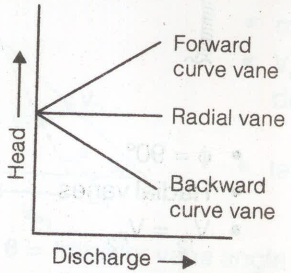
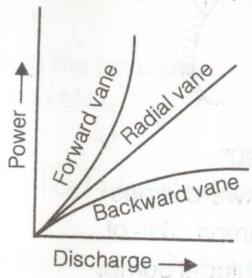
15. HYDRAULIC PUMPS

CENTRIFUGAL PUMP

- Centrifugal pump is reverse of inward flow reaction turbine. It works on principle of forced vortex motion. It has high discharging capacity and can be used for lifting highly viscous liquids e.g. sewage water, chemicals etc.
- Priming is an operation in which liquid is completely filled in the chamber of pump so that air or gas or vapour from the portion of pump is driven out & no air pocket is left.
- In volute pump cross sectional area results in developing a uniform velocity throughout the casing & free vortex is formed.
- Centrifugal pump has high output and high efficiency.
- Head Vs discharge and Power Vs discharge relationship





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Types of Pump	Range of Head
Low head pump	upto 15m head
Medium head pump	15 m to 40 m
High head pump	above 40 m

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Pump	Specific Speed
Radial Flow	10 to 80
Mixed Flow	80 to 160
Axial Flow	160 to 450

- The specific speed of a centrifugal pump may be defined as the speed in revolution per minute of a geometrically similar pump of such a size that under corresponding conditions it would deliver 1 liter of liquid per second against of a head of 1m.

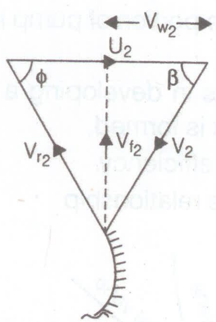
For multi stage $(H_m) = \dfrac{\text{Total head}}{\text{No. of stage}}$

Specific speed $(N_s) = \dfrac{N\sqrt{Q}}{(H_m)^{3/4}}$

- For optimum efficiency impeller should be designed such that whirl velocity at inlet is zero. It means discharge should enter in the pump radially $(V_{w_1} = 0)$.
- **Velocity Triangle**

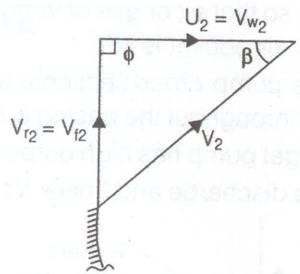
(i) At Outlet

(a)



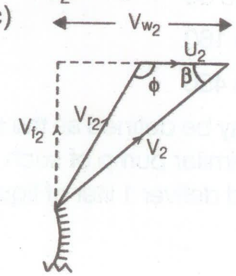
- $\phi < 90^\circ$
- Backward vanes
- Common case of centrifugal pump.
- More efficient
- $\tan\phi = \dfrac{V_{f_2}}{U_2 - V_{w_2}}$
- $V_{F_2} = V_2 \sin\beta.$
- $V_{w_2} = V_2 \cos\beta.$

(b)



- $\phi = 90^\circ$
- Radial vanes
- $V_{r_2} = V_{F_2}$
- $U_2 = V_{w_2}$

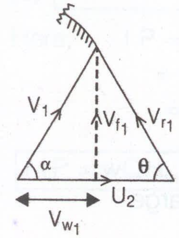
(c)



- $\phi < 90^\circ$
- Forward vanes
- $\tan(180^\circ - \phi) = \dfrac{V_{f_2}}{(V_{w_2} - U_2)}$
- $V_{f_2} = V_2 \sin\beta$
- $V_{w_2} = V_2 \cos\beta$

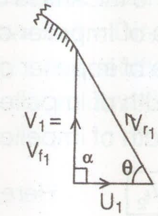
(ii) At Inlet

(a)



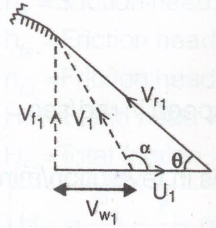
- $\alpha < 90^\circ$
- V_{w_1} is in the direction of U_1
- $\tan\theta = \dfrac{V_{f_1}}{U_1 - V_{W_1}}$
- $V_{w_1} = V_1 \cos\alpha$
- $V_{F_1} = V_1 \sin\alpha$

(b)



- $\alpha = 90^\circ$
- $V_{w_1} = 0$
- $\tan\theta = \dfrac{V_{f_1}}{U_1} = \dfrac{V_1}{U_1}$
- Common case of centrifugal pump.

(c)



- $\alpha > 90^\circ$
- V_{w_1} is in the opposite direction of U_1
- $\tan\theta = \dfrac{V_{f_1}}{|U_1| + |V_{w_1}|}$

Here, θ = Impeller vane angle of inlet (Angle between V_{r_1} and U_1)
φ = Impeller vane angle of outlet (Angle between V_{r_2} and U_2)
α = Angle between V_1 and U_1 at Inlet (Blade Angle)
β = Angle between V_2 and U_2 of outlet (Blade Angle)
V_1 = Absolute velocity of water at inlet
V_2 = Absolute velocity of water at Inlet.

V_{r1} and V_{r2} are relative velocity of water at Inlet and outlet respectively. U_1 and U_2 are tangential also called circumferential velocity at inlet and outlet respectively.

- **Design parameters of pump**

1. (a) $A_{f_1} = \pi D_1 B_1$ ... neglecting vane thickness
$= (1-k)\pi D_1 B_1$

Here, A_{f_1} = Area of flow at inlet
k = Vane thickness coefficient ≃ 5%

(b) $A_{f_2} = \pi D_2 B_2$ ... neglecting vane thickness
$= (1-k)\pi D_2 B_2$

Here, A_{f_2} = Area of flow of outlet
k = Vane thickness coefficient
D_1 = Dia of Impeller of Inlet
D_2 = Dia of impeller of outlet
B_1 = Width of impeller of inlet
B_2 = Width of impeller of outlet.

2. $\boxed{Q = A_{f_1} V_{f_1} = A_{f_2} V_{f_2}}$ Here, Q = Discharge

V_{f_1} and V_{f_2} is velocity of flow at inlet and outlet respectively.

3. Number of vanes may be 6 to 12.

4. $\boxed{\phi = \dfrac{U_2}{\sqrt{2gHm}}}$ Here, ϕ = Speed ratio
$g = 9.81\ m/s^2$
H_m = Manometric Head.

5. $\boxed{\psi = \dfrac{V_{f2}}{\sqrt{2gHm}}}$ Here, ψ = Flow ratio

6. $\boxed{\dfrac{D_2}{D_1} \simeq 0}$

7. $\boxed{\omega_{win} = \sqrt{\dfrac{8gHm}{D_2^2 - D_1^2}}}$ ω = Angular speed in rad/sec.

$= \dfrac{2\pi N}{60}$ 'N' is in revolution/min (r.p.m)

8. If $D_2 = 2D_1$ than $\boxed{D_2 = \dfrac{10.23}{\omega}\sqrt{H_m}}$

- **Powers In Pump**

(i) $\boxed{S.P = T\omega}$ Here, S.P = Shaft power

$$\omega = \frac{2\pi N}{60}\ \text{rad/sec.}$$

ω = Angular speed of shaft
T = Torque produced in the shaft.
N in r.p.m (revolution per minute)

(ii) (a) $\boxed{I.P = \dfrac{wQ}{g}[V_{w_2}U_2 - V_{w_1}U_1]}$

(b) $\boxed{I.P|_{max} = \dfrac{wQ}{g}[V_{w_2}U_2]}$ [Why $V_{w_1} = 0$]

(c) $\boxed{I.P = S.P - \text{Mechanical Frictional losses}}$

Here, I.P = Impeller power, θ = Discharge, $\omega = \rho g$
$= \gamma$ = unit weight or specific weights
= weight per unit volume (kN/m³)

(iii) $\boxed{M.P = wQH_m}$ ... kW

$$= \frac{\rho Q H_m}{75} \ \ldots \text{H.P}$$

Here, M.P = Manometric power,
$\omega = \rho g = \gamma$ in kN/m³ = Specific weight
H_m = Manometric head, ρ = Density in kg/m³.

- **S.P > I.P > M.P**
- **Manometric Head (H_m)**

(i) $H_m = h_s + h_d + h_{fs} + h_{fd} = H_s + H_d$
h_s = Suction head, h_d = delivery head
h_{fs} = Friction head loss in suction pipe
h_{fd} = Friction head loss in delivery pipe
Hs = Static head = $h_s + h_d$.
H_F = Total friction loss = $h_{fs} + h_{fd}$.

(ii) $\boxed{H_m = \dfrac{P_d}{\rho g} - \dfrac{P_s}{\rho g} \text{ or } \dfrac{P_d}{\gamma} - \dfrac{P_s}{\gamma}}$ P_d = Pressure in delivery pipe
P_s = Pressure in suction pipe.

(iii) $\boxed{H_m = \dfrac{V_{w_2} \cdot U_2}{g}}$

- **Efficiencies of the Pump**

(i) $\boxed{\eta_{mech.} = \dfrac{I.P}{S \cdot P}}$ where, η_{mech} = Mechanical efficiency

(ii) $\boxed{\eta_{man} = \dfrac{M \cdot P}{I \cdot P}}$ where, η_{man} = Manometric efficiency

$$= \frac{gH_m}{(V_{w_2}U_2 - V_{w_1}U_1)}$$

(iii) $\boxed{\eta_o = \dfrac{M \cdot P}{S \cdot P}}$ where, η_o = Overall efficiency
$\eta_o = \eta_{mech} \cdot \eta_{vol} \cdot \eta_{max}$
$\eta_o = \eta_{mech} \cdot \eta_{man}$ [where η_{vol} is neglected]

(iv) $\boxed{\eta_{vol} = \dfrac{Q - \Delta Q}{Q}}$ where, η_{vol} = Volumetric efficiency.

- **Model Relationship for Pumps**

Dimensionless Parameter	Dimensional Parameter
(i) $C_H = \dfrac{gH}{\omega^2 D^2}$	(i) $C_H = \dfrac{H}{N^2 D^2}$
(ii) $C_Q = \dfrac{Q}{\omega D^3}$	(ii) $C_Q = \dfrac{Q}{ND^3}$
(iii) $C_P = \dfrac{P}{\rho \omega^3 D^5}$	(iii) $C_P = \dfrac{P}{N^3 D^5}$
(iv) $N_S = \dfrac{\omega\sqrt{Q}}{(gH)^{3/4}}$	(iv) $N_S = \dfrac{N\sqrt{Q}}{(H)^{3/4}}$

Where, C_H = Head coefficient, C_Q = Discharge coefficient

C_P = Power coefficient, N_S = Specific speed of pump

P = Power

- Overall efficiency of pump(η_o) = $\dfrac{C_Q \cdot C_H}{C_P}$
- Net Positive Suction Head (NPSH)
- $\text{NPSH} = \left(\dfrac{P_{atm}}{\rho g} - h_s - h_{fs}\right) - \dfrac{P_v}{\rho g}$

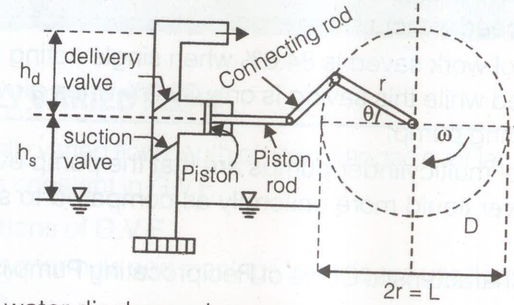
where, $\dfrac{P_{atm}}{\rho g}$ = Atmospheric pressure head $\simeq$ 10.3 m at mean sea level.

$\dfrac{P_v}{\rho g}$ = Vapour pressure head $\simeq$ 2.5 m at 20 °C for water and

10.3 m at 100 °C for water

- Thomas Cavitation Number $(\sigma) = \dfrac{\text{NPSH}}{H}$
- Critical Thomas number $(\sigma_c) = 1.042(H)^{4/3}$
- For no cavitation $\sigma \geq \sigma_c$ or NPSH $\geq \sigma_c$H.

RECIPROCATING PUMP



Volume of water discharged per second,

$$Q = \frac{ALN}{60}\ \text{m}^3/\text{sec}$$

A = Area of cylinder (in m²)

L = Length of cylinder (in m)

N = Crank speed (in rpm)

- If the head against which water is to be lifted is

$H_s = (h_s + h_d)$

h_s = suction head

h_d = delivery head

- Work done per second = $\gamma Q(h_s + h_d)$
- Reciprocating pumps are used to lift water against high head at low discharge.
- To increase discharge and to maintain it more uniform, double acting reciprocating pumps are used.

$Q \approx \dfrac{2ALN}{60}$, thus power also gets doubled.

- Slip in Percentage is given by

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

where C_d = coefficient of discharge

Slip is negative when (i) delivery pipe is small and suction pipe is long (ii) Pump is running at very high speed.

- **Indicator diagram** is a graph between the pressure head in the cylinder and the distance travelled by the piston flow inner dead centre for one complete revolution of the crank, work done by pump is proportional to area of indicator diagram.

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(iv) $N_S = \dfrac{\omega\sqrt{Q}}{(gH)^{3/4}}$	(iv) $N_S = \dfrac{N\sqrt{Q}}{(H)^{3/4}}$

Where, C_H = Head coefficient C_Q = Discharge coefficient
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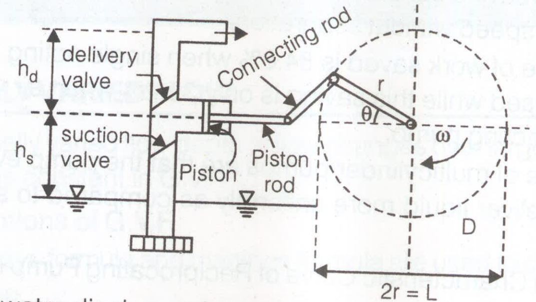

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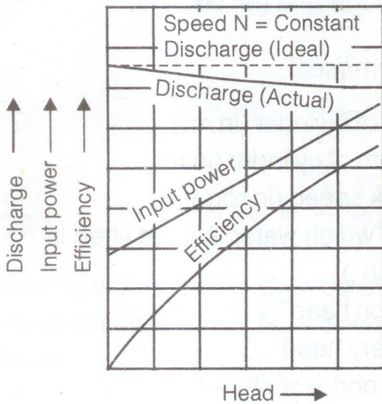
- **Indicator diagram** is a graph between the pressure head in the cylinder and the distance travelled by the piston flow inner dead centre for one complete revolution of the crank, work done by pump is proportional to area of indicator diagram.

- **Air Vessel** is used to obtain continuous supply of water at uniform rate, to save a considerable amount of work and to run the pump at a high speed without separation.

- Percentage of work saved is 84.8% when single acting pump with air vessel is used while this saving is only 39.2% when air vessel is used in double acting pump.
- Advantage of multicylinder pumps are that the pump even without air vessels deliver liquid more uniformly as compared to single cylinder pump.
- Operating Characteristic Curve of Reciprocating Pump is given below:



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