# LECTURE NOTES

**ON**

# HYDRAULIC MACHINES AND INDUSTRIAL FLUID POWER

**5TH SEMESTER MECHANICAL**

**BY**


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# Chapter-1 HYDRAULIC TURBINES

## Definition of hydraulic Machines :-

→ Hydraulic machines are defined as those machines which convert either hydraulic energy into Mechanical energy (which is further converted into energy of Electrical or Electrical energy.) or Mechanical energy into hydraulic energy.

→ The hydraulic machines, which convert the hydraulic energy into Mechanical energy, are called "Turbines".

→ The hydraulic machines which convert the mechanical energy into hydraulic energy are called "pumps".

The study of hydraulic machines consists of study of "turbines" and "pumps".

Here, turbines consists of mainly study of

i) Pelton turbine

ii) francis Turbine

iii) Kaplan turbine

Pumps consist of study of

i) Centrifugal pump

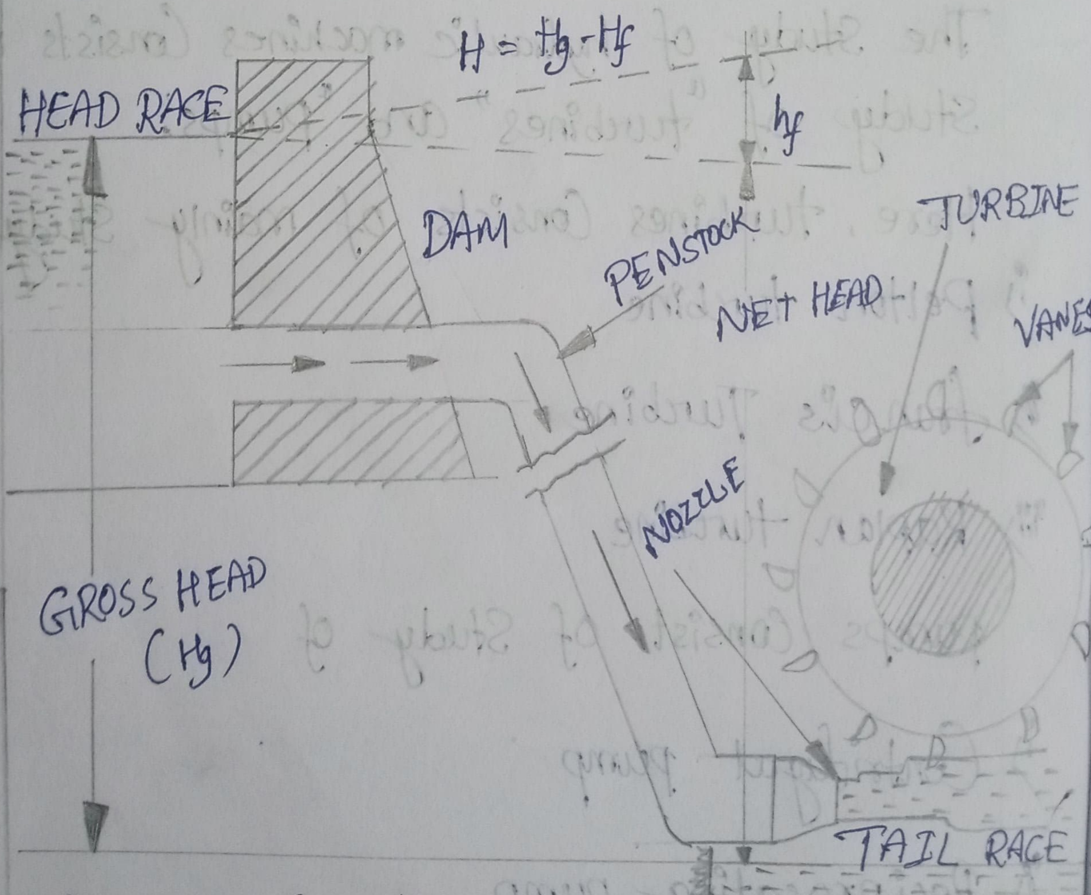
ii) Reciprocating pump

## TURBINES :-

→ Turbines are defined as the hydraulic machines which Convert hydraulic Energy into Mechanical Energy.

→ This Mechanical Energy is used in running an electric generator which is directly Coupled to the Shaft of the turbine. the Mechanical Energy is Converted into Electrical Energy.

→ The electric power which is obtained from the hydraulic Energy (Energy of water) is known as "Hydro-electric Power".




(Layout of a hydro-electric power plant)

# General Layout of a Hydro-electric power plant

General Layout of a hydro-electric power plant which Consists of :

i) A dam Constructed across a river to store water.

ii) Pipes of large diameters Called "penstocks" which carry water under pressure from the storage reservoir to the turbines.

→ These pipes are made of Steel or reinforced Concrete.

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

iv) Tail race, which is a channel which carries the water away from the turbines after the water has worked on the turbines.

The Surface of water in the tail race is also known as tail race.

## Definitions of Heads :-

### 1. Gross Head :-

The difference between the head race Level and tail race Level when no water flowing is known as "Gross Head".

It is denoted by "Hg".

2) Net head :-

→ It is defined as the head available at the inlet of the turbine.

→ It is also called as effective head.

→ When water is flowing from head race to the turbine, a loss of head due to bend or pipe fittings loss at the entrance of penstock etc.

→ It is denoted by "$H_f$".

→ If "$h_f$" is the head loss due to friction between penstocks and water than the net head on turbine in $\boxed{H = H_g - h_f}$

Where as, $H_g$ = Gross Head

$$h_f = \frac{4fLV^2}{D \times 2g}$$

where as, V = Velocity of flow in penstock

L = Length of penstock

D = Diameter of penstock

3) Efficiencies of a turbine :-

a) Hydraulic Efficiency, $\eta_h$

b) Mechanical Efficiency, $\eta_m$

c) Volumetric Efficiency, $\eta_v$

d) Overall Efficiency, $\eta_o$

a) Hydraulic Efficiency :-

→ It is the ratio of power Given by water to the runner to the power Supplied at inlet of the turbine.

→ It is denoted by $\eta_h$

$$\eta_h = \frac{\text{Power Delivered to runner}}{\text{Power Supplied at inlet}}$$

$$= \frac{R.P.}{W.P.}$$

Where R.P. = power delivered to runner i.e. runner power.

$$= \frac{\frac{W}{g}\left[V_{w_1} \pm V_{w_2}\right] \times U}{1000} \text{ KW for pelton turbine.}$$

$$= \frac{\frac{W}{g}\left[V_{w_1}u_1 \pm V_{w_2}U_2\right]}{1000} \text{ KW for Radial flow turbine}$$

W.P. = Power Supplied at inlet of the turbine and also Called water power.

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

Where, as

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

= $\rho g Q$ in which

$Q$ = Volume of water/s

$V_{w_1}$ = Velocity of whirl at inlet

$V_{w_2}$ = Velocity of whirl at outlet.

$u$ = Tangential velocity of vane

$u_1$ = Tangential velocity of vane at inlet for radial vane

$u_2$ = Tangential velocity of vane at outlet for radial vane

$H$ = Net Head on the turbine.

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

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

for water $\rho = 1000 \ kg/m^3$

$$\therefore W.P. = \frac{1000 \times 9.81 \times Q \times H}{1000}$$

$$W.P. = g \times Q \times H \ (Kw)$$

b) Mechanical Efficiency ($\eta_m$): -

The ratio of the power available at the shaft of the turbine to the power delivered to the runner is defined as the "Mechanical Efficiency".

It is denoted by $\eta_m$

$$\eta_m = \frac{\text{Powere at the Shaft of the turbine}}{\text{powere Delivered by watere to the runner}}$$

$$= \frac{S.P.}{R.P.}$$

c) Volumetric Efficiency :-

The ratio of volume of watere actualy striking the runnere to the Volume of water supplied to the turebine is defined as "Volumetric Efficiency".

→ It is denoted by "$\eta_v$".

$$\eta_v = \frac{\text{Volume of watere actually striking the runnere}}{\text{Volume of watere Supplied to the turbine}}$$

d) Overall Efficiency :-

It is defined as the ratio of power available at the shaft of the turbine to the powere Supplied by the water at the inlet of the turebine.

→ It is denoted by "$\eta_o$"

$$\eta_o = \frac{\text{Volume available at the shaft of the turebine}}{\text{Powere Supplied at the inlet of the turebine}}$$

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

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

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

Shaft water is Commonly represented by P.

but water power in kW

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

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

Where, P = Shaft power

## *) Classification of hydraulic turbines :-

The hydraulic turbines are classified according to the type of Energy available at the inlet of the turbine, direction of flow through the vanes, Head at the inlet of the turbine and Specific Speed of the turbines.

1. According to the type of Energy at inlet:

a) Impulse turbine,

b) Reaction turbine.

2) According to the direection of flow through reunner :

a) Tangential flow turbine b) Radial flow turbine

c) Axial flow turbine d) Mixed flow turbine

3) According to the head at the inlet of turbine :

a) High Head turbine c) Low Head turbine

b) Medium Head turbine

4) According to the Specific Speed of the turbine

a) Low Specific Speed turbine

b) Medium Specific Speed turbine

c) High Specific Speed turbine

*) Impulse Turbine :-

The inlet of the turbine, the Energy available is only Kinetic Energy, the turbine is known as "Impulse Turbine".

*) Reaction Turbine :-

The inlet of the turbine, the water possesses Kinetic Energy as well as preessuree Energy, the turbine is known as "Reaction turbine".

*) Pelton Wheel (ore turebine) :-

→ The pelton wheel ore pelton turebine is a tangential flow impulse turbine, the water strikes the bucket along the tangent of the runnere.

→ The pressure at the inlet and outlet of the turbine is atmospherce.

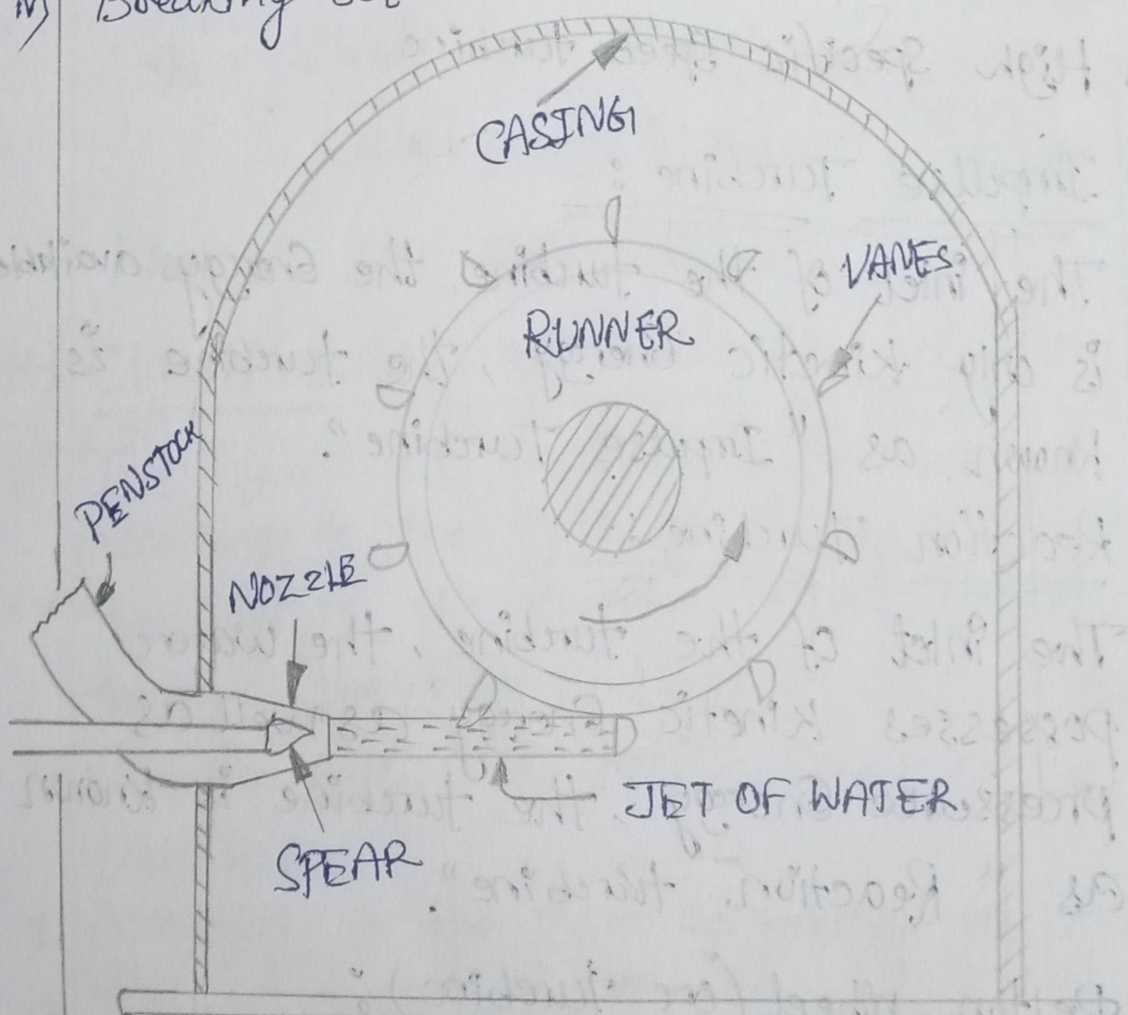
** The main parts of the pelton turbines are

i) Nozzle & flow regulating arrangement

ii) Runnere and buckets.

iii) Casing

iv) Breaking jet

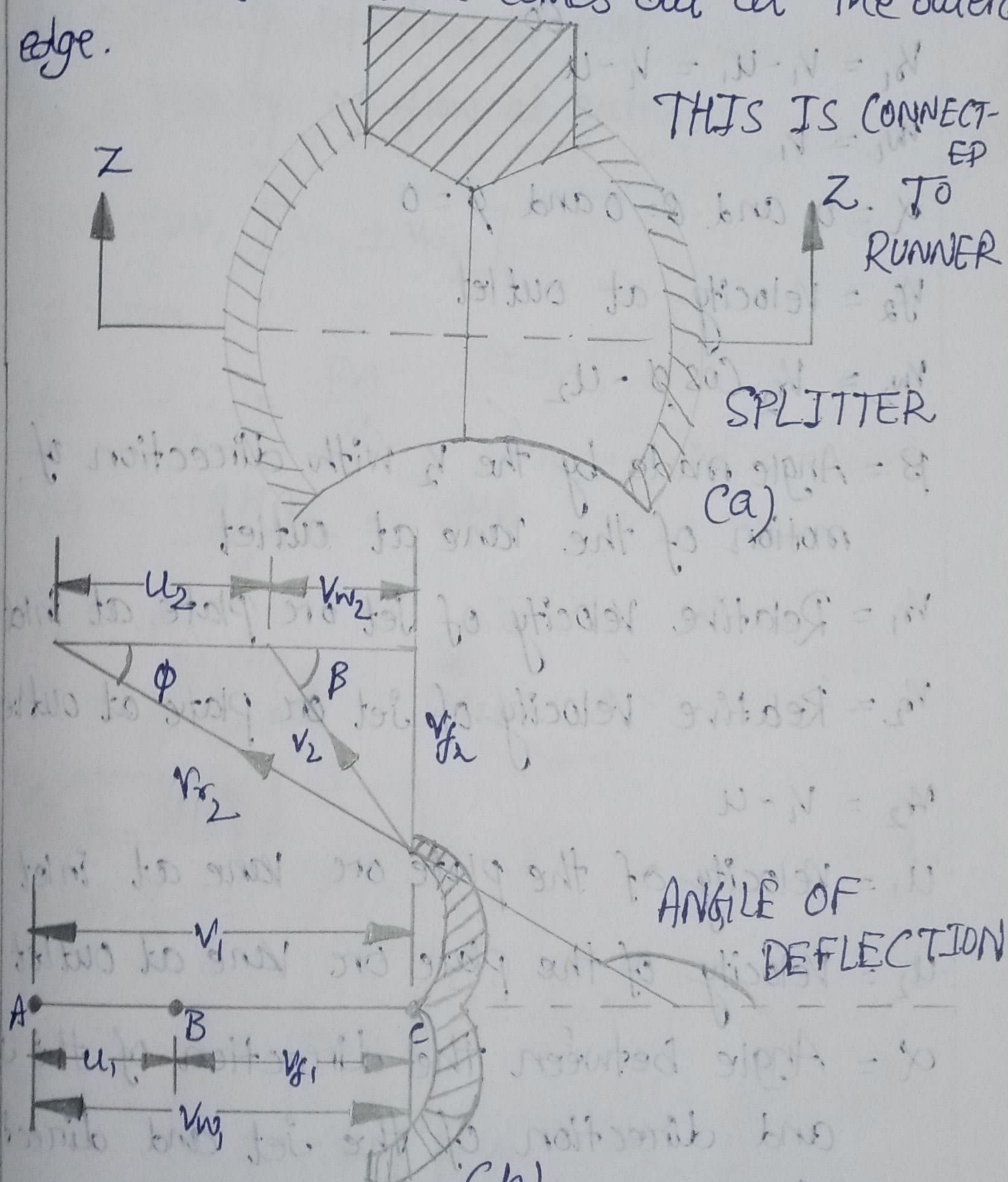



(Diagram of Pelton turbine)

## *) Velocity triangles and workdone for pelton wheel :-

→ The shape of the vanes or buckets of the pelton wheels. The jet of water from the nozzle strikes the bucket at the splitter, which splits up the jet into the two parts.

→ These parts of the jet, glides over the inner surfaces and comes out at the outer edge.




( Diagram of shape of bucket )

H = Net head acting on the pelton wheel

$H = H_g - H_f$

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

$V_1$ = Velocity of jet at inlet

$V_1 = \sqrt{2gH}$

$u = u_1 = u_2 = \dfrac{\pi DN}{60}$

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

$V_{w_1} = V_1$

$\alpha = 0$ and $\theta = 0$ and $\phi = 0$

$V_{f_2}$ = Velocity at outlet

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

$\beta$ = Angle made by the $V_2$ with direction of motion of the vane at outlet

$V_{r_1}$ = Relative velocity of jet or plate at inlet

$V_{r_2}$ = Relative velocity of jet or plate at outlet

$V_{r_2} = V_1 - u$

$u_1$ = Velocity of the plate or vane at inlet

$u_2$ = Velocity of the plate or vane at outlet

$\alpha$ = Angle between the direction of the jet and direction of the jet and direction of motion of the plate [Guide blade angle]

$\theta$ = Vane angle at inlet or

Angle made by the Relative velocity or

Angle made by the with direction of motion at inlet.

$\phi$ = Angle made by the relative velocity or angle made by the direction of motion at inlet

$V_{w_1}$ = Velocity of whirl at inlet

$V_{w_2}$ = Velocity of whirl at outlet

$V_{f_1}$ = Velocity of flow at inlet

$V_{f_2}$ = Velocity of flow at outlet

$$F = \rho a v_1 [V_{w_1} \pm V_{w_2}]$$

$$F = ma$$

$$M = \frac{F}{a} = \frac{PA}{a}$$

$$M = \frac{\rho g H A}{a} = \rho H A$$

$$F_r = ma$$

$$= \frac{\rho H A}{\cancel{a}} \times a = \rho H A a$$

$$\rho H A \frac{V}{S} = \frac{\rho A H}{t} \times V$$

$$= \rho a v_i V$$

$$= \rho a v_1 [V_{w_1} + V_{w_2}]$$

Where as,

$\rho$ = Density of water

$a$ = Area of jet

Net workdone $= \frac{W_{net}}{Sec}$

$W = \frac{F \times S}{Sec}$

$= F_x \times U$

$= \rho a v_1 [V_{w_1} + V_{w_2}] \times U$ —— (i)

(watt)

If we devide by 1000 in equ-(i) then in kW.

Workdone per unit weight of water of striking per sec $= \frac{m \times g}{S}$

$= \rho a v_1 \times g$

$W_{net} = \frac{\cancel{\rho a v_1} [V_{w_1} + V_{w_2}] U}{\cancel{\rho a v_1} \times g}$

$W_{net} = \frac{[V_{w_1} + V_{w_2}]}{g \times U}$

$K.E = \frac{1}{2} m v^2$

$= \frac{1}{2} (\rho a v_1) \times v_1^2$

<u>Hydraulic Efficiency ($\eta_h$):-</u>

$\eta_h = \frac{\text{Workdone per second}}{\text{K.E. of jet per second}}$

$$= \frac{\rho a v_1 [V_{w_1} + V_{w_2}] U}{\frac{1}{2} \rho a v_1 \times v_1^2}$$

$$\Rightarrow \eta_h = \frac{2U [V_{w_1} + V_{w_2}]}{v_1^2}$$

$V_{w_1} = v_1$

$V_{r_1} = v_1 - u_1 = (v_1 - U)$

$V_{r_2} = (v_1 - u)$

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

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

$= (v_2 - u) \cos\phi - u$

$$\eta_h = \frac{2U [v_1 + (v_1 - u) \cos\phi - U]}{v_1^2}$$

$$\eta_h = \frac{2U [(v_1 - u) + (v_1 - u) \cos\phi]}{v_1^2}$$

$$\eta_h = \frac{2U [v_1 - u] (1 + \cos\phi)}{v_1^2}$$

The efficiency will be maximum

$$\frac{d}{du}(\eta_h) = 0$$

$$\Rightarrow \frac{d}{du}\left[\frac{2U (v_1 - u)(1 + \cos\phi)}{v_1^2}\right] = 0$$

$$\Rightarrow \frac{1+\cos\phi}{V_1^2} \frac{d}{du}(2uV_1 - 2u^2) = 0$$

$$\Rightarrow \frac{d}{du}[2uV_1 - 2u^2] = 0$$

$$\Rightarrow \frac{d[2uV_1]}{du} - \frac{d[2u^2]}{du} = 0$$

$$\Rightarrow 2V_1 \times 1 - 2 \times 2u = 0$$

$$\Rightarrow 2V_1 - 4u = 0$$

$$\Rightarrow 2V_1 = 4u \Rightarrow u = \frac{2V_1}{\cancel{4}\,2} \Rightarrow \frac{V_1}{2} = u$$

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

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

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

$$= \frac{2u(2u-u)(1+\cos\phi)}{4u^2}$$

$$= \frac{(2 \times u \times u)(1+\cos\phi)}{\underset{2}{\cancel{4}}u^2}$$

$$\boxed{\eta_h = \frac{1+\cos\phi}{2}}$$

$$\boxed{V_1 = C_V\sqrt{2gH}}$$

$C_V$ = Co-efficient of velocity.

$C_V = 0.98$ or $0.99$

$$\boxed{u = \phi\sqrt{2gH}}$$

$\phi$ = Speed ratio = 0.432 or 0.48

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

$m = \frac{D}{d}$ ; D = pitch diameter of pelton wheel

d = jet diameter

<u>Number of buckets :-</u>

$$Z = \frac{15+D}{2d} = 15 + 0.5m$$

Basically one jet is there.

Q1 A pelton wheel has a mean bucket speed of 10 meters per sec with a jet of water flowing at the rate of 700 litre/s under a head of 30 meters. The buckets deflect the jet through an angle of 160°. Calculate the power given by water to the runner and the hydraulic efficiency of the turbine. Assume Co-efficient of velocity 0.98

Given data :-

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

Discharge, $Q = 700$ litres/s

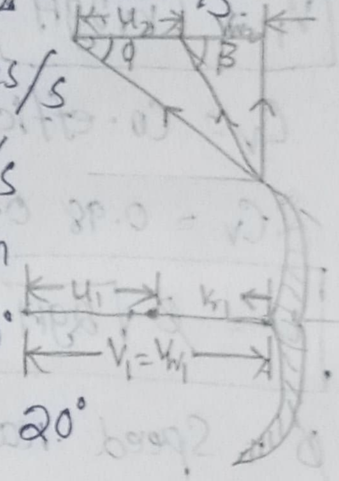
$= 0.7$ m$^3$/s

Head of water, $H = 30$m

Angle of deflection, $= 160°$

Angle, $\phi = 180° - 160° = 20°$

Co-efficient of velocity $(C_v) = 0.98$



The velocity of jet, $V_1 = C_v\sqrt{2gH}$

$= 0.98\sqrt{2 \times 9.81 \times 30}$

$= 23.77$ m/s

$\therefore V_{r_1} = V_1 - u_1$

$= 23.77 - 10 = 13.77$ m/s

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

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

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

$= 13.77\cos 20° - 10$

$= 2.93$ m/s

Workdone $= 0.7 \times 1000\left[23.77 + 2.93\right]10$

$= 186900$ Nm/s

$\therefore$ power Given to turbine $= \dfrac{186900}{1000}$

$P = 186.9$ KW

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

$$\eta_h = \frac{2[23.77 + 2.93]\,10}{23.77^2}$$

$\eta_h = 0.945 = 94.5\%$ (Ans)

Qn A pelton wheel is to be designed for the following specifications.

S.P. = 11,772 KW , Speed (N) = 750 r.p.m.

Head = 380 meters , Overall Efficiency = 86%

jet diameter is not exceed to one-sixth of the wheel diameter.

Determine :-

i) The wheel diameter.

ii) The number of jets required.

iii) Diameter of the jet.

Assume $C_v = 0.985$ Speed ratio ($\phi$) = 0.45

Given data :-

Shaft power (S.P.) = 11,772 KW

Head (H) = 380m

Speed (N) = 750 r.p.m.

Overall efficiency ($\eta_o$) = 86% or 0.86

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

Co-efficient of velocity $(C_v) = 0.985$

Speed ratio $(\phi) = 0.45$

Velocity of jet $(v_1) = C_v \sqrt{2gH}$

$$= 0.985 \sqrt{2 \times 9.81 \times 380}$$

$$= 85.05 \text{ m/s}$$

The Velocity of wheel, $U = U_1 = U_2$

$= \text{Speed ratio} \times \sqrt{2gH}$

$= 0.45 \times \sqrt{2 \times 9.81 \times 380}$

$= 38.85 \text{ m/s}$

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

$\Rightarrow 0.86 = \frac{11772}{\rho g Q H}$

$\Rightarrow Q = \frac{11772 \times 1000}{1000 \times 9.81 \times 380 \times 0.86} = 3.67$

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

$\Rightarrow D = \frac{U \times 60}{\pi N} = \frac{38.85 \times 60}{\pi \times 750} = 0.98$

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

$= \frac{0.98}{6} = 0.164 m$

Number of jet $= \frac{Q}{q}$

$Q = A \times V_1$

$Q = \pi/4 \times d^2 \times 85.05$

$Q = \pi/4 \times 0.163^2 \times 85.05$

$Q = 1.77 \ m^3/s$

$\therefore$ Number of jets $= \dfrac{\text{Total discharge}}{\text{Discharge of one jet}}$

$= \dfrac{Q}{q} = \dfrac{3.67}{1.77} = 2.07$

$= 2$ jets

3) A pelton wheel is having a mean bucket diameter of 1m and is running at 1000 r.p.m. the net head on the pelton wheel is 700m. If the side clearance angle is 15° and discharge through nozzle is 0.1 $m^3/s$ find

i) power available at the nozzle

ii) hydraulic efficiency of the turbine.

Given data :-

Diameter of wheel, D = 1m

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

Side clearance Angle, $\phi = 15°$

Net Head on turbine, H = 700m

Discharge, $Q = 0.1 \ m^3/s$

$U = \dfrac{\pi D N}{60} = \dfrac{\pi \times 1 \times 1000}{60} = 52.35 \ m/s$

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

$$= \frac{1000 \times 9.81 \times 0.1 \times 700}{1000}$$

$$= 686.7 \text{ KW}$$

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

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

$$= 1\sqrt{2 \times 9.81 \times 700}$$

$$= 117.19 \text{ m/s}$$

$$\boxed{V_1 = V_{w_1}}$$

$$V_{w_2} = 117.19 \times \cos 15^\circ = 52.38$$

$$V_{w_2} = 60.81$$

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

$$= \frac{2(117.19 - 52.38)(1 + \cos 15^\circ)52.38}{117.19^2}$$

$$= 0.9719 = 97.19\%$$

*) <u>Radial flow</u> <u>Reaction turbine</u> :-

<u>Radial flow</u> :- Those turbines for which the water flows in the radial directions

is Called as "Radial flow" Turbine.

→ These are classified as 3 types

i) Inward Radial flow turbine

ii) Outward Radial flow turbine

**Inward Radial flow turbine :-**

If the water flows from outwards to inwards through the runner of turbine, the turbine is known as "Inward Radial flow turbine."

**Outward Radial flow turbine :-**

If the water flows from Inwards to outwards through the runner of turbine, the turbine is known as "Outward Radial flow turbine."

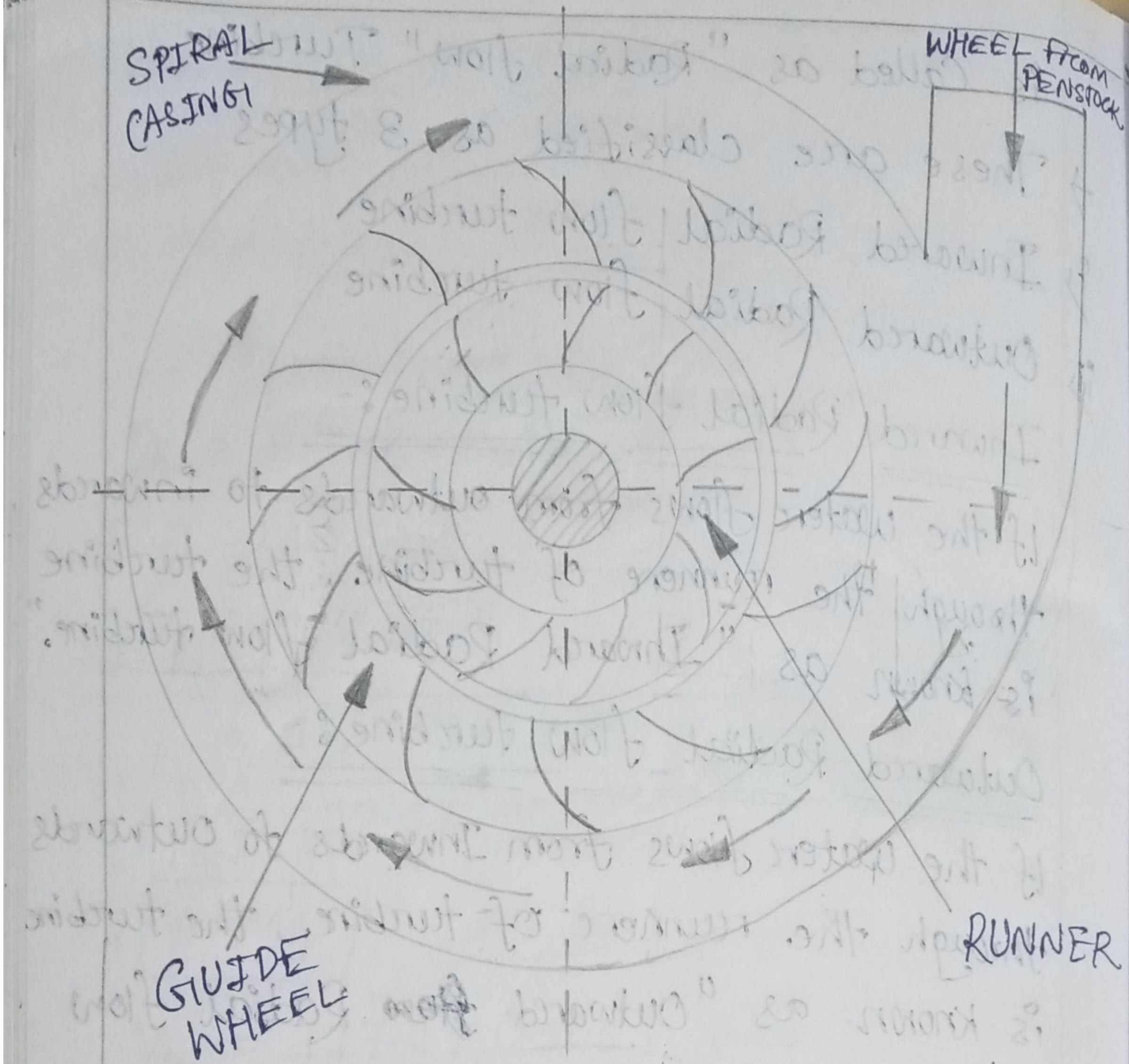
**Reaction turbine :-**

The turbine posses Kinetic Energy as well as pressure Energy is Called as "Reaction turbine".

**Main parts of a Radial flow reaction turbine :-**

i) Casing        iii) Runner

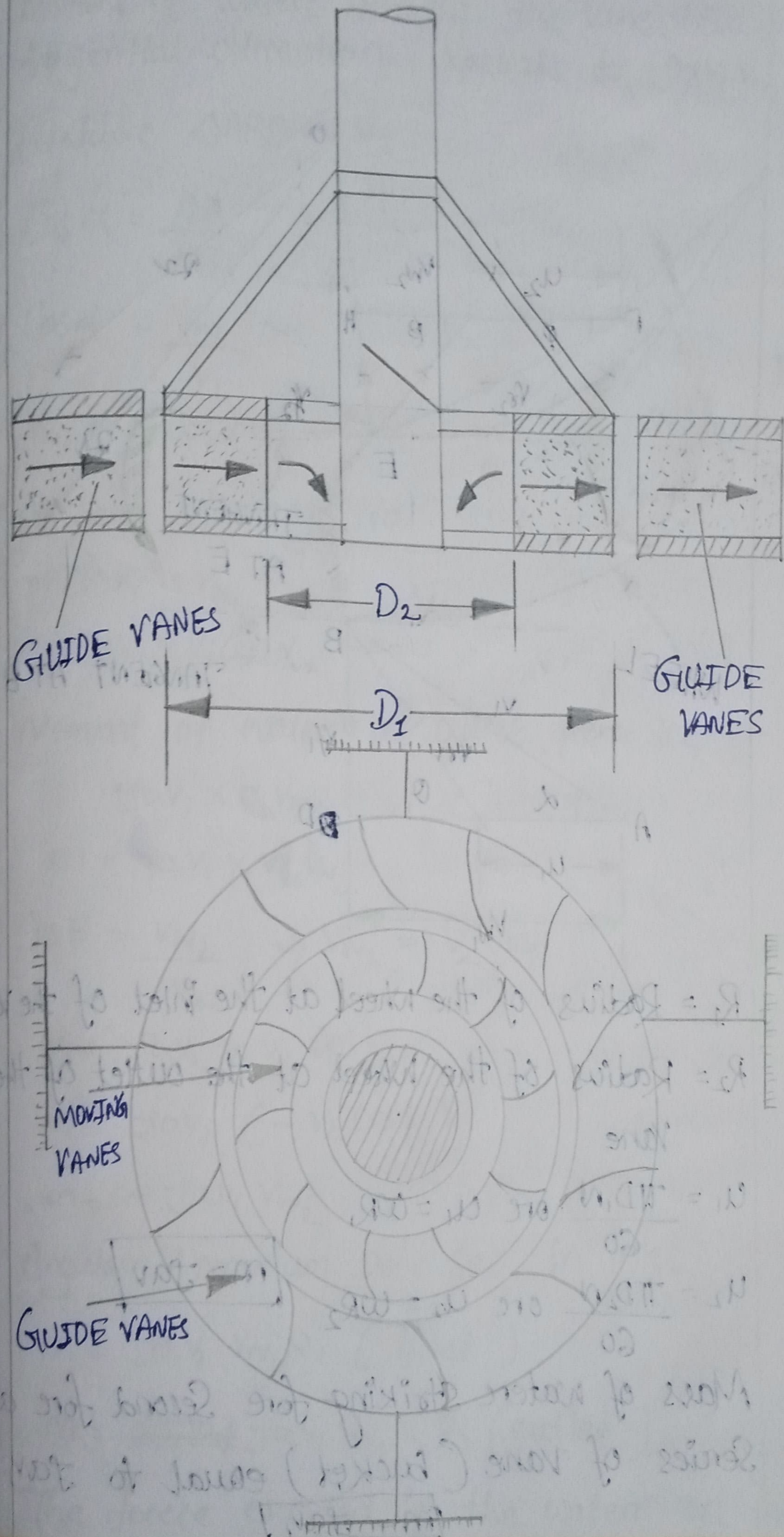
ii) Guide Mechanism   iv) Spiral

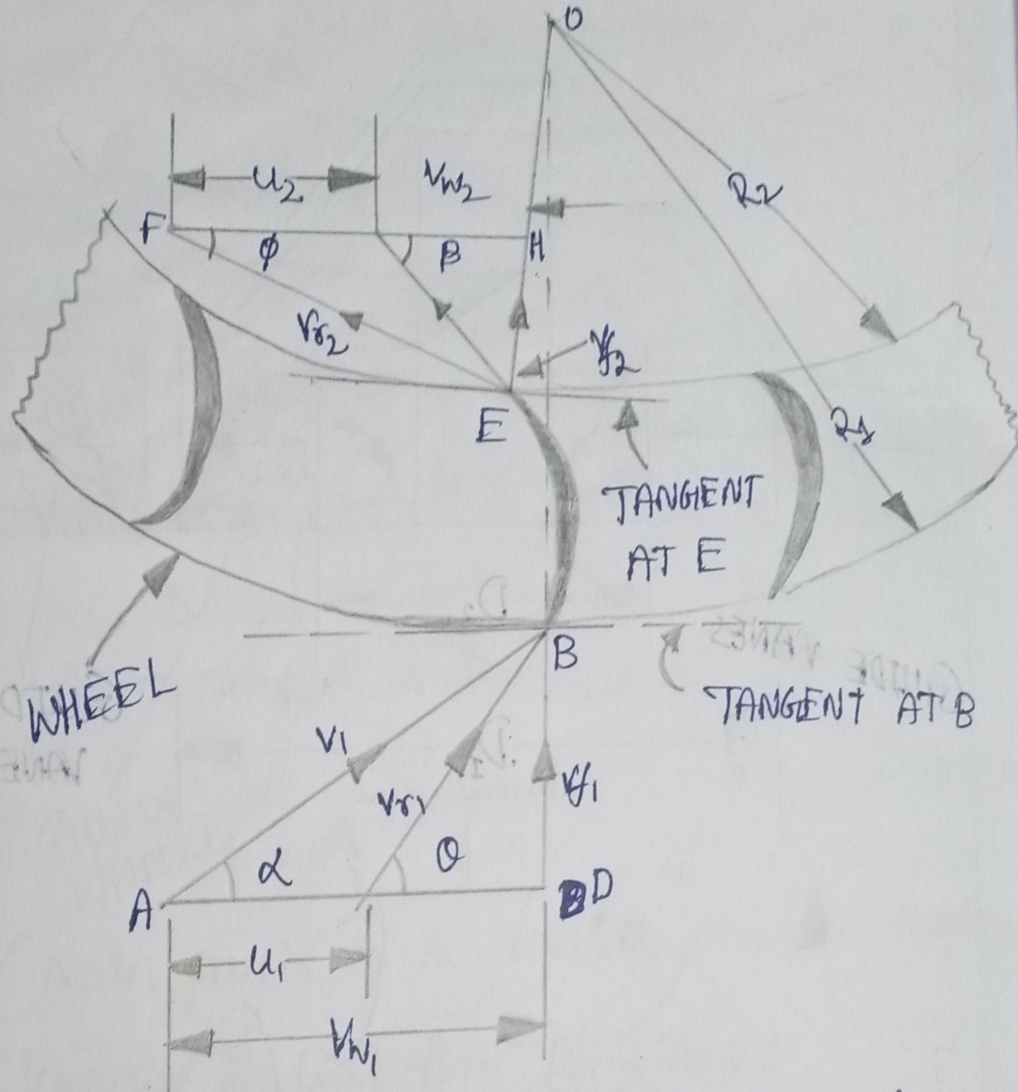



(Main parts of a radial reaction turbines)

*) Inward Radial flow Turbines :-

Inward Radial flow turbine, in which case the water from the casing enters the stationary guiding wheel. The guiding wheel consists of guide the moving vanes in the inward radial direction and is discharged at the inner diameter of the runner. The outer diameter of the runner is the inlet and inner diameter is the outlet.


GUIDE VANES
GUIDE VANES
$D_2$
$D_1$
MOVING VANES
GUIDE VANES




$R_1$ = Radius of the wheel at the inlet of the vane

$R_2$ = Radius of the wheel at the outlet of the vane

$$u_1 = \frac{\pi D_1 N}{60} \text{ ore } u_1 = \omega R_1$$

$$\boxed{m = \rho a v}$$

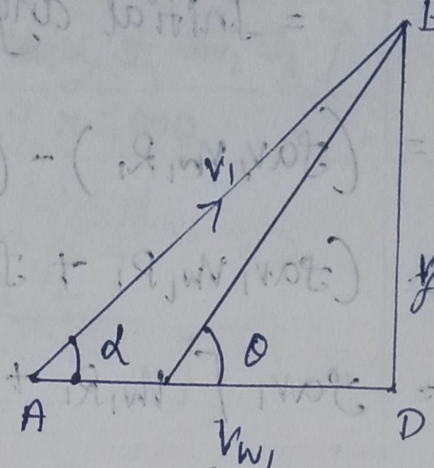
$$u_2 = \frac{\pi D_2 N}{60} \text{ ore } u_2 = \omega R_2$$

Mass of watere striking fore second fore a series of vane (bucket) equal to $\rho a v_1$

$$\boxed{m = \rho a v_1}$$

Moment of water striking the vane in tangential direction equals to $\rho a V_1 V_{w_1}$.

Consider $\triangle ABD = \frac{V_{w_1}}{V_1}$




$\Rightarrow \cos\alpha = \frac{AD}{AB}$

$\Rightarrow \cos\alpha = \frac{V_{w_1}}{V_1}$

$\Rightarrow V_{w_1} = V_1 \cos\alpha$

Moment of tangential

$= \rho a V_1 \times V_{w_1}$

$= \rho a V_1 \times V_1 \cos\alpha$

Moment of water at outlet per sec

$\rho a V_1 \times V_{w_2}$

$m = \rho a V_1 \times V_{w_2}$

$\cos\beta = \frac{V_{w_2}}{V_2} \Rightarrow V_{w_2} = V_2 \cos\beta$

$m = \rho a V_1 \times V_2 \cos\beta$

$m = \rho a V_1 (-V_2 \cos\beta)$

$m = -\rho a V_1 V_{w_2}$

Angular Momentum per sec in inlet

$m = \rho a V_1 V_{w_1} R_1$ (inlet)

$m = \rho a V_1 V_{w_2} \times R_2$ (outlet)

The force exerted by the water on the

Wheel

$T$ = Rate of change of Angular momentum

= Initial angular momentum - final angular momentum

$$= (\rho a v_1 V_{w_1} R_1) - (-\rho a v_2 V_{w_2} R_2)$$

$$= (\rho a v_1 V_{w_1} R_1 + \rho a v_1 V_{w_2} R_2)$$

$$= \rho a v_1 [V_{w_1} R_1 + V_{w_2} R_2]$$

Workdone per sec on the wheel

$$= T \cdot \omega$$

$$W/s = \rho a v_1 [V_{w_1} R_1 + V_{w_2} R_2] \omega$$

$$= \rho a v_1 [V_{w_1} R_1 \omega + V_{w_2} R_2 \omega]$$

$$= \rho a v_1 [V_{w_1} u_1 + V_{w_2} u_2]$$

$$\boxed{Q = AV}$$

$$W = \rho Q [V_{w_1} u_1 \pm V_{w_2} u_2]$$

Workdone per sec per unit weight of water

$$= \frac{W/s}{w/s} = \frac{V_{w_1} u_1 \pm V_{w_2} u_2}{g}$$

positive sign is taken if Angle is acute angle

Negative sign is taken if Angle is obtuse angle.

If $\beta$ is $90^\circ$ $\boxed{V_{w_2} = 0}$ $= \frac{V_{w_1} u_1}{g}$

hydraulic Efficiency :-

$$\eta_h = \frac{R.P.}{W.P.} = \frac{\frac{\rho a v_1 [V_{w_1} u_1 \pm V_{w_2} u_2]}{g \times 1000}}{\omega \times H}$$

$$\eta_h = \frac{\frac{\cancel{\rho a v_1} [V_{w_1} u_1 + V_{w_2} u_2]}{g \times \cancel{1000}}}{\frac{\cancel{\rho a v_1} \times H}{\cancel{1000}}}$$

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

If discharge is equally to radial $V_{w_2} = 0$

i) Speed ratio :-

The Speed ratio is defined as $= \frac{u_1}{\sqrt{2gH}}$

Where as,

$u_1$ = Tangential velocity of wheel at inlet.

ii) flow ratio :-

The ratio of the Velocity of flow at inlet ($V_{f_1}$) to the Velocity $\sqrt{2gH}$ is known as "flow ratio". $= V_{f_1} / \sqrt{2gH}$

Where as,

$H$ = Head of turbine

Discharge of the turbine :-

$Q = \pi D_1 B_1 \times V_{f_1}$ (inlet)

$= \pi D_2 B_2 \times V_{f_2}$ (outlet)

Where as

$D_1$ = Diameter of $R_1$

$B_1$ = Width of $R_1$

Radial discharge :-

If $\beta = 90°$, $V_{w_2} = 0$

If $\alpha = 90°$, $V_{w_1} = 0$

The angle made by absoulte velocity with the tangent on the wheel is 90° and the Component of the whirel velocity is zero.

Q4) An inward flow reaction turbine has external and internal diameters is 0.9m & 0.45m Respectively. The turbine is running 200 r.p.m. and width of turbine at inlet is 200 mm. The velocity of flow through the runner is Constant and is equal to 1.8 m/s. The Guide blades make an angle of 10° to the tangent of the wheel and the discharge at the outlet of the turbine is Radial. Draw the inlet and outlet velocity triangles and determine:

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

ii) The velocity of whirl at inlet.

iii) The relative velocity at inlet

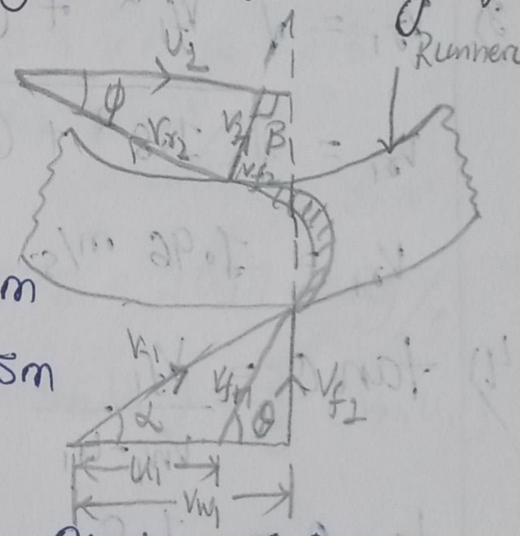
iv) The runner blade angles.

v) Width of the runner at outlet

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

vii) Head at inlet of the turbine.

viii) power developed and hydraulic efficiency of the turbine.




Given data :-

External dia, $D_1 = 0.9\,m$

Internal dia, $D_2 = 0.45m$

Speed, $N = 200$ r.p.m.

Width at inlet $(B_1) = 200mm = 0.2m$

velocity of flow ; $V_{f1} = V_{f2} = 7.8\,m/s$

Guide blade angle, $\alpha = 10°$

Discharge at outlet = Radial

$\therefore \beta = 90°$ and $V_{w2} = 0$

$$u_1 = \frac{\pi D_1 N}{60} = \frac{\pi \times 0.9 \times 200}{60} = 9.47\ m/s$$

$$u_2 = \pi D_2 N/60 = \pi \times 0.45 \times 200/60 = 4.71\,m/s$$

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

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

$$\Rightarrow V_1 = \frac{V_{f_1}}{\sin\alpha} = \frac{1.8}{\sin 10^\circ} = 10.36 \text{ m/s}$$

$$\Rightarrow \cos\alpha = \frac{V_{w_1}}{V_1}$$

$$\Rightarrow V_{w_1} = V_1 \cos\alpha$$

$$\Rightarrow V_{w_1} = 10.36 \times \cos 10^\circ$$

$$\rightarrow V_{w_1} = 10.20 \text{ m/s}$$

3) $$V_{r_1} = \sqrt{V_{f_1}^2 + (V_{w_1} - U_1)^2}$$

$$V_{r_1} = \sqrt{1.8^2 + (10.20 - 9.41)^2}$$

$$V_{r_1} = 1.96 \text{ m/s}$$

4) $$\tan\theta = \frac{V_{f_1}}{V_{w_1} - U_1}$$

$$\tan\theta = \frac{1.8}{10.20 - 9.41}$$

$$\tan\theta = 2.30$$

$$\theta = \tan^{-1} 2.30$$

$$\theta = 66.50^\circ$$

5) $\tan\phi = \dfrac{V_{f_2}}{U_2}$

$\tan\phi = \dfrac{1.8}{4.71}$

$\tan\phi = 0.38$

$\phi = \tan^{-1} 0.38 = 20.80^\circ$

6) $\pi D B_1 \cancel{V_{f_1}} = \pi D_2 B_2 \cancel{V_{f_1}}$

$\Rightarrow D_1 B_1 = D_2 B_2$

$\Rightarrow B_2 = \dfrac{D_1 B_1}{D_2} = \dfrac{0.9 \times 0.2}{0.45} = 0.4 \text{ mm} = 400\text{m}$

Discharge $(Q) = \pi D_1 B_1 V_{f_1}$

$= \pi \times 0.9 \times 0.2 \times 1.8$

$= 1.0178 \ m^3/s$

$\therefore$ Mass $= \rho \times Q$

$= 1000 \times 1.0178 = 1017.8 \text{ kg/s}$

$H - \dfrac{V_2^2}{2g} = \dfrac{1}{g}(V_{w_1} u_1 + V_{w_2} u_2) = \dfrac{1}{g}(V_{w_1} u_1)$

$\Rightarrow H = \dfrac{1}{g} V_{w_1} u_1 + \dfrac{V_2^2}{2g}$

$\Rightarrow H = \dfrac{1}{9.81} \times 10.20 \times 9.47 + \dfrac{1.8^2}{2 \times 9.81}$

$\Rightarrow H = 9.94\text{m}$

7) Power Developed $(P) = \dfrac{W}{1000}$

$= \rho Q (V_{w_1} u_1)/1000$

$$= \frac{1000 \times 1.0178 \,(10.20 \times 9.41)}{1000}$$

$= 97.69$ KW

hydraulic efficiency ($\eta_h$) :-

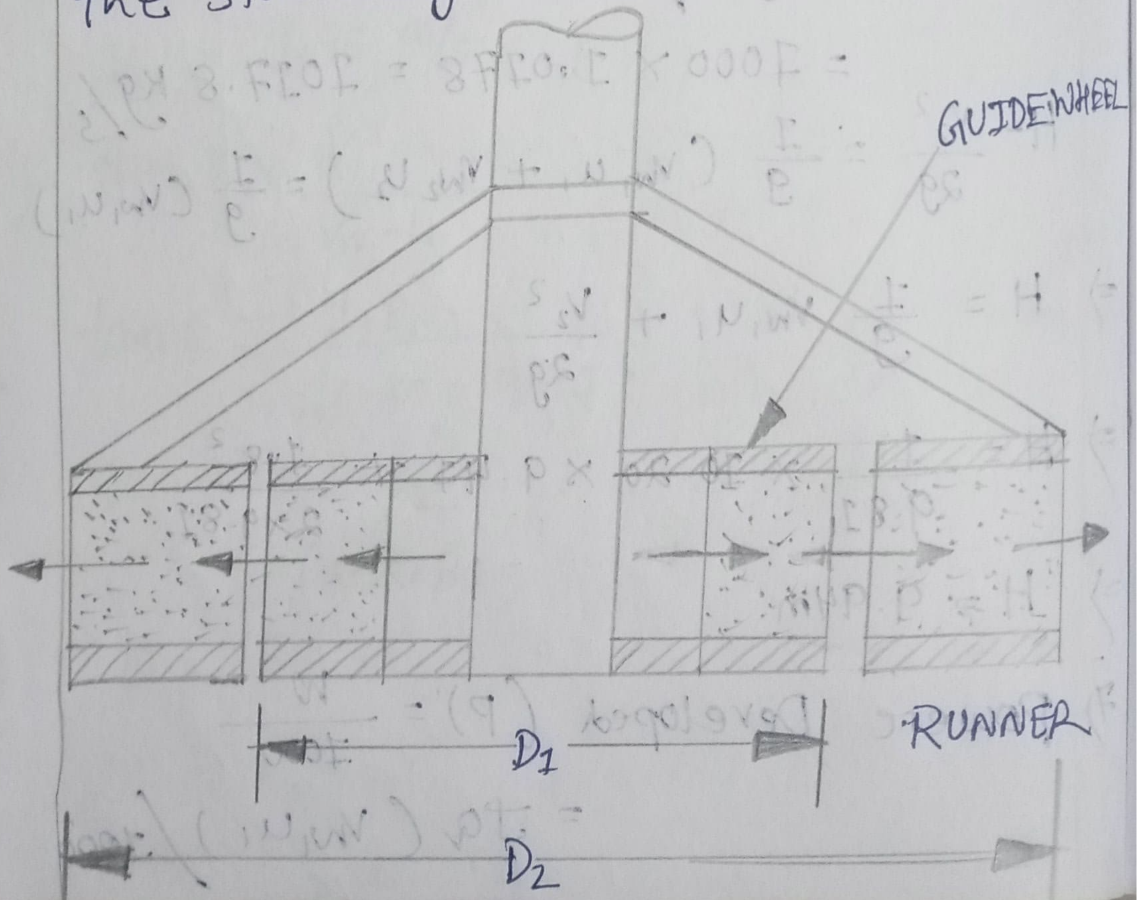
$$\eta_h = \frac{V_{w_1} u_1}{gH}$$

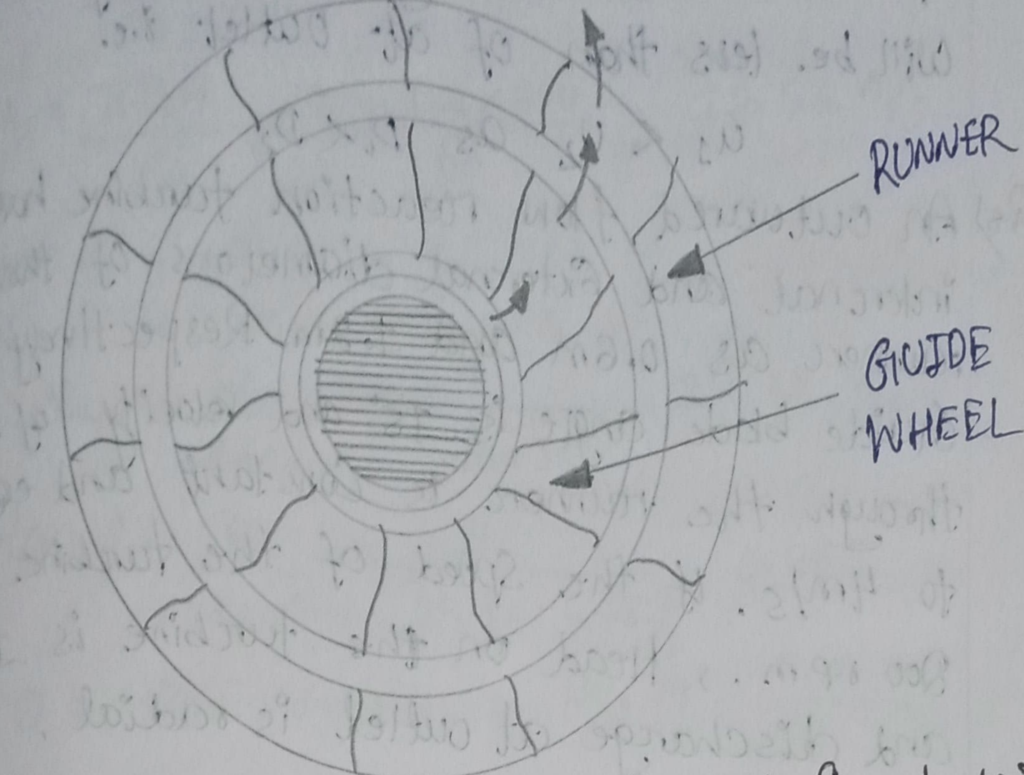
$$\eta_h = \frac{10.20 \times 9.41}{9.81 \times 9.94} = 0.9843$$

$= 98.43\%$

*) **Outward Radial flow Reaction Turbine :-**

Outward Radial flow Reaction turbine is in which the water from casing enters the stationary Guide wheel.






(Diagram of Outwared Radial flow turbine)

→ The Guide Wheel Consists of Guide wheel or Vanes which direct water to enter the runner which is around the Stationary Guide wheel.

→ The water flow through the Vanes of the runner in the Outward radial direction and is discharges at the outer diameter of the reunnere.

→ The inner diameter of the reunner is inlet and outer diameter is the outlet.

→ The Workdone by the water on the runner per second, the horse power developed and hydraulic efficiency will be obtained from the velocity triangles.

→ Inlet of the reunnere is at the inner diameter of the runner, the tangential velocity at inlet will be less than of at outlet i.e.

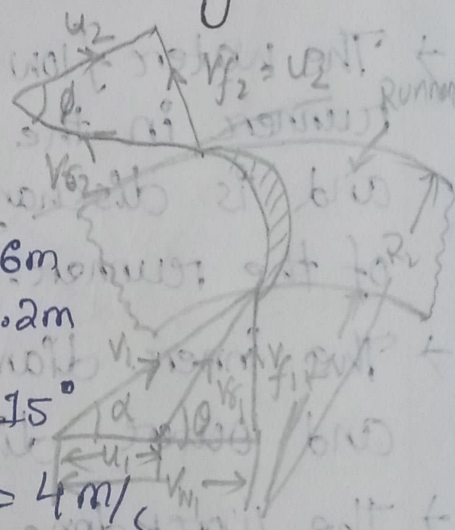
$u_1 < u_2$ as $D_1 < D_2$

Q5) An outward flow reaction turbine has internal and external diameters of the runner as 0.6m and 1.2m Respectively. The Guide blade angle is 15° and velocity of flow through the runner is constant and equals to 4m/s. If the speed of the turbine is 200 r.p.m., Head on the turbine is 10m and discharge at outlet is radial, determine:

i) The runner vane angles at inlet and outlet

ii) work done by water on the runner per sec per unit weight of water striking per sec.

iii) hydraulic efficiency.




Given data:-

Internal diameter, $D_1 = 0.6m$

External diameter, $D_2 = 1.2m$

Guide blade angle, $\alpha = 15°$

Velocity of flow, $V_{f_1} = V_{f_2} = 4 m/s$

Speed (N) = 200 r.p.m.

Head (H) = 10m

Discharge at outlet = radial

$V_{w_2} = 0$, $V_{f_2} = V_2$

$$u_1 = \frac{\pi D_1 N}{60} = \frac{\pi \times 0.6 \times 200}{60} = 6.28 \text{ m/s}$$

$$u_2 = \frac{\pi D_2 N}{60} = \frac{\pi \times 1.2 \times 200}{60} = 12.56 \text{ m/s}$$

$$\tan\alpha = \frac{V_{f_1}}{V_{w_1}}$$

$$\therefore V_{w_1} = \frac{V_{f_1}}{\tan\alpha} = \frac{4}{\tan 15^\circ} = 14.92 \text{ m/s}$$

$$\tan\theta = \frac{V_{f_1}}{V_{w_1} - u_1} = \frac{4}{14.92 - 6.28} = 0.462$$

$$\theta = \tan^{-1} 0.462 = 24.79^\circ$$

$$\tan\phi = \frac{V_{f_2}}{u_2} = \frac{4}{12.56} = 0.3183$$

$$\phi = \tan^{-1} 0.3183 = 17.65^\circ$$

ii) $$\frac{W}{W/s} = \frac{1}{g} V_{w_1} u_1$$

$$= \frac{1}{9.81} \times 14.92 \times 6.28 = 9.551 \text{ Nm/N}$$

iii) $$\eta_h = \frac{V_{w_1} u_1}{gH} = \frac{14.92 \times 6.28}{9.81 \times 10}$$

$$\eta_h = 0.955$$

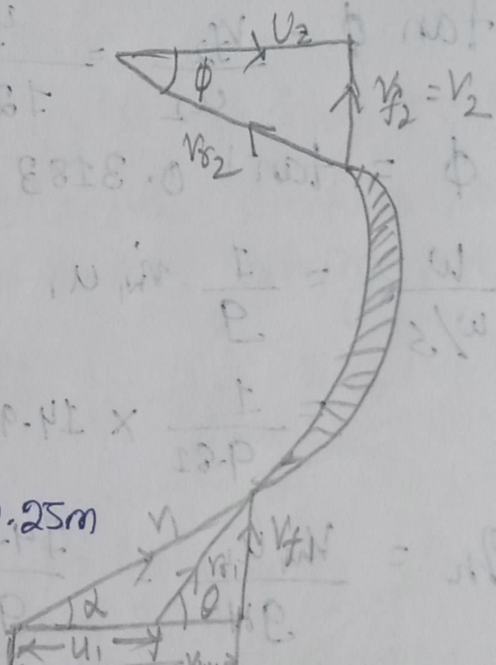
$$\eta_h = 95.5\ \%$$

Q6) The internal and external diameter of an outward flow reaction turbine are 2m and 2.75m respectively. the turbine is running at 250 r.p.m. and rate of flow of water through the turbine is $5 m^3/s$. The width of the runner is constant at inlet and outlet and is equals to 250mm. The Head on the turbine is 150m. Neglecting thickness of the vanes and taking discharge Radial at outlet determine :-

i) Vane angles at inlet and outlet

ii) Velocity of flow at inlet and outlet

<u>Given data:-</u>




$d_1 = 2m$

$d_2 = 2.75m$

$N = 250$ r.p.m.

$Q = 5 m^3/s$

$B_1 = B_2 = 250mm = 0.25m$

$H = 150m$

Discharge at outlet = radial

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

$$u_1 = \frac{\pi D_1 N}{60} = \frac{\pi \times 2 \times 250}{60} = 26.17 \ m/s$$

$$u_2 = \frac{\pi D_2 N}{60} = \frac{\pi \times 2.75 \times 250}{60} = 35.99 \ m/s$$

$Q = \pi D_1 B_1 V_{f_1} = \pi D_2 B_2 V_{f_2}$

$\therefore V_{f_1} = \frac{Q}{\pi D_1 B_1} = \frac{5}{\pi \times 2 \times 0.25} = 3.183 \text{ m/s}$

$V_{f_2} = \frac{Q}{\pi D_2 B_2} = \frac{5}{\pi \times 2.75 \times 0.25} = 2.134 \text{ m/s}$

$H - \frac{V_2^2}{2g} = \frac{V_{w_1} u_1}{g}$ $\boxed{V_2 = V_{f_2}}$

$\therefore 150 - \frac{2.315^2}{2 \times 9.81} = \frac{V_{w_1} \times 26.17}{9.81}$

or $149.73 = \frac{V_{w_1} \times 26.17}{9.81}$

$\therefore V_{w_1} = \frac{149.73 \times 9.81}{26.17} = 56.12 \text{ m/s}$

i) $\tan\theta = \frac{V_{f_1}}{V_{w_1} - u_1} = \frac{3.18}{56.12 - 26.18} = 0.1062$

$\theta = \tan^{-1} 0.1062$

$= 6.06°$

$\tan\phi = \frac{V_{f_2}}{u_2} = \frac{2.134}{35.99} = 0.059$

$\phi = \tan^{-1} 0.059 = 3.37°$

ii) $V_{f_1} = 3.183$ m/s and $V_{f_2}$ 2.315 m/s

*) Francis turbine :-

The inward flow reaction turbine having radial discharge at outlet is known as

"Francis Turbine": The turbine name of "J. B. Francis".

$$\boxed{V_{w_2} = 0}$$

Workdone per water on the runner per sec

$$\frac{W}{w/s} = \rho Q (V_{w_1} u_1)$$

Workdone per sec per unit weight of water striking/sec = $\boxed{\frac{V_{w_1} u_1}{g}}$

hydraulic efficiency ($\eta_h$) =

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

Importants Relations for francis turbine:-

1) The ratio of width of the wheel to its diameter is

$$N = \frac{B_1}{D_1}$$

The value of N varies from 0.10 to 0.4.

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

and varies from 0.15 to 0.30

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

## Axial flow Reaction Turbine :-

→ If the water flows parallel to the axis of the rotation of the shaft, the turbine is known as "Axial flow reaction turbine".

→ The axial flow reaction turbine, the shaft of the turbine is vertical. The lower end of the shaft is made larger which is known as "hub" or "boss".

→ The vanes are fixed on the hub and hub acts as a runner for axial flow reaction turbine.

→ Axial flow are 2 types
i) Propeller turbine, ii) Kaplan turbine

### Propeller turbine :-

When the vanes are fixed to the hub and they are not adjustable, the turbine is known as "Propeller turbine".

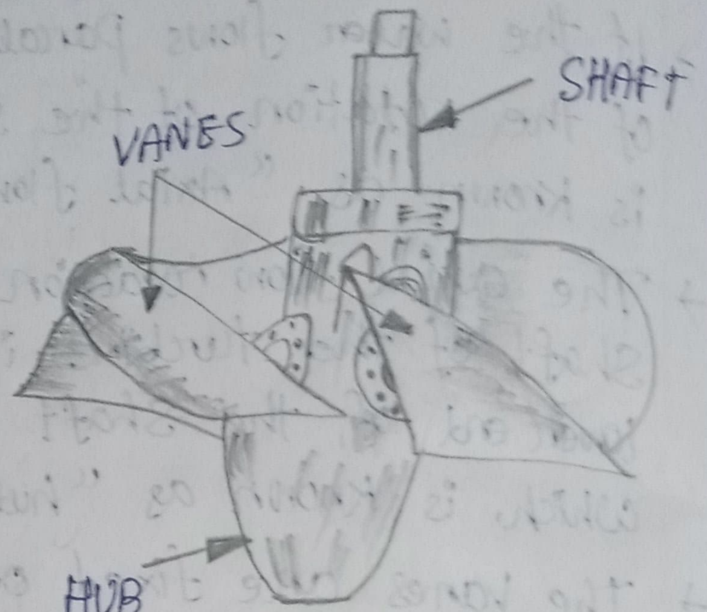
### Kaplan turbine :-

If the vanes on the hub are adjustable the turbine is known as "Kaplan turbine". the turbine of Kaplan turbine is the name of V. Kaplan.

Main parts of a kaplan turbines :-
i) Scroll Casing ii) Guide Vanes Mechanism

iii) hub with vanes or runner of the turbine

iv) Draft tube




(Kaplan turbine runner)

Main parts of a Kaplan turbine. The water from Penstock, enters the Scroll Casing and then moves to the Guide Vanes. from the Guide Vanes, the water turns through 90° and flows axially through the runner the discharge through the runner, is

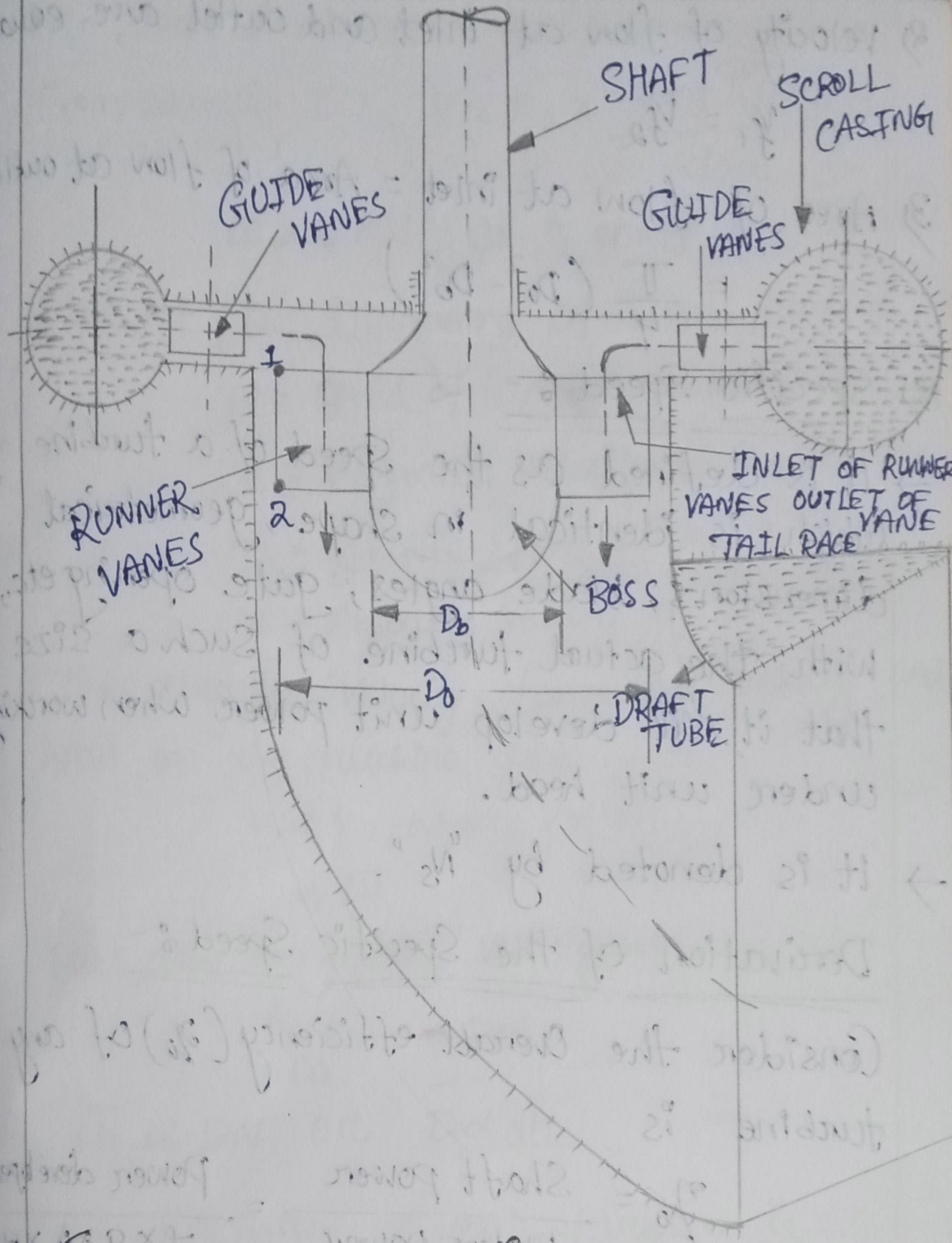
$$Q = \frac{\pi}{4}\left(D_o^2 - D_b^2\right) \times V_{f_1}$$

Where as,

$D_o$ = Outer diameter of the runner,

$D_b$ = Diameter of hub

$V_{f_1}$ = Velocity of flow at inlet.




(Main parts or Components of kaplan turbine)

Some important point for propeller (kaplan turbine).

1) The peripheral velocity at inlet and outlet are equal

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

2) Velocity of flow at inlet and outlet are equal

$V_{f1} = V_{f2}$

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

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

*) <u>Specific Speed</u> :-

→ It is defined as the speed of a turbine which is identical in shape, geometrical dimensions, blade angles, gate opening etc., with the actual turbine of such a size that it will develop unit power when working under unit head.

→ It is denoted by "$N_s$".

<u>Derivation of the Specific Speed</u> :-

Consider the overall efficiency ($\eta_o$) of any turbine is

$$\eta_o = \frac{\text{Shaft power}}{\text{Water power}} = \frac{\text{Power developed}}{\frac{\rho \times g \times Q \times H}{1000}}$$

$$= \frac{P}{\frac{\rho \times g \times Q \times H}{1000}} \quad \text{—(i)}$$

Where as, H = Head under which the turbine is working

Q = Discharge through turbine

P = Powere developed ore shaft powere

From equation (i), $P = \eta_0 \times \frac{\rho \times g \times Q \times H}{1000}$

$\propto Q \times H$ (as $\eta_0$ and $\rho$ are Constant)

Now let D = Diameter of actual turbine

N = Speed of actual turbine

u = Tangential velocity of the turbine

$N_s$ = Specific Speed of the turbine

V = Absoulte velocity of water,

The absoulte velocity, tangential velocity and head on the turbine are

$u \propto V$, where $V \propto \sqrt{H}$

$\propto \sqrt{H}$

But the tangential velocity u is

$u = \frac{\pi D N}{60} \quad \propto DN$

$\sqrt{H} \propto DN$ ore $D \propto \frac{\sqrt{H}}{N}$

The discharge through turbine is

Q = Area x Velocity

But Area $\propto B \times D \propto D^2$

and velocity $\propto \sqrt{H}$

$\therefore Q \propto D^2 \times \sqrt{H}$

$\propto \left(\frac{\sqrt{H}}{N}\right)^2 \times \sqrt{H} = \propto \frac{H}{N^2} \times \sqrt{H}$

$= \dfrac{H^{3/2}}{N^2}$

Now the value of Q in eqn (ii)

$$P \propto \frac{H^{3/2}}{N^2} \times H \propto \frac{H^{5/2}}{N^2}$$

$$\boxed{P = K \frac{H^{5/2}}{N^2}}$$

Where K = Constant of proportionality

If P = 1, H = 1, the Speed N = Specific Speed $N_s$,

Substituting these values in the above equation,

$$1 = \frac{K \times 1^{5/2}}{N_s^2} \quad \text{or} \quad N_s^2 = K$$

$$P = N_s^2 \frac{H^{5/2}}{N^2} \quad \text{or} \quad N_s^2 = \frac{N^2 P}{H^{5/2}}$$

$$\therefore N_s = \sqrt{\frac{N^2 P}{H^{5/2}}} = \frac{N\sqrt{P}}{H^{5/4}}$$

| S. No | Specific Speed (M.K.S) | (S.I) | Types of turbine |
|---|---|---|---|
| 1 | 10 to 35 | 8.5 to 30 | Pelton wheel with single jet |
| 2 | 35 to 60 | 30 to 51 | Pelton wheel with two or more jets |
| 3 | 60 to 300 | 51 to 225 | Francis turbine |
| 4 | 300 to 1000 | 255 to 860 | Kaplan or propeller turbine |

Example of Axial flow turbine

Q8) A kaplan turbine working under a head of 20m develops 11772 kw shaft power. the outer diameter of the runner is 3.5m and hub diameter 1.75m. The Guide blade angle at the extreme edge of the runner is 35°. The hydraulic and overall efficiencies of the turbines are 88% and 84% respectively. If the velocity of whirl is zero at outlet, determine :

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

ii) Speed of the turbine.

Given data :-

Head, $H = 20m$

Shaft power, $S.P. = 11772$ kw

outer dia of runner, $D_0 = 3.5m$

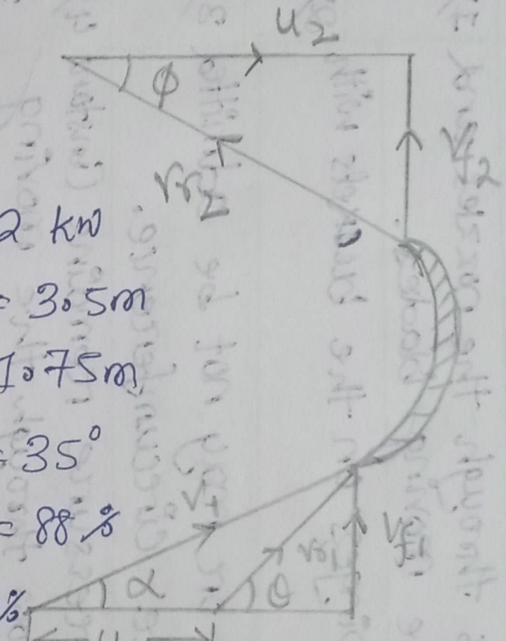
hub dia, $D_b = 1.75m$

Guide blade angle, $\alpha = 35°$

hydraulic efficiency, $\eta_h = 88\%$

Overall efficiency, $\eta_0 = 84\%$

Velocity of whirl at outlet $= 0$.




$$\eta_0 = \frac{S.P.}{W.P.}$$

$$W.P. = \frac{1 \times g \times Q \times H}{1000}$$

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

$$0.84 = \frac{11772 \times 1000}{1000 \times 9.81 \times Q \times 20}$$

$$\Rightarrow Q = \frac{11772 \times 1000}{0.84 \times 1000 \times 9.81 \times 20}$$

$$Q = 71.428 \ m^3/s$$

$$Q = \pi/4 \left( D_o^2 - D_b^2 \right) \times V_{f1}$$

$$71.428 = \pi/4 \left( 3.5^2 - 1.75^2 \right) \times V_{f1}$$

$$71.428 = \pi/4 \left( 12.25 - 3.062 \right) V_{f1}$$

$$71.428 = 7.216 \ V_{f1}$$

$$\Rightarrow V_{f1} = \frac{71.428}{7.216} = 9.89 \ m/s$$

from inlet velocity triangle,

$$\tan\alpha = \frac{V_{f2}}{V_{w_1}}$$

$$\therefore V_{w_1} = \frac{V_{f2}}{\tan\alpha} = \frac{9.89}{\tan 35^\circ} = 14.12 \ m/s$$

using the relation for hydraulic efficiency,

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

$$0.88 = \frac{14.12 \times u_1}{9.81 \times 20}$$

$$\Rightarrow u_1 = \frac{0.88 \times 9.81 \times 20}{14.12} = 12.22 \text{ m/s}$$

$$\tan\theta = \frac{V_{f_1}}{V_{w_1} - u_1} = \frac{9.89}{(14.12 - 12.22)} = 5.20$$

$$\theta = \tan^{-1} 5.20$$

$$\theta = 79.71^\circ$$

$u_1 = u_2 = 12.22$ m/s and $V_{f_1} = V_{f_2} = 9.89$ m/s

$$\tan\phi = \frac{V_{f_2}}{u_2} = \frac{9.89}{12.22} = 0.8093$$

$$\phi = \tan^{-1} 0.8093 = 38.98^\circ$$

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

$$12.22 = \frac{\pi \times 3.5 \times N}{60}$$

$$\Rightarrow N = \frac{60 \times 12.22}{\pi \times 3.5} = 66.68 \text{ r.p.m.}$$

(Ans)

(12) A turbine is operate under a head of 25m at 200 r.p.m. the discharge is $9 m^3/s$. If the efficiency of overall or overall efficiency is 90°;
determine -
i) Specific speed of the machine,
ii) power generated,
iii) Type of turbine;

Given data :-

Head (H) = 25 m
Speed, (N) = 200 r.p.m.
discharge (Q) = $9 m^3/s$
$\eta_o = 90\% = 0.90$

$$P = \frac{\eta_o \times \rho g Q H}{1000} = 0.90 \times \frac{1000 \times 9.81 \times 9 \times 25}{1000}$$

$$= 1986.72 \text{ kW}$$

$$N_s = \sqrt{\frac{P \times N^2}{H^{5/2}}} = \sqrt{\frac{1986.72 \times 200^2}{(25)^{5/2}}} = 159.46 \text{ r.p.m.}$$

It is a francis flow turbine

# Difference between Impulse & Reaction turbine

| S.NO | Impulse Turbine | Reaction Turbine |
|---|---|---|
| 1) | The water flows through the nozzles and impinges on the moving blades. | 1) The water flows first through Guide Mechanism and then through the moving blades. |
| 2) | The water impinges on the buckets with kinetic Energy. | 2) The water glides over the moving vanes with pressure and kinetic Energy. |
| 3) | The water may or may not be admitted over the whole circumference. | 3) The water must be admitted over the whole circumference. |
| 4) | The water pressure remains Constant during its flow through the moving blades. | 4) The water is reduced during its flow through the moving blades. |
| 5) | The relative velocity of water while gliding over the blades remains Constant. | 5) The relative velocity of water while gliding over the blade increases. |
| 6) | The blades are Symmetrical. | 6) The blades are not Symmetrical |
| 7) | The number of stages required are less for the Same power developed. | 7) The number of stages required are more of the Same power developed. |

# Chapterc-2 Centrifugal pumps

## Introduction :-

→ It Converet Mechanical Energy into hydraulic Energy arce called "pumps".

→ These arce classified as 2 types

i) Reciprocating pump

ii) Centrifugal pump

→ If the Mechanical Energy is Converted, into preessurce Energy by means of Centrifugal forcee acting on the fluid, the hydraulic machine is called "Centrifugal forcee."

→ It act as a reeversed of an inwared radial flow reaction turbine.

## *) Main parts of a Centrifugal pump :-

The following arce the main parts of a Centrifugal pump arce

i) Impellere

ii) Casing

iii) Suction pipe with a foot valve and a straineer

iv) Deliverey pipe

### Impellere :-

→ The rotating parct of a Centrifugal pump is called "Impellere".

→ It consists of a series of a backward curved vanes.

→ The Impeller is mounted on a shaft which is connected to the shaft of an electric motor.

Casing :-

→ The casing of a centrifugal pump is similar to the casing of a reaction turbine.

→ It is an air-tight passage surrounding the impeller and is designed in such a way that the kinetic energy of the water discharged at the outlet of the impeller is converted into pressure energy before the water leaves the casing and enters the delivery pipe.

→ Casing are classified as 3 types

i) Volute casing

ii) Vortex casing

iii) Casing with guide blades

Volute Casing :-

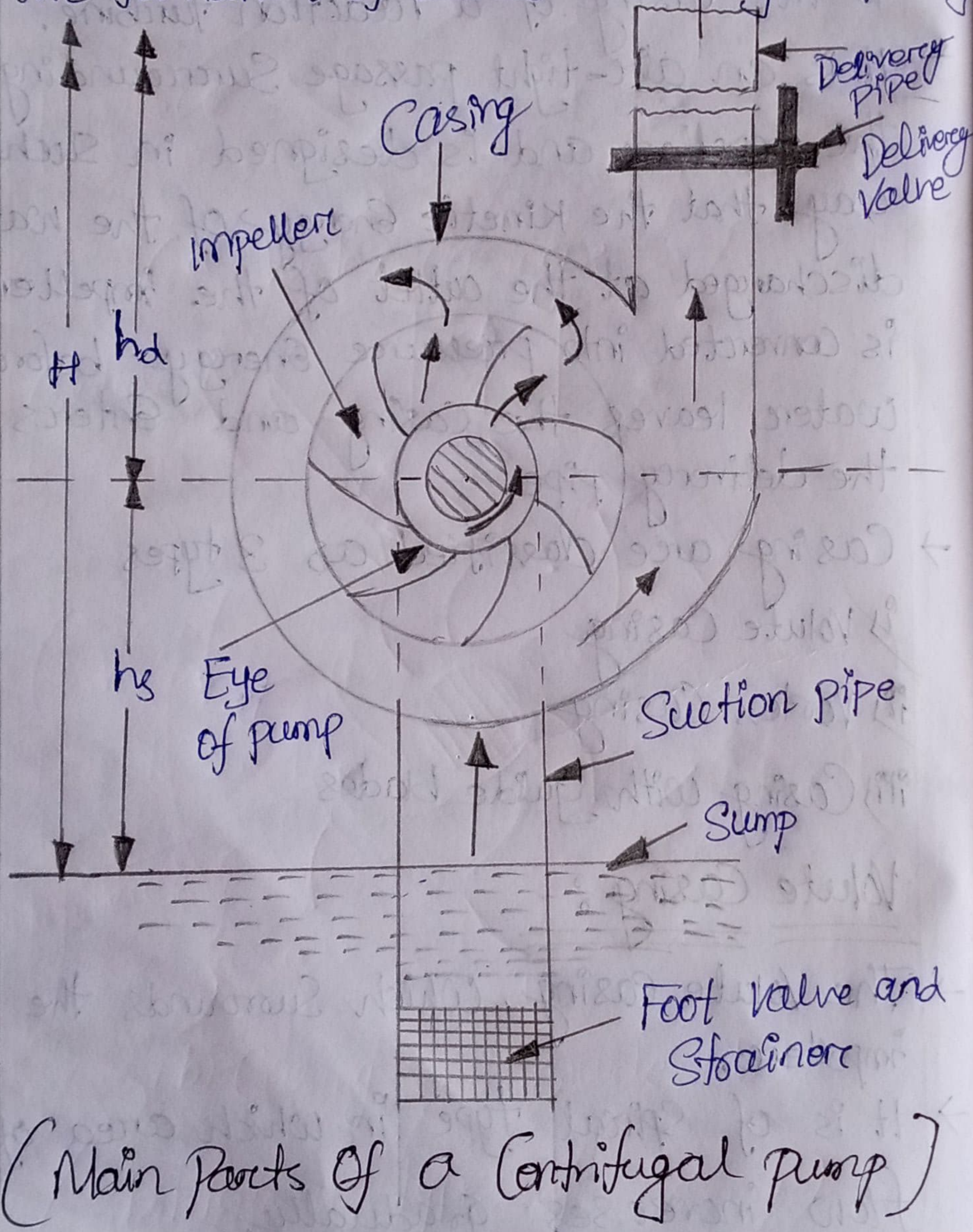
→ The volute casing, which surrounds the impeller.

→ It is of spiral type in which area of flow increases gradually.

→ The increase in area of flow decreases the velocity of flow.

→ The decrease in velocity increases the pressure of the water flowing through the casing.

→ It has been observed that in case of volute casing, the efficiency of pump increases slightly as a large amount of energy is lost due to the formation of eddies in this type of casing.




(Main Parts Of a Centrifugal pump)

## Vortex Casing :-

If a circulare chambere is introduced between the casing and the impellere, the casing is known as "vortex casing".

→ By introducing the circulare chambere, the loss of energy due to the formation of eddies is reduced to a considerable extent.

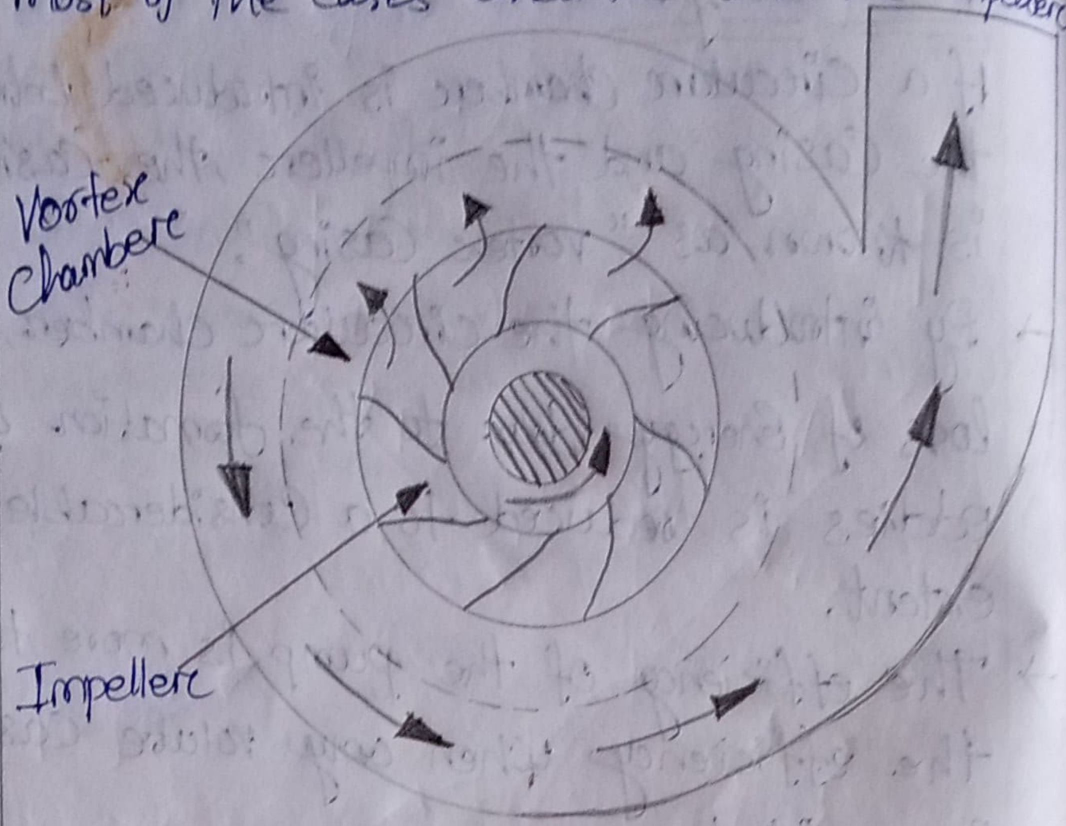
→ The efficiency of the pump is more than the efficiency when only volute casing is provided.

## Casing with Guide Blades :-

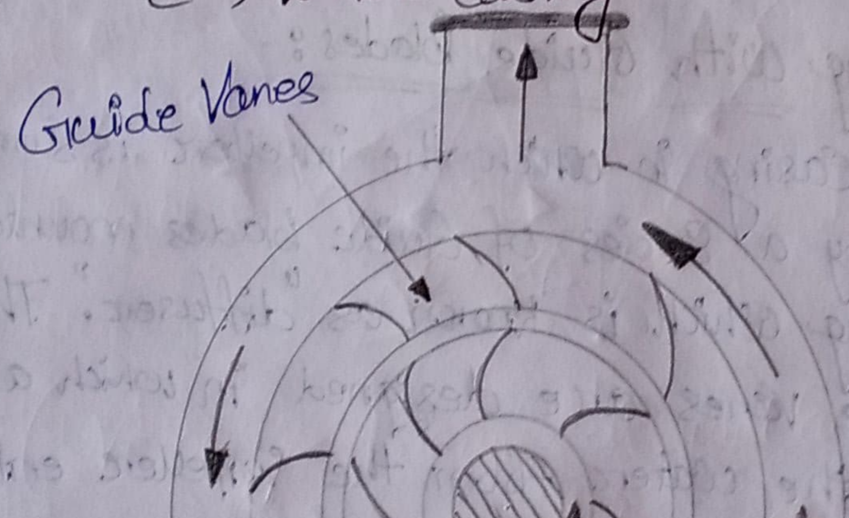
→ This casing in which the impellere is surrounded by a series of guide blades mounted on a reing which is known as "diffusere." The guide vanes are designed in which a way that the watere from the impellere enters the guide vanes without stock.

→ Also the area of the guide vanes increases thus reducing the velocity of flow through guide vanes and consequently increasing the pressure of watere.

→ The watere from the guide vanes then passes through the surrounding casing which is most of the cases concentric with the impeller.




(a) Vortex Casing




(b) Casing with Guide Blades

( Different types of Casing )

3) <u>Suction Pipe with a foot - Valve and a Strainere</u> :-

→ A pipe whose one end is Connected to the inlet of the pump and other end dips into water in a sump is known as "Suction pipe".

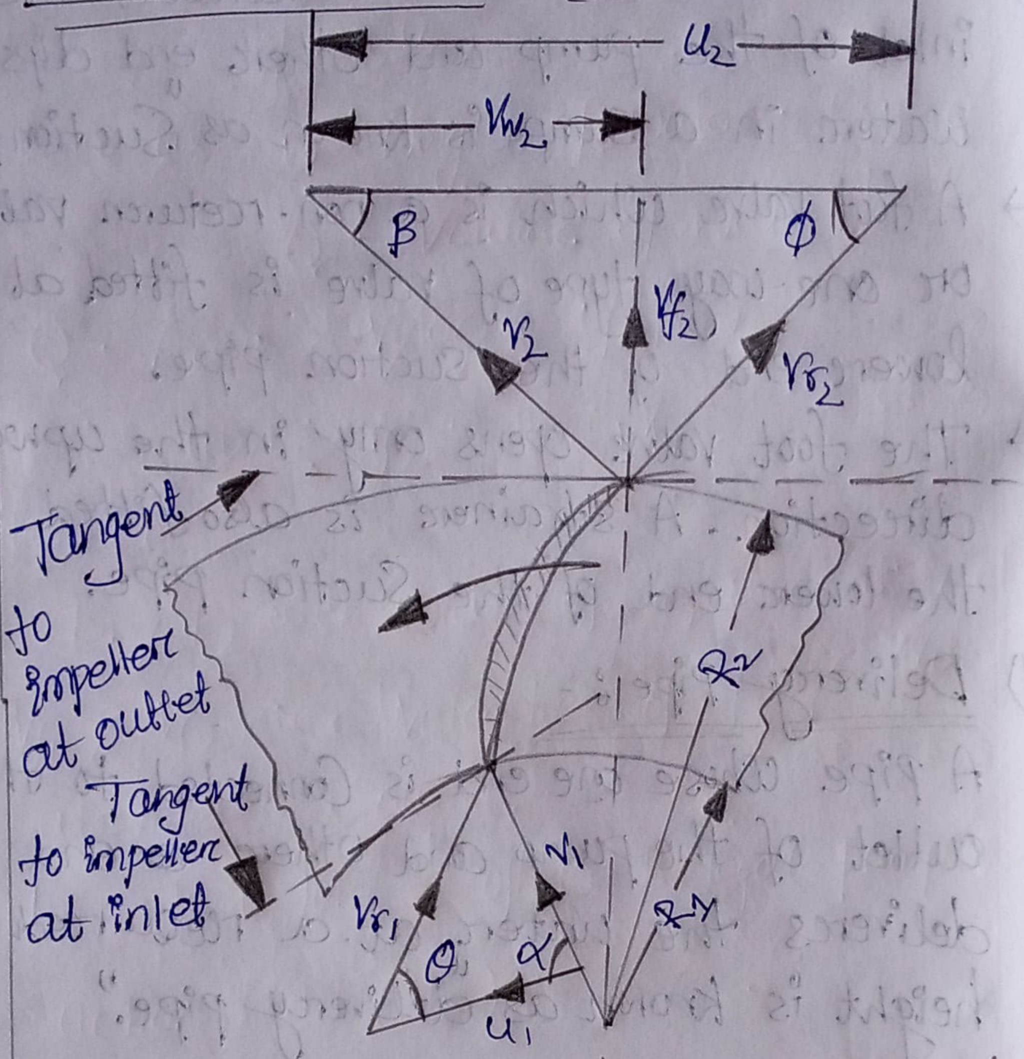
→ A foot valve which is a non-return valve or one-way type of valve is fitted at the lower end of the suction pipe.

→ The foot valve opens only in the upward direction. A strainer is also fitted at the lower end of the Suction pipe.

4) <u>Delivery Pipe</u> :-

A pipe whose one end is Connected to the outlet of the pump and other end delivers the water at a required height is known as "delivery pipe."

## *) Workdone by the Centrifugal pump (or by impeller) on water




(Velocity triangles at inlet and outlet)

Due to the flow is radial the water enter the Impeller radially at inlet for best efficiency of pump. The absoulte velocity of water at inlet make an angle of $90°$ with the direction of motion of Impeller at inlet

$$\alpha = 90°, \; V_{w_1} = 0$$

Wherce as,

$N$ = Speed of the Impellere in r.p.m.,

$D_1$ = Diametere of Impellere at inlet,

$u_1$ = Tangential velocity of impellere at inlet,

$= \pi D_1 N/60$

$D_2$ = Diametere of Impellere at outlet

$u_2$ = Tangential velocity of Impellere at outlet

$= \pi D_2 N/60$

$V_1$ = Absoulte velocity of watere at inlet,

$V_{r_1}$ = Relative velocity of watere at inlet,

$\alpha$ = Angle made by Absoulte velocity ($V_1$) at inlet with the direction of motion of vane,

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

$V_2$, $V_{r_2}$, $\beta$ and $\phi$ are the Correesponding values at outlet.

The work done by the watere on the reunnere pere sec pere unit weight of the watere striking pere sec is equals to

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

Work done by the impellere on the water pere second pere unit weight of water striking pere second.

$= -\left[\frac{1}{g}\left(V_{w_1} u_1 \pm V_{w_2} U_2\right)\right]$

- Work done by impeller on water per second

$$= \frac{W}{g} \cdot V_{w_2} U_2$$

Where as, $W$ = weight of water = $\rho \times g \times Q$

$Q$ = Volume of water

$Q$ = Area × velocity of flow

$$= \pi D_2 B_2 \times V_{f_2} = \pi D_1 B_1 V_{f_1}$$

Where

$B_1$ & $B_2$ are width of impeller at inlet & outlet

$V_{f_1}$ & $V_{f_2}$ are velocities of flow at inlet and outlet.

**Important terms of centrifugal pump :-**

1) **Suction head ($h_s$) :-**

→ It is the vertical height of the centre line of the centrifugal pump above the water surface in the tank, or pump from which water is to be lifted.

→ This height is also called a "suction lift".

→ It is denoted by "$h_s$".

2) **Delivery Head ($h_d$) :-**

→ The vertical distance between the centre line of the pump and the water surface in the tank to which water is delivered is known as "delivery head."

→ It is denoted by "$h_d$".

3) **Static head ($H_s$):-**

→ The sum of suction head and delivery head is known as "static head".

→ It is represented by "$h_s$".

$$\boxed{H_s = h_s + h_d}$$

4) **Manometric head ($H_m$):-**

→ The manometric head is defined as the head against which a centrifugal pump has to work.

→ It is denoted by "$h_m$".

a) $H_m$ = Head imparted by the impeller to the water - Loss of head in the pump

$= \frac{V_{w_2} U_2}{g}$ - Loss of head in impeller and casing

$= \frac{V_{w_2} U_2}{g}$ ..... if loss of pump is zero

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

$$= \left(\frac{P_0}{\rho g} + \frac{V_0^2}{2g} + Z_0\right) - \left(\frac{P_i}{\rho g} + \frac{V_i^2}{2g} + Z_i\right)$$

Wherere as

$P_o$ = preressure head at outlet of the pump

$= h_d$

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

= Velocity head in delivery pipe

$= V_d^2/2g$

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

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

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

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

Wherere, $h_s$ = Suction head,

$h_d$ = Delivery head,

$h_{fs}$ = frictional head loss in suction pipe,

$h_{fd}$ = frictional head loss in delivery pipe, and

$V_d$ = Velocity of waterer in delivery pipe.

## *) Efficiencies of a Centrifugal pump :-

### a) Manometric Efficiency :- ($\eta_{mm}$)

The ratio of the manometric head to the head imparted by the impeller to the water is known as "Manometric Efficiency."

→ It is denoted by "$\eta_{mm}$".

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

$$= \frac{H_m}{\left(\frac{V_{w_2} U_2}{g}\right)}$$

$$\boxed{\eta_{mm} = \frac{g H_m}{V_{w_2} U_2}}$$

The power Given to water at outlet of the pump

$$= \frac{W H_m}{1000} \text{ KW}$$

The power at the impeller

$$= \frac{\text{Workdone by impeller per second}}{1000} \text{ KW}$$

$$= \frac{W}{g} \times \frac{V_{w_2} U_2}{1000} \text{ KW}$$

$$\eta_{man} = \frac{\frac{W \times H_m}{1000}}{\frac{W}{g} \times \frac{V_{w_2} U_2}{1000}} = \frac{g \times H_m}{V_{w_2} U_2}$$

## b) Mechanical Efficiency :- ($\eta_m$) :-

→ The power at the shaft of the centrifugal pump is more than the power available at the impeller of the pump. The ratio of the power available at the impeller to the power at the shaft of the centrifugal pump is known as "Mechanical Efficiency."

→ It is denoted by "$\eta_m$".

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

The power at the impeller in kW

$$= \frac{\text{Workdone by impeller per second}}{1000}$$

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

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

Where, S.P. = Shaft power

## c) Overall Efficiency ($\eta_o$) :-

It is defined as ratio of power output of the pump to the power input to the pump.

→ The power output of the pump in KW
→ It is denoted by "$\eta_o$".

$$\boxed{\eta_o = \frac{\text{Weight of water lifted} \times H_m}{1000} \qquad \eta_o = \frac{W H_m}{1000}}$$

Power input to the pump

= power supplied by the electric motor

= S.P. of the pump

$$\therefore \boxed{\eta_o = \frac{\left(\frac{W H_m}{1000}\right)}{S.P.}}$$

also $\boxed{\eta_o = \eta_{man} \times \eta_m}$

Q1) The internal and External diameters of the impeller of a Centrifugal pump are 200mm and 400mm respectively. The pump is running at 1200 r.p.m. The vanes angles of the impeller at inlet and outlet are 20° and 30° respectively.

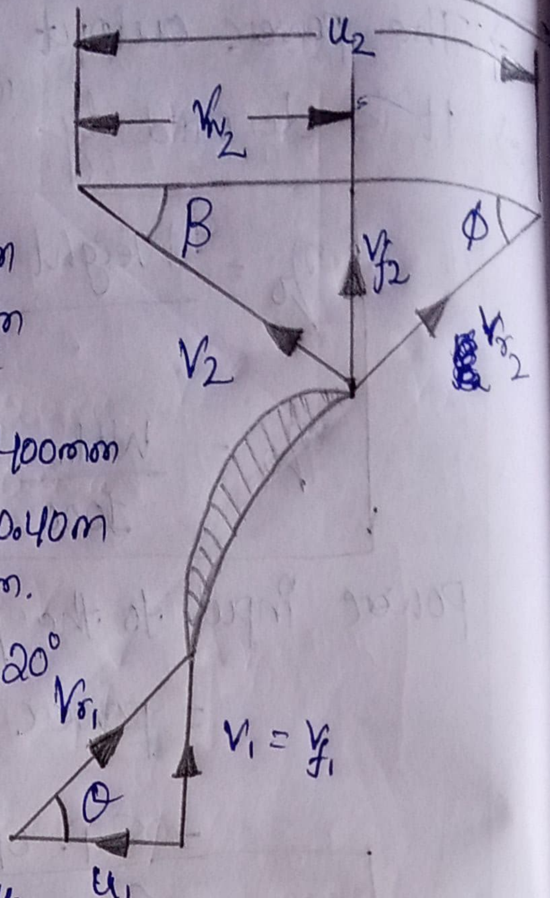
The water enters the impeller radially and Velocity of flow is Constant.

Determine the workdone by the impeller per unit weight of water.

Given data :-

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

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




Speed (N) = 1200 r.p.m.

Vane angle at inlet, $\theta = 20°$

Vane angle at outlet, $\phi = 30°$

Water enters radially

Means, $\alpha = 90°$ and $V_{w_1} = 0$

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

Tangential velocity of impeller at inlet and outlet are

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

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

From velocity outlet triangle,

$$\tan\theta = \frac{V_{f_1}}{u_1}$$

$$\tan\theta = V_{f_1} / 12.56$$

$\therefore V_{f_1} = 12.56 \tan\theta$

$V_{f_1} = 12.56 \tan 20^\circ$

$V_{f_1} = 4.57 \text{ m/s}$

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

From outlet velocity triangle,

$$\tan\phi = \frac{V_{f_2}}{u_2 - V_{w_2}} = \frac{4.57}{25.13 - V_{w_2}}$$

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

$$\Rightarrow 25.13 - V_{w_2} = \frac{4.57}{\tan 30^\circ}$$

$$\Rightarrow 25.13 - V_{w_2} = 7.915$$

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

The work done by impeller per kg of water per second is

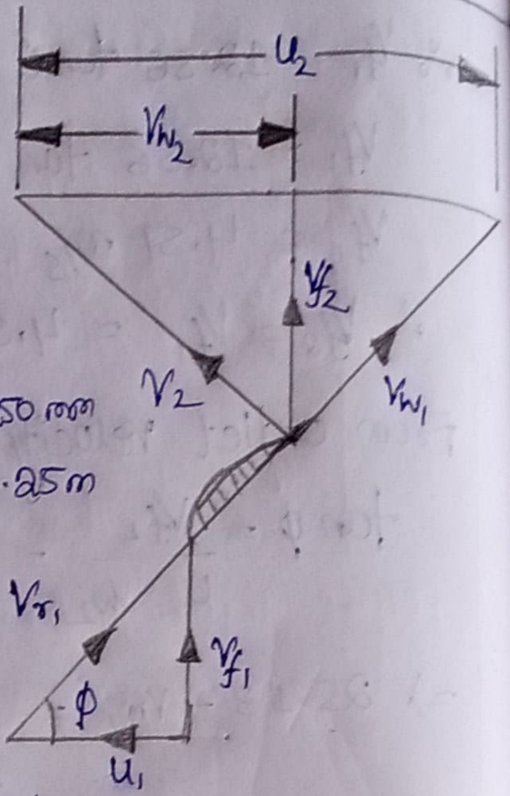
$$= \frac{1}{g} V_{w_2} u_2 = \frac{17.215 \times 25.13}{9.81}$$

$$= 44.1$$

Nm/N (Ans)

(Q2) A centrifugal pump is to discharge 0.118 m³/s at a speed of 1450 r.p.m., against a head of 25 m. The impeller diametere is 250 mm, its width at outlet is 50 mm and manometeric efficiency is 75% Deteremine the vane angle at the outere periphery of the impeller.

Given data:-




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

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

Head, $H_m = 25\ m$

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

Width at outlet $(B_2) = 50\ mm = 0.05\ m$

Manometric efficiency,

$\eta_{mm} = 75\% = 0.75$

Let vane angle at outlet $= \phi$

Tangential velocity of impeller at outlet,

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

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

$$\therefore V_{f_2} = \frac{Q}{\pi D_2 B_2} = \frac{0.118}{\pi \times 0.25 \times 0.05} = 3\ m/s$$

$$\eta_{mm} = \frac{gH_m}{V_{w_1} u_2} = \frac{9.81 \times 25}{V_{w_2} \times 18.98}$$

$$\therefore V_{w_2} = \frac{9.81 \times 25}{0.75 \times 18.98} = 17.23\ m/s$$

From outlet velocity triangle,

$$\tan\phi = \frac{V_{f_2}}{(u_2 - V_{w_2})} = \frac{3}{(18.98 - 17.23)} = 1.7143$$

$\phi = \tan^{-1} 1.7143$

$\phi = 59.74°$

Q3) A centrifugal pump deliveres watere aganist a net head of 14.5 meteres and a design speed of 1000 r.p.m. The vanes are curved back to an angle of 30° with the periphery. The impeller diametere is 300 mm and outlet width 50 mm. Determine the discharege of the pump if manometric efficiency is 95%.

Given data :-

Net head, $H_m = 14.5 m$

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

Vane angle at outlet, $\phi = 30°$

Impellere diametere means the diametere of the impellere at outlet

∴ Diametere, $D_2 = 300 mm = 0.30 m$

Outlet width, $B_2 = 50 mm = 0.05 m$

Manometric efficieny, $\eta_{mm} = 95\% = 0.95$

Tangential velocity of impellere at outlet,

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

$$\eta_{mm} = \frac{g H_m}{V_{w_2} U_2}$$

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

$\therefore V_{w_2} = \dfrac{0.95 \times 14.5}{0.95 \times 15.70} = 9.532 \text{ m/s}$

$\tan \phi = \dfrac{V_{f_2}}{(u_2 - V_{w_2})}$

$\Rightarrow \tan 30° = \dfrac{V_{f_2}}{(15.70 - 9.54)}$

$\Rightarrow \tan 30° = \dfrac{V_{f_2}}{6.16}$

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

$\therefore$ Discharge, $Q = \pi D_2 B_2 \times V_{f_2}$

$= \pi \times 0.30 \times 0.05 \times 3.556$

$= 0.1675 \text{ m}^3/\text{s}$ (Ans)

*) **Minimum Speed for Starting A Centrifugal pump :-**

If the pressure rise in the impeller is more than or equal to manometric head (hm), the Centrifugal pump will start delivering water.

The Centrifugal Head is equals to

$$= \frac{w^2 r_2^2}{2g} - \frac{w^2 r_1^2}{2g}$$

$w r_2$ = Tangential velocity of Empeller at outlet is equal to $u_2$

$w_1$ = Tangential velocity of empeller at inlet. is equal to $u_1$

So, Head due to preesurcised in empeller is $\frac{u_2^2}{2g} - \frac{u_1^2}{2g}$

If the manometreic head is greater than equal to head due to preesurcised in empeller. is less than equal to manometric head

$$\frac{u_2^2}{2g} - \frac{u_1^2}{2g} = h_m$$

If we consider that is minimum speed Head due to preesurcised is equal to manometer head

$$\boxed{\frac{u_2^2}{2g} - \frac{u_1^2}{2g} = h_m} \text{ --- (i)}$$

$$\eta_{mm} = \frac{g H_m}{V_{w_2} u_2}$$

$$H_m = \eta_{mm} \times \frac{V_{w_2} u_2}{g}$$

$$\boxed{H_m = \frac{V_{w_2} u_2 \times \eta_{mm}}{g}}$$

Substuting the of $h_m$ value in equ - (i)

$$\frac{u_2^2}{2g} - \frac{u_1^2}{2g} = \frac{V_{w_2} u_2 \times \eta_{mm}}{g}$$

$$\boxed{u_2 = \frac{\pi D_2 N}{60}} \Rightarrow \boxed{u_1 = \frac{\pi D_1 N}{60}} \text{ --- ?}$$

$$\Rightarrow \frac{\left(\frac{\pi D_2 N}{60}\right)^2}{2g} - \frac{\left(\frac{\pi D_1 N}{60}\right)^2}{2g} = \frac{V_{w_2} \times \frac{\pi D_2 N}{60} \times \eta_{mn}}{g}$$

$$\Rightarrow \frac{\pi N D_2^2}{120} - \frac{\pi N D_1^2}{120} = \eta_{mn} \times V_{w_2} D_2$$

$$\Rightarrow \frac{\pi N}{120}\left(D_2^2 - D_1^2\right) = \eta_{mn} \times V_{w_2} D_2$$

$$\boxed{N = \frac{120 \times \eta_{mn} \times V_{w_2} \times D_2}{\pi \times (D_2^2 - D_1^2)}}$$

Above equation gives the minimum starting speed of the centrifugal pump.

*) Multistage Centrifugal pumps for High Discharge

*) Multistage Centrifugal pumps :-

→ If a centrifugal pump consists of two or more impellers, the pump is called a "multistage centrifugal pump".

→ The impellers mounted on the same shaft or on different shafts.

A multistage pump is classified as two important functions :

1. To produce a high head
2. To discharge a large Quantity of liquid.

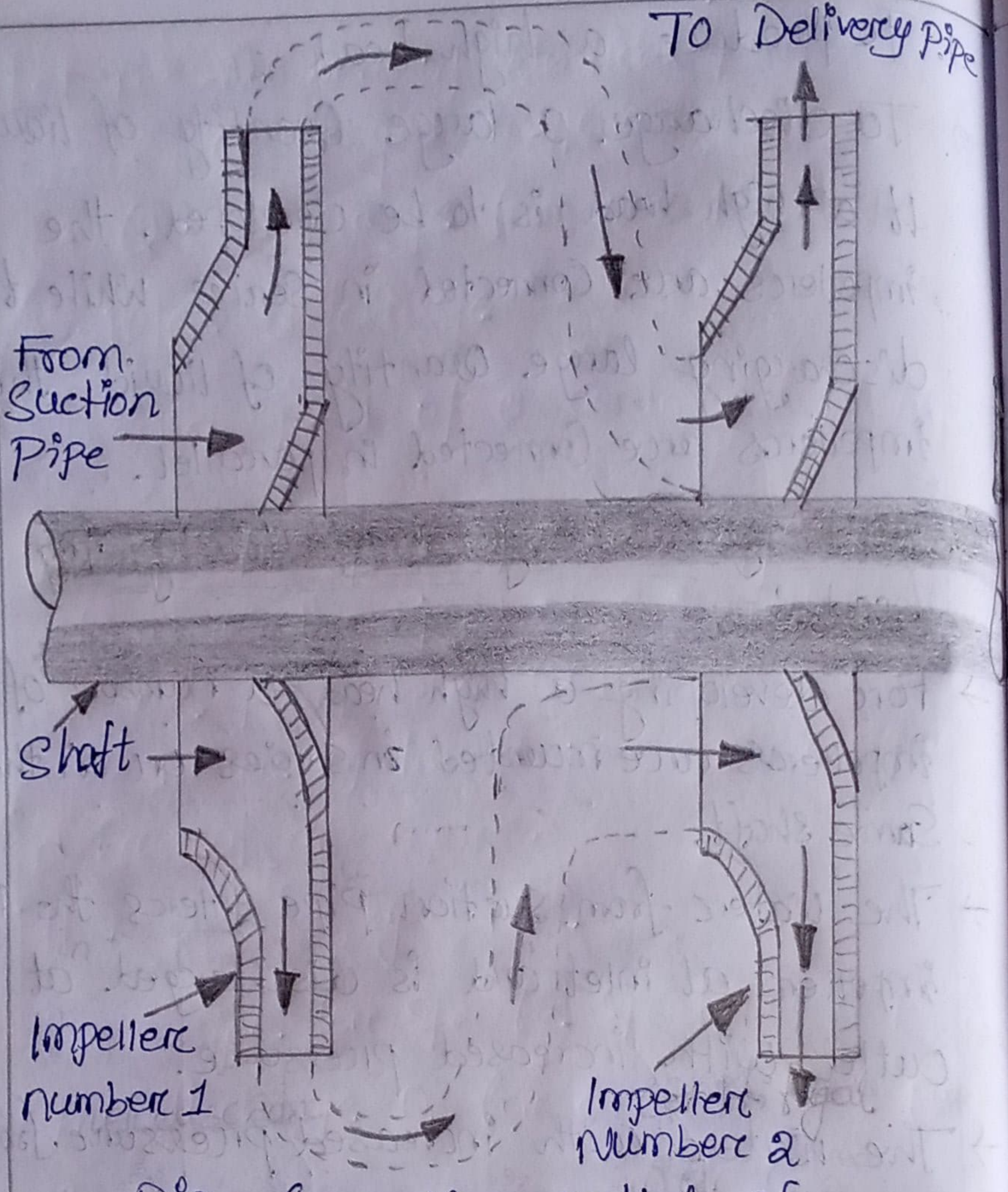
If a high head is to be developed, the impellers are connected in series while for discharging large Quantity of liquid, the impellers are connected in parallel.

* **Multistage Centrifugal pumps for high head :-**

→ For developing a high head, a number of impellers are mounted in series or on the same shaft.

→ The water from suction pipe enters the 1st impeller at inlet and is discharged at outlet with increased pressure.

→ The water with increased pressure from the outlet of the 1st impeller is taken to the inlet of the pressure 2nd impeller with the help of a connecting pipe. at outlet of the 2nd impeller the pressure of water will be more than the pressure of water at the outlet of the 1st impeller. Thus if more impellers are mounted on the same shaft, the pressure at the outlet will be increased further




Pipe Connecting outlet of 1st Impeller to Inlet of 2nd Impeller

(Two-stage pumps with impellers in Series)

Let $n$ = Number of identical impellers mounted on the same Shaft.

$H_m$ = Head developed by each impeller.

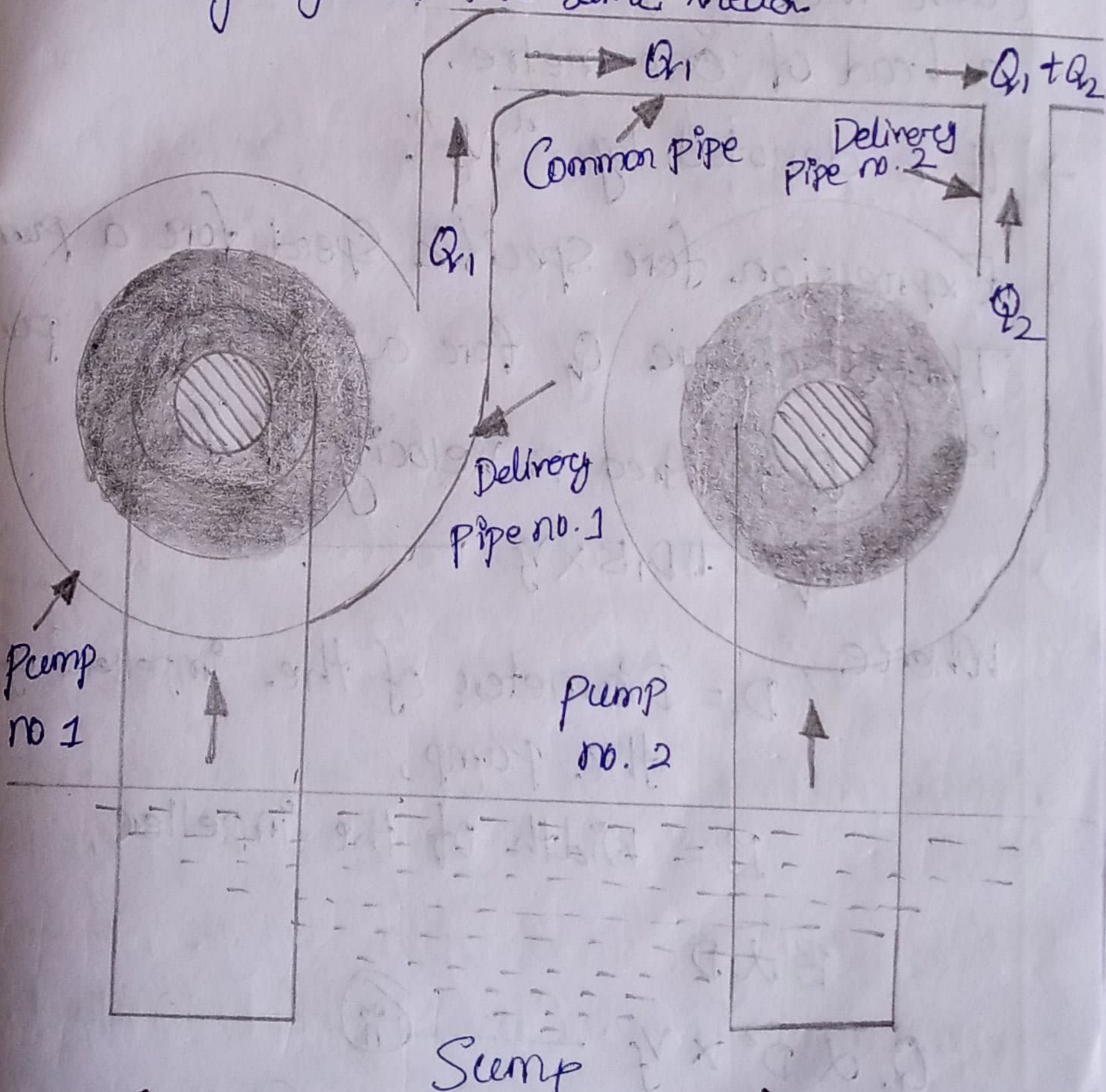
Then total head developed

$= n \times H_m$

The discharge passing through each impeller is Same.

<u>multistage Centrifugal pumps fore High discharge</u> :-

The Pump Should be Connected in parcallel. Each of the pumps lifts fore water from a Common pump and discharges water to a Common pipe to which the delivery pipes of each pump in Conneeted. Each of the pump is working aganist the same head




( pump in parcallel )

Let $n$ = Numbere of identical pumps arreanged in parallel.

$Q$ = Discharege from one pump

$\therefore$ Total discharege $= n \times Q$

## * Specific Speed of a Centrifugal Pump ($N_s$):

The Specific Speed of a Centrifugal pump is defined as the Speed of a geometrically similar pump which would deliver one cubic meter of liquid per second aganist a head of one metre.

→ It is denoted by "$N_s$".

Expression for Specific Speed for a Pump.

The discharge $Q$ for a centrifugal pump is

$$Q = \text{Area} \times \text{velocity of flow}$$

$$Q = \pi D_1 B \times V_f \quad \text{——— (i)}$$

Where, $D$ = Diameter of the impeller of the pump,

$B$ = Width of the impeller,

$$B \propto D$$

$$Q \propto D^2 \times V_f \quad \text{——— (ii)}$$

$$u = \frac{\pi D N}{60} \propto DN \quad \text{——— (iii)}$$

Now the tangential velocity ($u$) and velocity of flow ($V_1$) are related to the manometric head ($H_m$) is

$$u \propto V_f \propto \sqrt{H_m} \quad \text{--- (iv)}$$

Substituting the value of $u$ in eqn (iii),

$$\sqrt{H_m} \propto DN$$

$$= D \propto \frac{\sqrt{H_m}}{N}$$

Substuting the values of D in equn (ii),

$$Q \propto \frac{H_m}{N^2} \times V_f$$

$$Q \propto \frac{H_m}{N^2} \sqrt{H_m}$$

$$Q \propto \frac{H_m^{3/2}}{N^2}$$

$$Q = K \frac{H_m^{3/2}}{N^2} \quad \text{--- (v)}$$

Where K is a Constant of proportionality.

If $H_m = 1m$ and $Q = 1\ m^3/s$, $N = N_s$

Substuting the values of K in equation (v),

$$Q = N_s^2 \frac{H_m^{3/2}}{N^2}$$

$$N_s^2 = \frac{N^2 Q}{H_m^{3/2}} \Rightarrow N_s = \sqrt{\frac{N^2 Q}{H_m^{3/2}}}$$

$$\therefore N_s = N\sqrt{Q} / H_m^{3/4}$$

## *) Characteristic Curves of Centrifugal Pumps:

→ Characteristic Curves of Centrifugal are defined those Curves which are plotted from the results of a number of tests on the Centrifugal pump.

→ These curves are necessary to predict the behaviour and performance of the pump when the pump is working under different flow rate, head and Speed.

→ The following important Characteristic Curves for pumps:

1. Main Characteristic Curves,
2. Operating characteristic curves,
3. Constant efficiency or Muschel curves.

### Main Characteristic Curves :-

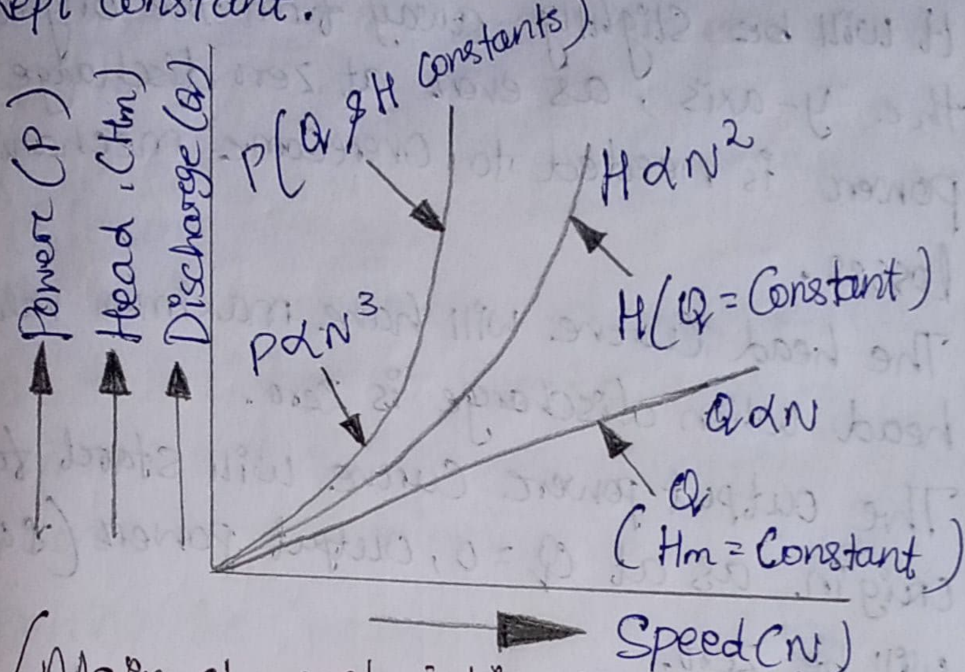
→ The main Characteristic Curves of a Centrifugal pump Consists of variation of head, Power and discharge with respect to Speed.

→ For plotting curves of manometric head versus Speed, discharge is kept Constant.

→ for plotting Curves of discharge versus Speed manometric head ($H_m$) is kept Constant.

→ For plotting curves of power versus speed, the manometric head and discharge are

Kept Constant.




(Main characteristic curves of a pump)

For plotting the graph of Hm versus speed (N), the discharge is kept constant.

$\sqrt{H_m}/DN$ is a constant

$H_m \alpha N^2$

$\frac{Q}{D^3N}$ = Constant

$Q \alpha N$

Operating characteristic curves :-

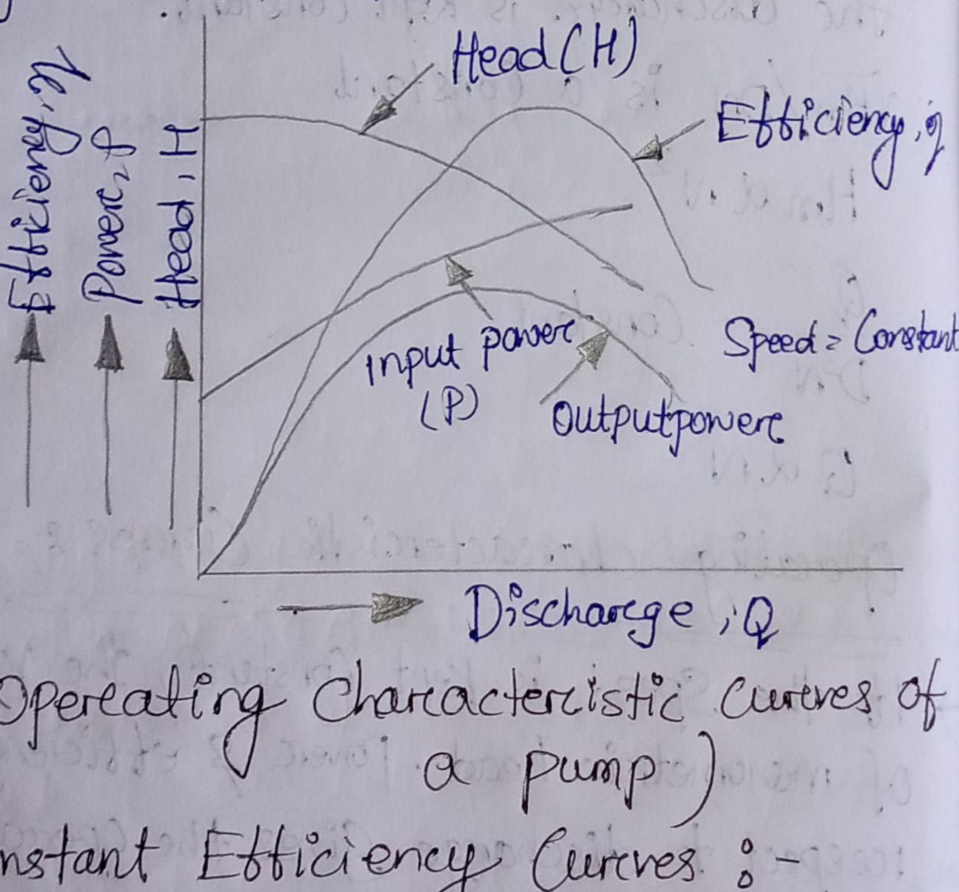
- If the speed is kept constant, the variation of manometric head, power & efficiency with respect to discharge gives the operating characteristics of the pump.
- The input power curve for pumps shall not pass through the origin.

→ It will be slightly away from the origin on the y-axis, as even at zero discharge some power is needed to overcome mechanical losses.

→ The head curve will have maximum value of head when discharge is zero.

→ The output power curve will start from origin as at $Q = 0$, output power ($\rho Q g H$) will be zero.

The efficiency curve will start from origin at $Q = 0$; $\eta = 0$




(Operating characteristic curves of a pump)

*) **Constant Efficiency Curves :-**

Constant efficiency curves for a pump, the head versus curves and efficiency veresus discharege curves fore different speed are used.

→ The head versus discharge curves for different speeds.

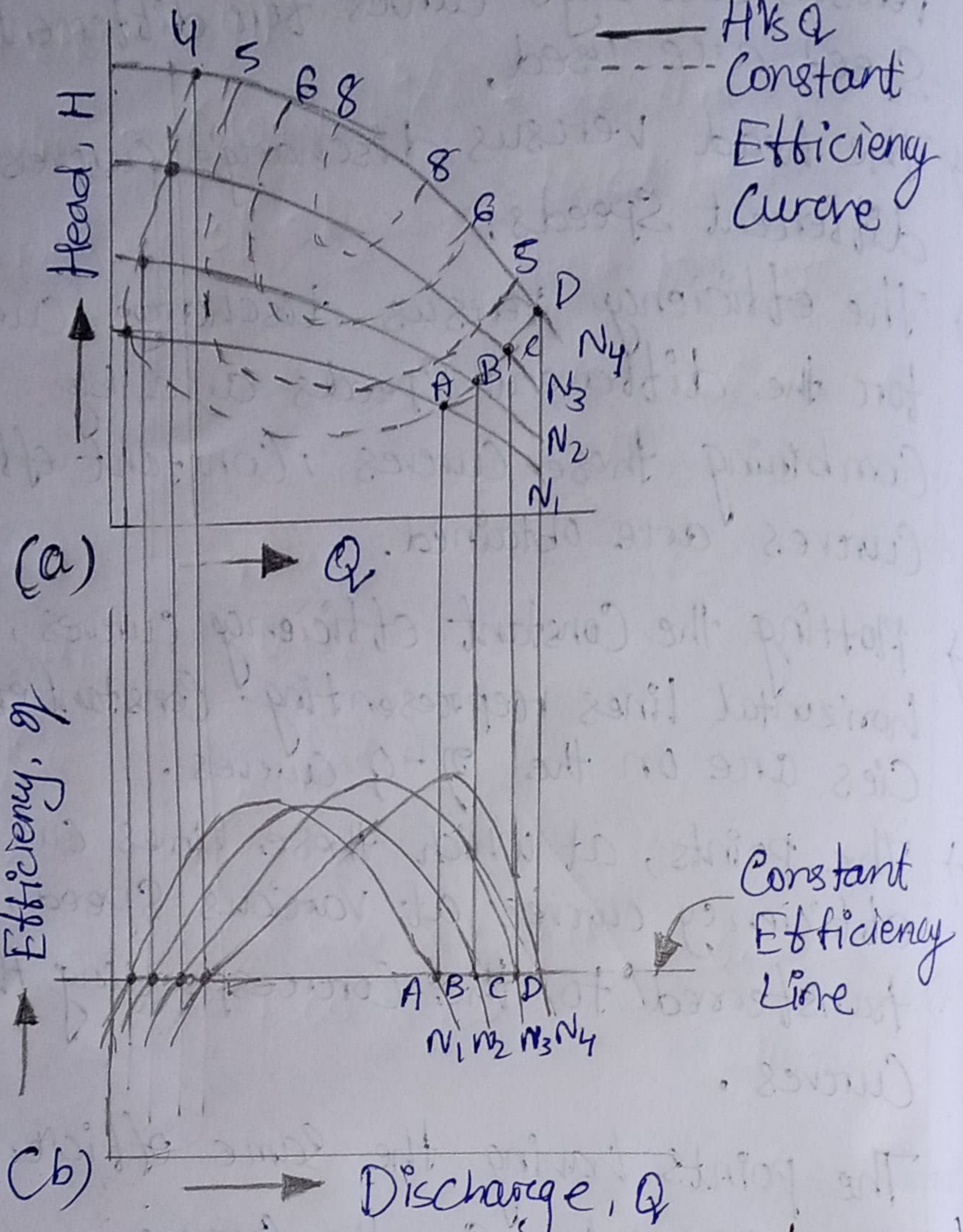
→ The efficiency veresus discharge curves for the different speeds are by combining these curves, constant efficiency curves are obtained.

→ Plotting the constant efficiency curves, horizontal lines representing constant efficiencies are on the $\eta - Q$ curves.

→ The points, at which these lines cut the efficiency curves at various speed, are transferred to the corresponding H-Q curves.

The points having the same efficiency are joined by smooth curves.

These smooth curves represents the iso-efficiency curves.




(Constant Efficiency Curves of a pump)

Q4) A single-stage centrifugal pump with impeller diameter of 30 cm rotates a 2000 r.p.m. and lifts $3m^3$ of water per second to a height of 30m with an efficiency of 75% find the number of stages and diameter of each impeller of a similar multistage pump of lift $5m^3$ of water per second to a height of 200 meters when rotating at

1500 r.p.m.

Given data :-

Single-stage pump :

Diametere of impellere, $D_1 = 30\,cm = 0.30\,m$

Speed, $N_1 = 2000$ r.p.m.

Discharge, $Q_1 = 3\,m^3/s$

Height, $Hm_1 = 30\,m$

Efficiency, $\eta_{mm} = 75\% = 0.75$

multistage similare pump :-

Discharegre, $Q_2 = 5\,m^3/s$

Total height $= 200m$

Height $= Hm_2$

Speed, $N_2 = 1500$

Diameter of each impellere $= D_2$

$$\left(\frac{N\sqrt{Q}}{Hm^{3/4}}\right)_1 = \left(\frac{N\sqrt{Q}}{Hm^{3/4}}\right)_2$$

$$\therefore \frac{N_1\sqrt{Q_1}}{Hm_1^{3/4}} = \frac{N_2\sqrt{Q_2}}{Hm_2^{3/4}}$$

$$\Rightarrow \frac{2000\sqrt{3}}{30^{3/4}} = \frac{1500\sqrt{5}}{Hm_2^{3/4}}$$

$$\Rightarrow Hm_2^{3/4} = \frac{1500\sqrt{5} \times 30^{3/4}}{2000\sqrt{3}} = \frac{1500}{2000} \times \sqrt{\frac{5}{3}}$$

$= 12.818 = 12.411$

$\therefore H_{m_2} = (12.411)^{4/3} = 28.71 \text{ m}$

$\therefore$ Number of stages $= \dfrac{\text{Total head}}{\text{Head per stage}}$

$= \dfrac{20}{28.71} = 6.96 \approx 7$

$$\frac{\sqrt{H_{m_1}}}{D_1 N_1} = \frac{\sqrt{H_{m_2}}}{D_2 N_2}$$

$$\Rightarrow \frac{\sqrt{30}}{0.30 \times 2000} = \frac{\sqrt{28.71}}{D_2 \times 1500}$$

$$\Rightarrow D_2 = \frac{0.30 \times 2000 \times \sqrt{28.71}}{1500 \times \sqrt{30}} = 0.3913$$

$= 391.3 \text{ mm}$

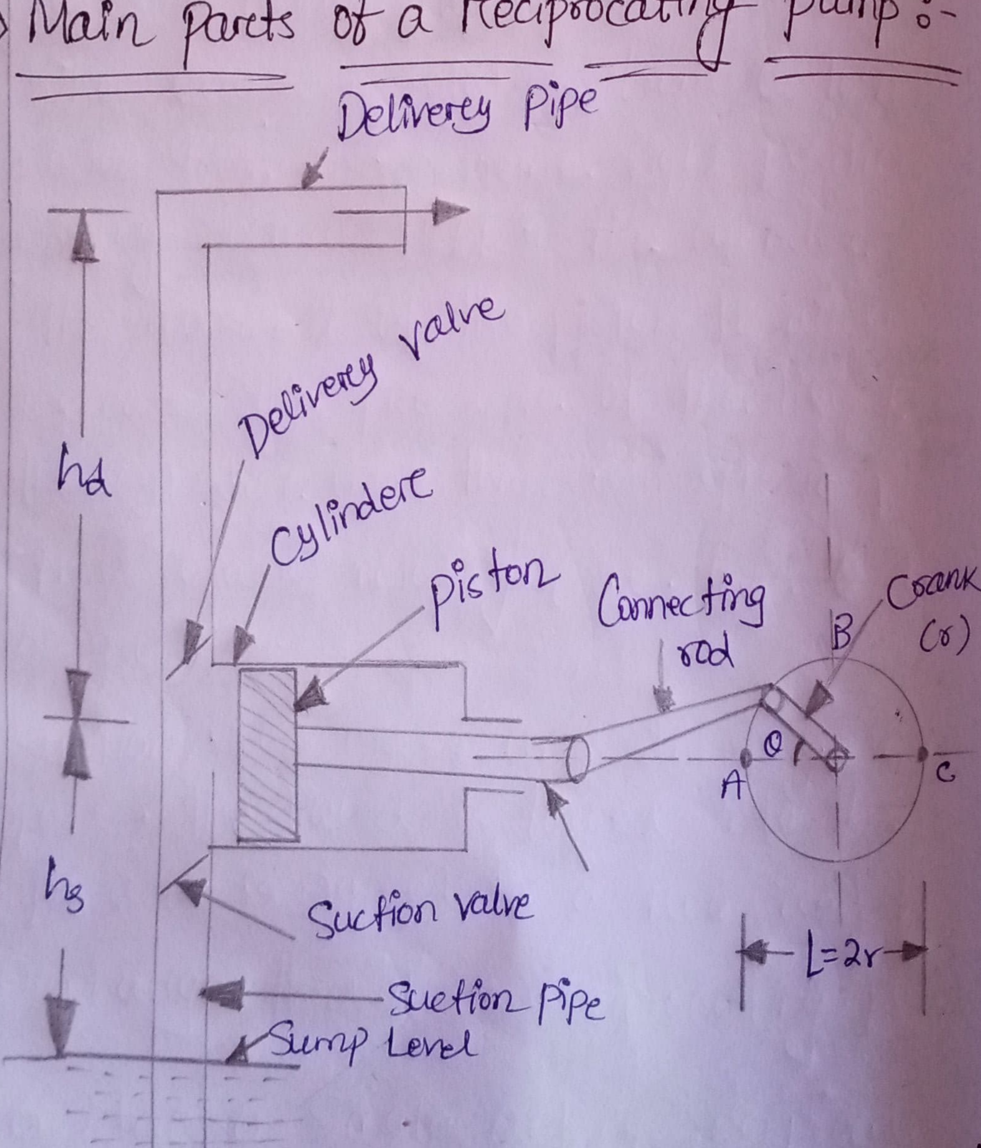
(Ans)

# Chaptere-3 Reciprocating Pumps

## *) Introduction of reeciprocating pump :-

If the Mechanical Energy is Converted into hydraulic Energy or preessure Energy by Sucking the liquid into a cylindere in which a piston is reciprocating, which exerts the thrust on the liquid and increases its hydraulic Energy, the pump is known as "reciprocating Pump".

## *) Main parts of a reeciprocating pump :-




( Main Parts of a reeciprocating Parts )

The Main parts of a reciprocating pump are

i) A cylindere with piston, piston rod, connecting rod and a crank,

ii) Suction pipe,

iii) Deliverey pipe,

iv) Suction valve

v) Deliverey valve

## * Clasification of reciprocating pumps :-

The reciprocating pumps are classified as

1) According to the acting or Contact of water

2) According to number of cylinder

according to the Contact of water :

i) Single - acting pump,

ii) Double - acting pump

according to number of cylinder :

i) Single cylinder pump

ii) Double cylinder pump

iii) Triple cylinder pump

## * Working of a reciprocating pump :-

→ A single acting reciprocating pump, which consists of a piston which moves forwards and backwards in a close fitting cylinder.

→ The movement of the piston is obtained by connecting the piston rod to crank by means of a connecting rod. The crank is rotated by means of an electric motor.

→ Suction and delivery pipes with suction valve and delivery valve are connected to cylinder.

→ The suction and delivery valves are one way valves or non-return valves, which allow the water to flow in one direction only.

→ Suction valve allows water from suction pipes to the cylinder which delivery valve allows water from cylinder to delivery pipe only.

→ When crank starts rotating, the piston moves to and fro in the cylinder. When crank is at A, the piston is at the extreme left position in the cylinder. the crank is rotating from A to C ( i.e., from $\theta = 0^\circ$ to $\theta = 180^\circ$ ), the piston is moving towards right in the cylinder.

→ The movement of the piston towards right creates a partial vaccum in cylinder.

→ When crank is rotating from C to A, the piston from its extreme right position starts moving towards left in the cylinder.

→ The movement of the piston towards left increases the pressure of the liquid inside the cylinder more than atmospheric pressure.

→ Thus crank rotating ( i.e. from $\theta = 180^\circ$ to $\theta = 360^\circ$ ).

→ Suction valve closes and delivery valve opens. the liquid is forced into the delivery pipe and is raised to a required height.

*) **Discharge through a reciprocating pump :-**

Consider a single acting reciprocating pump

Let D = Diameterr of the cylinderr

A = Cross - sectional areea of the piston
ore cylinderr

$= \pi/4 \times D^2$

$r$ = Radius of crank

N = r.p.m. of the crank

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

$h_s$ = Height of the axis of the cylinderr
from waterr surface in sump.

$h_d$ = Height of delivery outlet above
the cylinderr axis

Volume of watere deliverred in one revolution,
ore discharge of waterr in one reevolution

$= \text{Area} \times \text{Length of stroke}$

$= A \times L$

Numberr of revolution perr Second, $= \frac{N}{60}$

∴ Discharge of the pump perr second,

$Q$ = Discharge in one reevolution × No. of
reevolution perr second

$= A \times L \times \frac{N}{60} = \frac{ALN}{60}$

Weight of watere deliverred perr Second,

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

## * Workdone by Reciprocating Pump :-

Workdone by the reciprocating pump per second is

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

$$= W \times (h_s + h_d)$$

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

$$W = \frac{\rho g A L N}{60}$$

Substituting the value of W in eqn (i)

$$\text{Workdone per second} = \frac{\rho g A L N}{60} \times (h_s + h_d)$$

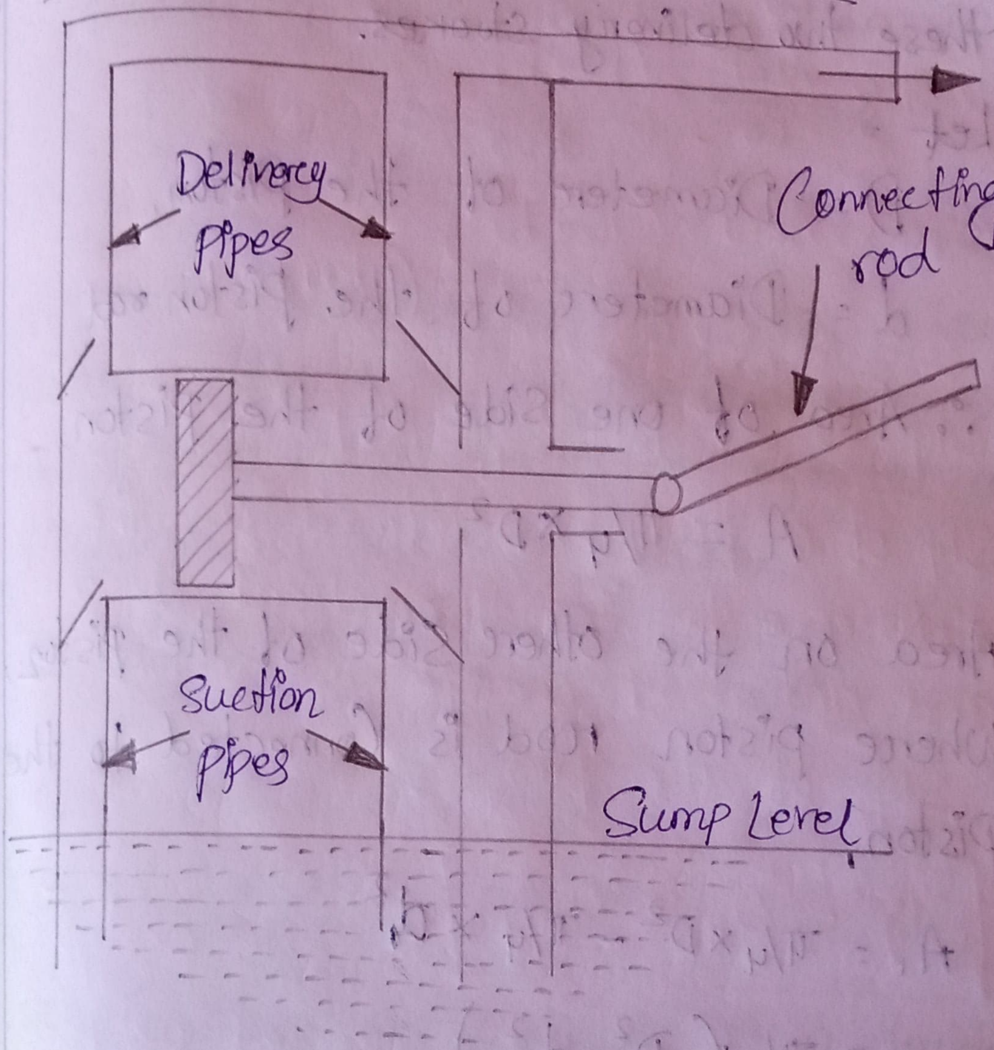
∴ Power required to drive the pump, in KW

$$P = \frac{\text{Workdone per second}}{1000}$$

$$= \frac{\rho g A L N \times (h_s + h_d)}{60 \times 1000}$$

$$= \frac{\rho g A L N \times (h_s + h_d)}{60{,}000} \text{ KW}$$

# Discharge, Workdone and Power required to drive a Double-acting pump.




- In double acting pump, the water is acting on both sides of the piston. We require two suction pipes and two delivery pipes for double-acting pump.
- When there is a suction stroke on one side of the piston, there is at the same time a delivery stroke on the other side of the piston.
- Thus for one complete revolution of the crank

there are two delivery strokes and water is delivered to the pipes by the pump during these two delivery strokes.

Let =

D = Diameter of the Piston,

d = Diameter of the piston rod

∴ Area of one side of the piston,

$$A = \pi/4 \times D^2$$

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

$$A_1 = \pi/4 \times D^2 - \pi/4 \times d^2$$

$$= \pi/4 (D^2 - d^2)$$

∴ Volume of water delivered in one revolution of crank

$= A \times$ Length of stroke $+ A_1 \times$ Length of stroke

$= AL + A_1L$

$= (A + A_1)L$

$= \left[\pi/4 \times D^2 + \pi/4 (D^2 - d^2)\right] \times L$

∴ Discharge of pump per second

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

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

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

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

$$Q = 2 \times \frac{\pi}{4} \times D^2 \times \frac{L \times N}{60}$$

$$Q = \frac{2ALN}{60} \quad \text{—— (i)}$$

eqn-(i) Gives the discharge of a double-acting reciprocating pump. This discharge is two times the discharge of a single-acting pump.

## Work done by double-acting reciprocating pump :-

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

= $\rho g$ × Discharge per second × Total height

$= \rho g \times$ Discharge per second $\times$ Total height

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

$= 2\rho g \times \frac{ALN}{60} \times \frac{(h_s + h_d)}{1000}$

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

## * Slip of reciprocating Pump :-

Slip of a pump is defined as the difference between theoretical discharge and actual discharge is known as "slip of the pump".

$$\boxed{Slip = Q_{th} - Q_{act}}$$

But slip is mostly expressed as percentage slip which is

$$\text{percentage Slip} = \frac{Q_{th} - Q_{act}}{Q_{th}} \times 100$$

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

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

$$\left(\because \frac{Q_{act}}{Q_{th}} = C_d\right)$$

where $C_d$ = Co-efficient of discharge

*) **Negative slip of the reciprocating pump :-**

→ Slip is equal to the difference of theoretical discharge and actual discharge.

→ If actual discharge is more than the theoretical discharge, the slip of the pump will become -ve. In this case, the slip of the pump is known as negative slip.

→ Negative slip occures when delivery pipe is long and pump is running at high speed.

Q1) A single acting reciprocating pump, running at 50 r.p.m., delivers 0.01 $m^3/s$ of water the diametere of the piston is 200mm and stroke length 400mm. Determine :

i) The theoretical discharge of the pump.

ii) Co-efficient of discharge,

iii) Slip and the percentage slip of the pump.

Given data :-

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

Actual discharge, $Q_{act} = 0.01\ m^3/s$

Dia of piston, $D = 200mm = 0.20m$

$\therefore$ Area, $A = \pi/4 \times D^2$

$= \pi/4 \times (0.20)^2$

$= 0.03146\ m^2$

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

i) Theoretical discharge for single-acting reciprocating pump is

$$Q_{th} = \frac{A \times L \times N}{60} = \frac{0.03146 \times 0.40 \times 50}{60}$$

$$= 0.01048\ m^3/s$$

ii) Co-efficient of discharge is,

$$C_d = \frac{Q_{act}}{Q_{th}} = \frac{0.01}{0.01048} = 0.954$$

iii) Slip $= Q_{th} - Q_{act}$

$= 0.01048 - 0.01 = 0.00048$

And percentage slip

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

$$= \frac{slip}{Q_{th}} \times 100$$

$$= \frac{0.00048}{0.01048} \times 100$$

$$= 4.58\ \%$$

Q2) A double-acting reciprocating pump, running at 40 r.p.m., is discharging 1.0 m³ of water per minute. The pump has a large stroke of 400 mm. The diameter of the piston is 200mm. The delivery and suction head are 20m and 5m Respectively. Find the slip of the pump and power required to drive the pump.

Given data :-

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

actual discharge, $Q_{act} = 1.0\ m^3/min$

$$= \frac{1.0}{60}\ m^3/s = 0.01666\ m^3/s$$

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

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

∴ Area, $A = \pi/4 \times D^2 = \pi/4 \times 0.20^2 = 0.031416\ m^2$

Suction head, $h_s = 5m$

delivery head, $h_d = 20m$

Theoretical discharge fore double acting Pump is

$$Q_{th} = \frac{2ALN}{60} = \frac{2 \times 0.031416 \times 0.4 \times 40}{60}$$

$$Q_{th} = 0.01675 \ m^3/s$$

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

$$= 0.01675 - 0.01666$$

$$= 0.00009 \ m^3/s$$

Powere reeauirced to dreive the double acting pump is

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

$$P = \frac{2 \times 1000 \times 9.81 \times 0.031416 \times 0.4 \times 40 \ (5 + 20)}{60{,}000}$$

$$P = 4.109 \ KW$$

*) <u>Establish relation between Slip & Co-efficient of discharge</u> :-

Slip is expreessed as $\text{Slip} = Q_{th} - Q_{act}$

but Slip is mostly Expreessed as

$$\% \ \text{Slip} = \frac{Q_{th} - Q_{act}}{Q_{th}} \times 100$$

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

$(1 - C_d) \times 100$

$$\left( \because \frac{Q_{act}}{Q_{th}} = C_d \right)$$

where $C_d$ = Co-efficient of discharge

$C_d$ is known as Co-efficient of discharge and is defined as the reatio of the actual discharge to the theoretical discharge.

$$C_d = \frac{Q_{act}}{Q_{th}}$$

$C_d$ value reanges between 95-98%

percentage slip is of the ordere of 2% fore pumps in good Conditions.

* Variation of velocity and acceleration in the Suction and deliverey pipes due to acceleration of the piston :-

→ When creank starts rotating, the piston moves forwards and backwards in the cylindere at the extreme left position and right position of the piston in the cylindere.

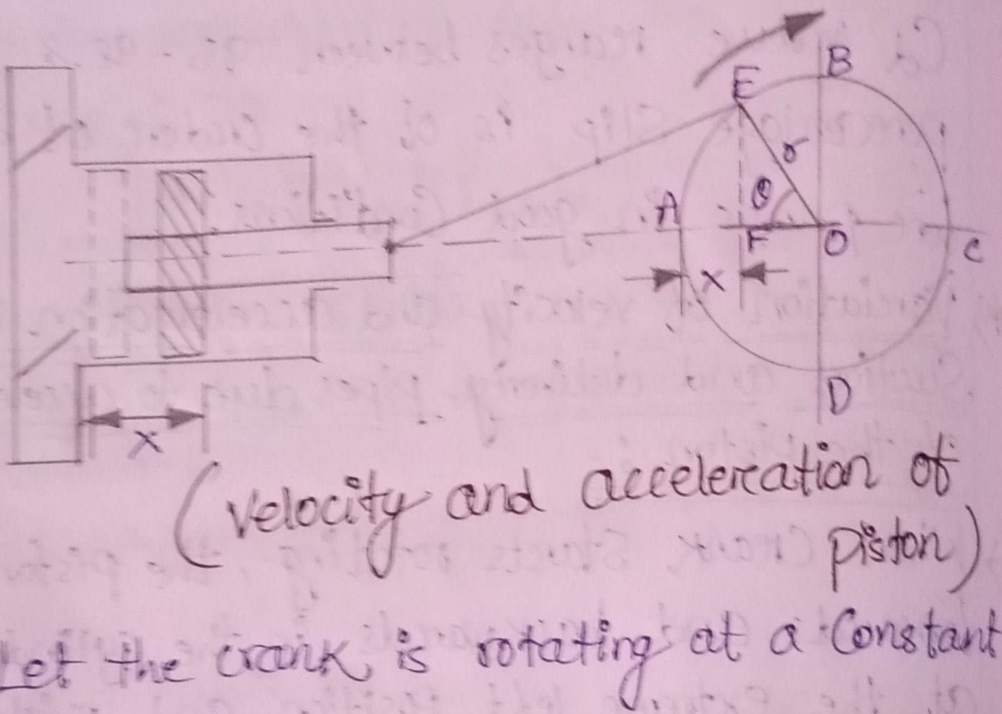
→ the velocity of flow is zero. The velocity of the piston is maximum at the centre of the cylindere.

→ The piston will be having an acceleration and at the end of each stroke, the piston will be having a retardation. the velocity of flow of watere in the suction & deliverey pipe will

will not be uniform.

→ This accelerative or retarding head will change the pressure inside the cylinder.

→ If the ratio of length of connecting rod to the radius of crank is very large, then the motion of the piston can be assumed as simple harmonic the cylinder of a reciprocating single acting pump fitted with a piston which is connected to the crank.




(velocity and acceleration of piston)

Let the crank is rotating at a constant angular speed.

Let $\omega$ = Angular speed of the crank in rad/s,

$A$ = Area of the ~~pipe~~ cylinder,

$a$ = Area of the pipe (suction or delivery),

$l$ = Length of the pipe (suction or delivery),

$r$ = radius of the crank.

→ The crank is at A and the piston in the cylinder is at a position by dotted lines.

→ The crank is rotating with an angular velocity $\omega$ and Let in time "t" seconds, the crank turns through an angle $\theta$ from A. The displacement of the piston in time "t" is "x".

Now $\theta$ = angle turned by crank in radians in time "t"

$= \omega t$

The distance $x$ travelled by the piston is

$x$ = Distance AF

$= AO - FO \quad (\because AO = r,\ FO = r\cos\theta)$

$= r - r\cos\theta$

$= r - r\cos(\omega t) \quad (\because$ from (i), $\theta = \omega t)$ —(ii)

The velocity of the piston is with respect to "t".

$\therefore$ velocity of piston, $V = \dfrac{dx}{dt}$

$= \dfrac{d}{dt}(r - r\cos(\omega t))$

$= 0 - r(-\sin\omega t) \times \omega$

$= \omega r \sin\omega t.$

The Volume of water flowing into cylinder per second is equal to the volume of water flowing from the pipe per second.

$\therefore$ Velocity of water in cylinder × Area of Cylinder

= Velocity of water in pipe × Area of pipe

$V \times A = v \times a$

Where as $v$ = Velocity of water in pipe

$$v = \frac{V \times A}{a} = \frac{A}{a} \times V$$

$$= \frac{A}{a} \omega r \sin \omega t$$

the acceleration of water in pipe is

$\therefore$ Acceleration of water in pipe

$$= \frac{dv}{dt} = \frac{d}{dt}\left(\frac{A}{a} \omega r \sin \omega t\right)$$

$$= \frac{A}{a} \omega^2 r \cos \omega t$$

Mass of water in pipe = $\rho \times$ Volume of water in pipe

$= \rho \times$ (Area of pipe $\times$ Length of pipe)

$= \rho \times [a \times l]$

$= \rho a l$

$\therefore$ force required to accelerate the water in the pipe

= mass of water in pipe $\times$ acceleration of water in pipe

$$= \rho a l \times \frac{A}{a} \omega^2 r \cos \omega t$$

$\therefore$ Intensity of pressure due to acceleration

$$= \frac{\text{force required to accelerate the water}}{\text{Area of pipe}}$$

$$= \frac{\rho a l \times \frac{A}{a} \omega^2 r \cos \omega t}{a} = \rho l \times \frac{A}{a} \omega^2 r \cos \omega t$$

$= \rho l \times \frac{A}{a} \omega^2 r \cos\theta \quad (\because \omega t = \theta)$

$\therefore$ pressure head ($h_a$) due to acceleration

$$h_a = \frac{\text{Intensity of pressure due to acceleration}}{\text{Weight density of liquid}}$$

$$= \frac{\rho l \times \frac{A}{a} \omega^2 r \cos\theta}{\rho g}$$

$$= \frac{l}{g} \times \frac{A}{a} \omega^2 r \cos\theta$$

The pressure head due to acceleration in the suction and delivery pipes is by "s" and "d"

$$h_{as} = \frac{l_s}{g} \times \frac{A}{a_s} \omega^2 r \cos\theta$$

$$h_{ad} = \frac{l_d}{g} \times \frac{A}{a_d} \omega^2 r \cos\theta$$

The pressure head ($h_a$) due to acceleration varies with $\theta$.

the value of "$h_a$" for different values of $\theta$ are:

1. When $\theta = 0$, $h_a = \frac{l}{g} \times \frac{A}{a} \omega^2 r$, as $\cos 0^\circ = 1$
2. When $\theta = 90^\circ$, $h_a = 0$ as $\cos 90^\circ = 0$
3. When $\theta = 180^\circ$, $h_a = -\frac{l}{g} \times \frac{A}{a} \omega^2 r$, as $\cos 180^\circ = -1$

$$(h_a)_{max} = \frac{l}{g} \times \frac{A}{a} \omega^2 r$$

## * Effect of Variation of velocity on friction in the suction and delivery pipes :-

The velocity of water in suction or delivery pipe is

$$V = \frac{A}{a} \omega r \sin \omega t$$

$$= \frac{A}{a} \omega r \sin\theta \quad \text{——— (i)}$$

Loss of head due to friction in pipes is

$$h_f = \frac{4flV^2}{d \times 2g} \quad \text{——— (ii)}$$

Where, $f$ = Co-efficient of friction,

$l$ = Length of pipe,

$d$ = Diameter of pipe,

$v$ = Velocity of water in pipe.

Substituting equation (i) into equation (ii):

$$h_f = \frac{4fl}{d \times 2g} \times \left[\frac{A}{a} \omega r \sin\theta\right]^2$$

→ The variation of $h_f$ with $\theta$ is parabolic.

→ The loss of head due to friction in suction and delivery pipes is by using subscripts "s" for suction pipe and "d" for delivery pipe

$$h_{fs} = \frac{4fl_s}{d_s \times 2g} \times \left[\frac{A}{a_s} \omega r \sin\theta\right]^2$$

$$h_{fd} = \frac{4fl_d}{d_d \times 2g} \times \left[\frac{A}{a_d} \omega r \sin\theta\right]^2$$

The loss of head due to friction in pipes which varies with $\theta$ as :

1. When $\theta = 0°$, $\sin\theta = 0$ $\therefore h_f = \frac{4fl}{d \times 2g} \times 0 = 0$

2. When $\theta = 90°$, $\sin 90° = 1$ $\therefore h_f = \frac{4fl}{d \times 2g} \times \left[\frac{A}{a} \omega r\right]^2$

3. When $\theta = 180°$, $\sin 180° = 0$ $\therefore h_f = 0$

$\therefore$ Maximum value of loss of head due to friction ;

$$(h_f)_{max} = \frac{4fl}{d \times 2g} \times \left[\frac{A}{a} \omega r\right]^2$$

3) The cylinder bore diameter of a single acting reciprocating pump is 150 mm and its stroke is 300 mm. The pump runs at 50 r.p.m. and lifts water through a height of 25 m. The delivery pipe is 22 m long and 100 mm in diameter. Find the theoretical discharge and the theoretical power required to run the pump. If the actual discharge is 4.2 litres/s, find the percentage slip. Also determine the acceleration head at the beginning and middle of the delivery stroke.

Given data :-

Dia. of cylinder, $D = 150\text{mm} = 0.15\text{m}$

$\therefore$ Area, $A = (\pi/4) \times 0.15^2 = 0.01767\ m^2$

Stroke, $L = 300 mm = 0.3 m$

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

Total height through which water is lifted,

$$H = 25 m$$

Length of delivery pipe, $l_d = 22 m$

Diameter of delivery pipe, $d_d = 100 mm = 0.1 m$

Actual discharge, $Q_{act} = 4.2$ litres/s

$$= \frac{4.2}{1000} m^3/s$$

$$= 0.0042 m^3/s$$

i) Theoretical discharge ($Q_{th}$)

Theoretical discharge for a single-acting reciprocating pump is

$$Q_{th} = \frac{A \times L \times N}{60} = \frac{0.01767 \times 0.3 \times 50}{60}$$

$$= 0.0044175 m^3/s$$

$$= 0.0044175 \times 1000 \text{ litres/s}$$

$$= 4.4175 \text{ litres/s}$$ (Ans)

ii) Theoretical power ($P_t$)

Theoretical power is

$$P_t = \frac{(\text{Theoretical weight of water litre/s}) \times \text{Total height}}{1000}$$

$$= \frac{\rho \times g \times Q_{th} \times H}{1000} = \frac{1000 \times 9.81 \times 0.0044175 \times 25}{1000}$$

$= 1.0833$ KW. (Ans)

iii) Percentage slip

The percentage slip is

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

$$= \left(\frac{4.4175 - 4.2}{4.4175}\right) \times 100$$

$$= 4.92\%$$ (Ans)

iv) Acceleration head at the beginning of delivery stroke.

The acceleration head in the delivery pipe is

$$h_{ad} = \frac{l_d}{g} \times \frac{A}{a_d} \omega^2 r \times \cos\theta$$

Where $a_d$ = Area of delivery pipe

$$= \pi/4 \times (0.1)^2$$

$$= 0.007854$$

$\omega$ = Angular Speed $= \frac{2\pi N}{60} = \frac{2 \times \pi \times 50}{60}$

$r$ = Crank radius $= 5.236$

$$= \frac{L}{2} = \frac{0.3}{2} = 0.15 \text{ m}$$

$$\therefore h_{ad} = \frac{22}{9.81} \times \frac{0.01767}{0.007854} \times 5.236^2 \times 0.15 \times \cos\theta$$

$$= 20.75 \times \cos\theta$$

At the beginning of delivery stroke, $\theta = 0$ and $\cos\theta = 1$

$\therefore h_{ad} = 20.75\,m$ (Ans)

v) Acceleration head at the middle of delivery stroke.

At the middle of delivery stroke,

$\theta = 90^\circ$ and $\cos\theta = 0$

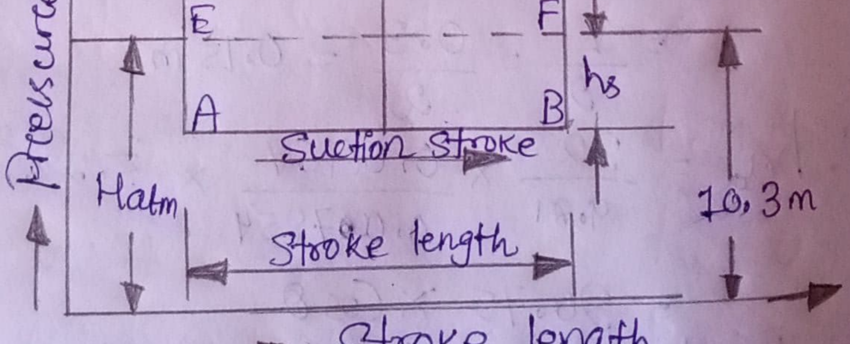
$\therefore h_{ad} = 20.75 \times 0 = 0$ (Ans)

## *) Indicator diagram :-

→ The indicator diagram for a reciprocating pump is defined as the graph between the pressure head in the cylinder and the distance travelled by piston from inner dead centre for one complete revolution of the crank.

→ The pressure head is taken as ordinate and stroke length as abscissa.

### Ideal Indicator diagram :-




(Ideal indicator diagram)

→ The graph between pressure head in the cylinder and stroke length of the piston for one complete revolution of the crank under ideal conditions is known as "Ideal indicator diagram".

→ EF represents the atomspheric pressure head equal to 10.3 m of water.

Let $H_{atm}$ = Atmospheric pressure head
= 10.3 m of water,

L = Length of the stroke,

$h_s$ = Suction head, and

$h_d$ = Delivery head.

→ During suction stroke, the pressure head in the cylinder is constant and equal to suction head ($h_s$). which is below the atmospheric pressure head ($h_{atm}$) by a height of $h_s$.

The pressure head during suction stroke is represented by a horizontal line AB which is below the line EF by a height of "$h_s$".

→ During delivery stroke, the pressure head in the cylinder is constant and equal to delivery head ($h_d$), which is above the atmospheric head by height of ($h_d$)., the pressure head during delivery stroke is represented by a horizontal line CD which is above the line EF by a height of $h_d$. One complete revolution of the crank, the pressure head in the cylinder is diagram A-B-C-D-A.

This diagram is known as "Ideal indicator diagram."

Workdone by the pump per second

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

$$= K \times L (h_s + h_d)$$

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

But area of indicator diagram

$$= AB \times BC$$

$$= AB \times (BF + FC)$$

$$= L \times (h_s + h_d)$$

Workdone by pump $\propto$ Area of indicator diagram.

<u>Effect of acceleration in suction and delivery pipes on Indicator diagram:-</u>

The pressure head due to acceleration in the suction pipe is

$$h_{as} = \frac{l_s}{g} \times \frac{A}{a_s} \omega^2 r \cos\theta$$

When $\theta = 0^\circ$, $\cos\theta = 1$, and $h_{as} = \frac{l_s}{g} \times \frac{A}{a_s} \omega^2 r$

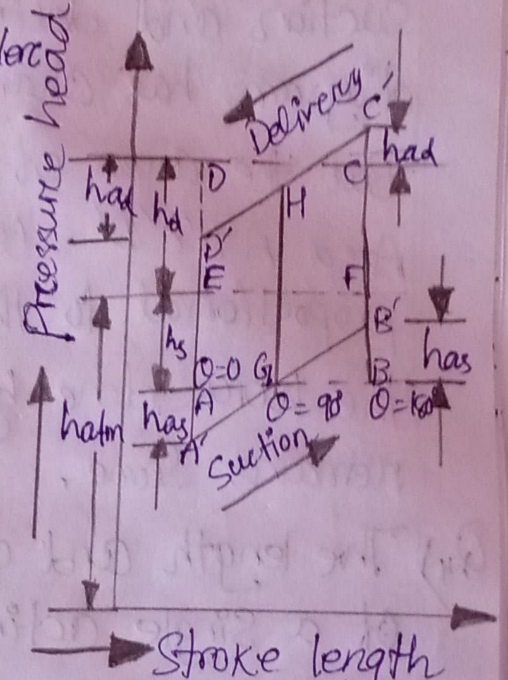
When $\theta = 90^\circ$, $\cos\theta = 0$, and $h_{as} = 0$

When $\theta = 180^\circ$, $\cos\theta = -1$, and $h_{as} = -\frac{l_s}{g} \times \frac{A}{a_s} \times \omega^2 r$

The pressure head inside the cylinder during

Suction stroke will not be equal to "$h_s$". but it will be equal to the sum of "$h_s$" and "$h_{as}$". When $\theta = 0°$, '$h_{as}$' is +ve and the pressure head in the cylinder will be ($h_s + h_{as}$) below the atmospheric pressure head.

At the middle of suction stroke $\theta = 90°$ and $h_{as} = 0$ pressure head in the cylinder will be $h_s$ below the atmospheric pressure head. At the end of suction stroke, $\theta = 180°$ and $h_{as}$ is -ve and the pressure head in the cylinder will be ($h_s$ - $h_{as}$) below the atmospheric pressure head.




For suction stroke, the indicator diagram will be $A'GB'$.

(fig of Effect of acceleration on indicator diagram).

Also the area of $A'AG$ = Area of $BGB'$.

At the begining of delivery stroke, $h_{ad}$ is +ve and the pressure head in the cylinder will be ($h_d + h_{ad}$) above the atmospheric pressure head. At the middle of the delivery stroke, $h_{ad} = 0$ and pressure head in the cylinder is equal to $h_d$ above the atmospheric pressure head.

At the end of delivery stroke, $h_{ad}$ is -ve and pressure in the cylinder will be $(h_d - h_{ad})$ above the atmospheric pressure head. And the indicator diagram for delivery stroke is by the line $C'HD'$. Also, the area of $CC'H$ = Area of $DD'H$

→ It is now clear that due to acceleration in suction and delivery pipe, the indicator diagram has changed from ABCD to A'B'C'D'. But the area of indicator diagram ABCD = Area A'B'C'D'. Workdone, by pump is proportional to the area of indicator diagram the work done by the pump on the water remains same.

Q4) The length and diameter of a suction pipe of a single acting reciprocating pump are 5m and 10 cm respectively. The pump has a plunger of diameter 15cm and a stroke length of 35 cm. The centre of the pump is 3m above the water surface in the pump. The atmospheric pressure head is 10.3m of water and pump is running at 35 r.p.m. Determine:

i) pressure head due to acceleration at the beginning of the suction stroke,

ii) maximum pressure head due to acceleration, and

iii) pressure head in the cylinder at the beginning and at the end of the stroke.

Solution :-

Length of Suction pipe, $l_s = 5m$

Diametere of Suction pipe, $d_s = 10cm = 0.1m$

$\therefore$ Area, $a_s = \pi/4 \times d_s^2$

$= \pi/4 \times 0.1^2 = 0.007854\ m^2$

Stroke length, $L = 35\ cm = 0.35m$

$\therefore$ Crank radius, $r = \frac{L}{2} = \frac{0.35}{2} = 0.175m$

Suction head, $h_s = 3m$

Atmospheric pressure head, $H_{atm} = 10.3$ m of watere.

Speed of pump, $N = 35$ r.p.m.

Angular speed of the crank, is

$$\omega = \frac{2\pi N}{60} = \frac{2\times\pi\times 35}{60} = 3.665 \text{ rad/s.}$$

i) The pressure head due to acceleration in the Suction pipe is

$$h_{as} = \frac{l_s}{g} \times \frac{A}{a_s} \times \omega^2 r \cos\theta$$

$\theta = 0$ and $\cos\theta = 1$

$$\therefore h_{as} = \frac{l_s}{g} \times \frac{A}{a_s} \times \omega^2 r = \frac{5}{9.81} \times \frac{0.01767}{0.007854} \times 3.665^2 \times 0.175$$

$$\therefore h_{as} = 2.695\ m$$

ii) maximum pressure head due to acceleratio in Suction pipe is

$$(h_{as})_{max} = \frac{l_s}{g} \times \frac{A}{a_s} \times \omega^2 r = 2.695m.$$

iii) Pressure head in the cylinder at the beginning of Suction stroke

$= h_s + h_{as} = 3.0 + 2.695 = 5.695$

This pressure head in the cylinder is below the atmospheric pressure head.

∴ Absoulte pressure head in the cylinder at the beginning of suction stroke

= Atmospheric Pressure head - 5.695

$= 10.3 - 5.695$

$= 4.605$ m of water (abs.)

Pressure head in the cylinder at the end of Suction Stroke

$= h_s - h_{as} = 3.0 - 2.695 = 0.305$ m below atmospheric pressure head

$= 10.3 - 0.305 = 9.995$ m of water (abs.)

* Maximum Speed of a reciprocating pump :-

→ Maximum Speed of a reciprocating Pump is determined from the pressure, in the cylinder during Suction and delivery stroke, Should not fall below the vapour pressure of the liquid, flowing through suction and delivery pipe.

→ If the pressure in the cylinder is below the vapour pressure, the dissolved gases will be liberated from the liquid and Cavitation will take place.

→ The Countionus flow of liquid will not exist which Means Separation of liquid will take place.

→ The pressure at which Separation takes place is known as Separation pressure and the head Corresponding to Separation pressure is called Separation pressure head.

It is denoted by "$h_{sep}$".

*) For water, the limiting value of separation pressure head ($h_{sep}$) is 7.8 below atmospheric pressure head or $10.3 - 7.8 = 2.5$ m abs.

The Separation may take place during the Suction stroke or during delivery stroke. The maximum Speed of the reciprocating pump during suction and delivery strokes is

a) Maximum Speed during suction stroke :-

→ From the indicator diagram, it is clear that the absolute pressure head during Suction stroke is minimum at the beginning of the stroke. the Seperation Can take place at the beginning the stroke only.

→ the abs. pressure head in the cylinder at the beginning of the stroke will be equal to Separation pressure head ($h_{sep}$).

$$\therefore h_{sep} = H_{atm} - (h_s + h_{as}) \text{(abs.)}$$

$$h_{as} = h_{atm} - h_s - h_{sep} \quad \text{—— (i)}$$

the values of $h_{sep}$ and $h_s$ (Suction head) hence '$h_{as}$' the pressure head due to acceleration at the beginning of suction stroke Can be obtained.

The value of 'has' is ——— (ii)

$$h_{as} = \frac{l_s}{g} \times \frac{A}{a_s} \times \omega^2 r$$

equating the two values of has

$$h_{atm} - h_s - h_{sep} = \frac{l_s}{g} \times \frac{A}{a_s} \times \omega^2 r \quad \text{——— (iii)}$$

From equation - (iii), the values of the ω can be obtained.

→ This speed is the maximum speed of the reciprocating pump without separation during suction stroke.

b) **Maximum speed during delivery stroke:-**

During delivery stroke, the probability of separation is only at the end of the delivery stroke. The pressure head in the cylinder at the end of the delivery stroke

$$= (H_{atm} + H_d) - h_{ad} \text{ m (abs.)}$$

If separation is to be avoided, the above pressure head should be more than the separation pressure head ($h_{sep}$).

$$h_{sep} = (H_{atm} + h_d) - h_{ad}$$

$$h_{ad} = (H_{atm} + h_d) - h_{sep}$$

But '$h_{ad}$' the pressure head due to acceleration at the end of the delivery stroke is

$$h_{ad} = \frac{l_d}{g} \times \frac{A}{a_d} \times \omega^2 \times r$$

equating the two values of $h_{ad}$,

$$(H_{atm} + h_d) - h_{sep} = \frac{l_d}{g} \times \frac{A}{a_d} \times \omega^2 r \quad \text{--- (iv)}$$

From the above equation (iv), the value of $\omega$ and speed N, can be calculated.

This is the maximum speed of the reciprocating pump without separation during delivery stroke only.

The minimum of the two speeds (a) and (b) is the maximum speed of the reciprocating pump without separation during suction & delivery strokes.

Q5) Find the maximum speed of a single acting reciprocating pump to avoid separation which occurs at 3.0m of water (abs.) The pump has a cylinder of diameter 10 cm and a stroke length of 20 cm. The pump draws water from a sump and delivers to a tank. The water level in the sump is 3.5m below the pump axis and in the tank the water is 13m above the pump axis. The diameter and length of the suction pipe are 4 cm & 5m while of delivery pipe the diameter and length are 3 cm, 20m respectively. Take atmospheric pressure head = 10.3 of water.

Given data :-

Separation pressure head, $h_{sep}$ = 3.0 m of water (abs)

Dia. of cylinder, D = 10 cm
= 0.10 m

stroke length, L = 20 cm
= 0.20 m

$\therefore$ Crank radius, $r = \frac{L}{2} = \frac{0.20}{2} = 0.10$ m

Suction head, $h_s$ = 3.5 m

Delivery head, $h_d$ = 13 m

Dia. of suction pipe, $d_s$ = 4 cm = 0.04 m

Length of suction pipe, $l_s$ = 5 m

Dia. of delivery pipe, $d_d$ = 3 cm = 0.03 m

Length of delivery pipe, $l_d$ = 20 m

Atmos. pressure head, $H_{atm}$ = 10.3 m

a) Maximum speed during suction stroke without separation is from the relation,

$$H_{atm} - h_s - h_{sep} = \frac{l_s}{g} \times \frac{A}{a_s} \times \omega^2 r$$

$$10.3 - 3.5 - 3.0 = \frac{5}{9.81} \times \frac{\pi/4 \times D^2}{\pi/4 \times d_s^2} \times \omega^2 \times 0.10$$

$$3.8 = 0.5096 \times \frac{\pi/4 \times (0.10)^2}{\pi/4 \times (0.04)^2} \times \omega^2 \times 0.10$$

$$3.8 = 0.5096 \times \frac{0.07853}{0.01256} \times \omega^2 \times 0.10$$

$$3.8 = 0.5096 \times 6.25 \times \omega^2 \times 0.10$$

$$= 0.3185 \times \omega^2$$

$$\Rightarrow \omega^2 = \frac{3.8}{0.3185}$$

$$\Rightarrow \omega^2 = 11.930$$

$$\Rightarrow w = \sqrt{11.930} = 3.454$$

$$\therefore \omega = \frac{2\pi N}{60} = 2.994$$

$$N = \frac{60 \times 2.994}{2\pi} = 28.59 \text{ r.p.m.}$$

$\therefore$ maximum speed without separation = 28.59 r.p.m. (Ans)

# Chaptere- 5 Hydraulic Control System :-

*) What is a hydraulic System ?

→ A hydraulic System is a drive technology where a fluid is used to move the Energy from e.g. an electric motore to an actuatore, Such as a hydraulic Cylindere.

→ The fluid is theoretically uncompressible and the fluid path Can be flexible in the Same way as an electric Cable.

Its merit and demerits :-

Merit → Powere, Accuracy, efficiency and ease of maintenance.

Demerits → Leakage problem.

*) Hydraulic Accumulatore :-

→ The hydraulic accumulatore is a device used fore Storing the Energy of a liquid in the form of pressure Energy, which may be Supplied fore any Sudden ore interemittent requirement.

→ In Case of hydraulic lift ore the hydraulic crane, a large amount of Energy is required when lift ore Crane is moving upward. This Energy is Supplied from hydraulic accumulatore.

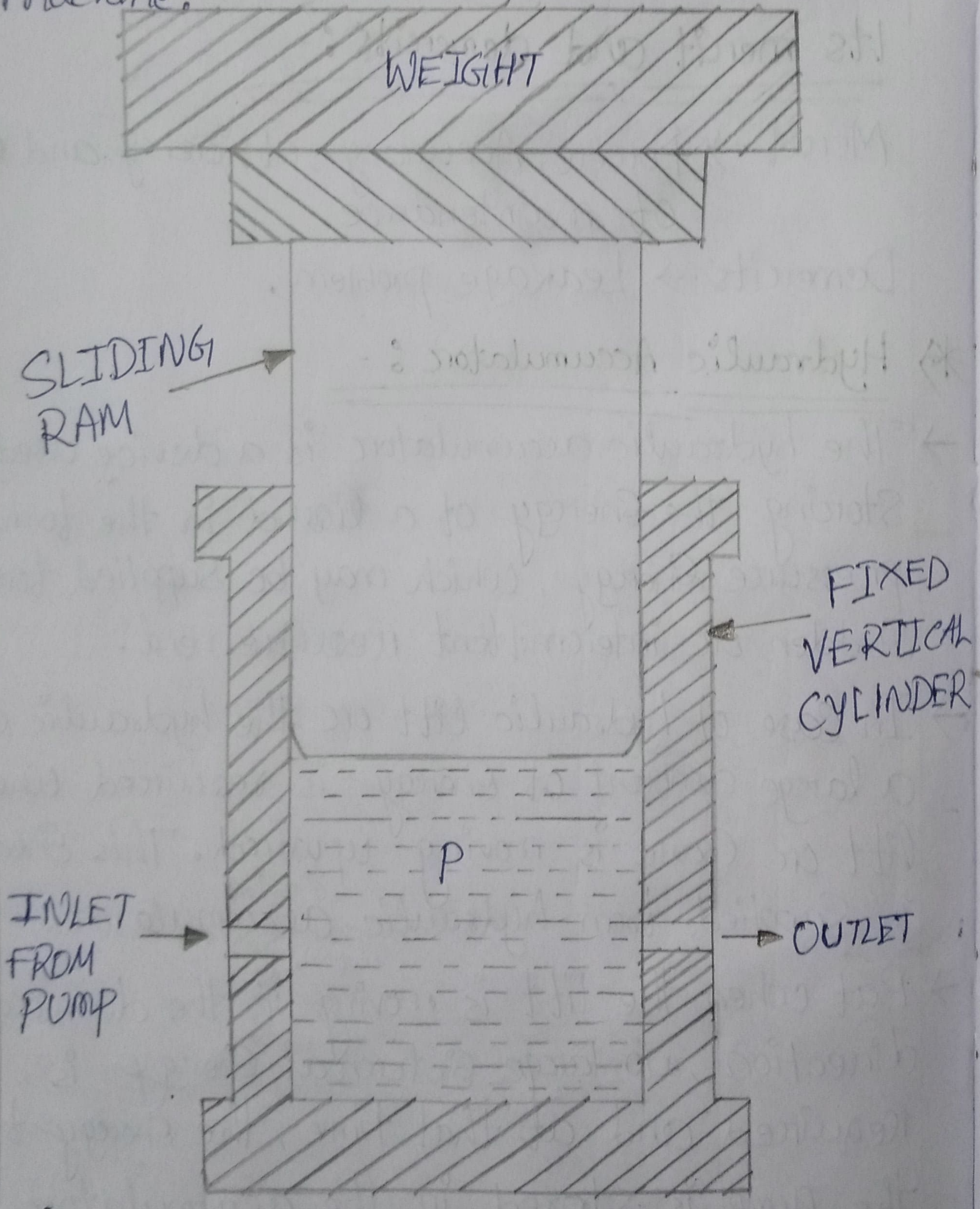
→ But when the lift is moving in the downward direction, no large External Energy is required and at that time, the Energy from the pump is stored in the accumulatore.

→ a hydraulic accumulator which consists of a fixed vertical cylinder containing a sliding ram.

→ A heavy weight is placed on the ram.

→ The inlet of the cylinder is connected to the pump. which continuously supplies water under pressure to the cylinder.

→ The outlet of the cylinder is connected to the machine.




( The hydraulic accumulator )

→ The ram is at the lowermost position in the beginning.

→ The pump supplies water under pressure continuously. If the water under pressure is not required by the machine, the water under pressure will be stored in the cylinder.

→ This will raise the ram on which a heavy weight is placed. When the ram is at the uppermost position, the cylinder is full of water and accumulator has stored the maximum amount of pressure energy.

→ When the machine requires a large amount of energy, the hydraulic accumulator will supply this energy and ram will move in the downward direction.

## Capacity of hydraulic Accumulator :-

It is defined as the maximum amount of hydraulic energy stored in the accumulator.

Let, $A$ = Area of sliding ram,

$L$ = Stroke or lift of the ram,

$p$ = Intensity of water pressure supplied by the pump,

$W$ = Weight placed on the ram

$W$ = Intensity of pressure × Area of ram

$$W = p \times A$$

The workdone in lifting the ram

$= W \times$ Lift of ram

$= WL$

$= P \times A \times L$

The workdone in lifting the ram is also the Energy Stored in the accumulator. And Energy Stored is equal to the Capacity of the accumulator.

$\therefore$ Capacity of accumulator

= Workdone in lifting the ram

$= P \times A \times L$

But, $A \times L$ = Volume of accumulator

$\therefore$ Capacity of accumulator

$= P \times$ Volume of accumulator

→ Pressure Control Valves :-

i) Controlling the rate of flow

ii) Controlling the pressure level

Types of pressure Control Valves :-

i) Relief valves

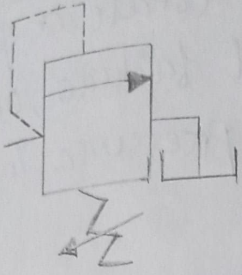
ii) unloading Valve

iii) Sequence Valve

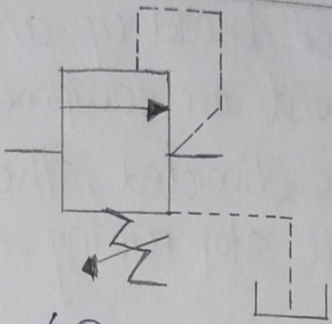
iv) Reducing valve

v) Counter balance Valve

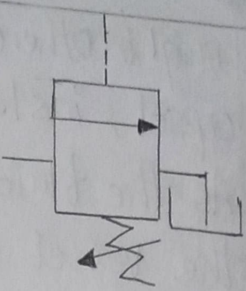
vi) brake Valve



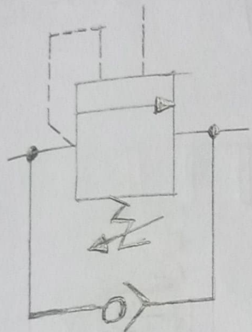
(Relief valve)



(Reducing valve)

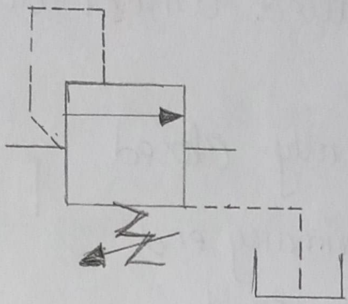


(unloading valve)

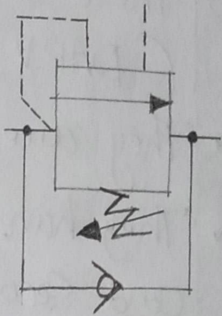


(Counter balance valve)

(may be interenally or Exterenally piloted)



(Sequence Valve)



(Brake Valve)

( ANSI Symbols )

## [ Pressure Control Valves ]

→ Pressure Control valves are used in hydraulic Systems to Control actuators force and to determine and Select pressure levels at Which Certain machine operations must occurs.

Actuator Force = Pressure(p) × Area (a)

**Relief Valve :-**

→ A relief valve or pressure relief valve is a type of Safety valve used to Control or limit the pressure in a System ; pressure might otherewise build up and create a process upset, instrument ore equipment failure, ore fire. As the fluid is directed, the pressure inside the vessel will stop rising.

Simple Type :-

→ These are called direct acting relief valves (DARV).

→ They are normally closed

→ They have a primary end and secondary port.

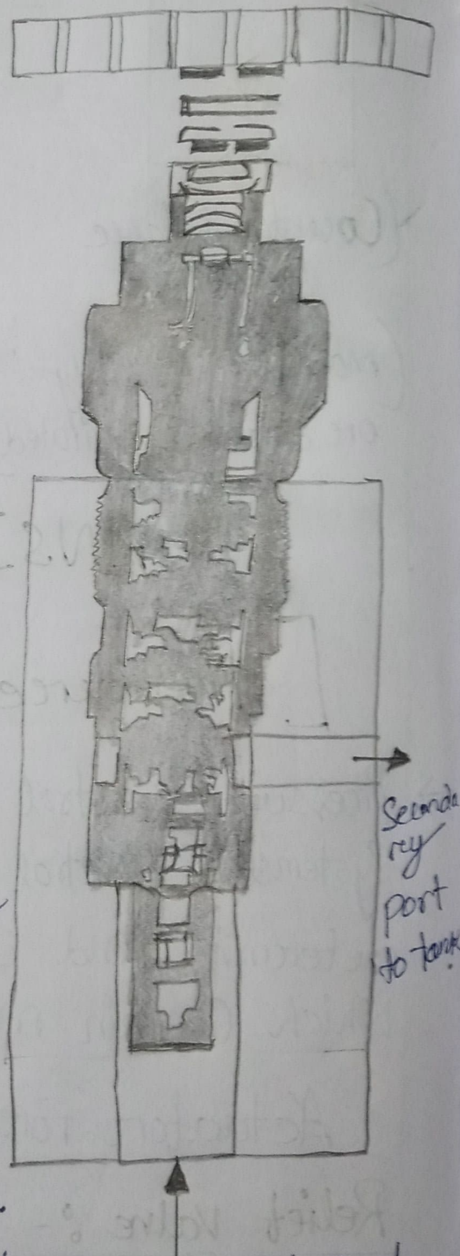
→ The primary port is exposed to the system pressure and is blocked by the poppet.

→ The secondary port is connected to the line that connects the tank ore reservoire.

→ The poppet lies exposed to in the intersectional passage between the primary & secondary ports.

→ The adjustable knob on the top of the valve can vary the compression in the spring.

→ Direct acting pressure relief valves are the one in which the poppet is closed by




(Direct acting relief valve)

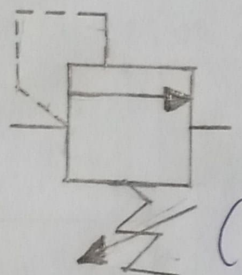
the direct action of the mechanical Spring force that oppose the excess fluid pressure.

→ When the system pressure reaches the spring force, it opens and allows half the fluid to pass to the tank, which is term as "Cracking Pressure".

→ The cracking pressure can be adjusted by increasing the spring force.

→ The application of the direct acting relief valves are limited due to the difficulty in designing strong compression spring that can be used for long period of time.

ANSI Symbol



(Relief valve)

## Pilot operated Relief valve :- (Compound)

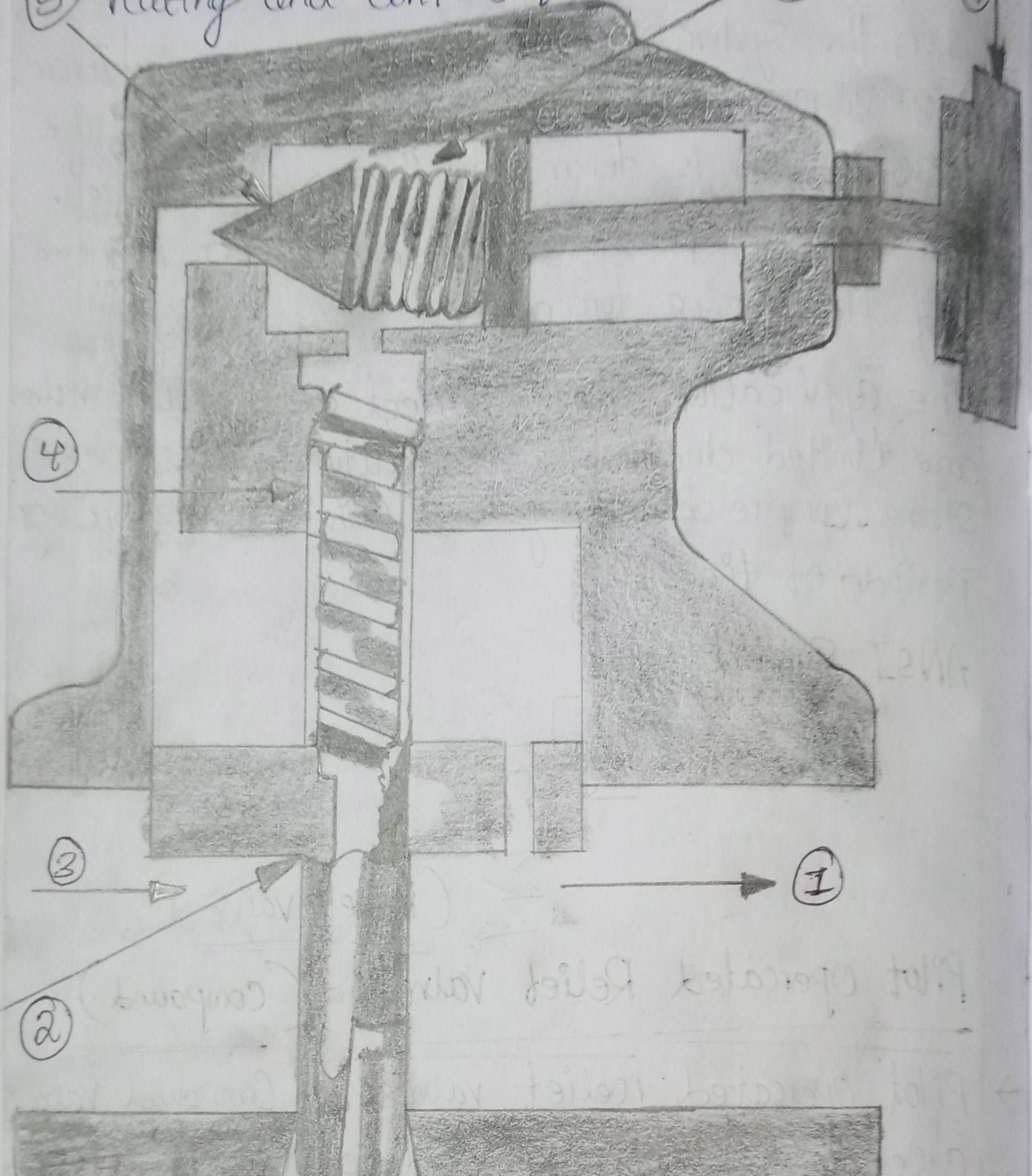
→ Pilot operated relief valves are compound valves are designed to accommodate high pressure and high flow capacity with same frame size as the direct acting valves.

→ These valves are bulit in two stages.

→ The first stage is the same as direct acting relief valve and controls pressure. They have a poppet, spring and knob that can adjust the pressure level.

→ The second stage has a main spool that is held normally closed by light non-adjustable spring.

They arce large enough to handle maximum flow reating and Controle flow.




( Pilot Opercated Relief Valve )

<1> Flow leaves, <2> Main Spool, <3> Flow enteres, <4> Non adiustable spreing, <5> Poppet <6> Adiustable Spreing, <7> Adjustable KNOB, <8> Orifice to tank

## Pressure regulation valves :-

→ Pressure regulation valves are called as "pressure reducing valve".

→ Pressure reducing valve is normally open pressure control valve and are used to limit pressure in this valve the primary port is connected to the secondary port.

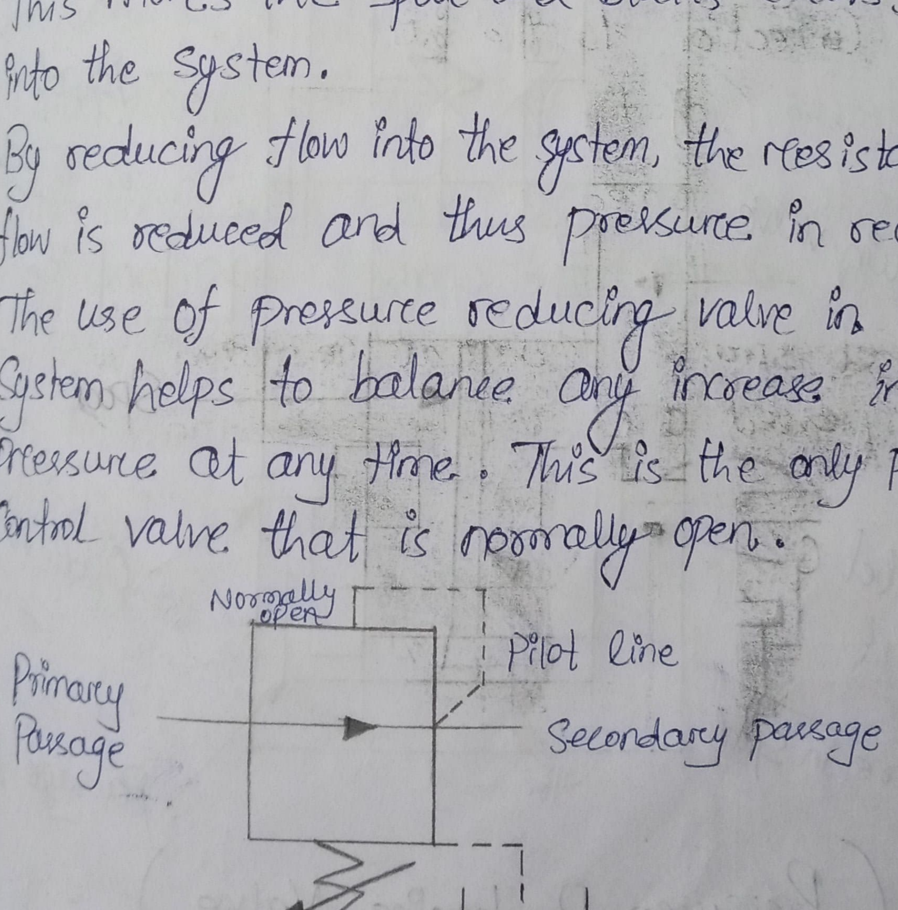
→ A pilot line is taken from the bottom of the spool.

→ When there is an increase in pressure due to resistance to flow in the line the flow is direted through the pilot line.

→ This moves the spool and blocks excess flow into the system.

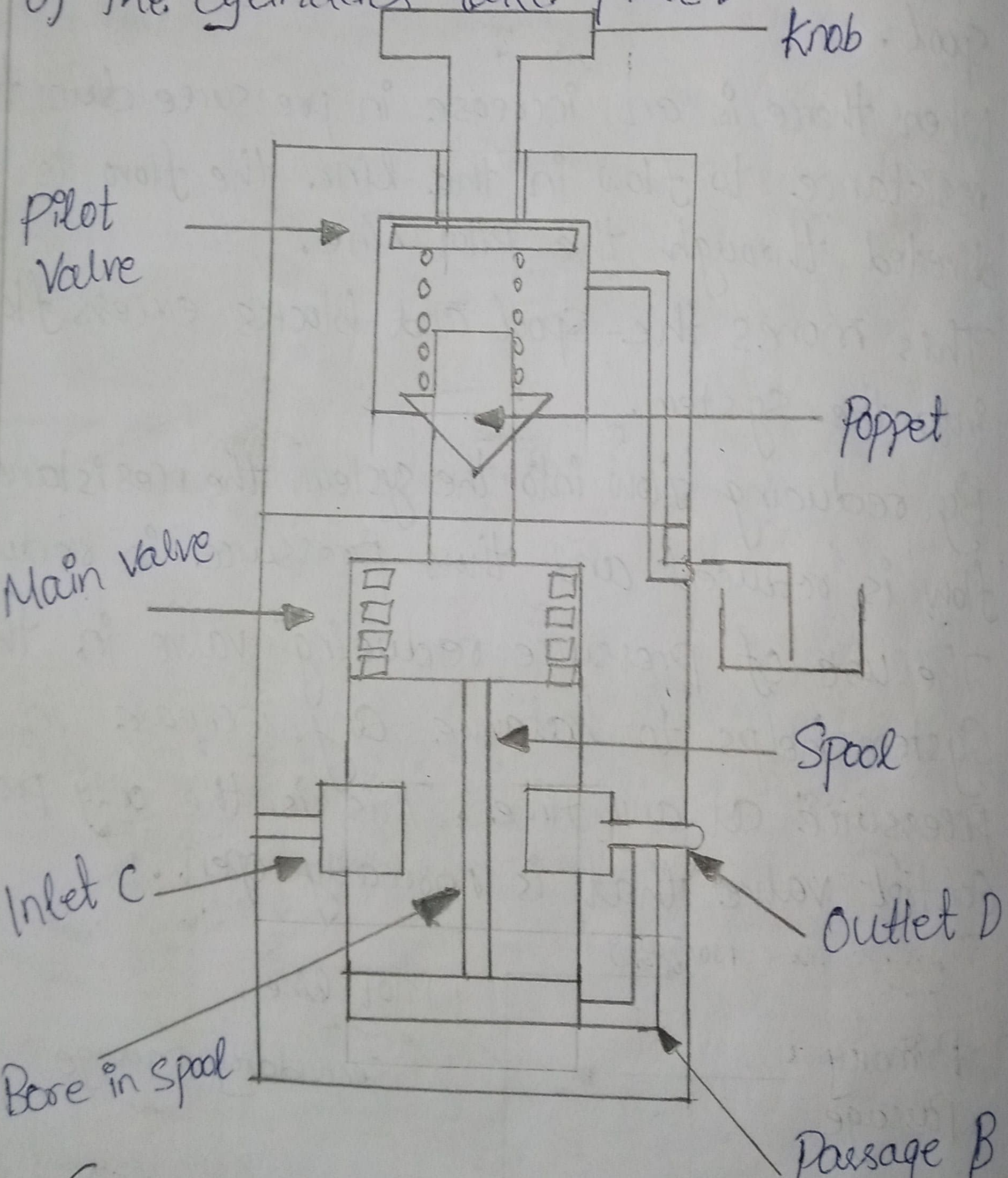
→ By reducing flow into the system, the resistance to flow is reduced and thus pressure in reduced.

→ The use of pressure reducing valve in the system helps to balance any increase in pressure at any time. This is the only pressure control valve that is normally open.




(Reducing Valve)

* When there are two clamping cylinderes used with relatively difference in the requirement of clamping forcees, the actuator that requires less pressure closes as end when the clamping Cylindere Continues and when both the cylinders reach theire pressure rating, the flow is reversed by changing the position of the direction Control valve and retraction of the cylinderes take place.




(Pressure Reducing Valve)

* These valves limit pressure on a branch circuit to a lesser amount than required in a main circuit.

→ For example, in a system, a branch-circuit pressure is limited to 300 Psi, but a main circuit must operate at 800 Psi. A relief valve in a main circuit is adjusted to a setting above 800 Psi to meet a main circuit's requirements.

→ However, it would surpass a branch-circuit pressure of 300 Psi. Therefore, besides a relief valve in a main circuit, a pressure-reducing valve must be installed in a branch circuit and set at 300 Psi.

## Direction Control valves :-

Direction Control Valves (DCV) are used to start, stop and and change the direction of flow in a fluid power system.

→ In hydraulic control system, the DCV are classified in 3 types :-

1) 3/2 DCV,
2) 5/2 DCV,
3) 5/3 DCV,

## *) 3/2-way valve :-

→ The 3/2-way valve is a signal-generating valve, with the characteristic that a signal on the output side of the valve can be generated and also cancelled.

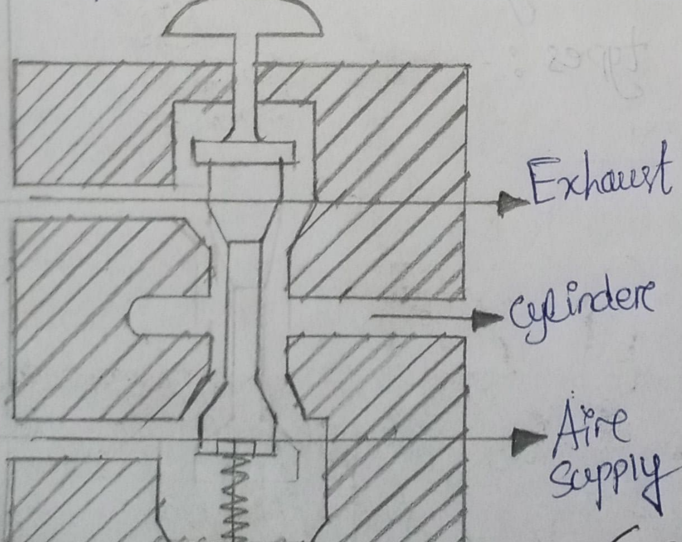
→ The 3/2-way valve has three ports and two positions. the addition of the exhaust port enables the signal generated via the passage through the 3/2-way valve to be cancelled.

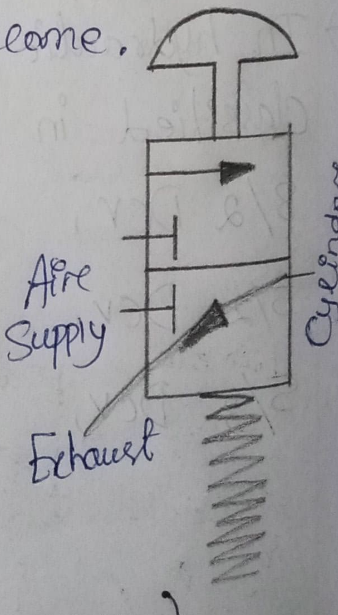
→ The valve connects the output signal to exhaust and thus to atmosphere in the initial position.

→ A spring forces a ball against the valve seat preventing the compressed air from flowing the air connection to the working line.

→ Actuation of the valve plunger causes the ball to be forced away from the seat.

→ In doing this, the opposing force of the reset spring and that generated from the compressed air must be overcome.



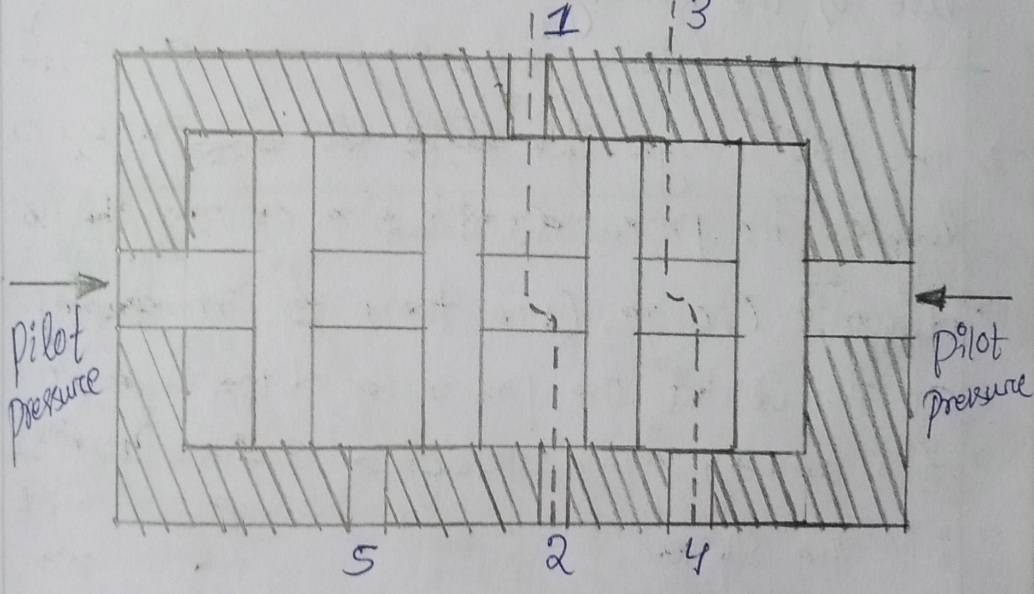



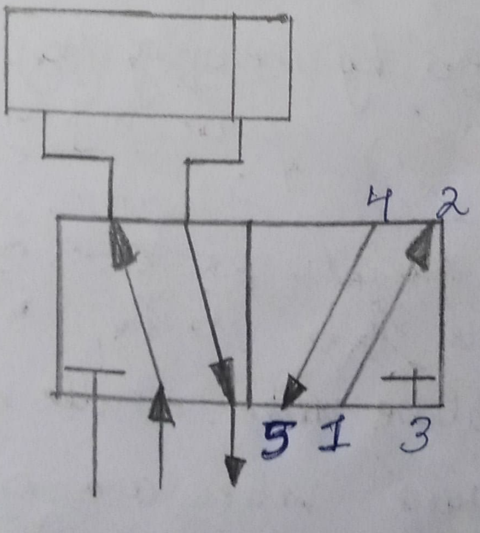

(3/2 way)

## * 5/2 - way valve :-

→ The 5/2 way valve, has five ports and two opposition.

→ The 5/2 way valve is used primary as a Control element fore the Control of cylinderes. An Example of the 5/2 way valve, the longitudinal Slide valve, uses a pilot spool as a Control Component. This Connects or Separates the Corresponding lines by means of longitudinal movements. The required actuating force is lower because there are no opposing forces due to Compressed air or Spring.



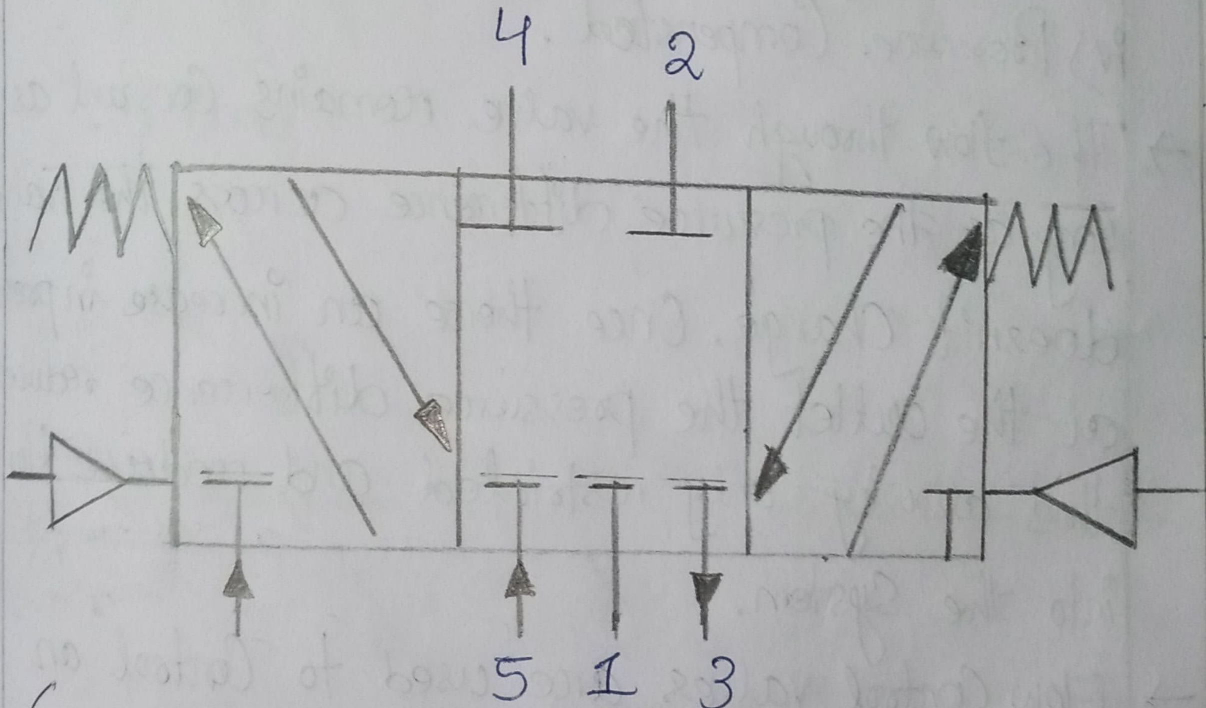
- All forms of actuation can be used with longitudinal slide valves, i.e. manual, mechanical, electrical or pneumatic.
- These types of actuation can also be used resetting the valve to its starting position.
- The actuation travel is considerable larger than with seat valves.
- Sealing presents a problem in this type of slide valve.
- The type of fit known in hydraulics as metal to metal reaires the spool to fit precisely in the bore of the housing.

## * 5/3 - way Valve :-

→ The 5/3 valve has five working ports and three switching positions.

→ With these valves, double acting cylinders can be stopped with in the stroke range. This means a cylinder piston under pressure in mid-position is briefly clamped in the normally closed position and in the open position can be moved un-pressurised.

→ If no signals are applied at either of the two control ports, the valve remains spring-centered in mid position.




(5/3 way Double pilot valve - normally closed)

## *) Flow Control Valves :-

→ Flow Control valves are used to reduce the rate of flow.

→ The reduction in rate of flow results in reduction of speed and increase in pressure.

→ This results in increase in pressure and temperature of the system.

→ The flow control valves are classified as

i) Fixed or Non-adjustable,

ii) Fixed or Adjustable,

iii) Throttling,

iv) Pressure Compensted.

→ The flow through the valve remains Constant as long as the pressure difference across the valve doesn't Change. Once there an increase in pressure at the outlet the pressure difference reduces, thus allowing only restricted and reduce flow into the System.

→ Flow Control valves are used to Control an actuator's Speed by metering flow.

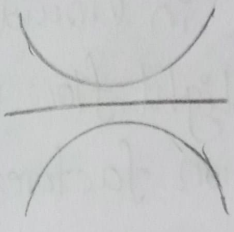
→ Metering is measuring or regulating the flow rate to or from an actuator.

## *) Non-pressure Compensated Pressure Control Valves :-

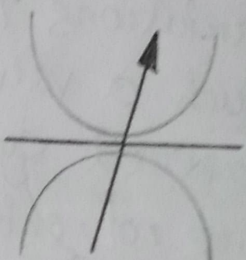
→ Gate valve, Globe valve, Needle valve, Check Valve, Diaphagram valves and Butterfly valves are non-pressure compensated valves which are just used to control or limit the volume of flow entering the system.

They do not deliver fixed volume into the system once there is any change in pressure difference.

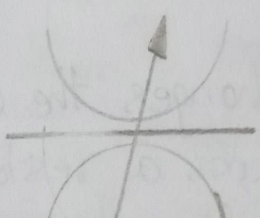
But as most of the system is subjected to pressure variations, the application of non-pressure compensated flow control valves in hydraulic systems in is limited. Thus more emphasis on these valves is not given.



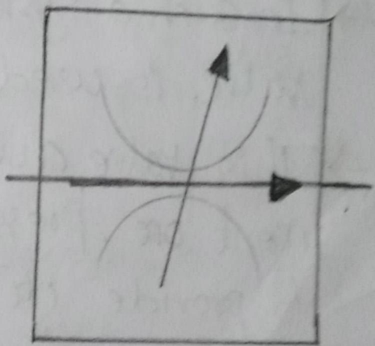
Fixed



Adjustable



Throtting



Pressure Compensated.

<u>Pressure/Temperature</u> <u>Compensated flow control flow</u>:-

The flow-control valves previously discussed do not compensate for changes in fluid temperature or pressure and are considered non-compensating valves.

Flow reate through these valves Can vary at a fixed Setting if eithere the pressure ore the fluid's temperatere changes.

→ Viscosity is the internal reesistance of a fluid that can stop it from flowing.

→ A liquid that flows easily has a high Viscocity.

→ Viscocity changes, which can reesult from temperatere changes, Can Cause low Vareiations through a valve.

→ Such a valve can be used in liquid - powereed systems where slight flow variations are not critical Consideration factores.

→ Howevere, Some systems require extremely accurate control of an actuating device.

→ In such a system, a Compensated flow-Control valve is used.

→ This valve automatically changes the adjustment ore pressure drop across a restriction to provide ore a Constant flow at a given Setting.

→ A valve meteres a Constant flow regardless of variation in system pressere.

→ A Compensated flow-Control valve is used mainly to meter fluid flowing into a circuit; however, it can be used to metere fluid as it leaves a circuit.

For clarity, this manual will refere to this valve as a flow regulatore.

The schematic representation of this valve is included in the appendix.

## Flow Control Methods :-

The Flow can be controlled eithere by one of the following methods.

i) Metere-in circuit (iii) Bleed-off circuit

ii) Metere-out circuit

## 1) Metere-in circuit :-

With this circuit, a flow-control valve is installed in a pressure line that leads to work cylindere.

All flow entering a work cylindere is first metered through a flow-control valve.

Since this metereing action involves reducing flow from a pump to a work cylindere, a pump must delivere more fluid than is required to actuate a cylindere at the desired speed.

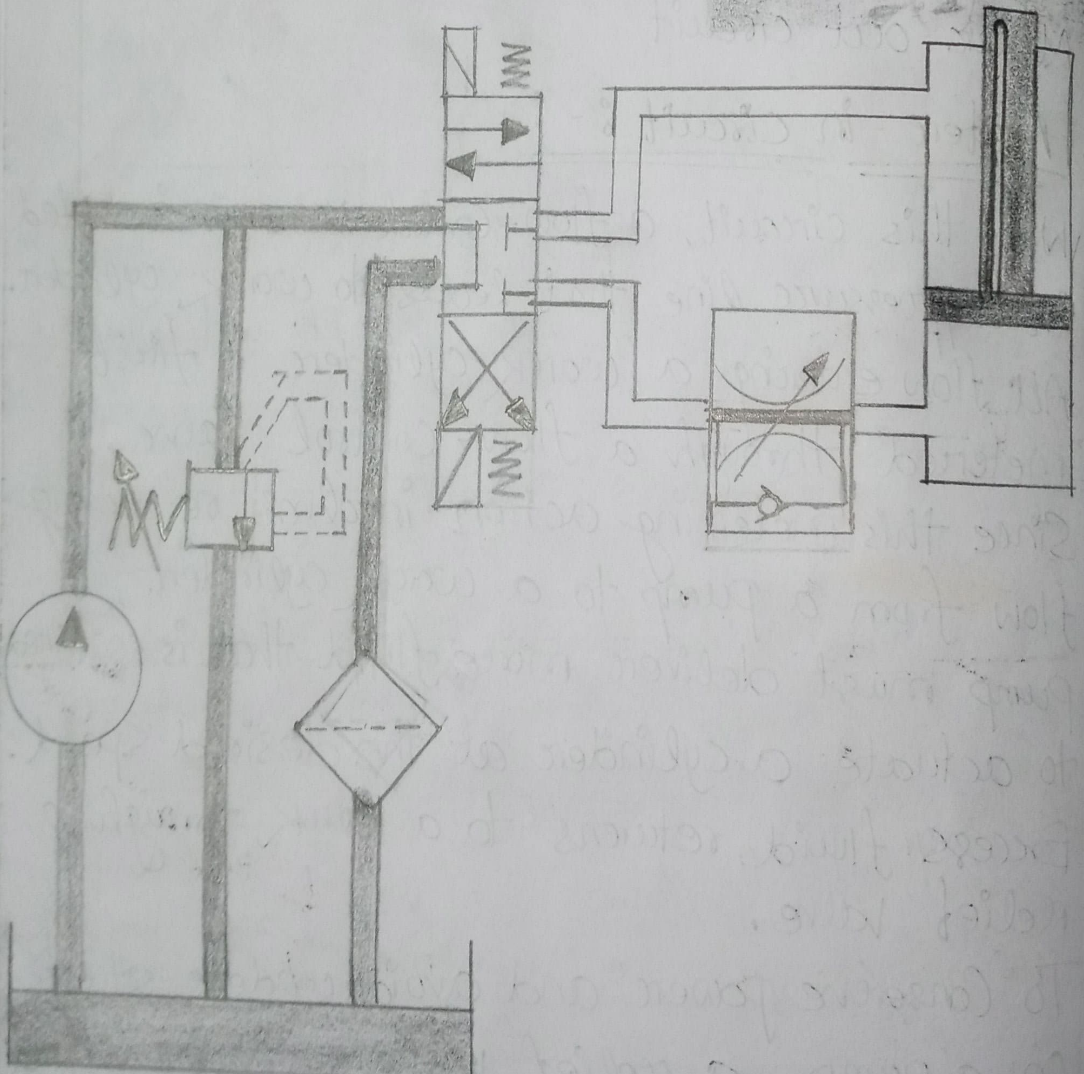
Excess fluid returns to a tank through a relief valve.

To Conserve powere and avoid undere stress on a pump, a relief valve's setting should be only slightly highere than a working Pressure's, which a cylindere requires.

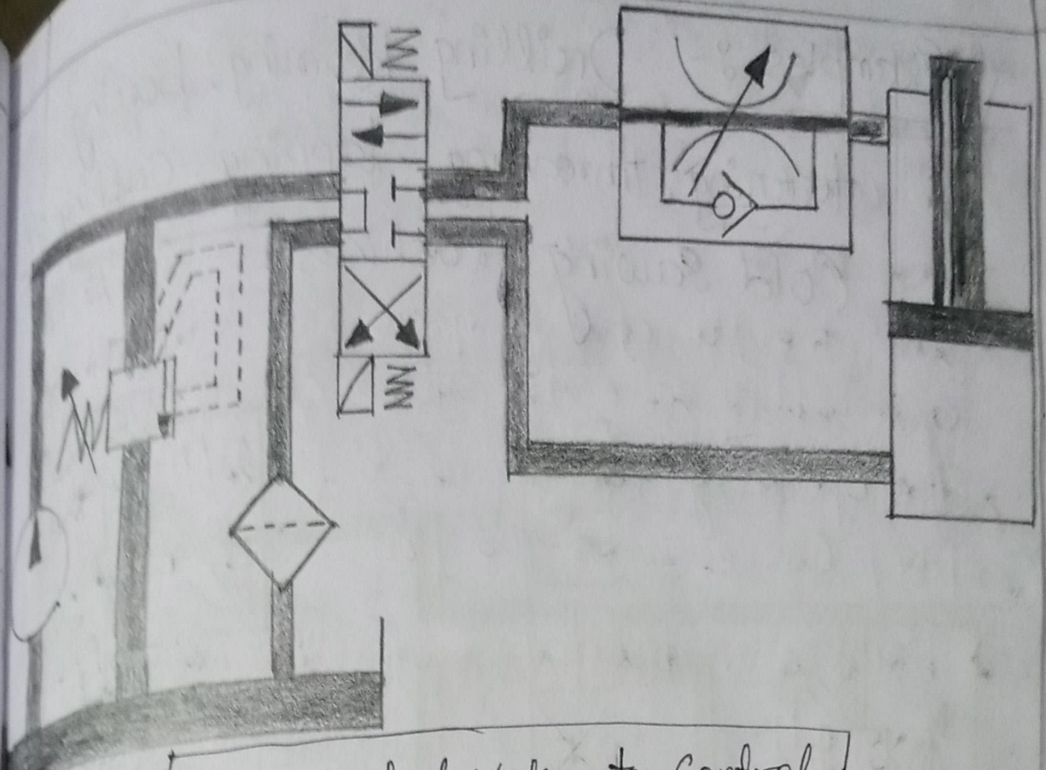
## Applications :-

→ Ametere-in circuit is ideal in applications where a load always offeres a positive resistance to flow dureing a Controlled stroke.

Examples :- Grindere tables,
welding machines,
milling machines,
rotarcy hydraulic motor drives.



Flow Control valve to Control Extension stroke



Flow Control valve to Control Returen stooke

Meter-in Circuits Diagrams

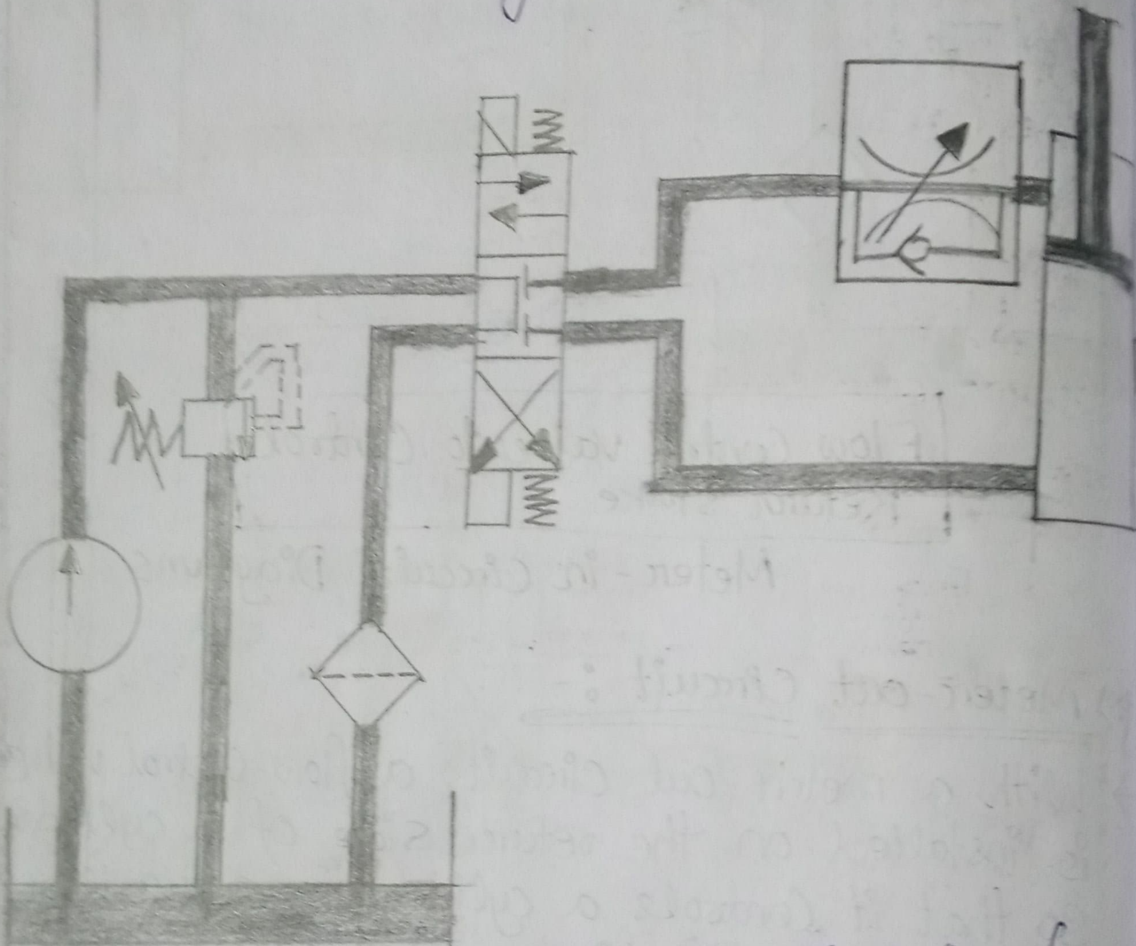
## * Meter-out Circuit :-

→ With a meter-out circuit, a flow-control valve is installed on the returen side of a cylinder so that it Controls a cylindere's actuation by metering its discharge flow. A relief valve is set slightly above the opereating pressuree that is requireed by the type of work..

### Application :-

→ This type of circuit is ideal fore overhauling load applications in which a workload tends to pull an opereating piston faster than a pump's deliveryewould warrant.

Examples :- Drilling, Reaming, boring, turning, threading, tapping, Cutting off Cold sawing machines.



The above circuit meters the oil coming from the rod side.

(Metere-out circuit Diagram)

## Bleed-off circuit :-

- A typical bleed-off circuit is not installed directly in a feed line.
- A valve regulates flow to a cylinder by diverting an adjustable portion of a pump's flow to a tank.
- Since fluid delivered to a work cylinder does not have to pass through a flow-control valve, excess fluid does not have to be dumped through a relief valve.
- This type of circuit usually involves less heat generation because pressure on a pump equals the work resistance during a feed operation.

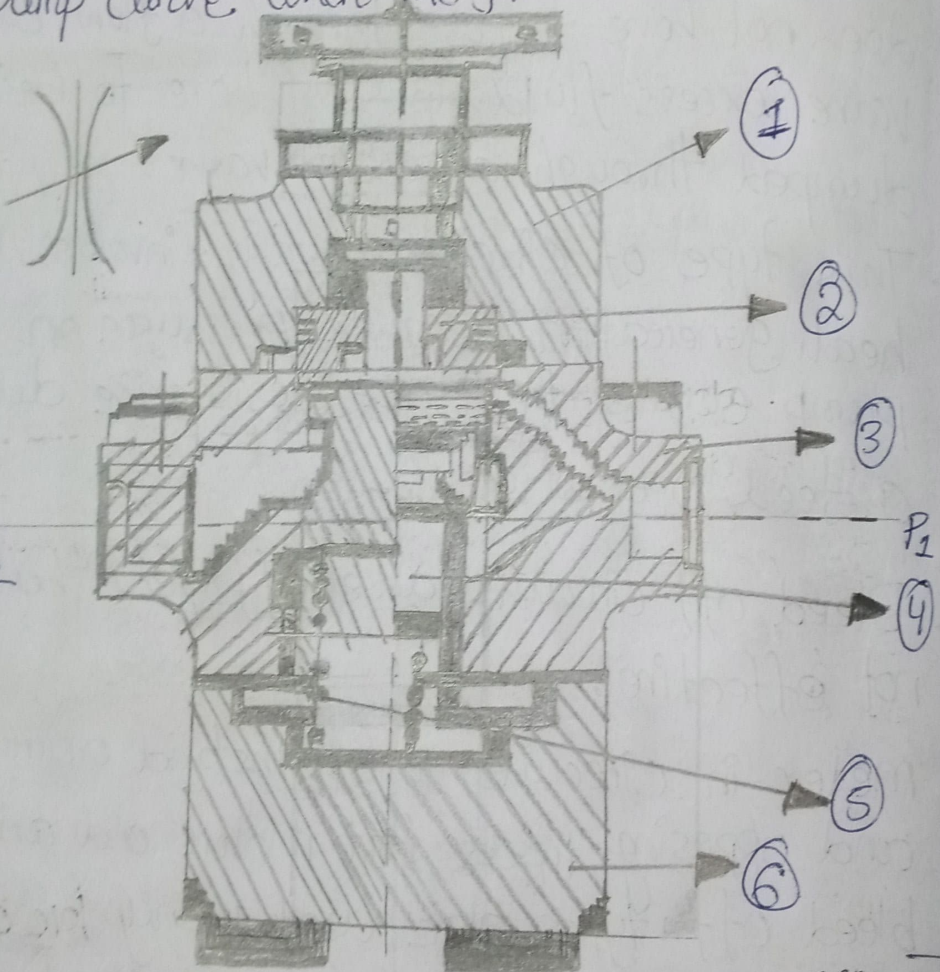
- Bleed off circuits are efficient but not effective.
- metere-in circuits can withstand overrunning and opposing loads but metere-out and bleed off types are useful only for opposing loads. The choice of flow control valve depends on the application.

*) Throttle Valves :-

→ Throttling valves are a type of valve that can be used to start, stop and regulate the flow of fluid through a rotodynamic pump.

→ When the flow of a pump is regulated using a throttling valve, the system curve is changed.

→ The operating point moves to the left on the pump curve when the flow is decreased.




[Throttle Valve with axial V-notch orifice]

1) Top Cover
2) Guide Sleeve
3) Valve body
4) Spool
5) Spring
6) bottom Cover

## * Fluid Powere Pumps :-

→ **Pumps :-**

→ The pumps are the heart of the hydraulic System. Pumps transform the mechanical Energy that receive from the prime mover (electric motor) into fluid Energy.

*) **Pump classification :-**

→ There are two broad classifications of pumps. They are

- Hydrodynamic or non-Positive displacement pumps
- Hydrostatic or positive displacement pumps

*) **Hydrodynamic or non-positive displacement position:**

→ These are low pressure, high volume flow pumps. They are used only for fluid transport and are not used in fluid power industry.

Examples of these pumps are :-

i) Centrifugal pumps (Impeller type)

ii) Axial pumps (propeller pumps)

*) **Hydrostatic or positive displacement pumps :-**

→ The Hydrostatic or positive displacement pumps eject a fixed volume of flow into the hydraulic System per revolution of pump Shaft rotation.

→ Based on the nature of the sliding motion between the relative parts and based on construction these pumps are broadly classified as :-

1) Rotary pumps 2) Reciprocating pumps.

1) Rotary Pumps :-

→ In rotary pumps (Gear pump, Vane pump, Screw pump, Gerotor pump) the drivers are coupled with the prime mover and rotate inside a housing.

→ The driven element (gear, screw, lobe) rotates in the opposing direction. At the inlet they move away from each other creating partial vacuum at the inlet and move towards each other at the inlet creating high pressure to push the liquid into the discharge line.

→ In vane pumps, the vanes move out of their radial slots near the inlet and move in near the discharge port.

2) Reciprocating Pump :-

→ In Reciprocating Pumps (Piston and cylinder arrangement) the piston moves away at the inlet valve resulting in partial vacuum.

→ This pushes the fluid into the cylinder from the reservoir, as the atmospheric pressure is large.

→ When the piston is reversed the valve that opened during suction is closed and this increase in pressure opens the discharge valve and pushes the fluid into the discharge line.

Examples :- Piston Pump
(Radial, Inline, Axial types)

* Gear pumps :-

→ There are two different types of gear pumps.

i) Internal Gear pumps
ii) External Gear pumps

* External Gear pumps :-

→ Mostly external gear pumps are used.
→ They have meshing gears of equal size.
→ The drive gear is coupled with the drive shaft of the electric motor.
→ This gear drives the other gear.
→ As they rotate the fluid is trapped and carried between the teeth of the driver and driven gears and the external casing, which is in close contact with the gears.
→ The pump creates flow and as they pass through the components of the systems, pressure is generated and transmits it to the actuator.

The displacement of the geare pump increases with an increase in input rpm.

**Volumetric Displacement and theoretical flow reate :-**

Theoretically the displaced volume and flow rate can be

$D_o$ is the addendum circle diametore of the geare teeth,

$D_i$ is the base circle diametere of the geare

$W$ the width of geares,

$N$ Speed of revolution of the prime movere,

$Q_T$ theoretical Pump flow reate,

$V_D$ displaced volume of the pump.

Volumetric displacement $= V_D$

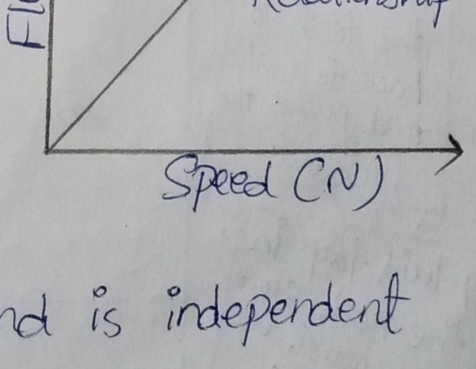
$$= \pi/4 \left( D_o^2 - D_i^2 \right) W \; (mm^3).$$

Theoretical flowreate $Q_T = V_D N \; (mm^3/min)$.

The theoretical equations Show the pump flow depends on

i) The size of geare

ii) The speed of revolution.




The pump flow varies directly with Speed and is independent of other parameters.

## *) Volumetric Efficiency :- ($\eta_v$)

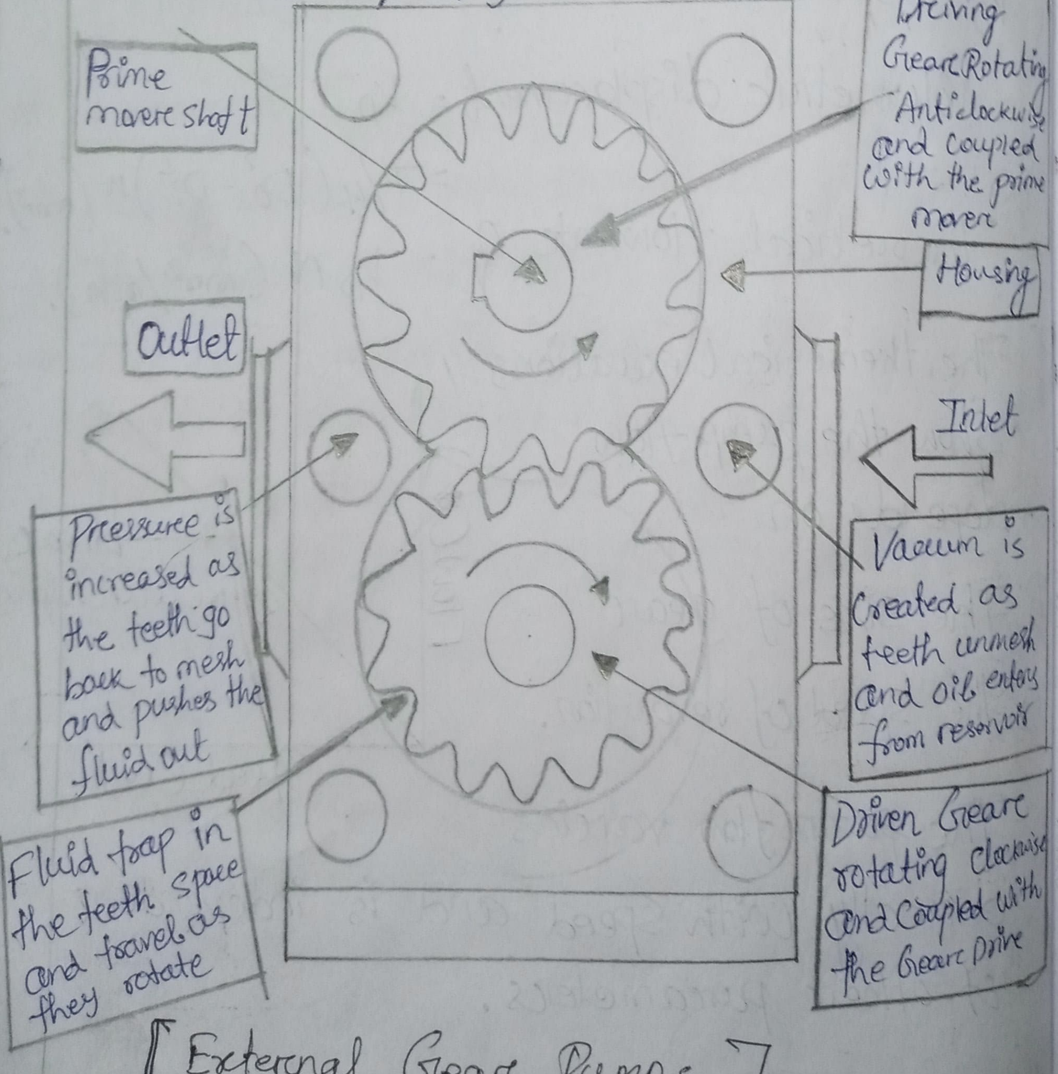
→ There is a small clearance between the tip of the gear and the housing.

→ The ratio of actual flow rate to theoretical flow rate is termed as volumetric efficiency.

→ The efficiency of the positive displacement pump is usually more than 90% at rated pressure.

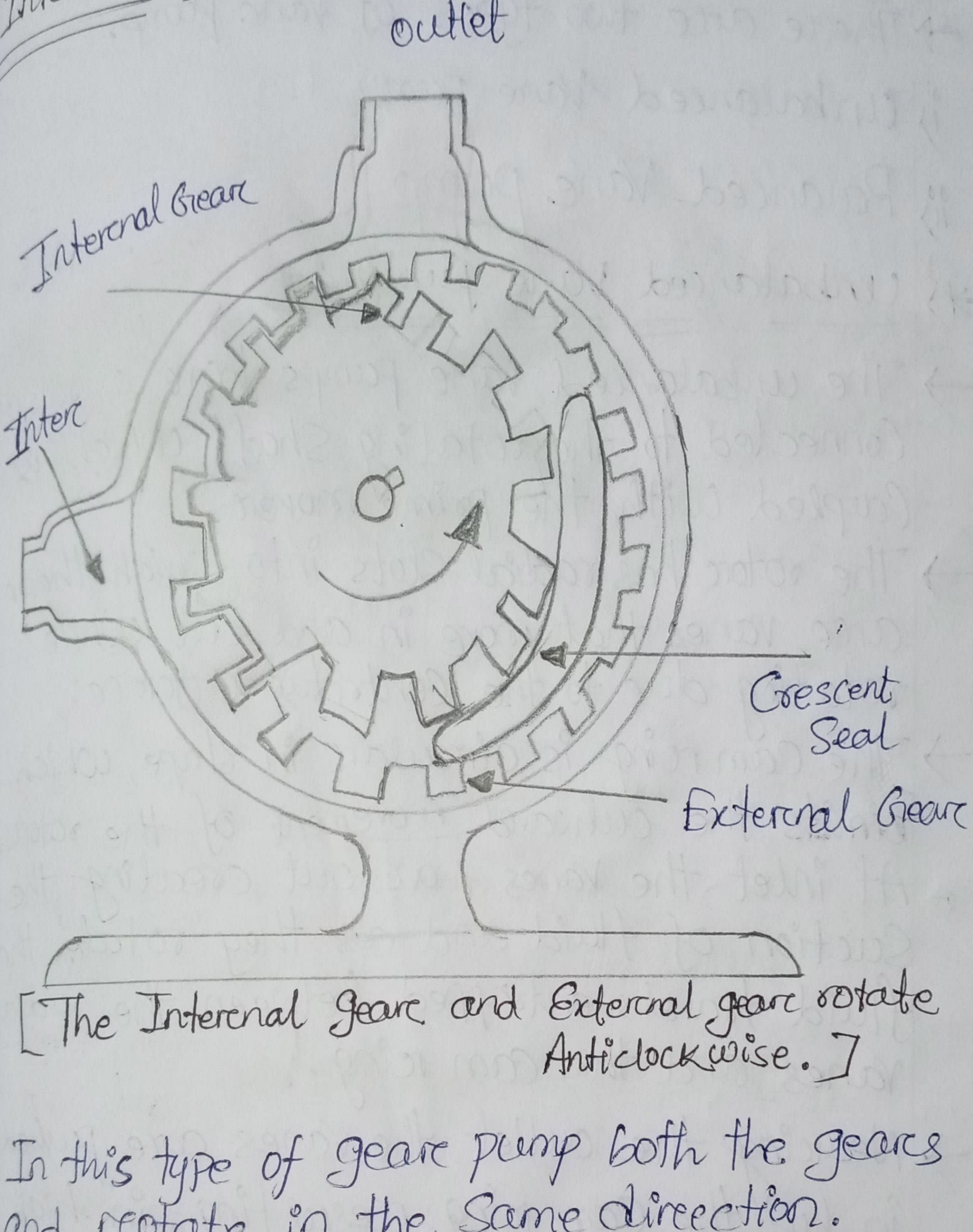
Volumetric efficiency ($\eta_v$)

$$= (Q_A / Q_T) \times 100\%$$




[ External Gear Pumps ]

## *) Intercnal Geare Pumps :-




[The Intercnal geare and Extercnal geare rotate Anticlockwise.]

→ In this type of geare pump both the gears and reotate in the Same direection.
→ As the geares move away neare the inlet, the oil is trapped in the gear space and travels areound the creescent Seal and neare the inlet as the two gears Come closere pressurce reises and the oil is pushed into the outlet poret.

## Vane Pumps :-

→ There are two types of Vane pump.

i) Unbalanced Vane pump

ii) Balanced Vane pump

## Unbalanced Vane pump :-

→ The unbalanced vane pumps have a rotor connected to the rotating shaft which is coupled with the prime mover.

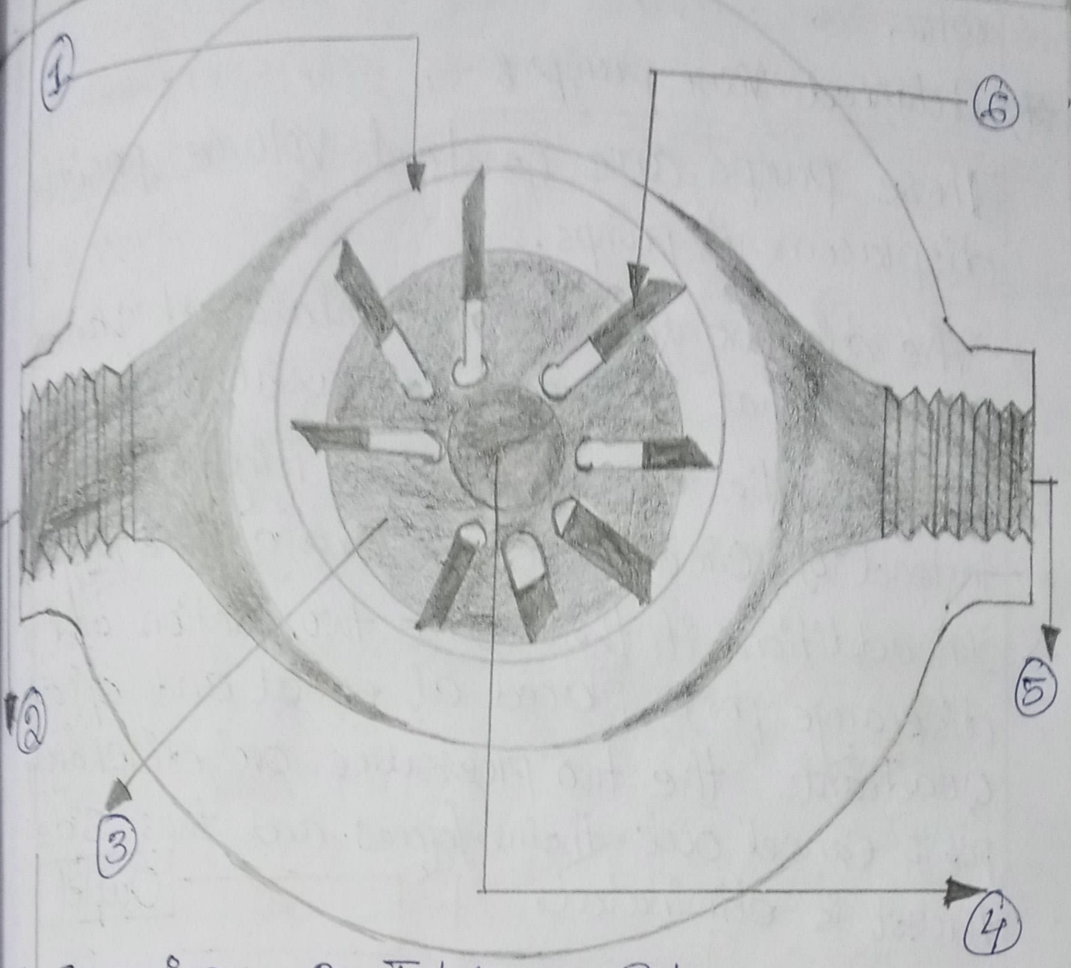
→ The rotor has radial slots into which there are vanes that move in and out while rotating due to the centrifugal force.

→ The cam ring is circular in shape, which limits the outward movement of the rotors. At inlet the vanes move out creating the suction of fluid and as they rotate the fluid travels entrapped between the radial vanes and the cam ring.

→ Nearing the outlet the vanes are pushed in by the cam ring resulting in high pressure. This results in pushing or discharge of liquids out into the discharge line.

## Advantage :-

→ The advantage of the unbalanced vane pump is that as the eccentricity between the cam ring and rotor is changed the volume of fluid pumped can be proportionately




1) Camreing , 2) Inlet, 3) Rotor,
4) Rotating Shaft , 5) outlet , 6) Vane

[ Unbalanced Vane pump ]

changed.

*) Disadvantage :-

→ The suction side of the fluid is at atmospheric pressure. But at the discharge end the fluid is at system pressure and as a result it imparts the side axial thrust on the rotore.

→ This unbalanced force creates changes in the displacement volume and failure of the rotor.

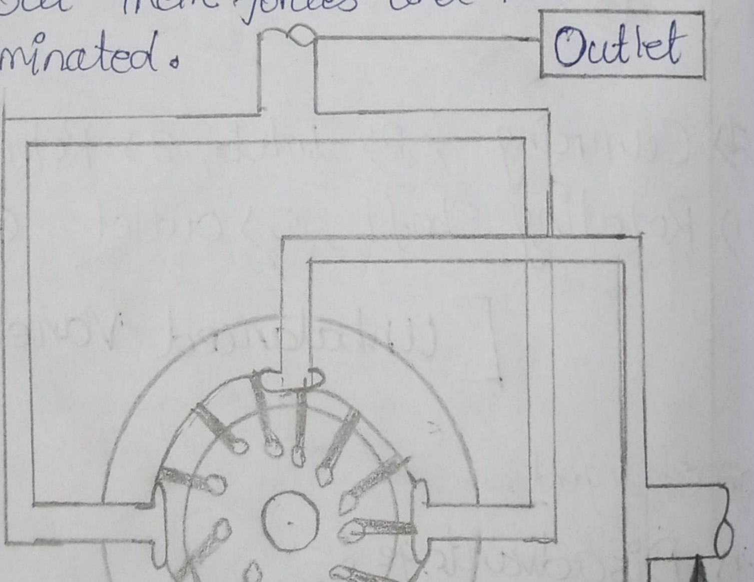
## *) Balanced Vane pump :-

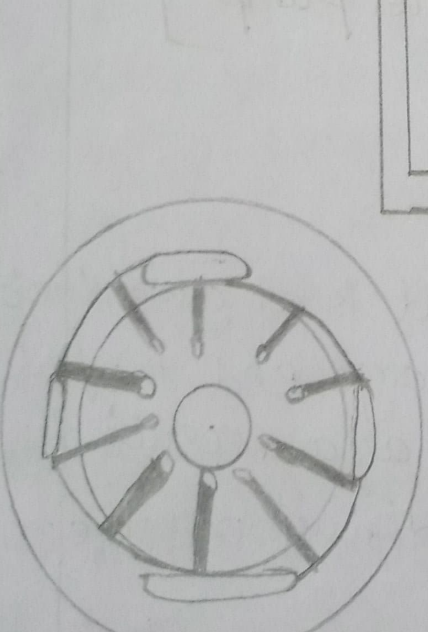
These pumps are Constant volume positive displacement pumps.

→ The disadvantage of the unbalanced Vane pump is that it experiences axial thrust.

→ Changing the shape of the cam ring elliptical instead of Circular can eliminate side thrust.

→ In addition, if there are two suction and discharge ports placed at equal and opposite quadrants, the two pressure or discharge ports cancel out their forces and thus side thrust is eliminated.



[ Balanced Vane Pump ]

→ The displacement of fluid and the basic operation of the pump are similare to the unbalanced pump except that there are two suction and pressure/discharge ports.

→ The pump remains a constant volume discharge pump and hence they cannot be used as a variable discharge pump. But most of the industrial applications use only constant volume positive displacement balanced vane pumps.

*) Volumetric displacement of unbalanced vane pumps :-

→ There is an eccentricity between the centerline of rotor and centerline of cam ring. If the eccentricity is zero than there is no flow.

Let,
$D_c$ = diametere of cam ring
$D_R$ = diametere of rotor
$L$ = width of rotor
$N$ = rotor speed in rpm
$V_D$ = pump volumetric displacement
$e$ = eccentricity
$e_{max}$ = maximum eccentricity
$V_{Dmax}$ = maximum pump volumetric displacement

The maximum possible eccentricity

$$e_{max} = (D_c - D_R)/2$$

Then,

Thus maximum possible eccentricity

$$V_{Dmax} = \pi/4 \left( D_C^2 - D_R^2 \right) L$$

Rearranging,

$$V_{Dmax} = \frac{\pi}{4} \left( D_C + D_R \right) \left( D_C - D_R \right) L$$

Substituting $e_{max}$

$$V_{Dmax} = \frac{\pi}{4} \left( D_C + D_R \right) 2 \, e_{max} L$$

Actual volumetric displacement occurs when $e_{max} = e$,

The value of volumetric displacement is

$$V_D = \frac{\pi}{2} \left( D_C + D_R \right) 2 \, e \, L$$

*) **Piston pumps :-**

The piston pumps are reciprocating pumps. They are classified as

i) Axial piston pump

ii) In-line piston pump

iii) Radial piston pump