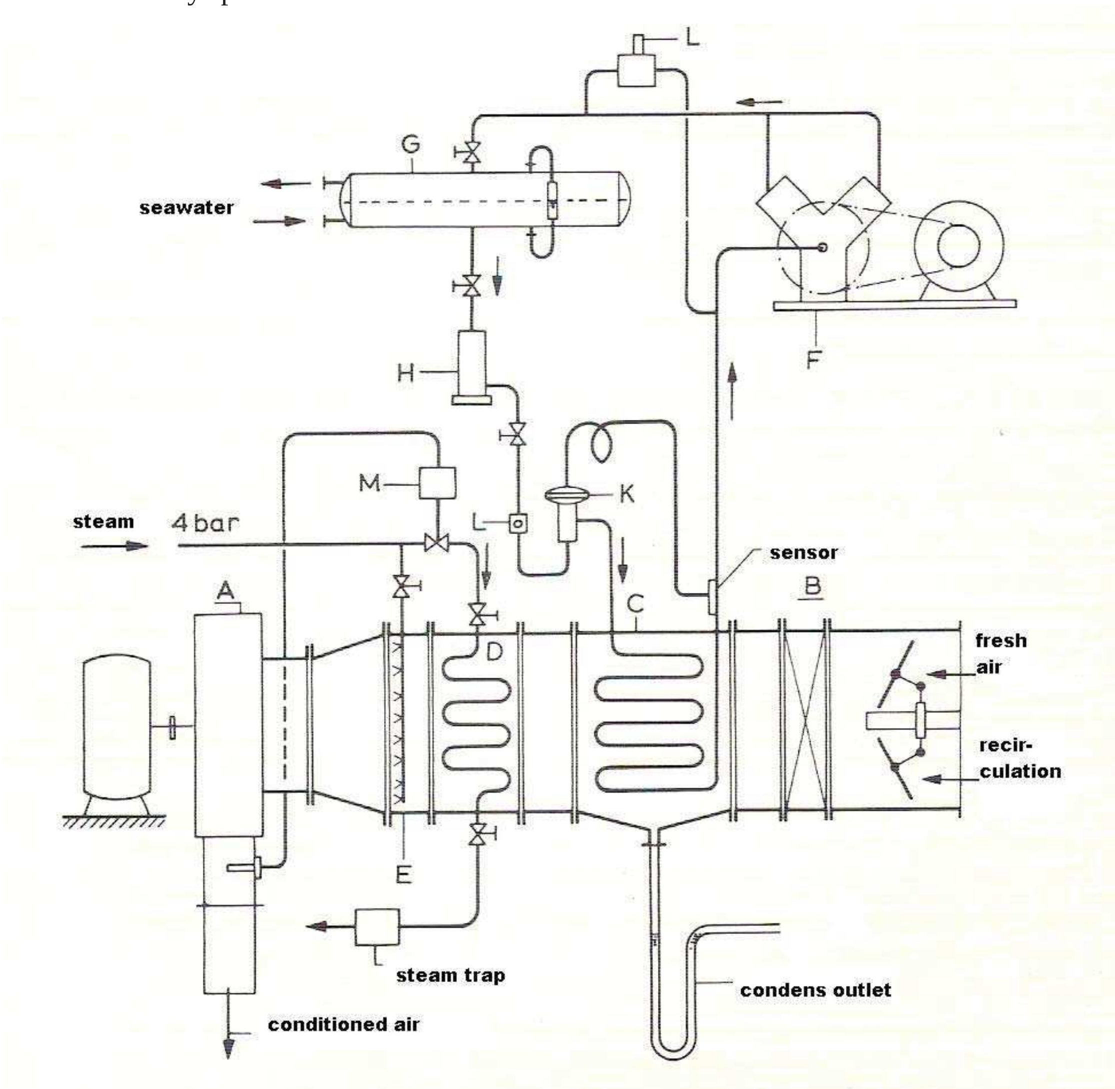
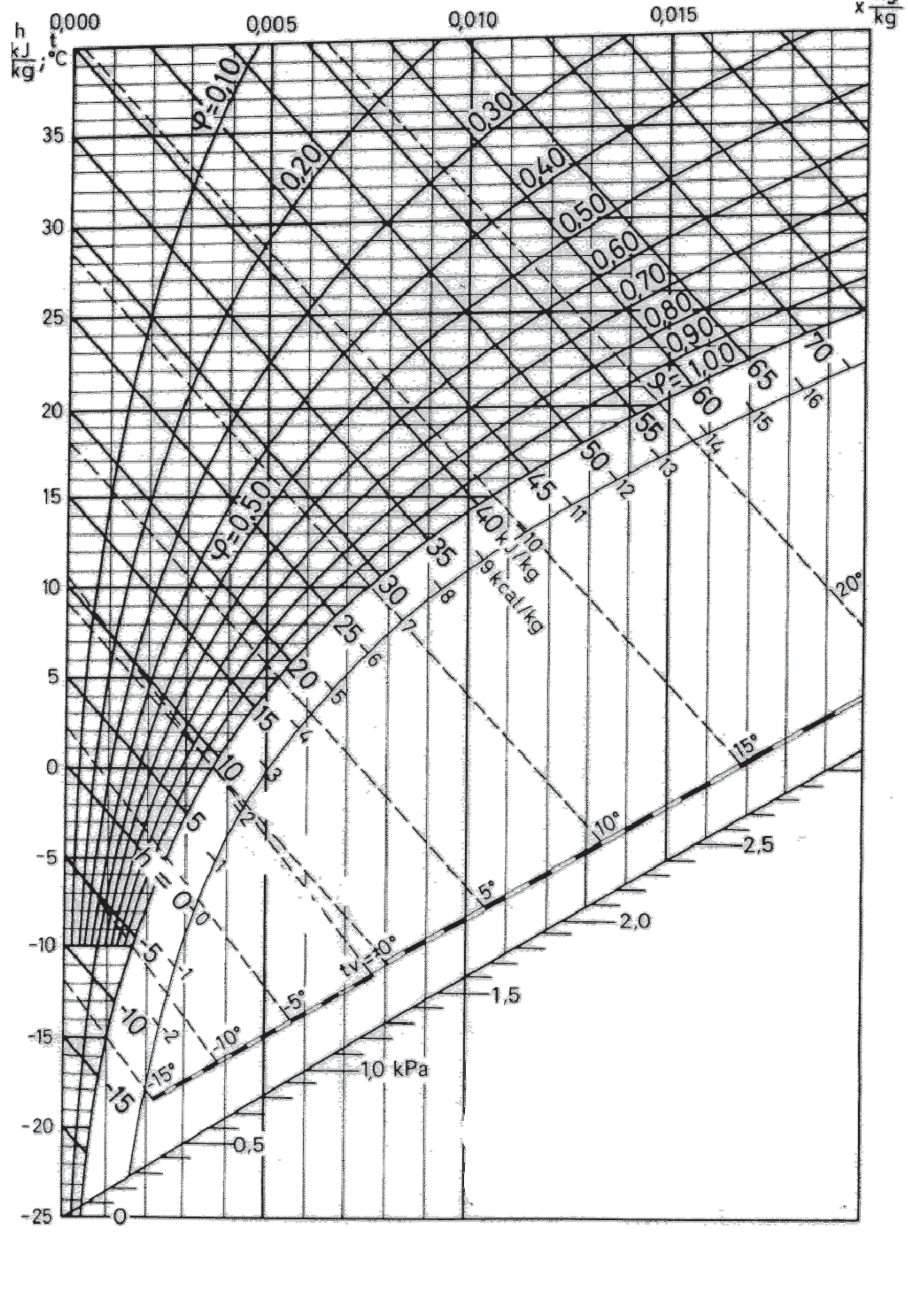


figure 8 AC installation

Questions Air-conditioning:

1. Why are air-conditioning systems used on board ships?
2. Why must the air sucked in be filtered before the ventilator?
3. By what means can air be heated up?
4. Why is it necessary to inject water or steam if air is heated up?
5. How can air be cooled down?
6. If the air-conditioning is running in a hot climate where does the human sweat go to?
7. Why do we use re-circulation air?
8. What is absolute humidity of air?
9. What is relative humidity of air?
10. Show using a Mollier's chart how air can be dried using a cooling machine.
11. Describe 2 methods how to determine the humidity of air.
12. Determine the relative humidity if a dry-wet bulb thermometer indicates 10 and 12^oC.
13. Same question for 20 and 22^oC.
14. If steam of 2600 kJ/kg is blown in air of 25^oC with x = 0.002 resulting in x = 0.02. What will be the new temperature and relative humidity?
15. If water is sprayed in air of 25^oC with x = 0.002, while the water temperature remains the same, rising the relative humidity to 80% what will be the new temperature and absolute humidity?
16. Two streams of air are mixed. 40% air of 40^oC with a relative humidity of 70% and 60% air of 20^oC with a relative humidity of 50%. Determine the enthalpy, temperature, absolute and relative humidity of the mixture.
17. Give four possible methods how the air condition can be brought in to the comfort zone.
18. Which spaces must be excluded in the re-circulation system of an AC installation



h
kJ/kg ; t °C
x kg/kg
0,000
0,005
0,010
0,015
35
30
25
20
15
10
5
0
-5
-10
-15
-20
-25
φ=0,10
0,20
0,30
0,40
0,50
0,60
0,70
0,80
0,90
φ=1,00
φ=0,50
h = 0
5
10
15
20
25
30
35
40 kJ/kg
45
50
55
60
65
70
9 kcal/kg
tv=±0°
5°
10°
15°
20°
-5°
-10°
-15°
0,5
1,0 kPa
1,5
2,0
2,5
0

Air compressors

Introduction

The object of all compressors is to raise the pressure of a gas with the minimum expenditure of energy.

There are four principle types of air compressors:

• Reciprocating compressors	Gas is compressed by positive displacement pistons in cylinders. Flow being controlled by valves
• Turbo machinery	Gas is driven by high speed impellers rotating in confined case
• Rotary Machines	Gas is compressed by rotors provided with lobes, gears, vanes or helical screw Near positive displacement
• Ejectors	Gas is moved using kinetic energy induced by high velocity jet through nozzles

When considering turbo machinery a number of different designations are used:

• Pumps	mainly for liquids
• Fans	move gases against small pressure differences with little change in density
• Blowers	move gases with some slight pressure differences
• Compressors	used to move gases and provide significant pressure increases

Reciprocating compressors

Reciprocating compressors are often used with air reservoirs to provide compressed air for starting diesel engines, driving air tools etc. Reservoirs have to be used because reciprocating compressors provide a pulsating air delivery.

Figure 1 shows a hypothetical indicator diagram for a single stage -single acting reciprocating compressor.

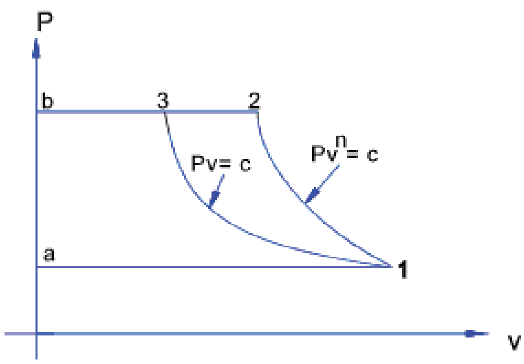


figure 1 hypothetical indicator diagram

a to 1 Air is drawn into the cylinder on the suction stroke
1 to 2 The suction valve is closed and air is compressed according to the law $pv^n = c$
2 to b The delivery valve opens and air is delivered under pressure
b to a The delivery valve closes and the suction valve opens

The theoretical work done on the air per cycle is the area enclosed by [a-1-2-b- a] which equals:

$$W = P_2V_2 + \frac{P_2 V_2 - P_1 V_1}{n - 1} - P_1 V_1$$

$$= (P_2V_2 - P_1 V_1)\left(1 + \frac{1}{n - 1}\right)$$

$$= \left(\frac{n}{n - 1}\right)(P_2V_2 - P_1 V_1)$$

If C is the rate at which the cycles are repeated then the rate at which energy is imparted to the air:

$$\text{Power} = \frac{n}{n - 1}\ (P_2V_2 - P_1 V_1)\ C$$

The ideal compression requiring the minimum amount of work is the perfect reversible isothermal compression process which obeys Boyle's law pv = c. This is represented by 1-3.The work saved per cycle is [1-2-3-1].

The objective is to minimize the air compressor work that means to approach the reversible process i.e. minimize the friction, and turbulence.

Practical way to do this is to make v (specific volume) small by maintaining t (temperature) at low temperature during compression. In other words, to reduce the work input to a compressor, air should be cooled as it is compressed.

The compressor isothermal efficiency is a measure of the departure from the ideal compression process and is defined as:

$$\eta_{iso} = \frac{\text{Calculated isothermal power}}{\text{Power imparted to air}}$$

Clearance Volume effect

A practical single stage compressor cylinder will have a small clearance at the end of the stroke. This clearance will have a significant effect on the work done per cycle.

A section of a typical reciprocating single-stage, single-acting compressor cylinder is shown in figure 2 Inlet and discharge valves are located in the clearance space and connected through ports in the cylinder head to the inlet and discharge connections.

During the suction stroke the compressor piston starts its downward stroke and the air under pressure in the clearance space rapidly expands until the pressure falls below that on the opposite side of the inlet valve. This difference in pressure causes the inlet valve to open into the cylinder until the piston reaches the bottom of its stroke. During the compression stroke the piston starts upward, compression begins, when the pressure has reached the same pressure as the compressor intake. The spring-loaded inlet valve then closes. As the piston continues upward, air is compressed until the pressure in the cylinder becomes great enough to open the discharge valve against the pressure of the valve springs and the pressure of the discharge line. From this point, to the end of the stroke, the air compressed within the cylinder is discharged at practically constant pressure.

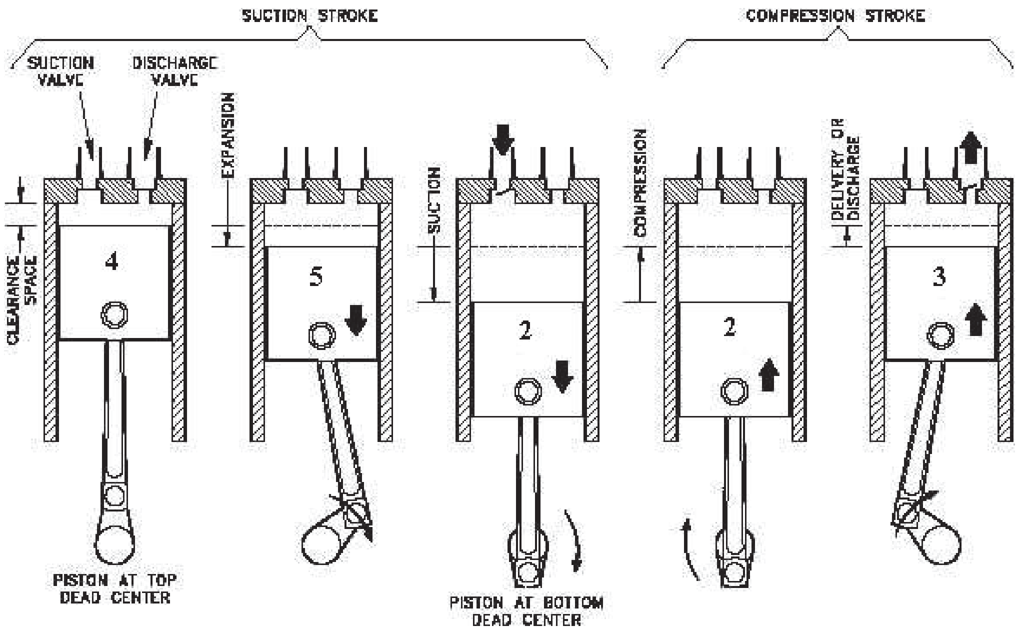


figure 2 Single-Acting Air Compressor Cylinder

The compressor cycle of figure 2 can be drawn in a p-v as shown in figure 3.

In operation the air in the clearance volume expands to 5 before any fresh air is drawn into the cylinder. The stroke is from 1 to 2 with a swept volume of (V_2 - V_1) but the suction is only from 5 to 2 giving a volume of (V_2 - V_5) taken into the cylinder on each stroke.

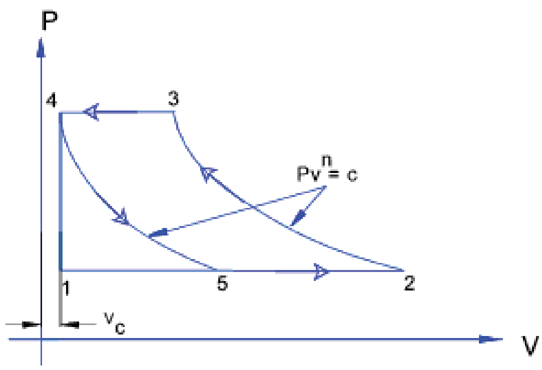


figure 3 Effect of Clearance Volume

The volumetric efficiency obtained from the hypothetical indicator diagram is:

$$\eta_v = \frac{V_2 - V_5}{V_2 - V_1} = \frac{\text{Volume of air pumped /cycle}}{\text{Stroke volume of cylinder}} = \frac{\text{mass of air delivered/cycle}}{\text{mass of air to fill stroke volume}}$$

Assuming compression curve 2 to 3 and the expansion curve 4 to 5 follow the same law $pv^n = c$ then:

$$\left(\frac{V_2}{V_3}\right) = \left(\frac{P_3}{P_2}\right)^n = \left(\frac{P_4}{P_5}\right)^n = \left(\frac{V_5}{V_4}\right)$$

The volumetric ratio of compression (V_2 / V_3) = the volumetric ratio of expansion $(V_5 / V_4) = r_v$. The volumetric efficiency:

$$\eta_v = \frac{V_2 - V_5}{V_2 - V_1} = \frac{(V_c + V_l) - r_v V_c}{V_l}$$

That is:

$$\eta_v = 1 - (r_v - 1)\frac{V_c}{V_l} = 1 - \left[\left(\frac{P_3}{P_2}\right)^{1/n} - 1\right]\frac{V_c}{V_l}$$

It is clear that the smaller the clearance volume V_c the larger the volumetric efficiency will be. In practice it is possible to get the clearance volume down to 3 to 5% of the stroke.

When clearance is taken into account the work done per cycle:

$$= \left(\frac{n}{n-1}\right)(P_3V_3 - P_2V_2) - \left(\frac{n}{n-1}\right)(P_4V_4 - P_5V_5)$$

The hypothetical power of a single stage compressor (kW working on c cycles /s):

$$\text{Power} = \left(\frac{n}{n-1}\right)\frac{c}{1000}\left[(P_3V_3 - P_2V_2) - (P_4V_4 - P_5V_5)\right]$$

The actual compressor diagrams differ from hypothetical diagrams because of valve opening and closing delays and component inertia. A typical actual indicator diagram is shown in figure 3.

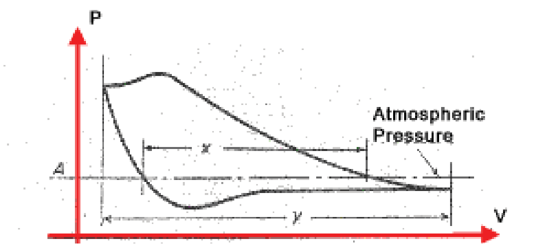


Figure 3 Actual air compressor diagram

A good approximation of the volumetric efficiency is indicated by the ratio of x to y measured at the atmospheric pressure line.

The actual performance of a reciprocating compressor used as pump is measured by the ratio:

$$= \frac{\text{equivalent volume of free air delivered per cycle}}{\text{vol swept by piston}}$$

Multi-stage

When air at high pressure is required, multi-staged compression is more efficient than using a single stage compressor. Also single stage compressors delivering high pressures result in high gas temperatures which effect the lubrication and increase the risk of burning.

Effect of Cooling

Isentropic process: No cooling during compression
Polytropic process: Involve some cooling
Isothermal process: Involve maximum cooling

Which process yields the minimum required compressor work?

We can plot them in a p-v diagram as follows:

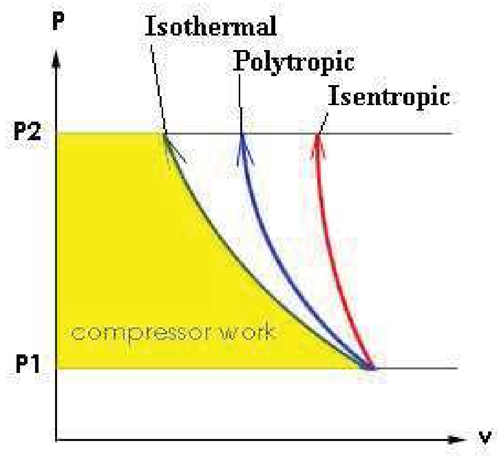


figure 4 p-v diagram

where:

The isentropic compression process (n=k)
The polytrophic compression process (1<n<k)
The isothermal compression process (n=1)

k is the ratio of the specific heat of the gas at constant pressure to the specific heat at constant volume k = cp/cv
For air compressors at compression n = 1.25 – 1.4, at expansion n = 1 to 1.2

The shaded area represents the air compressor work required during compression process of an isothermal process.

Because the area from each line to the left is the required air compressor work, we can see that an isothermal process requires lower amount of energy than polytropic process and isentropic process respectively.

Now we understand that we can save the energy required for compression if we could have some cooling during compression.
Next is to see how to save the energy in polytrophic compression process which is the case of most air compressors.

We know that the minimum air compressor work is achieved with isothermal compression. In practical way, we try to achieve that by involving some cooling during the compression process that leads to a polytropic compression process.

Normally, this can be achieved by dividing air compression into 2 stages. The first stage builds up the pressure from p_1 to p_x then the compressed air is cooled by the intercooler and the second stage compressor builds up the pressure again from p_x to the final pressure p_2. See the following figures to understand how the energy can be saved by using intercooling between each stage.

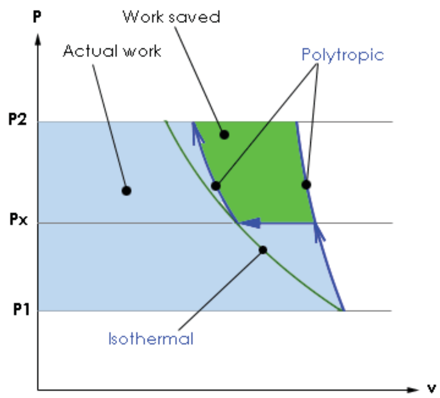


figure 5 p-v diagram of polytropic compression process with intercooling

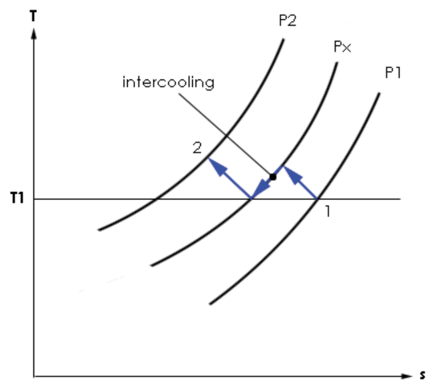


figure 6 T-s diagram of polytropic compression process with intercooling

We can see from figures 5 and 6 that the amount of compressor work saved is related to the pressure p_x.

What is the optimal value of p_x that yields maximum compressor work saved?
The total compressor work, for this case, is the summation of compressor work of each stage as follows:

$$w_{total} = w_1 + w_2$$

We can see that w_{total} will be lowest when $w_1 = w_2$. Thus:

$$p_1/p_x = p_2/p_x \quad \text{or}$$

$$P_x = \sqrt{P_1 \cdot P_2}$$

That means <u>the pressure ratio of each stage should be identical to get the lowest amount of work required for air compression</u>.

Reciprocating compressors

The reciprocating air compressor, illustrated in figure 7, is the most common design employed today. The reciprocating compressor normally consists of the following elements:

- The compressing element, consisting of air cylinders, heads and pistons, and air inlet and discharge valves.
- A system of connecting rods, piston rods, crossheads, and a crankshaft and flywheel for transmitting the power developed by the driving unit to the air cylinder piston.
- A self-contained lubricating system for bearings, gears, and cylinder walls, including a reservoir or sump for the lubricating oil, and a pump, or other means of delivering oil to the various parts. On some compressors a separate force-fed lubricator is installed to supply oil to the compressor cylinders.
- A regulation or control system designed to maintain the pressure in the discharge line and air receiver (storage tank) within a predetermined range of pressure.
- An unloading system, which operates in conjunction with the regulator, to reduce or eliminate the load put on the prime mover when starting the unit.

Air compressor operation

Figure 7 shows the general operation of the air compressor. The air compressor has two cylinders; air from the low pressure cylinder feeds into an intercooler and then into the high pressure cylinder.

The compression cycle starts with the low pressure piston A at the top of its stroke. When the piston moves down, it draws air through the air filter B and inlet valve C into the cylinder. The air filter keeps dirt out of the compressor.

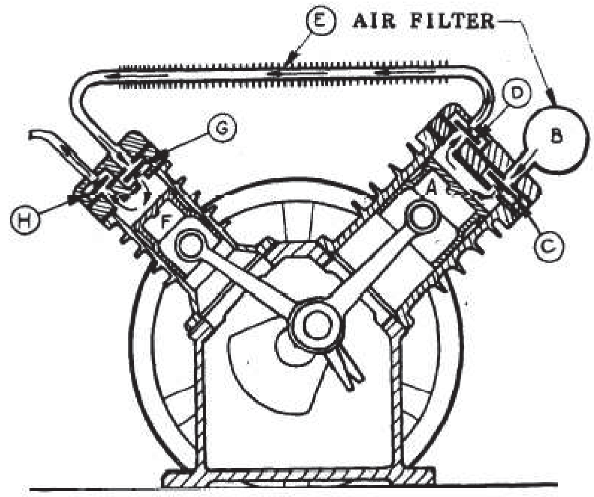


Figure 7 Reciprocating Air Compressor

On the upstroke, inlet C closes and the piston A pushes air out through the exhaust valve D and into the intercooler E. Compressing the air increases its temperature. The intercooler E dissipates some of that heat to the surrounding atmosphere before passing the compressed air on to the high pressure stage for further compression. The high pressure stage works in the same way as the low pressure stage does, except that the high pressure piston goes up when the low pressure piston goes down. This way, the low pressure piston is drawing air in when the high pressure piston is pushing air out. Compressed air from the high pressure stage goes through the capacity control system to the air receiver tank. The capacity control system regulates the loading of the air compressor and engine speed.

Figure 8 shows the cross section drawings of a single acting reciprocating air compressor with one crank and two stages.

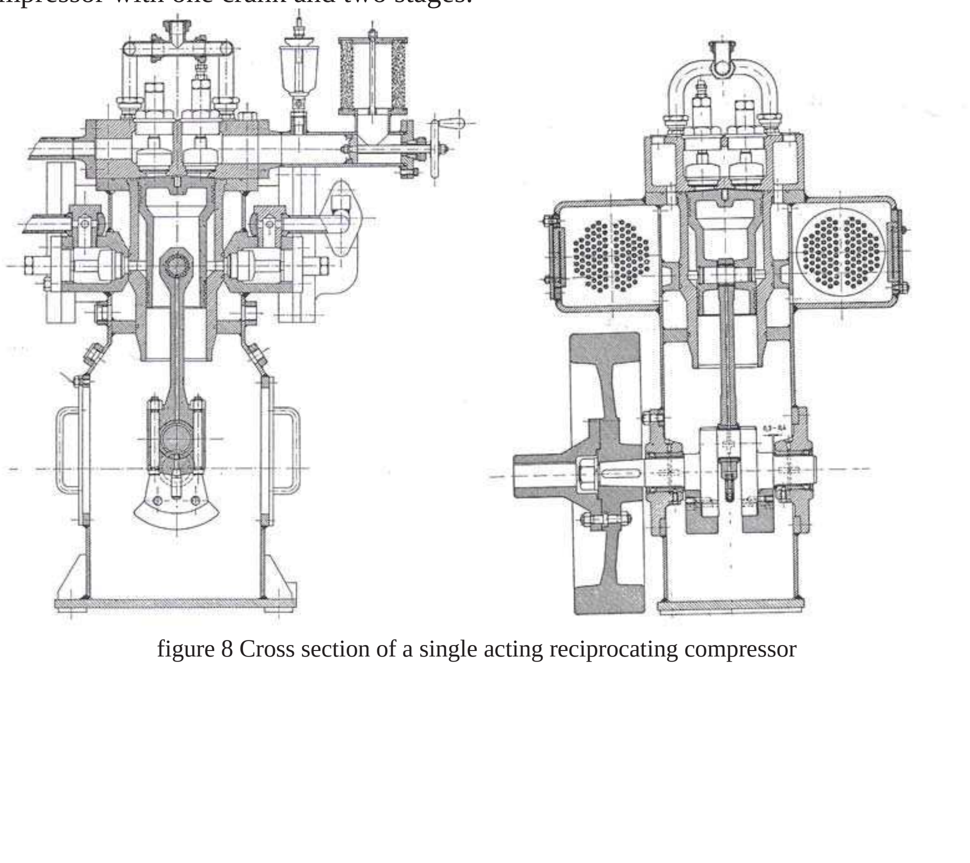


figure 8 Cross section of a single acting reciprocating compressor

When the piston moves downwards the volume above the piston will increase and when the pressure difference is big enough the suction valve will open and air will be flow inside. During the upward movement of the piston the air above the piston will be compressed and the suction valve is closed during this motion and when the pressure has increased sufficient the discharge valve will open. The air will pass the inter cooler and flow into the second stage of the compressor.

The second stage is created by the lower portion of the piston which has a smaller diameter. The air is flowing in the second stage during the upward movement of the piston. During the downward movement the air is compressed and will when the pressure is high enough open the discharge valve. The air will flow to the air bottles passing the after cooler.

Automatic operation

On board starting air compressors and working air compressors are normally equipped with accessories in order to run the compressor automatic. The compressor will be started by a differential pressure switch which measures the pressure inside the air receiver. When the pressure will reach a pre-set level the switch will start the compressor. When the pressure has increased to the cut-off level the switch will stop the compressor.

In order to minimise the currents to the electro motor during starting the compressor should not start to compress immediately as this will require a big torque to be delivered by the electro motor. The compressor has to start in a so called no-load condition. This condition is in most systems obtained by keeping the suction valve(s) of the first stage open during starting by an unloader. This will allow the air which is sucked in to escape to the atmosphere again during the compression stroke resulting in no compression taking place.

When the compressor has reached its working rpm the timer in the electrical circuit will cut off the unloader after which the compressor will start to operate.

In order to avoid flow back of compressed air from the air receiver to the compressor when it is stopped a non return valve is fitted in the discharge line.

Lubrication

Reciprocating air compressors need lubrication for the crankshaft and connecting rod bearings. The cylinder liner is mostly lubricated by splash lubrication. This is created by the rotation of the connecting rod through the oil sump. The lubricating oil pressure is created by a gear pump normally fitted at the end of the crankshaft.

Air contains moisture which becomes water during compression and cooling. This also happens in the cylinder. To avoid that the lubrication would be lost special lubricating oil is applied which has the capacity to emulsify with water but will still keep a lubricating effect.

The single acting reciprocating compressor has an lubricator connected at the air inlet as the lubricating oil from the crankcase can not reach the upper part of the piston.

Rotary and Turbo Compressors

Rotary or turbo-compressors deal with larger flow rates of air than reciprocating compressors but usually at lower delivery pressures. Rotary compressors can be driven by high speed electric motors, steam turbines, and internal combustion engines. They are usually multi-stage machines of the centrifugal or axial-flow types.

In centrifugal compressors a number of impellers are mounted on a common rotor in a robust casing. Air from the atmosphere enters the eye of the first impeller it then acquires kinetic energy from the rotating impellers. The air is directed from the periphery of the impeller into stationary diffuser vanes which are designed to convert the kinetic energy of the gas to increased pressure. The gas is directed inwards to the eye of the next impeller and the process is repeated as it passes through each stage the pressure being progressively increased.

In the axial-flow compressor, the air is compressed while continuing its original direction of flow. The rotor has fixed blades that force the air rearward much like an aircraft propeller. In front of the first rotor stage are the inlet guide vanes. These vanes direct the intake air toward the first set of rotor blades. Directly behind each rotor stage is a stator. The stator directs the air rearward to the next rotor stage. Each consecutive pair of rotor and stator blades constitutes a pressure stage.

Higher duty rotary compressors are usually provided with water cooling with intercoolers. The volumetric efficiency of turbo-compressors is usually defined by the ratio:

$$= \frac{\text{equivalent volume of free atmospheric air finally delivered}}{\text{volume of free atmospheric air entering the compressor}}$$

Although minimum work input is usually achieved with a constant temperature (isothermal) reversible process, compression in rotary compressors is most often assessed relative to the reversible adiabatic process (isentropic -constant s processes). The pv diagram below shows the different processes.

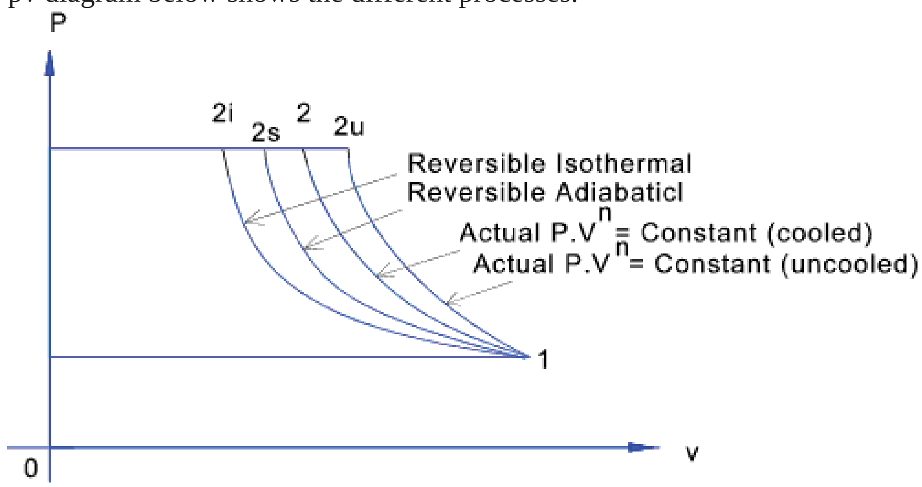


figure 9 p-v diagram for ratory compressors

An ideal compression process with no losses would be adiabatic and real processes are compared to this by having using the adiabatic- isentropic efficiency which is defined as.:

$$\eta_s = \frac{\text{Calculated power for reversible adiabatic compression}}{\text{indicated power of compressor}}$$

The rotary compressor is adaptable to direct drive by induction motors or multi cylinder gasoline or diesel engines. The units are compact, relatively inexpensive, and require a minimum of operating attention and maintenance. They occupy a fraction of the space and weight of a reciprocating machine of equivalent capacity. Rotary compressor units are classified into three general groups, slide vane-type, lobe-type, and liquid seal ring-type. The rotary slide vane-type, as illustrated in figures 10 and 11, has longitudinal vanes, sliding radially in a slotted rotor mounted eccentrically in a cylinder. The centrifugal force carries the sliding vanes against the cylindrical case with the vanes forming a number of individual longitudinal cells in the eccentric annulus between the case and rotor. The suction port is located where the longitudinal cells are largest. The size of each cell is reduced by the eccentricity of the rotor as the vanes approach the discharge port, thus compressing the air.

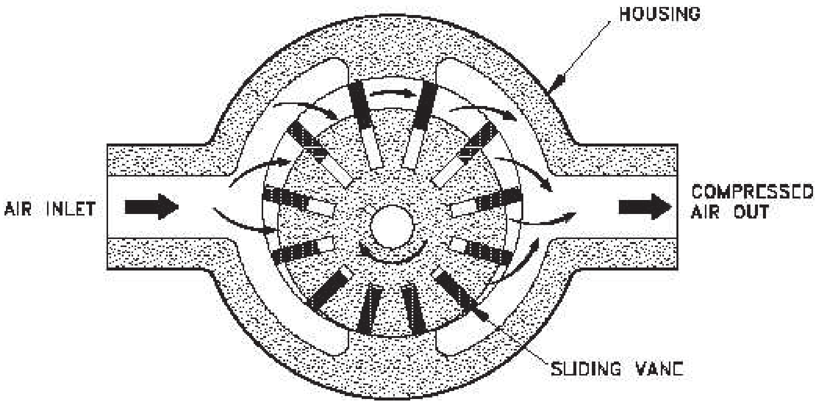


figure 10 Rotary Slide Vane Air Compressor

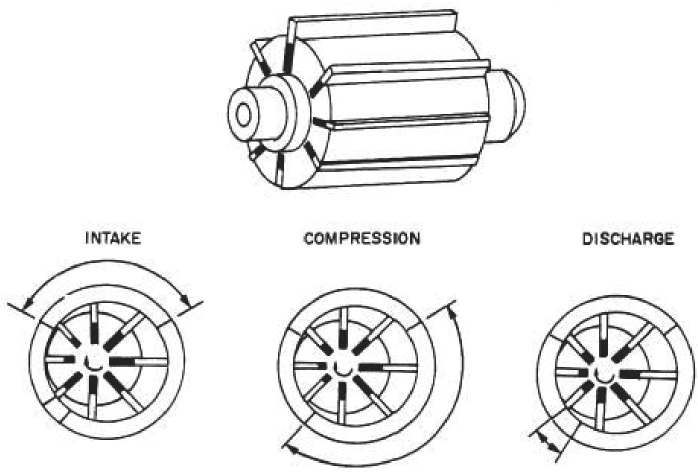


figure 11 The compression process

Rotary lobe compressor

The rotary lobe-type, illustrated in figure 12, features two mating lobe-type rotors mounted in a case. The lobes are gear driven at close clearance, but without metal-to-metal contact. The suction to the unit is located where the cavity made by the lobes is largest. As the lobes rotate, the cavity size is reduced, causing compression of the vapor within. The compression continues until the discharge port is reached, at which point the vapor exits the compressor at a higher pressure.

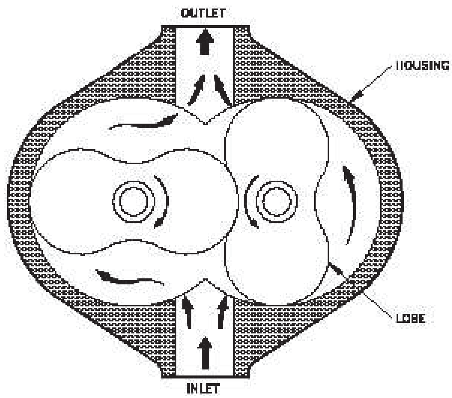


Figure 12 Rotary lobe-type compressor

The rotary liquid seal compressor

As the name implies, liquid ring vacuum compressors require a liquid to create a seal inside the pump. The liquid ring compressors are sealed with either water or oil. The operating principles of the two are the same: an impeller, which is offset so the impeller is not in the centre of the pump housing, rotates and traps pockets of air in the space between the impeller fins.

The impeller is typically made of brass and has fixed fins. As the impeller turns, there is a pocket of air that is trapped in the space between each of the fins. This space is first increased (the suction side of the pump) and then decreased (discharge side of the pump). As the air is compressed, it is then pushed out of the pumps discharge.

The rotary liquid seal compressor is frequently used in specialized applications for the compression of extremely corrosive and exothermic gasses.

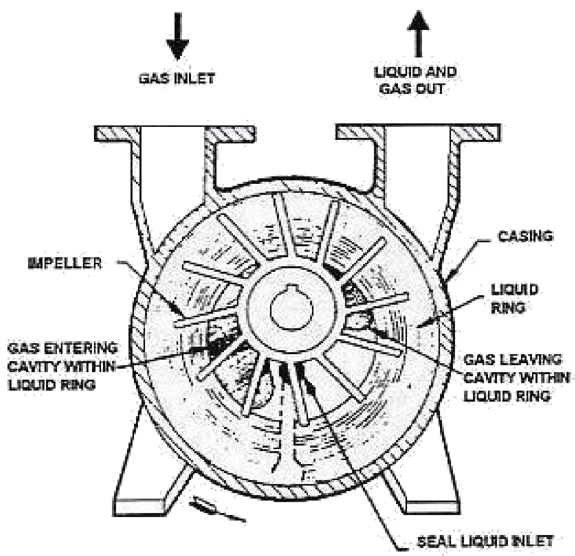


figure 13 Rotary Liquid seal compressor

Rotary Helical Screw type

Figure 14 shows a cross section of a single stage rotary helical screw compressor. Within the compressor body there are two screws with matting profile: a female and a male screw, female having concave inlets and the male with convex helical inlets. The screws rotate in opposite directions with the female screw receiving the driving power and transmitting this power to the male screw through a set of synchronization gears.

The screw element's main parts, the male and female rotors, move towards each other while the volume between them and the housing decreases. Each screw element has a fixed, integrated pressure ratio that is dependent on its length, the pitch of the screw and the form of the discharge port. To attain the best efficiency the pressure ratio must be adapted to the required working pressure.

This is an example on how air is compressed in a screw compressor. In figure 15 air fills the space between the rotors, but for each turn the space does decreases more and more.

The screw compressor is not equipped with valves and has no mechanical forces that cause unbalance. This means it can work at a high shaft speed and combine a large flow rate with small exterior dimensions. An axial acting force, dependent on the pressure difference between the inlet and outlet, must be taken up by the bearings.

Throughout this process, there is no contact between the screws. This means no wear, total reliability, and non-pulsating air delivery.

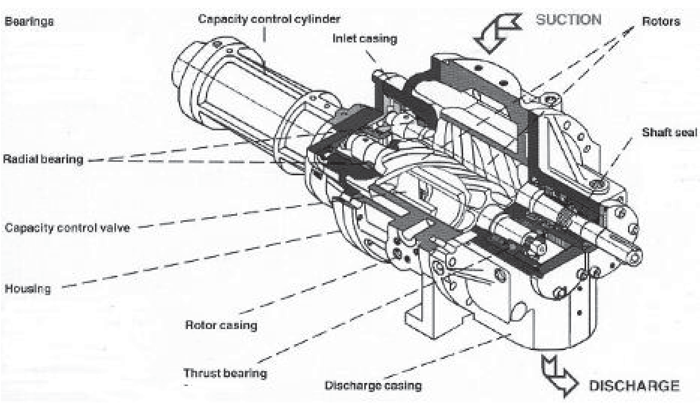


figure 14 Rotary Screw Compressor

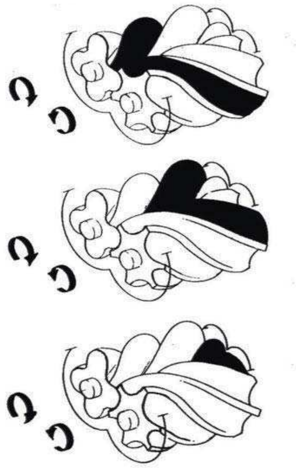


figure 15 Compression process

Centrifugal compressors

The centrifugal compressor, originally built to handle only large volumes of low pressure gas and air, has been developed to enable it to move large volumes of gas with high discharge pressures. However, centrifugal compressors are now most frequently used for medium volume and medium pressure air delivery. One advantage of a centrifugal pump is the smooth discharge of the compressed air.
The centrifugal force utilized by the centrifugal compressor is the same force utilized by the centrifugal pump. The air particles enter the eye of the impeller, designated D

in figure 16. As the impeller rotates, air is thrown against the casing of the compressor. The air becomes compressed as more and more air is thrown out to the casing by the impeller blades. The air is pushed along the path designated A, B, and C. The pressure of the air is increased as it is pushed along this path.

Note in figure 16 that the impeller blades curve forward, which is opposite to the backward curve used in typical centrifugal liquid pumps. Centrifugal compressors can use a variety of blade orientation including both forward and backward curves as well as other designs. There may be several stages to a centrifugal air compressor, as in the centrifugal pump, and the result would be the same; a higher pressure would be produced. The air compressor is used to create compressed or high pressure air for a variety of uses.

Figure 17 shows a single-stage centrifugal compressor. The gas molecules leave the blade tips at high velocity and enter the diffuser passage that encircles the impeller.

This passage of increased area collects and slows the molecules down, guiding them on in a slower mass of higher pressure. From the diffuser the mass moves on to the discharge passage, where the molecules are slowed down even more by the spiral shell diffusing shape of the discharge passage. By the time the gas arrives at the storage container, the high-speed energy of the molecules has been transformed into the stored energy of high pressure.

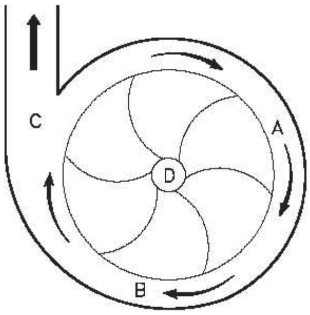


figure 16 Simplified Centrifugal Pump

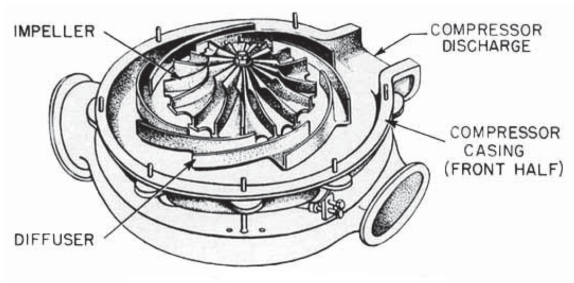


figure 17 Centrifugal Compressor with diffuser

Ejectors

An ejector as shown in figure 18 is a pump-like device that uses the Venturi effect of a converging-diverging nozzle to convert the pressure energy of a motive fluid to velocity energy which creates a low pressure zone that draws in and entrains a suction fluid and then recompresses the mixed fluids by converting velocity energy back into pressure energy. The motive fluid may be a liquid, steam or any other gas. The entrained suction fluid may be a gas, a liquid, a slurry, or a dust-laden gas stream.

The Venturi effect, a particular case of Bernoulli's principle, applies to the operation of this device. Fluid under high pressure is converted into a high-velocity jet at the throat of the convergent-divergent nozzle which creates a low pressure at that point. The low pressure draws the suction fluid into the convergent-divergent nozzle where it mixes with the motive fluid.

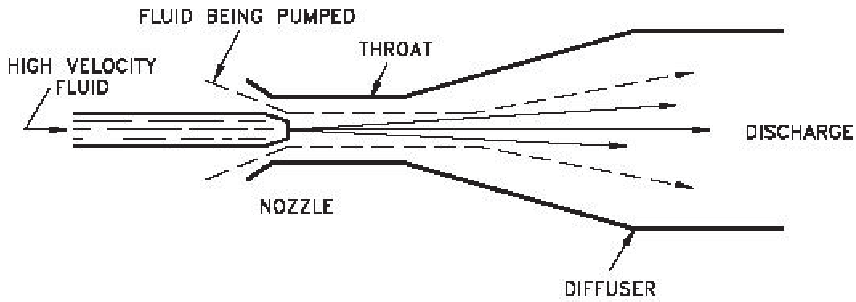


figure 18 Ejector

In essence, the pressure energy of the inlet motive fluid is converted to kinetic energy in the form of velocity head at the throat of the convergent-divergent nozzle. As the mixed fluid then expands in the divergent diffuser, the kinetic energy is converted back to pressure energy at the diffuser outlet in accordance with Bernoulli's principle.

Compressor Coolers

The amount of moisture that air can hold is inversely proportional to the pressure of the air. As the pressure of the air increases, the amount of moisture that air can hold decreases. The amount of moisture that air can hold is also proportional to the temperature of the air. As the temperature of the air increases, the amount of moisture it can hold increases. However, the pressure change of compressed air is larger than the temperature change of the compressed air. This causes the moisture in the air to condense.

Moisture in compressed air systems can cause serious damage. The condensed moisture can cause corrosion, water hammers, and freeze damage; therefore, it is important to avoid moisture in compressed air systems. Coolers are used to minimize the problems caused by heat and moisture in compressed air systems.

Coolers used on the discharge of a compressor are called aftercoolers. Their purpose is to remove the heat generated during the compression of the air. The decrease in temperature promotes the condensing of any moisture present in the compressed air. This moisture is collected in condensate traps that are either automatically or manually drained. If the compressor is multi-staged, there may be an intercooler, which is usually located after the first stage discharge and before the second stage suction.

The principle of the intercooler is the same as that of the aftercoolers. The result is drier, cooler, compressed air. The structure of a particular cooler depends on the pressure and volume of the air it cools. Figure 19 illustrates a typical compressor air cooler. Air coolers are used because drier compressed air helps prevent corrosion and cooler compressed air allows more air to be compressed for a set volume.

Item e in figure 7 shows another type of air cooler. In this type the air is cooled by the air circulating around the tube. The fins on the pipe increase the surface area which increases the cooling effect.

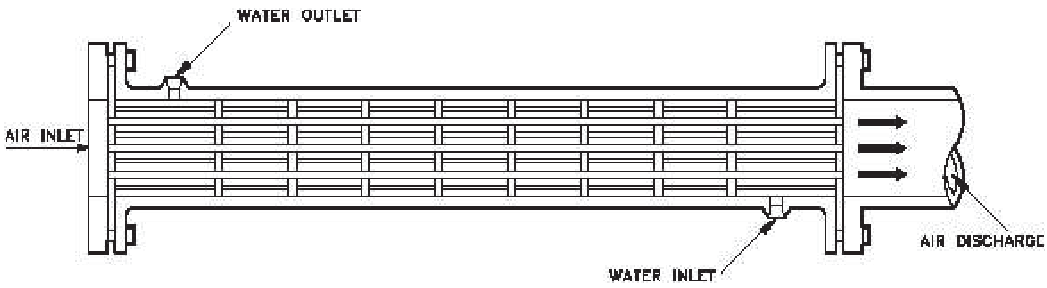


Figure 19 Compressor Air Cooler

Motors

There are many kinds of air motors used for powering tools and mechanisms which use compressed air. These are specially designed units which are very compact and are able to operate at high speeds with built in torque limitation.

Typical designs of air motors include rotary vane, axial piston, radial piston, turbine, V-type, and diaphragm. Rotary vane, axial- and radial-piston air motors are most commonly used for industrial applications. These designs operate with highest efficiency and longevity when using lubricated air.

Unlike steam air cannot, conveniently, be used expansively because the resulting cooling effect would result in freezing of the moisture being carried in the air. If the moisture in the air is removed then the air can be used more flexibly.

The efficiency of air motors based on non-expansion cycles is about 20%. With the efficiency or compressors being about 60% then pneumatic drive systems have efficiencies of less than 12%. This compares unfavourable with internal combustion or electric motor drive systems.

The primary advantages justifying the use of pneumatic drive systems are:

- Safety — air motors can safely be used in locations with explosive risk resulting from ignition sources due to electrical devices
- Convenience — air motors are generally very compact and include built in overload protection
- Capital Costs — air motors are often very low cost units
- Maintenance/Operation — air motors cost little in maintenance and can be easily operated by semi-skilled operatives
- Installation — most ships have compressed air systems installed

Hazards of Compressed Air

People often lack respect for the power in compressed air because air is so common and is often viewed as harmless. At sufficient pressures, compressed air can cause serious damage if handled incorrectly.

To minimize the hazards of working with compressed air, all safety precautions should be followed closely. Small leaks or breaks in the compressed air system can cause minute particles to be blown at extremely high speeds.

Always wear safety glasses when working in the vicinity of any compressed air system. Safety goggles are recommended if contact lenses are worn. Compressors can make an exceptional amount of noise while running.

The noise of the compressor, in addition to the drain valves lifting, creates enough noise to require hearing protection.

Pressurized air can do the same type of damage as pressurized water. Treat all operations on compressed air systems with the same care taken on liquid systems.

Closed valves should be slowly cracked open and both sides should be allowed to equalize prior to opening the valve further. Systems being opened for maintenance should always be depressurized before work begins.

Great care should be taken to keep contaminants from entering air systems. This is especially true for oil. Oil introduced in an air compressor can be compressed to the point where detonation takes place in a similar manner as that which occurs in a diesel engine. This detonation can cause equipment damage and personnel injury.

Safety regulations applicable for air compressors on board

1. Each ship must be equipped with two starting air compressors which can work independently from each other. Each starting air compressor must be able to fill the starting air receivers from complete empty till the required working pressure within 1 hour.
2. The total capacity of the starting air receivers must be sufficient to start a reversible main engine alternating ahead and astern 12 times. For main engines equipped with a pitch propeller or a clutch gearbox this may be reduced to 6 times.
3. Each air receiver must be equipped with the following equipments:
    - Pressure gauge with a red marking indicating the allowable working pressure
    - Safety valve with a setting of 1.2 times the working pressure
    - Water drain at the lowest point
    - Each pipe connection to the air receiver must have a valve directly mounted on the receiver
    - Each receiver should as far as possible be equipped with a manhole or inspection opening
4. Each stage of an air compressor must be equipped with a safety valve with a setting of the working pressure at that point.
5. Non return valves must be fitted in the discharge line of each compressor to the receivers.
6. If the bridge control for the main engine is pneumatic each compressor feeding this system must have an independent automatic starting and stopping device.

The reason that the set point for the safety valve of the air receivers is 1.2 times the working pressure is that the temperature in the engine room is not constant. If the setting would be equal to the working pressure and the temperature would rise the safety valves would open. The setting of the safety valve of the last stage of the compressor must avoid that the pressure will become higher as the working pressure.

Questions Air compressors

1. What do we mean with the compression ratio of a compressor stage?
2. For what reason is the compression ratio of the stages in a 2 stage compressor kept equal?
3. A two stage air compressor has equal compression ratios for each stage. The suction pressure is 1 bar, the gauge on the discharge line reads 35 bar. Determine the pressure indicated by the gauge on the inter cooler.
4. Draw the theoretical p-v diagram for a two stage air compressor with inter cooling.
   The data of the compressor are:

Displaced volume first stage	: 1 dm^3
Clearance volume first stage	: 0.02 dm3
Displaced volume second stage	: 0.2 dm3
Clearance volume second stage	: 0.01 dm3
Suction pressure (abs.)	: 1 bar
Intermediate pressure (abs.)	: 5.6 bar
Discharge pressure (abs.)	: 31 bar

5. Explain by drawing an indicator diagram that by cooling the air between first and second stage the total work is reduced.
6. Explain why an air compressor should be started without load.
7. Explain why a reciprocating air compressor cannot deliver air directly to a consumer.
8. Explain the working principle of a rotary slide vane compressor.
9. Can the same construction as a rotary slide vane compressor be used as air motor? Support your answer.
10. What causes the clearance volume effect?
11. Explain the consequence of the clearance volume effect.
12. From a single stage air compressor the following data is known:

Discharge pressure (abs.)	p_d	= 7 bar
Suction pressure (abs.)	p_s	= 1 bar
Polytrophic exponent n	n	= 1.3
Rotation c	c	= 12 Hz
Displaced volume	V_s	= 0.002 m3
Clearance volume	V_c	= 0.05 V_s

   Calculate: Volumetric efficiency
   Work done
   Hypothetical power
13. List the safety requirements for starting air compressors.
14. Why are air receivers equipped with water drains?
15. Why can oil in an air compressor be dangerous?
16. Describe the working principle of an ejector.
17. What is the function of the diffuser in a centrifugal compressor?
18. Why must the compressor in fig. 8 have a separate lubricator on the intake?

Pumps

"Head" is a very convenient term in defining pumps. Pressure is not as convenient a term because the amount of pressure that the pump will deliver is dependent upon the weight (specific gravity) of the liquid being pumped, the specific gravity changes with the fluid temperature and concentration.

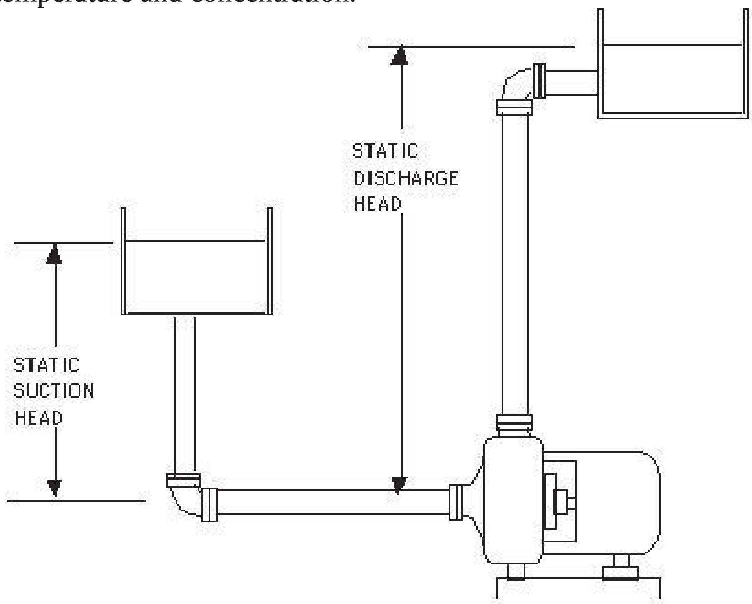


Figure 1 Static head

Figure 1 shows what is meant by static discharge head. Please note that this is always measure from the center-line of the pump to the highest liquid level.

To calculate the head accurately the total head on both the suction and discharge sides of the pump must be looked at. In addition to the static head there is a second head caused by resistance in the piping, fittings and valves called friction head and a third head caused by any pressure that might be acting on the liquid in the suction or discharge tanks including atmospheric pressure. This third head is called "surface pressure head".

Once all of these heads are known it becomes simple. Subtracting the suction head from the discharge head gives the head that is remaining and is the amount of head that the pump must be able to generate at its rated flow:

System head = total discharge head - total suction head or H = hd – hs

The total discharge head is made from three separate heads:

hd = hsd + hpd + hfd

hd = total discharge head
hsd = discharge static head
hpd = discharge surface pressure head
hfd = discharge friction head

The total suction head also consists of three separate heads:

hs = hss + hps – hfs

hs = total suction head
hss = suction static head
hps = suction surface pressure head
hfs = suction friction head

Calculations are made in either "meters of liquid, gauge" or "meters of liquid, absolute". Absolute means that you have added atmospheric pressure (head) to the gauge reading. Normally head readings are made in gauge readings and we switch to the absolute readings only when we want to calculate the net positive suction head available (NPSHA) to find out if the pump is going to cavitate.

Figure 2 demonstrates that the discharge head is still measured to the liquid level, but note that it is now below the maximum height of the piping.

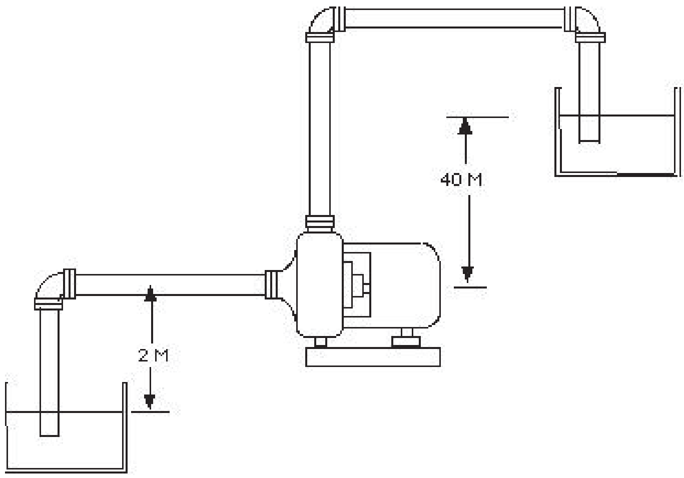


figure 2 Siphon effect

Although the pump must deliver enough head to get up to the maximum piping height it will not have to continue to deliver this head when the pump is running because of the "siphon effect". There is of course a maximum siphon effect. It is derived from the formula to convert pressure to head:

$$\text{Head(meters} = \frac{\text{pressure(bar)} \times 9.8}{\text{specific gravity}}$$

Since atmospheric pressure at sea level is one bar the maximum siphon distance is 9.8 meters if the friction in the piping is ignored.

Total suction head calculation

The suction head is negative because the liquid level in the suction tank is below the centerline of the pump e.g.:

$hss = - 2$ meters

The suction tank is open so the suction surface pressure equals atmospheric pressure:

$hps = 0$ meters gauge

In these examples no calculation of the suction friction head itself will be done. To calculate the friction head you can look at a chart that shows the equivalent length of pipe for each of the fittings and add this number to the length of the piping in the system to determine the total friction loss. An example is given in figure …

For this example, the total friction head on the suction side of the pump is:

$hfs = 1.5$ meters at rated flow

The total suction head will be a gauge value because atmosphere was given as 0:

$hs = hss + hps - hfs = - 2 + 0 - 1.5 = - 3.5$ meters of liquid gauge at rated flow

The total discharge head calculation is similar

The static discharge head is:

$hsd = 40$ meters

The discharge tank is also open to atmospheric pressure, so:

$hpd = 0$ meter, gauge

Suppose the discharge friction head is:

$hfd = 7$ meters at rated flow

The total discharge head is:

$hd = hsd + hpd + hfd = 40 + 0 + 7 = 47$ meters of liquid gauge at rated flow

The total system head calculation becomes:

$Head = hd - hs = 47 - (-3.5) = 50.5$ meters of liquid at rated flow

The next example (figure 3) involves a few more calculations.

In this example pumping takes place from a vacuum receiver that is very similar to the hotwell found in many condenser applications

Specifications:

- Transferring 300 m3/hr weak acid from the vacuum receiver to the storage tank

- Specific Gravity of the acid = 0.98
- Viscosity = equal to water
- Piping = all 150 mm Schedule 40 steel pipe
- Discharge piping rises 15 meters vertically above the pump centerline and then runs 135 meters horizontally. There is one 90° elbow in this line
- Suction piping has 1.5 meters of pipe, one gate valve, and one 90° elbow all of which are 150 mm in diameter.
- The minimum level in the vacuum receiver is 2 meters above the pump centerline.
- The pressure on top of the liquid in the vacuum receiver is 500 mm of mercury, vacuum.

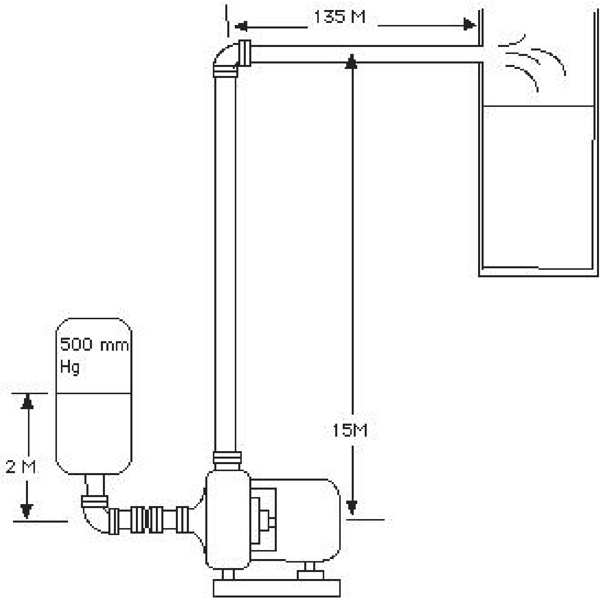


figure 3 Closed suction container

To calculate suction surface pressure use the following formula:

$$\text{millimeters of mercury} \times \frac{0.014}{\text{specific gravity}} = \text{meters of liquid}$$

Begin by dividing the system into two different sections using the pump as the dividing line.

Total suction head calculation

The suction side of the system shows a minimum static head of 2 meters above suction centerline. Therefore, the static suction head is:

hss = 2 meters

Using the first conversion formula, the suction surface pressure is:

hps = 500 x (0.014/0.98) = 7.14 meters of vacuum

The suction friction head fs, equals the sum of all the friction losses in the suction line. The metric pipe friction loss table in figure 5 shows that the friction loss in the 150 mm. pipe at 300 m^3/hr. is 9 meters per 100 meters of pipe.

$$\text{In 1.5 meters of pipe friction loss} = \frac{1.5}{100} \times 9 = 0.14 \text{ meters}$$

In figure 6 the equivalent values can be found for the fittings:

Fitting	Equivalent length of straight pipe
150 mm normal bend elbow	3.4 meters
150 mm Gate valve	2.1 meters

In a pumping application there could be other valves and fittings that experience friction losses:

- Check valves
- Foot valves
- Strainers
- Sudden enlargements
- Shut off valves
- Entrance and exit losses
- Etc...

The loss in the suction fittings becomes:

in 5.5 meters of pipe friction loss = 5.5/100 x 9 = 0.50 meters

The total friction loss on the suction side is:

hfs = 0.14 + 0.50 = 0.64 meters at 300 m^3/hr

The total suction head then becomes:

hs = hss + hps - hfs = 2 - 7.14 - 0.64 = - 5.78 meters gauge at 300 m^3/hr

The total discharge head calculation:

- Static discharge head = hsd = 15 meters
- Discharge surface pressure = hpd = 0 meters gauge
- Discharge friction head = hfd = sum of the following losses :

Friction loss in 150 mm pipe at 300 m3/hr, from the charts is 9 meters per hundred meter of pipe.

$$\text{In 150 meters of pipe the friction loss} = \frac{150}{100} \times 9 = 13.5 \text{ meters}$$

$$\text{Friction loss in a 150 mm elbow} = \frac{3.4}{100} \times 9 = 0.31 \text{ meters}$$

The discharge friction head is the sum of the above losses, that is:

hfd = 13.5 + .31 = 13.81 meters at 300 m^3/hr

The total discharge head then becomes:

hd = hsd + hpd + hfd = 15 + 0 + 13.81 = 28.81 meters at 300 m^3/hr.

Total system head calculation:

H = hd - hs = 28.81 - (-5.78) = 34.59 meters at 300 m^3/hr

The next example will be the same as the one of figure 3 except that there is an additional 3 meters of pipe and another 90° flanged elbow in the vertical leg. The total suction head will be the same as in the previous example. Take a look at figure 4:

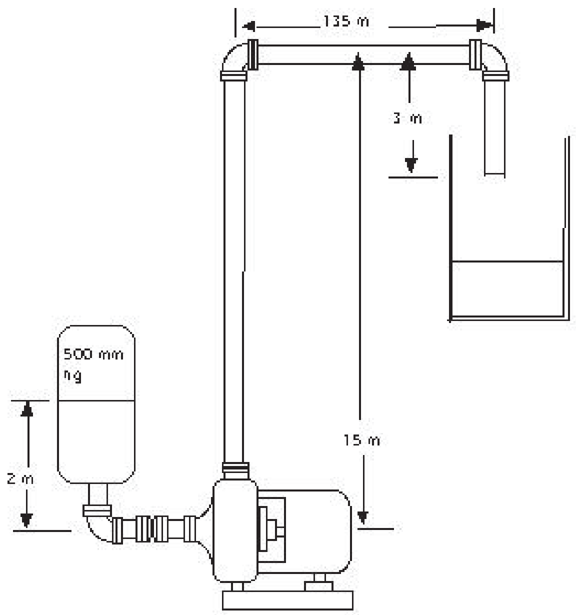


figure 4 Extra piping installed

Nothing has changed on the suction side of the pump so the total suction head will remain the same:

hs = - 5.78 meters at 300 m^3/hr

Total discharge head calculation

The static discharge head (hsd) will change from 15 meters to 12 meters since the highest liquid surface in the discharge is now only 12 meters above the pump centerline. This value is based on the assumption that the vertical leg in the discharge tank is full of liquid and that as this liquid falls it will tend to pull the liquid up and over the loop in the pipe line. This arrangement is called a siphon leg.

The discharge surface pressure is unchanged:

hpd = 0 meters

The friction loss in the discharge pipe will be increased by the additional 3 meters of pipe and the additional elbow.

- In 3 meters of pipe the friction loss = 3 /100 x 9 = 0.27 meters
- The friction loss in the additional elbow = 3.4 /100 x 9 = 0.31 meters

The friction head will then increase as follows:

hfd = 0.27 + 0.31 = 0.58 at 300 m^3/hr.

The total discharge head becomes:

hd = hsd + hpd + hfd = 12 + 13.81+ 0 + 0.58 = 26.39 meters at 300 m3/hr

Total system head calculation

Head = hd – hs = 26.39 - (-5.78) = 32.17 meters at 300 m3/hr.

Calculating the friction loss in metric size piping

Some notes for the metric pipe friction chart shown below

- The chart is calculated for fresh water at 15°C.
- Use actual bores rather than nominal pipe size.
- For stainless steel pipe multiply the numbers by 1.1.
- For steel pipe multiply the numbers by 1.3
- For cast iron pipe multiply the numbers by 1.7
- The losses are calculated for a fluid viscosity similar to fresh water

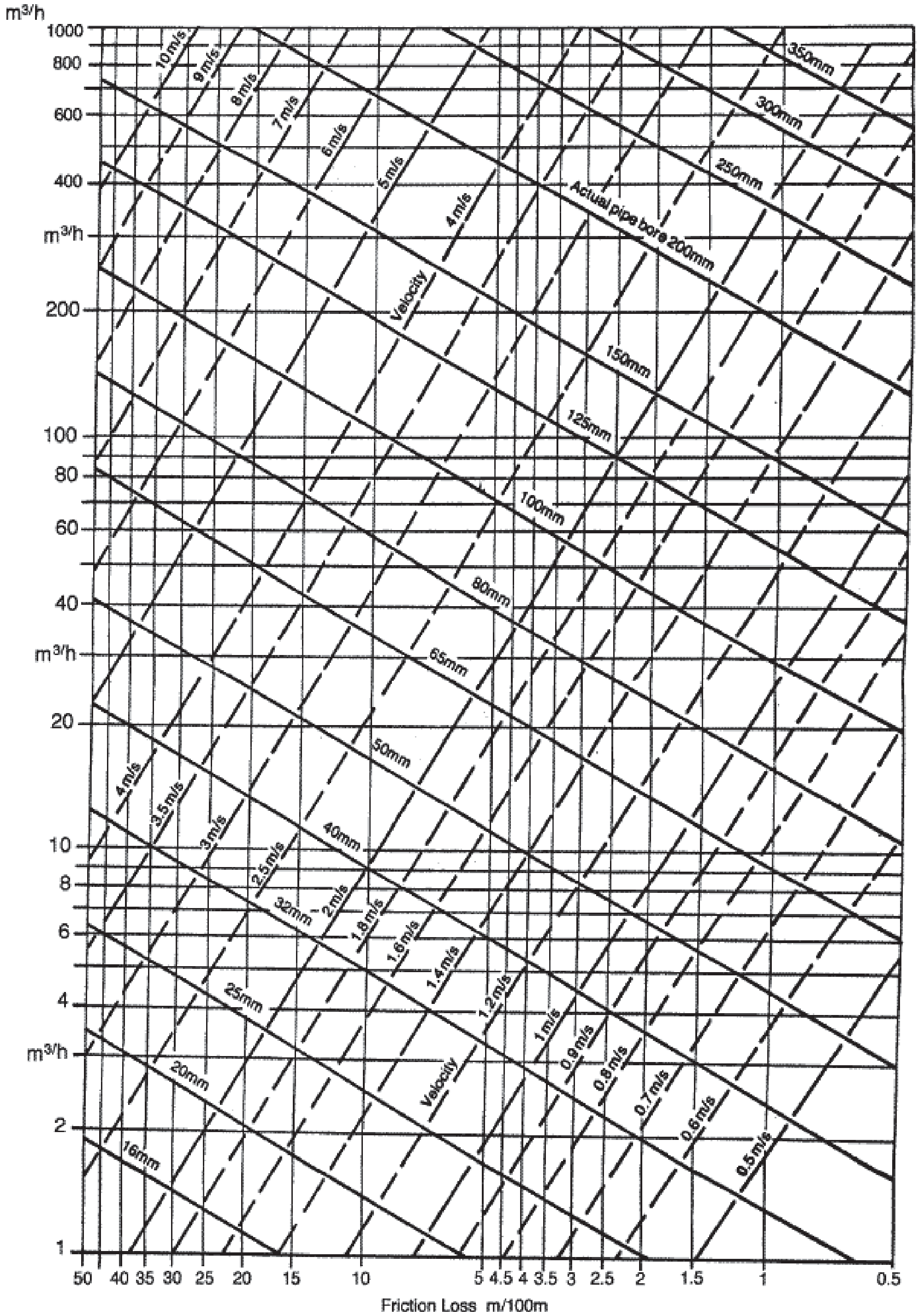
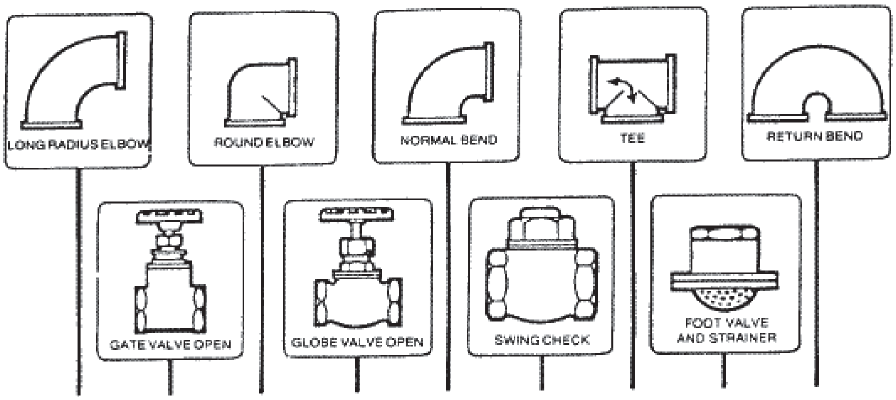


figure 5 Friction loss indicated as m/100 at different pipe sizes and velocities



Pipe size mm — Equivalent length of straight pipe in metres, for calculating friction loss

20	0.3	0.3	0.6	6.7	0.5	1.5	1.5	1.5	1.5
25	0.3	0.3	0.8	8.2	0.5	2.0	1.8	2.3	2.0
32	0.3	0.6	0.9	11.3	0.8	2.6	2.4	2.7	2.6
40	0.4	0.6	1.1	13.4	0.9	3.1	2.7	3.4	3.1
50	0.5	0.8	1.4	17.4	1.1	4.0	3.4	4.6	4.0
65	0.6	0.9	1.7	20.1	1.4	5.2	4.3	5.5	4.6
80	0.8	1.1	2.1	26.0	1.5	6.1	5.2	6.7	5.5
100	1.1	1.5	2.7	34.0	2.1	8.2	6.7	8.8	7.3
125	1.2	1.8	3.7	43.0	2.7	10.0	8.2	11.0	9.5
150	1.5	2.1	4.3	49.0	3.4	12.2	10.0	14.0	11.0
200	2.1	3.1	5.5	67.0	4.3	16.5	13.4	18.0	15.0
250	2.4	3.7	7.3	85.4	5.5	20.0	16.5	22.0	19.0
300	3.1	4.3	8.5	98.0	6.7	24.4	20.0	27.4	23.0

figure 6 Chart to calculate the loses through various type valves and fittings

Suction lift

Suction lift exists when the source of supply is below the centerline of the pump. Thus the Static suction lift is the vertical distance in meter from the centerline of the pump to the free level of the liquid to be pumped as can be seen in figure 2.

Centrifugal pumps have the capability of creating a vacuum in a suction pipe which enables them to suck water from below their setting level.

Maximum suction lift is determined by the formula: $H_{max} = A - NPSH - H_f - H_v - H_s$

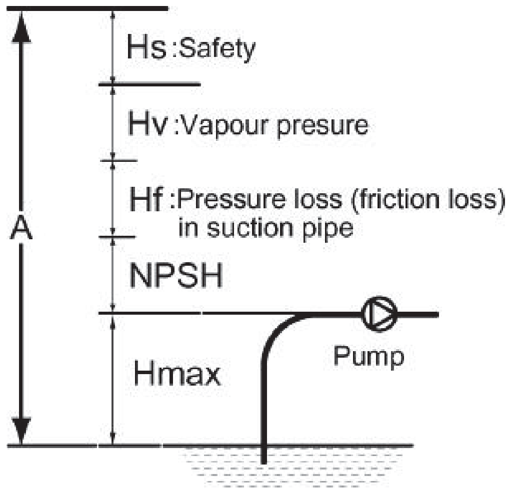


Figure 7 Suction lift

Geodesic suction height (A)

This is the height created by the hydrostatic pressure. When the suction pipe of a pump will suction from an open liquid container and the pump would be able to create complete vacuum inside, which is not possible in reality, the atmospheric pressure would be able to press the liquid upwards. The height the liquid could be pressed up is:

$h = p/(\rho.g)$ meter. if : $g = 9.81\ m/s^2$
$\rho = 1{,}000\ kg/m^3$
$p = 1bar\ (1.10^5\ Pa)$

the hydrostatic height would be $1.10^5/(1{,}000*9.81) = 10.19$ m

Friction loss (Hf)

This is created in the suction pipe, strainer and valves. The value increases with increasing flow thereby reducing the available suction lift. This is identical to the dynamic head on the discharge side of the pump

Vapour pressure (H_v)

The vapour pressure of a fluid is the pressure at which the fluid will boil at ambient temperature. If the pressure within a fluid falls below the vapour pressure of the fluid, gas bubbles will form within the fluid (local boiling of the fluid will occur).

If a fluid which contains gas bubbles is allowed to move through a pump, it is likely that the pump will increase the pressure within the fluid so that the gas bubbles collapse. This will occur within the pump and reduce the flow of delivered fluid. The collapse of the gas bubbles may cause vibrations which could result in damage to the pipe work system or the pump. This effect is known as cavitation.

To avoid cavitation the pressure within the fluid must be higher than the fluid vapour pressure at all times.

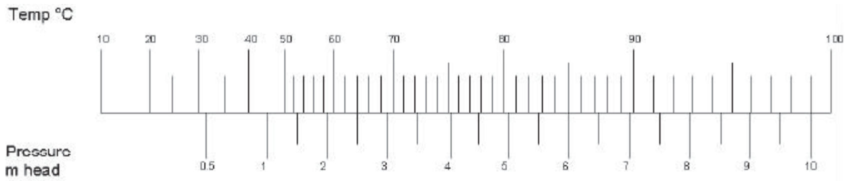


figure 8 Vapour values for water

Safety margin (H_S)

A safety margin of usually 0.5 to 1 m is being acceptable.

Net Positive Suction Head (NPSH)

Net positive suction head is the term that is usually used to describe the absolute pressure of a fluid at the inlet to a pump minus the vapour pressure of the liquid. The resultant value is known as the Net Positive Suction Head available. The term is normally shortened to the acronym $NPSH_a$, the ‘a’ denotes ‘available’.

A similar term is used by pump manufactures to describe the energy losses that occur within many pumps as the fluid volume is allowed to expand within the pump body. This energy loss is expressed as a head of fluid and is described as $NPSH_r$ (Net Positive Suction Head requirement) the ‘r’ suffix is used to denote the value is a requirement.

Different pumps will have different NPSH requirements dependant on the impellor design, impellor diameter, inlet type, flow rate, pump speed and other factors.

A pump performance curve will usually include a NPSH requirement graph expressed in metres head so that the $NPSH_r$ for the operating condition can be established.

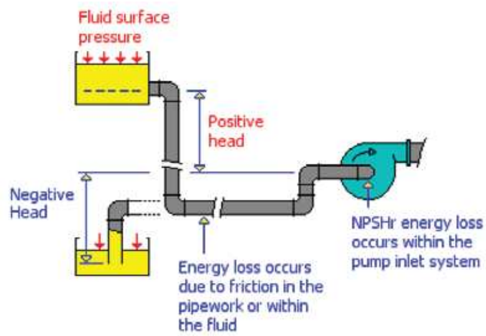


figure 9 Finding NPSH

The fluid pressure at a pump inlet will be determined by the pressure on the fluid surface, the frictional losses in the suction pipework and any rises or falls within the suction pipework system.

The elements used to calculate $NPSH_a$ are all expressed in absolute head units. The $NPSH_a$ is calculated from:

$$H_{\text{fluid surface}} - H_{\text{negative head}} - H_{\text{pipework friction loss}} - H_{\text{vapour pressure}}$$

or

$$H_{\text{fluid surface}} + H_{\text{positive head}} - H_{\text{pipework friction loss}} - H_{\text{vapour pressure}}$$

In a system where the fluid needs to be lifted to the pump inlet, the negative head reduces the motive force to move the fluid to the pump. In these instances it is essential to size the supply pipe work and isolating valves generously so that high frictional losses do not reduce the $NPSH_a$ below the $NPSH_r$.

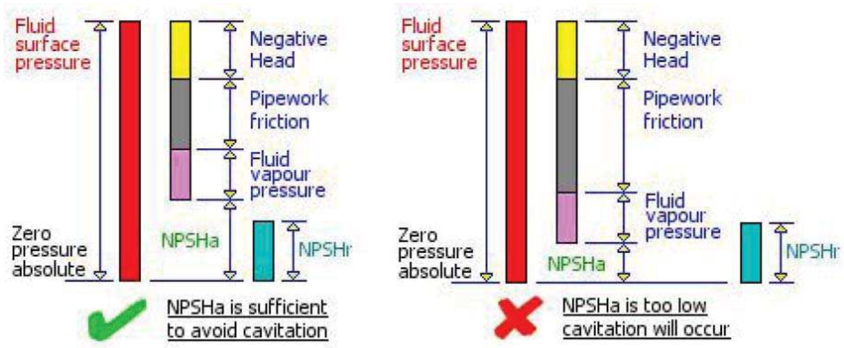


figure 10 Different NPSH conditions

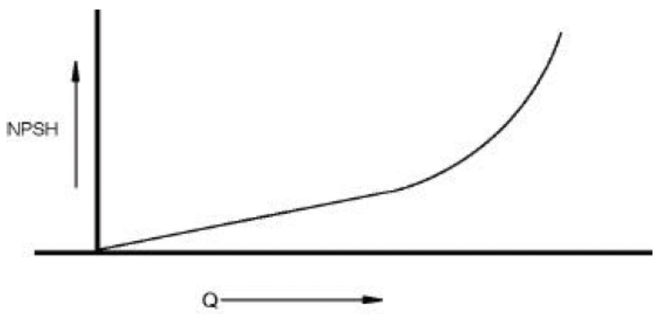


figure 11 NPSH development

Some general points about suction conditions are as follows:

- It is good practice to keep suction pipes as short as is practical
- Suction pipes must be totally airtight. If there are any leaks the pump will be unable to create the vacuum condition for suction to occur
- Suction pipes must be straight and laid to rise continuously to the pump. If there are any leaks in the pipe air pockets will form and the system will become air locked
- Suction pipes must be generously sized, one size larger than the delivery pipe being standard practice
- All suctions should be fitted with foot valves

Horizontal shaft centrifugal pump construction

These are by far the most common generic type of electric or engine powered pumps. Figure 12 shows a typical volute-centrifugal pump in cross section. In this type of pump the casing and frame are usually cast iron or cast steel, while the impeller may be bronze or steel.

Critical parts of the pump are the edges of the entry and exit to the impeller as a major source of loss is back-leakage from the exit of the impeller around the front of it to the entry. To prevent this, good quality pumps, including the one in the diagram, have a closely fitting wear ring fitted into the casing around the front rim of the impeller; this is subject to some wear by particulate matter in the water and can be replaced when the clearance becomes large enough to cause significant loss of performance.

Another wearing part is a stuffing box packing where the drive shaft emerges from the back of the impeller casing. This needs to be periodically tightened to minimize leakage, although excessive tightening increases wear of the packing.

The back of the pump consists of a bearing pedestal and housing enclosing two deep-groove ball-bearings. This particular pump is oil lubricated, it has a filler, dip-stick and drain plug.

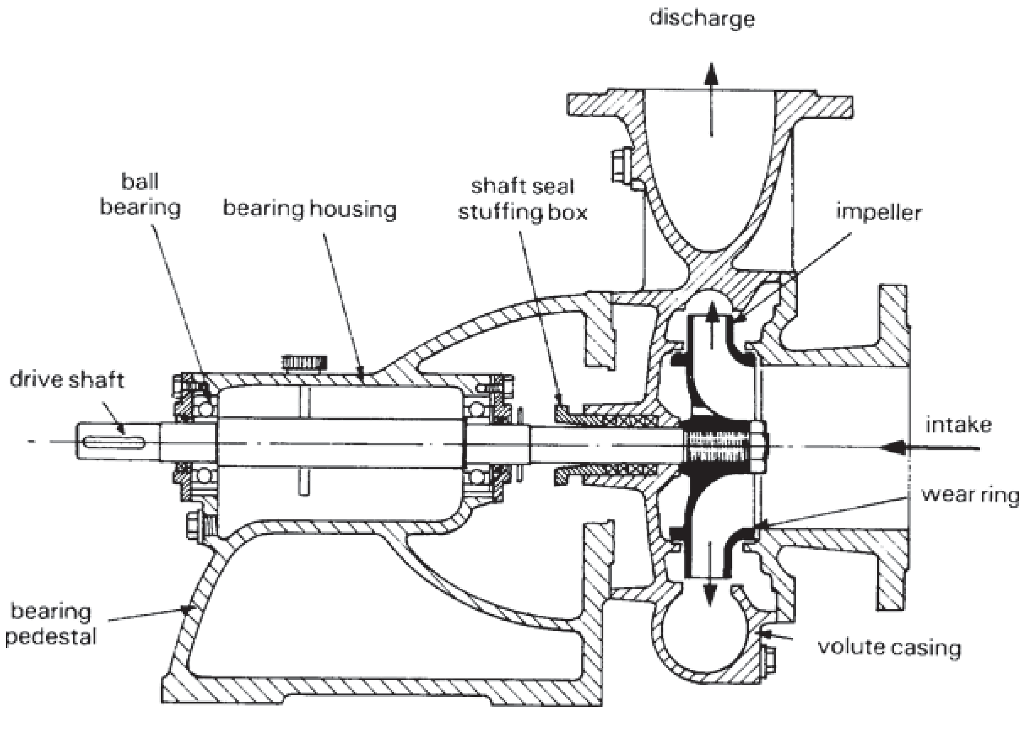


figure 12 Horizontal centrifugal pump

Working Mechanism of a Centrifugal Pump

Its purpose is to convert energy of a prime mover (an electric motor or turbine) first into velocity or kinetic energy and then into pressure energy of a fluid that is being pumped. The energy changes occur by virtue of two main parts of the pump, the impeller and the volute or diffuser. The impeller is the rotating part that converts driver energy into the kinetic energy. The volute or diffuser is the stationary part that converts the kinetic energy into pressure energy.

Fluid enters the inlet duct (D). As the shaft (A) rotates, the impeller (B), which is connected to the shaft, also rotates. The impeller consists of a number of blades that project the fluid outward when rotating. This centrifugal force gives the fluid a high velocity.

Next, the moving fluid passes through the pump case (C) and then into the volute (E). The volute chamber has a uniformly increasing area. This increasing area decreases the fluids velocity, which converts the velocity energy into pressure energy. Two characteristics a pump produces are pressure head and volumetric flow.

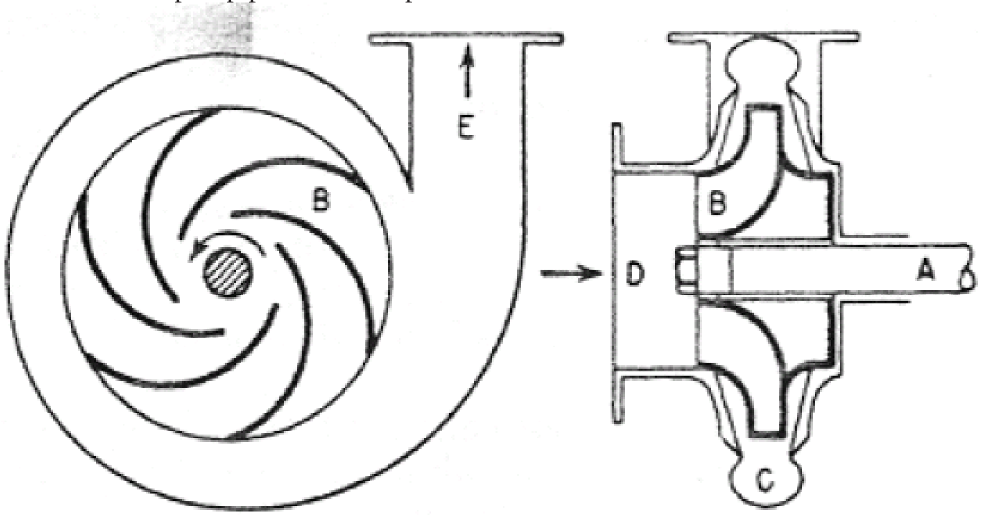


Figure 13 Cross section of a centrifugal pump

Speed conditions at the impeller

With the current of a liquid through the channels of a rotary impeller it is to be differentiated between absolute and relative movement. The movement of the liquid particles is called absolute, if they can be noticed by an outside of the impeller standing observer. The relative movement of the liquid particles notices an observer, who moves with the impeller.

In figure 14 speed conditions in the impeller are represented for a backwards curved blading. The current enters with the relative velocity w_1 the blade channel. With an inner diameter D_1 and rotation n the impeller has the peripheral speed $u_1 = \pi D_1 n$. From the vectorial addition of the relative velocity w_1 and the peripheral speed u_1 the absolute speed c_1 results. The shape of the blades at the inner diameter of the impeller is normally designed in such a way that the liquid will enter with a radial flow. Giving $c_1 = c_{m1}$ an angle α_1 between c_1 and u_1 of 90^0.

At the outer diameter of the impeller the liquid has the peripheral speed $u_2 = \pi D_2 n$ and the relative speed w_2. As resulting the absolute outgoing speed c_2 results, which is substantially larger due to the transfer of energy than c_1.

For purposes of continuation the design at the outer diameter of the impeller is such that the absolute radial outgoing speed $c_{m2} = c_{m1} = c_1$. Because of this there will be no acceleration of the liquid in the radial direction and therefore no force will act on the blades in that direction.

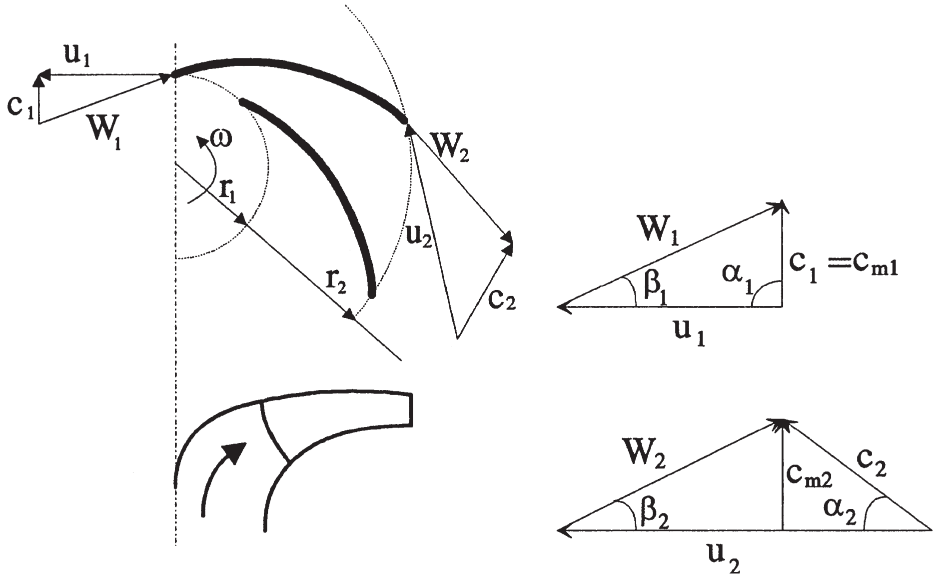


figure 14 Speed conditions in the impeller

Generation of Centrifugal Force

The liquid enters the suction nozzle and then into eye (center) of the impeller. When the impeller rotates, it spins the liquid sitting in the cavities between the vanes outward and provides centrifugal acceleration.

As liquid leaves the eye of the impeller a low-pressure area is created causing more liquid to flow toward the inlet. Because the impeller blades are curved, the fluid is pushed in a tangential and radial direction by the centrifugal force. This force acting inside the pump is the same one that keeps water inside a bucket that is rotating at the end of a string.

The amount of energy given to the liquid is proportional to the *velocity* at the edge or blade tip of the impeller. The faster the impeller revolves or the bigger the impeller is, then the higher will be the velocity of the liquid at the blade tip and the greater the energy imparted to the liquid.

Therefore, the pressure developed is equal to the velocity energy at the periphery of the impeller expressed by the formula of Euler:

$p_E = \rho\ (u_2 * c_2 * \cos\alpha_2 – u_1 * c_1 * \cos\alpha_1)$

as $\alpha_1 = 90^0$, $\cos \alpha_1 = 0$ we can state that the ideal developed pressure is:
$p_E = \rho * u_2 * c_2 * \cos\alpha_2$ in N/m^2

The ideal pressure head H_E becomes:

$H_E = p_E /(\rho * g) = (u_2 * c_2 * \cos\alpha_2)/g$ in meter

Thickness of the blades

The formula of Euler is only applicable for an ideal pump with an indefinite number of very thin blades which will have no influence on the movement of the liquid. However in practice the impeller only has a limited number of blades with a certain thickness. The thickness of the blades will reduce the inlet and outlet openings of the impeller. The flow of the liquid in the impeller will not be ideal. Part of the liquid will collide with the blades when entering the impeller resulting in a disturbance of the ideal flow.

The influence of the thickness, number and shape of the blades on the ideal pressure head can only be determined by complicated experiments. Results of these experiments have shown that the factor (k) is in between 0.7 – 0.85.

The reduced head is called the theoretical pressure head:

$$p_{TH} = k.p_E \qquad \text{or} \qquad H_{TH} = k.H_E$$

The theoretical pressure or head is the pressure of the liquid when leaving the impeller.

Influence of the blade angle $ß_2$ on the pressure head

The angle of outlet $ß_2$ can theoretically be selected freely within a wide range. An angle $ß_2 > 90°$ leads to backwards curved blades. $ß_2 = 90°$ means radial ending blades and $ß_2 < 90°$ means forward curved blades. With equation and velocity triangles results that the specific blade work is the larger, the smaller $ß_2$ is.

From figure 15 it can be determined that a small angle $ß_2$ means a large absolute speed c_2. The transformation of this speed energy in pressure energy in the peeler is connected with substantial losses due to high turbulence. It is better to chose $ß_2 > 90°$ for getting a lower c_2.

A large angle $ß_2$ has the disadvantages that it requires, to have the same delivery head, a larger circumferential speed and so it causes larger wheel friction losses.

Because of the higher difference of pressure between entrance and exit of the impeller, larger gap leakages are caused. However, these disadvantages are neglect able in relation with the crucially better hydraulic efficiency. Therefore in centrifugal pumps only backwards curved blades with angles of outlet $ß_2 = 140°$-$160°$ are used.

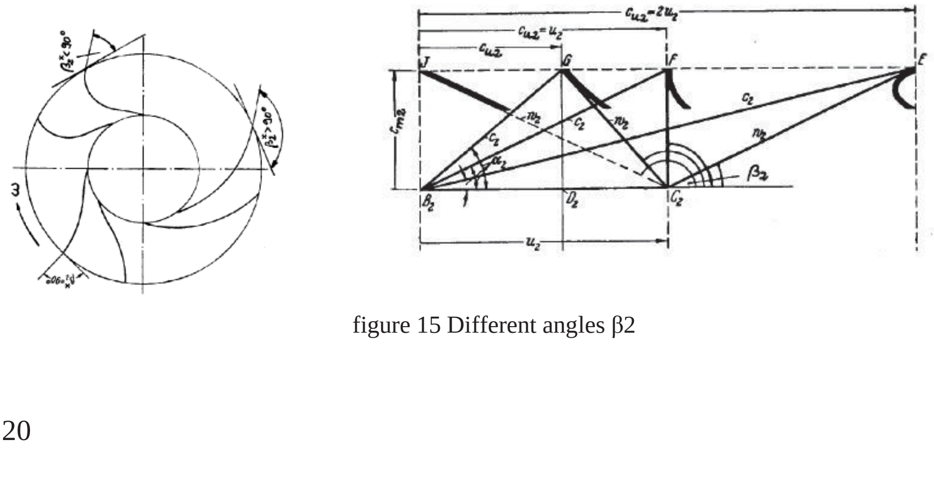


figure 15 Different angles β2

Conversion of Kinetic Energy to Pressure Energy

This kinetic energy of a liquid coming out of an impeller is harnessed by creating a resistance to the flow. The first resistance is created by the pump volute (casing) or diffuser that catches the liquid and slows it down. In the discharge nozzle, the liquid further decelerates and its velocity is converted to pressure according to Bernoulli's principle.

Therefore, the head (pressure in terms of height of liquid) developed is approximately equal to the velocity energy at the periphery of the impeller expressed by the following formula:

$$p_{th} = k * p_{euler} = k * \rho * u_2 * c_2 * \cos\alpha_2$$

For $c_2 * \cos\alpha_2$ we can also write c_{2u} meaning c_2 in the direction of u_2. This gives the formula:

$$p_{th} = k * p_{euler} = k * \rho * u_2 * c_{2u}$$

A part of the pressure will be lost due to the pressure difference between discharge and suction side of the impeller. The gap between the impeller en the casing should therefore be as small as possible.

In the volute or diffuser also losses will take place due to turbulence and resistance.

To find the effective pressure head of the pump the friction and impulse losses must be deducted from the theoretical pressure head. The friction losses are equal to the power of 2 with the speed of the liquid. The impulse losses are difficult to calculate and are determined by testing.

The effective pressure head (p_{eff}) is the pressure the pump can deliver without mounting suction and discharge piping to the pump. The ratio between the effective pressure head and the theoretical pressure head is called the hydraulic efficiency η_{hpump} of the pump.

$$\eta_{h\ pump} = p_{eff}/p_{th} \qquad \text{or} \qquad \eta_{h\ pump} = H_{eff}/H_{th}$$

Average values for the hydraulic efficiency are 0.75 – 0.9. When the pump is in operation the effective pressure can be found by taking the difference between the reading of the suction and discharge pressure gauge on the pump (minus the static height between the pressure gauges).

Capacity

Capacity means the flow rate with which liquid is moved or pushed by the pump to the desired point in the process. It is commonly measured in cubic meters per hour (m3/hr). The capacity usually changes with the changes in operation of the process. For example, a boiler feed pump is an application that needs a constant pressure with varying capacities to meet a changing steam demand.

The capacity depends on a number of factors like:

- Process liquid characteristics i.e. density, viscosity
- Size of the pump and its inlet and outlet sections
- Impeller size
- Impeller rotational speed RPM
- Size and shape of cavities between the vanes
- Pump suction and discharge temperature and pressure conditions

For a pump with a particular impeller running at a certain speed in a liquid, the only items on the list above that can change the amount flowing through the pump are the pressures at the pump inlet and outlet. The effect on the flow through a pump by changing the outlet pressures is graphed on a pump curve.

The theoretical capacity Q_{th} of an impeller pump is the amount of liquid, flowing through the impeller. Due to leakage of water inside the pump from discharge to suction the effective capacity Q_{eff} will be smaller. This is indicated by the volumetric efficiency η_{vol}.

To determine the theoretical capacity you have to determine the total opening on the circumference of the impeller. For the outside diameter of the impeller this is:

$$A_2 = (\pi*D_2 - z*a_2) * b_2$$

A_2 = total opening on outside diameter of impeller
D_2 = outer diameter of impeller
z = number of blades
a_2 = thickness of the blades
b_2 = wide of the blades

The theoretical capacity $Q_{th} = A_2 * c_{2rad} = (\pi*D_2 - z*a_2) * b_2 * c_{2rad}$

The effective capacity of the impeller pump $Q_{eff} = \eta_{vol} * (\pi*D_2 - z*a_2) * b_2 * c_{2rad}$

Power

The theoretic power P_{th} to drive a pump can be determined using the theoretical capacity Q_{th} and the pressure p_{th}. The theoretical capacity is the amount of liquid that is flowing through the impeller and is set in motion with a certain velocity equal to the theoretical pressure.

The driving motor must however deliver a higher power due to mechanical losses like friction in bearings. The mechanical efficiency η_{mech} shows how much power is delivered to the impeller:

$$P_{th} = p_{th} * Q_{th} \text{ and } P_{eff} = P_{th} / \eta_{mech}$$

Combination of these formulas gives:

$$P_{eff} = p_{th} * Q_{th} / \eta_{mech} = (p_{eff} * Q_{th}) / (\eta_{h\,pump} * \eta_{vol} * \eta_{mech})$$

The total of $\eta_{h\ pump}$ * η_{vol} * η_{mech} is the pump efficiency η_{pump} resulting in:

$$P_{eff} = (p_{eff} * Q_{th}) / \eta_{pump}$$

The pump efficiency of an impeller pump can vary from 0.25 to 0.8.

Q-H curve

The relation between p_{eff} at different capacities for a pump is displayed by the maker in a so called Q-H curve. This curve is applicable for one number of revolutions only.

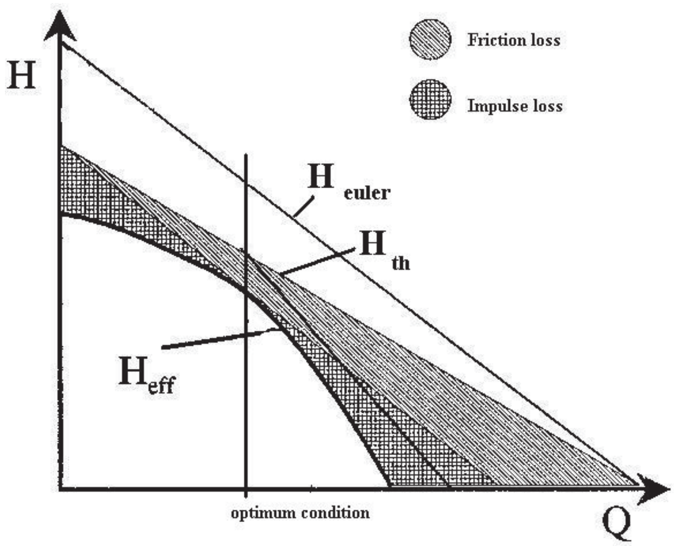


figure 16 Q-H curve

Figure 16 indicates a Q-H curve of a centrifugal pump. There is a linear relation between pressure and head so interchanging these values will not change the shape of the curve.

The highest line H_{euler} is the development of the head for an ideal impeller according the formula of Euler. Multiplying H_{euler} with the factor k gives the line for the theoretical head H_{th}.

The friction losses are increasing with the square of the capacity. The impulse losses are taking place at the entry of the impeller as result of turbulence and no perfect entry of the impeller. The lowest line indicates H_{eff}.

The vertical line shows the capacity at which there are no impulse losses. The pump efficiency is maximum at that time.

Affinity laws

The centrifugal pump is a very capable and flexible machine. Because of this it is unnecessary to design a separate pump for each job. The performance of a centrifugal pump can be varied by changing the impeller diameter or its rotation speed. Either change produces approximately the same results. Reducing impeller diameter is probably the most common change and is usually the most economical. The speed can be altered by changing pulley diameters or changing the speed of the driver.

When the driven speed or impeller diameter of a centrifugal pump changes, operation of the pump changes in accordance with three fundamental laws. These laws are known as the "laws of affinity". They state that:

- Capacity varies directly as the change in speed or impeller diameter

$$\frac{Q1}{Q2} = \frac{n1}{n2} = \frac{d1}{d2}$$

- Head varies as the square of the change in speed or impeller diameter

$$\frac{H_1}{H_2} = \left(\frac{u_1}{u_2}\right)^2 = \left(\frac{n_1}{n_2}\right)^2$$

- Power varies with the power of three of the change in speed or impeller diameter

$$\frac{P_1}{P_2} = \left(\frac{n_1}{n_2}\right)^3$$

Figure 17 shows the relation of Q, H and Power for different impeller diameters.

Since the transmission flow changes in a linear, the pump head pressure in a square relation to the rotation speed, corresponding points (i.e. A and B) are located on a parabola with the vertex at the point of origin. All points of such a parabola have geometrically similar curves for different rotational pump speeds. If for one point of the parabola non-impact entry of the liquid into the blades channels is considered, then a non-impact entry applies to all other points of the parabola.

Within its field of validity (maximum deviation approx 20 to 25% from the beginning rotation speed) the affinity laws offer possibility to calculate Q-H curves for other rotational speeds from a measured Q-H curve. All these Q-H curves are congruent.

One could conclude that the efficiency is the same for all maps on such parabola. This however is not the case. Equal efficiency is found in elliptical curves that are closed within themselves. The fluctuation is a result of the fact that the power consumption includes losses that do not change exactly with the 3rd power of the rotation speed. These are the mechanical losses in the bearings and shaft seals.

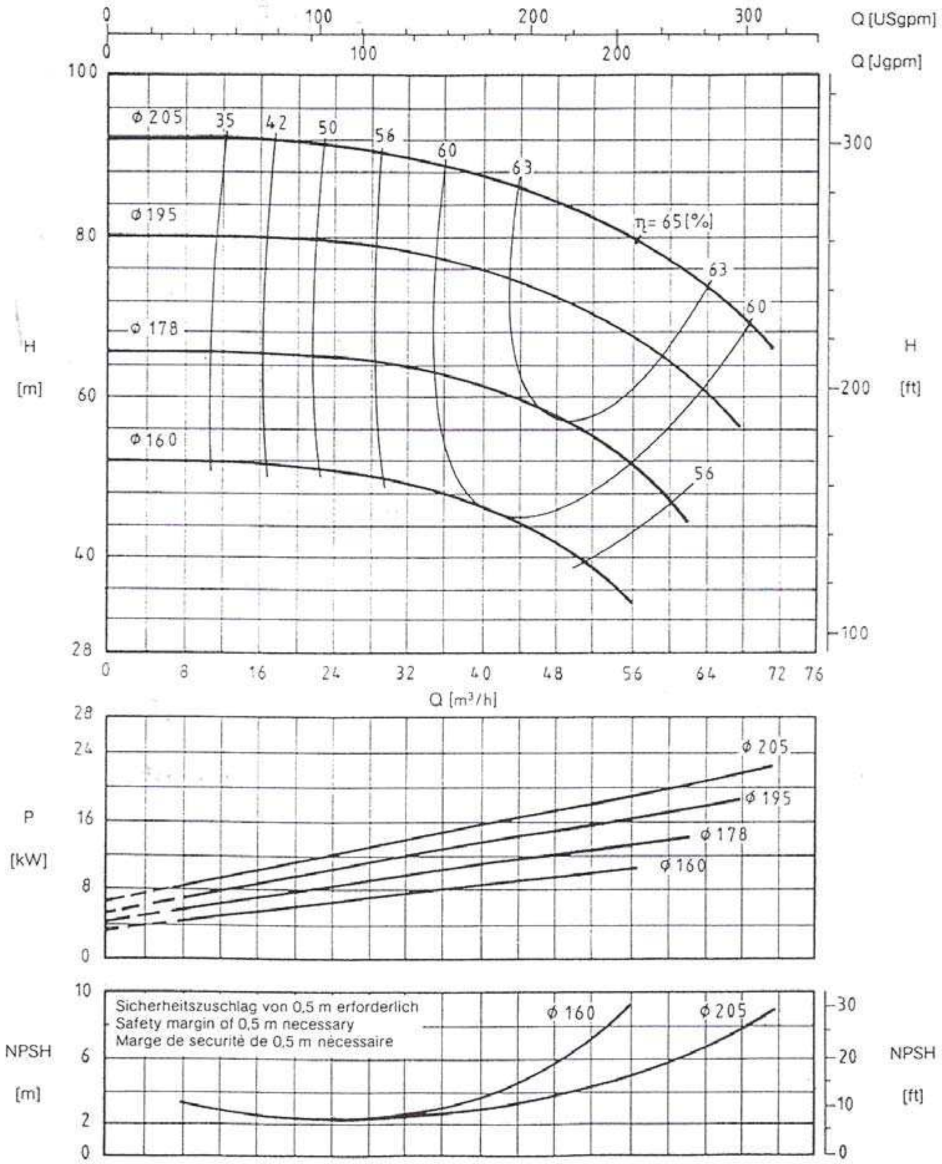


figure 17 Composite curves

The characteristic curve of the system

The pump head pressure of a plant generally consists of a static pump head pressure H_{stat} and the loss of head H_v in the connected pipe lines, valves and filters.

It was determined that both frictional losses and minor losses in the pump were proportional to the square of the flow velocity. The same is true for these losses in the piping system. Since flow velocity is directly proportional to the volumetric flow rate, the system head loss must be directly proportional to the square of the volumetric flow rate. From this relationship, it is possible to develop a curve of system head loss versus volumetric flow rate. The head loss curve for a typical piping system is in the shape of a parabola as shown in figure 18.

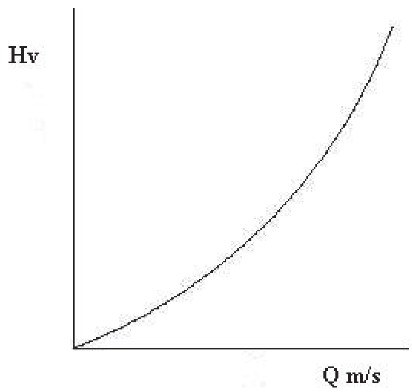


figure 18 Losses in the piping

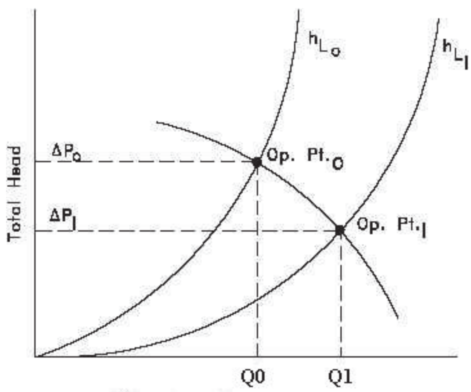


figure 19 System operating point

Operating point

The point at which a pump operates in a given piping system depends on the flow rate and head loss of that system. For a given system, volumetric flow rate is compared to system head loss on a system characteristic curve. By graphing a system characteristic curve and the pump characteristic curve on the same coordinate system, the point at which the pump must operate is identified. For example, in figure 19, the operating point for the centrifugal pump in the original system is designated by the intersection of the pump curve and the system curve (h_{Lo}).

The system has a flow rate equal to Q_0 and a total system head loss equal to ΔP_0. In order to maintain the flow rate, the pump head must be equal to ΔP_0.

In the system described by the system curve (h_{L1}), a valve has been opened in the system to reduce the system's resistance to flow. For this system, the pump maintains a large flow rate at a smaller pump head (Q_1) (ΔP_1).

System use of multiple centrifugal pumps

In order to increase the volumetric flow rate in a system or to compensate for large flow resistances, centrifugal pumps are often used in parallel or in series. figure 20 shows two identical centrifugal pumps operating at the same speed in parallel.

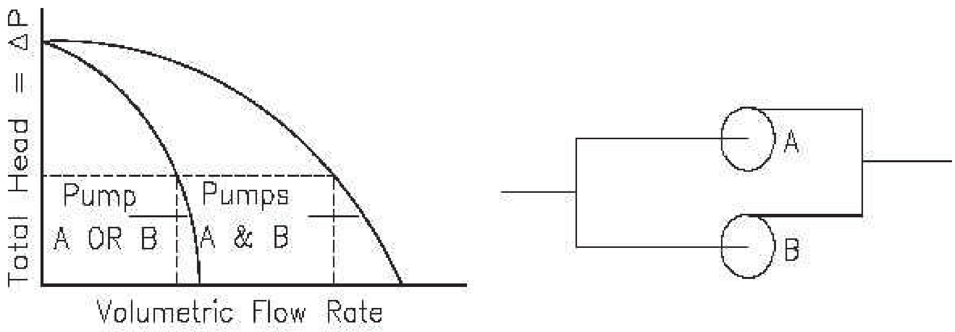


figure 20 Parallel operation of centrifugal pumps

Since the inlet and the outlet of each pump shown in figure 20 are at identical points in the system, each pump must produce the same pump head. The total flow rate in the system, however, is the sum of the individual flow rates for each pump.

When the system characteristic curve is considered with the curve for pumps in parallel, the operating point at the intersection of the two curves represents a higher volumetric flow rate than for a single pump and a greater system head loss. As shown in figure 21, a greater system head loss occurs with the increased fluid velocity resulting from the increased volumetric flow rate. Because of the greater system head, the volumetric flow rate is actually less than twice the flow rate achieved by using a single pump.

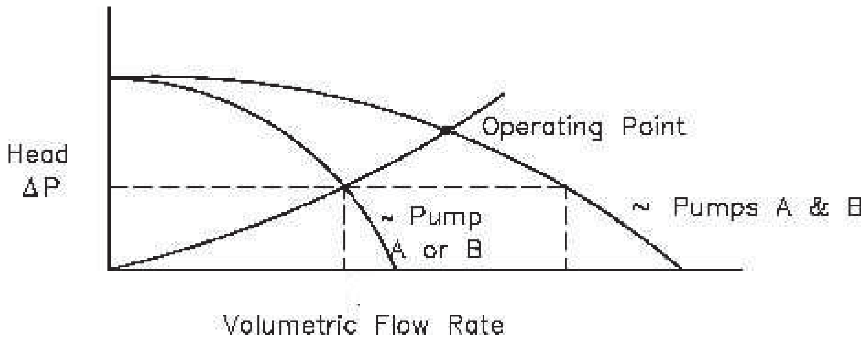


Figure 21 Operating point of two parallel centrifugal pumps

Centrifugal pumps in series

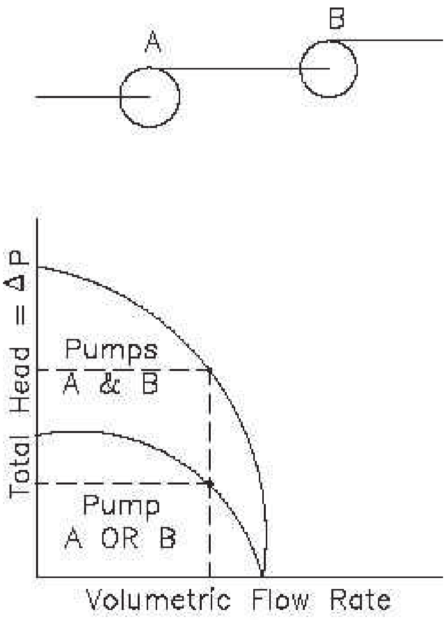


figure 21 Two pumps in series

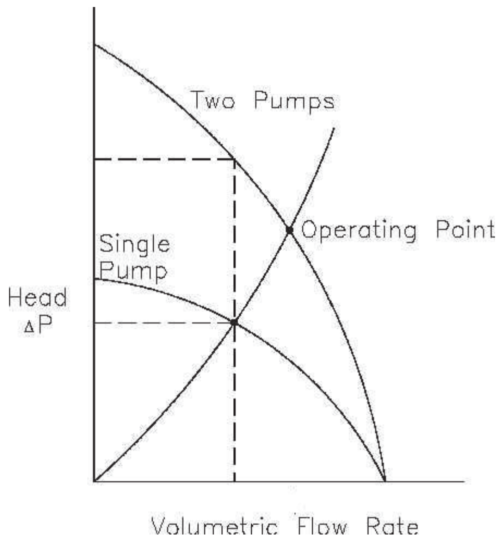


figure 22 operating point two pumps in series

Centrifugal pumps are used in series to overcome a larger system head loss than one pump can compensate for individually. As illustrated in figure 22, two identical centrifugal pumps operating at the same speed with the same volumetric flow rate contribute the same pump head.

Since the inlet to the second pump is the outlet of the first pump, the head produced by both pumps is the sum of the individual heads. The volumetric flow rate from the inlet of the first pump to the outlet of the second remains the same.

As shown in figure 22, using two pumps in series does not actually double the resistance to flow in the system. The two pumps provide adequate pump head for the new system and also maintain a slightly higher volumetric flow rate.

Figure 23 shows a multi stage centrifugal pump with 4 impellers in series. These pumps can create high pressures and can be used for example as booster pump for fire fighting monitors or as boiler feed water pump.

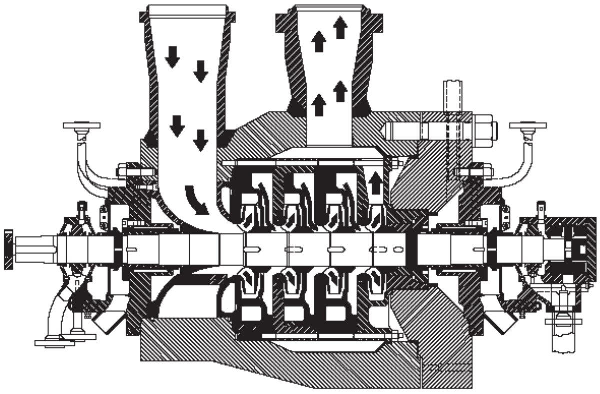


figure 23 Multi stage centrifugal pump

Impeller types

The impellers of centrifugal pumps can be divided according entry of the liquid in:

- Radial impellers (figure 24a & d)
- Axial impellers (figure 24e)
- Mixed radial and axial impeller (figure 24b & c)
- Double entry impellers (figure 24c)

or by construction of the impeller:

- Closed impeller (figure 24a & b)
- Semi open impeller (figure 24d)
- Open impeller (figure 24e)

A closed impeller pump is usually highly efficient in moving water but cannot handle liquids containing solid particles. This impeller type is enclosed on both sides by a cover plate or a shroud. It will develop high pressures.

A semi-open impeller has a plate or shroud on only one side. This impeller type is used for handling water with limited amounts of solids.

Open impeller pumps have no plates or shrouds attached to the impeller and are used primarily to handle high solids content liquids. Large open impeller pumps can be used to handle liquid at 10 to 15 percent solids content.

Double sided impellers are used in large pumps and have the advantage of no axial force working on the impeller and shaft.

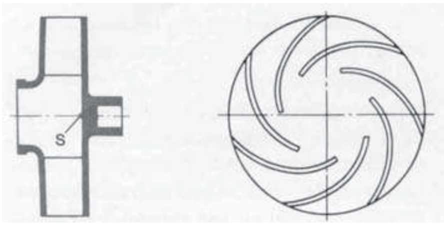


figure 24a radial closed impeller

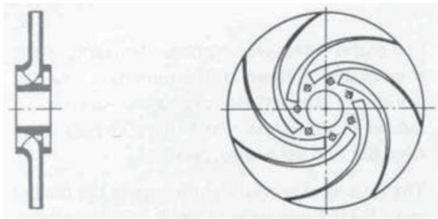


figure 24b axial closed impeller

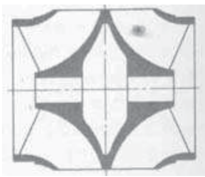


figure 24c double entry impeller

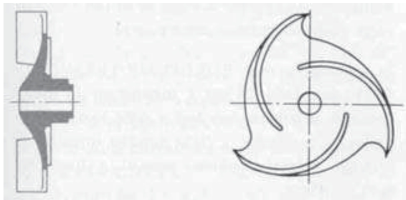


figure 24d semi open radial impeller

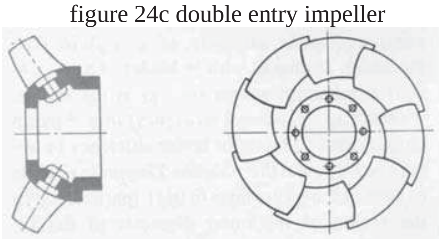


figure 24e open axial impeller

Centrifugal pump classification by flow

Centrifugal pumps can be classified based on the manner in which fluid flows through the pump. The manner in which fluid flows through the pump is determined by the design of the pump casing and the impeller. The three types of flow through a centrifugal pump are radial flow, axial flow, and mixed flow.

Radial flow pumps

In a radial flow pump, the liquid enters at the center of the impeller and is directed out along the impeller blades in a direction at right angles to the pump shaft. The impeller of a typical radial flow pump and the flow through a radial flow pump are shown in Figure 25.

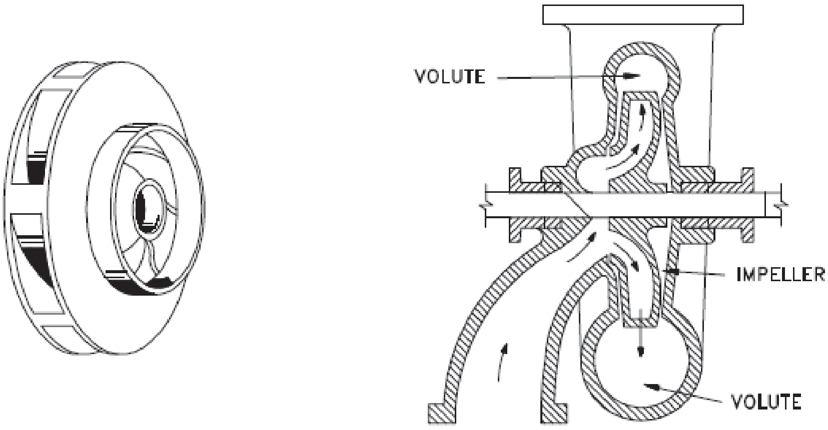


Figure 25 Radial flow centrifugal pump

Axial Flow Pumps

In an axial flow pump, the impeller pushes the liquid in a direction parallel to the pump shaft. Axial flow pumps are sometimes called propeller pumps because they operate essentially the same as the propeller of a boat. The impeller of a typical axial flow pump and the flow through a radial flow pump are shown in Figure 26.

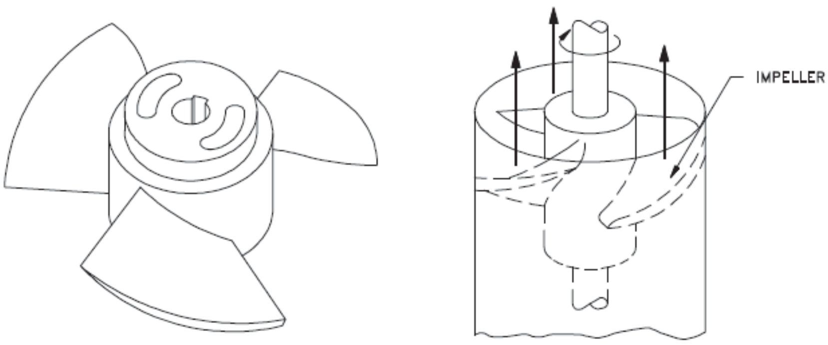
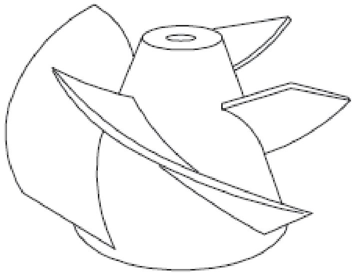


figure 26 Axial flow centrifugal pump

Mixed flow pumps

Mixed flow pumps borrow characteristics from both radial flow and axial flow pumps. As liquid flows through the impeller of a mixed flow pump, the impeller blades push the liquid out away from the pump shaft and to the pump suction at an angle greater than 90°. The impeller of a typical mixed flow pump and the flow through a mixed flow pump are shown in Figure 27.



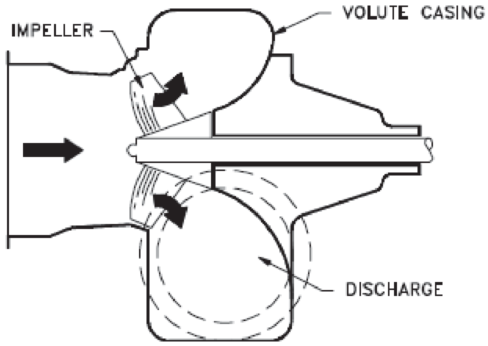


figure 27 Mixed flow centrifugal pump

Vertical placed centrifugal pump

Figure 28 shows a vertical placed centrifugal pump with double entry. The advantages of vertical mounted pumps are:

- Limited floor space required
- Split pump casing for easy access during maintenance
- Electric motor is mounted on a flange. The shaft of the motor and pump are therefore in line and no adjustments need to be made

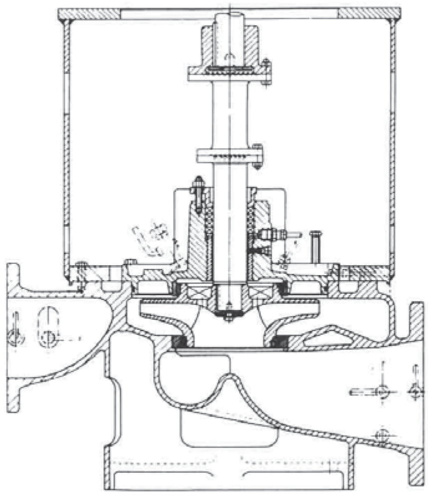


figure 28 Vertical mounted centrifugal pump

Operation of a centrifugal pump

A centrifugal pump is not self priming. This is the result of:

- The gap between the impeller and the casing resulting in an opening between the suction and discharge side of the pump
- If the pump is filled with air the impeller is not able to create sufficient pressure difference. In density of air is 1,000 times smaller as water which results in a too small p_{euler}. No flow will start resulting in no lowering of pressure on the suction side

There are 4 ways to make a centrifugal pump self priming:

- Make sure that the pump is placed below the liquid level on the suction side. This is shown in figure 1.
- Install a foot valve at the bottom of the suction line (figure 29)
- Install a vacuum installation to the volute casing of the pump. Before starting the pump the air will be removed by the vacuum unit

- The pump is designed in such a way that after stopping the pump the casing will remain full of water. This will enable the pump to self prime at the next start up (figure 30)

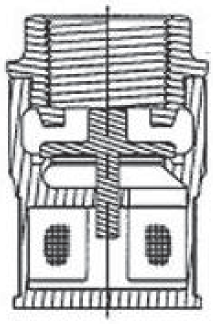


figure 29 Foot valve

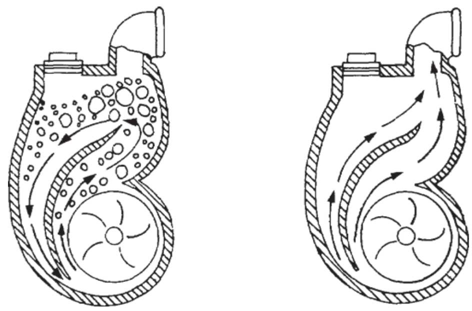


figure 30 Casing construction to keep pump filled up

Only the suction valve is opened before starting. The discharge valve of the pump is kept closed. This is to minimise the electrical current to the motor. No extreme high pressures will be developed as no water is flowing through the pump. This can be seen from the composite curve figure 17. The power consumption of the pump is lowest at low flow. The pressure is limited to the effective pressure of the pump.

As soon as the pump has reached its full speed the discharge valve is opened slowly to increase the flow through the pump.

Capacity control of a centrifugal pump is always regulated with the discharge valve. Reducing the flow of the suction valve will increase the resistance on the suction side. Too much resistance might result in the pressure at the pump to go below the vapour pressure creating bubbles which hamper the flow and might cause cavitation.

Positive displacement pumps

Positive displacement pumps operate on a different principle than centrifugal pumps. Positive displacement pumps physically entrap a quantity of liquid at the suction of the pump and push that quantity out the discharge of the pump.

Introduction

A positive displacement pump is one in which a definite volume of liquid is delivered for each cycle of pump operation. This volume is constant regardless of the resistance to flow offered by the system the pump is in, provided the capacity of the power unit driving the pump or pump component strength limits are not exceeded. The positive displacement pump delivers liquid in separate volumes with no delivery in between, although a pump having several chambers may have an overlapping delivery among individual chambers, which minimizes this effect. The positive displacement pump differs from centrifugal pumps, which deliver a continuous flow for any given pump speed and discharge resistance.

Positive displacement pumps can be grouped into three basic categories based on their design and operation. The three groups are reciprocating pumps, rotary pumps, and diaphragm pumps.

Principle of operation

All positive displacement pumps operate on the same basic principle. This principle can be most easily demonstrated by considering a reciprocating positive displacement pump consisting of a single reciprocating piston in a cylinder with a single suction port and a single discharge port as shown in Figure 31. Check valves in the suction and discharge ports allow flow in only one direction.

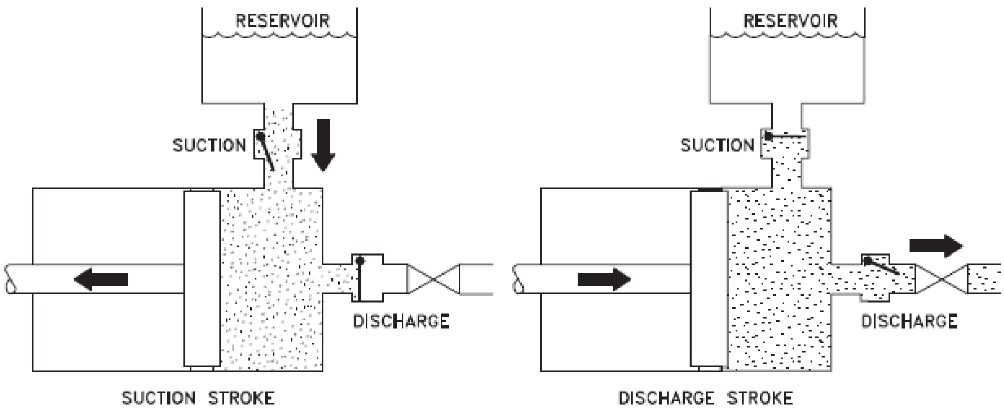


figure 31 Reciprocating Positive Displacement Pump Operation

During the suction stroke, the piston moves to the left, causing the check valve in the suction line between the reservoir and the pump cylinder to open and admit water from the reservoir. During the discharge stroke, the piston moves to the right, seating the check valve in the suction line and opening the check valve in the discharge line. The volume of liquid moved by the pump in one cycle (one suction stroke and one discharge stroke) is equal to the change in the liquid volume of the cylinder as the piston moves from its farthest left position to its farthest right position.

Reciprocating pumps

Reciprocating positive displacement pumps are generally categorized in four ways: direct-acting or indirect-acting; simplex or duplex; single-acting or double-acting; and power pumps.

Direct-acting and indirect-acting pumps

Some reciprocating pumps are powered by prime movers that also have reciprocating motion, such as a reciprocating pump powered by a reciprocating steam piston. The piston rod of the steam piston may be directly connected to the liquid piston of the pump or it may be indirectly connected with a beam or linkage. Direct-acting pumps have a plunger on the liquid (pump) end that is directly driven by the pump rod (also the piston rod or extension thereof) and carries the piston of the power end. Indirect-acting pumps are driven by means of a beam or linkage connected to and actuated by the power piston rod of a separate reciprocating engine.

Simplex and duplex pumps

A simplex pump, sometimes referred to as a single pump, is a pump having a single liquid (pump) cylinder. A duplex pump is the equivalent of two simplex pumps placed side by side on the same foundation.

The driving of the pistons of a duplex pump is arranged in such a manner that when one piston is on its upstroke the other piston is on its down stroke, and vice versa. This arrangement doubles the capacity of the duplex pump compared to a simplex pump of comparable design.

Single-acting and double-acting pumps

A single-acting pump is one that takes suction, filling the pump cylinder on the stroke in only one direction, called the suction stroke, and then forces the liquid out of the cylinder on the return stroke, called the discharge stroke. A double-acting pump is one that, as it fills one end of the liquid cylinder, is discharging liquid from the other end of the cylinder. On the return stroke, the end of the cylinder just emptied is filled, and the end just filled is emptied. One possible arrangement for single-acting and double-acting pumps is shown in Figure 32.

Power Pumps

Power pumps convert rotary motion to low speed reciprocating motion by reduction gearing, a crankshaft, connecting rods and crossheads. Plungers or pistons are driven by the crosshead drives. Rod and piston construction, similar to duplex double-acting steam pumps, is used by the liquid ends of the low pressure, higher capacity units. The higher pressure units are normally single-acting plungers, and usually employ three (triplex) plungers. Three or more plungers substantially reduce flow pulsations relative to simplex and even duplex pumps.

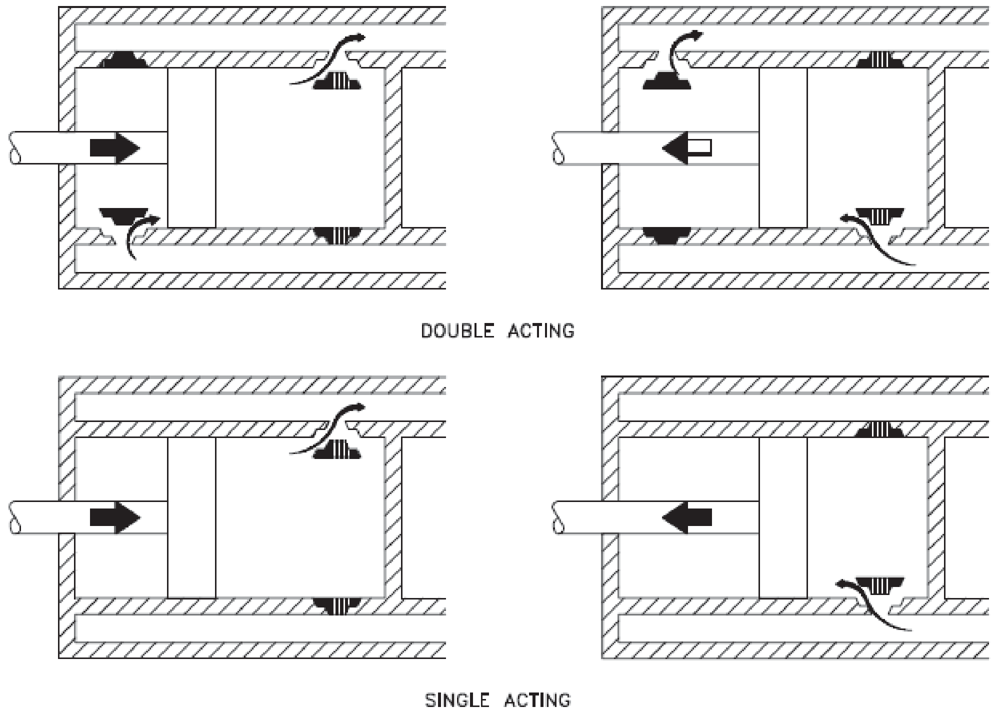


figure 32 Single-acting and double-acting pumps

Power pumps typically have high efficiency and are capable of developing very high pressures. They can be driven by either electric motors or turbines. They are relatively expensive pumps and can rarely be justified on the basis of efficiency over centrifugal pumps. However, they are frequently justified over steam reciprocating pumps where continuous duty service is needed due to the high steam requirements of direct-acting steam pumps.

In general, the effective flow rate of reciprocating pumps decreases as the viscosity of the fluid being pumped increases because the speed of the pump must be reduced. In contrast to centrifugal pumps, the differential pressure generated by reciprocating pumps is independent of fluid density. It is dependent entirely on the amount of force exerted on the piston.

Rotary pumps

Rotary pumps operate on the principle that a rotating vane, screw, or gear traps the liquid in the suction side of the pump casing and forces it to the discharge side of the casing. These pumps are essentially self-priming due to their capability of removing air from suction lines and producing a high suction lift. In pumps designed for systems requiring high suction lift and self-priming features, it is essential that all clearances between rotating parts, and between rotating and stationary parts, be kept to a minimum in order to reduce slippage. Slippage is leakage of fluid from the discharge of the pump back to its suction.

Due to the close clearances in rotary pumps, it is necessary to operate these pumps at relatively low speed in order to secure reliable operation and maintain pump capacity over an extended period of time. Otherwise, the erosive action due to the high velocities of the liquid passing through the narrow clearance spaces would soon cause excessive wear and increased clearances, resulting in slippage.

There are many types of positive displacement rotary pumps, and they are normally grouped into three basic categories that include gear pumps, screw pumps, and moving vane pumps.

Simple gear pump

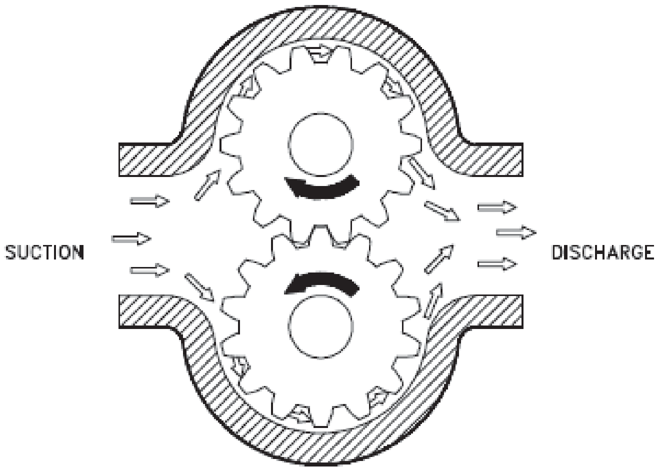


figure 33 Simple gear pump

There are several variations of gear pumps. The simple gear pump shown in Figure 33 consists of two spur gears meshing together and revolving in opposite directions within a casing. Only a few thousandths of an inch clearance exists between the case and the gear faces and teeth extremities. Any liquid that fills the space bounded by two successive gear teeth and the case must follow along with the teeth as they revolve. When the gear teeth mesh with the teeth of the other gear, the space between the teeth is reduced, and the entrapped liquid is forced out the pump discharge pipe. As the gears revolve and the teeth disengage, the space again opens on the suction side of the pump, trapping new quantities of liquid and carrying it around the pump case to the discharge. As liquid is carried away from the suction side, a lower pressure is created, which draws liquid in through the suction line.

With the large number of teeth usually employed on the gears, the discharge is relatively smooth and continuous, with small quantities of liquid being delivered to the discharge line in rapid succession. If designed with fewer teeth, the space between the teeth is greater and the capacity increases for a given speed; however, the tendency toward a pulsating discharge increases. In all simple gear pumps, power is applied to the shaft of one of the gears, which transmits power to the driven gear through their meshing teeth.

There are no valves in the gear pump to cause friction losses as in the reciprocating pump. The high impeller velocities, with resultant friction losses, are not required as in the centrifugal pump. Therefore, the gear pump is well suited for handling viscous fluids such as fuel and lubricating oils.

Other gear pumps

There are two types of gears used in gear pumps in addition to the simple spur gear. One type is the helical gear. A helix is the curve produced when a straight line moves up or down the surf ace of a cylinder. The other type is the herringbone gear. A herringbone gear is composed of two helixes spiralling in different directions from the center of the gear. Spur, helical, and herringbone gears are shown in Figure 34.

The helical gear pump has advantages over the simple spur gear. In a spur gear, the entire length of the gear tooth engages at the same time. In a helical gear, the point of engagement moves along the length of the gear tooth as the gear rotates. This makes the helical gear operate with a steadier discharge pressure and fewer pulsations than a spur gear pump.

The herringbone gear pump is also a modification of the simple gear pump. Its principal difference in operation from the simple spur gear pump is that the pointed center section of the space between two teeth begins discharging before the divergent outer ends of the preceding space complete discharging. This overlapping tends to provide a steadier discharge pressure. The power transmission from the driving to the driven gear is also smoother and quieter.

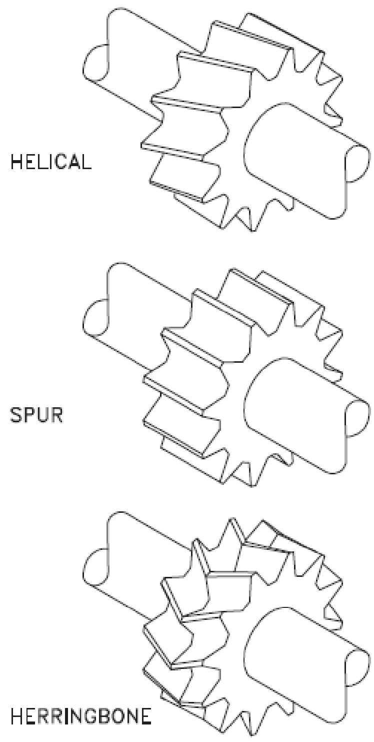


figure 34 Types of gears used in pumps

Lobe type pump

The lobe type pump shown in Figure 35 is another variation of the simple gear pump. It is considered as a simple gear pump having only two or three teeth per rotor; otherwise, its operation or the explanation of the function of its parts is no different. Some designs of lobe pumps are fitted with replaceable gibs, that is, thin plates carried in grooves at the extremity of each lobe where they make contact with the casing. The gib promotes tightness and absorbs radial wear.

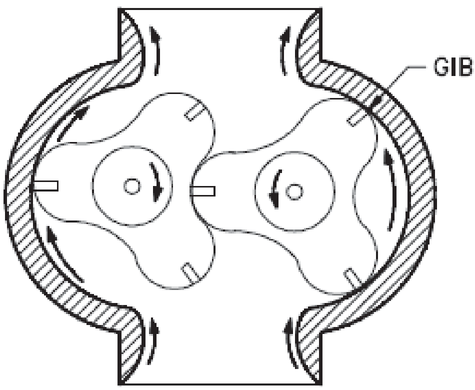


figure 35 lobe type pump

Screw-type positive displacement rotary pump

There are many variations in the design of the screw type positive displacement, rotary pump. The primary differences consist of the number of intermeshing screws involved, the pitch of the screws, and the general direction of fluid flow. Two common designs are the two-screw, low-pitch, double-flow pump and the three-screw, high-pitch, double-flow pump.

Two-screw, low-pitch, screw pump

The two-screw, low-pitch, screw pump consists of two screws that mesh with close clearances, mounted on two parallel shafts. One screw has a right-handed thread, and the other screw has a left-handed thread. One shaft is the driving shaft and drives the other shaft through a set of herringbone timing gears. The gears serve to maintain clearances between the screws as they turn and to promote quiet operation. The screws rotate in closely fitting duplex cylinders that have overlapping bores. All clearances are small, but there is no actual contact between the two screws or between the screws and the cylinder walls.

The complete assembly is shown in Figure 36. Liquid is trapped at the outer end of each pair of screws. As the first space between the screw threads rotates away from the opposite screw, a one-turn, spiral-shaped quantity of liquid is enclosed when the end of the screw again meshes with the opposite screw. As the screw continues to rotate, the entrapped spiral turns of liquid slide along the cylinder toward the center discharge space while the next slug is being entrapped. Each screw functions similarly, and each pair of screws discharges an equal quantity of liquid in opposed streams toward the center, thus eliminating hydraulic thrust. The removal of liquid from the suction end by the screws produces a reduction in pressure, which draws liquid through the suction line.

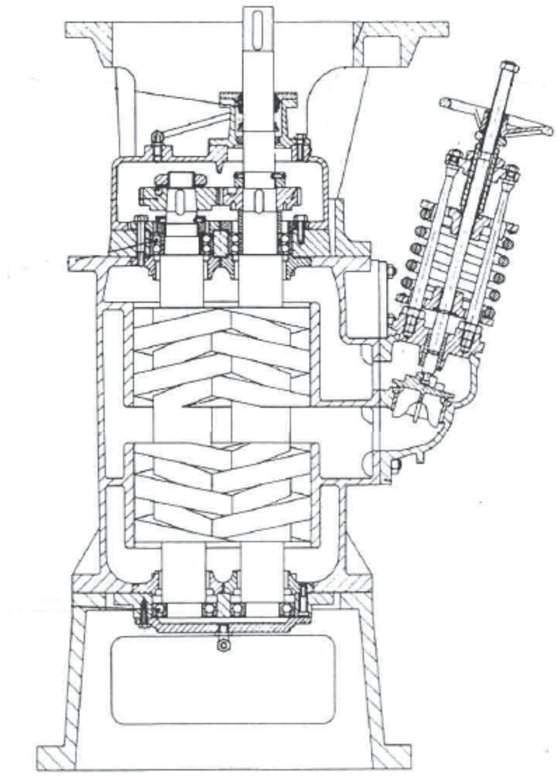
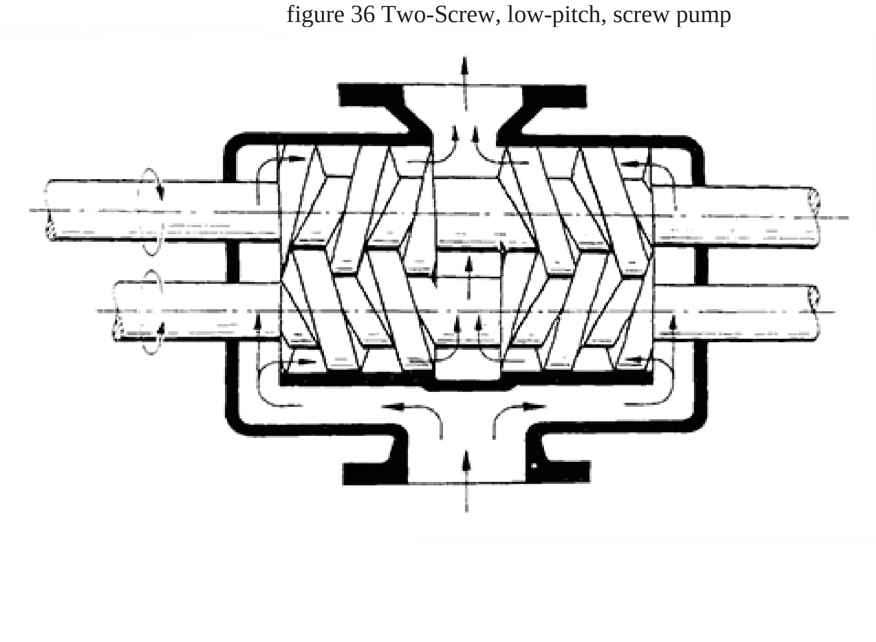
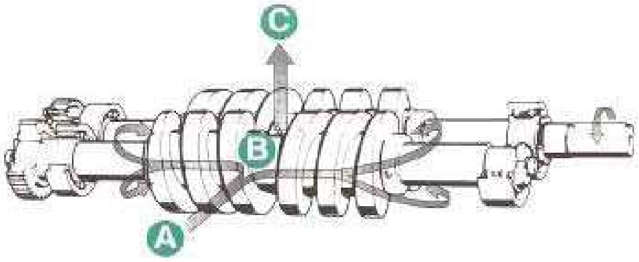


figure 36 Two-Screw, low-pitch, screw pump





Three-screw, high-pitch, screw pump

The three-screw, high-pitch, screw pump, shown in Figure 37, has many of the same elements as the two-screw, low-pitch, screw pump, and their operations are similar. Three screws, oppositely threaded on each end, are employed. They rotate in a triple cylinder, the two outer bores of which overlap the center bore. The pitch of the screws is much higher than in the low pitch screw pump; therefore, the center screw, or power rotor, is used to drive the two outer idler rotors directly without external timing gears. Pedestal bearings at the base support the weight of the rotors and maintain their axial position. The liquid being pumped enters the suction opening, flows through passages around the rotor housing, and through the screws from each end, in opposed streams, toward the center discharge. This eliminates unbalanced hydraulic thrust. The screw pump is used for pumping viscous fluids, usually lubricating, hydraulic, or fuel oil.

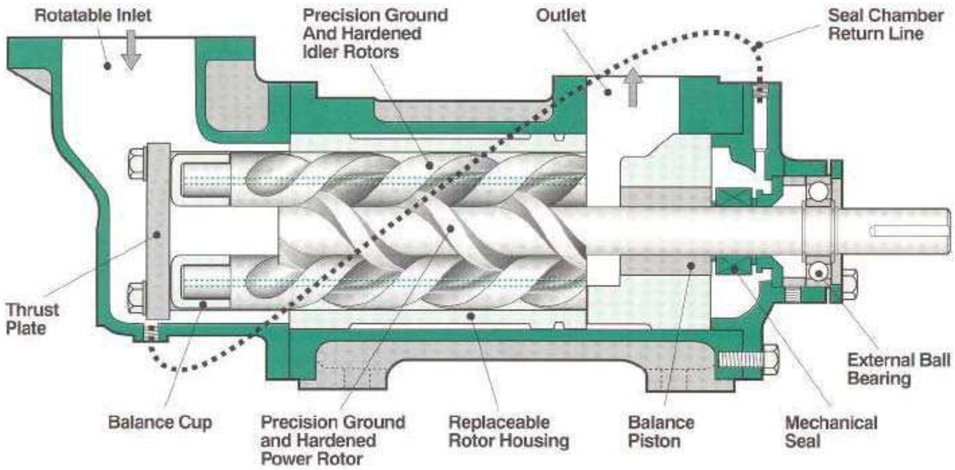


figure 37 Three-Screw, high-pitch, screw pump

Rotary moving vane pump

The rotary moving vane pump shown in Figure 38 is another type of positive displacement pump used. The pump consists of a cylindrically bored housing with a suction inlet on one side and a discharge outlet on the other. A cylindrically shaped rotor with a diameter smaller than the cylinder is driven about an axis placed above the centerline of the cylinder. The clearance between rotor and cylinder is small at the top but increases at the bottom. The rotor carries vanes that move in and out as it rotates to maintain sealed spaces between the rotor and the cylinder wall. The vanes trap liquid or gas on the suction side and carry it to the discharge side, where contraction of the space expels it through the discharge line. The vanes may swing on pivots, or they may slide in slots in the rotor.

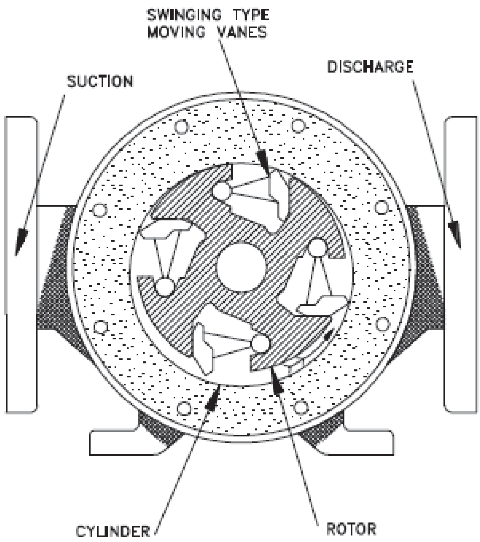


figure 38 Rotary moving vane pump

Diaphragm pumps

Diaphragm pumps are also classified as positive displacement pumps because the diaphragm acts as a limited displacement piston. The pump will function when a diaphragm is forced into reciprocating motion by mechanical linkage, compressed air, or fluid from a pulsating, external source. The pump construction eliminates any contact between the liquid being pumped and the source of energy. This eliminates the possibility of leakage, which is important when handling toxic or very expensive liquids. Disadvantages include limited head and capacity range, and the necessity of check valves in the suction and discharge nozzles. An example of a diaphragm pump is shown in Figure 39.

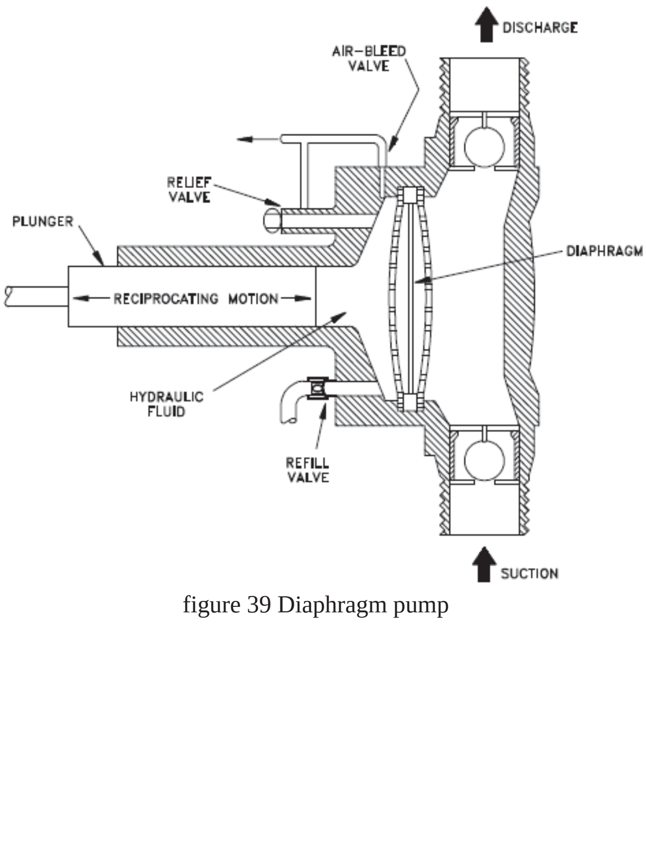


figure 39 Diaphragm pump

Positive displacement pump characteristic curves

Positive displacement pumps deliver a definite volume of liquid for each cycle of pump operation. Therefore, the only factor that affects flow rate in an ideal positive displacement pump is the speed at which it operates. The flow resistance of the system in which the pump is operating will not affect the flow rate through the pump. Figure 40 shows the characteristic curve for a positive displacement pump.

The dashed line in Figure 40 shows actual positive displacement pump performance. This line reflects the fact that as the discharge pressure of the pump increases, some amount of liquid will leak from the discharge of the pump back to the pump suction, reducing the effective flow rate of the pump. The rate at which liquid leaks from the pump discharge to its suction is called slippage.

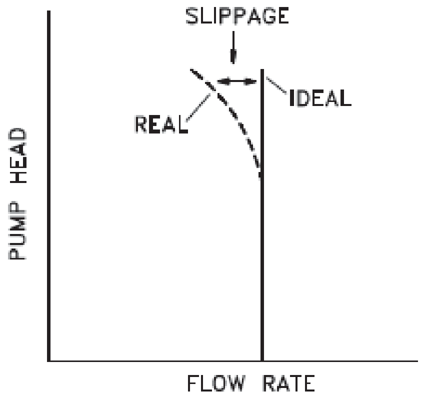


figure 40 Positive displacement pump characteristic curve

Positive displacement pump protection

Positive displacement pumps are normally fitted with relief valves on the upstream side of their discharge valves to protect the pump and its discharge piping from over pressurization. Positive displacement pumps will discharge at the pressure required by the system they are supplying. The relief valve prevents system and pump damage if the pump discharge valve is shut during pump operation or if any other occurrence such as a clogged strainer blocks system flow.

Sewage systems

Sewage from vessels, also known as “black water,” generally means human body wastes and the wastes from toilets and other receptacles intended to receive or retain body wastes. Next to “black water” there is also “grey water” which is defined as drainage from dishwater, shower, laundry, bath and washbasin drains.

On most ships, sewage is treated using a marine sanitation device that biologically treats and disinfects the waste prior to discharge. Some ships have installed Advanced Wastewater Treatment systems (AWTs) to treat sewage and often grey water. These AWTs provide higher levels of biological treatment, solids removal, and disinfection as compared to traditional marine sanitation devices.

The International Convention for the Prevention of Pollution from Ships

The principal international convention addressing discharge standards for vessel sewage is Annex IV to the International Convention for the Prevention of Pollution from Ships (known as MARPOL 73/78, or simply MARPOL). Although Annex IV was adopted in 1973, it did not come into effect until September 2003, after ratification by the requisite number of states (and corresponding shipping fleet tonnage).

MARPOL Annex IV contains regulations regarding the discharge of sewage into the sea, ships' equipment and systems for the control of sewage discharge, a provision for facilities at ports and terminals for the reception of sewage, and requirements for survey and certification. Annex IV defines sewage as drainage and other wastes from any form of toilets and urinals; drainage from medical premises (dispensary, sick bay, etc.) via wash basins, wash tubs and scuppers located in such premises; drainage from spaces containing living animals; or other waste waters when mixed with the drainages defined above.

MARPOL Annex IV generally requires ships to be equipped with either a sewage treatment plant, a sewage comminuting and disinfecting system, or a sewage holding tank. More specifically, the discharge of sewage into the sea is prohibited except when the ship has in operation an approved sewage treatment plant or is discharging comminuted and disinfected sewage using an approved system at a distance of more than three nautical miles (nm) from the nearest land; or is discharging sewage which is not comminuted or disinfected at a distance of more than 12 nm from the nearest land. There is a standard for the maximum rate of discharge of untreated sewage from holding tanks when at a distance equal or greater than 12 nm from the nearest land. Annex IV also establishes certain sewage reception facility standards and responsibilities for ports. Vessels that comply with Annex IV are issued an International Sewage Pollution Prevention Certificate (ISPPC).

All sewage treatment plants installed on existing ships on or after 1 January 2010, and on new ships whose keels are laid on or after this date, must meet the new MEPC.159(55) guidelines which contain amendments to MARPOL Annex IV.

The discharge limits for the installations installed after 1 January 2010 are stricter and additional requirements like allowable pH value have been introduced. The main reasons for the amended rules are that in certain areas of the world the density of ship is high which might create high concentrations of sewage waste and secondly the modern cruise ships have grown enormous in size and passenger capacity. As most of these cruise vessels operate in the same areas you can imagine that if no additional measures would be taken the concentration of sewage waste would go out of control.

For certain areas in the world like Alaska, which is a favorite cruise destination, the local authorities have set even stricter rules.

Sanitary system

Onboard ships normally there are no toilets which use a conventional cistern tank with a float switch for flushing. This system is unpractical as ships pitch and roll and water will spill everywhere. A marine toilet should be dry when not in use. The following systems can be found onboard:

- Sea or fresh water pressurized flushing or so called wet systems
- Vacuum system or so called dry system

Wet systems use pressurized sea or fresh water. The pressure is maintained by the hydrofoor system. The valve used to flush the toilet is called a flush-o-meter and is designed in such way that only a minimum amount of water is needed for each flushing. The sewage is pushed out of the toilet by the pressurized water to the sewage tank or directly overboard.

The diaphragm valve is in the ready position with all of the parts labelled. In this position the upper and lower chambers contain the same amount of pressure. Therefore, the diaphragm remains seated on the seat.

When the handle is moved in any direction, the plunger opens the relief valve, which allows the water from the upper chamber to flow into the lower chamber and causes the diaphragm to rise off of its seat. Water now continues to flow down the barrel and into the fixture.

As the valve lifts the diaphragm, water begins to flow slowly through the bypass orifice until the pressure rises enough to equalize the pressure in the upper and lower chambers, seating the valve.

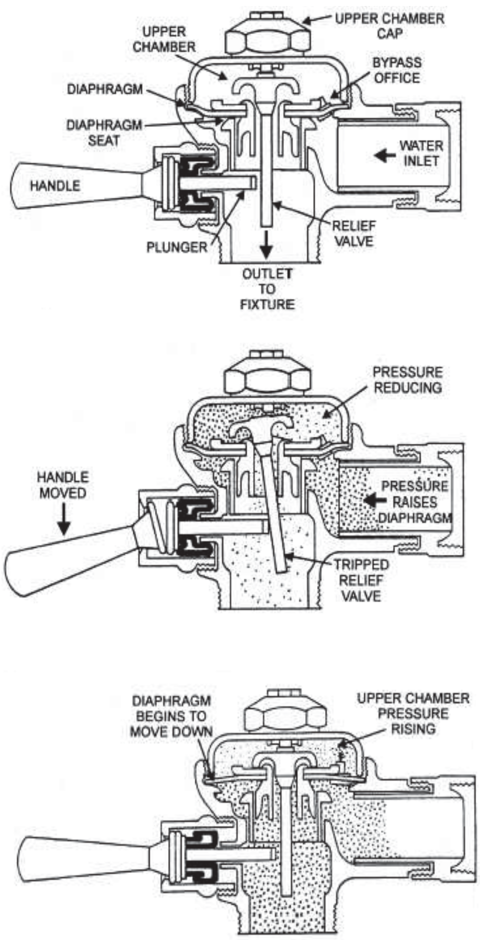


figure …. Operation of a flush-o-meter

By this sequence of operation the opening and especially the closing of the flush-o-meter will not go instantly which may cause sudden pressure changes in the piping, which can be accompanied by strong banging sounds.

The choice between fresh or sea water has consequences for the systems onboard. Sea water is free but more aggressive so more durable piping materials are needed and pipes might clog up due to salt deposits. Fresh water does not have these consequences but the consumption of fresh water will require larger capacity of the fresh water generators and/or fresh water tanks.

Vacuum sewage system

In vacuum systems sewage transport is done by vacuum (air) instead of water and gravity. The advantage of a vacuum sewage system over a wet system is the reduction of flush water. A wet system will produce around 3.5 – 4 litres of liquids during each flush. In a vacuum system this is normally below 1 litre. A lower amount of water will help to reduce the size of the holding or sewage tank. On vessels like ferries and cruise liners where the number of people onboard can go up to thousands this would mean a large reduction in sewage tank capacity.

System load (flushing) must be divided evenly between main lines. To avoid sewage back flow due to pitching, rolling and trim horizontal pipes must be sloped, transport pockets at every 25 – 30 meter and/or non-return valves installed.

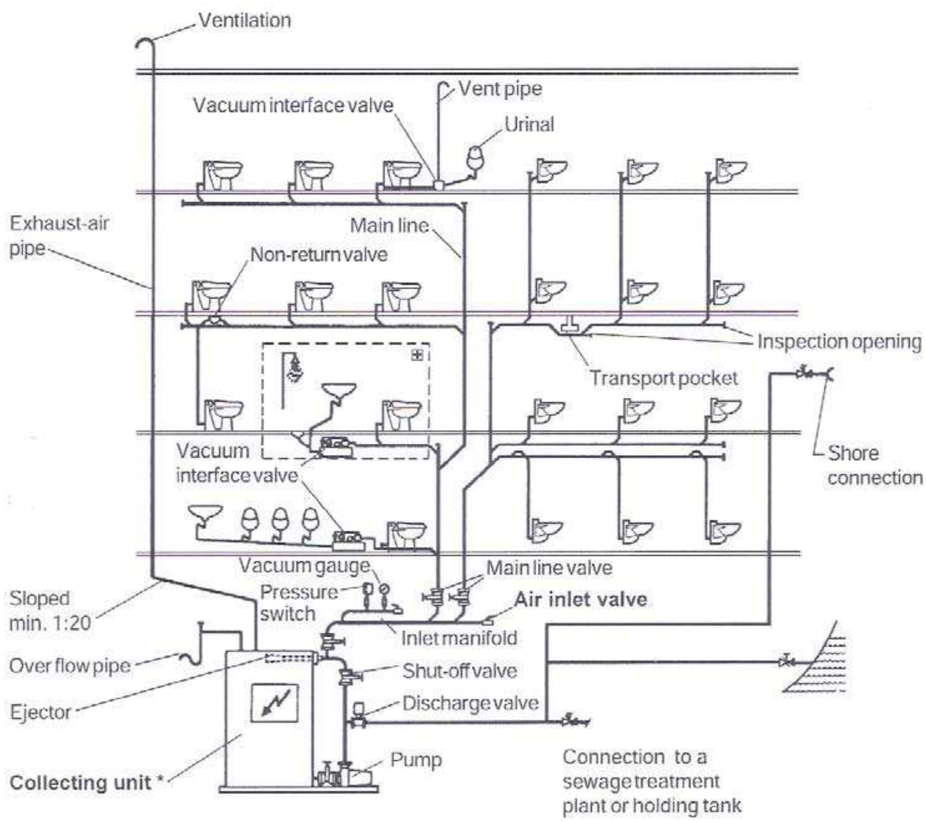


figure … Vacuum sewage system

The piping system is kept under vacuum constantly. At toiler bowls valves are installed which are normally closed and can be operated to flush the toilet. Sinks, urinals, showers etc. are not equipped with valves but connected to the system via vacuum interface valves. This type of valve operates as soon as a certain static head of grey water is acting on it. The valve will open and the liquid is sucked in the system.

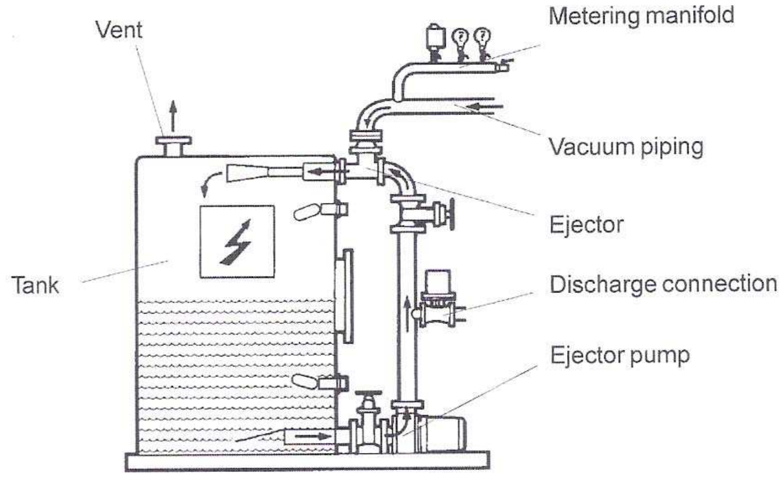


figure … Collecting tank unit

The vacuum in the system is generated by an ejector pump circulating water through an ejector. (figure …) The vacuum is only generated to the piping, the sewage collecting tank itself is under atmospheric pressure. The vacuum is controlled by a pressure switch located at the metering manifold. This switch will start and stop the pump to maintain a certain level of vacuum.

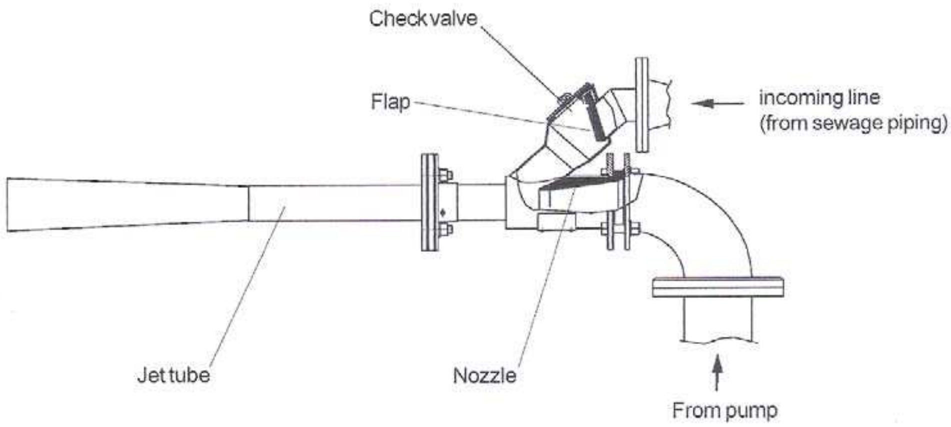


figure … Ejector system

Inside the tank level switches control the level. When the high level switch is activated the motor controlled valve at the discharge connection is opened and sewage is pumped out by the ejector pump to the sewage processing unit. As soon as the level has dropped to the low level switch the discharge valve will close again.

On vessels like ferries or cruise liners where large vacuum systems are installed a vacuum collecting tank as shown in figure … can be installed. In this instance the whole collecting tank is kept under vacuum.

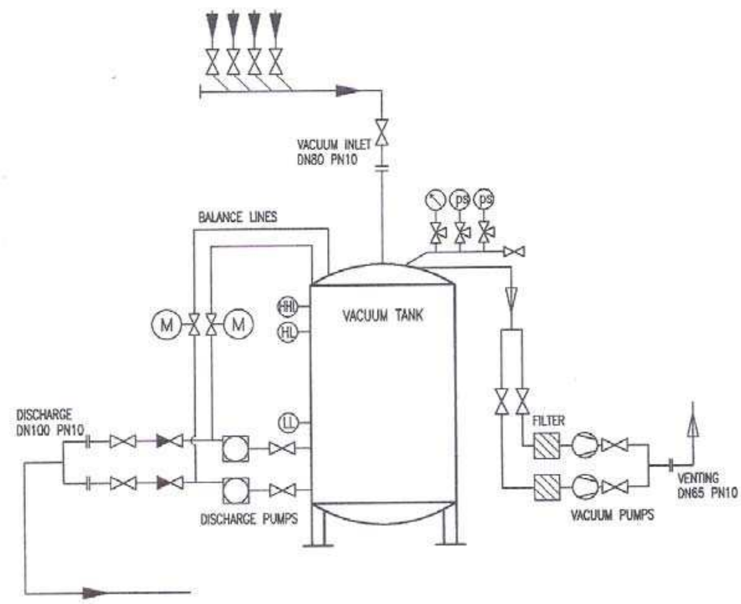


figure … Vacuum collecting tank unit

Traditional Marine Sanitation Devices (MDS)

On ships with traditional MSDs, sewage is treated using biological treatment and chlorination. Some ships do not treat their sewage biologically, but instead use maceration and chlorination.

Biological-chlorination MSDs operate similarly to land-based biological treatment systems for municipal wastewater treatment. The treatment system typically includes aerobic biological treatment to remove biochemical oxygen demand and some nutrients, clarification and filtration to remove solids, and final chlorine disinfection to destroy pathogens (see figure). The system also may include screening to remove grit and debris.

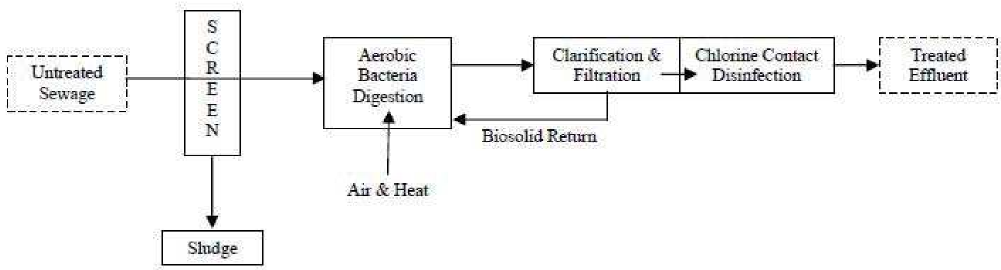


figure …… Flow diagram of MDS sewage treatment unit

Maceration-chlorination systems use screening to remove grit and debris, maceration for solids size reduction, and chlorine disinfection to oxidize and disinfect the waste.

Chlorine is either added (sodium hypochlorite) or generated by mixing the sewage with sea water and then passing this solution between electrolytic cells to produce hypochlorite.

Operating Principle MDS

In biological sewage treatment plants biopopulation coverts organic substances existing in waste water to carbon dioxide and water without danger of methane gas production. The biological sewage treatment plants have four chambers (two aeration chambers, a settling chamber and a disinfection chamber). The biological aerobic process takes place in two aeration chambers. In the settling chamber sludge is separated from waste water and in the disinfection chamber discharged effluent is neutralized.

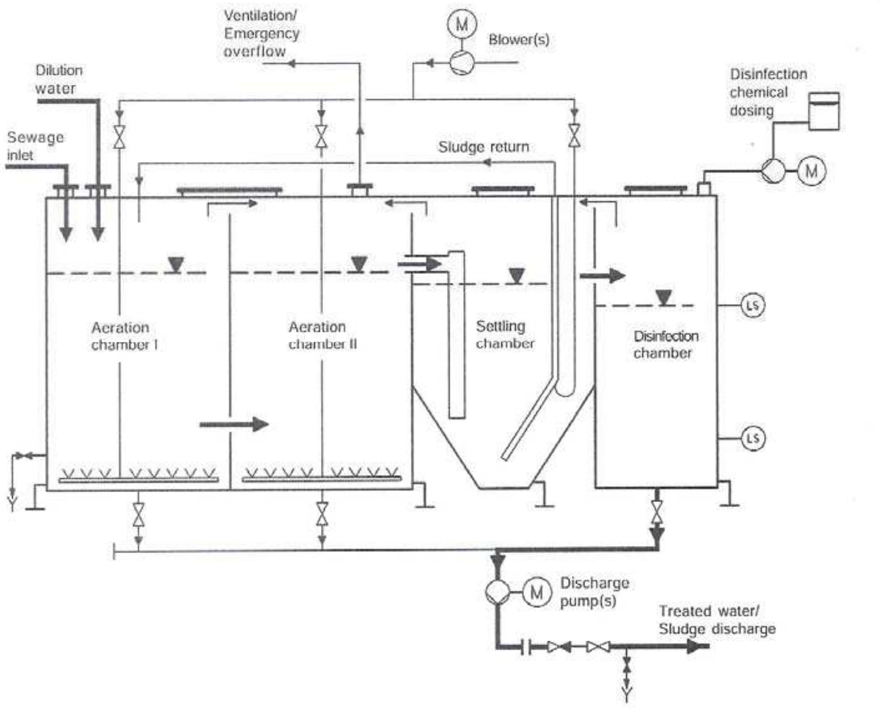


Figure … Biological-chlorination MDS

Aeration and blowers

Waste water is led to the aeration chamber I by gravity or vacuum. Bacterial growth is stimulated by oxygen of air. Required air is produced by the air blower and led to the aeration chamber I and II via installed aerators. An aerobic process continues in the aeration chamber II. Inorganic solids (for instance plastic) are stopped in the aeration chamber I. Air flow can be adjusted between chambers I and II by air valves.

Settling

Activated sludge is separated in the settling chamber by gravity and clarified water flows to the disinfection chamber. Activated sludge is pumped back to the aeration chamber I by an air driven ejector pump.

Air flow for the sludge ejector is adjusted so that the sludge return from the settling chamber is about 1/3 of the pipe section. Rest of the air flow produced by the blower is used for aeration and divided equally between tanks I and II.
Sludge has to be removed frequently from the process to maintain a good biological balance in the sewage treatment unit. Sludge is removed by the discharge pump.

Disinfection

Disinfection chemical (e.g. Sodium hypochlorite (NaClO), solution, active chlorine 10%) is added to the clarified water in the disinfection chamber to meet IMO's requirements regarding presence of coliform bacteria in treated water. Residual chlorine must be kept between 2 ppm and 5 ppm. Residual chlorine can be adjusted by the dosing pump's setting and/or timer settings. Treated water can be pumped to sea or ashore by the discharge pump.

Advanced Wastewater Treatment Systems (AWT)

On some cruise vessels, especially many of those traveling to Alaska, sewage and often grey water are treated using AWTs. AWTs generally provide improved screening, biological treatment, solids separation (using filtration or flotation), and disinfection (using ultraviolet light) as compared to traditional MSDs. The AWTs currently used by cruise ships operating in Alaskan waters are discussed in this subsection.

Membrane Bioreactor (MBR)

One of the Advanced Wastewater Treatment Systems is the Membrane Bioreactor (MBR) system. The MBR uses aerobic biological treatment followed by ultra-filtration using membranes and ultraviolet (UV) disinfection.

Membrane

A membrane is defined as a material that forms a thin wall capable of selectively resisting the transfer of different constituents of a fluid and thus effecting a separation, of the constituents. Thus, membranes should be produced with a material of reasonable mechanical strength that can maintain a high throughput of a desired permeate with a high degree of selectivity. The optimal physical structure of the membrane material is based on a thin layer of material with a narrow range of pore size and a high surface porosity. This concept is extended to include the separation of dissolved solutes in liquid streams and the separation of gas mixtures for membrane filtration.

Membranes can be classified by:

- the driving force used for the separation of impurities, such as pressure, temperature, concentration gradient, partial pressure, electrical potential etc.
- the structure and chemical composition
- the mechanism of separation
- the construction geometry of the membrane

Microfiltration (MF) and ultrafiltration (UF) are low pressure driven processes, where feed water is driven through a micro-porous synthetic membrane and divided into permeate, which passes through the membrane, and retenate or reject containing the non-permeating species. In wastewater treatment applications, these membranes process are more effective in removal of particles and microorganisms. Whereas, reverse osmosis (RO) is a high pressure driven process designed to remove salts and low molecular organic and inorganic pollutants. Nanofiltration (NF) operates at a pressure range in between RO & UF targeting removal of divalent ion impurities.

Figure ….. illustrates the size range of various impurities and the application range of the membrane processes.

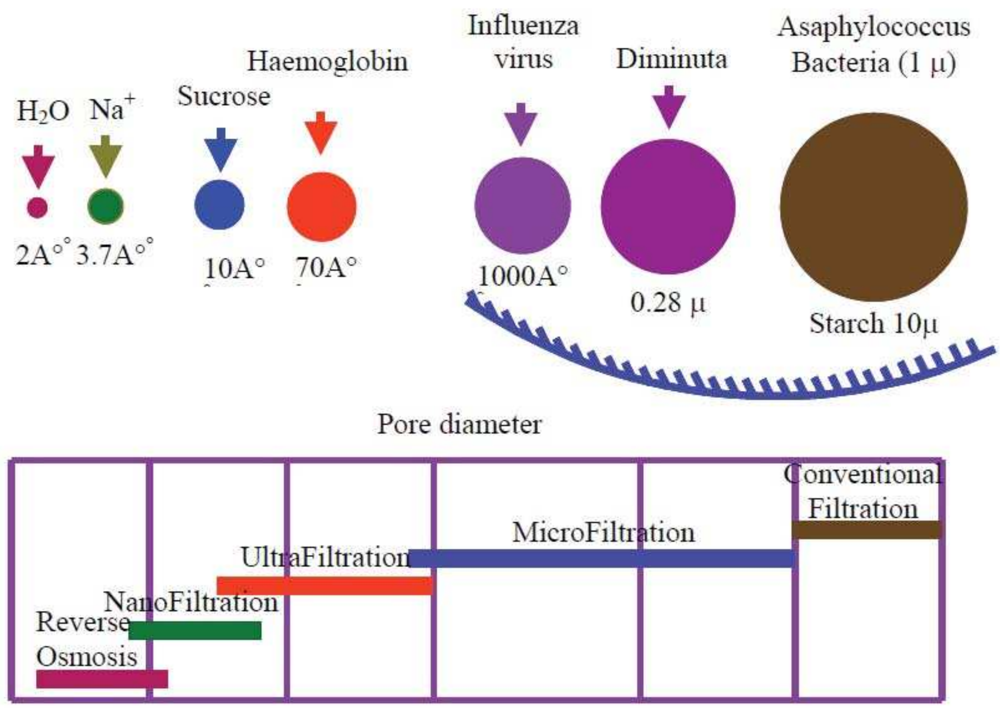


figure … Membrane filtration spectrum

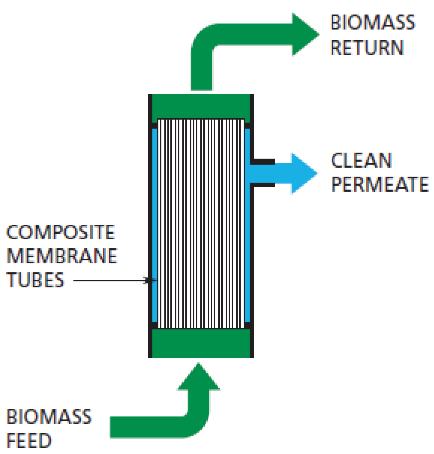


figure …. Ultra filtration membrane module

MBR unit

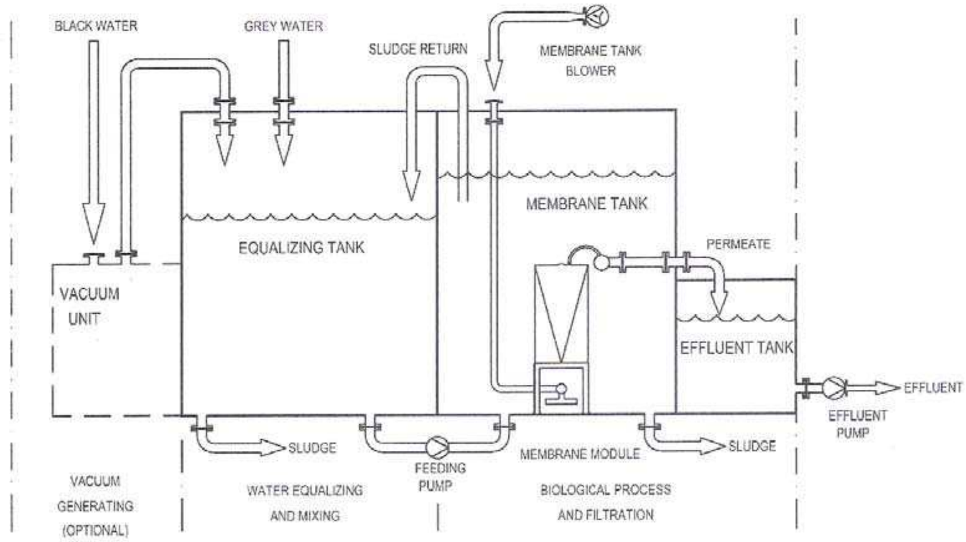


figure … Membrane bioreactor sewage system

Wastewater collecting, equalizing and mixing

Knowledge on the ship operational profile, source and amount of wastewater and the collection methods of the wastewater streams, among others, is the key to the most optimum process. Because wastewater is produced unevenly during the day as the galley is not operating the whole day and different number of crew are awake or asleep at certain times. A constant feed to the MBR unit would give the most optimum treatment.

Pre-treatment

Foreign objects (towels, rubber gloves, rings etc.) in wastewater have to be removed at the front end of the process. Heavy objects are separated and waste is macerated.

Biological process

In a biological treatment process, organics are turned into carbon dioxide, water and biomass. Oxygen supply for the biomass is secured through air diffusers.

Membrane filtration

Clean water is separated from the biomass by membrane filtration. Membrane filtration is a physical barrier, producing treated water without solids. Pressure difference for the membrane filtration is created either by slight vacuum or by gravity head. Treated water do not need any further disinfection and can be discharged directly to the sea.

Sludge

The bioprocess of the MBR produces surplus sludge that needs to be removed from the process. The biomass concentration in the MBR bioprocess is kept constant and sludge removal from the biological process by a pump or by the airlift in the membrane tank to the equalizing tank is controlled by a timer. Sludge discharge from the equalizing tank overboard is manually carried out when allowed by the operating area. Sludge holding capacity of the tank is approximately one week.

Equalizing tank

The purpose of the equalizing tank is to equalize the influent flow prior to feeding to the main MBR treatment stage and provide a source of influent for the feeding pump. Third function of the tank is to store the sludge before disposal.

The equalizing tank is also aerated intermittently by timer control. The purpose of this aeration is to keep the content of the tank completely mixed and thus prevent the sedimentation of the solids to the bottom.

MBR reactor tank or membrane module

The aeration blower is on all the time to avoid that solids will deposit on the nozzles thereby reducing the air distribution to the membranes.

The flowrate, and thus the pressure difference on the membranes is controlled by backpressure in the pipeline going to the effluent tank.

The pH in the MBR reactor tank is continuously measured and controlled. The pH must be maintained between 6.5 and 8. The pH adjustment is done automatically by adding a sodium hydroxide ($NaOH$) solution.

The effluent tank

The level in the effluent tank is controlled by level switches. The pump will start when the high level switch is activated and stop when the low level switch is activated.

Ultraviolet Disinfection

Ultraviolet (UV) light has become an established water treatment disinfection technology due to its extremely effective ability to kill or inactivate many species of disease-causing microorganisms. Ultraviolet light disinfection is effective on bacteria, protozoan parasites and, can also be effective for most viruses, providing sufficiently high UV dosage rates are used.

Ultraviolet energy is found in the electromagnetic spectrum between visible light and x-rays and can best be described as invisible radiation. In order to kill microorganisms, the UV rays must actually strike the cell. UV energy penetrates the outer cell membrane, passes through the cell body, and disrupts its DNA, preventing reproduction. UV treatment does not alter water chemically; nothing is being added except energy. The sterilized microorganisms are not removed from the water. UV disinfection does not remove dissolved organics, in-organics or particles in the water.

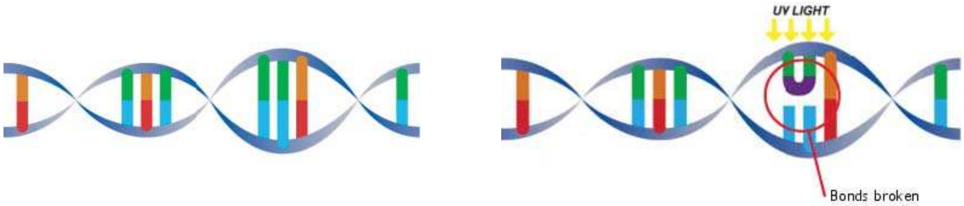


figure …. DNA structure before and after it was treated by UV light

The degree of inactivation by ultraviolet radiation is directly related to the UV dose applied to the water. The dosage, a product of UV light intensity and exposure time, is measured in microwatt second per square centimeter (μws/cm^2). The accompanying table lists dosage requirements to destroy common microorganisms.

The UV units for water treatment consist of a specialized low pressure mercury vapor lamp that produces ultraviolet radiation at 254 nm, or medium pressure UV lamps that produce a polychromatic output from 200 nm to visible and infrared energy. The optimal wavelengths for disinfection are close to 260 nm. The UV lamp never contacts the water, it is either housed in a quartz glass sleeve inside the water chamber or mounted external to the water which flows through the transparent UV tube. It is mounted so that water can pass through a flow chamber, and UV rays are admitted and absorbed into the stream

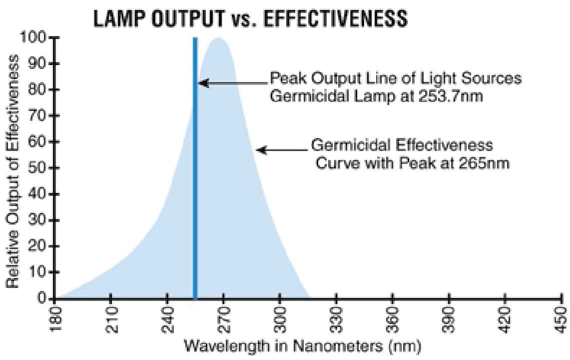


figure …. Graph showing the effectiveness of UV light in disrupting DNA

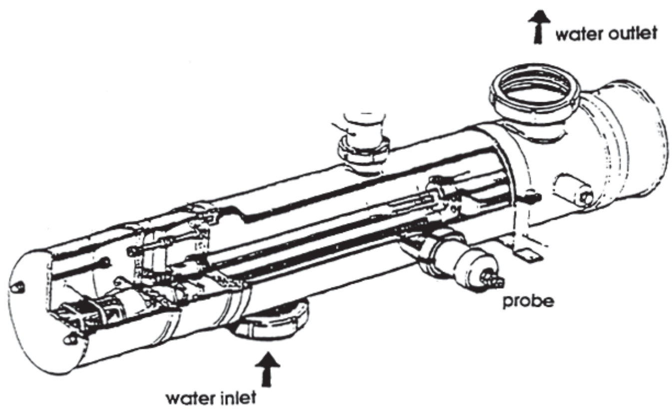


figure …UV water disinfection system

Incinerators

Stricter legislation with regard to pollution of the sea, limits and, in some instances, completely bans the discharge of untreated waste water, sewage, waste oil and sludge. The ultimate situation of no discharge can be achieved by the use of a suitable incinerator. When used in conjunction with a sewage plant and with facilities for burning oil sludge, the incinerator forms a complete waste disposal system.

The International Convention for the Prevention of Pollution from Ships

The construction and operation of an incinerator unit has to comply with regulations of the MARPOL convention.

Annex V contains regulations for the prevention of pollution by garbage from ships. The Annex defines different categories of garbage. One of the categories is incinerator ash. Incinerator ash may be disposed overboard unless it is ash from plastic products which may contain toxic or heavy metal residues.

Annex VI contains regulations for the prevention of air pollution from ships. The Annex defines the operating limits and standards for incinerator units. Each unit installed onboard should have an IMO type approval. Annex VI prohibits incineration of:

a. cargo residues which are categorized by MARPOL Annex I, II and III
b. polychlorinated biphenyls (PCB's)
c. refined petroleum products containing halogen compounds

Incinerator ashes must be declared in the garbage record book. If they have been dumped overboard or delivered to a shore reception facility.

The incinerator (see fig ….)

Solid material, usually in sacks, is burnt by an automatic cycle of operation. Liquid waste is stored in a tank, heated and then pumped to the sludge burner where it is burnt in an automatic cycle. After use the ash box can be emptied overboard.

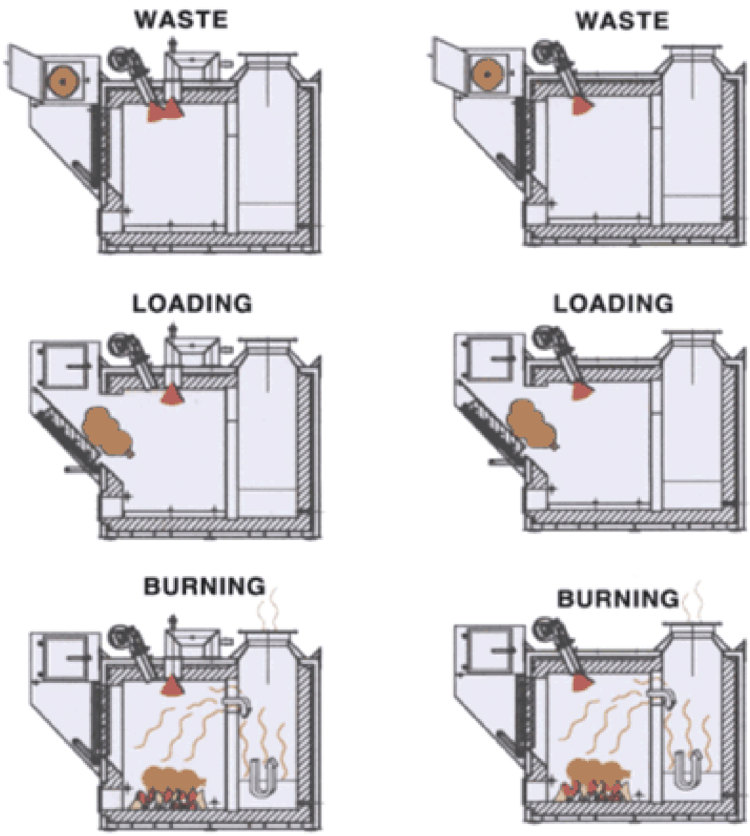
The incinerator is designed with a primary combustion chamber for burning sludge oil and/or solid waste, and a secondary combustion chamber for burning out un-combusted exhaust gases. The primary combustion chamber is equipped with a diesel oil burner (5) called primary burner. The function of this burner is to help heating up the incinerator and is used to ignite the sludge burner. The primary burner is also used when only solid waste is incinerated.

Solid waste is loaded into the incinerator through the charging door (1). Ashes which have to be removed from the incinerator can be cleared through the ash clearing door (6).

Sluice

To avoid flash backs or toxic fumes to come out of the incinerator when charging solid waste, a sluice can be fitted on the unit. This is essential when the incinerator is used in continues operation.

The sluice has an outer and an inner door. When opening the outer door the inner door will separate the sluice from the primary combustion chamber. As soon as the garbage is loaded in the sluice and the outer door is closed the inner door can be opened and the garbage will drop down in the primary combustion chamber.



Simultaneous burning of oil sludge and solid

Solid waste burning

fig … Operating modes incinerator

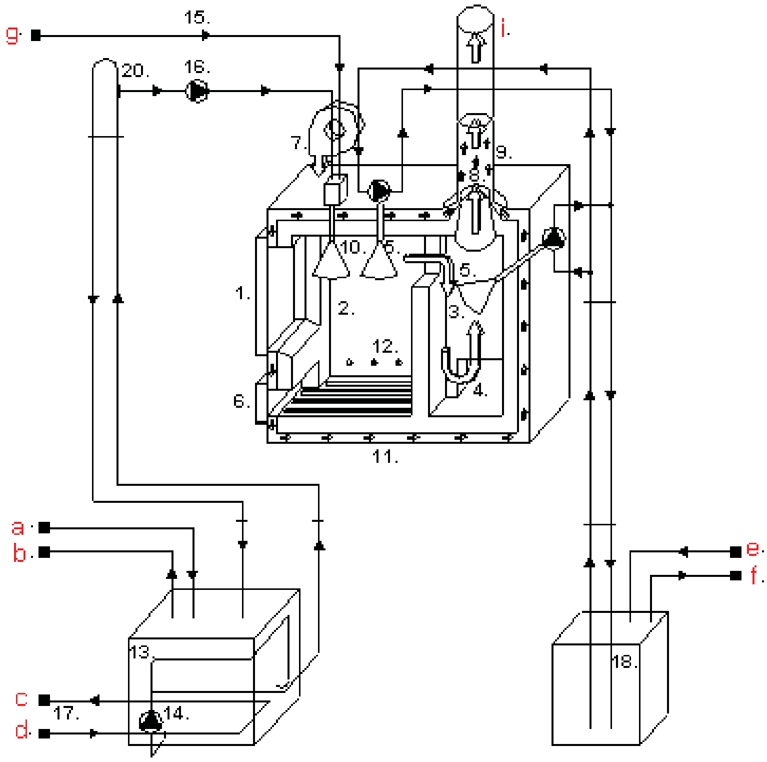


fig … Lay out of incinerator plant

Components

1. Charging Door
2. Combustion chamber
3. Afterburning chamber
4. Secondary Afterburning Chamber
5. Oil Burner with Built-in Pump
6. Ash Cleaning Door
7. Air Blower
8. Induced Draught Air Ejector
9. Damper
10. Sludge Burner
11. Double Air-cooling Wall
12. Combustion Air Inlets
13. Oil Sludge Mixing Tank
14. Mill Pump
15. Compressed Air
16. Sludge Dosing Pump
17. Heating Element
18. Diesel Oil Tank
20. Self-cleaning strainer for sludge oil

Connections

a. Sludge Oil Inlet
b. Steam Inlet
c. Steam Outlet
d. Sludge Oil Ventilation Outlet
e. Diesel Oil Inlet
f. Diesel Oil Ventilation Outlet
g. Compressed Air Inlet
h. Electrical Power Supply
i. Flue Gas Outlet

Sludge burner

The air necessary for the combustion of the sludge is delivered by the air blower (7). The air is passing through the double air cooled wall (11) of the incinerator. This has a dual purpose, the air is heated up for better combustion and at the same time the air flow cools down the outer wall. At the sludge burner (10) the air will start to swirl by a static swirler.

The oil sludge is atomized using pressurized air (15). The holes for the oil sludge are bigger as the holes in the self cleaning strainer (20) so there is no possibility that the holes will clog up.

Even sludge containing approx. 50% of water can be burnt without flame failure.

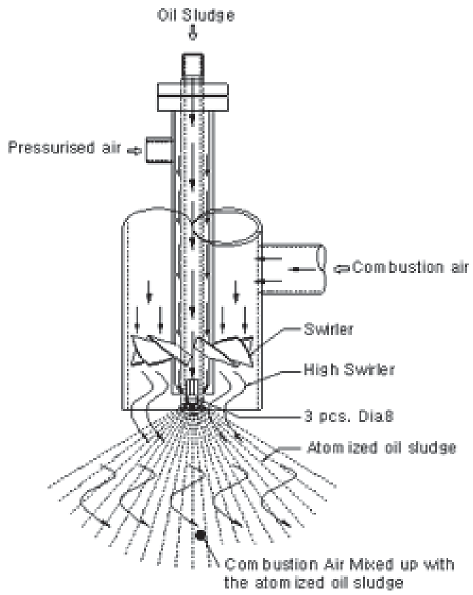


fig … Sludge burner

Primary combustion chamber

The incinerator is designed to burn solid waste and/or oil. The heat from the primary burner will dry out and start burning the solid waste and/or ignite the sludge oil. The very large heat transmission area in the primary combustion chamber optimizes the drying and burning of the solid waste. At the lower portion of the chamber combustion air inlets (12) are located to bring air to the solid waste which is burning on the incinerator floor.

Secondary combustion chamber

The primary and the secondary combustion chamber are separated by wall made of ceramic heavy duty refractory. In the secondary combustion chamber the gases from the primary combustion chamber will burn out.

The flue gasses go through the afterburning chamber (3) and secondary afterburning chamber (4) having a residence time of 2secs at 850 ^{o}C or 1sec. at 950 ^{o}C, self secondary burning for un-combusted flue gases is accomplished to minimize harmful emission. Any gases not completely burned from primary chamber will be combusted definitively. For instance CO will only be transformed in CO_2 above 700 ^{o}C

The function of the oil burner (5) placed in the secondary combustion chamber is to heat up the incinerator before operation and will also start when the temperature of the flue gas would drop below 850 ^{o}C.


fig … Flue gas flow

Induced Draught Air Ejector

A part of the air coming from the air blower (7) is flowing to the induced draught air injector (8). The purpose of the ejector is to create a low pressure at the smoke stack. This will result in a low pressure in the combustion chambers. Due to this no flue gas draft fan is needed. The second function of the ejector is to shock cool the flue gas in order to eliminate the formation of harmful dioxins.

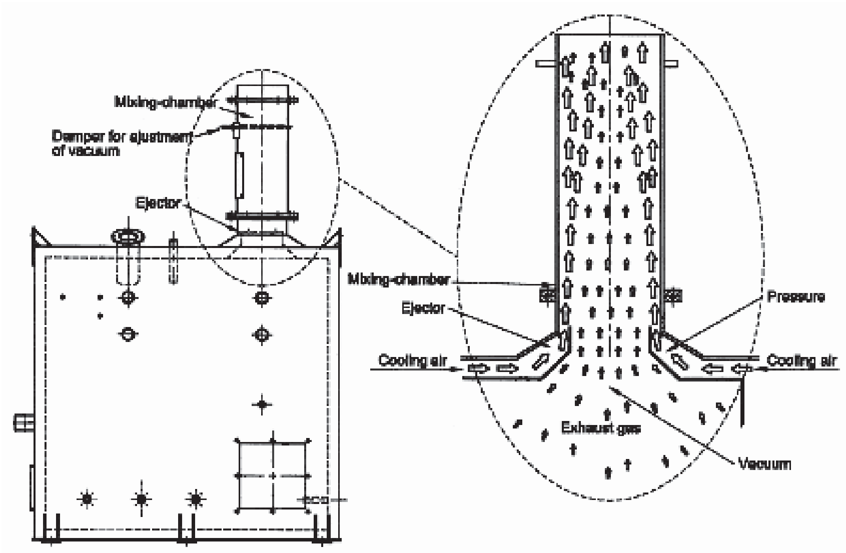


fig … Induced draught air ejector

Self-cleaning strainer for sludge oil

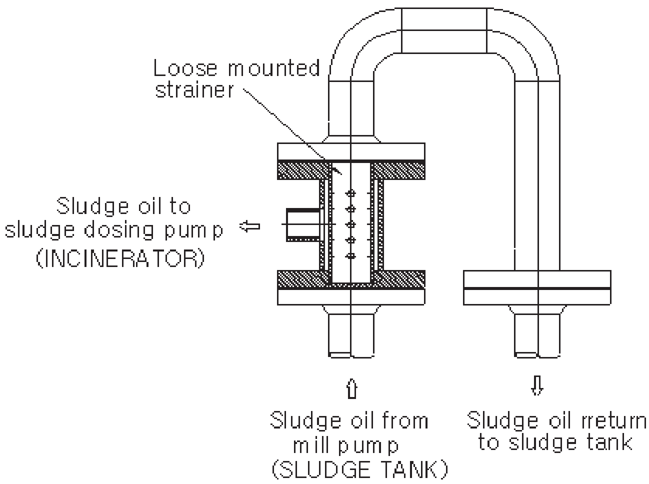


fig … Self-cleaning strainer for sludge

The pressurized sludge oil supplied from the Mill pump (14) flows through 6mm strainer holes in the strainer (20) to the sludge burner for burning. Oversize particles are returned to the sludge tank for shattering by the Mill pump. There is no possibility for blockage of the strainer which keeps maintenance and cleaning work to a minimum and operation time to a maximum.

Mill pump

The Mill pump (14) fitted on the oil sludge mixing tank (13) is a centrifugal pump of which the impeller is equipped with knives. They comminute of particles contained in the sludge oil and effective mixture of the sludge oil.

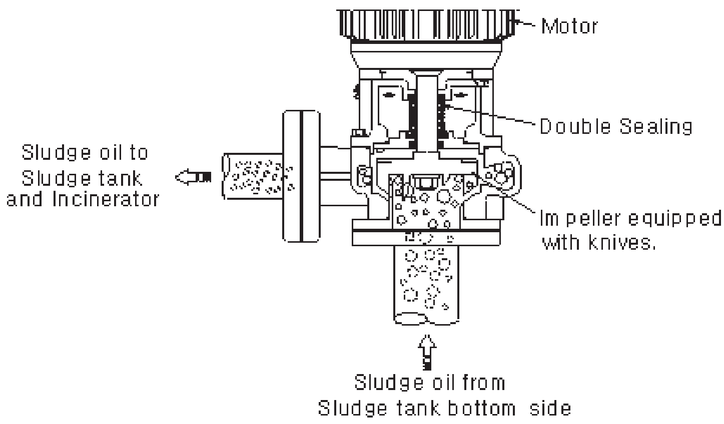


fig … Mill pump

Start-up of the incinerator

1. Start air blower (7) and close damper (9)
2. Start oil burner (5) in secondary combustion chamber to heat up incinerator
3. At 400 ^{o}C start oil burner (5) in primary combustion chamber to further increase the temperature
4. At 600 ^{o}C start sludge oil burner (10). It is ignited by oil burner (5), damper (9) opens
5. Oil burner (5) in primary combustion chamber switches of after 30 sec.
6. At 850 ^{o}C oil burner (5) in secondary combustion chamber switches off

Oil Water Separators

The bilge is the lowest part of the ship and the point of accumulation of a complex cocktail of compounds including soaps, detergents, solvents, soot and other particular matter and sometimes microbial contamination.

The major part of the cocktail will consist of water coming from leakages, repairs and cleaning activities. From time to time the bilges have to be pumped out to avoid problems as, free surface effect influencing the stability and otherwise the water level might rise above the engine room flooring with serious consequences for both men and machinery.

In order to minimize damage to the environment the contaminated bilge water cannot be pumped out overboard untreated. Annex I of the MARPOL Convention contains clear regulations regarding the discharge of bilge water from machinery spaces.

Each vessel of 400 GT and over must have a holding tank, normally called the sludge tank, to collect oil residues, such as those resulting from the purification of fuel and lubricating oils and oil leakages in the machinery spaces.

From the sludge tank no pipelines are allowed to go directly overboard other than the standard discharge connection. This connection has a standardized discharge flange connection in order that a connection can be made to a shore reception facility anywhere in the world.

Each vessel of 400 GT and over must have a Bilge Separator or Oil Water Separator installed. The basic requirements for the Bilge Separator itself are:

- The maximum oil contents which may be pumped overboard is 15 parts per million (15 ppm).
- The separator must be equipped with a 15 ppm alarm.
- The separator must be equipped with an automatic stopping device in case the concentration would be more as 15 ppm.

Next to these general requirements the IMO adopted in July 2003 Resolution MEPC 107(49) with additional specific requirements for all Bilge Separators installed on or after 1 January 2005.

The most important additional requirements are:

- The installations must be tested according standardized protocols in order to make sure that they will meet the standards of MARPOL Annex I.
- Must be able to handle emulsions of oil, water and other substances.
- The 15 ppm alarm must be protected against tampering by means of a seal. Tampering will become visible by the breakage of the seal.

Emulsification

The basic concept of many Oil Water Separators is based on gravity separation devices such as parallel plate coalescers and centrifuges. The principle of operation exploits the difference in buoyancy of two or more immiscible liquids or materials dispersed in each other. Particular matters or oil which will not separate from the water under the influence of gravity will be referred as emulsion.

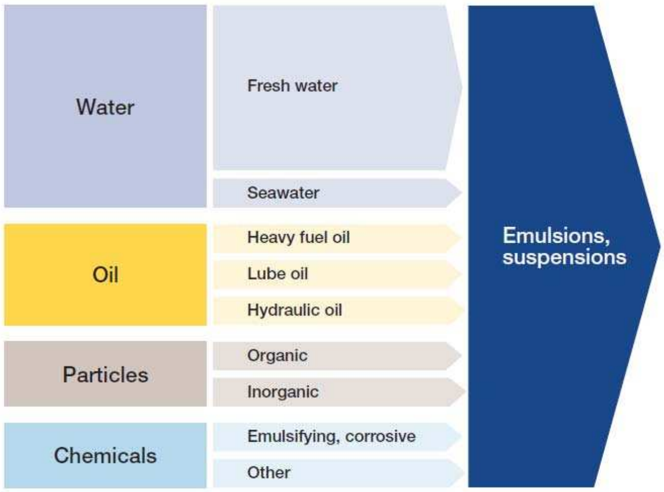


fig … Bilge water composition

Most oils are less dense in water, and if oil and water are mixed then the oil will simply float to the surface. In emulsions, the oil is dispersed as liquid droplets through the continuous phase of the water. Those droplets want to combine together again to form a single blob of oil. To prevent them from doing this, emulsions contain a surfactant which coats the surface of each drop and prevents the droplets from coalescing. In practice, a mixture of oil, water and added surfactant are put through a blender, for example, and the product is an emulsion.

However the oil is still less dense than the water. So each drop is prone to floating upwards. This process is called creaming - the oil droplets will gradually form a dense layer at the top. An emulsion is therefore described as unstable with respect to creaming (creaming is just upside down sedimentation). To prevent creaming, many emulsion products, like cleaning agents, contain additives called stabilizers that inhibit creaming. Stabilizers work by increasing the viscosity of the continuous phase in which the oil droplets are immersed, or by inducing some kind of interaction between droplets.

The properties of emulsions depend on virtually everything - the continuous phase, temperature, average droplet size, droplet size distribution, the amount of oil dispersed in the water (called the oil volume fraction), the oil itself, additives and other substances that are present. On top of that a ship is a floating object which can roll and pitch creating motion of the mixture in the bilges which results in a continuous mixing of the liquids.

A 15 ppm Bilge Separator must be capable of handling any oily mixtures from the machinery space bilges and be effective over the complete range of oils which might be carried on board ship, and deal satisfactorily with oil of very high relative density, or with a mixture presented to it as an emulsion. Cleansing agents, emulsifiers, solvents or surfactants used for cleaning purposes may cause the bilge water to emulsify.

Inverto Bilge Separator

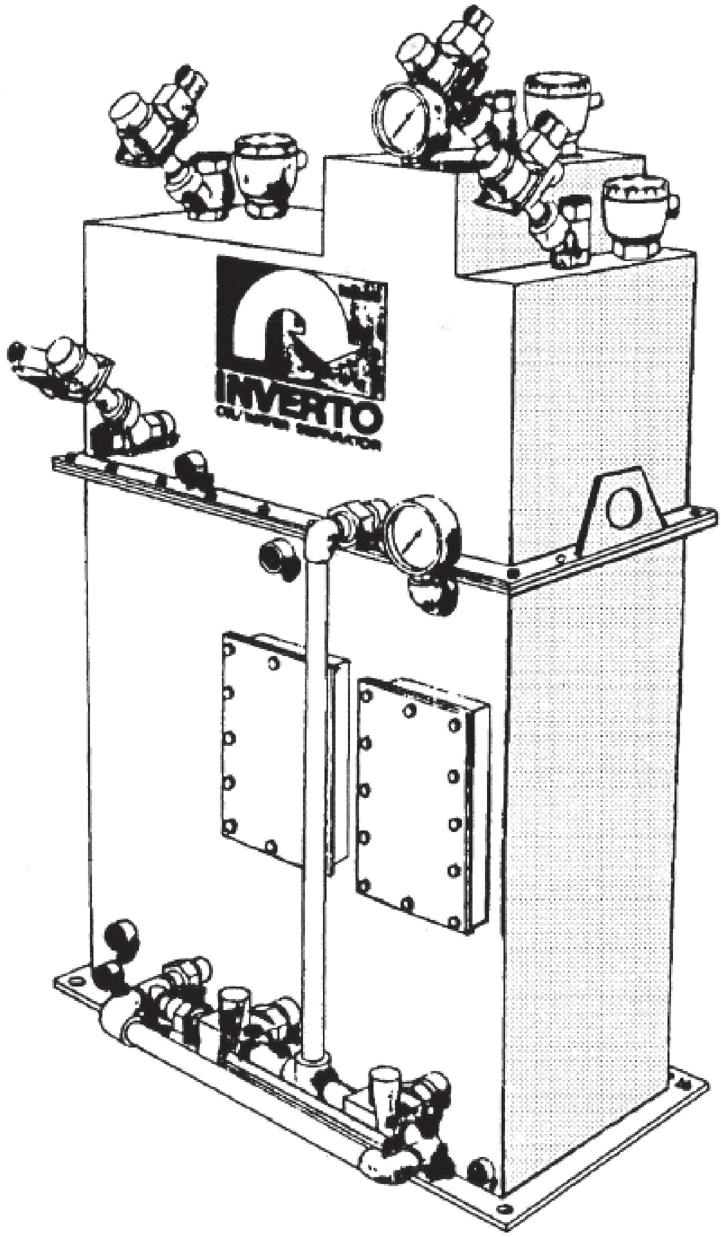


fig … Inverto 15 ppm Oil Bilge Separator

The Inverto oil bilge separator complies with all the IMO requirements. The clean water discharge will contain less then the maximum of 15 parts per million (ppm). During tests results with a maximum of 3 ppm were measured.

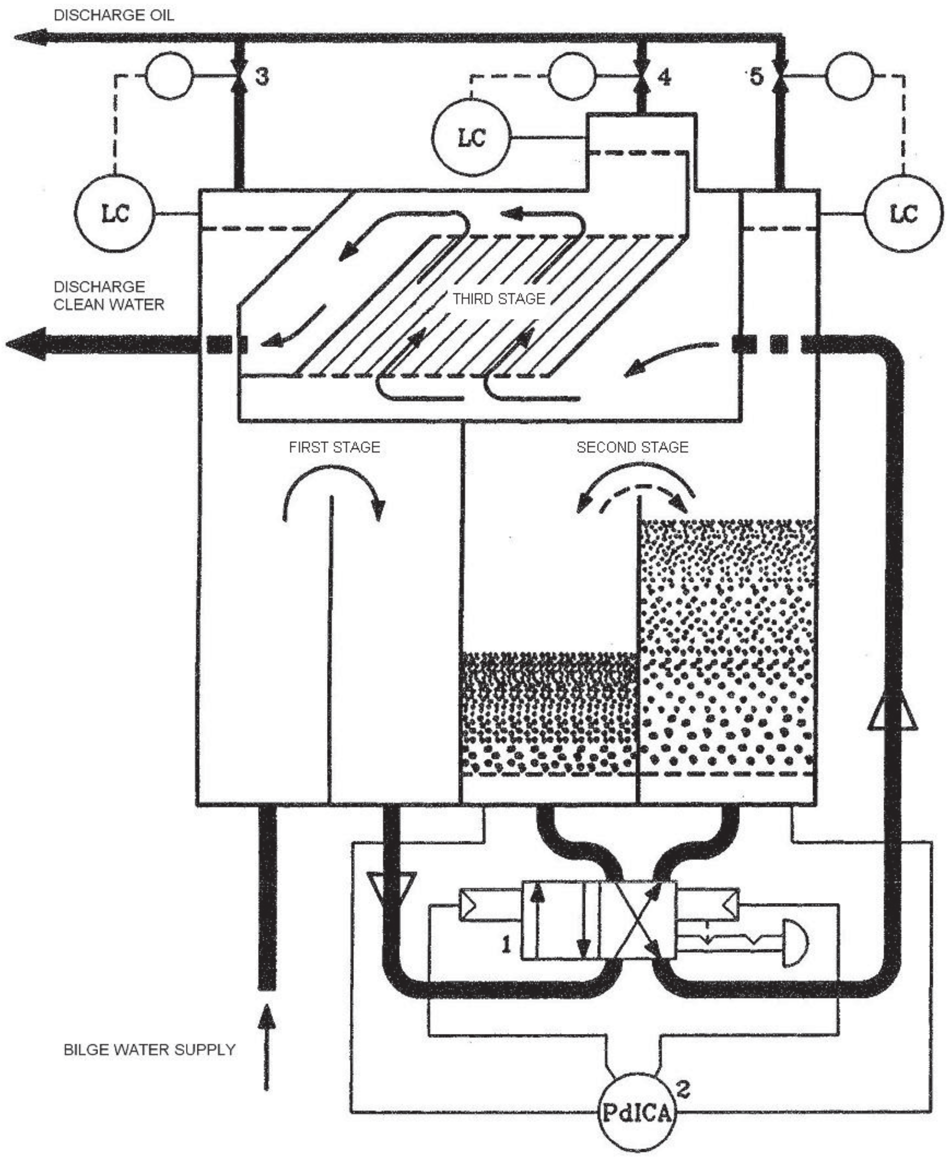


fig … Cross section Inverto Oil Bilge Separator

The unit must always be completely filled with liquid. In the first stage the bilge water is forced to flow over a vertical baffle plate. The flow of the liquid is altered 180 degrees. This creates a slow down in the flow, allowing larger oil droplets to rise to the top, as oil has a lower density as water. Separation in the first stage is based on gravity. If more oil collects at the top the interface between water and oil will come down. The level controller (LC) will be activated and will open valve #3 of the first phase and at the same time close the clean water discharge. Oil will flow out of the separator to the sludge tank allowing the interface to rise. The LC will give another signal to the valve #3 to close and the clean water discharge to open.

The second stage consists of two identical sections. Depending on the position of shuttle valve #1 the bilge water will be added to the left or right side. The supply to this stage takes place through small holes in order to increase the speed of the liquid. Both sections are filled with four layers of porous grains with different sizes and density. As a result of the high velocity of the entering liquid the grains will start to whirl. This will result in coalescence (sticking of oil to a porous surface) and coagulation (adhesion between oil droplets). As soon as the oil droplets are sufficient in size, they will be released and float to the top. While rising to the top they might bump to other grains with oil and take that oil with them.

In the other section of the second stage the flow is downward. The water, which is already partly cleaned, will compact the grains together, creating a filter. The small oil droplets which still can be in the water will remain on the surface and will coagulate to bigger droplets and rise to the top. The water which will flow out at the bottom is almost clean and any oil still present will be in larger coagulated droplets.

If the interface between oil and water of the second stage will come down the level controller (LC) will activate valve #5 and the clean water discharge until the sufficient oil has been discharged.

In the third stage the separation will take place by gravity force and surface tension. The oil droplets will stick to the bottom side of the plates and if they are big enough slowly rise to the top. This section also has a level controller to control the interface level.

Dirt and other particles which might have entered the oil bilge separator will be separated and collected in the downstream on top of the porous grains of the second stage. An increase of dirt and particles on top of the porous grains will create clogging up. This results in a smaller flow of water going down which creates a drop in pressure at the bottom side. The pressure difference between the two sections of the second stage will increase. This is measured by the Pressure differential Indicator Controller and Alarm (PdICA). The PdICA will change over valve #1 resulting in an interchange of the flow through the second phase. The majority of the dirt and other particles will rise to the top and are removed with the oil.

Facet Oil Bilge Separator

The integrated vertical helical rotary pump draws the water/oil mixture from the bilge and delivers it into the inlet equalization chamber with a low velocity. The flow is then split into two almost vertical upward streams in which the gross oil is accelerated towards the oil collection area. The water, now only containing low concentration of oil, deflects towards the coalescing plate inlet section.

In the plates, oil particles down to 15 micron are fully removed from the water-phase and transferred to the oil collection area through weep holes. A scavenging/dry run prevention line is provided between the water outlet compartment and the suction side of the pump.

The level control device initiates automatic evacuation of separated oil to the sludge tank. The electrical level controller operates a solenoid valve. This 2/2 valve controls the air to the diaphragm of the oil discharge valve. The level controller consists of three probes with different lengths. The middle probe will give the signal to the solenoid valve to allow the air to open the oil discharge valve. As soon as the interface inside the separator has gone up, the shortest probe will give a signal to close the valve. The longest probe acts as the alarm signal in case the valve is not opened and the interface will go down more.

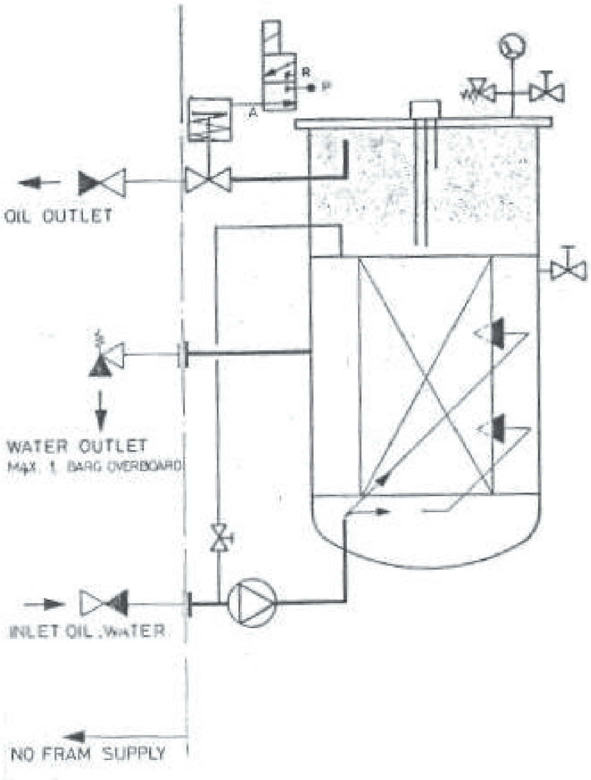


fig … Facet 15 ppm Oil Bilge Separator

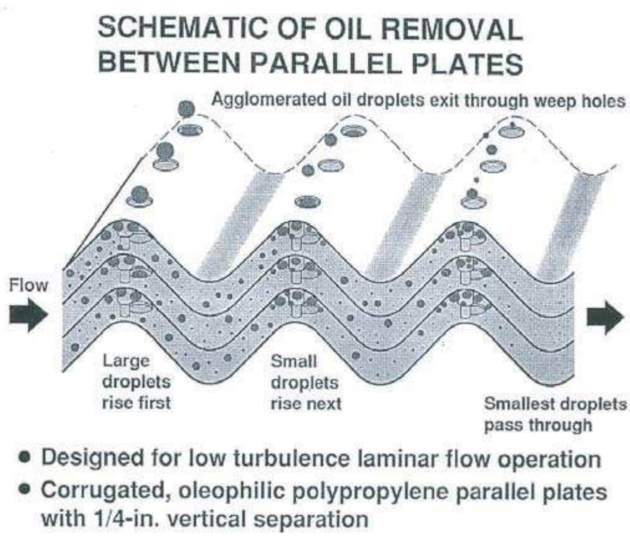


fig … Coalescing plate pack

The high efficiency of the coalescing plate packs is obtained by their use of both gravity separation and hydrodynamic coalescence. The plate shape, spacing and influent velocity have been combined to give complete removal of all oil particles of 20 microns and over. The plate shape promotes hydrodynamic collision of oil particles, and these collisions also mean many particles between 5 and 20 microns are captured by the plates.

Alfa Laval Bilge Separator

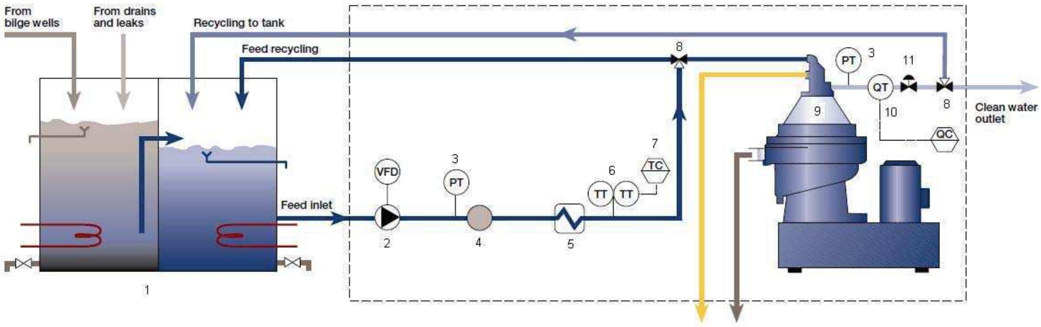


fig … Alfa Laval Bilge Separator

1. 2 stage bilge water settling tank
2. Feedpump with variable flow control
3. Pressure transmitter
4. Strainer
5. Pre Heater
6. Temperature transmitter
7. Temperature controller
8. 3 way change over valve
9. Centrifugal separator
10. Oil in water monitor
11. Constant pressure modulating valve

<u>Operating principle</u>

A feed pump with variable frequency drive control pumps oily water from the bilge tank water settling tank to the system.

Bilge water then passes through a strainer which traps large particles from the fluid before entering a heat exchanger, which raises the fluid temperature to 60^o- 70^o C, for optimum separation efficiency.

A 3 way change over valve then directs the fluid to the separation stage if all process conditions, such as feed temperature, feed pressure and separator speed, fall within preset process values. If any process condition is not met, the valve re-circulates the fluid back to the bilge water settling tank.

A high speed centrifugal separator continuously processes large volumes of bilge water. The oil outlet continuously discharges separated oil and emulsions. Solids that collect at the separator bowl periphery are discharged intermittently. Discharge occurs at preset intervals, which is generally set at 20 minutes, depending on the installation. Solids are then directed to the sludge tank.

A build-in water pump, or paring disc, continuously discharges separated bilge water through the clean water outlet. The destination of the separated bilge water depends upon its oil content, which is continuously monitored at a sampling point by an oil-in water monitor.

If oil content is below the pre-set ppm alarm limit, the separated bilge water can be pumped either directly overboard or to a clean bilge water holding tank for discharge overboard later. If oil content exceeds the ppm alarm limit, the effluent is re-circulated for re-processing.

<u>The separation process</u>

Figure … shows the inside of the centrifugal separator. The bilge water is fed into the separator through the centre pipe and enters the bowl at the bottom side. The bowl contains a disk stack of metal disks at an angle of 45 degree. The bilge water rises through holes in the disks.

The bowl with the disks is rotates with a gravitational force of 6,000 G generated at 8,000 rpm. Oil and emulsions are separated from the oily water and discharged continuously through the oil outlet. The high centrifugal force will separate the heavy from the lighter parts of the solution. Oil droplets with the lowest density will move to the centre of the separator. At that area coalescence will take place, smaller droplets will collide and become bigger droplets. The centrifugal force is strong enough to break emulsions into the different components. Oil, water and solid particles are discharged separately from the separator.

Solids with a higher density will move outwards and collide with other particles, called flocculation and collected at the separator bowl periphery are discharged intermittently at preset intervals and are directed to a collecting tank for sludge or waste oil..

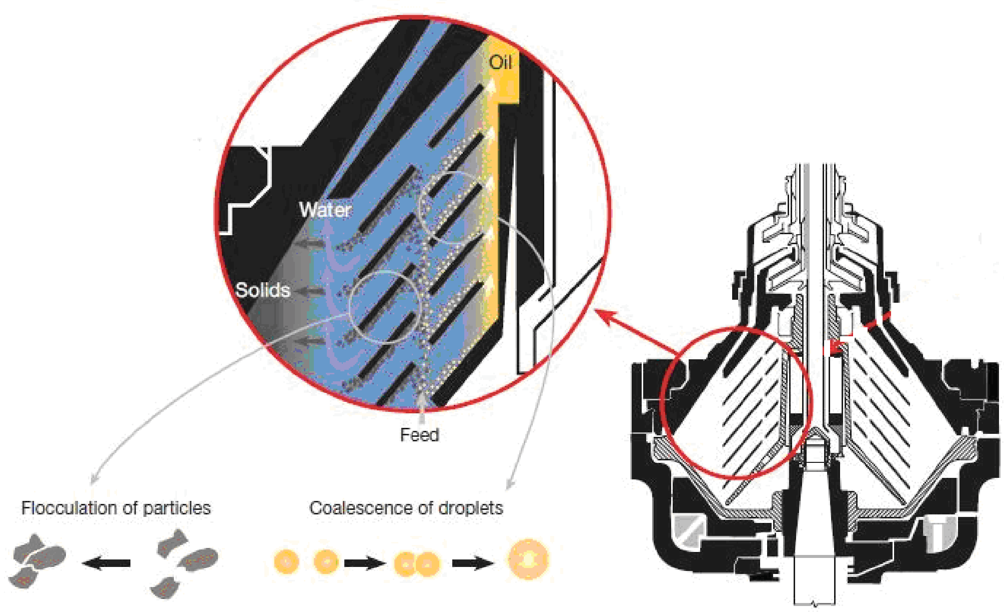


Fig … Separation process inside separator

Heli-Sep Bilge Separator

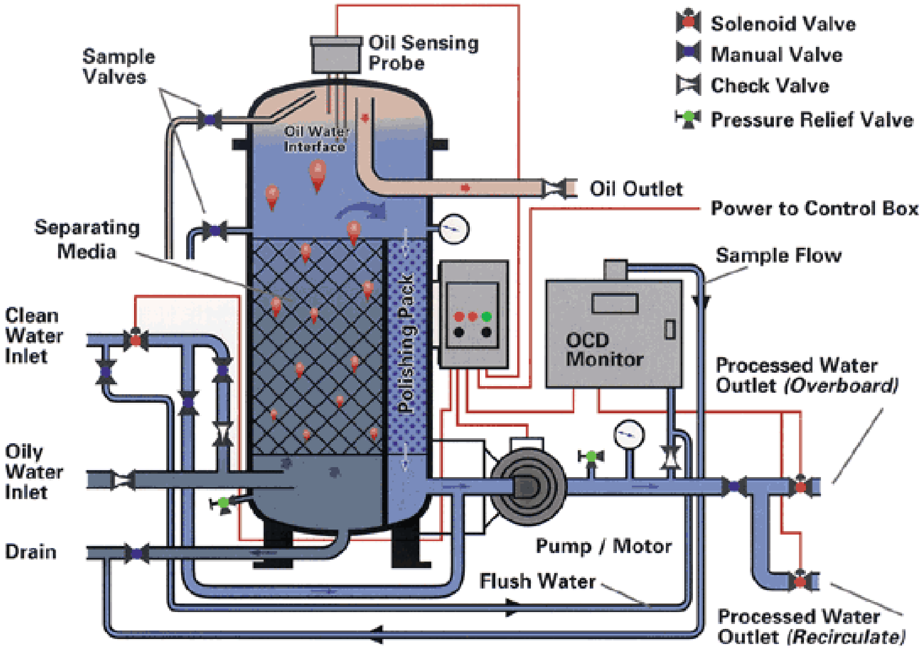


fig … Heli-Sep 15 ppm Oil Bilge Separator

The Heli-Sep Oil Bilge-Bilge Separator is a single vertical cylindrical vessel utilizing system generated vacuum, fluid velocity reduction, differential specific gravity, and coalescence to separate and remove non-soluble oil, solids and entrained air from oily water.

Oily water is pulled into the system by the vacuum created by low shear positive displacement pump located on the treated water discharge. The bilge water enters the separator at point 1 to the first stage.

As the water enters the vessel solids begin to fall out of suspension and drop to the bottom, the water flows into the inclined corrugated matrix where the oil impinges on the surface and additional solids fall out. At 2 the free oil droplets coalesce and become larger, and float to the top of the separator for collection and discharge while dirt and other particles settle on the bottom for removal to the drain.

To remove the remaining oil from the water the liquid is pulled through a polishing pack which consists of oleophilic material that scrubs the remaining oil out of the water.

fig … Heli-Sep Oil Bilge Separator

The oil discharge sensor 3 initiates the automatic removal of oil from the separator in the same way as described by the Facet oil bilge separator.

Before the water can be pumped overboard the oil content is measured by the Oil Contents Discharge Monitor (OCD Monitor). The oil level in ppm is displayed on a LED display.

The function of the OCD not only to measure the oil content but also to avoid that water with more than 15 ppm will be pumped overboard. If the oil level exceeds the set point the OCD close the overboard valve and open the recirculate valve until the oil content falls below the set point, simultaneously triggering alarm.

The oil bilge separator and the OCD Monitor can be cleaned by flushing clean water through the units. The polishing pack can be cleaned by back-flushing clean water from bottom to top.

Handling emulsions

The above described Bilge Separators are all based on gravity separation and will, except for the Alfa-Laval separator, not be able to separate emulsions. In order to remove or break the emulsions a second separation unit can be placed in series with the Bilge Separator. In general this second stage separation is done by filtering techniques.

The Spir-O-Later

This unit is placed in series with the Heli-Sep Bilge Separator.

1. Processed water from the first stage is pumped through the second stage at the optimum flow and pressure for the membranes.
2. The membranes repel oil at the surface and attract water. Their 0.01 micron pores present a physical barrier that rejects oil molecules and particulates while allowing water to pass.
3. Water permeating the membranes is clean with less than 5 ppm and can be discharged overboard.
4. Concentrated oil waste—the “reject”—is sent to the sludge tank.

Fig … Second stage filter unit

How the membrane works

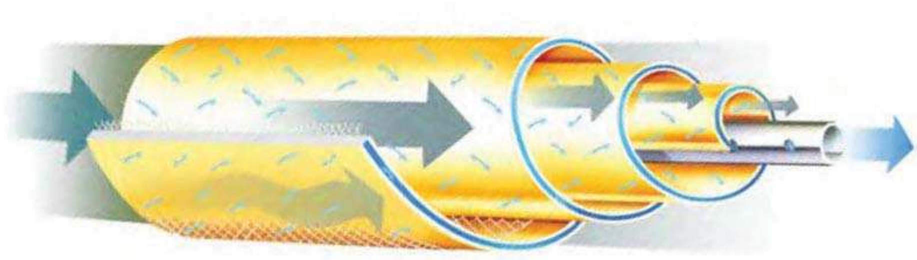


fig … Water is pumped from the outside through the membrane to come out clean through the central core

The flow continuously pushes bilge water between the layers of membranes. The smaller water molecules are forced into the center of the membrane while the larger molecules of oil remain outside. The two parallel streams of fluids come out the end of the membrane – clean water in the center, ready to be freely discharged and the oily residue on the outside to be removed.

The unit contains multiple membrane sheets, spirally wound around a central core (permeate tube). Each sheet consists of oleophobic (oil repelling) outer layers and a hydrophilic (water attracting) central layer to collect the separated clean water. The spiral design provides a large surface area for more effective demulsification.

15 ppm Alarm

Figure … shows the location of the 15 ppm alarm. It is placed in the outlet of the separator connected to the pipeline to the overboard discharge. The position of the alarm sensor is very sensitive as it must be able to measure the real level of oil in the discharge line. The processed water from the separator will flow continuously through the sensor and from the sensor it is returned to the bilge.

The sensor has a clean water supply. This is used to flush the sensor and check and if needed to adjust the 0 ppm point of the alarm. This is not the same as calibrating the alarm, as this can only be done by the manufacturer.

The oil content in the processed water is measured using an optical sensor. The combination of light scattered and absorbed in the sample stream. The sensor signals are then processed by a microprocessor to produce linearised output.

The microprocessor continuously monitors the condition of the sensor components and associated electronics to ensure that calibration accuracy is maintained over time and extremes of environmental conditions.

The two valve handles for sample and clean water are mechanically interlocked. If the valves are set to let clean water flow through the sensor the overboard discharge is automatically closed. The overboard will only be available when the valve handles are placed in normal position.

The display unit does not only display the actual value of the oil content of the processed water, it is also possible to track back the values of earlier dates. According MEPC 107(49) the unit must be able to store data for at least 18 months.

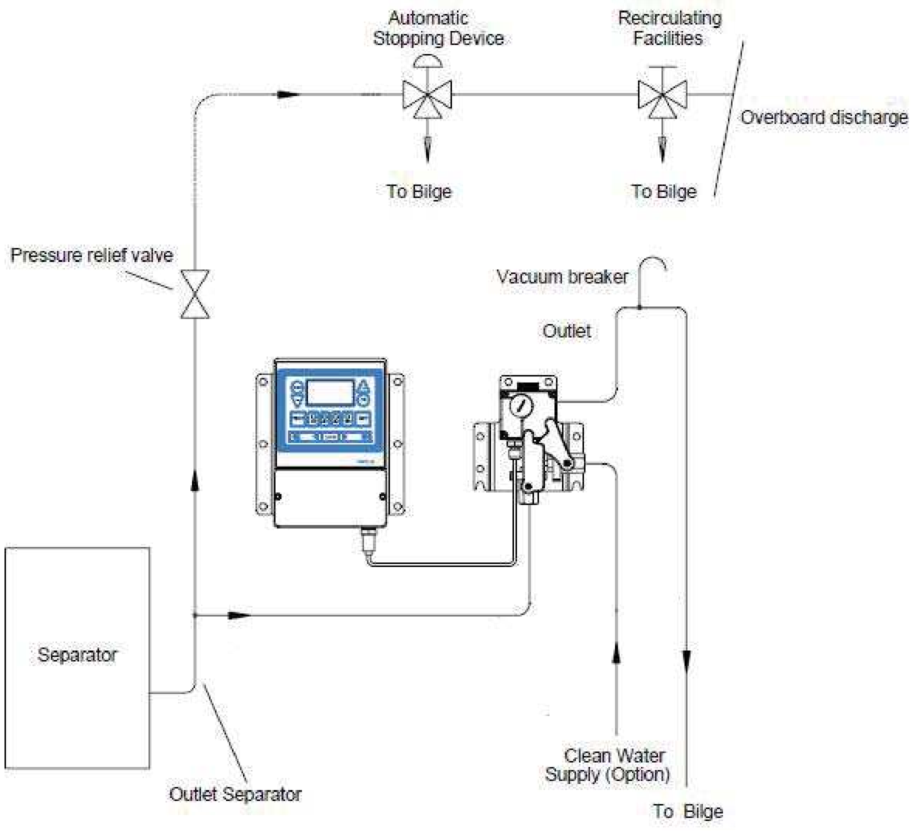


fig .. 15 ppm bilge separator alarm

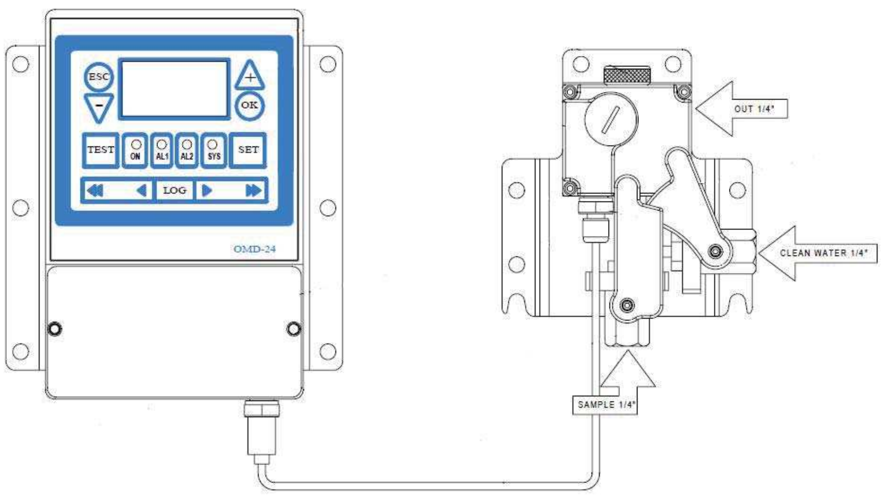


fig … 15 ppm Alarm display and sensor