SREENIVASA INSTITUTE OF TECHNOLOGY AND MANAGEMENT STUDIES

(AUTONOMOUS)

DEPARTMENT OF MECHANICAL ENGINEERING

COURSE MATERIAL

Subject Name	Fluid Mechanics and Hydraulic Machines
Subject Code	23MEC241T
Semester	IV Semester
Academic Year	2025-26
Regulation	R23

Unit-V

Hydraulic Turbines, Centrifugal pumps and Reciprocating pumps

TOPICS TO BE COVERED

- Derivation of Units&
- Specific quantities
- Surge tank
- Water hammer

LECTURE 8

Geometric similarity

GEOMETRIC SIMILARITY

- The geometric similarity must exist between the model and its proto type. the ratio of all corresponding linear dimensions in the model and its proto type are equal.

Let L_m = length of model

b_m = Breadth of model

D_m = Diameter of model

A_m = Area of model

$\forall_m$ = Volume of model

And $L_p, b_p, D_p, A_p, \forall_p$= Corresponding values of the proto type.

For geometrical similarity between model and prototype, we must have the relation,

$$\frac{L_P}{L_m} = \frac{b_P}{b_m} = \frac{D_P}{D_m} = L_r$$

Where L_r is called scale ratio.

For area's ratio and volume's ratio the relation should be,

$$\frac{A_P}{A_m} = \frac{L_P \times b_P}{L_m \times b_m} = L_r \times L_r = L_r{}^2$$

$$\frac{\forall_P}{\forall_m} = \left(\frac{L_P}{L_m}\right)^3 = \left(\frac{b_P}{b_m}\right)^3 = \left(\frac{D_P}{D_m}\right)^3 = L_r{}^3$$

PERFORMANCE OF HYDRAULIC TURBINES

- In order to predict the behavior of a turbine working under varying conditions of head, speed, output and gate opening , the results are expressed in terms of quantities which may be obtained when the head on the turbine is reduced to unity. The conditions of the turbine under unit head are such that the efficiency of the turbine remains unaffected.

The three important unit quantities are:

1. Unit speed,
2. Unit discharge, and
3. Unit power

1. Unit Speed: it is defined as the speed of a turbine working under a unit head. It is denoted by 'N_u'. The expression of unit speed (N_u) is obtained as:

Let N = Speed of the turbine under a head H

H = Head under which a turbine is working

u = Tangential velocity.

The tangential velocity, absolute velocity of water and head on turbine are related as:

$$u \propto V \qquad \text{Where } V \propto \sqrt{H}$$

$$\propto \sqrt{H} \quad \text{\_\_\_\_\_\_\_\_\_\_\_\_} (1)$$

Also tangential velocity (*u*) is given by

$$u = \frac{\pi D N}{60}$$ Where D = Diameter of turbine.

For a given turbine, the diameter (D) is constant

$$u \propto N \quad \text{Or} \quad N \propto u \quad \text{Or} \quad N \propto \sqrt{H} \quad \left(\because From\,(1),\; u \propto \sqrt{H}\right)$$

∴ $$N = K_1\sqrt{H} \quad \text{\_\_\_\_\_\_\_\_\_\_ (2)}$$ Where K_1 is constant of proportionality.

If head on the turbine becomes unity, the speed becomes unit speed or

When $$H = 1, \;\; N = N_u$$

Substituting these values in equation (2), we get

$$N_u = K_1\sqrt{1.0} = K_1$$

Substituting the value of K_1 in equation (2)

$$N = N_u\sqrt{H} \quad or \quad N_u = \frac{N}{\sqrt{H}} \quad \text{\_\_\_\_\_\_\_\_\_\_ \textbf{(I)}}$$

2. Unit Discharge: It is defined as the discharge passing through a turbine, which is working under a unit head (i.e. 1 m). It is denoted by $'Q_u'$ the expression for unit discharge is given as:

Let H = head of water on the turbine

Q = Discharge passing through turbine when head is H on the turbine.

The discharge passing through a given turbine under a head *'H'* is given by,

Q=Area of flow × Velocity

But for a turbine, area of flow is constant and velocity is proportional to $\sqrt{H}$

$$Q \propto velocity \propto \sqrt{H}$$

Or $Q = K_2\sqrt{H}$ ____________ (3)

Where K_2 is constant of proportionality

If H = 1, $Q = Q_u$ (By definition)

Substituting these values in equation (3) we get

$$Q_u = K_2\sqrt{1.0} = K_2$$

Substituting the value of K_2 in equation (3) we get

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

$Q_u = \frac{Q}{\sqrt{H}}$ _____________ **(II)**

3. Unit Power: It is defined as the power developed by a turbine working under a unit head (i.e. under a head of 1m). It is denoted by $'P_u'$. The expression for unit power is obtained as:

Let H= Head of water on the turbine
P= Power developed by the turbine under a head of H
Q= Discharge through turbine under a head H

The overall efficiency (η_0) is given as

$$\eta_0 = \frac{Power\ developed}{Water\ power} = \frac{P}{\frac{\rho g Q H}{1000}}$$

$$P = \eta_0 \times \frac{\rho g Q h}{1000}$$

$$\propto\ Q \times H \quad \propto\ \sqrt{H} \times H \qquad (\because\ Q \propto \sqrt{H})$$

$$\propto\ H^{\frac{3}{2}}$$

$P = K_3 H^{3/2}$ ________ (4) Where K_3 is a constant of proportionality

When H=1 m, $P = P_u$

$$\therefore \quad P_u = K_3 (1)^{3/2} \quad = K_3$$

Substituting the value of K_3 in equation (4) we get

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

$P_u = \frac{P}{H^{3/2}}$ ____________ **(III)**

UNIT QUANTITIES

Use of Unit Quantities $(N_u\ ,\ Q_u\ ,\ P_u)$**:**

If a turbine is working under different heads, the behaviour of the turbine can be easily known from the values of the unit quantities i.e. from the value of unit speed, unit discharge and unit power.

Let H_1, H_2, H_3, -------- are the heads under which a turbine works,

N_1, N_2, N_3, --------- are the corresponding speeds,
Q_1, Q_2, Q_3, -------- are the discharge and
P_1, P_2, P_3, --------- are the power developed by the turbine.

Using equation I, II, III respectively,

$$\left.\begin{aligned} N_u &= \frac{N_1}{\sqrt{H_1}} = \frac{N_2}{\sqrt{H_2}} = \frac{N_3}{\sqrt{H_3}} \\ Q_u &= \frac{Q_1}{\sqrt{H_1}} = \frac{Q_2}{\sqrt{H_2}} = \frac{Q_3}{\sqrt{H_3}} \\ P_u &= \frac{P_1}{H_1^{3/2}} = \frac{P_2}{H_2^{3/2}} = \frac{P_3}{H_3^{3/2}} \end{aligned}\right\} \text{————— (IV)}$$

Hence, if the speed, discharge and power developed by a turbine under a head are known, then by using equation (IV) the speed, discharge, power developed by the same turbine a

CHARACTERISTIC CURVES OF HYDRAULIC TURBINES:

- Characteristic curves of a hydraulic turbine are the curves, with the help of which the exact behaviour and performance of the turbine under different working conditions can be known. These curves are plotted from the results of the tests performed on the turbine under different working conditions.

1) Speed (N)

2) Head (H)

3) Discharge (Q)

4) Power (P)

5) Overall Efficiency ( ) and

6) Gate opening.

MAIN CHARACTERISTIC CURVES OR CONSTANT HEAD CURVES

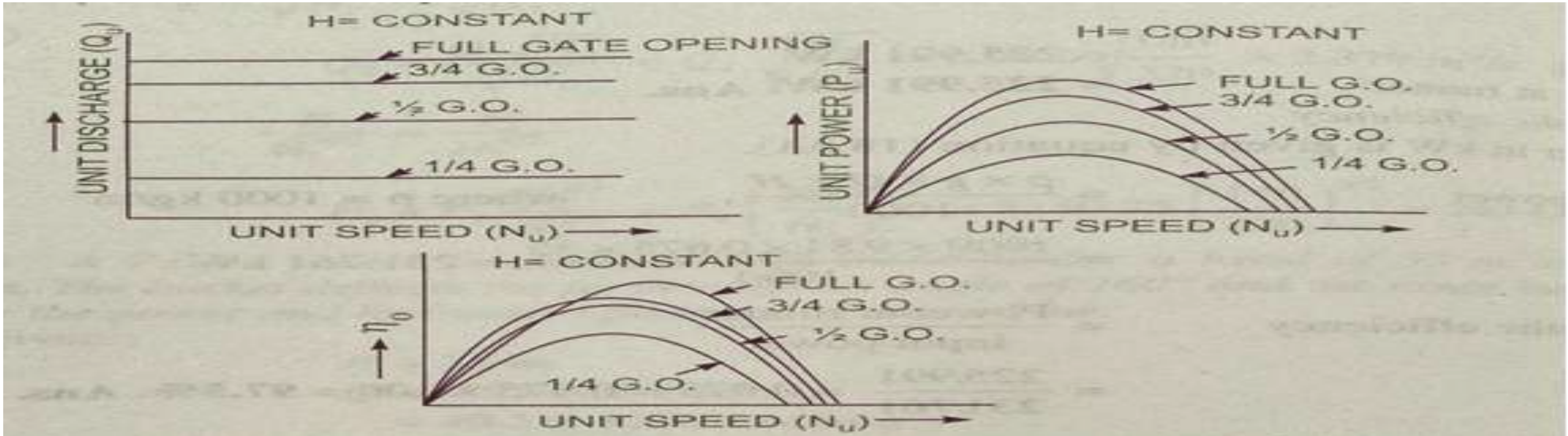
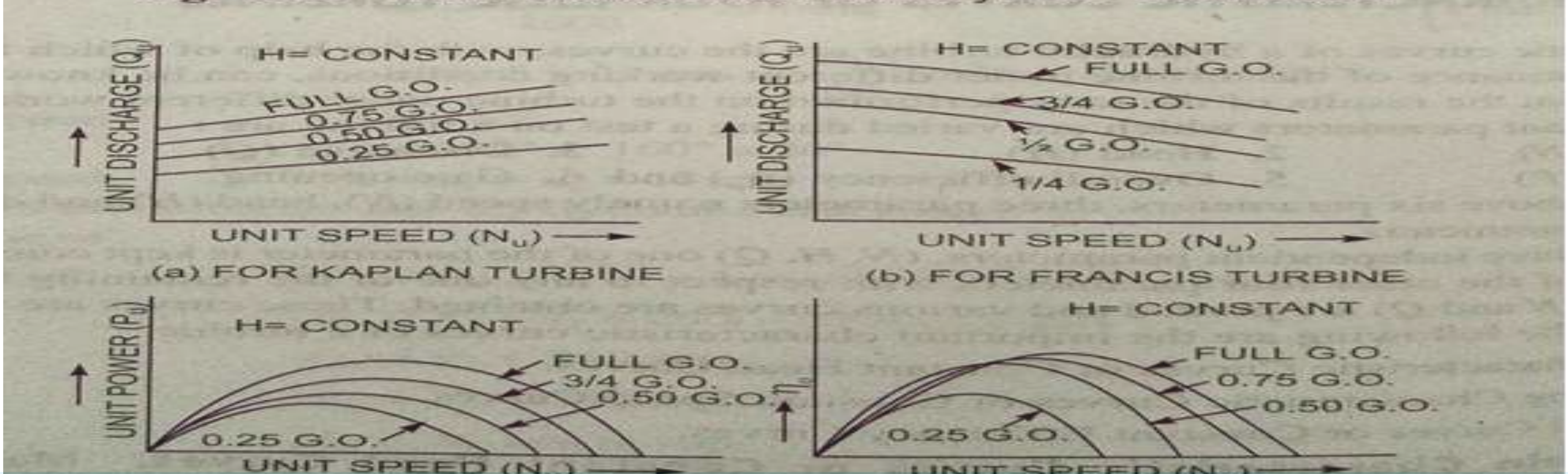


Fig. 18.35 *Main characteristic curves for a Pelton wheel.*



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.

Precaution against Cavitation:

•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 2.5m of water.

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

Effects of Cavitation

- The metallic surfaces are damaged and cavities are formed on the surfaces.

Due to sudden collapse of vapour bubbles, considerable noise and vibrations are produced.

- The efficiency of a turbine decreases due to cavitation. Due to pitting action, the surface of the turbine blades becomes rough and the force exerted by the water on the turbine blades decreases. Hence, the work done by water or output horse power becomes less and efficiency decreases.

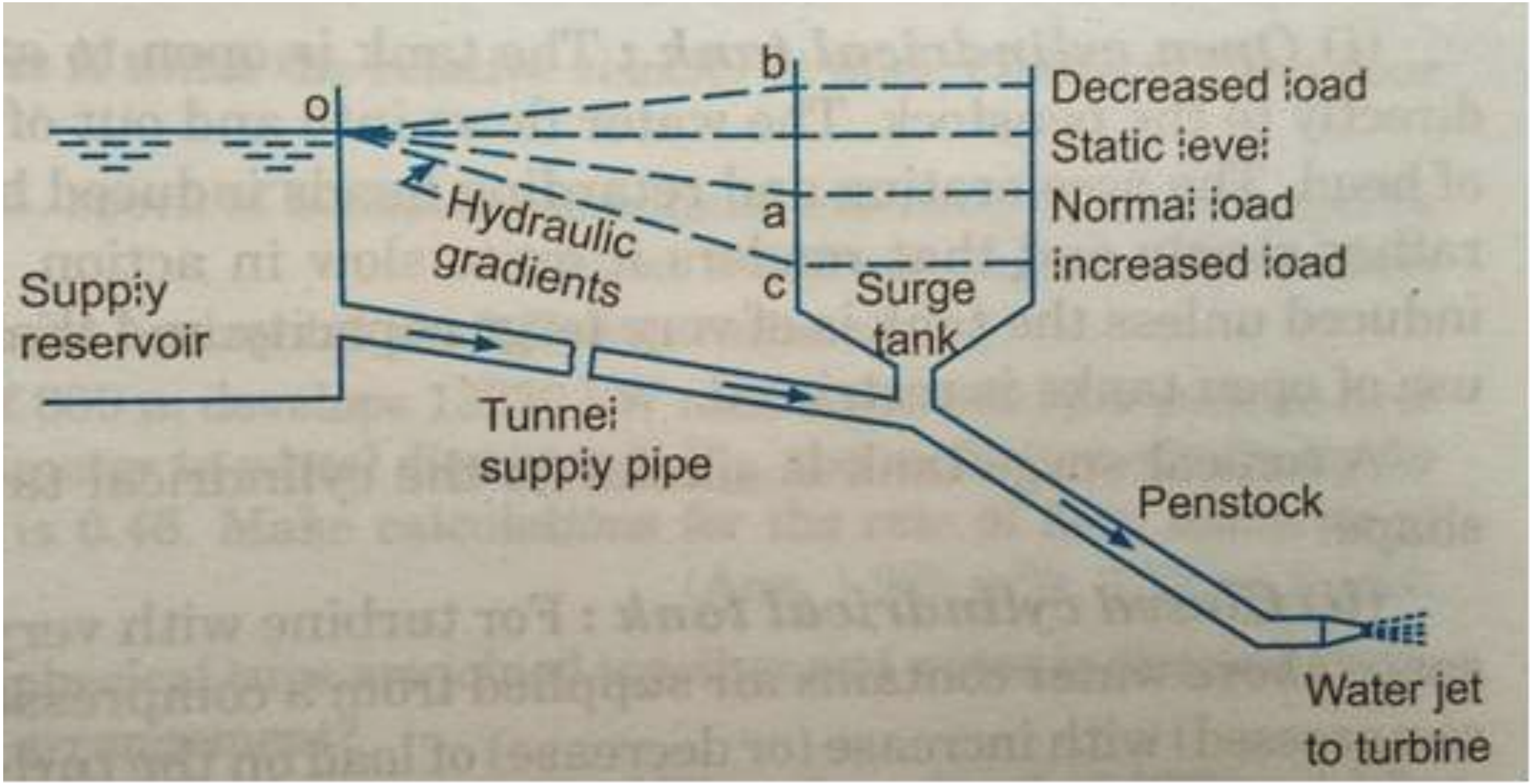
TOPICS TO BE COVERED

- Surge tank
- Water hammer
- Applications
- Assignment questions

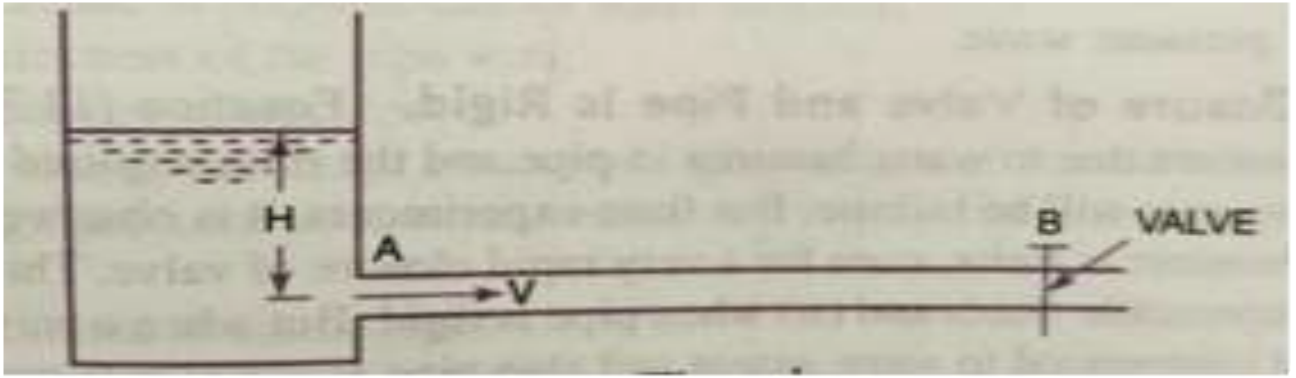
LECTURE 9

Surge Tank & Water Hammer

SURGE TANK


b
o
Decreased load
Static level
Normal load
Increased load
a
c
Hydraulic gradients
Supply reservoir
Surge tank
Tunnel supply pipe
Penstock
Water jet to turbine

WATER HAMMER



The pressure rise due to water hammer depends up on:

1. Velocity of flow of water in pipe.
2. The length of pipe.
3. Time taken to close the valve.
4. Elastic properties of the material of the pipe.

The following cases of water hammer in pipes will be considered.

1. Gradual closure of valve
2. Sudden closure of valve considering pipe in rigid
3. Sudden closer of valve considering pipe elastic.

UNIT – V

Centrifugal Pumps:

- Classification, working- work done.
- Manometric head and efficiencies.
- Specific speed.
- Performance characteristic curves, NPSH.

Reciprocating Pumps:

- Working, discharge.
- Slip and Indicator diagram.

COURSE OUTLINE

LECTURE	LECTURE TOPIC	KEY ELEMENTS	Learning objectives
1	Introduction to Pumps	Classification of centrifugal pumps	Understand types of pumps (B2)
2	Centrifugal Pump- work done	Manometric Head & η	Evaluate η of pump (B5)
3	Specific Speed of Centrifugal Pump	Performance	Understand the performance of pump (B2)
4	Characteristic curves of pump NPSH	Performance	Understand the performance of pump (B2) Evaluate NPSH (B5)
5	Problems on Centrifugal pumps		
6	Reciprocating pumps	Working, Q & η	Evaluate η of pump (B5)
7	Slip & Indicator diagram	Positive displacement	Evaluate slip of pump (B5) Understand Indicator diagram (B2)

TOPICS TO BE COVERED

- Introduction
- Main parts of centrifugal pump
- Heads on Centrifugal pump
- Efficiencies

LECTURE 1

Introduction to pumps

INTRODUCTION TO CENTRIFUGAL PUMPS

- The hydraulic machines which convert the mechanical energy in to hydraulic energy are called pumps.
- The hydraulic energy is in the form of pressure energy. If the mechanical energy is converted in to pressure energy by means of centrifugal force acting on the fluid, the hydraulic machine is called centrifugal pump.
- The centrifugal pump acts as a reversed of an inward radial flow reaction turbine. This means that the flow in centrifugal pumps is in the radial outward directions.
- The centrifugal pump works on the principle of forced vertex flow which means that when a certain mass of liquid is rotated by an external torque, the rise in pressure head of the rotating liquid takes place.

MAIN PARTS OF A CENTRIFUGAL PUMP

- **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

- **Casing:** It 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 kinetic energy of the water discharged at the outlet of the impeller is converted in to pressure energy before the water leaves the casing and enters the delivery pipe.
- The following three types of the casing are commonly adopted.

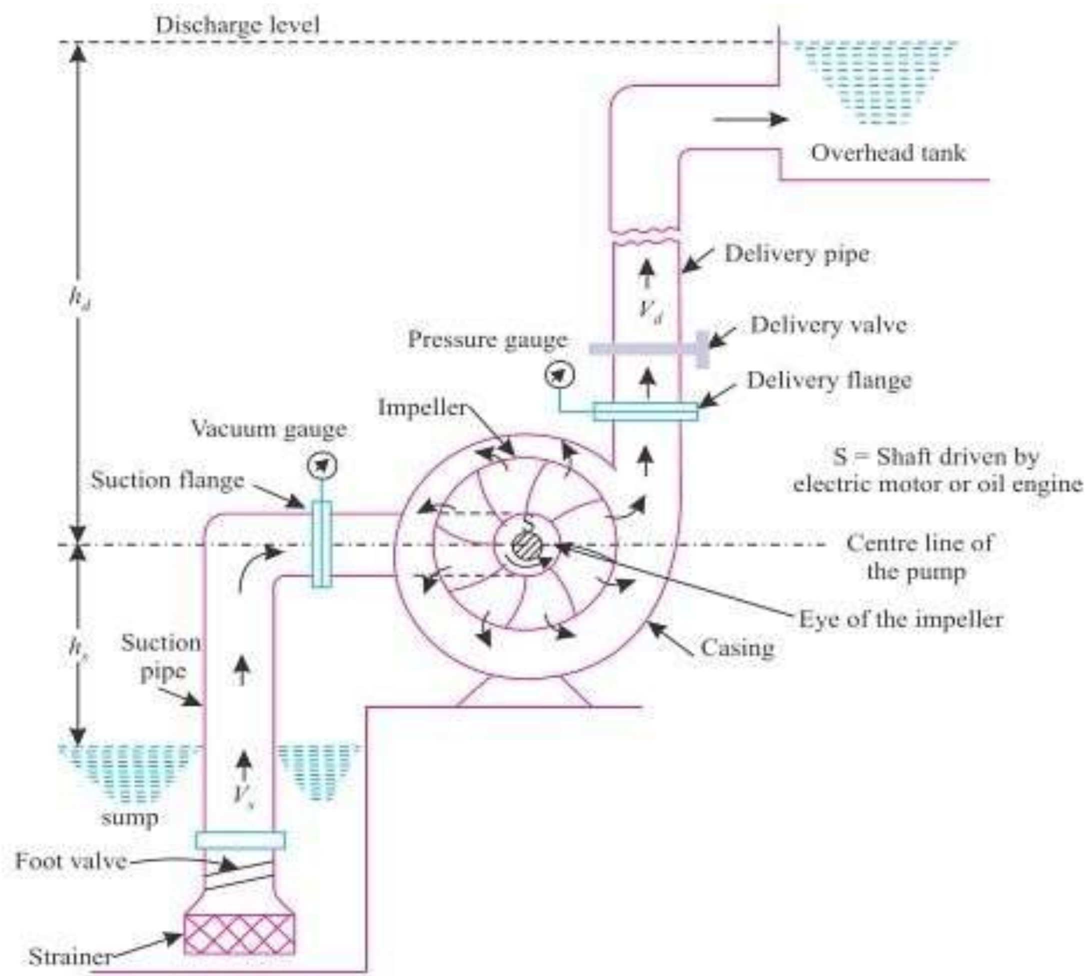
➢ Volute

➢ Vortex

➢ Casing with guide blades

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

CENTRIFUGAL PUMP PARTS



HEADS OF A CENTRIFUGAL PUMPS

- **Suction Head** $(\boldsymbol{h_s})$: It is the vertical height of the centre line of centrifugal pump, above the water surface in the tank or sump from which water is to be lifted. This height is also called suction lift $'h_s'$.
- **Delivery Head**$(\boldsymbol{h_d})$**:** The vertical distance between the centre line of the pump and the water surface in the tank to which water is delivered is known as delivery head. This is denoted by $'h_d'$.
- **Static Head**$(\boldsymbol{H_s})$**:** The sum of suction head and delivery head is known as statics head$'H_s'$.
- $$H_s = h_s + h_d$$
- **Manometric Head**$(\boldsymbol{H_m})$**:** Manometric head is defined as the head against which a centrifugal pump has to work. It is denoted by H_m.

a) H_m = Head imparted by the impeller to the water – Loss of head in the pump $= \frac{V_{w_2} u_2}{g} -$ Loss of head in impeller and casing

$= \frac{V_{w_2} u_2}{g}$ ………… If loss of head in pump is zero.

b) H_m = Total head at outlet of pump – Total head at the inlet of the pump $= \left(\frac{P_0}{\rho g} + \frac{{V_0}^2}{2g} + Z_0\right) - \left(\frac{P_i}{\rho g} + \frac{{V_i}^2}{2g} + Z_i\right)$

Where $\frac{P_0}{\rho g}$ = Pressure head at outlet of the pump = h_d

$\frac{{V_0}^2}{2g}$ = Velocity head at outlet of the pump

= Velocity head in delivery pipe $= \frac{{V_d}^2}{2g}$

Z_0 = Vertical height of the outlet of the pump from datum line, and

$\frac{P_i}{\rho g}, \frac{V_i{}^2}{2g}, Z_i$ = Corresponding values of pressure head, velocity head and datum head at the Inlet of the pump, i.e. $h_s, \frac{V_s{}^2}{2g}$ $and\ Z_s$ respectively.

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

Where h_s = Suction head,

h_d = Delivery head,

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

h_{f_d} = Frictional head loss in delivery pipe

V_d = Velocity of water in delivery pipe.

EFFICIENCIES OF CENTRIFUGAL PUMP

- **Manometric Efficiency**$(\boldsymbol{\eta_{man}})$**:** The ratio of the Manometric head to the head imparted by the impeller to the water is known as

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

$$= \frac{H_m}{\left(\frac{V_{w_2}u_2}{g}\right)} = \frac{gH_m}{V_{w_2}u_2}$$

- **Mechanical Efficiency**$(\boldsymbol{\eta_m})$**:** The power at the shaft of the centrifugal pump is more 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 centrifugal pump is known as mechanical efficiency.

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

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

$$= \frac{W}{g} \times \frac{V_{w_2} \times u_2}{1000}$$

$$\eta_m = \frac{\frac{W}{g}\left(\frac{V_{w_2} \times u_2}{1000}\right)}{S.P}$$

Where S.P. = Shaft power.

- **Overall Efficiency(η_0):** It is defined as the ratio of power output of the pump to the power input to the pump.

The power output of the pump in kW $= \frac{Weight\ of\ water\ lifted \times H_m}{1000} = \frac{WH_m}{1000}$

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

= S.P. Of the pump

∴

$$\eta_0 = \frac{\left(\frac{WH_m}{1000}\right)}{S.P.}$$

$$\eta_0 = \eta_{man} \times \eta_m$$

TOPICS TO BE COVERED

- Work done by Centrifugal pump
- Velocity Triangle diagram
- Derivation

LECTURE 2

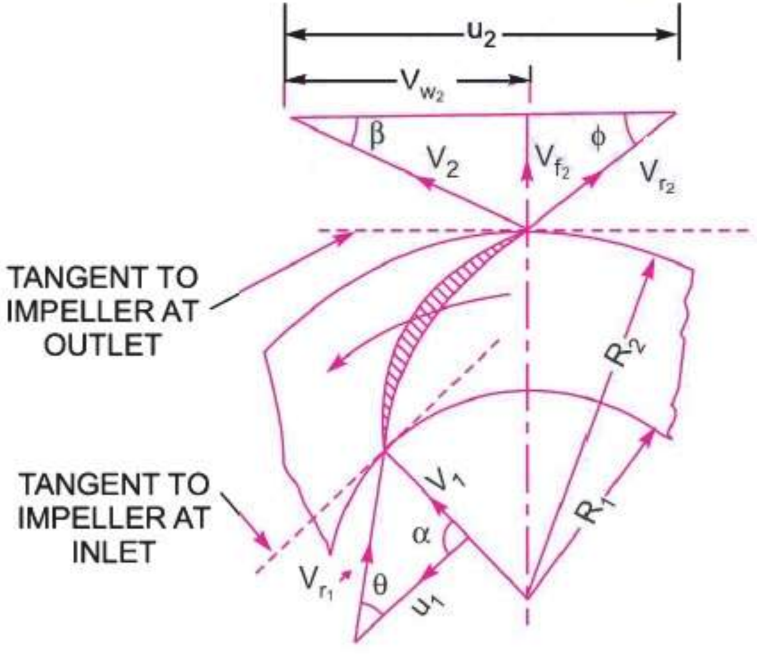
Work done by Centrifugal pump

WORK DONE BY CENTRIFUGAL PUMP

DERIVATION

- In the centrifugal pump, work is done by the impeller on the water.
- The expression for the work done by the impeller on the water is obtained by drawing velocity triangles at inlet and outlet of the impeller in the same way as for a turbine.

VELOCITY TRIANGLE DIAGRAM



- The water enters the impeller radially at inlet for best efficiency of the pump, which means the absolute velocity of water at inlet makes an angle of 90^0 with the direction of motion of the impeller at inlet.
- Hence angle $\alpha = 90^0$ and $V_{w_1} = 0$ for drawing the velocity triangles the same notations are used as that for turbines.
- 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 $= \frac{\pi D_1 N}{60}$

$D_2 =$ Diameter of 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

θ = Angle made by relative velocity (V_{r_1})at inlet with the direction of motion of vane

And V_2, V_{r_2} , β and Ø are the corresponding valves at outlet.

- As the water enters the impeller radially which means the absolute velocity of water at inlet is in the radial direction and hence angle $\alpha = 90^0$ and $V_{w_1} = 0$.
- 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 equation.

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

∴Work done by the impeller on the water per second per unit weight of water striking/second

$$= -[workdone\ in\ case\ of\ a\ turbine]$$

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

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

$$= \frac{1}{g}V_{w_2}u_2 \quad \text{\_\_\_\_\_\_\_\_\_\_\_\_ (1)} \qquad \left(\because V_{w_1} = 0\right)$$

- Work done by the impeller on water per second= $\frac{W}{g} \times V_{w_2}u_2$

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

Q = Area × Velocity of flow

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

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

- Where B_1 and B_2 are width of impeller at inlet and outlet and V_{f_1} And V_{f_2} are velocities of flow at inlet and outlet
- **Head imparted to the water by the impeller or energy given by impeller to water per unit weight per second**

$$\mathbf{H = \frac{1}{g} V_{w_2} u_2}$$

TOPICS TO BE COVERED

- Specific speed of centrifugal pump
- Multi stage centrifugal pump
  - To produce high heads
  - To produce high discharge

LECTURE 3

Specific speed of Centrifugal pump

SPECIFIC SPEED OF CENTRIFUGAL PUMP

- The specific speed of a centrifugal pump is defined as the speed of a geometrically similar pump, which would deliver one cubic meter of liquid per second against a head of one meter.
- It is denoted by $'N_s'$.
- The discharge Q for a centrifugal pump is given by the relation

$$Q = Area \times Velocity\ of\ flow$$

$$= \pi D \times B \times V_f \quad \text{Or} \qquad Q \propto D \times B \times V_f \text{ \_\_\_\_\_\_\_ (1)}$$

Where D = Diameter of the impeller of the pump and

B = Width of the impeller

- We know that $B \propto D$
- From equation (1) we have $Q \propto D^2 \times V_f$ __________ (2)

- We also know that the tangential velocity is given by

$$u = \frac{\pi DN}{60} \propto DN \_\_\_\_\_\_\_\_\_\_ (3)$$

- Now the tangential velocity (u) and velocity of flow (V_f) are related to Manometric head (H_m) as $u \propto V_f \propto \sqrt{H_m}$ ____________ (4)
- Substituting the value of (u) in equation (3), we get

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

- Substituting the values of D in equation (2)

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

$$\propto \frac{H_m}{N^2} \times \sqrt{H_m} \qquad \left[\because From\ eq\ (4) V_f \propto \sqrt{H_m}\right]$$

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

$$Q = K \ \frac{{H_m}^{3/2}}{N^2} \_\_\_\_\_\_\_\_\_\_\_\_ (5)$$

Where K is a constant of proportionality

- If $H_m = 1m$ and $Q = 1m^3/sec$ N becomes N_s
- Substituting these values in equation (5), we get

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

$$\therefore \qquad K = {N_s}^2$$

- Substituting the value of K in equation (5), we get

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

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

MULTISTAGE CENTRIFUGAL PUMPS

1. TO PRODUCE HIGH HEADS

- If a high head is to be developed, the impellers are connected in series (or on the same shaft) while for discharging large quantity of liquid, the impellers (or pumps) are connected in parallel.

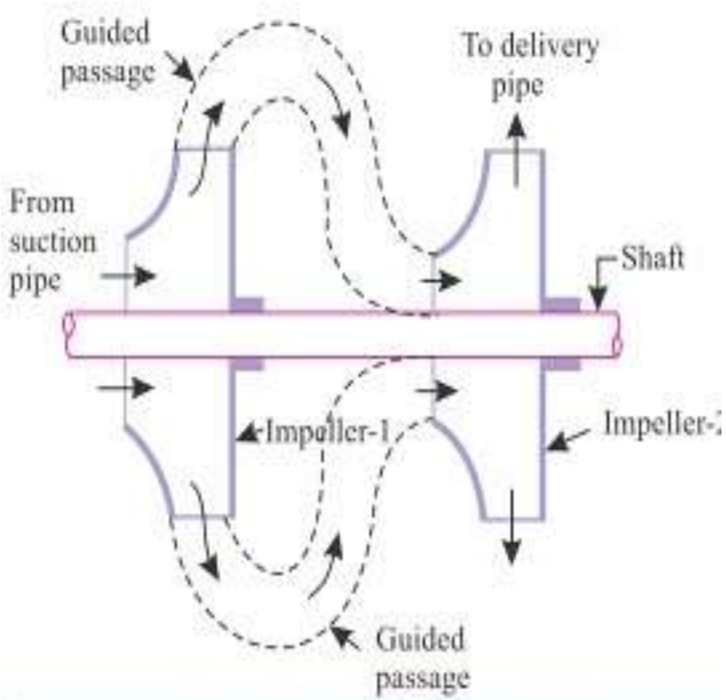


Fig. 3.23. Two-stage pump-impellers in series.

- For developing a high head, a number of impellers are mounted in series on the same shaft.
- The water from suction pipe enters the 1^{st} impeller at inlet and is discharged at outlet with increased pressure.
- The water with increased pressure from the outlet of the 1^{st} impeller is taken to the inlet of the 2^{nd} impeller with the help of a connecting pipe.
- At the outlet of the 2^{nd} impeller the pressure of the water will be more than the water at the outlet of the 1^{st} impeller.
- Thus if more impellers are mounted on the same shaft, the pressure at the outlet will be increased further.
- Let n= Number of identical impellers mounted on the same shaft,

  $H_m =$ Head developed by each impeller.
- Then total Head developed $= n \times H_m$
- The discharge passing through each impeller is same.

MULTISTAGE CENTRIFUGAL PUMPS

2. TO PRODUCE HIGH DISCHARGE

- For obtaining high discharge, the pumps should be connected in parallel.
- Each of the pumps lifts the water from a common sump and discharges water to a common pipe to which the delivery pipes of each pump is connected.
- Each of the pumps is working against the same head.

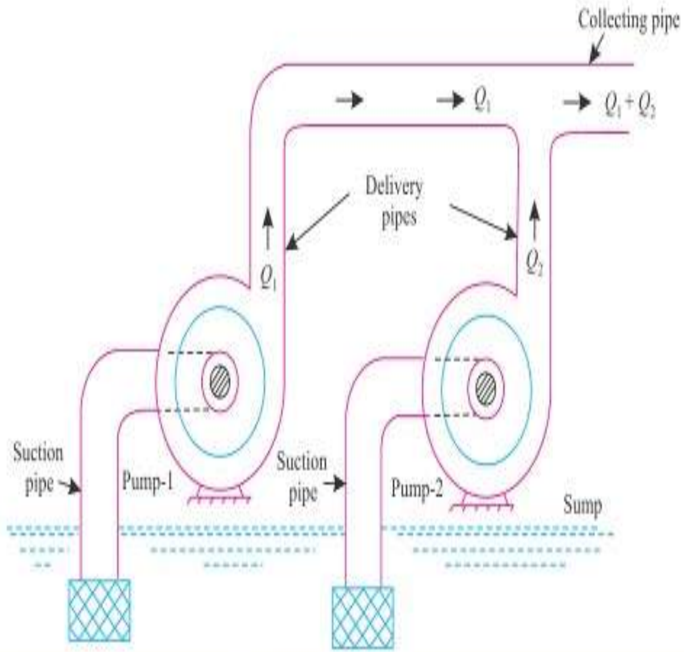


Fig. 3.24. Pumps in parallel.

- Let n = Number of identical pumps arranged in parallel.

  Q = Discharge from one pump.

  $\therefore$ Total Discharge = $\boldsymbol{n} \times \boldsymbol{Q}$

TOPICS TO BE COVERED

- Performance characteristic curves
- Main characteristic curves
- Operating characteristic curves
- Constant efficiency or Muschel curves
- NPSH

LECTURE 4

Performance characteristic curves

PERFORMANCE CHARACTERISTIC CURVES

- The characteristic curves of a centrifugal pump 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 behavior and performance of the pump, when the pump is working under different flow rate, head and speed.
- The following are the important characteristic curves for the pumps:

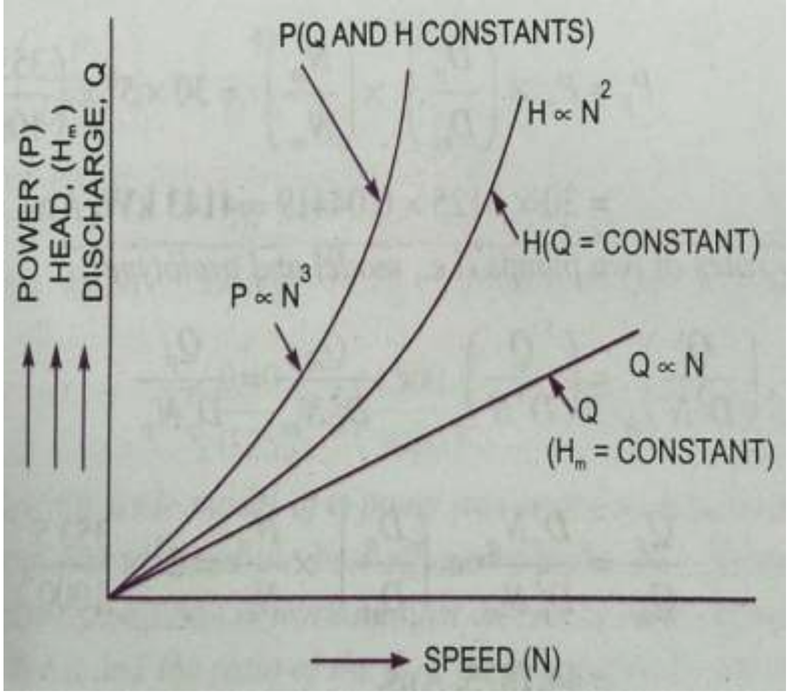
➢ Main characteristic curves.

➢ Operating characteristic curves and

➢ Constant efficiency or Muschel curves at different flow rate, head and speed.

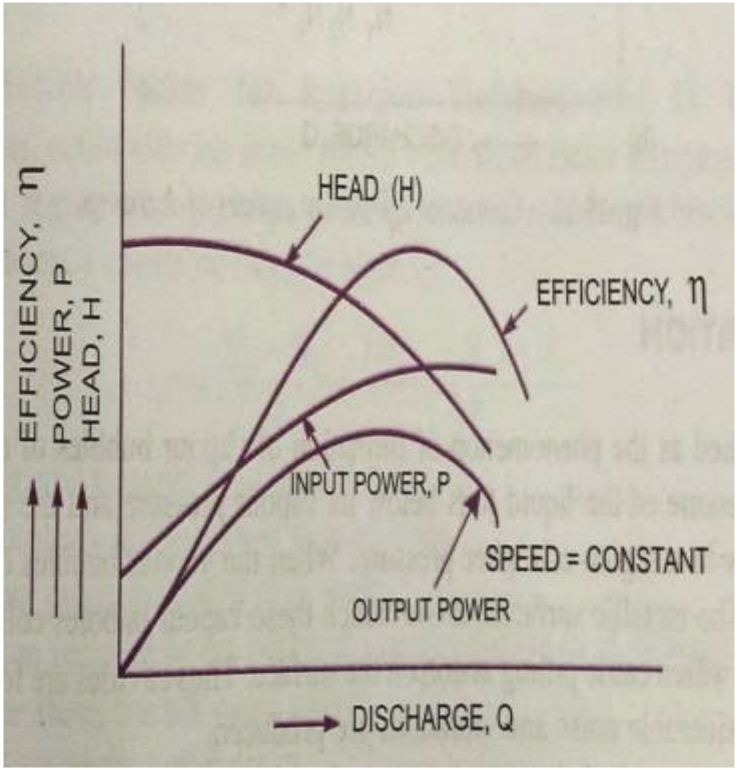
MAIN CHARACTERISTIC CURVES

- The main characteristic curves of a centrifugal pump consists of a head (Manometric head H_m) 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 kept constant.
- For plotting curves power versus speed, Manometric head and discharge are kept constant.



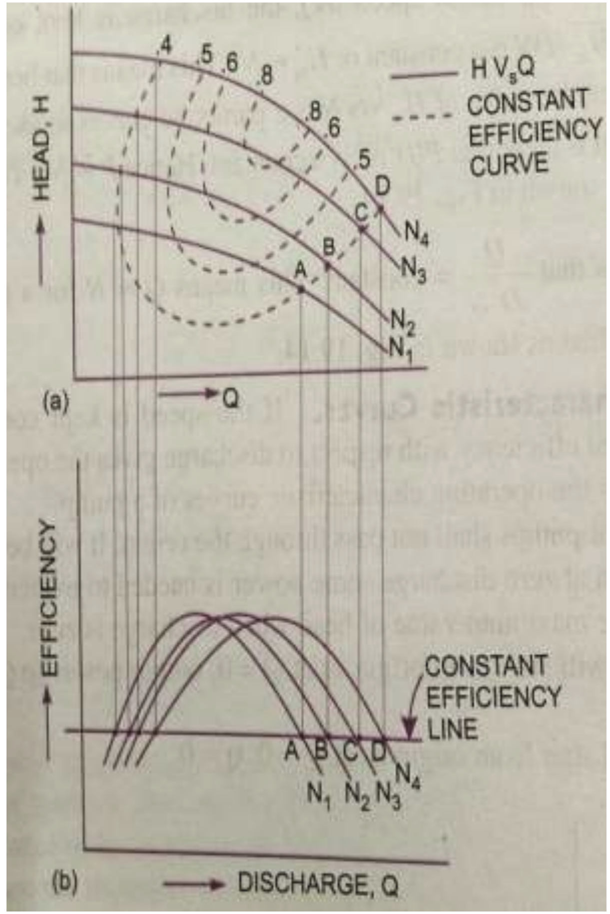
OPERATING CHARACTERISTIC CURVES

- If the speed is kept constant, the variation of Manometric head, power and efficiency with respect to discharge gives the operating characteristics of the pump.
- The input power curve for pumps shall not pass through the origin. It will be slightly away from the origin on the y-axis, as even at zero discharge some power is needed to overcome mechanical losses.
- The head curve will have maximum value of head when the discharge is zero.



CONSTANT EFFICIENCY OR MUSCHEL CURVES

- For obtaining constant efficiency curves for a pump, the head versus discharge curves and efficiency v/s discharge curves for different speeds are used.
- By combining these curves $(H \sim Q\ curves\ \eta \sim$



NET POSITIVE SUCTION HEAD (NPSH)

- The term NPSH is very commonly used selection of a pump. The minimum suction conditions are specified in terms NPSH.
- It is defined as the absolute pressure head at the inlet to the pump minus the vapour pressure head plus velocity head.

∴ NPSH = Absolute pressure head at inlet of pump – vapour pressure head (absolute units) + Velocity head

$$= \frac{p_1}{\rho g} - \frac{p_v}{\rho g} + \frac{{v_s}^2}{2g} \quad \_\_\_\_\_\_\_\_ (1)$$

(∵

- $$NPSH = \frac{p_a}{\rho g} - \frac{{v_s}^2}{2g} - h_s - h_{f_s} - \frac{p_v}{\rho g} + \frac{{v_s}^2}{2g}$$

$$= \frac{p_a}{\rho g} - \frac{p_v}{\rho g} - h_s - h_{f_s}$$

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

$$\left(\because \frac{p_a}{\rho g} = H_a = Atmospheric\ pressure\ head,\right)$$

$$= \left[\left(H_a - h_s - h_{f_s}\right) - H_v\right] \_\_\_\_\_\_\_\_ (2) \qquad \left(\because \frac{p_v}{\rho g} = H_v = \right.$$

TOPICS TO BE COVERED

- Example problems on centrifugal pump

LECTURE 5

Problems on Centrifugal Pump

PROBLEM 1

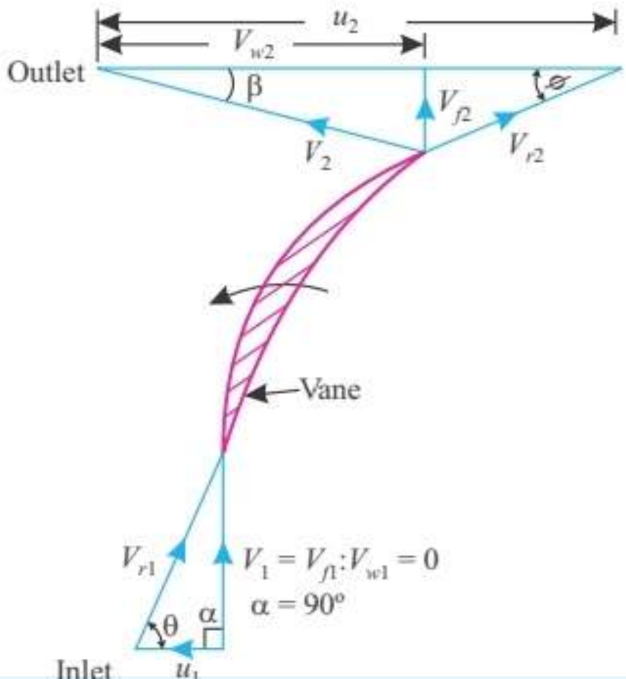
Example 3.1. *The impeller of a centrifugal pump has an external diameter of 450 mm and internal diameter of 200 mm and it runs at 1440 r.p.m. Assuming a constant radial flow through the impeller at 2.5 m/s and that the vanes at exit are set back at an angle 25°, determine:*

(*i*) *Inlet vane angle,*

(*ii*) *The angle, absolute velocity of water at exit makes with the tengent, and*

(*iii*) *The work done per N of water.*

Solution. Internal diameter of the impeller, D_1 = 200 mm = 0.2 m



External diameter of the impeller, D_2 = 450 mm = 0.45 m

Speed of impeller, N = 1440 r.p.m.

Velocity of flow, $V_{f1} = V_{f2} = 2.5$ m/s

Vane angle at outlet, $\phi = 25°$

(*i*) Inlet vane angle, θ:

Tangential velocity of impeller at inlet,

$$u_1 = \frac{\pi D_1 N}{60} = \frac{\pi \times 0.2 \times 1440}{60} = 15.08 \text{ m/s}$$

From velocity triangle at *inlet,* we have:

$$\tan\theta = \frac{V_{f1}}{u_1}, \text{ or, } \tan\theta = \frac{2.5}{15.08} = 0.1658$$

$$\therefore \quad \theta = \tan^{-1} 0.1658 = \mathbf{9.4°} \textbf{ (Ans.)}$$

(*ii*) The angle, absolute velocity of water at exit makes with the tangent, β:

Tangential velocity of impeller at outlet,

$$u_2 = \frac{\pi D_2 N}{60} = \frac{\pi \times 0{\cdot}45 \times 1440}{60} = 33.93 \text{ m/s}$$

From velocity triangle at *outlet,* we have:

$$V_{w2} = u_2 - \frac{V_{f2}}{\tan\phi}, \text{ or, } V_{w2} = 33.93 - \frac{2.5}{\tan 25°} = 28.57 \text{ m/s}$$

Now,
$$\tan\beta = \frac{V_{f2}}{V_{w2}} = \frac{2.5}{28.57} = 0.0875$$

$$\therefore \quad \beta = \tan^{-1} 0.0875 = \mathbf{5°} \textbf{ (Ans.)}$$

(*iii*) Work done per N of water:

$$\textit{Work done per N of water} = \frac{V_{w2}u_2}{g} = \frac{28.57 \times 33.93}{9.81} = \mathbf{98.81 \text{ Nm}} \textbf{ (Ans.)}$$

PROBLEM 2

Example 3.2. *A centrifugal pump is to discharge 0.118 m^3/s at a speed of 1450 r.p.m against a head of 25 m. The impeller diameter is 250 mm, its width at outlet is 50 mm and manometric efficiency is 75 percent. Determine the vane angle at the outer periphery of the impeller.* **[PTU]**

Solution. Discharge, $Q = 0.118\ m^3/s$

Speed $N = 1450$ r.p.m.

Speed, $H_{mano} = 25$ m

Diameter of impeller at outlet,

$D_2 = 250\ mm = 0.25\ m$

Width at outlet, $B_2 = 50\ mm = 0.05\ m$

Manometric efficiency, $\eta_{mano} = 75\%$

Vane angle at outlet, ϕ:

Tangential velocity of impeller at outlet,

$$u_2 = \frac{\pi D_2 N}{60} = \frac{\pi \times 0.25 \times 1450}{60} = 18.98\ m/s$$

Discharge, $Q = \pi D_2 B_2 \times V_{f2}$

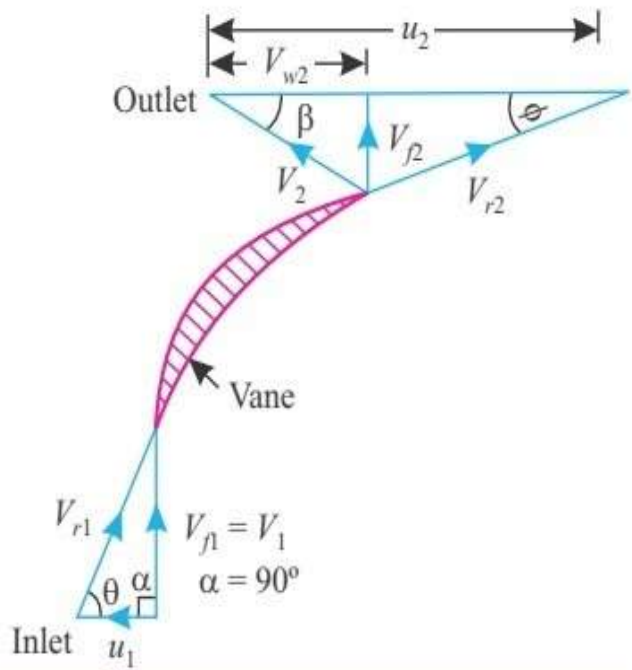


Fig. 3.8

$$\therefore \qquad V_{f2} = \frac{Q}{\pi D_2 B_2} = \frac{0.118}{\pi \times 0.25 \times 0.05} = 3.0 \text{ m/s}$$

Manometric efficiency, $\eta_{mano} = \dfrac{gH_{mano}}{V_{w2} u_2}$

or,
$$0.75 = \frac{9.81 \times 25}{V_{w2} \times 18.98}, \quad \text{or,} \quad V_{w2} = \frac{9.81 \times 25}{0.75 \times 18.98} = 17.23 \text{ m/s}$$

From velocity triangle at *outlet,* we have:

$$\tan \phi = \frac{V_{f2}}{u_2 - V_{w2}} = \frac{3.0}{18.98 - 17.23} = 1.7143$$

$$\therefore \qquad \phi = \tan^{-1} 1.7143 = \mathbf{59.74°}$$ **(Ans.)**

PROBLEM 3

Example 3.3. *The impeller of a centrifugal pump having external and internal diameters 500 mm and 250 mm respectively, width at outlet 50 mm and running at 1200 r.p.m. works against a head of 48 m. The velocity of flow through the impeller is constant and equal to 3.0 m/s. The vanes are set back at an angle of 40° at outlet. Determine:*

(*i*) *Inlet vane angle,*

(*ii*) *Work done by the impeller on water per second, and*

(*iii*) *Manometric efficiency.*

Solution. External diameter of impeller, D_2 = 500 mm = 0.05 m

Internal diameter, D_1 = 250 mm = 0.25 m

Width at outlet, B_2 = 50 mm = 0.05 m

Speed, N = 1200 r.p.m.

Head, H_{mano} = 48 m

Velocity of flow, $V_{f1} = V_{f2}$ = 3.0 m/s

Vane angle at outlet, ϕ = 40°

(*i*) Inlet vane angle, θ:

Refer to Fig. 3.8. From velocity triangle at *inlet*, we have:

$$\tan\theta = \frac{V_{f1}}{u_1} \quad \text{where, } u_1 = \frac{\pi D_1 N}{60} = \frac{\pi \times 0.25 \times 1200}{60} = 15.7 \text{ m/s}$$

Substituting the values of V_{f1} and u_1, we get:

$$\tan\theta = \frac{3.0}{15.7} = 0.191 \qquad \therefore \theta = \tan^{-1} 0.191 = \mathbf{10.81°} \ \textbf{(Ans.)}$$

(*ii*) Work done by the impeller:

Work done by the impeller on water per second is given by eqn. (3.1) as

$$= \frac{W}{g} V_{w2} u_2 = \frac{wQ}{g} \times V_{w2} u_2 \qquad \text{...(i)}$$

where, $Q = \pi D_2 B_2 \times V_{f2} = \pi \times 0.5 \times 0.05 \times 3.0 = 0.2356 \text{ m}^3/\text{s}$

Also, from velocity triangle at *outlet*, we have:

$$\tan\phi = \frac{V_{f2}}{u_2 - V_{w2}} = \frac{3.0}{31.41 - V_{w2}}, \quad \text{or,} \quad \tan 40° = \frac{3.0}{31.41 - V_{w_2}}$$

$$\left(\text{where, } u_2 = \frac{\pi D_2 N}{60} = \frac{\pi \times 0.5 \times 1200}{60} = 31.41 \text{ m/s}\right)$$

or, $$31.41 - V_{w2} = \frac{3.0}{\tan 40°}, \quad \text{or,} \quad V_{w_2} = 31.41 - \frac{3.0}{\tan 40°} = 27.83 \text{ m/s}$$

Substituting the values in eqn (*i*), we get the work done by the impeller

$$= \frac{9.81 \times 0.2356}{9.81} \times 27.83 \times 31.41 = \mathbf{205.95\ kNm}\ \textbf{(Ans.)}$$

$$(\because w = 9.81 \text{ kN/m}^3)$$

(*iii*) Manometric efficiency, (η_{mano}):

Manometric efficiency is given by eqn. (3.9) as:

$$\eta_{mano} = \frac{gH_{mano}}{V_{w2}u_2} = \frac{9.81 \times 48}{27.83 \times 31.41} = 0.5386 \text{ or } \mathbf{53.86\%}\ \textbf{(Ans.)}$$

PROBLEM 4

Example 3.4. *A centrifugal pump running at 800 r.p.m. is working against a total head of 20.2 m. The external diameter of the impeller is 480 mm and outlet width 60 mm. If the vanes angle at outlet is 40° and manometric efficiency is 70 percent, determine:*

(i) *Flow velocity at outlet,*
(ii) *Absolute velocity of water leaving the vane,*
(iii) *Angle made by the absolute velocity at outlet with the dirction of motion at outlet, and*
(iv) *Rate of flow through the pump.*

Solution. Speed, N = 800 r.p.m.; Head, H_{mano} = 20.2 m;
External diameter, D_2 = 480 mm = 0.48 m; Width at outlet, B_2 = 60 mm = 0.06 m;
Outlet vangle angle, ϕ = 40°; Manometric efficiency, η_{mano} = 70 %.

Refer to Fig. 3.9.

(i) Flow velocity at outlet, V_{f2} :

Tangential velocity of impeller at outlet,

$$u_2 = \frac{\pi D_2 N}{60} = \frac{\pi \times 0.48 \times 800}{60}$$

$$= 20.1 \text{ m/s}$$

Also, manometric efficiency is given by,

$$\eta_{mano} = \frac{gH_m}{V_{w2} u_2} \quad \text{...(Eqn 3.9)}$$

or, $$0.70 = \frac{9.81 \times 20.2}{V_{w2} \times 20.1}$$

or, $$V_{w2} = \frac{9.81 \times 20.2}{0.70 \times 20.1} = 14.08 \text{ m/s}$$

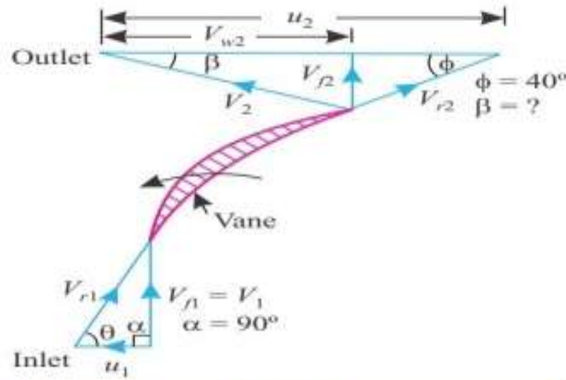


Fig. 3.9

From velocity triangle at outlet, we have:

$$\tan \phi = \frac{V_{f2}}{u_2 - V_{w2}} = \frac{V_{f2}}{(20.1 - 14.08)}$$

or, $$V_{f_2} = \tan \phi \,(20.1 - 14.08)$$
$$= \tan 40° \,(20.1 - 14.08) = \mathbf{5.05 \text{ m/s}} \textbf{ (Ans.)}$$

(ii) Absolute velocity of water leaving the vane, V_2 :

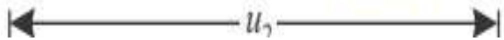
(*iii*) Angle made by the absolute velocity at outlet with the direction of motion, β:

From velocity triangle at *outlet*, we have:

$$\tan \beta = \frac{V_{f2}}{V_{w2}} = \frac{5.05}{14.08} = 0.3586 \quad \therefore \ \beta = \tan^{-1}(0.3586) = \mathbf{19.7^\circ} \ \textbf{(Ans.)}$$

(*iv*) Rate of flow through the pump, Q:

$$Q = \pi D_2 B_2 \times V_{f2} = \pi \times 0.48 \times 0.06 \times 5.05 = \mathbf{0.457\ m^3/s} \ \textbf{(Ans.)}$$



PROBLEM 5

Example 3.5. *A centrifugal pump impeller runs at 80 r.p.m. and has outlet vane angle of 60°. The velocity of flow is 2.5 m/s throughout and diameter of the impeller at exit is twice that at inlet. If the manometric head is 20 m and the manometric efficiency is 75 percent, determine:*

(i) The diameter of the impeller at the exit, and

(ii) Inlet vane angle.

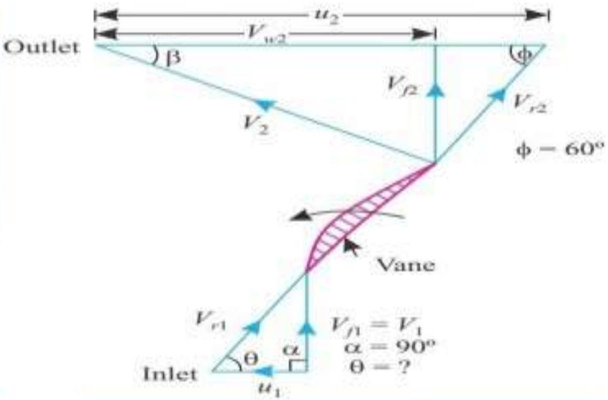


Fig. 3.10

Solution. Speed, N = 80 r.p.m.; Outlet vane angle, $\phi = 60°$; Velocity of flow, $V_{f1} = V_{f2} = 2.5$ m/s; manometric head, H_{mano} = 20 m; Manometric efficiency, $\eta_{mano} = 75\%$;

Diameter of the impeller at outlet,

$$D_2 = 2D_1 \text{(diameter at inlet)}$$

(i) The diameter of the impeller at the exit, D_2:

$$\eta_{mano} = \frac{gH_{mano}}{V_{w2}u_2} \quad \text{.... Eqn. (3.9)}$$

$$0.75 = \frac{9.81 \times 20}{V_{w2}u_2}, \quad \text{or,} \quad V_{w2}u_2 = \frac{9.81 \times 20}{0.75} = 261.6 \quad \text{...(}i\text{)}$$

From velocity triangle at outlet (Fig. 3.10), we have

$$\tan\phi = \frac{V_{f2}}{u_2 - V_{w2}}$$

or, $$u_2 - V_{w2} = \frac{V_{f2}}{\tan\phi}, \quad \text{or,} \quad V_{w2} = u_2 - \frac{V_{f2}}{\tan\phi}$$

or, $$V_{w2} = u_2 - \frac{2.5}{\tan 60°} = u_2 - 1.44$$

Substituting this value of V_{w2} in (*i*), we get:

$$(u_2 - 1.44)\,u_2 = 261.6, \quad \text{or} \quad u_2^2 - 1.44u_2 - 261.6 = 0$$

or, $$u_2 = \frac{1.44 \pm \sqrt{1.44^2 + 4 \times 261.6}}{2} = \frac{1.44 \pm 32.38}{2} = 16.91 \text{ m/s (ignoring –ve sign)}$$

Also, tangential velocity of impeller at outlet, $u_2 = \frac{\pi D_2 N}{60}$, or, $D_2 = \frac{60u_2}{\pi N}$

∴ $$D_2 = \frac{60 \times 16.91}{\pi \times 80} = 4.037 \text{ m} \simeq 4 \text{ m}$$ **(Ans.)**

From velocity triangle at *inlet*, we have:

$$\tan\theta = \frac{V_{f1}}{u_1} = \frac{2.5}{8.455} = 0.2957$$

$$\therefore \quad \theta = \tan^{-1}(0.2957) = \mathbf{16.47°}$$ **(Ans.)**

PROBLEM 6

Example 3.6. *A centrifugal pump impeller having external and intenal diameters 480 mm and 240 mm respectively is running at 100 r.p.m. The rate of flow through the pump is 0.0576 m^3/s and velocity of flow is constant and equal to 2.4 m/s. The diameters of the suction and delivery pipes are 180 mm and 120 mm respectively and suction and delivery heads are 6.2 m (abs.) and 30.2 m of water respectively. If the power required to drive the pump is 23.3 kW and the outlet vane angle is 45°, determine:*

(*i*) *Inlet vane angle,*
(*ii*) *The overall efficiency of the pump, and*
(*iii*) *The manometric efficiency of the pump.*

Solution. External diameter of the impeller,

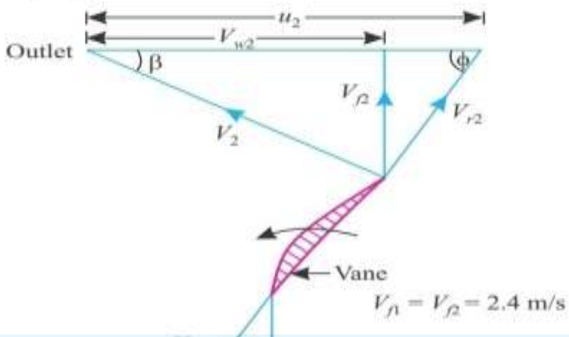
D_2 = 480 mm = 0.48 m,
Internal diameter, D_1 = 240 mm = 0.24 m,
Speed, N = 1000 r.p.m.,
Discharge, Q = 0.0567 m^3/s;
Velocity of flow, V_{f1} = V_{f2} = 2.4 m/s

The diameter of suction pipe,

D_s = 180 mm = 0.18 m

The diameter of delivery pipe,

D_d = 120 mm = 0.12 m
Suction head, h_s = 6.2 m (abs.)
Delivery head, h_d = 30.2 m (abs.)
Shaft power, P = 23.3 kW
Outlet vane angle, ϕ = 45°



$$u_2 = \frac{\pi D_2 N}{60} = \frac{\pi \times 0.48 \times 1000}{60} = 25.13 \text{ m/s}$$

From velocity triangle at outlet (Fig. 3.11), we have:

$$\tan\phi = \frac{V_{f2}}{u_2 - V_{w2}}, \text{ or, } \tan 45° = \frac{2.4}{25.13 - V_{w2}}, \text{ or, } 25.13 - V_{w2} = \frac{2.4}{\tan 45°} = 2.4$$

$$\therefore \quad V_{w2} = 25.13 - 2.4 = 22.73 \text{ m/s}$$

Substituting the values in the above equation, we get:

$$\eta_{mano} = \frac{9.81 \times 25.03}{22.73 \times 25.13} = 0.43 \text{ or } \mathbf{43\%} \text{ (Ans.)}$$

PROBLEM 7

Example 3.7. *It is required to deliver 0.048 m^3/s of water to a height of 24 m through a 150 mm diameter pipe and 120 m long, by a centrifugal pump. If the overall efficiency of the pump is 75 percent and co-efficient of friction, f = 0.01 for thr pipe line, find the power required to drive the pump.*

Solution. Rate of flow, $Q = 0.048\ m^3/s$; Height, $H_{stat} = h_s + h_d = 24$ m

Diameter of pipe, $D_s = D_d = D = 150$ mm, or, 0.15 m,

Length, $L_s + L_d = L = 120$ m

Overall efficiency, $\eta_o = 75\%$

Co-efficient of friction, $f = 0.01$

Power required to drive the pump, P:

$$\text{Velocity of water in pipe, } V_s = V_d = V = \frac{Q}{\text{Area of pipe}} = \frac{0.048}{\frac{\pi}{4} \times 0.15^2} = 2.7 \text{ m/s}$$

Loss of head due to friction in pipe,

$$(h_{fs} = h_{fd}) = \frac{4fLV^2}{D \times 2g} = \frac{4 \times 0.01 \times 120 \times 2.7^2}{0.15 \times 2 \times 9.81} = 11.89 \text{ m}$$

The manometric head (H_{mano}) is given by eqn. (3.7) as:

$$H_{mano} = (h_s + h_d) + (h_{fs} + h_{fd}) + \frac{V_d^2}{2g}$$

$$= 24 + 11.89 + \frac{2.7^2}{2 \times 9.81} = 36.26 \text{ m}$$

Using the relation : $\eta_0 = \dfrac{wQH_{mano}}{P}$, we get:

$$0.75 = \frac{9.81 \times 0.048 \times 36.26}{P} \qquad (\because w = 9.81 \text{ kN/m}^3)$$

or $$P = \frac{9.81 \times 0.048 \times 36.26}{\ } = 22.76 \text{ kW} \quad \textbf{(Ans.)}$$

TOPICS TO BE COVERED

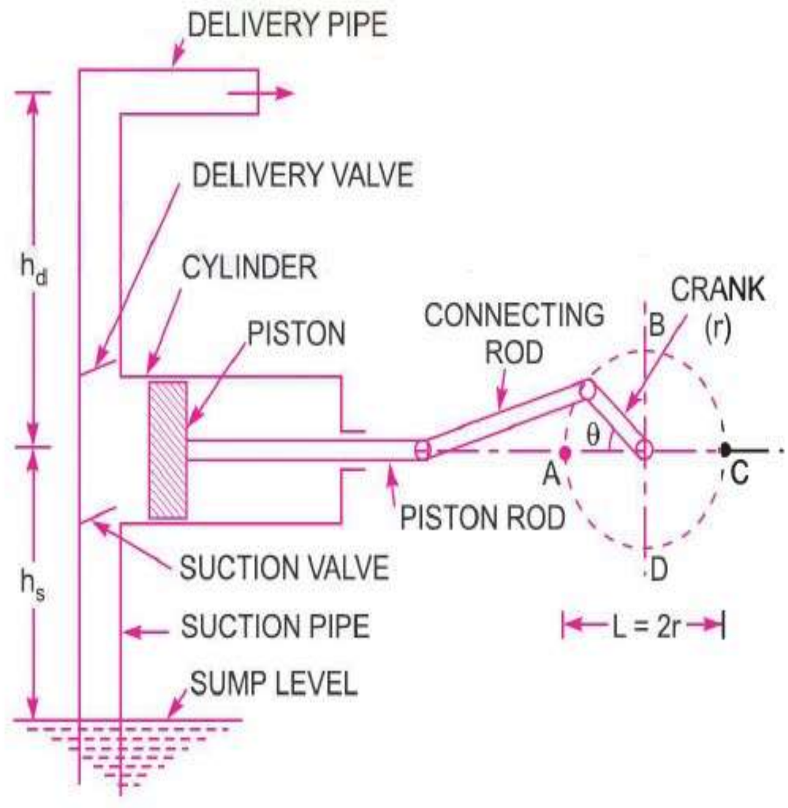
- Introduction
- Discharge through a reciprocating pump
- Work done by a reciprocating pump

LECTURE 6

Reciprocating pump

RECIPROCATING PUMPS

- The mechanical energy is converted in to hydraulic energy (pressure energy) by sucking the liquid in to a cylinder in which a piston is reciprocating, which exerts the thrust on the liquid and increases its hydraulic energy (pressure energy) the pump is known as reciprocating pump.



- A single acting reciprocating pump consists of a piston, which moves forwards and backwards in a close fitting cylinder.
- The movement of the 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 or 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.

- When the crank starts rotating, the piston moves to and fro in the cylinder. When the crank is at A the piston is at the extreme left position in the cylinder.
- As the crank is rotating from A to C (i.e. from $\theta = 0\ to\ 180^0$) the piston is moving towards right in the cylinder. The movement of the piston towards right creates a partial vacuum in the cylinder.
- But on the surface of the liquid in the sump atmospheric pressure in acting, which is more than the pressure inside the cylinder.
- Thus the liquid is forced in the suction pipe from the sump.
- This liquid opens the suction valve and enters the cylinder.

- When crank is rotating from C to A (i.e. from $\theta = 180^0\ to\ 360^0$), the piston from its extreme right position starts moving towards left in the cylinder.
- The movement of the piston towards the left increases the pressure on the liquid inside the cylinder more than atmospheric pressure.
- Hence the suction valve closes and delivery valve opens.
- The liquid is forced in to the delivery pipe and is raised to the required height.

DISCHARGE THROUGH A RECIPROCATING PUMP

- Consider a single acting reciprocating pump.

Let D = Diameter of cylinder

A = Cross-sectional area of piston or cylinder $= \frac{\pi}{4} D^2$

r = Radius of crank

N = r.p.m. of the crank

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

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

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

- Volume of water delivered in one revolution or Discharge of water in one revolution = Area × Length of stroke $= A \times L$

- Number of revolutions per second $= \frac{N}{60}$
- Discharge of pump per second Q = Discharge in one revolution × No. of revolutions per sec $= A \times L \times \frac{N}{60}$

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

- Weight of water delivered per second $W = \rho \times g \times Q$

$$= \frac{\rho g ALN}{60} \qquad \text{\_\_\_\_\_\_ (1)}$$

WORK DONE BY RECIPROCATING PUMP

- Work done per second = Weight of water lifted per second ×Total height through which water is lifted $= W \times [h_s + h_d]$ ______(2)

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

- From equation (1) weight of water is given by

$$W = \frac{\rho g A L N}{60}$$

- Substituting the value of W in equation (2), we get
- Work done per second $= \frac{\rho g A L N}{60} \times (h_s + h_d)$
- Power required to drive the pump in kW

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

$$P = \frac{\rho g \times ALN \times (h_s + h_d)}{\ldots} kW$$

TOPICS TO BE COVERED

- Slip of Reciprocating pump
- Negative Slip of Reciprocating pump
- Indicator diagram
- Ideal Indicator diagram

LECTURE 7

Slip & Indicator diagram of Reciprocating pump

SLIP OF RECIPROCATING PUMP

- Slip of a pump is defined as the difference between the theoretical discharge and actual discharge of a pump.
- The actual discharge of pump is less than the theoretical discharge due to leakage.
- The difference of the theoretical discharge and actual discharge is known as slip of the pump.
- Hence $slip = \ Q_{th} - Q_{act}$
- But slip is mostly expressed as percentage slip

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

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

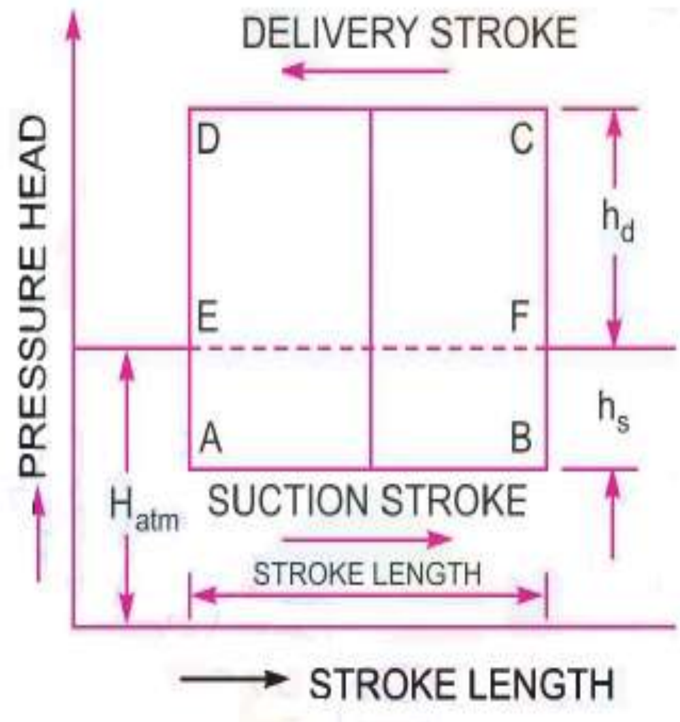
Where C_d = Co-efficient of discharge.

NEGATIVE SLIP OF THE RECIPROCATING PUMP

- Slip is equal to the difference of theoretical discharge and actual discharge.
- If actual discharge is more than the theoretical discharge, the slip of the pump will become –ve.
- In that case the slip of the pump is known as negative slip.
- Negative slip occurs when the delivery pipe is short, suction pipe is long and pump is running at high speed.

INDICATOR DIAGRAM

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



IDEAL INDICATOR DIAGRAM

- The graph between pressure head in the cylinder and the stroke length of piston for one complete revolution of the crank under ideal conditions is known as ideal indicator diagram.
- Line EF represents the atmospheric pressure head equal to 10.3 meters of water.
- Let H_m = Atmospheric pressure head = 10.3 m of water

  L = Length of the stroke

  h_s = Suction head and

  h_d = Delivery head
- During suction stroke, the pressure head in the cylinder is constant and equal to suction head(h_s), which is below the atmospheric pressure head(H_{atm}) by a height ofh_s.

- The pressure head during suction stroke is represented by a horizontal line AB which is below the line EF by a height of $'h_s'$
- During delivery stroke, the pressure head in the cylinder is constant and equal to delivery head (h_d), which is above the atmospheric head by a height of $'h_d'$.
- Thus the pressure head during the delivery stroke is represented by a horizontal line CD, which is above the line EF by a height of h_d.
- Thus for one complete revolution of crank, the pressure head in the cylinder is represented by the diagram ABCD.
- This diagram is known as ideal indicator diagram.

- The work done by the pump per second $= \frac{\rho gALN}{60} \times (h_s + h_d)$

$$= K \times L(h_s + h_d)$$

$$\propto L \times (h_s + h_d) \quad \_\_\_\_(1)$$

Where $K = \frac{\rho gAN}{60} = constant$

- Area of Indicator diagram $= AB \times BC = AB \times (BF + FC) = L \times (h_s + h_d)$
- Substituting this value in equation (1), we get

Work done by pump ∝ Area of Indicator diagram

TOPICS TO BE COVERED

- Example problems on Reciprocating pump
- Applications
- Assignment questions

LECTURE 8

Problems on Reciprocating Pump

PROBLEM 1

- A single acting reciprocating pump running at 50 r.p.m. delivers $0.01\, m^3/s$ of water. The diameter of piston is 200mm and stroke length in 400mm. Determine **i)** The theoretical discharge of pump **ii)** Co-efficient of discharge **iii)** Slip and the percentage slip of pump.

Sol: Given data

The speed of the pump, N = 50 rpm

Actual discharge, $Q_{act} = 0.01\, m^3/s$

Dia. Of piston, D = 200mm = 0.2m

Area, $A = \frac{\pi D^2}{4} = \frac{\pi}{4}(0.2)^2 = 0.031416\, m^2$

i) The theoretical discharge, $Q_{th} = \frac{ALN}{60} = \frac{0.031416 \times 0.4 \times 50}{60}$

$$= \mathbf{0.01047 m^3/s}$$

ii) The Co-efficient of discharge, $C_d = \frac{Q_{act}}{Q_{th}} = \frac{0.01}{0.01047}$

$$= \mathbf{0.955}$$

iii) Slip, $Q_{th} - Q_{act} = 0.01047 - 0.01$

$$= \mathbf{0.00047 m^3/s}$$

Percentage Slip $= \left(\frac{Q_{th}-Q_{act}}{Q_{th}}\right) \times 100 = \frac{0.01047-0.01}{0.01047} \times 100$

$$= \mathbf{4.489\%}$$

PROBLEM 2

- A double acting reciprocating pump, running at 40 r.p.m. is discharging 1.0m3 of water per minute. The pump has a stroke of 400 mm. the diameter of piston is 200 mm. the delivery and suction head are 20m and 5m respectively. Find the slip of the pump and power required to drive the pump.

Sol: Given data,

Speed of the pump, N = 40 r.p.m.

Actual discharge, $Q_{act} = 1\, m^3/s = \frac{1}{60} = 0.01666 m^3/s$

Stroke, L = 400mm = 0.4m

Diameter of piston, D = 200 mm = 0.2m

$\therefore \quad Area, A = \frac{\pi}{4} D^2 = \frac{\pi}{4} (0.2)^2 = 0.031416 m^2$

Suction Head, $h_s = 5m$

Delivery head, $h_d = 20\, m$

- Theoretical discharge for double acting pump

$$Q_{th} = \frac{2ALN}{60} = \frac{2 \times 0.31416 \times 0.4 \times 40}{60}$$

$$= \mathbf{0.01675 m^3/s}$$

- Slip= $Q_{th} - Q_{act} = 0.01675 - 0.1666$

$$= \mathbf{0.00009 m^3/s}$$

- Power required to drive the double acting pump

$$P = \frac{2 \times \rho g \times ALN \times (h_s + h_d)}{60{,}000}$$

$$= \frac{2 \times 1000 \times 9.81 \times 0.031416 \times 0.4 \times 40 \times (5+20)}{60{,}000}$$

$$= \mathbf{4.109 kW}$$

APPLICATIONS

- Centrifugal pumps are used in variety of applications. Almost 70-80% centrifugal pumps are used in industry or for domestic purpose.
- Water supply and irrigation
- Chemical, food, Petrochemical industries
- Mining, domestic appliances
- Reciprocating pumps are mainly used in oil and gas industry.
- They can also be used sugar industries, soap and detergent industries, water treatment plants and Food & beverages.
- Have huge demand in cryogenic applications.

ASSIGNMENT QUESTIONS

- A centrifugal pump delivers water to a height of 22 m at a speed of 800 rpm. The velocity of flow is constant at a speed of 2m/s and the outlet vane angle is 450. If the pump discharges 225 litres of water / second, find the diameter of the impeller and width of the impeller.
- A centrifugal pump having outer diameter equal to two times the inner diameter and running at 1200 rpm works against a total head of 75m. The velocity of flow through the impeller is constant and is equal to 3m/s. The vanes are set back at an angle of 300 at outlet. If the outer diameter of the impeller is 600 mm and width at outlet is 50 mm, determine (i) vane angle at inlet (ii) work done per second by impeller (iii) manometric efficiency.
- What is the working principle of a reciprocating pump? Explain its working with the help of an indicator diagram.

- A single acting reciprocating pump having cylinder diameter of 150 mm and stroke 300 mm is used to raise water through a total height of 30m. Find the power required to drive the pump, if the crank rotates at 60 rpm.
- A fluid is to be lifted against a head of 120m. The pumps that run at a speed of 1200 rpm with rated capacity of 300 litres/sec are available. How many pumps are required to pump the water if specific speed is 700.