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ASPIRE TO EXCEL

ISO 21001

DEPARTMENT OF MECHANICAL ENGINEERING

SUBJECT NOTES

SUB NAME: FLUID MACHINERY

SUBCODE : MET42 **YEAR & SEM :** II/IV

PREPARED BY,

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SVCET

1. A jet of water having a velocity of 20 m/s strikes a curved vane, which is moving with a velocity of 10 m/s. The jet makes an angle of 20° with the direction of motion of vane at inlet and leaves at **an angle of 130° to the direction of motion of vane an outlet.** Calculate :
   (i) Vane angles, so that the water enters and leaves the vane without shock.
   (ii) Work done per second per unit weight of water striking the vane per second

Given:

$V_1 = 20$ m/s

$u_1 = 10$ m/s

$\alpha = 20°$

$\beta = 180° - 130° = 50°$

$u_1 = u_2$

$V_{r1} = V_{r2}$

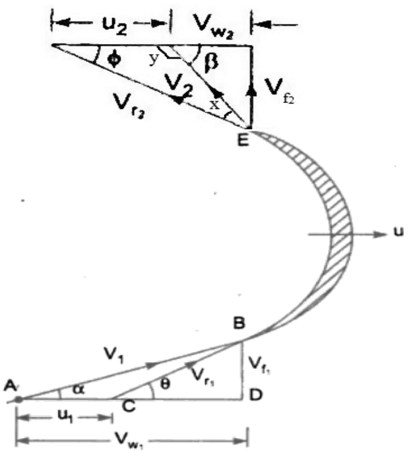
To Find:

$\theta = ?$

$\emptyset = ?$

W.D per unit weight =?

Solution:



a. From inlet velocity triangle

$$\sin\alpha = \frac{V_{f1}}{V_1}$$

$V_{f1} = 6.84$ m/s

$$\cos\alpha = \frac{V_{w1}}{V_1}$$

$V_{w1} = 18.794$ m/s

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

$\boldsymbol{\theta}$ **= 37° 52.5'**

$$\sin\theta = \frac{V_{f1}}{Vr_1}$$

$V_{r1} = V_{r2} = 11.14$ m/s

b. From outlet Velocity triangle

Applying sine rule

$$\frac{u_2}{\sin x} = \frac{Vr_2}{\sin y}$$

$y = 180 - \beta = 130°$

$x = 43°26'$

$x + y + \emptyset = 180°$

$\emptyset$ **= 6°33'**

c. Work done per unit weight:

$$\cos\emptyset = \frac{V_{w2} + u_2}{V_{r2}}$$

$V_{w2} = 1.067$ m/s

$$\text{W.D} = \frac{[V_{w1} \pm V_{w2}] \text{ X } u}{g}$$

W.D = 20.24 Nm/N

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2. A jet of water having a velocity of 40 m/s strikes a curved vane, which is moving with a velocity of 20 m/s. The jet makes an angle of 30° with the direction of motion of vane at inlet and leaves at **an angle of 90° to the direction of motion of vane outlet**. Draw the velocity triangles at inlet and outlet determine the vane angles at inlet and outlet so that the water enters and leaves the vane without shock.

Given:

$V_1 = 40$ m/s

$u_1 = 20$ m/s

$\alpha = 30°$

$\beta = 180° - 90° = 90°$

$u_1 = u_2$

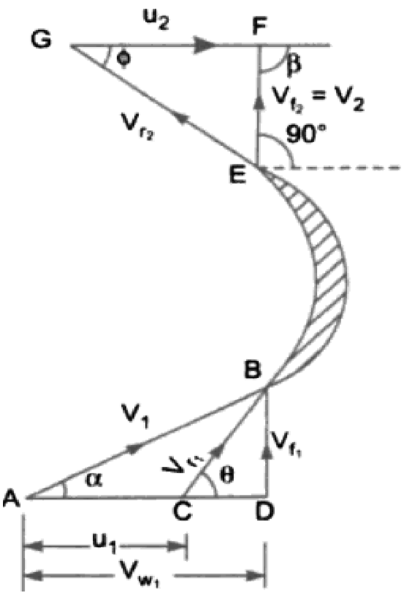
$V_{r1} = V_{r2}$

To Find:

θ =?

$\emptyset$ =?

Solution:



d. From inlet velocity triangle

$$\sin\alpha = \frac{V_{f1}}{V_1}$$

$V_{f1} = 20$ m/s

$$\cos\alpha = \frac{V_{w1}}{V_1}$$

$V_{w1} = 34.64$ m/s

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

$\boldsymbol{\theta}$ **= 53° 47.4'**

$$\sin\theta = \frac{V_{f1}}{Vr_1}$$

$V_{r1} = V_{r2} = 24.78$ m/s

e. From outlet Velocity triangle

$$\cos\emptyset = \frac{u_2}{V_{r2}}$$

$\emptyset$ = 36° 10'

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3. A jet of water of diameter 50 mm, having a velocity of 20 m/s strikes a curved vane which is moving with a velocity of 10 m/s in **the direction of the jet**. The jet leaves the vane of an **angle of 60° to the direction of motion of vane at outlet**. Determine:
   (i) The force exerted by the jet on the vane in the direction of motion
   (ii) Work done per second of the jet

Given:

d = 0.05 m

$V_1 = 20$ m/s

$u_1 = 10$ m/s

$\alpha = 0°$

$\beta = 180° – 60° = 120°$

$u_1 = u_2$

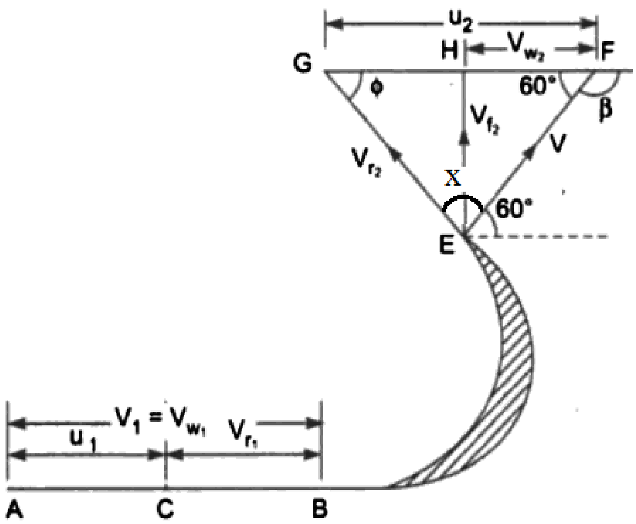
$V_{r1} = V_{r2}$

To Find:

Force exerted by the jet =?

W.D per second =?

Solution:



From inlet velocity triangle

$V_{r1} = V_1 – u_1 = 10$ m/s

$V_{w1} = V_1 = 20$ m/s

$V_{r2} = V_{r1} = 10$ m/s

From outlet velocity triangle

$$\frac{u_2}{\sin x} = \frac{Vr_2}{\sin 60}$$

$x = 60°$

$x + 60 + \emptyset = 180°$

$\emptyset = 60°$

$$\cos \emptyset = \frac{u_2 - V_{w2}}{V_{r2}}$$

$V_{w2} = 5$ m/s

a. Force exerted by the jet:

$F_X = \rho a V_{r1} [V_{w1} - V_{w2}]$

$F_X =$ **294.45 N**

b. Work done per second:

$W.D = \rho a V_{r1} [V_{w1} - V_{w2}] \times u$

W.D = 2944.5 Nm/s

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4. A jet of water having a velocity of 15 m/s strikes a curved vane which is moving with a velocity of 5 m/s. The vane symmetrical and is so shaped that **the jet is deflected through 120°.** Find the angle of the jet at inlet of the vane so that there is no shock. What is the absolute velocity of the jet at outlet in magnitude and direction and work done per unit weight of water? Assume the vane to be smooth.

Given:

$V_1 = 20$ m/s

$u_1 = 10$ m/s

$\emptyset = \theta$

$\emptyset + \theta = 180° - 120° = 60°$

$\theta = 30°$

$\emptyset = 30°$

$u_1 = u_2$, $V_{r1} = V_{r2}$

To Find:

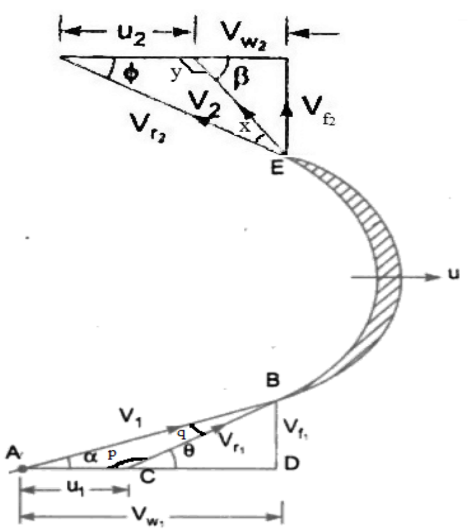
α =?

V_2 =?

β^* =?

W.D per unit weight =?

Solution:



From inlet velocity triangle

Applying sine rule

$$\frac{u_1}{\sin q} = \frac{V_1}{\sin p}$$

$p = 180° - 30° = 150°$

$q = 9.596°$

$p + q + \alpha = 180°$

α = 20° 24'

Applying sine rule

$$\frac{u_1}{\sin q} = \frac{V_{r1}}{\sin \alpha}$$

$V_{r1} = V_{r2} = 10.46$ m/s

From outlet velocity triangle

$\cos \emptyset = \frac{V_{w2} + u_2}{V_{r2}}$, **$V_{w2}$ = 4.06 m/s**

$\sin \emptyset = \frac{V_{f2}}{V_{r2}}$, **$V_{f2}$ = 5.23 m/s**

$V_2 = \sqrt{(V^2_{f2} + V^2_{w2})} =$ **6.62 m/s**

$$\tan \beta = \frac{V_{f2}}{V_{w2}}$$

$\beta = 52°10'$

$\beta^* = 180 - \beta = 127°50'$

Work done per unit weight:

$$W.D = \frac{[V_{w1} \pm V_{w2}] \times u}{g}$$

W.D = 9.225 Nm/N

5. A jet of water moving at 12 m/s impinges on vane shaped to **deflect the jet through 120°** when stationary. If the vane is moving at 5 m/s, find the angle of the jet so that there is no shock at inlet. What is the absolute velocity of the jet at exit in magnitude and direction and the work done per second per unit weight of water striking per second? Assume that the vane is smooth.

Given:

V_1 = 12 m/s

u_1 = 5 m/s

$\emptyset = \theta$

$\emptyset + \theta$ = 180° – 120° = 60°

θ = 30°

$\emptyset$ = 30°

$u_1 = u_2$, $V_{r1} = V_{r2}$

To Find:

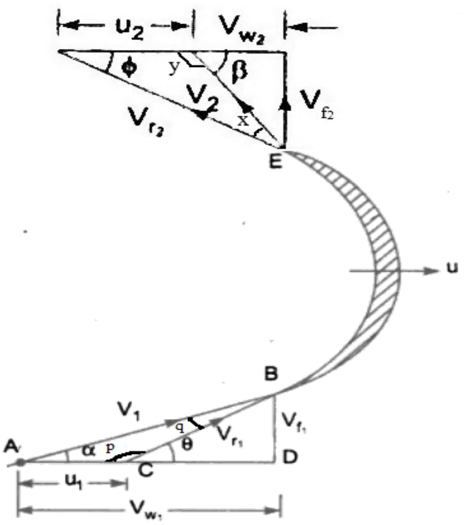
α =?

V_2 =?

β^* =?

W.D per unit weight =?

Solution:



From inlet velocity triangle

Applying sine rule

$$\frac{u_1}{\sin q} = \frac{V_1}{\sin p}$$

p = 180° - 30° = 150°

q = 12.02°

p + q + α = 180°

α = 17° 59'

Applying sine rule

$$\frac{u_1}{\sin q} = \frac{V_{r1}}{\sin \alpha}$$

$V_{r1} = V_{r2}$ = 7.41 m/s

From outlet velocity triangle

$$\cos \emptyset = \frac{V_{w2} + u_2}{V_{r2}}, \; \mathbf{V_{w2} = 1.417\ m/s}$$

$$\sin \emptyset = \frac{V_{f2}}{V_{r2}}, \; \mathbf{V_{f2} = 3.705\ m/s}$$

$V_2 = \sqrt{(V^2_{f2} + V^2_{w2})}$ = **3.96 m/s**

$$\tan \beta = \frac{V_{f2}}{V_{w2}}$$

β = 69°4.2'

β^* = 180 – β = 110°55.8'

Work done per unit weight:

$$\text{W.D} = \frac{[V_{w1} \pm V_{w2}]\ X\ u}{g}$$

W.D = 6.537 Nm/N

6. A jet of water having a velocity of 15 m/s, strikes a curved vane which is moving with a velocity of 5 m/s **in the same direction as that of jet at inlet**. The vane is so shaped that **the jet is deflected through 135°.** The diameter of jet is 100 mm. Assuming the vane to be smooth, find: (i) Force exerted by the jet on the vane in the direction of motion,(ii)Power exerted on the vane,(iii) Efficiency of the vane.

Given:

V_1 = 15 m/s

u_1 = 5 m/s

$\theta = 0$

$\emptyset + \theta$ = 180° – 135° = 45°

$\emptyset$ = 45°

$u_1 = u_2$, $V_{r1} = V_{r2}$

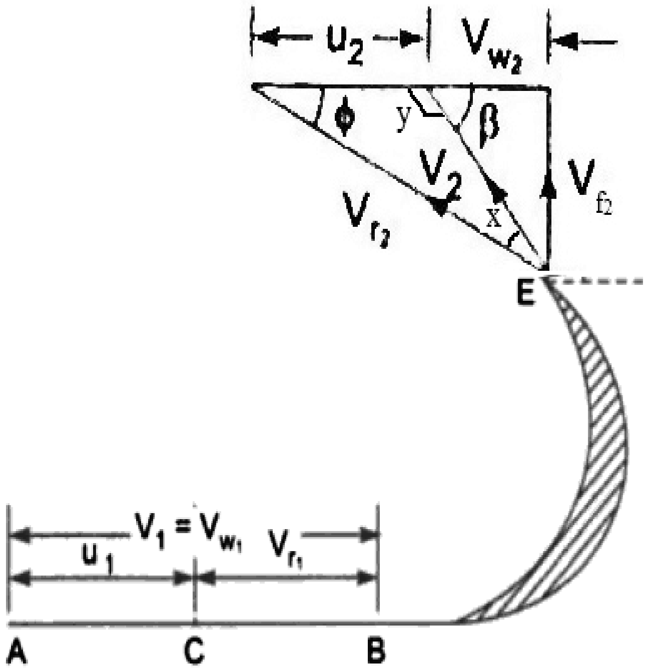
To Find:

Force exerted by the jet =?

Power exerted by the vane =?

Efficiency of the vane =?

Solution:



From inlet velocity triangle

$V_{r1} = V_1 - u_1 = 10$ m/s

$V_{w1} = V_1 = 15$ m/s

$V_{r2} = V_{r1} = 10$ m/s

From outlet velocity triangle

$$\cos \emptyset = \frac{V_{w2} + u_2}{V_{r2}}$$

$\mathbf{V_{w2} = 2.07}$ **m/s**

a. Force exerted by the jet

$F_X = \rho a V_{r1} [V_{w1} + V_{w2}]$

F_X = **1340.6 N**

b. Power of the vane

$P = \frac{F_X \text{ x } u}{1000}$ = **6.703 kW**

c. **Efficiency:**

$\text{Efficiency} = \frac{W.D}{K.E}$

$W.D = \rho a V_{r1} [V_{w1} + V_{w2}] \text{ x } u = 6703$ Nm/s

$K.E = \frac{1}{2} m V_1^2 \quad = \frac{1}{2}(\rho a V_1)V_1^2 = \frac{1}{2}\rho a V_1^3$

K.E = 13253.6 Nm/s

Efficiency = 50.5 %

7. A jet of water having a velocity of 35 m/s impinges on a series of vanes moving with a velocity of 20 m/s. The jet makes an **angle of 30° to the direction of motion of vanes when entering and leaves at an angle of 120°.** Draw the velocity triangles and find: (a) the angles of vane tips so that water enters and leaves without shock, (b) the work done per unit weight of water entering the vanes (c) the efficiency

Given:

$V_1 = 35$ m/s

$u_1 = 20$ m/s

$\alpha = 30°$

$\beta = 180° - 120° = 60°$

$u_1 = u_2$

$V_{r1} = V_{r2}$

To Find:

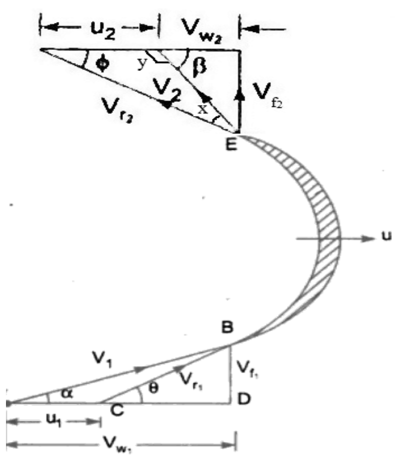
θ =?

$\emptyset$ =?

W.D per unit weight =?

Efficiency of the vane =?

Solution:



From inlet velocity triangle

$$\sin \alpha = \frac{V_{f1}}{V_1}$$

V_{f1} = 17.50 m/s

$\cos\alpha = \frac{V_{w1}}{V_1}$

V_{w1} = 30.31 m/s

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

$\boldsymbol{\theta}$ **= 60°**

$\sin\theta = \frac{V_{f1}}{Vr_1}$

$V_{r1} = V_{r2}$ = 20.25 m/s

From outlet Velocity triangle

Applying sine rule

$\frac{u_2}{\sin x} = \frac{Vr_2}{\sin y}$

$y = 180 - \beta = 120°$

x = 58.75°

x + y + ∅ = 180°

∅ = 1.25°

Work done per unit weight:

$\cos\emptyset = \frac{V_{w2} + u_2}{V_{r2}}$

V_{w2} = 0.24 m/s

$\text{W.D} = \frac{[V_{w1} \pm V_{w2}] \text{ X } u}{g}$

W.D = 62.28 Nm/N

Efficiency:

$\text{Efficiency} = \frac{W.D}{K.E}$

$\text{K.E} = \frac{1}{2}mV_1^2$ ------ (per unit weight m = 1/g)

K.E = 62.43

Efficiency = 99.74 %

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8. A jet of water having a velocity of 30 m/s strikes a series of radial curved vanes mounted on a wheel which is rotating at 200 r.p.m. The jet makes an angle of 20° with tangent to the wheel at inlet and leaves the wheel with a velocity of 5 m/s at an angle of 130° to the tangent to the wheel at outlet. **Water is flowing outward in radial direction.** The outer and inner radii of the wheel are 0.5 m and 0.25 m respectively. Determine : (a) vane angle at inlet and outlet (b) work done per unit weight of water (c) efficiency of the wheel

Given:

V_1 = 30 m/s

N = 200 r.p.m

$\omega = \frac{2\pi N}{60}$ = 20.94 rad/s

$\alpha = 20°$

$\beta = 180° - 130° = 50°$

V_2 = 5 m/s

R_1 = 0.5 m

R_2 = 0.25 m

To Find:

θ =?

∅ =?

W.D per unit weight =?

Efficiency of the vane =?

Solution:

$u_1 = \omega R_1$ = 10.47 m/s

$u_2 = \omega R_2$ = 5.235 m/s

Refer diagram in above problem

V_2 is given so easy to find data in velocity triangles

From inlet velocity triangle

$\boldsymbol{\theta}$ **= 30° 4.2'**

From outlet Velocity triangle

∅ = 24° 23'

Work done per unit weight:

$\text{W.D} = \frac{[V_{w1} u_1 + V_{w2} u_2]}{g}$ --------- **(radial)**

W.D = 31.8 Nm/N

Efficiency:

$\text{Efficiency} = \frac{W.D}{K.E}$

$\text{K.E} = \frac{1}{2}mV_1^2$ ------ (per unit weight m = 1/g)

K.E = 45.87

Efficiency = 69.32 %

Pelton Wheel

Pelton wheel is well suited for operating under high heads. A pelton turbine has one or more nozzles discharging jets of water which strike a series of buckets mounted on the periphery of a circular disc. The *runner* consists of a circular disc with a number of buckets evenly spaced round its periphery. The *buckets* have a shape of a double semi-ellipsoidal cup. The pelton bucket is designed to deflect the jet back through 165° which is the maximum angle possible without the return jet interfering with the next bucket.

Pelton wheels are easier to fabricate and are relatively cheaper. The turbines are in general, not subjected to the cavitations effect. The turbines have access to working parts so that the maintenance or repairs can be affected in a shorter time.

Traditionally, micro hydro pelton wheels were always single jet because of the complexity and the cost of flow control governing of more than one jet.

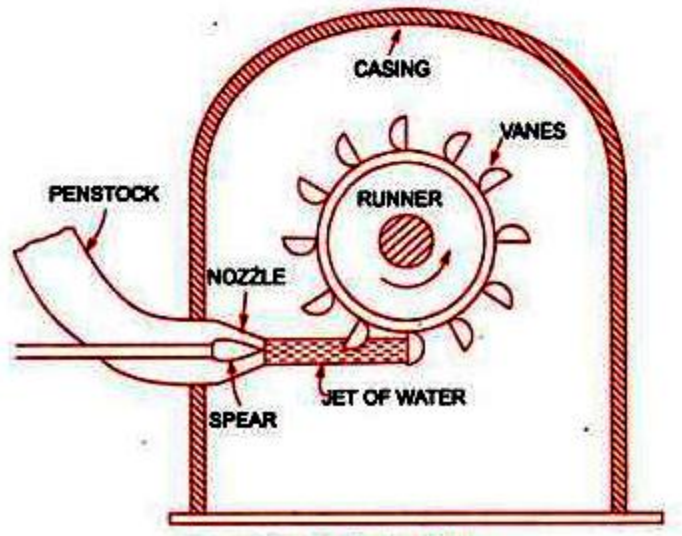
Advantages of multi-jet:

- Higher rotational speed
- Smaller runner
- Less chance of blockage

Disadvantages of multi-jet:

- Possibility of jet interference on incorrectly designed systems
- Complexity of manifolds

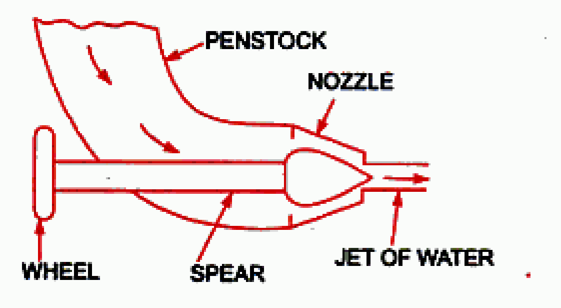
THE MAIN PARTS OF THE PELTON TURBINE ARE



Pelton turbine.

1. Nozzle and Flow regulating arrangement (spear)
2. Runner and buckets(vanes)
3. Casing
4. Breaking jet

1. Nozzle and flow Regulating Arrangement. The amount of water striking the buckets (vanes) of the runner is controlled by providing a spear in the nozzle as shown in Fig. 18.2. The spear is a conical needle which is operated either by a hand wheel or automatically in an axial direction depending upon the size of the unit. When the spear is pushed forward into the nozzle the amount of water striking the runner is reduced. On the other hand, if the spear is pushed back, the amount of water striking the runner increases.



Nozzle with a spear to regulate flow.

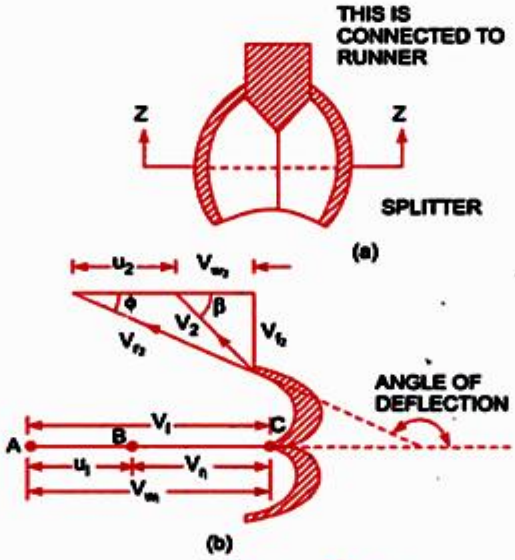
2. **Runner with Buckets.** Fig. 18.3 shows the runner of a Pelton wheel. It consists of a circular disc on the periphery of which a number of buckets evenly spaced are fixed. The shape of the buckets is of a double hemispherical cup or bowl. Each bucket is divided into two symmetrical parts by a dividing wall which is known as splitter.

The jet of water strikes on the splitter. The splitter divides the jet into two equal parts and the jet comes out at the outer edge of the bucket. The buckets are shaped in such a way that the jet gets deflected through 160° or 170°. The buckets are made of cast iron, cast steel bronze or stainless steel depending upon the head at the inlet of the turbine.

3. **Casing.** Fig. 18.4 shows a Pelton turbine with a casing. The function of the casing is to prevent the splashing of the water and to discharge water to tail race. It also acts as safeguard against accidents. It is made of cast iron or fabricated steel plates. The casing of the Pelton wheel does not perform any hydraulic function.

4. **Breaking Jet.** When the nozzle is completely closed by moving the spear in the forward direction, the amount of water striking the runner reduces to zero. But the runner due to inertia goes on revolving for a long time. To stop the runner in a short time, a small nozzle is provided which directs the jet of water on the back of the vanes. This jet of water is called breaking jet.

Velocity Triangles and Work done for Pelton Wheel. shows the shape of the vanes or buckets of the Pelton wheel. The jet of water from the nozzle strikes the bucket at the splitter, which splits up the jet into two parts. These parts of the jet, glides over the inner surfaces and comes out at the outer edge. Fig. 18.5 (*b*) shows the section of the bucket at *z-z*. The splitter is the inlet tip and outer edge of the bucket is the outlet tip of the bucket. The inlet velocity triangle is drawn at the splitter and outlet velocity triangle is drawn at the outer edge of the bucket, by the same method as explained in Chapter 17.



Shape of bucket.

Let $\quad H$ = Net head acting on the Pelton wheel

$= H_g - h_f$

where H_g = Gross head and $h_f = \frac{4fLV^2}{D^* \times 2g}$

where D^* = Dia. of Penstock, N = Speed of the wheel in r.p.m.
D = Diameter of the wheel, d = Diameter of the jet.

Then V_1 = Velocity of jet at inlet $= \sqrt{2gH}$

$$u = u_1 = u_2 = \frac{\pi DN}{60}.$$

The velocity triangle at inlet will be a straight line where

$$V_{r_1} = V_1 - u_1 = V_1 - u$$

$$V_{w_1} = V_1$$

$$\alpha = 0 \text{ and } \theta = 0$$

From the velocity triangle at outlet, we have

$$V_{r_2} = V_{r_1} \text{ and } V_{w_2} = V_{r_2} \cos\phi - u_2.$$

The force exerted by the jet of water in the direction of motion is given by equation (17.19) as

$$F_x = \rho a V_1 [V_{w_1} + V_{w_2}] \quad \text{...(18.8)}$$

As the angle β is an acute angle, +ve sign should be taken. Also this is the case of series of Vanes, the mass of water striking is $\rho a V_1$ and not $\rho a V_{r_1}$. In equation (18.8), 'a' is the area of the jet which is given as

$$a = \text{Area of jet} = \frac{\pi}{4}d^2.$$

Now work done by the jet on the runner per second

$$= F_x \times u = \rho a V_1 [V_{w_1} + V_{w_2}] \times u \text{ Nm/s} \quad \text{...(18.9)}$$

Power given to the runner by the jet

$$= \frac{\rho a V_1 [V_{w_1} + V_{w_2}] \times u}{1000} \text{ kW} \quad \text{...(18.10)}$$

Work done/s per unit weight of water striking/s

$$= \frac{\rho a V_1 [V_{w_1} + V_{w_2}] \times u}{\text{Weight of water striking/s}}$$

$$= \frac{\rho a V_1 [V_{w_1} + V_{w_2}] \times u}{\rho a V_1 \times g} = \frac{1}{g}[V_{w_1} + V_{w_2}] \times u \quad \text{...(18.11)}$$

The energy supplied to the jet at inlet is in the form of kinetic energy and is equal to $\frac{1}{2}mV^2$

$\therefore$ K.E. of jet per second $= \frac{1}{2}(\rho a V_1) \times V_1^2$

$\therefore$ **Hydraulic efficiency,** $\eta_h = \frac{\text{Work done per second}}{\text{K.E. of jet per second}}$

$$= \frac{\rho a V_1 [V_{w_1} + V_{w_2}] \times u}{\frac{1}{2}(\rho a V_1) \times V_1^2} = \frac{2[V_{w_1} + V_{w_2}] \times u}{V_1^2} \quad \text{...(18.12)}$$

Now $V_{w_1} = V_1, V_{r_1} = V_1 - u_1 = (V_1 - u)$

$\therefore$ $V_{r_2} = (V_1 - u)$

and $V_{w_2} = V_{r_2} \cos\phi - u_2 = V_{r_2} \cos\phi - u = (V_1 - u)\cos\phi - u$

Substituting the values of V_{w_1} and V_{w_2} in equation (18.12),

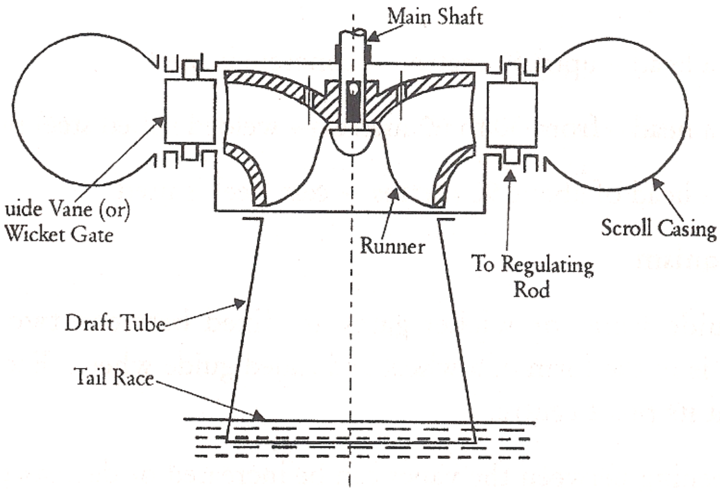
$$\eta_h = \frac{2[V_1 + (V_1 - u)\cos\phi - u] \times u}{V_1^2}$$

$$= \frac{2[V_1 - u + (V_1 - u)\cos\phi] \times u}{V_1^2} = \frac{2(V_1 - u)[1 + \cos\phi]u}{V_1^2}. \quad \text{...(18.13)}$$

Maximum Efficiency $= \frac{1+\cos\phi}{2}$ ....$u = \frac{V1}{2}$

Francis Turbine

Francis turbine is a mixed flow type, in which water enters the runner radially at its outer periphery and leaves axially at its centre. Fig.5.6 illustrates the Francis turbine. The runner blades are profiled in a complex manner and the casing is scrolled to distribute water around the entire perimeter of the runner. The water from the penstock enters a *scroll case* which completely surrounds the runner. The purpose of the scroll case is to provide an even distribution of water around the circumference of the turbine runner, maintaining an approximately constant velocity for the water so distributed. The function of *guide vane* is to regulate the quantity of water supplied to the runner and to direct water on to the runner at an angle appropriate design. A *draft tube* is a pipe or passage of gradually increasing cross sectional area which connects the runner exit to the tail race.



Description of main parts

Francis turbine consists mainly of the following parts

a) Spiral or scroll casing
b) Guide mechanism
c) Runner and turbine main shaft
d) Draft tube

Spiral casing of scroll casing

The casing of the francis turbine is designed in a spiral form with a gradually increasing area. The advantages of this design are

i) Smooth and even distribution of water around the runner.
ii) Loss of head due to the formation of eddies is avoided.
iii) Efficiency of flow of water to the turbine is increased.

- In big units stay vanes are provided which direct and water to the guide vanes.
- The casing is also provided with inspection holes and pressure gauge connection.
- The selection of material for the casing depends upon the head of water to be supplied

For a head – up to 30 metres – concrete is used.

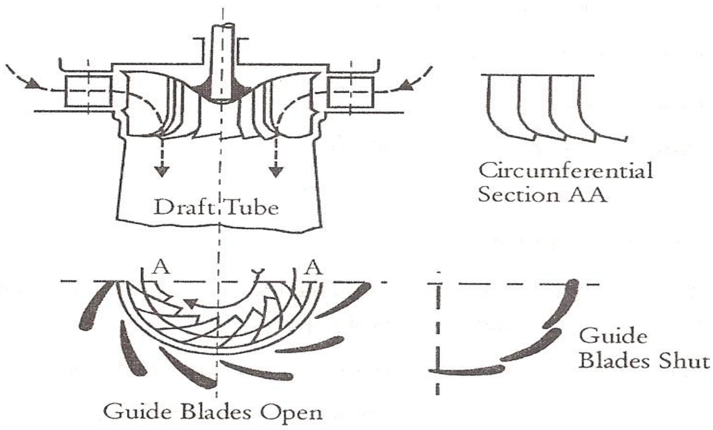
For a head – from 30 to 60 metres – welded rolled steel plates are used.

For a head of above 90 metres – cast steel is used.

Guide mechanism

The guide vanes of wicket gates are fixed between two rings. This arrangement is in the form of a wheel and called guide wheel. Each vane can be rotated about its pivot centre.

The opening between the vanes can be increased or decreased by adjusting the guide wheel. The guide wheel is adjusted by the regulating shaft which is operated by a governor.



The guide mechanism provides the required quantity of water to the runner depending upon the load conditions. The guide vanes are in general made of cast steel.

Runner and turbine main shaft

The flow in the runner of a modern Francis turbine is partly radial and partly axial. The runners may be classified as

i) Slow ii) Medium iii) Fast

The runner may be cast in one piece or made of separate steel plates welded together. The runners are made of CI for small output, cast steel or stainless steel or bronze for large output. The runner blades should be carefully finished with high degree of accuracy.

The runner may be keyed to the shaft which may be vertical or horizontal. The shaft is made of steel and is forged it is provided with a collar for transmitting the axial thrust.

Draft tube

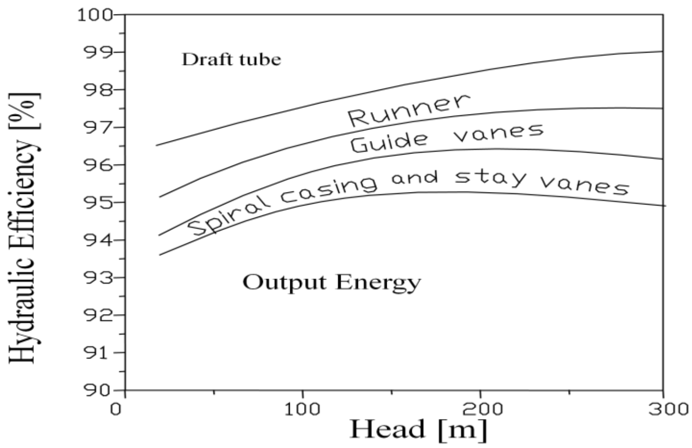
The water after doing work on the runner passes on to the tail race through a tube called draft tube. It is made of riveted steel plate or pipe or a concrete tunner. The cross – section of the tube increases gradually towards the outlet. The draft tube connects the runner exit to the tail race. This tube should be drowned approximately 1 metre below the tail race water level.

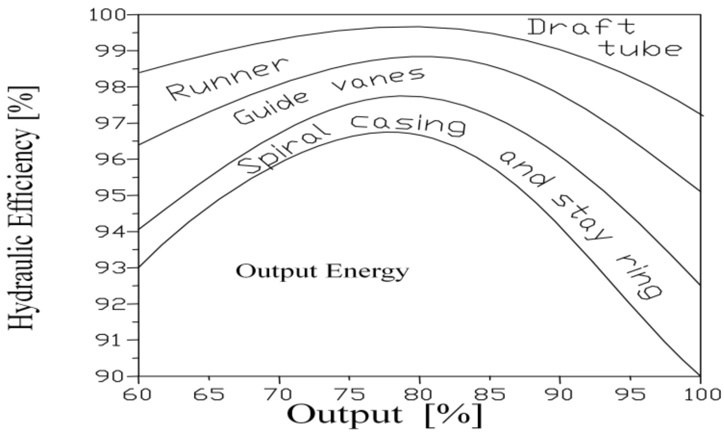
Working Principles of Francis Turbine

The water is admitted to the runner through guide vanes or wicket gates. The opening between the vanes can be adjusted to vary the quantity of water admitted to the turbine. This is done to suit the load conditions.

The water enters the runner with a low velocity but with a considerable pressure. As the water flows over the vanes the pressure head is gradually converted into velocity head. This kinetic energy is utilized in rotating the wheel. Thus the hydraulic energy is converted into mechanical energy. The outgoing water enters the tail race after passing through the draft tube. The draft tube enlarges gradually and the enlarged end is submerged deeply in the tail race water. Due to this arrangement a suction head is created at the exit of the runner.

<u>Losses in Francis Turbines</u>





VELOCITY TRIANGLE

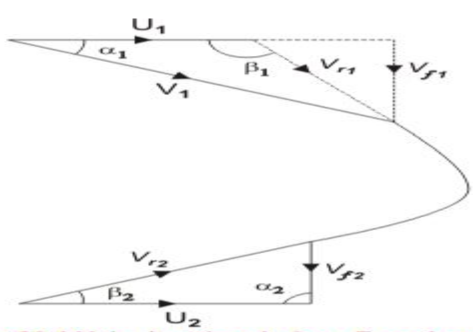
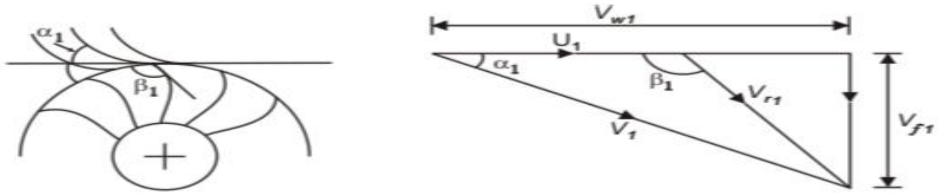


Figure 29.1 Velocity triangle for a Francis runner

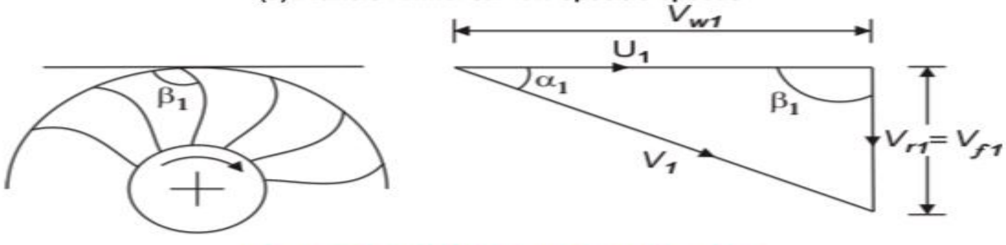
Work done per second on the runner per second will be = $\rho Q[V_{w1}U_1]$

Work done per second per unit weight of water striking/s = $\frac{1}{g}[V_{w1}U_1]$

Hydraulic Efficiency will be given by = $\frac{Vw1U1}{gH}$



(a) Francis runner for low specific speeds



(b) Francis runner for normal specific speeds

Kaplan Turbine

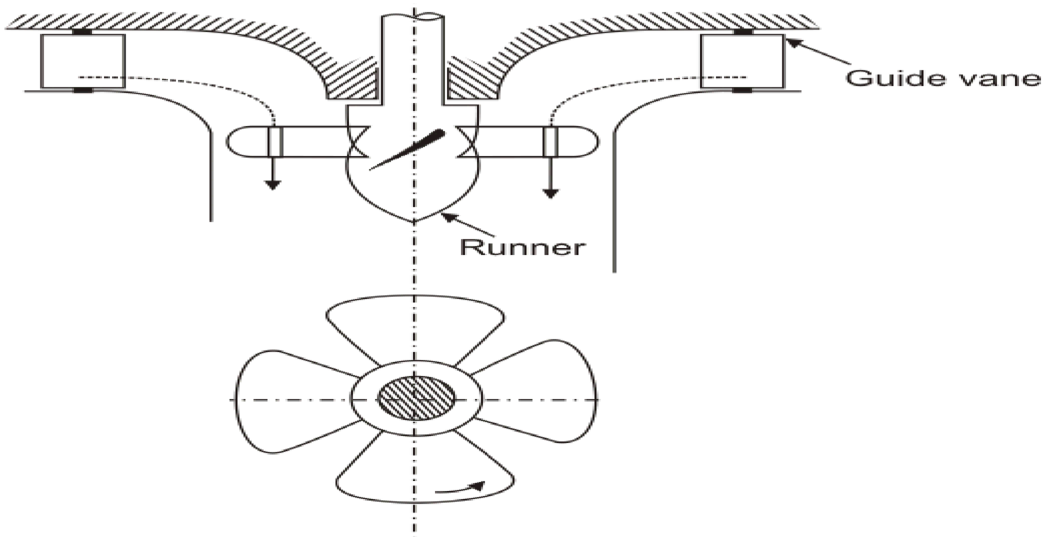


Fig. 30.3 A propeller of Kaplan turbine

It is an axial flow turbine which is suitable for relatively low heads. It will be seen that the main components of Kaplan turbine such as scroll casing, guide vanes, and the draft tube are similar to those of a Francis turbine.

This type of turbine evolved from the need to generate power from much lower pressure heads than are normally employed with the Francis turbine. To satisfy large power demands very large volume flow rates need to be accommodated in the Kaplan turbine, i.e. the product QHE is large. The overall flow configuration is from radial to axial. Figure 9.16 is a part sectional view of a Kaplan turbine in which the flow enters from a volute into the inlet guide vanes which impart a degree of swirl to the flow determined by the needs of the runner. The flow leaving the guide vanes is forced by the shape of the passage into an axial direction and the swirl becomes essentially a free vortex.

The vanes of the runner are similar to those of an axial-flow turbine rotor but designed with a twist suitable for the free-vortex flow at entry and an axial flow at outlet. Because of the very high torque that must be transmitted and the large length of the blades, strength considerations impose the need for large blade chords.

As a result, pitch/chord ratios of 1.0 to 1.5 are commonly used by manufacturers and, consequently, the number of blades is small, usually 4, 5 or 6. The Kaplan turbine incorporates one essential feature not found in other turbine rotors and that is the setting of the stagger angle can be controlled.

At part load operation the setting angle of the runner vanes is adjusted automatically by a servo mechanism to maintain optimum efficiency conditions. This adjustment requires a complementary adjustment of the inlet guide vane stagger angle in order to maintain an absolute axial flow at exit from the runner.

Basic equations

Most of the equations presented for the Francis turbine also apply to the Kaplan (or propeller) turbine, apart from the treatment of the runner. Figure 9.17 shows the velocity triangles and part section of a Kaplan turbine drawn for the mid-blade height. At exit from the runner the flow is shown leaving the runner without a whirl velocity, i.e. c 3 D 0 and constant axial velocity.

The theory of free-vortex flows was expounded in Chapter 6 and the main results as they apply to an incompressible fluid are given here. The runner blades will have a fairly high degree of twist, the amount depending upon the strength of the circulation function K and the magnitude of the axial velocity. Just upstream of the runner the flow is assumed to be a free-vortex and the velocity components are accordingly:

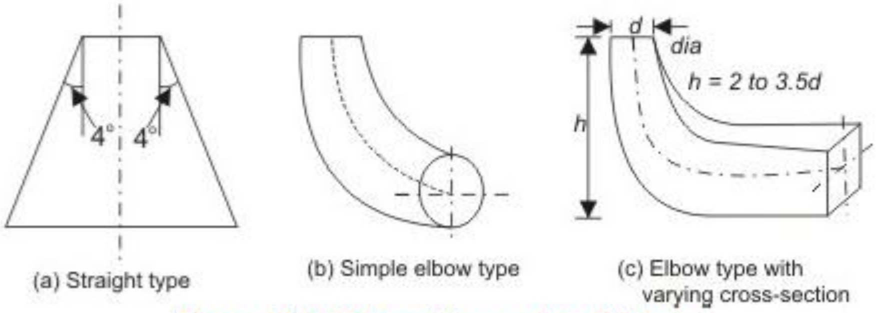


Figure 30.4 Different types of draft tubes

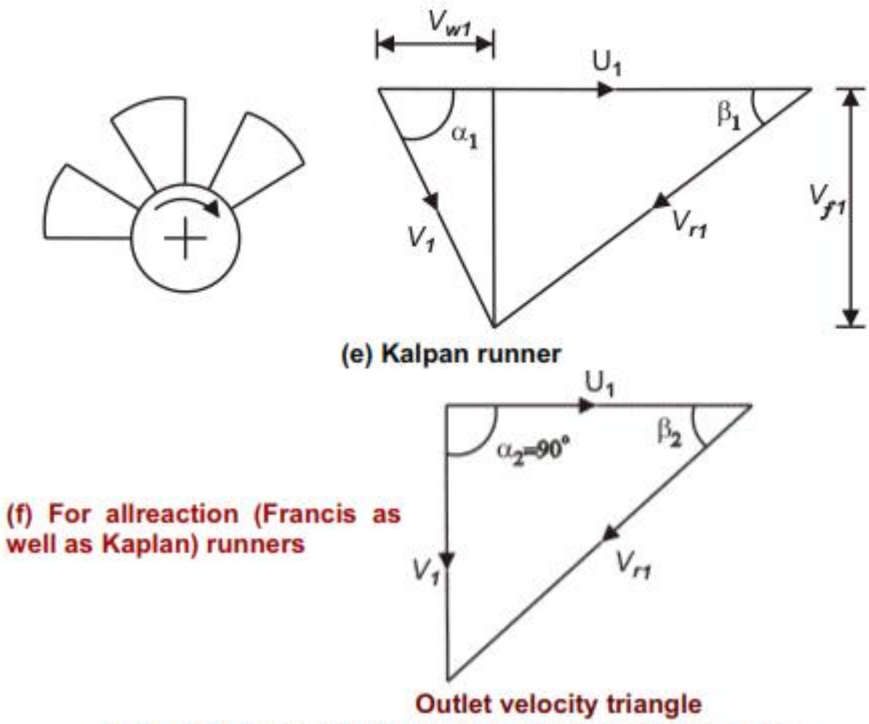
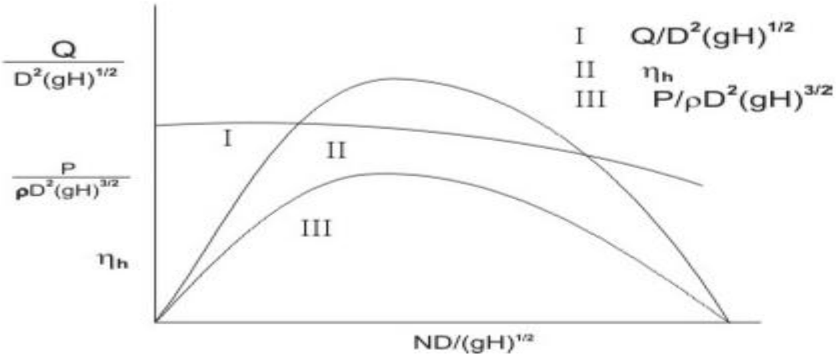


Fig. 30.2 Evolution of Kaplan runner form Francis one

Performance Characteristics of Reaction Turbine

It is not always possible in practice, although desirable, to run a machine at its maximum efficiency due to changes in operating parameters. Therefore, it becomes important to know the performance of the machine under conditions for which the efficiency is less than the maximum. It is more useful to plot the basic dimensionless performance parameters is derived earlier from the similarity principles of fluid machines. Thus one set of curves, as shown in Fig. 31.1, is applicable not just to the conditions of the test, but to any machine in the same homologous series under any altered conditions.



Performance characteristics of a reaction turbine (in dimensionless parameters)

Below figure is one of the typical plots where variation in efficiency of different reaction turbines with the rated power is shown.

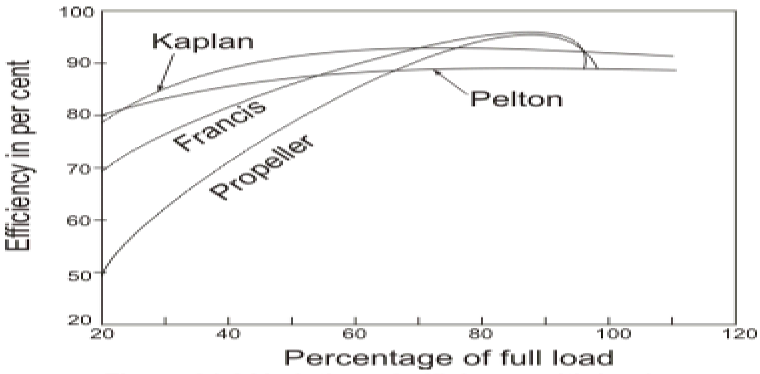


Figure 31.2 Variation of efficiency with load

Characteristic Curves

The turbines are generally designed to work at particular values of H, Q, P, N and η_o which are known as the designed conditions. It is essential to determine exact behaviour of the turbines under the varying conditions by carrying out tests either on the actual turbines or on their small scale models. The results of these tests are usually graphically represented and the resulting curves are known as characteristic curves.

- -constant head characteristic curves
- -constant speed characteristic curves
- -constant efficiency characteristic curves

In order to obtain constant head characteristics curves the tests are performed on the turbine by maintaining a constant head and a constant gate opening and the speed is varied by changing the load on the turbine. A series of values of N are thus obtained and corresponding to each value of N, discharge Q and the output power P are measured. A series of such tests are performed by varying the gate opening, the head being maintained constant at the previous value. From the data of the tests the values of Qu, Pu, nu and η_o are computed for each gate opening. Then with Nu as abscissa the values of Qu, Pu and η_o for each gate opening are plotted. The curves thus obtained for pelton wheel and the reaction turbines for four different gate openings are shown in Fig.

Governing Mechanism of Turbines

Usually hydraulic turbines are coupled to the generators. The power output of the generator is actually the load on the turbine. When the load changes the speed or the turbine also change.

The governing of a turbine is the operation by which the speed of the turbine is kept constant by controlling the discharge irrespective of the fluctuations of load on the turbine.

Functions of governor

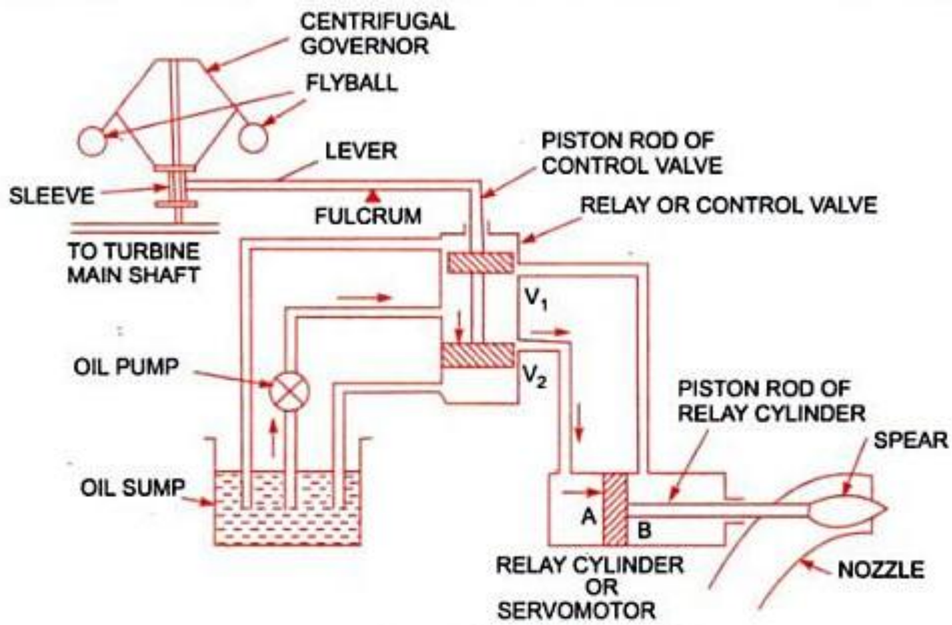
The main functions of a governor are

i) Noticing the speed variations quickly.
ii) Operating the different components rapidly and effectively.
iii) Controlling the discharge to maintain constant speed.
iv) Matching the speed of the turbine and generator.
v) Setting the amount of load to the turbine unit.
vi) To alert the operator under extreme conditions.

Working:

Fig Shows governing mechanism of a pelton wheel. It consists of a spring loaded centrifugal or pendulum. This governor is driven by the turbine main shaft by a belt.

As the speed increases, the balls of the governor moves outward and as speed falls the balls move inwards. The pendulum operates a small control valve and the spear is operated by a servomotor.



Governing of Pelton turbine.

If the load decreases the speed of the turbine will increases. As the speed increases, the pendulum starts moving faster. This causes an upward movement of the sleeve attached to the governor. As the control valve is moved in the upward, direction, high pressure oil is admitted to the left hand side of the servomotor piston. This causes the spear to move deep into the nozzle hole reducing the discharge of water directed towards the vanes. Now the oil from the right hand end of the servomotor cylinder is returned to the oil sump.

When the load on the turbine increases, the main level moves upward. This pulls the piston of the control valve upward. Now oil under pressure enters the right hand side of the servo – motor piston. The piston moves to the left and the spear also moves to the left and there is an increase in flow through nozzle. The increased discharge increases the output and the normal speed is maintained.

Draft Tube:

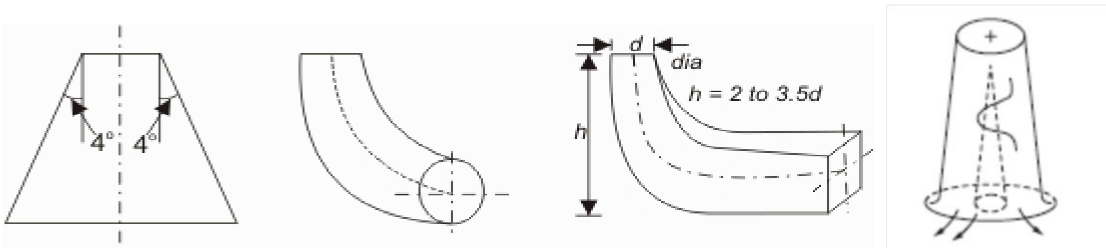
- The draft tube is pipe of gradually increasing area which connects the outlet of the runner to the tail race.
- It is used for discharging water from the exit of the turbine to the tail race.
- This pipe of gradually increasing area is called draft tube.
- One end of the draft tube is connected to the outlet of the runner while the other end is sub-merged below the level of water in the tail race.

Functions of Draft Tube:

- It permits a negative head to be established at the outlet of the runner and thereby increase the net head on the turbine.
- The turbine may be placed above the tail race without any loss of net head and hence turbine may be inspected properly.
- It converts a large proportion of the kinetic energy ($V_2^2/ 2g$) rejected at the outlet of the turbine into useful pressure energy.
- Without the draft tube the kinetic energy rejected at the outlet of the turbine will go waste to the tail race.
- Hence by using draft tube net head on the turbine increases.
- The turbine develops more power and also the efficiency of the turbine increases.
- If the reaction turbine is not fitted with a draft tube, the pressure at the outlet is equal to the atmospheric pressure.

Types of Draft Tube:

- Conical Draft Tubes
- Simple elbow Tubes
- Elbow Draft tubes with circular inlet and rectangular outlet.
- Moody's bell mouthed tube



(a) Straight type (b) Simple elbow type (c) Elbow type with varying cross-section Moody's bell mouthed tube

Draft Tube Theory:

Consider a capital draft-tube as shown in figure

H_s = vertical height of draft tube above the tail race.

Y= distance of bottom of draft tube from tail race.

Applying Bernoulli's equation to inlet (section 1-1) and outlet (section 2-2) of the draft tube and taking section 2-2 as datum line, we get

$$\frac{P_1}{\rho g} + \frac{V_1^2}{2g} + (H_s + y) = \frac{P_2}{\rho g} + \frac{V_2^2}{2g} + 0 + h_f \qquad \text{---------- (1)}$$

Where h_f = loss of energy between sections 1-1 and 2-2.

But $$\frac{P_2}{\rho g} = \text{Atmospheric pressure head} + y$$

$$= \frac{P_a}{\rho g} + y$$

Substituting this value in equation (1), we get

$$\frac{P_1}{\rho g} + \frac{V_1^2}{2g} + (H_s + y) = \frac{P_a}{\rho g} + y + \frac{V_2^2}{2g} + h_f$$

$$\boxed{\frac{P_1}{\rho g} = \frac{P_a}{\rho g} - H_s - \left(\frac{V_1^2}{2g} - \frac{V_2^2}{2g} - h_f\right)}$$

Efficiency of Draft Tube:

The efficiency of draft tube is defined as the ratio of actual conversation of kinetic head into pressure head in the draft tube to the kinetic head at the inlet of the draft tube.

$$\boxed{\eta_d = \frac{\left(\frac{V_1^2}{2g} - \frac{V_2^2}{2g} - h_f\right)}{\frac{V_1^2}{2g}}}$$

1. A water turbine has a velocity of 6 m/s at the entrance to the draft tube and a velocity of 1.2 m/s at the exit. For friction losses of 0.1 m and tail water 5 m below the entrance to the draft-tube, find the pressure head at the entrance.

Given:

Velocity at inlet (V_1) = 6 m/s
Velocity at outlet (V_2) = 1.2 m/s
Friction loss (h_f) = 0.1 m
H_s = 5 m

To find:

Pressure Head at entrance $\left(\frac{P1}{\rho g}\right)$ =?

Solution:

Atmospheric pressure (P_a = 1.0135 bar = 1.0135 x 10^5 N/m^2)

$$\frac{P_a}{\rho g} = 10.3 \text{ m}$$

$$\boxed{\frac{P_1}{\rho g} = \frac{P_a}{\rho g} - H_s - \left(\frac{V_1^2}{2g} - \frac{V_2^2}{2g} - h_f\right)}$$

$$\frac{P_1}{\rho g} = \mathbf{-\ 6.6616\ m}$$

2. A conical draft tube having diameter at the top as 2.0 m and pressure head at 7m of water(vacuum), discharge water at the outlet with a velocity 0f 1.2 m/s at the rate of 25 m^3/s. If atmospheric pressure head is 10.3 m of water and losses between the inlet and outlet of the draft-tubes are negligible, find the length of draft-tube immersed in water. Total length of tube is 5 m.

Given:

Diameter at top, D_1 = 2 m
Pressure head $\left(\frac{P_1}{\rho g}\right)$ = 7m (vacuum) = 10.3 – 7 = 3.3 m (abs)
Velocity at outlet, V_2 =1.2 m/s
Q = 25 m^3/s
H_f = negligible
Total length of the tube = 5 m
$\frac{P_a}{\rho g} = 10.3$ m

To find:

Length of the tube immersed in water (y) =?

Solution:

$Q = A_1V_1$
V_1 = 7.957 m/s

$$\boxed{\frac{P_1}{\rho g} = \frac{P_a}{\rho g} - H_s - \left(\frac{V_1^2}{2g} - \frac{V_2^2}{2g} - h_f\right)}$$

H_s = 3.8464m

Y = total length – H_s = 5 – 3.8464 = **1.1536m**

3. A conical draft tube having inlet and outlet diameter 1 m and 1.5 m discharges water at outlet with a velocity of 2.5 m/s. The total length of the draft tube is 6 m and 1.2 m of the length of the draft tube is immersed in water. If the atmospheric pressure head is 10.3 m of water and loss of head due to friction in the draft tube is equal to 0.2 x velocity head at outlet of the tube, find: (i) pressure head at inlet,(ii) efficiency of the draft tube.

Given:

Diameter at inlet (D_1) = 1.0 m

Diameter at outlet (D_2) = 1.5 m

Velocity at outlet (V_2) = 2.5 m/s

Total length of tube, $H_s + y = 6$ m

y = 1.2 m

$H_s = 4.80$ m

$$\frac{P_a}{\rho g} = 10.3 \text{ m}$$

Loss of head due to friction (h_f) = 0.2 x velocity of head at outlet

$$= 0.2 \times \frac{V_2^2}{2g} = 0.0637 \text{ m}$$

To find:

Pressure Head at entrance $\left(\frac{P1}{\rho g}\right)$ =?

η_d =?

Solution:

$Q = A_2V_2$

$A_2 = \frac{\pi}{4} \times D_2^2$

$Q = 4.4178$ m^3/s

$Q = A_1V_1$

$A_1 = \frac{\pi}{4} \times D_1^2$

$V_1 = 5.625$ m/s

(i) Pressure head at inlet

$$\boxed{\frac{P_1}{\rho g} = \frac{P_a}{\rho g} - H_s - \left(\frac{V_1^2}{2g} - \frac{V_2^2}{2g} - h_f\right)}$$

$$\frac{P_1}{\rho g} = \mathbf{4.27\ m\ (abs)}$$

(ii) Efficiency of draft tube

$$\boxed{\eta_d = \frac{\left(\frac{V_1^2}{2g} - \frac{V_2^2}{2g} - h_f\right)}{\frac{V_1^2}{2g}}}$$

η_d = **76.3 %**

Pelton Wheel

Formulas:

- Work done by the jet per second = $\rho a V_1 [V_{w1} \pm V_{w2}] \times u$
- Runner power = $\dfrac{\rho a V_1 [V_{w1} \pm V_{w2}] \times u}{1000}$
- Water power = $\dfrac{W \times Q \times H}{1000}$
- Kinetic energy = $\frac{1}{2} m V_1^2 \quad = \frac{1}{2}(\rho a V_1) V_1^2 = \frac{1}{2}\rho a V_1^3$
- Hydraulic Efficiency = $\dfrac{\text{work done by jet per second}}{\text{Kinetic energy}} = \dfrac{2 [V_{w1} \pm V_{w2}] \times u}{V_1^2}$
- Velocity of jet at inlet is given by $V_1 = C_v\sqrt{2gH}$, $H = \dfrac{V_1^2}{2g}$

  C_V = Co-efficient of velocity = 0.98 or 0.99

  - Gross head $H_g = h_f + H$
  - Gross Head $H_g = h_f + H$ + head lost in nozzle -------(if nozzle loss consider)
  - Head lost in nozzle = $\dfrac{V_1^2}{2g}\left(\dfrac{1}{C_v^2} - 1\right)$
  - Efficiency of nozzle = $(H_g - h_f)/ H_g$
- Velocity of wheel is given by $(u) = C_u\sqrt{2gH}$,

  C_u = speed ratio = 0.43 to 0.48

  - $u = u_1 = u_2 = \dfrac{\pi D N}{60}$
- **Jet ratio(m)** = D/d

  D = pitch diameter, d = jet diameter
- Number of buckets(Z) = $15 + \dfrac{D}{2d}$
- No of jet = $\dfrac{Q}{q}$

  Q = total discharge, q = discharge by single jet
- Mechanical Efficiency = $\eta_m = \dfrac{\textit{Power at the shaft of the turbine}}{\textit{Power deliverd by the water to the runner}} = \dfrac{S.P}{R.P}$
- Volumetric Efficiency = $\eta_v = \dfrac{\textit{Volume of water actually striking the runner}}{\textit{Volume of water supplied to the turbine}}$
- Overall efficiency = $\eta_o = \dfrac{\textit{Shaft Power}}{\textit{Water Power}} = \dfrac{S.P}{W.P}$

  $\eta_o = \eta_m \times \eta_h$

  - $V_{r1} = V_{r2}$

1. A pelton wheel has a mean bucket speed of 10 meters per second with a jet of water flowing at the rate 700 litres/s under a head of 30 meters. The buckets deflect jet through an angle of 160°. Calculate the power given by water to the runner and the hydraulic efficiency of the turbine. Assume co-efficient of velocity as 0.98.

Given:

$u = u_1 = u_2 = 10$ m/s

$Q = 700$ litres/s $= 0.7\ m^3/s$

$H = 30$ m

$\emptyset = 180° - 160° = 20°$

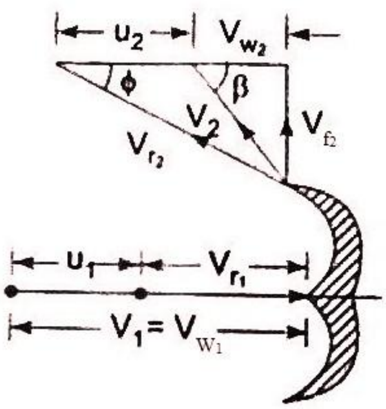
$C_V = 0.98$

To find:

Runner power =?

η_h =?

Solution:



$V_1 = C_V\sqrt{2gH} = 23.77$ m/s

From inlet velocity triangle

$V_{r1} = V_1 - u_1 = 13.77$ m/s

$V_{w1} = V_1 = 23.77$ m/s

From outlet velocity triangle

$V_{r2} = V_{r1} = 13.77$ m/s

$$\cos\emptyset = \frac{V_{w2} + u_2}{V_{r2}}$$

$V_{w2} = 2.94$ m/s

a. Runner Power

$$\boxed{\text{Runner power} = \frac{\rho a V_1\,[V_{w1} \pm V_{w2}]\ X\ u}{1000}}$$

$Q = aV_1 = 0.7\ m^3/s$

R.P = 186.97 kW

b. Hydraulic Efficiency

$$\eta_h = \frac{2\,[V_{w1} \pm V_{w2}]\ X\ u}{V_1^2}$$

$\boldsymbol{\eta_h}$ **= 94.54 %**

2. A Pelton wheel is to be designed for the following specification: shaft power = 11,772; Head = 380 meters; Speed = 750 R.p.m; Overall efficiency = 86 %; Jet diameter is not to exceed one-sixth of the wheel diameter. Determine: (i) the wheel diameter, (ii) the number of jets required, and (iii) diameter of the jet. Take $C_{v1} = 0.985$ and $C_{u1} = 0.45$.

Given:

S.P = 11,772 kW

H = 380 m

N =750 r.p.m

$\eta_o = 0.86$

d = 1/6 D

$C_V = 0.985$

$C_u = 0.45$

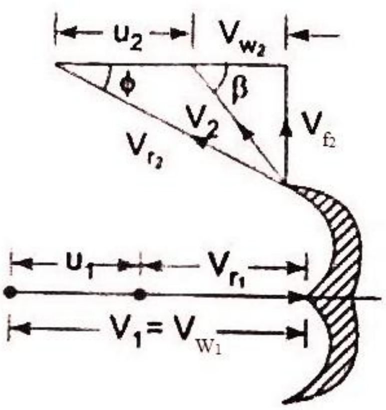
To find:

Wheel Diameter (D) =?

Jet Diameter (d) =?

No of Jets =?

Solution:



$V_1 = C_V\sqrt{2gH}$ = 85.05 m/s

$u = u_1 = u_2 = C_u\sqrt{2gH}$ = 38.85 m/s

a. The wheel Diameter

$$u = u_1 = u_2 = \frac{\pi D N}{60}$$

D = 0.989 m

b. Jet Diameter

d = 1/6 D

d= 0.165 m

c. No of jets

q = Area of jet x velocity of jet = 1.818 m^3/s

$$\eta_o = 0.86 = \frac{S.P}{W.P}$$

W.P = 13688 kW

Water power = $\frac{W \times Q \times H}{1000}$, where,

W = ρg = 1000 x 9.81

Q = 3.672 m^3/s

No of jet = $\frac{Q}{q}$ = **2 jets**

3. The penstocks supply water from a reservoir to the pelton wheel with a gross head of 500m. One third of the gross head is lost in friction in the penstock. The rate of flow of water through the nozzle fitted at the end of the penstock is 2.0m^3/s. The angle of deflection of the jet is 165°. Determine the power given by the water to the runner and also hydraulic efficiency of the pelton wheel. Take speed ratio = 0.45 and C_V = 1.0.

Given:

H_g = 500 m

$h_f = \frac{H_g}{3}$ = 166.7 m

Net head H = $H_g - h_f$ = 333.30 m

Q = 2.0 m^3/s

$\emptyset$ =180° – 165° = 15°

Speed ration C_u =0.45

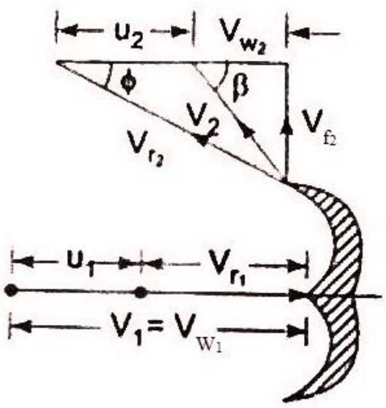
Co-efficient of velocity C_V = 1.0

To find:

Runner power =?

η_h =?

Solution:



$V_1 = C_V\sqrt{2gH}$ = 80.86 m/s

$u = u_1 = u_2 = C_u\sqrt{2gH}$ = 36.387 m/s

From inlet velocity triangle

$V_{r1} = V_1 - u_1$ = 44.473 m/s

$V_{w1} = V_1$ = 80.86 m/s

From outlet velocity triangle

$V_{r2} = V_{r1}$ = 44.473 m/s

$$\cos \emptyset = \frac{V_{w2} + u_2}{V_{r2}}$$

V_{w2} = 6.57 m/s

a. Runner Power

$$\boxed{\text{Runner power} = \frac{\rho a V_1 [V_{w1} \pm V_{w2}] \times u}{1000}}$$

Q = aV_1 = 2.0 m^3/s

R.P = 6362.63 kW

b. Hydraulic Efficiency

$$\eta_h = \frac{2 [V_{w1} \pm V_{w2}] \times u}{V_1^2}$$

η_h **= 97.31 %**

4. A Pelton wheel is having a mean bucket diameter of 1 m and is running at 1000 r.p.m. The net head on the Pelton wheel is 700 m. If the side clearance angle is 15° and discharge through nozzle is 0.1m^3/s, find: (i) power available at the nozzle, and (ii) Hydraulic efficiency of the turbine.

Given:

D = 1.0 m

N = 1000 r.p.m

H = 700 m

$\emptyset$ = 15°

Q = 0.1 m^3/s

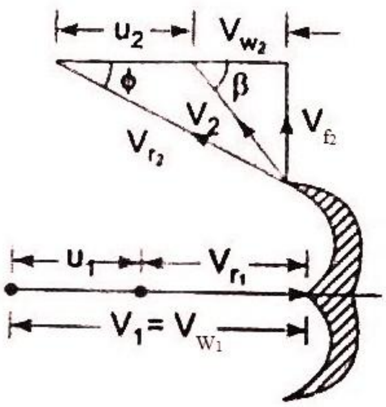
C_V = 1 ------ (not given)

To find:

Power available at nozzle =?

η_h =?

Solution:



$V_1 = C_V\sqrt{2gH}$ = 117.19 m/s

$u = u_1 = u_2 = \frac{\pi D N}{60} = 52.36$ m/s

From inlet velocity triangle

$V_{r1} = V_1 - u_1$ = 64.83 m/s

$V_{w1} = V_1$ = 117.19 m/s

From outlet velocity triangle

$V_{r2} = V_{r1}$ = 64.83 m/s

$\cos \emptyset = \frac{V_{w2} + u_2}{V_{r2}}$

V_{w2} = 10.26 m/s

a. **Power available at nozzle**

Water power = $\frac{W \times Q \times H}{1000}$, where,

W = ρg = 1000 x 9.81

W.P = **686.7 kW**

b. **Hydraulic Efficiency**

$$\eta_h = \frac{2\,[V_{w1} \pm V_{w2}]\ X\ u}{V_1^2}$$

$\boldsymbol{\eta_h}$ **= 98.18 %**

5. A Pelton wheel is working under a gross head of 400 m. The water is supplied through penstock of diameter 1 m and length 4 km from reservoir to the Pelton wheel. The co-efficient of friction for the penstock is given as 0.008. The jet of water of diameter 150mm strikes the buckets of the wheel and gets deflected through an angle of 165°. The relative velocity of the water at outlet is reduced by 15% due to friction between inside surface of the bucket and water. If the velocity of the buckets is 0.45 times the jet velocity at inlet and mechanical efficiency as 85 % determine: (i) power given to the runner, (ii) shaft power (iii) hydraulic efficiency and overall efficiency.

Given:

H_g = 400 m
D = 1.0 m
L = 4000 m
f = 0.008
d = 0.15 m
$\emptyset$ =180° – 165° = 15°
$V_{r2} = 0.85\ V_{r1}$
u = 0.45 x jet velocity
η_m = 0.85

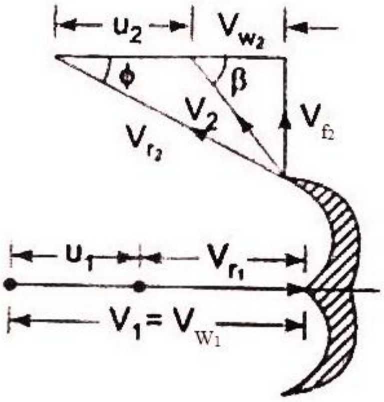
To find:

Runner power =?

Shaft power =?

η_h =?

Solution:



Let, V^* = velocity of penstock

V_1 = Velocity of jet of water

Using continuity equation

Area of penstock x V^* = Area of jet x V_1

$$\frac{\pi}{4} \times D^2 \times V^* = \frac{\pi}{4} \times d^2 \times V_1$$

$V^* = 0.0225V_1$ -------- (1)

Gross head $H_g = h_f + H$

$$400 = \frac{4fLV^{*2}}{2gD} + \frac{V_1^2}{2g} \text{ -------- (2)}$$

Sub (1) in (2), we get

$\mathbf{V_1 = 85.83\ m/s}$

u = 0.45 x jet velocity = 38.62 m/s

From inlet velocity triangle

$V_{r1} = V_1 - u_1 = 47.21$ m/s

$V_{w1} = V_1 = 85.83$ m/s

From outlet velocity triangle

$V_{r2} = 0.85V_{r1} = 40.13$ m/s

$$\cos \emptyset = \frac{V_{w2} + u_2}{V_{r2}}$$

$V_{w2} = 0.143$ m/s

$a = \frac{\pi}{4} \times d^2 = 0.01767\ m^2$

a. Runner Power

$$\boxed{\text{Runner power} = \frac{\rho a V_1 [V_{w1} \pm V_{w2}] \text{ X } u}{1000}}$$

R.P = 5033.54 kW

b. Shaft Power

$$\eta_m = \frac{S.P}{R.P}$$

Shaft power = 4278.5 kW

c. Hydraulic Efficiency

$$\eta_h = \frac{2\ [V_{w1} \pm V_{w2}] \text{ X } u}{V_1^2}$$

$\boldsymbol{\eta_h}$ **= 90.14 %**

6. A Pelton wheel nozzle, for which $C_V = 0.97$, is 400m below the water surface of a lake. The jet diameter is 80 mm, the pipe diameter is 0.6 m, its length is 4 km and f = 0.032 in the formula $\mathbf{h_f = fLV^2/ 2g \times d}$. The buckets, deflects the jet through 165° and they run at 0.48 times the jet speed, bucket friction reducing the relative velocity at outlet by 15% of relative velocity at inlet. Mechanical efficiency = 90%. Find the flow rate and the shaft power developed by the turbine.

Given:

$C_V = 0.97$

$H_g = 400$ m

d = 0.08 m

D = 0.6 m

L = 4000 m

f = 0.032

$\emptyset = 180° - 165° = 15°$

u = 0.48 x jet speed

$V_{r2} = 0.85V_{r1}$

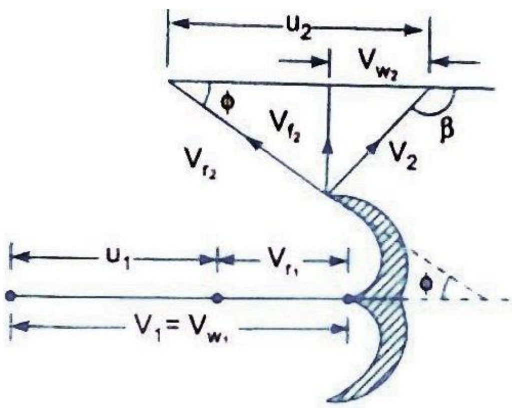
$\eta_m = 0.90$

To find:

Flow rate (Q) =?

Shaft power =?

Solution:



Let, V^* = velocity of penstock

V_1 = Velocity of jet of water

Using continuity equation

Area of penstock x V^* = Area of jet x V_1

$$\frac{\pi}{4} \times D^2 \times V^* = \frac{\pi}{4} \times d^2 \times V_1$$

$V^* = 0.0177V_1$ -------- (1)

Gross Head **(H_g) = h_f + H + head lost in nozzle**

$$400 = \frac{fLV^{*2}}{2gD} + \frac{V_1^2}{2g} + \frac{V_1^2}{2g}\left(\frac{1}{C_v^2} - 1\right) \text{ --------- (2)}$$

Sub (1) in (2), we get

V_1 = 83.47 m/s

u = 0.48 x jet velocity = 40.06 m/s

From inlet velocity triangle

$V_{r1} = V_1 - u_1 = 43.41$ m/s

$V_{w1} = V_1 = 83.47$ m/s

From outlet velocity triangle

$V_{r2} = 0.85V_{r1} = 36.898$ m/s

$$\cos \emptyset = \frac{V_{w2} + u_2}{V_{r2}}$$

V_{w2} = - 4.42 m/s

$a = \frac{\pi}{4} \times d^2 = 0.0050265 \text{ m}^2$

a. **Flow rate**

Q =a V_1 = 0.419 m^3/s

b. **Shaft Power:**

$$\eta_m = \frac{S.P}{R.P}$$

$$\boxed{\text{Runner power} = \frac{\rho a V_1 [V_{w1} \pm V_{w2}] \text{ X u}}{1000}}$$

R.P = 1326.865 kW

S.P = 1326.865 x 0.90 = 1194.18 kW

7. A 137 mm diameter jet of water issuing from a nozzle impinges on the buckets. The head available at the nozzle is 400m. Assuming C_V = 0.97, speed ratio as 0.46, and reduction in relative velocity while passing through buckets as 15%, find: (i) the force exerted by the jet on buckets in tangential direction, (ii) the power developed.

Given:

d = 0.137 m

$\emptyset$ =180° – 165° = 15°

H = 400 m

C_V = 0.97

C_u = 0.46

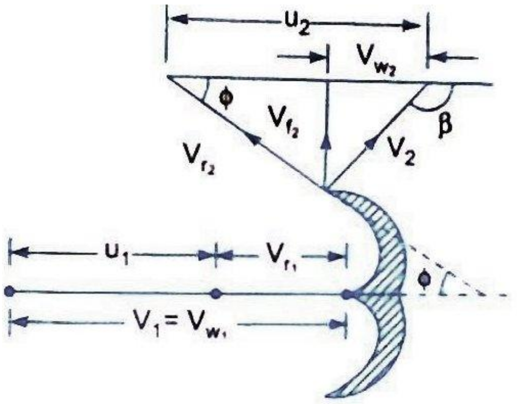
$V_{r2} = 0.85\ V_{r1}$

To find:

Force Exerted =?

Runner power =?

Solution:



$V_1 = C_V\sqrt{2gH} = 85.93$ m/s

$u = u_1 = u_2 = C_u\sqrt{2gH} = 40.75$ m/s

From inlet velocity triangle

$V_{r1} = V_1 - u_1 = 45.18$ m/s

$V_{w1} = V_1 = 85.93$ m/s

From outlet velocity triangle

$V_{r2} = 0.85V_{r1} = 38.40$ m/s

$$\cos \emptyset = \frac{V_{w2} + u_2}{V_{r2}}$$

V_{w2} = -3.658 m/s

a. **Force exerted**

$F_X = \rho a V_1 [V_{w1} \pm V_{w2}]$

= **104206 N**

b. **Runner Power**

$$\boxed{\text{Runner power} = \frac{\rho a V_1 [V_{w1} \pm V_{w2}] \text{ X u}}{1000}}$$

R.P = 4246.4 kW

8. Two jets strike the buckets of a Pelton wheel, which is having shaft power at 15450 kW. The diameter of each jet is given as 200 mm. If the net head on the turbine is 400 m. find the overall efficiency of the turbine. Take $C_V = 1.0$.

Given:

No of jets = 2

S.P = 15450 kW

d = 0.20 m

H = 400 m

$C_V = 1.0$

To Find:

Overall efficiency = η_o =?

Solution:

$V_1 = C_V\sqrt{2gH} = 88.58$ m/s

Discharge of each jet (q) = a x V_1 = 2.78 m^3/s

Total Discharge (Q) = 2 x q = 5.56 m^3/s

Water power = $\frac{\text{W x Q x H}}{1000}$, where,

W = ρg = 1000 x 9.81

W.P = 21817.44 **kW**

$\eta_o = \frac{S.P}{W.P}$ = **70.8 %**

9. The water available for a Pelton wheel is 4 cumec and the total head from the reservoir to the nozzle is 250m. The turbine has two runners with two jets per runner. All the four jets have the same diameters. The pipe line is 3000 m long. The efficiency of power transmission through the pipe line and the nozzle is 91% and efficiency of each runner is 90%. The velocity co-efficient of each nozzle is 0.975 and co-efficient of friction '4f' for the pipe is 0.0045. Determine: (i) The power developed by the turbine (ii) The diameter of the jet, and (iii) The diameter of the pipe line.

Given:

Q = a V_1 = 4 cumec = 4 m^3/s

H_g = 250 m

No of jets = 2 x 2 = 4

L =3000 m

Efficiency of pipeline or nozzle = 0.91

Efficiency of runner or η_h = 0.90

C_V= 0.975

4f = 0.0045

To Find:

Power developed =?

D =?

d =?

Solution:

Efficiency of nozzle = $(H_g - h_f)/ H_g$

h_f = 22.5 m

Net Head H = $H_g - h_f$ = 227.5 m

$V_1 = C_V\sqrt{2gH} = 65.14$ m/s

a. **Power developed**

Hydraulic Efficiency = $\frac{\text{work done by jet per second}}{\text{Kinetic energy}}$

Kinetic energy = $\frac{1}{2}mV_1^2 = \frac{1}{2}(\rho a V_1)V_1^2$

= 8486.44 x 10^3 Nm

W.D by the jet = 0.90 x K.E = 7637.8 x 10^3 Nm

Power developed = W.D/ 1000 = **7637 kW**

b. **Diameter of the jet**

No of jets = Q/q

q = 1.0 m^3/s

q= $\frac{\pi}{4}$ x d^2 x V_1

d = 0.14 m

c. **Diameter of pipe line**

Q = A x V^*

4 = $\frac{\pi}{4}$ x D^2 x V^*

$V^* = 5.09/D^2$ ------- (1)

$h_f = \frac{4fLV^{*2}}{2gD}$

$22.5 = \frac{0.0045 \times 3000 \times (5.09/D^2)^2}{2 \times 9.81 \times D}$

$22.5 = \frac{0.0045 \times 3000 \times 5.09}{2 \times 9.81 \times D^5}$

$D^5 = 0.7933$

D = $(0.7933)^{1/5}$ = **0.995 m**

10. The following data is related to Pelton wheel: Head at the base of the nozzle = 80 m; Diameter of the jet = 100 mm; Discharge of the nozzle – 0.30 m^3/s; Power at the shaft = 206 kW; Power absorbed in mechanical resistance = 4.5 kW; Determine (i) Power lost in nozzle and (ii) Power lost due to hydraulic resistance in the runner.

Given:

H = 80 m

d = 0.1 m

Q = 0.30 m^3/s

S.P = 206 kW

Power absorbed in mechanical resistance = 5.5 kW

To Find:

Power lost in nozzle =?

Power lost due to hydraulic resistance =?

Solution:

$Q = a\,V_1$

$V_1 = 38.197$ m/s

Power at the nozzle

Water power = $\frac{\text{W x Q x H}}{1000}$, where,

W = ρg = 1000 x 9.81

W.P **= 235.44 kW**

Power at the jet = K.E/ 1000

Kinetic energy = $\frac{1}{2} m V_1^2 \quad = \frac{1}{2}(\rho a V_1) V_1^2$

Power at the jet **= 218.85 kW**

a. **Power lost in nozzle**

Power at the Nozzle = Power at the jet + Power lost in nozzle

Power lost in nozzle = **16.59 kW**

b. **Power lost due to hydraulic resistance:**

Power at the jet = Power at the shaft + Power absorbed in mechanical resistance + Power lost in hydraulic resistance

Power lost in hydraulic resistance = **8.35 kW**

Design of Pelton Wheel:

Design of Pelton wheel means the following data is to be determined

- Diameter of the jet(d)
- Diameter of the wheel(D)
- Width of the buckets which is = 5 x d
- Depth of the buckets which is = 1.2 x d
- Number of buckets on the wheel

Size of the buckets means the width and the depth of the buckets

11. A Pelton wheel is to be designed for a head of 60m when running at 200 r.p.m. The Pelton wheel develops 95.6475 kW shaft Power. The velocity of the buckets = 0.45 times the velocity of the jet, overall efficiency = 0.85 and co-efficient of the velocity is equal to 0.98.

Given

H =60 m

N = 200 r.p.m

S.P = 95.6475 kW

u = 0.45 x jet velocity

$\eta_o = 0.85$

$C_V = 0.98$

To Find

Diameter of the jet (d) =?
Diameter of the wheel (D) =?
Width of the buckets which is =?
Depth of the buckets which is =?
Number of buckets on the wheel =?

Solution:

a. **Diameter of jet (d) :**

$\eta_o = \frac{S.P}{W.P} = 0.85$

W.P = 81.3 kW

Water power = $\frac{W \times Q \times H}{1000}$, where,

$W = \rho g = 1000 \times 9.81$

$Q = 0.1912\ m^3/s$

$V_1 = C_V\sqrt{2gH} = 33.62$ m/s

$Q = A\ V_1$

$Q = \frac{\pi}{4} \times d^2 \times V_1$

d = 0.085 m

b. **Diameter of wheel**:

$u_1 = 0.45 \times V_1 = 15.13$ m/s

$u = u_1 = u_2 = \frac{\pi D N}{60}$

D = 1.44 m

c. **Width of the buckets:**

Width of the buckets which is = 5 x d = **0.425 m**

d. **Depth of the buckets:**

Depth of the buckets which is = 1.2 x d = **0.102 m**

e. **No of buckets:**

$(Z) = 15 + \frac{D}{2d}$ = **24**

---

12. The three Pelton turbine is required to generate 10,000 kW under a net head of 400 m. the blade angle at outlet is 15° and the reduction in the relative velocity while passing over the blade is 5%, If the overall efficiency of the wheel is 80 %, $C_v = 0.98$ and speed ratio = 0.46, then find: (i) the diameter of the jet, (ii) total flow in m^3/s and (iii) the force exerted by a jet on the buckets. If the jet ratio is not less than 10, find the speed of the wheel for a frequency of 50 hertz/sec and the corresponding wheel diameter.

Given:

No of jets = 3

S.P = 10000 kW

H = 400 m

$\emptyset = 15°$

$V_{r2} = 0.95\ V_{r1}$

$\eta_o = 0.80$

$C_V = 0.98$

$C_u = 0.46$

f = 50 hertz/sec

$\frac{D}{d} = 10$

To Find:

d =?

Q =?

F_X =?

N =?

D =?

Solution:

a. **Discharge:**

$\eta_o = \frac{S.P}{W.P} = 0.80$

W.P = 8000 kW

Water power = $\frac{W \times Q \times H}{1000}$, where,

$W = \rho g = 1000 \times 9.81$

$\mathbf{Q = 3.18\ m^3/s}$

b. **Diameter of jet:**

$V_1 = C_V\sqrt{2gH} = 87$ m/s

$Q = A\ V_1$

$Q = \frac{\pi}{4} \times d^2 \times V_1$

d = 0.125 m

c. **Force exerted by a jet on the wheel**

$u_1 = C_u\sqrt{2gH} = 40.75$ m/s

From inlet velocity triangle

$V_{r1} = V_1 - u_1 = 46.25$ m/s

$V_{w1} = V_1 = 87$ m/s

From outlet velocity triangle

$V_{r2} = 0.95V_{r1} = 44$ m/s

$\cos \emptyset = \frac{V_{w2} + u_2}{V_{r2}}$

$V_{w2} = 1.75$ m/s

$F_X = \rho a V_1\ [V_{w1} \pm V_{w2}]$

= **94.075 KN**

d. **Wheel diameter and speed**

$\frac{D}{d} = 10$

D = 1.25 m

$u = u_1 = u_2 = \frac{\pi D N}{60}$

N = 620 r.p.m

Now using the relation

N = (60 x f) / p

p = 4.85

Take the next whole no, p = 5

N = (60 x f) / p

N = 600 r.p.m

$u = u_1 = u_2 = \frac{\pi D N}{60}$

D = 1.3 m

UNIT III

CENTRIFUGAL PUMP

Introduction

Centrifugal pumps are classified as rotodynamic type of pumps in which dynamic pressure is developed which enables the lifting of liquids from a lower to a higher level. The basic principle on which a centrifugal works is that when a certain mass of liquid is made to rotate by an external force, it is thrown away from the central axis of rotation and a centrifugal head is impressed which enable it to rise to a higher level. Now, if more liquid is constantly made available at the centre of rotation, a continuous supply of liquid at a higher level may be ensured. Since in these pumps the lifting of the liquid is due to centrifugal action, these pumps are called 'centrifugal pumps'

Advantages of centrifugal pumps over reciprocating pumps

The main advantage of a centrifugal pump is that its discharging capacity is very much greater than a reciprocating pump which can handle relatively small quantity of liquid only. A centrifugal pump can be operated at very high speeds without any danger of separation and cavitation. The maintenance cost of a centrifugal pump is low and only periodical check up is sufficient. But for a reciprocating pump the maintenance cost is high because the parts such as valves etc., may need frequent replacement.

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 vortex flow which means that when a certain mass of liquid is rotated by an external torque, the rise is pressure head of the rotation 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

(i.e., rise in pressure head = $v^2/2g$ or $\omega^2 r^2 / 2g$)

This at the outlet of the impeller where radius is more, the rise in pressure head will be more and the liquid will be discharged at the outlet with a high pressure head. Due to this high pressure head, the liquid can be lifted to a high level.

Classification of Centrifugal Pumps

a) Single stage
b) Multi stage

Component Parts of a Centrifugal Pump

The main component parts of a centrifugal pump are:

- impeller
- casing
- suction pipe
- delivery pipe

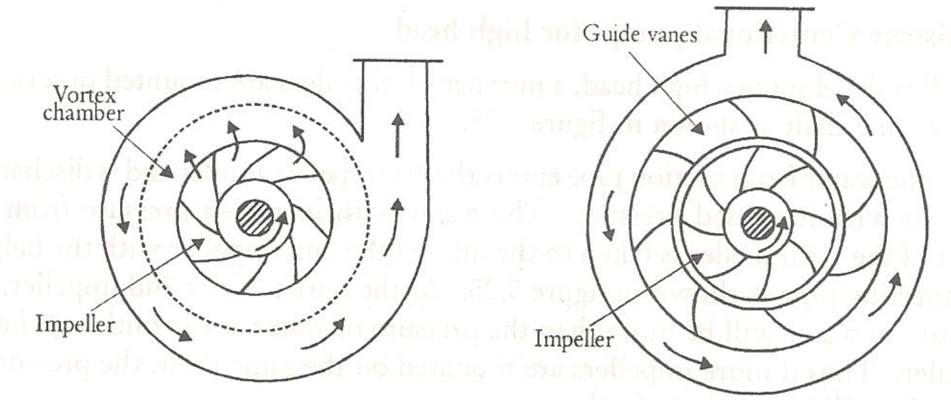
1. **Impeller**

The rotating part of a centrifugal pump is called ‘impeller’. It consists of a series of backward curved vanes. The impeller is mounted on a shaft which is connected to the shaft of an electric motor.

2. **Casing**

The casing of a centrifugal pump is similar to the casing of a reaction turbine. It is an air – tight passage surrounding the impeller and is designed in such a way that the 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.



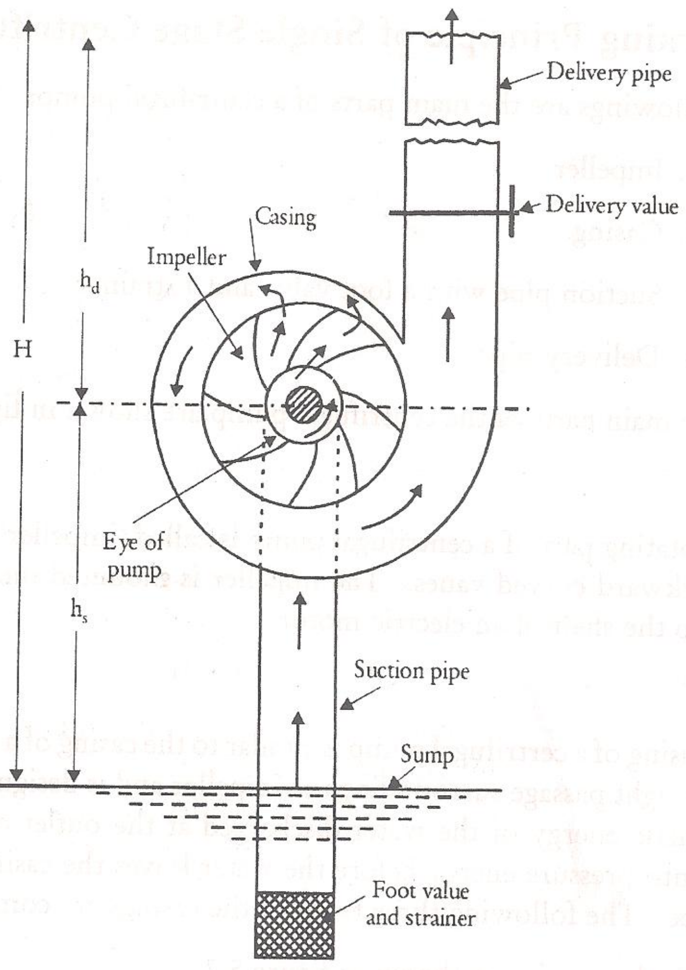
(a) Vortex casing (b) Casing with guide blades

a) Volute casing

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, 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 is 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 delivers the water at a required height is known as delivery pipe.

Work done by the Impeller

The expression of the work done by the impeller of a centrifugal pump on the liquid flowing through it may be derived in the same way as for a turbine. The liquid enters the impeller at its centre and leaves at its periphery. Fig.6.2 shows a portion of the impeller of a centrifugal pump with one vane and the velocity triangles at the inlet and outlet tips of the vane. V is absolute velocity of liquid, u is tangential velocity of the impeller, V_r is relative velocity of liquid, Vf is velocity of flow of liquid, and V_w is velocity of whirl of the liquid at the entrance to the impeller. Similarly V_1, u_1, V_{r1}, V_{f1} and V_{w1} represent their counterparts at the exit point of the impeller.

Velocity triangle of an impeller vane

θ = the impeller vane angle at the entrance

ϕ = the impeller vane angle at the outlet

α = the angle between the directions of the absolute velocity of entering liquid and the peripheral velocity of the impeller at the entrance

β = the angle between the absolute velocity of leaving liquid and the peripheral velocity of the impeller at the exit point

Work done per second by the impeller on the liquid may be written as

$$\text{Work done} = \frac{W\ (V_{w1}\, u_1 - V_w\, u)}{g}$$

Where W kg of liquid per second passes through the impeller since the liquid enters the impeller radially $\alpha = 90$ and hence $V_w = 0$. Thus equation (6.1) becomes

$$\text{Work done} = \frac{W\ (V_{w1} u_1)}{g}$$

<u>**Performance of Pumps- Characteristic Curves**</u>

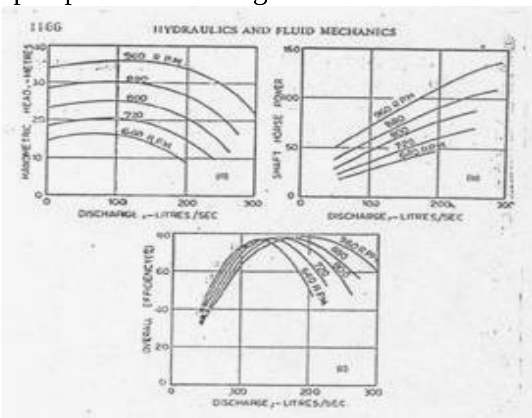
A pump is usually designed for one speed, flow rate and head in actual practice, the operation may be at some other condition of head on flow rate, and for the changed conditions, the behaviour of the pump may be quite different. Therefore, in order to predict the behaviour and performance of a pump under varying conditions, tests are performed and the results of the tests are plotted. The curves thus obtained are known as the characteristic curves of the pump. The following three types of characteristic curves are usually prepared for the centrifugal pumps:

(a) Main and operating characteristics.
(b) Constant efficiency or Muschel curves.
(c) Constant head and constant discharge curves.

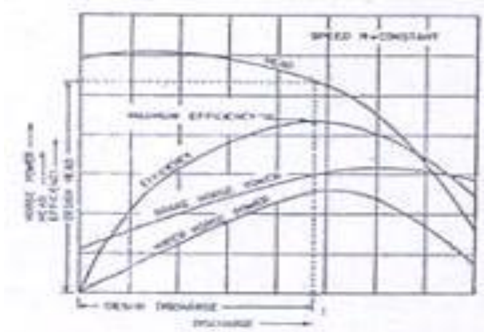
Main and Operating Characteristics

In order to obtain the main characteristic curves of a pump it is operated at different speeds. For each speed the rate of flow Q is varied by means of a delivery valve and for the different values of Q the corresponding values of manometric head H_m, shaft H.P., P , and overall efficiency η are measured or calculated. The same operation is repeated for different speeds of the pump. Then Q v/s H_m ; Q v/s P and Q v/s η curves for different speeds are plotted, so that three sets of curves, as shown in Fig.6.4 are obtained, which represent the *main characteristics* of a pump. The main characteristics are useful in indicating the performance of a pump at different speeds.

During operation a pump is normally required to run at a constant speed, which is its designed speed, (same as the speed of the driving motor). As such that particular set of main characteristics which corresponds to the designed speed is mostly used in the operations of a pump and is, therefore, known as the *operating characteristics*. A typical set of such characteristics of a pump is shown in Fig.



Main characteristics of centrifugal pump



Operating characteristics of centrifugal pump

<u>Reciprocating Pump</u>

The pump is 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.

Classification of Reciprocation 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 ii) Double – acting pump.

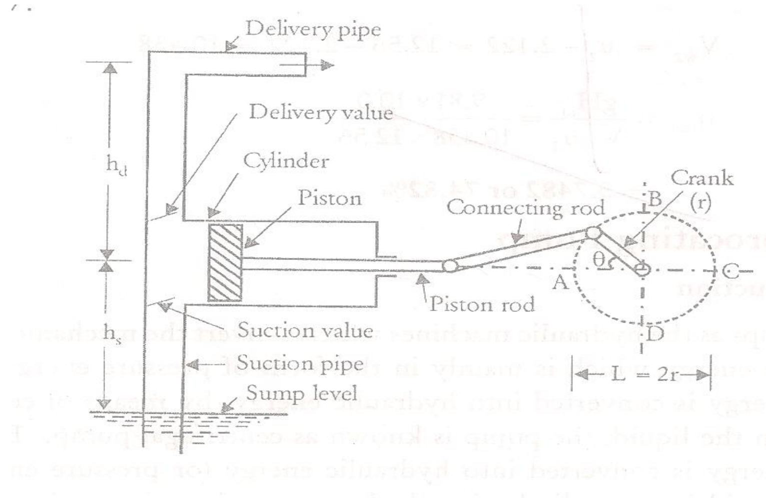
According to the number of cylinder provided, the pumps are classified as:

i) Single cylinder pump ii) Double cylinder pump

iii) Triple cylinder pump

Working Principle of a Reciprocating Pump

The following are the 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
5. Delivery valve

Shows a single action reciprocation pump, which consists of a piston which moves forwards and backwards in a close fitting cylinder. The movement of the piston is connecting the piston road 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 crank starts rotating, the piston moves top and pro 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 θ = 0 to θ = 180 degree), 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 if 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 θ = 180 degree to θ = 360 degree), 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 closes and delivery valve opens. The liquid is forced into the delivery pipe and is raised to a required height.

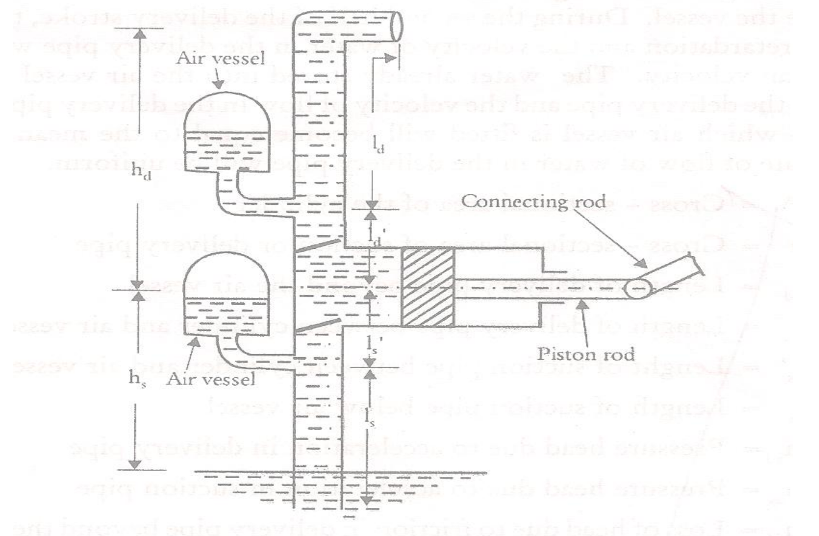
Air vessels

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

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

i) To obtain a continuous supply of liquid at a uniform rate
ii) To save a considerable amount of work in overcoming the frictional Resistance in the suction and delivery pipes

iii) To run the pump at a high speed without separation.



The single acting reciprocating to which air vessels are fitted to the suction and delivery pipes. The air vessels act like an intermediate reservoir. During the first half of the suction stroke, the piston moves with acceleration, which means the velocity of water in the suction pipe is more than the mean velocity and hence the discharge of water entering the cylinder will be more than the mean discharge. This excess quantity of water will be supplied from the air vessel to the cylinder in such a way that the velocity in the suction pipe below the air vessel is equal to mean velocity of flow. During the second half of the suction pipe is less than the mean velocity of flow. Thus the discharge entering the cylinder will be less than the mean discharge. The velocity of water in the suction pipe due to air vessel is equal to mean velocity of flow and discharge required in cylinder is less than the mean discharge. Thus the excess water flowing in suction pipe will be stored in to air vessel, which will be supplied during the first half of the next suction stroke.

When the air vessel if fitted to the delivery pipe, during the first half of delivery stroke the piston moves with acceleration and forces the water into the delivery pipe with a velocity more than the mean velocity. The quantity of water in excess of the mean discharge will flow into the air vessel. This will compress the air inside the vessel. During the second half of the delivery stroke, the piston moves with retardation and the velocity of water in the delivery pipe will be less than the mean velocity. The water already stored into the air vessel will start flowing into the delivery pipe and the velocity of flow in the delivery pipe beyond the point to which air vessel is fitted will become equal to the mean velocity. Hence the rate of flow of water in the delivery pipe will be uniform.

Ideal and Actual Indicator Diagram

The indicator diagram for a reciprocating pump is defined as the graph between the pressure head in the cylinder and the distance traveled by piston from inner dead centre for one complete revolution of the crank. As the maximum distance traveled 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

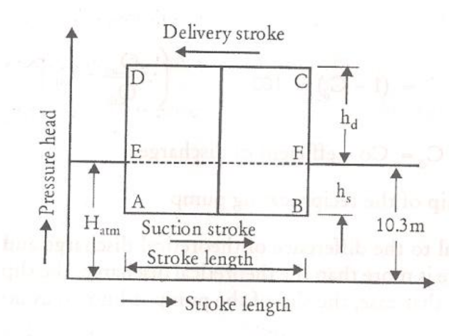
The graph between pressure head in the cylinder and stroke length of the piston for one complete revolution of the crank under ideal conditions is known as ideal indicator diagram. Shows the ideal indicator diagram, in which line EF represents the atmospheric pressure head equal to 10.3 of water.

Let H_{atm} = 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 (h_s), which is below the atmospheric pressure head (H_{atm}) by a height of h_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 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 the crank, the pressure head in the cylinder is represented by the diagram A – B – C – D.

Now, we know that the done by the pump per second

$$= p \times g \times ALN / 60 \times (h_s + h_d)$$

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

[where K = pgAN / 60 = Constant]

$$=\infty\; L \times (h_s + h_d) \qquad \ldots \text{(i)}$$

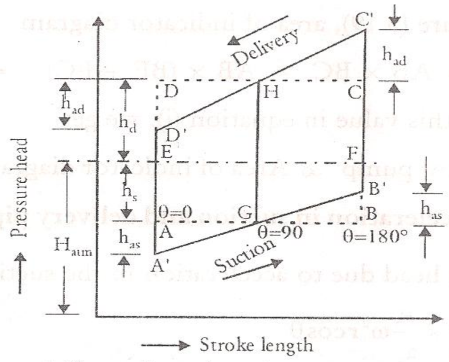
But from figure, area of indicator diagram

$$= AB \times BC = AB \times (BF + FC) \quad = L \times (h_s + h_d)$$

Substituting this value in equation (i), we get

Work done by pump ∞ Area of indicator diagram

Effect of acceleration in suction and delivery pipes on indicator diagram



The pressure head due to acceleration in the suction pipe is given by

$h_{as} = 1_s / g \times A / a_s\ \omega^2 r\cos\theta$

When $\theta = 0°$. $\cos\theta = 1$, and $h_{as} = 1_s / g \times A / a_s\ \omega^2 r$

When $\theta = 90°$, $\cos\theta = 0$, and $h_{as} = 0$

When $\theta = 180°$, $\cos\theta = -1$ and $h_{as} = - 1_s / g \times A / a_s\ \omega^2 r$

Thus, the pressure head inside the cylinder during suction stroke will not be equal to 'h_s' as was the case for ideal indicator diagram, but it will be equal to the sum of 'h_s' and 'h_{as}'. As the beginning of suction stroke $\theta = 0°$, 'h_{as}' is + ve and hence the pressure head in the cylinder will be $(h_s + h_d)$ below the atmospheric pressure head. At the middle of suction stroke $\theta = 90°$ and $h_{as} = 0$ and hence pressure head in the cylinder will be h_s below the atmospheric pressure head. All the end of suction stroke, $\theta = 180°$ and h_{as} is –ve and hence the pressure head in the cylinder will be $(h_s - h_d)$ below the atmospheric pressure head. For suction stroke, the indicator diagram will be shown by A 'GB'. Also the area of A' AG = Area of BGB'.

Similarly, the indicator diagram for the delivery stroke can be drawn. At the beginning of delivery stroke, h_{ad} is + ve and hence the pressure head in the cylinder will be $(h_d + h_{ad})$ above the atmospheric pressure head. At the middle of the delivery stroke, $h_{ad} = 0$ and hence pressure head in the cylinder is equal h_d above the atmospheric pressure head. At the end of the delivery stroke, h_d is –ve and hence pressure in the cylinder will be $(h_d + h_{ad})$ above the atmospheric pressure head. And thus the indicator diagram for delivery stroke is represented by the line C'HD'. Also the are of CC'H = Area of DD'H.

From figure, it is now clear that due to acceleration is suction and delivery pipe, the indicator diagram has changed from ABCD to A'B'C'D'. But the area of indicator diagram ABCD = Area A'B'C'D'. Now from equation (s.22), work done, by pump is proportional to the area of indicator diagram. Hence the work had done by the pump on the water remains same.

CENTRIFUGAL PUMP

1. The internal and external diameters of the centrifugal pump are 200mm and 400mm respectively. The pump is running at 1200 r.p.m. The vane angles of the impeller at inlet and outlet are 20° and 30° respectively. The water enters the impeller radially and velocity of flow is constant. Determine the work done by the impeller per unit weight of water.

Given:

$D_1 = 0.2$ m

$D_2 = 0.4$ m

N = 1200 r.p.m

$\theta = 20°$

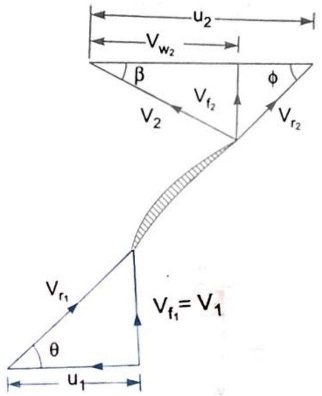
$\emptyset = 30°$

$V_{f1} = V_{f2}$

To Find:

W.D per unit weight =?

Solution:



$$u_1 = \frac{\pi D_1 N}{60} = 12.56 \text{ m/s}$$

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

From Inlet Velocity triangle

$$\tan\theta = \frac{V_{f1}}{u_1}$$

$V_{f1} = 4.57$ m/s

From outlet Velocity triangle

$V_{f1} = V_{f2} = 4.57$ m/s

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

$V_{w2} = 17.215$ m/s

Work done per unit weight:

$$W.D = \frac{V_{w2} \times u_2}{g}$$

W.D = **44.1 Nm/N**

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

Given:

$Q = 0.118 \text{ m}^3/\text{s}$

N = 1450 r.p.m

$H_m = 25$ m

$D_2 = 0.25$ m

$B_2 = 0.05$ m

$\eta_{man} = 0.75$

To find:

$\emptyset$ =?

Solution:

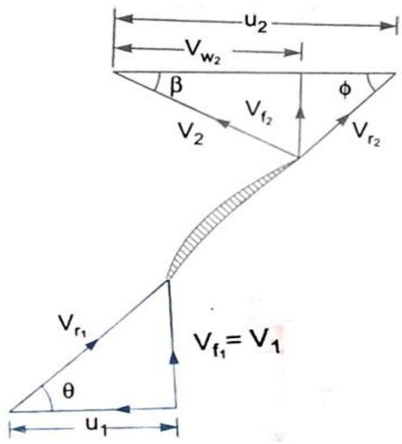
$$u_2 = \frac{\pi D_2 N}{60} = 18.98 \text{ m/s}$$

$$\eta_{man} = \frac{g \times H_m}{V_{w2} \times u_2}$$

$V_{w2} = 17.23$ m/s

$Q = \pi D_2 B_2 V_{f2}$

$V_{f2} = 3.0$ m/s



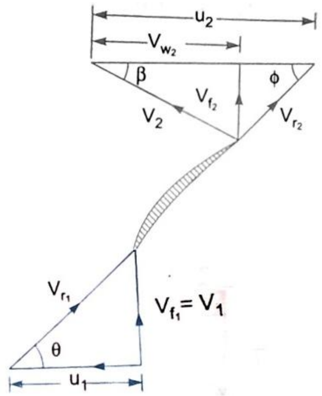
$$\tan\emptyset = \frac{V_{f2}}{u_2 - V_{w2}}$$

$\emptyset$ = **59° 44'**

3. A centrifugal pump delivers water against a net head of 14.5 m and a design speed of 1000 r.p.m. The vanes are curved back to an angle of 30° with the periphery. The impeller diameter is 300 mm and outlet width is 50mm. determine the discharge of the pump if manometric efficiency is 95 %.

Given:

$H_m = 14.5$ m
$N = 1000$ r.p.m
$\emptyset = 30°$
$D_2 = 0.3$ m
$B_2 = 0.05$ m
$\eta_{man} = 0.95$



To Find:

Q =?

Solution:

$$u_2 = \frac{\pi D_2 N}{60} = 15.70 \text{ m/s}$$

$$\eta_{man} = \frac{g \text{ x } H_m}{V_{w2} \text{ x } u_2}$$

$V_{w2} = 9.54$ m/s

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

$V_{f2} = 3.556$ m/s

$Q = \pi D_2 B_2 V_{f2}$

$Q = 0.1675$ m^3/s

---

4. A centrifugal pump having outer diameter equal to two times the inner diameter and running at 1000 r.p.m works against a total head of 40m. The velocity of flow through the impeller is constant and equal to 2.5 m/s. the vanes are set back at an angle of 40° at outlet. If the outer diameter of the impeller is 500mm and width at outlet is 50mm, determine: (i) vane angle at inlet (ii) work done by the impeller on water per second and (iii) manometric efficiency.

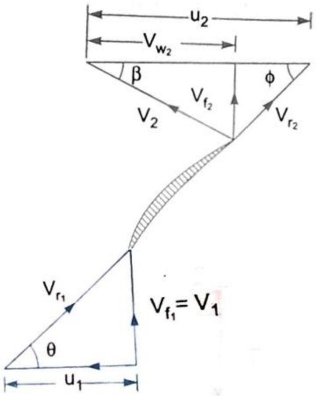
Given:

$D_2 = 2 \text{ x } D_1$
$N = 1000$ r.p.m
$H_m = 40$ m
$V_{f1} = V_{f2} = 2.5$ m/s
$\emptyset = 40°$
$D_2 = 0.5$ m
$B_2 = 0.05$ m

To Find:

θ =?
W.D per second =?
η_{man} =?

Solution:



$D_1 = 0.25$ m

$$u_1 = \frac{\pi D_1 N}{60} = 13.09 \text{ m/s}$$

$$u_2 = \frac{\pi D_2 N}{60} = 26.18 \text{ m/s}$$

$Q = \pi D_2 B_2 V_{f2} = 0.1963$ m^3/s

From Inlet Velocity triangle

$$\tan\theta = \frac{V_{f1}}{u_1}$$

θ = 10° 48’

From outlet Velocity triangle

$V_{f1} = V_{f2} = 2.5$ m/s

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

$V_{w2} = 23.2$ m/s

Work done per second:

W.D per second = ρ Q V_{w2} u_2 = **119227.9 Nm/s**

Manometric efficiency:

$$\eta_{man} = \frac{g \text{ x } H_m}{V_{w2} \text{ x } u_2} = \mathbf{64.4\ \%}$$

5. A centrifugal pump discharge 0.15m^3/s of water against a head of 12.5 m, the speed of the impeller being 600 r.p.m. The outer and inner diameters of impeller are 500 mm and 250 mm respectively and the vanes are bent back at 35 ° the tangent at exit. If the area of flow remains 0.07 m^2 from inlet to outlet, calculate: (i) Manometric efficiency of the pump (ii) vane angle at inlet (iii) loss of head at inlet to impeller when the discharge is reduced by 40% without changing the speed.

Given:

$Q = 0.15\ m^3/s$

$H_m = 12.5\ m$

$N = 600$ r.p.m

$D_2 = 0.5\ m$

$D_1 = 0.25\ m$

$\emptyset = 35°$

$A_1 = A_2 = 0.07\ m^2$

To find:

η_{man} =?

θ =?

Loss of head at inlet when discharge is reduced by 40% =?

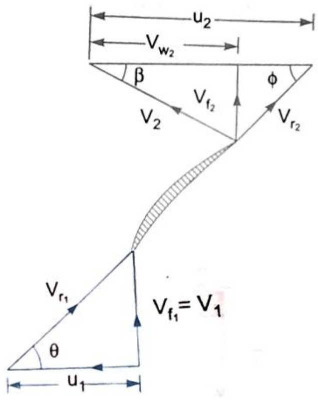
Solution:

$Q = A_1 V_{f1}$

$V_{f1} = V_{f2} = 2.14$ m/s

$$u_1 = \frac{\pi D_1 N}{60} = 7.85 \text{ m/s}$$

$$u_2 = \frac{\pi D_2 N}{60} = 15.70 \text{ m/s}$$



From outlet Velocity triangle

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

$V_{w2} = 12.64$ m/s

a. Manometric efficiency:

$$\eta_{man} = \frac{g \times H_m}{V_{w2} \times u_2} = \mathbf{61.8\%}$$

b. **Vane angle at inlet:**

From Inlet Velocity triangle

$$\tan \theta = \frac{V_{f1}}{u_1}$$

$\boldsymbol{\theta = 15°12'}$

c. Loss of head at inlet when discharge is reduced by 40%

Discharge is reduced by 40%. Hence the new discharge is given by

$Q^* = 0.6\ Q = 0.09\ m^3/s$

$Q^* = A \times V_{f1}^*$

$V_{f1}^* = 1.284$ m/s

$$\tan \theta = \frac{V_{f1}^*}{u_1^*}$$

'$u_1^* = 4.808$ m/s

$$\text{Head loss at inlet} = \frac{(u_1 - u_1^*)^2}{2g} = \mathbf{0.5\ m}$$

---

6. The outer diameter of an impeller of the centrifugal pump is 400 mm and outlet widths is 50mm. the pump is running at 800 r.p.m. and working against a total head of 15m. The vanes angle at outlet is 40° and manometric efficiency is 75 %. Determine: (i) velocity of flow at outlet (ii) velocity of water leaving the vane (iii) angle made by the absolute velocity at outlet with the direction of motion at outlet (iv) Discharge

Given:

$D_2 = 0.4$ m

$B_2 = 0.05$m

$N = 800$ r.p.m

$H_m = 15$ m

$\emptyset = 40°$

$\eta_{man} = 0.75$

To Find:

V_{f2} =?

V_2 =?

β =?

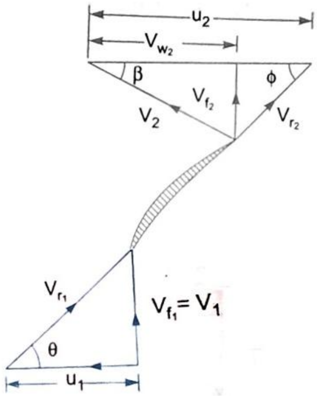
Q =?

Solution:

$$u_2 = \frac{\pi D_2 N}{60} = 16.75 \text{ m/s}$$

$$\eta_{man} = \frac{g \times H_m}{V_{w2} \times u_2}$$

$V_{w2} = 11.71$ m/s



From outlet Velocity triangle

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

V_{f2} = **4.23 m/s**

$$\tan \beta = \frac{V_{f2}}{V_{w2}}$$

β = **19° 48'**

$$\cos \beta = \frac{V_{w2}}{V_2}$$

V_2 = **12.45 m/s**

$Q = \pi D_2 B_2 V_{f2}$ = **0.265 m^3/s**

---

7. A centrifugal pump is running at 1000 r.p.m. The outlet vane angle of the impeller is 45 ° and velocity of flow at outlet is 2.5 m/s. The discharge through the pump is 200liters/s where the pump working against a total head of 20 m. if the manometric efficiency of the pump is 80%. Determine (i) the diameter of the impeller and (ii) the width of the impeller at outlet.

Given:

N =1000 r.p.m

$\phi = 45°$

$V_{f2} = 2.5$ m/s

Q = 200 liters/s = 0.2 m^3/s

$H_m = 20$ m

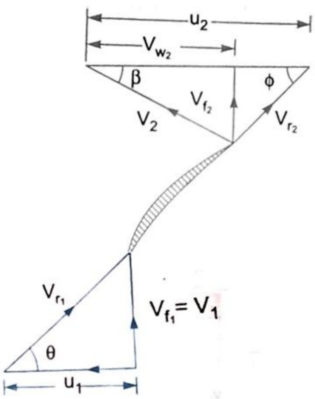
$\eta_{man} = 0.80$

To find:

D_2 =?

B_2 =?

Solution:



$$\eta_{man} = \frac{g \times H_m}{V_{w2} \times u_2}$$

$V_{w2} \times u_2 = 245.25$ -------- (1)

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

$u_2 - V_{w2} = 2.5$ -------- (2)

Substitute equation (1) in (2)

$u_2 = 16.96$ or -14.46 ------- (Negative value not possible)

$$u_2 = \frac{\pi D_2 N}{60}$$

D_2 = 0.324 m

$Q = \pi D_2 B_2 V_{f2}$

B_2 = 0.0786 m

8. A centrifugal pump has a following dimension: inlet radius = 80mm; outlet radius = 160mm; width of the impeller at inlet = 50 mm; ß₁= 0.45 radians; ß₂= 0.25 radians; width of the impeller at outlet = 50 mm. assuming shock less entry determine the discharge and the head developed by the pump when the impeller rotates at 90radians/second.

Given:

$R_1 = 0.08$ m

$R_2 = 0.16$ m

$B_1 = 0.05$ m

$B_2 = 0.05$ m

$\beta_1 = \theta = 0.45 \text{ radians} = 0.45 \times \dfrac{180}{\pi} = 25.78°$

$\beta_2 = \emptyset = 0.25 \text{ radians} = 0.25 \times \dfrac{180}{\pi} = 14.32°$

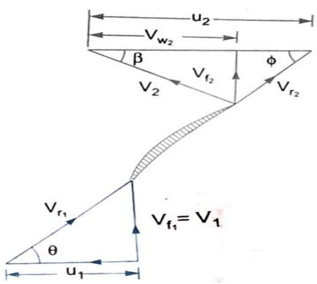
$\omega = 90$ rad/s

To find:

Discharge (Q) =?

Head (H_m) =?

Solution:



$u_1 = \omega \times R_1 = 7.2$ m/s

$u_2 = \omega \times R_2 = 14.4$ m/s

From Inlet Velocity triangle

$\tan\theta = \dfrac{V_{f1}}{u_1}$

$V_{f1} = 3.478$ m/s

a. **Discharge:**

$Q = \pi D_1 B_1 V_{f1} = \mathbf{0.0874\ m^3/s}$

b. **Head Developed:**

$Q = \pi D_2 B_2 V_{f2}$

$V_{f2} = 1.7387$ m/s

$\tan\emptyset = \dfrac{V_{f2}}{u_2 - V_{w2}}$

$V_{w2} = 7.951$ m/s

$H_m = \dfrac{V_{w2} \times u_2}{g}$

$\mathbf{H_m = 11.142\ m}$

---

9. The internal and external diameter of the centrifugal pump which is running at 1000 r.p.m. is 200mm and 400mm respectively. The discharge through the pump is 0.04m³/s and velocity of flow is constant and equal to 2.0m/s. the diameter of the suction and delivery pipes are 150mm and 100mm respectively and suction and delivery heads are 6 m and 30m of water respectively. If the outlet vane angle is 45° and the power required to drive the pump is 16.186 kW, determine: (i) the vane angle of the impeller at inlet (ii) the overall efficiency of the pump (iii) manometric efficiency of the pump.

Given:

N = 1000 r.p.m

$D_1 = 0.2$ m

$D_2 = 0.4$ m

$Q = 0.04\ m^3/s$

$V_{f1} = V_{f2} = 2.0$ m/s

$D_S = 0.15$ m

$D_d = 0.10$ m

$h_S = 6$ m

$h_d = 30$ m

$\emptyset = 45°$

P = 16.186 kW

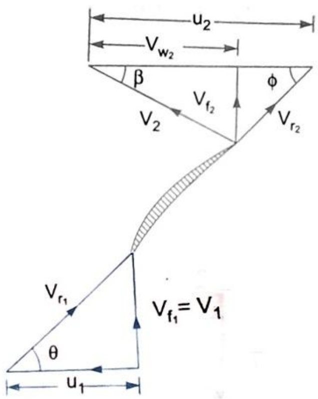
To Find:

θ =?

η_o =?

η_{man} =?

Solution:



$u_1 = \dfrac{\pi D_1 N}{60} = 10.47$ m/s

$u_2 = \dfrac{\pi D_2 N}{60} = 20.94$ m/s

a. **From Inlet Velocity triangle**

$\tan\theta = \frac{V_{f1}}{u_1}$

***θ* = 10° 48'**

b. Overall efficiency

$Q = A_S V_S = A_d V_d$

$A_S = \frac{\pi}{4} D_S^2 = 0.01767\ m^2$

$A_d = \frac{\pi}{4} D_d^2 = 0.007853\ m^2$

$V_S = 2.26$ m/s

$V_d = 5.09$ m/s

$H_m = h_S + h_d + \frac{V_d^2}{2g} - \frac{V_S^2}{2g}$

$H_m = 25.06$ m

Water power = $\frac{W \times Q \times H_m}{1000}$, where,

$W = \rho g = 1000 \times 9.81$

W.P = 9.833 kW

$\eta_o = \frac{W.P}{S.P}$ = **60.74 %**

c. Manometric efficiency:

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

$V_{w2} = 18.94$ m/s

$\eta_{man} = \frac{g \times H_m}{V_{w2} \times u_2}$ **= 61.98 %**

---

10. Find the power required to drive a centrifugal pump which delivers $0.04m^3/s$ of water to a height of 20m through a 15 cm diameter pipe and 100m long. The overall efficiency of the pump is 70% and co-efficient of friction 'f' = 0.15 in the formula $h_f = \frac{4fLV2}{d\ X\ 2g}$.

Given:

$Q = 0.04\ m^3/s$

$H_S = h_s + h_d = 20$ m

$D = D_S = D_d$ = o.15 m

$L = L_S + L_d = 100$ m

$\eta_o = 0.70$

f = 0.15

To Find:

Shaft Power =?

Solution:

$Q = A_S V_S = A_d V_d$

$V_S = V_d = 2.26$ m/s

$h_f = h_{fs} + h_{fd} = \frac{4fLV2}{d\ X\ 2g}$.

= 10.41 m

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

= 30.67 m

Water power = $\frac{W \times Q \times H_m}{1000}$, where,

$W = \rho g = 1000 \times 9.81$

W.P = 12.03 kW

$\eta_o = \frac{W.P}{S.P}$

S.P = **17.19 kW**

---

11. Find the rise in pressure in the impeller of a centrifugal pump through which water is flowing at a rate of $0.01m^3/s$. the internal and external diameters of the impeller ate 15cm and 30cm respectively. The widths of the impeller at inlet and outlet are 1.2cm and 0.6cm. The pump is running at 1500 r.p.m. the water enters the impeller radially at inlet and impeller vane angle at outlet is 45°. Neglect losses through the impeller.

Given:

$Q = 0.01\ m^3/s$

$D_1 = 0.15$ m

$D_2 = 0.30$ m

$B_1 = 0.012$m

$B_2 = 0.006$ m

N = 1500 r.p.m

$\emptyset = 45°$

To Find:

Pressure Raise =?

Solution: $Q = \pi D_1 B_1 V_{f1}$

$V_{f1} = 1.768$ m/s

$Q = \pi D_2 B_2 V_{f2}$

$V_{f2} = 1.768$ m/s

$u_2 = \frac{\pi D_2 N}{60} = 23.56$ m/s

$$\boxed{\text{Pressure raise} = \frac{1}{2g}[V_{f1}^2 + u_2^2 - V_{f2}^2 \operatorname{cosec}^2\emptyset]}$$

= 28.13 m

12. The diameters of an impeller of a centrifugal pump at inlet and outlet are 30cm and 60cm respectively. Determine the minimum starting speed of the pump if its works against a head of 30m.

<u>Given:</u>

$D_1 = 0.30$ m

$D_2 = 0.60$ m

$H_m = 30$ m

<u>To Find:</u>

Minimum starting speed (N) =?

<u>Solution:</u>

$u_1 = \frac{\pi D_1 N}{60} = 0.0157$ N

$u_2 = \frac{\pi D_2 N}{60} = 0.03141$ N

Equations for minimum speed

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

N = 891.8 r.p.m

---

13. The diameters of an impeller of a centrifugal pump at inlet and outlet are 30cm and 60cm respectively. The velocity of flow at outlet is 2.0 m/s and the vanes are set back at an angle of 45° at outlet. Determine the minimum starting speed of the pump if the manometric efficiency is 70 %.

<u>Given:</u>

$D_1 = 0.30$ m
$D_2 = 0.60$ m
$V_{f2} = 2.0$ m/s
$\emptyset = 45°$
$\eta_{man} = 0.70$

<u>To Find</u>:

Minimum Starting speed (N) =?

<u>Solution:</u>

$u_1 = \frac{\pi D_1 N}{60} = 0.0157$ N

$u_2 = \frac{\pi D_2 N}{60} = 0.03141$ N

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

$V_{w2} = u_2 - 2 = 0.03141$ N – 2 ------ (1)

$\eta_{man} = \frac{g \text{ x } H_m}{V_{w2} \text{ x } u_2}$

$H_m = 0.07135(0.03141$ N – 2) 0.03141 N

Equations for minimum speed

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

N = 137.22 r.p.m

---

14. A centrifugal pump with 1.2 diameter runs at 200 r.p.m and pumps 1880 liters/s, the average lift being 6m.The angle which the vane makes at exit with tangent to the impeller is 26° and the radial velocity of flow is 2.5 m/s. determine the manometric efficiency and the least speed to start pumping against a head of 6m, the inner diameter of the impeller being 0.6m.

<u>Given:</u>

$D_2 = 1.2$ m
N =200 r.p.m
Q = 1880 liters/s = 1.88 m^3/s
$H_m = 6$ m
$V_{f2} = 2.5$ m/s
$\emptyset = 26°$
$D_1 = 0.6$ m

<u>To Find:</u>

Manometric efficiency =?

Minimum Speed =?

<u>Solution:</u>

a. **<u>Manometric efficiency:</u>**

$u_2 = \frac{\pi D_2 N}{60} = 12.56$ m/s

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

$V_{w2} = 7.43$ m/s

$\eta_{man} = \frac{g \times H_m}{V_{w2} \times u_2}$

= 63 %

b. Least speed to start the pump:

$u_1 = \frac{\pi D_1 N}{60} = 0.03141\ N$

$u_2 = \frac{\pi D_2 N}{60} = 0.06283\ N$

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

N = 200 r.p.m

---

15. A three stage centrifugal pump has impeller 40cm in diameter and 2cm wide at outlet. The vanes are curved back at outlet at 45° and reduce the circumferential area by 10 %. The manometric efficiency is 90 % and the overall efficiency is 80 %. Determine the head generated by the pump when running at 1000 r.p.m. delivering 50 liters per second. What should be the shaft horse power?

Given:

$n = 3$

$D_2 = 0.40$ m

$B_2 = 0.02$ m

$\emptyset = 45°$

Reduction in area at outlet = 10% = 0.1

$\eta_{man} = 0.90$

$\eta_o = 0.80$

N = 1000 r.p.m

$Q = 0.05\ m^3/s$

To Find:

Head generated by the pump =?

Shaft Power =?

Solution:

Area of flow at outlet = $0.9\ \pi D_2 B_2 = 0.02262\ m^2$

$V_{f2} = Q/A_2 = 2.21$ m/s

$u_2 = \frac{\pi D_2 N}{60} = 20.94$ m/s

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

$V_{w2} = 18.73$ m/s

a. Head generated by the pump:

$\eta_{man} = \frac{g \times H_m}{V_{w2} \times u_2}$

H_m = 35.98 m

Total head generated by the pump = n x H_m

= 3 x 35.98 = **107.94 m**

b. Shaft Power:

Water power = $\frac{W \times Q \times H_m}{1000}$, where,

$W = \rho g = 1000 \times 9.81$

W.P = 52.94 kW

$\eta_o = \frac{W.P}{S.P}$

S.P = **66.175 kW**

---

16. A four stage centrifugal pump has four identical impellers, keep to the same shaft. The shaft running at 400 r.p.m. and the total manometric head developed by the multistage pump is 40m. The discharge through the pump is $0.2m^3/s$. the vanes of each impeller having outlet angle as 45°. If the width and diameter of each impeller at outlet is 5cm and 60cm respectively, find the manometric efficiency.

Given:

n =4

N = 400 r.p.m

Total head = 40 m

For each stage H_m = 10 m

$Q = 0.2\ m^3/s$

$\emptyset = 45°$

$D_2 = 0.60$ m

$B_2 = 0.05$ m

To Find:

$u_2 = \frac{\pi D_2 N}{60} = 12.56$ m/s

$Q = \pi D_2 B_2 V_{f2}$

$V_{f2} = 2.122$ m/s

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

$V_{w2} = 10.438$ m/s

$\eta_{man} = \frac{g \times H_m}{V_{w2} \times u_2}$

= 74.82 %

Unit IV

CLASSIFICATION OF COMPRESSOR

Air compressors may be classified as follows:

1) *According to design and principle of operation*
   a) Reciprocating compressors
   b) Rotary compressors

2) *According to action*
   a) Single acting compressors
   b) Double acting compressors

3) *According to number of stages*
   a) Single stage compressors
   b) Multistage compressors

4) *According to pressure limit*
   a) Low pressure compressor
   b) Medium pressure compressors
   c) High pressure compressors

5) *According to capacity*
   a) Low capacity compressors (Volume delivered 0.12m^3/s or less)
   b) Medium capacity compressors (volume delivered 0.15m^3/s to 5m^3/s)
   c) High capacity compressors (Volume delivered is above 5m^3/s)

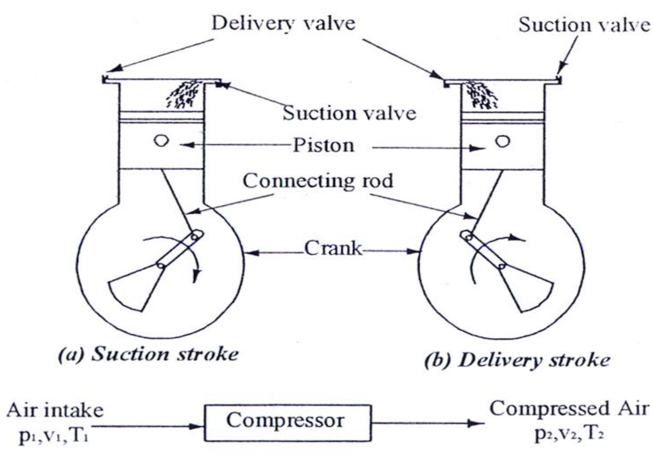
Single stage compressor:

In single stage compressor, the compression of the air from the initial pressure to the final pressure is carried out in one cylinder only.

Multistage compressor:

In multistage compressor, the compression of the air from the initial pressure to the final pressure is carried out in more than one cylinder.

WORKING OF SINGLE STAGE RECIPROCATING AIR COMPRESSOR



Single stage reciprocating air compressor

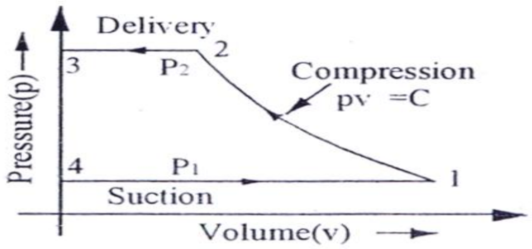
In a single stage compressor, the compression of air from the initial pressure to final pressure is carried out in one cylinder only. A schematic diagram of single stage, single acting compressor is shown in fig

It consists of a cylinder, piston, connecting rod, crank, inlet and discharge valves. When the piston moves downward i.e. during suction stroke, the pressure of air inside the cylinder falls below the atmospheric pressure. So the inlet valve opens and the air from atmospheric is sucked into the cylinder until the piston reaches the bottom dead center. During this stroke delivery valve remains closed. When the piston moves upwards both valves are closed. So the pressure inside the cylinder goes on increasing till it reaches required discharge pressure. At this stage, the discharge value opens and the compressed air is delivered through this valve. Thus the cycle is repeated.

The work done by a single stage reciprocating air compressor without clearance volume

a) Work done isothermal compression (pv = c):

The p-v diagram for a single stage single acting reciprocating air compressor is show is fig. the sequences of operation as represented on the diagram are as follows:



Isothermal compression

Process 4-1 => Represents the suction of air at pressure p_1

Process 1-2 => Air is compressed isothermally from pressure p_1 to pressure p_2

Process 2-3 => Represents the discharge of air at pressure p_2

Work done = Area 1 – 2 – 3 – 4 – 1

$$W = W_{comp} + W_{Delivery} - W_{Suction}$$

$$= p_1 v_1 \text{In}\left[\frac{v_1}{v_2}\right] + p_2 v_2 - p_1 v_1 \qquad [\because \textit{For Isothermal process } W_{(comp)} = p_1 v_1 \text{In}\left[\frac{v_1}{v_2}\right]]$$

$$= p_1 v_1 \text{In}\left[\frac{v_1}{v_2}\right] + p_2 v_2 - p_2 v_2 \qquad [\because \textit{For Isothermal process } p_1 v_2 - p_1 v_1]$$

$$W = p_1 v_1 \text{In}\left[\frac{v_1}{v_2}\right]$$

We known that,

$$p_1 v_1 = p_2 v_2$$

$$\frac{v_1}{v_2} = \frac{p_2}{p_1}$$

Substituting in equation (1)

$$W = p_1 v_1 \text{In}\left[\frac{p_2}{p_1}\right]$$

We known that,

$$Pv = mRT$$

So

$$\boxed{W = mRT \text{ In}\left[\frac{p_2}{p_1}\right]}$$

b) Work done during polytrophic compression (pv^n = constant);

The *p-v* diagram for a single stage single acting reciprocating air compressor is shown in fig.

The sequences of operations as represented on the diagram are as follows:

Process 4-1 => Represents the suction of air at pressure p_1
Process 1-2 => Air is compressed Polytropically from pressure p_1 to pressure p_2
Process 2-3 => Represents the discharge of air at pressure p_2

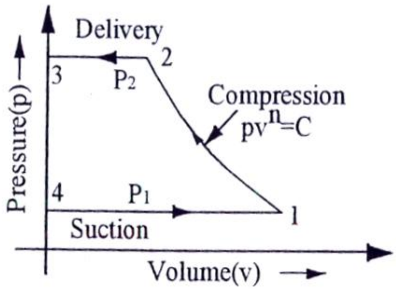


Fig. Polytropic compression

Work done = Area 1 – 2 – 3 – 4 – 1

$$W = W_{comp} + W_{Delivery} - W_{Suction} \qquad \left[\because \text{For Polytropic process} \;\; W_{comp} = \frac{p_2 v_2 - p_1 v_1}{n-1}\right]$$

$$W = \frac{p_2 v_2 - p_1 v_1}{n-1} + p_2\, v_2 - p_1 v_1$$

$$= \frac{p_2 v_2 - p_1 v_1 + (n-1)(p_2 v_2) - (n-1)(p_1 v_1)}{n-1}$$

$$= \frac{p_2 v_2 - p_1 v_1 + n p_2 v_2 - p_2 v_2 - n\, p_1 v_1 + p_1 v_1}{n-1}$$

$$= \frac{n p_2\, v_2 - n\, p_1 v_1}{n-1} = \frac{n\,(p_2 v_2 - p_1 v_1)}{n-1}$$

$$\boxed{W = \frac{n}{n-1}\,[p_2\, v_2 - p_1 v_1]}$$

$$p_1 v_1 = mRT_1; \qquad p_2\, v_2 = mRT_2$$

$$W = \frac{n}{n-1}\,[mRT_2 - mRT_1]$$

$$W = \frac{n}{n-1}\, mR\, T_1\left[\frac{T_2}{T_1} - 1\right]$$

For polytropic process,

$$\frac{T_2}{T_1} = \left(\frac{P_2}{P_1}\right)^{\frac{n-1}{n}}$$

$$W = \frac{n}{n-1}\, mR\, T_1\left[\left(\frac{P_2}{P_1}\right)^{\frac{n-1}{n}} - 1\right]$$

$$W = \frac{n}{n-1}\, p_1 v_1\left[\left(\frac{P_2}{P_1}\right)^{\frac{n-1}{n}} - 1\right] \qquad [\because pv = mRT]$$

c) Work done during isentropic compression (or) adiabatic compression [$pv^\gamma = C$]:

The *p-v* diagram for a single stage single acting reciprocating air compressor is shown in fig.

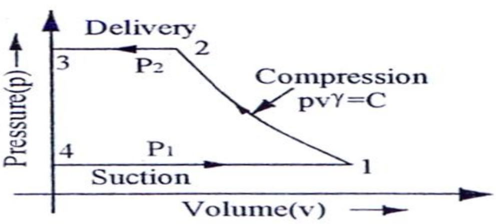


Fig. Adiabatic compression

The sequences of operation as represented on the diagram are as

Follows:

Process 4-1 => Represents the suction of air at pressure p_1

Process 1-2 => Air is compressed Polytropically from pressure p_1 to pressure p_2

Process 2-3 => Represents the discharge of air at pressure p_2

Work done = Area 1-2-3-4-1

$$W = W_{comp} + W_{Delivery} - W_{Suction} \qquad \left[\begin{array}{l} \because \text{For isentropic process} \\ W_{comp} = \frac{p_2 v_2 - p_1 v_1}{\gamma - 1} \end{array}\right]$$

$$= \frac{p_2 v_2 - p_1 v_1}{\gamma - 1} + p_2 v_2 - p_1 v_1$$

$$= \frac{p_2 v_2 - p_1 v_1 + (\gamma - 1)(p_2 v_2) - (\gamma - 1)(p_1 v_1)}{\gamma - 1}$$

$$= \frac{p_2 v_2 - p_1 v_1 + \gamma p_2 v_2 - p_2 v_2 - \gamma p_1 v_1 + p_1 v_1}{\gamma - 1}$$

$$= \frac{\gamma (p_2 v_2 - p_1 v_1)}{\gamma - 1}$$

$$\boxed{W = \frac{\gamma}{\gamma - 1}[p_2 v_2 - p_1 v_1]}$$

$p_1 v_1 = mRT_1; \qquad p_2 v_2 = mRT_2$

$$W = \frac{\gamma}{\gamma - 1}[mRT_2 - mRT_1]$$

$$W = \frac{\gamma}{\gamma - 1} mR\, T_1 \left[\frac{T_2}{T_1} - 1\right]$$

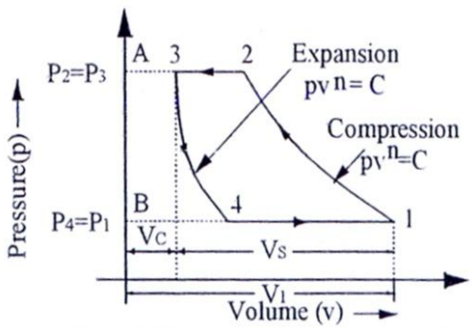
For polytropic process,

$$\frac{T_2}{T_1} = \left(\frac{P_2}{P_1}\right)^{\frac{\gamma - 1}{\gamma}}$$

$$W = \frac{\gamma}{\gamma - 1} p_1 v_1 \left[\left(\frac{P_2}{P_1}\right)^{\frac{\gamma - 1}{\gamma}} - 1\right] \quad [\because pv = mRT]$$

Efficiency and work done by single stage reciprocating air compressor with clearance volume

When the piston reaches top dead centre in the cylinder there is a dead space between piston top and the cylinder head. This space is known as clearance space and the volume occupied by this space is known as clearance volume (vc). Consider a reciprocating air compressor with clearance volume (vc) as shown in fig



P_1, $v1$, $T1$ => Initial pressure, Initial temperature, initial volume of air respectively

P_2, $v2$, $T2$ => Corresponding values for the final conditions

V_c => clearance volume

V_s => Stroke volume = V_1 - V_c

n => Polytropic index for compression and expansion

Work done by the compressor per cycle

W = Area 1-2-3-4-1 = (Area 1-2-A-B-1) – (Area 3-A-B-4-3)

W = Work done during compression – work done during expansion

$$W = \frac{n}{n-1} p_1 v_1 \left[\left(\frac{P_2}{P_1}\right)^{\frac{n-1}{n}} - 1 \right] W = \frac{n}{n-1} p_1 v_4 \left[\left(\frac{P_3}{P_4}\right)^{\frac{n-1}{n}} - 1 \right]$$

we known that,

p_1-p_4; p_1-p_3

$$W = \frac{n}{n-1} p_1 v_1 \left[\left(\frac{P_2}{P_1}\right)^{\frac{n-1}{n}} - 1 \right] W = \frac{n}{n-1} p_1 v_4 \left[\left(\frac{P_3}{P_4}\right)^{\frac{n-1}{n}} - 1 \right]$$

$$W = \frac{n}{n-1} p_1 \left[\left(\frac{P_2}{P_1}\right)^{\frac{n-1}{n}} - 1 \right] [v_1 - v_4]$$

$$W = \frac{n}{n-1} p_1 [v_1 - v_4] \left[\left(\frac{P_2}{P_1}\right)^{\frac{n-1}{n}} - 1 \right]$$

$$\boxed{W = \frac{n}{n-1} p_1 v_a \left[\left(\frac{P_2}{P_1}\right)^{\frac{n-1}{n}} - 1 \right]}$$

Where, $V_a = V_1 - V_4$ is the actual volume of free air delivered per cycle

$$W = \frac{n}{n-1} mR\, T_1 \left[\left(\frac{P_2}{P_1}\right)^{\frac{n-1}{n}} - 1 \right] \qquad [\because P_1 V_a = mRT_1]$$

VOLUMETRIC EFFICIENCY

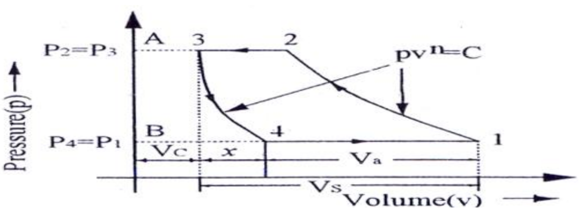
Volumetric efficiency is defined as the ratio of volume of free air sucked into the compressor per cycle to the stoke volume of the cylinder.

$$\eta_{vol} = \frac{Volume\ of\ Free\ taken\ per\ cycle}{Stroke\ volume\ of\ the\ cylinder}$$

$$\eta_{vol} = \frac{v_a}{v_s}$$

From $P - V$ diagram

$V_a = v_s - x$, $X = v_4 - vc$



volumetric efficiency

$$\eta_{vol} = \frac{v_s - x}{v_s} = \frac{v_S - (v_4 - v_C)}{V_s} = \frac{v_S - v_C\left[\frac{v_4}{v_C} - 1\right]}{V_s}$$

$$\boxed{\eta_{vol} = 1 - \frac{v_C}{V_s}\left[\frac{v_4}{v_C} - 1\right]}$$

Compression and expansion follows $PV^n = C$

$$P_3V_3^n = P_4V_4^n$$

$$\frac{v_4}{V_3} = \left(\frac{P_3}{P_4}\right)^{\frac{1}{n}}$$

From p-v diagram we known that,

$$V_3 = v_C,\ P_4 = P_1,\ P_3 = P_2$$

$$\frac{v_4}{V_C} = \left(\frac{P_3}{P_4}\right)^{1/n} \Rightarrow \frac{v_4}{V_C}\left(\frac{P_2}{P_1}\right)^{1/n}$$

Apply $\frac{v_4}{V_C}$ value in Equation (10)

$$\eta_{vol} = 1 - \frac{v_C}{V_s}\left[\left(\frac{P_2}{P_1}\right)^{\frac{1}{n}} - 1\right]$$

Clearance ratio is defined as the ratio of clearance volume to swept volume.

Clearance ratio, C = $\frac{v_C}{V_s}$

$$\eta_{vol} = 1 - c\left[\left(\frac{P_2}{P_1}\right)^{\frac{1}{n}} - 1\right]$$

Centrifugal compressors

In this type of compressor air enters axially and leaves radially.

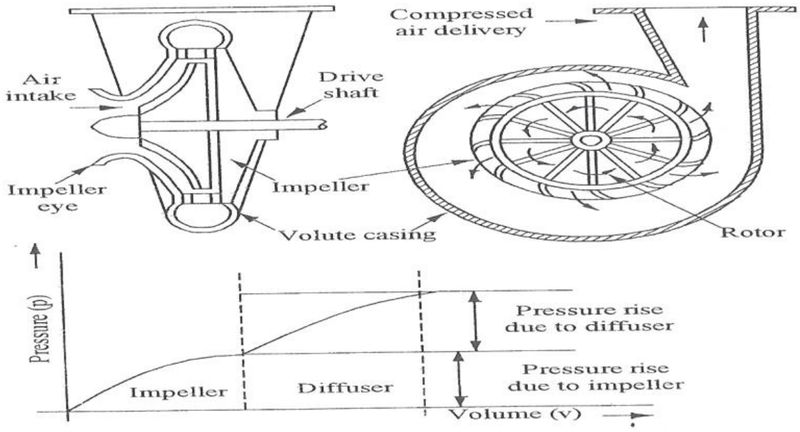
Construction:

- The arrangement of centrifugal compressor is shown in Fig.
- It consists of a rotating impeller, a casing and a diffuser.
- The impeller consists of a disc on which radial blades are attached. The impeller is surrounded by casing.
- The diffuser is other important part of the compressor, which is used to convert kinetic energy of air into pressure energy.
- The air coming out from the diffuser is collected in the casing and taken out from the outlet of the compressor.
- The impeller of a centrifugal compressor can be run at speeds of 20,000 to 30,000*rpm*

Working:

- When the power is given to compressor, the impellor, the impeller rotates, and it sucks the air.
- This air enters axially with low velocity. The velocity and pressure of the air passing through the impeller are partially increased.

- Then the air is entered into diffuser. In the diffuser, kinetic energy is converted into pressure energy. So the pressure of air is further increased.
- Finally the air at high pressure is delivered to the receiver. Nearly half of the total pressure rise is achieved in the impeller and remaining half in the diffuser.
- The change of pressure and velocity of air passing through the impeller and diffuser are shown in Fig



Applications:

- Centrifugal compressors are suitable for super charging I.C. engines, refrigeration and low-pressure units

Vane blower

Construction:

- The arrangement of vane blower is shown in Fig.
- It consists of rotating drum, spring-loaded vanes, inlet and outlet ports and casing.

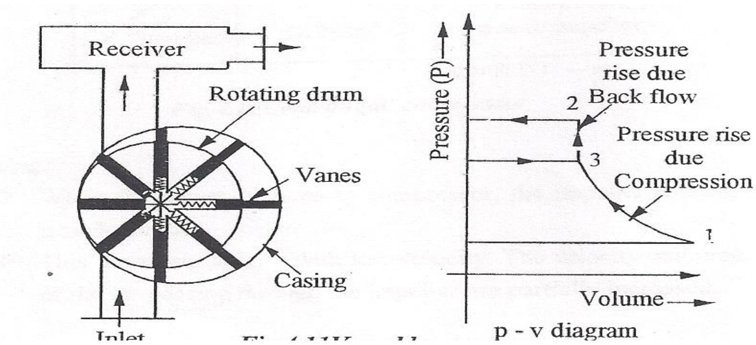


Fig. Vane blower

- The rotor is located eccentrically inside the casing.
- The rotor carries a set of spring-loaded vanes.
- These spring-loaded vanes are made of fiber of carbon.

Working:

- When the power is given to a vane blower, rotating drum (rotor) rotates, and the air is trapped between two consecutive vanes.
- As the rotation takes place the trapped air first expands and then gradually compressed due to decreasing volume between the rotor and outer casing.
- This partially compressed air is delivered to the receive.
- When the outlet is opened, there is a back flow of high-pressure air from receiver will rush back and mixed up with the entrapped air. So partially compressed air pressure is further increased. Finally high-pressure air is delivered from the receiver.
- In vane blower the pressure of air is increased first by decreasing the volume and then by back flow of air as shown in p-v diagram Fig

c) **<u>Roots blower:</u>**

Construction:

- The arrangement of Roots blower is shown in fig. which is simply a development of the gear pump.
- It consists of two rotors, which are aligned in different parallel axis.
- One of the rotors is connected to the drive and the second rotor is driven from the first.
- The two rotors rotates in opposite directions i.e. one in clockwise direction and the other in anti clockwise direction.
- The lobes of the rotors are of epicycloids, hypocycloid, or involutes profile to ensure correct matching.
- There must be small clearance between the lobe and casing to reduce the wear of moving parts.

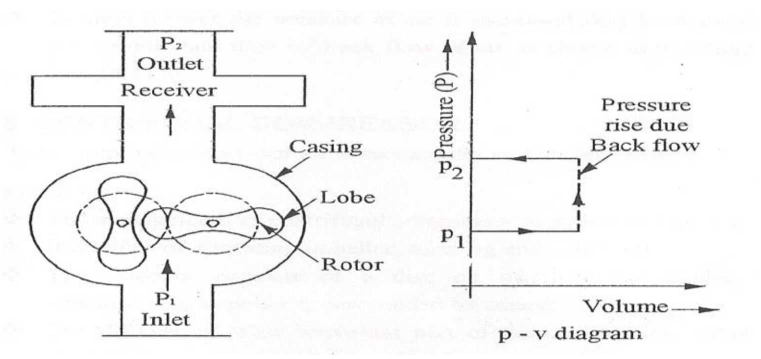


Fig. Roots blower

- When the power is given to the roots blower, rotors rotates and the air at atmospheric pressure is trapped between the lobe and casing.
- The trapped air moves along the casing and discharged into the receiver.
- The flow area from entry to exit remains constant. So, there is no developing in pressure.
- When the exit port opens, some high-pressure air from receiver will rush back and mixed up with entrapped air until the pressure is equalized.
- Thus the pressure of the entrapped air is increased by back flow of air.
- The *p-v* diagram for this type of compressor is shown in Fig.

Differentiate between Reciprocating and Rotary compressor.

SL.NO	Rotary Compressor	Reciprocating Compressor
1.	Simple in construction	Complicated construction
2.	Speed is high	Speed is low
3.	It is suitable for large rates of flow at low discharge pressure	It is suitable for low rates of flow at very discharge pressure
4.	Maintenance cost is less	Maintenance cost is high
5.	There is no balancing problem	Balancing is major problem
6.	Simple lubrication system	More complicated
7.	Small in size for the same discharge compared with reciprocating compressors	Large in size for the same discharge compa with rotary compressor
8.	Uniform delivery of air	Delivery is not uniform

UNIT - 4

1. A single stage, single acting reciprocating air compressor has a bore of 200mm and a stroke of 300mm. It receives air at 1 bar and 20° C and delivers it at 5.5 bar. If the compression follows the law $pv^{1.3}$ = C and clearance volume is 5 % of the stroke volume. Determine: 1. the mean effective pressure and 2. Power required to drive the compressor if it runs at 500 r.p.m.

Given:

$D = 0.2m$

$L = 0.3m$

$P_1 = 1 \text{ bar} = 1 \times 10^5 \text{ N/m}^2$

$T_1 = 293 \text{ K}$

$P_2 = 5.5 \text{ bar} = 5.5 \times 10^5 \text{ N/m}^2$

$n = 1.3$

$V_C = 5\% \, v_s$

$N = 500 \text{ r.p.m}$

To Find:

$P_m = ?$

$P = ?$

Solution

Stroke Volume

$V_s = \frac{\pi}{4} \times D^2 \times L = 0.00942 \text{ m}^3$

Clearance volume

$V_C = 5\% \, V_s = 0.00047 \text{ m}^3$

Initial Volume of air,

$V_1 = V_C + V_s = 0.00989 \text{ m}^3$

Expanded clearance Volume,

$V_4 = V_C (P_2/P_1)^{1/n} = 0.00174 \text{ m}^3$

Work Done

$$W = \frac{n}{n-1} \times \boldsymbol{p_1 v_1} \left[\left(\frac{P_2}{P_1}\right)^{\frac{n-1}{n}} - 1 \right]$$

$= 1702 \text{ Nm}$

a. **Mean Effective Pressure**

$$P_m = \frac{work\ done}{stroke\ volume} = \mathbf{1.807\ bar}$$

b. **Power required to drive the compressor**

$$P = \frac{W \times N}{60} = \mathbf{14.183\ kW}$$

---

2. A two stage single acting reciprocating air compressor draws in air at a pressure of 1 bar and 17° C and compresses it to a pressure of 60 bar. After compression in the L.P cylinder, the air is cooled at a constant pressure of 8 bar to a temperature of 37° C. The L.P cylinder has a diameter of 150 mm and both the cylinder have 200mm stroke. If the law of compression is $pv^{1.35}$ = C, find the power of the compressor, when it runs at 200 r.p.m. Take R = 287 J/kg K.

Given:

$P_1 = 1 \times 10^5 \text{ N/m}^2$

$P_2 = 8 \times 10^5 \text{ N/m}^2$

$P_3 = 60 \times 10^5 \text{ N/m}^2$

$T_1 = 290 \text{ K}$

$T_3 = 310 \text{ K}$

$D = 0.15 \text{ m}$

$L = 0.2 \text{ m}$

$n = 1.35$

$N = 200 \text{ r.p.m}$

To Find:

Power =?

Solution:

Volume of L.P cylinder

$V_1 = \frac{\pi}{4} \times D^2 \times L = 0.0035 \text{ m}^3$

Volume of H.P cylinder

$$\frac{P_1V_1}{T_1} = \frac{P_2V_2}{T_3}$$

$V_2 = 0.00047 \text{ m}^3$

Work done by L.P Cylinder

$$W_{L.P} = \frac{n}{n-1} \boldsymbol{p_1 v_1} \left[\left(\frac{P_2}{P_1}\right)^{\frac{n-1}{n}} - 1 \right]$$

$W_{L.P} = 965 \text{ Nm}$

Work Done By H.P Cylinder

$$W_{H.P} = = \frac{n}{n-1} \, p_2 v_2 \left[\left(\frac{p_3}{p_2}\right)^{\frac{n-1}{n}} - 1 \right]$$

$W_{H.P} = 996 \text{ Nm}$

Total Work done by the Compressor

$W = W_{L.P} + W_{H.P}$

$W = 1961 \text{ Nm}$

a. **Power**

$$P = \frac{W \times N}{60} = \mathbf{6.54\ kW}$$

3. A centrifugal compressor runs at 15000 r.p.m and produce a stagnation pressure ratio of 4 between the impeller inlet and outlet. The stagnation conditions of air at the compressor intake are 1.01325 bar and 25° C respectively. The absolute velocity at the compressor inlet is axial. If the compressor has radial blades at the exit such that V_{f2} = 135 m/s and the total to total efficiency of the compressor is 0.78, draw the velocity triangle at exit of the rotor and compute the slip as well as the slip co-efficient. Rotor diameter at outlet is 580mm. Compute the absolute velocity at the compressor exit also.

Given:

N = 15000 r.p.m

$P_{o2} / P_{o1} = 4$

P_{o1} = 1.01325 bar

T_{o1} = 298 K

$\eta_{t\text{-}t} = 0.78$

D_2 = 0.58 m

V_{f2} = 135 m/s

To find:

Slip =?

Slip co-efficient =?

V_2 =?

Solution:

$$U_2 = \frac{\pi D_2 N}{60} = 455.53 \text{ m/s}$$

$$\frac{T_{02}'}{T_{01}} = \left\{\frac{P_{02}}{P_{01}}\right\}^{\frac{\gamma-1}{\gamma}}$$

T_{02}' = 442.826K

$$\eta_{t\text{-}t} = \frac{T_{02}' - T_{01}}{T_{02} - T_{01}} = 0.78$$

T_{02} = 483.647 K

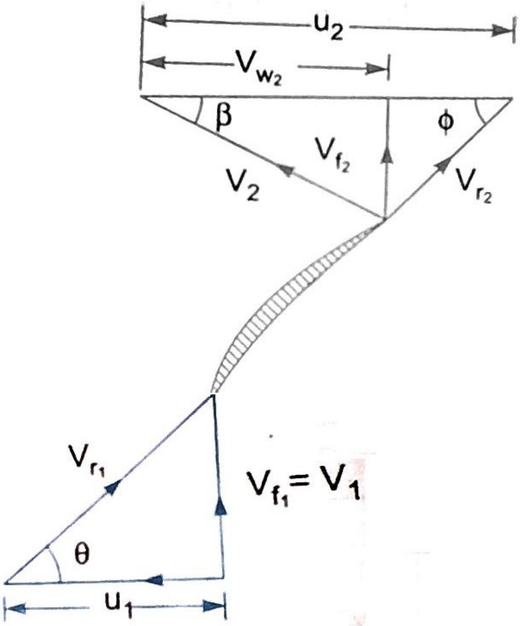
Actual work

$W_{act} = C_p (T_{02} - T_{01})$ = 186.603 KJ/kg

We know that

$W_{act} = V_{w2} U_2$

V_{w2} = 409.6394 m/s



a. **Slip:**

Slip = $U_2 - V_{w2}$ = **45.89 m/s**

b. **Slip co-efficient:**

Slip co-efficient = V_{w2}/ U_2 = **0.89925**

c. **Absolute velocity at exit:**

$V_2 = \sqrt{(V_{f2}^2 - U_2^2)}$ = **431.3113 m/s**

---

4. A centrifugal compressor compresses 30 kg of air per second. It runs at 15000 r.p.m. the air enters the compressor axially. The radius at exit of blade is 300 mm. The relative velocity of air at the tip is 100 m/s. The relative air angle at exit is 80°. Find the power and ideal head developed.

Given:

m = 30 kg/s

N = 15000 r.p.m

D_2 = 0.6 m

V_{r2} = 100 m/s

$\emptyset = 80°$

To find

Power developed =?

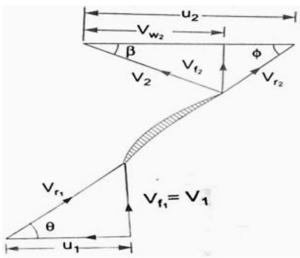
Ideal Head =?

Solution

$$U_2 = \frac{\pi D_2 N}{60} = 471.2388 \text{ m/s}$$

$$\cos \emptyset = \frac{U_2 - V_{w2}}{V_{r2}}$$

V_{w2} = 453.8739 m/s



a. **Power developed**

P = m x V_{w2} x U_2 = **6416.491 kW**

b. **Ideal head developed**

= V_{w2} x U_2 = **213.883 KJ/kg**

5. The inlet stagnation conditions of a centrifugal compressor are 102 Kpa, 335 K. The hub and tip diameters at the impeller inlet are 100 mm, 250 mm respectively. The compressor runs at 120r.p.s and delivers 5 kg/s of air. Determine the air angle at inlet of the inducer blade and inlet mach number-relative velocity based.

Given:

P_{01} = 102 Kpa = 102 KN/m^2

T_{01} = 335 K

D_H = 0.1 m

D_T = 0.25 m

N = 7200 r.p.m

m = 5 kg/s

To find:

θ =?

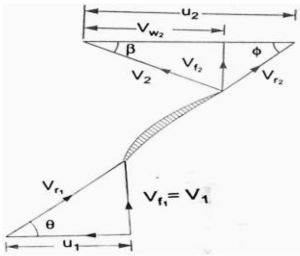
M_I =?

Solution

$$D_m = \frac{D_H + D_T}{2} = 0.175 \text{ m}$$

$$B = \frac{D_H - D_T}{2} = 0.075 \text{ m}$$

$$\rho_1 = \frac{P_{01}}{R \times T_{01}} = 1.060897 \text{ kg/m}^3$$



Mass flow rate

$$m = \rho_1[\pi\, D_m B]\, V_{f1}$$

V_{f1} = 114.3 m/s

$$U_1 = \frac{\pi\, D_m N}{60} = 65.9734 \text{ m/s}$$

a. **Air angle at inlet**

$$\tan\theta = \frac{V_{f1}}{U_1}, \ \boldsymbol{\theta = 60°}$$

b. **Inlet mach number relative velocity based**

$\cos\theta = U_1 / V_{r1}$

V_{r1} = 132 m/s

$$M_I = \frac{V_{r1}}{\sqrt{\gamma R T_{01}}} = \mathbf{0.366}$$

---

6. A centrifugal blower runs at a speed of 3000 r.p.m. Its impeller outer diameter being 75 cm. The impeller blades are designed for a constant radial velocity of 57 m/s from the inlet to the outlet. There are no guide vanes so that the absolute velocity at inlet is axial. If the degree of reaction is 0.58 draw velocity triangle at the outlet and compute the exit blade angle. Find also the power input to the machine assuming a total –to-total efficiency of 0.75 and the exit pressure at the inlet to be 1 atm, and the total temperature to be 25°C.

Given:

N = 3000 r.p.m

D_2 = 0.75 m

$V_{f1} = V_{f2}$ = 57 m/s

Rd = 0.58

$\eta_{t\text{-}t}$ = 0.75

P_{01} = 101 Kpa = 101 KN/m^2

T_{01} = 298 K

To find:

$\emptyset$ =?

P =?

P_{02} =?

Solution

$$U_2 = \frac{\pi D_2 N}{60} = 117.8097 \text{ m/s}$$

Degree of reaction

$$Rd = 1 - \frac{1}{2}\left(\frac{V_{w2}}{U_2}\right)$$

V_{w2} = 98.96 m/s

a. **From Velocity Triangle**

$$\tan\emptyset = \frac{V_{f2}}{U_2 - V_{w2}}$$

$\emptyset$ = **71.7°**

b. **Power input to the blower**

$W_{act} = V_{w2}\, U_2$ = 11.658 KJ/kg

Power input to the blower = **11.658 KW/ Kg/s**

c. **The exit total pressure**

$W_{act} = C_p\,(T_{02} - T_{01})$

$(T_{02} - T_{01})$ = 11.6 K

$$\eta_{t\text{-}t} = \frac{T_{02}' - T_{01}}{T_{02} - T_{01}} = 0.75$$

$(T_{02}' - T_{01})$ = 8.7003

T_{02}' = 306.7 K

$$\frac{T_{02}'}{T_{01}} = \left\{\frac{P_{02}}{P_{01}}\right\}^{\frac{\gamma-1}{\gamma}}$$

P_{01} = **112.0623 KN/m^2**

7. Air flows through a blower; its total pressure is increased by 15 cm of water head. The inlet total pressure and temperature are 105 Kpa and 15°C. The total – to – total efficiency is 75 %. Evaluate (i) exit total pressure and temperature (ii) isentropic and actual change in total head enthalpy. Assume 50 % reaction.

Given

Height increased = 0.15 m

P_{01} = 105 KN/m^2

T_{01} = 288 K

η_{t-t} = 0.75

To find:

P_{02} =?

T_{02} =?

Isentropic change in enthalpy =?

Actual change in enthalpy =?

Solution

$H = \frac{P}{\rho g}$

Pressure increased = 1471 N/m^2

P_{02} = P_{01} + pressure increased

P_{02} = **106.47 KN/m^2**

$$\frac{T_{02}'}{T_{01}} = \left\{\frac{P_{02}}{P_{01}}\right\}^{\frac{\gamma-1}{\gamma}}$$

T_{02}' = 289.14 K

$$\eta_{t-t} = \frac{T_{02}' - T_{01}}{T_{02} - T_{01}} = 0.75$$

T_{02} = **289.528K**

Isentropic change in enthalpy

= C_p (T_{02}' – T_{01}) = **1.152 KJ/kg**

Actual Change in enthalpy

= C_p (T_{02} – T_{01}) = **1.535 KJ/kg**

---

8. A single acting reciprocating air compressor has cylinder diameter and stroke of 200 mm and 300 mm respectively. The compressor sucks air at 1 bar and 27° C and delivers at 8 bar while running at 100 r.p.m. find: 1. Indicated power of the compressor 2. Mass of air delivered by the compressor per minute 3. Temperature of air delivered by the compressor. The compression follows the law $pv^{1.25}$ = C.

Given:

D = 0.2 m

L = 0.3 m

$P_1 = 1 \times 10^5$ N/m^2

T_1 = 300 K

$P_2 = 8 \times 10^5$ N/m^2

N = 100 r.p.m

n = 1.25

R = 0.287 KJ/kg K

To find:

Indicated power =?

Mass delivered per minute =?

Temperature of air delivered =?

Solution;

$V_1 = \frac{\pi}{4} \times D^2 \times L = 0.0094$ m^3

a. Indicated power

$$W = \frac{n}{n-1} \times \boldsymbol{p_1 v_1} \left[\left(\frac{P_2}{P_1}\right)^{\frac{n-1}{n}} - 1 \right]$$

= 2425 Nm

$\text{Power} = \frac{W \times N}{60}$ = 4.042 kW

b. Mass of air delivered per minute

P_1V_1 = m R T_1

m = 0.0109 kg

Mass delivered per minute = m x N

= 1.09 kg

c. Temperature of air delivered

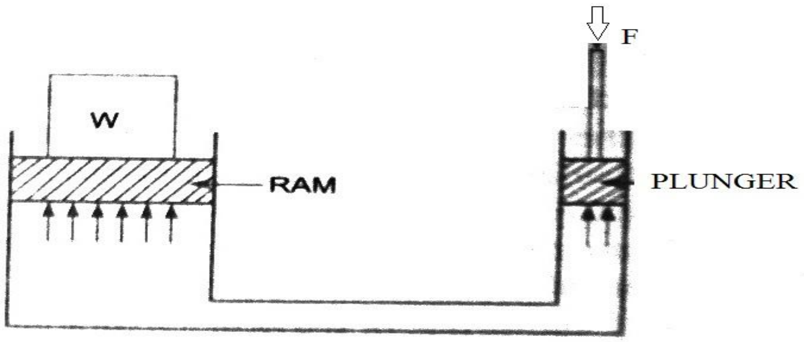
$(T_2/T_1) = (P_2/P_1)^{n-1/n}$

T_2 = 454.8 K

Fluid system

Hydraulic press

- The hydraulic press is a device used for lifting heavy weights by the application of a much smaller force.
- It is based on Pascal's law, which states that the intensity of pressure in a static fluid is transmitted equally in all direction



Construction

- It consists of two cylinders of different diameter.
- One of the cylinders is of large diameter and contains a ram.
- The other cylinder is of smaller diameter and contains a plunger.
- The two cylinders connected by pipe.
- The cylinder and pipe contain a liquid through which pressure is transmitted.

Working

- When a small force F is applied on the plunger in the downward direction,
- A pressure is produced on the liquid in contact with the plunger.
- This pressure is equally transmitted equally in all direction and acts on the ram in the upward direction.
- The heavier weight placed on the ram is then lifted up.

Let

W = Weight to be lifted

F = force applied on the plunger

A =Area of ram

a = area of plunger

p = pressure intensity produced by force F

$$= \frac{Force\ (F)}{Area\ of\ plunger} = \frac{F}{a}$$

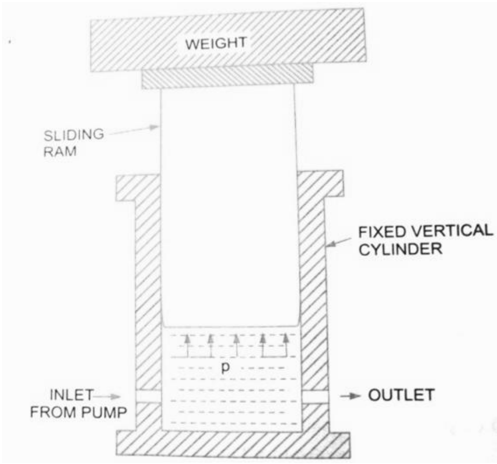
Pressure intensity of ram (p) = $\frac{W}{A}$

Equating the pressure intensity, $\frac{F}{a} = \frac{W}{A}$

$$\boxed{\mathbf{W = \frac{F}{a} \times A}}$$

Hydraulic accumulator

- The hydraulic accumulator is device used for storing the energy of the liquid in the form of pressure energy, which may be supplied for any sudden or intermittent requirement.
- In case of hydraulic lift or hydraulic crane, a large amount of energy is required when lift or crane is moving upward.
- This energy is supplied from accumulator.



Construction

- It consists of fixed vertical cylinder containing a sliding ram.
- A heavy weight is placed on the ram.
- The inlet of the cylinder connected to the pump.
- The outlet of the cylinder is connected to the machine

Working

- The ram is at the lowermost position in the beginning.
- The pump supplies water under pressure continuously.
- If the water under pressure is not required by the machine, the water under pressure will be stored in the cylinder.
- This will raise the ram on which a heavy weight is placed.
- When the ram is at uppermost position, the cylinder is full of water and accumulator is stored the maximum amount of pressure energy.
- When they require a large amount of energy, the hydraulic accumulator will supply this energy and ram will move in downward direction.

Capacity of the accumulator

- It is defined as the maximum amount of hydraulic energy stored in the accumulator.

  Weight = P x A

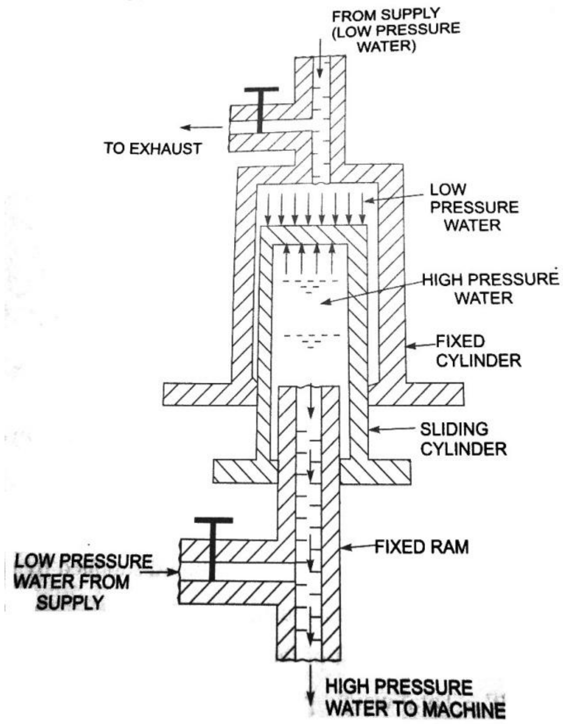
  $\boxed{\text{Capacity} = P \times A \times L}$

  Where, p = intensity pressure, A = area of the sliding ram, L = stroke or lift of the ram.

  Volume = A x L

Hydraulic intensifier

- The device which is used to increase the intensity of pressure of water by means of hydraulic energy available from large amount of water at a low pressure is called hydraulic intensifier.
- Such a device is needed when the hydraulic machines such as hydraulic press requires water at very high pressure which cannot be obtained from the main supply directly.



Construction:

- It consists of fixed ram through which the water, under a high pressure, flows to the machine.
- A hollow inverted sliding cylinder, containing water under high pressure, is mounted over the fixed ram.
- The inverted sliding cylinder is surrounded by another fixed inverted cylinder which contains water from the main supply at a low pressure.

Working

- A large quantity of water at low pressure from supply enters the inverted fixed cylinder.
- The weight of this water pressures the sliding cylinder in the downward direction.
- The water in the sliding cylinder gets compressed due to the downward movement of the sliding cylinder and its pressure is thus increased.
- The high pressure water is forced out of the sliding cylinder through the fixed ram, to the machine.

Let,

p = intensity of pressure of water supply to the fixed cylinder
A = external area of the sliding cylinder
a = area of the end of the fixed ram
P^* = intensity of pressure of water in the sliding cylinder

Force exerted by low pressure water = p x A

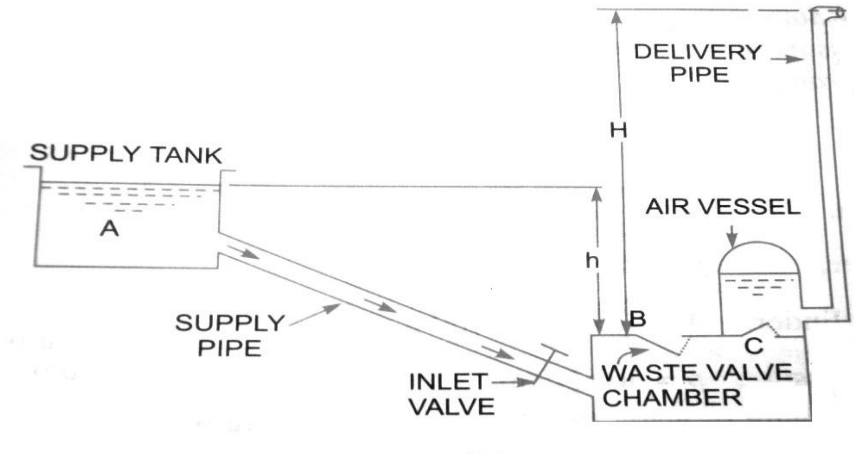
Force exerted by high pressure water = P^* x a

Equating the force,

$$\boxed{p \times A = P^* \times a}$$

Hydraulic Ram

- The hydraulic ram is a pump which raises water without any external power for its operation.
- When large quantity of water is available at small height, a small quantity of water is raised to a greater height with the help of hydraulic ram.
- It works under the principle of hydraulic hammer.



Construction

- It consists of supply tank which large quantity of water is available at small height.
- A supply pipe connected in between the supply tank and the chamber
- Inlet valve is at the end of the supply pipe
- Delivery pipe is connected to the chamber having a small quantity of water at a greater height.

Working

- When the inlet valve fitted to the supply pipe is opened, water flows from supply tank to chamber, which has two valves B and C.
- The valve B is called waste valve and valve C is called delivery valve. The valve C is fitted to the air vessel.
- As the water is coming into the chamber from supply tank, the level of water raises in the chamber and waste valve B moving upward.
- A stage comes, when the waste valve B suddenly closes.
- This sudden closure of waste valve creates high pressure inside the chamber. This high pressure opens the delivery valve C.
- The water from the chamber enters the air vessel and compresses the air inside the air vessel.
- This compressed air exerts force on the air vessel and small quantity of water is raised to greater height.
- When the water in the chamber loses its momentum, the waste valve B opens in downward direction and the flow of water from supply tank starts flowing to the chamber and cycle will be repeated.

W = weight of water is flowing per second into the chamber

w = weight of water raised per second

h = height of water in the supply tank above the chamber

H = height of water raised from the chamber.

Energy supplied by the supply tank to ram = W x h

Energy delivered by the ram = w x H

D' Aubuisson's Efficiency = $\frac{w\ X\ H}{W\ X\ h} = \frac{q\ X\ H}{Q\ X\ h}$

Rankine Efficiency = $\frac{w\ X(H-h)}{(W-w)\ X\ h} = \frac{q\ X(H-h)}{(Q-q)\ X\ h}$

Where q = discharge of delivery pipe

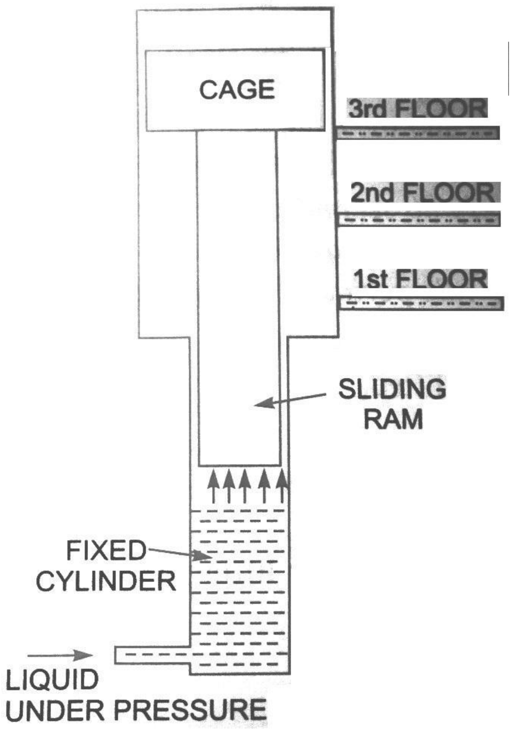
Q = discharge through supply pipe

Hydraulic Lift

The hydraulic lift is a device used for carrying passenger or goods from one floor to another in multi-storeyed building. The hydraulic lifts are two types, namely

a. Direct acting hydraulic lift
b. Suspended hydraulic lift

a. Direct acting hydraulic lift



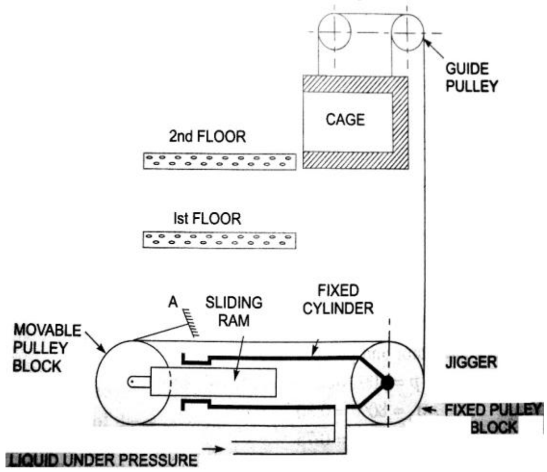
Construction:

- It consists of a ram, sliding in fixed cylinder.
- At the top of the sliding ram, cage is fitted.

Working:

- The liquid under pressure flows into the fixed cylinder.
- This liquid exerts force on the sliding ram, which moves vertically up and thus raises the cage to the required height.
  - The cage is moving down by removing the liquid from the fixed cylinder

b. Suspended hydraulic lift



Construction:

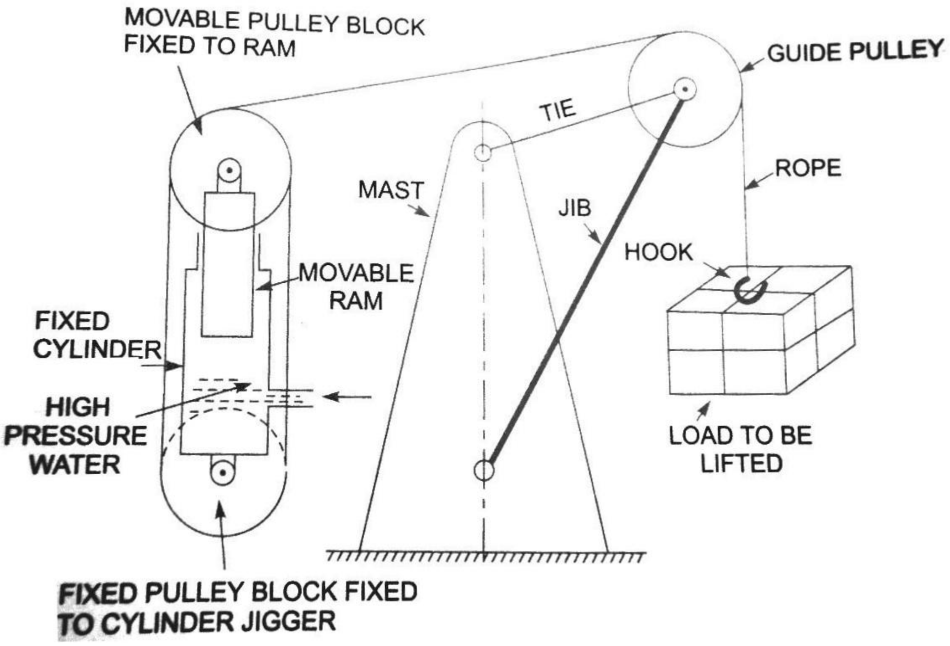
- It consists of cage, which is suspended from a wire rope.
- A jigger consisting of fixed cylinder, a sliding ram and a set of two pulley blocks is provided at the foot
- One of the pulley blocks is movable and other is fixed.
- Movable pulley is connected at the end of the sliding ram.

Working:

- When water under high pressure is admitted into the fixed cylinder of the jigger, the sliding ram is forced to move towards left.
- Thus increase the distance between two pulley blocks.
- The wire rope connected to the cage is pulled and cage is lifted.
- When the water is taken away from the cylinder the cage is down, due decrease the distance from the two pulleys.

Hydraulic crane:

- Hydraulic crane is a device, used for raising or transferring heavy loads.
- It is widely used in workshop, ware house and dock sidings.



Construction:

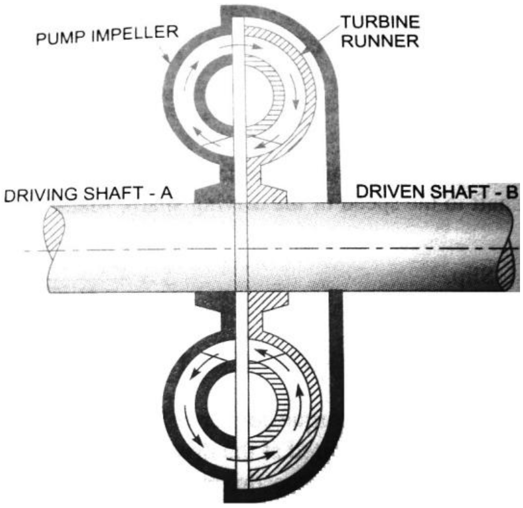
- It consists of mast, tie, jib, guide pulley and a jigger.
- The jib and tie are attached to the mast.
- The jigger which consists of a movable ram sliding in the fixed cylinder is used for lifting or lowering the heavy loads.
- One end of the ram is contact with water and the other end is connected to a set of movable pulley block.
- Another pulley block is called the fixed pulley block is attached to the fixed cylinder, is not having any movement
- The wire rope, one end is connected to movable pulley, other is connected to hook through fixed pulley and guide pulley.

Working:

- The water under high pressure is admitted into the cylinder of the jigger.
- This water forces the sliding ram to move vertically up.
- Due to the movement of the ram in the vertical up direction, the movable pulley block attached to the ram also moves upward.
- This increases the distance between two pulley blocks and hence the wire passing over the guide pulley is pulled by the jigger. This raises the load attached to the hook.

Fluid or Hydraulic coupling

- It is device used for transmitting power from driving shaft to driven shaft with the help of fluid.
- There is no mechanical connection between two shafts.



Construction:

- It consists of radial pump impeller mounted on the driving shaft A and radial flow reaction turbine mounted on the driven shaft B.
- Both the impeller and runner are in identical shape and they together form a casing which is completely enclosed and filled with oil.

Working:

- In the beginning, both the shafts A and B are at rest. When the driving shaft A is rotated, the oil starts moving from the inner radius to the outer radius of the pump impeller.
- The pressure energy and kinetic energy of the oil get increases at the outer radius of the pump impeller.
- This oil of increased energy enters the runner of the reaction turbine at the outer radius of the turbine runner and flows inwardly to the inner radius of the turbine runner.
- The oil, while flowing through the runner, transfers its energy to the blades of the runner and makes the runner to rotate.
- The oil, from the runner flows back to the pump impeller, thus having a continuous circulation.

Efficiency of the fluid coupling = $\frac{power\ output}{power\ input}$

$= \frac{power\ transmitted\ to\ the\ shaft\ B}{power\ Available\ to\ the\ shaft\ A}$

Power at any shaft = $\frac{2\pi NT}{60}$

Let N_A = speed of shaft A,

N_B = speed of shaft B,

T_A = Torque at shaft A,

T_B = Torque at shaft B.

Efficiency = $(N_B \times T_B) / (N_A \times T_A)$

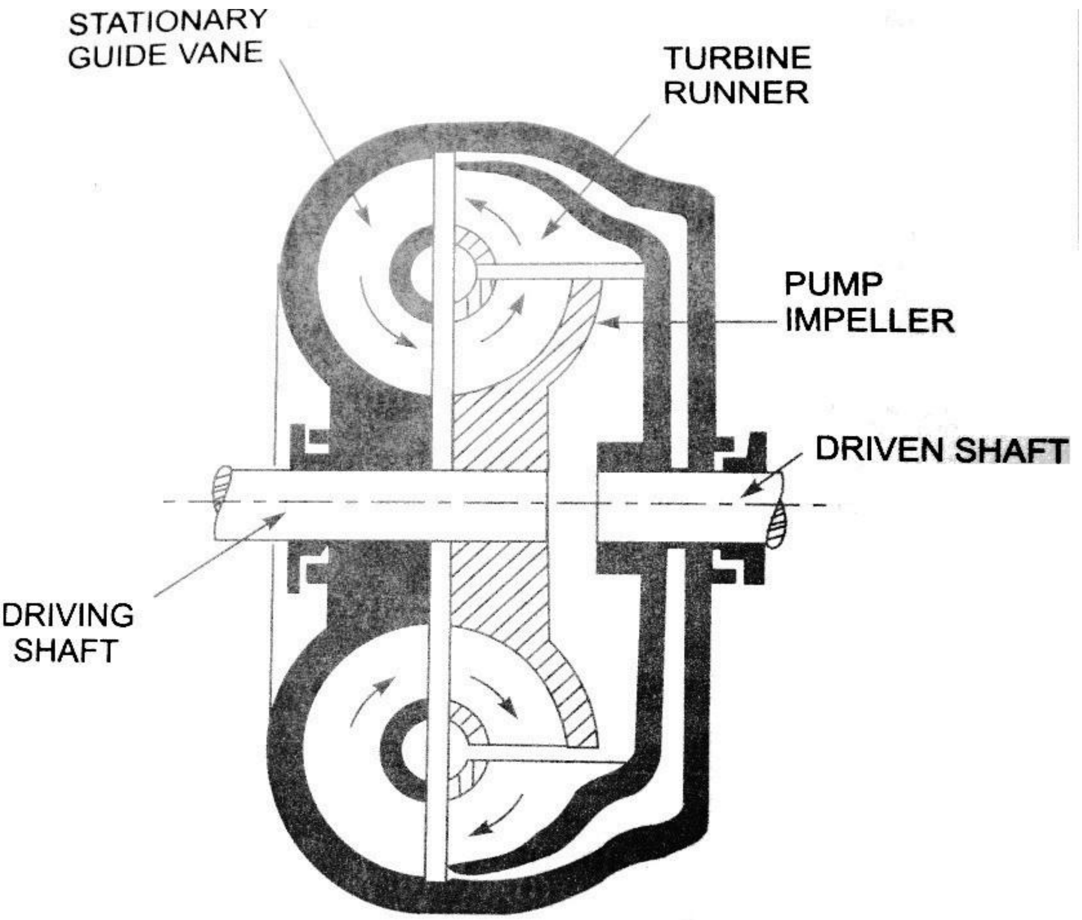
$T_A = T_B$ (torque transmitted is same)

$\boxed{\text{Efficiency} = N_B / N_A}$

$\boxed{\text{Slip} = (N_A - N_B) / N_A}$

Hydraulic Torque Converter

- It is a device used for transmitting increased torque at the driven shaft.
- The torque transmitted at the driven shaft may be more or less than the torque available at the driving shaft.
- The torque at the driven shaft may be increased about five times the torque available at the driving shaft with an efficiency of 90 %.



Construction:

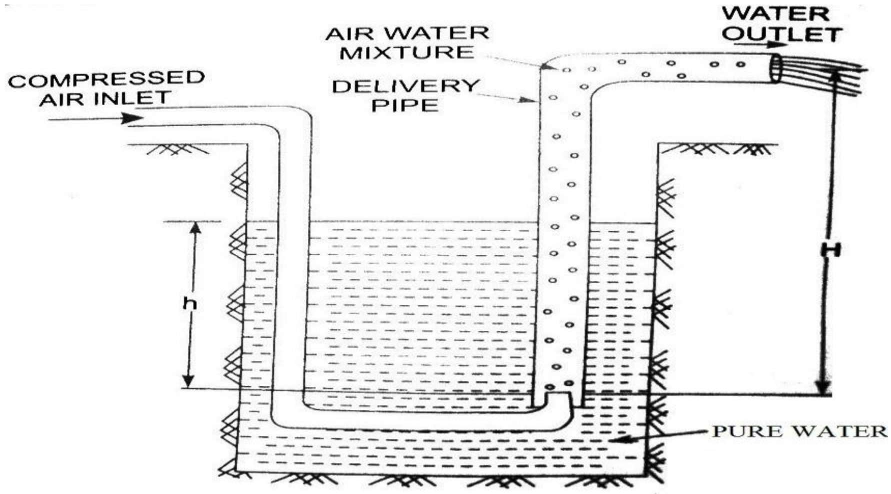
- It consists of driving shaft and driven shaft.
- Stationary guide vanes, pump impeller and turbine impeller

Working

- The power at any shaft is proportional to the product of the torque and the speed of the shaft.
- Hence if torque at the driven shaft is to be increased, the corresponding value of the speed at the same shaft should be decreased.
- The speed of the driven shaft is decreased by decreasing the velocity of the oil, which is allowed to flow from the pump impeller to the turbine runner and then through stationary guide vane.
- Due to decrease in speed at the driven shaft, the torque increases

The Air Lift Pump:

- The air lift pump is a device which is used for lifting water from well or sump by using compressed air.
- The compress air is made to mix with water. The density of the mixture of air and water is reduced.
- The density of the mixture is much less than that of pure water. Hence a very small column of pure water can balance a very long column of air water mixture.
- Thus is the principle on which the airlift pump works.



Construction:

- It consists of delivery pipe having air water mixture.
- Pure water was in a fixed tank, compressed air at inlet pipe and water at outlet pipe.

Working:

- The compressed air is introduced through one or more nozzles at the foot of the delivery pipe, which is fixed in the well from which water is to be lifted.
- In the delivery pipe, a mixture of air and water is formed.
- The density of air water mixtures becomes very less as compared to the density of pure water.
- Hence a small column of pure water will balance a very long column of air water mixture.
- This air water mixture will be discharged out of delivery pipe. The flow will continue as long as there is supply of compressed air.

Advantage:

- The air lift pump is not having any moving parts below water level and hence there are no chances of suspended solid particles damaging the pump.
- This pump can raise more water through a bore hole of given diameter than any other pump

Disadvantage:

- The efficiency of the pump is low only 20 to 40 %.
- The discharge is minimum compared to other pump.

FLUID SYSTEM

1. A hydraulic press has a ram of 300 mm diameter and a plunger of 45 mm diameter. Find the weight lifted by the hydraulic press when the force applied at the plunger is 50 N. If the stroke of the plunger is 100 mm, find the distance travelled by the weight in 100 strokes. Determine the work done during 100 strokes.

Given:

D = 0.3 m

d = 0.045 m

F = 50 N

Stroke of the plunger(X) = 0.1 m

No of strokes = 100

To Find:

W =?

W.D =?

Solution:

$A = \frac{\pi}{4} D^2 = 0.07068\ m^2$

$a = \frac{\pi}{4} d^2 = 0.00159\ m^2$

$W = \frac{F}{a} \times A =$ **2222.64 N**

Let distance moved by the ram = Y

A x Y = a x X

Y = 0.00225 m

Distance moved in 100 strokes = 100 x Y = **0.225 m**

W.D = Weight x Distance moved = **500 Nm**

---

2. A hydraulic press has a ram of 200 mm diameter and a plunger of 30 mm diameter. It is used to lifting a weight of 3KN. Find the force required at the plunger. If a lever is used for applying force on the plunger, find the force required at the end of the lever if the ratio l/L is 1/10.

Given:

D = 0.2 m

d = 0.03 m

W = 3000 N

$\frac{l}{L} = \frac{1}{10}$

To Find:

F =?

F' =?

Solution:

$A = \frac{\pi}{4} D^2 = 0.0314\ m^2$

$a = \frac{\pi}{4} d^2 = 7.068 \times 10^{-4}\ m^2$

$F = \frac{W}{A} \times a =$ **67.52 N**

$F' = \frac{W}{A} \times a \times \frac{l}{L} =$ **6.752 N**

---

3. A hydraulic press has a ram of 150 mm diameter and a plunger of 20 mm diameter. The stroke of the plunger is 200 mm and weight lifted is 800 N. If the distance moved by the weight is 1m in 20 min determines: (a) The force applied on the plunger, (b) Power required to drive the plunger, (c) Number of strokes performed by the plunger.

Given:

D = 0.15 m

d = 0.02 m

W = 800 N

Stroke of the plunger(X) = 0.2 m

Distance moved by the weight (Y) = 1 m

Time = 20 min = 120 seconds

To Find:

F =?

P =?

No of strokes performed by the plunger =?

Solution:

$A = \frac{\pi}{4} D^2 = 0.01767\ m^2$

$a = \frac{\pi}{4} d^2 = 3.1416 \times 10^{-4}\ m^2$

a. Force applied on the plunger

$F = \frac{W}{A} \times a =$ **14.22 N**

b. Power required to drive the plunger:

$W.D = \frac{\text{Weight x Distance moved}}{\text{Time}} = 0.6667$ Nm/s

$\text{Power} = \frac{W.D}{1000} =$ **0.000667 kW**

c. **No of strokes performed by the plunger:**

$\text{No of strokes} = \frac{a \times X}{A \times Y} =$ **281**

---

4. Determine the length of stroke for an accumulator having a displacement of 115 litres. The diameter of the plunger is 350 mm.

Given:

$V = 0.115\ m^3$

D= 0.350 m

To Find:

L =?

Solution:

$A = \frac{\pi}{4} D^2 = 0.09621\ m^2$

V = A x L

L = 1.195 m

5. The water is supplied at a pressure of 14 N/cm^2 to an accumulator, having a ram of diameter 1.5 m. If the total lift of the ram is 8 m, determine: (a) The capacity of the accumulator, (b) Total weight placed on the ram.

Given:

$p = 14 \times 10^4\ N/m^2$

D = 1.5 m

L = 8 m

To find:

Capacity =?

Total weight =?

Solution:

a. **Capacity of the accumulator:**

= p x A x L = **1979 kNm**

b. **Total weight placed on the ram:**

W = p x L = **247380 N**

---

6. Total weight (including the self weight of ram) placed on the sliding ram of a hydraulic accumulator is 40 KN. The diameter of the ram is 500 mm. If the frictional resistance against the movement of the ram is 5% of the total weight, determine the intensity of pressure of water when: (a) The ram is moving up with a uniform velocity, (b) The ram is moving down with uniform velocity.

Given:

W = 40000 N

D = 0.5 m

Frictional resistance = 0.05

To find:

Intensity pressure- when ram moving up =?

Intensity pressure- when ram moving down =?

Solution:

a. **Intensity pressure when ram moving up:**

Actual weight when ram moving up

= W x 1.05 = 42000 N

$W_{act} = p \times A$

$A = \frac{\pi}{4} D^2 = 0.1963\ m^2$

P = $\mathbf{21.4 \times 10^4\ N/m^2}$

b. **Intensity pressure when ram moving down:**

Actual weight when ram moving down

= W x 0.95 = 38000 N

$W_{act} = p \times A$

$A = \frac{\pi}{4} D^2 = 0.1963\ m^2$

P = $\mathbf{19.35 \times 10^4\ N/m^2}$

---

7. An accumulator is loaded with 40 KN weight. The ram has a diameter of 30 cm and stroke of 6 m. Its friction may be taken as 5%. It takes 2 min to **fall through** its full stroke. Find the total work supplied and power delivered to the hydraulic appliance by the accumulator, when 7.5 lit/s is being delivered by a pump, while the accumulator descends with the stated velocity.

Given:

W = 40000 N

D = 0.3 m

Distance moved (L) = 6 m

$Q = 0.0075\ m^3/s$

Frictional resistance = 0.05

Time = 120 sec

To find:

Total Work =?

Power Delivered to the appliance=?

Solution:

Work of accumulator

Actual weight = W x 0.95 = 38000 N

$$W.D_{acc} = \frac{\text{Actual Weight x Distance moved}}{\text{Time}} = 1900\ Nm/s$$

Pump Work

$W_{act} = p \times A$

$A = \frac{\pi}{4} D^2 = 0.07068\ m^2$

$p = 542857\ N/m^2$

$W.D_{pump} = p \times Q = 4071.35\ Nm/s$

a. Total Work = $W.D_{acc} + W.D_{pump}$ = 5971.25 Nm/s

b. Power delivered = Total Work / 1000 = 5.971 kW

8. An accumulator has a ram of diameter 250 mm and a lift of 8 m. The total weight on accumulator is 70 KN. The packing friction is 5% of the load on the ram. Find the power delivered to the machine if ram **falls through** the full height in 100 sec and at the same time the pumps are delivering 0.028 m^3 /s through the accumulator.

Given:

W = 70000 N

D = 0.25 m

Distance moved (L) = 8 m

Q = 0.028 m^3/s

Frictional resistance = 0.05

Time = 100 sec

To find:

Power Delivered to the appliance=?

Solution:

Work of accumulator

Actual weight = W x 0.95 = 66500 N

$$W.D_{acc} = \frac{\text{Actual Weight x Distance moved}}{\text{Time}} = 5320 \text{ Nm/s}$$

Pump Work

$W_{act} = p \times A$

$A = \frac{\pi}{4} D^2 = 0.049 \text{ m}^2$

$p = 1354727 \text{ N/m}^2$

$W.D_{pump} = p \times Q = 37932$ Nm/s

a. Total Work = $W.D_{acc} + W.D_{pump}$ = 43252 Nm/s

b. Power delivered = Total Work / 1000 = 43.252 kW

---

9. The diameters of fixed ram and fixed cylinder of an intensifier are 8 cm and 20 cm respectively. If the pressure of the water supplied to the fixed cylinder is 300 N/cm ^{2}, find the pressure of the water flowing through the fixed ram.

Given:

D = 0.2 m

d = 0.08 m

$p = 300 \times 10^4 \text{ N/m}^2$

To Find:

$P^* = ?$

Solution:

$A = \frac{\pi}{4} D^2 = 0.0314 \text{ m}^2$

$a = \frac{\pi}{4} d^2 = 5.026 \times 10^3 \text{ m}^2$

$P^* = \frac{p}{A} \times a =$ **1875×10^4 N/m^2**

---

10. The pressure intensity of water supplied to an intensifier is 20 N/cm ^{2} while the pressure intensity of water leaving the intensifier is 100 N/cm ^{2} . The external diameter of the sliding cylinder is 20 cm. Find the diameter of the fixed ram of the intensifier.

Given:

D = 0.2 m

$p = 20 \times 10^4 \text{ N/m}^2$

$P^* = 100 \times 10^4 \text{ N/m}^2$

To Find:

d =?

Solution:

$A = \frac{\pi}{4} D^2 = 0.0314 \text{ m}^2$

$P^* = \frac{p}{A} \times a$

$a = \frac{\pi}{4} d^2$

d = 8.94 cm

---

11. The water is supplied at the rate of 0.02m^3/s from a height of 3 m to a hydraulic ram which raises 0.002m^3/s to a height of 20 m from the ram. Determine D' Aubuisson's and Rankine's efficiencies of the hydraulic ram.

Given:

$Q = 0.02 \text{ m}^3/s$

h = 3 m

$q = 0.002 \text{ m}^3/s$

H = 20 m

To Find:

$\eta_{Aub} = ?$

$\eta_{Rank} = ?$

Solution:

a. D' Aubuisson's efficiency

$$\eta_{Aub} = \frac{q \times H}{Q \times h} = \mathbf{66.67\ \%}$$

b. Rankine's efficiency

$$\eta_{Rank} = \frac{q \times (H-h)}{(Q-q) \times h} = \mathbf{62.96\ \%}$$

12. The water is supplied at the rate of 3000 litres per minute from a height of 4m to a hydraulic ram which raises 300litres/minute to height of 30m from the ram. The length and diameter of the delivery pipe is 100m and 70mm respectively. Calculate the efficiency of the hydraulic ram if the co-efficient of friction f=0.009.

Given:

$Q = 0.05\ m^3/s$

$h = 4\ m$

$q = 0.005\ m^3/s$

$H = 30\ m$

$L = 100\ m$

$d = 0.07\ m$

$f = 0.009$

To Find:

$\eta_{Aub} = ?$

$\eta_{Rank} = ?$

Solution:

Due to Effect of friction

Effective head $= H + h_f$

$h_f = \frac{4fLV^2}{2gd}$

$Q = a\ V$

$a = \frac{\pi}{4} d^2$

$V = 1.3\ m/s$

$h_f = 4.43\ m$

Effective head = **34.43 m**

a. D' Aubuisson's efficiency

$\eta_{Aub} = \frac{q \times H}{Q \times h} = \mathbf{86.07\ \%}$

b. Rankine's efficiency

$\eta_{Rank} = \frac{q \times (H-h)}{(Q-q) \times h} = \mathbf{84.53\ \%}$

---

13. A hydraulic lift is required to lift a load of 8 KN through a height of 10m, once in every 80 seconds. The speed of the lift is 0.5 m per second. Determine: (a) Power required to drive the lift (b) Working period of lift in seconds, (c) Idle period of the lift in seconds.

Given:

$W = 8000N$

Distance moved $(H) = 10\ m$

$t = 80\ s$

$v = 0.5\ m/s$

To Find:

Power required =?
Work period of lift =?
Idle period of lift =?

Solution:

a. Power required to drive the lift:

$W.D_{lift} = \frac{\text{Weight x Distance moved}}{\text{Time}} = 1000\ Nm/s$

$\text{Power} = \frac{W.D}{1000} = \mathbf{1.0\ kW}$

b. Working period of the lift:

$= \frac{\text{Height of the lift}}{\text{Velocity of the lift}} = \mathbf{20\ sec}$

c. **Idle period of the lift:**

= total time – working period of time
= **60 sec**

---

14. A hydraulic lift is required to lift a load of 12 KN through a height of 10m, once in every 1.75 minutes. The speed of the lift is 0.75 m/s during working stroke of the lift, water from accumulator and the pump at a pressure of 400 N/cm^2 is supplied to the lift. If the efficiency of the pump is 8% and that of lift is 75%, find the power required to drive the pump and the minimum capacity of the accumulator. Neglect friction losses in the pipe.

Given:

$W = 12000\ N$

Distance moved $(H) = 10\ m$

$t = 105\ s$

$v = 0.75\ m/s$

$p = 400 \times 10^4\ N/m^2$

$\eta_{Pump} = 0.80$

$\eta_{Lift} = 0.75$

To Find:

Power required to drive the pump =?
Capacity of accumulator =?

Solution:

Total Work done $= W.D_{pump} + W.D_{acc}$

= Total Weight x distance moved per second

$= W \times v = 9000\ Nm/s$

Actual W.D = Total W.D/ η_{Lift} = 12000 Nm/s

$W.D_{pump} + W.D_{acc} = 12000$ -------- (1)

Working period of the lift

$= \frac{\text{Height of the lift}}{\text{Velocity of the lift}} = 13.33$ sec

Ideal period of the lift

= Total time – working period = 91.67 sec

$W.D_{acc} = \frac{W.D_{pump} \text{ x Ideal period}}{\text{working period}}$

$W.D_{acc} = 6.876\ W.D_{pump}$ --------- (2)

Solving eqn (1) and (2)

$W.D_{pump} = 1523.6$

a. **Power required to drive the pump**

$= W.D_{pump}/1000 = 1.523$ kW

Actual power required = power required/ η_{Pump}

= **1.90375 kW**

- **Minimum capacity of the accumulator**

$= W.D_{pump}$ x Ideal time

= 1523.6 x 91.67 = **139613.4 Nm**

---

15. Find the efficiency of a hydraulic crane, which is supplied 300 litres of water under a pressure of 60 N/cm^2 for lifting a weight of 12 KN through a height of 11m.

Given:

V = 0.30 m^3

p = 60 x 10^4 N/m^2

W =12000 N

Distance moved (H) = 11 m

To Find:

Efficiency =?

Solution:

Output of the crane = Weight x distance moved

= 132000 Nm

Input of the crane = p x A x L

V = A x L

Input of the crane = 18 x 10^4 Nm

Efficiency = output/input = **73.33 %**

---

16. The efficiency of a hydraulic crane, which is supplied water under a pressure of 70 N/cm^2 for lifting a weight through a height of 10m is 60 %. If the diameter of the ram is 150mm and velocity ratio is 6, find, (a)The weight lifted by the crane, (b) the volume of water required in litres to lift the weight.

Given:

Efficiency = 0.60

p = 70 x 10^4 N/m^2

D = 0.15m

Distance moved (H) = 10 m

Velocity ratio (H/L) = 6

L = 1.667 m

To Find:

W =?

V =?

Solution:

a. **Weight lifted by the crane:**

Output of the crane = Weight x distance moved

= W x 10

$A = \frac{\pi}{4} D^2 = 0.01767\ m^2$

Input of the crane = p x A x L

= 2.06 x 10^4 Nm

Efficiency = output/input

W = 1236.9 N

b. **Volume of water:**

V = A x L = 0.02945 m^3

= **29.45 Litres**

---

17. In a hydraulic coupling the speed of the driving and driven shafts are 800 r.p.m and 780 r.p.m respectively. Find (a) the efficiency of the hydraulic coupling, (b) the slip of the coupling.

Given:

N_A = 800 r.p.m

N_B = 780 r.p.m

To Find:

Efficiency =?

Slip =?

Solution:

Efficiency = N_B/N_A

= 97.5 %

Slip = $(N_A – N_B)/N_A$

= 0.025 = **2.5 %**

18. A hydraulic crane is lifting a weight of 12000 N through a height of 12m with a speed of 18m per minute once in every 2 minutes. The efficiency of the hydraulic crane is 65% and it is working under a pressure of 500 N/cm^2 of water. The crane is fed from an accumulator to which water is supplied by a pump. Find: (a) The capacity of the cylinder of the jigger in litres. (b) The capacity of the accumulator in litres (c) Minimum power required for the pump.

Given:

W = 12000 N

Efficiency = 0.65

p = 500 x 10^4 N/m^2

Distance moved (H) = 12 m

Time = 2 min

v = 18 m/min

To Find:

Volume of cylinder =?

Volume of accumulator =?

Minimum power required for the pump =?

Solution:

(a) **The capacity of the cylinder of the jigger in litres.**

Output of the crane = Weight x distance moved

= 12000 x 12

Input of the crane = p x A x L

Efficiency = output/input

Volume = A x L = 0.0443 m^3

= 44.3 Litres

(b) **The capacity of the accumulator in litres**

Volume of accumulator = volume of cylinder x $\frac{H}{v}$

= **29.53 litres**

(c) **Minimum power required for the pump.**

Power = work input per minute/ 1000

Work input per minute = $\frac{p \text{ x } A \text{ x } L}{t}$

$= \frac{500 \text{ x } 10^4 \text{ x } 0.0443}{120}$

= 1846 Nm/min

Power = 1.856 kW