

Centrifugal PumpsIntroduction :

- The hydraulic machines which convert the mechanical energy into hydraulic energy are called Pumps.
- The hydraulic energy is in the form of pressure energy.
- If the mechanical energy is converted into pressure energy by means of Centrifugal force acting on the fluid, the hydraulic machine is called "Centrifugal pump".
- The centrifugal pump acts as a reverse of an inward radial flow reaction turbine. This means that flow in centrifugal pumps is in the radial outward directions.
- The centrifugal pump works on the principle of forced vortex flow which means that when a certain mass of liquid is rotated by an external torque, the rise of pressure head of the rotating liquid takes place.
- The rise in pressure head at any point of the rotating liquid is proportional to the square of the tangential velocity of the liquid at that point (i.e., rise in Pressure Head = $\frac{V^2}{2g}$)
- Thus at the outlet of the impeller, where radius is more, the rise in pressure head will be more and the liquid will be discharged at the outlet with a high pressure head. Due to this high pressure head, the liquid can be lifted to a high level. Ex: Sewage water, Chemicals etc.

Centrifugal Pump

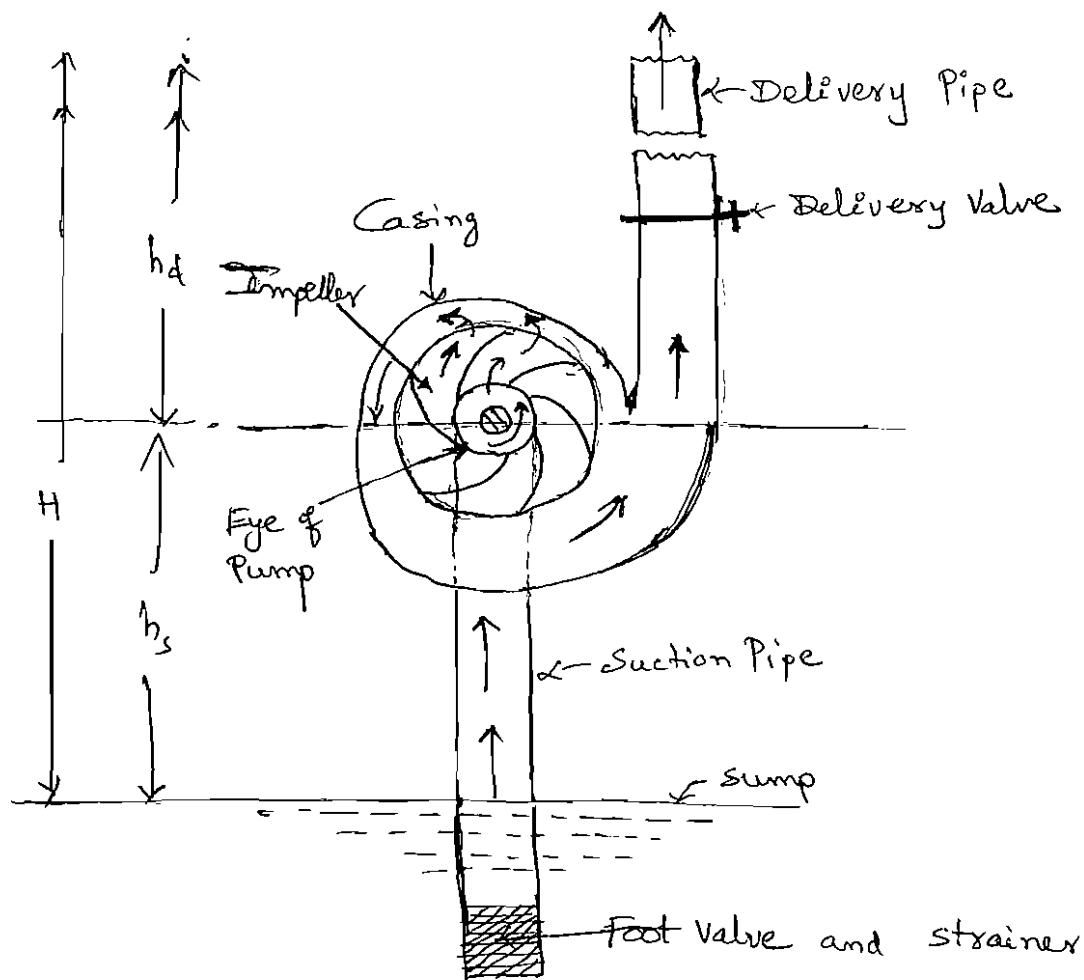


Fig1: Main Parts of a Centrifugal Pump

- Priming is an operation in which liquid is completely filled in the chamber of pump so that air or gas or vapour from the portion of the pump is driven & no air pocket is left.
- In volute pump cross sectional area results in developing a uniform velocity throughout the casing & free vortex is formed.
- Centrifugal pump has high output and high efficiency.

Main Parts of a Centrifugal Pump:

The following are the main parts of a centrifugal pump.

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

1. Impeller:- The rotating part of a centrifugal pump is called 'Impeller'. It consists of a series of backward curved vanes. The impeller is mounted on a shaft which is connected to the shaft of an electric motor.

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

The following three types of casings are commonly adopted:

(a) Volute Casing as shown in fig. 1

(b) Vortex Casing as shown in fig. 2

(c) Casing with guide blades as shown in fig. 3.

② Volute Casing: Fig. 1 shows the volute casing, which surrounds the impeller. It is of spiral type in which

area of flow increases gradually. The increase in area of flow decreases the velocity of flow. The decrease in velocity increases the pressure of water flowing through the casing. It has been observed that in case of Volute Casing, the efficiency of the pump increases slightly as a large amount of energy is lost due to the formation of eddies in this type of casing.

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

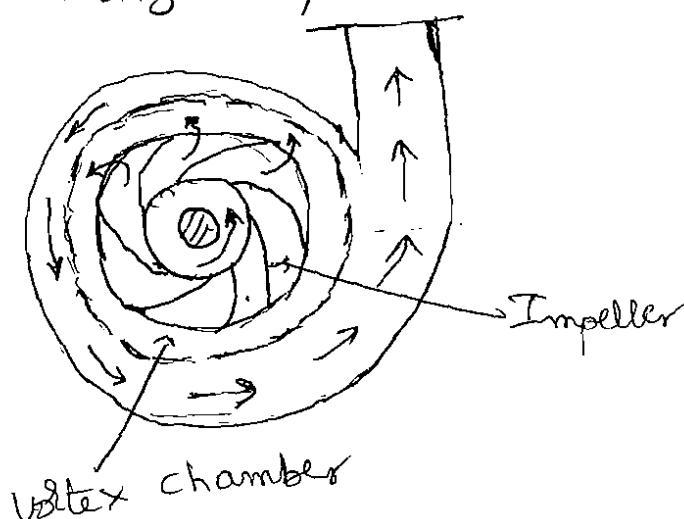


fig 2 Vortex Casing.

(c) Casing with Guide Blades: This casing is shown in

Fig. 3. in which the impeller is surrounded by a series of guide blades mounted on a ring which is known as diffuser. The guide vanes are designed in such a way that the water from the impeller enters the guide vanes without shock. Also the area of the guide vanes increases, thus reducing the velocity of flow through guide vanes and consequently increasing the pressure of water.

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

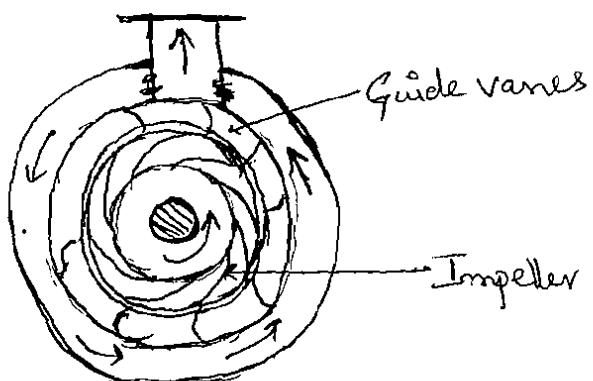


Fig 3. Casing with Guide Blades.

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

* Workdone by the Centrifugal Pump (Or By Impeller) on Water

In case of the centrifugal pump, work is done by the impeller on the water. The expression for the workdone by the impeller

on the water is

obtained by drawing

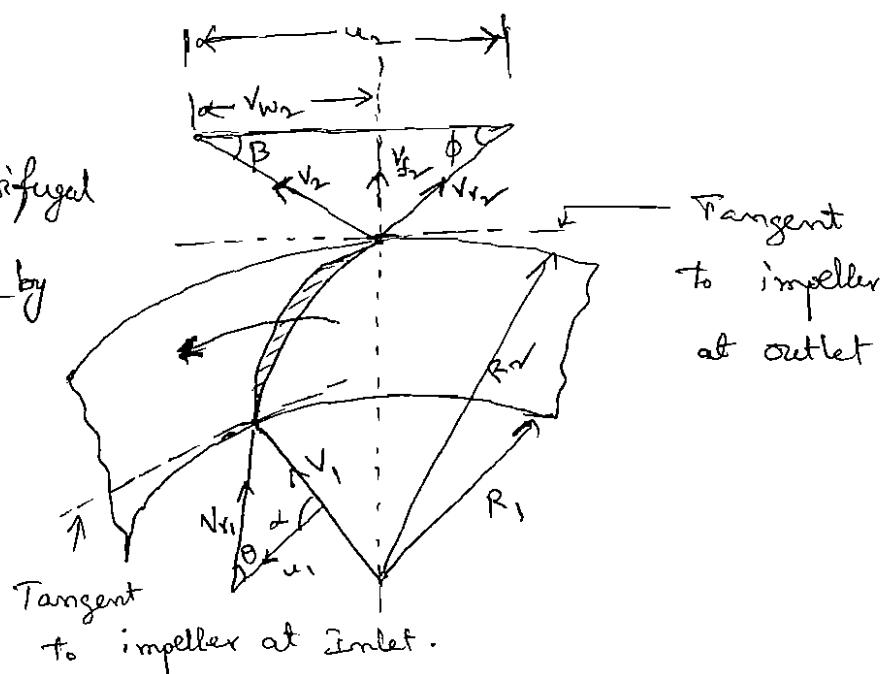


Fig 4: Velocity triangles at inlet & outlet.

velocity triangles at inlet and outlet of the impeller in the same way as for a turbine. The water enters the impeller radially at inlet for the best efficiency of pump, which means the absolute velocity of water at inlet makes an angle of 90° with the direction of motion of the impeller at inlet. Hence angle $\alpha = 90^\circ$. and $N_w = 0$. for drawing the velocity triangles, the same notations are used as that for turbines. fig. 4 shows the velocity triangles at the inlet and outlet tips of the vanes fixed to impeller.

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

D_1 = Diameter of impeller at inlet

u_1 = Tangential velocity of impeller at inlet

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

D_2 = Dia. of the impeller at outlet

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

V_1 = Absolute velocity of water at inlet

V_{r_1} = Relative Velocity of water at inlet

α = Angle made by absolute velocity (V_1) at inlet with the direction of motion of vane.

β = Angle made by relative velocity (V_{r_1}) at inlet with the direction of motion of vane, ~~and~~.

N_2 , V_{r_2} , β and ϕ are the corresponding values at outlet.

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

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

\therefore work done by the impeller on the water per second
per unit weight of water striking per second

$$= - [\text{workdone in case of turbines}]$$

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

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

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

Workdone by impeller on water per second

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

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

where $Q = \text{Volume of water}$

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

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

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

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

UNIT-6

CENTRIFUGAL PUMPS & RECIPROCATING PUMPS

* Cavitation:

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

* precautions against cavitation:

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

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

* Effects of cavitation:

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

- iii) Due to sudden collapse of vapour bubble, considerable noise and vibrations are produced.
- iii) The efficiency of a turbine decreases due to cavitation. Due to pitting action, the surface of the turbine blade becomes rough and force exerted by water on the surface turbine blades decreases. Hence the work done by water or output horse power becomes less and thus efficiency decreases.

* Characteristic curves of pumps:

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

1. Main characteristic curves:

The main characteristics curves of a pump consists of variation of head, power and discharge with respect to speed. For plotting curves of manometric head versus speed, discharge is kept constant. For plotting curves of discharge versus speed, manometric head (H_m) is constant. And for plotting curves of power versus speed, the manometric head and discharge are kept constant.

Result: i) Power lost in the nozzle = 10.16 kW
 ii) Power lost due to hydraulic resistance
 in the runner = 10.69 kW.

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

A Characteristic Curves of Centrifugal pumps are defined as those curves which are plotted from the results of a number of tests on the Centrifugal Pump. These curves are necessary to predict the behaviour and performance of the pump when the pump is working under different flow rate, head and speed. The following are the important characteristic curves of pumps:-

1. Main characteristic curves
 2. Operating characteristic Curves, and
 3. Constant efficiency Or Muschel curves.
1. Main characteristic Curves:- It consists of variation of head (manometric head, H_m), Power and discharge with respect to speed.

For plotting the curves of discharge Versus speed ,

Manometric head (H_m) is kept constant

For plotting the curves of Power Versus speed , the manometric head and discharge are kept constant.

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

Fig. Shows main characteristic curves of a pump .

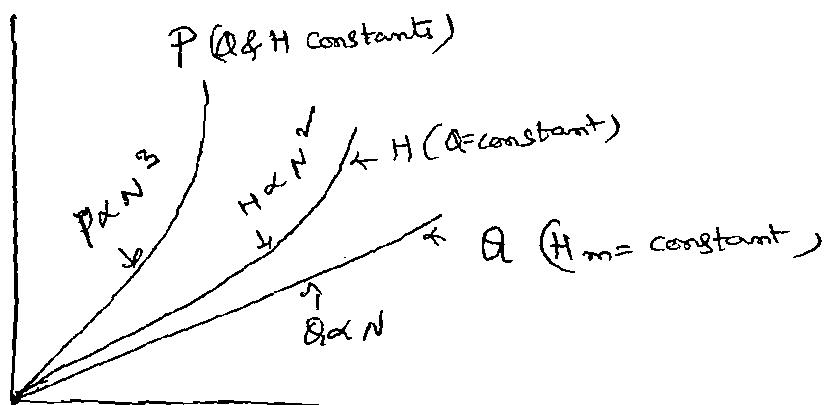
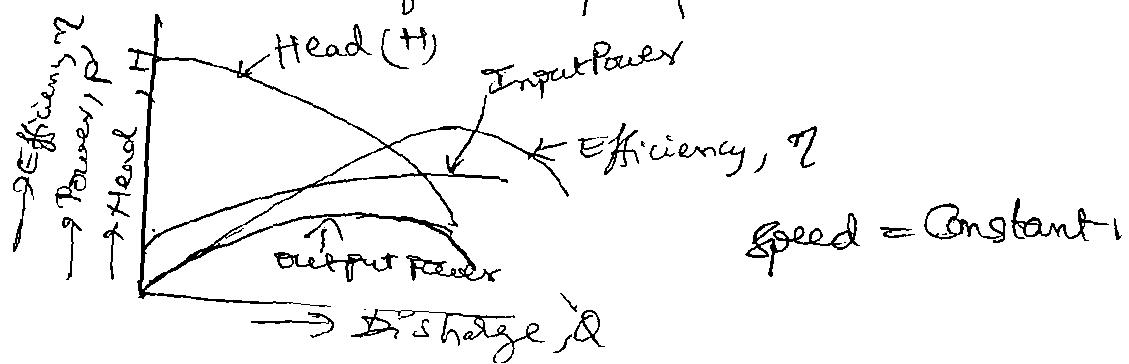


Fig Main characteristic Curves of a Pump.

2. Operating characteristic Curves: If the speed is kept constant , the variation of manometric head, Power & efficiency with respect to discharge gives the operating characteristics of the pump . fig shows the operating characteristic curves of a pump .



(5)

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

The Output Power Curve is starts from the origin

$$\Omega = 0, \text{ sign } H = 0$$

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

The efficiency is starts from the origin as at

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

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

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

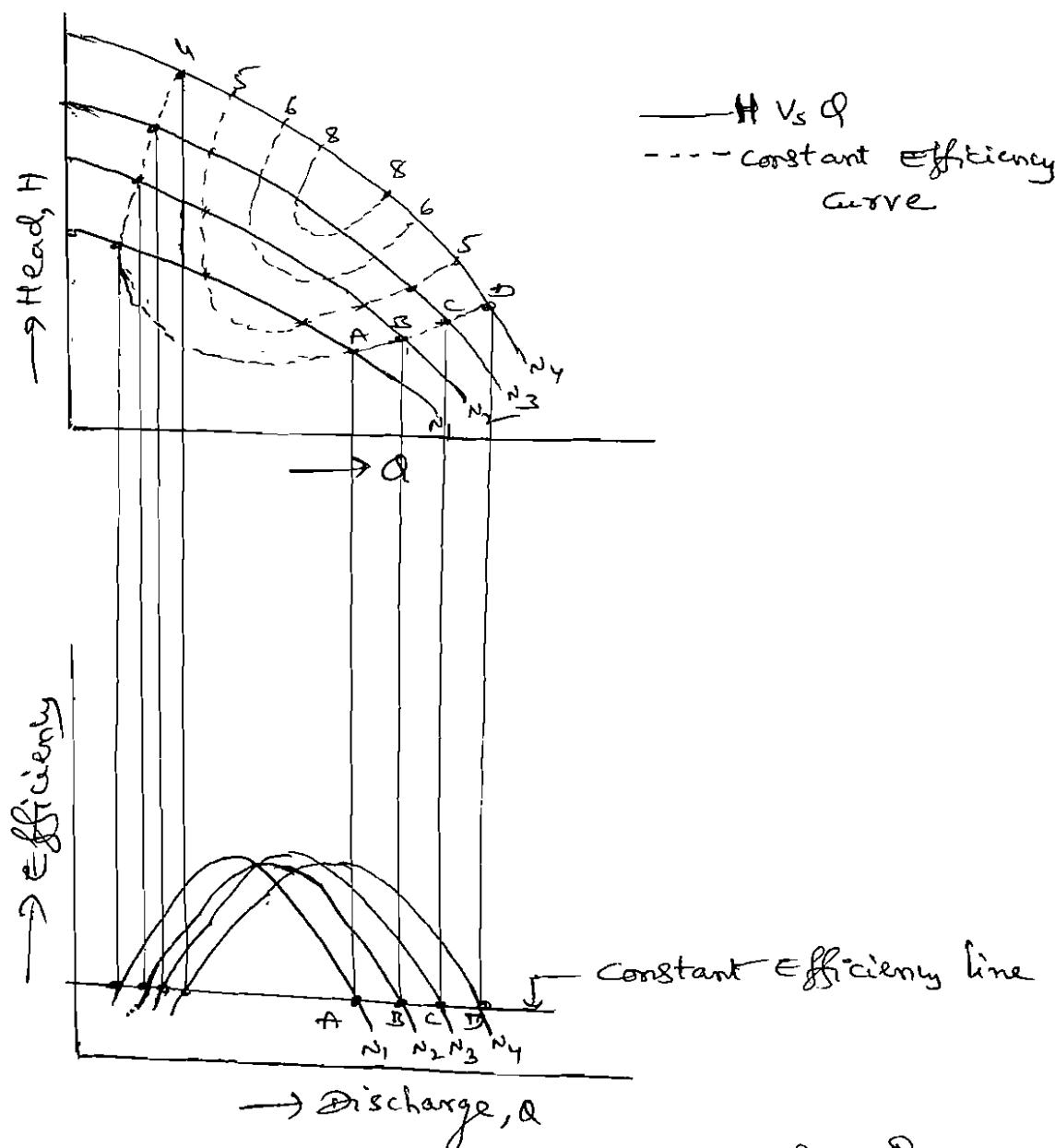


fig: Constant Efficiency Curves of a Pump .

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

i.s 50 cm. Determine

The theoretical discharge of the pump

Co-efficient of discharge and

slip and Percentage slip of the pump ,

Given Data:

Speed of the pump, $N = 30 \text{ rpm}$

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

Dia. of piston, $D = 25 \text{ cm} = 0.25 \text{ m}$

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

To find:

i) The theoretical discharge of the pump

ii) Co-efficient of discharge (C_d) and

iii) Slip and the percentage slip of the pump.

Formula used:

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

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

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

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

Calculation:

$$\begin{aligned} \therefore Q_{\text{th}} &= \frac{A L N}{60} = \frac{0.049 \times 0.5 \times 30}{60} \\ &= 0.0122 \text{ m}^3/\text{sec.} \end{aligned}$$

$$\text{i)} \quad C_d = \frac{Q_{act}}{Q_{th}} = \frac{0.012}{0.0122} = 0.98$$

$$\text{ii)} \quad \text{Slip} = Q_{th} - Q_{act} = 0.0122 - 0.012 \\ = 2 \times 10^{-4} \text{ m}^3/\text{s}$$

$$\text{iii)} \quad \text{Percentage Slip} = \frac{Q_{th} - Q_{act}}{Q_{th}} \times 100 \\ = \frac{0.0122 - 0.012}{0.0122} \times 100 \\ = 1.639 \%$$

Result: $\therefore Q_{th} = 0.0122 \text{ m}^3/\text{s}$

$$\therefore C_d = 0.98$$

$$\text{iv)} \quad \text{Slip} = 2 \times 10^{-4} \text{ m}^3/\text{sec}$$

$$\text{v)} \quad \text{Percentage Slip} = 1.639 \%$$

Definitions of Heads and efficiencies of a Centrifugal pump :-

* Suction Head (h_s) :-

It is the vertical height of the centre line of the centrifugal pump above the water surface in the tank or pump from which water is to be lifted. This height is called Suction lift and is denoted by ' h_s '.

* Delivery Head (h_d) :-

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

* Static Head (H_s) :-

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

$$H_s = h_s + h_d$$

* Manometric Head (H_m) :-

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

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

where h_s = suction head, h_d = delivery head,

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

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

V_d = velocity of water in delivery pipe.

* Efficiencies of a centrifugal pump :-

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

- a) manometric efficiency, η_{man}
- b) mechanical efficiency, η_{m}
- c) overall efficiency, η_o .

a) manometric efficiency (η_{man}) :-

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

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

$$= \frac{H_m}{\left(\frac{V_w^2 U_2}{g} \right)}$$

$$= \frac{g H_m}{V_w U_2}$$

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

The power at the impeller = work done by impeller per second $\frac{1000}{K_W}$

$$= \frac{W}{g} \times \frac{V_w U_2}{1000} K_W$$

$$\eta_{\text{man}} = \frac{\frac{W \times H_m}{1000}}{\frac{W}{g} \times \frac{V_w U_2}{1000}} = \frac{g \times H_m}{V_w U_2}$$

b) Mechanical Efficiency (η_m) :-

The power at the shaft of the centrifugal pump is more than the power available at the impeller of the pump. The ratio of the power available at the impeller to the power at the shaft of the c.p. is known as mechanical efficiency. It is written as.

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

The power at the impeller in kW = $\frac{\text{Work done by impeller per second}}{1000}$

$$= \frac{\omega}{g} \times \frac{V_{02} U_2}{1000}$$

$$\eta_m = \frac{\frac{\omega}{g} \left(\frac{V_{02} U_2}{1000} \right)}{S.P}$$

where S.P. = shaft power

c) Overall Efficiency (η_o) :-

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

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

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

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

= S.P. of the pump

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

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



*

Specific speed of a centrifugal pump (N_s) :-

The specific speed of a centrifugal pump is defined as the speed of a geometrically similar pump which would deliver one cubic metre of liquid per second against a head of one metre. It is denoted by N_s .

Expression for specific speed of a pump :-

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

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

$$= \pi D \times B \times V_f \xrightarrow{81} \textcircled{1}$$

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

where D = diameter of the impeller of the pump

B = width of the impeller

We know that $B \propto D$

From equ ① we have $Q \propto D^2 \times V_f \xrightarrow{\text{(ii)}}$

We also know that Tangential velocity is given by

$$V = \frac{\pi D N}{60} \propto DN \xrightarrow{\text{(iii)}}$$

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

$$V \propto V_f \propto \sqrt{H_m} \xrightarrow{\text{(iv)}}$$

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

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

Sub the values of D in equ (ii)

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

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

$$\therefore V_f \propto \sqrt{H_m}$$

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

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

where k is constant

If $H_m = 1m$ and $Q = 1 \text{ m}^3/\text{s}$, N becomes = Ns.

Sub these values in equ (v) we get

$$1 = k \frac{1^{3/2}}{Ns^2} = \frac{k}{Ns^2}$$

$$\therefore k = Ns^2$$

Sub value of k in equation (iv), we get

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

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

Net positive suction head (NPSH) :-

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

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

$\therefore \text{NPSH} = \text{Absolute pressure head at inlet of the pump} -$
 Vapour pressure head + Velocity head.

$$= \frac{P_i}{\rho g} - \frac{P_v}{\rho g} + \frac{V_s^2}{2g} \quad \rightarrow \textcircled{1}$$

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

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

Sub this equ. ① we get

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

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

$$= H_a - H_v - h_s - h_{fs}$$

$$\therefore \frac{P_a}{\rho g} = H_a$$

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

$$\text{NPSH} = [(H_a - h_s - h_{fs}) - H_v] \rightarrow \textcircled{2}$$

The right hand side of equ ② is the total suction head hence NPSH is equal to total suction head. Thus NPSH may also be defined as the total head required to make

the liquid flow through the suction pipe to the pump impeller.

Reciprocating Pumps

Introduction: The pumps as the hydraulic machine which convert the mechanical energy into hydraulic energy which is mainly in the form of pressure or If the mechanical energy is converted into hydraulic energy, by means of centrifugal force acting on the liquid, the pump is known as centrifugal pump. But if the mechanical energy is converted into hydraulic energy (or pressure energy) by forcing the liquid into a cylinder in which a piston is reciprocating (moving backwards and forwards), which exerts the thrust on the liquid and increases its hydraulic energy (pressure energy), the pump is known as Reciprocating pump.

Main Parts of a Reciprocating pump:-

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

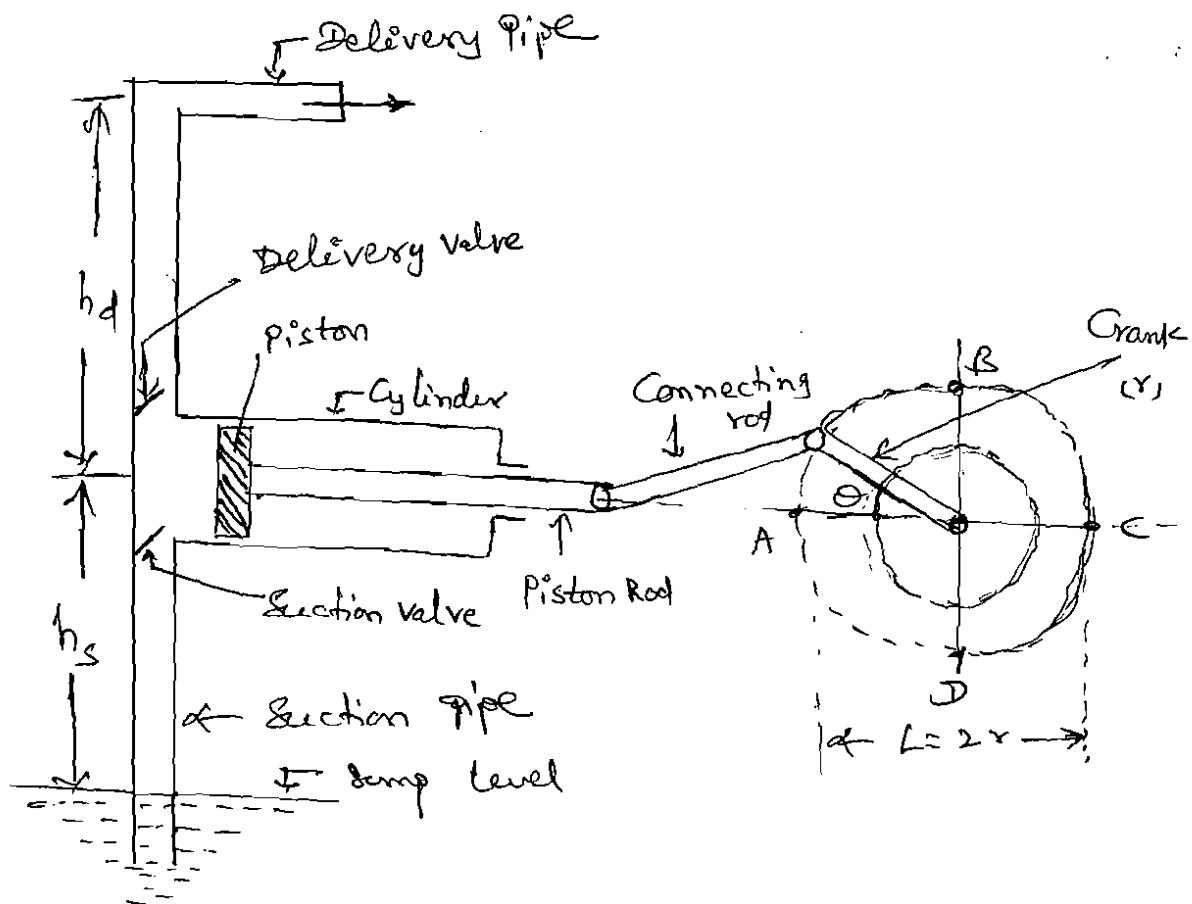


Fig: Main parts of a reciprocating pump.

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

Working of a R.P: Fig. Shows a single acting R.P,

which consists of a piston which moves forwards & backwards in a close fitting cylinder. The movement of piston is obtained by connecting the piston rod to crank by means of a connecting rod.

The crank is rotated by means of an electric motor.

Suction and delivery pipes with suction valve and delivery valve are connected to the cylinder. The suction and delivery valves are one way valves & non-return valves, which allow the water to flow in one-direction only. Suction valve allows water from suction pipe to the cylinder which delivery valve allows water from cylinder to delivery pipe only.

Discharge through a Reciprocating Pump:-

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

Let D = Dia. of the cylinder
 A = cross-sectional area of the piston cylinder

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

r = Radius of crank

N = r.p.m. of the crank

$$= 2 \times r$$

L = Length of the stroke

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

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

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

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

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

\therefore Discharge of the pump per second,

$$Q = \text{Discharge in one revolution} \times \text{No. of revolutions per second}$$

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

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

Weight of the water delivered per second

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

(5)

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

$$\begin{aligned} \text{Workdone per second} &= \frac{\text{Weight of water lifted per second} \times}{\text{Total height through which water is}} \\ &\quad \text{lifted} \\ &= W \times (h_s + h_d) \rightarrow (1) \end{aligned}$$

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

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

Substituting the value of W in eq i., we get

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

i.e. Power required to drive the pump, in kW

$$\begin{aligned} P &= \frac{\text{Workdone per Second}}{1000} = \frac{\rho g \times ALN \times (h_s + h_d)}{60 \times 1000} \\ &= \frac{\rho g \times ALN \times (h_s + h_d)}{60,000} \cdot kW. \end{aligned}$$

* Indicator diagrams

→ the Indicator diagram for a Reciprocating pump is defined as graph between the pressure head in the cylinder and the distance travelled by piston from inner dead centre for one complete revolution of the crank. At the maximum distance travelled by the piston is equal to the stroke length and hence the indicator diagram is a graph between pressure head and stroke length of the piston for one complete revolution. The pressure head is taken as ordinate and stroke length as abscissa.

Work done by pump = Area of indicator diagram.

* Thoma cavitation factor for Reaction Turbines

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

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

where H_b = Barometric pressure head in m of water,

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

H_v = Vapour pressure head in m of water,

H_s = Section pressure at the outlet of reaction turbine in m of water of height of turbine runner above the fast water surface,

H = net head on the turbine in m.

* Thoma's cavitation factor for centrifugal pump:

→ the mathematical expression for thoma's cavitation factor for centrifugal pump is given by

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

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

H_v = Vapour pressure head in m of water.

H_s = Suction pressure head in m of water.

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

H = Head developed by the pump.

→ the value of Thoma's cavitation factor (σ) for a particular type of turbine or pump is calculated from Eqs ① and ②. the value of Thoma's cavitation factor (σ) be compared with critical cavitation factor (σ_c) for that type of turbine pump.

If the value of σ is greater than σ_c , the cavitation will not occur in that turbine or pump. the critical cavitation factor (σ_c) may be obtained from tables or Empirical relationships.

→ the following Empirical relationships are used for obtaining the value of σ_c for different turbines:

$$\text{for Francis turbines, } \sigma_c = 0.625 \left(\frac{N_s}{380.78} \right)^2 \\ \approx 4.31 \times 10^{-8} N_s^2$$

$$\text{for propeller turbines, } \sigma_c = 0.28 + \left[\frac{1}{7.5} \left(\frac{N_s}{380.78} \right)^2 \right]$$

In the above Expression N_s is in (r.p.m., rev/min) units. If N_s is in (r.p.m., h.p., m) units the Empirical relationship would be as follows:

$$\text{for Francis turbines, } \sigma_c = 0.625 \left(\frac{N_s}{444} \right)^2 \approx 3.17 \times 10^{-8} N_s^2$$

$$\text{for propeller turbines, } \sigma_c = 0.28 + \left[\frac{1}{7.5} \left(\frac{N_s}{444} \right)^2 \right]$$

* Slop of reciprocating pump

→ Slop of a pump is defined as the difference between the theoretical discharge and actual discharge of the pump. The discharge of a single-acting pump given by,

Equation, $Q = AXL \times \frac{N}{60} = \frac{ALN}{60}$ and of a double-acting pump given by Equation,

$$Q = \left(\frac{\pi}{4} D^2 + \frac{\pi}{4} d^2 \right) \times \frac{L \times N}{60} = 2 \times \frac{\pi}{4} D^2 \times \frac{L \times N}{60} = \frac{2ALN}{60}$$
 are theoretical discharge. The.

actual discharge of a pump is less than the theoretical discharge due to leakage. The difference of the theoretical discharge and actual discharge is known as slop of the pump. Hence, Mathematically,

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

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

$$\begin{aligned}\text{Percentage Slop} &= \frac{Q_{th} - Q_{act}}{Q_{th}} \times 100 = \left(1 - \frac{Q_{act}}{Q_{th}} \right) \times 100 \\ &= (1 - C_d) \times 100 \quad \left[\because \frac{Q_{act}}{Q_{th}} = C_d \right]\end{aligned}$$

Where C_d = co-efficient of discharge.

* Negative Slop of the Reciprocating pump

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

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

" Negative Slop occurs when delivery pipe is short,

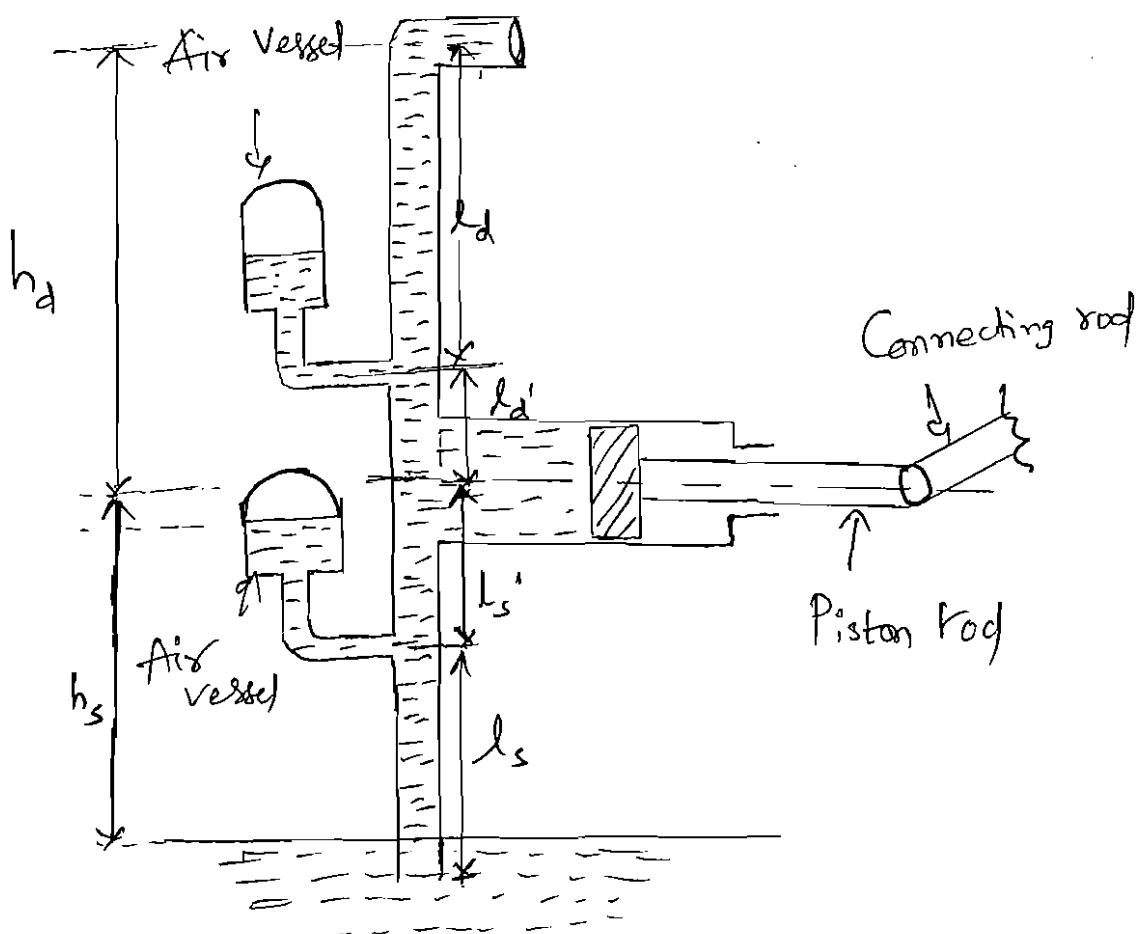
Suction pipe is long and pump is running at high speed."

* Air Vessel :-

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

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

- To obtain a continuous supply of liquid at a uniform rate,
- To save a considerable amount of work in overcoming the frictional resistance in the suction and delivery pipes, and
- To run the pump at a high speed without separation.



Eg: Air Vessels fitted to reciprocating pump

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

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

Let A = cross sectional area of the cylinder

a = cross-sectional area of suction & delivery pipe.

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

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

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

l_s = Length of the suction pipe below air vessel.

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

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

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

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

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

h'_{fs} = Loss of head due to friction in suction pipe between cylinder and air vessel.

(a) Pressure head in the cylinder during delivery stroke?

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

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

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

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

iii) At the end of the delivery stroke, $\theta = 180^\circ$, $\sin\theta = 0$ & $\cos\theta = -1$ and hence total pressure head

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

* Comparison between Centrifugal Pumps and Reciprocating Pumps.

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

Classification of Reciprocating Pumps :-

The reciprocating pumps may be classified as:

1. According to the water being in contact with one side, both sides of the piston, and
2. According to the number of cylinders provided,

If the water is in contact with one side of the piston, the pump is known as single-acting.

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

- i) Single-acting pump, and
- ii) Double-acting pump.

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

- i) Single cylinder pump
- ii) Double cylinder pump, and
- iii) Triple cylinder pump.