

Hydraulic machine and industrial fluid power

(c) Chapterwise distribution of portions.

1. Hydraulic Turbines.
2. Centrifugal Pumps.
3. Pneumatic system.
4. Hydraulic System.

(2) Course contents:-

1. Hydraulic turbines:-

- 1.1 Definition and classification of hydraulic turbine.
- 1.2 Construction and working principle of impulse turbine.
- 1.3 Velocity diagram of moving blades, work done and derivation of various efficiencies of impulse turbine.
- 1.4 Velocity diagram to moving blade, work done and variation of various efficiency of Kaplan turbine.
- 1.6 Numerical on above.
- 1.7 Distinguish between impulse turbine and reaction turbine.

2. Centrifugal pump:-

- 2.1 Construction and working principle of centrifugal pump.
- 2.2 Work done and derivation of various efficiency of centrifugal pump.

2.3 Numerical on above.

3. Reciprocating pump:-

- 3.1 Describe construction and amp. working of single acting Reciprocating pump.
- 3.2 Describe construction and amp. working of double acting Reciprocating pump.

3.3 Derive the formula for power required to drive the pump (single acting and amp, double acting)

3.4 Define slip.

3.5 State positive and amp, negative slip and amp, establish relation between slip and coefficient of discharge.

3.6 Solve numerical on above.

4. Pneumatic Control system:-

4.1 Element - Filter - regulation - lubrication Unit.

4.2 pressure control valves.

4.2.1 - Pn. relief valve.

4.2.2 - Pn. regulation valves.

4.3 direction control valve.

4.3.1 - $3/2$ DCV, $5/2$ DCV, $5/3$ DCV.

4.3.2 - Flow control valve.

4.3.3 - Throttle valve.

4.4 ISO symbol of pneumatic component.

4.5 pneumatic circuit:-

4.5.1 - Direct control of single acting cylinder.

4.5.2 - Operation of double acting cylinder.

4.5.3 - Operation of double acting cylinder with

metering in and metering out control.

5. Hydraulic Control system.

5.1 Hydraulic system, its merit and demerits.

5.2 Hydraulic accumulators.

5.2.1 = Pressure control valve

5.2.2 = Pressure relief valve

5.2.3 = Pressure Regulation valve

5.3 Direction control valve.

5.3.1 = $3/2$ DCV, $5/2$ DCV, $5/3$ DCV

5.3.2 = flow control valve.

5.3.3 = Throttle valve.

5.4 Fluid power pump:

5.4.1 - External and internal gear pump.

5.4.2 - Vane pump.

5.4.3 - Radial piston pump.

5.5 ISO Symbols for hydraulic components.

5.6 Actuators.

5.7 Hydraulic circuits.

5.7.1 = Direct control of single acting cylinder.

5.7.2 = operation of double acting cylinder.

5.7.3 = operation of double acting cylinder with metering in and metering out control.

5.8 ~~5.8~~ Comparison of hydraulic and pneumatic system.

:- Swagun

Hydraulic turbines

* Definition and classification of Hydraulic turbines:

→ Hydraulic (water) turbine are the machine which convert hydropower in to mechanical energy.

→ Hydraulic turbine may be considered as hydraulic motors or primemotors.

→ Mechanical energy developed by a hydraulic turbine is used in running an electrical generator which is directly coupled to the shaft of turbine. Thus power developed by electric generator is called as Hydro-electric power.

Classification of hydraulic turbine :-

→ Classification of hydraulic turbine are made on the basis of

① Energy conversion inside turbine runner :-

- (i) Impulse turbine
- (ii) Reaction turbine

② Direction of flow of water inside runner :-

- (i) Tangential flow
- (ii) Radial flow
- (iii) Axial flow
- (iv) mixed flow.

③ Head and quantity of water available :-

- (i) High head
- (ii) medium head
- (iii) low head.

④ Specific speed :-

- (i) 10-35
- (ii) 60-400
- (iii) 300-1000

* Classification on the basis of the Energy conversion.

Impulse Turbine

→ Total available energy of water in term of head.

$$\frac{P}{\rho g} + \frac{v^2}{2g} + Z = \text{Const}$$

Flow energy head K. Head P. head

In a Impulse turbine all the available energy of water is converted into kinetic head, by passing water through a contracting nozzle.

→ Water coming out of nozzle from a free jet which impinge on series of buckets of runner this rotate the runner.

→ this class of turbine work under atmospheric pressure.

Ex pelton wheel, Girard Turbine
Banki turbine, Jonval Turbine.

Reaction turbine

→ out of total available energy

$$\frac{P}{\rho g} + \frac{v^2}{2g} + Z = C$$

at inlet of runner only a part of total energy is converted into kinetic energy. and substantial part remains in form of pressure head.

→ As water ^{flow} through runner gradually pressure head is converted into kinetic head.

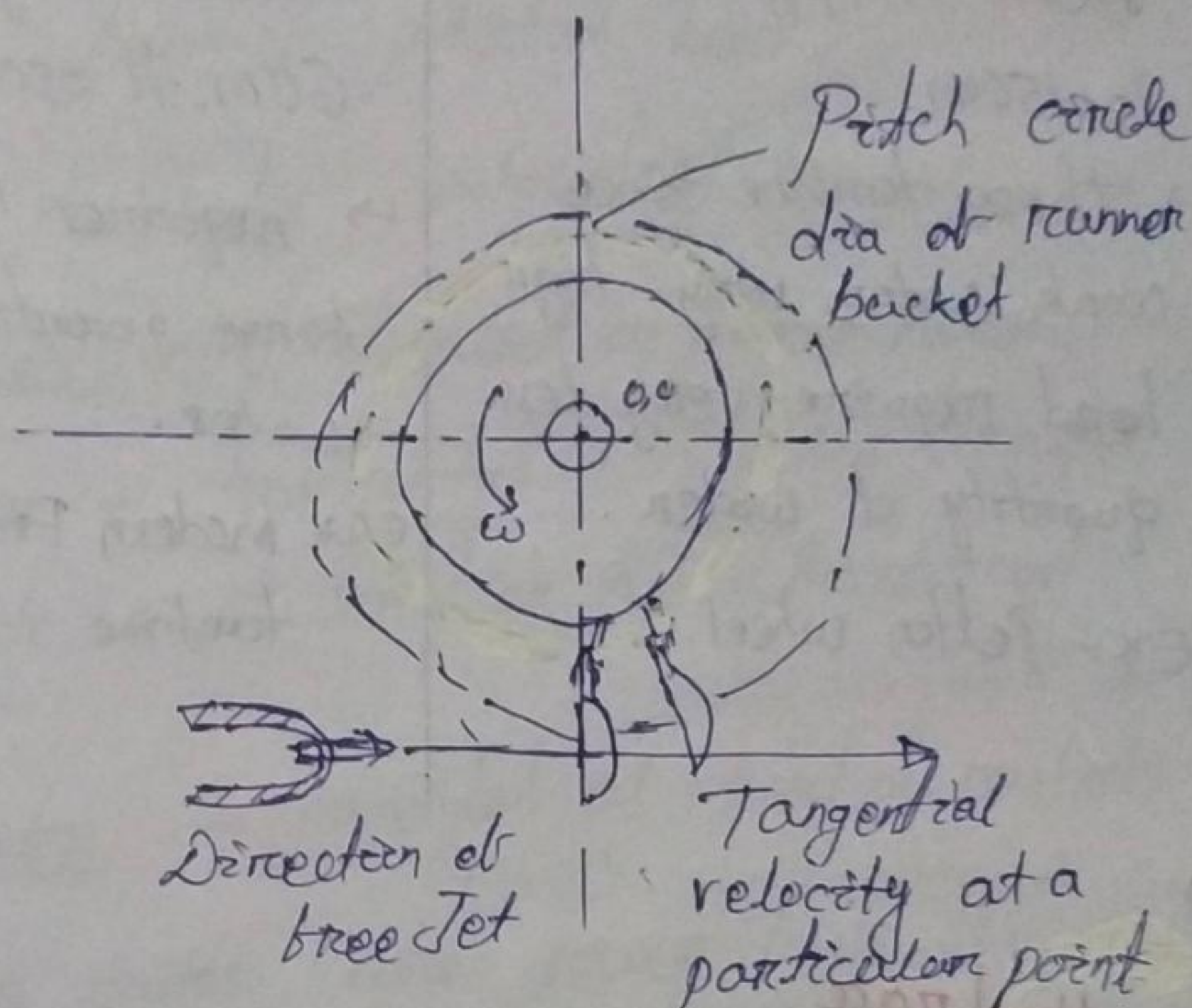
→ The difference of pressure between inlet and outlet runner is called as Reaction pressure and hence the turbine are known as Reaction turbine.

Ex Francis, Kaplan, ~~parallel~~ propeller, Founeyron

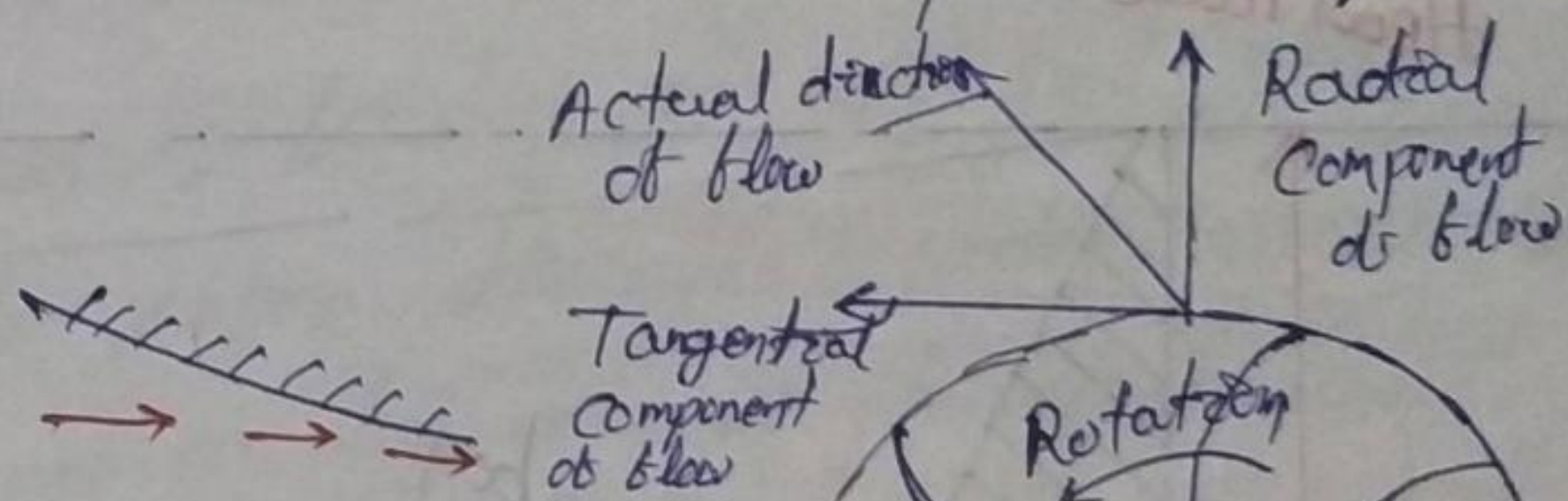
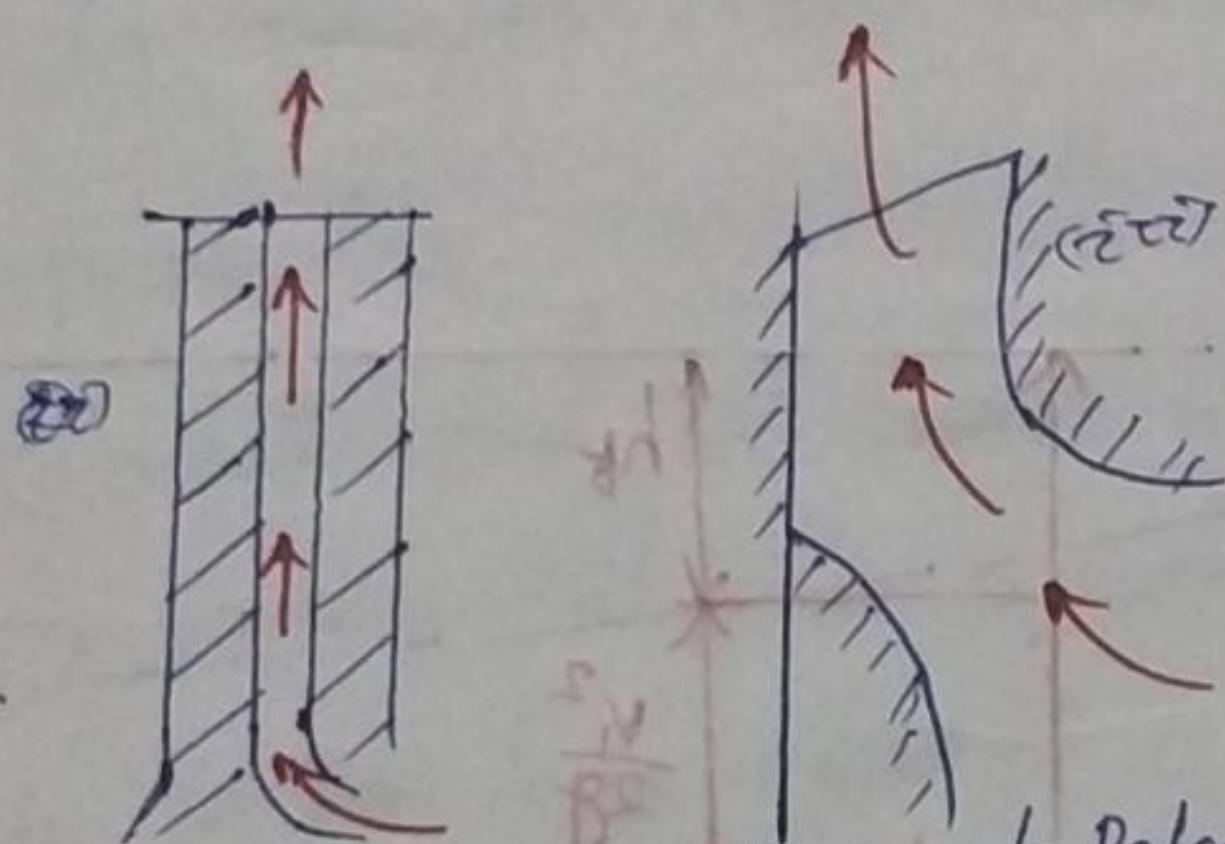
* Direction of flow of water:

(i) Tangential flow:

→ water flows along tangential to path of Reaction of Runner.



Ex: pelton wheel.



(ii) Radial flow:

→ water flow along radial direction and remain wholly and mostly perpendicular to the axis of rotation of Runner.

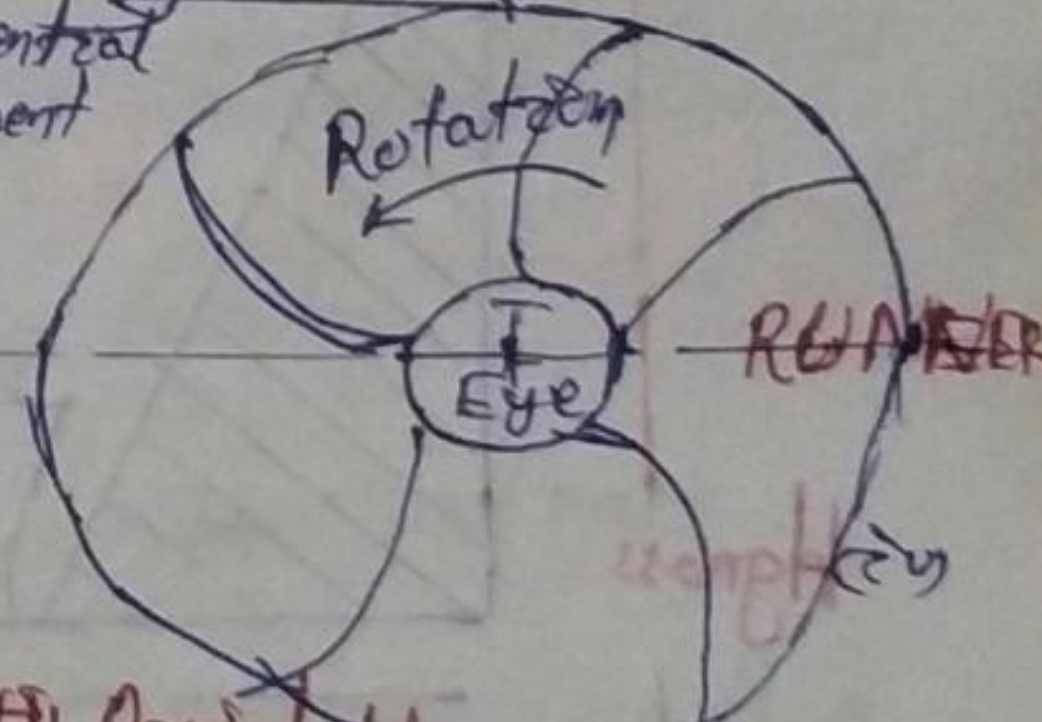
→ outward radial flow water enters at ^{eye} on centre of Runner and flow radially outwards.

Ex: Francis (old)

(iii) Mixed flow

→ In mixed flow Turbine water flow direction is in between radial and tangential direction.

Ex: Francis (modern)



(iv) Axial flow

→ In Axial flow turbine flow of water through runner is mainly along axis of rotation of turbine.

Ex: Jonval, propeller, Kaplan.

* Classification on basis of head and quantity of water available

High head

→ Head ranging from 500m. to 1500m.

→ These turbine since work under very high head require very less quantity of water.

Ex: Pelton wheel.

Medium head

→ Head ranging from 60m. to 250m.

→ Requires relatively large quantity of water.

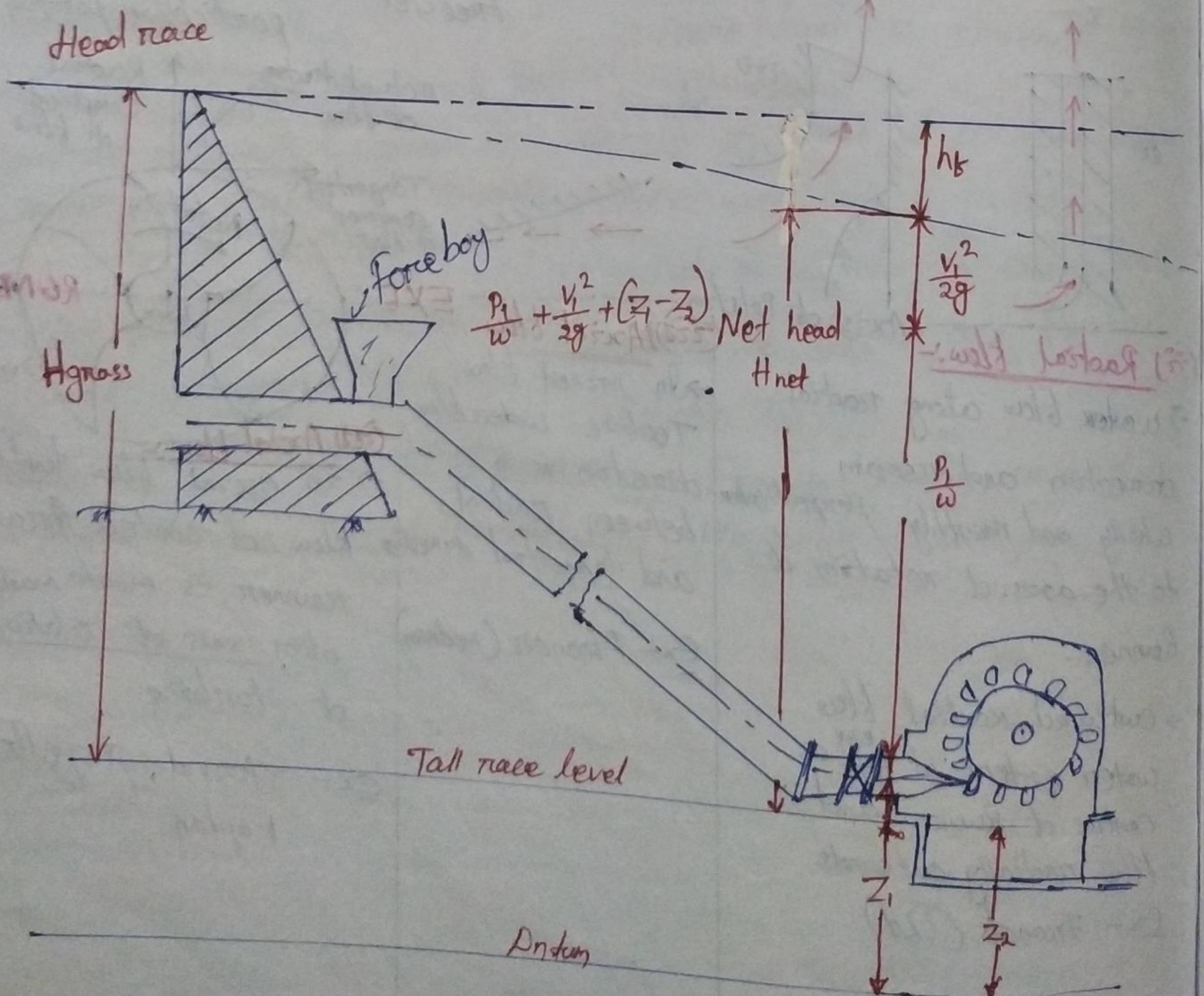
Ex: Modern Francis turbine

Low head

→ Capable of working under head less than 60m.

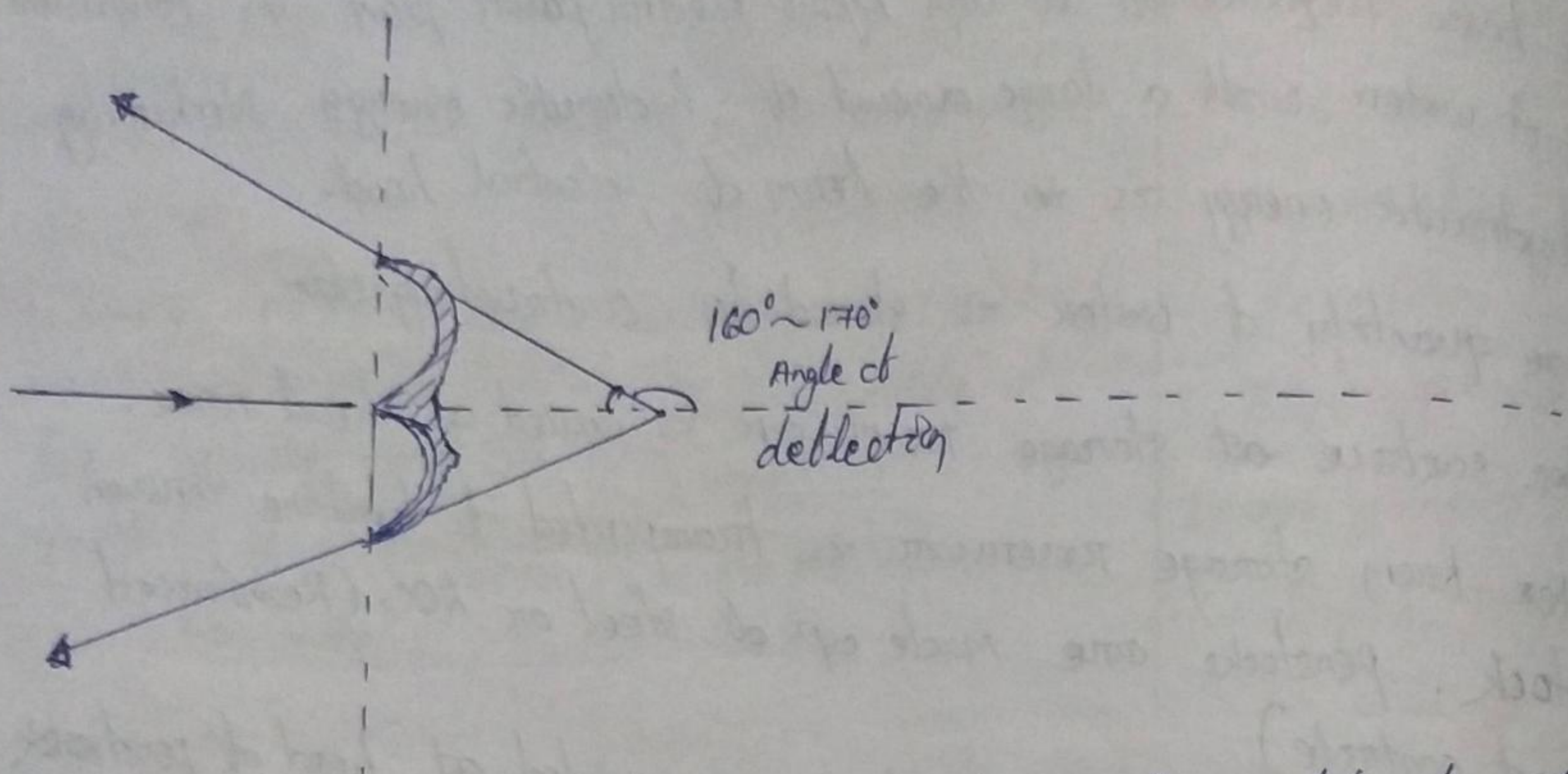
→ Requires very large quantity of water.
Ex: Kaplan, propeller.

*



* Construction and working principle of pelton turbine:-

- The basic requirement of any hydro-electric power plant is constant source of water with a large amount of hydraulic energy. Naturally the hydraulic energy is in the form of potential head.
- Large quantity of water is stored by constructing dam.
- Water surface at storage reservoir is called as head race.
- Water from storage reservoir is transported to turbine through penstock. penstocks are made up of steel or RCC. (Reinforced cement concrete)
- In smaller reservoirs borebay is provided at head of penstock.
- Bore bay temporarily stores water. when power house is located just at base of dam no borebay is required, since reservoir itself serves the purpose. However, water is carried to power house through a canal or long ~~conduit~~ ~~to power house~~ penstock it is always advisable to provide a borebay.
- Water passing through turbine is discharged to tail race.
- Runner of Pelton turbine consists of a circular disk with a number of bucket evenly spaced around periphery.
- Bucket are in shape of semi-ellipsoidal cups divided into two symmetrical parts by a splitter.
- The name of splitter is justified as it splits jet of water jet deflected by an angle approximately 160° to 170° .

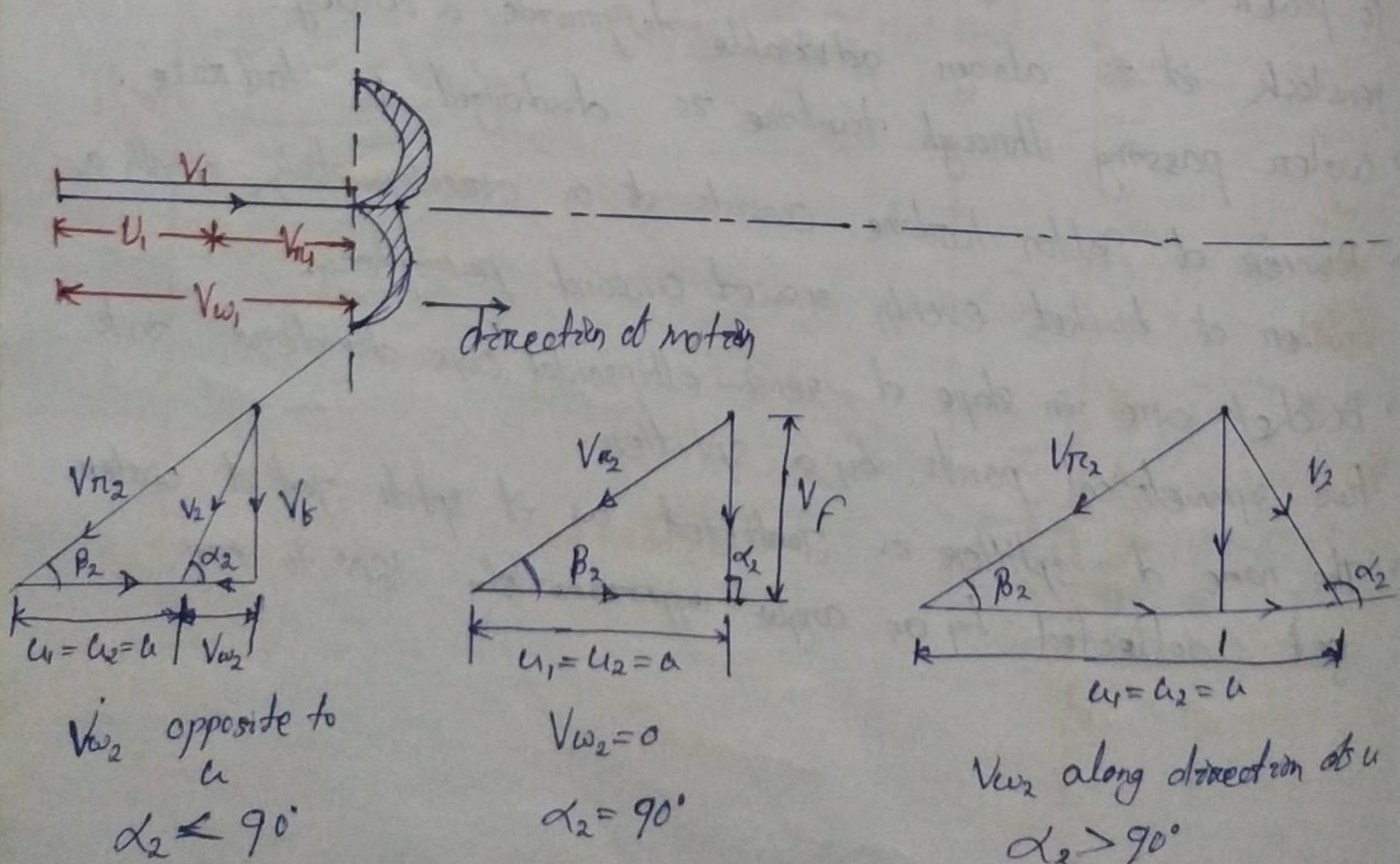


→ Nozzles are provided with a spear to control quantity of water

impinging on bucket.

⇒ when shaft of pelton wheel is horizontal then maximum two no. of jets are used, whereas for a vertical shaft pelton wheel no. of jets maximum 6.

* Velocity triangle +



V_1 = absolute velocity of incoming jet.

V_{r1} = Relative velocity at component at inlet.

U_1 = peripheral velocity. or tangential velocity.

V_2 = Absolute velocity at outlet jet.

V_{r2} = Relative velocity at exit.

U_2 = peripheral velocity of exit.

V_{wt} = vane component or tangential component. or whirl velocity.

α_1 = Guide blade angle or,

Angle between direction of jet and direction of motion of vane.

vane.

α_2 = Angle made by absolute velocity of jet with direction of motion of vane.

β_1 = vane angle or Blade angle at inlet or Angle made by relative velocity with direction of motion at inlet.

β_2 = vane angle at outlet.

or Angle made by relative velocity with direction of motion at outlet.

* $\rightarrow$ In a pelton turbine total energy of water is converted into K.E at entrance to turbine runner this velocity (absolute) with which jet approaches runner can be found out from Bernoulli's equation.

$$\frac{P}{\rho g} + \frac{V^2}{2g} + Z = \text{Constant}$$

or

$$\frac{P}{\rho g} + \frac{V^2}{2g} + Z = H_{\text{net}}$$

Flow energy $\rightarrow$ $\frac{P}{\rho g}$ $\leftarrow$ K.E $\leftarrow$ Potential head

$$\Rightarrow \frac{p}{\rho g} + \frac{v^2}{2g} + \frac{z}{g} = H_{net}$$

$$\Rightarrow \frac{v^2}{2g} = H_{net}$$

$$\Rightarrow v^2 = 2g \cdot H_{net}$$

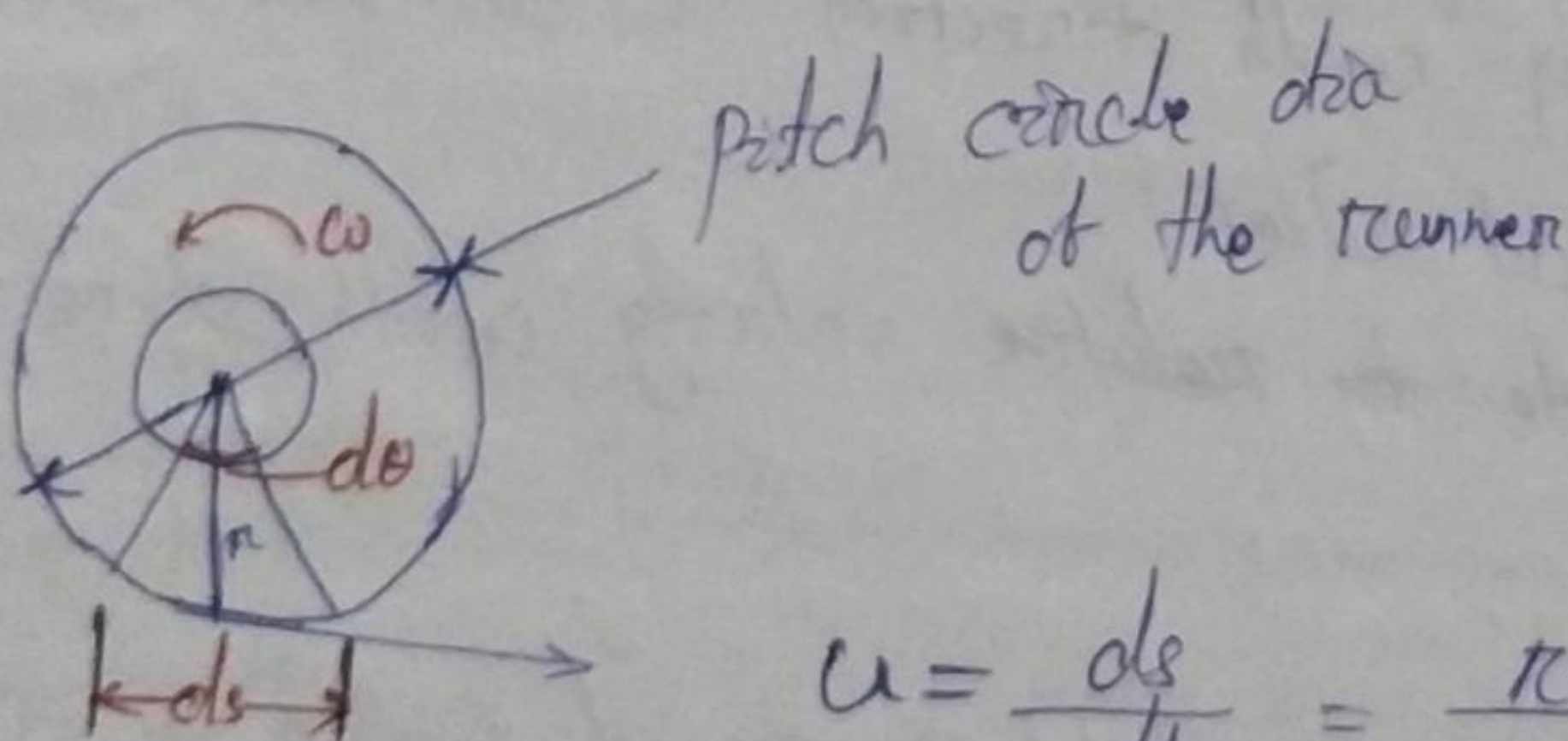
$$\Rightarrow v = \sqrt{2g \cdot H_{net}}$$

(Theoretical absolute velocity to get at inlet)

$$\Rightarrow \boxed{v_1 = C_v \sqrt{2g \cdot H_{net}}}$$

(C_v take care at friction in nozzle. 0.96 to 0.98)

→ Velocity of runner blade at any of tip is tangent to radius at the tip.



$$u = \frac{ds}{dt} = \frac{\pi d \omega}{dt}$$

$$u = \lim_{\Delta t \rightarrow 0} \frac{\pi d \omega}{dt}$$

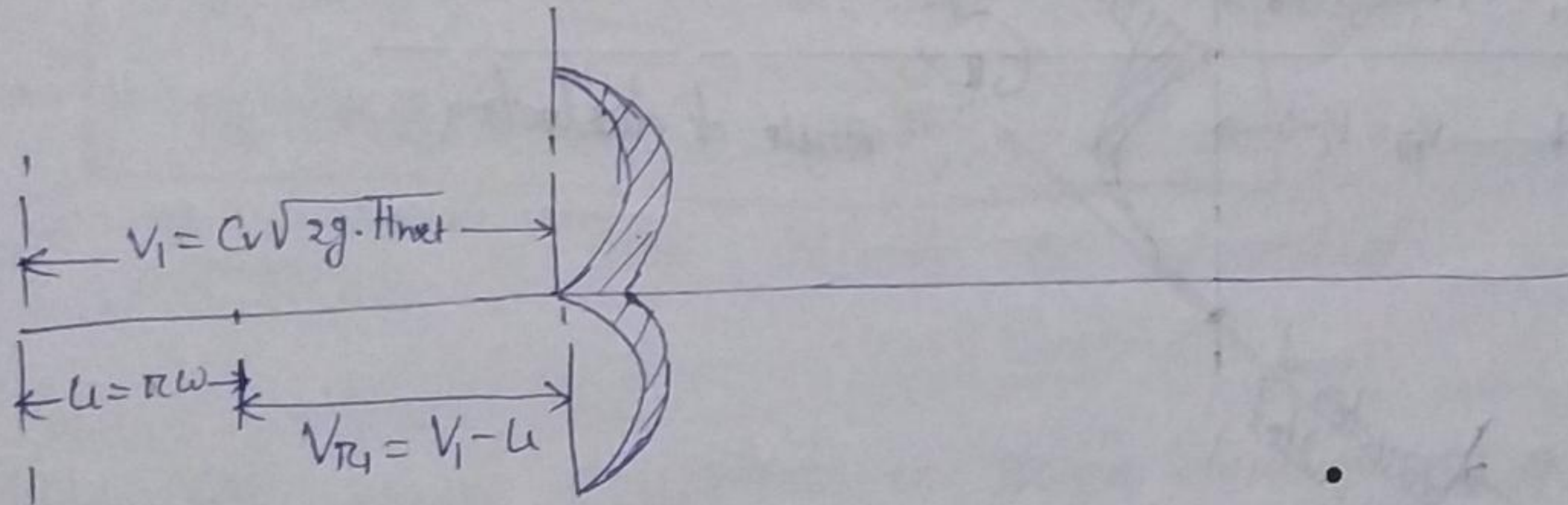
$$\boxed{u = \pi \omega r}$$

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

For a pelton wheel turbine $u_1 = u_2 = u$

* Now we are in very much self sufficient position to construct inlet velocity triangle of pelton turbine.

→ From a relative reference frame it we analyze



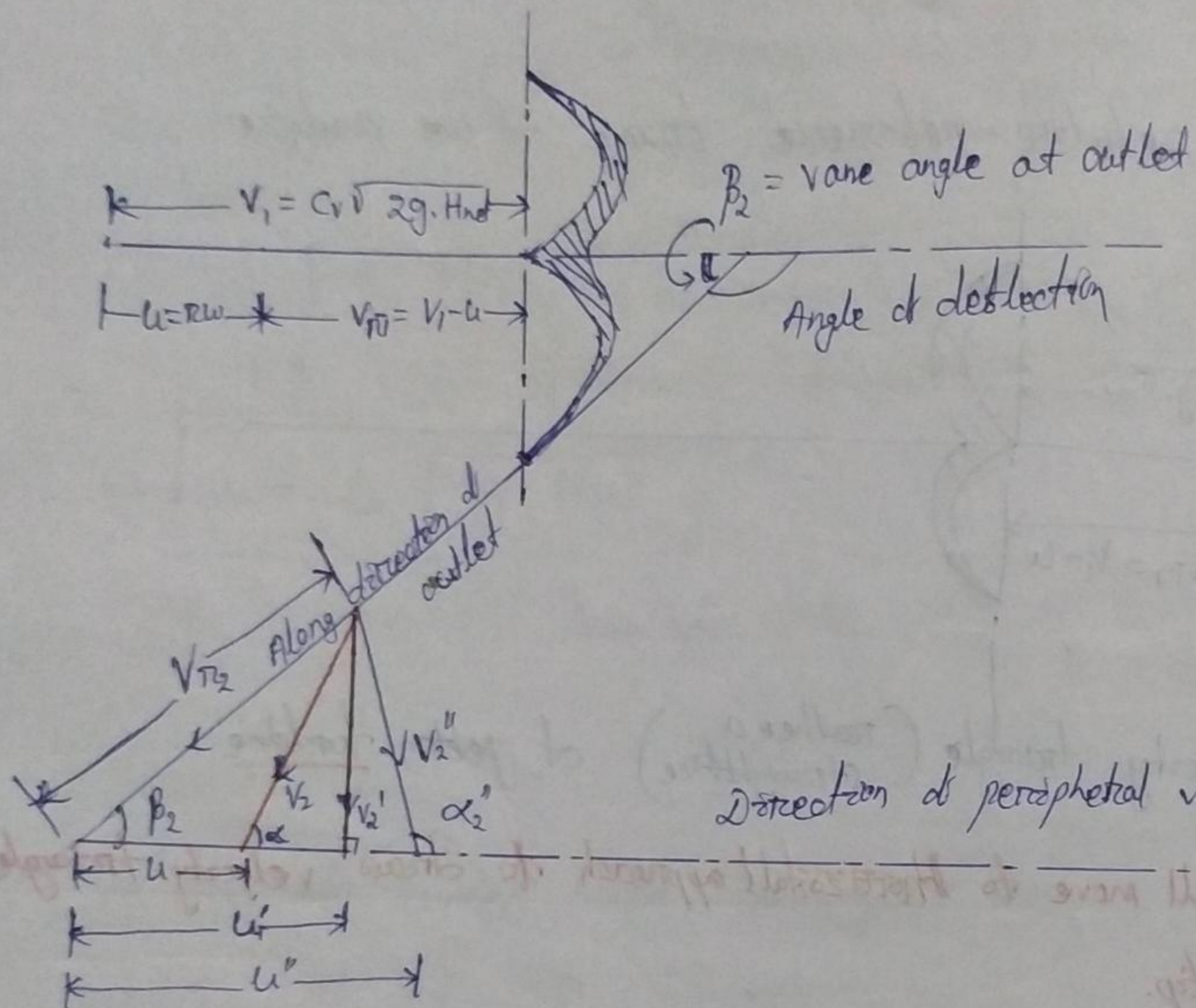
Inlet velocity triangle (rather a straightline) of pelton turbine.

No we will move to Michonchil approach to draw velocity triangle at outlet tip.

Relative velocity at outlet is as usually expressed as

$$\boxed{V_{r2} = k \cdot V_{r1}} \quad (k = \text{compensation factor of friction})$$

→ At head off (arrow) V_{r2} draw peripheral velocity 'u' in taken in order according to the Triangle law of vector addition the closing side triangle will represent. V_2 (Absolute velocity of jet at outlet tip) taken in reverse order.



- So you can see 3 varieties of velocity triangle may be created at outlet tip.
- And the three varieties of peripheral velocities ' u '.
- It may be a slow moving or medium moving or high speed moving runner, shape of outlet velocity Δ will change.
- one thing you mark β_2 is fixed, depending on ' u ' V_2 and α_2 (Angle made by absolute velocity at outlet) are changing.
- Now you might be thinking which velocity Δ you have to draw?

→ No, problem assume any Δ velocity Δ at outlet and draw then applying principle of trigonometry calculate $V_{r2} \cos \beta_2$

$$\begin{aligned} \text{If } V_{r2} \cos \beta_2 < u & : \alpha_2 < 90^\circ \\ V_{r2} \cos \beta_2 = u & : \alpha_2 = 90^\circ \\ V_{r2} \cos \beta_2 > u & : \alpha_2 > 90^\circ \end{aligned}$$

and accordingly redraw velocity Δ at outlet.

→ Now we should concentrate on power developed by a pelton turbine. As it is a rotodynamic m/c as usually power is being expressed in the form of

$$P = T \times \omega$$

∴ Where T = Twisting moment
 ω = Angular velocity

$$P = [F \times r] \times \omega$$

force perpendicular distance

$$P = [m_{\text{water}} \times \Delta V_w] \times r \times \omega$$

(mass flow rate) (rate of change of velocity along direction of motion) (radius) (angular velocity)

($\frac{\text{kg}}{\text{sec}}$) ($\frac{\text{m}}{\text{s}}$) (m) ($\frac{\text{rad}}{\text{s}}$)

$$P = f_{H_2O} \left(\frac{\text{kg}}{\text{m}^3} \right) \times Q \left(\frac{\text{m}^3}{\text{s}} \right) \times \Delta V_w \left(\frac{\text{m}}{\text{s}} \right) \times r \times \frac{\omega}{r}$$

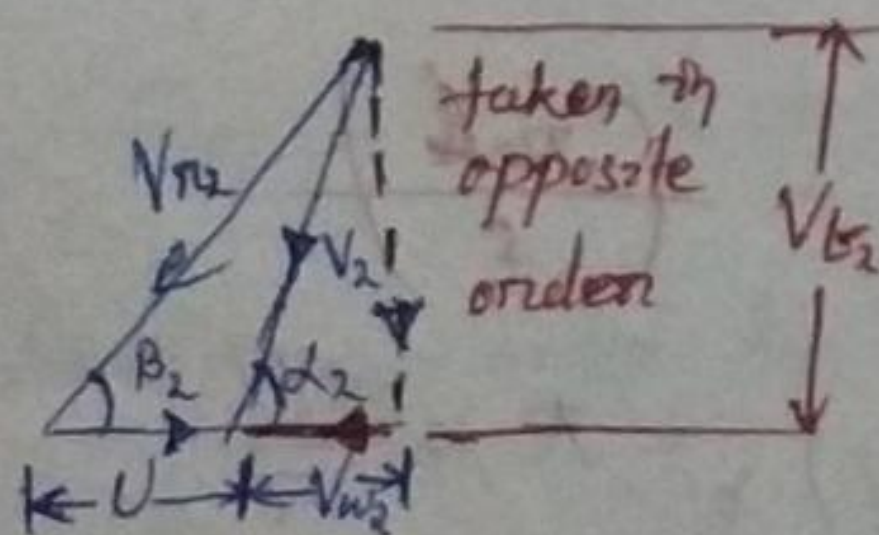
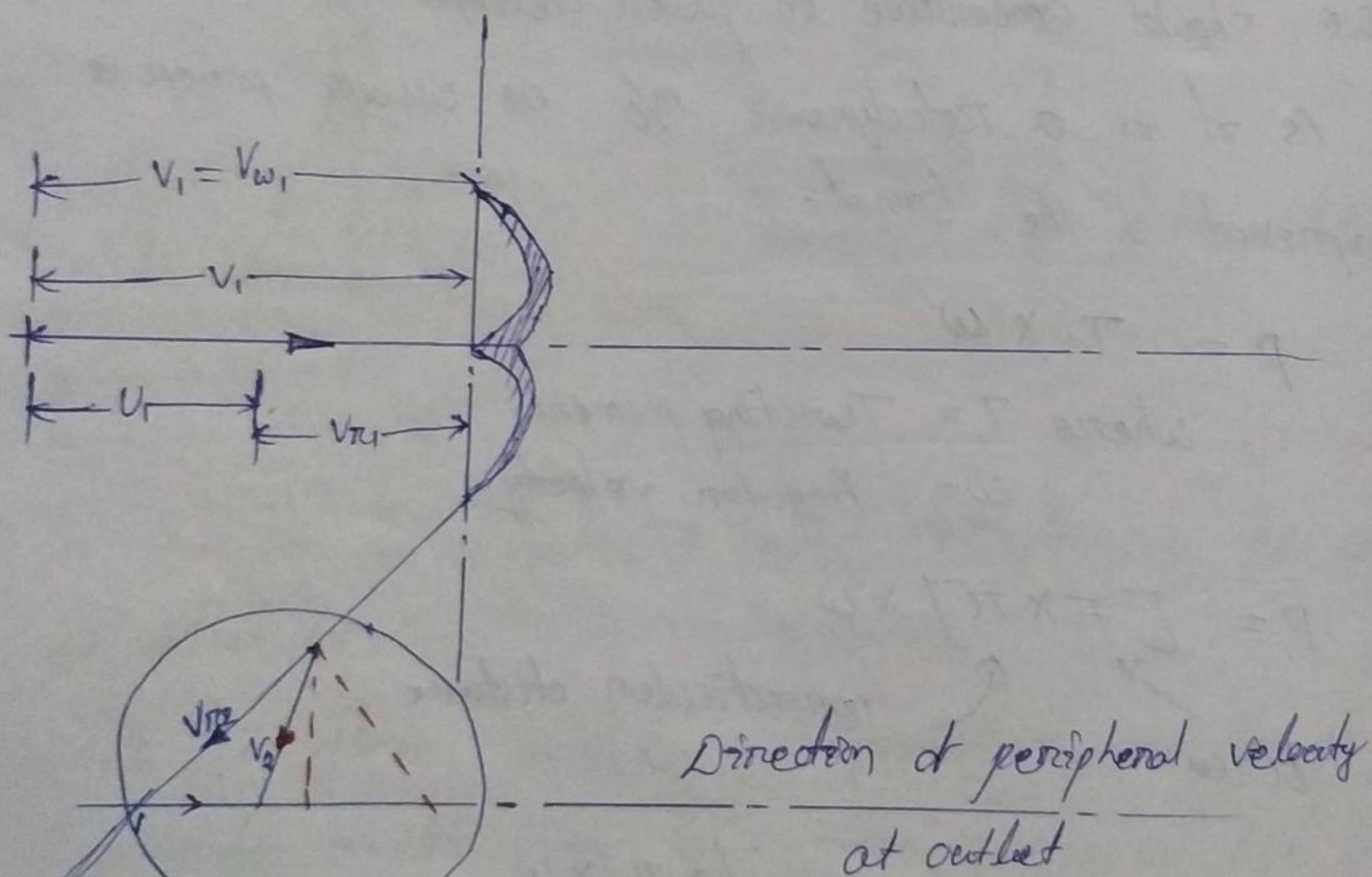
$$P = f \cdot Q \cdot (\Delta V_w) \cdot U$$

Now what is ΔV_w ?

$\rightarrow \Delta V_w$ is change in tangential component of absolute velocity along direction of motion of rotor.

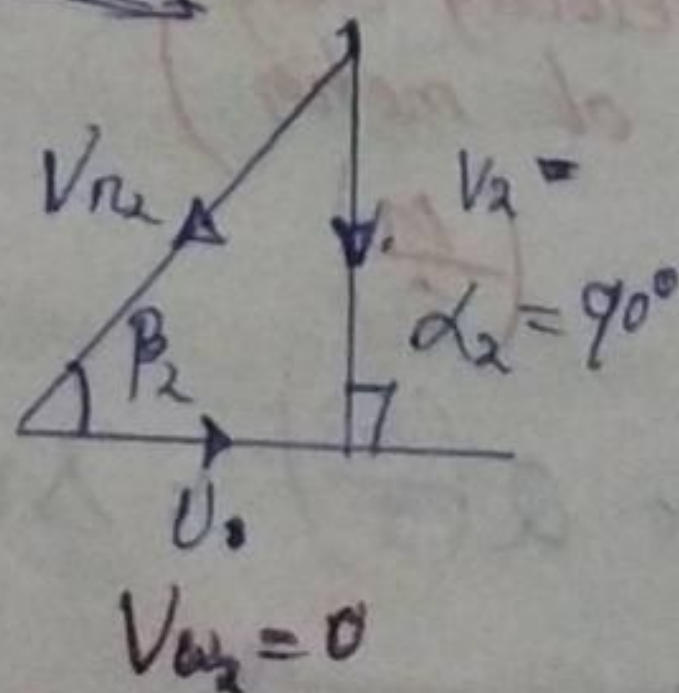
$$\text{Change} = (\text{Initial value}) - (\text{Final value})$$

$\rightarrow$ As turbine is the work producing machine and final value is less than initial value. So to get a (+ve) answer change is calculated over initial & final.



$$P = f \cdot Q [V_{w1} - (-V_{w2})]$$

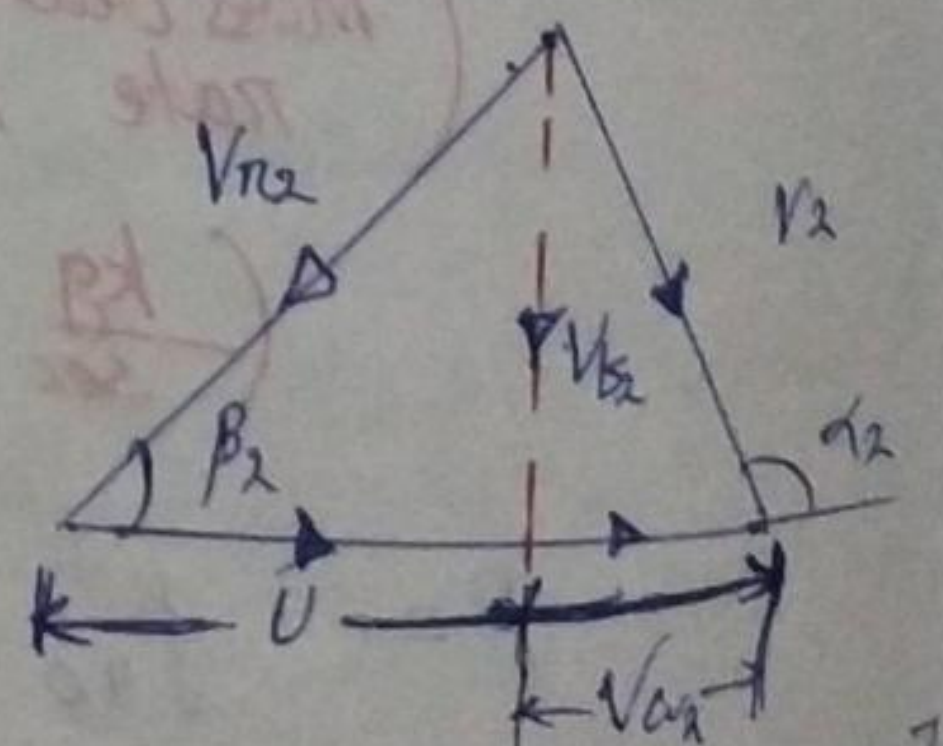
$$P = f \cdot Q [V_{w1} + V_{w2}]$$



$$V_{w2} = 0$$

$$P = f \cdot Q [V_{w1} - 0]$$

$$P = f \cdot Q \cdot V_{w1}$$



$$P = f \cdot Q [V_{w1} - V_{w2}]$$

In general

$$P = f \cdot Q [V_{w1} \pm V_{w2}] \times u$$

From intuition we have developed power of a pelton turbine is governed by 4 factors

- ① u
- ② v_1
- ③ β_2
- ④ k

→ Now we will visualize mathematically how u , v_1 , β_2 and bucket ratio k affected hydraulic efficiency η_{hyd} of pelton turbine.

$$\eta_{hyd} = \frac{\text{Power developed by runner}}{\text{power related to K.E of jet}}$$

$$= \frac{f \cdot Q \cdot [V_{w1} \pm V_{w2}] \times u}{\frac{1}{2} m v_1^2}$$

$$= \frac{f \cdot Q [V_{w1} \pm V_{w2}] \times u}{\frac{f \cdot Q \cdot v_1^2}{2}} \quad (\because V_1 = V_{w1})$$

$$= \frac{f \cdot Q [(V_{r1} + u) \pm (V_{r2} \cos \beta_2 - u)] \times u}{\frac{f \cdot Q \cdot v_1^2}{2}}$$

$$= \frac{f \cdot Q [V_{r1} + V_{r2} \cos \beta_2] \times u}{\frac{f \cdot Q \cdot v_1^2}{2}}$$

$$= \frac{f \cdot Q \cdot V_{r1} \left[1 + \frac{V_{r2}}{V_{r1}} \cos \beta_2 \right] \times u}{\frac{f \cdot Q \cdot v_1^2}{2}} \quad \left(\because V_{r2} = k \cdot V_{r1} \right)$$
$$k = \frac{V_{r2}}{V_{r1}}$$

$$= \frac{f \cdot Q \cdot V_{r1} [1 + k \cos \beta_2] \times u}{\frac{f \cdot Q \cdot v_1^2}{2}}$$

$$= \frac{f_R (v_1 - U) [1 + k \cos \beta_2] U}{\frac{f_R v_1^2}{2}}$$

$$\Rightarrow \frac{2 (v_1 - U) [1 + k \cos \beta_2] \times U}{v_1^2}$$

$$= \frac{2 [1 + k \cos \beta_2] \times (v_1 - U) \times U}{v_1^2}$$

$$= \frac{2 [1 + k \cos \beta_2] \times v_1 \left[1 - \frac{U}{v_1}\right] \times U}{v_1^2}$$

dividing v_1^2 on numerator and denominator

$$= \frac{2 [1 + k \cos \beta_2] \times \cancel{v_1} \left[1 + \frac{U}{\cancel{v_1}}\right] \times \frac{U}{\cancel{v_1}}}{\frac{\cancel{v_1}^2}{\cancel{v_1}^2}}$$

$$\eta_{\text{Hyd}} = \left[2 [1 + k \cos \beta_2] \times \left[1 - \frac{U}{v_1}\right] \times \frac{U}{v_1} \right]$$

No we can see mathematically

$$\eta_h = f \left(k; \beta_2, \frac{U}{v_1} \right)$$

→ Now as a general consequence k and β_2 are fixed or not compatible to alter, but we can very much effectively change the ratio $\frac{U}{v_1}$

→ Our next task should aim at optimizing $\frac{U}{v_1}$ to get maximum $\eta_{\text{hydraulic}}$

$$\frac{d \eta_h}{d \left(\frac{U}{v_1} \right)} = 0$$

$$\Rightarrow \frac{d \left[2 \left[1 + k \cos \beta_2 \right] \left(1 - \frac{U}{v_1} \right) \times \frac{U}{v_1} \right]}{d \left(\frac{U}{v_1} \right)} = 0$$

$$\Rightarrow 2 \left[1 + k \cos \beta_2 \right] \frac{d \left(\frac{U}{v_1} - \frac{U^2}{v_1^2} \right)}{d \left(\frac{U}{v_1} \right)} = 0$$

$$\Rightarrow 1 - 2 \left(\frac{U}{v_1} \right) = 0$$

$$\Rightarrow +2 \left(\frac{U}{v_1} \right) = +1$$

$$\Rightarrow \boxed{\frac{U}{v_1} = \frac{1}{2}}$$

$$\Rightarrow v_1 = 2U$$

* Thus to attain maximum hydraulic efficiency η_{hyd} .

$$\left[\text{Absolute velocity of incoming jet} \right] = \left[2 \text{ times peripheral velocity at Runner} \right]$$

Classification +

$$\eta_{hyd} = 2 \left[1 + k \cos \beta_2 \right] \left(1 - \frac{U}{v_1} \right) \frac{U}{v_1}$$

$$\text{as } \boxed{\text{If } \frac{U}{v_1} = 0}$$

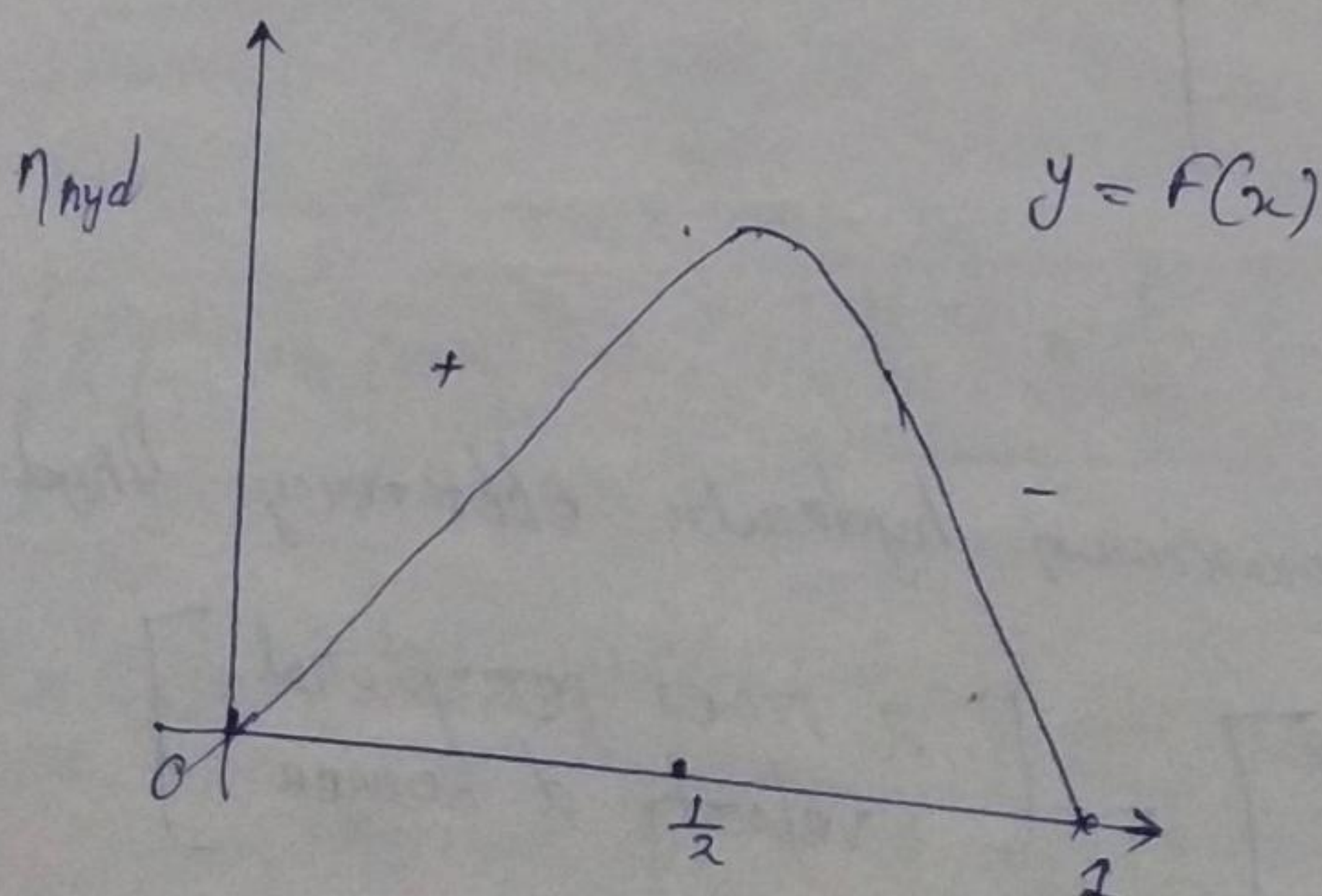
$$\text{obviously } \eta_{hyd} = 0$$

$$* \left[\frac{U}{V_1} = \frac{1}{2} \right]$$

$$\begin{aligned} &\Rightarrow 2(1+k \cos \beta_2) \left(1 - \frac{1}{2}\right) \times \frac{1}{2} \\ &= 2(1+k \cos \beta_2) \left(\frac{1}{2} - \frac{1}{4}\right) \\ &\Rightarrow 2[1+k \cos \beta_2] \left(\frac{2-1}{4}\right) \\ &= 2[1+k \cos \beta_2] \frac{1}{4} \\ &= \frac{1+k \cos \beta_2}{2} \end{aligned}$$

$$* \left[\frac{U}{V_1} = 1 \right]$$

$$\begin{aligned} &\Rightarrow 2[1+k \cos \beta_2] \left(1 - 1\right) \times 1 \\ &= 0 \end{aligned}$$



Variation of η_{hyd} w.r.t speed ratio $\frac{U}{V_1}$

Working Proportion for Pelton turbine

Horizontal shaft pelton turbine: max 2 no. jet.

Vertical shaft pelton turbine: max 6 no. jet.

$$\text{No. of jets} = \frac{Q \text{ across turbine}}{q \text{ across a jet}}$$

d : dia of a free jet.

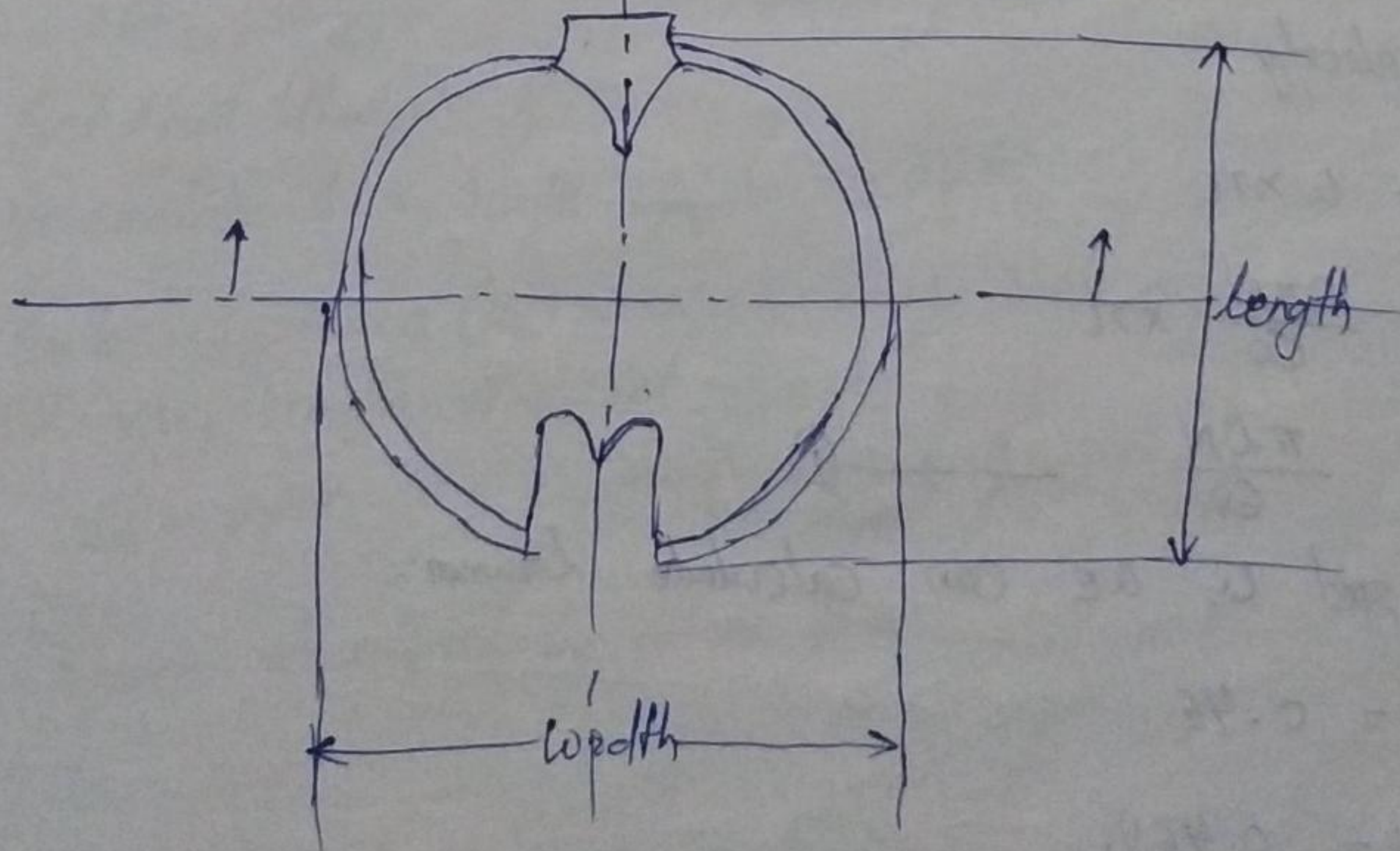
width of bucket $= 5d$

Depth of bucket $= 1.2d$

Length of bucket =

$$\text{Jet Ratio } m' = \frac{\text{Runner pitch circle dia}}{\text{diameter of jet}} = \frac{D}{d}$$

$$\begin{aligned} \text{No of Bucket } Z &= 15 + 0.5m' = \boxed{15.5m'} = \cancel{15.5 \times \frac{D}{d}} \\ &= \boxed{15 + 0.5 \times \frac{D}{d}} \end{aligned}$$



Top view

① A pelton wheel with single jet rotates at 600 rpm and working under a head of 500m. It ratio of vane velocity to jet velocity is 0.46. Find diameter of pelton wheel. Assume coefficient of velocity as 0.98.

Data given:

Number of jet = 1

Speed of runner (N) = 600 rpm

Net head = 500m.

$$\frac{U}{V_1} = 0.46 \quad (\text{speed ratio})$$

$$C_v = 0.98$$

To Find:

Diameter of pelton wheel 'D' = _____

Analysis:

As we know diameter of Runner is always associated with peripheral velocity.

$$U = W \times \pi$$

$$U = \frac{2\pi N}{60} \times \pi$$

$$U = \frac{\pi D N}{60} \quad \text{--- ①}$$

If we can get U, we can calculate Diameter.

$$\frac{U}{V_1} = 0.46$$

$$\Rightarrow U = 0.46 V_1$$

$$\Rightarrow U = 0.46 \times C_v \sqrt{2g \cdot H_{\text{net}}}$$

$$\Rightarrow U = 0.46 \times 0.98 \sqrt{2 \times 9.81 \times 500\text{m}}$$

$$\Rightarrow U = \underline{\underline{44.64\text{m}}}$$

∴ putting the 'U' in eqn (1)

$$44.64 = \frac{\pi D \times 600}{60}$$

$$\Rightarrow 44.64 = 31.41 D$$

$$\Rightarrow D = \frac{44.64}{31.41} = \underline{\underline{1.42 \text{ m}}}$$

② Following data refer to pelton wheel.

Tangential velocity of wheel = $30 \frac{\text{m}}{\text{s}}$

Head = 80 m.

Discharge (volumetric flow rate) = $0.04 \frac{\text{m}^3}{\text{sec}}$

Side clearance angle = 10°

Co-efficient of velocity $C_v = 0.98$

Determine power developed by wheel Neglect Frictional losses in bucket.

Data given:

$$U = 30 \frac{\text{m}}{\text{s}}$$

Net head (H_{net}) = 80 m.

Volumetric flow rate $Q = 0.04 \frac{\text{m}^3}{\text{sec}}$

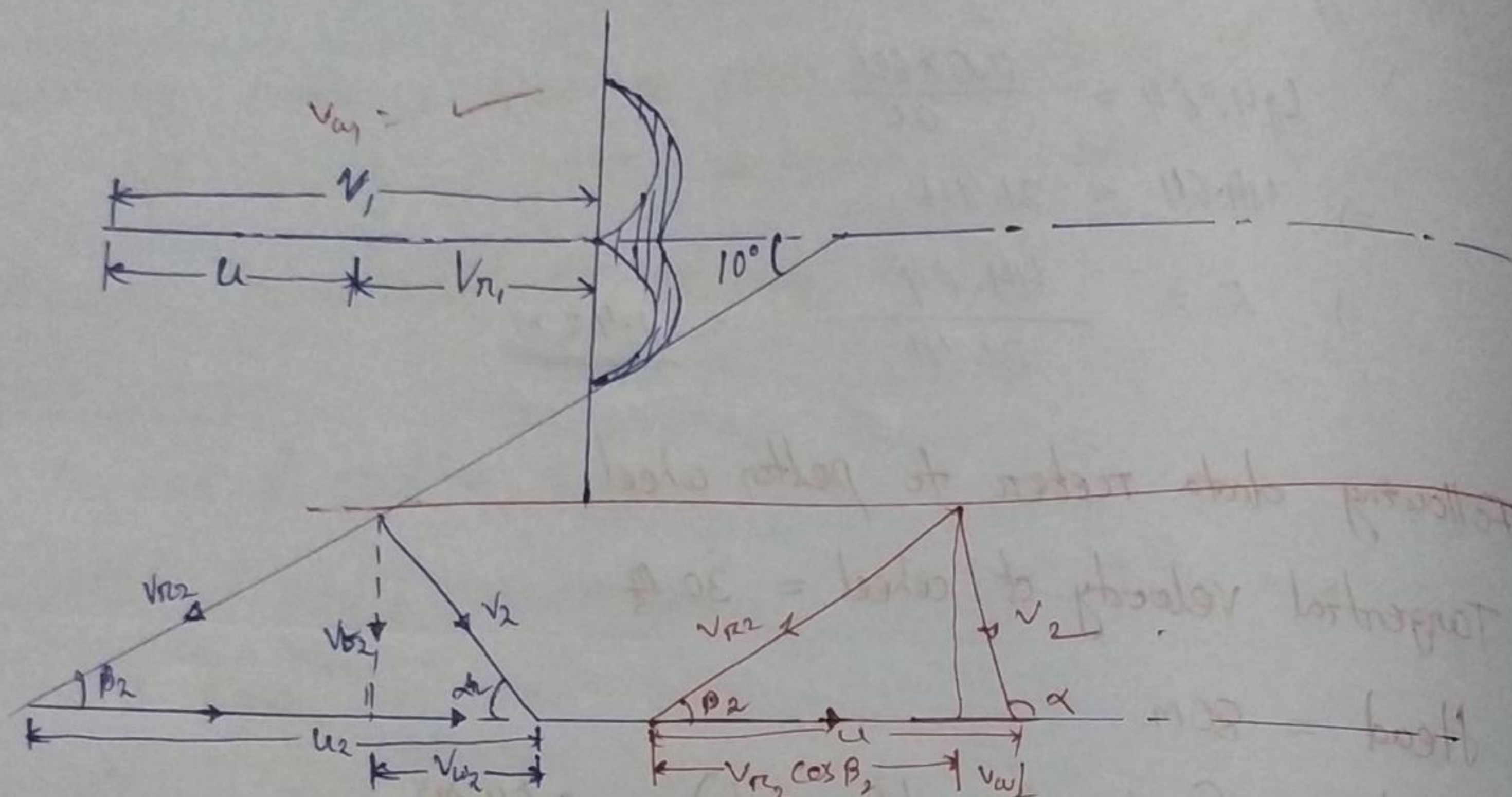
Side clearance (β_2) = ~~10°~~

or vane angle of outlet = 10°

$$C_v = 0.98$$

$$(\because K = 1)$$

To Find
power developed by wheel $P' = \underline{\hspace{2cm}}$



Here in this problem $V_{r1} = V_{r2}$

V_{r1} can be calculate from inlet velocity Δ .

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

$$V_{r1} = C_v \sqrt{2g h_{net}} - 30$$

$$V_{r1} = (0.98 \sqrt{2 \times 9.81 \times 80}) - 30$$

$$V_{r1} = 8.82 \text{ m/s} = V_{r2}$$

$$V_{r2} \cos \beta_2 = 8.82 \times \cos 10^\circ = \underline{8.68}$$

$$\boxed{V_{r2} \cos \beta_2 < u}$$

Now to construct velocity triangle at outlet we have to check

$$V_{r2} \cos \beta_2 \times u$$

$$V_{r2} \cos \beta_2 = u$$

$$V_{r2} \cos \beta_2 > u$$

Calculate $\checkmark$

$$V_{w1} = V_1 = C_v \sqrt{2g h_{net}} = \underline{38.82}$$

$$\boxed{V_{w2} = V_{r2} \cos \beta_2 - u_2 = -21.31 \times}$$

$$u_1 = u_2 = u$$

$$V_{w2} = 30 - V_{r2} \cos \beta_2 = 30 - 8.68 = 21.32$$

Now we can construct velocity Δ at outlet

As desired by question $P = T \times \omega$

$$P = \{F \times r\} \times \omega$$

$$\Rightarrow P = m \left(\frac{\text{kg}}{\text{sec}} \right) \times Q \Delta V_w \left(\frac{\text{m}}{\text{s}} \right) \times r (\text{m}) \times \frac{2\pi N}{60} \left(\frac{\text{rad}}{\text{sec}} \right)$$

$$\Rightarrow P = f \times Q [V_{w1} - V_{w2}] \times \frac{2\pi N}{60}$$

$$\Rightarrow P = f \times Q [V_{w1} - V_{w2}] \times u$$

$$\Rightarrow P = 1000 \frac{\text{kg}}{\text{m}^3} \times 0.04 \left(\frac{\text{m}^3}{\text{s}} \right) \times [38.82 - 21.31] \times 30$$

$$\Rightarrow \boxed{P = 72.15 \text{ kW}} \quad 21.012 \times 10^3 \text{ Watt} = 21 \text{ kW}$$

③ A pelton wheel having a wheel diameter of 1.2m. rotate at 600rpm. water is supplied at the rate of $0.6 \frac{\text{m}^3}{\text{sec}}$. under a head of 450m. If the bucket deflect the jet through an angle 160° . Find power develop and hydraulic efficiency (η_{hyd}) of turbine. Assume coefficient of velocity as 0.97. Neglect frictional losses in bucket.

Data given:

wheel diameter $D = 1.2 \text{ m}$.

wheel speed $N = 600 \text{ rpm}$

Volumetric flow rate $Q = 0.6 \frac{\text{m}^3}{\text{sec}}$

Net head = 450m.

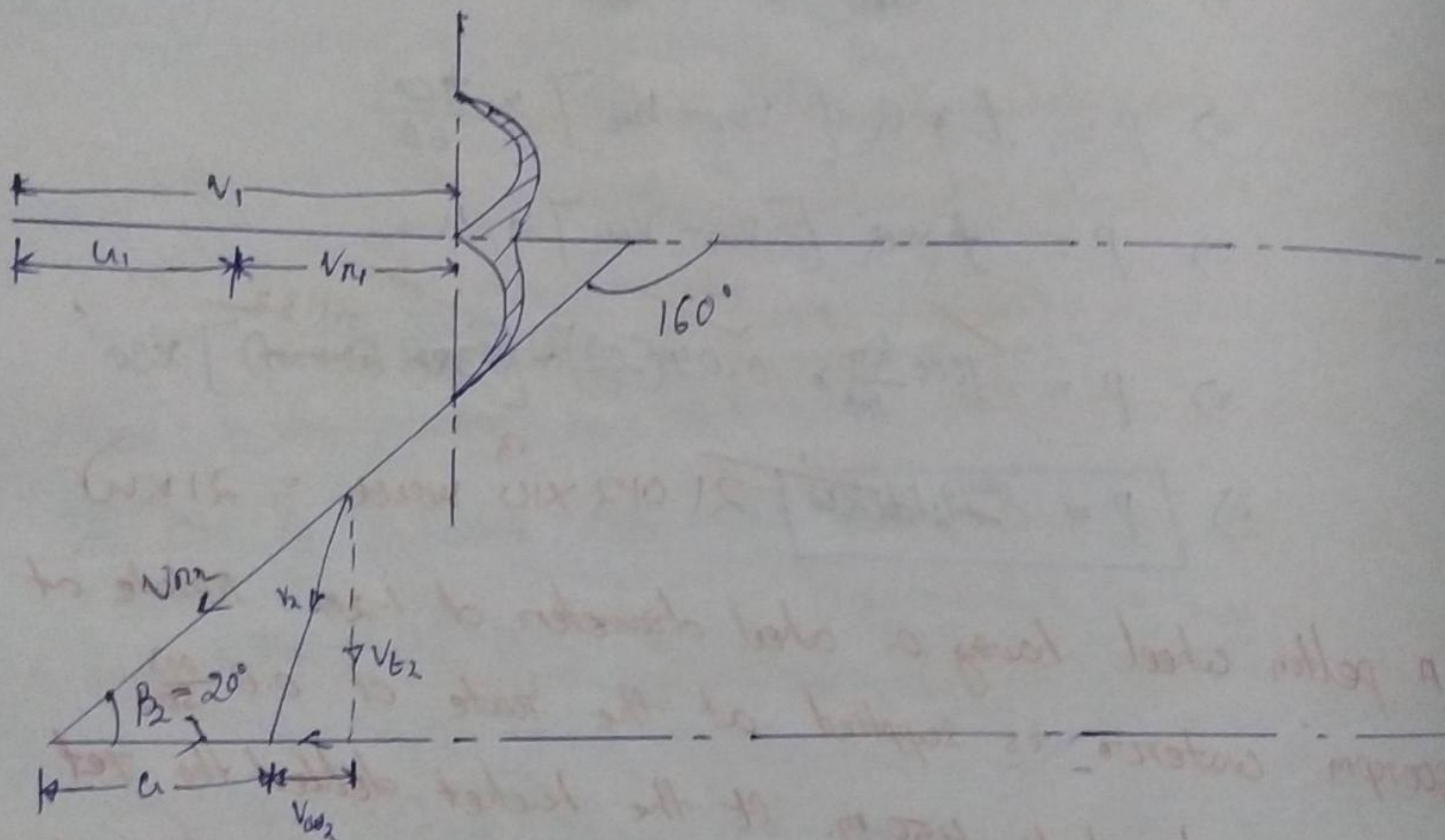
Angle of deflection of jet $\alpha = 160^\circ$

$C_v = 0.97$

To Find +

power develop by turbine $P = \text{---}$

hydraulic efficiency $\eta_{hyd} = \text{---}$



To construct velocity triangle inlet

$$u = \frac{\pi DN}{60} = \frac{\pi \times 1.2 \times 600}{60} = 37.699 \frac{m}{s}$$

Absolute velocity of jet at inlet

$$\begin{aligned} V_1 &= C_v \sqrt{2g h_{net}} \\ &= 0.97 \sqrt{2 \times 9.81 \times 450} \\ &= 91.14 \frac{m}{s} \end{aligned}$$

$$V_{n1} = V_1 - u = 91.14 - 37.699 = 53.441 \frac{m}{s}$$

As blade are frictionless $V_{n1} = V_{n2}$

$$\begin{aligned} V_{n2} \cos(\beta_2) &= \\ \Rightarrow 53.441 \cos(20) &= 50.21 \end{aligned}$$

To construct velocity triangle at outlet we must check

$$V_{r2} \cos \beta_2 < u$$

$$V_{r2} \cos \beta_2 = u$$

$$V_{r2} \cos \beta_2 > u$$

$$V_{r2} \cos \beta_2 = 53.441 \times (\cos(20)^\circ) = 50.218$$

$$V_{r2} \cos \beta_2 > u$$

Now complete velocity triangle at outlet

$$V_{w2} = V_{r2} \cos \beta_2 - u$$

$$= 53.441 \cos(20) - 37.699$$

$$V_{w2} = 12.519 \text{ (m/s)}$$

$$V_{w2} = V_1 = 91.14 \text{ m/s}$$

power developed by turbine

$$P = T \times \omega$$

$$= (F \times r) \times \omega$$

$$= (m \times \Delta V_w) \times r \times \omega$$

$$= f \cdot Q \cdot [V_{w1} + V_{w2}] \times r \times \frac{2\pi N}{60}$$

$$= f \cdot Q \cdot [V_{w1} + V_{w2}] \times \frac{\pi D N}{60}$$

$$= 1000 \times 0.6 [12.519 + 91.14] \times \frac{\pi \times 2 \times 600}{60}$$

$$= 2344704.38 \text{ kW}$$

$$\eta_{\text{hyd}} = \frac{\text{power developed by runner}}{\text{power related to K.E of jet}}$$

$$= \frac{f \cdot Q [V_{w1} + V_{w2}] u}{\frac{f \cdot Q V_1^2}{2}}$$

$$= \frac{2344704.38}{\frac{1000 \times 0.6 \times 91.14^2}{2}} = 0.94 = 94\%$$

Q. A pelton wheel operates under a head of 400m. with a speed ratio of 0.5 and flow ratio of 0.98. The bucket deflects the jet through an angle of 160° . Determine the power developed per unit weight of water flow.

Data given

Net head (H_{net}) = 400m.

with a speed ratio $\frac{U}{v_1} = 0.5 \Rightarrow U = 0.5 \times v_1$

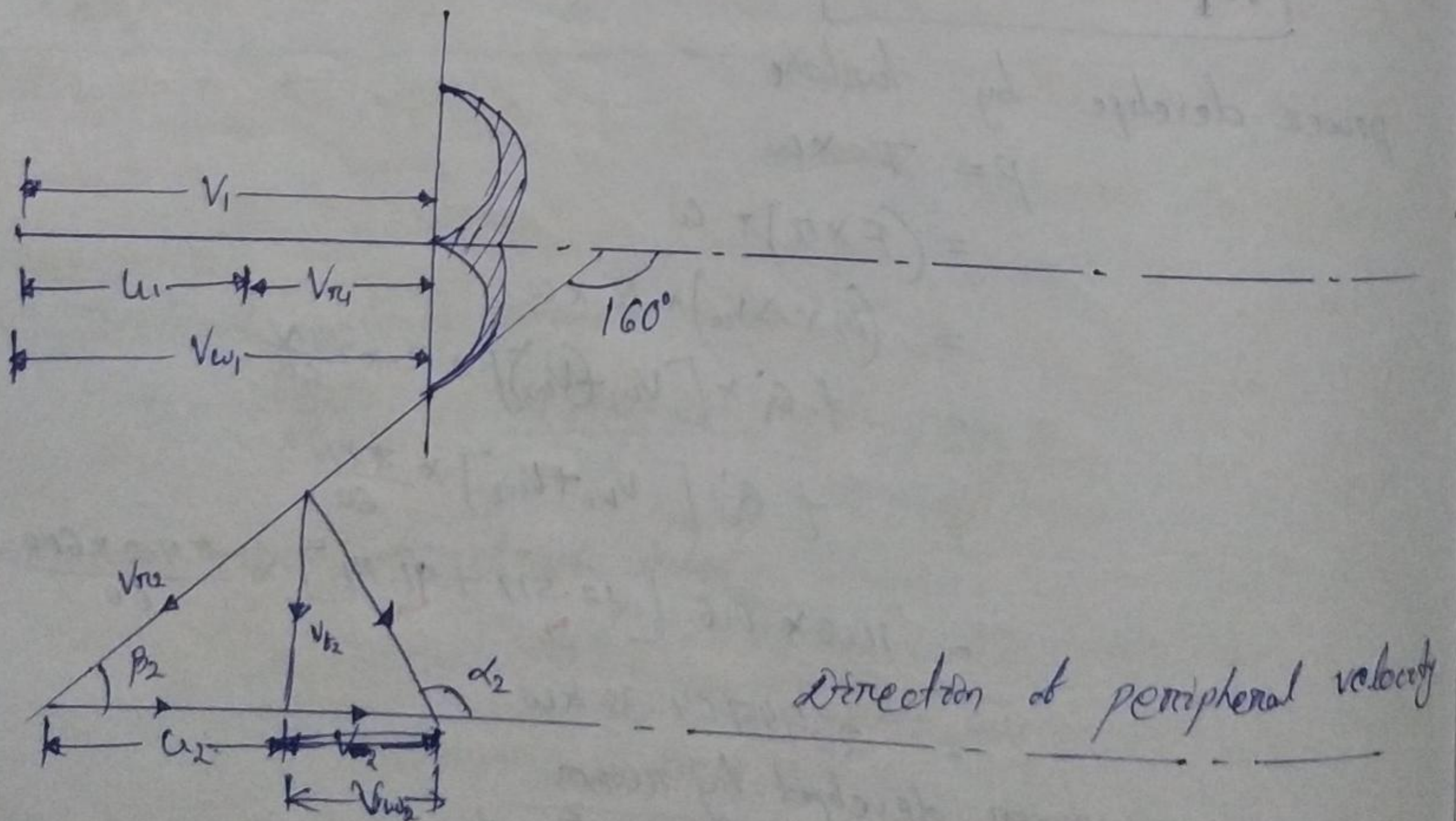
Co-efficient of ~~friction~~ velocity $C_v = 0.98$

Angle of deflection = 160°

$$\beta_2 = 180^\circ - 160^\circ = 20^\circ$$

To Find :

power developed $P =$



V_{r1} can be calculate from inlet velocity Δ .

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

$$V_1 = C_v \sqrt{2g \cdot h_{\text{net}}}$$

$$= 0.98 \sqrt{2 \times 9.81 \times 400} = 86.81$$

$$u_1 = 0.5 \times 86.81 = \underline{43.40 \frac{\text{m}}{\text{s}}} \quad (u_1 = u_2 = u)$$

$$V_{r1} = 86.81 - 43.40 = \underline{43.41 \frac{\text{m}}{\text{s}}}$$

$$\boxed{V_{r1} = V_{r2}}$$

$$V_{w1} = V_1 = \underline{43.41 \times 86.81 \frac{\text{m}}{\text{s}}}$$

$$V_{r2} \cos \beta_2 = 43.41 \cos 20^\circ = 40.79$$

$$V_{r2} \cos \beta_2 < u$$

$$\begin{aligned} V_{w2} &= \cancel{V_{r2} \cos \beta_2 - u_2} - V_{r2} \\ &= 43.41 \times \cos(20^\circ) - 43.40 \\ &= -2.60 \frac{\text{m}}{\text{s}} \end{aligned}$$

$$\begin{aligned} V_{w2} &= u_2 - V_{r2} \cos \beta_2 \\ &= 43.40 - 43.41 \cos(20^\circ) \\ V_{w2} &= 2.60 \frac{\text{m}}{\text{s}} \end{aligned}$$

power $P = T \times \omega$

$$= \{F \times r\} \times \omega$$

$$= \{(m \times \Delta V_w) \times r\} \times \omega$$

$$= m \times (V_{w1} - V_{w2}) \times \frac{\pi D N}{60}$$

$$= m \times [V_{w1} - V_{w2}] \times u$$

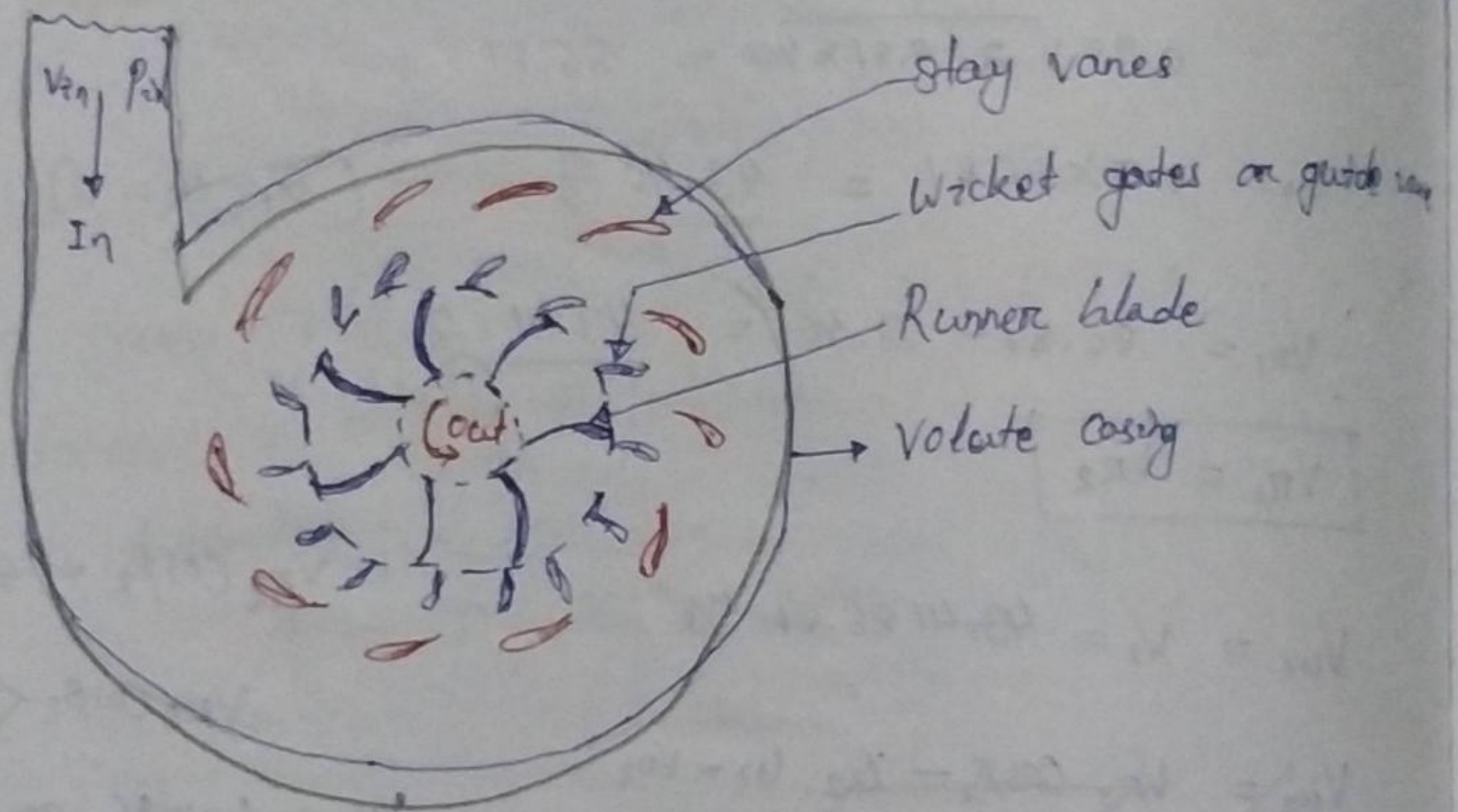
$$= \cancel{1000} \times [43.41 \times 86.81 - (-2.60)] \times 43.40$$

$$\boxed{P = 3880.39 \text{ kW}} = 3654.71 \text{ W}$$

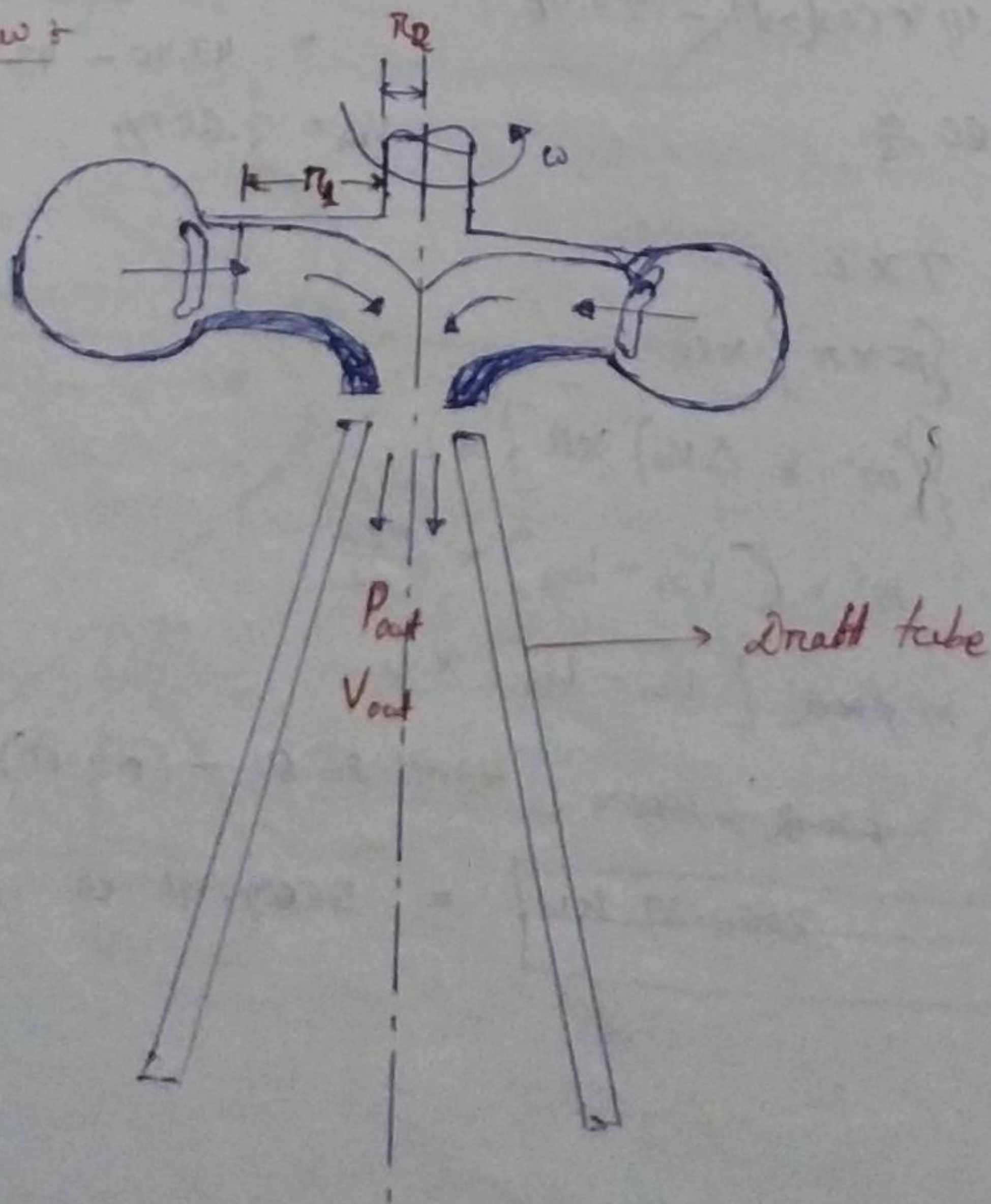
Francis turbine

* Top view and side view of a Francis turbine.

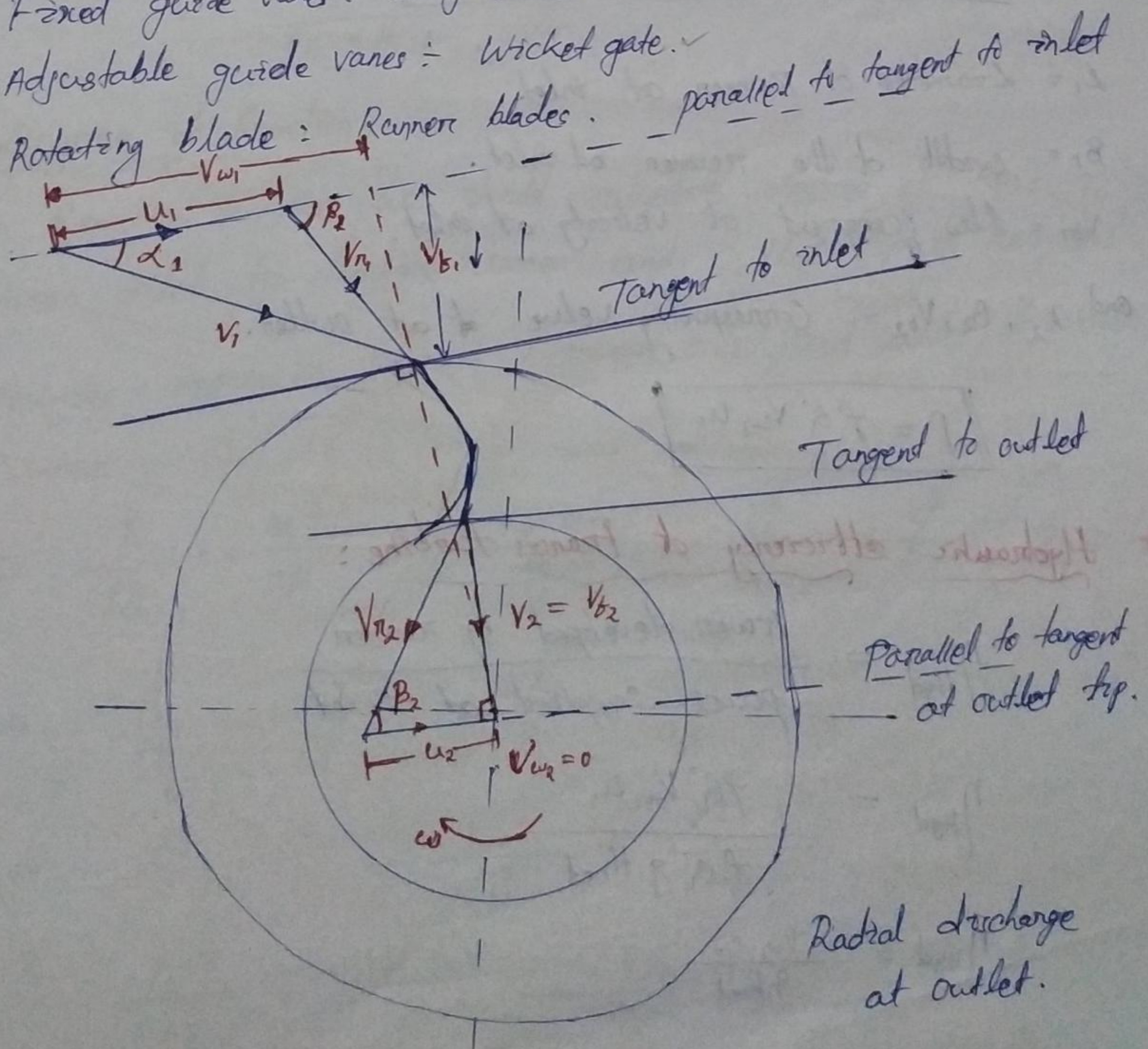
Top view:



Side view:



- In case of Reaction turbine energy available at inlet to turbine is both kinetic energy and pressure energy.
- Pressure energy is gradually change to K.E as fluid moves through runner.
- Francis turbine is an inward mixed flow reaction turbine.
- Fluid enters from penstock to a spiral casing which completely surrounding the runner. The cross-sectional area of casing decreases gradually along circumferences in order to volumetric flow rate constant through out circumferences.
- Fixed guide vanes: Stay vanes.
- Adjustable guide vanes: Wicket gate.
- Rotating blade: Runner blades.



Rate of work done by water jet on runner is given

$$P = T \times \omega$$

$$= (F \times r) \times \frac{v}{r}$$

$$= M \times \Delta V_w \times r \times \frac{v}{r}$$

$$= m \times \Delta V_w \times v$$

$$= f \cdot Q [V_{w1} - V_{w2}] \times u$$

$$= f \cdot Q [V_{w1} u_1 - V_{w2} u_2] \quad (\because V_{w2} = 0)$$

Volumetric flow rate is given by

$$Q = \pi D_1 \cdot B_1 \cdot V_{b1} = \pi D_2 \cdot B_2 \cdot V_{b2} \quad \left(\frac{m^3}{\text{sec}} \right)$$

D_1 = Diameter of runner at inlet.

B_1 = width of the runner at inlet.

V_{b1} = flow component of velocity at inlet.

and D_2, B_2, V_{b2} = corresponding value at outlet.

$$P = f \cdot Q \cdot V_{w1} \cdot u_1$$

* Hydraulic efficiency of Francis turbine:

$$\eta_{\text{Hyd}} = \frac{\text{power developed by runner}}{\text{power supplied at inlet}}$$

$$\eta_{\text{Hyd}} = \frac{f \cdot Q \cdot V_{w1} \cdot u_1}{f \cdot Q \cdot g \cdot H_{\text{net}}}$$

$$\eta_{\text{Hyd}} = \frac{V_{w1} \cdot u_1}{g \cdot H_{\text{net}}}$$

* Mechanical efficiency:

$$\eta_{\text{mech}} = \frac{\text{power available at turbine shaft}}{\text{power runner}}$$

$$\eta_{\text{mech}} = \frac{SP}{fQ V_{w1} U_1} \quad (SP = \text{shaft power})$$

* overall efficiency:

$$\eta_{\text{overall}} = \eta_{\text{hyd}} \times \eta_{\text{mech}}$$

$$= \frac{SP}{\text{power available at turbine inlet}}$$

$$= \frac{SP}{fQ \cdot g \cdot H_{\text{net}}}$$

* Degree of Reaction:

→ In Reaction turbine we should calculate degree of Reaction between inlet to turbine runner and outlet tip of runner.

→ Applying Bernoulli's theorem between inlet and outlet of turbine runner.

$$\Rightarrow \frac{P_1}{fg} + \frac{V_1^2}{2g} = \frac{V_{w1} \cdot U_1}{g} + \frac{P_2}{fg} + \frac{V_2^2}{2g} \quad (Z_1 = H_{\text{net}})$$

$$\Rightarrow \frac{P_1}{fg} - \frac{P_2}{fg} = \frac{V_{w1} U_1}{g} + \frac{V_2^2}{2g} - \frac{V_1^2}{2g}$$

$$\Rightarrow \frac{P_1 - P_2}{fg} = \frac{V_{w1} U_1}{g} + \frac{V_2^2 - V_1^2}{2g}$$

$$\Rightarrow \frac{P_1 - P_2}{fg} = \frac{V_{w1} \cdot U_1}{g} - \left[\frac{V_1^2 + V_2^2}{2g} \right]$$

$$\text{Degree of Reaction 'R'} = \frac{P_1 - P_2}{\rho g} \div \frac{V_{w1} u_1}{g}$$

$$R = \frac{\frac{V_{w1} u_1}{g} - \left[\frac{V_1^2 - V_2^2}{2g} \right]}{\frac{V_{w1} u_1}{g}}$$

$$R = \frac{\frac{V_{w1} u_1}{g} - \left[\frac{V_1^2 - V_2^2}{2g} \right]}{\frac{V_{w1} u_1}{g}} = \frac{V_{w1} u_1}{g} - \frac{\left[\frac{V_1^2 - V_2^2}{2g} \right]}{\frac{V_{w1} u_1}{g}}$$

$$R = 1 - \frac{V_1^2 - V_2^2}{2g} \times \frac{g}{V_{w1} u_1}$$

$$R = 1 - \frac{V_1^2 - V_2^2}{2(V_{w1} u_1)}$$

But it is very much conducive to express degree of Reaction in terms of angle.

guide blade angle α'

Vane angle β'

Degree of Reaction is given by

$$R = 1 - \frac{V_1^2 - V_2^2}{2 V_{w1} u_1}$$

Now let us express V_1, V_2, V_{w1} and u_1 in terms of α and β

From inlet velocity triangle

$$\sin \alpha_1 = \frac{V_{t1}}{V_1}$$

$$\Rightarrow \frac{V_{t1}}{V_1} = \csc \alpha_1 \Rightarrow \frac{V_1}{V_{t1}} = \cos \alpha_1$$

$$\Rightarrow V_{t1} =$$

$$\Rightarrow V_1 = V_{t1} \csc \alpha_1$$

Now V_2 :

$$V_2 = V_{t2}$$

$\frac{V_{w1}}{V_{t1}}$

$$\tan \alpha_1 = \frac{V_{t1}}{V_{w1}}$$

$$\Rightarrow \frac{V_{w1}}{V_{t1}} = \cot \alpha_1$$

$$\Rightarrow V_{w1} = V_{t1} \cot \alpha_1$$

To calculate u_1

From inlet velocity triangle $\tan \beta_1 = \frac{V_{t1}}{(V_{w1} - u_1)}$

$$\Rightarrow \frac{V_{w1} - u_1}{V_{t1}} = \cot \beta_1$$

$$\Rightarrow V_{w1} - u_1 = V_{t1} \cot \beta_1$$

$$\Rightarrow -u_1 = V_{t1} \cot \beta_1 - V_{w1}$$

$$= -u_1 = -(V_{w1} - V_{t1} \cot \beta_1)$$

$$= u_1 = V_{w1} - V_{t1} \cot \beta_1$$

$$R = 1 - \frac{V_1^2 - V_2^2}{2 V_{w1} u_1}$$

$$= 1 - \frac{V_{t1}^2 \operatorname{cosec}^2 \alpha_1 - V_{t2}^2}{2 V_{t1} \cot \alpha_1 \times V_{w1} - V_{t1} \cot \beta_1}$$

$$(V_2 = V_{t2} = V_{t1})$$

$$= 1 - \frac{V_{t1}^2 \operatorname{cosec}^2 \alpha_1 - V_{t1}^2}{2 V_{t1} \cot \alpha_1 \times (V_{t1} \cot \alpha_1 - V_{t1} \cot \beta_1)}$$

$$= 1 - \frac{V_{t1}^2 (\operatorname{cosec}^2 \alpha_1 - 1)}{2 V_{t1} \cot \alpha_1 \times V_{t1} (\cot \alpha_1 - \cot \beta_1)}$$

$$\begin{cases} \operatorname{cosec}^2 \alpha - \cot^2 \alpha = 1 \\ \operatorname{cosec}^2 \alpha - 1 = \cot^2 \alpha \end{cases}$$

$$= 1 - \frac{V_{b1}^2 \cot^2 \alpha_1}{2 V_{b1}^2 \cot \alpha_1 (\cot \alpha_1 - \cot \beta_1)}$$

$$R = 1 - \frac{\cot^2 \alpha_1}{2(\cot \alpha_1 - \cot \beta_1)}$$

* Working proportion for Francis turbine:-

$$\text{Flow ratio } \psi = \frac{V_{b1}}{\sqrt{2gH}} = 0.15 \leq \psi \leq 0.3$$

$$\text{Speed ratio } k_u = \frac{U}{\sqrt{2gH}} = 0.6 \leq k_u \leq 0.9$$

Q.1 The external and internal diameters of an inward flow Reaction turbine are 1.2m and 0.6m. respectively. Head on turbine is 8m. The flow component of velocity is constant and given by $0.3\sqrt{2gH}$. Its width of turbine at outlet is 300mm. Find the discharge and width of turbine at inlet.

Data given:-

$$D_1 = 1.2 \text{ m.}$$

$$D_2 = 0.6 \text{ m}$$

$$H_{\text{net}} = 8 \text{ m.}$$

$$V_{b1} = V_{b2} = 0.3\sqrt{2gH} = 0.3\sqrt{2 \times 9.81 \times 8} = 3.75$$

$$\text{width of turbine at outlet } B_2 = 300 \text{ mm} = 0.3 \text{ m}$$

To Find:-

$$\text{Volumetric flow rate } Q \left(\frac{\text{m}^3}{\text{sec}} \right) = \underline{\hspace{2cm}}$$

$$\text{width of turbine at inlet } B_1 = \underline{\hspace{2cm}}$$

Volumetric flow rate

$$Q = \pi D_2 B_2 V_{b2}$$

$$\Rightarrow Q = \pi \times 0.6 \times 0.3 \times 3.75$$

$$\Rightarrow Q = 2.12 \frac{m^3}{sec}$$

width of turbine at inlet B_1 -

$$Q = \pi D_1 B_1 V_{b1}$$

$$\Rightarrow 2.12 = \pi \times 1.2 \times B_1 \times 3.75$$

$$\Rightarrow B_1 = \frac{2.12}{\pi \times 1.2 \times 3.75}$$

$$\Rightarrow B_1 = 0.14 \text{ m.}$$

Q.2 A francis turbine has to be developed designed to developed 370kw under a head of 70m. while running at speed equal to 750 rpm.

- Ratio of width of runner at inlet to outer dia of runner = 0.1
- Ratio of inner dia to outer dia = 0.5
- Hydraulic efficiency = 95%
- mechanical efficiency = 84%
- Circumferential area occupied by thickness of vanes = 5%
- Flow ratio (ψ) = 0.15

Assume velocity flow constant. Calculate

(i) guide vane angle at inlet

(ii) runner vane angle at inlet.

(iii) runner vane angle at outlet.

Data given

$$RP = 370 \text{ kW} = 370 \times 10^3 \text{ W}$$

$$H_{\text{net}} = 70 \text{ m.}$$

$$N = 750 \text{ RPM.}$$

$$\frac{B_1}{D_1} = 0.1 \Rightarrow B_1 = 0.1 \times D_1$$

$$\frac{D_2}{D_1} = 0.5$$

$$\eta_{\text{hyd}} = 95\% = 0.95$$

$$\eta_{\text{mech}} = 84\% = 0.84$$

$$\text{Flow ratio } \psi = 0.157 = 0.15$$

To Find

$$\alpha_1 =$$

$$\beta_1 =$$

$$\beta_2 =$$

Analysis

$$\eta_{\text{hyd}} = \frac{RP}{\text{power available at runner inlet}}$$

$$\eta_{\text{hyd}} = \frac{370 \times 10^3 \text{ W}}{\rho \cdot Q \cdot g \cdot H_{\text{net}}}$$

$$\Rightarrow 0.95 = \frac{370 \times 10^3}{\rho \cdot Q \cdot g \cdot H_{\text{net}}}$$

$$\Rightarrow 0.95 (\rho \cdot Q \cdot g \cdot H_{\text{net}}) = 370 \times 10^3$$

$$\Rightarrow Q = \frac{370 \times 10^3}{0.95 \times 1000 \times 9.81 \times 70}$$

$$\Rightarrow Q = 0.567 \frac{\text{m}^3}{\text{s}}$$

~~$$Q = \pi D_1 B_1 V_{b1}$$~~

→ From expression of flow rate $\psi = \frac{V_{b1}}{\sqrt{2g \cdot H_{net}}}$

$$\Rightarrow 0.15 = \frac{V_{b1}}{\sqrt{2 \times 9.81 \times 70}}$$

$$\Rightarrow V_{b1} = 0.15 \times 37.05 = 5.55 \frac{m}{s} \quad [V_{b1} = V_{b2} = V_2]$$

$$Q = \pi D_1 B_1 V_{b1}$$

$$\Rightarrow 0.567 = \pi \times D_1 \times 0.1 D_1 \times 5.55$$

$$\Rightarrow 0.567 = \pi D_1^2 \cdot 0.1 \times 5.55$$

$$\Rightarrow 0.567 = 1.74 D_1^2$$

$$\Rightarrow D_1^2 = \frac{0.567}{1.74}$$

$$\Rightarrow D_1 = \sqrt{\frac{0.567}{1.74}}$$

$$\Rightarrow \boxed{D_1 = 0.57 \text{ m}}$$

$$\frac{D_2}{D_1} = 0.5$$

$$\Rightarrow D_2 = 0.5 \times 0.57$$

$$\Rightarrow \boxed{D_2 = 0.285 \text{ m}}$$

and

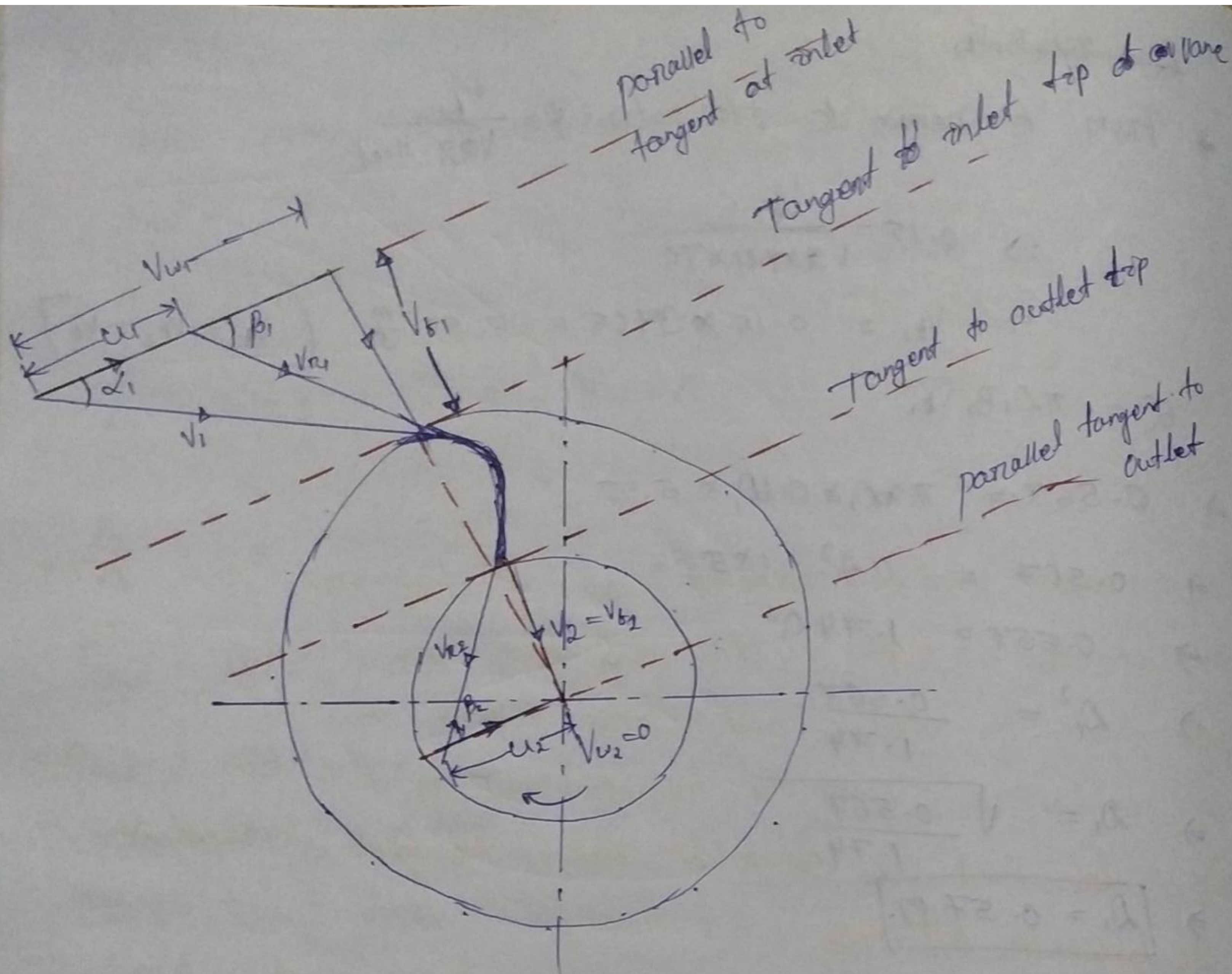
$$B_1 = 0.1 \times 0.57$$

$$\boxed{B_1 = 0.057 \text{ m}}$$

Now we can calculate ω_1 and ω_2

$$\omega_1 = \frac{\pi D_1 N}{60} = \frac{\pi \times 0.57 \times 750}{60} = 22.38 \frac{m}{s}$$

$$\omega_2 = \frac{\pi D_2 N}{60} = \frac{\pi \times 0.285 \times 750}{60} = 11.19 \frac{m}{s}$$



→ To Find out guide blade angle α and Vane angle β .
Velocity component should be determination

$$RP = f \cdot Q \cdot V_{w1} \cdot l_v$$

$$\Rightarrow 370 \times 10^3 = 1000 \times 0.567 \times V_{w1} \times 22.38$$

$$\Rightarrow V_{w1} = \frac{370 \times 10^3}{1000 \times 0.567 \times 22.38}$$

$$\Rightarrow V_{w1} = 29.15 \frac{m}{sec}$$

$$(i) \tan \alpha_1 = \frac{V_{b1}}{V_{w1}}$$

$$\Rightarrow \alpha_1 = \tan^{-1} \left(\frac{V_{b1}}{V_{w1}} \right)$$

$$= \tan^{-1} \left(\frac{5.55}{29.15} \right)$$

$$\boxed{\alpha_1 = 10.77^\circ}$$

$$(82) \quad \tan \beta_1 = \frac{V_{b1}}{(V_{w1} - u_1)}$$

$$\Rightarrow \beta_1 = \tan^{-1} \left(\frac{5.55}{29.15 - 22.38} \right) = \boxed{39.44^\circ}$$

(83)

$$\tan \beta_2 = \frac{V_{b2}}{u_2} \quad (\because V_{b1} = V_{b2})$$

$$\Rightarrow \beta_2 = \tan^{-1} \left(\frac{5.55}{11.19} \right) = \boxed{26.38^\circ}$$

③ A reaction turbine operating under head of 70m. Run at 400rpm. Its diameter at inlet is 1m. and flow area is 0.35m². The angle made by absolute and relative velocity at inlet are 15° and 45° respectively. with the tangential velocity. for radial discharge at outlet. Find the volume flow rate, the power developed and the hydraulic efficiency.

Data given

$$H_{net} = 70\text{m.}$$

$$N = 400\text{rpm}$$

$$D_1 = 1\text{m.}$$

$$\text{Flow area at inlet } A_{1,B1} = 0.35\text{m}^2$$

$$\alpha_1 = 15^\circ \quad \alpha_2 = 90^\circ$$

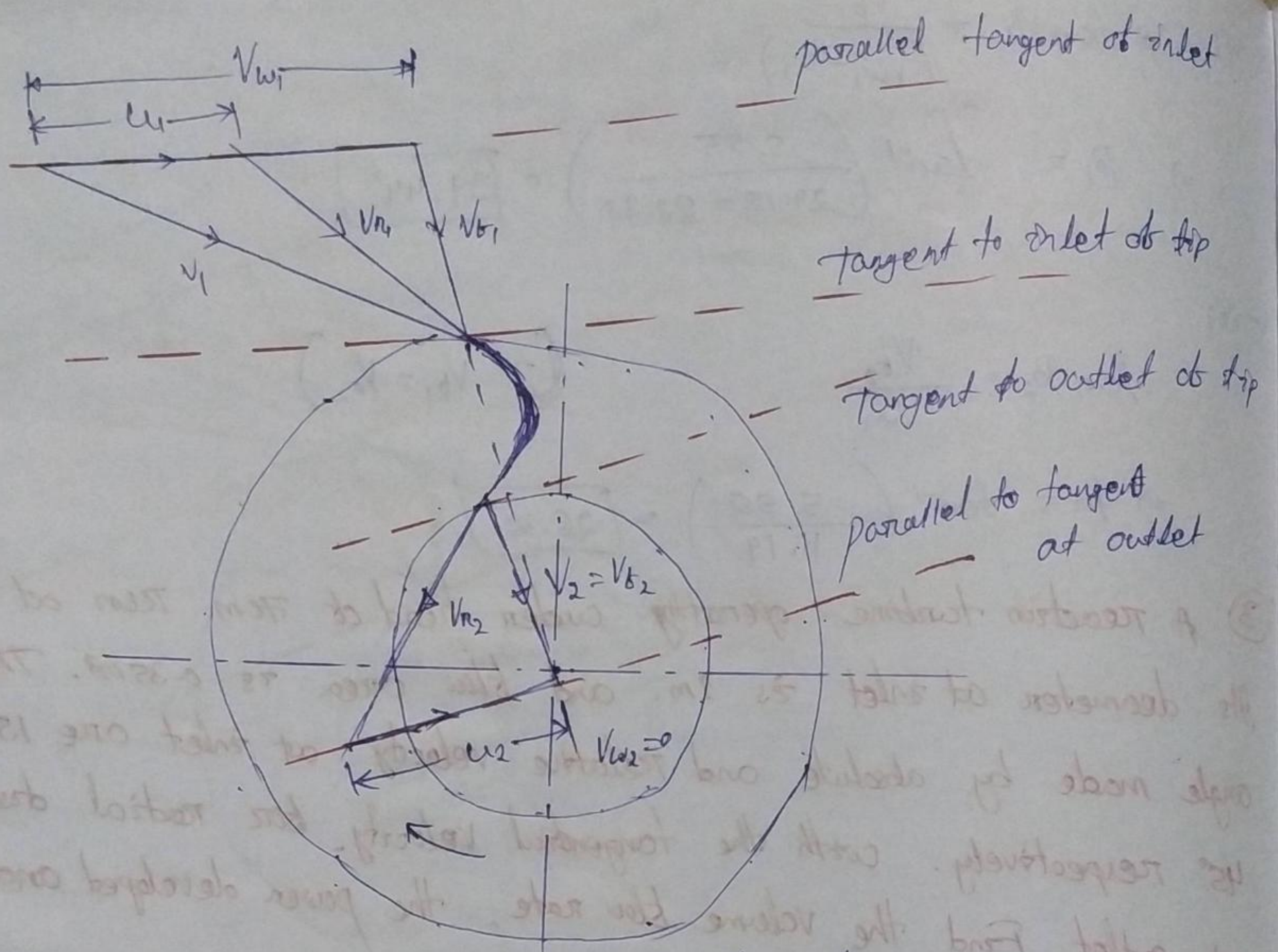
$$\beta_1 = 45^\circ$$

To Find

$$\text{Volume flow rate } Q =$$

$$\text{Runner power } RP =$$

$$\eta_{hyd} =$$



From Geometry on inlet velocity triangle

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

$$= \frac{\pi \times 1 \times 400}{60} = 20.94 \text{ m/s}$$

$$\tan \alpha_1 = \frac{V_{b1}}{V_{w1}} \Rightarrow V_{b1} = \tan \alpha_1 \cdot V_{w1}$$

$$\tan \beta_1 = \frac{V_{b1}}{(V_{w1} - u_1)} \Rightarrow V_{b1} = \tan \beta_1 \cdot (V_{w1} - u_1)$$

From above two expression

$$V_{b1} = \tan \alpha_1 \cdot V_{w1} = \tan \beta_1 \cdot (V_{w1} - u_1)$$

$$\Rightarrow \tan \alpha_1 \cdot V_{w1} = \tan \beta_1 \cdot V_{w1} - \tan \beta_1 \cdot u_1$$

$$\Rightarrow \tan(15^\circ) \cdot V_{w1} = \tan \beta_1 \cdot V_{w1} - \tan \beta_1 \cdot u_1$$

$$\Rightarrow 0.267 \times V_{w1} = V_{w1} - 20.94$$

$$\Rightarrow 0.267 V_{w1} - V_{w1} = -20.94$$

$$\Rightarrow -0.733 V_{w1} = -20.94$$

$$\Rightarrow 0.733 V_{w1} = 20.94$$

$$\Rightarrow V_{w1} = \frac{20.94}{0.733}$$

$$\Rightarrow \boxed{V_{w1} = 28.56 \frac{m}{s}}$$

so that $V_{b1} = \tan \alpha_1 \cdot V_{w1}$

$$= \tan(15^\circ) \times 28.56$$

$$\boxed{V_{b1} = 7.65 \frac{m}{s}}$$

(i) volumetric flow rate

$$Q' = \pi D_1 B_1 V_{b1}$$

$$= 0.35 m^2 \times 7.65 \frac{m}{s}$$

$$\boxed{Q' = 2.677 \frac{m^3}{s}}$$

(ii) Runner power (RP) = $\rho \cdot Q' \cdot V_{w1} \cdot u_1$

$$= 1000 \frac{kg}{m^3} \times 2.677 \frac{m^3}{s} \times 28.56 \frac{m}{s} \times 20.94 \frac{m}{s}$$

$$= 1600970.213 W$$

$$= 1600.97 kW$$

(iii) Hydraulic efficiency

$$\eta_{hyd} = \frac{RP}{\text{power available at inlet of Runner}}$$

$$= \frac{1600.97}{1000 \times 9.81 \times 2.677 \times 9.81 \times 70}$$

$$= \frac{\rho \cdot Q' \cdot V_{w1} \cdot u_1}{\rho \cdot g \cdot H_{net}}$$

$$= \frac{V_{w1} \cdot u_1}{g \cdot H_{net}} = \frac{28.56 \times 20.94}{9.81 \times 70} = 0.87 = 87\%$$

④ A Francis turbine developed 160kW under a head of 10m. and at a speed of 200rpm and gives an efficiency of 80%. The speed ratio is 0.3 and flow ratio is 0.9, hydraulic efficiency of turbine is 85%. For radial discharge. Find

- Diameter of wheel
- width of wheel at inlet.
- Guide blade angle at inlet.
- Wheel vane angle at inlet.

Data given:-

$$P = 160 \text{ kW} = 160 \times 10^3 \text{ W}$$

$$H_{\text{net}} = 10 \text{ m.}$$

$$N = 200 \text{ rpm}$$

$$\eta_o = \frac{\text{shaft power}}{\text{power available at runner inlet}} = 80\%$$

$$\text{speed ratio } K_u = 0.3$$

$$\text{flow ratio } \psi = 0.9$$

$$\eta_{\text{hyd}} = 85\% = 0.85$$

$$\text{Radial discharge } \alpha_2 = 90^\circ$$

To Find :-

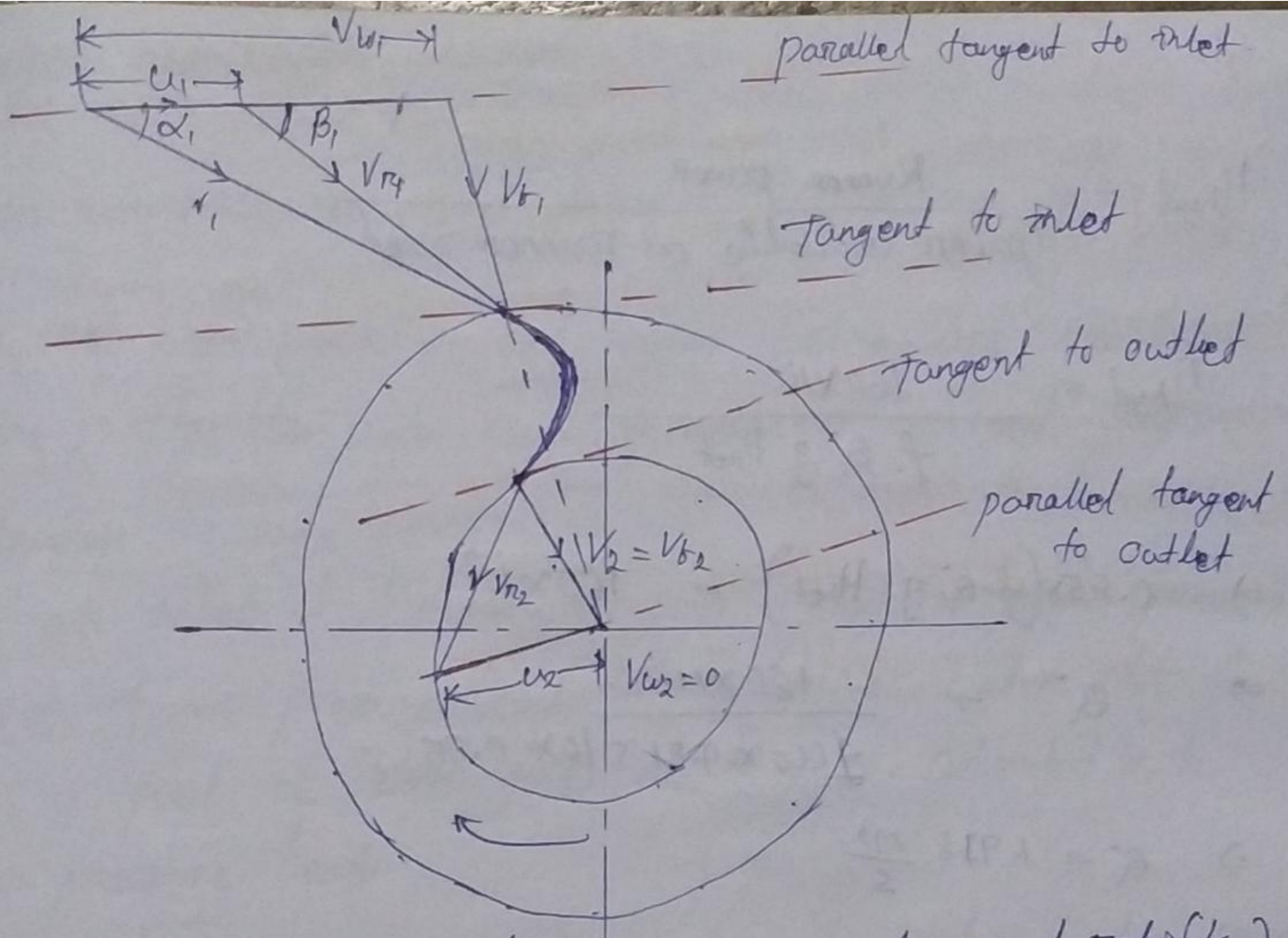
$$(a) \quad D_1 =$$

$$D_2 =$$

$$(b) \quad B_1 =$$

$$(c) \quad \alpha_1 =$$

$$(d) \quad \beta_1 =$$



→ It is quite evident from expression of speed ratio (K_u) and flow ratio (ψ) u_1 and V_{b1} can be determined.

$$K_u = \frac{u_1}{\sqrt{2gH_{net}}}$$

$$\Rightarrow 0.3 = \frac{u_1}{\sqrt{2 \times 9.81 \times 10}} \quad \text{--- (1)}$$

$$\Rightarrow u_1 = 0.3 \times \sqrt{2 \times 9.81 \times 10}$$

$$\Rightarrow \boxed{u_1 = 4.202 \text{ m/s}}$$

$$\psi = \frac{V_{b1}}{\sqrt{2gH}}$$

$$\Rightarrow 0.9 = \frac{V_{b1}}{\sqrt{2 \times 9.81 \times 10}} \quad \text{--- (2)}$$

$$\Rightarrow V_{b1} = 0.9 \times 14.00 \Rightarrow \boxed{V_{b1} = 12.6 \text{ m/s}}$$

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

$$\Rightarrow 4.202 = \frac{\pi D_1 \cdot 200}{60}$$

$$\Rightarrow 4.202 = 10.47 D_1$$

$$\Rightarrow D_1 = \frac{4.202}{10.47} = 0.40 \text{ m}$$

$$\eta_{hyd} = \frac{\text{Runner power}}{\text{power available at runner inlet}}$$

$$\eta_{hyd} = \frac{160 \times 10^3}{f \cdot Q \cdot g \cdot H_{net}}$$

$$\Rightarrow 0.85 \times (f \cdot Q \cdot g \cdot H_{net}) = 160 \times 10^3$$

$$\Rightarrow Q = \frac{160 \times 10^3}{1000 \times 9.81 \times 10 \times 0.85}$$

$$\Rightarrow Q = 1.918 \frac{m^3}{s}$$

$$Q = \pi D_1 B_1 V_{B1}$$

$$\Rightarrow B_1 = \frac{1.918}{\pi \times 0.40 \times 12.6}$$

$$\Rightarrow B_1 = 0.12 \text{ m}$$

$$\text{Runner power } 160 \times 10^3 = f \times Q \times V_{w1} \times \omega$$

$$\Rightarrow 160 \times 10^3 = 1000 \times 1.918 \times V_{w1} \times 4.202$$

$$\Rightarrow V_{w1} = \frac{160 \times 10^3}{1000 \times 1.918 \times 4.202}$$

$$\Rightarrow V_{w1} = 19.85$$

From inlet velocity triangle

$$\tan \alpha_1 = \frac{V_{B1}}{V_{w1}}$$

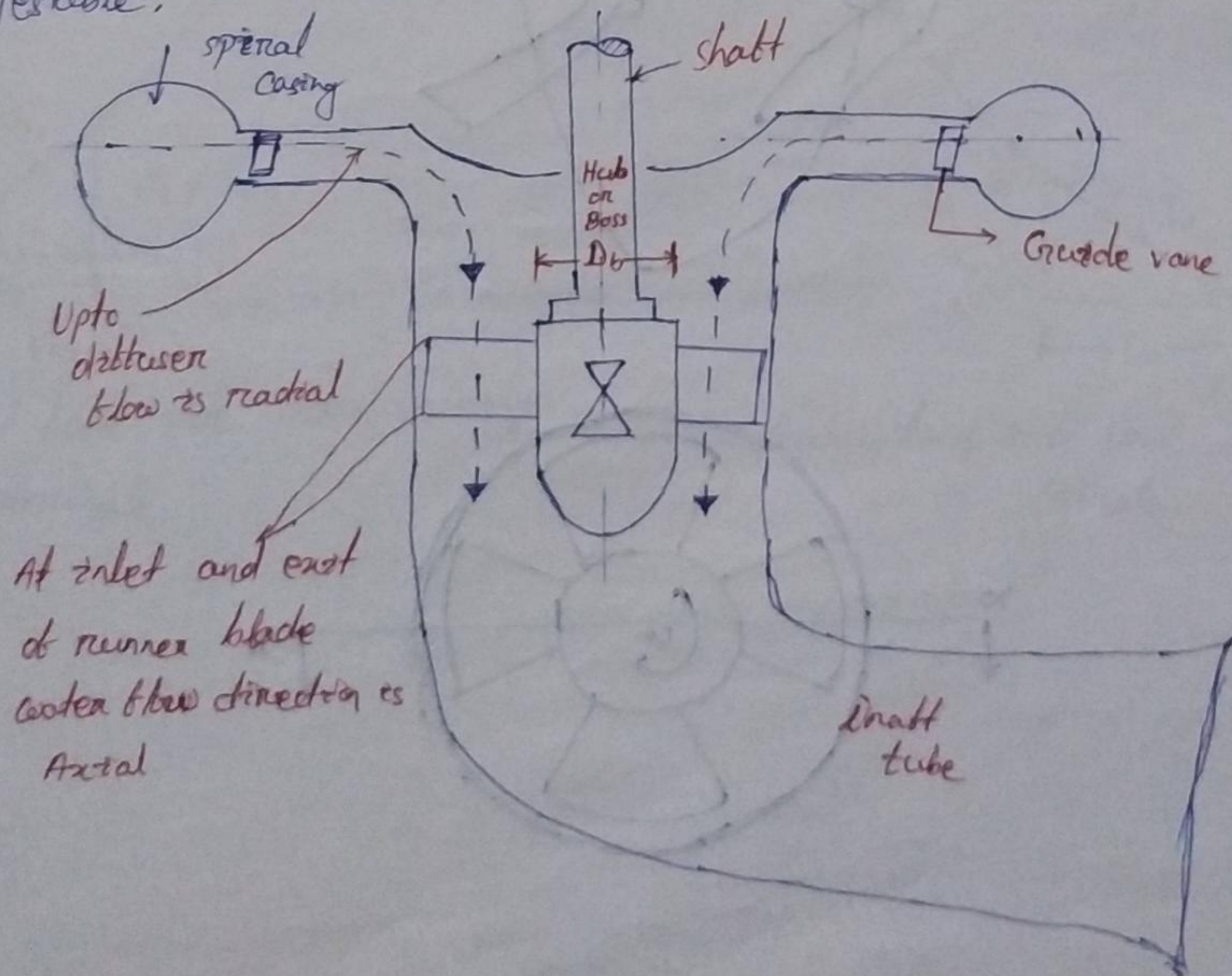
$$\Rightarrow \alpha_1 = \tan^{-1} \left(\frac{12.6}{19.85} \right) = \boxed{32.40^\circ}$$

$$\tan \beta_1 = \frac{V_{B1}}{V_{w1} - u_1}$$

$$\Rightarrow \beta_1 = \tan^{-1} \left(\frac{12.6}{19.85 - 4.202} \right) \Rightarrow \boxed{\beta_1 = 38.84^\circ}$$

* Propeller and Kaplan turbine :-

- Power available at inlet of a turbine runner $\propto Q \cdot H$
- Since we know propeller and kaplan turbine are low head turbine, the flow rate must increase in required proportionate to develop the same power.
- Head at inlet of turbine is sum of static pressure head and dynamic pressure head. During flow of water through runner a part of static pressure head is converted in to dynamic pressure head.
- Propeller turbine are nearly identical to kaplan turbine except blade of propeller turbine are fixed on hub and are not adjustable, where as blades of kaplan turbine are adjustable.

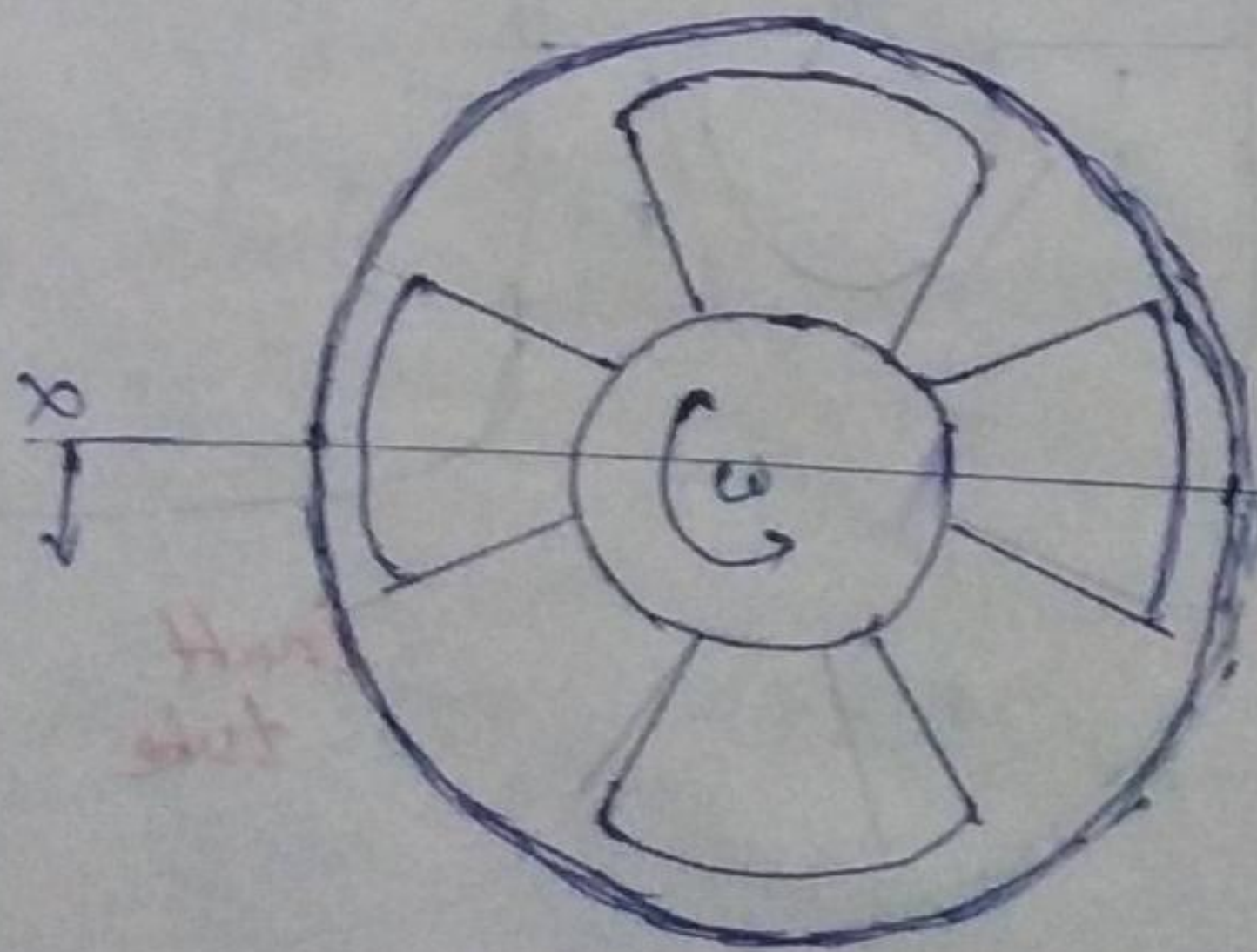
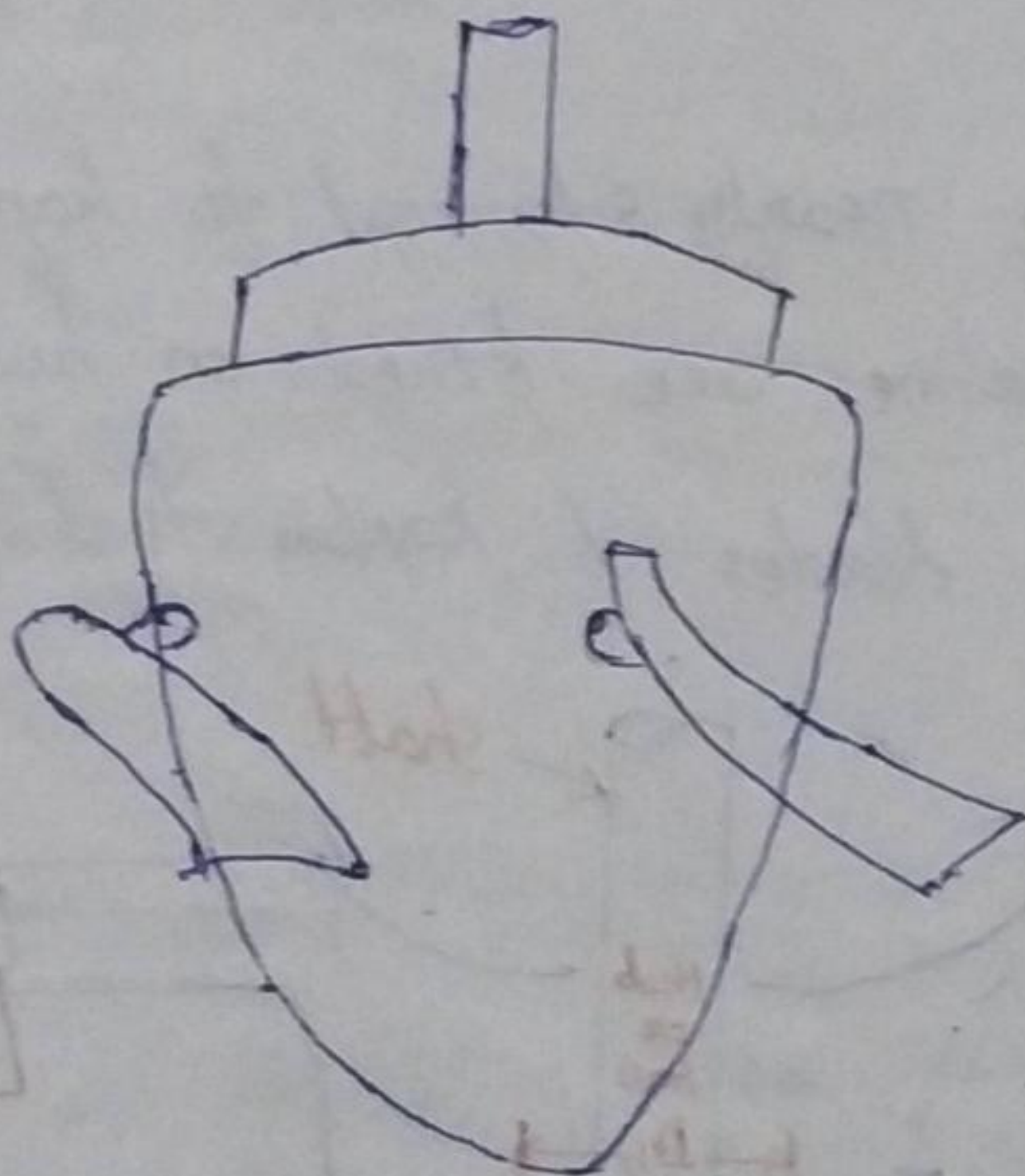


Note:-

- Kaplan co-axial flow reaction turbine.

→ Kaplan turbine installations in India.

1. Bhakra Nangal project --- Punjab --- Nangla hydro channel.
2. Hirakud dam project --- Odisha --- Mahanadi river.
3. Nizam Sagar project --- A.P. --- Narmada river.
4. Radhargarh hydro electric project --- Maharashtra --- Bhagabati river.
Kelhapur
5. Tungabhadra hydro electric project --- Karnataka --- Tungabhadra river.
Tungabhadra



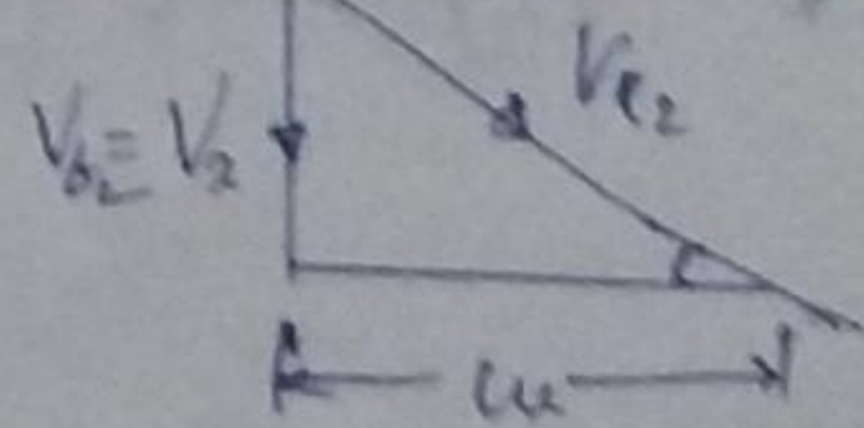
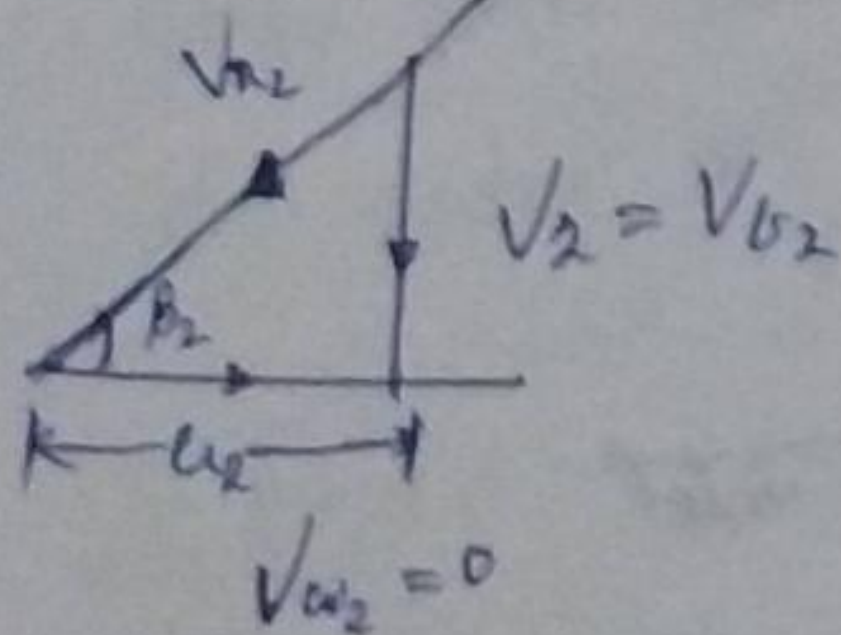
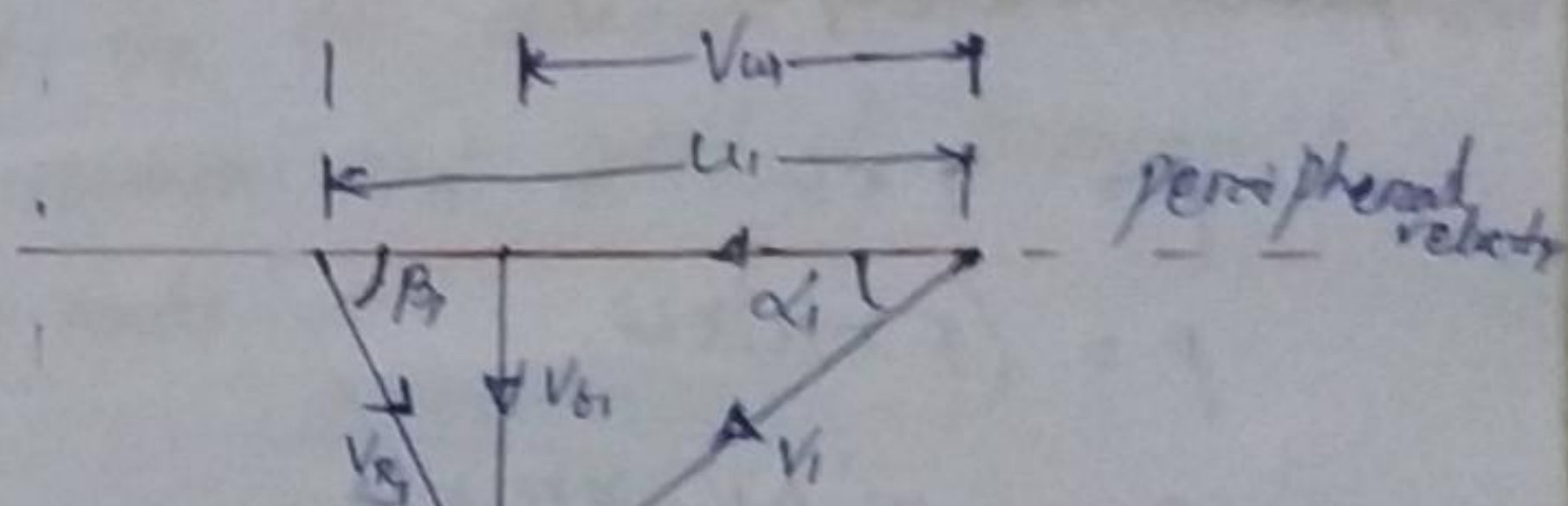
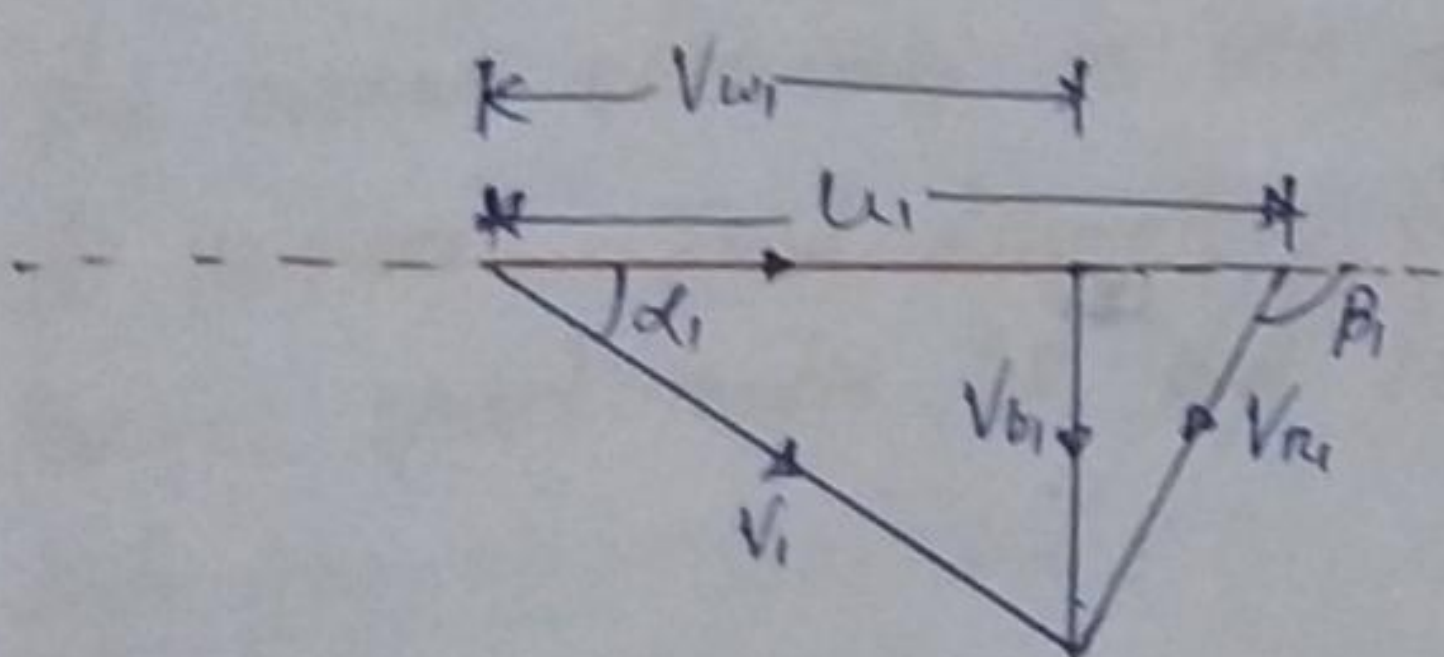
peripheral velocity

$$u_1 = u_2 = \frac{\pi D_{\text{mean}} N}{60}$$

average dia or mean dia

$$D = \frac{D_{\text{runner}} + D_{\text{boss}}}{2}$$

velocity triangle



left hand side vane velocity triangle.

Right hand side vane velocity triangle.

* For Kaplan turbine the volumetric flow rate is

$$Q = \frac{\pi}{4} [D^2 - D_b^2] \times V_{b1}$$

where D = Diameter of runner (outer dia of runner)

D_b = Diameter of hub (inner dia of runner)

power developed by turbine runner

$$P = T \times \omega$$

$$P = (F \times r) \times \omega$$

$$P = m \cdot \Delta V \omega \times r \times \frac{\omega}{r}$$

$$P = m \cdot \Delta V \omega$$

$$P = m (V_{w1} - V_{w2}) \omega$$

$$P = f \times Q [V_{w1} \omega_1 - V_{w2} \omega_2]$$

$$P = f \times Q \times V_{w1} \omega_1$$

* Hydraulic efficiency

$$\eta_{hyd} = \frac{\text{Runner power}}{\text{power available at turbine inlet}}$$

$$\eta_{hyd} = \frac{f \times Q \times V_{w1} \omega_1}{f \times Q \times g \cdot H}$$

$$\eta_{hyd} = \frac{V_{w1} \omega_1}{g \cdot H}$$

* mechanical efficiency

$$\eta_{mech} = \frac{\text{Shaft power}}{\text{Runner power}}$$

* overall efficiency

$$\eta_{\text{overall}} = \frac{\text{s.p}}{\text{power available at turbine inlet}}$$

$$\boxed{\eta_o = \eta_{\text{mech}} \times \eta_{\text{hyd}}}$$

Numerical

Q.1 A Kaplan turbine operating under a head of 10m develops 8000kW. The overall efficiency is 85% and the boss diameter is 0.4 times diameter of runner. Find Diameter of runner and speed of runner. Assume speed ratio 1.8 and flow ratio 0.6.

Data given

$$\text{Head } H = 10 \text{ m.}$$

$$P = 8000 \text{ kW} = 8000000 \text{ W}$$

$$\eta_{\text{overall}} = 85\% = 0.85$$

$$D_{\text{boss}} = 0.4 D_{\text{runner}}$$

$$\text{speed ratio } K_u = \frac{u_1}{\sqrt{2gH}} = 1.8$$

$$\text{flow ratio } \psi = \frac{V_{f1}}{\sqrt{2gH}} = 0.6$$

To Find

$$D_{\text{runner}} =$$

$$N_{\text{runner}} =$$

Analysis

At first we must utilize speed ratio and flow ratio

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

$$\Rightarrow 1.8 = \frac{u_1}{\sqrt{2 \times 9.81 \times 10}}$$

$$\Rightarrow u_1 = 1.8 \times \sqrt{2 \times 9.81 \times 10}$$

$$\Rightarrow u_1 = 25.21 \text{ m/s}$$

$$\text{flow rate} \quad \psi = \frac{V_{b1}}{\sqrt{2g \cdot H}}$$

$$\Rightarrow 0.6 = \frac{V_{b1}}{\sqrt{2 \times 9.81 \times 10}}$$

$$\Rightarrow V_{b1} = 0.6 \times \sqrt{2 \times 9.81 \times 10}$$

$$\Rightarrow V_{b1} = 8.404 \text{ m/s}$$

$$\eta_{\text{overall}} = \frac{\text{S.P}}{\text{power available at runner inlet}}$$

$$\Rightarrow 0.85 = \frac{8000000 \text{ W}}{f \cdot Q \cdot g \cdot H}$$

$$\Rightarrow 0.85 \times f \cdot Q \cdot g \cdot H = 8000000$$

$$\Rightarrow Q = \frac{8000000}{0.85 \times 1000 \times 9.81 \times 10}$$

$$\Rightarrow Q = 95.94 \text{ m}^3/\text{sec}$$

and Volumetric flow rate is given by expression

$$Q = \frac{\pi}{4} [D_{\text{runner}}^2 - D_{\text{boss}}^2] V_{b1}$$

$$Q = \frac{\pi}{4} [D_{\text{runner}}^2 - (0.4 D_{\text{runner}})^2] V_{b1}$$

$$\Rightarrow 95.94 = \frac{\pi}{4} [D_{\text{runner}}^2 - 0.16 D_{\text{runner}}^2] \times 8.404$$

$$\Rightarrow 95.94 = [D_{\text{runner}}^2 (1 - 0.16)] \times 6.60$$

$$\Rightarrow D_{\text{runner}}^2 = \frac{95.94}{0.84 \times 6.60}$$

$$\Rightarrow D_{\text{runner}}^2 = 17.30$$

$$\Rightarrow D_{\text{runner}} = \sqrt{17.30}$$

$$\Rightarrow \boxed{D_{\text{runner}} = 4.15 \text{ m}}$$

$$\therefore D_{\text{boss}} = 0.4 \times 4.15 = \boxed{1.66 \text{ m}}$$

speed of runner can be calculate from the expression

$$D_{\text{mean}} = \frac{D_{\text{run}} + D_{\text{boss}}}{2}$$

$$= \frac{4.15 + 1.66}{2} = 2.905 \text{ m.}$$

$$u_1 = \frac{\pi D_{\text{mean}} N}{60}$$

$$\Rightarrow u_1 = \frac{\pi \times 2.905 \cdot N}{60}$$

$$\Rightarrow 25.21 = 0.15 N$$

$$\Rightarrow N = \frac{25.21}{0.15}$$

$$\Rightarrow N = 168.06 \approx 167 \text{ rpm}$$

upto this level we did not require velocity triangle so we did not draw velocity triangle.

Q.2

Following data refer to Kaplan turbine
Diameter of boss = 0.4 times external diameter

$$\text{Head} = 8 \text{ m}$$

$$\text{speed ratio} = 2$$

$$\text{flow ratio} = 0.6$$

$$\text{speed} = 80 \text{ RPM}$$

$$\text{shaft power} = 11 \text{ MW}$$

Find overall efficiency

Data given

$$D_{\text{boss}} = 0.4 D_{\text{runner}}$$

$$H = 8 \text{ m}$$

$$\text{speed ratio } k_u = \frac{u_1}{\sqrt{2gH}} = 2$$

$$\text{flow ratio } \psi = \frac{V_{f1}}{\sqrt{2gH}} = 0.6$$

$$N = 80 \text{ rpm}$$

$$SP = 11 \text{ MW} = 11000000 \text{ W}$$

To Find:

overall efficiency $\eta_o =$

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

$$Q = \frac{\pi}{4} [D_{run}^2 - D_{boss}^2] V_{b1}$$

D_{run} and D_{boss} has to be calculate.

Expression:

speed rotation $K_u = \frac{u_1}{\sqrt{2gH}}$

$$\Rightarrow K_u = \frac{u_1}{\sqrt{2 \times 9.81 \times 8}}$$

$$\Rightarrow u_1 = 2 \times \sqrt{2 \times 9.81 \times 8}$$

$$\Rightarrow u_1 = 25.05 \text{ m/s}$$

flow ratio $\phi = \frac{V_{b1}}{\sqrt{2gH}}$

$$\Rightarrow 0.6 = \frac{V_{b1}}{\sqrt{2 \times 9.81 \times 8}}$$

$$\Rightarrow V_{b1} = 0.6 \times \sqrt{2 \times 9.81 \times 8}$$

$$\Rightarrow V_{b1} = 7.51 \text{ m/s}$$

$$D_{mean} = \frac{D_{run} + D_{boss}}{2}$$

$$= \frac{D_{run} + 0.4 D_{run}}{2}$$

$$D_{mean} = 0.7 D_{run}$$

$$u_1 = \frac{\pi D_{mean} N}{60}$$

$$\Rightarrow 25.05 = \frac{\pi \times 0.7 D_{run} \times 80}{60}$$

$$\Rightarrow 25.05 = 2.93 D_{run}$$

$$\Rightarrow D_{run} = \frac{25.05}{2.93} = 8.549 \text{ m}$$

$$\text{and } D_{\text{boss}} = 0.4 \times 8.54 \\ = 3.416 \text{ m}$$

$$Q = \frac{\pi}{4} [(8.54)^2 - (3.416)^2] \times 7.51$$

$$Q = 362.25 \frac{\text{m}^3}{\text{sec}}$$

$$\therefore \eta_o = \frac{\text{S.P.}}{f Q \rho g H}$$

$$\Rightarrow \eta_o = \frac{11000000 \text{ W}}{1000 \times 362.25 \times 9.81 \times 8}$$

$$\Rightarrow \eta_o = 0.38 = 38\%$$

In this problem, we didn't deal the requirement of velocity triangle.

Q.3 A Kaplan turbine develops 20 MW when running at 130 RPM under a head of 20 m. The outer diameter of runner is 4 m and hub diameter is 1.8 m. The hydraulic and overall efficiency of turbine is 90% and 85% respectively. Determine the discharge through the turbine and blade angle or vane angle at inlet and outlet, measured at mean radius of runner.

Data given

$$P = 20 \text{ MW} = 20000 \text{ kW}$$

$$N = 130 \text{ RPM}$$

$$H = 20 \text{ m.}$$

$$D_{\text{runner}} = 4 \text{ m.}$$

$$D_{\text{boss}} = 1.8 \text{ m}$$

$$\eta_{\text{hyd}} = 90\%$$

$$\eta_o = 85\%$$

To Find +

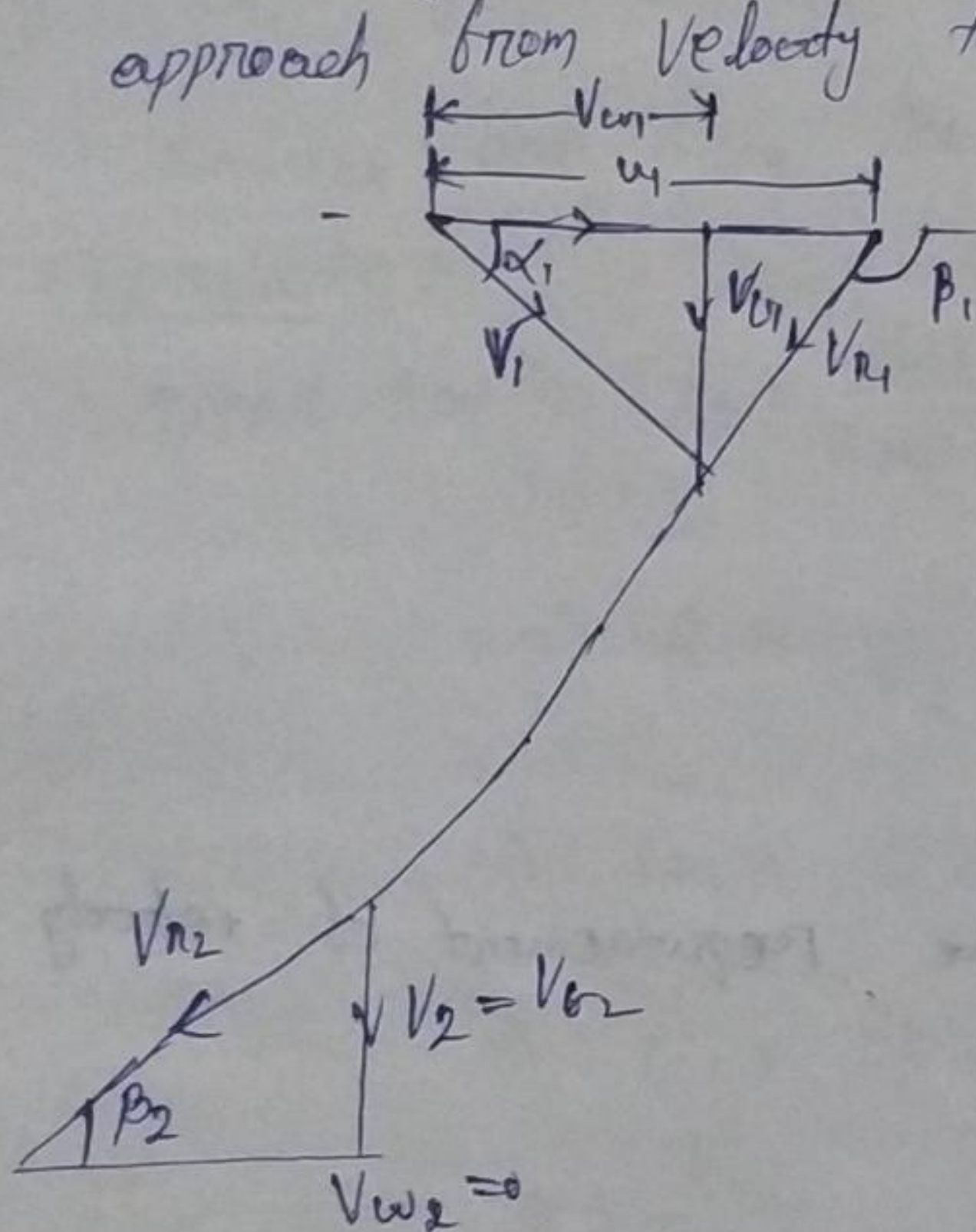
$$Q' =$$

$$\beta_1 =$$

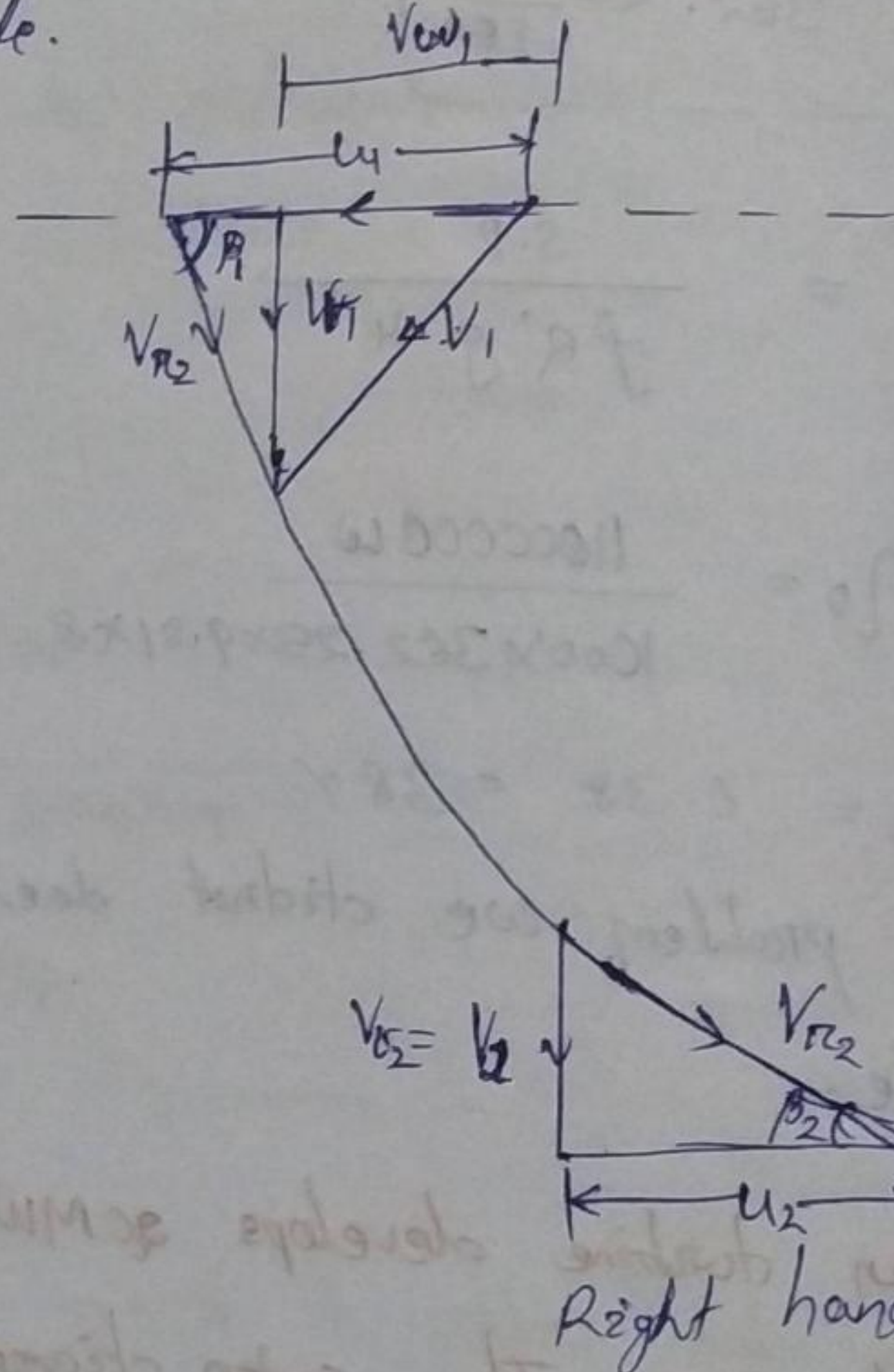
$$\beta_2 =$$

Analysis

Since angle are required to find out we will initiate our approach from Velocity triangle.



left hand vane



Right hand vane

To determine β_1 and β_2 we must calculate u_1, V_{w1}, V_1, V_2 and u_2

$$\eta_o = \frac{SP}{f Q g H}$$

$$\Rightarrow 0.85 = \frac{20000000}{1000 \times Q' \times 9.81 \times 20}$$

$$\Rightarrow 0.85 = \frac{101.93}{Q'}$$

$$\Rightarrow Q' = \frac{101.93}{0.85}$$

$$\Rightarrow Q' = 119.91 \approx 120 \frac{m^3}{s}$$

$$Q = \frac{\pi}{4} [(D_{mean})^2 - (D_{boss})^2] \times V_{b1}$$

$$\Rightarrow 120 = \frac{\pi}{4} [(4)^2 - (1.8)^2] \times V_{b1}$$

$$\Rightarrow V_{b1} = \frac{120}{\frac{\pi}{4} [16 - 3.24]}$$

$$\Rightarrow V_{b1} = 11.97$$

from assumption of Reaction turbine we can consider

$$V_{b1} = V_{b2} = V_2 = 11.97 \text{ m/s}$$

$$u_1 = \frac{\pi D_{mean} N}{60}$$

$$u_1 = \frac{\pi \left(\frac{4+1.8}{2} \right) \times 130}{60}$$

$$u_1 = 19.73 \text{ m/s} \quad (u_1 = u_2)$$

To determine V_{w1} we must consider η_{hyd}

$$\eta_{hyd} = \frac{f \cdot Q \cdot V_{w1} \cdot u_1}{f \cdot Q \cdot g \cdot H}$$

$$\Rightarrow \eta_{hyd} = \frac{V_{w1} \times 19.73}{9.81 \times 20}$$

$$\Rightarrow 0.9 = 0.100 V_{w1}$$

$$\Rightarrow V_{w1} = \frac{0.9}{0.100}$$

$$\Rightarrow V_{w1} = 9 \text{ m/s}$$

determine the angle

$$\tan \beta_1 = \frac{V_{b1}}{(u_1 - V_{w1})}$$

$$\Rightarrow \beta_1 = \tan^{-1} \left(\frac{11.97}{19.73 - 9} \right)$$

$$\Rightarrow \boxed{\beta_1 = 48.72^\circ}$$

$$\tan \beta_2 = \frac{V_{b2}}{u_2}$$

$$\Rightarrow \beta_2 = \tan^{-1} \left(\frac{11.97}{19.73} \right) \Rightarrow \boxed{\beta_2 = 31.24^\circ}$$

MCA ON HYDRAULIC TURBINE

- ① Velocity triangle are used to analyze _____
- (a) Flow of water along blades of turbine
 - (b) Measure discharge of flow.
 - (c) Angle of deflection of jet.
 - ☒ (d) Flow of water, measure of discharge, angle of deflection.
- ② In which the following turbine inlet and outlet blade velocity are equal?
- (a) Francis turbine.
 - (b) Kaplan turbine
 - ☒ (c) Pelton turbine.
 - (d) propeller turbine.
- ③ Tangential velocity of blade of pelton wheel is proportional to _____
- (a) speed of wheel.
 - (b) Angular velocity of wheel.
 - ☒ (c) RPM of wheel.
 - (d) speed, Angular velocity, RPM of the wheel.
- ④ The value of coefficient of velocity is _____
- ☒ (a) 0.98
 - (b) 0.65
 - (c) 0.85
 - (d) 0.33
- ⑤ In pelton wheel, relative velocity of jet at inlet with respect to tangential velocity of vane is _____
- (a) Difference between inlet jet velocity and blade velocity
 - (b) Sum of inlet jet velocity and blade velocity
 - ☒ (c) Inlet jet velocity.
 - (d) Blade velocity.

⑥ In which of the following turbine inlet wheel velocity and Inlet jet velocity are equal in magnitude?

- ☒ (a) pelton turbine (b) propeller turbine
(c) Kaplan turbine (d) Francis turbine.

⑦ The work done per unit weight of water jet striking runner blades of pelton turbine is given by expression —

☒ (a) $[V_{w1} + V_{w2}] \frac{u}{g}$

(b) $V_{w1} \times \frac{u}{g}$

(c) $[V_{w1} + V_{w2}] g$

(d) $[V_{w1} + V_{w2}] u$

⑧ The expression for water power in pelton wheel is —

☒ (a) $\rho \times Q \times g \times H$ (kw)

(b) $(g \times Q \times H \times \rho)$ kw

(c) $(g \times Q)$ kw

(d) $(g \times H)$ kw

⑨ The hydraulic efficiency of pelton turbine will be maximum when blade velocity equal to —

☒ (a) $\frac{V}{2}$ (b) $\frac{V}{3}$ (c) $\frac{V}{4}$ (d) $\frac{V}{5}$

⑩ In pelton turbine — is defined as the ratio between power developed delivered by runner and power supplied at inlet of turbine.

⇒ (C) Hydraulic efficiency

⑪ The maximum hydraulic efficiency of pelton turbine is —

(a) 80 %

(b) 70 %

☒ (c) 50 %

(d) 88 %

⑫ In Pelton turbine product of mechanical efficiency and hydraulic efficiency is known as _____

~~and~~ overall efficiency.

⑬ In pelton turbine _____ is ratio of volume of water actually striking the runner and volume of water supplied by jet as Mechanical efficiency ~~or~~ Volumetric efficiency.

a) Hydraulic efficiency d) overall efficiency.

⑭ Among the following which turbine has least efficiency?

☒ a) Pelton turbine

b) Kaplan turbine

c) Francis turbine

d) propeller turbine.

⑮ The expression for maximum hydraulic efficiency of pelton turbine is given by _____

☒ a) $\frac{1+k \cos \beta_2}{2}$

⑯ The ratio of pitch diameter of pelton wheel 'D' to diameter of jet 'd' is known as _____

a) speed ratio

☒ b) jet ratio

c) velocity ratio

d) Co-efficient of Velocity.

⑰ Find the diameter of runner wheel 'D' if jet ratio 'm' and diameter of jet 'd' are given as 10 and 125 mm.

Ans:- $M = 10 \text{ m.}$

$d = 0.125 \text{ m}$

$$m = \frac{D}{d} \Rightarrow 10 = \frac{D}{0.125}$$

$$\Rightarrow D = 1.25$$

☒ a) 1.25 m.

18) If diameter of jet is 85 mm and diameter of runner is 1.5 m, then calculate width of bucket —

Ans $d = 85 \text{ mm}$

$D = 1.5 \text{ m}$

width of bucket $= 5d$

$= 5 \times 85 = 425 \text{ mm}$

d) 425 mm

19) If diameter of jet is 85 mm and diameter of runner is 1.5 m, then depth of bucket is —

$D = 1.5 \text{ m}$

$d = 85 \text{ mm}$

depth of bucket $= 1.2d$

$= 1.2 \times 85 \text{ mm}$

$= 102 \text{ mm}$

d) 102 mm

20) If diameter of jet is 85 mm and diameter of runner is 1.5 m, then calculate number of buckets on pelton wheel approximately

$d = 85 \text{ mm} = 85 \times 10^{-3} \text{ m}$

$D = 1.5 \text{ m}$

Number of bucket $= 15 + 0.5M$

$M = \frac{D}{d} = \frac{1.5}{85 \times 10^{-3}} = 17.64$

No. of bucket $\Rightarrow 15 + 0.5 \times 17.64 = 23.82$

Ans 23

21) Main part of radial flow reaction turbine are —

(a) Casing

(b) Guide mechanism

(c) Draft tube

d) All of the mentioned.

22) Discharge through radial flow reaction turbine is —

(a) $\pi d_1 \times b_1 \times V_{t1}$

(b) $\pi d_2 \times b_2 \times V_{t2}$

(c) $\pi d_1 \times b_2 \times V_{t2}$

☒ (d) Both $\pi d_1 \times b_1 \times V_{t1}$ and $\pi d_2 \times b_2 \times V_{t2}$

23) Radial flow Reaction turbine contain spiral casing which area —

(a) Remains constant

☒ (b) Gradually decreases

(c) Gradually increases

(d) Suddenly decrease

24) The casing of radial flow Reaction turbine is made of spiral shape, so that water may enter the runner —

(a) Variable acceleration

(b) Constant acceleration

(c) Variable velocity

☒ (d) Constant velocity.

25) — allow the water to strike the vanes fixed on runner without shock at inlet

(a) Casing ☒ (b) Guide vane (c) Runner (d) Draft tube.

26) Runner blade are made up of —

☒ (a) Cast steel

(b) Cast iron

(c) Wrought iron

(d) steel.

27) The pressure at the exit of runner of Reaction turbine is generally — than atmospheric pressure.

(a) Greater

☒ (b) lesser

(c) Constant

(d) Equal

②8 — is a pipe of gradually increasing area used for discharging water from exit of the turbine to the tail race.
⇒ (a) Draft tube.

②9 — and — of radial flow reaction turbine are always full of water.

- ☒ (a) Casing and runner
(b) Casing and penstock
(c) Runner and penstock
(d) Runner and draft tube.

③0 — governs the flow of water entering the runner blades.
(a) Casing (c) Draft tube
(b) Guide vanes (d) Runner

☒ (a) Guide vanes
③1 spiral casing of Reaction turbine will regulate the flow?

- ☒ (a) True
(b) False

③2 Inward radial flow Reaction turbine is a turbine in which water flows across the blade of runner —

- (a) Radial direction ☒ (b) Radially inward
(c) Radially outward (d) Axial direction.

③3 Which of the following is inward radial flow reaction turbine?

- (a) Pelton turbine
☒ (b) Francis turbine
(c) Axial turbine
(d) Kaplan turbine

③4 What is the water flow direction in the runner in a Francis turbine?

- ⇒ ☒ (a) Radial and then axial.

- 35 Which of the following is true in case of flow of water before it enters the runner of a francis turbine.
- (a) Available head is entirely converted to velocity head.
 - (b) " " " " " " to pressure head.
 - (c) " " " " " " neither converted to pressure head nor velocity head.
 - (d) Available head is partly converted into pressure head and partly velocity head.

37 Why does the cross-sectional area of spiral casing gradually decreases along the circumference of the francis turbine from the entrance to the top?

→ (a) To ensure constant velocity of water during runner entry.

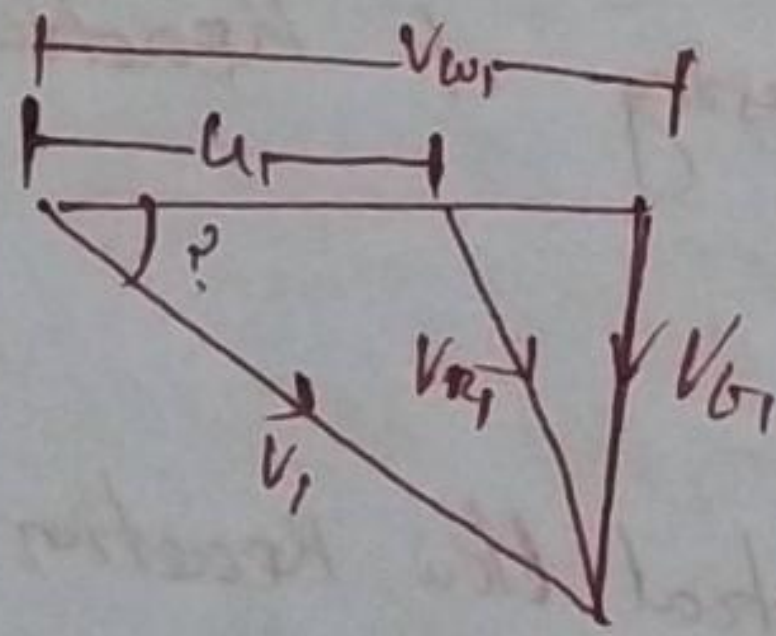
38 In the figure shown below, which of the following angle replace the question mark?

(a) Guide ~~vane~~^{vane} angle at inlet.

(b) Blade angle at inlet.

(c) Vane angle at inlet.

(d) Blade angle at outlet.



39 Which of the following efficiency of francis turbine is described as the ratio between the power produce by runner to the power supplied by water at the inlet?

→ (a) Hydraulic efficiency.

40 Which of the following efficiency of francis turbine is defined as the ratio of between the power available at the shaft of the turbine to the power produce by the runner.

(a) Mechanical efficiency.

41 Which of the following efficiency of francis turbine is defined as the ratio between the power available to the shaft to the power supplied by water at the inlet?

(a) Overall efficiency

Q42) The whirl velocity at inlet of francis turbine is given to be 20 m/s . The blade velocity is given as 35 m/s . What is the hydraulic efficiency for a head of 100 m ?

$$\eta_{\text{hyd}} = \frac{fQ \cdot V_w \cdot u_1}{fQ \cdot g \cdot H} = \frac{20 \times 35}{9.81 \times 100} = 0.71 = 71\%$$

Ans 71%

Q43) The desired hydraulic efficiency of a turbine is 80% at a whirl velocity 20 m/s and head 100 m . What should be the blade velocity of the turbine at inlet in m/s ?

$$\eta_{\text{hyd}} = \frac{V_w \cdot u_1}{g \cdot H}$$

Ans $\Rightarrow 40 \text{ m/s}$

$$\Rightarrow 0.8 = \frac{20 \times u_1}{9.81 \times 100}$$

$$\Rightarrow 0.8 = 0.02 \times u_1$$

$$\Rightarrow u_1 = \frac{0.8}{0.02} = 40 \text{ m/s}$$

Q44) The input water power of the francis turbine is 1.25 times the runner power. What would be the hydraulic efficiency of the turbine?

(a) 60 (b) 70 (c) 80 (d) 90

$$\frac{100}{125} = 0.8 = 80\%$$

Q45) The volume flow rate into a francis turbine is $0.25 \text{ m}^3/\text{s}$. What is the volume of water that does not flow over the runner blade. What is the mechanical efficiency of the turbine.

(a) 65

(b) 75

(c) 80

(d) mechanical efficiency cannot be found out from the given information.

PUMP

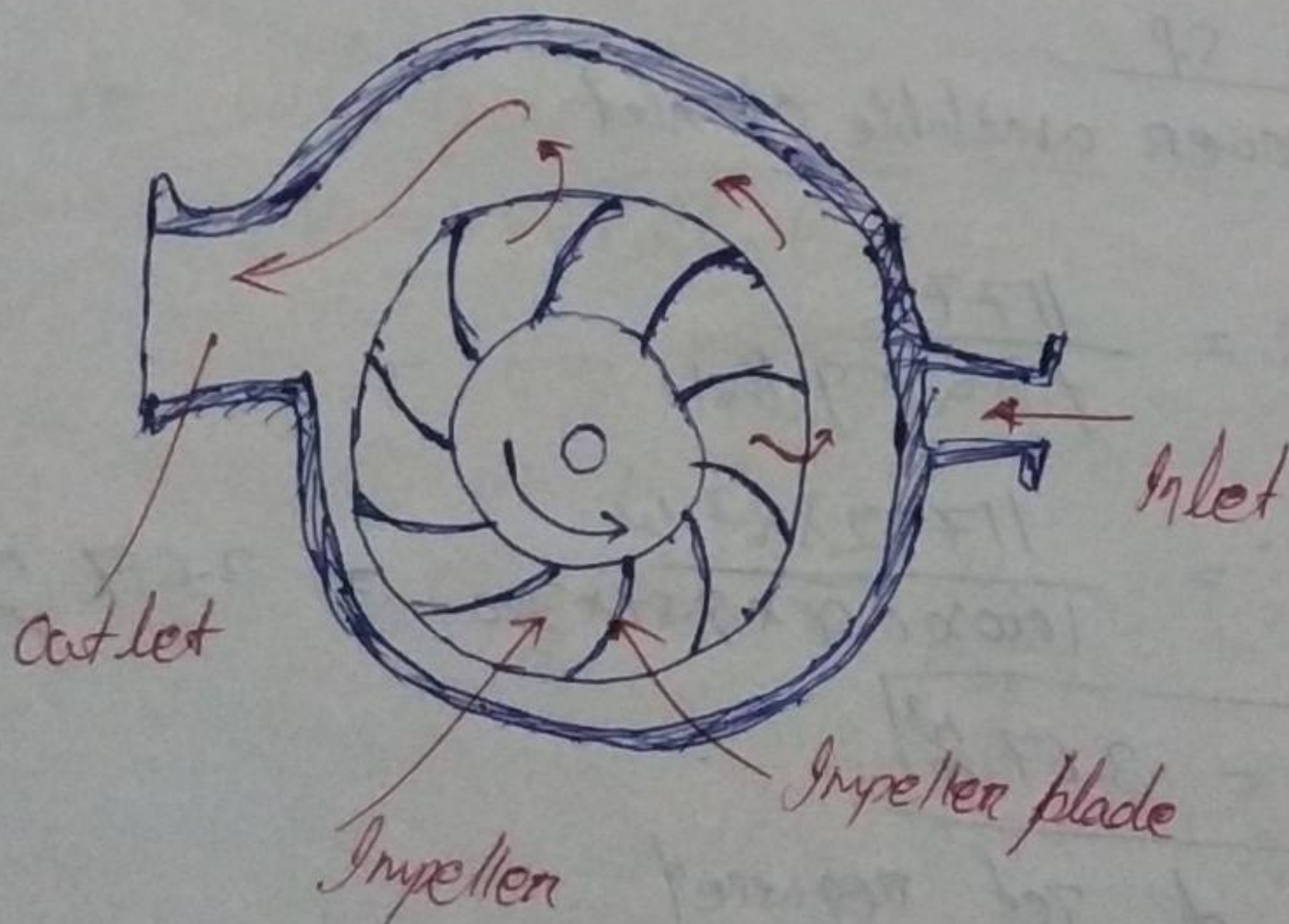
* what is pump?

→ A hydrodynamic pump machine is a device which convert the mechanical energy held by a device into potential and kinetic energy in fluid.

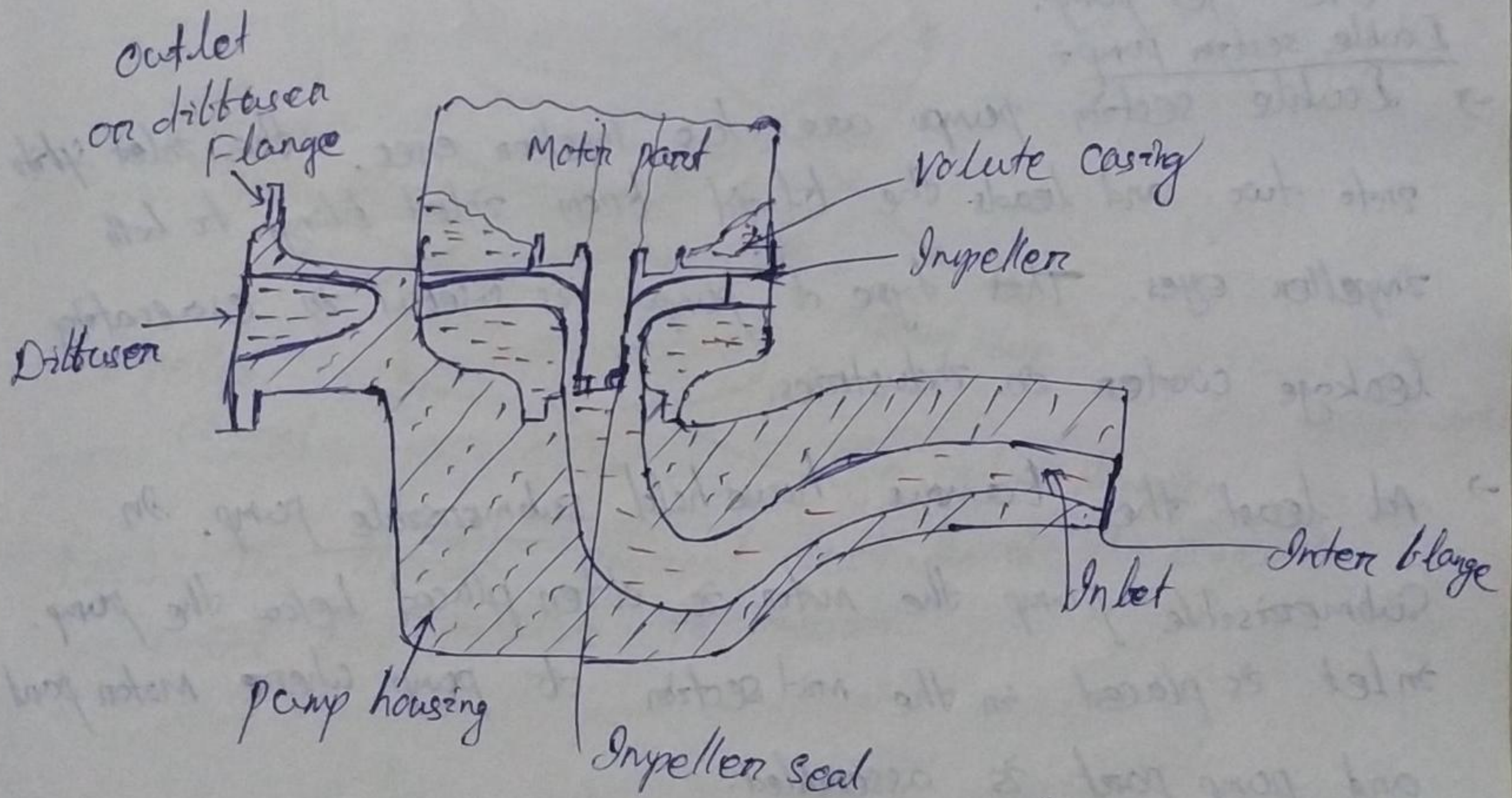
Centrifugal pump

→ I welcome you to the pride of mechanical engineers 'The centrifugal pump'. Starting from our home to all industries. For a burning field to power plant we will not find a place where there are no centrifugal pump.

→ From beginner point of view at first we must acquainted with shape, size, construction, working principle of centrifugal pump so let us begin with



→ Mechanical energy (torque) is transferred from motor shaft to Impeller. The Impeller thus gains high centrifugal force and hence by velocity and in technical term high kinetic energy. The kinetic energy is transferred to pressure and this pressure rise drives the fluid through the system.



Motor portion of assembly is not seen here.

→ Inlet guides the fluid to impeller eye. The design of inlet depends on pump type. The most common type of inlet at inlet are.

- ① Inline pump
- ② End suction pump.
- ③ Double suction pump.
- ④ Submersible pump.

→ In line pump are constructed to be mounted on a straight pipe. hence the name inline. The inlet & suction leads the fluid into impeller eye.

→ End suction pump have a very short and straight inlet section just below the eye of impeller. you can also tell impeller eye is placed in continuation to inlet blange.

exit jet pump.

Double section pump

→ Double section pumps are two impeller eyes. the inlet splits into two and leads the fluid from inlet blange to both impeller eyes. This type of pump is useful in evacuating leakage water in industries.

→ At least the famous household submersible pump. In submersible pump the motor is often placed below the pump. inlet is placed in the mid section of pump where motor part and pump part is assembled.

→ The design as inlet aims at creating a uniform velocity profile into the impeller.

Impeller

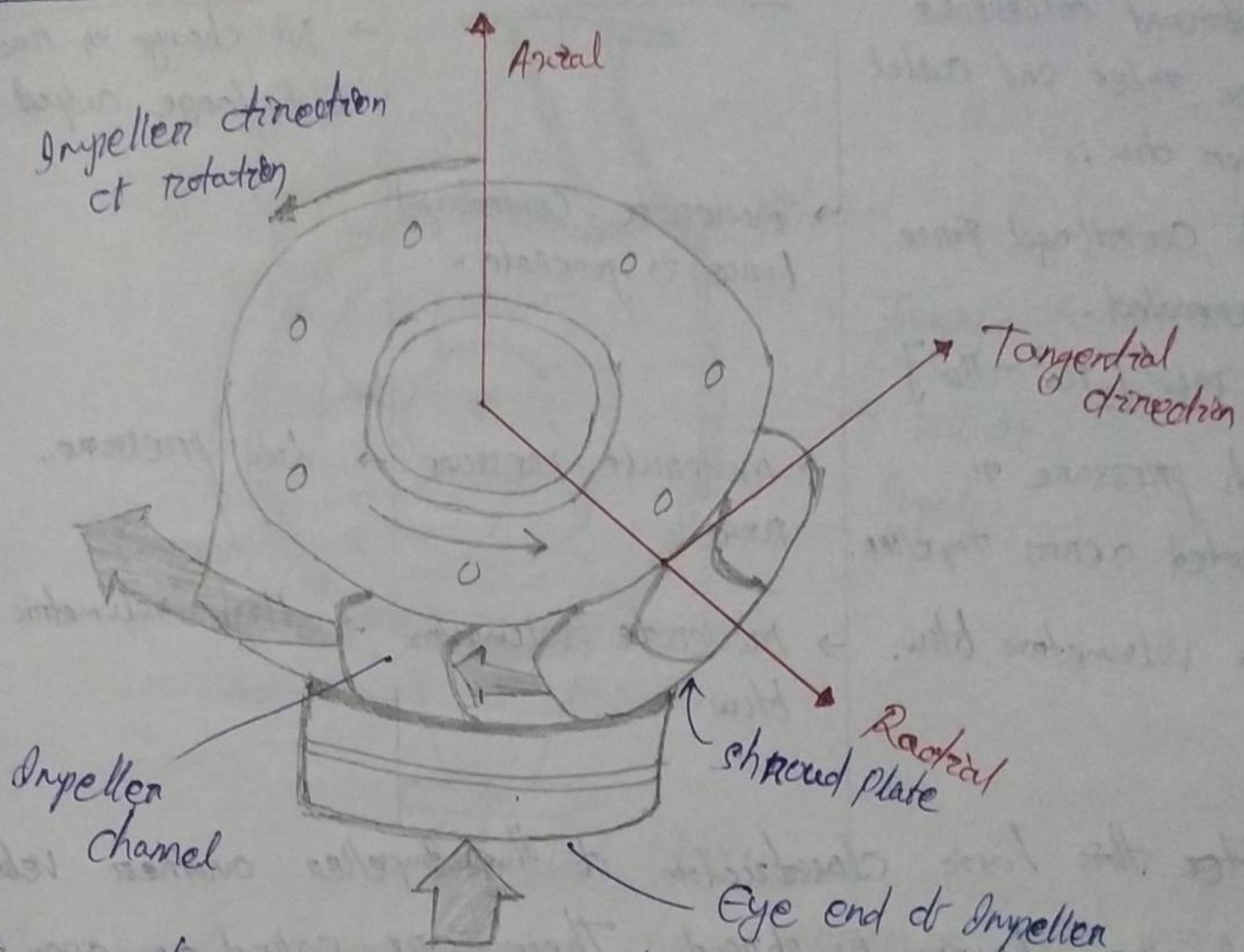
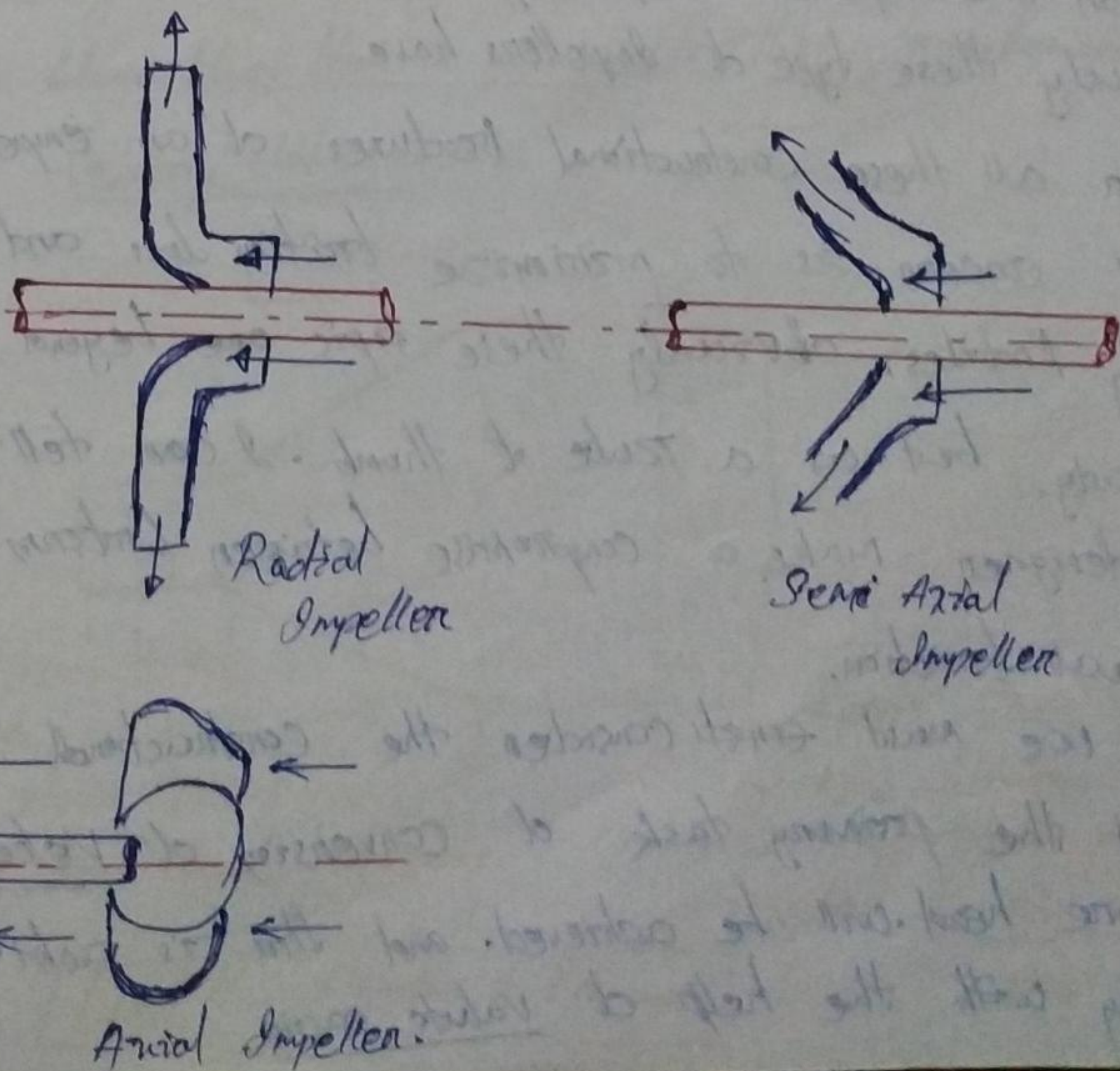


Fig 3: The impeller component definition of the direction and flow relative to impeller.



→ Radial Impeller

- Significant difference between inlet and outlet impeller dia.
- High Centrifugal Force is generated.
$$F_c = m\omega^2 [r_o^2 - r_i^2]$$
- High pressure is generated across impeller.
- Low volumetric flow.

Semi axial Impeller

- moderate centrifugal force is generated.
- Moderate pressure rise.
- moderate volumetric flow.

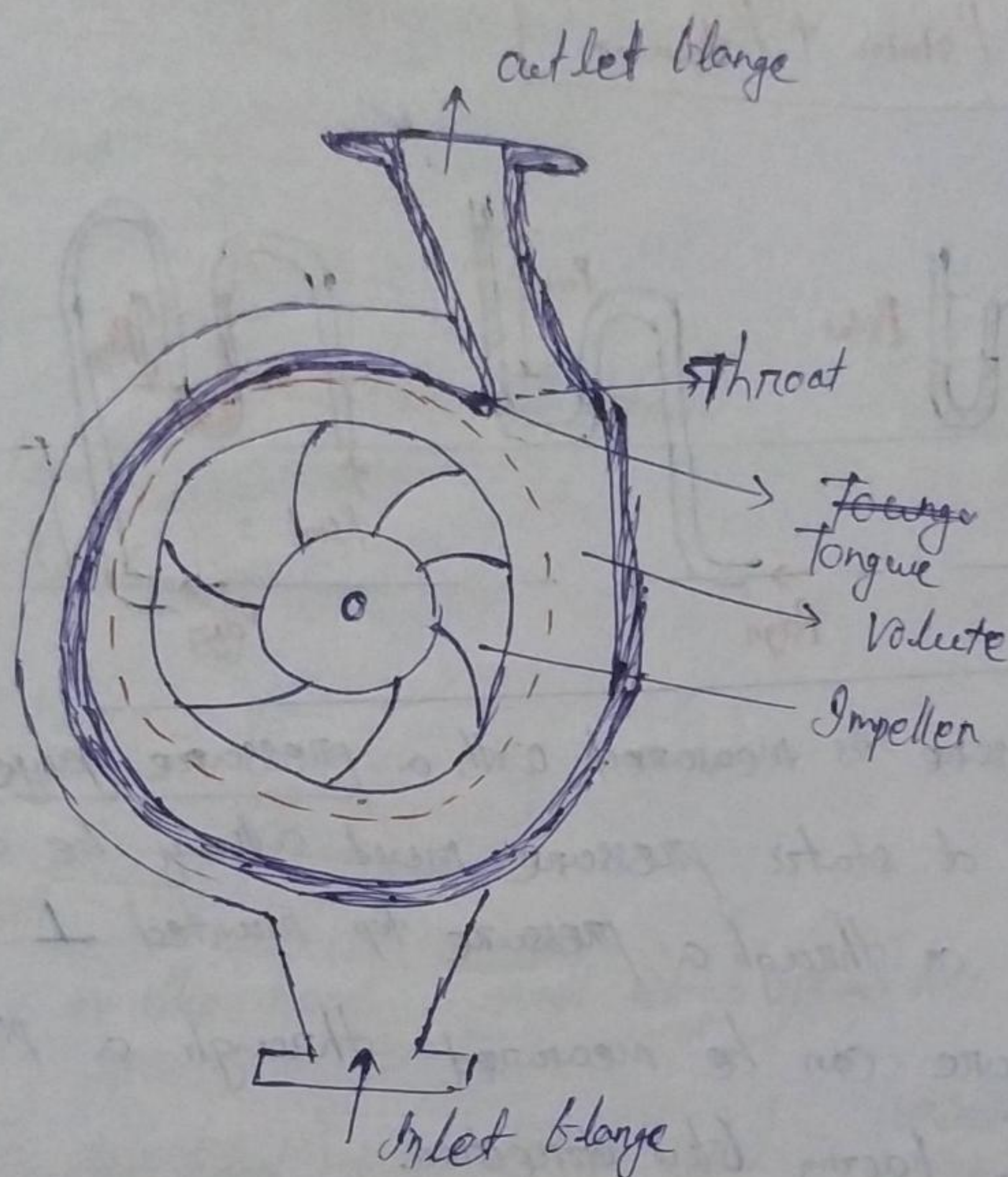
Axial Impeller

- No change in radial dia but large output with.
- Low pressure.
- High volumetric flow.

* After this basic classification of the impeller another velocity of impellers having no shroud. These are called as open impellers you can choose an open impeller where it is necessary to clean the impeller frequently and risk to blocking, obviously these type of impellers have.

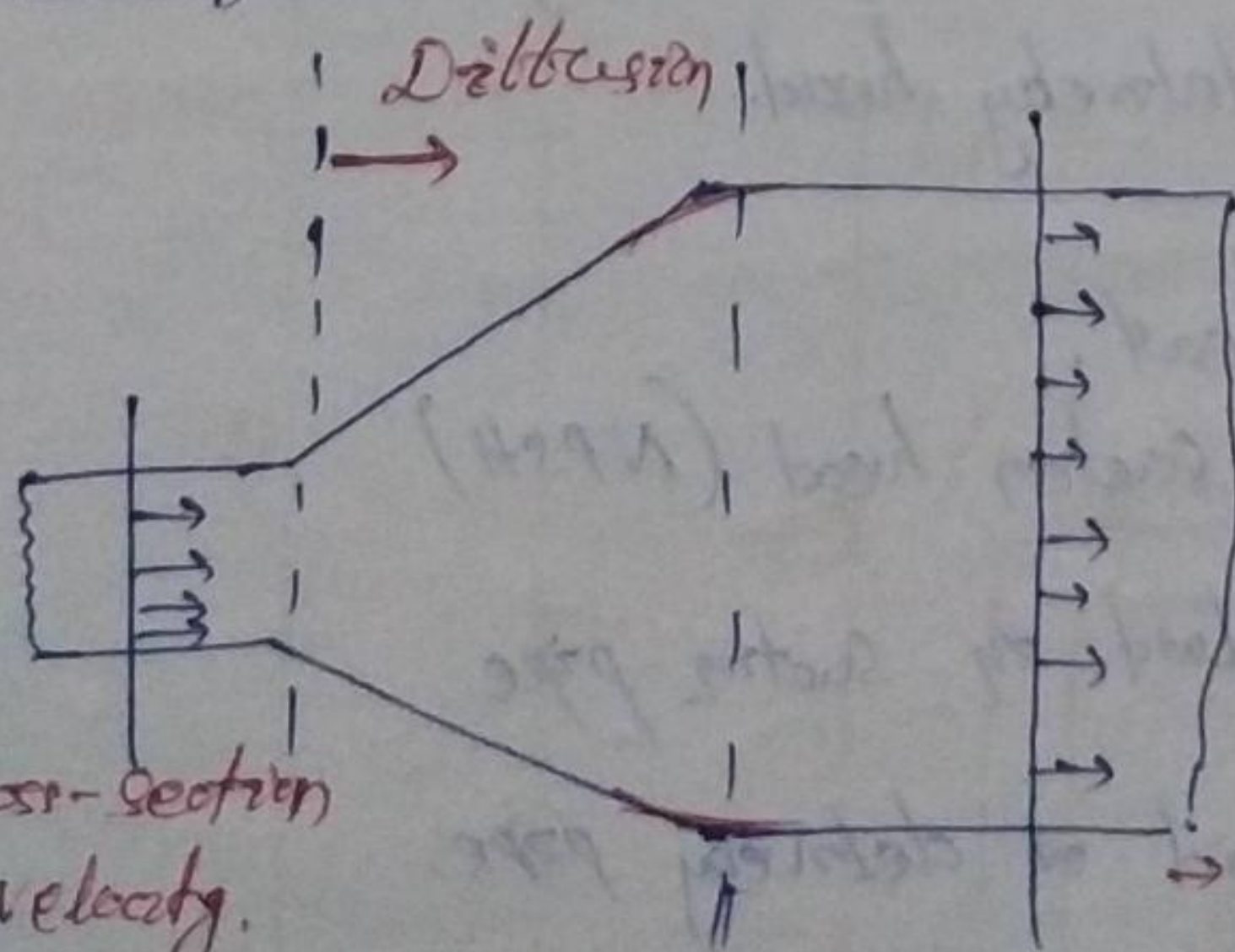
* After all these constructional features of an impeller one biggest concern is to minimize friction loss and non uniform velocity profiles, obviously these topics are beyond our scope of study, but as a rule of thumb, I can tell you the designer make a compromise between uniform velocity and wall friction.

* Now we must ~~not~~ consider the constructional features so that the primary task of conversion of velocity head to pressure head will be achieved, and this is achieved solely with the help of vanes.



→ The volute casing converts dynamic pressure rise on the impeller to static pressure.

→ The velocity is gradually reduced when cross sectional area of fluid flow is increased. This transformation is called velocity diffusion.

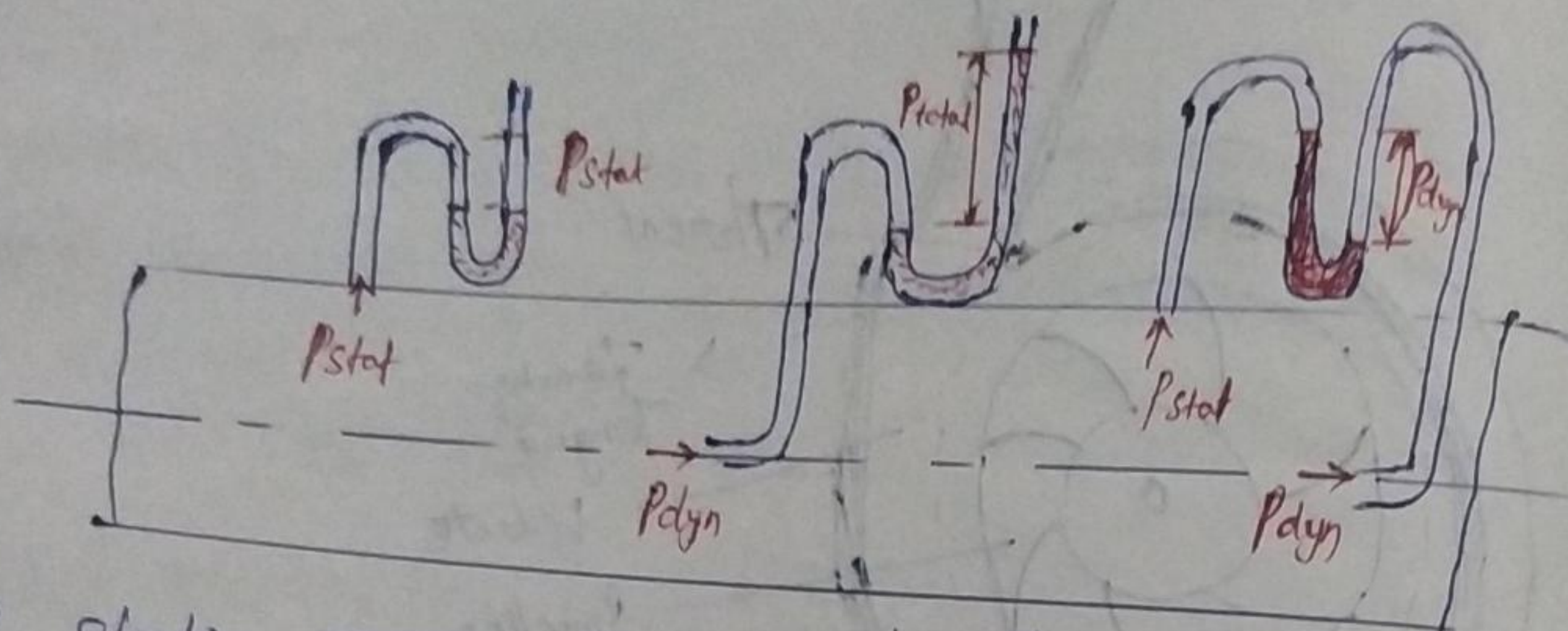


- Small cross-section
- High velocity.
- Low static pressure
- High dynamic pressure.

- Large cross section
- Low velocity
- High static pressure
- Low dynamic pressure.

$$P_{\text{total}} = P_{\text{static}} + P_{\text{dynamic}}$$

$$P_{total} = P_{static} + P_{dynamic}$$



- Static pressure is measured with a pressure gauge and the measurement of static pressure must always be done in static fluid. or through a pressure tap mounted $\perp$ to flow direction.
- Total pressure can be measured through a pressure tap with opening facing flow direction.
- The dynamic pressure can be found measuring pressure difference between total pressure and static pressure.
Ex → pitot tube.

Head of pump

→ In our calculation we use 4 categories of head:

- ① Suction and delivery head.
- ② Total head.
- ③ Manometric head.
- ④ Net positive suction head (NPSH)

V_s = Velocity of fluid in suction pipe

V_d = Velocity of fluid in delivery pipe.

h_s = Suction lift.

h_d = delivery lift.

Total static or vertical lift = $h_s + h_d$

1) suction and delivery head +

$$H_s = h_s + h_{fs} + \frac{V_s^2}{2g}$$

$$= \frac{P_s}{\gamma} + \frac{V_s^2}{2g} \quad \left(\begin{array}{l} \text{(static head)} \\ \text{(dynamic head)} \end{array} \right)$$

$$H_d = h_d + h_{fd} + \frac{V_d^2}{2g}$$

$$= \frac{P_d}{\gamma} + \frac{V_d^2}{2g}$$

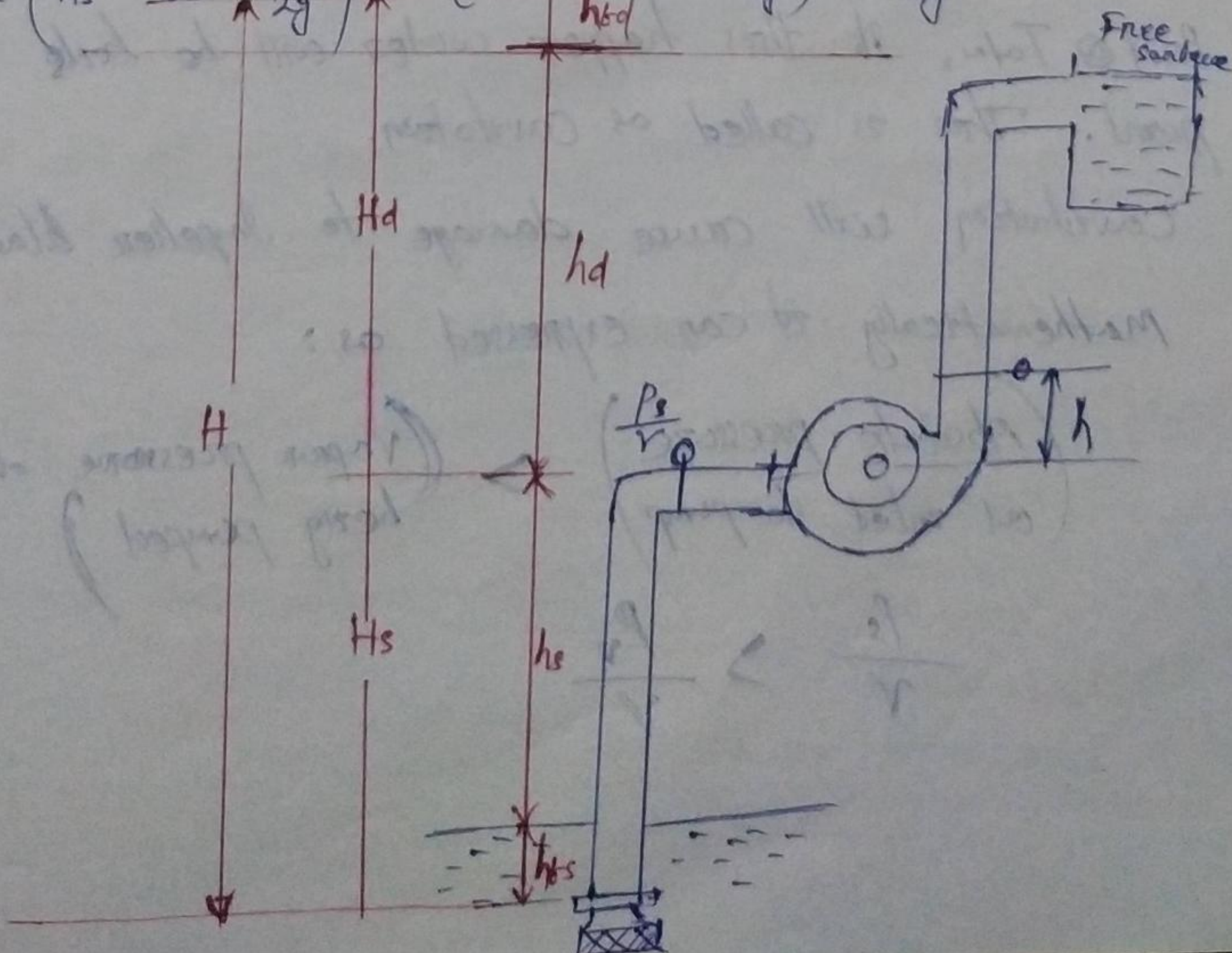
2) Total head + (H)

→ total head is the head against which pump has to work.

$$H = (\text{Suction head}) + (\text{Delivery head}) - (\text{velocity head in suction pipe})$$

$$H = \left(\frac{P_s}{\gamma} + \frac{V_s^2}{2g} \right) + \left(\frac{P_d}{\gamma} + \frac{V_d^2}{2g} \right) - \frac{V_s^2}{2g}$$

$$= \left(h_s + h_{fs} + \frac{V_s^2}{2g} \right) + \left(h_d + h_{fd} + \frac{V_d^2}{2g} \right) - \frac{V_s^2}{2g}$$



3/ Monometric head (HM)

→ monometric head is the difference of total energy at inlet and exit at pump.

$$\text{Fluid energy at suction} = \frac{P_s}{\rho} + \frac{V_s^2}{2g}$$

$$\text{Fluid energy at delivery flange} = \frac{P_d}{\rho} + \frac{V_d^2}{2g} + h$$

h = difference in elevation of two pressure gauge.

$$HM = \left(\frac{P_d}{\rho} + \frac{V_d^2}{2g} + h \right) - \left(\frac{P_s}{\rho} + \frac{V_s^2}{2g} \right) = \frac{P_d}{\rho} - \frac{P_s}{\rho}$$

4/ Net positive suction head (NPSH) :-

→ In the ~~system~~ ~~pipe~~ suction pipe, there is always negative or vacuum pressure at eye and entry of impeller blade.

→ This vacuum is inevitable otherwise fluid will not be sucked from atmosphere.

→ But in no case the vacuum will coincide with $P_{sat} @ T_{atm}$, if this happens water will be boiled at that point. This is called as cavitation.

→ Cavitation will cause damage to impeller blades.

Mathematically it can be expressed as:

$$\left(\text{Absolute pressure at inlet to pump} \right) > \left(\text{Vapor pressure of liquid being pumped} \right)$$

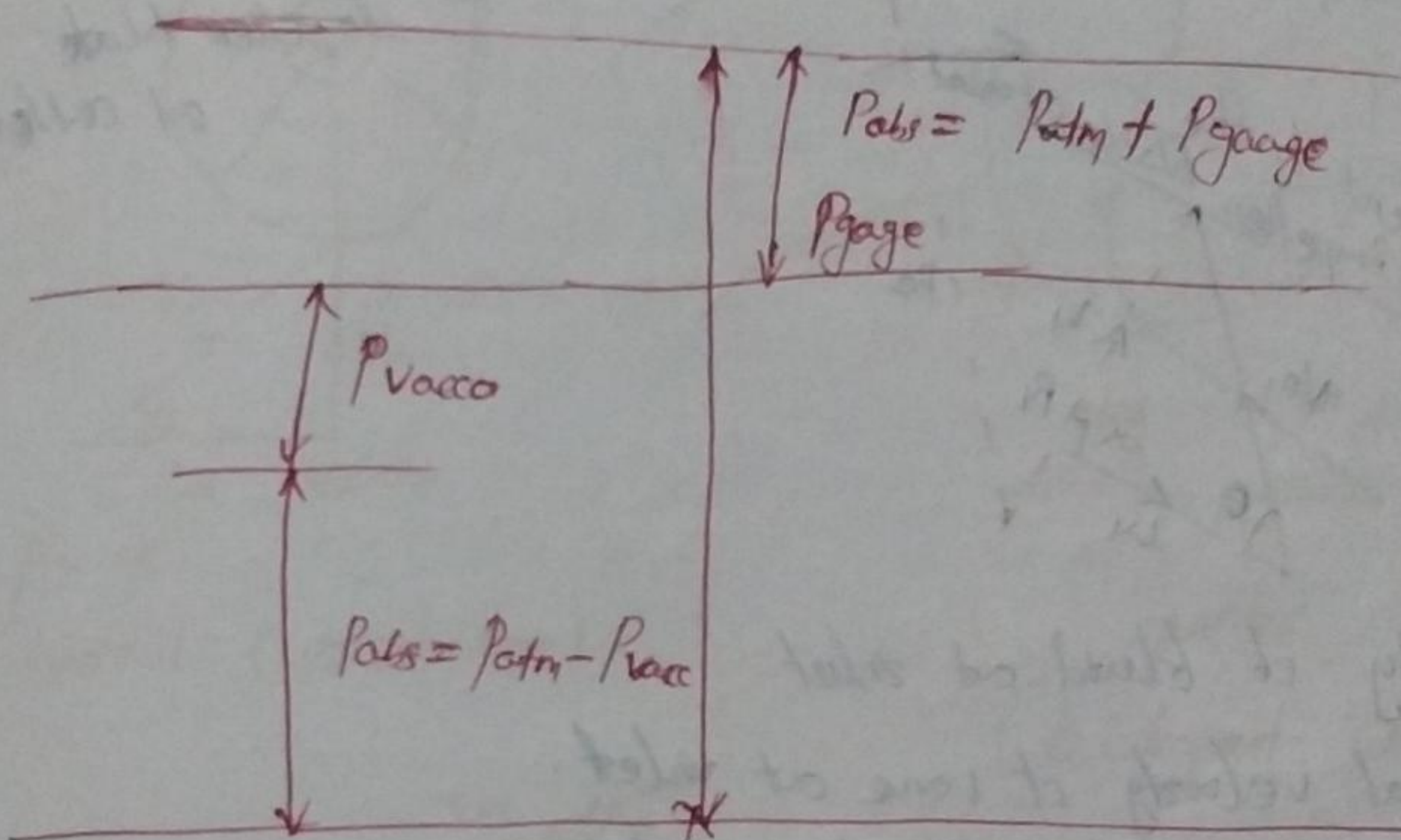
$$\frac{P_s}{\rho} > \frac{P_v}{\rho}$$

since absolute pressure in suction pipe is below $P_{atm} \left(\frac{P_s}{\rho} \right)$

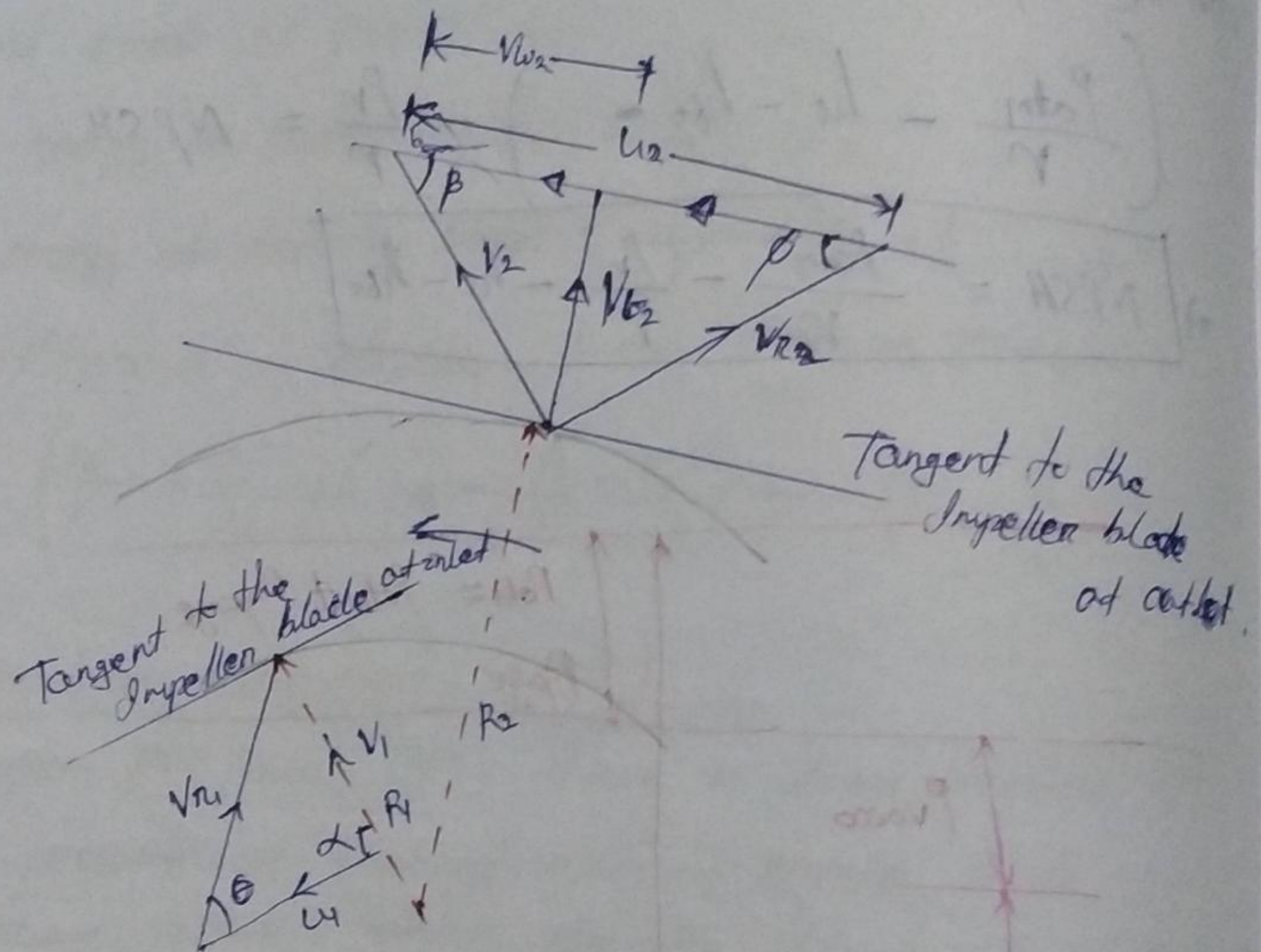
can be expressed as

$$\left(\frac{P_{atm}}{\rho} - h_s - h_{fs} - \frac{P_r}{\rho} \right) = NPSH$$

$$\Rightarrow \boxed{NPSH = \frac{P_{atm}}{\rho} - \frac{P_r}{\rho} - h_s - h_{fs}}$$



Velocity triangle +



V_1 = velocity of fluid at inlet

u_1 = Tangential velocity of vane at inlet.

V_{r1} = relative velocity of fluid at inlet.

α = Angle between fluid direction and V_1 .

θ = Vane angle at inlet = Angle made by the V_{r1} and direction of motion of the blade at inlet.

V_2 = velocity of fluid at outlet.

u_2 = Tangential velocity of vane at outlet.

V_{r2} = relative velocity at outlet.

β = Angle between the fluid direction and vane motion at outlet.

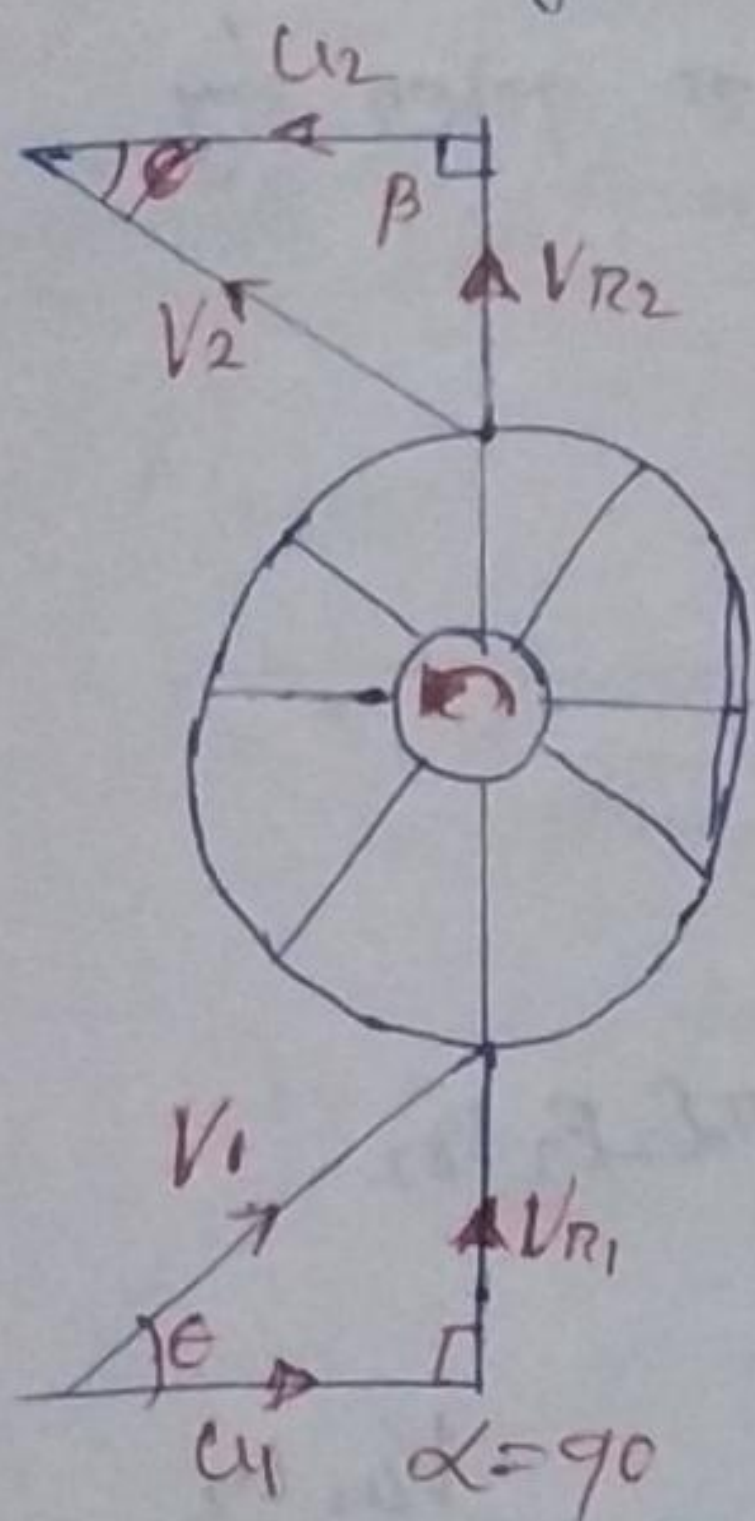
ϕ = Vane angle at outlet.

V_{t1} = Flow component of velocity at inlet.

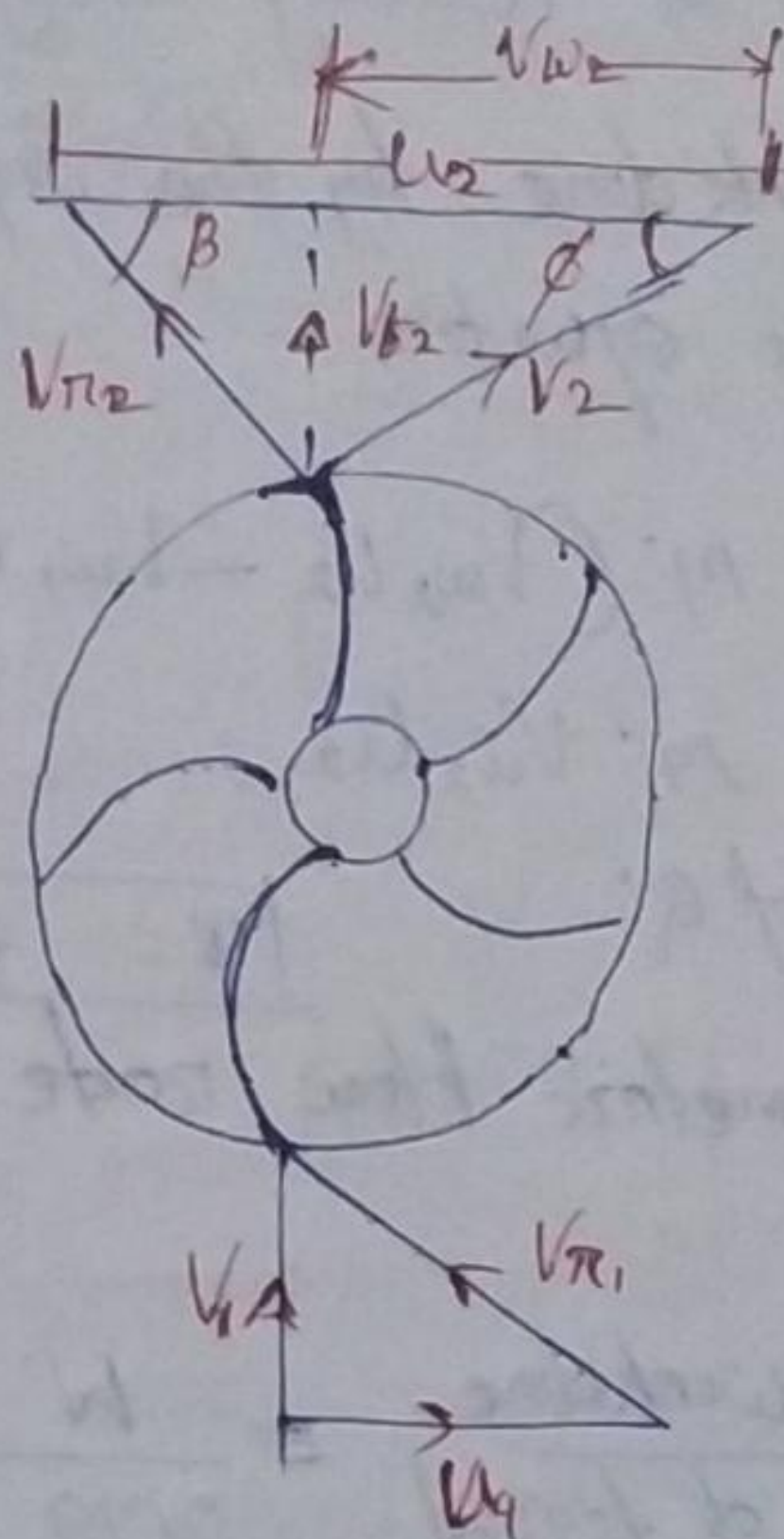
V_{t2} = Flow component of velocity at outlet.

V_{w1} = Velocity of whirl at inlet

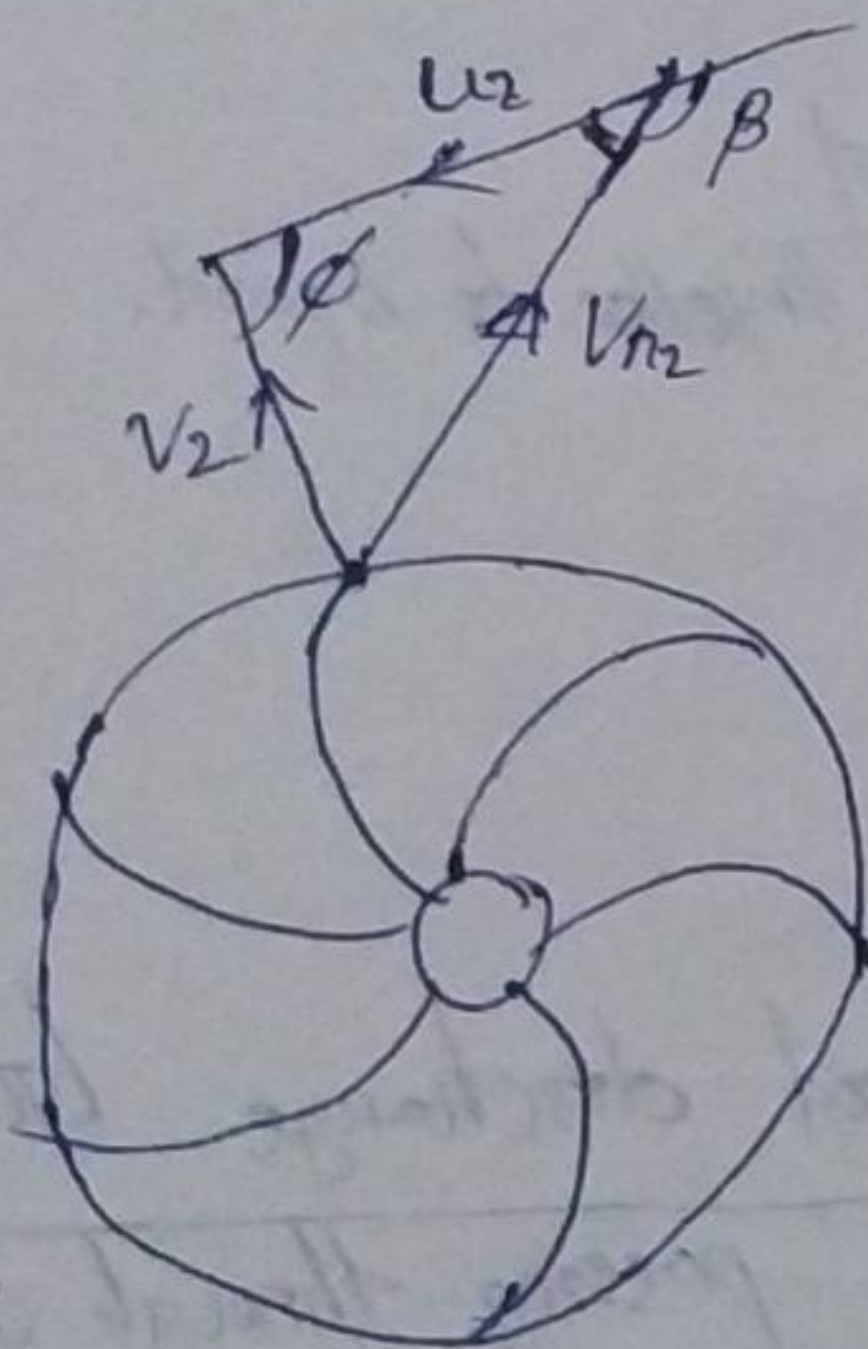
V_{w2} = Velocity of whirl at outlet.



$\beta = 90^\circ$
(Radial curved vane)



$\beta < 90^\circ$
(Backward curved vane)



$\beta > 90^\circ$
(Forward curved vane)

Work done by the centrifugal pump:-

- Centrifugal pump is a reverse of radially inward flow reaction turbine.
- As shown in velocity diagram $V_{w1} = 0$
- Rate of work done by the Impeller on liquid is given by Euler's equation

$$W = m \cdot (V_{w2} u_2 - V_{w1} u_1)$$

$$W = m \cdot V_{w2} u_2$$

$$m = \rho Q$$

$$W = \rho Q \cdot V_{w2} u_2$$

where volumetric flow rate $Q = \pi D_1 B_1 V_{f1} = \pi D_2 B_2 V_{f2}$

$$\frac{\text{Rate of work done}}{\text{weight of liquid}} = \frac{W}{mg} = \frac{m \cdot V_{w2} u_2}{m \cdot g} = \frac{V_{w2} u_2}{g}$$

* Efficiency of the centrifugal pump +
manometric efficiency:

$$\eta_{\text{manometric}} = \frac{\text{Manometric head}}{\text{Head imparted by Impeller to liquid.}}$$

$$= \frac{H_m}{\frac{V_{w2} u_2}{g}}$$

Volumetric efficiency:

$$\eta_{\text{volumetric}} = \frac{\text{Quantity of liquid discharge from pump}}{\text{Quantity of liquid passing through Impeller.}}$$

$$= \frac{Q}{Q + \Delta Q}$$

Mechanical efficiency +

$$\eta_{\text{mech}} = \frac{\text{Energy available at impeller}}{\text{Energy supplied to shaft.}}$$

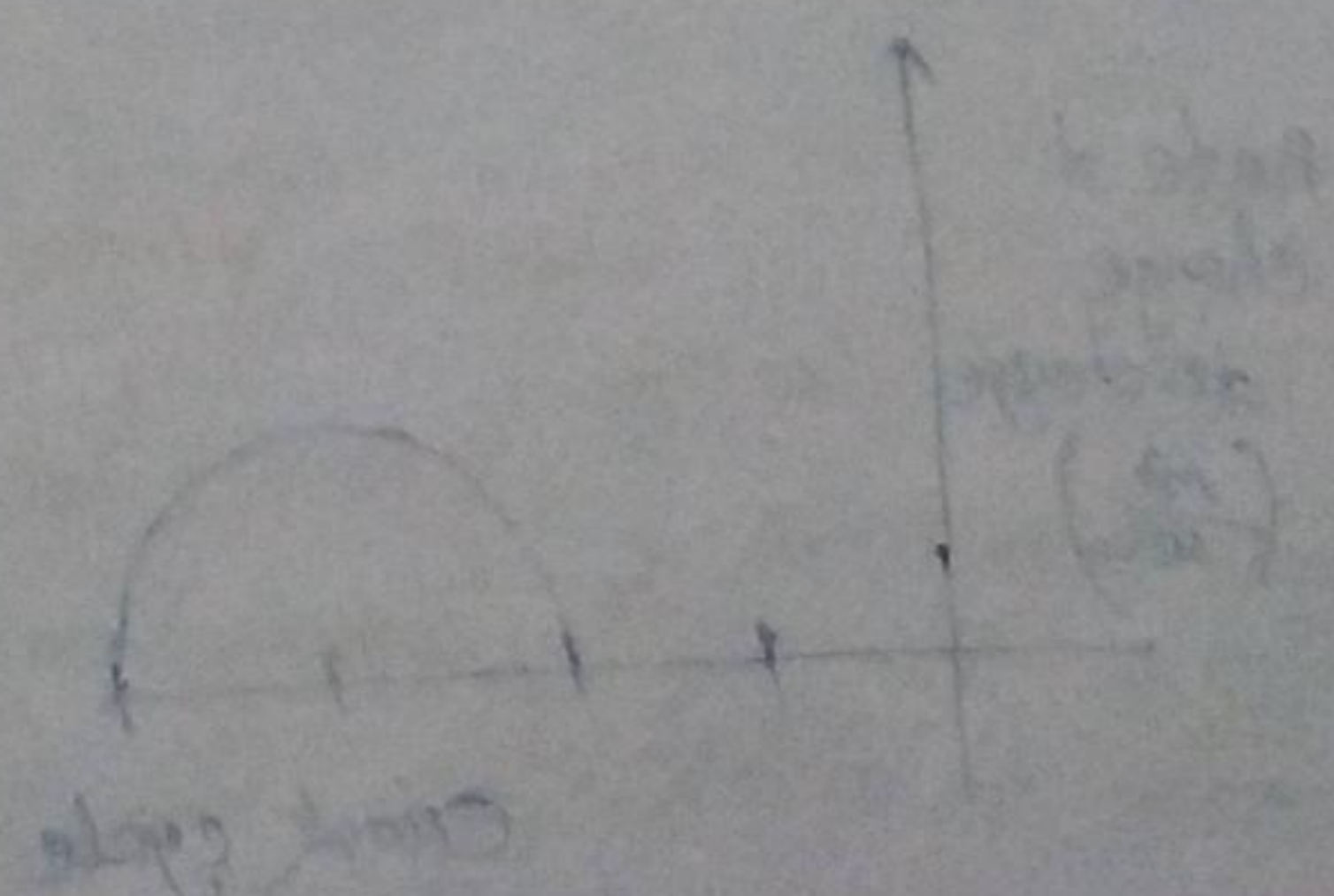
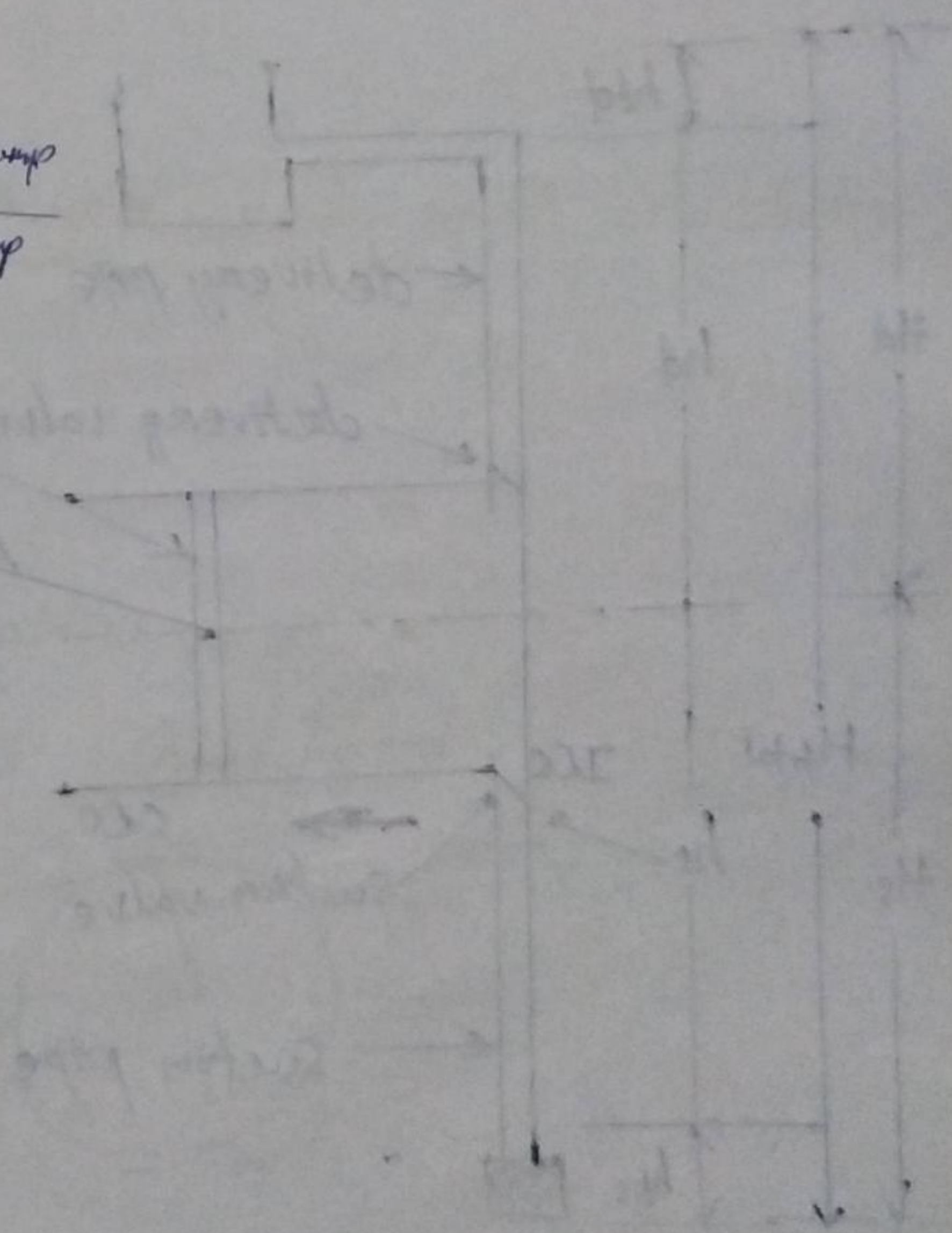
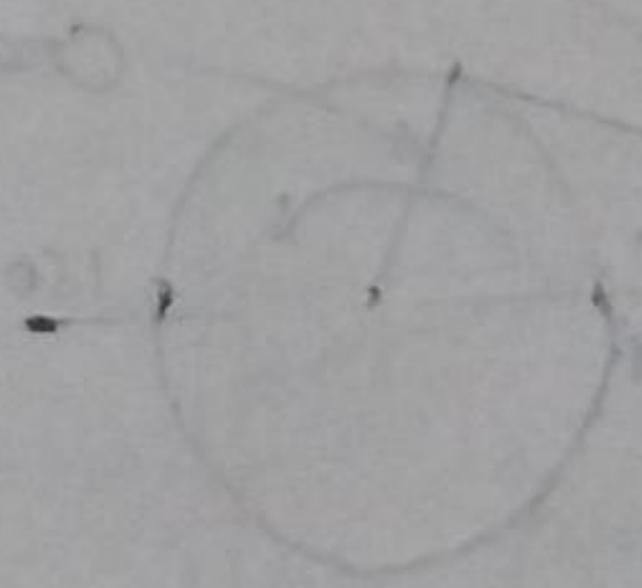
$$= \frac{f Q V w_2 u_2}{\text{power supplied to shaft}}$$

overall efficiency +

$$\eta_{\text{overall}} = \eta_{\text{mano}} \times \eta_{\text{mech}}$$

$$\eta_o = \frac{\text{power output from the pump}}{\text{power input to the pump}}$$

$$\eta_o = \frac{f \rho g H_m}{P}$$



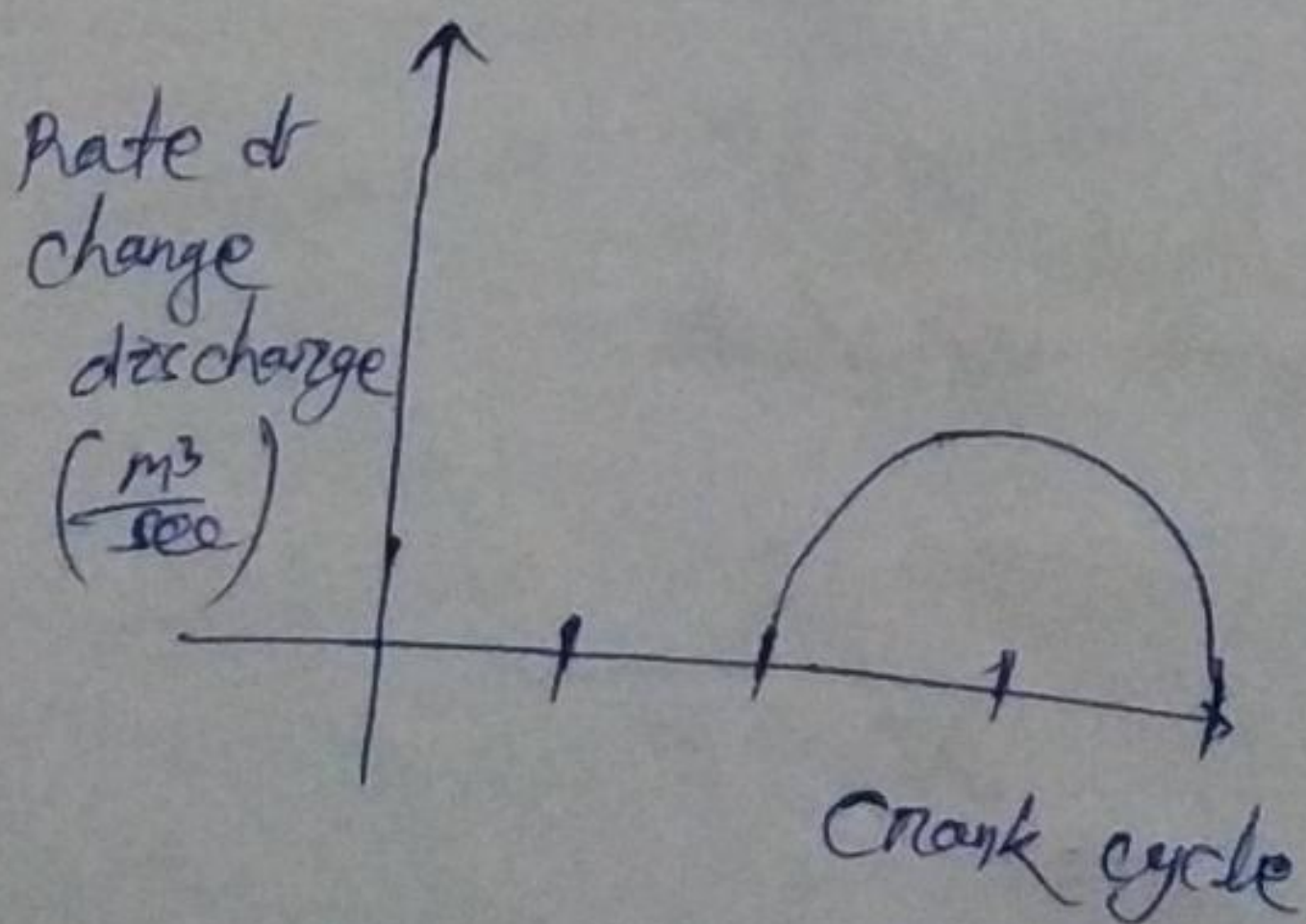
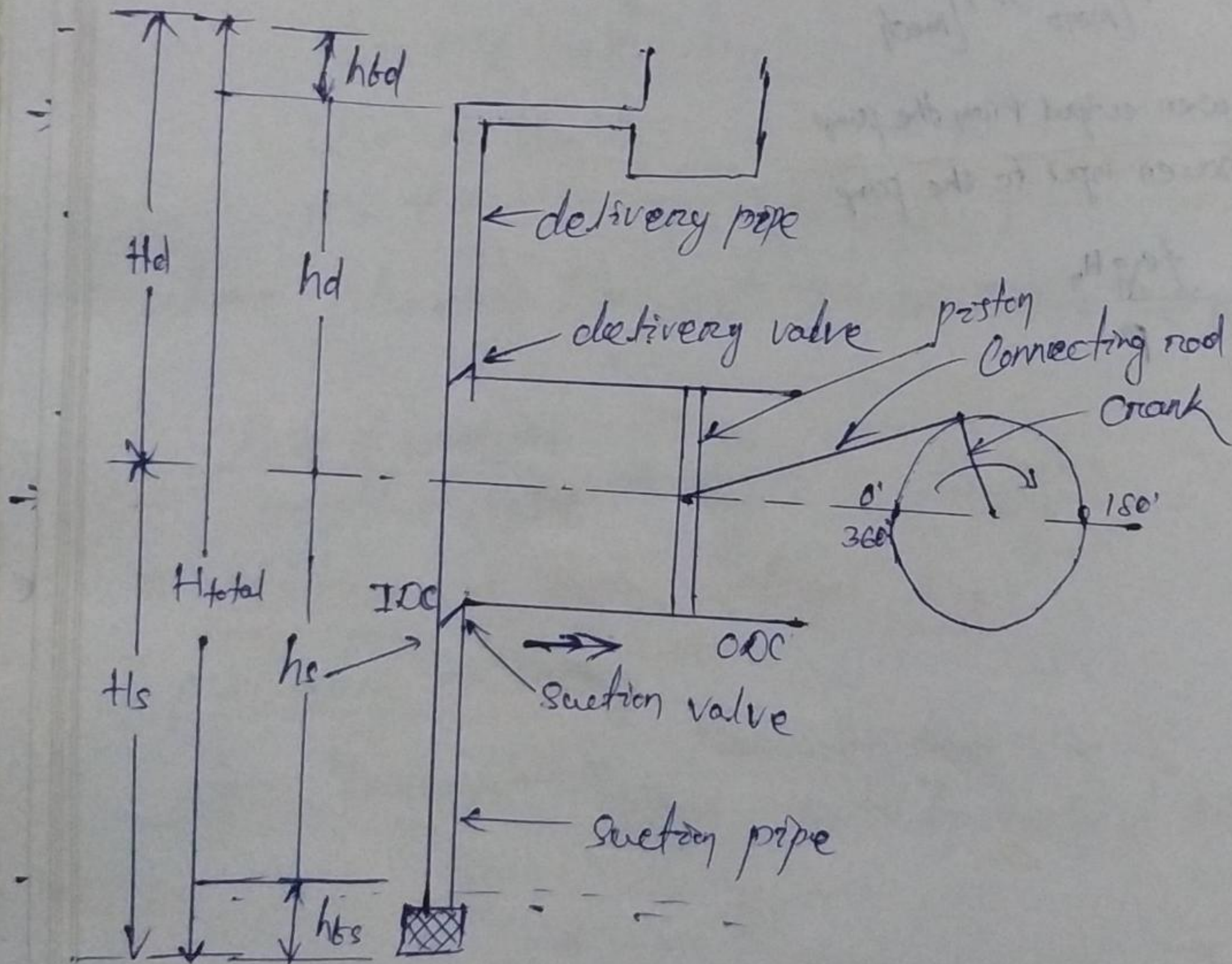
3

Reciprocating pump

→ It is a positive displacement pump.

→ Important feature — High pressure rise.
Low pressure rise.

Single acting reciprocating pump



suction Head + (H_s)

$$H_s = h_s + h_{fs} + \frac{v_s^2}{2g}$$

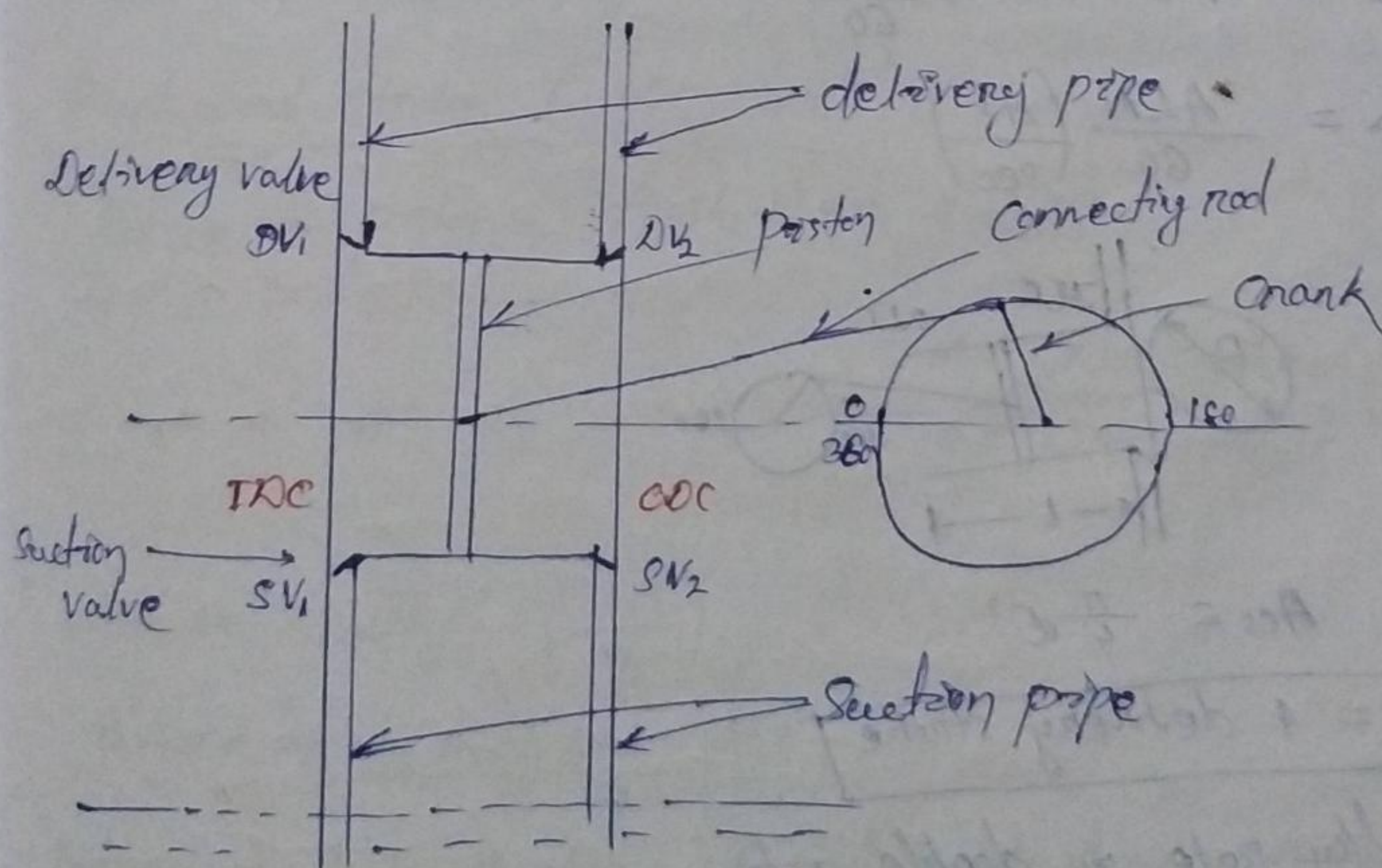
h_s = suction lift

h_{fs} = head loss due to friction in suction pipe.

delivery head (H_d)

$$H_d = h_d + h_{fd} + \frac{v_d^2}{2g}$$

* Double acting reciprocating pump:-



* Requirement of power to drive pump +

- Motor power (electrical power)
 - Brake power (mechanical power)
 - power developed by pump
- } Mechanical loss
} Hydraulic loss

Power developed by pump

$$\text{Power (P) (Watt)} = f \left(\frac{\text{kg}}{\text{m}^3} \right) \times g \left(\frac{\text{m}}{\text{sec}} \right) \times Q \left(\frac{\text{m}^3}{\text{sec}} \right) \times (H_s + H_d) \text{ (m)}$$

$$\left(\frac{\text{Joule}}{\text{sec}} \right)$$

$$P = f Q g H_{\text{total}}$$

$$H_{\text{total}} = h_s + h_{fs} + \frac{V_s^2}{2g} + h_d + h_{fd} + \frac{V_d^2}{2g}$$

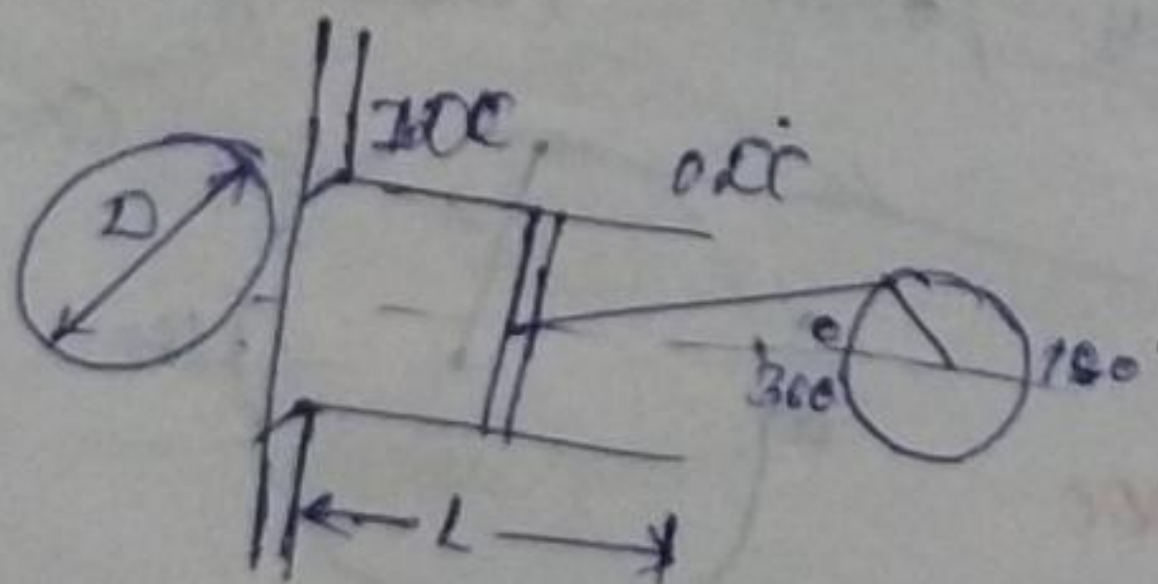
$$\approx 2 \text{ m/s}$$

Hydraulic power developed by reciprocating pump

Volumetric flow rate (discharge) for a single acting:

$$Q_{sa} = A \text{ (m}^2\text{)} \times L \text{ (m)} \times \frac{N}{60}$$

$$Q_{sa} = \frac{A L N}{60} \left(\frac{\text{m}^3}{\text{sec}} \right)$$



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

$$[1 \text{ rev} = 1 \text{ delivery stroke}]$$

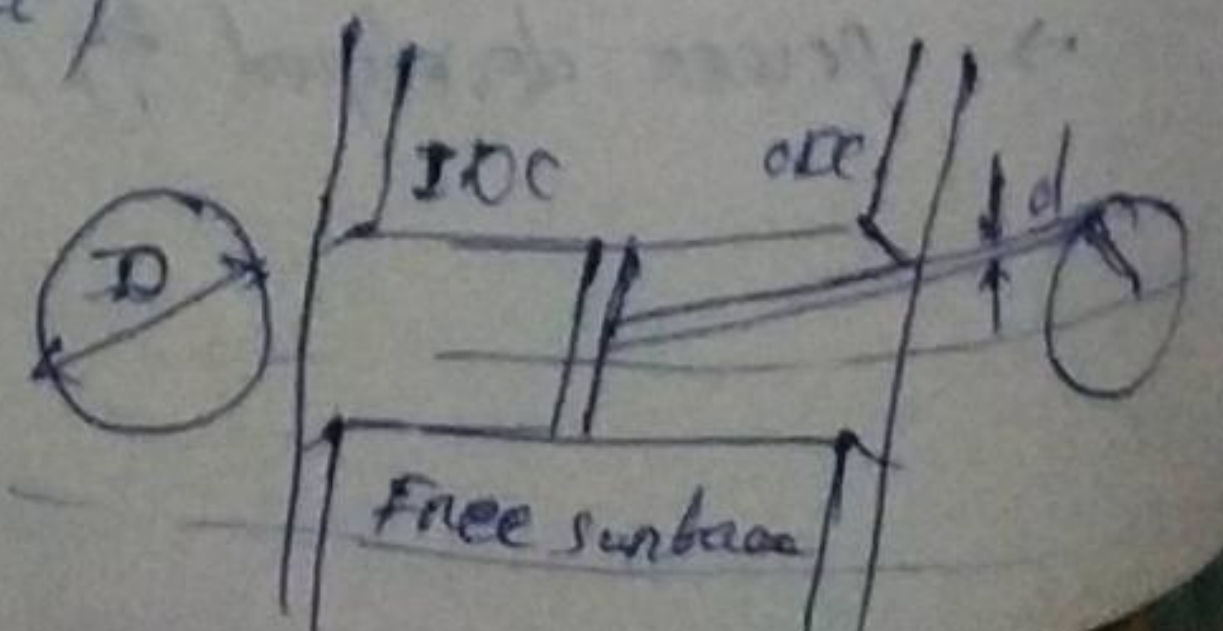
Volumetric flow rate in double acting

$$Q_{th,da} = (A - a) \text{ (m}^2\text{)} \times L \times \frac{N}{60} \times 2$$

$$Q_{th,da} = \frac{2(A - a) \times L \times N}{60} \left(\frac{\text{m}^3}{\text{sec}} \right)$$

where (D) = diameter of piston

(d) = diameter of piston rod



force acting on piston +

Force
(single acting)

Force on piston
(Double acting)

Force on piston single acting +

$$\begin{aligned} \text{Force on piston}_{SA} &= P \times A \\ &= f \left(\frac{\text{kg}}{\text{m}^3} \right) \times g \left(\frac{\text{m}}{\text{sec}^2} \right) \times H_{\text{total}} (\text{m}) \times A \end{aligned}$$

Double acting during (forward stroke (DCC → IDC))

$$\text{Force on piston} = f g A h_d + f g (A-a) h_s$$

Backward stroke (IDC → DCC)

$$\text{Force on piston} = f g A h_s + f g (A-a) h_d$$

A = Cross sectional area of piston
 a = Cross sectional area of piston rod.

$$\dot{Q}_{th, SA} = \frac{A L N}{60} \left(\frac{\text{m}^3}{\text{sec}} \right)$$

$$\dot{Q}_{th, DA} = \frac{2(A-a) L N}{60} \left(\frac{\text{m}^3}{\text{sec}} \right)$$

But in actual practice theoretical discharge Never happens.

→ This is due to the leakage along, → Valve.
→ piston ring.
→ glands.
→ Coupling.

→ Slip:- The difference between theoretical discharge and Actual discharge is called as slip.

Mathematically,

$$\text{slip} = Q_{th} - Q_{act} \left(\frac{m^3}{\text{sec}} \right)$$

percentage of slip =

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

Non dimensional slip =

A Non dimensional parameter Co-efficient of discharge (C_d)

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

$$C_d = 0 < C_d < 1 \quad (\text{Typically value of } 0.9)$$

Q.1

A single acting reciprocating pump has a bore of 25 cm and stroke 40 cm. runs 30 rpm. It discharge a water at a rate of $0.009 \frac{m^3}{\text{sec}}$. The suction and delivery head are 7 m and 15 m respectively. Determine (i) Theoretical discharge.

(ii) Slip

(iii) percentage slip.

(iv) Co-efficient of discharge

(v) Force required to work the piston during suction and delivery stroke

(vi) Ideal power required drive the pump.

Data given +

single acting

$$\text{bore } (D) = 25 \text{ cm} = 25 \times 10^{-2} \text{ m} = 0.25 \text{ m}$$

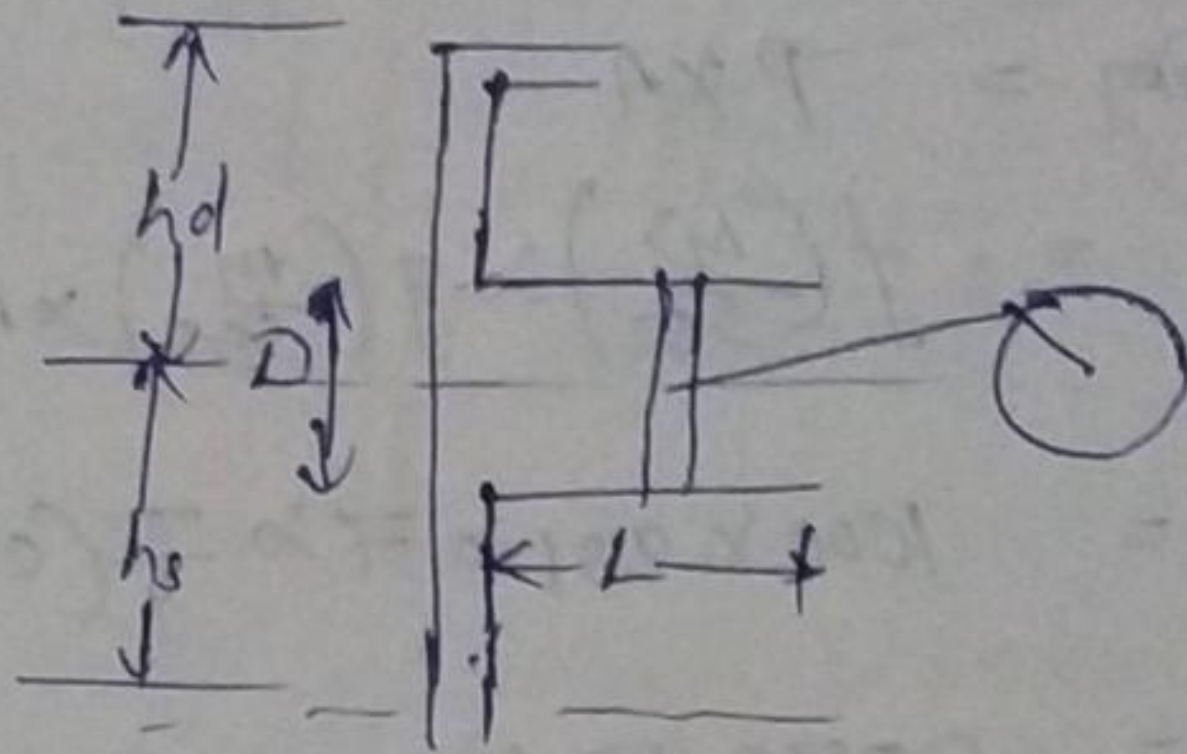
$$\text{stroke } (L) = 40 \text{ cm} = 0.4 \text{ m}$$

$$N = 30 \text{ rpm}$$

$$Q_{\text{act}} = 0.009 \frac{\text{m}^3}{\text{sec}}$$

$$h_s = 7 \text{ m}$$

$$h_d = 15 \text{ m}$$



To Find :

(i) $Q_{th} = \underline{\hspace{2cm}}$

(ii) $\text{slip} = \underline{\hspace{2cm}}$

(iii) $\% \text{ slip} = \underline{\hspace{2cm}}$

(iv) $C_d = \underline{\hspace{2cm}}$

(v) $F_{\text{friction}} = \underline{\hspace{2cm}}$

$F_{\text{delivery}} = \underline{\hspace{2cm}}$

(vi) $\text{power} = \underline{\hspace{2cm}}$

Analysis +

(i) Theoretical discharge

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

$$= \frac{\frac{\pi}{4} (0.25)^2 \times 0.4 \times 30}{60}$$

$$Q_{th} = \frac{9.81 \times 10^{-3}}{\frac{\text{m}^3}{\text{sec}}}$$

(ii) $\text{slip} = Q_{th} - Q_{\text{act}}$

$$= 9.81 \times 10^{-3} - 0.009$$

$$= 8.1 \times 10^{-4} \left(\frac{\text{m}^3}{\text{sec}} \right)$$

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

$$= \frac{8.1 \times 10^{-4}}{9.81 \times 10^{-3}} \times 100 = 8.25 \%$$

(iv) Co-efficient of discharge (C_d)

$$C_d = \frac{Q_{act}}{Q_{th}} = \frac{0.009}{9.81 \times 10^{-3}} = \underline{0.9}$$

(v) $F_{section} = P \times A$

$$= f \left(\frac{m^3}{sec} \right) \times g \left(\frac{m}{sec^2} \right) \times h_s (m) \times A$$

$$= 1000 \times 9.81 \times 7 \times \frac{\pi}{4} (0.25)^2$$

$$= \underline{3370.83 \text{ N}}$$

(vi) $F_{delivery} = f g h_d A$

$$= 1000 \times 9.81 \times 15 \times \frac{\pi}{4} (0.25)^2$$

$$= \underline{7223.20 \text{ N}}$$

(vii) power (P) = $f g A f g H_{total}$

$$H_{total} = (h_s + h_d)$$

$$= 7 + 15 = 22$$

$$P = 1000 \times 9.81 \times 10^{-3} \times 9.81 \times 22$$

$$P = \underline{2117.19 \text{ W}} = \underline{2.11 \text{ kW}}$$

② A single acting reciprocating pump having a bore of 20cm and stroke of 30cm runs at 50rpm. The pump is raising a liquid height of 20cm above the pump level at a rate of ~~20cm~~ $0.0078 \frac{m^3}{sec}$. The efficiency is 75%.

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

(ii) slip

(iii) Coefficient of discharge.

(iv) Ideal power

(v) Actual power consumption by the pump.

Data given =

A single acting

$$D = 200 \text{ mm} = 0.2 \text{ m}$$

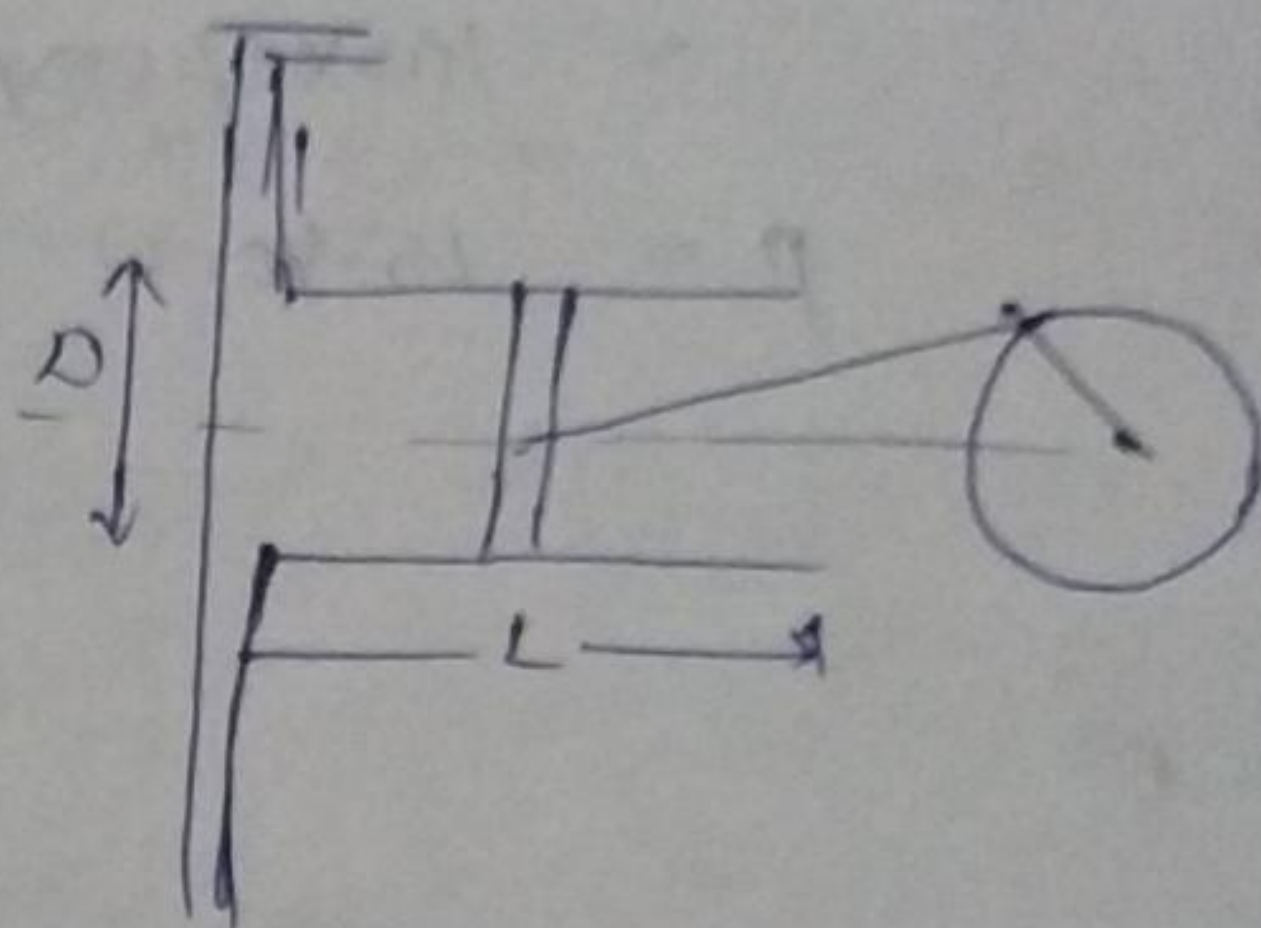
$$L = 300 \text{ mm} = 0.3 \text{ m}$$

$$N = 50 \text{ rpm}$$

$$H_{\text{total}} = 20 \text{ cm} = 0.2 \text{ m}$$

$$Q_{\text{act}} = 0.0078 \frac{\text{m}^3}{\text{s}}$$

$$\eta = 75\% = 0.75$$



To Find +

(i) $Q_{\text{th}} = \underline{\hspace{2cm}}$

(ii) $\text{slip} = \underline{\hspace{2cm}}$

(iii) $C_d = \underline{\hspace{2cm}}$

(iv) Ideal power = $\underline{\hspace{2cm}}$

(v) Actual power = $\underline{\hspace{2cm}}$

Analysis +

(i) $Q_{\text{th}} = \frac{A L N}{60}$

$$A = \frac{\pi}{4} (0.2)^2 = 0.0314 \text{ m}^2$$

$$Q_{\text{th}} = \frac{0.0314 \times 0.3 \times 50}{60} = 7.85 \times 10^{-3} \frac{\text{m}^3}{\text{sec}}$$

(ii) $\text{slip} = Q_{\text{th}} - Q_{\text{act}}$

$$= 7.85 \times 10^{-3} - 0.0078 = 5 \times 10^{-5} \frac{\text{m}^3}{\text{sec}}$$

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

$$= \frac{5 \times 10^{-5}}{7.8 \times 10^{-3}} \times 100 = 0.64\%$$

(iii)

$$C_d = \frac{Q_{\text{act}}}{Q_{\text{th}}} = \frac{0.0078}{7.85 \times 10^{-3}} = 0.99$$

$$(2v) \text{ power } p = f \times Q \times \rho \times g \times H_{\text{total}}$$

$$= 1000 \times 7.85 \times 10^{-3} \times 9.81 \times 0.2$$

$$p = 15.40 \text{ W}$$

- ③ A Double acting reciprocating pump run at 50 rpm.
 as a piston 30 cm and a piston rod 5 cm and a stroke
 40 cm. Such a suction and delivery are 4 m. and 14 m. respectively
 Determine (i) Theoretical discharge
 (ii) power required to drive the pump.

Data given

Double acting

$$\text{piston dia } (D) = 30 \text{ cm} = 0.3 \text{ m}$$

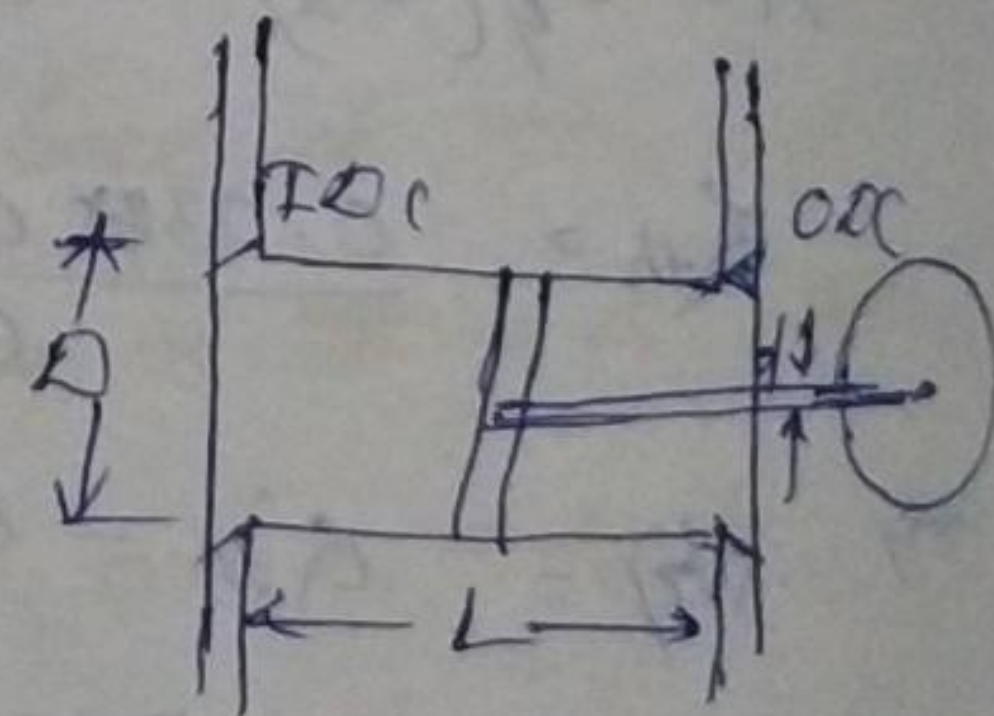
$$\text{piston rod } (d) = 5 \text{ cm} = 0.05 \text{ m}$$

$$L = 40 \text{ cm} = 0.4 \text{ m}$$

$$N = 50 \text{ rpm}$$

$$h_s = 4 \text{ m}$$

$$h_d = 14 \text{ m}$$



To Find

$$(i) Q_{th} =$$

$$(ii) \text{ power } p =$$

Analysis

$$(i) Q_{th} = \frac{2(A-a) \times L \times N}{60}$$

$$\text{piston Area } A = \frac{\pi}{4} (D)^2 = \frac{\pi}{4} (0.3)^2 = 0.07 \text{ m}^2$$

$$\text{piston rod area (a)} = \frac{\pi}{4} d^2 = \frac{\pi}{4} (0.05)^2 = 1.96 \times 10^{-3} \text{ m}^2$$

$$Q_{th} = \frac{2(0.07 - 1.96 \times 10^{-3}) \times 0.4 \times 50}{60}$$

$$Q_{th} = 0.045 \frac{\text{m}^3}{\text{sec}}$$

$$(ii) \text{ power (P)} = \rho \cdot Q_{th} \cdot g \cdot H_{total}$$

$$H_{total} = h_s + h_d = 4 + 14 = 18 \text{ m}$$

$$P = 1000 \times 9.81 \times 0.045 \times 18$$

$$P = 7946.1 \text{ W} = 7.94 \times 10^3 \text{ W} = \underline{\underline{7.94 \text{ kW}}}$$

Q.4 A double acting reciprocating pump runs at 45 rpm. & discharging $0.009 \frac{\text{m}^3}{\text{s}}$ of water. The pump has a stroke of 40 cm. The diameter of piston is 20 cm. The suction and delivery head are 3 m and 14 m respectively. Find the slip of the pump and power required to drive the pump. Neglect the piston rod area.

Data given:

A double acting

$$N = 45 \text{ rpm.}$$

$$Q_{act} = 0.009 \frac{\text{m}^3}{\text{sec}}$$

$$L = 40 \text{ cm} = 0.4 \text{ m}$$

$$D = 20 \text{ cm} = 0.2$$

$$h_s = 3 \text{ m}$$

$$h_d = 14 \text{ m.}$$

Neglect the piston rod area.

To Find:

$$\text{Slip} =$$

$$\text{power (P)} =$$

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

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

(piston rod area is neglected)

$$A = \frac{\pi}{4}(D)^2 = \frac{\pi}{4}(0.2)^2 = 0.031 \text{ m}^2$$

$$Q_{th} = \frac{2 \times 0.031 \times 0.4 \times 45}{60}$$

$$Q_{th} = 0.0186 \frac{\text{m}^3}{\text{sec}}$$

$$\text{Slip} = 0.0186 - 0.009 = 9.6 \times 10^{-3} \frac{\text{m}^3}{\text{sec}}$$

power required

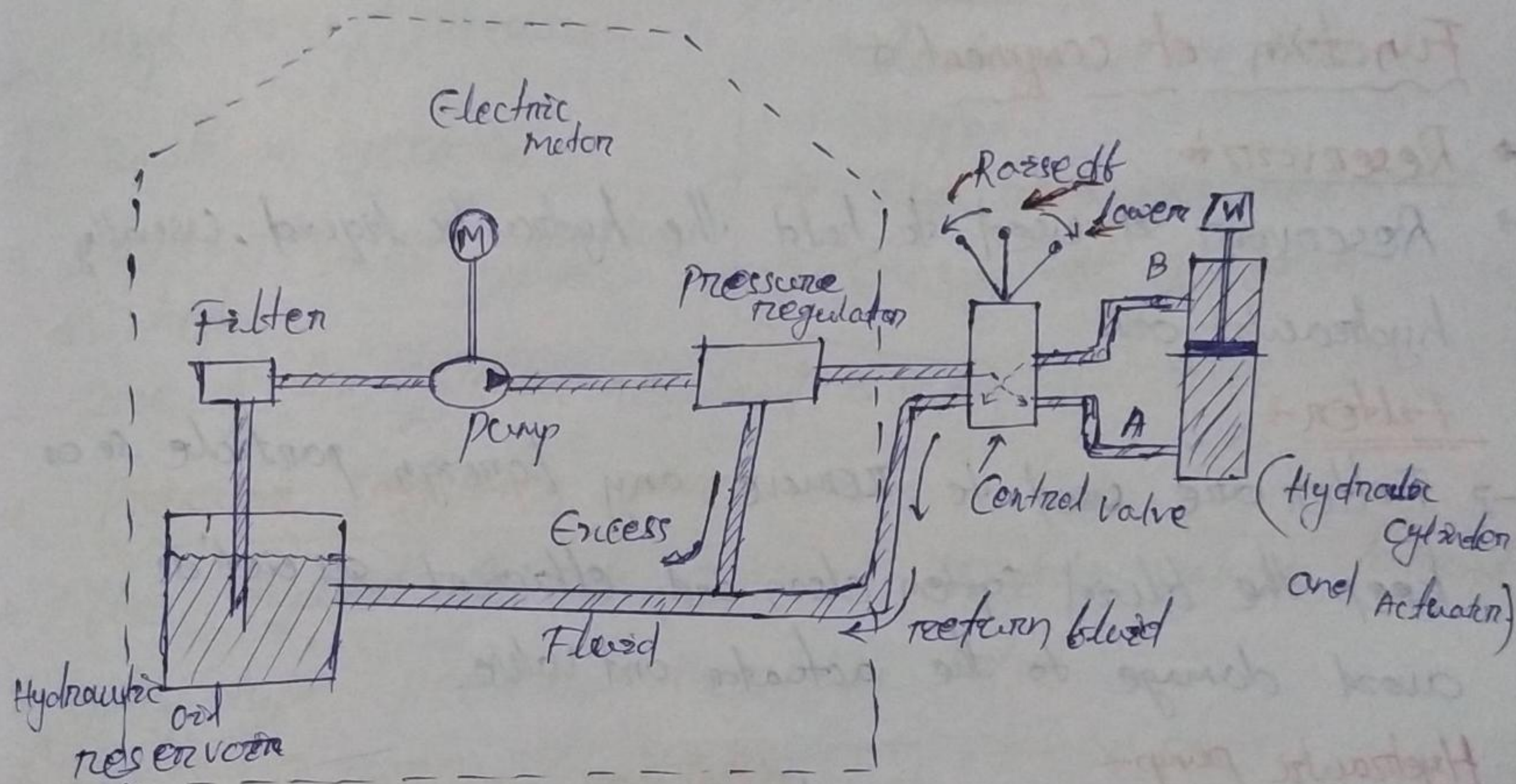
$$H_{total} = h_s + h_d = 3 + 14 = 17$$

$$P = \rho \times Q_{th} \times g \times H_{total}$$

$$= 1000 \times 0.0186 \times 9.81 \times 17$$

$$P = \underline{\underline{3.10 \text{ kW}}}$$

Hydraulic control system



Component common to many Hydraulic system.

Typical lay out of a common hydraulic system:

- Hydraulic fluid is pumped through a filter so that the No. of debris is communicated is hydraulic circuit.
- pump is running with the help of 3 ϕ induction motor.
- A hydraulic actuator is connected to system which can be Raise or lower with the help of a pressure control valve.
- The fluid is return to a reservoir after working in the actuator.
- Suppose the hydraulic actuator is under at a certain point. at the but pump pressure rises. the fluid. Ultimate fluid high pressure will generate in pipe line in burst.

→ To safeguard the hydraulic system a pressure regulation² valve is required which will bypass the excess fluid.

Function of component +

* Reservoir *

→ Reservoir is used to hold the hydraulic liquid, usually hydraulic oil.

Filter +

→ Filter are used to remove any foreign particle so as keep the fluid system clean and efficient. as well as avoid damage to the actuator and valve.

Hydraulic pump +

→ hydraulic pump is used to force the fluid from the reservoir to rest of the cylinder. hydraulic circuit by converting mechanical energy into hydraulic energy.

Pressure regulator +

→ Regulate (i.e. maintain) the required level of pressure in hydraulic fluid and redirect the excess fluid back to the storage tank.

Control valve +

→ It is used to control the direction, pressure and flow rate of the fluid flowing through the circuit.

Hydraulic actuator +

→ The hydraulic actuator is converted by the fluid power into mechanical power to do useful work.

Fluid +

→ It is used to generally high density incompressible oil.
piston systems
Carries the hydraulic oil from one place to another.

Merit

- High force / pressure
- Smooth in operation
- density of hydraulic fluid is high.
- size of the cylinder required is small

demerit

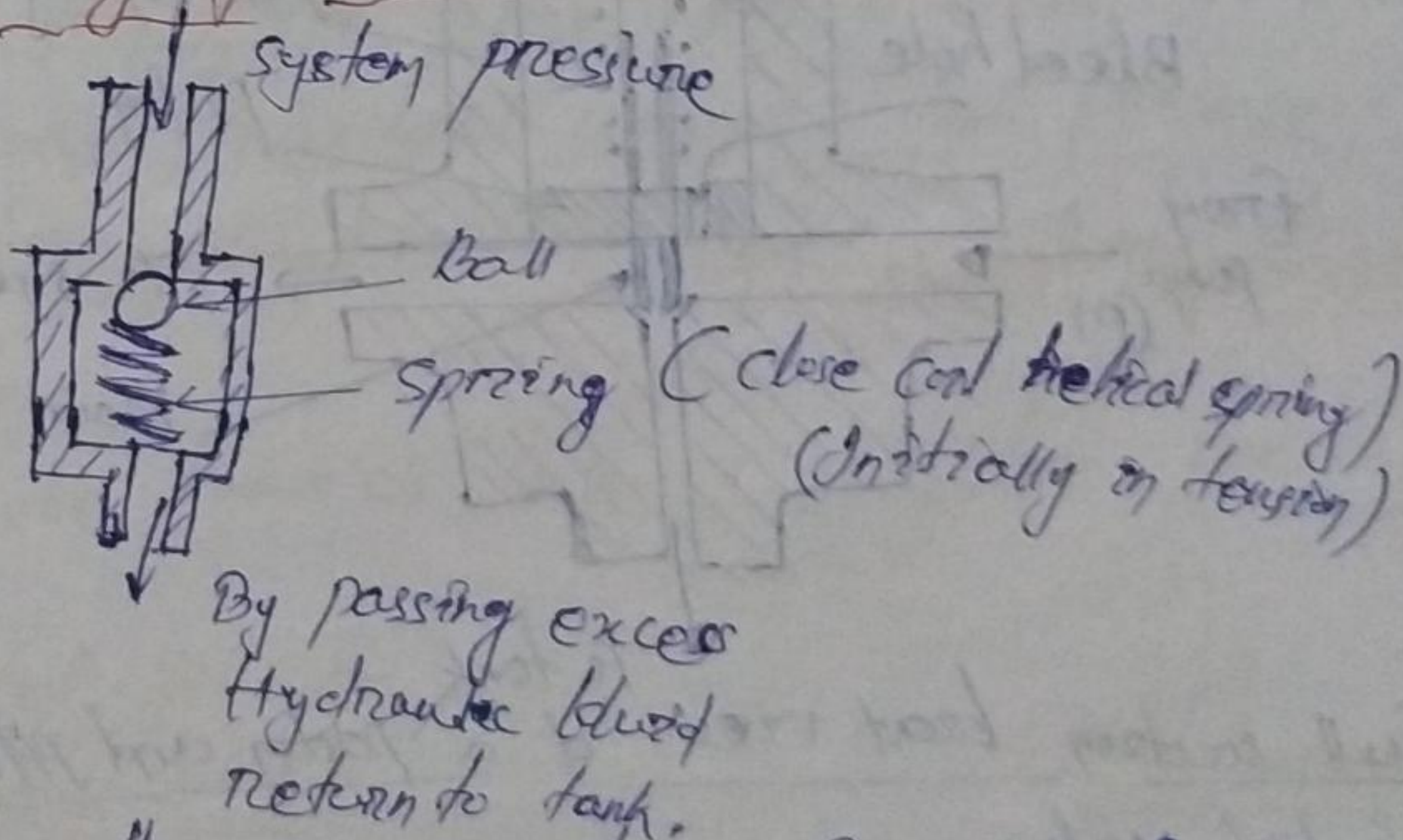
- Leakage, fire hazard, slippery.
- Floor area required is high
- Energy cost is medium.

Pressure regulation valve

- The simplest form of pressure regulation valve is ball and spring type pressure regulation valve.

Ball and pressure regulation

- Ball and spring type pressure regulation valve



- System force in the pipe exerts a force $= p \times A_{piston}$ when the force in the pipe line is greater than the compressive force of spring the valve will crack open by passing in the fluid to return tank.
- Higher the pipe pressure more is the valve opening.
- Cracking pressure is set by spring compressing constant.
- Difference between the cracking pressure and full flow pressure pressure override.

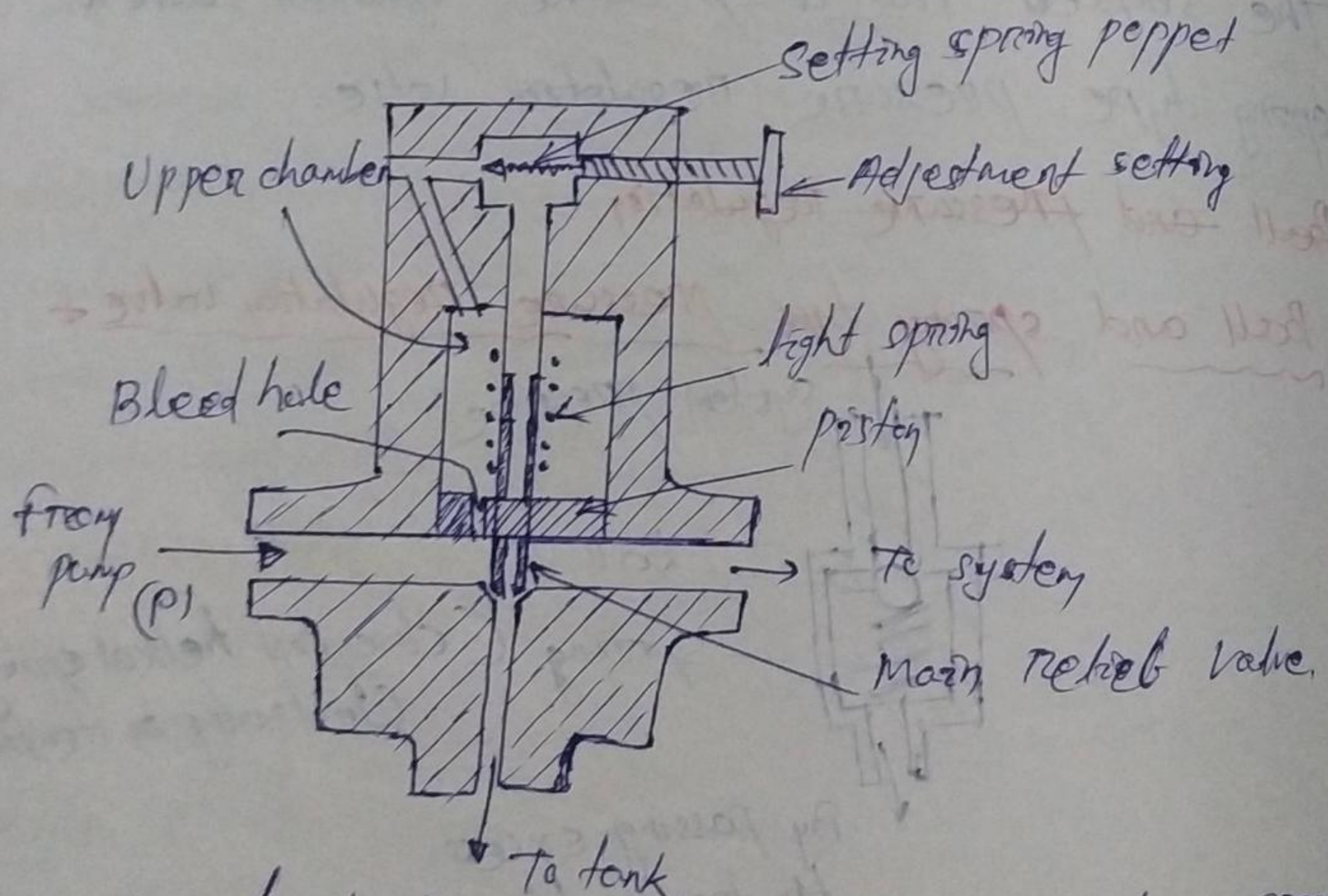
= Mathematically

Cranking pressure - full flow pressure (p.m. overcrack)
 $50 \text{ bar} - 30 \text{ bar} = 20 \text{ bar}$.

→ When pressure over control pressure is required relief valve is required.

→ A popular type of pressure relief valve is Balanced piston and poppet. (pressure relief valve)

* Piston and poppet type pressure relief valve



Full section front view of a piston and poppet type pressure relief valve.

① In case of working on low pressure valve.

→ The piston inside the upper chamber of the valve is free moving normally held at the lower position.

→ The piston pressure at the bottom at the top side of the piston is same. So there is no bleed flow across the poppet.

- The poppet is held in closed position by an adjustment spring.
- In this case bleed from the pump closed to the system by passing the main relief valve.

② Rise in pressure →

- When there is Rise in bleed pressure due to less demand in system high pressure bleed flows to upper chamber via bleed hole.
- The increase pressure on cracking pressure open the poppet valve. Those high pressure bleed passes through control valve whole of piston and main relief valve to the tank.

Case-3

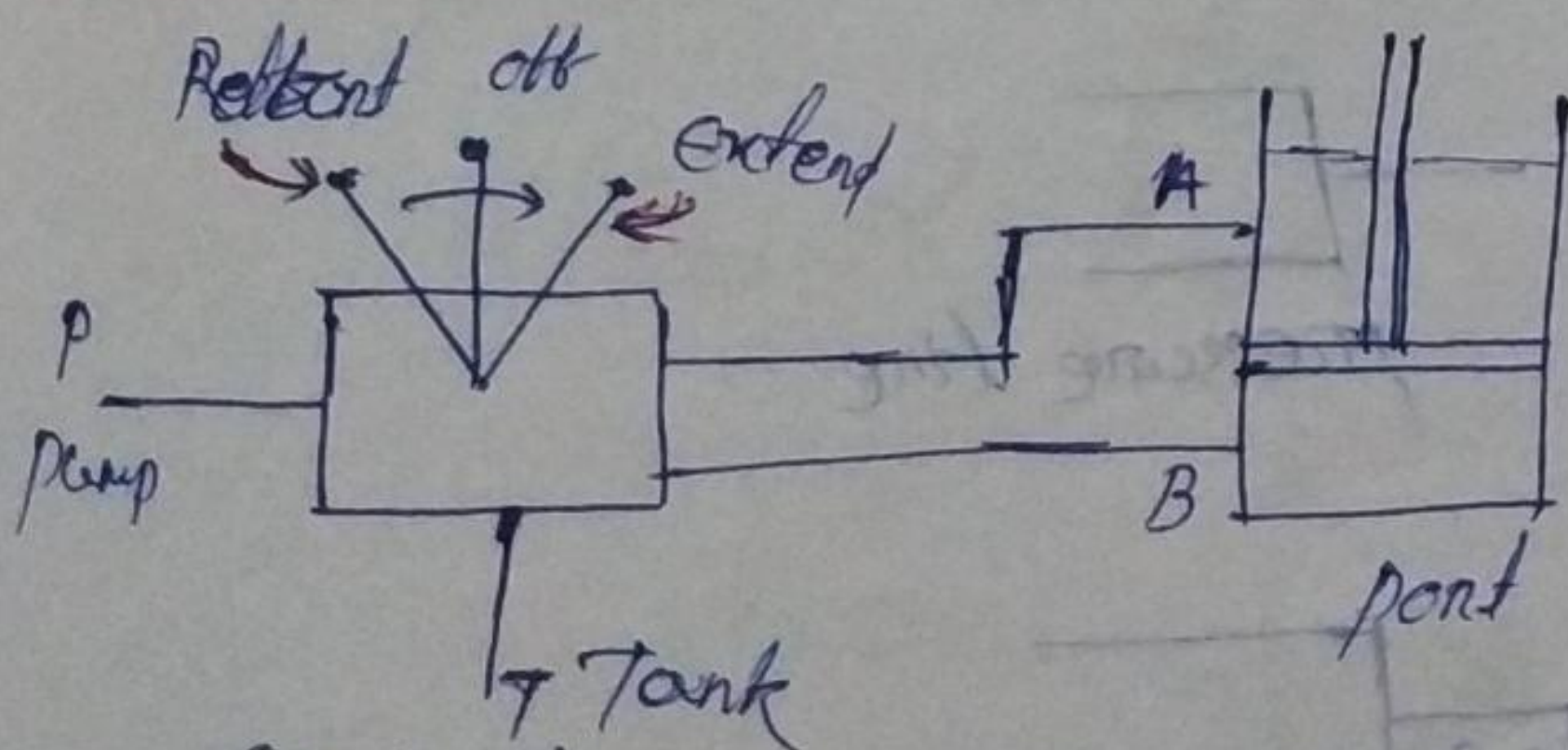
- When there is given a rise in pressure the bleed passes through the poppet along with due to high pressure main relief valve gives it. Then by high pressure bleed directly flow from pump to tank and stabilizing it

hydraulic system.

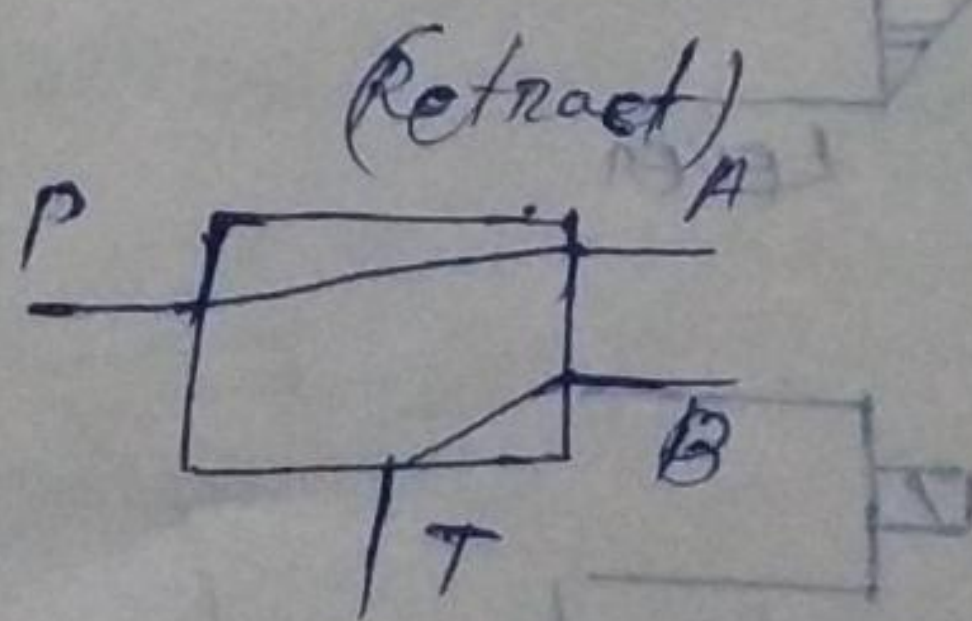
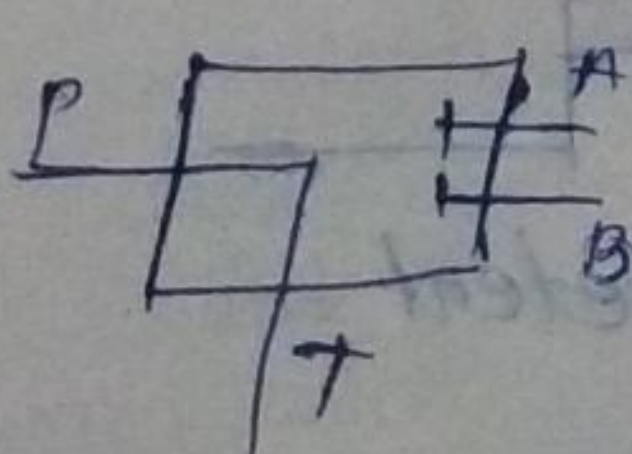
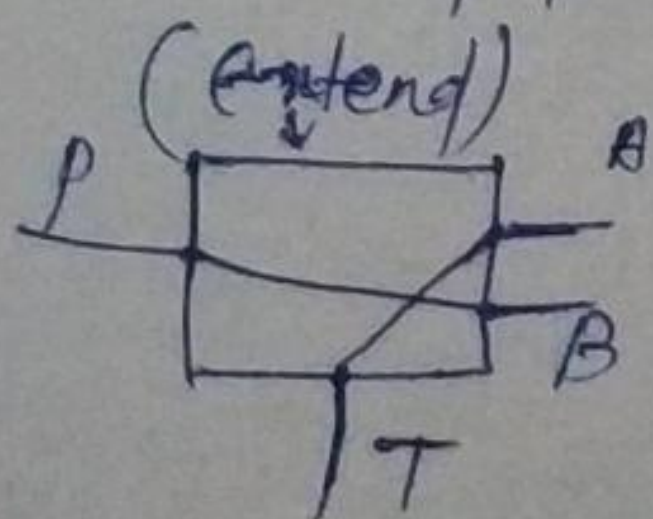
- This is otherwise known as Balanced piston type pressure relief valve.

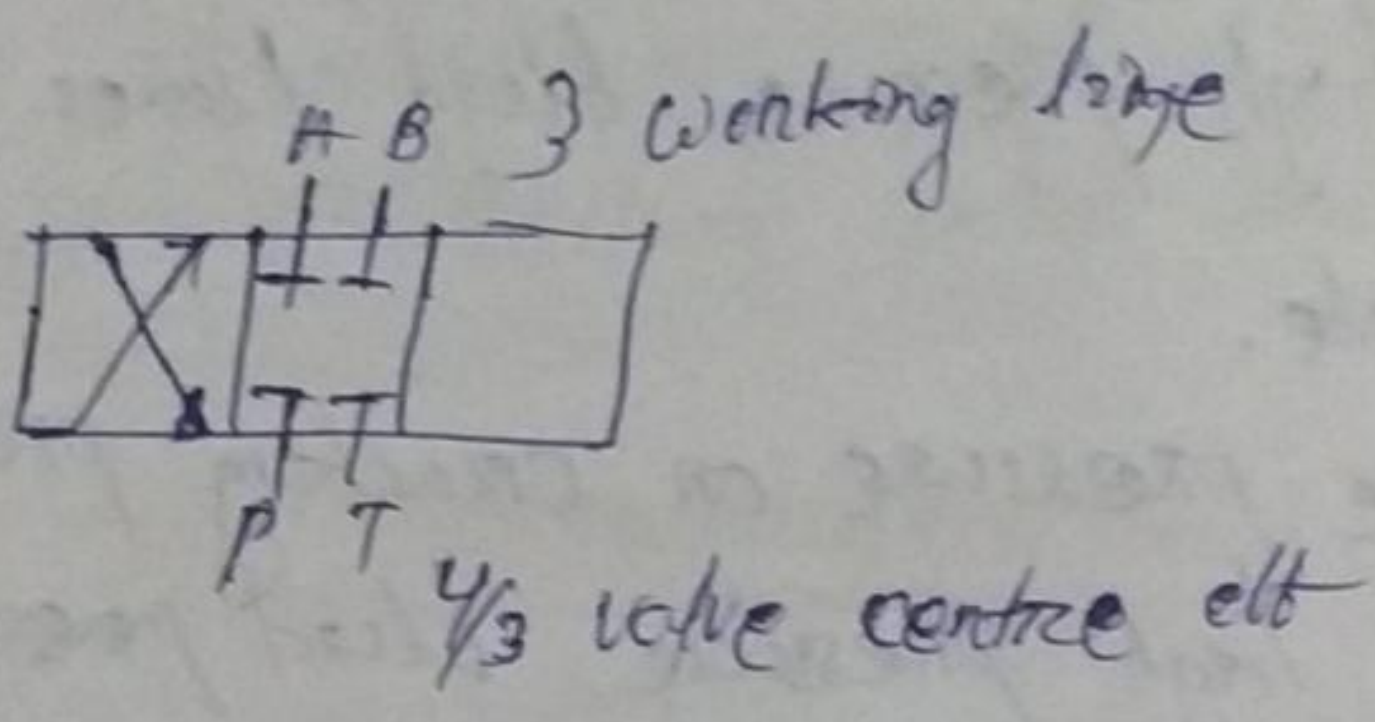
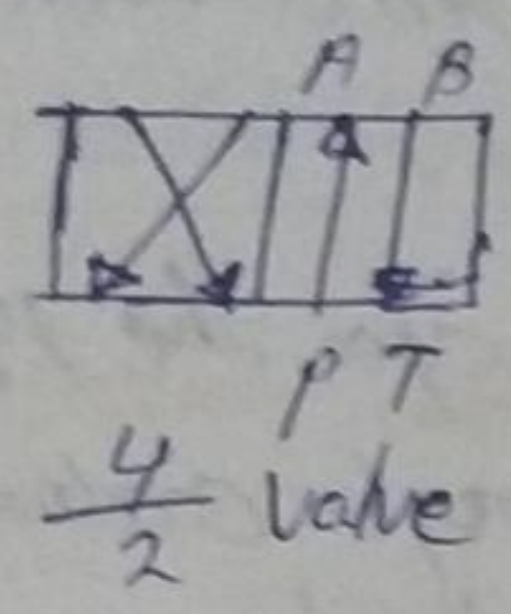
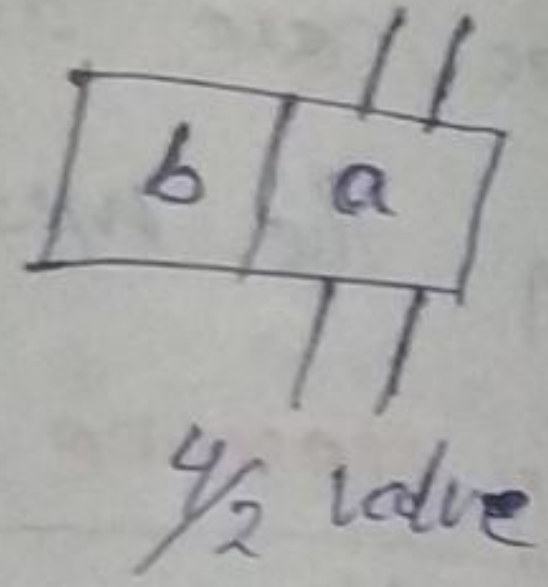
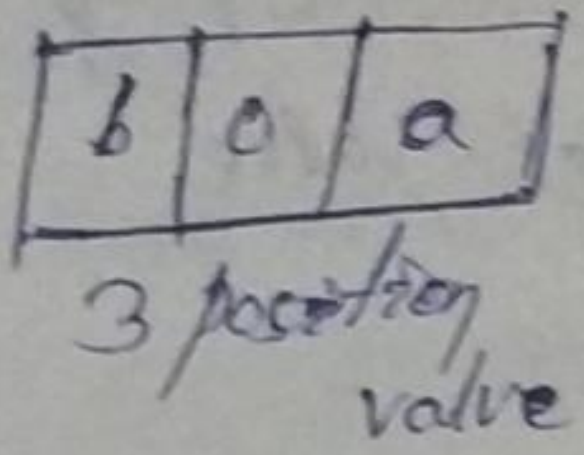
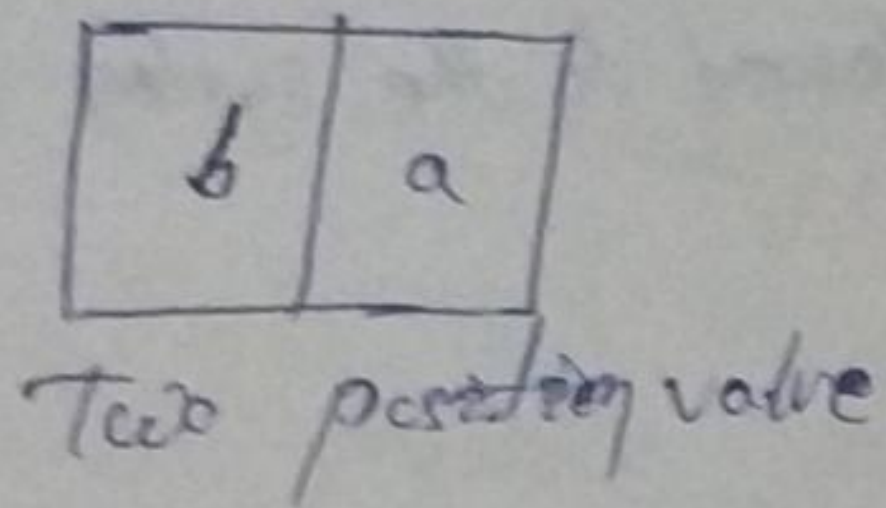
* pressure relief valve →

Direction control valve →



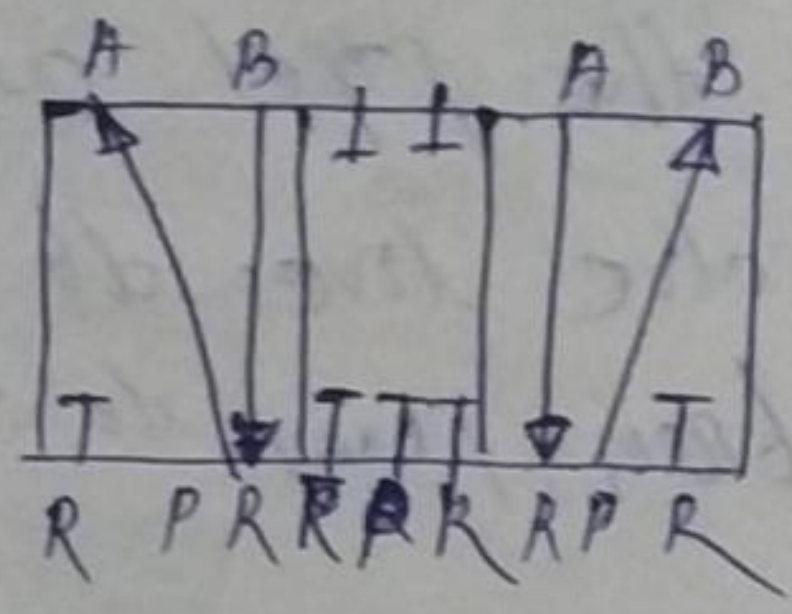
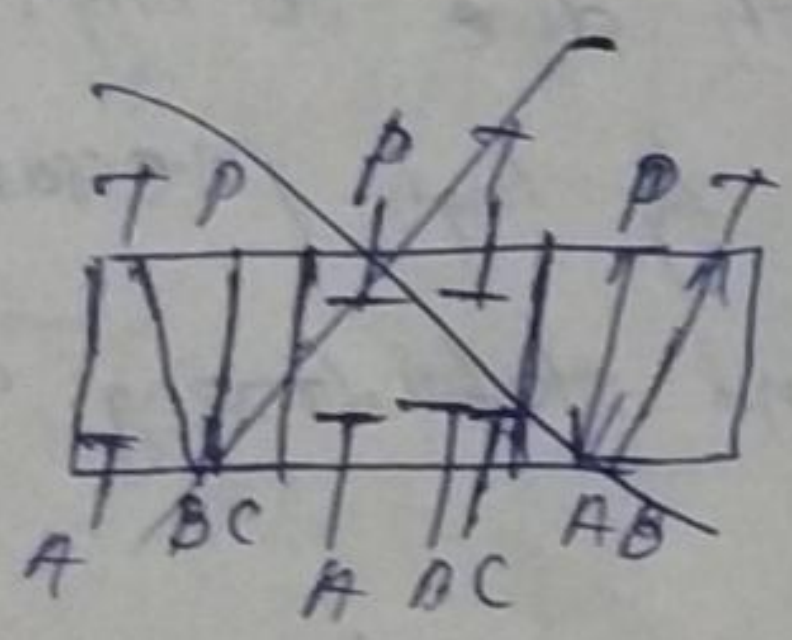
4/3 direction control valve
4 port 3 position



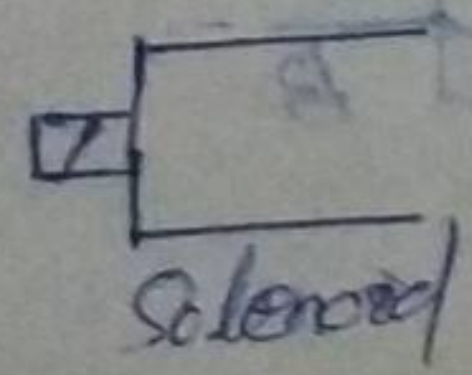
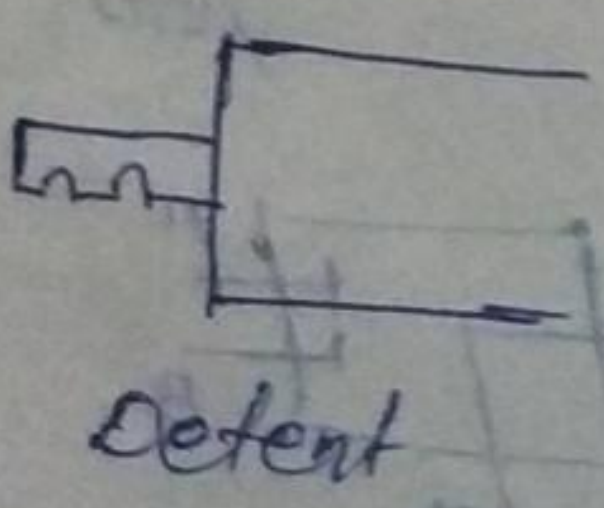
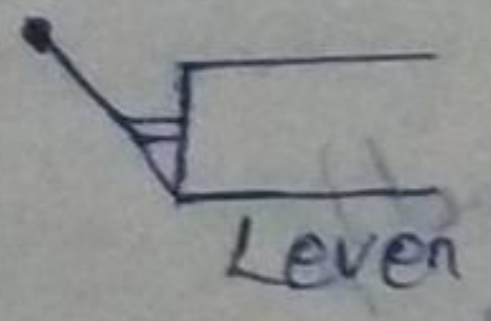
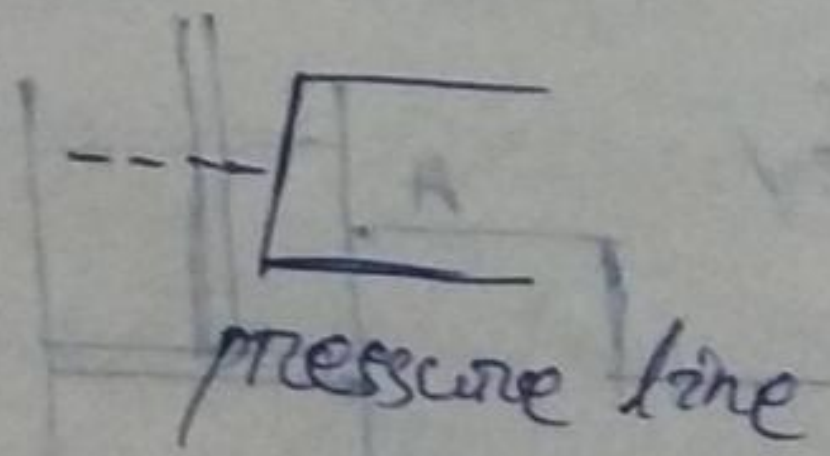
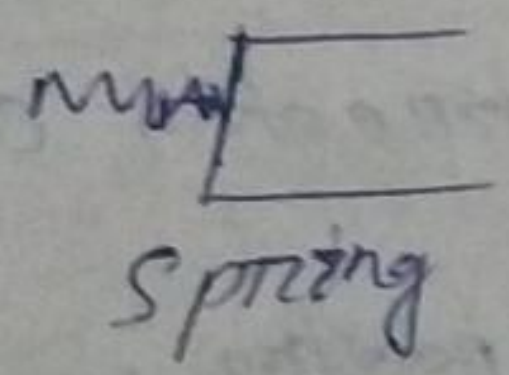
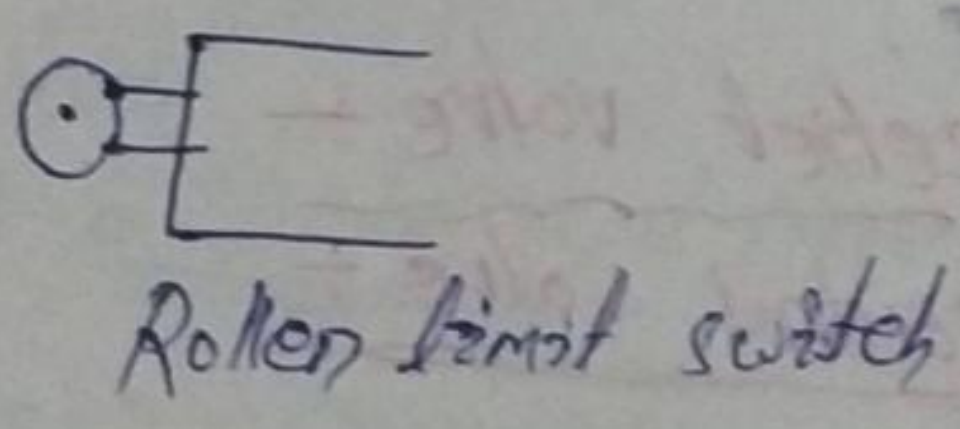
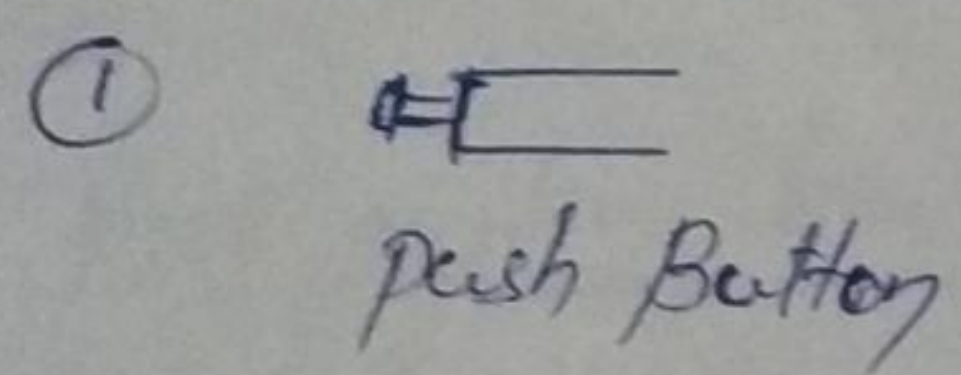


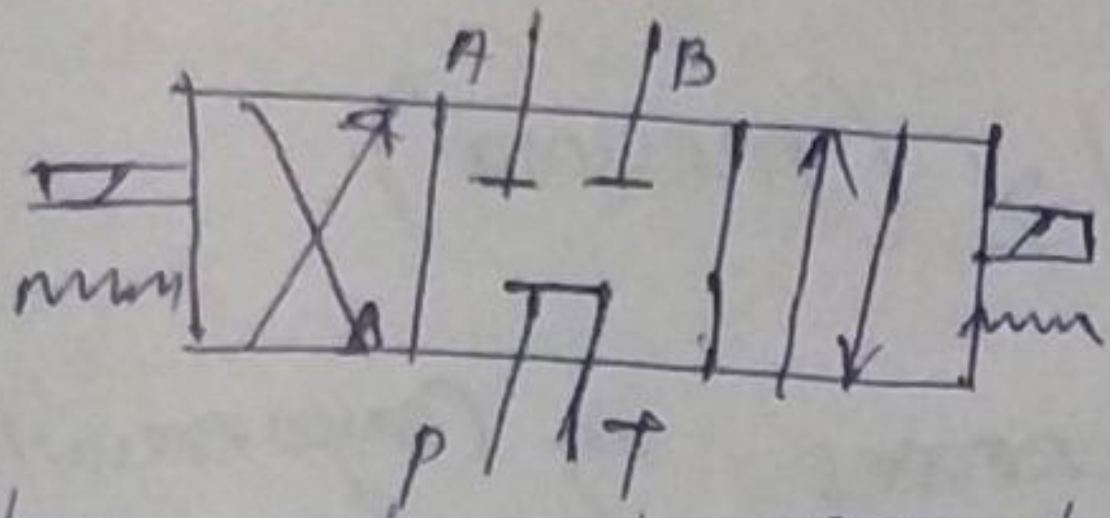
5/3 direction control valve

5 no of port
3 no of position



Valve actuation system:

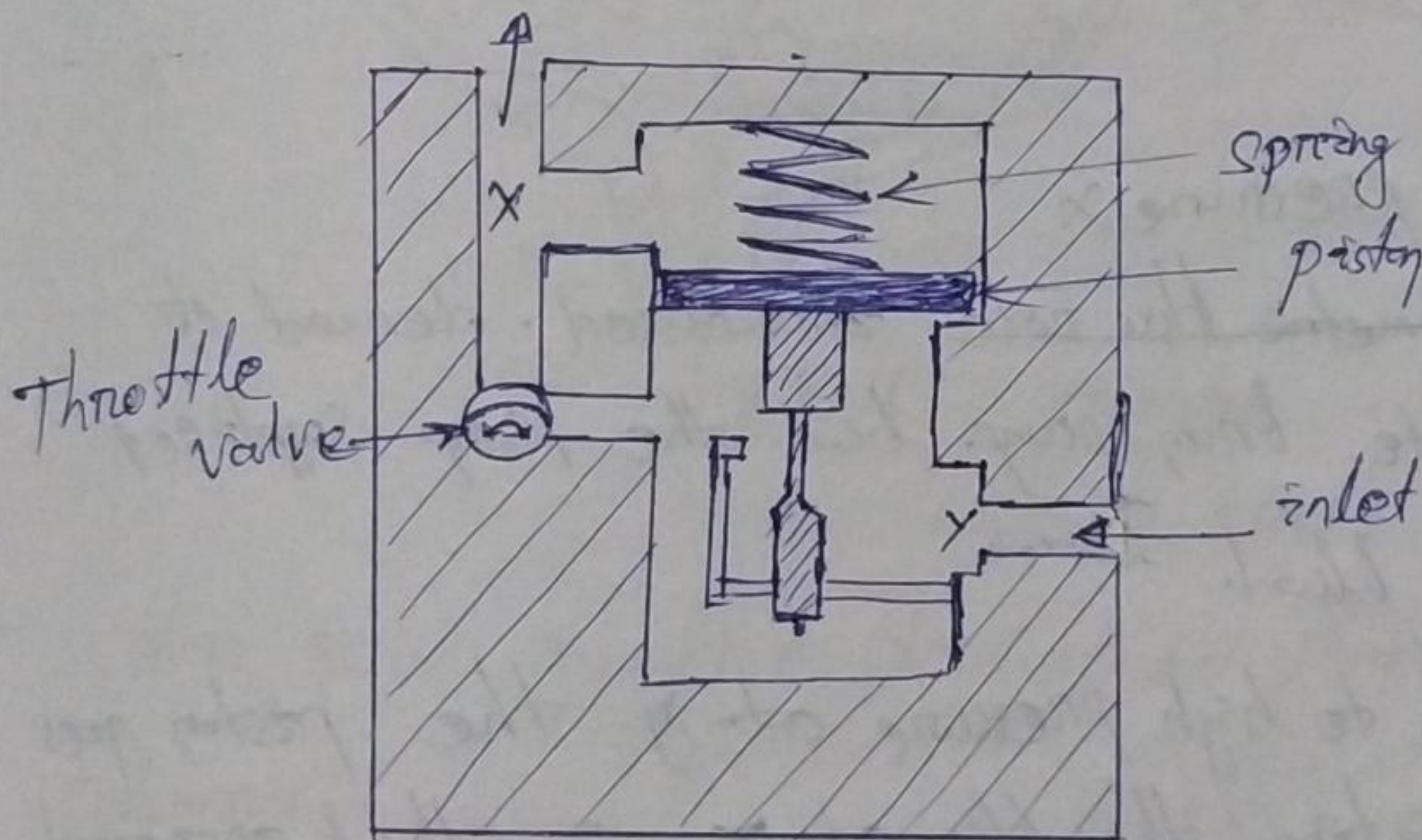




→ 4/3 valve solenoid operated spring return to centre pressure line unload to tank and load ~~to~~ Locked centre position

* Flow control valve +

outlet to actuator



pressure compensated flow control valve

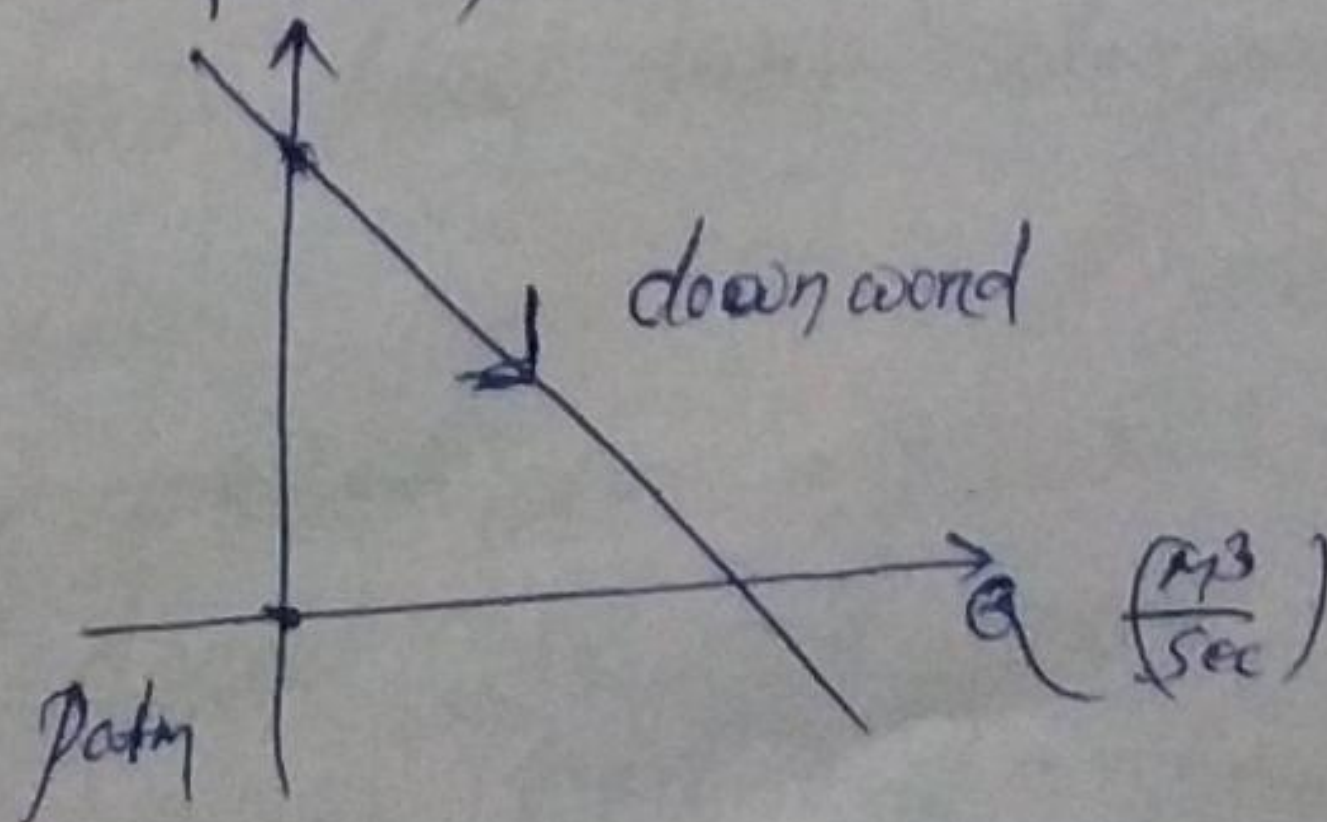
purpose ÷ fixed volumetric flow rate

$$Q = \frac{m^3}{sec} = \text{Constant}$$

How the purpose will be met +

By compensating / Adjusting pressure difference

$$Q \left(\frac{m^3}{sec} \right) \propto \frac{1}{\Delta p} \Delta p (\text{bar})$$



Case-1

at port x pressure $>$ port y (pressure)

→ When pressure at port x increases in comparison to pressure $-y$. The compresses the piston downward. There by the make the piston pressure stabilize at both x and y port. increased volumetric flow rate is returned at port x so that pressure at x and y comes to same level.

Case-2

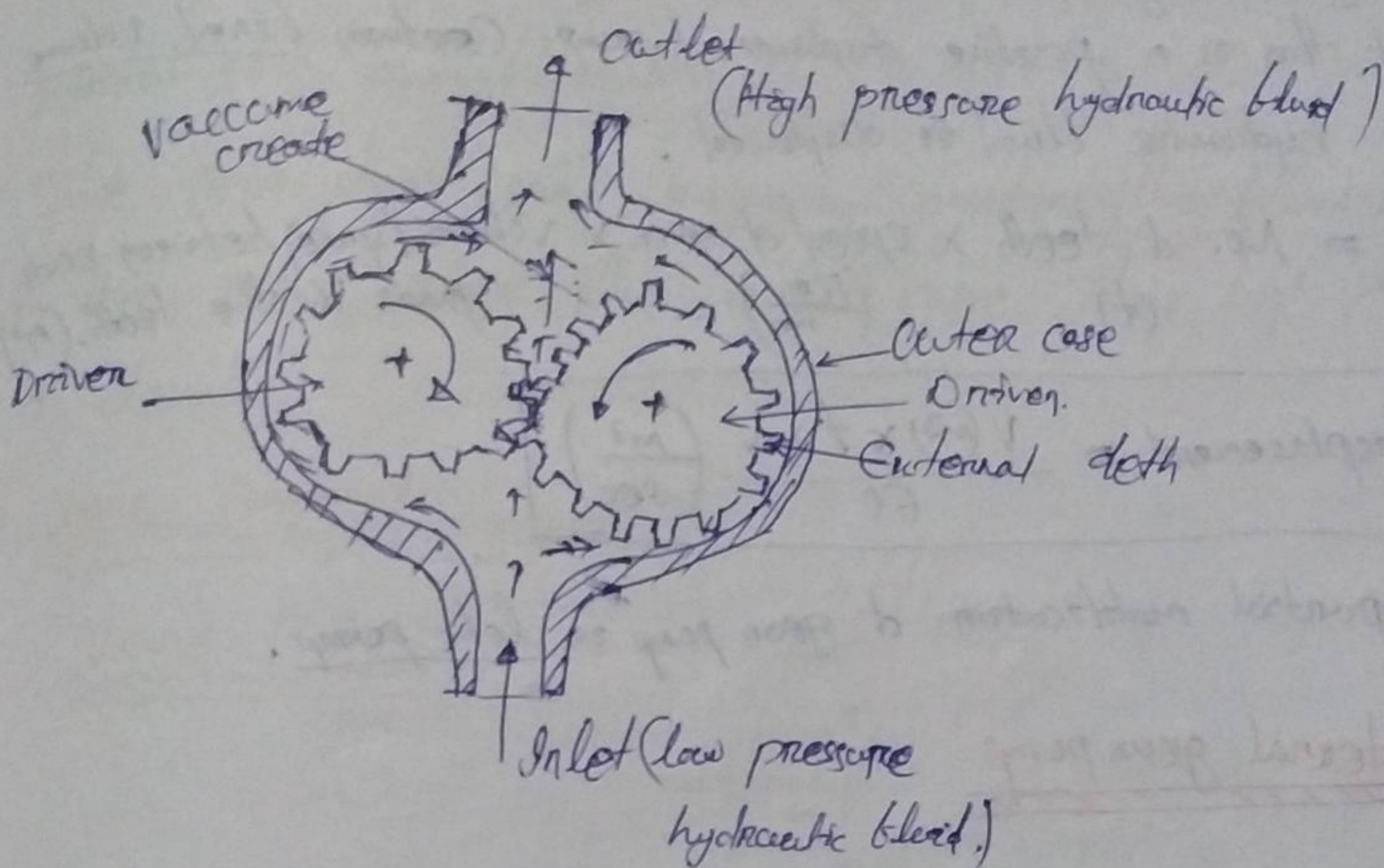
If pressure $y >$ pressure x

→ There is the ~~volumetric flow rate~~ increased. demand of volumetric flow rate from pump. but the pump supplied constant volume of blood.

→ in this case due to high pressure at y the piston goes at their by restricting the flow at x . so that increased volume blood is experienced at port y and pressure differential minimize.

* Fluid power pump :-

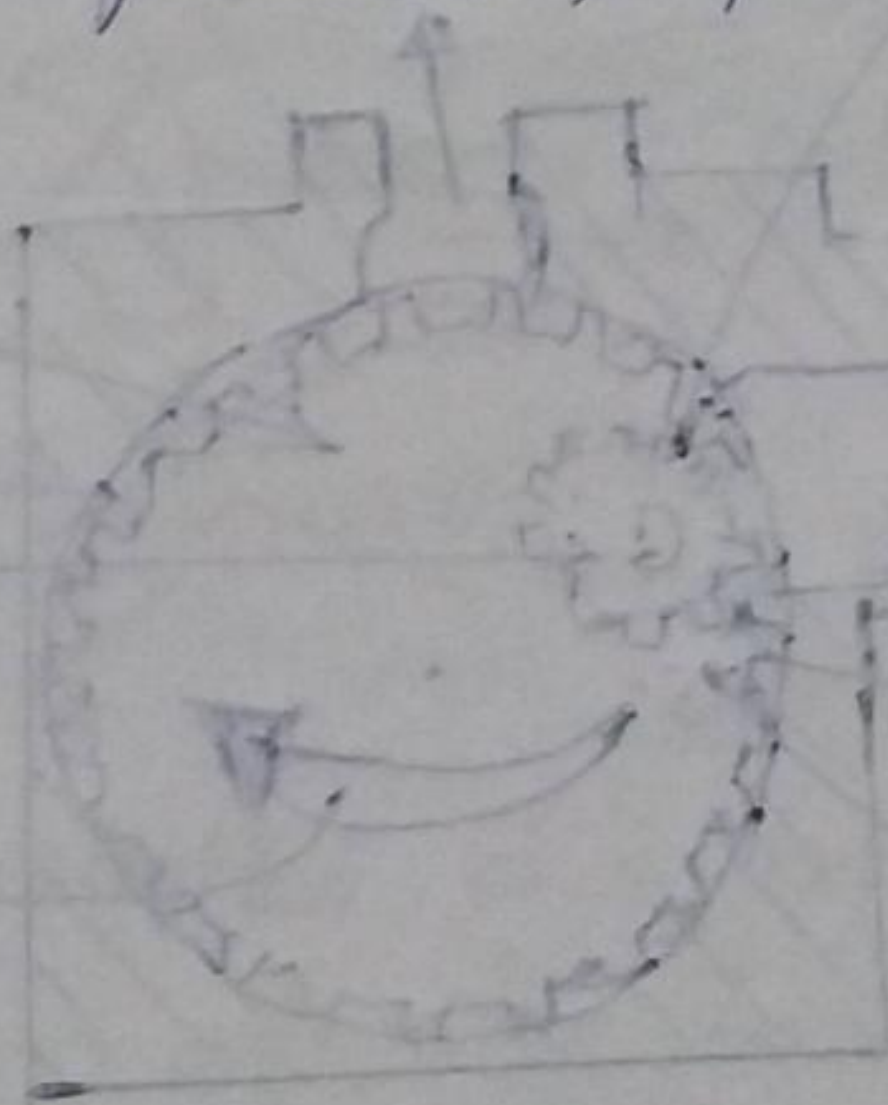
Gear pump :-



→ This is a positive displacement pump.

Component :-

- outer case -
- Driver gear.
- Driven gear
- side pressure plate.
- gland and packing.



* Working principle :-

→ As the driver and driven gear rotates with a high velocity partially vacuum is create at the upper part.

→ Then by the hydraulic fluid from inlet part is forced to moved along 3 direction.

(i) Along pitch line of meshed gear.

(ii) Left side space between case and driver.

(iii) Right side space between case and driven.

→ High pressure up to the range of 150 bar may be generated in the gear.

Mathematically,

As this is a positive displacement pump certain fixed volume of hydraulic fluid is displaced.

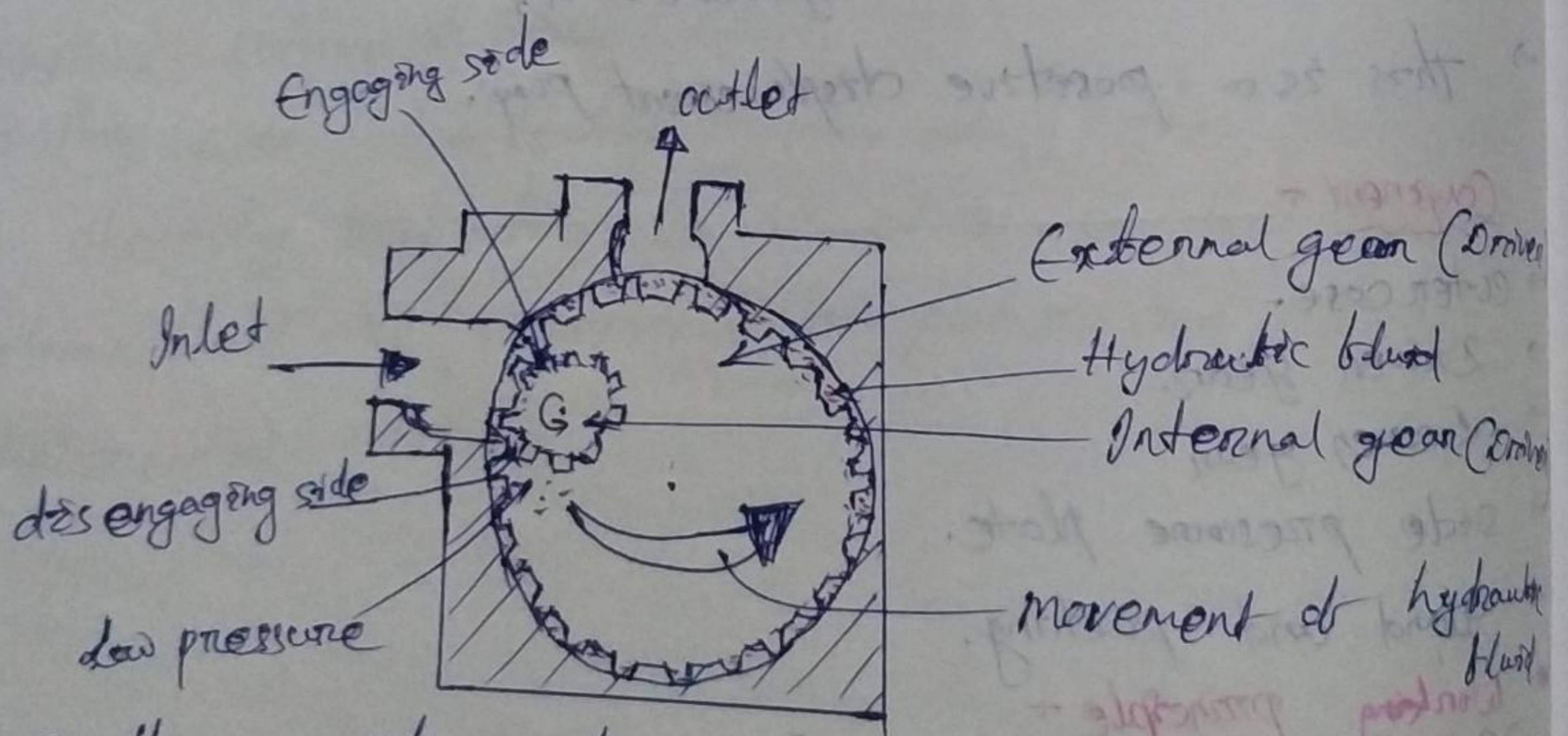
= No. of teeth $\times$ speed of gear $\times$ volume (space) between each pair of the teeth. (m³)

(t) $\left(\frac{rev}{min}\right)$

$$\text{displacement} = \frac{V (m^3) \times t \times N}{60} \left(\frac{m^3}{sec} \right)$$

→ A partial modification of gear pump is lobe pump.

Internal gear pump



→ Another varieties of gear pump is Internal gear pump.

→ Internal gear pump is a less leakage in comparison to gear pump. Where as pressure develop in

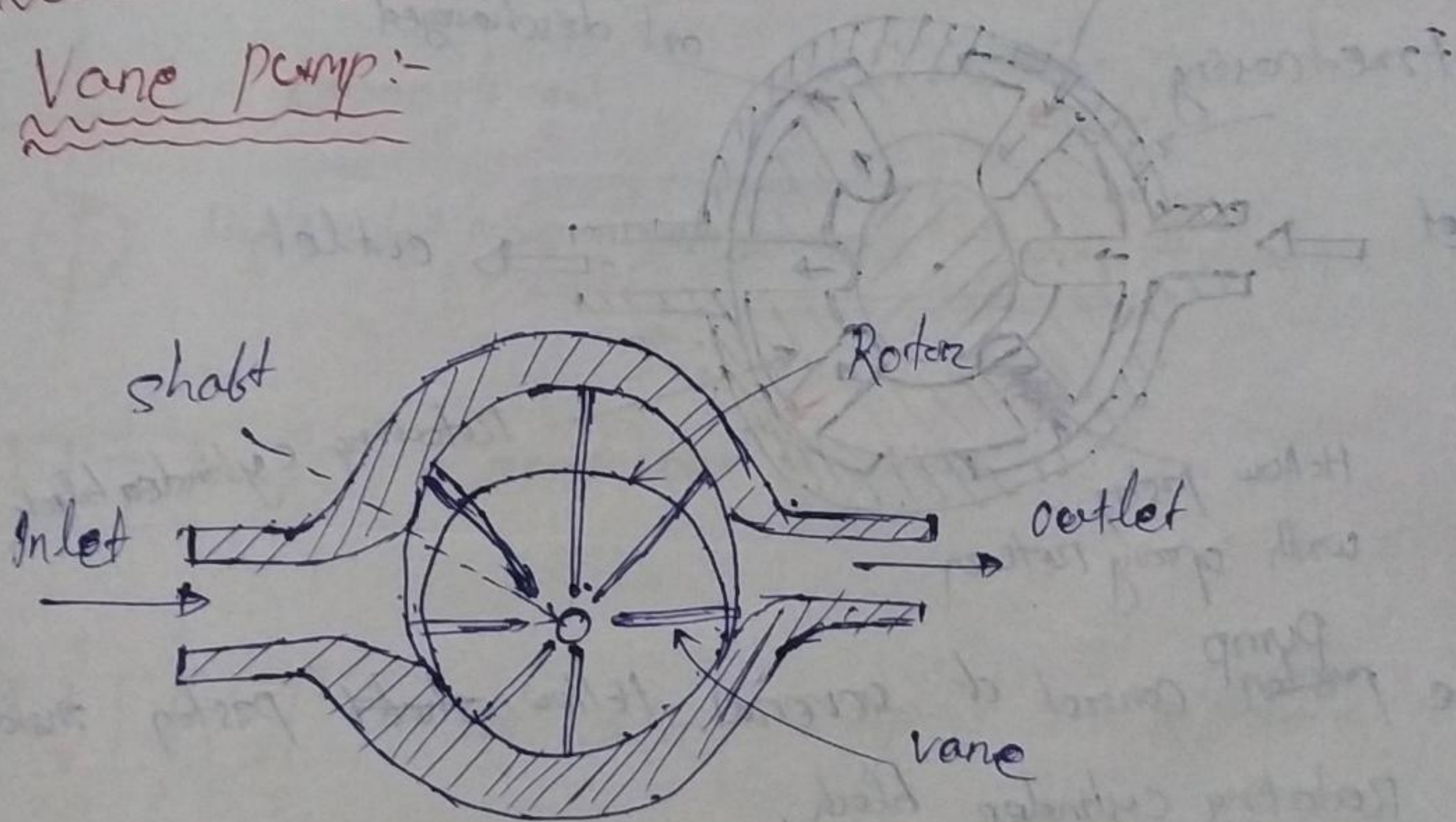
internal gear pump is less than gear pump.

→ External and internal gear are mounted on a body bearing. External gear is immersed inside the hydraulic fluid.

- External gear rotated counter clockwise, then by external gear also rotated counter clockwise.
- The partial vacuum is created at the bottom of the internal gear and fluid flow in the direction.
- The gear teeth are engaging position the fluid is forced to move along the outlet vane. Thus high pressure is developed.

* Vane Component :-

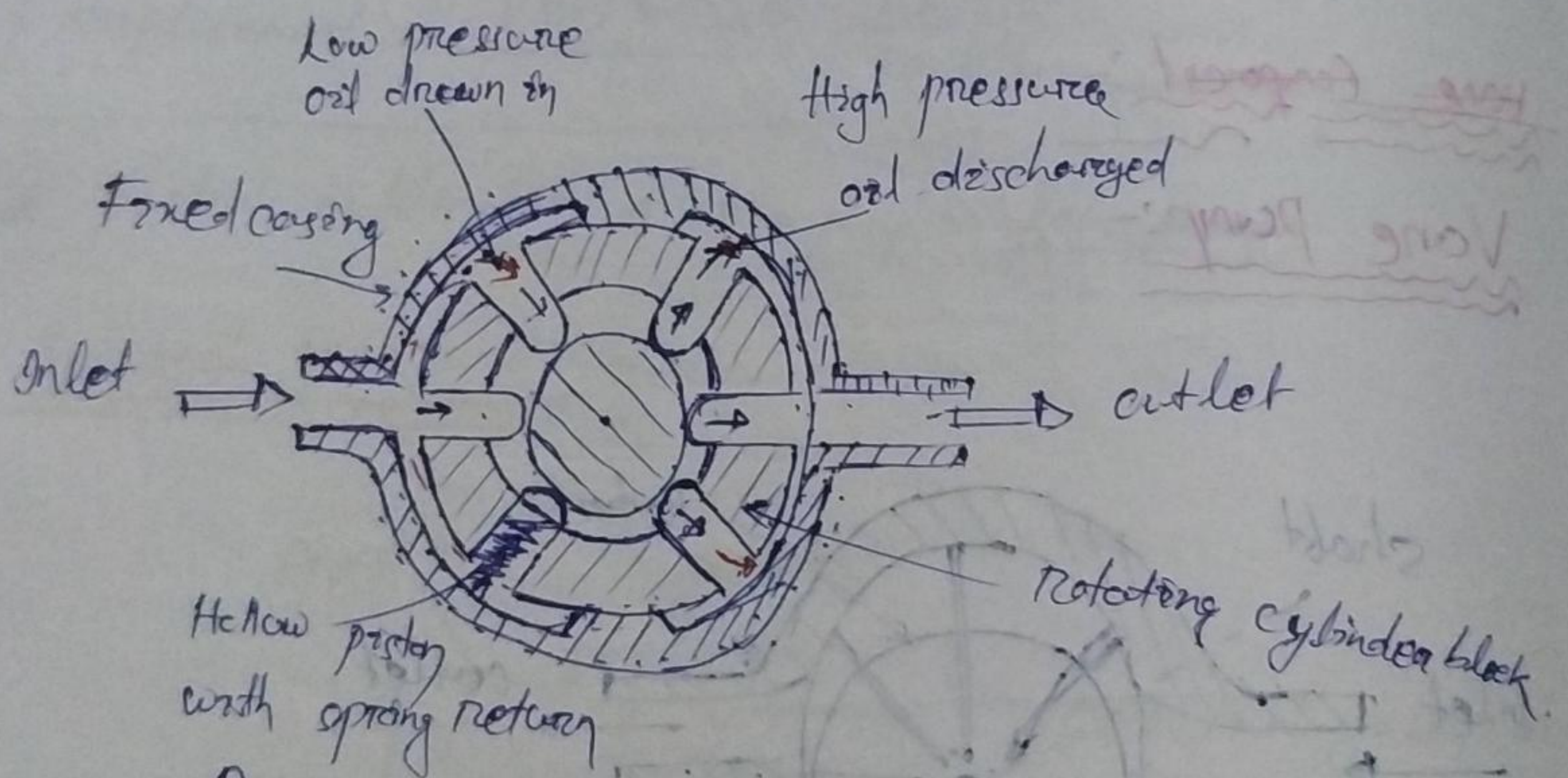
Vane Pump :-



- Vane pump is constituted of a rotor and a bearing.
- The rotor is offset within the bearing.
- When rotor is given power fluid is forced to move along the upper side of bearing and rotor.
- There is a fixed positive volume bearing at created at the pump upper part within bearing and rotor, thus fluid is forced to move along the outlet vane.
- To maintain the offset successfully vanes hold the rotor against bearing.

- The disadvantage of vane pump is bleed separated due to vane or unbalanced force with in vane.
- Another improvement over vane pump is radial piston pump where imbalance force due to taken care.

* Radial piston pump:-



- The ~~piston~~ ^{pump} consist of several hollow ~~shaft~~ piston inside a rotating cylinder block.
- Here the inlet port is connected to a inlet radial chamber when the piston pump across inlet radial chamber hydraulic oil is pushed into the piston against spring force.
- As the cylinder block is rotating when it come across outlet radial chamber. The spring create a pulse as to return to go original position. Thus high pressure bleed comes out from the hollow piston.

→ to flow the make the flow continuous multiple pistons are arranged. 13

→ Here during one revolution of cylinder block 3 pistons are acting as inlet piston and 3 pistons are acting as outlet piston.

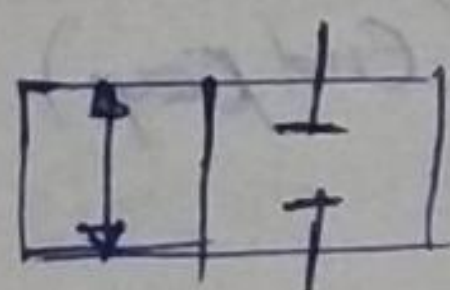
ISO symbol for hydraulic component

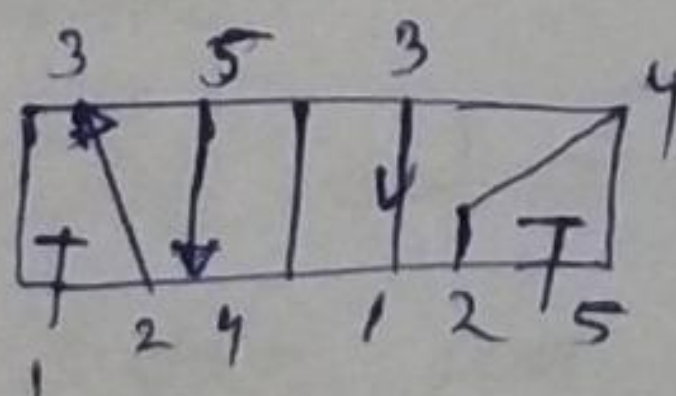
———— Supply line, Return line

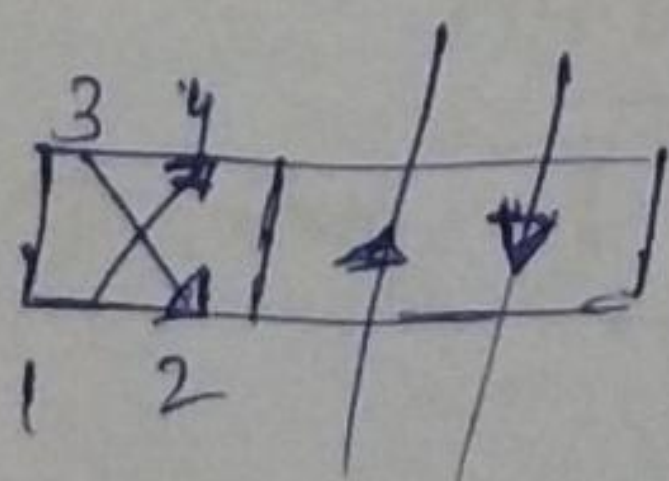
—U— Hose assembly

┌┴┐ Drain to tank

⊙ Hydraulic pump (Arrow head showing direction)

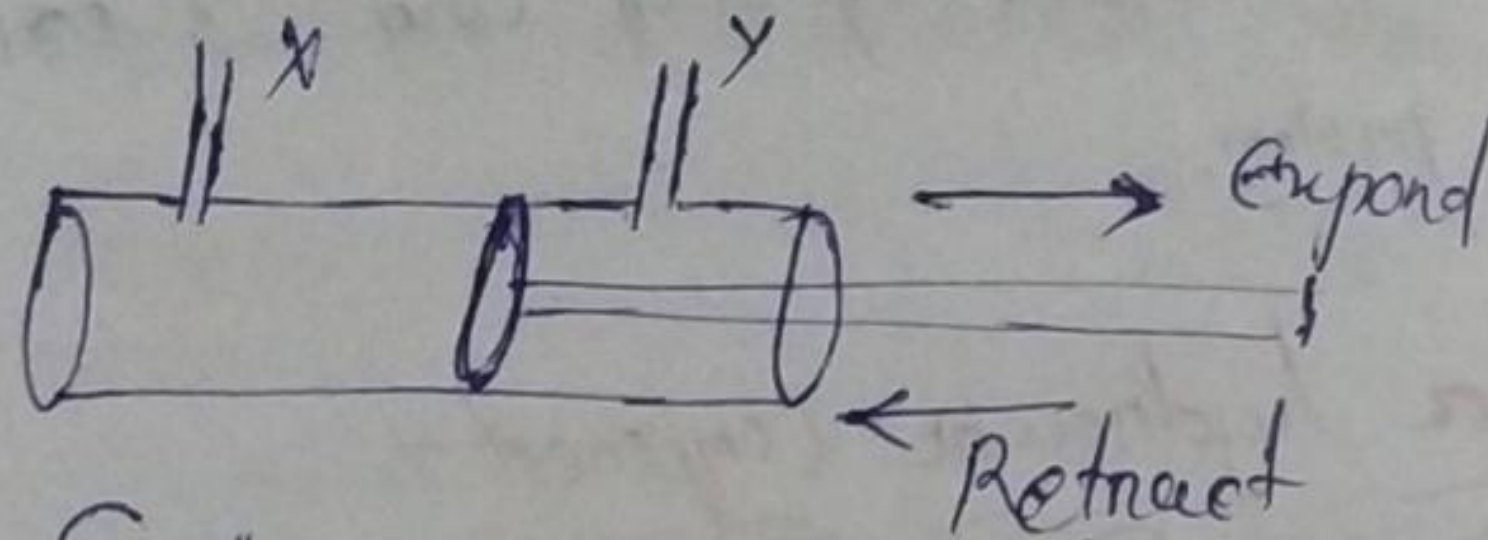
 $\frac{2}{2}$ 2 port and 2 position

 $\frac{5}{2}$ valve 5 port and 2 position

 $\frac{4}{2}$ Directional Control Valve 4 port, 2 position.

Actuator

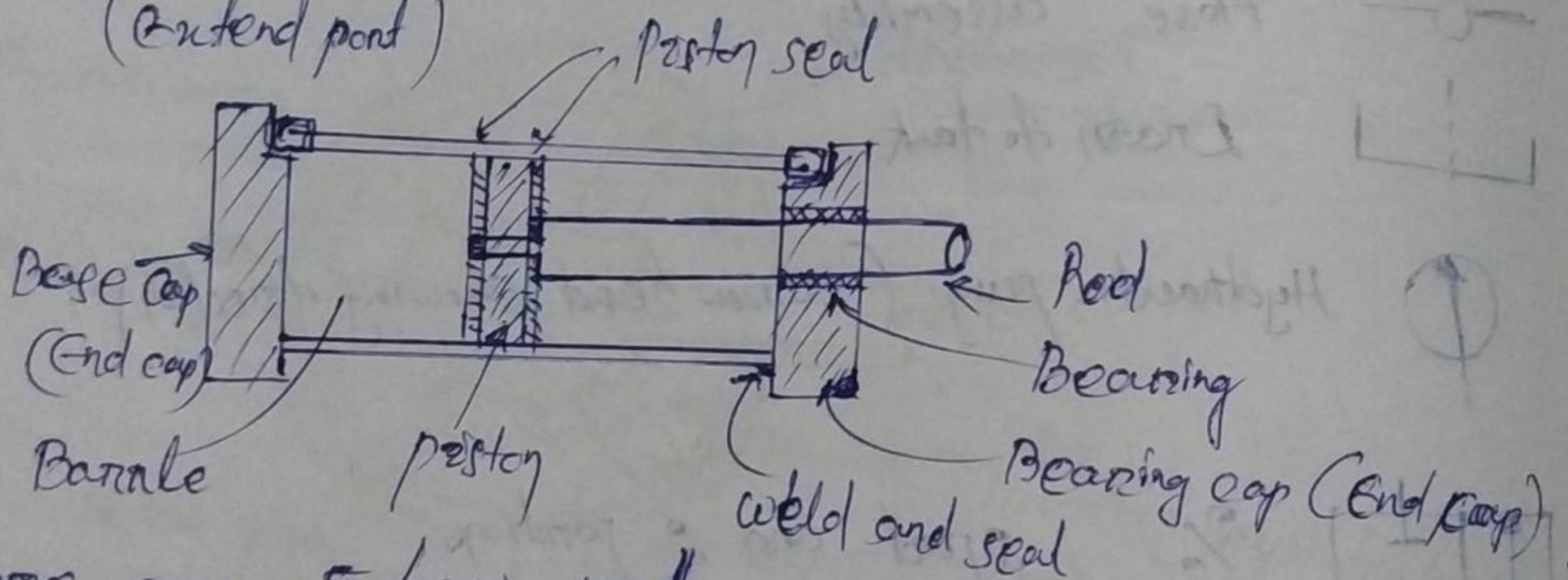
Basic linear actuator is a cylinder on ram.



(A basic cylinder actuator)

A typical construction in a double acting cylinder

(Extend port)



→ There are 5 basic part

(i) Two end cap:-

Base cap bearing cap

(ii) piston connection

Extend port Retract port

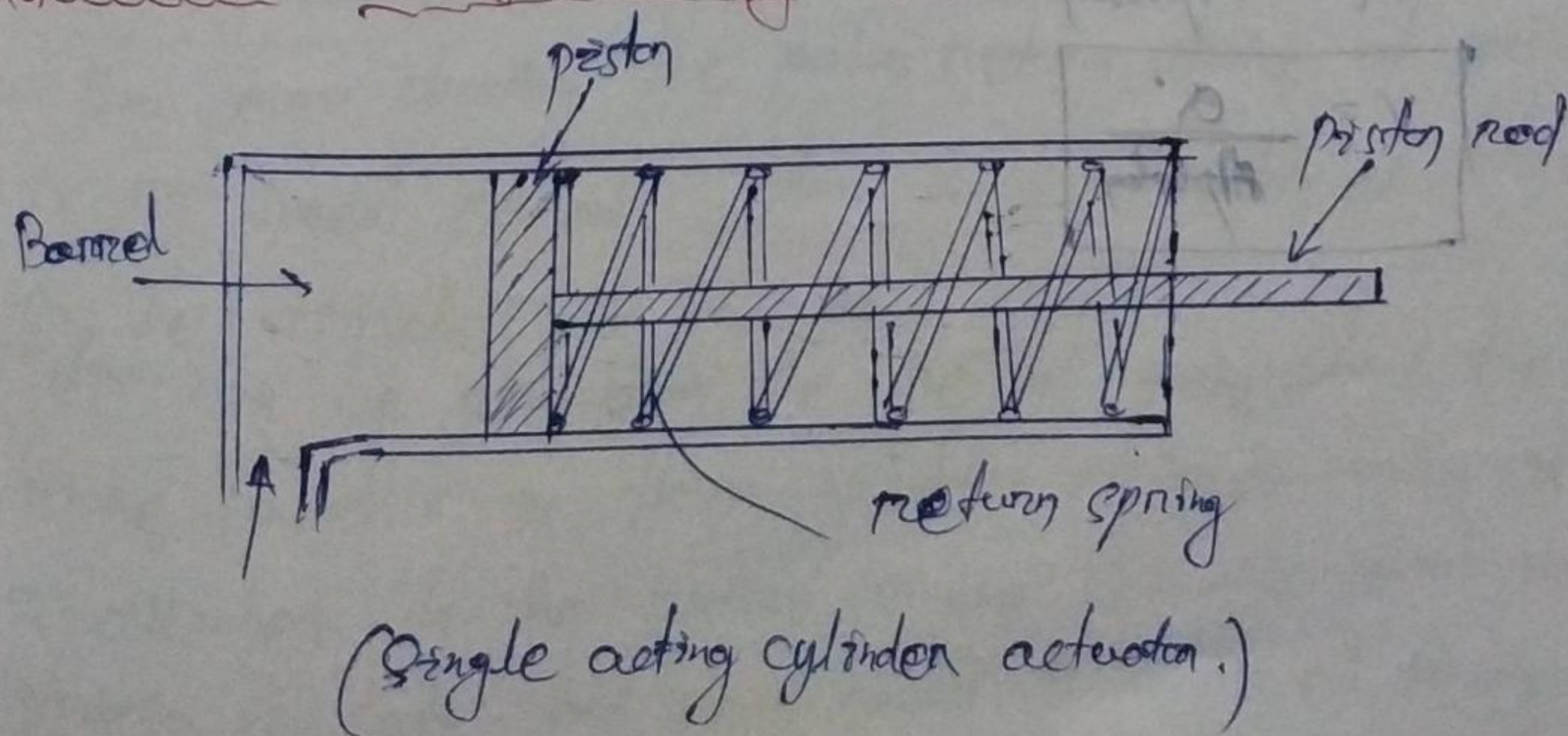
(iii) cylinder barrel.

(iv) piston

(v) piston rod

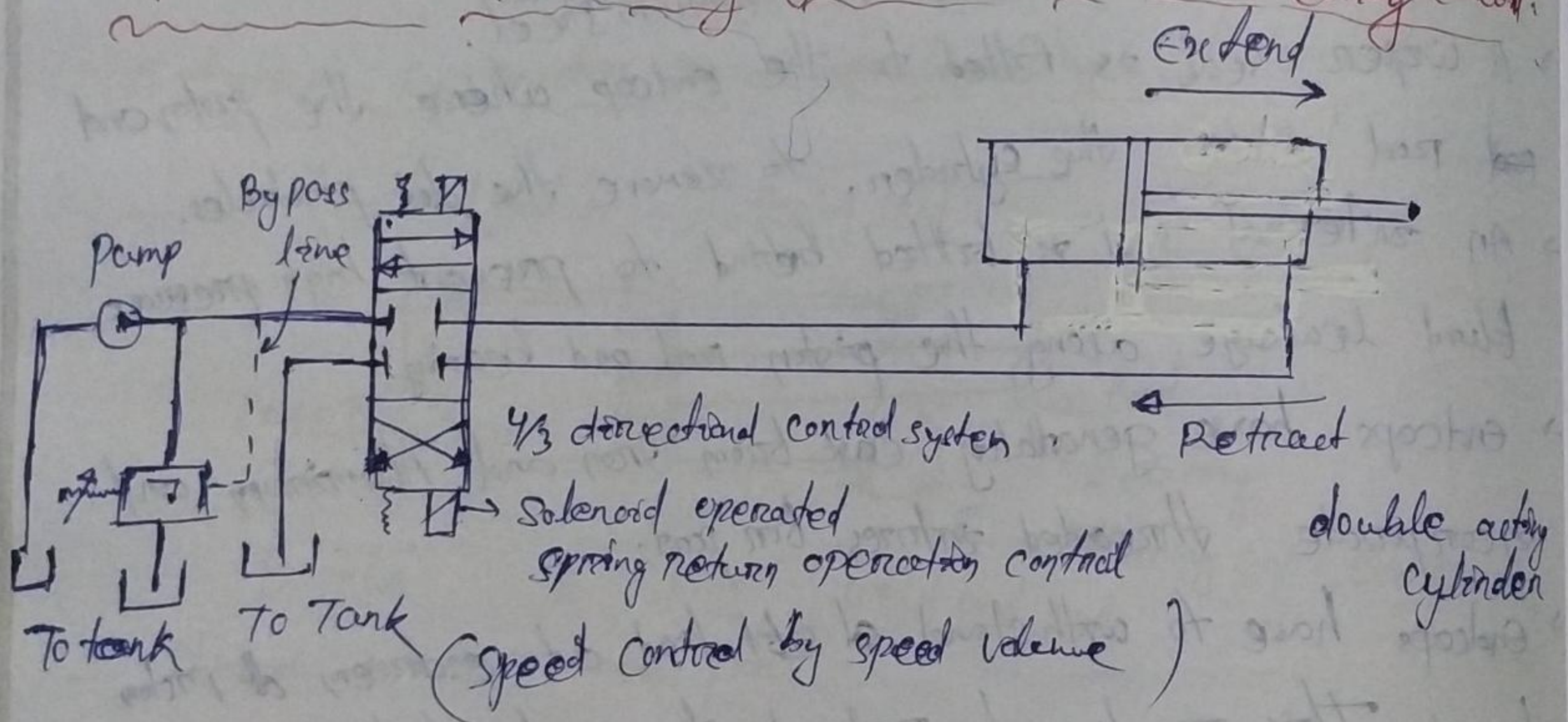
- End caps are secured to barrel by welding ^{or} by them connecting.
- The barrel is generally manufactured seamless drawing the piston finish made one provided.
- Again piston rod is exposed to atmosphere to prevent corrosion.
- The piston rod is made up of chromium steel.
- A wiper seal is fitted to the endcap where the piston and rod enter the cylinder, to remove the dirt particles.
- An internal seal is fitted behind to prevent high pressure fluid leakage along the piston rod and bearing.
- Endcaps have generally cast from Iron and Aluminium and incorporate threaded entries for port.
- Endcaps have to withstand a soft load at extremity of piston travel. This is due to rate of change of velocity.
- This end of the travel shock load can be reduced with cushioning pad build in to the end caps.

Hydraulic circuit Operation of single acting cylinder actuator:-



- Single acting cylinders are simple to drive and especially used in pneumatic circuit.
- The piston ^{position} ~~move~~ to the initial ~~with~~ the piston help of spring action. for quicker return of piston spring constant has to be increasing.

operation of double acting cylinder with metering circuit.

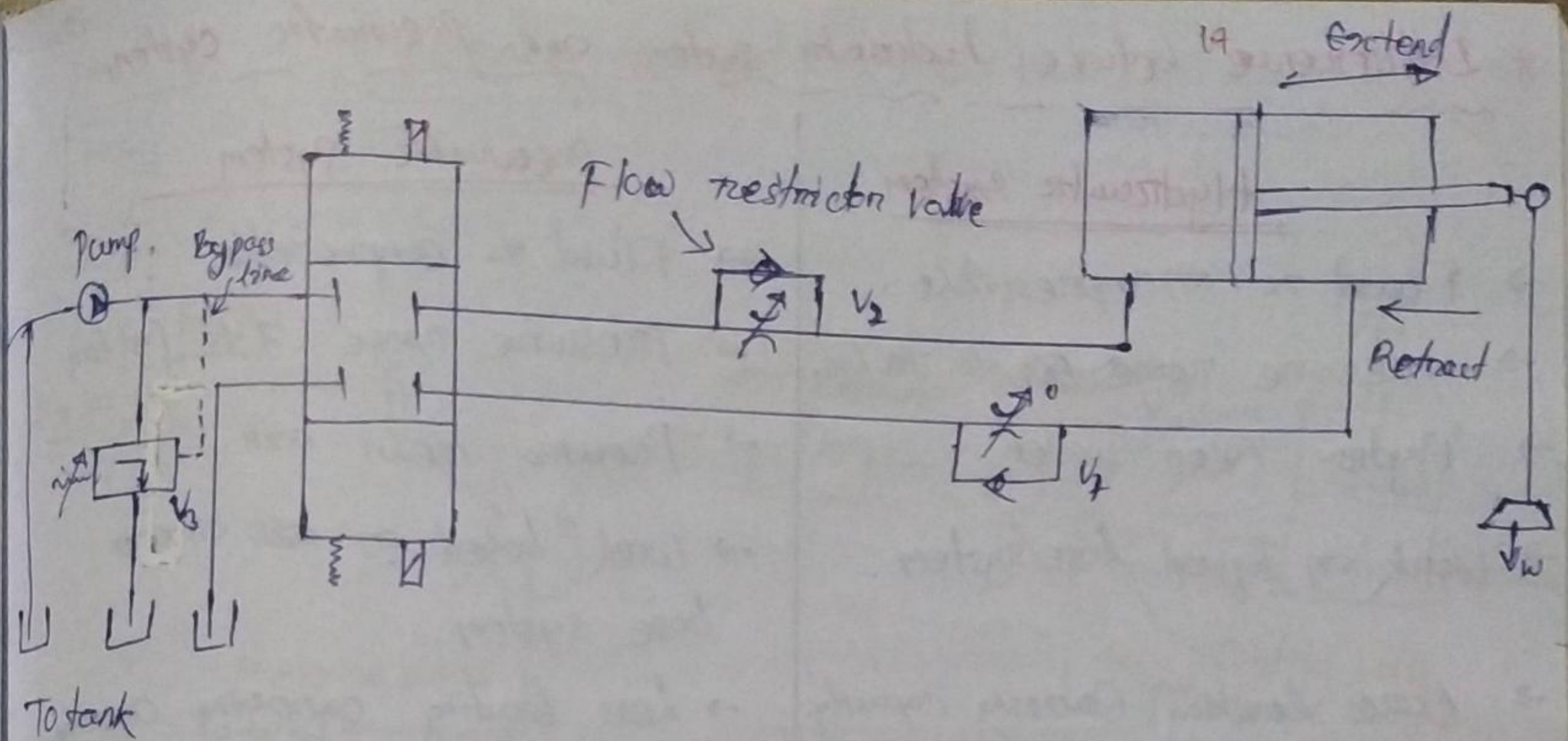


objective: speed control of piston during extend and retract.
 $\left(\frac{N}{\text{sec}}\right)$

Volumetric flow rate across pump: $Q' \left(\frac{m^3}{\text{sec}}\right)$

$$Q' = A_{\text{piston}} \times V$$

$$V = \frac{Q'}{A_{\text{piston}}}$$



(speed control by metering in and metering out)

→ In the first circuit speed is controlled by varying cross-sectional area of piston which may not be feasible at all time. So we get a constant speed during extend stroke and returned stroke.

Metering in:

→ To control the ~~para~~^{speed} of the piston during extend stroke volumetric flow rate (Q) from the pump is varied with the help of flow restriction valve (V_2). Thus velocity during extend stroke can be controlled. This is called as metering in.

Metering out:

→ However during the ~~piston~~ return stroke of piston there is no speed control. This can be achieved by attaching a dead weight

can be attached to the piston rod.

→ This can be slow down the return stroke. But the extension stroke is faster. So there is flow restriction valve is ~~control~~ attached to the return stroke pipe line, which varies the volumetric flow rate and speed control is achieved during return stroke speed control by the flow restriction valve is metering out.

* Difference between hydraulic system and pneumatic system

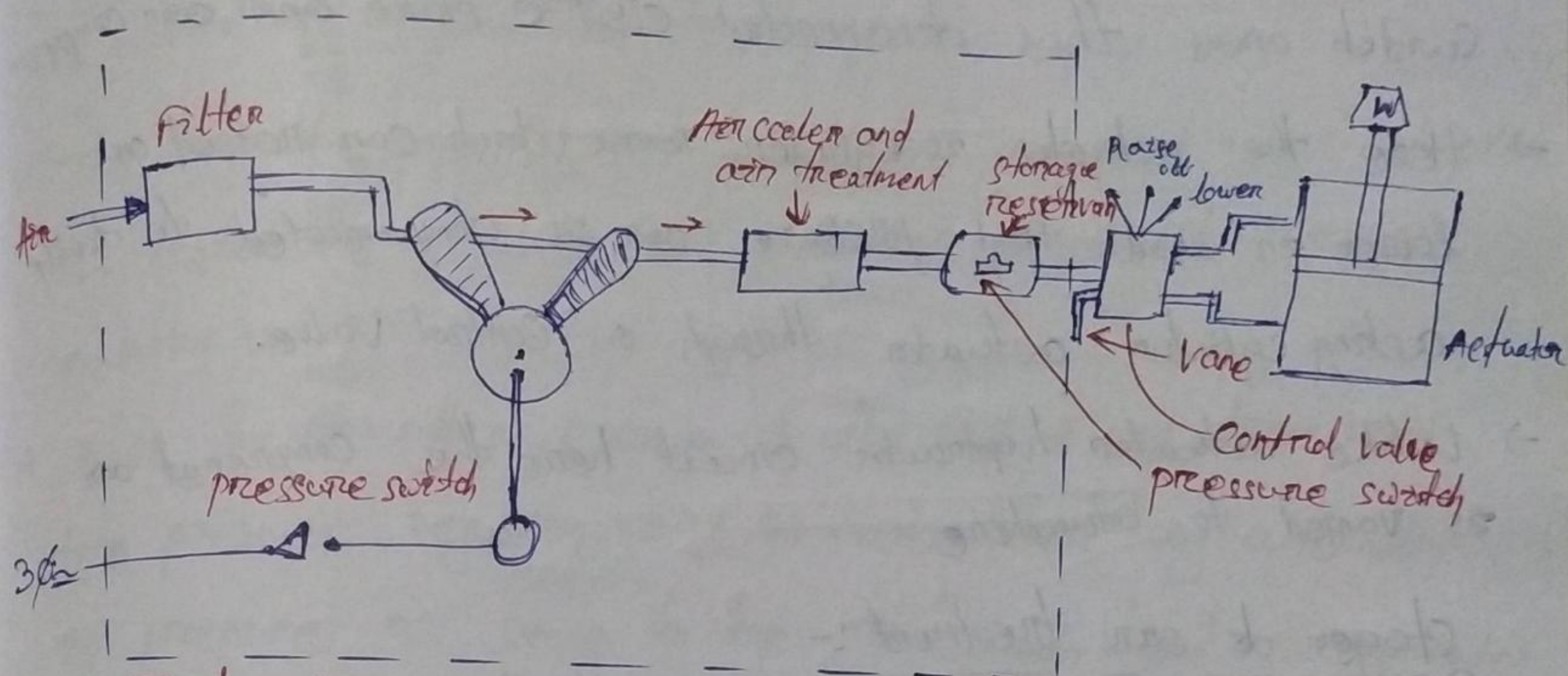
Hydraulic system

- Fluid is incompressible.
- pressure range 500 to 700 bar.
- Hydraulics mean water.
- Work on liquid base system.
- More loading carrying capacity.
- Separate lubrication is not required.
- Bulky system.
- Slower system. (Due to viscous oil)
- Close loop system.
- more complex (return line)

pneumatic system

- Fluid is compressible.
- pressure range 7 to 15 bar.
- Pneumatics mean air.
- Work based on gas base system.
- Less loading carrying capacity.
- Separate lubrication with special arrangement is required.
- Less Bulky system.
- Faster ~~oil~~ system.
- open loop system.
- Less complex (No return line)

Pneumatic circuit



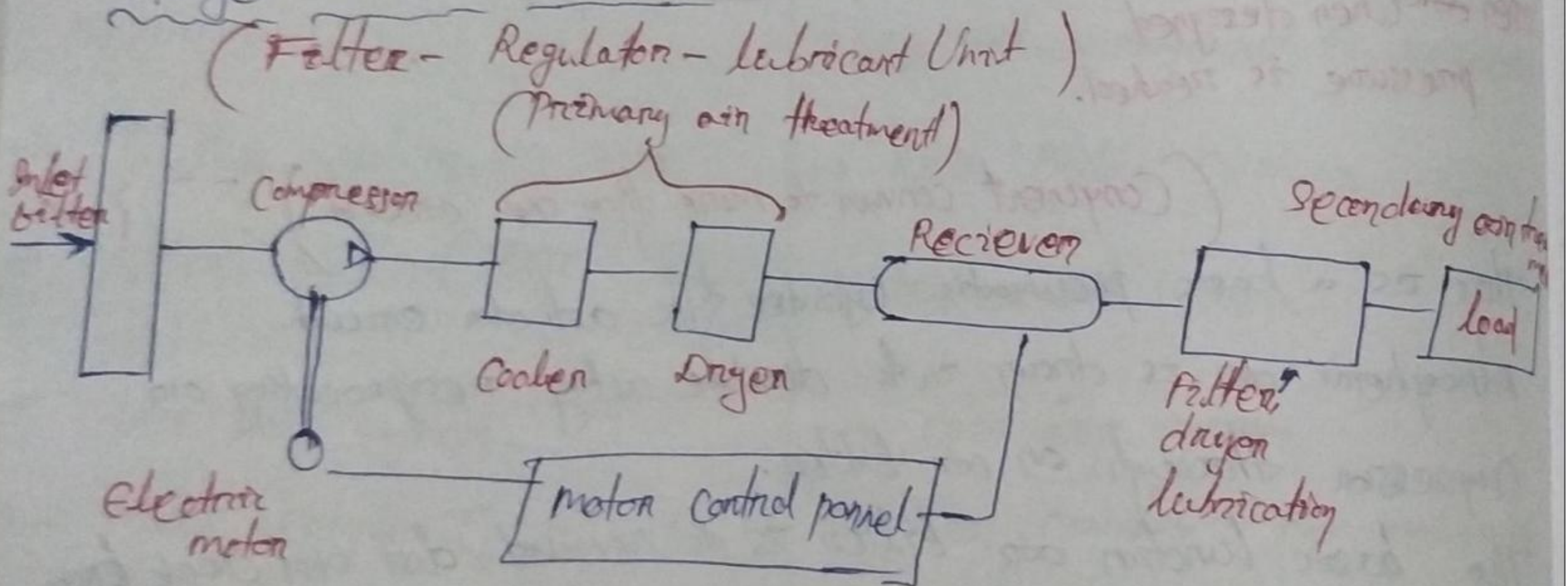
open when designed pressure is reached.

(Component common to more than one circuit)

- This is a basic pneumatic cylinder type actuator circuit.
- Atmospheric air is drawn into double acting reciprocating air compressor through an air filter.
- The basic function air filter is to remove dust and dirt from air.
- Double acting reciprocating air compressor raises the pressure typically in the range of 7-15 pressure ratio. The compression process is associated with ^{raise} the temperature.
- The remove water vapour from compressed air cooling arrangement is done. Thus water vapour condense to liquid water and is collected. Along with this little brake lubricant is added to the compressed air in this air cooler and air treatment unit.

- High pressure air is storage reservoir. When the desired pressure is raised in the reservoir, the pressure switch opens thus disconnected electric motor and air compressor.
- Here the actuator is cylinder which can raised or lower in load. High pressure air is communicated to double acting cylinder actuator through a control valve.
- Unlike actuator hydraulic circuit here the compressed air is vented to atmosphere.

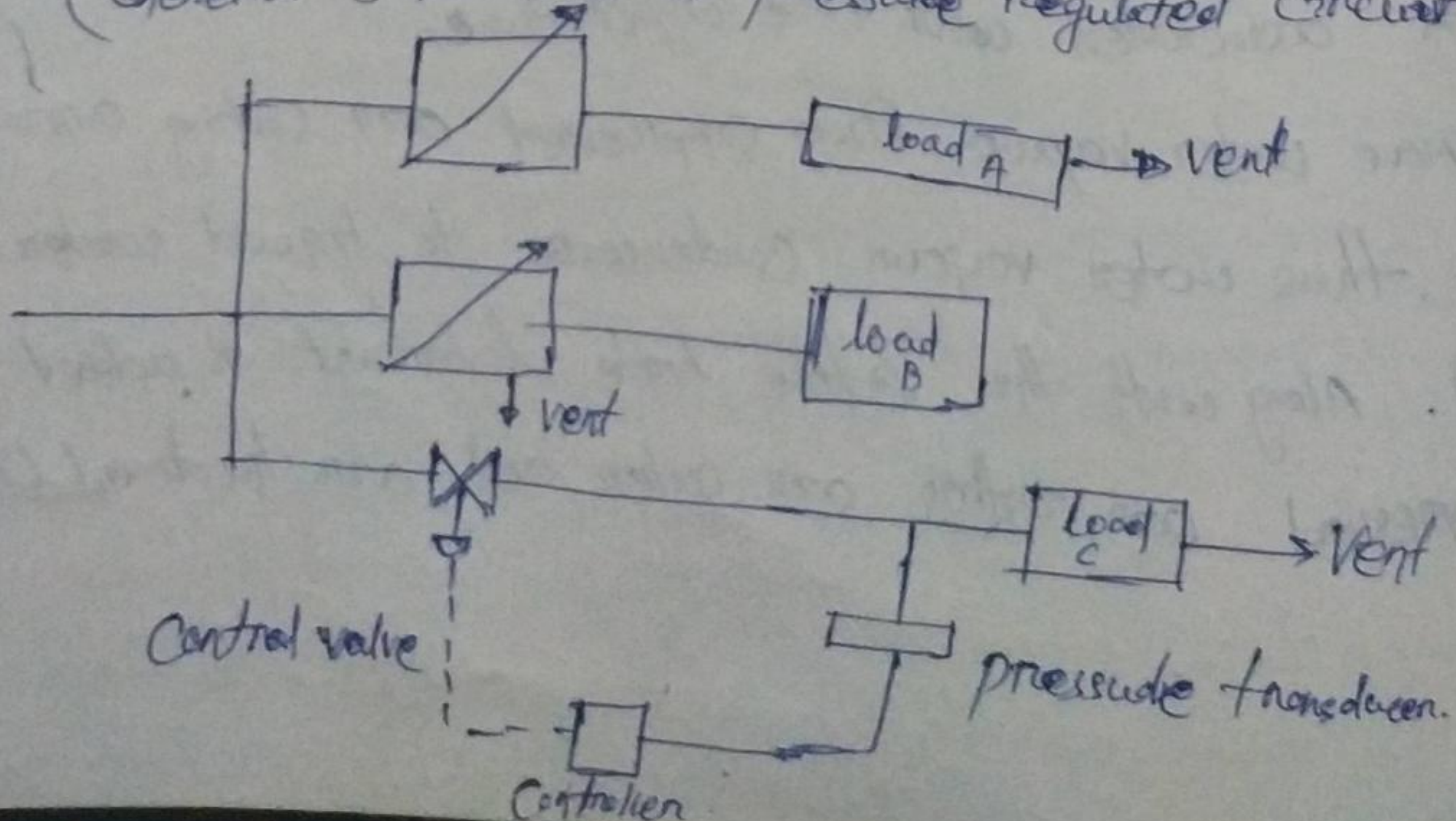
Stages of air treatment :-



(Typical layout of a Filter, lubrication, regulator power unit.)

* Pressure regulation in pneumatic circuit :-

(Generic 3 varieties of pressure regulated circuit.)



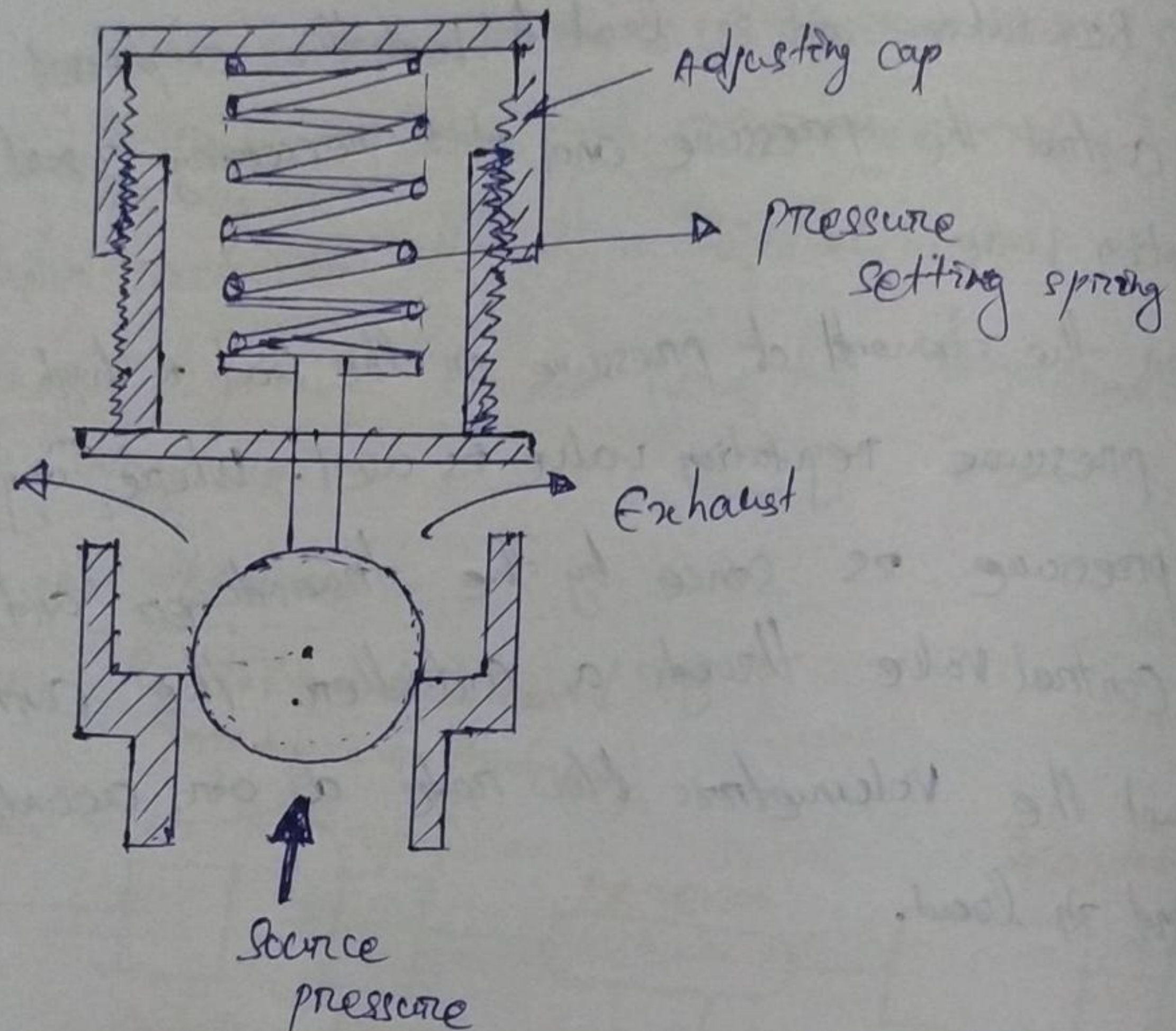
→ pressure regulator used in load A is necessary control the pressure by restricting volumetric flow rate of air after doing the activities, in load vent to atmosphere.

→ The Regulator used in load B vent the compressed air to atmosphere thus control the pressure and it is necessary 3 port pressure regulation valve.

→ When the demand of pressure in the load is high transducer type pressure regulation valve is used. Where any alternation in pressure is sense by the transducer and communicated to control valve through a controller. The controller valve adjust the volumetric flow rate of air, according to pressure demand in load.

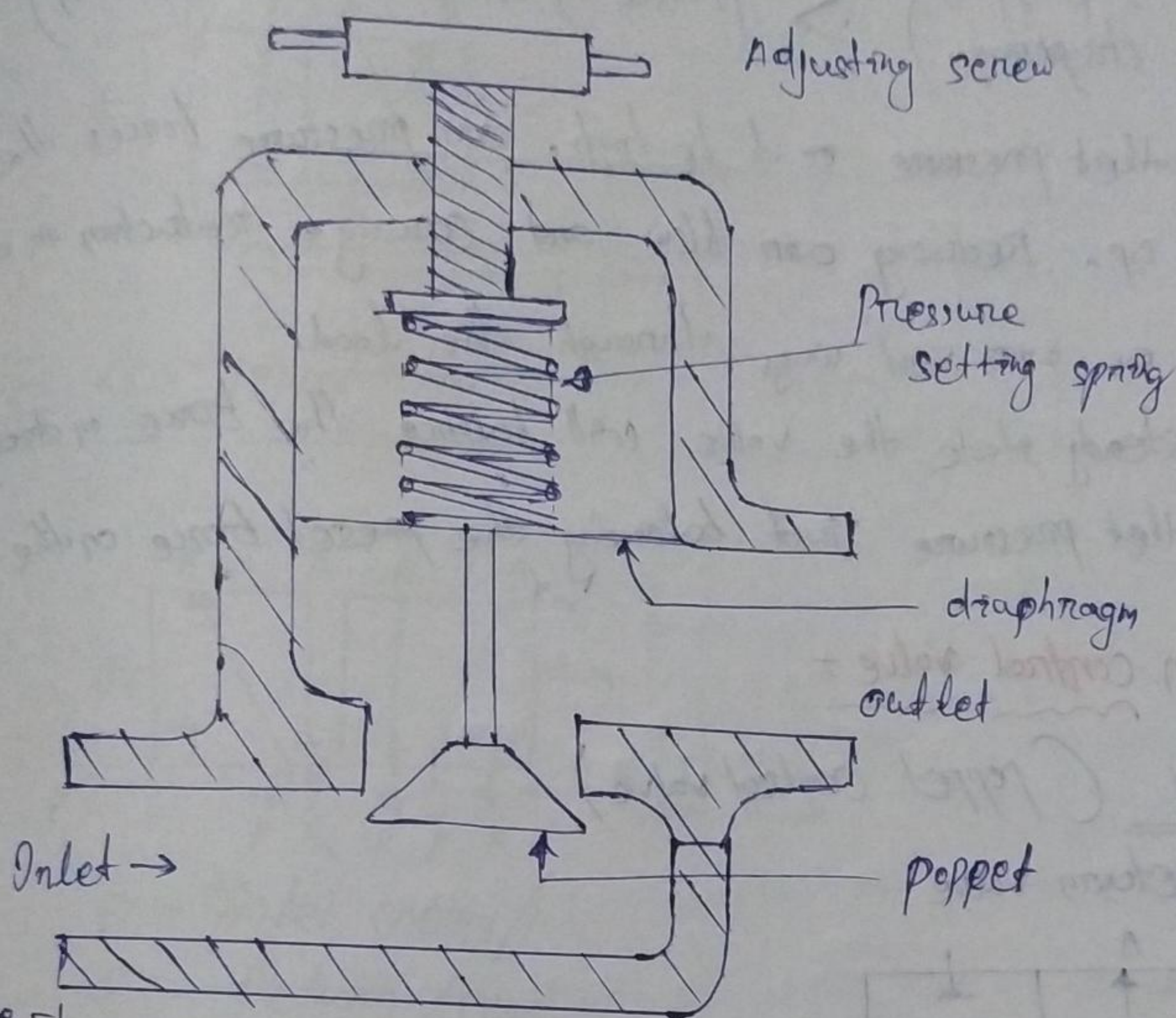
* Pressure control valve +

(i) Relief valves +



- A simplest pressure regulating valve is a ball valve.
- The ball held in position by pressure setting spring.
- When pressure from the pipe line from tank rises force exhausted by compressed air is more and it lift the ball up against the spring force. Thus relieving high pressure air to exhaust on atmosphere.

Non-relieving pressure regulators



Case - 1

→ (Pressure exerted by the spring on diaphragm) = (Force exerted by outlet air on diaphragm.)

* This is the equilibrium condition, where poppet is in closed position delivering constant quantity of air from inlet to outlet.

→ outlet pressure sensed by diaphragm which is preloaded by a pressure setting spring.

Case - 2

(Force by outlet air on diaphragm) > (Force by spring on diaphragm)

→ outlet pressure sensed by diaphragm which is preloaded by a pressure setting spring.

→ The spring force on diaphragm and poppet down, opening the valve to admit more air and raise output pressure.

Case-3

(Force by spring by diaphragm) > (Force by outlet air on diaphragm)

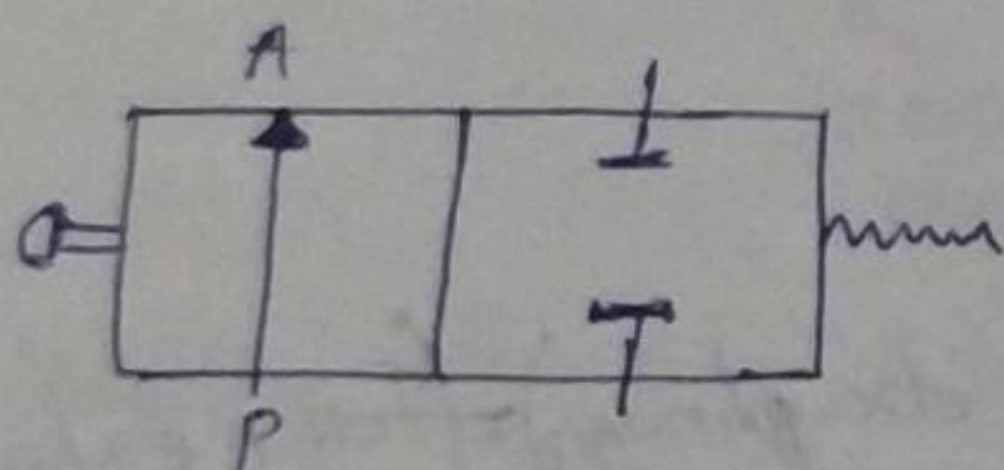
→ The outlet pressure is to be high, air pressure forces the diaphragm up, reducing air flow and causing a reduction in air pressure as air vent ways through the load.

→ In a steady state the valve will balance the force on diaphragm from outlet pressure just balancing the preset force on the spring.

* Direction control valve +

* $\frac{2}{2}$ DCV (poppet control valve.)

→ Non return valve

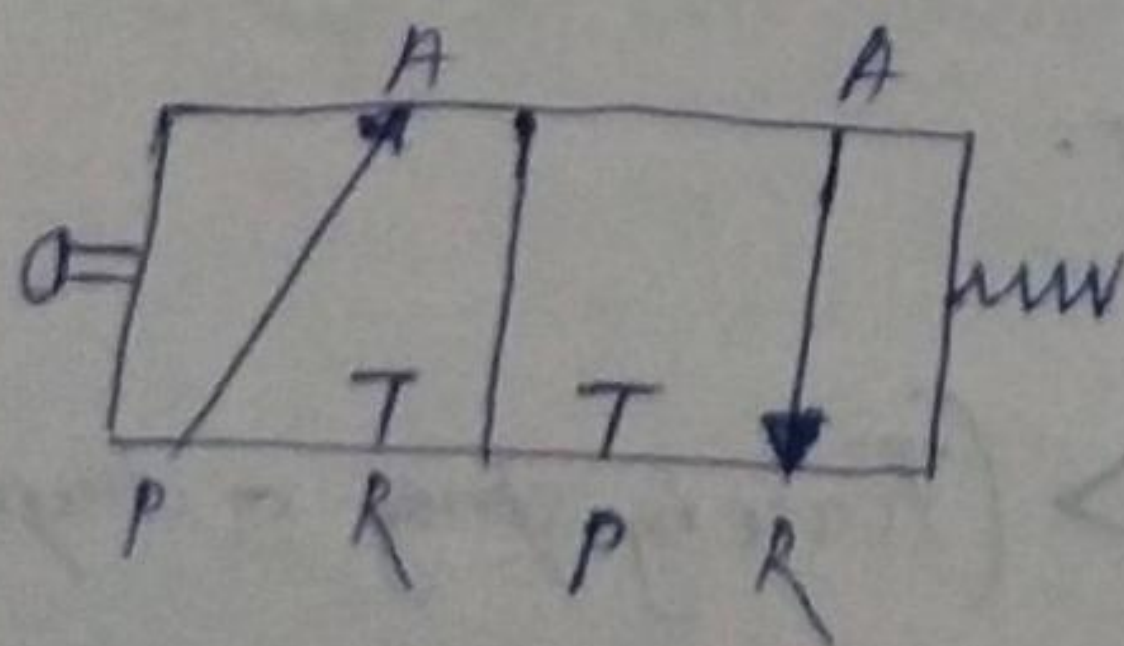


Next point $\frac{2}{2}$ valve (poppet type)

No. of position

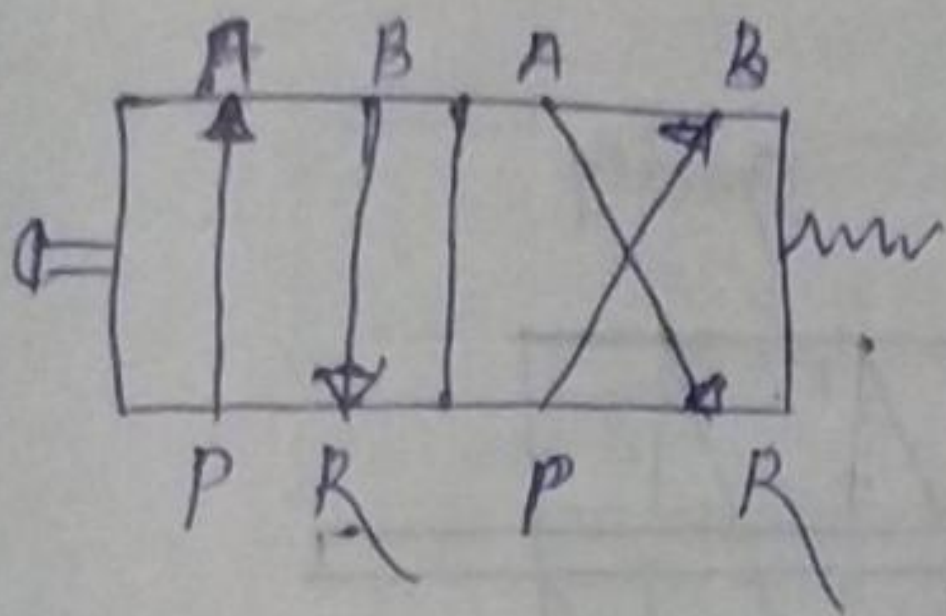
→ Poppet pushed bottom activated - spring returned.

* $\frac{3}{2}$ DCV poppet type:-

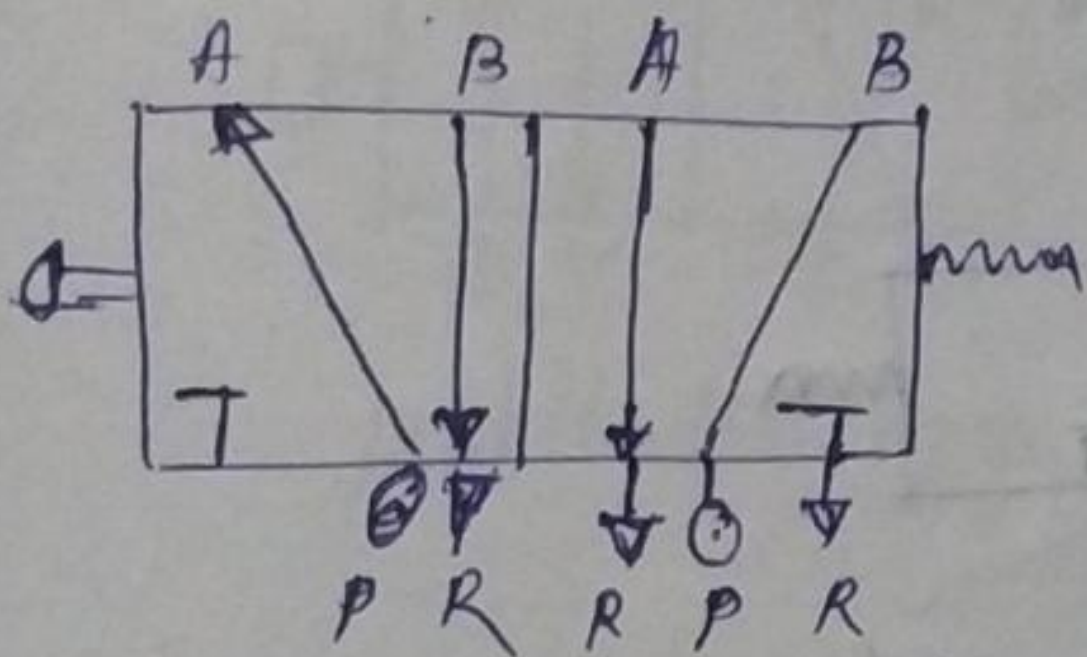


$\frac{3}{2}$ poppet push button activated spring return

* $\frac{4}{2}$ DCV poppet type



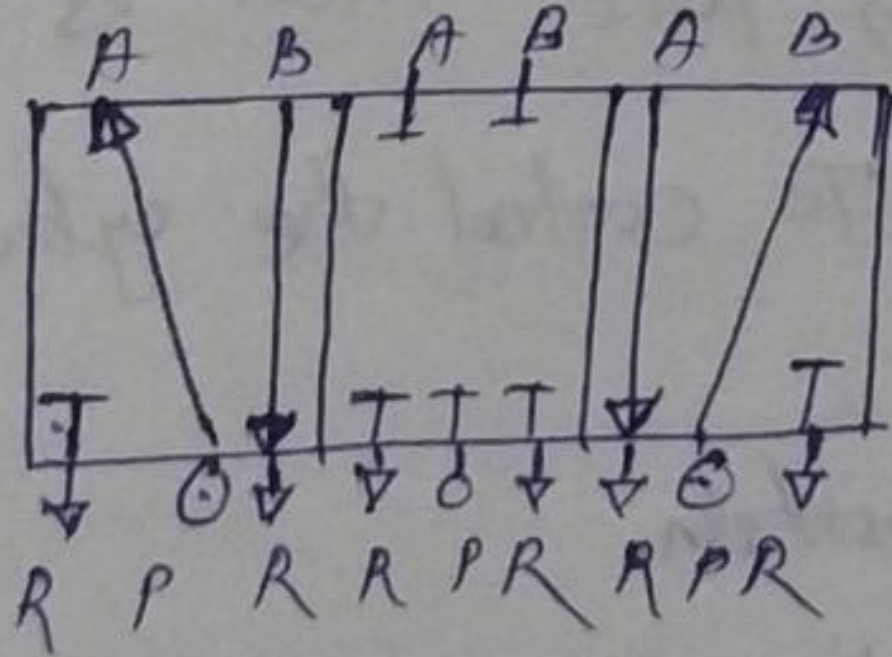
* $\frac{5}{2}$ DCV poppet type



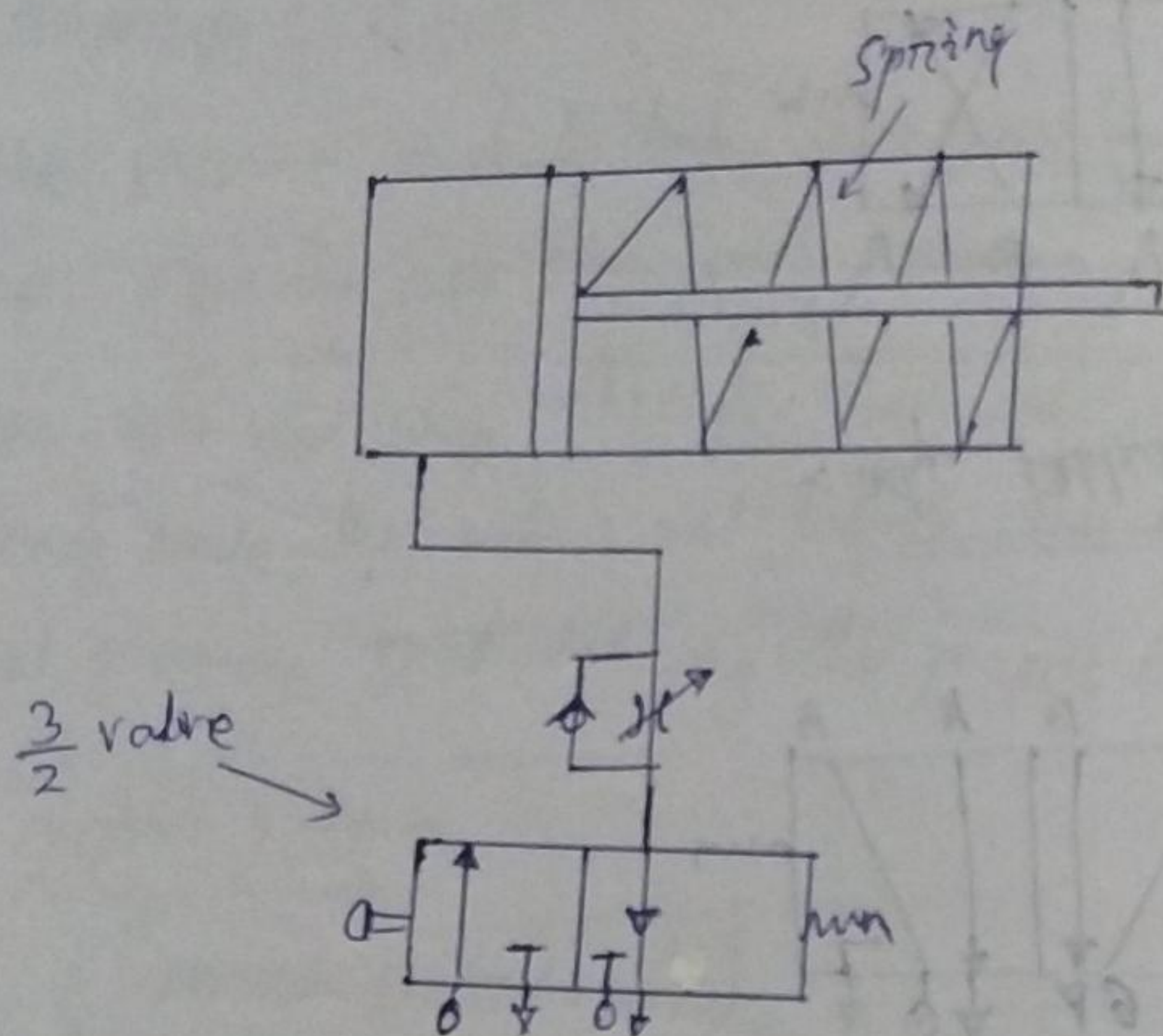
$\ominus$ = Inlet energy/p

∇ = Return/vent

* $\frac{5}{3}$ DCV spool valve

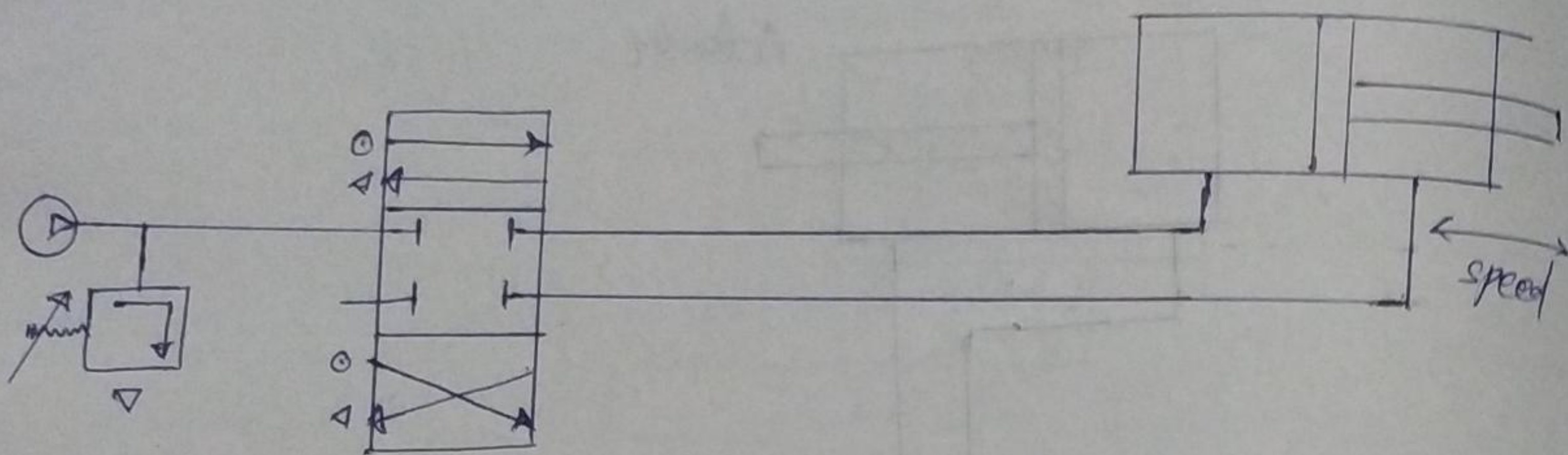


* Direction control of single acting cylinder :-

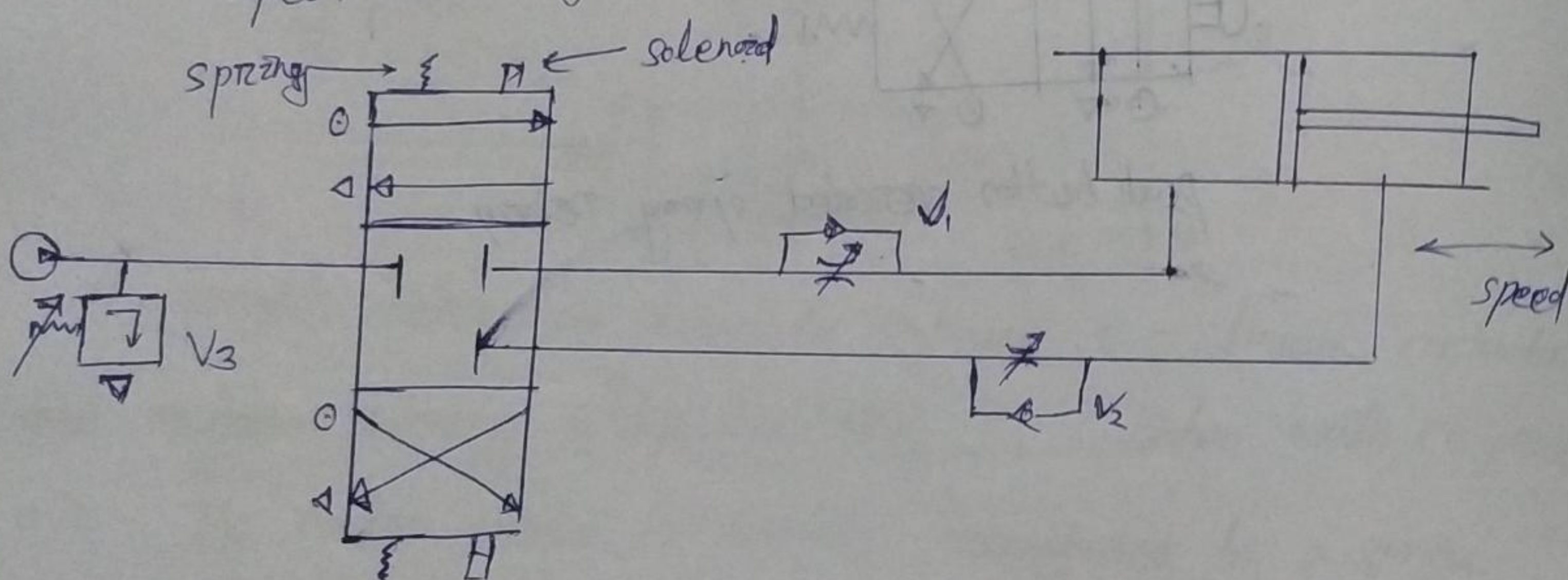


- A single acting pneumatic cylinder is a linear actuator and realizes working stroke by filling the cylinder with compressed air. The return stroke is usually accomplished by a spring.
- The cylinder has one connection port that is used to fill or vent the cylinder. To control the cylinder 3/2 valve is used.
- 3/2 mean 3 port and 2 position.
- one port connected to source the source of compressed air, one port is required as exhaust and the third port connected to the cylinder.
- The valve are two position - Filling or venting the cylinder.

* operation of double acting cylinder with metering in / metering out control.



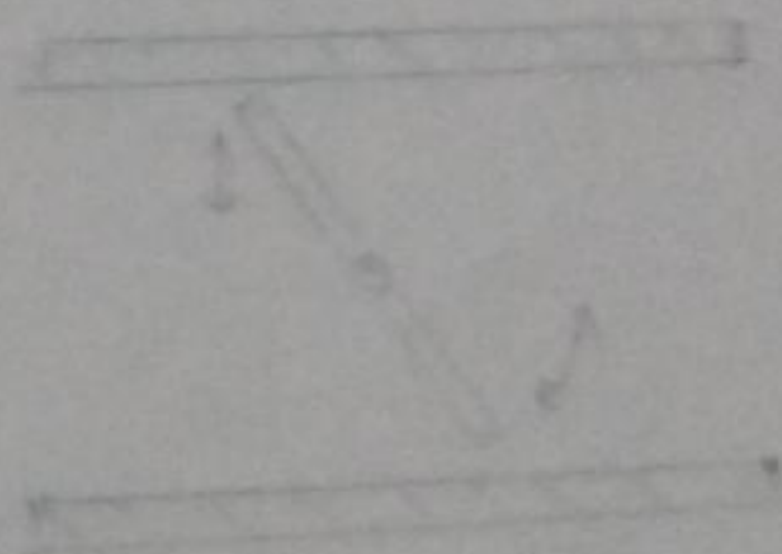
speed control by volume of compressed air.



speed control of pneumatic double acting cylinder with the metering in and metering out.

- The pump output is higher than needed by any individual actuator so, a flow restriction is used to set the flow to reach actuator. This is known as the meter in circuit. and force is needed to move a load.
- Check valve V_1 is gives a full speed retraction and check valve V_2 provided a small back pressure to avoid load running away. The full pump delivery is produced when the pressure reaches the setting of relief valve V_3 . So there is a waste of energy and unnecessary production of heat in the fluid.

- If the load can run away from the actuator, the third speed control method. motor and circuit must be used.
- The again give a controlled extension speed and full retracting speed (V₁). As before the pump delivery fluid at a pressure set by a relief valve, leading to heat generation.



* Flow control valve

→ All valve work by putting a variable restriction along the flow valve.

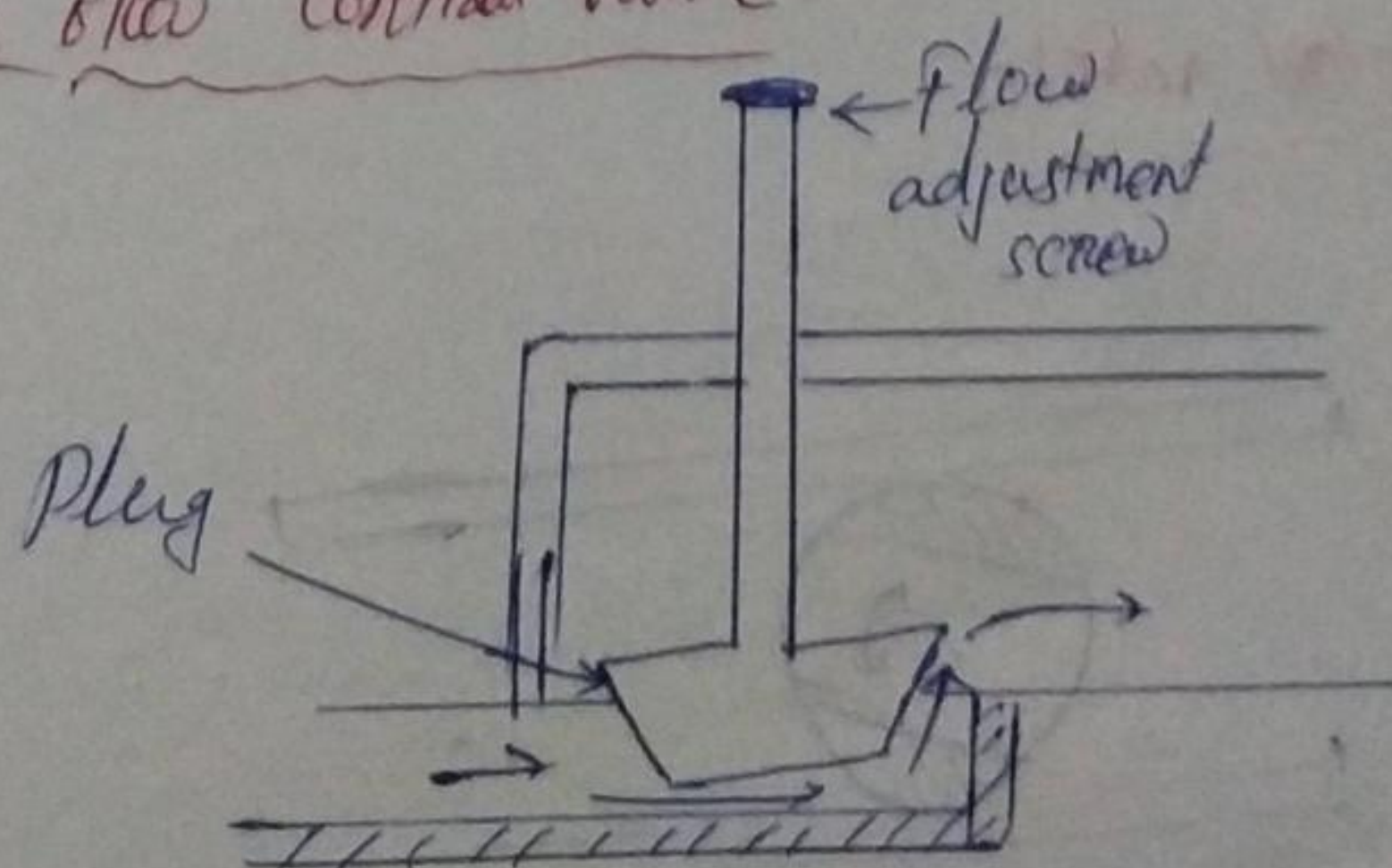
⇒ Three basic type of flow control valve.

(i) plug type.

(ii) Butterfly type.

(iii) Ball type.

(i) plug type flow control valve:



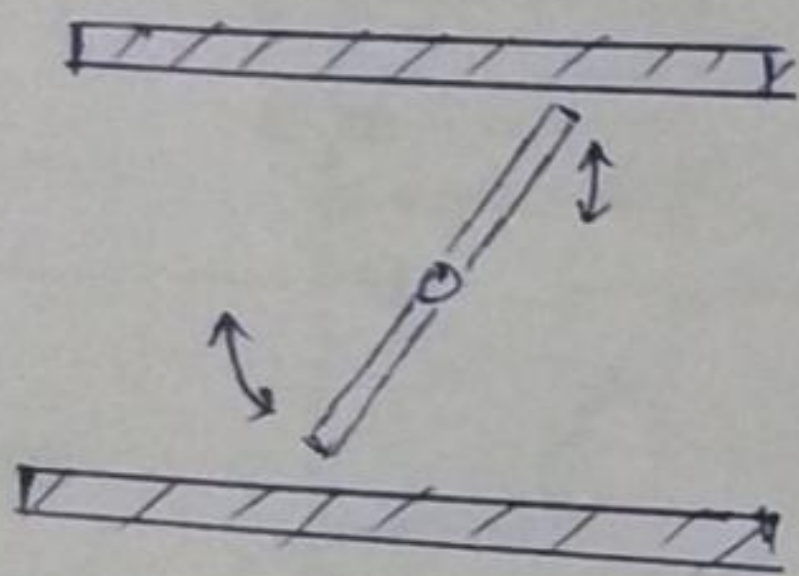
→ The plug valve is commonly used valve. It is also termed as glove valve.

→ This valve has plug which can be adjust in vertical direction by setting flow adjustment screw.

→ The adjustment of plug control the fluid flow in the pipe line.

→ The valve body is made of glass or rubber. The plug is made of plastic and or glass.

(ii) Butterfly type flow control valve



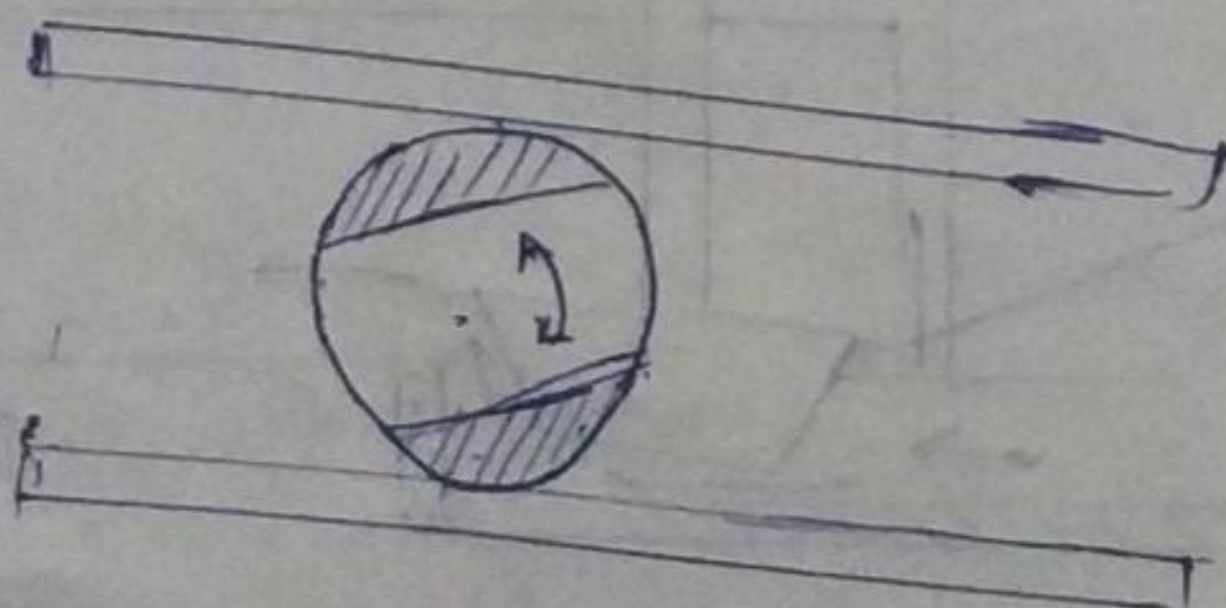
→ It consists of a disc which can rotate inside the pipe.

→ Butterfly valve can be made to any size and is widely used to the flow control valve gas.

→ Butterfly valve has favoured because of their lower cost and higher cost and lighter weight.

→ The disc always present in the flow therefore a pressure drop is induced regardless of the valve position.

(iii) Ball type flow control valve



2 Ball valve are also used where simultaneous operating and closing of ~~three~~ two or three valve by quick action is needed.