# UNIT - V

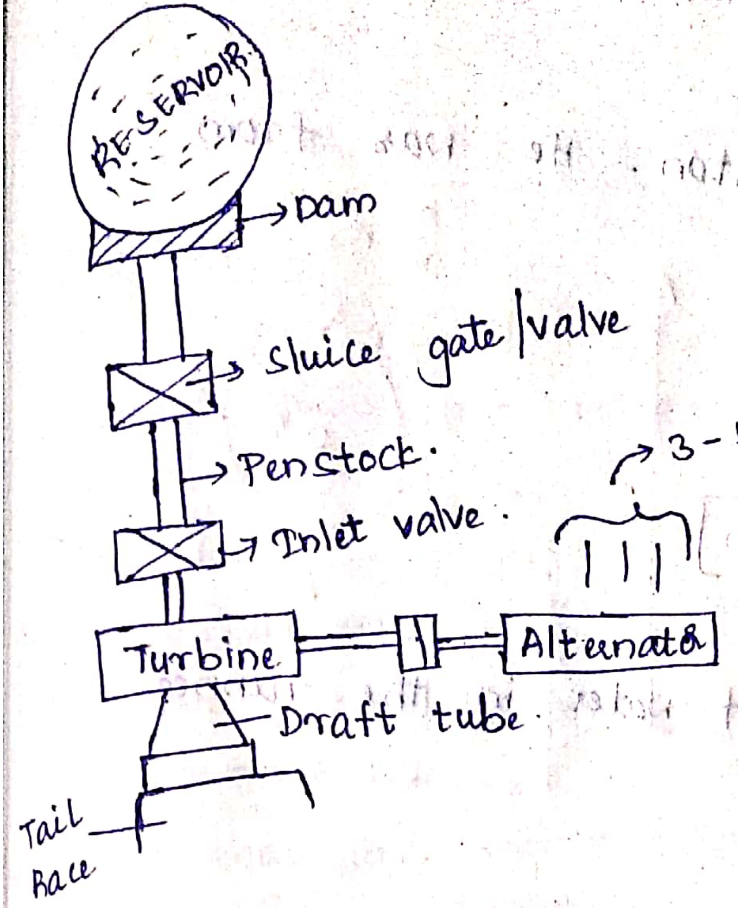
# HYDRAULIC TURBINES

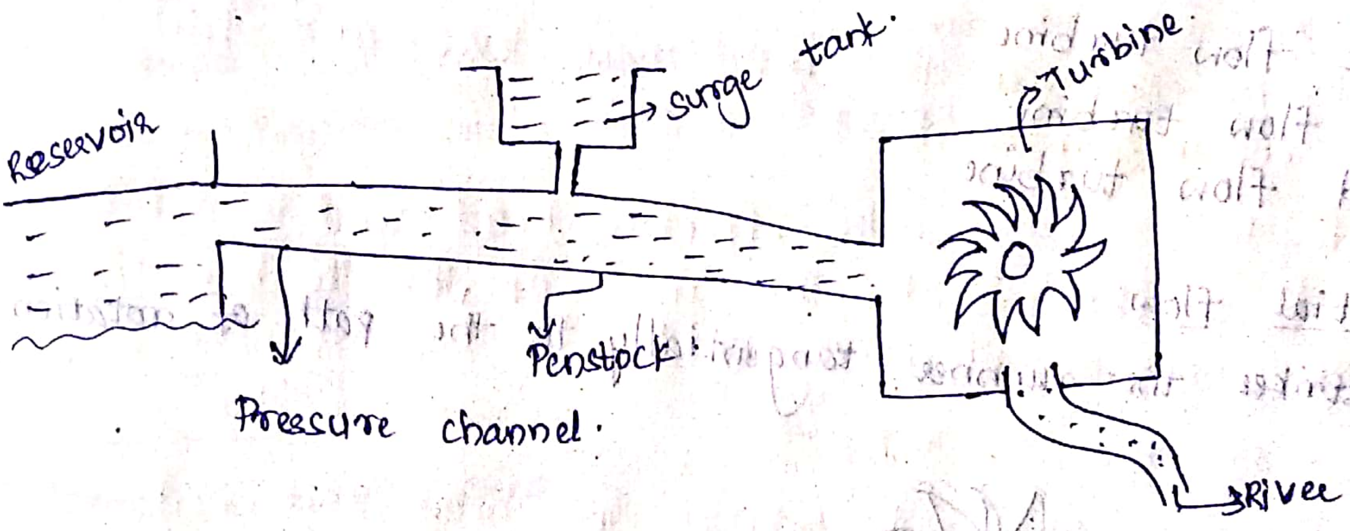
→ Layout of a typical hydropower installation -
→ Heads and efficiencies.
→ classification of turbines
→ ***Pelton wheel
→ ***Francis turbine
→ ***Kaplan turbine

1 MH.

→ Working, working proportions, velocity diagram, workdone and efficiency, hydraulic design, draft tube
→ Theory and efficiency, Governing of turbines
→ surge tanks
→ Unit and specific quantities, selection of turbines, performance characteristics.
→ Geometric similarity
→ cavitation.

Flow sheet of hydroelectric power plant:




classification of turbines:

1. Impulse turbine:
Requires high head and small quantity of flow
Eg: pelton wheel.

2. Reaction turbine:
Requires low head and high rate of flow.
Eg: Fransis turbine and Kaplan turbine

Classification of

According to the name of the originator

1. Pelton turbine (or) Pelton wheel

It was discovered by Allen pelton. He was from california USA.

2. Francis turbine

[Bichens Francis]

3. Kaplan turbine [Dr. victor kaplan]

According to direction of flow of water in the runnee

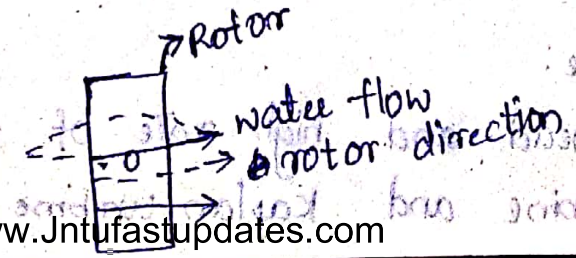
There are four types of flow.

1. Tangential flow turbine.
2. Radial flow turbine
3. Axial flow turbine
4. Mixed flow turbine

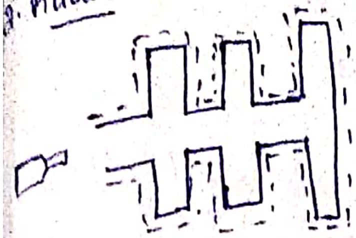
1. Tangential flow:

Water strikes the runnee tangentially to the path of rotation.

3. Axial flow turbine:



3. Radial flow.



4. Mixed flow

when the flow is both radial and axial flow then it is known as mixed flow. Water enters with radial flow and leaves with axial flow

According to water head and quantity of water available.

1. High head and small quantity of water flow.
   (H > 250m) pelton.
2. Medium head and medium quantity of water flow.
   (60m to 250m) Fransis.
3. Low head and large quantity of water flow.
   (< 60m) kaplan and propeller turbines.

According to specific speed of the turbine

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

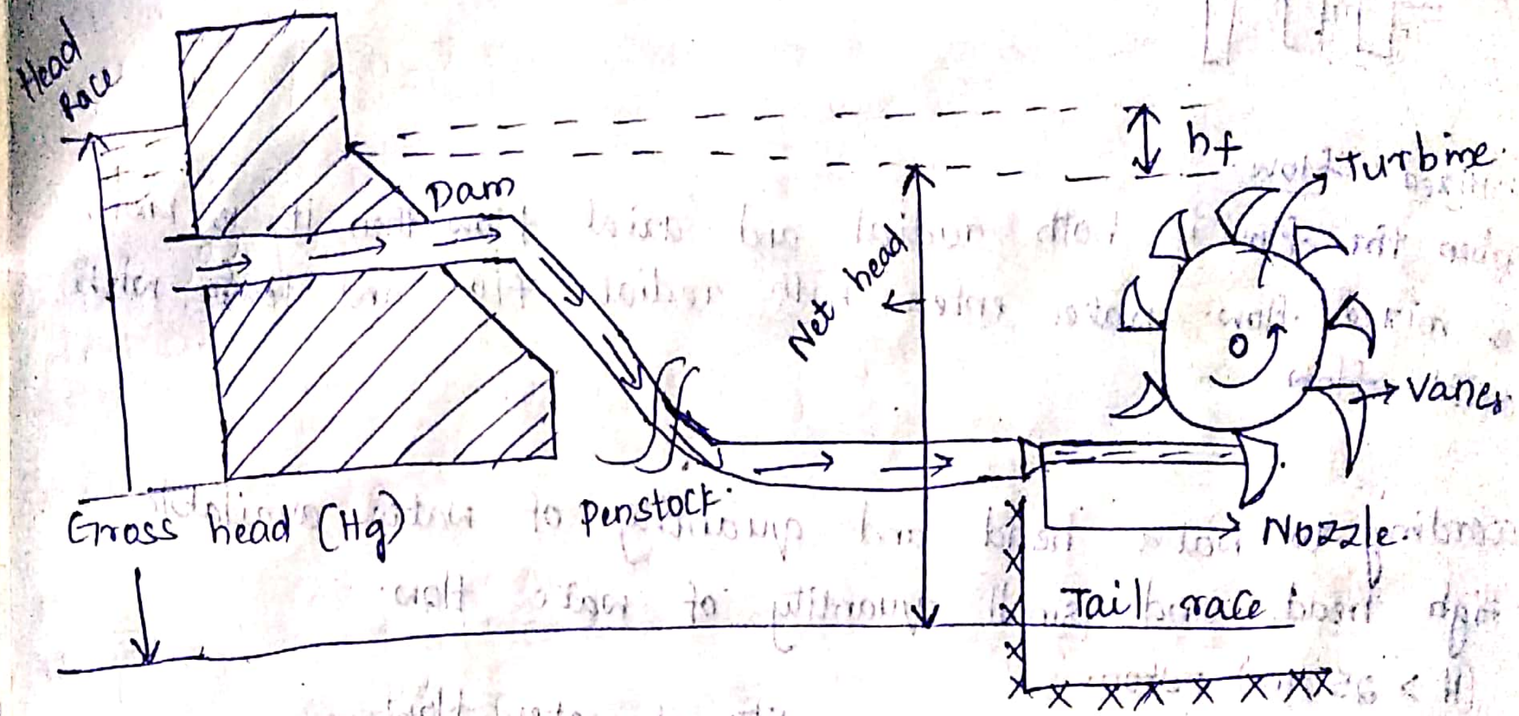
N = Normal working speed

P = Power output

H = The net or effective head in m.

1. Low specific speed: Ex: pelton wheel < 60
2. Medium specific speed (60 to 400) speed Ex: Francis
3. High specific speed > 400. Ex: kaplan turbine.

# Heads and efficiencies of a turbine:




**Gross head:**
The difference between the head race level and tail race level is known gross head.

**Net head:** This is also called effective head and is defined as the head available at the inlet of the turbine is known as net or effective head.

$\therefore$ Net head $H = H_g - H_f$

$H_f$ = Total loss of head between the head race and entrance of the turbine.

$$H_f = \frac{4fLV^2}{2gd}$$

**Efficiency.**

1. Hydraulic efficiency $(\eta_h) = \frac{\text{Power delivered to runner}}{\text{Power supplied at inlet}}$

$$\eta_h = \frac{R.P}{W.P} = \frac{\text{Rotor power}}{\text{Water power}}$$

$$B.P = \frac{\frac{W}{g}\left[V_{w_1} \pm V_{w_2}\right] \times u}{1000} \text{ kW} \rightarrow \text{Tangential flow}$$

$$B.P = \frac{\frac{W}{g}\left[V_{w_1} \times u \pm V_{w_2} \times u\right]}{1000} \text{ kW} \rightarrow \text{Radial flow}$$

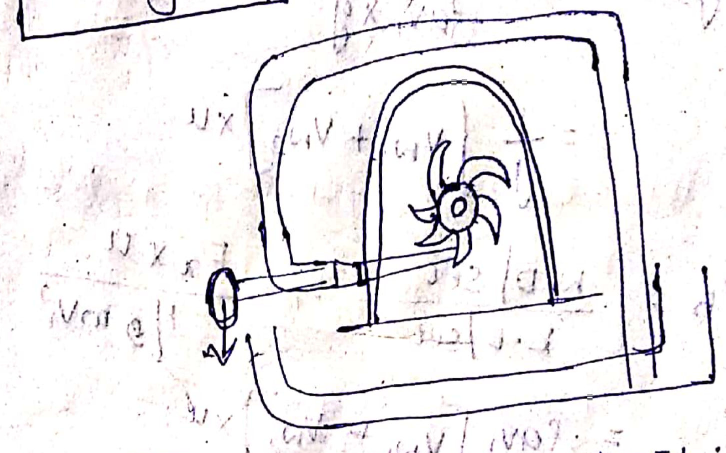
Water power $W.P = \frac{W \times H}{1000}$ kW.

W = Weight of water strikes the vanes of the turbine per sec

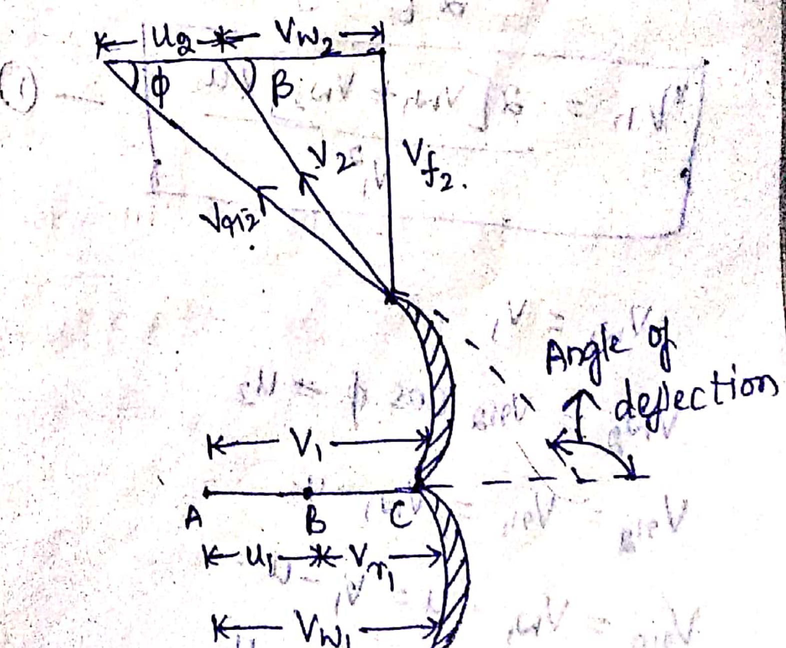
$= \rho g q$

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

$$= \frac{\cancel{1000} \times g \times Q \times H}{\cancel{1000}} \Rightarrow \boxed{W.P = gQH}$$



VELOCITY TRIANGLE AND WORKDONE FOR PELTON WHEEL:




$H = H_g - h_f$ = Net head.

$H_g$ = Gross head

$h_f$ =

$$h_f = \frac{4fLV^2}{2gD}$$

D = Diameter of penstock.

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

$V_{w_1} = V_1$ ,

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

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

$\theta = \alpha = 0°$

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

Work Done $W.D = F_x \times u$

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

$$\text{Weight} = \rho a v_1 \times g$$

$$\frac{W.D}{\text{Weight}} = \frac{\cancel{\rho a v_1}[V_{w_1} + V_{w_2}] \times u}{\cancel{\rho a v_1} \times g}$$

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

$$\eta_h = \frac{W.D/\text{sec}}{K.E/\text{sec}} = \frac{F_x \times u}{\frac{1}{2} m V_1^2}$$

$$= \frac{\cancel{\rho a v_1}[V_{w_1} + V_{w_2}] \times u}{\frac{1}{2}[\cancel{\rho a v_1}] V_1^2}$$

$$\boxed{\eta_h = \frac{2[V_{w_1} + V_{w_2}] \times u}{V_1^2}} \quad — (1)$$

$V_{w_1} = V_1$

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

$V_{r_2} = V_{r_1} = V_{w_1} - u_1$

$V_{r_2} = V_{w_1} - u_1 = V_1 - u_1$

$V_{w_2} = (V_1 - u)\cos\phi - u_1$

The substitute the '$V_{w_2}$' value in equ ①

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

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

$$\boxed{\eta_h = \frac{2(V_1 - u)(1 + \cos\phi) \times u}{V_1^2}} \quad \text{— ②}$$

$$\frac{d}{du}\left[\eta_h\right] = 0$$

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

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

$$\frac{d}{du}\left[V_1 u - u^2\right] = 0.$$

$$V_1 - 2u = 0.$$

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

substitute 'u' value in equ ②

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

$$= \frac{\cancel{2}\left(\frac{2V_1 - V_1}{\cancel{2}}\right)(1 + \cos\phi) \times \frac{V_1}{2}}{V_1^2}$$

$$\boxed{\eta_{h_{max}} = \frac{1}{2}\left[1 + \cos\phi\right]}$$

Efficiency formulas:

1. Mechanical efficiency ($\eta_m$) = $\frac{\text{Power at the shaft of a turbine}}{\text{Power delivered by water to the runner}}$

$= \frac{S.P}{R.P} = \frac{\text{shaft power}}{\text{Runner power}}$

2. volumetric efficiency ($\eta_v$) = $\frac{\text{volume of water actually striking the runner}}{\text{volume of water supplied to the turbine}}$

3. overall efficiency ($\eta_o$) = $\frac{\text{volume available at the shaft of the turbine}}{\text{Power supplied at the inlet of the turbine}}$

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

$= \eta_m \times \eta_h$

Relation between mechanical, volumetric and overall efficiencies are $\boxed{\eta_o = \eta_m \times \eta_h}$

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

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

Points to be remembered in pelton wheel:

1. Velocity at inlet $V_1 = C_v \sqrt{2gH}$

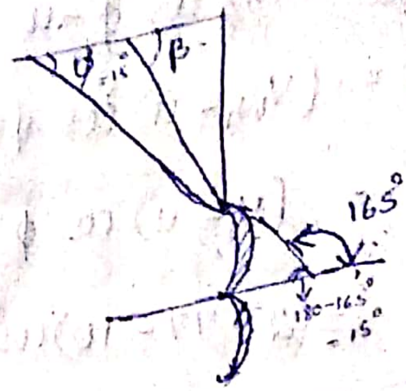
$C_v = 0.98$ (or) $0.99$

2. $U = \phi \sqrt{2gH}$

$\phi$ = speed ratio = 0.43 to 0.48

3. Angle of deflection of the jet through buckets is taken

as 165° if no angle of deflection is given




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

N = speed in rpm  D = Mean diameter

5. Jet ratio (m) $= \frac{D}{d} = \frac{\text{Mean diameter}}{\text{Diameter of the nozzle}}$

For most cases m = 12

6. No. of buckets (z) $= 15 + \frac{D}{2d}$

$$\boxed{z = 15 + 0.5\,m}$$

7. No. of jets $= \frac{\text{Total rate of flow through the turbine}}{\text{Rate of flow through a single jet.}}$

Problems.

1. A pelton wheel has a mean bucket speed of 10 m/s with a jet of water flowing at the rate of 700 lit/sec under a head of 30 m. The buckets deflect the jet through an angle of 165°. calculate the power given by water to the runner and the hydraulic efficiency of the turbine. Assume coefficient of velocity as 0.98.

sol: Given Mean bucket $U = 10$ m/sec, discharge $Q = 700$ lit/sec

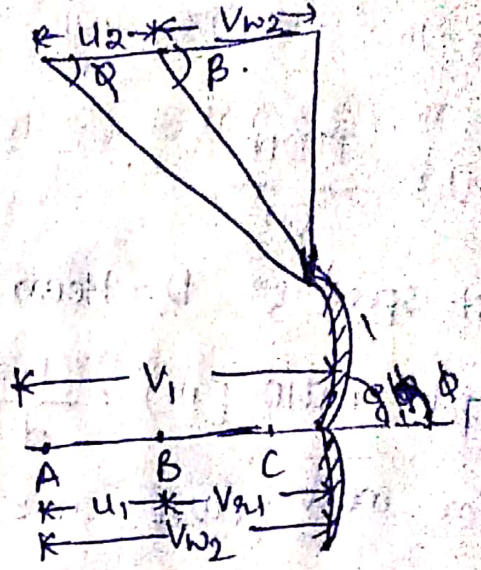
$= 0.7\ m^3/sec$

$H = 30$ m, Angle of deflection $= 165°$, $C_v = 0.98$

$\phi = 180 - 165° \Rightarrow \phi = 15°$

$V_{w_1} = V_1 = C_v \sqrt{2gH} = 0.98 \sqrt{2 \times 9.81 \times 30}$

$$\boxed{V_{w_1} = 23.77\ m/s.}$$




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

$$= (V_{w_1} - u)\cos\phi - u$$

$$= (V_1 - u)\cos\phi - u$$

$$= (23.77 - 10)\cos 15^\circ - 10$$

$$\boxed{V_{w_2} = 3.3 \text{ m/sec}}$$

$$\text{Power} = \frac{W.D}{1000} \text{ kW}$$

$$= \frac{F_x \times u}{1000} \text{ kW}$$

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

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

$$= 0.7 [23.77 + 3.3] \times 10$$

$$\boxed{\text{Power} = 189.49 \text{ kW}}$$

$$\eta_h = \frac{2[V_{w_1} + V_{w_2}] \times u}{V_1^2} = \frac{2 \times [23.77 + 3.3] \times 10}{(23.77)^2}$$

$$\eta_h = 0.95$$

$$\boxed{\eta_h = 95\%}$$

2. 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 700 m if the side clearance angle is $15^\circ$ and discharge through the nozzle is 0.1 $m^3/sec$ find (i) Power available at the nozzle
(ii) Hydraulic efficiency of the turbine.

Sol: Given data, $D = 1\,m$, $N = 1000$ r.p.m, $H = 700\,m$
$Q = 0.1\ m^3/sec$, $\phi = 15^\circ$.

(i) Water power $W.P = \frac{\rho g Q H}{1000} = 9.81 \times 0.1 \times 700 = 686.7$

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

$V_1 = C_v\sqrt{2gH} = 0.98\sqrt{2 \times 9.81 \times 700} \Rightarrow V_1 = 114.84$

$U = \frac{\pi D N}{60} = \frac{\pi \times 1 \times 1000}{60} \Rightarrow U = 52.35$ m/sec

$$\eta_h = \frac{2(114.84 - 52.35)(1 + \cos 15^\circ) \times 52.35}{114.84^2} \times 100$$

$\boxed{\eta_h = 97\%}$

Two jets strikes the buckets of a pelton wheel which is having shaft power as 15450 kW. The diameter of each jet is given as 200 mm if the net head on the turbine 400 m find the over all efficiency of the turbine. Take $C_v = 1$

Sol: Given,

Shaft power. $S.P = 15450\ KW$, $d = 200\ mm = 0.2\ m$

$H = 400\ m$, $C_v = 1$.

W.K.T

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

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

$$= \frac{15450}{9.81 \times 5.56 \times 400}$$

$$= 0.708$$

$$= 70.8\%$$

$$\therefore Q = aV_1$$

$$= \frac{\pi}{4} d^2 \left(C_v \sqrt{2gH}\right)$$

$$= \frac{\pi}{4}(0.2)^2 \left(1 \times \sqrt{2 \times 9.81 \times 400}\right)$$

$$= 2.78\ m^3/sec$$

Total discharge from two jets

$$Q = 2 \times 2.78$$

$$= 5.56\ m^3/sec$$

Design of pelton wheel (Impulse)

1. Diameter of the jet (d)
2. Diameter of the wheel (D)
3. Width of buckets (5×d)
4. Depth of buckets (1.2×d)
5. No. of buckets ($z = 1.5 + 0.5m$) $m = \frac{D}{d}$

1. A pelton wheel is to be designed for a head of 60 m when running at 200 r.p.m. The pelton wheel develops 95.6475 KW shaft power. The velocity of buckets = 0.45 times the velocity of jet, overall efficiency = 0.85 & $C_v = 0.98$?

Sol: Given data,

$N = 200$ r.p.m ; $H = 60$ m, S.P $= 95.6475$ kW

$C_v = 0.98$ ; $\eta_o = 0.85$ ;

$V_1 = C_v\sqrt{2gH} = 0.98\sqrt{2\times9.81\times60} = 33.62$ m/s

$u = 0.45\times V_1$.

$u = 0.45\times33.62 = 15.13$ m/sec

$u = \frac{\pi DN}{60} \Rightarrow D = \frac{15.13\times60}{200\times3.14} = 1.444$ m

$W.P = \frac{S.P}{\eta_o} = \frac{95.6475}{0.85} = 119.56$ kW.

$gQH = 119.56 \Rightarrow Q = 0.2\ m^3/sec$.

$Q = aV_1 \Rightarrow a = \frac{Q}{V_1} = \frac{0.2}{33.62}$

$\frac{\pi}{4}d^2 = \frac{0.2}{33.62} \Rightarrow d = 0.087$ m

$= 87$ mm

Width of buckets $= 5\times d = 5\times0.087 = 0.435$ m

Depth of buckets $= 1.2\times0.087 = 0.1044$ m.

No. of buckets $\left(z = 15 + 0.5\frac{D}{d}\right) = 15 + 0.5\times\frac{1.44}{0.087}$

$= 23.27$

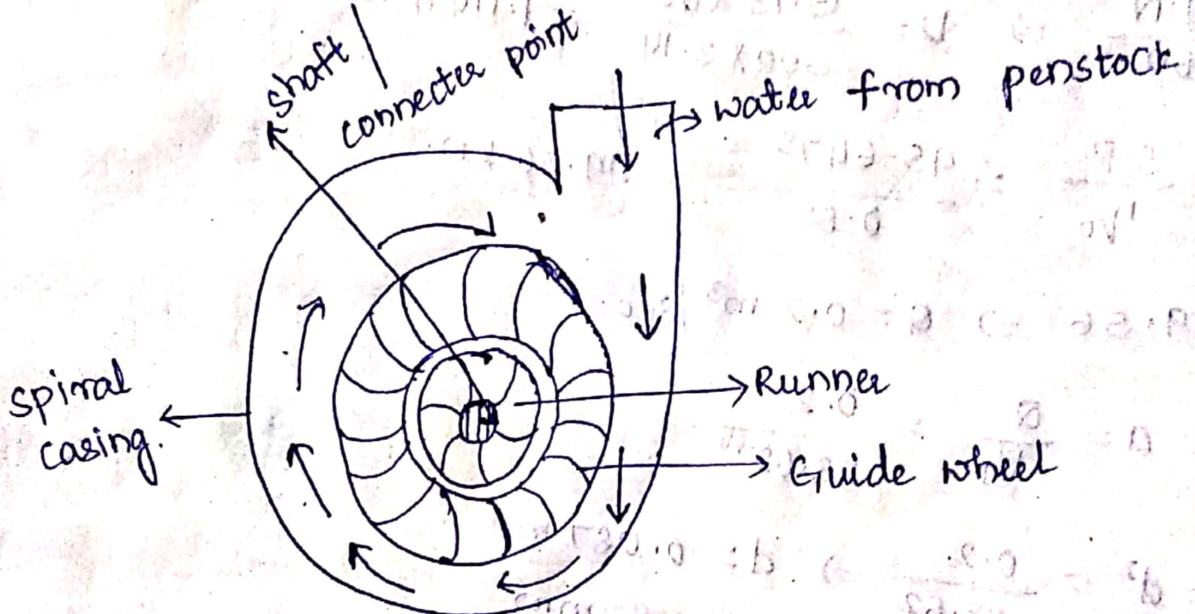
No. of buckets $(z) = 24$.

Reaction radial flow turbine:

When the water flows in radial direction the turbine is known as radial flow reaction turbine;

1. If the flow is outward to inward towards the axis of rotation is called inward flow turbine

2. If the water flow is inward to outward toward the axis of rotation is called outward flow turbine.

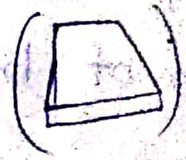


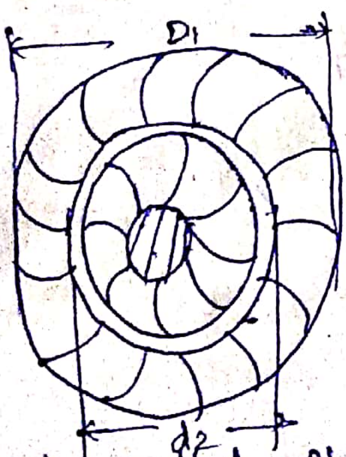

Radial flow turbine.

Parts of radial flow turbine.

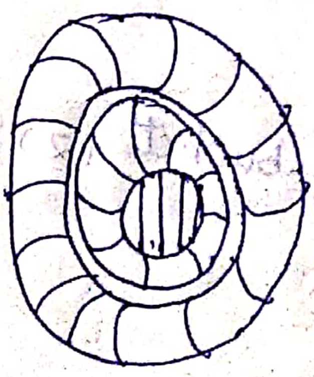
1. Runner 2. Guide wheel 3. spiral casing 4. Draft tube

Gradually increa






Inward radial flow



outward radial flow

Work done and efficiency of Inward radial flow turbine :

Work done $= F_x \times u$ $\qquad \left( u_1 = \frac{\pi D N}{60}, \frac{\pi D_2 N}{60} \right)$

$$= \rho a V_1 \left[ V_{w_1} \pm V_{w_2} \right] \times u$$

$$\text{work done} = \rho a V_1 \left[ V_{w_1} \times u_1 + V_{w_2} \times u_2 \right]$$

$$\text{work done / weigth} = \frac{1}{g} \left[ V_{w_1} u_1 \pm V_{w_2} \times u_2 \right]$$

If $\beta = 90^\circ$, $V_{w_2} = 0$

$$= \frac{1}{g} \left[ V_{w_1} u_1 \right]$$

The obtained equation is known as Euler's equation or the fundamental equation is known as Hydro dynamic machines. This equation is given by switz scientist L. Euler

$$\eta_h = \frac{B.P.}{W.P}$$

$$= \frac{\frac{W}{1000g} \left[ V_{w_1} \times u_1 \pm V_{w_2} u_2 \right]}{\frac{W \times H}{1000}}$$

$$= \frac{\cancel{W}}{\cancel{1000}g} \left[ V_{w_1} \times u_1 \pm V_{w_2} u_2 \right] \times \frac{\cancel{1000}}{\cancel{W} \times H}$$

$$= \frac{\left[ V_{w_1} u_1 + V_{w_2} u_2 \right]}{gH}$$

$$\beta = 90^\circ, V_{w_2} = 0 \Rightarrow \boxed{\eta_h = \frac{V_{w_1} u_1}{gH}}$$

## Definitions:

1. Speed ratio: The speed ratio is defined as

$$\phi = \frac{u_1}{\sqrt{2gH}}$$

$u_1$ = Tangential velocity

2. Flow ratio: The ratio of the velocity of flow at inlet $V_{f_1}$ to the velocity given $\sqrt{2gH}$ is known as flow ratio.

$$\text{Flow ratio} = \frac{V_{f_1}}{\sqrt{2gH}}$$ (H = Head of the turbine)

3. Discharge of the turbine:

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

where $D_1$ = Diameter of the runner at inlet

$B_1$ = width of the runner at inlet

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

$D_2, B_2, V_{f_2}$ are corresponding values at outlet.

4. Head (H): Head of the turbine.

$$(H) = \frac{P_1}{\rho g} + \frac{V_1^2}{2g}$$

5. Radial discharge: This means the angle made by absolute velocity with the tangent on the wheel is $90^\circ$ and is '0' (zero). Radial component of the whirl velocity discharge at outlet means $\beta = 90^\circ$.

6. If there is no loss of energy when water flows through the vanes then we have the below equation.

$$H - \frac{V_2^2}{2g} = \frac{1}{g}\left[V_{w_1} u_1 \pm V_{w_2} u_2\right]$$

Problems

1. An inward flow reaction turbine, has external and internal diameters as 1m and 0.5m respectively. The velocity of flow through the runner is constant and is equal to 1.5 m/sec. Determine (i) Discharge through the runner (ii) width of the turbine at outlet. If the width of the turbine at inlet is 200 mm.

Sol: Given $D_1 = 1m$, $D_2 = 0.5\ m$, $B_1 = 200\ mm$

$V_{f_1} = 1.5\ m/sec = V_{f_2}$

(i) $Q = \pi D_1 B_1 \times V_{f_1} = \cancel{\pi D_2 B_2 \times V_{f_2}}\ \pi \times 1 \times 0.2 \times 1.5 = 0.942$

(ii) $Q = \pi D_2 B_2 \times V_{f_2} \Rightarrow 0.942 = \pi \times 0.5 \times B_2 \times 1.5 \Rightarrow \boxed{B_2 = 0.39m}$

## OUTWARD FLOW REACTION TURBINE:

$u_1 < u_2$

$D_1 < D_2$

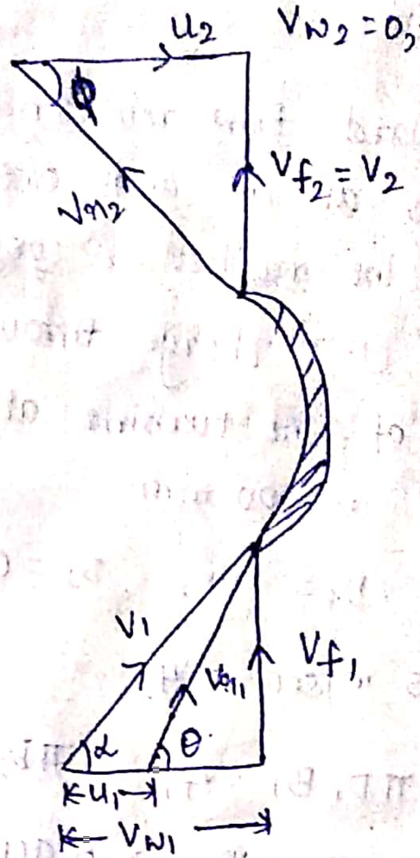
2. The Internal and external diameters of an outward flow reaction turbine are 2m and 2.75 m respectively. The turbine is running at 250 r.p.m and rate of flow of water through the turbine is 5 m³/sec. The width of the runner is constant at inlet and outlet and is equal to 250 mm. The head on the turbine is 150 m 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

Sol) Given data, $D_1 = 2m$, $D_2 = 2.75\ m$

$Q = 5\ m^3/sec$, speed $N = 250$ r.p.m

$B_1 = B_2 = 250\ mm = 0.25\ m$

$H = 150\ m$




find outs:

$\theta = ?,\ \phi = ?,\ V_{f_1} = ?\ ,\ V_{f_2} = ?$

$$\tan\theta = \frac{V_{f_1}}{V_{w_1}} \Rightarrow \theta = \tan^{-1}\left(\frac{V_{f_1}}{V_{w_1} - u}\right)$$

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

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

$$Q = \pi D_1 B_1 V_{f_1}$$

$$\Rightarrow V_{f_1} = \frac{Q}{\pi D_1 B_1} = \frac{5}{\pi \times 2 \times 0.25} \Rightarrow \boxed{V_{f_1} = 3.18\ m/sec}$$

$$\Rightarrow V_{f_2} = \frac{Q}{\pi D_2 B_2} = \frac{5}{\pi \times 2.75 \times 0.25} \Rightarrow \boxed{V_{f_2} = 2.31\ m/s}$$

$$H - \frac{V_2^2}{2g} = \frac{1}{g}\left[V_{w_1} - u_1\right] \qquad \left[\because V_2 = V_{f_2}\right]$$

$$150 - \frac{(2.31)^2}{2 \times 9.81} = \frac{1}{9.81}\left[V_{w_1} \times 26.17\right]$$

$$\boxed{V_{w_1} = 56.12\ m/s}$$

$$\theta = \tan^{-1}\left(\frac{3.18}{56.12 - 26.17}\right) \Rightarrow \boxed{\theta = 6°\ 3'\ 38''}$$

$$\phi = \tan^{-1}\left(\frac{V_{f_2}}{u_2}\right) = \tan^{-1}\left(\frac{2.31}{35.99}\right) \Rightarrow \boxed{\phi = 3°\ 40'}$$

# FRANCIS TURBINE:

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

In the modern Francis turbine the water enters the runner of the turbine in radial direction at outlet and leaves in the axial direction at the inlet of the ~~to~~ runner. Thus, the Francis turbine is a mixed flow type turbine.

→ Workdone by water on the runner per second will be

$$W.D|sec = \rho Q \left[V_{w_1} u_1\right]$$

→ Workdone per unit weight of water striking per second.

$$= \frac{1}{g}\left[V_{w_1} u_1\right]$$

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

→ The ratio of width of the wheel to its diameter is given as $n = \frac{B_1}{D_1}$

n varies from 0.10 to 0.40.

→ Flow ratio $= \frac{V_{f_1}}{\sqrt{2gH}}$ This ratio varies from 0.15 to 0.30

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

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

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

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

Sol: Note: when hydraulic loss is given like pacentage of available energy we have to take it as a value by multiplying with H.

Ex: 22%, 20%.

0.22 H, 0.20H.

Given, $\eta_0 = 75\%$, S.P = 148.25 kW, H = 7.62m, N = 150 rpm

$u_1 = 0.26\sqrt{2gH} = 0.26\sqrt{2\times9.81\times7.62} \Rightarrow \boxed{u_1 = 3.17 \text{ m/s}}$

$V_{f_1} = 0.96\sqrt{2gH} = 0.96\sqrt{2\times9.81\times7.62} \Rightarrow \boxed{V_{f_1} = 11.73 \text{ m/s}}$

Find out:

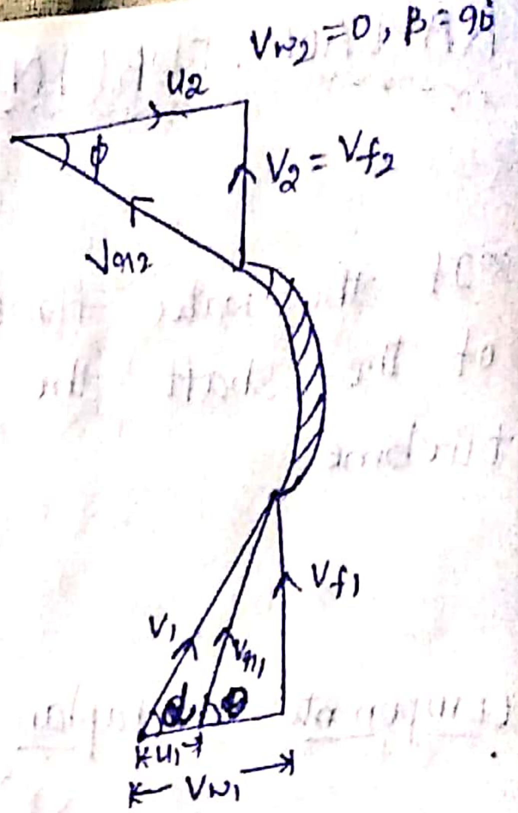
$\alpha = ?$, $\theta = ?$, $D_1 = ?$, $B_1 = ?$

$\tan\theta = \dfrac{V_{f_1}}{(V_{w_1} - u_1)}$ ; $\tan\alpha = \dfrac{V_{f_1}}{V_{w_1}}$

$$Q = \pi D_1 B_1 V_{f_1}$$

$$B_1 = \frac{Q}{\pi D_1 V_{f_1}}$$

$$u_1 = \frac{\pi D N}{60}, \quad D_1 = \frac{60 u_1}{\pi N}$$




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

$$\eta_h = \frac{T.H - \text{Head loss}}{\text{Head at inlet}}$$

$$\eta_h = \frac{H - 0.22H}{H}$$

$$= \frac{H(1-0.22)}{H} \Rightarrow \eta_h = 0.78 \Rightarrow \boxed{\eta_h = 78\%}$$

$$\eta_h = \frac{V_{w_1} u_1}{gH} \Rightarrow 0.78 = \frac{V_{w_1} \times 3.17}{9.81 \times 7.62} \Rightarrow \boxed{V_{w_1} = 18.39 \text{ m/s}}$$

$$\theta = \tan^{-1}\left[\frac{V_{f_1}}{V_{w_1} - u_1}\right] = \tan^{-1}\left[\frac{11.78}{18.39 - 3.17}\right] \Rightarrow \boxed{\theta = 37° \, 44'}$$

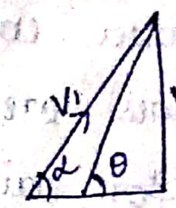
$$\alpha = \tan^{-1}\left[\frac{V_{f_1}}{V_{w_1}}\right] = \tan^{-1}\left(\frac{11.78}{18.39}\right) \Rightarrow \boxed{\alpha = 32° \, 38'}$$

$$D_1 = \frac{60 u_1}{\pi N} = \frac{60 \times 3.17}{\pi \times 150} \Rightarrow \boxed{D_1 = 0.40 \text{ m}}$$

$$Q = a V_1$$

$$a = \frac{\pi}{4}(D_1)^2 = \frac{\pi}{4}(0.40)^2 \Rightarrow a = 0.12$$

$$\boxed{Q = 2.73 \text{ m}^3/\text{s}}$$




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

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

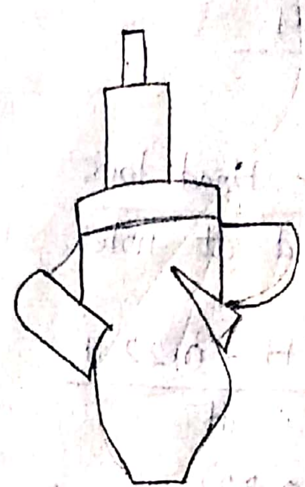
$$V_1 = \frac{11.73}{\sin 32° 38'}$$

$$\boxed{V_1 = 21.75}$$

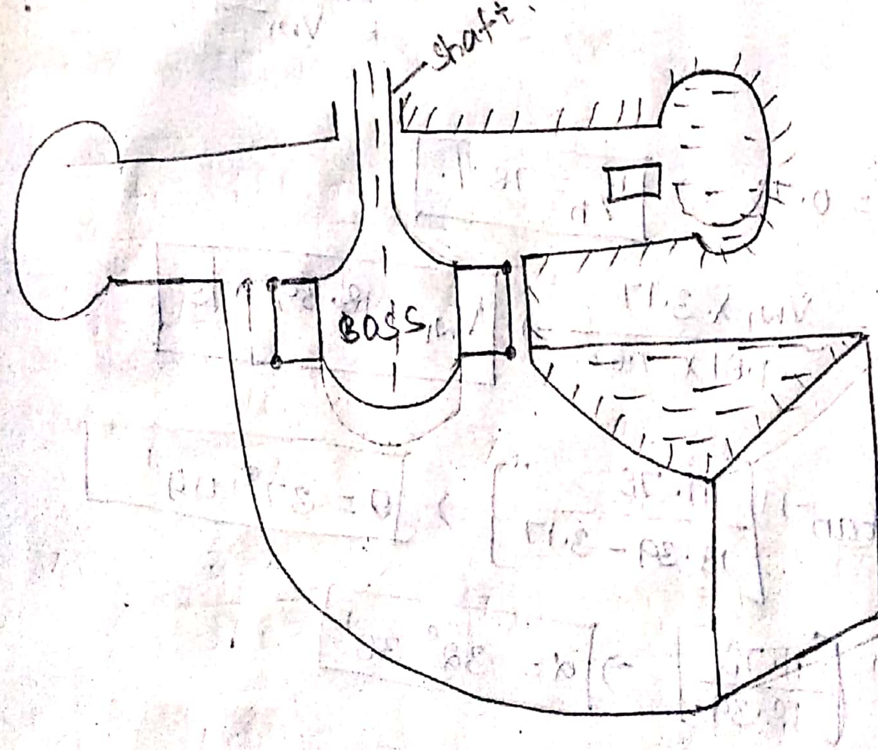
$$B_1 = \frac{Q}{\pi D_1 V_{f_1}} = \frac{2.73}{\pi \times 0.40 \times 11.73} \Rightarrow \boxed{B = 0.185 \text{ m}}$$

# KAPLAN TURBINE (OR) AXIAL FLOW REACTION TURBINE

So If the water flows parallel to the axis of rotation of the shaft the turbine is known as axial flow turbine.



Components of kaplan turbine:




For axial flow reaction turbine the shaft of the turbine is vertical. The lower end of the shaft is made with Hub (or) boss. The vanes are fixed on the hub hence hub acts as the runner for axial flow reaction turbine.

The main parts of the kaplan turbine are

1. scroll casing 2. Guide vanes mechanism 3. Hub with vanes (or) boss with vanes (or) runner of the turbine

4. Draft tube.

Formulas:

1. Discharge through the runner is obtained as

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

where $D_o$ = outer diameter of the runner.

$D_b$ = Diameter of the hub, and.

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

2. The peripheral velocity at inlet and outlet are $u_1 = u_2 = \frac{\pi D_o N}{60}$

velocity of flow at inlet and outlet are equal.

3 Area of flow at inlet = Area of flow at outlet $= \frac{\pi}{4}\left(D_o^2 - D_b^2\right)$

1. A kaplan turbine working under a head of 20 m develops 11772 kW shaft power. The outer diameter of the runner is 3.5 m and hub diameter is 1.75 m. The guide blade angle at the extreme edge of the runner is 35°. The hydraulic and overall efficiencies of the turbines are 88% & 84%. If the velocity of whirl is 0 at outlet, determine

(i) runner vane angles at inlet & outlet at the extreme edge of the runner.

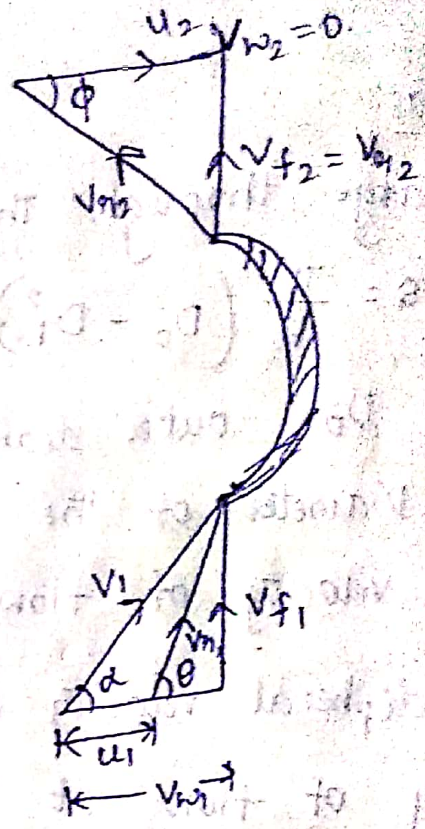
(ii) speed of the runner.

Sol: Given $D_o = 3.5$ m, $D_b = 1.75$ m, $\alpha = 35°$, $H = 20$ m

Power of the shaft $= 11772$ kW

$V_2 = 0$

$Q = \frac{\pi}{4}\left(D_0^2 - D_b^2\right) V_{f_1}$




Water power

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

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

$$\boxed{Q = 71.42\ m^3/s.}$$

$$Q = \frac{\pi}{4}\left(D_0^2 - D_b^2\right) \times V_{f_1}$$

$$71.42 = \frac{\pi}{4}\left((3.5)^2 - (1.75)^2\right) \times V_{f_1}.$$

$$\boxed{V_{f_1} = 9.89\ m/sec}$$

$$V_{f_1} = V_{f_2} = 9.89\ m/sec.$$

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

$$V_{w_1} = \frac{9.9}{\tan 35^\circ} \Rightarrow \boxed{V_{w_1} = 14.13\ m/sec}$$

$$\eta_h = \frac{V_{w_1} u_1}{g \times H} \Rightarrow 0.88 = \frac{14.13 \times u_1}{9.81 \times 20}.$$

$$\boxed{u_1 = 12.21\ m/sec}$$

$$\tan\theta = \frac{V_{f_1}}{V_{w_1} - u_1} = \frac{9.9}{14.13 - 12.21}$$

$$\boxed{\theta = 79^\circ 1^\circ}$$

$$\phi = \tan^{-1}\left(\frac{V_{f_2}}{u_2}\right)$$

$= \tan^{-1}\left[\frac{9.9}{12.21}\right]$

$\boxed{\phi = 39^\circ 2'}$

(ii) $u_1 = u_2 = \frac{\pi D_0 N}{60}$

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

$N = 66.62 \Rightarrow \cancel{N = 66}$

$\boxed{N = 67\ r.p.m}$

## Draft tubes:

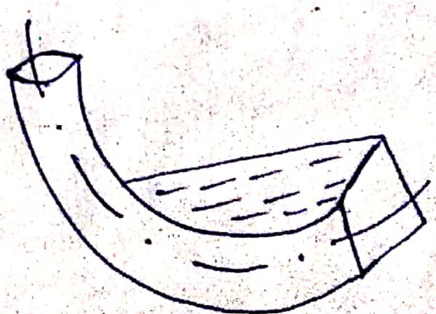
Types of Draft tubes:

1. conical draft tube


2. simple Elbow draft tube

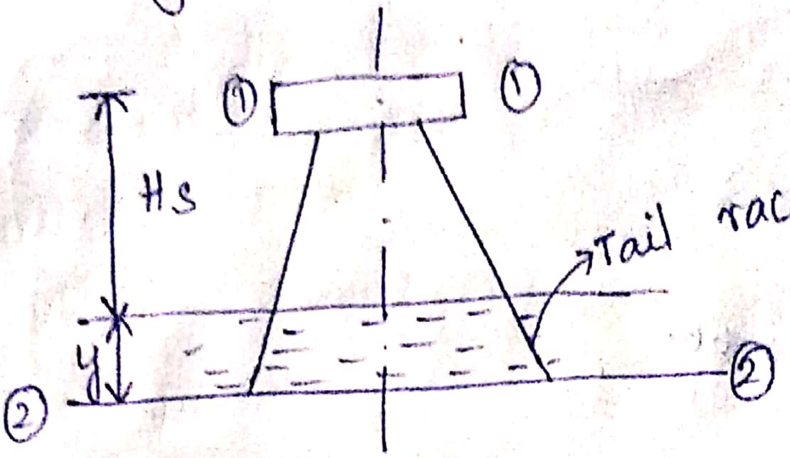
3. Moody spreading draft tu

4. Tube with circular inlet and rectangular outlet



## Draft tube theory:

Consider a graphical draft tube as shown in figure.




$$\frac{P_1}{\rho g} + \frac{V_1^2}{2g} + H_s + y = \frac{P_2}{\rho g} + \frac{V_2^2}{2g} + 0 + h_f.$$

$$\frac{P_2}{\rho g} = \text{Atmospheric pressure} + y.$$

$$\frac{P_2}{\rho g} = \frac{P_a}{\rho g} + y.$$

$$\frac{P_1}{\rho g} + \frac{V_1^2}{2g} + H_s + \cancel{y} = \frac{P_a}{\rho g} + \cancel{y} + \frac{V_2^2}{2g} + h_f$$

$$\boxed{\frac{P_1}{\rho g} + \frac{V_1^2}{2g} + H_s = \frac{P_a}{\rho g} + \frac{V_2^2}{2g} + h_f}$$

**Efficiency of the draft tube:**

$$\eta_d = \frac{\left[\frac{V_1^2}{2g} - \frac{V_2^2}{2g}\right] - h_f}{\left[\frac{V_1^2}{2g}\right]}$$

A water turbine has a velocity of 6 m/sec at the entrance to the draft tube and the velocity of 1.2 m/sec at the exit. For frictional losses of 0.1 m and a tail water 5m below the entrance to the draft tube. Find the pressure head at the entrance.

sol: $V_1 = 6\,m/sec$, $V_2 = 1.2\,m/sec$, $h_f = 0.1\,m$, $H_s = 5\,m$

$$\frac{P_1}{\rho g} + \frac{V_1^2}{2g} + H_s = \frac{P_a}{\rho g} + \frac{V_2^2}{2g} + h_f$$

$$\frac{P_1}{\rho g} = \frac{P_a}{\rho g} + \frac{V_2^2}{2g} + h_f - \left[\frac{V_1^2}{2g} + H_s\right]$$

$$= \frac{P_a}{\rho g} + \frac{(1.2)^2}{2\times9.81} + 0.1 - \left[\frac{(6)^2}{2\times9.81} + 5\right]$$

$$= \frac{P_a}{\rho g} + (-6.66)$$

If $\frac{P_a}{\rho g} = 0$ then $\frac{P_1}{\rho g} = 0 - 6.66 = -6.66$ (vaccum pressure)

If $\frac{P_a}{\rho g} = 10$ then $\frac{P_1}{\rho g} = 10 - 6.66 = 3.34.$

Specific speed:

It is defined as the speed of the turbine which is identical in shape, geometric dimensions, blade angles, gate opening etc.

Derivation of the specific speed:

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

$\eta_0$ = overall efficiency.

Q = Discharge through the turbine.

$\Rightarrow P \propto Q \times H.$

$u \propto V$

$V \propto \sqrt{H}$

$u \propto \sqrt{H}$

$u = \frac{\pi D N}{60}.$

$u \propto DN.$

$\sqrt{H} \propto D.$

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

$\Rightarrow Q = A \times V.$

$A \propto B \times D.$

$A \propto D \times D.$

$A \propto D^2.$

$V \propto \sqrt{H}$

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

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

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

$P \propto \left[\frac{H^{3/2}}{N^2}\right] \times H$

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

$$P = k \cdot \frac{H^{5/2}}{N^2}$$

If $P=1$, $H=1$, $N = N_s$.

$$1 = k \cdot \frac{(1)^{5/2}}{N_s^2}$$

$$N_s^2 = k$$

$$\therefore P = N_s^2 \times \frac{(H)^{5/2}}{N^2}$$

$$\Rightarrow N_s^2 = \frac{N^2 P}{(H)^{5/2}} \Rightarrow N_s = \frac{\sqrt{N^2 P}}{\sqrt{H^{5/2}}}$$

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

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1. A turbine is to operate under a head of 25 m at 200 rpm, the discharge is 9 cumics. If the efficiency is 90%. Determine (i) specific speed of the machine. (ii) power generated (iii) Type of the turbine.

Sol: Given data, $H = 25$ m, $N = 200$ r.p.m.

$Q = 9$ cumics $= 9\ m^3/sec$.

$$\boxed{1 \text{ cumic} = 1\ m^3/sec}$$

$\eta_o = 90\% = 0.90$

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

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

$$\boxed{P = 1986.52 \text{ kW}}$$

$$N_s = \frac{200\sqrt{1986.52}}{(25)^{5/4}} \Rightarrow \boxed{N_s = 159.4 \text{ r.p.m}}$$

(iii) As the specific speed lies between 51 & 225.
The turbine is Francis turbine.

2. The turbine develops 9000 kW when running at a speed of 140 r.p.m under a head of 30 m. Determine the specific speed of the turbine.

sol: $N_s = \frac{N\sqrt{P}}{(H)^{5/4}} = \frac{140\sqrt{9000}}{(30)^{5/4}} \Rightarrow \boxed{N_s = 189.16 \text{ r.p.m}}$

## Unit Quantities:

The following are the three important unit quantities which must be studied under a unit head.

(i) unit speed. (ii) Unit power (iii) Unit discharge.

(i) Unit speed:

$$N_u = \frac{N}{\sqrt{H}}$$

(ii) Unit discharge: $Q_u = \frac{Q}{\sqrt{H}}$.

(iii) Unit power : $P_u = \frac{P}{H^{3/2}}$.

(i) Unit speed:

$u \propto V$

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

$u = \frac{\pi D N}{60}$.

$u \propto N$

$N \propto \sqrt{H}$

$N = k\sqrt{H}$

if $N = 1$, $N = N_u$

$\Rightarrow N_u = k\sqrt{1}$

$\therefore N_u = k$

$\therefore N = N_u\sqrt{H}$

(ii) Unit discharge:

$Q = A \times V.$

$Q \propto V \propto \sqrt{H}.$

$Q = k\sqrt{H}$

if $Q = Q_u,\ H = 1.$

$Q_u = k\sqrt{1}.$

$Q_u = k.$

$Q = Q_u \sqrt{H}$

$$\boxed{Q_u = \frac{Q}{\sqrt{H}}}$$

~~(iii) Unit power:~~

$\cancel{\eta_o = \frac{P}{\frac{\rho g Q H}{1000}}}$

$\cancel{P =}$

(iii) Unit power:

$$\eta_o = \frac{P}{\frac{\rho g Q H}{1000}}$$

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

$P \propto Q \times H.$

$P \propto \sqrt{H} \times H.$

$P \propto H^{3/2}$

$P = k \times H^{3/2}.$

if $P = P_u,\ H = 1.$

$P_u = k \times (1)^{3/2}.$

$P = P_u \times (H)^{3/2}.$

$$\boxed{P_u = \frac{P}{H^{3/2}}}$$

1\. A turbine develops 9000 kW when running at 10 r.p.m. The head on the turbine is 30 m. If the head on the turbine is reduced to 18 m. Determine the speed and power developed by the turbine.

Sol:

## Characteristic curves of the turbines:

characteristic curves are used for knowing exact behaviour of the turbines under different working conditions.

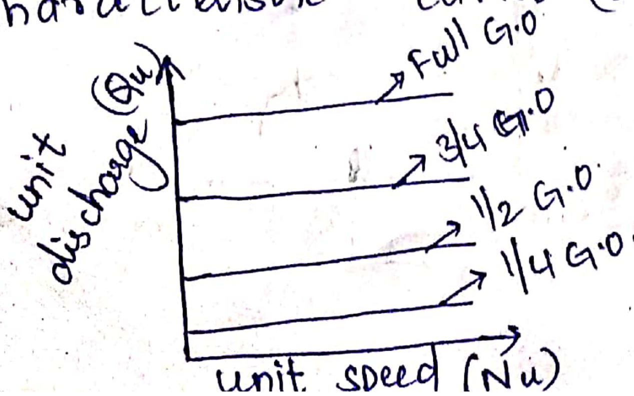
Important parameters which are varied during a test on a turbine:

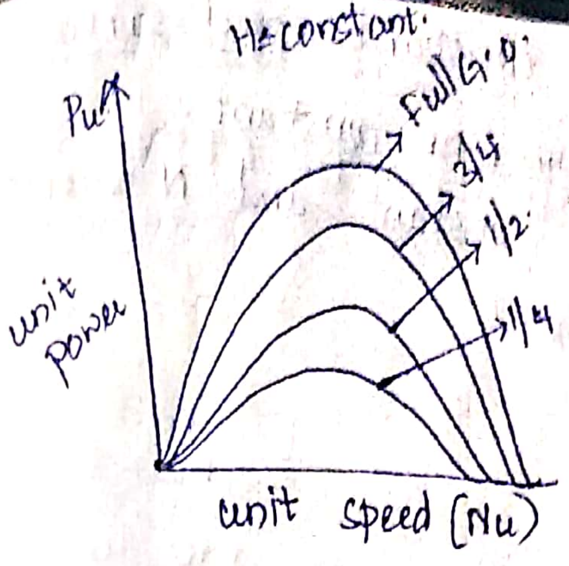
1. Speed (N):
2. Head (H)
3. Discharge (Q)
4. Power (P)
5. over all efficiency ($\eta_o$)
6. Gate opening ($G_o$)

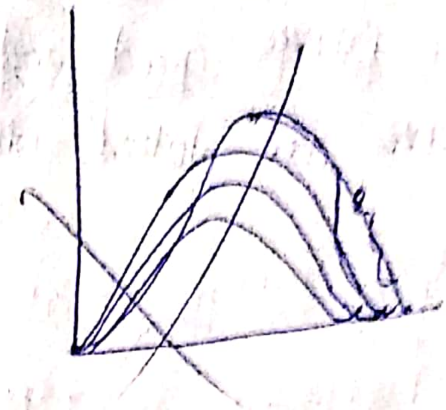
In these parameter N, H, Q are independent parameters. here H is constant

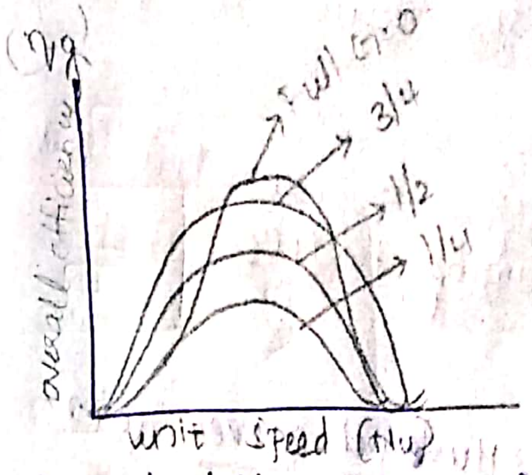
Types of characteristic curves:

1. Main characteristic curves (or) constant head curves.





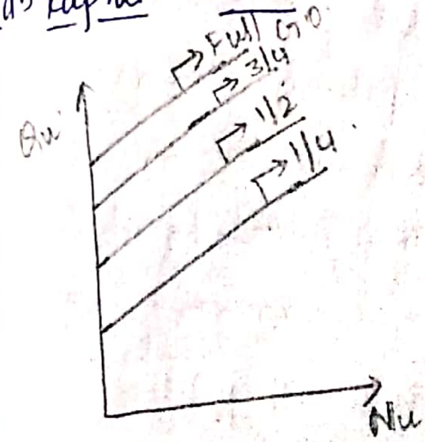



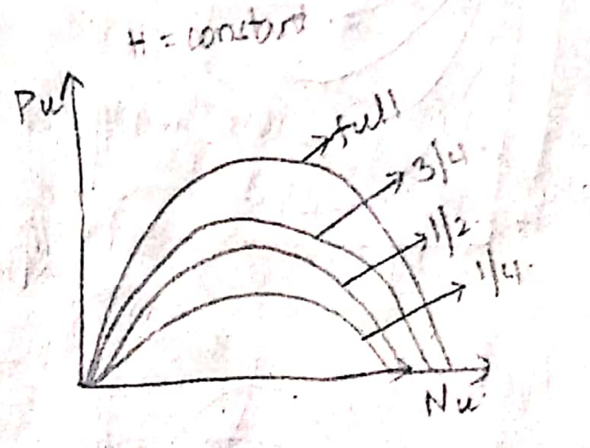



Main characteristic curves of pelton turbine

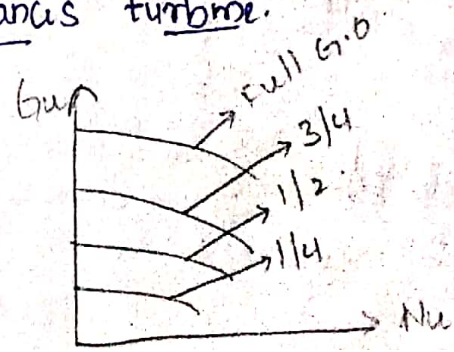
## characteristic curves for reaction turbine:

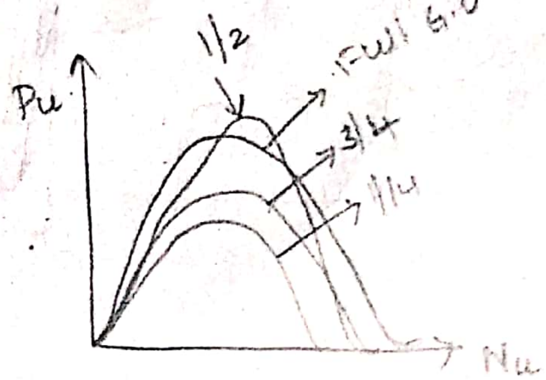
(a.) Kaplan turbine:




(b.) Francis turbine: H = constant

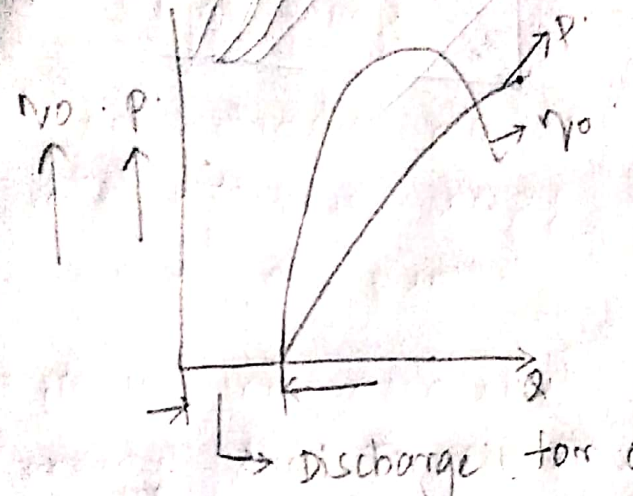


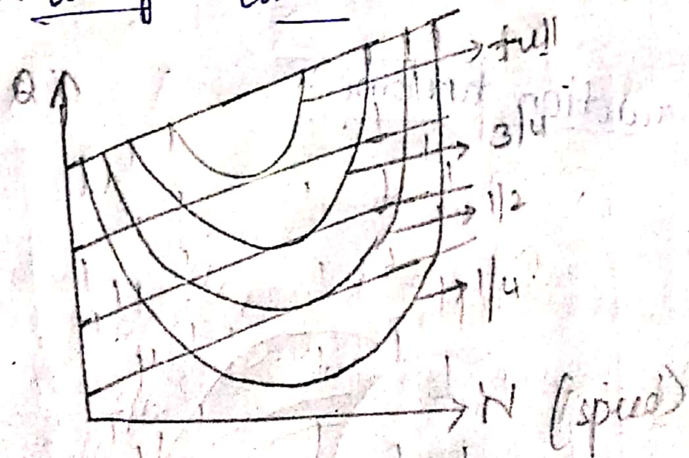
2. Operating characteristic curves or constant speed curves:

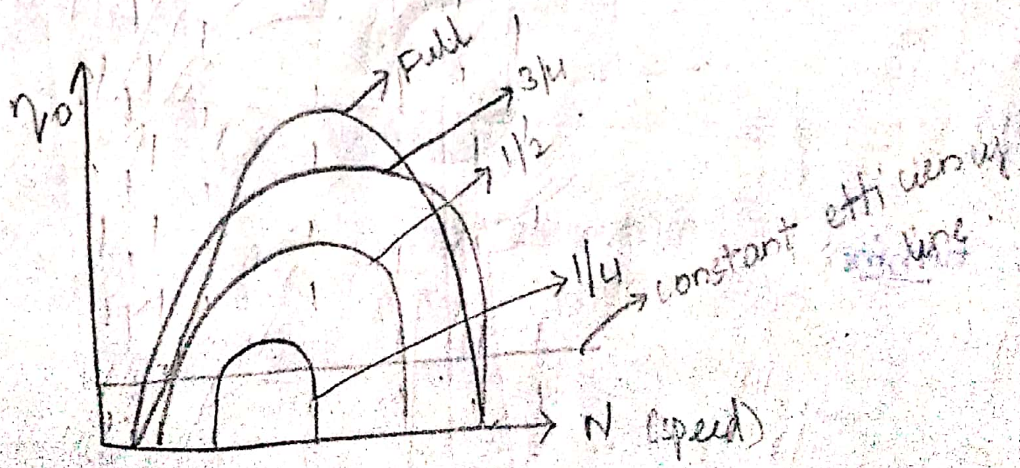
In these curves speed and head remains constant.

These curves are plotted when H is constant and N is constant




3. Constant efficiency curves (or) Husshel curves (or) Iso-efficiency curves:



# Centrifugal Pumps

Mr. P. Siva Sankar
Asst. Professor, CE.

## Introduction :

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

→ The hydraulic energy is in the form of pressure energy.

→ If the mechanical energy is converted into pressure energy by means of Centrifugal force acting on the fluid, the hydraulic machine is called "Centrifugal pump".

→ The centrifugal pump acts as a reverse of an inward radial flow reaction turbine. This means that flow in centrifugal pumps is in the radial outward directions.

→ The Centrifugal pump works on the principle of forced vortex flow which means that when a certain mass of liquid is rotated by an external torque, the rise of Pressure head of the rotating liquid takes place.

→ The rise in pressure head at any point of the rotating liquid is proportional to the Square of the tangential velocity of the liquid at that point (i.e., rise in Pressure Head $= \frac{V^2}{2g}$ )

→ Thus at the outlet of the impeller, where radius is more, the rise in pressure head will be more and the liquid will be discharged at the outlet with a high pressure head. Due to this high pressure head the liquid can be lifted to a high level. Ex: Sewage water, Chemicals etc.

## Centrifugal Pump :

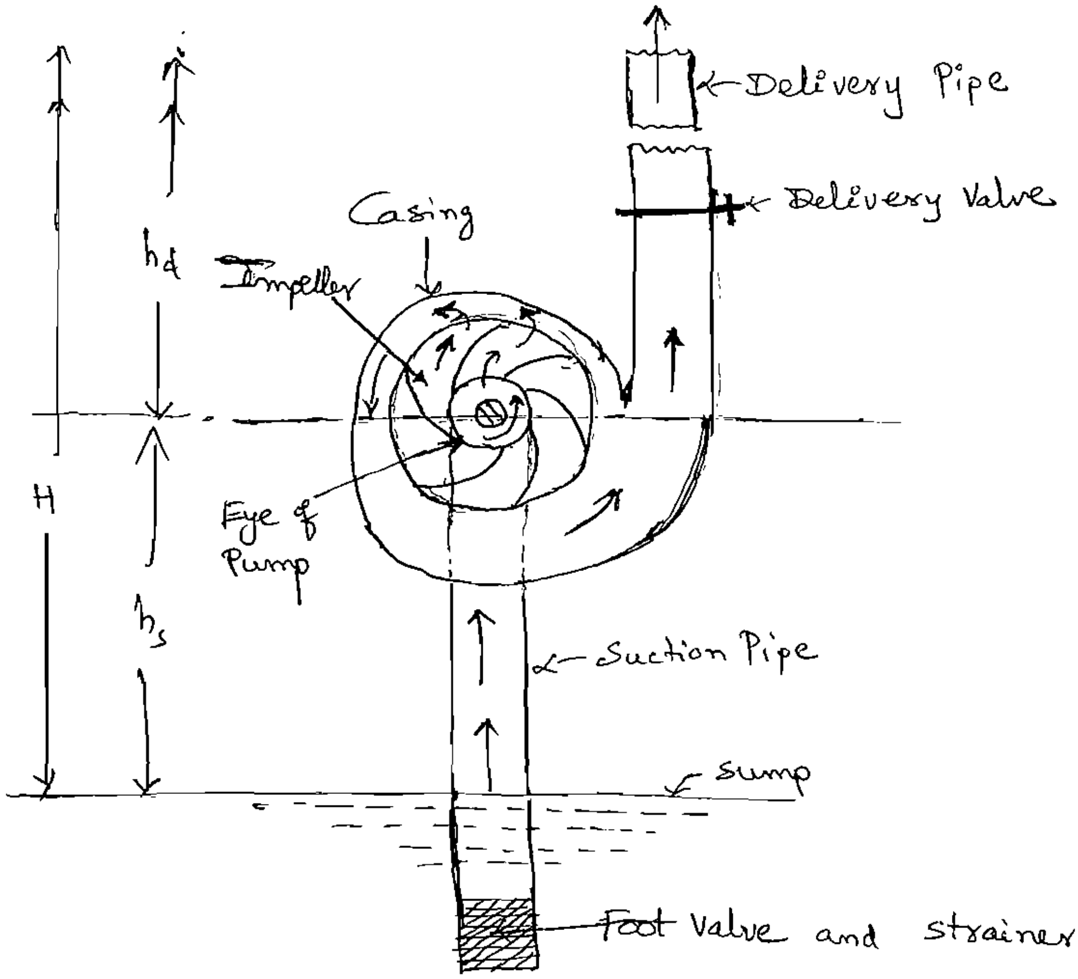



Fig1: Main Parts of a Centrifugal Pump

→ Priming is an operation in which liquid is completely filled in the chamber of pump so that air & gas or vapour from the portion of ~~the~~ pump is driven & no air pocket is left.

→ In volute pump cross sectional area results in developing a uniform velocity throughout the casing & free vortex is formed.

→ Centrifugal pump has high output and high efficiency.

## Main Parts of a Centrifugal Pump:

The Following are the main parts of a Centrifugal Pump.

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

1. Impeller:- The rotating part of a Centrifugal pump is Called 'Impeller'. It consists of a series of backward Curved vanes. The impeller is mounted on a shaft Which is connected to the shaft of an electric motor.

2. Casing:- The casing of a Centrifugal pump is similar to the casing of a reaction turbine. It is an air-tight passage surrounding the impeller and is designed in such a way that the K.E. of the water discharged at the outlet of the impeller is converted, into pressure energy before the water leaves the casing and enters the delivery pipe.

The Following three types of casings are commonly adopted:

(a) Volute casing as shown in fig. 1
(b) Vortex Casing as shown in fig. 2
(c) Casing with guide blades as shown in fig. 3.

(a) Volute Casing: Fig. 1 shows the volute casing, which surrounds the impeller. It is of spiral type in which area of flow increases gradually. The increase in area of flow decreases the velocity of flow. The decrease in velocity increases the pressure of water flowing through the casing. It has been observed that in case of volute casing, the efficiency of the pump increases slightly as a large amount of energy is lost due to the formation of eddies in this type of casing.

b, Vortex Casing: If a circular chamber is introduced between the casing and impeller as shown in fig (2). the casing is known as vortex casing. By introducing the circular chamber, the loss of energy due to the formation of eddies is reduced to a considerable extent. Thus the efficiency of the pump is more than the efficiency when only volute casing is provided.

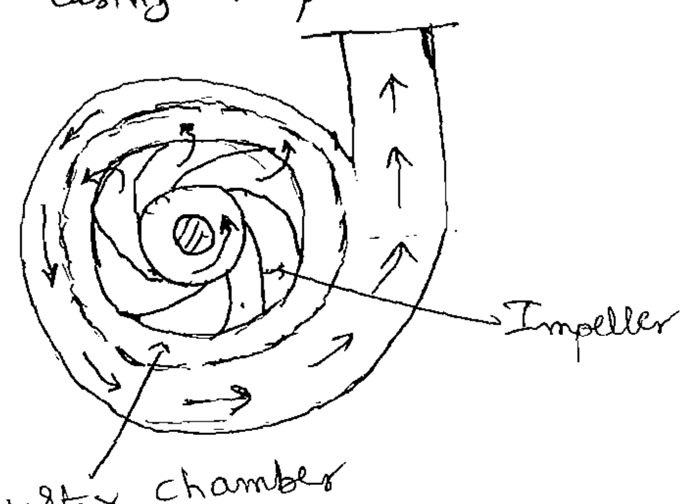



fig 2 Vortex Casing.

(c) Casing with Guide Blades: This casing is shown in Fig. 3. in which the impeller is surrounded by a series of guide blades mounted on a ring which is known as diffuser. The guide vanes are designed in such a way that the water from the impeller enters the guide vanes without stock. Also the area of the guide vanes increases, thus reducing the velocity of flow through guide vanes and consequently increasing the pressure of water.

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

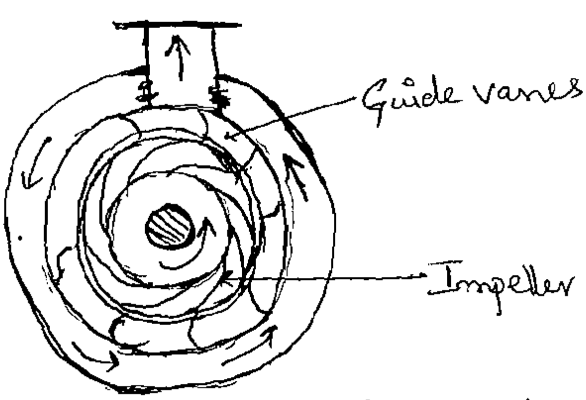



Fig 3. Casing with Guide Blades.

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

## ★ Workdone by the Centrifugal Pump (or By Impeller) on water

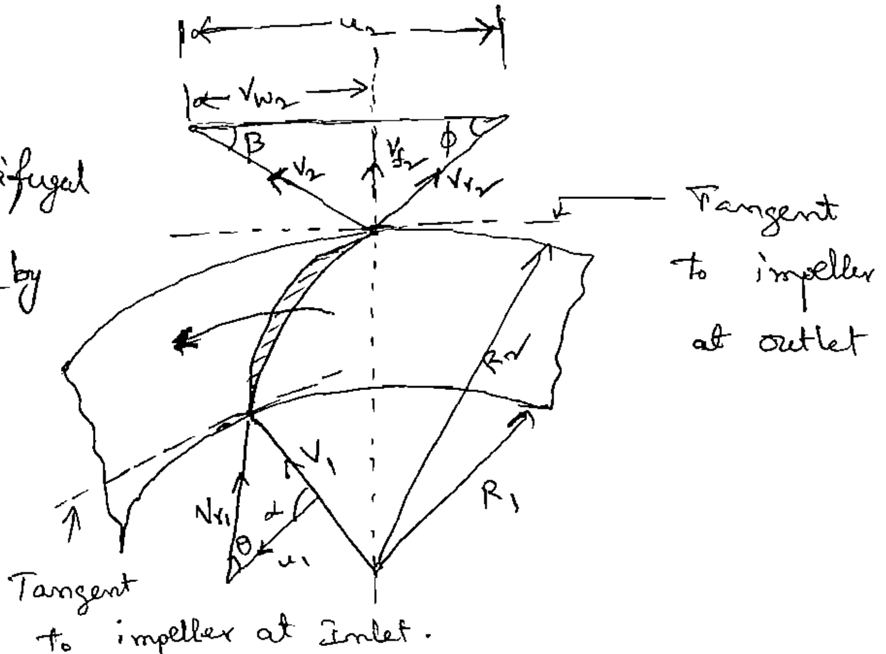



Fig 4: Velocity triangles at inlet & outlet.

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

fig. 4 shows the velocity triangles at the inlet and outlet tips of the vanes fixed to impeller.

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

$D_1$ = Diameter of impeller at inlet

$u_1$ = Tangential velocity of impeller at inlet

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

$D_2$ = Dia. of the impeller at outlet

$u_2$ = Tangential velocity of impeller at outlet $= \frac{\pi D_2 N}{60}$

$V_1$ = Absolute velocity of water at inlet

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

$\alpha$ = Angle made by absolute 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 corresponding values at outlet.

A centrifugal pump is the reverse of a radially inward flow reaction turbine. But in case of a radially inward flow reaction turbine, the workdone by the water on the runner per second per unit weight of the water striking per second is given by eq. as

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

∴ Workdone by the impeller on the water per second per unit weight of water striking per second

$$= -\left[\text{Workdone in case of turbines}\right]$$

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

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

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

Workdone by impeller on water per second

$$= \frac{W}{g}V_{w_2} \cdot u_2$$

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

where $Q$ = Volume of water

$Q$ = Area x velocity of flow

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

$$= \pi D_2 B_2 V_{f_2}$$

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

UNIT-6

# CENTRIFUGAL PUMPS & RECIPROCATING PUMPS

* Cavitation:

Cavitation is defined as the phenomenon of formation of vapour bubbles of a flowing liquid in a region where the pressure of the liquid falls below it's vapour pressure and the sudden collapsing of these vapour bubbles in a region of higher pressure. When the vapour bubbles collapse, a very high pressure is created. These cavities are formed on the metallic surface and also considerable noise and vibrations are produced.

* precautions against cavitation:

i.) The pressure of the flowing liquid in any part of the hydraulic system should not be allowed to fall below it's vapour pressure. If the flowing liquid is water, then the absolute pressure head should not be below 2.5 mts of water.

ii) The special materials or coatings such as aluminium-bronze and stainless steel, which are cavitation resistant materials, should be used.

* Effects of cavitation:

i.) The metallic surfaces are damaged and cavities are formed on the surfaces.

ii) Due to sudden collapse of vapour bubble, Considerable noise and vibrations are produced.

iii) The efficiency of a turbine decreases due to cavit-tion. Due to pitting action, the surface of the turbin blade becomes rough and force exerted by water on the Surface turbine blades decreases. Hence the w done by water or output horse power becomes less and thus efficiency decreases.

## * Characteristic curves of pumps:

Characteristic curves of centrifugal pumps are defined as those curves which are plotted from the results of a number of tests on the pump. The following are the important characteristics curves for pumps.

### 1. Main Characteristic curves:

The main Characteristics curves of a 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 (Hm) is Constant. And for plotting curves of power versus speed, the manometric head and discharge are kept constant

Result : i) Power lost in the nozzle = 10.16 kW

ii) Power lost due to hydraulic resistance in the runner = 10.69 kW.

3. (a) Explain the characteristics curves of a pumps and their significance !

A Characteristic Curves of Centrifugal pumps are defined as 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 are the important characteristic curves of pumps :-

1. Main characteristic curves
2. Operating characteristic Curves, and
3. Constant efficiency Or Muschel curves.

1. Main characteristic Curves:- It consists of variation of head (manometric head, $H_m$), Power and discharge with respect to speed.

For plotting the curves of discharge versus speed, manometric head ($H_m$) is kept constant.

For plotting the curves of power versus speed, the manometric head and discharge are kept constant.

For plotting the graph $H_m$ versus speed (N), the discharge is kept constant. It is clear that $H_m \propto N^2$.

Fig. shows main characteristic curves of a pump.

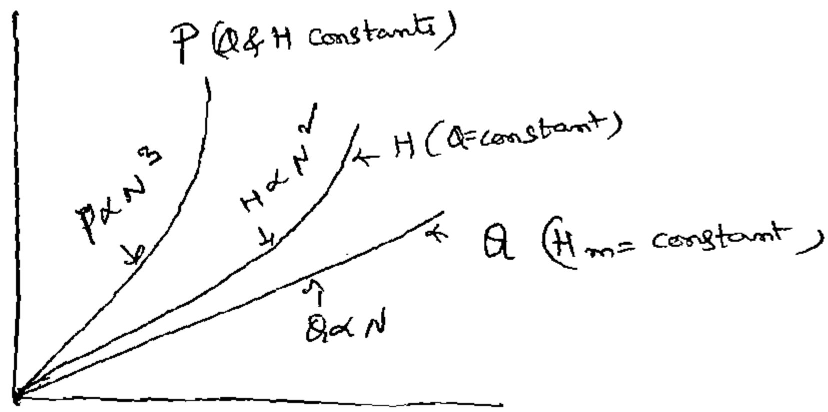
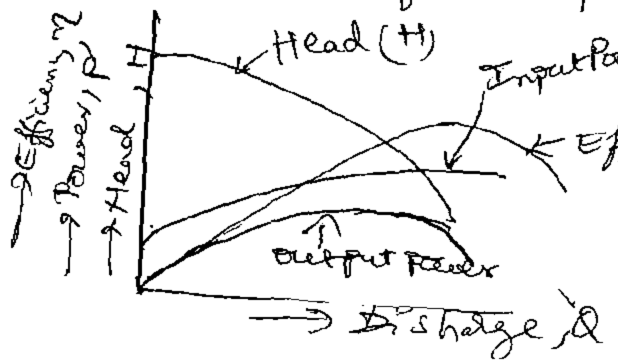



Fig: Main characteristic curves of a pump.

2. 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. Fig shows the operating characteristic curves of a pump.




Speed = Constant.

Here the Input Power curve is not starts from the origin, i.e, $Q = 0$

The Output Power Curve is starts from the origin

$Q = 0$ , $\rho g Q H = 0$

The head curve will have max. value of head, when $Q=0$

The Efficiency is starts from the origin as at

$Q = 0$, $\eta = 0$ $\left[\because \eta = \frac{\text{Output}}{\text{Input}}\right]$

3. Constant Efficiency Curves:- It consists of a plot for discharge Vs Head and another plot for discharge Vs efficiency. for different speed are used.

For plotting the constant efficiency curves (also known as Iso-efficiency curves), horizontal lines representing constant efficiencies are drawn on the $\eta \sim Q$ curves. The point, at which these lines cut the efficiency curves at various speeds are transferred to the corresponding $H \sim Q$ curves. The points having the same efficiency are then joined by smooth curves. These smooth curves represent the iso efficiency curves.

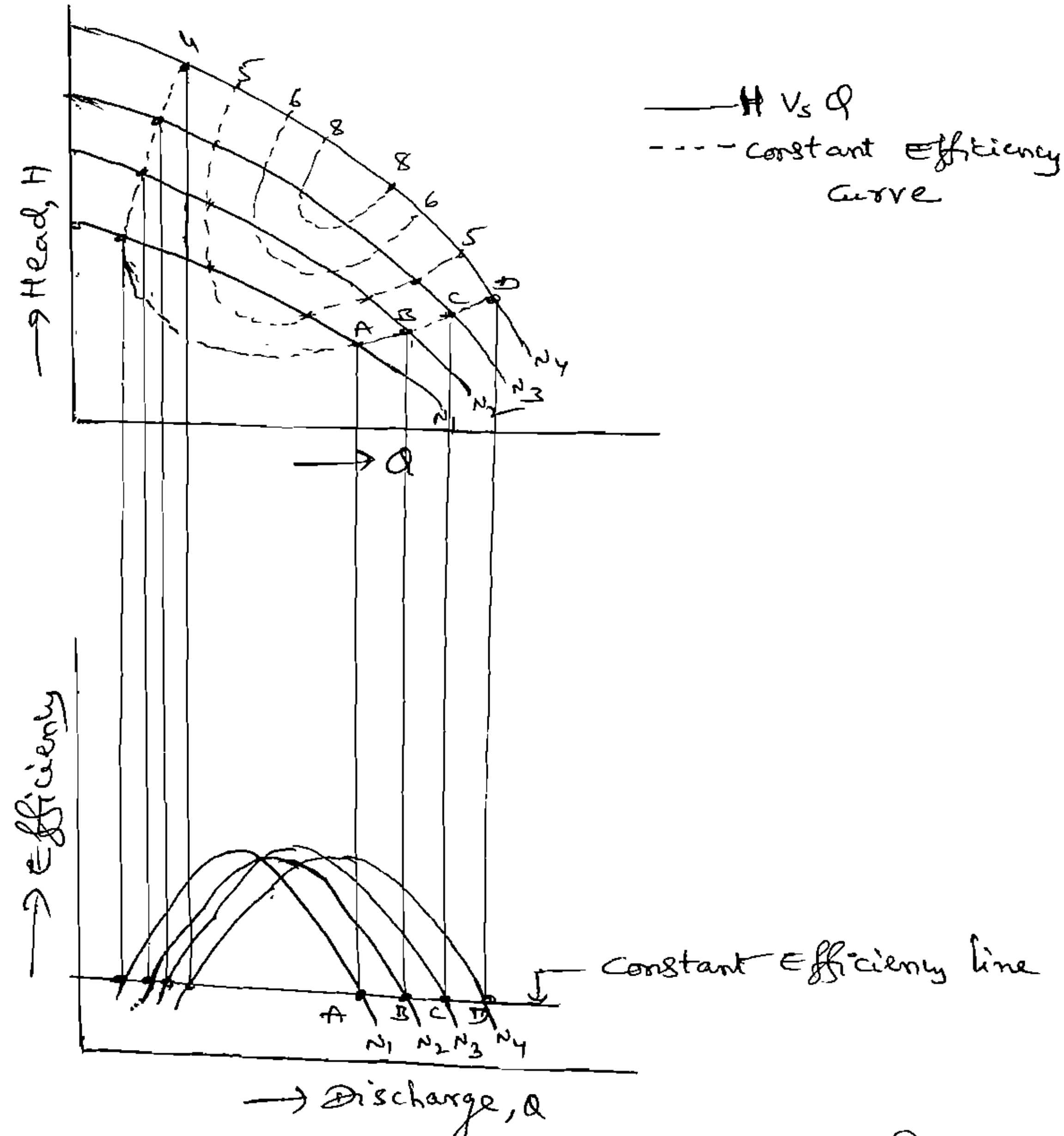



fig: Constant Efficiency Curves of a Pump.

3(b) A Single acting reciprocating pump running at 30 r.p.m., delivers 0.012 $m^3/sec$ of water. The diameter of the piston is 25 cm and stroke length is 50 cm. Determine

i) The theoretical discharge of the pump

ii) Co-efficient of discharge and

iii) slip and Percentage of Slip of the pump.

Given Data:

Speed of the pump, $N = 30$ rpm.

Actual Discharge, $Q_{act} = 0.012\ m^3/s$

Dia. of Piston, $D = 25\ cm = 0.25\ m$

∴ Area, $A = \frac{\pi}{4}(D)^2 = 0.049\ m^2$

To find:

i) The theoretical discharge of the pump

ii) Coefficient of discharge ($C_d$) and

iii) Slip and the percentage slip of the pump.

Formula used:

i) Theoretical discharge $Q_{th} = \frac{A \times L \times N}{60}\ m^3/s$

ii) Co-efficient of discharge, $C_d = \frac{Q_{act}}{Q_{the}}$

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

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

Calculation: i) $Q_{th} = \frac{ALN}{60} = \frac{0.049 \times 0.5 \times 30}{60}$

$= 0.0122\ m^3/sec.$

ii) $C_d = \dfrac{Q_{act}}{Q_{th}} = \dfrac{0.012}{0.0122} = 0.98$

iii) $Slip = Q_{th} - Q_{act} = 0.0122 - 0.012$

$= 2 \times 10^{-4} \; m^3/s$

iv) $Percentage\ slip = \dfrac{Q_{th} - Q_{act}}{Q_{th}} \times 100$

$= \dfrac{0.0122 - 0.012}{0.0122} \times 100$

$= 1.639\ \%$

Result:

i) $Q_{th} = 0.0122 \; m^3/s$

ii) $C_d = 0.98$

iii) $Slip = 2 \times 10^{-4} \; m^3/sec$

iv) $Percentage\ slip = 1.639\ \%$

# Definitions of Heads and efficiencies of a Centrifugal Pump :-

* **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 called suction lift and is denoted by '$h_s$'.

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

The vertical distance b/w the centre line of the pump and the water surface in the tank to which water is delivered is known as delivery head.

* **Static Head ($H_s$) :-**

The sum of suction head and delivery head is known as static head. This is represented by '$H_s$' and is written as

$$H_s = h_s + h_d.$$

* **Manometric Head ($H_m$) :-**

The manometric head is defined as the head against which a centrifugal pump has to work. It is denoted by '$H_m$'. It is given by following expressions:

$$H_m = h_s + h_d + h_{f_s} + h_{f_d} + \frac{V_d^2}{2g}$$

where $h_s$ = suction head, $h_d$ = Delivery head,

$h_{f_s}$ = Frictional head loss in suction pipe,

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

$V_d$ = Velocity of water in delivery pipe.

## * Efficiencies of a centrifugal pump :-

In case of a C.P, the power is transmitted from the shaft of the electric motor to the shaft of the pump and then to the impeller. From the impeller, the power is given to the water. Thus power is decreasing from the shaft of the pump to the impeller and then to the water. The following are the important efficiencies of a centrifugal pump:

a). manometric efficiency, $\eta_{man}$ b). mechanical efficiency, $\eta_m$ an

c) overall efficiency, $\eta_o$.

a). manometric efficiency ($\eta_{man}$) :-

The ratio of the manometric head to the head imparted by the impeller to the water is known as manometric efficiency. Mathematically, it is written as

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

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



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

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

The power at the impeller $= \frac{\text{workdone by impeller per second}}{1000}$ kw

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

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

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 c.p is known as mechanical efficiency. It is written as.

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

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

$$= \frac{W}{g} \times \frac{V_{w2} u_2}{1000}$$

$$\eta_m = \frac{\frac{W}{g}\left(\frac{V_{w2} u_2}{1000}\right)}{S.P}$$

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.

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

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

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

= S.P of the pump

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

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

*

## 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 metre of liquid per second against a head of one metre. It is denoted by '$N_s$'

## Expression for specific speed of a pump :-

The discharge (Q) for a centrifugal pump is given by the relation

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

$$= \pi D \times B \times V_f \longrightarrow ①$$

(or)

$$= D \times B \times V_f$$

where $D$ = Diameter of the impeller of the pump

$B$ = width of the impeller

We know that $B \propto D$

From equ ① we have $Q \propto D^2 \times V_f \longrightarrow$ (ii)

We also know that Tangential velocity is given by

$$U = \frac{\pi D N}{60} \propto DN \longrightarrow (iii)$$

Now the tangential velocity (U) and velocity of flow ($V_f$) are related to the manometric head ($H_m$) as

$$U \propto V_f \propto \sqrt{H_m} \longrightarrow (iv)$$

Sub the value of U in equ (iii) we get

$$\sqrt{H_m} \propto DN \text{ (or) } D \propto \frac{\sqrt{H_m}}{N}$$

Sub the values of D in equ (ii)

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

$$\propto \frac{H_m}{N^2} \times \sqrt{H_m} \qquad \therefore V_f \propto \sqrt{H_m}$$

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

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

where K is constant

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

Sub these values in equ (v) we get

$$1 = K \frac{1^{3/2}}{N_s^2} = \frac{K}{N_s^2}$$

$$\therefore K = N_s^2$$

Sub value of K in equation (v), we get

$$Q = N_s^2 \frac{H_m^{3/2}}{N^2} \quad (or) \quad N_s^2 = \frac{N^2 Q}{H_m^{3/2}}$$

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

## Net positive Suction Head (NPSH) :-

The term NPSH is very commonly used in the pump industry. Actually the minimum suction conditions are more frequently specified in terms of NPSH.

The net positive suction Head (NPSH) is defined as the absolute pressure head at the inlet to the pump, minus the vapour pressure head plus the velocity head.

∴ NPSH = Absolute pressure head at inlet of the pump − vapour pressure head + velocity head.

$$= \frac{P_1}{\rho g} - \frac{P_v}{\rho g} + \frac{V_s^2}{2g} \longrightarrow (1)$$

The absolute pressure head at inlet of the pump is given by as

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

Sub this equ. (1) we get

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

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

$$= H_a - H_v - h_s - h_{fs} \qquad \because \frac{P_a}{\rho g} = H_a$$

$$\because \frac{P_v}{\rho g} = H_v$$

$$NPSH = \left[(H_a - h_s - h_{fs}) - H_v\right] \longrightarrow (2)$$

The right hand side of equ (2) is the total suction head. Hence NPSH is equal to total suction head. Thus NPSH may also be defined as the total head required to make the liquid flow through the suction pipe to the pump impeller.

# Reciprocating Pumps

**Introduction:** The pumps as the hydraulic machin which convert the mechanical energy into hydrauli energy which is mainly in the form of pressure ene If the mechanical energy is converted into hydraul energy, by means of centrifugal force acting on the liquid, the pump is known as centrifugal pump. But if the mechanical energy is converted into hydraulic energy (or pressure energy) by sucking the liquid into a cylinder in which a piston is reciprocating (moving backwards and forwards), which exerts the thrust on the liquid and increase its hydraulic energy (Pressure energy), the pump is known as reciprocating pump.

## Main Parts of a Reciprocating pump:-

The following are the main parts of a reciprocating pump as shown in fig.

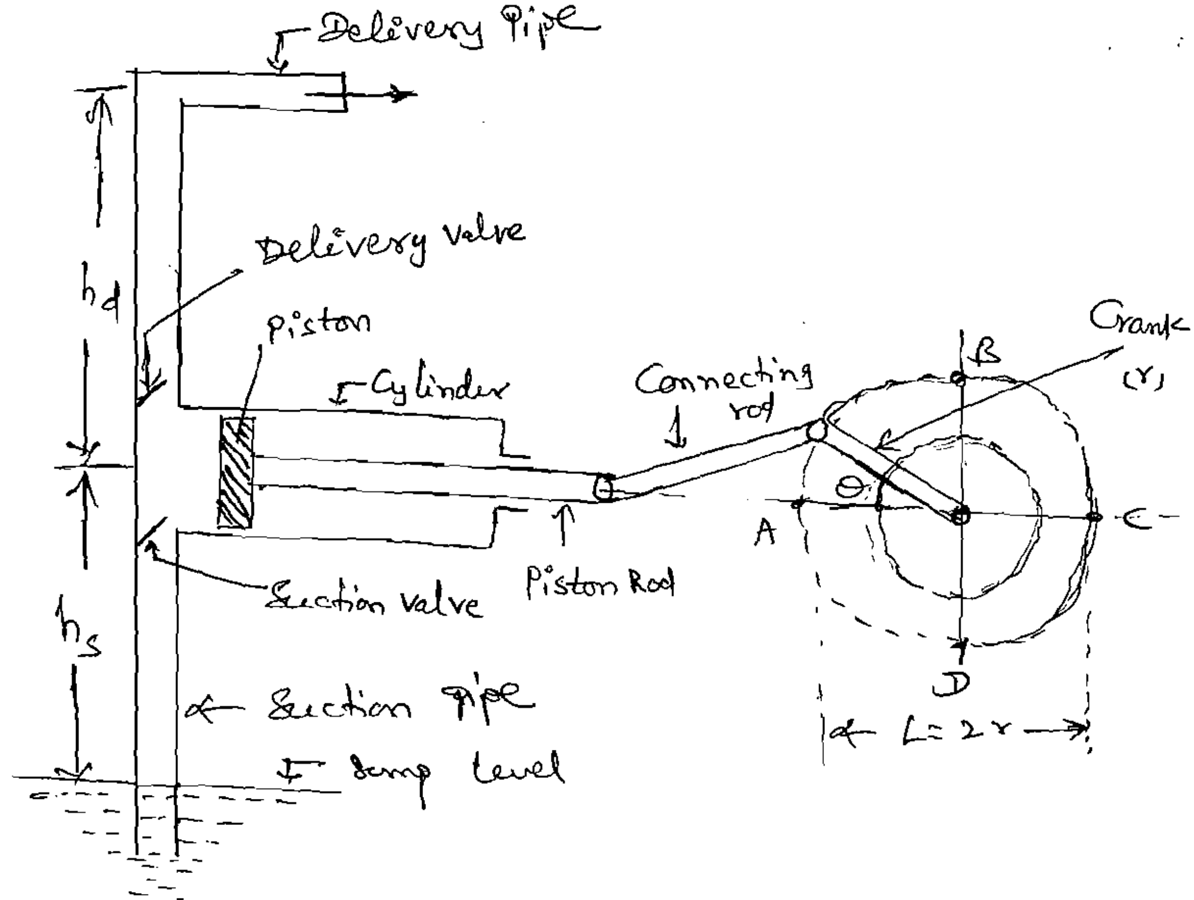


Fig: Main parts of a Reciprocating pump.

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

Working of a R.P: Fig. Shows a single acting R.P, which consists of a piston which moves forwards & backwards in a close fitting cylinder. The movement of piston is obtained by connecting the piston rod to crank by means of a connecting rod.

(49)

The crank is rotated by means of an electric motor. Suction and delivery pipes with suction valve and delivery valve are connected to the cylinder. The suction and delivery valves are one way valves & non-return valves, which allow the water to flow in one-direction only. Suction valve allows water from suction pipe to the cylinder which delivery valve allows water from cylinder to delivery pipe only.

## Discharge through a Reciprocating Pump:-

Consider a single acting reciprocating pump as shown in Fig. (In previous page).

Let $D$ = Dia. of the cylinder

$A$ = Cross-sectional area of the piston cylinder

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

$r$ = Radius of crank

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

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

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

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

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

= Area × Length of stroke = $A \times L$

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

∴ Discharge of the pump per second,

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

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

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

Weight of the water delivered per second

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

Workdone by Reciprocating Pump:- Workdone by the R.P. per second is given by the reaction as

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

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

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

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

Substituting the value of W in eq i, we get

$$\text{Workdone per second} = \frac{\rho g\, ALN}{60} \times (h_s + h_d)$$

∴ Power required to drive the pump, in kW

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

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

* Indicator diagram:-

→ The Indicator diagram for a Reciprocating pump is defined as 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. As the maximum distance travelled by the piston is Equal to the stroke length and hence the indicator diagram is a graph between pressure head and stroke length of the piston for one complete revolution. the pressure head is taken as ordinate and stroke length as abcissa.

Workdone by pump ∝ Area of indicator diagram.

* Thomas cavitation factor for Reaction Turbines:-

→ prof. D. Thoma suggested a dimensionless number, called after his name thoma's cavitation factor σ (sigma), which can be used for determining the region where cavitation takes place in reaction turbines. the mathematical Expression for the thoma's cavitation factor is given by.

$$\sigma = \frac{H_b - H_s}{H} = \frac{(H_{atm} - H_v) - H_s}{H} \longrightarrow (1)$$

where $H_b$ = Barometric pressure head in m of water,

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

$H_v$ = Vapour pressure head in m of water,

$H_s$ = Suction pressure at the outlet of reaction turbine in m of water or height of turbine runner above the tail water surface,

$H$ = net head on the turbine in m.

* Thoma's cavitation factor for centrifugal pumps:

→ The mathematical Expression for thoma's cavitation factor for centrifugal pump is given by

$$\sigma = \frac{(H_b) - H_s - h_{Ls}}{H} = \frac{(H_{atm} - H_v) - H_s - h_{Ls}}{H} \longrightarrow ②$$

where, $H_{atm}$ = Atmospheric pressure head in m of water or absolute pressure head at the liquid surface in pump,

$H_v$ = Vapour pressure head in m of water.

$H_s$ = Suction pressure head in m of water.

$h_{Ls}$ = Head lost due to friction in suction pipe, and.

$H$ = Head developed by the pump.

→ The value of Thoma's cavitation factor ($\sigma$) for a particular type of turbine or pump is calculated from Eqns ① and ②. This value of Thoma's cavitation factor ($\sigma$) is compared with critical cavitation factor ($\sigma_c$) for that type of turbine pump. If the value of $\sigma$ is greater than $\sigma_c$, the cavitation will not occur in that turbine or pump. The critical cavitation factor ($\sigma_c$) may be obtained from tables or Emperical relationships.

→ The following Emperical relationships are used for obtaining the value of $\sigma_c$ for different turbines:

For Francis turbines, $\sigma_c = 0.625\left(\frac{N_s}{380.78}\right)^2$

$$\simeq 431 \times 10^{-8} N_s^2$$

For propeller turbines, $\sigma_c = 0.28 + \left[\frac{1}{7.5}\left(\frac{N_s}{380.78}\right)^3\right]$

In the above Expression $N_s$ is in (r.p.m., kW, m) units. If $N_s$ is in (r.p.m., h.p., m) units the Emperical relationships would be as follows.

For Francis turbines, $\sigma_c = 0.625\left(\frac{N_s}{444}\right)^2 \simeq 317 \times 10^{-8} \times N_s^2$

For propeller turbines, $\sigma_c = 0.28 + \left[\frac{1}{7.5}\left(\frac{N_s}{444}\right)^3\right]$

## * Slip of reciprocating pump:-

→ Slip of a pump is defined as the difference between the theoretical discharge and actual discharge of the pump. The discharge of a single-acting pump given by Equation, $Q = A \times L \times \frac{N}{60} = \frac{ALN}{60}$ and of a double-acting pump given by Equation,

$Q = \left(\frac{\pi}{4}D^2 + \frac{\pi}{4}D^2\right) \times \frac{L \times N}{60} = 2 \times \frac{\pi}{4}D^2 \times \frac{L \times N}{60} = \frac{2ALN}{60}$ are theoretical discharge. The actual discharge of a pump is less than the theoretical discharge due to leakage. The difference of the theoretical discharge and actual discharge is known as slip of the pump. Hence, Mathematically,

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

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

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

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

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 that case, the slip of the pump is known as negative slip.

"Negative Slip occurs when delivery pipe is short,

Suction pipe is long and pump is running at high speed."

## * Air Vessels :-

→ An air vessel is a closed chamber containing compressed air in the top portion and liquid (or water) at the bottom of the chamber. At the base of the chamber there is an opening through which the liquid may flow into the vessel or out from the vessel. When the liquid enters the air vessel, the air gets compressed further and when the liquid flows out the vessel, the air will expand in the chamber.

→ An air vessel is fitted to the suction pipe and to the delivery pipe at a point close to the cylinder of a single-acting reciprocating pump:

(i) To obtain a continuous supply of liquid at a uniform rate,

(ii) To save a considerable amount of work in overcoming the frictional resistance in the suction and delivery pipes, and

(iii) To run the pump at a high speed without separation.

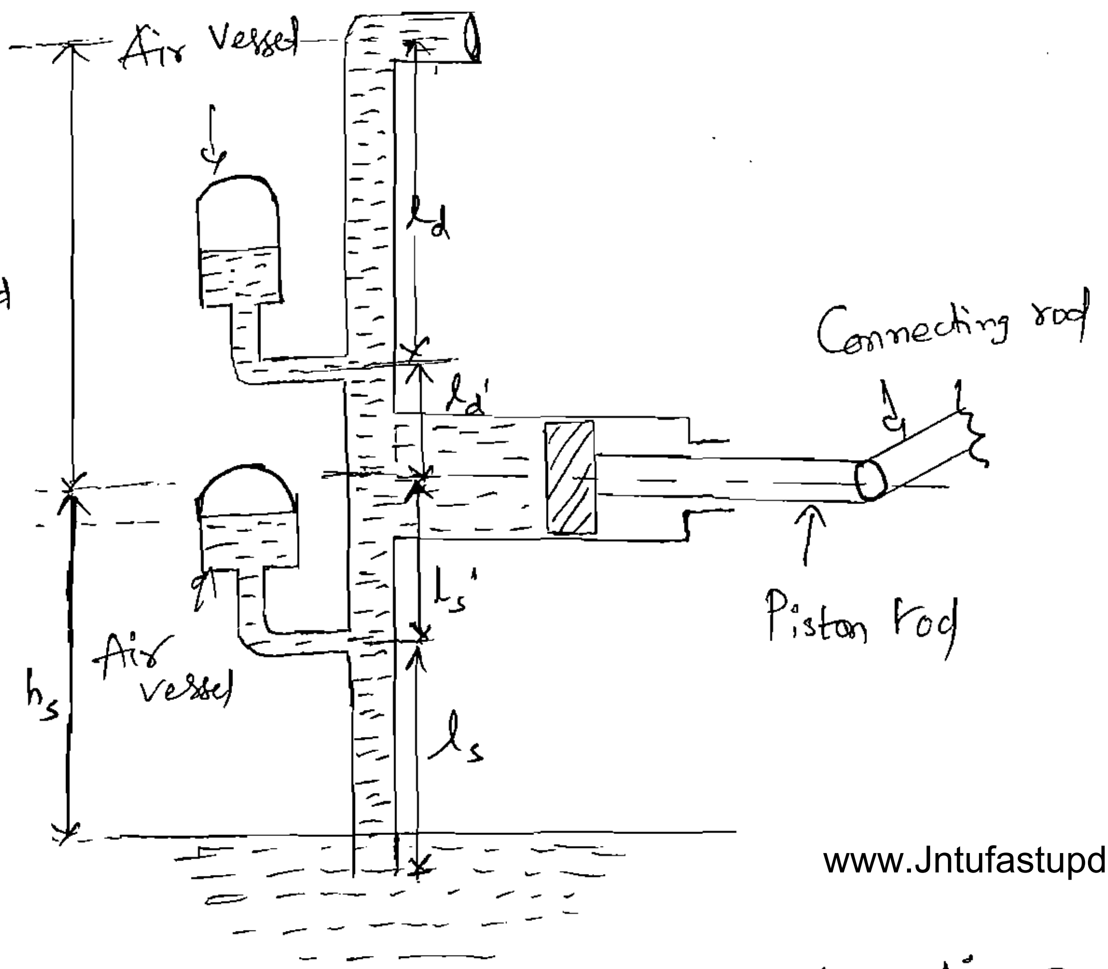




Fig: Air Vessels fitted to reciprocating pump

②

Fig. shows the single-acting reciprocating pump to which air vessels are fitted to the suction and delivery pipes. The air vessels act like an intermediate reservoir. During the first half of the suction stroke, the piston moves with acceleration, which means the velocity of water in the suction pipe is more than the mean velocity and hence the discharge of water entering the cylinder will be more than the mean discharge. This excess quantity of water ~~entering the cylinder will be more than the mean discharge~~ will be supplied from the air vessel to the cylinder.

During the second half of the suction stroke, the piston moves with retardation and hence velocity of flow in the suction pipe is less than the mean velocity of flow. Thus, the discharge entering the cylinder will be less than the mean discharge. Thus, the excess water flowing in suction pipe will be stored into air vessel, which will be supplied during the first half of the next suction stroke.

Let $A$ = cross sectional area of the cylinder

$a$ = Cross-sectional area of suction & delivery pipe.

$l_d$ = Length of the delivery pipe beyond the air vessel.

$l_d'$ = Length of the delivery pipe between cylinder & air vessel.

$l_s'$ = Length of the suction pipe b/w cylinder & air vessel

$l_s$ = Length of the suction pipe below air vessel.

$h_{ad}$ = Pressure head due to acceleration in delivery Pipe.

$h_{as}$ = Pressure head due to acceleration in suction pipe.

$h_{fd}$ = Loss of head due to friction in delivery pipe ~~by~~ beyond the air vessel.

$h_{fd}'$ = Loss of head due to friction in delivery pipe between cylinder and air vessel.

$h_{fs}$ = Loss of head due to friction in suction pipe below the air vessel

$h_{fs}'$ = Loss of head due to friction in suction Pipe between cylinder and air vessel.

(a) Pressure head in the cylinder during delivery stroke:-

i) At the beginning of the delivery stroke, $\theta = 0^\circ$, $\sin\theta = 0$ & $\cos\theta = 1$ and hence total pressure head

$$= h_d + \frac{l_d'}{g} \times \frac{A}{a_d} \omega^2 r + \frac{4 f l_d}{d_d \times 2g} \times \left(\frac{A}{a_d} \times \frac{\omega r}{\pi}\right)^2 + \frac{1}{2g}\left(\frac{A}{a_d} + \frac{\omega}{\pi}\right.$$

ii) In the middle of the stroke, $\theta = 90^\circ$, $\sin\theta = 1$ and $\cos\theta = 0$ and total pressure head.

$$= h_d + \frac{4f \times l_d'}{d_d \times 2g} \times \left(\frac{A}{a_d} \omega r\right)^2 + \frac{4f \times l_d}{d_d \times 2g} \times \left(\frac{A}{a_d} \times \frac{\omega r}{\pi}\right)^2 + \frac{1}{2g}\left(\frac{A}{a_d} + \frac{\omega}{\pi}\right.$$

iii) At the end of the delivery stroke, $\theta = 180^\circ$, $\sin\theta = 0$ & $\cos\theta = -$ and hence total pressure head

$$= h_d - \frac{l_d'}{g} \times \frac{A}{a_d} \omega^2 r + \frac{4f \times l_d}{d_d \times 2g} \times \left(\frac{A}{a_d} \times \frac{\omega r}{\pi}\right)^2 + \frac{1}{2g}\left(\frac{A}{a_d} + \frac{\omega r}{\pi}\right)^2$$

* Comparison between Centrifugal Pumps and Reciprocating Pumps.

| Centrifugal Pump | Reciprocating Pump. |
|---|---|
| 1. The discharge is continuous & smooth | 1. The discharge is fluctuating & pulsating |
| 2. It can handle large quantity of liquid. | 2. It handles small quantity of liquid only. |
| 3. It can be used for lifting highly viscous liquids. | 3. It is used only for lifting pure water or less viscous liquids, |
| 4. It is used for large discharge through smaller heads. | 4. It is meant for small discharge and high heads. |
| 5. Cost of Centrifugal pump is less as compared to reciprocating pump. | 5. Cost of reciprocating pump is approximately four times the cost of Centrifugal pump. |
| 6. Centrifugal pump runs at high speed. They can be coupled to electric motor. | 6. Reciprocating pump runs at low speed. Speed is limited due to consideration of separation & cavitation. |
| 7. The operation of C.P. is smooth & without much noise. The maintenance cost is low. | 7. The operation of R.P. is complicated and with much noise. The maintenance cost is high. |
| 8. C.P. needs smaller floor area & installation cost is low. | 8. R.P. requires large floor area and installation cost is high. |
| 9. Efficiency is high | 9. Efficiency is low. |

Classification of Reciprocating Pumps :-

The reciprocating pumps may be classified as:

1. According to the water being in contact with one or both sides of the piston, and

2. According to the number of cylinders provided,

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

i) Single-acting pump, and

ii) Double-acting pump.

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

i) Single cylinder pump

ii) Double cylinder pump, and

iii) Triple cylinder pump.