Turbomachines - 18AE46

Old VTU Question's Answers

Module – 5

Syllabus:
Hydraulic pumps: Centrifugal and axial pumps. Manometric head, suction head, delivery head; manometric efficiency, hydraulic efficiency, volumetric efficiency, overall efficiency; multi stage pumps. Characteristics of pumps.
Hydraulic turbines: Classification; Module quantities; Pelton wheel, Francis turbine, Kaplan turbine and their velocity triangles. Draft tubes and their function. Characteristics of hydraulic turbines.

Part – A Questions

1. With the help of a neat sketch, explain the parts and working principle of a centrifugal pump.
or
With a neat sketch, explain the terminology of centrifugal PUMP.

4.15 MAIN PARTS OF A CENTRIFUGAL PUMP

Figure 4.12 illusrates the following main parts of a centrifugal pump.

1. Impeller
2. Casing
3. Suction pipe
4. Delivery pipe
5. Delivery valve or check valve or regulating valve.

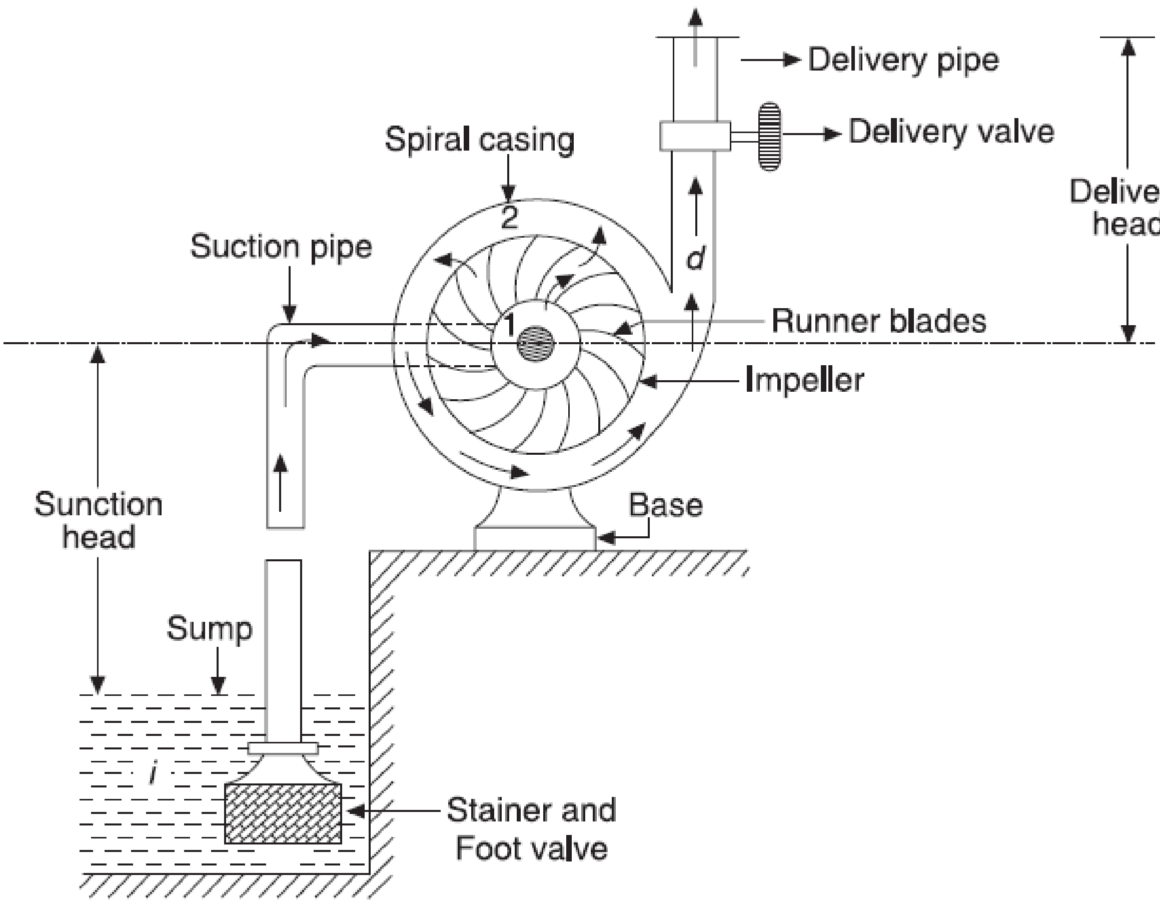


Figure 4.12 Main parts of a centrifugal pump.

These devices producing the head due to the centrifugal action. The centrifugal pump mainly consists of (a) ***Impeller*** by means of which water will be lifted from low level to high level and is mount on a shaft and (b) ***Casing*** which directs the fluid and also it rises the static head. For machines of small capacity, the impeller will be cast in Bronze, but where the pump has to handle large capacity or to be handle corrosive fluids, the impeller may be of cast or stainless steel. Other alloys like aluminium - bronze are also sometimes used.

The dynamic pressure generated by the forced vortex motion of the impeller lifts water from low level to high level. The velocity of water leaving the impeller is high & fixed blades are often used around the impeller to act as a diffuser, which rises the static pressure. The flow of water through the diverging passages between the guide vanes brings about the desired reduction of velocity and raise of pressure.

The pumps provided with diffusers are called ***diffusion pumps*** or ***turbine pumps*** since they resembles Francis turbines which has inlet guide vanes. Usually, these are meant for high heads at small discharge. Pumps with volute chamber is ***volute pumps***. Volute chamber is a outer housing provided around the impeller and it has continuously increasing flow area towards the outward flow of the fluid which slows down the exit velocity, thereby the static pressure increases.

Centrifugal pumps based on type of vane are of (i) Backward curved blades (ii)Radial type and (iii) Forward curved type. They may be of single suction or double suction type and may be single and multistage. When high heads are required then the pumps are connected in series and when high discharge is required then the pumps are to be connected in parallel.

8.2 Terminology of Centrifugal Pump

(1) **Suction Head (h_S)**: It is the vertical height of the centre line of the pump above the water surface in the sump. This height is also called suction lift and is denoted as h_S as in Fig..8.1.

(2) **Delivery Head (h_d)**: It is the vertical height between the centre line of the pump and the water surface in the tank to which water is delivered . It is denoted as h_d as in Fig.8.1.

(3) **Static Head (H_s)**:

Static head is the vertical distance between the liquid level in the sump and the delivery tank. It is denoted by H_S. Therefore the static head, $H_S = h_S + h_d$

(4) **Manometric Head (H_m) or Effective Head** : It is the total head or lift that must be produced by the pump to satisfy the external requirements. It includes all the losses like frictional losses, Leakage losses etc. If there is no losses in the impeller, then the manometric head will be equal to the head imparted to the liquid by the impeller (i.e.,Euler's head).

$$\therefore \quad H_m = \frac{U_2 V_{u2}}{g} \text{ - loss of head in the impeller \& casings.}$$

$$= H_e \text{- losses} \qquad (8.1)$$

Also, $$H_m = h_S + h_d + h_{fs} + h_{fd} + V_d^2/2g = H_S + h_{fs} + h_{fd} + V_d^2/2g \qquad (8.1a)$$

Where h_{fs} and h_{fd} are the frictional loss due to friction in suction and delivery pipes respectively, and V_d is the velocity of the fluid in the delivery pipe.

In other words, the actual head against which the pump works is known as the manometric head (H_m). It is, therefore in general, equal to Euler's head minus the various hydraulic losses.

By applying the Bernoulli's equation between the sump and the deliver tank level, one can write,

$$H_m = \frac{p_d - p_s}{\rho g} = h_s + h_d + h_{fs} + h_{fd} + V_s^2/2g = H_e - (V_s^2 - V_d^2)/2g \qquad (8.2)$$

Where h_{fs} and h_{fd} are the frictional losses due to friction in the suction and delivery pipes respectively.

2. Draw the centrifugal pump and explain the terminologies:
(i) Manometric efficiency (ii) Mechanical efficiency (iii) Volumetric efficiency (iv) Overall efficiency (v) Hydraulic efficiency (vi)NPSH / Net positive suction head (vii) Operating characteristic curves

4.18.1 Manometric Efficiency (η_{mano})

It is the ratio of the manometric head to the head imparted by the impeller to the fluid.

$$\left.\begin{aligned}\eta_{mano} &= \frac{H_m}{V_{w2}u_2/g} = \frac{H_m g}{V_{w2}u_2} \\ &= \frac{\text{Power delivered by the pump, measured in head}}{\text{Power imparted by the impeller (head imparted by impeller)}}\end{aligned}\right] \qquad (4.37)$$

$$= \frac{H_m}{H_m + \text{losses}} = \frac{H_m}{H_m + H_L} = \frac{\text{Water Power}}{\text{Power of the impeller}}$$

Head imparted by the runner (impeller) is as per the ideal velocity triangle.

4.18.2 Mechanical Efficiency (η_m)

It is the ratio of the power actually delivered by the impeller to the power supplied at the shaft.

$$\eta_m = \frac{\text{Mechanical energy supplied to rotor}}{\text{Mechanical energy supplied to shaft}}$$

$$\eta_m = \frac{V_{w2}\,u_2 / g_c}{V_{w2}\,u_2 / g_c + \text{Mechanical losses}} = \frac{\text{Impeller power}}{\text{Shaft power (SP)}} \qquad (4.38a)$$

Power at the impeller = Power actually delivered by the impeller = Work done by impeller per second.

4.18.3 Hydraulic Efficiency (η_H)

$$\eta_H = \frac{\text{Useful hydrodynamic energy in fluid}}{\text{Mechanical energy supplied to rotor}} \qquad (4.38b)$$

4.18.4 Volumetric Efficiency (η_v)

$$\eta_v = \frac{\text{Actual discharge}}{\text{Theoretical discharge}} \qquad (4.38c)$$

4.18.5 Overall Efficiency (η_o)

It is the ratio of the power output of the pump to the power input to the pump.

$$\eta_o = \frac{\text{Water power}}{\text{Shaft power}} = \frac{WP}{SP} \quad (4.39)$$

$$\eta_o = \frac{\text{Water Power}}{\text{Impeller Power}} \times \frac{\text{Impeller Power}}{\text{Shaft Power}} = \eta_{mano} \times \eta_m \quad (4.40)$$

9.3.2 Operating Characteristic Curves :

If the speed is constant the variation of manometric head, power and efficiency with response to discharge gives the operating characteristics of the pump. Fig.9.7 shows the operating characteristics of the pump.

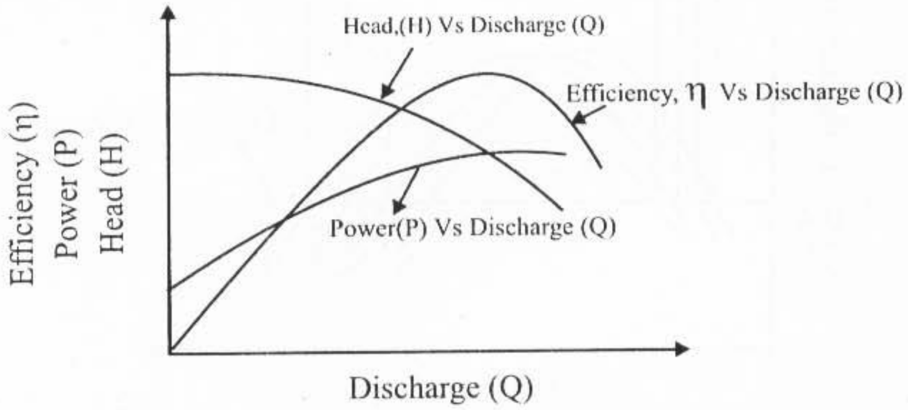


Fig.9.7 Operating characteristics of the pump

In order to overcome the initial torque, a small amount of power is consumed even though there is no flow and hence power curve will not start from origin. Head corresponding to no flow condition is called the shut-off head and when flow starts, the head developed by the pump reduces parabolically due to the various losses through the system. Even when pump requires some power for no flow condition, the water power delivered by the pump is zero and hence the efficiency starts from origin as shown in Fig. 9.7.

8.10.1 Net Positive Suction Head (NPSH)

Net Positive Suction Head (NPSH) is the head required at the pump inlet to avoid the water from cavitation or boiling. The cavitation is likely to occur on suction side of the pump as the lowest pressure can exist in this region. The NPSH is defined as,

$$NPSH = \frac{p_s}{\rho g} + \frac{V_s^2}{2g} - \frac{p_{Vap}}{\rho g} \quad (8.16)$$

Where V_s is the velocity of the water in suction side, p_s and p_{Vap} are the pressure at the inlet and the vapour pressure in absolute units. NPSH indicates the height of the pump axis from the water surface in the sump to which it can be installed to avoid cavitation problem.

3. Show that the pressure rise in the impeller of a centrifugal pump when the frictional and other losses in the impeller are neglected is given, $\frac{1}{2g}\left[Vf_1^2+U_2^2-Vf_2^2\ \mathrm{cosec}^2\beta_2\right]$, Where Vf_1 and Vf_2 are the flow velocities at inlet and outlet of the impeller, U_2 = tangential speed of impeller at exit, β_2 = Exit blade angle.

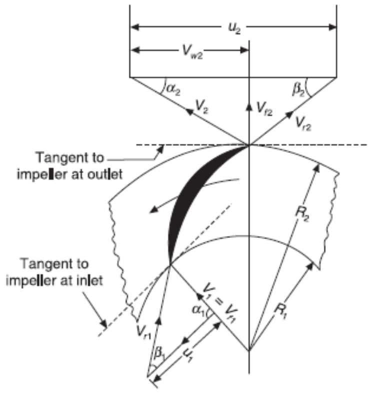


Figure 4.18 Velocity triangles for centrifugal pumps.

4.20 PRESSURE RISE IN PUMP, IMPELLER AND MANOMETRIC HEAD

Applying Bernolli's equation to inlet and outlet,

Energy at inlet + work done by impeller = Energy at outlet.

From Eq. (4.48), $V_1 = V_{f1}$, $Z_1 = Z_2$ and neglecting losses

$$\frac{p_1}{\rho g}+\frac{V_{r1}^2}{2g}+\frac{(u_2^2-u_1^2)}{2g}=\frac{p_2}{\rho g}+\frac{V_{r2}^2}{2g};\quad \frac{(p_2-p_1)}{\rho g}=\frac{(V_{r1}^2-V_{r2}^2)}{2g}+\frac{(u_2^2-u_1^2)}{2g} \qquad (4.53)$$

Substituting Eqs. (4.44) and (4.45) in RHS of (4.53),

$$=\frac{\cancel{u_1^2}+V_1^2-V_2^2-\cancel{u_2^2}+2u_2V_{w2}+\cancel{u_2^2}-\cancel{u_1^2})}{2g}=\frac{V_1^2-V_2^2+2u_2V_{w2}}{2g}$$

$$=\frac{V_{f1}^2-V_2^2+2V_{w2}u_2}{2g}=\frac{V_{f1}^2-(V_{w2}^2+V_{f2}^2)+2V_{w2}u_2}{2g} \qquad (\because\ V_2^2=V_{w2}^2+V_{f2}^2)$$

$$=\frac{V_{f1}^2-V_{w2}^2-V_{f2}^2+2V_{w2}u_2}{2g}$$

$$=\frac{V_{f1}^2-V_{f2}^2-(u_2-V_{f2}\cot\beta_2)^2+2u_2(u_2-V_{f2}\cot\beta_2)}{2g}$$

$$(\because\ V_{w2}=u_2-V_{f2}\cot\beta_2)$$

$$=\frac{V_{f1}^2-V_{f2}^2-u_2^2-V_{f2}^2\cot^2\beta_2+2u_2V_{f2}\cot\beta_2+2u_2^2-2u_2V_{f2}\cot\beta_2}{2g}$$

$$=\frac{V_{f1}^2-V_{f2}^2+u_2^2-V_{f2}^2\cot^2\beta_2}{2g}=\frac{V_{f1}^2+u_2^2-V_{f2}^2(1+\cot^2\beta_2)}{2g}$$

$$\therefore\quad \frac{(p_2-p_1)}{\rho g}=\frac{(V_{f1}^2+u_2^2-V_{f2}^2\ \mathrm{cosec}^2\beta_2)}{2g}=\text{pressure rise (m)} \qquad (4.54)$$

4. Derive an expression for minimum speed for starting centrifugal pumps.

When the pump is started, there will be no flow until the pressure difference in the impeller is large enough to overcome the manometric head. If the impeller is rotating and if there is no flow, then the water is rotating in a forced vertex.

Centrifugal pressure head for no flow of water = $(u_2^2-u_1^2)/2g$

Unless this pressure head is equal to or greater than the manometric head, the pump will not deliver water. By this the minimum speed can be determined.

The flow will commence only if $(u_2^2 - u_1^2)/2g \geq H_m$ (4.55)

i.e. $$\frac{1}{2g}\left[\frac{\pi d_2 N}{60}\right]^2 - \frac{1}{2g}\left[\frac{\pi d_1 N}{60}\right]^2 \geq H_m \qquad \left[\eta_{mano} = \frac{gH_m}{V_{w2}u_2}\right]$$

i.e. $$\frac{1}{2g}\left[\frac{\pi d_2 N}{60}\right]^2 - \frac{1}{2g}\left[\frac{\pi d_1 N}{60}\right]^2 \geq \eta_{mano}\frac{V_{w2}u_2}{g} \geq \eta_{mano}\frac{V_{w2}}{g}\left[\frac{\pi d_2 N}{60}\right]$$

For minimum speed, using the equal sign,

$$\frac{\pi^2 N^2}{2g}\left[\frac{(d_2^2 - d_1^2)}{3600}\right] = \eta_{mano}\frac{V_{w2}}{g}\frac{\pi d_2 N}{60} \tag{4.56}$$

$$\therefore \qquad N = \frac{\overset{60}{\cancel{3600}} \times 2\eta_{mano} V_{w2} d_2}{\pi \times (d_2^2 - d_1^2) \times \cancel{60}} = \text{minimum speed for a centrifugal pump.}$$

5. Explain the condition for using multistage pumps in series and parallel.

If the centrifugal pump consists of two or more impellers, then the pump is called *multistage pump*. Depending on whether (i) to produce high head or (ii) to discharge more quantity of water, the rotors on the same shaft or on different shafts are mounted. Multistage pump is used for following reasons: (a) the limitation of using bigger size impeller which increases the cost (b)the outlet blade angle not adviceable to increase more than 45^0 which leads to the reduced overall efficiency and (c) a single impeller can be used to develop a head of 50 m generally.

8.8.1 Pumps in Series

To develop high head, the impellers should be connected in series on the same shaft as shown in Fig.8.6 (a).

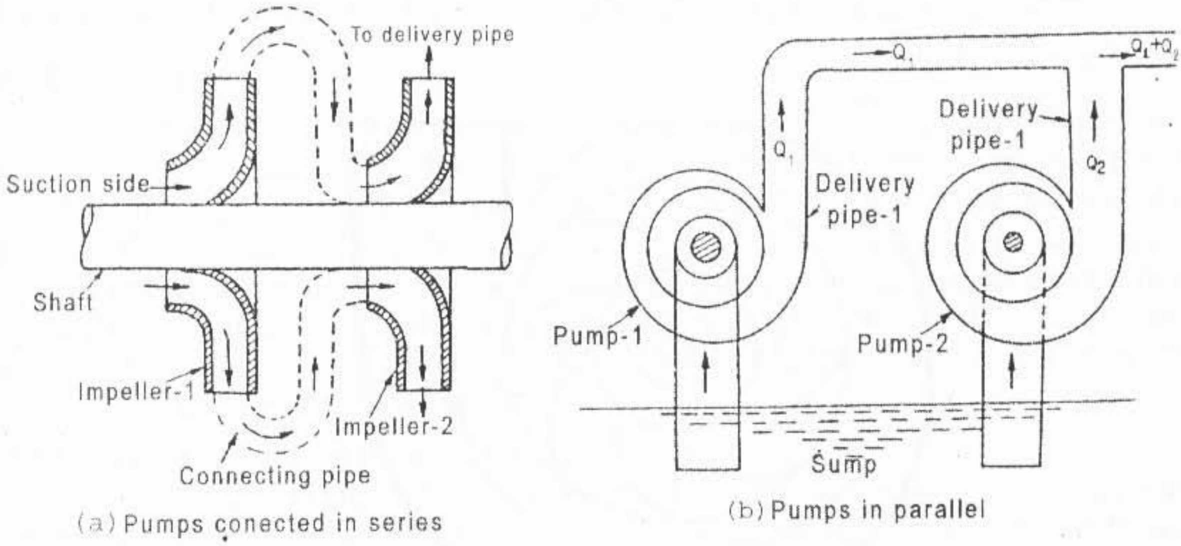


Fig 8.6 Multistage pumps

The water enters the Impeller-1 through the suction pipe and the pressure increases in it. The high pressure water from the Impeller-1 is then enter the Impeller-2 where the pressure increases further. The flow generally is same in both the impeller. The head produced by the combined impellers will be higher than either of one, but need not be sum of them. If n be the number of identical impellers and each produce a head of H_m, then the total head produced is

$$\text{Total head developed}, H_T = n\ H_m \qquad (8.14)$$

8.8.2 Pumps in Parallel

In order to obtain large discharge, the pumps are to be connected in parallel as shown in Fig.8.6(b). Each pump lifts water from a common sump and delivers to the common pipe to which the delivery end of each are connected. Each of the pump works against the same head. For identical pumps working under the same head the discharge will be the sum of the flow rates of each pump. If n be the number of identical impellers, each delivers the same flow rates which works under same head, then the total discharge from the multistage pumps is

$$\text{Total discharge}, Q_T = n\ Q \qquad (8.15)$$

6. A centrifugal pump having 120cm diameter pumps 1880 lps running at 200rpm. If the radial flow velocity is 2.5m/s, exit vane angle tangent to the impeller is 26^0, determine the manometric efficiency and the minimum speed to start the pump against a head of 6m. The ID of impeller is 60cm.

or

A centrifugal pump with 1.2m diameter runs at 200rpm and pumps 1.88 m^3/s, the average lift being 6m. The angle which the vane make at exit with the tangent to the impeller is 26^0 and the radial velocity is 2.5m/s. Determine the manometric efficiency and the least speed to start pumping if the inner diameter of the impeller is 0.6m.

Solution: Data:

Impeller outer diameter:	$d_2 = 1.2$ m
Speed:	$N = 200$ rpm
Discharge:	$Q = 1.88$ m^3/s
Total head:	$H_m = 6$ m
Vane angle at exit:	$\beta_2 = 26°$
Radial velocity at exit:	$V_{f2} = 2.5$ m/s
Impeller inlet diameter:	$d_1 = 0.6$ m

To determine: η_{man}, N_L

$$u_2 = \frac{\pi d_2 N}{60} = \frac{\pi \times 1.2 \times 200}{60} = 12.57 \text{ m/s}$$

$$u_1 = \frac{\pi d_1 N}{60} = \frac{\pi \times 0.6 \times 200}{60} = 6.285 \text{ m/s}$$

From exit velocity triangle,

$$\tan \beta_2 = \frac{V_{f2}}{u_2 - V_{w2}} = \frac{2.5}{12.57 \times V_{w2}} = \tan 26°$$

$\therefore \quad V_{w2} = 7.44$ m/s

(a) **Manometric efficiency (η_{man}):**

$$\eta_{man} = \frac{gH_m}{V_{w2}\, u_2} = \frac{9.81 \times 6 \times 100}{7.44 \times 12.57} = 63\% \qquad \textbf{Ans.}$$

(b) **Minimum speed (N_L):**

We have $$\frac{u_2^2 - u_1^2}{2g} = H_m$$

$$\left[\frac{\pi d_2 N_L}{60}\right]^2 - \left[\frac{\pi d_1 N_L}{60}\right]^2 = 2gH_m = 2 \times 9.81 \times 6$$

or $$\frac{\pi^2 N_L^2}{60^2}(d_2^2 - d_1^2) = 2 \times 9.81 \times 6$$

$\therefore$ $$N_L = \frac{2 \times 9.81 \times 6 \times 60 \times 60}{\pi^2 (d_2^2 - d_1^2)} = \frac{2 \times 9.81 \times 6 \times 3000}{\pi^2 (1.2^2 - 0.6^2)}$$

$$N_L = 199.39 \text{ rpm} \qquad \textbf{Ans.}$$

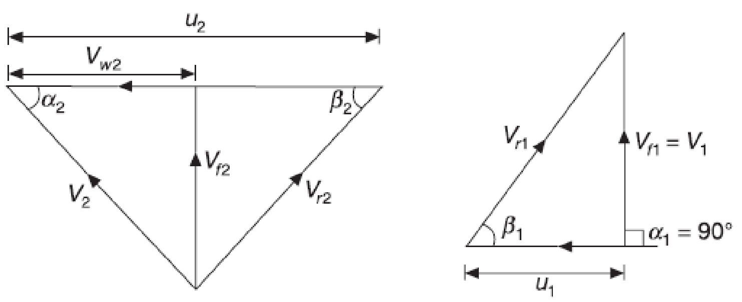
7. A centrifugal pump delivers 50 lit/s of water against a head of 24 m, running at 1500 rpm. The velocity of flow 2.4 m/s is constant and the blades are set back to 30°. The inner diameter as half the outer dia. If manometric efficiency is 80% from the blade angle at inlet and power required to drive the pump.

EXAMPLE 4.32 A centrifugal pump delivers 50 lit of water per second against a total head of 24 m at 1500 rpm. The velocity of flow is maintained constant at 2.4 m/s and blades are curved backward at 30° to tangent at exit. The inner diameter is half of the outer diameter, if the manometric efficiency is 80%. Find (a) the blade angle at inlet, (b) the power required to drive the pump, and (c) the torque. Assume radial entry.

Solution: Data:

Discharge:	$Q = 50$ lit/s
Head:	$H_m = 24$ m
Speed:	$N = 1500$ rpm
Velocity of flow:	$V_{f1} = V_{f2} = 2.4$ m/s
Vane angle at exit:	$\beta_2 = 30°$
Inner diameter of blade:	$d_1 = 0.5 d_2$
Manometric efficiency:	$\eta_{man} = 80\%$
Radial entry:	$\alpha_1 = 90°$, $V_{w1} = 0$, $V_1 = V_{f1}$

To determine: β_1, P, T



$$u_2 = \frac{\pi d_2 N}{60} = \frac{\pi \times d_2 \times 1500}{60} = 78.54 d_2 \text{ m/s}$$

$$\eta_{\text{man}} = 0.8 = \frac{H_m g}{V_{w2}\, u_2} = \frac{9.81 \times 24}{V_{w2} \times 78.54 d_2}$$

$$\tan \beta_2 = \frac{V_{f2}}{u_2 - V_{w2}} = \frac{2.4}{78.54 d_2 - \dfrac{3.75}{d_2}} = \tan 30°$$

$\therefore$ $$d_2 = 0.2464 \text{ m/s}$$

$\therefore$ $$d_1 = 0.1232 \text{ m/s}$$

$$V_{w2} = \frac{3.75}{d_2} = \frac{3.75}{0.2464} = 15.22 \text{ m/s}$$

$\therefore$ $$u_1 = \frac{\pi d_1 N}{60} = \frac{\pi \times 0.1232 \times 1500}{60} = 9.677 \text{ m/s}$$

$$u_2 = 78.54 \times 0.2464 = 19.354 \text{ m/s}$$

(a) **Vane angle at inlet (β_1):**

$$\tan \beta_1 = \frac{V_{f1}}{u_1} = \frac{2.4}{2.677} = 0.248$$

$$\beta_1 = 13.93° \qquad \textbf{Ans.}$$

(b) **Power (P):**

$$P = \frac{\rho Q V_{w2} u_2}{g_c} = 1000 \frac{\text{kg}}{\text{m}^3} \times 0.05 \frac{\text{m}^3}{\text{s}} \times 15.22 \frac{\text{m}}{\text{s}} \times 19.354 \frac{\text{m}}{\text{s}} \times \frac{\text{N-s}^2}{\text{kg-m}}$$

$$= 14730 \text{ N-m/s} \quad (\text{or W}) \qquad \textbf{Ans.}$$

(c) **Torque (T):**

$$\omega = \frac{2\pi N}{60} = \frac{2\pi \times 1500}{60} = 157.1 \text{ rad/s} \quad (\omega = \text{angular velocity})$$

$$P = T\omega$$

$\therefore$ $$T = \frac{P}{\omega} = \frac{14.730}{157.1} = \frac{\text{kN-m}}{\text{s}} \cdot \frac{\text{s}}{\text{rad}} = 0.09378 \text{ kN-m} \qquad \textbf{Ans.}$$

8. A centrifugal pump has impeller diameter of 30 cm and a constant area of flow 210 cm^2. The pump runs at 1440 rpm and delivers 90 lps against a head of 25 m. If there is no whirl velocity at entry, find the pressure rise in terms of pressure head across the impeller and hydraulic efficiency of pump. The vanes at exit are bent back at 22^0 with reference to tangential speed.

<u>**Solution**</u> :-Given: $D_2 = 0.3$m

$A_f = 210\text{cm}^2 = 0.021\text{m}^2$

$\beta_2 = 22^o$, $Q = 0.09 \text{ m}^3/\text{s}$

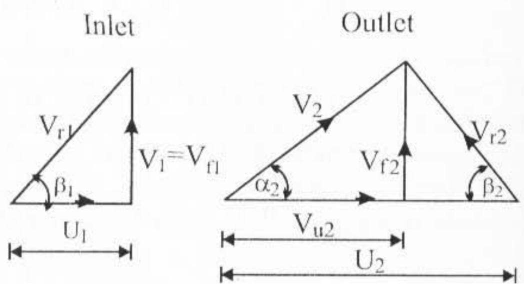
$H_m = 25$ m, $V_{u1} = 0$

Velocity of flow, $V_f = Q/A_f = 0.09/0.021$

$V_f = 4.29 \text{ m/s} = V_1$

$U_2 = \pi D_2 N/60 = \pi \times 0.3 \times 1440/60$

$U_2 = 22.62$ m/s



$V_{u2} = U_2 - V_f \cot\beta_2 = 22.62 - 4.29 \cot(22^0) = 12$ m/s

Euler's head transferred by the impeller,

$$H_e = U_2 V_{u2}/g = 22.62 \times 12/9.81 = \underline{27.7}\ \text{m}$$

And, $V_2 = \sqrt{V_{u2}^2 + V_f^2} = \sqrt{12^2 + 4.29^2} = 12.74$ m/s

Hydraulic efficiency (η_H): $\eta_H = H_m/H_e = 25/27.7 = 0.903 = \eta_{mano}$

$$\eta_{mano} = 90.3\ \%$$

Pressure rise in the Impeller :-

Apply the Bernoulli's equation between inlet & outlet of a pump, we get, (neglecting other losses)

$$\frac{p_1}{\rho g} + \frac{V_1^2}{2g} + \text{Work done on the pump} = \frac{p_2}{\rho g} + \frac{V_2^2}{2g}$$

$$\frac{p_1}{\rho g} + \frac{V_1^2}{2g} + H_e = \frac{p_2}{\rho g} + \frac{V_2^2}{2g}$$

$$\therefore \frac{p_2 - p_1}{\rho g} = H_e - \frac{(V_2^2 - V_1^2)}{2g} = 27.7 - \frac{(12.74^2 - 4.29^2)}{2 \times 9.81}$$

$(p_2 - p_1)/(\rho g) = 20.4$ m of water

or $p_2 - p_1 = 20.4 \times 9810\ \text{N/m}^2$

$(\Delta p_o) = 2.0$ bar.

9. A four stage centrifugal pump has four identical impellers keyed to the same shaft running at 500 rpm. The total manometric head developed is 40m, discharge 0.3m^3/s. If the outlet vane angle is 45^0 for impeller of 5cm outlet diameter, determine the manometric efficiency.

or

A four stage centrifugal pump has four identical impellers keyed to the same shaft running at 500rpm. The total manometric head developed is 40m, discharging 0.3m^3/s. If the outlet vane angle is 45^0 for each impeller of 5cm width and 50cm outlet diameter, determine the manometric efficiency.

EXAMPLE 4.41 A four-stage centrifugal pump has four identical impellers keyed to the same shaft. The speed of the shaft is 500 rpm. The total manometric head developed from four impellers is 50 m. The width at exit is 5 cm and the diameter at exit is 50 cm. The whirl velocity at exit is 10 m/s, and the radial flow velocity at exit is 2 m/s. Calculate (a) the discharge, (b) the exit vane angle, and (c) the manometric efficiency.

Solution: Data:

Number of stages:	$n = 4$ series
Speed:	$N = 500$ rpm
Total manometric head:	$H_{mT} = 50$ m
Width at exit:	$b_2 = 5$ cm
Diameter at exit:	$d_2 = 60$ cm
Whirl velocity at exit:	$V_{w2} = 10$ m/s
Radial velocity at exit:	$V_{f2} = 2$ m/s

To determine: Q, β_2, η_{man}

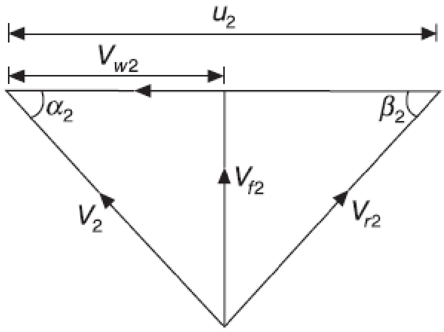
$$u_2 = \frac{\pi d_2 N}{60} = \frac{\pi \times 0.6 \times 500}{60} = 15.71 \text{ m/s}$$

(a) **Discharge (Q):**

$$Q = \pi d_2 b_2 V_{f2} = \pi \times 0.6 \times 0.05 \times 2 = 0.1885 \text{ m}^3/\text{s}$$

(b) **Exit vane angle (β_2):**

From exit velocity triangle,



$$\tan \beta_2 = \frac{V_{f2}}{u_2 - V_{w2}} = \frac{2}{(15.71 - 10)}$$

$$\beta_2 = 19.3° \qquad \textbf{Ans.}$$

(c) **Manometric efficiency (η_{man}):**

$$H_m = \frac{H_{mT}}{4} = \frac{50}{4} = 12.5 \text{ m}$$

$$\eta_{man} = \frac{gH_m}{V_{w2} u_2} = \frac{9.81 \times 12.5 \times 100}{10 \times 15.71} = 78.1\% \qquad \textbf{Ans.}$$

10. An axial flow pump is required to discharge 1.25m^3/s of water while running at 500RPM, the total head is 3.9m. If the speed ratio is 2.3, flow ratio = 0.51, hydraulic efficiency = 0.87 and the overall pump efficiency is 0.82, determine: i) Power delivered to the water and the power input ii) The impeller hub diameter and tip diameter.

Solution :- Given: $Q = 1.25$ m^3/s, $N = 500$ RPM, $H_m = 3.9$m. $\phi = 2.3$, $\psi = 0.51$, $\eta_H = 0.87$, $\eta_0 = 0.82$.

Note :- For axial-flow pump, water enters at inlet always axially for maximum energy transfer. Therefore the velocity at the inlet is zero, i.e, $V_{u1} = 0$.

Speed ratio $= \phi = U_2/(2gH_m)^{1/2}$

$$U_2 = \phi\sqrt{2gH_m} = 2.3 \times \sqrt{2 \times 9.81 \times 3.9} = 20.1 \text{ m/s}$$

And $$V_f = \psi\sqrt{2gH_m} = 0.51 \times \sqrt{2 \times 9.81 \times 3.9} = 4.46 \text{ m/s}$$

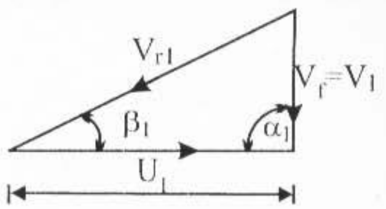
Tip diameter, $D_2 = 60 \times U_2/(\pi N) = 60 \times 20.1/(\pi \times 500) = 0.768$ m

And also, Discharge, $Q = \frac{\pi}{4}(D^2 - d^2)V_f$

$$1.25 = \frac{\pi}{4}(0.768^2 - d^2) \times 4.46$$

Diameter of the hub, $d = 0.484$ m

Inlet velocity triangle at the hub

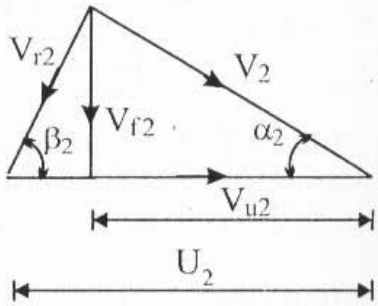


Outlet velocity triangle at the tip

(i) **Power of water at delivery point (WP)**:

$WP = \rho Q g H_m/1000 = 9810 \times 1.25 \times 3.9/1000 = \underline{47.8}$ kW

$$\eta_0 = \frac{\text{Water Power at exit}}{\text{Shaft power input to the impeller,}}$$



Input shaft power, $SP = WP/\eta_O = 47.8/0.82$

$$SP = \underline{58.32}\ \text{kW}$$

(ii) **Tip diameter,** $D = 0.768$ m

& hub diameter, $d = \underline{0.484\text{ m}}$

(iii) **Rotor I/L and O/L angles β_1 & β_2 :-**

As the rotor blade angles vary from the root to the tip, only angles at root and tip will be calculated here.

$\therefore$ Tangential velocity of rotor at hub,

$$U_1 = (d/D)U_2 = (0.484/0.768) \times 20.1$$

$$U_1 = 12.67\text{ m/s}$$

As axial flow velocity is constant, we have

From I/L Vel. Δ^{le} :- $\tan\beta_1 = V_f/U_1 = 4.46/12.67 = 0.352$

$$\therefore \beta_1 = \tan^{-1}(0.352) = \underline{19.4^o}.$$ Ans

From O/L Vel. Δ^{le} :- $H_e = U_2V_{u2}/g = H_m/\eta_H$

$$20.1 \times V_{u2}/9.81 = 3.9/0.87 \Rightarrow V_{u2} = 2.188\text{ m/s}$$

$\therefore \tan\beta_2 = V_f/(U_2 - V_{u2}) = 4.46/(20.1 - 2.188) = 0.249$

$$\therefore \qquad \tan\beta_2 = \underline{13.98^o}.$$ Ans

Also, $\tan\alpha_2 = V_f/V_{u2} = 4.46/2.188 = 2.038$

$\therefore$ Diffuser inlet angle, $\alpha_2 = \underline{63.87^o}$. Ans

11. Elaborate the working principle of Pelton wheel with figure.

It is an impulse turbine working under a high head (above 300 m) and handling low quantity of water. The specific speed is in the range of 8.5 to 51 rpm. This turbine is named after L.A. Pelton of USA.

The water flows from the reservoir to the turbine through the penstock. The end of the penstock is fitted with one or more nozzles. The entire pressure energy (static energy) of water is converted into kinetic energy in the nozzle. The high velocity water jet emerging from the nozzle strikes the bucket (blades) attached to the periphery of the rotor and sets the bucket into rotary motion. Here, water flows in the tangential direction, doing work. (The buckets change the direction of the jet, resulting in change in momentum that sets the wheel into rotary motion.) The kinetic energy of the jet is completely transferred to the rotating wheel, i.e. the velocity of water at the exit of the runner is just sufficient to enable it to move out the runner. The static pressure of water at the entrance and exit of the bucket is same and it is equal to atmospheric pressure. Water is discharged at the tail race after doing work on the runner.

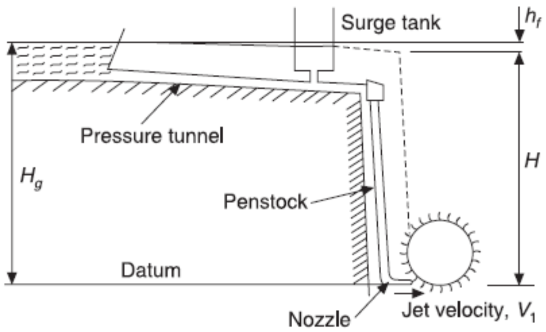


Figure 7.2 High head (Pelton wheel) hydraulic power plant.

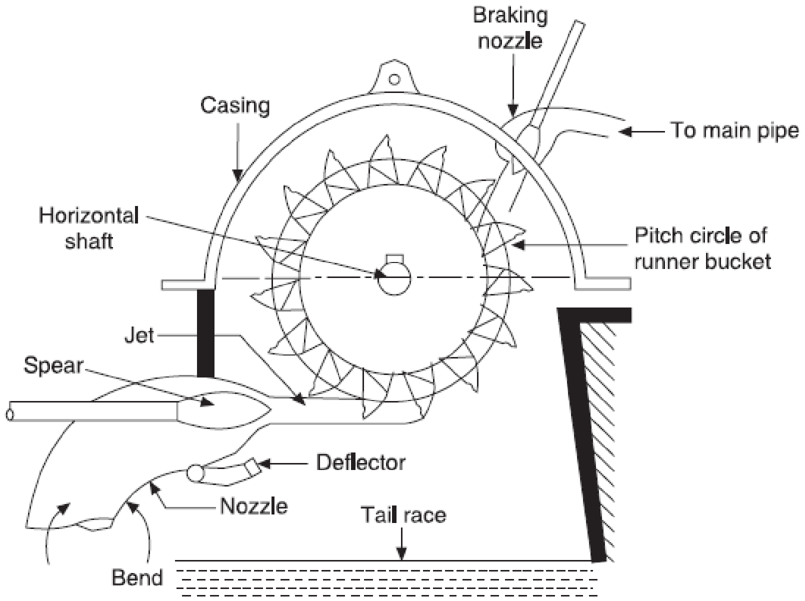


Figure 7.3 Pelton wheel with its main components.

12. Elaborate the working principle of Kaplan turbine with figure.

Figure 7.11(b) shows two views of a Kaplan Turbine. The extension of the shaft is called boss or hub. The runner vanes are fixed around the circumference of the boss. The boss is similar to the propeller of a ship (Figure 7.11(a)). The runner is enclosed in a spiral casing.

Kaplan turbine is a reaction turbine working under low head (2.5 m to 50 m) and handling a large quantity of water. The specific speed is in the range of 255 to 860 rpm. Water enters and leaves the turbine axially, otherwise it is exactly similar to the Francis turbine. This turbine is named after Kaplan, an Australian engineer.

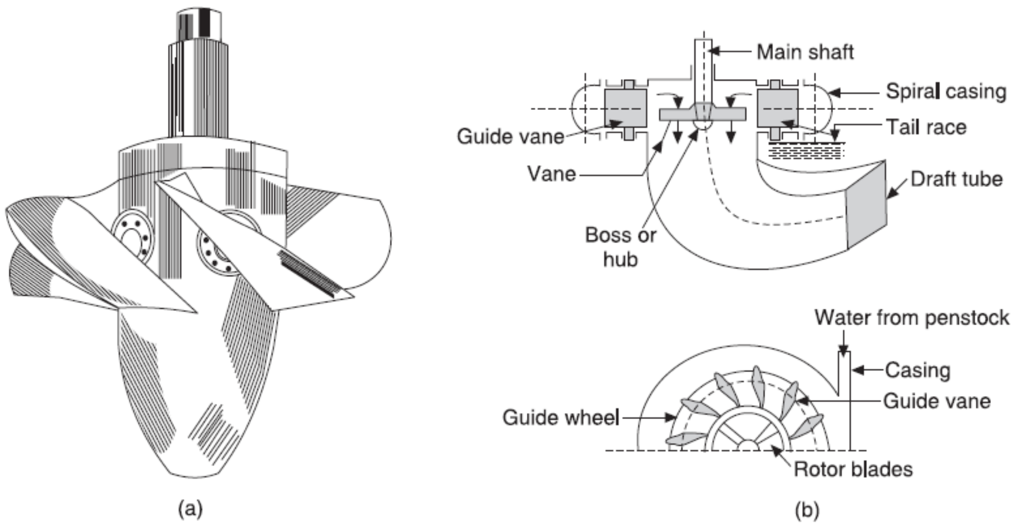


Figure 7.11 Kaplan turbine: (a) Runner. (b) Turbine with components.

Part – B Questions

1. Classify and explain draft tube, and what are its functions?

2. Briefly discuss the classification of hydraulic turbines.

3. Draw the Pelton wheel turbine and explain the efficiencies and head used in Pelton turbine.

4. Draw the inlet and exit velocity triangles for a Pelton wheel turbine. Formulate an expression for the maximum hydraulic efficiency.

5. Velocity triangle and workdone by the pelton wheel expression for maximum utilization factor of hydraulic turbine.

6. Show that the hydraulic efficiency of pelton wheel is maximum when peripheral wheel velocity is half

7. the absolute velocity of jet at inlet. Further deduce that $\eta_{b,max} = \left(\frac{1+K\,Cos\,\beta_2}{2}\right)$ **where K is friction co-efficient and β_2 is outlet blade angle.**

8. Draw the Kaplan turbine and explain the operation with velocity triangles.

9. A Pelton wheel has a water supply rate of 5m^3/s at a head of 256m and runs at 500RPM. Assuming a turbine efficiency of 0.85, a coefficient of velocity for nozzle as 0.985, nozzle speed ratio of 0.46, determine: i) Power output ii) Specific speed iii) Number of jets iv) Diameter of the wheel v) Jet diameter.

10. For an impulse turbine (Pelton wheel) show that the hydraulic efficiency is maximum at U = (V_1/2), where U is the peripheral speed and V_1 is the inlet jet velocity. Consider the effect of friction.

11. Show that the hydraulic efficiency of Pelton wheel is maximum when peripheral wheel velocity is half the absolute velocity of jet at inlet. Further deduce that $\eta_{t\,max} = \left(\frac{1+K\cos\beta_2}{2}\right)$**, where K is friction co-efficient and β_2 is outlet blade angle.**

12. A Kaplan turbine a 5MW generator at 150rpm under a head of 5.5m. The generator and overall efficiencies are respectively 93% and 88%. The tip diameter of the runner is 4.5m and the hub diameter is 2m. Assuming 94% hydraulic efficiency and no exit whirl, determine inlet and outlet.

13. A furnaces turbine has wheel diameter of 1 m at the entrance and 0.5 m at the exit. The guide vane angle is 15°. The water at exit leaves the vane without any tangential component. The vane angle at the entrance is 90°. The head is 30 m and the radial component of the flow is constant, what would be the speed of the wheel in rpm and vane angle at exit.

14. An inward flow reaction turbine with radial discharge with an overall efficiency of 80% is required to develop 147 KW. The total head is 8m. Peripheral velocity of wheel is 0.96 $\sqrt{2gH}$, the radial velocity of flow is 4.51 m/s. The wheel is to make 150 rpm and hydraulic losses in the turbine is 22% of available energy. Determine: (i) Angle of guide blade at inlet (ii) Wheel vane angle at inlet (iii) Dia meter and width of wheel at inlet

15. An inward flow Francis turbine operates at 486rpm and uses 100m3/min of water. The draft tube diameter at inlet and outlet are 0.8m and 1.5m respectively. The length of the draft tube is 30m. The

available head is 81m. Assuming ηv = 0.98, ηm = 0.97 and ηH = 0.92, find the runner tip diameter, power output and speed ratio if the flow ratio ψ = 0.2, the blade at the inlet is inclined 120^0 to the wheel tip velocity.

16. Two inward flow reaction turbines have same runner diameter of 60cm and same hydraulic efficiency. They work under the same head and have the same flow velocity of 6m/s. Runner B has an inlet vane angle of 110^0 and runs at 600rpm. The runner A has an inlet vane angle of 65^0. For both turbines, discharge is radial at outlet. Determine speed of the runner A, and draw the velocity triangle for both runners.