KAPLAN TURBINE

A Kaplan turbine is a type of propeller turbine which was developed by the Austrian engineer V. Kaplan. It is an axial flow turbine, which is suitable for relatively low heads, and hence requires a large quantity of water to develop large amount of power. It is also a reaction type of turbine and hence it operates in an entirely closed conduit from the head race to tail race. The main components of a Kaplan turbine such as scroll casing, stay ring, arrangement of guide vanes and the draft tube.[similar to those of a Francis turbine]. Between the guide vanes and the runner, the water in a Kaplan (or propeller) turbine turns through a right-angle into the axial direction and then passes through the runner.

The runner of a Kaplan (or propeller) turbine has four or six blades and it closely resembles a ship's propeller. The blades or vanes attached to a hub or boss are so shaped that water flows axially through the runner.

NOTE: Ordinarily the runner blades of a propeller turbine are fixed, but the Kaplan turbine runner blades can be turned about their own axis, so that their angle of inclination may be adjusted while the turbine is in motion.

This adjustment of the runner blades is usually carried out automatically by means of a servomotor operating inside the hollow coupling of turbine and generator shaft. When both guide-vane angle and runner-blade angle may thus be varied, a high efficiency can be maintained over a wide range of operating conditions. In other words even at part load, when a lower discharge is flowing through the runner, a high efficiency can be attained in the case of a Kaplan turbine.

It may be explained with the help of Fig. 4.4, in which inlet and outlet velocity triangles for a Kaplan turbine runner working at constant speed under constant head at full load and at part load are shown. It will be observed that although the corresponding changes in the flow through the turbine runner does affect the shape of the velocity triangles, yet as the blade angles are simultaneously adjusted, the water under all the working conditions flows through the runner blades without shock. As such the eddy losses which are inevitable in Francis and propeller turbines are almost completely eliminated in a Kaplan turbine.

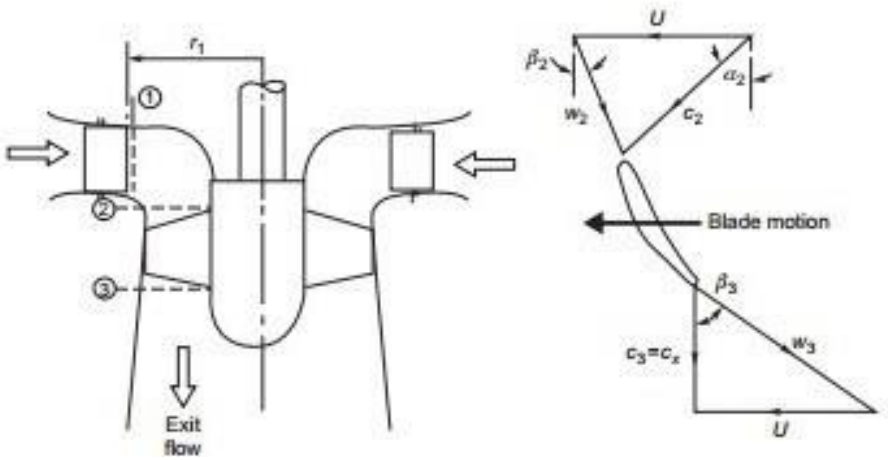


Fig. 4.4 Section Kaplan turbine and velocity triangles

4.1 DRAFT TUBE

A draft tube is a pipe or passage of gradually increasing cross-sectional area which connects the runner exit to the tail race. It may be made of cast or plate steel or concrete. It must be airtight and under all conditions of operation its lower end must be submerged below the level of water in the tail race.

The draft tube has two purposes as follows:

1. It permits a negative or suction head to be established at the runner exit, thus making it possible to install the turbine above the tail race level without loss of head.
2. It converts a large proportion of velocity energy rejected from the runner into useful pressure energy i.e., it act as a recuperator of pressure energy.

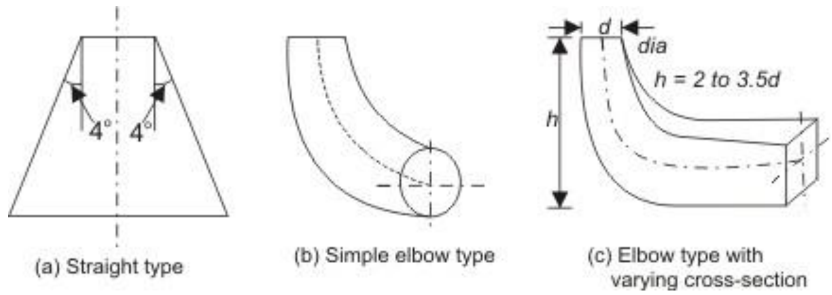


Fig. 4.5 Different types of draft tubes

4.2 GOVERNING OF TURBINES

The governing of a turbine is defined as the operation by which the speed of the turbine is kept constant under all conditions of working.

It is done automatically by means of a governor, which regulates the rate of flow through the turbine.

4.3 CAVITATION

- When the pressure in any part of the flow passage reaches the vapour pressure of the flowing liquid, it starts vapourizing and small bubbles of vapour form in large numbers.

- These bubbles are carried along by the flow, and on reaching the high pressure zones these bubbles suddenly collapse as the vapour condenses to liquid again.
- Due to sudden collapsing of the bubbles or cavities the surrounding liquid rushes into fill them.
- The liquid moving from all directions collides at the centre of the cavity, thus giving rise high local pressure, which may be as high as 686.7 MN/m^2.
- Any solid surface in the vicinity is also subjected to these intense pressures.
- The alternate formation and collapse of vapour bubbles may cause severe damage to the surface which ultimately fails by fatigue and the surface becomes badly scoured and pitted. This phenomenon such as penstocks, gates, valves, spillways etc.

4.4 SPECIFIC SPEED

- The performance or operating conditions for a turbine handling a particular fluid are usually expressed by the values of *P* and *H* and for a pump by *N*, *Q* and *H* . It is important to know the range of these operating parameters covered by a machine of a particular shape (homologous series) at high efficiency. Such information enables us to select the type of machine best suited to a particular application, and thus serves as a starting point in its design. Therefore a parameter independent of the size of the machine *D* is required which will be the characteristic of all the machines of a homologous series.
- The Specific Speed N_s of a turbine is the speed in rotations per minute (r.p.m.) at which a similar model of the turbine would run under a head of 1ft. when of such size as to develop 1 H.P. Each type of Turbine (Pelton Wheel, Francis etc.) has it's own characteristic limits of N_s .

$$n_s = n\frac{\sqrt{P}}{H^{\frac{5}{4}}}$$

Where: n_s = specific speed

n= revolution per minute

P=Power (K.W)

H= head (m)

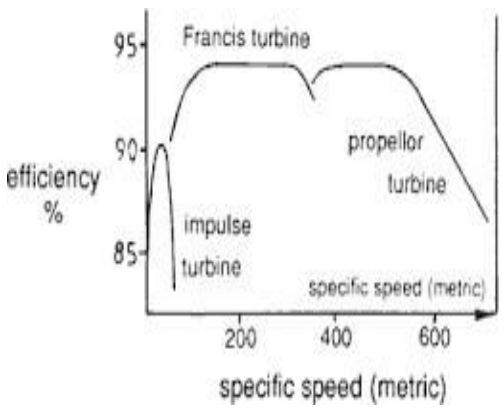


Fig. 4.6 Comparison of N_s for different turbines

4.5 UNIT QUANTITIES

The unit operating conditions for a turbine are those under which that particular turbine would run when working under a head of 1 ft. (or unit head in any other system) assuming there no change in efficiency. This allows the performance of a given turbine to be compared when working under different heads and enables the characteristic curves to be drawn, showing the efficiency at all running conditions.

4.9.1 Unit Speed

If N_u is the Unit speed and N the speed under a head H

$$v = \frac{\pi D N}{60}$$

And $v \propto V \propto H$. Therefore $N \propto \sqrt{H}$. Or $\frac{N}{\sqrt{H}} = C = \frac{N_u}{\sqrt{H_u}}$

(Where u represents unit conditions and C represents a Constant)

Unit speed:

$$N_u = \frac{N}{\sqrt{H}}$$

4.9.2 Unit Quantity

The Unit quantity of a Turbine is the flow through the turbine when operating under a head of 1 ft. assuming similar conditions.

Let,

- Q be the flow under a head H.
- Therefore Q is the area of flow X velocity.

And since the area is constant, and velocity is $\propto \sqrt{H}$ $Q \propto \sqrt{H}$. Or $\frac{Q}{\sqrt{H}} = C$ (where C is a Constant)

$$\therefore \quad \frac{Q}{\sqrt{H}} = \frac{Q_u}{\sqrt{H_u}}$$

4.9.3 Unit Power

The Unit Power of a given turbine is the power output of the turbine when operating under a head of 1 ft. assuming no change in efficiency.

If P is the output under a head H

Then:

$$P = \frac{W\ H}{550} \times \eta$$

If η is unchanged $W = wQ$

$$\therefore \quad W \propto \sqrt{H}$$

And:

$$P \propto \sqrt{H} \times H \quad \propto H^{\frac{3}{2}} \qquad \therefore \quad \frac{P}{H^{\frac{3}{2}}} = C = \frac{P_u}{H_u^{\frac{3}{2}}}$$

(where C is a Constant)

But $H_u = 1$

Unit Power

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

4.6 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.

The important parameters which are varied during a test on a turbine are:

1. Speed(N) 2. Head(H) 3. Discharge(Q) 4. Power(P) 5. Overall efficiency(Θ_{h0})

6. Gate opening

Out of the above six parameters, three parameters namely speed(N), Head(H) and discharge(Q) are independent parameters.

The following are the important characteristic curves of a turbine

1. Main Characteristic Curves or Constant Head Curve
2. Operating Characteristic Curves or Constant Speed Curve
3. Muschel Curves or Constant Efficiency Curve

4.10.1 Constant head curves

Maintaining a constant head, the speed of the turbine is varied by admitting different rates of flow by adjusting the percentage of gate opening. The power *P* developed is measured mechanically. From each test the unit power P_u, the unit speed N_u, the unit discharge Q_u and the overall efficiency Θ_o are determined. The characteristic curves drawn are

a) Unit discharge vs unit speed

b) Unit power vs unit speed

c) Overall efficiency vs unit speed

4.10.2 Constant speed curves

In this case tests are conducted at a constant speed varying the head *H* and suitably adjusting the discharge *Q*. The power developed *P* is measured mechanically. The overall efficiency is aimed at its maximum value.

The curves drawn are

P	vs	Q
Θ_o	vs	Q
Θ_o	vs	P_u
Θo max	vs	% Full load

4.10.3 Constant efficiency curves

These curves are plotted from data which can be obtained from the constant head and constant speed curves. The object of obtaining this curve is to determine the zone of constant efficiency so that we can always run the turbine with maximum efficiency. This curve also gives a good idea about the performance of the turbine at various efficiencies.

Numericals

Problem.

The following data is for a Francis Wheel.

Radial velocity is constant	No whirl at exit.		
Flow rate	0.189 m^3/s		
$D_1 = 0.6$ m	$D_2=0.4$ m	k =0.85	$h_1=50$ mm
$\alpha_1=110^o$	N=562 rev/min		

Head difference from inlet to outlet is 32 m. Entry is shockless. Calculate

i. the guide vane angle
ii. the diagram power
iii. the hydraulic efficiency
iv. the outlet vane angle
v. the blade height at outlet.

Solution.

$u_1 = \pi N D_1 = 17.655$ m/s $v_{r1} = Q/(\pi D_1 h_1 k) = 0.189/(\pi \times 0.6 \times 0.05 \times 0.85) = 2.35$ m/s

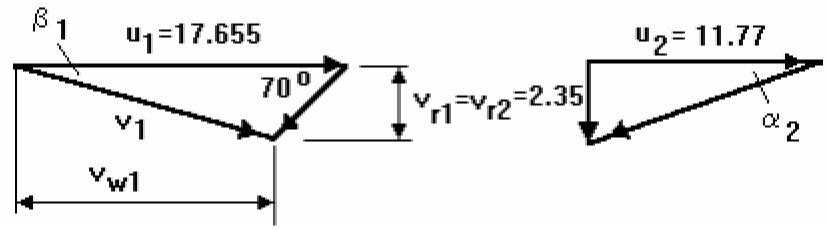


Fig. 19

v_{w1} and β_1 may be found by scaling or by trigonometry.

$v_{w1} = 16.47$m/s $\beta_1 = 8.12^o$ $u_2 = \pi N D_2 = 11.77$ m/s

$\alpha_2 = \tan^{-1}(2.35/11.77) = 11.29^o$

D.P. = m $u_1 v_{w1}$=189 (17.655 x 16.47) = 54 957 Watts

W.P.= mgΔH = 189 x 9.81 x 32 = 59 331 Watts

η_{hyd} = 54 957/59 331 = 92.6%

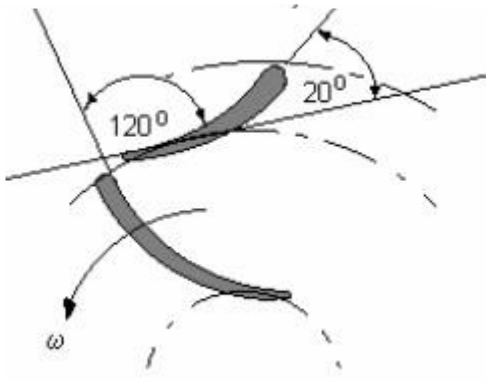
since $v_{r1} = v_{r2}$ then $D_1 h_1 = D_2 h_2$

h_2 = 0.6 x 0.05/0.4 = 0.075 m

Problem.

The runner (rotor) of a Francis turbine has a blade configuration as shown. The outer diameter is 0.4 m and the inner diameter is 0.25 m. The vanes are 65 mm high at inlet and 100 mm at outlet. The supply head is 20 m and the losses in the guide vanes and runner are equivalent to 0.4 m. The water exhausts from the middle at atmospheric pressure. Entry is shockless and there is no whirl at exit. Neglecting the blade thickness, determine :

i. the speed of rotation.
ii. the flow rate.
iii. the output power given a mecanical efficiency of 88%.
iv. the overall efficiency.
v. The outlet vane angle.



Solution.

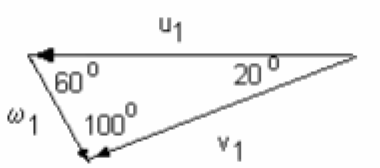


Fig.21

$$v_{w1} = \frac{u_1 \sin(\alpha_1)\sin(\beta_1)}{\sin(\alpha_1 - \beta_1)\tan\beta_1}$$

$$v_{w1} = \frac{u_1 \sin(120)\sin(20)}{\sin(100)\tan(20)} = 0.826 u_1$$

The inlet vector diagram is as shown. Values can be found by drawing to scale.
Since all angles are known but no flow rate, find v_{w1} in terms of u_1

$\Delta H - h_L = u1 v_{w1}/g$

$20 - 0.4 = 19.6 = u1 v_{w1}/g$

$19.6 = 0.826\ u_1^2/g$

$u_1 = 15.26$ m/s

$u_1 = \pi N D_1/60$

$N = 15.26 \times 60/(\pi \times 0.4) = 728.5$ rev/min

$$v_{r1} = \frac{u_1 \sin(\alpha_1)\sin(\beta_1)}{\sin(\alpha_1 - \beta_1)} = \frac{15.26 \sin(120)\sin(20)}{\sin(100_1)} = 4.589\text{m/s}$$

$Q = v_{r1} \times \pi D_1 h_1 = 12.6 \times \pi \times 0.4 \times 0.065 = 0.375\ m^3/s$

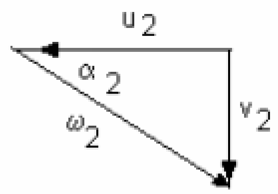
m = 375 kg/s

$v_{w1} = 0.826\ u_1 = 12.6$ m/s

Diagram Power = m u_1 v_{w1} = 375 x 15.26 x 12.6 = 72.1 kW

Output power = 0.88 x 72.1 = 63.45 kW

OUTLET TRIANGLE



$u_2 = \pi N D_2/60 = \pi \times 728.5 \times 0.25/60 = 9.54$m/s

$Q = v_{r2} \times \pi D_2 h_2$

$0.375 = v_{r2} \times \pi \times 0.25 \times 0.1$

$v_{r2} = 4.775$ m/s = v_2 if no whirl.

$\tan \alpha_2 = 4.775/9.54 = 0.5$

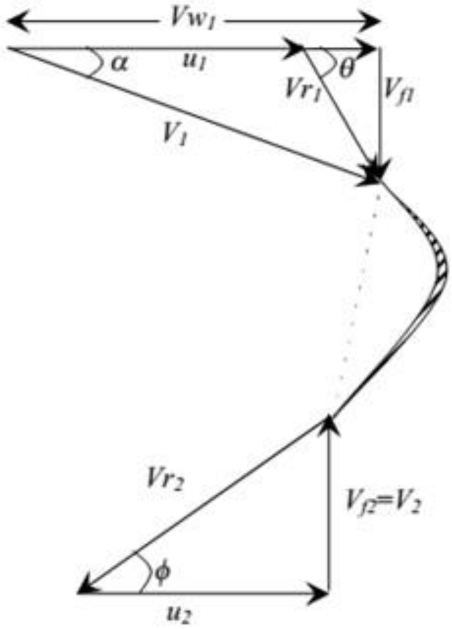
$\alpha_2 = 26.6^o$.

Problem.

The following data is given for a Francis turbine. Net Head = 60 m; speed N = 700 rpm; Shaft power = 294.3 kW; η_o = 84%; η_h = 93%; flow ratio = 0.2; breadth ratio n = 0.1; Outer diameter of the runner = 2 x inner diameter of the runner. The thickness of the vanes occupies 5% circumferential area of the runner, velocity of flow is constant at inlet and outlet and discharge is radial at outlet. Determine: (i) Guide blade angle (ii) Runner vane angles at inlet and outlet (iii) Diameters of runner at inlet and outlet (iv) Width of wheel at inlet

Solution.

$H = 60$ m; $N = 700$ rpm; $P = 294.3$ kW; $\eta_o = 84\%$; $\eta_h = 93\%$;



flow ratio = $\dfrac{V_{f1}}{\sqrt{2gH}} = 0.2$

$V_{f1} = 0.2\sqrt{2\times10\times60} = 6.928\,\text{m/s}$

Breadth ratio $\dfrac{B_1}{D_1} = 0.1$

$D_1 = 2 \text{ x } D_2$

$V_{f1} = V_{f2} = 6.928$ m/s

Thickness of vanes =

5% of circumferential area of runner

$\therefore$ Actual area of flow = $0.95\ \pi\ D_1 B_1$

Discharge at outlet = Radial and hence

$V_{w2} = 0$ and $V_{f2} = V_2$

We know that the overall efficiency is given by

$$\eta_0 = \frac{P}{\rho g Q H}; 0.84 = \frac{294.3\times10^3}{1000\times10\times Q\times60}$$

$Q = 0.584\ \text{m}^3/\text{s}$

$Q = 0.95\ \pi\ D_1 B_1 V_{f1} = 0.95\ \pi\ D_1 \text{ x } (0.1\ D_1) \text{ x } 6.928 = 0.584$

Hence $D_1 = 0.531$ m (Ans)

$\dfrac{B_1}{D_1} = 0.1$ and $B_1 = 53.1$ mm (Ans)

$$u_1 = \frac{\pi D_1 N}{60} = \frac{\pi\times0.531\times700}{60} = 19.46\,\text{m/s}$$

Hydraulic efficiency $\eta_h = \dfrac{V_{w1}u_1}{gH}; 0.93 = \dfrac{V_{w1} \times 19.46}{10 \times 60}$

V_{w1} = 28.67 m/s

From Inlet velocity triangle $\tan\alpha = \dfrac{V_{f1}}{V_{w1}} = \dfrac{6.928}{28.67} = 0.242$

Hence Guide blade angle = α = 13.58° (Ans)
We know that the overall efficiency is
given by

$$\eta_0 = \frac{P}{\rho g Q H}; 0.84 = \frac{294.3 \times 10^3}{1000 \times 10 \times Q \times 60}$$

Q = 0.584 m³/s

$Q = 0.95\ \pi\ D_1 B_1 V_{f1} = 0.95\ \pi\ D_1 \times (0.1\ D_1) \times 6.928 = 0.584$

Hence D_1 = 0.531 m (Ans)

$\dfrac{B_1}{D_1} = 0.1$ and B_1 = 53.1 mm (Ans)

$$u_1 = \frac{\pi D_1 N}{60} = \frac{\pi \times 0.531 \times 700}{60} = 19.46 \text{m/s}$$

Hydraulic efficiency $\eta_h = \dfrac{V_{w1}u_1}{gH}; 0.93 = \dfrac{V_{w1} \times 19.46}{10 \times 60}$

V_{w1} = 28.67 m/s

From Inlet velocity triangle $\tan\alpha = \dfrac{V_{f1}}{V_{w1}} = \dfrac{6.928}{28.67} = 0.242$

Hence Guide blade angle = α = 13.58° (Ans)

Problem.

A Kaplan turbine develops 9000 kW under a net head of 7.5 m. Overall efficiency of the wheel is 86% The speed ratio based on outer diameter is 2.2 and the flow ratio is 0.66. Diameter of the boss is 0.35 times the external diameter of the wheel. Determine the diameter of the runner and the specific speed of the runner.

Solution.

P = 9000 kW; H = 7.5 m; η_o = 0.86; Speed ratio = 2.2; flow ratio = 0.66;

$D_b = 0.35\ D_o$;

$$\frac{u_1}{\sqrt{2gH}} = 2.2$$

$$u_1 = 2.2\sqrt{2 \times 10 \times 7.5} = 26.94 \text{m/s}$$

$$\frac{V_{f1}}{\sqrt{2gH}} = 0.66$$

$$V_{f_1} = 0.66\sqrt{2\times10\times7.5} = 8.08\,\text{m/s}$$

$$\eta_0 = \frac{P}{\rho g Q H}; 0.86 = \frac{9000\times10^3}{1000\times10\times Q\times7.5}$$

$Q = 139.5\ \text{m}^3/\text{s}$

$$Q = \frac{\pi}{4}\left(D_o^2 - D_b^2\right)\times V_{f1} \Rightarrow \frac{\pi}{4}\left(D_o^2 - [0.35D_o]^2\right)\times8.08 = 139.5$$

$D_o = 5.005\ \text{m (Ans)}$

$$u = \frac{\pi D_o N}{60} = \frac{\pi\times5.005\times N}{60} = 26.94\,\text{m/s}$$

N=102.8 rpm (Ans)

$$N_s = \frac{N\sqrt{P}}{H^{5/4}} = \frac{102.8\sqrt{9000}}{7.5^{5/4}} = 785.76\,\text{rpm (Ans)}$$

Problem.

A Kaplan turbine working under a head of 25 m develops 16,000 kW shaft power. The outer diameter of the runner is 4 m and hub diameter is 2 m. The guide blade angle is 35°. The hydraulic and overall efficiency are 90% and 85% respectively. If the velocity of whirl is zero at outlet, determine runner vane angles at inlet and outlet and speed of turbine.

Solution.

$H = 25$ m; $P = 16{,}000$ kW; $D_b = 2$ m; $D_o = 4$ m; $\alpha = 35°$; $\eta_h = 0.9$;

$\eta_o = 0.85$; $V_{w2} = 0$; $\theta = ?$; $\phi = ?$; $N = ?$

$$\eta_0 = \frac{P}{\rho g Q H};\ 0.85 = \frac{16000\times10^3}{1000\times10\times Q\times25}$$

$Q = 75.29\ \text{m}^3/\text{s}$

$$Q = \frac{\pi}{4}\left(D_o^2 - D_b^2\right)\times V_{f1} \Rightarrow \frac{\pi}{4}\left(4^2 - 2^2\right)\times V_{f1} = 75.29$$

$V_{f1} = 7.99$ m/s

From inlet velocity triangle,

$$\tan\alpha = \frac{V_{f1}}{V_{w1}}$$

$$V_{w1} = \frac{7.99}{\tan 35} = 11.41\,\text{m/s}$$

From Hydraulic efficiency

$$\eta_h = \frac{V_{w1}u_1}{gH}$$

$$0.9 = \frac{11.41 \times u_1}{10 \times 25}$$

u_1 = 19.72 m/s

$$\tan\theta = \frac{V_{f1}}{u_1 - V_{w1}} = \frac{7.99}{19.72 - 11.41} = 0.9614$$

θ = 43.88° (Ans)

For Kaplan turbine, $u_1 = u_2$ = 19.72 m/s and $V_{f1} = V_{f2}$ = 7.99 m/s

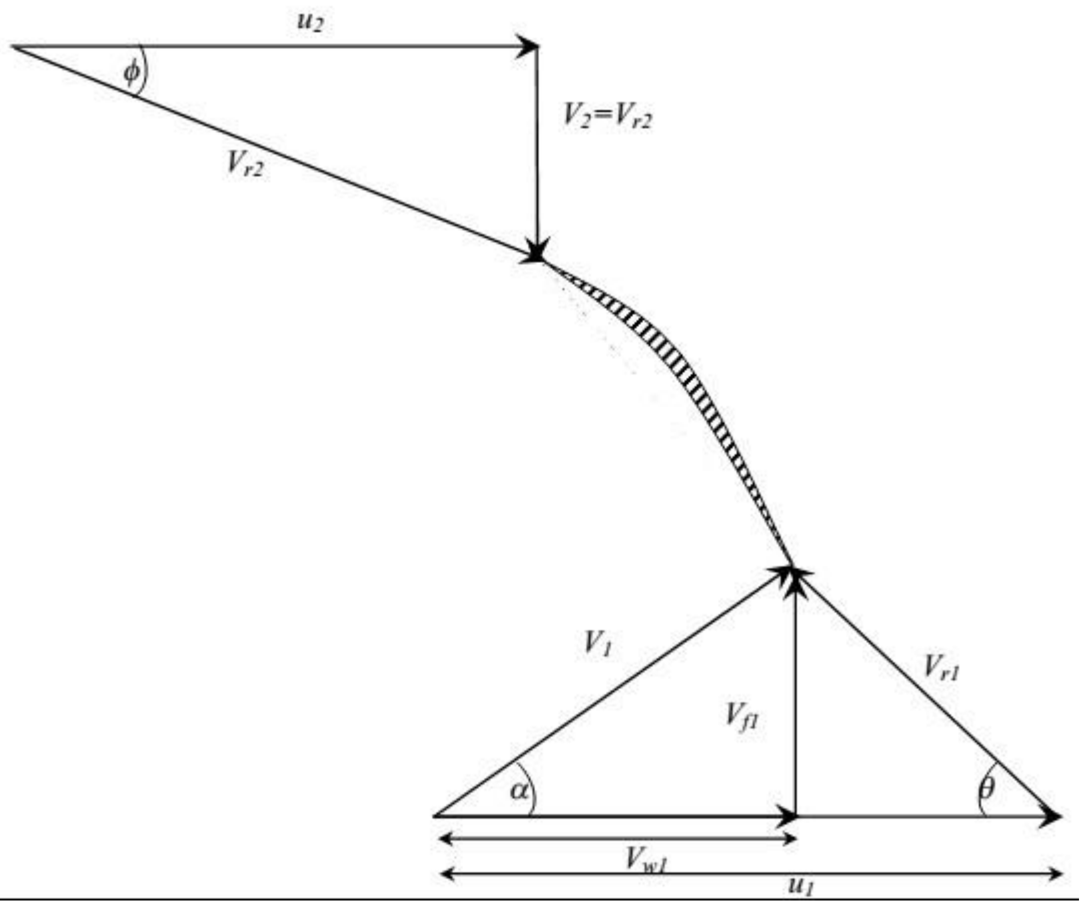
From outlet velocity triangle

$$\tan\phi = \frac{V_{f2}}{u_2} = \frac{7.99}{19.72} = 0.4052$$

ϕ = 22.06° (Ans)

$$u_1 = u_2 = \frac{\pi D_o N}{60} = \frac{\pi \times 4 \times N}{60} = 19.72\,\text{m/s}$$

N = 94.16 rpm (Ans)



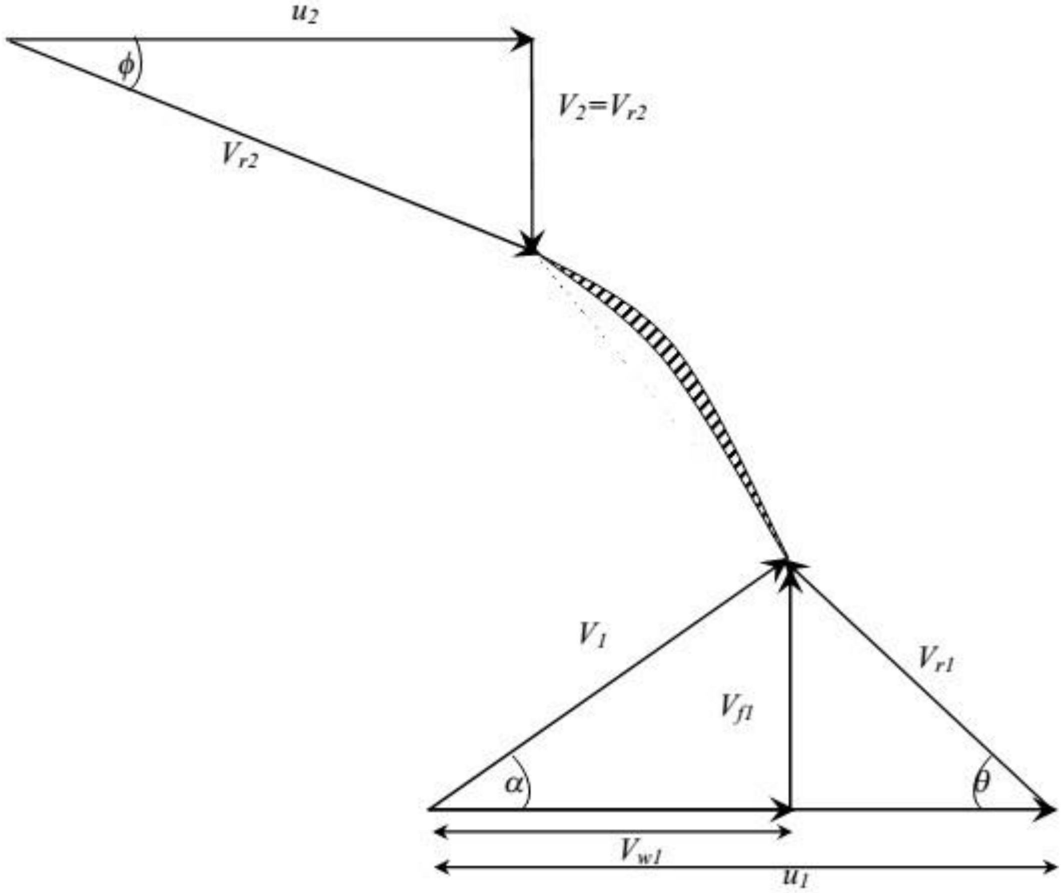
Problem.

. A Kaplan turbine works under a head of 22 m and runs at 150 rpm. The diameters of the runner and the boss are 4.5 m and 12 m respectively. The flow ratio is 0.43. The inlet vane angle at the extreme edge of the runner is 163°19'. If the turbine discharges radially at outlet, determine the discharge, the hydraulic efficiency, the guide blade angle at the extreme edge of the runner and the outlet vane angle at the extreme edge of the manner.

Solution.

H = 22 m; N = 150 rpm; D_o = 4.5 m; D_b = 2 m; θ = 163°19′; $V_{\omega 2} = 0$;

$V_2 = V_{f2} = V_{f1}$; Q = ?; η_h = ?; α = ?; ϕ = ?; $\dfrac{V_{f1}}{\sqrt{2gH}} = 0.43$,



$$u_1 = u_2 = \frac{\pi D_o N}{60} = \frac{\pi \times 4.5 \times 150}{60} = 35.34 \text{ m/s}$$

$$V_{f1} = 0.43\sqrt{2\times 10\times 22} = 9.02 \text{ m/s}$$

$$\tan(180-\theta) = \frac{V_{f1}}{(u_1 - V_{w1})}$$

$$\tan(180-163°19') = \frac{9.02}{(35.34 - V_{w1})} = 0.2997$$

V_{w1} = 5.24 m/s

Hydraulic efficiency is given by

$$\eta_h = \frac{V_{w1}u_1}{gH} = \frac{5.24\times35.34}{10\times22} = 84.17\%$$

$$\tan\alpha = \frac{V_{f1}}{V_{w1}} = \frac{9.02}{5.24} = 1.72$$

α = 59.85° (Ans)

$$\tan\phi = \frac{V_{f2}}{u_2} = \frac{9.02}{35.34} = 0.2552$$

ϕ = 14.32° (Ans)

Problem.

)A kaplan turbine is to be designed to develop 7,350 kW. The net available head is 5.5 m. Assume that the speed ratio as 0.68 and the overall efficiency as 60%. The diameter of the boss is ⅓ rdof the diameter of the runner. Find the diameter of the runner, its speed and its specific speed.

Solution.

P = 7350 kW, H = 5.5 m

$$\frac{V_{f1}}{\sqrt{2gH}} = 0.68 \quad \text{and hence} \quad V_{f1} = 0.68\sqrt{2\times10\times5.5} = 7.13\,\text{m/s}$$

$$\frac{u_1}{\sqrt{2gH}} = 2.09 \quad \text{and hence} \quad u_1 = 2.2\sqrt{2\times10\times5.5} = 23.07\,\text{m/s}$$

$$\eta_0 = \frac{P}{\rho g Q H};\ 0.6 = \frac{7350\times10^3}{1000\times10\times Q\times5.5}$$

Q = 222.72 m^3/s

$$Q = \frac{\pi}{4}\left(D_o^2 - D_b^2\right)\times V_{f1} \Rightarrow \frac{\pi}{4}\left(D_o^2 - \left[\frac{D_o}{3}\right]^2\right)\times7.13 = 222.72$$

D_o = 6.69 m (Ans)

$$u = \frac{\pi D_o N}{60} = \frac{\pi\times6.69\times N}{60} = 23.07\,\text{m/s}$$

N=65.86 rpm (Ans)

$$N_s = \frac{N\sqrt{P}}{H^{5/4}} = \frac{65.86\sqrt{7350}}{5.5^{5/4}} = 670.37\,\text{rpm (Ans)}$$

Problem.

The head at the base of the nozzle of a Pelton wheel is 640 m. The outlet vane angle of the bucket is 15 o. The relative velocity at the outlet is reduced by 15% due to friction along the vanes. If the discharge at outlet is without whirl find the ratio of bucket speed to the jet speed. If the jet diameter is 100 mm while the wheel diameter is 1.2 m, find the speed of the turbine in rpm, the force exerted by the jet on the wheel, the Power developed and the hydraulic efficiency. Take C_v=0.97.

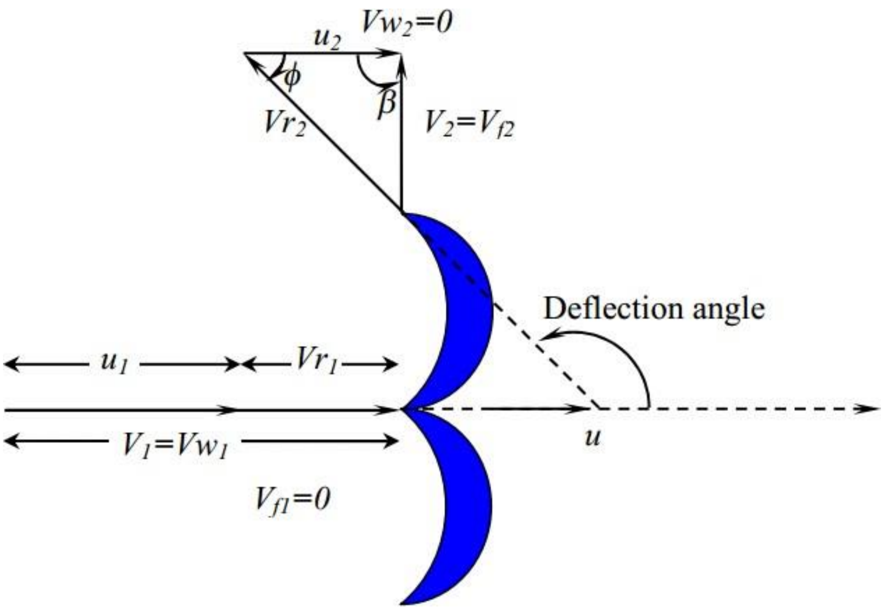
Solution.

H = 640 m; ϕ=15°; V_{r1} = 0.85 V_{r2}; V_{w2} = 0; d = 100 mm; D = 1.2 m;

C_v = 0.97; K_u = ?; N = ?; F_x = ?; P = ?; η_h = ?

We know that the absolute velocity of jet is given by

$$V = C_v\sqrt{2gH} = 0.97\sqrt{2\times10\times640} = 109.74 \text{ m/s}$$



Let the bucket speed be u

Relative velocity at inlet = V_{r1} = V_1-u = 109.74-u

Relative velocity at outlet = V_{r2} = (1-0.15)V_{r1} = 0.85(109.74-u)

But $V_{r2}\cos\phi$ = u ⇨ 0.85(109.74-u)cos15

Hence u = 49.48 m/s

But $u = \dfrac{\pi DN}{60}$ and hence

$$N = \frac{60u}{\pi D} = \frac{60\times49.48}{\pi\times1.2} = 787.5 \text{ rpm (Ans)}$$

Jet ratio = m = $\dfrac{u}{V} = \dfrac{49.48}{109.74} = 0.45$

Weight of water supplied = γ Q = $10\times1000\times\dfrac{\pi}{4}\times0.1^2\times109.74^2 = 8.62$ kN/s

Force exerted = $F_x = \rho a V_1 (V_{w1} - V_{w2})$

But V_{w1} = V_1 and V_{w2} = 0 and hence

$$F_x = 1000 \times \frac{\pi}{4} \times 0.1^2 (109.74)^2 = 94.58 \text{ kN}$$

Work done/second = F_x x u = 94.58 x 49.48 = 4679.82 kN/s

Kinetic Energy/second = $\frac{1}{2}\rho a V_1^3 = \frac{1}{2} \times 1000 \times \frac{\pi}{4} \times 0.1^2 \times 109.74^3 = 5189.85 \text{ kN/s}$

Hydraulic Efficiency = $\eta_h = \frac{\text{Work done/s}}{\text{Kinetic Energy/s}} = \frac{4679.82}{5189.85} \times 100 = 90.17\%$

Problem.

A peltonwheel turbine is having a mean runner diameter of 1.0 m and is running at 1000 rpm. The net head is 100.0 m. If the side clearance is 20° and discharge is 0.1 m3/s, find the power available at the nozzle and hydraulic efficiency of the turbine.

Solution.

D = 1.0 m; N = 1000 rpm; H = 100.0 m; ϕ = 20°; Q = 0.1 m^3/s; WD/s = ? and η_h = ?

Assume C_v = *0.98*

We know that the velocity of the jet is given by

$$V = C_v \sqrt{2 g H} = 0.98 \sqrt{2 \times 10 \times 1000} = 43.83 \text{ m/s}$$

The absolute velocity of the vane is given by

$$u = \frac{\pi D N}{60} = \frac{\pi \times 1 \times 1000}{60} = 52.36 \text{ m/s}$$

This situation is impracticable and hence the data has to be modified. Clearly state the assumption as follows:

Assume H = 700 m (Because it is assumed that the typing and seeing error as 100 for 700)

Absolute velocity of the jet is given by

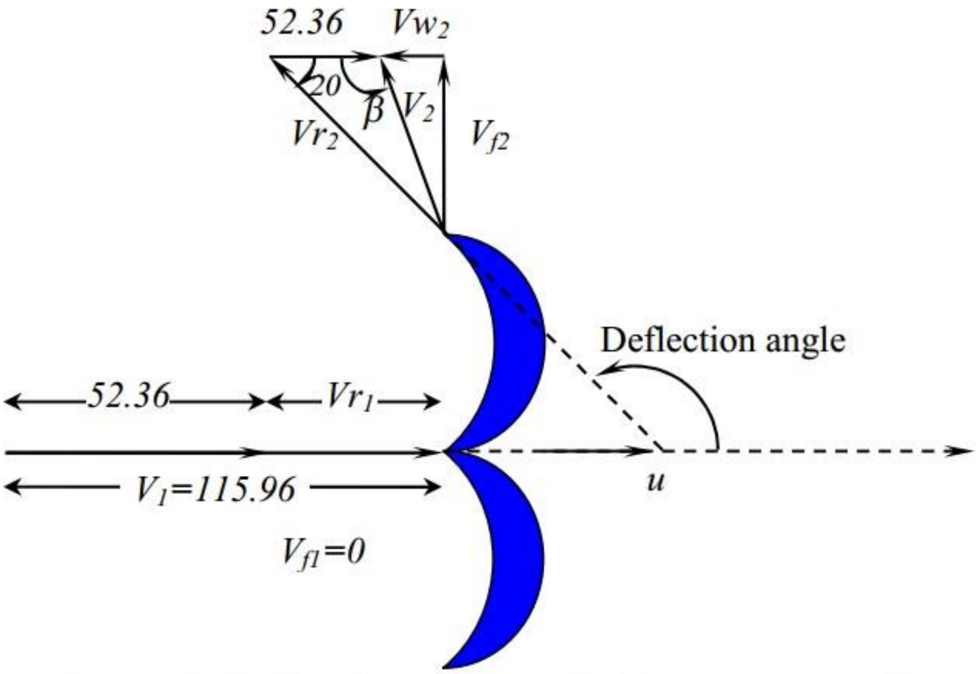
$$V = C_v \sqrt{2 g H} = 0.98 \sqrt{2 \times 10 \times 700} = 115.96 \text{ m/s}$$

Power available at the nozzle is the given by work done per second

WD/second = γ Q H = ρ g Q H = 1000x10x0.1x700 = 700 kW

Hydraulic Efficiency is given by

$$\eta_h = \frac{2u}{V_1^2}(V_1 - u)[1 + \cos\phi] = \frac{2 \times 52.36}{115.96^2}(115.96 - 52.36)(1 + \cos 20) = 96.07\%$$



Problem.

A Pelton wheel has a mean bucket speed of 10 m/s with a jet of water flowing at the rate of 700 lps under a head of 30 m. The buckets deflect the jet through an angle of 160°. Calculate the power given by water to the runner and the hydraulic efficiency of the turbine. Assume the coefficient of nozzle as 0.98.

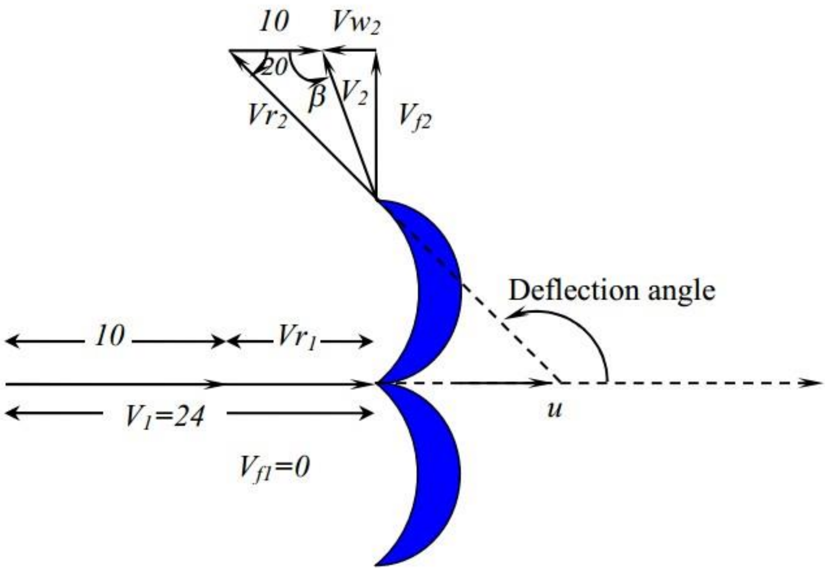
Solution.

u = 10 m/s; Q = 0.7 m^3/s; ϕ = 180-160 = 20°; H = 30 m; C_v = 0.98;

WD/s = ? and η_h = ?

Assume g = 10m/s^2

$$V = C_v\sqrt{2gH} = 0.98\sqrt{2\times10\times30} = 24\,\text{m/s}$$



$V_{r1} = V_1 - u = 24 - 10 = 14$ m/s

Assuming no shock and frictional losses we have $V_{r1} = V_{r2} = 14$ m/s

$V_{w2} = V_{r2} \; Cos\ \phi - u = 14 \times Cos\ 20 - 10 = 3.16$ m/s

We know that the Work done by the jet on the vane is given by

$$\text{WD/s} = \rho a V_1 \left[V_{w1} + V_{w2}\right] u = \rho Q u \left[V_{w1} + V_{w2}\right] \text{ as } Q = aV_1$$

$$= 1000 \times 0.7 \times 10 \left[24 + 3.16\right] = 190.12 \text{ kN-m/s (Ans)}$$

$$\text{IP/s} = \text{KE/s} = \frac{1}{2} \rho a V_1^3 = \frac{1}{2} \rho Q V_1^2 = \frac{1}{2} \times 1000 \times 0.7 \times 24^2 = 201.6 \text{ kN-m/s}$$

Hydraulic Efficiency = Output/Input = 190.12/201.6 = 94.305%

It can also be directly calculated by the derived equation as

$$\eta_h = \frac{2u}{V_1^2}(V_1 - u)\left[1 + \cos\phi\right] = \frac{2 \times 10}{24^2}(24 - 10)\left[1 + \cos 20\right] = 94.29\% \text{ (Ans)}$$

Exercise problems

1. The external and internal diameters of an inward flow reaction turbines are 1.2 m and 0.6 m respectively. the head on the turbine is 22 m and velocity of flow through the runner is constant and equals to 2.5 m/s. The guide blade angle is 10º and the runner vanes are radial at inlet. If the discharge at the outlet is radial, determine:
i) speed of the turbine, ii) vane angle at the outlet of the runner and iii)hydraulic efficiency.

2. A pelton wheel is to be designed for a head of 60 m when running under a head at 200 rpm. The pelton wheel develops 95.6475 kW shaft power. The velocity of the buckets is equal to
0.45 times the velocity of the jet, overall efficiency = 0.85 and coefficient of the velocity is = 0.98.

3. A reaction turbine works at 450 rpm under a head of 120 metres. Its diameter at inlet is 120 cm and the flow area is 0.4 m^2. The angles made by the absolute and relative velocities at inlet are 20 and 60 respectively with the tangential velocity. Determine:
i)the volume flow rate, ii) power developed and iii) hydraulic efficiency, by assuming whirl at outlet to be zero.

4. A Kaplan turbine runner is to be designed to develop 7357.5 kW shaft power. The net available head is 5.5 m. Assume that the speed ratio is 2.09 and flow ratio is 0.68 and the overall efficiency is 60%. The diameter of the boss is one third of the diameter of the runner. Find the diameter of the runner, its speed and its specific speed.

5. CENTRIFUGAL PUMPS

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 convened into pressure energy by means of centrifugal force acting on the fluid, the hydraulic machine is called centrifugal pump.

But if the mechanical energy is converted into hydraulic energy (or pressure energy) by sucking 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.***

Table 5.1 Comparison between Centrifugal pumps and Reciprocating pumps

Centrifugal pumps	Reciprocating pumps
1. The discharge is continuous and smooth.	1. The discharge is fluctuating and 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 and cavitation.
7. The operation of centrifugal pump is smooth and without much noise. The maintenance cost is low.	7. The operation of reciprocating pump is complicated and with much noise. The maintenance cost is high.
8. Centrifugal pump needs smaller floor area and installation cost is low.	8. Reciprocating pump requires large floor area and installation cost is high.
9. Efficiency is high.	9. Efficiency is low.

The centrifugal pump acts as a reverse 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 vortex 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. The rise in pressure head at any point of the rotating liquid is proportional to the square of tangential velocity of the liquid at that point. 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 he lifted to a high level.

5.1.1 MAIN PARTS OF A CENTRIFUGAL PUMP

The followings 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. All the main parts of the centrifugal pump are shown in Fig. 5.1.

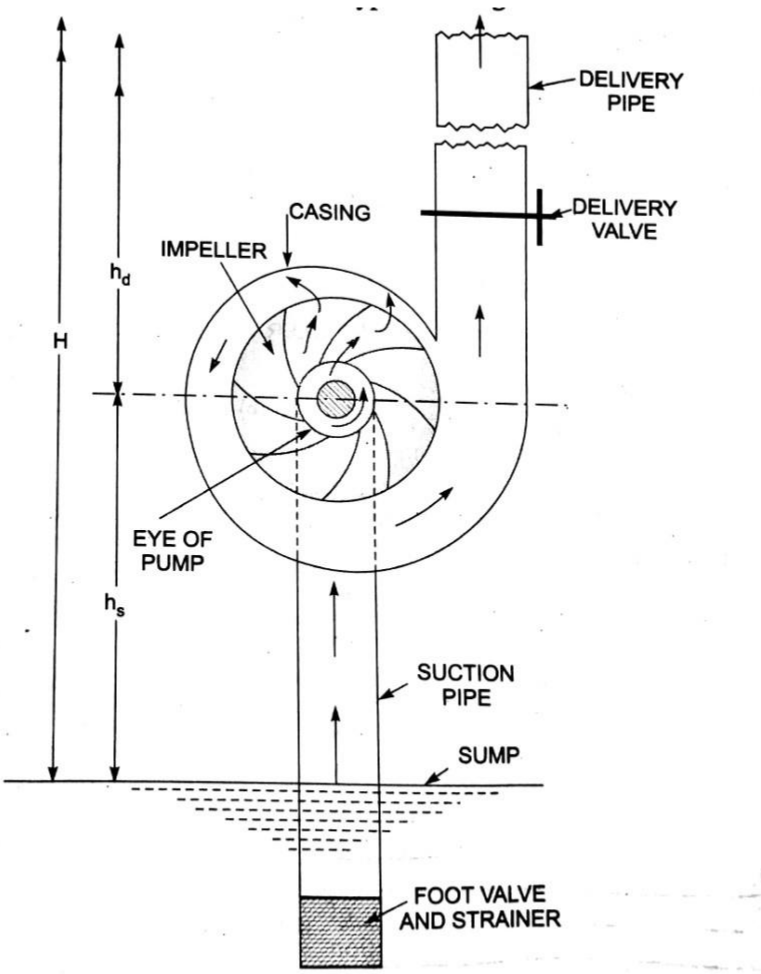


Fig.5.1 Main parts of a Centrifugal pump

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 kinetic energy 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 the casings are commonly adopted :

(a) Volute casing (b) Vortex casing (c) Casing with guide blades

(a) 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 the 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 the impeller, then 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.

(c) Casing with Guide Blades

This casing 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 which a way that the water from the impeller enters the guide vanes without stock. Also the area of the guide vanes increases, thus reducing the velocity of flow through guide vanes and consequently increasing the pressure of water. The water from the guide vanes then passes through the surrounding casing which is in most of the cases concentric with the impeller.

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 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.

4. Delivery pipe

A pipe whose one end is connected to the outlet of the pump and other end derive the water at a required height is known as delivery pipe.

5.1.2 WORK DONE AND EFFICIENCY OF CENTRIFUGAL PUMP

As the impeller rotates, the fluid is drawn into the blade passage at the impeller eye, the centre of the impeller. The inlet pipe is axial and therefore fluid enters the impeller with very little whirl or tangential component of velocity and flows outwards in the direction of the blades. The fluid receives energy from the impeller while flowing through it and is discharged with increased pressure and velocity into the casing. To convert the kinetic energy or fluid at the impeller outlet gradually into pressure energy, diffuser blades mounted on a diffuser ring are used.

The stationary blade passages so formed have an increasing cross-sectional area which reduces the flow velocity and hence increases the static pressure of the fluid. Finally, the fluid moves from the diffuser blades into the volute casing which is a passage of gradually increasing cross-section and also serves to reduce the velocity of fluid and to convert some of the velocity head into static head. Sometimes pumps have only volute casing without any diffuser.

Figure 5.2 shows an impeller of a centrifugal pump with the velocity triangles drawn at inlet and outlet. The blades are curved between the inlet and outlet radius. A particle of fluid moves along the broken curve shown in Figure 5.2.

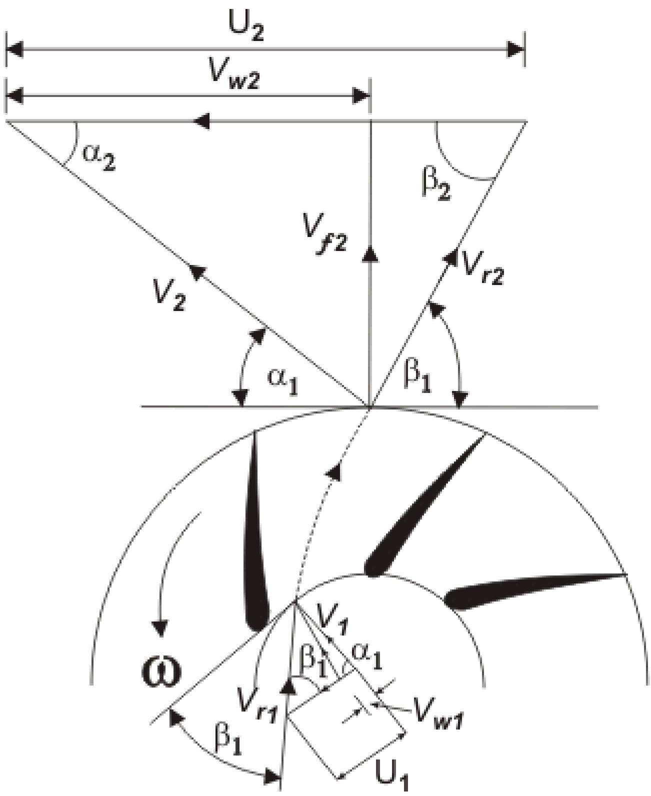


Fig. 5.2 Velocity triangles of centrifugal pump

Let β_1 be the angle made by the blade at inlet, with the tangent to the inlet radius, while β_2 is the blade angle with the tangent at outlet. V_1 and V_2 are the absolute velocities of fluid at inlet an outlet respectively, while V_{r1} and V_{r2} are the relative velocities (with respect to blade velocity) at inlet and outlet respectively. Therefore,

$$\text{Work done on the fluid per unit weight} = (V_{w2}U_2 - V_{w1}U_1)/g$$

A centrifugal pump rarely has any sort of guide vanes at inlet. The fluid therefore approaches the impeller without appreciable whirl and so the inlet angle of the blades is designed to produce a right-angled velocity triangle at inlet (as shown in Fig. 5.2). At conditions other than those for which the impeller was designed, the direction of relative velocity V_r does not coincide with that of a blade. Consequently, the fluid changes direction abruptly on entering the impeller. In addition, the eddies give rise to some back flow into the inlet pipe, thus causing fluid to have some whirl before entering the impeller. However, considering the operation under design conditions, the inlet whirl velocity V_{w1} and accordingly the inlet angular momentum of the fluid entering the impeller is set to zero.

Therefore,

$$\text{Work done on the fluid per unit weight} = (V_{w2}U_2)/g$$

From this equation, it can be observed that the work done is independent of the inlet radius. The difference in total head across the pump known as manometric head, is always less than the quantity $(V_{w2}U_2)/g$ because of the energy dissipated in eddies due to friction.

The ratio of manometric head *H* and the work head imparted by the rotor on the fluid $(V_{w2}U_2)/g$ (usually known as Euler head) is termed as manometric efficiency η_m. It represents the effectiveness of the pump in increasing the total energy of the fluid from the energy given to it by the impeller. Therefore, we can write

$$\eta_m = (gH)/(V_{w2}U_2)$$

The Overall efficiency η_o of pump is defined as,

$$\eta_o = (\rho QgH)/P$$

where, *Q* is the volume flow rate of the fluid through the pump, and *P* is the shaft power, i.e. the input power to the shaft. The energy required at the shaft exceeds $(V_{w2}U_2)/g$ because of friction in the bearings and other mechanical parts. Thus a mechanical efficiency is defined as,

$$\eta_{mech} = (\rho Q V_{w2}U_2)/P$$

So that,

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

5.1.3 MULTISTAGE CENTRIFUGAL PUMPS

If a centrifugal pump consists of two or more impellers, the pump is called a multistage centrifugal pump. The impellers may be mounted on the same shaft or on different shafts. A multistage pump is having the following two important functions:

1. To produce a high head, and
2. To discharge a large quantity of liquid.

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.

5.1.3.1 Multistage Centrifugal Pumps for High Heads.

For developing. a high head, a number of impellers are mounted in series or on the same shaft as shown in Fig 5.3. The water from suction pipe enters the 1st impeller at inlet and is discharged at outlet with increased pressure. The water with increased pressure from the outlet of the 1st impeller is taken to the inlet of the 2nd impeller with the help of a connecting pipe as shown in Fig. 5.4. At the outlet of the 2nd impeller, the pressure of water will be more than the pressure of water at the outlet of the 1st impeller. Thus if more impellers are mounted on the same shaft, the pressure at the outlet will be increased further.

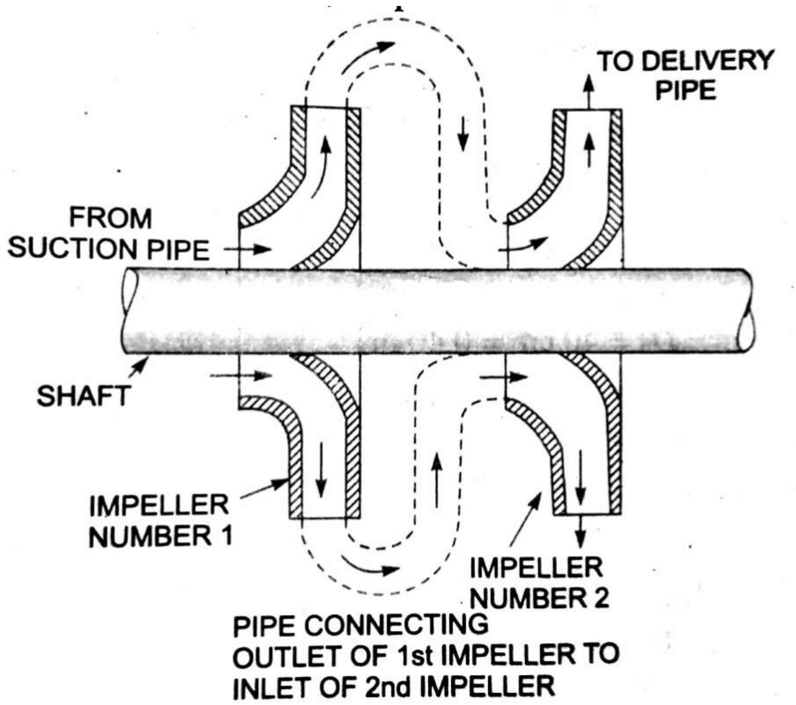


Fig.5.3 Two stage pump with impellers in Series

Let n= Number of identical impellers mounted on the same shaft,

H_m = Head developed by each impeller.

Then total head developed = n x H_m

The discharge passing through each impeller is same.

5.1.3.2 Multistage Centrifugal Pumps for High Discharge.

For obtaining high discharge, the pumps should be connected in parallel as shown in Fig. 5.4. Each of the pumps lifts the water from a common pump and discharges water to a common pipe to which the delivery pipes of each pump is connected. Each of the pump is working against the same head.

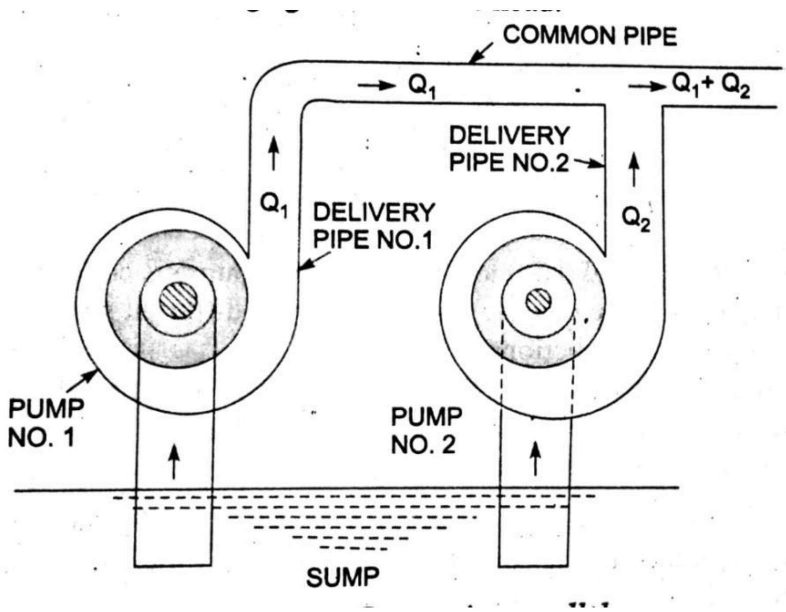


Fig.5.4 Two stage pump with impellers in Parallel

Let

n= Number of identical pumps arranged in parallel.

Q= Discharge from one pump.

Total discharge = n x Q

5.1.4 SPECIFIC SPEED OF A PUMP

Specific speed is an index used to predict desired pump or turbine performance. i.e. it predicts the general shape of a pumps impeller. It is this impeller's "shape" that predicts its flow and head characteristics so that the designer can then select a pump or turbine most appropriate for a particular application. Once the desired specific speed is known, basic dimensions of the unit's components can be easily calculated.

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.

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

where, N_s is specific speed (dimensionless)

N is pump rotational speed (rpm)

Q is flowrate (l/s) at the point of best efficiency

H_m is total head (m) per stage at the point of best efficiency

5.1.5 NET SUCTION SPECIFIC SPEED

The net suction specific speed is mainly used to see if there will be problems with cavitation during the pump's operation on the suction side. It is defined by centrifugal and axial pumps' inherent physical characteristics and operating point. The net suction specific speed of a pump will define the range of operation in which a pump will experience stable operation. The higher the net suction specific speed, then the smaller the range of stable operation, up to the point of cavitation at 8500 (unit less). The envelope of stable operation is defined in terms of the best efficiency point of the pump.

5.1.6 CHARACTERISTIC CURVES OF CENTRIFUGAL PUMPS

Characteristic curves of centrifugal pumps are defined those curves which are plotted form 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 followings are the important characteristic curves for pumps :

1. Main characteristic curves.
2. Operating characteristic curves and
3. Constant efficiency or Muschel curves

I Main Characteristic Curves.

The main characteristic curves of a centrifugal pump consists of variation of 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. And for plotting curves of power versus speed, the manometric head and discharge are kept constant.

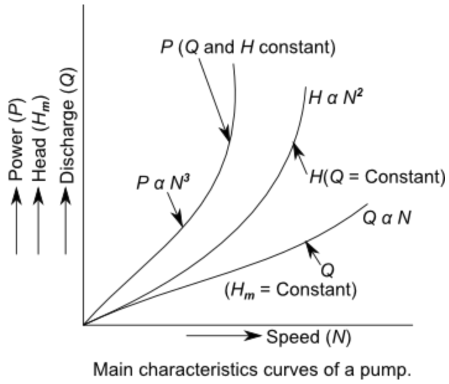


Fig. 5.5 Main characteristic curves

II 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.

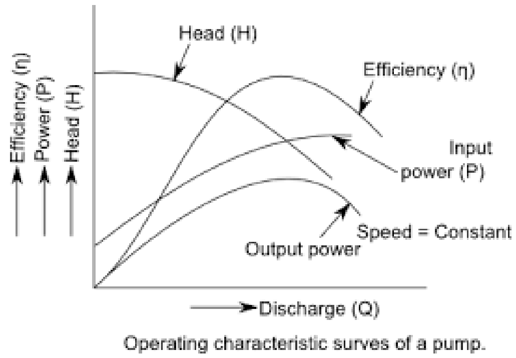


Fig. 5.6 Operating characteristic curves

III Constant Efficiency Curves.

For obtaining constant efficiency curves for a pump, the head versus discharge curves and efficiency versus discharge curves for different speed are used.

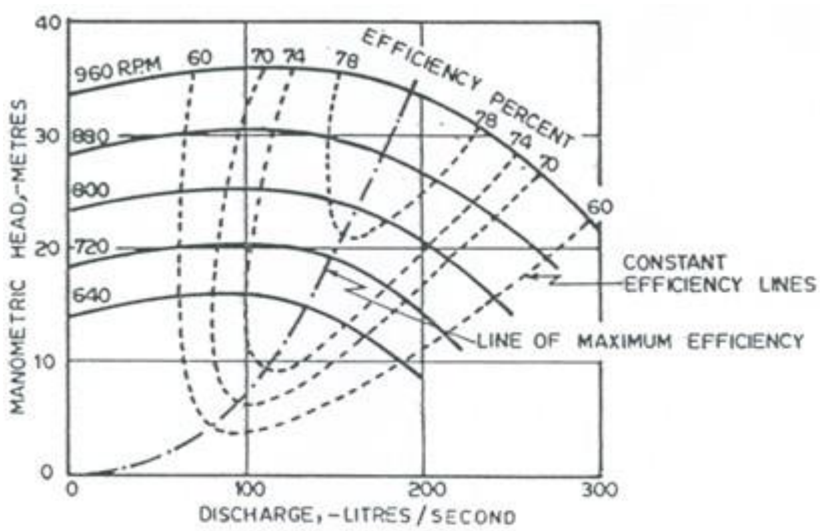


Fig. 5.7 Constant Efficiency curves

RECIPROCATING PUMP

The pumps are the hydraulic machines which convert the mechanical energy into hydraulic energy which is mainly in the form of pressure energy.

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 sucking 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.***

5.2.1 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. Suction valve, and
5. Delivery valve

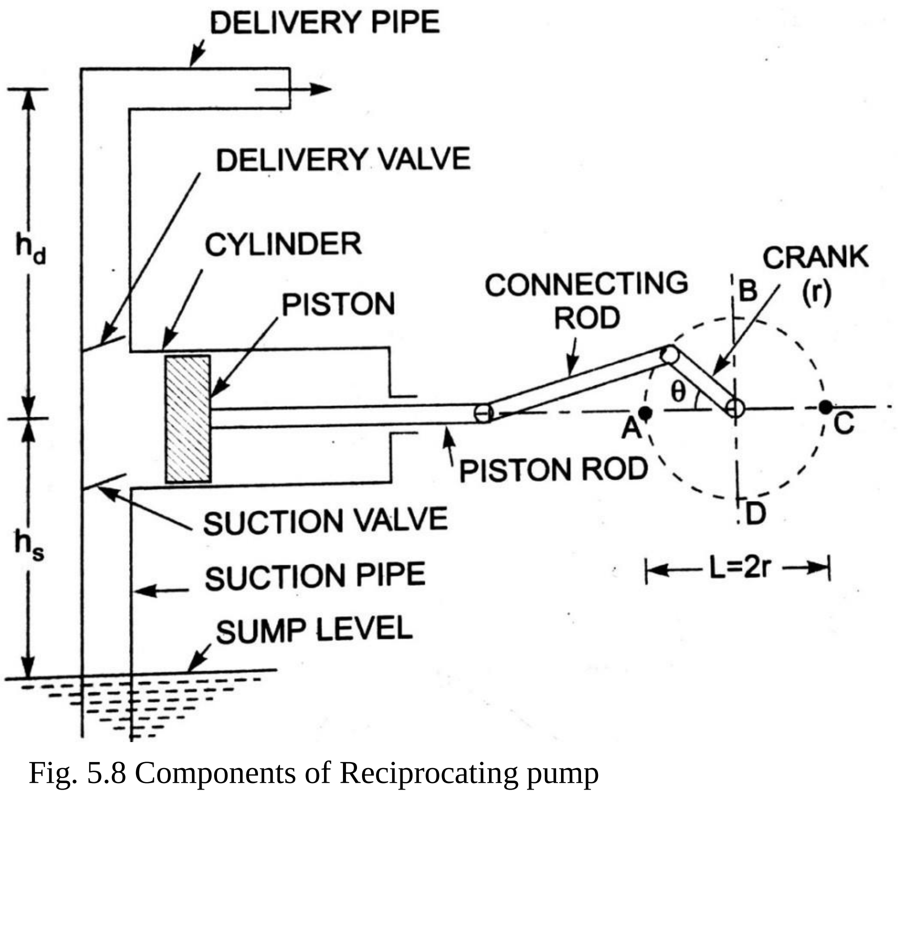


Fig. 5.8 Components of Reciprocating pump

5.2.2 WORKING OF A RECIPROCATING PUMP

- Fig. shows a single acting reciprocating pump, which 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 whereas delivery valve allows water from cylinder to delivery pipe only.
- When crank starts rotating, the piston moves to and fro in the cylinder.
- When 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 ɳ = 0 to ɳ = 180°), 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 is 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 ɳ = 180° to ɳ = 360°), the piston from its extreme right position starts moving towards left in the cylinder.
- The movement of the piston towards left increases the pressure of the liquid inside the cylinder more than atmospheric pressure.
- Hence suction valve closes and delivery valve opens.
- The liquid is forced into the delivery pipe and is raised to a required height.

5.2.3 DISCHARGE THROUGH A RECIPROCATING PUMP

Consider a single acting (Water is acting on one side of the piston only) reciprocating pump as shown in fig

Let D = Diameter of the cylinder

A = Cross-sectional area of the piston or cylinder

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

r = Radius of crank

N = r.p.m. of the crank

L = Length of the stroke = 2 x 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 x Length of stroke

= A x L

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

Discharge of the pump per second, Q= Discharge in one revolution x No. of revolution per second

=A x L x $\frac{N}{60} = \frac{ALN}{60}$

Weight of water delivered per second,

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

5.2.4 WORK DONE BY RECIPROCATING PUMP

Work done by the reciprocating pump per second is given by the reaction as,

Work done per second

= Weight of water lifted per second x Total height through which water is lifted

= W x $(h_s + h_d)$ … (i)

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

Substituting the value of W in equation (i), we get

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

Power required to drive the pump, in kW

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

5.2.5 SLIP OF RECIPROCATING PUMP

Slip of a pump is defined as the difference between the theoretical discharge and actual discharge of the pump. The actual discharge of a pump is less than the theoretical discharge due to leakage.

Hence, mathematically.

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

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

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

$$= \left(1 - \frac{Q_{act}}{Q_{th}}\right)$$

$$= (1 - C_d) \times 100$$

where C_d= Co-efficient of discharge

5.2.6 NEGATIVE SLIP OF THE RECIPROCATING PUMP

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

5.2.7 CLASSIFICATION OF RECIPROCATING PUMPS

The reciprocating pumps may be classified as:

1. According to the water being in contact with one side or both sides of the piston
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.

5.2.8 INDICATOR DIAGRAM

- The indicator diagram for a reciprocating pump is defined as 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.

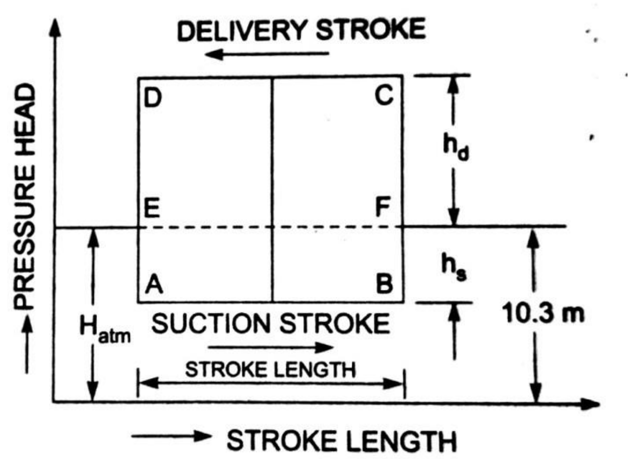


Fig. 5.9 IDEAL INDICATOR DIAGRAM

Work done by the pump per second

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

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

$$\square\ L \times (h_s + h_d)$$

From indicator diagram

Area of indicator diagram = AB × BC

= AB × (BF + FC)

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

η $\boxed{\text{Work done by the pump per second} \quad η \text{ Area of indicator diagram}}$

Numericlas

Problem.

A centrifugal pump has a 100 mm diameter suction pipe and a 75 mm diameter delivery pipe. When discharging 15 l/s of water, the inlet water mercury manometer with one limb exposed to the atmosphere recorded a vacuum deflection of 198 mm; the mercury level on the suction side was 100 mm below the pipe centerline. The delivery pressure gauge, 200 mm above the pump inlet, recorded a pressure of 0.95 bar. The measured input power was 3.2 kW. Calculate the pump efficiency. (See fig.1)

Solution.

Manometric head = rise in total head

$$Hm\ η \frac{p_2}{ηg} \quad \frac{V_2^2}{2g} \quad z \quad η\frac{ηp_1}{ηηg} \quad \frac{V_1^2}{2g}η \quad η\ 1\text{bar}\ η 10.198\text{m of water}$$

$$\frac{p_2}{ηg} η 0.95*10.198\ η 9.65 m of\ water$$

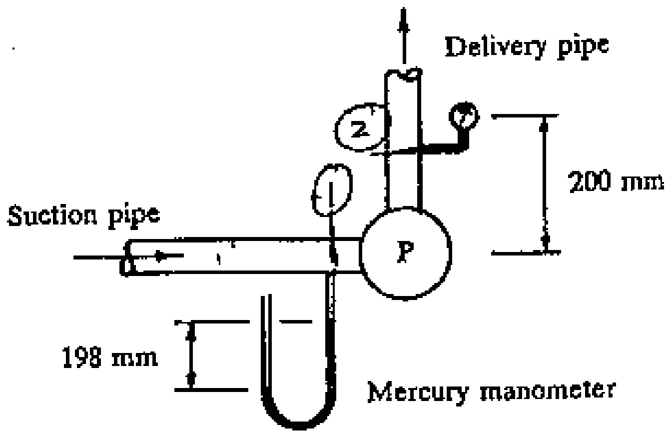
$$V_2\ η 3.39 m/s; \qquad \frac{V_2^2}{2g} η 0.588m$$

$$V_1\ η 1.91 m/s; \qquad \frac{V_1^2}{2g} η 0.186m$$

Then Hm η9.69 0.588 0.2 -(-2.793 0.186) η13.09m

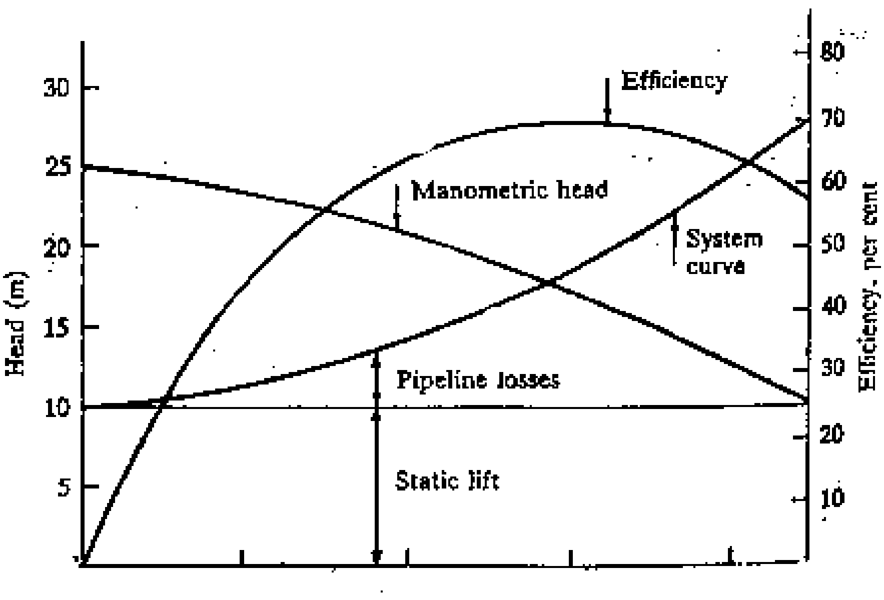
$$\text{Efficiency } (η)\ η \frac{\text{output power}}{\text{input power}}\ η \frac{ηgQHm(watts)}{3200(watts)}$$

$$ηη \frac{3.2η0.015η13.09}{3.2} η 0.602\ (60.2\ percent)$$



Problem. (Pipeline selection in pumping system design)

As existing pump, having the tabulated characteristics is to be used to pump raw sewage to a treatment plant through a static lift of 20 m. An uPVC pipeline 10 km long is to be used. Allowing for minor losses totaling 10 $V^2/2g$ and taking an effective roughness of 0.15 mm



because of sliming, select a suitable commercially available pipe size to achieve a discharge of 60 l/s. Calculate this power consumption.

Discharge (l/s)	0	10	20	30	40	50	60	70
Total head (m)	45	44.7	43.7	42.5	40.6	38	35	31
Overall efficiency (per cent)		35	50	57	60	60	53	40

Solution.

At 60 l/s, total head = 35.0 m, therefore the sum of the static lift and pipeline losses must not exceed 35.0 m.

Try 300 mm diameter: A=0.0707 m^2 V=0.85 m/s

Re = 2.25*100000 ; k/D = 0.0005 η =0.019= f

Friction head loss $\eta \dfrac{0.019\eta 10000 \eta 0.85^2}{0.3\eta 19.62} \eta 23.32m$

Hs + h_f = 43.32 (> 35) pipe diameter too small

Try 300 mm diameter: A=0.0962 m^2 V=0.624 m/s;

Re = 1.93*100000 ; k/D = 0.00043 η =0.0185

H_f=10.48m;   hm $\eta \dfrac{10\ \eta 0.624^2}{19.612} \eta 0.2m$

Hs + h_f + hm =30.68 (< 35 m) O.K

The pump would deliver approximately 70 l/s through the 350 mm pipe and to regulate the flow to 60 l/s an additional head loss of 4.32 m by valve closure would be required.

Power consumption P $\eta \dfrac{1000\eta 9.81\eta 0.06\ \eta 35}{0.55\ \eta 1000} \eta 38.85kW$

Problem. (Pumps in parallel and series)

Two identical pumps having the tabulated characteristics are to be installed in a pumping station to deliver sewage to a settling tank through a 200 mm uPVC pipeline 2.5 km long. The static lift is 15 m. Allowing for minor head losses of $10.0V^2/2g$ and assuming an effective roughness of 0.15 mm calculate the discharge and power consumption if the pumps were to be connected: (a) in parallel, and (b) in series.

Pump Characteristics

discharge (l/s)	0	10	20	30	40
Total head (m)	30	27.5	23.5	17	7.5
Overall efficiency (per cent)	η	44	58	50	18

Solution.

The ‘system curve’ is computed as in the previous examples; this is, of course , independent of the pump characteristics. Calculated system heads (H) are tabulated below for discrete discharges (Q)

$H \;\eta Hs_T \quad h_f \quad h_m$

Q (l/s)	10	20	30	40
H (m)	16.53	20.8	27.37	36.48

(a) Parallel operation

The predicted head v. discharge curve for dual pump operation in parallel mode is obtained as described,.i.e. by doubling she discharge over the range of heads (since the pumps are identical in this case). The system and efficiency curves are added as shown in fig. From the intersection of the characteristic and system curves the following results are obtained:

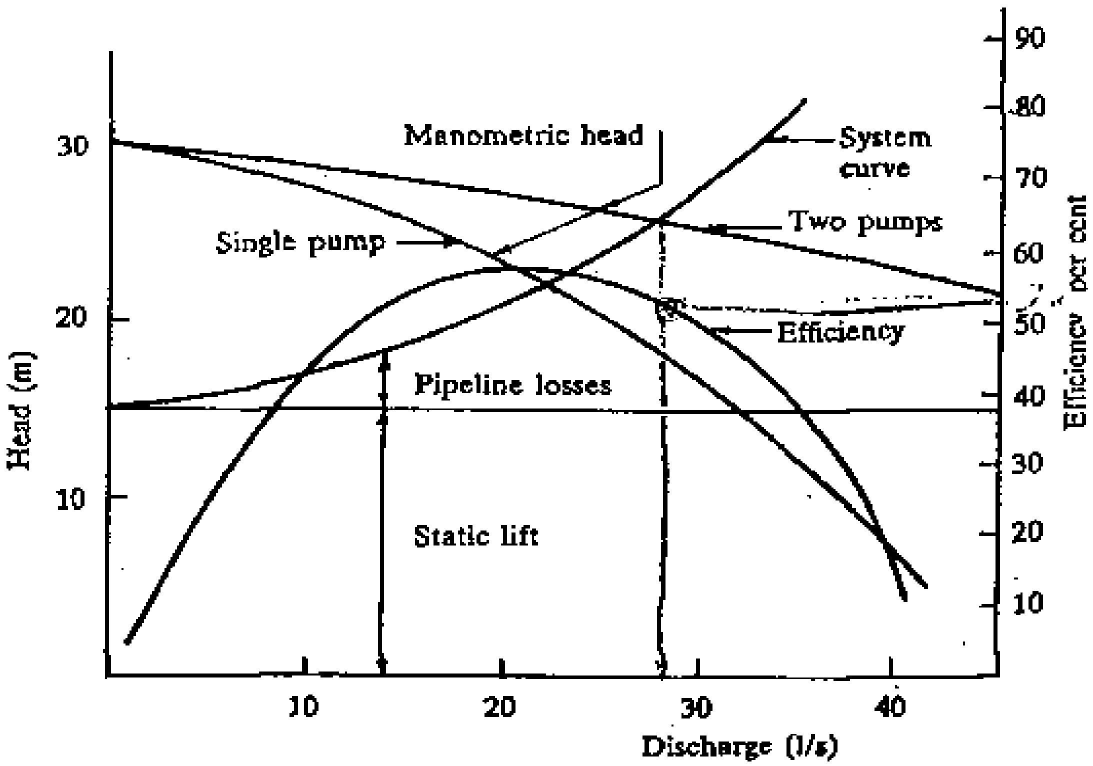


Fig. Parallel operation

Single pimp operation; Q = 22.5 l/S; H_m = 24 m ; η = 0.58 Power consumption = 9.13 kW

Parallel operation, Q = 28.5 l/S; H_m = 26 m ; η = 0.51

(Corresponding with 14.25 l/s per pump)

Power input = 14.11 kW

(b) Series operation

Using the method described in section 6.3 (b) and plotting the dual-pump characteristic curve, intersection with the system curve yields (see Fig.)

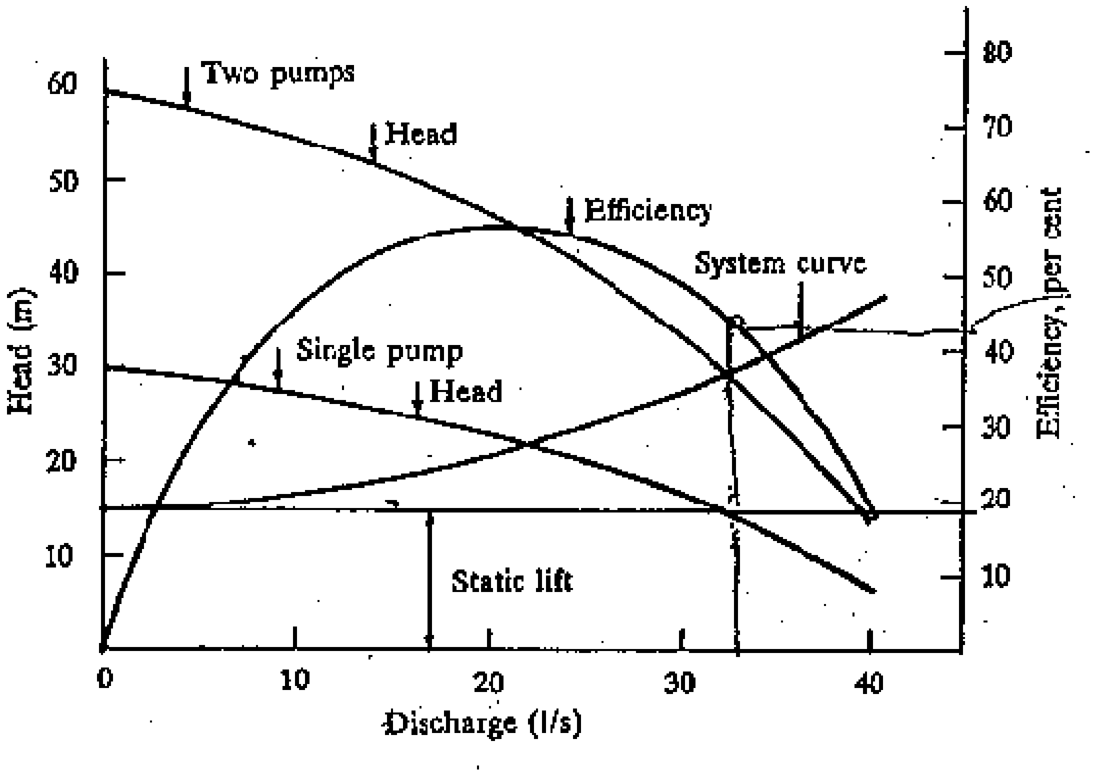


Fig. Series operation

Q=32.5 l/S; H_m = 28 m ; η = 0.41

Power input = 21.77kW

Note that for this particular pipe system, comparing the relative power consumptions the parallel operation is more efficient in producing an increase in discharge than the series operation.

Problem.

A centrifugal pump has the following data :

Rotor inlet diameter	D_1 = 40 mm
Rotor outlet diameter	D_2 = 100 mm
Inlet vane height	h_1 = 60 mm
Outlet vane height	h_2 = 20 mm
Speed	N =1420 rev/min
Flow rate	Q = 0.0022 m^3/s
Blade thickness coefficient	k = 0.95

The flow enters radially without shock.

The blades are swept forward at 30° at exit.

The developed head is 5 m and the power input to the shaft is 170 Watts.

Determine the following.

i. The inlet vane angle
ii. The diagram power
iii. The manometric head
iv. The manometric efficiency
v. The overall efficiency.
vi. The head produced when the outlet valve is shut.
vii. The speed at which pumping commences for a static head of 5 m.

Solution.

$u_1 = \pi N D_1 = 2.97$ m/s

$u_2 = \pi N D_2 = 7.435$ m/s

$v_{r_1} = Q/k\pi D_1 h_1 = 0.307$ m/s

$v_{r2} = Q/k\pi D_2 h_2 = 0.368$ m/s

Since the flow enters radially $v_1 = v_{r_1} = 0.307$ m/s and $v_{W1} = 0$

From the inlet vector diagram the angle of the vane that produces no shock is found as follows:

$\tan\alpha_1 = 0.307/2.97$ hence $\alpha_1 = 5.9°$.

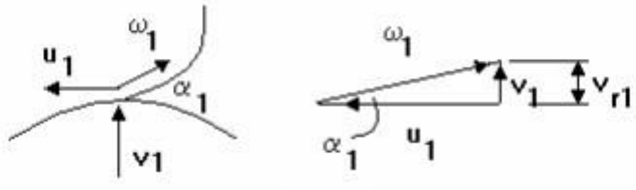


Fig. 41
Inlet vector diagram

From the outlet vector diagram we find :

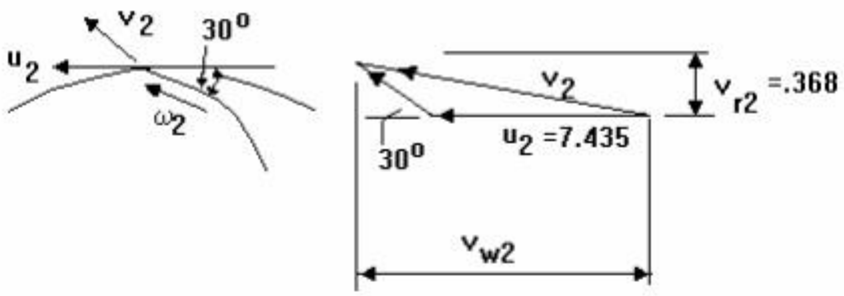


Fig. 42
Outlet vector diagram

$v_{W_2} = 7.435 + 0.368/\tan 30° = 8.07$ m/s

D.P.= $m u_2 v_{W_2}$

D.P.= 2.2 x 7.435 x 8.07 = 132 Watts

W.P. = $mg\Delta h$ = 2.2 x 9.81 x 5 = 107.9 Watts

Δh_m = W.P./D.P.= 107.9/132 = 81.7%

$\Delta h_m = u_2 v_{W_2}/g$ = 7.435 x 8.07/9.81 =6.12 m

$\Delta_m = \Delta h/\Delta h_m$ = 5/6.12 = 81.7%

$\eta_{o/a}$ = W.P./S.P. = 107.9/170 = 63.5%

When the outlet valve is closed the static head is $\Delta h = \frac{u_2^2}{g} = \frac{7.435^2}{9.81} = 5.63\text{ m}$

Problem.

A water pump has a suction lift of 5 m. The friction head in the suction pipe is 0.3m. The kinetic head is negligible. The water temperature is 16°C. Atmospheric pressure is 1.011 bar (10.31 m water). Determine the NPSH .

Solution.

$h_{suc} = 5 + 0.3 = 5.3$ m
$p_s = 0.01817$ bar (from tables)
NPSH = $(1.011 - 0.01817) \times 10^5/(1000 \times 9.81) - 5.3$
NPSH = 4.821 metres of water