INSTITUTE OF TEXTILE TECHNOLOGY CHOUDWAR

Sem: 5th Mechanical Engineering

Hydraulic Machines & Industrial Fluid Power

Chapter-1
Hydraulic Turbines

S K Panigrahi
Lect Mechanical
ITT, Choudwar

18

Hydraulic Machines—Turbines

18.1. INTRODUCTION

Hydraulic machines are defined as those machines which convert either hydraulic energy (energy possessed by water) into mechanical energy (which is further converted into electrical energy) or mechanical energy into hydraulic energy. The hydraulic machines, which convert the hydraulic energy into mechanical energy, are called *turbines* while the hydraulic machines which convert the mechanical energy into hydraulic energy are called *pumps*. Thus the study of hydraulic machines consists of study of turbines and pumps. Turbines consists of mainly study of Pelton turbine, Francis Turbine and Kaplan Turbine while pumps consist of study of centrifugal pump and reciprocating pumps.

18.2. TURBINES

Turbines are defined as the hydraulic machines which convert hydraulic energy into mechanical energy. This mechanical energy is used in running an electric generator which is directly coupled to the shaft of the turbine. Thus the mechanical energy is converted into electrical energy. The electric power which is obtained from the hydraulic energy (energy of water) is known as *Hydro-electric power*. At present the generation of hydro-electric power is the cheapest as compared by the power generated by other sources such as oil, coal etc.

18.3. GENERAL LAYOUT OF A HYDRO-ELECTRIC POWER PLANT

Fig. 18.1 shows a general lay-out of a hydro-electric power plant which consists of :

(*i*) A dam constructed across a river to store water.

(*ii*) Pipes of large diameters called penstocks, which carry water under pressure from the storage reservoir to the turbines. These pipes are made of steel or reinforced concrete.

(*iii*) Turbines having different types of vanes fitted to the wheels.

(*iv*) Tail race, which is a channel which carries water away from the turbines after the water has worked on the turbines. The surface of water in the tail race is also known as tail race.

18.4. DEFINITIONS OF HEADS AND EFFICIENCIES OF A TURBINE

1. Gross Head. The difference between the head race level and tail race level when no water is flowing is known as Gross Head. It is denoted by 'H_g' in Fig. 18.1.

2. Net Head. It is also called effective head and is defined as the head available at the inlet of the turbine. When water is flowing from head race to the turbine, a loss of head due to friction between the water and penstocks occurs. Though there are other losses also such as loss due to bend, pipe fittings, loss at the entrance of penstock etc., yet they are having small magnitude as compared to head loss due to friction. If 'h_f' is the head loss due to friction between penstocks and water than net heat on turbine is given by

$$H = H_g - h_f \qquad ...(18.1)$$

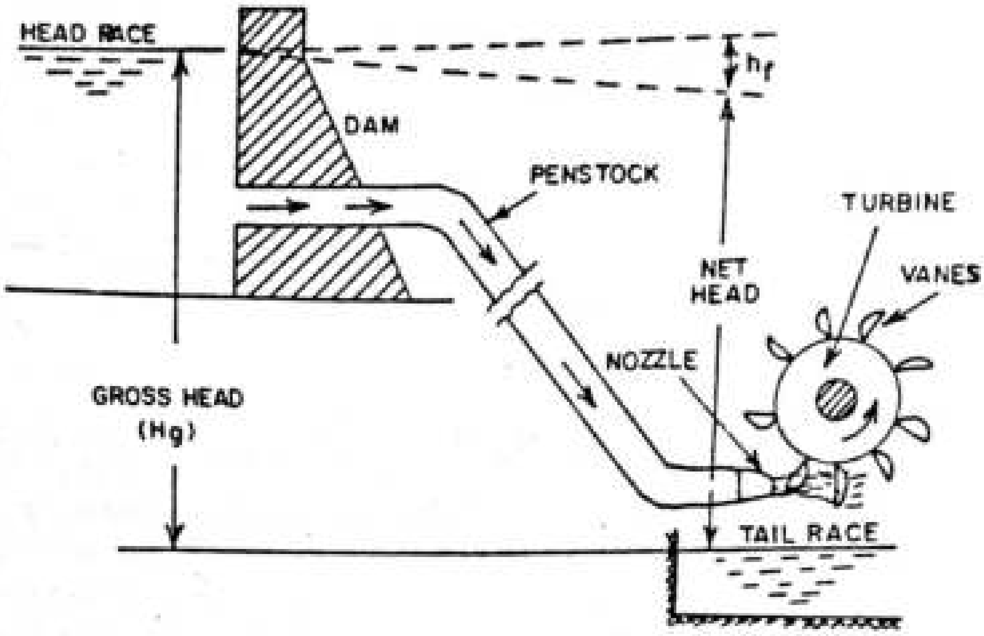


Fig. 18.1. Layout of a Hydro-electric Power Plant.

where H_g = Gross head, $h_f = \dfrac{4 \times f \times L \times V^2}{D \times 2g}$,

in which V = Velocity of flow in penstock

L = Length of penstock

D = Diameter of penstock.

3. Efficiencies of a Turbine. The followings are the important efficiencies of a turbine.

(a) Hydraulic Efficiency, η_h
(b) Mechanical Efficiency, η_m
(c) Volumetric Efficiency, η_v and
(d) Overall Efficiency, η_o

(a) **Hydraulic Efficiency** (η_h). It is defined as the ratio of power given by water to the runner of a turbine (runner is a rotating part of a turbine and on the runner vanes are fixed) to the power supplied by the water at the inlet of the turbine. The power at the inlet of the turbine is more and this power goes on decreasing as the water flows over the vanes of the turbine due to hydraulic losses as the vanes are not smooth. Hence the power delivered to the runner of the turbine will be less than the power available at the inlet of the turbine. Thus, mathematically, the hydraulic efficiency of a turbine is written as

$$\eta_h = \frac{\text{Power delivered to runner}}{\text{Power supplied at inlet}} = \frac{\text{R.P}}{\text{W.P.}} \quad \text{...(18.2)}$$

where R.P. = Power delivered to runner *i.e.* runner power

$$= \frac{W}{g}\frac{[V_{w_1} \pm V_{w_2}] \times u}{1000} \text{ kW} \quad \text{...for Pelton Turbine}$$

$$= \frac{W}{g}\frac{[V_{w_1}u_1 \pm V_{w_2}u_2]}{1000} \text{ kW} \quad \text{...for a radial Flow Turbine}$$

W.P. = Power supplied at inlet of turbine and also called water power

$$= \frac{W \times H}{1000} \text{ kW} \quad \text{...(18.3)}$$

where W = Weight of water striking the vanes of the turbine per second

$= \rho g \times Q$ in which Q = Volume of water/s

V_{w_1} = Velocity of whirl at inlet

V_{w_2} = Velocity of whirl at outlet

u = Tangential velocity of vane

u_1 = Tangential velocity of vane at inlet for radial vane

u_2 = Tangential velocity of vane at outlet for radial vane

H = Net head on the turbine.

Power supplied at the inlet of turbine in S.I. units is known as water power. It is given by

$$\text{W.P.} = \frac{\rho \times g \times Q \times H}{1000} \text{ kW} \quad \text{...(18.3 A)}$$

For water, $\rho = 1000 \text{ kg/m}^3$

$$\therefore \quad \text{W.P.} = \frac{1000 \times g \times Q \times H}{1000} = g \times Q \times H \text{ kW} \quad \text{...(18.3 B)}$$

The relation (18.3 B) is only used when the flowing fluid is water. If the flowing fluid is other than the water, then relation (18.3 A) is used.

(*b*) **Mechanical Efficiency** (η_m). The power delivered by water to the runner of a turbine is transmitted to the shaft of the turbine. Due to mechanical losses, the power available at the shaft of the turbine is less than the power delivered to the runner of a turbine. The ratio of the power available at the shaft of the turbine (known as S.P. or B.P.) to the power delivered to the runner is defined as mechanical efficiency. Hence, mathematically, it is written as

$$\eta_m = \frac{\text{Power at the shaft of the turbine}}{\text{Power delivered by water to the runner}} = \frac{\text{S.P.}}{\text{R.P.}} \quad \text{...(18.4)}$$

(*c*) **Volumetric Efficiency** (η_v). The volume of the water striking the runner of a turbine is slightly less than the volume of the water supplied to the turbine. Some of the volume of the water is discharged to the tail race without striking the runner of the turbine. Thus the ratio of the volume of the water actually striking the runner to the volume of water supplied to the turbine is defined as volumetric efficiency. It is written as

$$\eta_v = \frac{\text{Volume of water actually striking the runner}}{\text{Volume of water supplied to the turbine}} \quad \text{...(18.5)}$$

(*d*) **Overall Efficiency** (η_o). It is defined as the ratio of power available at the shaft of the turbine to the power supplied by the water at the inlet of the turbine. It is written as :

$$\eta_o = \frac{\text{Volume available at the shaft of the turbine}}{\text{Power supplied at the inlet of the turbine}} = \frac{\text{Shaft power}}{\text{Water power}} = \frac{\text{S.P.}}{\text{W.P.}}$$

$$= \frac{\text{S.P.}}{\text{W.P.}} \times \frac{\text{R.P.}}{\text{R.P.}} \qquad \text{(where R.P. = Power delivered to runner)}$$

$$= \frac{\text{S.P.}}{\text{R.P.}} \times \frac{\text{R.P.}}{\text{W.P.}}$$

$$= \eta_m \times \eta_h \qquad \left(\because \begin{array}{l} \text{From equation (18.4),} \quad \dfrac{\text{S.P.}}{\text{R.P.}} = \eta_m \\ \text{and from equation (18.2),} \quad \dfrac{\text{R.P.}}{\text{W.P.}} = \eta_h \end{array} \right) \quad \text{...(18.6)}$$

If shaft power (S.P.) is taken in kW then water power should also be taken in kW. Shaft power is commonly represented by P. But from equation (18.3 A),

$$\text{Water power in kW} = \frac{\rho \times g \times Q \times H}{1000} \qquad \text{where } \rho = 1000 \text{ kg/m}^3$$

$$\therefore \qquad \eta_o = \frac{\text{Shaft power in kW}}{\text{Water power in kW}} = \frac{P}{\left(\dfrac{\rho \times g \times Q \times H}{1000}\right)} \qquad \text{...(18.6 A)}$$

where P = Shaft power.

18.5. CLASSIFICATION OF HYDRAULIC TURBINES

The hydraulic turbines are classified according to the type of energy available at the inlet of the turbine, direction of flow through the vanes, head at the inlet of the turbine and specific speed of the turbines. Thus the followings are the important classification of the turbines :

1. According to the type of energy at inlet :
   (*a*) Impulse turbine, and (*b*) Reaction turbine.
2. According to the direction of flow through runner :
   (*a*) Tangential flow turbine, (*b*) Radial flow turbine,
   (*c*) Axial flow turbine, and (*d*) Mixed flow turbine.
3. According to the head at the inlet of turbine :
   (*a*) High head turbine, (*b*) Medium head turbine, and
   (*c*) Low head turbine.
4. According to the specific speed of the turbine :
   (*a*) Low specific speed turbine, (*b*) Medium specific speed turbine, and
   (*c*) High specific speed turbine.

If at the inlet of the turbine, the energy available is only kinetic energy, the turbine is known as **impulse turbine.** As the water flows over the vanes, the pressure is atmospheric from inlet to outlet of the turbine. If at the inlet of the turbine, the water possesses kinetic energy as well as pressure energy, the turbine is known as **reaction turbine.** As the water flows through the runner, the water is under pressure and the pressure energy goes on changing into kinetic energy. The runner is completely enclosed in an air-tight casing and the runner and casing is completely full of water.

If the water flows along the tangent of the runner, the turbine is known at **tangential flow turbine.** If the water flows in the radial direction through the runner, the turbine is called **radial flow turbine.** If the water flows from outwards to inwards, radially, the turbine is known as **inward** radial flow turbine, on the other hand, if water flows radially from inwards to outwards, the turbine is known as **outward** radial flow turbine. If the water flow through the runner along the direction parallel to the axis of rotation of the runner, the turbine is called **axial flow** turbine. If the water flows through the runner in the radial direction but leaves in the direction parallel to the axis of rotation of the runner, the turbine is called **mixed flow** turbine.

18.6. PELTON WHEEL (OR TURBINE)

The Pelton wheel or Pelton turbine is a tangential flow impulse turbine. The water strikes the bucket along the tangent of the runner. The energy available at the inlet of the turbine is only kinetic energy. The pressure at the inlet and outlet of the turbine is atmosphere. This turbine is used for high heads and is named after L.A. Pelton, an American Engineer.

Fig. 18.1 shows the lay-out of a hydro-electric power plant in which the turbine is Pelton wheel. The water from the reservoir flows through the penstocks at the outlet of which a nozzle is fitted. The nozzle increases the kinetic energy of the water flowing through the penstock. At the outlet of the nozzle, the water comes out in the form of a jet and strikes the buckets (vanes) of the runner. The main parts of the Pelton turbine are :

1. Nozzle and flow regulating arrangement (spear), 2. Runner and buckets,
3. Casing, and 4. Breaking jet.

1. Nozzle and Flow Regulating Arrangement. The amount of water striking the buckets (vanes) of the runner is controlled by providing a spear in the nozzle as shown in Fig. 18.2. The spear is a conical needle which is operated either by a hand wheel or automatically in an axial direction depending upon the size of the unit. When the spear is pushed forward into the nozzle the amount of water striking the runner is reduced. On the other hand, if the spear is pushed back, the amount of water striking the runner increases.

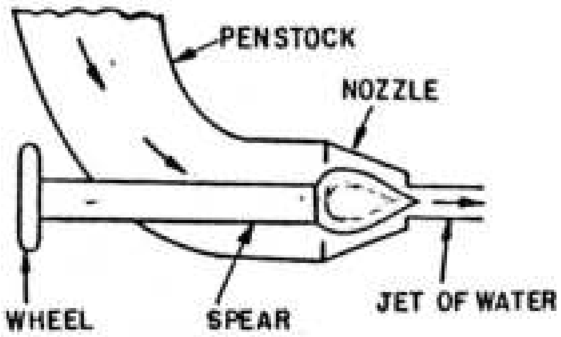


Fig. 18.2. Nozzle with a spear to regulate flow.

2. Runner with Buckets. Fig. 18.3 shows the runner of a Pelton wheel. It consists of a circular disc on the periphery of which a number of buckets evenly spaced are fixed. The shape of the buckets is of a double hemispherical cup or bowl. Each bucket is divided into two symmetrical parts by a dividing wall which is known as splitter.

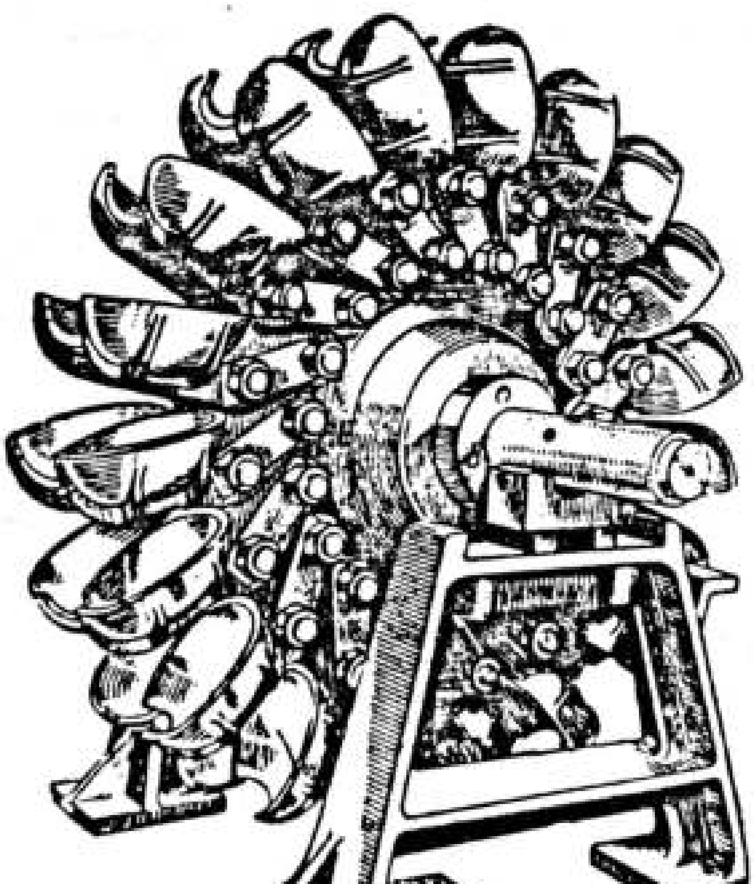


Fig. 18.3. Runner of a Pelton Wheel.

The jet of water strikes on the splitter. The splitter divides the jet into two equal parts and the jet comes out at the outer edge of the bucket. The buckets are shaped in such a way that the jet gets deflected through 160° or 170°. The buckets are made of cast iron, cast steel bronze or stainless steel depending upon the head at the inlet of the turbine.

3. Casing. Fig. 18.4 shows a Pelton turbine with a casing. The function of the casing is to prevent the splashing of the water and to discharge water to tail race. It also acts as a safeguard against accidents. It is made of cast iron or fabricated steel plates. The casing of the Pelton wheel does not perform any hydraulic function.

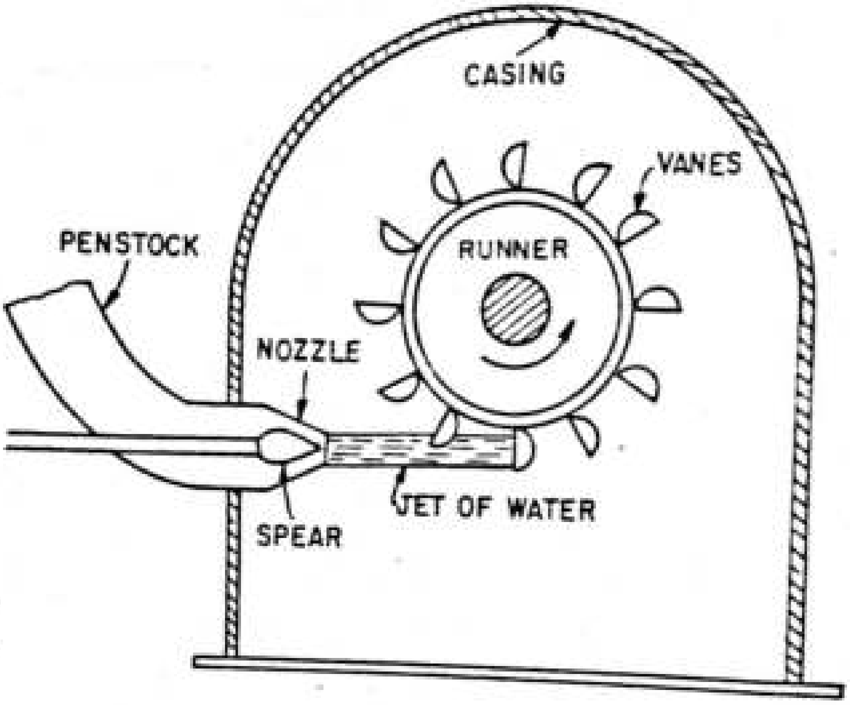


Fig. 18.4. Pelton Turbine.

4. Breaking Jet. When the nozzle is completely closed by moving the spear in the forward direction the amount of water striking the runner reduces to zero. But the runner due to inertia goes on revolving for a long time. To stop the runner in a short time, a small nozzle is provided which directs the jet of water on the back of the vanes. This jet of water is called breaking jet.

18.6.1. Velocity Triangles and Work done for Pelton Wheel. Fig. 18.5 shows the shape of the vanes or buckets of the Pelton wheel. The jet of water from the nozzle strikes the bucket at the splitter, which splits up the jet into two parts. These parts of the jet, glides over the inner surfaces and comes out at the outer edge. Fig. 18.5 (b) shows the section of the bucket at z-z. The splitter is the inlet tip and outer edge of the bucket is the outlet tip of the bucket. The inlet velocity triangle is drawn at the splitter and outlet velocity triangle is drawn at the outer edge of the bucket, by the same method as explained in Chapter 17.

Let H = Net head acting on the Pelton wheel

$= H_g - h_f$

where H_g = Gross head and $h_f = \dfrac{4fLV^2}{D^* \times 2g}$

where D^* = Dia. of Penstock, N = Speed of the wheel in r.p.m.

D = Diameter of the wheel, d = Diameter of the jet.

Then V_1 = Velocity of jet at inlet $= \sqrt{2gH}$

$$u = u_1 = u_2 = \frac{\pi DN}{60}.$$

The velocity triangle at inlet will be a straight line where

$$V_{r_1} = V_1 - u_1 = V_1 - u$$

$$V_{w_1} = V_1$$

$$\alpha = 0 \text{ and } \theta = 0$$

From the velocity triangle at outlet, we have

$$V_{r_2} = V_{r_1} \text{ and } V_{w_2} = V_{r_2} \cos \phi - u_2.$$

The force exerted by the jet of water in the direction of motion is given by equation (17.19) as

$$F_x = \rho a V_1 [V_{w_1} + V_{w_2}]. \quad \text{...(18.8)}$$

As the angle β is an acute angle, +ve sign should be taken. Also this is the case of series of Vanes, the mass of water striking is $\rho a V_1$ and not $\rho a V_{r_1}$. In equation (18.8), 'a' is the area of the jet which is given as

$$a = \text{Area of jet} = \frac{\pi}{4} d^2.$$

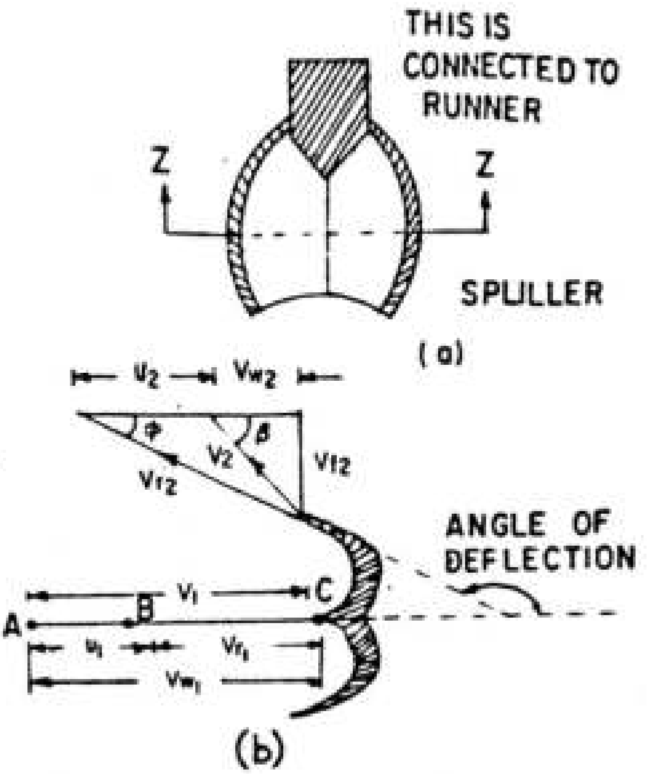


Fig. 18.5 Shape of Bucket

Now work done by the jet on the runner per second

$$= F_x \times u = \rho a V_1 [V_{w_1} + V_{w_2}] \times u \text{ Nm/s} \quad \text{...(18.9)}$$

Power given to the runner by the jet

$$= \frac{\rho a V_1 [V_{w_1} + V_{w_2}] \times u}{1000} \text{ kW} \quad \text{...(18.10)}$$

Work done/s per unit weight of water striking/s

$$= \frac{\rho a V_1 [V_{w_1} + V_{w_2}] \times u}{\text{Weight of water striking/s}}$$

$$= \frac{\rho a V_1 [V_{w_1} + V_{w_2}] \times u}{\rho a V_1 \times g} = \frac{1}{g} [V_{w_1} + V_{w_2}] \times u \quad \text{...(18.11)}$$

The energy supplied to the jet at inlet is in the form of kinetic energy and is equal to $\frac{1}{2} mV^2$

$\therefore$ K.E. of jet per second $= \frac{1}{2} (\rho a V_1) \times V_1^2$.

$\therefore$ **Hydraulic efficiency,** $\eta_h = \dfrac{\text{Work done per second}}{\text{K.E. of jet per second}}$

$$= \frac{\rho a V_1 [V_{w_1} + V_{w_2}] \times u}{\frac{1}{2} (\rho a V_1) \times V_1^2} = \frac{2 [V_{w_1} + V_{w_2}] \times u}{V_1^2} \quad \text{...(18.12)}$$

Now $V_{w_1} = V_1$, $V_{r_1} = V_1 - u_1 = (V_1 - u)$

$\therefore$ $V_{r_2} = (V_1 - u)$

and $V_{w_2} = V_{r_2} \cos \phi - u_2 = V_{r_2} \cos \phi - u = (V_1 - u) \cos \phi - u$

Substituting the values of V_{w_1} and V_{w_2} in equation (18.12),

$$\eta_h = \frac{2[V_1 + (V_1 - u)\cos\phi - u] \times u}{V_1^2}$$

$$= \frac{2[V_1 - u + (V_1 - u)\cos\phi] \times u}{V_1^2} = \frac{2(V_1 - u)[1 + \cos\phi]\,u}{V_1^2}. \qquad \text{...(18.13)}$$

The efficiency will be maximum for a given value of V_1 when

$$\frac{d}{du}(\eta_h) = 0 \qquad \text{or} \qquad \frac{d}{du}\left[\frac{2u(V_1 - u)(1 + \cos\phi)}{V_1^2}\right] = 0$$

or $$\frac{(1 + \cos\phi)}{V_1^2}\frac{d}{du}(2uV_1 - 2u^2) = 0 \qquad \text{or} \qquad \frac{d}{du}[2uV_1 - 2u^2] = 0 \qquad \left(\because \frac{1 + \cos\phi}{V_1^2} \neq 0\right)$$

or $$2V_1 - 4u = 0 \qquad \text{or} \qquad \boxed{u = \frac{V_1}{2}} \qquad \text{...(18.14)}$$

Equation (18.14) states that hydraulic efficiency of a Pelton wheel will be maximum when the velocity of the wheel is half the velocity of the jet of water at inlet. The expression for maximum efficiency will be obtained by substituting the value of $u = \frac{V_1}{2}$ in equation (18.13).

$$\therefore \qquad \text{Max. } \eta_h = \frac{2\left(V_1 - \frac{V_1}{2}\right)(1 + \cos\phi) \times \frac{V_1}{2}}{V_1^2}$$

$$= \frac{2 \times \frac{V_1}{2}(1 + \cos\phi)\frac{V_1}{2}}{V_1^2} = \frac{(1 + \cos\phi)}{2}. \qquad \text{...(18.15)}$$

18.6.2. Points to be Remembered for Pelton Wheel

(*i*) The velocity of the jet at inlet is given by $V_1 = C_v\sqrt{2gH}$

where C_v = Co-efficiency of velocity = 0.98 or 0.99

H = Net head on turbine.

(*ii*) The velocity of wheel (u) is given by $u = \phi\sqrt{2gH}$

where ϕ = Speed ratio. The value of speed ratio varies from 0.43 to 0.48.

(*iii*) The angle of deflection of the jet through buckets is taken at 165° if no angle of deflection is given.

(*iv*) The mean diameter or the pitch diameter D of the Pelton wheel is given by

$$u = \frac{\pi DN}{60} \text{ or } D = \frac{60\,u}{\pi N}.$$

(*v*) **Jet Ratio.** It is defined as the ratio of the pitch diameter (D) of the Pelton wheel to the diameter of the jet (d). It is denoted by 'm' and is given as

$$m = \frac{D}{d} \ (= 12 \text{ for most cases}) \qquad \text{...(18.16)}$$

(*vi*) Number of buckets on a runner is given by

$$Z = 15 + \frac{D}{2d} = 15 + 0.5\,m \qquad \text{...(18.17)}$$

where m = Jet ratio.

(*vii*) **Number of Jets.** It is obtained by dividing the total rate of flow through the turbine by the rate of flow of water through a single jet.

Problem 18.1. *A Pelton wheel has a mean bucket speed of 10 metres per second with a jet of water flowing at the rate of 700 litres/s under a head of 30 metres. The buckets deflect the jet through an angle of 160°. Calculate the power given by water to the runner and the hydraulic efficiency of the turbine. Assume co-efficient of velocity as 0.98.* (AMIE, Summer, 1980)

Sol. Given :

Speed of bucket, $u = u_1 = u_2 = 10$ m/s

Discharge, $Q = 700$ litres/s $= 0.7$ m^3/s, Head of water, $H = 30$ m

Angle of deflection $= 160°$

$\therefore$ Angle, $\phi = 180 - 160 = 20°$

Co-efficient of velocity, $C_v = 0.98$.

The velocity of jet, $V_1 = C_v\sqrt{2gH} = 0.98\sqrt{2 \times 9.81 \times 30} = 23.77$ m/s

$\therefore$ $V_{r_1} = V_1 - u_1 = 23.77 - 10 = 13.77$ m/s

$V_{w_1} = V_1 = 23.77$ m/s

From outlet velocity triangle,

$V_{r_2} = V_{r_1} = 13.77$ m/s

$V_{w_2} = V_{r_2} \cos\phi - u_2$

$= 13.77 \cos 20 - 10.0 = 2.94$ m/s

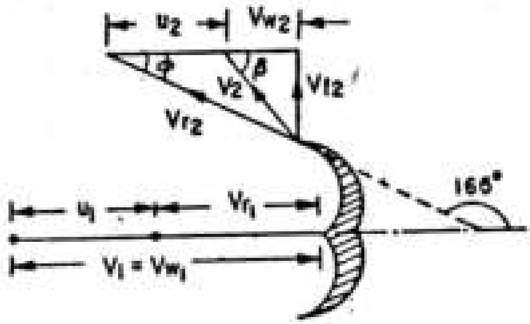


Fig. 18.6

Work done by the jet per second on the runner is given by equation (18.9) as

$= \rho a V_1 [V_{w_1} + V_{w_2}] \times u$

$= 1000 \times 0.7 \times [23.77 + 2.94] \times 10$

$(\because \; aV_1 = Q = 0.7 \text{ m}^3\text{/s})$

$= 186970$ Nm/s

$\therefore$ Power given to turbine $= \dfrac{186970}{1000} = $ **186.97 kW. Ans.**

The hydraulic efficiency of the turbine is given by equation (18.12) as

$$\eta_h = \frac{2\,[V_{w_1} + V_{w_2}] \times u}{V_1^2} = \frac{2\,[23.77 + 2.94] \times 10}{23.77 \times 23.77} = 0.9454 \text{ or } \mathbf{94.54\%}. \textbf{ Ans.}$$

Problem 18.2. *A Pelton wheel is to be designed for the following specifications : Shaft power = 11,772 kW ; Head = 380 metres ; Speed = 750 r.p.m. ; Overall efficiency = 86% ; Jet diameter is not to exceed one-sixth of the wheel diameter. Determine :*

(i) The wheel diameter, *(ii) The number of jets required, and*

(iii) Diameter of the jet.

Take $K_{v_1} = 0.985$ and $K_{u_1} = 0.45$. (AMIE, Winter, 1980)

Sol. Given :

Shaft power, S.P. $= 11,772$ kW

Head, $H = 380$ m

Speed, $N = 750$ r.m.p.

Overall efficiency, $\eta_0 = 86\%$ or 0.86

Ratio of jet dia. to wheel dia. $= \frac{d}{D} = \frac{1}{6}$

Co-efficient of velocity, $K_{v_1} = C_v = 0.985$

Speed ratio, $K_{u_1} = 0.45$

Velocity of jet, $V_1 = C_v \sqrt{2gH} = 0.985 \sqrt{2 \times 9.81 \times 380} = 85.05$ m/s

The velocity of wheel,
$$u = u_1 = u_2$$
$$= \text{Speed ratio} \times \sqrt{2gH} = 0.45 \times \sqrt{2 \times 9.81 \times 380} = 38.85 \text{ m/s}$$

But
$$u = \frac{\pi DN}{60}$$

$\therefore$
$$38.85 = \frac{\pi DN}{60}$$

or
$$D = \frac{60 \times 38.85}{\pi \times N} = \frac{60 \times 38.85}{\pi \times 750} = \mathbf{0.989\ m.} \quad \textbf{Ans.}$$

But
$$\frac{d}{D} = \frac{1}{6}$$

$\therefore$ Dia. of jet,
$$d = \frac{1}{6} \times D = \frac{0.989}{6} = \mathbf{0.165\ m.} \quad \textbf{Ans.}$$

Discharge of one jet, q = Area of jet × Velocity of jet
$$= \frac{\pi}{4} d^2 \times V_1 = \frac{\pi}{4} (.165)^2 \times 85.05 \text{ m}^3\text{/s} = 1.818 \text{ m}^3\text{/s} \quad \text{...(i)}$$

Now
$$\eta_0 = \frac{\text{S.P.}}{\text{W.P.}} = \frac{11772}{\frac{\rho g \times Q \times H}{1000}}$$

$$0.86 = \frac{11772 \times 1000}{1000 \times 9.81 \times Q \times 380}$$ where Q = Total discharge

$\therefore$ Total discharge,
$$Q = \frac{11772 \times 1000}{1000 \times 9.81 \times 380 \times 0.86} = 3.672 \text{ m}^3\text{/s}$$

$\therefore$ Number of jets
$$= \frac{\text{Total discharge}}{\text{Discharge of one jet}} = \frac{Q}{q} = \frac{3.672}{1.818} = \mathbf{2\ jets.} \quad \textbf{Ans.}$$

Problem 18.3. *The penstock supplies water from a reservoir to the Pelton wheel with a gross head of 500 m. One-third of the gross head is lost in friction in the penstock. The rate of flow of water through the nozzle fitted at the end of the penstock is 2.0 m³/s. The angle of deflection of the jet is 165°. Determine the power given by the water to the runner and also hydraulic efficiency of the Pelton wheel. Take speed ratio = 0.45 and C_v = 1.0.* (AMIE, Fluid Power Engg., 1988 ; Osmania University, 1992)

Sol. Given :

Gross head, $H_g = 500$ m

Head lost in friction, $h_f = \frac{H_g}{3} = \frac{500}{3} = 166.7$ m

$\therefore$ Net head, $H = H_g - h_f = 500 - 166.7 = 333.30$ m

Discharge, $Q = 2.0$ m³/s

Angle of deflection $= 165°$

$\therefore$ Angle, $\phi = 180 - 165 = 15°$

Speed ratio $= 0.45$

Co-efficient of velocity, $C_v = 1.0$

Velocity of jet, $V_1 = C_v\sqrt{2gH} = 1.0 \times \sqrt{2 \times 9.81 \times 333.3} = 80.86$ m/s

Velocity of wheel, $u = \text{Speed ratio} \times \sqrt{2gH}$

or $u = u_1 = u_2 = 0.45 \times \sqrt{2 \times 9.81 \times 333.3} = 36.387$ m/s

$\therefore$ $V_{r_1} = V_1 - u_1 = 80.86 - 36.387 = 44.473$ m/s

Also $V_{w_1} = V_1 = 80.86$ m/s

From outlet velocity triangle, we have

$V_{r_2} = V_{r_1} = 44.473$

$V_{r_2} \cos \phi = u_2 + V_{w_2}$

or $44.473 \cos 15 = 36.387 + V_{w_2}$

or $V_{w_2} = 44.473 \cos 15 - 36.387 = 6.57$ m/s.

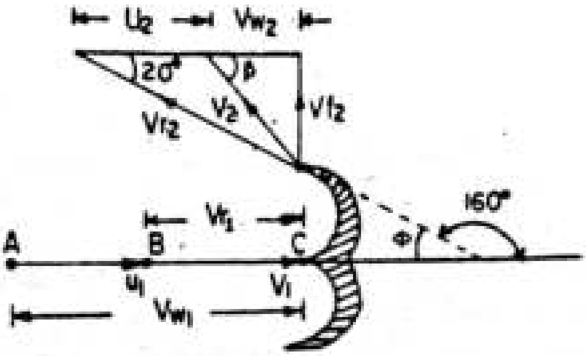


Fig. 18.7

Work done by the jet on the runner per second is given by equation (18.9) as

$= \rho a V_1[V_{w_1} + V_{w_2}] \times u = \rho Q[V_{w_1} + V_{w_2}] \times u$

$(\because \; aV_1 = Q)$

$= 1000 \times 2.0 \times [80.86 + 6.57] \times 36.387 = 6362630$ Nm/s

$\therefore$ Power given by the water to the runner in kW

$$= \frac{\text{Work done per second}}{1000} = \frac{6362630}{1000} = \mathbf{6362.63\ kW.} \quad \textbf{Ans.}$$

Hydraulic efficiency of the turbine is given by equation (18.12) as

$$\eta_h = \frac{2[V_{w_1} + V_{w_2}] \times u}{V_1^2} = \frac{2\,[80.86 + 6.57] \times 36.387}{80.86 \times 80.86} = 0.9731 \text{ or } \mathbf{97.31\%}. \quad \textbf{Ans.}$$

Problem 18.4. *A Pelton wheel is having a mean bucket diameter of 1 m and is running at 1000 r.p.m. The net head on the Pelton wheel is 700 m. If the side clearance angle is 15° and discharge through nozzle is 0.1 m³/s, find :*

(i) Power available at the nozzle, and (ω, ρ,) *(ii) Hydraulic efficiency of the turbine.*

Sol. Given :

Diameter of wheel, $D = 1.0$ m

Speed of wheel, $N = 1000$ r.p.m.

$\therefore$ Tangential velocity of the wheel, $u = \dfrac{\pi DN}{60} = \dfrac{\pi \times 1.0 \times 1000}{60} = 52.36$ m/s

Net head on turbine, $H = 700$ m

Side clearance angle, $\phi = 15°$

Discharge, $Q = 0.1$ m³/s

Velocity of jet at inlet, $V_1 = C_v\sqrt{2gH} = 1 \times \sqrt{2 \times 9.81 \times 700}$

$(\because$ Value of C_v is not given. Take it $= 1.0)$

or $V_1 = 117.19$ m/s.

(*i*) Power available at the nozzle is given by equation (18.3) as

$$\text{W.P.} = \frac{W \times H}{1000} = \frac{\rho \times g \times Q \times H}{1000} = \frac{1000 \times 9.81 \times 0.1 \times 700}{1000} = \mathbf{686.7\ kW.} \quad \textbf{Ans.}$$

(*ii*) Hydraulic efficiency is given by equation (18.13) as

$$\eta_h = \frac{2(V_1 - u)(1 + \cos\phi)\, u}{V_1^2}$$

$$= \frac{2(117.19 - 52.36)\,(1 + \cos 15) \times 52.36}{117.19 \times 117.19}$$

$$= \frac{2 \times 64.83 \times 1.966 \times 52.36}{117.19 \times 117.19} = 0.9718 = \mathbf{97.18\%}. \quad \textbf{Ans.}$$

Problem 18.5 *A Pelton wheel is working under a gross head of 400 m. The water is supplied through penstock of diameter 1 m and length 4 km from reservoir to the Pelton wheel. The co-efficient of friction for the penstock is given as .008. The jet of water of diameter 150 mm strikes the buckets of the wheel and gets deflected through an angle of 165°. The relative velocity of water at outlet is reduced by 15% due to friction between inside surface of the bucket and water. If the velocity of the buckets is 0.45 times the jet velocity at inlet and mechanical efficiency as 85%, determine :*

(*i*) *Power given to the runner,* (*ii*) *Shaft power,*

(*iii*) *Hydraulic efficiency and overall efficiency.*

Sol. Given :

Gross head, $H_g = 400$ m

Diameter of penstock, $D = 1.0$ m

Length of penstock, $L = 4$ km $= 4 \times 1000 = 4000$ m

Co-efficient of friction, $f = .008$

Diameter of jet, $d = 150$ mm $= 0.15$ m

Angle of deflection $= 165°$

∴ Angle, $\phi = 180 - 165 = 15°$

Relative velocity at outlet, $V_{r_2} = 0.85\, V_{r_1}$

Velocity of bucket, $u = 0.45 \times$ Jet velocity

Mechanical efficiency, $\eta_m = 85\% = 0.85$

Let V^* = Velocity of water in penstock, and

V_1 = Velocity of jet of water.

Using continuity equation, we have

Area of penstock $\times V^*$ = Area of jet $\times V_1$

or

$$\frac{\pi}{4} D^2 \times V^* = \frac{\pi}{4} d^2 \times V_1$$

∴

$$V^* = \frac{d^2}{D^2} \times V_1 = \frac{0.15^2}{1.0^2} \times V_1 = .0225\, V_1 \qquad \ldots(i)$$

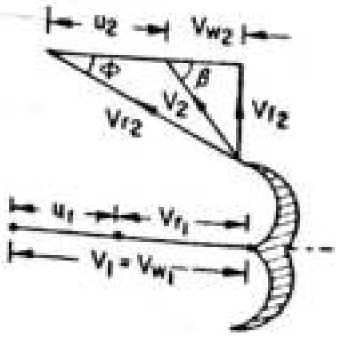


Fig. 18.8

Applying Bernoulli's equation to the free surface of water in the reservoir and outlet of the nozzle, we get

$$H_g = \text{Head lost due to friction} + \frac{V_1^2}{2g}$$

or
$$400 = \frac{4fLV^{*2}}{D \times 2g} + \frac{V_1^2}{2g} = \frac{4 \times .008 \times 4000 \times V^{*2}}{1.0 \times 2 \times 9.81} + \frac{V_1^2}{2g}$$

Substituting the value of V^* from equation (*i*), we get

$$400 = \frac{4 \times .008 \times 4000}{2 \times 9.81} \times (0.0225\, V_1)^2 + \frac{V_1^2}{2g}$$

$$= .0033\, V_1^2 + .051\, V_1^2 \quad \text{or} \quad 400 = .0543\, V_1^2$$

$\therefore$
$$V_1 = \sqrt{\frac{400}{.0543}} = 85.83 \text{ m/s.}$$

Now velocity of bucket, $u_1 = 0.45\, V_1 = 0.45 \times 85.83 = 38.62$ m/s

From inlet velocity triangle, $V_{r_1} = V_1 - u_1 = 85.83 - 38.62 = 47.21$ m/s

$$V_{w_1} = V_1 = 85.83 \text{ m/s}$$

From outlet velocity triangle, $V_{r_2} = 0.85 \times V_{r_1} = 0.85 \times 47.21 = 40.13$ m/s

$$V_{w_2} = V_{r_2} \cos\phi - u_2 = 40.13 \cos 15 - 38.62$$

$$= 0.143 \text{ m/s} \qquad (\because\ u = u_1 = u_2 = 38.62)$$

Discharge through nozzle is given as

$$Q = \text{Area of jet} \times \text{Velocity of jet} = a \times V_1$$

$$= \frac{\pi}{4} d^2 \times V_1 = \frac{\pi}{4} (.15)^2 \times 85.83 = 1.516 \text{ m}^3\text{/s}$$

Work done on the wheel per second is given by equation (18.9) as

$$= \rho a V_1 [V_{w_1} + V_{w_2}] \times u = \rho Q [V_{w_1} + V_{w_2}] \times u$$

$$= 1000 \times 1.516 [85.83 + .143] \times 38.62 = 5033540 \text{ Nm/s}$$

(*i*) Power given to the runner in kW

$$= \frac{\text{Work done per second}}{1000} = \frac{5033540}{1000} = \mathbf{5033.54\ kW.} \quad \textbf{Ans.}$$

(*ii*) Using equation (18.4) for mechanical efficiency,

$$\eta_m = \frac{\text{Power at the shaft}}{\text{Power given to the runner}} = \frac{\text{S.P.}}{5033.54}$$

$\therefore$
$$\text{S.P.} = \eta_m \times 5033.54 = 0.85 \times 5033.54 = \mathbf{4278.5\ kW.} \quad \textbf{Ans.}$$

(*iii*) Hydraulic efficiency is given by equation (18.12) as

$$\eta_h = \frac{2[V_{w_1} + V_{w_2}] \times u}{V_1^2}$$

$$= \frac{2\,[85.83 + .143] \times 38.62}{85.83 \times 85.83} = 0.9014 = \mathbf{90.14\%.} \quad \textbf{Ans.}$$

Overall efficiency is given by equation (18.6) as

$$\eta_o = \eta_m \times \eta_h = 0.85 \times .9014 = 0.7662 \quad \text{or} \quad \mathbf{76.62\%.} \quad \textbf{Ans.}$$

Problem 18.8. *Two jets strike the buckets of a Pelton wheel, which is having shaft power as 15450 kW. The diameter of each jet is given as 200 mm. If the net head on the turbine is 400 m, find the overall efficiency of the turbine. Take $C_v = 1.0$.*

Sol. Given :

Number of jets $= 2$

Shaft power, $\text{S.P.} = 15450 \text{ kW}$

Diameter of each jet, $d = 200 \text{ mm} = 0.20 \text{ m}$

$\therefore$ Area of each jet, $a = \frac{\pi}{4} d^2 = \frac{\pi}{4} (.2)^2 = 0.031416 \text{ m}^2$

Net head, $H = 400 \text{ m}$

Co-efficient of Velocity, $C_v = 1.0$

Velocity of each jet, $V_1 = C_v \sqrt{2gH} = 1.0 \times \sqrt{2 \times 9.81 \times 400} = 88.58 \text{ m/s}$

Discharge of each jet $= a \times V_1 = .031416 \times 88.58 = 2.78 \text{ m}^3\text{/s}$

$\therefore$ Total discharge, $Q = 2 \times 2.78 = 5.56 \text{ m}^3\text{/s}$

Power at the inlet of turbine,

$$\text{W.P.} = \frac{\rho \times g \times Q \times H}{1000} \text{ kW}$$

$$= \frac{1000 \times 9.81 \times 5.56 \times 400}{1000} = 21817.44 \text{ kW}$$

$\therefore$ Overall efficiency is given as

$$\eta_o = \frac{\text{S.P.}}{\text{W.P.}} = \frac{15450}{21817.44} = 0.708 = \mathbf{70.8\%}. \quad \textbf{Ans.}$$

Problem 18.12. *Determine the power given by the jet of water to the runner of a Pelton wheel which is having tangential velocity as 20 m/s. The net head on the turbine is 50 m and discharge through the jet water is 0.03 m³/s. The side clearance angle is 15° and take C_v = 0.975.*

Sol. Given :

Tangential velocity of wheel, $u = u_1 = u_2 = 20$ m/s

Net head, $H = 50$ m

Discharge, $Q = 0.03 \text{ m}^3/\text{s}$

Side clearance angle, $\phi = 15°$

Co-efficient of velocity, $C_v = 0.975$

Velocity of the jet, $V_1 = C_v \times \sqrt{2gH}$

$= 0.975 \times \sqrt{2 \times 9.81 \times 50}$

$= 30.54$ m/s.

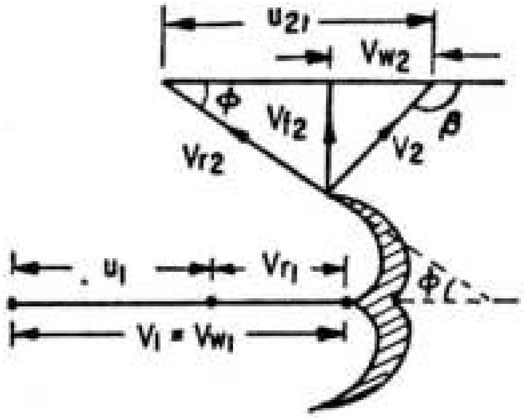


Fig. 18.9

From inlet triangle, $V_{w_1} = V_1 = 30.54$ m/s

$V_{r_1} = V_{w_1} - u_1 = 30.54 - 20.0 = 10.54$ m/s.

From outlet velocity triangle, we have

$V_{r_2} = V_{r_1} = 10.54$ m/s

$V_{r_2} \cos \phi = 10.54 \cos 15 = 10.18$ m/s.

As $V_{r_2} \cos \phi$ less than u_2, the velocity triangle at outlet will be as shown in Fig. 18.9.

$\therefore$ $V_{w_2} = u_2 - V_{r_2} \cos \phi = 20 - 10.18 = 9.82$ m/s.

Also as β is an obtuse angle, the work done per second on the runner,

$= \rho a V_1 [V_{w_1} - V_{w_2}] \times u = \rho Q\, [V_{w_1} - V_{w_2}] \times u$

$= 1000 \times .03 \times [30.54 - 9.82] \times 20 = 12432$ Nm/s

$\therefore$ Power given to the runner in kW $= \dfrac{\text{Work done per second}}{1000} = \dfrac{12432}{1000} = \mathbf{12.432\ kW.}$ **Ans.**

18.7. RADIAL FLOW REACTION TURBINES

Radial flow turbines are those tubines in which the water flows in the radial direction. The water may flow radially from outwards to inwards (*i.e.*, towards the axis of rotation) or from inwards to outwards. If the water flows from outwards to inwards through the runner, the turbine is known as inward radial flow turbine. And if the water flows from inwards to outwards, the turbine is known as outward radial flow turbine.

Reaction turbine means that the water at the inlet of the turbine possesses kinetic energy as well as pressure energy. As the water flows through the runner, a part of pressure energy goes on changing into kinetic energy. Thus the water through the runner is under pressure. The runner is completely enclosed in an air-tight casing and casing and the runner is always full of water.

18.7.1. Main Parts of a Radial Flow Reaction Turbine. The main parts of a radial flow reaction turbine are :

1. Casing,
2. Guide mechanism,
3. Runner, and
4. Draft tube.

1. Casing. As mentioned above that in case of reaction turbine, casing and runner are always full of water. The water from the penstocks enters the casing which is of spiral shape in which area of cross-section of the casing goes on decreasing gradually. The casing completely surrounds the runner of the turbine. The casing as shown in Fig. 18.10 is made of spiral shape, so that the water may enter the runner at constant velocity through out the circumference of the runner. The casing is made of concrete, cast steel or plate steel.

2. Guide Mechanism. It consists of a stationary circular wheel all round the runner of the turbine. The stationary guide vanes are fixed on the guide mechanism. The guide vanes allow the water to strike the vanes fixed on the runner without shock at inlet. Also by a suitable arrangement, the width between two adjacent vanes of a guide mechanism can be altered so that the amount of water striking the runner can be varied.

3. Runner. It is a circular wheel on which a series of radial curved vanes are fixed. The surface of the vanes are made very smooth. The radial curved vanes are so shaped that the water enters and leaves the runner without shock. The runners are made of cast steel, cast iron or stainless steel. They are keyed to the shaft.

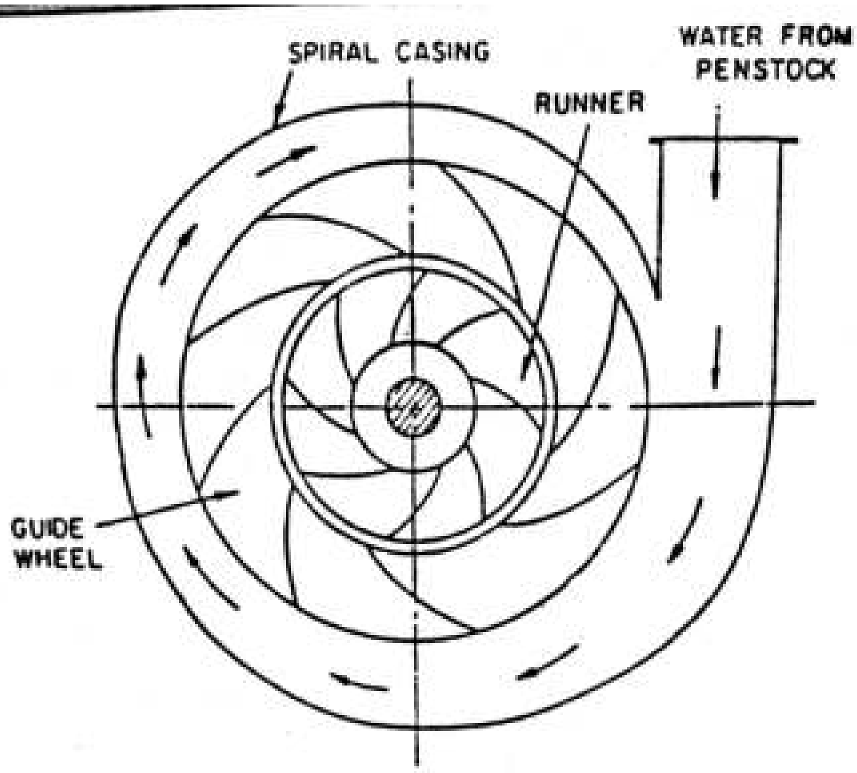


Fig. 18.10. Main parts of a Radial Reaction Turbine.

4. Draft-tube. The pressure at the exit of the runner of a reaction turbine is generally less than atmospheric pressure. The water at exit cannot be directly discharged to the tail race. A tube or pipe of gradually increasing area is used for discharging water from the exit of the turbine to the tail race. This tube of increasing area is called draft tube.

18.7.2. Inward Radial Flow Turbine. Fig. 18.11 shows inward radial flow turbine, in which case the water from the casing enters the stationary guiding wheel. The guiding wheel consists of guide vanes which

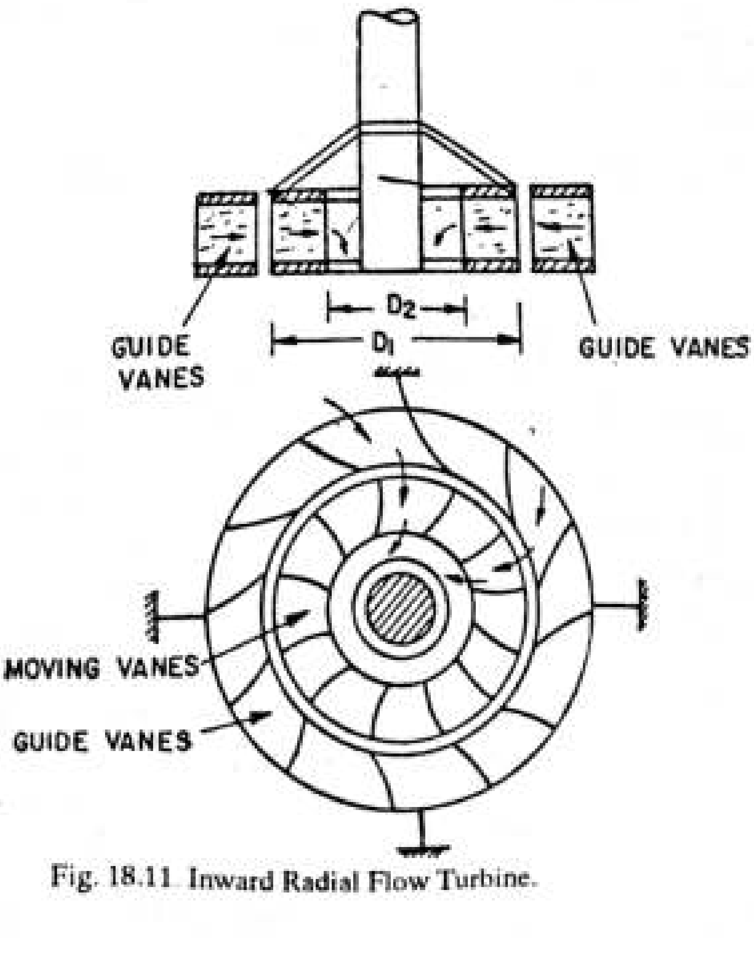


Fig. 18.11. Inward Radial Flow Turbine.

direct the water to enter the runner which consists of moving vanes. The water flows over the moving vanes in the inward radial direction and is discharged at the inner diameter of the runner. The outer diameter of the runner is the inlet and the inner diameter is the outlet.

Velocity Triangles and Work done by Water on Runner. In Chapter 17 (Art. 17.4.6), we have discussed in detail the force exerted by the water on the radial curved vanes fixed on a wheel. From the force exerted on the vanes, the work done by water, the horse power given by the water to the vanes and efficiency of the vanes can be obtained. Also we have drawn velocity triangles at inlet and outlet of the moving radial vanes in Fig. 17.23. From the velocity triangles, the work done by the water on the runners, horse power and efficiency of the turbine can be obtained.

The work done per second on the runner by water is given by equation (17.26) as

$$= \rho a V_1 [V_{w_1} u_1 \pm V_{w_2} u_2]$$

$$= \rho Q [V_{w_1} u_1 \pm V_{w_2} u_2] \qquad (\because \; aV_1 = Q) \qquad ...(18.18)$$

where V_{w_1} = Velocity of whirl at inlet,

V_{w_2} = Velocity of whirl at outlet,

u_1 = Tangential velocity of wheel at inlet

$= \dfrac{\pi D_1 \times N}{60}$, where D_1 = Outer dia. of runner,

u_2 = Tangential velocity of wheel at outlet

$= \dfrac{\pi D_2 \times N}{60}$ where D_2 = Inner dia. of runner, N = Speed of the turbined in r.p.m.

The work done per second per unit weight of water per second

$$= \frac{\text{Work done per second}}{\text{Weight of water striking per second}}$$

$$= \frac{\rho Q [V_{w_1} u_1 \pm V_{w_2} u_2]}{\rho Q \times g} = \frac{1}{g} [V_{w_1} u_1 \pm V_{w_2} u_2] \qquad ...(18.19)$$

In equation (18.19), +ve sign is taken if angle β is an acute angle. If β is an obtuse angle then – ve sign is taken. If $\beta = 90°$, then $V_{w_2} = 0$ and work done per second per unit weight of water striking/s becomes as

$$= \frac{1}{g} V_{w_1} u_1 \qquad ...(18.20)$$

Hydraulic efficiency is obtained from equation (18.2) as

$$\eta_h = \frac{\text{R.P.}}{\text{W.P.}} = \frac{\dfrac{W}{1000\, g} [V_{w_1} u_1 \pm V_{w_2} u_2]}{\dfrac{W \times H}{1000}} = \frac{(V_{w_1} u_1 \pm V_{w_2} u_2)}{gH} \qquad ...(18.20\text{ A})$$

where R.P. = Runner power *i.e.* power delivered by water to the runner

W.P. = Water power

If the discharge is radial at outlet, then $V_{w_2} = 0$

$$\therefore \qquad \eta_h = \frac{V_{w_1} u_1}{gH} \qquad ...(18.20\text{ B})$$

18.7.3. Definitions. The following terms are generally used in case of reaction radial flow turbines which are defined as :

(*i*) **Speed Ratio.** The speed ratio is defind as $= \frac{u_1}{\sqrt{2gH}}$

where u_1 = Tangential velocity of wheel at inlet.

(*ii*) **Flow Ratio.** The ratio of the velocity of flow at inlet (V_{f_1}) to the velocity given $\sqrt{2gH}$ is known as flow ratio or it is given as

$$= \frac{V_{f_1}}{\sqrt{2gH}}$$ where H = Head on turbine.

(*iii*) **Discharge of the Turbine.** The discharge through a reaction radial flow turbine is given by

$$Q = \pi D_1 B_1 \times V_{f_1} = \pi D_2 \times B_2 \times V_{f_2} \quad ...(18.21)$$

where D_1 = Diameter of runner at inlet,
B_1 = Width of runner at inlet,
V_{f_1} = Velocity of flow at inlet, and
D_2, B_2, V_{f_2} = Corresponding values at outlet.

If the thickness of vanes are taken into consideration, then the area through which flow takes place is given by $(\pi D_1 - n \times t)$

where n = Number of vanes on runner and t = Thickness of each vane.

The discharge Q, then is given by $Q = (\pi D_1 - n \times t)\, B_1 \times V_{f_1}$ ...(18.22)

(*iv*) The head (H) on the turbine is given by $H = \frac{p_1}{w} + \frac{V_1^2}{2g}$ ...(18.23)

where p_1 = Pressure at inlet.

(*v*) **Radial Discharge.** This means the angle made by absolute velocity with the tangent on the wheel is 90° and the component of the whirl velocity is zero. Radial discharge at outlet means $\beta = 90°$ and $V_{w_2} = 0$, while radial discharge at ~~out~~inlet means $\alpha = 90°$ and $V_{w_1} = 0$.

(*vi*) If there is no loss of energy when water flows through the vanes then we have

$$H - \frac{V_2^2}{2g} = \frac{1}{g}[V_{w_1} u_1 \pm V_{w_2} u_2]. \quad ...(18.24)$$

Problem 18.14. *An inward flow reaction turbine has external and internal diameters as 1 m and 0.5 m respectively. The velocity of flow through the runner is constant and is equal to 1.5 m/s. Determine :*

(*i*) *Discharge through the runner, and*

(*ii*) *Width of the turbine at outlet if the width of the turbine at inlet = 200 mm.*

Sol. Given :

External diameter of turbine, $D_1 = 1$ m

Internal diameter of turbine, $D_2 = 0.5$ m

Velocity of flow at inlet and outlet, $V_{f_1} = V_{f_2} = 1.5$ m/s

Width of turbine at inlet, $B_1 = 200$ mm $= 0.20$ m

Let the width at outlet $= B_2$

Using equation (18.21) for discharge,

$$Q = \pi D_1 B_1 \times V_{f_1} = \pi \times 1 \times 0.20 \times 1.5 = 0.9425 \text{ m}^3/\text{s}$$

Also $\quad \pi D_1 B_1 V_{f_1} = \pi D_2 B_2 V_{f_2}$ or $D_1 B_1 = D_2 B_2 \qquad (\because \ \pi V_{f_1} = \pi V_{f_2})$

$$\therefore \quad B_2 = \frac{D_1 \times B_1}{D_2} = \frac{1 \times 0.20}{0.5} = 0.40 \text{ m} = \mathbf{400 \text{ mm}}. \quad \textbf{Ans.}$$

Problem 18.15. *An inward flow reaction turbine has external and internal diameters as 0.9 m and 0.45 m respectively. The turbine is running at 200 r.p.m. and width of turbine at inlet is 200 mm. The velocity of flow through the runner is constant and is equal to 1.8 m/s. The guide blades make an angle of 10° to the tangent of the wheel and the discharge at the outlet of the turbine is radial. Draw the inlet and outlet velocity triangles and determine :*

(*i*) *The absolute velocity of water at inlet of runner,*

(*ii*) *The velocity of whirl at inlet,* (*iii*) *The relative velocity at inlet,*

(*iv*) *The runner blade angles,* (*v*) *Width of the runner at outlet,*

(*vi*) *Mass of water flowing through the runner per second,*

(*vii*) *Head at the inlet of the turbine,*

(*viii*) *Power developed and hydraulic efficiency of the turbine.*

Sol. Given :

External Dia.,	$D_1 = 0.9$ m
Internal Dia.,	$D_2 = 0.45$ m
Speed,	$N = 200$ r.p.m.
Width at inlet,	$B_1 = 200$ mm $= 0.2$ m
Velocity of flow,	$V_{f_1} = V_{f_2} = 1.8$ m/s
Guide blade angle,	$\alpha = 10°$
Discharge at outlet	= Radial
$\therefore$	$\beta = 90°$ and $V_{w_2} = 0$

Tangential velocity of wheel at inlet and outlet are :

$$u_1 = \frac{\pi D_1 N}{60} = \frac{\pi \times .9 \times 200}{60} = 9.424 \text{ m/s}$$

$$u_2 = \frac{\pi D_2 \times N}{60} = \frac{\pi \times .45 \times 200}{60} = 4.712 \text{ m/s.}$$

(*i*) Absolute velocity of water at inlet of the runner *i.e.* V_1

From inlet velocity triangle,

$$V_1 \sin \alpha = V_{f_1}$$

$$\therefore \quad V_1 = \frac{V_{f_1}}{\sin \alpha} = \frac{18}{\sin 10} = 10.365 \text{ m/s.}$$

(*ii*) Velocity of whirl at inlet, *i.e.*, V_{w_1}

$$V_{w_1} = V_1 \cos \alpha = 10.365 \times \cos 10$$

$$= \mathbf{10.207 \text{ m/s.}} \quad \textbf{Ans.}$$

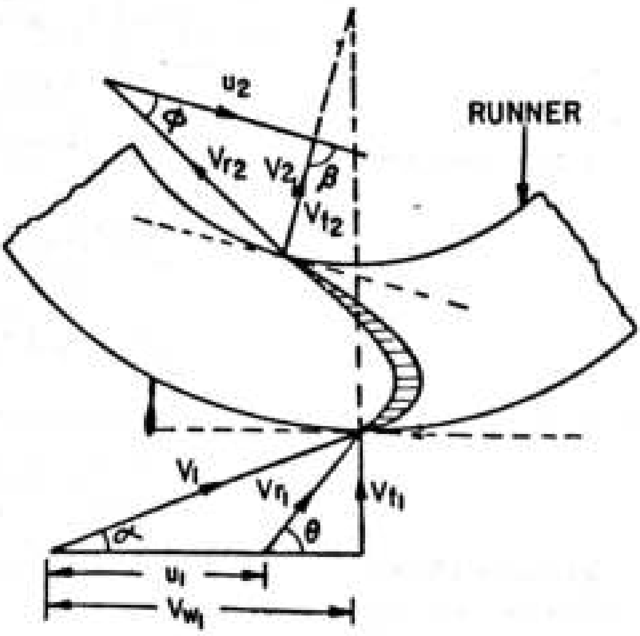


Fig. 18.12

(iii) Relative velocity at inlet, *i.e.*, V_{r_1}

$$V_{r_1} = \sqrt{V_{f_1}^2 + (V_{w_1} - u_1)^2} = \sqrt{1.8^2 + (10.207 - 9.424)^2}$$

$$= \sqrt{3.24 + .613} = \mathbf{1.963\ m/s.}\quad \textbf{Ans.}$$

(iv) The runner blade angles means the angle θ and ϕ

Now $$\tan\theta = \frac{V_{f_1}}{(V_{w_1} - u_1)} = \frac{1.8}{(10.207 - 9.424)} = 2.298$$

$\therefore$ $$\theta = \tan^{-1} 2.298 = 66.48^\circ \text{ or } \mathbf{66^\circ\ 29'}.\quad \textbf{Ans.}$$

From outlet velocity triangle, we have

$$\tan\phi = \frac{V_{f_2}}{u_2} = \frac{1.8}{4.712} = \tan 20.9$$

$\therefore$ $$\phi = 20.9 \text{ or } \mathbf{20^\circ\ 54.4'}.\quad \textbf{Ans.}$$

(v) Width of runner at outlet, *i.e.*, B_2

From equation (18.21), we have

$$\pi D_1 B_1 V_{f_1} = \pi D_2 B_2 V_{f_2} \quad\text{or}\quad D_1 B_1 = D_2 B_2 \qquad (\because\ \pi V_{f_1} = \pi V_{f_2} \text{ as } V_{f_1} = V_{f_2})$$

$\therefore$ $$B_2 = \frac{D_1 B_1}{D_2} = \frac{0.90 \times 0.20}{0.45} = 0.40 \text{ m} = \mathbf{400\ mm.}\quad \textbf{Ans.}$$

(vi) Mass of water flowing through the runner per second.

The discharge, $$Q = \pi D_1 B_1 V_{f_1} = \pi \times 0.9 \times 0.20 \times 1.8 = 1.0178 \text{ m}^3/\text{s}.$$

$\therefore$ **Mass** $$= \rho \times Q = 1000 \times 1.0178 \text{ kg/s} = \mathbf{1017.8\ kg/s.}\quad \textbf{Ans.}$$

(vii) Head at the inlet of turbine, *i.e.*, H.

Using equation (18.24), we have

$$H - \frac{V_2^2}{2g} = \frac{1}{g}(V_{w_1} u_1 \pm V_{w_2} u_2) = \frac{1}{g}(V_{w_1} u_1) \qquad (\because\ \text{Here } V_{w_2} = 0)$$

$$H = \frac{1}{g} V_{w_1} u_1 + \frac{V_2^2}{2g} = \frac{1}{9.81} \times 10.207 \times 9.424 + \frac{1.8^2}{2 \times 9.81} \qquad (\because\ V_2 = V_{f_2})$$

$$= 9.805 + 0.165 = \mathbf{9.97\ m.}\quad \textbf{Ans.}$$

(viii) Power developed, *i.e.*, P $$= \frac{\text{Work done per second on runner}}{1000}$$

$$= \frac{\rho Q\,[V_{w_1} u_1]}{1000} \qquad \text{[Using equation (18.18)]}$$

$$= 1000 \times \frac{1.0178 \times 10.207 \times 9.424}{1000} = \mathbf{97.9\ kW.}\quad \textbf{Ans.}$$

Hydraulic efficiency is given by equation (18.20 B) as

$$\eta_h = \frac{V_{w_1} u_1}{gH} = \frac{10.207 \times 9.424}{9.81 \times 9.97} = 0.9834 = \mathbf{98.34\%}.\quad \textbf{Ans.}$$

Problem 18.16. *A reaction turbine works at 450 r.p.m. under a head of 120 metres. Its diameter at inlet is 120 cm and the flow area is 0.4 m². The angles made by absolute and relative velocities at inlet are 20° and 60° respectively with the tangential velocity. Determine :*

(a) *The volume flow rate,*

(b) *The power developed, and*

(c) *Hydraulic efficiency.*

Assume whirl at outlet to be zero. (AMIE, Fluid Power—Summer, 1979)

Sol. Given :

Speed of turbine, $N = 450$ r.p.m.

Head, $H = 120$ m

Diameter at inlet, $D_1 = 120$ cm $= 1.2$ m

Flow area, $\pi D_1 \times B_1 = 0.4$ m^2

Angle made by absolute velocity at inlet, $\alpha = 20°$

Angle made by the relative velocity at inlet, $\theta = 60°$

Whirl at outlet, $V_{w_2} = 0$

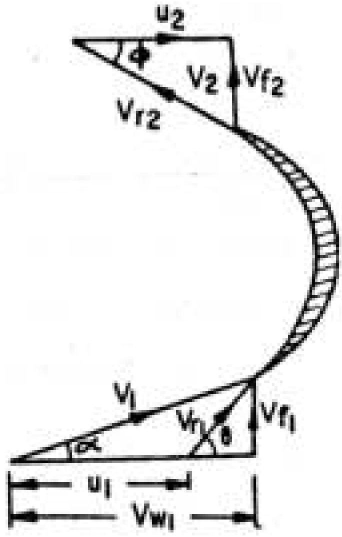


Fig. 18.13

Tangential velocity of the turbine at inlet,

$$u_1 = \frac{\pi D_1 N}{60} = \frac{\pi \times 1.2 \times 450}{60} = 28.27 \text{ m/s}$$

From inlet velocity triangle,

$$\tan \alpha = \frac{V_{f_1}}{V_{w_1}} \text{ or } \tan 20 = \frac{V_{f_1}}{V_{w_1}} \quad \text{or} \quad \frac{V_{f_1}}{V_{w_1}} = \tan 20 = 0.364$$

$\therefore \quad V_{f_1} = 0.364\, V_{w_1}$ ...(*i*)

Also
$$\tan \theta = \frac{V_{f_1}}{V_{w_1} - u_1} = \frac{0.364\, V_{w_1}}{V_{w_1} - 28.27} \qquad (\because\ V_{f_1} = 0.364\, V_{w_1})$$

or
$$\frac{0.364\, V_{w_1}}{V_{w_1} - 28.27} = \tan \theta = \tan 60° = 1.732$$

$$\therefore \quad 0.364\, V_{w_1} = 1.732\,(V_{w_1} - 28.27) = 1.732\, V_{w_1} - 48.96$$

or
$$(1.732 - 0.364)\, V_{w_1} = 48.96$$

$$\therefore \quad V_{w_1} = \frac{48.96}{(1.732 - 0.364)} = 35.789 = 35.79 \text{ m/s.}$$

From equation (*i*), $V_{f_1} = 0.364 \times V_{w_1} = 0.364 \times 35.79 = 13.027$ m/s.

(*i*) Volume flow rate is given by equation (18.21) as $Q = \pi D_1 B_1 \times V_{f_1}$

But $\pi D_1 \times B_1 = 0.4$ m^2 (given)

$Q = 0.4 \times 13.027 =$ **5.211 m³/s. Ans.**

(*ii*) Work done per sec on the turbine is given by equation (18.18),

$$= \rho Q[V_{w_1} u_1] \qquad (\because\ V_{w_2} = 0)$$

$$= 1000 \times 5.211\,[35.79 \times 28.27] = 5272402 \text{ Nm/s}$$

$\therefore$ Power developed in kW $= \dfrac{\text{Work done per second}}{1000} = \dfrac{5272402}{1000} =$ **5272.402 kW. Ans.**

(*iii*) The hydraulic efficiency is given by equation (18.20 B) as

$$\eta_h = \frac{V_{w_1} u_1}{gH} = \frac{35.79 \times 28.27}{9.81 \times 120} = 0.8595 = \mathbf{85.95\%}. \quad \textbf{Ans.}$$

Problem 18.17. *As inward flow reaction turbine has external and internal diameters as 1.0 m and 0.6 m respectively. The hydraulic efficiency of the turbine is 90% when the head on the turbine is 36 m. The velocity of flow at outlet is 2.5 m/s and discharge at outlet is radial. If the vane angle at outlet is 15° and width of the wheel is 100 mm at inlet and outlet, determine : (i) the guide blade angle, (ii) Speed of the turbine, (iii) Vane angle of the runner at inlet, (iv) Volume flow rate of turbine and (v) power developed.*

Sol. Given :

External diameter, $D_1 = 1.0$ m

Internal diameter, $D_2 = 0.6$ m

Hydraulic efficiency, $\eta_h = 90\% = 0.90$

Head, $H = 36$ m

Velocity of flow at outlet, $V_{f_2} = 2.5$ m/s

Discharge is radial, $V_{w_2} = 0$

Vane angle at outlet, $\phi = 15°$

Width of wheel, $B_1 = B_2 = 100 \text{ mm} = 0.1$ m

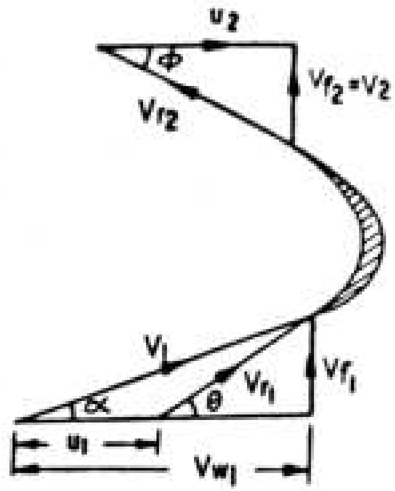


Fig. 18.14

Using equation (18.20 B) for hydraulic efficiency as

$$\eta_h = \frac{V_{w_1} u_1}{gH} \quad \text{or} \quad 0.90 = \frac{V_{w_1} \cdot u_1}{9.81 \times 36}$$

$\therefore$ $$V_{w_1} u_1 = 0.90 \times 9.81 \times 36 = 317.85 \quad \text{...(i)}$$

From outlet velocity triangle, $\tan \phi = \dfrac{V_{f_2}}{u_2} = \dfrac{2.5}{u_2}$

$\therefore$ $$u_2 = \frac{2.5}{\tan \phi} = \frac{2.5}{\tan 15} = 9.33$$

But $$u_2 = \frac{\pi D_2 N}{60} = \frac{\pi \times 0.6 \times N}{60}$$

$\therefore$ $$9.33 = \frac{\pi \times 0.6 \times N}{60} \quad \text{or} \quad N = \frac{60 \times 9.33}{\pi \times 0.6} = 296.98. \quad \textbf{Ans.}$$

$\therefore$ $$u_1 = \frac{\pi \times D_1 \times N}{60} = \frac{\pi \times 1.0 \times 296.98}{60} = 15.55 \text{ m/s.}$$

Substituting this value of 'u_1' in equation (i),

$$V_{w_1} \times 15.55 = 317.85$$

$\therefore$ $$V_{w_1} = \frac{317.85}{15.55} = 20.44 \text{ m/s}$$

Using equation (18.21), $$\pi D_1 B_1 V_{f_1} = \pi D_2 B_2 V_{f_2} \quad \text{or} \quad D_1 V_{f_1} = D_2 V_{f_2} \quad (\because B_1 = B_2)$$

$\therefore$ $$V_{f_1} = \frac{D_2 \times V_{f_2}}{D_1} = \frac{0.6 \times 2.5}{1.0} = 1.5 \text{ m/s.}$$

(*i*) *Guide blade angle* (α).

From inlet velocity triangle, $$\tan \alpha = \frac{V_{f_1}}{V_{w_1}} = \frac{1.5}{20.44} = 0.07338$$

$\therefore$ $$\alpha = \tan^{-1} 0.07338 = 4.19° \text{ or } 4° \, 11.8'. \quad \textbf{Ans.}$$

(*ii*) *Speed of the turbine*

$$N = 296.98 \text{ r.p.m.}$$

(*iii*) *Same angle of runner at inlet* (θ)

$$\tan\theta = \frac{V_{f_1}}{V_{w_1} - u_1} = \frac{1.5}{(20.44 - 15.55)} = 0.3067$$

$\therefore$ $$\theta = \tan^{-1} .3067 = \mathbf{17.05° \text{ or } 17° \, 3'}. \quad \textbf{Ans.}$$

(*iv*) Volume flow rate of turbine is given by equation (18.21) as

$$= \pi D_1 B_1 V_{f_1} = \pi \times 1.0 \times 0.1 \times 1.5 = \mathbf{0.4712 \ m^3/s}. \quad \textbf{Ans.}$$

(*v*) *Power developed* (in kW)

$$= \frac{\text{Work done per second}}{1000} = \frac{\rho Q \, [V_{w_1} u_1]}{1000} \quad \text{[Using equation (18.18) and } V_{w_2} = 0]$$

$$= 1000 \times \frac{0.4712 \times 20.44 \times 15.55}{1000} = \mathbf{149.76 \ kW}. \quad \textbf{Ans.}$$

18.8. FRANCIS TURBINE

The inward flow reaction turbine having radial discharge at outlet is known as Francis Turbine, after the name of J.B. Francis an American engineer who in the beginning designed inward radial flow reaction type of turbine. In the modern Francis turbine, the water enters the runner of the turbine in the radial direction at outlet and leaves in the axial direction at the inlet of the runner. Thus the modern Francis Turbine is a mixed flow type turbine.

The velocity triangle at inlet and outlet of the Francis turbine are drawn in the same way as in case of inward flow reaction turbine. As in case of Francis turbine, the discharge is radial at outlet, the velocity of whirl at outlet (*i.e.*, V_{w_2}) will be zero. Hence the work done by water on the runner per second will be

$$= \rho Q [V_{w_1} u_1]$$

And work done per second per unit weight of water striking/s $= \dfrac{1}{g} [V_{w_1} u_1]$

Hydraulic efficiency will be given by, $\eta_h = \dfrac{V_{w_1} u_1}{gH}$.

18.8.1. Important Relations for Francis Turbines. The following are the important relations for Francis Turbines :

1. The ratio of width of the wheel to its diameter is given as $n = \dfrac{B_1}{D_1}$. The value of n varies from 0.10 to .40.

2. The flow ratio is given as,

   Flow ratio $= \dfrac{V_{f_1}}{\sqrt{2gH}}$ and varies from 0.15 to 0.30.

3. The speed ratio $= \dfrac{u_1}{\sqrt{2gH}}$ varies from 0.6 to 0.9.

Problem 18.23. *A Francis turbine with an overall efficiency of 75% is required to produce 148.25 kW power. It is working under a head of 7.62 m. The peripheral velocity = 0.26 $\sqrt{2gH}$ and the radial velocity of flow at inlet is 0.96 $\sqrt{2gH}$. The wheel runs at 150 r.p.m. and the hydraulic losses in the turbine are 22% of the available energy. Assuming radial discharge, determine :*

(i) The guide blade angle,
(ii) The wheel vane angle at inlet,
(iii) Diameter of the wheel at inlet, and
(iv) Width of the wheel at inlet.

(AMIE, Fluid Power—Winter, 1975)

Sol. Given :

Overall efficiency, $\eta_o = 75\% = 0.75$

Power produced, S.P. = 148.25 kW

Head, $H = 7.62$ m

Peripheral velocity, $u_1 = 0.26\sqrt{2gH} = 0.26 \times \sqrt{2 \times 9.81 \times 7.62} = 3.179$ m/s

Velocity of flow at inlet, $V_{f_1} = 0.96\sqrt{2gH} = 0.96 \times \sqrt{2 \times 9.81 \times 7.62} = 11.738$ m/s.

Speed, $N = 150$ r.p.m.

Hydraulic losses = 22% of available energy

Discharge at outlet = Radial

$$V_{w_2} = 0 \text{ and } V_{f_2} = V_2$$

Hydraulic efficiency is given as

$$\eta_h = \frac{\text{Total head at inlet} - \text{Hydraulic loss}}{\text{Head at inlet}}$$

$$= \frac{H - .22\,H}{H} = \frac{0.78\,H}{H} = 0.78$$

But
$$\eta_h = \frac{V_{w_1} u_1}{gH}$$

$$\therefore \quad \frac{V_{w_1} u_1}{gH} = 0.78$$

$$\therefore \quad V_{w_1} = \frac{0.78 \times g \times H}{u_1}$$

$$= \frac{0.78 \times 9.81 \times 7.62}{3.179} = 18.34 \text{ m/s.}$$

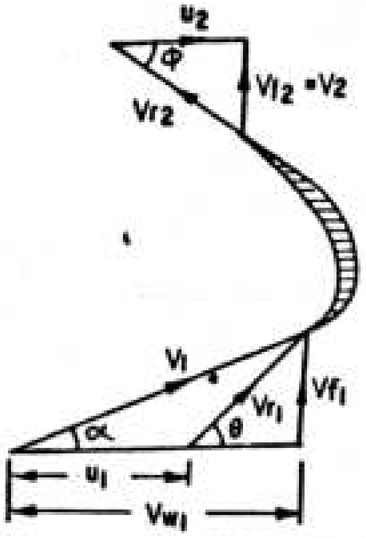


Fig. 18.21

(*i*) *The guide blade angle, i.e.,* α. From inlet velocity triangle,

$$\tan\alpha = \frac{V_{f_1}}{V_{w_1}} = \frac{11.738}{18.34} = 0.64$$

$$\therefore \quad \alpha = \tan^{-1} 0.64 = 32.619° \text{ or } \mathbf{32° \ 37'}. \quad \textbf{Ans.}$$

(*ii*) *The wheel vane angle at inlet, i.e.,* θ

$$\tan\theta = \frac{V_{f_1}}{V_{w_1} - u_1} = \frac{11.738}{18.34 - 3.179} = 0.774$$

$$\therefore \quad \theta = \tan^{-1} .774 = 37.74 \text{ or } \mathbf{37° \ 44.4'}. \quad \textbf{Ans.}$$

(*iii*) *Diameter of wheel at inlet* (D_1).

Using the relation,
$$u_1 = \frac{\pi D_1 N}{60}$$

$$D_1 = \frac{60 \times u_1}{\pi \times N} = \frac{60 \times 3.179}{\pi \times 50} = \mathbf{0.4047 \text{ m.}} \quad \textbf{Ans.}$$

(*iv*) *Width of the wheel at inlet* (B_1)

$$\eta_o = \frac{\text{S.P.}}{\text{W.P.}} = \frac{148.25}{\text{W.P.}}$$

But $$\text{W.P.} = \frac{WH}{1000} = \frac{\rho \times g \times Q \times H}{1000} = \frac{1000 \times 9.81 \times Q \times 7.62}{1000}$$

$\therefore$ $$\eta_o = \frac{148.25}{\frac{1000 \times 9.81 \times Q \times 7.62}{1000}} = \frac{148.25 \times 1000}{1000 \times 9.81 \times Q \times 7.62}$$

or $$Q = \frac{148.25 \times 1000}{1000 \times 9.81 \times 7.62 \times \eta_o} = \frac{148.25 \times 1000}{1000 \times 9.81 \times 7.62 \times 0.75} = 2.644 \text{ m}^3/\text{s}$$

Using equation (18.21), $Q = \pi D_1 \times B_1 \times V_{f_1}$

$\therefore$ $$2.644 = \pi \times .4047 \times B_1 \times 11.738$$

$\therefore$ $$B_1 = \frac{2.644}{\pi \times .4047 \times 11.738} = \mathbf{0.177 \text{ m.}} \quad \textbf{Ans.}$$

Problem 18.24. *The following data is given for a Francis Turbine. Net head $H = 60$ m ; Speed $N = 700$ r.p.m ; shaft power = 294.3 kW ; $\eta_o = 84\%$; $\eta_h = 93\%$; flow ratio = 0.20 ; breadth ratio $n = 0.1$; Outer diameter of the runner = 2 × inner diameter of runner. The thickness of vanes occupy 5% of circumferential area of the runner, velocity of flow is constant at inlet and outlet and discharge is radial at outlet. Determine :*

(i) Guide blade angle, *(ii) Runner vane angles at inlet and outlet,*
(iii) Diameters of runner at inlet and outlet, and (vi) Width of wheel at inlet.

Sol. Given :

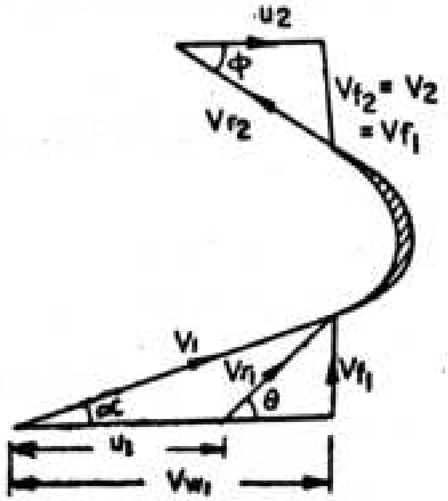


Fig. 18.22

Net head, $H = 60$ m.

Speed, $N = 700$ r.p.m.

Shaft power = 294.3 kW

Overall efficiency, $\eta_o = 84\% = 0.84$

Hydraulic efficiency, $\eta_h = 93\% = 0.93$

Flow ratio, $\dfrac{V_{f_1}}{\sqrt{2gH}} = 0.20$

$\therefore$ $$V_{f_1} = 0.20 \times \sqrt{2gH}$$
$$= 0.20 \times \sqrt{2 \times 9.81 \times 60} = 6.862 \text{ m/s}$$

Breadth ratio, $\dfrac{B_1}{D_1} = 0.1$

Outer diameter, $D_1 = 2 \times \text{Inner diameter} = 2 \times D_2$

Velocity of flow, $V_{f_1} = V_{f_2} = 6.862$ m/s.

Thickness of vanes = 5% of circumferential area of runner

$\therefore$ Actual area of flow $= 0.95\, \pi D_1 \times B_1$

Discharge at outlet = Radial

$\therefore$ $V_{w_2} = 0$ and $V_{f_2} = V_2$

Using relation, $\eta_o = \dfrac{\text{S.P.}}{\text{W.P.}}$

$$0.84 = \frac{294.3}{\text{W.P.}}$$

$$\therefore \qquad \text{W.P.} = \frac{294.3}{0.84} = 350.357 \text{ kW.}$$

But $$\text{W.P.} = \frac{WH}{1000} = \frac{\rho \times g \times Q \times H}{1000} = \frac{1000 \times 9.81 \times Q \times 60}{1000}$$

$$\therefore \qquad \frac{1000 \times 9.81 \times Q \times 60}{1000} = 350.357$$

$$\therefore \qquad Q = \frac{350.357 \times 1000}{60 \times 1000 \times 9.81} = 0.5952 \text{ m}^3\text{/s:}$$

Using equation (18.21), Q = Actual area of flow × Velocity of flow

$$= 0.95\, \pi D_1 \times B_1 \times V_{f_1}$$

$$= 0.95 \times \pi \times D_1 \times 0.1\, D_1 \times V_{f_2} \qquad (\because \; B_1 = 0.1\, D_1)$$

or $$0.5952 = 0.95 \times \pi \times D_1 \times 0.1 \times D_1 \times 6.862 = 2.048\, D_1^2$$

$$\therefore \qquad D_1 = \sqrt{\frac{0.5952}{2.048}} = 0.54 \text{ m}$$

But $$\frac{B_1}{D_1} = 0.1$$

$$\therefore \qquad B_1 = 0.1 \times D_1 = 0.1 \times .54 = .054 \text{ m} = 54 \text{ mm}$$

Tangential speed of the runner at inlet,

$$u_1 = \frac{\pi D_1 N}{60} = \frac{\pi \times 0.54 \times 700}{60} = 19.79 \text{ m/s.}$$

Using relation for hydraulic efficiency,

$$\eta_h = \frac{V_{w_1} u_1}{gH} \quad \text{or} \quad 0.93 = \frac{V_{w_1} \times 19.79}{9.81 \times 60}$$

$$\therefore \qquad V_{w_1} = \frac{0.93 \times 9.81 \times 60}{19.79} = 27.66 \text{ m/s.}$$

(*i*) *Guide blade angle* (α)

From inlet velocity triangle, $\tan \alpha = \dfrac{V_{f_1}}{V_{w_1}} = \dfrac{6.862}{27.66} = 0.248$

$$\therefore \qquad \alpha = \tan^{-1} 0.248 = 13.928° \text{ or } 13° \; 55.7'. \quad \textbf{Ans.}$$

(*ii*) *Runner vane angles at inlet and outlet* (θ *and* ϕ)

$$\tan \theta = \frac{V_{f_1}}{V_{w_1} - u_1} = \frac{6.862}{27.66 - 19.79} = 0.872$$

$$\therefore \qquad \theta = \tan^{-1} 0.872 = 41.09° \text{ or } 41° \; 5.4'. \quad \textbf{Ans.}$$

From outlet velocity triangle, $\tan \phi = \dfrac{V_{f_2}}{u_2} = \dfrac{V_{f_1}}{u_2} = \dfrac{6.862}{u_2}$ ...(*i*)

But $$u_2 = \frac{\pi D_2 N}{60} = \frac{\pi \times D_1}{2} \times \frac{N}{60} \qquad \left(\because \; D_2 = \frac{D_1}{2} \text{ given}\right)$$

$$= \pi \times \frac{.54}{2} \times \frac{700}{60} = 9.896 \text{ m/s.}$$

Substituting the value of u_2 in equation (*i*),

$$\tan \phi = \frac{6.862}{9.896} = 0.6934$$

$$\therefore \qquad \phi = \tan^{-1} .6934° = 34.74 \text{ or } \mathbf{34° \; 44.4'}. \quad \textbf{Ans.}$$

(*iii*) *Diameters of runner at inlet and outlet*

$$D_1 = 0.54 \text{ m}, D_2 = \mathbf{0.27 \ m}. \quad \textbf{Ans.}$$

(*iv*) *Width of wheel at inlet*

$$B_1 = \mathbf{54 \ mm}. \quad \textbf{Ans.}$$

18.9. AXIAL FLOW REACTION TURBINE

If the water flows parallel to the axis of the rotation of the shaft, the turbine is known as axial flow turbine. And if the head at the inlet of the turbine is the sum of pressure energy and kinetic energy and during the flow of water through runner a part of pressure energy is converted into kinetic energy, the turbine is known as reaction turbine.

For the axial flow reaction turbine, the shaft of the turbine is vertical. The lower end of the shaft is made larger which is known as 'hub' or 'boss'. The vanes are fixed on the hub and hence hub acts as a runner for axial flow reaction turbine. The following are the important type of axial flow reaction turbines :

1. Propeller Turbine, and 2. Kaplan Turbine.

When the vanes are fixed to the hub and they are not adjustable, the turbine is known as propeller turbine. But if the vanes on the hub are adjustable the turbine is known as a *Kaplan Turbine*, after the name of V. Kaplan, an Austrian Engineer. This turbine is suitable where a large quantity of water at low heads is available. Fig. 18.25 shows the runner of a Kaplan turbine, which consists of a hub fixed to the shaft. On the hub, the adjustable vanes are fixed as shown in Fig. 18.25.

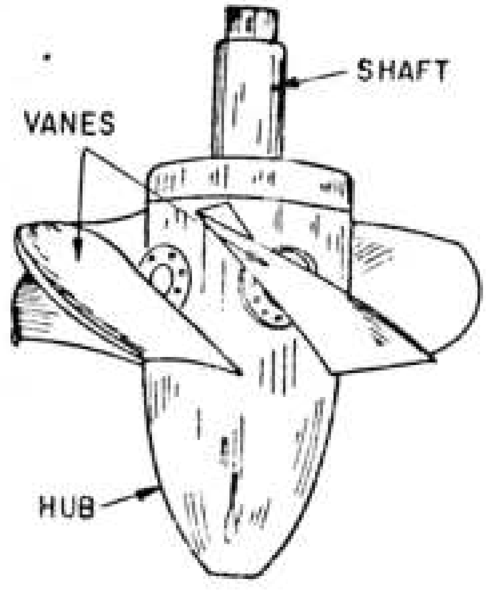


Fig. 18.25. Kaplan turbine runner.

The main parts of a Kaplan turbine are :

1. Scroll casing,
2. Guide vanes mechanism,
3. Hub with vanes or runner of the turbine, and
4. Draft tube.

Fig. 18.26 shows all main parts of a Kaplan turbine. The water from penstock enters the scroll casing and then moves to the guide vanes. From the guide vanes, the water turns through 90° and flows axially through the runner as shown in Fig. 18.26. The discharge through the runner is obtained as

$$Q = \frac{\pi}{4}(D_o^2 - D_b^3) \times V_{f_1} \quad \text{...(18.25)}$$

where D_o = Outer diameter of the runner,

D_b = Diameter of hub

V_{f_1} = Velocity of flow at inlet.

The inlet and outlet velocity triangles are drawn at the extreme edge of the runner vane corresponding to the points 1 and to 2 as shown in Fig. 18.26.

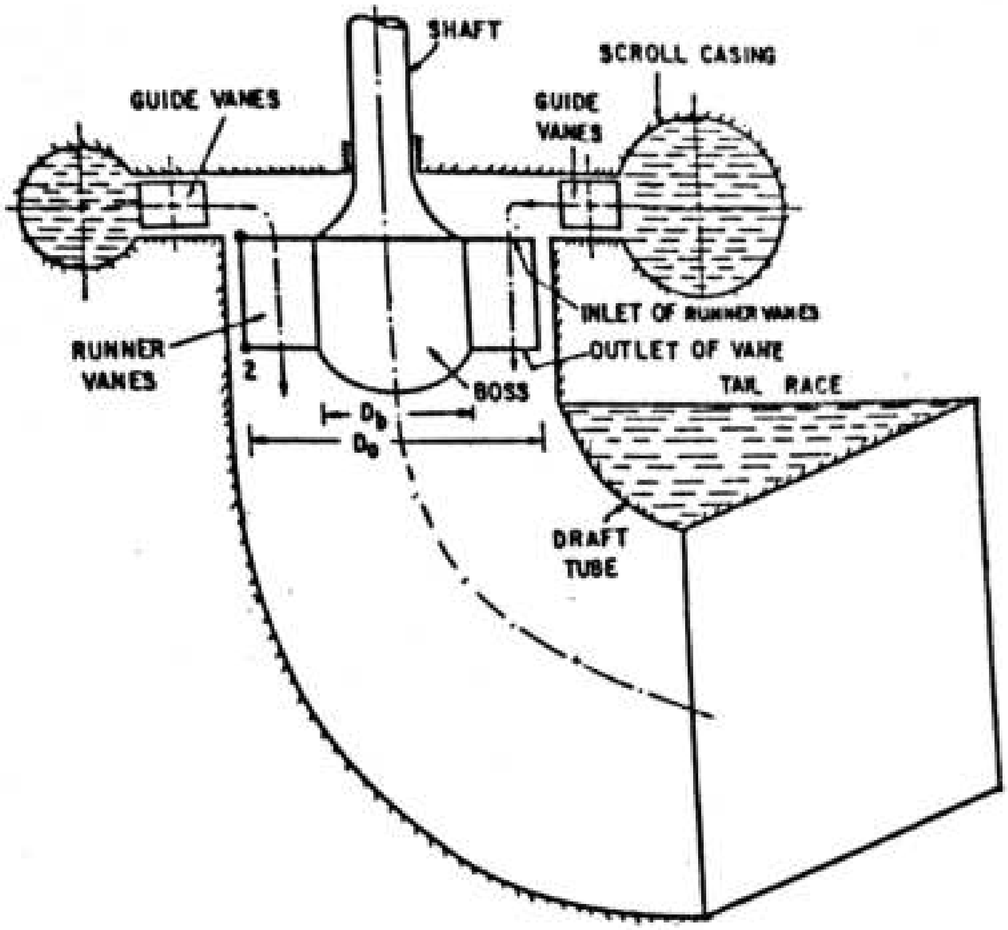


Fig. 18.26. Main components of Kaplan turbine.

18.9.1. Some Important Points for Propeller (Kaplan Turbine). The following are the important points for propeller or Kaplan turbine :

1. The peripheral velocity at inlet and outlet are equal

$\therefore$ $$u_1 = u_2 = \frac{\pi D_o N}{60}$$ where D_o = Outer dia. of runner.

2. Velocity of flow at inlet and outlet are equal

$\therefore$ $$V_{f_1} = V_{f_2}.$$

3. Area of flow at inlet = Area of flow at outlet

$$= \frac{\pi}{4}(D_o^2 - D_b^2).$$

Problem 18.27. *A Kaplan turbine working under a head of 20 m develops 11772 kW shaft power. The outer diameter of the runner is 3.5 m and hub diameter 1.75 m. The guide blade angle at the extreme edge of the runner is 35°. The hydraulic and overall efficiencies of the turbines are 88% and 84% respectively. If the velocity of whirl is zero at outlet, determine :*

(i) Runner vane angles at inlet and outlet at the extreme edge of the runner, and

(ii) Speed of the turbine.

Sol. Given :

Head, $H = 20$ m

Shaft power, S.P. = 11772 kW

Outer dia. of runner, $D_o = 3.5$ m

Hub diameter, $D_b = 1.75$ m

Guide blade angle, $\alpha = 35°$

Hydraulic efficiency, $\eta_h = 88\%$

Overall efficiency, $\eta_o = 84\%$

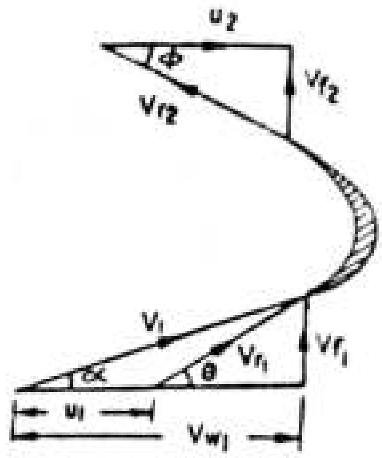


Fig. 18.27

Velocity of whirl at outlet = 0. Using the relation, $\eta_o = \dfrac{\text{S.P.}}{\text{W.P.}}$

where W.P. $= \dfrac{WP}{1000} = \dfrac{\rho \times g \times Q \times H}{1000}$ we get,

$$0.84 = \frac{11772}{\dfrac{\rho \times g \times Q \times H}{1000}}$$

$$= \frac{11772 \times 1000}{1000 \times 9.81 \times Q \times 20} \qquad (\because \; \rho = 1000)$$

$\therefore$
$$Q = \frac{11772 \times 1000}{0.84 \times 1000 \times 9.81 \times 20} = 71.428 \text{ m}^3/\text{s}.$$

Using equation (18.25),
$$Q = \frac{\pi}{4}(D_o^2 - D_b^2) \times V_{f_1}$$

or
$$71.428 = \frac{\pi}{4}(3.5^2 - 1.75^2) \times V_{f_1} = \frac{\pi}{4}(12.25 - 3.0625)\, V_{f_1} = 7.216\, V_{f_1}$$

$\therefore$
$$V_{f_1} = \frac{71.428}{7.216} = 9.9 \text{ m/s}.$$

From inlet velocity triangle, $\tan \alpha = \dfrac{V_{f_1}}{V_{w_1}}$

$\therefore$
$$V_{w_1} = \frac{V_{f_1}}{\tan \alpha} = \frac{9.9}{\tan 35} = \frac{9.9}{.7} = 14.14 \text{ m/s}$$

Using the relation for hydraulic efficiency,

$$\eta_h = \frac{V_{w_1} u_1}{gH} \qquad (\because \; V_{w_2} = 0)$$

$$0.88 = \frac{14.14 \times u_1}{9.81 \times 20}$$

$\therefore$
$$u_1 = \frac{0.88 \times 9.81 \times 20}{14.14} = 12.21 \text{ m/s}.$$

(*i*) **Runner vane angles at inlet and outlet at the extreme edge of the runner are given as :**

$$\tan \theta = \frac{V_{f_1}}{V_{w_1} - u_1} = \frac{9.9}{(14.14 - 12.21)} = 5.13$$

$\therefore$
$$\theta = \tan^{-1} 5.13 = 78.97° \text{ or } \mathbf{78° \, 58'}. \quad \textbf{Ans.}$$

For Kaplan turbine,
$$u_1 = u_2 = 12.21 \text{ m/s} \quad \text{and} \quad V_{f_1} = V_{f_2} = 9.9 \text{ m/s}$$

$\therefore$ From outlet velocity triangle, $\tan \phi = \dfrac{V_{f_2}}{u_2} = \dfrac{9.9}{12.21} = 0.811$

$\therefore$
$$\phi = \tan^{-1} .811 = 39.035° \text{ or } \mathbf{39° \, 2'}. \quad \textbf{Ans.}$$

(*ii*) Speed of turbine is given by $$u_1 = u_2 = \frac{\pi D_o N}{60}$$

$$12.21 = \frac{\pi \times 3.5 \times N}{60}$$

$$\therefore \quad N = \frac{60 \times 12.21}{\pi \times 3.50} = \mathbf{66.63\ r.p.m.}\quad \textbf{Ans.}$$

Problem 18.28. *A Kaplan turbine develops 24647.6 kW power at an average head of 39 metres. Assuming a speed ratio of 2, flow ratio of 0.6, diameter of the boss equal to 0.35 times the diameter of the runner and an overall efficiency of 90%, calculate the diameter, speed and specific speed of the turbine.*

(AMIE, Summer, 1981)

Sol. Given :

Shaft power, $\quad$ S.P. = 24647.6 kW

Head, $\quad H = 39$ m

Speed ratio, $\quad u_1/\sqrt{2gH} = 2.0$

$\therefore \quad u_1 = 2.0 \times \sqrt{2gH} = 2.0 \times \sqrt{2 \times 9.81 \times 39} = 55.32$ m/s

Flow ratio, $\quad \dfrac{V_{f_1}}{\sqrt{2gH}} = 0.6$

$\therefore \quad V_{f_1} = 0.6 \times \sqrt{2gH} = 0.6 \times \sqrt{2 \times 9.81 \times 39} = 16.59$ m/s

Diameter of boss $\quad = 0.35 \times$ Diameter of runner

$\therefore \quad D_b = 0.35 \times D_o$

Overall efficiency, $\quad \eta_o = 90\% = 0.90$

Using the relation, $\quad \eta_o = \dfrac{\text{S.P.}}{\text{W.P.}} \qquad$ where W.P. $= \dfrac{\rho \times g \times Q \times H}{1000}$

$$\therefore \quad 0.90 = \frac{24647.6}{\dfrac{\rho \times g \times Q \times H}{1000}} = \frac{24647.6 \times 1000}{1000 \times 9.81 \times Q \times 39}$$

$$\therefore \quad Q = \frac{24647.6 \times 1000}{0.9 \times 1000 \times 9.81 \times 39} = 71.58\ \text{m}^3/\text{s}.$$

But from equation (18.25), we have

$$Q = \frac{\pi}{4}(D_o^2 - D_b^2) \times V_{f_1}$$

$$\therefore \quad 71.58 = \frac{\pi}{4}[D_o^2 - (.35\,D_o)^2] \times 16.59 \qquad (\because\ D_b = 0.35\,D_o,\ V_{f_1} = 16.59)$$

$$= \frac{\pi}{4}[D_o^2 - .1225\,D_o^2] \times 16.59$$

$$= \frac{\pi}{4} \times .8775\,D_o^2 \times 16.59 = 11.433\,D_o^2$$

$$(i) \therefore \quad D_o = \sqrt{\frac{71.58}{11.433}} = \mathbf{2.5\ m.}\quad \textbf{Ans.}$$

$$\therefore \quad D_o = 0.35 \times D_o = 0.35 \times 2.5 = \mathbf{0.875\ m.}\quad \textbf{Ans.}$$

(*ii*) Speed of the turbine is given by $u_1 = \dfrac{\pi D_o N}{60}$

$$\therefore \quad 55.32 = \frac{\pi \times 2.5 \times N}{60}$$

$$\therefore \quad N = \frac{60 \times 55.32}{\pi \times 2.5} = \textbf{422.61 r.p.m.} \quad \textbf{Ans.}$$

(*iii*) Specific speed* is given by $N_s = \frac{N\sqrt{P}}{H^{5/4}}$ where P = Shaft power in kW

$$\therefore \quad N_s = \frac{422.61 \times \sqrt{24647.6}}{(39)^{5/4}} = \frac{422.61 \times 156.99}{97.461} = \textbf{680.76 r.p.m.} \quad \textbf{Ans.}$$

Problem 18.29. *A Kaplan turbine runner is to be designed to develop 9100 kW. The net available head is 5.6 m. If the speed ratio = 2.09, flow ratio = 0.68, overall efficiency 86% and the diameter of the boss is 1/3 the diameter of the runner. Find the diameter of the runner, its speed and the specific speed of the turbine.*

Sol. Given :

Power, $P = 9100$ kW

Net head, $H = 5.6$ m

Speed ratio $= 2.09$

Flow ratio $= 0.68$

Overall efficiency, $\eta_o = 86\% = 0.86$

Diameter of boss $= \frac{1}{3}$ of diameter of runner

or $$D_b = \frac{1}{3} D_o$$

Now speed ratio $= \frac{u_1}{\sqrt{2gH}}$

$$\therefore \quad u_1 = 2.09 \times \sqrt{2 \times 9.81 \times 5.6} = 21.95 \text{ m/s}$$

Flow ratio $= \frac{V_{f_1}}{\sqrt{2gH}}$

$$\therefore \quad V_{f_1} = 0.68 \times \sqrt{2 \times 9.81 \times 5.6} = 7.12 \text{ m/s}$$

The overall efficiency is given by, $\eta_o = \frac{P}{\left(\frac{\rho \times g.Q.H}{1000}\right)}$

or $$Q = \frac{P \times 1000}{\rho \times g \times H \times \eta_o} = \frac{9100 \times 1000}{1000 \times 9.81 \times 5.6 \times 0.86}$$

$$(\because \rho g = 1000 \times 9.81 \text{ N/m}^3)$$

$$= 192.5 \text{ m}^3\text{/s.}$$

The discharge through a Kaplan turbine is given by

$$Q = \frac{\pi}{4}[D_o^2 - D_b^2] \times V_{f_1}$$

or $$192.5 = \frac{\pi}{4}\left[D_o^2 - \left(\frac{D_o}{3}\right)^2\right] \times 7.12 \qquad \left(\because D_b = \frac{D_o}{3}\right)$$

$$= \frac{\pi}{4}\left[1 - \frac{1}{9}\right] D_o^2 \times 7.12$$

$$\therefore \qquad D_o = \sqrt{\frac{4 \times 192.5 \times 9}{\pi \times 8 \times 7.12}} = \mathbf{6.21\ m.} \quad \textbf{Ans.}$$

The speed of turbine is given by, $u_1 = \frac{\pi DN}{60}$

$$\therefore \qquad N = \frac{60 \times u_1}{\pi \times D} = \frac{60 \times 21.95}{\pi \times 6.21} = \mathbf{67.5\ r.p.m.} \quad \textbf{Ans.}$$

The specific speed is given by, $N_s = \frac{N\sqrt{P}}{H^{5/4}} = \frac{67.5 \times \sqrt{9100}}{5.6^{5/4}} = \mathbf{746.}$ **Ans.**

33.7 Difference between an Impulse Turbine and a Reaction Turbine

Following are the few points of difference between a reaction turbine and an impulse turbine :

S.No.	*Impulse turbine*	*Reaction turbine*
1.	The entire available energy of the water, is first converted into kinetic energy.	The available energy, of the water, is not converted from one form to another.
2.	The water flows through the nozzles and impinges on the buckets, which are fixed to the outer periphery of the wheel.	The water is guided by the guide blades to flow over the moving vanes.
3.	The water impinges on the buckets, with kinetic energy.	The water glides over the moving vanes, with pressures energy.
4.	The pressure of the flowing water remains unchanged, and is equal to the atmospheric pressure.	The pressure of the flowing water is reduced after gliding over the vanes.
5.	It is not essential that the wheel should run full. Moreover, there should be free access of air between the vanes and the wheel.	It is essential that the wheel should always run full, and kept full of water.
6.	The water may be admitted over a part of the circumference or over the whole circumference of the wheel.	The water must be admitted over the whole circumference of the wheel.
7.	It is possible to regulate the flow without loss.	It is not possible to regulate the flow without loss.
8.	The work is done by the change in the kinetic energy of the jet.	The work is done partly by the change in the velocity head, but almost entirely by the change in pressure head.

INSTITUTE OF TEXTILE TECHNOLOGY CHOUDWAR

Hydraulic Machines & Industrial Fluid Power

Chapter-2
Centrifugal Pumps

Centrifugal Pumps

19.1. INTRODUCTION

The hydraulic machines which convert the mechanical energy into hydraulic energy are called pumps. The hydraulic energy is in the form of pressure energy. If the mechanical energy is converted into pressure energy by means of centrifugal force acting on the fluid, the hydraulic machine is called centrifugal pump.

The centrifugal pump acts as a reversed of an inward radial flow reaction turbine. This means that the flow in centrifugal pumps is in the radial outward directions. The centrifugal pump works on the principle of forced vortex flow which means that when a certain mass of liquid is rotated by an external torque, the rise in pressure head of the rotating liquid takes place. The rise in pressure head at any point of the rotating liquid is proportional to the square of tangential velocity of the liquid at that point $\left(i.e., \text{ rise in pressure head } = \frac{V^2}{2g} \text{ or } \frac{\omega^2 r^2}{2g} \right)$. Thus at the outlet of the impeller where radius is more, the rise in pressure head will be more and the liquid will be discharged at the outlet with a high pressure head. Due to this high pressure head, the liquid can be lifted to a high level.

19.2. MAIN PARTS OF A CENTRIFUGAL PUMP

The followings are the main parts of a centrifugal pump :

1. Impeller.
2. Casing.
3. Suction pipe with a foot valve and a strainer.
4. Delivery pipe.

All the main parts of the centrifugal pump are shown in Fig. 19.1.

1. Impeller. The rotating part of a centrifugal pump is called 'impeller'. It consists of a series of backward curved vanes. The impeller is mounted on a shaft which is connected to the shaft of an electric motor.

2. Casing. The casing of a centrifugal pump is similar to the casing of a reaction turbine. It is an air-tight passage surrounding the impeller and is designed in such a way that the kinetic energy of the water discharged at the outlet of the impeller is converted into pressure energy before the water leaves the casing and enters the delivery pipe. The following three types of the casings are commonly adopted :

(*a*) Volute casing as shown in Fig. 19.1.

(*b*) Vortex casing as shown in Fig. 19.2 (*a*).

(*c*) Casing with guide blades as shown in Fig. 19.2 (*b*).

(*a*) **Volute Casing.** Fig. 19.1 shows the volute casing, which surrounds the impeller. It is of spiral type in which area of flow increases gradually. The increase in area of flow decreases the velocity of flow. The decrease in velocity increases the pressure of the water flowing through the casing. It has been observed that in case of volute casing, the efficiency of the pump increases slightly as a large amount of energy is lost due to the formation of eddies in this type of casing.

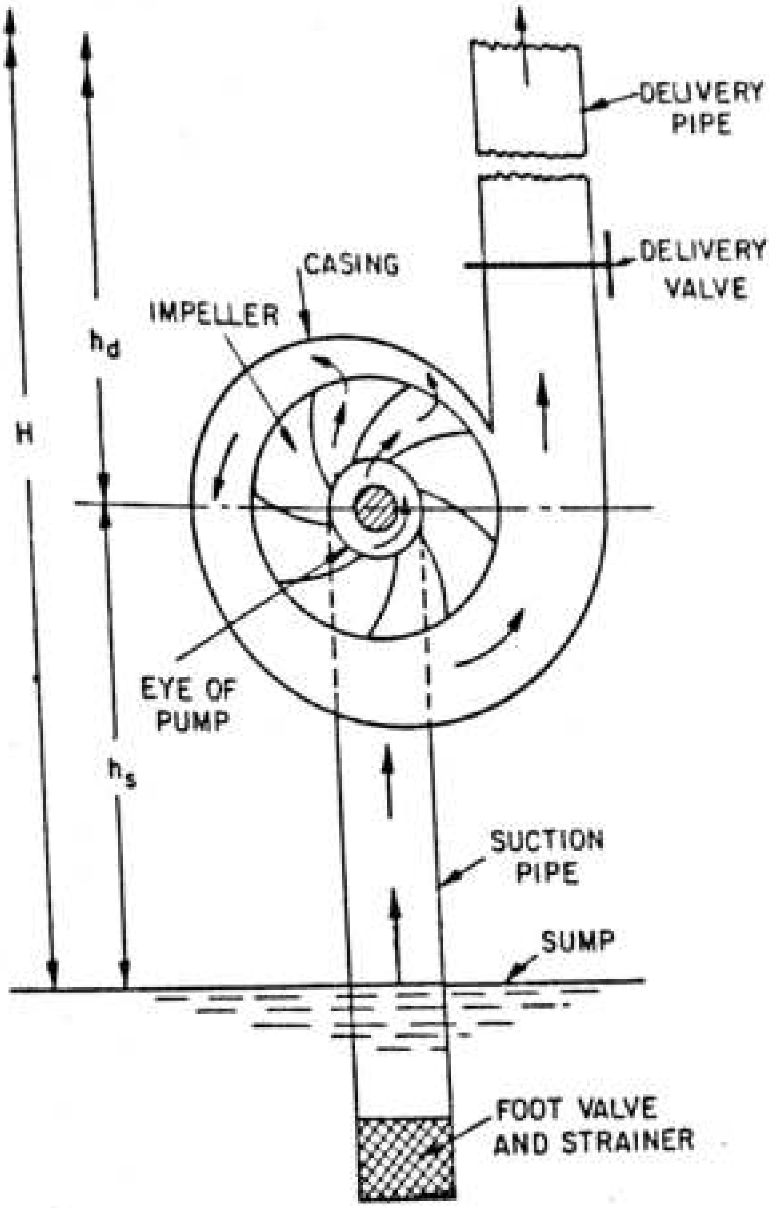


Fig. 19.1. Main parts of a centrifugal pump.

(*b*) **Vortex Casing.** If a circular chamber is introduced between the casing and the impeller as shown in Fig. 19.2 (*a*), the casing is known as Vortex Casing. By introducing the circular chamber, the loss of energy due to the formation of eddies is reduced to a considerable extent. Thus the efficiency of the pump is more than the efficiency when only volute casing is provided.

(*c*) **Casing with Guide Blades.** This casing is shown in Fig. 19.2 (*b*) in which the impeller is surrounded by a series of guide blades mounted on a ring which is known as diffuser. The guide vanes are designed in which a way that the water from the impeller enters the guide vanes without stock. Also the area of the guide vanes increases, thus reducing the velocity of flow through guide vanes and consequently increasing the pressure of water. The water from the guide vanes then passes through the surrounding casing which is in most of the cases concentric with the impeller as shown in Fig. 19.2 (*b*).

3. Suction Pipe with a Foot-valve and a Strainer. A pipe whose one end is connected to the inlet of the pump and other end dips into water in a sump is known as suction pipe. A foot valve which is a non-return valve or one-way type of valve is fitted at the lower end of the suction pipe. The foot valve opens only in the upward direction. A strainer is also fitted at the lower end of the suction pipe.

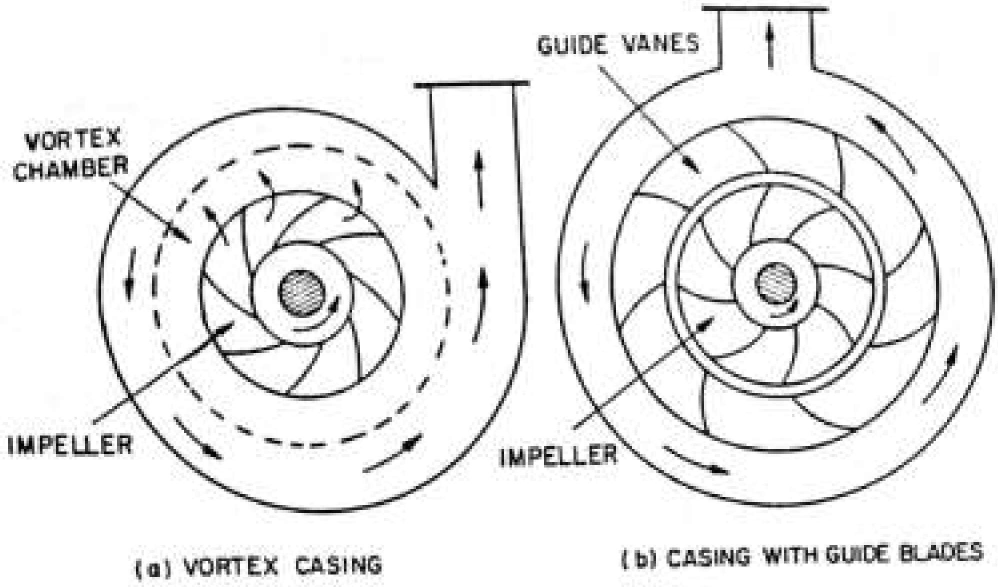


Fig. 19.2. Different types of casing.

4. Delivery Pipe. A pipe whose one end is connected to the outlet of the pump and other end delivers the water at a required height is known as delivery pipe.

19.3. WORK DONE BY THE CENTRIFUGAL PUMP (OR BY IMPELLER) ON WATER

In case of the centrifugal pump, work is done by the impeller on the water. The expression for the work done by the impeller on the water is obtained by drawing velocity triangles at inlet and outlet of the impeller in the same way as for a turbine. The water enters the impeller radially at inlet for best efficiency of the pump, which means the absolute velocity of water at inlet makes an angle of 90° with the direction of motion of the impeller at inlet. Hence angle $\alpha = 90°$ and $V_{w_1} = 0$. For drawing the velocity triangles, the same notations are used as that for turbines. Fig. 19.3 shows the velocity triangles at the inlet and outlet tips of the vane fixed to an impeller.

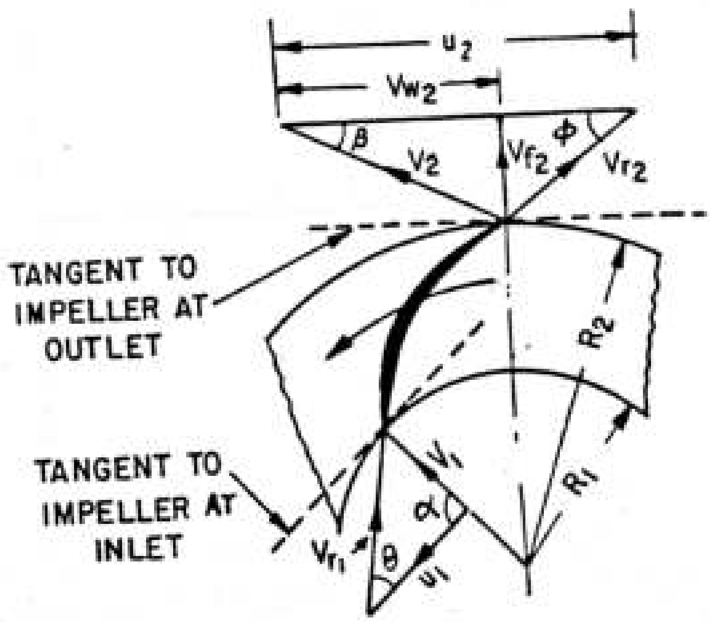


Fig. 19.3. Velocity triangles at inlet and outlet.

Let N = Speed of the impeller in r.p.m.,

D_1 = Diameter of impeller at inlet,

u_1 = Tangential velocity of impeller at inlet,

$= \dfrac{\pi D_1 N}{60}$

D_2 = Diameter of impeller at outlet,

u_2 = Tangential velocity of impeller at outlet

$= \dfrac{\pi D_2 N}{60}$

V_1 = Absolute velocity of water at inlet,

V_{r_1} = Relative velocity of water at inlet,

α = Angle made by absolute velocity (V_1) at inlet with the direction of motion of vane,

θ = Angle made by relative velocity (V_{r_1}) at inlet with the direction of motion of vane, and

V_2, V_{r_2}, β and ϕ are the corresponding values at outlet.

As the water enters the impeller radially which means the absolute velocity of water at inlet is in the radial direction and hence angle $\alpha = 90°$ and $V_{w_1} = 0$.

A centrifugal pump is the reverse of a radially inward flow reaction turbine. But in case of a radially inward flow reaction turbine, the work done by the water on the runner per second per unit weight of the water striking per second is given by the equation (18.19) as

$$= \frac{1}{g}[V_{w_1} u_1 - V_{w_2} u_2]$$

$\therefore$ Work done by the impeller on the water per second per unit weight of water striking per second

$$= -[\text{Work done in case of turbine}]$$

$$= -\left[\frac{1}{g}(V_{w_1} u_1 - V_{w_2} u_2)\right] = \frac{1}{g}[V_{w_2} u_2 - V_{w_1} u_1]$$

$$= \frac{1}{g} V_{w_2} u_2 \qquad (\because \; V_{w_1} = 0 \text{ here}) \qquad ...(19.1)$$

Work done by impeller on water per second

$$= \frac{W}{g} \cdot V_{w_2} u_2 \qquad ...(19.2)$$

where W = Weight of water = $\rho \times g \times Q$

where Q = Volume of water

and Q = Area × Velocity of flow = $\pi D_1 B_1 \times V_{f_1}$

$$= \pi D_2 B_2 \times V_{f_2} \qquad ...(19.2\text{ A})$$

where B_1 and B_2 are width of impeller at inlet and outlet and V_{f_1} and V_{f_2} are velocities of flow at inlet and outlet.

Equation (19.1) gives the head imparted to the water by the impeller or energy given by impeller to water per unit weight per second.

19.4. DEFINITIONS OF HEADS AND EFFICIENCIES OF A CENTRIFUGAL PUMP

1. *Suction Head* (h_s). It is the vertical height of the centre line of the centrifugal pump above the water surface in the tank or pump from which water is to be lifted as shown in Fig. 19.1. This height is also called suction lift and is denoted by 'h_s'.

2. *Delivery Head* (h_d). The vertical distance between the centre line of the pump and the water surface in the tank to which water is delivered is known as delivery head. This is denoted by 'h_d'.

3. *Static Head* (H_s). The sum of suction head and delivery head is known as static head. This is represented by 'H_s' and is written as

$$H_s = h_s + h_d. \qquad ...(19.3)$$

4. *Manometric Head* (H_m). The manometric head is defined as the head against which a centrifugal pump has to work. It is denoted by 'H_m'. It is given by the following expressions :

(*a*) H_m = Head imparted by the impeller to the water – Loss of head in the pump

$$= \frac{V_{w_2} u_2}{g} - \text{Loss of head in impeller and casing} \qquad ...(19.4)$$

$$= \frac{V_{w_2} u_2}{g} \text{ ... if loss of pump is zero} \quad ...(19.5)$$

(b) H_m = Total head at outlet of the pump − Total head at the inlet of the pump

$$= \left(\frac{P_o}{\rho g} + \frac{V_o^2}{2g} + Z_o\right) - \left(\frac{p_i}{\rho g} + \frac{V_i^2}{2g} + Z_i\right) \quad ...(19.6)$$

where $\frac{p_o}{\rho g}$ = Pressure head at outlet of the pump = h_d

$\frac{V_o^2}{2g}$ = Velocity head at outlet of the pump

= Velocity head in delivery pipe = $\frac{V_d^2}{2g}$

Z_o = Vertical height of the outlet of the pump from datum line, and

$\frac{p_i}{\rho g}, \frac{V_i^2}{2g}, Z_i$ = Corresponding values of pressure head, velocity head and datum head at the inlet of the pump,

i.e., $h_s, \frac{V_s^2}{2g}$ and Z_s respectively.

(c) $$H_m = (h_s + h_d) + h_{f_s} + h_{f_d} + \frac{V_d^2}{2g} \quad ...(19.7)$$

where h_s = Suction head, h_d = Delivery head,

h_{f_s} = Frictional head loss in suction pipe, h_{f_d} = Frictional head loss in delivery pipe, and

V_d = Velocity of water in delivery pipe.

5. *Efficiencies of a Centrifugal Pump.* In case of a centrifugal pump, the power is transmitted from the shaft of the electric motor to the shaft of the pump and then to the impeller. From the impeller, the power is given to the water. Thus power is decreasing from the shaft of the pump to the impeller and then to the water. The followings are the important efficiencies of a centrifugal pump :

(a) Manometric efficiency, η_{man} (b) Mechanical efficiency, η_m and

(c) Overall efficiency, η_o.

(a) **Manometric Efficiency** (η_{man}). The ratio of the manometric head to the head imparted by the impeller to the water is known as manometric efficiency. Mathematically, it is written as

$$\eta_{man} = \frac{\text{Manometric head}}{\text{Head imparted by impeller to water}}$$

$$= \frac{H_m}{\left(\frac{V_{w_2} u_2}{g}\right)} = \frac{gH_m}{V_{w_2} u_2} \quad ...(19.8)$$

The power at the impeller of the pump is more than the power given to the water at outlet of the pump. The ratio of the power given to water at outlet of the pump to the power available at the impeller, is known as manometric efficiency.

The power given to water at outlet of the pump $= \dfrac{WH_m}{1000}$ kW

The power at the impeller $= \dfrac{\text{Work done by impeller per second}}{1000}$ kW

$$= \frac{W}{g} \times \frac{V_{w_2} \times u_2}{1000} \text{ kW}$$

$$\therefore \qquad \eta_{man} = \frac{\dfrac{W \times H_m}{1000}}{\dfrac{W}{g} \times \dfrac{V_{w_2} \times u_2}{1000}} = \frac{g \times H_m}{V_{w_2} \times u_2}$$

(*b*) **Mechanical Efficiency** (η_m). The power at the shaft of the centrifugal pump is more than the power available at the impeller of the pump. The ratio of the power available at the impeller to the power at the shaft of the centrifugal pump is known as mechanical efficiency. It is written as

$$\eta_m = \frac{\text{Power at the impeller}}{\text{Power at the shaft}}$$

The power at the impeller in kW $= \dfrac{\text{Work done by impeller per second}}{1000}$

$$= \frac{W}{g} \times \frac{V_{w_2} u_2}{1000} \qquad \text{[Using equation (19.2)]}$$

$$\eta_m = \frac{\dfrac{W}{g}\left(\dfrac{V_{w_2} u_2}{1000}\right)}{\text{S.P.}} \qquad ...(19.9)$$

where S.P. = Shaft power.

(*c*) **Overall Efficiency** (η_o). It is defined as the ratio of power output of the pump to the power input to the pump. The power output of the pump in kW

$$= \frac{\text{Weight of water lifted} \times H_m}{1000} = \frac{WH_m}{1000}$$

Power input to the pump = Power supplied by the electric motor

= S.P. of the pump.

$$\therefore \qquad \eta_o = \frac{\left(\dfrac{WH_m}{1000}\right)}{\text{S.P.}} \qquad ...(19.10)$$

Also $$\eta_o = \eta_{man} \times \eta_m. \qquad ...(19.11)$$

Problem 19.1. *The internal and external diameters of the impeller of a centrifugal pump are 200 mm and 400 mm respectively. The pump is running at 1200 r.p.m. The vane angles of the impeller at inlet and outlet are 20° and 30° respectively. The water enters the impeller radially and velocity of flow is constant. Determine the work done by the impeller per unit weight of water.*

Sol. Given :

Internal diameter of impeller, $D_1 = 200$ mm $= 0.20$ m

External diameter of impeller, $D_2 = 400$ mm $= 0.40$ m

Speed, $N = 1200$ r.p.m.

Vane angle at inlet, $\theta = 20°$

Vane angle at outlet, $\phi = 30°$

Water enters radially* means, $\alpha = 90°$ and $V_{w_1} = 0$

Velocity of flow, $V_{f_1} = V_{f_2}$

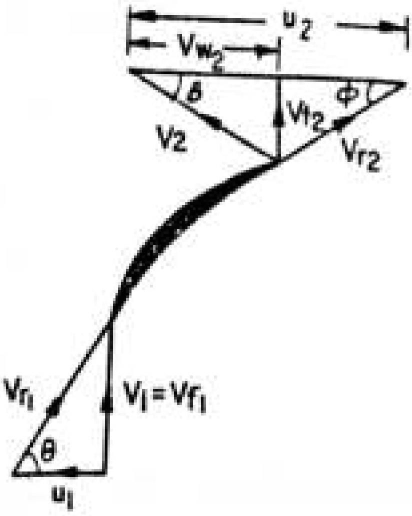


Fig. 19.4

Tangential velocity of impeller at inlet and outlet are,

$$u_1 = \frac{\pi D_1 N}{60} = \frac{\pi \times 0.20 \times 1200}{60} = 12.56 \text{ m/s}$$

and

$$u_2 = \frac{\pi D_2 N}{60} = \frac{\pi \times 0.4 \times 1200}{60} = 25.13 \text{ m/s.}$$

From inlet velocity triangle, $\tan\theta = \frac{V_{f_1}}{u_1} = \frac{V_{f_1}}{12.56}$

$$\therefore \quad V_{f_1} = 12.56 \tan\theta = 12.56 \times \tan 20 = 4.57 \text{ m/s}$$

$$\therefore \quad V_{f_2} = V_{f_1} = 4.57 \text{ m/s.}$$

From outlet velocity triangle, $\tan\phi = \frac{V_{f_2}}{u_2 - V_{w_2}} = \frac{4.57}{25.13 - V_{w_2}}$

or

$$25.13 - V_{w_2} = \frac{4.57}{\tan\phi} = \frac{4.57}{\tan 30} = 7.915$$

$$\therefore \quad V_{w_2} = 25.13 - 7.915 = 17.215 \text{ m/s.}$$

The work done by impeller per kg of water per second is given by equation (19.1) as

$$= \frac{1}{g} V_{w_2} u_2 = \frac{17.215 \times 25.13}{9.81} = \textbf{44.1 Nm/N.} \quad \textbf{Ans.}$$

Problem 19.2. *A centrifugal pump is to discharge 0.118 m³/s at a speed of 1450 r.p.m. against a head of 25 m. The impeller diameter is 250 mm, its width at outlet is 50 mm and manometric efficiency is 75%. Determine the vane angle at the outer periphery of the impeller.* (AMIE, Winter, 1982)

Sol. Given :

Discharge, $Q = 0.118 \text{ m}^3/\text{s}$

Speed, $N = 1450$ r.p.m.

Head, $H_m = 25$ m

Diameter at outlet, $D_2 = 250 \text{ mm} = 0.25$ m

Width at outlet, $B_2 = 50 \text{ mm} = 0.05$ m

Manometric efficiency, $\eta_{man} = 75\% = 0.75$.

Let vane angle at outlet $= \phi$

Tangential velocity of impeller at outlet,

$$u_2 = \frac{\pi D_2 N}{60} = \frac{\pi \times 0.25 \times 1450}{60} = 18.98 \text{ m/s}$$

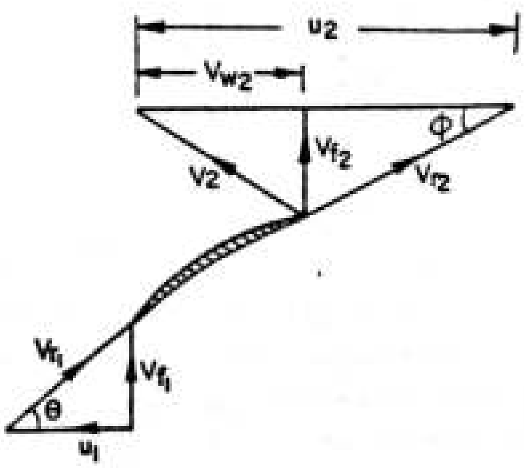


Fig. 19.5

*If in the problem, this condition is not given even then the water is assumed to be entering radially unless stated otherwise.

Discharge is given by $$Q = \pi D_2 B_2 \times V_{f_2}$$

$\therefore$ $$V_{f_2} = \frac{Q}{\pi D_2 B_2} = \frac{0.118}{\pi \times 0.25 \times .05} = 3.0 \text{ m/s.}$$

Using equation (19.8), $$\eta_{man} = \frac{gH_m}{V_{w_2} u_2} = \frac{9.81 \times 25}{V_{w_2} \times 18.98}$$

$\therefore$ $$V_{w_2} = \frac{9.81 \times 25}{\eta_{man} \times 18.98} = \frac{9.81 \times 25}{0.75 \times 18.98} = 17.23.$$

From outlet velocity triangle, we have

$$\tan \phi = \frac{V_{f_2}}{(u_2 - V_{w_2})} = \frac{3.0}{(18.98 - 17.23)} = 1.7143$$

$\therefore$ $$\phi = \tan^{-1} 1.7143 = 59.74° \text{ or } \mathbf{59° \, 44'}. \quad \textbf{Ans.}$$

Problem 19.3. *A centrifugal pump delivers water against a net head of 14.5 metres and a design speed of 1000 r.p.m. The vanes are curved back to an angle of 30° with the periphery. The impeller diameter is 300 mm and outlet width 50 mm. Determine the discharge of the pump if manometric efficiency is 95%.*

(AMIE, Winter, 1983 ; Osmania University, 1992)

Sol. Given :

Net head, $H_m = 14.5$ m

Speed, $N = 1000$ r.p.m.

Vane angle at outlet, $\phi = 30°$

Impeller diameter means the diameter of the impeller at outlet

$\therefore$ Diameter, $D_2 = 300 \text{ mm} = 0.30$ m

Outlet width, $B_2 = 50 \text{ mm} = 0.05$ m

Manometric efficiency, $\eta_{man} = 95\% = 0.95$

Tangential velocity of impeller at outlet,

$$u_2 = \frac{\pi D_2 N}{60} = \frac{\pi \times 0.30 \times 1000}{60} = 15.70 \text{ m/s.}$$

Now using equation (19.8), $$\eta_{man} = \frac{gH_m}{V_{w_2} \times u_2}$$

$\therefore$ $$0.95 = \frac{9.81 \times 14.5}{V_{w_2} \times 15.70}$$

$\therefore$ $$V_{w_2} = \frac{0.95 \times 14.5}{0.95 \times 15.70} = 9.54 \text{ m/s.}$$

Refer to Fig. 19.5. From outlet velocity triangle, we have

$$\tan \phi = \frac{V_{f_2}}{(u_2 - V_{w_2})} \quad \text{or} \quad \tan 30° = \frac{V_{f_2}}{(15.70 - 9.54)} = \frac{V_{f_2}}{6.16}$$

$\therefore$ $$V_{f_2} = 6.16 \times \tan 30° = 3.556 \text{ m/s}$$

$\therefore$ Discharge, $$Q = \pi D_2 B_2 \times V_{f_2}$$
$$= \pi \times 0.30 \times 0.05 \times 3.556 \text{ m}^3\text{/s} = \mathbf{0.1675 \text{ m}^3\text{/s}}. \quad \textbf{Ans.}$$

Problem 19.4. *A centrifugal pump having outer diameter equal to two times the inner diameter and running at 1000 r.p.m. works against a total head of 40 m. The velocity of flow through the impeller is constant and equal to 2.5 m/s. The vanes are set back at an angle of 40° at outlet. If the outer diameter of the impeller is 500 mm and width at outlet is 50 mm, determine :*

(*i*) *Vane angle at inlet,* (*ii*) *Work done by impeller on water per second, and*

(*iii*) *Manometric efficiency.*

(Delhi University, May 1998)

Sol. Given :

Speed, $N = 1000$ r.p.m.

Head, $H_m = 40$ m

Velocity of flow, $V_{f_1} = V_{f_2} = 2.5$ m/s

Vane angle at outlet, $\phi = 40°$

Outer dia. of impeller, $D_2 = 500$ mm $= 0.50$ m

Inner dia. of impeller, $D_1 = \dfrac{D_2}{2} = \dfrac{0.50}{2} = 0.25$ m

Width at outlet, $B_2 = 50$ mm $= 0.05$ m

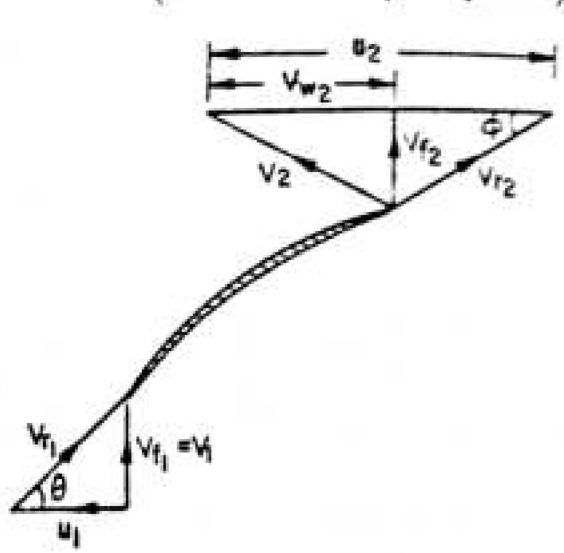


Fig. 19.6

Tangential velocity of impeller at inlet and outlet are

$$u_1 = \frac{\pi D_1 N}{60} = \frac{\pi \times 0.25 \times 1000}{60} = 13.09 \text{ m/s}$$

and

$$u_2 = \frac{\pi D_2 N}{60} = \frac{\pi \times 0.50 \times 1000}{60} = 26.18 \text{ m/s.}$$

Discharge is given by, $Q = \pi D_2 B_2 \times V_{f_2} = \pi \times 0.50 \times .05 \times 2.5 = 0.1963 \text{ m}^3\text{/s.}$

(*i*) **Vane angle at inlet** (θ).

From inlet velocity triangle $\tan\theta = \dfrac{V_{f_1}}{u_1} = \dfrac{2.5}{13.09} = 0.191$

$\therefore$ $\theta = \tan^{-1} .191 = 10.81°$ or **10° 48′. Ans.**

(*ii*) Work done by impeller on water per second is given by equation (19.2) as

$$= \frac{W}{g} V_{w_2} u_2 = \frac{\rho \times g \times Q}{g} \times V_{w_2} \times u_2$$

$$= \frac{1000 \times 9.81 \times 0.1963}{9.81} \times V_{w_2} \times 26.18 \qquad \text{...(}i\text{)}$$

But from outlet velocity triangle, we have

$$\tan\phi = \frac{V_{f_2}}{u_2 - V_{w_2}} = \frac{2.5}{(26.18 - V_{w_2})}$$

$$\therefore \quad 26.18 - V_{w_2} = \frac{2.5}{\tan\phi} = \frac{2.5}{\tan 40} = 2.979$$

$$\therefore \quad V_{w_2} = 26.18 - 2.979 = 23.2 \text{ m/s.}$$

Substituting this value of V_{w_2} in equation (*i*), we get the work done by impeller as

$$= \frac{1000 \times 9.81 \times 0.1963}{9.81} \times 23.2 \times 26.18$$

$$= \mathbf{119227.9 \text{ Nm/s.}} \quad \textbf{Ans.}$$

(iii) **Manometric efficiency** (η_{man}). Using equation (19.8), we have

$$\eta_{man} = \frac{gH_m}{V_{w_2} u_2} = \frac{9.81 \times 40}{23.2 \times 26.18} = 0.646 = \mathbf{64.6\%}. \quad \textbf{Ans.}$$

Problem 19.5. *A centrifugal pump discharges 0.15 m³/s of water against a head of 12.5 m, the speed of the impeller being 600 r.p.m. The outer and inner diameters of impeller are 500 mm and 250 mm respectively and the vanes are bent back at 35° to the tangent at exit. If the area of flow remains 0.07 m² from inlet to outlet, calculate :*

(i) Manometric efficiency of pump, *(ii) Vane angle at inlet, and*

(iii) Loss of head at inlet to impeller when the discharge is reduced by 40% without changing the speed.

(AMIE, Summer, 1988)

Sol. Given :

Discharge, $Q = 0.15$ m³/s

Head, $H_m = 12.5$ m

Speed, $N = 600$ r.p.m.

Outer dia., $D_2 = 500$ mm $= 0.50$ m

Inner dia., $D_1 = 250$ mm $= 0.25$ m

Vane angle at outlet, $\phi = 35°$

Area of flow, $= 0.07$ m²

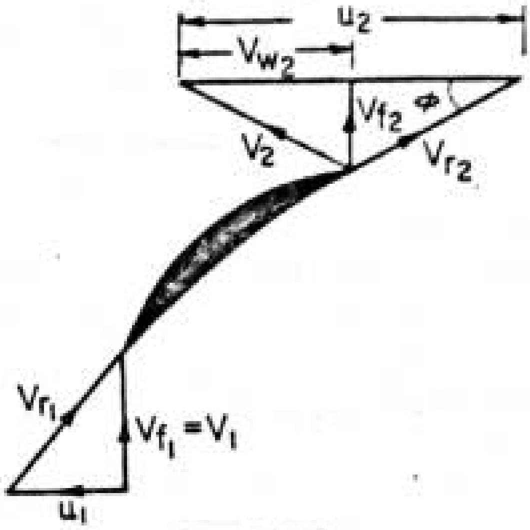


Fig. 19.6 *(a)*

As area of flow is constant from inlet to outlet, then velocity of flow will be constant from inlet to outlet.

Discharge = Area of flow × Velocity of flow

or 0.15 = 0.07 × Velocity of flow

$\therefore$ Velocity of flow $= \dfrac{0.15}{0.07} = 2.14$ m/s

$\therefore$ $V_{f_1} = V_{f_2} = 2.14$ m/s

Tangential velocity of impeller at inlet and outlet are

$$u_1 = \frac{\pi D_1 N}{60} = \frac{\pi \times 0.25 \times 600}{60} = 7.85 \text{ m/s}$$

and

$$u_2 = \frac{\pi D_2 N}{60} = \frac{\pi \times 0.50 \times 600}{60} = 15.70 \text{ m/s}$$

From outlet velocity triangle, $V_{w_2} = u_2 - \dfrac{V_{f_2}}{\tan \phi} = 15.70 - \dfrac{2.14}{\tan 35°} = 12.64$ m/s

(i) Manometric efficiency of the pump

Using equation (19.8), we have $\eta_{man} = \dfrac{g \times H_m}{V_{w_2} \times u_2} = \dfrac{9.81 \times 12.5}{12.64 \times 15.7} = \mathbf{0.618}$ or **61.8%**. **Ans.**

(ii) Vane angle at inlet (θ)

From inlet velocity triangle, $\tan \theta = \dfrac{V_{f_1}}{u_1} = \dfrac{2.14}{7.85} = 0.272$

$\therefore$ $\theta = \tan^{-1} 0.272 = \mathbf{15° \ 12'}$. **Ans.**

19.6. MULTISTAGE CENTRIFUGAL PUMPS

If a centrifugal pump consists of two or more impellers, the pump is called a multistage centrifugal pump. The impellers may be mounted on the same shaft or on different shafts. A multistage pump is having the following two important functions :

1. To produce a high head, and 2. To discharge a large quantity of liquid.

If a high head is to be developed, the impellers are connected in series (or on the same shaft) while for discharging large quantity of liquid, the impellers (or pumps) are connected in parallel.

19.6.1. Multistage Centrifugal Pumps for High Heads. For developing a high head, a number of impellers are mounted in series or on the same shaft as shown in Fig. 19.12.

The water from suction pipe enters the 1st impeller at inlet and is discharged at outlet with increased pressure. The water with increased pressure from the outlet of the 1st impeller is taken to the inlet of the 2nd impeller with the help of a connecting pipe as shown in Fig. 19.12. At the outlet of the 2nd impeller, the pressure of water will be more than the pressure of water at the outlet of the 1st impeller. Thus if more impellers are mounted on the same shaft, the pressure at the outlet will be increased further.

Let n = Number of identical impellers mounted on the same shaft,

H_m = Head developed by each impeller.

Then total head developed

$$= n \times H_m \quad \text{...(19.16)}$$

The discharge passing through each impeller is same.

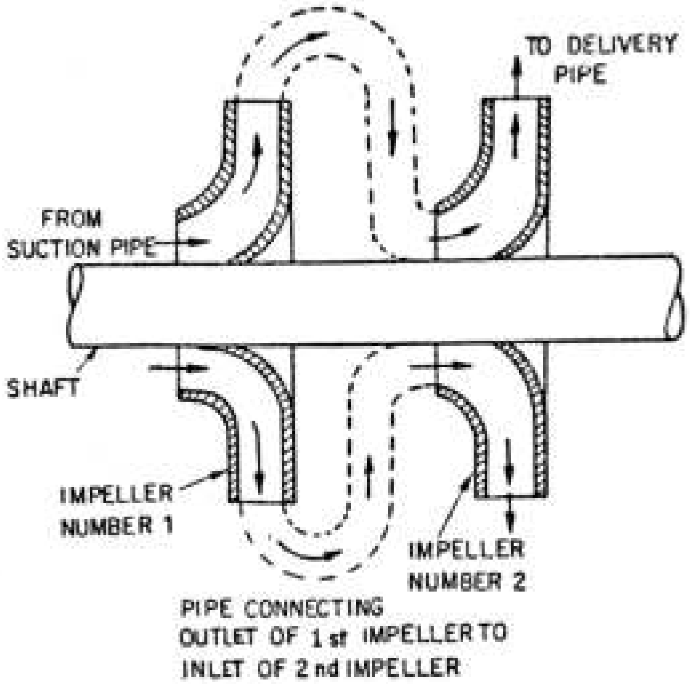


Fig. 19.12. Two-stage pumps with impellers in series.

19.6.2. Multistage Centrifugal Pumps for High Discharge. For obtaining high discharge, the pumps should be connected in parallel as shown in Fig. 19.13. Each of the pumps lifts the water from a common sump and discharges water to a common pipe to which the delivery pipes of each pump is connected. Each of the pump is working against the same head.

Let n = Number of identical pumps arranged in parallel.

Q = Discharge from one pump.

$\therefore$ Total discharge $= n \times Q$. ....(19.17)

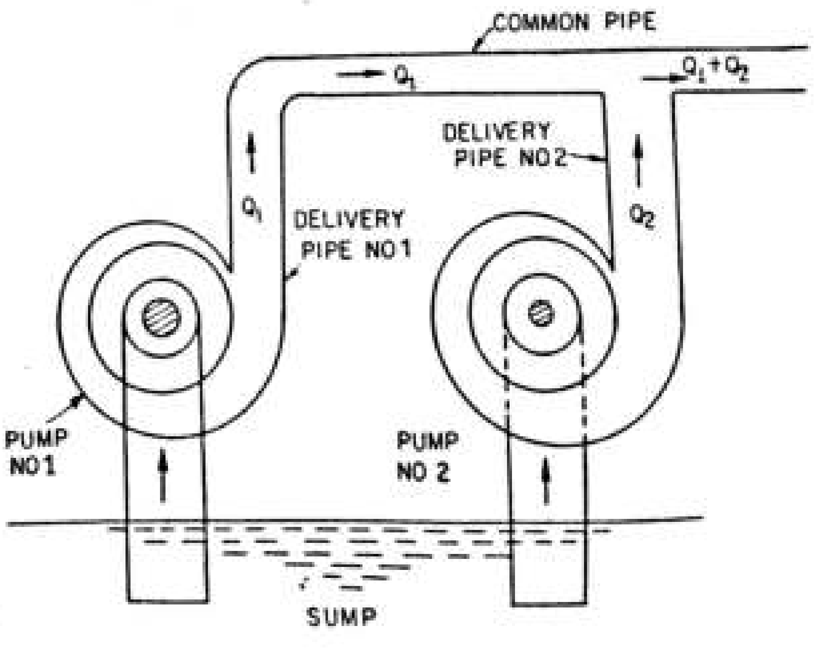


Fig. 19.13. Pumps in parallel.

INSTITUTE OF TEXTILE TECHNOLOGY CHOUDWAR

Hydraulic Machines & Industrial Fluid Power

Chapter-3
Reciprocating Pumps

20.1. INTRODUCTION

In the last chapter, we have defined the pumps as the hydraulic machines which convert the mechanical energy into hydraulic energy which is mainly in the form of pressure energy. If the mechanical energy is converted into hydraulic energy, by means of centrifugal force acting on the liquid, the pump is known as centrifugal pump. But if the mechanical energy is converted into hydraulic energy (or pressure energy) by sucking the liquid into a cylinder in which a piston is reciprocating (moving backwards and forwards), which exerts the thrust on the liquid and increases its hydraulic energy (pressure energy), the pump is known as reciprocating pump.

20.2. MAIN PARTS OF A RECIPROCATING PUMP

The followings are the main parts of a reciprocating pump as shown in Fig. 20.1 :

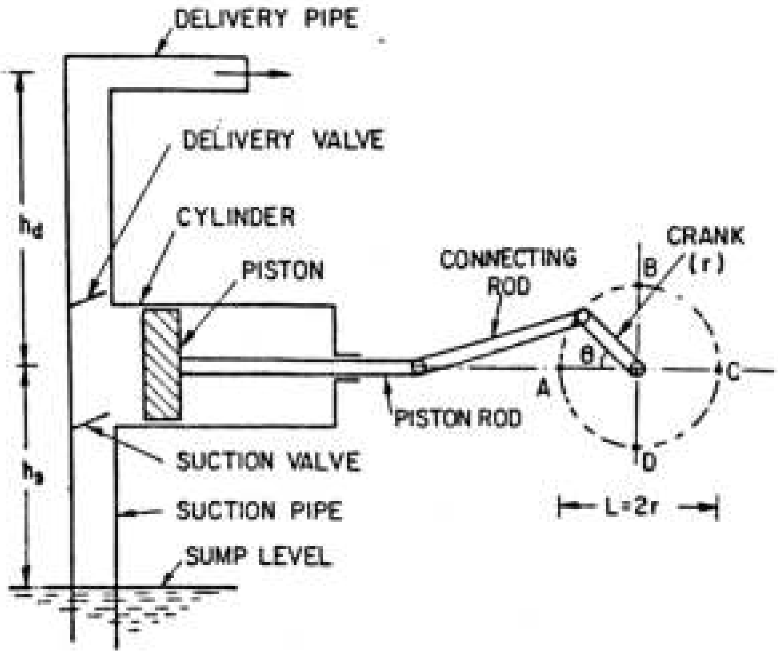


Fig. 20.1. Main parts of a reciprocating pump.

1. A cylinder with a piston, piston rod, connecting rod and a crank,
2. Suction pipe,
3. Delivery pipe,
4. Suction valve, and
5. Delivery valve.

20.3. WORKING OF A RECIPROCATING PUMP

Fig. 20.1 shows a single acting reciprocating pump, which consists of a piston which moves forwards and backwards in a close fitting cylinder. The movement of the piston is obtained by connecting the piston rod to crank by means of a connecting rod. The crank is rotated by means of an electric motor. Suction and delivery pipes with suction valve and delivery valve are connected to the cylinder. The suction and delivery valves are one way valves or non-return valves, which allow the water to flow in one direction only. Suction valve allows water from suction pipe to the cylinder which delivery valve allows water from cylinder to delivery pipe only.

When crank starts rotating, the piston moves to and fro in the cylinder. When crank is at A, the piston is at the extreme left position in the cylinder. As the crank is rotating from A to C, (*i.e.*, from $\theta = 0$ to $\theta = 180°$), the piston is moving towards right in the cylinder. The movement of the piston towards right creates a partial vacuum in the cylinder. But on the surface of the liquid in the sump atmospheric pressure is acting, which is more than the pressure inside the cylinder. Thus the liquid is forced in the suction pipe from the sump. This liquid opens the suction valve and enters the cylinder.

When crank is rotating from C to A (*i.e.*, from $\theta = 180°$ to $\theta = 360°$), the piston from its extreme right position starts moving towards left in the cylinder. The movement of the piston towards left increases the pressure of the liquid inside the cylinder more than atmospheric pressure. Hence suction valve closes and delivery valve opens. The liquid is forced into the delivery pipe and is raised to a required height.

20.3.1. Discharge through a Reciprocating Pump. Consider a single* acting reciprocating pump as shown in Fig. 20.1.

Let D = Diameter of the cylinder

A = Cross-sectional area of the piston or cylinder

$= \frac{\pi}{4} D^2$

r = Radius of crank

N = r.p.m. of the crank

L = Length of the stroke $= 2 \times r$

h_s = Height of the axis of the cylinder from water surface in sump.

h_d = Height of delivery outlet above the cylinder axis (also called delivery head)

Volume of water delivered in one revolution or discharge of water in one revolution

= Area × Length of stroke

$= A \times L$

Number of revolution per second, $= \frac{N}{60}$

$\therefore$ Discharge of the pump per second,

Q = Discharge in one revolution × No. of revolution per second

$$= A \times L \times \frac{N}{60} = \frac{ALN}{60} \quad \text{...(20.1)}$$

Weight of water delivered per second,

$$W = \rho \times g \times Q = \frac{\rho g ALN}{60}. \quad \text{...(20.2)}$$

20.3.2. Work done by Reciprocating Pump. Work done by the reciprocating pump per second is given by the relation as

Work done per second = Weight of water lifted per second × Total height through which water is lifted

$$= W \times (h_s + h_d) \qquad \ldots(i)$$

where $(h_s + h_d)$ = Total height through which water is lifted.

From equation (20.2), Weight, W, is given by

$$W = \frac{\rho g \times ALN}{60}.$$

Substituting the value of W in equation (*i*), we get

Work done per second $$= \frac{\rho g \times ALN}{60} \times (h_s + h_d) \qquad \ldots(20.3)$$

$\therefore$ Power required to drive the pump, in kW

$$P = \frac{\text{Work done per second}}{1000} = \frac{\rho g \times ALN \times (h_s + h_d)}{60 \times 1000}$$

$$= \frac{\rho g \times ALN \times (h_s + h_d)}{60{,}000} \text{ kW} \qquad \ldots(20.4)$$

20.3.3. Discharge, Work done and Power required to Drive a Double-acting Pump. In case of double acting pump, the water is acting on both sides of the piston as shown in Fig. 20.2. Thus we require two suction pipes and two delivery pipes for double-acting pump. When there is a suction stroke on one side of the piston, there is at the same time a delivery stroke on the other side of the piston. Thus for one complete revolution of the crank there are two delivery strokes and water is delivered to the pipes by the pump during these two delivery strokes.

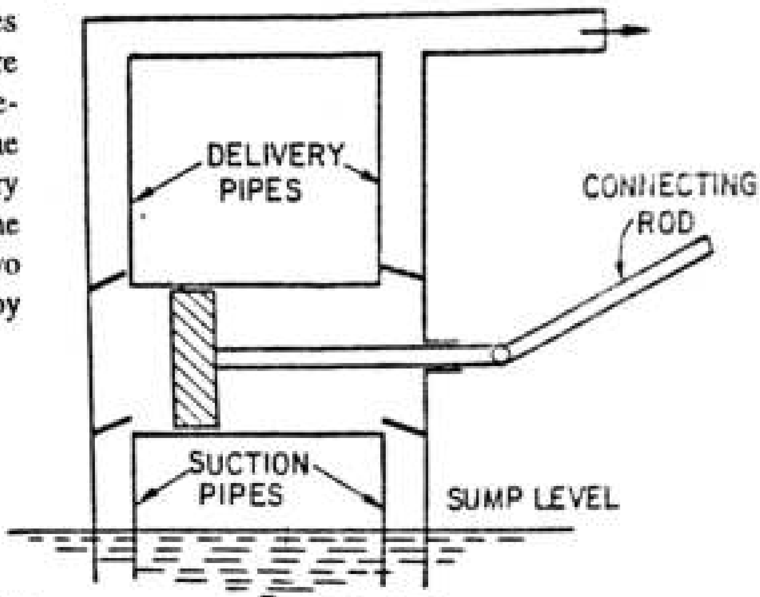


Fig. 20.2

Let D = Diameter of the piston,

d = Diameter of the piston rod

$\therefore$ Area on one side of the piston,

$$A = \frac{\pi}{4} D^2.$$

Area on the other side of the piston, where piston rod is connected to the piston,

$$A_1 = \frac{\pi}{4} D^2 - \frac{\pi}{4} d^2 = \frac{\pi}{4} (D^2 - d^2).$$

$\therefore$ Volume of water delivered in one revolution of crank

$$= A \times \text{Length of stroke} + A_1 \times \text{Length of stroke}$$

$$= AL + A_1 L = (A + A_1)L = \left[\frac{\pi}{4} D^2 + \frac{\pi}{4} (D^2 - d^2)\right] \times L$$

$\therefore$ Discharge of pump per second

= Volume of water delivered in one revolution × No. of revolution per second

$$= \left[\frac{\pi}{4} D^2 + \frac{\pi}{4} (D^2 - d^2)\right] \times L \times \frac{N}{60}$$

If 'd' the diameter of the piston rod is very small as compared to the diameter of the piston, then it can be neglected and discharge of pump per second,

$$Q = \left(\frac{\pi}{4}D^2 + \frac{\pi}{4}D^2\right) \times \frac{L \times N}{60} = 2 \times \frac{\pi}{4}D^2 \times \frac{L \times N}{60} = \frac{2ALN}{60} \qquad ...(20.5)$$

Equation (20.5) gives the discharge of a double-acting reciprocating pump. This discharge is two times the discharge of a single-acting pump.

Work done by double-acting reciprocating pump

Work done per second = Weight of water delivered × Total height

= ρg × Discharge per second × Total height

$$= \rho g \times \frac{2ALN}{60} \times (h_s + h_d) = 2\rho g \times \frac{ALN}{60} \times (h_s + h_d) \qquad ...(20.6)$$

∴ Power required to drive the double-acting pump in kW,

$$P = \frac{\text{Work done per second}}{1000} = 2\rho g \times \frac{ALN}{60} \times \frac{(h_s + h_d)}{1000}$$

$$= \frac{2\rho g \times ALN \times (h_s + h_d)}{60{,}000}. \qquad ...(20.7)$$

20.4. SLIP OF RECIPROCATING PUMP

Slip of a pump is defined as the difference between the theoretical discharge and actual discharge of the pump. The discharge of a single acting pump given by equation (20.1) and of a double-acting pump given by equation (20.5) are theoretical discharge. The actual discharge of a pump is less than the theoretical discharge due to leakage. The difference of the theoretical discharge and actual discharge is known as slip of the pump. Hence, mathematically.

$$\text{Slip} = Q_{th} - Q_{act} \qquad ...(20.8)$$

But slip is mostly expressed as percentage slip which is given by,

$$\text{Percentage slip} = \frac{Q_{th} - Q_{act}}{Q_{th}} \times 100 = \left(1 - \frac{Q_{act}}{Q_{th}}\right) \times 100$$

$$= (1 - C_d) \times 100 \qquad \left(\because \frac{Q_{act}}{Q_{th}} = C_d\right) \qquad ...(20.9)$$

where C_d = Co-efficient of discharge.

20.4.1. Negative Slip of the Reciprocating Pump. Slip is equal to the difference of theoretical discharge and actual discharge. If actual discharge is more than the theoretical discharge, the slip of the pump will become –ve. In that case, the slip of the pump is known as negative slip.

Negative slip occurs when delivery pipe is short, suction pipe is long and pump is running at high speed.

20.5. CLASSIFICATION OF RECIPROCATING PUMPS

The reciprocating pumps may be classified as :

1. According to the water being in contact with one side or both sides of the piston, and
2. According to the number of cylinders provided.

If the water is in contact with one side of the piston, the pump is known as single-acting. On the other hand, if the water is in contact with both sides of the piston, the pump is called double-acting. Hence, classification according to the contact of water is :

(*i*) Single-acting pump, and (*ii*) Double-acting pump.

According to the number of cylinder provided, the pumps are classified as :

(*i*) Single cylinder pump, (*ii*) Double cylinder pump, and

(*iii*) Triple cylinder pump.

Problem 20.1. *A single acting reciprocating pump, running at 50 r.p.m., delivers 0.01 m^3/s of water. The diameter of the piston is 200 mm and stroke length 400 mm. Determine :*

(i) The theoretical discharge of the pump, (ii) Co-efficient of discharge, and

(iii) Slip and the percentage slip of the pump.

Sol. Given :

Speed of the pump, $N = 50$ r.p.m.

Actual discharge, $Q_{act} = .01\ m^3/s$

Dia. of piston, $D = 200\ mm = .20\ m$

$\therefore$ Area, $A = \frac{\pi}{4}(.2)^2 = .031416\ m^2$

Stroke, $L = 400\ mm = 0.40\ m.$

(*i*) Theoretical discharge for single-acting reciprocating pump is given by equation (20.1) as

$$Q_{th} = \frac{A \times L \times N}{60} = \frac{.031416 \times .40 \times 50}{60} = \mathbf{0.01047\ m^3/s.} \quad \textbf{Ans.}$$

(*ii*) Co-efficient of discharge is given by

$$C_d = \frac{Q_{act}}{Q_{th}} = \frac{0.01}{.01047} = \mathbf{0.955.} \quad \textbf{Ans.}$$

(*iii*) Using equation (20.8), we get

$$\text{Slip} = Q_{th} - Q_{act} = .01047 - .01 = \mathbf{0.00047\ m^3/s.} \quad \textbf{Ans.}$$

And percentage slip

$$= \frac{(Q_{th} - Q_{act})}{Q_{th}} \times 100 = \frac{(.01047 - .01)}{.01047} \times 100$$

$$= \frac{.00047}{.01047} \times 100 = \mathbf{4.489\%.} \quad \textbf{Ans.}$$

Problem 20.2. *A double-acting reciprocating pump, running at 40 r.p.m., is discharging 1.0 m^3 of water per minute. The pump has a stroke of 400 mm. The diameter of the piston is 200 mm. The delivery and suction heads are 20 m and 5 m respectively. Find the slip of the pump and power required to drive the p ıp.*

Sol. Given :

Speed of pump, $N = 40$ r.p.m.

Actual discharge, $Q_{act} = 1.0\ m^3/min = \frac{1.0}{60}\ m^3/s = 0.01666\ m^3/s$

Stroke, $L = 400\ mm = 0.40\ m$

Diameter of piston, $D = 200\ mm = 0.20\ m$

$\therefore$ Area, $A = \frac{\pi}{4}D^2 = \frac{\pi}{4}(.2)^2 = 0.031416\ m^2$

Suction head, $h_s = 5\ m$

Delivery head, $h_d = 20\ m.$

Theoretical discharge for double acting pump is given by equation (20.5) as,

$$Q_{th} = \frac{2ALN}{60} = \frac{2 \times .031416 \times 0.4 \times 40}{60} = .01675\ m^3/s.$$

Using equation (20.8), $\text{Slip} = Q_{th} - Q_{act} = .01675 - .01666 = \mathbf{.00009\ m^3/s.}$ **Ans.**

Power required to drive the double acting pump is given by équation (20.7) as,

$$P = \frac{2 \times \rho g \times ALN \times (h_s + h_d)}{60,000} = \frac{2 \times 1000 \times 9.81 \times .031416 \times .4 \times 40 \times (5 + 20)}{60,000}$$

$= \mathbf{4.109\ kW.}$ **Ans.**

INSTITUTE OF TEXTILE TECHNOLOGY CHOUDWAR

Hydraulic Machines & Industrial Fluid Power

Fluid Power Fundamentals

Fluid Power Fundamentals

After completing this chapter, you will understand

- *The fundamentals of fluid power,*
- *Different fluid power systems,*
- *The physics behind their application,*
- *Fluids and their properties*

INTRODUCTION

The fluids have been used to help human from ancient times. Even before man developed an understanding of the science and knowledge of how it can be usefully used, it had been used as an application to reduce his burden. The primitive application known is the water wheel used in irrigation and the driving of ship with the aid of wind, using logs of wood to cross the river etc. But once the science of fluid and industrial revolution joined hands the application of the fluids have been wide and ever growing and now it is used in all the fields of engineering, biomedical, space, automobile, defense, agriculture and all industrial sectors.

The greatest advantage of this system is its versatility to be controlled by a feather touch and drive a large power (in tones) and its precision in its application when used in repeated loading with close tolerances (in microns). In this fast growing computer/ electronic world, it is still advantageous and easy to control this powerful muscle remotely, smoothly, efficiently, safely and precisely to accomplish useful work.

The development in the designing of a hydraulic and pneumatic system is today integrating with recent developments namely the electronics and computers. The use of fluid power system in industrial sectors had helped in producing quality components at less cost and less time. Fluid power is not used only in industrial sectors but also in household applications. They are available in small size which are portable and easy to operate.

FLUID POWER

The technology of generating, controlling and transmitting power using pressurized fluids is termed as *fluid power*. Fluids are either gas or liquids. They are termed *hydraulics* for liquids and *pneumatics* for gases. Hydraulic systems use petroleum oil, synthetic oil, water etc., while pneumatics use air as the most prime medium.

BASIC METHODS OF TRANSMITTING POWER:

There are basically three methods of transmitting power. They are

(*i*) Electrical – power transmission over large distance.

(*ii*) Mechanical – power transmission to short distance

(*iii*) Fluid power – power transmission for intermediate distance.

Most of the applications use a combination of these three methods to obtain most efficient overall system.

FLUID SYSTEMS

The type of systems used for transporting fluids from one place to another via pipe accessories in household and industrial applications are termed as *Fluid Transport System.*

Fluid Power Systems are specifically used to perform work. In this system, the prime mover (electric motors) are coupled to components (pumps or compressors) to supply pressurized fluid to produce translation motion (using Cylinders) or rotary motion (using motors)

FLUID POWER PHYSICS

ENERGY

Fluid power is one of the methods of energy transfer. Energy transfer is from the power source to an actuator.

Energy is defined as the ability to do work (Watts).

Work is force through distance (N-m)

Power is the rate of doing work (Nm/s)

LAW OF CONSERVATION OF ENERGY

"Energy can neither be created nor be destroyed but can be changed from one from to another". In fluid power, energy not used is converted to heat. The heat generated in the system is controlled using heat exchangers.

FLOW

The Centrifugal pumps are non-positive displacement pumps, which are used, in fluid transport. Flow in hydraulic systems is generated using positive displacement pumps. Flow is required at the actuator *to make it go. The rate of flow* (depends on pump) determines speed of the actuator. For a constant flow rate, depending on the actuator volume the speed of the actuator changes (Variable speed of extension and retraction is due to difference in area of the piston)

PRESSURE

Pressure **P** = **Force / Area**

Pressure in a hydraulic system comes from resistance to flow. Pump produces flow and not pressure. If the flow is restricted

(*i*) When passing through the components of the system (pipe, elbows, etc.) or

(*ii*) Loads induced by the actuator,

then it results in pressure.

Pascal's Law:

"Pressure applied on a confined fluid at rest is transmitted undiminished in all directions and acts with equal force on equal areas, and at right angles to them."

Load Induced Pressure

Load induced pressure is defined as pressure generated from the load of force on the actuator.

Problem: A double acting cylinder is used to push and pull 10000 N and they have 10 mm^2 and 5mm^2 area on the piston and rod side areas. Determine the pressure induced due to loads.

Solution:

Force F = 10000 N,

Cylinder Piston area A_1=10 mm^2,

Piston Rod area A_2 = 5 mm ^{2}

Pressure during extension $P_1 = F / A_1 = 10000/10 = 1000$ N/mm^2

Pressure during retraction $P_2 = F / A_2 = 10000/5 = 2000$ N/mm^2

PRESSURE DROP

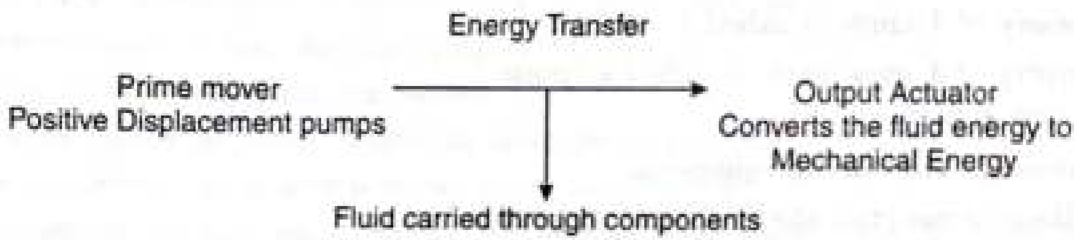
Pressure that was not used directly for doing work is called as *pressure drop* or *resistive pressure*. Excessive pressure drop results in increased heat generation. This pressure drop should be added to the system pressure to calculate overall pressure requirements during designing a system.

FLUIDS

The most important component in the hydraulic system is the fluid. They are primarily used as

- Lubricants – Fluids as a lubricant allow the relatively sliding blocks to move with less friction and wear of parts.
- Energy transfer – They transfer the energy from the input to output devices as they are incompressible.
- Heat transfer – The heated fluids enters and radiates the heat energy out and keeps the system cooler.
- Sealant – The fluid between the sliding spool of the valves and the outer cylinder acts as sealant because of its viscosity.

Hydraulic fluids are incompressible and take the shape of the container. This advantage is used in the fluid power system.



VELOCITY

Velocity is the distance transferred per unit time (m/s). This depends on the area of the conductor. As the area of the conductor increases (depends on the inside diameter of the conductor) the velocity decreases and vice versa.

VISCOSITY

Viscosity is defined as the measure of a liquid's resistance.

The viscosity depends on (i) thickness and (ii) temperature.

- A thicker fluid offers more resistance and hence more viscosity.
- As the temperature increases the viscosity of the fluid decreases.

Viscosity of the fluid is determined using ***Viscosimeter (or Viscometer).***

Procedure to determine Viscosity:

The Fluid whose viscosity is to be determined is kept in a beaker and heated in the oil bath. The temperature is noted using the thermometer. At the required temperatures say Room temperature, 45° , 50°, 55°, etc…the standard sized knob at the bottom is opened and the time required to collect 60 ml of the fluid is noted, which is called as the *Saybolt Universal Seconds* (SUS). As the temperature increases the time required to collect the fluid decreases indicating the decrease in viscosity with increase in temperature. The procedure to determine the viscosity is illustrated next page.

The *kinematic viscosity* (ν) in SUS and centistokes (cS) are empirically related as below:

ν (cS) = 0.226t-195/t, t $\leq$100 SUS or seconds

ν (cS) = 0.220t-135/t, t >100 SUS or seconds

Pumps

After completing this chapter, you will come to know

- *The different components of hydraulic system.*
- *Different pumps used in hydraulic system*
- *Their functionality, advantage and disadvantages*

COMPONENTS OF HYDRAULIC SYSTEM

There are six basic components in the hydraulic system. They are:

(*i*) Tank (Reservoir) to hold the liquid, usually hydraulic oil

(*ii*) A pump to force liquid into the system

(*iii*) An electric motor or power source to drive the pump (prime mover)

(*iv*) Valves to control liquid direction, pressure and flow rate

(*v*) An actuator to convert liquid energy into useful work (linear force or rotary torque)

(*vi*) Piping to carry liquid to all the locations

PUMPS

The pumps are the heart of the hydraulic system. Pumps transform the Mechanical energy they receive from the prime mover (electric motor) into Fluid energy.

PUMPING THEORY

All pumps operate on the principle that a partial vacuum is created at the inlet of the pump due to internal operation of the pump. This allows atmospheric pressure to push the fluid out of the reservoir and into the pump intake. The pump then mechanically pushes the fluid out into the discharge line.

PUMP CLASSIFICATION

There are two broad classifications of pumps. They are

- Hydrodynamic or non-positive displacement pumps
- Hydrostatic or positive displacement pumps

Hydrodynamic or non-positive displacement pumps

These are low pressure, high volume flow pumps. They are used only for fluid transport and are not used in fluid power industry. In these pumps there is large clearance between rotating and stationary elements and hence the discharge rate is low. Their flow rate depends not only on rotational speed but also on external resistance. Their discharge decreases with increase in external resistance. It is also possible to completely stop the pump discharge at maximum pressure (resistance) during pump operation. This maximum pressure is termed as "shutoff head". These pumps are larger in size. They have relatively small volumetric efficiency compared to positive pumps and low-pressure discharge output. Examples of these pumps are:

- Centrifugal pumps (*Impeller type*)
- Axial pumps (*Propeller pumps*)

Hydrostatic or positive displacement pumps

The Hydrostatic or positive displacement pumps eject a fixed volume of flow into the hydraulic system per revolution of pump shaft rotation. These pumps overcome external pressure (from mechanical loads and resistance to flow due to friction) and are small and compact in size. They have large volumetric efficiency and high-pressure discharge output. Based on the nature of the sliding motion between the relative parts and based on construction these pumps are broadly classified as:

1. Rotary pumps
2. Reciprocating pumps.

Rotary pumps

In rotary pumps (Gear pump, Vane pump, Screw pump, Lobe pump, Gerotor pump) the drivers are coupled with the prime mover and rotate inside a housing. The driven element (gear, screw, lobe) rotates in the opposing direction. At the inlet they move away from each other creating partial vacuum at the inlet and move towards each other at the inlet creating high pressure to push the liquid into the discharge line. In vane pumps, the vanes move out of their radial slots near the inlet and move in near the discharge port.

Reciprocating Pumps

In Reciprocating Pumps (Piston and Cylinder arrangement) the piston moves away at the inlet valve resulting in partial vacuum. This pushes the fluid into the cylinder from the reservoir, as the atmospheric pressure is large. When the piston is reversed the valve that opened during suction is closed and this increase in pressure opens the discharge valve and pushes the fluid into the discharge line. Eg. Piston Pump (Radial, Inline, Axial types)

GEAR PUMPS

There are two different types of gear pumps. They are

(i) Internal Gear Pumps and (ii) External Gear Pumps.

EXTERNAL GEAR PUMP

Mostly external gear pumps are used. They have *meshing gears of equal size*. The drive gear is coupled with the drive shaft of the electric motor. This gear drives the other gear. As they rotate the fluid is trapped and carried between the teeth of the driver and driven gears and the external casing, which is in close contact with the gears. The pump creates flow and as they pass through the components of the systems, pressure is generated and transmits it to the actuator. The displacement of the gear pump increases with an increase in input rpm.

Volumetric Displacement And Theoretical Flow Rate

Theoretically the displaced volume and flow rate can be determined as below:

D_o is the addendum circle diameter of the gear teeth,

D_i is the base circle diameter of the gear

W the width of gears,

N speed of revolution of the prime mover,

Q_T theoretical pump flow rate,

V_D displaced volume of the pump.

Volumetric displacement $V_D = \pi/4\,(D_o^2 - D_i^2)\,W$ (mm^3).

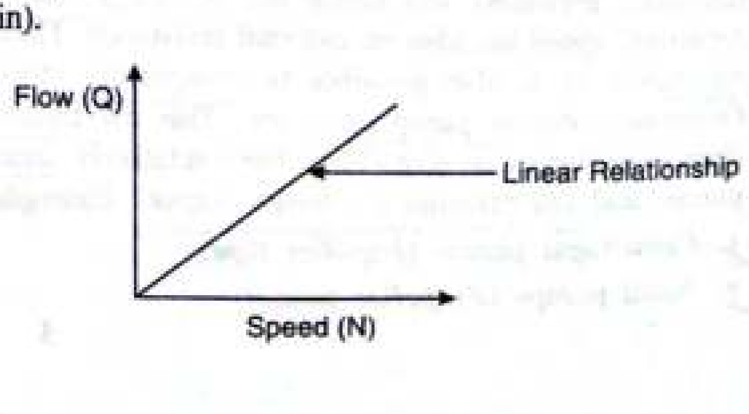
Theoretical flow rate $Q_T = V_D N$ (mm^3/min).

The theoretical equations show the pump flow depends on

(*i*) The size of gears and

(*ii*) The speed of revolution.

The pump flow varies directly with speed and is independent of other parameters. This is shown in the adjacent figure



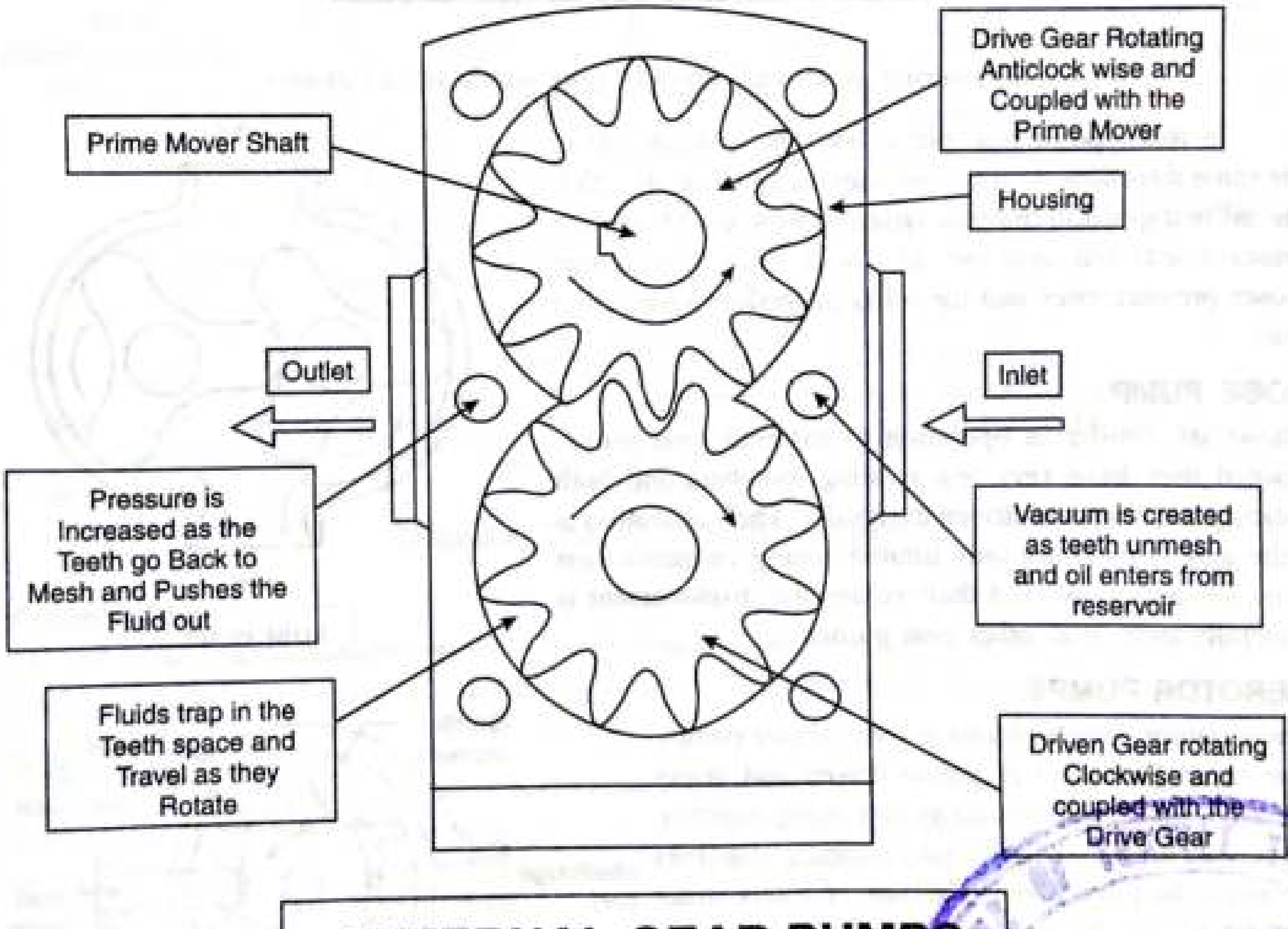
Volumetric Efficiency (η_v)

There is a small clearance between the tip of the gear and the housing (approximately $1/1000^{th}$ of an inch for practical reasons of rotation). This may carry back some oil without discharging at the outlet port. As a result of this leakage, termed as 'pump slippage' the actual flow rate Q_A is less than theoretical flow rate. The ratio of actual flow rate to theoretical flow rate is termed as volumetric efficiency. The efficiency of the positive displacement pumps is usually more than 90% at rated pressure.

$$\text{Volumetric efficiency } \eta_v = (Q_A / Q_T) \times 100\ \%$$

Rated Pressure

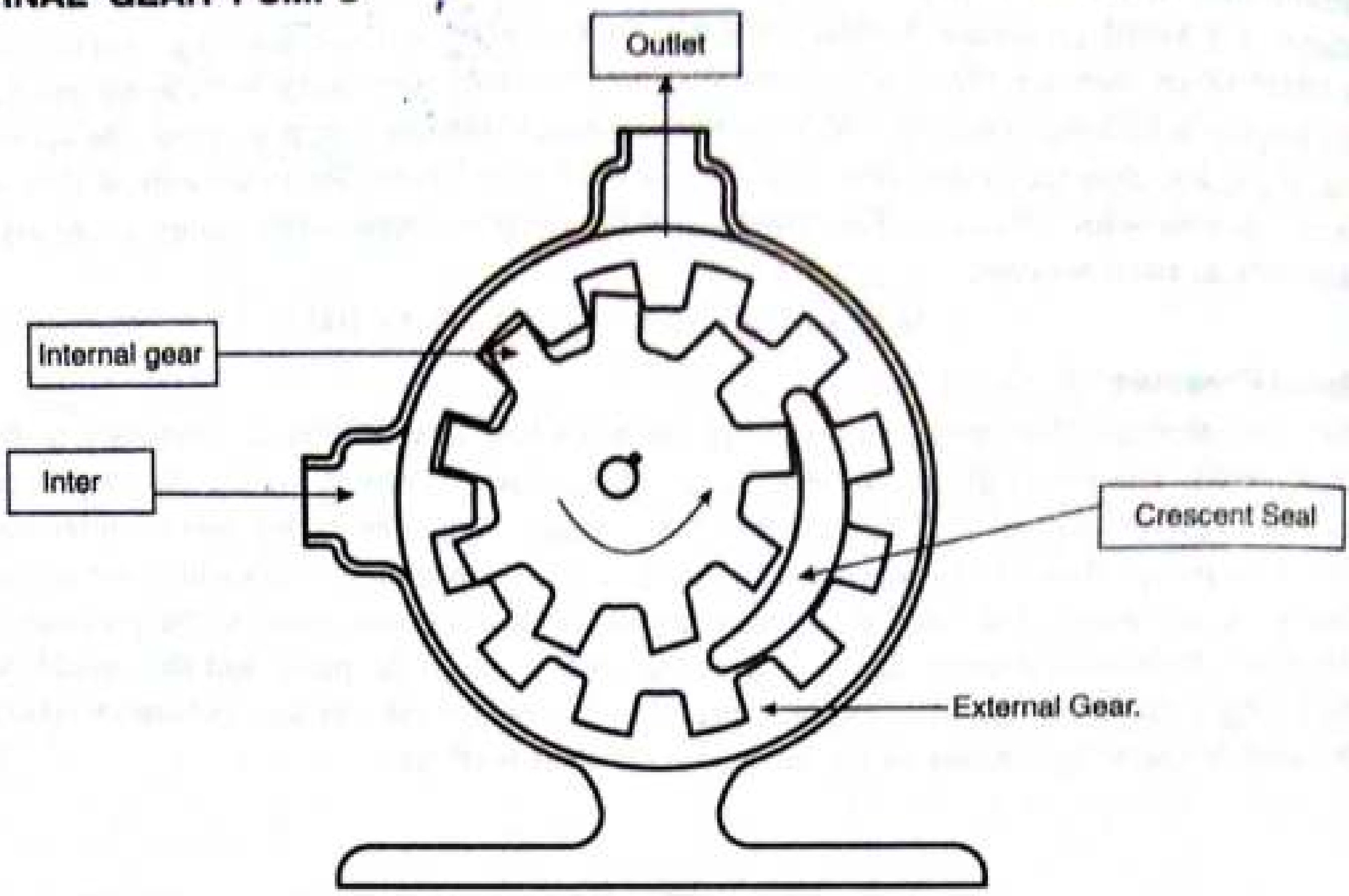
The high pressures (discharge pressure) are created when a large load or resistance to flow is encountered. The actual discharge depends on the discharge pressure. As the discharge pressure increases, the amount of internal leakage also increases, thus decreasing pump outlet volume. Hence the pumps should be operated at pressures less than the pressure, which will cause mechanical damage to the pump. The rated pressure of a positive displacement pump is the pressure below which no mechanical damage due to overpressure will occur to the pump and this would help to have long reliable service life. Too high pressures not only results in less volumetric efficiency but also damages the housing of the pump and the shaft bearings.



EXTERNAL GEAR PUMPS

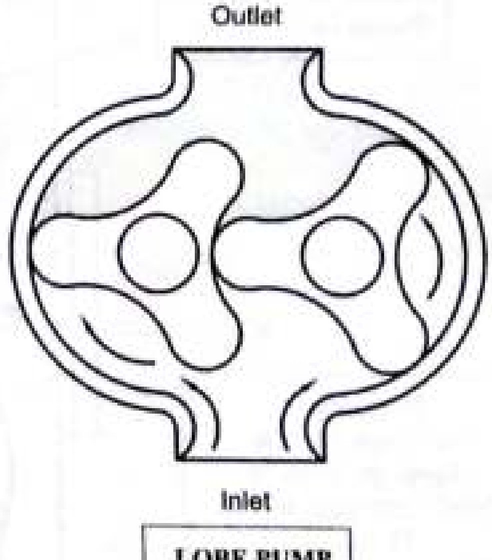
11145

INTERNAL GEAR PUMPS



The Internal gear and external gear rotate Anticlockwise

In this type of gear pump both the gears rotate in the same direction. As the gears move away near the inlet, the oil is trapped in the gear space and travels around the crescent seal and near the inlet as the two gears come closer pressure rises and the oil is pushed into the outlet port.



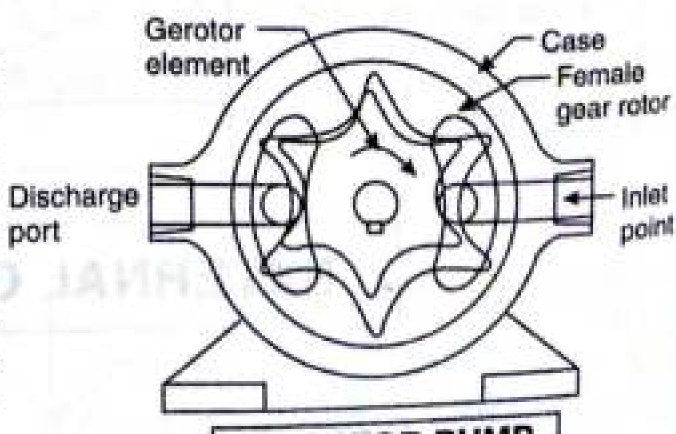
LOBE PUMP

LOBE PUMP

These are similar in operation to external gear pump. Instead they have very few rotating members and both rotating members are driven externally. Their operation is quite and as there are only limited mating elements their flow has pulsations. But their volumetric displacement is generally more than other gear pumps.

GEROTOR PUMPS

These pumps operate similar to internal gear pumps. The inner gear rotor is power driven and draws the outer gear rotor around as they mesh together. This forms inlet and discharging pumping chambers between the rotor lobes. The tips of rotors make contact to seal the pumping chambers from each other. The inner gear has one teeth less than the outer gear and the volumetric displacement is determined by the space formed by the extra tooth in the outer rotor.



GEROTOR PUMP

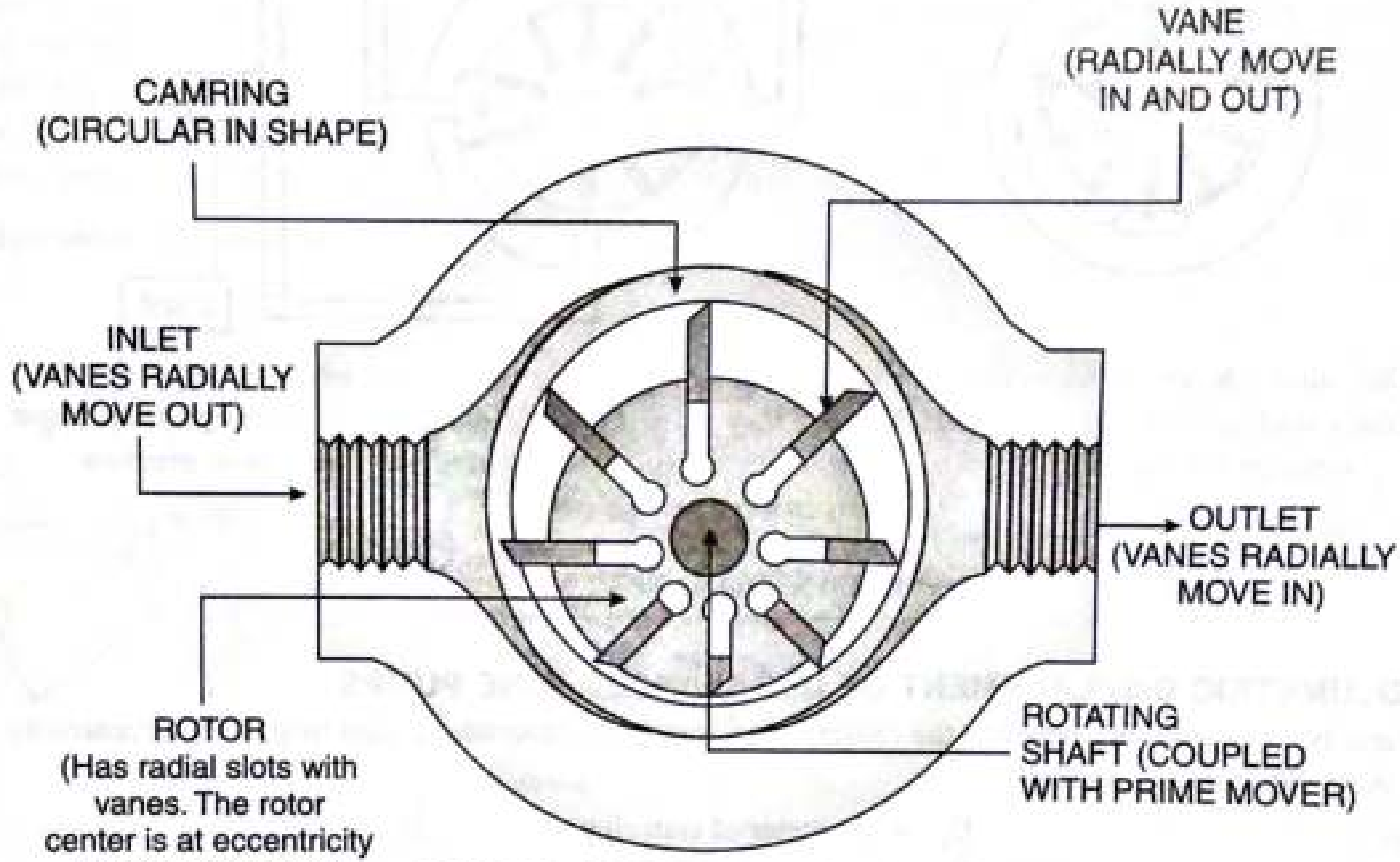
VANE PUMPS

There are two types of vane pump. They are

- Unbalanced Vane pump
- Balanced Vane pump

UNBALANCED VANE PUMP

The unbalanced vane pumps have a rotor connected to the rotating shaft which is coupled with the prime mover. The rotor has radial slots into which there are vanes that move in and out while rotating due to the centrifugal force. The cam ring is *circular* in shape, which limits the outward movement of the rotors. At inlet the vanes move out creating the suction of fluid and as they rotate the fluid travels entrapped between the radial vanes and the cam ring. Nearing the outlet the vanes are pushed in by the cam ring resulting in high pressure. This results in pushing or discharge of liquids out into the discharge line.



UNBALANCED VANE PUMP

Advantage:

The advantage of the unbalanced vane pump is that as the eccentricity between the cam ring and rotor is changed the volume of fluid pumped can be proportionately changed.

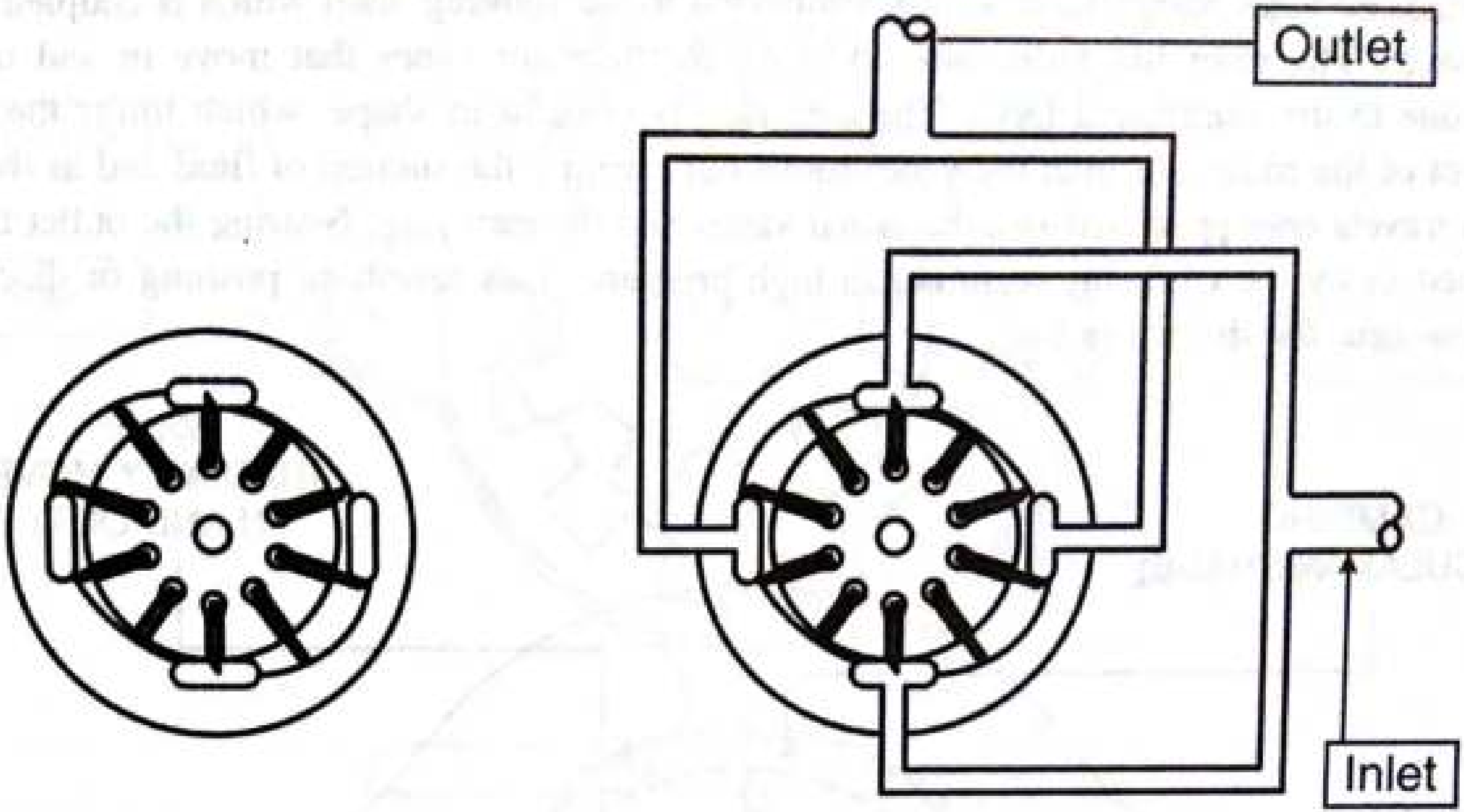
Disadvantage:

The suction side of the fluid (almost half of the mechanism) is at atmospheric pressure. But at the discharge end the fluid is at system pressure and as a result it imparts the side axial thrust on the rotor. This unbalanced force creates changes in the displacement volume and failure of the rotor.

BALANCED VANE PUMP:

These pumps are Constant Volume Positive displacement pumps.

The disadvantage of the unbalanced vane pump is that it experiences axial thrust. Changing the shape of the cam ring elliptical instead of circular can eliminate side thrust. In addition if there are two suction and discharge ports placed at equal and opposite quadrants, the two pressure or discharge ports cancel out their forces and thus side thrust is eliminated. The displacement of fluid and the basic operation of the pump are similar to the unbalanced pump except that there are two suction and pressure/ discharge ports. The pump remains a constant volume discharge pump and hence they cannot be used as a variable discharge pump. But most of the industrial applications use only constant volume positive displacement balanced vane pumps.



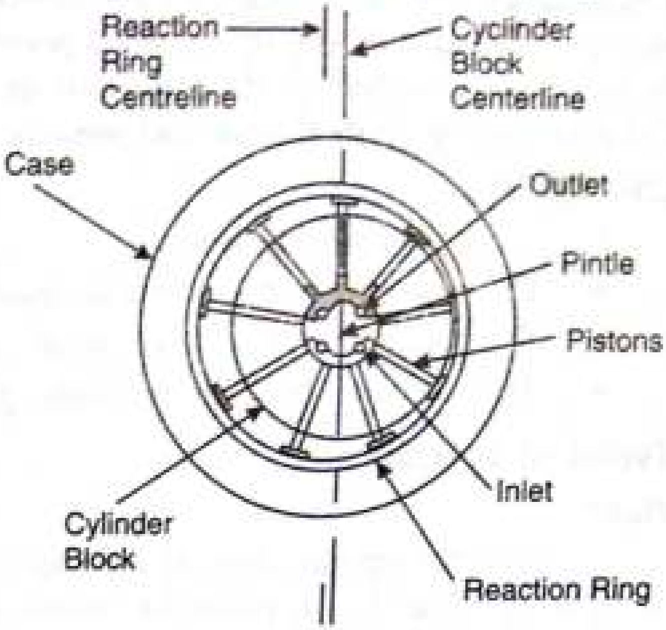
The pump shown above rotates anti-clockwise. As the vanes move out near the inlet, suction occurs and as they move in near the inlet they force the fluid out. The positioning of inlet and outlet ports radially opposite protects the pressure surges and thus the name pressure compensated vane pump

BALANCED VANE PUMP

RADIAL PISTON PUMPS

The pintle directs the fluid in and out of the cylinders. In this type the piston remain in contact with the reaction ring due to centrifugal force and backpressure on the pistons. To provide pumping action, the reaction ring moves eccentrically with respect to the pintle or shaft axis. As cylinder barrel rotates, the pistons on one side move out. This draws fluid as each cylinder passes the suction port. When the piston moves to maximum eccentricity, it is forced back by the reaction ring. This forces the fluid into the discharge port of the pintle.

Operation of a radial piston pump



RADIAL PISTON PUMPS

Fixed Vs Variable Pumps:

There are two types of positive displacement pumps. Based on the volume displaced per revolution they are termed as

(*i*) Fixed displacement pumps and

(*ii*) Variable displacement pumps

In *fixed displacement pumps*, the output per revolution of the rotor remains constant independent of the speed. As the speed of the prime mover is increased the output rate of this pump can be increased. Ex. Gear pump.

In *variable displacement pumps* the output per revolution can be varied although the rpm of the prime mover remains constant. They can be established by

(*i*) Varying the angle of the swash plate in the piston pumps or

(*ii*) Increasing the eccentricity in the vane pumps.

Increasing the rpm of the prime mover can increase the output in variable displacement pumps.

Pressure Compensated Pumps

Reducing the flow into the line can always compensate increase in Pressure. Only variable displacement pumps are pressure compensated pumps (Vane and Piston pumps). Gear pumps are non pressure compensated pumps. The pressure compensation in a piston pump is achieved as follows:

When the pressure is increased in the discharge line it would affect the pump. In pressure compensated piston pump the increase in pressure is felt at the end of the stroke and the flow is pressurized and diverted through the compensator spool upwards and enters the destroking piston and the swash plate is made vertical. This reduces the suction and discharge, which result in reduced pressure at the outlet.

Problem 2.1. *A hydraulic press has a ram of 30 cm diameter and a plunger of 4.5 cm diameter. Find the weight lifted by the hydraulic press when the force applied at the plunger is 500 N.*

Sol. Given :

Dia. of ram, $D = 30 \text{ cm} = 0.3 \text{ m}$

Dia. of plunger, $d = 4.5 \text{ cm} = 0.045 \text{ m}$

Force on plunger, $F = 500 \text{ N}$

Find weight lifted $= W$

Area of ram, $A = \frac{\pi}{4} D^2 = \frac{\pi}{4} (0.3)^2 = 0.07068 \text{ m}^2$

Area of plunger, $a = \frac{\pi}{4} d^2 = \frac{\pi}{4} (0.045)^2 = .00159 \text{ m}^2$

Pressure intensity due to plunger

$$= \frac{\text{Force on plunger}}{\text{Area of plunger}} = \frac{F}{a} = \frac{500}{.00159} \text{ N/m}^2.$$

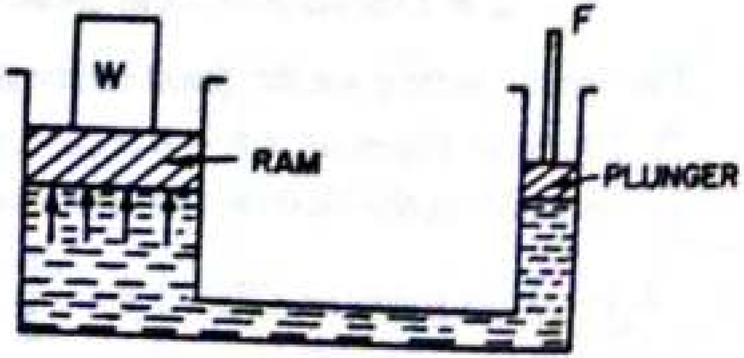


Fig. 2.3

Due to Pascal's law, the intensity of pressure will be equally transmitted in all directions. Hence the pressure intensity at the ram

$$= \frac{500}{.00159} = 314465.4 \text{ N/m}^2$$

But pressure intensity at ram

$$= \frac{\text{Weight}}{\text{Area of ram}} = \frac{W}{A} = \frac{W}{.07068} \text{ N/m}^2$$

$$\therefore \quad \frac{W}{.07068} = 314465.4$$

$\therefore$ Weight $= 314465.4 \times .07068 = \mathbf{22222 \text{ N} = 22.222 \text{ kN}}$. **Ans.**

INSTITUTE OF TEXTILE TECHNOLOGY CHOUDWAR

Hydraulic Machines & Industrial Fluid Power

Pressure Control Valves

Pressure Control

The primary concern in a hydraulic system is either:

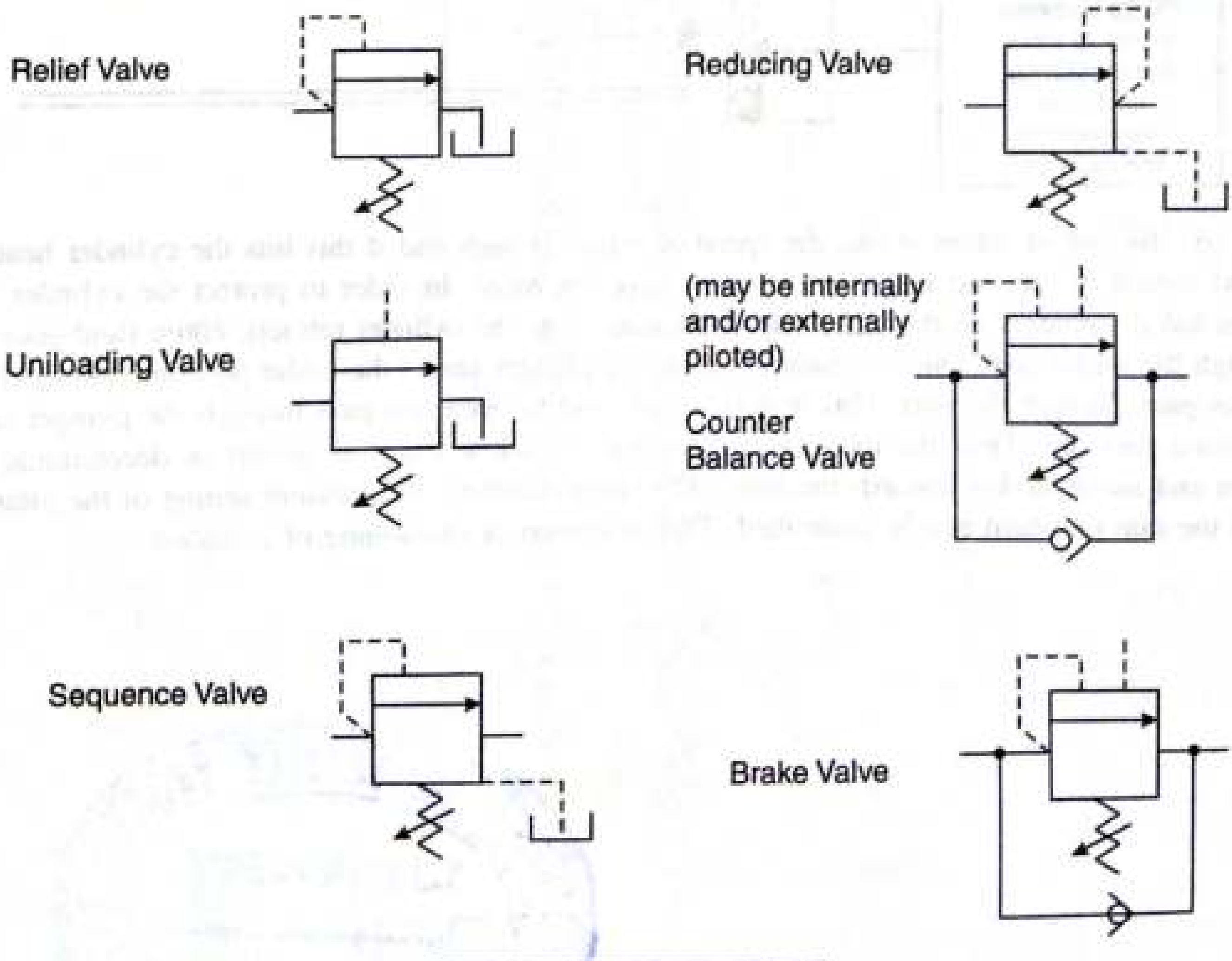
(*i*) Controlling the rate of flow. (*ii*) Controlling the pressure level.

The method of controlling the pressure by passing the fluid through the orifice or flow control device does not give good accuracy. To control precisely the level of pressure six different devices has been generally used. They are:

(*i*) Relief Valve (*ii*) Unloading valve (*iii*) Sequence valve

(*iv*) Reducing valve (*v*) Counterbalance valve (*vi*) Brake valve

They are shown with ANSI Symbols below. Their ANSI symbols resemble each other and only their position in the circuit differentiates their functionality.



Pressure Control Valves

RELIEF VALVES

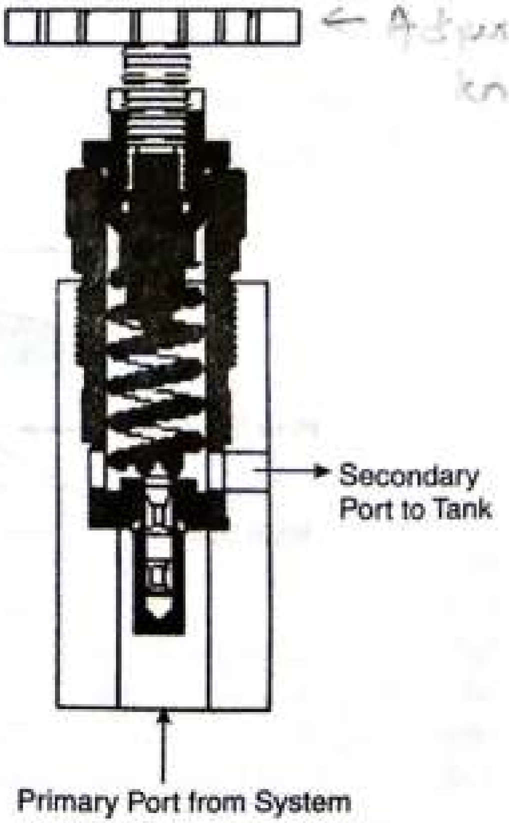
Relief valves are the most common type of pressure-control valves. The relief valves' function may vary, depending on a system's needs. They can provide overload protection for circuit components or limit the force or torque exerted by a linear actuator or rotary motor. The internal design of all relief valves is basically similar. The valves consist of two sections: a body section containing a piston that is retained on its seat by a spring(s), depending on the model, and a cover or pilot-valve section that hydraulically controls a body piston's movement. The adjusting screw adjusts this control within the range of the valves. Valves that provide emergency overload protection do not operate as often since other valve types are used to load and unload a pump. However, relief valves should be cleaned regularly by reducing their pressure adjustments to flush out any possible sludge deposits that may accumulate. Operating under reduced pressure will clean out sludge deposits and ensure that the valves operate properly after the pressure is adjusted to its prescribed setting.

SIMPLE TYPE (DARV)

The figure shows a simple-type relief valve. This valve is installed so that one port is connected to the pressure line or the inlet and the other port to the reservoir. The ball is held on its seat by thrust of the spring, which can be changed by turning the adjusting screw. When pressure at the valve's inlet is insufficient to overcome spring force, the ball remains on its seat and the valve is closed, preventing flow through it. When pressure at the valve's inlet exceeds the adjusted spring force, the ball is forced off its seat and the valve is opened. Liquid flows from the pressure line through the valve to the reservoir. This diversion of flow prevents further pressure increase in the pressure line. When pressure decreases below the valve's setting, the spring reseats the ball and the valve is again closed. The pressure at which a valve first begins to pass flow is the cracking pressure of a valve. The pressure at which a valve passes its full-rated capacity is the full-flow pressure of a valve. Because of spring rate, a full-flow pressure is higher than a cracking pressure. This condition is referred to as pressure override. A disadvantage of a simple-type relief valve is its relatively high-pressure override at its rated capacity.

This can be summarized as below:

- These are called direct acting relief valves (DARV).
- They are normally closed
- They have a primary and secondary port.
- The primary port is exposed to the system pressure and is blocked by the poppet.
- The secondary port is connected to the line that connects the tank or reservoir.
- The poppet lies in the intersectional passage between the primary and secondary ports.
- The poppet opens and closes at the predetermined pressure levels that can be set by the compression spring that seat pushes the poppet down and close the primary port.
- The adjustable knob on the top of the valve can vary the compression in the spring.
- Direct Acting Pressure Relief Valves are the one in which the poppet is closed by the direct action of the mechanical spring force that oppose the excess fluid pressure.

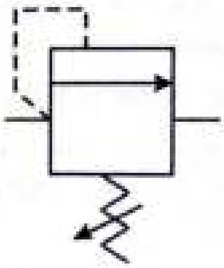


← Adjustable knob

Direct Acting Relief Valve

- When the system pressure reaches the spring force, it opens and allows half the fluid to pass to the tank, which is termed as *Cracking Pressure*.
- The cracking pressure can be adjusted by increasing the spring force.
- The application of the direct acting relief valves are limited due to the difficulty in designing strong compression spring that can be used for long period of time.

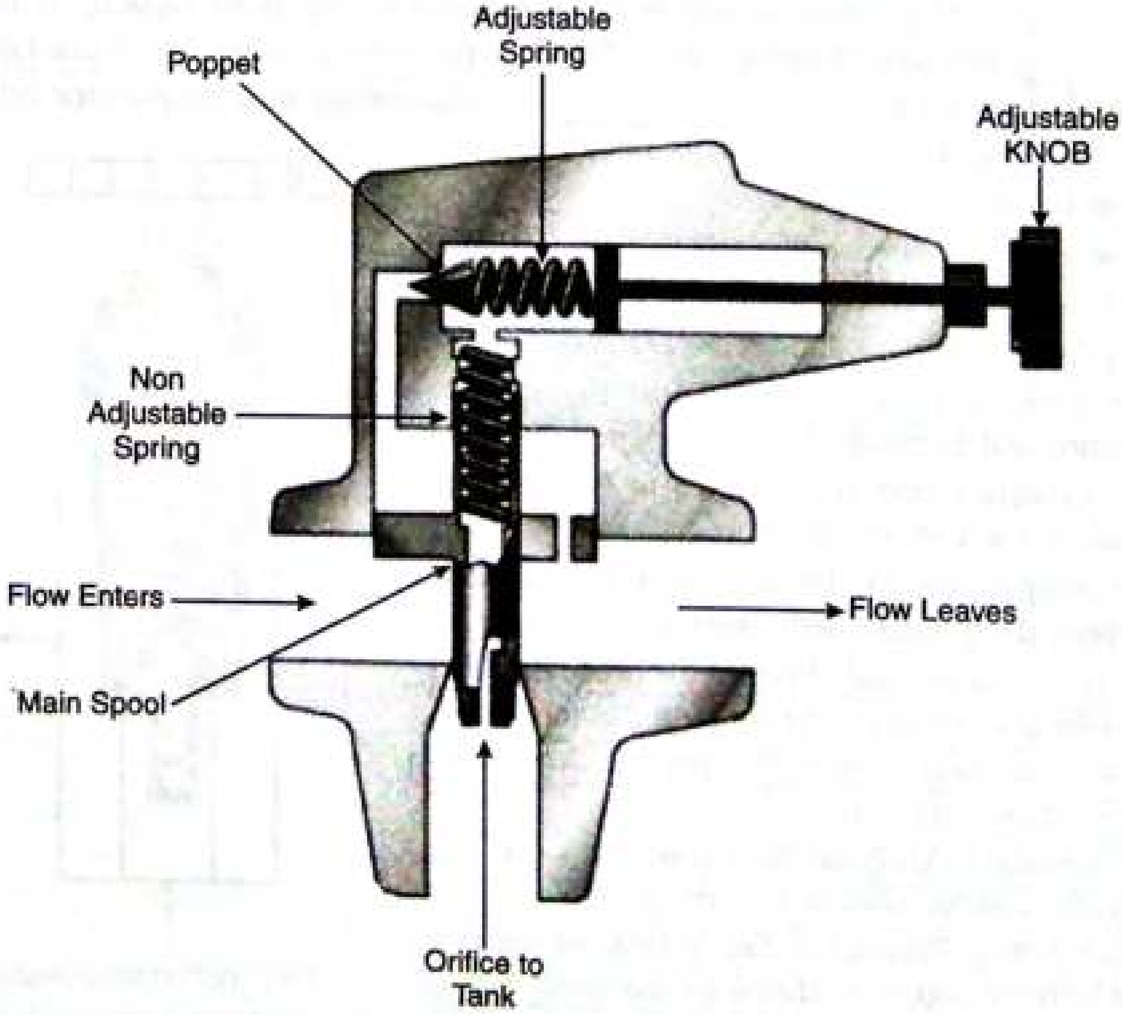
ANSI Symbol



Relief Valve

PILOT OPERATED (COMPOUND) RELIEF VALVE

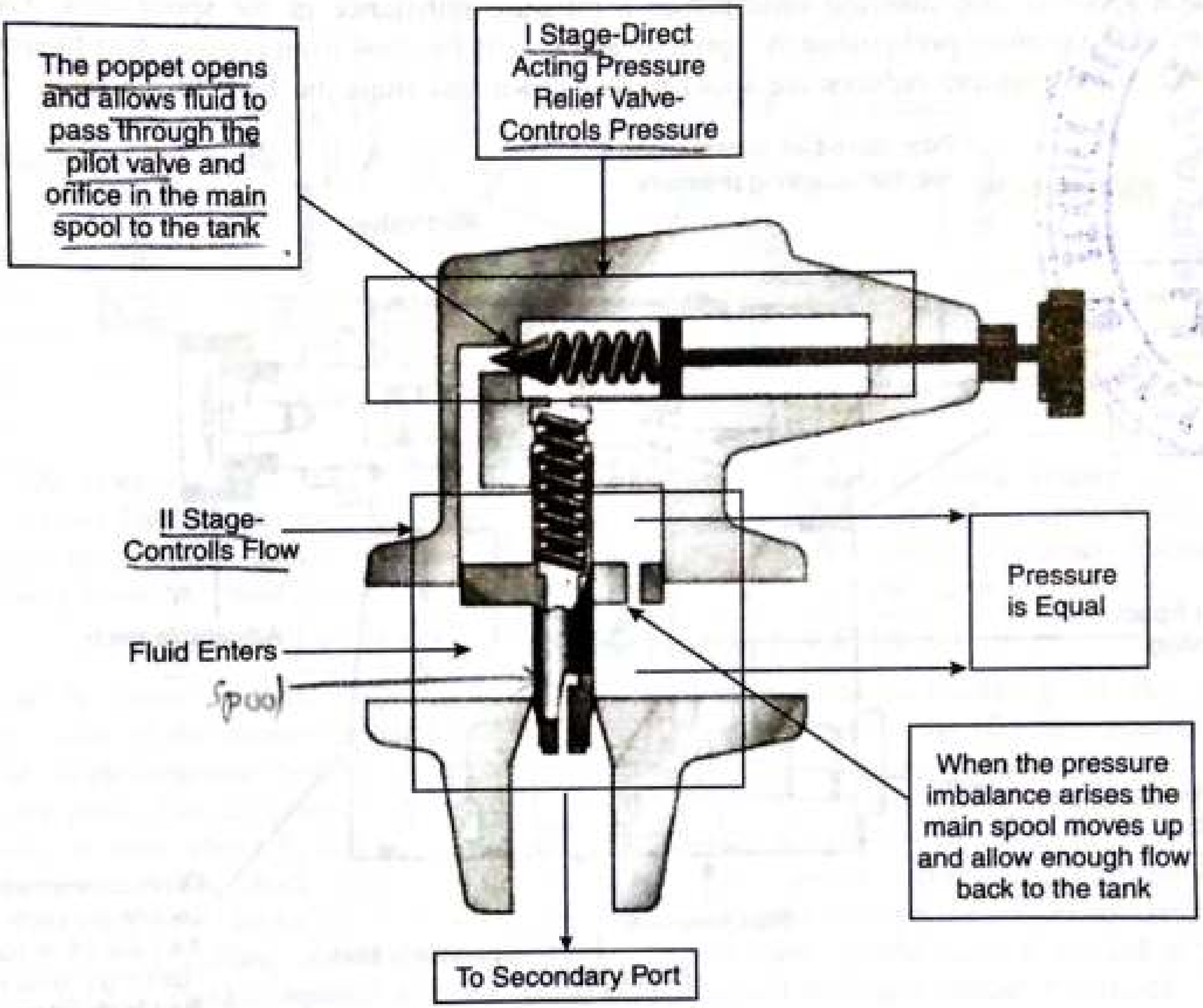
- Pilot Operated Relief Valves or Compound valves are designed to accommodate high pressure and high flow capacity with same frame size as the direct acting valves.
- These valves are built in two stages.
- The first stage is the same as direct acting relief valve and controls pressure. They have a poppet, spring and knob that can adjust the pressure level.
- The second stage has a main spool that is held normally closed by light non-adjustable spring. They are large enough to handle maximum flow rating and controls flow.



Pilot Operated Relief Valve

They function and control the pressure as follows:

- As long as the system pressure is less than the leaving pressure set on the control knob, pressure on the main spring line and above the spool are the same. Once the system pressure increases, the pilot relief poppet opens and a restricted flow of the fluid above the spool is diverted through the pilot valve and the orifice in the main spool back to the tank. The flow of fluid above the spool through the orifice in the main spool creates the pressure imbalance between the pump line and the area above the spool. This result in moving up the main spool and diverting enough flow of the fluid to the tank. Once the pressure decreases, the spool retracts back to its balanced position.



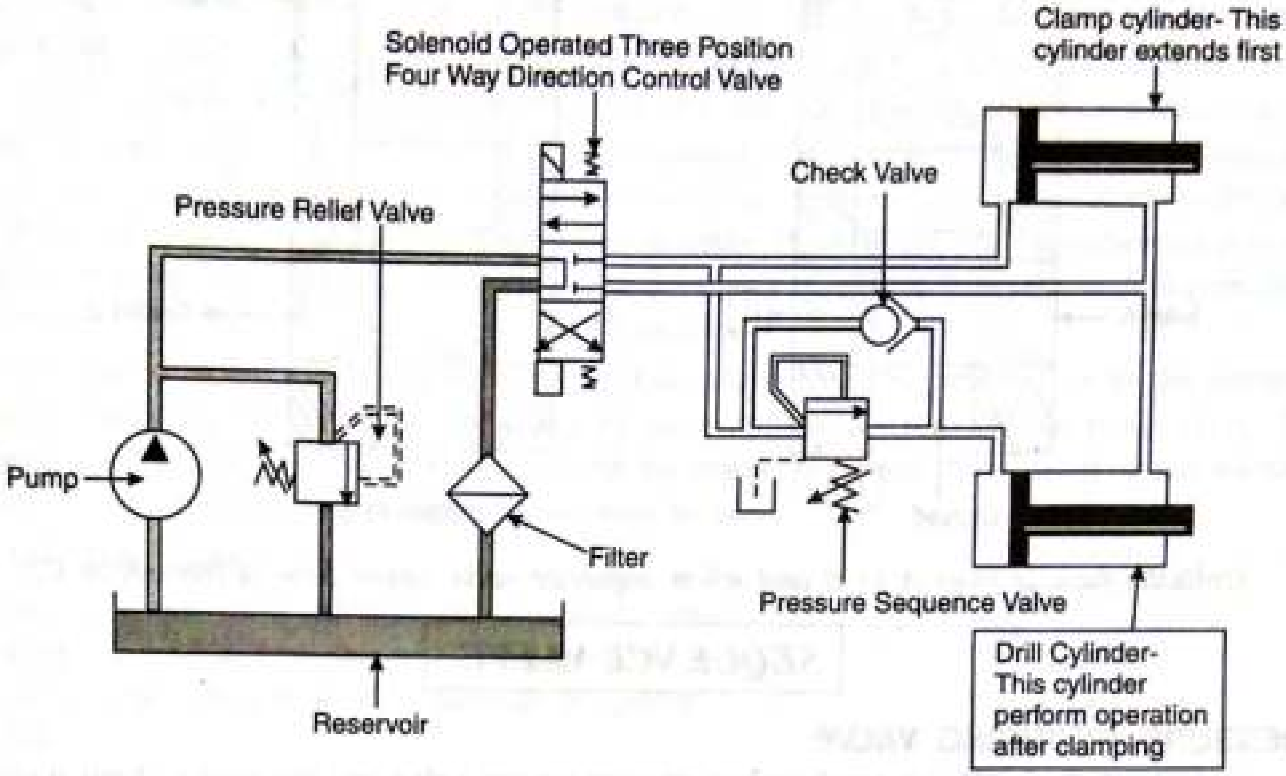
Operation of Pilot Operated Relief Valve

11, 145-

PRESSURE SEQUENCE VALVE

A Sequence valve is a normally closed pressure control valve. They are similar in construction to the poppet relief valve. Their position is the one that differentiate its functionality. If the system pressure reaches the maximum and the flow through the primary port of the valve is diverted through the secondary port to the tank, it is named as *Relief Valve*. These valves are used to safeguard the system from high pressure.

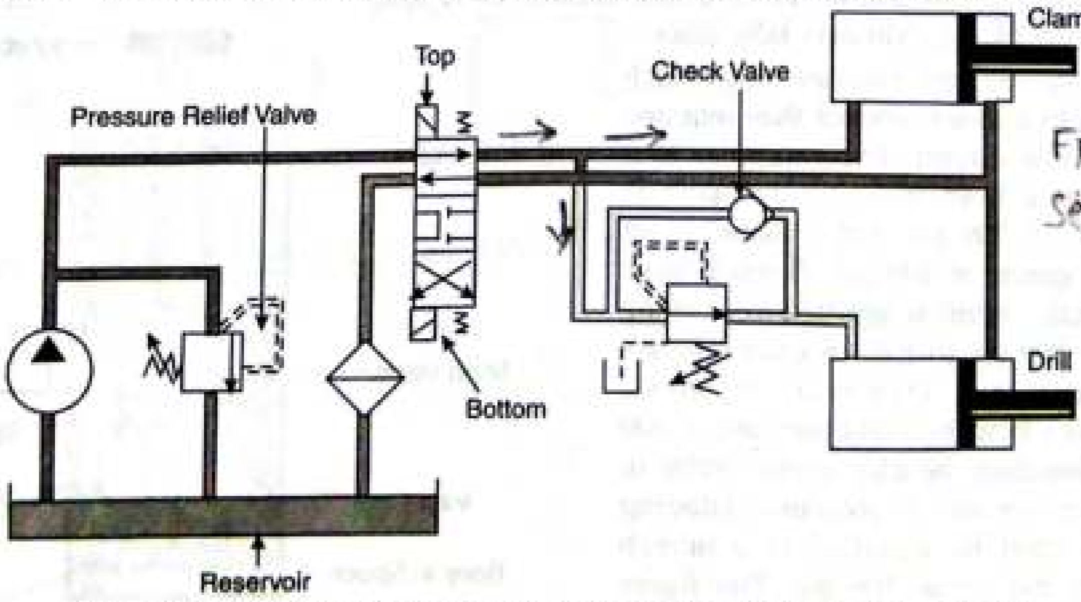
Alternatively, if **the end of one operation indicated by an increase in pressure results in diverting the flow from the primary port of the poppet valve to the succeeding operation**, these valves are termed as *Sequence valves* and they are positioned between the two actuators. Their spring setting is higher than the pressure level required for the first operation but less than the system pressure. These valves are used when two actuators are operated in the single position of the direction control valve. The clamp and drill circuit given below can best explain these valves.



The pressure relief valve setting is higher than the sequence valve setting. The relief valve drains back to the tank if it reaches system pressure (at the end of all operations or any increase in pressure during the operations) while the sequence valve performs. Succeeding operation from the fluid that comes from the secondary port of the valve.

APPLICATION OF SEQUENCE VALVE

In the above example, the fluid from the pump is diverted to the clamping cylinder and the primary port of the sequence valve in the top position (explained in the direction control valve chapter) of the solenoid operated direction control valve. The clamp cylinder extends first and clamps the work piece. Let the pressure required to hold the job be P_1. At the end of clamping operation, the pressure rises above P_1 and the pressure sequence valve diverts the flow from the pump to the drilling actuator. The check valve kept in parallel with the sequence valve during the extension of the drilling actuator blocks the flow and allows the fluid to pass through the sequence valve alone. At the end of the drilling operation, the position of the direction control valve is shifted to bottom position and the fluid is equally diverted to the clamping and drilling actuators to retract.

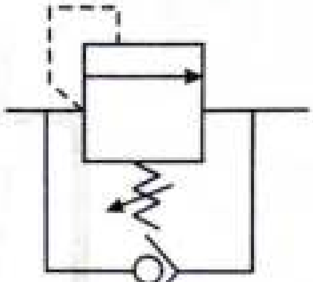


First = Clamping
Second = Drilling

(Note the position of the Pressure Sequence Valve in the circuit)

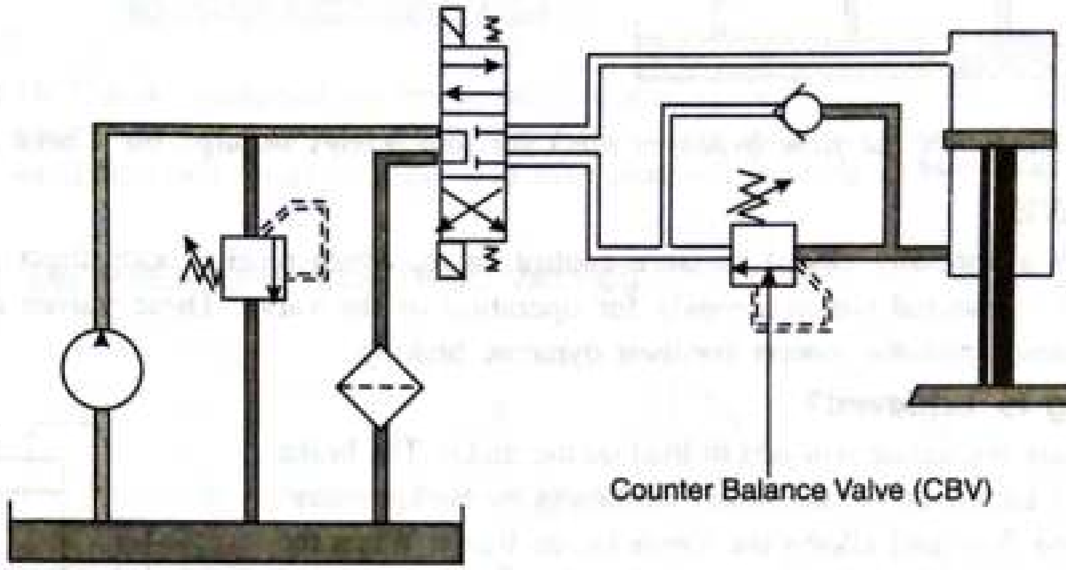
COUNTER BALANCE VALVE (CBV)

A counterbalance valve is a normally closed pressure valve used mostly in cylinders that are held vertical. These valves are positioned on the outlet of the rod side flow to counter the free extension of the cylinder. Also they are used to protect the overrunning loads. These valves are set to pressure more than the system load pressure, and if the cylinder has to extend, the pressure has to rise and open the counter balance valve and then only the cylinder extends.

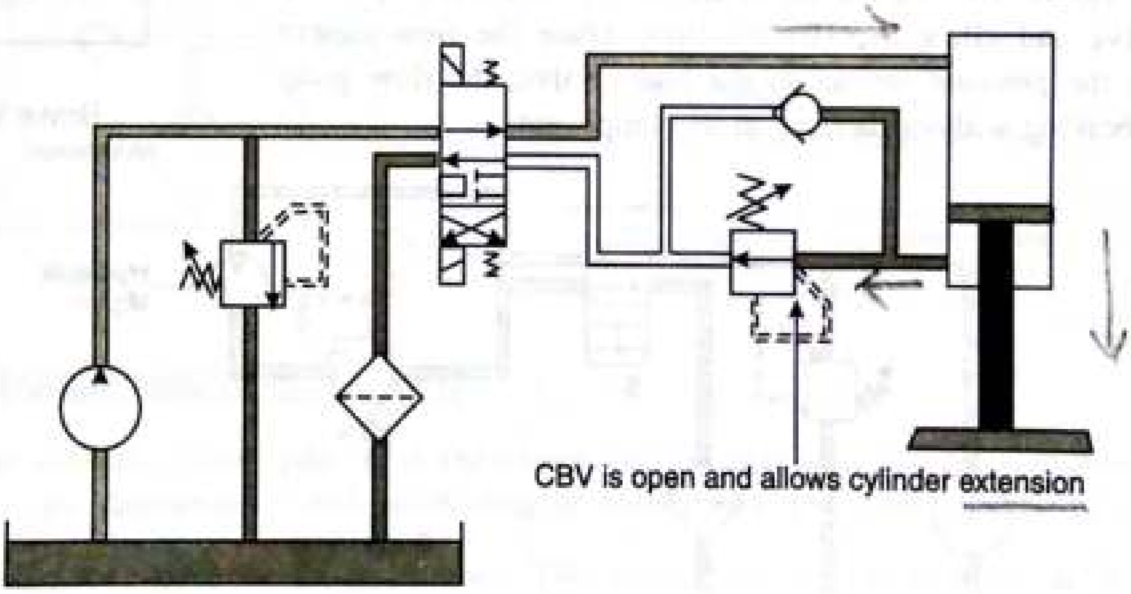


Counter Balance Valve

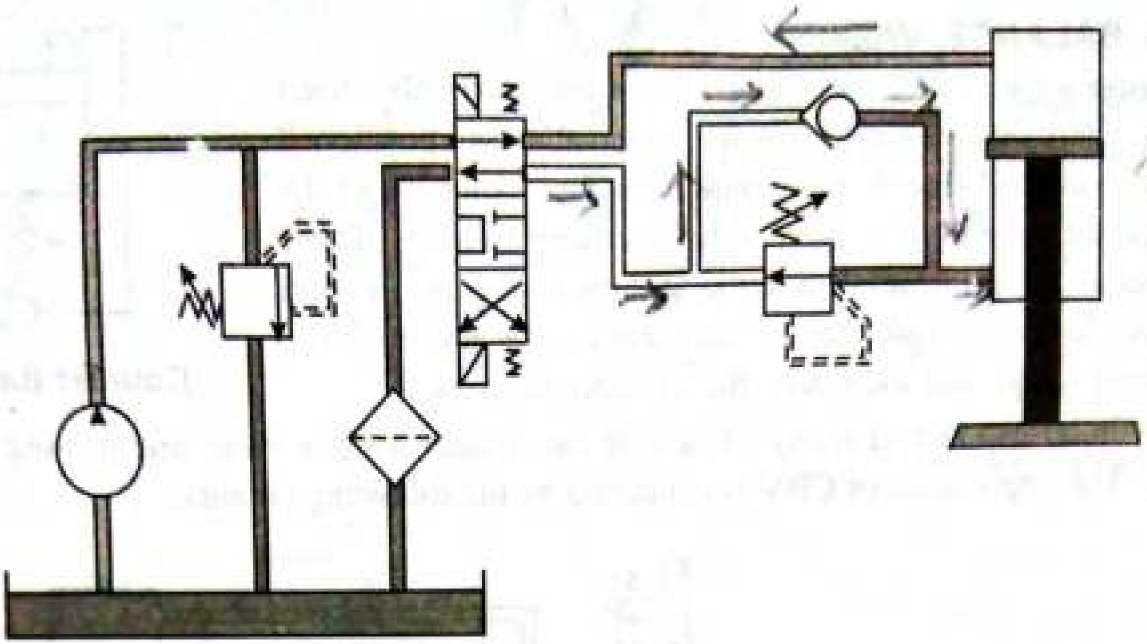
The cylinder can retract freely bypassing the counter balance valve and passing through the check valve. The application of CBV is explained by the following circuits.



The counter balance valve is closed and balances the cylinder weight



During extension the pressure set valve (more than load pressure) of the CBV is reached and the CBV opens to allow the extension of the cylinder



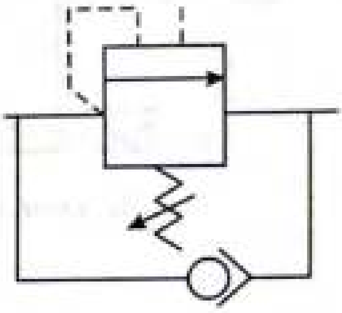
During retraction the flow bypasses the CBV and passes through the Check Valve

BRAKE VALVE

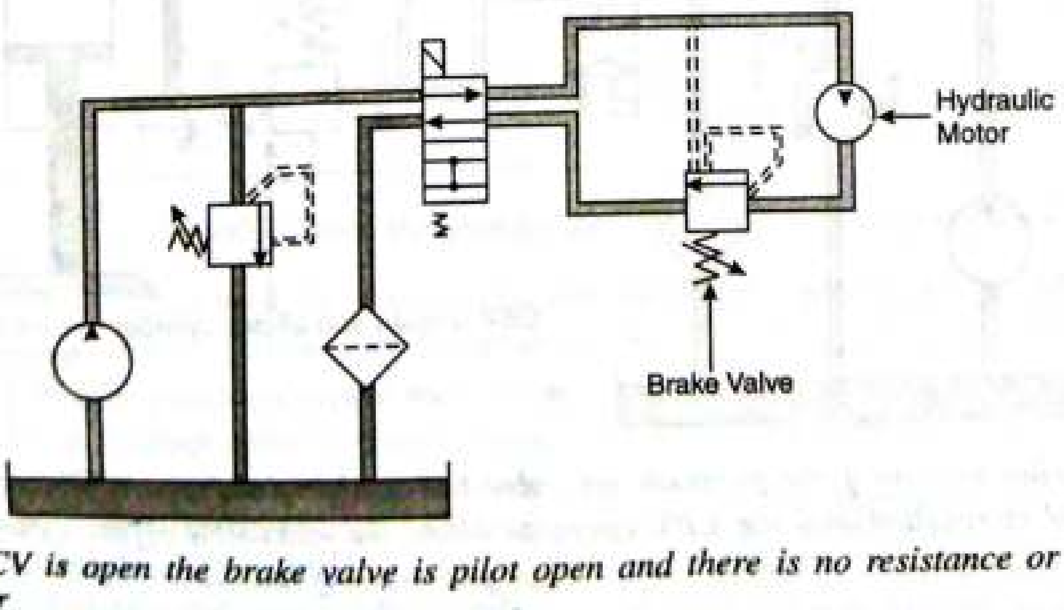
Brake valve is a normally closed pressure control valve, which receive both direct and remote pilots and are connected simultaneously for operation of the valve. These valves are used in circuits that have hydraulic motors for their dynamic braking.

How braking is achieved?

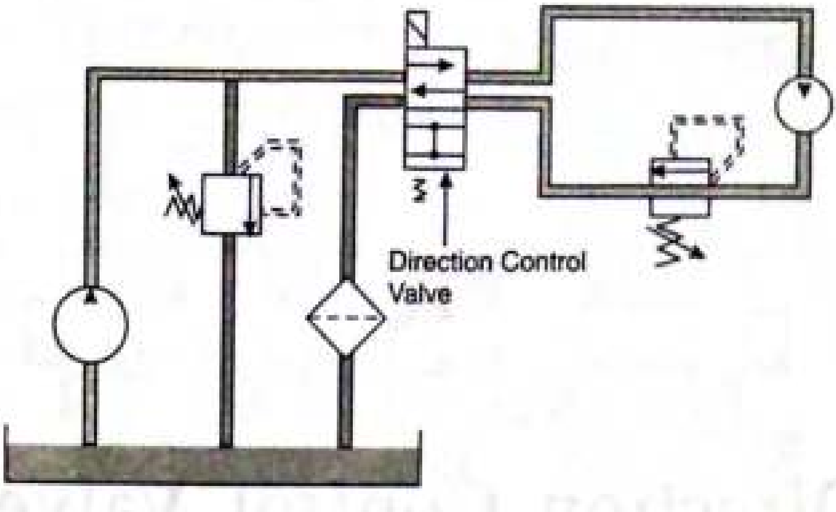
Any downstream resistance will add to load on the motor. The brake valve is usually kept open (which justify eliminates the backpressure) by the pilot line flow and allows the motor to run freely. When the direction control valve is de energized the pilot pressure is lost and the valve closes. But the inertia of the motor will drive open the brake valve and allow a restricted flow. Once the flow cannot overcome the pressure setting of the brake valve, the flow gives dynamic braking without damage to its components.



Brake Valve

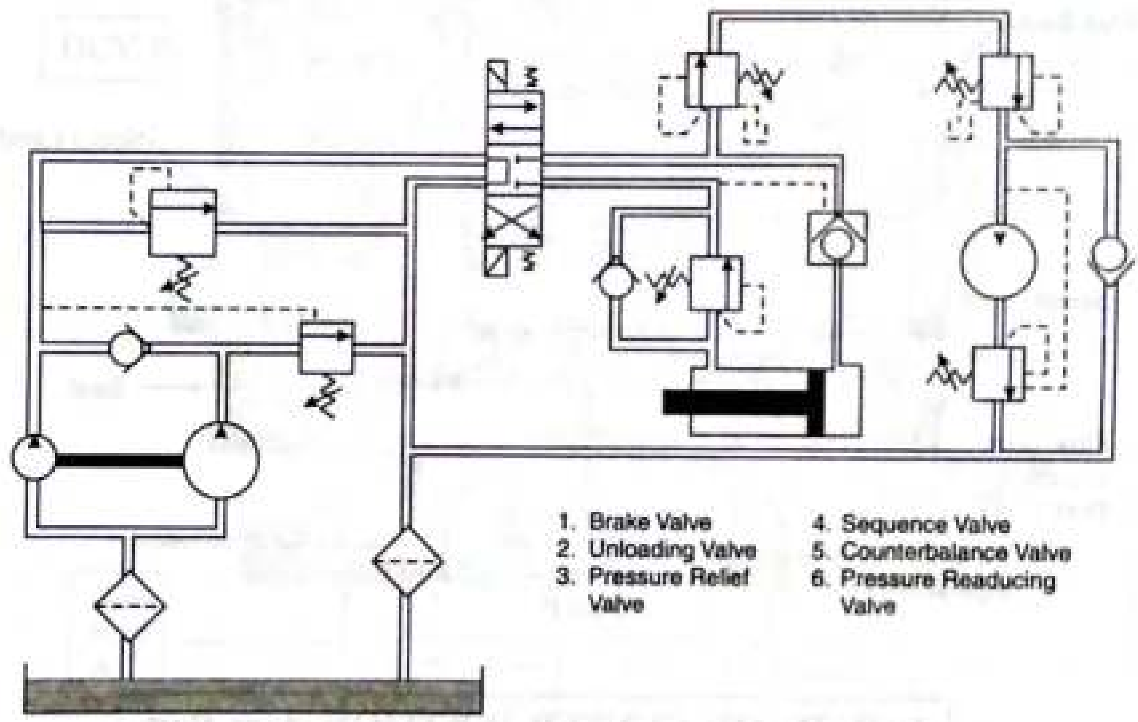


When the DCV is open the brake valve is pilot open and there is no resistance or backpressure on the motor



When the DCV is de energized the brake valve is pilot open by the inertial rotation of the motor and when it cannot overcome the spring force of the valve the spool of the brake valve oscillates and tends to close and give dynamic braking to the motor

EXERCISE ON PRESSURE CONTROL VALVES

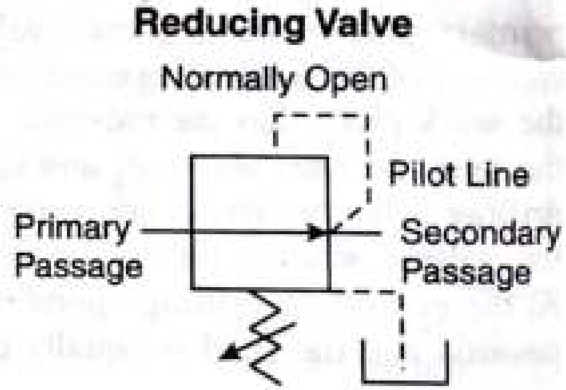


Identify the different valves placed in the above circuit. This would help in understanding the functionality and positioning of valves in a hydraulic system.

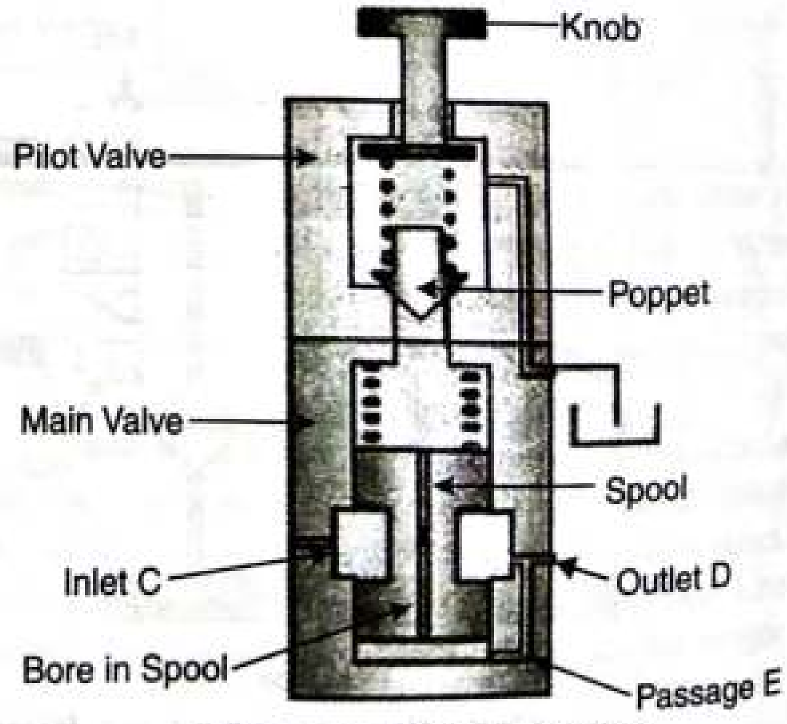
The lines inside the values should be trimmed. The pipeline should start and stop at the valves.

PRESSURE REDUCING VALVE

Pressure reducing valve is a normally open pressure control valve and are used to limit pressure. In this valve the primary port is connected to the secondary port. A pilot line is taken from the bottom of the spool. When there is an increase in pressure due to resistance to flow in the line, the flow is diverted through the pilot line. This moves the spool and blocks excess flow into the system. By reducing flow into the system, the resistance to flow is reduced and thus pressure is reduced. The use of pressure reducing valve in the system helps to balance any increase in pressure at any time. This is the only pressure control valve that is normally open. The significance of this valve can be explained as follows:



- When there are two clamping cylinders used with relatively difference in the requirement of clamping forces, the actuator that requires less pressure closes as and when the clamping force is reached and does not allow any excess flow into the actuator. The flow to the higher rated clamping cylinder continues and when both the cylinders reach their pressure rating, the flow is reversed by changing the position of the direction control valve and retraction of the cylinders take place.
- These valves limit pressure on a branch circuit to a lesser amount than required in a main circuit. For example, in a system, a branch-circuit pressure is limited to 300 psi, but a main circuit must operate at 800 psi. A relief valve in a main circuit is adjusted to a setting above 800 psi to meet a main circuit's requirements. However, it would surpass a branch-circuit pressure of 300 psi. Therefore, besides a relief valve in a main circuit, a pressure-reducing valve must be installed in a branch circuit and set at 300 psi. The figure below shows a pressure-reducing valve.



Pressure Reducing Valve

In a pressure-reducing valve, adjusting the spring's compression obtains the maximum branch-circuit pressure. The spring also holds spool in the open position. Liquid from the main circuit enters the valve at the inlet port C, flows past the valve spool, and enters the branch circuit through the outlet port D. Pressure at the outlet port acts through the passage E to the bottom of spool. If the pressure is insufficient to overcome the thrust of the spring, the valve remains open.

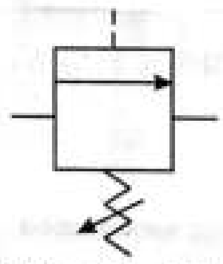
If the pressure at the outlet port and under the spool exceeds the equivalent thrust of the spring, the spool rises and the valve is partially closed. This increases the valve's resistance to flow, creates a greater pressure drop through the valve, and reduces the pressure at the outlet port. The spool will position itself to limit maximum pressure at the outlet port regardless of pressure fluctuations at the inlet port, as long as workload does not cause back flow at the outlet port. Back flow would close the valve and pressure would increase.

During the period the actuator is protected from the increase in pressure rise by the pressure-reducing valve, the blocked fluid delivered by the pump passes through the relief valve. The reducing valve protects only the actuator and not the pump. To protect the pump from any increase or raise in system pressure Unloading Valves may be used.

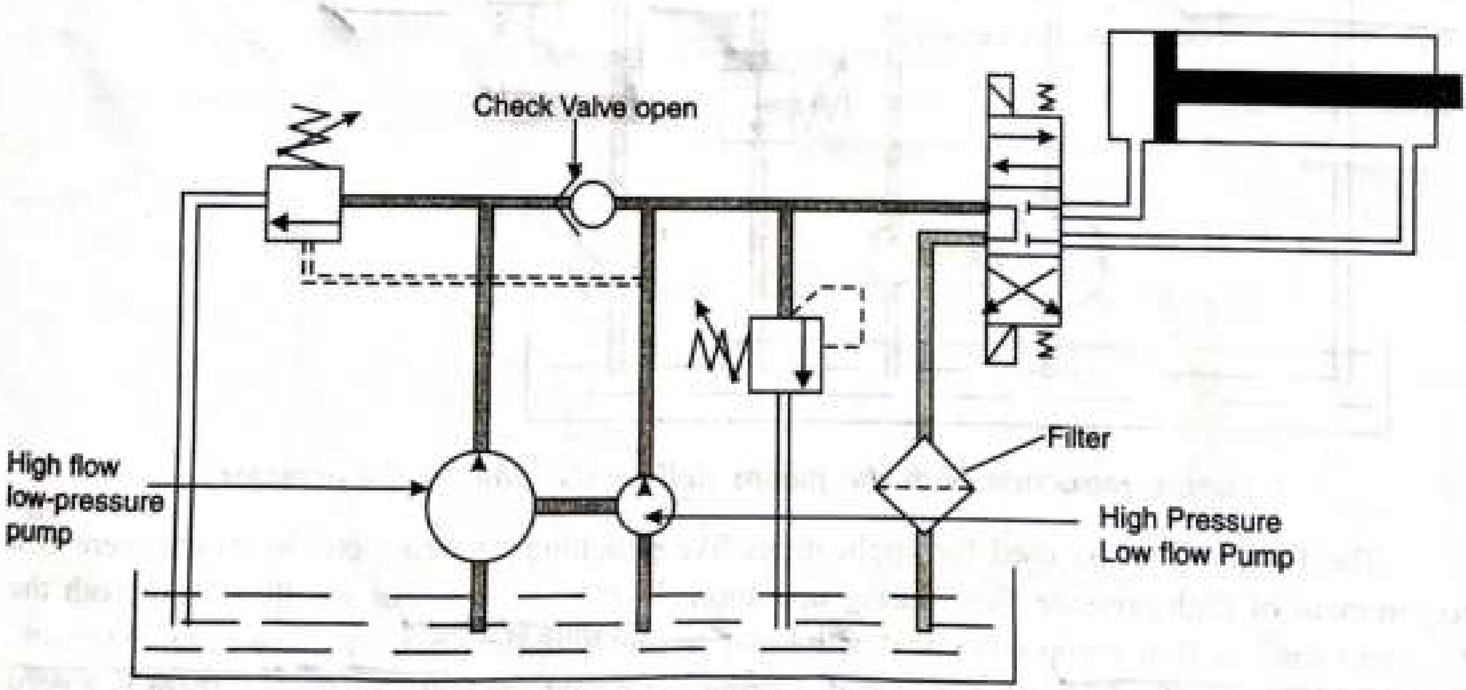
UNLOADING VALVE

An Unloading valve is a remotely pilot operated normally closed pressure valve that directs the flow from the pump back to the tank when there is an increase in system pressure.

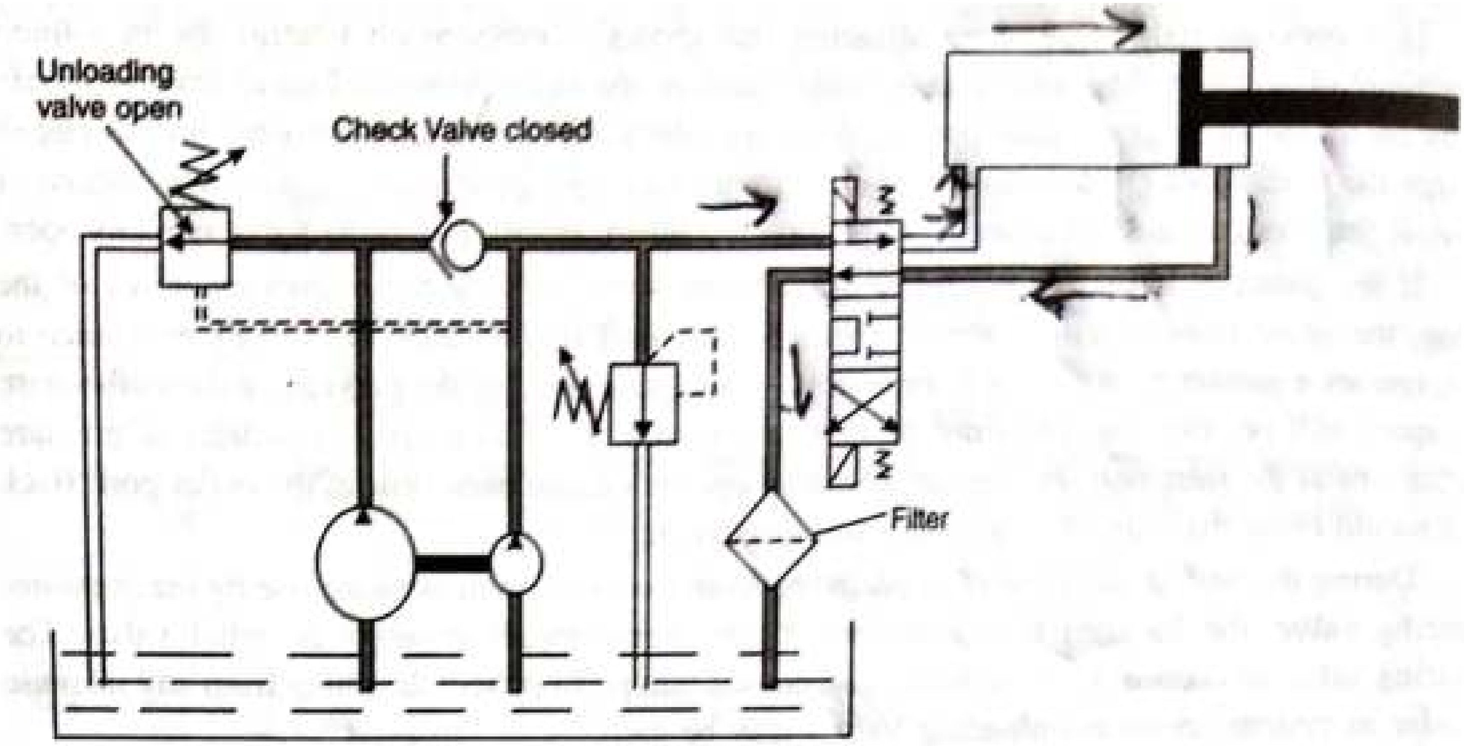
The application of the unloading valve can best explained by the below circuit.



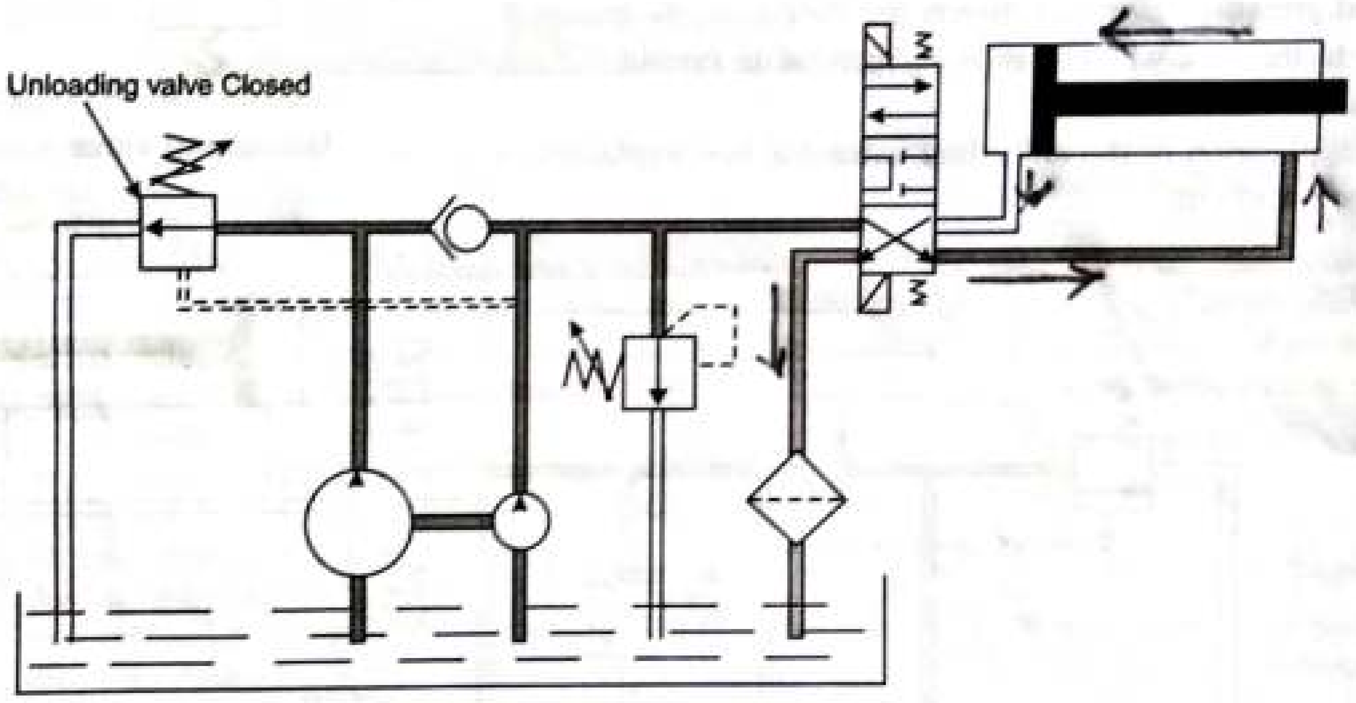
Unloading Valve



Double Pump (Hi-Lo) Hydraulic circuit



When cylinder extends, both pump work and when resistance occurs, the low-pressure pump is unloaded at less pressure to the tank.



During retraction both the pumps deliver the flow to the actuator

The Hi-Lo circuit is used for applications like punching, shearing etc. in which there is a requirement of high-pressure flow during operation. In the initial stages the flow from both the high-pressure low flow pump is coupled with low-pressure high flow pump to have faster extension. In this stage, the unloading valve is closed. During the actual punching operation, there is a need for high-pressure flow for relatively small extension of the cylinder. This can be well managed by the discharge from high-pressure pump. In this stage, if the pressure increase is felt on low pressure pump it would result in failure. This is managed by the opening of the unloading valve that discharges the low-pressure pump at minimum pressure, without any increase in pressure up to relief valve setting thus protecting the pump. When the return stroke occurs, the unloading valve closes and once again the pump flows are coupled.

INSTITUTE OF TEXTILE TECHNOLOGY CHOUDWAR

Hydraulic Machines & Industrial Fluid Power

Flow Control Valves

Flow Control Valves

FLOW CONTROL VALVES

Flow control valves are used to reduce the rate of flow. The reduction in rate of flow results in reduction of speed and increase in pressure. Once the flow is blocked, allowing partial fraction to enter the system, the remaining fluid has to pass through the pressure relief valve setting. This results in increase in pressure and temperature of the system. If a pressure compensated pump is used then the pressure increase can be managed. The flow control valves are classified as

(i) Fixed or Non-adjustable

(ii) Adjustable

(iii) Throttling and

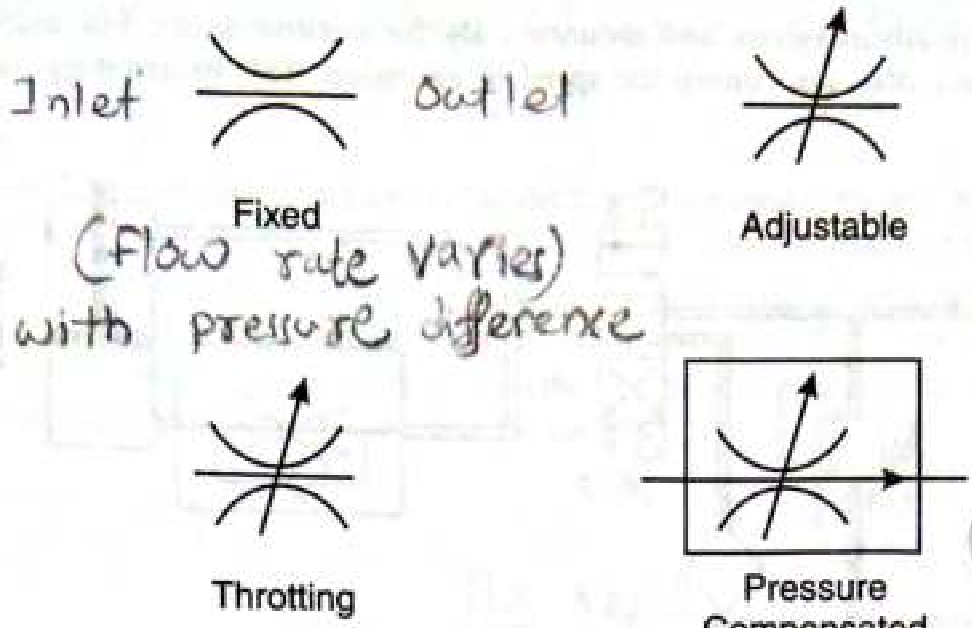
(iv) Pressure Compensated.

The flow through the valve remains constant as long as the pressure difference across the valve does not change. Once there an increase in pressure at the outlet the pressure difference reduces, thus allowing only restricted and reduced flow into the system.

Flow-control valves are used to control an actuator's speed by metering flow. Metering is measuring or regulating the flow rate to or from an actuator. A water faucet is an example of a flow-control valve. Flow rate varies as a faucet handle is turned clockwise or counterclockwise. In a closed position, flow stops. Many flow-control valves used in fluid-powered systems are similar in design and operation to a water faucet's. In hydraulic circuits, flow-control valves are generally used to control the speed of hydraulic motors and work spindles and the travel rates of tool heads or slides. Flow-control valves incorporate an integral pressure compensator, which causes the flow rate to remain substantially uniform regardless of changes in workload. A non-pressure, compensated flow control, such as a needle valve or fixed restriction, allows changes in the flow rate when pressure drop through it changes. Variations of the basic flow-control valves are the flow-control-and-check valves and the flow-control-and-overload relief valves. Models in the flow-control-and-check-valve series incorporate an integral check valve to allow reverse free flow. Models in the flow-control-and-overload-relief-valve series incorporate an integral relief valve to limit system pressure. Some of these valves are gasket-mounted, and some are panel-mounted.

Non-Pressure Compensated Pressure Control Valves

Gate valve, Globe valve, Needle valve, Check valve, Diaphragm valves and Butterfly valves are non-pressure compensated valves which are just used to control or limit the volume of flow entering the system. They do not deliver fixed volume into the system once there is any change in Pressure difference. But as most of the system is subjected to pressure variations, the application of non-pressure compensated flow control valves in hydraulic systems is limited. Thus more emphasis on these valves is not given.



(flow rate const
regardle
pressure differe

PRESSURE/ TEMPERATURE COMPENSATED FLOW CONTROL VALVE

Compensated Flow: The flow-control valves previously discussed do not compensate for changes in fluid temperature or pressure and are considered non-compensating valves. Flow rate through these valves can vary at a fixed setting if either the pressure or the fluid's temperature changes. Viscosity is the internal resistance of a fluid that can stop it from flowing. A liquid that flows easily has a high viscosity. Viscosity changes, which can result from temperature changes, can cause low variations through a valve. Such a valve can be used in liquid-powered systems where slight flow variations are not critical consideration factors. However, some systems require extremely accurate control of an actuating device. In such a system, a compensated flow-control valve is used. This valve automatically changes the adjustment or pressure drop across a restriction to provide a constant flow at a given setting. A valve meters a constant flow regardless of variation in system pressure. A compensated flow-control valve is used mainly to meter fluid flowing into a circuit; however, it can be used to meter fluid as it leaves a circuit. For clarity, this manual will refer to this valve as a flow regulator. The schematic representation of this value is included in the appendix.

FLOW CONTROL METHODS

The flow can be controlled either by one of the following methods.

(i) Meter-in circuit *(ii)* Meter-out circuit *(iii)* Bleed-off circuit

(I) METER-IN CIRCUIT

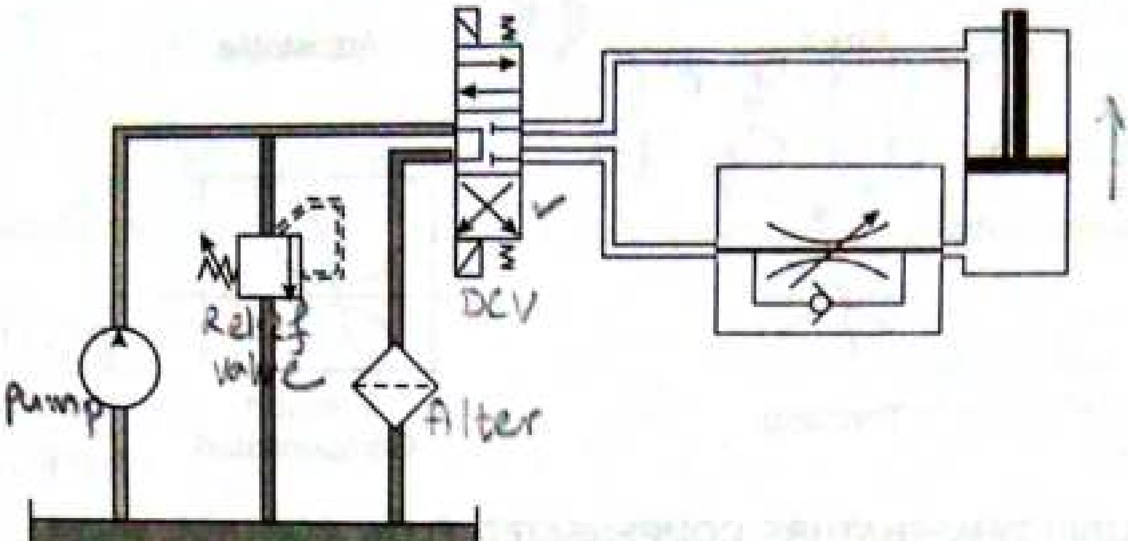
With this circuit, a flow-control valve is installed in a pressure line that leads to a work cylinder. All flow entering a work cylinder is first metered through a flow-control valve. Since this metering action involves reducing flow from a pump to a work cylinder, a pump must deliver more fluid than is required to actuate a cylinder at the desired speed. Excess fluid returns to a tank through a relief valve. To conserve power and avoid undue stress on a pump, a relief valve's setting should be only slightly higher than a working pressure's, which a cylinder requires.

Application

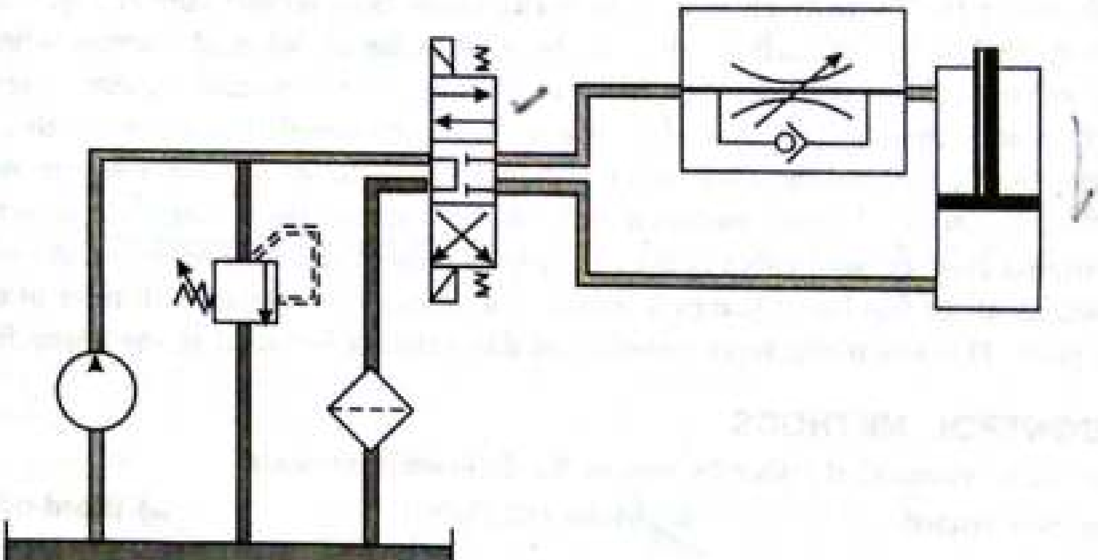
A meter-in circuit is ideal in applications where a load always offers a positive resistance to flow during a controlled stroke. Examples would be feeding grinder tables, welding machines, milling machines, and rotary hydraulic motor drives. A flow-control-and-check valve used in this type of circuit would allow reverse free flow for the return stroke of a cylinder, but it would not provide control of return stroke speed.

Drawback

The meter-in circuit is advantageous and accurate only for positive loads. For loads that are overrunning the actuator does not control the speed of extension. This overrunning load creates cavity and vacuum.



FLOW CONTROL VALVE TO CONTROL EXTENSION STROKE



FLOW CONTROL VALVE TO CONTROL RETURN STROKE

METER IN CIRCUITS

(ii) METER-OUT CIRCUIT

With a meter-out circuit, a flow-control valve is installed on the return side of a cylinder so that it controls a cylinder's actuation by metering its discharge flow. A relief valve is set slightly above the operating pressure that is required by the type of work.

Application

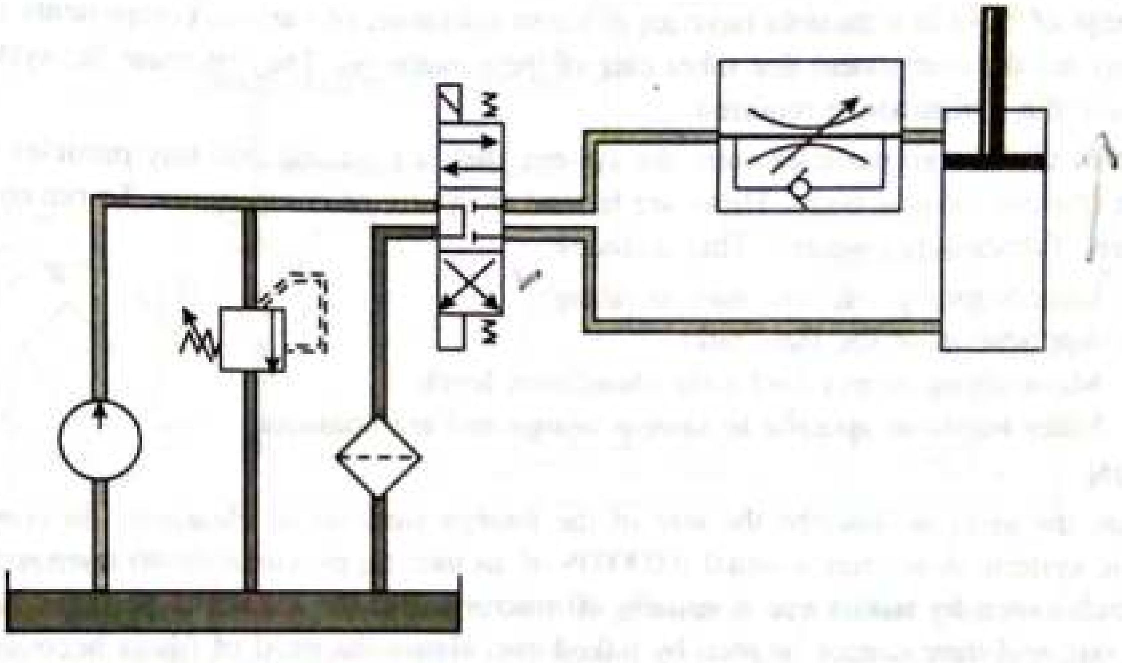
This type of circuit is ideal for overhauling load applications in which a workload tends to pull an operating piston faster than a pump's delivery would warrant. Examples would be for drilling, reaming, boring, turning, threading, tapping, cutting off, and cold sawing machines. A flow-control-and-check valve used in this circuit would allow reverse free flow, but it would not provide a control of return stroke speed.

Drawback

These types of circuits though they protect cylinder from overrunning, there is pressure intensification if the flow control valve is on the rod side of the cylinder and this damages the rod seals.

Note: *Have a close look at the positioning of the flow control valve and check valve in the meter-in and meter-out circuits. Only change would be in the way the check valves are connected.*

Both meter-in and meter-out circuits are effective but not efficient as the excess controlled flow passes through the relief valve back to the tank.



The above circuit meters the oil coming from the Rod Side

METER-OUT CIRCUIT

(iii) BLEED-OFF CIRCUIT

A typical bleed-off circuit is not installed directly in a feed line. A valve regulates flow to a cylinder by diverting an adjustable portion of a pump's flow to a tank. Since fluid delivered to a work cylinder does not have to pass through a flow-control valve, excess fluid does not have to be dumped through a relief valve. This type of circuit usually involves less heat generation because pressure on a pump equals the work resistance during a feed operation.

Bleed off circuits are efficient but not effective. Meter-in circuits can withstand overrunning and opposing loads but meter-out and bleed off types are useful only for opposing loads. The choice of flow control method and flow control valve depends on the application.

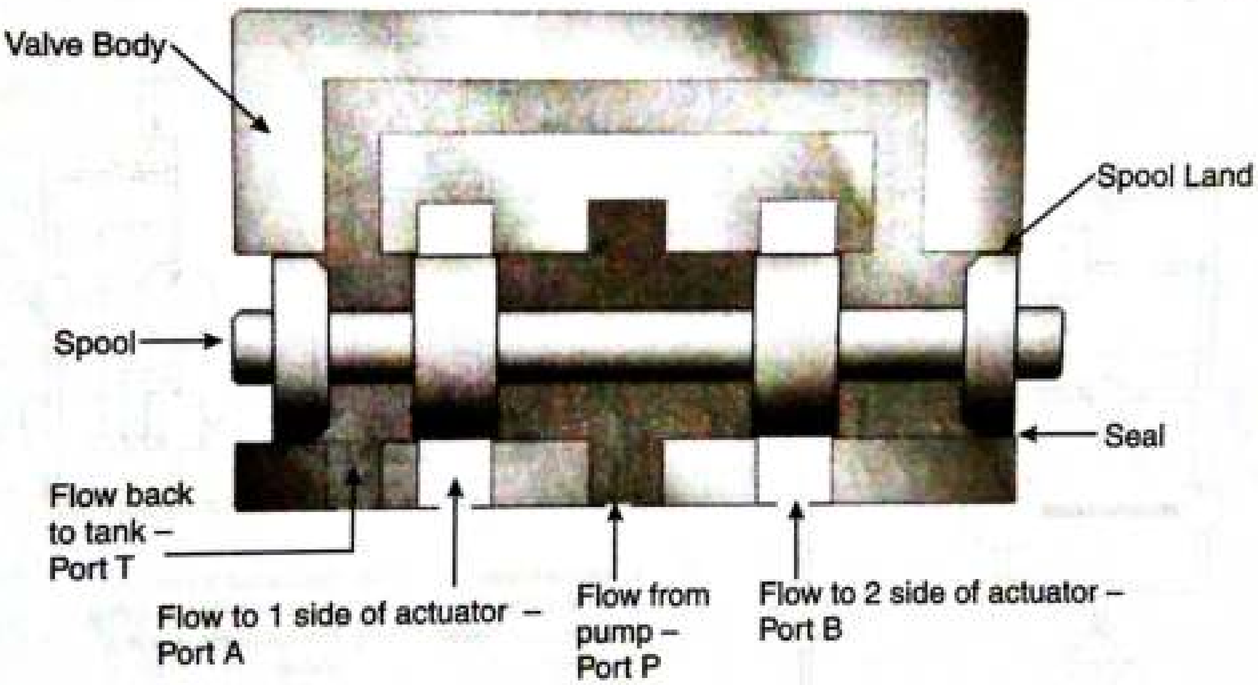
INSTITUTE OF TEXTILE TECHNOLOGY CHOUDWAR

Hydraulic Machines & Industrial Fluid Power

Direction Control Valves

Direction Control Valves

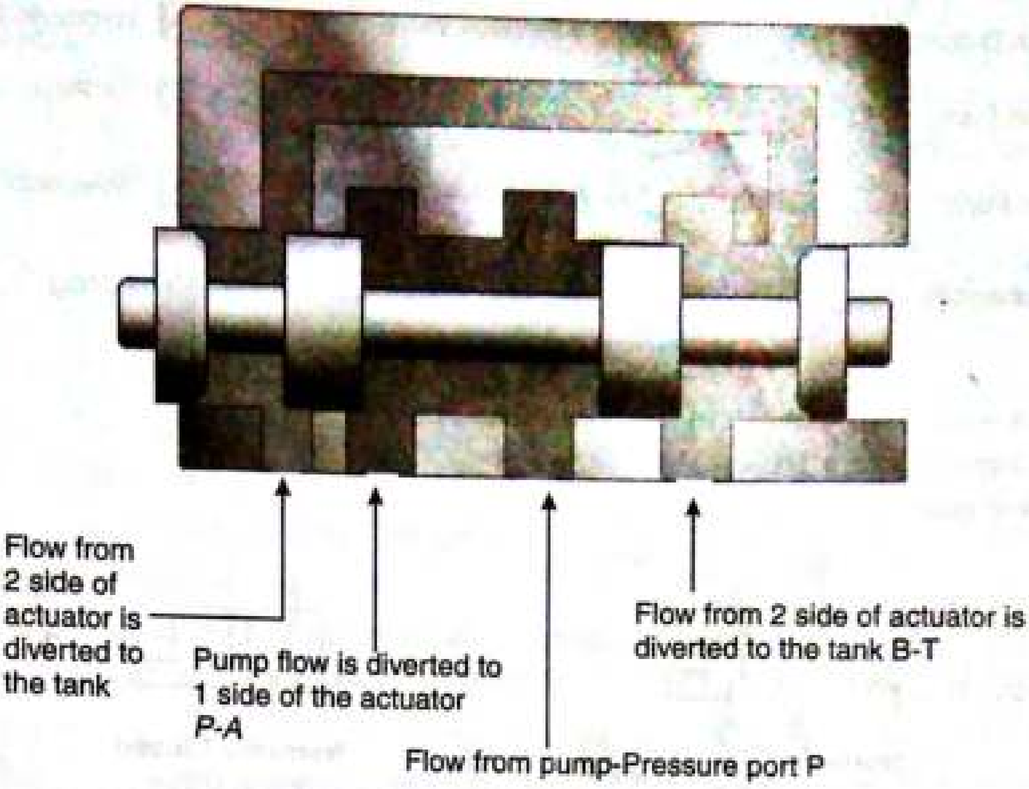
Direction control valves (DCV) are used to ***start, stop and change the direction*** of flow in a fluid power system. This can be demonstrated by the figures below.



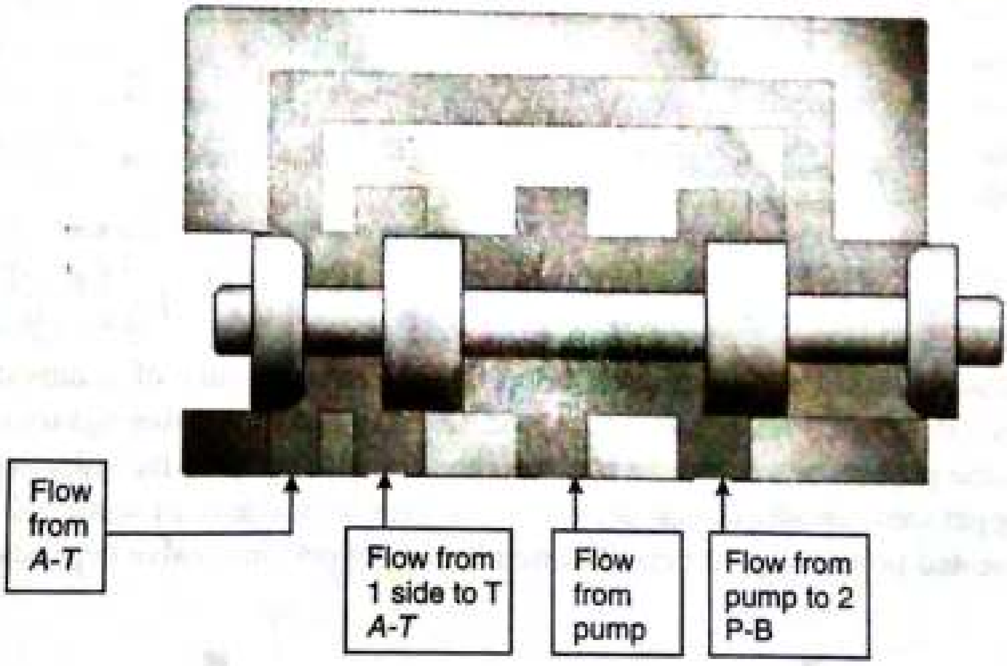
DCV IN ITS CENTER POSITION-stops flow

The DCVs may be

- Poppet type, in which a piston or ball moves on and off a seat.
- Rotary-spool type, in which a spool rotates about its axis.
- Sliding-spool type, in which a spool slides axially in a bore. In this type, a spool is often classified according to the flow conditions created when it is in the normal or neutral position. A closed-center spool blocks all valve ports from each other when in the normal position. In an open-center spool, all valve ports are open to each other when the spool is in the normal position.

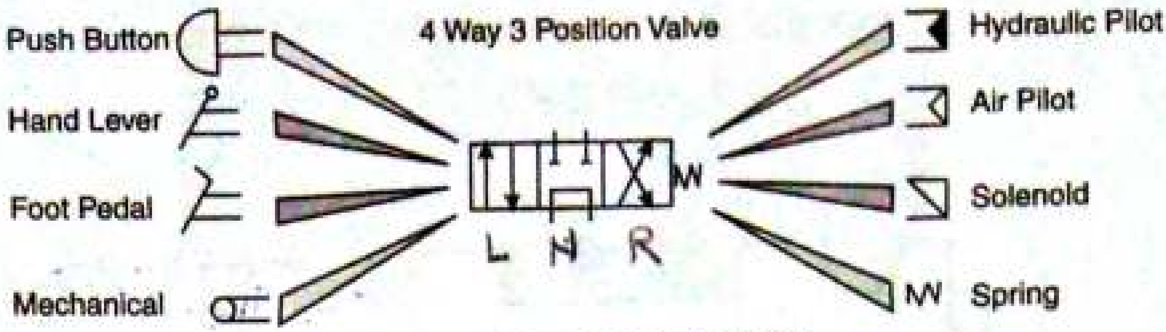


DCV IN ITS LEFT SIDE POSITION - *allow flow to pass to 1 side*



DCV IN ITS RIGHT SIDE POSITION-*allows flow to pass to 2 side*

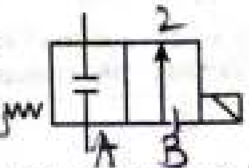
- Directional-control valves may also be designated according to the method used to actuate the valve element. A poppet-type valve is usually hydraulically operated. A rotary-spool type may be manually (lever or plunger action), mechanically (cam or trip action), or electrically (solenoid action) operated. A sliding-spool type may be manually, mechanically, electrically, or hydraulically operated, or it may be operated in combination. The different types of actuation is shown in the figure shown in the next page (methods of acheating DCV)
- Directional-control valves may also be classified according to the number of positions of the valve elements or the total number of flow paths provided in the extreme position. For example, a three-position, four-way valve has two extreme positions and a center or neutral position. In each of the two extreme positions, there are two flow paths, making a total of four flow paths.



METHODS OF ACTUATING DCV

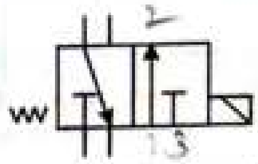
- Direction control valves are also designated as
  (*i*) Normally open DCV
  (*ii*) Normally closed DCV

2 ports
2 position

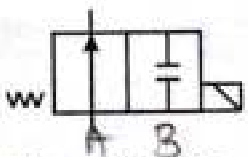


Normally Closed (2/2 DCV)
Spring Offset
Solenoid Operated 2-way

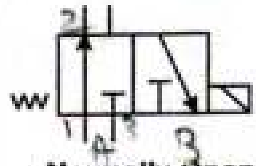
3 ports
2 position



Normally Closed (3/2 DCV)
Spring Offset
Solenoid Operated 3-way



Normally Open
Spring Offset
Solenoid Operated 2-way

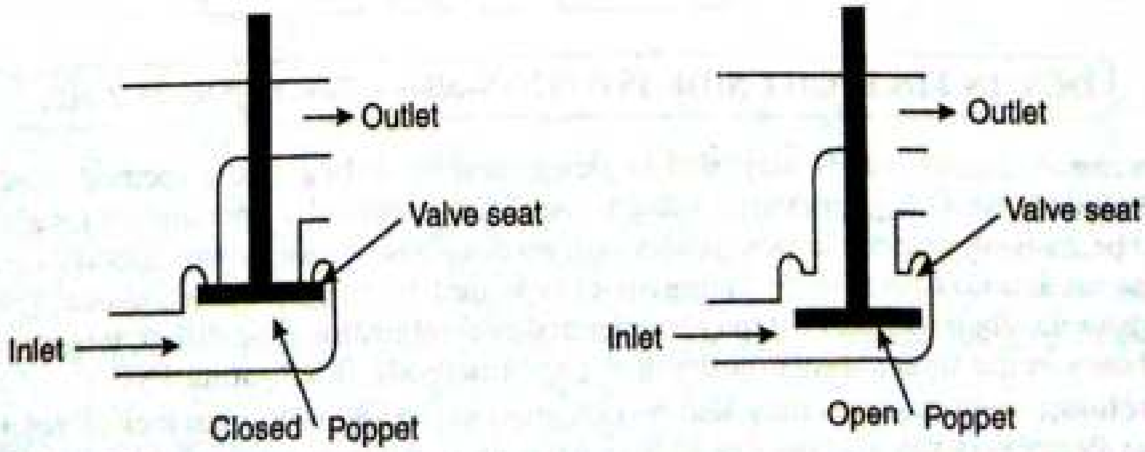


Normally Open
Spring Offset
Solenoid Operated 3-way

Spring offset → spring action automatically returns the spool to the normal offset position as the shifter force is released.

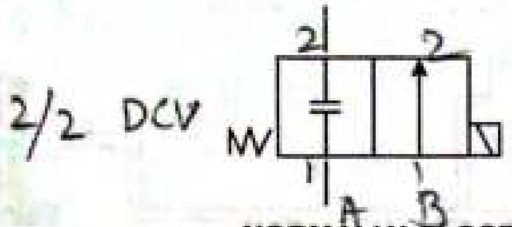
POPPET VALVES

The Figure below shows a simple poppet valve. It consists primarily of a movable poppet that closes against a valve seat. Pressure from the inlet tends to hold the valve tightly closed. A slight force applied to the poppet stem opens the poppet. The action is similar to the valves of an automobile engine. The poppet stem usually has an *O*-ring seal to prevent leakage. In some valves, the poppets are held in the seated position by springs. The number of poppets in a valve depends on the purpose of the valve.

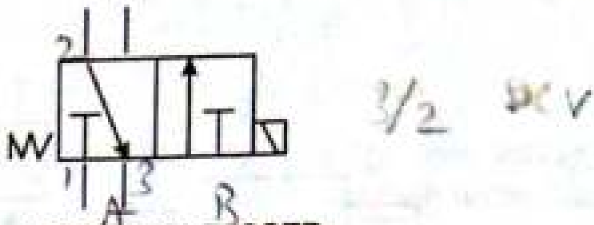


TWO-WAY DIRECTION CONTROL VALVE – SPOOL TYPE

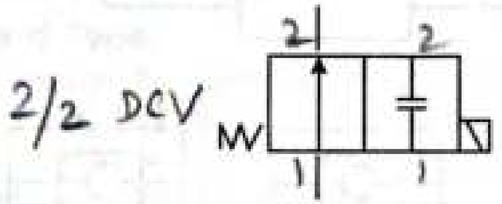
A two-way valve is generally used to control the direction of fluid flow in a hydraulic circuit and is a sliding-spool type. The below ANSI symbols show the different two-way, directional-control valves. As the spool moves back and forth, it either allows or prevents fluid flow through the valve. In either shifted position in a two-way valve, a pressure port is open to one cylinder port, but the opposite cylinder port is not open to a tank. A tank port on this valve is used primarily for draining.



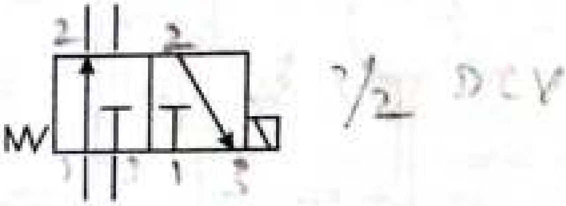
NORMALLY CLOSED
Spring Offset
Solenoid Operated 2-way



NORMALLY CLOSED
Spring Offset
Solenoid Operated 3-way



NORMALLY OPEN
Spring Offset
Solenoid Operated 2-way



NORMALLY OPEN
Spring Offset
Solenoid Operated 3-way

TWO-WAY DIRECTION CONTROL VALVE

FOUR-WAY VALVES

Four-way, directional-control valves are used to control the direction of fluid flow in a hydraulic circuit, which controls the direction of movement of a work cylinder or the rotation of a fluid motor. These valves are usually the sliding-spool type. A typical four-way, directional-control valve has four ports:

- One pressure port is connected to a pressure line.
- One return or exhaust port is connected to a reservoir.
- Two working ports are connected by lines to an actuating unit.

Four-way valves consist of a rectangular cast body, a sliding spool, and a way to position a spool. A spool is precision-fitted to a bore through the longitudinal axis of a valve's body. The lands of a spool divide this bore into a series of separate chambers. Ports in a valve's body lead into a chamber so that a spool's position determines which ports are open to each other and which ones are sealed off from each other. Ports that are sealed off from each other in one position may be interconnected in another position. Spool positioning is accomplished manually, mechanically, electrically, or hydraulically or by combing any of the four. The figure shown at the start of the chapter shows how the spool position determines the possible flow conditions in the circuit. The four ports are marked P, T, A, and B:

- P is connected to the flow source
- T to the tank and
- A and B to the respective ports of the work cylinder, hydraulic motor, or some other valve in the circuit.

In the first diagram it shows the pressure port is blocked and in second diagram it shows the spool is in such a position that port P is open to port A, and port B is open to port T. Ports A and B are connected to the ports of the cylinder. Flow through port P-A cause the piston of the cylinder to move to the right (extension). Return flow from the cylinder passes through ports B and T. In diagram B, port P is open to port B, and the piston moves to the left. Return flow from the cylinder passes through ports A and T.

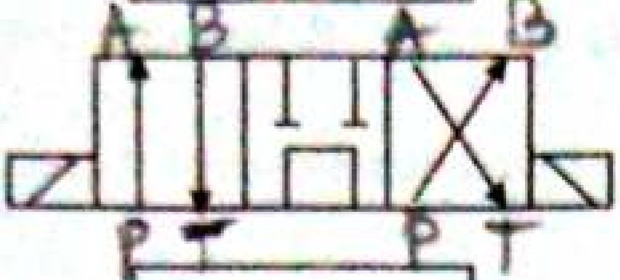
The below table lists some of the classifications of directional-control valves. These valves could be identified according to the-

- Number of spool positions.
- Number of flow paths in the extreme positions.
- Flow pattern in the center or crossover position.
- Method of shifting a spool.
- Method of providing spool return

	Classification	*Description*
Path-of-flow type	Two way	Allows a total of two possible flow paths in two extreme spool positions
	Four way	Allows a total of four possible flow paths in two extreme spool positions
Control type	Manual operated Pilot operated	Hand lever is used to shift the spool. Hydraulic pressure is used to shift the spool.
	Solenoid operated	Solenoid action is used to shift the spool.
	Solenoid controlled pilot operated	Solenoid action is used to shift the integral pilot spool, which directs the pilot flow to shift the main spool.
Position type	Two position	Spool has two extreme positions of dwell.
	Three position	Spool has two extreme positions plus one intermediate or center position.
Spring type	Spring offset	Spring action automatically returns the spool to the normal offset position as soon as shifter force is released. (Spring offset is always a two-way valve.)
	No spring	Spool is not spring-loaded; it is moved only by shifter force, and it remains where it is shifted (may be two- or three-position type, but three-position type uses detent).
	Spring centered	Spring action automatically returns the spool to the center position as soon as the shifter force is released. (Spring-centered is always a three-position valve.)
Spool type	Open center Closed center Tandem center Partially closed center Semi-open center	These are five of the more common spool types. They refer to the flow pattern allowed when the spool is in the center position (three-position valves) or in the cross-over position (two-position valves).

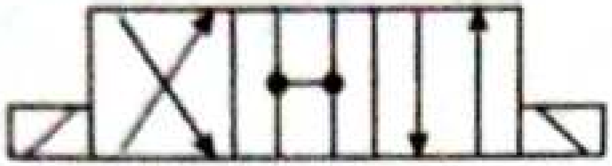
The various ANSI symbols for the corresponding DCV are shown below

Open Center

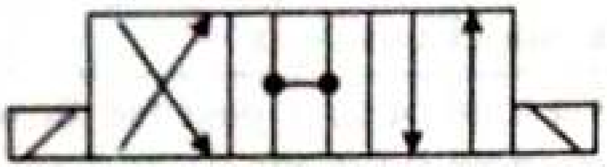


4/3 DCV
4 ports, 3 positions

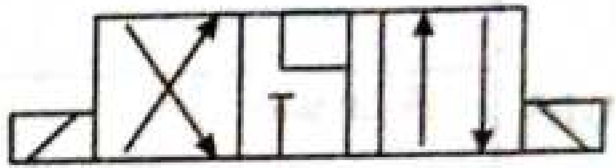
Open Center



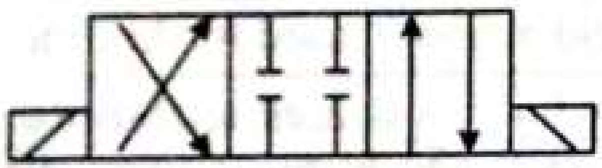
Open Type



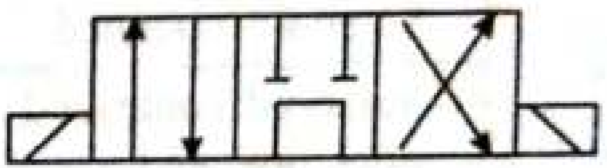
FloatType



Closed Type



Tandem Type



INSTITUTE OF TEXTILE TECHNOLOGY CHOUDWAR

Hydraulic Machines & Industrial Fluid Power

ANSI Symbols

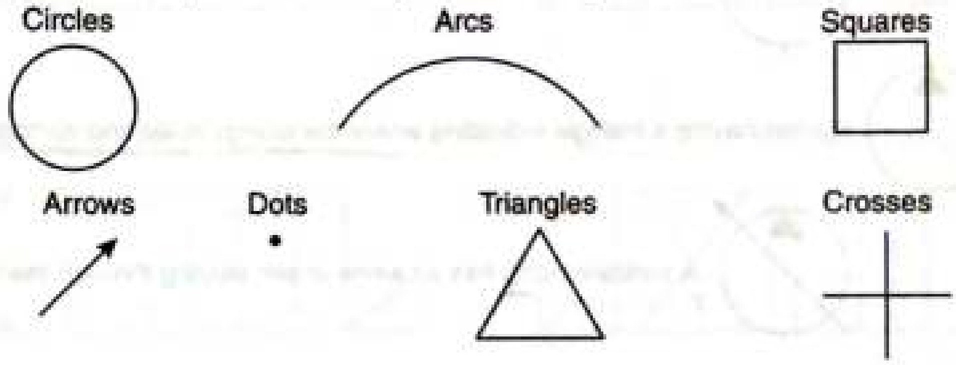
Hydraulic Symbols – ANSI Symbols

Symbols are used to draw and represent a hydraulic system on paper or in computer. It becomes necessary to give unique representation to various components in the system and when drawn their positioning in exact locations is more important.

The main reasons for using symbols are:

- These symbols are required for Technical Communication.
- They are language independent as the symbols used are the same worldwide.
- These symbols emphasize the function of the component.
- Every symbol uses elementary form to understand logically their method of operation.

Some of the elementary forms or shapes used in symbols are:



The various symbols used in the hydraulic system uses these elementary shapes and form the symbol. The various symbols mostly used in hydraulic circuits throughout the world to represent the system is explained below dividing them according to the components as flow lines, Reservoirs, Direction control valves, Pressure control Valves, Flow control Valves, Cylinders, Motors, Check valves, filters and heat exchangers.

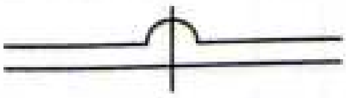
Flow Lines:

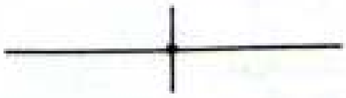
The steel tubes, pipes or flexible hoses used in hydraulic circuits are shown as either continuous lines or dashed lines. They are straight or curved to show their flexibility. They are best explained below.

Continuous lines indicate working, pilot supply, return or electrical lines


Dashed lines indicate a pilot, drain, purge, or bleed line


Flexible lines indicate a hose which usually connects moving parts


Crossing lines use loops at cross over

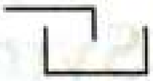

Lines joining may use a dot at the junction

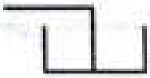
Reservoirs:

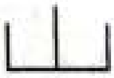
Reservoirs are used to store oil and based on the shape of the reservoir container and their position these reservoirs are divided into many types and are shown below.


Vented reservoirs are shown as rectangles without top lines


Pressurized reservoirs are shown as capsules

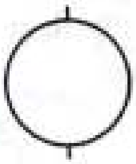

Above oil level return-line reservoir


Below oil level return-line reservoir

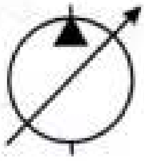

Common reservoir symbol minimizes the need to draw a number of lines into one reservoir

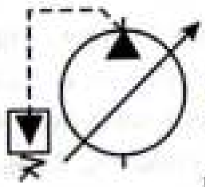
Pumps:

Pumps are the energy producing devices to the fluid power system. Hence they are represented by a circle and filled triangle, which moves towards the discharge. Here based on the nature of displacement they are split and their corresponding graphic symbols are shown below


Rotary devices are shown as circles


Pumps having a triangle indicating where the energy is leaving component


A variable pump has an arrow drawn sloping through thecircle


A pressure-compensated pump is drawn with a control symbol and connected adjustable spring. The control symbol's triangle indicates the source of energy

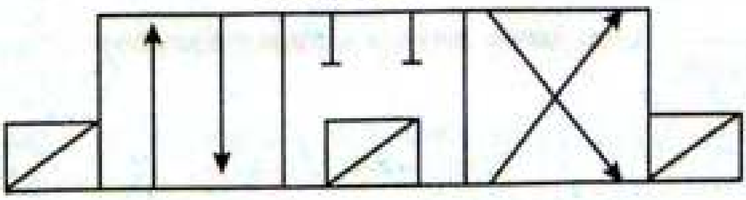

The drain line indicates internal leakage that returns to tank

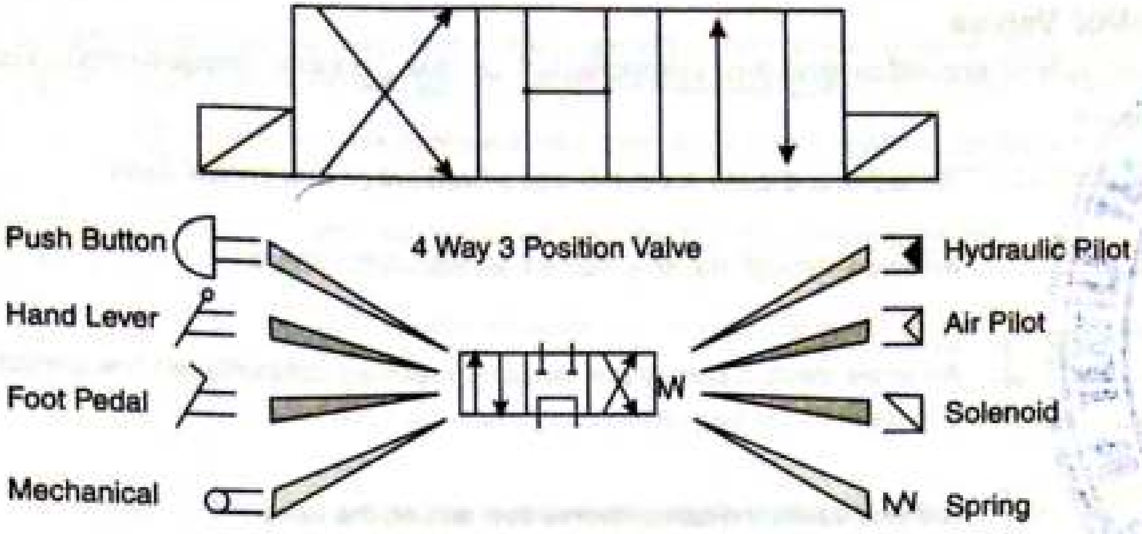

Bi-directional pumps have two energy flow triangles

Direction Control Valves

There are different types of direction control Valves and based on their number of ports, positions and methods of operation they are divided into 2/3/4 Way, 2/3 position, method of operation (Manual/ spring/lever/mechanical/pilot (hydraulic/pneumatic), electrical etc.) their symbols change and they are shown below.

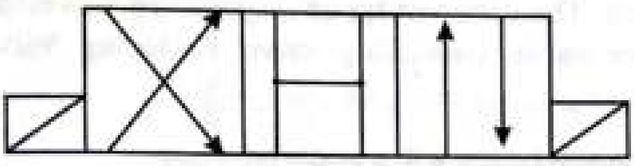
Open Center



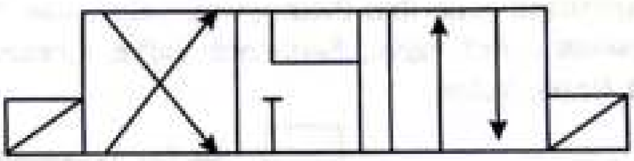


METHODS OF ACTUATING DCV

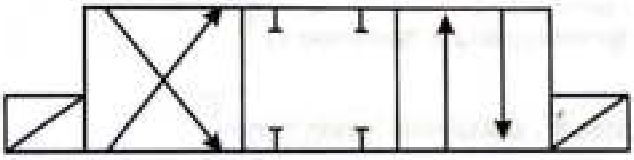
OPEN TYPE



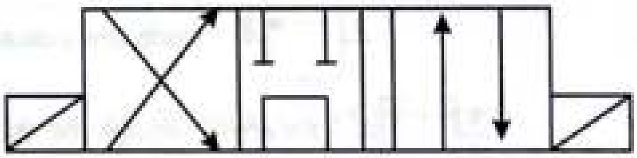
FLOAT TYPE

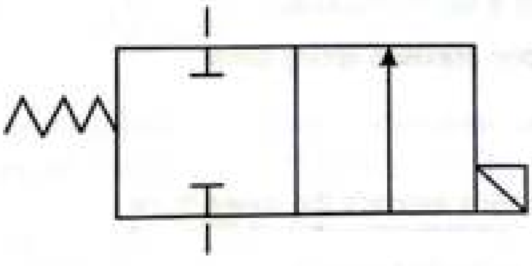


CLOSED TYPE

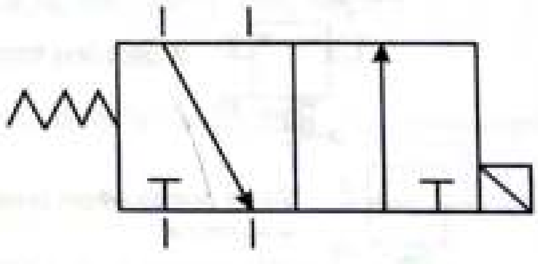


TANDEM TYPE

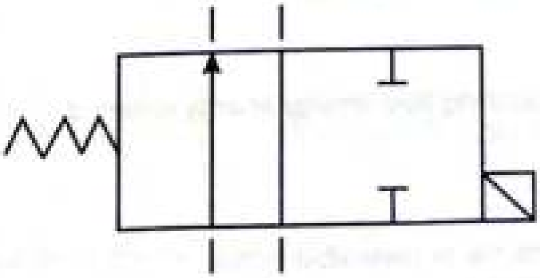




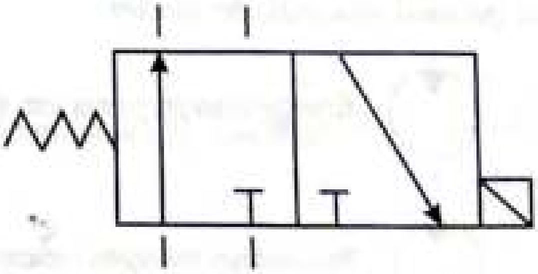
NORMALLY CLOSED
Spring Offest
Solenoid Operated 2-way



NORMALLY CLOSED
Spring Offset
Solenoid Operated 3-way



NORMALLY OPEN
Spring Offset
Solenoid Operated 2-way



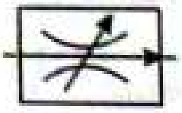
NORMALLY OPEN
Spring Offset
Solenoid operated 3-way

Flow Control Valves

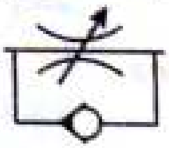
Flow control valves are either pressure compensated or non pressure compensated. They can be shown as below

An upper and lower arc symbolize a fixed orifice flow control valve

An arrow through the arcs indicate an adjustable orifice



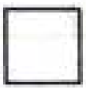
An arrow inside a control box indicates pressure compensated flow control



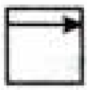
A check valves indicates reverse flow around the valve

Pressure Control Valves:

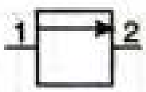
Pressure control valves are usually closed valves except pressure reducing valve and they are constructed such that their spring values can be altered. The different types of pressure valves are Pressure relief valve, Sequence valve, Counterbalance valve, Unloading valve, Reducing Valve, and Brake valve.



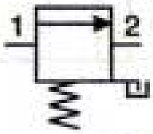
Symbol for pressure controlvalve begins with a single envelope



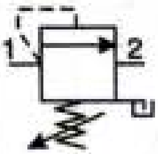
An arrow depictsthe direction of flow through the valve



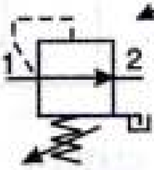
Ports are indicated as 1 (primary) and 2 (secondary)



An arrow through the spring indicates an adjustable spring force



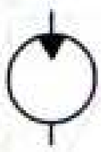
A pilot line from primary port opposes spring force



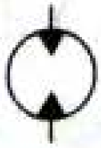
Valve opens when pressure in pilot line exceeds the spring force

Motors

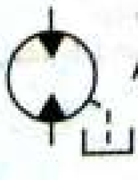
Motors are the devices that convert fluid energy to mechanical energy by producing torque. They are usually bi-directional and sometimes unidirectional. They receive energy and hence the energy triangle is pointed towards the center.



Energy triangle points into the circle indicating fluid energy energy entering



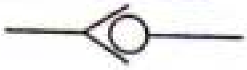
Two energy triangles indicate a bi-directional or reversible motor



A dashed line leaving the circle indicates a drain line to tank

Check Valves

Check valves are Zero leakage direction control valves. They allow reverse flow by additionally including a pilot line to open during the return flow. They are placed in many critical locations.



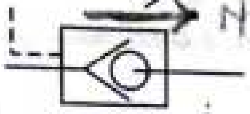
Check valves are drawn with small circles inside an open triangle

Free flow

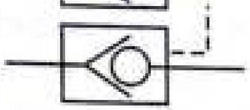
Free flow is opposite the direction the triangle is pointed

As the circle moves into the triangle, the flow is blocked

No flow



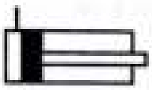
Pilot to open is indicated witha pilot line directed to the triangle



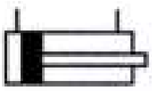
Pilot to close is indicated by directing pilot line to back of the circle

Cylinders

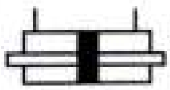
The cylinders are actuators which convert the fluid energy into reciprocating mechanical energy and based on the piston and their characteristics they are divided into many types and are best explained by the below figure.



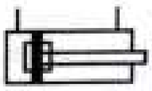
Single Acting (one line)



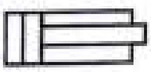
Double acting (two lines)



Double rod (two lines and two rods)



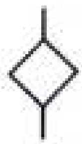
The internal rectangle indicates a cushion device at end of the stroke



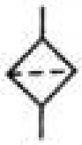
If diameter of rod is larger than usual, the symbol must reflect it

Filters:

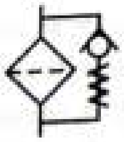
Filters are positioned at various locations and based on their location they are positioned separately or integrally with check valves.



A square standing on end symbolizes a fluid conditioning device



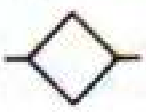
A dotted line across opposite corners indicates a filter or strainer



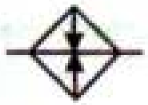
A check valve across the ports indicates a bypass filter

Heat Exchangers:

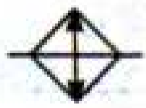
Heat exchangers are present in all the large industrial applications which are operated for long period of time.



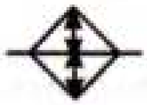
A Square standing on end symbolizes a fluid conditioning device



Arrows pointing in indicate a heater

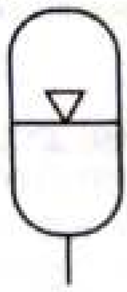


Arrows pointing out indicate a cooler



Arrows pointing in and out indicate a temperature controller

Accumulators



A cylindrical shell with a line separating oil and gas with a weight to indicate that it is pressurized by either gas/gravity/Spring force

INSTITUTE OF TEXTILE TECHNOLOGY CHOUDWAR

Hydraulic Machines & Industrial Fluid Power

Accumulators

ACCUMULATORS

The accumulators are devices that are used to store energy of the fluid (potential energy) in the hydraulic system under pressure created by an external source (Pump/surges/increase in system pressure etc.) against the dynamic force (weight or gravity, pressurized gas or mechanical force by springs). There are three basic different types of accumulators. They are:

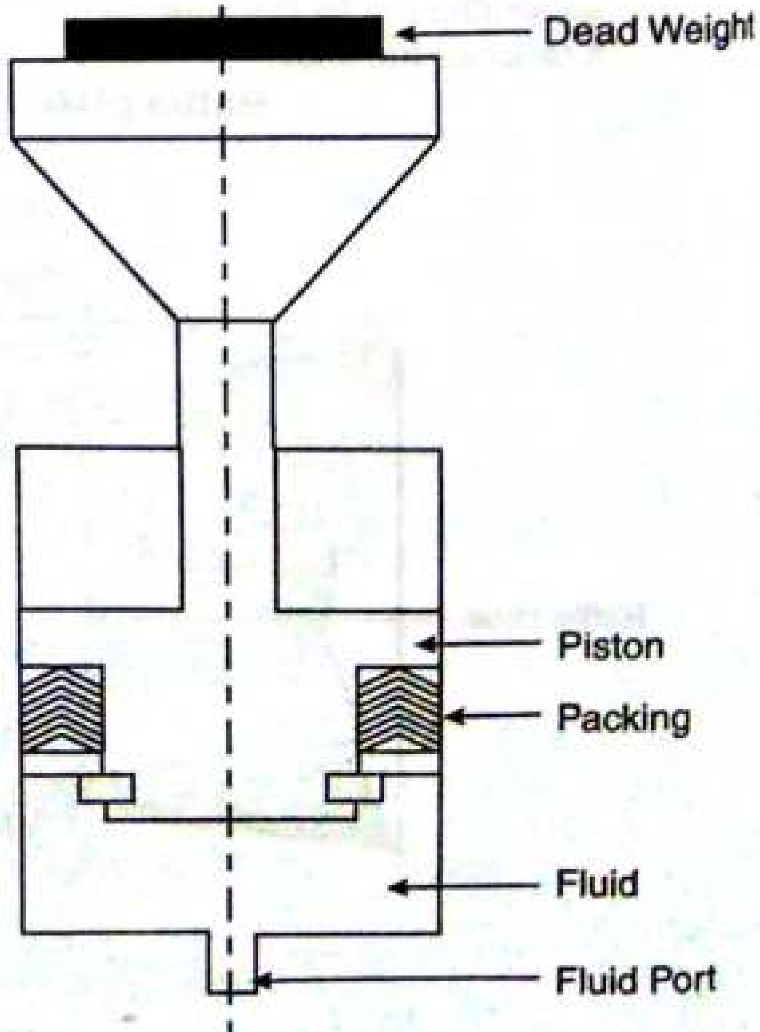
(i) Weight loaded or gravity type

(ii) Spring loaded or mechanical type

(iii) Gas loaded or pressurized or hydro pneumatic type

(i) WEIGHT LOADED ACCUMULATOR

This is the oldest type of accumulator. They consist of strong vertically mounted cylindrical shell, which incorporates the piston with packing to prevent leakage. A dead weight is attached at the top of the piston, which gives a large force of gravity that provides the potential energy in the accumulator.

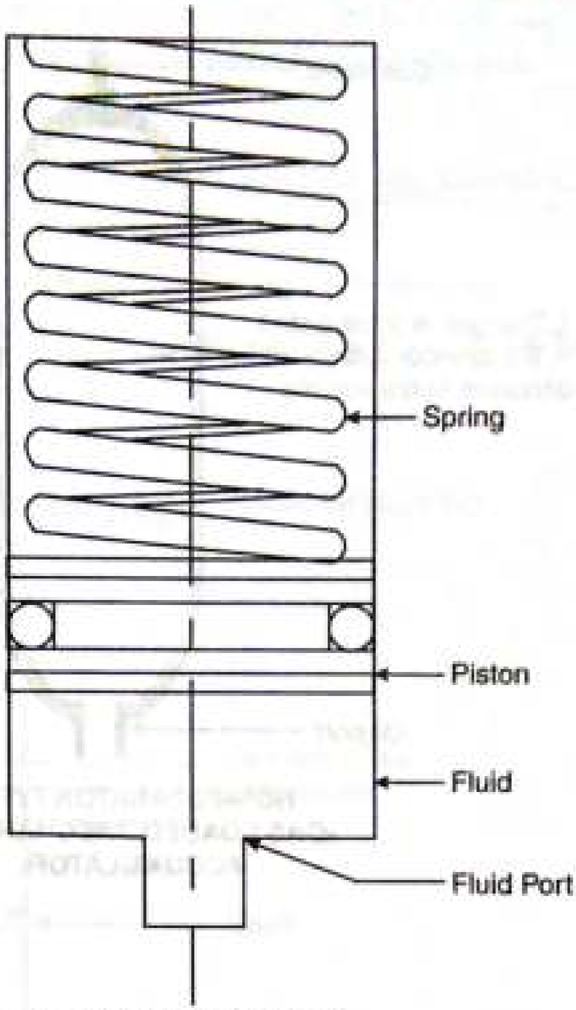


Advantage: This type of accumulator creates a constant fluid pressure (pressure may change only when the weight is changed and this is decided by the user) throughout the full volume output of the unit regardless of the rate and quantity of output. The other two types of accumulators lack this quality and there is change in pressure and the fluid output pressure decreases as a function of the volume output of the accumulator.

Disadvantage: These types are extremely large in size and weight and hence they are not portable and not used in mobile applications.

ii) SPRING LOADED ACCUMULATOR

Spring-loaded accumulators are similar in construction to that of weight loaded type of accumulator. In this type instead of loading the piston with weights, it is preloaded with spring compression. The spring force acts against the piston and forces the fluid into the system when needed. The pressure level of this type is dependent on the size and preloading of the spring. The pressure exerted by the spring is large when fully compressed and when it extends the force applied is less. Hence the pressure exerted on the fluid is not a constant. This type delivers a relatively small volume of oil at low pressures. To generate large pressures, the size of spring required is large and hence they tend to be heavy for high pressures, large volume applications. This type of accumulators are not used in applications which require large cycle rates because the spring would be subjected to fatigue and it would lose it elasticity sooner. Thus this type is mostly used for applications, which require less pressure/volume and that do not have surges and cyclic loadings.



Spring Type Accumulator

iii) GAS LOADED ACCUMULATOR

- These are the accumulators that are used in almost all the industrial applications. They use compressed gas to give the dynamic force and hence they are also called as "*hydro-pneumatic accumulators*".
- These accumulators obey the Boyle's Gas law, which states that for a constant temperature process, the pressure of the gas varies inversely with its volume. Thus when the gases are compressed more (*say half the initial volume*), the pressure is increased (*doubled in this case*) and the compressibility of gases accounts for the storage of potential energy. This energy forces the oil out of the accumulator when gas expands due to reduction in system pressure.
- *The important attention to be given to gas charged accumulators is that they have to be discharged before any service is taken up.*
- These accumulators are of two types based on the method they separate oil and gas. They are:

  (*i*) Non separator type (*ii*) Separator type

Non-separator type Accumulator:

In this type the accumulator is made of an enclosed shell and has oil port opening at the bottom and a gas-charging valve on the top. The oil remains at the bottom and gas remains at the top and there is no physical separator between and the gas directly pushes the oil.

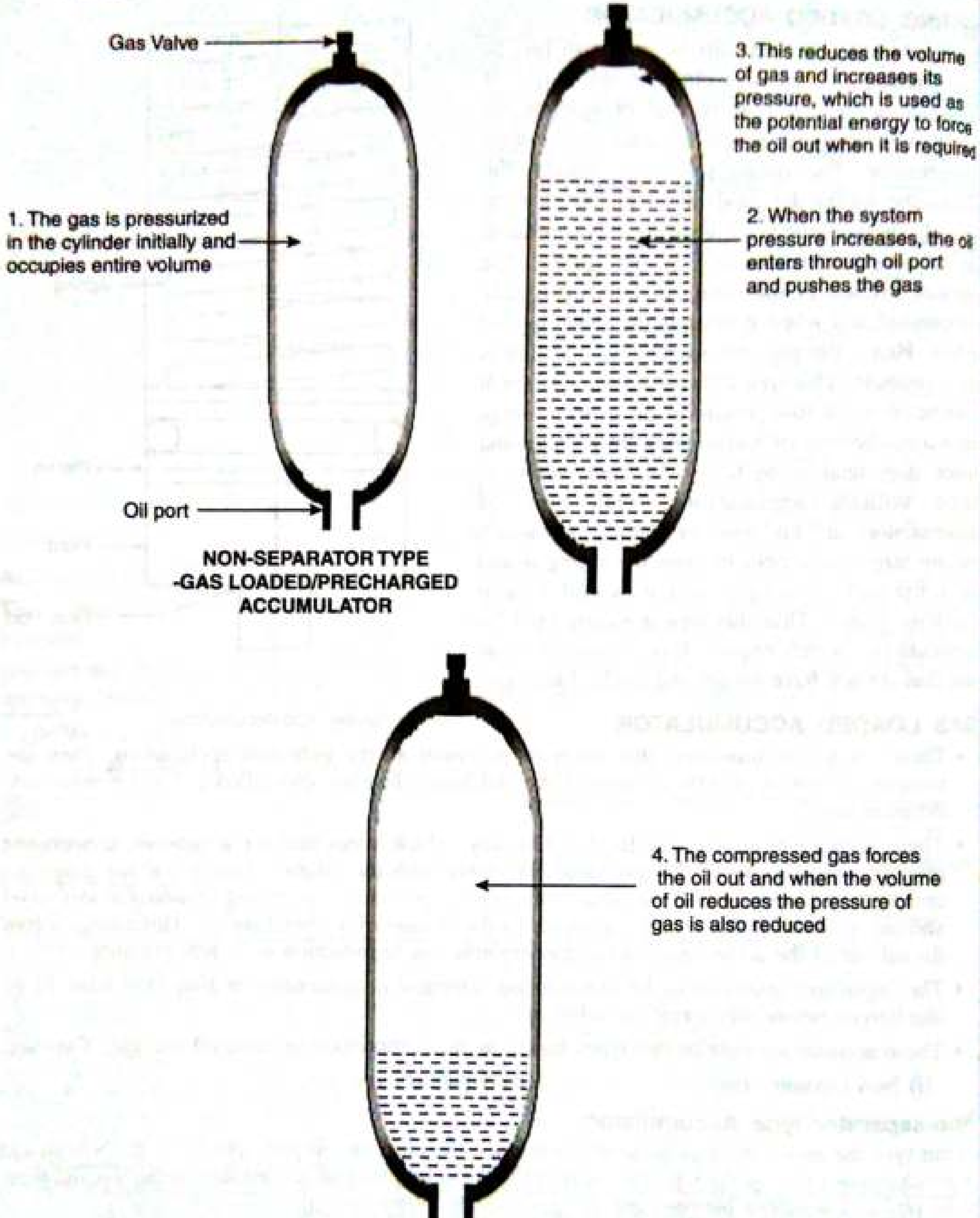
Advantages: This type has the ability to handle large volumes of oil.

Disadvantages:

(*i*) They have to be vertically mounted

(*ii*) The gas is entrapped in oil as there is no separator

(*iii*) This cannot be used for high speed pumps as the entrapped air may cause cavitations (*iv*) The entrapped air makes oil compressible and thus results in spongy operation of hydraulic actuators.



NON-SEPARATOR TYPE -GAS LOADED/PRECHARGED ACCUMULATOR

Separator Type Accumulators:

This is the most common accepted gas charged accumulator. In these types there is a physical barrier between the oil and the gas. The barrier effectively utilizes the compressibility of the gas. The three major classifications based on the type of separator are

(*i*) Piston type (*ii*) Diaphragm type (*iii*) Bladder type

Piston type Accumulators:

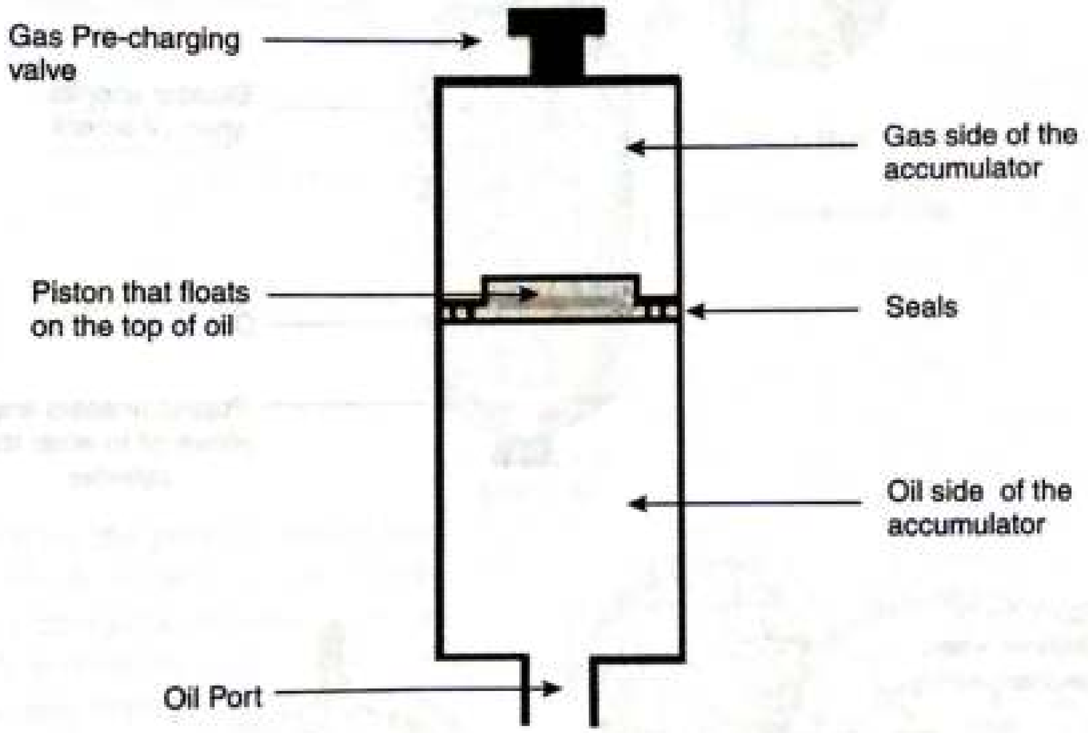
In this type the accumulator has a cylinder and a freely floating piston with O-ring seals. The piston serves as the barrier between the gas and oil. There is also a threaded lock ring that provides a safety feature. This prevents the operator from disassembling the unit while it is pre-charged.

Disadvantages:

- These types of accumulators are expensive to manufacture and have practical size limitations.
- Piston and seal friction is also a problem in low-pressure systems.
- There is always appreciable leakage over a long period requiring frequent pre-charging
- They cannot be used as pressure pulsation dampers and shock absorbers because of the inertia of the piston and friction of the seals.

Advantage:

This type has the ability to handle very high or low temperature system fluids through the utilization of compatible O-ring seals.



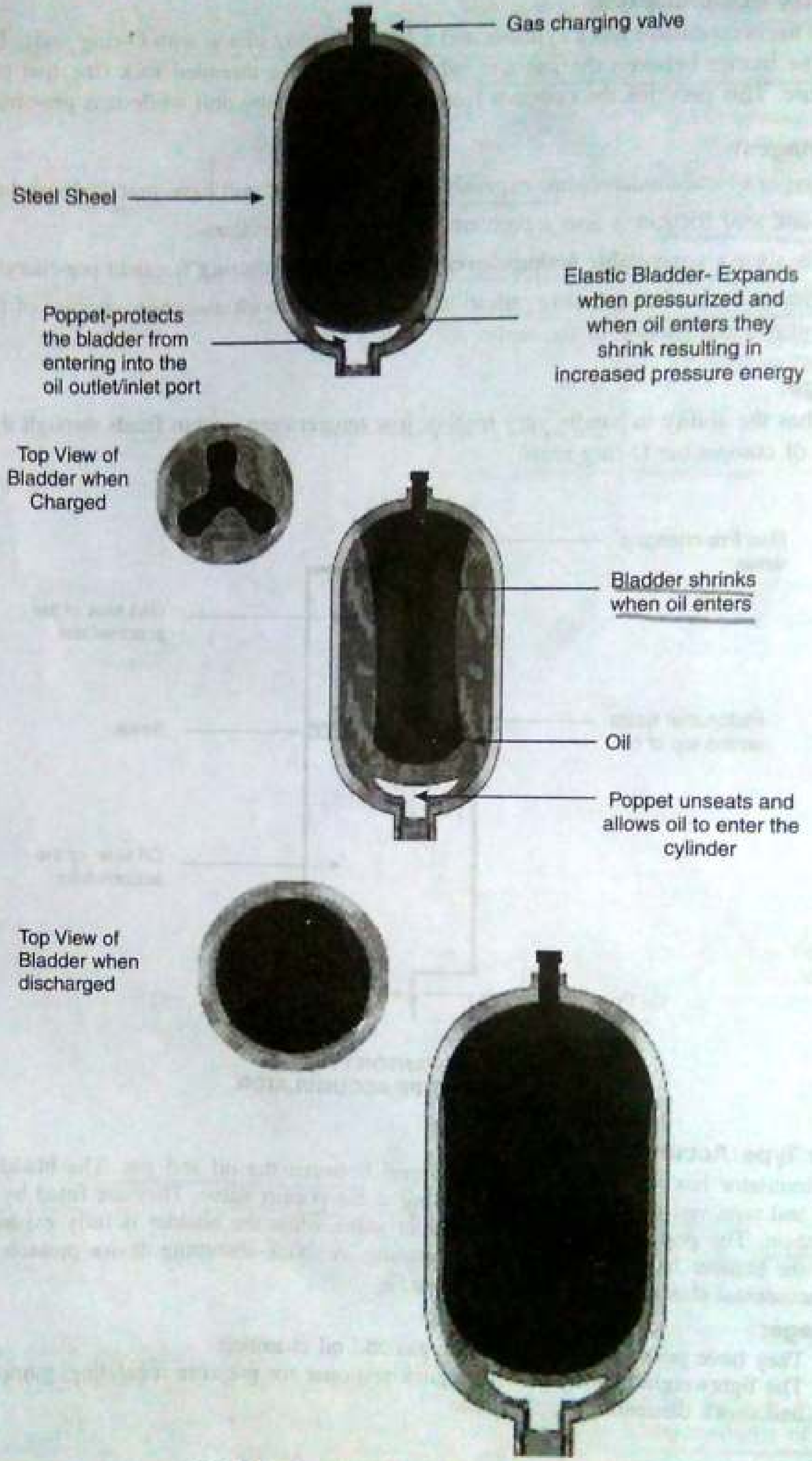
SEPARTOR TYPE-
PISTON TYPE ACCUMULATOR

Bladder Type Accumulators

This accumulator has an elastic barrier (bladder) between the oil and gas. The bladder can be installed and removed through the shell opening at the poppet valve. They are fitted by means of vulcanization. The poppet valve closes the inlet valve when the bladder is fully expanded. This prevents the bladder from entering into the opening. A shock-absorbing device protects the valve against accidental shocks during quick opening.

Advantage:

- They have positive sealing between gas and oil chambers.
- The lightweight bladder provides quick response for pressure regulating, pump pulsation and shock dampening applications



BLADDER TYPE ACCUMULATOR

INSTITUTE OF TEXTILE TECHNOLOGY CHOUDWAR

Hydraulic Machines & Industrial Fluid Power

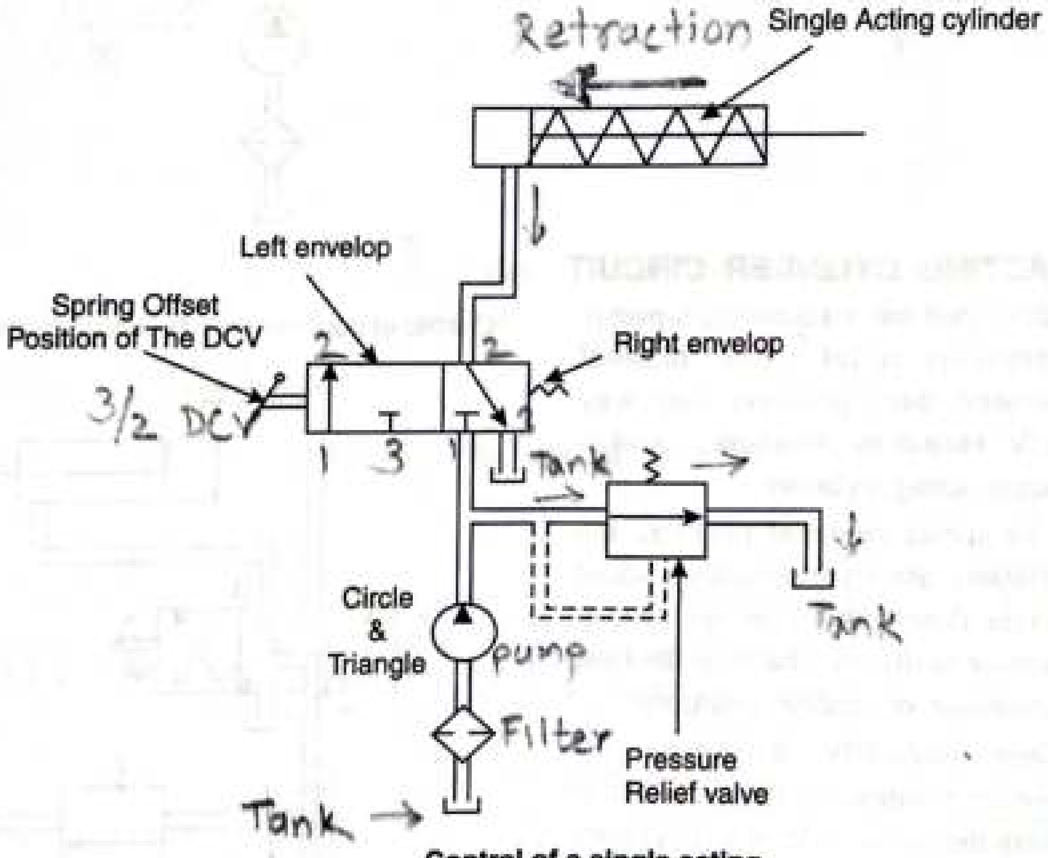
Hydraulic Circuits

Hydraulic Circuits

This chapter had been made more illustrative to have better understanding of the systems. The basic components of the hydraulic system remain the same. But, the combinations of the components are different and their locations make the difference in type of application. After completing this chapter it must be simple to understand how to build a small and simple hydraulic system. It should be kept in mind, designing of circuits for any application can be made as a fusión of these circuits forthwith.

SINGLE ACTING CYLINDER CIRCUIT

To start with, a simple circuit with minimum number of components and simple in operation is taken.

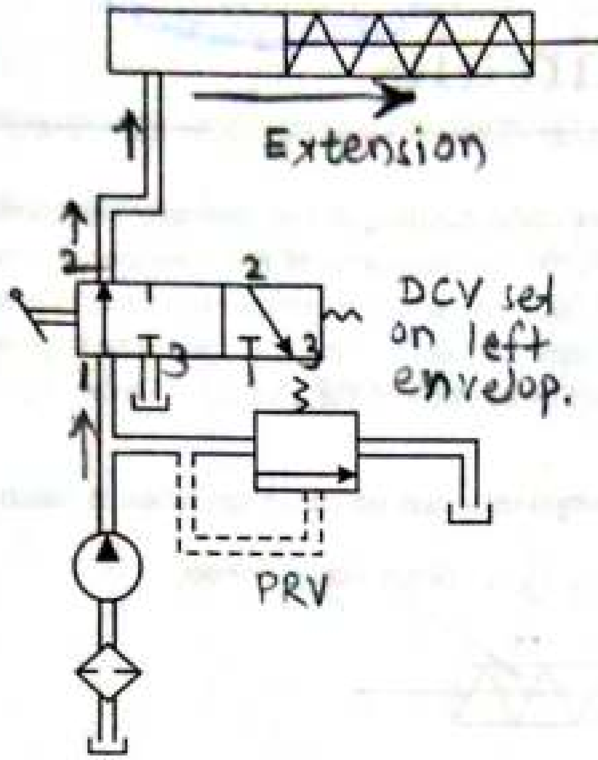


Control of a single acting hydraulic cylinder

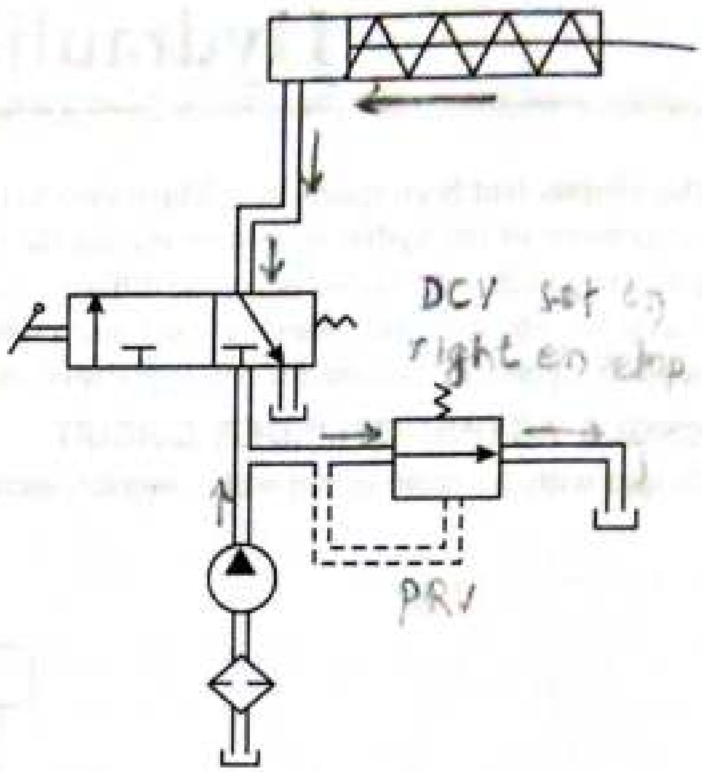
- It has a reservoir, a pump, a *two position- three way manual operated spring centered* direction control valve, a pressure relief valve and a single acting cylinder.
- In its normal position, the fluid is diverted back to the tank through the pressure relief valve. (*See the pressure relief valve is open to the tank*). In this position the pump operates at maximum pressure.
- When the direction control valve is manually pushed to left envelop, flow enters the piston side of the single acting cylinder. (*See the Pressure relief valve is closed and cylinder extends and note the position of the direction control valve*)

- At the end of extension stroke, DCV is pushed to right envelop resulting in the pump flow diverted through the PRV to the tank.
- The fluid in the cylinder is pushed by the spring on the rod side to retract and the flow goes to the tank. This can be seen in the below figure.

Control of single Octing hydraulic cylinder



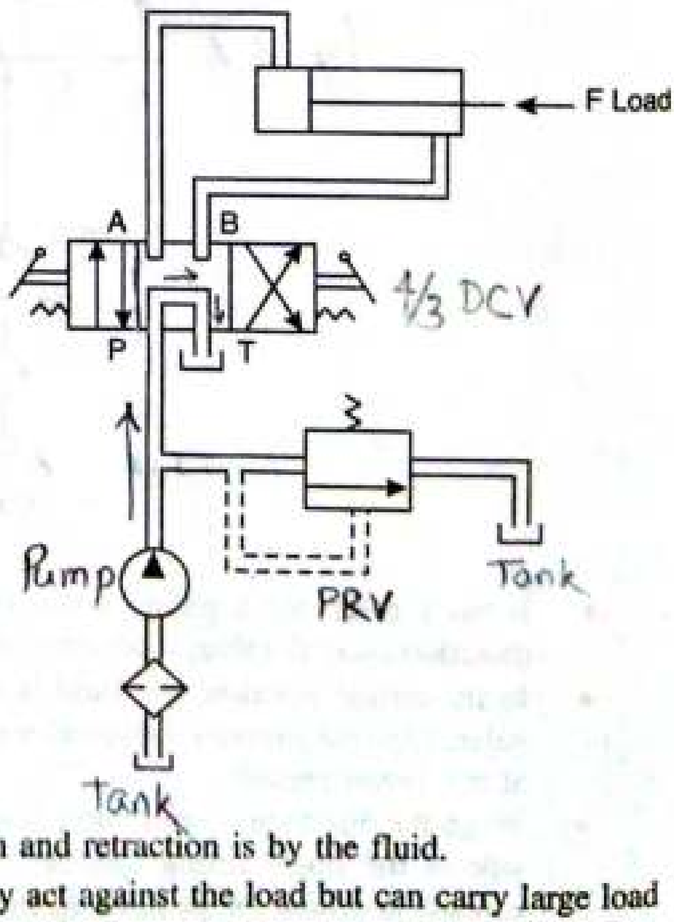
Control of single acting hydraulic cylinder



DOUBLE ACTING CYLINDER CIRCUIT

- This circuit has a reservoir, a pump, a pressure relief valve, manual operated three position four way DCV (tandem position), and a double acting cylinder
- In its spring centered position, the cylinders are hydraulically locked and the flow without any increase in pressure is diverted back to the tank (advantage of tandem position)
- When manually shifted to left envelop position of the DCV, flow enters the piston side of the cylinder and extension stroke takes place, against the load
- At the end of extension stroke, the DCV can be shifted to the right envelope in which case the flow is directed to the rod side of the actuator and retraction takes place.

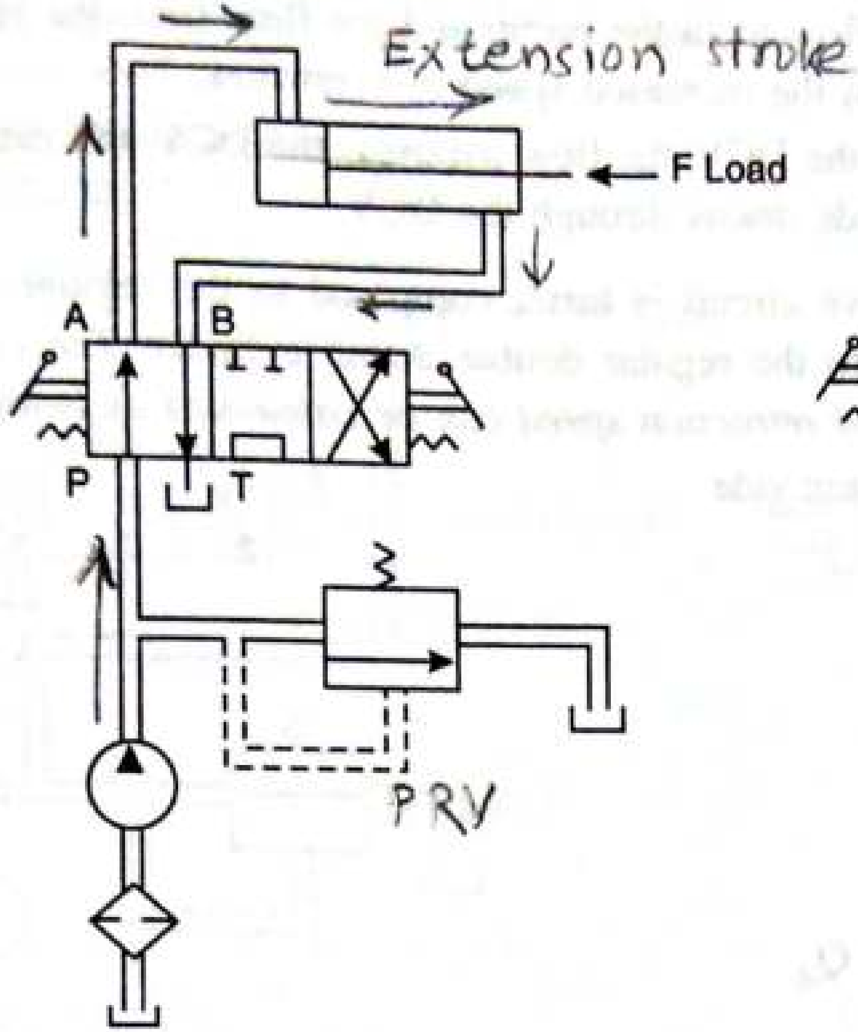
Control of double-acting hydraulic cylinder



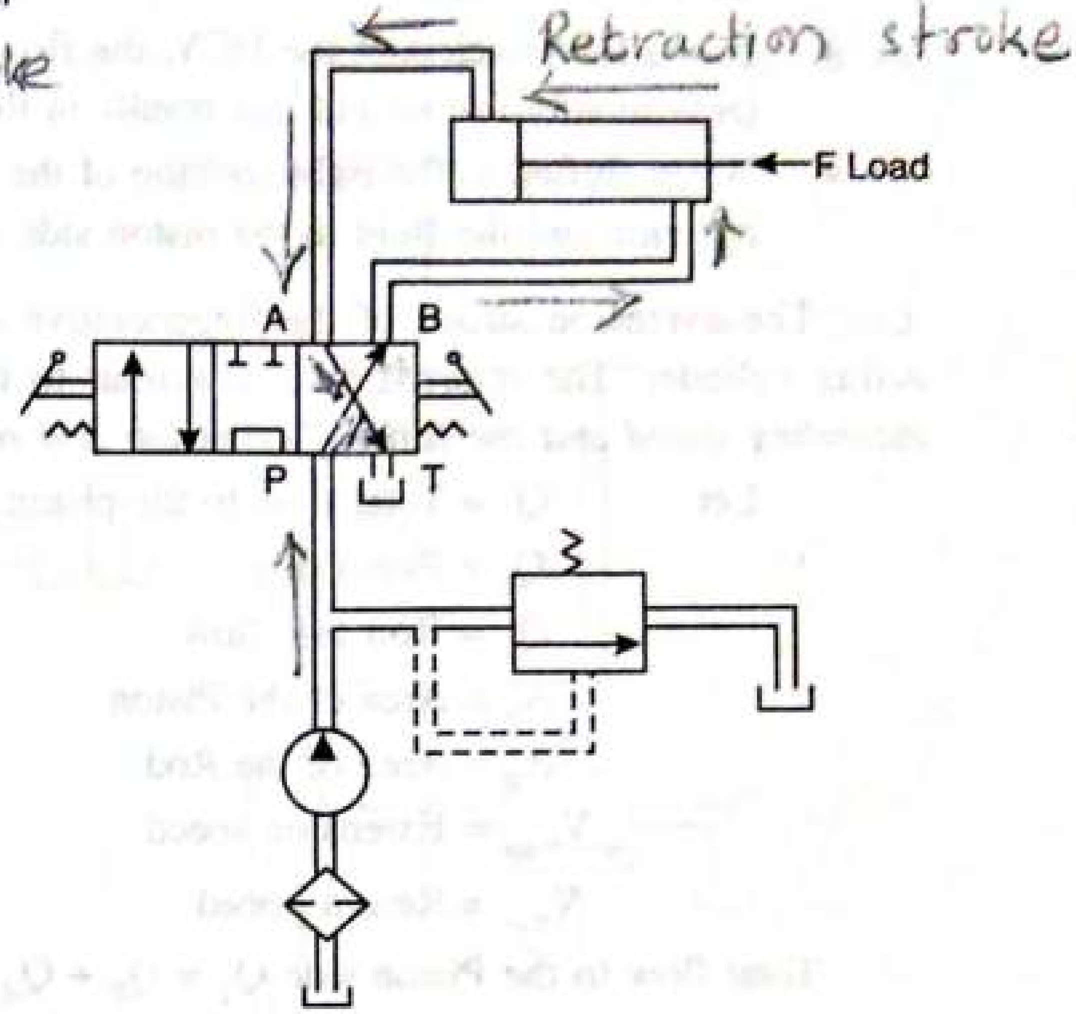
- In the double acting cylinder both extension and retraction is by the fluid.
- The extension of the cylinder is slow as they act against the load but can carry large load as the area on the piston side is more

- The retraction in double acting cylinder is fast as the area on the rod side is less and the same pump flow enters the side.
- At the end of extension and return strokes if the flow is not stopped there is increase in pressure and flow is diverted through the PRV.

Control of 0 double-acting hydraulic cylinder



In the above figure, DCV is shifted to Left envelop to allow fluid to enter Piston Side



In the above figure, DCV is shifted to right envelop to allow fluid to enter Rod Side

INSTITUTE OF TEXTILE TECHNOLOGY CHOUDWAR

Hydraulic Machines & Industrial Fluid Power

Pneumatic System

Pneumatic System Components

AIR GENERATION AND DISTRIBUTION

AIR PREPARATION

For the continuing performance of control systems and working elements of pneumatics it is necessary to guarantee that the air supply is:

- at the required pressure,
- dry and
- clean

If these conditions are not fulfilled, then short to medium term degeneration of the system will be accelerated. The effect is downtime on the machinery in addition to increased costs for repair or replacement of parts.

The generation of compressed air starts off with compression. The compressed air flows through an entire series of components before reaching the consuming device. The type of compressor and its location to a lesser or greater degree affect the amount of dirt particles, oil and water, which enter into a pneumatic system. The equipment to be considered in the generation and preparation of air include:

- Inlet filter
- Air compressor
- Air reservoir
- Air dryer
- Air filter with water separator
- Pressure regulator
- Air lubricator
- Drainage points

Poorly prepared compressed air will inevitably lead to malfunctions and may manifest itself in the system as follows:

- Rapid wear of seals and moving parts in the cylinders and values
- Oiled-up values
- Contaminated silencers
- Corrosion in pipes, valves, cylinders and other components
- Flushing out of lubrication of moving components

In the case of leakage, escaping compressed air may impair the materials to be processed (e.g. food).

Pressure level Usually required in Pneumatic Systems:

As a rule, pneumatic components are designed for a maximum operating pressure of 800 to 1000 kPa (8-10 bar). Practical experience has shown, however, that approximately 600 kPa (6 bar) should be used for economic operation. Pressure losses of between 10 and 50 kPa (0.1 and 0.5 bar) must be expected due to the restrictions, bends, leaks and pipe-runs, depending on the size of the piping system and the method of layout. The compressor's system should provide at least 650 to 700 kPa (6.5 to 7 bar) for a desired operating pressure level of 600 kPa (6 bar).

AIR COMPRESSORS

The selection from the various types of compressors available is dependent upon quantity of air, pressure, quality and cleanliness and how dry the air should be. There are varying levels of these criteria depending on the type of compressor. The compressors are selected based on the SCFM. The compressors are sized to deliver the pressure and flow required for the system. The pressure level of the selected compressor should be at least 10 % more than the required system pressure. The compressor is placed in a clean, cool and oil free room and if it is noisy it is placed in a separate room.

Reciprocating piston compressors

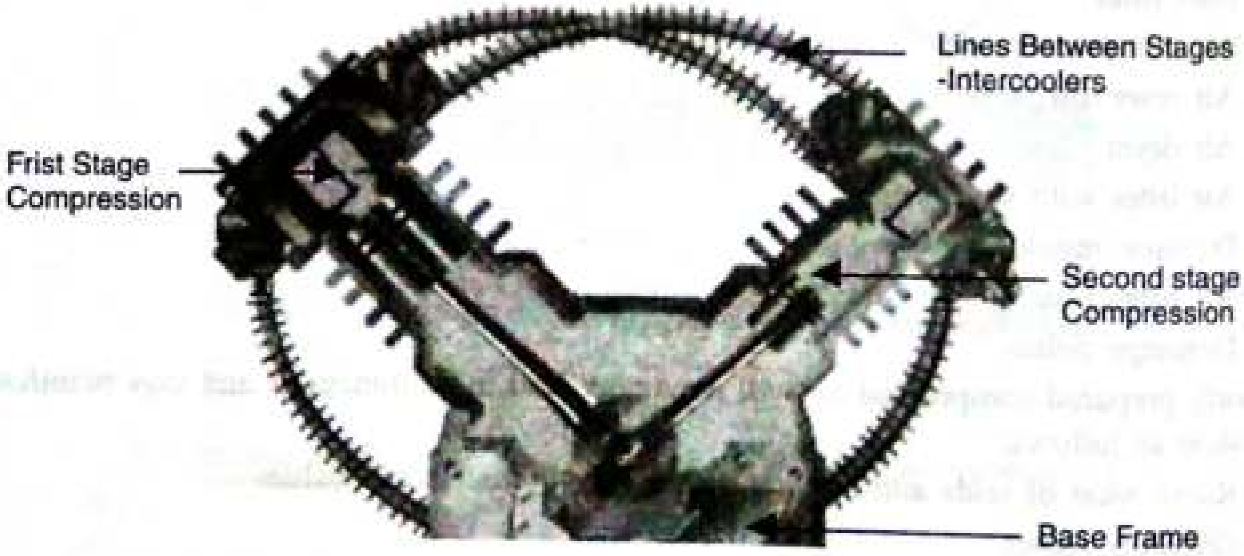
A piston compresses the air drawn in via an inlet valve. The air is passed on via outlet valve. These produce compressed air with pressure pulsations

Reciprocating compressors are very common and provide a wide range of pressures and delivery rates. For higher pressures multistage compression is used with inter-cooling between each stage of compression.

The optimum ranges of pressures for reciprocating compressors are approximately:

Up to 400 kPa (4 bar)	Single stage	125 psi
Up to 1500 kPa (15 bar)	Double stage	175 psi
Over 1500 kPa (> 15 bar)	Treble or multi stage	

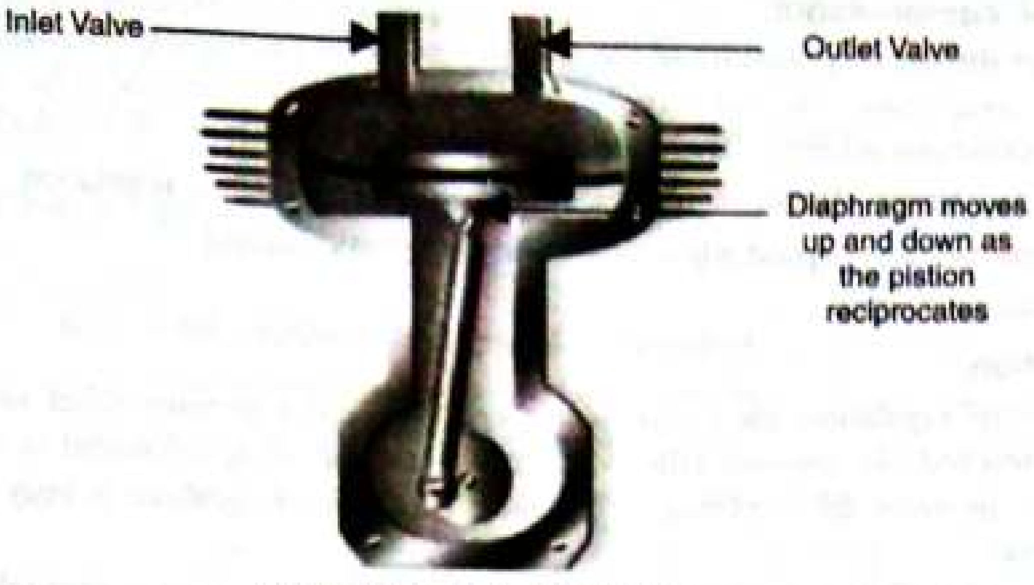
The compressor with stages has the first stage with large cylinder and subsequently the second and third stages are smaller and intercoolers are placed to achieve maximum efficiency.



TWO STAGE RECIPROCATING COMPRESSOR

Diaphragm Compressor

The diaphragm compressor belongs to the reciprocating piston compressor group. The compressor chamber is separated from the piston by a diaphragm. The advantage of this is that no oil can enter into the airflow from the compressor. The diaphragm compressor is therefore used where oil is to be excluded from the air supply, for example in the food, pharmaceutical and chemical industries.



DIAPHRAGM COMPRESSOR

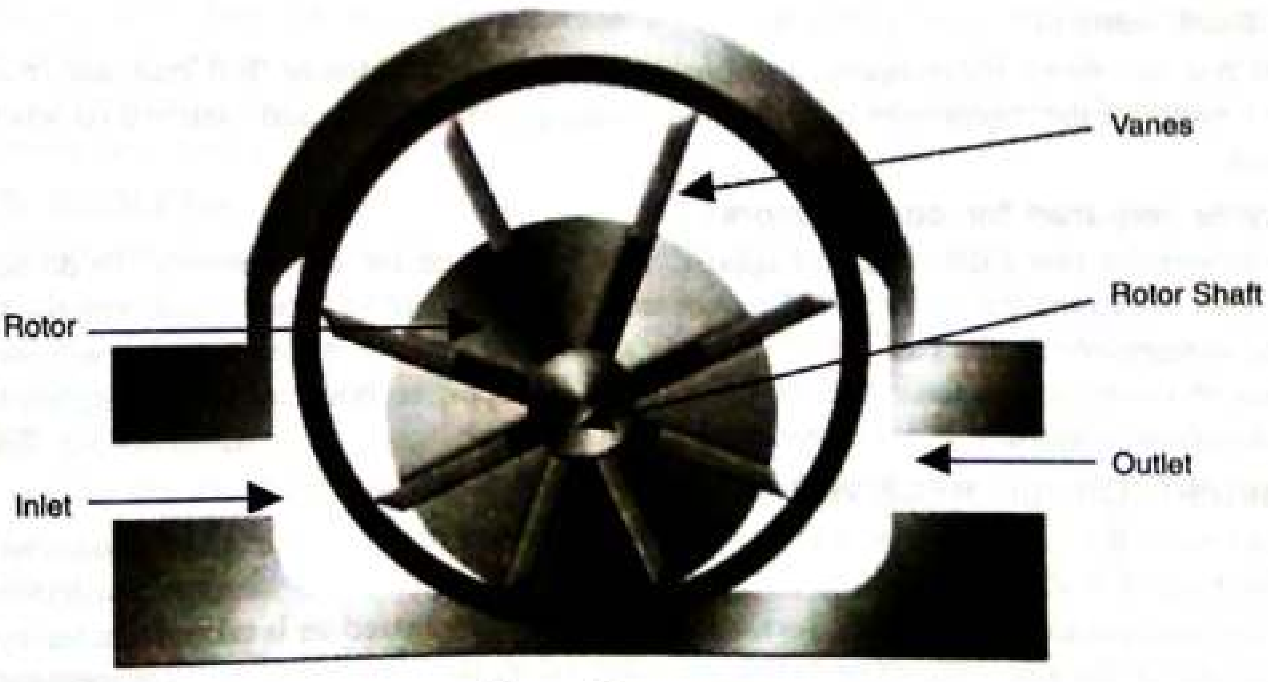
Rotary piston compressor

The rotary group of compressors uses rotating elements to compress and increase the pressure of the air. During the compression process, the compression chamber is continually reduced.

Screw compressor

Two screw-shaped shafts (rotors) turn in opposite directions. The meshed profile of the two shafts causes the air to flow, which is then compressed.

Vane Compressors



Vane Compressors

The vane compressors are similar to the construction of unbalanced vane pump. As the vanes move away near the inlet port the air is sucked in. As they move close near the outlet there is an increase in pressure and the compressed air is discharged at the outlet.

Flow compressor

These are particularly suitable for large delivery quantities. Flow compressors are made in axial and radial form. The air is made to flow through one or several turbine wheels. The kinetic energy is converted into pressure energy. In the case of an axial compressor, the air is accelerated in the axial direction of flow by means of blades.

RESERVOIRS OR AIR RECEIVER

The reservoir or the air receiver is a pressure vessel designed based on the ASME codes to store compressed air. A reservoir is configured downstream of a compressor to stabilize compressed air. A reservoir compensates the pressure fluctuations when the compressed air is taken from the system. If the pressure in the reservoir drops below a certain value, the compressor will compensate until the set higher value is reached again. This has the advantage that the compressor does not need to operate continuously.

The large surface area of the reservoir cools the air. Thus, a portion of the moisture in the air is separated directly from the reservoir as water, which has to be regularly drained via a drain cock.

The size of a compressed air reservoir depends on the:

- Delivery volume of the compressor
- Air consumption for the applications
- Network size (any additional requirements)

- Type of compressor cycle regulation
- Permissible pressure drop in the supply network

The selection of the receiver size is based on the below formula:

$$\text{Receiver Size (in gallons)} = \frac{K \times cfm \times 14.7 \times 7.78}{Psig + 14.7}$$

Where,

K = 1 for continuously operated compressors

= 3 for intermittently operated compressors

7.48 is used to convert cubic feet into gallons

(cfm is the required flow rate required by the system.)

RECEIVER

INLET FILTERS

Inlet filters are the first defense components that protect the compressor, which in turn also protect the entire pneumatic system for any foreign particle. Usually compressors are placed in cool, clean and oil free environment. The high rated filters, which can prevent fine particles entering into the compressor, are mostly used.

INTER COOLERS

Intercoolers are tubes that connect the outlet of one stage of the compressor to the inlet of the successive stage. These tubes have thin fins, which help in dissipating the heat from the compressed air. Cooling the air increases the efficiency of the compressor. The forced air is made to blow around the compressor and the tubes to increase the rate of cooling.

AFTER COOLERS

After coolers are the heat removing components that is placed inline between the compressor and the air receiver. The properly sized after coolers are used to cool and remove the heat and moisture from the compressed air. There are two types of after coolers based on the type of cooling medium. They are:

- Water cooled after coolers
- Air cooled after coolers

Water-cooled After coolers:

These water coolers have a bundle of bronze tubes through which water enters and leaves the coolers. Hot compressed air is made to enter the space around the tubes and leave the cooler. (*The construction is similar and their advantage and disadvantage are similar to the water-cooled heat exchangers used in hydraulics*)

Air-cooled After coolers:

In the air-cooled after coolers the hot air is made to pass through the bundle of steel stubs and air is forced around the tubes to remove the heat. (*The construction is similar and their advantage and disadvantage are similar to the water-cooled heat exchangers used in hydraulics*)

PRESSURE SWITCHES

Compressors that are used with intermittent service always store compressed air in large air receivers. When the air is supplied to the system, the pressure inside the receiver decreases and this may not satisfy the requirement for the system. In such cases the pressure switches, which are similar to electrical switches, are used to switch ON and OFF the compressors. Initially the electrical circuit is normally open and this would allow the compressor to run. When the pressure inside the receiver increases, the pressure moves the poppet and closes the electrical circuit. This passes the signal to switch OFF the compressor. When the pressure reduces, the poppet is released and the contact is not maintained. This breaks open the signal and allows the compressor to switch ON.

AIR DRYERS

Condensate (water) enters into the air network through the air intake of the compressor. The accumulation of condensate depends largely on the relative air humidity. The relative air humidity is dependent on the air temperature and the weather situation.

The absolute humidity is the mass vapour, actually contained in one m^3 of air. The saturation quantity is the mass of water vapour, which one m^3 of air can absorb at the respective temperature.

The following formula applies if the relative air humidity is specified in percent:

$$\text{Relative humidity} = \frac{\text{Absolute humidity} \cdot 100\%}{\text{Saturation quantity}}$$

Since the saturation quantity is dependent on temperature, the relative humidity changes with the temperature, even if the absolute humidity remains constant. If the dew point is reached, the relative humidity increases to 100%.

Dew point

The dew point temperature is the temperature at which relative humidity is 100%. The lower the dew point the more the water will condense and reduce the amount entrapped in the air.

The service life of pneumatic systems is considerably reduced if excessive moisture is carried through the air system to the components. Therefore it is important to fit the necessary air-drying equipment to reduce the moisture content to a level, which suits the application, and the components used. There are three auxiliary methods of reducing the moisture content in air:

- Low temperature drying
- Adsorption drying
- Absorption drying

Pressure dew point

To be able to compare different types systems, the operating pressure of the systems must be taken into account. The term 'pressure dew point' is used in this context. The pressure dew point is the air temperature reached during drying at operating pressure. The pressure dew point of the dried air should be approx. 2 to 3 °C under the coolest ambient temperature. The pressure dew point is defined as:

"*The temperature at which the water vapor begins to condense out of the air at given temperature*".

The factors that affect dew point are

(*i*) Temperature

(*ii*) Air – pressure

(*iii*) Vapor content or percent relative humidity

The additional cost of installing air drying equipment can be amortized over a short period due to the reduction in maintenance costs, reduced downtime and increased reliability of the system.

Low temperature drying

The most common type of dryer today is the *refrigeration dryer*. With refrigerated drying, the compressed air is passed through a heat-exchanger system through which a refrigerant flows. The aim is to reduce the temperature of the air to a dew point, which ensures that the water in the air condenses and drops out in the quantity required.

The air entering into the refrigeration dryer is pre-cooled in a heat exchanger by the escaping cold air. It is then cooled in the cooling unit to temperatures between + 2 and + 5 °C. The dried compressed air is filtered. Before the compressed air is output into the network, the air is heated to bring the air back to ambient conditions. The advantages of the refrigerant air dryers are:

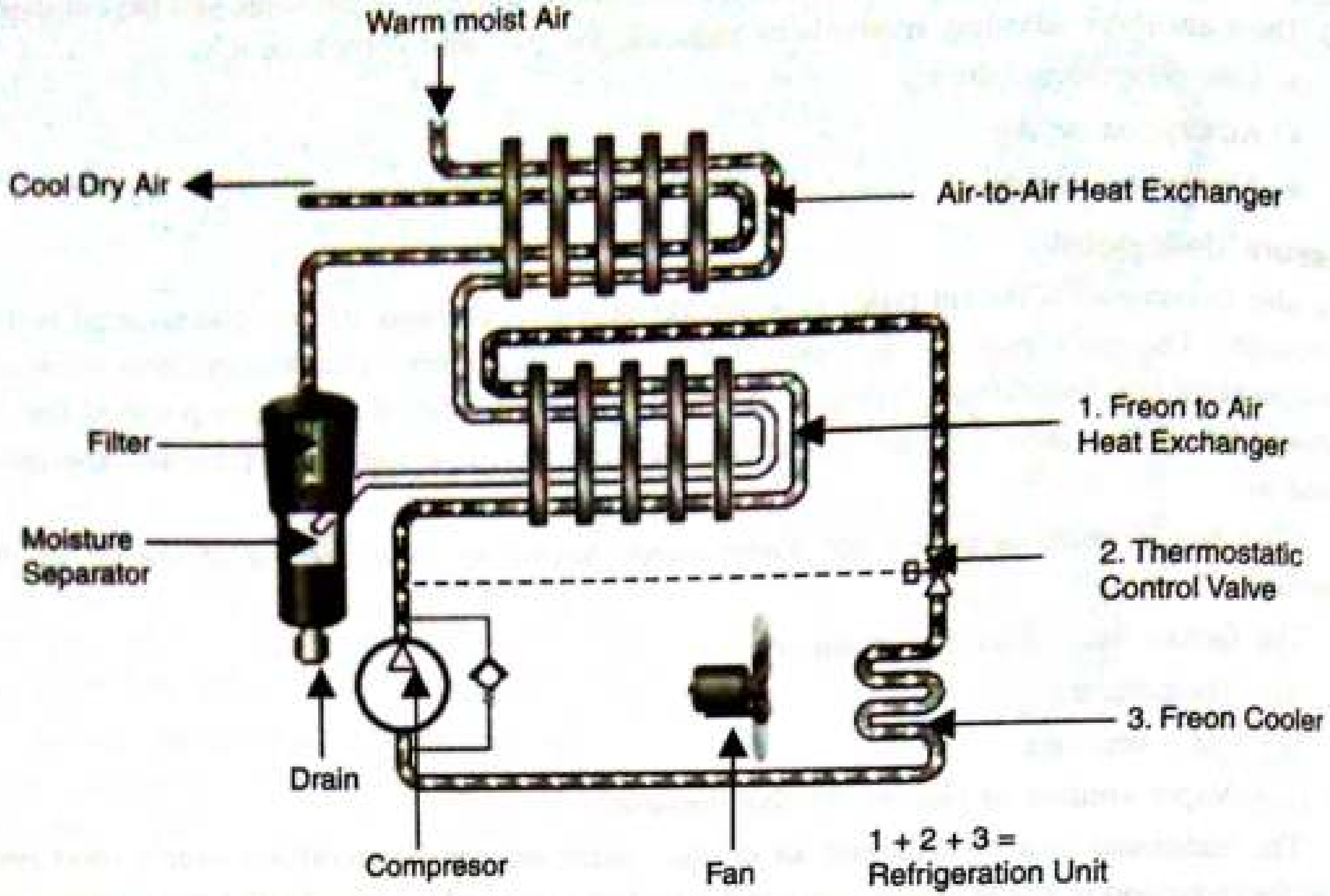
(*i*) Low initial cost

(*ii*) Low operational cost

(*iii*) Not damaged by oil vapors

The main disadvantage of the this type of driers is limited due point capacity of 33^0 F.

In this case the Warm hot air enters the system and cool dry air leaves the system. As the warm hot air enters, the cool dry air travels in the opposite direction and as a result maximum heat is transferred to the out going air in the air-to-air heat exchanger unit. Then as the incoming air passes through the freon to air heat exchanger, the air is once again made to pass in the opposite direction to the freon flow path. Here the air gets cooled and the water vapor is condensed and enters the moisture separator unit. The moisture is collected at the bottom of the unit and drained as and when enough water is stored and clean air passes through the filter at the top back into the air-to-air heat exchanger unit. In this way the air is removed from moisture and the warm air entering the drying unit to condense heats the clean air passing out. Thus the refrigeration dryer performs its operation. Using refrigeration methods, it is possible to achieve dew points of between + 2 and + 5°C.



REFRIGERATION DRYER

Adsorption dryers

Adsorption: Water is deposited on the surface of solids.

The drying agent is a granular material (gel) consisting almost entirely of silicon dioxide or activated alumina.

Usually two tanks are used. When the gel in one tank is saturated, the airflow is switched to the dry, second tank and the first tank is regenerated by hot air drying.

The lowest equivalent dew points (down to -90°C) can be achieved by means of adsorption drying. These types of dryers are used only when no moisture in the system is tolerated. There are two major types of desiccants dryers used namely the Heatless type and Heated type. In both the cases, the air is made to pass through two towers. First the air is made to pass through the drying tower upwards through the desiccants and made to dry. Then the air moves down through the pipe and enters the top of regenerating tower. Now the air is made to pass through the cool desiccants downwards in the regenerating tower and they are made to enter the system. Thus the air is heated in the drying tower and removed from moisture in the regeneration tower and clean air is made to enter into the system.

Absorption dryers or Deliquescent Air Dryers

Absorption: A solid or liquid substance bonds a gaseous substance.

Absorption drying is a purely chemical process. Absorption drying is not of major significance in present-day practice, since the operating costs are too high and the efficiency too low for most applications. Some of the deliquescent dryers used are magnesium chloride or the dehydrated chalk. calcium chloride etc. are used to absorb the moisture in the air. The advantages are low initial cost, low operational cost, less loss in pressure as the air passes from the inlet to the outlet. Oil vapor and oil particles are also separated in the absorption dryer. The moisture in the compressed air forms a compound with the drying agent in the tank. This causes the drying agent to break down: it is then discharged in the form of a fluid at the base of the tank.

SERVICE UNIT

The individual functions of compressed air preparation, i.e. filtering, regulating and lubricating, can be fulfilled by individual components. These functions have often been combined into one unit, i.e. the service unit. Service units are connected upstream of all pneumatic systems.

Generally, the use of a lubricator is not necessary in advanced systems. This is to be used for specific requirements only, primarily in the power section of a system. Compressed air in a control section should not be lubricated.

AIR FILTERS

Condensed water, contamination and excess oil can lead to wear on the moving parts and seals of pneumatic components. These substances can escape as a result of leakage. Without the use of filters, for example, products to be processed in the food, pharmaceutical or chemical industries could become contaminated and therefore rendered useless.

The selection of the correct filter plays an important role in determining the quality and performance of the working system, which is to be supplied with compressed air. One characteristic of compressed-air filters is the pore size. The pore size of the filter element indicates the minimum particle size, which can be filtered out of the compressed air. The collected condensate must be drained before the level exceeds the maximum condensate mark otherwise it will be re-introduced in the air stream.

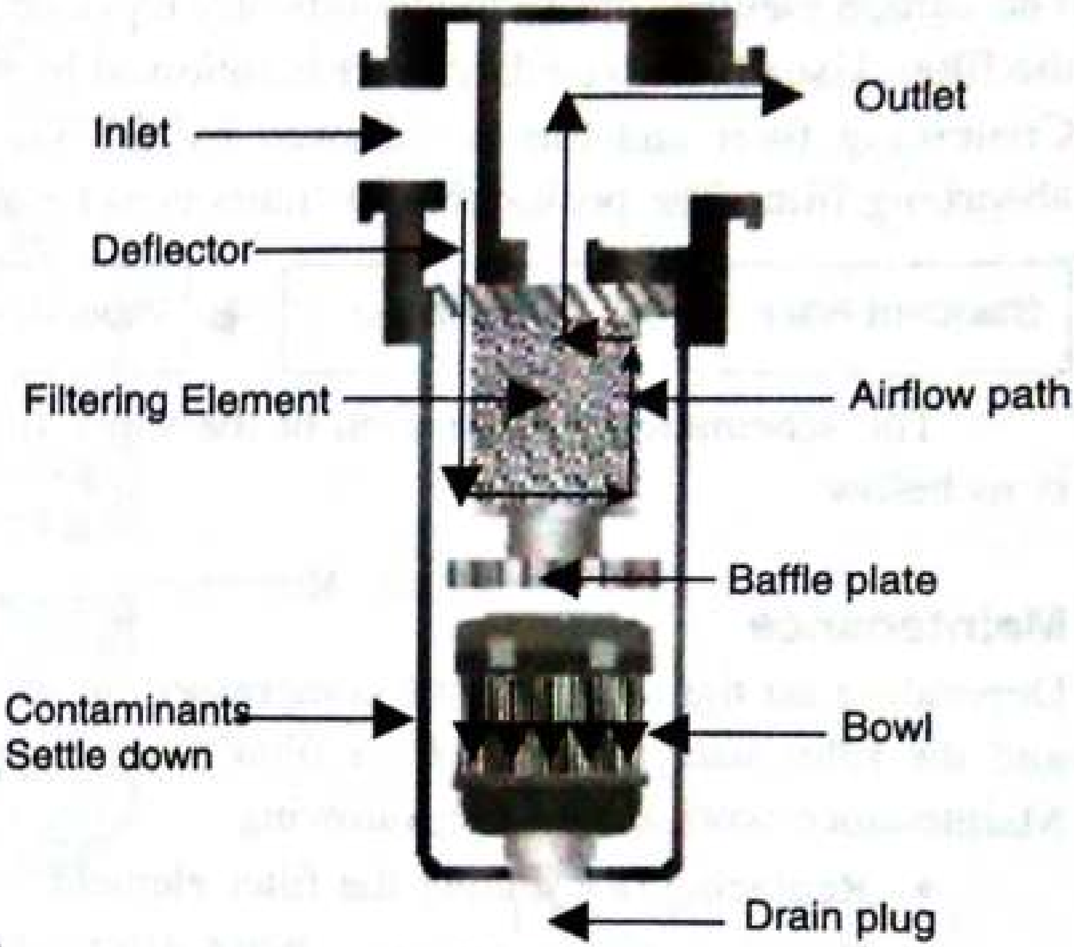
If a large amount of condensate accumulates, it is advisable to fit an automatic drain in place of the manually operated drain cock. However, in such cases, the cause of the accumulated condensate is to be established. For example, an unsuitable pipe layout may be the cause of the condensate accumulation.

The automatic drain uses a float to determine the level of condensate in the bowl and when the limit is reached a control piston opens a valve seat that ejects the condensate under air pressure via a drain line. If the float reaches the minimum level of condensate, the seat valve is closed and the process stopped. The filter bowl can also be emptied manually.

There are three types of filters namely the Standard filter, Coalescing filter and Vapor Absorbing filters.

Standard Filter:

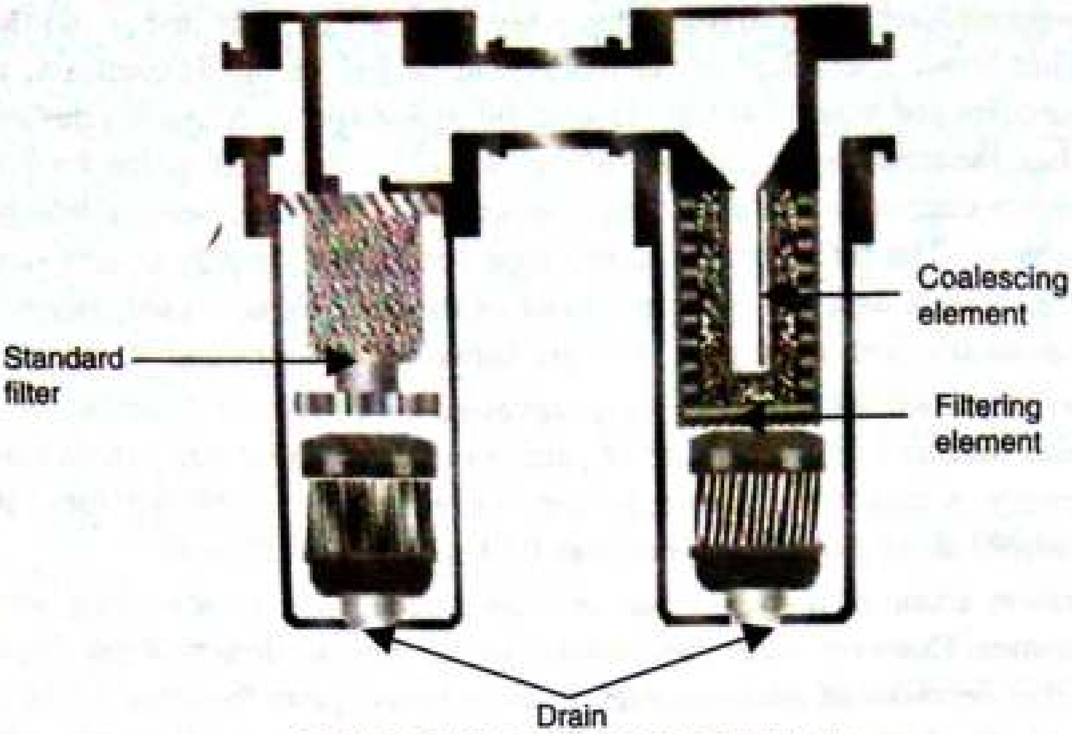
The standard filter has a bowl, inlet and outlet port, a deflector element that deflects the inlet air to enter the filtering element, a baffle plate that separates the contaminants from clean air. As the air passes through the filtering element any foreign particles up to 5 μm is removed and clean air move out of the filter. The moisture collected at the bottom is removed by opening the drain plug.



STANDARD FILTER

Coalescing Filter:

This filter has the similar construction but the air passes through the coalescing element, which can remove major portion of the oil vapor and water moisture mist from the air. This can be placed at the upstream of standard filter. If both these are used in the system it is possible to get clean air into the system. The constructional features of the coalescing filter is described in the below diagram.



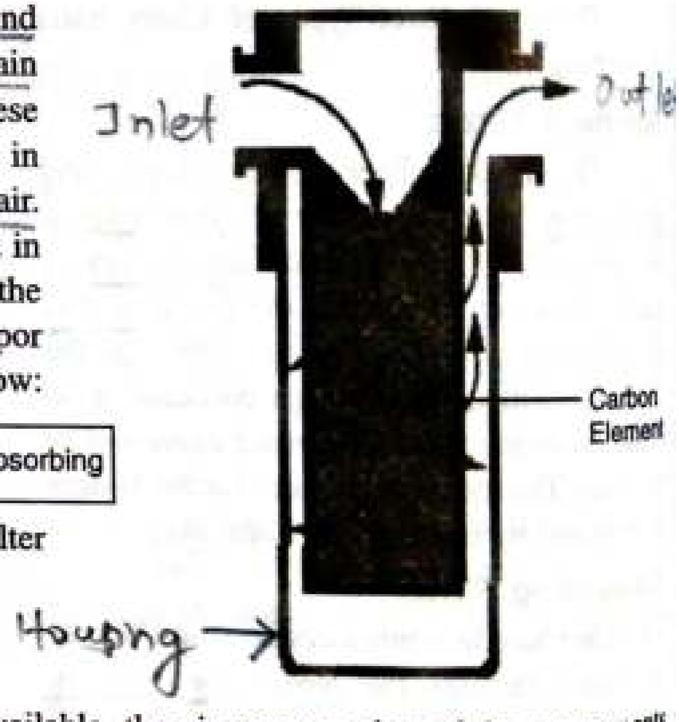
COALESCING AND STANDARD FILTER

Vapor Absorbing Filter:

The vapor-absorbing filter has the housing, inlet and outlet port and carbon filtering element and the drain to remove the oil that is collected at the bottom. These filters are capable of removing only the oil droplets in the air and do not remove water vapor from the air. The carbon element has to be periodically replaced in the filter. Usually the standard filter is followed by the Coalescing filter and this is followed by the Vapor absorbing filter. The positioning of filters is as below:

Standard Filter → Coalescing → Vapor Absorbing

The schematic representation of the vapor filter is as below:



Maintenance

Depending on the nature of the compressed air available, the air consumption of the components and the filter size, compressed-air filter require a greater or lesser amount of maintenance work. Maintenance work means the following:

- Replacing or cleaning the filter element
- Draining the condensate

When cleaning is required, the manufacturer's specifications must be observed concerning the cleaning agents to be used.

AIR REGULATORS

The compressed air generated by the compressor will fluctuate. Changes in the pressure level in the pipe system can adversely affect the switching characteristics of valves, the running times of cylinders and the timing characteristics of flow control and memory valves.

A constant pressure level is thus a prerequisite for the trouble-free operation of a pneumatic control. In order to provide constant pressure conditions, regulators are fitted in a central position in the compressed air network to ensure that there is a constant supply pressure (secondary pressure) irrespective of the pressure fluctuations in the main loop (primary pressure). The pressure reducer or pressure regulator is fitted downstream of the compressed air filter and has the task of keeping the operating pressure constant, regardless of pressure fluctuations or air consumption in the system. The air pressure should be matched to individual requirements upstream of each plant section.

The system pressure which has proved in practice to be the best economic and of technical compromise between compressed-air generation and the efficiency of the components is approximately:

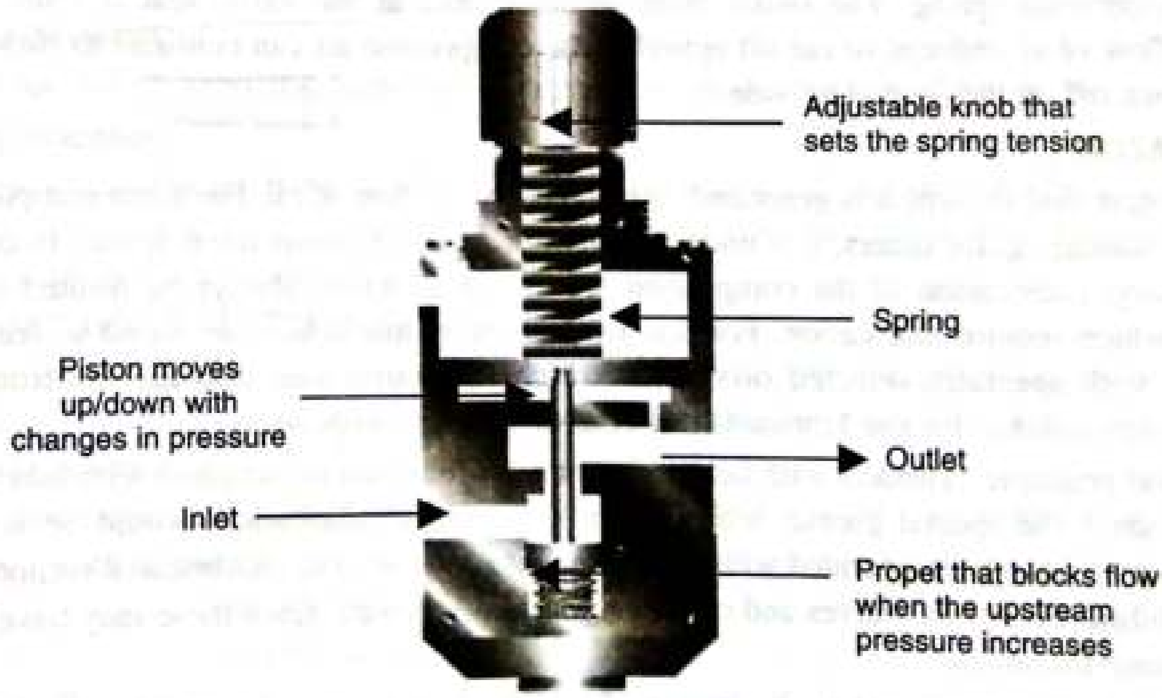
- 600 kPa (6 bar) in the power section and
- 300 to 400 kPa (4 bar) in the control section.

A higher operating pressure would lead to inefficient energy utilization and increased wear, whereas a lower operating pressure would lead to poor efficiency, particularly in the power section.

Operational principle

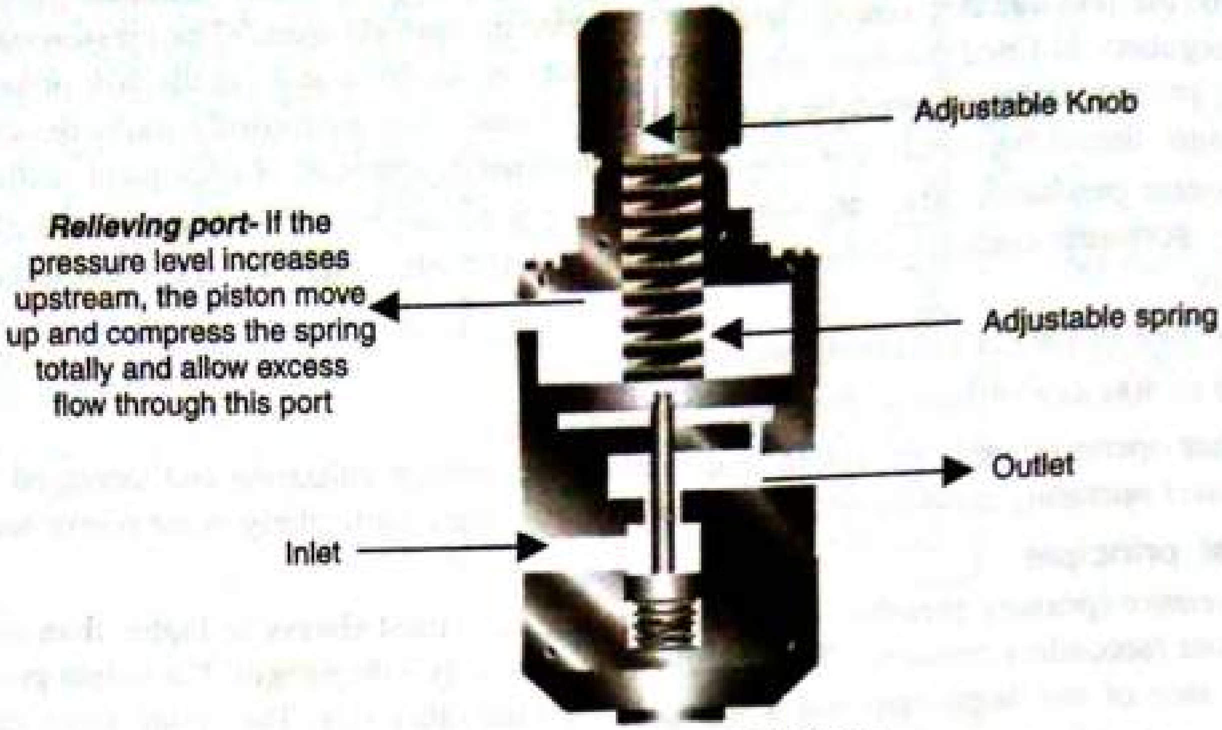
The input pressure (primary pressure) at the pressure regulator must always be higher than out the output pressure (secondary pressure). The pressure is regulated by a diaphragm. The output pressure acts on one side of the diaphragm and a spring acts on the other side. The spring force can be adjusted by means of an adjusting screw.

When the output pressure increase for example during cylinder load changes, the diaphragm moves against the spring force causing the outlet cross-sectional area at the valve seat to be reduced or closed entirely. The centerpiece of the diaphragm then opens and the compressed air can flow to atmosphere through the vent holes in the housing.



NON RELIEVENG AIR REGULATOR

When the output pressure decreases the spring force opens the valve. Regulation of the preset output pressure in thus a continual opening and closing of the valve seat caused by the flow of air. The operating pressure is indicated on a gauge. There are two types are regulators. They are i) Non-relieving and ii) Relieving air regulator. In the non-relieving air regulator as shown in the above diagram the pressure fluctuations are maintained by the adjustable spring force. This requires a strong spring. To eliminate this a relieving regulator as shown below is used:



RELIEVING AIR REGULATOR

In this type the excess air pulsations if not equaled by the spring is relieved through the Relieving port.

Operational principle

If no air is drawn off on the secondary side, the pressure rises and presses the diaphragm against the compression spring. The outlet cross-sectional area at the valve seat is reduced or closed and the flow of air reduced or cut off entirely. The compressed air can continue to flow only when air is drawn off on the secondary side.

AIR LUBRICATOR

As a rule the compressed air, which is generated, should be dry, i.e. free of oil. For some components lubricated air is damaging, for others, it is undesirable, but for power components it may in certain cases be necessary. Lubrication of the compressed air should therefore always be limited to the plant sections, which require lubrication. For this purpose, mist lubricators are fitted to feed the compressed air with specially selected oils. Oils, which are introduced into the air from the compressor, are not suitable for the lubrication of control system components.

As a general principle cylinders with heat-resistant seals must not be supplied with lubricated compressed air, since the special grease, which forms the original lubrication, would be washed out. If systems, which have been operated with lubrication, are converted to unlubricated compressed air, the original lubrication of the valves and cylinders must be renewed, since these may have been flushed out in some instances.

Lubrication of the compressed air by means of mist lubricators may be necessary in certain cases:

(*i*) Where extremely rapid oscillating motions are required

(*ii*) With cylinder of large diameter, lubricators should where possible be installed only directly upstream of the consuming cylinders

The following problems may occur as a result of excessive lubrication:

- Malfunction of components.
- Increased environmental problems
- Seizing of components after prolonged downtime

Operational principle

The compressed air passing through the lubricator causes a pressure drop between the oil reservoir and the upper part of the lubricator. The pressure difference is sufficient to force the oil upwards through a duct, where it then drips into a nozzle, which can be seen through an inspection glass. Here the oil is atomized and taken up by the air stream to a greater or lesser extent. Thus the oil carried with the compressed air into the system helps to lubricate the system components.

The FILTER REGULATOR LUBRICATOR (FRL) is a trio used in all the pneumatic systems and as a service unit they come as a combinational unit. The simple representative form of this FRL is shown in the preceeding diagram for easy understanding.

The following points should be observed in everyday practice:

- As far as possible compressor oils should be prevented from entering the compressed-air network (oil separators should be fitted)
- For operation fit components which can also function with nonlubricated compressed air
- Once a system has been operated and run-with oil, the lubrication must be continued since the original lubrication of the components will have been flushed away by the oil.

Maintenance of air service units

The following routine service measures are necessary on a regular basis.

Air filter:

The condensate level must be checked regularly, as the height specified on the sight glass must not be exceeded. The accumulated condensate could otherwise be drawn into the compressed air pipelines. The drain screw on the bowl must be opened to drain the condensate. The filter cartridge in the filter must also be cleaned if it is dirty.

Air regulator:

This requires no servicing, provided it is preceded by a compressed air filter.

Air lubricator:

If fitted check the oil level in the sight glass and top up, if necessary, to the level indicated. The plastic filter and lubricator bowl must not be cleaned with trichlorothylene. Only mineral oils may be used for the lubricator.

ACTUATORS AND OUTPUT DEVICES

An actuator is an output device for the conversion of supply energy into useful work. The output signal is controlled by the control system, and the actuator responds to the control signals via the control element.

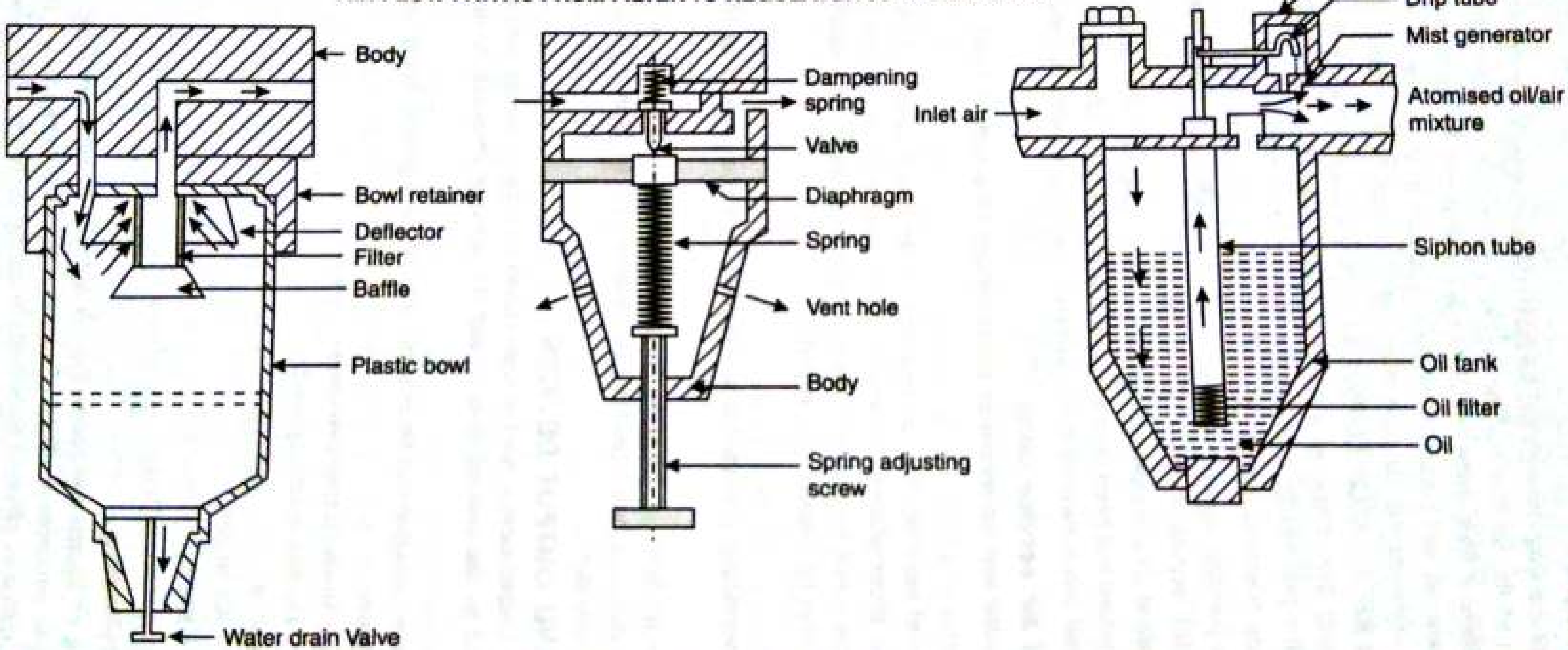
The pneumatic actuator can be described under two group, linear and rotary:

- Linear motion
  - Single-acting cylinders
  - Double-acting cylinders
- Rotary motion
  - Air motor
  - Rotary cylinders
  - Rotary actuator

Single-acting cylinders

With single-acting cylinders compressed air is applied on only one side of the piston face. The other side is open to atmosphere. The cylinder can produce work in only one direction. The return movement of the piston is effected by a built-in spring or by the application of an external force. The spring force of the built-in spring is designed to return the piston to its start position with a reasonably high speed under no load conditions.

AIR FLOW PATH IS FROM FILTER TO REGULATOR TO LUBRICATOR



FILTER REGULATOR LUBRICATOR- FRL TRIO

ACTUATORS AND OUTPUT DEVICES

An actuator is an output device for the conversion of supply energy into useful work. The output signal is controlled by the control system, and the actuator responds to the control signals via the control element.

The pneumatic actuator can be described under two group, linear and rotary:

- Linear motion
  - Single-acting cylinders
  - Double-acting cylinders
- Rotary motion
  - Air motor
  - Rotary cylinders
  - Rotary actuator

Single-acting cylinders

With single-acting cylinders compressed air is applied on only one side of the piston face. The other side is open to atmosphere. The cylinder can produce work in only one direction. The return movement of the piston is effected by a built-in spring or by the application of an external force. The spring force of the built-in spring is designed to return the piston to its start position with a reasonably high speed under no load conditions.

For single-acting cylinders with built-in spring, the stroke is limited by the natural length of the spring. Single-acting cylinder are therefore only available in stroke lengths of up to approximately 80 mm. The construction and simplicity of operation of the single-acting cylinder make it particularly suitable for compact, short stroke length cylinder for the following types of applications:

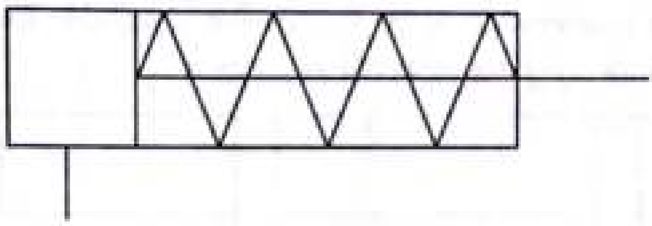
- Transferring
- Branching
- Converging
- Allocating
- Clamping
- Ejecting

Construction

The single-acting cylinder has a single piston seal, which is fitted on the air supply side. Sealing is by a flexible material that is embedded in a metal or plastic piston. During motion, the sealing edges slide over the cylinder-bearing surface. There are varying designs of single-acting cylinders including:

- Diaphragm cylinder
- Rolling diaphragm cylinder

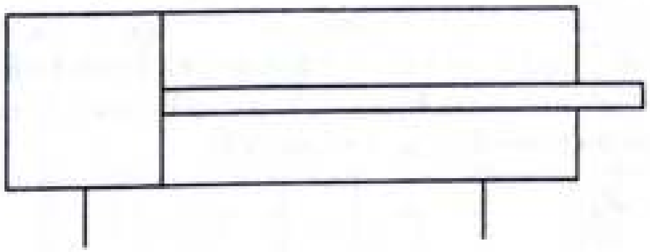
With a diaphragm cylinder, a built-in diaphragm made of rubber, plastic or metal performs the task of the piston. The piston rod is mounted centrally on the diaphragm. There is no sliding seal, but merely friction as a result of the tensile stress of the diaphragm. They are used in short stroke applications, for clamping, embossing and lifting operations.



SINGLE ACTING CYLINDER

Double-acting cylinders

The construction principle of a double-acting cylinder is similar to that of the single-acting cylinder. However, there is no return spring, and the two ports are used alternatively as supple and exhaust ports. The double-acting cylinder has the advantage that the cylinder is able to carry out work in both direction of motion. Thus, installation possibilities are universal. The force transferred by the piston rod is somewhat greater for the forward stroke than for the return stroke as the effective piston surface is reduced on the piston rod size by the cross-sectional area of the piston rod.



DOUBLE ACTING CYLINDER

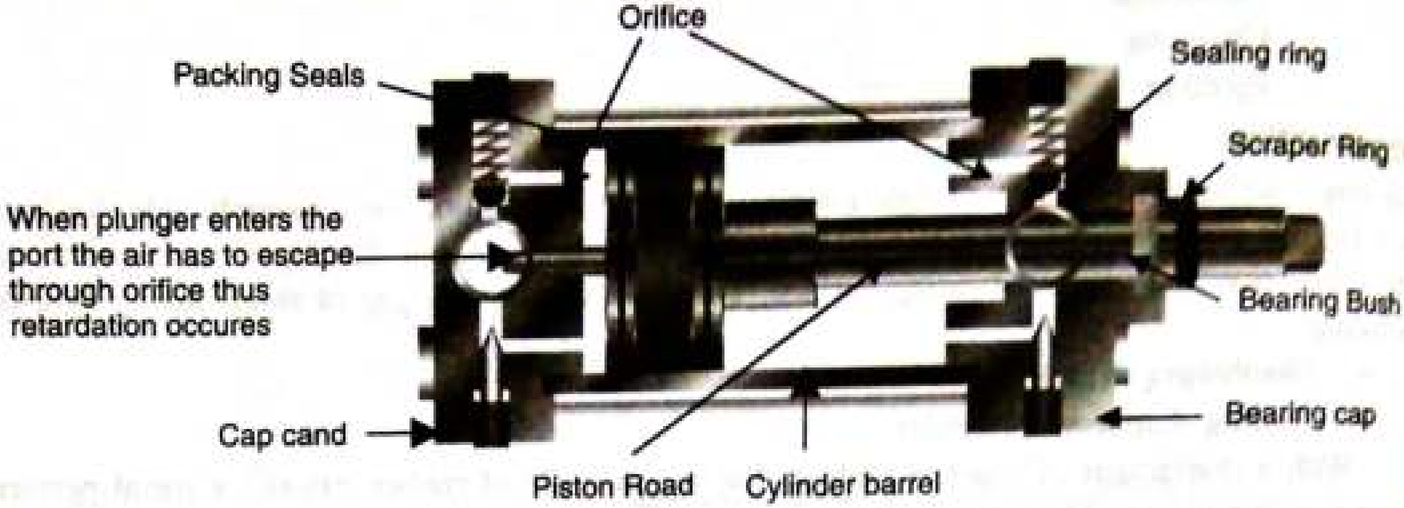
Cylinder with end position cushioning

If a cylinder moves large masses, cushioning is used in the end positions to prevent sudden damaging impacts. Before reaching the end position, a cushioning piston interrupts the direct flow path of the air to the outside. Instead a very small and often adjustable exhaust aperture is open. For the last part of the stroke the cylinder speed is progressively reduced. If the passage adjustment is too small, the cylinder may not reach the end position due to the blockage of air. With very large forces and high acceleration extra measures must be taken such as external shock absorbers to assist the load deceleration.

To achieve correct declaration:

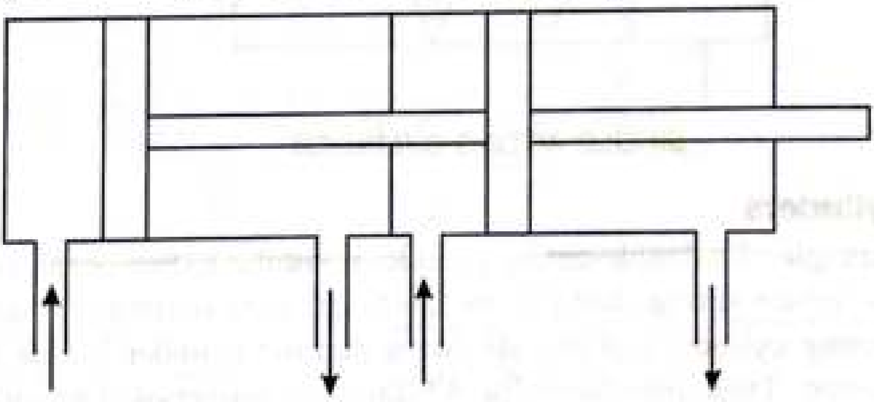
- The regulating screw should first be screwed in fully and
- Backed off in order to allow the adjustment to be increased slowly to the optimum value.

The below figure shows the tie rod cylinder with end cushions:



Tandem double-acting cylinder

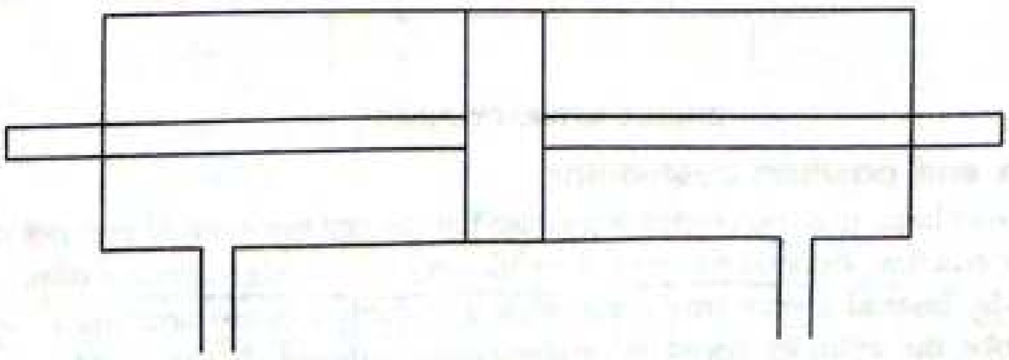
The tandem cylinder incorporates the features of two double-acting cylinders, which have been joined to form single unit. By this arrangement and with the simultaneous loading of both pistons, the force on the piston rod is almost doubled. This design is suitable for such applications where a large force is required but the cylinder diameter is restricted.



TANDEM CYLINDER

Cylinder with through piston rod (Double Rod Cylinder)

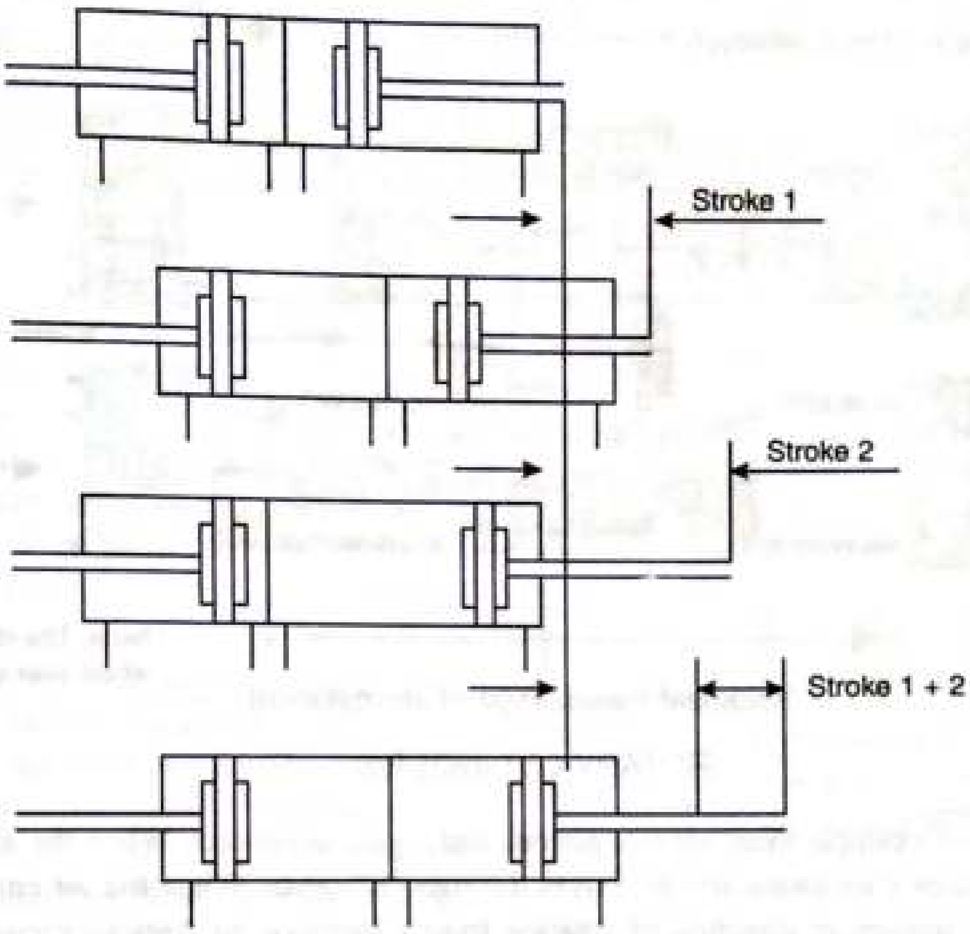
This cylinder has a piston rod on both sides, which is a through piston rod. The guide of the piston rod is better, as there are two bearing points. The force is identical in both direction of movement. The through piston rod can be hollow, in which case it can be used to conduct various media, such as compressed air. A vacuum connection is also possible.



DOUBLE ROD CYLINDER/CYLINDER WITH THROUGH PISTON ROD

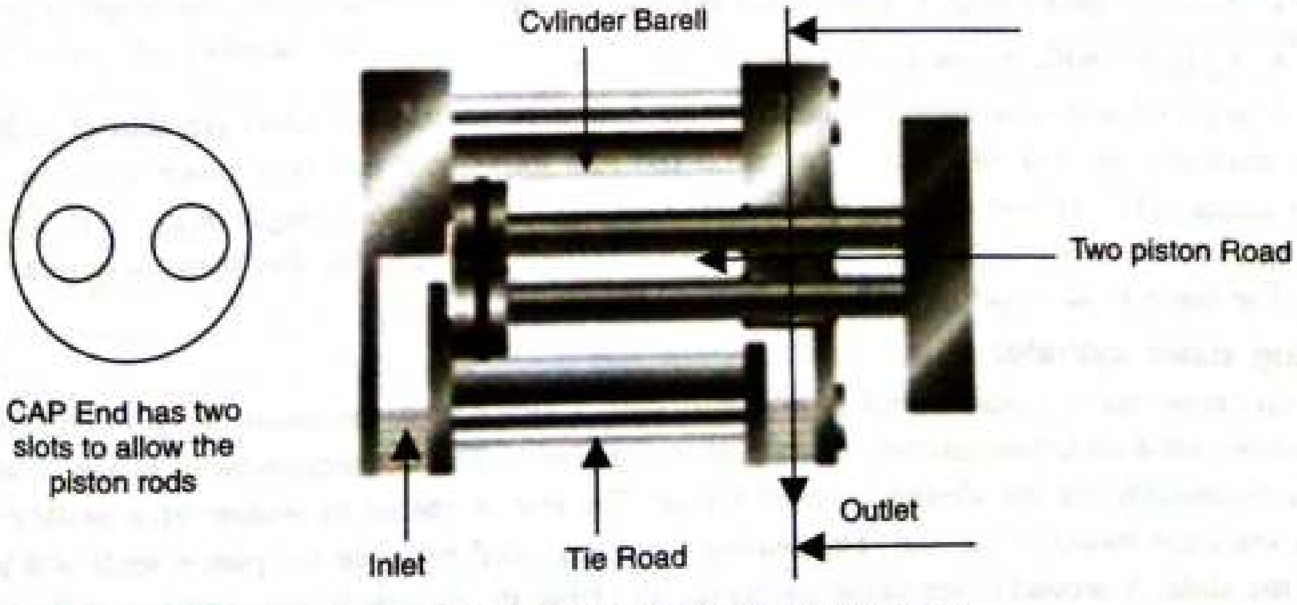
Multi-position cylinders

The multi-position cylinder consists of two or several double-acting cylinders, which are interconnected. The individual cylinder advance when pressure is applied. In the case of two cylinders with different stroke lengths, four positions are obtained. The various strokes that can be obtained by using two double acting cylinders to get multi-positional cylinder is shown in the below figure.



Double Rod Non-Rotating Cylinders:

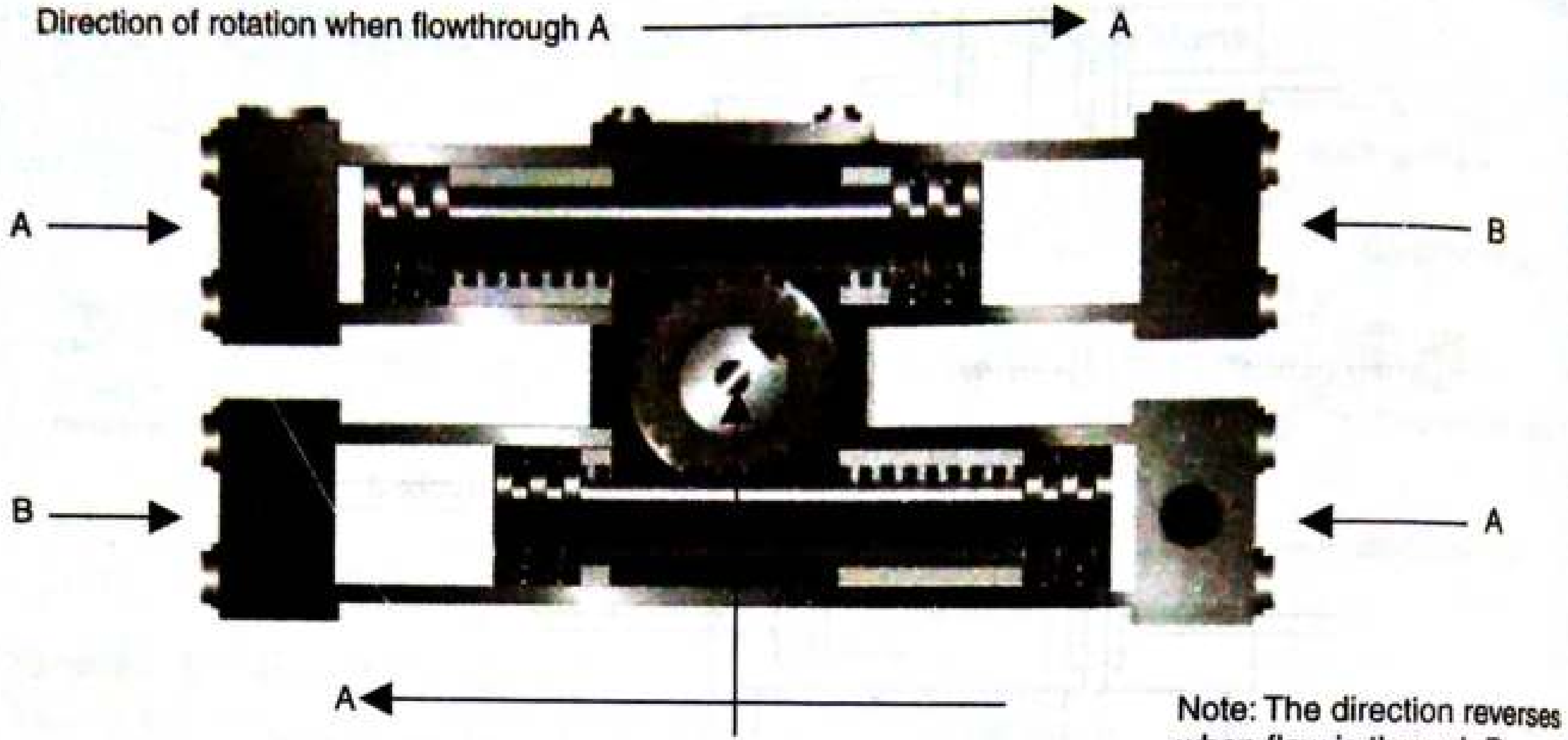
In certain applications the rotation of piston and piston rod is not permitted. In such cases instead of having a single rod from the piston, double rod is used and the rod side cap end is provided with holes to accommodate the rods. This helps in preventing the piston and rod from rotating.



DOUBLE ROD NON-ROTATING CYLINDER

Rotary Cylinders

With this design of double-acting cylinder, the piston rod has a gear tooth profile. The piston rod drives a gear wheel, and a rotary movement results from a linear movement. The range of rotation varies from 45°, 90°, 180°, and 270° to 360°. The torque is dependent on pressure, piston surface and gear ratio.



Rack and Pinion Type of arrangement

ROTARY CYLINDERS

In the above example there are two pistons and a gear in contact. When the air enters through "A" the gear rotates Clockwise and air leaves through "B" ports. When the air enters through "B" ports the gears changes in direction of rotation from Clockwise to Anticlockwise and air leaves through "A" ports.

MOTORS

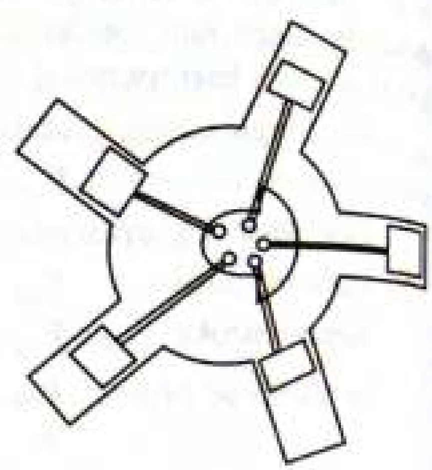
Devices, which transform pneumatic energy into mechanical rotary movement with the possibility of continuous motion, are known as pneumatic motors. The pneumatic motor with unlimited angle of rotation has becomes one of the most widely used working elements operating on compressed air. Pneumatic motors are categorized according to design:

- Piston motors
- Sliding-vane motors
- Gear motors
- Turbines (high flow)

Piston motors

This type of design is further subdivided into radial and axial piston motors. The crankshaft of the motor is driven by the compressed air via reciprocating pistons and connecting rods. To ensure smooth running several pistons are required. The power of the motor depends on input pressure, number of pistons, piston area, stroke and piston speed.

The working principle of the axial piston motor is similar to that of the radial piston motor. The force from 5 axially arranged cylinders is converted into a rotary motion via a swash plate. Compressed air is applied to two pistons simultaneously, the balanced torque providing smooth running of the motor. These pneumatic motors are available in clockwise or anti-clockwise rotation. The maximum speed is around 5000 rpm, the power range at normal pressure being 1.5 - 19kW (2-25 hp).



AIR PISTON MOTOR-RADIAL TYPE

Sliding vane motors

Because of their simple construction and the low weight, sliding vane motors are used for hands tools. An eccentric rotor is contained in bearing in a cylindrical chamber. Slots are arranged in the rotor. The vanes are guided in the slots of the rotor and forced outwards against the inner wall of the cylinder by centrifugal force. With other designs, the vanes are moved via springs. This ensures that the individual chambers are sealed. The rotor speed is between 3000 and 8500 rpm. Here too, clockwise or anti-clockwise units are available. Power ranges available are 0.1-17 KW (0.14 -24 hp).

Gear motors

In this design, torque is generated by the pressure of the air against the teeth profiles of two meshed gear wheels. One of the gear wheels is secured to the motor shaft. Gear motors are produced with spur or helical gearing. These gear motors are used in applications with a very high power rating (up to 44 kW/60 hp). The direction of rotation is also reversible for these motors.

Turbines (flow motors)

Turbine motors can be used only where a low power is required. The speed range is very high. For example, the Dentists' air drill operates at 500,00 rpm. The working principle is the reverse of the flow compressor.

Characteristics of pneumatic motors are:

(i) Smooth regulation of speed and torque	(ii) Small size (weight)
(iii) Overload safe	(iv) Insensitive to dust, water, heat, cold
(v) Explosion proof	(vi) Large speed selection
(vii) Maintenance minimal	(viii) Direction of rotation easily reversed

DIRECTIONAL CONTROL VALVES

Configuration and construction

Directional control valves are devices, which influence the path taken by an air stream. Normally this involves one or all of the following:

- Opening the passage of air and directing it to particular air lines,

- Canceling air signals as required by blocking their passage and/or
- Relieving the air to atmosphere via an exhaust port.

The directional control valve is characterized by

(i) Its number of controlled connections or ways,

(ii) The number of switching positions and

(iii) The method of actuation.

However, these symbols do not provide any information about the constructional design, but merely indicate the function of the valve.

The **normal position** on valves with existing reset, e.g. spring, refers to the switching position assumed by the moving parts of the valve, if the valve is not connected. The **initial position** is the switching position assumed by the moving parts of a valve after the valve has been installed in a system and the system pressure has been switched on and possibly also the electrical voltage, and with which the designated switching program starts.

The constructional principle of a directional control valve is an important factor as far as the service life, switching time, type of actuation, connection methods and size are concerned.

Design are categorized as follows:

- Poppet valves:
  - -Ball seat valve
  - -Disc seat valve
- Slide valves:
  - -Longitudinal slide valve (spool valve)
  - -Longitudinal flat slide valve
  - -Plate slide valve

Poppet valves

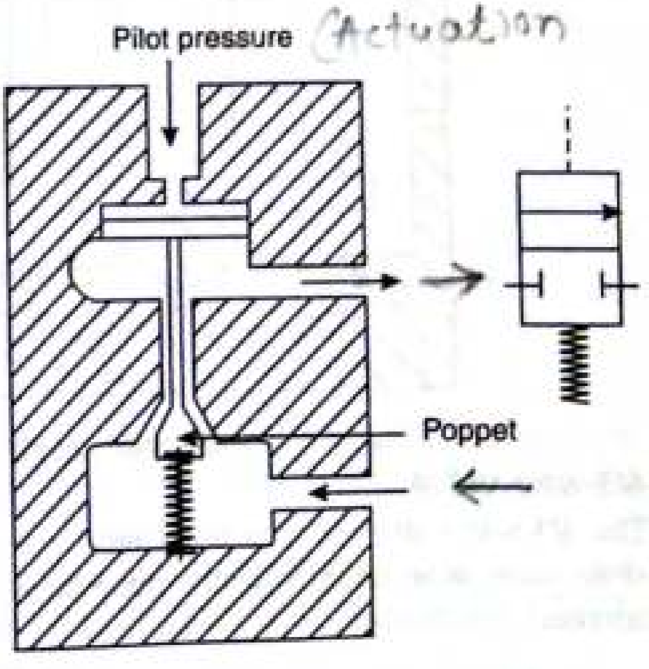
With poppet valves the connections are opened and closely by means of balls, discs, plates or cones. The valve seats are usually sealed simply using flexible seals. Seat valves have few parts, which are subject to wear, and hence they have a long service life. They are insensitive to dirt and are robust. The actuation force, however, is relatively high as it is a necessary to overcome the force of the built-in reset spring and the air pressure.

Slides valves

In slide valves, the individual connections are linked together or closed by means of spools, flat slide or plate slide valves.

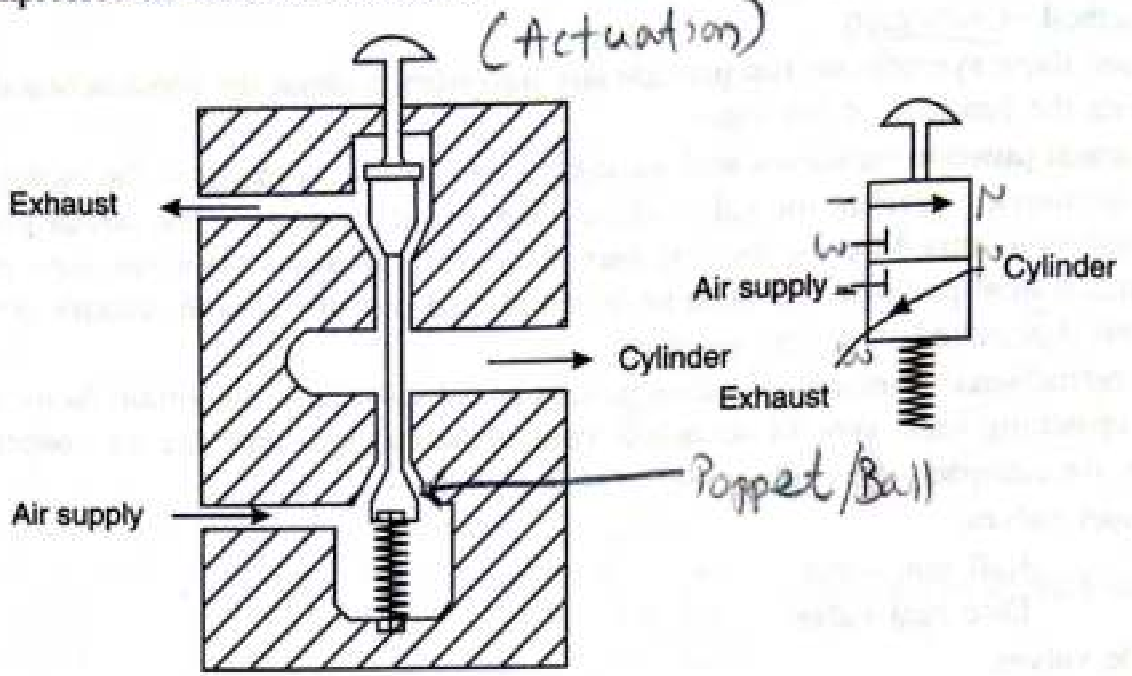
2/2-way valve

The 2/2-way valve has two ports and two positions (open, closed). It is rarely used except as an on-off valve, since its only function is to enable signal flow through and cannot release the air to atmosphere once in the closed position in contrast to the 3/2-way valve. The 2/2-way valve is normally of the ball seat construction. This valve can be operated manually, mechanically or pneumatically.



3/2-way valve

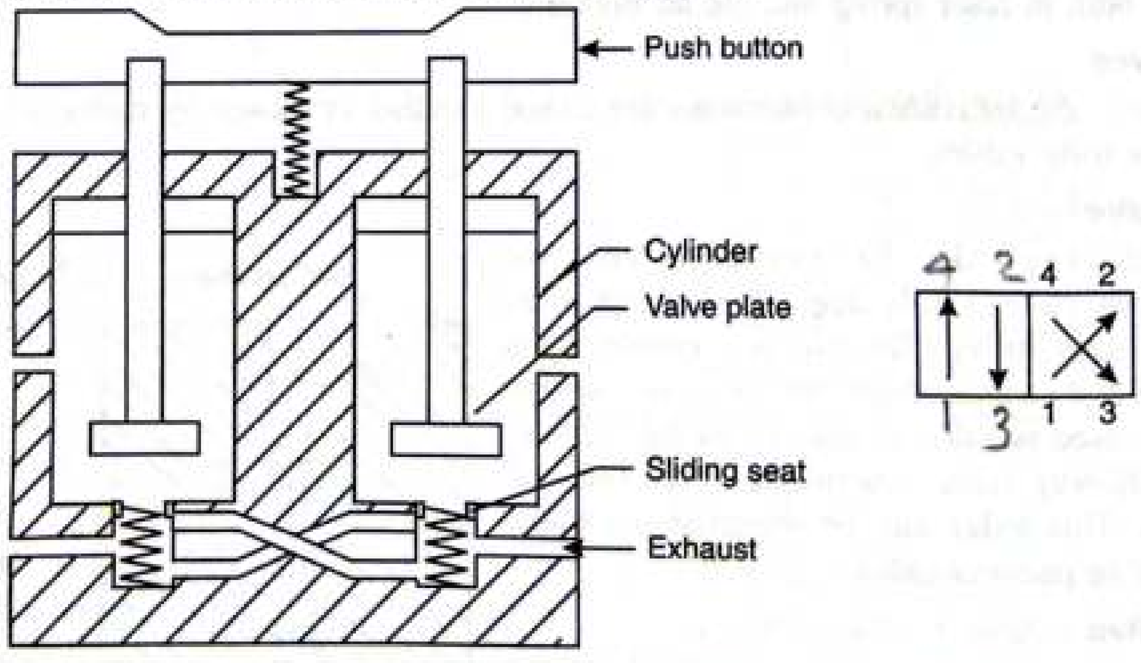
The 3/2-way valve is a signal-generating valve, with the characteristic that a signal on the output side of the valve can be generated and also cancelled. The 3/2-way valve has three ports and two positions. The addition of the exhaust port enables the signal generated via the passage through the 3/2-way valve to be cancelled. The valve connects the output signal to exhaust and thus to atmosphere in the initial position. A spring forces a ball against the valve seat preventing the compressed air from flowing the air connection to the working line. Actuation of the valve plunger causes the ball to be forced away from the seat. In doing this, the opposing force of the reset spring and that generated from the compressed air must be overcome.



3/ 2 WAY VALVE

4/2-way valve

The 4/2-way valve has four ports and two positions. A disc-set 4/2-way valve is similar in construction to the combination of two 3/2-way valves, one valve normally closed and the other normally open. The valve has a non-overlapping exhaust connection and is returned to its start position by the spring. The valves are used for controls employing double-acting cylinders. The actuating methods and types of construction available for the 4/2-way valve include push button, single air pilot, double air pilot, roller lever actuated, spool and sliding plate.



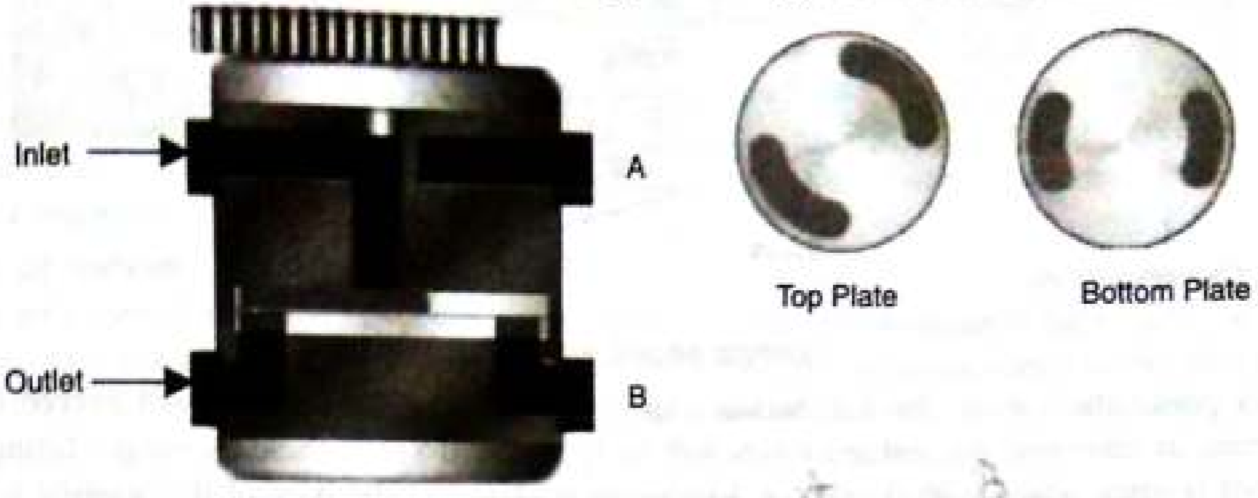
4/ 2 WAY VALVE

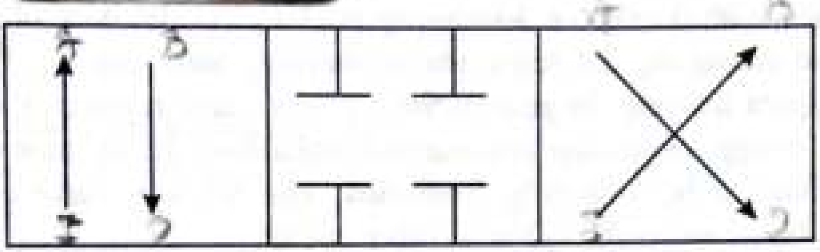
4/3-way valve

The 4/3-way valve has four ports and three positions. An example of the 4/3 way valve is the plate slide valve with hand or foot actuation. By turning two discs, channels are connected with one another.

In the circuits using these valves, the lines of the 4/3-way valve are closed in the middle position. This enables the piston rod of a cylinder to be stopped in any position over its stroke range, although intermediate positions of the piston rod cannot be closed be located with accuracy. Owing to the compressibility of air, another position will be assumed if the load on the piston rod changes.

The ANSI symbol and the rotational disc type 4/3- way DCV is shown in the below diagram.

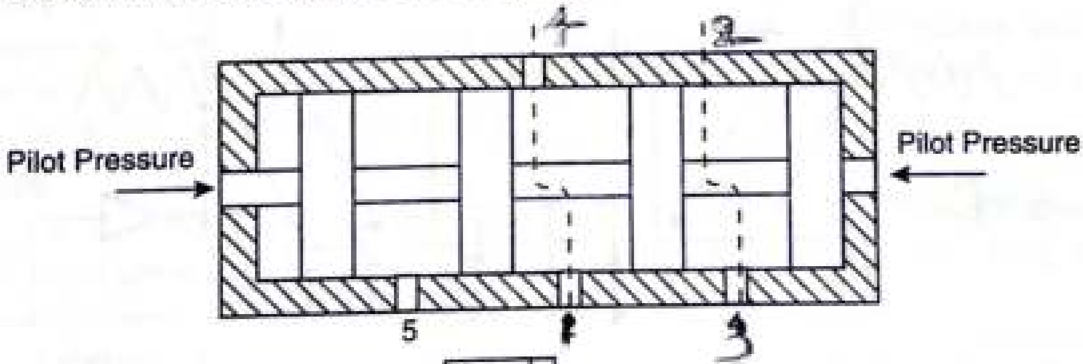


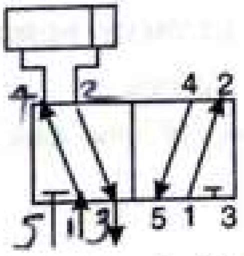


4/3 Way dcv-The Method of Actuation can be any one of the types

5/2-way valve

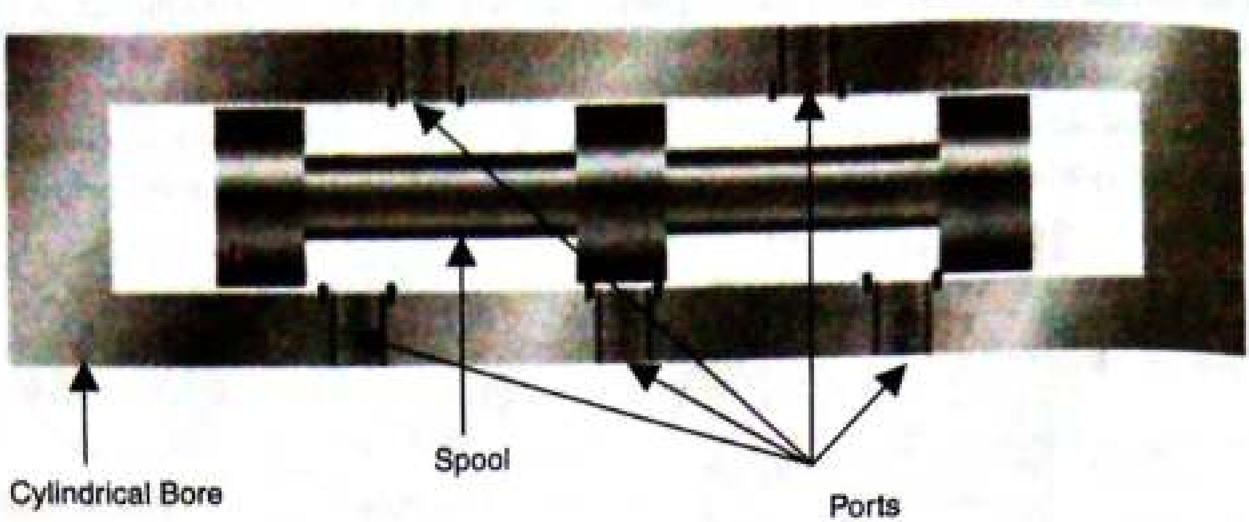
The 5/2-way valve has five ports and two positions. The 5/2-way valve is used primarily as a control element for the control of cylinders. An example of the 5/2-way valve, the longitudinal slide valve, uses a pilot spool as a control component. This connects or separates the corresponding lines by means of longitudinal movements. The required actuating force is lower because there are no opposing forces due to compressed air or spring.





All forms of actuation can be used with longitudinal slide valves, i.e. manual, mechanical, electrical or pneumatic. These types of actuation can also be used resetting the valve to its starting position.

The actuation travel is considerable larger than with seat valves. Sealing presents a problem in this type of slide valve. The type of fit known in hydraulics as metal to metal (lapped spool), requires the spool to fit precisely in the bore of the housing.

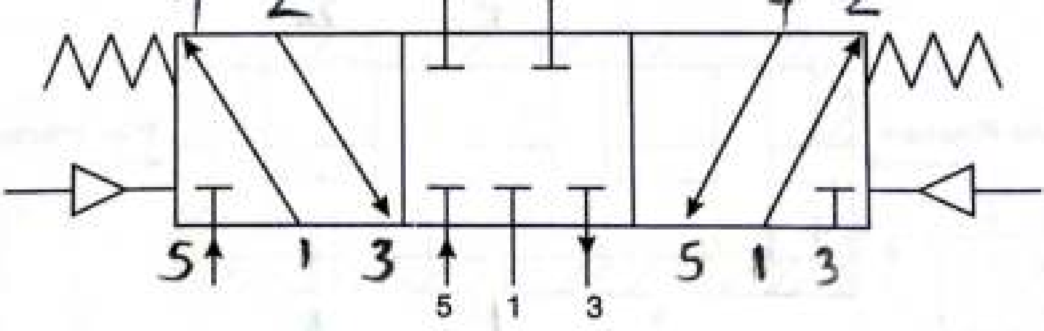


LAPPED SPOOL 5/2 WAY DCV

In pneumatic valves, the gap between spool and housing bore should not exceed 0.002 - 0.004 mm, as otherwise the leakage losses will be too great. To save these expensive fitting costs, the spool is often sealed with 0-rings or double-cup packings or the bore of the housing is sealed with 0-rings. To avoid damaging the seals, the connecting ports can be distributed around the circumference of the spool housing. In general the 5/2-way valve replaces the 4/2-way valve. The 5/2-way valve has advantages in passage construction and allows the exhaust of both extension and retraction air for cylinder to be separately controlled. The 5/2-way valve circuit carries out the same primary control functions as the 4/2-way valve circuit.

5/3-way valve

The 5/3-way valve has five working ports and three switching positions. With these valves, double-acting cylinders can be stopped within the stroke range. This means a cylinder piston under pressure in mid-position is briefly clamped in the normally closed position and in the normally open position the piston can be moved unpressurised. If no signals are applied at either of the two control ports, the valve remains springs-centered in mid position.

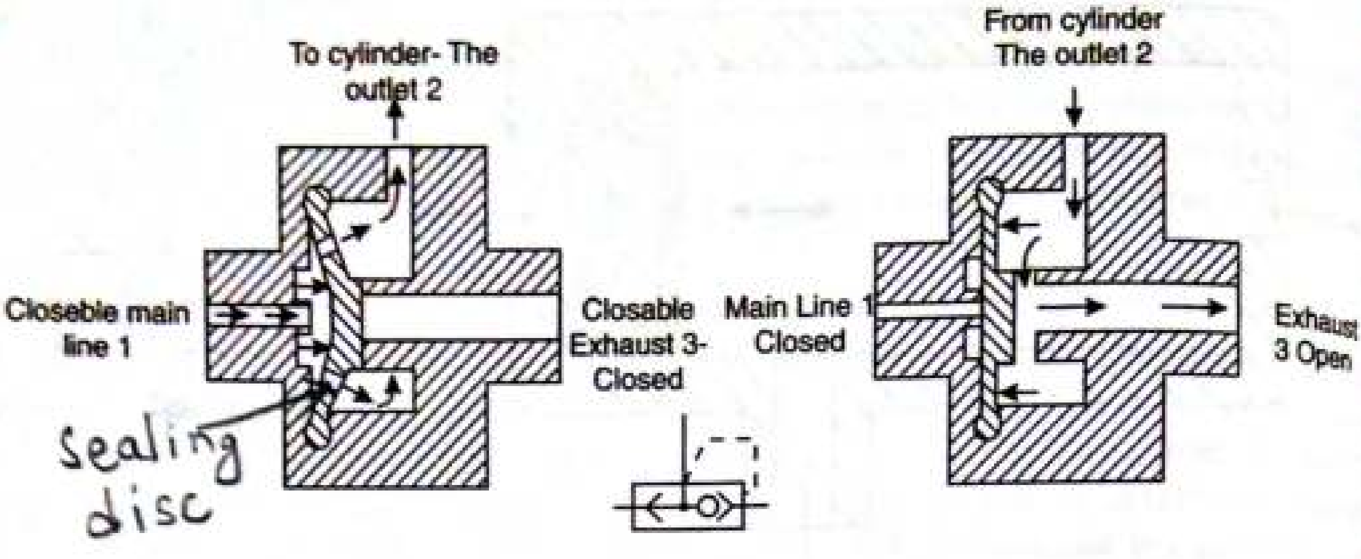


5 /3 WAY DOUBLE PILOT VALVE- NORMALLY CLOSED

Quick Exhaust Valve

Quick-exhaust valves are used to increase the piston speed of cylinders. This enables lengthy return times to be avoided, particularly with single-acting cylinders. The principle of operation is to allow the cylinder to retract at its near maximum speed by reducing the resistance to flow of the exhausting air during motion of the cylinder. To reduce resistance, the air is expelled to atmosphere close to the cylinder via a large orifice opening. The valve has a closable supply connection 1,a closable exhaust 3 and an outlet 2.

If pressure is applied at port 1, then the sealing disc covers the exhaust 3, whereby the compressed air passes from 1 to 2. If pressure is no longer applied at 1, then the air from 2 moves the sealing disc against port 1 and closes this, whereby the exhaust air immediately vents to atmosphere. There is no need for the air to pass through a long and possibly restricted path to the directional control valve via the connecting lines. It is advantageous to mount the quick-exhaust valve directly on the cylinder or as near to it as possible.



QUICK EXHAUST VALVE

Shut –off Valves:

Shut-off valves are non-adjustable valves, which release or shut off flow in both directions. Typical examples are the stopcock and ball cock.

FLOW CONTROL VALVES

Flow control valves influence the volumetric flow of the compressed air in both directions. The throttle valve is a flow control valve.

Throttle valve, bi-directional

Throttle valves are normally adjustable and the setting can be locked in position. Throttle valves are used for speed control of cylinders. Care must be taken that the throttle valve does not close fully, cutting off air from the system.

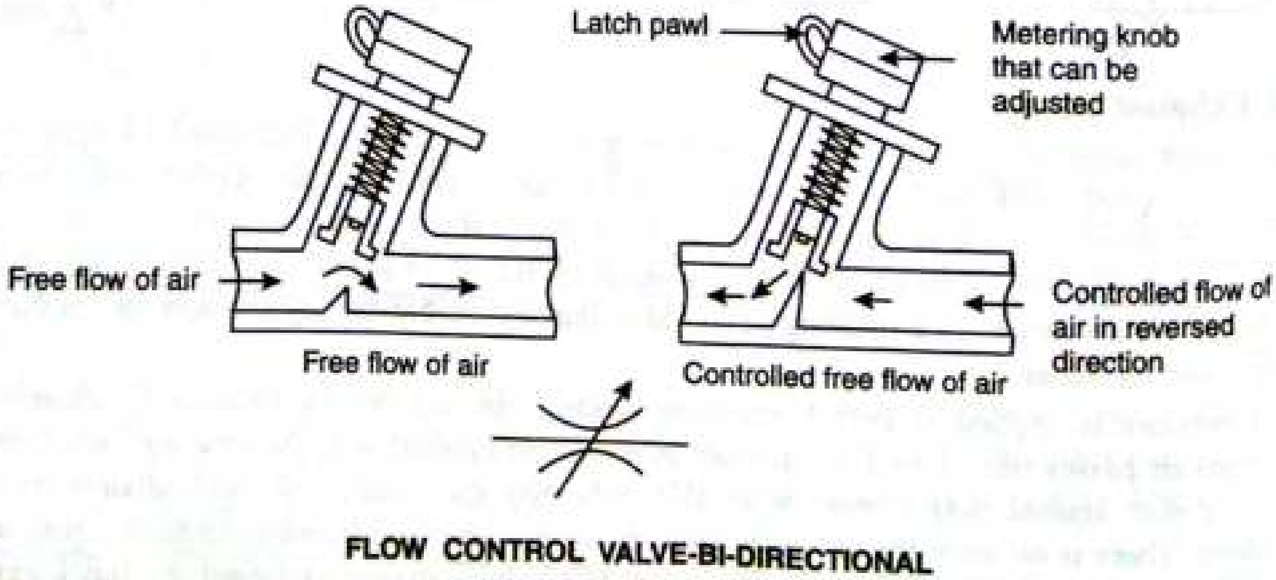
Characteristics of flow control valves according to construction principles.

Throttle valve:

In the throttle valve, the length of the throttling section is greater than its diameter.

Diaphragm valve:

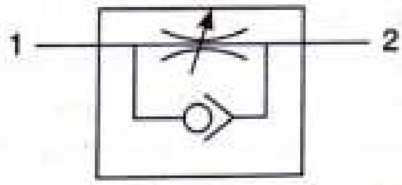
In the diaphragm valve, the length of the throttling section is less than its diameter.



FLOW CONTROL VALVE-BI-DIRECTIONAL

One-way flow control valve

In the case of the one-way flow control valve, the airflow is throttled in one direction only. A check valve blocks the flow of air in the bypass leg and the air can flow only through the regulated cross-section. In the opposite direction, the air can flow freely through the opened check valve. These valves are used for speed regulation of actuators and if possible should be mounted directly on the cylinder.



ONE-WAY FLOW CONTROL VALVE

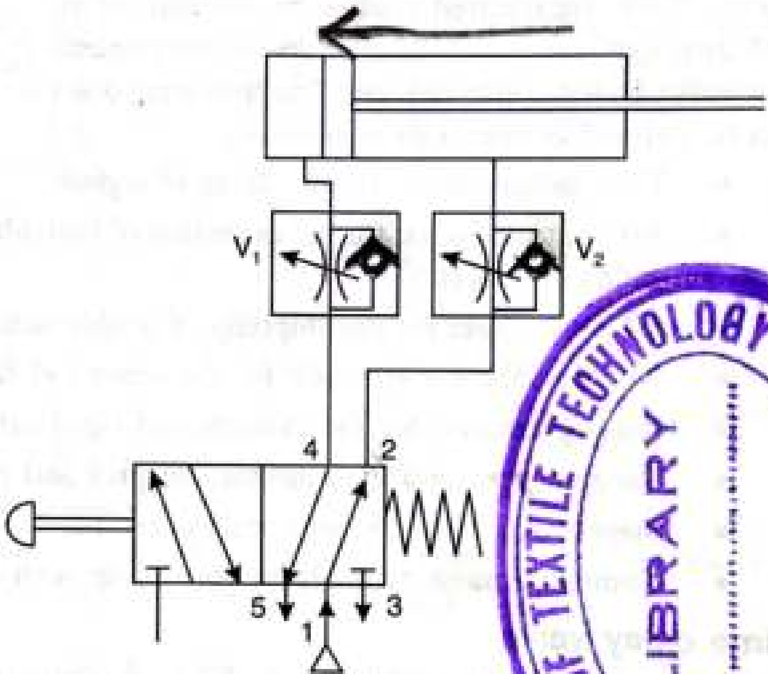
Fundamentally, there are two types of throttling circuits for double-acting cylinders:

- Supply air throttling
- Exhaust air throttling

Supply air throttling

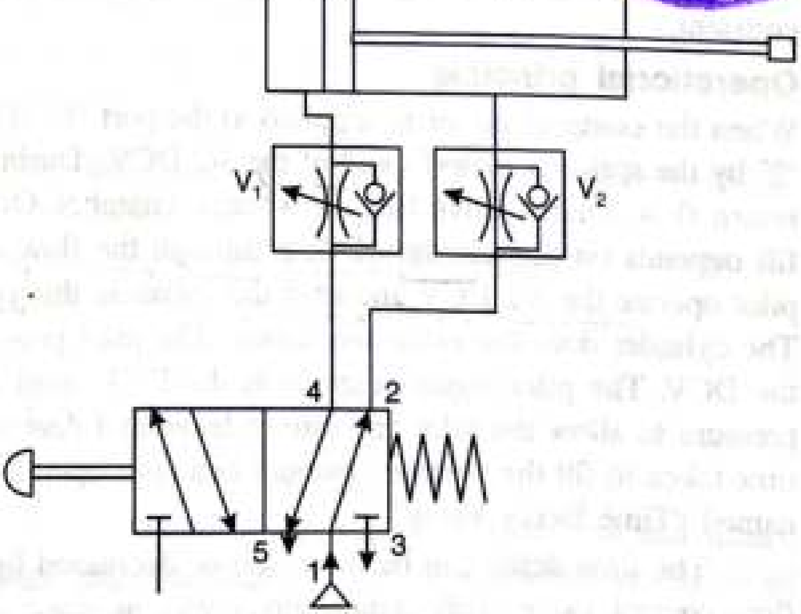
For supply air throttling, one-way flow control valves are installed so that the air entering the cylinder is throttled. The exhaust air can escape freely through the check valve of the throttle valve on the outlet side of the cylinder. The slightest fluctuation in the load on the piston rod, such as for example when passing a limit switch, lead to very large irregularities in the feed speed.

A load in the direction of movement of the cylinder accelerates the cylinder beyond the set value. Therefore supply air throttling can be used for single-acting and small volume cylinders.



Exhaust air throttling

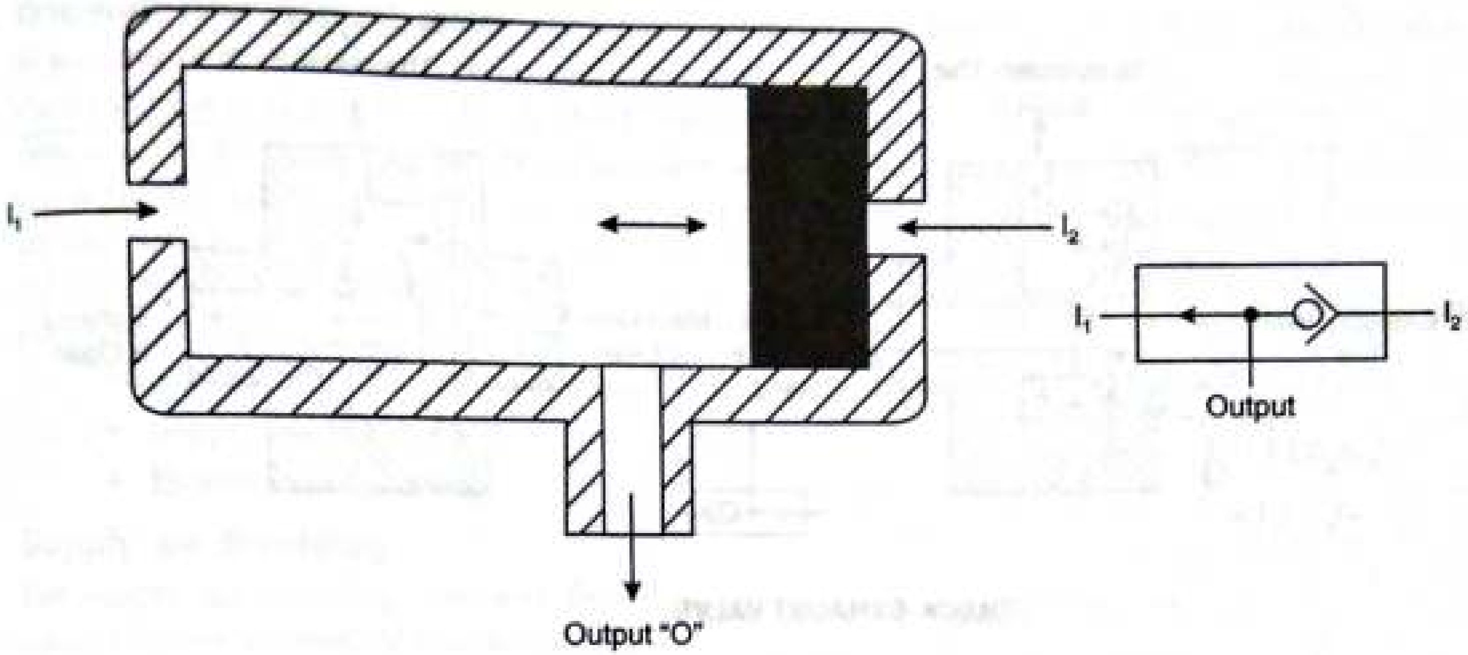
With exhaust air throttling, the supply air flows freely to the cylinder and the exhaust air is throttled. In this case, the piston is loaded between two cushions of air. The first cushion effect is the supply pressure to the cylinder and the second cushion is the exhausting air being restricted at the one-way flow control valve orifice. Arranging throttle relief valves in this way contributes substantially to the improvement of feed behaviour. Exhaust air throttling should be used for double-acting cylinder. In the case of miniature cylinders, supply and exhaust air flow control is to be selected because of the reduced air quantity.



Shuttle valve: Logic OR function

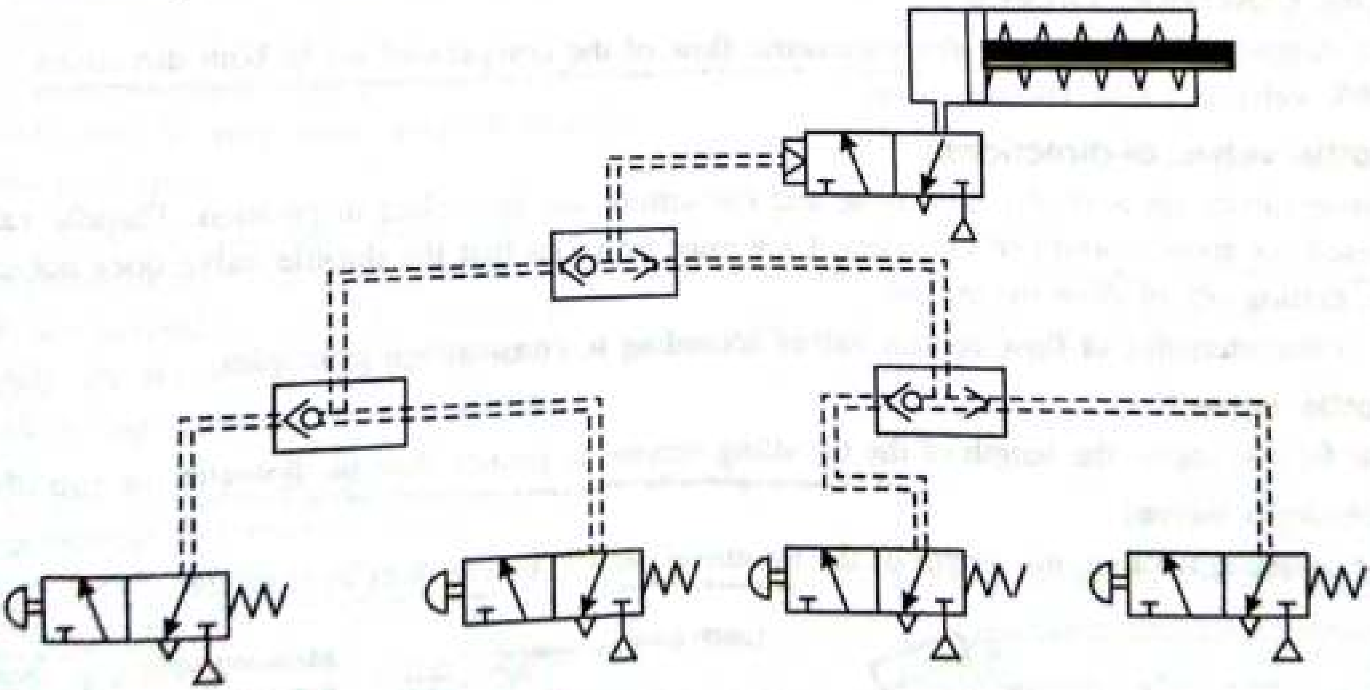
This non-return element has two inlets "I_1" and "I_2" and one outlet "O". If compressed air is applied to the first inlet "I_1", the valve seat seals the opposing inlet "I_2", the air flows from "I_1" to "O". Inlet "I_1" is closed, if air passes from "I_2" to "O". A signal is generated at the outlet. When the air flow is reserved, i.e. a cylinder or valve is exhausted, the seat remains in its previously assumed position because of the pressure conditions. This valve is also called and OR element. If a cylinder or control valve is to be actuated from two or more positions, one or more shuttle valves should be used.

The shuttle valve automatically allows the high pressure to the output port while it blocks the low-pressure inlet. The spool is free floating with an open center action. At either end of the spool it has two inlet ports. The Schematic representation of the Shuttle Valve and its ANSI symbol is shown in the below figure.



SHUTTLE VALVE

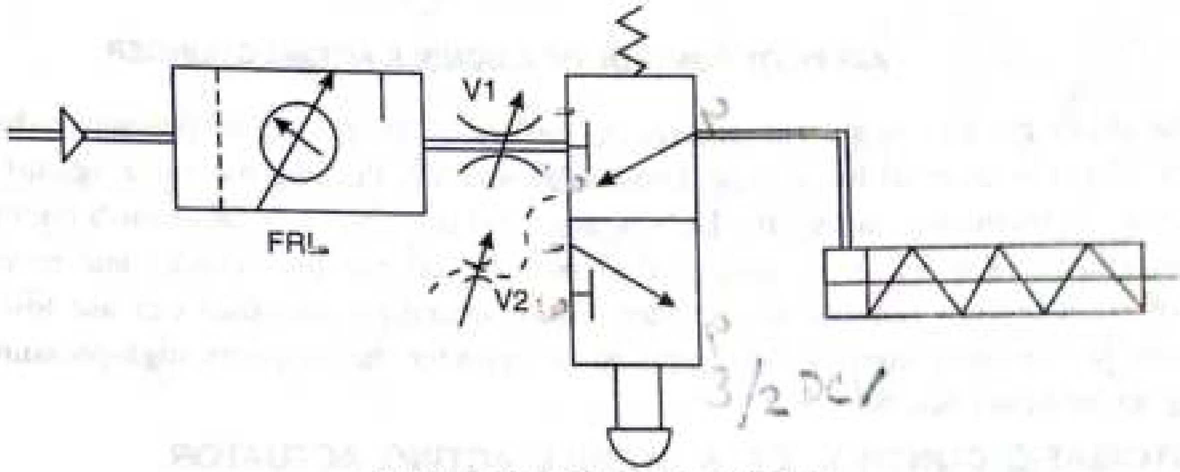
Shuttle valves can be linked to create additional logic OR conditions e.g. as shown below: if any of the four push buttons are operated, the cylinder is to extend.



Pneumatic Circuits

CONTROL OF SINGLE ACTING CYLINDER

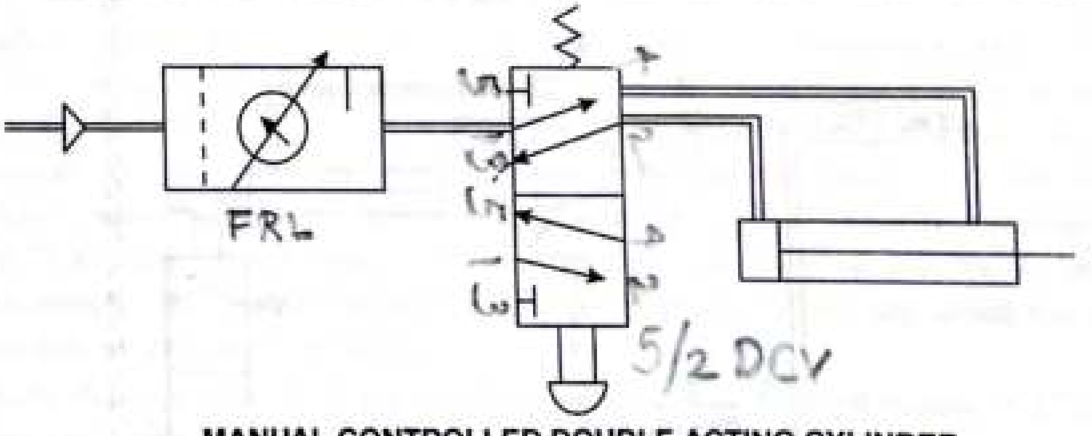
The circuits consists of a 3/2 way push button operated DCV to control a single acting cylinder. Initially the flow is blocked and when the push button is operated the cylinder extends and when the button is released the cylinder retracts by the compression spring located at the rod end of the cylinder. The adjustable flow control valves "V1" and "V2" can control the rate of extension and retraction of the cylinder.



Control of single-acting cylinder

MANUAL CONTROLLED DOUBLE ACTING CYLINDER

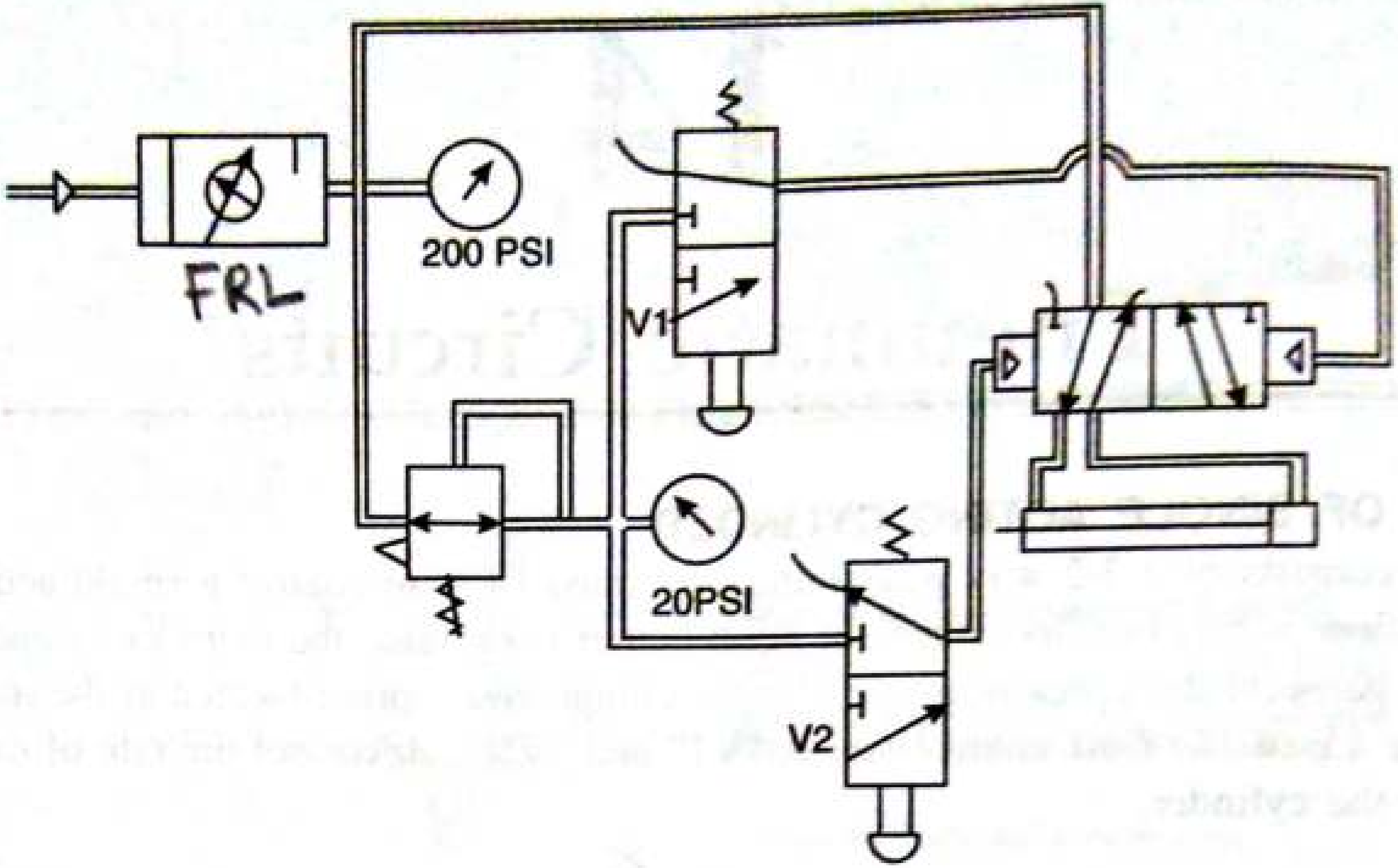
The below circuit has a four way two position push button, a FRL and double acting cylinder. In spring offset position the cylinder retracts and when the push button is actuated the cylinder extends.



MANUAL CONTROLLED DOUBLE ACTING CYLINDER

AIR PILOT CONTROL OF DOUBLE ACTING ACTUATOR

In large systems, the pressure required for operation is high. In such cases shifting of DCVs to change the direction of the cylinders are difficult. To overcome this difficulty air pilot operated DCVs are mostly used. The shifting of air pilot DCVs can be controlled by relatively less pressurized push buttons. This is seen in the below circuit.



AIR PILOT CONTROL OF A DOUBLE ACTING CYLINDER

In the above circuit, the system receives the airflow at 200 psi and by pressure reducer a less pressure of 20 psi is diverted to the push buttons V1 and V2. Pushing the valve against 20 psi is easy compared to manually shifting the DCV against 200 psi. Thus activating push button V1 and maintaining push button V2 in its spring-offset position extends the cylinder and reversing the actuation of push buttons retracts the cylinder. Thus, operating personnel can use low-pressure push buttons to remotely control the operation of cylinder that requires high-pressure air for performing its intended function.

COMPARISON OF HYDRAULICS, PNEUMATICS AND HYDROPNEUMATICS

Sl.No	*Hydraulics*	*Pneumatics*	*Hydro pneumatics*
1	Pump is necessary	Compressor and Reservoir is necessary	No pump is required; small oil reservoir is sufficient
2	Operating pressure varies from low to very high pressure 20000 psi	Operating pressure is mostly 6 bar	Suitable for low pressure applications; using oil-air intensifier medium pressures can be generated 3000 psi
3	Resistant to fluctuating loads	Non-resistant to fluctuating loads	Resistant to fluctuating loads
4	Weight to pressure ratio is small	Weight to pressure ratio is large	Weight to pressure ratio is large
5	System rigidity is good	System rigidity is poor	System rigidity is good
6	Speed is limited	Very high speed is possible	High speed is possible
7	Cavitation is a major Problem	No such problem	No cavitation but possibility of air and oil mixing possible
8	Suitable for feed movement in machine tools	Unsuitable for feed movement in machine tools	Suitable for feed movement in machine tools
9	Moderate operating cost	Very low operating cost	Very low operating cost
10	Overall cost is moderate to high	Overall cost is low	Overall cost is low to moderate