HYDRAULIC MACHINES

6 Que.
PRE

1. overview of hydroelectric projects (HEP's)

2. Hydraulic Turbines → Impulse turbiness
   → Reaction Turbines.

3. Hydraulic pumps → centrifugal pump
   → Reciprocating pump.

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Hydraulic Machines

Energy producing

↓

Energy of Fluid

↓

Mechanical Energy

↓ Electrical generator

Electrical Enery

Ex. Hydraulic Turbine
Steam Turbine
Gas Turbine
Wind Turbine

Energy Consuming

↓

Electrical Energy

↓ Electric Motor

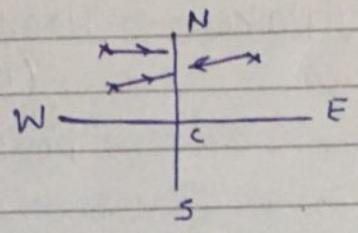
Mechanical Energy

↓ pumps, Fans
blowers, Compresser

Hydraulic Energy

① OVERVIEW OF HEP'S

1) Generation
2) Transmission — pencil
3) Distribution.



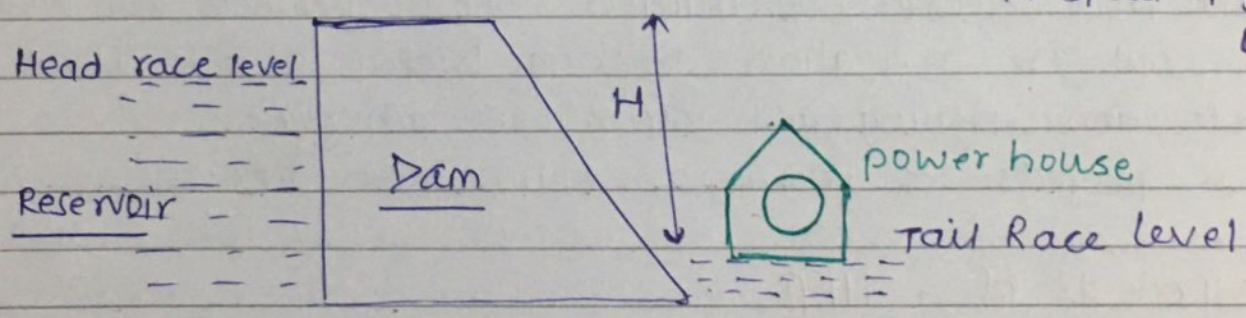
Supply	Demand	
500 MW	500 MW	$\Rightarrow f = 50$ Hz.
500 MW	600 MW	$\Rightarrow f < 50$ Hz
500 MW	400 MW	$\Rightarrow f > 50$ Hz.

Demand ↑ → price ↑
Demand ↓ → price ↓

$$N = \frac{60 \times f}{P}$$

P → pair of poles.

N → speed of generator (rpm)



Water power = $mgh = \rho Q g H$

load ↑, P_{gen} ↑ ⇒ Q ↑

$\Downarrow$

$\rho g Q H$

$\Downarrow$ Varies, others are fixed.

Waterways

Trash rack → Intake → Head race Tunnel → forebay
(Debris) (HRT) or power channel
⇓
कचरा

forebay ↓ penstock → Surge Tank → Turbine → Tail race → River

In HEP, potential energy of water is utalised to drive a turbine which intern runs the generator to produce electricity.

Apart from electricity production these projects can also be used for irrigation, drinking Water supply, flood moderation, fishing and recreational activities.

Such projects are known as Multipurpose projects.

Components of a HEP

① **Reservoir** Water available in catchment area is stored in reservoir so as to meet requirement of plant throughout the year.

Reservoir's can be natural as well as Artificial.

Natural reservoirs are – lakes in mountains

Artificial reservoirs are made by constructing a dam accross a river.

② **Trash rack** It is used to obstruct debris from going into the intake.

NOTE ① Foreway and Surge tank, both serve same purpose
② Surge tank shall be located close to the turbine.

③ Waterway It is a passage through which water is carried from reservoir to the power house.
It consists of Head race Tunnel (HRT) or canal, Forebay, penstock, Surge tank.

④ Forebay It is an enlarge sec^n @ the end of power channel (HRT)
it's function is to store temporary the water rejected by plant when the load is reduced and to meet the instaneous increase demand of water due to sudden increase in load.

It also helps in absorbing the sudden rise in pressure due to closer of vaulves, when load on turbine ↓
Thus it prevents HRT from water hammer effect.

⑤ Pen stock It is a closed conduit for supplying water under pressure from forebay to turbine
It is subjected to water hammer pressure due to fluatuation in turbine load. and to reduce this, a Surge TANK is provided.

⑥ Surge Tank It is a small reservoir fitted @ some opening in penstock to recieve the rejected flow when the load of turbine ↓ and thus it prevents pen stock from water hammer effect.

⑦ Tail Race it is a waterway to carry water ~~from~~ discharged by turbine to a suitable point where it can be safely disposed off. or stored for pumping back into the original reservoir.

Classification of HEP

Based on availability of Head.

① High Head plants :- Head > 250 m.
② Medium Head plants :- 30 < H < 250 m.
③ low head plants :- H < 30 m.

This classification is not based on any scientific criteria.

Based on load capacity

① Base load plant → Supply constant power

Ex: Thermal power plant, Hydropower plants with storage

② peak load plants Supply power during peak hours.

A storage type HEP is ideally suited for this purpose as it can be started in few minutes.

Storage & pondage

storage & pondage of water is req^r for regulation of flow of water so as to make it available in requicite ~~deficit~~ quantity. to meet the power demand @ a given time.

Storage:- it is impounding of considerable amount of excess runoff during seasons of surplus flow for use in dry season. This is accomplished by constructing a dam accross a river

pondage:- it is a regulating body of water in the form of relatively small pond or reservoir provided @ the plant. pondage is used to regulate the variable water flow to meet power demand.

Ex: forebay and surge tank.

- storage & pondage can be obtained from flow duration curve.

Based on function

i) Run of river plant (ROR)

No storage of water is done and whatever water in the river is available is utalisied. to run the turbine. RoR plants are suitable for perrinial rivers.

- RoR plants can be with or without pondage.

With pondage ⟶ peak load

Without pondage ⟶ Base load

plants with pondage are provided with a weir or barrage to accomodate sufficient storage to take care of load flauctuations.

ii) Storage plants These plants have storage provided by constructing Dam. These plants can meet peak load demands

iii) Pumped storage plants

This type of plant is provided when there is a shortage of water.

These plants generate power during peak hours. During off peak hours water is pumped back from Tail race channel to the reservoir.

These plants use reversible turbines which functions as both turbine in peak hrs as well as pump during off peak hours.

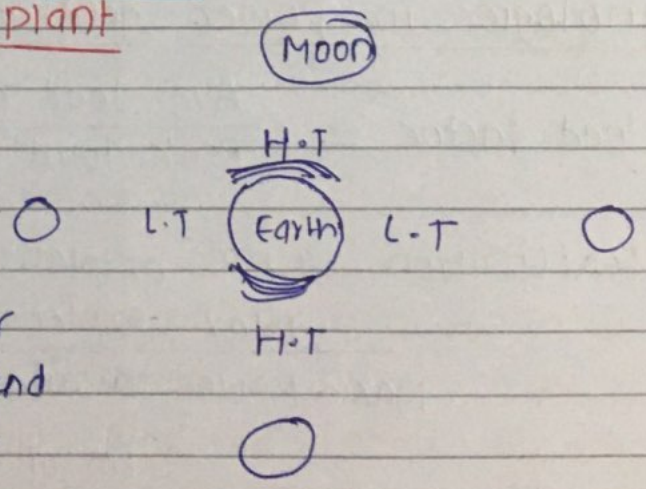
Such plants convert low value of peak energy into high value on peak energy hence economically viable.

* Suitable for peak load applications.

Based on Source of Energy

① river Based power plant

② Tidal power plant



These plants tap the energy of earth's rotation. These is rise in sea water during High tide period and fall during L.T period.

Water rises and falls twice a day hence Head on turbine varies with time cyclically.

- During H.T when water level along cost rises, electricity can be produced.
  electricity can also be produced when ~~H.T~~ water is retreating by changing the dir^n of rotation of turbine.

Turbines used are – Bulb turbine
Tubular turbine.

Advantages	limitations
1) Water	1) Very high initial cost
2) low Running cost	2) long gestation period.
3) Renewable source of energy	3) Far from load centre ⇓ High transmission cost
4) NO GHG emmisions	4) resettlement & issues.
5) Quick starting & closure ⇓ Suitable for peak load applications.	5) environmental flows.

Terminologies in power generation

(i) load factor $= \dfrac{\text{Avg. load during a certain period}}{\text{Peak load during that period}}$

(ii) Utilisation factor or plant use factor

$$\frac{\text{Max. power utilised}}{\text{Max. power available (= installed capacity)}}$$

(iii) Diversity factor $= \dfrac{\text{Sum of individual Max. demands}}{\text{Simultaneous Max. demand}}$

iv) Demand factor = $\frac{\text{Simultaneous Max. Demand}}{\text{Sum of individual Max. demands}}$

$= \frac{1}{\text{Diversity factor}}$

(v) primary power (firm power)

it is the net amount of power that is available from a plant on a garunteed basis.

vi) secondary power

excess power available above primary power during peak hours.

Qu. load on a hydal plant varies from a Min. of 10000 KW to a Max. of 35000 KW. Two turbo-generators of capacity 22000 KW each have been installed. Calculate

i) Total installed capacity of plant

ii) Max. demand

iii) Load factor

iv) utilisation or plant use factor

i) $2 \times 22000 = 44000$ KW

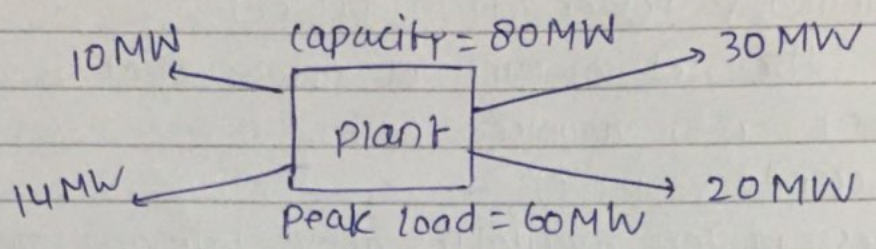
ii) 35000 KW

iii) $\frac{\frac{10000+35000}{2}}{35000} = 64.28\%$

iv) $\frac{35000}{44000} = 79.54\%$

Qu. peak load on a powerplant is 60 MW. The loads having Max. demands of 30 MW, 20 MW, 10 MW and 14 MW are connected to power plant capacity of power plant is 80 MW, Annual load factor is 0.5. Estimate.

i) Avg. load on power plant
ii) Energy supplied / year in (kWH)
iii) Demand factor
iv) Diversity factor



i) $\text{load factor} = \dfrac{\text{avg. load}}{\text{peak load}}$

$$0.5 = \frac{\text{avg. load}}{60}$$

$$\text{Avg. load} = 30\ MW$$

ii) $30000\ KW \times 8760 = 262800 \times 10^3\ KW\cdot h$

$\{8760 = 365 \times 24$

(iii) $\dfrac{60}{74} = 0.81$

iv) $\dfrac{74}{60} = 1.23$

2. HYDRAULIC TURBINES

Hydraulic Energy $\xrightarrow{\text{Hydraulic Turbine}}$ Mechanical Energy $\xrightarrow{\text{electrical generator}}$ Electrical Energy

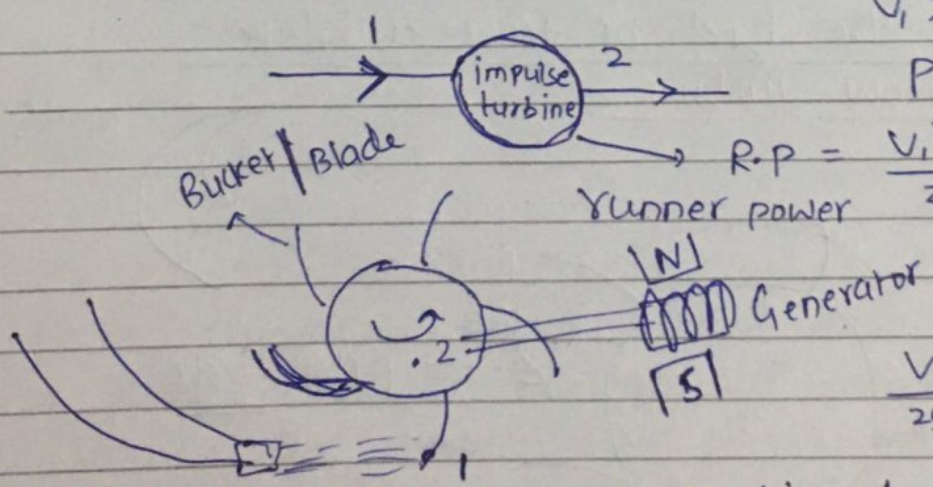
Hydraulic Turbines are the Devices which convert hydraulic energy into mechanical energy. This mechanical energy is further converted into electrical energy. by a generator

Classification of Hydraulic Turbines

① According to the action of Water on moving blade.

A) Impulse Turbine, in impulse turbine, energy available @ entrance to the turbine is in the form of kinetic energy and water strikes the bucket with kinetic energy, causing an impact or ~~impulse~~ impulse action. Throughout the movement of water through the runner, pressure remains atmospheric and kinetic energy of water is extracted in form of runner power

Pelton wheel is the most commonly used impulse turbine.



$V_1 > V_2$

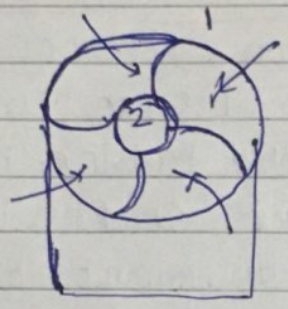
$P_1 = P_2 = P_{atm}$

$R.P = \frac{V_1^2 - V_2^2}{2g}$.

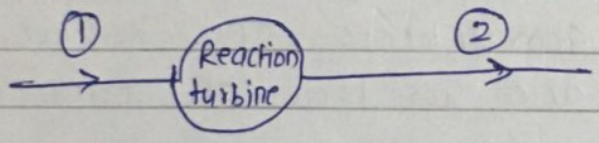
$\frac{V_1^2}{2g} - \frac{V_2^2}{2g}$

B) Reaction Turbine In reaction turbine Runner blades are completly submerged in water and energy available @ the entrance to turbine is in the form of both P.E and K.E

While flowing through runner, both KE & PE are extracted in form of runner power and @ exit both K.E & P.E are low.
flow in reaction turbine is inside in Air tight casing.
Comenly used reaction turbines are:-
Francis Turbine, Kaplam, propeller Turbine



$$R.P = \left(\frac{P_1 - P_2}{\rho g}\right) + \left(\frac{V_1^2 - V_2^2}{2g}\right)$$

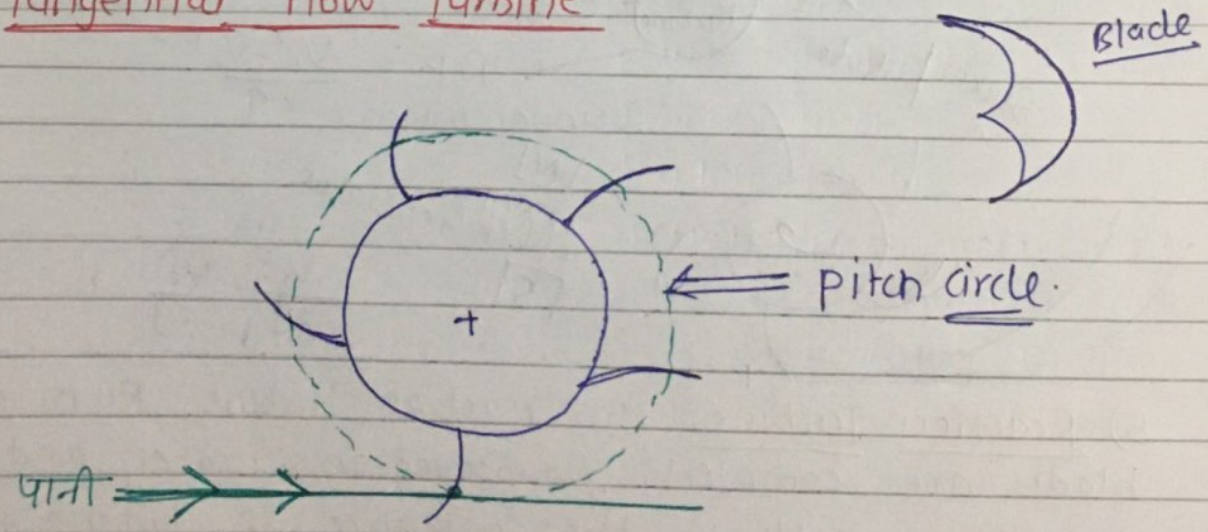


$V_1 >> V_2$

$P_1 >> P_2$

2) According to the Dir^n of flow of Water

① Tangential Flow Turbine



Ex pelton Wheel.

Water flows in a dir^n tangential to the path of rotation of runner

② Radial Flow

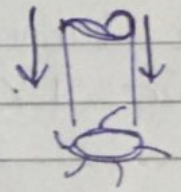
Water flows along the radial dir^n and remains mainly in the plane normal to the axis of rotation of runner

it can be i) Inward Radial Flow:- Water enters @ outer circumferance and flows radially inwards towards center of runner

Ex: old francis and Thomsan turbine

ii) outward Radial Flow:- Water enters @ center and flows radially outwards towards the outer periferi of runner

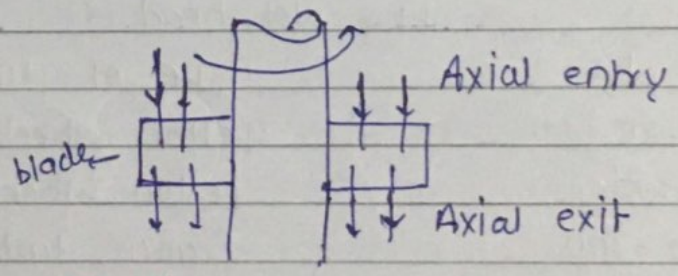
Ex: Fourneyron Turbine



③ Axial Flow Turbine

In this, flow of water through runner is mainly along the dir^n parallel to the axis of rotation of runner.

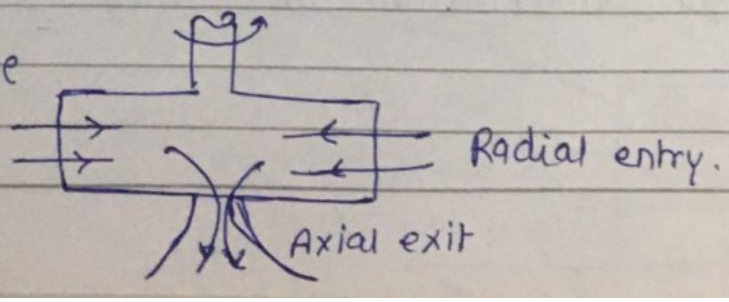
Ex: Kaplon Turbine, propeller turbine



④ Mixed Flow Turbine

In this type of turbine Water enters the runner @ Outer periferi in radial dir^n and leaves it at center in the dir^n parallel to axis of rotation of runner

Ex: Modern Francis turbine



③ Based on Head and discharge.

① pelton wheel → works under high head hence requires low discharge

$$P = \rho g Q H$$

② francis turbine works under medium head and hence requires Medium discharge.

③ Kaplon and propeller turbine work under low head hence require high discharge.

④ on Basis of specific speed (N_s)

specific speed is the speed of a geometrically similar turbine which while working under a head of 1m. produces 1 KW power

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

N → speed of runner (rpm)
P → power output in (KW)
H → Net head

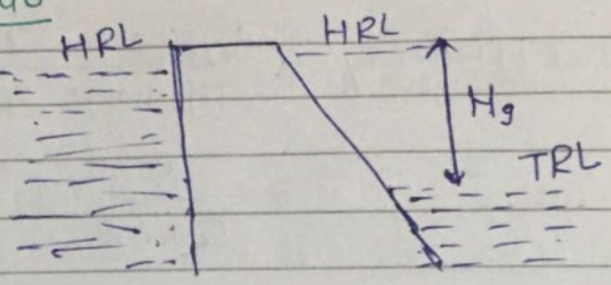
N_s	Type of turbine
10-35	pelton wheel with single jet
>35	pelton wheel with multijets
60-300	francis turbine
300-900	Kapton turbine
>900	special type of runners are used

NOTE calculation of specific speed is independent of size of turbine hence this is the most scientific criteria for classification of turbine

Head and efficiency of Turbine

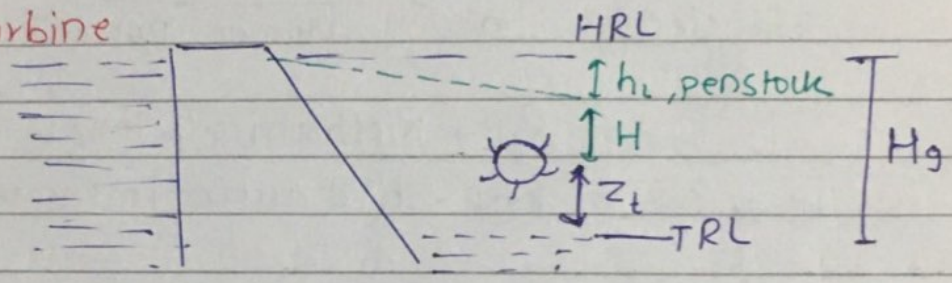
① Gross Head

It is the diff^n of elevation b/w HRL and TRL



② NET Head

for Impulse Turbine →



$$\text{Net head } (H) = H_g - h_{L,penstock} - Z_t$$

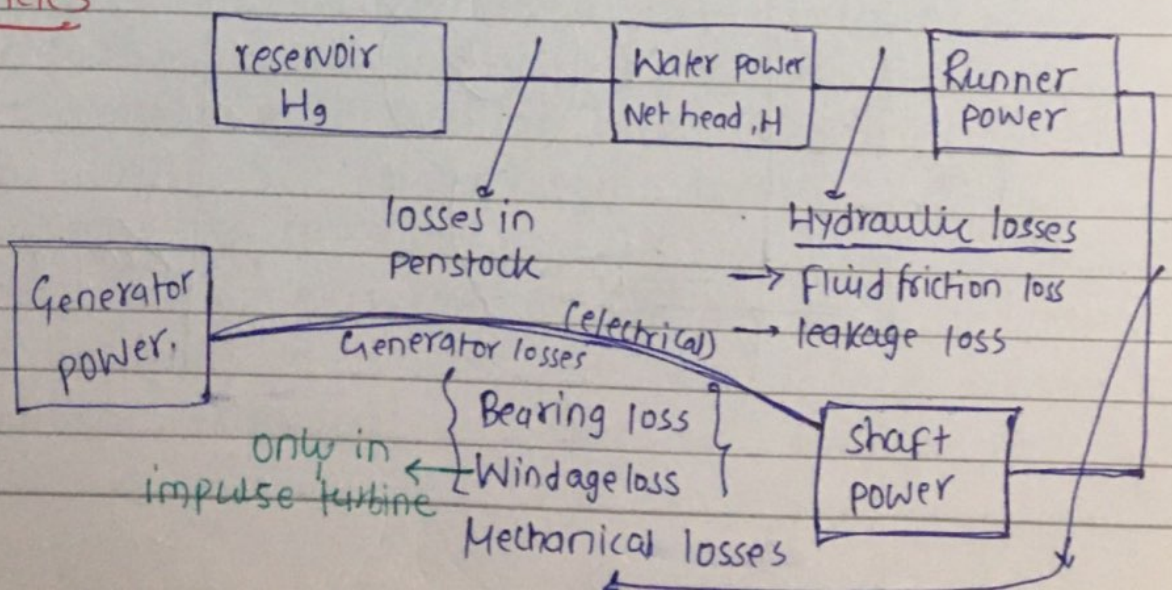
Z_t → height of turbine above TRL

for Reaction turbine

$$H = H_g - h_{L,penstock} - \frac{V_{dt}^2}{2g}$$

V_{dt} → velocity @ exit of draft tube

Efficiencies



Hydraulic Efficiency $\eta_h = \frac{\text{Runner power}}{\text{Water power}} = \frac{RP}{\rho g Q H}$

Volumetric Efficiency $= \frac{\text{Discharge striking the Runner}}{\text{Discharge Supplied to the Runner}}$

$= \frac{Q}{Q + \Delta Q}$ $\Delta Q \rightarrow$ leekage loss.

Mechanical Efficiency $\eta_m = \frac{\text{Shaft power}}{\text{Runner power}}$

S.P = R.P − Mechanical losses

R.P = W.P − hydraulic losses.

Overall Efficiency

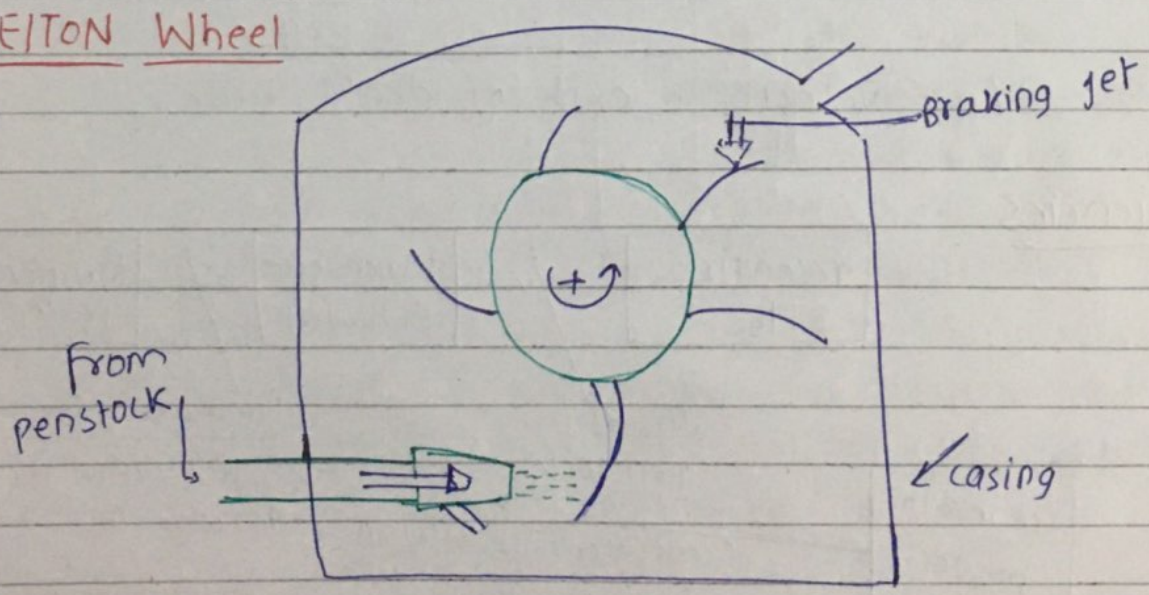
$\eta_o = \frac{\text{Shaft power}}{\text{Water power}}$

Shaft power → Brake horse power (BHP)

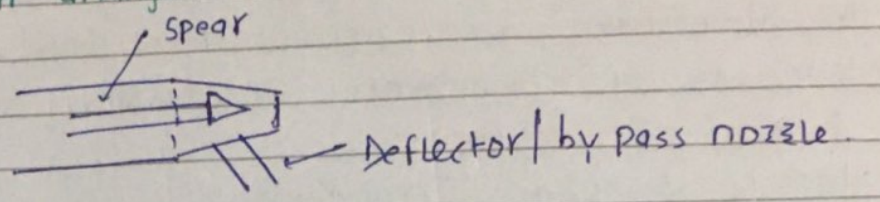
Water power → Water horse power (WHP)

$\eta_o = \eta_h \times \eta_m$

PELTON Wheel

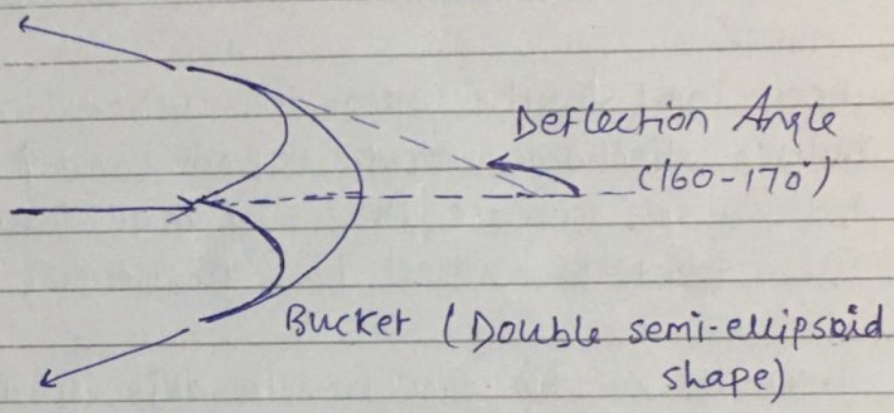


Nozzle spear arrangement



load ↑ ⇒ P_{gen} ↑ ⇒ Q↑ } Governing.
load ↓ ⇒ P_{gen} ↓ ⇒ Q↓

→ High H, low Q
→ Tangential flow
→ Impulse
→ $V_2 << V_1$
→ $P_1 = P_2 = P_{atm}$



Major components of a pelton wheel

1) Nozzle spear arrangement
2) Bucket
3) Casing
4) Brakling Jet

pelton wheel turbine is a tangential flow, high head impulse turbine. in this turbine water jet impinges on bucket attach to the shaft of turbine which causes force on bucket. This force creats torque necessary for rotation of shaft.
all the head is converted to kinetic head before the jet impinges on bucket. and the kinetic energy of water is extracted in form of runner power

While moving on bucket, water is always in contact with atmosphere and hence throughout it's operation water is at atmosphere pressure.

Major components.

① Nozzle & spear Arrangement

It is a guide mechanism which controls quantity of water passing through nozzle and striking the bucket. It helps in meeting the variable demand of power

When load ↓ the spear is pushed into nozzle to reduce discharge and when load ↑ spear is pulled back to increase the discharge. Thus it maintains the speed of wheel (or frequency) constant

Deflector or by pass nozzle is used to control the water hammer pressure rise due to sudden reduction in discharge.

② Bucket

It has the shape of double semi-ellipsoid. Advantage of having this shape is that net axial thrust gets reduced.

③ Casing

In pelton wheel casing doesn't perform any hydraulic function. It's purpose is to prevent the splashing of water and to guide the water to the tail race. it also safeguards against accidents.

④ Braking jet

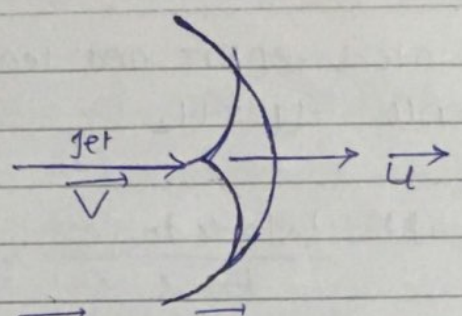
it is provided to stop the runner in shortest possible time. it is in the form of a jet striking the bucket from opposite dir^n.

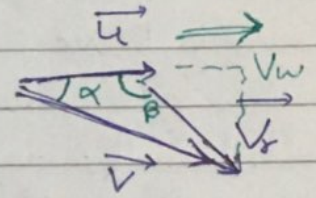
NOTE ⇓ Not used for speed reduction, only used to stop the runner

NOTE • ordinarily pelton wheel has single jet and horizontal Shaft. However no. of jets depends on specific speed. for specific speed > 35, Generally multijets are used.

• World wide Max. 6 jets have been employed without interferance.

Power output in Pelton Wheel



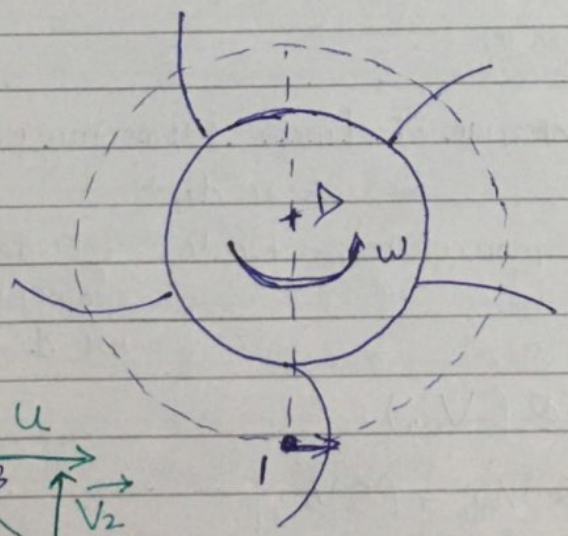


$$\vec{V_{jg}} = \vec{V_{jb}} + \vec{V_{bg}}$$

$$\vec{V} = \vec{V_r} + \vec{u}$$

$\alpha \rightarrow$ Angle b/w $\vec{V}$ & $\vec{u}$

$\beta \rightarrow$ Angle b/w $\vec{V_r}$ & $-\vec{u}$



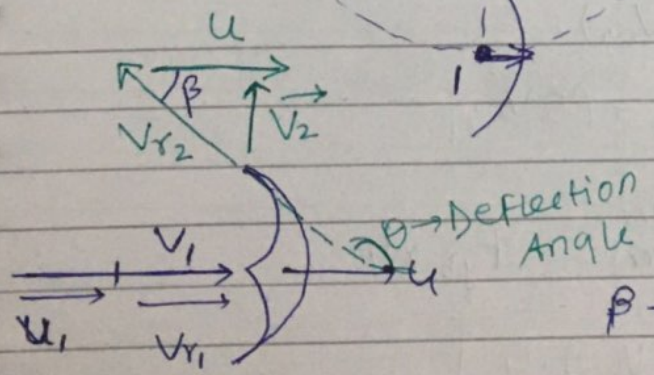
$D \rightarrow$ Pitch circle diameter.

$u \rightarrow$ peripheral / tangential velocity of wheel.

$$= \omega \times D/2$$

$$= \frac{2\pi N}{60} \times D/2$$

$$u_1 = \frac{\pi D N}{60} = u_2$$



$\beta \rightarrow$ Angle b/w V_r & $-u$

$= 180°$

$$\vec{V_1} = \vec{V_{r_1}} + \vec{u_1}$$

- V_1 and V_2 are absolute velocity of jet at inlet and ~~outlet~~ exit
- V_{r_1} & V_{r_2} are the relative velocities of jet w.r.t bucket at inlet & exit
- u → tangential velocity of bucket
- V_{w_1} and V_{w_2} → tangential component of absolute velocity at inlet and exit.

In pelton wheel waters enters and leavs @ same Diameter (D) hence $u_1 = u_2$

$$\text{power output} = \frac{\text{Work done}}{\text{time}} = \frac{F_{jb} \times \text{Displacement in dir}^n \text{ of force}}{\text{time}}$$

$$= F_{\text{Jet on bucket}} \times u$$

$$F_{\text{Jet on bucket}} = -F_{\text{bucket on jet}}$$

$F_{\text{bucket on jet}}$ = Rate of change of Linear Momentum of jet in $+x$ dir^n

= Final linear momentum in $+x$ dir^n − Initial Linear Momentum in $+x$ dir^n

$$F_{bj} = \rho Q(-V_{w_2}) - \rho Q(V_{w_1})$$

$$F_{\text{jet on bucket}} = -F_{bj} = -[-\rho Q V_{w_2} - \rho Q V_{w_1}]$$

$$= \rho Q(V_{w_1} + V_{w_2})$$

power output (Runner power) = $F_{jb} \times u$

$$\boxed{R.P = \rho Q(V_{w_1} + V_{w_2}) \cdot u}$$

$Q \rightarrow$ Discharge hitting the bucket

$= A \times V_1$

$A \rightarrow$ Area of jet

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

where $d \rightarrow$ dia of jet

$V_1 \rightarrow$ absolute velocity of jet

* if it was a single jet, we would have taken V_{r_1}

It is assumed that at every instant, there is a bucket in front of jet.

Fulgr's head (He)

$$H_e = \frac{\text{Runner power}}{\text{Wt. of fluid}} = \frac{\rho Q (V_{w_1} + V_{w_2}) u}{\rho Q g}$$

$$\boxed{H_e = \frac{(V_{w_1} + V_{w_2}) u}{g}}$$

$$R.P = \rho Q (V_{w_1} + V_{w_2}) u$$

From velocity Δ^{le}

$V_{w_1} = V_1$

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

$K \rightarrow$ bucket friction cofficient $= \frac{V_{r_2}}{V_{r_1}}$

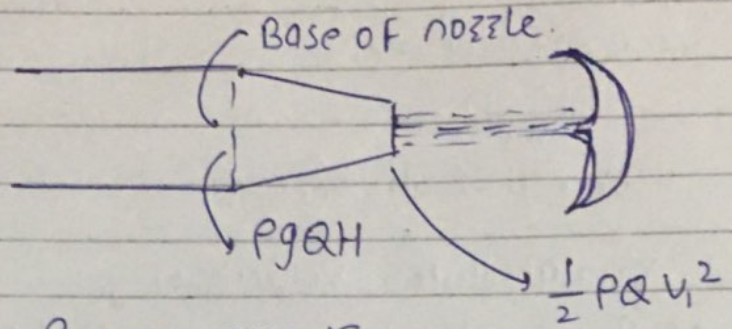
$V_{r_2} = K V_{r_1}$

$V_{r_2} = K (V_1 - u)$

$V_{w_2} = K (V_1 - u) \cos\beta - u$

$$\boxed{R.P = \rho Q [(V_1 - u)(1 + K \cos\beta)] u}$$

Hydraulic Efficiency (η_h)



$$\eta_h = \frac{\text{Runner power}}{\frac{1}{2}\rho Q V_1^2}$$

$$\eta_h = \frac{\text{Runner power}}{\rho g Q H}$$ ⇒ when nozzle is considered as an integral part of turbine.

$$\eta_h = \frac{\rho Q (V_1 - u)\, u\, (1 + K\cos\beta)}{\rho g Q H}$$

$$\eta_h = \frac{(V_1 - u)\, u\, (1 + K\cos\beta)}{gH}$$

for max. η_h

$$\frac{d\eta_h}{du} = 0 \Rightarrow V_1 - 2u = 0$$

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

$$(\eta_h)_{max} = \frac{\left(V_1 - \frac{V_1}{2}\right)\frac{V_1}{2}(1 + K\cos\beta)}{gH}$$

$$\eta_{h.max} = \frac{V_1^2 (1 + K\cos\beta)}{4gH}$$

$V_1 \rightarrow$ absolute velocity of jet

$$V_1 = K_{v_1}\sqrt{2gH}$$

$\beta = 180 - \theta$

$K_{v_1} \rightarrow$ cofficient of Velocity of nozzle.

$K_{v_1} \Rightarrow 0.97 - 0.99$

$$\eta_{h_{max.}} = \frac{K_{v_1}^2 \cdot 2gH(1 + K\cos\beta)}{gH}$$

$$= \frac{K_{v_1}^2 (1 + K\cos\beta)}{2}$$

if losses in nozzle are neglected ; $K_{v_1} = 1$

Bucket friction is neglected (ideal Flow) ; $K = 1$

$$\boxed{(\eta_{h.max}) = 1 + \frac{\cos\beta}{2}}$$ *** $\beta \rightarrow$ Blade angle

$\beta = 180 - \theta$

$$\eta_{h,max} = \frac{1 - \cos\theta}{2}$$ *** $\theta \rightarrow$ deflection Angle.

if $\theta = 180°$

$$\boxed{\eta_{h.max.} = 100\%}$$ ***

In practice, however the deflection is limited to about 165° so that the water leaving the bucket may not hit the following bucket

Wheel efficiency η_w

$$\eta_w = \frac{\text{Runner power}}{\text{power input to wheel}}$$

$$\eta_w = \frac{\rho Q (V_{w_1} + V_{w_2}) u}{\frac{1}{2} \rho Q V_1^2}$$

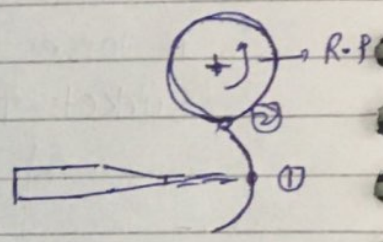
Nozzle efficiency η_n

$$\eta_n = \frac{\frac{1}{2}\rho Q V_1^2}{\rho g Q H} = \frac{V_1^2}{2gH} = \frac{K V_1^2 \cdot 2gH}{2gH} = K V_1^2$$

$$\eta_n = K V_1^2$$

$$\eta_h = \eta_n \times \eta_w$$ **.

Hydraulic losses



1. Nozzle losses $= \rho g Q H - \frac{1}{2}\rho Q V_1^2$

2. Bucket friction losses $= (h_{L,bf})$

Applying energy eq^n b/w ① & ②

$$\frac{V_1^2}{2g} = \frac{V_2^2}{2g} + h_{L,bf} + \frac{(V_{w_1} + V_{w_2})u}{g}$$

$$\frac{V_{r_1}^2}{2g} = \frac{V_{r_2}^2}{2g} + h_{L,bf}$$

$$h_{L,bf} = \frac{V_{r_1}^2 - V_{r_2}^2}{2g}$$

3. K.E of water leaving the bucket

Mechanical losses

S.P = R.P − Mechanical losses

- → Bearing friction loss
- → Windage loss.

Working proportion of pelton wheel.

(i) $V_1 = K_{V_1}\sqrt{2gH}$

K_{V_1} = coeff. of velocity of jet

$K_{V_1} = 0.97 - 0.99$

(ii) $u = K_u\sqrt{2gH}$

$K_u \rightarrow$ speed ratio

$K_u = 0.43 - 0.47$

(iii) $Q = \frac{\pi}{4} d^2 \times V_1$; for single jet

$Q = n \times \frac{\pi}{4} d^2 \times V_1$; for n. jets

(iv) Jet ratio (m)

$m = \frac{D}{d}$ $D \rightarrow$ Pitch circle dia of wheel, $d \rightarrow$ Dia of jet

v) Noumber of bucket (Z)

$Z = 0.5m + 15$ $m \rightarrow$ jet ratio

(vi) Selection of Wheel

$u = K_u\sqrt{2gH}$

$\frac{\pi D N}{60} = K_u\sqrt{2gH}$

$\Rightarrow N = \_\_\_$

Also, speed of generator is calculated as.

$N = \frac{60 \times f}{P}$ f; frequency (= 50 Hz in India)

P: no. of pair of poles

$N = \frac{120 \times f}{P'}$ P' $\rightarrow$ No. of poles.

N $\rightarrow$ synchronous speed.

Governing of Turbine

Governer regulates the inflow of water into a turbine. A hydro turbine is directly coupled to the electric generator hence is reqᵛ to maintain synchronous speed @ all loads. Change in load necessiates speed adjustment.

The req. adjustments in form of control of flow into the turbine is done by governer. in pelton wheel governing is done through nozzle and sphere arrangement by moving the sphere (Needle Valve)

Runaway speed

When discharge is hitting the bucket and load suddenly reduces to 0, wheel rotates @ max. speed and this speed is known as runaway speed.

Specific speed (N_s)

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

$N \rightarrow$ speed of wheel (rpm)
$P \rightarrow$ power output for a single jet (kW)
$H \rightarrow$ Net Head (m)

for n - no. of jets

$$Q_{n\text{-jet}} = n \times \frac{\pi}{4} \times d^2 \times V_1$$

$$P_{n\text{-jet}} = n \times P_{\text{single jet}}$$

$$(N_s)_{n\text{-jet}} = \frac{N\sqrt{n P_{\text{single jet}}}}{(H)^{5/4}}$$

$$= \sqrt{n}\,\frac{N\sqrt{P}}{H^{5/4}} = \sqrt{n}\,(N_s)_{\text{single jet}}$$

Que. In a 2 jet pelton Wheel working @ Full load, each of the nozzles are issuing 10 cm dia jets. The nozzles are so shaped so as to avoid any further contraction of jet. if the load is suddenly reduced to 36 % of full load, the altered jet dia will be?

A) 1.8 cm
B) 3 cm
C) 3.6 cm
d) 6 cm.

$$Q = n \times \frac{\pi}{4} d^2 \times V_1$$

$$Q \propto d^2$$

As load is reduced to 36 % of full load

$$\frac{Q_2}{Q_1} = 0.36$$

$$\frac{Q_2}{Q_1} \propto \left(\frac{d_2}{d_1}\right)^2$$

$$0.36 = \left(\frac{d_2}{10}\right)^2 \Rightarrow \boxed{d_2 = 6\,cm}$$

Que. Following observations were obtained from test

Head @ Base of nozzle = 32 m
Discharge of nozzle = $0.18\ m^3/s$
Area of jet = $7500\ mm^2$
Power available @ shaft = 44 KW
Mechanical efficiency = 94 %

Calculate power lost in KW in ~~first~~ i) Nozzle ii) Runner iii) mechanical friction

$$V = \frac{Q}{A} = \frac{0.18}{0.0075} = 24\ m/s$$

$$R.P = \frac{44}{\eta_m} = \frac{44}{0.94} = 46.80\ KW$$

i) power lost in nozzle $= \rho g Q H - \frac{1}{2}\rho Q V_1^2$

$= 1000 \times 9.81 \times 0.18 \times 32 - \frac{1}{2} \times 1000 \times 0.18 \times 24^2$

$= 4.67$ KW

ii) power lost in runner $= \frac{1}{2}\rho Q V_1^2 - R.P$

$= \frac{1}{2} \times 1000 \times 0.18 \times 24^2 - \frac{44}{0.94}$

$= 5.03$ KW

iii) power lost in Mechanical friction $= R.P - S.P$

$= \frac{44}{0.94} - 44$

$= 2.808$ KW.

Que. pelton wheel is producing a shaft power of 1 MW while working under a net head of 150 m. speed N = 450 rpm. $K_{v_1} = 0.97$
K = 0.9, θ deflection angle = 165° β = 180 - 165 = 15°
Mechanical loss = 5% of Velocity head of jet pitch circle Dia of runner = 1.2 m
calculate i) mechanical losses expressed in m of water ii) Hydraulic losses in m of water indicating all components seprately iii) Hydraulic efficiency, mechanical efficiency & overall efficiency of turbine.

$V_1 = K_{v_1}\sqrt{2gH}$

$V_1 = 0.97\sqrt{2 \times 9.81 \times 150}$

$V_1 = 52.62$ m/s

Mechanical loss $= 0.05 \times \frac{V_1^2}{2g} = 7.05$ m

Hydraulic losses :

i) nozzle loss $= H - \frac{V_1^2}{2g} = 150 - \frac{(52.62)^2}{2 \times 9.81}$

$= 8.87 m$.

ii) Bucket friction loss

$$h_{L.bf} = \frac{V_{r_1}^2 - V_{r_2}^2}{2g}$$

$V_{r_1} = V_1 - u$

$V_{r_1} = 24.36$ m/s

$V_{r_2} = K V_{r_1}$

$V_{r_2} = 0.9 \times 24.36$

$V_{r_2} = 21.924$ m/s.

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

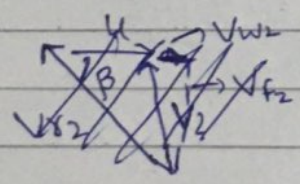
$= \frac{3.14 \times 1.2 \times 450}{60}$

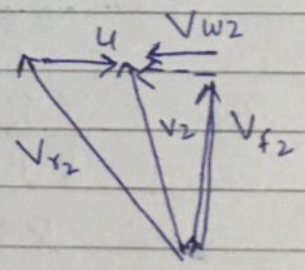
$u = 28.26$

$$h_{L.bf} = \frac{24.36^2 - 21.924^2}{2 \times 9.81}$$

$h_{L.bf} = 5.74 m$.

iii) KE of water leaving the bucket $= \frac{V_2^2}{2g}$.



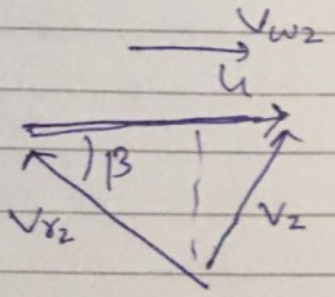


$V_2^2 = V_{w_2}^2 + V_{f_2}^2$

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

$= 21.92 \cos 15 - 28.26$

$= -7.086$ m/s



$V_{f_2} = V_{r_2} \sin\beta$

$= 21.92 \sin 15$

$= 5.6$ m/s

$$V_2^2 = V_{w_2}^2 + V_{f_2}^2$$
$$= (7.086)^2 + (5.6)^2$$
$$V_2 = 9.03 \text{ m/s}$$

KE of Water Leaving the Bucket

$$\frac{V_2^2}{2g} = \frac{(9.03)^2}{2 \times 9.81} = 4.16 \text{ m}.$$

Total hydraulic losses.

$$= 8.875 + 5.74 + 4.16$$
$$= 18.81 \text{ m}$$

iii) $$\eta_h = \frac{R.P}{W.P} = \frac{H_e}{H}$$

$R.P\ (H_e) =$ ~~[illegible]~~ $H -$ hydraulic losses.

$$= 150 - 18.81$$
$$= 131.19 \text{ m}$$

$$\eta_h = \frac{131.19}{150} = 0.8746.$$

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

$S.P =$ R.P - mechanical losses.

$$= 131.19 - 7.057$$
$$= 124.14 \text{ m}.$$

$$\eta_m = \frac{124.14}{139.18} = 0.946$$

$$\eta_o = \eta_m \times \eta_h = 0.946 \times 0.875$$
$$= 82.7\ \%$$

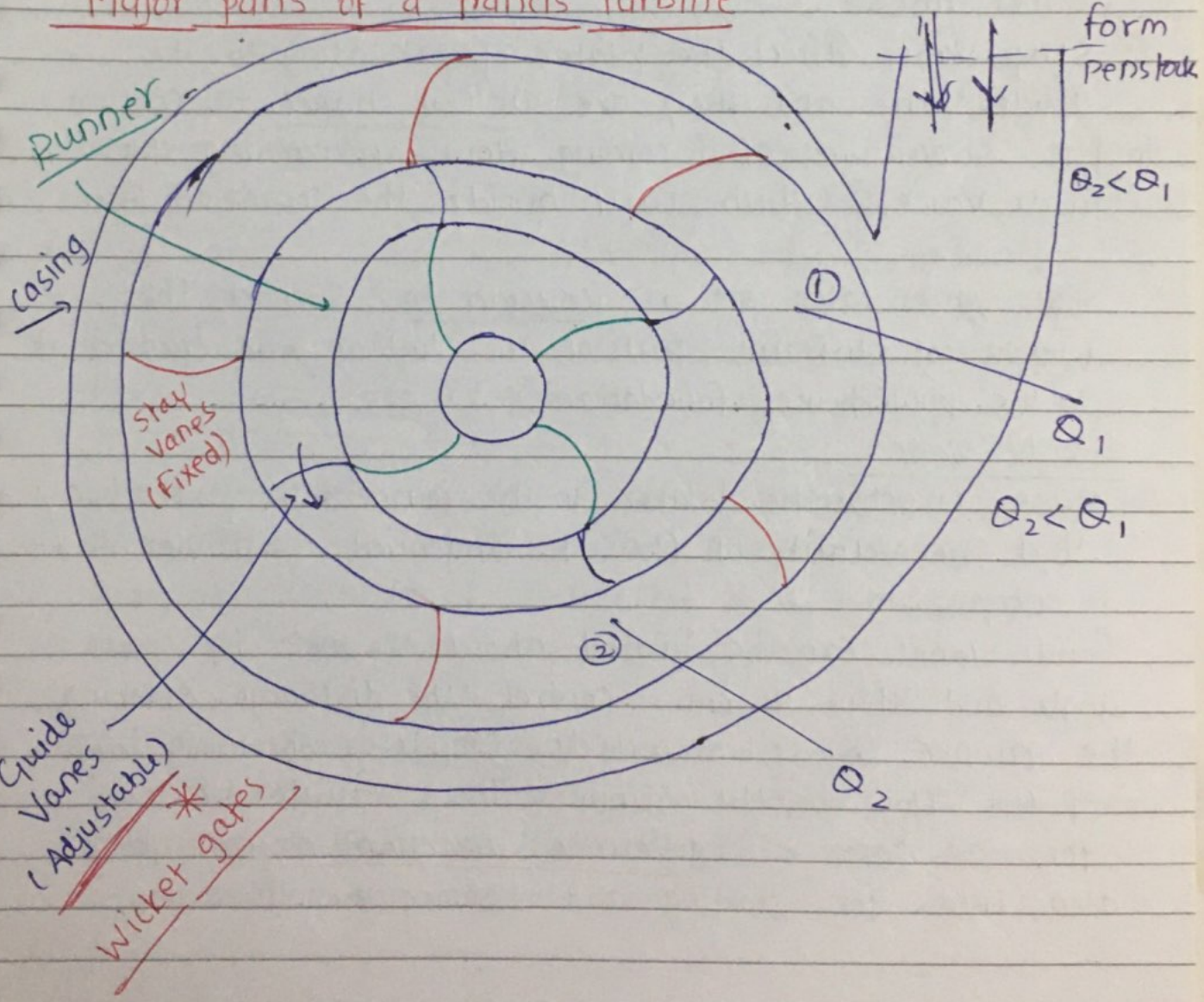
Reaction Turbine

In this energy at the entrance to runner is in form of both K.E and P.E, while passing through runner passage both pressure & velocity ↓ and @ exit pressure is -ve and velocity is low.

In reaction turbines runner blades are completely submerged in water

In reac^n turbines the flow entering the casing is turned towards the runner by the stay vanes and then it passes through guide vanes and runner blades.

Major parts of a francis turbine



Casing It is a closed passage of gradually reducing c/s enclosing the runner. It's function is to distribute the flow over the periferi of the runner in such a way that the entrance velocity to the runner vanes is same all around it
Since the flow is pressuried flow, casing should be an air-tight casing

Stay Vanes
These are optional and are generally provided for large turbines. and these are fixed vanes outside the Guide vanes
Stay vanes direct the water from casing to the guide vane and they are air foil shaped to confirm to the streamlines of incoming flow approaching the Guide Vane which will avoide the sepration losses.

Stay vanes also act as column to support the weight of different parts of the turbine and transfer it to the powerhouse foundation

Guide Vanes
These directs the water to the runner in such a way that the velocity of flow is tangential to runner blade @ entry
Guide vanes can be turned about the pivot by small angle and thus it can control the discharge entering the runner. This also includes complete shutting off of the flow to the runner. Thus Guide Vanes form a part of governing mechanism and are also used for starting and stopping the turbine.

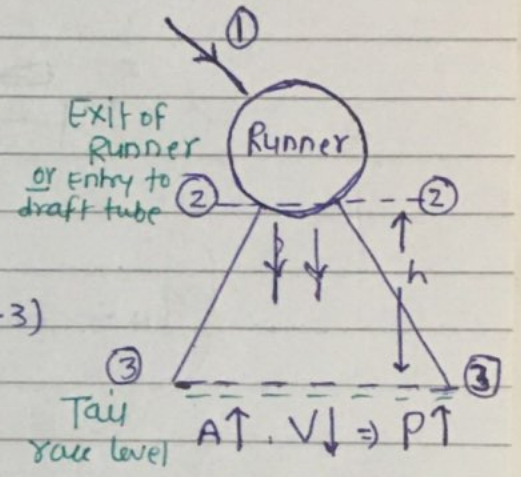
iv) Runner

Rotating component of turbine. When water is allowed to flow through the summer, it's angular momentum is reduced, thereby producing the torque, which rotats the runner. From the runner flow is directed to a draft tube and it is always desirable that water leaves the runner with no angular momentum.

DRAFT TUBE

KE is significant

$$R \cdot P = \left(\frac{P_1 - P_2}{\rho g}\right) + \left(\frac{V_1^2 - V_2^2}{2g}\right)$$



$A\uparrow . V\downarrow \Rightarrow P\uparrow$

$V_3 < V_2$

$P_3 > P_2$

$P_2 < P_{atm}$

Applying Energy Eq^n b/w sec^n (2-2) & (3-3)

$$\frac{P_2}{\rho g} + \frac{V_2^2}{2g} + h = \frac{P_2}{\rho g} + \frac{V_3^2}{2g} + h_{L,dt}$$

$$\boxed{\frac{P_2}{\rho g} = -h - \frac{V_2^2}{2g} + \frac{V_3^2}{2g} + h_{L,dt}}$$

In reaction turbines K.E @ exit of runner is significant. This K.E can be converted into pressure energy by gradual expansion of flow through a gradually diverging device known as draft tube.

Draft tube serves 2 purposes

i) it converts KE @ exit of runner into useful pressure energy

ii) Turbine can be located above the tail race without compromising on available head.

Efficiency of Draft tube

$$= \frac{\text{Actual conversion of kinetic head into pressure head}}{\text{Max. possible recovery of kinetic head}}$$

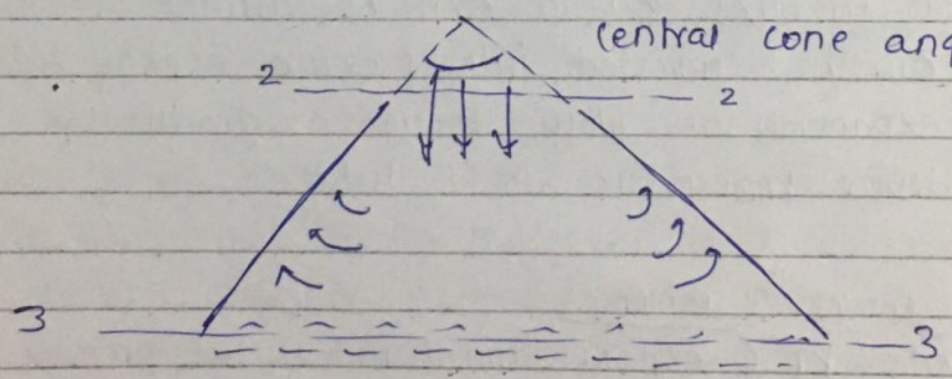
$$= \frac{\frac{V_2^2}{2g} - \frac{V_3^2}{2g} - h_{L,dt}}{\frac{V_2^2}{2g} - \frac{V_3^2}{2g}}$$

$$= 1 - \frac{\text{loss inside the Draft tube}}{\frac{V_2^2 - V_3^2}{2g}}$$

$$\eta_{dt} = 1 - \frac{h_{L,dt}}{\frac{V_2^2 - V_3^2}{2g}}$$

Alternatively,

$$\eta_{dt} = \frac{\text{Actual conversion of kinetic head into pressure head}}{\text{Kinetic head at entrance of Draft tube.}}$$



central cone angle of straight divergent draft tube should not be more than 8° otherwise flow sepration may occure and losses in draft tube him be more.

The lower end of draft tube should be kept Submerged in water so that air can't enter through it.

Cavitation in Reaction Turbines

$P_{abs} < P_{vap.}$ ⇒ cavitation may occure

$$\left(\frac{P_2}{\rho g}\right)_{abs.} = \left(\frac{P_3}{\rho g}\right)_{abs} - h - \frac{V_2^2}{2g} + \frac{V_3^2}{2g} + h_{L,dt}$$

$$\left(\frac{P_2}{\rho g}\right)_{abs.} = \frac{P_{atm}}{\rho g} - h - \frac{V_2^2}{2g} + \frac{V_3^2}{2g} + h_{L,dt}$$

for no cavitation,

$$\left(\frac{P_2}{\rho g}\right)_{abs} > \left(\frac{P_v}{\rho g}\right)$$

$$\frac{P_{atm}}{\rho g} - h - \frac{V_2^2}{2g} + \frac{V_3^2}{2g} + h_{L,dt} > \frac{P_v}{\rho g} \quad \text{—— (1)}$$

chances of cavitation are high if –

$P_{atm} \downarrow$

$h \uparrow$

$V_2 \uparrow$

$T \uparrow \Rightarrow P_v \uparrow$

from eq^n (1) $\quad h < \frac{P_{atm}}{\rho g} - \frac{P_v}{\rho g} - \frac{V_2^2}{2g} + \frac{V_3^2}{2g} + h_{L,dt}$

Max. height @ which turbine can be located above TRL

$$(h)_{max} = \frac{P_{atm} - P_v}{\rho g} - \frac{V_2^2 - V_3^2}{2g} + h_{L,dt}$$

Alternatively

check for cavitation can be done by using

Thoma's cavitation factor (σ) = $\frac{H_a - H_s - H_v}{H}$

$H_a \rightarrow$ Atmospheric pressure head
$H_s \rightarrow$ Draft head (height of turbine above TRL)
$H_v \rightarrow$ Vapor pressure head
$H \rightarrow$ net head

for no cavitation ; $\sigma > \sigma_c$

σ_c : critical cavitation factor

$\sigma_c = f(N_s)$

$N_s \uparrow \Rightarrow \sigma_c \uparrow \rightarrow$ chances of cavitation $\uparrow$

σ is a ~~meager~~ measure of margin against cavitation susceptibility

In reaction turbine pressure is Min. @ exit of runner or at the entry to draft tube hence chances of cavitation are max @ the exit of runner

Que. Giving that atmospheric pressure head = 9 m. Vapor pressure head = 1 m., net head = 40 m cavitation cofficient $\sigma = 0.15$. the height @ which Turbine can be located above TRL is

a) 6m b) 4m c) 3m d) 2m ✓

Que. Francis turbine operating under a net head of 20 m is tentatively proposed to be set up with a Draft head of 2 m. @ a certain location.

local atmospheric head = 10 m. and vapor pressure head is 0.18m. critical cavitation parameter for given turbine can be taken as 0.4

i) estimate weather there is cavitation susceptibility.

ii) if a safety margin of 0.5 m for draft height is mendatory, what is the max. height above tail water @ which turbine can be installed.

$$\sigma = \frac{10 - 2 - 0.18}{20}$$

$$\sigma = 0.391 < \sigma_c = 0.4$$

hence, cavitation susceptibility.

$$0.4 = \frac{10 - x - 0.18}{20}$$

$$\underline{x = 1.82 \text{ m}}$$

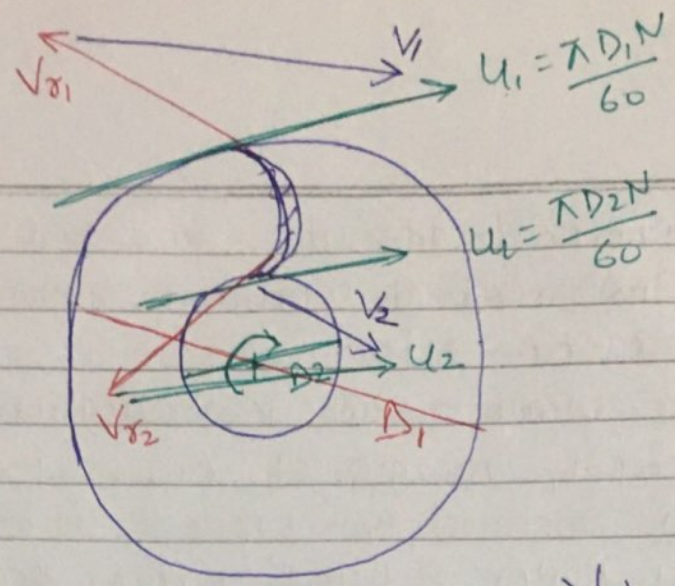
after margion of 0.5 m $\Rightarrow (h_s)_{max} = 1.32$ m

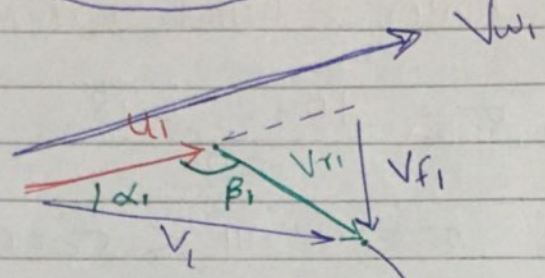
Que. Conical draft tube of a reac^n turbine has dia of 1.5 m and 2 m at it's ends the turbine is set 7m above Tail level when discharge from turbine is 12 m^3/sec the loss of energy in draft tube is 0.35 x velocity head @ runner exit take Atmospheric pressure head as 10.3 m of water

Calculate i) pressure head @ entrance of draft tube

ii) efficiency of draft tube.

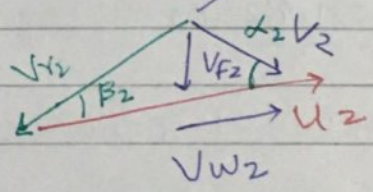
RUNNER POWER





V_1 & $u_1 \rightarrow \alpha_1 \rightarrow$ Guide Vane Angle

V_{r_1} & $-u_1 \rightarrow \beta_1 \rightarrow$ Runner Vane Angle



- u_1 and u_2 are periferal or tangential velocity of Runner @ inlet and exit
- V_{r_1} and V_{r_2} are relative velocity of water @ inlet and exit

NOTE For shockless entry and exit V_{r_1} & V_{r_2} should be tangential to the blade

- α_1 and α_2 are Guide vane angle @ inlet & exit
- β_1 and β_2 runner vane angle @ inlet & exit
- V_{w_1} and V_{w_2} are tangential components of absolute velocity @ inlet and exit (Whirl velocity)
- V_{f_1} and V_{f_2} → radial Component of absolute velocity @ inlet and exit. (Flow Velocity)

moment of momentum principle is newton's 2nd law applied for a rotating fluid mass.
In a steady state flow the rate of change of moment of momentum of a system = net torque acting on the system.

$$\text{Runner power} = \omega \times T_{\text{water on runner}}$$

$$T_{\text{water on runner}} = -T_{\text{runner on water}}$$

$T_{\text{runner on water}}$ = Rate of change of angular momentum of water in c/w dir^n

= Angular momentum flux out in ↱ dir^n − Angular momentum flux in ↱ dir^n

$$= \rho Q V_{w_2} r_2 - \rho Q V_{w_1} r_1$$

$$T_{\text{water on runner}} = -[\rho Q V_{w_2} r_2 - \rho Q V_{w_1} r_1]$$
$$= \rho Q [V_{w_1} r_1 - V_{w_2} r_2]$$

$$\text{Runner power} = \omega \times T_{\text{water on runner}}$$
$$= \rho Q [V_{w_1} r_1 - V_{w_2} r_2] \omega$$
$$= \rho Q [V_{w_1} (\omega r_1) - V_{w_2} (\omega r_2)]$$

$$R.P = \rho Q [V_{w_1} u_1 - V_{w_2} u_2] \quad \alpha_2 < 90^\circ$$

$$R.P = \rho Q [V_{w_1} u_1 + V_{w_2} u_2] \quad \alpha_2 > 90^\circ$$

if $\alpha_2 = 90^\circ$ Radial discharge @ outlet $V_{w_2} = 0$

$$V_2 = V_{f_2}$$

practically it is observed that hydraulic efficiency of francis turbine is max. when discharge is radial @ the outlet, i.e ~~water~~ $\alpha_2 = 90°$, $V_{w2} = 0$, $V_2 = V_{f2}$
मतलब water has no angular momentum

$$R.P = \rho Q V_{w1} u_1$$

Euler's Head (H_e)

$$= \frac{\text{Runner power}}{\text{wt. of fluid}}$$

$$H_e = \frac{\rho Q (V_{w1} u_1 - V_{w2} u_2)}{\rho Q g} = \frac{V_{w1} u_1 - V_{w2} u_2}{g}$$

if $\alpha_2 = 90°$

$$H_e = \frac{V_{w1} u_1}{g}$$

For max η_n

for old francis

Discharge is purely radial @ outlet

Modern francis Discharge is purely axial @ outlet

Radial Discharge @ outlet

$\alpha_2 = 90°$, $V_{w2} = 0$

$V_2 = V_{f2}$

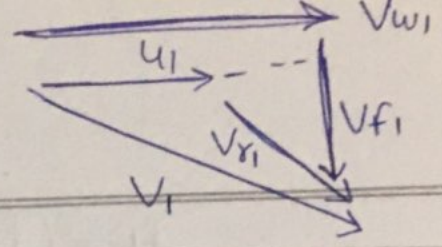
***

Degree of Reaction in Reaction Turbines

$$R.P = \left(\frac{P_1 - P_2}{\rho g}\right) + \left(\frac{V_1^2 - V_2^2}{2g}\right)$$

$\Downarrow$ Reaction effect

$\Downarrow$ Impulse effect



$$H_e = \frac{V_{w_1}u_1 - V_{w_2}u_2}{g}$$

from inlet velocity triangle

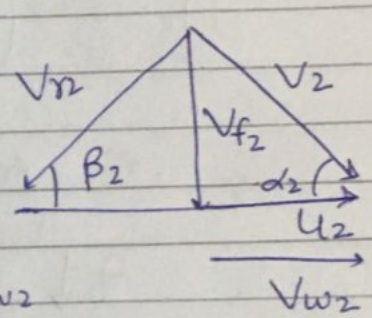
$$V_1^2 = V_{w_1}^2 + V_{f_1}^2$$

$$V_1^2 = V_{w_1}^2 + V_{r_1}^2 - (V_{w_1} - u_1)^2$$

$$V_1^2 = \cancel{V_{w_1}^2} + V_{r_1}^2 - \cancel{V_{w_1}^2} - u_1^2 + 2u_1V_{w_1}$$

$$V_{w_1}u_1 = \frac{V_1^2 + u_1^2 - V_{r_1}^2}{2}$$

from exit velocity triangle



$$V_2^2 = V_{w_2}^2 + V_{f_2}^2$$

$$V_2^2 = V_{w_2}^2 + V_{r_2}^2 + (u_2 - V_{w_2})^2$$

$$V_2^2 = V_{w_2}^2 + V_{r_2}^2 + u_2^2 - V_{w_2}^2 + 2u_2V_{w_2}$$

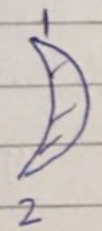
$$V_{w_2}u_2 = \frac{V_2^2 + u_2^2 - V_{r_2}^2}{2}$$

$$H_e = \left(\frac{V_1^2 + u_1^2 - V_{r_1}^2}{2g}\right) - \left(\frac{V_2^2 + u_2^2 - V_{r_2}^2}{2g}\right)$$

$$H_e = \left(\frac{V_1^2 - V_2^2}{2g}\right) + \underbrace{\left(\frac{u_1^2 - u_2^2}{2g}\right) + \left(\frac{V_{r_2}^2 - V_{r_1}^2}{2g}\right)}_{\text{Pressure}} \quad \text{— (1)}$$

Applying Energy EQ^n b/w inlet and exit of Runner (Neglecting losses in runner)

$$\frac{P_1}{\rho g} + \frac{V_1^2}{2g} + Z_1 = \frac{P_2}{\rho g} + \frac{V_2^2}{2g} + Z_2 + H_e$$



$$H_e = \left(\frac{P_1 - P_2}{\rho g}\right) + \left(\frac{V_1^2 - V_2^2}{2g}\right) \quad \text{—— (2)}$$

From eqn (1) & (2)

$$\left(\frac{P_1 - P_2}{\rho g}\right) + \left(\frac{V_1^2 - V_2^2}{2g}\right) = \left(\frac{V_1^2 - V_2^2}{2g}\right) + \left(\frac{u_1^2 - u_2^2}{2g}\right) + \left(\frac{V_{r_2}^2 - V_{r_1}^2}{2g}\right)$$

$$\left(\frac{P_1 - P_2}{\rho g}\right) = \left(\frac{u_1^2 - u_2^2}{2g}\right) + \left(\frac{V_{r_2}^2 - V_{r_1}^2}{2g}\right) \quad \text{—— (3)}$$

$$H_e = \underbrace{\left(\frac{V_1^2 - V_2^2}{2g}\right)}_{\text{Impulse effect}} + \underbrace{\left(\frac{u_1^2 - u_2^2}{2g}\right) + \left(\frac{V_{r_2}^2 - V_{r_1}^2}{2g}\right)}_{\text{pressure effect}}$$

$$A \downarrow \Rightarrow V_r \uparrow \Rightarrow P \downarrow$$

Degree of Reaction → $R = \dfrac{\frac{P_1 - P_2}{\rho g}}{H_e}$

impulse turbine → $P_1 = P_2 = P_{atm}$.

Degree of reaction = 0

* Kaplan and Francis & propeller turbine are not pure (100%) reaction turbines.

- $\frac{V_1^2 - V_2^2}{2g}$ represents decrease in K.E of Fluid
- $\frac{u_1^2 - u_2^2}{2g}$ represents change in energy in terms of pressure head due to centrifugal action
- $\frac{V_{r_2}^2 - V_{r_1}^2}{2g}$ represents change in energy in terms of pressure head due to acceleration of fluid while flowing through the runner passage.

Efficiency of Reaction Turbine

① Hydraulic efficiency

$$\eta_h = \frac{R.P}{W.P} = \frac{\rho Q V_{w_1} u_1}{\rho Q g H}$$

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

$$\eta_h = \frac{H_e}{H_{(net)}}$$

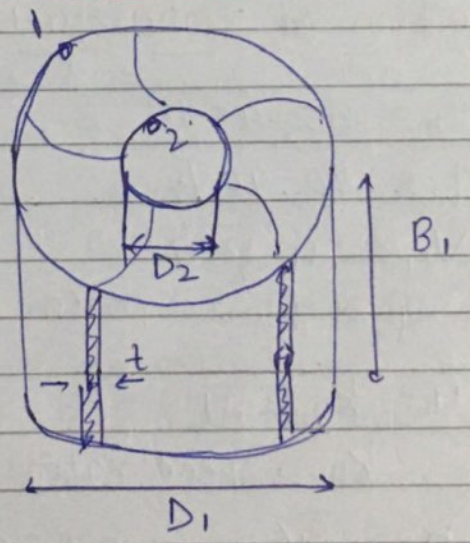
② mechanical efficiency

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

③ overall Efficiency

$$\eta_o = \eta_h \eta_m \qquad \eta_o = \frac{S.P}{W.P}$$

Discharge through the turbine



D_1 & D_2 → Dia at inlet & exit

B_1 & B_2 → width of runner @ inlet & exit

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

in n → no. of vanes

t → thickness of vane

$$Q = (\pi D_1 - nt) B_1 \times V_{f_1} = K \pi D_1 B_1 V_{f_1}$$

K → Blade thickness coffi. = 0.95

For old Francis turbine

Radial entry $Q = \pi D_1 B_1 \times V_{f1}$

Radial exit $Q = \pi D_2 B_2 \times V_{f2}$

Modern Francis turbine

Radial entry $Q = \pi D_1 B_1 \times V_{f1}$

Axial exit $Q = \frac{\pi D_2^2}{4} \times V_{f2}$

Unless otherwised mentioned it is assumed that

i) Water leaves the runner with no angular momentum or discharge is radial @ the outlet i.e $\alpha_2 = 90^\circ$

$V_{w2} = 0$, $V_2 = V_{f2}$

iii) Flow velocity V_f is constant i.e $V_{f1} = V_{f2}$

Working proportion of Francis turbine

1) $\eta_h = 85-95\%$

2) $\eta_o = 80-90\%$

3) $V_{f1} = \psi\sqrt{2gH}$

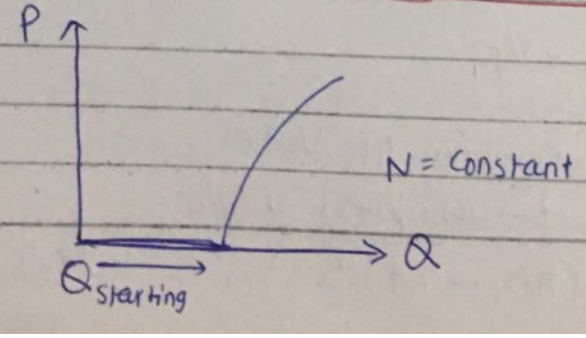
$\psi \rightarrow$ Flow ratio $= 0.15-0.3$

4) $u_1 = K_u\sqrt{2gH}$

$K_u \rightarrow$ Speed ratio $\doteq 0.6-0.9$

5) $\eta_o = B_1/D_1 = 0.1-0.45$

Operating charecterics of Francis turbine



- A Turbine is used by coupling it to generator for power generator and hence it's operating speed is constant

- A min. Discharge, $Q_{starting}$ is required to overcome the inertia of turbine unit, which requires a min. Torque to overcome it before it starts moving from the ~~idlest~~ idle position.

For PRE

1) PE & KE Both
2) Runner blades are completely Submerged in Water
3) Kaplan → Axial
   modern → mixed
   old Francis → Radial
4) N_s → 60-300
   300-900
5) casing → Air tight
   c/s is gradually reducing to keep entering Velocity same @ all blades
6) Stay Vanes → Fixed
   Air Foil shape
   behaves @ column.
7) Guide Vanes → Adjustable
   form a part of governing mechanism
8) Runner →
9) Draft tube →
10) $$\eta_{dt} = \frac{\frac{V_2^2}{2g} - \frac{V_3^2}{2g} - h_{l,dt}}{\frac{V_2^2}{2g} - \frac{V_3^2}{2g}}$$

Draft tube sepration angle $\not> 8^\circ$

(11) cavitation chances

for no cavitation $\left(\frac{P_2}{\rho g}\right)_{abs} > \left(\frac{P_{vap}}{\rho g}\right)$

$$h_{max.} = \frac{P_{atm} - P_v}{\rho g} - \frac{V_2^2 - V_3^2}{2g} + h_{i,dt}$$

Also $\sigma > \sigma_c$

$$\sigma = \frac{H_a - H_s - H_v}{H}$$

(12) $Q = \pi D_1 B_1 V_{f1}$

(13) $Q_{starting}$

Que A Francis turbine has dia of 1.5 m. and rotates at 450 rpm, velocity of flow is 10 m/s and discharge is radial @ outlet, Turbine develops 13 mW of power with a discharge of 12 cumec and diff^n of piezometric head b/w the entrance and exit from the runner is 70 m. Find

i) Absolute velocity and guide vane angle @ inlet

ii) Angle of Runner blade @ entry.

iii) loss of head in the runner.

$D_1 = 1.5$ m

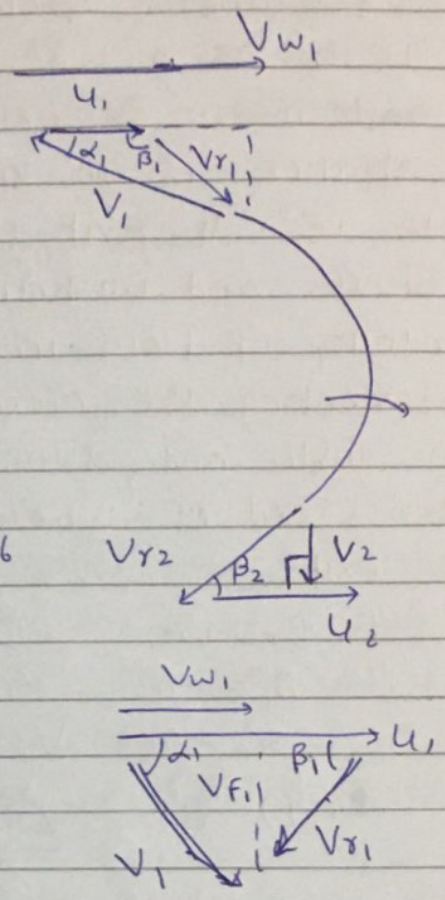
$N = 450$ rpm

$V_{f1} = 10$ m/s $= V_{f2} = V_2$

R.P $= 13$ mW

$Q = 12 m^3/s$

$$\left(\frac{P_1}{\rho g}+Z_1\right)-\left(\frac{P_2}{\rho g}+Z_2\right)=70\ m$$



(i) $u_1 = \dfrac{\pi D_1 N}{60}$

$= \dfrac{\pi \times 1.5 \times 450}{60} = 35.34\ m/s$

$V_1 = \sqrt{V_{w_1}^2 + V_{f_1}^2}$

$R.P = \rho Q V_{w_1} u_1 = 13\times10^6\ W$

$\Rightarrow 1000 \times 12 \times V_{w_1} \times 35.34 = 13\times10^6$

$V_{w_1} = 30.65\ m/s$

$V_1 = 32.24\ m/s$

$\tan\alpha_1 = \dfrac{V_{f_1}}{V_{w_1}} = \dfrac{10}{30.65}$

$\alpha_1 = 18.06°$

(ii) $\tan\beta_1 = \dfrac{V_{f_1}}{u_1 - V_{w_1}} = \dfrac{10}{35.34 - 30.65} \Rightarrow \beta_1 = 64.85°$

(iii) $h_{L,runner}$

by energy EQ^n

$$\frac{P_1}{\rho g}+\frac{V_1^2}{2g}+Z_1 = \frac{P_2}{\rho g}+\frac{V_2^2}{2g}+Z_2+h_{L,runner}+\frac{V_{w_1}u_1}{g}$$

$$70 + \frac{V_1^2}{2g} - \frac{V_2^2}{2g} - \frac{V_{w_1}u_1}{g} = h_{L,runner}$$

$$70 + \frac{32.24^2}{2\times9.8} - \frac{10^2}{2\times9.8} - \frac{30.65\times35.34}{9.8} = h_{L,runner}$$

$$\boxed{h_{L,runner} = 7.47\ m}$$

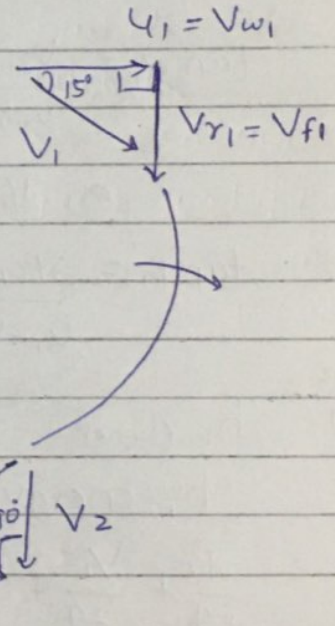
Que A Francis turbine working under a head of 40 m as a wheel dia of 1 m. @ the entrance and 0.5 m @ exit. The runner vane angle @ entrance is 90° and guide blade angle @ inlet is 15°. At exit water leaves the vane without shock and without any tangential velocity. Velocity of flow in runner is constant, Neglecting the effect of Draft tube and the losses in guide and runner passage, Determine the speed of wheel in rpm and the vane angle @ exit.

$D_1 = 1\,m \qquad D_2 = 0.5\,m \qquad H = 40\,m$

$\beta_1 = 90° \quad \alpha_1 = 15°$

$\alpha_2 = 90° \quad V_1 = V_{f2} \qquad V_{w2} = 0$

i) $H = \dfrac{V_{w1} u_1}{g} + \dfrac{V_2^2}{2g}$



$V_2 = V_{f2} = V_{f1}$

$\tan\alpha_1 = \dfrac{V_{f1}}{V_{w1}} = \dfrac{V_{f1}}{u_1}$

$V_{f1} = u_1 \tan\alpha_1$

$H = \dfrac{u_1^2}{g} + \dfrac{(u_1 \tan 15°)^2}{2g}$

$u_1 = 19.46\ m/s$

$u_1 = \dfrac{\pi D_1 N}{60}$

$19.46 = \dfrac{3.14 \times 1 \times N}{60}$

$N = 371.7\ rpm$

* Kaplon turbine operates @ higher speed and hence it's runner is smaller in size as compare to that of a francis turbine

$$\tan\beta_2 = \frac{V_{f2}}{u_2} \qquad u_2 = \frac{\pi D_2 N}{60}$$

$$\tan\beta_2 = \frac{V_{f1}}{9.72} \qquad u_2 = 9.72 \text{ m/s}$$

$$\tan\beta_2 = \frac{u_1 \tan 15}{9.72}$$

$$\beta_2 = 28.18^\circ$$

Kaplon and propeller turbines

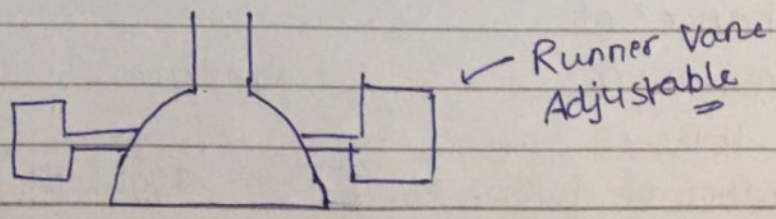
Both kaplon & propeller turbines are axial flow turbines and the runner vanes are connected to the boss or hub

The runner vane of propeller turbine are fixed while that of kaplon turbine can be turned and hence part load efficiency of kaplon turbine is higher as compare to that of a propeller turbine.

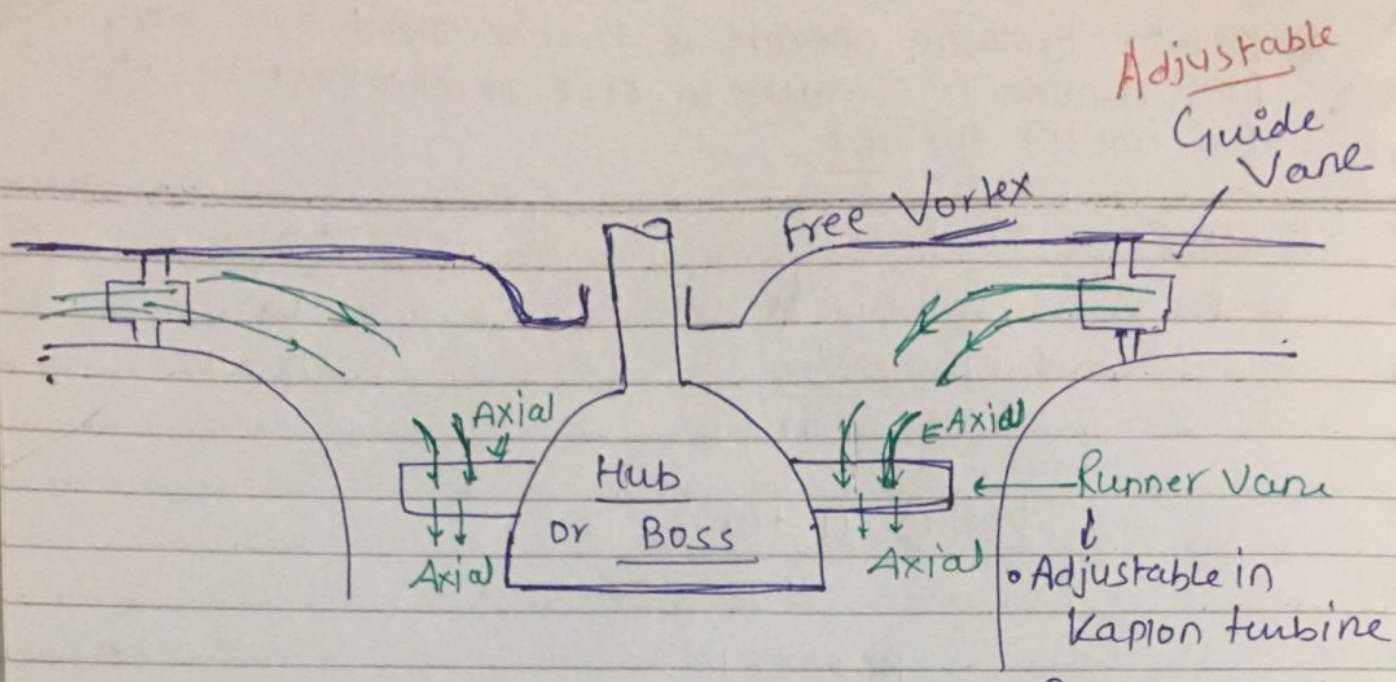
francis turbine has 16-24 no. of blades while kaplon and propeller turbine have 4-6 only. hence there is lesser contact surface area in kaplon and propeller turbine therefore frictional losses are lesser in kaplon and propeller turbine

space b/w the guide vane outlet and runner is known as whirl chamber and in this chamber flow changes from radial to axial

- Reaction turbine
- Axial flow turbine
- Kaplon turbine → runner ~~blades~~ vane are adjustable.



Kaplon turbine



- Adjustable in Kaplon turbine
- fixed in propeller

free Vortex ⇒ $V_w \cdot r$ = constant

*

Pelton	Francis	Kaplon / propeller
Tangential	Mixed / Radial	Axial
Impulse	Reaction	Reaction
High H, Low Q	Medium H, Medium Q	low H, High Q
N_s is low (8-30)	~~30-450~~ 60-300	N_s = 300-900
part load η is high	$P_1 > P_{atm}$	part load η is max
$P_1 = P_2 = P_{atm}$	$P_2 < P_{atm}$	—"—
		Kaplon ⇓ propeller ⇓ poor
• Based on fluctuations in TRL	Location of turbine is decided by dT	—"—
	Draft tube is always kept submerged in water so that air cannot enter	
chances of cavitation low	medium	Max.
location of turbine above TRL is decided by fluctuation in TRL	height of Draft tube and hence by possibility of cavitation	

Model Analysis

To predict the behaviour of a turbine under varying cond. unit quantities are defined

To estimate unit quantities it is assumed that overall efficiency is constant and for this velocity triangle under working head and other heads should be similar.

Such turbines are called homologous turbines

$$\frac{u_1}{u_2} = \frac{V_1}{V_2} = \frac{V_{r1}}{V_{r2}} = \frac{V_{w1}}{V_{w2}} = \frac{V_{f1}}{V_{f2}}$$

Unit speed (N_u)

It is the theortical speed @ which turbine would run while operating under unit head

100 m head $\rightarrow N_1$

1 m head $\rightarrow N_u$

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

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

$$u \propto \sqrt{H}$$

so, $\sqrt{H} \propto DN$

$$\frac{\sqrt{H}}{DN} = \text{constant} \Rightarrow \boxed{\frac{H}{D^2 N^2} = \text{constant}} \; *$$

for same turbine

D is constant

$$\sqrt{H} \propto N \; *$$

$$\frac{N_2}{N_1} = \sqrt{\frac{H_2}{H_1}}$$

$H_1 = H$, ; $N_1 = 1$

$H_2 = 1$; $N_2 = N_u$

$$\Rightarrow \frac{N_u}{N} = \sqrt{\frac{1}{H}} \Rightarrow N_u = N/\sqrt{H}$$

Unit Discharge (Q_u)

Discharge passing through a turbine operating under unit head

$$Q = \frac{\pi}{4} D^2 \times V_f$$

$$Q \propto D^2 \cdot V_f \qquad \{ V_f = \psi \sqrt{2gH}$$

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

$$\boxed{\frac{Q}{D^2\sqrt{H}} = \text{constant}} \; *$$

For Same turbine $D = \text{constant}$

$$\frac{Q}{\sqrt{H}} = \text{const.}$$

$$\frac{Q_2}{Q_1} = \sqrt{\frac{H_2}{H_1}}$$

$H_1 = H, \quad Q_1 = Q$

$H_2 = 1m, \quad Q_2 = Q_u$

$$\frac{Q_u}{Q} = \sqrt{\frac{1}{H}}$$

$$Q_u = \frac{Q}{\sqrt{H}} \Rightarrow \frac{Q}{D^2\sqrt{H}} = \text{constant}$$

$$\sqrt{H} \propto DN$$

$$\frac{Q}{D^3 N} = \text{constant} \; *$$

Unit power (P_u)

$$P = \eta_o \rho g Q H$$

$$P \propto QH \qquad P \propto D^2\sqrt{H} \cdot H \qquad (Q \propto D^2\sqrt{H})$$

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

$$\frac{P}{D^2 \cdot H^{3/2}} = \text{constant} \quad *$$

for same turbine $D \to$ const. $\Rightarrow \frac{P}{H^{3/2}} = \text{const}$

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

$$\frac{P_2}{P_1} = \left(\frac{H_2}{H_1}\right)^{3/2}$$

$H_1 = H$; $P_1 = P$

$H_2 = 1m$; $P_2 = P_u$

$$\frac{P_u}{P} = \left(\frac{1}{H}\right)^{3/2} \Rightarrow P_u = \frac{P}{H^{3/2}} \quad *$$

Now,

$$\frac{P}{D^2 H^{3/2}} = \text{constant}$$

$$\sqrt{H} \propto DN$$

$$\frac{P}{D^2 (DN)^{3/2}} = \text{constant}$$

$$\boxed{\frac{P}{D^5 N^3} = \text{constant}}$$

for same turbine

$\frac{H}{D^2 N^2} = \text{const}$	$\frac{\sqrt{H}}{DN} = \text{const}$	$N \propto \sqrt{H}$
$\frac{Q}{D^3 N} = \text{const}$	$\frac{Q}{D^2 \sqrt{H}} = \text{const.}$	$Q \propto \sqrt{H}$
$\frac{P}{D^5 N^3} = \text{const}$	$\frac{P}{D^2 (H)^{3/2}} = \text{const.}$	$P \propto H^{3/2}$

Specific Speed (N_s)

It is the rotational speed of a homologous turbine so reduced inside that it produces 1 KW power working under a head of 1m. Under conditions of Max. efficiency.

$$\frac{P}{D^2 H^{3/2}} = \text{const.}$$

$$\sqrt{H} \propto DN$$

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

$$\frac{P}{\left(\frac{\sqrt{H}}{N}\right)^2 \cdot H^{3/2}} = \text{const.}$$

$$\frac{N^2 P}{H^{5/2}} = \text{const.}$$

$$\frac{N\sqrt{P}}{H^{5/4}} = \text{Constant} \quad ***$$

$$\left(\frac{N\sqrt{P}}{H^{5/4}}\right)_1 = \left(\frac{N\sqrt{P}}{H^{5/4}}\right)_2$$

$H_2 = 1m$; $P_2 = 1\,KN$; $N_2 = N_s$

$$\frac{N\sqrt{P}}{H^{5/4}} = \frac{N_s\sqrt{1}}{(1)^{5/4}} = N_s$$

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

- 10-35
- 60-300
- 300-900

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

N; speed of runner (RPM)
H; Net Head (m)
P; S.P in (KW)

Specific Head has Dimensions.

Non-Dimensional specific speed

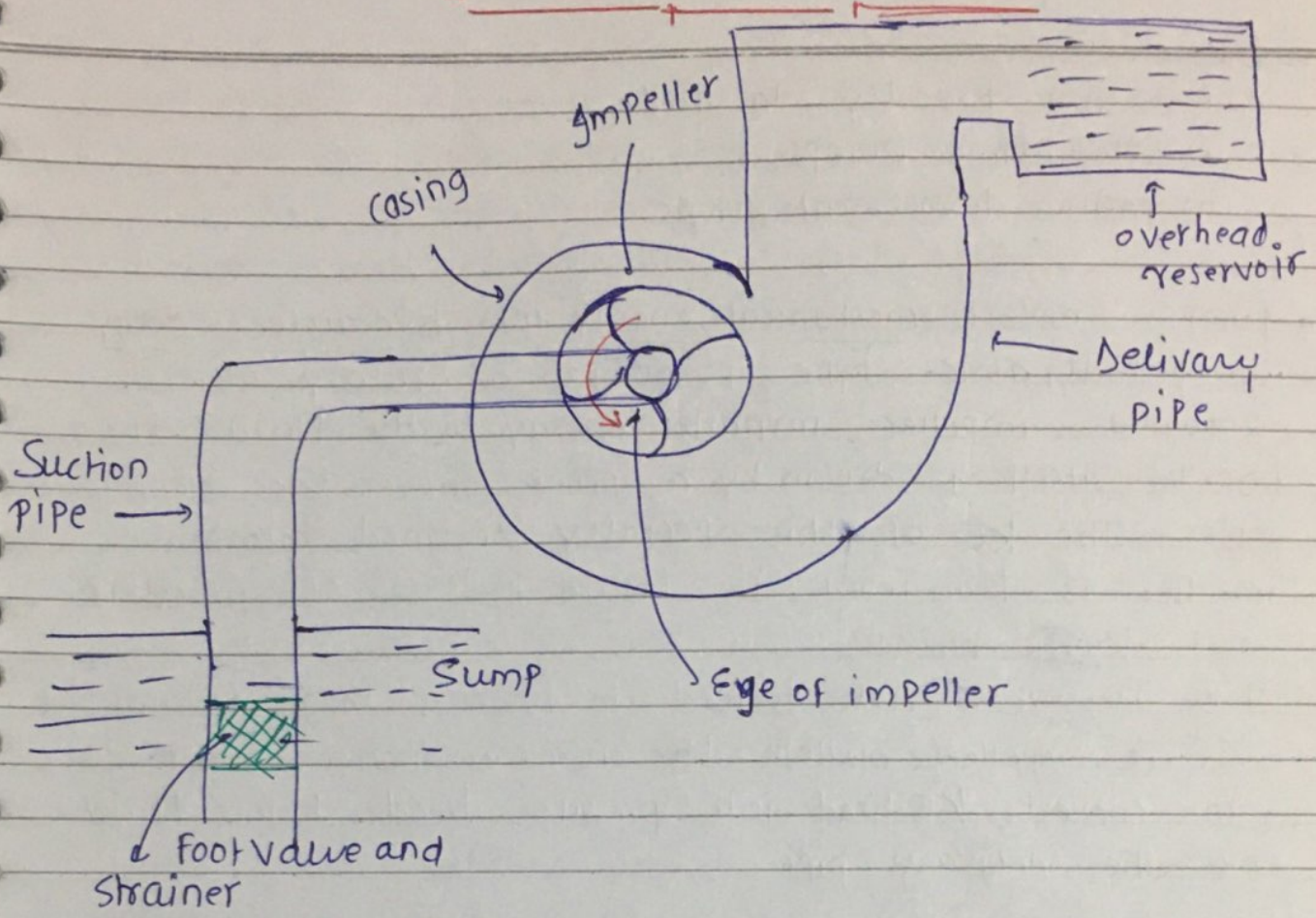
$$N_s = \frac{N\sqrt{P/\rho}}{(gH)^{5/4}}$$

$H \rightarrow m$
$P \rightarrow W$
$N \rightarrow rpm$

***

- Specific speed is same for all turbines of same type.
- specific speed is related to shape & type of turbine but not it's size and therefore it is true charecteristic of a turbine and hence classification of turbine on the basis of N_s is the most scientific criteria

CENTRIFUGAL PUMPS



PUMPS

Electrical Energy $\xrightarrow{\text{Electrical Motor}}$ Mechanical Energy $\xrightarrow{\text{pump}}$ Hydraulic Energy $\rightarrow$ pressure Energy

KE
PE

PE & KE $\rightarrow$ potential Energy

pumps are the devices which are used for transportation of a liquid from one location to another

Ex: pumping of Water from a well or Sump to an overhead reservoir

pumps are also used to circulate a liquid in a close system

Ex: circulation of cooling Water and lubricating oil to Various parts of a machine.

pumps are broadly classified as –
i) Rotodynamic pumps
ii) positive displacement pumps

- pumps Convert mechanical energy into hydraulic energy.
- In rotodynamic type of pumps a rotary element known as impeller imparts energy to the Fluid, the impeller itself is drivan by a prime mover like electric motor. The KE of this specially designed rotating impeller is transferred to liquid in terms of pressure and Kinetic energy.

It is However not desired to have a large amount of KE @ impeller outlet therefore arrangement is made to convert KE head into pressure head before Fluid enters the delivary pipe.

Centrifugal pumps are rotodynamic type of pumps in which dynamic pressure is developed which enables the liquid to be lifted from lower to higher level

Action of Centrifugal pump is the reverse of old francis turbine i.e (Radially outward)

Centrifugal pumps work on principle of forced Vortex motion in which when an external torque is applied, rise of pressure takes place in the rotating fluid. Which enables the lifting of liquid.

Components

① **Impeller** It is the rotor with a series of curved vanes it is mounted on a shaft coupled to electric motor electric motor imparts the required energy for rotating the impeller

Impellers are of 3- types

① closed impeller

It is used for handling non-viscous liquids which are free from Debris. it has better efficiency

② semi-open impeller

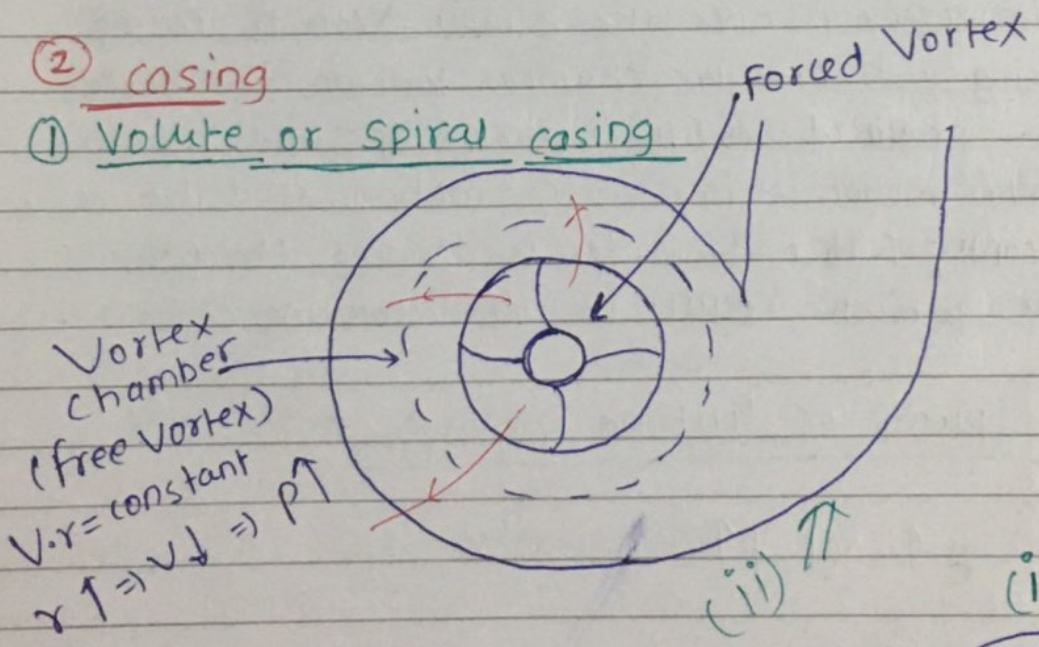
It is used for handling viscous fluids such as Seewage water, paper pulp and liquids containing small amount of suspended particles.

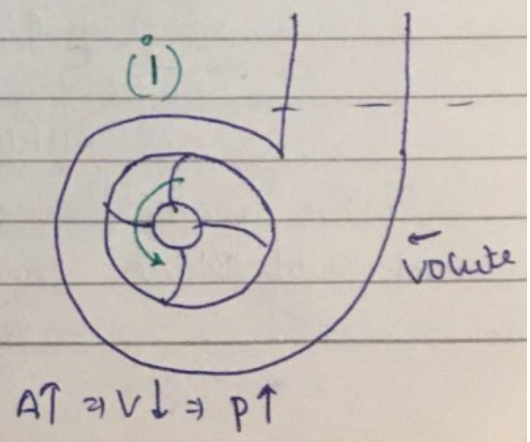
③ open impeller

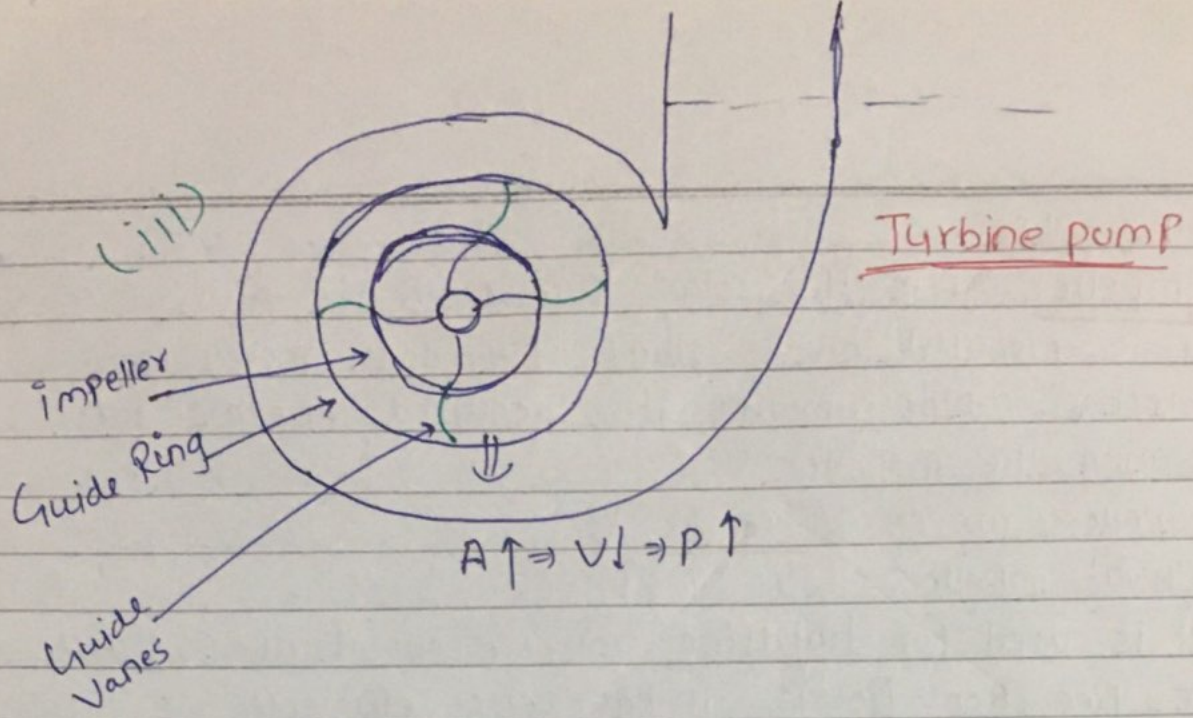
It is used to perform rough duties like handling abrasive liquids, mixture of water & sand.

② casing

① Volute or spiral casing





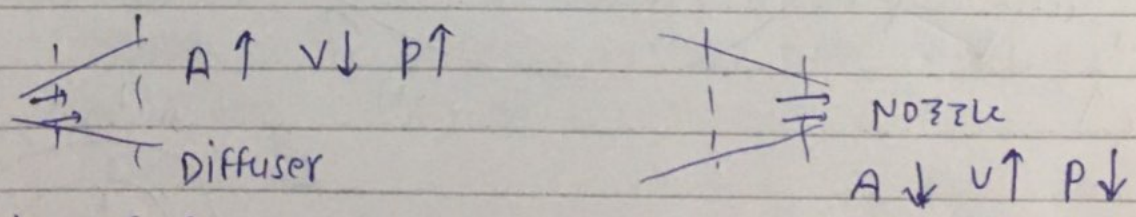


Shape of casing is such that Area of flow around periferi of impeller gradually increases towards the delivery side. This gradually increased area causes decrease in velocity and hence increase in pressure.

ii) Volute casing with Vortex chamber

It is an improvement over the simple Volute casing. in this casing, a circular chamber known as Vortex Chamber is provided in b/w impeller ~~of~~ and casing. In this Chamber motion is free vortex motion and hence KE↓ significantly while liquide is flowing through this chamber which results in corresponding rise in pressure.

iii) Diffuser pump or turbine pump



In this type of casing impeller is surrounded by a series of guide blades mounted on a ring called diffuser ring

Guide Vanes provide gradually increasing passage for the flow of fluid where velocity ↓ and pressure ↑

Guide Vanes are so designed that liquid coming out of an impeller enters the guide passage smoothly After the exist from guide passage water enters the casing which can be circular or spiral.

Based on number of entrance to impeller pumps can be single suction pump or Double suction pump In single suction pump fluid is admitted from the suction pipe on one side of impeller and hence there is an axial thrust. However in double suction pump fluid enters from both sides of impeller and hence axial thrust is neutralised.

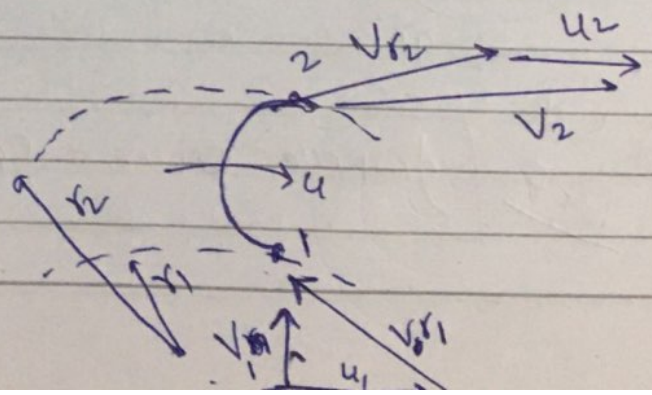
On the Basis of working head centrifugal pumps can be classified as -

i) Low Lift pump
- Head < 15 m
- impeller with Volute casing is used

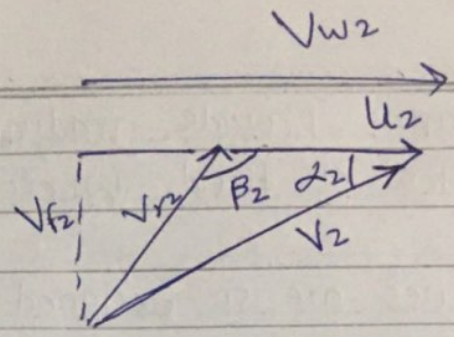
ii) Medium Lift pump $15 < H < 40$ m
casing with diffuser ring is used

iii) High Lift pump $H > 40$ m
Multistage pumping is used

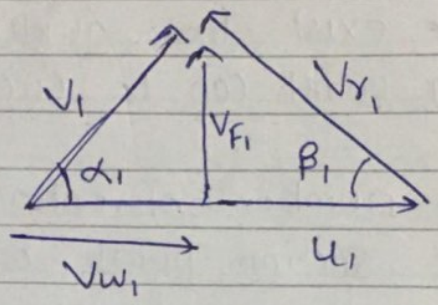
Impeller power
(Work done by impeller on water)



Exit velocity Angle



Inlet velocity Angle



Work done by impeller on Water –

$= \omega \times T_{impeller\ on\ Water}$

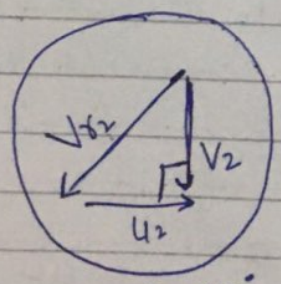
$= \omega \times$ Rate of change of angular momentum in clockwise dirⁿ

$= \omega \times$ (Angular momentum flux out in ↻ dirⁿ − Angular momentum flux in ↻ dirⁿ)

$= \omega \times (\rho Q V_{w_2} r_2 - \rho Q V_{w_1} r_1)$

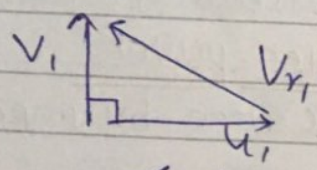
$= \boxed{\rho Q (V_{w_2}(\omega r_2) - V_{w_1}(\omega r_1))}$

Impeller power $= \rho Q [V_{w_2} U_2 - V_{w_1} U_1]$



$\alpha_2 = 90°$

Discharge is radial @ entry



$\alpha_1 = 90°$

$V_1 = V_{f_1}$; $V_{w_1} = 0$

*** PRE $\boxed{\text{Impeller power} = \rho Q V_{w_2} U_2}$

Euler's Head (H_e) $= \dfrac{\text{impeller power}}{\text{Wt. of fluid}}$

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

$$\boxed{H_e = \frac{V_{w_2} u_2}{g}} *$$

practically it's observed that impeller efficiency is Max. when entry to impeller blade is radial i.e $\alpha_1 = 90^\circ$, $V_{w_1} = 0$ and $V_1 = V_{f_1}$

$$H_e = \frac{V_{w_2} u_2 - V_{w_1} u_1}{g}$$

$$= \underbrace{\frac{V_2^2 - V_1^2}{2g}}_{\Delta(KE)} + \underbrace{\frac{u_2^2 - u_1^2}{2g} + \frac{V_{r_1}^2 - V_{r_2}^2}{2g}}_{\Delta(\text{press. Energy})}$$

$\dfrac{V_2^2 - V_1^2}{2g}$ → represents increase in KE of fluid

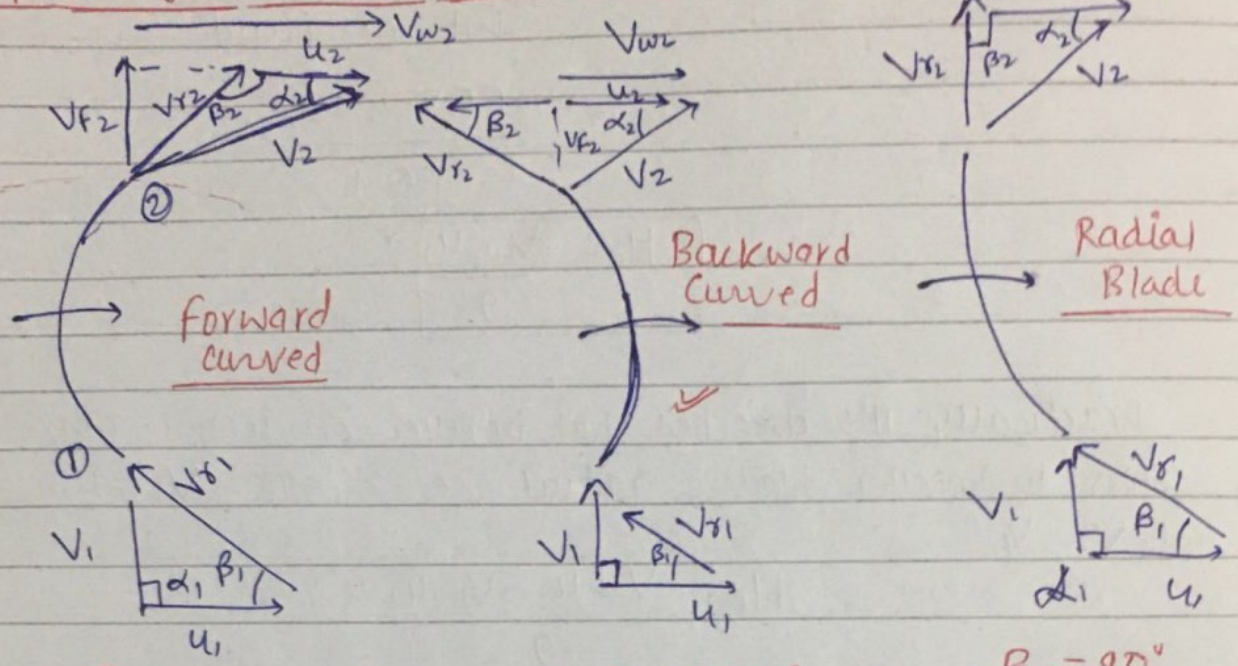
$\dfrac{u_2^2 - u_1^2}{2g}$ → represents increase in pressure energy due to Centrifugal action

$\dfrac{V_{r_1}^2 - V_{r_2}^2}{2g}$ → represents increase in pressure energy due to retardation of fluid while flowing through the blade passage.

* Energy transfer takes place in impeller whereas energy transformation (KE to P.E) takes place in casing.

~~Not for PPT~~

Velocity Triangle at outlet in diff^n cases



$\beta_2 > 90°$	$\beta_2 < 90°$	$\beta_2 = 90°$
Slow runner	Fast runner	Medium speed Runner
$Q\uparrow \rightarrow H\uparrow \rightarrow P\uparrow\uparrow$	$Q\uparrow \rightarrow H\downarrow = P$	

IMP

- if outlet tip of blade curves in dir^n of Blade motion then the Blade is said to be Forward curved blade ($\beta > 90°$)
  Forward curved Vane provide very low efficiency.

- for forward curved blade with $Q\uparrow$, Head developed $\uparrow$ but power req. $\uparrow\uparrow$ exponentially and hence there is a chance of burn out of motor. therefore these are not used practically.

- if outlet tip of blade curves in dir^n opposite to that of blade motion then blade is said to be Backward curved blade ($\beta_2 < 90°$), Fast runner.
  - $Q\uparrow$, Head developed $\downarrow$ and P req. remains nearly constant.

These provides Max. efficiency Hence most of the Centrifugal pumps are these only

- if outlet tip of blade makes an angle of 90° with the dir^n of blade motion then it is called Radial blade
- Medium speed runner

Relation B/W Head developed and Discharge, power

Head imparted by impeller to water $= V_{w_2} u_2 / g$

Neglecting losses in pump

$$\text{Head developed} = \frac{V_{w_2} u_2}{g}$$

from velocity triangle $\tan\beta_2 = \dfrac{V_{f_2}}{u_2 - V_{w_2}}$

$$u_2 - V_{w_2} = V_{f_2} \cdot \cot\beta_2$$

$$V_{w_2} = u_2 - V_{f_2} \cot\beta_2$$

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

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

$$V_{w_2} = u_2 - \frac{Q}{\pi D_2 B_2} \cot\beta_2$$

$$H = \left(u_2 - \frac{Q}{\pi D_2 B_2} \cot\beta_2\right) \frac{u_2}{g}$$

$$H = \frac{u_2^2}{g} - \frac{Q u_2}{\pi D_2 B_2 g} \cot\beta_2$$

$$H = K_1 - K_2 Q$$

$$K_1 = \frac{u_2^2}{g} \qquad K_2 = \frac{u_2}{\pi D_2 B_2 g} \cot\beta_2$$

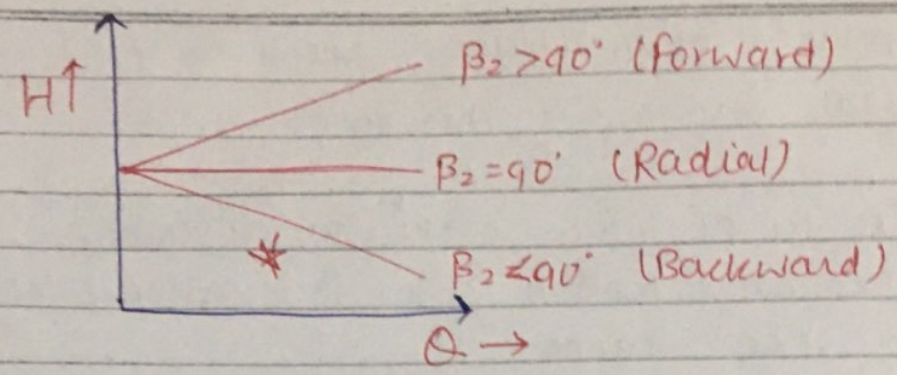
for forward Curved blade, $\beta_2 > 90°$; $\cot\beta_2 = -ve$

$K_2 = -ve$

For Backward curved blade ; $\beta_2 < 90°$

$\cot\beta_2 = +ve$, $K_2 = +ve$

for radial blade , $\beta_2 = 90°$ $\cot\beta_2 = 0$ $K_2 = 0$



Neglecting losses,

power input = $\rho g Q H$

$$P \propto QH$$

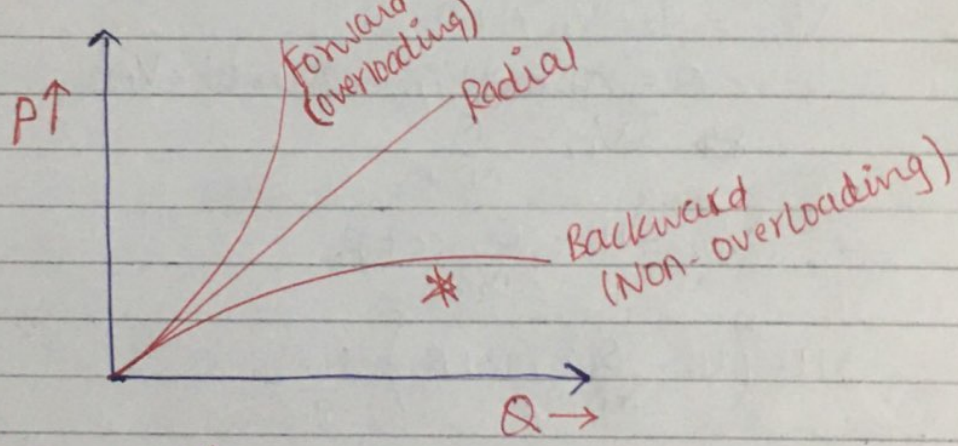
$$P \propto Q(K_1 - K_2 Q)$$

$$P \propto K_1 Q - K_2 Q^2$$

forward curved blade ; $\beta_2 > 90° \Rightarrow K_2 = -ve$

Backward curved blade ; $\beta_2 < 90° \Rightarrow K_2 = +ve$

Radial Blade ; $\beta_2 = 90°$, $P \propto K_1 Q$



PRE

if $Q = 0$

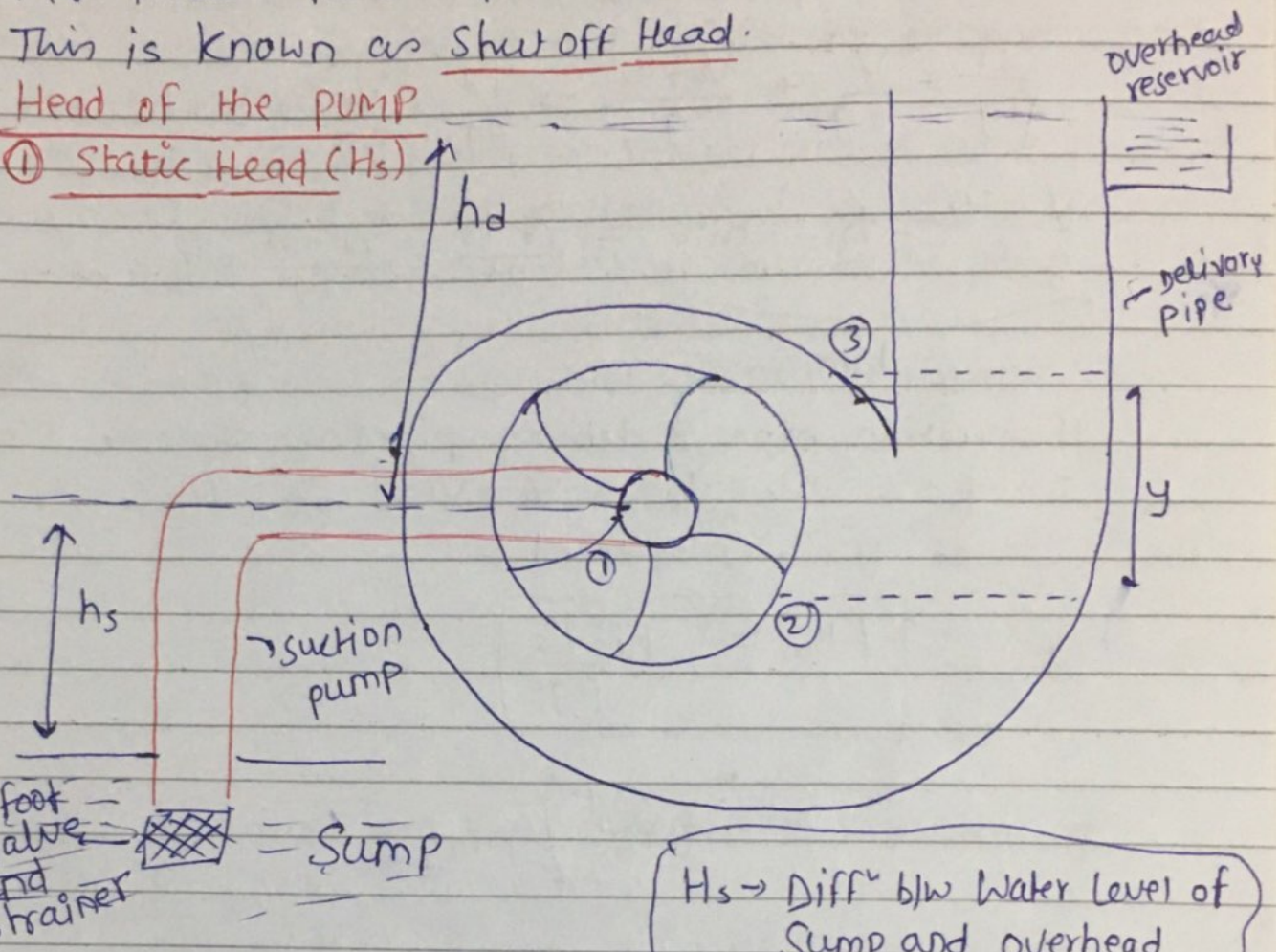
$H = \frac{u_2^2}{g}$ → Shut off Head

for a given angle β_2, theortical Head He Varies linearly with Q. when $Q = 0$ there is a finite +ve Value of He. this indicates that if delivary Value of a running pump is Completly closed shut and pump is kept running,

a +ve pressure head is produced by pump
This is known as Shut off Head.

Head of the pump

① Static Head (H_s)



$H_s \rightarrow$ Diff^n b/w Water level of Sump and overhead reservoir where water is to be stored after pumping

$h_s \rightarrow$ suction head
$h_d \rightarrow$ delivery head.

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

② Manometric head (H_m)

Demand $\rightarrow$ $H_m = h_s + h_d + h_{fs} + h_{fd} + \frac{V_d^2}{2g}$ —— ①

Net Head imparted by impeller to water

$$= \frac{V_{w2} u_2}{g} - h_{L,i} - h_{L,c} \quad \text{—— ②}$$

$$H_m = \frac{V_{w2} u_2}{g} - h_{L,i} - h_{L,c}$$

Applying energy eq^n b/w points (1-1) and (2-3)

$$\frac{P_s}{\rho g} + \frac{V_s^2}{2g} + \frac{V_{w2} u_2}{g} = \frac{P_d}{\rho g} + \frac{V_d^2}{2g} + y + h_{L,i} + h_{L,c}$$

$$\underbrace{\frac{V_{w2} u_2}{g} - h_{L,i} - h_{L,c}}_{H_m} = \frac{P_d - P_s}{\rho g} + \frac{V_d^2 - V_s^2}{2g} + y$$

if suction pipe & delivary pipes are of same size.

$d_s = d_d \Rightarrow V_s = V_d$

if y is Neglected.

$$\boxed{H_m = \frac{P_d - P_s}{\rho g}}$$

$$= \left(\frac{\Delta P}{\rho g}\right)_{pump}$$

$$= \left(\frac{\Delta P}{\rho g}\right)_{impeller} + \left(\frac{\Delta P}{\rho g}\right)_{casing}$$

$$\boxed{\begin{matrix} P_3 > P_2 \\ V_3 < V_2 \end{matrix}} \qquad \boxed{\begin{matrix} P_2 > P_1 \\ V_2 > V_1 \end{matrix}} \qquad V_{r_2} < V_{r_1}$$

$H_m \rightarrow$ it is the total head that must be produced by pump to satisfy external req.

The external req. in to lift the water to the req level (H_s) meet the energy req. of losses in suction and delivary pipe and to deliver the water @ a velocity V_d ~~hence~~.

hence we get the eq^n ①

This external req. has to be met by pump hence net head imparted by pump must cater for manometric head.
that is eq^n ②

point ① is in suction pipe very close to pump
point ② is @ exit of impeller.
point ③ is in delivery pipe very close to the pump

pressure rise accross impeller

Applying Energy eq^n b/w inlet and exit of impeller

$$\frac{P_s}{\rho g} + \frac{V_s^2}{2g} + Z_1 + \frac{V_{w2}u_2}{g} = \frac{P_2}{\rho g} + \frac{V_2^2}{2g} + Z_2 + h_{L,i}$$

$$\frac{P_2 - P_s}{\rho g} = (\Delta P)_i = \frac{V_s^2}{2g} - \frac{V_2^2}{2g} - h_{L,i} + \frac{V_{w_2} u_2}{g}$$

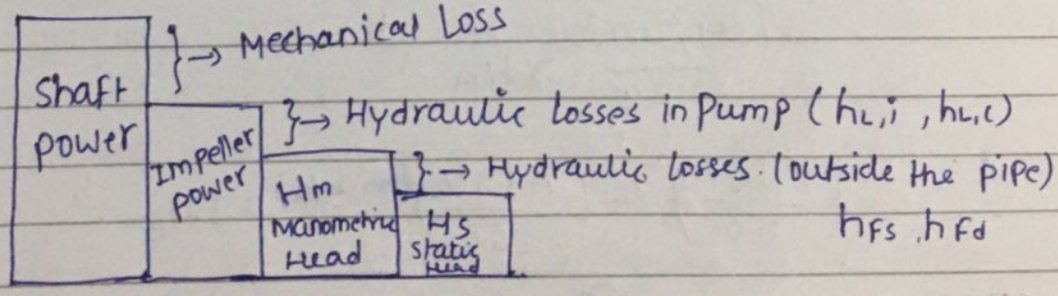
pressure rise in casing

Applying Energy eq^n b/w entry and exit of casing

$$\frac{P_2}{\rho g} + \frac{V_2^2}{2g} = \frac{P_d}{\rho g} + \frac{V_d^2}{2g} + y + h_{L,c}$$

$$\left(\frac{P_d - P_2}{\rho g}\right) = (\Delta P)_c = \frac{V_2^2 - V_d^2}{2g} - y - h_{L,c}$$

Losses and Efficiencies



1) Mechanical Efficiency $\eta_m = \dfrac{\text{impeller power}}{\text{shaft power}}$

2) Manometric Efficiency $\eta_{mano} = \dfrac{\text{Head developed } (H_m)}{\text{Head imparted by impeller to water } (H_e)}$

$$= \frac{H_m}{V_{w_2} u_2 / g} \Rightarrow \boxed{\eta_{mano} = \frac{g H_m}{V_{w_2} u_2}}$$

3) Volumetric efficiency $\eta_{vol.} = \dfrac{\text{Quality of water delivery by pump}}{\text{Quantity of water passing through impeller}}$

4) Overall efficiency $\eta_o = \dfrac{\rho g Q H_m}{\text{shaft power}}$

Minimum starting speed of pump to deliver liquid:-

When the pump is started there will be no flow of water until the head imparted by impeller is large enough to overcome the manometric head

$$H_e \geq H_m$$

$$\left(\frac{V_2^2 - V_1^2}{2g}\right) + \left(\frac{u_2^2 - u_1^2}{2g}\right) + \left(\frac{V_{r_1}^2 - V_{r_2}^2}{2g}\right) \geq H_m$$

When the flow has not started ; V_1, V_2, V_{r_1}, $V_{r_2} = 0$

$$\frac{u_2^2 - u_1^2}{2g} \geq H_m$$

$$\left(\frac{\pi D_2 N}{60}\right)^2 - \left(\frac{\pi D_1 N}{60}\right)^2 \geq 2g H_m$$

$$\boxed{N \geq \sqrt{\frac{2g H_m}{\left(\frac{\pi}{60}\right)^2 (D_2^2 - D_1^2)}}}$$

Inlet ⇓ inner Outlet ⇓ outer

Diameter of Impeller and pipes

(i) Outer dia of impeller (D_2) $U_2 = \frac{\pi D_2 N}{60} = K_u \sqrt{2gH_m}$

(2) Inner Dia of impeller (D_1)

$$\boxed{D_1 = \frac{D_2}{2}}$$

(3) Dia of Suction pipe (d_s)

$$Q = \frac{\pi}{4} d_s^2 \times V_s \Rightarrow d_s = \sqrt{\frac{4Q}{\pi V_s}}$$

4) Diameter of delivery pipe (d_d)

$$Q = \frac{\pi}{4} d_d^2 \times V_d \Rightarrow d_d = \sqrt{\frac{4Q}{\pi V_d}}$$

5) Least dia of impeller ($D_{2,min}$)

$$\frac{u_2^2 - u_1^2}{2g} \geq H_m \Rightarrow \left(\frac{\pi D_2 N}{60}\right)^2 - \left(\frac{\pi D_1 N}{60}\right)^2 \geq 2gH_m.$$

$$\left(\frac{\pi N}{60}\right)^2 D_2^2 \left(1 - \frac{1}{4}\right) \geq H_m$$

$$D_1 = \frac{D_2}{2}$$

Que. A centrifugal pump lifts 100 l/sec of a liquid against a net head of 150 kPa. Break power req. is 18 kW when the liquid is water with a specific wt. of 9.8 kN/m³. What would be the break power if the liquid is a solvent having a relative density of 0.8 instead of water with all other factors remaining the same as before.

Ans: 18 kW.

Cavitation if $P_{abs} < P_{v.p}$ cavitation occurs. (may)
cavitation can be noted experimently by sudden drop in efficiency and head.

In centrifugal pumps pressure is suction pressure @ the entry to the pump and hence chances of cavitation are Max. @ the entry to the pump.

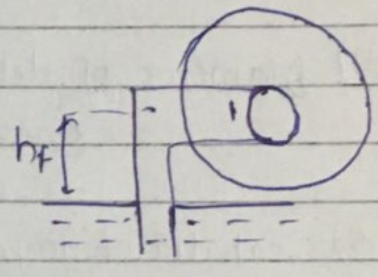
for no cavitation

$$\left(\frac{P_1}{\rho g}\right)_{abs.} > \left(\frac{P_{vap}}{\rho g}\right)$$

Applying Energy EQ^n b/w inlet of pump and surface of sump

$$\frac{P_{atm}}{\rho g} = \frac{P_1}{\rho g} + \frac{V_s^2}{2g} + h_s + h_{fs}$$

$$\frac{P_1}{\rho g} = \frac{P_{atm}}{\rho g} - \frac{V_s^2}{2g} - h_s - h_{fs}$$



for no cavitation $\frac{P_1}{\rho g} > \frac{P_{vap}}{\rho g}$

$$\boxed{\frac{P_{atm}}{\rho g} - \frac{V_s^2}{2g} - h_s - h_{fs} > \frac{P_{vap}}{\rho g}}$$

IMP.

Chances of cavitation can be reduced by

i) $P_{atm} \uparrow$
ii) $V_s \downarrow \Rightarrow d_s \uparrow$
iii) $h_s \downarrow$
iv) $h_{fs} \downarrow$ → should be
v) $P_{vap} \downarrow$ ($T \downarrow$)

Max. Suction lift ($h_{s,max}$)

$$h_s < \frac{P_{atm} - P_v}{\rho g} - \frac{V_s^2}{2g} - h_{fs}$$

$$(h_s)_{max} = \frac{P_{atm} - P_v}{\rho g} - \frac{V_s^2}{2g} - h_{fs}$$

for Water $\frac{P_v}{\rho g} = 0.5m$ $\frac{P_{atm}}{\rho g} = 10.3$

6–8 m

Net positive suction head (NPSH)

$$(NPSH)_{avail} = \left(\frac{P_s}{\rho g} + \frac{V_s^2}{2g}\right) - \frac{P_v}{\rho g}$$

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

$$* \boxed{(NPSH)_{avail.} = \frac{P_{atm} - P_v}{\rho g} - h_s - h_{fs}}$$

for No cavitation;

$$(NPSH)_{avail} > (NPSH)_{req.}$$

(NPSH) available is the energy available, in forms of pressure & KE @ the inlet to the pump above vapor pressure of water. it is a measure of margin against Liquid Vaporization and hence Cavitation Susceptibility

(NPSH) req. is decided by Manufecturer.

Thoma's Cavitation Factor (σ)

$$\sigma = \frac{NPSH}{H_m}$$

for no cavitation $\sigma > \sigma_c$

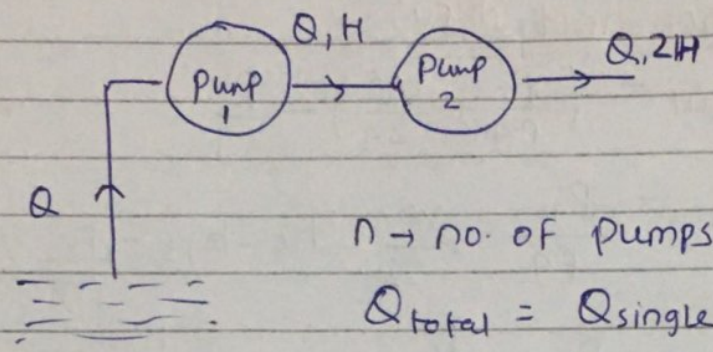
Critical Cavitation factor $= f(N_s)$

$N_s \uparrow \Rightarrow \sigma_c \uparrow \Rightarrow$ chances of Cavitation.

Multistage pumping

i) pumps in Series — Q remains same

H ↑

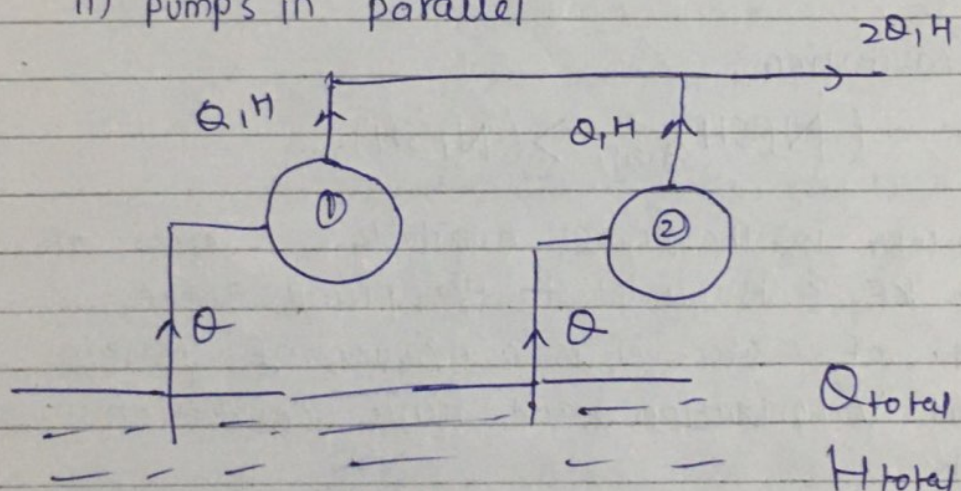


n → no. of pumps

$Q_{total} = Q_{single}$

$H_{total} = n \times H_{single}$

ii) pumps in parallel



$Q_{total} = n \times Q_{single}$

$H_{total} = H_{single}$.

Specific Speed (N_s)

speed of a geometrically similar pump so reduced in size that it would deliver $1 m^3$ of liquid per second against a head of 1 m.

Water condition of max. efficiency –

$$\frac{H}{N^2 D^2} = \text{const} \quad , \quad \frac{Q}{ND^3} = \text{const.}$$

$$\frac{P}{N^3 D^5} = \text{const.}$$

$$\frac{Q}{D^3 N} = \text{const.}$$

$$\frac{\sqrt{H}}{DN} = \text{const}$$

$$\frac{Q}{\left(\frac{\sqrt{H}}{N}\right)^3 N} = \text{const.}$$

$$\frac{N^2 Q}{(H)^{3/2}} = \text{const}$$

$$\frac{N\sqrt{Q}}{(H)^{3/4}} = \text{const.}$$

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

if $Q_2 = 1\ m^3/s$; $H_2 = 1m$

$N_2 = N_s$

$$\frac{N\sqrt{Q}}{H^{3/4}} = \frac{N_s\sqrt{1}}{H^{3/4}}$$

$$\boxed{N_s = \frac{N\sqrt{Q}}{(H_m)^{3/4}}}$$

N :- speed of pump in rpm
Q :- Discharge in m^3/s (Lit/s)
H_m :- Manometric Head in (m)

Non-Dimensional Specific Speed (Shape Number)

$$\boxed{N_s = \frac{N\sqrt{Q}}{(gH_m)^{3/4}}}$$

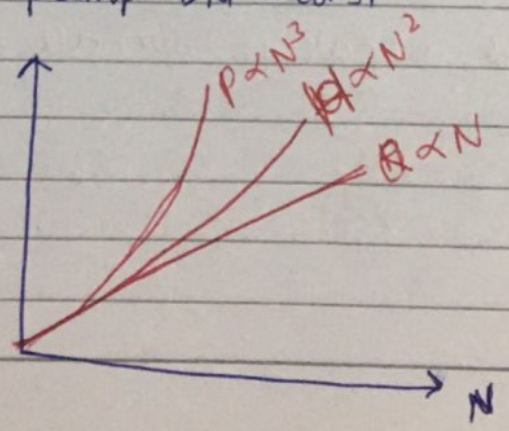
$N \rightarrow rad/s$
$Q \rightarrow m^3/s$
$H_m \rightarrow m$

While calculating N_s of a double Suction pump, Q is taken as half of total discharge

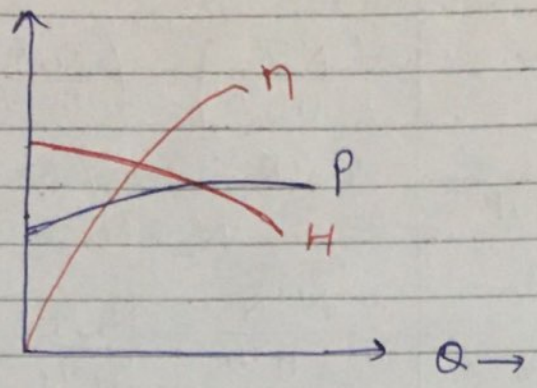
Main characteristic curve

for same pump Dia = const.

$H \propto N^2$
$Q \propto N$
$P \propto N^3$



operating characteristic curve



priming

Before starting a pump it's impeller and suction pipe have to be filled with water to remove any air or gas from the waterways of pump. this is known as priming. if pump casing is not filled with water @ start of pump, impeller rotates in air filled casing this can't create the necessary differential head of water b/w sump and inlet as density of air is much less than that of water. Consequently the pump doesn't do it's work of pumping of water.

A 1-way valve called foot valve is provided @ entrance of suction pipe to prevent the water from draining out to sump when a running pump is closed.

Que: 2 centrifugal pumps worked in parallel with a common delivery head of 18 m. first pump delivers 90 l/s @ an efficiency of 60%, 2nd pump delivers 60 l/s @ an efficiency of 50%. what is overall efficiency of system?

A) 62.5%
B) 55.5%
C) 58%
D) 52%

$H_1 = H_2 = H_{Total} = 18\,m$

$Q_{total} = Q_1 + Q_2$

$= 90 + 60$

$= 150$ lps.

power input to the 1st pump $= \dfrac{\rho g Q_1 H}{\eta_1}$

power input to 2nd pump $= \dfrac{\rho g Q_2 H}{\eta_2}$

$$\eta_0 = \frac{\rho g Q_{total} H_{total}}{P_i/\eta_1 + P_i/\eta_2} = \frac{\rho g Q_{total} H}{\dfrac{\rho g Q_1 H}{\eta_1} + \dfrac{\rho g Q_2 H}{\eta_2}}$$

$$= \frac{150}{\dfrac{90}{60} + \dfrac{60}{50}}$$

$= 55.5\%$

Que. Determine the no. of impellers req^r for a multistage pump to lift 4000 lit/minute against a total head of 80 m. @ a speed of 750 rpm. Given that specific speed for each impeller should be b/w 720 - 780 units

A) 6 B) 5 C) 4 D) 3

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

$$750 = \frac{750\sqrt{4000/60}}{H^{3/4}} \Rightarrow H = 16.4\,m$$

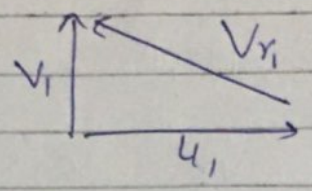
pumps are connected in series *

No. of impellers req^r $= \dfrac{80}{16.4} \simeq 5$

Assumption

Unless o/w mentioned, it is assumed that

i) Absolute velocity is radial @ the inlet



$\alpha_1 = 90°$, $V_{w1} = 0$

$V_1 = V_{f1}$

ii) Velocity of flow is constant.

Qu. Water enters radially through a centrifugal pump whose impeller has dia = 30 cm. and width = 15 cm the corresponding dimension @ outer periferi are 60 cm and 7.5 cm, The blades are curved backward @ 30° to tangent @ exit, Discharge is 0.225 cumec if the rotational speed of impeller is 1200 rpm. and the pump operates under the manometric head of 115 m. Calculate ① theortical head developed and the manometric efficiency ② The pressure rise across impeller, assuming losses = 10% of velocity head @ exit. ③ pressure rise and loss of head in volute casing ④ vane angle @ inlet. ⑤ assuming overall efficiency = 65%, calculate the power req. to drive the pump and the corresponding mechanical efficiency.

i) Theortical head developed (H_e)

$$H_e = \frac{V_{w_2} u_2}{g}$$

$$u_2 = \frac{\pi D_2 N}{60} = \frac{3.14 \times 0.6 \times 1200}{60} = 37.7 \text{ m/s}$$

$\underline{V_{w2}} \Rightarrow \tan\beta_2 = \dfrac{V_{f2}}{U_2 - V_{w2}}$

$$V_{f2} = \frac{Q}{\pi D_2 B_2} = \frac{0.225}{3.14 \times 0.6 \times 0.075} = 1.59 \text{ m}$$

$$\tan 30^\circ = \frac{1.59}{37.7 - V_{w2}}$$

$$V_{w2} = 34.94 \text{ m/s}$$

$$H_e = \frac{V_{w2} U_2}{g} = \frac{34.94 \times 37.7}{9.81} = 134.27 \text{ m}$$

$$\eta_{mano.} = \frac{H_m}{H_e} = \frac{115}{134.27} = 85.64\%$$

ii) pressure rise accross impeller

Apply Energy EQ^n b/w ① & ②

$$\frac{P_1}{\rho g} + \frac{V_1^2}{2g} + Z_1 + H_e = \frac{P_2}{\rho g} + \frac{V_2^2}{2g} + Z_2 + H_{L,i}$$

$\left(\dfrac{P_2 - P_1}{\rho g}\right) \rightarrow$ pressure rise accross impeller.

$$\left(\frac{P_2 - P_1}{\rho g}\right) = \frac{V_1^2}{2g} - \frac{V_2^2}{2g} + H_e - 0.1\frac{V_2^2}{2g}$$

$$V_1 = V_{f1} = V_{f2} = 1.592 \text{ m/s}$$

$$V_2 = \sqrt{V_{w2}^2 + V_{f2}^2} = \sqrt{34.94^2 + 1.592^2} = 34.978 \text{ m/s}$$

$$(\Delta P)_i = 65.814 \text{ m}$$

iii) $$H_m = \frac{(\Delta P)_i}{\rho g} + \frac{(\Delta P)_c}{\rho g}$$

$$115 \text{ m} = \frac{65.814}{\cancel{1000 \times 9.81}} + \frac{(\Delta P)_c}{\rho g} \Rightarrow (\Delta P)_c = 49.19 \text{ m}$$

$$H_m = \frac{V_{w_2} u_2}{g} - h_{L,i} - h_{L,c}$$

$$115 = 134.27 - 6.23 - h_{L,c}$$

$$h_{L,c} = 13.05\,m$$

iv) $\beta_1 = ?$

$$\tan\beta_1 = \frac{V_{f_1}}{u_1} = \frac{1.592}{u_2/2}$$

$$\beta_1 = 4.83^\circ$$

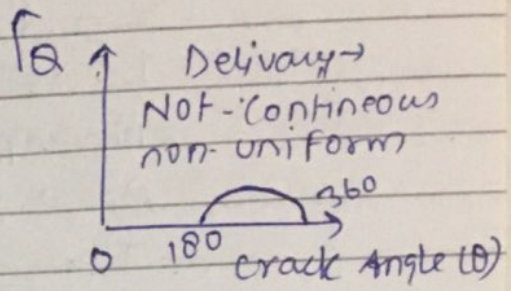
v) power req^r to drive the pump $= \dfrac{\rho g Q H_m}{\eta_o}$

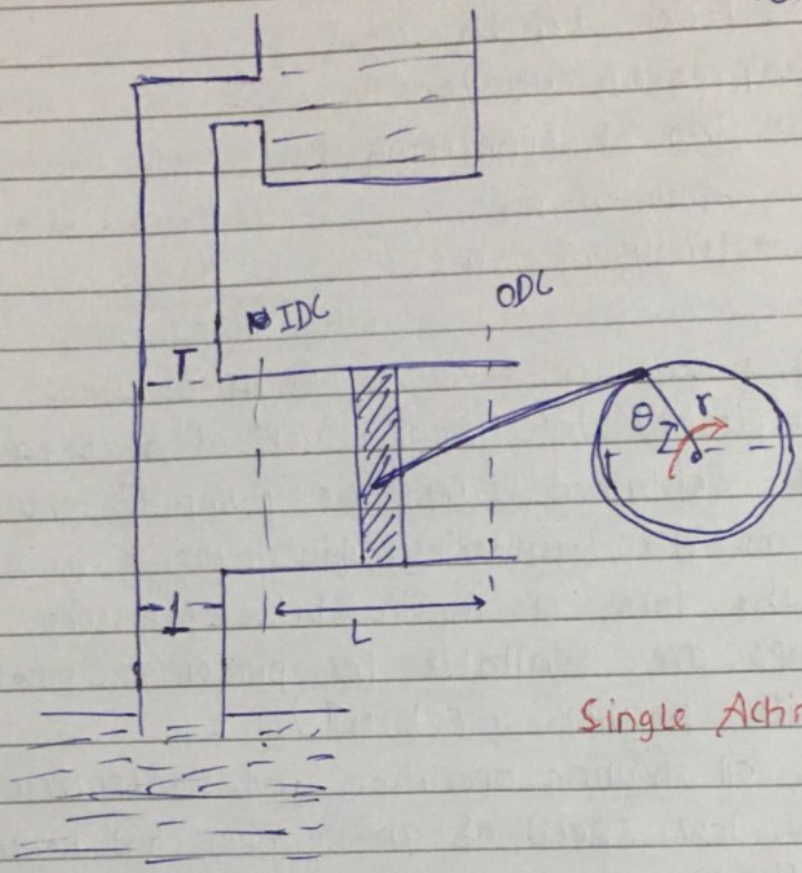
$$= 390.15\ kW$$

$$\eta_o = 85.64 \times \eta_{mech}$$

$$\eta_{mech} = 75.89\ \%$$

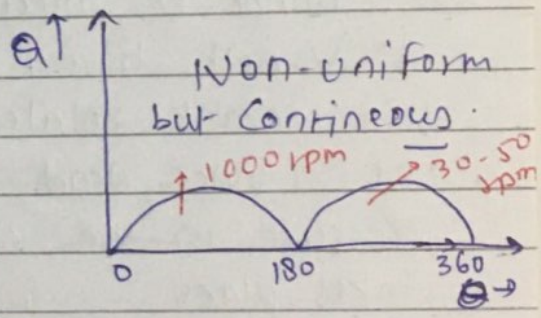
Reciprocating pump

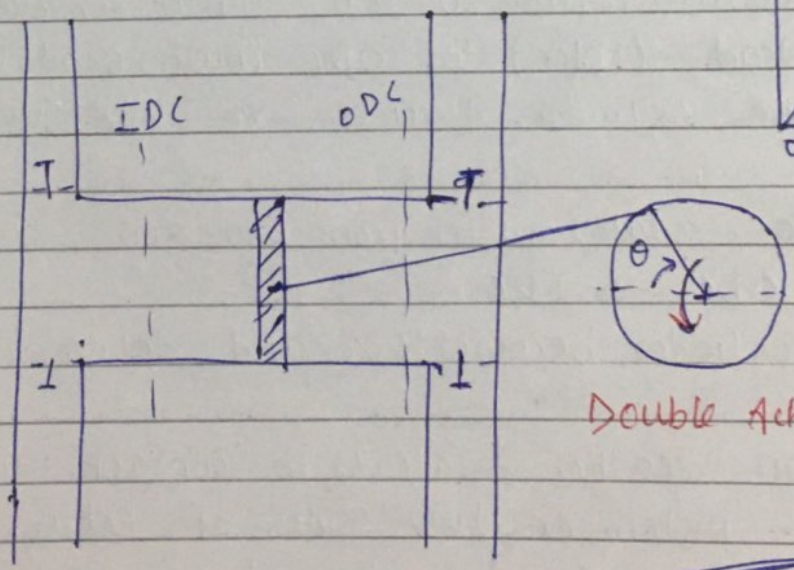




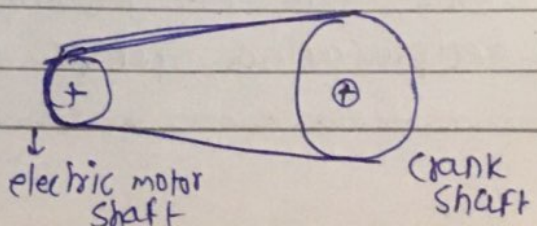
θ; Angle Tu

Single Acting reciprocating pump





Double Acting Reciprocating pump.



θ - Angle turned by crank
r → crank radius
L → Stroke Length
A → Area of cylinder
A_p → Area of connecting rod
a → Area of pipe
l → Length of pipe

In reciprocating pumps or +ve displacement pumps energy transfer is accomplished by changing Volⁿ of a certain mass of fluid contained within the pump by physically displacing one of it's boundaries. This increase in energy is menifested in the form of rise in static pressure.
Reciprocating pumps are suitable for pumping smaller quantity @ greater heights. ($Q \downarrow H \uparrow$)
It is used in oil drilling operation and in concrete pumps
It works @ a low speed of 30-50 rpm and hence it cannot be directly coupled to the electric motor.
A V-belt drive is used to reduce the speed.
When crank rotates from 0-180°, piston moves from IDC (Inner dead center) to ODC (Outer Dead center)
pressure in the cylinder becomes -ve and suction takes place
As the crank rotates ~~piston~~ from 180-360°. Piston moves from ODC → IDC.
pressure in cylinder becomes +ve and delivery takes place.
Depending on weather water is in contact with one side of piston or two sides of piston, reciprocating pumps can be classified as

i) Single acting

ii) Double Acting.

* In single acting pump Discharge is non-continuous as well as non-uniform whereas in Double acting pump Discharge is continuous ~~and~~ but non-uniform.

power input in Reciprocating pump

for single Acting pump

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

$$P_{i/p} = \rho g Q_{th} (h_s + h_d)$$

*

$$P_{i/p} = \rho g \left(\frac{ALN}{60}\right)(h_s + h_d) \rightarrow \text{theortical}$$

for Double Acting pump

$$Q = \frac{ALN}{60} + \frac{(A - A_p)LN}{60}$$

if A_p is neglected

*

$$P_{i/p} = \rho g \left(\frac{2ALN}{60}\right)(h_s + h_d)$$

This is the theortical power input because here losses have not been considered

Slip, percentage slip, cofficient of Discharge (C_d)

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

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

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

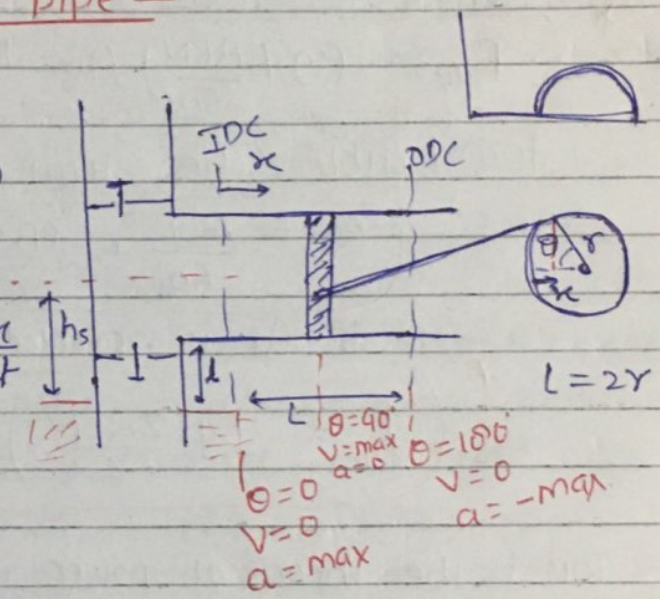
if $Q_{actual} > Q_{th}$

$C_d > 1$

slip → −ve.

-ve slip occurs, when Suction pipe is long, delivery pipe is short and speed of rotation is high.
in this case inertia force in Suction pipe will be large in comparison to force outside delivery valve due to which, delivery valve opens, before Suction Stroke is completed. Thus Some of liquid is pushed directly from Suction pipe to delivery pipe and hence pump will discharge more quantity then the swept Vol^m of pump

Effect of acceleration of piston on velocity and pressure in Suction and delivery pipe —



x = Displacement of piston
$= r - r\cos\theta$
$= r - r\cos\omega t$

Velocity of piston : $V_p = \frac{dx}{dt}$

$\frac{dx}{dt} = 0 - r(-\omega\sin\omega t)$

$V_p = r\omega\sin\omega t$

~~Acceleration of piston ; A_p~~

Velocity of water in cylinder $= r\omega\sin\omega t$

Velocity of water in pipe $(v) \Rightarrow$

$$A \times r\omega\sin\omega t = a \times v$$

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

Acceleration of water in pipe

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

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

mass of Water in pipe × Acceleration of water in pipe

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

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

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

$$\boxed{H_a = \frac{A}{a} \frac{l}{g} r\omega^2 \cos\omega t}$$ * Acceleration head.

Acceleration head during suction stroke

$$H_{as} = \frac{A}{a_s} \frac{l_s}{g} r\omega^2 \cos\omega t$$

Acceleration head during delivery stroke

$$H_{ad} = \frac{A}{a_d} \frac{l_d}{g} r\omega^2 \cos\omega t$$

Suction stroke	Velocity of Water in pipe	Acceleration of Water in pipe.
Begining ($\theta = 0$)	0	MAX (Acl^n)
Middle ($\theta = 90°$)	MAX	0
End ($\theta = 180°$)	0	MAX (retardati

*

$$\boxed{H_{as} \text{ is max. @ begining of suction stroke}}$$

Effect of Variation in Velocity in pipe on friction in pipe

frictional losses in pipe (h_f) $= \frac{f l v^2}{2gd}$

$f \rightarrow$ friction factor

$l \rightarrow$ length of pipe

$d \rightarrow$ Dia of pipe

$v \rightarrow$ velocity of Water

Suction stroke

$$h_{fs} = \frac{f l_s V_s^2}{2g d_s} = \frac{fl}{2gd_s}\left(\frac{A}{a_s} r\omega \sin\omega t\right)^2 \longrightarrow \begin{cases} \to 0 & \text{begining} \\ \to \max. & \text{middle} \\ \to 0 & \text{End} \end{cases}$$

$$(h_{fs})_{max} = \frac{fl}{2gd_s}\left(\frac{A}{a_s} r\omega\right)^2$$

Delivery stroke

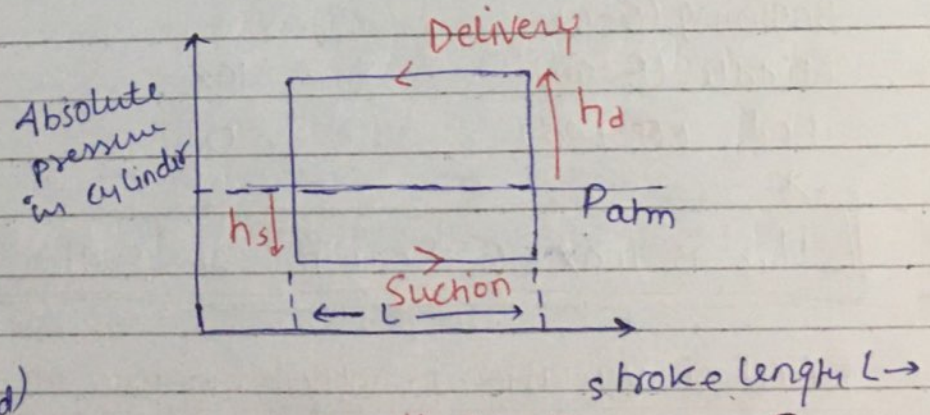
$$h_{fd} = \frac{f l_d}{2gd_d}\left(\frac{A}{a_d} r\omega \sin\omega t\right)^2 \longrightarrow \begin{cases} \to 0 & \text{Begining} \\ \to \max. & \text{middle} \\ \to 0 & \text{End.} \end{cases}$$

$$(h_{fd})_{max.} = \frac{f l_d}{2gd_d}\left(\frac{A}{a_d} r\omega\right)^2$$

NOTE Effect of friction head is max. @ middle of Suction Stroke and Middle of delivery Stroke

Indicator Diagram

It is the graph b/w absolute pressure in cylinder and stroke length



Ideal indicator diagrame

power input to reciprocating pump

$$P = \rho g\left(\frac{ALN}{60}\right)(h_s + h_d)$$

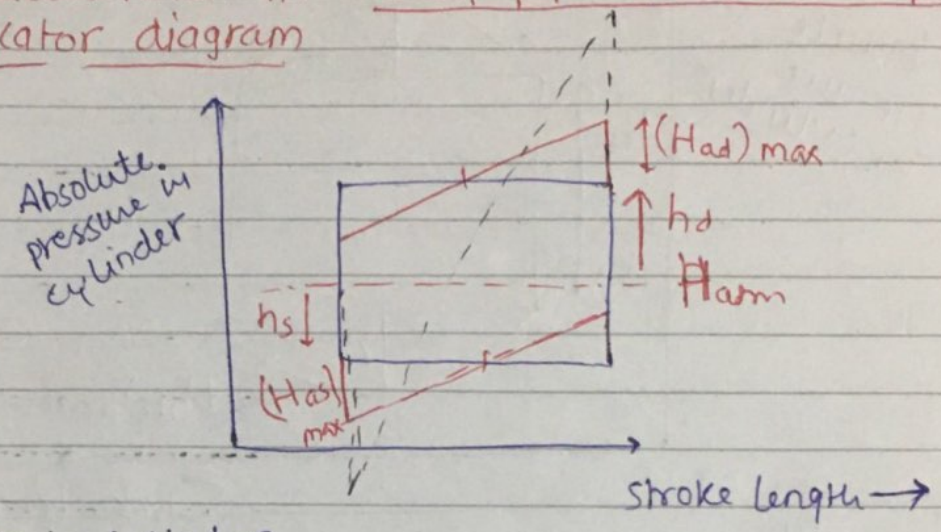
$$= \rho g \frac{AN}{60} L (h_s + h_d)$$

$$= KL(h_s + h_d)$$

$$= K \times \text{Area of indicator diagram}$$

* Area of indicator Diagram to some scale represents power input in a resiprocating pump

Effect of acceleration in delivery pipe and suction pipe on indicator diagram



pressure in cylinder

At begining of suction stroke = $H_{atm} - h_s - (H_{as})_{max}$

At end of suction stroke = $H_{atm} - h_s + (H_{as})_{max}$

At begining of delivery stroke = $H_{atm} + h_d + (H_{ad})_{max}$

At end of of delivery stroke = $H_{atm} + h_d - (H_{ad})_{max}$

Chances of cavitation are max. at begining of suction stroke.

For no cavitation, $H_{atm} - h_s - (H_{as})_{max} > H_{v.p}$

$\rightarrow \omega_1 =$

Cavitation can also occure @ end of delivery stroke

For no cavitation

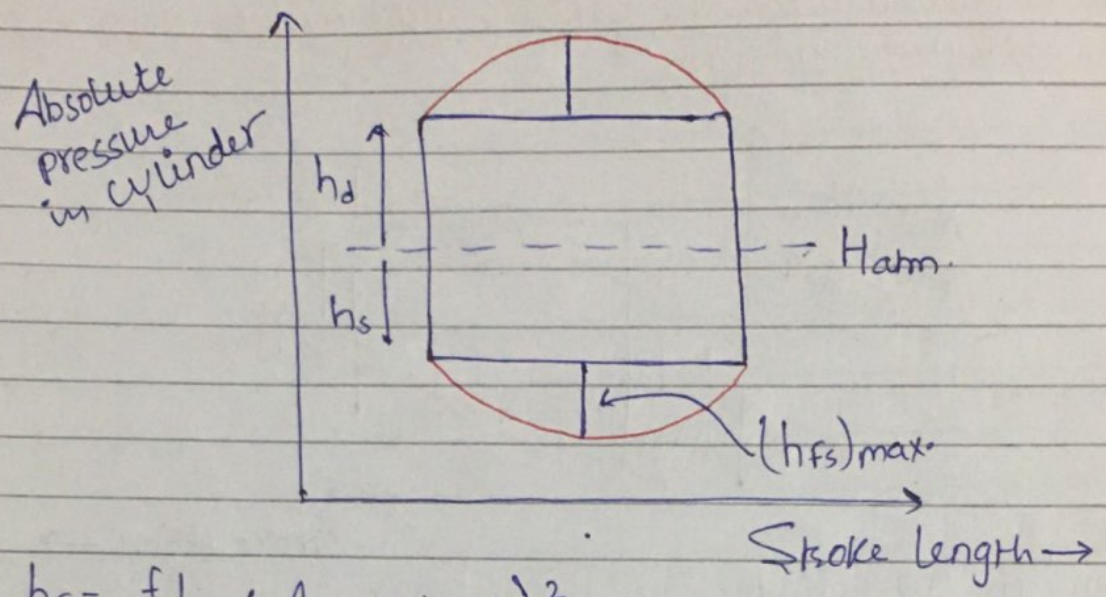
$$H_{atm} + h_d - (H_{ad})_{max} > H_{vp} \Rightarrow \omega_2$$

$$\text{Safe speed} = \text{Min}(\omega_1, \omega_2)$$

NOTE There is no effect of acceleration head on power input in a reciprocating pump

Due to acceleration head chances of cavitation increases.

Effect of friction in suction & Delivery pipe on indicator Diagram



$$h_f = \frac{fl}{2gd}\left(\frac{A}{a} r\omega \sin\omega t\right)^2$$

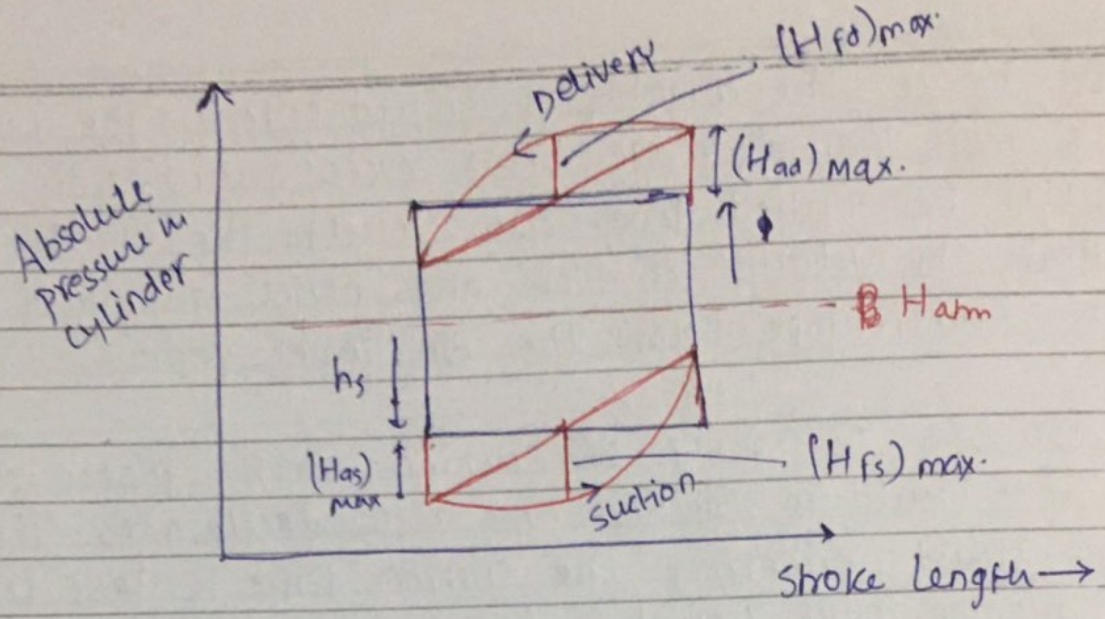
power input to pump = K × Area of indicator diagram.

$$= K \cdot L\,(h_s + h_d) + \frac{2}{3}(h_{fs})_{max} L + \frac{2}{3}(h_{fd})_{max} L$$

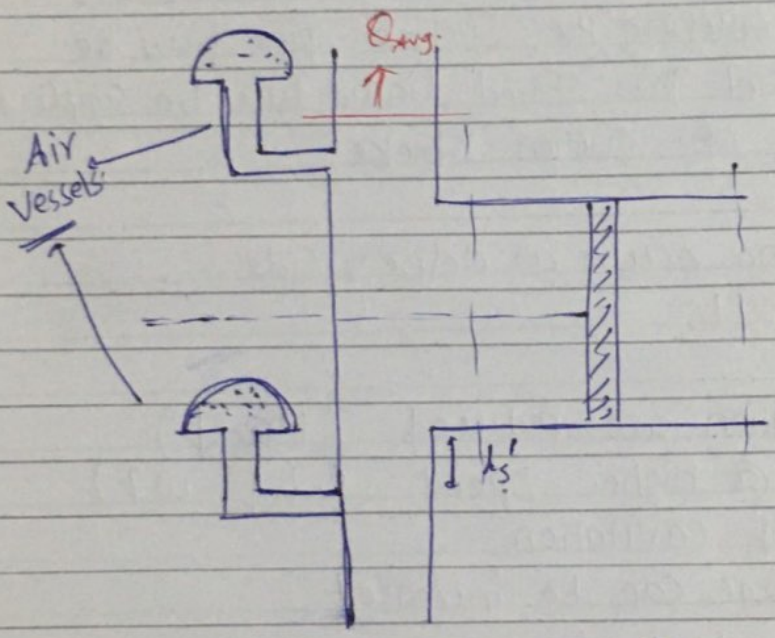
$$P_{i/p} = \rho g \left(\frac{ALN}{60}\right)\left(h_s + h_d + \frac{2}{3}(h_{fs})_{max} + \frac{2}{3}(h_{fd})_{max}\right)$$

* Power input to a reciprocating pump ↑ due to friction in Suction & delivery pipe
* There is no effect of friction head on possibility of cavitation

Effect of Acceleration and Friction in Suction and delivery pipe on indicator Diagram



Air Vessels



$$H_{atm} - h_s - (H_{as})_{max} > H_{v.p}$$

$$(H_{as})_{max} = \frac{A}{a_s}\left(\frac{l_s}{g}\right) r\omega^2$$

Air vessel is a large closed chamber fitted in suction and delivery pipe close to the cylinder.

During the first half of suction stroke piston accelerates and hence liquid in suction pipe is accelerated

and hence the quantity of liquid entering the cylinder is more than average. This excess quantity of liquid will be supplied from air vessel to the cylinder so that the velocity of flow and hence quantity of water in suction pipe below the Air vessel remains Uniform

During 2nd half of suction stroke piston decelerates and liquid in suction pipe also decelerates. The quantity of liquid entering the suction pipe is less than average. since liquid is flowing with an avg. velocity in suction pipe below air vessel, excess quantity of liquid entering the suction pipe will be stored into air vessel. This stored liquid will be supplied during first half of suction stroke

Similar phenomena occurs on delivery side.

Functions of Air vessels

1. Chances of cavitation are reduced ($H_{as} \downarrow$)
2. pump can be run @ higher speed ($h_s \downarrow$, $\omega \uparrow$) without danger of cavitation.
   Accordingly Discharge can be increased

$$Q = \frac{ALN}{60} \qquad \omega\uparrow \Rightarrow N\uparrow \Rightarrow Q\uparrow$$

3. suction head can be increased
4. Uniform Discharge can be obtained
5. power req. to drive the pump is reduced. because of reduction in power req. to overcome friction losses.

For single Acting pump % saving in work = 84.8 %.
Double Acting pump → 39.2 %

This power saved can be utalised to increase the delivery head.