SWAMI VIVEKANANDA SCHOOL OF ENGINEERING & TECHNOLOGY

SUBJECT NOTE – HYDRAULIC MACHINES & INDUSTRIAL FLUID POWER

SEMESTER – 5TH

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Hydraulic machine and Fluid Power (Industrial)

(1) Development of water turbines :

The hydraulic power was available mostly in rural and mountainous regions. As a result of this the mills directly run by water wheels, had to be installed near the power stability. After research, water turbines were designed which can operate under high head (highest head 1765 m in Austria) and can run at higher speeds.

(2) classification of water turbines :

(1) Impulse or velocity turbines
(2) Pressure or reaction turbines

(3) Hydroelectric Power Plant :

It consists of the following main components :

(1) Storage reservoir,
(2) Dam and its parts,
(3) Water ways,
(4) Water turbines, and electric generators

(1) Storage ~~tank~~ reservoir :

The water available from the catchment area is stored in the reservoir. The capacity of reservoir should be such that the water should be available for running the turbines, for producing the desired quantity of electric power, throughout the year. A reservoir may be natural or artificial.

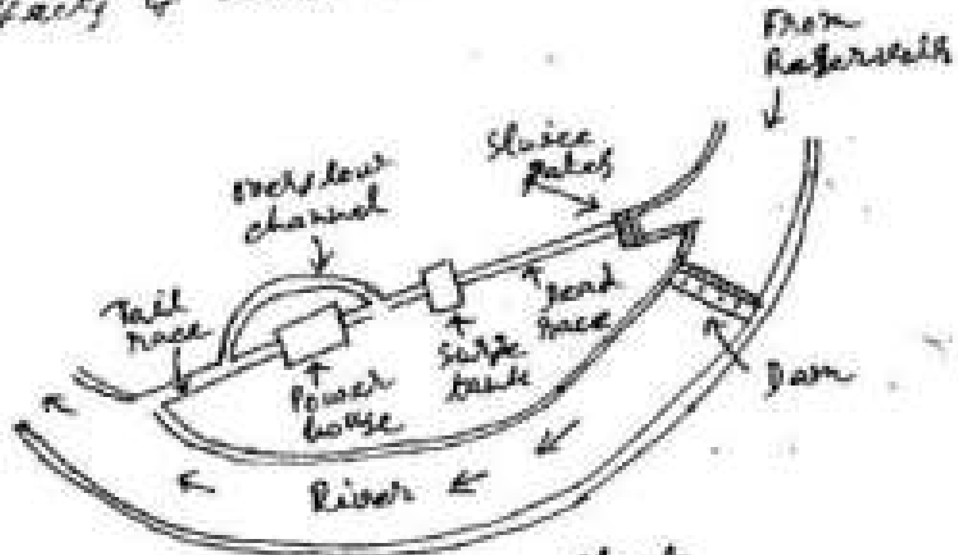
(2) Dams and its parts :

A dam is constructed across a river in order to divert the flow of water and impound in the reservoir formed on the upstream side. The size and type of dam depends upon the character of river, head of water, amount of discharge etc. Its shape and other components are decided by tests on model in the laboratory.

The dams are provided with gates for regulating the flow of water. There is also an arrangement of automatic overflow of excess water.

(3) Water ways :

A water ways is a passage through which the water is carried from the dam to the power house and then to the river. The upstream portion is known as head race and the downstream as tail race. It may consists of tunnels, canals, flumes, pipes, or any other suitable arrangement. A surge tank is also provided just on the upstream of the power house to control the pressure variation and eliminate the effects of water hammer.



Hydraulic Power Plant

(4) Water turbines and electric generators :

A place where water turbines and electric generators are installed, is called power house. Its design is very complicated and requires ... of skill.

The positions of a power house is decided by considering factors such as space available transport facilities etc. The size of a power house is decided by considering the factors such as supplying height number and size of units, type of units, electrical arrangements etc. The electrical generators are directly coupled with the turbines for better efficiency.

The turbines may be designed and laid either with their shafts horizontal or vertical. In horizontal shaft lay out, the whole installation lies on the same floor. Thus it is very easy to carry out inspection, service or any other modification in the plant. In a vertical shaft layout, it is convenient to connect incoming pipe and outgoing shaft tube. Moreover, the generators are placed well above the water surface, which makes their inspection, service and maintenance easier.

Impulse turbine

1 Introduction: An impulse turbine, as name indicates, is a turbine which runs by impulse of water. In an impulse turbine, the water from a dam is made to flow through a pipe line, and then through the guide mechanism and finally through the nozzle. In such a process, the entire available energy of the water is converted into kinetic energy, by passing it through nozzles, which are kept close to the runner. The water enters the running wheel in the form of a jet (or jets) which impinges on the buckets, fixed to the outer periphery of the wheel.

The jet of water impinges on the buckets with a high velocity, and after flowing over the vanes, leaves with a low velocity, thus imparting energy to the runner. The pressure of water, both the entering and leaving the vanes, is atmospheric. The commonest example of an impulse turbine is Pelton wheel.

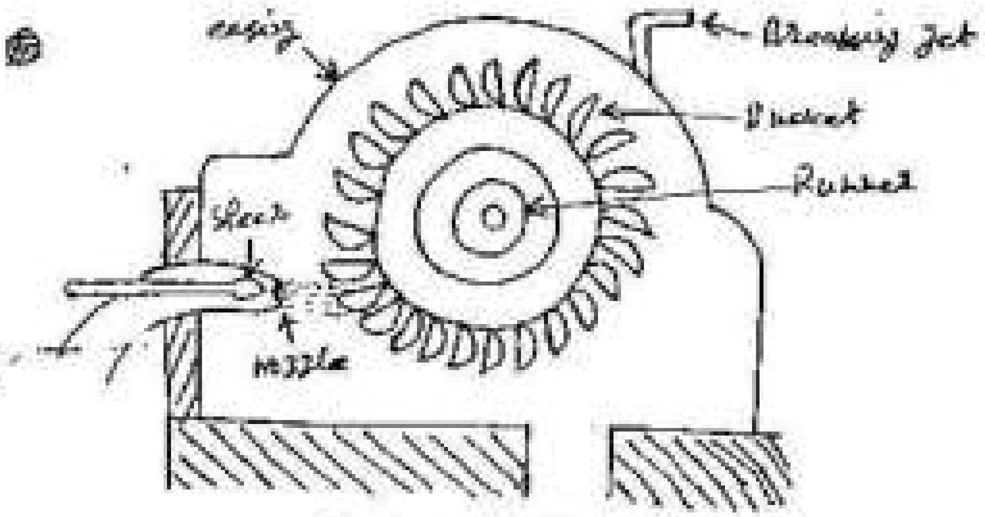
1 Pelton wheel:

The Pelton wheel is an impulse turbine used for high heads of water. It has following main components:

(1) Nozzle, (2) Runner and buckets, (3) casing and
(4) Breaking jet.

1 Nozzle: It is a circular guide mechanism, which guides the water to flow in a designed direction and also to regulate the flow of water. This water, in the form of jet, strikes the buckets. A conical needle or spear operates inside the nozzle in an axial direction. The main purpose of this spear, is to control or regulate the quantity of water flowing through the nozzle as shown in figure.

A little consideration will show, that when the spear is pushed forward into the nozzle, it reduces the area of jet. As a result of this, the quantity of water flowing through the jet is thus reduced. Similarly if the spear is pushed back out of the nozzle, it allows a greater quantity of water to flow through the jet. The movement of the spear is regulated by hand or by automatic governing arrangement, depending upon the requirement. Sometimes it is very essential to close the nozzle suddenly. This is done with the fall of spear, which may cause the pipe to burst due to sudden increase of pressure. In order to avoid such a mishap, an additional nozzle (known as bypass nozzle) is provided through which the water can pass, without striking the buckets. Sometimes a plate (known as deflector) is provided to the nozzle, which is used to deflect the water jet, and preventing, and preventing it from striking the buckets. The nozzle is kept very close to the buckets, in order to minimise the losses due to windage.



Parts of Pelton wheel

(ii) Runner and Buckets: The runner of a Pelton wheel essentially consists of a circular disc fixed to a horizontal shaft. On the periphery of the runner, a number of buckets are fixed uniformly. A buckets resembles to a hemispherical cup or bowl with a dividing wall (known as splitter) in its centre in the radial direction of the runner as shown in figure. The surface of the bucket is made very smooth. For low heads, the buckets are made of cast iron.

[illegible] the buckets are made of [illegible] [illegible] steel or other alloys. When the water is chemically impure, the buckets are made of special alloys. The buckets are generally bolted to the runner disc. But sometimes, the buckets and disc are cast as a single unit. Sometimes, all the buckets wear equally in a given time. But in actual practice, all the buckets do not wear equally. A few buckets get worn out and damaged early and need replacement. This can be done only if the buckets are bolted to the disc.

(ii) Casing: Strictly speaking the casing of Pelton wheel does not perform any hydraulic function. But it is necessary to safeguard the runner against accident, and also to prevent the splashing of water and lead the water to the tail race. The casing is generally made of cast or fabricated parts.

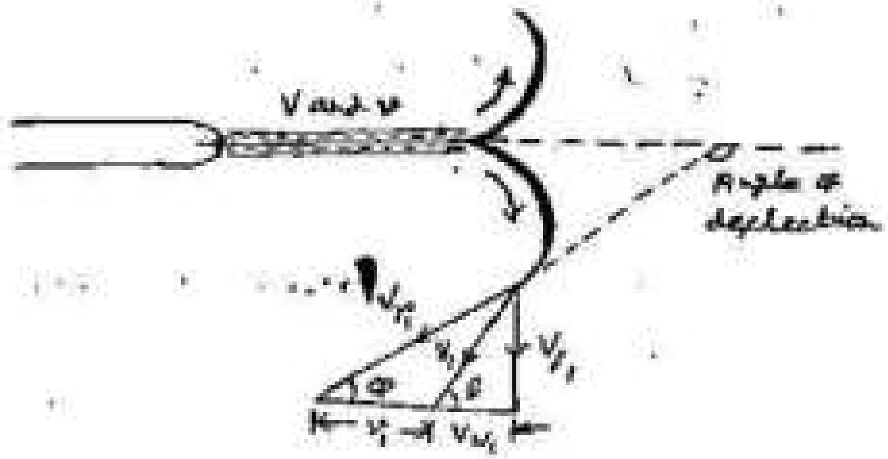
(iii) Braking Jet: Whenever the turbine has to be brought to rest, the nozzle is completely closed. It has been observed, that the runner goes on revolving for a considerable time, due to inertia, before it comes to rest. In order to bring the runner to rest in a short time, a small nozzle is provided in such a way, that it will direct a jet of water on the back of the buckets. It acts as a brake for reducing the speed of the runner.

(i) Work done by an impulse turbine:

The jet of water, issuing from the nozzle, strikes the bucket at its splitter. The splitter then splits up the jet into two parts. One part of the jet glides over the inside surface of one portion of the [illegible] and leaves it at its extreme edge. The other portion of the jet glides over the inside surface of the other portion and leaves it at its other extreme edge.

[illegible]

the two extreme edges where the divided jet leaves the bucket, from the two outlet tips.



Triangle of velocities

First of all, draw the velocity triangles at the splitter (which will be a straight line only) and any one of the outlet tips of the semispherical bucket as shown in figure. All the notations and theory of jet impinging on series of vanes is applicable in the case also.

Let, V = Absolute velocity of the entering water,

V_r = Relative velocity of water and bucket at inlet,

V_f = velocity of flow at inlet.

V_1, V_{r_1}, V_{f_1} = corresponding values at outlet, i.e., at the point of leaving,

D = Diameter of the wheel,

d = Diameter of the nozzle,

N = Revolutions of the wheel in r.p.m.,

ϕ = Angle of the outlet blade tip at outlet,

H = Total head of water, under which the wheel is working.

It will be interesting to know that inlet velocity triangle will be a straight line as shown in the figure. In this case, $\alpha = 0^\circ$, $\theta = 0^\circ$, $V_w = V$ and $V_r = V - v$

As a matter of fact, the shape of the outlet velocity triangle depends upon the value of V_{w_1}.

If V_{w_1} is in the same direction as that of the jet, its value is taken as positive. However, if V_{w_1} is in the opposite (as shown in figure) its value is taken as negative. The relation between these two velocity triangles, is

$$V_1 = v \text{ and } V_{r_1} = V_r = (V - v)$$

We know that force per kN of water in the direction of motion of the jet,

$$= \frac{1}{g}(V_w - V_{w_1})$$

W.D = no of gm × change of velo of whirl × but velo $= \frac{m}{g} \times (V_{w_1} V_{w_1}) \times V_b$ N·m/s

Power = $\frac{?}{?}$ watts (F_x × u) Nm/s

and work done = Force × Distance

$$= \frac{1}{g}(V_w \cdot v - V_{w_1} \cdot v_1)$$

$$= \frac{1}{g}(V_w - V_{w_1}) \times v \quad \ldots\ldots (\because v_1 = v)$$

$\therefore$ Hydraulic efficiency, $\eta_h = \frac{\text{workdone per kN of water}}{\text{kinetic energy of the jet}}$

$$= \frac{\frac{1}{g}(V_w - V_{w_1}) \times v}{\frac{V^2}{2g}}$$

$$= \frac{2(V_w - V_{w_1}) \times v}{V^2}$$

Now consider a case, in which the value of V_{w_1} is negative as shown in figure. Therefore work done per kN of water

$$= \frac{1}{g}[V_w - (-V_{w_1})] \times v = \frac{1}{g}(V_w + V_{w_1}) \times v$$

$$= \frac{V_w v}{g} + \frac{V_{w_1} v}{g} = \frac{V_w v}{g} + \frac{(V_{r_1} \cos\phi - v)v}{g}$$

$$\ldots [\because V_{w_1} = V_{r_1} \cos\phi - v,\ v_1 = v]$$

$$= \frac{v}{g}[V_w + (V - v)\cos\phi - v]$$

$$= \frac{v}{g}[V + (V - v)\cos\phi - v] \quad \ldots [\because V_w = V]$$

$$= \frac{v}{g}[V(1 + \cos\phi) - v(1 + \cos\phi)]$$

$$= \frac{v(V - v)(1 + \cos\phi)}{g}$$

We know that the hydraulic efficiency,

$$\eta_h = \frac{\text{Workdone per kN of water}}{\text{Energy supplied per kN of water}}$$

$$= \frac{\frac{v(V-v)(1+\cos\phi)}{g}}{\frac{V^2}{2g}}$$

$$= \frac{2v(V-v)(1+\cos\phi)}{V^2}$$

For maximum efficiency, differentiate the numerator of the above equation, with respect to v and equate it to zero (as the maximum efficiency will be, when the numerator will be maximum).

$$\frac{d}{dv}\left[2v(V-v)(1+\cos\phi)\right] = 0$$

or, $\frac{d}{dv}\left[(2Vv - 2v^2)(1+\cos\phi)\right] = 0$

or, $2V - 4v = 0$ or, $v = \frac{V}{2}$

It means that the velocity of the wheel, for maximum hydraulic efficiency, should be half of the jet velocity. Therefore maximum workdone/kN of water,

$$= \frac{v(V-v)(1+\cos\phi)}{g} = \frac{\frac{V}{2}\left(V-\frac{V}{2}\right)(1+\cos\phi)}{g}$$

$$= \frac{V^2}{4g}(1+\cos\phi) \text{ kN} \qquad \cdots\left[\text{Substituting } v = \frac{V}{2}\right]$$

$\therefore$ Maximum hydraulic efficiency,

$$\max \eta_h = \frac{\frac{V^2}{4g}(1+\cos\phi)}{\frac{V^2}{2g}} = \frac{1+\cos\phi}{2}$$

<u>Notes</u>: (1) It may be noted that the efficiency is maximum when $\cos\phi = 1$ i.e, $\phi = 180°$. But in actual practice, the jet is deflected through an angle of 160° to 165° only. Because if the jet is made to deflect through an angle of 180°, the water discharged from one bucket will have an impact on the bucket, in front of it.

(2) In actual practice, maximum efficiency takes place when the velocity of wheel is 0.46 times the velocity of the jet (i.e, $u = 0.46\,V$)

(3) The power generated by the turbine may be found out as usual by multiplying the discharge in m^3/s with the work done per kN of water.

④ <u>Design aspect of Pelton wheel</u>:

(1) <u>Velocity of jet</u>: The velocity of jet at inlet is given by,

$$V_1 = C_v\sqrt{2gH}$$

Where, C_v = co-efficient of velocity (0.98 or 0.99)
and H = Net head on turbine

(2) <u>Velocity of wheel</u>: The velocity of wheel (u) is given by;

$$u = K_u\sqrt{2gH}$$

Where, K_u = Speed ratio. It varies from 0.43 to 0.48.

(3) <u>Angle of deflection of the jet</u>:

The angle of defl^n of jet through the buckets is taken as 165° if no angle of defl^n is given.

(4) <u>Mean dia of the wheel (D)</u>:

The mean or pitch dia D of the Pelton wheel is given by,

$$u = \frac{\pi D N}{60} \quad \text{or} \quad D = \frac{60u}{\pi N}$$

[illegible]

(5) <u>Jet Ratio (m):</u> It is defined as the ratio of the dia (D) of pelton wheel to the dia of the jet (d). It is denoted by 'm' and is given as,

$$m = \frac{D}{d} \text{ (lies between 11 and 14 for max } \eta_0)$$

(6) <u>Size of buckets of a pelton wheel:</u>

Width of the bucket = 5×d

and depth of the bucket = 1.2×d (d = dia of the jet)

(7) <u>Number of buckets on the periphery of a pelton wheel:</u>

Theoritically, number of buckets, $= \frac{360°}{\alpha}$

But in actual practice,

$$\text{Number of buckets} = \left(\frac{D}{2d} + 15\right)$$

D = mean bucket diameter, and

d = diameter of the jet.

(8) <u>Power produced by an impulse turbine:</u>

If we know the quantity of water in KN, flowing through the jets per second and the amount of workdone per second, then the power produced by the turbine may be found out with the help of velocity triangles as usual.

$$P = w Q H$$

where, w = Specific weight of water (9.81 KN/m³)

Q = Discharge of the turbine in m³/s and

H = Head of water in metres

(9) <u>Efficiencies of an impulse turbine:</u>

(1) Hydraulic efficiency

(2) Mechanical efficiency and

(3) Overall efficiency.

(i) **Hydraulic efficiency:**

It is the ratio of work done on the wheel to the energy of the jet.

$$\eta_h = \frac{2v(V-v)(1+\cos\phi)}{V^2}$$

and maximum hydraulic efficiency,

$$\max \eta_h = \frac{(1+\cos\phi)}{2}$$

(ii) **Mechanical Efficiency:**

It has been observed that all the energy supplied to the wheel does not come out as useful work. But a part of it is dissipated in overcoming friction of bearings and other moving parts. Thus the mechanical efficiency is the ratio of actual work available at the turbine to the energy imparted to the wheel.

(iii) **Overall Efficiency:**

It is a measure of the performance of a turbine, and is the ratio of the actual power produced by the turbine to the energy actually supplied by the turbine.

$$\eta_o = \frac{P}{wQH}$$

(iv) **Governing of an impulse turbine (Pelton wheel):**

In actual practice, load on the generator (which is coupled to an impulse turbine) is always fluctuating from time to time. This fluctuating load on the generator has some effect on the turbine also, because the generator is directly coupled to the turbine. A little consideration will show that any change of load on the turbine ...

Q(3) A Pelton wheel develops 2000 kW under a head of 100 metres and with an overall efficiency of 85%. Find the diameter of the nozzle, if the coefficient of velocity for the nozzle is 0.98.

<u>Solution</u>: Given: $P = 2000$ kW, $H = 100$ m, $\eta_o = 85\%$ $= 0.85$ and $C_v = 0.98$.

Let d = diameter of the nozzle, and

Q = Discharge of the turbine.

We know that the velocity of the jet,

$$V = C_v\sqrt{2gH}$$

$$= 0.98 \times \sqrt{2 \times 9.81 \times 100} = 43.4 \text{ m/s}$$

and overall efficiency (η_o), $\left(\frac{\text{Power available at turbine shaft}}{\text{Power in the form the water jet}}\right)$

$$0.85 = \frac{P}{wQH} = \frac{2000}{9.81 \times Q \times 100} = \frac{2.04}{Q}$$

$$\therefore Q = 2.04/0.85 = 2.4 \text{ m}^3/\text{s}$$

Now the total discharge of the wheel should be equal to the discharge through the jet. i.e,

$$Q = V \times \frac{\pi}{4} \times (d)^2$$

or, $$2.4 = 43.4 \times \frac{\pi}{4} \times d^2 = 34.1\, d^2$$

$$\therefore d^2 = 2.4/34.1 = 0.0704$$

or, $$d = 0.265 \text{ m} = 265 \text{ mm}$$ Ans:

Q(4) A Pelton wheel, having semicircular buckets and working under a head of 140 m, is running at 600 r.p.m. The discharge through the nozzle is 500 lit/sec and diameter of the wheel is 600 mm. Find:

(a) Power available at the nozzle, and

(b) Hydraulic efficiency of the wheel, if coefficient of velocity is 0.98.

Solution: Given, $\phi = 180° - 180° = 0°$ (because of semi-circular buckets), $H = 140$ m; $N = 600$ r.p.m, $Q = 500$ lit/s $= 0.5$ m³/s

$D = 600$ mm $= 0.6$ m, and $C_v = 0.98$.

(a) Power available at the nozzle

$$P = wQH = 9.81 \times 0.5 \times 140 = 686.7 \text{ kW} \quad \text{Ans:}$$

(b) Hydraulic efficiency of the wheel,

We know that velocity of the Jet,

$$V = C_v \sqrt{2gH} = 0.98 \times \sqrt{2 \times 9.81 \times 140}$$

$$= 51.36 \text{ m/s}$$

and tangential velocity of the wheel,

$$v = \frac{\pi D N}{60} = \frac{\pi \times 0.6 \times 600}{60} = 18.85 \text{ m/s}$$

∴ Hydraulic efficiency of the wheel,

$$\eta_h = \frac{2v(V-v)(1+\cos\phi)}{V^2}$$

$$= \frac{2 \times 18.85 (51.36 - 18.85)(1+\cos 0°)}{(51.36)^2}$$

$$= 0.465 (1+1) = 0.929 = 92.9\% \quad \text{Ans:}$$

(4) In a hydraulic scheme, the distance between high level reservoir at the top of mountains and turbine is ~~level reservoir at the top of mountains and turbine is~~ 1.6 km and difference of ~~higher~~ their levels is 550 m. The water is brought in 4 penstocks each of diameter of 0.9 m connected to a nozzle of 200 mm dia at the end. Find:

(a) Power of each Jet, and

(b) Total power available at the reservoir, taking the value of Darcy's coefficient friction as 0.008.

Solution: Given: $l = 1.6$ km $= 1600$ m, $H = 550$ m,

No of Penstocks $(n) = 4$, Dia of each penstocks $(D) = 0.9$ m,

Dia of nozzle $(d) = 200$ mm $= 0.2$ m and $f = 0.008$.

(a) Power of each Jet

We know that area of each nozzle,

$$a = \frac{\pi}{4} \times d^2 = \frac{\pi}{4} \times (0.2)^2 = 0.0314 \ m^2$$

and velocity of jet, $V = C_v \sqrt{2gH}$

$$= 0.98 \times \sqrt{2 \times 9.81 \times 500} = 97.1 \ m/s$$

... (assuming $C_v = 0.98$)

and velocity of the jet.

∴ Discharge through end jet,

$$Q = a.V = 0.0314 \times 97.1 = 3.05 \ m^3/s$$

and Power of each jet, $= wQH = 9.81 \times 3.05 \times 500$

$$= 14\,960 \ kW \ \underline{Ans.}$$

(b) Total power available at the reservoir,

We also know that area of the penstock,

$$A = \frac{\pi}{4} \times (D)^2 = \frac{\pi}{4} \times (0.9)^2 = 0.636 \ m^2$$

and velocity of water in penstock,

$$v = \frac{Q}{A} = \frac{3.05}{0.636} = 4.8 \ m/s.$$

∴ Head lost due to friction in each penstock,

$$H_f = \frac{4flv^2}{2gd} = \frac{4 \times 0.008 \times 1600 \times (4.8)^2}{2 \times 9.81 \times 0.9}$$

$$= 66.8 \ m$$

and total power available at the reservoir (in 4 penstocks)

$$P = 4 \times wQ(H + H_f)$$

$$= 4 \times 9.81 \times 3.05 (500 + 66.8) \ kW$$

$$= 67\,836 \ kW \ \underline{Ans.}$$

(5) The Pykara Power house, in South India, is equipped with impulse turbines of Pelton type. Each turbine delivers a maximum power of 14250 kW when working under a head of 900 metres and running at 600 r.p.m.

Find the diameter of the jet and the mean diameter of the wheel. Take overall efficiency of the turbine as 89.2%.

<u>Solution</u>: Given, $P = 14250$ kW, $H = 900$ m, $N = 600$ r.p.m and $\eta_o = 89.2\% = 0.892$ (diameter of the jet)

Let, d = diameter of the jet, and

Q = Discharge of the turbine.

We know that overall efficiency of the turbine (η_o),

$$0.892 = \frac{P}{wQH} = \frac{14250}{9.81 \times Q \times 900} = \frac{1.61}{Q}$$

$$\therefore Q = 1.61/0.892 = 1.8 \ m^3/s$$

and velocity of the jet, $V = C_v \times \sqrt{2gH}$

$$= 0.98 \times \sqrt{2 \times 9.81 \times 900} = 130.2 \ m/s$$

--- (Assuming $C_v = 0.98$)

Now, the discharge through the turbine must be equal to discharge through the jet. i.e.,

$$Q = V \times \frac{\pi}{4} \times d^2$$

$$\text{or, } 1.8 = 130.2 \times \frac{\pi}{4} \times d^2 = 102.3 \ d^2$$

$$\therefore d^2 = 1.8/102.3 = 0.018$$

$$\text{or, } d = 0.134 \ m = 134 \ mm \quad \text{Ans.}$$

Mean diameter of the wheel,

Let, D = mean diameter of the wheel

We know that peripheral velocity of the wheel,

$$v = 0.46 \times V = 0.46 \times 130.2 = 59.9 \ m/s$$

Now the total discharge of the wheel should be equal to the discharge through the jet, i.e,

$$q = V \times \frac{\pi}{4} \times d^2$$

$$1.25 = 85.7 \times \frac{\pi}{4} \times d^2 = 67.5\, d^2$$

$m = 10 = \frac{D}{d}$ or,

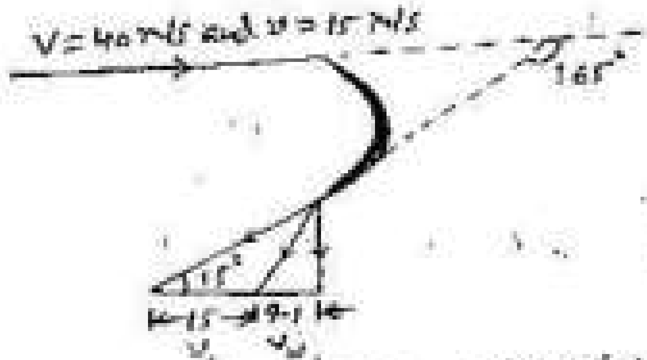
$= \frac{D}{136}$ $\therefore$ $d^2 = 1.25 / 67.5 = 0.0185$ or

$\Rightarrow D = 1360\ mm$ or, $d = 0.136\ m = 136\ mm$ Ans.

$= 1.36\ m$ Ans.

(7) A Pelton wheel has a mean bucket speed of 15 m/sec with a jet of water impinging with a velocity of 40 m/sec and discharging 450 liters/sec. If the buckets deflects the jet through an angle of 165°, find the force generated by the wheel.

Solution: Given: $v = 15\ m/sec$, $V = 40\ m/s$, $q = 450\ lit/s$ $= 0.45\ m^3/sec$ and $\phi = 180 - 165° = 15°$ (because the jet is deflected through 165°)



From the inlet triangle, we find that velocity of whirl inlet,

$$V_w = 40\ m/s \qquad [\because V = V_w]$$

and relative velocity, $V_r = V - v$ (ii)

$= 40 - 15 = 25\ m/s$

From the outlet triangle, we also find that,

$$V_{r_1} = V_r = 25\ m/s$$

and velocity of buckets, $v_1 = v = 15\ m/s$

∴ Velocity of whirls outlet,

$V_{w_1} = v_1 - V_{r_1} \cos\alpha = 15 - (25 \cos 15^\circ)$

$= 15 - (25 \times 0.9659)$ m/s

$= 15 - 24.1 = -9.1$ m/s

... (minus sign indicates that the direction of V_{w_1} is opposite to that of V or v)

∴ We know that workdone per kN of water,

$= \frac{v}{g}(V_w - V_{w_1}) = \frac{15}{9.81}[40 - (-9.1)]$

$= \frac{15}{9.81} \times 49.1$ kN-m/s

$= 75.1$ kN-m/s

and total work done $= 9.81 \times 0.45 \times 75.1$

$= 331.5$ kN-m/sec.

∴ Power generated by the wheel,

$P = 331.5$ kJ/sec $= 331.5$ kW Ans.

(ii) Depending upon magnitude of the peripheral speed (u), the unit may have a slow, medium or fast runner and the angle β and V_{w_2} will vary as follows:

(i) Slow runner $\beta < 90^\circ$ (V_{w_2} is $-ve$)

and $V_{w_2} = V_{r_2} \cos\alpha - u_2 = V_{r_2} \cos\alpha - u$ [∵ $u_1 = u_2 = u$ when $\beta < 90^\circ$]

(ii) Medium runner $\beta = 90^\circ$ ($V_{w_2} = 0$)

(iii) Fast runner $\beta > 90^\circ$ (V_{w_2} is $+ve$)

PART C

(2) (1) A Pelton wheel is receiving water from a penstock with a gross head of 510 m. One third of gross head is lost in friction in penstock. The rate of flow through the nozzle fitted at the end of the penstock is 2.2 m^3/s. The angle of deflection of the jet is 165°. Determine:

(i) The power given by water to the runner, and

(ii) Hydraulic efficiency of the Pelton wheel.

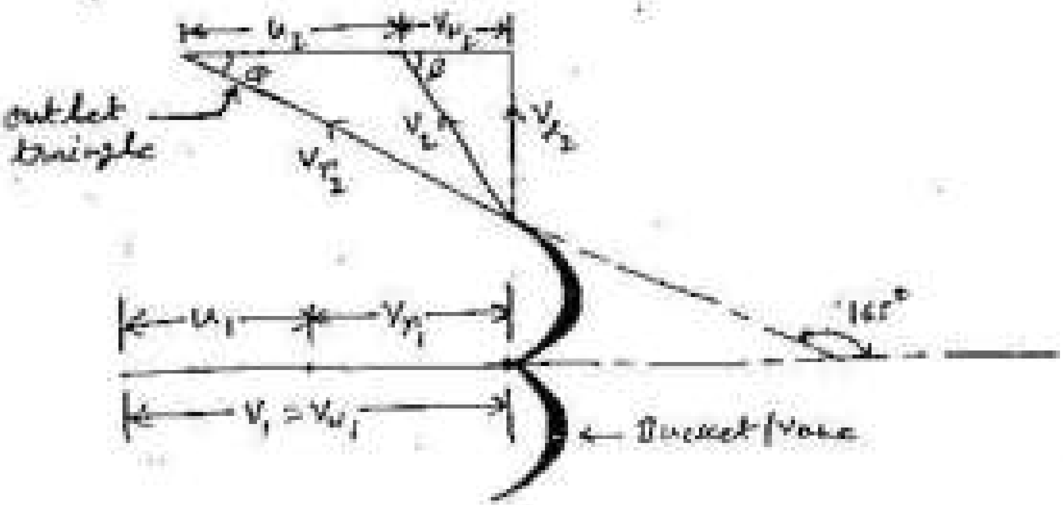
Take C_v (co-efficient of velocity) = 1.0 and speed ratio = 0.45.

Solution: Gross head, $H_g = 510$ m. Head lost in friction, $h_f = \frac{H_g}{3} = \frac{510}{3} = 170$ m

∴ Net head, $H = H_g - h_f = 510 - 170 = 340$ m

Discharge, $Q = 2.2\ m^3/s$

Angle of deflection = 165°



Angle, $\phi = 180° - 165° = 15°$

Co-efficient of velocity, $C_v = 1.0$

Speed ratio, $K_u = 0.45$

(i) The power given by water to runner:

Velocity of jet, $V_1 = C_v\sqrt{2gH} = 1.0\sqrt{2 \times 9.81 \times 340} = 81.67$ m/s

Velocity of wheel, $u_1 = K_u\sqrt{2gH} = 0.45\sqrt{2 \times 9.81 \times 340} = 36.75$ m/s

From figure, $V_{r_1} = V_1 - u_1 = V_1 - u = 81.67 - 36.75$ $[\because u_1 = u_2 = u]$

$= 44.92$ m/s

Also, $V_{w_1} = V_1 = 81.67$ m/s

From outlet velocity triangle, we have:

$V_{r_2} = V_{r_1} = 44.92$ m/s

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

or, $V_{w_2} = V_{r_2}\cos\phi - u = 44.32\cos 15^\circ - 36.75$

$= 6.64$ m/s

Work done by the jet on the runner per second,

$$= \rho a (V_{w_1} + V_{w_2}) \times u$$

$$= 1000 \times 2.2\,(81.67 + 6.64) \times 36.75$$

$$= 7139813 \text{ Nm/s}$$

∴ Power given by water to the runner $= 7139813$ J/s

or, $= 7139.8$ kW (Ans).

(ii) Hydraulic efficiency of the Pelton wheel, η_h:

$$\eta_h = \frac{2(V_{w_1} + V_{w_2}) \times u_1}{V_1^2}$$

$$= \frac{2(81.67 + 6.64) \times 36.75}{(81.67)^2} = 0.973$$

$$= 97.3\ \% \text{ Ans.}$$

Q(5) A single jet Pelton wheel runs at 300 r.p.m under a head of 510 m. The jet diameter is 200 mm, its deflection inside the bucket is 165° and its relative velocity is reduced by 15% due to friction. Determine:

(i) Water power,

(ii) Resultant force on the bucket, and

(iii) Overall efficiency.

Take: mechanical losses = 3%, co-efficient of velocity = 0.98, and speed ratio = 0.46.

Solution: Speed of the wheel, $N = 300$ r.p.m

Diameter of jet, $d = 200$ mm $= 0.2$ m

Net head, $H = 510$ m

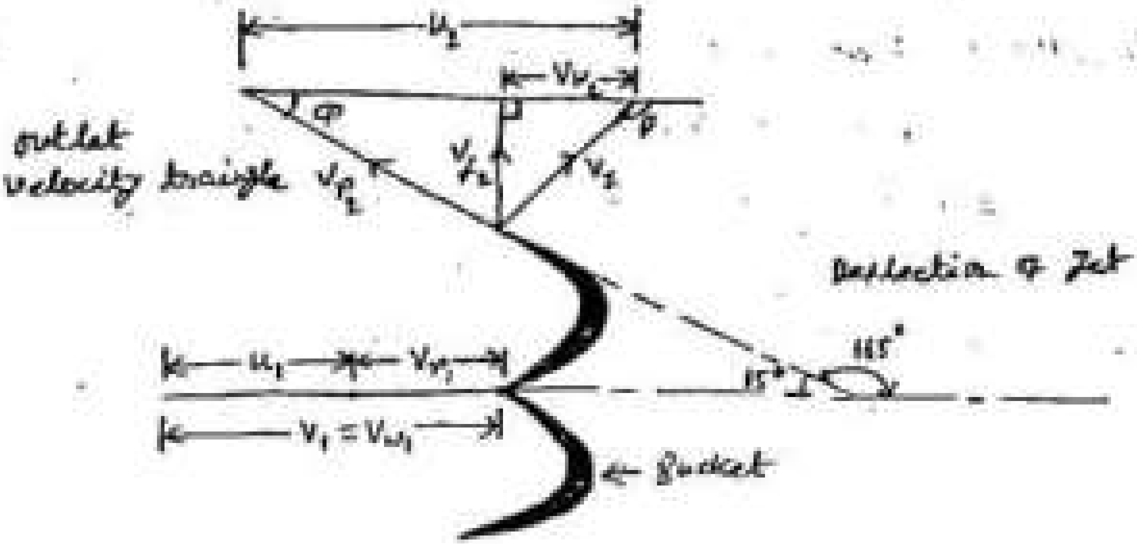
Angle of deflection of jet $= 165^\circ$

Reduction of relative velocity due to friction $= 15\%$

Mechanical losses $= 3\%$

Co-efficient of velocity, $C_v = 0.98$

Speed ratio, $K_u = 0.46$



(i) Water Power :

Velocity of Jet, $V_1 = C_v \sqrt{2gH} = 0.98\sqrt{2\times 9.81\times 510}$
$= 98$ m/s

∴ Discharge through the Pelton wheel,

$Q =$ Area of Jet $(a) \times$ Velocity (V_1)
$= \frac{\pi}{4} \times (0.2)^2 \times 98 = 3.078$ m³/s

∴ Water Power $= WQH = 9.81 \times 3.078 \times 510$ kW
$= 15399.5$ kW Ans:

(ii) Resultant force on the bucket :

peripheral speed of the wheel, $u = K_u\sqrt{2gH}$
$= 0.46\sqrt{2\times 9.81\times 510} = 46$ m/s

From figure at inlet to turbine:

$V_{w_1} = V_1 = 98$ m/s

$V_{r_1} = V_1 - u_1 = 98 - 46 = 52$ m/s [∵ $u_1 = u_2 = u$]

At exit from the turbine :

The blade angle at exit, $\phi = 180° - 165° = 15°$

$V_{r_2} = 0.85\, V_{r_1}$ ....(.... given)

or, $V_{r_2} = 0.85 \times 52 = 44.2$ m/s

As $V_{r_2}\cos\phi$ is less than blade speed U, the velocity triangle at outlet will be as shown in figure

∴ $V_{w_2} = u_2 - V_{r_2}\cos\phi$ [∵ $u_1 = u_2 = u$]
$= 46 - 44.2 \times \cos 15° = 3.31$ m/s

(ii) Resultant force on the bucket,

$$F = \rho Q (V_{w_1} - V_{w_2}) \qquad (\because \theta > 90^\circ)$$

$$= 1000 \times 3.078 (98 - 3.71) = 291455.8\ N. \quad \underline{\underline{Ans:}}$$

(iii) Brake Power, P:

Power developed by the wheel $= F \times u$

$= 291455.8 \times 46$ N-m/s or J/s or W

$= 291455.8 \times 46 \times 10^{-3}$ kW

$= 13406.97$ kW

$\therefore$ Brake Power (Power Produced at the shaft),

$P = 13406.97 \times (1 - 0.03)$

$= 13004.76$ kW $\underline{\underline{Ans:}}$

(iv) Overall efficiency, η_o:

$$\eta_o = \frac{\text{Brake Power}}{\text{Water Power}} = \frac{13004.76}{15399.5}$$

$= 0.844$ or 84.4%. $\underline{\underline{Ans:}}$

Imp.

(5) A Pelton wheel of 1.1 m mean bucket diameter works under a head of 550 m. The deflection of jet is 165° and its relative velocity is reduced over the bucket by 15 percent due to friction. If the diameter of jet is 100 mm and the water is to leave the bucket without any whirl, determine:

(i) Rotational speed of wheel.

(ii) Ratio of bucket speed to jet velocity

(iii) Impulsive force and power developed by the wheel,

(iv) Available power (water power),

(v) Power input to buckets, and

(vi) Efficiency of the wheel with power input to bucket as reference input. Take $C_v = 0.97$.

Solution: mean bucket diameter, $D = 1.1$ m

Net head, $H = 550$ m

Deflection of jet $= 165°$

Reduction of relative velocity due to friction = 15%

Diameter of Jet, $d = 100$ mm $= 0.1$ m

co-efficient of velocity, $C_v = 0.97$

(i) Rotational speed of wheel, N:

Velocity of Jet, $V_1 = C_v\sqrt{2gH}$

$= 0.97\sqrt{2\times9.81\times500} = 96.07$ m/s

Let, bucket speed $u_1 = u_2 = u$

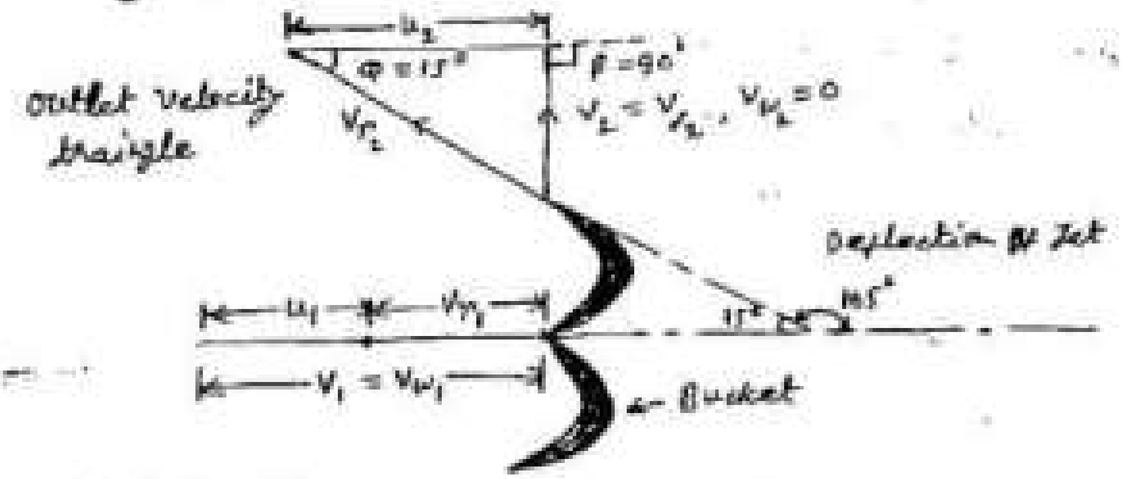
Relative velocity at inlet, $V_{r_1} = V_1 - u_1 = 96.07 - u$

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

$= 0.85(96.07-u)$ ---- (i)

The blade angle at exit, $\phi = 180 - 165^\circ = 15^\circ$

As the Jet leaves the bucket without any whirl, the velocity triangle at outlet will be as shown in figure. ($\beta = 90^\circ$),



$V_{r_2}\cos\phi = u$

or, $V_{r_2}\cos15^\circ = u$ ---- (ii)

From (i) and (ii), we get,

$0.85(96.07-u)\cos15^\circ = u$

or, $0.85(96.07-u)\times0.966 = u$

or, $78.88 - 0.821u = u$

or, $u = 43.31$ m/s

Also, $u = \frac{\pi DN}{60}$, or, $43.31 = \frac{\pi\times1.1\times N}{60}$

$\therefore$ Rotational speed of wheel, $N = \frac{43.31\times60}{\pi\times1.1} = 752$ r.p.m

$\frac{u}{V_1} = \frac{43.31}{96.07} = 0.4508$ Ans

(iii) Impulsive force and power developed by the wheel:

Discharge through the wheel

$$Q = \frac{\pi}{4} \times d^2 \times V_1 = \frac{\pi}{4} \times (0.1)^2 \times 96.07$$

$$= 0.7545 \ m^3/s$$

Impulsive force on the buckets,

$$F = \rho Q (V_{w_1} \pm V_{w_2}) = \rho Q V_{w_1} \quad \left(\because \begin{matrix} V_{w_2} = 0 \\ V_{w_1} = V_1 \end{matrix}\right)$$

$$= 1000 \times 0.7545 \times 96.07$$

$$= 72484.8 \ N \quad \text{Ans}$$

Power developed by the wheel,

$$P = F \times u = 72484.8 \times 43.31$$

$$= 3139316.7 \ Nm/s \text{ or } J/s \text{ or } W$$

$$= 3139.3 \ kW \quad \text{Ans}$$

(iv) Available power (water power):

Available power (water power) $= wQH$

$$= 9.81 \times 0.7545 \times 500$$

$$= 3700.8 \ kW \ \text{(Ans)}$$

(v) Power input to buckets:

$$= \frac{1}{2} m V_1^2 = \frac{1}{2} (\rho Q) \times V_1^2$$

$$= \frac{1}{2} \times 1000 \times 0.7545 \times (96.07)^2$$

$$= 3481808 \ Nm/s \text{ or } J/s \text{ or } W$$

$$= 3481.8 \ kW \quad \text{Ans}$$

(vi) Efficiency of wheel η_{wheel}:

$$\eta_{wheel} = \frac{\text{Power developed by wheel}}{\text{Power input to buckets}}$$

$$= \frac{3139.3}{3481.8} = 0.9016 \text{ or } 90.16\% \quad \text{Ans}$$

10) Design a Pelton wheel for a head of 350 m at a speed of 300 r.p.m. Take overall efficiency of the wheel as 85% and ratio of jet to wheel diameter as 1/10.

Solution : Given; H = 350 m, N = 300 r.p.m, η_o = 85% = 0.85 and $\frac{d}{D} = \frac{1}{10}$ or d = D/10

(1) Size of the wheel

Let, D = Diameter of wheel

We know that velocity of the jet,

$$V = C_v \times \sqrt{2gH} = 0.98 \times \sqrt{2 \times 9.81 \times 350}$$

$$= 81.2 \text{ m/s} \quad [\text{Assuming } C_v = 0.98]$$

and Peripheral velocity of the wheel,

$$v = 0.46\,V = 0.46 \times 81.2 = 37.4 \text{ m/s}$$

We also know that peripheral velocity (v)

$$37.4 = \frac{\pi D N}{60} = \frac{\pi \times D \times 300}{60} = 15.7 D$$

$$\therefore D = 37.4/15.7 = 2.4 \text{ m} \quad \text{Ans:-}$$

(2) Diameter of the jet,

We know that dia of the jet,

$$d = \frac{\text{Dia of wheel}}{10} = \frac{2.4}{10} = 0.24 \text{ m}$$

$$= 240 \text{ mm} \quad \text{Ans:}$$

(3) Width of the buckets

$$= 5 \times d = 5 \times 0.24 = 1.2 \text{ m} \quad \text{Ans:}$$

(4) Depth of the buckets,

$$= 1.2 \times d = 1.2 \times 0.24 = 0.48 \text{ m} \quad \text{Ans:}$$

(5) No of buckets,

$$= \frac{D}{2d} + 15 = \frac{2.4}{2 \times 0.24} + 15 = 20 \quad \text{Ans:}$$

Q. [illegible]

Head [illegible] = 150 m

Power to be developed = 600 kW,

Speed of wheel = 360 r.p.m.

Assume, reasonably, the missing data

<u>Solution</u>: Given; $H = 150$ m, $P = 600$ kW and $N = 360$ r.p.m.

(1) Diameter of the wheel

Let ... we know that velocity of the jet,

$$V = c_v \times \sqrt{2gH} = 0.985 \times \sqrt{2 \times 9.81 \times 150}$$
$$= 53.5 \text{ m/s}$$

(Assuming $c_v = 0.985$)

and peripheral velocity,

$$v = 0.46V = 0.46 \times 53.4 = 24.6 \text{ m/s}$$

(Assuming $v = 0.46V$)

We also know that peripheral velocity, (v),

$$24.6 = \frac{\pi D N}{60} = \frac{\pi \times D \times 360}{60} = 18.85 D$$

or, $D = 24.6/18.85 = 1.3$ m <u>Ans:</u>

(2) Diameter of the jet,

Let, d = dia of the jet,

$$0.85 = \frac{P}{w \cdot Q \cdot H} = \frac{600}{9.81 \times Q \times 150} =$$
$$= \frac{0.408}{Q} \quad (\text{Assuming } \eta_o = 85\%)$$

$$\therefore Q = 0.408/0.85 = 0.48 \text{ m}^3/\text{s}$$

Now the discharge through the wheel must be equal to the discharge through the jet, i.e.,

$$Q = V \times \frac{\pi}{4} \times d^2$$

or $0.48 = 53.4 \times \frac{\pi}{4} \times (d)^2 = 41.9 d^2$

$\therefore \quad d^2 = 0.458/41.7 = 0.011 \quad \text{or } d = 0.105 \text{ m}$

$= 105 \text{ mm}$ **Ans:**

(3) Width of the buckets,

$= 5 \times d = 5 \times 0.105 = 525 \text{ mm}$ **Ans:**

(4) Depth of buckets $= 1.2 \times d$

$= 1.2 \times 105 = 210 \text{ mm}$ **Ans:**

(5) No of the buckets,

$= \frac{D}{2d} + 15 = \frac{1.3}{2 \times 0.105} + 15$

$= 21$ **Ans:**

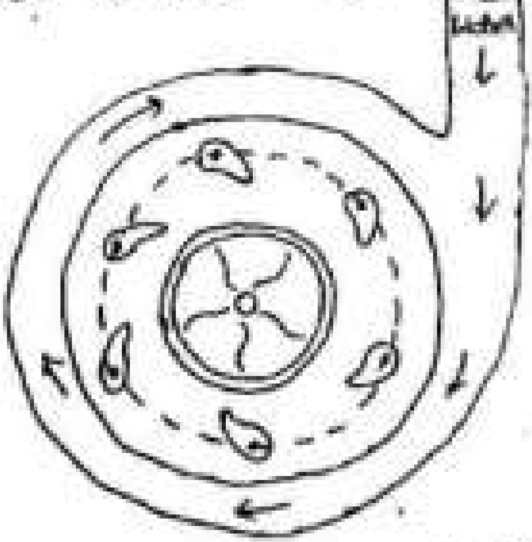
Reaction turbine

(i) **Introduction:** In a reaction turbine, the water enters the wheel under pressure and flows over the vanes. As the water, flowing over the vanes, is under pressure, therefore wheel of the turbine run full and may be submerged below the tail race or may discharge into the atmosphere. The pressure head of water while flowing over vanes is converted into velocity head and is finally reduced to the atmospheric pressure, before leaving the wheel.

(ii) **main components of reaction turbine:**

(1) spiral casing.
(2) Guide mechanism.
(3) Turbine runner.
(4) Draft tube.

(1) **spiral casing:** The water from a pipe line, is distributed around ~~a~~ the guide ring ~~around~~ in a casing. This casing is designed in such a way that its cross sectional area goes on reducing uniformly around the circumference. The cross sectional area is maximum at the entrance, and minimum at the tip as shown in figure. As a result of this casing will be spiral shape.



casing of reaction turbine

The spiral casing are provided with inspection holes and pressure gauges. The material of a casing depends upon the head of water, under which the turbine is working as discussed below:

concrete .... upto 30 m.

Welded rolled steel plate ... upto 100 m.

cast steel .... more that 100 m.

(ii) Guide mechanism:

The guide vanes are fixed between two rings in the form of a wheel. This wheel is fixed in the spiral casing. The guide vanes are properly designed in order to:

(1) Allow the water to enter the runner without shock (This is done by keeping the relative velocity, at inlet of the runner, tangential to the vane angle)

(2) Allow the water to flow over them, without forming eddies.

(3) Allow the required quantity of water to enter the turbine. (This is done by adjusting the opening of the vanes).

All the guide vanes can rotate about their respective pivots, which are connected to the Regulating ring by some mechanical means. The guide vanes may be closed or opened by rotating the Regulating shaft by means of two Regulating rods. The guide vanes may be closed or opened by rotating the Regulating shaft, thus allowing the required quantity of water to flow according to need. The Regulating shaft is operated by means of a governor, whose function is to govern the turbine

(... speed constant at varying loads)

The guide vanes are generally made of cast steel.

(ii) **Turbine Runner:**

The runner of a reaction turbine consists of runner blades fixed either to a shaft or rings, depending upon the type of turbine. The blades are properly designed, in order to allow the water to enter and leave the runner without shock.

The runner is keyed to shaft, which may be vertical or horizontal. If the shaft is vertical it is called vertical turbine. Similarly if the shaft is horizontal, it is called horizontal turbine. The surface of the runner is made very smooth. The runner may be cast in one piece or may be made of separate steel plates and welded together. For low heads the runner may be made of cast iron. But for high heads, the runner is made of steel or alloy. When the water is chemically impure, the runner is made of special alloy.

(iii) **Draft tube:** The water are passing through the runner, flows down through a tube is called draft tube. It is generally drowned approximately 1 m below the tail race level.

A draft tube has the following functions:

It increases the head of water by an amount equal to height of the runner outlet above the tail race.

It increases efficiency of the turbine ... [illegible]

(B) Classification of Reaction turbines:

Depending upon the direction of flow of water through the wheel it may be classified:-

(1) Radial flow turbines.
(2) Axial flow turbines.
(3) Mixed flow turbines

(I) Radial flow turbines:

In such turbines the flow of water is radial (along the radius of the wheel). It is further sub devided by -

(1) Inward flow turbines: In such turbines, the water enters the wheel at the outer periphery, and then flows inwards (towards the centre of the wheel)

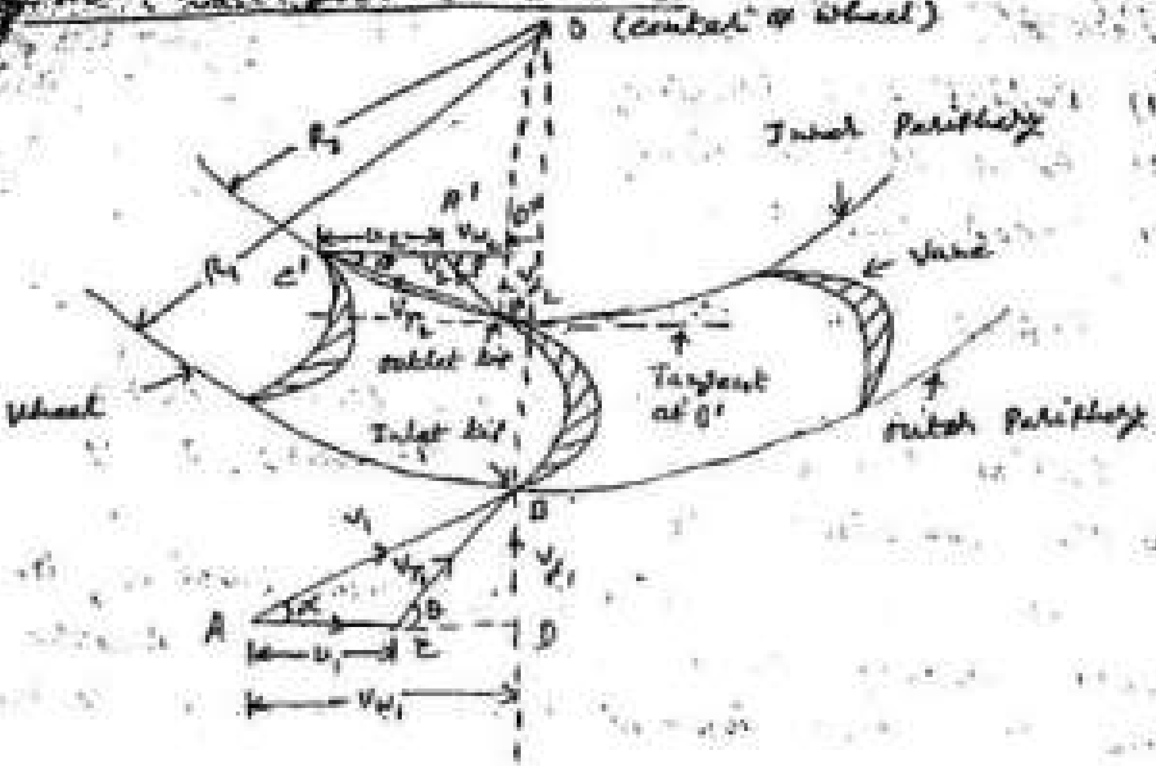
(2) Outward flow turbines: In such turbines water enters at the centre of the wheel, and then flows outwards (towards the outer periphery of the wheel).

(II) **Axial flow turbines**: In such turbines, the water flows parallel to the axis of the wheel. Such turbines are also called parallel flow turbines.

(III) **Mixed flow turbines**: These are latest types of turbines, in which the flow is partly radial and partly axial.

(ii) Working and efficiency of Francis turbine:

Work done by the runner:



$$\text{Workdone} = \rho Q (V_{w_1} u_1 \pm V_{w_2} u_2)$$

$$= \frac{wQ}{g} (V_{w_1} u_1 \pm V_{w_2} u_2)$$

Where, Q = Discharge through the runner, m^3/s

The maximum output under given conditions is obtained when $V_{w_2} = 0$

Thus maximum workdone is given by,

$$\text{Workdone} = \frac{wQ}{g} (V_{w_1} u_1)$$

This discharge in this case is radial. For radial discharge, the absolute velocity at exit is radial.

(b) Hydraulic efficiency (η_h):

If H is the net head, then input to the turbine $= wQH$

$$\eta_h = \frac{\text{Power developed by the runner}}{\text{Power supplied to the turbine (Water Power)}}$$

$$= \frac{\frac{wQ}{g}(V_{w_1} u_1)}{wQH} = \frac{V_{w_1} u_1}{gH}$$

However, if the velocity of whirl at the exit is not zero, then

$$\eta_h = \frac{V_{w_1} u_1 \pm V_{w_2} u_2}{gH}$$

The hydraulic efficiency of the Francis turbine varies from 85 to 90 percent.

(ii) mechanical efficiency (η_m) :

$$\eta_m = \frac{\text{Shaft Power } (P)}{\text{Power Developed by the Runner}}$$

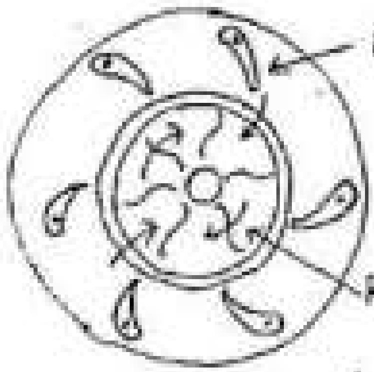
(iii) overall efficiency (η_o) :

$$\eta_o = \frac{\text{Shaft Power}}{\text{Water Power}} = \frac{P}{WQH}$$

and $\eta_o = \eta_h \times \eta_m$

The overall efficiency varies from 80 to 90 percent.

(1) Inward flow reaction turbine :



An inward flow reaction turbine, as the name indicates, is that reaction turbine in which the water enters in wheel at the outer periphery and then flows inwards over the vanes (towards the centre of the wheel) as shown in figure.

An inward flow reaction turbine in its simplest form, consists of fixed guide blades, which guide the water to enter into the revolving wheel at correct angle, for the shockless entry of water. (This is done by adjusting the vane angle tangentially to the relative velocity of the water and the revolving wheel.) The water while sliding over the vanes, exerts some force on the revolving wheel, to which the vanes are fixed. This force causes the revolving wheel to revolve.

It may be noted that whenever the load on the turbine is decreased, it causes the shaft to rotate at a higher speed. The higher centrifugal force, which increases due to higher speed, tends to reduce the quantity of water flowing over the vanes, and thus the velocity of water at the entry is also reduced. It will ultimately lead to reduce the power produced by the turbine. This is the advantage of an inward flow reaction turbine, that it adjusts automatically according to the required load on the turbine. The highest efficiency is obtained when the velocity of the leaving water is as small as possible.

Now the work done or any other detail of the turbine runner may be found out by drawing the inlet and outlet velocity triangles, as shown in figure.

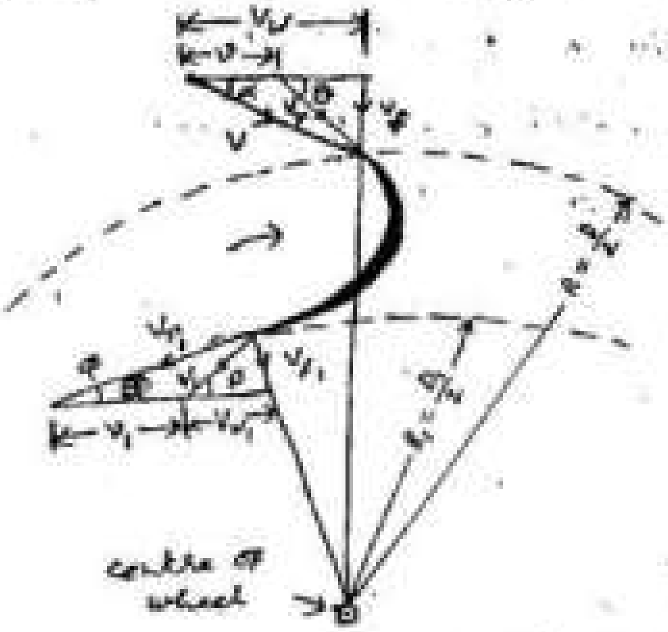
Let V = Absolute velocity of the entering water,

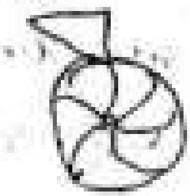
D = Outer diameter of the wheel,

N = Revolutions of the wheel per minute,

v = Tangential velocity of wheel at inlet (also known as peripheral velocity of wheel at inlet),

$$v = \frac{\pi D N}{60}$$





"Triangle of velocities for inward flow Reaction Turbine"

V_r = Relative velocity of water to the wheel, at inlet,

V_f = Velocity of flow at inlet,

$V_1, \beta_1, v_1, V_{r_1}, V_{f_1}$ = corresponding values at the outlet.

α = Angle at which the water enters the wheel (also known as guide blade angle),

β = Angle at which water leaves the wheel,

θ = Angle at the blade tip at inlet (Also known as vane angle at outlet),

H = Total head of water, under which the turbine is working,

W = Weight of the water entering the wheel in kN/s

From the inlet triangle, we find that,

$$V_w = V\cos\alpha \text{ and } V_f = V\sin\alpha$$

and from the outlet triangle, we find that,

$$V_{w_1} = V_1\cos\beta \text{ and } V_{f_1} = V_1\sin\beta$$

We know that the force per kN of water,

$$= \frac{1}{g} \times (\text{change of velocity of whirl})$$

$$= \frac{1}{g} \times (V_w + V_{w_1})$$

[illegible]

[illegible] work done per kg of water

$= \text{Force} \times \text{Distance}$

= (Velocity of whirl at inlet × Tangential velocity of wheel at inlet) − (Velocity of whirl at outlet × Tangential velocity of wheel at outlet)

$$= \frac{1}{g}(V_w \cdot v - V_{w_1} v_1) = \frac{V_w \cdot v}{g} - \frac{V_{w_1} \cdot v_1}{g} \quad \text{---(1)}$$

Notes: (1) If there is no loss of energy, then,

$$\frac{V_w \cdot v}{g} - \frac{V_{w_1} \cdot v_1}{g} = H - \frac{V_1^2}{2g}$$

(2) If the discharge of the turbine is radial, then $\beta = 90^\circ$, $V_{w_1} = 0$ and $V_1 = V_{f_1}$

Work done per kg of water $= \dfrac{V_w \cdot v}{g}$

and work done $\dfrac{V_w \cdot v}{g} = H - \dfrac{V_1^2}{2g} = H - \dfrac{V_{f_1}^2}{2g}$

(1) Q. An inward flow reaction turbine, having an external diameter of 1.5 metre runs at 400 r.p.m. The velocity of flow at inlet is 10 m/s. If the blade angle is 15°, find (a) absolute velocity of water, (b) velocity of whirl at inlet, (c) inlet vane angle of the runner, and (d) Relative velocity at inlet.

Solution: Given: $D = 1.5$ m, $N = 400$ r.p.m, $V_f = 10$ m/s and $\alpha = 15°$

(a) Absolute velocity of water:

From the inlet velocity triangle, we find that absolute velocity of water,

$$V = \frac{V_f}{\sin 15°} = \frac{10}{0.2588} = 38.64 \text{ m/s} \quad \underline{\text{Ans.}}$$

(b) Velocity of whirl at inlet:

From the inlet velocity triangle, we find that the velocity of whirl at inlet,

$$V_w = V \cos 15° = 38.64 \times 0.9659 = 37.32 \text{ m/s} \quad \underline{\text{Ans.}}$$

(c) Inlet vane angle of the runner,

Let, θ = Inlet vane angle of the runner,

$$v = \frac{\pi D N}{60} = \frac{\pi \times 1.5 \times 400}{60} = 31.42 \text{ m/s}$$

and from the inlet velocity triangle, we find that

$$\tan\theta = \frac{V_f}{V_w - v} = \frac{10}{37.32 - 31.42} = 1.695$$

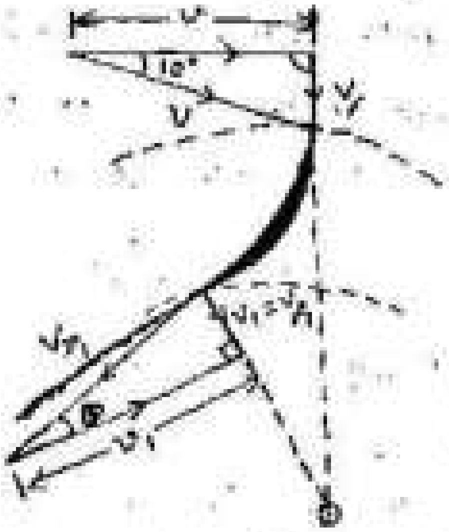
or, $\theta = 59.5°$ Ans.

(d) Relative velocity at inlet,

From the inlet velocity triangle, we also find that Relative velocity at inlet,

$$V_r = \frac{V_f}{\sin 59.5°} = \frac{10}{0.8616} = 11.61 \text{ m/s} \quad \underline{\text{Ans.}}$$

① Q9. An inward flow reaction turbine has outer and inner diameters of the wheel as 1 metre and 0.5 metre respectively. The vanes are radial at inlet and the discharge is radial at outlet and the water enters the vanes at an angle of 10°. Assuming the velocity of flow to be constant and equal to 3 m/s, find (a) speed of the wheel, and (b) vane angle at outlet.



Solution: Given; $D = 1$ m, $D_1 = 0.5$ m, $\alpha = 10°$, and $V_f = V_{f_1} = 3$ m/s

(i) Speed of the wheel:

Let, N = Speed of the wheel.

Since the vanes are radial at inlet and outlet, therefore the velocities of whirl at inlet and outlet will be zero. And the shapes of the two triangles will be as shown in figure.

From the inlet triangle of velocities, we find that tangential velocity of the wheel at inlet,

$$v = \frac{V_f}{\tan 10°} = \frac{3}{0.1763} = 17 \text{ m/s}$$

We also know that the tangential velocity of wheel at inlet (v),

$$17 = \frac{\pi D N}{60} = \frac{\pi \times 1 \times N}{60} = 0.0524 N$$

or, $N = 324.4$ r.p.m. Ans.

(ii) Vane angle at outlet,

Let, ϕ = Vane angle at outlet,

We know that tangential velocity of wheel at outlet,

$$v_1 = \frac{\pi D_1 N}{60}$$

$$= \frac{\pi \times 0.5 \times 324.4}{60} = 8.5 \text{ m/s}$$

From the outlet triangle of velocities, we find that

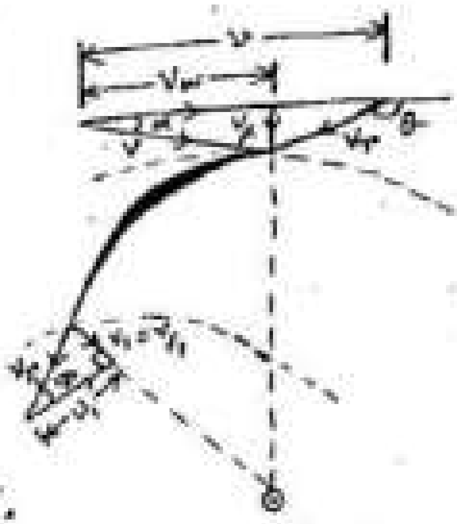
$$\tan\alpha = \frac{V_{f1}}{V_1} = \frac{3}{8.5} = 0.3529$$

or, $\alpha = 19.4°$ **Ans:**

(2) (7) An inward flow reaction turbine is supplied water at the rate of 600 lit/sec with a velocity of flow of 6 m/sec. The velocity of periphery and velocity of whirl at inlet is 24 m/s and 18 m/s respectively. Assuming the discharge to be radial at outlet, and the velocity of flow to be constant, find

(1) Vane angle at inlet, and

(2) head of water on the wheel.



Solution: Given; $Q = 600$ lit/sec $= 0.6\ m^3/sec$,

$V_f = 6$ m/s, $v = 24$ m/s,

$V_w = 18$ m/s and $V_{f1} = V_f$

(1) Vane angle at inlet:

Let, θ = Vane angle at inlet.

From the triangle of velocities, we find that

$$\tan(180° - \theta) = \frac{V_f}{v - V_w} = \frac{6}{24-18} = 1.0$$

$$180° - \theta = 45° \quad \text{or,} \quad \theta = 135° \quad \textbf{Ans:}$$

(2) Head of water on the wheel:

Let, H = Head of water on the wheel.

We know that, $\frac{V_w \cdot v}{g} = H - \frac{V_1^2}{2g}$ (∵ $V_1 = V_{f1}$)

or, $\frac{18 \times 24}{9.81} = H - \frac{V_{f1}^2}{2g}$

or, $44.0 = H - \frac{V_f^2}{2 \times 9.81} = H - \frac{(6)^2}{2 \times 9.81}$

or, $44 = H - 1.8$

or, $H = 45.8$ m **Ans:**

(ii) (4) An inward flow reaction turbine is working under a head of 25 metres and running at 300 revolution per minute. The velocity of periphery of the wheel is 30 m/s and velocity of flow is 4 m/s. If the hydraulic losses are 20% of the available head, and the discharge is radial, find:

(a) Guide blade at inlet.
(b) Wheel angle at inlet.
(c) diameter of the wheel.

Solution: Given, $H = 25$ m, $N = 300$ r.p.m., $u = 30$ m/s, $V_f = 4$ m/s and hydraulic losses are 20% of the available head $= 0.20 \times 25 = 5$ m.

(a) Guide blade angle at inlet

Let, α = Guide blade angle at inlet.

Since the discharge is radial, therefore velocity of whirl at outlet is zero.

We know that $\frac{V_w \cdot u}{g} = H - \frac{V_1^2}{2g}$ $\left(\because \text{hydraulic losses} = V_1^2/2g\right)$

$$\Rightarrow \frac{V_w \times 30}{9.81} = 25 - 5 = 20$$

$$\Rightarrow 3.06\, V_w = 20 \quad \text{or,} \quad V_w = 20/3.06 = 6.54 \text{ m/s}$$

Since V_w (6.54) is less than u (30), therefore shape of the inlet triangle will be as shown in fig.

Now from the inlet triangle, we find

$$\tan\alpha = \frac{V_f}{V_w} = \frac{4}{6.54} = 0.6116$$

or, $\alpha = 31.4°$ **Ans.**

(b) Wheel angle at inlet.

Let, θ = wheel angle at inlet.

From the inlet triangle of velocities, we also find that, $\tan(180-\theta) = \frac{V_f}{u - V_w} = \frac{4}{30-6.54} = 0.1705$

$\therefore$ $180 - \theta = 9.7°$

or, $\theta = 170.3°$ **Ans.**

(c) Diameter of the wheel

Let, D = Dia of the wheel.

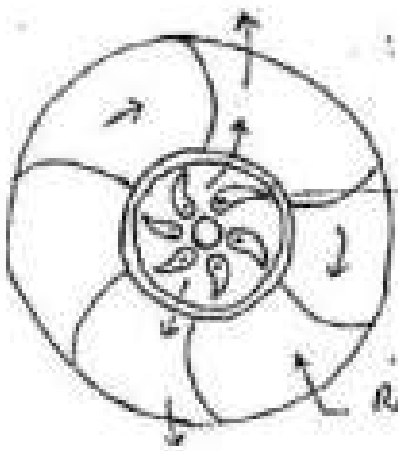
We know that the velocity of periphery at inlet (u),

$$30 = \frac{\pi D N}{60} = \frac{\pi \times D \times 300}{60} = 15.7 D$$

$\therefore D = 30/15.7 \approx 1.91$ m Ans:

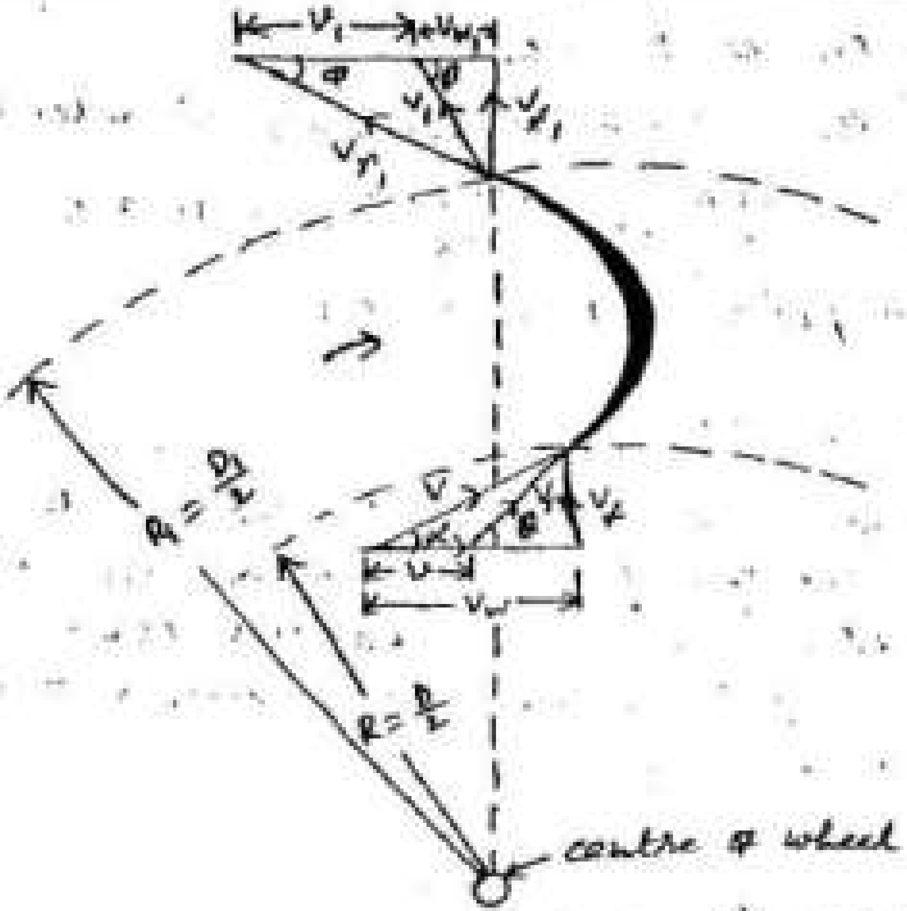
(B) Outward flow Reaction turbines:

An outward flow reaction turbine, as the name indicates, is that reaction turbine, in which the water enters at the centre of the wheel and then flows outwards over the vanes (towards the outer periphery of the wheel) as shown in figure.



An outward flow reaction turbine, in its simplest form, consists of fixed guide blades which guide the water to enter into the revolving wheel at correct angle, i.e. for shockless entry of water (This is done by adjusting the vane angle tangentially to the relative velocity of water and the revolving wheel). The water, while gliding over the vanes exerts some force on the revolving wheel to which the vanes are fixed. This force causes the revolving wheel to revolve. The only difference, between the inward and outward flow reaction turbine is that in case of an inward flow reaction turbine, the revolving wheel is inside the fixed guide blades whereas in the case of an outward flow reaction turbine, the revolving wheel is outside the fixed guide blades as shown in figure.

TRIANGLE OF VELOCITIES OF OUTWARD FLOW REACTION TURBINE



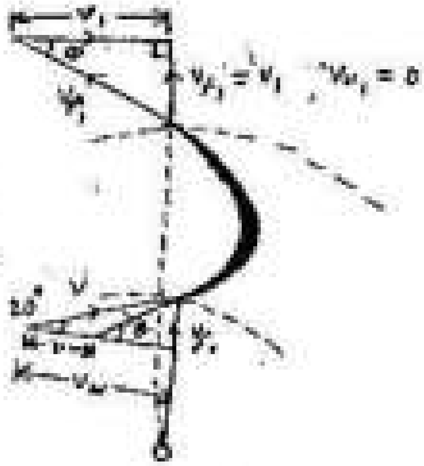
It may be noted, that whenever the load on the turbine is decreased, it causes the shaft to rotate at a higher speed. The centrifugal force, which increases due to the higher speed, tends to increase the quantity of water flowing over the vanes, and thus the wheel tends to run faster and faster. It is the only disadvantage of an outward flow reaction turbine. Thus every outward flow reaction turbine has to be governed by a turbine governor.

All the notations and relations for finding out various angles and other data will hold good for an outward turbine also.

REACTION TURBINE

47. An outward flow reaction turbine has inner and outer diameters of the wheel as 1 metre and 2 metres respectively. The water enters the vanes at an angle of 20° and leaves the vanes radially. If the velocity of flow remains constant at 10 m/s and the speed of the wheel be 300 r.p.m, find the vane angles at inlet and outlet.

Solution: Given: $D = 1\ m$, $D_1 = 2\ m$, $\alpha = 20°$; $V_f = V_{f_1} = 10$ m/s and $N = 300$ r.p.m.



Let, θ = Vane angle at inlet

We know that velocity of periphery at inlet,

$$v = \frac{\pi D N}{60} = \frac{\pi \times 1 \times 300}{60} = 15.71\ m/s$$

From inlet triangle of velocities, we find that the velocity of whirl at inlet,

$$V_w = \frac{V_f}{\tan 20°} = \frac{10}{0.364} = 27.5\ m/s$$

and $\tan\theta = \dfrac{V_f}{V_w - v} = \dfrac{10}{27.5 - 15.71} = 0.8482$

or, $\theta = 40.3°$ **Ans:**

Vane angle at outlet,

Let ϕ = vane angle at outlet

We know that the velocity of periphery at outlet,

$$v_1 = \frac{\pi D_1 N}{60} = \frac{\pi \times 2 \times 300}{60} = 31.42\ m/s$$

From the outlet triangle of velocities, we find that

$$\tan\phi = \frac{V_{f_1}}{v_1} = \frac{10}{31.42}$$

$$= 0.3183$$

or, $\phi = 17.7°$ **Ans:**

⑩ Discharge of a Reaction turbine :

The discharge of a reaction turbine may be found out either from the gross energy supplied to the turbine at inlet or outlet or from the actual velocity of flow at inlet or outlet as discussed below.

The gross power supplied to the turbine,

$= w Q H$ kW.

From the velocity of flow,

V_f = velocity of flow at inlet,
D = dia of the wheel at inlet,
and b = width of the wheel at inlet,

We know that water entering the wheel

$Q = \pi D b V_f$

Similarly water leaving the wheel,

$Q = \pi D_1 b_1 V_{f_1}$

When water entering and leaving is equal,

$\pi D b V_f = \pi D_1 b_1 V_{f_1}$

⑪ (a) An Inward flow reaction turbine has external and internal wheel diameters of 1.0 m and 0.5 m respectively. The water entering the wheel with a velocity of 30 m/s at an angle of 10°. The width of the wheel at inlet and outlet is 150 mm and 300 mm respectively. The vane angle is 90° and 25° at outlet. Determine

(i) tangential velocity of runner at inlet, and

(ii) absolute velocity of water at outlet.

Solution: Given, $D = 1$ m, $D_1 = 0.5$ m, $V = 30$, $\alpha = 10°$,
$b = 150$ mm $= 0.15$ m, $b_1 = 300$ mm $= 0.3$ m,
$\theta = 90°$, and $\phi = 25°$

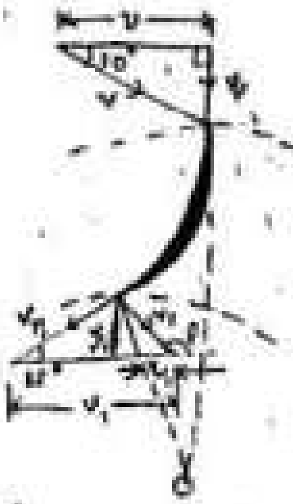
(i) Tangential velocity of runner at inlet,

From the inlet triangle of velocities,

$v = V\cos\alpha = 30 \times \cos 10°$

$= 30 \times 0.9848 = 29.5$ m/s **Ans.**

(ii) Absolute velocity of water at outlet.



From inlet triangle, find the velocity of flow at inlet,

$V_f = V \sin\alpha$
$= 30 \sin 10^\circ$
$= 30 \times 0.1736$
$= 5.21$ m/s

and tangential velocity of runner at outlet,

$$v_1 = v \times \frac{D_1}{D} = 29.5 \times \frac{0.5}{1.0} = 14.75 \text{ m/s}$$

Since the discharge of water at inlet and outlet is equal, therefore,

$$\pi D b V_f = \pi D_1 b_1 V_{f_1}$$

or, $\pi \times 1.0 \times 0.15 \times 5.21 = \pi \times 0.5 \times 0.3 \times V_{f_1}$

or, $V_{f_1} = \frac{0.7815}{0.15} = 5.21$ m/s

Now from the outlet triangle of velocities, we find that,

$$\tan\beta = \frac{V_{f_1}}{v_1 - V_{w_1}} \quad \text{or,} \quad \tan 25^\circ = \frac{5.21}{14.75 - V_{w_1}}$$

or, $14.75 - V_{w_1} = \frac{5.21}{\tan 25^\circ} = \frac{5.21}{0.4663} = 11.17$

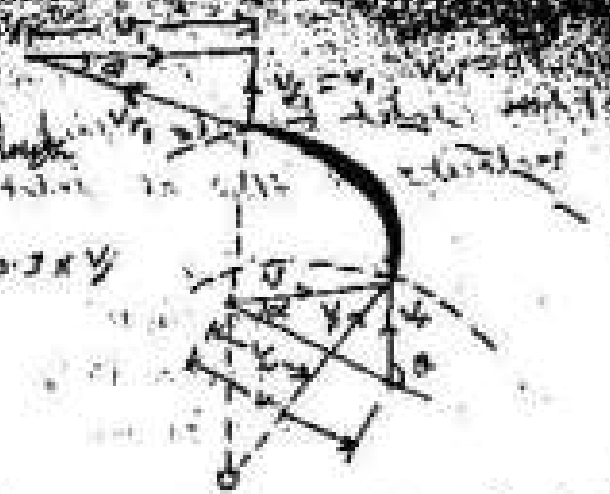
or, $V_{w_1} = 3.58$ m/s

and absolute velocity of water at outlet,

$$V_1 = \sqrt{V_{f_1}^2 + V_{w_1}^2} = \sqrt{(5.21)^2 + (3.58)^2} = 6.32 \text{ m/s} \quad \underline{\text{Ans.}}$$

(13) (a) An outward flow reaction turbine has internal and external diameters of 2.4 metres and 3 metres respectively. The turbine has a radial discharge of 6 m³/s, and is running at 200 r.p.m. The total head on the turbine, is 40 metres and width of the wheel at inlet and outlet is 300 mm. Neglecting thickness of the vanes, find (i) velocity of flow at inlet, (ii) velocity of flow at outlet, and (iii) velocity of whirl at inlet.

<u>Solution</u>: Given: $D = 2.4$ m, $D_1 = 3.0$ m, $Q = 6$ m³/s, $N = 200$ rpm, $H = 40$ m and $b = b_1 = 300$ mm $= 0.3$ m.



We know that the discharge through the vanes at inlet (Q),

$$Q = \pi D b V_f = \pi \times 2.4 \times 0.3 \times V_f$$

$$= 2.26\, V_f$$

$$\therefore V_f = 2.65 \text{ m/s} \quad \underline{\text{Ans.}}$$

(ii) Velocity of flow at outlet

Let, V_{f_1} = Velocity of flow at outlet,

Discharge through the vanes at outlet (Q),

$$Q = \pi D_1 b_1 V_{f_1} = \pi \times 3.0 \times 0.3 \times V_{f_1}$$

$$= 2.83\, V_{f_1}$$

$$\therefore V_{f_1} = 2.12 \text{ m/s} \quad \underline{\text{Ans.}}$$

(iii) Velocity of whirl at inlet,

Let, V_w = Velocity of whirl at inlet,

We know that the velocity of periphery at inlet,

$$u = \frac{\pi D N}{60} = \frac{\pi \times 2.4 \times 200}{60} = 25.13 \text{ m/s}$$

Since the discharge is radial, therefore velocity of whirl at the outlet is zero.

We know that, $\frac{V_w \cdot u}{g} = H - \frac{V_1^2}{2g}$

or, $\frac{V_w \times 25.13}{9.81} = 40 - \frac{(2.12)^2}{2 \times 9.81} \quad (\because V_1 = V_{f_1})$

$$2.56\, V_w = 39.8$$

or, $V_w = 39.8 / 2.56$

$$= 15.5 \text{ m/s} \quad \underline{\text{Ans.}}$$

(1) **Power Produced by a Reaction turbine :**

Some work is done per kN of water, when it flows over the vanes. The power produced by the turbine is given by the turbine :

$$P = wQH \text{ kW}$$

where, w = Specific weight of water,
Q = Discharge of the turbine in m^3/s
H = Head of water in metres.

Note : The power produced by a reaction turbine may also be found out from the relation :

P = Quantity of water flowing over the vane in kN
× work done per kN of water.

(2) **Efficiencies of a reaction turbine :**

In general, the term efficiency may be defined as the ratio of workdone to the energy supplied.
Following are the three types of efficiency of a turbine :
(1) Hydraulic (2) Mechanical (3) Overall efficiency.

(1) **Hydraulic efficiency :** It is the ratio of workdone on the wheel to the head of water (or energy) actually supplied to the turbine.

$$\eta_h = \frac{\text{Work done per kN of water}}{H} = \frac{\frac{V_w v}{g} - \frac{V_{w_1} v_1}{g}}{H}$$

If the discharge through the wheel is radial, then the velocity of whirl at outlet is zero i.e., $V_{w_1} = 0$

$$\therefore \eta_h = \frac{V_w v}{gH}$$

(2) **Mechanical Efficiency :** It is the ratio of the actual work available at the turbine to the energy imparted to the wheel. (in case of we know that total energy imparted to the wheel (in case of radial discharge)

= Weight of water in kN × Energy imparted per kN of water

$$= wQ \times \frac{V_w \times v}{g}$$

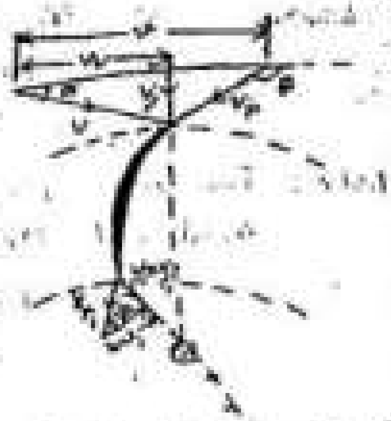
$$\therefore \text{Mechanical efficiency } \eta_m = \frac{P}{wQ \times \frac{V_w \times v}{g}}$$

where, P = Power available at the turbine.

(3) **Overall efficiency :** It is a measure of the performance of a turbine and is the ratio of power produced by the turbine to the energy actually supplied to the turbine i.e.,

$$\eta_o = \eta_h \times \eta_m = \frac{V_w v}{gH} \times \frac{P}{wQ \frac{V_w v}{g}} = \frac{P}{wQH}$$

Q) ... reaction turbine has ... velocity of whirl, velocity of flow and velocity of ... of 30 m/s, 3 m/s, and 24 m/s respectively. Assuming the discharge to be radial at outlet and hydraulic efficiency as 78%, determine the total head on the turbine and the inlet vane angle.



Solution: Given: $v = 30$ m/s, $V_f = 3$ m/s and $V_w = 24$ m/s respectively and $\eta_h = 78\% = 0.78$

Let, H = Total head on the turbine.

Since the discharge is radial at outlet, therefore velocity of whirl at outlet is zero.

We know that hydraulic efficiency (η_h),

$$0.78 = \frac{V_w \cdot v}{gH} = \frac{24 \times 30}{9.8\,H} = \frac{73.4}{H}$$

$\therefore$ $H = 73.4/0.78 = 94.1$ m. **Ans:**

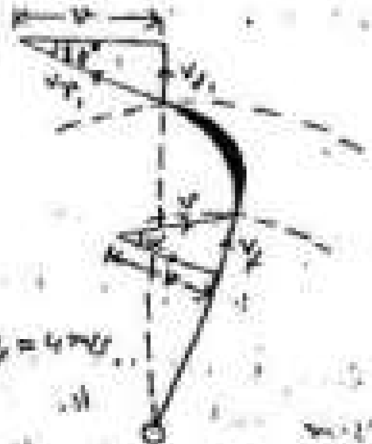
Let, θ = Inlet vane angle.

From the inlet triangle of velocities, we find that

$$\tan(180^\circ - \theta) = \frac{V_f}{v - V_w} = \frac{3}{30 - 24} = 0.5$$

or, $(180 - \theta) = 26.6^\circ$ or $\theta = 153.4^\circ$ **Ans:**

Q) An outward flow reaction turbine has tangential velocity at its inner rim of 12 m/s, and the ratio of radii is 0.8. The turbine is placed 1 m below the water surface in the tail race and the vane angles are 90° and 20° at inlet and outlet respectively. The radial velocity of flow at inlet and outlet is 4 m/s. Neglecting frictional losses and taking velocity of outflow as radial, find:



(a) Guide vane angle.
(b) Velocity of flow from guides,
(c) Total head of water, and
(d) Hydraulic efficiency

Solution: Given: $v = 12$ m/s, $\frac{r_1}{r_2} = 0.8$

or, $r_1 = 0.8\,r_2$, $\theta = 90^\circ$, $\phi = 20^\circ$ and $V_f = 4$ m/s.

Let, Guide blade angle = α

We know that, workdone per kN of water,

$\tan\alpha = \frac{V_f}{V_w} = \frac{V_f}{v} = \frac{4}{12} = 0.3333$, or, $\alpha = 18.4°$ Ans: (b)

Velocity of flow from guides;

From the inlet triangle of velocities, we also find that velocity of flow the guide (from the).

$$V = \sqrt{v^2 + V_f^2} = \sqrt{(12)^2 + (4)^2} \text{ m/s}$$

$$= 12.6 \text{ m/s} \quad \text{Ans:}$$

Total head of water,

we know that tangential velocity at the outer rim.

$$v_1 = v \times \frac{r_1}{r} = 12 \times \frac{1}{0.8} = 15 \text{ m/s}$$

and from the outlet triangle of velocities, we find that velocity of flow at outlet.

$$V_{f_1} = V_1 = v_1 \tan 20° = 15 \times 0.364 = 5.46 \text{ m/s}$$

We know that workdone per kN of water,

$$\frac{V_w \cdot v}{g} = \frac{12 \times 12}{9.81} = 14.7 \text{ kN-m/s}$$

∴ Total head of water at inlet

$= \text{Energy at outlet} + \text{work done}$

$= \left(\frac{V_1^2}{2g} + 1\right) + 14.68$ ... (The turbine is placed 1 m below the tail race.)

$$= \left[\frac{(5.46)^2}{2 \times 9.81} + 1\right] + 14.7 = 2.5 + 14.7 = 17.2 \text{ m} \quad \text{Ans:}$$

Hydraulic efficiency,

Since the turbine is placed 1 m below the water surface in the tail race, therefore net head of water available for the turbine,

$$\therefore H = 17.2 - 1 = 16.2 \text{ m}$$

and hydraulic efficiency,

$$\eta_h = \frac{V_w \cdot v}{gH} = \frac{12 \times 12}{9.81 \times 16.2} = 0.906$$

$$= 90.6\% \quad \text{Ans:}$$

[illegible] respectively [illegible]
[illegible] to be constant and radial discharge at outlet, find:

(a) Work done per kN of water,
(b) Power developed by the turbine,
(c) Head of water on the turbine, and
(d) Hydraulic efficiency of the turbine.

<u>Solution</u>: Given; $N = 180$ r.p.m, $D = 1$ m, $b = 135$ mm $= 0.135$ m
and $V_f = V_{f_1} = 7$ m/s

(a) Work done per kN of water.

We know that the tangential velocity of wheel at inlet,

$$v = \frac{\pi D N}{60} = \frac{\pi \times 1 \times 180}{60} = 9.42 \text{ m/s}$$

Since the discharge is radial at inlet, therefore velocity of whirl at inlet,

$$V_w = v = 9.42 \text{ m/s}$$

Moreover discharge is radial, so $V_{w_1} = 0$

We know that workdone per kN

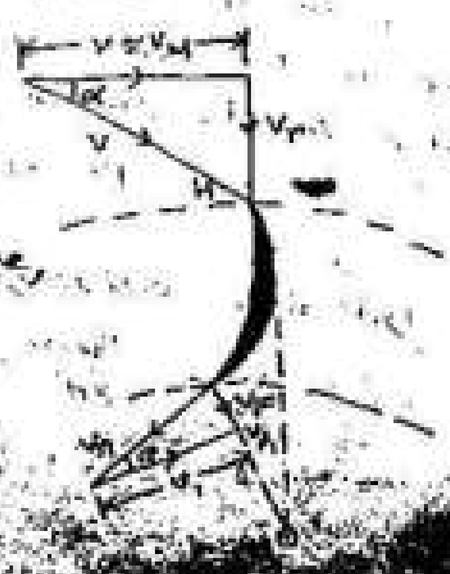
$$= \frac{V_w \cdot v}{g} = \frac{9.42 \times 9.42}{9.81}$$

$$= 9.05 \text{ kN-m} \quad \underline{\text{Ans:}}$$

(b) Power developed by the turbine,

We know that the discharge of the turbine,

$$Q = \pi D b V_f = \pi \times 1 \times 0.135 \times 7$$
$$= 1.272 \text{ m}^3/\text{s}$$



and power developed by the turbine,

P = weight of water flowing per second [illegible] kN × work done

Francis Turbine :

The Francis's turbine is an inward flow reaction, having radial discharge at outlet. It was the first turbine (inward flow reaction type) which was designed by Francis. It is widely used even in these days, for producing power under medium heads.

The modern Francis's turbine has a mixed flow (combination of radial and axial).

The size (or length) of the runner depends upon its specific speed. A Francis turbine having a higher specific speed, has a larger runner than that a lower specific speed. The runner of a Francis turbine may be cast in one piece, or made of separate steel plates welded together. The runners are made of cast iron for small output, cast steel for larger output and stainless steel or other non corrosive metal like bronze, when the water is chemically impure and there is a danger of corrosion. The blades of runner are carefully finished.

All the relations, for finding out the various angles and other data which were used for inward flow reaction turbine will hold for Francis turbine also.

Q. A Francis turbine, working under a head of 14 metres, has guide blade angle of 20° and radial vanes at inlet. The ratio of inlet and outlet diameters is 2 to 1. The velocity of flow of water at exit is 4 m/s. Assuming the velocity of flow to be constant, determine the peripheral velocity of wheel at inlet and the vane angle at outlet.

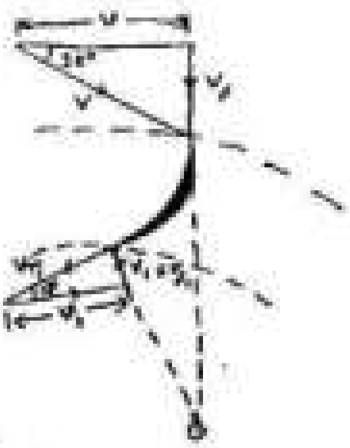
Solution: Given: $H = 14$ m, $\alpha = 20°$, $\theta = 90°$, $D = \frac{2}{1} D_1 = 2.0\, D_1$,

$V_{f_1} = 4$ m/s and $V_f = V_{f_1} = 4$ m/s.

Let,

Peripheral velocity of wheel at inlet $= v$

From the inlet triangle, we find that,



$$\tan 20° = \frac{V_f}{v} \Rightarrow 0.3640 = \frac{4}{v}$$

$$\therefore\ v = 4/0.3640 = 11 \text{ m/s} \quad \underline{\text{Ans.}}$$

Let, ϕ = vane angle at outlet,

As it is Francis turbine, therefore its discharge will be radial.

Moreover as the outer diameter of the

turbine is 2/3 rd of the inlet diameter. Therefore, the peripheral velocity of wheel at outlet,

$$v_1 = \frac{2}{3}u = \frac{2}{3} \times 11 = 7.33 \text{ m/s}$$

Now from the outlet triangle, we find that,

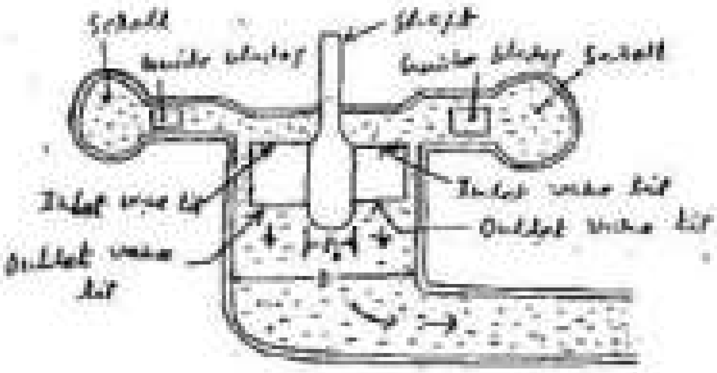
$$\tan\alpha = \frac{V_{f1}}{v_1} = \frac{4}{7.33} = 0.5457$$

$$\text{or, } \alpha = 28.6^\circ \text{ Ans.}$$

(ii) Kaplan turbine :

The Kaplan turbine is an axial flow reaction turbine, in which the flow of water is parallel to the shaft. A Kaplan turbine is used where a large quantity of water is available at low head.

The runner of a Kaplan turbine resembles with the propeller of a ship. That is why, a Kaplan turbine is also called propeller turbine. The water from the scroll flows over the guide blades and then over the vanes. The water while gliding over the vanes, exerts some force on the shaft of the turbine, which causes the shaft to revolve.



KAPLAN TURBINE

The runner of a modern Kaplan turbine has the following two major differences with that of Francis' turbine :

(i) In Francis' turbine runner, the water enters radially, whereas in a Kaplan turbine runner the water strikes the blades axially.

(ii) In a Francis' turbine runner, the number of blades is generally between 16 and 24 whereas in a Kaplan turbine runner the number of blade is generally between 3 to 8. This reduces the frictional resistance to water.

... of a Kaplan turbine ...

The blades of a Kaplan turbine which can be adjusted to [illegible] the passage area the two blades.

The runner of a Kaplan turbine is known as boss, which is nothing but an extension of the shaft (at its lower end) as shown in figure.

Let, D = Diameter of turbine,

D_b = Diameter of the boss, and

V_f = Velocity of flow at inlet.

∴ Discharge through the turbine,

$$Q = V_f \times \frac{\pi}{4}(D^2 - D_b^2)$$

All the relations of Kaplan turbine are the same as that of inward or outward flow reaction turbines. All relations for finding out the data hold good for a Kaplan turbine also.

Q (7) A Kaplan turbine operating under a net head of 20 metres develops 20000 kW with an overall efficiency of 86 percent. The speed ratio is 2.0 and flow ratio is 0.6. The hub diameter of the wheel is 0.35 times the outside diameter of the wheel. Find the diameter and speed of the turbine.

Solution: Given: $H = 20$ m, $P = 20000$ kW, $\eta_o = 86\% = 0.86$, $V_f/\sqrt{2gH} = 0.6$

$u/\sqrt{2gH} = 2.0$ or $u = 2 \times \sqrt{2 \times 9.81 \times 20} = 39.6$ m/s, and $D_b = 0.35\,D$.

or, $V_f = 0.6 \times \sqrt{2 \times 9.81 \times 20} = 11.9$ m/s

Let, D = Diameter of the turbine, and

Q = Discharge through the turbine.

We know that overall efficiency of the turbine (η_o)

$$0.86 = \frac{P}{wQH} = \frac{20000}{9.81 \times Q \times 20} = \frac{101.9}{Q}$$

or, $Q = 101.9/0.86 = 118.5$ m³/s

and discharge through the turbine (Q)

$$118.5 = V_f \times \frac{\pi}{4}(D^2 - D_b^2) = 11.9 \times \frac{\pi}{4}[D^2 - 0.35^2 D^2] = 8.2\,D^2$$

∴ $D^2 = 118.5/8.2 = 14.45$

or, $D = 3.8$ m. Ans.

Let, N = speed of the turbine in r.p.m.

At inlet, we know that peripheral velocity (u),

$$27.6 = \frac{\pi D N}{60} = \frac{\pi \times 3.9 \times N}{60} = 0.2\,N$$

$$\therefore N = 27.6/0.2 = 138 \text{ r.p.m.} \quad \underline{\text{Ans.}}$$

(3) A propeller turbine runner has an outer diameter of 4.5 meters and an inner diameter of 2.5 m and develops 21 000 kW when running at 140 r.p.m. under a head of 20 m. The hydraulic efficiency is 94% and overall efficiency is 88%. Find discharge through the turbine, and the guide blade angle at inlet.

Solution: Given: $D = 4.5$ m, $D_b = 2.5$ m, $P = 21\,000$ kW, $N = 140$ r.p.m, $H = 20$ m, $\eta_h = 94\% = 0.94$ and $\eta_o = 88\% = 0.88$.

Let, Q = Discharge through the turbine

We know that overall efficiency of the turbine (η_o),

$$0.88 = \frac{P}{wQH} = \frac{21\,000}{9.81 \times Q \times 20} = \frac{107}{Q}$$

$$\therefore Q = 107/0.88 = 121.6 \text{ m}^3/\text{s} \quad \underline{\text{Ans.}}$$

Guide blade angle at inlet.

Let, α = Guide blade angle at inlet,
V_w = Velocity of whirl at inlet, and
V_f = Velocity of flow at inlet.

We know that peripheral velocity at inlet,

$$u = \frac{\pi D N}{60} = \frac{\pi \times 4.5 \times 140}{60} = 33 \text{ m/s}$$

and hydraulic efficiency (η_h),

$$0.94 = \frac{V_w \cdot u}{gH} = \frac{V_w \times 33}{9.81 \times 20} = 0.168\,V_w$$

$$\therefore V_w = 0.94/0.168 = 5.6 \text{ m/s}$$

and discharge of the propeller (Q),

We know that discharge

$$121.6 = V_f \times \frac{\pi}{4}(D^2 - D_b^2) = V_f \times \frac{\pi}{4}\left[(4.5)^2 - (2.5)^2\right]$$

$$= 11\,V_f$$

$$\therefore V_f = 121.6/11 = 11.1 \text{ m/s}$$

From the inlet triangle velocities, we know that,

$$\tan\alpha = \frac{V_f}{V_w} = \frac{11.1}{5.6} = 1.9821$$

$$\therefore \alpha = 63.2^\circ \quad \underline{\text{Ans.}}$$

(i) Centrifugal Pump :

A pump, in general, may be defined as a machine, when driven from some external source, lifts water or some other liquid from a lower level to higher level. Or in other words, a pump may also be defined as a machine, which converts mechanical energy into pressure energy. The pump which raises water or a liquid from a lower level to higher level by the action of centrifugal force, is known as a centrifugal pump.

The action of a centrifugal pump is that of a reversed reaction turbine, the water at high pressure, is allowed to enter the casing which gives mechanical energy at its shaft, whereas in a pump, the mechanical energy is fed into the shaft and water enters the impeller (attached to the rotating shaft) which increases the pressure energy of the outgoing fluid. The water enters the impeller radially and leaves the vanes axially.

(ii) Types of casing for the impeller of a centrifugal pump :

(A centrifugal pump consists of an impeller, to which curved vanes are fitted. The impeller is enclosed in a water tight casing similar to that of a turbine, having a delivery pipe in one of its sides)

The casing for a centrifugal pump is so designed that the kinetic energy of the water is converted into pressure energy before the water leaves the casing. This considerably increases the efficiency of the pump. Following are the three types of the casings or chambers of centrifugal pumps.

(1) Volute casing (spiral casing),
(2) Vortex casing, and
(3) Volute casing with guide blades.

(i) Volute casing. (ii) Vortex casing (iii) Volute casing with guide blades.

Volute casing (spiral casing): In a volute casing, the impeller is surrounded by a spiral casing as shown in fig (a). Such a casing provides a gradual increase in the area of flow, which decreases the velocity of water, with a corresponding increase in pressure.

A considerable loss taking place due to the formation of eddies in this type of casing.

(b) **Vortex casing:** It is an improved type of volute casing, in which the spiral casing is combined with a circular chamber as shown in fig (b). In a vortex casing, the eddies are reduced to a considerable extent and an increased efficiency is obtained.

(c) **Volute casing with guide blades:** In this type of casing, there are guide blades surrounding the impeller as shown in figure (c). These guide blades are arranged at such an angle that the water enters without shocks and enters a passage of increasing area, through which the water passes and reaches the delivery pipe.

The ring of guide blades is called diffuser and is very efficient.

(3) **Piping system of a centrifugal pump:** The successful working of a centrifugal pump depends upon the correct selection and layout of its piping system. The extreme care should always be taken in selecting the sizes of the pipes and their arrangement. In general, a centrifugal pump has (i) suction pipe, and (ii) delivery pipe.

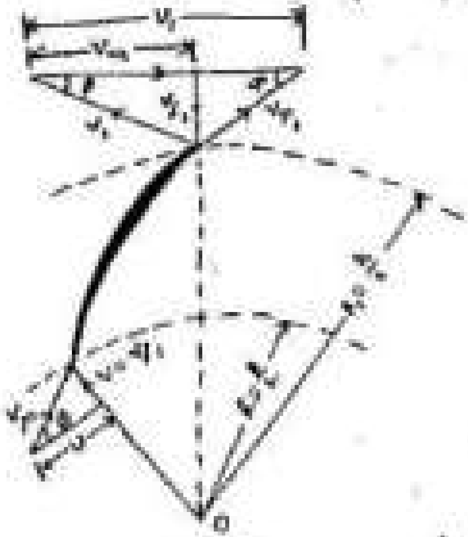
(a) **Suction pipe:** The suction pipe of a centrifugal pump plays an important role in the successful and smooth working of the pump. A poorly designed suction pipe causes insufficient net positive suction head (NPSH), vibration, noise, water hammer, excessive wear etc. While laying the pipe, a great care should be taken to make it air tight. A strainer foot-valve is connected at the bottom of the suction pipe to avoid the entry of foreign matter. Since the pressure at the inlet of the pump is suction (negative), it is therefore essential that the losses in the suction pipe should be as small as possible. For this purpose bends in the suction pipe are avoided, and its diameter is taken more than delivery pipe.

Sometimes to reduce the axial thrust the suction pipe is branched into two parts and the liquid is allowed to enter the impeller from both sides. Such a pump is called double suction pump.

(5) Delivery Pipe: A check valve is provided in the delivery pipe near the pump, in order to protect the pump from burst and also to regulate the discharge from the pump. The size and length of the delivery pipe depends upon the requirement.

(2) Workdone by a centrifugal pump:

The workdone or the power required to drive the pump, may be found out by drawing the inlet and outlet



Inlet and outlet triangle velocities

Consider a centrifugal pump lifting water from a lower level to a higher level. Now draw the inlet and outlet triangle of velocities as shown in figure.

V = Absolute velocity of the entering water,

D = Diameter of the impeller at inlet (inner dia),

v = Tangential velocity of impeller at inlet (also known as peripheral velocity at inlet),

V_r = Relative velocity of water to the wheel at inlet,

V_f = Velocity of flow at inlet,

V_1, D_1, v_1, V_{r1}, V_{f1} = corresponding values at the outlet.

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

θ = Vane angle at inlet,

β = Angle at which the water leaving the impeller,

ϕ = Vane angle at outlet.

Since the water enters the impeller radially, therefore the velocity of whirl at inlet ($V_w = 0$).

$\therefore$ Workdone per kN of water $= \frac{V_{w1} v_1}{g}$ kN-m

Where V_{w1} and v_1 are in m/s.

... and the basis of transfer of mechanical energy the pumps can be broadly classified as follows:

(1) Rotodynamic Pumps :

(i) Radial flow pumps.
(ii) Axial flow pumps.
(iii) Mixed flow pumps.

(2) Positive displacement pumps :

In rotodynamic pumps, increase in energy level is due to a combination of centrifugal energy, pressure and kinetic energy.

- The energy transfer in a radial flow pump occurs mainly when the flow is in its radial path.
- In an axial flow pump, the energy transfer occurs when the flow is in its axial direction.
- The energy transfer in a mixed flow pump takes place when the flow comprises radial as well as axial components.

The radial flow type pumps are commonly called centrifugal pumps.

(2) Classification of centrifugal pumps :

On the basis of characteristic features, the centrifugal pumps are classified as follows :

(1) Types of casing :
(i) Volute pump (ii) Turbine pump or diffusion pump.

(2) Working head :
(i) Low lift centrifugal pumps --- they work against heads upto 15 m.
(ii) Medium lift centrifugal pumps --- used to build up heads of high as 40 m.
(iii) High lift centrifugal pumps --- employed to deliver liquids at heads above 40 m.

(3) Liquid handled :
(i) Closed impeller pump. (ii) Semi open impeller pump (non-clog pump).
(iii) Open impeller pump.

(4) Number of impellers per shaft :
Single stage centrifugal pump --- has one impeller, usually a low lift pump.

(iv) Multi-stage centrifugal pump ---- has two or more impellers and pressure is built in steps. used usually for high working heads and the number of stages depends on the head required.

(5) Number of entrances to the impeller :

(i) Single entry or single suction pump ---- water is admitted on one side of the impeller.

(ii) Double entry or double suction pump ---- water is admitted from both sides of the impeller, axial thrust is neutralised.
---- employed for pumping large quantities of fluid.

(6) Relative direction of flow through impeller :

(i) Radial flow pump ----- normally radial flow impellers are used in all centrifugal pumps.

(ii) Axial flow pump ----- designed to deliver large quantities of water at comparatively low heads, ideally suited for irrigation purposes.

(iii) Mixed flow pump ----- mostly employed for irrigation purposes.

⑦ Component parts of a centrifugal pump :

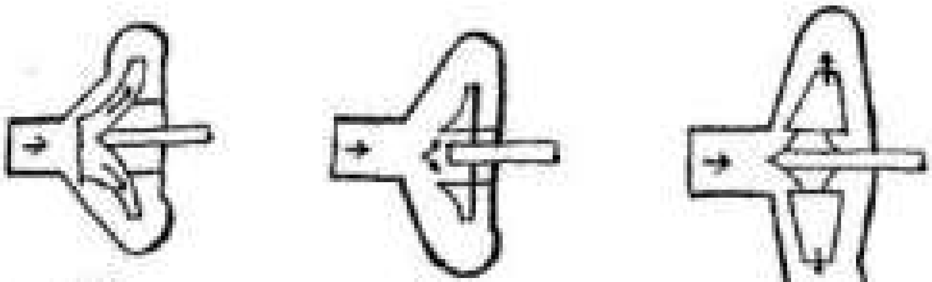
It consists of the following main components :

(1) Impeller (2) casing (3) suction pipe (4) delivery pipe.

(1) Impeller : An impeller is a wheel (or rotor) with a series of backward curved vanes (or blades). It is mounted on a shaft which is usually coupled to an electric motor.

They are following three types :

(i) Shrouded or closed impeller : In this type impeller vanes are provided with metal cover plates or shrouds on both the sides. It provides better guidance for the liquid and has a high efficiency. It is employed when the liquid to be pumped is pure and relatively free from debris.



(a) (b) (c)

(ii) **Semi open impeller:** A semi open impeller is one in which vanes have only the base plate and no crown plate. This impeller can be used even if the liquids contain some debris.

(iii) **Open impeller:** Such an impeller in fig, the vanes have neither the crown plate nor the base plate i.e., the vanes are open on both sides. Such impellers are employed for pumping liquids which contain suspended solid matter (eg, sewage, paper pulp, water containing sand or grit).

⑩ **Derivation of workdone by the centrifugal pump (impeller):**

ω = Angular velocity = $\frac{2\pi N}{60}$ rad/s,

u_1 = Tangential velocity of the impeller at inlet.

$= \frac{\pi D_1 N}{60} = \frac{2\pi R_1 N}{60} = \omega R_1$,

u_2 = Tangential velocity of the impeller at outlet,

$= \frac{\pi D_2 N}{60} = \frac{2\pi R_2 N}{60} = \omega R_2$,

While passing through the impeller, the velocity of whirl changes and there is a change of moment of momentum.

Torque on the impeller = Rate of change of moment of momentum

Moment of momentum at inlet = 0

moment of momentum at outlet = $\frac{W}{g}(V_{w_2} R_2)$

$\therefore$ Torque = $\frac{W}{g}(V_{w_2} R_2)$

Workdone per second = Torque × Angular velocity (1)

$= \frac{W}{g}(V_{w_2} R_2) \times \omega = \frac{W}{g}(V_{w_2} u_2)$ $[\because u_2 = \omega R_2]$

Workdone per unit weight per second of liquid = $\frac{V_{w_2} u_2}{g}$

From this equation have been developed assuming flow at inlet to be radial (i.e., $V_{w_1} = 0$). If the flow is not radial, the expression for workdone may be written as:

Workdone per second = $\frac{W}{g}(V_{w_2} u_2 - V_{w_1} u_1)$

∴ workdone per second per unit weight of liquid

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

It is known as the Euler momentum equation for centrifugal pump.

The term $\frac{1}{g}(V_{w_2} u_2 - V_{w_1} u_1)$ is referred to as Euler head H_e (Theoretical head)

(ii) Total head at outlet of the pump − Total head at inlet of the pump:

$$H_{man} = \left(\frac{p_2}{w} + \frac{V_{d}^2}{2g} + Z_2\right) - \left(\frac{p_1}{w} + \frac{V_1^2}{2g} + Z_1\right)$$

Where, $\frac{p_2}{w}$ = Pressure head at outlet of pump = h_d,

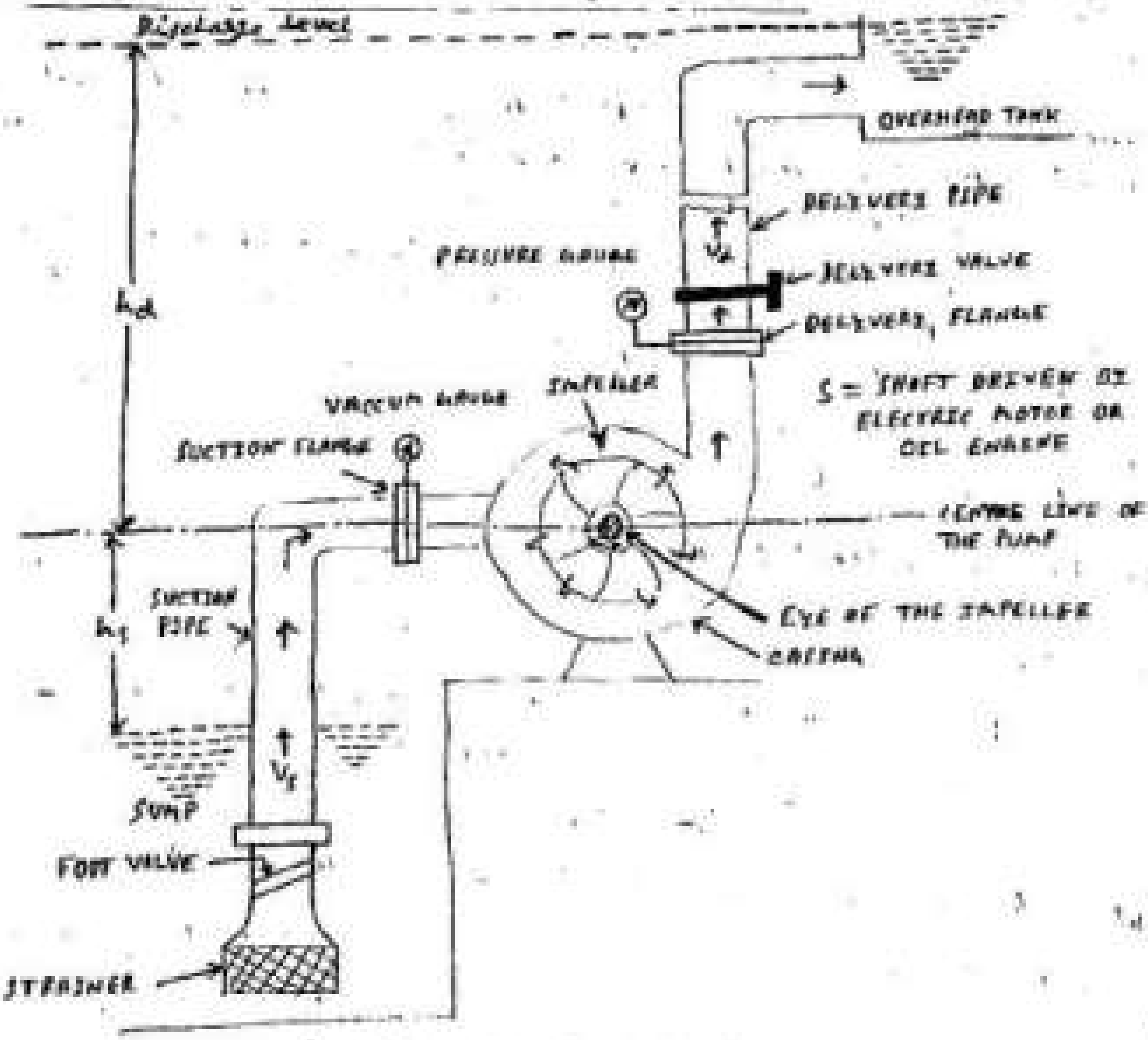
$\frac{V_2^2}{2g}$ = Velocity head at outlet of the pump,

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

Z_2 = Vertical height of the pump outlet from the datum line, and

(iii) Total gross or effective head: It is equal to the static head plus all the head losses occurring in flow, before, through and after the impeller.

⑧ Component parts of a centrifugal pump:



VOLUTE TYPE CENTRIFUGAL PUMP COMPONENT PARTS

(2) Efficiencies of a centrifugal pump :-

The various efficiencies of a centrifugal pump are :
(i) manometric efficiency (η_{mano}), (ii) volumetric efficiency (η_v),
(iii) mechanical efficiency (η_m) and (iv) overall efficiency (η_o)

(i) Manometric efficiency (η_{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 efficiency. Thus,

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

$$\text{or, } \eta_{mano} = \frac{H_{mano}}{\left(\frac{V_{w_2} u_2}{g}\right)} = \frac{g H_{mano}}{V_{w_2} u_2}$$

(ii) Volumetric efficiency (η_v) :

The ratio of quantity of liquid discharged per second from the pump to quantity passing per second through the impeller is known as volumetric efficiency. Thus,

$$\eta_v = \frac{\text{Liquid discharged per second from the pump}}{\text{Quantity of liquid passing per second through the imp}}$$

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

where, Q = Actual liquid discharged at the pump outlet per second, and

q = Leakage of liquid per second from the impeller and casing) (through the clearances between the impeller and casing).

(iii) Mechanical efficiency (η_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. Thus,

$$\eta_m = \frac{\text{Power delivered by the impeller to the liquid}}{\text{Power input to the pump shaft (P)}}$$

$$\text{or, } \eta_m = \frac{w(Q+q)(V_{w_2} u_2/g)}{P}$$

$$= \frac{P - P_{mech.\ losses}}{P}$$

(iv) Overall efficiency (η_o) :

The ratio of power output of the pump to the power input to the pump is known as overall efficiency. Thus,

$$\eta_o = \frac{\text{Power output of the pump}}{\text{Power input to the pump (shaft}} = \frac{w Q H_{mano}}{P}$$

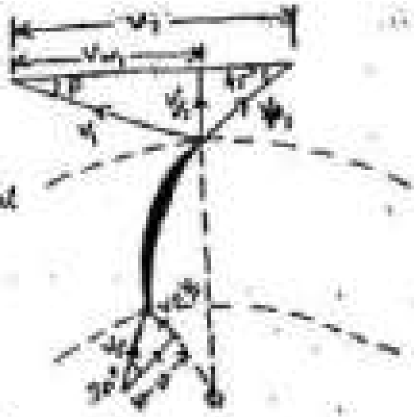
$$\text{Also, } \eta_o = \eta_{mano} \times \eta_v \times \eta_m = \frac{H_{mano}}{(V_{w_2} u_2/g)} \times \frac{Q}{(Q+q)} \times \frac{w(Q+q)(V_{w_2} u_2/g)}{P}$$

$$= \frac{w Q H_{mano}}{P}$$

Q. A centrifugal pump has external and internal impeller diameters as 600 mm and 300 mm respectively. The vane angles at inlet and outlet are 30° and 45° respectively. If the water enters the impeller at 2.5 m/s, find (a) speed of the impeller in r.p.m, (b) work done per kN of water.

Solution: Given: $D_1 = 600$ mm $= 0.6$ m, $D = 300$ mm $= 0.3$ m, $\theta = 30°$, $\phi = 45°$ and $V = 2.5$ m/s.

(a) Speed of the impeller in r.p.m



Let, N = ...

From the inlet triangle of velocities, we find that the tangential velocity of impeller at inlet,

$$v = \frac{V}{\tan 30°}$$

$$= \frac{2.5}{0.5774} = 4.33 \text{ m/s}$$

We know that the velocity of impeller at inlet (v),

$$4.33 = \frac{\pi D N}{60} = \frac{\pi \times 0.3 \times N}{60} = 0.0157\,N$$

$$\therefore N = 275.8 \text{ r.p.m} \quad \text{Ans.}$$

(b) Work done per kN of water,

From the outlet triangle of velocities we find that the tangential velocity at outlet,

$$v_1 = v \times \frac{D_1}{D} = 4.33 \times \frac{0.6}{0.3} = 8.66 \text{ m/s}$$

and velocity of whirl at outlet,

$$V_{w_1} = v_1 - \frac{V_{f_1}}{\tan 45°} = 8.66 - \frac{2.5}{1} = 6.16 \text{ m/s}$$

$$\text{--- } (\because V_{f_1} = V_f = V)$$

Since the tangential velocity of impeller at outlet v_1 (8.66) is more than velocity of whirl at outlet V_{w_1} (6.16), therefore shape of the outlet triangle will be as shown in figure.

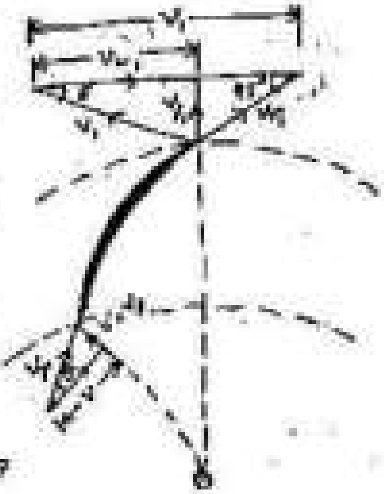
We know that workdone per kN of water,

$$W = \frac{V_{w_1} \cdot v_1}{g} = \frac{6.16 \times 8.66}{9.81}$$

$$= 5.44 \text{ kN-m} = 5.44 \text{ kJ} \quad \text{Ans.}$$

(Q) Calculate the vane angle at the inlet of a centrifugal pump impeller having 200 mm diameter at inlet and 400 mm diameter at outlet. The impeller vanes are set back at angle 45° to the outer rim and the entry of the pump is radial. The pump runs at 1000 r.p.m. and the velocity of flow through the impeller is constant at 3 m/s. Also calculate the workdone per kN of water and the velocity as well as direction of the water at outlet.

Solution: Given: $D = 200$ mm $= 0.2$ m, $D_1 = 400$ mm $= 0.4$ m,
$\phi = 45°$, $N = 1000$ r.p.m. and $V_f = V_{f_1} = 3$ m/s



Let, θ = Vane angle at inlet,

We know that the tangential velocity of impeller at inlet,

$$v = \frac{\pi D N}{60} = \frac{\pi \times 0.2 \times 1000}{60}$$
$$= 10.5 \text{ m/s}$$

From the inlet triangle of velocities, we find that,

$$\tan\theta = \frac{V_f}{v} = \frac{3}{10.5} = 0.2857$$
$$\therefore \theta = 15.9° \quad \underline{\text{Ans.}}$$

Velocity of water at outlet:

Velocity of impeller at outlet, we know that tangential velocity

$$v_1 = \frac{\pi D_1 N}{60} = \frac{\pi \times 0.4 \times 1000}{60} = 20.9 \text{ m/s}$$

From the outlet triangle of velocities, we find that the velocity of whirl at outlet,

$$V_{w_1} = v_1 - \frac{V_{f_1}}{\tan 45°} = 20.9 - \frac{3}{1} = 17.9 \text{ m/s}$$

and, $$V_1 = \sqrt{V_{f_1}^2 + V_{w_1}^2} = \sqrt{(3)^2 + (17.9)^2} = 18.1 \text{ m/s} \quad \underline{\text{Ans.}}$$

Direction of water at outlet,

Let, β = Angle at which the water leaves the impeller at outlet,

From the outlet triangle of velocities, we also find that,

$$\tan\beta = \frac{V_{f_1}}{V_{w_1}} = \frac{3}{17.9} = 0.1676$$
$$\text{or, } \beta = 9.5° \quad \underline{\text{Ans.}}$$

Workdone per kN of water,

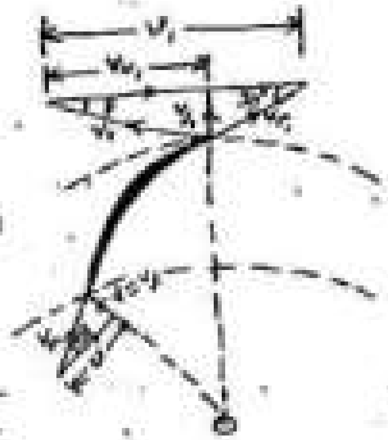
$$W = \frac{V_{w_1} \cdot v_1}{g} = \frac{17.9 \times 20.9}{9.81} = 38.14 \text{ kN-m}$$
$$= 38.14 \text{ kJ} \quad \underline{\text{Ans.}}$$

(1) A centrifugal pump delivers water against a head of 14.5 m while running at 1000 r.p.m. The vanes are curved back at an angle of 30° with the periphery. The impeller diameter is 300 mm and outlet width 50 mm. If manometric efficiency of the pump is 85%, find the discharge of the pump.

Solution: Given: $H_m = 14.5$ m, $N = 1000$ r.p.m., $\theta = 30$°, $D_2 = 300$ mm $= 0.3$ m, $b_2 = 50$ mm $= 0.05$ m and $\eta_{man} = 85\% = 0.85$

Let, V_{w_2} = velocity of whirl at outlet.

We know that tangential velocity of the impeller at outlet,



$$u_2 = \frac{\pi D_2 N}{60} = \frac{\pi \times 0.3 \times 1000}{60} = 15.7 \text{ m/s}$$

and manometric efficiency (η_{man}),

$$0.85 = \frac{H_m}{\frac{V_{w_2} \cdot u_2}{g}} = \frac{14.5}{\frac{V_{w_2} \times 15.7}{9.81}}$$

or, $$0.85 = \frac{9.06}{V_{w_2}}$$

$$\therefore V_{w_2} = 9.06/0.85 = 10.7 \text{ m/s}$$

From the outlet triangle of velocities, we find that,

$$\tan 30° = \frac{V_{f_2}}{u_2 - V_{w_2}} = \frac{V_{f_2}}{15.7 - 10.7}$$

or, $$V_{f_2} = 5 \times 0.5774 = 2.89 \text{ m/s}$$

$\therefore$ Discharge through the pump,

$$Q = \pi D_2 b_2 V_{f_2}$$

$$= \pi \times 0.3 \times 0.05 \times 2.89 = 0.136 \text{ m}^3/\text{s} \quad \text{Ans.}$$

(2) A centrifugal pump delivers 30 litres of water per second to a height of 18 metres through a pipe 90 metres long and of 100 mm diameter. If the overall efficiency of the pump is 75%, find the power required to drive the pump. Take f = 0.012.

Solution: Given: $Q = 30$ lit/s $= 0.03$ m³/s, $H = 18$ m, $l = 90$ m,

$d = 100$ mm $= 0.1$ m, $\eta_o = 75\% = 0.75$

and $f = 0.012$.

We know that cross sectional area of pipe,

$$a = \frac{\pi}{4} \times (d)^2 = \frac{\pi}{4} \times (0.1)^2 = 7.854 \times 10^{-3} \text{ m}^2$$

and velocity of water, $$v = \frac{Q}{a} = \frac{0.03}{7.854 \times 10^{-3}} = 3.82 \text{ m/s}$$

We know also that manometric head, 71

$H_m = H +$ Loss of head in pipe + Loss of head at outlet

$$= 18 + \frac{4 f L v^2}{2 g d} + \frac{v^2}{2g}$$

$$= 18 + \frac{4\times 0.012\times 90\times (3.82)^2}{2\times 9.81\times 0.1} + \frac{(3.82)^2}{2\times 9.81} \text{ m}$$

$$= 18 + 32.1 + 0.74 = 50.84 \text{ m}$$

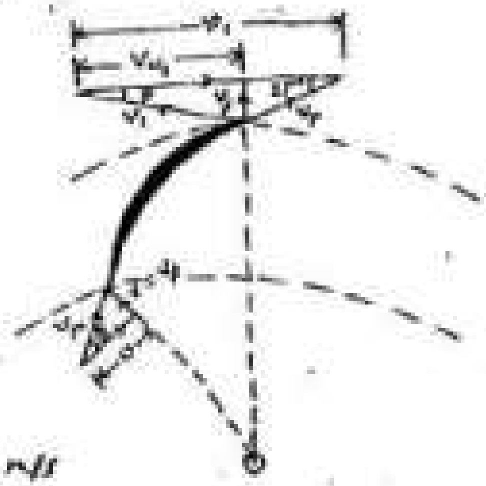
and Power required to drive the pump,

$$P = \frac{w q H_m}{\eta_o} = \frac{9.81\times 0.03\times 50.84}{0.75} = 19.9 \text{ kW} \quad \textbf{Ans:}$$

Q(8) A centrifugal pump of 1.5 metre diameter runs at 210 r.p.m and pumps 180 litres of water per second. The angle which the vane makes, at exit, with the tangent to the impeller is 25°. Assuming radial entry and velocity of flow throughout as 2.5 m/s, determine the power required to drive the pump. If manometric efficiency of the pump is 65 per cent, find the average lift of the pump.

Solution: Given: $D_1 = 1.5$ m, $N = 210$ r.p.m, $q = 180$ lit/sec $= 0.18$ m³/s, and $\eta_{mano} = 65\% = 0.65$, $\phi = 25°$, $V_f = V_{f1} = 2.5$ m/s,

Power required to drive the pump:



We know that, at outlet,

$$v_1 = \frac{\pi D_1 N}{60} = \frac{\pi\times 1.5\times 210}{60} = 16.5 \text{ m/s}$$

From the outlet velocity triangle,

$$V_{w1} = v_1 - \frac{V_f}{\tan 25°} \text{ m/s}$$

$$= 16.5 - \frac{2.5}{0.4663} = 11.1 \text{ m/s}$$

∴ Power required to drive the pump,

$$P = \frac{w q \cdot V_{w1} \cdot v_1}{g} = \frac{9.81\times 0.18\times 11.1\times 16.5}{9.81} \text{ kW}$$

$$= 33.0 \text{ kW} \quad \textbf{Ans:}$$

Let, H_m = Average lift of the pump (or manometric head)

We know that,

$$\eta_{mano} = \frac{H_m}{V_{w1} \cdot v_1 / g} = \frac{H_m}{\frac{11.1\times 16.5}{9.81}} = \frac{H_m}{18.7}$$

$$\therefore H_m = 0.65\times 18.7$$

$$= 12.2 \text{ m} \quad \textbf{Ans:}$$

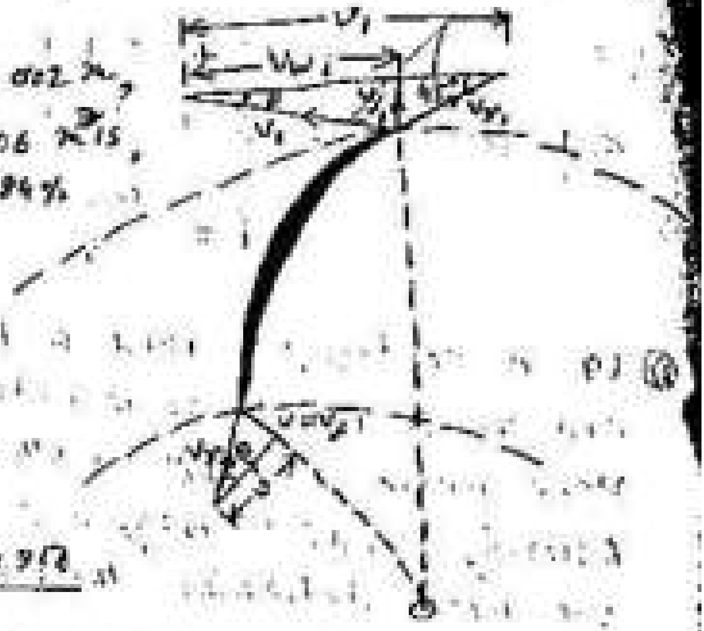
[illegible] impeller of a three stage centrifugal pump [illegible] diameter of 375 mm and width 20 mm. The pump is delivering 3600 litres of water per minute at 900 r.p.m. The vanes are curved back at 45° to the tangent at outlet. If the manometric efficiency is 84%, find the manometric head generated by the pump.

<u>Solution</u>; Given: No. of stages = 3,

$D_1 = 375\ mm = 0.375\ m$, $b_1 = 20\ mm = 0.02\ m$,

$Q = 3600\ lit/min = 60\ lit/sec = 0.06\ m^3/s$,

$N = 900\ r.p.m.$, $\phi = 45°$ and $\eta_{mano} = 84\% = 0.84$



Let H_m = manometric head of each stage

We know that tangential velocity of the impeller at outlet,

$$u_1 = \frac{\pi D_1 N}{60} = \frac{\pi \times 0.375 \times 900}{60} = 17.7\ m/s$$

and velocity of flow at outlet,

$$V_{f_1} = \frac{Q}{\pi D_1 b_1} = \frac{0.06}{\pi \times 0.375 \times 0.02} = 2.55\ m/s$$

From the outlet triangle of velocities,

$$V_{w_1} = u_1 - \frac{V_{f_1}}{\tan\phi} = 17.7 - \frac{2.55}{\tan 45°} = 15.15\ m/s$$

and

$$\eta_{mano} = 0.84 = \frac{H_m}{\frac{V_{w_1} \cdot u_1}{g}} = \frac{H_m}{\frac{15.15 \times 17.7}{9.81}} = \frac{H_m}{27.3}$$

$$\therefore H_m = 0.84 \times 27.3 = 22.9\ m$$

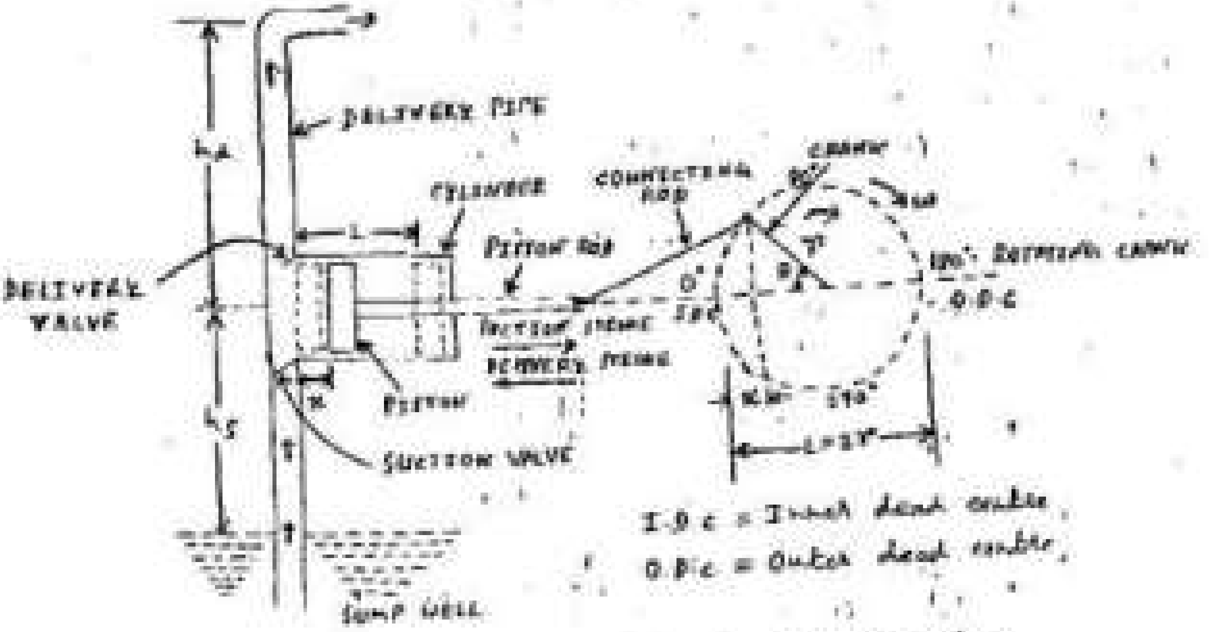
$\therefore$ Total manometric head due to three stage

$= 3 \times 22.9 = 68.7\ m$ <u>Ans</u>:

Main components and working of a Reciprocating Pump :-

The main parts of a Reciprocating pump are :

(1) Cylinder (2) Piston
(3) Suction valve (4) Delivery valve
(5) Suction Pipe (6) Delivery Pipe
(7) Crank and connecting rod mechanism operated by a power source e.g. steam engine, internal combustion engine or an electric motor.



Schematic view of single acting Reciprocating Pump

Working of a single acting Reciprocating pump :

As shown in figure, a single acting reciprocating pump has one suction pipe and one 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. The pump operates as follows :

Let us suppose that initially the crank is inner dead centre (I.D.C) and crank rotates in the clockwise 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 liquid is forced from the sump into the left side of the piston. When the crank is at the outer dead centre (O.D.C) the suction stroke is completed and the left side of cylinder is full of liquid.

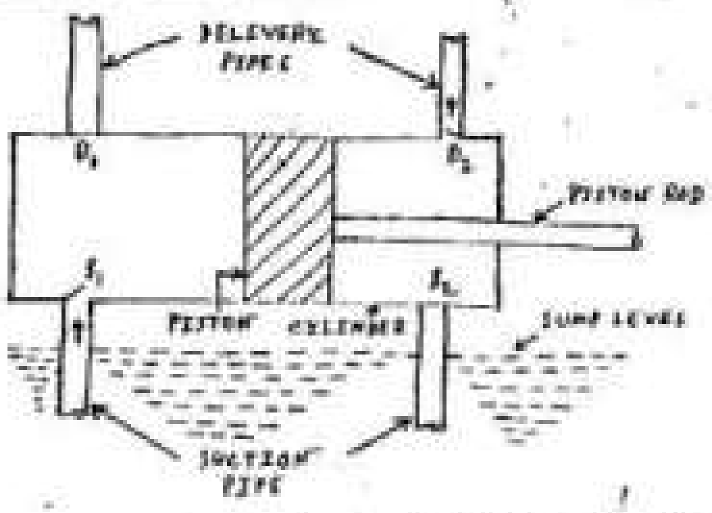
When the crank further turns from O.D.C to I.D.C, the piston moves inward to the left and high pressure is

up in the cylinder. The delivery valve opened and liquid is forced into the delivery pipe. The liquid is carried to the discharge tank through the delivery pipe.
At the end of delivery stroke the crank comes to the I.D.C and the piston is at the extreme left position.

(B) Working of a double acting reciprocating pump:

In this figure a double acting reciprocating pump, suction and delivery strokes occur simultaneously. When the crank rotates from I.D.C. in the clockwise direction, a vacuum is created on the left side of piston and the liquid is sucked in from the sump through valve S_1. At the same time, the liquid on the right side of the piston is pressed and a high pressure causes the delivery valve D_2 to open and the liquid is passed on to the discharge tank. This operation continues till the crank reaches O.D.C.

DOUBLE ACTING RECIPROCATING PUMP

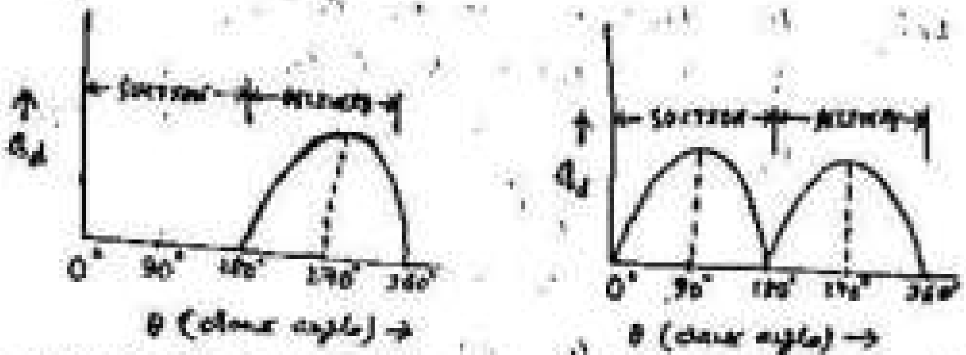


With further rotation of the crank the liquid is sucked in from the sump through the suction valve S_2 and is delivered to the discharge tank through the delivery valve D_1. When the crank reaches I.D.C, the piston is extreme left position. Thus the cycle is completed and as the crank further rotates, cycles are repeated.

Because of continuous delivery strokes, a double acting reciprocating pump gives more uniform discharge (as compared to single acting pump which pumps the liquid intermittently). To get a more uniform feed, invariably a multicylinder arrangement having two or more cylinders is employed.

This figures show the variations of discharge through delivery pipe (Q_d) with crank angle (θ) for

Single acting and double acting pumps, respectively.



Q_d V/S θ variation for single-acting pump — Q_d V/S θ variation for double acting pump

(ii) Discharge, workdone and power required to driving reciprocating pump :

(a) Single acting reciprocating pump :

Consider a single acting reciprocating pump shown in figure (above).

Let, D = Diameter of the cylinder, m

A = cross sectional area of the piston/cylinder $= \frac{\pi}{4} D^2$ m²

r = Radius of crank, m

N = Speed of the crank, r.p.m.

L = Length of the stroke $(= 2r)$, m

h_s = Height of the centre of the cylinder above the liquid surface, m and

h_d = Height to which the liquid is raised above the centre of the cylinder, m.

Volume of liquid sucked in during suction stroke $= A \times L$

$\therefore$ Discharge of the pump per second, $Q = A \times L \times \frac{N}{60}$

Weight of water delivered per second, $W = wQ = \frac{wLAN}{60}$

Work done per second = Weight of water lifted/sec × total height through which liquid is lifted

$= W(h_s + h_d) = \frac{wLAN}{60}(h_s + h_d)$

$\therefore$ Power required to drive the pump $= \frac{wLAN}{60 \times 1000}(h_s + h_d)$ kW.

(Where, w = weight density of liquid in N/m²)

(i) **Double acting Reciprocating Pump :** From Hydraulic Calculation

Let, D = Diameter of the Piston,

d = Diameter of the Piston Rod,

A_{PR} = Cross sectional area of the Piston Rod

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

Area on one side of the Piston, $A = \frac{\pi}{4} D^2$

Area on other side of the Piston where Piston Rod is connected to the Piston,

$$A' = A - A_{PR} = \frac{\pi}{4} D^2 - \frac{\pi}{4} d^2 = \frac{\pi}{4} (D^2 - d^2)$$

Volume of liquid delivered in one revolution of crank

$$= AL + A'L = (A + A')L$$

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

$\therefore$ Discharge of the Pump per second $= \left[\frac{\pi}{4} D^2 + \frac{\pi}{4} (D^2 - d^2)\right] L \times \frac{N}{60}$

If the diameter of the Piston Rod 'd' is very small as compared to the diameter of the Piston 'D' then it can be neglected and hence Q of the Pump per second will become, $Q = \left(\frac{\pi}{4} D^2 + \frac{\pi}{4} D^2\right) \times \frac{LN}{60} = 2 \times \frac{\pi}{4} D^2 \times \frac{LN}{60}$

$$= \frac{2ALN}{60}$$

Evidently the output of double acting pump is two times that of a single acting pump.

Work done per second = Weight of water delivered × total height through which liquid is lifted

$$= \left(w \times \frac{2ALN}{60}\right) \times (h_s + h_d) = \frac{2wALN}{60} (h_s + h_d)$$

Power required to drive the Pump, $P = \frac{2wALN}{60 \times 1000} (h_s + h_d)$ kW

(Here, w = weight density of liquid in N/m^3)

(ii) **Co-efficient of discharge and slip of Reciprocating Pump :**

(a) **Co-efficient of discharge :**

In a Reciprocating Pump, the actual discharge (Q_{act}) is always slightly different from the theoretical discharge (Q_{th}) due to following reasons :

(i) Leakage through the valves, glands and piston packing,

(ii) Improper operation of the valves (suction and discharge), and

(iii) Partial filling by the liquid.

The ratio between actual discharge and theoretical discharge is known as the co-efficient of discharge (C_d) of the pump. That is,

$$C_d = \frac{\text{Actual discharge}}{\text{Theoretical discharge}} = \frac{Q_{act}}{Q_{th}}$$

When the value of C_d is expressed in percentage, is known as 'volumetric efficiency' of the pump. Volumetric efficiency depends upon the dimensions of the pump and its value ranges from 85-98%.

(b) Slip: The difference between the theoretical and actual discharge is called slip of the pump. That is

$$\text{slip} = Q_{th} - Q_{act}$$

But the slip is mostly expressed in percentage which is given by,

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

$$= (1 - C_d) \times 100$$

The percentage slip for the pumps maintained in good condition is of the order of 2% or even less.

(c) Negative slip: In most of the reciprocating pumps
Q_{act} is less than Q_{th}, in such a case the value of C_d
is less than unity and slip of the pump is 'positive'.
However in some cases Q_{act} may be more than Q_{th}, in
such a case C_d is more than unity and the slip will be
'negative'. The suction will be negative when there is a
direct connection between the suction and delivery sides
before the end of a suction stroke. This happens if the
momentum of liquid in the suction pipe is large enough
to open the delivery valve before the beginning of delivery
stroke. The negative slip is possible in case of long suction
pipe and a short delivery pipe, especially when these
are operating at high speeds.

Q. (b) A single acting reciprocating pump, running at 50 r.p.m., delivers 0.00736 m³/s of water. The diameter of the piston is 200 mm and stroke length 300 mm. The suction and delivery heads are 3.5 m and 11.5 m respectively. Determine:

(i) Theoritical discharge.
(ii) Co-efficient of discharge.
(iii) Percentage slip of the pump and
(iv) Power required to run the pump.

<u>Solution:</u> Speed of the pump, $N = 50$ r.p.m.

Actual discharge, $Q_{act} = 0.00736 \; m^3/s$

Diameter of the piston, $D = 200 \; mm = 0.2 \; m$.

Area, $A = \frac{\pi}{4} \times (0.2)^2 = 0.0314 \; m^2$

Stroke length, $L = 300 \; mm = 0.3 \; m$.

Suction head, $h_s = 3.5 \; m$

Delivery head, $h_d = 11.5 \; m$.

(i) Theoritical discharge, $Q_{th} = \frac{ALN}{60} = \frac{0.0314 \times 0.3 \times 50}{60}$

$= 0.00785 \; m^3/s$ Ans:

(ii) Co-efficient of discharge, C_d:

$$C_d = \frac{Q_{act}}{Q_{th}} = \frac{0.00736}{0.00785} = 0.937 \quad \text{Ans:}$$

(iii) Percentage slip of the pump:

$$\% \text{ slip} = \frac{Q_{th} - Q_{act}}{Q_{th}} \times 100 = \frac{0.00785 - 0.00736}{0.00785} \times 100$$

$$= 6.24 \% \quad \text{Ans:}$$

(iv) Power required to run the pump, P:

$$P = \frac{w A L N}{60 \times 1000} (h_s + h_d) \; kW$$

$$= \frac{9810 \times 0.0314 \times 0.3 \times 50}{60 \times 1000} (3.5 + 11.5)$$

$$= 1.155 \; kW \quad \text{Ans:}$$

(i) A single acting reciprocating pump operating at 120 r.p.m. has a piston diameter of 200 mm and stroke of 300 mm. The suction and delivery heads are 4 m and 20 m, respectively. If the efficiency of both suction and delivery strokes is 75%, determine the power required by the pump.

Solution: Given: $N = 120$ r.p.m., $D = 200$ mm $= 0.2$ m, $L = 300$ mm $= 0.3$ m, $h_s = 4$ m, $h_d = 20$ m,

$\eta_{(\text{suction and delivery strokes, each})} = 75\%$

Power required by the pump,

$$q = \frac{ALN}{60} = \frac{\frac{\pi}{4}\times(0.2)^2\times 0.3\times 120}{60} = 0.0188 \text{ m}^3/\text{s}$$

Power required to drive the pump,

$$P = \frac{wq(h_s+h_d)}{\eta} = \frac{9810\times 0.0188\times(4+20)}{0.75}$$

$$= 5901.7 \text{ W or } 5.9 \text{ kW} \quad \textbf{Ans.}$$

(2) A "three throw" pump has cylinders of 250 mm diameter and stroke of 500 mm each. The pump is required to deliver 0.1 m³/s at a head of 100 m. Friction losses are estimated to be 1 m in suction pipe and 19 m in delivery pipe. Velocity of water in delivery pipe is 1 m/s, overall efficiency is 85% and the slip is 3%. Determine:

(i) Speed of the pump, and

(ii) Power required to drive the pump.

Solution: Diameter of each cylinder, $D = 250$ mm $= 0.25$ m

Stroke length of each cylinder, $L = 500$ mm $= 0.5$ m

Actual discharge, $Q_{act} = 0.1$ m³/s,

Static head, $(h_s + h_d) = 100$ m

Friction loss in suction pipe, $h_{fs} = 1$ m,

Friction loss in delivery pipe, $h_{fd} = 19$ m,

Velocity of water in delivery pipe, $V_d = 1$ m/s

Overall efficiency of the pump, $\eta_o = 85\%$

Percentage slip $= 3\%$

(i) Speed of the pump, N:

A three throw pump uses three equal cylinders with their cranks connected to cranks set 120° apart driven by a common shaft.

$$Q_{th} = 2 \times \frac{A L N}{60} = 2 \times \frac{\pi}{4} \times (0.25)^2 \times [illegible] \times \frac{N}{60}$$

$$= 0.001227\, N$$

Actual discharge, $Q_{act} = C_d \times Q_{th} = 0.97 \times 0.001227\, N$

$$= 0.00119\, N$$

But $Q_{act} = 0.1\ m^3/s$

$$\therefore\ 0.1 = 0.00119\, N \quad \text{or,} \quad N = \frac{0.1}{0.00119}$$

$$= 84\ \text{r.p.m. (Ans)}$$

(ii) Power required to run the pump, P :

Total head against which pump has to work,

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

$$= 100 + (1 + 19) + \frac{(1.0)^2}{2 \times 9.81} = 120.05\ m$$

$$\text{Water power} = \frac{w\, Q_{act}\, H}{1000}\ kW = \frac{9810 \times 0.1 \times 120.05}{1000}$$

$$= 117.77\ kW$$

Power required to drive the pump,

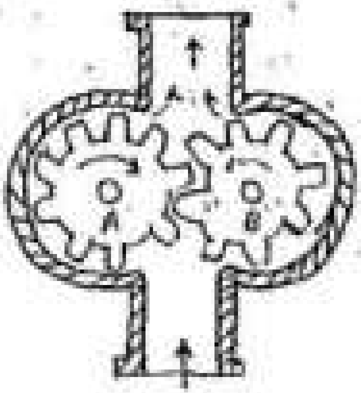
$$P = \frac{\text{Water power}}{\text{Overall efficiency}} = \frac{117.77}{0.85}$$

$$= 138.55\ kW\ \text{Ans.}$$

HYDRAULIC CONTROL SYSTEM

(1) External Gear Pump:

An external gear pump in its simplest form consists of two identical intermeshing spur wheels A and B working with a fine clearance inside the casing. The wheels are so designed that they form fluid tight joint at the point of contact as shown in figure. One of the wheels is keyed to the driving shaft, and the other revolves as a driven wheel.

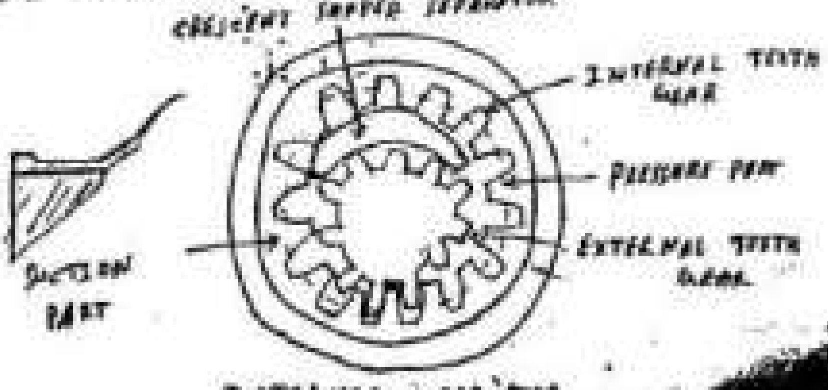


External Gear Pump

The pump is filled with the liquid before it is started. The liquid is trapped in between shafts. As the gears rotate, the liquid is carried to the discharge end round the teeth and is drawn to the casing. The rotating gears built up sufficient pressure to force the liquid into the delivery pipe. A little consideration will show that each tooth of the gear acts like a piston or plunger of a reciprocating pump to force liquid into the discharge pipe.

(2) Internal Gear Pump:

An internal gear pump in its simplest form, consists of two spur wheels intermeshing internally. The wheels are so designed that on one side, they form a fluid tight joint at the point of contact, and a space for crescent on the other as shown in figure.



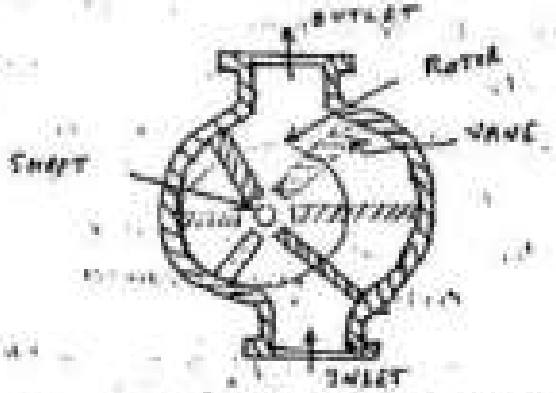
INTERNAL GEAR PUMP

crescent shaped partition between the two wheels to act as a seal between suction and discharge. This is done by placing the inner wheel eccentrically in the outer wheel. The inner wheel is keyed to the driving shaft and the outer revolves as a driven wheel.

The pump is first filled with the liquid before it is started. As the wheels rotate, the teeth come out of the mesh, near the suction end. As a result of this the space between the two wheels increases and the liquid flows into the space. As the wheels continue to rotate, the liquid is trapped between the teeth and crescent of both the wheels and flows to the discharge end. A little consideration will show at each tooth of the gear, like external gear pump, acts like a piston or plunger of a reciprocating pump to force the liquid inside the discharge pipe.

(b) Vane Pump :

A vane pump in its simplest form consists of a disc rotating eccentrically in the pump casing. The disc has a number of slots (generally 4 to 8) containing vanes, which are free to slide radially into the slots. When the motor rotates the disc, the vanes are pressed against the casing due to centrifugal force and forms a liquid tight seal. As the disc rotates the liquid is trapped in the pockets formed between the vanes and the casing. The vanes build up sufficient pressure to force the liquid into the delivery pipe.



In some design the springs are used to press the vanes against the casing. But in some other design springs are not used. In this type, the vanes are slipped out as the disc rotates due to centrifugal

(1) <u>Radial Piston Pump</u> :

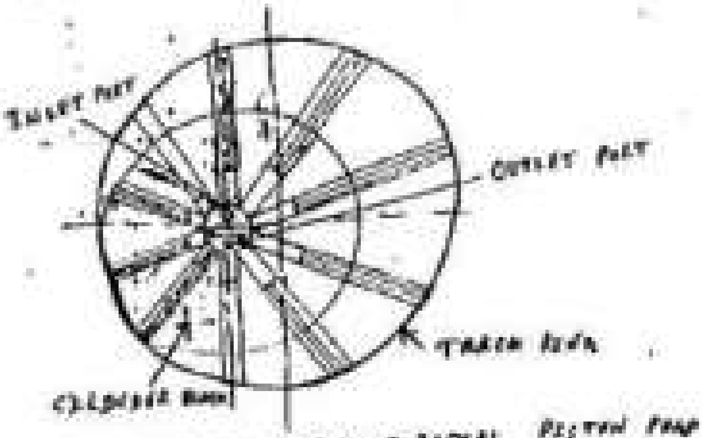
They are available in two basic types :
(a) Radial piston pump with stationary cylinder block.
(b) Radial piston pump with rotating cylinder block.

In a stationary cylinder block type radial piston pump, the cylinder block is kept stationary. Reciprocating motion is imparted to the pistons by a rotating cam. The outer end of each cylinder is connected to the is connected [redacted] to the inlet and discharge check valves, which are opened by the suction of the receding pistons and pressure of the advancing pistons build up respectively. A schematic diagram of a radial piston pump is shown in figure.

The rotating cylinder block and the pump housing with the thrust ring are positioned eccentrically which allows the radially placed pistons to move to and fro movement.

In a radial piston pump with rotating cylinder block, the thrust ring may be is connected to the driven shaft, the outer ring along with the fixed or rotating with the cylinder block along with the rotating cylinder the pistons confined by thrust ring, reciprocate into the slots provided in the cylinder block.

The cylinder block and outer ring are eccentrically placed. The pistons are thrown out by centrifugal force and pushed back because of eccentricity between rotor ring and cylinder block.



SCHEMATIC VIEW OF RADIAL PISTON PUMP

The principle of a constant delivery radial piston pump is shown in figure (3). The pumps are quite robust and work at high r.p.m with high flow rate. The pressure sometimes is as high as 450 to 500 bar.

(3) ... pre-filling is advisable to avoid cavitation.

(4) This pumps are found to fill majority of the hydraulic ... used in the market.

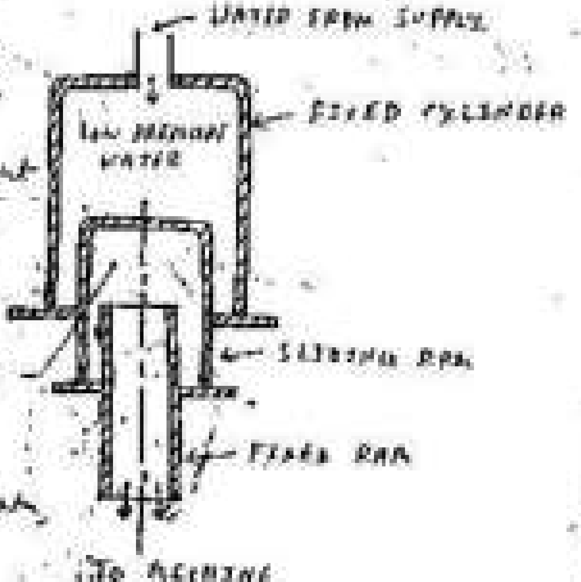
(5) Operating parameters are found to be quite high, viz. Pressure rating - from 20 to 400 bar at ... , flow rate - 20 cm³ to 50 cm³/rotation, noise level - about 75 to 90 dB (A).

(6) In most designs, pump rotation can be then reversed.

(2) Hydraulic Intensifier:

It is a device to increase the intensity of pressure of water, by means of energy available from a larger quantity of water at a low pressure.

A hydraulic intensifier in its simplest form, consists of a fixed ram through through which the water, under a high pressure, flows to the machine. A hollow sliding ram is mounted externally on the fixed ram, which contains water under a high pressure as shown in figure. This sliding ram is encased in a fixed cylinder which contains water under a low pressure from the supply as shown in figure.



The water under a low pressure presses the sliding ram on the top thus forcing it downwards to the fixed ram. This downward movement of the sliding ram increases the intensity of pressure of water in the sliding ram.

Let, A = External area of the sliding ram

a = Area of the end fixed ram

p = Intensity of pressure (of low pressure water) in the fixed cylinder, and

p^* = Intensity of pressure (of high pressure water) in the sliding ram.

We know that the total upward force

$= \text{Intensity of Pressure} \times \text{Area} = p \times a$ ---- (i)

Similarly, total downward force $= P \times A$ ---- (ii)

Since the two forces are equal, therefore equating equation (i) and (ii),

$$p \times a = P \times A$$

$$\therefore \quad P = \frac{a}{A} \times p$$

Note: Sometimes, there is a loss due to friction (k) at each Packing. In such case pressure of water in the sliding ram,

$$P = \frac{A}{a} \times p (1-k)^2$$

BEL HYDRAULIC SYSTEMS (P-349) - S R MAJUMDER

(iii) Hydraulic Accumulators :

(iii) **Accumulators:**

The accumulators are devices that are used to store energy of the fluid (Potential energy) in the hydraulic system under pressure created by an external source (pump against the dynamic /surges/ increase in system pressure etc.) against the dynamic force (weight or gravity, pressurized gas or mechanical force by springs). There are three basic different types of accumulators.

They are:

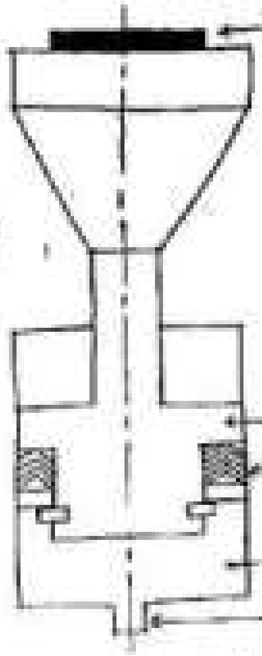
(i) Weight loaded or gravity type.

(ii) Spring loaded or mechanical type.

(iii) Gas loaded or pressurized or hydro pneumatic type.

(i) Weight loaded accumulator:

This is the oldest type accumulator. They consist of strong vertically mounted cylindrical shell which incorporates the piston with packing to prevent [redacted] leakage. A dead weight is attached at the top of the piston, which gives a large force of gravity that provides the potential energy in the accumulator.

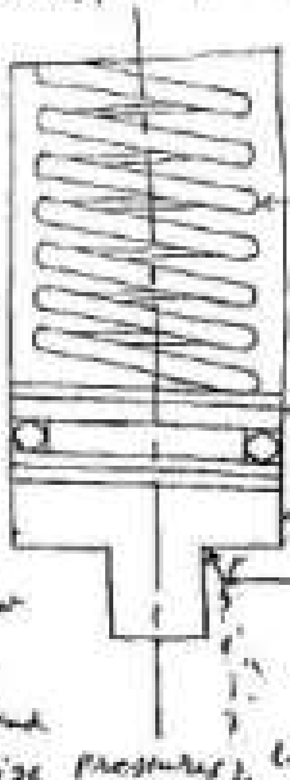


Advantage: This type of accumulator creates a constant fluid pressure (pressure may change only when the weight is changed out this is decided by the user) throughout the full volume output of the unit regardless of the rate and quantity of output. The other two types of accumulators have this quality and there is change in pressure and the fluid output pressure decreases as a function of the volume output of the accumulator.

Disadvantage: These types are extremely large in size and weight and hence they are not portable and not used in mobile application.

(ii) Spring loaded accumulator:

Spring loaded accumulators are similar in construction to weight loaded type of accumulators. In this type instead of loading the piston with weights, it is preloaded by loading the spring compression. The spring with spring compression, the piston and force acts against the piston and forcing the fluid into the system when needed. The pressure level of this type is dependent on the size and preloading of the spring. The pressure exerted by the spring is large when fully compressed and when it extends fully compressed is less. Hence the the force applied is less. Hence the pressure exerted on the fluid is not a constant. This type delivers a relatively small volume of oil at low pressures. To deliver large pressures, the size of spring required is large and hence they tend to be heavy for large pressures, large volume applications. This type of accumulator are not used



... which require sufficient space ... subjected to rubbing and it would lose its elasticity ... Thus this type is mostly used for applications which require less pressure / volume and that do not have surges and cyclic loading.

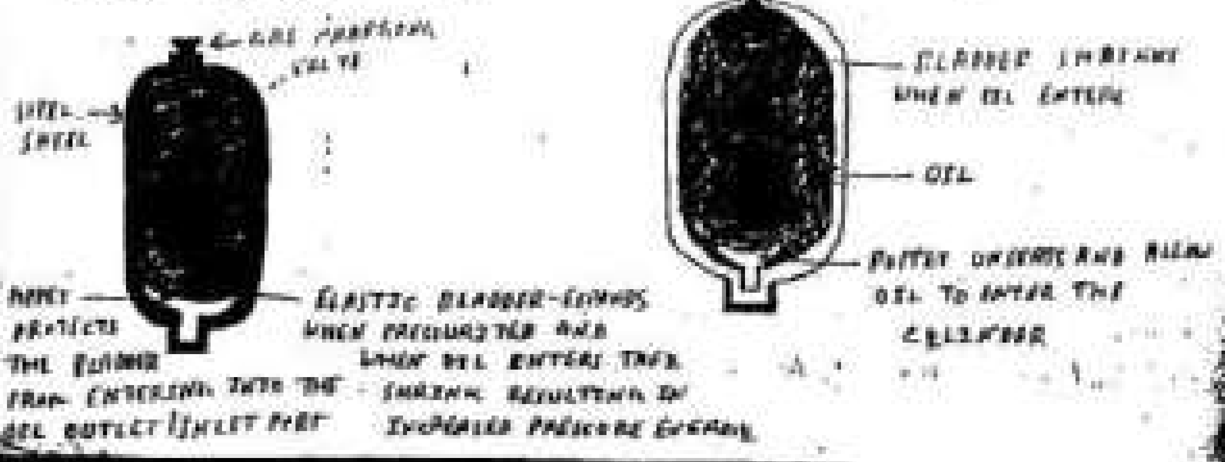
② Gas loaded accumulator :

- These accumulators are used in almost all the industrial applications. They use compressed gas to give the dynamic force and hence they are also called as "Hydro-Pneumatic accumulators".
- These accumulators obey the Boyle's gas law, which states that for a constant temperature process, the pressure of the gas varies inversely with its volume. Thus when the gases are compressed more (say half the initial volume), the pressure is increased (doubled in this case) and the compressibility of gases accounts for the storage of potential energy. This energy forces the oil out of the accumulator when gas expands due to reduction in system pressure.
- Four types of hydro pneumatic accumulator are used:

(i) Non-separator type.
(ii) Piston.
(iii) Diaphragm.
(iv) Bladder.

③ Bladder type Accumulators :

This accumulator has an elastic barrier (bladder) between the oil and gas. The bladder can be installed and removed through the seal opening at the poppet valve. They are filled by means of vulcanization. The poppet valve closes the inlet valve when the bladder is fully expanded. This prevents the bladder from entering into the opening. A shock-absorbing device protects the valve against accidental shocks during quick opening.



(ii) **Advantage:**

- They have positive sealing between gas and oil chambers.
- They light-weight bladder provides quick response as pressure regulating, pump pulsation and shock dampening applications.

(iii) **Hydraulic system:**

A hydraulic fluid power system may be described as a means of power transmission in which a relatively incompressible fluid is used as the power transmitting media. The primary purpose of hydraulic system is the transfer of energy from one location to another and the conversion of this energy to useful work.

Hydraulic power is usually generated by pumps and the energy generated is converted to useful work by hydraulic cylinders or other actuators (linear or rotary).

The transmission of this energy is accomplished by movement of the hydraulic fluid through metal tubes or elastomeric hoses, while the control of the power is achived by means of valves.

(iv) **Merits of hydraulic system:**

(1) To transfer hydraulic energy.
(2) To lubricate all parts.
(3) To avoid corrosion.
(4) To remove impurities and abrasion.
(5) To dissipate heat.

○ Properties of Hydraulic fluids :

(1) Proper lubricity
(2) Stable viscosity characteristics.
(3) Stable chemically and physically
(4) System compatibility
(5) Good heat dissipation.
(6) High bulk modulus.
(7) Adequate low temperature properties.
(8) Flash point.
(9) Low foaming tendency.
(10) Fire resistant.
(11) Prevent rust formation.
(12) Low in volatility
(13) Good demulsibility.
(14) Low coefficient of expansion.
(15) Low specific gravity.
(16) Non-toxic, easy to handle and available.

◉ Direction control valve :

(1) Seat valve or Poppet valve.
(2) Spool valve or sliding valve.

◉ Many other types of valves are often used in hydraulic system:

(1) Non return valves or check valves.
(2) Flow control valves.
(3) Pressure control valves.

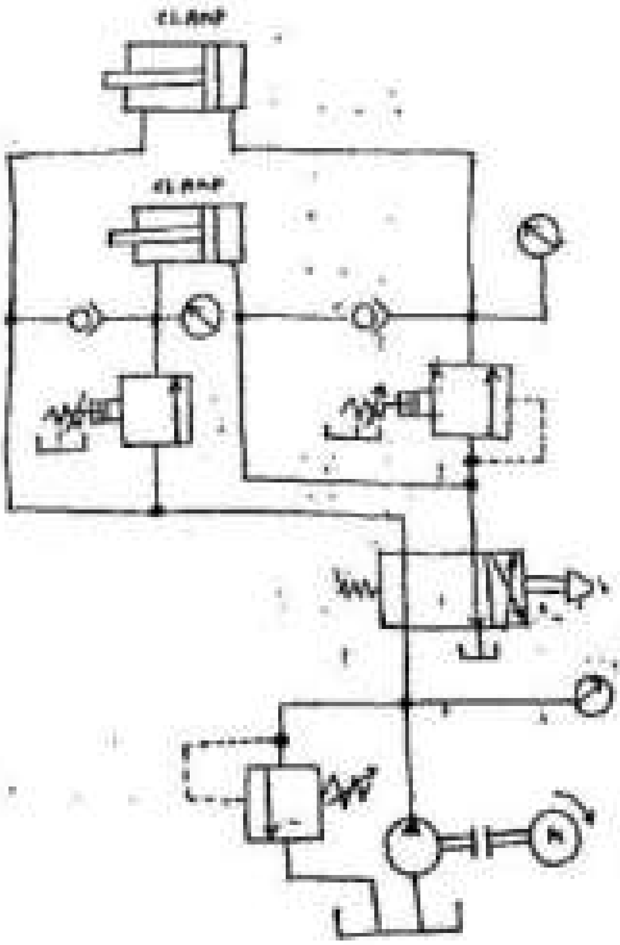
◉ Check valve: Check valves can be direct acting or pilot operated. A pilot operated version is used where the no flow characteristic of the valve is desired only for a portion of the system cycle. The main function of check valve is to block the reverse flow of oil or gas in a fluid power circuit. It are also used in a hi-low hydraulic [illegible] to isolate the high pressure from the low pressure

① [illegible] control valve : (Pneumatic control valves)

(i) [illegible] valve
(ii) Sequence valve.
(iii) Unloading valve.
(iv) Pressure reducing valve.

② **Sequence valve:**

A pressure sequence valve is used in a hydraulic system to cause various operations in a sequential order, one after another. For example a pressure sequence valve is used in a clamping and machining circuit may permit the clamping operation to take place first and when the clamping cylinder is fully extended, the machining cylinder is actuated. This means this valves will cause actions to take place in a definite order and also to maintain a predetermined minimum pressure in the primary line when the secondary operation by to be occur. Hence fluid flows freely through the primary passage to operate the first phase. As soon as spring setting of the valve is reached the valve spool lifts and all flow is diverted to the secondary port to operate the next phase of the system.



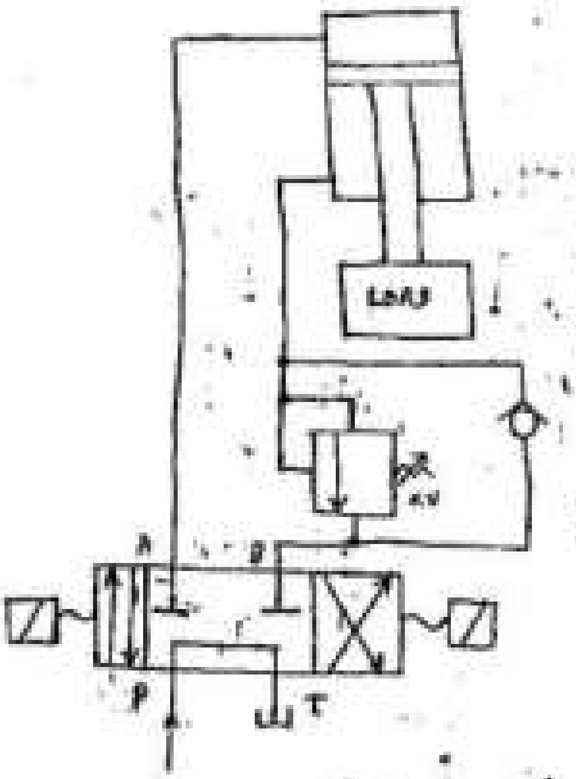
SEQUENCE CIRCUIT

(i) Counter balance valve :

A counter balance valve is used to maintain control over a vertical cylinder so that it will not fall freely because of gravity. The primary port of the valve is connected to the lower cylinder port and the secondary port to the directional control valve. The pressure setting is slightly higher than is required to hold the load from falling.

When the pump delivery is directed to the top of the cylinder, the cylinder piston is forced down causing pressure at the primary port to increase and rise the spool, opening a flow path for discharge through the secondary port to the D.C valve and subsequently to the tank. In cases where it is necessary to remove back pressure at the cylinder and increase the force potential at the bottom of the stroke, this valve too can be actuated remotely, when the cylinder is being raised, the integral check valve opens to permit free flow for returning the cylinder.

This valve is also called a back pressure valve. Application of this valve is shown in a schematic circuit diagram.



COUNTER BALANCE VALVE IN HYDRAULIC CIRCUIT

(1) Pressure control valves:

It performs the following functions:

(1) Limiting maximum system pressure as a safety measure.

(2) Regulating / reducing pressure in certain portions of the circuit.

(3) Unloading system pressure.

(4) Assisting sequential operation of actuators in a circuit with pressure control.

(5) Any other pressure related function by virtue of pressure control.

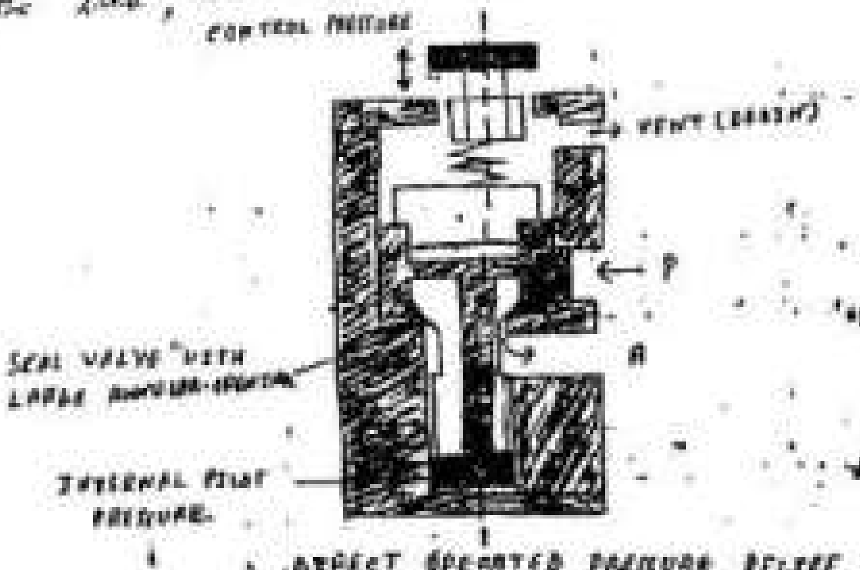
The operation of a pressure control valve is based mainly on a balance between pressure and a mechanical load, e.g. spring force biased against the oil pressure. The valve can assume various positions between fully closed and fully open conditions depending on the flow and pressure differential.

(2) Types of pressure control valves:

They are classified according to their function, type of connection, size and pressure operating range. Relief valve, sequence valve, sequence valve, unloading valve, pressure reducing valve etc.

(3) Pressure Relief valve:

Pressure relief valves are found in every hydraulic system. It is a normally closed valve connected between the pressure line and the oil reservoir. Its main purpose is to limit the pressure in a system to a prescribed maximum by diverting some or all of the pump output to the tank, when the designed set pressure is reached.



DIRECT OPERATED PRESSURE RELIEF VALVE

overcome the force of ... closed and hence it is only ... normally closed valve. When the preset pressure is ... ball unseats and allows flow through the outlet ... tank. In most of these valves an adjusting screw is provided to vary the spring force. Thus the valve can be ... to open at any pressure within the specified range. The pressure at which the valve first opens is called the cracking pressure. As the flow through the valve increases, the poppet is forced further, the resulting pressure increasing considerably ...

Direction control valve :

It is important to know how to reverse the direction of a hydraulic actuator. What mechanism is needed to start, stop & reverse a cylinder movement, change in the direction of the cylinder or a motor is executed by direction control valve. They are employed in a hydraulic system to determine the direction of the fluid in the hydraulic circuit. Sometimes they also ... a ... control.

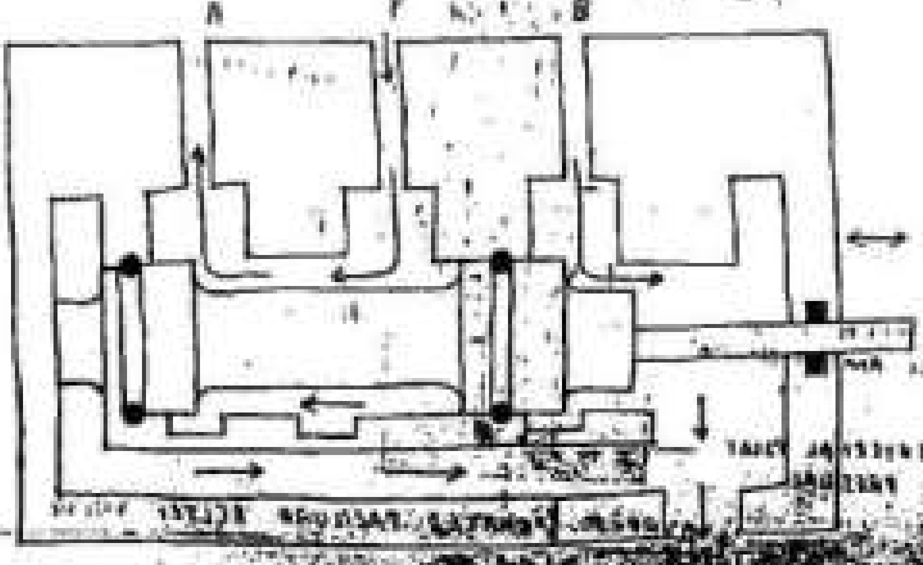
Many different constructional features inside a valve body are also possible to ... or ... the oil supply to a system.

Construction :

Two types of generally used for common direction control valves :

(i) Seat valve or poppet valve.

(ii) Spool valve or sliding valve.



DESIGN ... SPOOL VALVE

[illegible] ... cases, all the flow paths through the valve [illegible] ... is effected by sliding the internal piston ... [illegible]

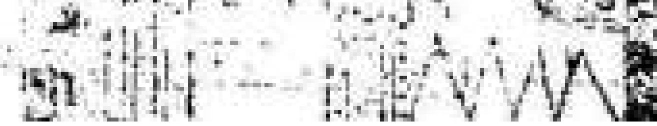
(ii) Poppet valve:

Here a poppet [illegible] like a plate is made to sit over a specially constructed, usually machined and polished seat. The advantage of this type of valve is that it has a high response and relative insensitivity to contamination. They are suited to high pressure. Sizing, however, they are less suitable for large valve sizes but since the opening load becomes excessive they may be more suitable for indirect actuation. The lift required is detected by the poppet angle.

(iii) Spool valve:

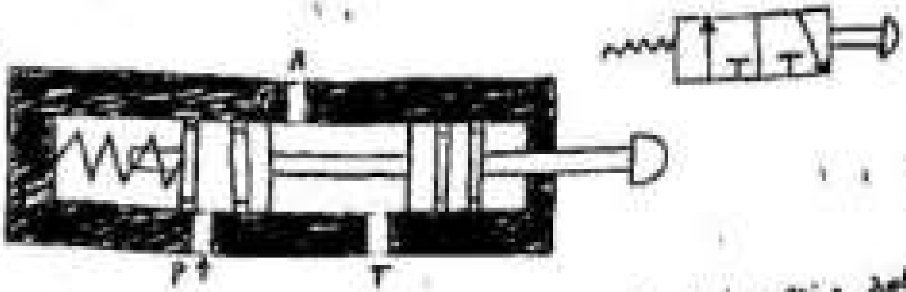
(i) Rotary spool: With a rotary spool the hydraulic fluid is directed through longitudinal grooves machined into a rotatable piston. As a rotational movement is necessary to effect changeover, this type of spool is used predominantly in manually operated valves. They are more appropriate for lower pressures.

(ii) Sliding spool: In this type of valve, there is a small piston called a spool inside the valve casing, which slides inside the casing thereby opening or closing the paths made specially inside the valve body and according to the position of the axially displaced piston (spool), the ports gets the supply. They are generally balanced permanently during operation, only spring force, frictional and flow forces are to be overcome.



3/2 DCV (Direction control valve) :

A three way direction control valve is shown in figure:

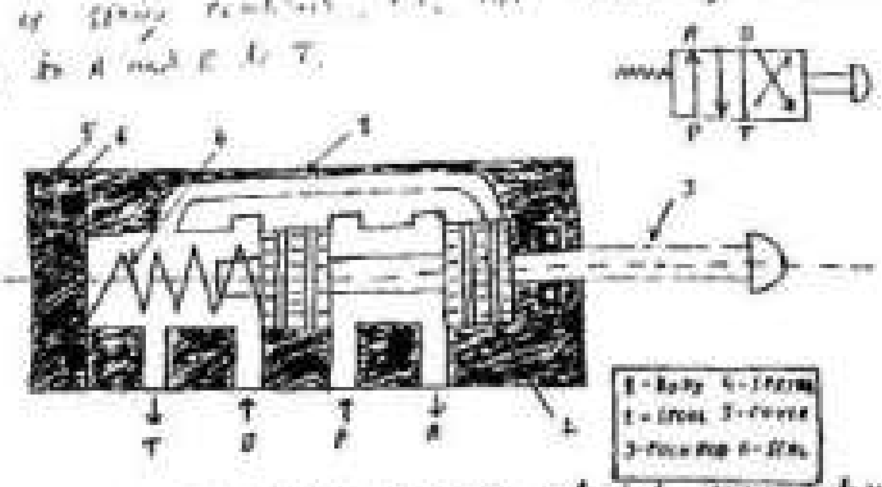


3/2 Direction control spool valve manually actuated spring return with symbol

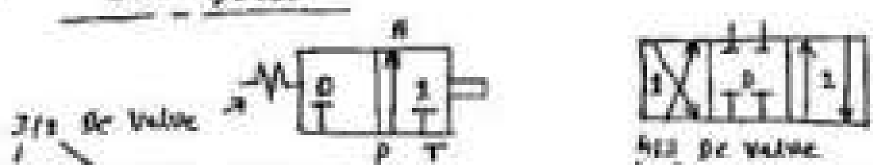
It has three ports at primary P, A and T, i.e. pump port P, actuator port A and tank port T. In one extreme position of the spool, the pressure oil flows from P to A to move the actuator, the tank port T remaining closed. In the other extreme position, the oil from P is closed and the oil from the actuator is allowed to flow to A and the oil from the actuator is allowed to flow through A to T and then to the tank. Therefore the valve alternately connects or disconnects oil supply to the cylinder by the sliding spool. As the valve has 3 ports (openings) and the valve spool has two distinct positions, the valve is called a 3-way 2-position direction control valve, i.e., 3/2 DC valve.

4/2 DCV (Direction control valve) :

A 4 way direction control valve is shown in figure. The valve consists of 4 ports P, A, B and T. Now when the valve is spring centred, i.e. not manually actuated, P connects to A and B to T.



4/2 Direction control spool valve manually actuated spring return with symbol



... are manually actuated ... the ... directly ... ports ... and ... is ... closed ... through ... is released ... the ... pression force of the spring brings the spool back to the original position. The valve is called a 4/2 directional ... valve ... the valve ... and ... of two distinct positions ...

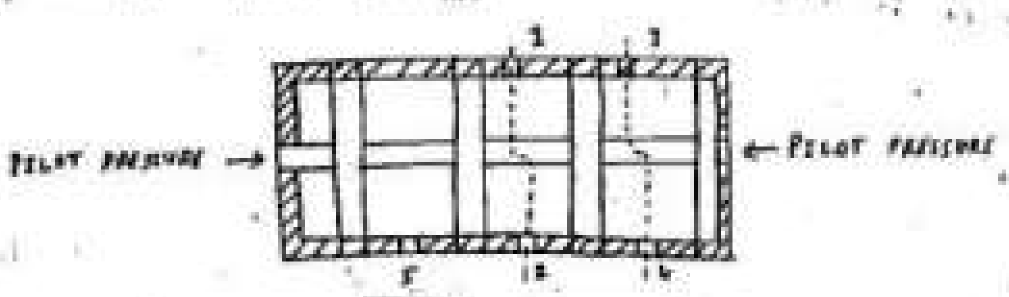
4/3 way (DC) valve :

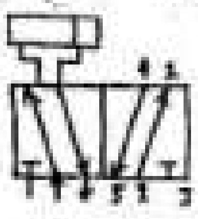
The 4/3 way valve has four ports and three positions. An example of the 4/3 way valve is the plate slide valve with hand or foot actuation. By turning two discs, channels are connected with one another.

In this circuit using these valves the lines of the 4/3 way valve are closed in the middle position. This enables the piston rod of a cylinder to be stopped in any position over its stroke range, although intermediate positions of the piston rod cannot be closed be located with accuracy. Owing to the compressibility of air, another position will be assumed if the load on the piston rod changes.

5/2 way (DC) valve :

The 5/2 way valve has five ports and two positions. The 5/2 way valve is used primarily as a control element for the control of cylinders. An example of the 5/2 way valve, the longitudinal slide valve, uses a pilot spool as a control component. This connects or separates the corresponding lines by means of longitudinal movements. The required actuating force is lower, because there are no opposing forces due to compressed air or spring.





... type of actuation can be used with ... i.e. manual, mechanical, electrical or pneumatic ... type of actuation can also be used according to valve ... its starting position.

The actuation travel is considerable larger than with seat valves. Sealing presents a problem in this type of slide valve. This type of fit known in hydraulics as metal to metal (lapped spool), requires the spool to fit precisely in the bore of the housing.

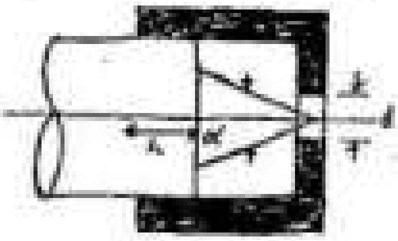
⑩ Throttle valve :

A throttle valve controls the flow rate Q by increasing or decreasing the area of the flow path. It can either be fixed or adjustable throttle. Q depends upon the pressure differential between two ports (ΔP). With changes in pressure, for the same opening the flow is different. As water resistance due to the load to be moved by the actuator has influence on the changes in pressure, the velocity of flow cannot be kept constant with a normal throttle valve.

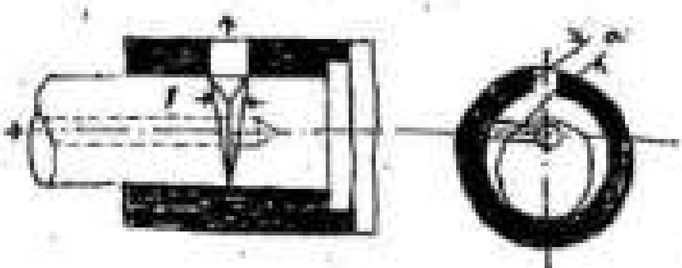
The l/d ratio of the throttle orifice may influence Q and pressure drop (ΔP) as stated below :

(i) l/d ratio is large – Q and ΔP independent on the viscosity of oil due to length of the throttle and resultant friction.

(ii) l/d ratio is small – Q and ΔP independent of the viscosity of oil as throttle length is almost zero.



NEEDLE THROTTLE



THROTTLE FORM

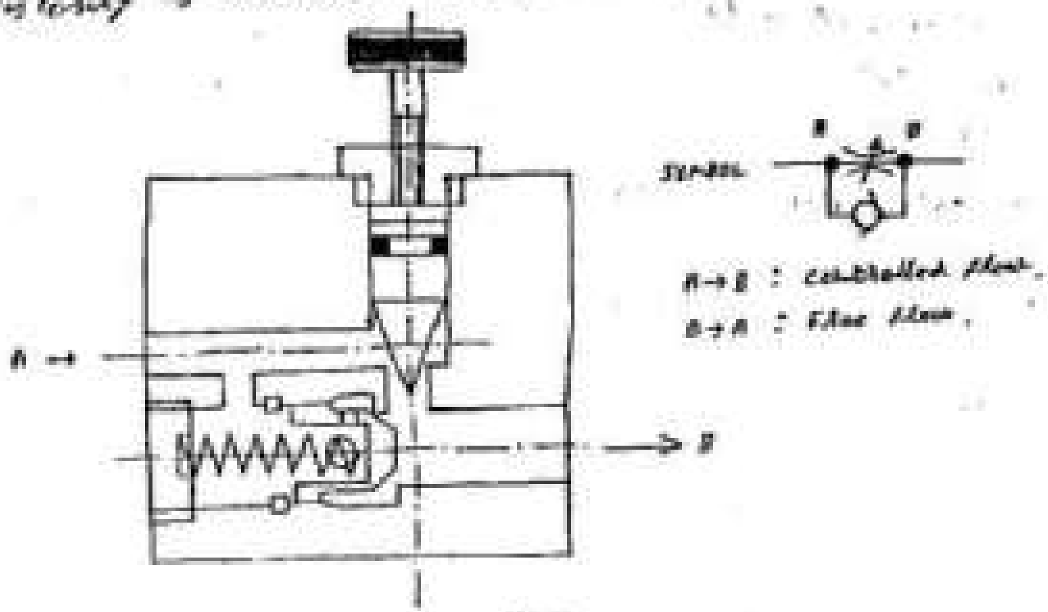
In a control technique system if the resistance due to work is kept constant, a constant speed is possible by a throttle valve only. Very often a non return valve can be in-built with a throttle valve to provide regulated flow in one direction and full flow or free flow from the opposite direction.

⑧ Flow Regulation (or control) valve (Pressure drop Independent)

Precision machine tools and other equipment needs a constant speed free from influence of external resistance and temperature. A pressure compensated flow control valve can meet such requirements of a hydraulic system which will provide a stepless adjustable speed control over a wide spectrum. The function of such a valve is to allow a constant predetermined amount of oil (lit/min) independent of pressure drop and temperature across the valve, as demanded above. The above function can be achieved if the two conditions as given below, are fulfilled:

(a) constant pressure drop on the adjustable throttle path.

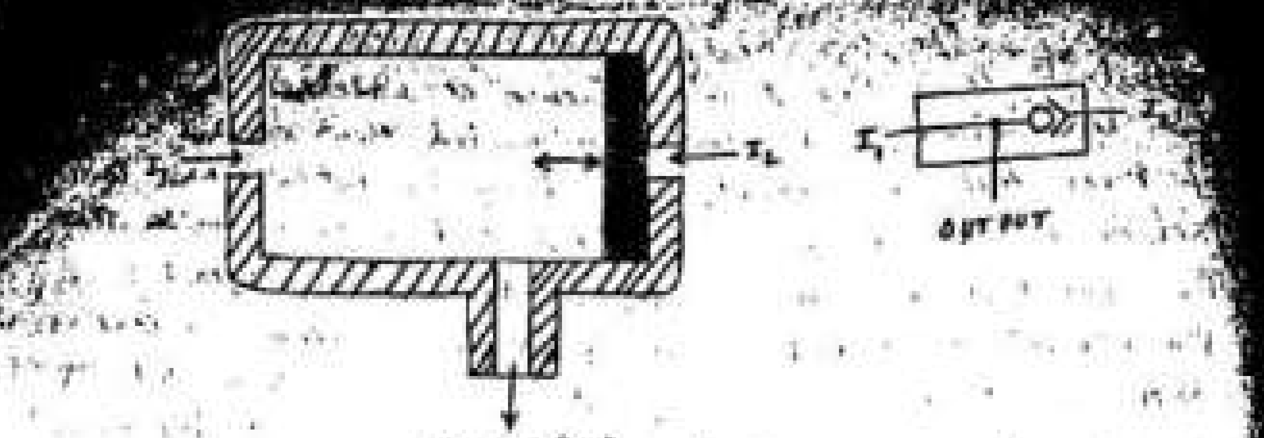
(b) The form of the throttle should be such that the influence of viscosity is minimum or insignificant.



NON RETURN FLOW CONTROL VALVE

⑨ Shuttle valve : logic or function :

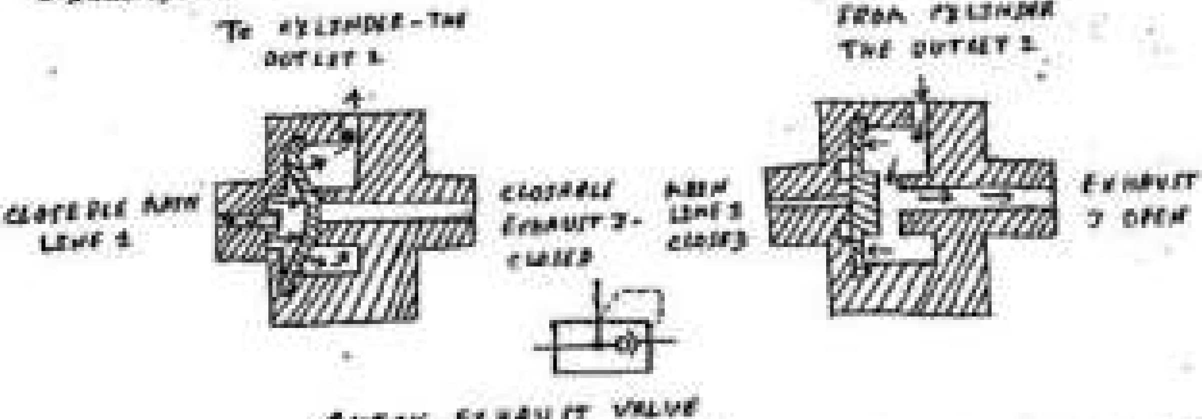
This non return element has two inlets "I_1" and "I_2" and one outlet "O". If compressed air is applied to the first inlet "I_1", the valve seat seals the opposite inlet "I_2", the air flows from "I_1" to "O". Inlet "I_2" is closed. If air passes from "I_2" to "O". A signal is generated at the outlet. When the air flow is reversed, i.e. a cylinder or valve is exhausted, the seat remains in the previously assumed position because of the pressure conditions. This valve is also called and OR element. If a cylinder or control valve is to be actuated two or more positions, one or more shuttle valves should be used.



The shuttle valve automatically allows the high pressure to the output port while it blocks the low pressure inlet. The spool is then floating with an open center action. At either end of the spool it has two inlet ports. The schematic representation of the shuttle valve and its ANSI symbol is shown in fig above figure.

③ **Quick exhaust valve :**

Quick exhaust valves are used to increase the piston speed of cylinders. This enables lengthy return times to be avoided, particularly with single acting cylinders. The principle of operation is to allow the cylinder to retract at its near maximum speed by reducing the resistance to flow of the exhausting air during motion of the cylinder. To reduce resistance, the air is expelled to atmosphere close to the cylinder via a large orifice opening. The valve has a closable supply connection 1, a closable exhaust 3 and an outlet 2.



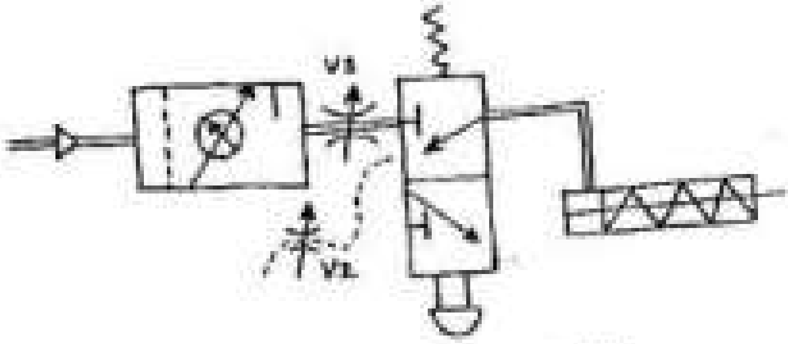
QUICK EXHAUST VALVE

If pressure is applied at port 1, then the sealing disc covers the exhaust 3, whereby the compressed air passes from 1 to 2. If pressure is no longer applied at 1, then the air from 2 moves the sealing disc against port 1 and closes this, whereby the exhaust air immediately vents to atmosphere.

There is no need for the air to pass through a long possibly restricted path to the directional control valve via the operating lines. It is advantageous to mount the quick exhaust valve directly on the cylinder or as near to it as possible.

⑱ control of single acting cylinder:

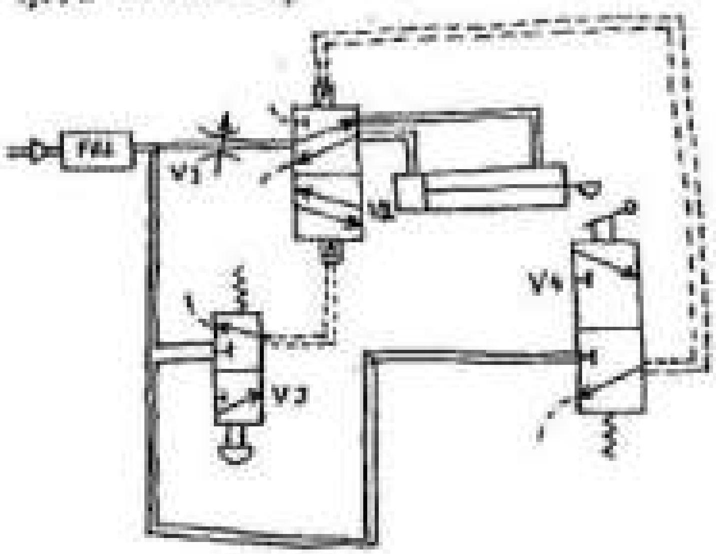
The circuit consists of a 3/2 way push button operated DCV to control a single acting cylinder. Initially the flow is blocked and when the push button is operated the cylinder extends and when the button is released the cylinder retracts by the compression spring located at the cylinder rod end of the cylinder. The adjustable flow control valves "V1" and "V2" can control the rate of extension and retraction of the cylinder.



CONTROL OF SINGLE ACTING CYLINDER

⑲ Semi automatic control of a double acting actuator (cylinder):

It is possible to achieve automatic retraction of the cylinder when the cylinder has completed its function during its extension stroke. The below circuit shows such a possibility of semi automating a cylinder.



CYLINDER CYCLE TIMING SYSTEM

[illegible] below circuit the 5/2 way pilot [illegible] 'V2' controls the cylinder movement. The DCV is controlled by two push buttons 'V3' and 'V4'. The flow is allowed only through the flow control valve 'V1'. When the push button 'V3' is operated to spring offset position, it pilot actuates the DCV 'V2' to allow the compressed air to enter the piston side of the cylinder to extend. At the end of the extension stroke, the piston can operates the push button 'V4'. This pilot actuates the DCV 'V2' to shift its position and allow the air to enter the rod side and retract the cylinder.

Thus in applications which require automatic retraction of the cylinder the above circuit design is used. It is possible to achive the control. The timing of extension and retraction can be achived by keeping a flow control valve in the upstream of the actuator.