BASICS OF TURBO MACHINERY

Syllabus:

Impulse momentum principle, Euler's equation of motion, force exerted by jet on stationary and moving plates, velocity diagrams, energy conversion in turbomachinery, concept of degree of reaction, application of model testing, work done and efficiency.

Learning objectives:

- **To study the force exerted by water by jet on different vanes**

Learning outcomes:

Students will be able to

- Understand the principle on which Turbo machinery works.
- Study the behavior of various types of vanes under the action of Jet impingement.
- Derive the equations for Forces exerted by jet impingement up on the various types of vanes in motion and rest, by applying Impulse momentum equation.

Introduction to Fluid Machines

A fluid machine is a device which converts the energy stored in a fluid into mechanical energy or vice versa. The energy stored in the fluid will be in the form of potential energy, kinetic energy and intermolecular energy. The mechanical energy is transmitted by a rotating shaft. Machines using liquids (water) are termed as hydraulic machines.

Classification of Turbomachines:

- **Based on the Direction of Energy Conversion:**

  The device in which the potential, kinetic or intermolecular energy in the fluid is converted into the mechanical energy is called a *turbine*.

  The device in which the mechanical energy of the moving parts is transferred to a fluid to increase its stored energy by increasing either its potential or velocity are known as *pumps, compressors, fans or blowers*.

- **Based on the Principle of Operation:**

  Positive Displacement Machine: It is the machine whose function depends on the change in volume of the fluid within its boundary.

  This principle in general is utilized in the reciprocating motion of the piston within a cylinder while entrapping some amount of fluid in it. The machine producing mechanical energy is termed as *reciprocating engine* while the machine increasing the energy of the fluid from the mechanical energy is termed as a *reciprocating pump or compressor.*

  Rotodynamic Machines: These machines work on the principle of fluid dynamics. They are distinguished from the positive displacement machines by considering the relative motion between the fluid and the moving part of the machine.

The rotating element of the machine consisting of number of blades or vanes is known as rotor or impeller while the fixed part is known as stator.

For the turbines, the work is done by the fluid on the rotor, while in pumps and compressors the work is done by the impeller on the fluid.

These are sub-classified based on the direction of fluid flow:

- ***Axial flow machines***: The flow of the fluid is parallel to the axis of rotation of rotor.
- ***Radial flow machines***: The flow of the fluid is radial to the rotor.

  In turbines, the flow is radially inward i.e. towards the center of the rotor. Hence radial flow turbines are also called as radially inward floe turbines.

  <u>Example</u>: Francis Turbine

  In pumps and compressor, the flow is radially outward i.e. away from the center of the rotor. Hence these radially flow pumps or compressors are also called as radially ouward flow machines.

  <u>Example</u>: Centrifugal pump, Centrifugal compressor

- **Based on the Fluid Used:** The machine transferring the mechanical energy of the rotor to the energy of the fluid is termed as a ***pump***, when it uses liquid and a ***compressor*** or a ***fan*** or a ***blower*** when it uses gases.

  The ***compressor*** increases the static pressure of the gas.

  The ***fan*** or the ***blower*** causes the large flow of gas increasing the kinetic energy of the gas by utilizing the mechanical energy at the rotor. In these machines the rise in static pressure is small.

  Hydraulic Turbine: Water is used to produce mechanical energy.

  Steam Turbine**:** Steam is used to produce mechanical energy.

  Gas Turbine: Hot gases generated in combustion of fuel are used to produce mechanical energy.

Impulse Momentum Principle:

The state of rest or uniform motion of a body changes in the direction of an externally applied force, and the magnitude of the force equals the rate of change on momentum.

$$F = \frac{d(mv)}{dt} = m\frac{d(v)}{dt} + v\frac{d(m)}{dt}$$

For constant mass flow rate, $dm = 0$.

Further momentum change may occur due to change in the magnitude of velocity or in its direction or due to both. If v_1 and v_2 represent average velocities at two sections of a fluid stream and t denotes the time interval, then

$$F = m\frac{d(v)}{dt} = m\frac{(v_2 - v_1)}{t} \Rightarrow F \times t = m(v_2 - v_1)$$

The product $(F \times t)$ is called the impulse and from the above equation, the impulse equals the change in momentum.

This equation is rewritten as $F = \frac{m}{t}(v_2 - v_1) = \dot{m}(v_2 - v_1)$

where the mass flow rate of the fluid can be written as $\dot{m} = \rho Q$.

Finally the force exerted by the body on the fluid by virtue of change in the velocity of the fluid is $F = \rho Q(v_2 - v_1)$.

From Newton's third law of motion, action and reaction are equal and opposite to each other.

Therefore, the force exerted by the fluid on the body is $F = -\rho Q(v_2 - v_1) = \rho Q(v_1 - v_2)$.

This force acts in the direction of motion of the incident jet.

Force exerted by fluid jet on a stationary flat plate:

Let a jet of diameter 'd' and velocity 'v' issue from a nozzle and impinges onto a stationary flat plate held perpendicular to the flow direction.

Assumptions:

- The plate is smooth and consequently no force of friction between the plate and the jet.
- No energy loss due to impact of jet.
- Jet moves on and off the plate with same velocity.
- Negligible variation in elevation of the incoming and outgoing jets.
- Constant pressure; pressure everywhere is atmospheric.

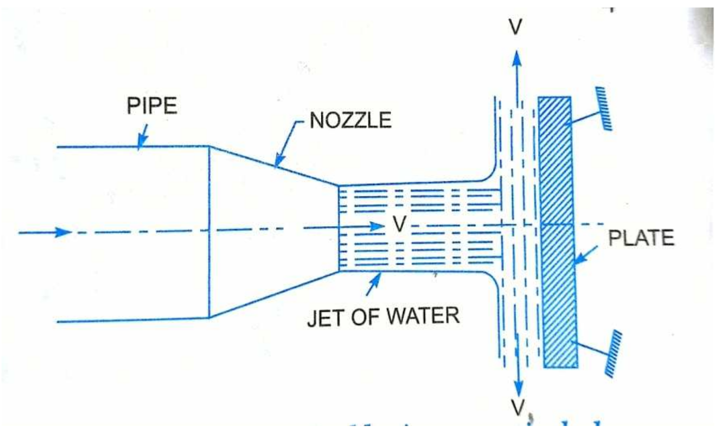


Fig. 3 Impact of jet on a flat vertical stationary plate

If there are no friction losses, the jet after striking the plate will leave with velocity 'v' tangentially i.e., the jet will get deflected through 90°.

The quantity of fluid striking the plate, $Q = a \times v$,

where cross-sectional area of jet, $a = \frac{\pi}{4} d^2$

From impulse momentum equation,

Force exerted by the jet in a direction normal to the plate is

$$F_n = \rho Q(v_1 - v_2) = \rho a v_1 (v_1 - v_2)$$

But the initial velocity is $v_1 = v$. Further the jet gets deflected through 90^0 and accordingly the component of velocity of leaving jet in a direction normal to the plate is zero i.e. $v_2 = 0$.

Therefore, $F_n = \rho a v^2$

Force exerted by a jet on a stationary inclined plate

Let a stationary smooth plate be held inclined at an angle θ to the direction of flow of jet of diameter 'd' and velocity 'v', as shown in figure. It is assumed that there exists no frictional resistance at the plate and there are no impact losses.

The discharge from the nozzle is divided in to two jets after impacting the plate.

Let Q1 be the discharge in upward direction and Q2 be the discharge in downward direction.

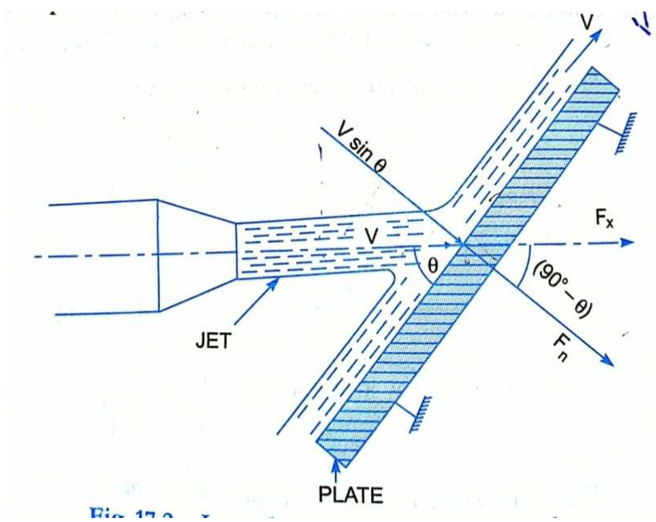


Fig. 3 Impact of jet on a flat stationary inclined plate

Initial velocity of jet in the direction normal to the plate, $v_1 = v \sin\theta$

Final velocity of jet in the direction normal to the plate, $v_2 = 0$

Normal force on the plate, $F_n = \rho a v_1 \left(v_1 - v_2\right) = \rho a v\left(v \sin\theta\right) = \rho a v^2 \sin\theta$

This normal force can be resolve into two components:

- Component parallel to the direction of jet, $F_x = F_n \sin\theta = \rho a v^2 \sin^2\theta$
- Component normal to the direction of jet, $F_y = F_n \sin\theta = \rho a v^2 \sin\theta \cos\theta$

Upon striking the stationary inclined plate, the jet splits into two streams with flow rates Q1 and Q2. Due to absence of friction, the resultant force parallel to the plate is zero. Consequently the impulse momentum equation in the direction parallel to the plate can be written as:

$$\left(\rho Q_1 v - \rho Q_2 v\right) - \rho Q v \cos\theta = 0 \Rightarrow Q_1 - Q_2 = Q\cos\theta$$

Also from continuity equation, total discharge, $Q = Q_1 + Q_2$

On solving these two equations, we get

$$Q_1 = \frac{Q}{2}\left(1+\cos\theta\right), \quad Q_2 = \frac{Q}{2}\left(1-\cos\theta\right)$$

Ratio of discharge, $\dfrac{Q_1}{Q_2} = \dfrac{1+\cos\theta}{1-\cos\theta}$

- Note:
- If $\theta = 90^0$, the plate is perpendicular to the flow of water jet. In such a case with water impinging on the vertical stationary flat plate, ratio of dicsharges is same i.e. equal discharge is observed in vertically upward and downward

directions.

Jet Striking a Symmetrical Stationary Curved Vane At Centre

Let a horizontal jet of cross sectional area ‘a’ and having velocity ‘V’ be striking a smooth stationary symmetrical curved vane at its centre on concave side. Let θ be the angle between the two tangents drawn to the vane at its out let tips as shown.

After striking the vane the jet will divide itself and the each divided jets glides over the vane and leaves the vanes at its both ends.

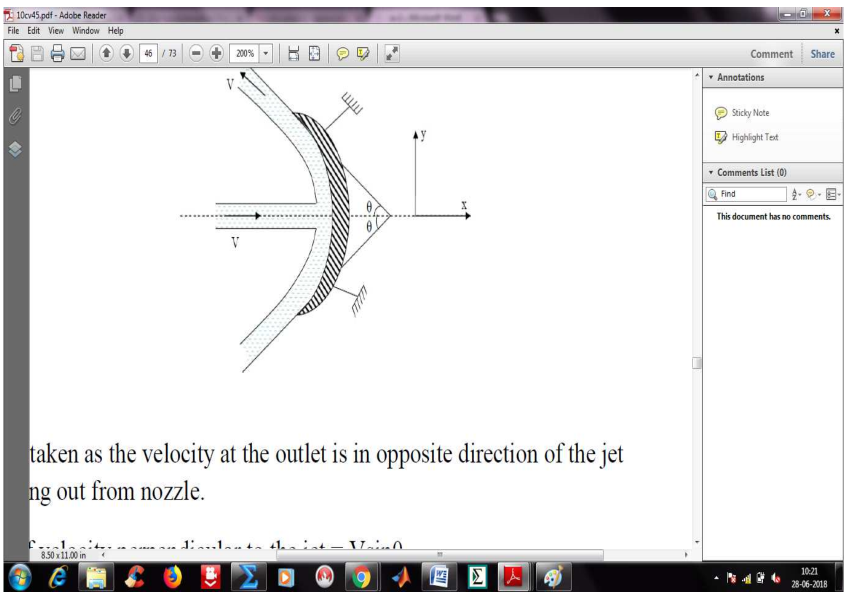


Fig. 4 Jet striking a symmetrical vane at the centre

Let the flow rate entering be ‘Q’ As the vane is symmetrical, the divided two jets at exits will have the same discharge and equal to Q/2.

Force exerted by plate on fluid in x-direction is given by,

$$F_x = \rho \frac{Q}{2}\left(-\cos\theta V - \cos\theta V\right) - \rho Q V = -\rho Q V\left(1+\cos\theta\right)$$

Therefore force exerted by fluid on the vane in x-direction is

$$F_X = -F_x = \rho Q V\left(1+\cos\theta\right)$$

Similarly, Force exerted by the plate on fluid in y-direction is given by

$$F_y = \rho \frac{Q}{2}\left(V\sin\theta - V\sin\theta\right) - 0 = 0$$

Therefore force exerted by fluid on vane in y-direction is zero.

Resultant force, $F_R = \sqrt{F_X{}^2 + F_Y{}^2}$

$$F_R = F_X = \rho Q V[1+\cos\theta] = \rho a V^2[1+\cos\theta]$$

Note:

- For flat plate, θ=90°; $F_R = \rho a V^2$
- For a symmetrical curved plate, the resultant force is $F_R = \rho a V^2[1+\cos\theta]$

  It can be observed that $\rho a V^2[1+\cos\theta] > \rho a V^2$. Hence the force acting on a curved vane is more than the force acting on a flat plate.

Jet striking an unsymmetrical stationary curved vane, tangentially at one of the tips

Let a jet of cross-sectional area ‘a’ and velocity ‘v’, strike tangentially at one of the tips, called the inlet tip, of a smooth curved vane as shown in figure. Let the tangent at inlet tip makes an angle θ with the horizontal and that at the outlet tip make an angle Ø with the axis.

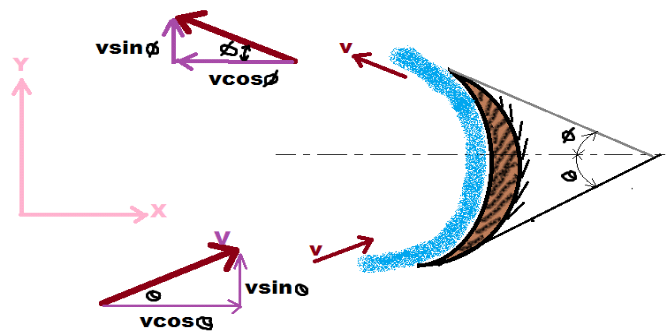


Fig. 5 Jet striking an unsymmetrical vane at the tip

Now from Impulse-Momentum principle,

Force exerted by the fluid on the vane in x-direction =

[Momentum at entry –Momentum at exit] in x-direction

$$F_x = \rho Q\left[V\cos\theta - \left(-V\cos\phi\right)\right] = \rho QV[\cos\theta + \cos\phi]$$

Therefore force exerted by fluid on vane in x-direction, $F_x = \rho a V^2[\cos\theta + \cos\phi]$

Force exerted by the fluid on the vane in y-direction =

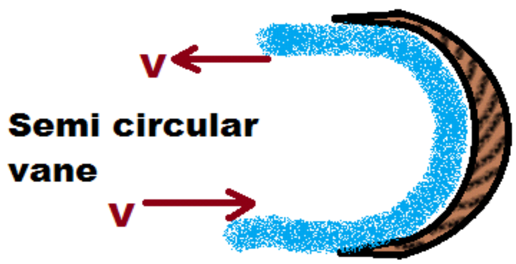
[Momentum at entry –Momentum at exit] in y-direction

$$F_y = \rho Q\left[V\sin\theta - V\sin\phi\right] = \rho QV[\sin\phi - \sin\theta]$$

Therefore force exerted by fluid on vane in y-direction, $F_y = \rho a V^2[\sin\theta - \sin\phi]$

Resultant force, $F = \sqrt{F_x^{\,2} + F_y^{\,2}}$

Direction of the resultant force is given by $\beta = \tan^{-1}\left(\dfrac{F_y}{F_x}\right)$



Special Case

- If $\phi = \theta = 0^0$, the vane becomes Semi circular & the incoming and outgoing jets are parallel to each other (x-axis) and opposite in direction. In this case y-component of force is zero.

  $F_y = 0$

  Fig. 6 Semi-circular vane

  Hence force acting on the semi-circular vane in x-direction is $F_x = 2\rho a V^2$.

__**Note**__: Effective velocity is the velocity of jet relative to the plate, in the direction of the jet strike.

Suppose when the vane is allowed to move with a velocity 'u' in the direction of jet, effective velocity of incoming jet is (V-u), as relative velocity w.r.t plate is (V-u).

Force exerted by jet on moving flat vertical vane:

Let a jet of cross sectional area 'a' moving with absolute velocity 'V' strikes the flat vertical vane moving with velocity 'u' as shown in fig. 7.

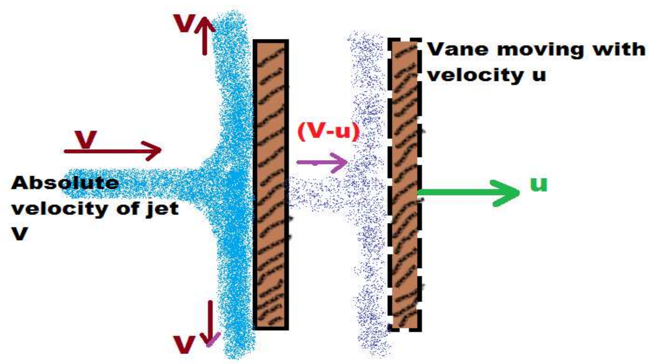


Fig. 7 jet striking on a moving flat plate

Since both the Moving vane and jet are in same direction, relative velocity of jet w.r.t vane is (V-u).

The mass of fluid striking the plate, $Q = \rho a\left(V - u\right)$.

Therefore force exerted by jet on this moving vane along x-direction i.e. normal to the plate is

$$F_x = \rho a(V - u)^2$$

Since the jet leaves along the plate vertically parallel to the surface, the force exerted is zero.

y-direction is $F_y = 0$

Resultant force, $F_R = \rho a(V - u)^2$

Work done = Resultant force × Displacement of the plate per unit time (i.e. plate velocity)

$$W = F_R \times u = \rho a(V - u)^2 u$$

Force Exerted By Jet on Moving Flat Inclined Vane

Let a stationary smooth plate be held inclined at an angle θ to the direction of flow of jet of diameter ‘d’ and velocity ‘V’, as shown in figure.

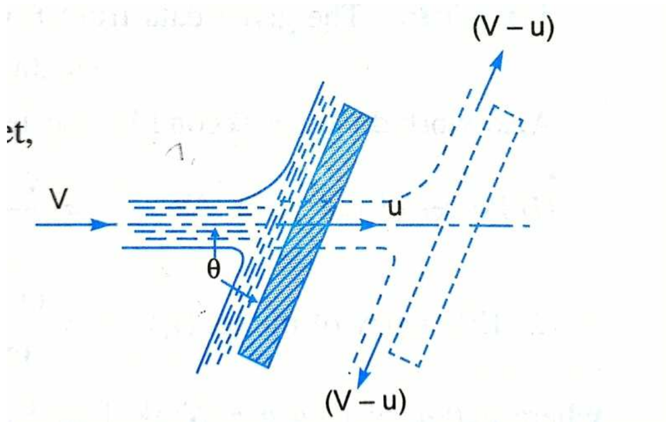


Fig. 8 Jet striking a moving inclined plate

Since both the Moving vane and jet are in same direction, relative velocity of jet w.r.t vane is (V-u).

The mass of fluid striking the plate, $Q = \rho a\left(V-u\right)$.

Initial velocity of jet in the direction normal to the plate, $v_1 = \left(V-u\right)\sin\theta$

Final velocity of jet in the direction normal to the plate, $v_2 = 0$

Normal force on the plate,

$$F_n = \rho a v_1\left(v_1 - v_2\right) = \rho a\left(V-u\right)\left(\left(V-u\right)\sin\theta - 0\right) = \rho a\left(V-u\right)^2 \sin\theta$$

This normal force can be resolve into two components:

- Component parallel to the direction of jet, $F_x = F_n \sin\theta = \rho a\left(V-u\right)^2 \sin^2\theta$
- Component normal to the direction of jet, $F_y = F_n \cos\theta = \rho a\left(V-u\right)^2 \sin\theta\cos\theta$

Work done = Resultant force × Displacement of the plate per unit time (i.e. plate velocity)

$$W = F_n \times u = \rho a u (V-u)^2 \sin\theta$$

Force on the curved plate when the plate is moving in the direction of jet

Consider a curved vane moving with velocity 'u' as shown in figure. The jet of diameter'd' impinges on the moving vane at the centre with velocity 'V'.

Hence relative velocity = V-u.

If the plate is smooth and the loss of energy due to impact of jet is zero, then the velocity with which the jet will be leaving the curved vane is = V-u.

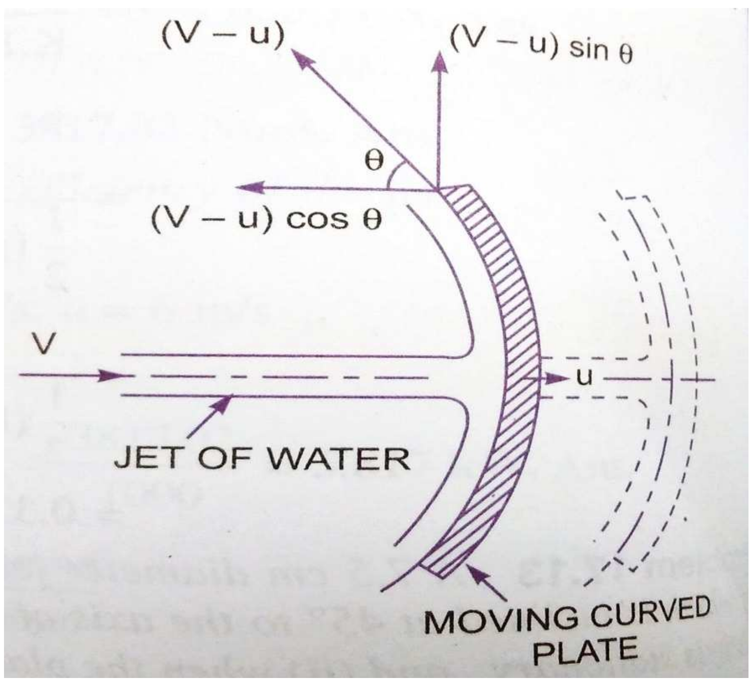


Fig. 9 jet impinging on a moving curved vane

From the Impulse-Momentum principle,

Force exerted by Jet on the curved plate in the direction of jet = rate of change of momentum

$$F_x = \rho a(V-u)\left[(V-u)-(-(\mathrm{V}-\mathrm{u})\cos\theta)\right]$$
$$F_x = \rho a(V-u)^2\left[1+\cos\theta\right]$$

Work done by the jet on the vane per second, = Force exerted x velocity of the vane

$$W = F_x \times u = \rho a u(V-u)^2\left[1+\cos\theta\right]$$

Kinetic energy supplied by the jet per second (Input energy),

$$I.E = \frac{1}{2}\dot{m}V^2 = \frac{1}{2}\rho Q V^2 = \frac{1}{2}(\rho a V)V^2 = \frac{1}{2}(\rho a V^3)$$

Efficiency of the jet, $\eta = \dfrac{work\ \ done\ \ by\ \ the\ \ jet}{input\ \ K.E\ \ of\ \ \ jet} = \dfrac{W}{I.E} = \dfrac{\rho a u(V-u)^2\left[1+\cos\theta\right]}{\frac{1}{2}(\rho a V^3)}$

On simplification, we get

$$\eta_{\max} = \frac{2uV(V-u)\left[1+\cos\theta\right]}{V^3} = \frac{2u(2u-u)\left[1+\cos\theta\right]}{(2u)^2} = \frac{1}{2}(1+\cos\theta)$$

$$\eta = \frac{2u(V-u)^2\left[1+\cos\theta\right]}{V^3}$$

For maximum efficiency of the jet, $\dfrac{d\eta}{du} = 0$

On simplification, we get

$$\frac{2\left[1+\cos\theta\right]}{V^3}\left[V^2-4uV+3u^2\right]=0$$

$$V^2-4uV+3u^2=0\Rightarrow V=u, V=3u$$

Case (i): If V = u, then the efficiency of jet is zero. This indicates that the jet will never hit the plate and hence force acting on the plate is zero.

Case (ii): If V = 3u, then the velocity of jet is higher than the velocity of plate. Hence the jet strikes the plate and the plate experiences a net force acting on it. At this condition, the maximum efficiency of the jet,

$$\eta_{\max}=\frac{2u\left(V-u\right)^2\left[1+\cos\theta\right]}{V^3}=\frac{2u\left(3u-u\right)^2\left[1+\cos\theta\right]}{\left(3u\right)^3}=\frac{8}{27}\left(1+\cos\theta\right)$$

For semi-circular vane, $\theta=0^0$. In this case, maximum efficiency is

$$\eta_{\max}=\frac{8}{27}\left(1+\cos\theta\right)=\frac{16}{27}=59.26\%$$

Jet striking a series of Curved vanes:

Jet striking a single moving vane is not feasible practically because the distance between the vane and the nozzle outlet from which the jet is issued will be constantly increasing at the rate of ‘u’ per second. This requires a continuous lengthening of the jet.

This difficulty is overcome by the arrangement of a series of such vanes at equal spacing on the periphery of a large wheel which is capable of rotating in a vertical plane. Thus as the wheel rotates, each vane will become normal to the jet in turn, so that the entire fluid of the incoming jet issued from the nozzle will be utilized in striking the vanes.

Mass of the fluid striking the plate, $m = \rho a V$

Force exerted by the je ton the series of vanes,

$$F = \rho a V\left(\left(V-u\right)-\left(V-u\right)\cos\theta\right) = \rho a V\left(V-u\right)\left(1+\cos\theta\right)$$

Work done by the jet on the wheel per second,

$$W = F \times u = \rho a u V\left(V-u\right)\left(1+\cos\theta\right)\; W = F \times u = \rho a u V\left(V-u\right)\left(1+\cos\theta\right)$$

Kinetic energy supplied by the jet per second (Input energy),

$$I.E = \frac{1}{2}\dot{m}V^2 = \frac{1}{2}\rho Q V^2 = \frac{1}{2}\left(\rho a V\right)V^2 = \frac{1}{2}\left(\rho a V^3\right)$$

Efficiency of the jet, $\eta = \dfrac{work\ done\ by\ the\ jet}{input\ K.E\ of\ jet} = \dfrac{W}{I.E} = \dfrac{2u\left(V-u\right)\left[1+\cos\theta\right]}{V^3}$

On simplification, we get $\eta = \dfrac{2u\left(V-u\right)\left[1+\cos\theta\right]}{V^3}$

For maximum efficiency of the jet, $\dfrac{d\eta}{du} = 0$

On simplification, we get

$$\frac{2\left[1+\cos\theta\right]}{V^3}\left[V-2u\right] = 0$$

$$V - 2u = 0 \Rightarrow V = 2u$$

If the jet velocity is half the velocity of the vane, maximum efficiency could be obtained.

Hence maximum efficiency is obtained as

$$\eta_{\max} = \frac{2uV\left(V-u\right)\left[1+\cos\theta\right]}{V^3} = \frac{2u\left(2u-u\right)\left[1+\cos\theta\right]}{\left(2u\right)^2} = \frac{1}{2}\left(1+\cos\theta\right)$$

For semi-circular vane, $\theta=0^0$. In this case, maximum efficiency is

$$\eta_{\max} = \frac{1}{2}\left(1+\cos\theta\right) = 100\%$$

Force exerted by the jet of water on an unsymmetrical moving curved plate when the jet strikes tangentially at one of the tips:

Consider a curved vane moving with velocity ‘u’. The jet of diameter ‘d’ is striking the vane tangentially at inlet as shown in figure. Here the relative velocity will be equal to vector difference of velocity of jet and velocity of plate.

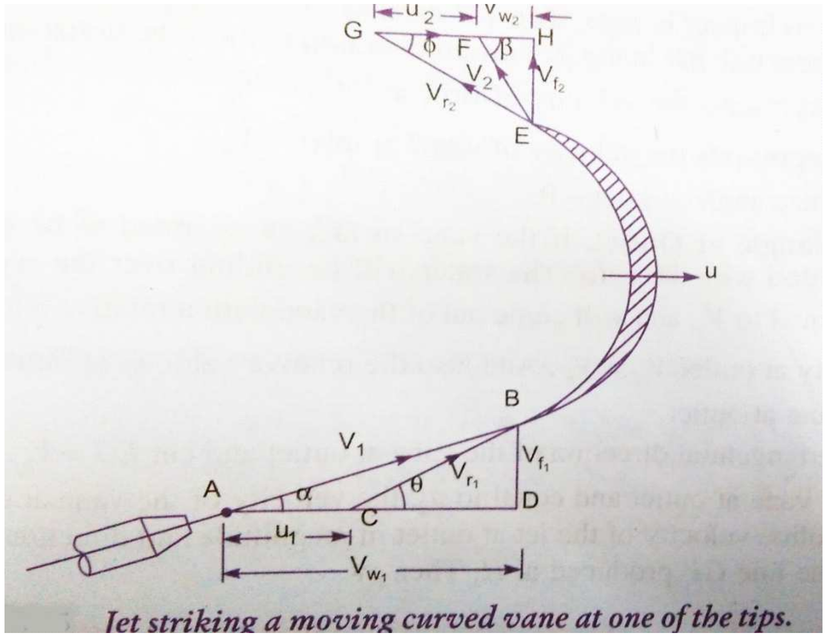


Fig. 10 Jet striking the curved vane tangentially

Let V_1 is the absolute velocity of jet.

u_1 is the velocity of plate at inlet.

V_{r1} is the relative velocity of jet and plate at inlet.

α is the angle between the direction of jet and the direction of motion of the plate.

Θ is the angle made by relative velocity with the direction of motion of vane at inlet. It is also called "angle of vane" at inlet.

V_{w1} is the component of velocity V_1 in the direction of motion of the vane at inlet. It is also known as whirl velocity at inlet.

V_{f1} is the component of velocity V_1 perpendicular to the direction of motion of vane at inlet. It is also called flow velocity at inlet.

V_2 is the absolute velocity of jet leaving the vane at the outlet.

V_{r2} is the relative velocity of jet and plate at outlet.

ϕ is the angle made by relative velocity with the direction of motion of vane at outlet.

β is the angle between the direction of jet and the direction of motion of the plate at outlet also called guide blade angle.

V_{w2} is the component of velocity V_2 in the direction of motion of the vane at outlet.

V_{f2} is the component of velocity V_2 perpendicular to the direction of motion of vane at outlet.

Velocity Triangles:

From impulse-momentum principle,

Force exerted by the jet on the vane in the direction of jet = change in momentum of the fluid

Relative velocity at inlet with which jet strikes the plate in the direction of jet is =

$$V_{r1}\cos\theta = V_{w1} - u_1 = V_{w1} - u$$

Final velocity with which jet leaves the vane in the direction of jet =

$$V_{r2}\cos\phi = V_{w2} + u_2 = V_{w2} + u \quad (\because u_1=u_2=u)$$

Therefore, $F_x = \rho a V_{r1}\left[V_{w1} - u_1 - \{-(V_{w2}+u_2)\}\right] = \rho a V_{r1}\left[V_{w1} + V_{w2}\right]$

The above equation is true only when $\beta < 90^0$.

When $\beta = 90^0$. Then $V_{w2}=0$.

$$F_x = \rho a V_{r1}\left[V_{w1}\right]$$

When β is obtuse angle $\beta > 90^0$, then $F_x = \rho a V_{r1}\left[V_{w1} - V_{w2}\right]$

In general $F_x = \rho a v_{r1}\left[v_{w1} \pm v_{w2}\right]$

Work done per sec = $W = F_x \times u = \rho a V_{r1}\left[V_{w1} \pm V_{w2}\right]\times u$

Work done per sec. per unit weight of water striking the vane per sec,

$$W = \frac{\rho a v_{r1}\left[v_{w1} \pm v_{w2}\right]\times u}{\rho g} = \frac{a v_{r1}\left[v_{w1} \pm v_{w2}\right]\times u}{g}$$

Efficiency of Jet:

The work done by the jet on the vane is the output of the jet where as the initial kinetic energy of the jet is the input.

$$\eta = \frac{\rho a V_{r1}\left(V_{w1} \pm V_{w2}\right) \times u}{\frac{1}{2}\rho a V_1 \times V_1^2}.$$

This is also no possible in practice. But the corresponding practical case is a series of such vanes fixed radially to the rim of a wheel. such vanes are provided for radial flow hydraulic turbines.

Force exerted on a series of radial vanes:

A radial flow is one in which a fluid particle during its flow through the vane of a rotating wheel remains in a plane normal to the axis of rotation and its distance from the axis of rotation is continuously changing.

Consider a series of radial curved vanes as shown in fig.11. For a radial curved vane, the radius of the vane at inlet and outlet is different and hence tangential velocities of the radial vane at inlet and outlet will not be equal.

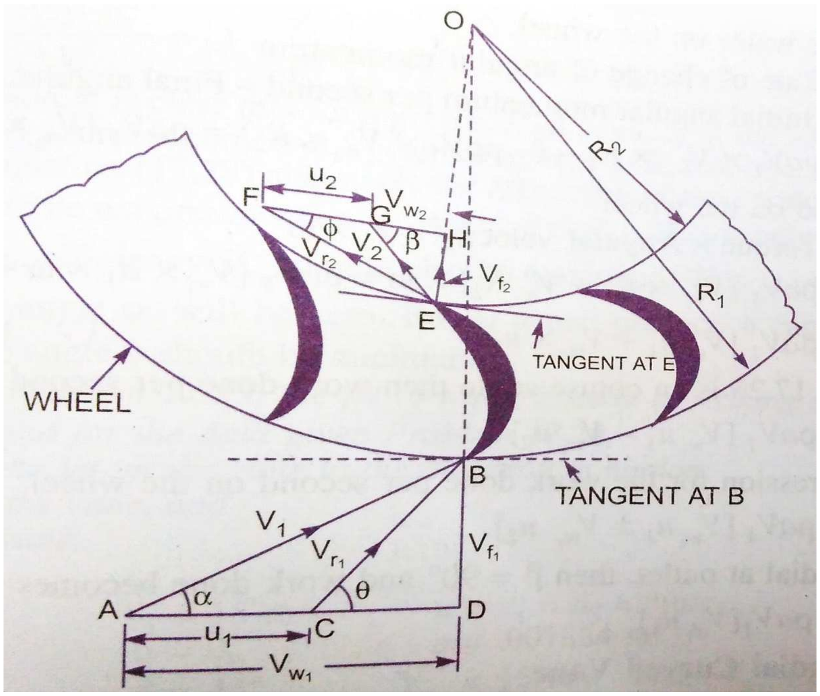


Fig. 11 Curved vane mounted radially

The jet of water strikes the vanes and the wheel starts rotating at a constant angular speed.

Let R_1 is the radius of the wheel at inlet of the vane.

R_2 is the radius of the wheel at the outlet of the vane.

ω is the angular speed of the wheel.

Then tangential velocities at inlet and outlet are $u_1 = R_1\omega,\ and\ u_2 = R_2\omega$ respectively.

Mass of water striking the vane per sec for a series of vanes = $\dot{m} = \rho a V_1$

Momentum of water striking the vanes in the tangential direction per sec at inlet

= mass of water per second x component of V_1 in the tangential direction = $\rho a V_1 \times V_{w1}$

Similarly, momentum of water at outlet per sec = $-\rho a V_1 \times V_{w2}$

Negative sign is taken as the velocity V_2 at outlet is in opposite direction.

Now, Angular momentum at inlet = momentum at inlet x radius at inlet = $\rho a V_1 \times V_{w1} \times R_1$

Angular momentum at out let = momentum at out let x radius at outlet =

$-\rho a V_1 \times V_{w2} \times R_2$

Torque Exerted by the water on the wheel, T = Rate of change of angular momentum.

= [Initial angular momentum per sec – final angular momentum per sec]

$$T = \rho a V_1 \times V_{w1} \times R_1 - (-\rho a V_1 \times V_{w2} \times R_2) = \rho a V_1 \left[V_{w1} \times R_1 + V_{w2} \times R_2 \right]$$

Work done per sec on the wheel = Torque x angular velocity

$$W = \rho a V_1 \left[V_{w1} R_1 + V_{w2} R_2 \right] \times \omega = \rho a V_1 \left[V_{w1} u_1 + V_{w2} u_2 \right]$$

The above equation is for $\beta < 90^0$.

When $\beta = 90^0$, $T = \rho a V_1 \times V_{w1} u_1$

When β is obtuse angle i.e. $\beta > 90^0$, $W = \rho a V_1 \left[V_{w1} R_1 - V_{w2} R_2 \right] \times \omega = \rho a V_1 \left[V_{w1} u_1 - V_{w2} u_2 \right]$

Efficiency of the radial curved vane:

The work done per second on the wheel is the output of the system where as the initial kinetic energy per second of the jet is the input. Hence,

Efficiency, $\eta = \dfrac{Workdone}{Kineticenergy}$

$$\eta = \frac{\rho a V_1 \left(V_{w1} u_1 \pm V_{w2} u_2 \right)}{\frac{1}{2} \rho a V_1 \times V_1^2} = \frac{2 \left(V_{w1} u_1 \pm V_{w2} u_2 \right)}{V_1^2}$$

Euler's equation:

The work done by the fluid on the vane is given as

$$W = \rho a V_1 \left[V_{w1} R_1 \pm V_{w2} R_2 \right] \times \omega = \rho a V_1 \left[V_{w1} u_1 \pm V_{w2} u_2 \right]$$

This work done per unit weight is defined as the Euler's equation of head.

$$H_{Eu} = \frac{W}{mg} = \frac{\rho a V_1 \left[V_{w1} u_1 \pm V_{w2} u_2 \right]}{mg} = \frac{\left[V_{w1} u_1 \pm V_{w2} u_2 \right]}{g}$$

For the power generating machines, work is done by the fluid on the vane. Hence Euler's equation for Turbines is $H_{Eu} = \frac{W}{mg} = \frac{\rho a V_1 \left[V_{w1} u_1 \pm V_{w2} u_2 \right]}{mg} = \frac{\left[V_{w1} u_1 \pm V_{w2} u_2 \right]}{g}$

For the power consuming machines like pumps, compressors, fans and blowers, work is done on the fluid. Hence Euler's equation is $H_{Eu} = \frac{\left[V_{w2} u_2 \pm V_{w1} u_1 \right]}{g}$

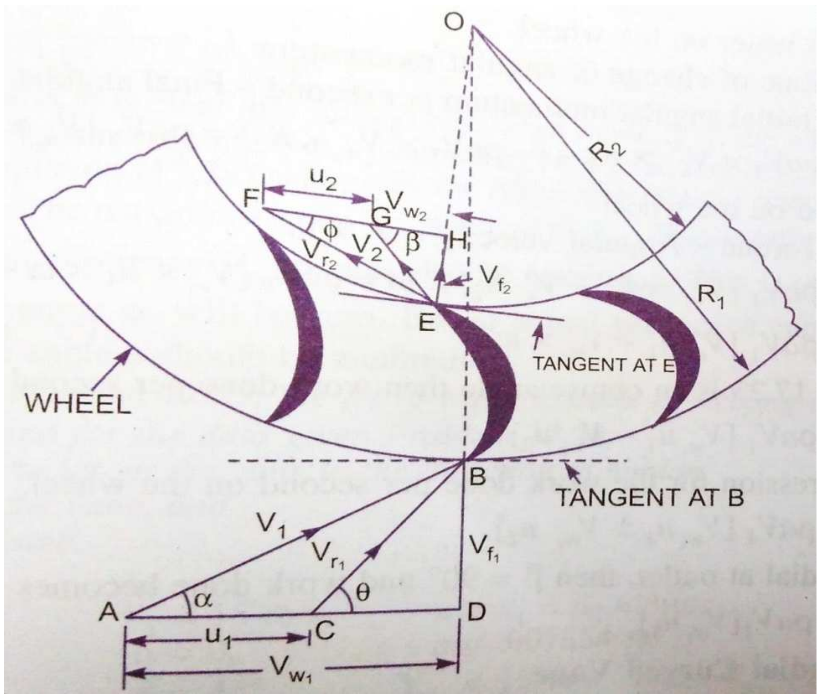
Energy Transfer in Turbomachinery

From the Euler's equation, $H_{Eu} = \frac{W}{mg} = \frac{\rho a V_1 \left[V_{w1} u_1 \pm V_{w2} u_2 \right]}{mg} = \frac{\left[V_{w1} u_1 \pm V_{w2} u_2 \right]}{g}$

We can write, from the inlet velocity triangle ABD, $V_1^2 = V_{f1}^2 + V_{w1}^2$

Again from triangle BCD, $V_{r1}^2 = V_{f1}^2 + \left(u_1 - V_{w1} \right)^2 = V_{f1}^2 + u_1^2 + V_{w1}^2 - 2 u_1 V_{w1}$

From these two equations we can write, $V_{r1}^2 = V_1^2 + u_1^2 - 2 u_1 V_{w1} \Rightarrow u_1 V_{w1} = \frac{V_1^2 + u_1^2 - V_{r1}^2}{2}$



Again we can write,

From the exit velocity triangle EFH, $V_{r2}^2 = V_{f2}^2 + \left(u_2 + V_{w2}\right)^2 = V_{f2}^2 + u_2^2 + V_{w2}^2 + 2u_2 V_{w2}$

From the triangle EGH, $V_2^2 = V_{f2}^2 + V_{w2}^2$

From the above equations, we can write, $V_{r2}^2 = V_2^2 + u_2^2 + 2u_2 V_{w2} \Rightarrow u_2 V_{w2} = \dfrac{V_{r2}^2 - u_2^2 - V_2^2}{2}$

From the Euler's equation, $H_{Eu} = \dfrac{\left[V_{w1}u_1 + V_{w2}u_2\right]}{g} = \dfrac{1}{g}\left[\dfrac{V_1^2 + u_1^2 - V_{r1}^2}{2} + \dfrac{V_{r2}^2 - u_2^2 - V_2^2}{2}\right]$

Finally, total head utilized from the fluid, $H_{Eu} = \frac{V_1^2 - V_2^2}{2g} + \frac{u_1^2 - u_2^2}{2g} + \frac{V_{r2}^2 - V_{r1}^2}{2g}$

The term $\left(\frac{V_1^2 - V_2^2}{2g}\right)$ represents the dynamic head or change in kinetic energy due to absolute velocity of jet per unit weight.

The term $\left(\frac{u_1^2 - u_2^2}{2g} + \frac{V_{r2}^2 - V_{r1}^2}{2g}\right)$ represents the static head. It is the sum of change in kinetic energy due to rotor velocity and kinetic energy change due to change in the jet velocity relative to the vane while flowing in between the rotor vanes per unit weight.

Similarly, total head gained by the fluid, $H_{Eu} = \frac{V_2^2 - V_1^2}{2g} + \frac{u_2^2 - u_1^2}{2g} + \frac{V_{r1}^2 - V_{r2}^2}{2g}$

Turbomachinery Unit-II

Centrifugal Pumps

Objectives:

- To understand the working principles of Hydraulic Centrifugal pumps and draw the performance characteristics of hydraulic pumps.

Syllabus:

Centrifugal Pumps: Component parts and Working, classification, Workdone by the Impeller,Manometric head, Losses and Efficiencies, specific Speed, Pumps in series and parallel,performance characteristic curves, NPSH

LEARNING OUTCOMES:

Students will be able to:

- Understand the working mechanism and components of a centrifugal pump.
- Classify the centrifugal pumps.
- Draw the velocity triangles and calculate the work done on the impeller of the Centrifugal pump.
- Understand the heads and efficiencies of centrifugal pump.
- Learn the construction of performance characteristic curves.
- Understand the Specific speed and its importance.

Definition:

- A device which raises or transfers liquids at the expense of power input is called Pump.
- A machine designed to elevate, deliver and move various liquids called Pump.
- A unit that transfers the mechanical energy of motor or an engine into potential and kinetic energy of liquid is called pump.

Note:

- By their action, the pump requires that energy must be expanded and as such they belong to the category of power absorbing machines.
- Since the temperature gradients are minimal, pumps are the non thermal machines.

Classification of Pumps:

1. Dynamic Pumps
    a) Centrifugal Pump

    b) Turbine
    c) Propeller
    d) Jet
2. Positive displacement Pumps
    a) Reciprocating pump
        i. Piston / Plunger
        ii. Diaphragm
    b) Rotor Pump
        i. Gear type
        ii. Lobe type
        iii. Vane type
        iv. Screw type
        v. Rotary plunger

Selection criterion of Pump:

- Pressure and capacity of liquid being handled.
- Properties such as viscosity, temperature, corrosiveness and grittiness etc of the flowing liquid.
- Initial and maintenance cost.
- Pump duty i.e whether the pump is to transfer the liquid or to meter it also.
- Availability of space, size and position of locating the pump.
- Speed of rotation and power required.
- Standardization with respect to the types and makes of pumps already available at the site.
- Scale up problems.

Applications of Pumps:

Some notable applications of pump installation are in the field of

i. Agriculture and irrigation works.
ii. Municipal water works and drainage system.
iii. Condensing water, condensate, boiler feed, sump drain and such other services in steam power plants.
iv. Hydraulic control systems.
v. Oil pumping.
vi. Transfer of raw materials, materials in manufacture and the finished products in industry.

CENTRIFUGAL PUMPS

INTRODUCTION:

- The hydraulic machines which convert the mechanical energy into hydraulic energy (in the form of pressure energy) are called **PUMPS**.

- If the mechanical energy is converted in to pressure energy by means of centrifugal force acting on a fluid, then the hydraulic machine is called **CENTRIFUGAL PUMP**.

WORKING MECHANISM OF A CENTRIFUGAL PUMP:

- A centrifugal pump is one of the simplest pieces of equipment in any pumping process plant.
- Its purpose is to convert energy of a prime mover (an electric motor or a turbine) firstinto velocity or kinetic energy and then into pressure energy of a fluid that is being pumped.

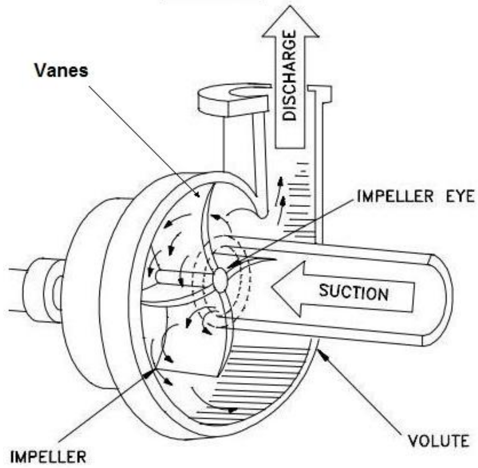


Fig. 1 Working of Centrifugal pump

- The energy changes occur by virtue of two main parts of the pump, the impellerand the volute or diffuser. The impeller is the rotating part that converts driver energy intothe kinetic energy. The volute or diffuser is the stationary part that converts the kineticenergy into pressure energy.

Note**:** All of the forms of energy involved in a liquid flow system are expressed in terms of height of liquid column i.e. head.

Generation of Centrifugal Force:

- The liquid enters the suction nozzle and then into eye (center) of a revolving device known as an impeller.
- When the impeller rotates, it spins the liquid sitting in the cavities between the vanes outward and provides centrifugal acceleration as shown in **figure 5.1**.
- As liquid leaves the eye of the impeller a low-pressure area is created causing more liquid to flow toward the inlet. Because the impeller blades are curved, the fluid is pushed in a tangential and radial direction by the centrifugal force.

Conversion of Kinetic Energy to Pressure Energy:

- The key point is that the energy created by the centrifugal force is *kinetic energy*. And the amount of energy given to the liquid is proportional to the *velocity* at the edge or vane tip of the impeller.
- The faster the impeller revolves or the bigger the impeller is, then the higher will be the velocity of the liquid at the inlet vane tip and the greater the energy imparted to the liquid.

- This kinetic energy of a liquid coming out of an impeller is harnessed by creating a *resistance* to the flow (decreasing the velocity of flow by increasing the passage cross sectional area).
- The first resistance is created by the pump volute (casing) that slows the liquid down, due to its increasing passage cross sectional area as shown in figure 5.2. In the discharge nozzle, the liquid further decelerates and its velocity is converted to pressure according to Bernoulli's principle.

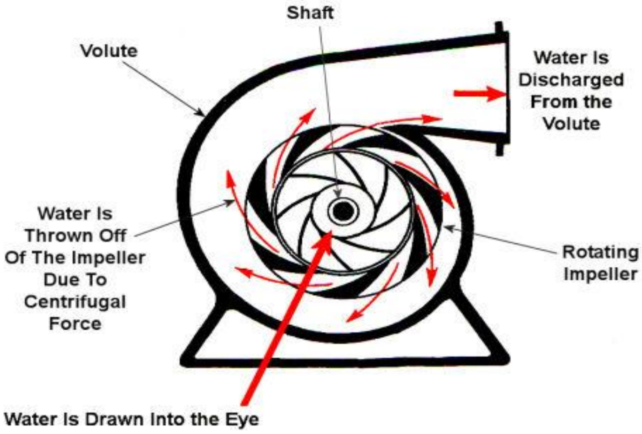


Fig. 2 Working of Centrifugal Pump

GENERAL COMPONENTS OF CENTRIFUGAL PUMPS:

A centrifugal pump has two main components:

i. A stationary component comprised of a casing, casing cover, and bearings.
ii. A rotating component comprised of an impeller and a shaft

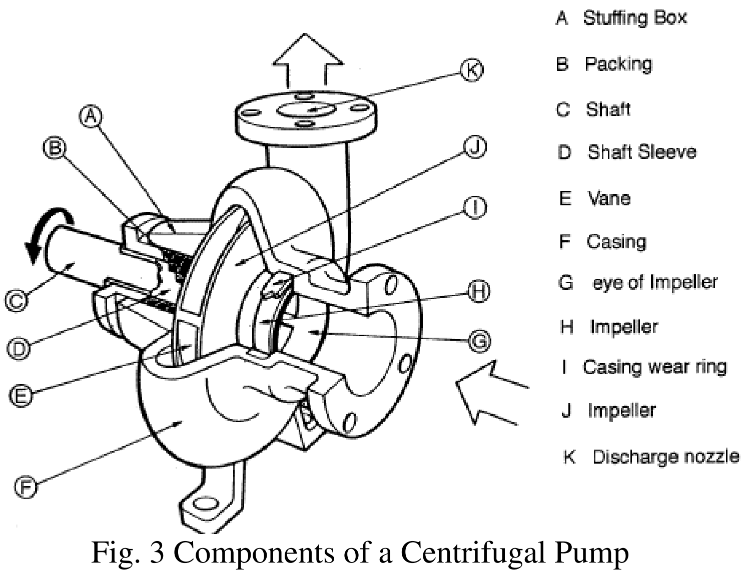


Fig. 3 Components of a Centrifugal Pump

The general components, both stationary and rotary, are depicted in **Figure**

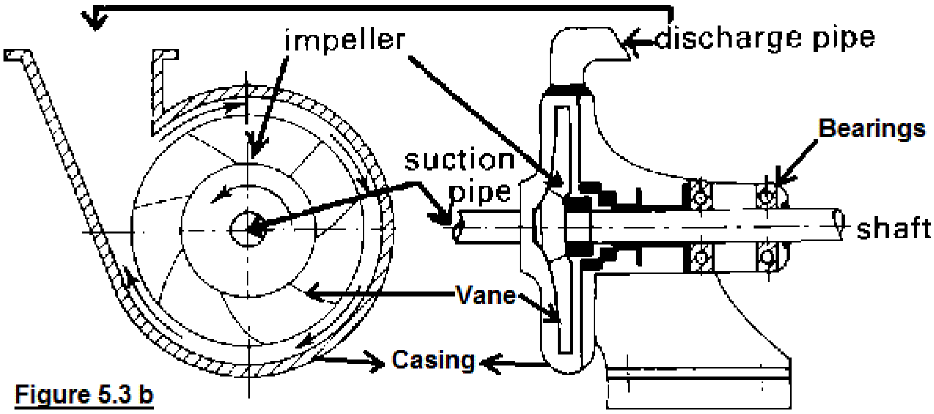


Fig. 3(b) Impeller components

Components of centrifugal pump showing in line diagram (Figure 5.3b) with front view and sectional side view.

i. **Stationary Components:**

Casing: Casings are generally of three types: **volute, vortex and circular**.

- *Volute casings*build a higher head.A *volute* is a curved funnel increasing in area to the discharge port as shown in Fig 5.4. As the area of the cross-section increases, the volute reduces the speed of the liquid and increases the pressure of the liquid. But large amount of energy is lost in volute casing due to formation of eddies.
- In *Vortex casing* a circular chamber is introduced between the casing and the impeller as shown in **figure 5.4**. By introducing the circular chamber, the loss of energy due to formation of eddies is reduced.The vortex casing may be with or without guide blades/vanes. The volute casing will not have guide vanes/blades.
- *Circular casings*are used for low head and high capacity.Circular casingis as shown in **Fig 5.4**, whichhave stationary *diffusion vanes* surrounding the impeller periphery that convert velocity energy to pressure energy. Hence this type of casing is also called Diffuser ring casing.
- Conventionally, thediffusers are applied to multi-stage pumps.

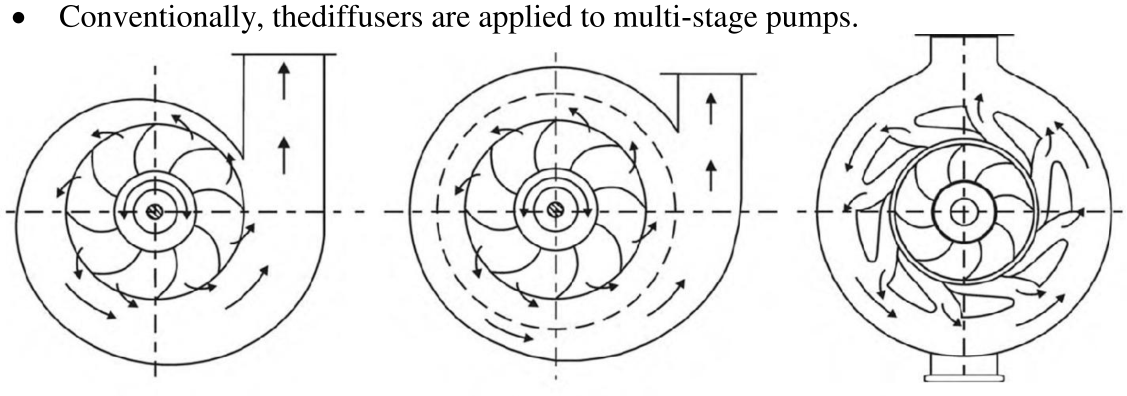


Fig. 4 Types of Casing

Suction and Discharge pipes:

The suction and discharge pipes are part of the casings itself. The suction pipe carries liquid from ground to the eye of an impeller while the delivery pipe carries the liquid from casing (impeller outlet) to the delivery tank. They commonly have the following configurations.

End suction/Top discharge- The suction nozzle is located at the end of and concentric to, the shaft while the discharge nozzle is located at the top of the case perpendicular to the shaft. This pump is always of an overhung type
Top suction Top discharge-The suction and discharge nozzles are located at the top of the case perpendicular to the shaft. This pump can either be an overhung type or between-bearing type but is always a radially split case pump.
Side suction / Side discharge - The suction and discharge nozzles are located at the sides of the case perpendicular to the shaft.

Seal Chamber and Stuffing Box
Seal chamber and Stuffing box both refer to a chamber, either integral with or separate from the pump case housing that forms the region between the shaft and casing where sealing media are installed. When the sealing is achieved by means of a mechanical seal, the chamber is commonly referred to as a Seal Chamber. When the sealing is achieved by means of packing, the chamber is referred to as a Stuffing Box. Both the seal chamber and the stuffing box have the primary function of protecting the pump against leakage at the point where the shaft passes out through the pump pressure casing.

Bearing housing
The bearing housing encloses the bearings mounted on the shaft. The bearings keep the shaft or rotor in correct alignment with the stationary parts under the action of radial and transverse loads. The bearing house also includes an oil reservoir for lubrication, constant level oiler, jacket for cooling by circulating cooling water.

Rotating Components:
Impeller
The impeller is the main rotating part that provides the centrifugal acceleration tothe fluid. They are often classified in many ways.

- Based on major direction of flow in reference to the axis of rotation
  - Radial flow
  - Axial flow
  - Mixed flow
- Based on suction type
  - Single-suction: Liquid inlet on one side.
  - Double-suction: Liquid inlet to the impeller symmetricallyfrom both sides.
- Based on mechanical construction **(Figure 5.5**)
  - Closed: Shrouds or sidewall enclosing the vanes. (For pumping pure liquids)
  - Open: No shrouds or wall to enclose the vanes.(For pumping semi-solids like mud)
  - Semi-open or vortex type. (For Pumping Stringy materials like debris, mango pulp...etc)

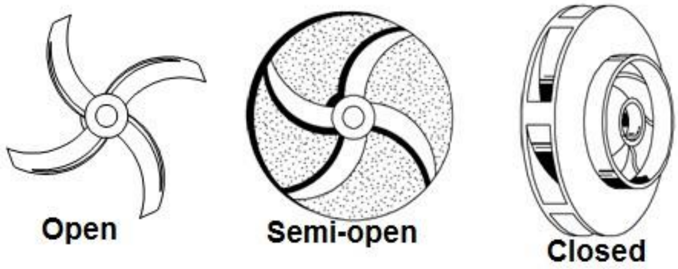


Fig. 5 Types of Impellers

The basic purpose of a centrifugal pump shaft is to transmit the torques encountered when starting and during operation while supporting the impeller and other rotating parts. It must do this job with a deflection less than the minimum clearance between the rotating and stationary parts.

Shaft Sleeve

Pump shafts are usually protected from erosion, corrosion, and wear at the seal chambers and leakage joints, internal bearings, and in the waterways by renewable sleeves.

TYPES OF CENTRIFUGAL PUMPS:

Centrifugal pumps are classified as given below

- *Based on casing design*: **volute, vortex and circular** type casing centrifugal pumps
- *Based on suction and discharge pipes locations*: **End suction/Top discharge, Top suction Top discharge&Side suction / Side discharge** type centrifugal pumps
- *Based on direction of flow in reference to the axis:* **Radial flow, Axial flow, Mixed** flow type centrifugal pumps
- *Based on number of suctions*: **Single suction, Double suction** type centrifugal pumps
- *Based on impeller design*: **Open, closed, semi-open** impeller type centrifugal pumps

WORK DONE BY THE IMPELLER:

In centrifugal pump work is done by the impeller on the water. The expression for work done by the impeller is obtained by drawing the velocity triangles at inlet and outlet of the impeller.

- Let 'α' be the angle made by absolute velocity of water entering at inlet of vane of impeller w.r.t the tangent at inlet.
- 'β' be the angle made by absolute velocity of water leaving at outlet of vane of impeller w.r.t the tangent at outlet.
- 'θ' be the angle made by the vane of impeller at inlet w.r.t the tangent at inlet. (*Blade angle at inlet*)
- '$\emptyset$' be the angle made by the vane of impeller at outlet w.r.t the tangent at outlet. (*Blade angle at outlet*)
- 'v_1' be the absolute velocity of jet at inlet of the vane & 'v_2' be the absolute velocity of jet at outlet of the vane.
- 'v_{f1}' be the flow velocity over the vane at inlet & 'v_{f2}' be the flow velocity over the vane at outlet. 'v_f' is the vertical component of absolute velocity 'v'. The flow velocity 'v_f' is responsible for flow of water over the vane.
- 'v_{w1}' be the whirl velocity over the vane at inlet & 'v_{w2}' be the whirl velocity over the vane at outlet. 'v_w' is the horizontal component of absolute velocity 'v'. The whirl velocity 'v_f' is responsible for motion of vane.
- 'v_{r1}' be the relative velocity of water entering the vane at inlet & 'v_{r2}' be the relative velocity of water leaving the vane at outlet as shown in figure.
- 'u_1' & 'u_2' be the tangential velocity of impeller at inlet & out let respectively.
- 'N' be speed of impeller in r.p.m.
- 'D_1' & 'D_2' be the diameter of impeller at inlet & outlet respectively.

When the pump is working at a rotational speed of N rpm,

Peripheral velocity of blade, at inlet, $u_1 = \frac{\pi D_1 N}{60}$

Peripheral velocity of blade, at outlet, $u_2 = \frac{\pi D_2 N}{60}$

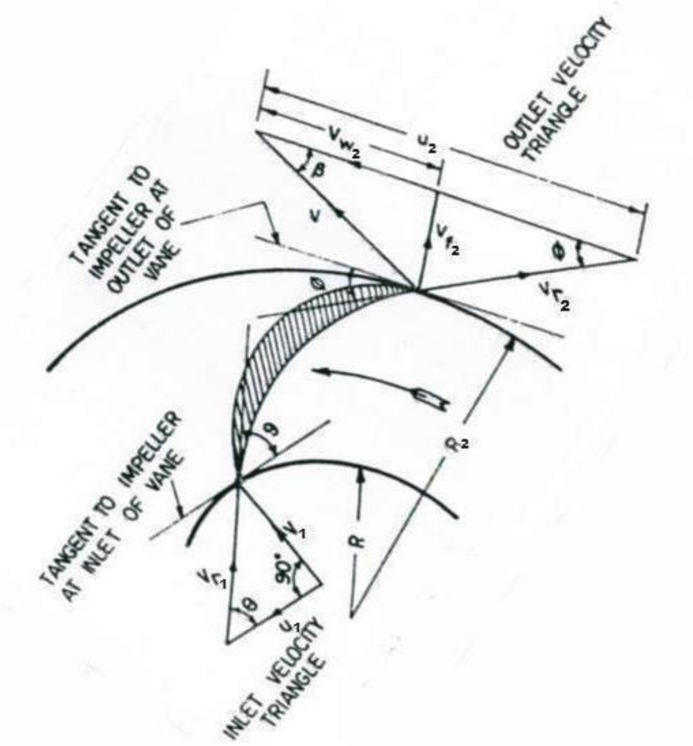


Fig. 6 Velocity Triangles

Discharge 'Q' is given by $Q = \pi D_1 b_1 V_{f1} = \pi D_2 b_2 V_{f2}$

b_1=width/breadth of vane at inlet and b_2=width/breadth of vane at outlet.

Work done by the impeller is given by $W = \rho Q\left[V_{w2}u_2 - V_{w1}u_1\right]$

Since the entry to the pump is radial $\alpha = 90^0$. Hence at the inlet,

Whirl component of velocity, $V_{w1} = V_1 \cos\alpha = 0$

Flow component of velocity, $V_{f1} = V_1 \sin\alpha = V_1$

Therefore work done/Output is $W = \rho Q V_{w2} u_2$

Head imparted to the impeller or Euler's head, $H_{Eu} = \dfrac{V_{w2}u_2}{g}$

HEADS AND EFFICIENCIES OF CENTRIFUGAL PUMP:

Static Suction Head (**hs**):

Head resulting from lifting of the liquid relative to the pump center line. If the liquid level is above pump centerline, $\mathbf{h_s}$is positive. If the liquid level is below pump centerline, $\mathbf{h_s}$is negative. Negative $\mathbf{h_s}$condition is commonly denoted as a "suction lift" condition.

Static Discharge Head (**hd**):

It is the vertical distance between the pump centerline and the point of free discharge or the surface of the liquid in the discharge tank.

Friction Head (**hf**):

The head required to overcome the resistance to flow in the pipe and fittings. It is dependent upon the size (diameter & length) of pipe, condition and type of pipe, number and type of pipe fittings, flow rate, and nature of the liquid.

(Note: The Subscripts'**s**'refers to suction conditions and'**d**'refers to discharge conditions.)

Hence, h_{f1} & h_{f2} are the head lost due to friction in suction and delivery pipes respectively. And h_f is the total head lost due to friction.

Vapor Pressure Head (**hvp**):

Vapor pressure is the pressure at which a liquid and its vapor co-exist in equilibrium at a given temperature. The vapor pressure of liquid can be obtained from vapor pressure tables. When the vapor pressure is converted to head, it is referred to as vapor pressure head, **hvp**.

Velocity Head (**hv**):

It is the energy of a liquid due to the result of its motion at some velocity '**v**'. It is the equivalent head through which the water would have to fall to acquire the same velocity, or in other words, the head necessary to accelerate the water to that velocity v. **The velocity head is usually insignificant and can be ignored** in most high head systems. However, it can be a large factor and must be considered in low head systems. Therefore 'h_{vs}' & 'h_{vd}' are the velocity heads in suction and delivery pipes respectively.

Pressure Head (**hp**)*:*

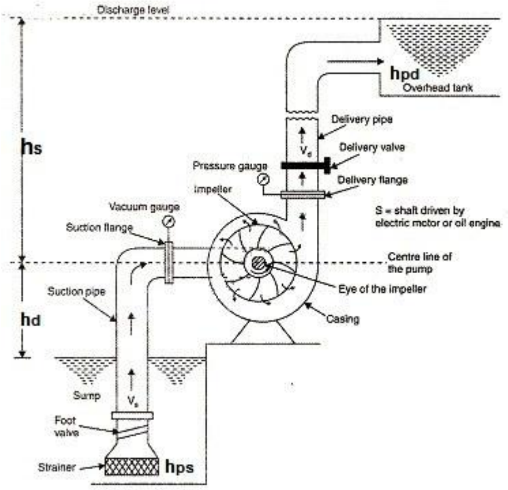
Pressure Head must be considered when a pumping system either begins at the suction point (Sump, Ground water point) or terminates in a tank which is under some pressure other than atmospheric. Simply pressure head at the sump/suction point is h_{ps} at that of delivery is h_{pd}

Total Suction Head (**Hs**):

The total suction head is the reading of the gauge on the suction flange, converted to height of liquid. Therefore **Hs = hps+ hs+ hvs– hfs**

Total Discharge Head (**Hd**):

The total discharge head is the reading of a gauge at the discharge flange, converted to length of liquid. Therefore **Hd= hpd+ hd+ hvd+ hfd**



*Total Differential Head (**HT**):*

It is the total discharge head minus the total suction head.

$$H_T = H_d + H_s \text{ (with a suction lift)}$$

$$H_T = H_d - H_s \text{ (with a suction head)}$$

*Manometric Head (**H_m**):*

*Manometric efficiency(**η_{man}**):* The ratio of the manometric head to the head imparted by the impeller to the water.

$$\eta_{man} = \frac{ManometricHead}{Workdoneonwater}$$

$$= \frac{wQH_m}{\rho Q v_{w1} u_1} = \frac{gH_m}{v_{w1} u_1}$$

*Mechanical efficiency (**η_m**)*: The ratio of power given by the impeller on water to the power available at the impeller shaft (S.P).

$$\eta_m = \frac{\rho Q[v_{w2} u_2]}{S.P}$$

Overall efficiency (**η_o**): The ratio of power output of pump to the input shaft power of pump.

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

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

NPSH (Net Positive Suction Head):

NPSH is one of the most widely used and least understood terms associated withpumps. Understanding the significance of NPSH is very much essential duringinstallation as well as operation of the pumps.When discussing centrifugal pumps, the two most important NPSH terms are **NPSHr** (Net Positive Suction Head Required) and**NPSHa** (Net Positive Suction Head Available).

- We know that, Pumps can pump only liquids, not vapors and rise in temperature and fall in pressure induces vaporization.
- This makes it clear that if we are to pump a fluid effectively, it must be kept always in the liquid form and thus, the pump always needs to have a sufficientamount of suction head present to prevent this vaporization at the lowest pressure point in the pump.
- The manufacturer usually tests the pump with water at different capacities, created bythrottling the suction side. When the first signs of vaporization induced cavitation occur,the suction pressure is noted (the term cavitation is discussed in detail later). Thispressure is converted into the head. This head number is published on the pump curveand is referred as the "net positive suction head required (NPSHr) or sometimes in short as the NPSH.
- Thus the **Net Positive Suction Head (NPSH) is the total head at the suction flange of the pump less the vapor pressure converted to fluid column height of the liquid**.
- *NPSHr is a function of pump design*: (Reason for decrease in pressure in the impeller)
    1. NPSH required is a function of the pump design and is determined based onactual pump test by the vendor. As the liquid passes from the pump suction to the eyeof the impeller, the velocity increases and the pressure decrease.
    2. There are alsopressure losses due to shock and turbulence as the liquid strikes the impeller.
    3. Thecentrifugal force of the impeller vanes further increases the velocity and decreases thepressure of the liquid.
- *NPSHr increases as capacity increases:* The NPSH required varies with speed and capacity within any particular pump. The NPSH required increase as the capacity is increasing because the velocity of the liquid is increasing, and as anytime the velocity of a liquid goes up, the pressure or head comes down.
- *NPSHa*: **N**et **P**ositive **S**uction **H**ead **A**vailable is a function of the system design in which the pump operates. It is the excess pressure of the liquid in feet absolute over its vapor pressure as it arrives at the pump suction, to be sure that the pump selected does not cavitate.Therefore, $NPSHa_S = hp_S + h_S - hvp_S - hf_S$

Important Notes on NPSH:

- Simply NPSH refers to NPSHr and it is not NPSHa.
- NPSH or NPSHr is a function of pump design where asNPSHa is the function of system design in which the pump has to operate.
- **NPSH =** $\mathbf{hvp_s}$ at the given operating temperature.
- The NPSH is always positive since it is expressed in terms of absolute fluid column height. The term "Net" refers to the actual pressure head at the pump suction flange and not the static suction head.

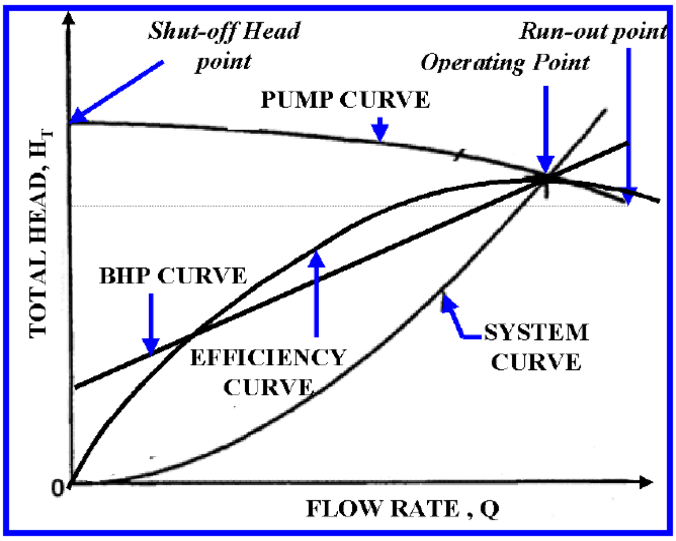
- ❖ It is to be noted that the net positive suction head required (NPSHr) number shown on the pump curves is for fresh water at 20°C and not for the fluid or combinations of fluids being pumped.
- ❖ Any discussion of NPSH or cavitation is only concerned about the suction side of the pump. There is almost always plenty of pressure on the discharge side.

Significance of NPSHr and NPSHa:

The NPSH available must always be greater than the NPSH required for the pump to operate properly. It is normal practice to have at least 2 to 3 feet of extra NPSH available at the suction flange to avoid any problems at the duty point.

PERFORMANCE CHARACTERISTIC CURVES:

- ✓ The capacity and pressure needs of any system can be defined with the help of a graph called a ***system curve***.
- ✓ Similarly the capacity *vs.* pressure variation graph for a particular pump defines its characteristic ***pump performance*** curve as shown in figure.



The pump suppliers try to match the system curve supplied by the user with a pump curve that satisfies these needs as closely as possible. A pumping system operates where the pump curve and the system resistance curve intersect. The intersection of the two curves defines the operating point of both pump and process. However, it is impossible for one operating point to meet all desired operating conditions.

Developing a system curve:

The *system resistance or system head curve* is the change in flow with respect to head of the system. It must be developed by the user based upon the conditions ofservice. These include physical layout, process conditions, and fluid characteristics. It represents the relationship between flow and hydraulic losses in a system in a graphic form and, since friction losses vary as a square of the flow rate, the system curve is parabolic in shape. Hydraulic losses in piping systems are composed of pipe friction losses, valves, elbows and other fittings, entrance and exit losses, and losses from changes in pipe size by enlargement or reduction in diameter.

Developing a Pump performance Curve:

A pump's performance is shown in its characteristics performance *curve* where its capacity i.e. flow rate is plotted against its developed head. The pump performance curve also shows its efficiency (BEP), required input power (in BHP), NPSH, speed (in RPM), and other information such as pump size and type, impeller size, etc. This curve is plotted for a constant speed (rpm) and a given impeller diameter (or series of diameters). It is generated by tests performed by the pump manufacturer. Pump curves are based on a specific gravity of 1.0. Other specific gravities must be considered by the user.

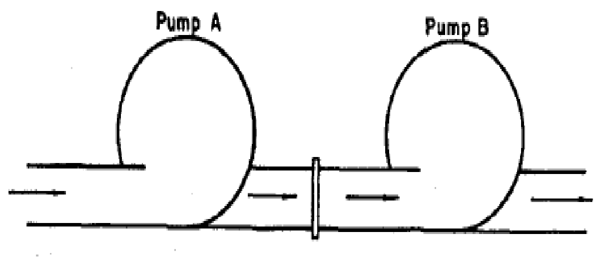
MULTISTAGE CENTRIFUGAL PUMPS

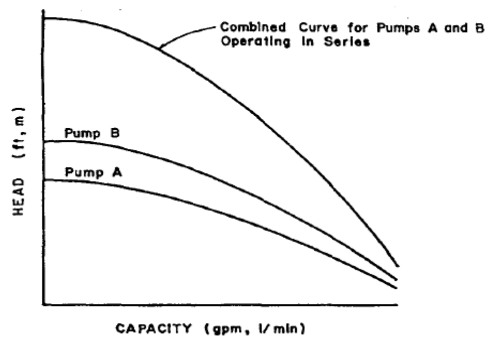
If centrifugal pump consist of two or more impellers then the pump is called multistage centrifugal pump. Multistage centrifugal pumps are used either to produce a high head or to discharge a large quantity of liquid than that of single centrifugal pump.

- If a high head is to be developed, the impellers are connected in series (or on same shaft)
- If a high discharge is to be developed, the impellers are connected in parallel (or on individual shaft/individual pump).

Pumps operating in series:

- To connect two pumps in series means that the discharge from the first pump is piped into the inlet side of the second pump (Figure).
- In this type of arrangement all the flow successively passes from one pump to the next with each pump adding more energy to the water. This is a typical arrangement in multi-stage pump where the same discharge passes through all stages and each builds additional head.



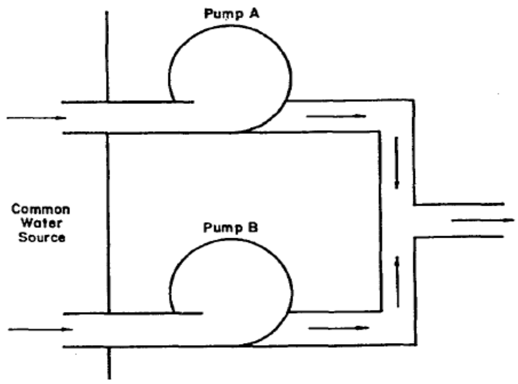


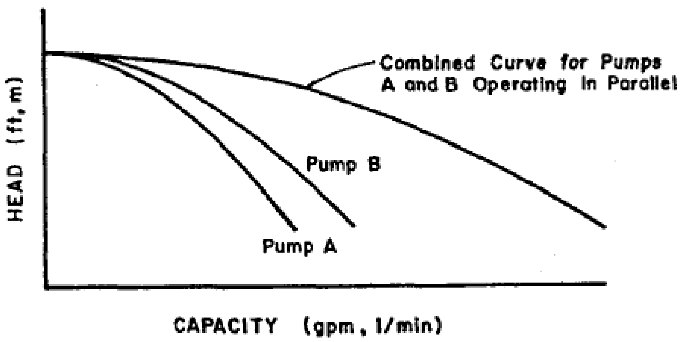
Figures showing

a) Two pumps connected In a series
b) Head versus discharge characteristic curves for two pumps operating in series.

Pumps operating in parallel:

- To connect two pumps in parallel means that the discharge from eachindividual (separate) pump is piped into single pipe line at the end. Figure presents a parallel configuration of two pumps.





Figures shows a) Two pumps connected in parallelb) Head versus discharge characteristic curves for two pumps operating in parallel.

SPECIFIC SPEED:

a) SPECIFIC SPEED AS A MEASURE OF THE GEOMETRIC SIMILARITY OF PUMPS

Specific speed (N_s) is a non-dimensional design index that identifies the geometricsimilarity of pumps. It is used to classify pump impellers as to their type and proportions.Pumps of the same Ns but of different size are considered to be geometrically similar,one pump being a size- factor of the other.

b) SPECIFIC SPEED CALCULATION:

$$Ns = \frac{N \times Q^{0.5}}{H^{0.75}}$$

Where N is speed in r.p.m, Q is Discharge,H is head in ‘m’

As per the above formula, it is defined as the speed in revolutions per minute at which a geometrically similar impeller would operate if it were of such a size as to deliver one gallon per minute flow against one-foot head. The understanding of this definition is of design engineering significance only, however, and specific speed should be thought of only as an index used to predict certain pump characteristics.

c) SPECIFIC SPEED AS A MEASURE OF THE SHAPE OR CLASS OF THE IMPELLERS:

The specific speed determines the general shape or class of the impellers. As the specific speed increases, the ratio of the impeller outlet diameter, D2, to the inlet or eye diameter, D1, decreases. This ratio becomes 1.0 for a true axial flow impeller. *Radial flowimpellers* develop head principally through centrifugal force. Radial impellers are generally low flow high head designs. Pumps of higher specific speeds develop head partly by centrifugal force and partly by axial force. A higher specific speed indicates a pump design with head generation more by axial forces and less by centrifugal forces. An axial flow or propeller pump with a specific speed of 10,000 or greater generates its head exclusively through axial forces. Axial flow impellers are high flow low head designs.

Specific speed identifies the approximate acceptable ratio of the impeller eye diameter (D1) to the impeller maximum diameter (D2) in designing a good impeller.

Ns: 500 to 5000; D1/D2 > 1.5 - radial flow pump

Ns: 5000 to 10000; D1/D2 < 1.5 - mixed flow pump

Ns: 10000 to 15000; D1/D2 = 1 - axial flow pump

TROUBLES & REMEDIES:

There are two Basic Requirements for Trouble-Free Operation of CentrifugalPumps.

- The **first** requirement is that no cavitation of the pump occurs throughout the broad operating range.
- The **second** requirement is that a certain minimum continuous flow is always maintained during operation.

A clear understanding of the concept of cavitation, its symptoms, its causes, and its consequences is very much essential in effective analyses and troubleshooting of the cavitation problem.

Just like there are many forms of cavitation, each demanding a unique solution, there are a number of unfavorable conditions which may occur separately or simultaneously when the pump is operated at reduced flows. Some include:

o Cases of heavy leakages from the casing, seal, and stuffing box
o Deflection and shearing of shafts
o Seizure of pump internals
o Close tolerances erosion
o Separation cavitation
o Product quality degradation
o Excessive hydraulic thrust
o Premature bearing failures

Each condition may dictate a different minimum flow low requirement. The final decision on recommended minimum flow is taken after careful “techno-economical” analysis by both the pump user and the manufacturer. The consequences of prolonged conditions of cavitation and low flow operation can be disastrous for both the pump and the process. Such failures in hydrocarbon services have often caused damaging fires resulting in loss of machine, production, and worst of all, human life. Thus, such situations must be avoided at all cost whether involving modifications in the pump and its piping or altering the operating conditions. Proper selection and sizing of pump and its associated piping can not only eliminate the chances of cavitation and low flow operation but also significantly decrease their harmful effects.

Turbomachinery Unit-III

Hydraulic Turbines

Syllabus: Classification and working of turbines, Work done and efficiency of Pelton wheel Turbine , Work done and efficiency of Francis Turbine, Work done and efficiency of Kaplan Turbine, Draft tube- theory functions and efficiency, Surge tank

Performance under unit head –unit Quantities of Hydraulic Turbine, Performance under Specific conditions Hydraulic Turbine, Selection of Hydraulic Turbine, Performance characteristic curves of Hydraulic Turbine

Course objectives:

- To study the working and performance characteristics of various hydraulic turbines

Learning outcomes:

Students will be able to

- Explain the working of various components of hydroelectric power plants.
- Analyze the power developed and efficiency of different turbines
- Design a hydraulic turbine based on the given conditions

Degree of Reaction (R): It is the parameter which characterizes the proportions of changes in the dynamic head and static head in the rotor of a fluid machine.

It is defined as the ratio of energy transfer by the change in static head to the total energy transfer in the rotor.

In the case of Impulse machines, R = 0, because there is no static pressure change in the rotor. A machine with any degree of reaction must have an enclosed rotor so that the fluid can expand in all directions.

Classification of Hydraulic Turbines:

1. **According to action of the water flowing through the turbine runners**
   a) **Impulse Turbine**
      - In Impulse Turbine all the available energy of water is converted into kinetic energy or velocity head by passing it through a converging nozzle provided at the end of the penstock.
      - Pelton wheel, Turgo impulse wheel, Girrard turbine, Banki Turbine ad Jonval turbine are the examples for an Impulse Turbine
      - Degree of reaction is zero
      - Impulse machines always an axial flow or tangential flow
      - Paddle wheel is best example for impulse machine.

   b) **Reaction Turbine**
      - In Reaction Turbine, at the entrance to the runner, only a part of the available energy of water is converted into kinetic energy and a substantial part remains in the form of pressure energy.
      - Pressure at inlet to the turbine is much higher than pressure at the outlet of the reaction turbine
      - Examples for reaction turbine are Fourneyron Turbine, Thomson Turbine, Francis Turbine, Propeller, Kaplan Turbine etc.
      - Degree of reaction R is $R>0$ or $0<R<1$ for reaction turbine.
      - Reaction machines are always radial flow machines.
      - Lawn sprinkler is the best example for reaction turbine
2. **According to the main direction of flow of water in the runner**
   a) **Tangential flow turbine**
      - In tangential flow Turbine the water flows along the tangent to the path of rotation of the runner.
      - Pelton wheel is a tangential flow Turbine.

   b) **Radial flow Turbine**
      - In radial flow turbine the water flows along the radial direction and remains wholly and mainly in the plane normal to the axis of rotation, as it passes through the runner.
      - In an **Inward Radial flow Turbine** the water enters at the outer circumference and flows radially inwards towards the center of the runner.
      - Old Francis Turbine, Thomson Turbine, Girard radial flow Turbine are examples for Inward Radial flow Turbine.

- In an **Outward Radial flow Turbine** water enters at the center and flows radially outwards towards outer periphery of the runner.
- Fourneyron Turbine is an example for Outward Radial flow Turbine

c) Axial flow Turbine

- In Axial flow turbine the flow of water through the runner is wholly and mainly along the direction parallel to the axis of rotation of the runner
- Girrad axial flow turbine, propeller turbine, Kaplan turbine are examples for axial flow turbine

d) Mixed flow turbine

- In mixed flow turbine water enters the runner at outer periphery in the radial direction and leaves it at the center in the direction parallel to the axis of rotation of the runner.
- Modern francis turbine is the example for mixed flow turbine

3. According to the head available

a) High head turbine:

- Head available is more than 300 m
- Example: Pelton wheel.

b) Medium head turbine:

- head available is in between 60 m – 250m
- example : Modern Francis Turbine

c) Low head Turbine:

- Head available is less than 60m.
- Example: Kaplan and other Propeller Turbines

4. According to specific speed of the Turbine

a) High specific speed Turbine:

- Specific speed varying from 300 to 1000rpm
- Example: Kaplan and other Propeller Turbines

b) Medium specific speed Turbine:

- Specific speed varying from 60 to 400rpm
- example : Modern Francis Turbine

c) low specific speed Turbine:

- specific speed less than 60 rpm
- Example: Pelton wheel.

Pelton wheel Turbine

- It is an impulse machine, in which the change in static head in the rotor is zero.
- The Pelton wheel or Pelton turbine is a tangential flow Impulse Turbine.
- The water strikes the bucket along the tangent of the runner.
- The energy available at the inlet of turbine is only kinetic energy.
- The pressure at the inlet and outlet of turbine is atmospheric.
- This turbine is used for high Heads.

Main components of Pelton turbine are:

1. **Guide mechanism / Nozzle and Flow regulating arrangement(Spear):**

- The amount of water striking the buckets of the runner is controlled by providing a spear in the nozzle as shown.

- 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.

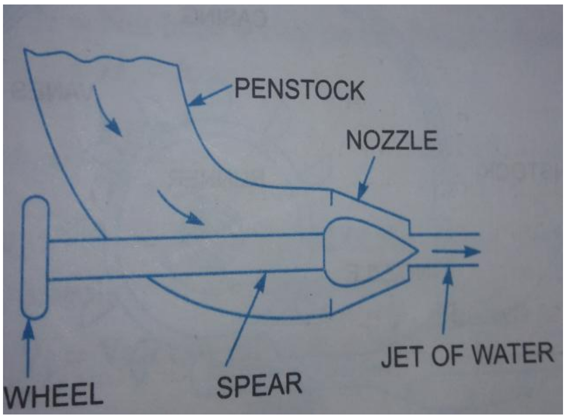


Fig. 1 Guide mechanism

2. **Buckets and Runner:**

- It consists of a circular disc on the periphery of which a number of buckets evenly spaced are fixed.
- The shape of the bucket is of a double hemispherical cup or bowl.
- Each bucket is divided into two symmetrical parts by dividing a wall which is known as splitter. The splitter divides the jet into two equal parts.
- The buckets are shaped in such a way that the jet gets deflected through 160^0 or 170^0.
- Buckets are made of cast Iron, cast steel, bronze or stainless steel depending upon the head at the inlet of the turbine.

Fig. 2 Arrangement of buckets

3. **Casing:**

- The function of the casing is to prevent the splashing of the water and to discharge the water to tail race.
- It also acts as safe guard against accidents.
- It is made of cast iron or fabricated steel plates.

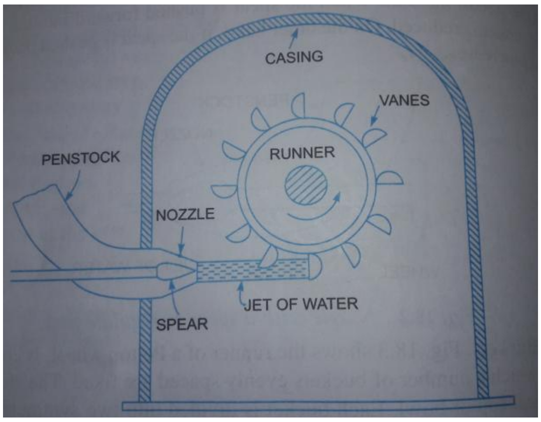


Fig. 3 Turbine casing

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.
- Due to inertia, the runner 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

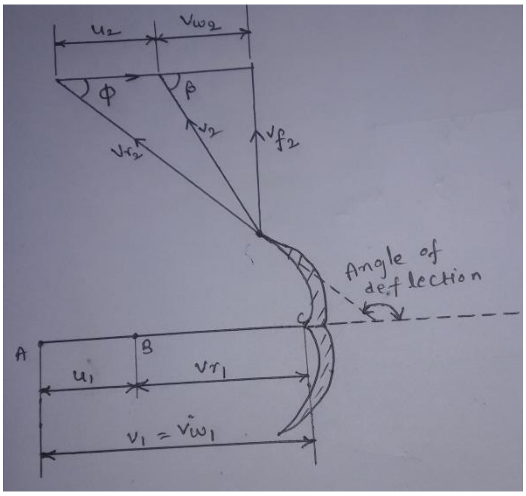


Fig. 4 Velocity triangle for Impulse turbine

Notations:

u1, u2 – blade velocities at inlet and outlet respectively

Vr1, Vr2 – relative velocities at inlet and outlet respectively

Vf1, Vf2 – Flow velocities at inlet and outlet respectively

Vw1, Vw2 – tangential; or whirl component of velocity at inlet and outlet respectively.

Let H = Net head acting on Pelton wheel = $H_g - h_f$.

where H_g = Gross head

$h_f = \dfrac{4flV^2}{2gd}$ = loss of head due to friction

Let d = diameter of penstock;
N = speed of wheel in RPM;
V = velocity of fluid in penstock;
f = coefficient of friction

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

Peripheral velocity, $u = u_1 = u_2 = \dfrac{\pi DN}{60}$

where D is diameter of Wheel.

Relative velocity at inlet, $V_{r1} = V_1 - u_1. = V_1 - u$

Whirl component of velocity at inlet, $V_{w1} = V_1$

$\alpha = 0$.

$\theta = 0$.

$V_{r1} = V_{r2}$

From outlet velocity triangle

Whirl component of velocity at inlet, $V_{w2} = V_{r2}\cos\phi - u_2$

The Force exerted by the jet of water in the direction of motion is $F_x = \rho a V_1 \left[V_{w1} + V_{w2}\right]$ (Positive sign will be taken when $\beta < 90^0$.

Where a = cross-sectional area of jet. Now

Work done by the jet on the runner per second = $P = F_x \times u = \rho a V_1 \left[V_{w1} + V_{w2}\right] \times u$

Power given to the runner by the jet = $F_x \times u = \dfrac{\rho a V_1 \left[V_{w1} + V_{w2}\right] \times u}{1000} kW$

Work done per sec. per unit weight of water striking the vane per sec = $\dfrac{\rho a V_1 \left[V_{w1} + V_{w2}\right] \times u}{\rho a V_1 \times g}$

$= \dfrac{\left[V_{w1} + V_{w2}\right] \times u}{g}$

Hydraulic Efficiency = $\dfrac{\text{work done per sec}}{\text{Kinetic energy of the jet per second}}$

Kinetic Energy of Jet striking the vane per sec. = $\dfrac{1}{2}\rho a v_1 \times v_1^2$

$$\therefore \eta_h = \frac{\rho a v_1 [v_{w1} + v_{w2}] \times u}{\frac{1}{2}(\rho a v_1) \times v_1^2} = \frac{2[v_{w1} + v_{w2}] \times u}{v_1^2} \text{---------------------(1)}$$

Condition for Maximum efficiency:

From Inlet velocity triangle,

$$v_{w1} = v_1, v_{r1} = v_1 - u_1 = v_1 - u$$

$$\therefore V_{r2} = v_1 - u \qquad (\because v_{r1} = v_{r2})$$

$$v_{w2} = v_{r2}\cos\phi - u_2 = (v_1 - u)\cos\phi - u$$

Substituting the values of v_{w1} and v_{w2} values in Equation (1)

$$\eta_h = \frac{2[v_1 + (v_1 - u)\cos\phi - u] \times u}{v_1^2} = \frac{2(v_1 - u)[1 + \cos\phi] \times u}{v_1^2}$$

The efficiency will be maximum for a given value of v_1 when $\frac{\partial}{\partial u}(\eta_h) = 0.$ i.e

$$\frac{\partial}{\partial u}\left(\frac{2u(v_1 - u)(1 + \cos\phi)}{v_1^2}\right) = 0. \qquad \text{or} \qquad \frac{1 + \cos\phi}{v_1^2}\frac{d}{du}\left(2uv_1 - 2u^2\right) = 0.$$

$$\frac{d}{du}\left(2uv_1 - 2u^2\right) = 0. \Rightarrow (2v_1 - 4u) = 0. \Rightarrow \text{u} = \frac{v_1}{2}.$$

$$\therefore \text{ Maximum efficiency } = \frac{2\left(v_1 - \frac{v_1}{2}\right)(1 + \cos\phi) \times \frac{v_1}{2}}{v_1^2} = \frac{2\left(\frac{v_1}{2}\right)(1 + \cos\phi) \times \frac{v_1}{2}}{v_1^2} = \frac{(1 + \cos\phi)}{2}$$

Important points to be remembered

- The velocity of jet at Inlet is given by $v_1 = c_v\sqrt{2gH}$
  Where c_v is coefficient of velocity = 0.98 or 0.99 and H = Net head.
- The velocity of wheel is given by $u = \phi\sqrt{2gH}$
  Where ϕ is the speed ratio. The value of speed ratio varies from 0.43 to 0.48
- The angle of deflection of the jet through the buckets is taken at 165^0, if no angle of deflection is given.

1. ***Radial Flow reaction Turbines:***

Radial flow turbines are those turbines in which the water flows in the radial direction. The water may flow radially from outwards to inwards or from inwards to outwards. If the water flows from outwards to inwards through the runner, the turbine is known as inward radial flow turbine. And if the water flows from inwards to outwards, the turbine is known outward radial flow turbine.

Reaction turbine means that the water at the inlet of the turbine possesses kinetic energy as well as pressure energy. As the water flows through the runner, a part of pressure energy goes on changing into kinetic energy. Thus the water through the runner is under pressure. The runner is completely enclosed in an air- tight casing and the runner is always full of water.

Main Parts of Radial flow Reaction Turbine:

The main parts of radial flow reaction turbine are:

1. Casing
2. Guide Mechanism
3. Runner and
4. Draft tube.

Casing:

The water from the penstock enters the casing which is of spiral shape in which the area of cross-section of the casing goes on decreasing gradually. The casing completely surrounds the runner of the turbine. The casing is made of concrete, cast steel or plate steel.

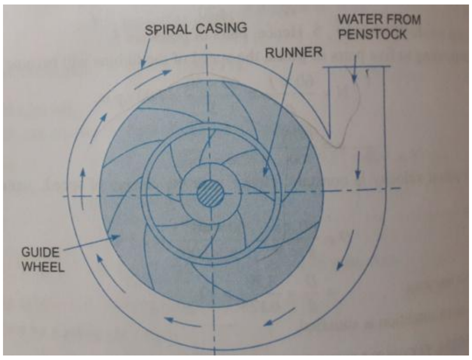


Fig. 5 Reaction turbine

Guide Mechanism:

It consists of stationary circular wheel all round the runner of the turbine. The stationary guide vanes are fixed on the guide mechanism. The guide vanes allow the water to strike the vanes fixed on the runner without shock at inlet. Also by a suitable arrangement, the width between two adjacent vanes of guide mechanism can be altered so that the amount of water striking the runner can be varied.

Runner:

It is a circular wheel on which a series of radial vanes are fixed. The surface of the vanes is made very smooth. The radial curved vanes are so shaped that the water enters and leaves the runner without shock. The runners are made of cast steel, cast Iron, or stainless steel.

Draft tube:

The pressure at the exit of the runner of a reaction turbine is generally less than atmospheric pressure. The water at the exit cannot be directly discharged to tail race. A tube or pipe of gradually increasing area is used for discharging water from the exit of the turbine to the tail race. This tube of increasing area is called draft tube.

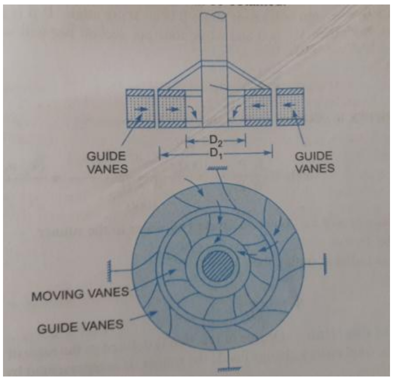


Fig. 7Inward radial flow reaction turbine

Velocity Triangles:

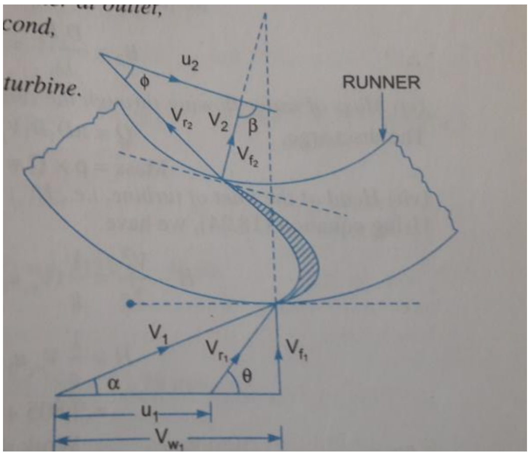


Fig. 8 Velocity triangles

2. Francis Turbine

The inward flow reaction turbine having radial discharge at outlet is known as Francis turbine. In modern Francis turbine, the water enters and the runner of the turbine in the radial direction at the outlet and leaves in axial direction at the inlet of the runner.

Inward flow reaction turbine:

Water from the casing enters the stationary guide wheel. The guiding wheel consists of guide vanes which direct the water to enter the runner which consists of moving vanes. The water flow over moving vanes in the inward radial direction and is discharged at the inner diameter of the runner.

The work done per second on the runner by water is $W = \rho a v_1 \left[v_{w1} u_1 \pm v_{w2} u_2 \right]$

v_{w1} is the velocity of whirl at inlet.

v_{w2} is the velocity of the whirl at outlet.

u_1 is the tangential velocity of the wheel at inlet.

u_2 is the tangential velocity of wheel at outlet.

The work done per second per unit weight of water striking the vane per second is

$$= \frac{\rho a v_1 \left[v_{w1} u_1 \pm v_{w2} u_2 \right]}{\left(\rho a v_1 \right) g} = \frac{1}{g} \left[v_{w1} u_1 \pm v_{w2} u_2 \right]$$

For Francis turbine β=90^0.

$\therefore$ Work done per second per unit weight of water striking the vane per second is = $\frac{1}{g} \left[v_{w1} u_1 \right]$

Hydraulic Efficiency is $\frac{R.P}{W.P} = \frac{\left[v_{w1} u_1 \pm v_{w2} u_2 \right]}{gH}$

For Francis turbine β=90^0.

Hydraulic Efficiency is $\eta_h = \dfrac{1}{gH}\left[v_{w1}u_1\right]$

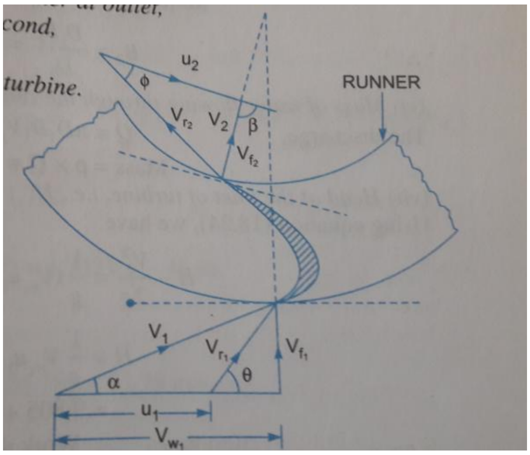


Fig. 9 Velocity triangles for Francis turbine

3. Outward flow Reaction Turbine:

Fig. shows outward flow reaction turbine in which the water from the casing enters the stationary guide wheel. The guide wheel consists of guide vanes which direct water to enter the runner which is around the stationary guide wheel. The water flows through the vanes of the runner in the outward radial direction and is discharged at the outer diameter of the runner. The inner diameter of the runner is the inlet and outer diameter is the outlet.

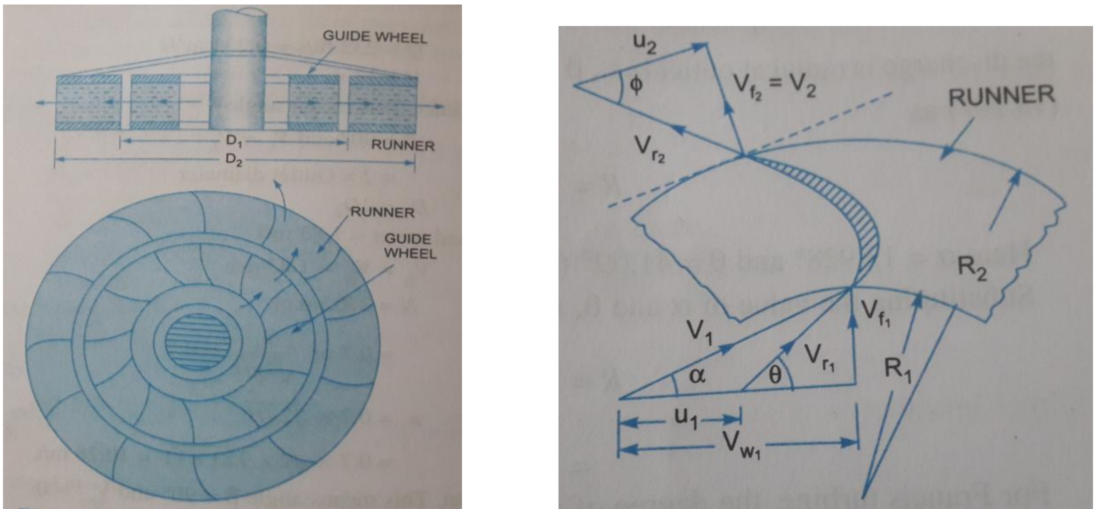
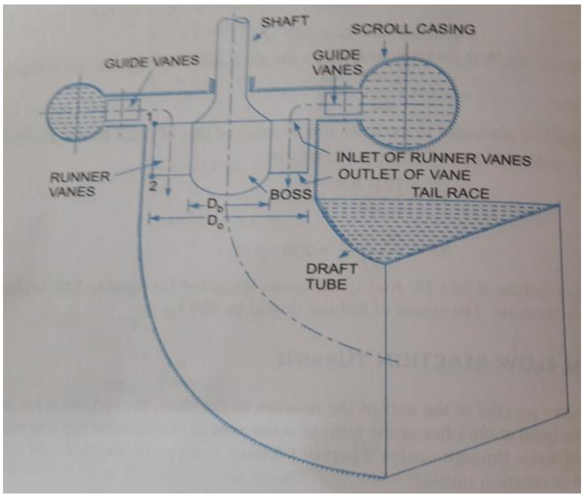


Fig. 10 Outward flow reaction turbine and velocity triangles with When $\beta=90^0$

4. Axial Flow turbines:

If the water flows parallel to the axis of the rotation of the shaft, the turbine is known as Axial flow Turbine. And if the head at the inlet of the turbine is the sum of pressure energy and kinetic energy and during the flow of water through the runner a part of pressure energy is converted to kinetic energy, this turbine is known as reaction turbine.

For the axial flow reaction turbine, the shaft of the turbine is vertical. The lower end of the shaft is made larger which is known as hub or boss. The vanes are fixed on the hub and hence, hub acts as a runner for axial flow reaction turbine.



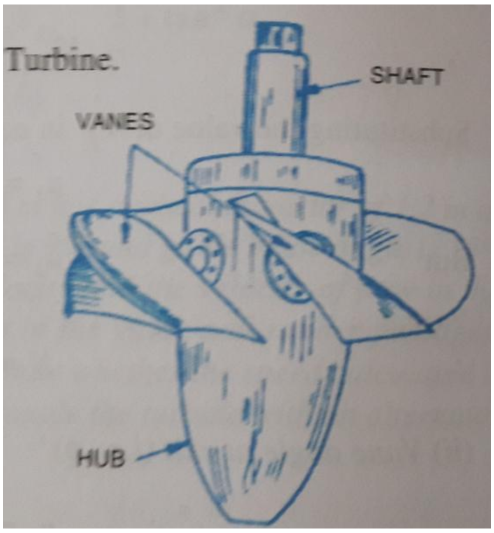


Fig. 11 Kaplan Turbine

The main parts of the Kaplan turbine are:

1. Scroll casing 2.Guide vane mechanism 3. Hub with vanes 4. Draft tube.

Fig. 11 shows all main parts of Kaplan Turbine. The water from penstock enters the scroll casing and then moves to the guide vanes. From the guide vanes, the water turns through 90^0 and flows axially through the runner.

The discharge through the runner, $Q = \frac{\pi}{4}\left[D_0^2 - D_b^2\right] \times v_{f1}$

Where D_0 is the outer diameter of the runner

D_b is the diameter of the hub.

V_{f1} is the velocity of flow at inlet.

5. Draft Tube:

The draft tube is a 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 runner 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 submerged below the level of water in the tail race.

Advantages of draft tube:

1. 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.

2. It converts a large portion of the kinetic energy 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, the net head on the turbine increases. The turbine develops more power and also the efficiency of the turbine increases.

Types of draft Tubes:

The following are the important types of draft- tubes which is commonly used.

1. Conical draft- tubes.
2. Simple elbow tubes.
3. Moody spreading tubes.
4. Elbow draft- tubes with circular inlet and rectangular outlet.

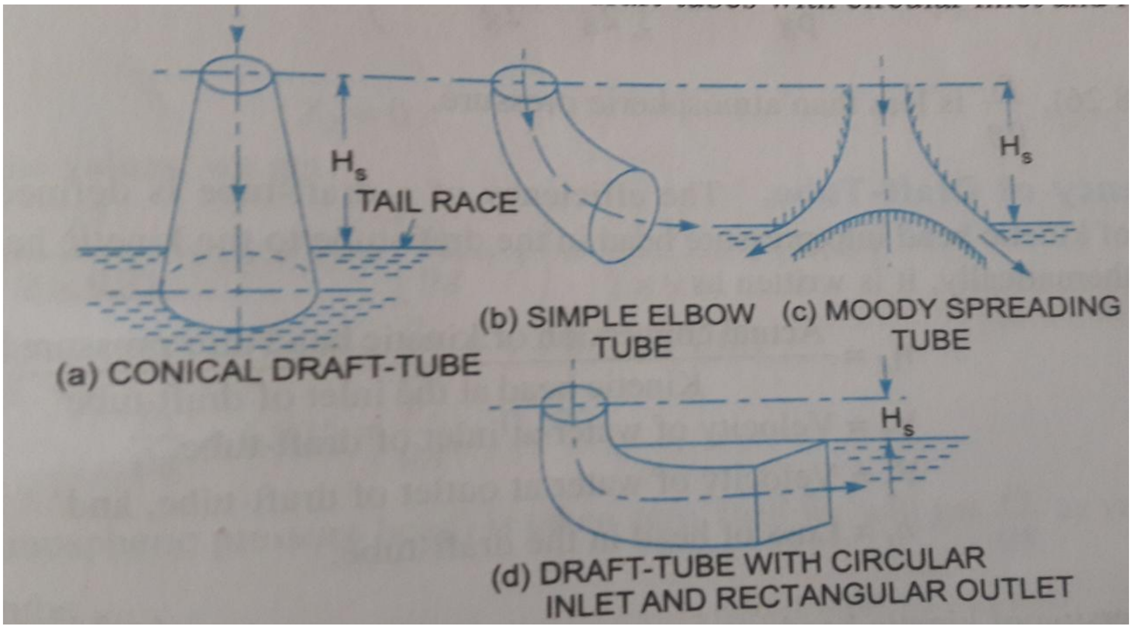


Fig. 12 Types of draft tubes

Draft tube Theory:

Consider a conical draft tube as shown in fig. 13.

H_s = vertical height of draft tube above tail race.

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

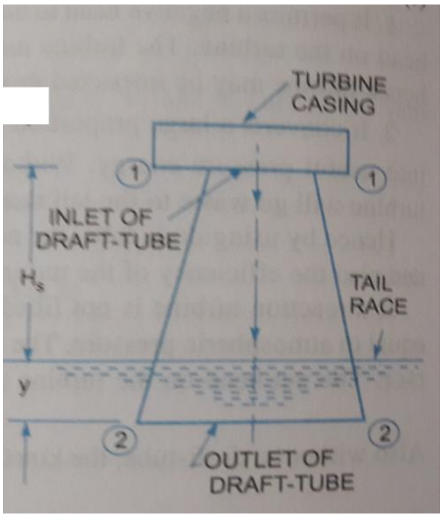


Fig. 13 Draft tube

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

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

Where h_f is loss of energy between section 1-1 and 2-2

$$\frac{p_2}{\rho g}=\frac{p_a}{\rho g}+y = \text{atmospheric pressure head} + \text{y}$$

Substituting the value of $\frac{p_2}{\rho g}$ in the above equation.

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

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

Efficiency of Draft Tube:

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

Efficiency of draft tube = Actual conversion of kinetic head into pressure head/kinetic head at the inlet of the draft tube.

V_1 = velocity of water at inlet of draft tube.

V_2 = Velocity of water at the outlet of draft tube.

h_f = loss of head at the inlet of draft tube.

Theoretical conversion of kinetic head into pressure head in draft tube $= \frac{V_1^2-V_2^2}{2g}$

Actual conversion of kinetic head into pressure head = $\frac{V_1^2-V_2^2}{2g}-h_f$

Therefore $\eta_d = \dfrac{\dfrac{V_1^2 - V_2^2}{2g} - h_f}{\dfrac{V_1^2 - V_2^2}{2g}}$

Specific Speed:

It is defined as the speed of the turbine which is identical in shape, geometrical dimensions, blade angles, gate openings etc., with the actual turbine but of such a size that it will develop unit power when working under unit head. It is denoted by N_s. It is used in comparing the different types of turbines as every type of turbine has different specific speed.

$$\eta_0 = \frac{S.P}{W.P} = \frac{P}{\rho g Q H}$$

$$P \propto Q \times H$$

Now let D= diameter of actual turbine.
N= speed of actual turbine
u = tangential velocity of turbine.
N_s = Specific speed of the turbine.
V = Absolute velocity of water.

Considering $V \propto u \propto \sqrt{H}$

$$u = \frac{\pi D N}{60} \Rightarrow u \propto DN$$

From the above equations: $\sqrt{H} \propto DN \quad \Rightarrow D \propto \dfrac{\sqrt{H}}{N}$

The discharge through of the turbine is given by Q = Area x velocity

$$Q \propto D^2 \sqrt{H} \Rightarrow Q \propto \frac{H}{N^2}\sqrt{H} \Rightarrow Q \propto \frac{H^{3/2}}{N^2}$$

Power α $P \propto \dfrac{H^{3/2}}{N^2} \times H \Rightarrow P \propto \dfrac{H^{5/2}}{N^2} \Rightarrow N \propto \dfrac{H^{5/4}}{\sqrt{P}}$

P= K$\dfrac{H^{5/2}}{N^2}$ where K is constant of proportionality.

If P =1, H=1, the speed N= specific speed N_s. substituting these values in the above equation, we get

$$1 = \frac{K(1)^{5/2}}{N_s^2}$$

Or $N_s^2 = K$

$$P = N_s^2 \frac{H^{5/2}}{N^2} \text{ or } N_s = \frac{N\sqrt{P}}{H^{5/4}}$$

Significance of specific speed:

Specific speed plays an important role for selecting the type of turbine. Also the performance of the turbine can be predicted by knowing the specific speed of the turbine.

Unit Quantities:

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

Important unit quantities are

i. Unit speed
ii. Unit discharge
iii. Unit power

Unit speed (Nu):
It is defined as the speed of the turbine working under a unit head.
N = Speed of a turbine under Head H.
u = tangential velocity
Tangential velocity, $u = k\sqrt{2gH} \Rightarrow u \propto \sqrt{H}$
Also $u = \dfrac{\pi DN}{60} \Rightarrow N \propto u \propto \sqrt{H}$
N = K_1 $\sqrt{H}$ where K_1 is proportionality constant.
If head on the turbine becomes unity, the speed becomes unit speed. Or when H =1, N = N_u
Therefore $N_u = K_1$

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

Unit discharge (Qu):
It is defined as the discharge passing through a turbine, which is working unit head.
H = head of water on turbine.
Q = discharge passing through the turbine when head on the turbine is H.
Q= area x velocity.
Q α velocity α $\sqrt{H}$
Q = $K_2\sqrt{H}$
Where K_2 is constant of proportionality.
If H =1, then Q = Q_u.
Therefore $Q_u = K_2$.
From the above

$$Q = Qu\sqrt{H} \Rightarrow Qu = \frac{Q}{\sqrt{H}}$$

Unit Power:
It is defined as the power developed by a turbine, working under a unit head. Denoted by P_u
Let H = head of water on the turbine.
P = power developed by the turbine under a head of H
Q = discharge through the turbine under head H

The overall efficiency $\eta_0 = \frac{S.P}{W.P} = \frac{P}{\rho g Q H}$

$$P \propto Q \times H \Rightarrow P \propto H^{1/2} \times H \Rightarrow P \propto H^{3/2}$$

$P = K_3\ H^{3/2}$
where K_3 is constant of proportionality.
When H=1m, $P=P_u$.
Therefore $P_u = K_3$.
Substituting the K_3 Value, $P = P_u\ H^{3/2}$

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

Cavitation:
Cavitation is defined as the phenomenon of formation of vapor bubbles of a flowing liquid in a region where the pressure of the liquid falls below its vapor pressure and the sudden collapsing of these vapor bubbles in a region of higher pressure. When the vapor bubbles collapse, a very high pressure is created. The metallic surfaces above which these bubbles collapse, is subjected to these high pressures, which cause pitting action on the surface. Thus cavities are formed on the metallic surface and also considerable noise and vibrations are produced.

Effects of cavitation:

(i) The metallic surfaces are damaged and cavities are formed on the surfaces.
(ii) Due to sudden collapse of vapor bubble, considerable noise and vibrations are produced.
(iii) The efficiency of a turbine decreases due to cavitation. Due to pitting action, the surface of the turbine blades becomes rough and the force exerted by water on the turbine blades decreases. Hence the work done by water or Output horse power becomes less and thus efficiency decreases.

Characteristic curves of hydraulic turbines:

Characteristic curves of hydraulic turbines are the curves, with the help of which the exact behavior and performance of the turbine under different working conditions can be known. These curves are plotted from the results of the tests performed on the turbine under different working conditions. The important parameters which can be varied during a test on a turbine are

(i) Speed (ii) Head (iii) Discharge (iv) Power (v) Overall efficiency and (vi) Gate opening.

The following are the important characteristic curves.

1. Main characteristic curves or constant head curves
2. Operating characteristic curves or constant speed curves.
3. Constant efficiency curves.

Main characteristic curves or Constant Head curves:

Main characteristic curves are obtained by maintaining a constant head and constant Gate openings on the turbine. The speed of the turbine is varied by changing the load on the turbine. For each values of the speed (N) the corresponding values of power (P) and discharge (Q) are obtained. Then the overall efficiency (η_0) for each value of the speed is calculated. From these readings, the values of unit speed (N_u), unit power (P_u), and unit discharge (Q_u) are determined. Taking N_u as abscissa, the values of the values of Q_u, P_u, P and η_0 are plotted. By changing the gate openings the values of Q_u, P_u, P and η_0 and N_u are determined and taking N_u as abscissa, the values of the values of Q_u, P_u, and η_0 are plotted.

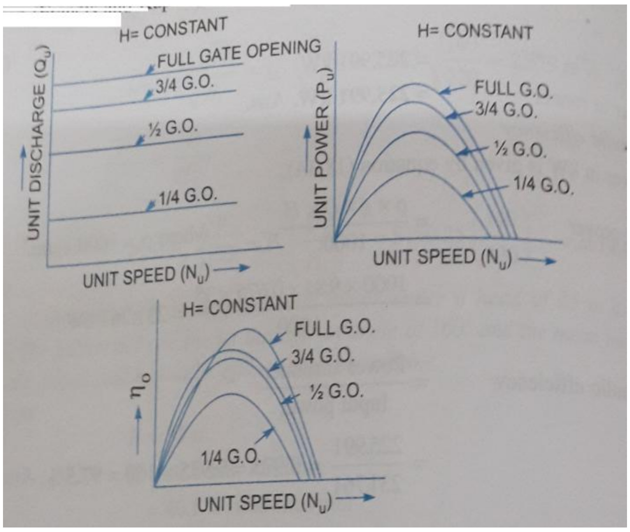


Fig. 14 Main characteristic curves for pelton wheel

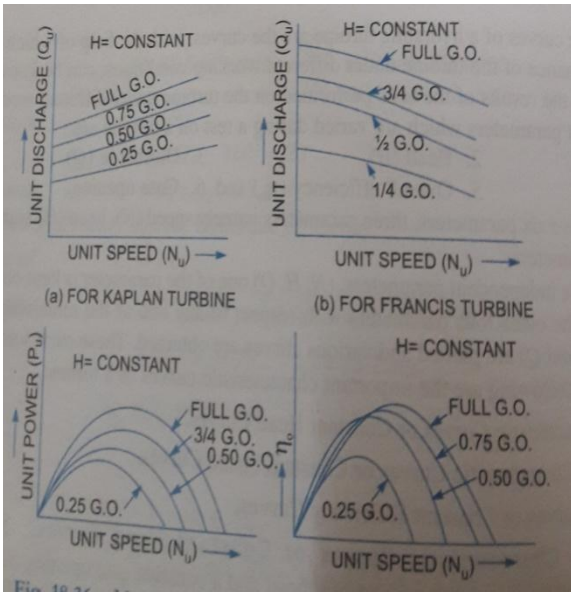


Fig. 15 Main characteristic curves for reaction turbine

Constant efficiency curves or Iso Efficiency curves:

These curves are obtained from the speed Vs efficiency and speed Vs discharge Curves for different Gate openings. For a given efficiency from the N_u Vs η_0 curves, there are two speeds. From N_u Vs Q_u curves corresponding to two values of speeds there are two values of discharge. Hence for a given efficiency, there are two values of discharge for a particular gate opening. This means for a given efficiency there are two values of speeds and two values of discharge for a given gate opening. If the efficiency is maximum there is only one value. These two values of speed and two values of discharge corresponding to particular gate opening are plotted. The procedure is repeated for different gate openings and the curves Q Vs N are plotted. The points having the same efficiency are plotted. These are iso efficiency curves.

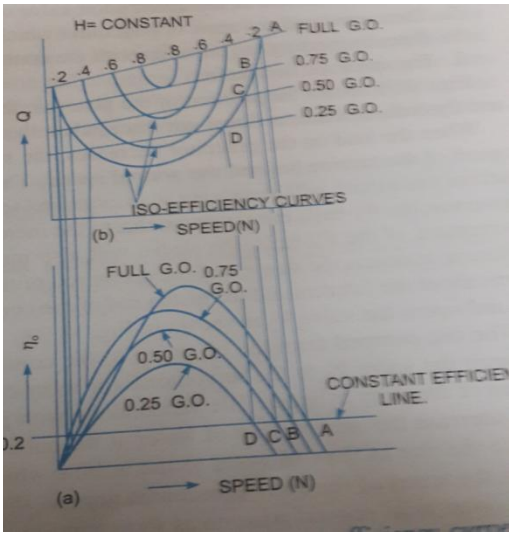


Fig. 16 Iso-efficiency curves

Governing of Pelton wheel Turbine

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

- Governing is done automatically by means of a governor, which regulates the rate of flow through the turbines according to the changing load conditions on the turbine.
- Governing of a turbine is necessary as a turbine is directly coupled to an electric generator, which is required to run at constant speed under all fluctuating conditions.
- The frequency of power generation by a generator of constant number of pair of poles under all varying conditions should be constant
- This is possible only when the speed of generator, under all changing load condition, is constant.
- The speed of the generator will be constant, when the speed of the turbine is constant.
- When the load on the generator decreases, the speed of the generator is increases beyond the normal speed (constant speed).
- Then speed of the turbine also increases beyond the normal speed.

If the turbine or the generator is to run at constant speed, the rate of flow of water to the turbine should be decreased till the speed becomes normal. This process by which the speed of the turbine is kept constant under varying condition of load is called governing.

Governing of Impulse Turbine (Pelton wheel):

Governing of impulse turbine is done by means of Oil pressure governor, which consist of following components

i. Oil sump.

ii. Gear pump / oil pump, which is driven by the power obtained from turbine shaft.
iii. Servomotor/ Relay cylinder.
iv. Control valve / Distribution valve / Relay valve
v. Centrifugal governor or pendulum, which is driven by belt or gear from the turbine shaft.
vi. Pipes connecting the oil sump with the control valve and control valve with servomotor.
vii. Spear rod or needle.

The following figure shows the position of the piston in the relay cylinder, position of control or relay valve and fly balls of the centrifugal governor, when the turbine is running at the normal speed.

When the load on the generator decreases, the speed of the generator increases. This increases the speed of the turbine beyond the normal speed. The centrifugal governor, which is connected to the turbine main shaft, will be rotating at an increased speed. Due to increase in the speed of centrifugal governor, the fly balls move upward due to increased centrifugal force on them. Due to the upward movement of the fly balls, the sleeve will also move upward.

A horizontal lever, supported over a fulcrum, connects the sleeve and the piston rod of the control valve. As the sleeve moves up, the lever turns about the fulcrum of the piston rod of the control valve moves downward. This closes the valve V_1 and opens the valve V_2 as shown in below figure.

The oil pumped from the oil pump to the control valve or relay valve, under pressure will flow through the valve V_2 to the servomotor and will exert force on the face A of the piston of the relay cylinder.

The piston along with the piston rod and spear will moves towards right. This will decrease the area of flow of water at the outlet of the nozzle. This decrease of area of flow will reduce the rate of flow of water to the turbine which consequently reduces the speed of turbine. When the speed of turbine becomes normal, the fly balls, sleeve, lever and piston rod of control valve come to its normal position as shown in figure below.

When the load on generator increases, the speed of the generator and hence turbine decreases. The speed of the centrifugal governor is also decreases and hence centrifugal force acting on the fly balls also reduces. This brings the fly balls in down ward direction. Due to this, the sleeve moves down ward and lever turns about the fulcrum, moving the piston rod of the control valve in the upward direction.

This closes the valve $V_2$and opens the valve V_1. The oil under pressure from the control valve will move through valve V_1 to the servomotor and will exert a force on face B of the piston. This will move the piston along with the piston rod and spear towards left, increasing the area of flow of water at the outlet of the nozzle. This will increase the rate of flow of water to the turbine and consequently, the speed of the turbine will also increase, till the speed of the turbine becomes normal.

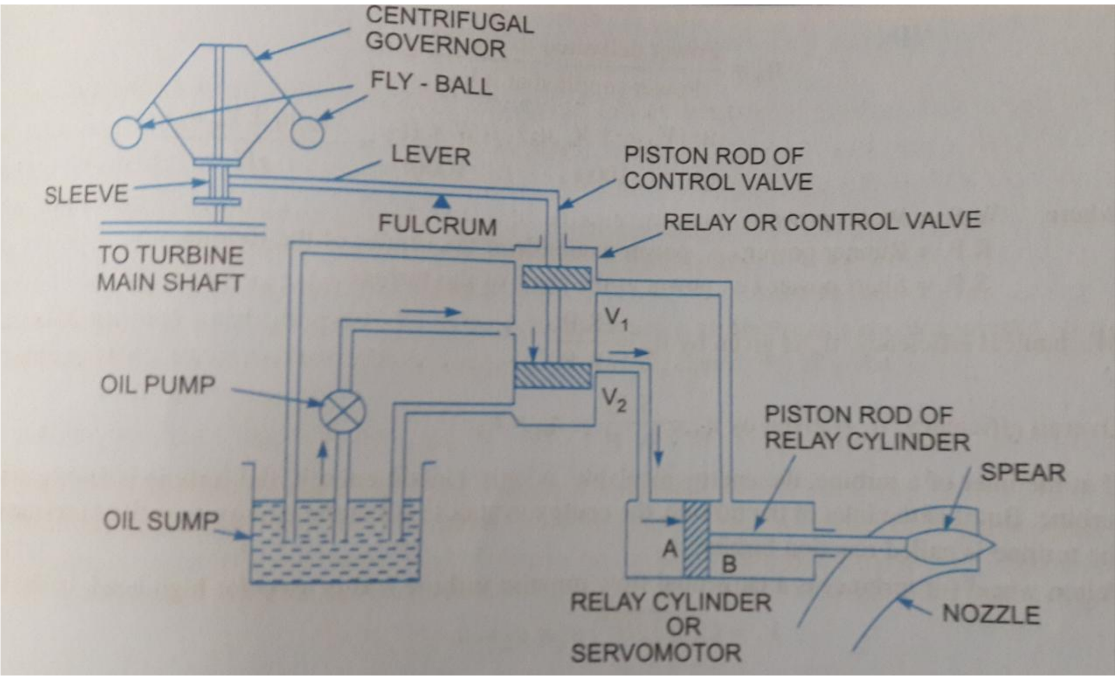


Fig. 17 Governing mechanism of Pelton wheel turbine

Unit _IV Steam Turbines

Introduction

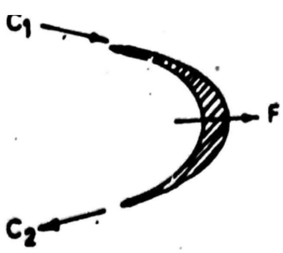
Steam turbine is a turbo machine. Turbo machines are those devices in which energy is transferred either to or from a continuously flowing fluid by the dynamic action of one or more moving blade rows. Here energy is transferred from fluid to blade rows and is decreasing along the flow direction.

Principle of operation of steam turbine:

Steam turbine is a prime mover which converts the heat energy of steam (at high pressure and temperature) into mechanical work.

The principle of operation of any turbine depends on Newton's second law of motion.

The motive power in a turbine is obtained by the change in momentum of high velocity jet impinging on a curved blade.



The steam from the boiler is expanded in a nozzle where due to fall in pressure of steam, thermal energy of steam is converted in kinetic energy of steam, resulting in the emission of a high velocity jet of steam which impinges on the moving vanes or blades, mounted on a shaft; here it undergoes a change in direction of motion which gives rise to a change in momentum and therefore, a force. This constitutes the driving force of the machine. This arrangement is shown in Fig.1

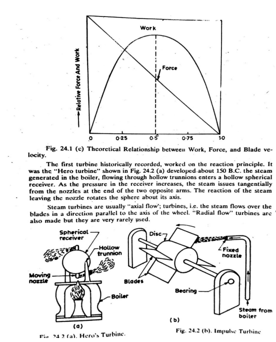


Fig. 24.1 (c) Theoretical Relationship between Work, Force, and Blade velocity.

The first turbine historically recorded, worked on the reaction principle. It was the "Hero turbine" shown in Fig. 24.2 (a) developed about 150 B.C. the steam generated in the boiler, flowing through hollow trunnions enters a hollow spherical receiver. As the pressure in the receiver increases, the steam issues tangentially from the nozzles at the end of the two opposite arms. The reaction of the steam leaving the nozzle rotates the sphere about its axis.

Steam turbines are usually "axial flow'; turbines, i.e. the steam flows over the blades in a direction parallel to the axis of the wheel. "Radial flow" turbines are also made but they are very rarely used.

Fig. 24.2 (a). Hero's Turbine.

Fig. 24.2 (b). Impulse Turbine

In brief, the turbine is a prime mover in which the working fluid enters at higher energy level and comes out at low energy level and in doing so it does work.

History : The first turbine historically recorded, worked on the reaction principle. It was the "Hero turbine" shown in the Fig .2 developed about 150 B.C. the steam generated in the boiler, flowing through hollow trunnions enters a hollow spherical receiver. As the pressure in the receiver increases, the steam issues tangentially from the nozzles at the end of the two opposite arms. The reaction of the steam leaving the nozzle rotates the sphere about its axis.

Types of steam turbines:

- On the basis of principles of operations, steam turbines are classified as:
    - Impulse turbine
    - Impulse reaction turbine
  - *Impulse turbine*: If the flow of steam through the nozzles and moving blades of a turbine takes place in such a manner that "the steam is expanded only in nozzles, and pressure at the outlet side of blades is equal to that at inlet side", i.e. drop in pressure of steam takes place only in nozzles and not in moving blades; such a turbine is termed as impulse turbine because it works on the principle of impulse. This is obtained by making the blade passage of constant cross sectional area.
  - *Impulse reaction turbine*: Expansion of steam takes place in nozzles as well as in moving blades. If the pressure of steam at the outlet from the moving blades of a turbine is less than that at inlet side of blades; this pressure drop suffered by steam while passing through the moving blade causes a further generation of kinetic energy within the blades, giving rise to reaction and adds to the propelling force which is applied through the rotor to the turbine shaft, such a turbine is termed as an impulse-reaction turbine because it works on the principle of impulse and reaction, both. This is achieved by varying the blade passage cross sectional area (converging type).
- On the basis of direction of flow, steam turbines may be classified as
    - Axial flow turbine (all power plants turbines are of this type)
    - Radial flow turbines

Steam turbines are usually ‘axial flow’; turbines, i.e. the steam flows over the blades in a direction parallel to the axis of xthe wheel. “Radial flow” turbines are also made but they are very rarely used.

The simple impulse turbine:

This type of turbine works on the principle of impulse and is shown in Fig. 3 It consists of a nozzle or a set of nozzles, a rotor mounted on a shaft, one set of moving blades attached to the rotor and casing etc.

The uppermost portion of the diagram shows a longitudinal section through the upper half of the turbine,

The middle portion shows the development of the nozzles and blading i.e. the actual shape of the nozzles and blading,

The bottom portion shows the variation of absolute velocity and absolute pressure during flow of steam through passage of nozzles and blades.

An example of this type of turbine is the De-Laval turbine.

approximately constant and equal to the condenser pressure.

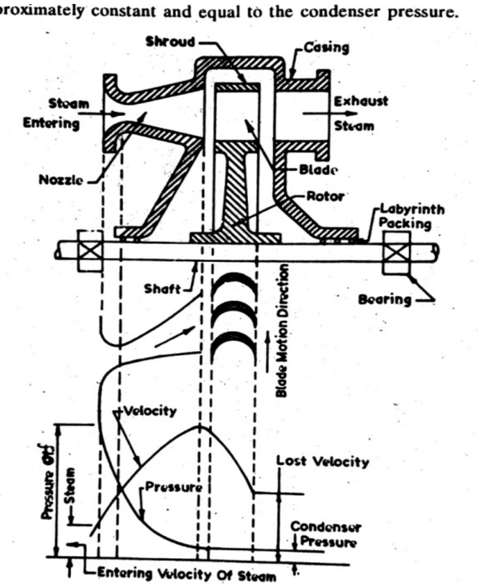


Fig. 24.3. Diagramatic Arrangement of a Simple Impulse Turbine.

Fig. Simple Impulse Turbine

It can be seen from the figure that the complete expansion of steam from the boiler pressure to the exhaust pressure of condenser pressure takes place only in one set of nozzles i.e. the pressure drop takes place only in nozzles. It is assumed that the pressure in the recess between nozzles and blades remain the same. The steam at condenser pressure or exhaust pressure enters the blade and comes out at the same pressure i.e. the pressure of the steam in the blade passages remains approximately constant and equal to the condenser pressure.

Compounding of impulse turbine:

This method is employed for reducing the rotational speed of the impulse turbine to practical limits. If the high velocity of steam is allowed to flow through one row of moving blades, it produces a rotor speed of about 30,000 rpm, which is too high for practical use. Not only this, the leaving loss is also very high. It is therefore, essential to incorporate, improvements in the simple impulse turbine as to make it more efficient and pragmatic.

This is achieved by making use of more than one set of nozzles, blades, rotors, in series, keyed to a common shaft, so that either the steam pressure or the jet velocity is absorbed by the turbine in stages. This also reduces the leaving loss. This process is called compounding of steam turbine. There are three main types of compound turbines:

- Pressure compounded impulse turbine
- Velocity compounded impulse turbine
- Pressure and Velocity compounded impulse turbine

Pressure compounded impulse turbine:

In this type of turbine, the compounding is done for pressure of steam only i.e. to reduce the high rotational speed of the turbine the whole expansion of steam is arranged in a number of steps by employing a number of simple impulse turbine in a series on the same shaft as shown in the Fig.4. Each of the simple impulse turbine consists of one set of nozzles and one row of moving blades is known as a stage of the turbine, and thus, this turbine consists of several stages. The exhaust from each row of moving blades enters the succeeding set of nozzles. Thus we can say that this arrangement is nothing but splitting up of the whole pressure drop from the steam chest pressure to the condenser pressure into a series of smaller pressure drops across several stages of impulse turbine, and hence, this turbine is called pressure compounded impulse turbine.

The pressure and velocity variation is also shown in the Fig.4 The nozzles are fitted into a diaphragm which is locked in the casing. This diaphragm separates one wheel chamber from another. All rotors are mounted on the same shaft and the blades are attached on the rotor.

The expansion of steam only takes place in the nozzles while pressure remains constant in the moving blades because each stage is a simple impulse turbine. It can be seen from the pressure curve that the space between any two consecutive diaphragms is filled with steam at constant pressure; the pressure on either side of the diaphragm is different. Since the diaphragm is a stationary part, there must be clearance between the rotating shaft and the diaphragm. The steam tends to leak through this clearance for which devices like labyrinth packing etc. are used.

Since the drop in pressure of steam per stage is reduced, the steam velocity leaving the nozzles and entering the moving blades is reduced which in turn reduces the blade velocity. Hence for economy and maximum work shaft speed is significantly reduced to suit practical purposes. Thus rotational speed may be reduced by increasing the number of stages according to ones need. The leaving velocity of the last stage of the turbine is, thus, much less compared to the de-Lavel turbine and, the leaving loss is not more than 1 to 2% of the initial total available energy. This turbine was invented by the late Prof. L. Rateau and so it is also known as Rateau turbine.

Velocity compounded impulse turbine:

In this turbine, the compounding is done for velocity of steam only i.e. velocity drop is arranged in many small drops through many moving rows of blades instead of single row of moving blades. It consists of a nozzle or a set of nozzles and rows of moving blades attached to the rotor or wheel and, rows of fixed blades attached to the casing as shown in fig.

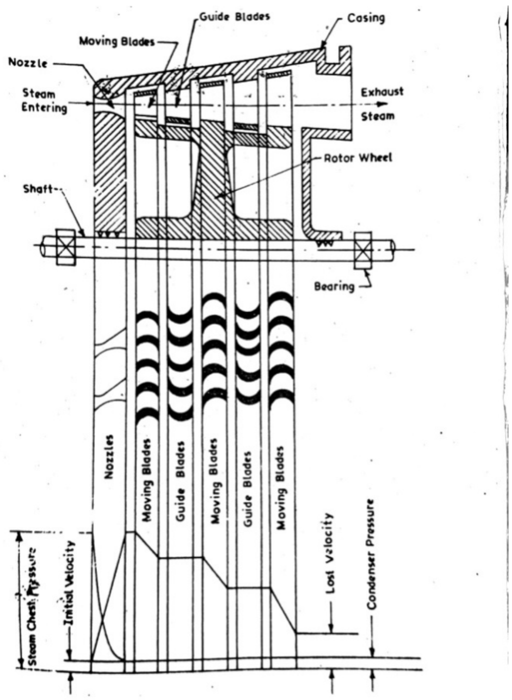


Fig. 24.5. Diagrammatic Arrangement of Velocity compounded Impulse Turbine.

The fixed blades are guide blades that guide the steam to the succeeding rows of moving blades suitable arranged between the moving blades but set in a reversed manner. In this turbine, three rows or rings of moving blades are fixed on a single wheel or rotor and this wheel is termed as the three row wheel. The arrangement consists of two rows of guide blades or fixed blades placed between the first and the second and the second and the third rows of moving blades.

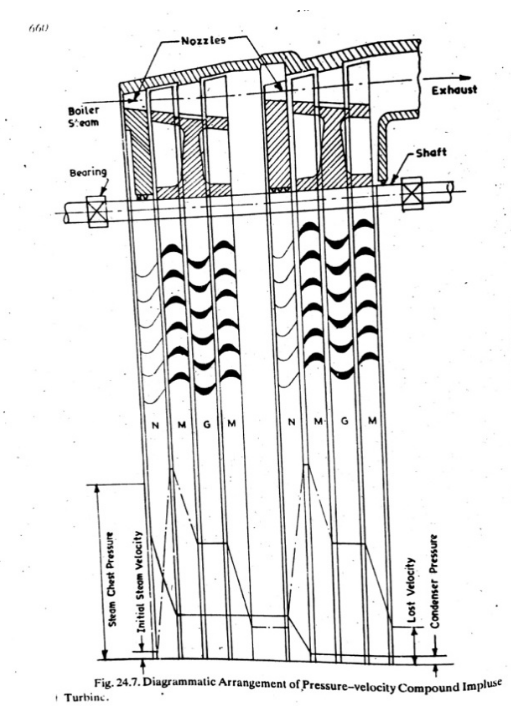


Fig. Velocity compounding - Principle

Fig. Pressure-Velocity compounding -

Principle

The expansion of steam from the steam chest pressure down to the exhaust pressure takes place in the nozzles only. There is no drop in pressure either in the moving blades or the fixed blades i.e. the pressure remains constant in the blades as in the simple impulse turbine. The steam velocity from the exit of the nozzle is very high similar to the simple impulse turbine. The steam with high velocity enters the first row of moving blades and, on passing through these blades, the velocity reduces slightly i.e. the steam gives up a part of its kinetic energy and reissues from this row of blades. It then enters the first row of guide blades which directs it to the second row of moving blades. A slight drop in velocity takes place in the fixed or guide blades due to friction. On passing through the second row of moving blades, there is a slight drop in velocity again i.e. steam gives up some more of its kinetic energy to the rotor. After this, it is again directed y the second row of guide blades to the third row of moving blades, again drop in velocity occurs and finally the steam leaves the wheel with a much reduced velocity in a more or less axial direction. Compared to the simple impulse turbine, the leaving velocity is small being about 2% of initial total available energy of steam.

So we can say that this arrangement is nothing more than the splitting up of the velocity gained from the exit of the nozzles into many small drops through several rows of fixed and moving blades. This type of turbine is also termed as **Curtis turbine**. Because of its low efficiency a three row wheel is used for driving small machines. It may be noted that a two row wheel is more efficient than the three row wheel.

Pressure-Velocity compounded impulse turbine:

This turbine is a combination of pressure-velocity compounding and is shown in fig. The arrangement in the figure is for two rotors. There are two wheels and two rotors and only two rows of moving blades are attached on each rotor because two row wheels are more efficient than three row wheels. The steam on passing through each wheel or rotor reduces in velocity i.e. drop in velocity is achieved by the many rows of moving blades and is velocity compounded. The whole pressure drop takes place in the two sets of nozzles i.e. the whole pressure drop is divided into small drops, hence it is pressure compounded.

In the first set of nozzles, there is a slight decrease in pressure which gives some kinetic energy to the steam and there is no pressure drop in the two rows of moving blades of the first wheel and in the first row of fixed blades. However, there is a velocity drop in moving blades which also occurs in the fixed blade due to friction. In second set of nozzles, the remaining pressure drop takes place, but the velocity here increases and, the drop in velocity take place in the moving blades of the second, wheel or rotor.

Compared to the pressure compounded impulse turbine this arrangement was more popular due to its simple construction, but because of its low efficiency it is rarely used now a day.

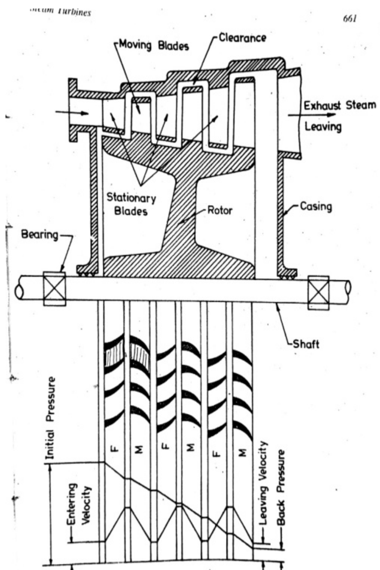


Fig 24.8. Diagrammatic Arrangement of Impulse-reaction Turbine.

Impulse reaction turbine:

This turbine utilizes the principle of impulse and reaction both and is shown in fig. There are a number of rows of moving blades attached to the rotor with an equal number of fixed blades attached to the casing. In this turbine, the fixed blades are set in a reverse manner compared to the moving blades, and correspond to nozzles mentioned in connection with the impulse turbine. Due to the position of the fixed row of blades at the entrance, in place of nozzles, steam is admitted for the whole circumference and hence there is an all round or complete admission. In passing through the first row of fixed blades, the steam undergoes a small drop in pressure and hence its velocity increases slightly. It then enters the first row of moving blades just as in the impulse turbine, and suffers a change in direction and therefore results in momentum. This momentum gives rise to an impulse on the blades.

But in this turbine, the passage of the moving blades is so designed that there is a small drop in pressure of steam in the moving blades which results in an increase in the kinetic energy of steam. This kinetic energy gives rise to a reaction in the direction opposite to that of added velocity. Thus, the gross propelling force or driving force is the vector sum of impulse and reaction forces. Commonly this type of turbine is known as Reaction turbine. The pressure and velocity variations are shown the fig. It can be seen in the figure that there is a gradual drop in pressure in both the moving and the fixed blades.

In this turbine as the pressure falls, the specific volume increases and hence the height of blades is increased in steps i.e. up to 4 stages it may remains constant then it may increase and remain constant for the next two stages.

In this turbine, the steam velocities are comparatively moderate and its maximum value is nearly equal to blade velocity. In general practice, to reduce the number of stages, the steam velocity is arranged greater than the blade velocity. The leaving loss is about 1 to 2% of the total initial available energy. This turbine is popular in power plants. An example of this type of turbine is the Parsons-Reaction turbine.

Pure Reaction turbine:

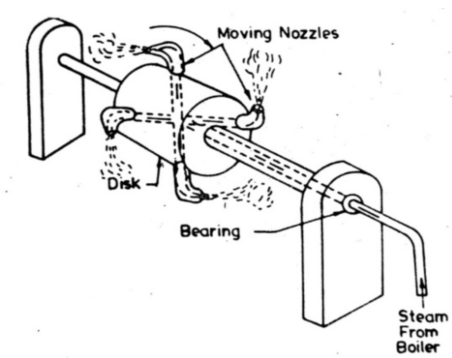


Fig. 24.9 (a). Principle of Pure Reaction Turbine

Pure reaction turbine is not possible in practice. However, the principle of pure reaction turbine needs explanation; fig. shows a reaction turbine in which steam expands through nozzles which are moving. High temperature and pressure steam from the boiler enters a hollow disc through a hollow shaft. The disc contains four radial openings through the tubes, the ends of which are shaped as nozzles. With the escape of steam through Fig. Principle of Pure Reaction Turbine

these tubes, it expands and there is an increase in steam velocity relative to the rotating disc.

This gives rise to a reaction force which sets the disc in rotation. Note that the disc and shaft rotate in the opposite direction of the steam jet.

Combination turbines:

A combination of two different types of turbines offers certain advantages. Such a turbine is called combination turbine. The most common combination are as under:

- Velocity compounded impulse wheel turbine followed by several single row impulse wheel turbine termed as Curtis and Rateau type.
- Velocity compounded impulse wheel turbine followed by several stages of impulse reaction blading, termed as Curtis-Parsons type. This is common practice in steam power plant to reduce number of stages.
- One or more single row impulse stages followed by several stages of impulse reaction blading, termed as Rateau-Parsons type. This is common in steam power plant to reduce number of stages.

Difference between Impulse and Reaction turbine:

IMPULSE TURBINE	REACTION TURBINE
• Pressure drops in nozzles and not in moving blade channels.	• Pressure drops in fixed blades as well as moving blades.
• Constant blade channels area.	• Varying blade channels area.
• Profile type blades.	• Aerofoil type blades.
• Restricted round or incomplete admission of steam.	• All round or complete admission.
• Diaphragm contains the nozzles.	• Fixed blades similar to moving blades attached to casing serve as nozzles and guide the steam.
• Not much power developed.	• Power developed is considerable.
• Occupies less space for same power.	• Occupies more space for same power.
• Lower efficiency.	• Higher efficiency.
• Suitable for small power requirements.	• Suitable for medium and high power requirements.
• Blade manufacturing is not difficult and thus not costly.	• Blade manufacturing process is difficult compared to impulse and hence costly.
• Velocity of steam is higher.	• Velocity of steam is less.
• In multi stage impulse turbine drop per stage varies from 2 to 3.	• In multi stage reaction turbine, pressure drop per stage varies from 1.5 to 2.

Velocity diagrams for impulse turbine:

The main parts of the impulse turbine are nozzles and blades. The steam is utilized in the nozzles to produce a jet of steam moving with a high velocity. The function of the blades is to change the direction of the high velocity steam and thus, the momentum of the jet or jets of steam so as to produce a force which propels the blades. Since this propelling force is caused by a change of momentum through a change in the direction of flow of the steam, it is essential to draw the diagrams showing the variation of velocity of steam during its flow through the blade passages.

It is an established fact that velocity is a vector quantity. It has a) magnitude (b) direction and (c) sense of direction. Thus we can represent velocity by a straight line and indicate (a) its magnitude by the length of that straight line drawn to a suitable scale, (b) its direction relative to the said straight line with reference to some fixed direction and (c) its sense of direction by an arrow placed on the straight line

Let suffixes 1 and 2 refer to inlet and outlet conditions of blades as shown in fig.

C_1	=	Absolute velocity of steam at inlet to the moving blade, i.e. exit velocity of nozzle
α_1	=	Angle of nozzle or angle made by the entering steam having velocity C_1 with tangent of the wheel at entrance of moving blade
C_{r1}	=	Relative velocity of steam with respect to the tip of the blade at inlet
C_{w1}	=	Tangential component of velocity C_1 known as velocity of whirl or the component of velocity responsible for the whirling or rotating of the turbine rotor at inlet
C_{f1}	=	Axial component of C_1 called velocity of flow
C_2	=	Absolute velocity of steam at outlet from the blade
C_{r2}	=	Relative velocity with respect to the tip of blade at outlet
α_2	=	Angle to the tangent of wheel at which the absolute velocity of steam C_2 leaves the blades
β_2	=	Outlet angle of the blade
C_{f2}	=	Outlet axial velocity of flow at outlet
C_{w2}	=	Tangential component of velocity C_2 known as the velocity of whirl at outlet
m	=	Mass flow rate, kg/s
u	=	Peripheral velocity of blade
k	=	Blade velocity coefficient
C_{r1}	=	C_{r2} for case of without friction

$kC_{r1} \quad = \quad C_{r2}$ for case of with friction

e blades.

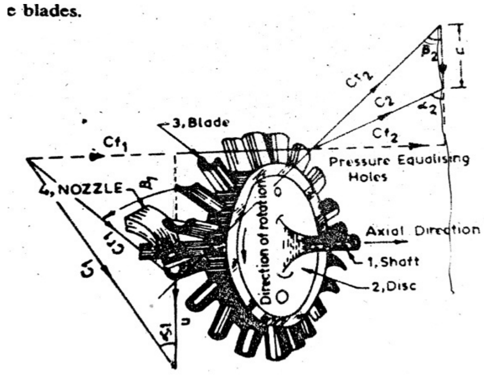


Fig. 24.10. Impulse Rotor.

Fig. Impulse Rotor

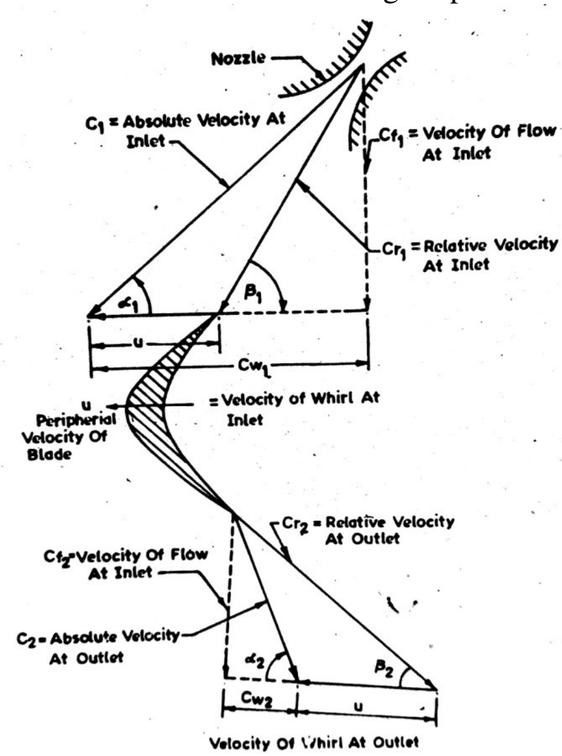


Fig. 24.11. Velocity Diagram for Impulse Turbine.

Fig. velocity diagram for impulse turbine

The jet of steam with absolute velocity C_1 impinges on the moving blade at an angle of α_1 to the tangent of blade. The tangential component of this jet does work on the blade, because it is in the same direction as the motion of the blade u; this tangential component is known as the velocity of whirl C_{w1}. The axial component C_{f1} of velocity C_1 does not work on the blade because it is perpendicular to the direction of the motion of the blade. This component C_{f1} is known as the velocity of flow because it is responsible for the flow through the turbine. The velocity of flow causes an axial thrust on the rotor.

Due to the motion of the blade u in the peripheral direction the entering steam jet with velocity C_1 will have a relative velocity C_{r1} to the blade which makes an angle of β_1 to the tangent of blade. This relative velocity may be found by subtracting the vectors of the steam velocity C_1 and the blade velocity u.

As the steam is to glide over the blade without shock, it follows that the surface of the blade at inlet must be parallel to the relative velocity C_{r1}. That is, the moving blade at inlet must be inclined to the tangent of the blade at an angle β_1. This is the principle on which the blade is designed.

A similar vector diagram is shown at the outlet tip of the moving blade. The steam glides off the blade and rushes out with a relative velocity C_{r2} inclined at an angle of β_2 to the tangent. Addition of the vector of blade velocity u and relative velocity C_{r2} gives the absolute velocity of the leaving steam at C_2 inclined at an angle α_2 to the tangent. After obtaining the vector C_2 its tangential component C_{w2}, the axial component C_{f2} may be drawn.

Combination of vector diagram:

The industrial method of constructing vector diagrams is to combine both the velocity diagrams at inlet as well as at the outlet of the blade into a single diagram as shown in fig. In this figure, both the velocity diagrams are drawn on a common base representing the blade velocity u.

When there is no friction, ($C_{r1} = C_{r2}$) and $\beta_1 = \beta_2$, then $C_{f1} = C_{f2}$.

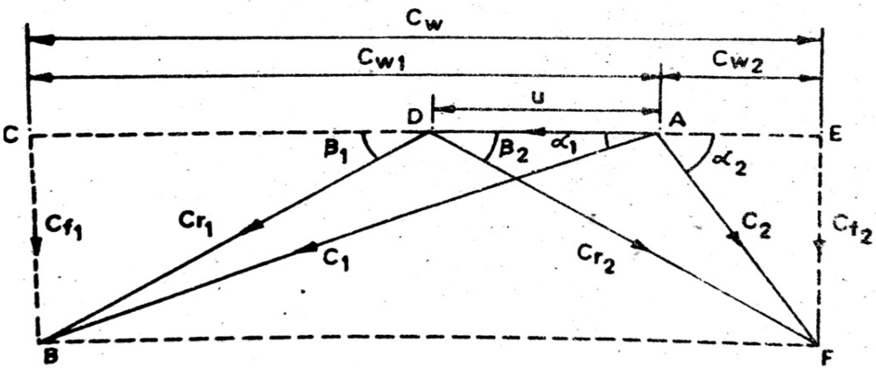


Fig. 24.12. Combined Velocity Diagram

Fig. Combined velocity diagrams

Effect of blade friction on velocity diagrams:

In actual practice, there is frictional resistance to the flow of steam jet over the blade, the effect of which is to cause a slowing down of the relative velocity C_{r2} which is in the range of 85 to 90% of C_{r1}. Thus due to frictional resistance

$$C_{r2} = K\, C_{r1}$$

where K is called the blade velocity coefficient and takes into account the blade losses due to friction.

$DF = DB' = K\, C_{r1} = C_{r2}$

The resultant force parallel to BF is inclined at an angle to the plane in which the wheel rotates. This resultant force may be resolved into two components as shown in fig.

- a useful thrust or tangential force (F_t) acting in the direction of blade motion
- an idle thrust or axial force (F_a) perpendicular to the wheel plane

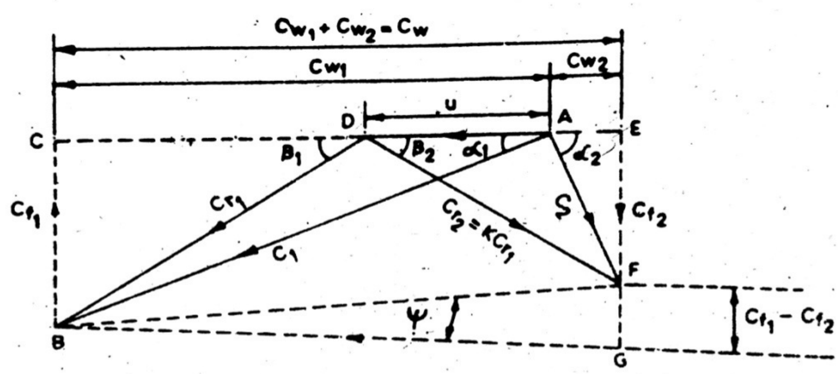


Fig. 24.13. Velocity Diagram showing Effect of Friction.

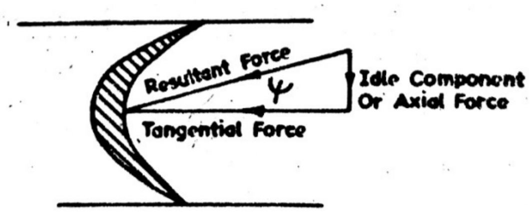


Fig. 24.14. Blade Force Diagram.

Fig. Combined velocity diagram with friction effect and blade force diagram

Work done on the blade:

The work done on the blade may be found out from the change of momentum of the steam jet during its flow over the blade. As earlier discussed, it is only the velocity of whirl which performs work on the blade since it acts in its direction of motion.

From Newton's second law of motion,

Force (tangential) on the wheel = mass of steam ×acceleration

= mass of steam/sec × change of velocity

= $\dot{m}_s$ (C_{w1} - C_{w2})

The value of C_{w2} is actually negative as the steam is discharged in the opposite direction to the blade motion, therefore due consideration should be given to the fact that values of C_{w1} and C_{w2} are to be added while doing the solution of the problem. (i.e., α_2 < 90^0).

Work done on the blades/se. = force × distance travelled/sec.

= $\dot{m}_s$ (C_{w1} - C_{w2}) × u

power per wheel = $\dot{m}_s$ (C_{w1} + C_{w2}) × u

= in KW

Blade or diagram efficiency =

=

=

If h_1 and h_2 be the total heats before and after expansion through the nozzles, then (h_1-h_2) is the heat drop through a stage of fixed blades ring and moving blades ring.

Stage efficiency, η_{stage} =

=

Nozzle efficiency =

Also, η_{stage} = Blade efficiency × nozzle efficiency

$= \times$

$=$

The axial thrust on the wheel is due to difference between the velocities of flow at entrance and outlet.

Axial force on the wheel = mass of steam × axial acceleration

$$= \dot{m}_s (C_{f1} - C_{f2})$$

The axial force on the wheel must be balanced or must be taken by a thrust bearing.

Energy converted to heat by blade friction = loss of kinetic energy during flow of blades

$$= \dot{m}_s ($$

Blade velocity Co-efficient:

In an impulse turbine, if friction is neglected the relative velocity will remain unaltered as it passes over blades. In practice the flow of steam over the blades is resisted by friction. The effect of the friction is to reduce the relative velocity of steam as it passes over the blades. In general there is a loss of 10 to 15% in the relative velocity. Owing to friction in the blades, C_{r2} is less than C_{r1} and we may write

$$C_{r2} = K\, C_{r1}$$

Where K is termed as blade velocity co-efficient

Expression for optimum value of the ratio of blade speed to steam speed (for a maximum efficiency) for a single stage impulse turbine:

From fig.

$$C_w = PQ = MP + MQ = C_{r1} \cos \beta_1 + C_{r2} \cos \beta_2$$

$$= C_{r1} \cos \beta_1$$

$$= C_{r1} \cos \beta_1 (1 + K.Z) \quad \text{Where } Z = \quad .... \text{(i)}$$

Generally, the angles β_1 and β_2 are nearly equal for impulse turbine and hence it can safely assumed that Z is constant.

But, $C_{r1} \cos \beta_1 = MP = LP - LM = C_1 \cos \alpha_1 - u$

From equation (i), $C_w = (C_1 \cos \alpha_1 - u)(1 + K.Z)$

We know that, Blade efficiency, $\eta_b =$ .... (ii)

$\eta_b =$

let us introduce a parameter blade velocity ratio $\rho =$ and $u = \rho C_1$

substitute 'u' value in above equation we get,

$= 2(\rho \cos \alpha_1 - \rho^2)(1+KZ)$ ....(iii)

Where $\rho =$ is the ratio of blade speed to steam speed and is commonly called as "Blade speed ratio".

For particular impulse turbine α_1, K and Z may assumed to be constant and from equation (iii) it can be seen clearly that η_b depends on the value of ρ only. Hence differentiating (iii),

$= 2(\cos \alpha_1 - 2\rho)(1+KZ)$

For a maximum or minimum value of η_{blade} this should be zero.

$\cos \alpha_1 - 2\rho = 0,$ $\rho =$

Now, $= 2(-2)(1+KZ) = -4(1+KZ)$

which is negative quantity and thus the value so obtained is the maximum.

Optimum value of ratio of blade speed to steam speed is

$\rho_{opt} =$

Substituting this value of ρ in equation (iii),we get

$(\eta_b)_{max} = 2 \times$ $(1+KZ)$

$=$ $(1+KZ)$

It is sufficiently accurate to assume symmetrical blades ($\beta_1 = \beta_2$) and no friction in fluid passage for the purpose of analysis.

$Z = 1$ and $K = 1$

$(\eta_b)_{max} =$

The work done per kg of steam is given by

$$W = (C_{w1} + C_{w2}) \times u$$

Substituting the value of $(C_{w1} + C_{w2})$ as C_w then

$$W = (C_1 \cos \alpha_1 - u)\ (1 + K.Z) \times u$$

$$= 2\ u\ (C_1 \cos \alpha_1 - u) \text{ when } K = 1 \text{ and } Z = 1$$

The maximum value of W can be obtained by substituting the value of $\cos \alpha_1$from equation.

$$\cos \alpha_1 = 2\rho = 2$$

$$W_{max} = 2\ u(2\ u - u) \quad = \quad 2\ u^2$$

It is obvious from the equation that the blade velocity should be approximately half of absolute velocity of steam jet coming out from the nozzle for the maximum work developed per kg of steam or for maximum efficiency. For the other values of blade speed the absolute velocity at outlet from the blade will increase, consequently, more energy will be carried away by the steam and efficiency will decrease.

For equiangular blades with no friction losses, optimum value of corresponds to the case, when the outlet absolute velocity is axial as shown in fig.

Since the discharge is axial, $\alpha_2 = 90^0$, so $C_2 = C_{f2}$ and $C_{w2} = 0$.

The variations of η_b or work developed per kg of steam with is shown in fig. This figure shows that:

- When = 0, the work done becomes zero as the distance travelled by the blade (u) is zero, even though the torque on the blade is maximum.
- The maximum efficiency is and maximum work done per kg of steam is 2 u^2 when .
- When = 1, the work done is zero as the torque acting on the blade becomes zero even though the distance travelled by the blade is maximum.

Reaction turbines (Impulse reaction Turbines):

The reaction turbines which are used these days are really impulse-reaction turbines. Pure reaction turbines are not in general use. The expansion of steam and heat drop occur both in fixed and moving blades.

Velocity diagram for reaction turbine blade:

Fig. shows the velocity diagram for reaction turbine blade. In case of an impulse turbine blade the relative velocity of steam either remains constant as the steam glides over the blades or is reduced slightly due to friction. In reaction turbine blades, the steam continuously expands at it flows over the blades. The effect of the continuous expansion of steam during the flow over the blade is to increase the relative velocity of steam.

$C_{r2} > C_{r1}$ for reaction turbine blade

$C_{r2} \leq C_{r1}$ for impulse turbine blade

Let AB = C1, be the jet velocity at the outlet from the nozzles or the fixed blades. AD = u is the blade velocity. Then DB will be the relative velocity at the inlet to the blade.

The angle is made appreciably less than . for pure impulse the relative velocity of the steam at the outlet from the blade would be DG = K Cr1 where k is blade velocity coefficient. So the final absolute velocity would be AG. The change in velocity suffered by the steam would be presented by BG.

The change in velocity would give rise to the impulse force *gb*, which is equal to *m* (GB) and parallel to GB

Due to expansion of steam in the moving blades, the steam velocity is increased from GD to GF. This increase in outlet relative velocity give s rise to a reaction force *fg* which is equal to *m*(FG) and parallel to FG. By combining the impulse force *gb* and reaction force *fg*, we get the resultant force *fb* which is equal to *m*(FB). Components of the resultant force is tangential and axial directions give the tangential and axial force respectively

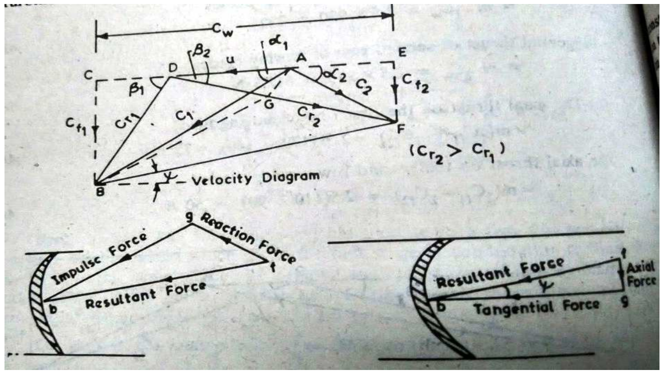


Fig. Combined velocity diagram and blade force diagram for Impulse - Reaction turbine

Degree of reaction:

The degree of reaction turbine stage is defined as the ratio of heat drop over moving blades to the total heat drop in the stage.

Thus the degree of reaction of reaction turbine is given by:

R_d =

=

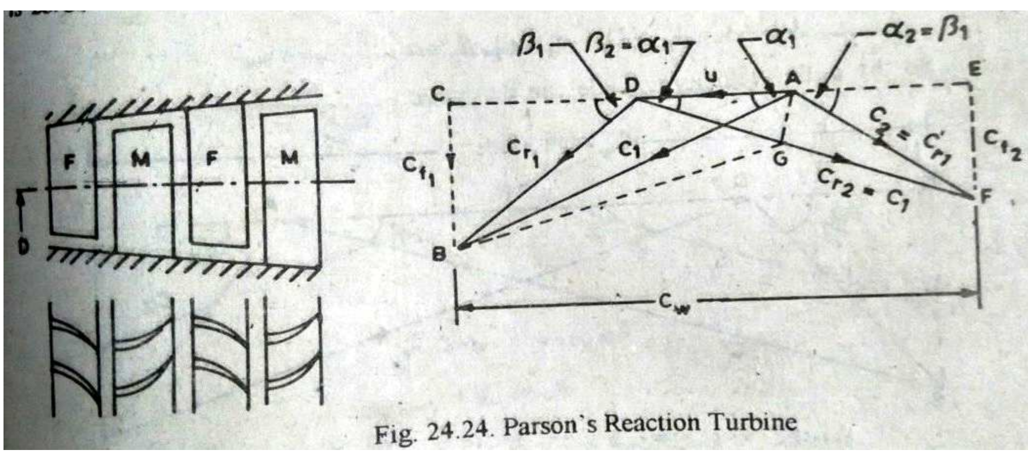


Fig. velocity diagram for Parsons reaction turbine

The heat drop in moving blades is equal to increase in relative velocity of steam passing through the blade.

$\Delta h_m =$

The total heat drop in the stage ($\Delta h_f + \Delta h_m$) is equal to the work done by the steam in the stage and it is given by

$$(\Delta h_f + \Delta h_m) = (C_{w1} - C_{w2}) \times u$$

$$R_d = \quad \ldots\ldots\ldots\ldots\ldots\ldots \text{(iv)}$$

Referring to fig.

$$C_{r2} = C_{f2} \operatorname{cosec} \beta_2 \text{ and } C_{r1} = C_{f1} \operatorname{cosec} \beta_1$$

$$\text{and } C_{w1} - C_{w2} = C_{f1} \cot \beta_1 + C_{f2} \cot \beta_2$$

The velocity of flow generally remains constant through the blades.

$$C_{f1} = C_{f2} = C_f$$

Substituting the values of C_{r1}, C_{r0} and ($C_{w1} - C_{w2}$) in eqn. (iv) we get,

$$R_d =$$

$$=$$

$$=$$

$$= (\cot \beta_2 - \cot \beta_1)$$

If the turbine is designed for 50% reaction ($\Delta h_f = \Delta h_m$) then the above equation can be written as

$$= (\cot \beta_2 - \cot \beta_1)$$

$$u = C_f (\cot \beta_2 - \cot \beta_1)$$

Also U can be written as

$$u = C_f (\cot \beta_2 - \cot \alpha_2)$$

$$u = C_f (\cot \alpha_1 - \cot \beta_1)$$

$C_{f1} = C_{f2} = C_f$ is assumed in writing the above equations.

Comparing the above equations we can write, $\beta_1 = \alpha_2$ and $\beta_2 = \alpha_1$, which means that moving blade and fixed blade must have the same shape if the degree of reaction is 50%. This condition gives symmetrical velocity diagrams. This type of turbine is known as Parson's reaction turbine. Velocity diagram for the blades of this turbine is given in fig.

Conditions for maximum efficiency:

The condition for maximum efficiency is derived by making the following assumptions:

- The degree of reaction is 50%
- The moving and fixed blades are symmetrical
- The velocity of steam at exit from the preceding stage is same as velocity of steam at the entrance to the succeeding stage.

Work done per kg of steam,

$$W = (c_{w1} + C_{w2}) \times u$$

$$= u\,[C_1 \cos \alpha_1 + (C_{r2} \cos \beta_2 - u)]$$

As $\beta_2 = \alpha_1$ and $C_{r2} = C_{r1}$ as per the assumptions

$$W = u\,[2C_1 \cos \alpha_1 - u]$$

$$=$$

$$= [2\rho \cos \alpha_1 - \rho^2], \qquad \text{where } \rho =$$

The K.E. supplied to the fixed blade =

The K.E. supplied to the moving blade =

Total energy supplied to the stage, $\Delta h =\ +\ $ as $C_{r2} = C_1$ for symmetrical triangles

$$\Delta h =\ +$$

$$= C_1^2 -$$

Considering ΔLMS from fig.

$$C_{r1}^2 = C_1^2 + u^2 - 2C_1\, u \cos \alpha_1$$

Substituting this value of C_{r1}^2 in above equation, we get

Total energy supplied to the stage,

$$\Delta h = C_1^2 - (C_1^2 + u^2 - 2C_1 u \cos \alpha_1)/2$$

$$= (C_1^2 - u^2 + 2C_1 u \cos \alpha_1)/2$$

$$=$$

$$= [1+2\rho \cos \alpha_1 - \rho^2]$$

The blade efficiency of the reaction turbine is given by,

$$\eta_b =$$

Substituting the value of W and Δh from above equations:

$$\eta_b =$$

$$=$$

$$=$$

$$=$$

$$= 2 -$$

The η_b becomes maximum when the value of $[1+2\rho \cos \alpha_1 - \rho^2]$ becomes maximum.

The required equation is:

$$(1+2\rho \cos \alpha_1 - \rho^2) = 0$$

$$2 \cos \alpha_1 - 2 \rho = 0$$

$$\rho = \cos \alpha_1$$

Substituting the value of ρ from above equation to eqn. the value of maximum efficiency is given by:

$(\eta_b)_{max} = 2 -$ $= 2[1 -]$ $=$

Hence $(\eta_b)_{max} =$

.

UNIT-V

CENTRIFUGAL COMPRESSORS

Learning Objectives:

- To study the working and performance characteristics of rotary compressors.

Syllabus:

Centrifugal compressors: Mechanical details and principle of operation – velocityand pressure variation. Energy transfer, impeller blade shape losses, velocity triangles, analysis of flow through compressors, slip factor, power input factor, pressure coefficient, adiabatic coefficient, compressor efficiency, surging and choking.

Axial Flow Compressors: Mechanical details and principle of operation – velocity triangles and energy transfer per stage, degree of reaction, work done factor -isentropic efficiency- pressure rise calculations – Polytropic efficiency, comparison of centrifugal and axial compressors. Surging and stalling

Learning Outcomes: At the end of the unit, the student will be able to

- Describe the working of rotary compressors

- Analyze the performance of centrifugal and axial flow compressors

Centrifugal Compressors

- **Construction:**

  A centrifugal compressor is a radial flow roto dynamic fluid machine that uses mostly air as the working fluid and utilizes the mechanical energy imparted to the machine from outside to increase the total internal energy of the fluid mainly in the form of increased static pressure head.

  A centrifugal compressor essentially consists of three components.

  - A stationary casing
  - A rotating impeller as shown in Fig. 1 (a) which imparts a high velocity to the air. The impeller may be single or double sided as show in Fig. 1 (b) and (c), but the fundamental theory is same for both.
  - A diffuser consisting of a number of fixed diverging passages in which the air is decelerated with a consequent rise in static pressure.

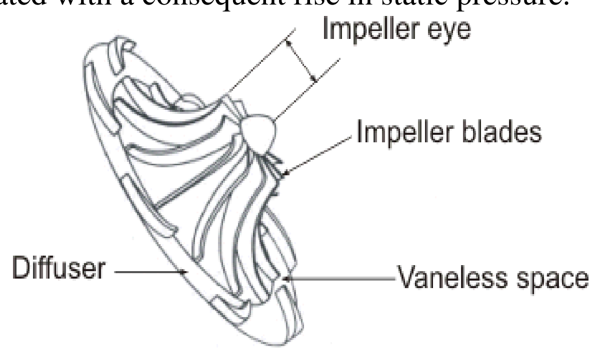


Fig. 1 (a)

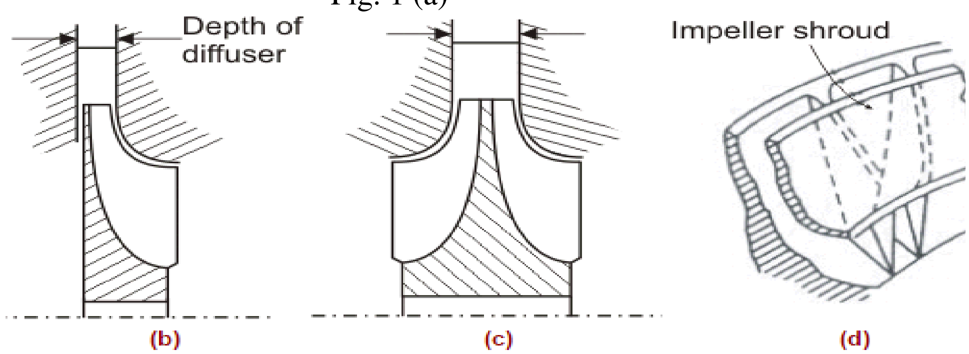


Fig. 1 Schematic views of a centrifugal compressor

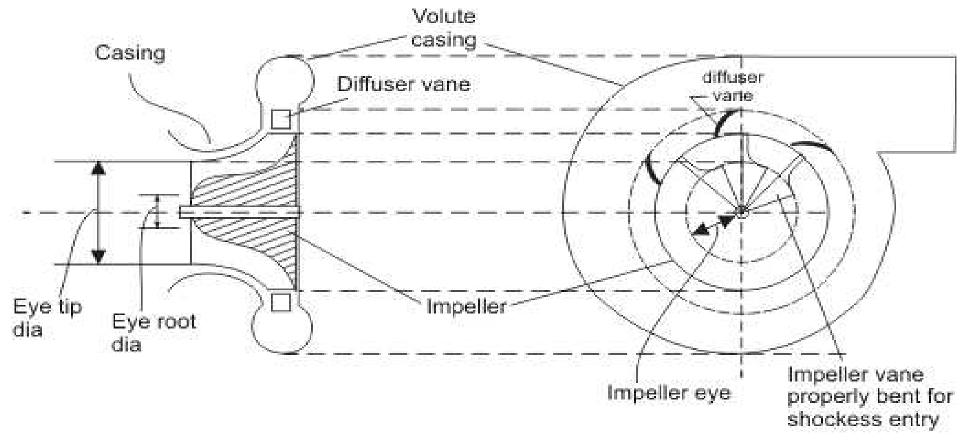


Fig. 2 Single entry and single outlet centrifugal compressor

Figure 2 is the schematic of a centrifugal compressor, where a single entry radial impeller is housed inside a volute casing.

- **Principle of operation:** Air is sucked into the impeller eye and whirled outwards at high speed by the impeller disk. At any point in the flow of air through the impeller the centripetal acceleration is obtained by a pressure head so that the static pressure of the air increases from the eye to the tip of the impeller. The remainder of the static pressure rise is obtained in the diffuser, where the very high velocity of air leaving the impeller tip is reduced to almost the velocity with which the air enters the impeller eye.

  Usually, about half of the total pressure rise occurs in the impeller and the other half in the diffuser. Owing to the action of the vanes in carrying the air around with the impeller, there is a slightly higher static pressure on the forward side of the vane than on the trailing face. The air will thus tend to flow around the edge of the vanes in the clearing space between the impeller and the casing. This results in a loss of efficiency and the clearance must be kept as small as possible. Sometimes, a shroud attached to the blades as shown in Figure.1(d) may eliminate such a loss, but it is avoided because of increased disc friction loss and of manufacturing difficulties.

  - The straight and radial blades are usually employed to avoid any undesirable bending stress to be set up in the blades. The choice of radial blades also determines that the total pressure rise is divided equally between impeller and diffuser.
  - Before further discussions following points are worth mentioning for a centrifugal compressor.
  - The pressure rise per stage is high and the volume flow rate tends to be low. The pressure rise per stage is generally limited to 4:1 for smooth operations.
  - Blade geometry is relatively simple and small foreign material does not affect much on operational characteristics.
  - Centrifugal impellers have lower efficiency compared to axial impellers and when used in aircraft engine it increases frontal area and thus drag.

Multistage is also difficult to achieve in case of centrifugal machines.

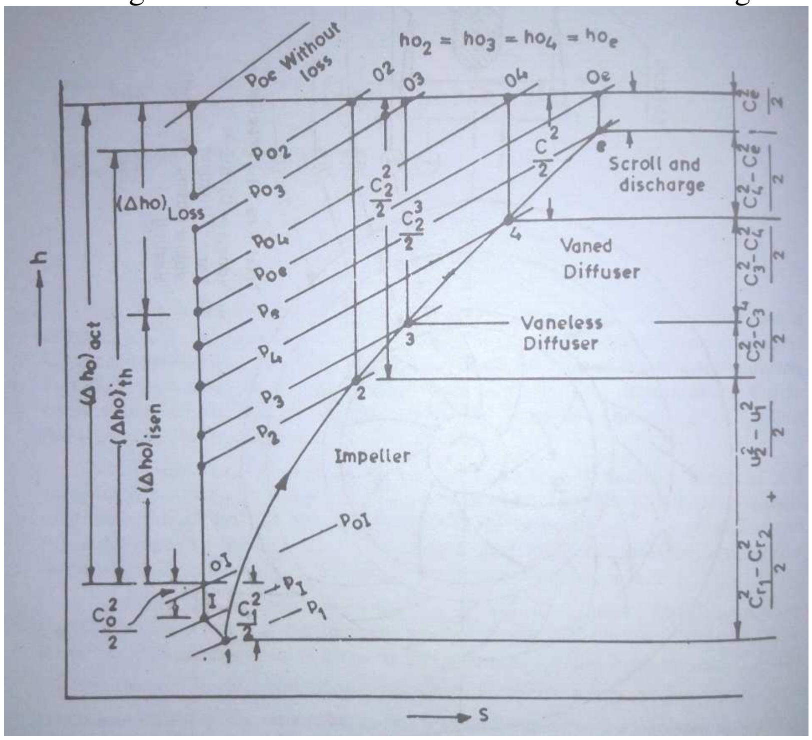


Fig. 3 Process representation on h-s chart

- **Pressure velocity variation curve:**

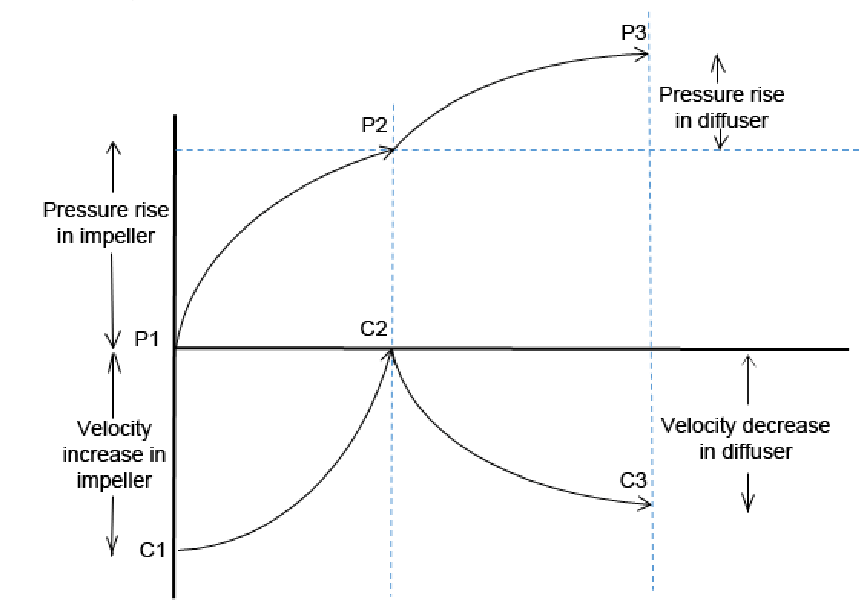


Fig.4: Pressure velocity variation curve

- **Advantages of centrifugal compressors:**
  - Low weight, easy to design and manufacture.
  - Suitable for continuous compressed air supply, such as cooling unit.

- The oil free in nature.
- They have fewer rubbing parts.
- High-flow rate than the positive displacement compressor.
- Relatively energy efficient.
- Wide range of rotational speed.
- Centrifugal compressors are reliable, low maintenance.
- Generating higher pressure ratio per stage as compared to axial flow compressor.
- It does not require special foundation.

- **Disadvantages of centrifugal compressors:**
  - Large frontal area for a given air flow rate compared to the axial flow compressor.
  - Unsuitable for very high compression, limited pressure.
  - They are sensitive to changes in gas composition.
  - They work at high speed, sophisticated vibration mounting needed.
  - Problem of surging, stalling and choking.

- **Applications of centrifugal compressors:**
  - Food and beverage industry - centrifugal compressor provides oil free compressed air for some sensitive application such as food processing.
  - Centrifugal compressor meets high demand of compressed air.
  - Gas turbines, in automobile turbochargers and supercharger.
  - Oil refineries, natural-gas processing.
  - Refrigeration, air-conditioning and HVAC.
  - Manufacturing process compressed air for pneumatic tools.

- **Different vane shapes:**
  - The impellers may be classified depending on the exit angle β_2 into
    (i) Backward curved vanes – $\beta_2<90^0$
    (ii) Radial vanes – $\beta_2 = 90^0$
    (iii) Forward curved vanes – $\beta_2> 90^0$
- **Velocity diagrams:**
  - Fig 5 shows theimpellerandvelocitydiagramsattheinletandoutlet.

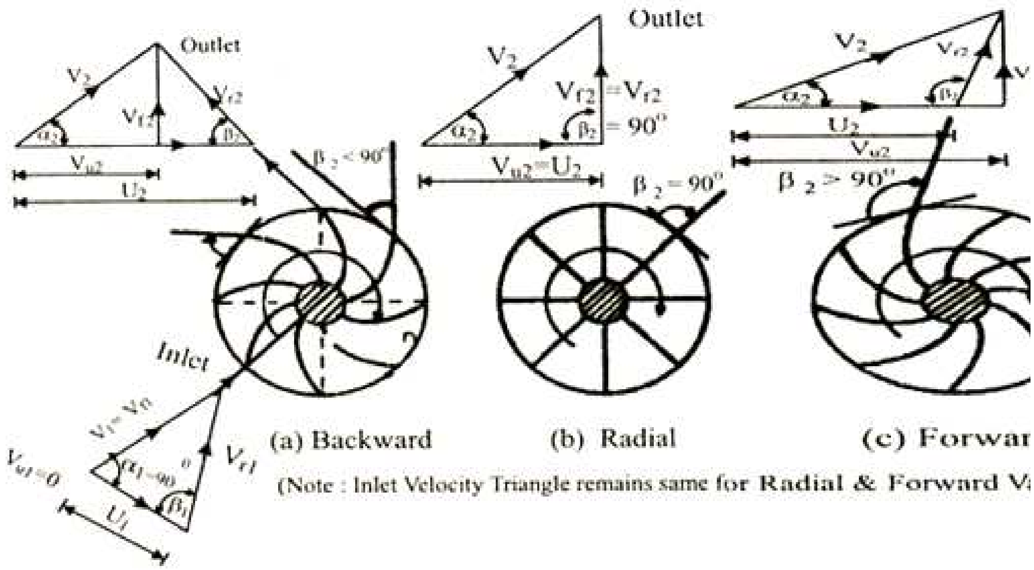


Fig 5: inlet and outlet velocity triangles for various types of impeller blades

V_1= absolute velocity at inlet (C_1)
V_2= absolute velocity at outlet (C_2)
V_{f1}= flow velocity at inlet (C_{f1})
V_{f2}= flow velocity at outlet (C_{f2})
V_{w1}= whirl velocity at inlet (C_{w1})
V_{w2}= whirl velocity at outlet (C_{w2})
V_{r1}= relative velocity at inlet (C_{r1})
V_{r2}= relative velocity at outlet (C_{r2})
U_1, U_2 are the corresponding blade velocities

Whentheairentersthe impellerinthe axialdirection then absolute velocityattheinlet,V_1=V_{f1}. TheabsolutevelocityV$_1$hasawhirlcomponent V_{w1}.Intheidealcase,aircomesoutfromtheimpellertipaftermakingan angleof 90º(i.e.,intheradialdirection),soV_{w2}= U_2.Thatis,thewhirlcomponentis exactlyequaltotheimpellertipvelocity.

- **Dimensionless parameters**
  - **Flow coefficient:** It is defined as the ratio of the mass flow rate to the mass flow rate referred to the impeller. Its value varies between 0.28 - 0.32.

Flow coefficient, $\phi = \dfrac{m}{\rho_2 A_2 u_2}$, $A_2 = \pi D_2 b_2$

Here mass flow rate, $m = \pi D_1 b_1 C_{f1} \rho_1 = \pi D_2 b_2 C_{f2} \rho_2$

where A_2 – flow area at the tip of the impeller, ρ_2 – density of the fluid at the tip of impeller

Flow coefficient, $\phi = \dfrac{m}{\rho_2 A_2 u_2} = \dfrac{\pi D_2 b_2 C_{f2} \rho_2}{\rho_2 \pi D_2 b_2 u_2} = \dfrac{C_{f2}}{u_2}$

- **Head Coefficient:** It is defined as the ratio of enthalpy increase in a stage to the kinetic energy corresponding to the tip peripheral velocity.

  Head coefficient, $\lambda = \dfrac{\Delta h}{u_2^2/2}$

- **Pressure coefficient:** It is defined as the ratio of isentropic enthalpy increase to the kinetic energy corresponding to the tip peripheral velocity.

  Pressure coefficient, $\Psi = \dfrac{\Delta h_{isen}}{u_2^2/2}$

  Also from the efficiency of compressor, $\eta_{isen} = \dfrac{\Delta h_{isen}}{\Delta h} \Rightarrow \Delta h_{isen} = \eta_{isen} \times \Delta h$

  Pressure coefficient, $\Psi = \dfrac{\Delta h_{isen}}{u_2^2/2} = \eta_{isen} \times \left(\dfrac{\Delta h}{u_2^2/2} \right) = \eta_{isen} \times \lambda$

- **Reaction:** It is defined as the temperature increase in the rotor to the temperature increase in the stage.

  Reaction, $\Omega = \dfrac{\Delta T_{rotor}}{\Delta T_{stage}}$

- **Slip and Slip Factor:**

  Slip: Under ideal conditions, the fluid particles follow the path of the blades resulting in such a relative velocity at impeller tip that makes an angle with tangential direction equal to the blade tip angle. This may be possible from an impeller with infinite number of blades of very less thickness. In this case the stream passes through the each blade channel following exactly the path of the blade. But in actual practice where the number of blades is finite, the air is trapped between the impeller vanes due to its inertia and thus the fluid is reluctant to move in the impeller.

  This results in a difference of pressures across the vane; there being a high pressure at the leading face and low pressure at the trailing face which gives rise to the relative velocity gradient. In high pressure side, the fluid leaves the vane tangentially and on the low pressure side, the fluid leaves the vane with certain circumferential component across the channel from one leading face to the other trailing face. Therefore the fluid slips with respect to the impeller during its passage through it.

  Slip factor: It is defined as the ratio of workdone under actual conditions (when no. of blades is finite) to the workdone under ideal conditions (when no. of blades is infinite).

Slip factor, $\mu = \dfrac{u_2 C_{w2} - u_1 C_{w1}}{u_2 C_{w2\infty} - u_1 C_{w1\infty}}$

For an impeller with no pre-whirl, $Cw1 = Cw1\infty = 0$

Finally, slip factor, $\mu = \dfrac{u_2 C_{w2}}{u_2 C_{w2\infty}} = \dfrac{C_{w2}}{C_{w2\infty}}$

- **Work done or Euler's work:**

  The theoretical torque will be equal to the rate of change of angular momentum experienced by the air. Considering a unit mass of air, this torque is given by theoretical torque,

  $$\tau = V_{w2} r_2 - V_{w1} r_1$$

  Where, V_{w2} is whirl component of V_2 and r_2 is impeller tip radius.

  Let ω = angular velocity. Then the theoretical work done on the air may be written as:

  2

  Theoretical work done $W = C_{w2} r_2\, \omega - C_{w1} r_1\, \omega$

  $$W = C_{w2} U_2 - C_{w1} U_1$$

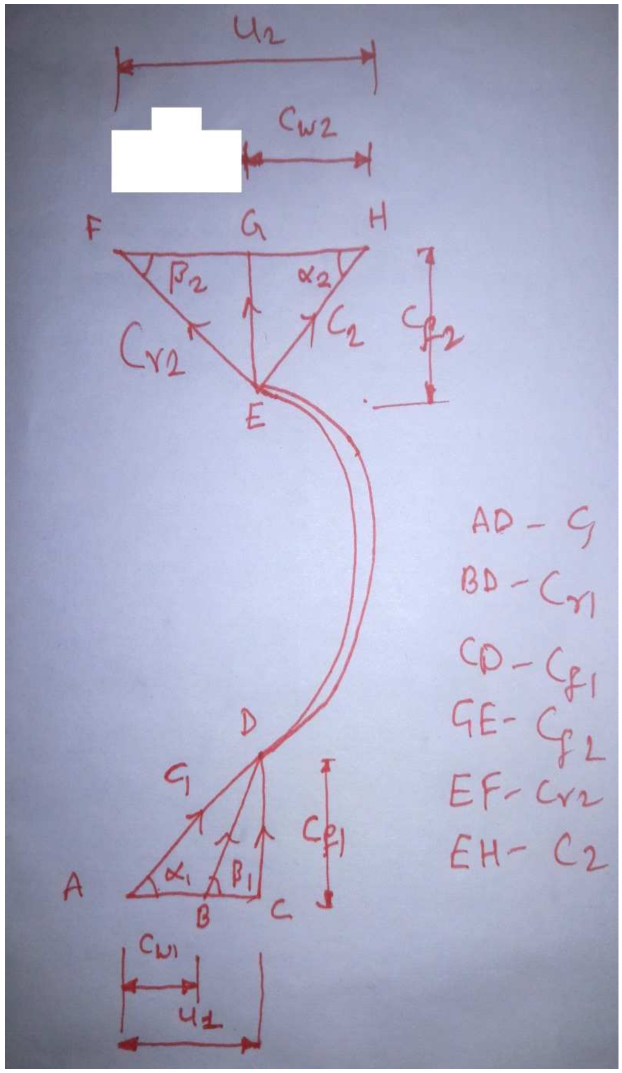


Fig. 6 Inlet and Outlet Velocity triangles for backward curved vane

From the inlet velocity triangle, ACD, $C_{w1} = C_1 \cos\alpha_1$

Using the principle of Cosine rule to ΔABD,

$$C_{r1}^2 = u_1^2 + C_1^2 - 2u_1C_1\cos\alpha_1 \Rightarrow C_{r1}^2 = u_1^2 + C_1^2 - 2u_1C_{w1}$$

On simplification, $$u_1C_{w1} = \frac{C_{r1}^2 - u_1^2 - C_1^2}{2}$$

Similarly, from the outlet velocity triangle, EGH, $C_{w2} = C_2 \cos\alpha_2$

Using the principle of Cosine rule to ΔEFH,

$$C_{r2}^2 = u_2^2 + C_2^2 - 2u_2C_2\cos\alpha_2 \Rightarrow C_{r2}^2 = u_2^2 + C_2^2 - 2u_2C_{w2}$$

On simplification, $$u_2C_{w2} = \frac{C_{r2}^2 - u_2^2 - C_2^2}{2}$$

Finally, work done, $w = C_{w2}u_2 - u_1 C_{w1} = \dfrac{C_{r1}^2 - C_{r2}^2}{2} + \dfrac{u_2^2 - u_1^2}{2} + \dfrac{C_2^2 - C_1^2}{2}$

UNIT-VI

AXIAL FLOW COMPRESSORS

- **Axial Flow Compressors**

The basic components of an axial flow compressor are a rotor and stator, the former carrying the moving blades and the latter the stationary rows of blades. The stationary blades convert the kinetic energy of the fluid into pressure energy, and also redirect the flow into an angle suitable for entry to the next row of moving blades. Each stage will consist of one rotor row followed by a stator row, but it is usual to provide a row of so called inlet guide vanes. This is an additional stator row upstream of the first stage in the compressor and serves to direct the axially approaching flow correctly into the first row of rotating blades. For a compressor, a row of rotor blades followed by a row of stator blades is called a stage.

Two forms of rotor have been taken up, namely drum type and disk type. A disk type rotor illustrated in Fig. 7 The disk type is used where consideration of low weight is most important. There is a contraction of the flow annulus from the low to the high pressure end of the compressor. This is necessary to maintain the axial velocity at a reasonably constant level throughout the length of the compressor despite the increase in density of air. Fig. 8illustrate flow through compressor stages. In an axial compressor, the flow rate tends to be high and pressure rise per stage is low. It also maintains fairly high efficiency.

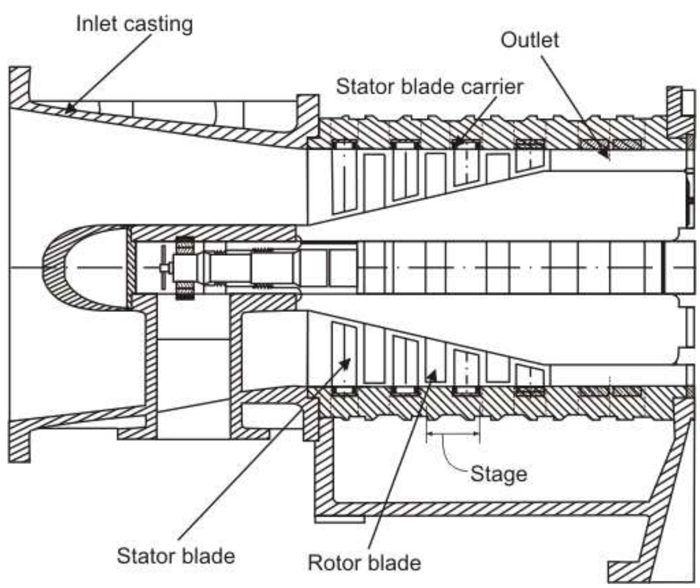


Fig. 7 Axial flow compressor

The basic principle of acceleration of the working fluid, followed by diffusion to convert acquired kinetic energy into a pressure rise, is applied in the axial compressor. The flow is considered as occurring in a tangential plane at the mean blade height where the blade peripheral velocity is U . This two dimensional approach means that in general the flow velocity will have two components, one axial and one peripheral denoted by subscript w , implying a whirl velocity. It is first assumed that the air approaches the rotor blades with an absolute velocity, V_1, at and angle $\alpha_1$to the axial direction. In combination with the peripheral velocity U of the blades, its relative velocity will be V_{r1} at and angle β_1 as shown in the upper velocity triangle (Fig. 9). After passing through the diverging passages formed between the rotor blades which do work on the air and increase its absolute velocity, the air will emerge with the relative velocity of V_{r2} at angle β_2 which is less than β_1. This turning of air towards the axial direction is, as previously mentioned, necessary to provide an increase in the effective flow area and is brought about by the camber of the blades. Since V_{r2} is less than V_{r1} due to diffusion, some pressure rise has been accomplished in the rotor. The velocity V_{r2} in combination with U gives the absolute velocity V_2 at the exit from the rotor at an angle α_2 to the axial direction. The air then passes through the passages formed by the stator blades where it is further diffused to velocity V_3 at an angle α_3 which in most designs equals to α_1 so that it is prepared for entry to next stage. Here again, the turning of the air towards the axial direction is brought about by the camber of the blades.

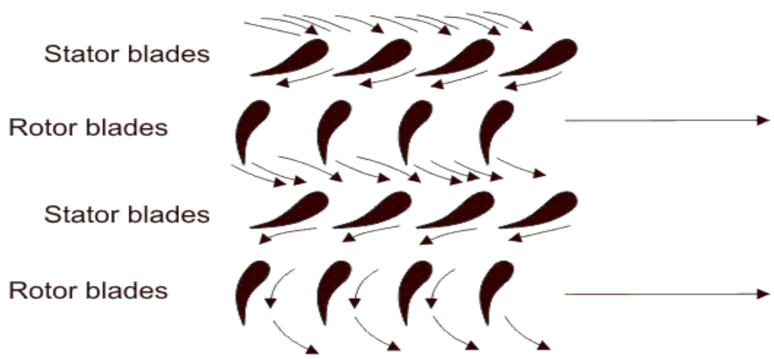


Fig. 8 Arrangement of vanes and flow directions

Velocity triangles

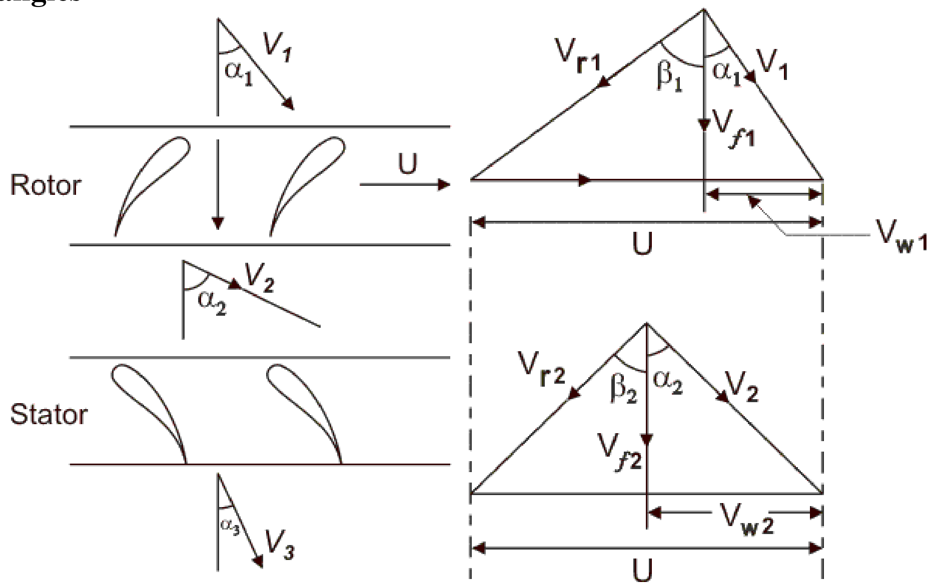


Fig. 9 Velocity triangles

Two basic equations follow immediately from the geometry of the velocity triangles. These are:

$$\frac{u}{V_f} = Tan\alpha_1 + Tan\beta_1$$ **-----------Eq 5.1**

$$\frac{u}{V_f} = Tan\alpha_2 + Tan\beta_2$$ **------Eq 5.2**

In which $V_f = V_{f1} = V_{f2}$ is the axial velocity, assumed constant through the stage. The work done per unit mass or specific work input, *w* being given by

$$W = u(V_{w2} - V_{w1})$$ **-------Eq 5.3**

This expression can be put in terms of the axial velocity and air angles to give

$$W = uV_f(\text{Tan}\,\alpha_2 - \text{Tan}\,\alpha_1)$$ **-------Eq 5.4**

or by using Eqs. 5.1 and 5.2

$$W = uV_f(\text{Tan}\,\beta_1 - \text{Tan}\,\beta_2)$$ **-------Eq 5.5**

This input energy will be absorbed usefully in raising the pressure and velocity of the air. A part of it will be spent in overcoming various frictional losses. Regardless of the losses, the input will reveal itself as a rise in the stagnation temperature of the air ΔT_o. If the absolute velocity of the air leaving the stage V_3 is made equal to that at the entry. V_1, the stagnation temperature rise ΔT_o will also be the static temperature rise of the stage, ΔT_s, so that

$$\Delta T_o = \Delta T_s = \frac{uV_f}{c_p}(\text{Tan}\,\beta_1 - \text{Tan}\,\beta_2)$$ **----- Eq 5.6**

In fact, the stage temperature rise will be less than that given in Eq. (5.6) owing to three dimensional effects in the compressor annulus. Experiments show that it is necessary to multiply the right hand side of Eq. (5.6) by a work-done factor λ which is a number less than unity. This is a measure of the ratio of actual work-absorbing capacity of the stage to its ideal

Assignment-Cum-Tutorial Questions

III B.Tech I Semester – 2019-20*Mechanical Engineering*

Turbomachinery

Unit –V & VI
Centrifugal and Axial Compressors

A) Objective Questions

- In a centrifugal compressor, assuming the same overall dimensions, blade inlet angle and rotational speeds, which of the following bladings will give the maximum pressure rise?
  (a) forward curved blade (b) backward curved blade
  (c) radial blades (d) all of the above [ ]
- In an RFC, the radial blades are employed [ ]
  (a) to avoid surging (b) to avoid stalling
  (c) to reduce cost (d) to increase efficiency
- In a centrifugal compressor terminology, the vaneless space refers to the space between [ ]
  (a) the inlet and blade inlet edge (b) blades in the impeller
  (c) diffuser exit and volute casing **(d)** impeller tip and the diffuser exit
- The slip in the RFCs is the reduction of [ ]
  (a) specific work (b) reaction (c) speed (d) utilization factor
- The effect of increase in the number of blades on the impeller in a centrifugal compressor is [ ]
  (a) increase in slip **(b)** decrease in slip
  (c) zero slip (d) maximum efficiency
- The purpose of inducer section in RFCs is [ ]
  (a) to have a high specific work (b) to increase the flow rate of fluid
  (c) to avoid surging **(d)** to have a smooth intake of flow
- The power input factor or work-done factor is [ ]
  (a) ideal work/actual work **(b)** actual work/theoretical work
  (c) isentropic work/actual work (d) ideal work/Euler work
- For the diffusion of a specified quantity of kinetic energy, the diffusion efficiency

is

(a) isentropic rise in static pressure/actual rise in static pressure

(b) isentropic rise in dynamic pressure/actual rise in dynamic pressure

(c) actual rise in static pressure/isentropic rise in static pressure

(d) actual rise in dynamic pressure/isentropic rise in dynamic pressure [ ]

- Diffusion efficiency depends on [ ]

(a) the rate of change of pressure (b) the rate of change of velocities

(c) the rate of change of temperature **(d)** the rate of change of areas

- AFCs have [ ]

(a) purely impulse stages (b) purely reaction stages

(c) impulse and reaction stages (d) alternate stages of impulse and reaction

- Pick the wrong statement. [ ]

(a) An AFC has larger frontal areas than an RFC.

(b) An AFC has higher pressure ratios than an RFC.

(c) An AFC has higher a weight-to-power ratio than an RFC.

(d) In an AFC, it is easier to have multi-staging than in an RFC.

- In a centrifugal air compressor the pressure developed depends on [ ]

(a) impeller tip velocity (b) inlet air temperature

(c) compression index **(d)** all of the above

- In a centrifugal air compressor the pressure ratio is increased by __________________.

(a) increasing the speed of the impeller keeping its diameter fixed

(b) increasing the diameter of the impeller keeping the speed constant

(c) reducing the inlet temperature, keeping the impeller diameter and speed fixed

(d) all of the above

- Surging basicaly implies [ ]

(a) unsteady, periodic and reversed flow

(b) forward motion of air at a speed above sonic velocity

(c) the surging action due to the blast of air produced in the compressor

(d) forward movement of aircraft

- Surging in an RFC is due to [ ]

  (a) the positive slope of the load line (b) the negative slope of the load line

  (c) the positive slope of the characteristic (d) the negative slope of the characteristic

- Stalling of the blades means [ ]

  (a) loss of drag (b) loss of lift

  (c) separation of flow **(d)** loss of performance

- Stalling of blades in an axial flow compressor is the phenomenon of [ ]

  (a) air stream blocking the passage (b) motion of air at sonic velocity

  (c) unsteady, periodic and reversed flow

  (d) air stream not able to follow blade contour

- Phenomenon of choking in compressor means

  (a) no flow of air

  (b) fixed mass flow rate regardless of pressure ratio

  (c) freducing mass flow rate with increase in pressure ratio

  (d) increased inclination of the chord with air stream

- **Assertion (A)**: The specific work input for an axial flow compressor is lower than that of the centrifugal compressor for the same pressure ratio.

  Reason (R): Isentropic efficiency of axial flow compressor is much higher than that of a centrifugal compressor. [ ]

  (a) Both A & R are individually true and R is the correct explanation of A.

  (b) Both A & R are individually true and R is not the correct explanation of A.

  (c) A is true but R is false (d) A is false but R is true

- The capacity of an air compressor is specified as 3 m^3/min. It means that the compressor is capable of [ ]

  (a) supplying 3 m^3 of compressed air per minute

  (b) compressing 3 m^3 of free air per minute

  (c) supplying 3 m^3 of compressed air per minute at NTP

  (d) compressing 3 m^3 of standard air per minute

- Centrifugal compressors are more suitable as compared to axial compressor for

  (a) high head, low flow rate (b) low head, low flow rate

  (c) low head, high flow rate (d) high head, high flow rate [ ]

- Consider the following statements:

  In centrifugal compressors, there is a tendency of increasing surge when

  1. the number of diffuser vanes is less than the number of impeller vanes
  2. the number of diffuser vanes is greater than the number of impeller vanes
  3. the number of diffuser vanes is equal to the number of impeller vanes
  4. mass flow is greatly in excess of that corresponding to the design mass flow

  Which of the these statements is / are correct? [ ]

  (a) 1 and 4 **(b)** 2 alone (c) 3 and 4 (d) 2 and 4

- For a multistage compressor, the polytropic efficiency is [ ]

  (a) the efficiency of all stages combined together

  (b) the efficiency of one stage

  (c) constant throughout all the stages

  (d) a direct consequence of the pressure ratio

B) Subjective Questions

- What is a slip factor and pressure coefficient?
- Describe briefly an axial flow compressor.
- What do you mean by surging and choking?
- Explain the construction and working of centrifugal compressor. Draw velocity triangles and obtain the expression for work done.
- Explain the construction and working of axial flow compressor. Draw velocity triangles and obtain the expression for work done.
- Compare (i) reciprocating and rotary air compressors (ii) reciprocating and centrifugal compressors (iii) centrifugal and axial flow compressors.
- Distinguish between the working principles of centrifugal, mixed and axial flow compressors.
- Derive the expression: $w = \frac{C_2^2 - C_1^2}{2} + \frac{C_{r1}^2 - C_{r2}^2}{2} + \frac{u_2^2 - u_1^2}{2}$ in the case of centrifugal compressor and explain its significance.

- A centrifugal air compressor delivers 18.2kg/s of air with a total head pressure ratio of 4 to 1. The speed of the compressor is 15000r.p.m. Inlet total head temperature is 15^0C, slip factor is 0.9, power input factor is 1.04 and 60% isentropic efficiency. Calculate the overall diameter of the impeller and power input.

- A centrifugal compressor running at 10000 r.p.m delivers 660 m^3/min of free air. The air compressed from 1 bar and 20^0C to a pressure ratio of 4 with an isentropic efficiency of 82%. Blades are radial at outlet of impeller and flow velocity of 62 m/s may be assumed through constant. The outer radius of impeller is twice the inner and the slip factor is assumed as 0.9. The area coefficient is 0.9 at the inlet. Calculate:
    - Final temperature of air
    - theoretical power
    - impeller diameters at inlet and outlet
    - breadth of impeller at inlet
    - impeller blade angle at inlet.
- A centrifugal compressor compresses 5m^3/s of air from 1 bar and 15^0C to 1.5bar. The index of compression is 1.5. The flow velocity at inlet and outlet of the machine is the same and equal to 70m/s. The inlet and outlet impeller diameters are 0.3m and 0.6m respectively. The blower rotates at 7000r.p.m.calculate
    - Blade angles at inlet and outlet of the impeller
    - The absolute angle at the tip of the impeller
    - The breadth of blade at inlet and outlet.
    - No diffuser employed and the whole pressure rise takes place in the impeller and the blades have negligible thickness.

- Air at a temperature of 300K flows in a centrifugal compressor running at 18000r.p.m. The other data is as follows.
    - Isentropic total head efficiency=0.76
    - Outer diameter of blade tip=550mm
    - Slip factor=0.82
    - Calculate: (i) The temperature rise of air passing through the compressor

      (ii) The static pressure ratio. Assume absolute velocities at inlet and outlet are same. Take Cp of air=1.005kJ/kg.K

- A single stage single sided centrifugal compressor delivers 18 kg/sec of air, with a pressure ratio 4 when running at 15000 r.p.m. The inlet conditions are 1 bar and

200C. The slip factor is 0.9. The power input factor is 1.04 and the isentropic efficiency of compressor is 75%. Calculate the (i) the final temperature of air (ii) power required (iii) the tip diameter of the impeller (iv) the blade angles at impeller eye which has root and tip diameter 15 cm and 25 cm respectively.

- The following data pertains to a centrifugal compressor:

Total pressure ratio	:	3.6:1
Diameter of inlet eye of impeller	:	35 cm
Axial velocity at inlet	:	140 m/s
Mass flow	:	12 kg/s
The velocity on the delivery duct	:	120 m/s
The tip speed of impeller	:	460 m/s
Speed of impeller	:	16000 rpm
Total head isentropic efficiency	:	80%
Pressure coefficient	:	0.73
Ambient conditions	:	1.013 bar and 15^0C

  Calculate: (i) The static pressure and temperature at inlet and outlet of compressor
  (ii) the static pressure ratio
  (iii) Work of compressor per kg of air
  (iv) The theoretical power required.

- Following particulars relate to a centrifugal compressor:
  Inlet diameter of impeller = 61.4 cm
  Outlet diameter of impeller = 123 cm, pressure ratio = 1.33
  Speed = 5000 r.p.m, velocity of flow = 61.6 m/sec, free air delivered = 1000 m3/min
  Index of compression = 1.6. Assuming that all the pressure rise takes place in the impeller, find the angles at which the air from impeller enters the casing, breadth of the impeller blade inlet and outlet.

- The following data relate to an axial flow compressor:

Blade velocity	:	240 m/s
Flow velocity	:	190 m/s
Inlet and exit angles of the air	:	45^0 and 14^0 respectively

  Calculate: (i) the pressure rise (ii) the work done per kg of air.

- An axial flow compressor having eight stages and with 50% reaction design compresses air in the pressure ratio of 4:1. The air enters the compressor at 20^0C and flows through it with a constant speed of 90 m/sec. The rotating blades of compressor rotate with a mean speed of 180 m/sec. Isentropic efficiency of the compressor may be taken as 82%. Calculate: (i) Work done by the machine (ii) Blade angles.

- An axial flow compressor compresses the air from an inlet condition of 1 bar and 290 K to a delivery pressure of 5 bar with an overall isentropic efficiency of 87%. The degree of reaction is 0.5 and the blade angles at inlet and outlet are 44^0 and 13^0 respectively. The mean blade speed and axial velocity are constant throughout the compressor. Assuming a blade velocity of 180 m/sec, and work done factor

0.85, calculate the number of stages.

- In an eight stage axial flow compressor, the overall stagnation pressure ratio achieved is 5:1 with an over all isentropic efficiency of 90%. The temperature and pressure at inlet are 20^0C and 1bar. The work is divided equally between the stages. The mean blade speed is 175m/s and 50% reaction design is used. The axial velocity through the compressor is constant and equal to 100m/s, calculate the power required and blade angles.
- A multistage axial flow compressor delivers 20 kg/s of air. The inlet stagnation condition is 1 bar and 17^0C. The power consumed by the compressor is 4.35 MW. Calculate:
  (i) Delivery pressure
  (ii) Number of stages
  (iii) Overall isentropic efficiency of the compressor. Assume temperature rise in the first stage is 15^0C, the polytropic efficiency of compression is 88% and the stage stagnation pressure ratio is constant

C) GATE/IES Questions

- Blower delivers gaseous fluids at pressure ratios ____________ (above/below) 1.15 and __________ (have/ have no)artificial cooling arrangement. [GATE-1992]

- For reversible adiabatic compression in a steady flow process, the work transfer per unit mass is [ ]
  [GATE-1996]

  (a) $\int pdv$ (b) $\int vdp$ (c) $\int Tds$ (d) $\int sdT$

- The specific speed of a centrifugal compressor is generally [ ]
  [GATE-1997]

  (a) higher than that of an axial compressor

  (b) less than that of a reciprocating compressor

  (c) independent of the type of compressor, but depends only on the size of the compressor

  (d) more than the specific speed of the reciprocating compressor but less that of the axial compressor

- Air enters a compressor at a temperature of 27^0C. The compressor pressure ratio is 4. Assuming an efficiency of 80%, the compressor work required in kJ/kg is [

]

(a) 160 (b) 172 (c) 182 (d) 225 [GATE-1998]

- A gas contained in a cylinder is compressed, the work required for compression being 5000 J. During the process, heat interaction of 200 kJ causes the surroundings to be heated. The change in internal energy of the gas during the process is [ ]

(a) -7000 kJ (b) -3000 kJ (c) 3000 kJ (d) 7000 kJ [GATE-2004]

- In a steady state steady flow process taking place in a device with a single inlet and a single outlet, the work done per unit mass flow rate is given by $w = -\int_{inlet}^{outlet} vdp$, where v - is the specific volume and p is pressure. The expression for w given above [ ]

(a) is valid only if the process is both reversible and adiabatic

(b) is valid only if the process is both reversible and isothermal

(c) is valid for any reversible process

(d) is incorrect; it must be $w = \int_{inlet}^{outlet} pdv$

[GATE-2008]

- A compressor undergoes a steady reversible flow process. The gas at inlet and outlet of the compressor is designated as state 1 and state 2 respectively. Potential and kinetic energy changes are to be ignored. The following notations are used: v = specific volume and p - pressure of the gas. The specific work required to be supplied to the compressor for this gas compression process is [ ][GATE-2009]

(a) $\int_1^2 pdv$ (b) $\int_1^2 vdp$ (c) $v_1(p_2 - p_1)$ (d)

$$-p_2\left(v_1 - v_2\right)$$