

**NARASARAOPET ENGINEERING COLLEGE (AUTONOMOUS)**  
**IMPACT OF JET ON VANES**

**FLUID JET:**

A fluid jet is a stream of fluid issuing from a nozzle with high velocity and hence a high kinetic energy.

**IMPULSE MOMENTUM PRINCIPLE (Newton's Second Law):**

“The sum of forces on the body equals the rate of change of momentum of the body in the direction of the force. In equation form (F and V are in the same direction).

$$\Sigma F dt = d(mV) \text{ (or) } \Sigma F = \rho Q (\Delta V)$$

**Force exerted by the jet on the plate:**

**F** = rate of change of momentum

$$= [\text{mass of jet /sec}] [\text{velocity of jet before striking the plate} - \text{velocity of jet after striking the plate}]$$
$$= \rho Q (\Delta V)$$

- **Kinetic Energy / Sec = K.E. =  $m \cdot v^2 / 2 = \rho a v^3 / 2$**
- **Work Done / Sec = Power =  $F_x \cdot u$**
- **Efficiency =  $\eta = W / K.E$**

**NOTE:** The plate is stationary; therefore, the work done on the plate is zero.

**CASES:**

**1. When the flat plate/vane is held normal to the jet:**

- (a) Stationary vane
- (b) Moving vane
- (c) Series moving vanes

**2. When the flat plate is held inclined to the jet:**

- (a) Stationary vane
- (b) Moving vane
- (c) Series moving vanes

**3. When the plate is curved ;jet strikes at the center:**

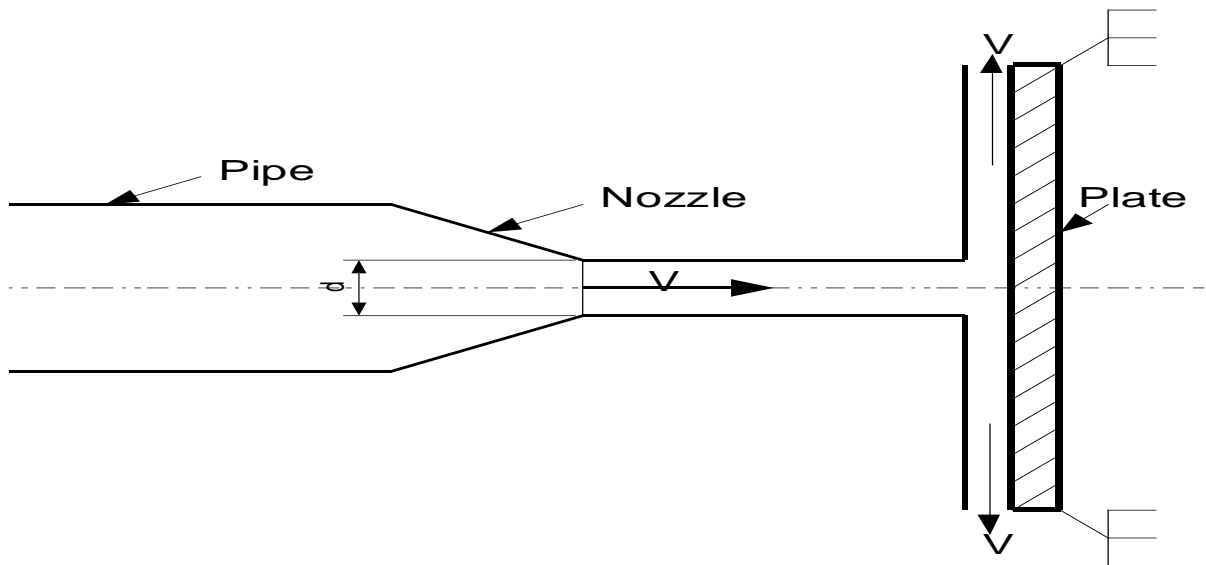
- (a) Stationary vane
- (b) Moving vane
- (c) Series moving vanes

**4. When the plate is curved :Jet strikes the plate at one end tangentially:**

- (a) Stationary vane
- (b) Moving vane
- (c) Series of moving vanes

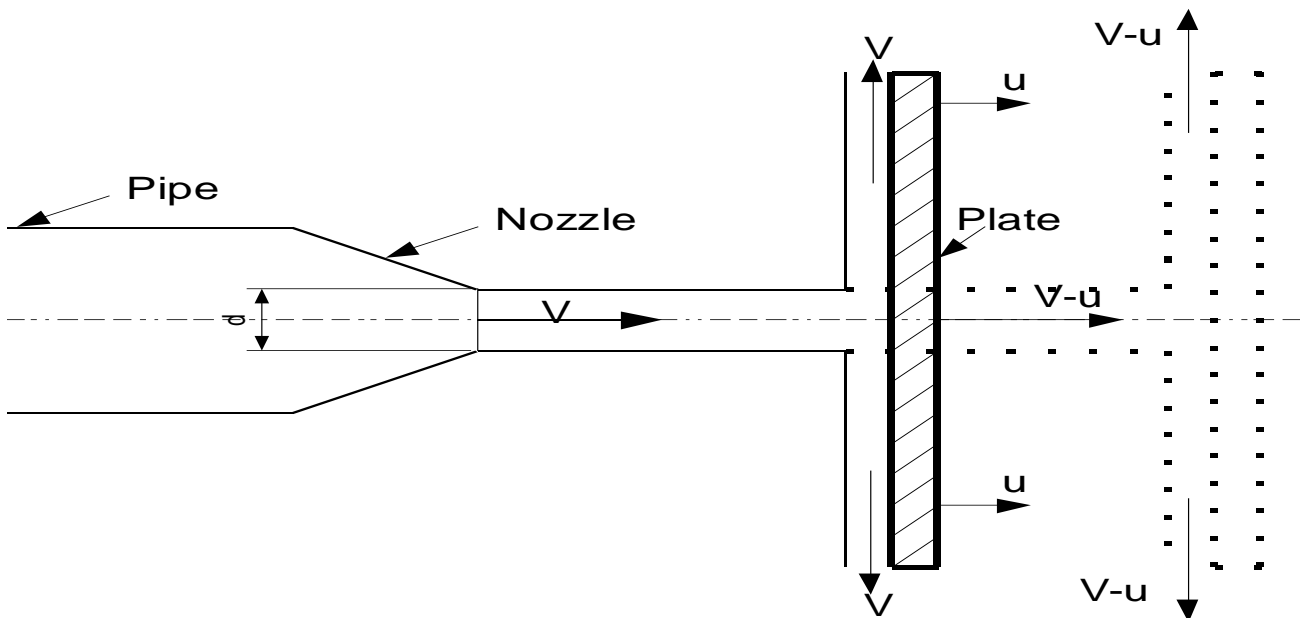
1. When the flat plate/vane is held normal to the jet:

(a) Stationary vane:



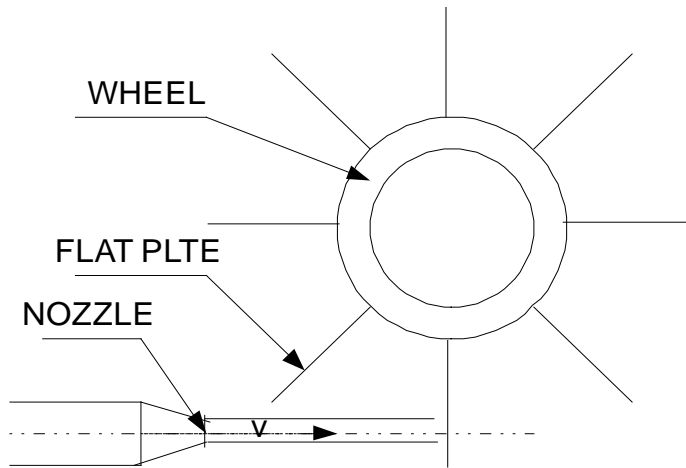
1. Force exerted by jet on vane in x- direction=  $F_x = \rho a v [ v - 0 ] = \rho a v^2$
2. Plate Velocity =  $u = 0$
3. Efficiency =  $\eta = W / \text{Error! Bookmark not defined.K.E} = 0$

(b) Moving vane:



1.  $F_x = \rho a (v-u) [(v-u) - 0] = \rho a (v-u)^2$
2.  $\eta = W / \text{K.E} = 2 (v-u)^2 u / v^3$
3.  $d\eta/du = 0 \Rightarrow \text{at } v = 3u; \eta_{\max} = 8/27$

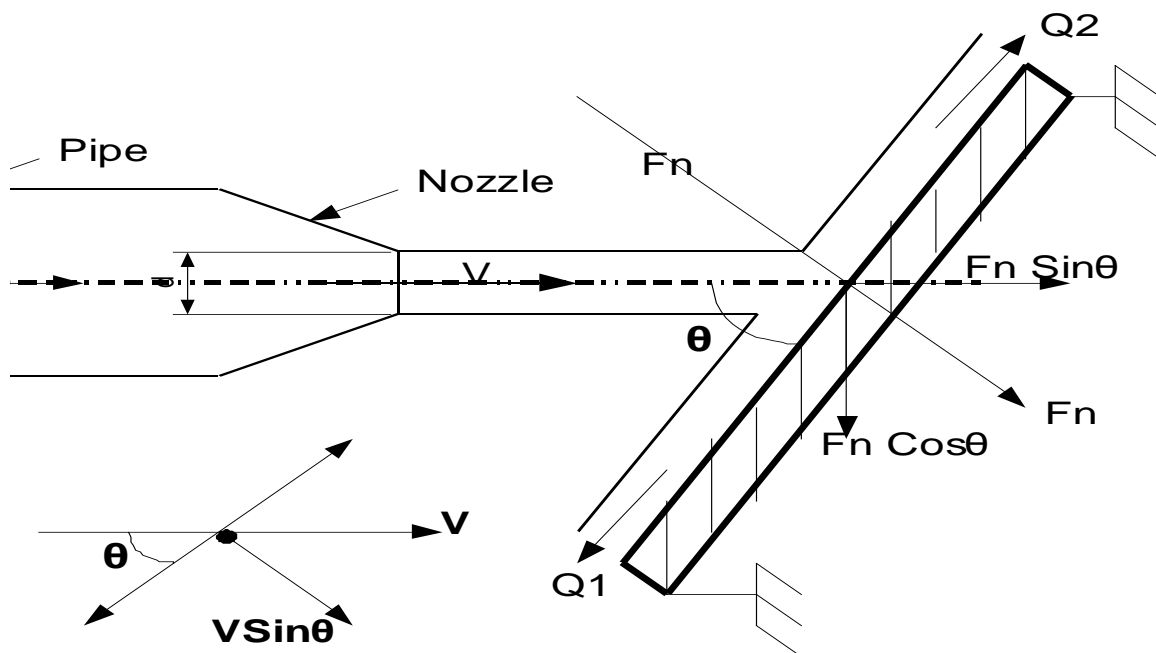
(c) Series moving vanes:



1.  $F_x = \rho a v [(v-u) - 0] = \rho a v (v-u)$
2.  $\eta = W / K.E = 2 (v-u) u / v^2$
3.  $d\eta/du = 0 \Rightarrow \text{at } v=2u; \eta_{\max} = 1/2 = 50\%$

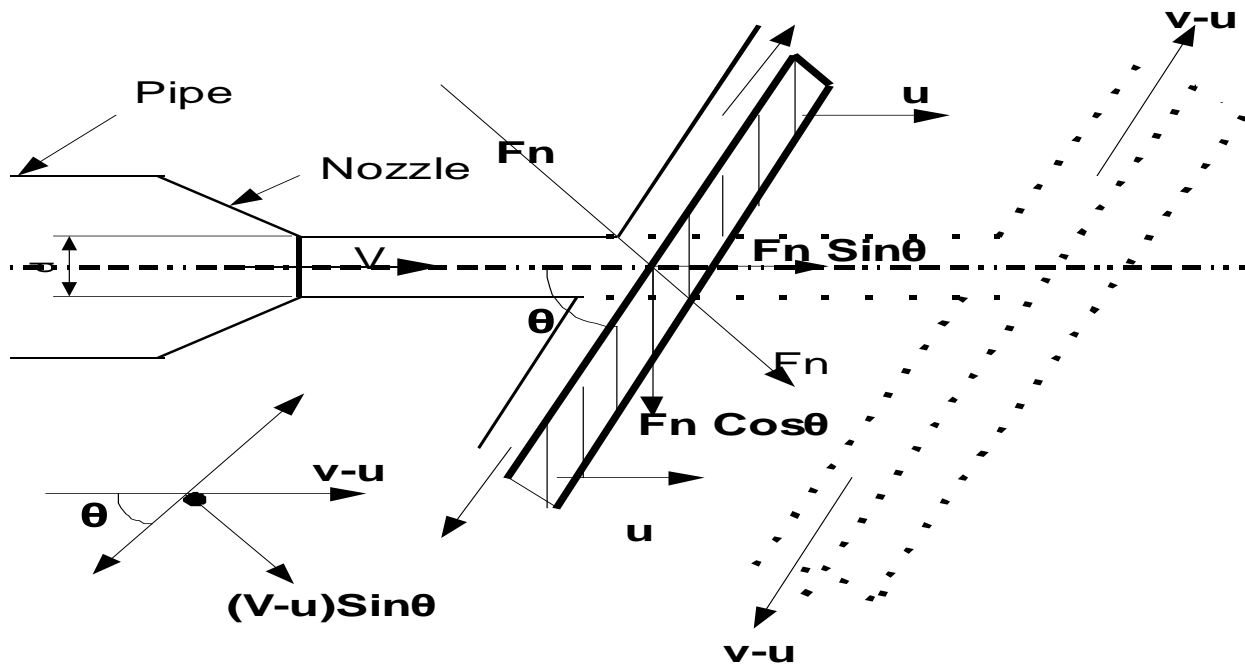
2. When the flat plate is held inclined to the jet:

(a) Stationary vane:



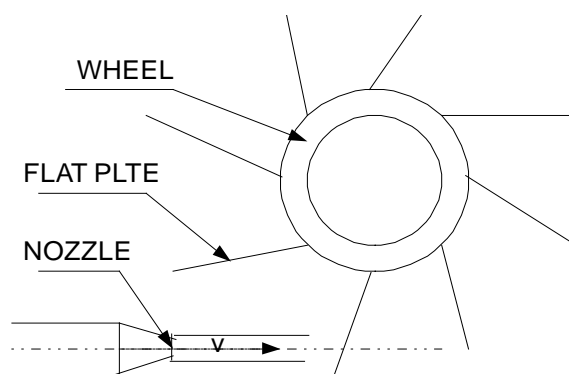
1.  $F_n = \rho a v [v \sin\theta - 0] = \rho a v^2 \sin\theta$
2.  $F_x = F_n \sin\theta = \rho a v^2 \sin^2\theta$
3.  $F_y = F_n \cos\theta = \rho a v^2 \sin\theta \cos\theta$
4. Ratio of discharges =  $Q1/Q2 = (1+\cos\theta) / (1- \cos\theta)$
5.  $u = 0, \eta = W/K.E = 0$

(b) Moving vane:



1.  $F_n = \rho a (v-u) [(v-u) \sin\theta - 0] = \rho a (v-u)^2 \sin\theta$
2.  $F_x = F_n \sin\theta = \rho a (v-u)^2 \sin^2\theta$
3.  $F_y = F_n \cos\theta = \rho a (v-u)^2 \sin\theta \cos\theta$
4.  $\eta = W / K.E = 2 u (v-u)^2 \sin^2\theta / v^3$
5.  $d\eta/du = 0 \Rightarrow$  at  $v = 3u$ ;  $\eta_{\max} = 8/27 \sin^2\theta$

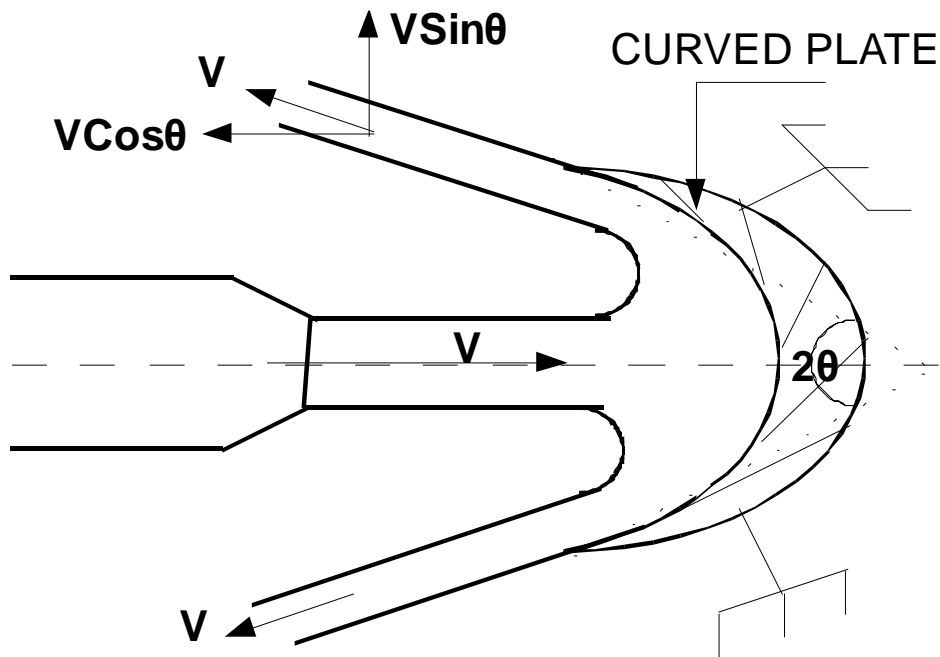
(c) Series moving vanes:



1.  $F_n = \rho a v [(v-u) \sin\theta - 0] = \rho a v (v-u) \sin\theta$
2.  $F_x = F_n \sin\theta = \rho a v (v-u) \sin^2\theta$ ,
3.  $F_y = F_n \cos\theta = \rho a v (v-u) \sin\theta \cos\theta$
4.  $\eta = W / K.E = 2 (v-u) u \sin^2\theta / v^2$
5.  $d\eta/du = 0 \Rightarrow$  at  $v=2u$ ;  $\eta_{\max} = 1/2 \sin^2\theta$

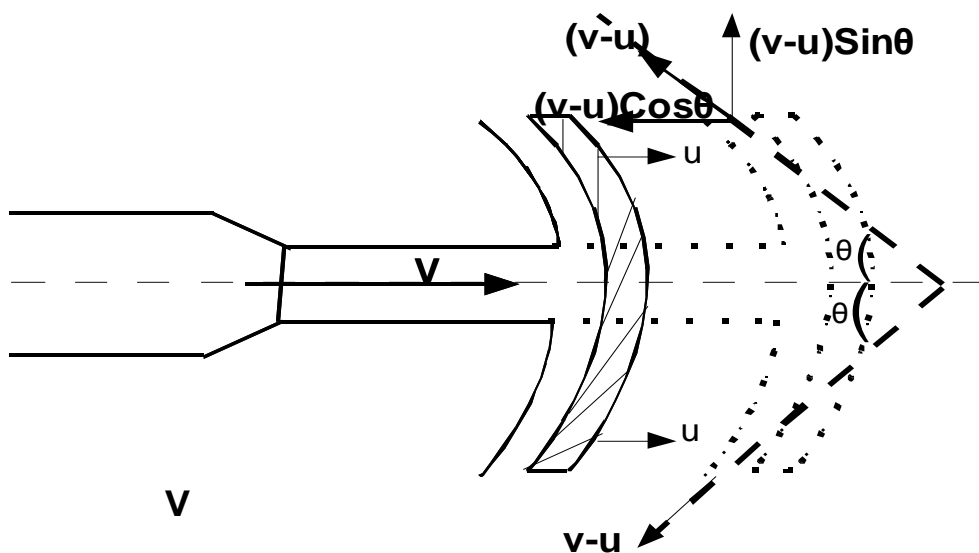
3. When the plate is curved jet strikes at the center:

(a) Stationary vane:



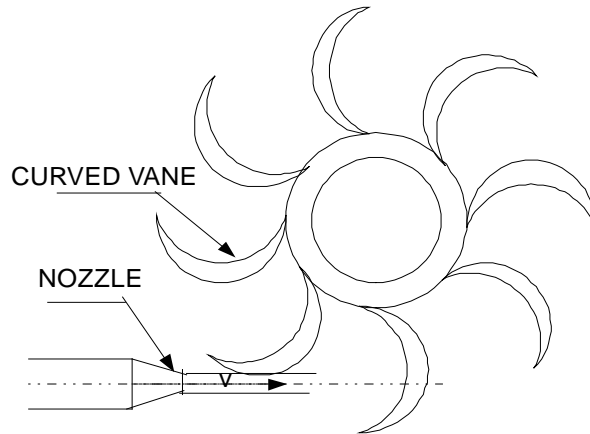
1.  $F_x = \rho a v[v - (-v \cos \theta)] = \rho a v^2 (1 + \cos \theta)$
2.  $F_y = \rho a v[0 - (v \sin \theta)] = -\rho a v^2 \sin \theta$
3.  $u = 0, \eta = W/K.E = 0$

(b) Moving vane:



1.  $F_x = \rho a (v-u) [(v-u) - (-(v-u) \cos \theta)] = \rho a (v-u)^2 (1 + \cos \theta),$
2.  $F_y = \rho a (v-u) [0 - ((v-u) \sin \theta)] = -\rho a (v-u)^2 \sin \theta$
3.  $\eta = W/ K.E = 2 (v-u)^2 (1 + \cos \theta)u / v^3$
4.  $d\eta/du = 0 \Rightarrow \text{at } v = 3u; \eta_{\max} = 8/27 (1 + \cos \theta)$

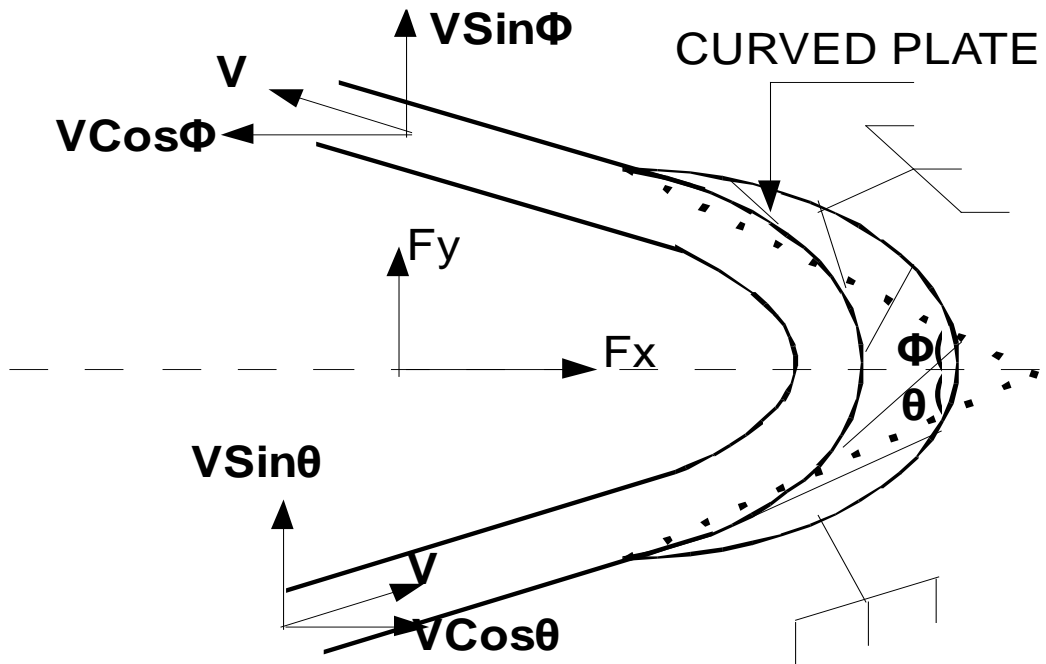
(c) Series moving vanes:



- 1  $F_x = \rho a v [(v-u) - (-(v-u) \cos\theta)] = \rho a v (v-u) (1+\cos\theta)$
- 2  $F_y = \rho a v [0 - ((v-u) \sin\theta)] = -\rho a v (v-u) \sin\theta$
- 3  $\eta = W / K.E = 2 v (v-u) (1+\cos\theta) u / v^3$
- 4  $d\eta/du = 0 \Rightarrow$  at  $v=2u$ ;  $\eta_{\max} = \frac{1}{2} (1+\cos\theta)$

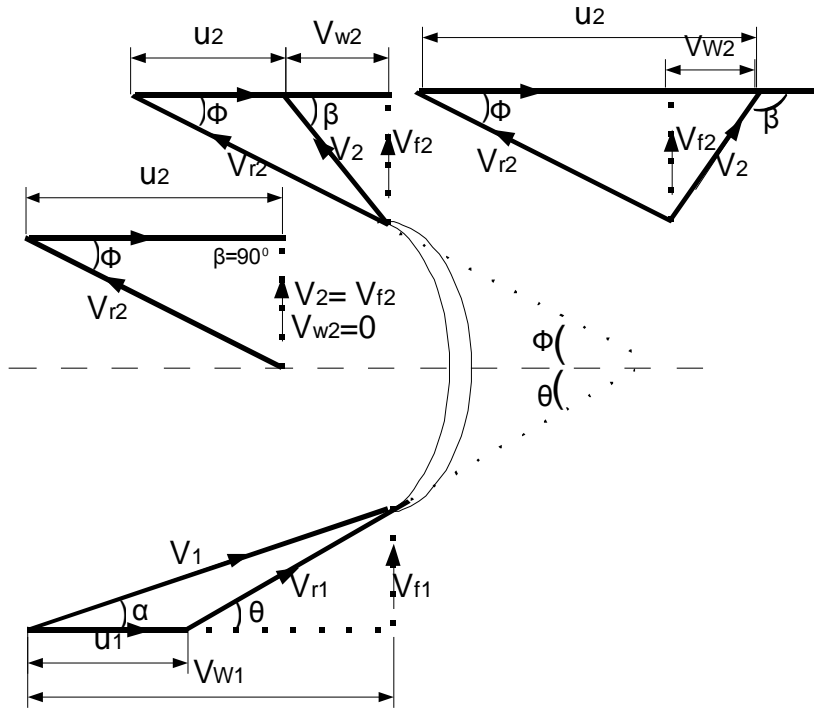
4. When the plate is curved : Jet strikes the plate at one end tangentially:

(a) Stationary vane:



- 1  $F_x = \rho a v [v \cos\theta - (-v \cos\Phi)] = \rho a v^2 (\cos\theta + \cos\Phi)$
- 2  $F_y = \rho a v [v \sin\theta - v \sin\Phi] = \rho a v^2 (\sin\theta - \sin\Phi)$
- 3 plate is symmetrical..... $\theta = \Phi$
- 4  $F_x = 2\rho a v^2 \cos\theta, F_y = 0$
- 5  $u = 0, \eta = W/K.E = 0$

(b) Moving vane:



1.  $F_x = \rho a v_{r1} [v_{w1} +/- v_{w2}]$
2.  $K.E = mv_{r1}^2/2, m = \rho a v_{r1}$
3.  $\eta = W / K.E = 2 [v_{w1} +/- v_{w2}]u / v_{r1}^2$

(C) Series moving vanes:

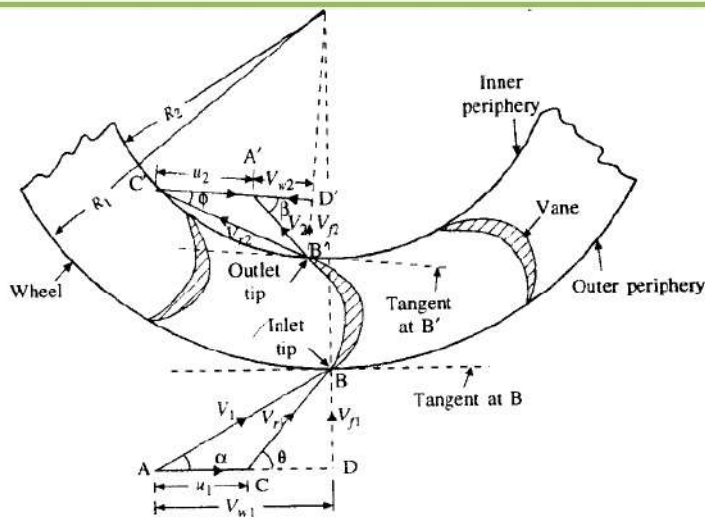


Fig. 18.12. Velocity diagrams for an inward flow reaction turbine.

1. Torque Exerted by water on the wheel =  $T = \rho a v_1 [v_{w1} R_1 +/- v_{w2} R_2]$
2. Work Done/sec =  $W = T\omega = \rho a v_1 [v_{w1} u_1 +/- v_{w2} u_2]$
3.  $u_1 = \omega R_1, u_2 = \omega R_2;$
4.  $K.E = mv_1^2/2, m = \rho a v_1$
5.  $\eta = W / K.E = 2 [v_{w1} u_1 +/- v_{w2} u_2] / v_1^2$

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**IMPACT OF JET ON VANES**

- **FLUID JET:** A fluid jet is a stream of fluid issuing from a nozzle with high velocity and hence a high kinetic energy.
- **Force exerted by the jet on the plate** =  $F = \text{rate of change of momentum} = \rho Q (\Delta V)$
- **Kinetic Energy / Sec = K.E.** =  $m \cdot v^2 / 2 = \rho a v^3 / 2$
- **Work Done / Sec = Power =  $F_x u$**

**NOTE:** The plate is stationary; therefore, the work done on the plate is zero.

S. No:	CASES	Force exerted by the jet on the plate = $F_x$	Efficiency = $\eta = W / \text{K.E}$	Maximum Efficiency = $\eta_{\max}$	
1	<b>When the flat plate is held normal to the jet (Vertical Plate)</b>	(a) Stationary vane	$\rho a v^2$	0	0
		(b) Moving vane	$\rho a (v-u)^2$	$2 (v-u)^2 u / v^3$	8/27 (29.63%) at $v = 3u$
		(c) Series moving vanes	$\rho a v (v-u)$	$2 (v-u) u / v^2$	1/2 (50%) at $v = 2u$
2	<b>When the flat plate is held inclined to the jet</b>	(a) Stationary vane	$\rho a v^2 \sin^2\theta$	0	0
		(b) Moving vane	$\rho a (v-u)^2 \sin^2\theta$	$2 (v-u)^2 u \sin^2\theta / v^3$	8/27 $\sin^2\theta$ at $v = 3u$
		(c) Series moving vanes	$\rho a v (v-u) \sin^2\theta$	$2 (v-u) u \sin^2\theta / v^2$	1/2 $\sin^2\theta$ at $v = 2u$
3	<b>When the plate is curved and Jet strikes at the center</b>	(a) Stationary vane	$\rho a v^2 (1+\cos\theta)$	0	0
		(b) Moving vane	$\rho a (v-u)^2 (1+\cos\theta)$	$2 (v-u)^2 u (1+\cos\theta) / v^3$	8/27 $(1+\cos\theta)$ at $v = 3u$
		(c) Series moving vanes	$\rho a v (v-u) (1+\cos\theta)$	$2 (v-u) u (1+\cos\theta) / v^2$	1/2 $(1+\cos\theta)$ at $v = 2u$
4	<b>When the plate is curved and Jet strikes the plate at one end tangentially:</b>	(a) Stationary vane	$\rho a v^2 (\cos\theta + \cos\Phi)$	0	0
		(b) Moving vane	$\rho a v_{r1} [v_{w1} +/- v_{w2}]$	$2 [v_{w1} +/- v_{w2}]u / v_{r1}^2$	-----
		(c) Series moving vanes <b>(Radial Vanes)</b>	$\rho a v_1 [v_{w1} R_1 +/- v_{w2} R_2]$	$2 [v_{w1} u_1 +/- v_{w2} u_2] / v_1^2$	-----



**Important Problems**

**1. CASE- 1 (a) : When the Stationary flat plate/vane is held normal to the jet:**

**Problem 17.2** Water is flowing through a pipe at the end of which a nozzle is fitted. The diameter of the nozzle is 100 mm and the head of water at the centre nozzle is 100 m. Find the force exerted by the jet of water on a fixed vertical plate. The co-efficient of velocity is given as 0.95.

**Solution.** Given :

Diameter of nozzle,  $d = 100 \text{ mm} = 0.1 \text{ m}$

Head of water,  $H = 100 \text{ m}$

Co-efficient of velocity,  $C_v = 0.95$

Area of nozzle,  $a = \frac{\pi}{4} (.1)^2 = .007854 \text{ m}^2$

Theoretical velocity of jet of water is given as

$$V_{th} = \sqrt{2gH} = \sqrt{2 \times 9.81 \times 100} = 44.294 \text{ m/s}$$

But 
$$C_v = \frac{\text{Actual velocity}}{\text{Theoretical velocity}}$$

$\therefore$  Actual velocity of jet of water,  $V = C_v \times V_{th} = 0.95 \times 44.294 = 42.08 \text{ m/s}$ .

Force on a fixed vertical plate is given by equation (17.1) as

$$F = \rho a V^2 = 1000 \times .007854 \times 42.08^2 \quad (\because \text{In S.I. units } \rho \text{ for water} = 1000 \text{ kg/m}^3)$$

$$= 13907.2 \text{ N} = \mathbf{13.9 \text{ kN. Ans.}}$$

**2. CASE-3 (a): Stationary Curved Plate and Jet Strikes at center.**

**Problem 17.5** A jet of water of diameter 50 mm moving with a velocity of 40 m/s, strikes a curved fixed symmetrical plate at the centre. Find the force exerted by the jet of water in the direction of the jet, if the jet is deflected through an angle of 120° at the outlet of the curved plate.

**Solution.** Given :

Diameter of the jet,  $d = 50 \text{ mm} = 0.05 \text{ m}$

$\therefore$  Area,  $a = \frac{\pi}{4} (.05)^2 = 0.001963 \text{ m}^2$

Velocity of jet,  $V = 40 \text{ m/s}$

Angle of deflection  $= 120^\circ$

From equation [17.6 (A)], the angle of deflection  $= 180^\circ - \theta$

$\therefore 180^\circ - \theta = 120^\circ$  or  $\theta = 180^\circ - 120^\circ = 60^\circ$

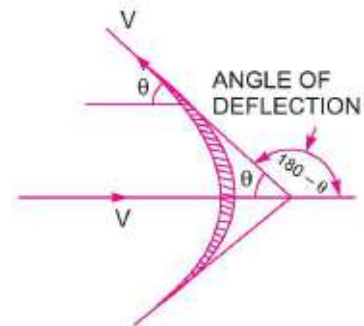


Fig. 17.5

Force exerted by the jet on the curved plate in the direction of the jet is given by equation (17.5) as

$$F_x = \rho a V^2 [1 + \cos \theta]$$

$$= 1000 \times .001963 \times 40^2 \times [1 + \cos 60^\circ] = \mathbf{4711.15 \text{ N. Ans.}}$$

3. CASE-3 (b): Moving Curved Plate and Jet Strikes at center.

**Problem 17.14** A jet of water of diameter 7.5 cm strikes a curved plate at its centre with a velocity of 20 m/s. The curved plate is moving with a velocity of 8 m/s in the direction of the jet. The jet is deflected through an angle of 165°. Assuming the plate smooth find :

(i) Force exerted on the plate in the direction of jet, (ii) Power of the jet, and (iii) Efficiency of the jet.

**Solution.** Given :

Diameter of the jet,  $d = 7.5 \text{ cm} = 0.075 \text{ m}$

$\therefore$  Area,  $a = \frac{\pi}{4} (.075)^2 = 0.004417$

Velocity of the jet,  $V = 20 \text{ m/s}$

Velocity of the plate,  $u = 8 \text{ m/s}$

Angle of deflection of the jet,  $= 165^\circ$

$\therefore$  Angle made by the relative velocity at the outlet of the plate,

$$\theta = 180^\circ - 165^\circ = 15^\circ.$$

(i) Force exerted by the jet on the plate in the direction of the jet is given by equation (17.17) as

$$\begin{aligned} &= F_x = \rho a(V - u)^2 (1 + \cos \theta) \\ &= 1000 \times .004417 \times (20 - 8)^2 [1 + \cos 15^\circ] = \mathbf{1250.38 \text{ N. Ans.}} \end{aligned}$$

(ii) Work done by the jet on the plate per second

$$= F_x \times u = 1250.38 \times 8 = 10003.04 \text{ N m/s}$$

$$\therefore \text{ Power of the jet} = \frac{10003.04}{1000} = \mathbf{10 \text{ kW. Ans.}}$$

$$(iii) \text{ Efficiency of the jet} = \frac{\text{Output}}{\text{Input}} = \frac{\text{Work done by jet/sec}}{\text{Kinetic energy of jet/sec}}$$

$$= \frac{1250.38 \times 8}{\frac{1}{2} (\rho a V) \times V^2} = \frac{1250.38 \times 8}{\frac{1}{2} \times 1000 \times .004417 \times V^3}$$

$$= \frac{1250.38 \times 8}{\frac{1}{2} \times 1000 \times .004417 \times 20^3} = 0.564 = \mathbf{56.4\% \text{ Ans}}$$

4. Case IV (a) : Stationary Curved Vane... Jet Strikes Tangentially at one end:

**Problem 17.15** A jet of water from a nozzle is deflected through 60° from its original direction by a curved plate which it enters tangentially without shock with a velocity of 30 m/s and leaves with a mean velocity of 25 m/s. If the discharge from the nozzle is 0.8 kg/s, calculate the magnitude and direction of the resultant force on the vane, if the vane is stationary.

**Solution.** Given :

Velocity at inlet,  $V_1 = 30 \text{ m/s}$

Velocity at outlet,  $V_2 = 25 \text{ m/s}$

Mass per second  $= 0.8 \text{ kg/s}$

Force in the direction of jet,

$$F_x = \text{Mass per second} \times (V_{1x} - V_{2x})$$

where  $V_{1x}$  = Initial velocity in the direction of  $x$   
 $= 30 \text{ m/s}$

$V_{2x}$  = Final velocity in the direction of  $x$

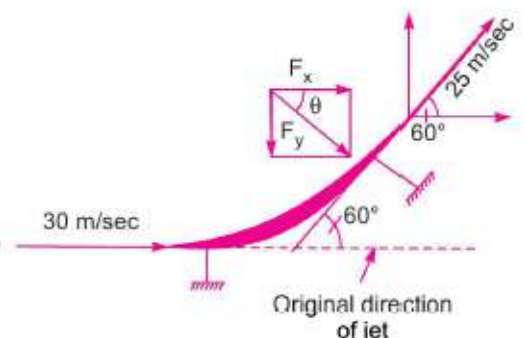


Fig. 17.13 (a)

$$= 25 \cos 60^\circ = 25 \times \frac{1}{2} = 12.5 \text{ m/s}$$

$$\therefore F_x = 0.8[30 - 12.5] = 0.8 \times 17.5 = 14.0 \text{ N}$$

Similarly, force normal to the jet,

$$F_y = \text{Mass per second} \times (V_{1y} - V_{2y}) \\ = 0.8 [0 - 25 \sin 60^\circ] = -17.32 \text{ N}$$

-ve sign means the force,  $F_y$ , is acting in the vertically downward direction.

$$\therefore \text{Resultant force on the vane} = \sqrt{F_x^2 + F_y^2} = \sqrt{14^2 + (-17.32)^2} = 22.27 \text{ N. Ans.}$$

The angle made by the resultant with x-axis

$$\tan \theta = \frac{F_y}{F_x} = \frac{-17.32}{14.0} = -1.237$$

-ve sign means the angle  $\theta$  is in the clockwise direction with x-axis as shown in Fig. 17.13 (a)

$$\therefore \theta = \tan^{-1} 1.237 = 51^\circ 28.6'. \text{ Ans.}$$

### 5. CASE IV (b): Moving Curved Vane... Jet Strikes Tangentially at one end: $\beta < 90^\circ$

**Problem 17.18** A jet of water having a velocity of 20 m/s strikes a curved vane, which is moving with a velocity of 10 m/s. The jet makes an angle of  $20^\circ$  with the direction of motion of vane at inlet and leaves at an angle of  $130^\circ$  to the direction of motion of vane at outlet. Calculate :

- Vane angles, so that the water enters and leaves the vane without shock.
- Work done per second per unit weight of water striking (or work done per unit weight of water striking) the vane per second.

**Solution.** Given :

Velocity of jet,  $V_1 = 20 \text{ m/s}$

Velocity of vane,  $u_1 = 10 \text{ m/s}$

Angle made by jet at inlet, with direction of motion of vane,  $\alpha = 20^\circ$

Angle made by the leaving jet, with the direction of motion  $= 130^\circ$

$$\therefore \beta = 180^\circ - 130^\circ = 50^\circ$$

In this problem,  $u_1 = u_2 = 10 \text{ m/s}$

$$V_{r1} = V_{r2}$$

(i) **Vane Angle** means angle made by the relative velocities at inlet and outlet, i.e.,  $\theta$  and  $\phi$ .

From Fig. 17.16, in  $\triangle ABD$ , we have  $\tan \theta = \frac{BD}{CD}$

$$= \frac{V_{f1}}{AD - AC} = \frac{V_{f1}}{V_{w1} - u_1} \quad \dots(i)$$

where  $V_{f1} = V_1 \sin \alpha = 20 \times \sin 20^\circ = 6.84 \text{ m/s}$

$$V_{w1} = V_1 \cos \alpha = 20 \times \cos 20^\circ = 18.794 \text{ m/s.}$$

$$u_1 = 10 \text{ m/s}$$

$$\therefore \tan \theta = \frac{6.84}{18.794 - 10} = .7778 \text{ or } \theta = 37.875^\circ$$

$$\therefore \theta = 37^\circ 52.5'. \text{ Ans.}$$

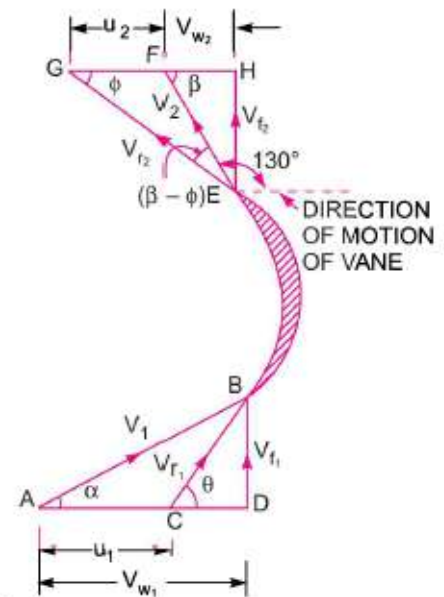


Fig. 17.16

From,  $\triangle ABC$ , 
$$\sin \theta = \frac{V_{f1}}{V_{r1}} \quad \text{or} \quad V_{r1} = \frac{V_{f1}}{\sin \theta} = \frac{6.84}{\sin 37.875^\circ} = 11.14$$

$\therefore V_{r2} = V_{r1} = 11.14 \text{ m/s.}$

From,  $\triangle EFG$ , applying sine rule, we have

$$\frac{V_{r2}}{\sin (180^\circ - \beta)} = \frac{u_2}{\sin (\beta - \phi)}$$

or 
$$\frac{11.14}{\sin \beta} = \frac{10}{\sin [\beta - \phi]} \quad \text{or} \quad \frac{11.14}{\sin 50^\circ} = \frac{10}{\sin [50^\circ - \phi]} \quad (\because \beta = 50^\circ)$$

$\therefore \sin (50^\circ - \phi) = \frac{10 \times \sin 50^\circ}{11.14} = 0.6876 = \sin 43.44^\circ$

$\therefore 50^\circ - \phi = 43.44^\circ \quad \text{or} \quad \phi = 50^\circ - 43.44^\circ = 6.56^\circ$

$\therefore \phi = 6^\circ 33.6' \text{ Ans.}$

(ii) Work done per second per unit weight of the water striking the vane per second is given by equation (17.21) as

$$= \frac{1}{g} [V_{w1} + V_{w2}] \times u \text{ Nm/N (+ ve sign is taken as } \beta \text{ is an acute angle)}$$

where  $V_{w1} = 18.794 \text{ m/s}$ ,  $V_{w2} = GH - GF = V_{r2} \cos \phi - u_2 = 11.14 \times \cos 6.56^\circ - 10 = 1.067 \text{ m/s}$

$u = u_1 = u_2 = 10 \text{ m/s}$

$\therefore$  Work done per unit weight of water

$$= \frac{1}{9.81} [18.794 + 1.067] \times 10 \text{ Nm/N} = \mathbf{20.24 \text{ Nm/N. Ans.}}$$

## 6. CASE IV (b): Moving Curved Vane... Jet Strikes Tangentially at one end: $\beta = 90^\circ$

**Problem 17.19** A jet of water having a velocity of 40 m/s strikes a curved vane, which is moving with a velocity of 20 m/s. The jet makes an angle of  $30^\circ$  with the direction of motion of vane at inlet and leaves at an angle of  $90^\circ$  to the direction of motion of vane at outlet. Draw the velocity triangles at inlet and outlet and determine the vane angles at inlet and outlet so that the water enters and leaves the vane without shock.

**Solution.** Given :

Velocity of jet,  $V_1 = 40 \text{ m/s}$

Velocity of vane,  $u_1 = 20 \text{ m/s}$

Angle made by jet at inlet,  $\alpha = 30^\circ$

Angle made by leaving jet  $= 90^\circ$

$\therefore \beta = 180^\circ - 90^\circ = 90^\circ$

For this problem, we have

$$u_1 = u_2 = u = 20 \text{ m/s}$$

Vane angles at inlet and outlet are  $\theta$  and  $\phi$  respectively.

From  $\triangle BCD$ , we have

$$\tan \theta = \frac{BD}{CD} = \frac{BD}{AD - AC} = \frac{V_{f1}}{V_{w1} - u_1}$$

where  $V_{f1} = V_1 \sin \alpha = 40 \times \sin 30^\circ = 20 \text{ m/s}$

$V_{w1} = V_1 \cos \alpha = 40 \times \cos 30^\circ = 34.64 \text{ m/s}$

$u_1 = 20 \text{ m/s}$

$\therefore \tan \theta = \frac{20}{34.64 - 20} = \frac{20}{14.64} = 1.366 = \tan 53.79^\circ$

$\therefore \theta = 53.79^\circ \text{ or } 53^\circ 47.4'. \text{ Ans.}$

Also from  $\triangle BCD$ ,

$$\sin \theta = \frac{V_{f1}}{V_{r1}} \text{ or } V_{r1} = \frac{V_{f1}}{\sin \theta} = \frac{20}{\sin 53.79^\circ}$$

$\therefore V_{r1} = 24.78$

But  $V_{r2} = V_{r1} = 24.78$

Hence, from  $\triangle EFG$ ,  $\cos \phi = \frac{u_2}{V_{r2}} = \frac{20}{24.78} = 0.8071 = \cos 36.18^\circ$

$\therefore \phi = 36.18^\circ \text{ or } 36^\circ 10.8'. \text{ Ans.}$

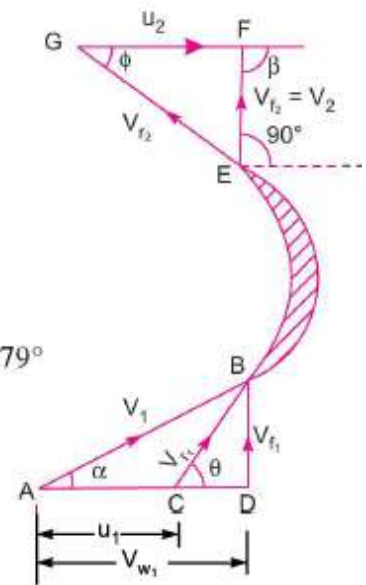


Fig. 17.17

7. CASE IV (b): Moving Curved Vane... Jet Strikes Tangentially at one end:  $\beta > 90^\circ$

**Problem 17.20** A jet of water of diameter 50 mm, having a velocity of 20 m/s strikes a curved vane which is moving with a velocity of 10 m/s in the direction of the jet. The jet leaves the vane at an angle of  $60^\circ$  to the direction of motion of vane at outlet. Determine :

- (i) The force exerted by the jet on the vane in the direction of motion.
- (ii) Work done per second by the jet.

**Solution.** Given :

Diameter of the jet,  $d = 50 \text{ mm} = 0.05 \text{ m}$

$\therefore$  Area,  $a = \frac{\pi}{4} (.05)^2 = .001963 \text{ m}^2$

Velocity of jet,  $V_1 = 20 \text{ m/s}$

Velocity of vane,  $u_1 = 10 \text{ m/s}$

As jet and vane are moving in the same direction,

$\therefore \alpha = 0$

Angle made by the leaving jet, with the direction of motion =  $60^\circ$

$\therefore \beta = 180^\circ - 60^\circ = 120^\circ$

For this problem, we have

$u_1 = u_2 = u = 10 \text{ m/s}$

$V_{r1} = V_{r2}$

From Fig. 17.18, we have

$V_{r1} = AB - AC = V_1 - u_1$   
 $= 20 - 10 = 10 \text{ m/s}$

$V_{w1} = V_1 = 20 \text{ m/s}$

$\therefore V_{r2} = V_{r1} = 10 \text{ m/s}$

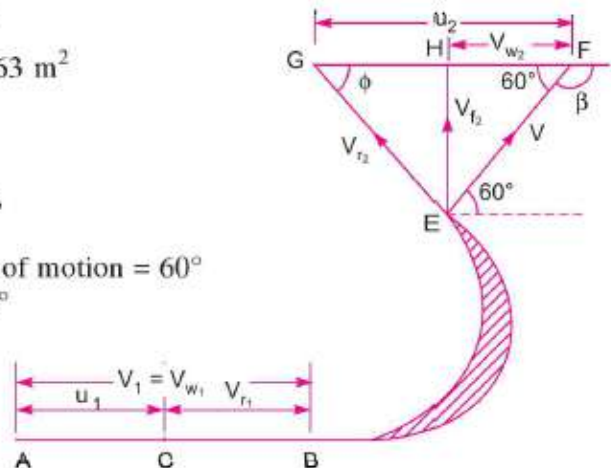


Fig. 17.18

Now in  $\triangle EFG$ ,

$$\begin{aligned} EG &= V_{r_2} = 10 \text{ m/s,} \\ GF &= u_2 = 10 \text{ m/s} \\ \angle GEF &= 180^\circ - (60^\circ + \phi) = (120^\circ - \phi) \end{aligned}$$

From sine rule, we have

$$\frac{EG}{\sin 60^\circ} = \frac{GF}{\sin (120^\circ - \phi)} \quad \text{or} \quad \frac{10}{\sin 60^\circ} = \frac{10}{\sin (120^\circ - \phi)}$$

or

$$\begin{aligned} \sin 60^\circ &= \sin (120^\circ - \phi) \\ \therefore 60^\circ &= 120^\circ - \phi \quad \text{or} \quad \phi = 120^\circ - 60^\circ = 60^\circ \end{aligned}$$

Now

$$\begin{aligned} V_{w_2} &= HF = GF - GH \\ &= u_2 - V_{r_2} \cos \phi = 10 - 10 \times \cos 60^\circ = 10 - 5 = 5 \text{ m/s.} \end{aligned}$$

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

$$\begin{aligned} F_x &= \rho a V_{r_1} [V_{w_1} - V_{w_2}] \quad (\text{-ve sign is taken as } \beta \text{ is an obtuse angle}) \\ &= 1000 \times .001963 \times 10 [20 - 5] \text{ N} = \mathbf{294.45 \text{ N. Ans.}} \end{aligned}$$

(ii) Work done per second by the jet

$$= F_x \times u = 294.45 \times 10 = 2944.5 \text{ N m/s}$$

$$= \mathbf{2944.5 \text{ W. Ans.}}$$

$$[\because \text{Nm/s} = \text{W (watt)}]$$

### 8. CASE IV (b): Moving Curved Vane... Jet Strikes Tangentially at one end: Symmetrical :

**Problem 17.21** A jet of water having a velocity of 15 m/s strikes a curved vane which is moving with a velocity of 5 m/s. The vane is symmetrical and is so shaped that the jet is deflected through  $120^\circ$ . Find the angle of the jet at inlet of the vane so that there is no shock. What is the absolute velocity of the jet at outlet in magnitude and direction and the work done per unit weight of water. Assume the vane to be smooth.

**Solution.** Given :

Velocity of jet,  $V_1 = 15 \text{ m/s}$

Velocity of vane,  $u_1 = 5 \text{ m/s}$

As vane is symmetrical. Hence angle  $\theta = \phi$

Angle of deflection of the jet  $= 120^\circ = 180^\circ - (\theta + \phi)$

$\therefore \theta + \phi = 60^\circ$  or each angle, i.e.,  $\theta = \phi = 30^\circ$

Let the angle of jet at inlet  $= \alpha$

Absolute velocity of jet at outlet  $= V_2$

Angle made by  $V_2$  at outlet with direction of motion of vane  $= \beta^*$ .

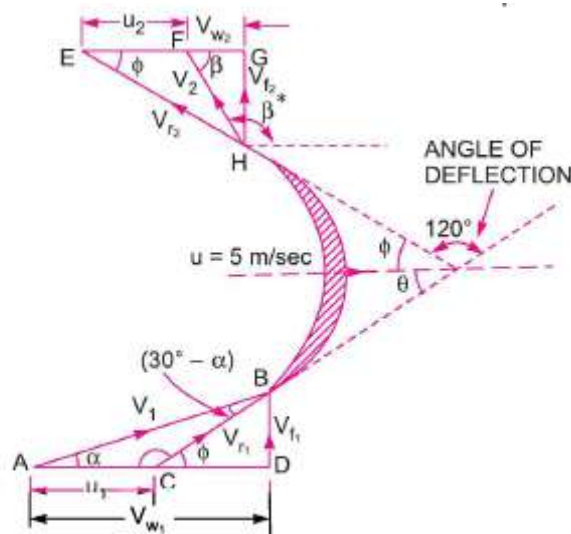


Fig. 17.19

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For this problem,

$$u_1 = u_2 = u = 5 \text{ m/s}$$

$$V_{r_1} = V_{r_2} \text{ (as vane is smooth)}$$

Applying the sine rule to  $\triangle ACB$ ,

$$\frac{AB}{\sin(180^\circ - \theta)} = \frac{AC}{\sin(30^\circ - \alpha)} \quad \text{or} \quad \frac{V_1}{\sin(180^\circ - 30^\circ)} = \frac{u_1}{\sin(30^\circ - \alpha)}$$

$$\text{or} \quad \frac{15}{\sin 30^\circ} = \frac{5}{\sin(30^\circ - \alpha)} \quad \text{or} \quad \sin(30^\circ - \alpha) = \frac{5 \sin 30^\circ}{15}$$

$$= \frac{1}{3} \times 0.5 = .1667 = \sin 9.596^\circ$$

$$\therefore 30^\circ - \alpha = 9.596^\circ \quad \text{or} \quad \alpha = 30^\circ - 9.596^\circ = 20.404^\circ \quad \text{or} \quad \mathbf{20^\circ 24'}. \text{ Ans.}$$

Also from sine rule to  $\triangle ACB$ , we have

$$\frac{AB}{\sin(180^\circ - 30^\circ)} = \frac{CB}{\sin \alpha} \quad \text{or} \quad \frac{V_1}{\sin 30^\circ} = \frac{V_{r_1}}{\sin 20.404^\circ}$$

$$\therefore V_{r_1} = \frac{V_1 \sin 20.404^\circ}{\sin 30^\circ} = 10.46 \text{ m/s}$$

$$\therefore V_{r_2} = V_{r_1} = 10.46 \text{ m/s}$$

From velocity  $\triangle HEG$  at outlet,

$$V_{r_2} \cos \phi = u_2 + V_{w_2} \quad \text{or} \quad 10.46 \cos 30^\circ = 5.0 + V_{w_2}$$

$$\therefore V_{w_2} = 10.46 \cos 30^\circ - 5.0 = 4.06 \text{ m/s}$$

$$\text{Also, we have} \quad V_{r_2} \sin \phi = V_{f_2} \quad \text{or} \quad V_{f_2} = 10.46 \sin 30^\circ = 5.23 \text{ m/s}$$

$$\text{In } \triangle HFG, \quad V_2 = \sqrt{V_{f_2}^2 + V_{w_2}^2} = \sqrt{5.23^2 + 4.06^2}$$

$$= \sqrt{27.353 + 16.483} = \mathbf{6.62 \text{ m/s. Ans.}}$$

$$\tan \beta = \frac{V_{f_2}}{V_{w_2}} = \frac{5.23}{4.06} = 1.288 = \tan 52.17^\circ$$

$$\therefore \beta = 52.17^\circ \quad \text{or} \quad 52^\circ 10.2'$$

$$\therefore \text{Angle made by absolute velocity at outlet with the direction of motion } \beta^*$$

$$= 180^\circ - \beta = 180^\circ - (52^\circ 10.2') = 127^\circ 49.8'$$

$$\therefore \beta^* = \mathbf{127^\circ 49.8}. \text{ Ans.}$$

Work done\* per unit weight of the water striking

$$= \frac{1}{g} [V_{w_1} + V_{w_2}] \times u \text{ Nm } (\because + \text{ve sign taken as } \beta \text{ is an acute angle})$$

$$= \frac{1}{9.81} [V_1 \cos \alpha + 4.06] \times 5 \quad (\because V_{w_1} = V_1 \cos \alpha)$$

$$= \frac{5}{9.81} [15 \cos 20.404^\circ + 4.06] = \mathbf{9.225 \text{ Nm/N. Ans.}}$$

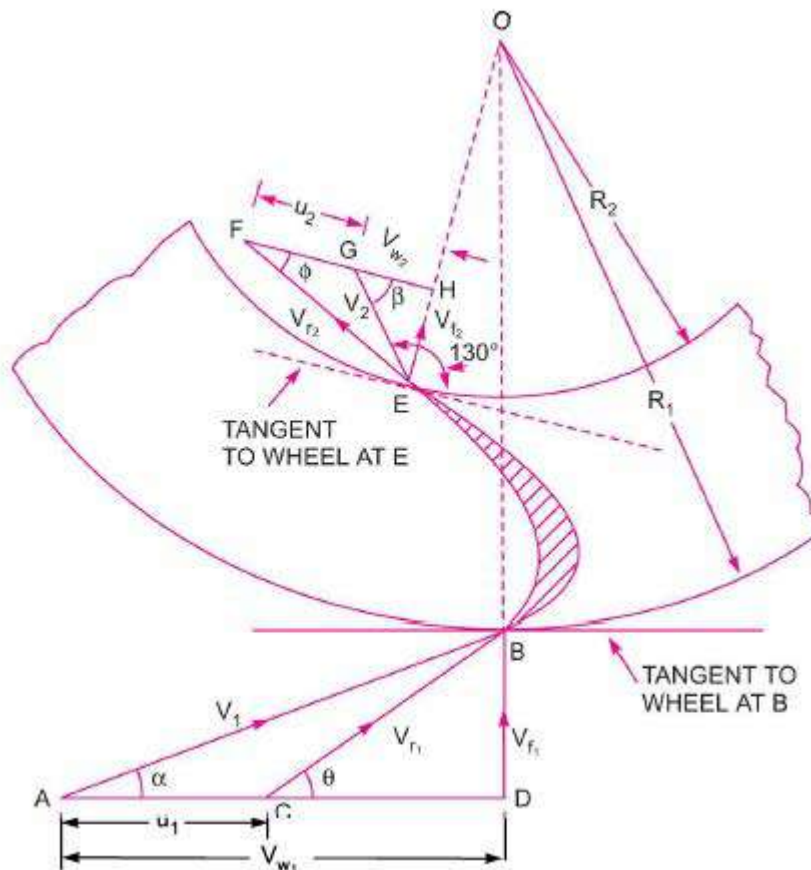
**9. CASE IV (c): Series of Moving Radial Curved Vanes... Jet Strikes Tangentially at one end:**

**Problem 17.26** *A jet of water having a velocity of 30 m/s strikes a series of radial curved vanes mounted on a wheel which is rotating at 200 r.p.m. The jet makes an angle of 20° with the tangent to the wheel at inlet and leaves the wheel with a velocity of 5 m/s at an angle of 130° to the tangent to the wheel at outlet. Water is flowing from outward in a radial direction. The outer and inner radii of the wheel are 0.5 m and 0.25 m respectively. Determine :*

- (i) Vane angles at inlet and outlet,                      (ii) Work done per unit weight of water, and  
(iii) Efficiency of the wheel.

**Solution.** Given :

Velocity of jet,	$V_1 = 30 \text{ m/s}$
Speed of wheel,	$N = 200 \text{ r.p.m.}$
$\therefore$ Angular speed,	$\omega = \frac{2\pi N}{60} = \frac{2\pi \times 200}{60} = 20.94 \text{ rad/s}$
Angle of jet at inlet,	$\alpha = 20^\circ$
Velocity of jet at outlet,	$V_2 = 5 \text{ m/s}$
Angle made by the jet at outlet with the tangent to wheel = $130^\circ$	
$\therefore$ Angle,	$\beta = 180^\circ - 130^\circ = 50^\circ$
Outer radius,	$R_1 = 0.5 \text{ m}$
Inner radius,	$R_2 = 0.25 \text{ m}$
$\therefore$ Velocity	$u_1 = \omega \times R_1 = 20.94 \times 0.5 = 10.47 \text{ m/s}$
And	$u_2 = \omega \times R_2 = 20.94 \times 0.25 = 5.235 \text{ m/s.}$



(i) **Vane angles** at inlet and outlet means the angle made by the relative velocities  $V_{r1}$  and  $V_{r2}$ , i.e., angle  $\theta$  and  $\phi$ .

From  $\triangle ABD$ ,  $V_{w1} = V_1 \cos \alpha = 30 \times \cos 20^\circ = 28.19 \text{ m/s}$

$$V_{f1} = V_1 \sin \alpha = 30 \times \sin 20^\circ = 10.26 \text{ m/s}$$

In  $\triangle CBD$ ,  $\tan \theta = \frac{BD}{CD} = \frac{V_{f1}}{AD - AC} = \frac{10.26}{V_{w1} - u_1} = \frac{10.26}{28.19 - 10.47} = 0.579 = \tan 30.07$

$\therefore \theta = 30.07^\circ$  or  **$30^\circ 4.2'$  Ans.**

From outlet velocity  $\Delta$ ,  $V_{w2} = V_2 \cos \beta = 5 \times \cos 50^\circ = 3.214 \text{ m/s}$

$$V_{f2} = V_2 \times \sin \beta = 5 \sin 50^\circ = 3.83 \text{ m/s}$$



In  $\triangle EFH$ , 
$$\tan \phi = \frac{V_{f2}}{u_2 + V_{w2}} = \frac{3.83}{5.235 + 3.214} = 0.453 = \tan 24.385^\circ$$

$\therefore \phi = 24.385^\circ$  or  **$24^\circ 23.1'$** . Ans.

(ii) Work done per second by water is given by equation (17.26)

$$= \rho a V_1 [V_{w1} u_1 + V_{w2} u_2]$$

(+ ve sign is taken as  $\beta$  is acute angle in Fig.17.24)

$\therefore$  Work done\* per second per unit weight of water striking per second

$$\begin{aligned} &= \frac{\rho a V_1 [V_{w1} u_1 + V_{w2} u_2]}{\text{Weight of water/s}} = \frac{\rho a V_1 [V_{w1} u_1 + V_{w2} u_2]}{\rho a V_1 \times g} \\ &= \frac{1}{g} [V_{w1} u_1 + V_{w2} u_2] \text{ Nm/N} = \frac{1}{9.81} [28.19 \times 10.47 + 3.214 \times 5.235] \\ &= \frac{1}{9.81} [295.15 + 16.82] = \mathbf{31.8 \text{ Nm/N. Ans.}} \end{aligned}$$

(iii) Efficiency,  $\eta$  is given by equation (17.28) as

$$\begin{aligned} \eta &= \frac{2 [V_{w1} u_1 + V_{w2} u_2]}{V_1^2} = \frac{2 [28.19 \times 10.47 + 3.214 \times 5.235]}{30^2} \\ &= \frac{2 [295.15 + 16.82]}{30 \times 30} = 0.6932 \text{ or } \mathbf{69.32\% \text{ Ans.}} \end{aligned}$$

# UNIT-2

## HYDRAULIC TURBINES

### 1. GENERAL LAYOUT OF A HYDROELECTRIC POWER PLANT

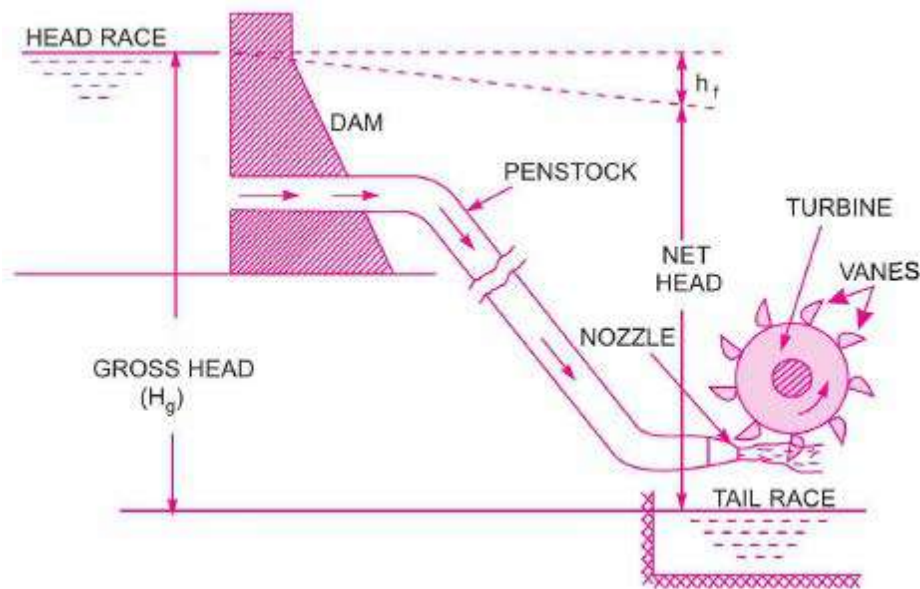


Fig. shows a general layout of a hydroelectric power plant which consists of :

- (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 channel is also known as tail race.

### 2. 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.

**2. Net Head.** It is also called effective head and is defined as the head available at the inlet of the turbine.

$$H = H_g - h_f$$

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

**Efficiencies of a Turbine.** The following are the important efficiencies of a turbine.

- (a) Hydraulic Efficiency,  $\eta_h$  (b) Mechanical Efficiency,  $\eta_m$   
 (c) Volumetric Efficiency,  $\eta_v$ , and (d) Overall Efficiency,  $\eta_o$

(a) **Hydraulic Efficiency ( $\eta_h$ ).**

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

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

$$= \frac{W}{g} \frac{[V_{w_1} u_1 \pm V_{w_2} u_2]}{1000} \text{ kW}$$

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

$$= \frac{W \times H}{1000} \text{ kW}$$

(b) **Mechanical Efficiency ( $\eta_m$ ).**

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

(c) **Volumetric Efficiency ( $\eta_v$ ).**

$$\eta_v = \frac{\text{Volume of water actually striking the runner}}{\text{Volume of water supplied to the turbine}}$$

(d) **Overall Efficiency ( $\eta_o$ ).**

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

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

### 3. CLASSIFICATION OF HYDRAULIC 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**.

If at the inlet of the turbine, the water possesses kinetic energy as well as pressure energy, the turbine is known as **reaction turbine**.

If the water flows along the tangent of the runner, the turbine is known as **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 through the runner along the direction parallel to the axis of rotation of the runner, the turbine is called **axial flow turbine**.

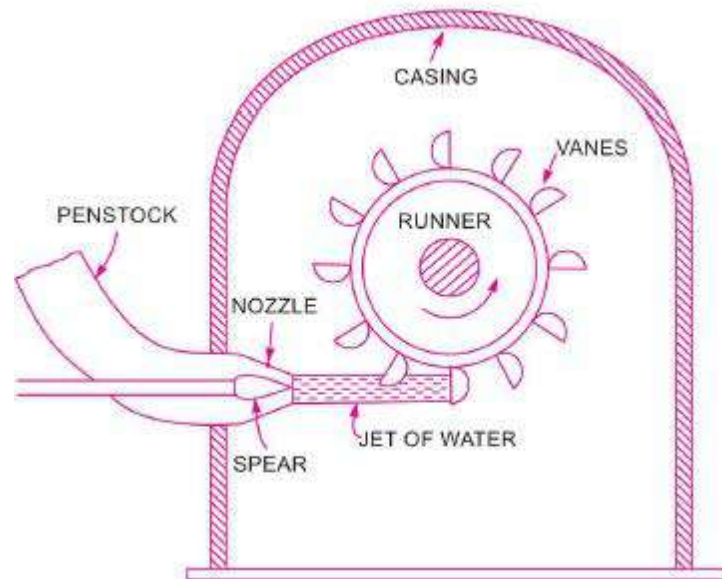
If the water flows through the runner in the radial direction but leaves in the direction parallel to axis of rotation of the runner, the turbine is called **mixed flow turbine**.

#### 4. Difference between Impulse and Reaction Turbines:

Impulse Turbine	Reaction Turbine
1. In impulse turbine only kinetic energy is used to rotate the turbine.	1. In reaction turbine both kinetic and pressure energy is used to rotate the turbine.
2. In this turbine water flow through the nozzle and strike the blades of turbine.	2. In this turbine water is guided by the guide blades to flow over the turbine.
3. All pressure energy of water converted into kinetic energy before striking the vanes.	3. In reaction turbine, there is no change in pressure energy of water before striking.
4. The pressure of the water remains unchanged and is equal to atmospheric pressure during process.	4. The pressure of water is reducing after passing through vanes.
5. Water may admitted over a part of circumference or over the whole circumference of the wheel of turbine.	5. Water may admitted over a part of circumference or over the whole circumference of the wheel of turbine.
6. In impulse turbine casing has no hydraulic function to perform because the jet is at atmospheric pressure. This casing serves only to prevent splashing of water.	6. Casing is absolutely necessary because the pressure at inlet of the turbine is much higher than the pressure at outlet. It is sealed from atmospheric pressure.
7. This turbine is most suitable for large head and lower flow rate. Pelton wheel is the example of this turbine.	7. This turbine is best suited for higher flow rate and lower head situation.

## 5. PELTON WHEEL (OR TURBINE)

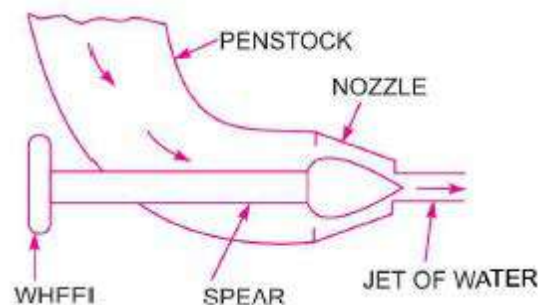
The Pelton wheel or Pelton turbine is a tangential flow impulse turbine. The water strikes the bucket along the tangent of the runner. The energy available at the inlet of the turbine is only kinetic energy. The pressure at the inlet and outlet of the turbine is atmospheric. This turbine is used for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



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. 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.



### 2. Runner with Buckets.

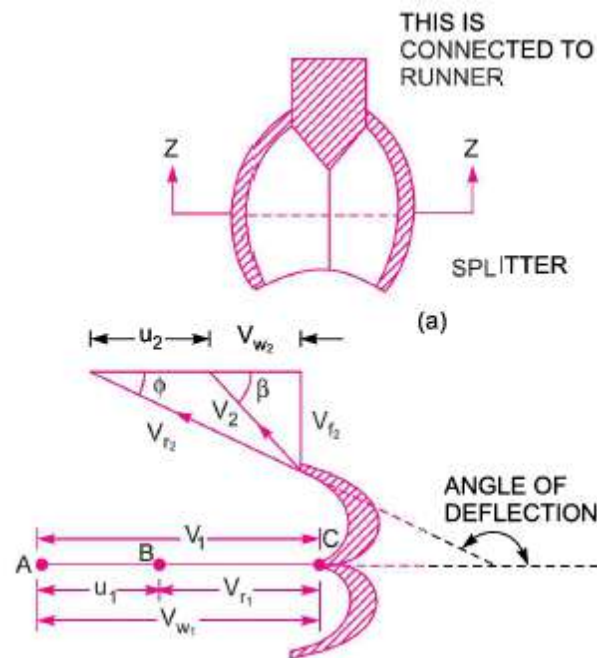
It consists of a circular disc on the periphery of which a number of buckets evenly spaced are fixed. The shape of the buckets is of a double hemispherical cup or bowl. Each bucket is divided into two symmetrical parts by a dividing wall which is known as splitter.

### 3. Casing.

The function of the casing is to prevent the splashing of the water and to discharge water to tail race. It is made of cast iron or fabricated steel plates. The casing of the Pelton wheel does not perform any hydraulic function.

**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.

### Velocity Triangles and Work done for Pelton Wheel.



Let  $H =$  Net head acting on the Pelton wheel  
 $= H_g - h_f$

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

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

$$V_{r1} = V_1 - u_1 = V_1 - u$$

$$V_{w1} = V_1$$

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

From the velocity triangle at outlet, we have

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

Now work done by the jet on the runner per second

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

Power given to the runner by the jet

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

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

$$= \frac{1}{g} [V_{w_1} + V_{w_2}] \times u$$

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

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

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

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

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

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

$$\text{or} \quad 2V_1 - 4u = 0 \quad \text{or} \quad u = \frac{V_1}{2} \quad \dots(18.14)$$

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  $u = \frac{V_1}{2}$

$$\therefore \text{Max. } \eta_h = \frac{(1 + \cos \phi)}{2}.$$

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

where  $C_v$  = Co-efficient 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  $\phi$  = 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  $165^\circ$  if no angle of deflection is given.

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

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

(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

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

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

$$Z = 15 + \frac{D}{2d} = 15 + 0.5m \quad \text{where } m = \text{Jet ratio}$$

(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  $160^\circ$ . Calculate the power given by water to the runner and the hydraulic efficiency of the turbine. Assume co-efficient of velocity as 0.98.

**Solution.** Given :

Speed of bucket,  $u = u_1 = u_2 = 10 \text{ m/s}$   
 Discharge,  $Q = 700 \text{ litres/s} = 0.7 \text{ m}^3/\text{s}$ , Head of water,  $H = 30 \text{ m}$   
 Angle of deflection  $= 160^\circ$   
 $\therefore$  Angle,  $\phi = 180^\circ - 160^\circ = 20^\circ$   
 Co-efficient of velocity,  $C_v = 0.98$ .

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

$\therefore$   $V_{r1} = V_1 - u_1 = 23.77 - 10$   
 $= 13.77 \text{ m/s}$

$V_{w1} = V_1 = 23.77 \text{ m/s}$

From outlet velocity triangle,

$V_{r2} = V_{r1} = 13.77 \text{ m/s}$

$V_{w2} = V_{r2} \cos \phi - u_2$   
 $= 13.77 \cos 20^\circ - 10.0 = 2.94 \text{ m/s}$

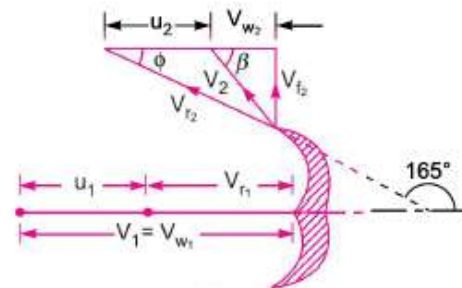


Fig. 18.6

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

$$\begin{aligned} &= \rho a V_1 [V_{w1} + V_{w2}] \times u \\ &= 1000 \times 0.7 \times [23.77 + 2.94] \times 10 \quad (\because aV_1 = Q = 0.7 \text{ m}^3/\text{s}) \\ &= 186970 \text{ Nm/s} \end{aligned}$$

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

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

$$\begin{aligned} \eta_h &= \frac{2[V_{w1} + V_{w2}] \times u}{V_1^2} = \frac{2[23.77 + 2.94] \times 10}{23.77 \times 23.77} \\ &= 0.9454 \text{ or } 94.54\%. \text{ Ans.} \end{aligned}$$



**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  $K_{u_1} = 0.45$

**Solution.** Given :

Shaft power, S.P. = 11,772 kW  
Head ,  $H = 380$  m  
Speed,  $N = 750$  r.p.m.

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

But  $u = \frac{\pi DN}{60} \quad \therefore \quad 38.85 = \frac{\pi DN}{60}$

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

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

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

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

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

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

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

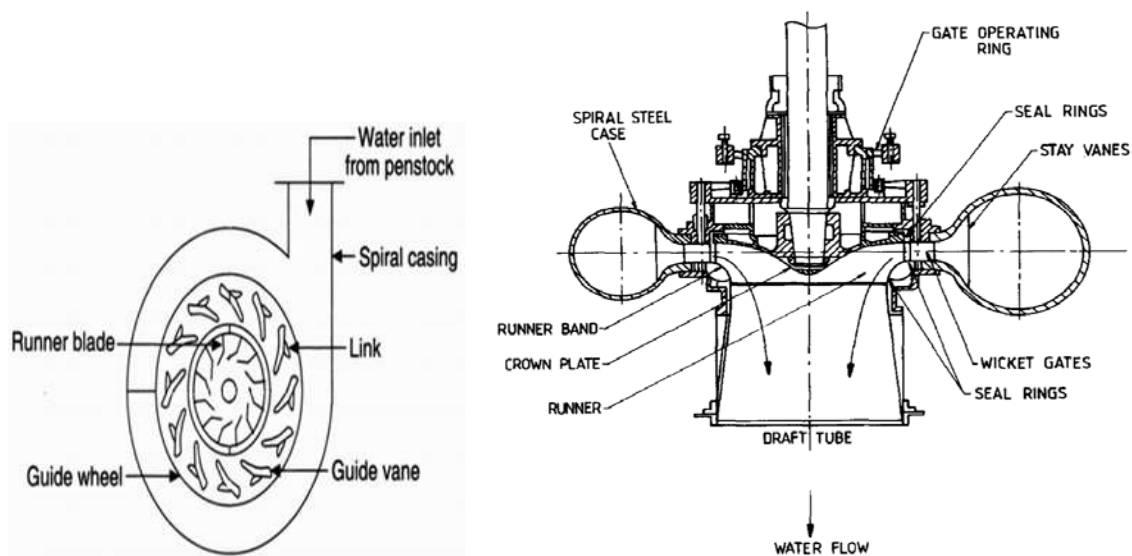
$\therefore$  Number of jets  $= \frac{\text{Total discharge}}{\text{Discharge of one jet}} = \frac{Q}{q} = \frac{3.672}{1.818} = \mathbf{2 \text{ jets. Ans.}}$

## 6. RADIAL FLOW REACTION TURBINES :

Radial flow turbines are those turbines 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.

## 7. 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 =  $\frac{1}{g} [V_{w_1} u_1]$

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

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

2. The flow ratio is given as, Flow ratio =  $\frac{V_{f_1}}{\sqrt{2gH}}$  and varies from 0.15 to 0.30.

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

**Problem 18.24** The following data is given for a Francis Turbine. Net head  $H = 60 \text{ m}$  ; Speed  $N = 700 \text{ r.p.m.}$ ; shaft power =  $294.3 \text{ kW}$  ;  $\eta_o = 84\%$  ;  $\eta_h = 93\%$ ; flow ratio =  $0.20$  ; breadth ratio  $n = 0.1$ ; Outer diameter of the runner =  $2 \times$  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 :

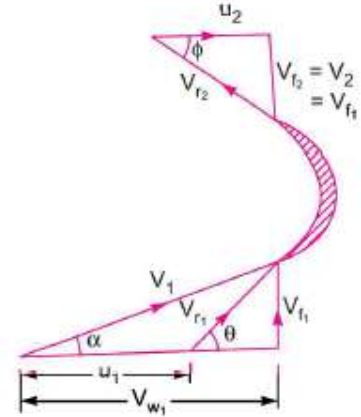
- (i) Guide blade angle, (ii) Runner vane angles at inlet and outlet,  
 (iii) Diameters of runner at inlet and outlet, and (iv) Width of wheel at inlet.

**Solution.** Given :

Net head,  $H = 60 \text{ m}$   
 Speed,  $N = 700 \text{ r.p.m.}$   
 Shaft power =  $294.3 \text{ kW}$   
 Overall efficiency,  $\eta_o = 84\% = 0.84$   
 Hydraulic efficiency,  $\eta_h = 93\% = 0.93$

Flow ratio,  $\frac{V_{f1}}{\sqrt{2gH}} = 0.20$

$\therefore V_{f1} = 0.20 \times \sqrt{2gH} = 6.862 \text{ m/s}$



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

Outer diameter,  $D_1 = 2 \times$  Inner diameter =  $2 \times D_2$

Velocity of flow,  $V_{f1} = V_{f2} = 6.862 \text{ m/s.}$

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

$\therefore V_{w2} = 0$  and  $V_{f2} = V_2$

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

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

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

But  $\text{W.P.} = \frac{WH}{1000} = \frac{\rho \times g \times Q \times H}{1000} = \frac{1000 \times 9.81 \times Q \times 60}{1000} = 350.357$

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

Using equation (18.21),  $Q = \text{Actual area of flow} \times \text{Velocity of flow}$

$$= 0.95 \pi D_1 \times B_1 \times V_{f1}$$

$$= 0.95 \times \pi \times D_1 \times 0.1 D_1 \times V_{f2} \quad (\because B_1 = 0.1 D_1)$$

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

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

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

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

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,

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

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

(i) Guide blade angle ( $\alpha$ )

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

$$\therefore \alpha = \tan^{-1} 0.248 = 13.928^\circ \text{ or } 13^\circ 55.7'. \text{ Ans.}$$

(ii) Runner vane angles at inlet and outlet ( $\theta$  and  $\phi$ )

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

$$\therefore \theta = \tan^{-1} 0.872 = 41.09^\circ \text{ or } 41^\circ 5.4'. \text{ Ans.}$$

From outlet velocity triangle,  $\tan \phi = \frac{V_{f_2}}{u_2} = \frac{V_{f_1}}{u_2} = \frac{6.862}{u_2} \quad \dots(i)$

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

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

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

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

$$\therefore \phi = \tan^{-1} .6934 = 34.74 \text{ or } 34^\circ 44.4'. \text{ Ans.}$$

(iii) Diameters of runner at inlet and outlet

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

(iv) Width of wheel at inlet  $B_1 = 54 \text{ mm. Ans.}$

## 8. AXIAL FLOW REACTION TURBINE (KAPLAN):

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 :

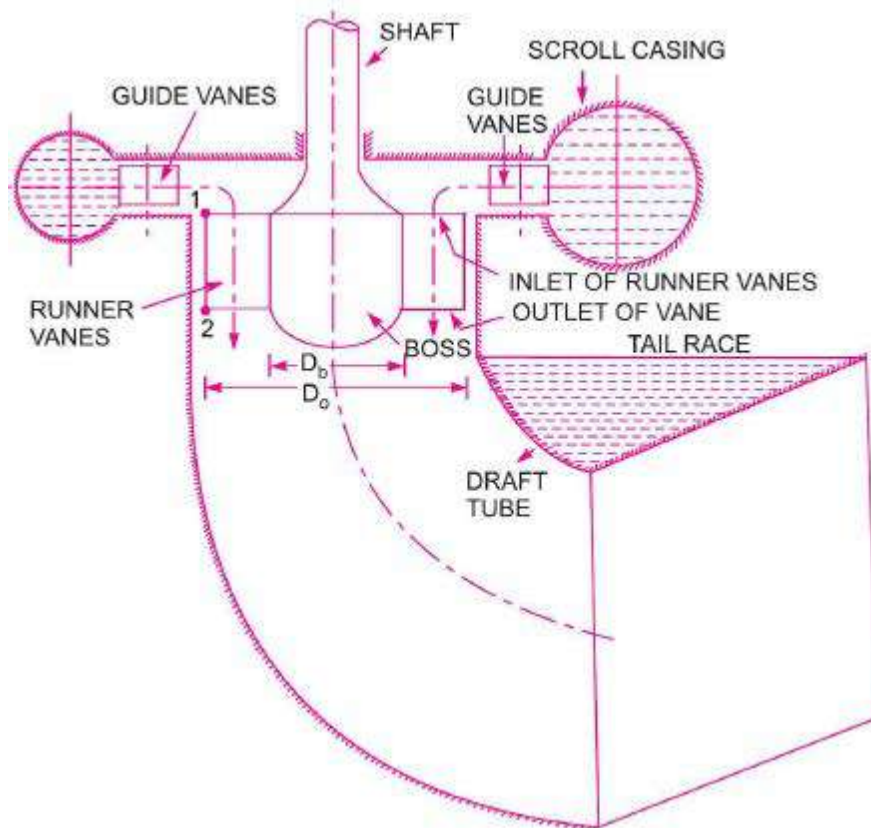
1. Propeller Turbine, and

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*

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.



The discharge through the runner is obtained as  $Q = \frac{\pi}{4}(D_o^2 - D_b^2) \times V_f$

where  $D_o$  = Outer diameter of the runner,  $D_b$  = Diameter of hub, and  $V_f$  = Velocity of flow at inlet.

1. The peripheral velocity at inlet and outlet are equal

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

2. Velocity of flow at inlet and outlet are equal

$$\therefore V_{f1} = V_{f2}$$

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 is 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 :

- Runner vane angles at inlet and outlet at the extreme edge of the runner, and
- Speed of the turbine.

**Solution.** Given :

Head,	$H = 20 \text{ m}$
Shaft power,	S.P. = 11772 kW
Outer dia. of runner,	$D_o = 3.5 \text{ m}$
Hub diameter,	$D_b = 1.75 \text{ m}$
Guide blade angle,	$\alpha = 35^\circ$
Hydraulic efficiency,	$\eta_h = 88\%$
Overall efficiency,	$\eta_o = 84\%$
Velocity of whirl at outlet	= 0.

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

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

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

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

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

Using equation  $Q = \frac{\pi}{4} (D_o^2 - D_b^2) \times V_{f1}$

$$\text{or } 71.428 = \frac{\pi}{4} (3.5^2 - 1.75^2) \times V_{f1} = \frac{\pi}{4} (12.25 - 3.0625) V_{f1} = 7.216 V_{f1}$$

$$\therefore V_{f1} = \frac{71.428}{7.216} = 9.9 \text{ m/s}.$$

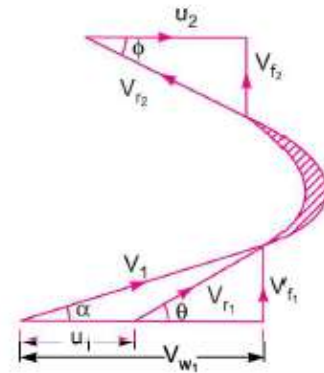


Fig. 18.27

From inlet velocity triangle,  $\tan \alpha = \frac{V_{f_2}}{V_{w_1}}$

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

Using the relation for hydraulic efficiency,

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

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

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

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

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

$$\therefore \theta = \tan^{-1} 5.13 = 78.97^\circ \text{ or } 78^\circ 58'. \text{ Ans.}$$

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

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

$$\therefore \phi = \tan^{-1} .811 = 39.035^\circ \text{ or } 39^\circ 2'. \text{ Ans.}$$

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

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

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

## 9. DRAFT-TUBE :

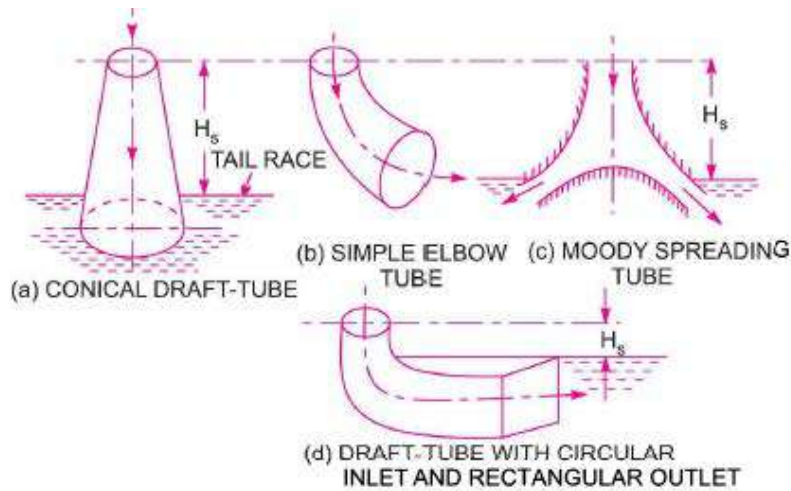
The draft-tube is a pipe of gradually increasing area which connects the outlet of the runner to the tail race. It is used for discharging water from the exit of the turbine to the tail race. This pipe of gradually increasing area is called a draft-tube. One end of the draft-tube is connected to the outlet of the runner while the other end is sub-merged below the level of water in the tail race. The draft-tube, in addition to serve a passage for water discharge, has the following two purposes also :

1. It permits a negative head to be established at the outlet of the runner and thereby increase the net head on the turbine. The turbine may be placed above the tail race without any loss of net head and hence turbine may be inspected properly.

2. It converts a large proportion of the kinetic energy ( $V_2^2/2g$ ) rejected at the outlet of the turbine into useful pressure energy. Without the draft tube, the kinetic energy rejected at the outlet of the turbine will go waste to the tail race.

**Types of Draft-Tubes.** The following are the important types of draft-tubes which are commonly used :

1. Conical draft-tubes,
2. Simple elbow tubes,
3. Moody spreading tubes, and
4. Elbow draft-tubes with circular inlet and rectangular outlet.



**Draft-Tube Theory.** Consider a capital draft-tube as shown in Fig.

Let  $H_s$  = Vertical height of draft-tube above the tail race,  
 $y$  = Distance of bottom of draft-tube from tail race.

Applying Bernoulli's equation to inlet (section 1-1) and outlet (section 2-2) of the draft-tube and taking section 2-2 as the datum line, we get

$$\frac{p_1}{\rho g} + \frac{V_1^2}{2g} + (H_s + y) = \frac{p_2}{\rho g} + \frac{V_2^2}{2g} + 0 + h_f \quad \dots(i)$$

where  $h_f$  = loss of energy between sections 1-1 and 2-2.

But  $\frac{p_2}{\rho g}$  = Atmospheric pressure head +  $y$   
 $= \frac{p_a}{\rho g} + y$ .

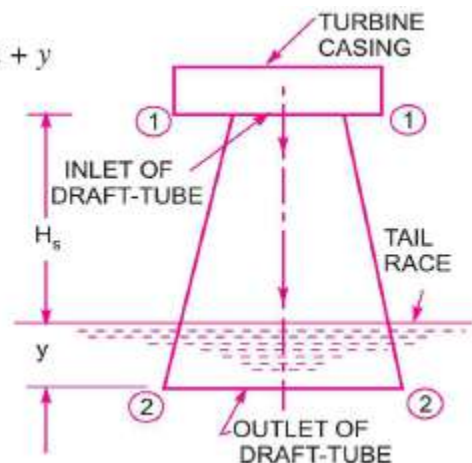
Substituting this value of  $\frac{p_2}{\rho g}$  in equation (i), we get

$$\frac{p_1}{\rho g} + \frac{V_1^2}{2g} + (H_s + y) = \frac{p_a}{\rho g} + y + \frac{V_2^2}{2g} + h_f$$

or  $\frac{p_1}{\rho g} + \frac{V_1^2}{2g} + H_s = \frac{p_a}{\rho g} + \frac{V_2^2}{2g} + h_f$

$\therefore \frac{p_1}{\rho g} = \frac{p_a}{\rho g} + \frac{V_2^2}{2g} + h_f - \frac{V_1^2}{2g} - H_s$

$$= \frac{p_a}{\rho g} - H_s - \left( \frac{V_1^2}{2g} - \frac{V_2^2}{2g} - h_f \right)$$





**Efficiency of Draft-Tube.** The efficiency of a draft-tube is defined as the ratio of actual conversion of kinetic head into pressure head in the draft-tube to the kinetic head at the inlet of the draft-tube. Mathematically, it is written as

$$\eta_d = \frac{\text{Actual conversion of kinetic head into pressure head}}{\text{Kinetic head at the inlet of draft-tube}}$$

Let

$V_1$  = Velocity of water at inlet of draft-tube,

$V_2$  = Velocity of water at outlet of draft-tube, and

$h_f$  = Loss of head in the draft-tube.

Theoretical conversion of kinetic head into pressure head in draft-tube =  $\left( \frac{V_1^2}{2g} - \frac{V_2^2}{2g} \right)$ .

Actual conversion of kinetic head into pressure head =  $\left( \frac{V_1^2}{2g} - \frac{V_2^2}{2g} \right) - h_f$

$$\therefore \eta_d = \frac{\left( \frac{V_1^2}{2g} - \frac{V_2^2}{2g} \right) - h_f}{\left( \frac{V_1^2}{2g} \right)}$$

# CENTRIFUGAL PUMPS

**Definition:-** A centrifugal Pump is a hydraulic machine used to raise the water (liquid) from lower level to higher level by creating a required pressure by means of centrifugal action. Centrifugal pumps are the machines which increase the pressure energy of a fluid. These may either lift the fluid or boost the pressure in a pipe line.

## classifications

centrifugal pumps are classified as

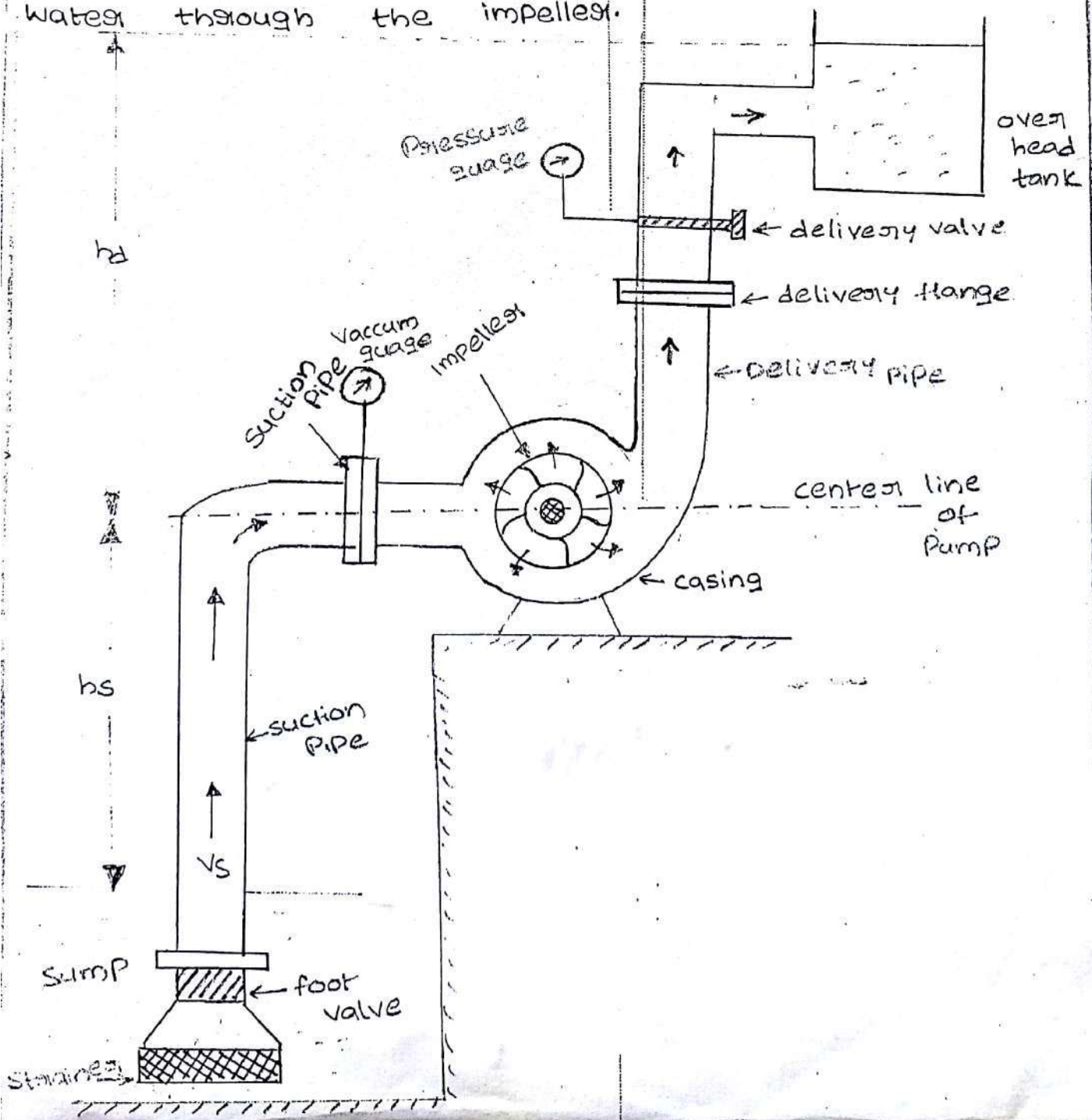
- i) According to type of casing:-
  - a) Volute pumps
  - b) Vortex pumps
  - c) Diffusion pumps
- ii) According to type of impeller:-
  - a) closed impeller pump
  - b) semi-closed impeller pump
  - c) open impeller pump
- iii) According to working head
  - a) low head pumps : up to 15 m water
  - b) Medium head pumps : 15 to 40 m of water
  - c) high head pumps : above 40 m of water
- iv) According to direction of flow through impeller:-
  - a) Radial flow pumps
  - b) Axial flow pumps
  - c) mixed flow pumps
- v) According to no. of impellers per shaft:-
  - a) single stage centrifugal pump
  - b) Multi stage centrifugal pump
- vi) According to no. of entrances to impeller:-
  - a) single entry (or) single suction pump
  - b) Double entry (or) Double suction pump

# COMPONENTS OF CENTRIFUGAL PUMP

A centrifugal pump consists of following components

- i) impeller
- (ii) casing
- (iii) suction Pipe
- (iv) Delivery Pipe

Impeller :- An impeller is a wheel (or) rotor with a series of backward curved vanes (blades). It is mounted on the shaft which is usually coupled to an electric motor. The function of impeller is to force the water into a rotating motion and the shroud around the impeller is to direct the water through the impeller.



casing: The casing is an air-tight chamber surrounding the pump impeller. It contains suction and discharge arrangements, supporting for bearings and facilitates to house the motor assembly. It has provision to fix stuffing box and house packing materials which prevent external leakage. The essential purposes of casing are

- \* To guide the water to and from the impeller
- \* To convert partially kinetic energy into pressure energy.

**Suction pipe:-** The pipe which connects the central edge of impeller to sump from which liquid is to be lifted is known as suction pipe. To prevent the entry of solid particles, into the pump, the suction pipe is provided with a strainer at its lowest end. The lowest end of the pipe is also fitted with a non-return foot valve which does not permit the liquid to drain out of the suction pipe when pump is not working; this also helps in priming.

**Delivery pipe:-** The pipe which is connected at its lowest end to the outlet of the pump and it delivers the liquid to the required height is known as delivery pipe. A regulating valve is provided on the delivery pipe to regulate the supply of water. The velocity of water in delivery pipe is equal (or) slightly higher than velocity of suction pipe.

### Working

A centrifugal pump works on the principle that when a certain mass of fluid is rotated by an external source, it is thrown away from the central axis of rotation and centrifugal head is impressed which enables it to rise to a higher level. The operation/working of centrifugal has the following steps.

- (i) The delivery valve is closed and the Pump is Primed that is, suction Pipe, casing and position of the delivery pipe upto delivery valve are completely filled with the liquid so that no air Pocket is left.
- (ii) Keeping the delivery valve still closed the electric motor is started to rotate the impeller. The rotation of the impeller causes strong suction or vacuum just at the eye of the casing.
- (3) The speed of the impeller is gradually increased till the impeller rotates at its normal speed and develops normal energy required for pumping the liquid.
- 4) After the impeller attains the normal speed the delivery valve is opened when the liquid is continuously sucked up the suction pipe. Due to impeller action the pressure head as well as velocity heads of the liquid are increased.
- 5) From casing, the liquid passes into pipe and is lifted to required height.
- 6) So long as motion is given to the impeller and there is supply of liquid to be lifted the process of lifting the liquid to the required height remains continuous.
- 7) When Pump is to be stopped delivery valve should be closed first, otherwise there may be some back flow from the reservoir.

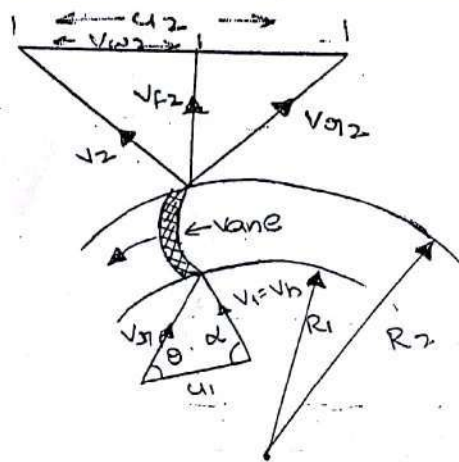
Work done:- The expression for work done (or) energy supplied by the impeller of a centrifugal pump may be derived in the same way as for turbine. The liquid enters the impeller at its center and leaves at its outer periphery.

Torque on impeller = rate of change of momentum

Moment of Momentum of inlet = 0

Moment of Momentum of outlet =  $PQ (V_{w2} R_2)$

$\therefore$  Torque =  $PQ (V_{w2} R_2)$



Work done/sec = Torque  $\times$  Angular velocity

$$\Rightarrow PQ V_{w2} R_2 \times \omega \Rightarrow PQ V_{w2} R_2 u_2$$

Work done/sec unit weight of liquid =  $\frac{1}{g} V_{w2} u_2$

If the flow is not radial at inlet

Work done =  $\frac{1}{g} [V_{w2} u_2 - V_{w1} u_1]$   $\rightarrow$  it is known as

Euler's momentum equation for centrifugal pump.

Volume of liquid =  $Q = \pi D_1 B_1 V_{f1} = \pi D_2 B_2 V_{f2}$

From outlet velocity triangle

$$V_{f2}^y = V_{a2}^y - (u_2 - V_{w2})^y \text{ and } V_{f2}^y = V_2^y - V_{w2}^y$$

$$V_2^y - V_{w2}^y = V_{a2}^y - u_2 - V_{w2}^y + 2u_2 V_{w2}$$

(or)

$$u_2 \cdot V_{w2} = \frac{1}{2} [V_2^y + u_2^y - V_{w2}^y]$$

Similarly,

from inlet velocity triangle  $u_1 V_{w1} = \frac{1}{2} (V_1^y + u_1^y - V_{a1}^y)$

$$\therefore \text{Work done} = \frac{V_2^y - V_1^y}{2g} + \frac{u_2^y - u_1^y}{2g} + \frac{V_{a1}^y - V_{a2}^y}{2g} \rightarrow \text{This eqn.}$$

is known as fundamental equation of centrifugal pump.

### Heads

i) static head: <sup>(Hs)</sup> The sum of suction head (hs) and delivery head (hd) is known as static head

$$\therefore H_s = h_s + h_d$$

**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 the water is to be lifted.

**Delivery head ( $h_d$ ):** - The vertical distance b/w the centreline of the pump and the water surface in the tank to which water is delivered is known as delivery head.

**Manometric head ( $H_m$ ):** - The head against which a centrifugal pump has to work is known as the manometric head. It is the head measured across the pump inlet and outlet flanges.

$H_m =$  head imparted by the impeller to liquid - loss of head in the pump

$$= \frac{V W_2 u_2}{g} - \text{losses}$$

$$H_m = h_s + h_d + h_{fs} + h_{fd} + \frac{V_d^2}{2g}$$

Where,

$h_{fs}, h_{fd}$  : friction head losses in suction and delivery pipes

$V_d =$  velocity in delivery pipe

$H_m =$  total head at outlet of pump - Total head at inlet

$$= \left[ \frac{P_2}{\omega} + \frac{V_2^2}{2g} + z_2 \right] - \left[ \frac{P_1}{\omega} + \frac{V_1^2}{2g} + z_1 \right]$$

losses

Various losses occurring in centrifugal pump are

i) Hydraulic losses:-

a) Shock or eddy losses at the entrance and exist from the impeller.

- b) losses due to friction in the impeller
- c) Friction and eddy losses in the Guide vanes / diffused & casing
- d) Friction and other minor losses in suction pipe
- e) Friction and other minor losses in delivery pipe.

### 2. Mechanical losses:-

- a) losses due to friction b/w the impeller and the liquid which fills the clearance spaces b/w the impeller and casing.
- b) losses pertaining to friction of the main bearings and glands.

### 3. leakage losses:-

The loss of energy due to leakage of liquid is known as leakage loss.

### EFFICIENCIES

Various efficiencies of centrifugal pump are

i) Manometric head efficiency ( $\eta_{mano}$ ):- The ratio of the Manometric head developed by the pump to the head imparted by the impeller to the liquid is known as Manometric (head) efficiency

$$\eta_{mano} = \frac{\text{Manometric head}}{\text{Head imparted by impeller to liquid}}$$

$$= \frac{H_m}{\frac{V_w u_2^2}{g}} \Rightarrow \frac{g \cdot H_m}{V_w u_2^2}$$

ii) Volumetric efficiency ( $\eta_v$ ):- The ratio of quantity liquid discharged / second from pump to quantity passing per second through impeller is known as volumetric efficiency.



$$\eta_v = \frac{Q}{Q+z}$$

Where,

$Q$  = Actual liquid discharge at pump outlet/  
second

$z$  = leakage<sup>or</sup> liquid per second.

Mechanical efficiency ( $\eta_m$ ) :- The ratio of the Power delivered by the impeller to the liquid to the Power input to the pump shaft is known as mechanical efficiency ( $\eta_m$ )

$$\eta_m = \frac{P - P_{\text{mech. losses}}}{P} = \frac{w(Q+z) \cdot (Vw_2/2g)}{P}$$

Overall efficiency :- The ratio of output of the Pump to the input of the pump is known as overall efficiency ( $\eta_o$ )

$$\eta_o = \frac{wQ H_{\text{mano}}}{P} \quad \text{and} \quad \eta_o = \eta_{\text{mano}} \times \eta_v \times \eta_m$$

Specific speed :- The specific speed of a centrifugal Pump is defined as the speed of a geometrically similar pump which would deliver unit quantity ( $1 \text{ m}^3/\text{sec}$ ) against a unit head ( $1 \text{ m}$ ). It is denoted by  $N_s$

discharge

(or)

$$Q = \text{Area} \times \text{velocity of flow} = \pi D B V_f$$

$$Q = D \times B \times V_f \rightarrow (1)$$

$$Q = D^2 \times V_f [B \propto D] \rightarrow (2)$$

$$\text{Tangential velocity } (u) = \frac{\pi D N}{60} \quad \text{(or)} \quad u \propto D N \rightarrow (3)$$

$$\text{also } u \propto V_f \propto \sqrt{H_{\text{mano}}} \rightarrow (4)$$

from (3) & (4)

$$\sqrt{H_{\text{mano}}} \propto D N \quad \text{[or]} \quad D \propto \frac{\sqrt{H_{\text{mano}}}}{N} \rightarrow (5)$$

From (2) and (5)

$$Q \propto \frac{H_{mano}}{N^2} \times V_f \propto \frac{H_{mano}}{N^2} \times \sqrt{H_{mano}}$$

$$Q \propto \frac{[H_{mano}]^{3/2}}{N^2}$$

$$Q = K \cdot \frac{[H_{mano}]^{3/2}}{N^2}, \text{ where } K = \text{constant}$$

When,

$$H_m = 1m \text{ and } Q = 1m^3/sec$$

$$1 = K \cdot \frac{(1)^{3/2}}{N_s^2} \quad (or) \quad K = N_s^2$$

$$\therefore Q = N_s^2 \cdot \frac{[H_{mano}]^{3/2}}{N^2} \quad (or) \quad N_s^2 = \frac{QN^2}{[H_{mano}]^{3/2}}$$

$$\therefore \text{Specific head speed } (N_s) = \frac{N\sqrt{Q}}{[H_{mano}]^{3/4}}$$

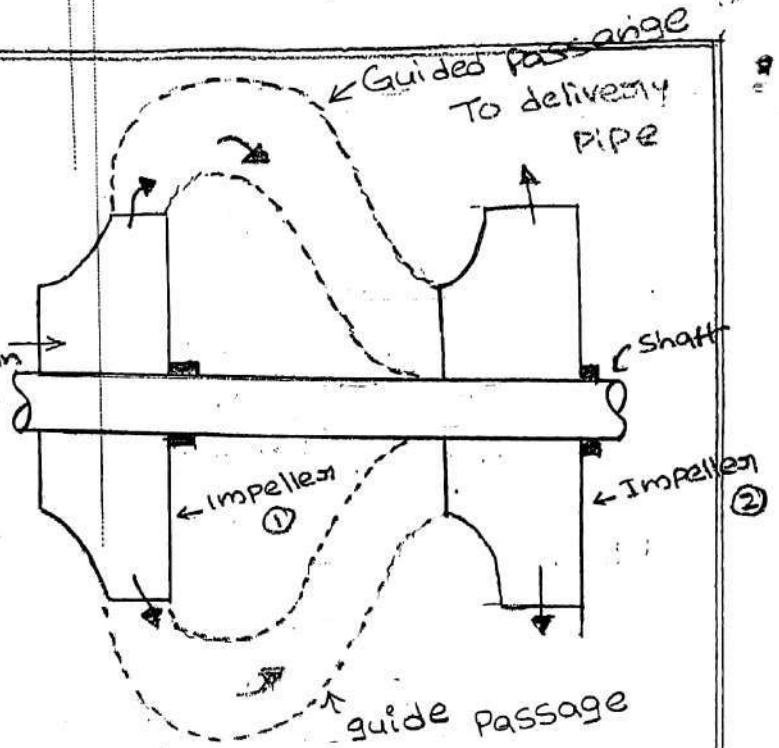
### MULTI-STAGE CENTRIFUGAL PUMPS:-

A Multi stage centrifugal pump is one which has two or more identical impellers mounted on the same shaft or different shafts. The important functions performed by these are

- (i) To produce heads greater than that permissible with a single impeller discharge remaining constant-series
- (ii) To discharge a large quantity of liquid, head remaining same-parallel arrangement

**Pumps in series:-** For obtaining a high head, a no. of impellers are mounted in series or on the same shaft. The below figure shows such an arrangement for a two-stage pump. The discharge from impeller - (1) passes through a guided passage and enters the impeller - (2), At the

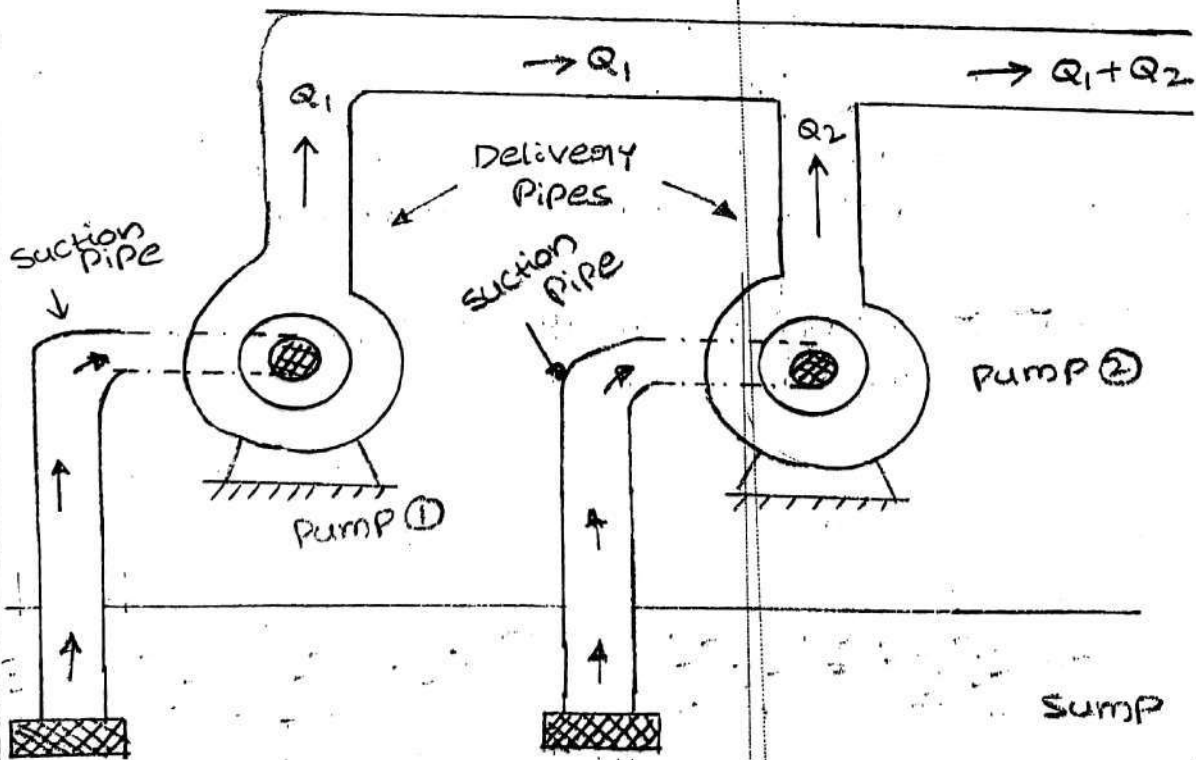
outlet of Impeller-②, the pressure of water will be more than the pressure of water at outlet of impeller -①. This if more no. of impellers are mounted on the same shaft the pressure at outlet is increased further.



If in each stage the manometric head imposed on the liquid is  $H_{mano}$  then for "n" identical impellers the total head developed will be  $H_{total} = n H_{mano}$ . However, the discharge passing through each impeller is same.

The series arrangement is employed for delivering a relatively small quantity of liquid against very high heads.

### PUMPS IN PARALLEL



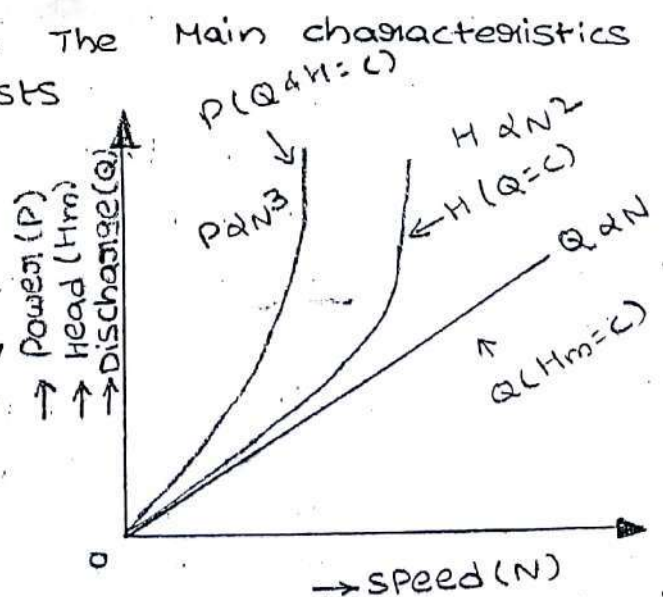
When a large quantity of liquid is required to be pumped against a relatively small head, two or more pumps are employed which are so arranged that each of these pumps working separately lifts the liquid from a common sump and delivers to common collecting pipe through which it is raised to required height. This arrangement is known as pumps in parallel. If  $Q$  is the discharge capacity of the pump and there are  $n$  identical pumps then total discharge will be

$$Q_{\text{total}} = n \cdot Q$$

### Characteristic Curves

Ordinarily a centrifugal pump is worked under maximum efficiency conditions. However when the pump runs at conditions different from the design conditions, it performs differently. Therefore the predict the behaviour of the pump under varying conditions of speeds, heads discharges (or) powers tests are usually conducted. The results obtained from these tests are plotted in the form of characteristic curves. These are

1) Main characteristic curves:- The main characteristics of a centrifugal pump consists of variation of head ( $H_m$ ), Power ( $P$ ), Discharge ( $Q$ ) w.r.t speed. For plotting curves of Manometric head vs speed, discharge is kept constant. For plotting discharge vs speed, Manometric head ( $H_m$ ) is kept constant. And for plotting curves of Power vs speed the Manometric head and discharge kept constant.



From, head coefficient  $\frac{H_m}{D^5 N^2} = \text{constant (or)} H_m \propto D^5 N^2$

This means the head developed by the pump is proportional to  $N^2$ . Hence, curve of  $H_m$  vs  $N$  is parabolic shape.

From, flow coefficient  $\frac{Q}{D^3 N} = \text{constant (or)} Q \propto D^3 N$

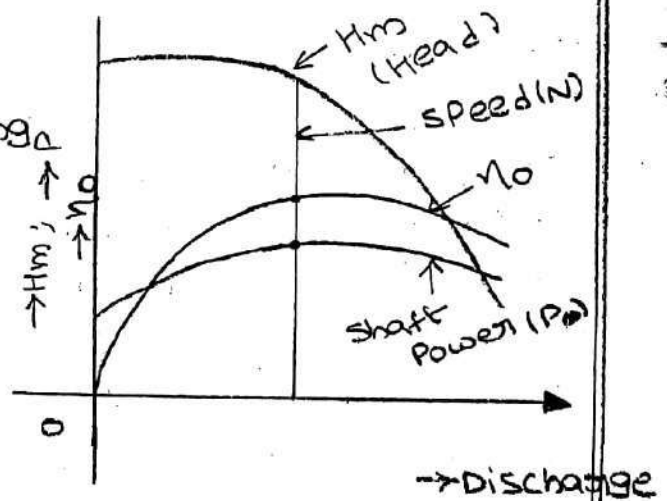
This means discharge is proportional to speed and the curve of  $Q$  vs  $N$  is straight line.

From, power coefficient  $\frac{P}{D^5 N^3} = \text{constant (or)} P \propto D^5 N^3$

This curve is  $P$  vs  $N$  is cubic curve.

### 2) operating characteristics:-

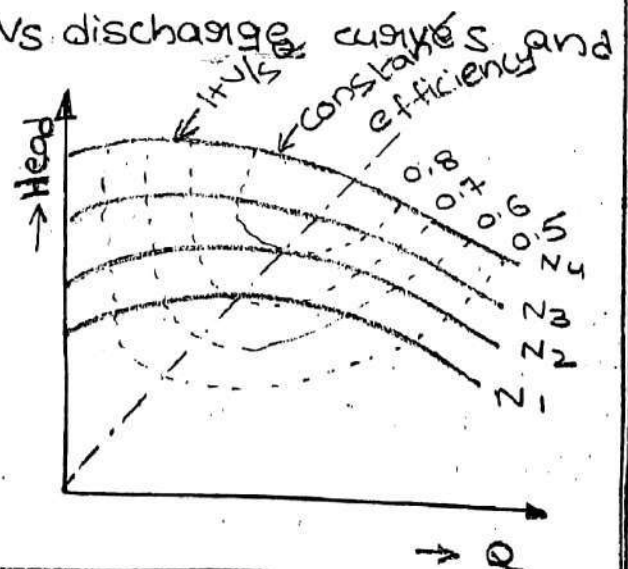
When a centrifugal pump operates at the design speed the maximum efficiency occurs. Evidently for optimum performance, the pump needs to be operated at the design speed. To obtain operating characteristics curves, the pump is run at the design speed and discharge is varied as in the case of main ch's curves.



### 3) constant efficiency curves:-

For obtaining constant efficiency curves for a pump, the head vs discharge curves for different speed curves are used.

For plotting the constant efficiency curves first  $\eta$  vs  $Q$  curves are drawn. These curves are transferred to the corresponding  $H$  vs  $Q$  curves. The points having the



same efficiency are then joined by smooth curves which represents the ISO efficiency (or) Mustel curves

The constant efficiency curves help to locate the regions where the pump would operate with Maximum efficiency.

Net Positive suction ~~pipe~~ Head (NPSH) :-

NPSH may be defined as "the difference b/w the net inlet head and the head corresponding to the vapour pressure of liquid". It may be also defined as "the net head that is required to make the liquid flow through the suction pipe from the sump to the impeller."

NPSH = Absolute pressure head - vapour pressure head + velocity head.

$$NPSH = \frac{P_1}{\rho g} + \frac{P_v}{\rho g} + \frac{V_s^2}{2g} \rightarrow \textcircled{1}$$

But Absolute pressure head, is given by,

$$\frac{P_1}{\rho g} = \frac{P_a}{\rho g} - \left[ \frac{V_s^2}{2g} + h_s + h_{fs} \right] \rightarrow \textcircled{2}$$

From  $\textcircled{1}$  &  $\textcircled{2}$

$$NPSH = \left[ \frac{P_a}{\rho g} - \left[ \frac{V_s^2}{2g} + h_s + h_{fs} \right] \right] - \frac{P_v}{\rho g} + \frac{V_s^2}{2g}$$

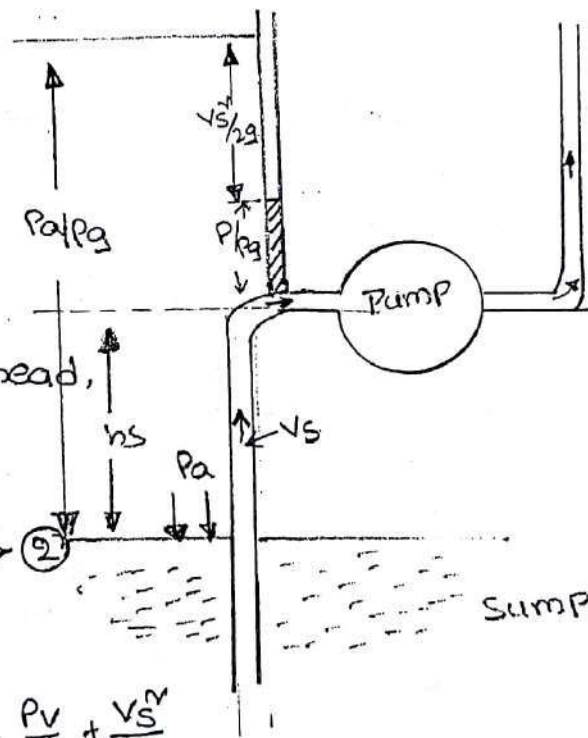
$$= \frac{P_a}{\rho g} - \frac{P_v}{\rho g} - h_s - h_{fs}$$

$$NPSH = H_a - H_v - h_s - h_{fs}$$

Where,  $H_a$ : atmospheric pressure head

$H_v$ : vapour pressure head

When pump is installed, the available NPSH is calculated from the above equation. In order to have evaluation free operation of centrifugal pump,



the available NPSH should be greater than required  
NPSH is given by manufacturer.

Model testing and Geometrically similar Pumps:-

In order to know performance of prototypes, models of centrifugal pumps are tested. Prototype are made of two geometrically similar pumps (1 & 2) the performance of these are predicted correctly if they ~~are~~ satisfy the following conditions.

1) Specific speed is same i.e.,  $\left[ \frac{N\sqrt{Q}}{(H_m)^{3/4}} \right]_1 = \left[ \frac{N\sqrt{Q}}{(H_m)^{3/4}} \right]_2$

2) Flow coefficient is same i.e.,  $\left[ \frac{Q}{D^3 N} \right]_1 = \left[ \frac{Q}{D^3 N} \right]_2$

3) Head coefficient is same i.e.,  $\left[ \frac{\sqrt{H_m}}{DN} \right]_1 = \left[ \frac{\sqrt{H_m}}{DN} \right]_2$

4) Power coefficient is same i.e.,  $\left[ \frac{P}{D^5 N^3} \right]_1 = \left[ \frac{P}{D^5 N^3} \right]_2$

### Problems

1) The diameter of centrifugal pump, which is discharging  $0.035 \text{ m}^3/\text{sec}$  of water against a total head of  $25 \text{ m}$  is  $0.05 \text{ m}$ . The pump is running at  $1200 \text{ r.p.m}$ . Find the head, discharge and ratio of powers of a geometrically similar pumps of dia  $0.3 \text{ m}$  when it is running at  $2000 \text{ r.p.m}$ .

Given data:-

$$Q_1 = 0.035 \text{ m}^3/\text{sec}$$

$$H_{m1} = 25 \text{ m}$$

$$D_1 = 0.05 \text{ m}$$

$$N_1 = 1200 \text{ r.p.m}$$

$$Q_2 = ?$$

$$H_{m2} = ?$$

$$D_2 = 0.3 \text{ m}$$

$$N_2 = 2000 \text{ r.p.m}$$

From head coefficients,

$$\frac{\sqrt{Hm_1}}{DN_1} = \frac{\sqrt{Hm_2}}{DN_2} \quad (\text{or}) \quad \frac{\sqrt{25}}{0.05 \times 1200} = \frac{\sqrt{Hm_2}}{0.3 \times 2000}$$

$$\Rightarrow Hm = 2500 \text{ m}$$

From flow coefficients,

$$\frac{Q_1}{D_1^3 N_1} = \frac{Q_2}{D_2^3 N_2} \Rightarrow \frac{0.035}{(0.05)^3 \times 1200} = \frac{Q_2}{(0.3)^3 \times 2000}$$

$$Q_2 = 12.6 \text{ m}^3/\text{sec}$$

From Power coefficients,

$$\frac{P_1}{D_1^5 N_1^3} = \frac{P_2}{D_2^5 N_2^3}$$

(or)

$$\frac{P_1}{(0.05)^5 (1200)^3} = \frac{P_2}{(0.3)^5 (2000)^3}$$

(or)

$$\frac{P_1}{P_2} = 2.78 \times 10^{-5}$$

2). A centrifugal Pump impeller whose external & internal diameters are 400mm and 200mm is running at 950 r.p.m. the rate of flow through the pump is 0.035 m<sup>3</sup>/sec. the suction and delivery heads are 5m and 25m. the diameters of suction and delivery pipes are 120mm and 80mm, if the outlet vane angle is 45°, the flow velocity is constant and equal to 1.8 m/sec and power required to drive the pump is 15KW, determine (i) inlet vane angle (ii) the overall efficiency (iii) the Manometric efficiency?



Given data :-

$$D_2 = 400 \text{ mm} = 0.4 \text{ m} ; N = 950 \text{ r.p.m}$$

$$D_1 = 200 \text{ mm} = 0.2 \text{ m} ; Q = 0.035 \text{ m}^3/\text{sec}$$

$$h_s = 5 \text{ m} ; h_d = 25 \text{ m} ; D_s = 120 \text{ mm} = 0.12 \text{ m} ; D_d = 80 \text{ mm} = 0.08 \text{ m}$$

$$\phi = 45^\circ ; P = 15 \text{ kW} ; V_{f1} = V_{f2} = 1.8 \text{ m/sec}$$

Tangential velocity at inlet,

$$u_1 = \frac{\pi D_1 N}{60} = \frac{\pi \times 0.2 \times 950}{60}$$

$$= 9.95 \text{ m/sec}$$

From inlet velocity triangle

$$\tan \theta = \frac{V_{f1}}{u_1} = \frac{1.8}{9.95} \Rightarrow \theta = 10.26^\circ$$

$$\text{Manometric head (Hm)} = \left[ \frac{P_2}{\rho} + \frac{V_2^2}{2g} + z_2 \right] - \left[ \frac{P_1}{\rho} + \frac{V_1^2}{2g} + z_1 \right]$$

Where,

$$\frac{P_2}{\rho} = h_d = 25 \text{ m} ; \frac{P_1}{\rho} = h_s = 5 \text{ m}$$

Assume,  $z_1 = z_2$  and  $V_1 = V_g$  and  $V_2 = V_d$

$$Q = Q_d V_d \Rightarrow 0.035 = \frac{\pi}{4} (0.08)^2 \cdot V_d \Rightarrow V_d = 6.963 \text{ m/sec}$$

$$Q = Q_s \cdot V_s \Rightarrow 0.035 = \frac{\pi}{4} (0.12)^2 \cdot V_s \Rightarrow V_s = 3.095 \text{ m/sec}$$

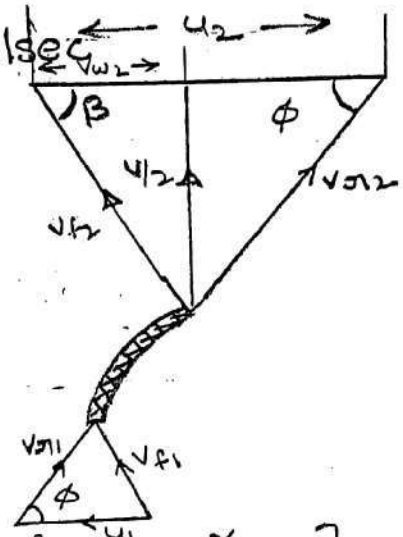
$$\therefore H_m = \left[ 25 + \frac{6.963^2}{2 \times 9.81} \right] - \left[ 5 + \frac{3.095^2}{2 \times 9.81} \right] \Rightarrow 27.47 - 5.488 = 21.98 \text{ m}$$

$$\eta_o = \frac{\rho Q H_m}{P} = \frac{9810 \times 0.035 \times 21.98}{15 \times 10^3} \times 100 \Rightarrow 50.3 \%$$

$$u_2 = \frac{\pi D_2 N}{60} = \frac{\pi \times 0.4 \times 950}{60} \Rightarrow 19.9 \text{ m/sec}$$

$$\tan \phi = \frac{V_{f2}}{u_2 - V_{w2}} \quad (\text{O.T.}) \quad \tan 45^\circ = \frac{1.8}{19.9 - V_{w2}} \Rightarrow V_{w2} = 18.1 \text{ m/sec}$$

$$\eta_{\text{mano}} = \frac{g \cdot H_m}{V_{w2} u_2} \Rightarrow \frac{9.81 \times 21.98}{18.1 \times 19.9} \times 100 \Rightarrow 59.86 \%$$



3. A centrifugal Pump impeller whose external dia and width at the outlet are 0.8 m & 0.1 m, is running at 550 r.p.m. the angle of impeller vanes at outlet is  $40^\circ$ . The Pump delivers of water at  $0.98 \text{ m}^3/\text{sec}$  under an effective head determine (i) Manometric efficiency (ii) The overall efficiency (iii) The Mechanical efficiency

Sol:- Given data:-

$D_2 = 0.8 \text{ m}$  ;  $B_2 = 0.1 \text{ m}$  ;  $N = 550 \text{ r.p.m}$  ;  $\phi = 45^\circ$  ;  $Q = 0.98 \text{ m}^3/\text{sec}$   
 $H_m = 35 \text{ m}$  ;  $P = 500 \text{ kW}$

We know that,

$$Q = \pi D_2 B_2 V_{f2}$$

$$0.98 = \pi \times 0.8 \times 0.1 \times V_{f2}$$

$\therefore$  flow velocity at outlet ( $V_{f2}$ ) = 3.9 m/sec

From outlet velocity triangle,

$$\tan \phi = \frac{V_{f2}}{u_2 - v_{w2}}$$

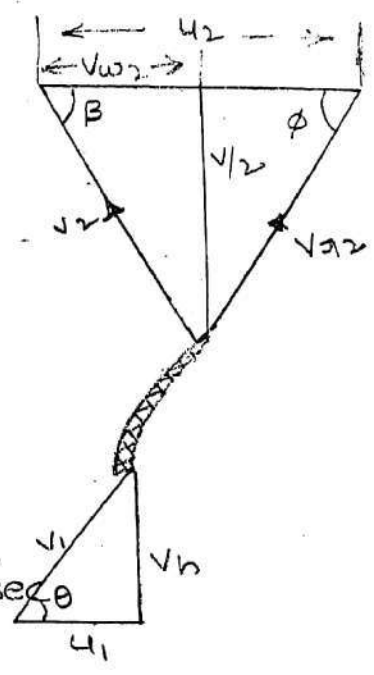
$$u_2 = \frac{\pi D_2 N}{60} \Rightarrow \frac{\pi \times 0.8 \times 550}{60} = 23.04 \text{ m/sec}$$

$$\tan 40^\circ = \frac{3.9}{23.04 - v_{w2}} \Rightarrow v_{w2} = 18.39 \text{ m/sec}$$

$$\eta_{\text{mano}} = \frac{g \cdot H_m}{v_{w2} \cdot u_2} \Rightarrow \frac{9.81 \times 35}{18.39 \times 23.04} = 81.03\%$$

$$\eta_o = \frac{W \cdot Q \cdot H_m}{P} = \frac{9810 \times 0.98 \times 35}{550 \times 10^3} = 61.18\%$$

$$\eta_o = \eta_{\text{mano}} \times \eta_m \Rightarrow \eta_m = \frac{\eta_o}{\eta_{\text{mano}}} = \frac{0.6118}{0.8103} = 75.5\%$$



(4). The outer dia. of an impeller of a centrifugal Pump is 400 mm. and outlet width 50 mm. The Pump is running at 800 r.p.m and is working against a total head of 15 m. the Vanes angle at outlet is  $40^\circ$  and  $\eta_{\text{mano}} = 75\%$ . Determine the (i) velocity of flow at outlet (ii) velocity of water leaving the vane (iii) Angle made by absolute velocity at outlet and (iv) discharge?

Sol: Given data:-

$$D_2 = 400 \text{ mm} = 0.4 \text{ m} ; B_2 = 50 \text{ mm} = 0.05 \text{ m} ; N = 800 \text{ r.p.m}$$

$$H_m = 15 \text{ m} ; \phi = 40^\circ ; \eta_{mano} = 0.75$$

$$u_2 = \frac{\pi D_2 N}{60} = \frac{\pi \times 0.4 \times 800}{60} = 16.755 \text{ m/sec}$$

$$\eta_{mano} = \frac{g \cdot H_m}{v \omega_2 \cdot u_2} \Rightarrow 0.75 = \frac{9.81 \times 15}{v \omega_2 \times 16.75}$$

(or)

$$v \omega_2 = 11.71 \text{ m/sec}$$

From outlet velocity triangle,

$$\tan \phi = \frac{v_{f2}}{u_2 - v \omega_2} \quad (\text{or}) \quad \tan 40^\circ = \frac{v_{f2}}{16.755 - 11.71}$$

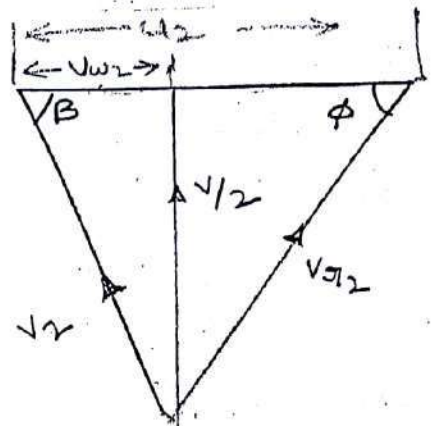
$$\Rightarrow v_{f2} = 4.23 \text{ m/sec}$$

$$v_2 = \sqrt{v \omega_2^2 + v_{f2}^2} = \sqrt{11.71^2 + 4.23^2} = 12.45 \text{ m/sec}$$

$$\sin \beta ; \frac{v_{f2}}{v_2} = \frac{4.23}{12.45} \Rightarrow \beta = 19.86^\circ$$

$$\text{discharge (Q)} = \pi D_2 B_2 v_{f2} = \pi \times 0.4 \times 0.05 \times 4.23 = 0.266 \text{ m}^3/\text{sec}$$

$$\sin \phi = \frac{v_{f2}}{v \omega_2} ; \sin 40^\circ \Rightarrow \frac{4.23}{v \omega_2} \Rightarrow v \omega_2 = 6.58 \text{ m/sec}$$



(5) A fluid is to be lifted against a head of 120 m. The pumps that run at a speed of 1200 r.p.m with a rated capacity of 300 lt/sec are available. How many pumps are required to pump the water of specific speed is 700?

Sol: Given data:-  $H_{total} = 120 \text{ m} ; N = 1200 \text{ r.p.m}$

$$Q = 300 \text{ lt/s} = 0.3 \text{ m}^3/\text{sec} ; N_s = 700 \text{ r.p.m}$$

We know that specific speed ( $N_s$ ) =  $\frac{N \sqrt{Q}}{(H_m)^{3/4}}$

$$\therefore 700 = \frac{1200 \times \sqrt{0.3}}{(H_m)^{3/4}} \Rightarrow H_m = 0.919 \text{ m}$$

$$\text{No. of stages} = \frac{H_{\text{total}}}{H_m} = \frac{120}{0.919} = 131$$

no. of Pumps  $\Rightarrow n = 131$ .

### Reciprocating Pumps

The reciprocating pump is a positive displacement as it sucks and raises the liquid by actual displacing with a piston/plunger that executes a reciprocating motion in a closely fitting cylinder. The amount of liquid pumped is equal to the volume displaced by the piston.

### CLASSIFICATION

- 1) According to the water being in contact with Piston
    - a) single acting Pump - Water contact with one side of Piston.
    - b) Double acting Pump - Water contact with both sides of the Piston.
  - 2). According to no. of cylinders:-
    - a) single cylinder      b) Double cylinder (2-throw)
    - c) Triple cylinder (3-throw)      d) Duplex-double acting (4-throw)
    - e) Quintuplex (five-throw)
- Pumps.

### Main components and working:-

The Main Parts of Reciprocating Pump are

- 1) cylinder      2) Piston      3) Suction Pipe
- 5) Delivery valve      6) Suction valve      4) Delivery Pipe
- 7) crank and connecting rod mechanism operated by a power source.

A single acting reciprocating pump has one suction pipe and delivery pipe. It is usually placed above the liquid level in the sump. When the crank rotates the piston moves backward and forward inside the cylinder.

Let the crank is initially at I.D.C and rotates in clock wise direction. As the crank rotates the piston moves towards right and a vacuum is created on the left side of the piston, The vacuum causes suction valve to open and consequently the liquid is forced ~~an~~ from the sump into the left side of the piston. When the crank is at O.D.C the suction stroke is completed and then left side of the cylinder is full of liquid.

When the crank further turns from O.D.C to I.D.C, the piston moves inward to the left & high pressure is built up in the cylinder. The delivery valve opens and liquid is forced into the delivery pipe and carried to the discharge tank. Thus one cycle is completed.

Work done:- let,

$D$  = dia. of the cylinder in meters

$A$  = Area of the piston/cylinder =  $\frac{\pi D^2}{4} \text{ m}^2$

$r$  = crank radius in meters

$L$  = length of stroke =  $2r$  in meters

$N$  = Speed of the crank in r.p.m

$h_s$  = suction head in meters

$h_d$  = delivery head in meters

Volume of water sucked in during suction

$$\text{Stroke} = AL$$

$\therefore$  Discharge of Pump ( $Q$ ) =  $AL \frac{N}{60} \text{ m}^3/\text{sec}$

Weight of water delivered ( $W$ ) =  $W \cdot Q = \frac{WALN}{60} \text{ N/sec}$

Work done = weight of water/sec  $\times$  total height lifted.

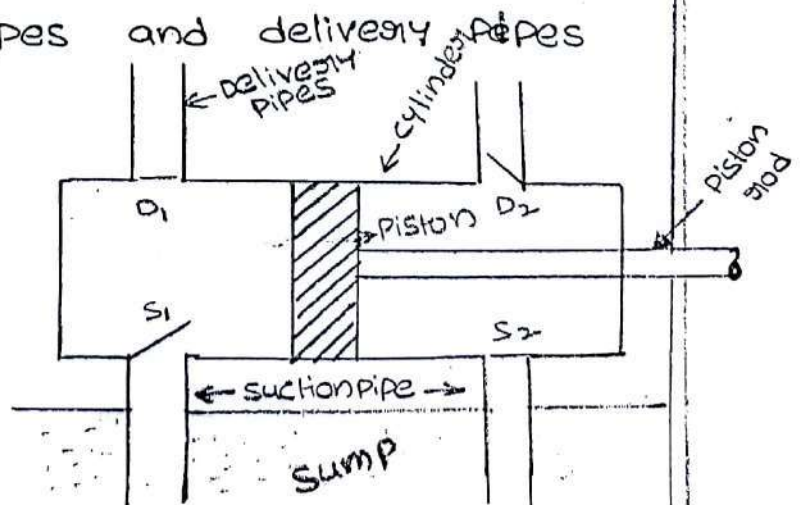
$$W(h_s + h_d) = \frac{WALN}{60} [h_s + h_d] \text{ J/sec}$$

$\therefore$  Power required to drive the pump =

$$\Rightarrow \frac{WALN (h_s + h_d)}{60 \cdot 1000} \text{ KW}$$

### Double acting Pump.

In this two suction pipes and delivery pipes are present. When the piston moves towards right side the liquid is sucked into the left side of piston and liquid at the right side of the piston is sucked into a tank. When the piston moves towards left the liquid is sucked into the right side of the piston and the liquid at the left side of the piston is delivered to the tank. Thus



for one revolution of the crank two delivery strokes are there.

The work done by double acting pump is two times the single acting pump.

$$W.D = \frac{2WALN}{60} \text{ (hsthd) } \text{ J/sec}$$

Coefficient of Discharge ( $C_d$ ):-

In reciprocating Pump, the actual discharge ( $Q_{AC}$ ) is always slightly different from the ( $Q_{th}$ ) theoretical discharge. due to \* leakage through valves, glands and piston packing. \* Imperfect operation of the valves \* Partial filling of cylinder by the liquid.

The ratio b/w the Actual discharge and theoretical discharge is known as the coefficient of discharge of pumps.

$$\therefore C_d = \frac{Q_{Actual}}{Q_{theoretical}}$$

Slip :- The difference b/w the theoretical discharge to Actual discharge is called slip of the pump.

$$\% \text{ slip} = \frac{Q_{th} - Q_{Actual}}{Q_{th}} \times 100 = \left[ 1 - \frac{Q_{AC}}{Q_{th}} \right] 100$$
$$\Rightarrow [1 - C_d] 100$$

Negative slip :- In some cases  $Q_{Actual}$  is greater than  $Q_{th}$  in such case  $C_d$  is greater than 1 and the slip will be negative. The negative slip is possible when \* There is a direct connection b/w the suction and delivery sides

\* Pumps having long <sup>suction</sup> & short delivery pipes  
\* high speed pumps

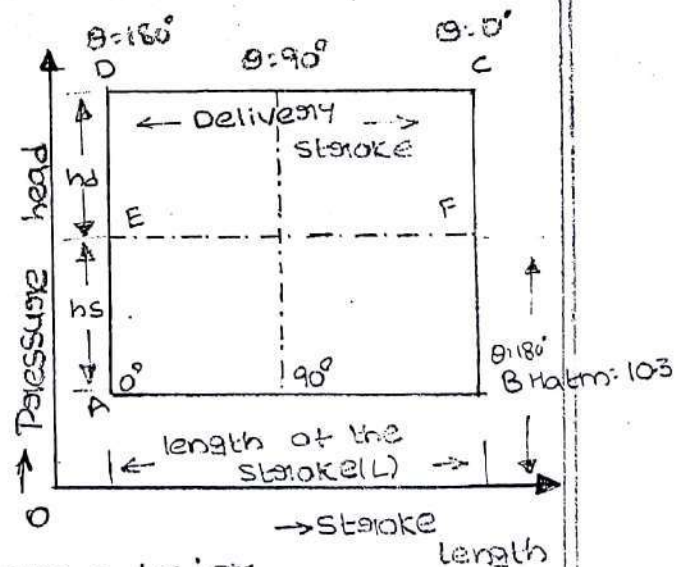
Indicator diagrams:- It is the diagram which shows the pressure head of the liquid in the cylinder corresponding to any position during the suction and delivery strokes. It is a graph between pressure head and stroke length of the piston for one complete revolution.

The indicator diagram obtained by neglecting the loss of head due to friction in the suction and delivery pipes and the effect of acceleration of piston is known as an ideal indicator diagram.

The line EF represents the atmospheric pressure head  $[H_{atm} = \frac{P_g}{\rho g}]$  equal to 10.3m of water.

During suction stroke, the pressure head in the cylinder is constant and equal to suction head ( $h_s$ ), which is below  $H_{atm}$  by a height of  $h_s$ . The pressure head during suction stroke is represented by a horizontal line AB which is below EF by a height ( $h_s$ ).

During the delivery stroke the head of pressure in the cylinder is constant and equal to delivery head ( $h_d$ ). The uniform pressure head (abs) throughout the delivery stroke is  $[h_d + H_{atm}]$ .



The work done by the pump (second) is

$$\Rightarrow \frac{WALN}{60} (h_s + h_d)$$

$$= k \cdot L(h_s + h_d) \quad \text{where } k = \frac{WAN}{60}$$

$$\propto L(h_s + h_d)$$

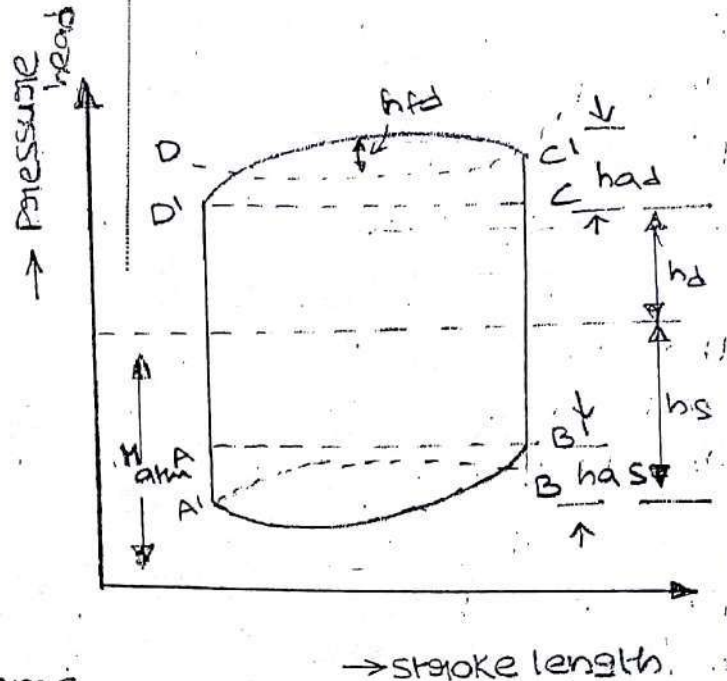
$$\propto \text{Area of indicator diagram} = AB \times BC = L(h_s + h_d)$$

$\therefore$  W.D by the pump  $\propto$  Area of indicator diagram



Including the effects of acceleration and friction the complete indicator diagram is as follows:

- $h_{as}$  - Acceleration head in suction
- $h_{ad}$  - Acceleration head in delivery
- $h_{fs}$  - friction head at suction
- $h_{fd}$  - friction head at delivery.



### Problems

1. A single acting reciprocating pump has a 15cm piston with a crank of radius 15 cm. The delivery pipe is 10cm diameter. At a speed of 60 RPM 310 lt/sec of water is lifted to a total height of 15m. Find the slip, coefficient of discharge and theoretical power in kw required to drive the pump?

Given data:  $D = 15 \text{ cm}$ ;  $r = 15 \text{ cm}$ ;  $L = 2r = 2(15) = 30 \text{ cm} = 0.3 \text{ m}$

$N = 60 \text{ R.P.M}$  ;  $Q_{AC} = 310 \text{ lt/sec} = 0.3 \text{ m}^3/\text{sec}$  ;  $h_s + h_d = 15 \text{ m}$

Area (A),  $\frac{\pi}{4} d^2 = \frac{\pi}{4} (0.15)^2 = 0.0177 \text{ m}^2$

$$Q_{th} = \frac{A L N}{60} = \frac{0.0177 \times 0.3 \times 60}{60} = 0.0053 \text{ m}^3/\text{sec}$$

coefficient of discharge ( $C_d$ ):  $\frac{Q_{th}}{Q_A} = \frac{0.31}{0.0053} = 58.49$

Slip (S) =  $Q_{th} - Q_{AC} = 0.0053 - 0.31 = -0.305 \text{ m}^3/\text{sec}$

Theoretical Power:  $\frac{W Q_{th}}{60} (h_s + h_d)$

$$\Rightarrow \frac{9810 \times 0.0053 \times 60 \times 15}{60 \times 1000} \Rightarrow 0.78 \text{ kw}$$

② A single acting reciprocating Pump having a bore of 150 mm and a stroke of 300 mm is raising water to height of 20 m above the sump level. The Pump has an actual discharge of 0.0052 m<sup>3</sup>/sec. The efficiency of Pump is 70%. If the speed is 60 r.p.m find (i) Theoretical discharge (ii) Theoretical Power (iii) Actual Power (iv) Percentage slip?

Sol: Given data:- D = 150 mm = 0.15 m ; L = 300 mm = 0.3 m  
 h<sub>s</sub> + h<sub>d</sub> = 20 m ; Q<sub>AC</sub> = 0.0052 m<sup>3</sup>/sec. η<sub>p</sub> = 70% = 0.7  
 N = 60 r.p.m ; A = π/4 D<sup>2</sup> = π/4 (0.15)<sup>2</sup> = 0.0177 m<sup>2</sup>

$$Q_{th} = \frac{ALN}{60} = \frac{0.0177 \times 0.3 \times 60}{60} = 0.0053 \text{ m}^3/\text{sec}$$

$$\begin{aligned} \text{Theoretical Power (P}_{th}\text{)} &= \frac{W Q_{th} (h_s + h_d)}{60 \times 1000} \\ &= \frac{9810 \times 0.0053 \times 60 \times 20}{60 \times 1000} \\ &= 1.04 \text{ kW} \end{aligned}$$

$$\text{Actual Power (P}_{act}\text{)} = \frac{P_{th}}{\eta_p} = \frac{1.04}{0.7} = 1.48 \text{ kW}$$

$$\begin{aligned} \text{Percentage of slip (\% slip)} &\Rightarrow \frac{Q_{th} - Q_{AC}}{Q_{th}} = \frac{0.0053 - 0.0052}{0.0053} \\ &\Rightarrow 0.00189 = 1.89\% \end{aligned}$$

③ A reciprocating Pump has a 20 cm dia cylinder and a stroke of 120 cm. The Pump delivers 5000 lit/min of water at a static head of 80 m. Friction losses in suction and delivery head pipe are 2 m and 18 m. The slip of Pump is 2% and efficiency is 90%. Determine the speed of the Pump and Power required neglecting velocity head in delivery pipe?

Sol:

Given data:  $D = 20\text{cm} = 0.2\text{m}$  ;  $L = 120\text{cm} = 1.2\text{m}$

$$Q_{AC} = 5000 \text{ lt/min} ; Q_{AC} = \frac{5}{60} = 0.0833 \text{ m}^3/\text{sec}$$

$$h_s + h_d = 80\text{m} ; h_{fs} = 2\text{m} ; h_{fd} = 18\text{m} ; \gamma_s = 2\% = 0.02$$

$$\eta_p = 90\% = 0.9$$

$$\% \text{ slip} = \frac{Q_{th} - Q_A}{Q_{th}} \quad \therefore 0.02 = 1 - \frac{Q_A}{Q_{th}}$$

$$0.02 = 1 - \frac{0.0833}{Q_{th}} \quad \Rightarrow \quad Q_{th} = 0.085 \text{ m}^3/\text{sec}$$

But,

$$Q_{th} = \frac{ALN}{60} = \frac{\frac{\pi}{4} (0.2)^2 \times 1.2 \times N}{60} = 0.085 \text{ m}^3/\text{sec}$$

$$\Rightarrow N = 135.3 \text{ r.p.m}$$

$$\text{Power required} = \frac{\gamma Q_{th} (h_s + h_d + h_{fs} + h_{fd})}{1000}$$

$$\Rightarrow \frac{9810 \times 0.085 (80 + 2 + 18)}{1000}$$

$$= 83.385 \text{ kW}$$

- (4). The dia. of the piston and stroke length of a reciprocating pump are 200mm and 300mm. The pump delivers water at a rate of 7.36 lt/sec. and the speed is 50 r.p.m. The suction and delivery heads are 3.5m and 11.5m. Determine
- coefficient of discharge
  - % slip
  - Power required?

Sol:

Given data:-  $D = 200\text{mm} = 0.2\text{m}$  ;  $L = 300\text{mm} = 0.3\text{m}$

$$Q_{AC} = 7.36 \text{ lt/sec} = 0.00736 \text{ m}^3/\text{sec} \quad N = 50 \text{ r.p.m.}$$

$$h_s = 3.5 ; h_d = 11.5 ; A = \frac{\pi}{4} D^2 = \frac{\pi}{4} (0.2)^2 = 0.0314 \text{ m}^2$$

$$Q_{th} = \frac{ALN}{60} = \frac{0.0314 \times 0.3 \times 50}{60} = 0.007854 \text{ m}^3/\text{sec}$$

$$(i) \text{ coefficient of discharge } (C_d) = \frac{Q_{AC}}{Q_{th}} = \frac{0.00736}{0.007814} = 0.9371$$

$$(ii) \% \text{ slip} = (1 - C_d) \times 100 = 6.3\%$$

$$(iii) \text{ Power required } (P) = \frac{\omega ALN}{60} (h_s + h_d)$$

$$\Rightarrow \omega Q_{th} (h_s + h_d)$$

$$\Rightarrow \frac{9810 \times 0.007854 \times (3.5 + 11.5)}{1000}$$

$$P \Rightarrow 1.156 \text{ KW}$$

(5) A single acting reciprocating pump, running at 60 r.p.m delivers  $0.53 \text{ m}^3$  of water per min. The dia. of piston is 200mm and stroke is 300mm. The suction and delivery heads are 4m & 12m. Determine (i) Theoretical discharge? (ii) coefficient of discharge (iii) percentage slip (iv) power required?

Soln Given data:  $N = 60 \text{ r.p.m}$ ;  $Q_{AC} = 0.53 \text{ m}^3/\text{min} = 0.0083 \text{ m}^3/\text{sec}$   
 $D = 200 \text{ mm} = 0.2 \text{ m}$ ;  $Q_{AC} = 0.53 \text{ m}^3/\text{sec} = 0.0083 \text{ m}^3/\text{sec}$   
 $L = 300 \text{ mm} = 0.3 \text{ m}$ ;  $h_s = 4 \text{ m}$ ;  $h_d = 12 \text{ m}$

$$Q_{th} = \frac{ALN}{60} = \frac{\pi/4 (0.2)^2 \times 0.3 \times 60}{60} = 0.00942 \text{ m}^3/\text{sec}$$

$$\text{coefficient of discharge } (C_d) = \frac{Q_{AC}}{Q_{th}} = \frac{0.0083}{0.00942} = 0.9377$$

$$\% \text{ slip} = \frac{Q_{th} - Q_A}{Q_{th}} = (1 - C_d) \times 100 \Rightarrow (1 - 0.9377) \times 100 = 6.23\%$$

$$\text{Power required } (P) = \frac{\omega Q_{th} (h_s + h_d)}{1000} \Rightarrow \frac{9810 \times 0.00942 \times (4 + 12)}{1000} \Rightarrow 1.48 \text{ KW}$$

## UNIT-4

### 1. INTRODUCTION TO HYDRAULICS AND PNEUMATICS

#### 1.1 Fluid Power and Its Scope

Fluid power is the technology that deals with the generation, control and transmission of forces and movement of mechanical element or system with the use of pressurized fluids in a confined system. Both liquids and gases are considered fluids. Fluid power system includes a hydraulic system (*hydra* meaning water in Greek) and a pneumatic system (*pneuma* meaning air in Greek). Oil hydraulic employs pressurized liquid petroleum oils and synthetic oils, and pneumatic employs compressed air that is released to the atmosphere after performing the work.

The following are the two types of hydraulic systems:

1. **Fluid transport systems:** Their sole objective is the delivery of a fluid from one location to another to accomplish some useful purpose. Examples include pumping stations for pumping water to homes, cross-country gas lines, etc.
2. **Fluid power systems:** These are designed to perform work. In fluid power systems, work is obtained by pressurized fluid acting directly on a fluid cylinder or a fluid motor. A cylinder produces a force resulting in linear motion, whereas a fluid motor produces a torque resulting in rotary motion.

#### 1.2 Classification of Fluid Power Systems

The fluid power system can be categorized as follows:

##### **Based on the control system**

- Open-loop system:** There is no feedback in the open system and performance is based on the characteristics of the individual components of the system. The open loop system is not accurate and error can be reduced by proper calibration and control.
- Closed-loop system:** This system uses feedback. The output of the system is fed back to a comparator by a measuring element. The comparator compares the actual output to the desired output and gives an error signal to the control element. The error is used to change the actual output and bring it closer to the desired value. A simple closed-loop system uses servo valves and an advanced system uses digital electronics.

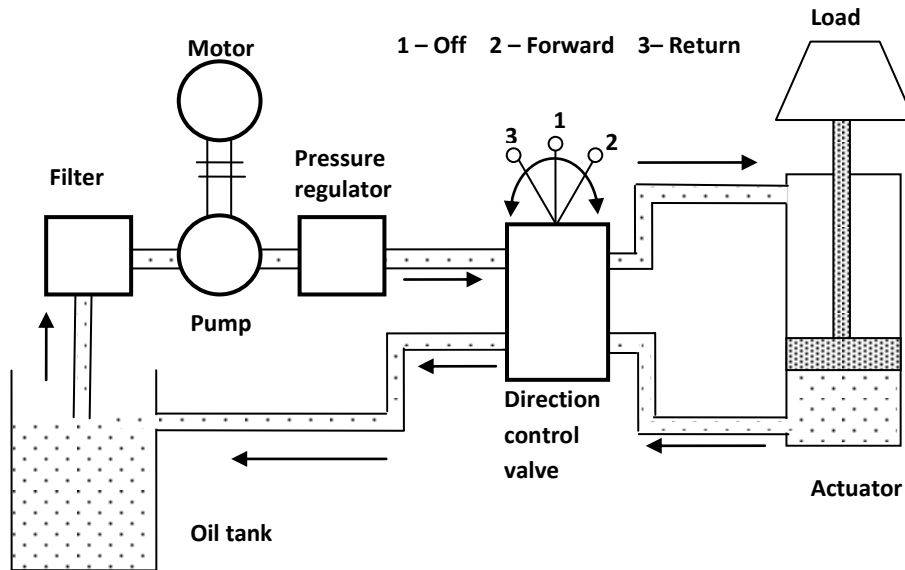
### 1.3 Advantages of a Fluid Power System

The advantages of a fluid power system are as follows:

- 1. Fluid power systems are simple, easy to operate and can be controlled accurately:** Fluid power gives flexibility to equipment without requiring a complex mechanism. Using fluid power, we can start, stop, accelerate, decelerate, reverse or position large forces/components with great accuracy using simple levers and push buttons. For example, in Earth-moving equipment, bucket carrying load can be raised or lowered by an operator using a lever. The landing gear of an aircraft can be retrieved to home position by the push button.
- 2. Multiplication and variation of forces:** Linear or rotary force can be multiplied by a fraction of a kilogram to several hundreds of tons.
- 3. Multifunction control:** A single hydraulic pump or air compressor can provide power and control for numerous machines using valve manifolds and distribution systems. The fluid power controls can be placed at a central station so that the operator has, at all times, a complete control of the entire production line, whether it be a multiple operation machine or a group of machines. Such a setup is more or less standard in the steel mill industry.
- 4. Low-speed torque:** Unlike electric motors, air or hydraulic motors can produce a large amount of torque while operating at low speeds. Some hydraulic and pneumatic motors can even maintain torque at a very slow speed without overheating.
- 5. Constant force or torque:** Fluid power systems can deliver constant torque or force regardless of speed changes.
- 6. Economical:** Not only reduction in required manpower but also the production or elimination of operator fatigue, as a production factor, is an important element in the use of fluid power.
- 7. Low weight to power ratio:** The hydraulic system has a low weight to power ratio compared to electromechanical systems. Fluid power systems are compact.
- 8. Fluid power systems can be used where safety is of vital importance:** Safety is of vital importance in air and space travel, in the production and operation of motor vehicles, in mining and manufacture of delicate products. For example, hydraulic systems are responsible for the safety of takeoff, landing and flight of aeroplanes and space craft. Rapid advances in mining and tunneling are the results of the application of modern hydraulic and pneumatic systems.

## 1.4 Basic Components of a Hydraulic System

Hydraulic systems are power-transmitting assemblies employing pressurized liquid as a fluid for transmitting energy from an energy-generating source to an energy-using point to accomplish useful work. Figure 1.1 shows a simple circuit of a hydraulic system with basic components.



**Figure 1.1** Components of a hydraulic system

Functions of the components shown in Fig. 1.1 are as follows:

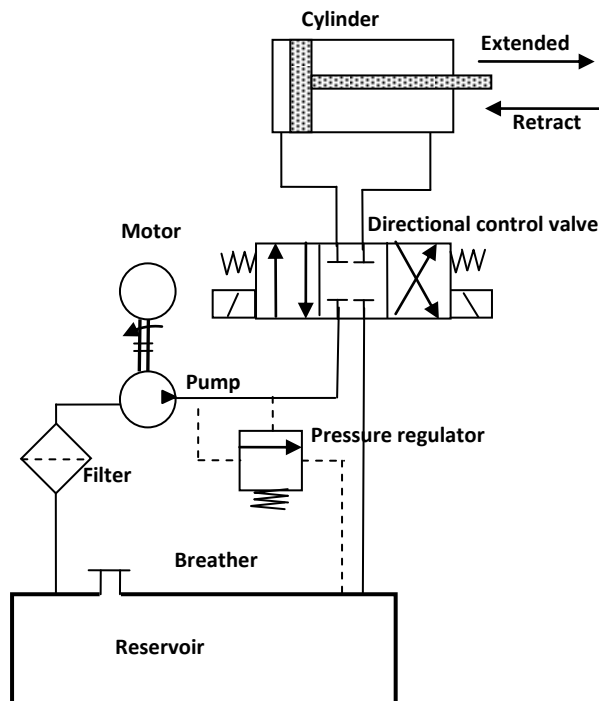
1. The hydraulic actuator is a device used to convert the fluid power into mechanical power to do useful work. The actuator may be of the linear type (e.g., hydraulic cylinder) or rotary type (e.g., hydraulic motor) to provide linear or rotary motion, respectively.
2. The hydraulic pump is used to force the fluid from the reservoir to rest of the hydraulic circuit by converting mechanical energy into hydraulic energy.
3. Valves are used to control the direction, pressure and flow rate of a fluid flowing through the circuit.
4. External power supply (motor) is required to drive the pump.
5. Reservoir is used to hold the hydraulic liquid, usually hydraulic oil.
6. Piping system carries the hydraulic oil from one place to another.
7. Filters are used to remove any foreign particles so as keep the fluid system clean and efficient, as well as avoid damage to the actuator and valves.
8. Pressure regulator regulates (i.e., maintains) the required level of pressure in the hydraulic fluid.

The piping shown in Fig. 1.1 is of closed-loop type with fluid transferred from the storage tank to one side of the piston and returned back from the other side of the piston to the tank. Fluid is drawn from the tank by a pump that produces fluid flow at the required level of pressure. If the fluid pressure exceeds the required level, then the excess fluid returns back to the reservoir and remains there until the pressure acquires the required level.

Cylinder movement is controlled by a three-position change over a control valve.

1. When the piston of the valve is changed to upper position, the pipe pressure line is connected to port A and thus the load is raised.
2. When the position of the valve is changed to lower position, the pipe pressure line is connected to port B and thus the load is lowered.
3. When the valve is at center position, it locks the fluid into the cylinder (thereby holding it in position) and dead-ends the fluid line (causing all the pump output fluid to return to tank via the pressure relief).

In industry, a machine designer conveys the design of hydraulic systems using a circuit diagram. Figure 1.2 shows the components of the hydraulic system using symbols. The working fluid, which is the hydraulic oil, is stored in a reservoir. When the electric motor is switched ON, it runs a positive displacement pump that draws hydraulic oil through a filter and delivers at high pressure. The pressurized oil passes through the regulating valve and does work on actuator. Oil from the other end of the actuator goes back to the tank via return line. To and fro motion of the cylinder is controlled using directional control valve.



**Figure 1.2** Components of a hydraulic system (shown using symbols).



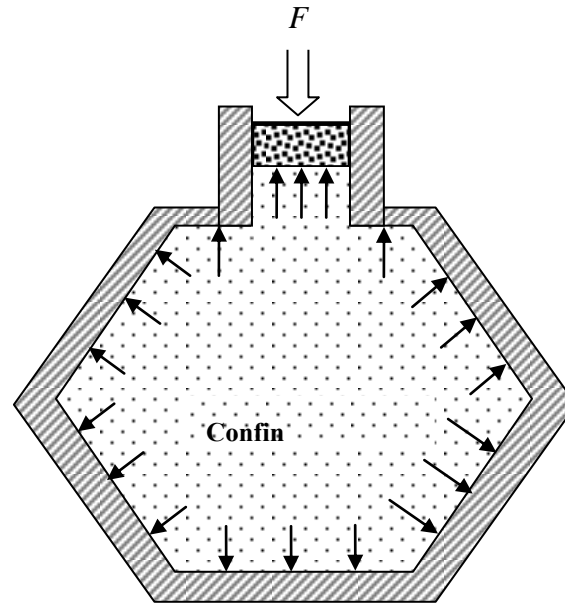
## 2. GOVERNING PRINCIPLES AND LAWS

### 2.1 Pascal's Law

Pascal's law states that the pressure exerted on a confined fluid is transmitted undiminished in all directions and acts with equal force on equal areas and at right angles to the containing surfaces. In Fig. 1.1, a force is being applied to a piston, which in turn exerts a pressure on the confined fluid. The pressure is equal everywhere and acts at right angles to the containing surfaces. Pressure is defined as the force acting per unit area and is expressed as

$$\text{Pressure} = \frac{F}{A}$$

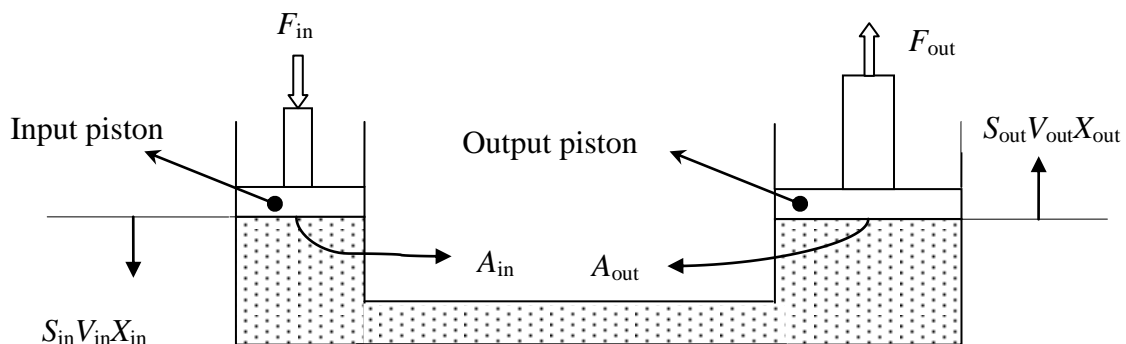
where  $F$  is the force acting on the piston,  $A$  is the area of the piston and  $p$  is the pressure on the fluid.



**Figure 1.1** Illustration of Pascal's law

### 400 Multiplication of Force

The most useful feature of fluid power is the ease with which it is able to multiply force. This is accomplished by using an output piston that is larger than the input piston. Such a system is shown in Fig. 1.2.



**Figure 1.2** Multiplication of force

This system consists of an input cylinder on the left and an output cylinder on the right that is filled with oil. When the input force is  $F_{in}$  on the input piston, the pressure in the system is given by

$$P = \frac{F_{out}}{A_{out}}$$

$$\Rightarrow F_{out} = PA_{out} = \frac{F_{in}}{A_{in}} A_{out} = \frac{A_{out}}{A_{in}} F_{in}$$

## 2.2 The Energy Equation

The Bernoulli equation discussed above can be modified to account for fractional losses ( $H_L$ ) between stations 1 and 2. Here  $H_L$  represents the energy loss due to friction of 1 kg of fluid moving from station 1 to station 2. As discussed earlier,  $H_p$  represents the energy head put into the flow by the pump. If there exists a hydraulic motor or turbine between stations 1 and 2, then it removes energy from the fluid. If  $H_m$  (motor head) represents the energy per kg of fluid removed by a hydraulic motor, the modified Bernoulli equation (also called the energy equation) is stated as follows for a fluid flowing in a pipeline from station 1 to station 2: The total energy possessed by 1 kg of fluid at station 1 plus the energy added to it by a pump minus the energy removed from it by a hydraulic motor minus the energy it loses due to friction equals the total energy possessed by 1 kg of fluid when it arrives at station 2. The energy equation is as follows, where each term represents a head and thus has the unit of length:

$$z_1 + \frac{p_1}{\gamma} + \frac{v_1^2}{2g} + H_p - H_m - H_L = z_2 + \frac{p_2}{\gamma} + \frac{v_2^2}{2g}$$

## 2.3 Elements of Hydraulic Systems and the Corresponding Bernoulli's Equation

The main elements of hydraulic systems are pump, motor, pipes, valves and fittings. Let us write the energy flow from point 1 to point 2 as shown in Fig. 1.22. After the fluid leaves point 1, it enters the pump where energy is added. A prime mover, such as an electric motor, drives the pump and the impeller of the pump transfers the energy to the fluid. Then the fluid flows through a piping system composed of a valve, elbows and the lengths of pipe in which energy is dissipated from the fluid and is lost. Before reaching point 2, the fluid flows through a fluid motor that removes some of the energy to drive an external device. The general energy equation accounts for all these energies.

In a particular problem, it is possible that not all of the terms in the general energy equation are required. For example, if there is no mechanical device between the sections of interest, the terms  $H_p$  and  $H_m$  will be zero and can be left out of the equation. If energy losses are so small that they can be neglected, the term  $H_L$  can be left out. If both these conditions exist, it can be seen that the energy equation reduces to Bernoulli's equation.

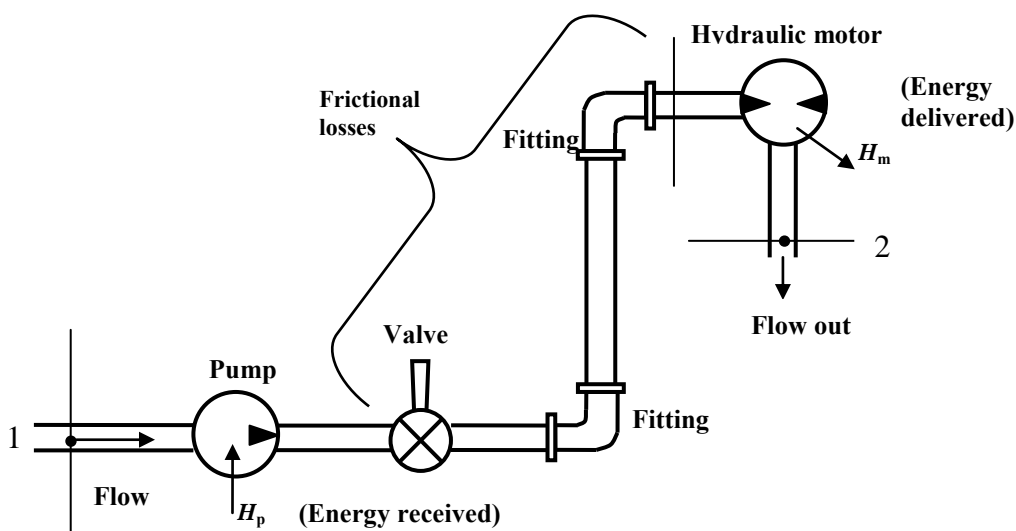
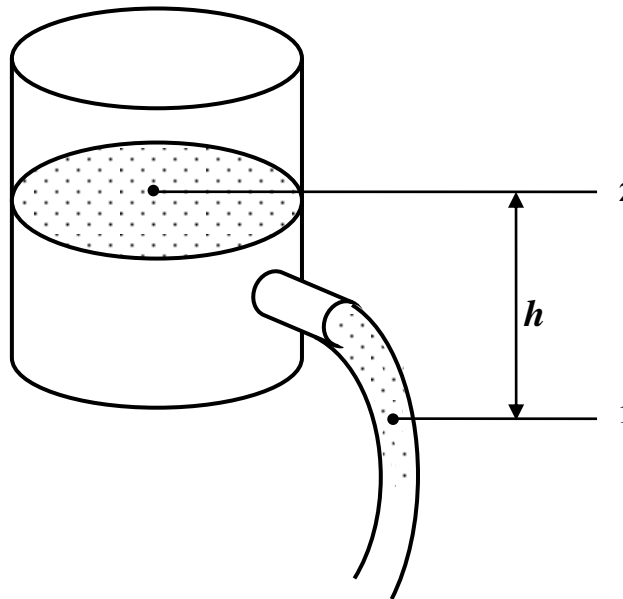


Figure 1.22 Elements of a hydraulic system

## 2.4 Torricelli's Theorem

Torricelli's theorem is Bernoulli's equation with certain assumptions made. Torricelli's theorem states that the velocity of the water jet of liquid is directly proportional to the square root of the head of the liquid producing it. This deals with the setup where there is a large tank with a narrow opening allowing the liquid to flow out (Fig. 1.30). Both the tank and the narrow opening (nozzle) are open to the atmosphere:

$$z_1 + \frac{p_1}{\gamma} + \frac{v_1^2}{2g} + H_p - H_m - H_L = z_2 + \frac{p_2}{\gamma} + \frac{v_2^2}{2g}$$



**Figure 1.30** Tank with a narrow opening (nozzle)

In this setup, certain assumptions are made:

1. Pressure is the same because the tank and the nozzle are open to the atmosphere, that is,  
 $p_1 = p_2$ .
2. Also, let  $z_2 - z_1 = h$ .
3. The fluid velocity of the tank (water level) is very much slower than the fluid velocity of the nozzle as the area of the liquid surface is much larger than that of the cross section of nozzle, that is,  $v_2 \ll v_1$ .
4. There is no pump or motor, that is,  $H_p = H_m = 0$ .
5. There are no frictional losses, that is,  $H_L = 0$ .

Keeping all these assumptions in mind, Bernoulli's equation gets reduced to

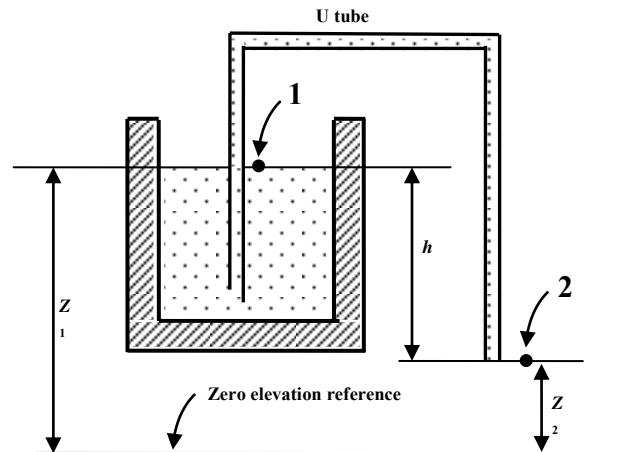
$$v_2 = \sqrt{2gh}$$

where  $v_2$  is the jet velocity (m/s),  $g$  is the acceleration due to gravity ( $m/s^2$ ) and  $h$  is the pressure head (m). Now if we do not consider an ideal fluid, then the friction head will be present ( $H_L$ ). In that case

$$v_2 = \sqrt{2g(h - H_L)}$$

This shows that the velocity of jet decreases if the friction losses are taken into account.

## 2.5 Siphon



**Figure 1.31** The siphon principle

A siphon is a familiar hydraulic device (Fig. 1.31). It is commonly used to cause a liquid to flow from one container in an upward direction over an obstacle to a second lower container in a downward direction. As shown in Fig. 1.31, a siphon consists of a U-tube with one end submerged below the level of the liquid surface, and the free end lying below it on the outside of the container. For the fluid to flow out of the free end, two conditions must be met:

1. The elevation of the free end must be lower than the elevation of the liquid surface inside the container.
2. The fluid must initially be forced to flow up from the container into the center portion of the U-tube. This is normally done by temporarily providing a suction pressure at the free end of the siphon. For example, when siphoning gasoline from an automobile gas tank, a person can develop this suction by momentarily sucking the free end of the hose. This allows atmospheric pressure in the tank to push the gasoline up the U-tube hose, as required. For continuous flow operation, the free end of the U-tube hose must lie below the gasoline level in the tank.

We can analyze the flow through a siphon by applying the energy equation between points 1 and 2 as shown in Fig. 1.31:

$$\frac{p_1}{\gamma} + \frac{v_1^2}{2g} + z_1 + H_p - H_m - H_L = \frac{p_2}{\gamma} + \frac{v_2^2}{2g} + z_2$$

The following conditions apply for a siphon:

1.  $p_1 = p_2 =$  atmospheric pressure.
2. The area of the surface of the liquid in the container is large so that the velocity  $v_1$  equals essentially 0.

## 3. DIRECTIONAL CONTROL VALVES

### 3.1 Introduction

In fluid power, controlling elements are called valves.

There are three types of valves:

1. **Directional control valves (DCVs):** They determine the path through which a fluid transverses a given circuit.
2. **Pressure control valves:** They protect the system against overpressure, which may occur due to a sudden surge as valves open or close or due to an increase in fluid demand.
3. **Flow control valves:** Shock absorbers are hydraulic devices designed to smooth out pressure surges and to dampen hydraulic shock.

### 3.2 Directional Control Valves

A valve is a device that receives an external signal (mechanical, fluid pilot signal, electrical or electronics) to release, stop or redirect the fluid that flows through it. The function of a DCV is to control the direction of fluid flow in any hydraulic system. A DCV does this by changing the position of internal movable parts.

To be more specific, a DCV is mainly required for the following purposes:

- To start, stop, accelerate, decelerate and change the direction of motion of a hydraulic actuator.
- To permit the free flow from the pump to the reservoir at low pressure when the pump's delivery is not needed into the system.
- To vent the relief valve by either electrical or mechanical control.
- To isolate certain branch of a circuit.

#### 3.2.1 Classification of DCVs based Fluid Path

- Check valves.
- Shuttle valves.
- Two-way valves.
- Three-way valves.
- Four-way valves.

#### 3.2.2 Classification of DCVs based on Design Characteristics

- An internal valve mechanism that directs the flow of fluid. Such a mechanism can either be a poppet, a ball, a sliding spool, a rotary plug or a rotary disk.
- Number of switching positions (usually 2 or 3).
- Number of connecting ports or ways.
- Method of valve actuation .

#### 3.2.3 Classification of DCVs based on the Control Method

- **Direct controlled DCV:**A valve is actuated directly on the valve spool.
- **Indirect controlled DCV:**A valve is actuated by a pilot line or using a solenoid or by the combination of electrohydraulic and electro-pneumatic means.

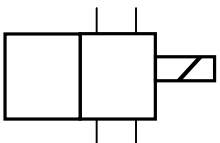
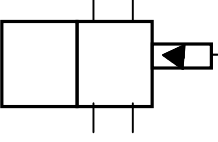
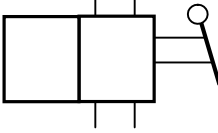
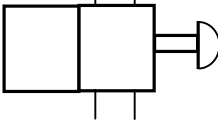
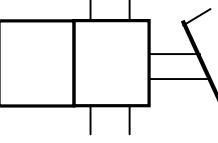
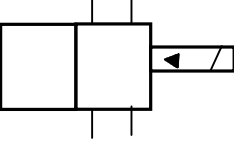
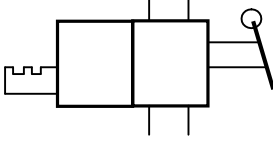
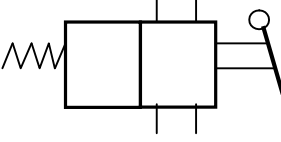
#### 3.2.4 Classification of DCVs based on the Construction of Internal Moving Parts

- **Rotary spool type:** In this type, the spool is rotated to change the direction of fluid.
- **Sliding spool type:** This consists of a specially shaped spool and a means of positioning the spool.

### 3.3 Actuating Devices

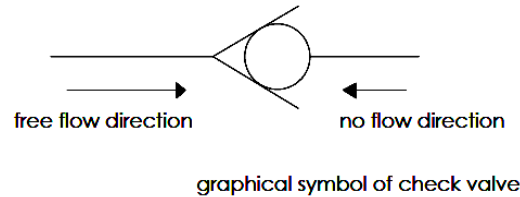
Direction control valves may be actuated by a variety of methods. Actuation is the method of moving the valve element from one position to another. There are four basic methods of actuation: Manual, mechanical, solenoid-operated and pilot-operated. Several combinations of actuation are possible using these four basic methods. Graphical symbols of such combinations are given in Table 1.3.

- **Manually operated:** In manually operated DCVs, the spool is shifted manually by moving a handle pushing a button or stepping on a foot pedal.
- **Mechanically operated:** The spool is shifted by mechanical linkages such as cam and rollers.
- **Solenoid operated:** When an electric coil or a solenoid is energized, it creates a magnetic force that pulls the armature into the coil. This causes the armature to push the spool of the valve.
- **Pilot operated:** A DCV can also be shifted by applying a pilot signal (either hydraulic or pneumatic) against a piston at either end of the valve spool.

	Solenoid operated
	Pilot operated
	Manual operated
	Push button
	Foot operated
	Pilot-operated solenoid
	Two-position detent
	Spring return

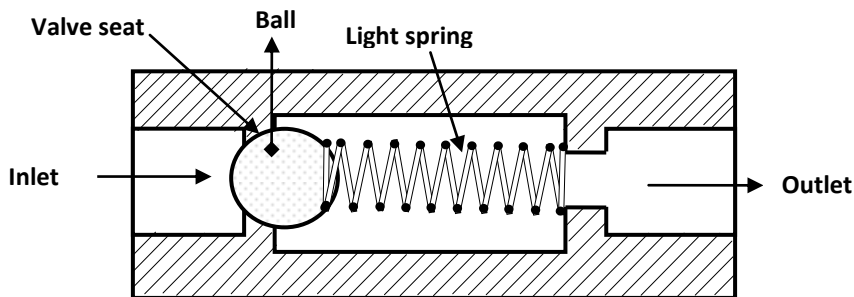
### 3.4 Check Valve

The simplest DCV is a check valve. A check valve allows flow in one direction, but blocks the flow in the opposite direction. It is a two-way valve because it contains two ports. Figure 1.1 shows the graphical symbol of a check valve along with its no-flow and free-flow directions.



**Figure 1.1** Graphical symbol of a check valve.

In Fig. 1.2, a light spring holds the ball against the valve seat. Flow coming into the inlet pushes the ball off the seat against the light force of the spring and continues to the outlet. A very low pressure is required to hold the valve open in this direction. If the flow tries to enter from the opposite direction, the pressure pushes the ball against the seat and the flow cannot pass through.

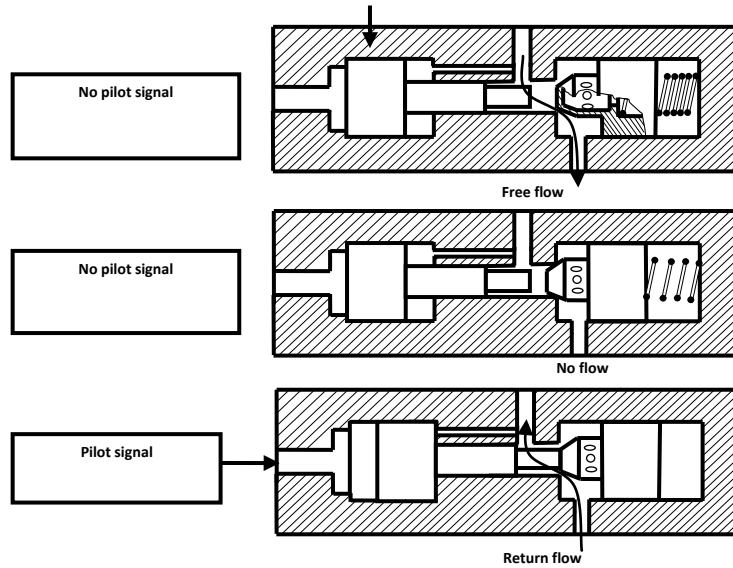


**Figure 1.2** Ball-type check valve.

### 3.5 Pilot-Operated check Valve

A pilot-operated valve along with its symbol is shown in Fig. 1.4. This type of check valve always permits free flow in one direction but permits flow in the normally blocked opposite direction only if the pilot pressure is applied at the pilot pressure point of the valve. The check valve poppet has the pilot piston attached to the threaded poppet stem by a nut.

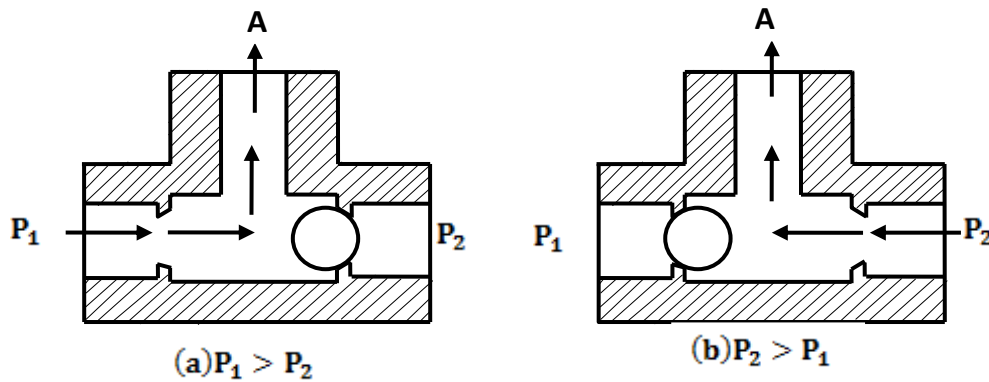
The light spring holds the poppet seated in a no-flow condition by pushing against the pilot piston. The purpose of the separate drain port is to prevent oil from creating a pressure build-up at the bottom of the piston. The dashed line in the graphical symbol represents the pilot pressure line connected to the pilot pressure port of the valve. Pilot check valves are used for locking hydraulic cylinders in position.



**Figure 1.4 Pilot-operated check valve**

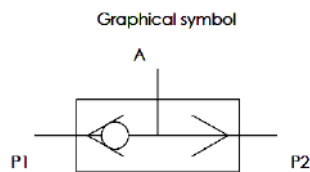
### 3.6 Shuttle Valve

A shuttle valve allows two alternate flow sources to be connected in a one-branch circuit. The valve has two inlets  $P_1$  and  $P_2$  and one outlet A. Outlet A receives flow from an inlet that is at a higher pressure. Figure 1.5 shows the operation of a shuttle valve. If the pressure at  $P_1$  is greater than that at  $P_2$ , the ball slides to the right and allows  $P_1$  to send flow to outlet A. If the pressure at  $P_2$  is greater than that at  $P_1$ , the ball slides to the left and  $P_2$  supplies flow to outlet A.



**Figure 1.5 Shuttle valve: (a) Flow from left to outlet and (b) flow from right to outlet in Fig. 1.5.**

One application for a shuttle valve is to have a primary pump inlet  $P_1$  and a secondary pump inlet  $P_2$  connected to the system outlet A. The secondary pump acts as a backup, supplying flow to the system if the primary pump loses pressure. A shuttle valve is called an “OR” valve because receiving a pressure input signal from either  $P_1$  or  $P_2$  causes a pressure output signal to be sent to A. Graphical symbol of shuttle valve is shown in Fig. 1.6.



**Fig 1.6 symbol of Shuttle valve**



### 3.7.1 2/2-Way DCV (Normally Closed)

Figure 1.7 shows a two-way two-position (normally closed) of spool type. A spool valve consists of a cylindrical spool that slides back and forth inside the valve body to connect or block flow between the ports. The larger diameter portion of the spool, the spool land blocks flow by covering the port. This particular valve has two ports labeled P and A. P is connected to the pump line and A is connected to the outlet to the system. Figure 1.7(a) shows the valve in its normal state and its corresponding symbol. The valve is held in this position by the force of the spring. In this position, the flow from the inlet port P is blocked from going to the outlet port A. Figure 1.7(b) shows the valve in its actuated state and its corresponding symbol. The valve is shifted into this position by applying a force to overcome the resistance of the spring. In this position, the flow is allowed to go to the outlet port.

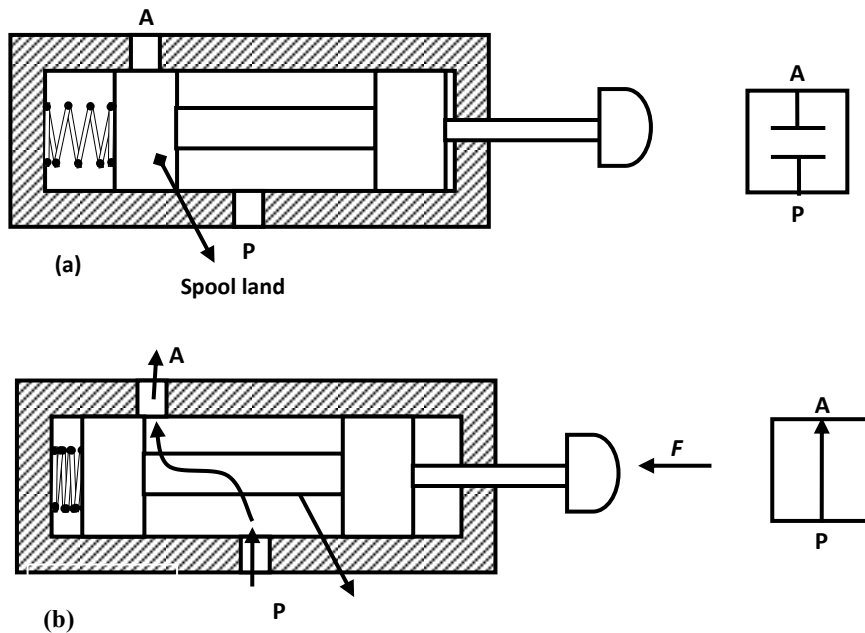
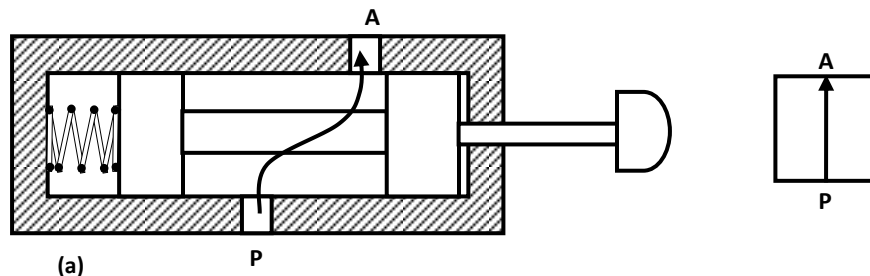
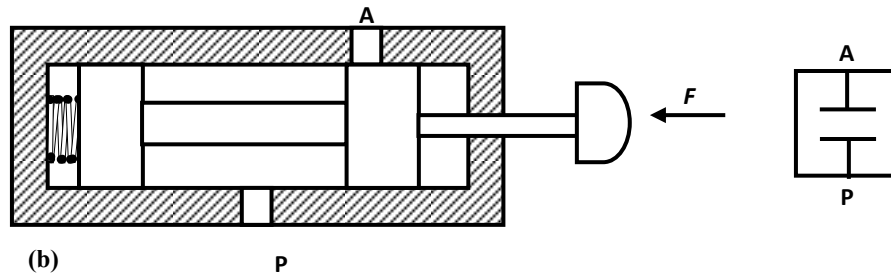


Figure 1.7 Two-way–two-position normally closed DCV. (a) Ports A and P are not connected when force is not applied (valve unactuated). (b) Ports A and P are connected when force is applied (valve actuated).

### 3.7.2 2/2-Way DCV (Normally Opened)

Figure 1.8 shows a two-way, two-position normally open DCV. The spring holds the valve in a position in which ports P and A are connected as shown in Fig 1.8.(a). When the valve is actuated, the flow is blocked from going to A as shown in Fig 1.8.(b). The complete graphic symbol for the given DCV is shown in Fig. 1.8(c).



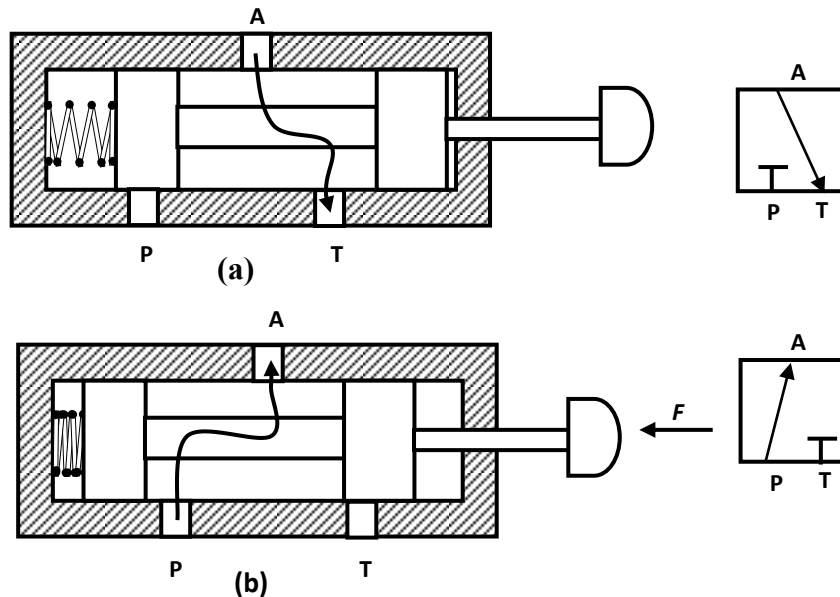


**Figure 1.8** 2/2 DCV normally opened. (a) Ports A and P are connected when force is not applied (valve unactuated). (b) Ports A and P are not connected when force is applied (valve actuated)

### 3.8 Three-Way Direction Control

#### 3.8.1 3/2-Way DCV (Normally Closed)

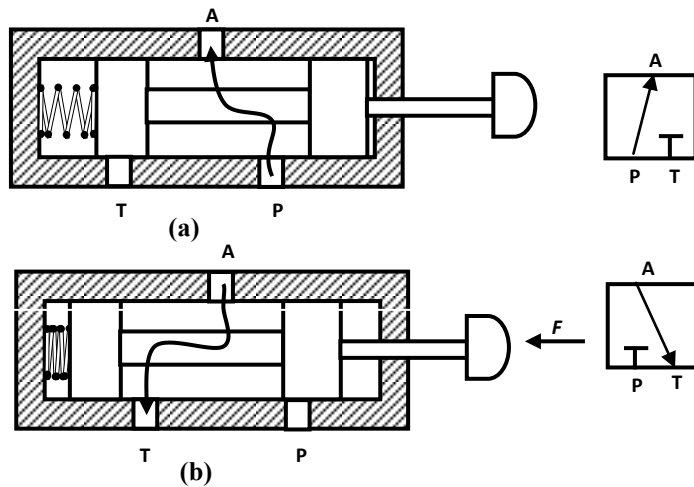
Three-way valves either block or allow flow from an inlet to an outlet. They also allow the outlet to flow back to the tank when the pump is blocked, while a two-way valve does not. A three-way valve has three ports, namely, a pressure inlet (P), an outlet to the system (A) and a return to the tank (T). Figure 1.10 shows the operation of a 3/2-way valve normally closed. In its normal position, the valve is held in position by a spring as shown in Fig. 1.10(a). In the normal position, the pressure port P is blocked and outlet A is connected to the tank. In the actuated position shown in Fig. 1.10(b), the pressure port is connected to the tank and the tank port is blocked.



**Figure 1.10** 3/2-way DCV (normally closed). (a) Ports A and T are connected when force is not applied (valve unactuated). (b) Ports A and P are connected when force is applied (valve actuated).

#### 3.8.2 3/2-Way DCV (Normally Open)

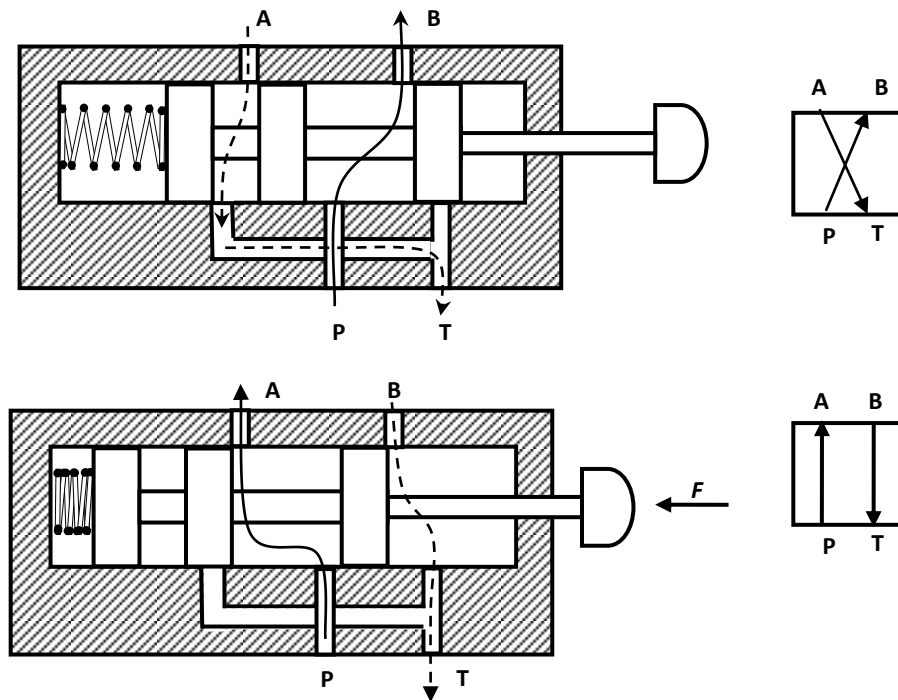
Figure 1.11 shows a three-way two-position DCV (normally open) with push button actuation and spring return. In the normal position, shown in Fig. 1.11(a), the valve sends pressure to the outlet and blocks the tank port in the normal position. In the actuated position, the pressure port is blocked and the outlet is vented to the tank.



(a) **Figure 1.11** 3/2-way DCV (normally opened). (a) Ports A and P are connected when force is not applied (valve unactuated). (b) Ports A and T are connected when force is applied (valve actuated).

### 3.9 Four-Way Direction Control Valves

Four-way DCVs are capable of controlling double-acting cylinders and bidirectional motors. Figure 1.16 shows the operation of a typical 4/2 DCV. A four-way has four ports labeled P, T, A and B. P is the pressure inlet and T is the return to the tank; A and B are outlets to the system. In the normal position, pump flow is sent to outlet B. Outlet A is connected to the tank. In the actuated position, the pump flow is sent to port A and port B connected to tank T. In four-way DCVs, two flows of the fluids are controlled at the same time, while two-way and three-way DCVs control only one flow at a time. Figure 1.16 (c) shows the complete graphic symbol for a four-way two-piston DCV.



**Figure 1.16** Four-way DCV.

### 3.10 Solenoid-Actuated Valve

A spool-type DCV can be actuated using a solenoid as shown in Figure. 1.24. When the electric coil (solenoid) is energized, it creates a magnetic force that pulls the armature into the coil. This causes the armature to push on the push pin to move the spool of the valve.

Like mechanical or pilot actuators, solenoids work against a push pin, which in turn actuates a spool. There are two types of solenoid designs used to dissipate the heat developed in electric current flowing in the coil. The first type dissipates the heat into surrounding air and is referred to as an “air gap solenoid.” In the second type “wet pin solenoid,” the push pin contains an internal passage way that allows the tank port oil to communicate between the housing of the valve and the housing of the solenoid. Wet pin solenoids do a better job in dissipating heat because the cool oil represents a good heat sink to absorb heat from the solenoid. As the oil circulates, the heat is carried into the hydraulic system where it can be easily dealt with.

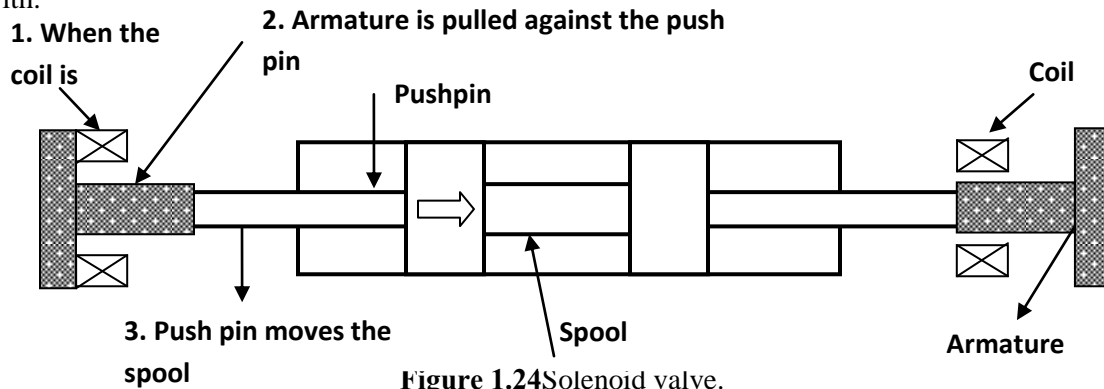


Figure 1.24 Solenoid valve.

Table 1.3 Comparison between AC and DC solenoids

Parameter	DC Solenoid	AC Solenoid
Switching time	50–60 ms	20 ms
Service life expectations	20–50 million cycles	10–20 million cycles
Max. switching frequency	Up to 4 cycles/s	Up to 2 cycles/s
Continuous operation	Unlimited	15–20 min for dry solenoids. 60–80 min for wet solenoids
Relative cost	1	1.2
Occurrence rate	10	2

### 3.11 Pilot-Operated Direction Control Valves

Pilot-operated DCVs are used in a hydraulic system operating at a high pressure. Due to the high pressure of the system, the force required to actuate the DCV is high. In such systems, operation at a high pressure uses a small DCV that is actuated by either a solenoid or manually. This pilot DCV in turn uses the pressure of the system to actuate the main DCV as shown in Figure. 1.25.

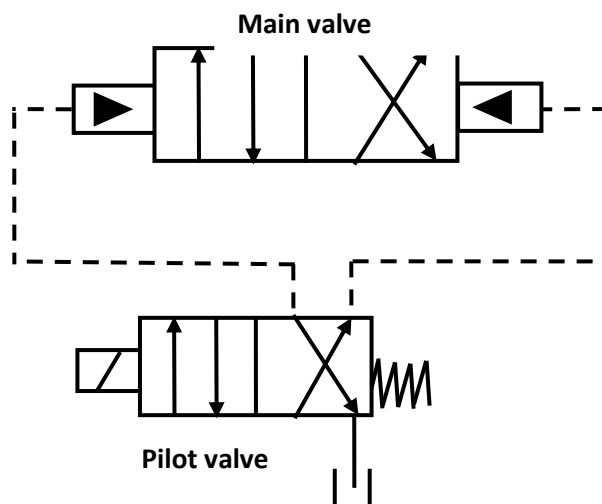


Figure 1.25 Pilot-operated DCVs.

## 4.PRESSURE-CONTROL VALVES

### 1.1 Introduction

Pressure-control valves are used in hydraulic systems to control actuator force (force = pressure  $\times$  area) and to determine and select pressure levels at which certain machine operations must occur.

"

Pressure controls are mainly used to perform the following system functions

- Limiting maximum system pressure at a safe level.
- Regulating/reducing pressure in certain portions of the circuit.
- Unloading system pressure.
- Assisting sequential operation of actuators in a circuit with pressure control.
- Any other pressure-related function by virtue of pressure control.
- Reducing or stepping down pressure levels from the main circuit to a lower pressure in a sub-circuit.

Pressure-control valves are often difficult to identify mainly because of the many descriptive names given to them. The function of the valve in the circuit usually becomes the basis for its name. The valves used for accomplishing the above-mentioned system functions are therefore given the following names:

- Pressure-relief valve.
- Pressure-reducing valve.
- Unloading valve
- Counterbalance valve.
- Pressure-sequence valve.
- Brake valve.

### 1.2 Pressure-Relief Valves

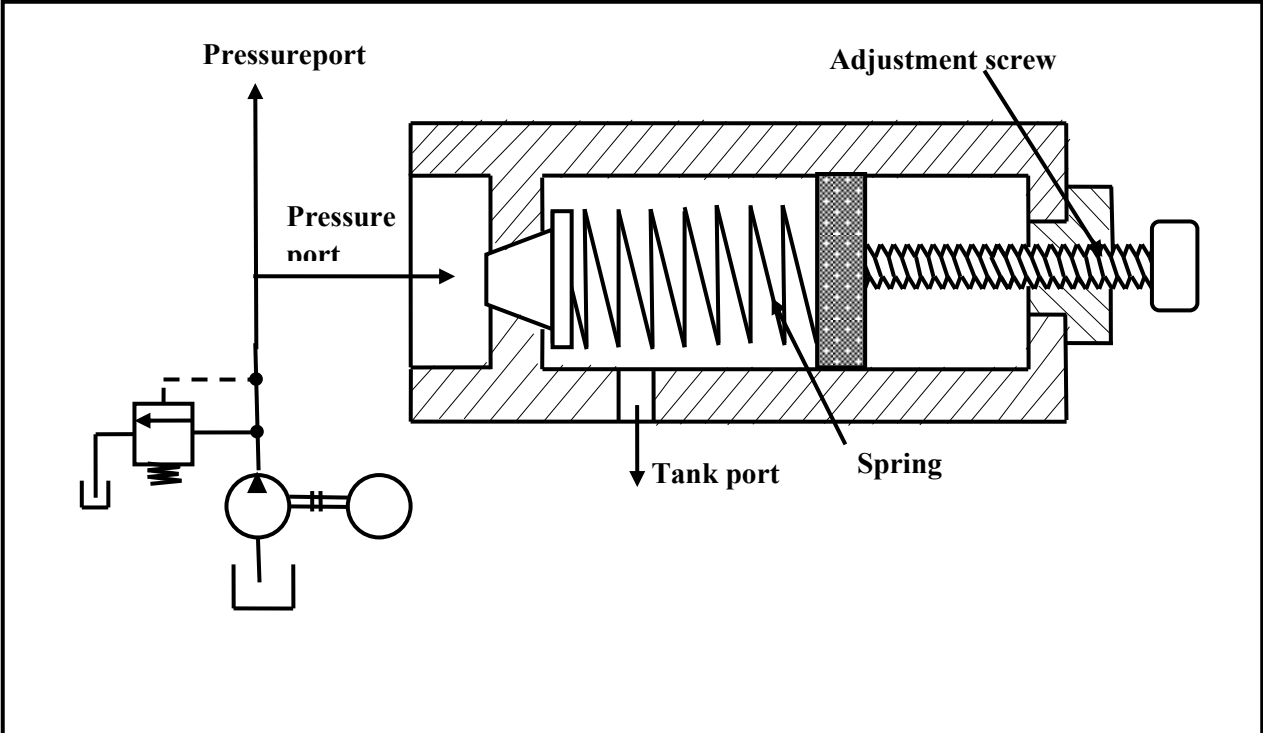
Pressure-relief valves limit the maximum pressure in a hydraulic circuit by providing an alternate path for fluid flow when the pressure reaches a preset level. All fixed-volume pump circuits require a relief valve to protect the system from excess pressure. Fixed-volume pumps must move fluid when they turn. When a pump unloads through an open-center circuit or actuators are in motion, fluid movement is not a problem. A relief valve is essential when the actuators stall with the directional valve still in shifted position.

There are two different designs of relief valves in use: direct-acting and pilot-operated. Both types have advantages and work better in certain applications.

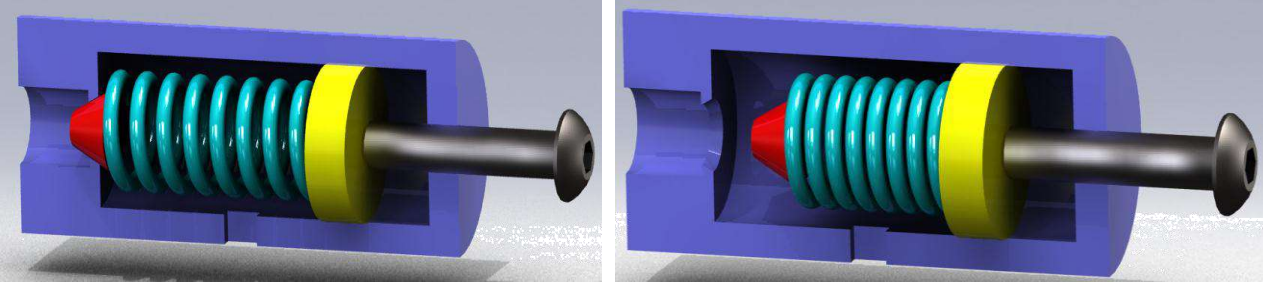
#### 1.2.1 Simple Pressure-Relief Valve

The most widely used type of pressure control valve is the pressure-relief valve because it is found in practically every hydraulic system. Schematic diagram of simple relief valve is shown in Fig. 1.1 and three-dimensional view is shown in Fig. 1.2. It is normally a closed valve whose function is to limit the pressure to a specified maximum value by diverting pump flow back to the tank. A poppet is held seated inside the valve by a heavy spring. When the system pressure reaches a high enough value, the poppet is forced off its seat. This permits flow through the outlet to the tank as long as this high pressure level is maintained. Note the external adjusting screw, which varies spring force and, thus, the pressure at which the valve begins to open (cracking pressure)(Fig. 1.3).

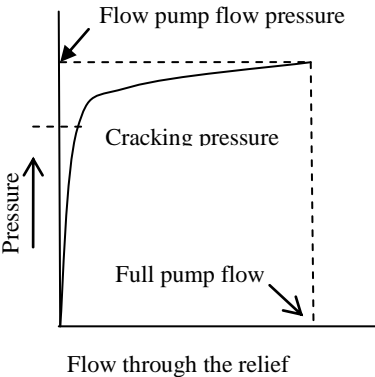
It should be noted that the poppet must open sufficiently to allow full pump flow. The pressure that exists at full pump flow can be substantially greater than cracking pressure. The pressure at full pump flow is the pressure level that is specified when referring to the pressure setting of the valve. It is the maximum pressure level permitted by the relief valve.



**Figure1.1** Simple pressure-relief valve.



**Figure 1.2**Three-dimensional view of simple pressure-relief valve.

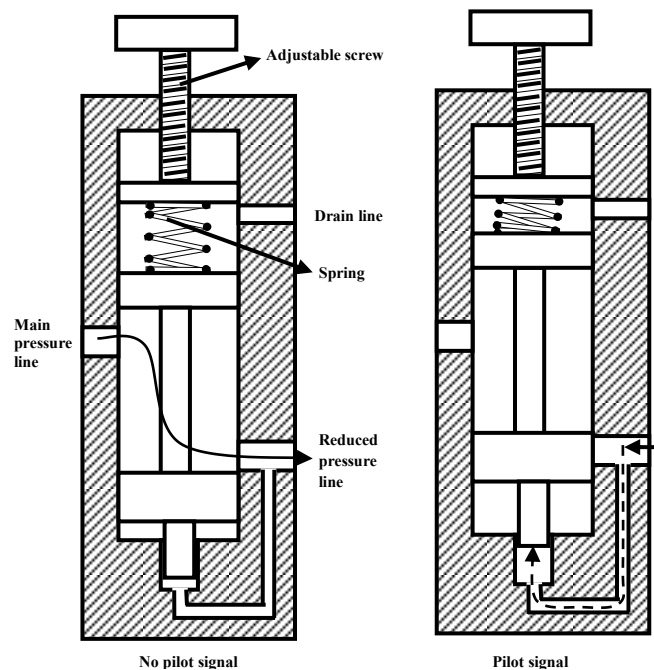


**Figure 1.3**Characteristics of a relief valve.

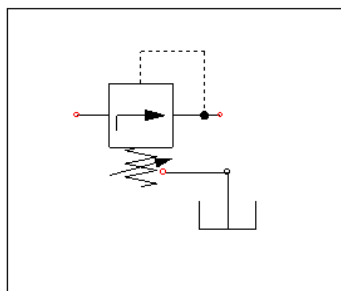
### 1.3 Pressure-Reducing Valve

The second type of valve is a pressure-reducing valve. This type of valve (which is normally open) is used to maintain reduced pressures in specified locations of hydraulic systems. It is actuated by downstream pressure and tends to close as this pressure reaches the valve setting. Schematic diagram of pressure reducing valve is shown in Fig. 1.6, symbolic representation is shown in Fig. 1.7 and three-dimensional view is shown in Fig. 1.8.

A pressure-reducing valve uses a spring-loaded spool to control the downstream pressure. If the downstream pressure is below the valve setting, the fluid flows freely from the inlet to the outlet. Note that there is an internal passageway from the outlet which transmits outlet pressure to the spool end opposite the spring. When the outlet (downstream) pressure increases to the valve setting, the spool moves to the right to partially block the outlet port. Just enough flow is passed to the outlet to maintain its preset pressure level. If the valve closes completely, leakage past the spool causes downstream pressure to build up above the valve setting. This is prevented from occurring because a continuous bleed to the tank is permitted via a separate drain line to the tank.

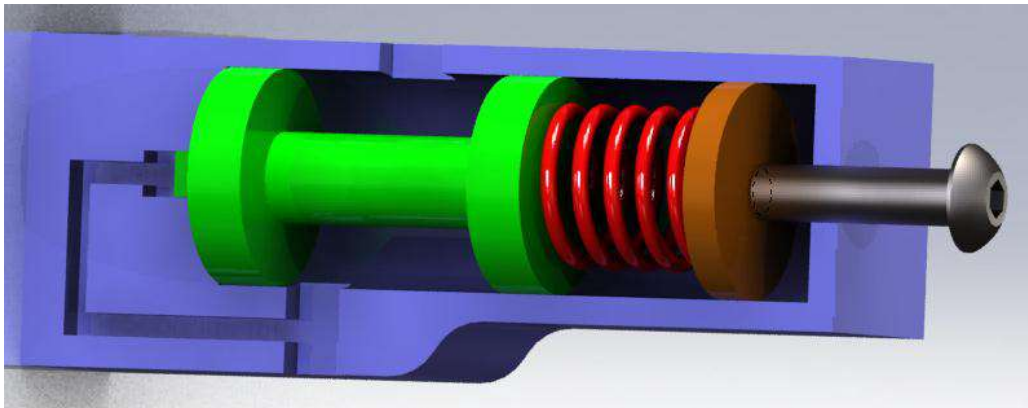


**Figure 1.6** Pressure-reducing valve.



**Figure 1.7** Symbolic representation of a pressure-reducing valve.

A reducing valve is normally open. It reads the downstream pressure. It has an external drain. This is represented by a line connected from the valve drain port to the tank. The symbol shows that the spring cavity has a drain to the tank.



**Figure 1.8** Three-dimensional view of a pressure-reducing valve.

#### **1.4 Unloading Valves**

Unloading valves are pressure-control devices that are used to dump excess fluid to the tank at little or no pressure. A common application is in high-low pump circuits where two pumps move an actuator at a high speed and low pressure. The circuit then shifts to a single pump providing a high pressure to perform work.

Another application is sending excess flow from the cap end of an oversize-rod cylinder to the tank as the cylinder retracts. This makes it possible to use a smaller, less-expensive directional control valve while keeping pressure drop low.

##### **1.4.1 Direct-Acting Unloading Valve**

A direct-acting unloading valve consists of a spool held in the closed position by a spring. The spool blocks flow from the inlet to the tank port under normal conditions. When a high-pressure fluid from the pump enters at the external-pilot port, it exerts force against the pilot piston. (The small-diameter pilot piston allows the use of a long, low-force spring.) When the system pressure increases to the spring setting, the fluid bypasses to the tank (as a relief valve would function). When the pressure goes above the spring setting, the spool opens fully to dump the excess fluid to the tank at little or no pressure.

##### **1.4.2 Pilot-Operated Unloading Valve**

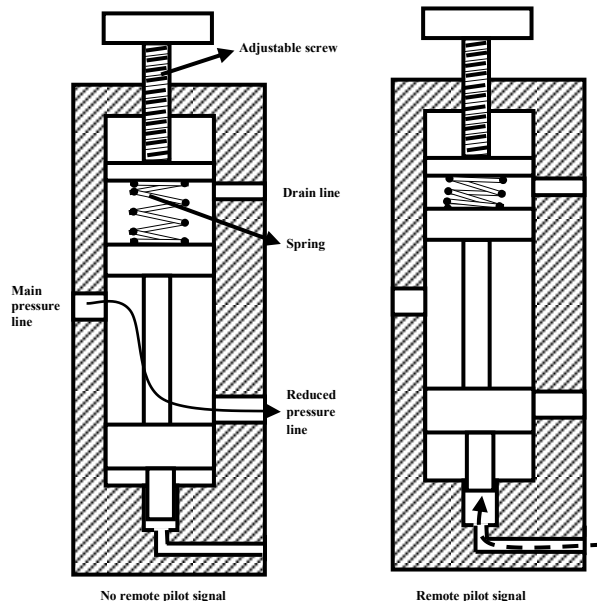
A pilot-operated unloading valve has less pressure override than its direct-acting counterpart. So it does not dump part of the flow prematurely.



A pilot-operated unloading relief valve is the same as a pilot-operated relief valve with the addition of an unloading spool. Without the unloading spool, this valve would function just like any pilot-operated relief valve. Pressure buildup in the pilot section would open some flow to the tank and unbalance the poppet, allowing it to open and relieve excess pump flow.

Schematic diagram of unloading valve is shown in Fig. 1.10. In a pilot-operated unloading valve; the unloading spool receives a signal through the remote-pilot port when pressure in the working circuit goes above its setting. At the same time, pressure on the spring-loaded ball in the pilot section starts to open it. Pressure drop on the front side of the unloading spool lowers back force and pilot pressure from the high-pressure circuit forces the spring-loaded ball completely off its seat. Now there is more flow going to the tank than what the control orifice can keep up with. The main poppet opens at approximately 20 psi. Now, all high-volume pump flow can go to the tank at little or no pressure drop and all horsepower can go to the low-volume pump to do the work. When pressure falls approximately 15% below the pressure set in the pilot section, the spring-loaded ball closes and pushes the unloading spool back for the next cycle.

An unloading valve requires no electric signals. This eliminates the need for extra persons when troubleshooting. These valves are very reliable and seldom require maintenance, adjustment or replacement. An unloading valve unloads the pump when the desired pressure is reached. It allows rapid discharge of pressurized oil near atmospheric pressure. As soon as the system pressure reaches the setting pressure that is available at the pilot port, it lifts the spool against the spring force. When the spool is held by the pilot pressure, the delivery from the pump goes to the tank. An unloading valve is used to perform operations such as stamping, coining, punching, piercing, etc.



**Figure 1.10** Unloading valve.

### 1.5 Counterbalance Valve

Schematic diagram of counterbalance valve is shown in Fig. 1.14. These normally closed valves are primarily used to maintain a back pressure on a vertical cylinder to prevent it from falling due to gravity. They are used to prevent a load from accelerating uncontrollably. This situation can occur in vertical cylinders in which the load is a weight. This can damage the load or even the cylinder itself when the load is stopped quickly at the end of the travel.

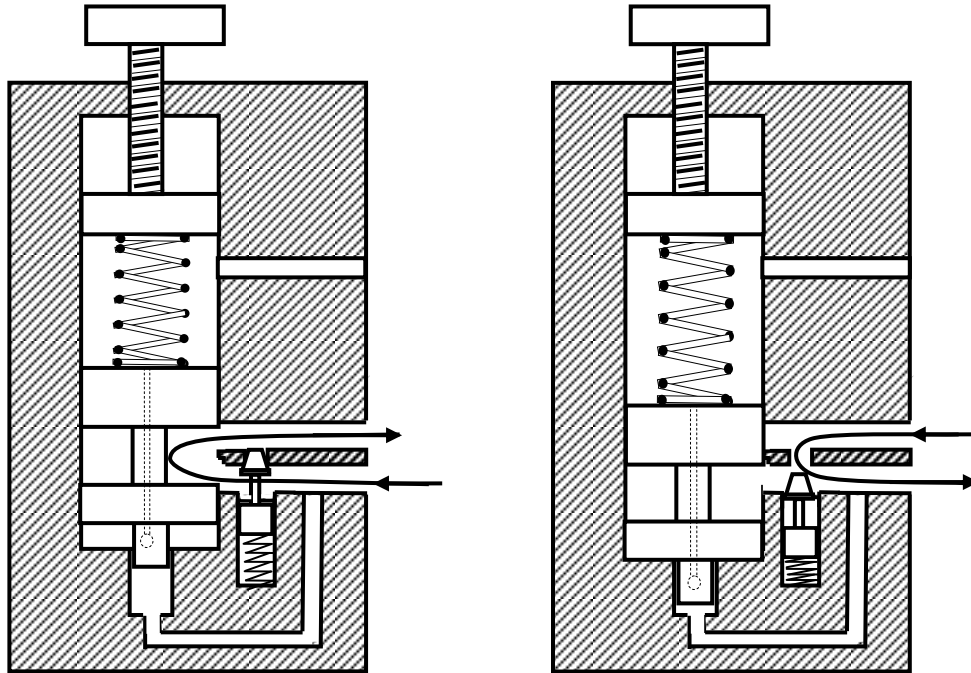


Figure 1.14 Counterbalance valve.

valve's primary port is connected to the cylinder's rod end and the secondary port to the directional control valve. The pressure setting is slightly higher than that required to keep the load from free-falling. When the pressurized fluid flows to the cylinder's cap end, the cylinder extends, increasing pressure in the rod end and shifting the main spool in the counterbalance valve. This creates a path that permits the fluid to flow through the secondary port via the directional control valve and to the reservoir. As the load is raised, the integral check valve opens to allow the cylinder to retract freely.

If it is necessary to relieve back pressure at the cylinder and increase the force at the bottom of the stroke, the counterbalance valve can be operated remotely. Counterbalance valves are usually drained internally. When the cylinder extends, the valve must open and its secondary port should be connected to the reservoir. When the cylinder retracts, it matters little that load pressure is felt in the drain passage because the check valve bypasses the valve's spool. Graphic symbol of a pressure-reducing valve is shown in Fig. 1.15.

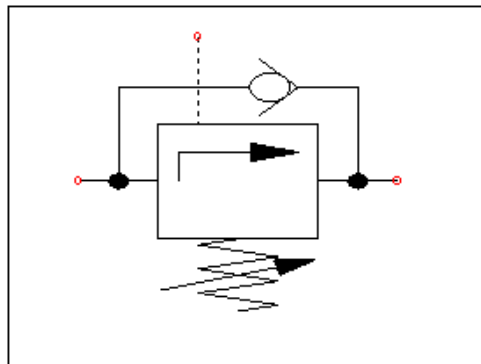


Figure 1.15 Symbolic representation of a counterbalance valve.

## 1.7 Pressure Sequence Valve

A sequence valve is a pressure-control valve that is used to force two actuators to operate in sequence. They are similar to pressure-relief valves. Schematic diagram of sequence valve is shown in Fig. 1.18. Instead of sending flow back to the tank, a sequence valve allows flow to a branch circuit, when a preset pressure is reached. The check valve allows the sequence valve to be bypassed in the reverse direction. The component enclosure line indicates that the check valve is an integral part of the component. The sequence valve has an external drain line; therefore, a line must be connected from the sequence valve's drain port to the tank. The symbol for a sequence valve is shown in Fig. 1.19.

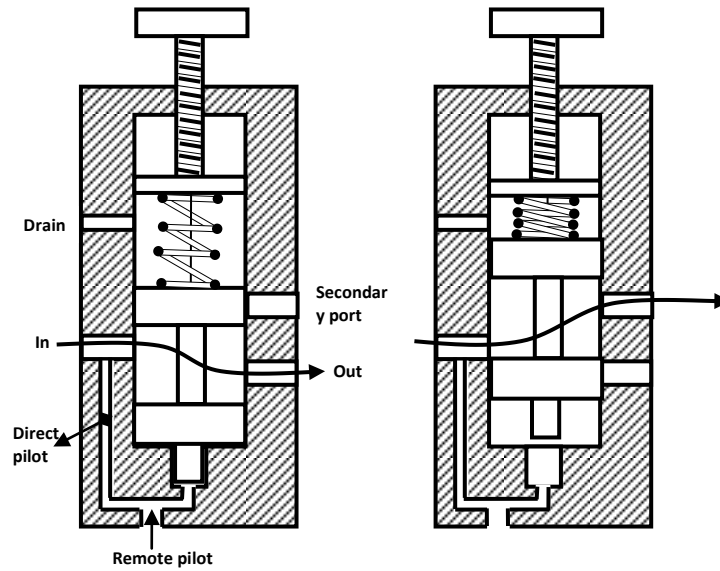


Figure 1.18 Sequence valve.

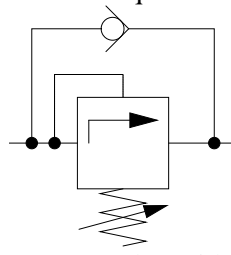


Figure 1.19 Sequence valve with a check valve.

## 5. FLOW-CONTROL VALVES

### 1.1 Introduction

Flow-control valves, as the name suggests, control the rate of flow of a fluid through a hydraulic circuit. Flow-control valves accurately limit the fluid volume rate from fixed displacement pump to or from branch circuits. Their function is to provide velocity control of linear actuators, or speed control of rotary actuators. Typical application include regulating cutting tool speeds, spindle speeds, surface grinder speeds, and the travel rate of vertically supported loads moved upward and downward by forklifts, and dump lifts. Flow-control valves also allow one fixed displacement pump to supply two or more branch circuits fluid at different flow rates on a priority basis. Typically, fixed displacement pumps are sized to supply maximum system volume flow rate demands. For industrial applications feeding two or more branch circuits from one pressurized manifold source, an oversupply of fluid in any circuit operated by itself is virtually assured. Mobile applications that supply branch circuits, such as the power steering and front end loader from one pump pose a similar situation. If left unrestricted, branch circuits receiving an oversupply of fluid would operate at greater than specified velocity, increasing the likelihood of damage to work, hydraulic system and operator.

#### 1.1.1 Functions of Flow-Control Valves

Flow-control valves have several functions, some of which are listed below:

1. **Regulate the speed of linear and rotary actuators:** They control the speed of piston that is dependent on the flow rate and area of the piston:

$$\text{Velocity of piston } (V_p) \text{ (m/s)} = \frac{\text{Flow rate in the actuator (m}^3/\text{s)}}{\text{Piston area (m}^2)} = \frac{Q}{A_p}$$

2. **Regulate the power available to the sub-circuits by controlling the flow to them:**

$$\begin{aligned} \text{Power (W)} &= \text{Flow rate (m}^3/\text{s)} \times \text{Pressure (N/m}^2) \\ \Rightarrow P &= Q \times p \end{aligned}$$

3. **Proportionally divide or regulate the pump flow to various branches of the circuit:** It transfers the power developed by the main pump to different sectors of the circuit to manage multiple tasks, if necessary.

#### 1.1.2 Classification of Flow-Control Valves

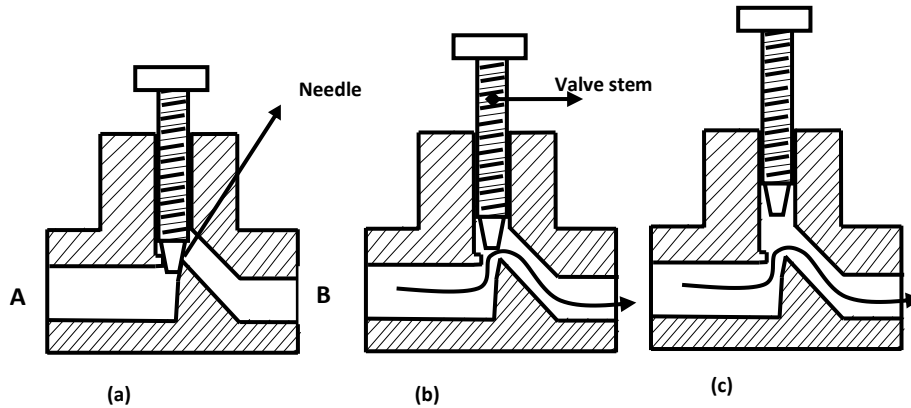
Flow-control valves can be classified as follows:

1. Non-pressure compensated.
2. Pressure compensated.

##### 1.1.2.1 Non-Pressure-Compensated Valves

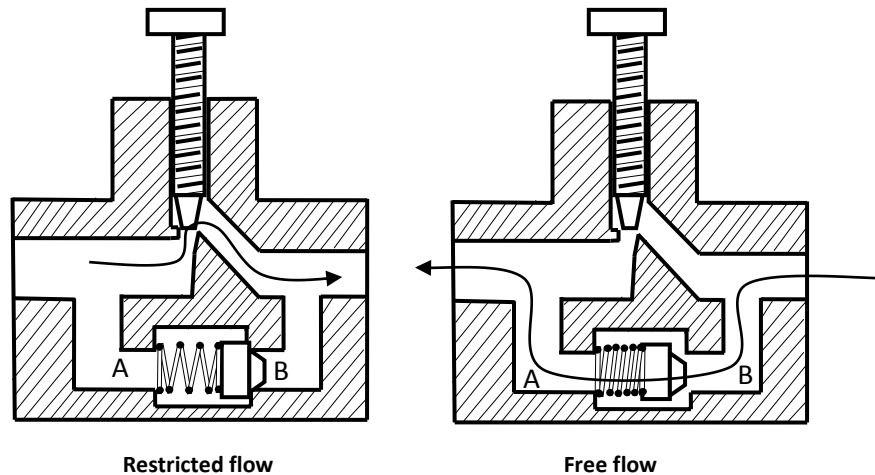
Non-pressure-compensated flow-control valves are used when the system pressure is relatively constant and motoring speeds are not too critical. The operating principle behind these valves is that the flow through an orifice remains constant if the pressure drop across it remains the same. In other words, the rate of flow through an orifice depends on the pressure drop across it.

The disadvantage of these valves is discussed below. The inlet pressure is the pressure from the pump that remains constant. Therefore, the variation in pressure occurs at the outlet that is defined by the work load. This implies that the flow rate depends on the work load. Hence, the speed of the piston cannot be defined accurately using non-pressure-compensated flow-control valves when the working load varies. This is an extremely important problem to be addressed in hydraulic circuits where the load and pressure vary constantly.



**Figure 1.3** Non-pressure-compensated needle-type flow-control valve. (a) Fully closed; (b) partially opened; (c) fully opened.

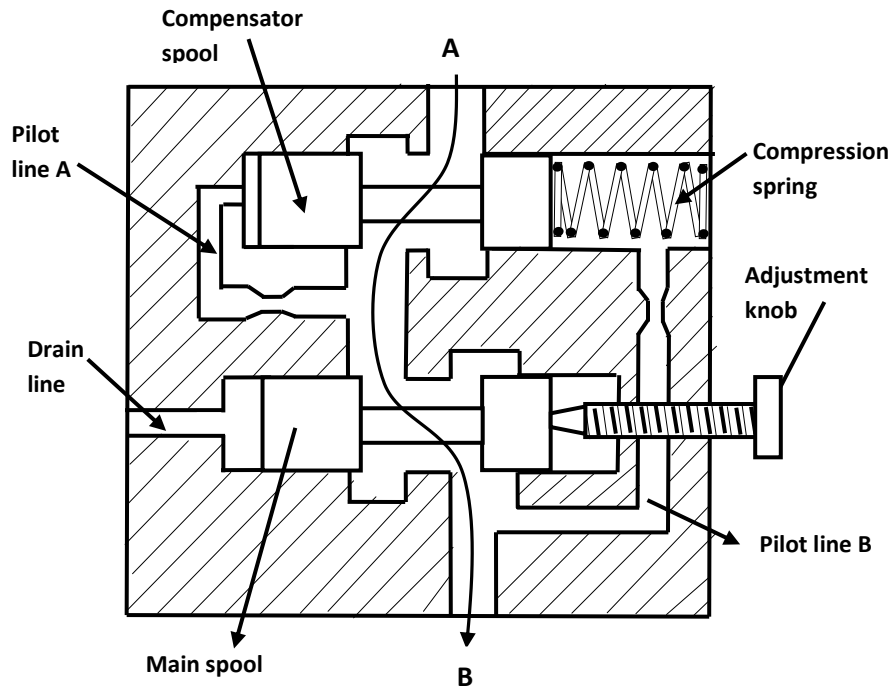
Schematic diagram of non-pressure-compensated needle-type flow-control valve is shown in Fig. 1.3. It is the simplest type of flow-control valve. It consists of a screw (and needle) inside a tube-like structure. It has an adjustable orifice that can be used to reduce the flow in a circuit. The size of the orifice is adjusted by turning the adjustment screw that raises or lowers the needle. For a given opening position, a needle valve behaves as an orifice. Usually, charts are available that allow quick determination of the controlled flow rate for given valve settings and pressure drops. Sometimes needle valves come with an integrated check valve for controlling the flow in one direction only. The check valve permits easy flow in the opposite direction without any restrictions. As shown in Fig. 1.4, only the flow from A to B is controlled using the needle. In the other direction (B to A), the check valve permits unrestricted fluid flow.



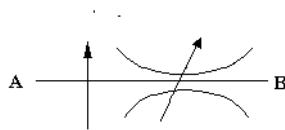
**Figure 1.4** Flow-control valve with an integrated check valve.

### 1.1.2.2 Pressure-Compensated Valves

Pressure-compensated flow-control valves overcome the difficulty caused by non-pressure-compensated valves by changing the size of the orifice in relation to the changes in the system pressure. This is accomplished through a spring-loaded compensator spool that reduces the size of the orifice when pressure drop increases. Once the valve is set, the pressure compensator acts to keep the pressure drop nearly constant. It works on a kind of feedback mechanism from the outlet pressure. This keeps the flow through the orifice nearly constant.



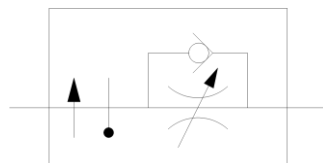
**Figure 1.5** Sectional view of a pressure-compensated flow-control valve.



**Figure 1.6** Graphic symbol of a pressure-compensated flow-control valve.

Schematic diagram of a pressure compensated flow-control valve is shown in Fig. 1.5 and its graphical symbol in Fig. 1.6. A pressure-compensated flow-control valve consists of a main spool and a compensator spool. The adjustment knob controls the main spool's position, which controls the orifice size at the outlet. The upstream pressure is delivered to the valve by the pilot line A. Similarly, the downstream pressure is ported to the right side of the compensator spool through the pilot line B. The compensator spring biases the spool so that it tends toward the fully open position. If the pressure drop across the valve increases, that is, the upstream pressure increases relative to the downstream pressure, the compensator spool moves to the right against the force of the spring. This reduces the flow that in turn reduces the pressure drop and tries to attain an equilibrium position as far as the flow is concerned.

Performance of flow-control valve is also affected by temperature changes which changes the viscosity of the fluid. Therefore, often flow-control valves have temperature compensation. Graphical symbol for pressure and temperature compensated flow-control valve is shown in Fig. 1.7.



**Figure 1.7** Pressure- and temperature-compensated flow-control valve.

## 6. ACCUMULATORS

### 1.1 Introduction

A hydraulic accumulator is a device that stores the potential energy of an incompressible fluid held under pressure by an external source against some dynamic force. This dynamic force can come from different sources. The stored potential energy in the accumulator is a quick secondary source of fluid power capable of doing useful work.

There are three basic types of accumulators:

**1. Weight-loaded or gravity accumulator:** Schematic diagram of weight loaded accumulator is shown in Fig. 1.1. It is a vertically mounted cylinder with a large weight. When the hydraulic fluid is pumped into it, the weight is raised. The weight applies a force on the piston that generates a pressure on the fluid side of piston. The advantage of this type of accumulator over other types is that it applies a constant pressure on the fluid throughout its range of motion. The main disadvantage is its extremely large size and heavy weight. This makes it unsuitable for mobile application.

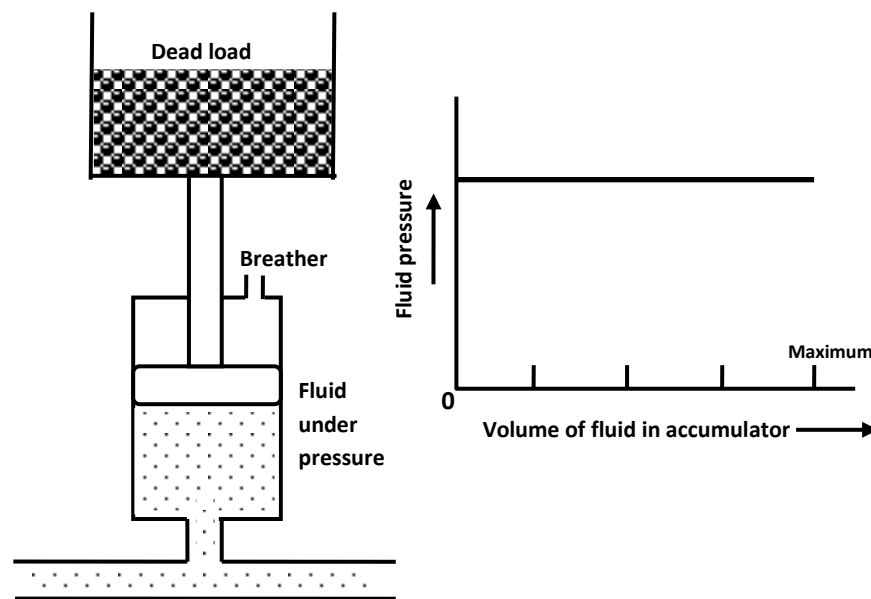


Figure 1.1 Dead weight accumulator.

**2. Spring-loaded accumulator:** A spring-loaded accumulator stores energy in the form of a compressed spring. A hydraulic fluid is pumped into the accumulator, causing the piston to move up and compress the spring as shown in Fig. 1.2. The compressed spring then applies a force on the piston that exerts a pressure on the hydraulic fluid.

This type of accumulator delivers only a small volume of oil at relatively low pressure. Furthermore, the pressure exerted on the oil is not constant as in the dead-weight-type accumulator. As the springs are compressed, the accumulator pressure reaches its peak, and as the springs approach their free lengths, the accumulator pressure drops to a minimum.

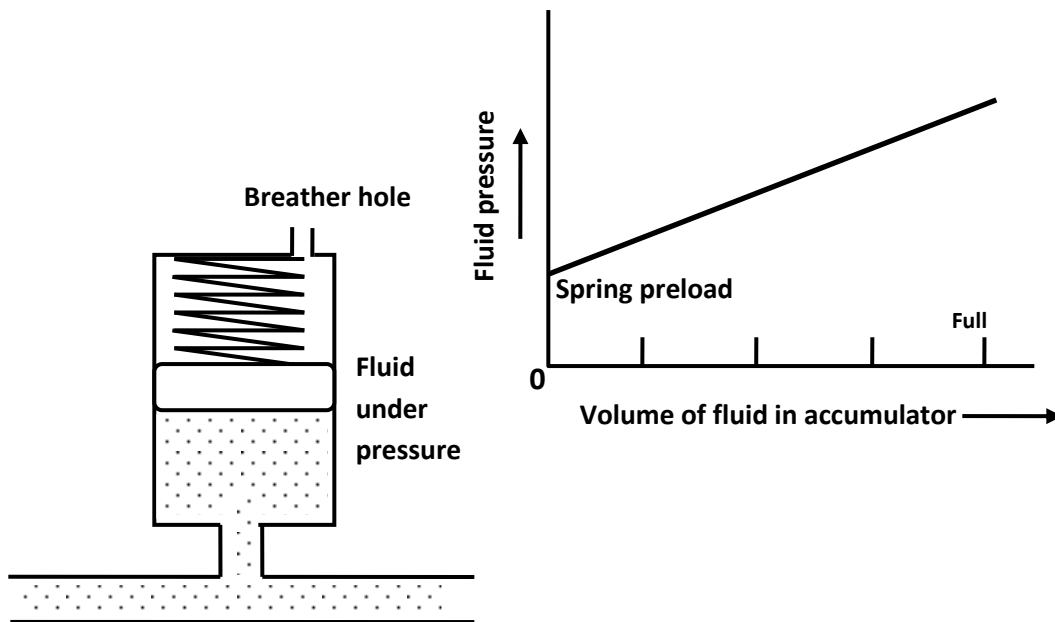


Figure 1.2 Spring-loaded accumulator.

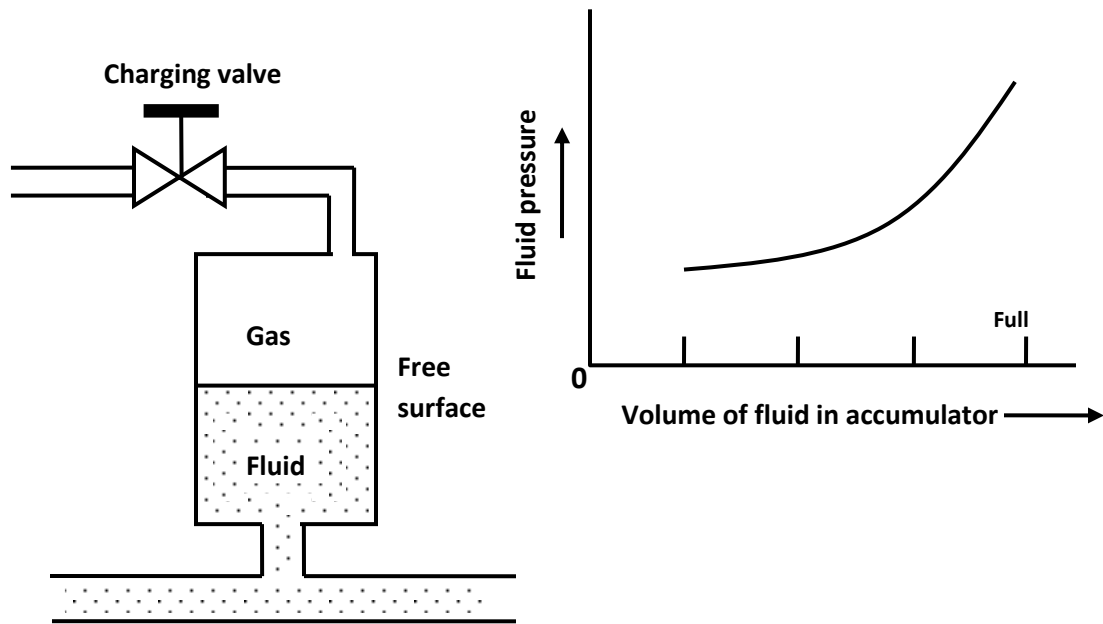
**3. Gas-loaded accumulator:** A gas-loaded accumulator is popularly used in industries. Here the force is applied to the oil using compressed air. Schematic diagram of a gas loaded accumulator is shown in Fig. 1.3. A gas accumulator can be very large and is often used with water or high water-based fluids using air as a gas charge. Typical application is on water turbines to absorb pressure surges owing to valve closure and on ram pumps to smooth out the delivery flow. The exact shape of the accumulator characteristic curve depends on pressure–volume relations:

- **Isothermal (constant temperature):** This occurs when the expansion or compression of the gas is very slow. The relationship between absolute pressure  $p$  and volume  $V$  of the gas is constant:
 
$$pV = \text{constant} \quad (1.1)$$
- **Isentropic (adiabatic processes):** This is where there is no flow of energy into or out of the fluid. The law that the gas obeys is given by  $pV^\gamma = \text{constant}$ , where  $\gamma$  is ratio of specific heat and is approximately equal to 1.4.
- **Polytropic:** This is somewhere between isothermal and isentropic. This gas change is governed by the law  $pV^n = \text{constant}$ , where  $n$  is somewhere between 1 and 1.4 and is known as the polytropic coefficient.

There are two types of gas-loaded accumulators:

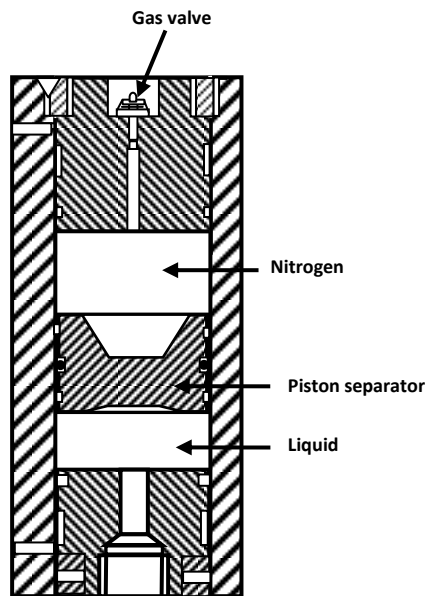
- **Non-separator-type accumulator:** Here the oil and gas are not separated. Hence, they are always placed vertically.
- **Separator-type accumulator:** Here the oil and gas are separated by an element. Based on the type of element used to separate the oil and gas, they are classified as follows:





**Figure 1.3** Gas-loaded accumulator.

**(a) Piston-type accumulator:** Schematic diagram of a piston type accumulator is shown in Fig. 1.4. It consists of a cylinder with a freely floating piston with proper seals. Its operation begins by charging the gas chamber with a gas (nitrogen) under a pre-determined pressure. This causes the free sliding piston to move down. Once the accumulator is pre-charged, a hydraulic fluid can be pumped into the hydraulic fluid port. As the fluid enters the accumulator, it causes the piston to slide up, thereby compressing the gas that increases its pressure and this pressure is then applied to the hydraulic fluid through the piston. Because the piston is free sliding, the pressure on the gas and that on the hydraulic fluid are always equal.



**Figure 1.4** Piston-type accumulator.

- (b) *Diaphragm accumulator*: In this type, the hydraulic fluid and nitrogen gas are separated by a synthetic rubber diaphragm. Schematic diagram of diaphragm accumulator is shown in Fig. 1.5. The advantage of a diaphragm accumulator over a piston accumulator is that it has no sliding surface that requires lubrication and can therefore be used with fluids having poor lubricating qualities. It is less sensitive to contamination due to lack of any close-fitting components.

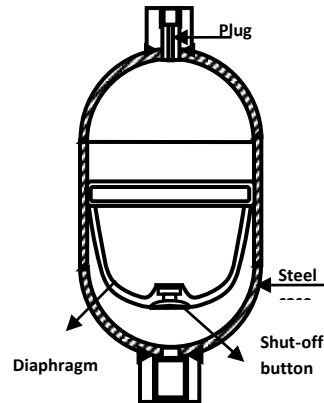


Figure 1.5 Diaphragm-type accumulator.

- (c) *Bladder accumulator*: It functions in the same way as the other two accumulators. Schematic diagram of bladder accumulator is shown in Fig. 1.6. Here the gas and the hydraulic fluid are separated by a synthetic rubber bladder. The bladder is filled with nitrogen until the designed pre-charge pressure is achieved. The hydraulic fluid is then pumped into the accumulator, thereby compressing the gas and increasing the pressure in the accumulator. The port cover is a small piece of metal that protects the bladder from damage as it expands and contacts the fluid port.

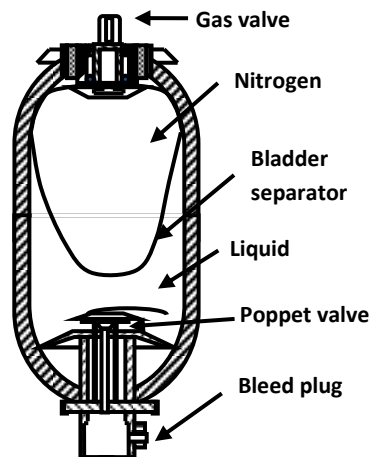


Figure 1.6 Bladder-type accumulator.

## 1.4 Reservoirs

The functions of a fluid reservoir in a power hydraulic system are as follows:

1. To provide a chamber in which any change in the volume of fluid in a hydraulic circuit can be accommodated. When the cylinder extends, there is an increased volume of fluid in the circuit and consequently there is a decrease in the reservoir level.
2. To provide a filling point for the system.
3. To serve as a storage space for the hydraulic fluid used in the system.
4. It is used as the location where the fluid is conditioned.
5. To provide a volume of fluid which is relatively stationary to allow entrained air to separate out and heavy contaminants to settle. The reservoir is where sludge, water and metal chips settle.
6. It is a place where the entrained air picked up by the oil is allowed to escape.
7. To accomplish the dissipation of heat by its proper design and to provide a radiating and convective surface to allow the fluid to cool.

A reservoir is constructed with steel plates. The inside surface is painted with a sealer to prevent rust due to condensed moisture. At the bottom, it contains a drain plug to allow the complete drainage of the tank when required. A removable head can be provided for easy access during cleaning. A vented breather cap is also included that contains an air filtering screen. This allows the tank to breathe as the oil level changes due to system demand requirements.

A baffle plate extends lengthwise across the center of the tank. The purpose of the baffle plate is to separate the pump inlet line from the return line to prevent the same fluid from recirculating continuously within the tank. The functions of a baffle plate are as follows:

1. To permit foreign substances to settle to bottom.
2. To allow entrained air to escape from oil.
3. To prevent localized turbulence in the reservoir.
4. To promote heat dissipation through reservoir walls.

The return line should enter the reservoir on the side of the baffle plate that is opposite to the pump suction line.

### 1.4.1 Features of a Hydraulic Reservoir

Schematic diagram of hydraulic reservoir is shown in Fig.1.11. There are many components mounted on reservoir and each one of them having specific features. Following are the features of a hydraulic reservoir:

1. **Filler cap (breather cap):** It should be airtight when closed but may contain the air vent that filters air entering the reservoir to provide a gravity push for proper oil flow.
2. **Oil level gauge:** It shows the level of oil in the reservoir without having to open the reservoir.
3. **Baffle plate:** It is located lengthwise through the center of the tank and is two-third the height of the oil level. It is used to separate the outlet to the pump from the return line. This ensures a circuitous flow instead of the same fluid being recirculated. The baffle prevents local turbulence in the tank and allows foreign material to settle, get rid of entrapped air and increase heat dissipation.
4. **Suction and return lines:** They are designed to enter the reservoir at points where air turbulence is the least. They can enter the reservoir at the top or at the sides, but their ends should be near the bottom of the tank. If the return line is above the oil level, the returning oil can foam and draw in air.

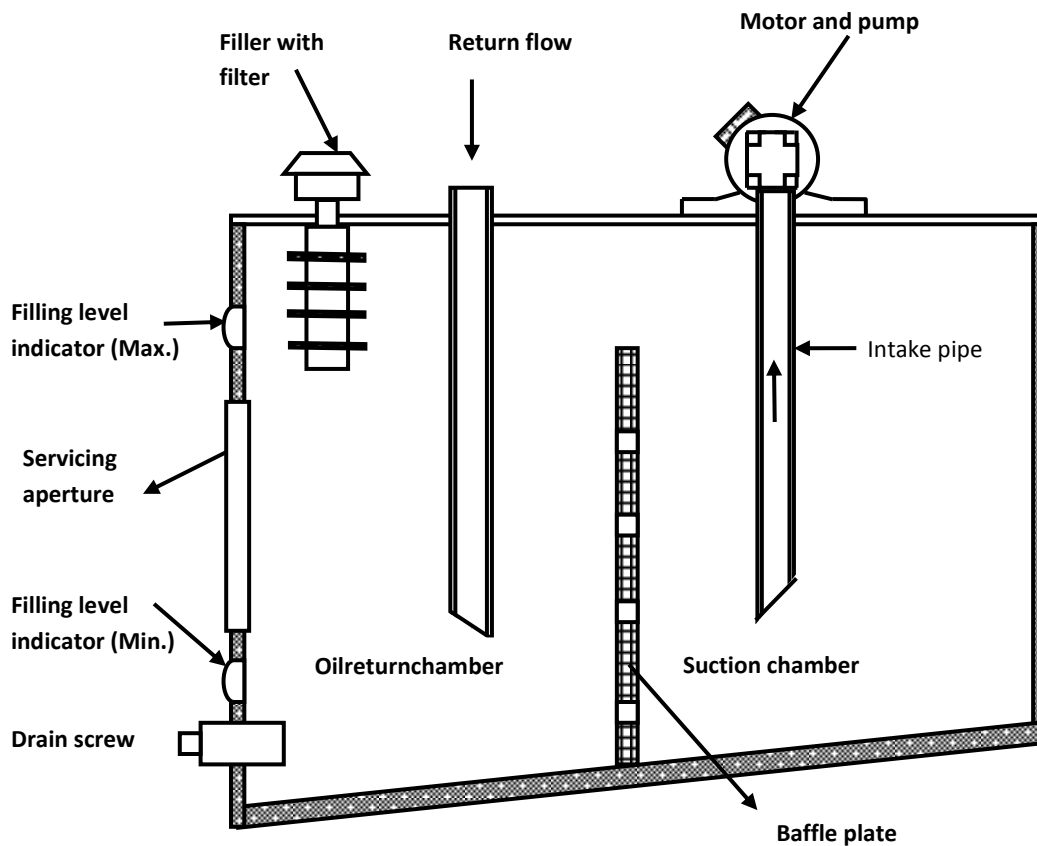


Figure 1.11 Hydraulic reservoir.

5. **Intake filter:** It is usually a screen that is attached to the suction pipe to filter the hydraulic oil.
6. **Drain plug:** It allows oil to be drained from the reservoir. Some drain plugs are magnetic to help remove metal chips from the oil.
7. **Strainers and filters:** Strainers and filters are designed to remove foreign particles from the hydraulic fluid. Strainers and filters are discussed in detail in Section 1.6.

## 1.4.2 Types of Reservoirs

Industrial reservoirs come in a variety of styles. Some of them are the following:

1. **Non-pressurized:** The reservoir may be vented to atmosphere using an air filter or a separating diaphragm. The type most commonly used in industry, normally, has an air breather filter, although in very dirty environments, diaphragms or air bags are used.
2. **Pressurized:** A pressurized reservoir usually operates between 0.35 and 1.4 bar and has to be provided with some method of pressure control; this may be a small air compressor maintaining a set charge pressure. In motor circuits where there is a little change in fluid volume in the reservoir, a simple relief valve may be used to limit the air pressure that alters with changes in temperature. The advantages of a pressurized reservoir are that it provides boost pressure to the main pump and prevents the ingress of atmospheric dirt.

### **30'INTRODUCTION TO PNEUMATICS**

#### **1.1 PNEUMATICS AND ITS MEANING.**

The English word pneumatic and its associate noun pneumatics are derived from the Greek “**pneuma**” meaning breath or air. Originally coined to give a name to the science of the motions and properties of air. Compressed air is a vital utility- just like water, gas and electricity used in countless ways to benefit everyday life. Pneumatics is application of compressed air (pressurized air) to power machine or control or regulate machines. Simply put, Pneumatics may be defined as branch of engineering science which deals with the study of the behavior and application of compressed air. Pneumatics can also be defined as the branch of fluid power technology that deals with generation, transmission and control of power using pressurized air. Gas in a pneumatic system behaves like a spring since it is compressible.

Any gas can be used in pneumatic system but air is the most usual, for obvious reasons. Exceptions are most likely to occur on aircraft and space vehicles where an inert gas such as nitrogen is preferred or the gas is one which is generated on board. Pure nitrogen may be used if there is a danger of combustion in a work environment. In Pneumatic control, compressed air is used as the working medium, normally at a pressure from 6 bar to 8 bar. Using Pneumatic Control, maximum force up to 50 kN can be developed. Actuation of the controls can be manual, Pneumatic or Electrical actuation. Signal medium such as compressed air at pressure of 1-2 bar can be used [Pilot operated Pneumatics] or Electrical signals [D.C or A.C source- 24V – 230V] can be used [Electro pneumatics]

#### **1.3 APPLICATIONS OF PNEUMATICS**

Pneumatic systems are used in many applications. New uses for pneumatics are constantly being discovered. In construction, it is indispensable source of power for such tools as air drills, hammers, wrenches, and even air cushion supported structures, not to mention the many vehicles using air suspension , braking and pneumatic tires.

In manufacturing, air is used to power high speed clamping, drilling, grinding , and assembly using pneumatic wrenches and riveting machines. Plant air is also used to power hoists and cushion support to transport loads through the plant.

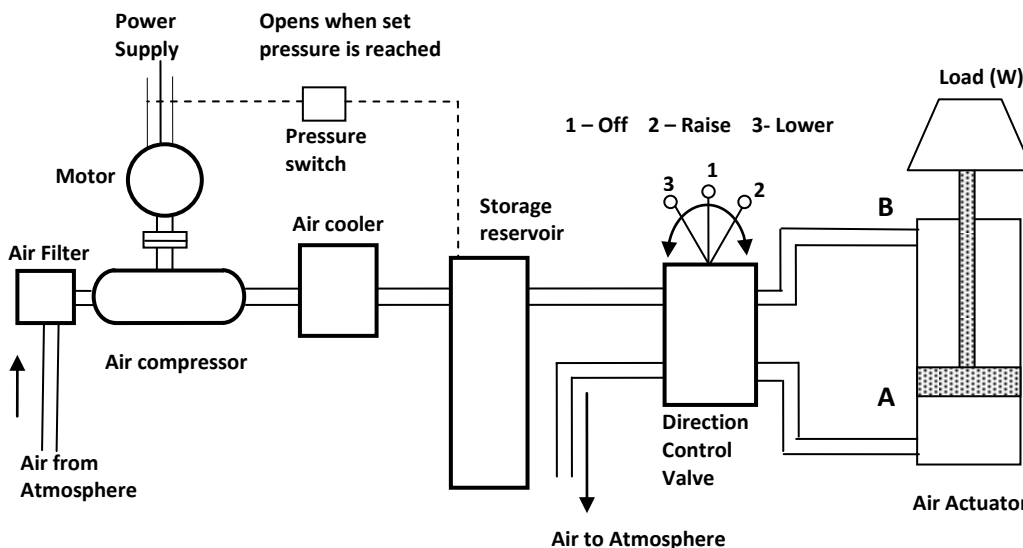
Many recent advances in air – cushion support are used in the military and commercial marine transport industry.

**Table 1.2:** Industrial applications of Pneumatics

Material Handling	Manufacturing	Other applications
Clamping	Drilling	Aircraft
Shifting	Turning	Cement plants
positioning	Milling	chemical plants
Orienting	Sawing	Coal mines
Feeding	Finishing	Cotton mills
Ejection	Forming	Dairies
Braking	Quality Control	Forge shops
Bonding	Stamping	Machine tools
Locking	Embossing	Door or chute control
Packaging	Filling	Turning and inverting parts
Feeding		
Sorting		
stacking		

### 1.6 BASIC COMPONENTS OF PNEUMATIC SYSTEMS

Pneumatic system carries power by employing compressed gas generally air as a fluid for transmitting the energy from an energy-generating source to an energy – use point to accomplish useful work. **Figure 1.6** shows the simple circuit of a pneumatic system with basic components.



**Figure 1.6** Components of Pneumatic System

Functions of components

- Pneumatic actuator converts the fluid power into mechanical power to do useful work
- Compressor is used to compress the fresh air drawn from the atmosphere.
- Storage reservoir is used to store a given volume of compressed air.
- Valves are used to control the direction, flow rate and pressure of compressed air.
- External power supply (Motor) is used to drive the compressor.
- Piping system carries the pressurized air from one location to another.

Air is drawn from the atmosphere through air filter and raised to required pressure by an air compressor. As the pressure rises, the temperature also rises and hence air cooler is provided to cool the air with some preliminary treatment to remove the moisture.

Then the treatment pressurized air needs to get stored to maintain the pressure. With the storage reservoir, a pressure switch is fitted to start and stop the electric motor when pressure falls and reached the required level, respectively.

The cylinder movement is controlled by pneumatic valve. one side of the pneumatic valve is connected to the compressed air and silencers for the exhaust air and the other side of the valve is connected to port A and Port B of the cylinder.

Position of the valve is as follows

1. **Raise:** To lift the weight , the compressed air supply is connected to port A and the port B is connected to the exhaust line, by moving the valve position to the “ Raise”
2. **Lower:** To bring the weight down, the compressed air line is connected to port B and port A is connected to exhaust air line , by moving the valve position to the “lower”
3. **Off:** The weight can be stopped at a particular position by moving the valve to position to “Off” position. This disconnects the port A and port B from the pressurized line and the retrieval line, which locks the air in the cylinder.

#### **Advantages of Pneumatic system**

- Low inertia effect of pneumatic components due to low density of air.
- Pneumatic Systems are light in weight.
- Operating elements are cheaper and easy to operate
- Power losses are less due to low viscosity of air
- High output to weight ratio
- Pneumatic systems offers a safe power source in explosive environment
- Leakage is less and does not influence the systems. Moreover, leakage is not harmful

#### **Disadvantages of Pneumatic systems**

- Suitable only for low pressure and hence low force applications
- Compressed air actuators are economical up to 50 kN only.
- Generation of the compressed air is expensive compared to electricity
- Exhaust air noise is unpleasant and silence has to be used.
- Rigidity of the system is poor
- Weight to pressure ratio is large
- Less precise. It is not possible to achieve uniform speed due to compressibility of air
- Pneumatic systems is vulnerable to dirt and contamination

## 1.7 COMPARISON BETWEEN HYDRAULIC AND PNEUMATIC SYSTEM.

Sl. No	Hydraulic system	Pneumatic system
1	It employs a pressurized liquid as fluid	it employs a compressed gas usually air as a fluid
2	Oil hydraulics system operates at pressures upto 700 bar.	Pneumatics systems usually operate at 5 to 10 bar.
3	Generally designed for closed systems	Pneumatic systems are usually designed as open system
4	System get slow down of leakage occurs	Leakage does not affect the system much more
5	Valve operations are difficult	Easy to operate the valves
6	Heavier in weight	Light in weight
7	Pumps are used to provide pressurized liquids	Compressors are used to provide compressed gas
8	System is unsafe to fire hazards	System is free from fire hazards
9	Automatic lubrication is provided	Special arrangements for lubrication needed.

## 1.8 COMPARISON OF DIFFERENT POWER SYSTEMS

Property	Electrical/Mechanical	Pneumatic	Hydraulic
Energy	IC engines, electrical energy is used to drive motors	Electrical energy is used to drive compressor and other equipments	I C Engines Electric Motor Air Turbine are used to drive hydraulic pumps.
Medium	. There is no medium, Energy is transferred through Levers, Gears, Shafts	Compressed air/gas in Pipes and hoses	Pressurized liquid in Pipes and hoses
Energy storage	Batteries	Reservoir, air tank	Accumulators
Regulations	Variable frequency drives	Pneumatic valves	Hydraulic valves
Transmitters	Transmitted through mechanical components like levers, gears, cams, screw cts	Transmitted through pneumatic cylinders , rotary drives and rotary actuators	Transmitted through hydraulic cylinders, and hydraulic rotary actuators.
Distribution system efficiency	Good	Limited (say up to 1000m)	Good (say up to 100 m)
Operating speed	Low	Limited (up to 1.5 m/s)	Limited (up to 0.5 m/s)
Positioning accuracy	Precision in terms of few micron can be achieved	. Precision in terms of few mm ( usually 0.1 mm) can be achieved	Precision in terms of few micron can be achieved
Stability	Good stability is possible using mechanical elements	. Low stability is due to High compressibility of air	Good stability is possible due to low compressibility



Forces	Mechanical elements break down if overloaded. Poor overloading capacity	Protected against overload with system pressure of 6 to 8 bar, forces up to 50 kN can be generated.	Protected against overload with high system pressure of 600 bar, very large forces can be generated.
Cost of energy	Lowest	Highest	Medium
Linear actuators	Short stroke length using mechanical/electrical transmission elements	Is possible using pneumatic actuators (cylinders), it can produce medium force.	Is possible using hydraulic actuators(cylinders), and it can produce heavy force
Rotary actuators	AC , DC, Servo motors and stepper motors can be used	Pneumatic rotary actuators can be used	Hydraulic motors and vane motors can be used
Controllable force	Possible with solenoid and DC motors, Needs cooling and hence complicated	medium force can be controlled easily	High force can be controlled
Work environment	Danger because of electric shock	Noise	Dangerous, unsightly and fire hazardous because of leakage.

## **1.1 FLUID CONDITIONERS**

The purpose of the fluid conditioners is to make the compressed air more acceptable and suitable fluid medium for the pneumatic system as well as the operating personal. The following five fluid conditioners are used in pneumatic systems

1. Air Filters
2. Air Regulators
3. Air Lubricator

### **4.1.1 AIR FILTERS**

The purpose of the air filter is to clean the compressed air of all impurities and any condensate it contains.

#### **a) Function of air filters**

- To remove all foreign matter and allow dry and clean air flow without restriction to regulator and then to the lubricator
- To condensate and remove water from the air
- To arrest fine particles and all solid contaminants from air

Filters are available in wide range starting from a fine mesh wire cloth (which strains heavy foreign particles) to elements made of synthetic material (which removes very small particles) Usually in line filter elements can remove contaminants in the 5-50 micron range.

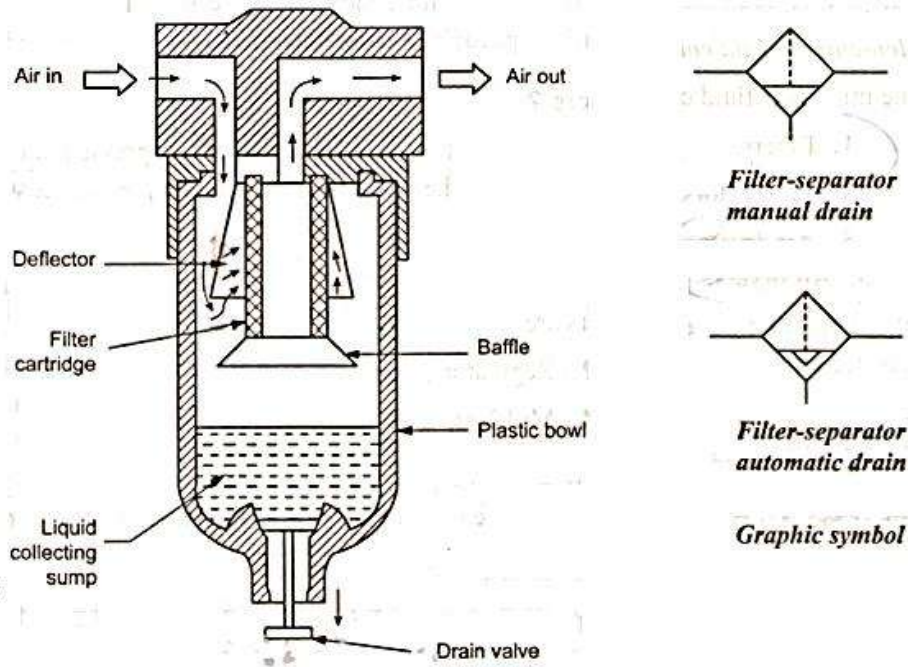
#### **d) Factor affecting selection of filters**

While selecting the filters, the following factors should be taken into account.

- Size of particles to be filtered from the system
- Capacity of the filter
- Accessibility and maintainability
- Life of the filter
- Ability to drain the condensate

#### **e) Construction**

The construction of typical cartridge type filter along with graphical symbols is shown in Figure 1.1. It consists of filter cartridge, Deflector, bowl, water drain valve. Filter bowl is usually made of plastic and transparent. For pressure more than 10 bar, bowl may be made of brass.



**Figure 1.1** Construction of a Air filter.

**f) Operation**

Air enters the inlet port of the air filter through angled louvers. This causes the air to spin as it enters the bowl. The centrifugal action of the rotating air causes the larger pieces of dirt and water particles to be thrown against the inner wall of the filter bowl. These contaminants then flow down into the bottom of the filter bowl.

A baffle prevents turbulent air from splashing water on to the filter element. The air, which has been pre-cleaned in this way, then passes through the filter element, where the fine dirt particles are filtered out. The size of the dirt particles which can be filtered out depends on mesh size of filter element (usually 5-50 microns). The compressed air then exits through the outer port.

The pressure difference between inlet and outlet will indicate the degree to which the filter element is clogged. Commercially available filters have many additional features like automatic drain facility, coalescing type filter element, service life indicator etc.

**1.1.2 AIR REGULATOR**

**a) Function:** The function of the air pressure regulator is to maintain working pressure virtually constant regardless of fluctuations of the line pressure and air consumption. When the pressure is too low, it results in poor efficiencies and when the pressure is too high, energy is wasted and equipment’s performance decay faster.

In pneumatic system, pressure fluctuations occur due to variation in supply pressure or load pressure. It is therefore essential to regulate the pressure to match the requirement of load regardless of variation in supply pressure or load pressure.

### c) Types of Pressure regulator

There are two types of Pressure regulators

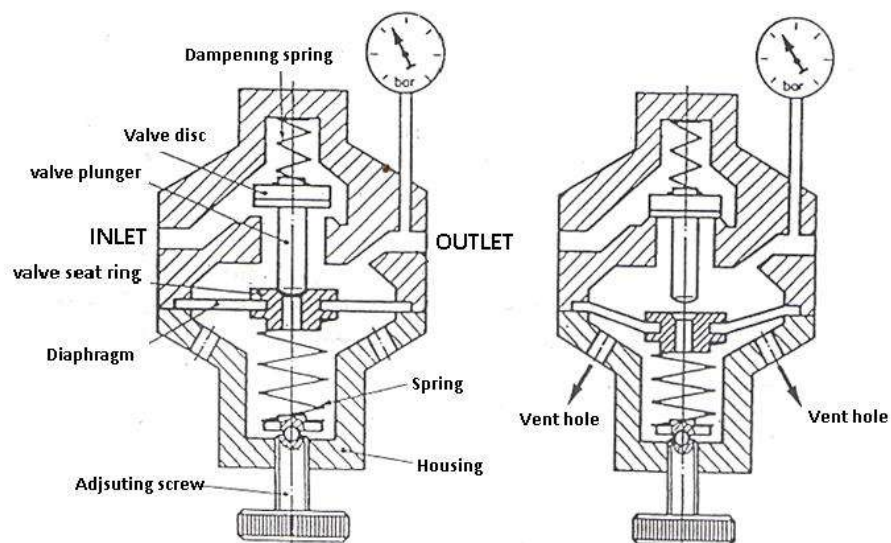
- i) Diaphragm type regulator
- ii) Piston type regulator

Diaphragm type regulator is commonly used in Industrial pneumatic system. There are two types of diaphragm type regulator

- i) Non-relieving or non-venting type.
- ii) Relieving or venting type

#### 1.1.2.1 Relieving or Venting Type Pressure regulator

A Relieving type pressure regulator is shown in **Figure 1.4**, Outlet pressure is sensed by a diaphragm preloaded with a adjustable pressure setting spring. The compressed air, which flows through a controlled cross section at the valve seat, acts on the other side of the diaphragm. The diaphragm has large surface area exposed to secondary (outlet) pressure and is quite sensitive to its fluctuations. The movement of diaphragm regulates the pressure.

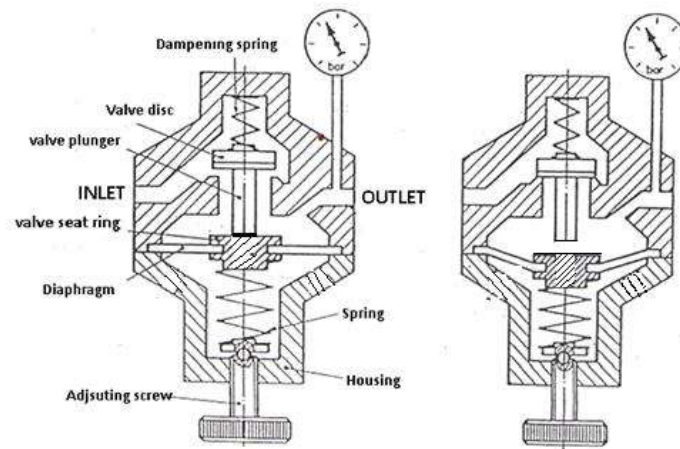


**Figure 1.4** venting type pressure regulator

#### 1.1.2.2 Non-Relieving or Non-Venting Type Pressure regulator

In this case compressed air cannot escape to the atmosphere in the event of high backpressure acting on the diaphragm, as there is no exit path provided in the diaphragm for the trapped air.

**Figure 1.5** shows the non-relieving venting type pressure regulator.



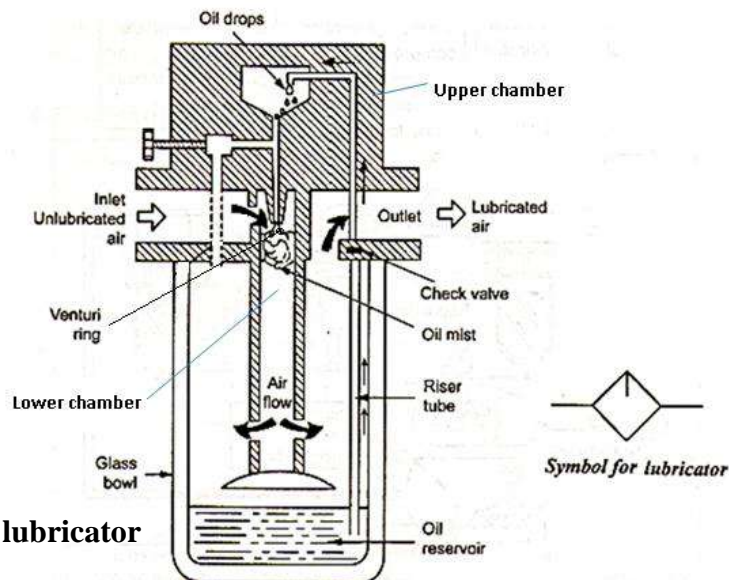
**Figure 1.5** Non-venting type pressure regulators

### 1.1.3 AIR LUBRICATOR

**Function:** The function of air lubricator is to add a controlled amount of oil with air to ensure proper lubrication of internal moving parts of pneumatic components. Lubricants are used to

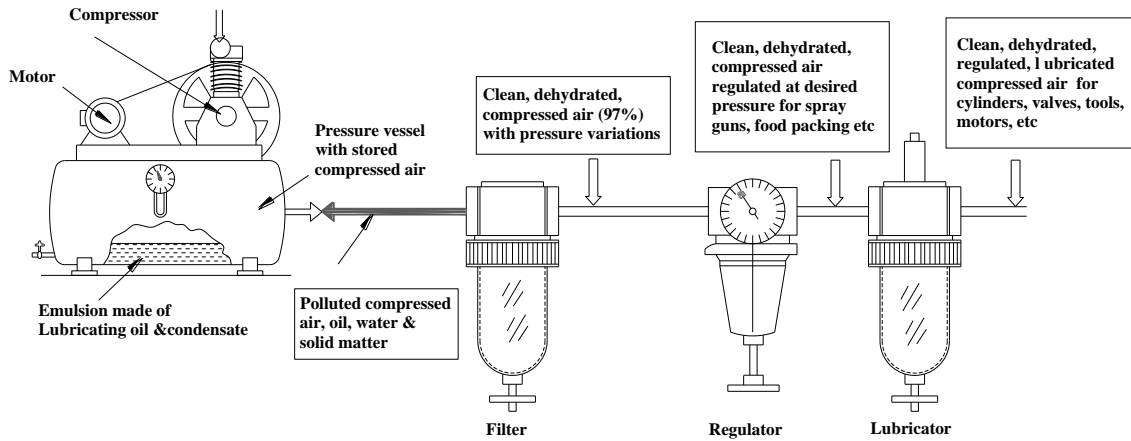
- To reduce the wear of the moving parts
- Reduce the frictional losses
- Protect the equipment from corrosion

**Operation:** The operation is similar to the principle of the carburettor. Schematic diagram is shown in **Figure 1.6**. As air enters the lubricator its velocity is increased by a venturi ring. The pressure at the venturi ring will be lower than the atmospheric pressure and the pressure on the oil is atmospheric. Due to this pressure difference between the upper chamber and lower chamber, oil will be drawn up in a riser tube. Oil droplets mix with the incoming air and form a fine mist. The needle valve is used to adjust the pressure differential between across the oil jet and hence the oil flow rate. The air – oil mixture is forced to swirl as it leaves the central cylinder so that large particles of oil go back to bowl and only the mist goes to outlet.



**Figure 1.6** Air lubricator

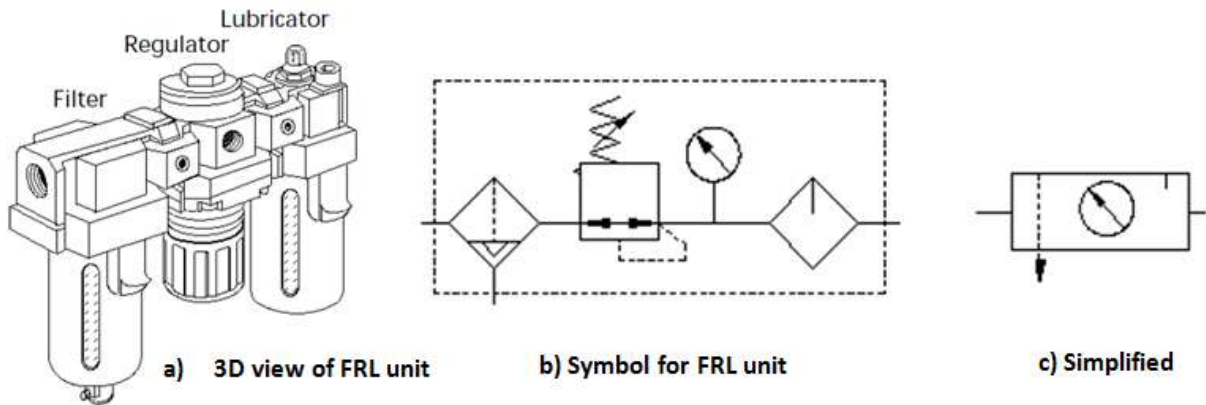
### 1.1.4 Filter Regulator Lubricator Unit (FRL Unit) /Service Unit



**Figure 1.7** Installation of FRL unit

In most pneumatic systems, the compressed air is first filtered and then regulated to the specific pressure and made to pass through a lubricator for lubricating the oil. Thus usually a filter, regulator and lubricator are placed in the inlet line to each air circuit. They may be installed as separate units, but more often they are used in the form of a combined unit. Figure 1.6 shows the schematic arrangement of installation of Filter, Regulator and Lubricator unit .

The combination of filter, regulator and lubricator is called FRL unit or service unit. Figure 1.7 (a) gives the three dimensional view of FRL unit. Figure 1.7(b) gives detailed symbol of FRL unit. Figure 1.7(c) gives simplified symbol of FRL unit.



**Figure 1.7** a) Three dimensional view of FRL unit b) detailed symbol c) simplified symbol of FRL

# **PNEUMATICS ACTUATORS**

## **1.1 PNEUMATICS ACTUATORS**

Pneumatic actuators are the devices used for converting pressure energy of compressed air into the mechanical energy to perform useful work. In other words, Actuators are used to perform the task of exerting the required force at the end of the stroke or used to create displacement by the movement of the piston. The pressurised air from the compressor is supplied to reservoir. The pressurised air from storage is supplied to pneumatic actuator to do work.

Their chief limitation is that the elastic nature of the compressed air makes them unsuitable for powering movement where absolutely steady forces or motions are required applied against a fluctuating load, or where extreme accuracy of feed is necessary. The air cylinder is also inherently limited in thrust output by the relatively low supply pressure so that production of high output forces can only be achieved by a large size of the cylinders.

## **1.2 TYPES OF PNEUMATICS ACTUATORS**

Pneumatic cylinders can be used to get linear, rotary and oscillatory motion. There are three types of pneumatic actuator: they are

- i) Linear Actuator or Pneumatic cylinders
- ii) Rotary Actuator or Air motors
- iii) Limited angle Actuators

### **1.2.1 Types of Pneumatic cylinders /Linear actuators**

Pneumatic cylinders are devices for converting the air pressure into linear mechanical force and motion. The pneumatic cylinders are basically used for single purpose application such as clamping, stamping, transferring, branching, allocating, ejecting, metering, tilting, bending, turning and many other applications.

The different classification scheme of the pneumatic cylinders are given below

#### **1. Based on application for which air cylinders are used**

- i) Light duty air cylinders
- ii) Medium duty air cylinders
- iii) Heavy duty air cylinders

#### **2. Based on the cylinder action**

- i) Single acting cylinder
- ii) Double acting cylinder
  - Single rod type double acting cylinder
  - Double rod type double acting cylinder

#### **3. Based on cylinder's movement**

- i) Rotating type air cylinder
- ii) Non rotating type air cylinder

#### 4. Based on the cylinder's design

- i) Telescopic cylinder
- ii) Tandem cylinder
- iii) Rod less cylinder
  - Cable cylinder,
  - Sealing band Cylinder with slotted cylinder barrel
  - Cylinder with Magnetically Coupled Slide
- iv) Impact cylinder
- v) Duplex cylinders
- vi) Cylinders with sensors

#### 1.2.1.1 Based on application for which air cylinders are used

Air cylinders can be classified according to their intended use, as light duty, medium duty or heavy duty types. In the main this merely governs the strength of the cylinder, and thus typical choice of material of construction and the form of construction. Comparison is given in [Table 1.1](#). It should be noted that classification by duty does not necessarily affect the output performance of the cylinder, as bore size for bore size; identical cylinder diameter will give the same thrust on the same line pressure, regardless of whether the cylinder is rated for light, medium or heavy duty. This form of rating, however, normally precludes the use of light classification for cylinders of large size (and thus high thrust); and medium classification for cylinders of even large size and very high thrust outputs.

**Table 1.1: Materials of construction for light, medium and heavy duty cylinders**

Components	Type of cylinder		
	Light duty	Medium duty	Heavy duty
Cylinder tubes	Hard drawn seamless aluminium or brass tubes Plastics	Hard drawn seamless brass tubes Aluminium, brass, iron or steel castings	Hard drawn seamless tubing, brass, bronze, iron or steel casting
End covers	Aluminium alloy castings Fabricated aluminium, brass, bronze	Aluminium brass, bronze, iron or steel castings, fabricated brass, bronze,	High tensile castings
Pistons	Aluminium alloy castings	Aluminium alloy castings, Brass, cast iron	Aluminium alloy castings, Brass, cast iron
Piston rods	EN 8 or similar steel ground and polished or chrome plated	EN 8 steel, ground and polished or chrome plated. Ground and polished stainless steel	Ground and polished stainless steel
Mounting brackets	Aluminium alloy casting	Aluminium, brass, iron castings	High tensile castings or fabricated



### 1.2.1.2 Based on the cylinder action

Based on cylinder action we can classify the cylinders as single acting and double acting. Single acting cylinders have single air inlet line. Double acting cylinders have two air inlet lines.

"

Advantages of double acting cylinders over single acting cylinders are

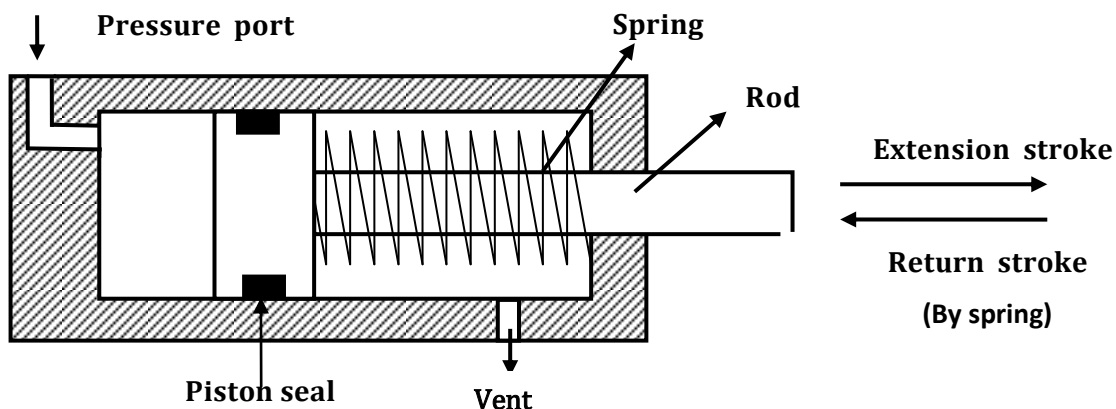
1. In single acting cylinder, compressed air is fed only on one side. Hence this cylinder can produce work only in one direction. But the compressed air moves the piston in two directions in double acting cylinder, so they work in both directions
2. In a single acting cylinder, the stroke length is limited by the compressed length of the spring. But in principle, the stroke length is unlimited in a double acting cylinder
3. While the piston moves forward in a single acting cylinder, air has to overcome the pressure of the spring and hence some power is lost before the actual stroke of the piston starts. But this problem is not present in a double acting cylinder.

#### A) Single acting cylinders.

Single acting cylinder has one working port. Forward motion of the piston is obtained by supplying compressed air to working port. Return motion of piston is obtained by spring placed on the rod side of the cylinder. Schematic diagram of single acting cylinder is shown in [Figure 1.1](#)

Single acting cylinders are used where force is required to be exerted only in one direction. Such as clamping, feeding, sorting, locking, ejecting, braking etc.,

Single acting cylinder is usually available in short stroke lengths [maximum length up to 80 mm] due to the natural length of the spring. Single Acting Cylinder exert force only in one direction. Single acting cylinders require only about half the air volume consumed by a double acting cylinder for one operating cycle.



[Figure 1.1](#) Construction features of single acting cylinder

There are varying designs of single acting cylinders including:

1. Diaphragm cylinder
2. Rolling diaphragm cylinder
3. Gravity return single acting cylinder
4. Spring return single acting cylinder

### i) Diaphragm cylinder

This is the simplest form of single acting cylinder. In diaphragm cylinder, piston is replaced by a diaphragm. The diaphragm is replaced by a diaphragm of hard rubber, plastic or metal clamped between the two halves of a metal casing expanded to form a wide, flat enclosure. Schematic diagram of diaphragm cylinder is shown in Figure 1.2. The operating stem which takes place of the piston rod in diaphragm cylinder can also be designed as a surface element so as to act directly as a clamping surface for example. Only short operating strokes can be executed by a diaphragm cylinder, up to a maximum of 50 mm. This makes the diaphragm type of cylinder particularly adaptable to clamping operations. Return stroke is accomplished by a spring built into the assembly or by the tension of diaphragm itself in the case of very short stroke.

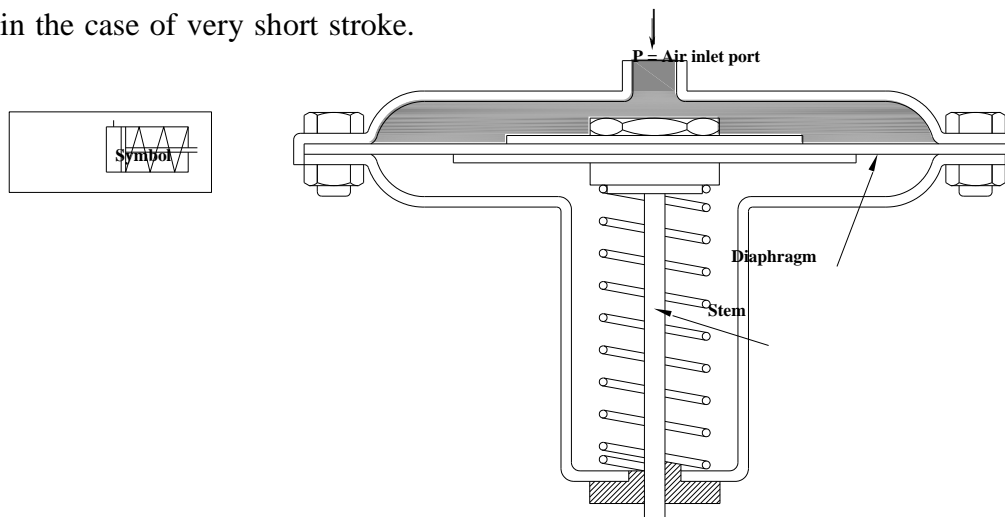
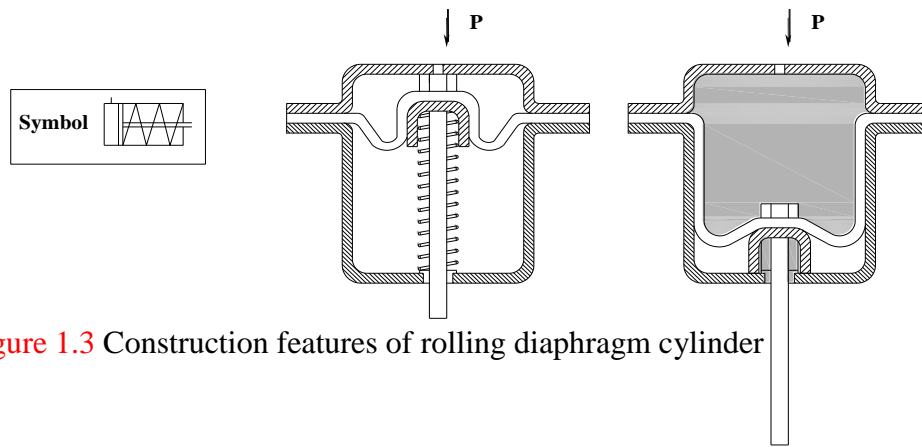


Figure 1.2 Construction features of diaphragm cylinder

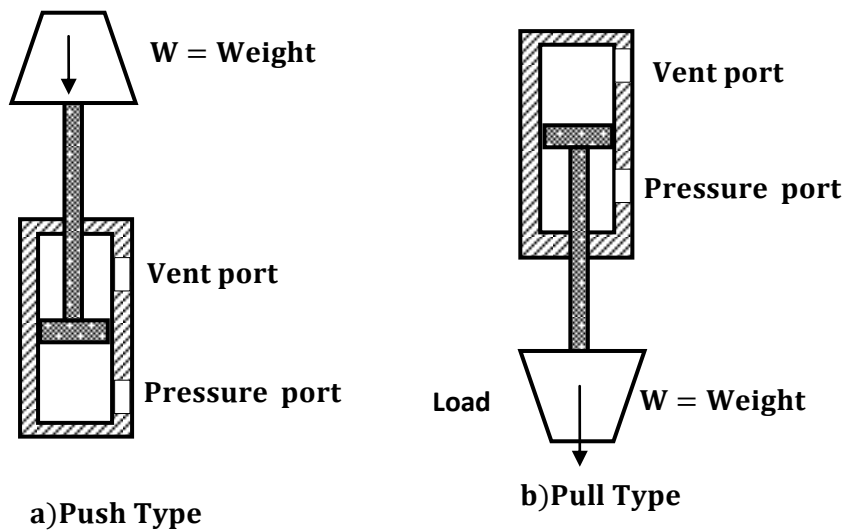
### ii) Rolling diaphragm cylinder

They are similar to diaphragm cylinders. Schematic diagram of Rolling diaphragm cylinder is shown in Figure 1.3. They too contain a diaphragm instead of piston, which in this instance rolls out along the inner walls of the cylinder when air pressure is applied to the device, thereby causing the operating stem to move outwards. Compared with the standard diaphragm type, a rolling diaphragm cylinder is capable of executing appreciably longer operating strokes (averaging from 50 mm to 800mm).



**Figure 1.3** Construction features of rolling diaphragm cylinder

**iii) Gravity Return Single Acting Cylinder**



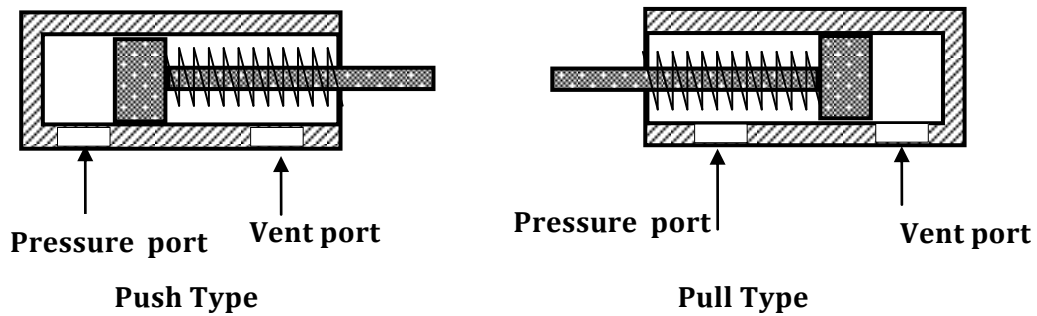
**Figure 1.4** Gravity return Single Acting Cylinder

Figure 1.4 shows gravity return type single acting cylinders. In a push type (a), the cylinder extends to lift a weight against the force of gravity by applying oil pressure at the blank end. The oil is passed through blank end port or pressure port. The rod end port or vent port is open to atmosphere so that air can flow freely in and out of the rod end of the cylinder. To retract the cylinder, the pressure is simply removed from the piston by connecting the pressure port to the tank. This allows the weight of the load to push the fluid out of the cylinder back to tank.

**iv) Spring Return Single Acting Cylinder**

Spring return single acting cylinder is shown in Figure 1.5 in part (a) push type the pressure is sent through pressure port situated at blank end of the cylinder. When the pressure is released, the spring automatically returns the cylinder to the fully retracted position. The vent port is open to atmosphere so that air can flow freely in and out of the rod end of the cylinder.

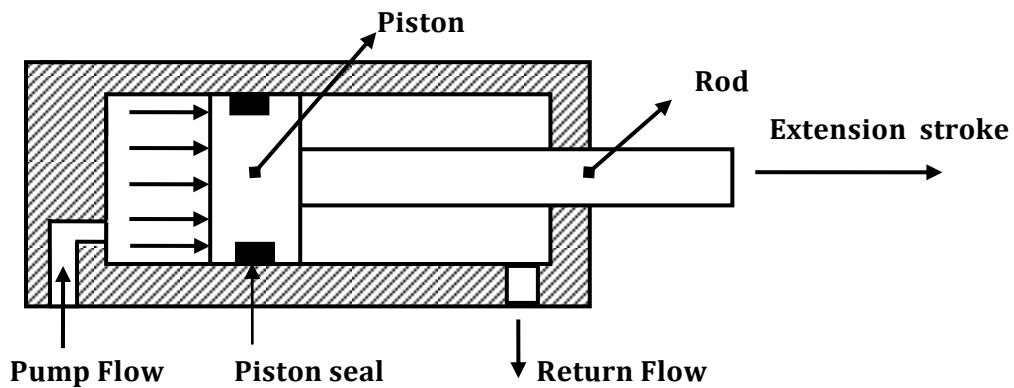
Part (b) shows a spring return single acting cylinder. In this design cylinder retracts when the pressure port is connected to the pump flow and extend whenever the pressure port is connected to the tank. Here pressure port is situated at rod end of the cylinder.



**Figure 1.5 Push and Pull type Single Acting Cylinder**

**B) Double acting cylinders.**

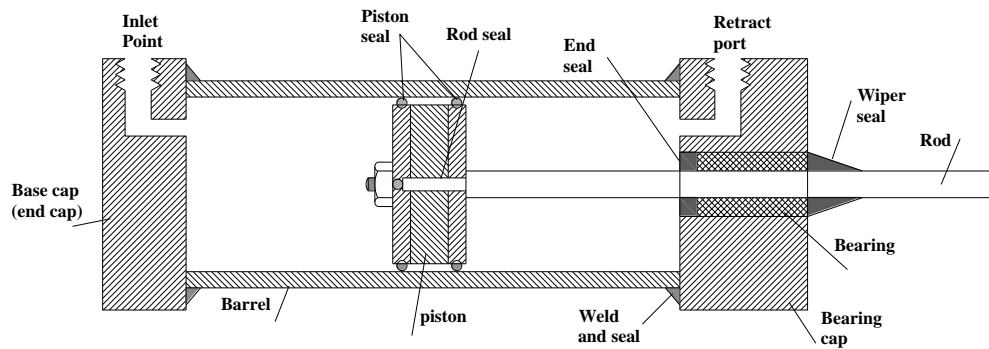
Schematic diagram of double acting cylinder is shown in Figure 1.6. Double Acting Cylinders are equipped with two working ports- one on the piston side and the other on the rod side. To achieve forward motion of the cylinder, compressed air is admitted on the piston side and the rod side is connected to exhaust. During return motion supply air admitted at the rod side while the piston side volume is connected to the exhaust. Force is exerted by the piston both during forward and return motion of cylinder. Double acting cylinders are available in diameters from few mm to around 300 mm and stroke lengths of few mm up to 2 meters



**Figure 1.6 Double acting cylinder**

**Construction of Double acting cylinder**

The construction features of double acting cylinder are shown in Figure 1.7. The construction of double acting cylinder is similar to that of a single cylinder. However, there is no return spring. In double acting cylinder, air pressure can be applied to either side ( supply and exhaust) of the piston, thereby providing a pneumatic force in both directions. The double acting cylinders are mostly commonly used in the application where larger stroke length is required.



**Figure 1.7** Construction features of double acting cylinder

The seven parts of the double acting cylinder are

1. Base cap with port connection
2. Bearing cap with port connection
3. Cylinder barrel
4. Piston
5. Piston rod
6. Scraper rings
7. Seals

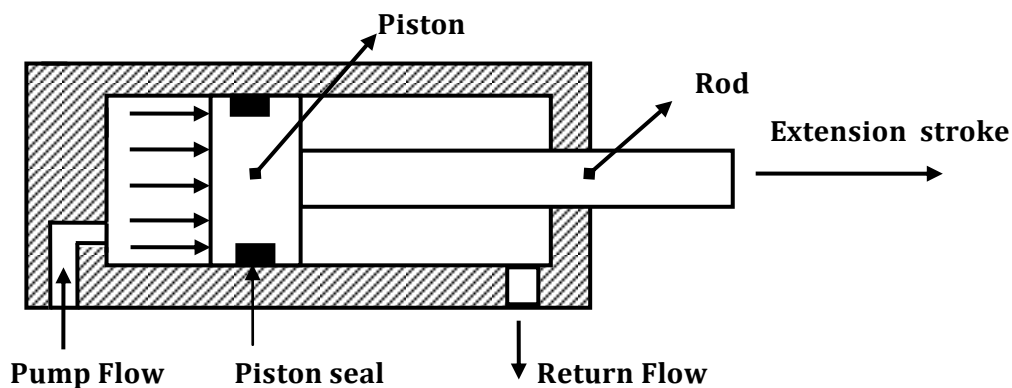
### Construction of Double acting cylinder

There are two types of double acting cylinders.

- i) Double acting cylinder with piston rod on one side.
- ii) Double acting cylinder with piston rod on both sides

#### i) Double acting cylinder with piston rod on one side.

Figure 1.8 shows the operation of a double acting cylinder with piston rod on one side. To extend the cylinder, pump flow is sent to the blank end port as in Figure 1.8 (a). Fluid from the rod end port returns to the reservoir. To retract the cylinder, the pump flow is sent to the rod end port and fluid from the blank end port returns to the tank as in Figure 1.8 (b).



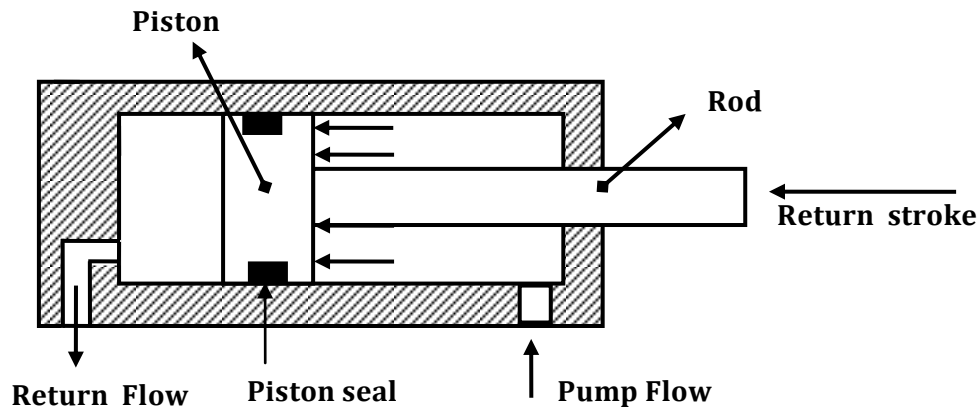


Figure 1.8 Double Acting Cylinder with piston rod on one side

ii) Double acting cylinder with piston rod on both sides

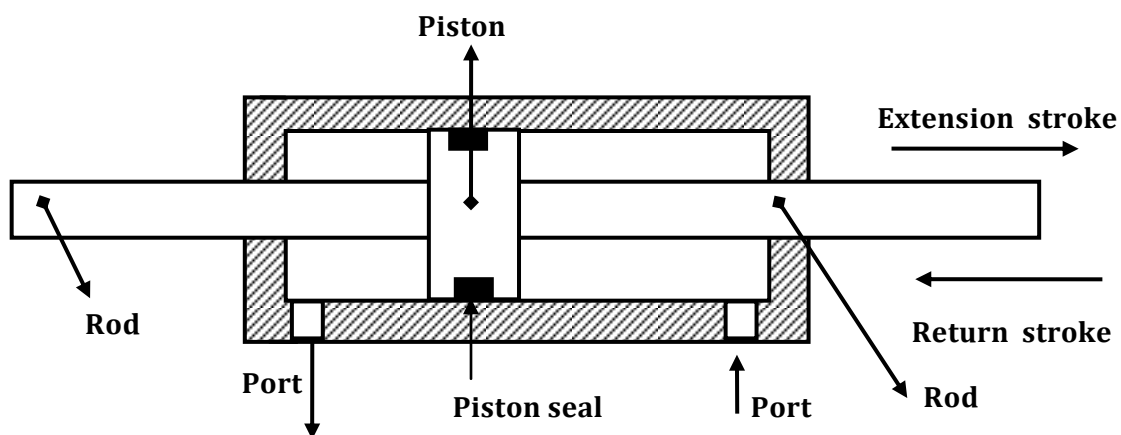
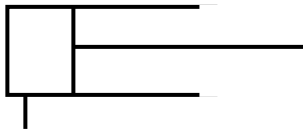
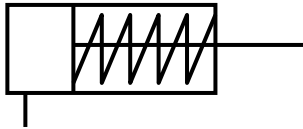
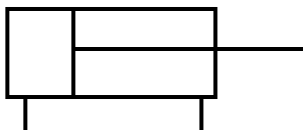
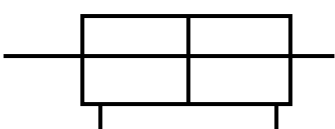
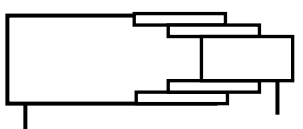
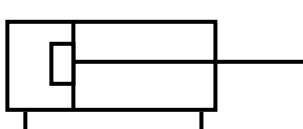
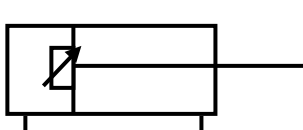
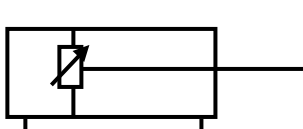


Figure 1.9 Double Acting Cylinder with piston rod on both side

A double acting cylinder with piston rod on both sides (Figure 1.9) is a cylinder with rod extending from both ends. This cylinder can be used in an application where work can be done by both ends of the cylinder, thereby making the cylinder more productive. Double rod cylinders can withstand higher side loads because they have an extra bearing one on each rod to withstand the loading. Double rod cylinders are used when there is bending load and accurate alignment and maximum strength is required. A further advantage is that rod is precisely located and may be used to guide the machine member coupled to it, dispensing with external guides or bearing in many cases, most standard production models are available either in single rod or double rod configuration A disadvantage of double rod configuration is that there is a reduction in maximum thrust due to the blanking effect of the rod cross section on the piston area and a slightly larger size of cylinder is required for a given duty. The thus will be the same on the ingoing stroke as that of a single rod double acting cylinder.

Table 1.3 Graphical symbols of cylinders

Sl No	Graphical Symbols	Explanation
1		<b>Single acting cylinder with unspecified return:</b> Air pushes the piston in one direction and the piston is return is unspecified. External dock or lever may push
2		<b>Single acting cylinder with spring return.</b> Air pushes the piston in one direction and piston returns by spring on rod side
3		<b>Double acting cylinder –single piston rod:</b> the force exerted by compressed air moves the piston in both direction.
4.		<b>Double acting cylinder –double piston rod</b> It has piston rods extending from both ends of the cylinder. It produces equal force and speed on both sides of the cylinder
1.		<b>Telescopic cylinder –double acting</b> is used where space is constraint. It is used for long stroke application like in pneumatic cranes, dump trucks, lift fork trucks, dipper wagon
7.		<b>Double acting cylinder – fixed cushion on one side,</b> Cushioning is used in the end position to prevent sudden impact which otherwise may damage parts.
8		<b>Double acting cylinder – variable cushion on one side–</b> fixed cushion on one side, cushioning is variable in one direction by adjusting the orifice opening.
9		<b>Double acting cylinder – variable cushion on both sides–</b> fixed cushion on one side, cushioning is variable in both direction.

## **PNEUMATIC CONTROL VALVES**

### **1.1 VALVES**

Valves are defined as devices to control or regulate the commencement, termination and direction and also the pressure or rate of flow of a fluid under pressure which is delivered by a compressor or vacuum pump or is stored in a vessel.

Valves of one sort or another, perform three main functions in pneumatic installation

- They control the supply of air to power units, example cylinders
- They provide signal which govern the sequence of operation
- They act as interlock and safety devices

Valves available for pneumatic control can be classified into four principal groups according to their function:

1. Direction control valve
2. Non return valves
3. Flow control valves
4. Pressure control valves

### **1.2 DIRECTION CONTROL VALVES**

Pneumatic systems like hydraulic systems also require control valves to direct and regulate the flow of fluid from the compressor to the various devices like air actuators and air motors. In order to control the movement of air actuators, compressed air has to be regulated, controlled and reversed with a predetermined sequence. Pressure and flow rates of the compressed air to be controlled to obtain the desired level of force and speed of air actuators.

The function of a directional control valve is to control the direction of flow in the pneumatic circuit. DCVs are used to start, stop and regulate the direction of air flow and to help in the distribution of air in the required line.

#### **6.2.1 TYPES OF DIRECTION CONTROL VALVES**

Directional valves control the way the air passes and are used principally for controlling commencement, termination and direction of air flow. The different classification schemes of the pneumatic cylinders are given below

##### **1. Based on construction**

###### **i) Poppet or seat valves**

- Ball seat valve
- Disc seat valve
- Diaphragm Valves

###### **ii) Sliding spool valves**

- Longitudinal slide valve
- Suspended spool valves
- Rotary spool valves



2. Based on the Number of ports

- i) Two way valves
- ii) Three way valves
- iii) Four way valves

3. Based on methods of actuation

- i) Mechanical
- ii) Electrical
- iii) Pneumatic

4. Based on Size of the port

Size refers to a valve's port size. The port sizes are designated as M5, G1/8, and G1/4 etc. M refer to Metric thread, G refer to British standard pipe (BSP) thread.

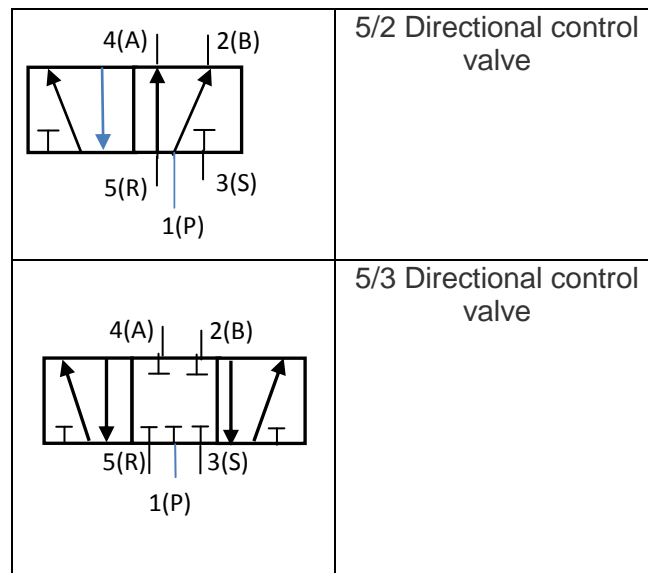
5. Based on mounting styles

- i) Sub base
- ii) Manifold
- iii) In-line
- iv) Valve island

**6.2.1.1 ISO DESIGNATION OF DIRECTION CONTROL VALVES**

**Table 1.2:** Port designation of DCV

Port and position	
	<p>2/2 Directional control valve</p>
	<p>3/2 Directional control valve (normally closed)</p>
	<p>3/2 Directional control valve (normally open)</p>
	<p>4/2 Directional control valve</p>



### 1.2.1.2 POPPET DIRECTION CONTROL VALVES

There are two different types of poppet valves, namely ball seat valve and disc seat valve.

#### A. Ball seat valve.

In a poppet valve, discs, cones or balls are used to control flow. Figure 1.1 shows the construction of a simple 2/2 normally closed valve. If the push button is pressed, ball will lift off from its seat and allows the air to flow from port P to port B. When the push button is released, spring force and air pressure keeps the ball back and closes air flow from port P to port B. Valve position are shown in Figure 1.1(a) 1.1 (b) 1.1(C)

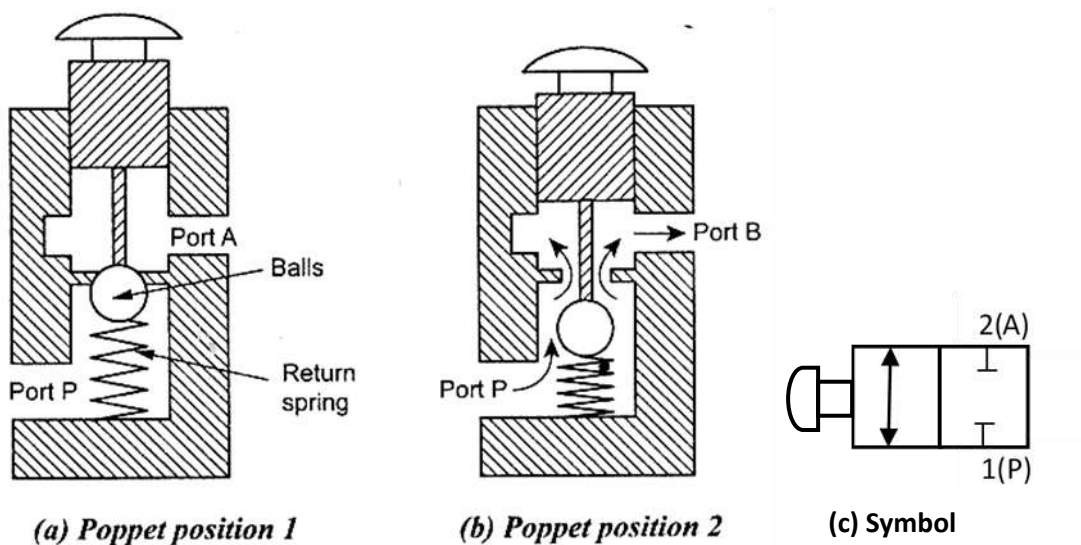
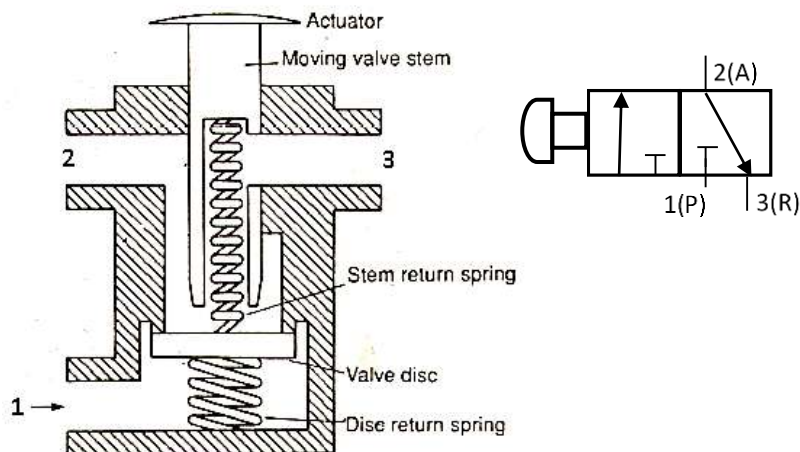


Figure 1.1 Two/Two Ball seat Poppet valve

### B. Disc seat poppet valve

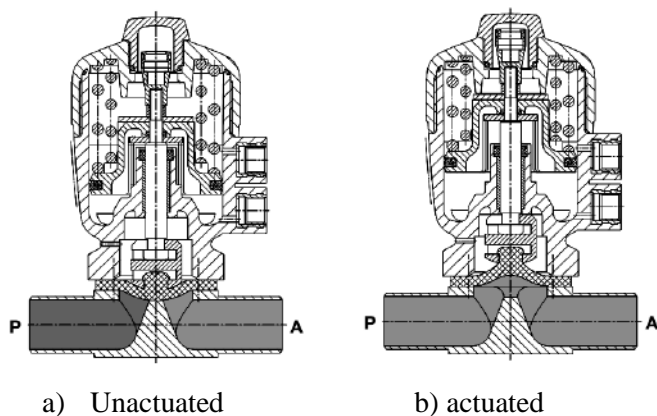
Figure 1.2 shows the construction of a disc type 3/2 way DCV. When push button is released, ports 1 and 3 are connected via hollow pushbutton stem. If the push button is pressed, port 3 is first blocked by the moving valve stem and then valve disc is pushed down so as to open the valve thus connecting port 1 and 3. When the push button is released, spring and air pressure from port 1 closes the valve.. Comparison between Ball seat and disc seat valve is given in [Table 1.3](#)



**Figure 1.2** Disc seat poppet valve

### C. Diaphragm valves

The diaphragm between the actuator and valve body hermetically isolates the fluid from the actuator. The valves are maintenance-free and extremely robust and can be retrofitted with a comprehensive range of accessories, e.g. electrical position feedback, stroke limitation or manual override. [Figure 1.4](#) shows unactuated and actuated position of diaphragm valves.



**Figure 1.4** Diaphragm valve: unactuated position, actuated position

### 6.2.1.3 SPOOL DIRECTION CONTROL VALVES

#### A. Hand operated 3/2 DCV

The cross sectional views of 3/2 DCV (normally closed) based on spool design is shown below. When the valve is not actuated, port 2 and 3 are connected and port 1 is blocked. When the valve is actuated then port 2 and 1 are connected and port 3 is blocked.

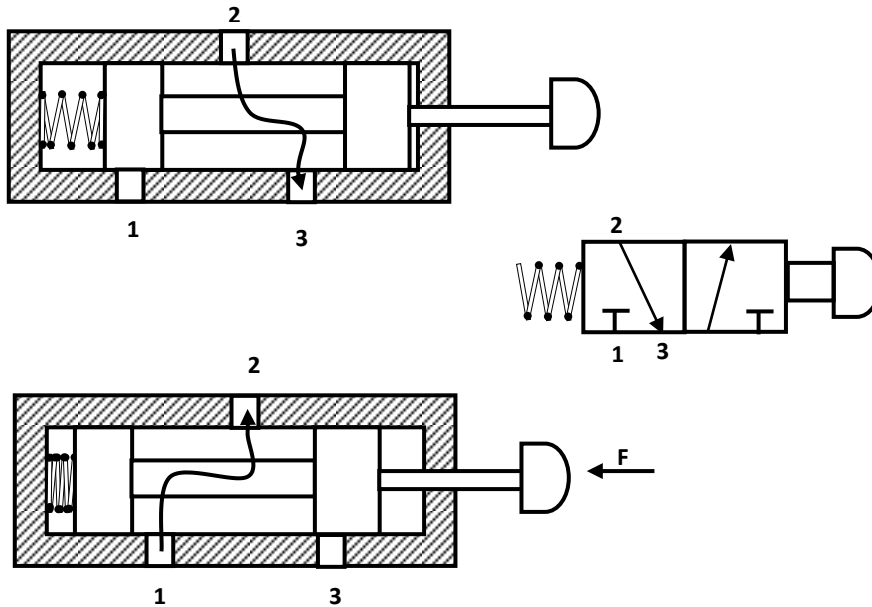


Figure 1.5 3/2 Directional control valve (Normally closed)

#### B. Pneumatically actuated 3/2 DCV

The cross – sectional views of pneumatically actuated NC type 3/2 DCV in normal position and actuated positions are shown in the Figure 1.7

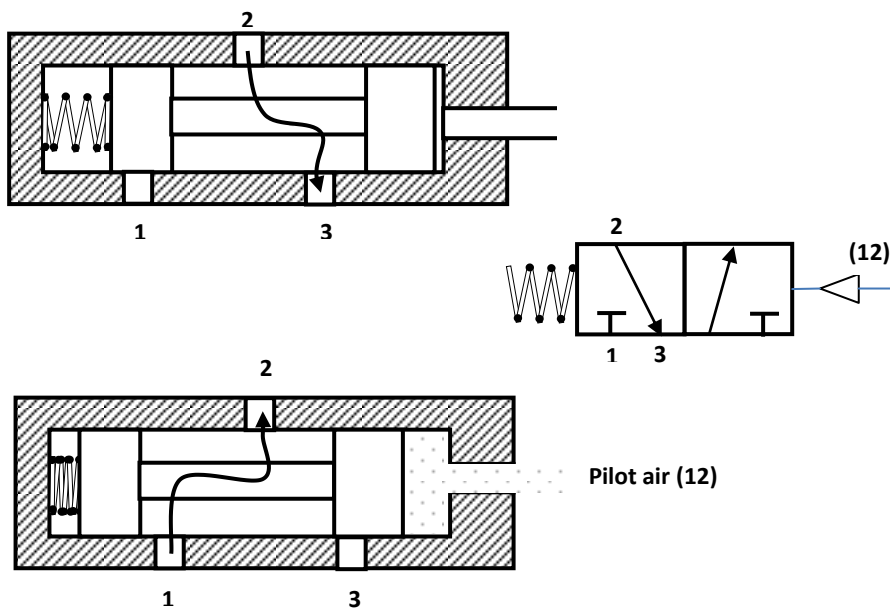
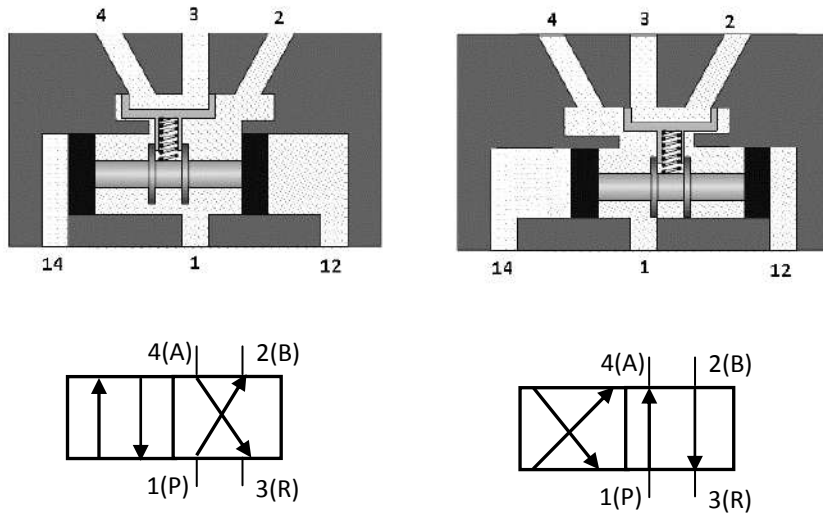


Figure 1.7 3/2 Directional control valve (pneumatically operated)

### C. Pneumatically actuated 4/2 DCV

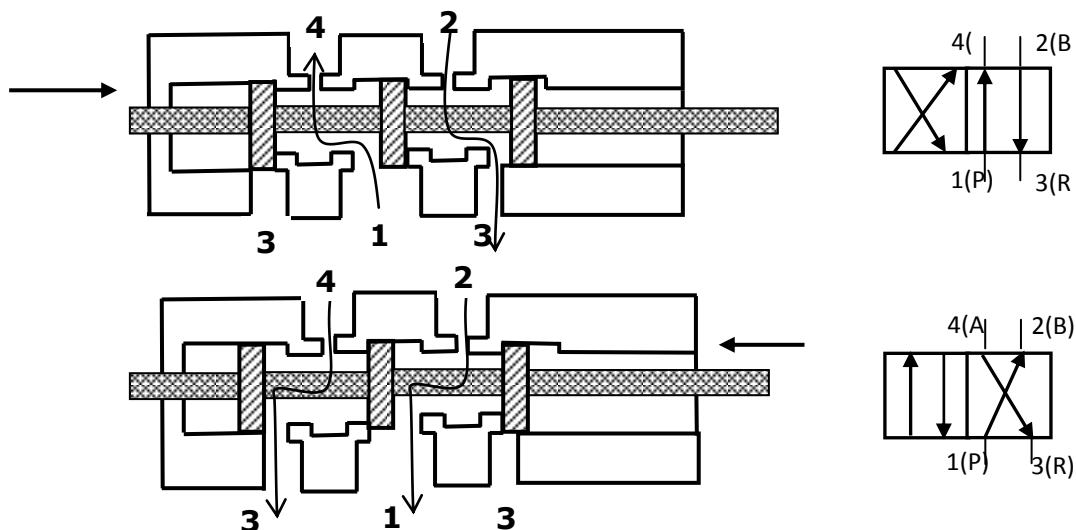
The valve shown in **Figure 1.9** is a 4/2 way valve pneumatically operated DCV. Switch over is effected by direct application of pressure. If compressed air is applied to pilot spool through control port 12, it connects port 1 with 2 and 4 is exhausted through port 3. If the pilot pressure is applied to port 14, then 1 is connected with 4 and line 2 exhausted through port 3. On disconnecting the compressed air from the control line, the pilot spool remains in its current position until spool receives a signal from the other control side.



**Figure 1.9** Schematic diagram of 4/2 way valve

### D. Suspended Disc Direction Control Valves

This valve is quite similar to 4/2 way spool valve. Schematic diagram is shown in **Figure 1.11**. In this design disc is used instead of a spool. This suspended disc can be moved by pilot pressure or by solenoid or by mechanical means. In this design, main disc connects port 1 to either port 4 or 2. The secondary seat discs seal the exhaust port 3 whichever is not functional. These valves are generally provided with manual override to manually move the cylinder.



**Figure 1.11** 4/2 Directional control valve (suspended disc type)

## 1.2.2 NON RETURN VALVES

Non return valves permit flow of air in one direction only, the other direction through the valve being at all times blocked to the air flow. Mostly the valves are designed so that the check is additionally loaded by the downstream air pressure, thus supporting the non-return action.

Among the various types of non-return valves available, those preferentially employed in pneumatic controls are as follows

- i) Check valve
- ii) Shuttle valve
- iii) Restrictor check valve
- iv) Quick exhaust valve
- v) Two pressure valve

### A. Check valve

The simplest type of non-return valve is the check valve (Figure 1.35 (a)), which completely blocks air flow in one direction while permitting flow in the opposite direction with minimum pressure loss across the valve. As soon as the inlet pressure in the direction of free flow develops a force greater than that of the internal spring, the check is lifted clear of the valve seat. The check in such valve may be plug, ball, plate or diaphragm.

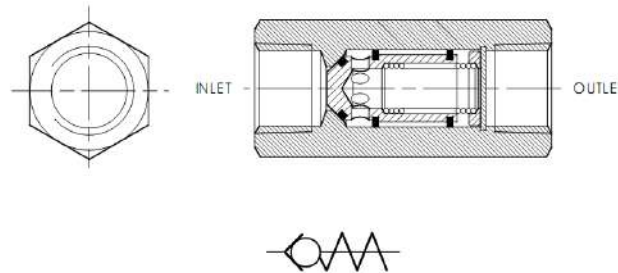


Figure 1.35 Check valve

### B. Shuttle valve

It is also known as a double control valve or double check valve. A shuttle valve has two inlets and one outlet. At any one time, flow is shut off in the direction of whichever inlet is unloaded and is open from the loaded inlet to the outlet (Figure 1.36). A shuttle valve may be installed, for example, when a power unit (cylinder) or control unit (valve) is to be actuated from two points, which may be remote from one other.

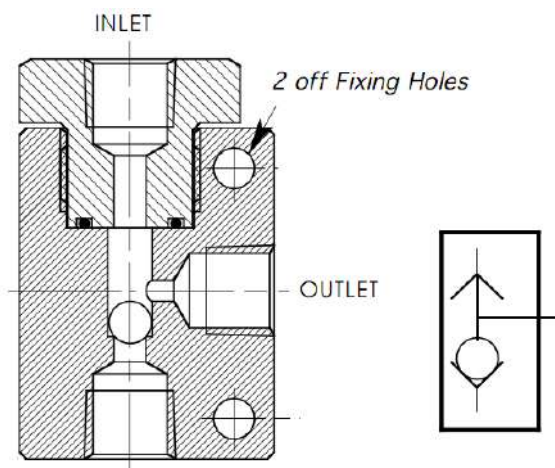


Figure 1.36 Shuttle valve

### C. Restrictor check valve

It also termed speed control valve for pneumatic applications are actually hybrid type of unit. By reason of their throttling function they are flow control valves and they are indeed used as flow control valves in pneumatics. Incorporation of check function also makes them non –return valves and it is as such that they are generally classified.

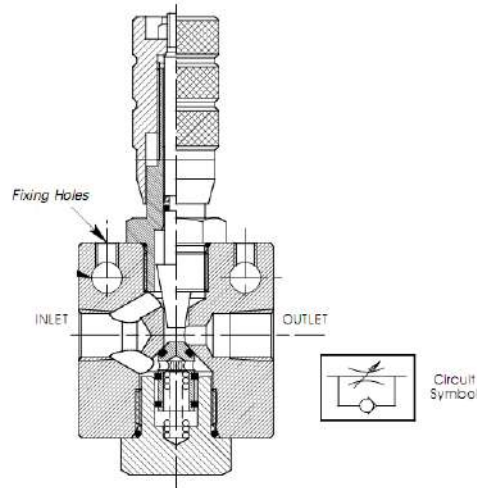


Figure 1.37 Functional diagram of restrictor check valve.

### D. Quick Exhaust Valves

A quick exhaust valve is a typical shuttle valve. The quick exhaust valve is used to exhaust the cylinder air quickly to atmosphere. Schematic diagram of quick exhaust valve is shown in Figure 1.38. In many applications especially with single acting cylinders, it is a common practice to increase the piston speed during retraction of the cylinder to save the cycle time. The higher speed of the piston is possible by reducing the resistance to flow of the exhausting air during the motion of cylinder. The resistance can be reduced by expelling the exhausting air to the atmosphere quickly by using Quick exhaust valve.

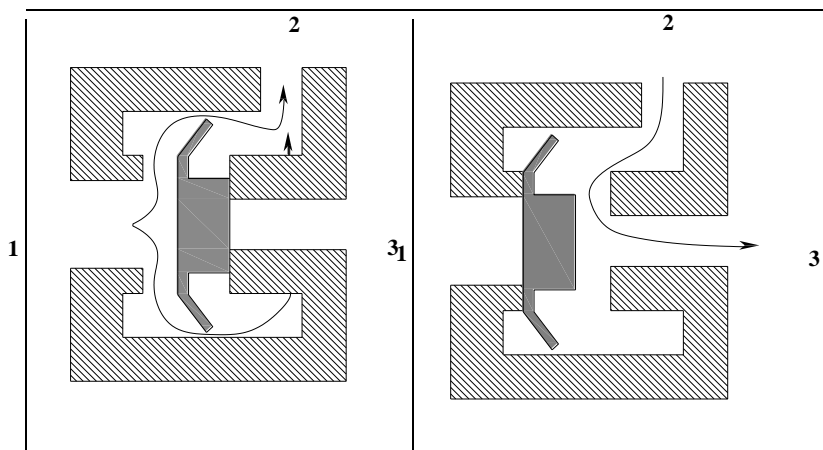


Figure 1.38 Functional diagram of quick exhaust valve.

### E. Two Pressure Valve

This valve is the pneumatic AND valve. It is also derivate of Non Return Valve. A two pressure valve requires two pressurised inputs to allow an output from itself. The cross sectional views of two pressure valve in two positions are given in Figure 1.40 As shown in the figure, this valve has two inputs 12 and 14 and one output 2. If the compressed air is applied to either 12 or input 14, the spool moves to block the flow, and no signal appears at output 2. If signals are applied to both the inputs 12 and 14, the compressed air flows through the valve, and the signal appears at output 2.

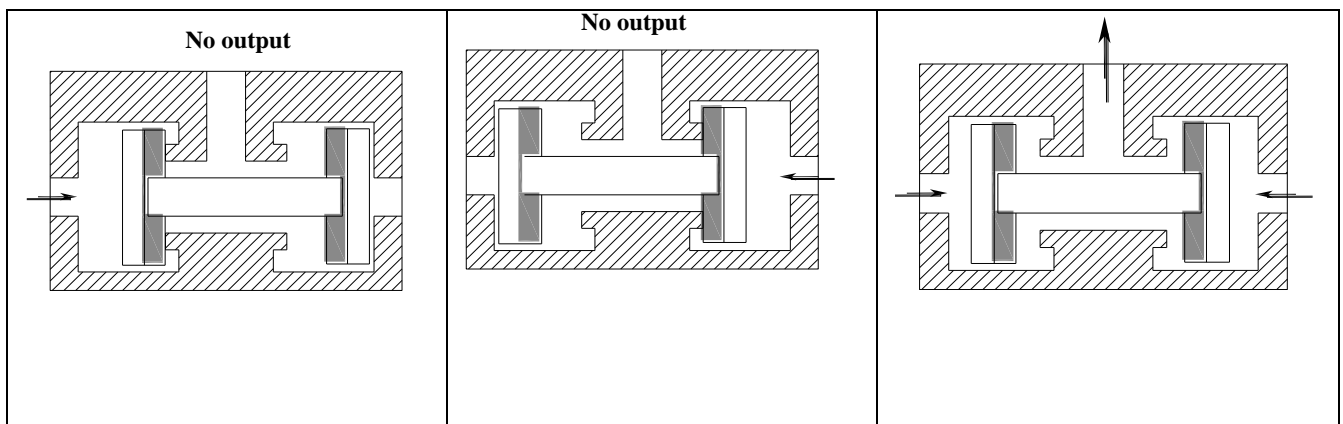


Figure 1.40 Two pressure valve.

### 1.2.3 FLOW CONTROL VALVES

Function of a flow control valve is self-evident from its name. A flow control valve regulates the rate of air flow. The control action is limited to the air flow passing through the valve when it is open, maintaining a set volume per unit of time. Figure 1.41(a) shows a variable restrictor type flow control valve (manifold type). Figure 1.41(b) shows a variable restriction type flow control valve (inline type). Figure 1.42 shows another design of Flow control valve, in which flow can be set by turning the knob.

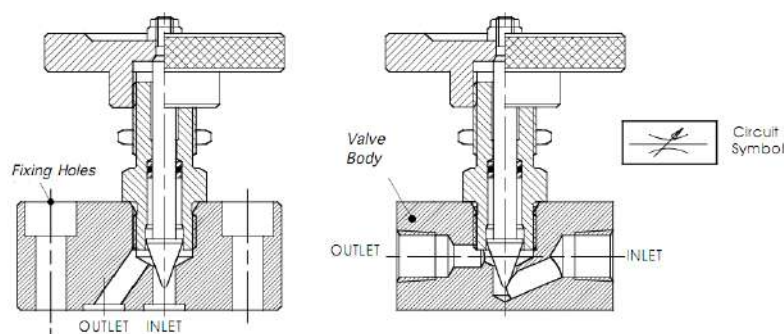
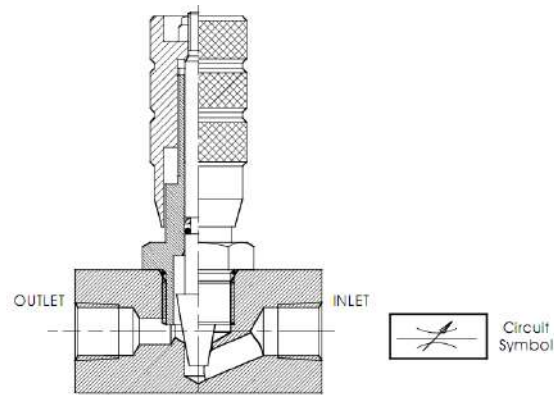


Figure 1.41 Flow control valve a) manifold b) inline





**Figure 1.42** Flow control valve (adjustable)

### 1.2.4 PRESSURE CONTROL VALVE.

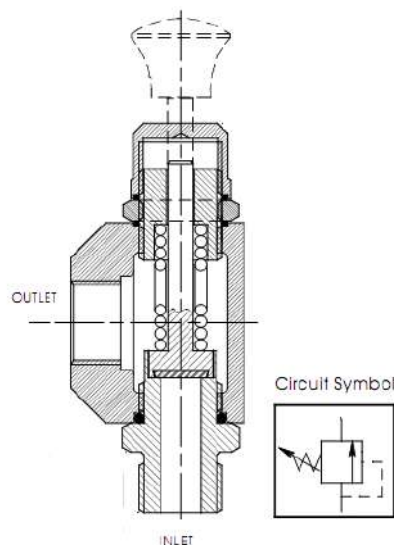
Compared with hydraulic systems, few pressure control valves are brought into use in pneumatics. Pressure control valves control the pressure of the air flowing through the valve or confined in the system controlled by the valve.

There are three types of pressure control valves

1. Pressure limiting valve
2. Pressure sequence valve
3. Pressure regulator or pressure reducing valve

#### **A. Pressure limiting valve.**

Prevents the pressure in a system from rising above a permissible maximum. Construction feature of pressure limiting valve is shown in **Figure 1.43**. It is a standard feature of compressed air production plant but is hardly ever used in pneumatic controls. These valves perform a safety relief function by opening to the atmosphere if a predetermined pressure is exceeded in the system, thus releasing the excess pressure. As soon as the pressure is thus relieved to the desired figure, the valve closed again by spring force.



**Figure 1.43** Pressure limiting valve

### **B. Pressure sequence valve**

Function of the sequence valve is very similar to that of a pressure limiting valve. It is however used for a different purpose. Outlet of the pressure sequence valve remains closed until pressure upstream of it builds up to a predetermined value. Only then the valve opens to permit the air from inlet to outlet. Sequence valve must be incorporated into a pneumatic control where a certain minimum pressure must be available for a given function and operation is not be initiated at any pressure lower than that. There are also used in systems containing priority air consumers, when other consumers are not to be supplied with air until ample pressure is assured.

### **C. Pressure reducing valve or regulator**

Pressure regulators, commonly called pressure-reducing valves, maintain constant output pressure in compressed-air systems regardless of variations in input pressure or output flow. Regulators are a special class of valve containing integral loading, sensing, actuating, and control components. Available in many configurations, they can be broadly classified as general purpose, special purpose, or precision.

## UNIT-6

### SINGLE ACTUATOR CIRCUITS

#### 1.1 Pneumatic circuit and pneumatic circuit diagram.

Pneumatic control systems can be designed in the form of pneumatic circuits. A pneumatic circuit is formed by various pneumatic components, such as cylinders, directional control valves, flow control valves, pressure regulator, signal processing elements such as shuttle valve, two pressure valve etc.

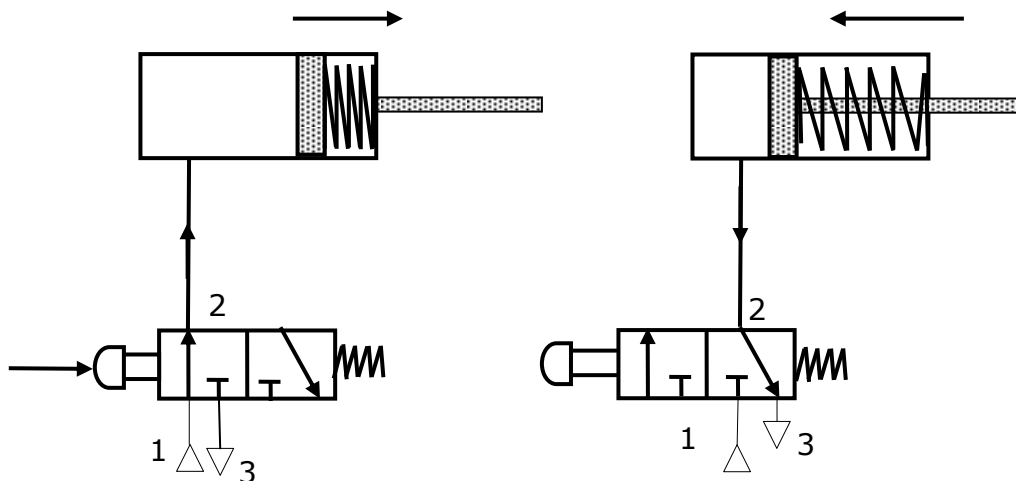
Pneumatic circuits have the following functions

- To control the entry and exit of compressed air in the cylinders.
- To use one valve to control another valve
- To control actuators or any other pneumatic devices

A pneumatic circuit diagram uses pneumatic symbols to describe its design. Some basic rules must be followed when drawing pneumatic diagrams.

#### 1.2 SINGLE ACTING CYLINDER CONTROL

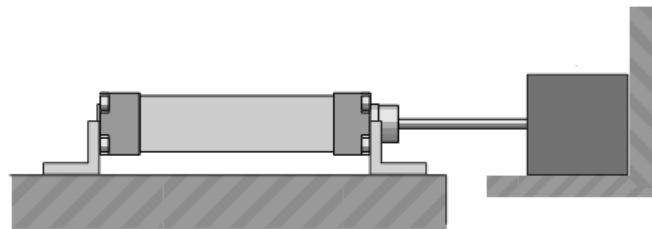
##### 1.2.1 DIRECT CONTROL OF SINGLE ACTING CYLINDER.



**Figure 1.1 Direct control of a single acting cylinder**

Pneumatic cylinders can be directly controlled by actuation of final directional control valve (Figure 1.1 ). These valves can be controlled manually or electrically. This circuit can be used for small cylinders as well as cylinders which operates at low speeds where the flow rate requirements are less. When the directional control valve is actuated by push button, the valve switches over to the open position, communicating working source to the cylinder volume. This results in the forward motion of the piston. When the push button is released, the reset spring of the valve restores the valve to the initial position [closed]. The cylinder space is connected to exhaust port there by piston retracts either due to spring or supply pressure applied from the other port.

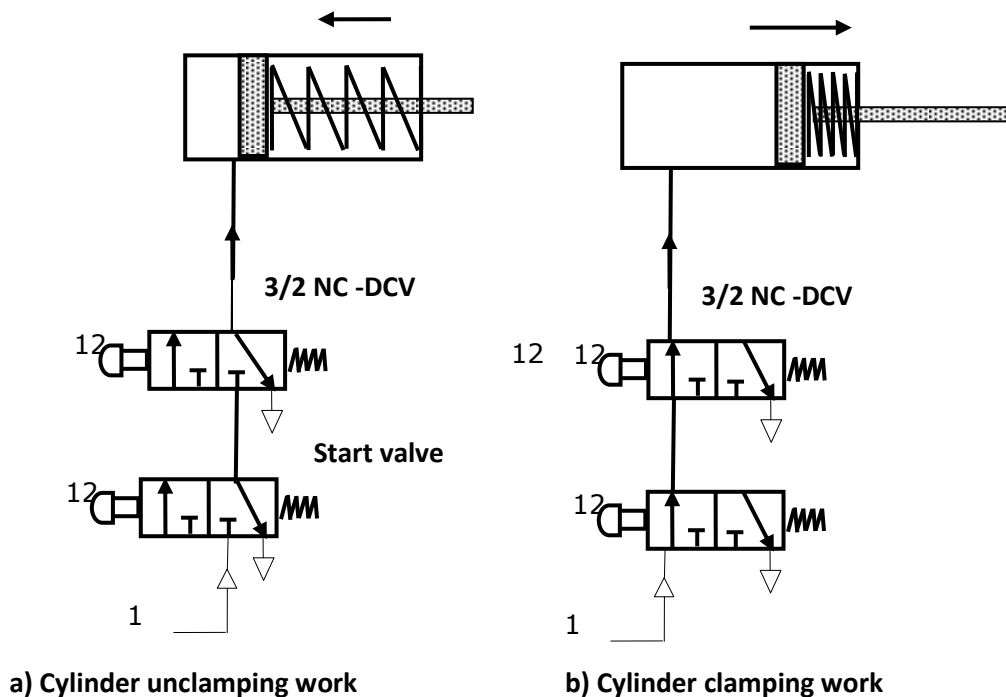
**Example 1:** A small single acting cylinder is to extend and clamp a work piece when a push button is pressed. As long as the push button is activated, the cylinder should remain in the clamped position. If the push button is released, the clamp is to retract. Use additional start button. Schematic diagram of the setup is shown in **Figure 1.2**



**Figure 1.2**

**Solution**

The control valve used for the single acting cylinder is the 3/2 way valve. In this case, since the cylinder is of small capacity, the operation can be directly controlled by a push button 3/2 way directional control valve with spring return.



**Figure 1.3**

When start button and 3/2 NC valve is operated, cylinder moves forward to clamp the work piece. When start button and 3/2 way valve is released cylinder comes back to the retracted position as shown in **Figure 1.3**

## 1.2.2 INDIRECT CONTROL OF SINGLE ACTING CYLINDER

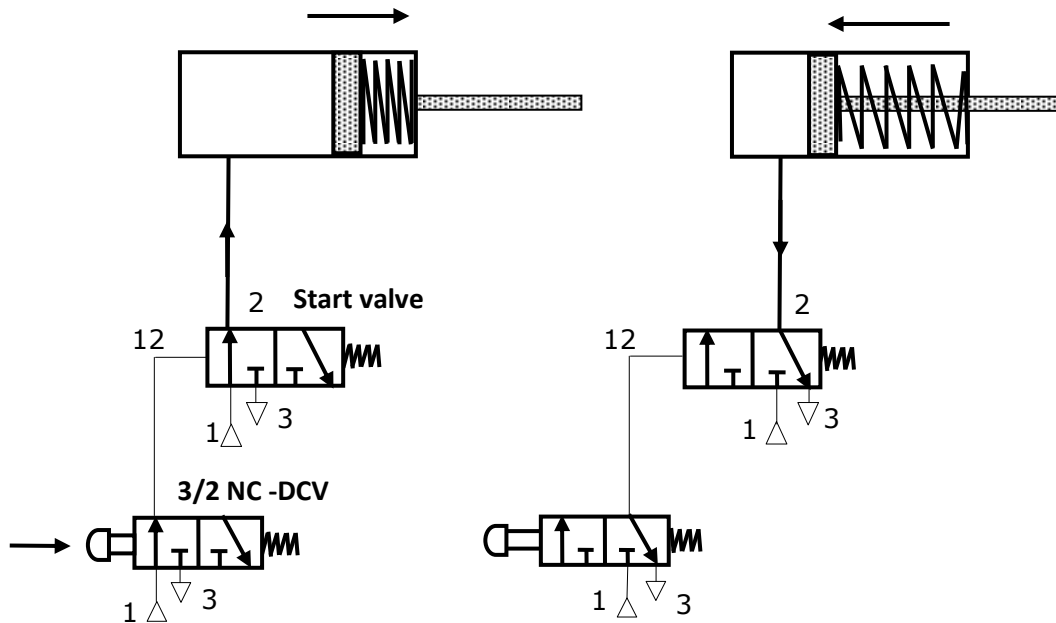


Figure 1.4 Indirect control of a single acting cylinder

This type of circuit (Figure 1.4) is suitable for large single cylinders as well as cylinders operating at high speeds. The final pilot control valve is actuated by normally closed 3/2 push button operated valve. The final control valves handle large quantity of air. When the push button is pressed, 3/2 normally closed valve generate a pilot signal 12 which controls the final valve thereby connecting the working medium to piston side of the cylinder so as to advance the cylinder. When the push button is released, pilot air from final valve is vented to atmosphere through 3/2 NC – DCV.

**Example 2:** A large single acting cylinder is to extend and clamp a work piece when a push button is pressed. As long as the push button is activated, the cylinder should remain in the clamped position. If the push button is released, the clamp is to retract. Use additional start button.

The control valve used for the single acting cylinder is the 3/2 way valve. In this case, since the cylinder is of large capacity, the operation cannot be directly controlled by a push button 3/2 way directional control valve with spring return. Indirect control is to be used as shown in the Figure 1.5

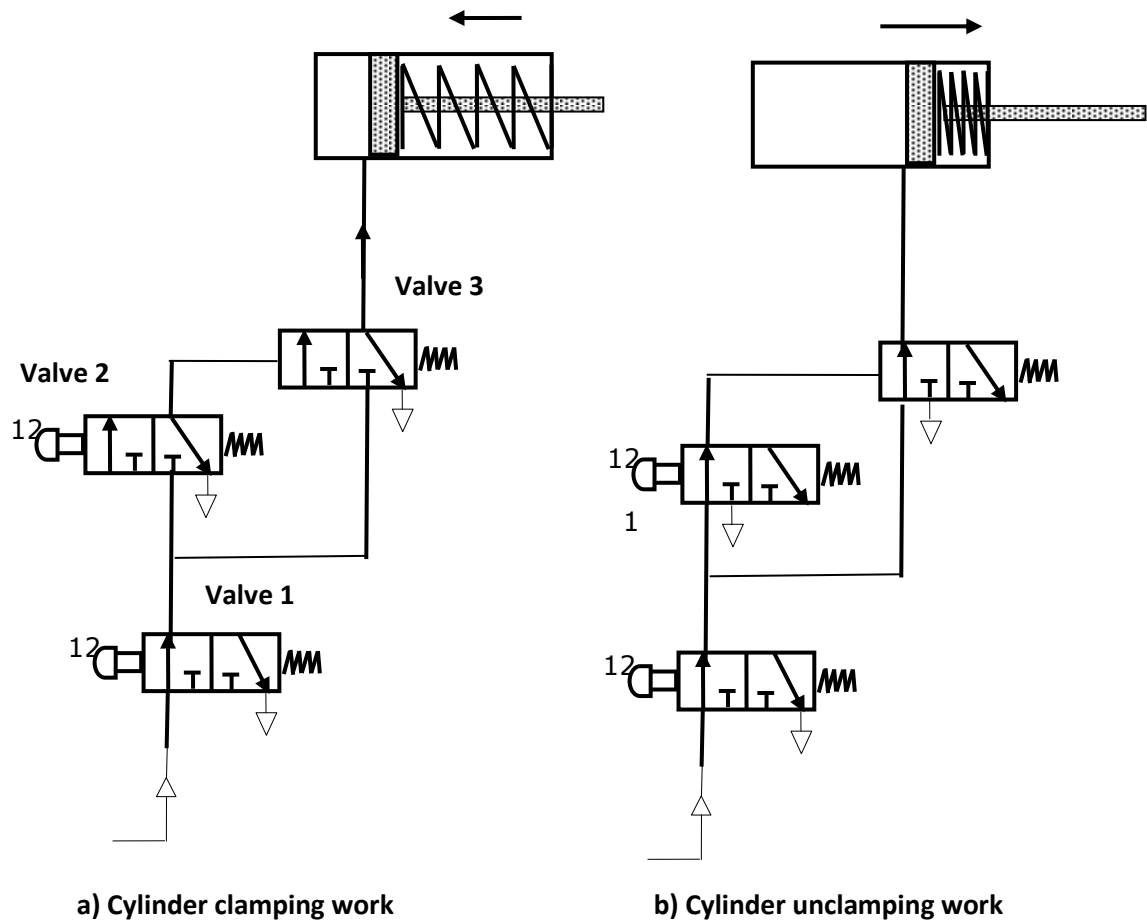


Figure 1.5

### 1.2.3 CONTROL OF SINGLE ACTING CYLINDER USING “OR” VALVE

Shuttle valve is also known as double control valve or double check valve. A shuttle valve has two inlets and one outlet (Figure 1.6). At any one time, flow is shut off in the direction of whichever inlet is unloaded and is open from the loaded inlet to the outlet. This valve is also called an OR valve. A shuttle valve may be installed for example, when the cylinder or valve is to be actuated from two points, which may be remote from one another.

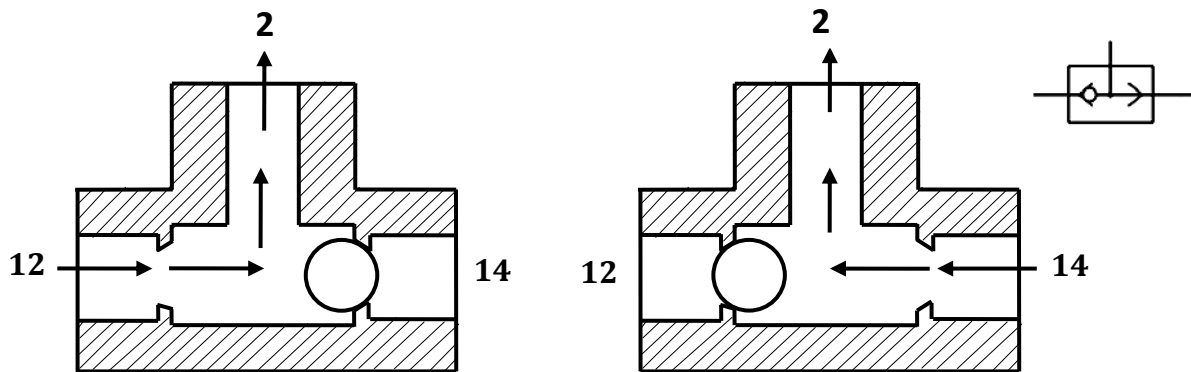
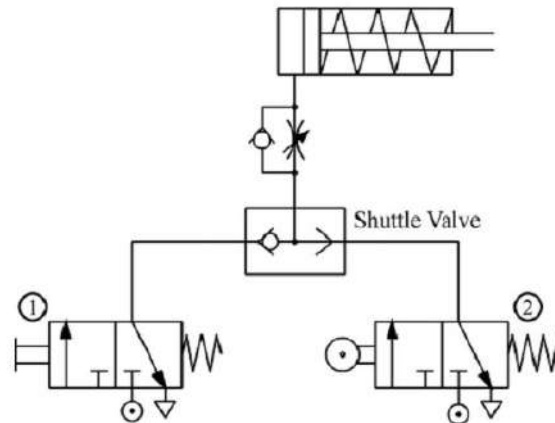


Figure 1.6 Shuttle valve ( OR valve)

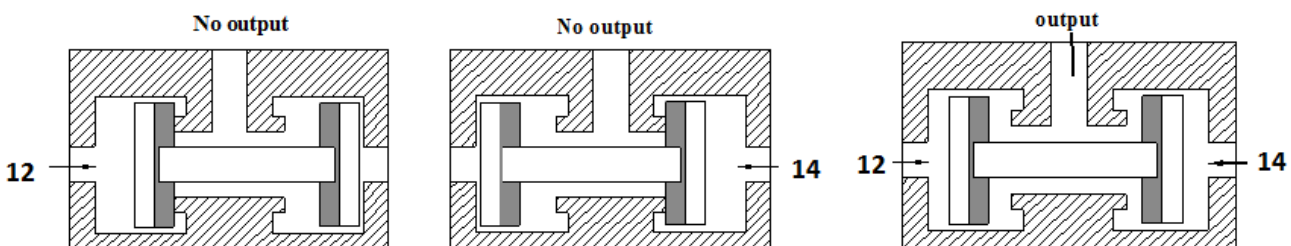
The single acting cylinder in **Figure 1.7** can be operated by two different circuits. Examples include manual operation and relying on automatic circuit signals, that is, when either control valve ① or control valve ② is operated, the cylinder will work. Therefore, the circuit in **Figure 1.7** possesses the OR function.



**Figure 1.7** Control of a single acting cylinder using OR valve

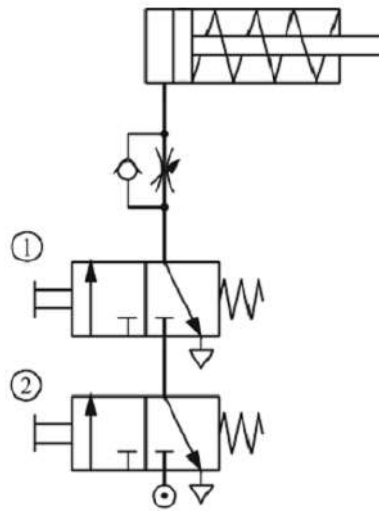
#### 1.2.4 CONTROL OF SINGLE ACTING CYLINDER USING “AND” VALVE

This valve is the pneumatic AND valve. It is also derivate of Non Return Valve. A two pressure valve requires two pressurised inputs to allow an output from itself. The cross sectional views of two pressure valve in two positions are given in **Figure 1.8** As shown in the **Figure 1.8**, this valve has two inputs 12 and 14 and one output 2. If the compressed air is applied to either 12 or input 14, the spool moves to block the flow, and no signal appears at output 2. If signals are applied to both the inputs 12 and 14, the compressed air flows through the valve, and the signal appears at output 2.



**Figure 1.8** control of a single acting cylinder using OR valve

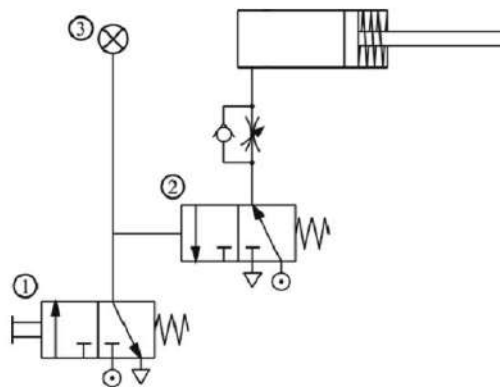
Another name for an AND function is interlock control. This means control is possible only when two conditions are satisfied. A classic example is a pneumatic system that works only when its safety door is closed and its manual control valve is operated. The flow passage will open only when both control valves are operated. **Figure 1.9** shows the circuit diagram of an AND function circuit. The cylinder will work only when both valve ① and ② are operated.



**Figure 1.9** Control of a single acting cylinder using AND valve

### 1.2.5 CONTROL OF SINGLE ACTING CYLINDER USING “NOT” VALVE

Another name for a NOT function is inverse control. In order to hold or lock an operating conveyor or a similar machine, the cylinder must be locked until a signal for cancelling the lock is received. Therefore, the signal for cancelling the lock should be operated by a normally open type control valve. However, to cancel the lock, the same signal must also cancel the locks on other devices, like the indication signal ③. **Figure 1.10** shows how the normally closed type control valve ① can be used to cut off the normally open type control valve ② and achieve the goal of changing the signal.

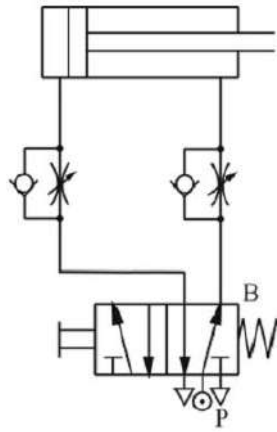


**Figure 1.10** Control of a single acting cylinder using NOT valve

### 1.3 DIRECT CONTROL OF DOUBLE ACTING CYLINDER

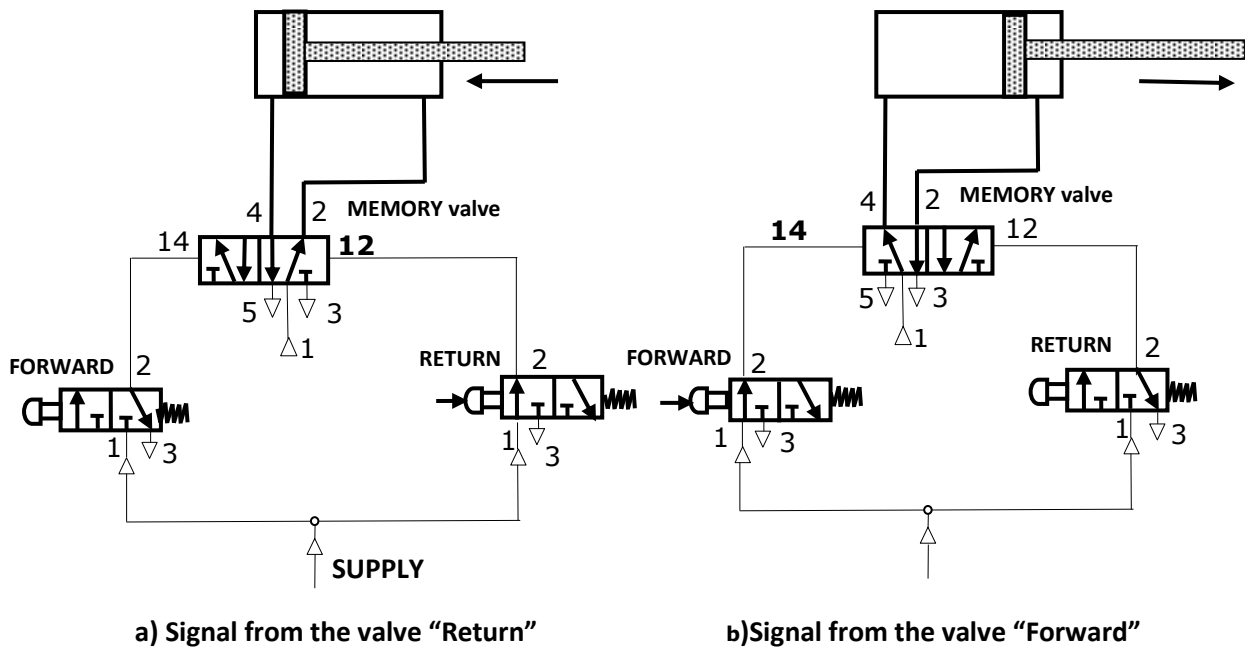
The only difference between a single acting cylinder and a double acting cylinder is that a double acting cylinder uses a 5/2 directional control valve instead of a 3/2 directional control valve (**Figure 1.11**). Usually, when a double acting cylinder is not operated, outlet ‘B’ and inlet ‘P’ will be connected. In this circuit, whenever the operation button is pushed manually, the double acting cylinder will move back and forth once





**Figure 1.11** Direct control of a double acting cylinder

### 1.3.1 IN DIRECT CONTROL OF DOUBLE ACTING CYLINDER USING MEMORY VALVE



**a) Signal from the valve "Return"**

**b) Signal from the valve "Forward"**

**Figure 1.14** Indirect control of Double acting cylinder using memory valve

When the 3/2 way valve meant for Forward motion (**Figure 1.14b**) is pressed, the 5/2 memory valve switches over through the signal applied to its pilot port 14. The piston travels out and remains in the forward end position. Double piloted valve is also called as the Memory valve because now even if this push button meant Forward is released the final 5/2 control valve remains in the actuated status as the both the pilot ports of 5/2 valves are exposed to the atmosphere pressure and the piston remains in the forward end position.

When the 3/2 way valve meant for return motion (**Figure 1.14a**) is pressed, the 5/2 way valve switches back to initial position through the signal applied to its pilot port 12. The piston then returns to its initial position and remains in the rear end position. Now even if the Return push button is released the status of the cylinder will not change.

## **MULTI ACTUATOR CIRCUITS**

### **1.1 SINGLE ACTUATOR CIRCUIT VERSUS MULTI ACTUATOR CIRCUITS**

Most of the practical pneumatic circuits use multi cylinders. They are operated in specific sequence to carry out the desired task. For example, to drill a wooden component first we need to clamp and then drill. We can only unclamp the cylinder, if and only if the drill is withdrawn away from the workpiece. Here sequencing of movement of clamp cylinders and cylinder which carries the drill is important. This sequencing is carried out by actuation of appropriate final control valves like directional control valves. The position of the cylinders is sensed by the sensors like limit switches, roller or cam operated valves.

Multi cylinder pneumatics circuits can be designed in various methods. There is no universal circuit design method that suits all types of circuits. Some methods are commonly used for compound circuits but would be too expensive for simple circuits. There are five common methods used by engineering and they are given below

- Classic method or Intuitive method
- Cascade method
- Step counter method
- Karnaugh–veitch method
- Combinational circuit design

### **1.3 CASCADE METHOD**

A Bi-stable memory valve or reversing valve can be used to eliminate signal conflicts. Signal conflict is avoided by allowing the signal to be effective only at times when they are needed. Two of the possible designs are possible.

- i) Cascade method
- ii) Shift register method

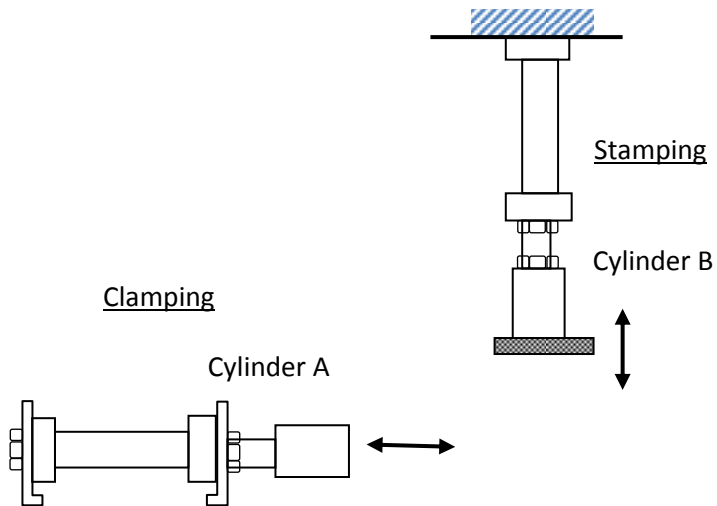
#### **1.3.1 Demonstration of Cascade method**

In order to develop control circuitry for multi cylinder applications, as done before in classic method, it is necessary to draw the motion diagram to understand the sequence of actuation of various signal input switches-limit switches and sensors. Motion diagram represents status of cylinder position - whether extended or retracted in a particular step

**Step 1: Write the statement of the problem:**

First cylinder A extends and brings under stamping station where cylinder B is located. Cylinder B then extends and stamps the job. Cylinder A can return back only cylinder B has retracted fully.

**Step 2: Draw the positional layout. (Figure 1.28)**



**Figure 1.28 Positional diagram**

**Step3: Represent the control task using notational form**

Cylinder A advancing step is designated as **A+**

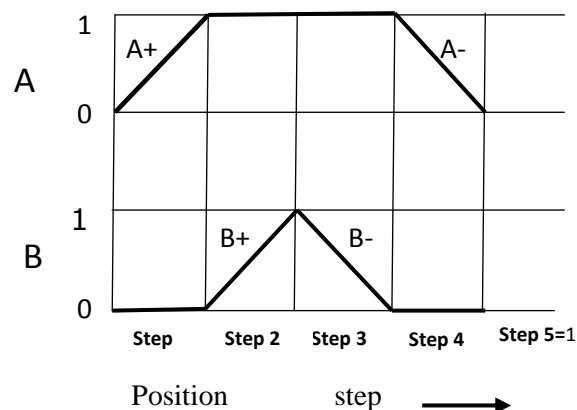
Cylinder A retracting step is designated as **A-**

Cylinder B advancing step is designated as **B+**

Cylinder B retracting step is designated as **B-**

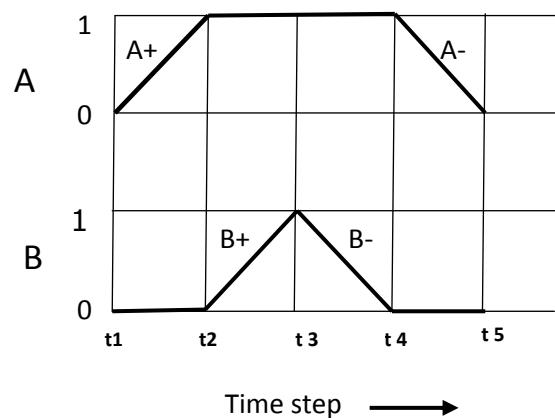
Given sequence for clamping and stamping is **A+B+B-A-**

**Step 4 Draw the Displacement –step diagram (Figure 1.29)**



**Figure 1.29 Displacement step diagram**

**Step 5 Draw the Displacement –time diagram (Figure 1.30)**



**Figure 1.30 Displacement time diagram**

**Step 6: Analyse and Draw Pneumatic circuit.**

**Step 6.1 Analyse input and output signals.**

**Input Signals**

Cylinder A – Limit switch at home position a0

Limit switch at home position a1

Cylinder B - Limit switch at home position b0

Limit switch at home position b1

**Output Signal**

Forward motion of cylinder A ( A+)

Return motion of cylinder A ( A-)

Forward motion of cylinder B ( B+)

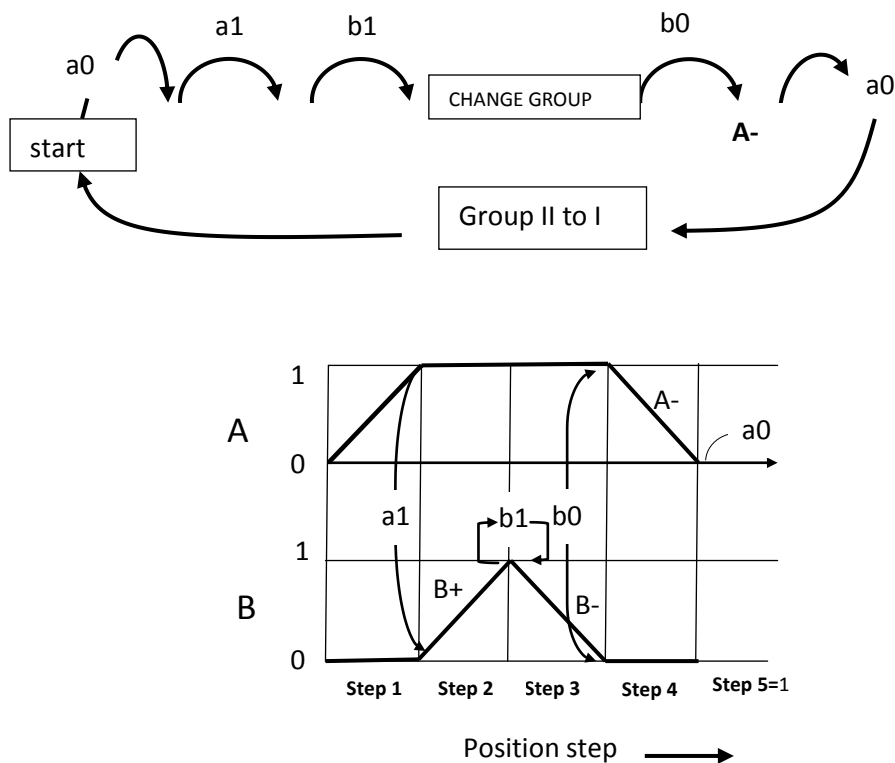
Return motion of cylinder B( B-)

**Step 6.2 Using the displacement time/step diagram link input signal and output signal. (Figure 1.31)**

Usually start signal is also required along with a0 signal for obtaining A+ motion.

1. A+ action generates sensor signal a1, which is used for B+ motion
2. B+ action generates sensor signal b1, which is used group changing.
3. B- action generates sensor signal b0, which is used for A- motion
4. A- action generates sensor signal a0, which is used for group changing

Above information (given in 6.2) is shown below graphically



**Figure 1.31 Displacement time diagram**

**Step 7 Draw the power circuit (Figure 1.32)**

- i) Divide the given circuits into groups. Grouping should be done such that there is no signal conflict. Do not put A+ and A- in the same group. Similarly B+ and B- should not be put in the same group. In other word A+ and A- should belong to different group to avoid signal conflict.

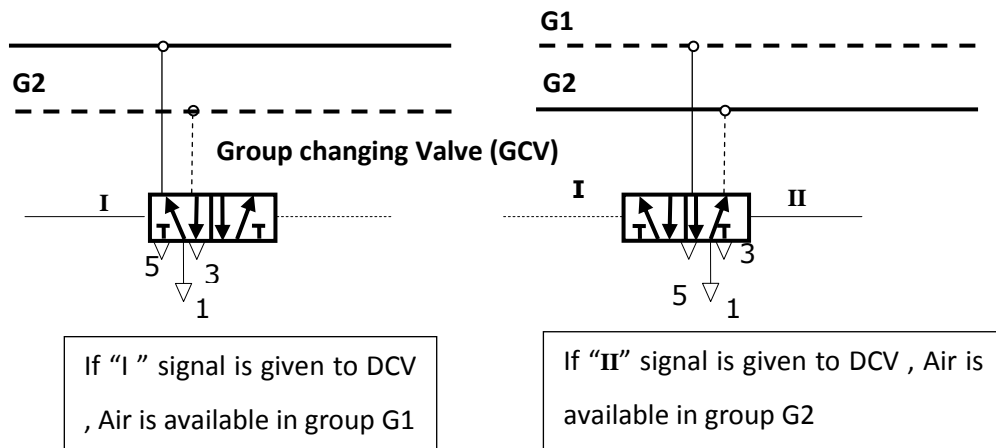
In our example of **A+ B+ B- A-** we can group as

A+ B+	B- A-
Group 1	Group 2

ii) Choose the number of **group changing valve** = no of groups -1

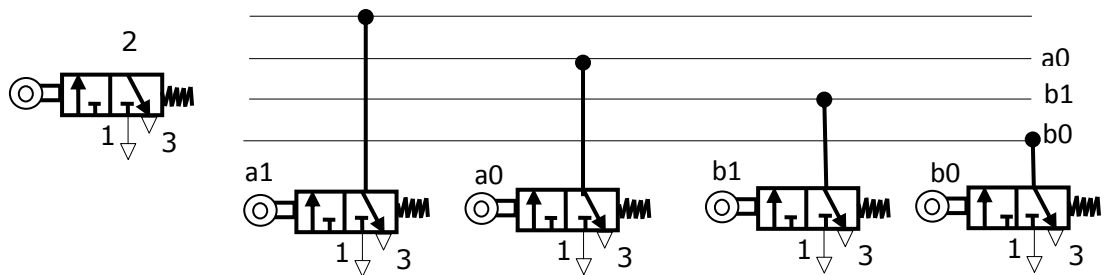
In our example, we have 2 groups so we need one group changing valve

Connect the group changing valve as follows. From the figure it is clear that when the control signals I and II are applied to group changing valve, the air (power) supply changes from Group 1(G1) to Group 2 (G2)



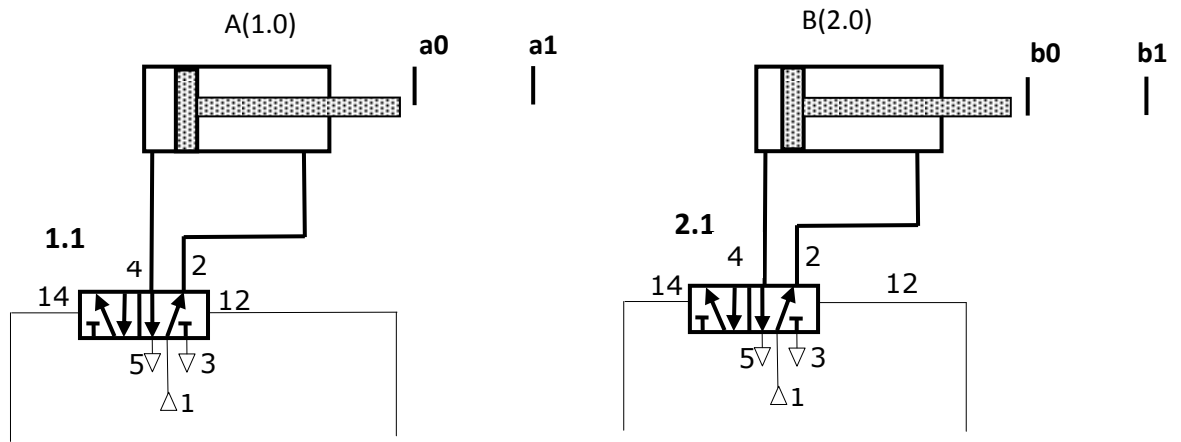
**Figure 1.32**

iii) Arrange the limit switch and start button as given below (**Figure 1.33**)



**Figure 1.33**

iv) Draw the power circuit (**Figure 1.34**)



**Figure 1.34**

**Step 8 Draw the control circuit (Figure 1.35)**

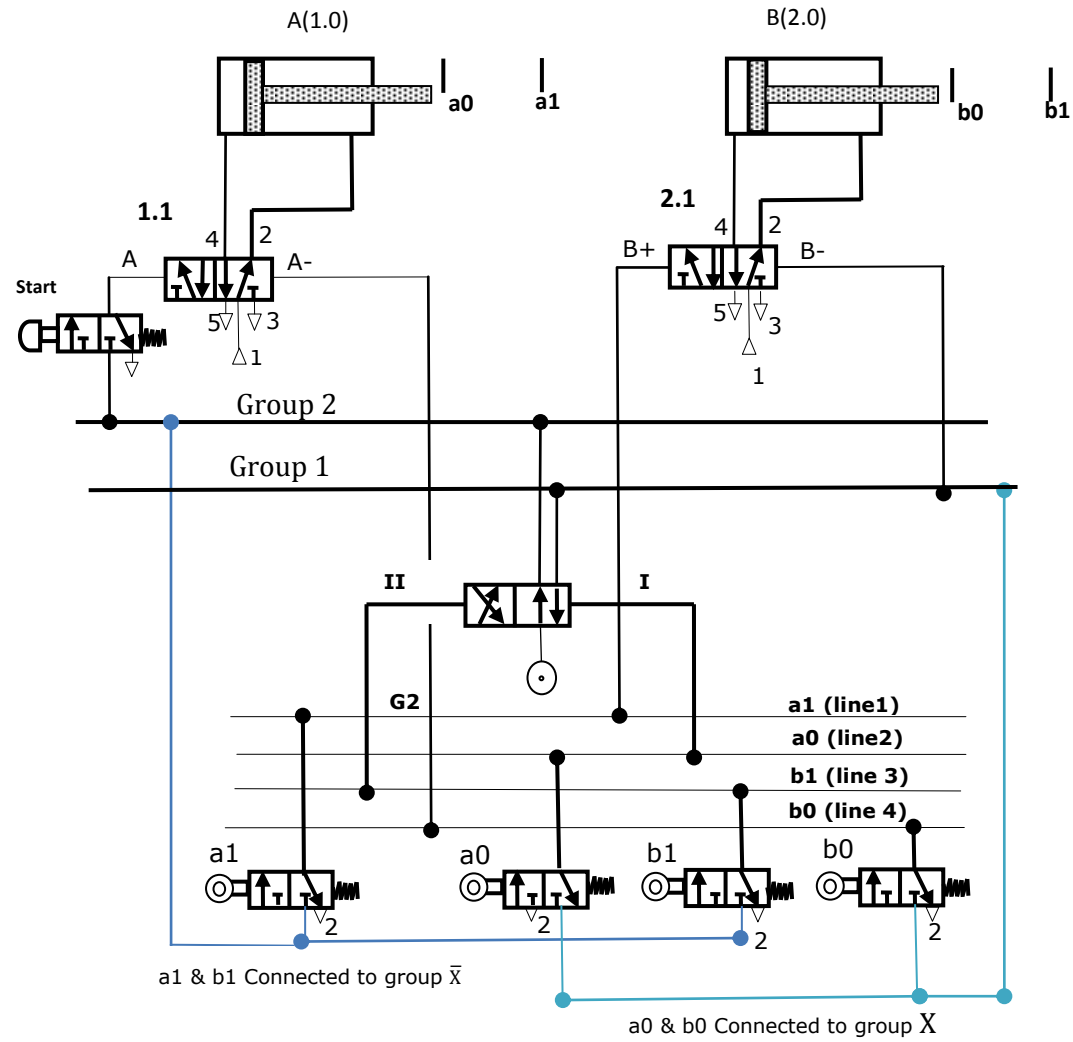


Figure 1.35 Pneumatic circuits for  $A + B + B - A -$



## Step 9 Analysis of pneumatic circuit

1. Assume that air is available in the line G2 to start with. (Say from last operation)
2. When the start button is pressed, Air supply from Group G2 is directed to line 2 through actuated limit switch a0. Now the air available in line 2, actuates the Group changing valve (GCV) to switch over to position I. This switching of the GCV causes air supply to change from G2 to G1.
3. Now the air is available in line G1. The air supply from group G1 is directed to port 14 of the valve 1.1. As there is no possibility of signal conflict here, valve 1.1 switches over causing the A+ action.
4. Sensor a1 is actuated as the result of A+ action, allowing the air supply from the Group G1 to reach to line 1 through a1. Now the air available reaches port 14 of valve 2.1. As there is no possibility of signal conflict here, valve 2.1 switches over, causing B+ action automatically.
5. Sensor b1 is actuated as result of B+ action, allowing the air supply in line 3. Air from line 3 allows the air to reach port 12 of Group changing valve (also called reversing valve). As a result, the Group changing valve switches over, causing the group supply to change from G1 to G2.
6. Now the air is available in G2. Air from G2 acts on port 12 of the Valve 2.1. As there is no possibility of signal conflict here, valve 2.1 switches over, causing B- action automatically.
7. Sensor is actuated as the result of B- action. Now the air is available in line 4. Air from line 4 reach port 12 of the valve 1.1 , As there is no possibility of signal conflict here, valve 2.1 switches over , causing A- action automatically.

The cascade system provides a straightforward method of designing any sequential circuit. Following are the important points to note:

- a) **Present** – the system must be set to the last group for start-up
- b) **Pressure drop**- Because the air supply is cascaded, a large circuit can suffer from more pressure drop.
- c) **Cost** – Costly due to additional reversing valves and other hardware.