Unit – I: Introduction to Turbo Machinery

Introduction to Turbo Machinery

Turbo machines (Hydraulic & Thermal), Classification of Turbo machines, Comparison with positive displacement machines, Fundamental equation governing turbo machines, Different losses associated with turbo-machinery, Applications of Turbo machines.

Impact of Jet

Impulse momentum principle and its applications, Force exerted on fixed and moving flat plate, hinged plate, curved vanes, series of flat plates and radial vanes, velocity triangles and their analysis, work done equations, vane efficiency.

Fluid Machines

Fluid machines are those devices that are used to either move fluid or extract energy from it.

Broadly speaking, fluid machines are divided into two groups:

1-Positive-displacement machines

2-Turbomachines

Positive-displacement machines are those devices that **force fluid into confined volumes**.

Examples - human heart, reciprocating pumps and compressors,

Turbomachines

Turbomachines are defined as all 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.

Example, ceiling fans, centrifugal pump

Introduction to Turbomachinery

A turbo machine is basically a rotating machine.

The rotating wheel is called a rotor /runner / impeller

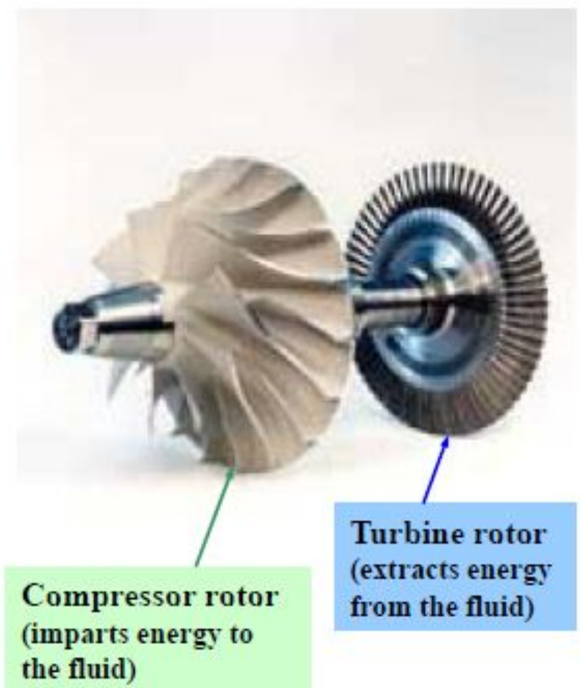
The rotor will be immersed in a fluid continuum

The fluid medium can be gas / steam /water / air

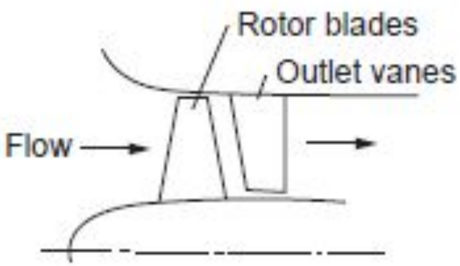
Energy transfer takes place either **Turbine rotor or fluid**

from rotor to fluid, *or*

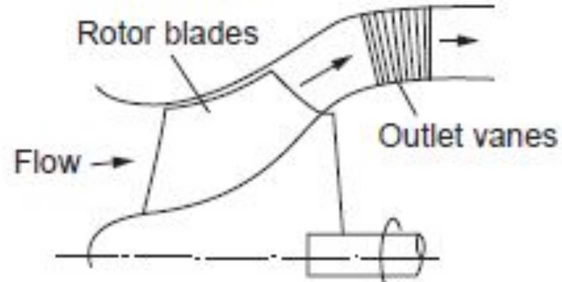
from fluid to rotor



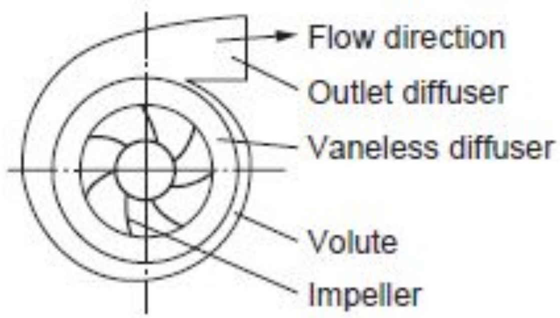
Examples of Turbo Machines



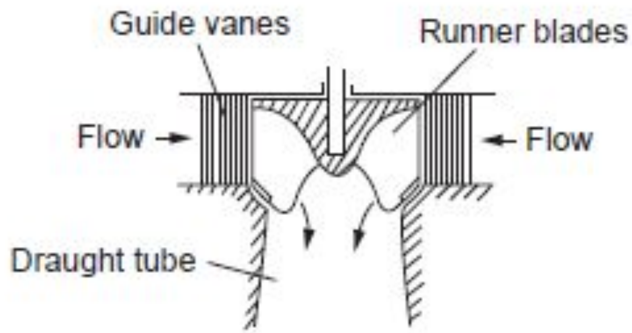
(a) Single stage axial flow compressor or pump



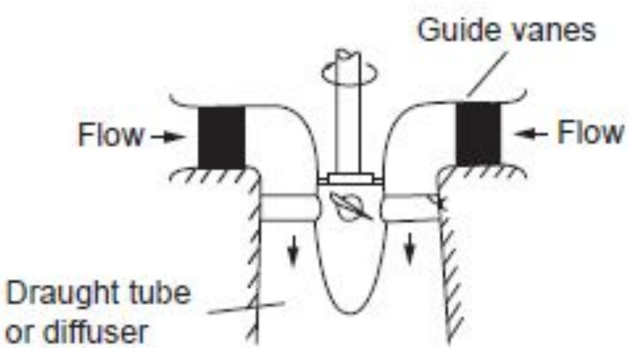
(b) Mixed flow pump



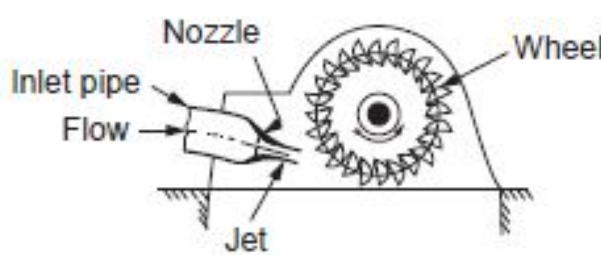
(c) Centrifugal compressor or pump



(d) Francis turbine (mixed flow type)



(e) Kaplan turbine



(f) Pelton wheel

Turbo machine - Definition

A turbo machine is a device where mechanical energy in the form of shaft work, is transferred either *to or from a continuously flowing* fluid by the dynamic action of rotating blade rows.

The interaction between the fluid and the turbo machine blades also results in fluid dynamic lift.

A turbo machine produces change in enthalpy of the fluid passing through it.

The word turbo is a Latin origin and implies that which spins or whirls around.

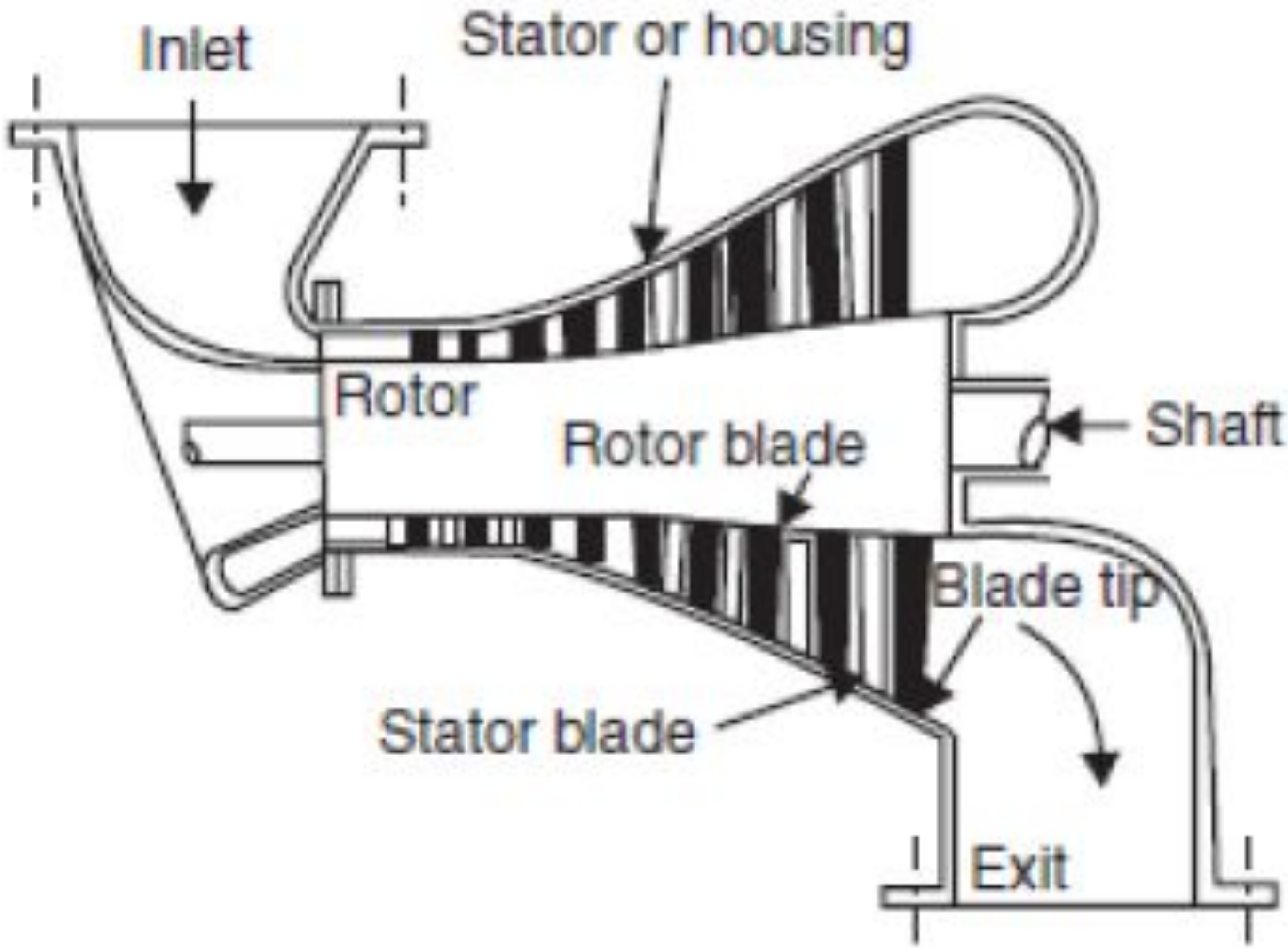
Parts of a turbo machine

The principle components of a turbo machine are:

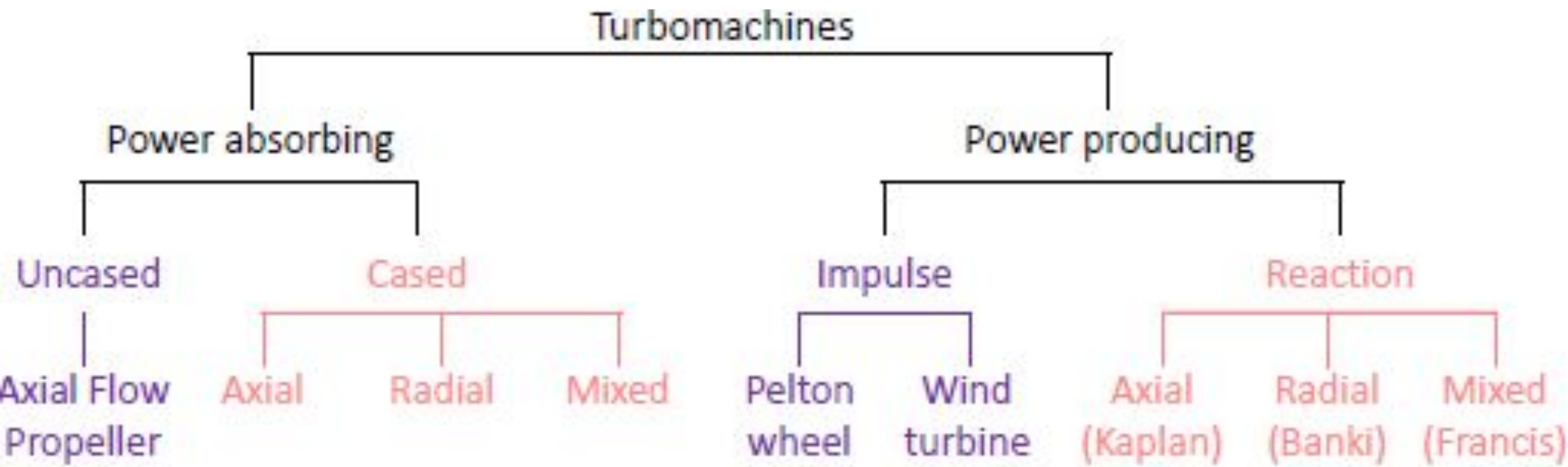
1. Rotating element (vane, impeller or blades) – operating in a stream of fluid.
2. Stationary elements – which usually guide the fluid in proper direction for efficient energy conversion process.
3. Shaft – This either gives input power or takes output power from fluid under dynamic conditions and runs at required speed.
4. Housing – to keep various rotating, stationery and other passages safely under dynamic conditions of the flowing fluid.

E.g. Steam turbine parts and Pelton turbine parts.

Parts of a turbo machine



Turbo machine - Classifications



CLASSIFICATION OF TURBO MACHINES

1. Based on energy transfer

a) Energy is given by fluid to the rotor - Power generating turbo machine E.g. Turbines

b) Energy given by the rotor to the fluid – Power absorbing turbo machine

c) E.g. Pumps, blowers and compressors

2. Based on fluid flowing in turbo machine

a) Water

b) Air

c) Steam

d) Hot gases

e) Liquids like petrol etc.

3. Based on direction of flow through the impeller or vanes or blades, withreference to the axis of shaft rotation

a) Axial flow – Axial pump, compressor or turbine

b) Mixed flow – Mixed flow pump, Francis turbine

c) Radial flow – Centrifugal pump or compressor

d) Tangential flow – Pelton water turbine

4. Based on condition of fluid in turbo machine

a) Impulse type (constant pressure) E.g Pelton water turbine

b) Reaction type (variable pressure) E.g. Francis reaction turbine

5. Based on position of rotating shaft

a) Horizontal shaft – Steam turbines

b) Vertical shaft – Kaplan water turbines

c) Inclined shaft – Modern bulb micro hydel turbines

Comparison of turbo machines with positive displacement machines

Turbo machines	Positive displacement machines
• It creates Thermodynamic & Dynamic action b/w rotating element & flowing fluid, energy transfer takes place if pressure and momentum changes	It creates Thermodynamic &Mechanical action b/w moving member&static fluid, energy transfer takes place with displacement of fluid
• It involves a steady flow of fluid & rotating motion of mechanical element	It involves a unsteady flow of fluid & reciprocating motion
• They operate at high rotational speed	They operate at low speed
• Change of phase during fluid flow causes serious problems in turbomachine	Change of phase during fluid flow causes less problems in Positive displacement machines
• Efficiency is usually less	Efficiency is higher
• It is simple in design	It is complex in design
• Due to rotary motion vibration problems are less	Due to reciprocating motion vibration problems are more
E.g. Hydraulic turbines, Gas turbines, Steam turbines etc.	E.g. I.C engines, Reciprocating air compressor, pumps etc.

Applications of Turbo Machines

Power Generation

- Hydro electric-
- Steam turbines-.
- Gas turbines-
- Wind mills-

Power Consumption

- Marine
- Steam turbine -.
- Gas turbines-
- Turbochargers-
- Superchargers

General

- Pumps-
- Air compressors-
- Fans-

Losses of Turbo machines

- ***Internal Losses***:
- ***External losses:***
- Hydraulic Loss (Zh)
- Disc Friction Loss (Zr)
- Return-Flow Loss (Za)
- Leakage Loss ΔV
- Impeller power loss
- Leakage power loss
- Casing power loss
- Leakage losses
- Mechanical losses:
- Generator losses

Fundamental equation governing turbo machines

Basic Physical laws of Fluid Mechanics and Thermodynamics used in Turbo machines are:

The *continuity of flow equation*

The *first law of thermodynamics and the steady flow energy equation*

The *momentum equation*

The *second law of thermodynamics*

Newton's Second Law of Motion

Unit – I: Introduction to Turbo Machinery

- Introduction and Impact of Jet

  Introduction to Turbomachines (Hydraulic & Thermal), Classification of Turbo machines, Applications of Turbomachines. Impulse momentum principle and its application to fixed and moving flat, inclined, and curved plate/vanes. Velocity triangles and their analysis, work done equations, vane efficiency (No numerical)

- Hydraulic Turbines

  Introduction to Hydro power plant, Classification of Hydraulic Turbines, Concept of Impulse and Reaction Turbines. Construction, Principle of Working, design aspects, velocity diagrams and its analysis of Pelton wheel, Francis, and Kaplan turbines, Degree of reaction, Draft tube: types and efficiencies, governing of hydraulic turbines, Cavitation in turbines.

Unit – I:Impact of Jet and Hydraulic Turbines

Introduction and Impact of Jet: Introduction to Turbomachines (Hydraulic & Thermal), Classification of Turbo machines, Applications of Turbomachines.

Impulse momentum principle and its application to fixed and moving flat, inclined, and curved plate/vanes. Velocity triangles and their analysis, work done equations, vane efficiency (No numerical)

Dynamic force or Dynamic action of fluid

Consider a stream of fluid entering a machine such as a hydraulic turbine or steam turbine, or a pump or a fan.

The stream of fluid has a more or less defined direction. For a force to be exerted by the fluid on the machine, the stream of fluid must undergo a change in its velocity either in its magnitude or direction or both.

When the fluid stream enters the machine, the machine exerts a force on the fluid bringing about a change in the velocity of fluid either in its magnitude or in its direction.

According to Newton's third law of motion for every action there is an equal and opposite reaction. Hence, the fluid stream exerts an equal and opposite force upon the machine that causes the change in velocity of fluid stream. This force exerted by the virtue of fluid in motion is called the **dynamic force of fluid.**

The dynamic force of fluid always involves a change in its velocity and thus a change in its momentum.

Hence, the force exerted by the machine on the fluid is the action and the force, in turn, exerted by the fluid on the machine is there reaction.

Momentum & Impulsive force

Momentum:

The capacity of a moving body to impart motion to other bodies is called momentum.

The momentum of a moving body is given by the product of mass and velocity of the moving body.

Momentum = Mass x Velocity

= m x V **Unit: kg m/s**

Impulsive Force and Impulse of Force:

A force acting over a short interval of time on a body is called impulsive force.

Eg: Kick given to a foot ball.

Impulse of a force is given by the product of magnitude of force and its time of action.

Impulse of a force = Force x Time interval **SI unit: Ns**

Newton's Second Law

The Rate of change of momentum of a body is equal to the resultant force acting on the body, and takes place in the direction of the force.

$$F \;\alpha\; \frac{(mV - mU)}{t}$$

$$F \;\alpha\; ma$$

a – acceleration

$$F = k\,ma$$

If m = 1 and a = 1 then F = 1

$$\therefore k = 1$$

$$\therefore \mathbf{F = ma}$$

SI unit of force: newton (N)

Impulse – Momentum Principle

- **Impulse – Momentum Principle:**

  From Newton's II Law

$$F = ma$$

$$F = m\,\frac{(V - U)}{t}$$

$$Ft = mV - mU$$

Impulse = Final momentum – Initial momentum

$\therefore$ Impulse of a force is given by the change in momentum caused by the force on the body.

Ft = m x Final velocity – m x Initial velocity

$$F_x\,t = m\,(V_x - U_x)$$

$$F_y\,t = m\,(V_y - U_y)$$

- *According to Impulse-momentum principle: "State of rest or of uniform motion of a body changes in the direction of an externally applied force, and that the magnitude of the force equals the rate of change of momentum".*

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

For constant mass flow rate, *dm*=0. The change in momentum may occur due to a change in the magnitude of velocity or in its direction or due to both.

For a constant mass flow rate, the momentum equation can be rewritten as,

$$\boldsymbol{F} = m\frac{d\boldsymbol{v}}{dt}$$

Dynamic force applied in x- direction = rate of change in momentum in x- direction,

i.e. $\boldsymbol{F}_x = m\frac{d\boldsymbol{v}_x}{dt}$; this is known as *linear momentum equation* in x direction.

$$\boldsymbol{F} = \frac{m}{t}(v_2 - v_1)$$

$$\boldsymbol{F} \times t = m(v_2 - v_1)$$

Here, the left hand side term is the product of force and time, (**F** X t) is called the *Impulse* of the applied force. Whereas, the right hand side term is the resulting *change in momentum*. The above equation can be rewritten as,

$$F = \dot{m}(v_2 - v_1) = \rho Q(v_2 - v_1)$$

The above equation represents, the force exerted by the body on the fluid.

- Newton's third law of motion, action and reaction are equal and opposite and therefore, *the force exerted by the fluid on the body is,*

$$F = -\rho Q(v_2 - v_1)$$

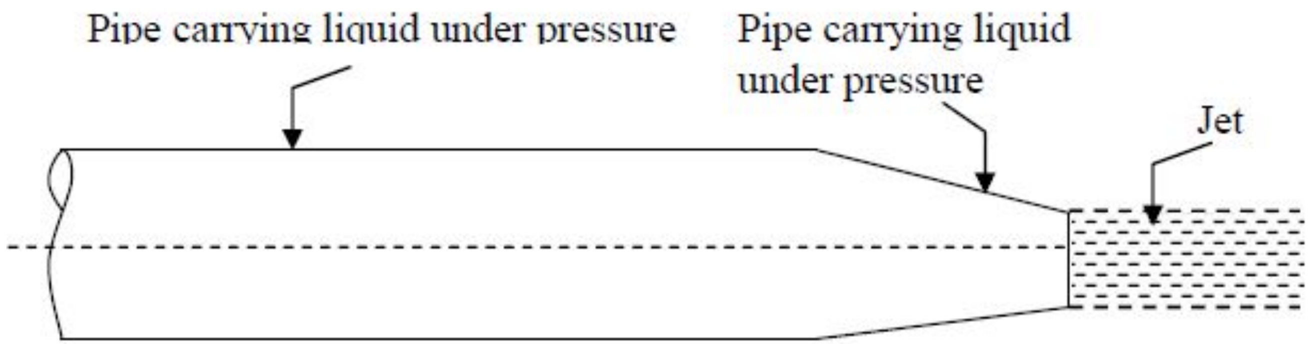
$$F = \rho Q(v_1 - v_2)$$

v_1 = initial velocity; and v_2= final velocity

Impact of Jet

A jet of fluid or liquid emerging out from a nozzle has some velocity and hence it possesses some kinetic energy.

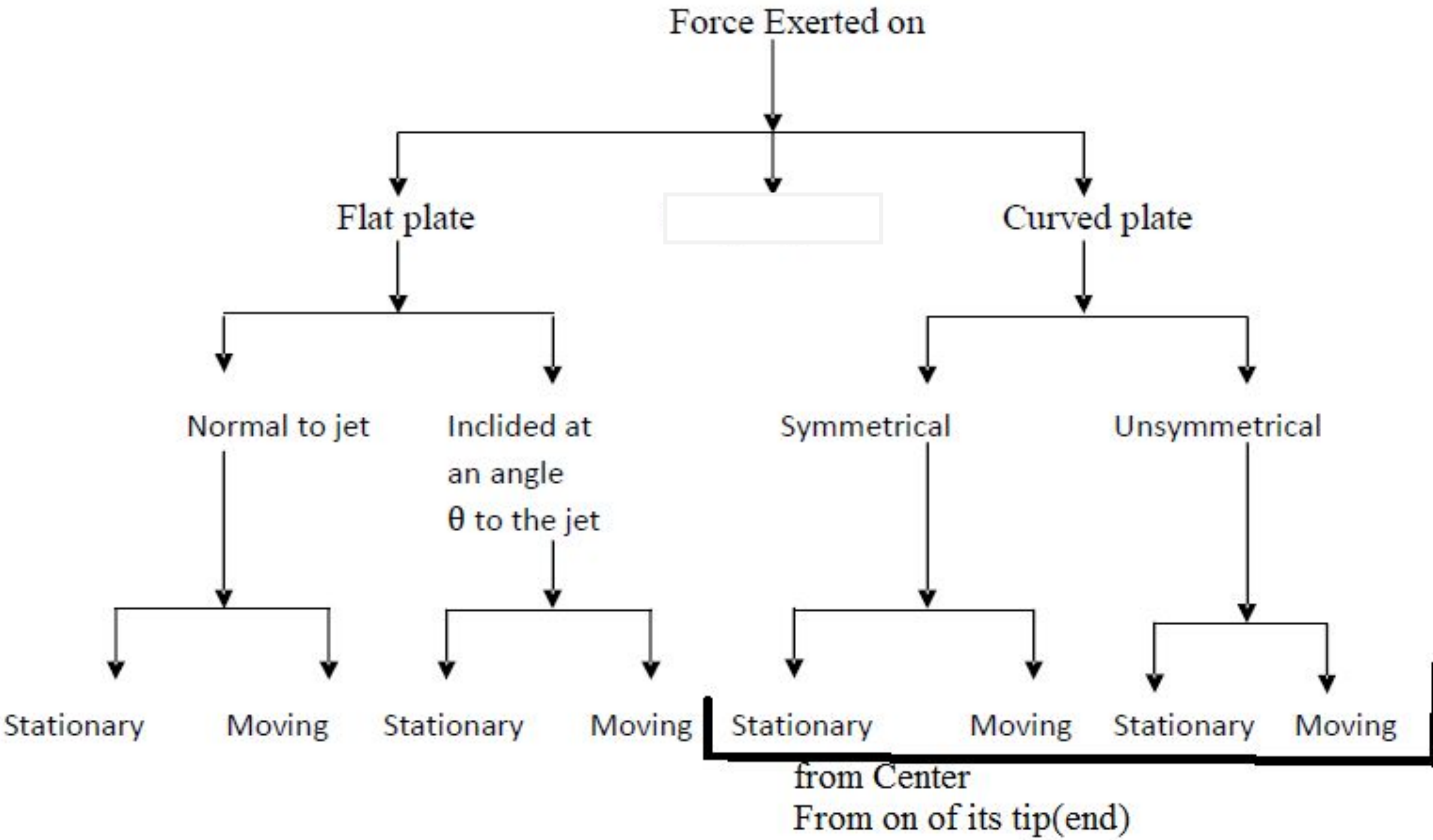
If this jet strikes a plate (either fixed or moving) , it will exert force on the plate. This impressed force is known as impact of jet and is referred to as Hydrodynamic force



Vane

Vanes or blades are plates of definite geometrical shape mounted on the periphery of rotor of a turbo machine (Pump / Turbine) in order to transfer energy from rotor to fluid or fluid to rotor.

Force exerted by jet on plate



Force exerted by jet on plate

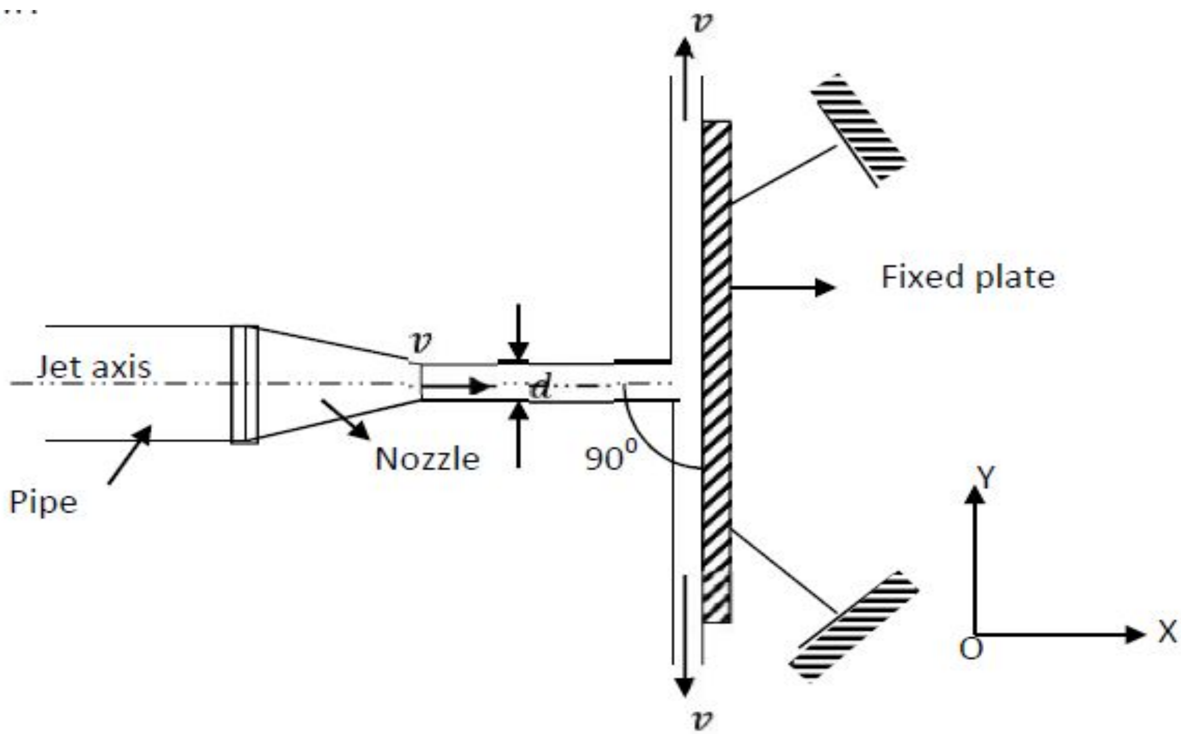
1. **Force exerted by the jet on a stationary plate**
   a) Plate is vertical to the jet
   b) Plate is inclined to the jet
   c) Plate is curved
2. **Force exerted by the jet on a moving plate**
   a) Plate is vertical to the jet
   b) Plate is inclined to the jet
   c) Plate is curved

Application of the Momentum Equation

- Impact of a jet on a plane surface
- Force due to flow round a curved vane
- Force due to the flow of fluid round a pipe bend.
- Reaction of a jet.

Force exerted on stationary plate held normal to jet.

Consider a jet of water strike **normally on the fixed plate held perpendicular to flow direction of jet(water)** as shown in figure Jet after striking the plate will deflected through 90°.so final velocity of fluid in the **direction of the jet after** striking plate will be **zero in the horizontal direction .**



$V1 = V$

$V2= \ 0$

Let,

V=velocity of jet(velocity of water jet before striking)

V_1 =velocity of water jet before striking to the plate = **V**

V_2 = velocity of water jet after striking to the plate = **0**

d=diameter of jet

ρ=density of fluid (water)

A=cross section area of jet= $\frac{\pi}{4}d^2$

$\dot{m}$=mass flow rate of fluid, kg/s=**ρAV**

We know that,

Impact of **jet(force exerted by jet on stationary flat plate)**

=Rate change of momentum in the direction of force (x dir)

$$= \frac{\text{Initial moment} - \text{Final moment}}{\text{time}}$$

$$= \frac{(\text{Mass} \times \text{Initial velocity})-(\text{Mass} \times \text{Final velocity})}{\text{time}}$$

=Mass/time[Velocity of jet before striking – velocity of jet after striking]

$$F = mass\ flow\,rate[V_1 - V_2]$$

$$F = \dot{m}[V - 0]$$

$$F = \rho A V[V]$$

$F = \rho A V^2$ **Force exerted by jet on stationary plate**

Work done = Force exerted by jet on stationary flat plate * distance or displacement of plate

Or

Work done per second =

= Force * distance travelled per second by plate

= Force * velocity of plate

Work done per second = $\rho A V^2 * 0$

Since plate is fixed (stationary) distance travelled = 0

Work done per second= 0

Efficiency = output/input

= Work done/ KE = 0

Force exerted on moving plate held normal to jet.

Consider, a jet of water strikes the **flat moving plate** with a uniform velocity **'u'** or U away from the jet. & jet is moving with velocity **'V'** before striking **(V>u)**

Jet will move in **vertical direction after striking** the plate will deflected through **90°**.So final velocity of fluid in the direction of the jet after striking to the plate **will be V- U in the horizontal direction**

Relative velocity with which jet is going to strike plate will be

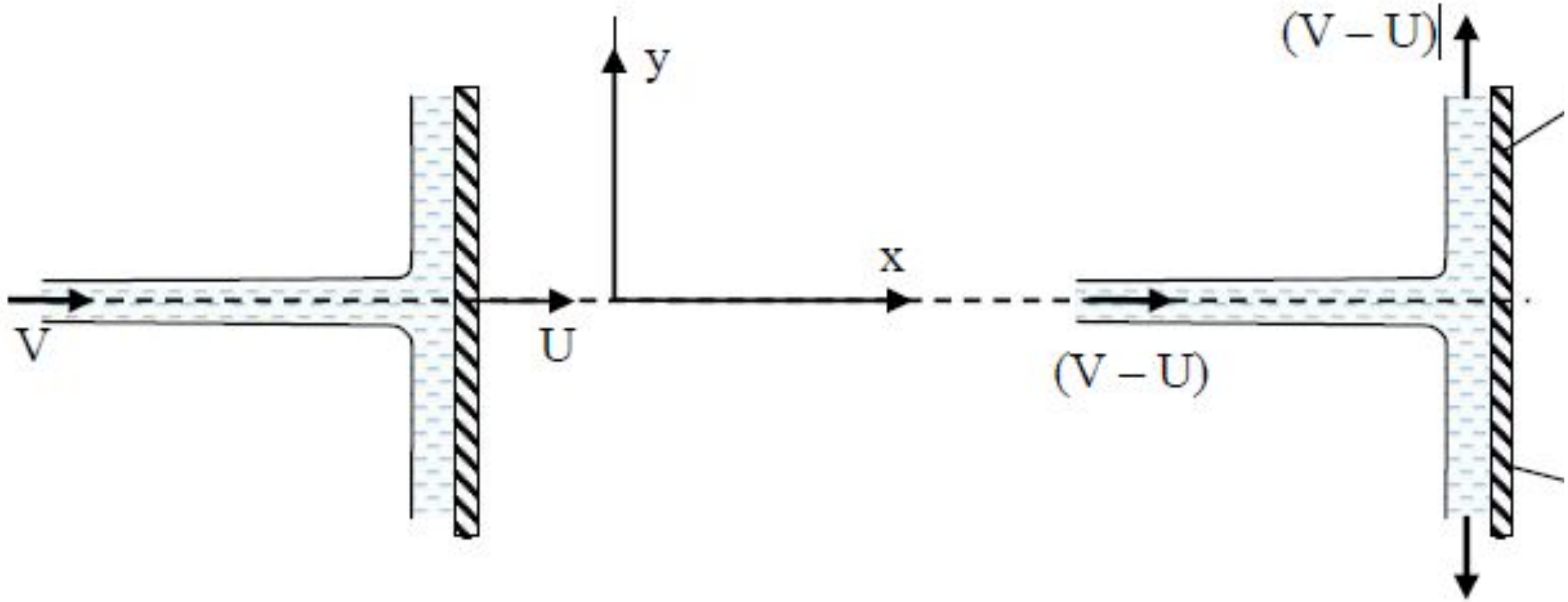
Relative velocity = velocity of jet- velocity of plate

= V - U or u

Relative velocity of jet w. r. t plate = V – u

Mass of water striking/ sec on the plate = ρ a (V - u)

Force exerted on moving plate held normal to jet.



V1 = (V –u) or V-U
V2= 0

Let,

u or U = Velocity of **plate**

V=velocity of **jet**(velocity of water jet before striking)

V_1 =velocity of water jet before striking to the plate in the horizontal direction = **V-u**

V_2 = velocity of water jet after striking to the plate in the horizontal direction = **0**

d=diameter of jet

ρ=density of fluid (water)

$\dot{m}$=mass flow rate of fluid, kg/s= ρAV

A=cross section area of jet = $\frac{\pi}{4}d^2$

it this case $\dot{m} = \rho a(V - u)$

∴ Mass of water striking the plate per second

=ρ A × velocity with which jet strikes the plate

= ρA(V-u)

Now, force exerted by jet(impact of water jet)**on moving flat plate** in the direction of jet,(in horizontal direction i. e. in x direction)

F= (Mass of jet strikes/sec) ×[Initial relative velocity of jet- Final relative velocity of jet in the direction of jet]

$$F = \rho A\left(V - u\right) \times \left[\left(V - u\right) - 0\right]$$

$$\therefore F = \rho A\left(V - u\right)^2$$

Force exerted by jet on moving flat plate normal to jet (moving away from jet)

$\therefore$ **Work done by jet on the plate(moving away from jet)**

W= Force × $\dfrac{\text{distance travelled by the plate}}{\text{time}}$

= **F× velocity of plate in direction of jet**

$$\therefore W = F * u \qquad \therefore W = \rho A (V-u)^2 u$$

Work done by jet on the plate (moving towards jet)

If plate is moving towards the jet,

$$F = \rho A[V-(-u)]^2 = \rho A (V+u)^2$$

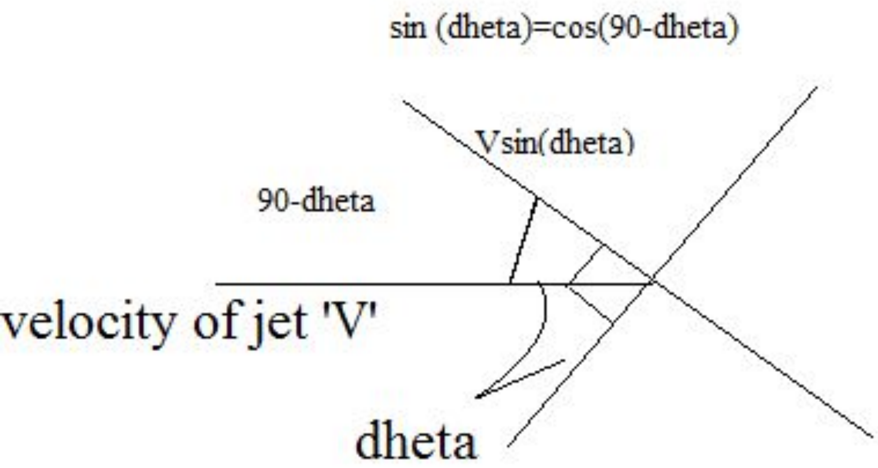
and

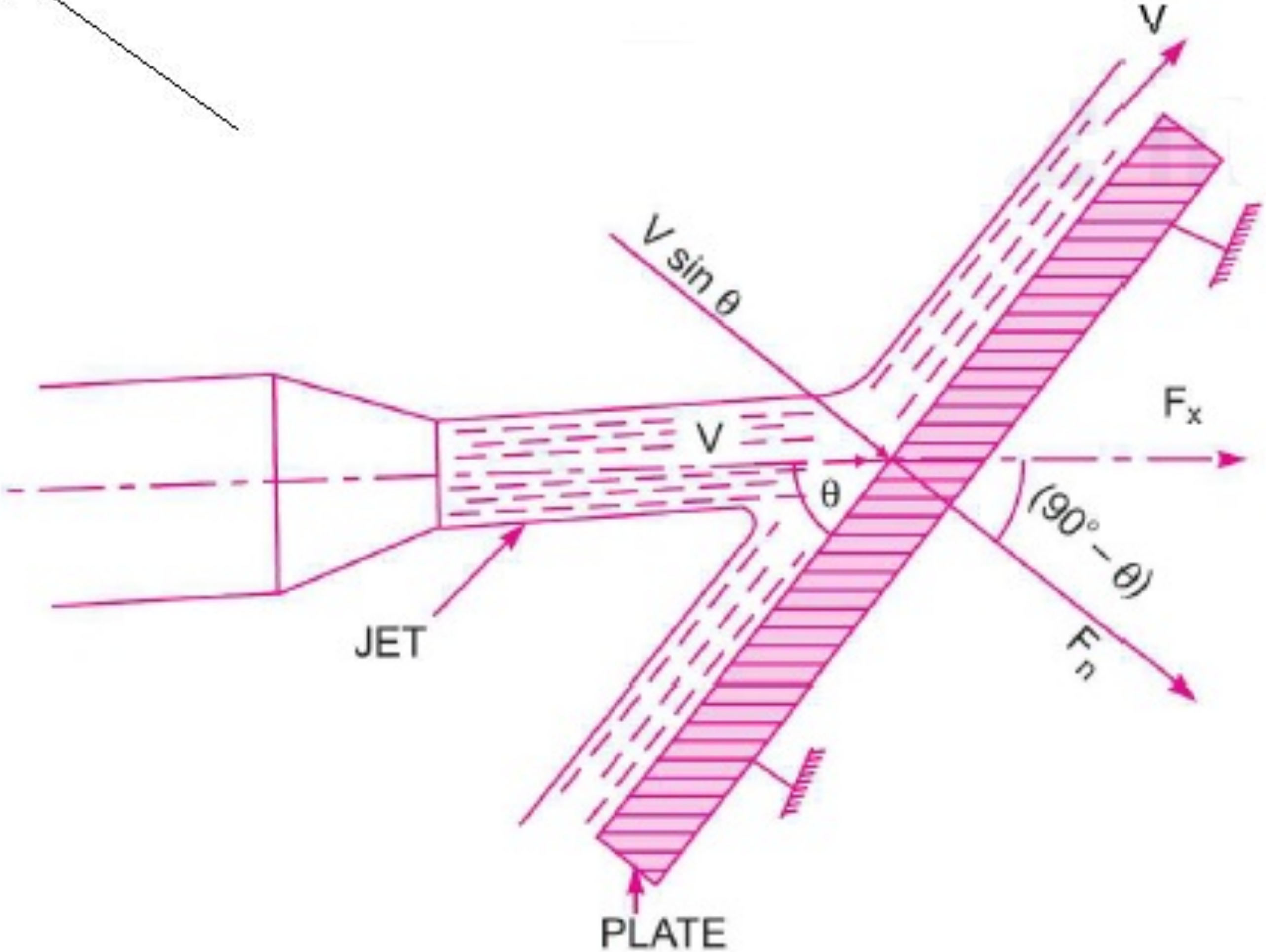
$$\therefore W = F * u \qquad \therefore W = \rho A[V+u]^2 u = \rho A u (V+u)^2$$

Work done by jet on the plate(moving towards jet)

$$\therefore W = \rho A u (V+u)^2$$

Force exerted by jet on **stationary flat inclined** plate



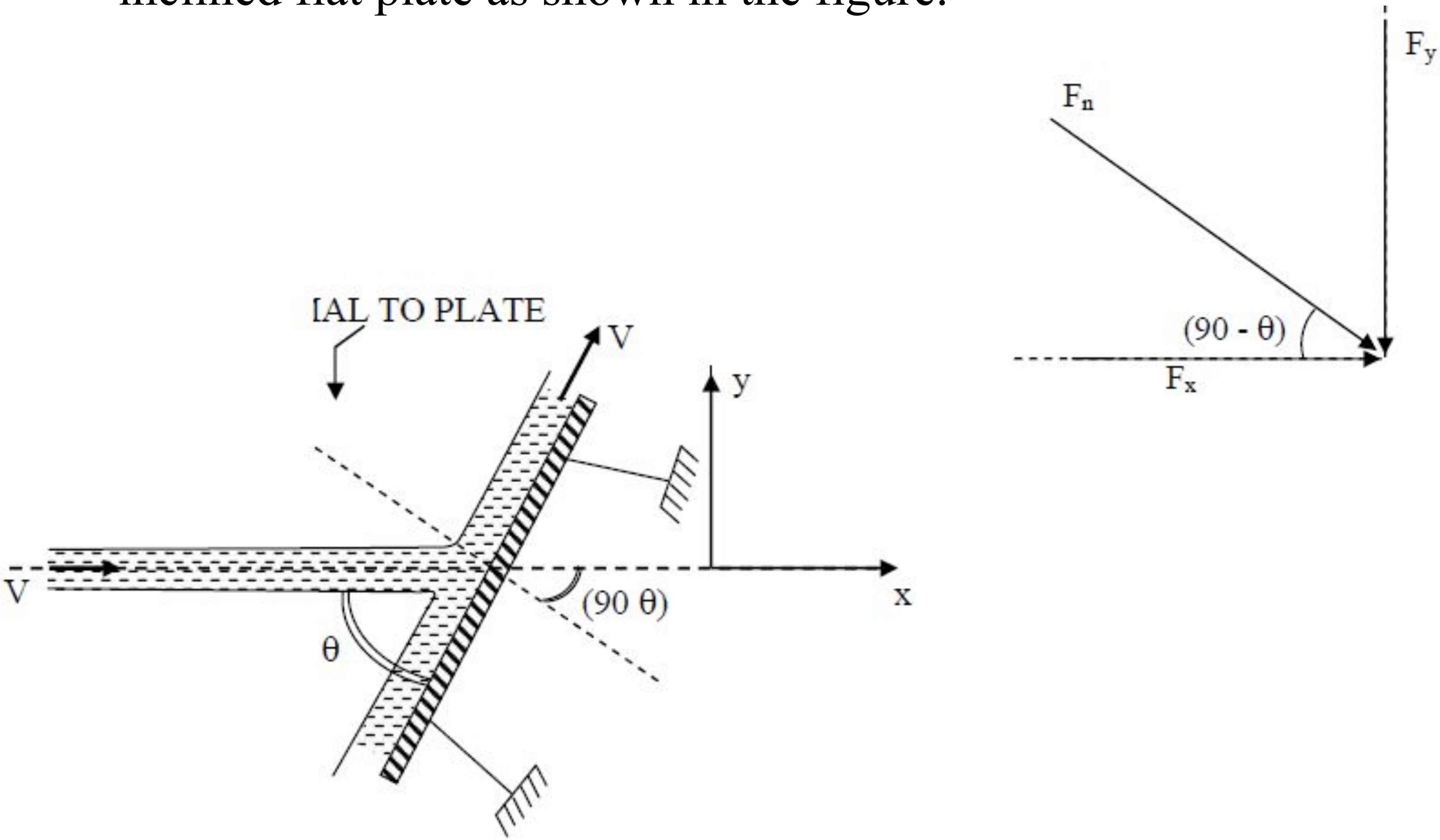


$V1 = V \sin\theta$

$V2 = 0$

Force exerted by jet on **stationary flat inclined** plate

Let a jet of water, coming out from the nozzle; strike an stationary inclined flat plate as shown in the figure.



Let,

θ=inclination of plate with the jet

= angle between jet and plate

V=velocity of jet(velocity of water jet before striking)

V_1 =velocity of water jet before striking to the plate in the horizontal direction = **V Sin θ**

V2 = velocity of water jet after striking to the plate in the horizontal direction = **0**

d=diameter of jet

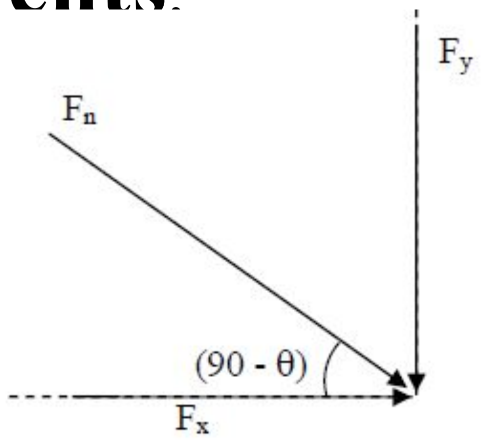
ρ=density of fluid (water)

A=cross section area of jet $= \frac{\pi}{4} d^2$

$\dot{m}$ =mass flow rate of fluid, kg/s

F_n =force exerted by the jet on the plate in the **direction normal to the plate**

If the force can be resolved into **two components**.
one in the **direction of the jet** &
second in **perpendicular to the direction of the flow**.
Then we have,



F_x = force exerted by the jet on the plate in the direction to the jet

F_y = force exerted by the jet on the plate in the direction perpendicular to direction of jet

Fn= Mass of the jet striking per second ×

[initial velocity of the jet before striking in the direction of F_n - final velocity of the jet after striking in the direction of F_n]

F_n= Mass per second ×[initial velocity - final velocity]

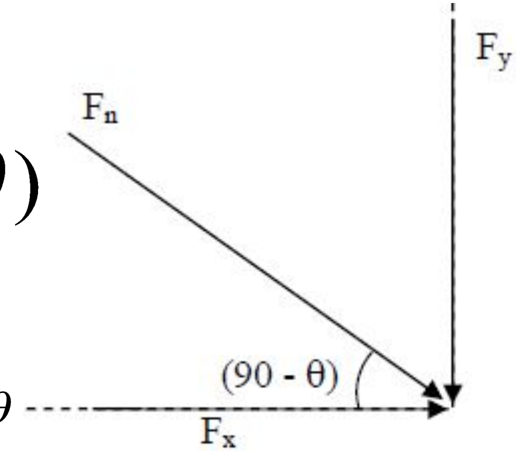
$$F = m^{\cdot} \times (V_1 - V_2)$$

$$F_n = (\rho A V) \times [V \sin\theta - 0]$$

$$\therefore F_n = \rho A V^2 \sin\theta$$

F_n =force exerted by the jet on the plate in the **direction normal to the plate**

$$F_n = \rho A V^2 \sin\theta$$



$$F_x = F_n \cos(90-\theta) \ \& \ F_y = F_n Sin(90-\theta)$$

$$\cos(90-\theta) = Sin\theta$$

$$\sin(90-\theta) = \cos\theta$$

$$F_x = F_n \sin\theta$$

$$F_y = F_n \cos\theta$$

$$\therefore F_x = \left(\rho A V^2 \sin\theta\right) \times \sin(\theta)$$

$$F_x = \rho A V^2 \sin^2\theta$$

$$F_y = \left(\rho A V^2 \sin\theta\right) \times \cos\theta$$

If θ is 90 degree then plate becomes vertical flat fixed plate then sin 90 =1 & Cos 90 =0 then equation of Fx and Fy can be written as

$$F_x = \rho A V^2 \qquad F_y = 0$$

Work done by jet on stationary inclined plate

=Force exerted by jet on stationary inclined flat plate * distance or displacement of plate

Or

Work done per second =

= Force * distance travelled per second by plate

= Force * velocity of plate

Work done per second $= \rho A V^2 \sin^2\theta * 0$

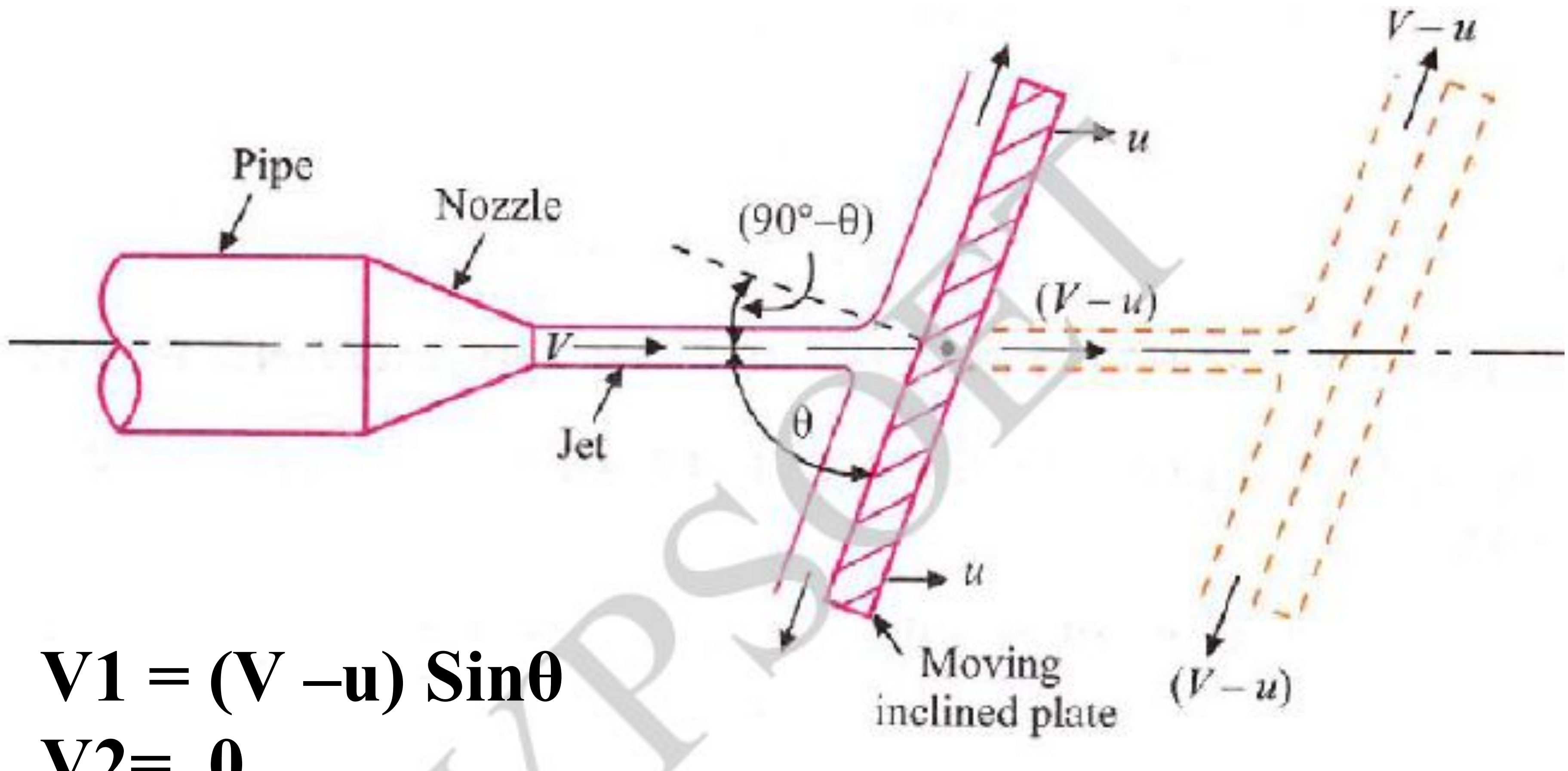
Since plate is fixed (stationary) distance travelled = 0

Work done per second= 0

Efficiency = 0

Force exerted by jet on moving inclined plate(away from jet)

Let a jet of water, coming out from the nozzle; strike an moving inclined flat plate as shown in the figure.



V1 = (V –u) Sinθ

V2= 0

Let,

u or U= Velocity of plate

θ=inclination of plate with the jet

= angle between jet and plate

V=velocity of jet(velocity of water jet before striking)

V_1 =velocity of water jet before striking to the plate in the horizontal direction = **$(V - u)\sin\theta$**

V2 = velocity of water jet after striking to the plate in the horizontal direction = **0**

d=diameter of jet

ρ=density of fluid (water)

A=cross section area of jet = $\frac{\pi}{4}d^2$

$\dot{m}$ =mass flow rate of fluid, kg/s =

Relative velocity of jet w. r .t . to plate = V-u

∴ Mass of water striking the plate per second

=ρA × velocity with which jet strikes the plate

$\dot{m} = \rho A(V - u)$

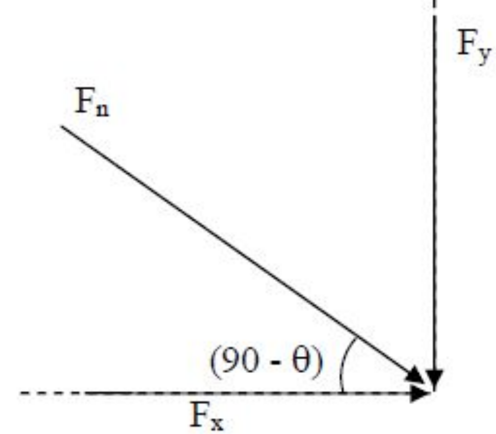
F_n =(Mass of jet strikes/sec) ×[Initial relative velocity of jet in the direction normal to plate - Final relative velocity of jet in the direction normal to plate]

$$F = \dot{m}(V_1 - V_2)$$

$$Fn = \rho A(V - u) \times [(V - u)\sin\theta - 0]$$

$$\therefore F_n = \rho A(V - u)^2 \sin\theta$$

F_n =force exerted by the jet on the moving plate in the **direction normal to the plate**



- Force F_x in the direction of jet,

$$F_x = F_n Cos(90-\theta) = F_n \sin\theta$$

$$Fx = \rho A(V-u)^2 \sin\theta \sin\theta$$

$$Fx = \rho A(V-u)^2 \sin^2\theta$$

- Force F_y perpendicular to the direction of jet,

$$F_y = F_n \cos\theta$$

$$Fy = \rho A(V-u)^2 \sin\theta \cos\theta$$

- Work done per second by jet on moving inclined flat plate

$$W = F_x \times u$$

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

$$\therefore W = \rho A u (V-u)^2 \sin^2\theta$$

- **Efficiency of jet = Output/input**
- Output= work done per second,
- input = Kinetic energy per second of water jet

$$\eta = \frac{\rho A u (V-u)^2 \cdot \sin^2\theta}{\frac{1}{2} m V^2} = \frac{\rho A u (V-u)^2 \cdot \sin^2\theta}{\frac{1}{2} (\rho A V) V^2} \qquad \eta = \frac{2u(V-u)^2 \sin^2\theta}{V^3}$$

Force exerted by jet on stationary symmetrical curved plate – from centre (Centrally)

Consider jet of the water striking on the curved fixed blade at the center of blade as shown in fig.

The jet after striking the plate comes out with the same velocity if the plate is smooth and there is no loss of energy due to the impact of jet, in the tangential direction of the curved vane.

Now velocity at the **outlet** of the plate can be resolved in to **two components**,

one in the direction of the jet and

perpendicular to the direction of the jet.

Let,

θ = angle made by jet with X axis at out let

V=velocity of jet(velocity of water jet before striking)

V_1 =velocity of water jet before striking to the plate in the horizontal direction

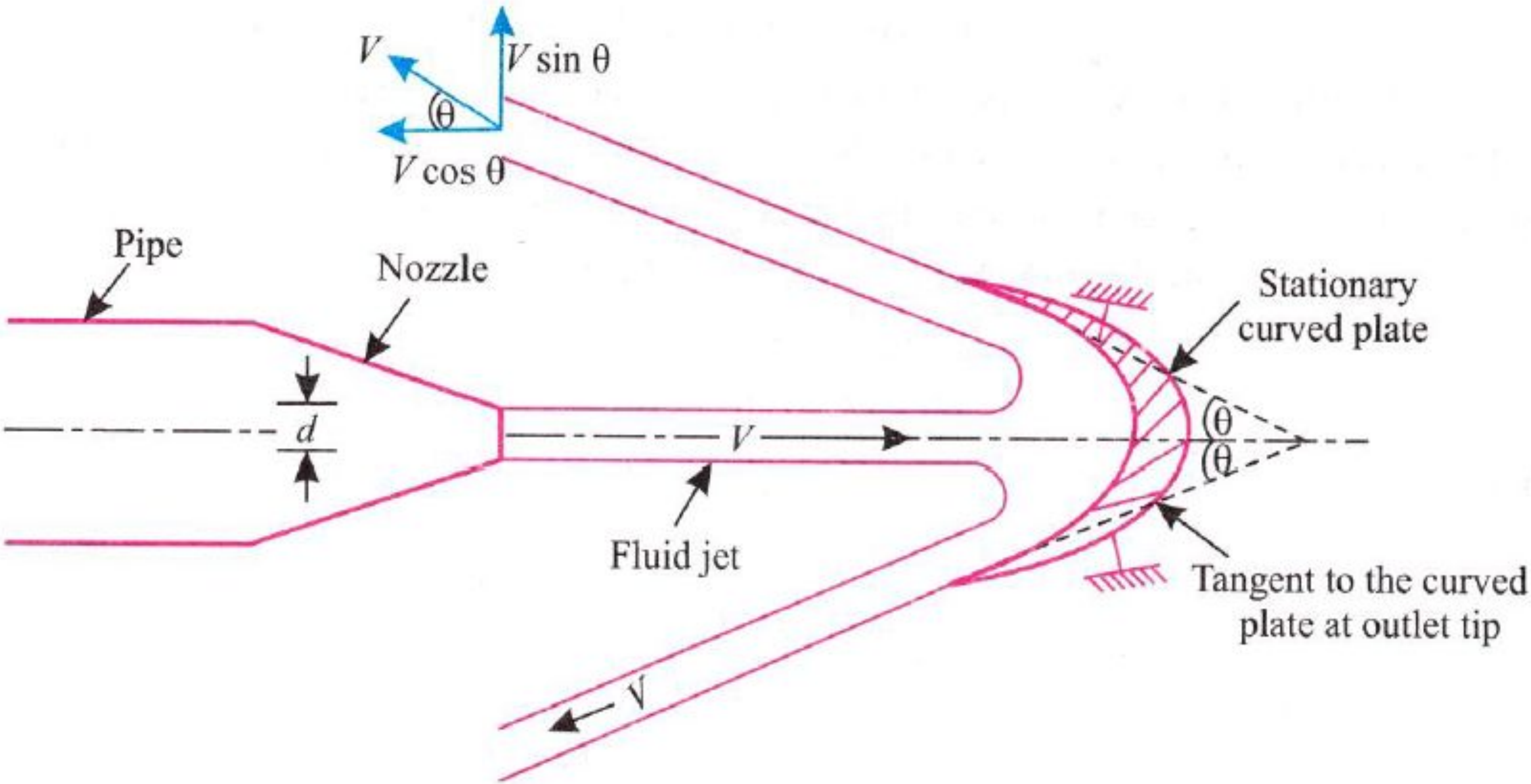
V2 = velocity of water jet after striking to the plate in the horizontal direction

d=diameter of jet

ρ=density of fluid (water)

A=cross section area of jet = $\frac{\pi}{4} d^2$

$\dot{m}$ =mass flow rate of fluid, kg/s


V
V sin θ
θ
V cos θ
Pipe
Nozzle
d
V
Fluid jet
Stationary
curved plate
θ
θ
Tangent to the curved
plate at outlet tip
V

The plate is smooth and there is no loss of energy due to impact of jet. Hence liquid leaving the plate with velocity V in the tangential direction of the curved plate. i.e.

Force exerted by jet in the direction of jet,

Force exerted by jet in the horizontal direction ,

F = (mass of water/sec) * $(V_1 - V_2)$= $\dot{m}$ * $(V_1 - V_2)$

$F_x = \dot{m} * (V_{1x} - V_{2x})$ ($F_{x=}$ Force in X direction)

where

V_{1x} =initial velocity of water jet in direction of jet = V

V_{2x} =final velocity water in direction of jet = $-V\cos\Theta$

negative sign as final velocity is opposite to direction jet velocity

$$F_x = (\rho AV)*[V-(-V\cos\Theta]$$

$$\mathbf{F_x = \rho AV^2[1+\cos\Theta]}$$ **Force exerted by jet in the horizontal direction**

Force exerted by jet on curved fixed plate in vertical direction

$$F_y = (\rho AV)*[V_{1y} - V_{2y}]$$

where

V_{1y} =initial velocity of water jet in vertical direction= 0

V_{2y} =final velocity water in vertical direction= +V sinΘ

positive sign as final velocity is direction jet velocity

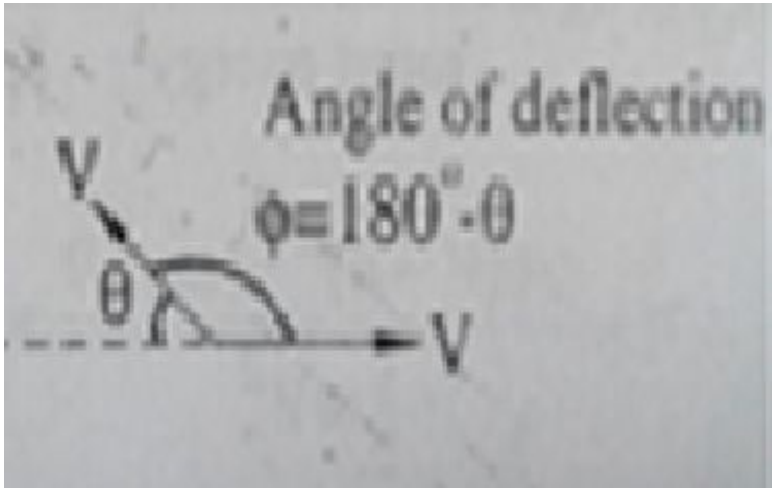
Force exerted by jet on curved fixed plate in vertical direction

$$F_y = (\rho AV)*[0 - V\sin\Theta]$$

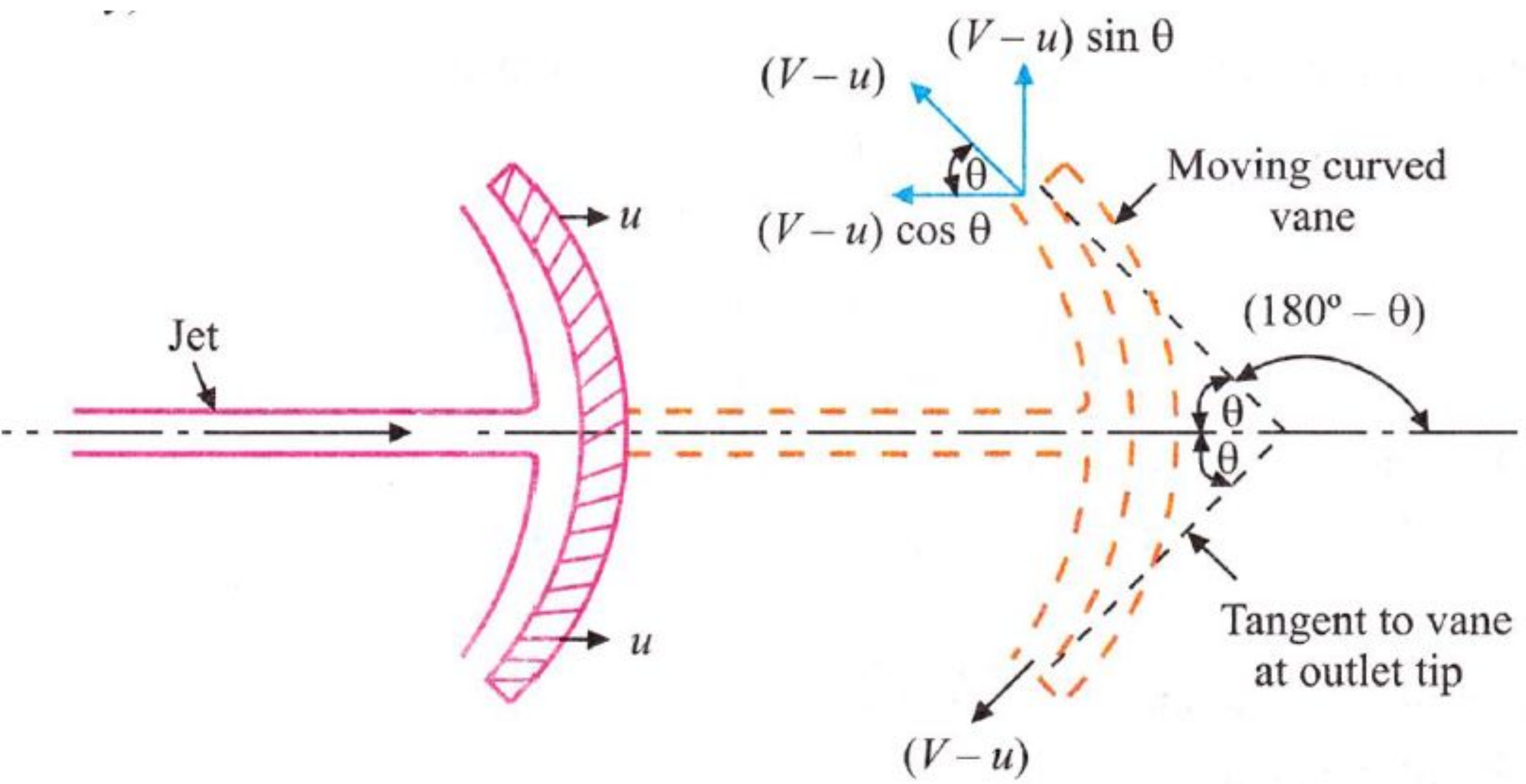
$$\mathbf{F_y = -\rho AV^2 \sin\Theta}$$

Negative sign = Fy acts downword

Angle of deflecton = 180°- Θ



Force exerted by jet on moving symmetrical curved plate -Centrally



Force exerted by jet on **moving symmetrical curved** plate -Centrally

consider jet of the water striking on the curved moving blade at the centre of blade as shown in fig.

The jet after striking the plate comes out with the same velocity if the plate is smooth and there is no loss of energy due to the impact of jet, in the tangential direction of the curved vane.

Now velocity at the **outlet of the plate** can be resolved in to two components,

one in the direction of the jet and

perpendicular to the direction of the jet.

Let,

V=velocity of jet(velocity of water jet before striking)

V_1 =velocity of water jet before striking to the plate in the horizontal direction

$V2$ = velocity of water jet after striking to the plate in the horizontal direction

θ = angle made by jet with X axis at out let

U or u= Velocity of plate

d=diameter of jet

ρ=density of fluid (water)

A=cross section area of jet = $\frac{\pi}{4}d^2$

$\dot{m}$=mass flow rate of fluid, kg/s **ρA [V-u]**

Relative velocity of jet w. r .t . to plate = V-u

Force exerted by jet on curved moving plate in the direction of jet,

$$\mathbf{F_x = (mass\ of\ water/sec) * (V_{1x} - V_{2x})}$$

where

V_{1x} =initial velocity of water jet with which strikes the plate in direction of jet **= V-u**

V_{2x} =final velocity of water jet in direction of jet = **-(V-u) cosΘ**

$$\mathbf{F_x = \rho A\,[V\text{-}u])*[(V\text{-}u)\text{-}\ (\text{-}[V\text{-}u\,]\cos\Theta\,]]}$$

$$F_x = \rho A(V-u)[(V-u)+(V-u)\cos\Theta]$$

$$F_x = \rho A(V-u)^2[1+\cos\Theta]$$

Force exerted by jet in horizontal direction

Force exerted by jet on curved moving plate in **vertical direction**

$$F_y = (\rho A[V\text{-}u])*[V_{1y} - V_{2y}]$$

where

V_{1y} =initial velocity of water jet in vertical direction= **0**

V_{2y} =final velocity water in vertical direction=**(V-u) sinΘ**

$$F_y = (\rho A[V\text{-}u])*[0 - (V\text{-}u)\sin\Theta]$$

$$\mathbf{F_y = -\rho A(V - u)^2 \sin\Theta}$$

Force exerted by jet on curved fixed plate in vertical direction

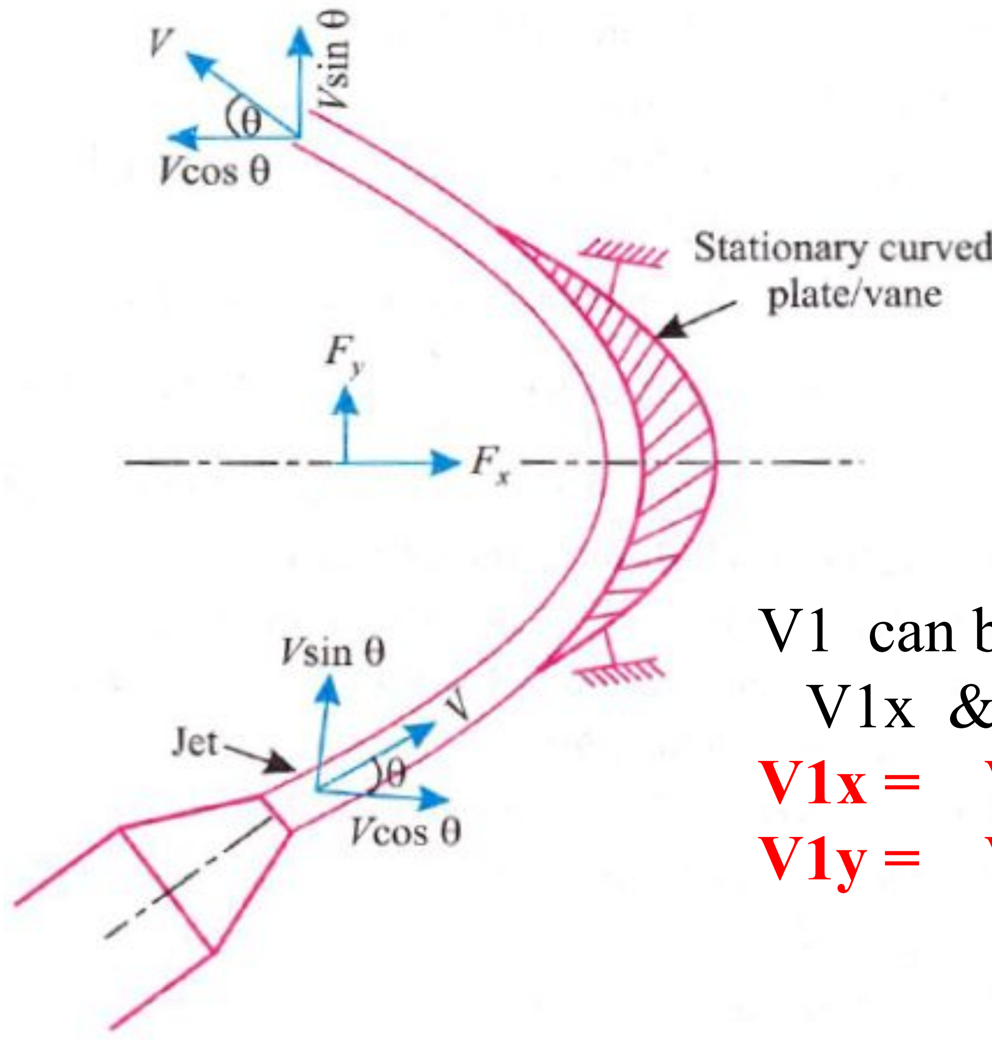
Force exerted by jet on stationary **curved plate** from one end of plate – **symmetrical plate**

consider a water jet striking on symmetrical curved plate **tangentially** at one end as shown in fig

V2 can be
V2x & V2y

V2x = - Vcos θ

V2y = Vsin θ



V1 can be
V1x & V1y

V1x = Vcos θ

V1y = Vsin θ

Let,

V= velocity of jet(velocity of water jet before striking)

V_1 =velocity of water jet before striking to the plate **from one end of tip (inlet tip)**

V2 = velocity of water jet after striking to the plate **from other end of tip (outlet tip)**

θ = angle between jet and x-axis at the tip of plate at inlet & angle between jet and y-axis at the tip of plate at outlet

d=diameter of jet

ρ=density of fluid (water)

A=cross section area of jet

$\dot{m}$ =mass flow rate of fluid, kg/s =ρav

Force exerted by jet on curved Fixed plate when jet strikes from one end in the direction of jet,

F_x = (mass of water/sec) * (V_{1x} - V_{2x})

where

V_{1x} =initial velocity of water jet with which strikes the plate in horizontal direction = $V\cos\Theta$

V_{2x} =final velocity of water jet in direction of jet = **$-V\cos\Theta$**

$$F_x = (\rho AV)*(V_{1x} - V_{2x}) = (\rho AV)*[V\cos\Theta - (-V\cos\Theta)]$$

$$= \rho AV*(2V\cos\Theta)$$

$$Fx = 2\rho AV^2\cos\Theta$$

Force exerted by jet in horizontal direction

Force exerted by jet on curved Fixed plate when jet strikes from one end in the perpendicular direction of jet,

$$F_y = (\rho AV)*[V_{1y} - V_{2y}]$$

where

V_{1y} =initial velocity of water jet in vertical direction= $V\sin\Theta$

V_{2y} =final velocity water in vertical direction = $\mathbf{V\sin\Theta}$

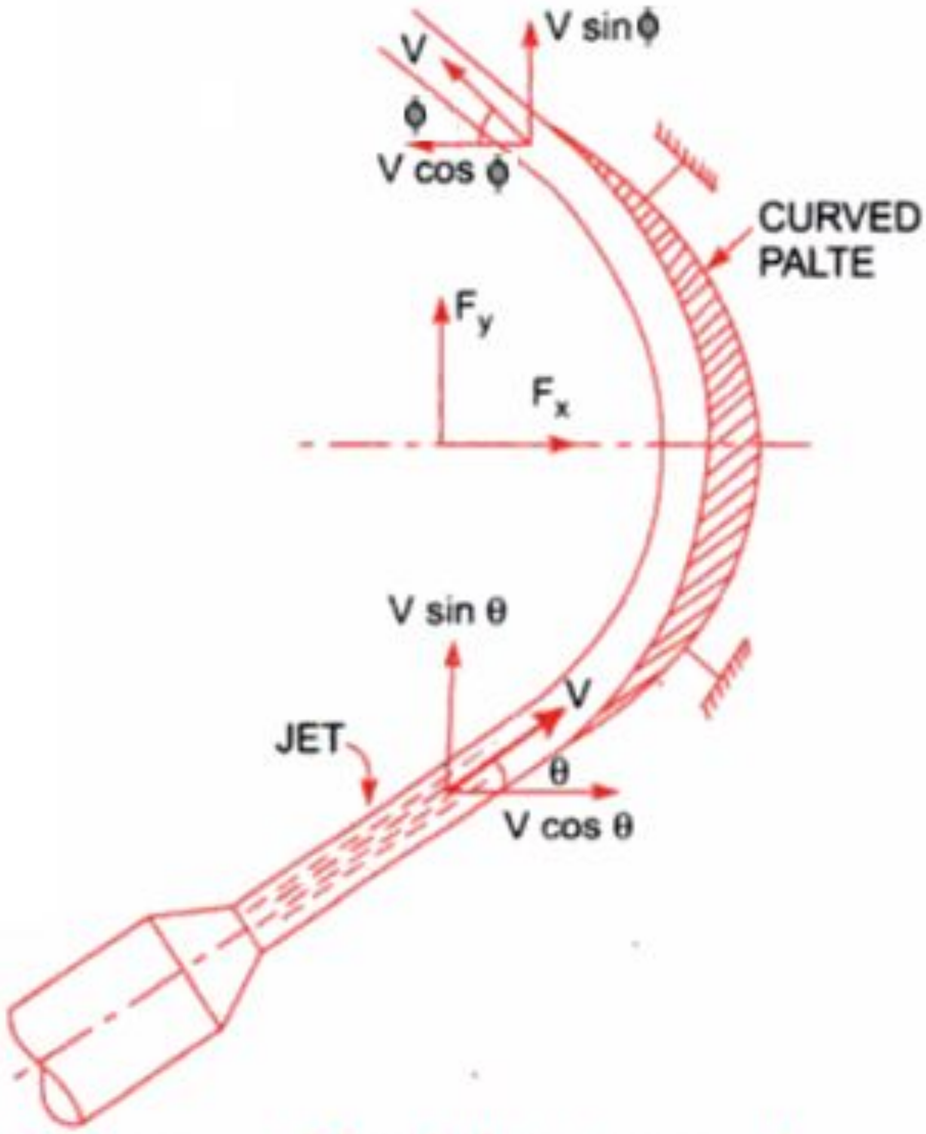
$$F_y = (\rho AV)*[V_{1y} - V_{2y}]$$

$$= (\rho AV)*[V\sin\Theta - V\sin\Theta]$$

$$\mathbf{Fy = 0}$$

Force exerted by jet on curved moving plate when jet strikes from one end in the perpendicular direction of jet = 0

Force exerted by jet on stationary curved plate from one end of plate – unsymmetrical plate



Jet striking curved fixed plate at one end.

Force exerted by jet on curved plate from one end of plate – unsymmetrical plate

consider a water jet striking on unsymmetrical curved plate tangentially at one end as shown in fig.

Let θ = angle between water jet and x-axis at of inlet tip.

ϕ = angle between water jet and x-axis at of outlet tip.

$F_x = (\rho AV)^* (V_{1x} - V_{2x}) = \rho AV * [V\cos\theta - (-V\cos\phi)]$

$$\mathbf{F_x = \rho AV^2[\cos\theta + \cos\phi]}$$

Force exerted by jet on curved plate in horizontal direction from one end of plate –plate is unsymmetrical

$F_y = (\rho AV)^*[V_{1y} - V_{2y}] = (\rho AV)^*[V\sin\theta - V\sin\phi]$

$$\mathbf{F_y = \rho AV^2[\sin\theta - \sin\phi]}$$

Force exerted by jet on curved plate in vertical direction from one end of plate –plate is unsymmetrical is given by above mail

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

Fig. shows a jet of water striking a moving curved plate/vane/blade tangentially at one of its tips. As the jet strikes tangentially, the loss of energy due to impact of the jet will be zero.

In this case as plate is moving, the velocity with which jet of water strikes is equals to the relative velocity of the jet with respect to the plate.

As the direction of jet velocity and vane velocity is not same, the relative velocity at inlet will be vector difference of the jet velocity and plate velocity at inlet.

$$\bar{v} - \bar{u}$$

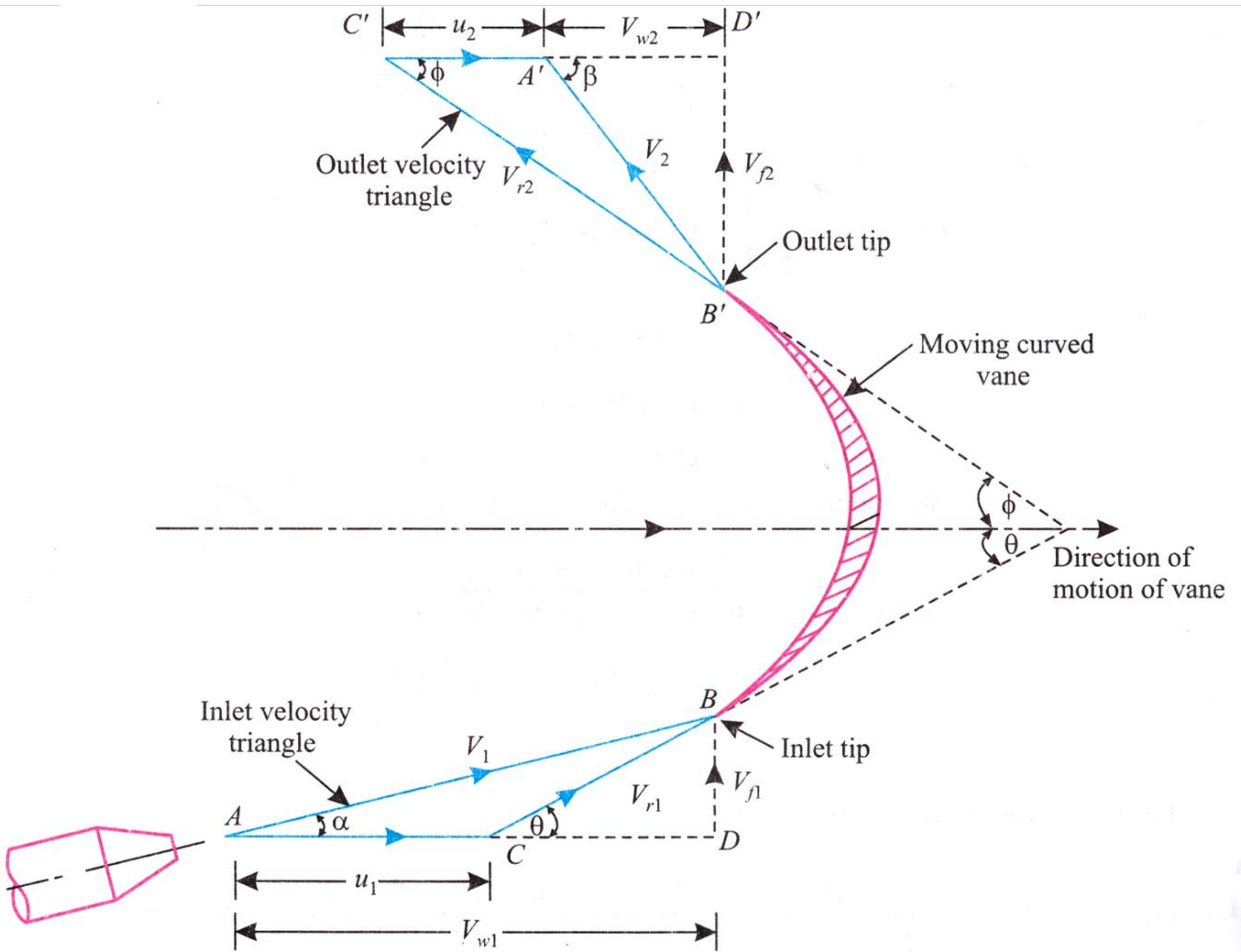
The fluid jet with an **absolute velocity V_1** strikes the blade at the inlet. The **relative** velocity of the jet $\mathbf{V_{r1}}$ at the **inlet** is obtained by **subtracting** vectorially the velocity **u** of the vane from V_1.

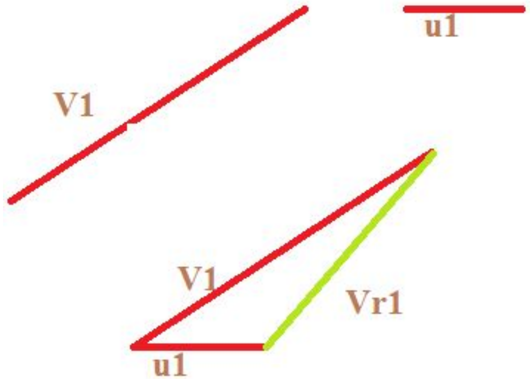
$$\overline{V_{r1} = v_1} - \bar{u}$$

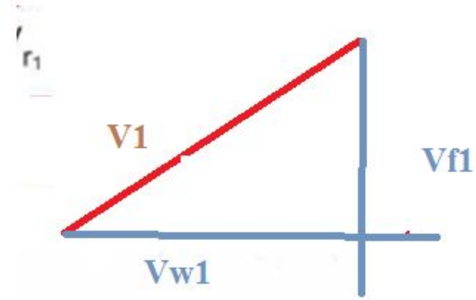
The jet strikes the blade without shock if α (refer Fig.) coincides with the inlet angle at the tip of the blade.

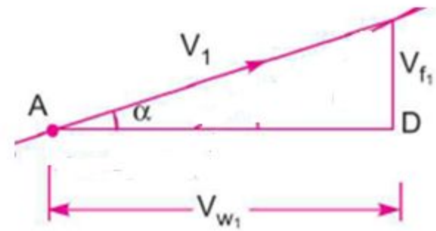
If friction is neglected and pressure remains constant, then the relative velocity at the outlet is equal to that at the inlet ($V_{r2} = V_{r1}$).

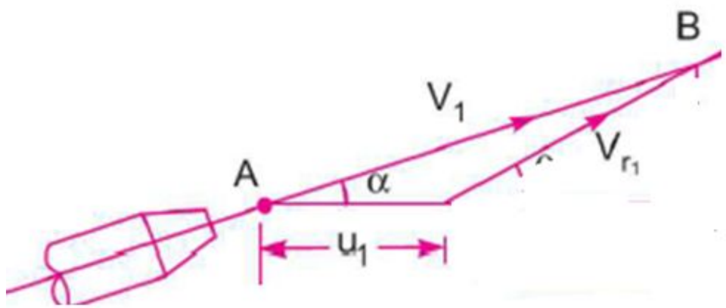
Jet striking unsymmetrical moving curved plate at one end

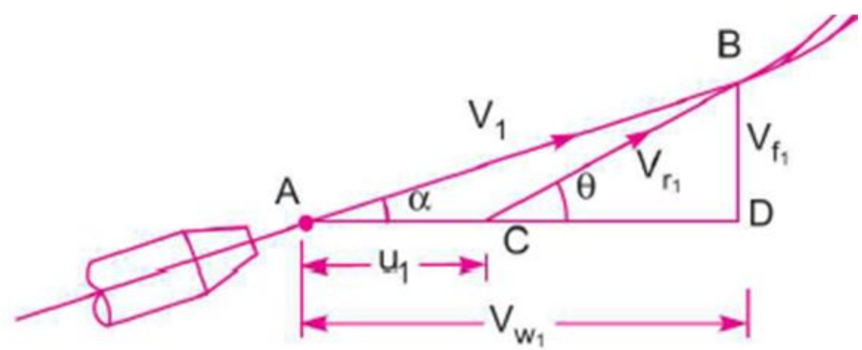


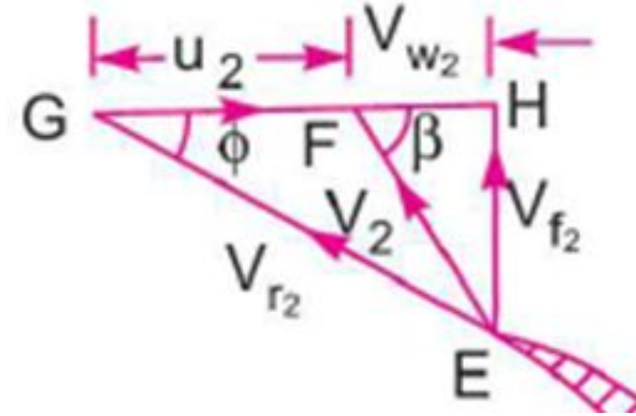

u1
V1
V1
Vr1
u1

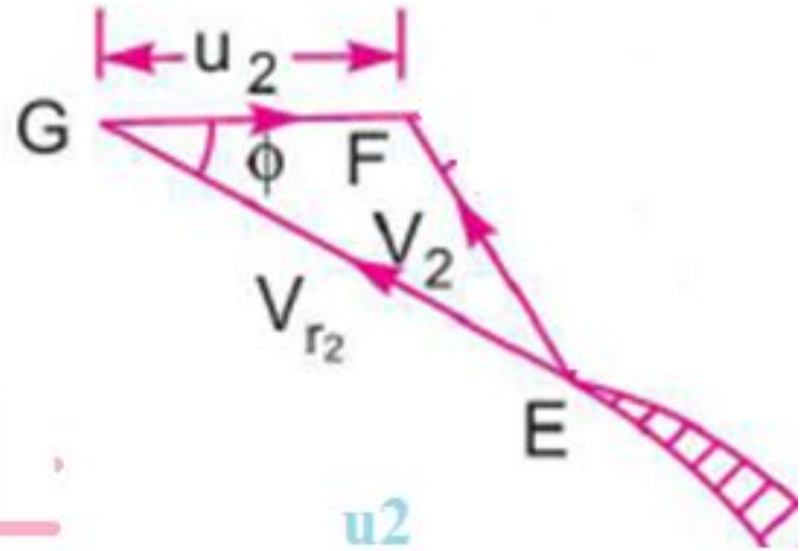

V1
Vf1
Vw1

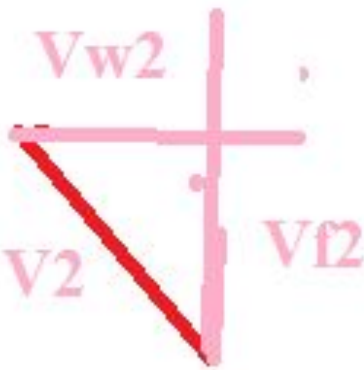

V1
Vf1
A
α
D
Vw1

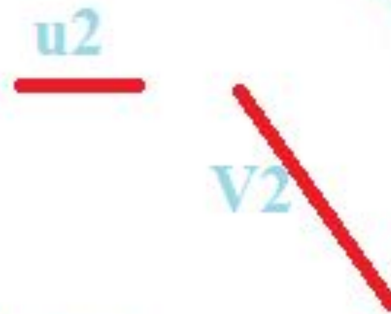

B
V1
Vr1
A
α
u1

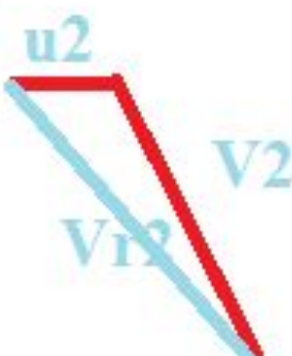

B
V1
Vf1
Vr1
A
α
θ
D
C
u1
Vw1


u2
Vw2
G
H
φ
F
β
V2
Vf2
Vr2
E


u2
G
φ
F
V2
Vr2
E

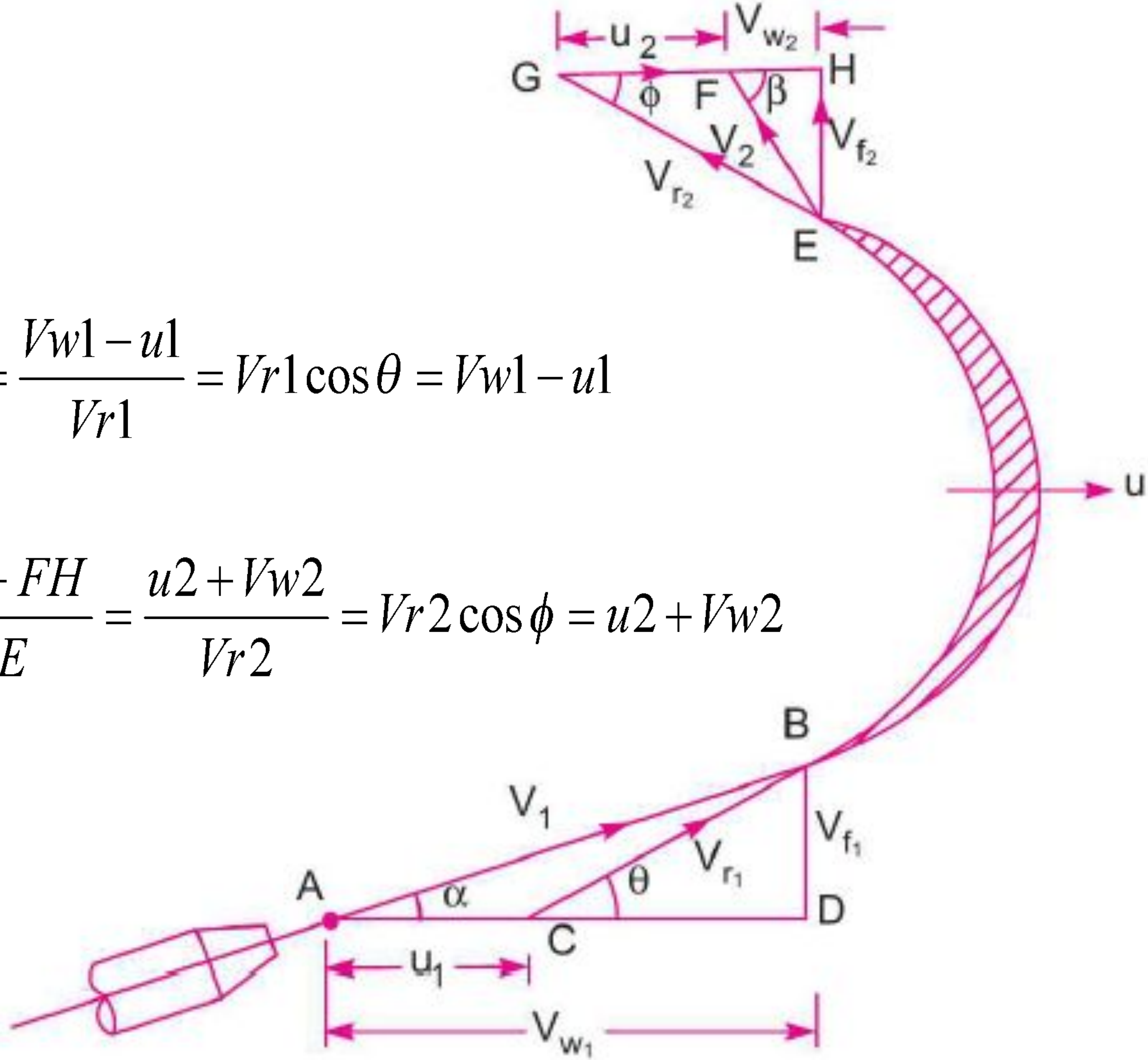

Vw2
V2
Vf2

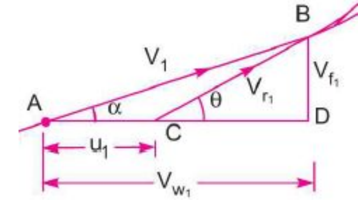

u2
V2


u2
V2
Vr2

$$\cos\theta = \frac{CD}{CB} = \frac{Vw1 - u1}{Vr1} = Vr1\cos\theta = Vw1 - u1$$

$$Cos\phi = \frac{GF + FH}{GE} = \frac{u2 + Vw2}{Vr2} = Vr2\cos\phi = u2 + Vw2$$



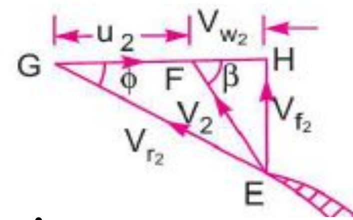


V_1 = Absolute velocity of the jet at inlet

V_2 = Absolute velocity of the jet at outlet

V_{r1} = Relative velocity of the jet and plate at inlet

V_{r2} = Relative velocity of the jet and plate at outlet



u_1 = Velocity of the vane at inlet

u_2 = Velocity of the vane at outlet

α = Angle between the direction of the jet and direction of motion of the plate at inlet or (Guide blade angle)

θ = Angle made by the relative velocity V_{r1}, with the direction of motion of the vane at inlet or (Vane/blade angle at inlet)

V_{w1} = The components of the velocity of the jet V_1 in the direction of motion

V_{f1} = The components of the velocity of the jet V_1 in the perpendicular to the direction of motion

- **V_{w1} = Velocity of whirl at inlet V_{f1} Velocity of flow at inlet**

β= Angle made by the velocity with the direction of motion of the vane at outlet

φ =Angle made by the relative velocity , with the direction of motion of the vane at outlet (Vane/blade angle at outlet)

V_{w2} = The components of the velocity V_2 , in the direction of motion of vane outlet

V_{f2} = The components of the velocity V_2, perpendicular to the direction of motion of the vane at outlet

V_{w2} = Velocity of whirl at outlet

V_{f2} = Velocity of flow at outlet

The triangles ABD and EGH are called the velocity triangles at inlet and outlet respectively.

If the vane is smooth and having velocity in the direction of motion at inlet and outlet equal then we have,

$u_1 = u_2 = u$ = velocity of vane in the direction of motion

and $V_{r1} = V_{r2}$

Mass of water striking the vane per second, = $\rho A V_{r1}$

- **Force exerted by the jet in the direction of motion,**
- *Fx= mass of water striking per sec * [Initial velocity with which jet strikes in the direction of motion - Final velocity of jet in the direction of motion]*
- Initial velocity with which jet strikes the vane = V_{r1}
- and,The component of this velocity in the direction of motion = V_{r1} **$\cos\theta = (V_{w1} - u_1)$**
- The component of the relative velocity at outlet in the direction of motion= $-V_{r2}$ **$\cos\varphi = -(V_{w2} + u_2)$**

$$F = \dot{m}\,(V1 - V2)$$

So,

$$\therefore F_x = \dot{m} \times [V_{r1}\cos\theta - (-V_{r2}\cos\varphi)]$$

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

As we know $u_1 = u_2$

$$F_x = \rho a V_{r1} \times [V_{w1} + V_{w2}] ---------(2.19)$$

Equaion 2.19 is true only when angle β shown in Fig. 2.9 is acute angle ($< 90°$).

- If $\beta = 90°$ then $V_{w2} = 0$ and equation 2.19 becomes,

$$F_x = \rho a V_{r1} V_{w1} ---------(2.20)$$

- If β is an obtuse angle ($> 90°$), the expression for F_x will become,

$$F_x = \rho a V_{r1} \times [V_{w1} - V_{w2}] ---------(2.21)$$

- In general,

$$F_x = \rho a V_{r1} \times [V_{w1} \pm V_{w2}] ---------(2.22)$$

- **Work done per second** on the vane by the jet,

$$W = Force \times distance\ travelled\ per\ sec\ in\ the\ direction\ of\ force$$

$$W = F_x \times u$$

$$W = \rho a u V_{r1} \times [V_{w1} \pm V_{w2}] ---------(2.23)$$

- **Work done per second** on the vane by the jet,

$$W = Force \times distance\ travelled\ per\ sec\ in\ the\ direction\ of\ force$$

$$W = F_x \times u$$

$$W = \rho a u V_{r1} \times [V_{w1} \pm V_{w2}] - - - - - - - - - -(2.23)$$

- Work done per second **per unit weight** of fluid striking per second ,

$$= \frac{\rho a u V_{r1} \times [V_{w1} \pm V_{w2}]}{(\rho a V_{r1}) \times g}$$

$$= \frac{1}{g}[V_{w1} \pm V_{w2}] \times u \quad \frac{N.m}{N} - - - - - - - - - -(2.24)$$

- Work done per second **per unit mass** of fluid striking per second ,

$$= \frac{\rho a u V_{r1} \times [V_{w1} \pm V_{w2}]}{(\rho a V_{r1})}$$

$$= u \times [V_{w1} \pm V_{w2}] \frac{N.m}{Kg} - - - - - - - - - -(2.25)$$

Force exerted by the jet on a moving plate

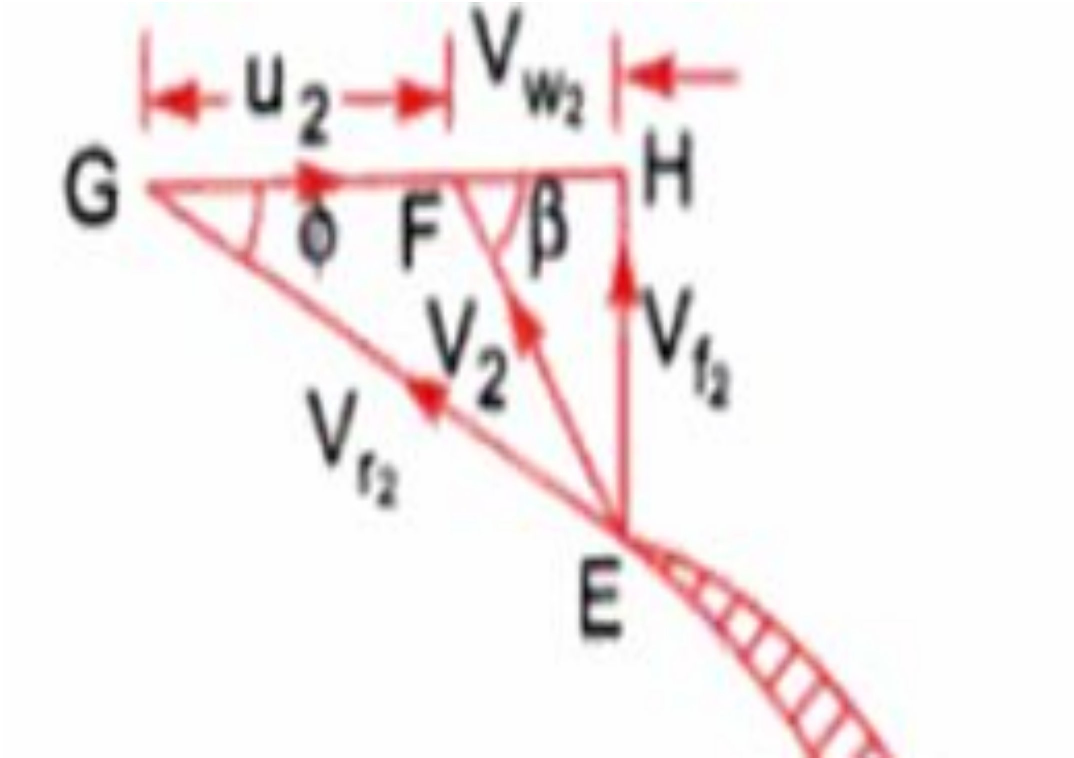
Considering Relative Velocity,

If $\beta < 90^0$

$$F_x = \rho a V_{r1} (V_{r1} \cos\theta + V_{r2} \cos\varphi)$$

OR

$$F_x = \rho a V_{r1} (V_{W1} + V_{W2})$$



Force exerted by the jet on a moving plate

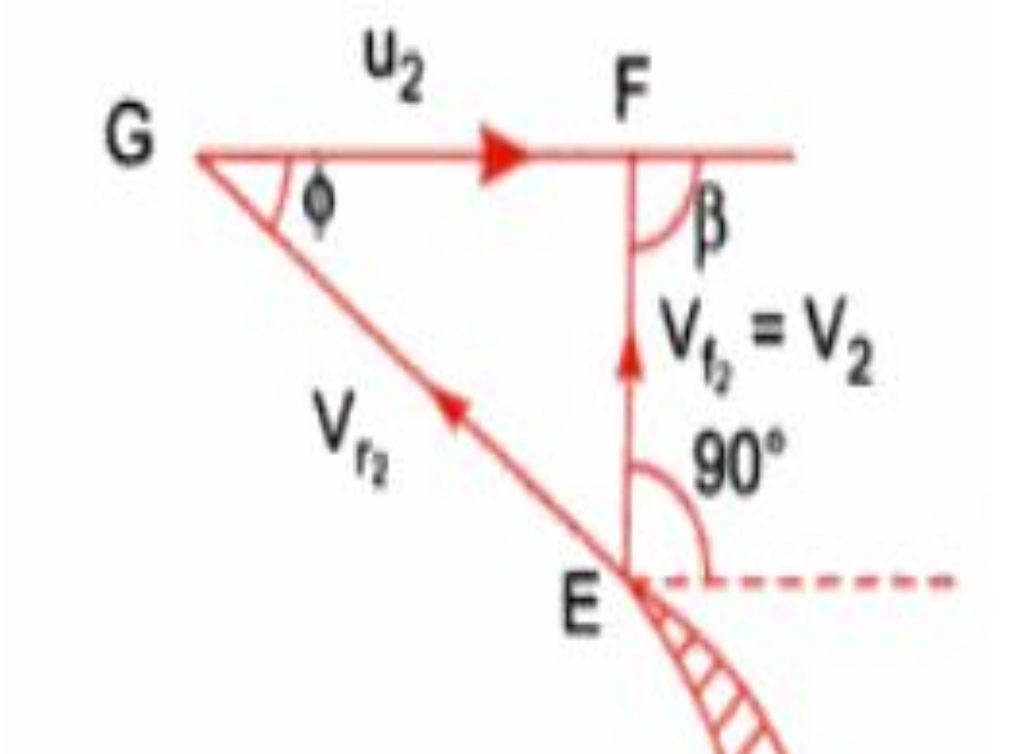
Considering Relative Velocity,

If $\beta = 90^0$

$$F_x = \rho a V_{r1} (V_{r1} \cos\theta - V_{r2} \cos\varphi)$$

OR

$$F_x = \rho a V_{r1} (V_{W1})$$



Force exerted by the jet on a moving plate

Considering Relative Velocity,

If β >90⁰

$$\cos\phi = \frac{GH}{GE} = \frac{GF - HF}{GE} = \frac{u2 - Vw2}{Vr2} = Vr2\cos\phi = u2 - Vw2$$

Fx= mass flow rate(v1- V2)

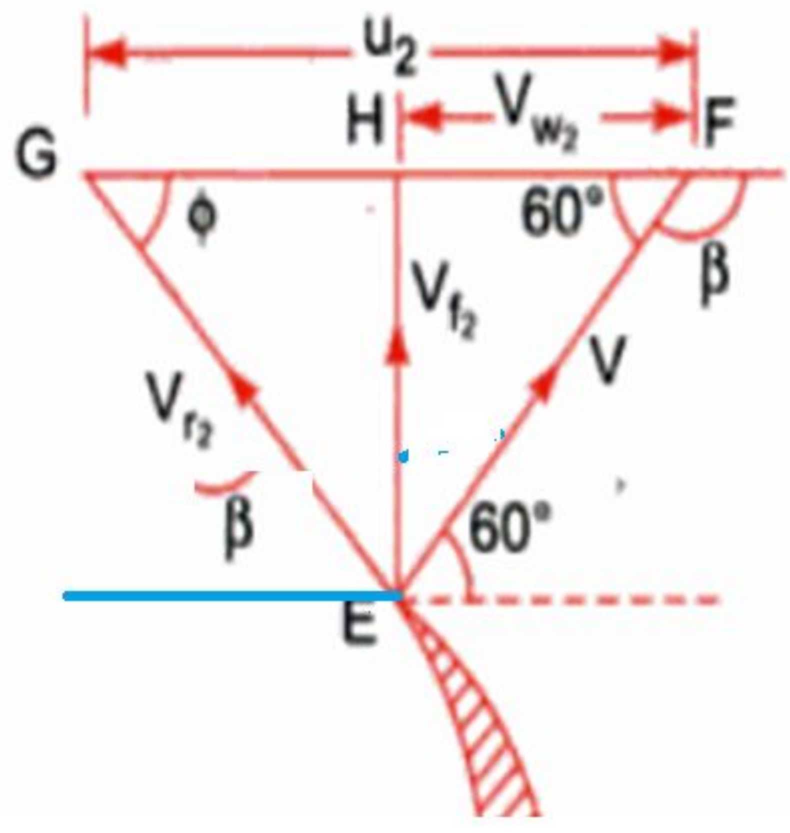
$$F_x = \rho a V_{r1} (V_{r1} \cos\theta - V_{r2} \cos\varphi)$$

$$F_x = \rho a V_{r1} (Vw1 - u1 - (-V_{r2} \cos\varphi))$$

$$Fx = \rho a V_{r1} (Vw1 - u1 + u2 - Vw2)$$

$$Fx = \rho a V_{r1} (Vw1 - u + u - Vw2)$$

$$F_x = \rho a V_{r1} (V_{W1} - V_{W2})$$



In General force exerted by jet on moving unsymmetrical curved plate from one end tangentially

$$\mathbf{F_x = \rho a V_{r1} (V_{W1} \pm V_{W2})}$$

- Power developed, P =work done in J/s

$$\therefore P = \rho A V_{r1}[V_{w1} \pm V_{w2}] \times u$$

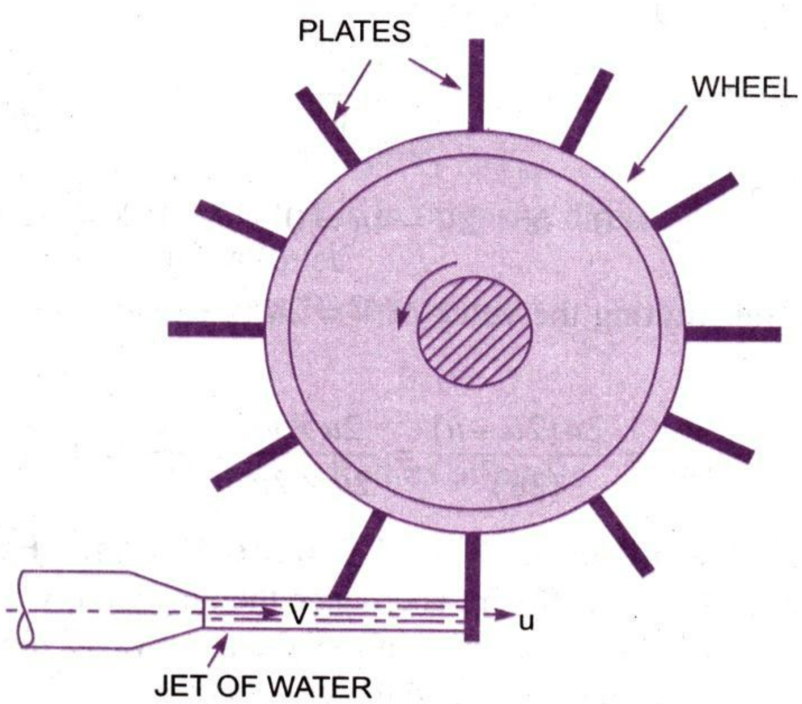
Angle of Deflection

$$\phi = 180 - (2\beta)$$

Force exerted by a jet of water on a series of flat vanes

The force exerted by a jet of water on a **single moving plate is not practically feasible. Its only a theoretical one.**

In actual practice, a **large number of plates/blades** are mounted on the circumference of a wheel at a fixed distance apart as shown in Figure



The **jet strikes a plate** and due to the force exerted by the jet on the plate, the **wheel starts moving and the 2nd plate mounted on the wheel appears before the jet**, which again **exerts the force on the 2nd plate**. Thus each plate appears successively before the jet and jet exerts force on each plate and the **wheel starts moving at a constant speed.**

- Let,

$$V = \text{Velocity of jet}$$
$$d = \text{Diameter of jet}$$
$$u = \text{Velocity of vane}$$

- In this case the mass of water coming out from the nozzle per second is always in contact with the plates, when all the plates are considered.
- Hence, mass of water per sec striking the series of plates $= \rho aV$

  Also,

  The jet strikes a plate with velocity $= (V - u)$

- After striking, the jet moves tangential to the plate and hence the velocity component in the direction of motion of plate is equals to zero.
- **Force exerted** by the jet in the direction of motion of plate,

$$F_x = \rho aV[(V - u) - 0]$$
$$F_x = \rho aV(V - u) \text{---------} (2.26)$$

- **Work done** by the jet on the series of plates per second,

$$W = Force \times Distance\ travelled\ per\ sec\ in\ the\ direction\ of\ force$$
$$W = F_x \times u$$

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

$$W = \rho a V u(V - u) ---------(2.27)$$

- **Kinetic energy** of the jet per second,

$$KE = \frac{1}{2}\dot{m}V^2$$

$$KE = \frac{1}{2}(\rho a V)V^2$$

$$KE = \frac{1}{2}\rho a V^3$$

- **Efficiency** of the wheel,

$$\eta = \frac{Work\ done\ per\ second}{Kinetic\ energy\ per\ second}$$

$$\eta = \frac{\rho a V u(V - u)}{\frac{1}{2}\rho a V^3}$$

$$\eta = \frac{2u(V - u)}{V^2} ---------(2.28)$$

Condition for maximum efficiency

- For a given jet velocity V, the efficiency will be maximum when,

$$\frac{d\eta}{du} = 0$$

$$\therefore \frac{d}{du}\left[\frac{2u(V-u)}{V^2}\right] = 0$$

$$\therefore \frac{d}{du}\left[\frac{2uV-2u^2}{V^2}\right] = 0$$

$$\therefore \frac{2V-4u}{V^2} = 0$$

$$\therefore 2V = 4u$$

$$\therefore \boldsymbol{u = \frac{V}{2}} ----------(\mathbf{2.29})$$

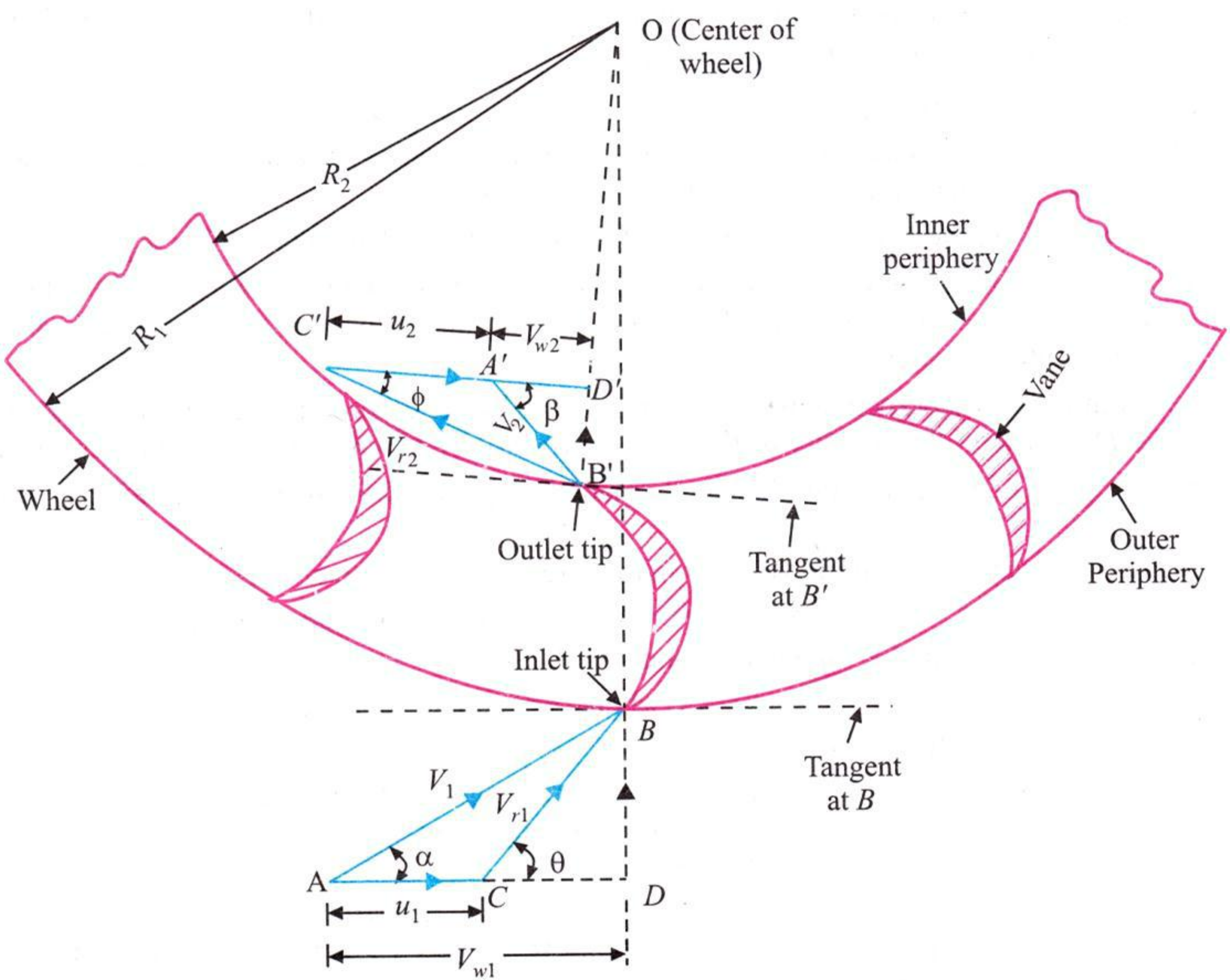
Maximum efficiency,

$$\eta_{max} = \frac{2u(V-u)}{V^2}$$

$$\eta_{max} = \frac{2u(2u-u)}{(2u)^2}$$

$$\eta_{max} = \frac{1}{2} = 50\% ----------(2.30)$$

Force Exerted by the Jet of Water on a Series of Radial Curved Vanes



For a radial curved vane, the radius of the vane at inlet and outlet is different and hence the tangential velocities of the radial vane at inlet and outlet will not be equal.

Consider a series of radial curved vanes mounted on a wheel as shown in Fig. The jet of water strikes the vanes and the wheel starts rotating at constant angular speed.

- Let,

  R_1 = Radius of wheel at inlet of the vane

  R_2 = Radius of wheel at outlet of the vane

  ω = Angular speed of the wheel

  Then,

$$u_1 = \omega R_1 \text{ and } u_2 = \omega R_2$$

- The *mass of water striking per second* for a series of vanes = The mass of water coming out from nozzle per sec = $\rho a V_1$

  Where,

  a = Area of jet, and

  V_1 = Velocity of jet

- *Momentum of water striking the vanes* in the tangential direction per sec at inlet = mass of water striking per sec X component of V_1 in the tangential direction

$$\therefore Momentum\ of\ water\ at\ inlet\ per\ sec = \rho a V_1 \times V_{w1} \qquad (\because V_{w1} = V_1 \cos\alpha)$$

- Similarly,

  Momentum of water at outlet per sec = $\rho a V_1$ X component of V_2 in the tangential direction

$$\therefore Momentum\ of\ water\ at\ outlet\ per\ sec = \rho a V_1 \times (-V_2 \cos\beta)$$

$$\therefore Momentum\ of\ water\ at\ outlet\ per\ sec = -\rho a V_1 \times V_{w2} \qquad (\because V_{w2} = V_2 \cos\beta)$$

- Now *angular momentum,*

$$Angular\ momentum\ per\ sec\ at\ inlet = Momentum\ at\ inlet \times Radius\ at\ inlet$$

$$= \rho a V_1 \times V_{w1} \times R_1$$

$$Angular\ momentum\ per\ sec\ at\ outlet = Momentum\ at\ outlet \times Radius\ at\ outlet$$

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

- *Torque exerted by the water* on the wheel,

$$T = Rate\ of\ change\ of\ angular\ momentum$$

$$T = [Initial\ angular\ momentum\ per\ sec - Final\ angular\ momentum\ per\ sec]$$

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

$$\therefore T = \rho a V_1 [V_{w1} R_1 + V_{w2} R_2] --------(2.31)$$

- *Work done per sec* on the wheel,

$$WD/sec = Torque \times Angular\ velocity$$

$$\therefore WD/sec = T \times \omega$$

$$\therefore WD/sec = \rho a V_1[V_{w1}R_1 + V_{w2}R_2] \times \omega$$

$$\therefore WD/sec = \rho a V_1[V_{w1}R_1\omega + V_{w2}R_2\omega]$$

$$\therefore WD/sec = \rho a V_1[V_{w1}u_1 + V_{w2}u_2] ---------(2.32)$$

- Equation 2.32 is valid only when, $\beta < 90$. If the angle β is an obtuse angle ($\beta > 90$) then,

$$WD/sec = \rho a V_1[V_{w1}u_1 - V_{w2}u_2] ---------(2.33)$$

- In general,

$$WD/sec = \rho a V_1[V_{w1}u_1 \pm V_{w2}u_2] ---------(2.34)$$

- If the discharge is radial at the outlet then, $\beta = 90°$ and hence $V_{w2} = 0$,

$$\therefore WD/sec = \rho a V_1[V_{w1}u_1] ---------(2.35)$$

- *Efficiency* of the radial curved vanes,

$$\eta = \frac{Work\ done\ per\ second}{Kinetic\ energy\ per\ second}$$

$$\therefore \eta = \frac{\rho a V_1[V_{w1}u_1 \pm V_{w2}u_2]}{\frac{1}{2}(\dot{m})V_1^2}$$

$$\therefore \eta = \frac{\rho a V_1[V_{w1}u_1 \pm V_{w2}u_2]}{\frac{1}{2}(\rho a V_1)V_1^2}$$

$$\therefore \eta = \frac{2[V_{w1}u_1 \pm V_{w2}u_2]}{V_1^2} \text{----------} (2.36)$$

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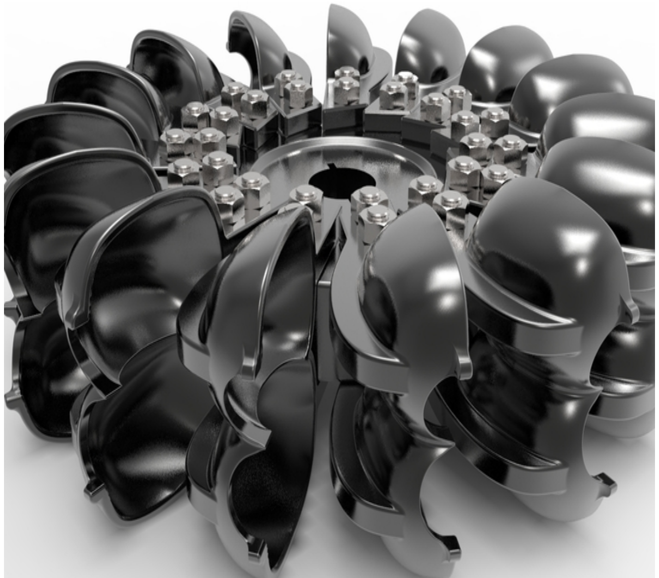
Unit –I B: Impulse Water Turbines

- **Hydraulic Turbines:**
- Introduction to Hydro power plant,
- Classification of Hydraulic Turbines,
- Concept of Impulse and Reaction Turbines.
- Construction, Principle of Working, design aspects, velocity diagrams and its analysis of Pelton wheel, Francis, and Kaplan turbines,
- Degree of reaction,
- Draft tube: types and efficiencies,
- governing of hydraulic turbines,
- Cavitation in turbines.



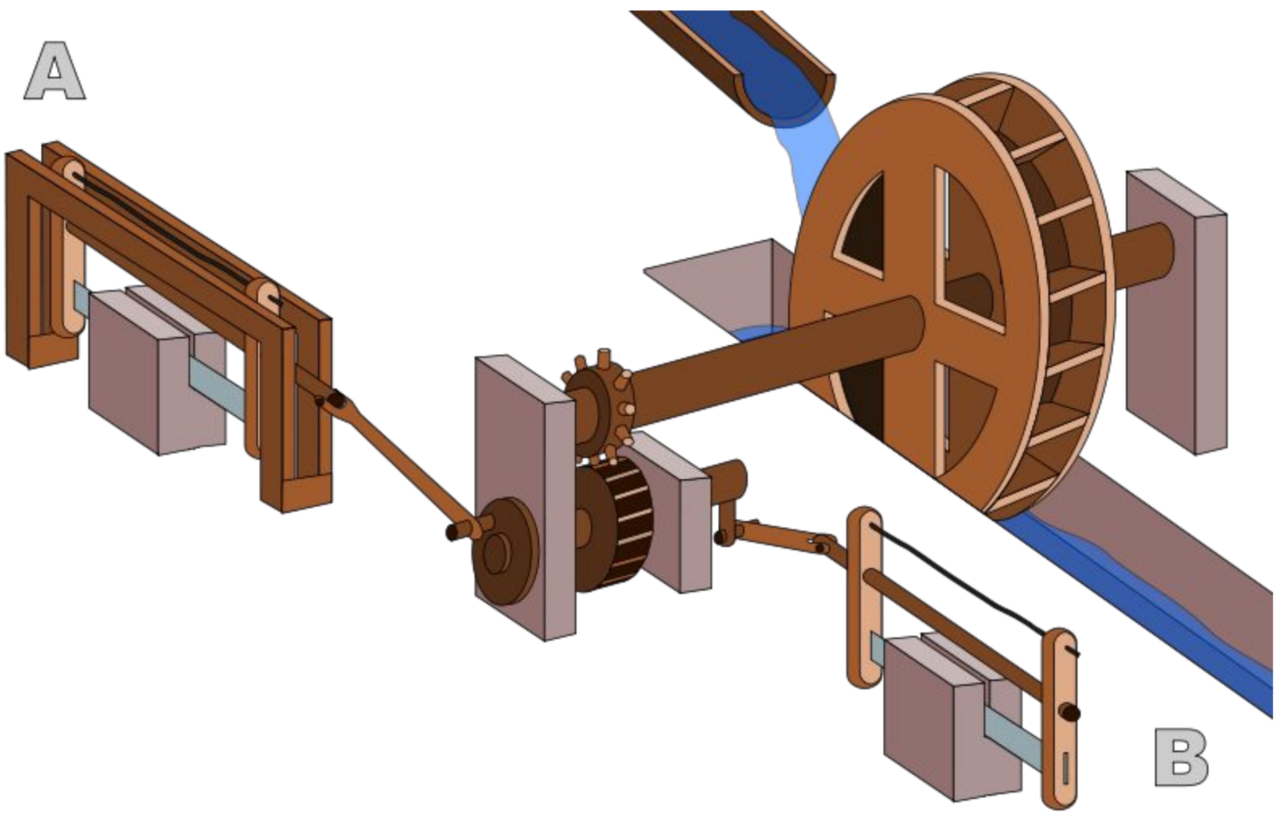
Turbine

TURBINES:- Turbines are defined as hydraulic machine which converts hydraulic energy into mechanical energy.

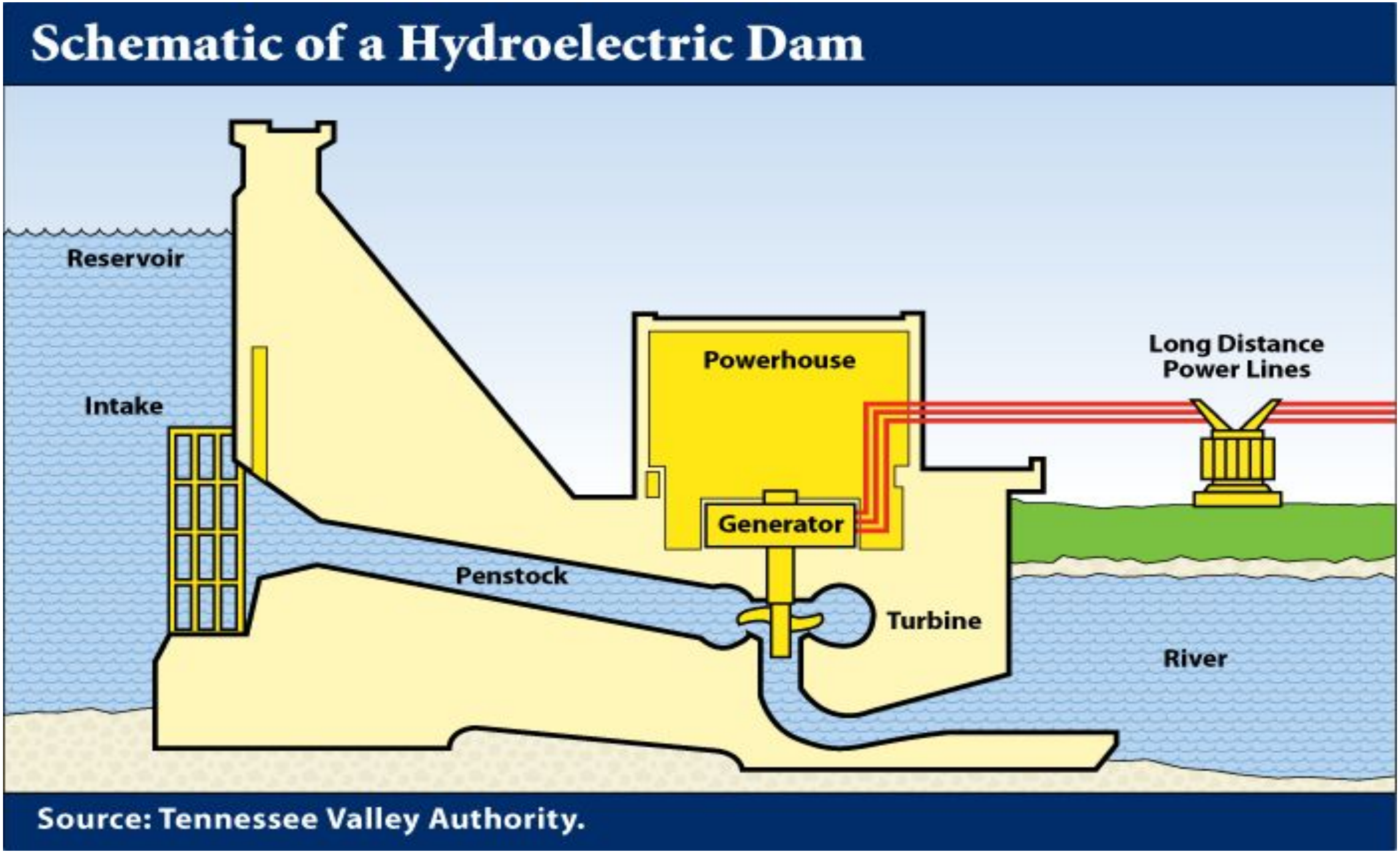
Turbine is a device that extracts energy from a fluid (converts the energy held by the fluid to mechanical energy)

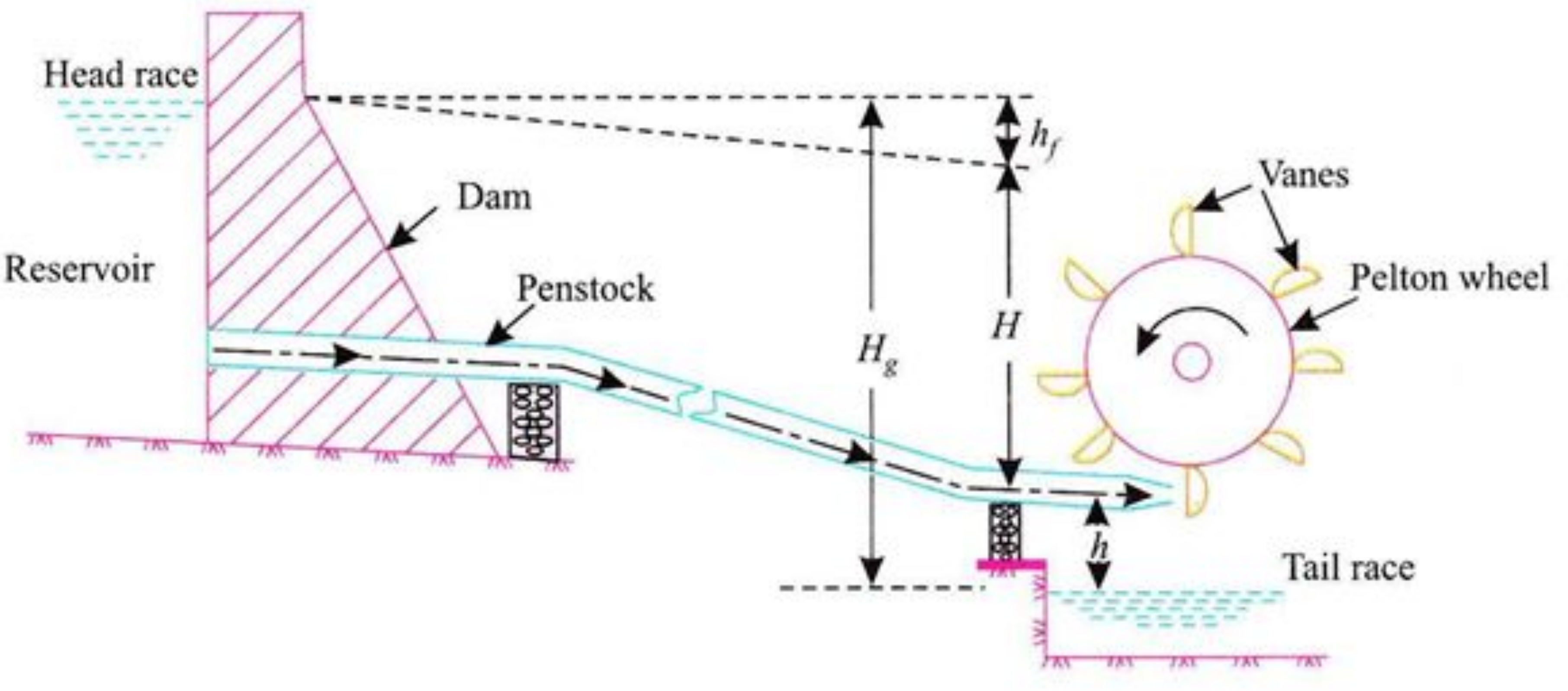
PRINCIPLE:-This mechanical energy is used in running an electric generator which is directly coupled to the shaft of turbine

Thus mechanical energy is converted into electrical energy.



Introduction to Hydro power plant





Head race
Dam
Reservoir
Penstock
h_f
H
H_g
Vanes
Pelton wheel
h
Tail race

Basic Elements in Hydro Electric Power Plant

1.Reservoir : Reservoir is used to Store Large amount water.

2.Dam : It is Structure of considerable height built across the reservoir. It develops a reservoir to store water & Built Head for power generation.

3. Trash Rack : It is provided for preventing entry of debris like dust, dirt etc. from the dam. Because it may damage to turbine blades & Chock the flow.

4.Gate : It is provided to control the flow of water from the reservoir to the turbine.

5.Penstock : A Pipe which carries water from reservoir to turbine house is called penstock. It is a large of 1 m to 2 m in diameter made up of concrete to withstand high pressure.

6.Turbine : Turbine is used to convert Kinetic Energy to Mechanical Energy.

Head of Turbine

Gross Head (Hg)

It is the difference between head race level and tail race level when no water is flowing. It is also known as ***total head*** of the turbine

Effective Head *or* Net Head (H)

Net head or effective head is the actual head available at the inlet of the turbine.

When water is flowing from head race to the turbine, a loss of head due to friction between water and penstock occurs. Though there are other losses also such as loss due to bend, pipe fittings, loss at entrance of the penstock, etc. These all having small magnitude as compared to head loss due to friction.

Euler's Head: It is defined as energy transfer per unit weight

$H = H_g - h_f$

Where , H = Net head

Hg = Gross head

h_f = Head Loss due to Friction= $\frac{4 \times f \times L \times V^2}{D \times 2g}$ $V = \sqrt{2gH}$

Where

F= cofficient of friction of penstock

V = Velocity of Flow in Penstock

L = Length of Penstock

D = Dia. of Penstock

g = acceleration due to gravity

Efficiency of Turbine

Hydraulic Efficiency

It is the ratio of the power developed by the runner of a turbine to the power supplied by the water at the inlet of a turbine.

Since the power supplied is hydraulic, and the probable loss is between the striking jet and vane it is rightly called hydraulic efficiency.

$$\eta_h = \frac{Power\ developed\ by\ the\ runner}{Power\ supplied\ by\ the\ water\ at\ the\ inlet}$$

$$\eta_h = \frac{Runner\ Power}{Water\ Power} = \frac{R.P.}{W.P.} ----------(3.3)$$

Mechanical Efficiency

The power delivered by water to the runner of a turbine is transmitted to the shaft of the turbine.

It is the ratio of the power available at the shaft of the turbine to the power developed by the runner of a turbine.

This depends on the slips and other mechanical problems that will create a loss of energy i.e. friction.

$$\eta_m = \frac{Power\ available\ at\ the\ shaft\ of\ the\ turbine}{Power\ developed\ by\ the\ runner\ of\ a\ turbine}$$

$$\eta_m = \frac{Shaft\ Power}{Runner\ Power} = \frac{S.P.}{R.P.} ----------(3.4)$$

Overall Efficiency

It is the ratio of the power available at the shaft to the power supplied by the water at the inlet of a turbine.

$$\eta_o = \frac{Shaft\ Power}{Water\ Power}$$

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

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

$$\eta_o = \eta_m \times \eta_h ---------(3.5)$$

Volumetric Efficiency

The volume of the water striking the runner of a turbine is slightly less than the

volume of the water supplied to the turbine. Some of the volume of the water is discharged to the tail race without striking the runner of the turbine.

Thus the ratio of the volume of the water actually striking the runner to the volume of water supplied to the turbine is defined as volumetric efficiency.

$$\eta_v = \frac{Volume\ of\ water\ actually\ striking\ the\ runner}{Volume\ of\ water\ supplied\ to\ the\ turbine} ---------(3.6)$$

Classification of Hydraulic Turbines

- **According to the type of energy at inlet or the action of the water on the blade**
  - **Impulse turbine**
    - In an *Impulse turbine*, all the available energy of the water is converted into kinetic energy or velocity head by passing it through a convergent nozzle provided at the end of penstock.
    - So at the inlet of the turbine, **only kinetic energy** is available.
    - Here the pressure of water flowing over the turbine blades remains constant. (i.e. atmospheric pressure)
    - **Examples:** *Pelton wheel*, Turgo-impulse turbine, Girard turbine, Banki turbine, Jonval.
  - **Reaction turbine**
    - In a *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.
    - So at the inlet of the turbine, water possesses **kinetic energy as well as pressure energy**.
    - As the water flows through the turbine blades, the change from pressure energy to kinetic energy takes place gradually.
    - For this gradual change of pressure, the runner must be completely enclosed in an air-tight casing and the passage should be full of water.
    - The difference of pressure between the inlet and outlet of the runner is called reaction pressure, and hence these turbines are known as reaction turbine.
    - **Examples:** *Francis turbine, Kaplan turbine, Propeller turbine*, Thomson turbine,

According to the direction of flow through runner

Tangential flow turbine

In *tangential flow*, the water strikes the runner in the direction of tangent to the path of rotation of runner. *OR* The water strikes the vane/bucket along the tangent of the runner.

Example: Pelton wheel

Radial flow turbine

In *radial flow*, water flows through the turbine along the direction normal to the axis of rotation (i.e. radial direction).

A radial flow turbine is further classified as inward or outward flow depending upon whether the flow is inward from the periphery to the center or outward from center to periphery.

Example: Old Francis turbine

Axial flow turbine

In an *axial flow*, water flows along the direction parallel to the axis of rotation of the runner.

Here water flows parallel to the turbine shaft.

Examples: Kaplan turbine, Propeller turbine

Mixed flow turbine

In mixed flow, water enters the runner in the radial direction and leaves in the direction parallel to the axis of rotation (i.e. axial direction).

Example: Modern Francis turbine.

According to the head at the inlet of the turbine

High head turbine

High head turbines which operates under high head (above 250m) and requires relatively less quantity of water.

Example: Pelton wheel turbine

Medium head turbine

Medium head turbines which operate under medium head (60m to 250m) and require medium flow rate.

Example: Modern Francis turbine

Low head turbine

Low head turbines which operate under head up to 30m and require very large quantity of water.

Example: Kaplan and Propeller turbine

According to specifice speed

Low specific speed turbine

For Pelton wheel turbine with single jet,

For Pelton wheel turbine with double jet,

Medium specific speed turbine

For Francis turbine,

High specific speed turbine

Kaplan and other Propeller turbine,

Pelton Wheel

It is impulse , tangential flow, high head and law specific turbine.
It require comparatively less quantity of water.
A pelton wheel name after the American engineer lester pelton who contriute much to its development.
The energy available at inlet is only kinetic energy. therefore it is a *tangential flow impulse turbine*.

It is used with heads of more than 500 m.

A head is the distance by which the water falls before it strikes the turbine blades

The flow of water is tangential to the runner. So it is a tangential flow impulse turbine.

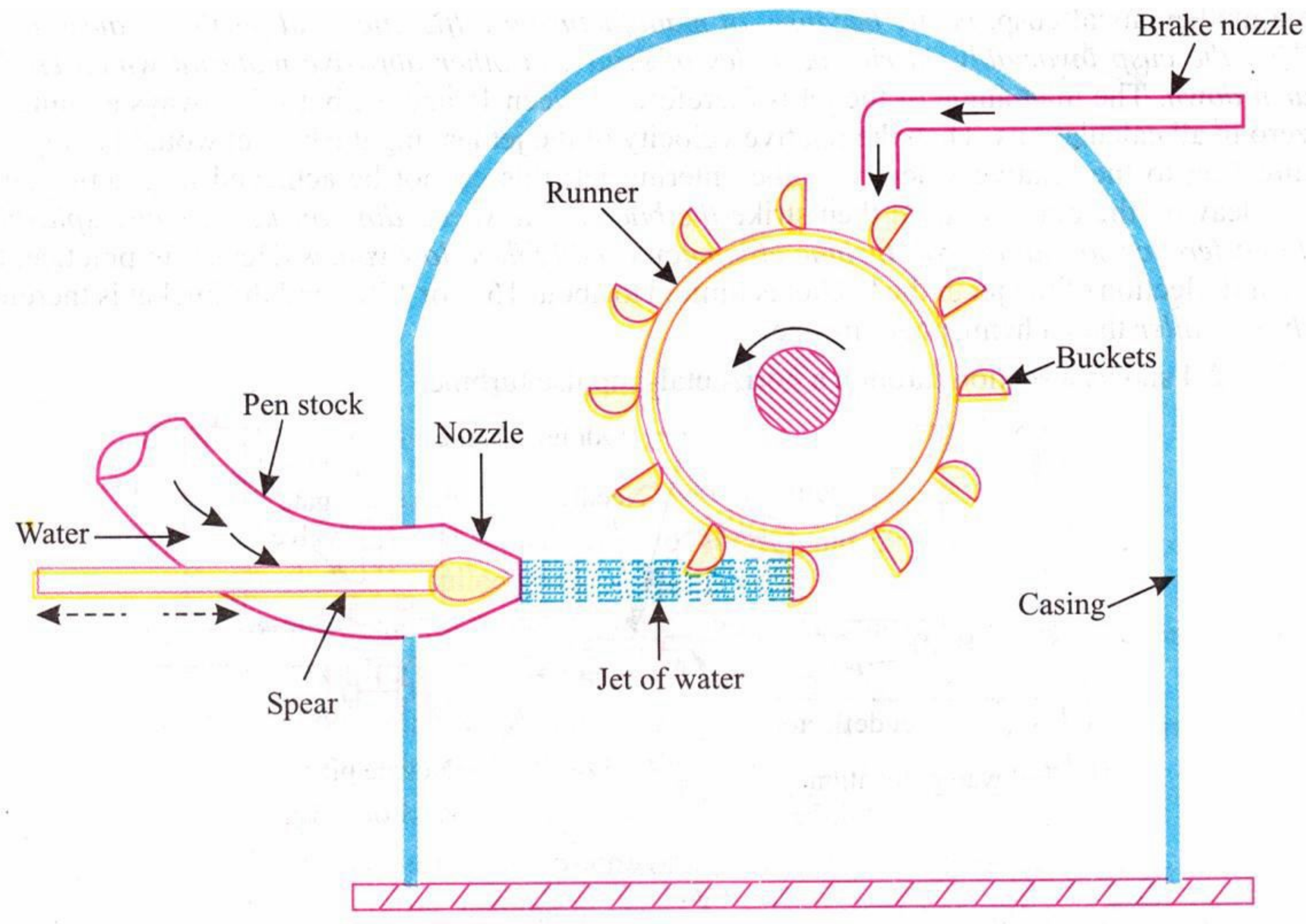
A Pelton's runner consists of a single wheel mounted on a horizontal shaft.

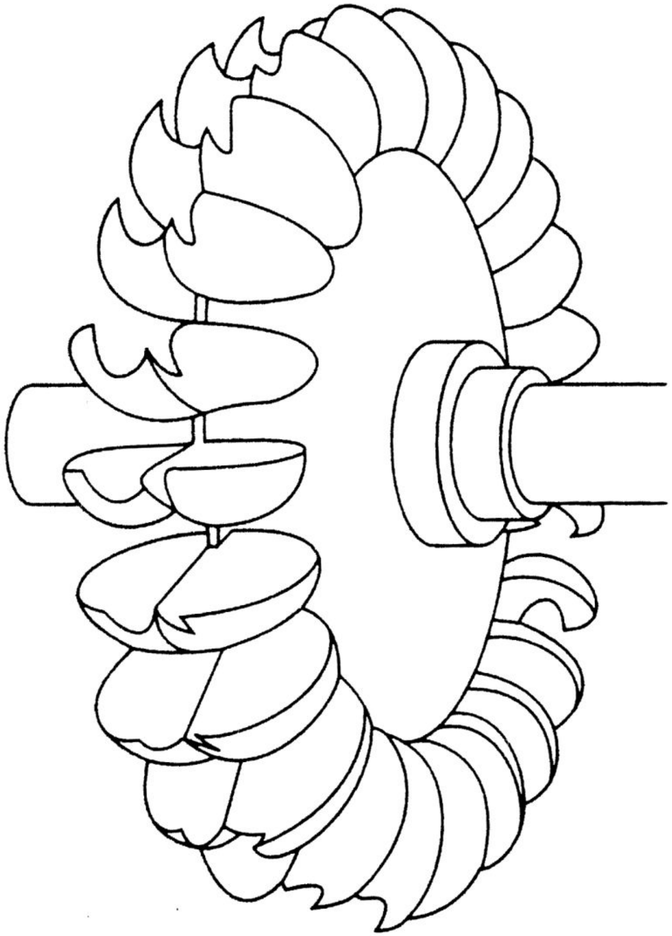
Water falls towards the turbine through a pipe called penstock and flows through a nozzle.

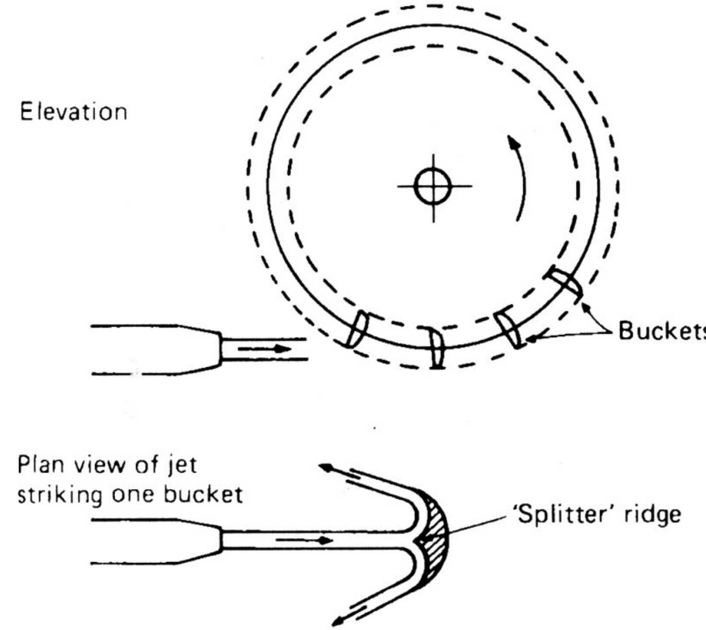
The high speed jet of water coming out from the nozzle hits the buckets (vanes) on the wheel and causes the wheel to rotate producing torque and power.

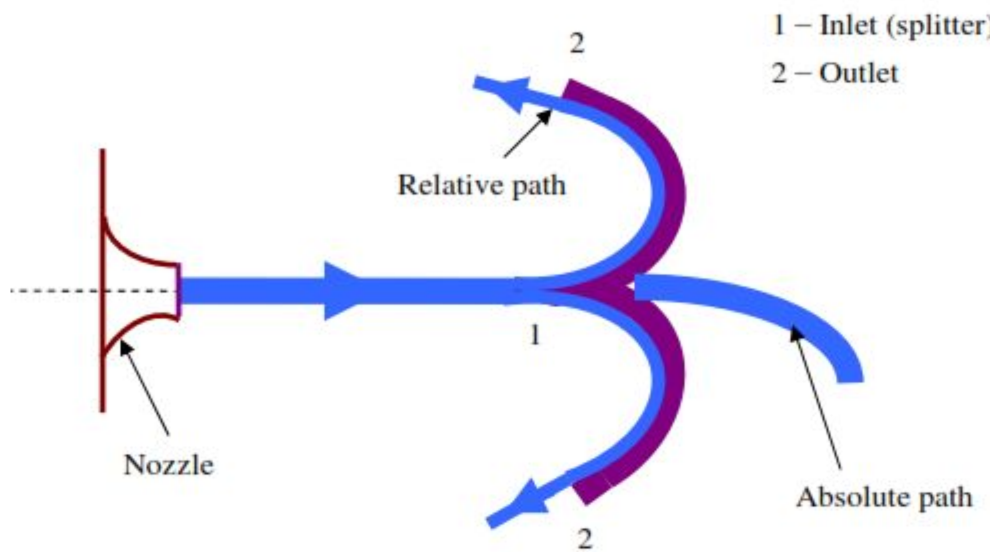
The Pelton wheel extracts energy from the impulse (momentum) of moving water as opposed to its weight like traditional overshot water wheel.

Pelton Wheel






Elevation
Buckets
Plan view of jet
striking one bucket
'Splitter' ridge


1 – Inlet (splitter)
2 – Outlet
2
Relative path
1
Nozzle
Absolute path
2

Pelton Wheel

Pelton wheel is generally used at a very high head and low discharge.

Components of Pelton Wheel

Nozzle and Flow Regulating Arrangement (Spear)

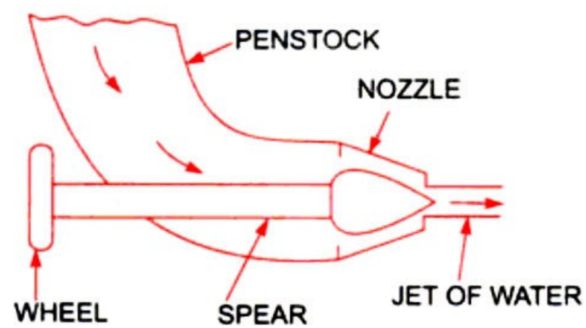
Runner and Buckets

Casing and

Breaking Jet

Nozzle and Flow Regulating Arrangement (Spear)

Depending on load fluctuations, the speed of the turbine is to be kept constant by controlling the quantity of water flowing through the nozzle.



Spear and needle arrangement with nozzle

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

The spear is a conical needle which is operated either by a hand wheel or automatically by governor in an axial direction depending upon the size of the unit.

When the spear is pushed forward into the nozzle, the amount of water striking the runner is reduced. On the other hand, if the spear is pushed back, the amount of water striking the runner increases.

•Runner and Buckets

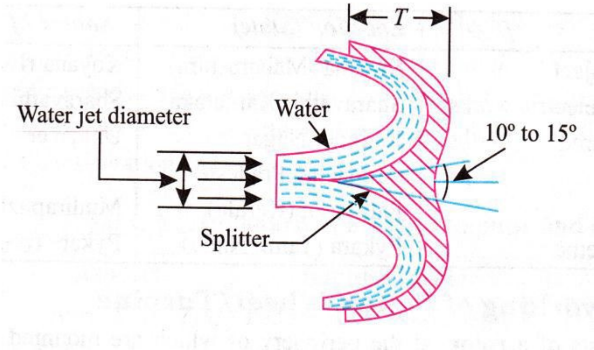
•It consists of a circular disc, on the periphery of which a number of buckets evenly spaced are fixed.

•The shape of the buckets is of a double hemispherical cup or bowl. Each bucket is divided into two symmetrical parts by a dividing wall which is known as **splitter**.

•The jet of water strikes on the splitter. The splitter divides the jet into two equal parts and the jet comes out at the outer edge of the bucket.

•The buckets are shaped in such a way that the jet gets deflected through 160° or 170°. Maximum work is obtained if the jet is deflected through 180° i.e. the bucket is semicircular.

•If semicircular bucket is used, an outgoing jet may strikes to the next incoming bucket and hence opposes the motion of the rotor. Hence the angle of jet deflection is generally kept 160° to 170°.



Casing

The function of the casing is to prevent the splashing of the water and to discharge water to the tailrace.

It also acts as a safe-guard against accidents.

Material: Cast iron or fabricated steel plates.

The casing of the Pelton wheel does not perform any hydraulic function.

Breaking Jet

When the nozzle is completely closed by moving the spear in the forward direction, the amount of water striking the runner reduces to zero.

But the runner due to inertia goes on revolving for a long time.

To stop the runner in a shorter time, a small nozzle is provided which directs the jet of water on the back of the vanes as shown in Fig This jet of water is called **breaking jet**.

- **Working Principle of Pelton Turbine**
- High speed water jets emerging from the nozzles (obtained by expanding high pressure water to the atmospheric pressure in the nozzle) strike a series of spoon-shaped buckets mounted around the edge of the pelton wheel. High pressure water can be obtained from any water body situated at some height or streams of water flowing down the hills.
- As water flows into the bucket, the direction of the water velocity changes to follow the contour of the bucket. These jets flow along the inner curve of the bucket and leave it in the direction opposite to that of incoming jet. When the water-jet contacts the bucket, the water exerts pressure on the bucket and the water is decelerated as it does a "u-turn" and flows out the other side of the bucket at low velocity.
- The change in momentum (direction as well as speed) of water jet produces an impulse on the blades of the wheel of Pelton Turbine. This "impulse" does work on the turbine and generates the torque and rotation in the shaft of Pelton Turbine.

- To obtain the optimum output from the Pelton Turbine the impulse received by the blades should be maximum. For that, change in momentum of the water jet should be maximum possible. This is obtained when the water jet is deflected in the direction opposite to which it strikes the buckets and with the same speed relative to the buckets.
- For maximum power and efficiency, the turbine system is designed such that the water-jet velocity is twice the velocity of the bucket. A very small percentage of the water's original kinetic energy will still remain in the water. However, this allows the bucket to be emptied at the same rate at which it is filled, thus allowing the water flow to continue uninterrupted.
- Often two buckets are mounted side-by-side, thus splitting the water jet in half (see photo). The high speed water jets emerging form the nozzles strike the buckets at splitters, placed at the middle of the buckets, from where jets are divided into two equal streams.
- This balances the side-load forces on the wheel, and helps to ensure smooth, efficient momentum transfer of the fluid jet to the turbine wheel.
- Because water and most liquids are nearly incompressible, almost all of the available energy is extracted in the first stage of the hydraulic turbine.
- Therefore, Pelton wheels have only one turbine stage, unlike gas turbines that operate with compressible fluid.

Net Heat avaiable at turbine

H_g = Gross Head

h_f = Head Loss due to Friction

$$= \frac{4 \times f \times L \times V^2}{D \times 2g} \qquad V = \sqrt{2gH}$$

Where

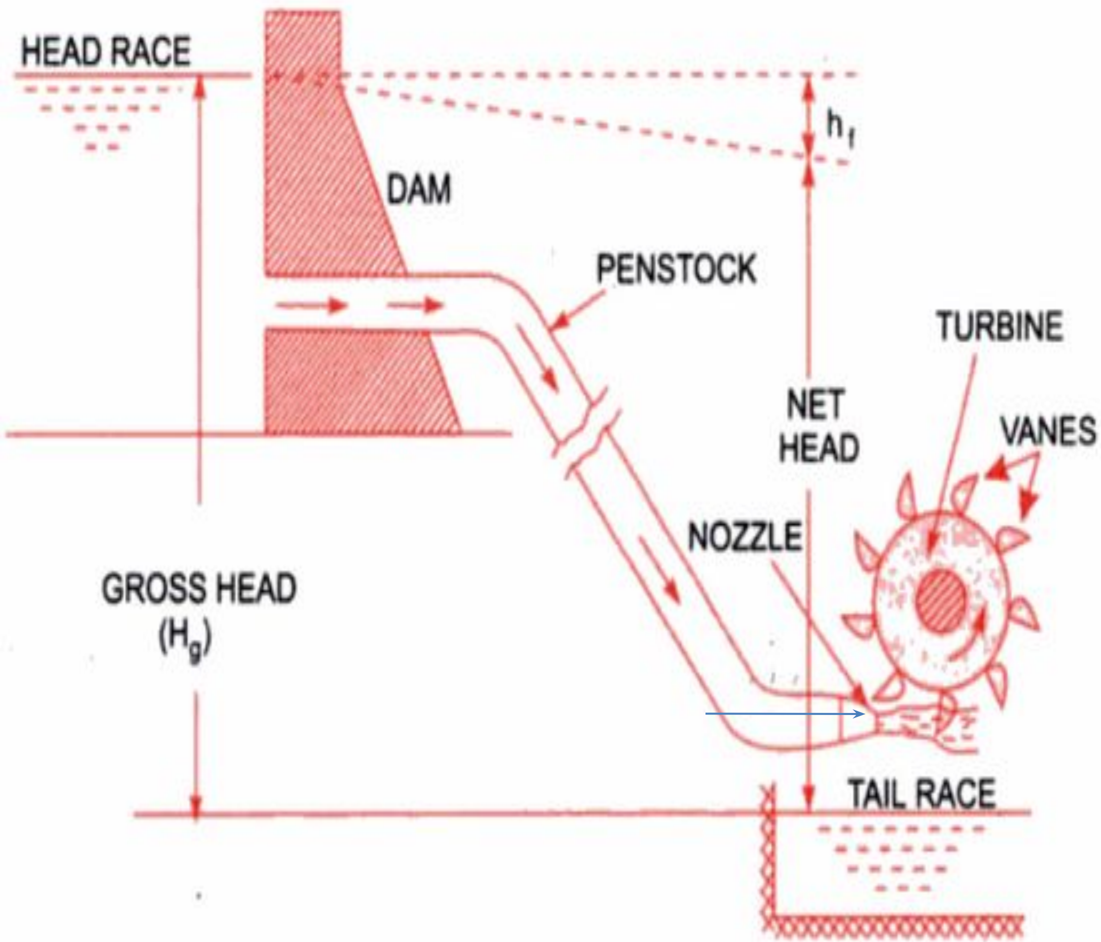
V = Velocity of Flow in Penstock

L = Length of Penstock

D = Dia. of Penstock

H = Net Head

$= H_g - h_f$



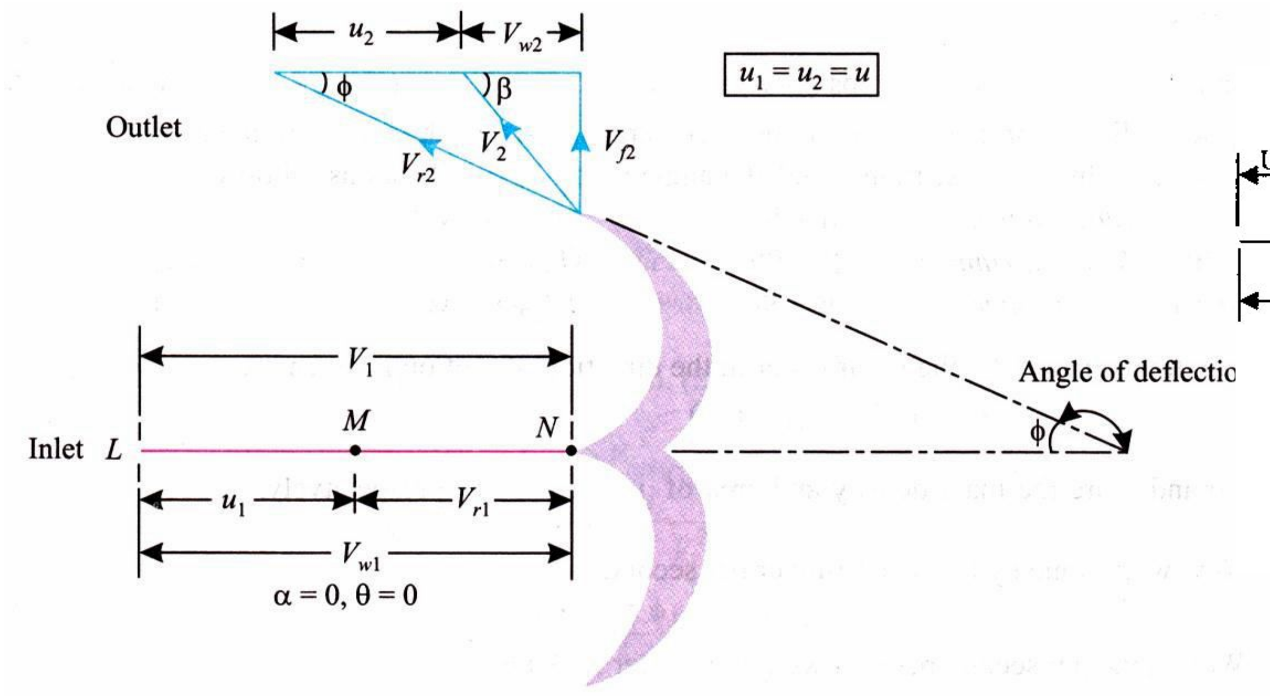
Layout of a hydro-eletric power plant.

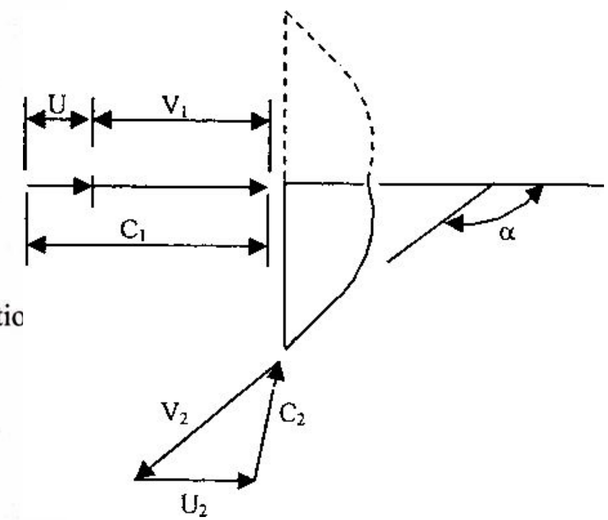
• Velocity Triangles, Work done and Efficiency of Pelton Wheel

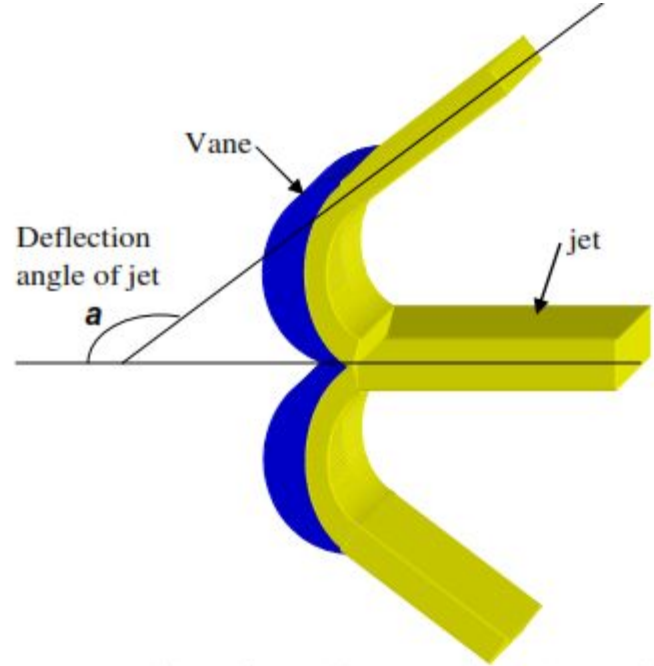
•The jet of water from the nozzle strikes the bucket at the splitter, which splits up the jet into two parts.

•These parts of the jet, glides over the inner surfaces and comes out at the outer edge of the bucket.

•The splitter is the inlet tip and outer edge of the bucket is the outlet tip of the bucket.







The velocity Δ at inlet will be a straight line,

$$V_{w1} = V_1$$

$$V_{r1} = V_1 - u_1 = V_1 - u$$

The velocity Δ at outlet,

$$V_{r1} = V_{r2}, \qquad V_{w2} = V_{r2}\cos\emptyset - u_2$$

The force exerted by jet of water in the direction of motion,

$$F_x = \rho a V_1[V_{w1} + V_{w2}]$$

Mass of water striking $= \rho a V_1$

Workdone by the jet on the runner / sec $= F_x \text{ x } u$ $= \rho a V_1[V_{w1} + V_{w2}] \text{x } u, \quad Nm/s$

Power given to the runner by jet $= \dfrac{\rho a V_1[V_{w1} + V_{w2}] \text{x } u}{1000}$, KW

$$\text{Workdone/s per unit weight of water} = \frac{\rho a V_1 [V_{w1} + V_{w2}] \text{x u}}{Weight\ of\ water\ striking/s}$$

$$= \frac{\rho a V_1 [V_{w1} + V_{w2}] \text{x u}}{\rho a V_1 \text{ x g}}$$

$$= \frac{[V_{w1} + V_{w2}] \text{x u}}{\text{g}}$$

$$\text{Hydraulic efficiency, } \eta_h = \frac{Workdone/s}{K.E\ of\ jet/s}$$

$$K.E\ of\ jet/s = \frac{1}{2} m V^2$$

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

$$= \frac{1}{2} \rho a {V_1}^3$$

$$\text{Hydraulic efficiency, } \eta_h = \frac{\rho a V_1 [V_{w1} + V_{w2}] \text{x u}}{\frac{1}{2} \rho a V_1{}^3}$$

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

$$\eta_h = \frac{[V_1 + (V_1 - u)\cos\emptyset - u] \text{ x u}}{V_1{}^2}$$

$$\eta_h = \frac{2(V_1 - u)\,(1 + \cos\emptyset) \text{x u}}{V_1{}^2}$$

he efficiency is maximum when $\frac{d\,(\eta_h)}{du} = 0$

$$\frac{d}{du}\left[\frac{2(V_1 - u)\,(1 + \cos\emptyset) \text{x u}}{V_1{}^2}\right] = 0$$

$$\frac{(1 + \cos\emptyset)}{V_1{}^2}\;\frac{d}{du}[2u(V_1 - u] = 0$$

$$\frac{d}{du}[2uV_1 - 2u^2] = 0$$

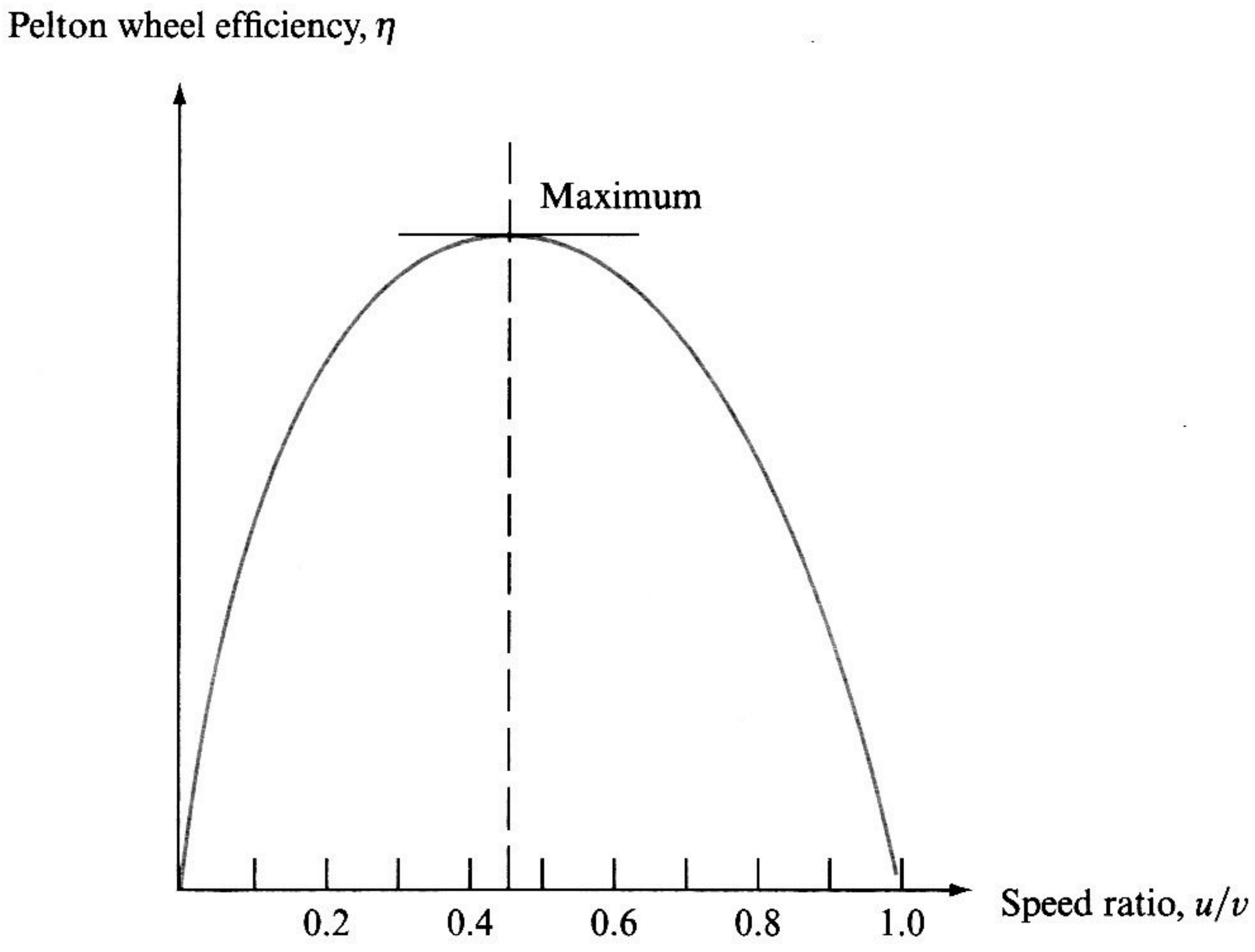
$$u = \frac{V_1}{2}$$

Expression for maximum efficiency of pelton wheel

$$\eta_{h_{max}} = \frac{2(V_1 - \frac{V_1}{2})\,(1 + \cos\emptyset)\mathrm{x}\,\frac{V_1}{2}}{{V_1}^2}$$

$$\eta_{h_{max}} = \frac{(1 + \cos\emptyset)}{2}$$

velocity at maximum hydraulic efficiency : $V = u1/2$



Design of Pelton wheel

Design of Pelton wheel

Design of the Pelton wheel means following data is to be determine:-

a) Diameter of the jet (d)

b) Diameter of wheel (D)

c) Width of bucket = 5 x d)

d) Depth of the bucket = 1.2 x d

e) No of buckets on the wheel = total rate of flow through the turbine/ rate of flow of water through a single jet of water through a single jet.

Design Aspects of Pelton Wheel

For design aspect following points should be considered:

1. The **velocity of jet** (V_1) at inlet of the turbine,

$$V_1 = C_v\sqrt{2gH} - - - - - - - - -(3.10)$$

Where, $C_v = Coefficient\ of\ velocity \cong 0.98\ to\ 0.99$

2. The **velocity of wheel** (u),

$$u = \varphi\sqrt{2gH} - - - - - - - - -(3.11)$$

Where, $\varphi = speed\ ratio \cong 0.43\ to\ 0.48$

3. The **angle of deflection** of the jet through bucket is taken at 165° (average of 160° to 170°), if no angle of deflection is given.

4. The **mean diameter or pitch diameter** (D) of the Pelton wheel is given by,

5. **Jet ratio** (m)**:** It is the ratio of pitch diameter (D) to diameter of jet (d).

$$m = \frac{D}{d} \quad (\cong 12\ in\ most\ of\ the\ cases) - - - - - - - - -(3.12)$$

6. **No. of buckets** (Z) on a runner is given by,

$$Z = 15 + \frac{D}{2d}$$

$$\therefore Z = 15 + 0.5m - - - - - - - - -(3.13)$$

7. **No. of jets:** It is obtained by dividing the total rate of flow through the turbine (Q) by the rate of flow of water through a single jet (q).

$$No.of\ jets = \frac{Q}{q} - - - - - - - - -(3.14)$$

8. **Working proportions for buckets:**
    1. Width of the bucket $= 3d\ to\ 5d \cong 5d$
    2. Depth of the bucket $= 0.8d\ to\ 1.2d \cong 1.2d$

- Size of bucket means width and depth of the buckets.

Working Design Proportions

8. ***Size of the buckets:*** The length, width and depth of buckets in terms of diameter of jet 'd' is shown in Fig. 5.10.

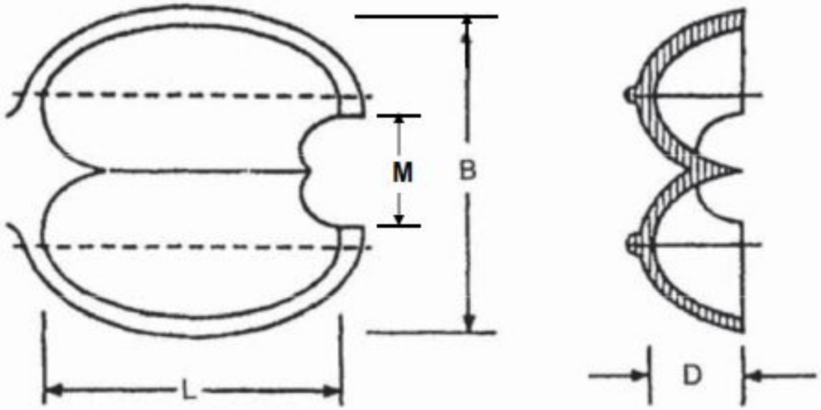
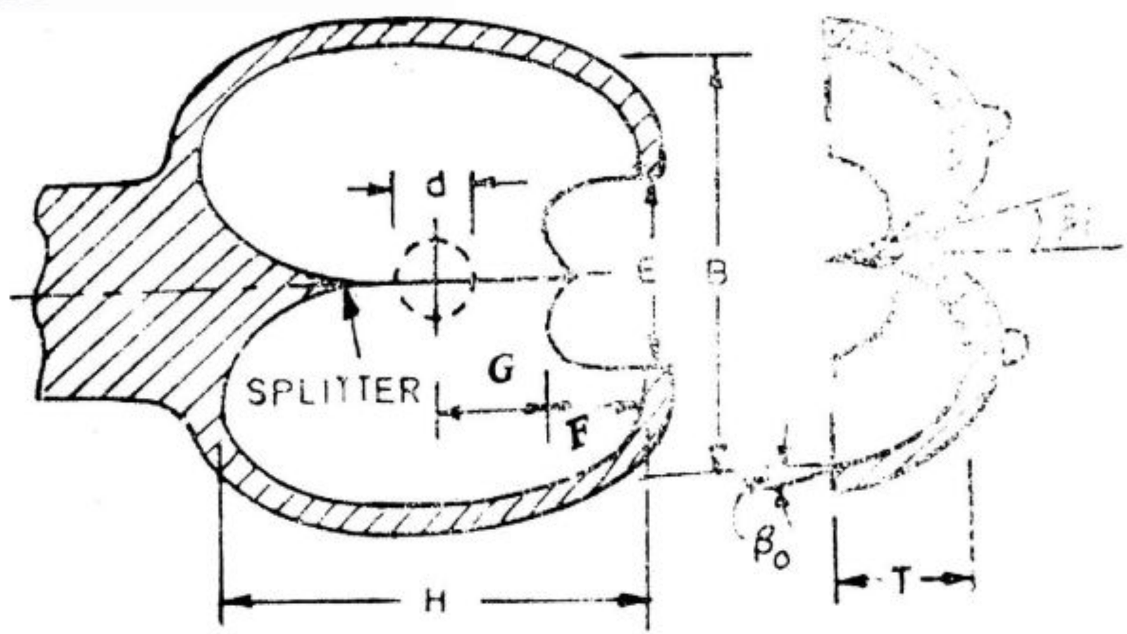
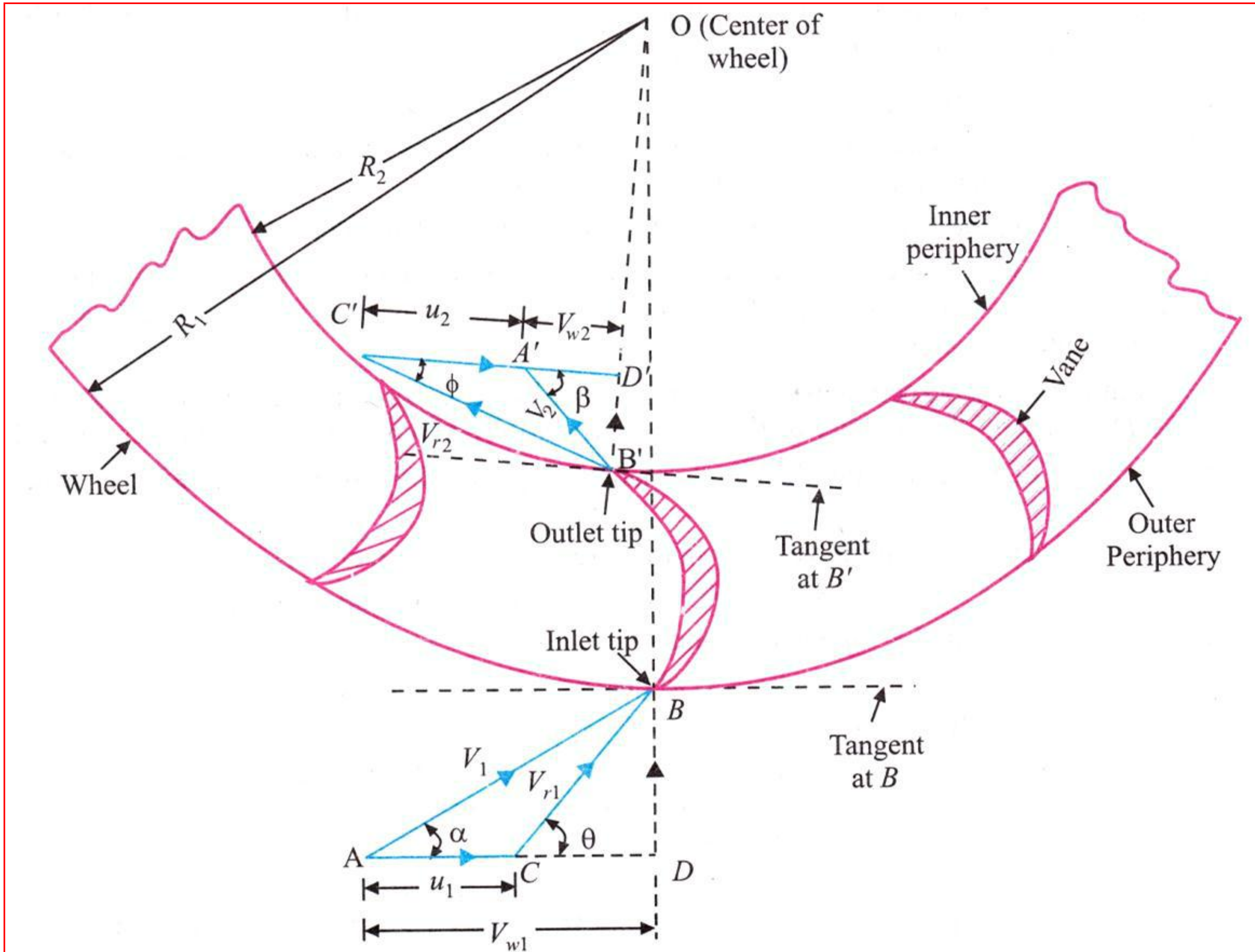


Fig. 5.10 Dimensions of bucket.

- Radial length of bucket L = 2 to 3d
- Axial width of bucket B = 3 to 5d
- Depth of bucket D = 0.8 to 1.2d



Force Exerted by the Jet of Water on a Series of Radial Curved Vanes



For a radial curved vane, the radius of the vane at inlet and outlet is different and hence the tangential velocities of the radial vane at inlet and outlet will not be equal.

Consider a series of radial curved vanes mounted on a wheel as shown in Fig. The jet of water strikes the vanes and the wheel starts rotating at constant angular speed.

- Let,

  R_1 = Radius of wheel at inlet of the vane

  R_2 = Radius of wheel at outlet of the vane

  ω = Angular speed of the wheel

  Then,

$$u_1 = \omega R_1 \text{ and } u_2 = \omega R_2$$

- The *mass of water striking per second* for a series of vanes = The mass of water coming out from nozzle per sec = $\rho a V_1$

  Where,

  a = Area of jet, and

  V_1 = Velocity of jet

- *Momentum of water striking the vanes* in the tangential direction per sec at inlet = mass of water striking per sec X component of V_1 in the tangential direction

$$\therefore Momentum\ of\ water\ at\ inlet\ per\ sec = \rho a V_1 \times V_{w1} \qquad (\because V_{w1} = V_1 \cos\alpha)$$

- Similarly,

  Momentum of water at outlet per sec = $\rho a V_1$ X component of V_2 in the tangential direction

$$\therefore Momentum\ of\ water\ at\ outlet\ per\ sec = \rho a V_1 \times (-V_2 \cos\beta)$$

$$\therefore Momentum\ of\ water\ at\ outlet\ per\ sec = -\rho a V_1 \times V_{w2} \qquad (\because V_{w2} = V_2 \cos\beta)$$

- Now *angular momentum,*

$$Angular\ momentum\ per\ \sec\ at\ inlet = Momentum\ at\ inlet \times Radius\ at\ inlet$$

$$= \rho a V_1 \times V_{w1} \times R_1$$

$$Angular\ momentum\ per\ \sec\ at\ outlet = Momentum\ at\ outlet \times Radius\ at\ outlet$$

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

- *Torque exerted by the water* on the wheel,

$$T = Rate\ of\ change\ of\ angular\ momentum$$

$$T = [Initial\ angular\ momentum\ per\ sec - Final\ angular\ momentum\ per\ sec]$$

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

$$\therefore T = \rho a V_1 [V_{w1} R_1 + V_{w2} R_2] --------(2.31)$$

- *Work done per sec* on the wheel,

$$WD/sec = Torque \times Angular\ velocity$$

$$\therefore WD/sec = T \times \omega$$

$$\therefore WD/sec = \rho a V_1[V_{w1}R_1 + V_{w2}R_2] \times \omega$$

$$\therefore WD/sec = \rho a V_1[V_{w1}R_1\omega + V_{w2}R_2\omega]$$

$$\therefore WD/sec = \rho a V_1[V_{w1}u_1 + V_{w2}u_2] - - - - - - - - - -(2.32)$$

- Equation 2.32 is valid only when, $\beta < 90$. If the angle β is an obtuse angle ($\beta > 90$) then,

$$WD/sec = \rho a V_1[V_{w1}u_1 - V_{w2}u_2] - - - - - - - - - -(2.33)$$

- In general,

$$WD/sec = \rho a V_1[V_{w1}u_1 \pm V_{w2}u_2] - - - - - - - - - -(2.34)$$

- If the discharge is radial at the outlet then, $\beta = 90°$ and hence $V_{w2} = 0$,

$$\therefore WD/sec = \rho a V_1[V_{w1}u_1] - - - - - - - - - -(2.35)$$

– *Efficiency* of the radial curved vanes,

$$\eta = \frac{Work\ done\ per\ second}{Kinetic\ energy\ per\ second}$$

$$\therefore \eta = \frac{\rho a V_1 [V_{w1} u_1 \pm V_{w2} u_2]}{\frac{1}{2}(\dot{m}) V_1^2}$$

$$\therefore \eta = \frac{\rho a V_1 [V_{w1} u_1 \pm V_{w2} u_2]}{\frac{1}{2}(\rho a V_1) V_1^2}$$

$$\therefore \eta = \frac{2[V_{w1} u_1 \pm V_{w2} u_2]}{V_1^2} \text{---------}(2.36)$$

Reaction Turbine

In reaction turbine, water at the inlet of the turbine possesses kinetic energy as well as pressure energy. As water flows through runner, a part of pressure energy goes on changing into kinetic energy.

Thus the water through runner is under pressure and the runner is completely enclosed in an air-tight casing.

Casing and the runner is always full of water.

Classification of reaction turbine

Radial Flow turbine

Inward radial flow reaction turbine

(Water flows from outward to inward)

Outward radial flow reaction turbine

(Water flows from inward to outward)

Mixed flow or Francis turbine

(Water enters radially but leaves axially)

Axial flow turbine (Water enters and leaves axially)

I. Kaplan turbine:- Runner blades are adjustable

II. Propeller turbine:- Runner blades are fixed

Main Components of a Radial Flow Reaction Turbine

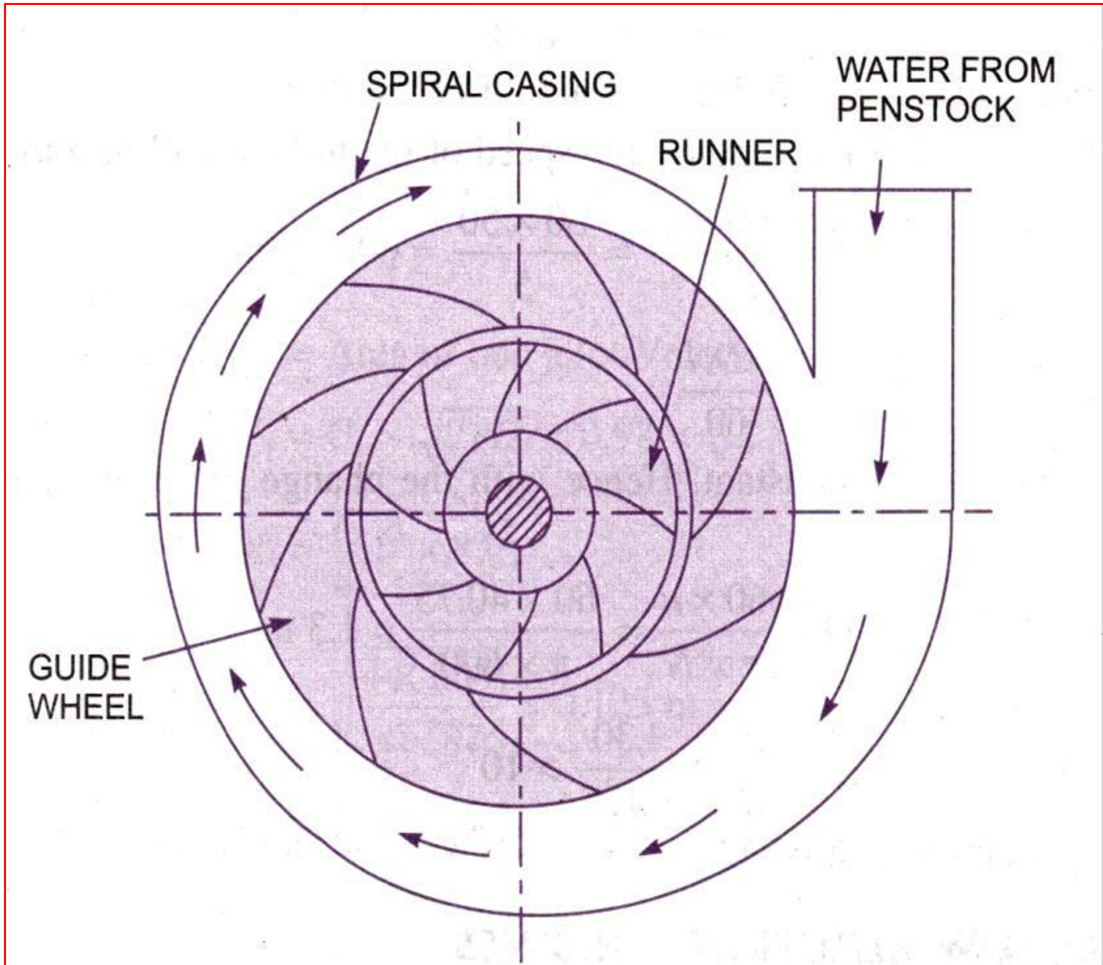
There are many components used in radial flow reaction turbine but the main components of radial flow reaction turbine are:

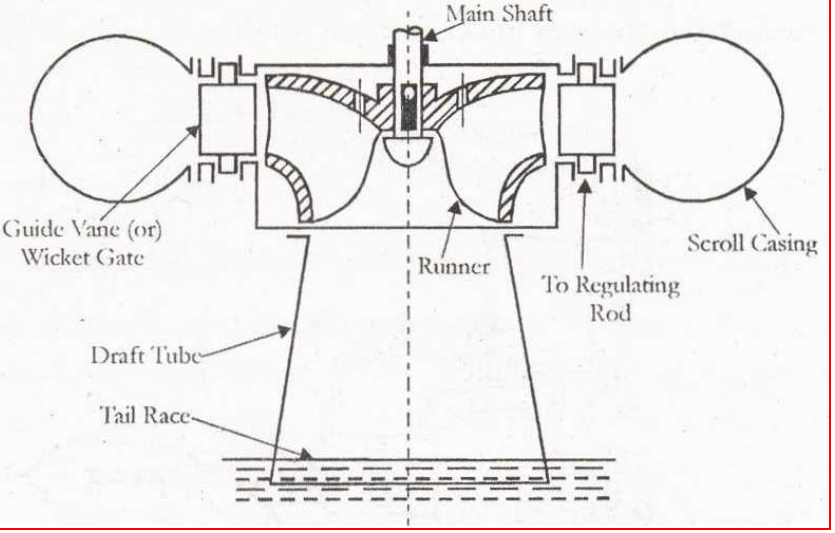
Casing

Guide Mechanism

Runner and

Draft tube




Main Shaft
Guide Vane (or)
Wicket Gate
Runner
To Regulating
Rod
Scroll Casing
Draft Tube
Tail Race

1. Casing

In case of reaction turbine, **casing and runner are always full of water.**

The cross-section area of this casing decreases uniformly along the circumference to keep the fluid velocity constant in magnitude along its path towards the guide vane.

This is so because the rate of flow along the fluid path in the volute decreases due to continuous entry of the fluid to the runner through the openings of the guide vanes.

2. Guide Mechanism or Guide Blades

It is a stationary circular wheel. Guide vanes are fixed on guide mechanism between two rings in form of wheel.

The guide vanes allow the water to strike the vanes fixed on the runner without shock at inlet.

The quantity of water passing through the guide blades depends on the position of the guide vanes

3. Runner

It is a circular wheel on which a series of radial curved vanes are fixed.

Surface of the vanes are made very smooth.

The radial curved vanes are so shaped that the water enters and leaves the runner without shock.

Material: Cast steel, Cast iron or Stainless steel.

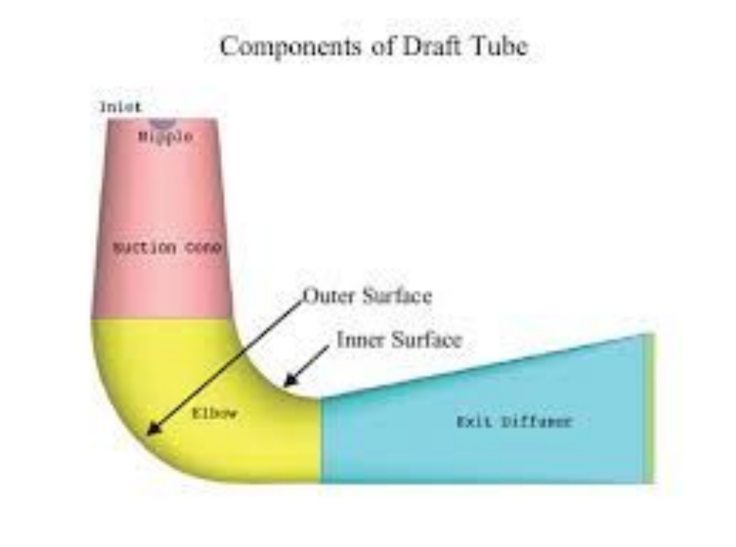
Runner is keyed to the shaft.

4. Draft Tube

The pressure at the exit of the runner of a reaction turbine is generally less than atmospheric pressure. Hence water at exit cannot be directly discharged to the tail race.

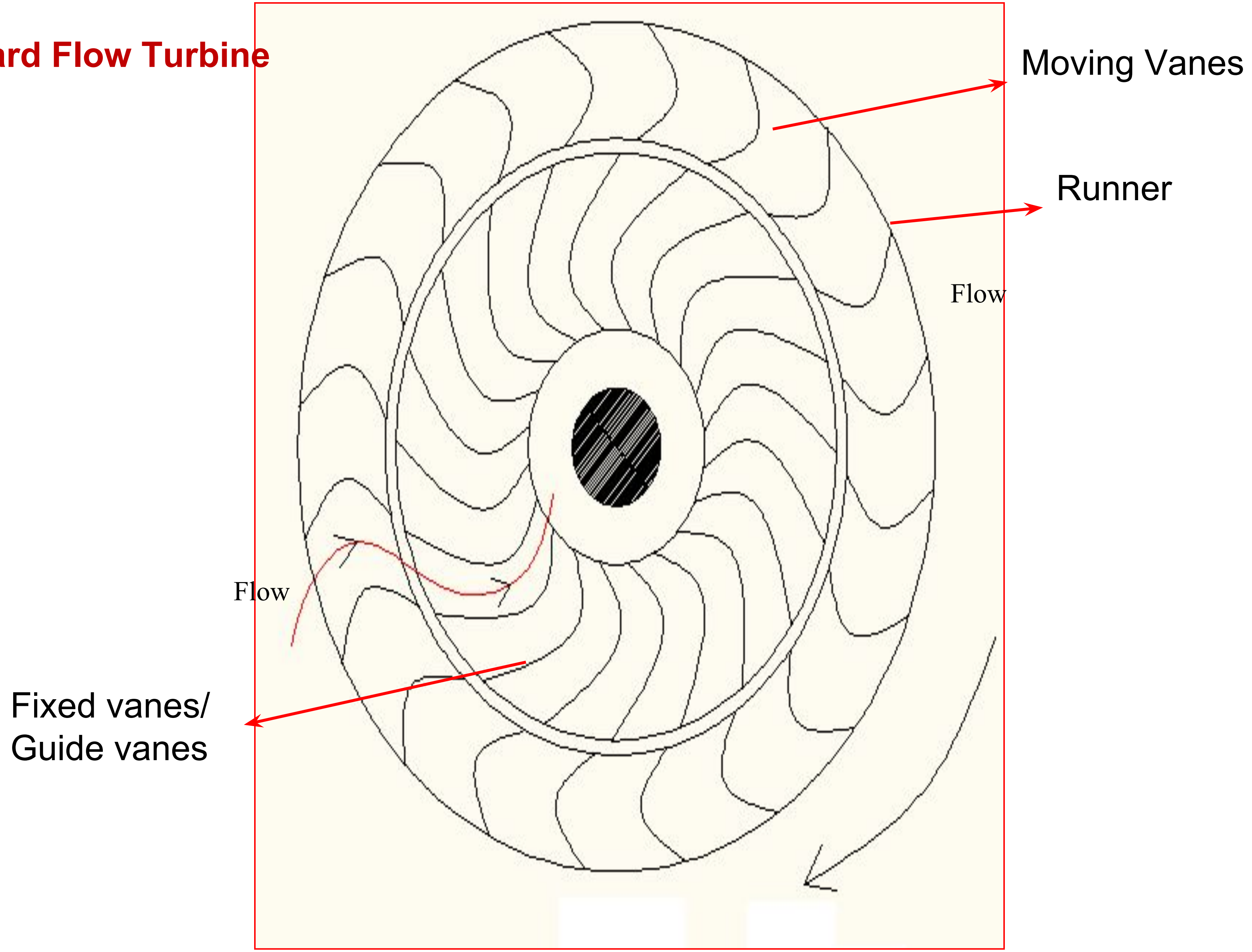
A tube or pipe of gradually increasing area is used for discharging water from the exit of the turbine to the tail race.

This tube of increasing area is called draft tube.

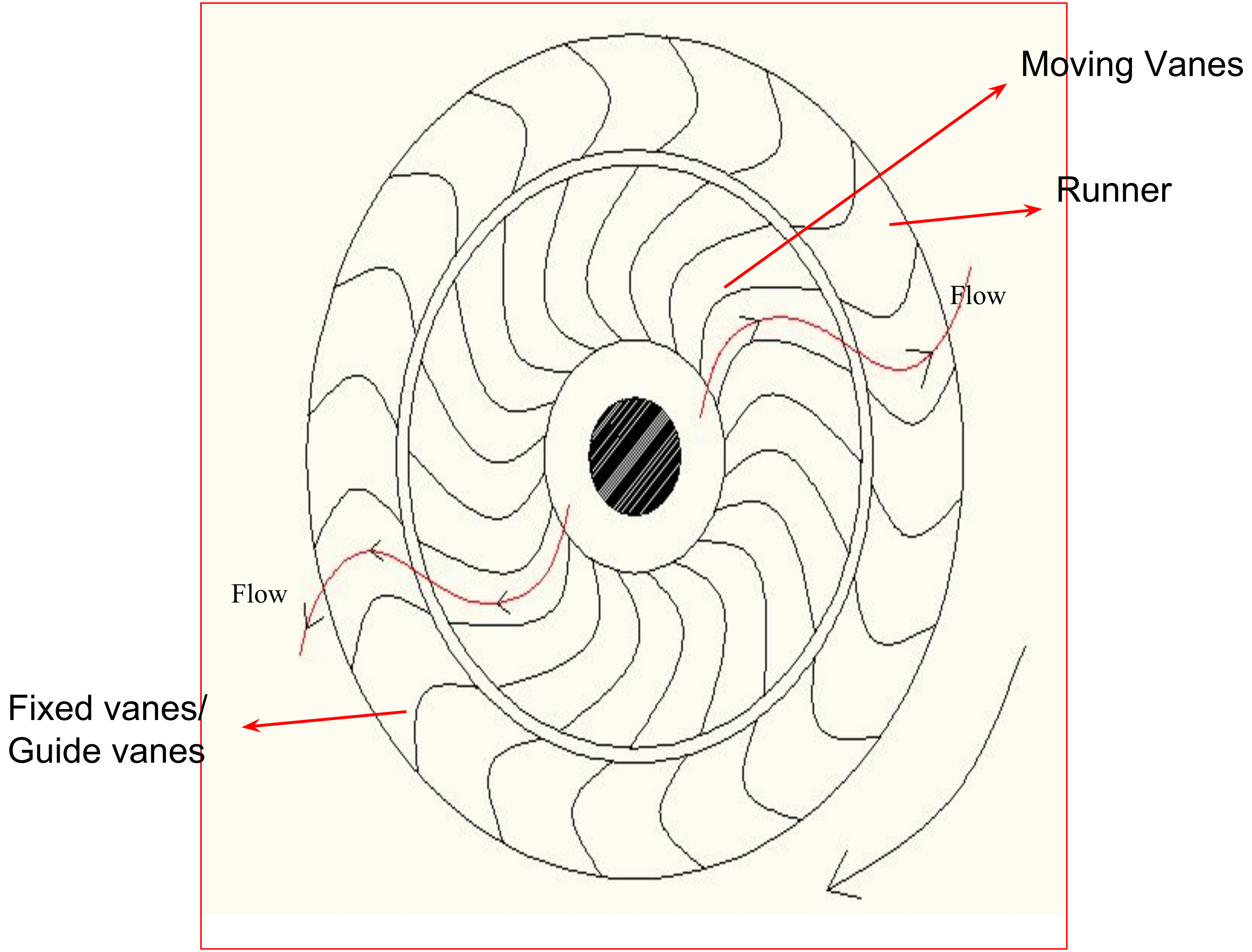


Inward and Outward Radial Flow Reaction Turbine

Inward Flow Turbine



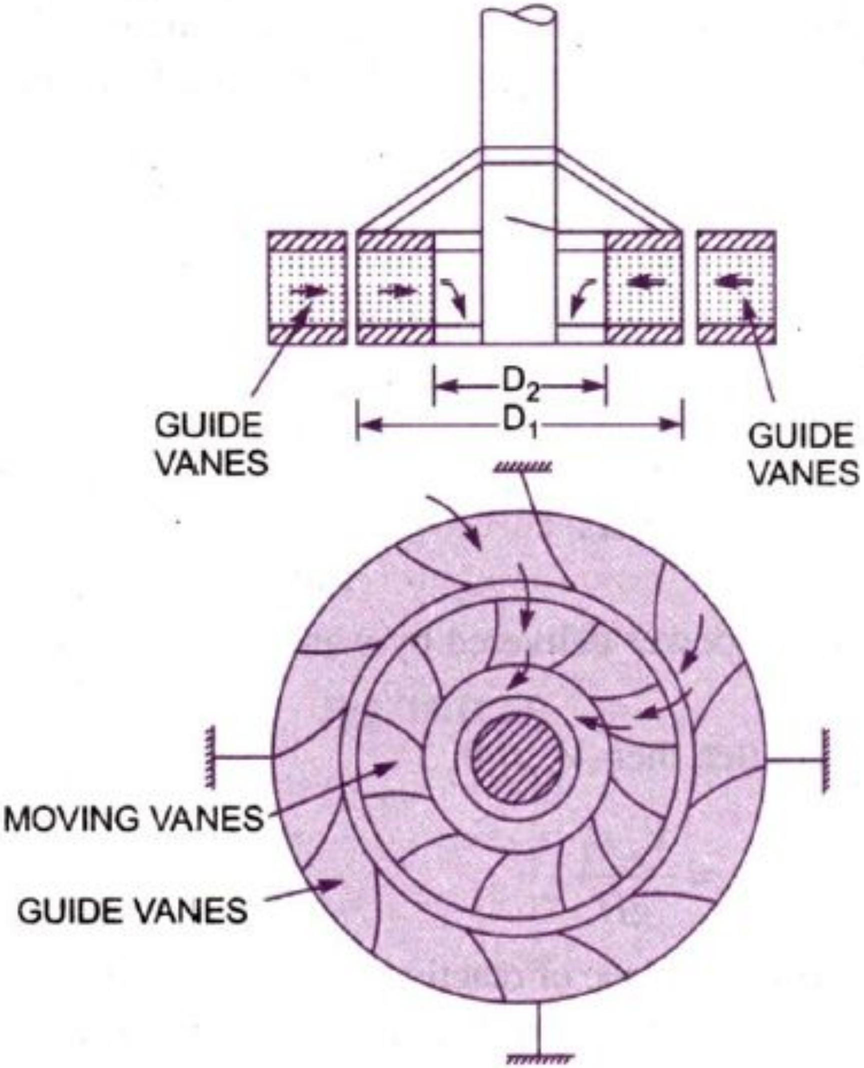
Outward Flow Turbine

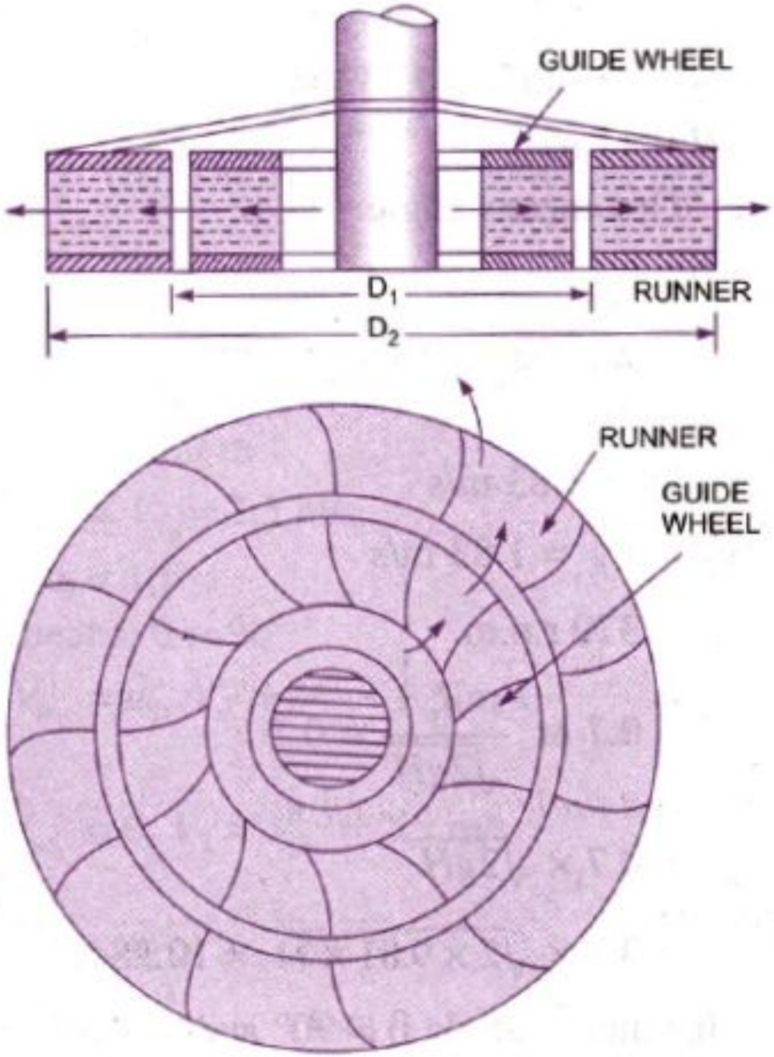


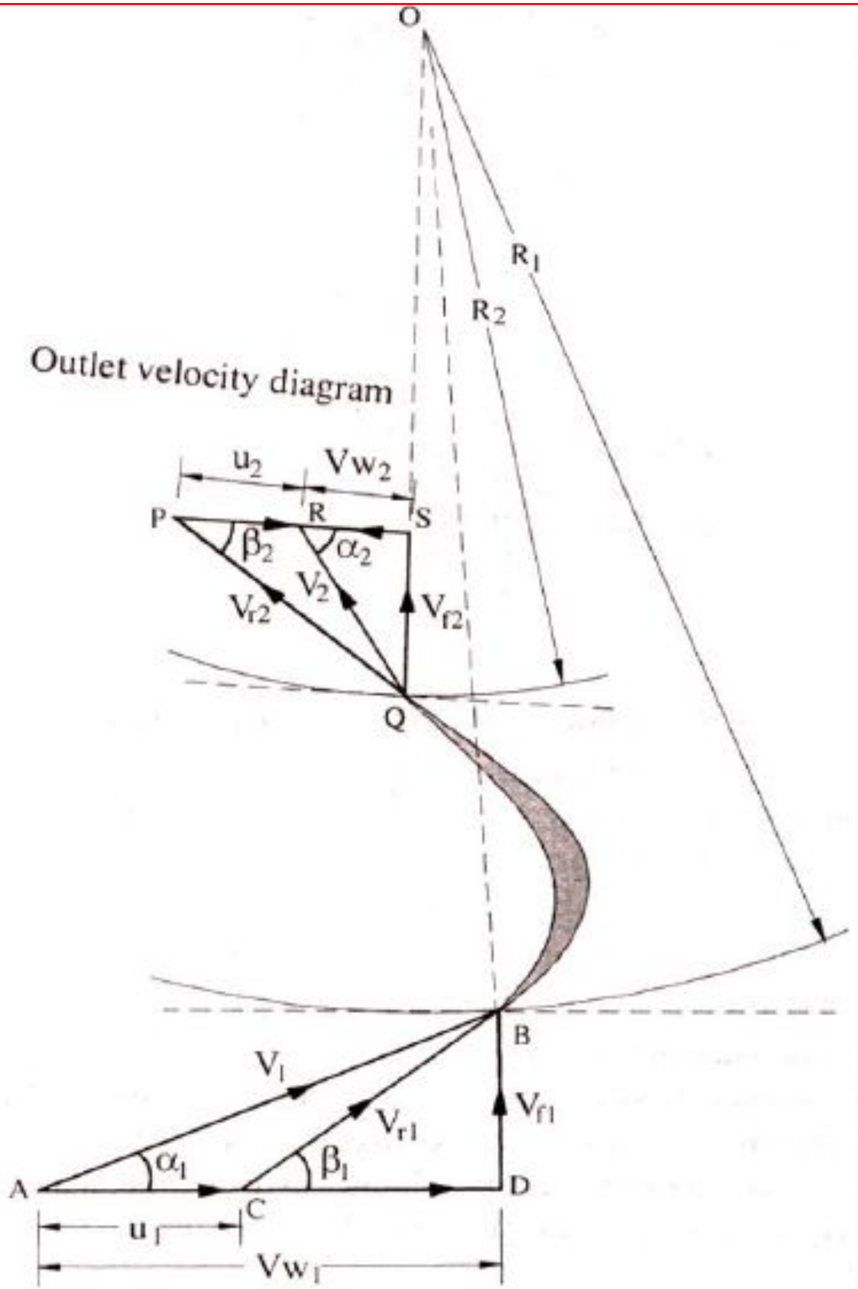
Inward and Outward Radial Flow Reaction Turbine

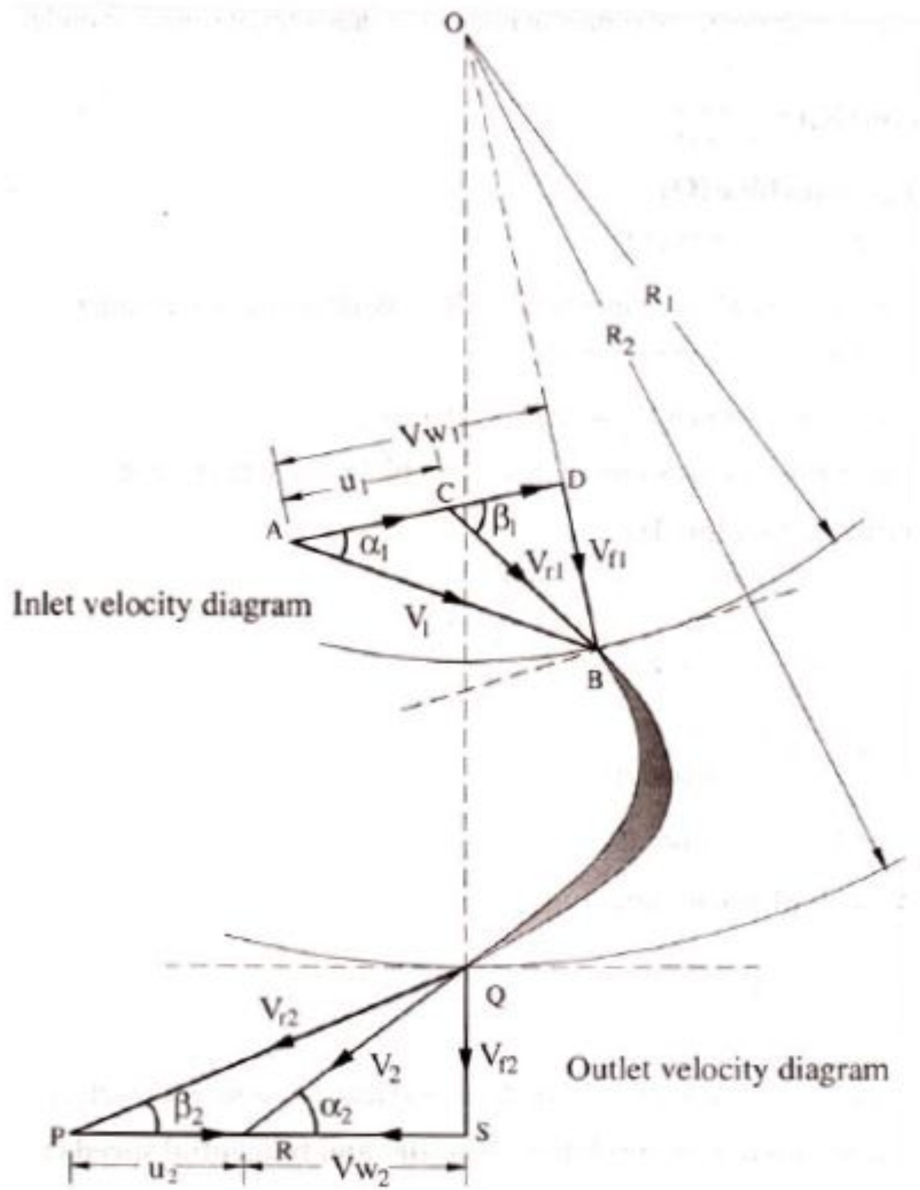
Inward Radial Flow Reaction Turbine	Outward Radial Flow Reaction Turbine
Water enters at the outer periphery, flows inward and towards the center of the turbine and discharges at the inner periphery.	Water enters at the inner periphery, flows outward and discharges at the outer periphery.
The outer diameter of the runner is inlet and the inner diameter is the outlet. $$\therefore D_1 > D_2$$ And hence, $$u_1 > u_2$$	The inner diameter of the runner is inlet and the outer diameter is the outlet. $$\therefore D_1 < D_2$$ And hence, $$u_1 < u_2$$

Inward and Outward Radial Flow Reaction Turbine






O
R1
R2
Outlet velocity diagram
u2
Vw2
P
R
S
β2
α2
Vr2
V2
Vf2
Q
B
V1
Vr1
Vf1
α1
β1
A
C
D
u1
Vw1


O
R1
R2
Vw1
u1
C
D
A
α1
β1
Vr1
Vf1
Inlet velocity diagram
V1
B
Q
Vr2
V2
Vf2
Outlet velocity diagram
β2
α2
P
R
S
u2
Vw2

Work done and hydraulic efficiency

(Work done and hydraulic efficiency are same for both inward and outward flow reaction turbines)

Work done per sec,

$$WD/sec = \rho a V_1 [V_{w1}u_1 \pm V_{w2}u_2] ---------(3.15)$$

Work done per unit weight,

$$WD/sec\ per\ unit\ weight = \frac{1}{g}[V_{w1}u_1 \pm V_{w2}u_2] ---------(3.16)$$

Hydraulic efficiency,

$$\eta_h = \frac{Runner\ Power}{Water\ Power}$$

$$\eta_h = \frac{\dot{m}\ (V_{w1}u_1 \pm V_{w2}u_2)}{\rho g Q H} = \frac{\rho Q\ (V_{w1}u_1 \pm V_{w2}u_2)}{\rho g Q H}$$

$$\eta_h = \frac{(V_{w1}u_1 \pm V_{w2}u_2)}{gH} ---------(3.17)$$

Francis turbine

A Francis turbine is:

It is inward flow R T who discharge is radial at outlet.

Mixed Flow Turbine:

Water enters radially and leaves axially to the direction of shaft.

Reaction Turbine:

At the inlet of the turbine both kinetic as well as pressure energy is available .It is generally operated under medium head and medium flow rate.

It is designed by an American engineer J. B. Francis in 1849.

<u>components of Francis Turbine</u>

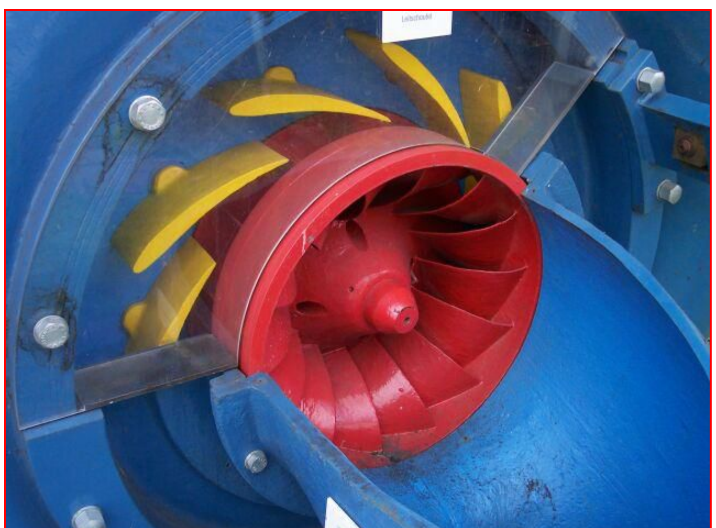
Penstock

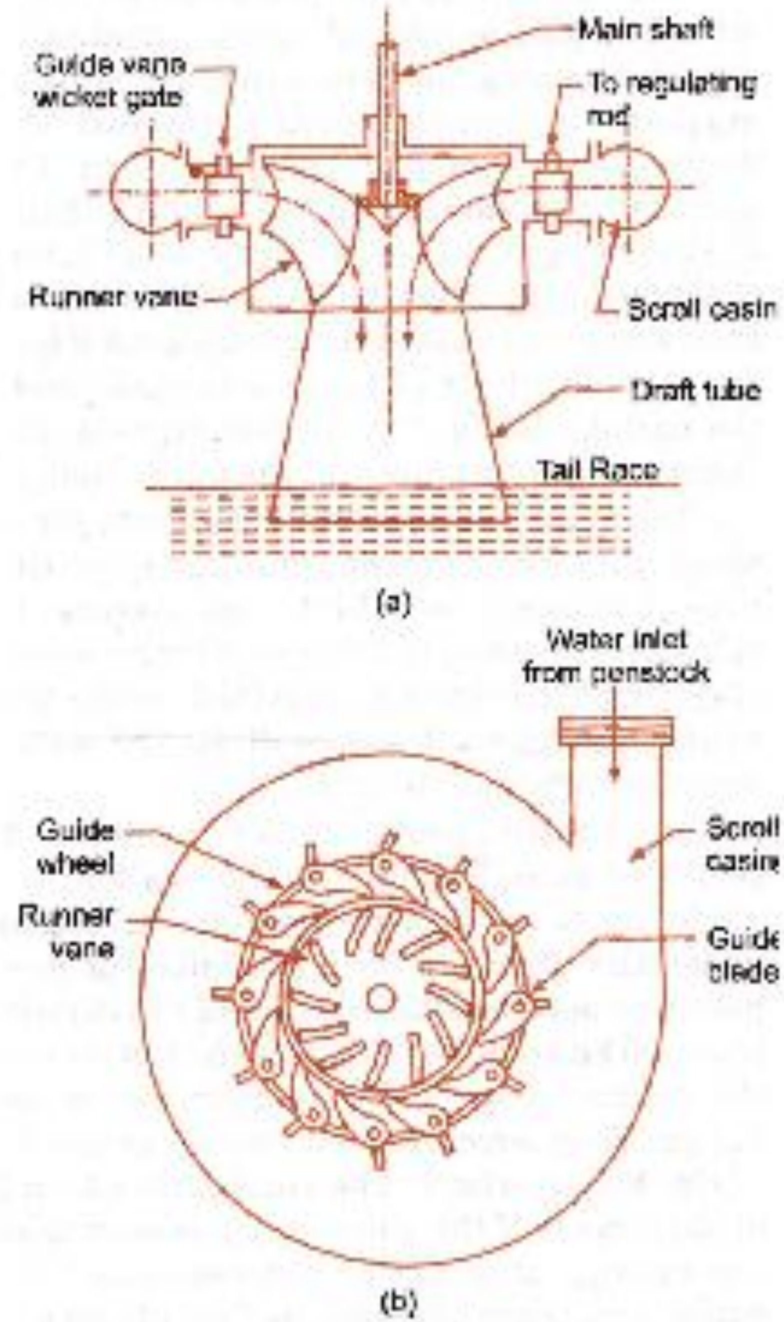
Spiral Casing

Runner

Guide Blades

Draft Tube




Main shaft
Guide vane wicket gate
To regulating rod
Runner vane
Scroll casin
Draft tube
Tail Race
(a)
Water inlet from penstock
Guide wheel
Runner vane
Scroll casin
Guide blade
(b)

Penstock

Penstock is a large diameter conduit, which carries water from a dam or a reservoir to the turbine house.

Since Francis turbine requires large volume of water than Pelton wheel, size of the penstock is bigger in the case of Francis turbine.

Spiral Casing

Water from the penstock enters into the spiral casing which completely surrounds the runner. This casing is also known as scroll casing or volute.

The cross-section area of this casing decreases uniformly along the circumference to keep the fluid velocity constant in magnitude along its path towards the guide vane.

This is so because the rate of flow along the fluid path in the volute decreases due to continuous entry of the fluid to the runner through the openings of the guide vanes.

Guide Blades

A series of airfoil shaped vanes called the guide vanes which are mounted on the casing.

Guide vanes are fixed between the two rings in form of a wheel; however they can swing about their own axis.

The basic purpose of the guide vanes is to convert a part of pressure energy at its entrance in to the kinetic energy and to direct the water or fluid on to the runner blades at an angle appropriate to the design.

Runner

It is the most important component of the Francis turbine.

The runner of a Francis turbine consists of a series of curved vanes evenly arranged around the circumference in the annular space between two plates.

The runner vanes are so shaped that water enters the runner radially at the outer periphery and leaves it axially at the inner periphery.
Most of the portion of pressure energy is converted into kinetic energy as water flows through the runner.
The driving force on the runner is both due to impulse (deviation in the direction of flow) and reaction (change in kinetic and pressure energy) effects.
The number of runner blades are usually varies between 16 to 24.The runner is keyed to the shaft which is usually of forged steel.

Draft Tube

It is a pipe or passage of gradually increasing cross-sectional area towards its outlet end. It connects the runner exit to the tail race.

As the pressure of reaction turbine decreases continuously as water passes through the guide vanes and the runner, it does below atmospheric pressure at the outlet of the runner.

Draft tube is used to discharge the water to the tail race by increasing pressure above atmospheric.

Draft tube must be submerged below the level of water in the tail race.

Working of a Francis Turbine

Water through the penstock under pressure enters the spiral casing which completely surrounds the runner.

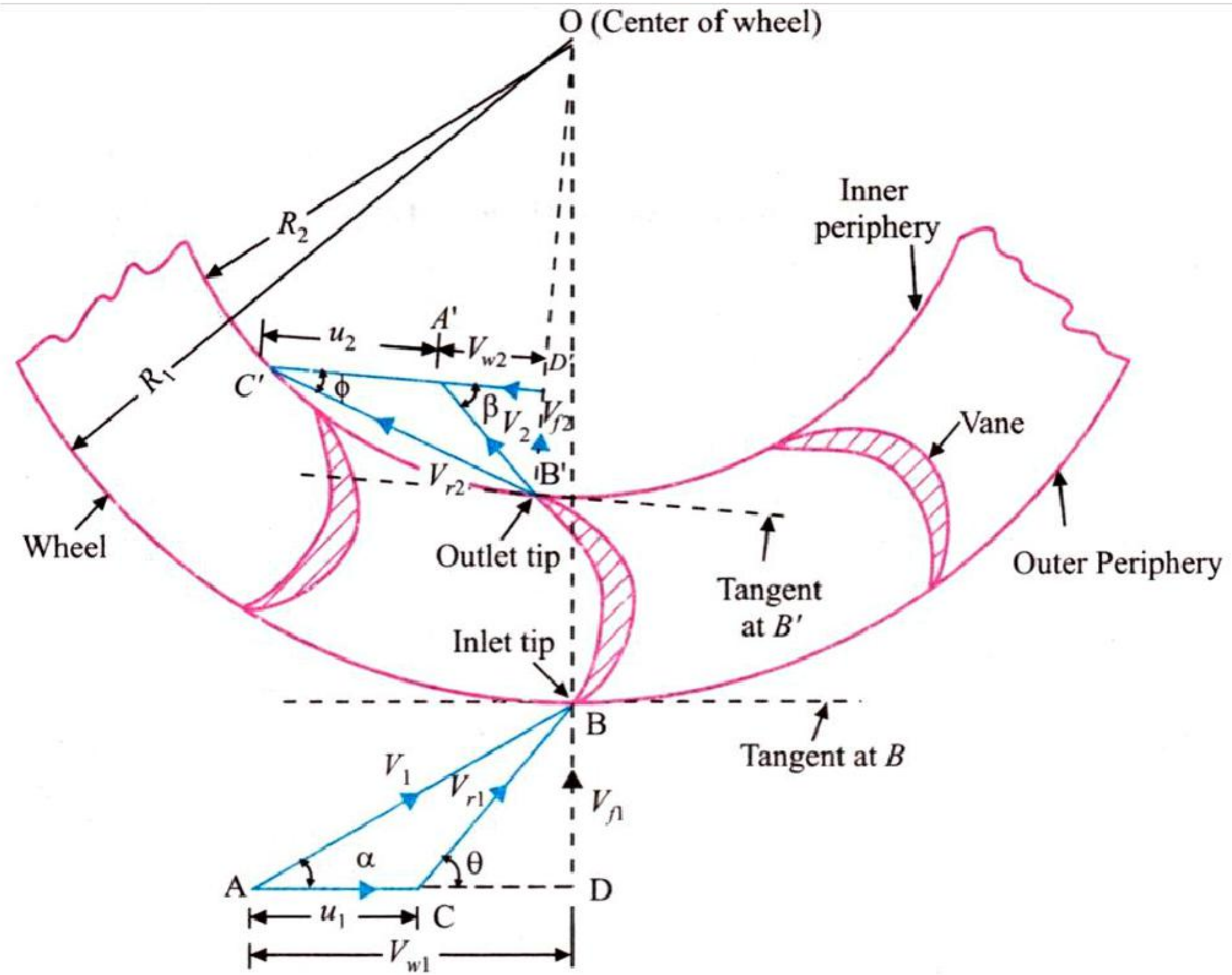
From casing water passes through a series of guide vanes, which directs the water to the runner at a proper angle.

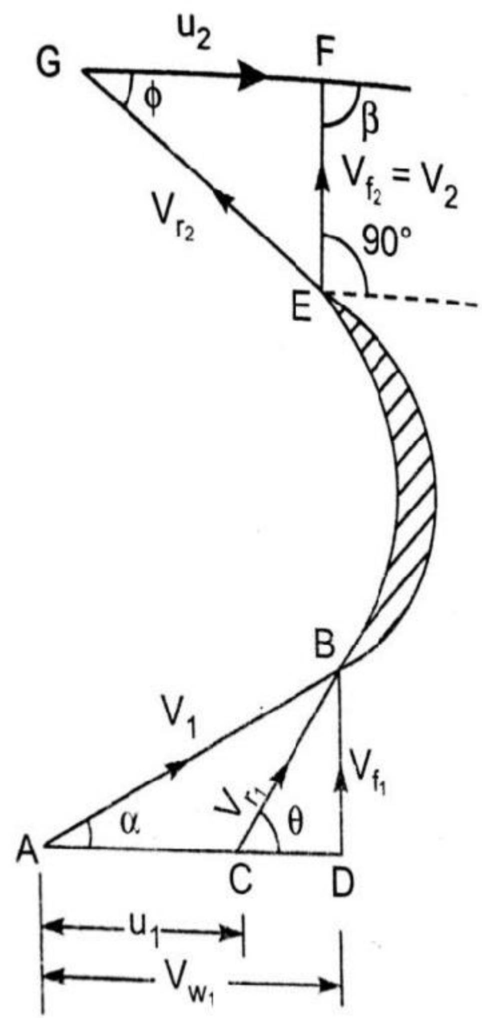
The pressure energy of water reduces continuously as it passes over the guide vanes and moving vanes.

The difference in pressure at stationary guide vanes and moving runner is responsible for the motion of the runner vanes.

Finally water is discharged to the tail race through a draft tube.

Velocity Triangles, Work done and Efficiency of Francis Turbine





General expression for **work done by runner** will be derived in the same manner as in the case of series of radial curved vanes (illustrated earlier in chapter-2 Impact of Jets, Fig. 2.11, Page No. 2.19)

$$WD/sec = \dot{m}(V_{w1}u_1 \pm V_{w2}u_2)$$

$$WD/sec = \rho a V_1(V_{w1}u_1 \pm V_{w2}u_2) ---------(3.18)$$

$$if\ \beta < 90^\circ \rightarrow +ve\ sign\ taken$$

$$if\ \beta > 90^\circ \rightarrow -ve\ sign\ taken$$

For maximum output, runner of the Francis turbine is so designed that there occurs a radial discharge at the outlet tip of the blades.

For radial discharge at the outlet, $\beta = 90^\circ$ and $V_{w2} = 0$, as shown in Fig. 3.14 (b).

$$\therefore \boldsymbol{WD/sec = \dot{m}\ (V_{w1}u_1)}, \qquad \boldsymbol{Nm/sec} ---------(3.19)$$

Hydraulic Efficiency

$$\eta_h = \frac{Runner\ Power}{Water\ Power}$$

$$\eta_h = \frac{\dot{m}\ (V_{w1}u_1)}{\rho g Q H} = \frac{\rho Q\ (V_{w1}u_1)}{\rho g Q H}$$

$$\boldsymbol{\eta_h = \frac{(V_{w1}u_1)}{gH}} ---------(3.20)$$

Working Proportions for Francis Turbine

1. Flow Ratio (K_f)

- Ratio of flow velocity at the inlet (V_{f1}) to theoretical velocity $(\sqrt{2gH})$ is called flow ratio. Its value lies between 0.15 to 0.30.

$$K_f = \frac{V_{f1}}{\sqrt{2gH}} \text{----------} (3.21)$$

2. Speed Ratio (K_u)

- Ratio of the peripheral velocity at the inlet (u_1) to theoretical velocity $(\sqrt{2gH})$ is called speed ratio. Its value lies between 0.6 to 0.9.

$$K_u = \frac{u_1}{\sqrt{2gH}} \text{----------} (3.22)$$

3. Breadth Ratio (n)

- Ratio of width of the runner (B) to outside diameter of the runner (D) is called breadth ratio. Its value ranges from 0.1 to 0.4.

$$n = \frac{B}{D} \text{----------} (3.23)$$

Total Discharge through Francis Turbine

Let,

$D_1 =$ Diameter of runner at inlet

$D_2 =$ Diameter of runner at outlet

$B_1 =$ Width of runner at inlet

$B_2 =$ Width of runner at outlet

$V_{f1} =$ Velocity of flow at inlet

$V_{f2} =$ Velocity of flow at outlet

$n =$ Number of vanes on runner

$t =$ Thickness of each vane

$Then,$ total discharge through the Francis turbine is given by,

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

$$= Area\ at\ outlet \times Velocity\ of\ flow\ at\ outlet$$

$$\therefore Q = \pi D_1 B_1 \times V_{f1} = \pi D_2 B_2 \times V_{f2} ---------(3.24)$$

If the thickness of the vanes are taken into consideration, then the area through which flow takes place is given by, $(\pi D_1 - nt)B_1$

Hence,

$$Q = (\pi D_1 - nt)B_1 \times V_{f1} = (\pi D_2 - nt)B_2 \times V_{f2} ---------(3.25)$$

Axial Flow Reaction Turbine

- In an axial flow reaction turbine, the water flows parallel to the axis of the rotation of the shaft.
- It is used under low head and high discharge conditions.
- 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.

Types of Axial Flow Reaction Turbine

Kaplan Turbine and Propeller Turbine

*When the vanes are fixed to the hub and they are **not adjustable,** the turbine is known as **Propeller turbine**.*

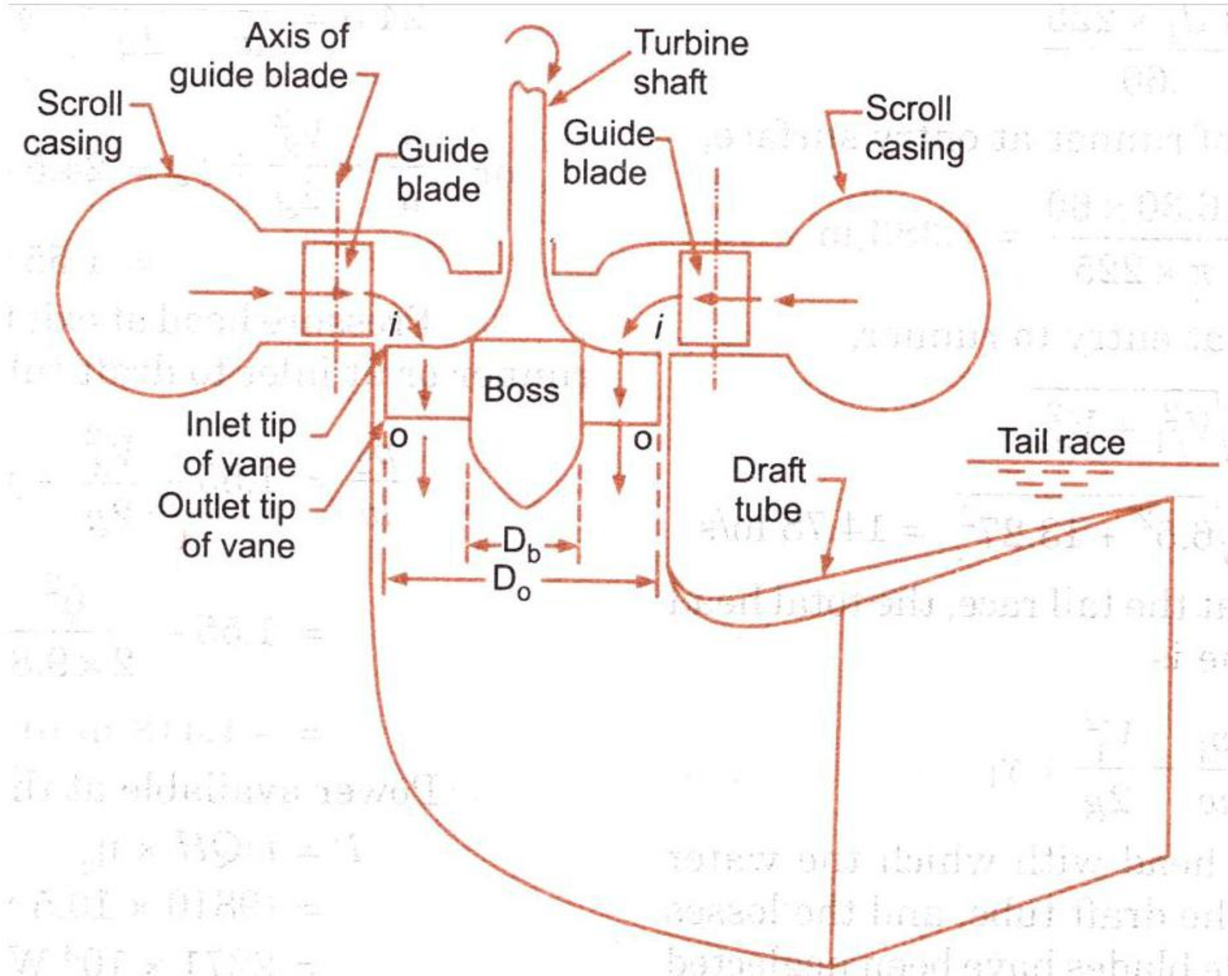
*If the vanes on the hub are **adjustable** the turbine is known as a **Kaplan turbine**.*

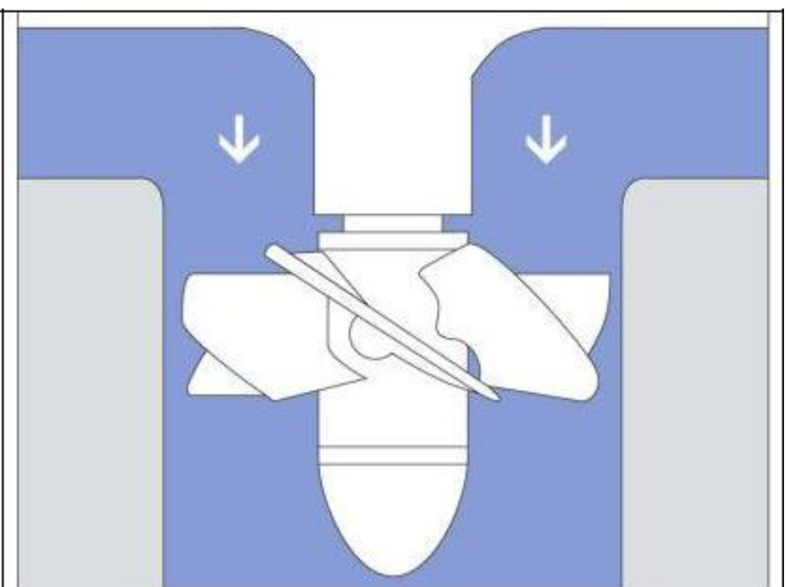
The runner blades are adjusted automatically by servo-mechanism so that at all loads the flow enters them without shock. This gives better part load efficiency for Kaplan turbine.

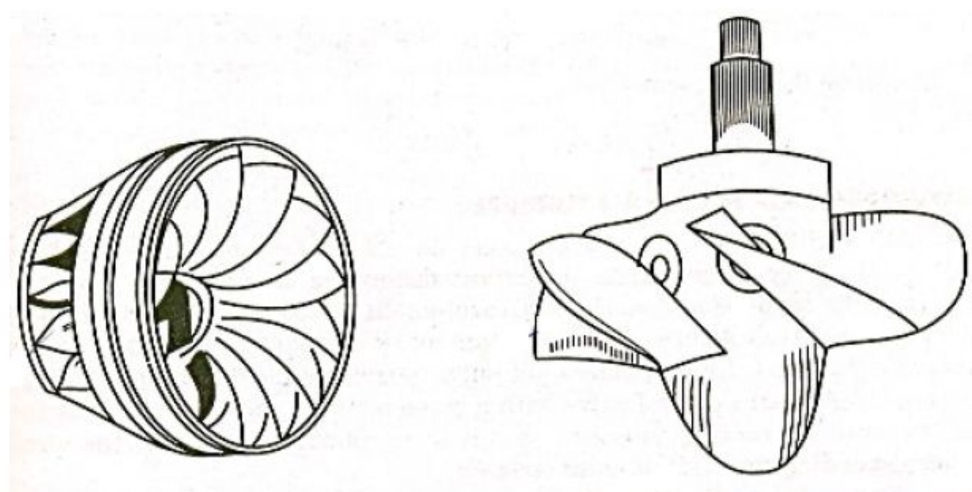
Components of Kaplan turbine and Propeller turbine are similar to that of the Francis turbine, only the ***runner is different***.

Main parts of the Kaplan & Propeller turbine are:

- Scroll casing
- Guide vane mechanism
- Hub with vanes or runner and
- Draft tube







(a) Francis turbine runner (b) Kaplan turbine runner

Kaplan Turbine

- The Kaplan turbine is purely axial flow reaction turbine in which the flow of water is parallel to the shaft.
- The number of vanes greatly reduces about 4 to 8 with the result that quantity of water handled for same diameter increases with lower losses on friction.
- Therefore, becomes more compact in construction and has higher rotational speed. Because of these characteristics, this turbine is used at such places where comparatively low head and large quantity of water is available.

The water from penstock enters the casing and then moves to the guide vanes. From the guide vanes, the water turns through 90° and flows axially through the runner as shown in Fig. 3.13.

Work done, Efficiency and Power Developed

Expressions for work done, efficiency and power developed by Kaplan & Propeller turbine are similar to that of Francis turbine.

Discharge through Runner of Kaplan & Propeller Turbine

The discharge through the runner is obtained by,

$$Q = \frac{\pi}{4}(D_o^2 - D_b^2) \times V_{f1} - - - - - - - - - - (3.27)$$

Where,

D_o = Outer diameter of the runner

D_b = Diameter of the hub

V_{f1} = Velocity of flow at inlet

Working Proportions of Kaplan and Propeller Turbine

1. The peripheral velocity at inlet and outlet are equal,

$$\therefore u_1 = u_2 = \frac{\pi D_0 N}{60} ----------(3.28)$$

2. Velocity of flow at inlet and outlet are equal,

$$\therefore V_{f1} = V_{f2} = K_f\sqrt{2gH} ----------(3.29)$$

3. Area of flow at inlet and outlet are equal,

$$\therefore A_1 = A_2 = \frac{\pi}{4}(D_o^2 - D_b^2)$$

Draft Tube

In case of reaction turbines, the water works on the turbine and while doing so, it imparts energy to the vanes and runner. Thus the water pressure reduces than the atmospheric pressure *i.e. vacuum*.

As the water flows from higher pressure to lower pressure, since the water pressure has *dropped below the atmospheric pressure, the water cannot come out of the turbine.*

Thus, a divergent tube is connected to the end of the turbine. This tube is called as a draft tube

A Draft tube connects the runner exit to tail race.

It is a pipe of gradually increasing area used for discharging water from exit of turbine to tail race.

By using draft tube, the kinetic energy rejected at the outlet of turbine is converted to useful pressure energy.

The turbine may be placed above the tail race and hence it can be inspected properly.

Draft tube is an integral part of reaction turbine. It is an air tight diverging conduit with cross-sectional area increasing along its length.

One end of this diverging tube is connected to runner exit and the other is located below the level of tail race

- It creates a negative head at the outlet of the runner thereby increasing the net head on the turbine.
- It converts a large proportion of rejected kinetic energy into useful pressure energy

function of the draft tube

The ***function of the draft tube*** are:

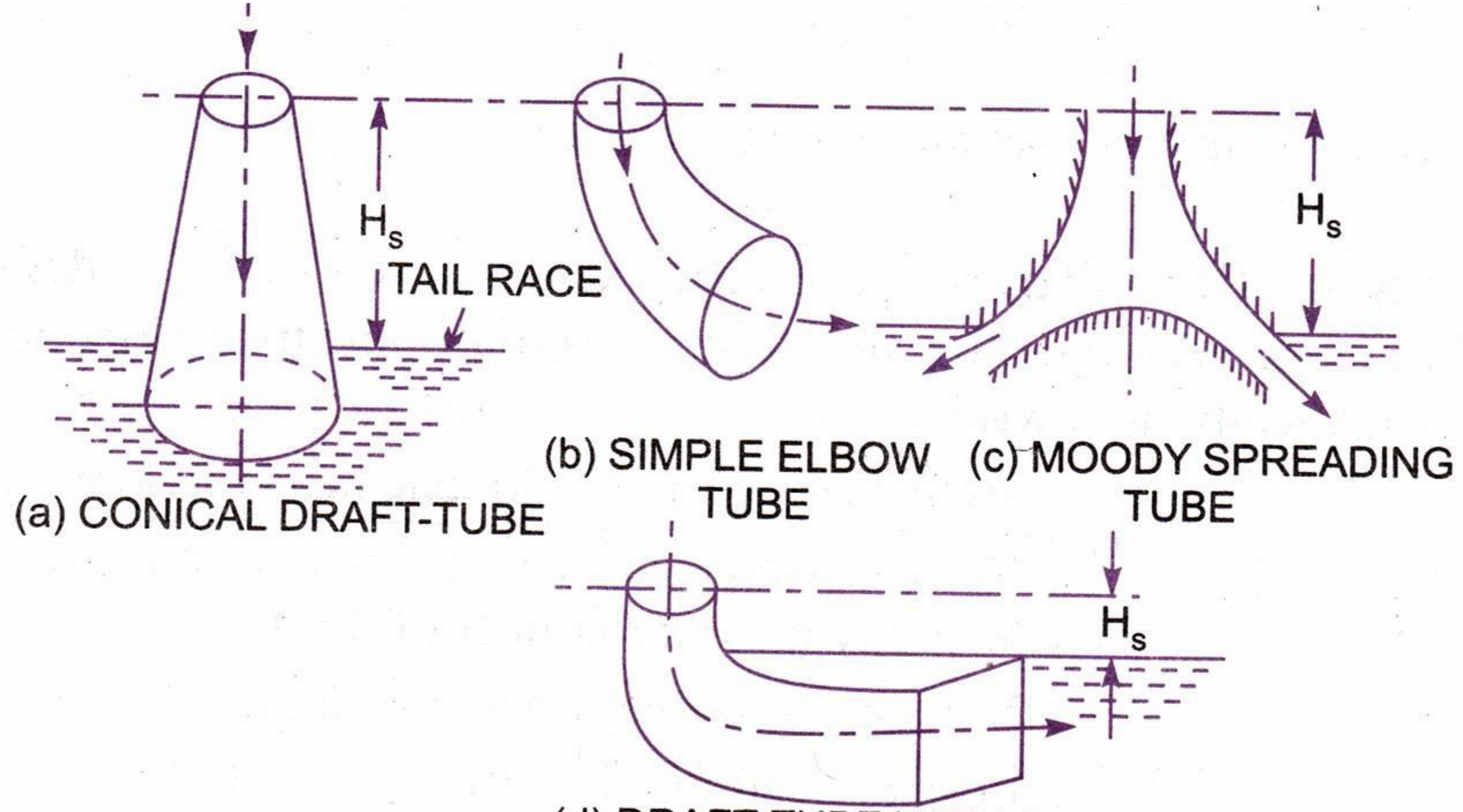
When water flows through turbine its kinetic energy and pressure energy is utilized to generate shaft power. Even though when water leaves the turbine it possesses high kinetic energy and negative pressure head.

If water is discharged through a draft tube having gradually increasing cross sectional area, the velocity is largely reduced at the outlet of the draft tube, and thus resulting in a gain in kinetic head and also increases the negative pressure head at the turbine exit so that net working head on the turbine increases. So output of turbine and efficiency also increases.

By providing a draft tube, a turbine can be installed above the tail race without loss of any head. This helps to make inspection and maintenance of a turbine easy.

Different types of draft tubes

- *Straight divergent tube or Conical draft tube*
- *Simple elbow tube*
- *Moody spreading tube*
- *Elbow tube with circular cross-section at inlet and rectangular at outlet*



(a) CONICAL DRAFT-TUBE (b) SIMPLE ELBOW TUBE (c) MOODY SPREADING TUBE

(d) DRAFT-TUBE WITH CIRCULAR INLET AND RECTANGULAR OUTLET

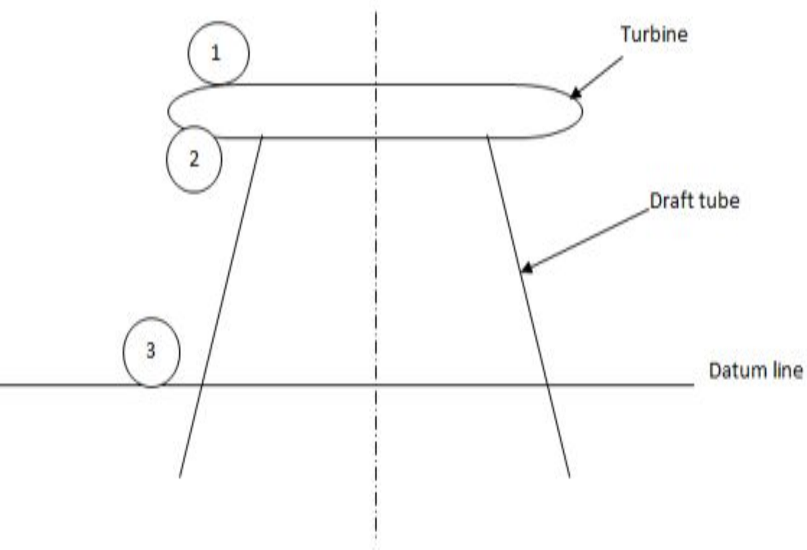
Draft Tube Theory

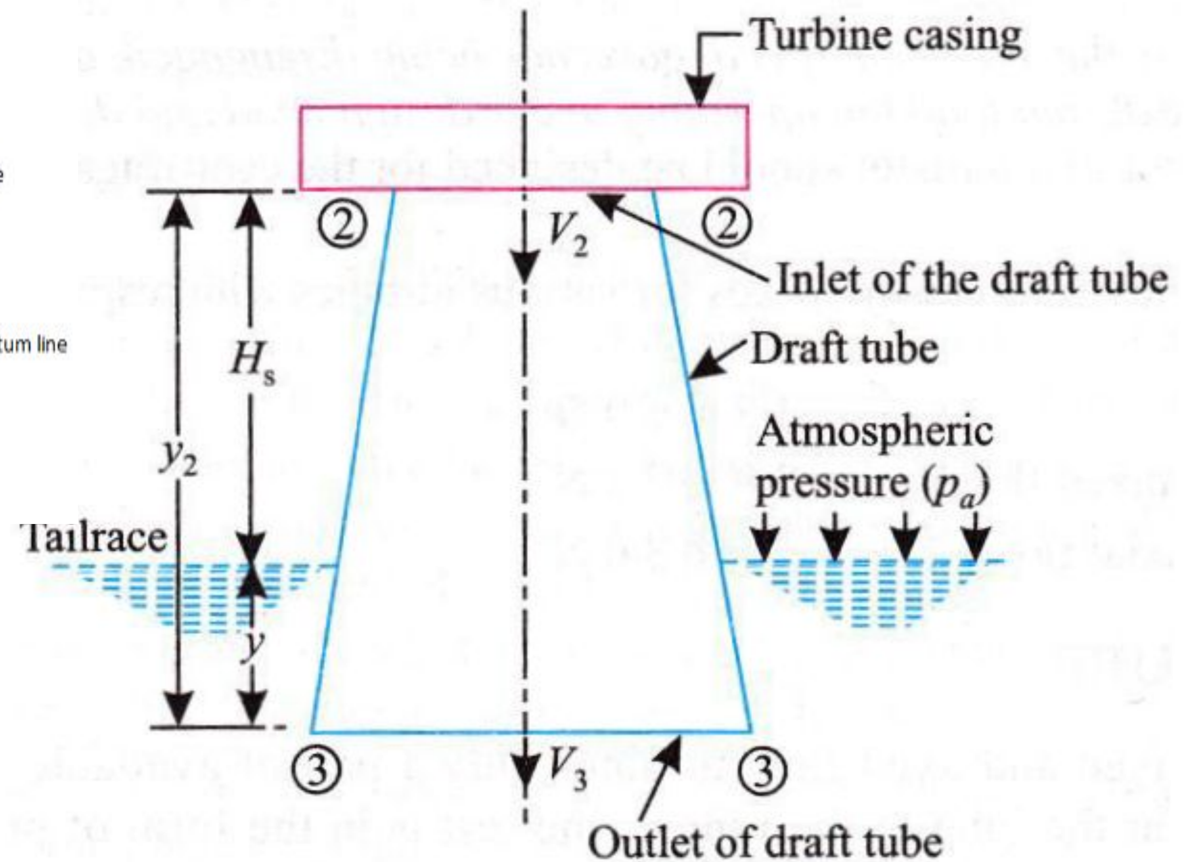
Let,

Vertical height of draft tube above the tail race = ***hs***

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

- **Applying Bernoulli's eqation to the inlet (section 2-2) and outlet (section 3-3) of the draft tube as shown in Figure**





- Assuming section 3-3 as a datum line, we get,

$$\frac{P_2}{\rho g}+\frac{V_2^2}{2g}+(H_s+y)=\frac{P_3}{\rho g}+\frac{V_3^2}{2g}+0+h_f---------(3.30)$$

Where,

h_f = Loss of energy between section 2-2 and 3-3.

But,

$$\frac{P_3}{\rho g}=Atmospheric\ pressure\ head+y$$

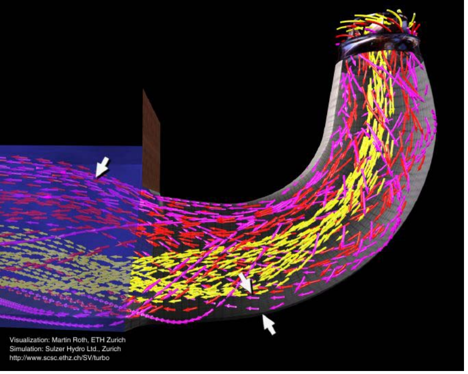
$$\therefore\frac{P_3}{\rho g}=\frac{P_a}{\rho g}+y$$

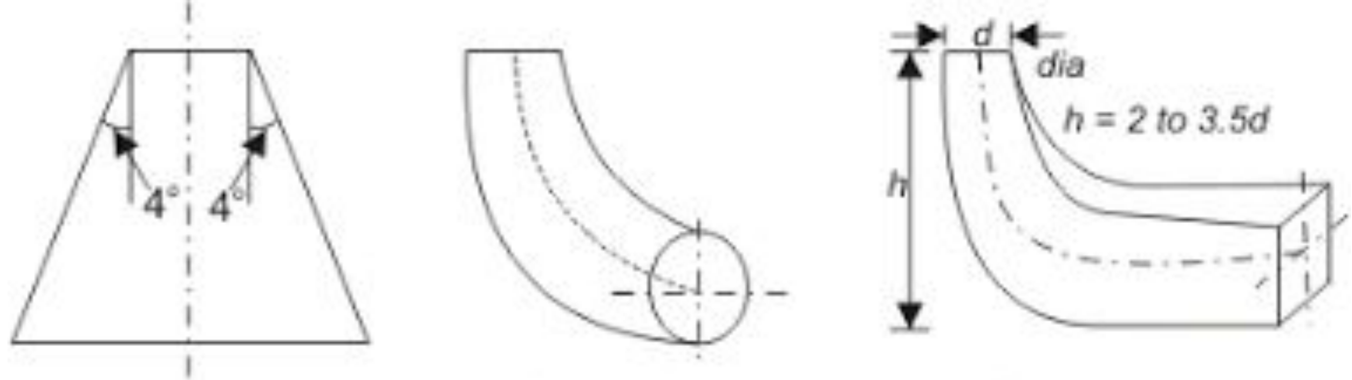
So,

$$\therefore\frac{P_2}{\rho g}+\frac{V_2^2}{2g}+(H_s+y)=\frac{P_a}{\rho g}+y+\frac{V_3^2}{2g}+h_f$$

$$\therefore\frac{P_2}{\rho g}+\frac{V_2^2}{2g}+H_s=\frac{P_a}{\rho g}+\frac{V_3^2}{2g}+h_f$$

$$\therefore\boldsymbol{\frac{P_2}{\rho g}=\frac{P_a}{\rho g}-H_s-\left(\frac{V_2^2}{2g}-\frac{V_3^2}{2g}-h_f\right)}---------(3.31)$$



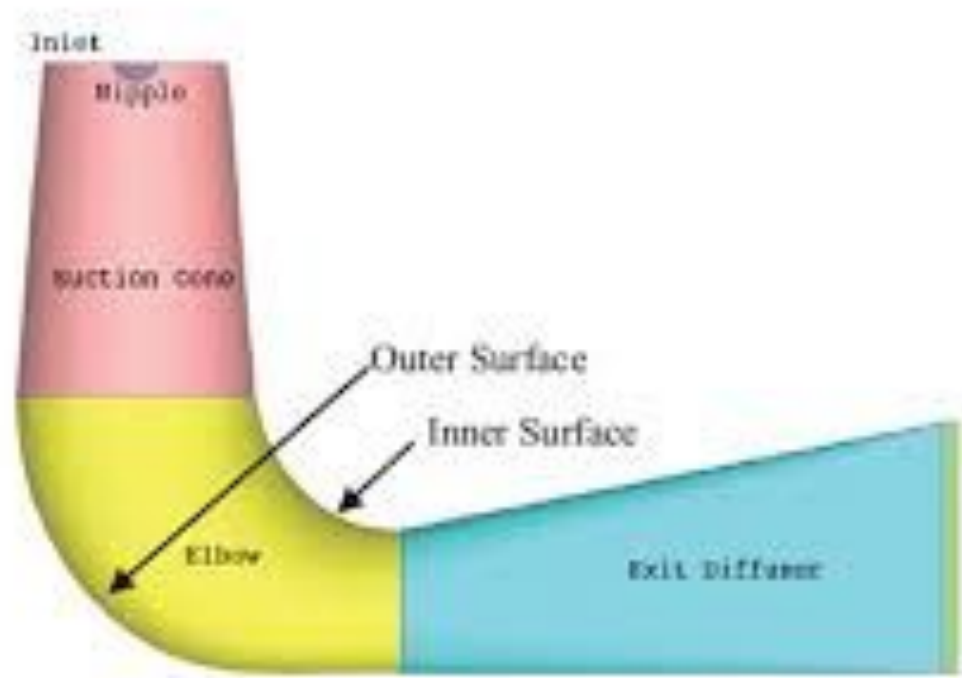


(a) Straight type

(b) Simple elbow type

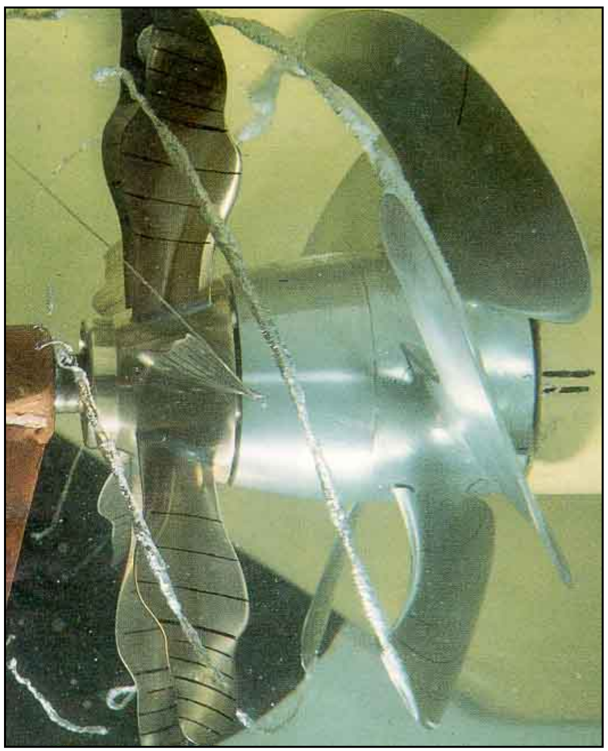
(c) Elbow type with varying cross-section

Components of Draft Tube



cavitation

The liquid enters hydraulic turbines at high pressure; this pressure is a combination of static and dynamic components. ... Thus, Cavitation can occur near the fast moving blades of the turbine where local dynamic head increases due to action of blades which causes static pressure to fall.



Cavitation: Thoma number

$$\sigma = \frac{\dfrac{p_A}{\rho g} - \dfrac{p_v}{\rho g} - h_s}{H}$$

s = Thoma number

p_a = atmospheric pressure

p_v = vapour pressure

h_s = elevation above tail water

H = total head

r = density

g = gravity

Specific speed (Ns)

It is the speed of a turbine which is identical in shape, geometrical dimension, blade angles, gate opening etc. with the actual turbine but of such a size that it will develop unit power when working under unit head.

It is used in comparing the different types of turbines as every type of turbine has different specific speed. It plays an important role for selecting the type of the turbine & predicting the performance of turbine.

Consider P as the power developed by the turbine.

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

- Where N = specific speed of turbine,
- H= Head,
- N= speedof the turbine, and
- P= power developed by the turbine
- **Use:**
- **To predict the performance of a turbine and to compare different types of turbine**

Derivation of the specific speed

The overall efficiency of any turbine is given by

$$\eta_0 = \text{shaft power / water power} = \text{Power developed} / (\rho g Q H / 1000)$$

where H = head under which the turbine is working, Q = discharge through the turbine, P = power developed or shaft power.

Now from above equation,

$$P = \eta_o \times \frac{\rho g Q H}{1000}$$

$$\propto Q \times H \text{ .... (1)}$$

Now let D = dia of actual turbine, N = speed of actual turbine, u = tangential velocity of the turbine, N_s= specific speed of the turbine, V = absolute velocity of water.

Now, $u \propto V$, and $V \propto H^{1/2}$. Hence

$$u \propto H^{1/2}$$

But $u = \pi D N / 60$, or

$$u \propto DN$$

Hence

$$\sqrt{H} \propto DN$$

or

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

Now

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

Hence eq (1) becomes

$$P \propto \frac{H^{5/2}}{N^2}$$

Or

$$P = K\,\frac{H^{5/2}}{N^2}$$

where K = constant of proportionality.
If P = 1, H = 1, then the speed N = specific speed N_s. putting this in above equation, we get

$$K = N_s^2$$

Hence

$$P = N_s^2\,\frac{H^{5/2}}{N^2}$$

Or

$$\boldsymbol{N_s = \frac{N\sqrt{P}}{H^{5/4}}}$$

In above eqn, if P is taken in metric horse power, the specific speed is obtained in MKS units. But if P is taken in kilowatts, the specific speed is obtained in SI units.

Governing of Hydraulic Turbines

The governing of a turbine is defined as the operation by which the speed of the turbine is kept constant under all working conditions (irrespective of the load variation)

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 loads conditions.

It 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.

The governor used in hydraulic turbines should be very strong as it has to deal with the water coming at very large force and huge quantity.

- **Functions:-**
- **Control speed of turbine with fluctuating load at synchronous speed of generator.**
- **Synchronous speed is the constant speed of the generator**

Why governing of hydraulic turbines is required?????

As turbine is directly coupled to the electric generator which is required to run at constant sped.

The load on turbine is not constant through out the day or hour, hence speed of turbine varies with respect to load at constant head and discharge .

Therefore in order to have constant speed of generator ,governing of turbine is required to maintain the constant speed of turbine with respect to load.

Main part for governing of pelton wheel….

1Centrifugal governor

2Oil pump-gear pump with oil sump

3Relay or control valve

4Servomotor with spear rod and spear

5Deflector mechanism.

Governor and linkages :

- A centrifugal governor is used as the measuring element of the closed loop control system. It is driven by the turbine shaft through bevel gears. The sleeve of the governor is connected through linkages to relay valve. The movement of is transferred through the lever to move the piston rod of relay valve
- **Oil pump and oil sump** :.
  System uses oil in servomotor or relay cylinder since the force required to actuate the spear valve would be enormous. For this reason, the system requires an oil sump to store the oil and an oil pump to regulate the oil supply in the mechanism.
  Oil pump is a positive displacement type of pump like gear pump or axial piston pump. lt's function is to pressurize the oil.

Relay or control valve :

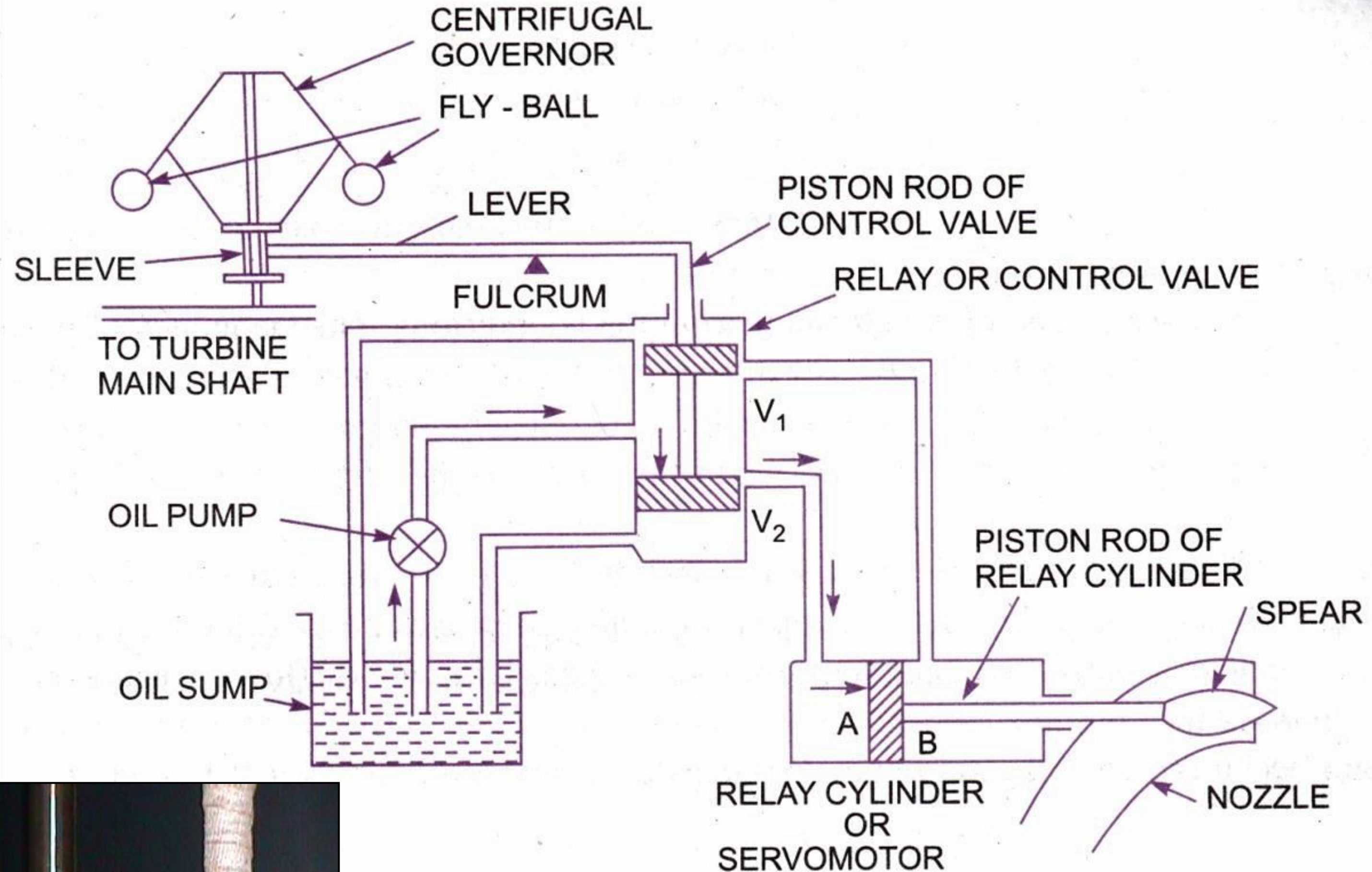
Relay valve is a spool valve having 5 ports. It is also called as control valve or distributor valve. It receives the pressurized oil from the oil pump which is diverted towards the ports to pipe A or pipe B. Through these pipes the oil is transferred to corresponding sides of double` acting servomotor cylinder. Simultaneously, the oil will be returned from the servomotor from the opposite pipe to the sump.

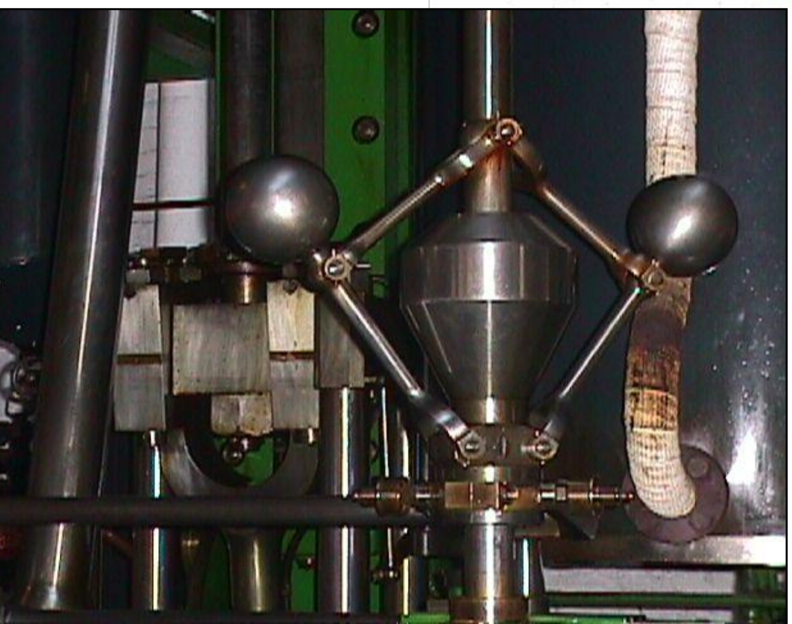
System uses oil in servomotor or relay cylinder since the force required to actuate the spear valve would be enormous. For this reason, the system requires an oil sump to store the oil and an oil pump to regulate the oil supply in the mechanism.

Oil pump is a positive displacement type of pump like gear pump or axial piston pump. lt's function is to pressurize the oil.

Servomotor or relay cylinder :

It is a double acting cylinder which acts as hydraulic actuator. It receives oil from relay valve say through pipe A. The piston of the cylinder will be displaced towards left, thus forcing the oil through the pipe B into the relay valve and finally to oil sump. It will simultaneously more the spear valve to the left and increase the area of flow through the nozzle.


CENTRIFUGAL GOVERNOR
FLY - BALL
LEVER
PISTON ROD OF CONTROL VALVE
SLEEVE
FULCRUM
RELAY OR CONTROL VALVE
TO TURBINE MAIN SHAFT
V_1
OIL PUMP
V_2
PISTON ROD OF RELAY CYLINDER
SPEAR
OIL SUMP
A
B
RELAY CYLINDER OR SERVOMOTOR
NOZZLE



Working:

When the load on the generator decreases, the speed of the generator increases. Hence speed of the turbine also increases beyond the normal speed.

The centrifugal governor which is connected to the turbine main shaft will be rotating at an increased speed and hence centrifugal force on the fly ball increases and it moves upward. Sleeve of the governor will also moves upward.

As the sleeve moves upward, a horizontal lever turns about the fulcrum and the piston rod of the control valve moves downward. This closes the valve V1 and opens the valve V2 as shown in fig

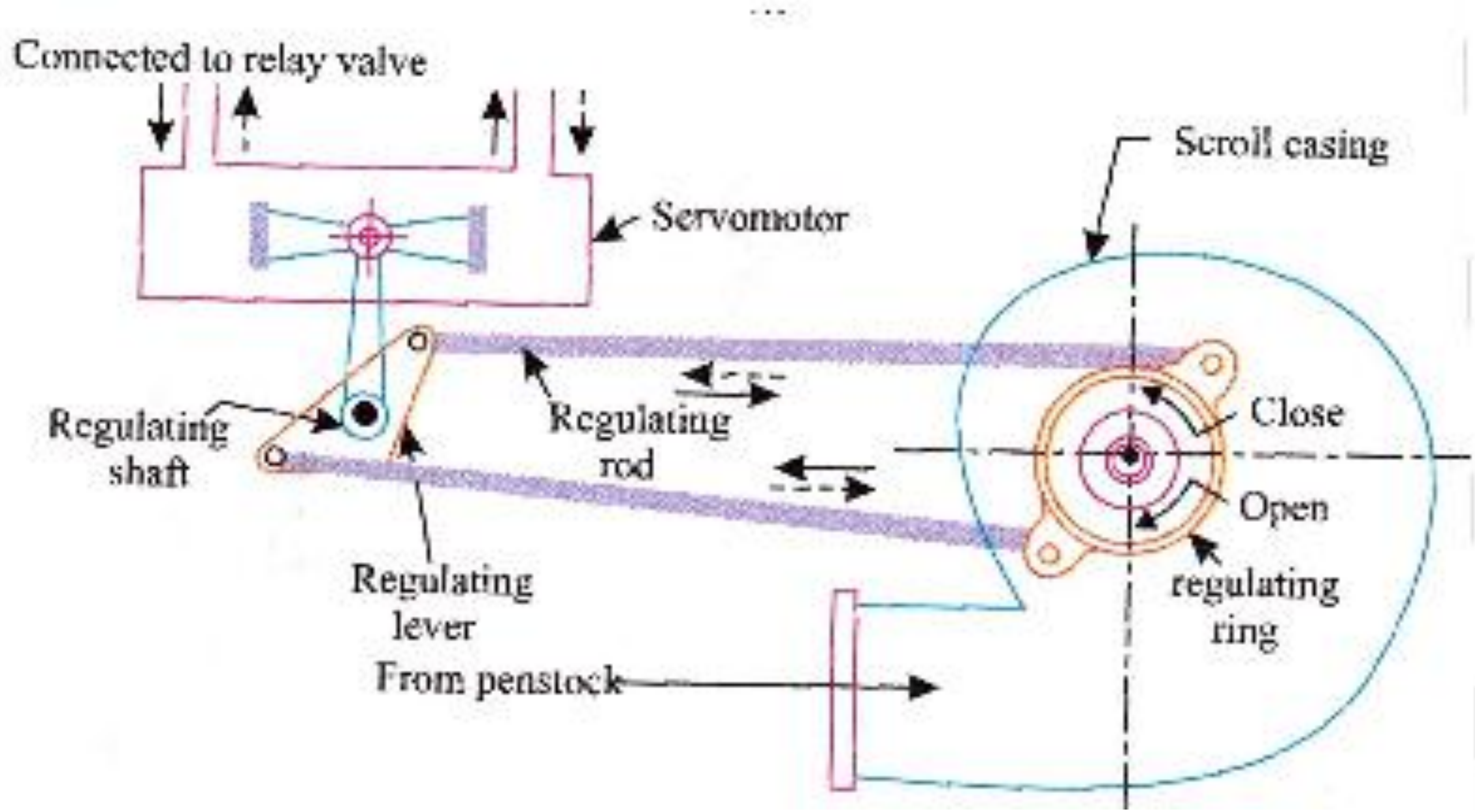
The oil pumped from the oil pump to the control valve under pressure will flow through the valve V2 to the servomotor and will exert force on the face A of the piston of the relay cylinder.

Piston along with piston rod and spear will move towards right. This will decrease the area of flow of water at the outlet of the nozzle and it will reduce the rate of flow to the turbine which consequently reduces the speed of the turbine.

Mean while bell crank lever moves downward, the jet deflector will operate and divert whole or part of the jet away from the buckets.

As soon as speed becomes normal, the fly balls, sleeves, lever and piston rod come to its normal position.

Governing of Francis Turbine



The guide blades of the Francis turbine are pivoted and connected by levers and links to the regulating ring. The regulating ring is attached with two regulating rods connected to the regulating lever. Thus regulating lever in turn is connected with regulating shaft, which is operated by the piston of servomotor

When load on the turbine decreases, speed tends to increase, which moves fly ball supwards and thus raises sleeves. Main lever on the other side of the fulcrum pushes down the control valve rod and opens port V_1. Oil under pressure enters the servomotor from left and pushes the piston to moves towards right.

When the piston of the servomotor moves towards right, regulating ring is rotated to decrease the passage between the guide vanes by changing guide vane angles. Thus quantity of water reaching the runner blades reduces and speed decreases to the normal speed.

Sudden reduction in passage between the guide blades may cause water hammer which can be prevented by providing a relief valve near the turbine which diverts the water directly to the tail race. Thus it functions similar to that of jet deflector as in Pelton wheel. Thus double regulation is also well performed in Francis turbine.

Characteristic curves of a Turbine

These are curves which are characteristic of a particular turbine which helps in studying the performance of the turbine under various conditions. These curves pertaining to any turbine are supplied by its manufacturers based on actual tests.

The data that must be obtained in testing a turbine are the following:

1. The speed of the turbine N
2. The discharge Q
3. The net head H
4. The power developed P
5. The overall efficiency
6. Gate opening (this refers to the percentage of the inlet passages provided for
water to enter the turbine)

The characteristic curves obtained are the following:

a)Constant head curves or main characteristic curves

b)Constant speed curves or operating characteristic curves

c)Constant efficiency curves or Muschel curves

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

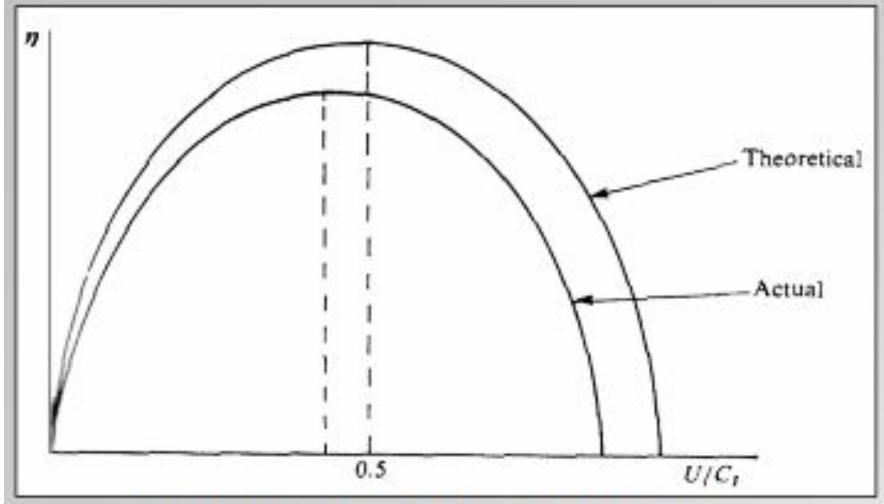
a) Unit discharge vs unit speed
b) Unit power vs unit speed
c) Overall efficiency vs unit speed

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

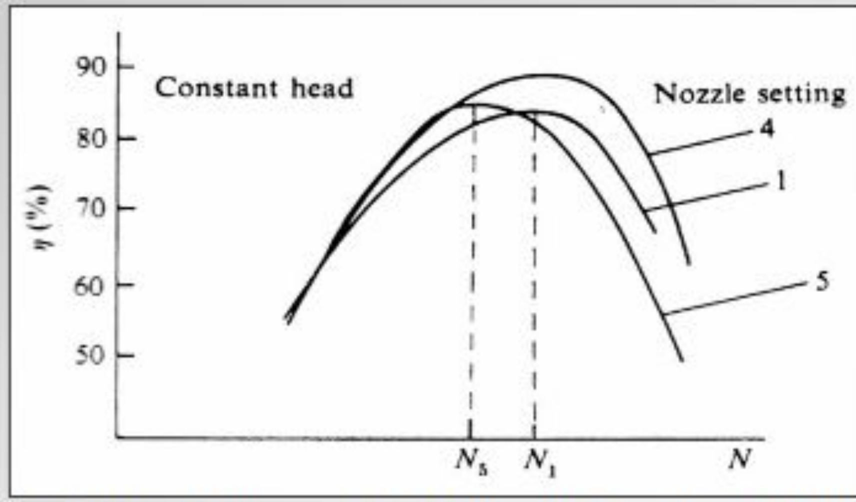
P	vs	Q
h_o	vs	Q
h_o	vs	P_u
$h_{o\,max}$	vs	% Full load

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

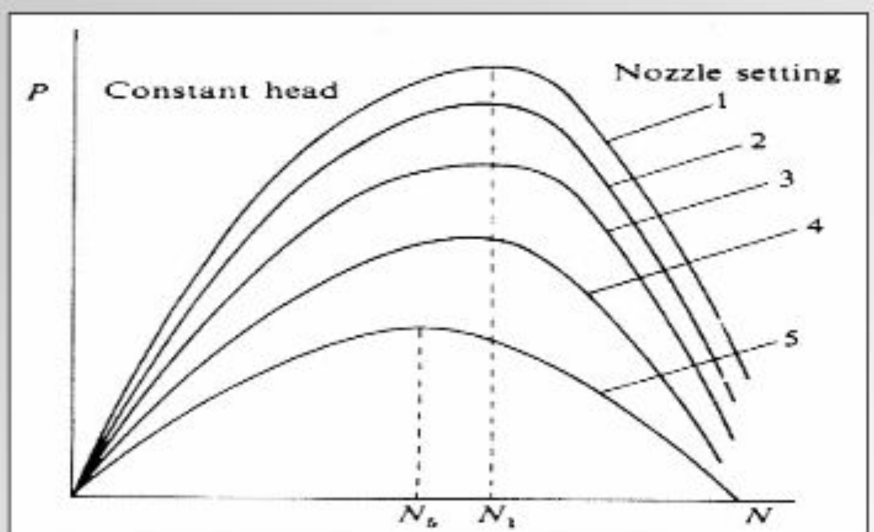
Characteristics Curves



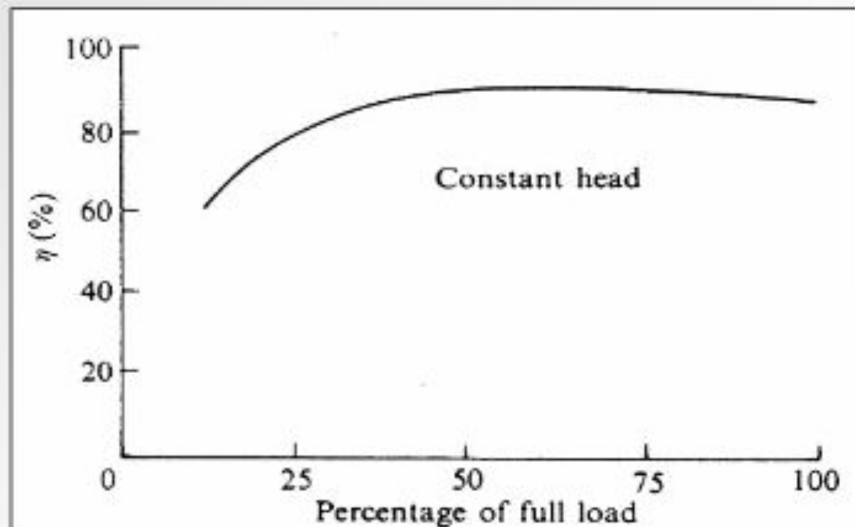
Efficiency and jet speed ratio



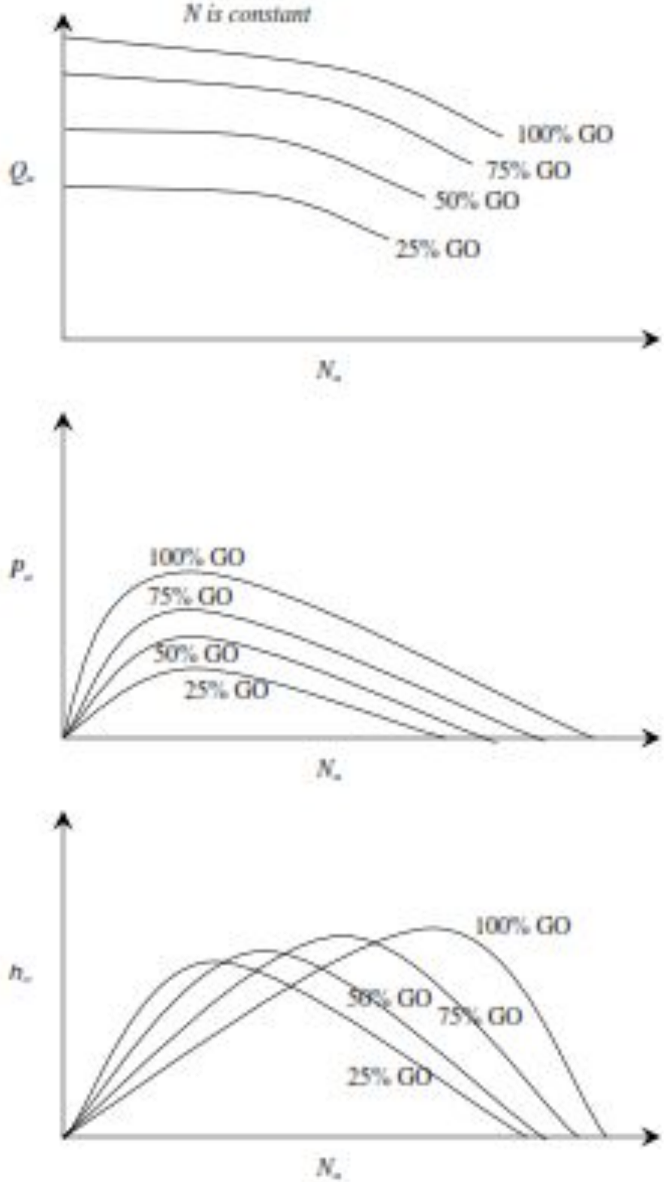
Efficiency vs speed at various nozzle setting



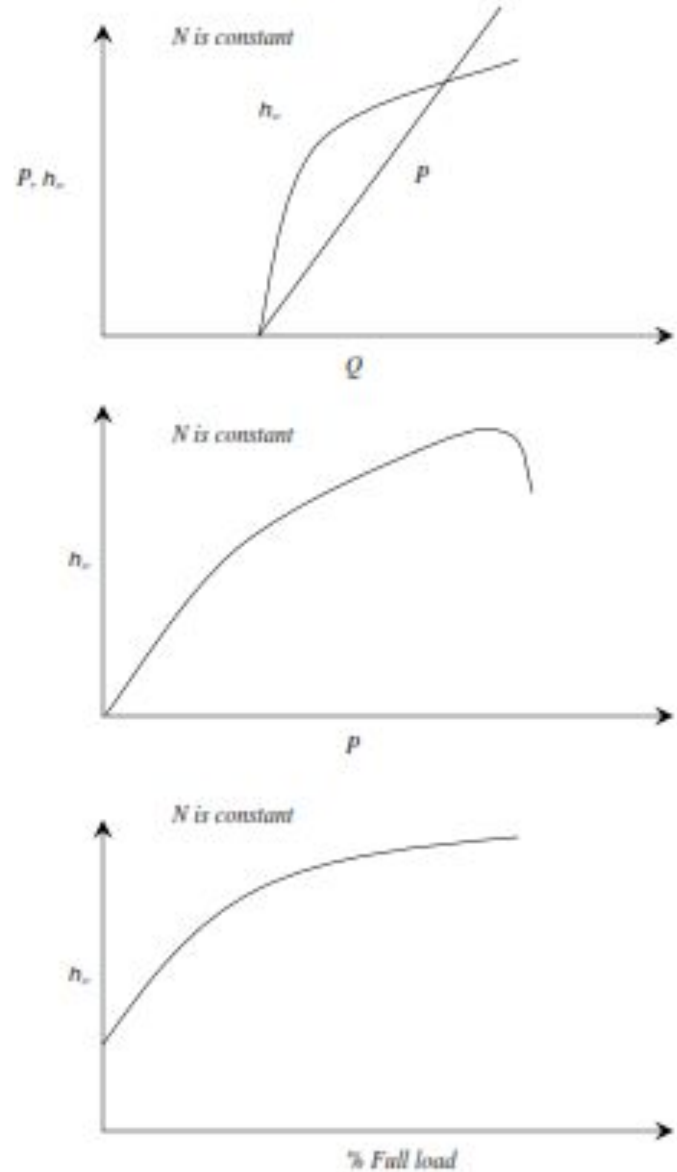
Power vs speed at various nozzle setting



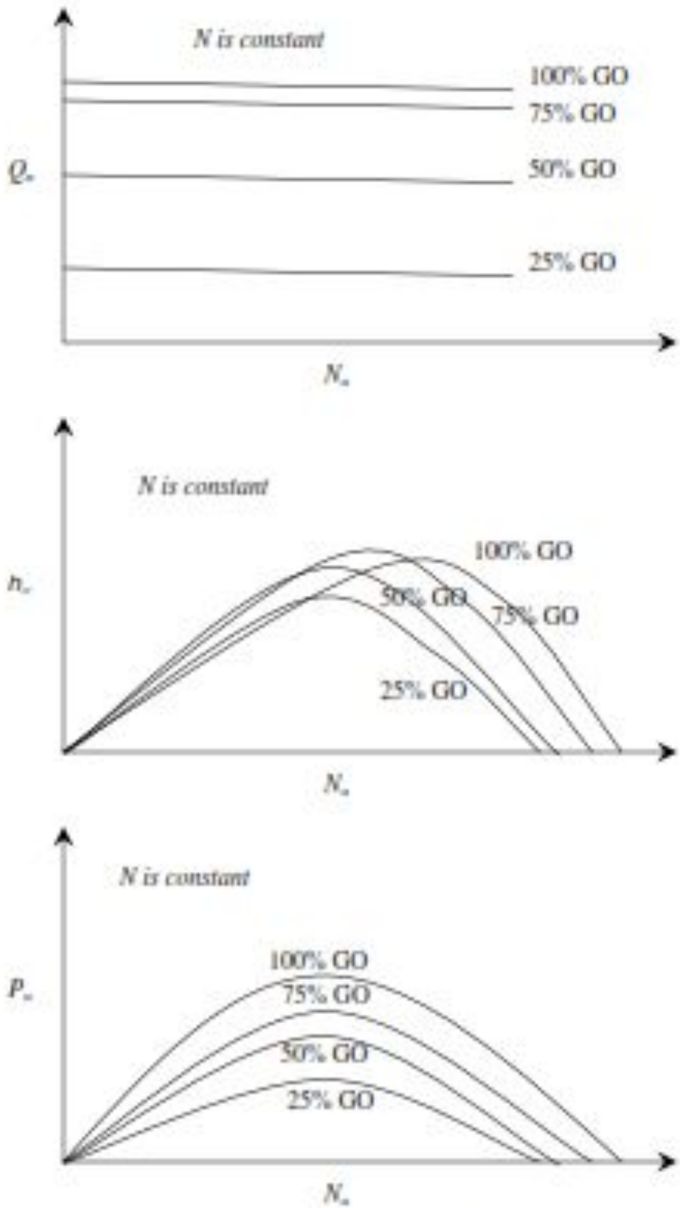
Variation of efficiency with load



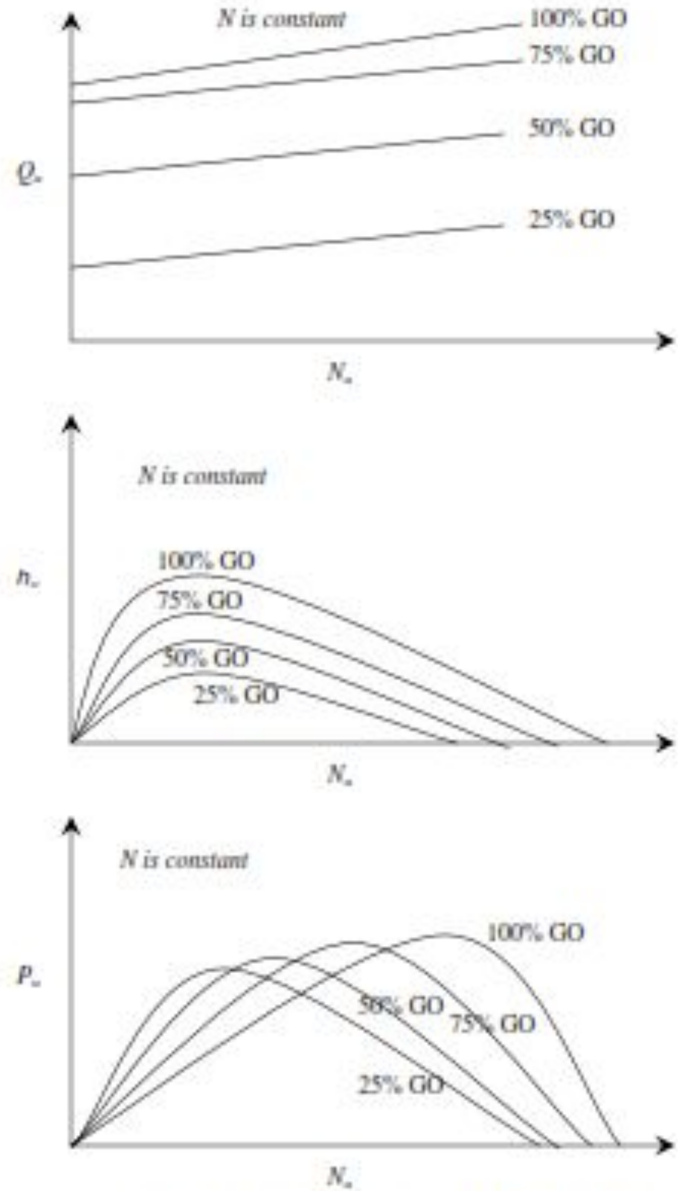
Main Characteristic curves of a Francis turbine



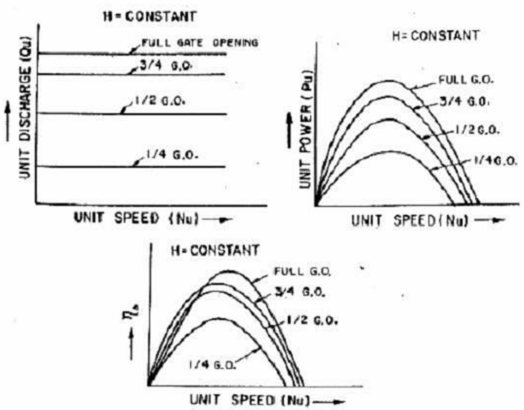
Operating Characteristic curves of a turbine



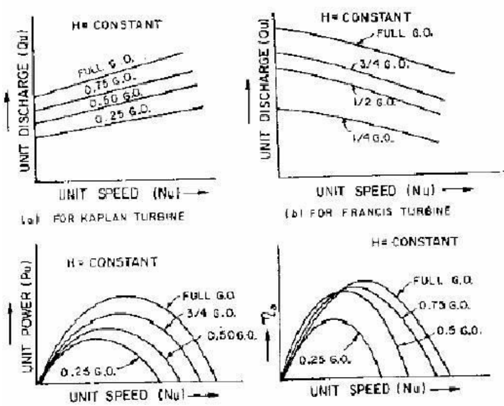
Main Characteristic curves of a Pelton turbine



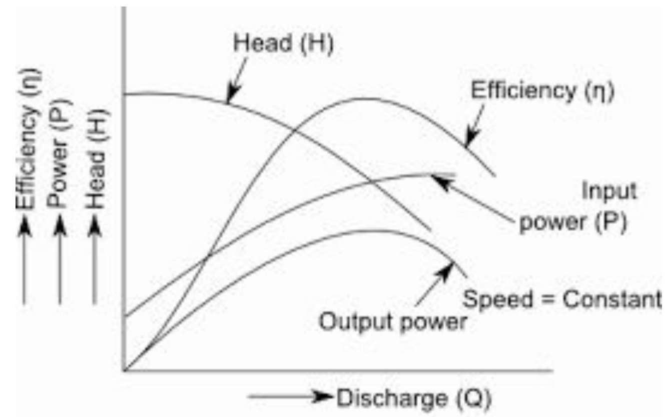
Main Characteristic curves of a Kaplan turbine



. Main characteristic curves for a Pelton wheel.



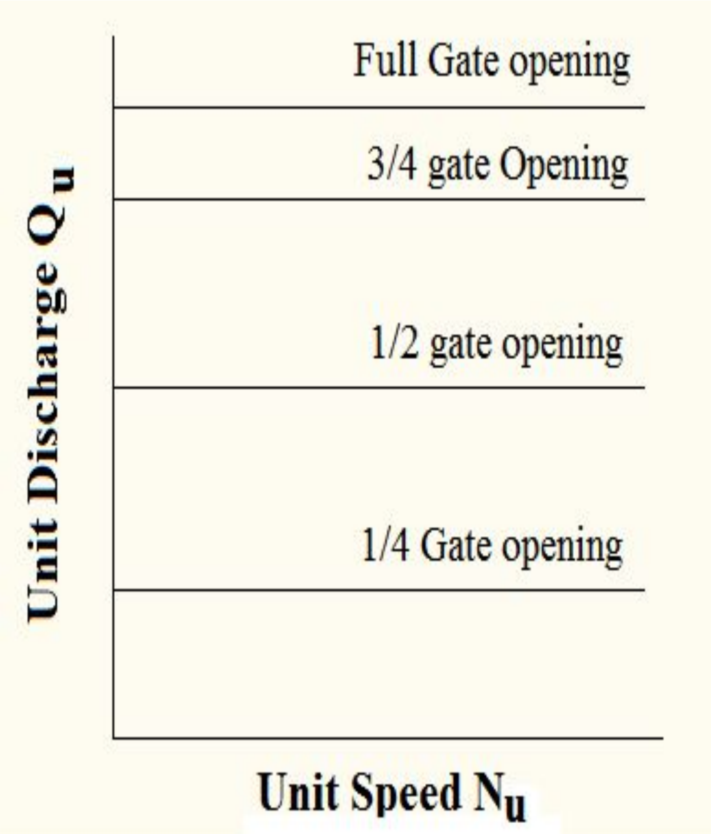
Main characteristic curves for reaction turbine.



Operating characteristic surves of a pump.

Main Characteristics

1)

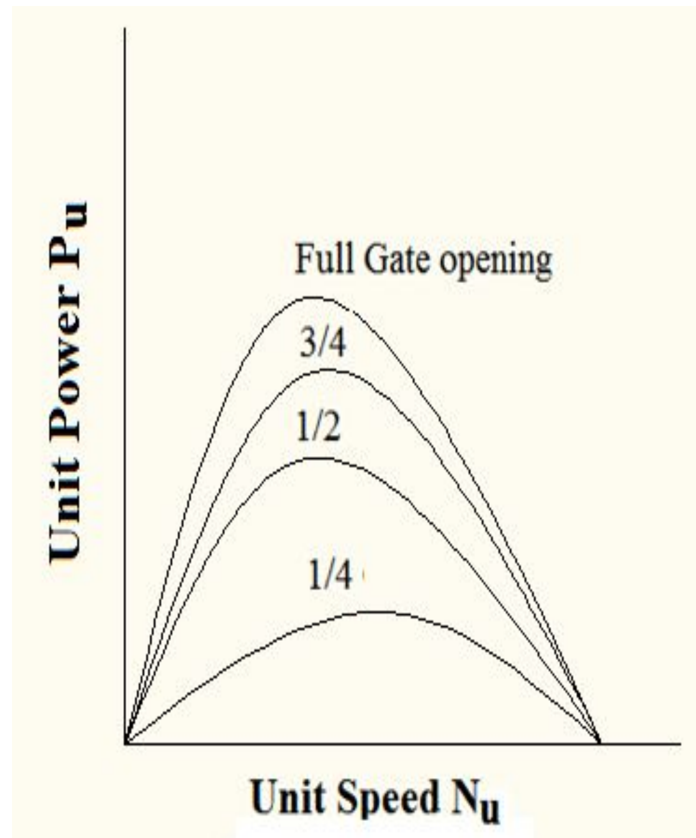


Unit speed vs Unit discharge

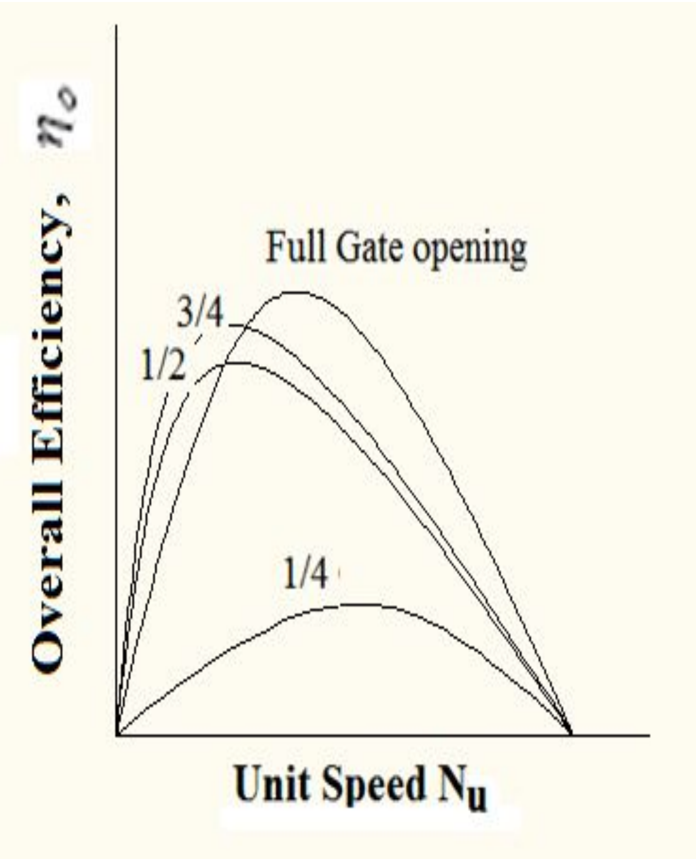
- Discharge unaffected by speed of the rotor
- Jet comes out as a free jet from the casing of the turbine

2)

- Power increases first to maximum with the increase in speed and then decreases with the increase in speed
- This is due to the fact that the as the speed of the rotor increases the jet does not have enough time to make an impact on the buckets.



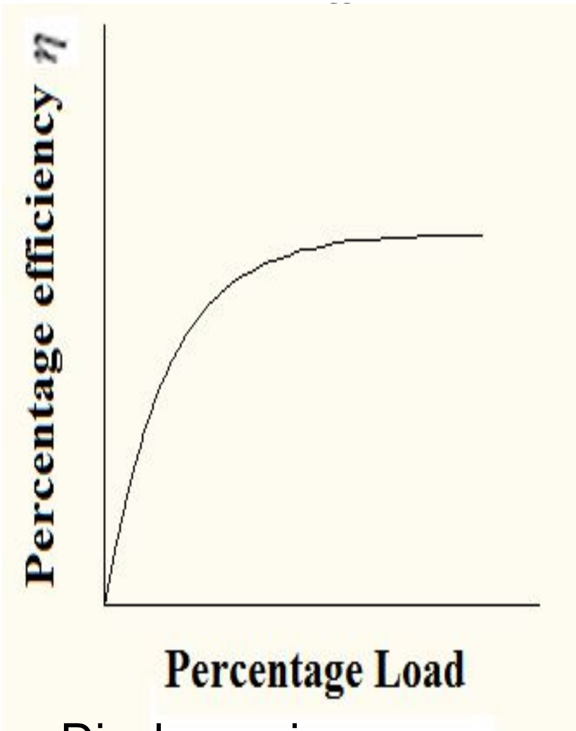
3) Unit Speed vs Overall Efficiency



- Similar to the Unit discharge vs unit speed graph.
- Two extreme speeds at which efficiency is zero.
- Highest speed called as runaway speed

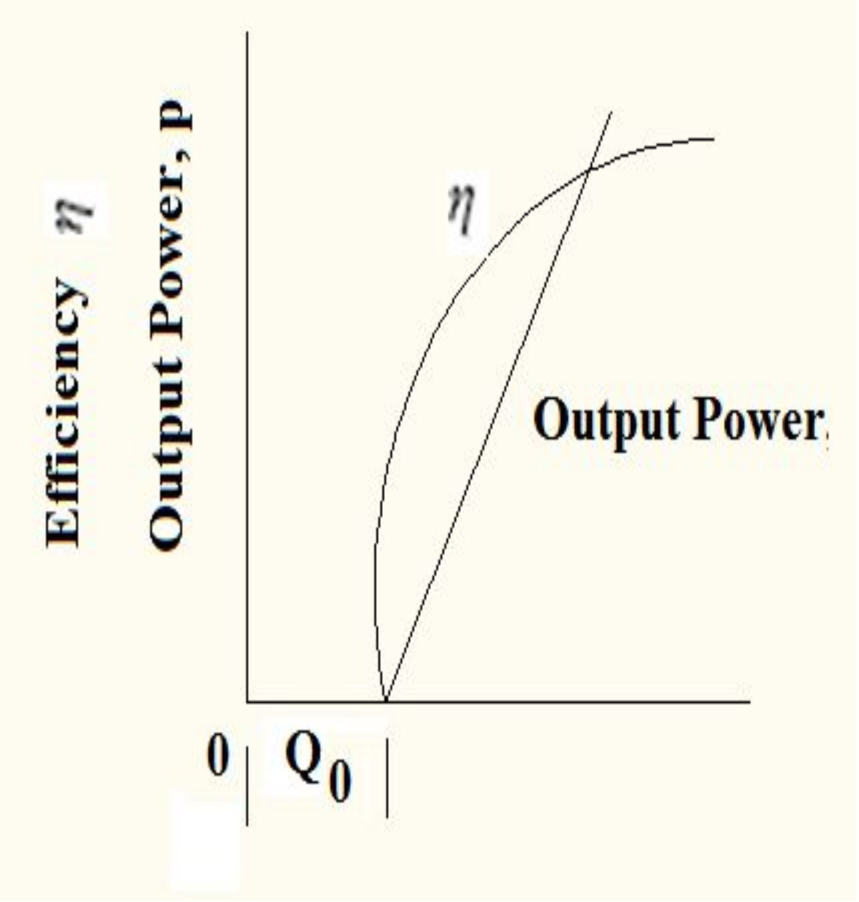
Operating Characteristics/Constant Speed characteristics

Load vs

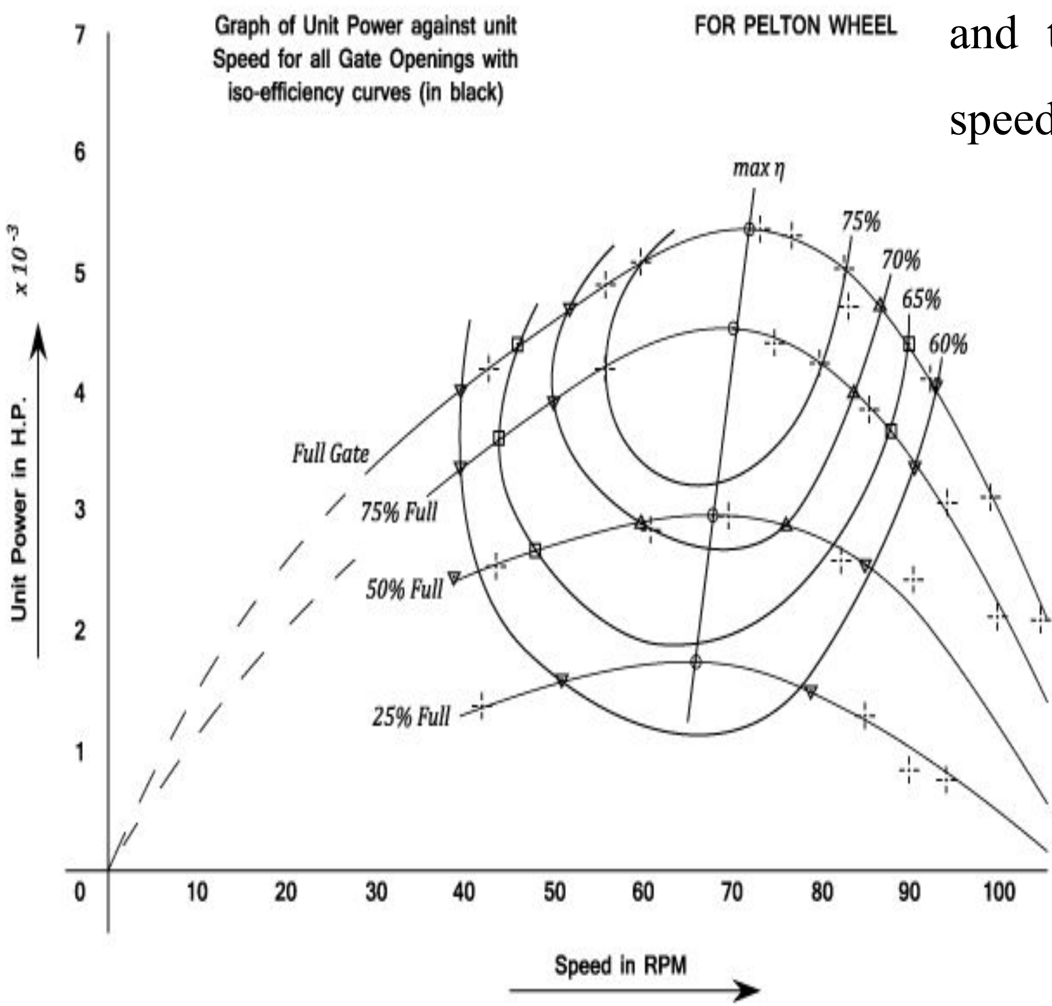
1) 

- Maximum Efficiency at 20 % to 30 % of load and constant afterwards

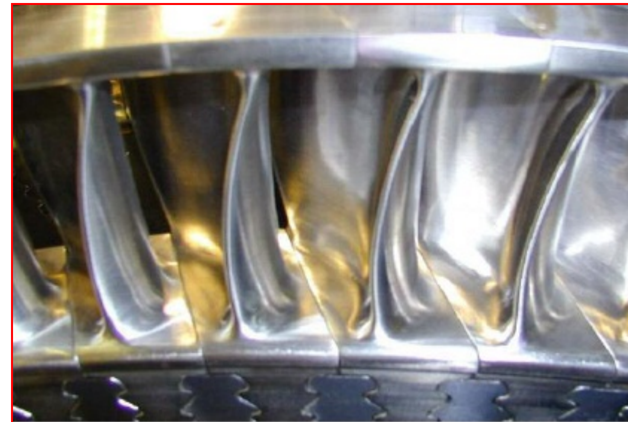
2)
- Discharge increases parabolically
- Output varies linearly
- Power is zero for initial discharge as it is lost in overcoming the friction and inertia effects of the rotating parts



Iso efficiency curves



- The Turbine is tested under a constant head H, for each of several gate openings, and the values of Power output, P and speed, N are reduced to unit conditions



Rotating Blade

STEAM TURBINE

Stationary blade

Unit –IV: Steam Turbines

Steam nozzles:

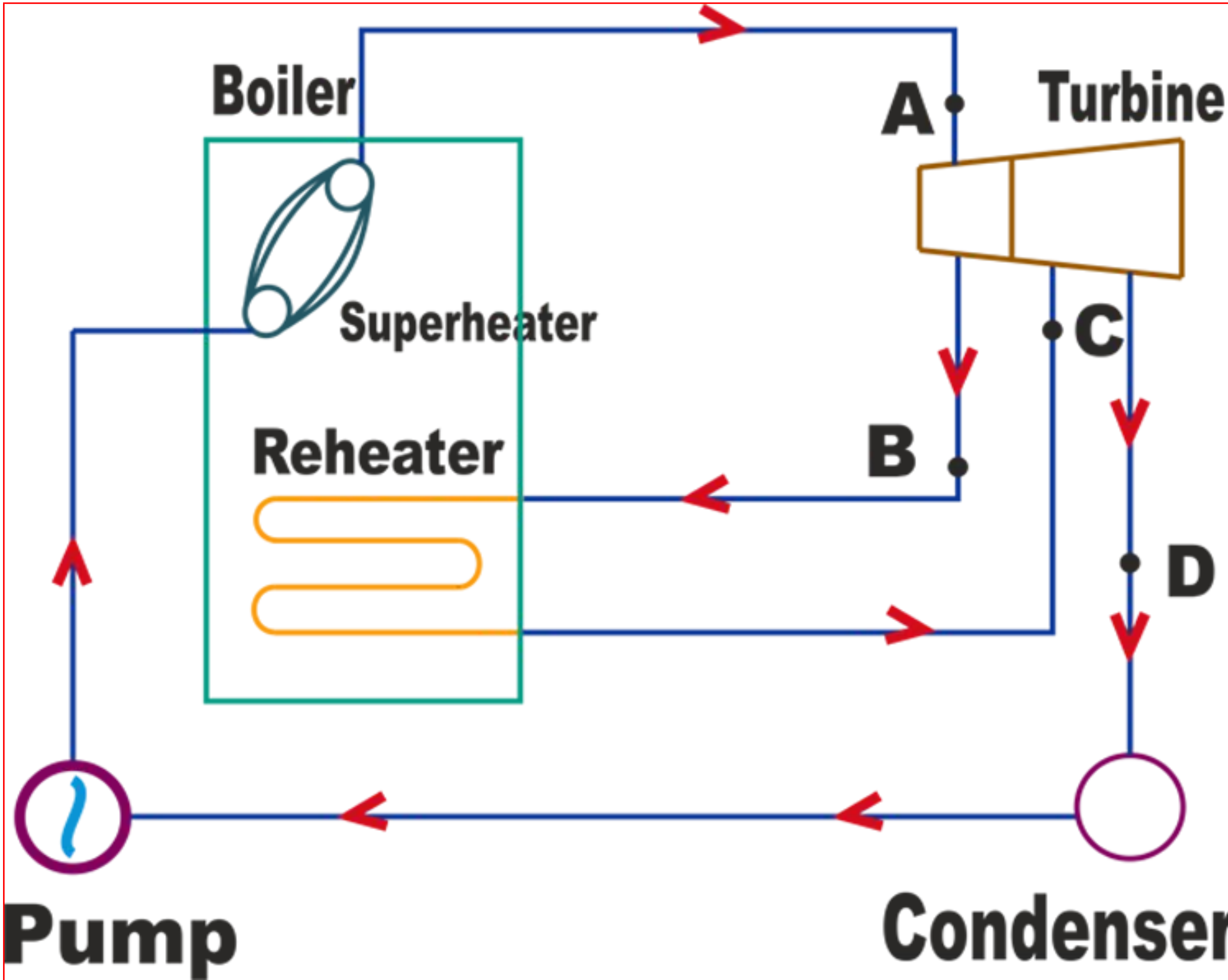
Equation for velocity and mass flow rate [No numerical treatment].

Steam Turbines:

Construction and working of Impulse and Reaction steam turbine,
velocity diagram, work done efficiencies,
Multi-staging,
compounding,
Degree of reaction,
losses in steam turbine,
governing of steam turbines

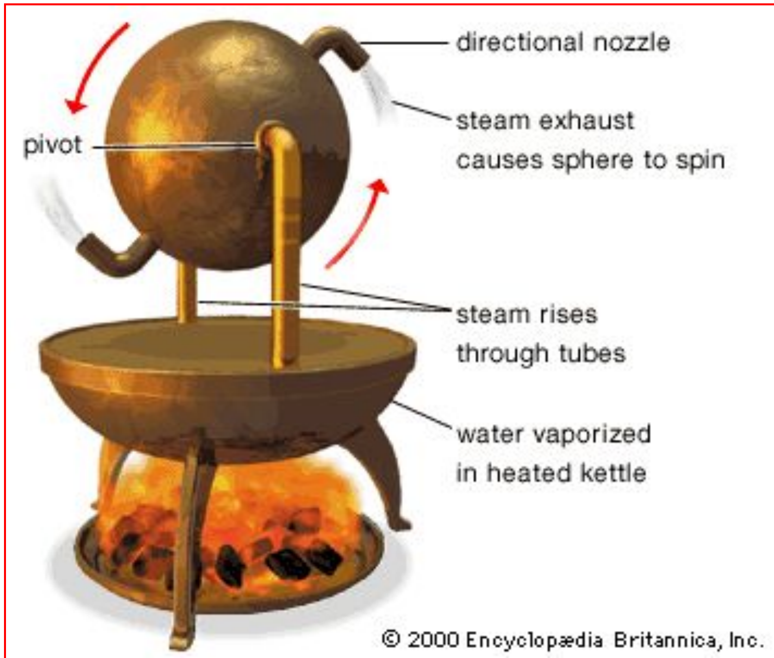
- **Study and Trial on Convergent-Divergent Air/Steam nozzle**
- **Study and Trial on steam Turbine and plotting the operating characteristics.**

Steam Power Plant



Some historical facts

- The first turbine was made by Hero of Alexandria in 120 BC
- In the end of XVIII century the Industrial Revolution began (in 1770 first reciprocating piston steam engine invented by Thomas Newcomen and invented by James Watt started its work)
- The first steam turbines were constructed in 1883 by Dr Gustaf de Laval and in 1884 by sir Charles Parsons
- In1896 Charles Curtis received a patent on impulse turbine



Steam Turbine Definition

- It is a prime mover in which rotary motion of steam is obtained by gradual change of momentum of the steam

Or

- It is a prime mover in which potential energy of steam in converted to kinetic energy and later on is turned is transformed in to mechanical energy i.e . rotation of shaft

Steam Turbine

The steam from the boiler is expanded in a nozzle, resulting in the emission of a high velocity jet. This jet of steam impinges on the moving vanes or blades, mounted on a shaft.

Here it undergoes a change of direction of motion which gives rise to a change in momentum and therefore a force.

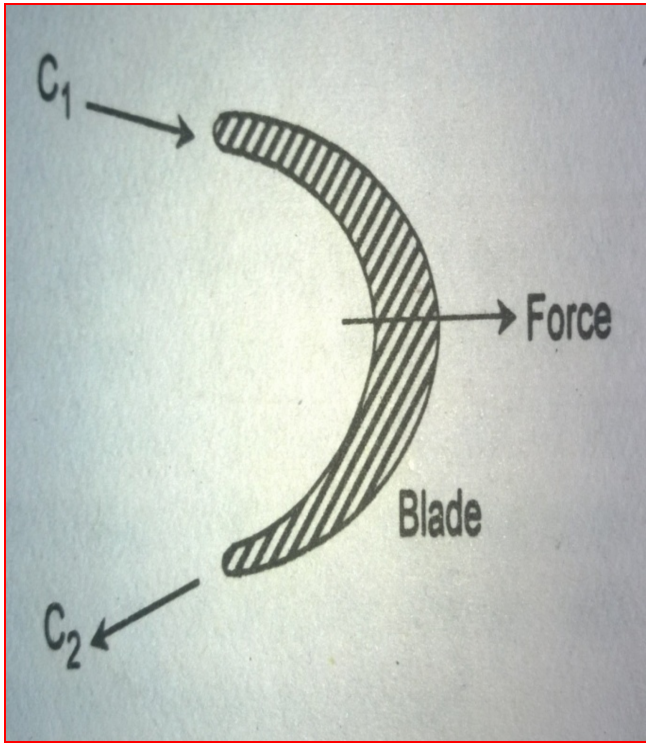
The motive power in a steam turbine is obtained by the rate of change in momentum of a high velocity jet of steam impinging on a curved blade which is free to rotate.

The conversion of energy in the blades takes place by impulse, reaction or impulse reaction principle.

PRINCIPLE OF OPERATION OF STEAM TURBINE

- The steam turbine depends completely upon the dynamic action of the steam.

- According to **Newton's second law of motion**, the force is proportional to the rate of change of momentum (mass x velocity). If the rate of change of momentum is caused in the steam by allowing a high velocity jet steam to pass over curved blade, the steam will impart a force to the blade. If the blade is free, it will move off (rotate) in the direction of force.



- In fig. C1 = Initial velocity

  C2 = Final velocity

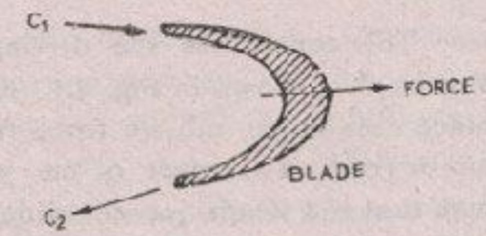
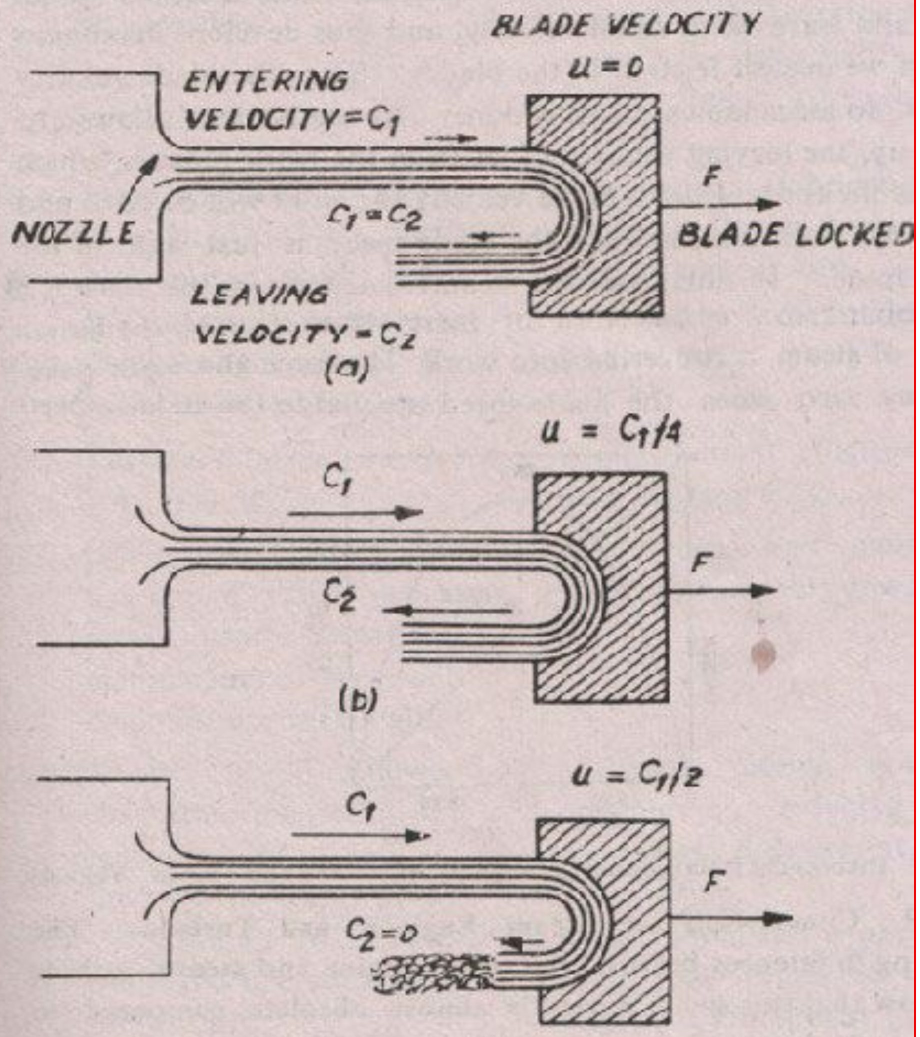
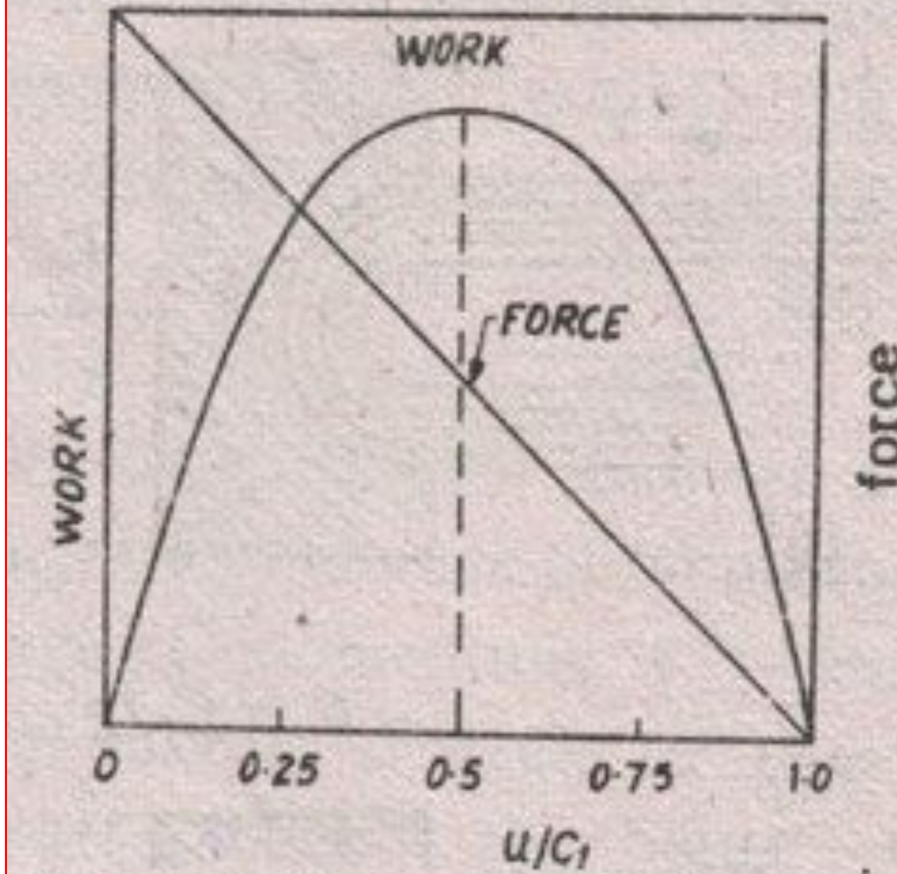


Fig. 4.1 (i) Principle of Working





Theoretical Relationship between Work, Force and Blade Velocity

The relationship between work, force and blade velocity

Classification of Steam Turbine

- **Based on action of steam**
  Impulse
  Reaction
  Combination of impulse and reaction
- **Based on number of pressure stages**
  Single stage
  Multi stage Impulse and Reaction turbines
- **Based on Direction of steam flow**
  Axial turbines
  Radial turbines
- **Based on no of cylinders**
  Single
  Double
  Three
  Four

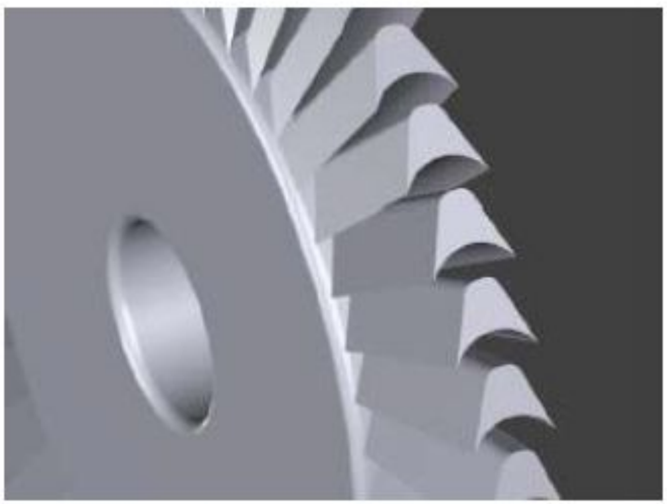
Main Classification Of Steam Turbine

- **Impulse Turbine**
- **Impulse-reaction Turbine or Reaction Turbine**

Types of Steam Turbine

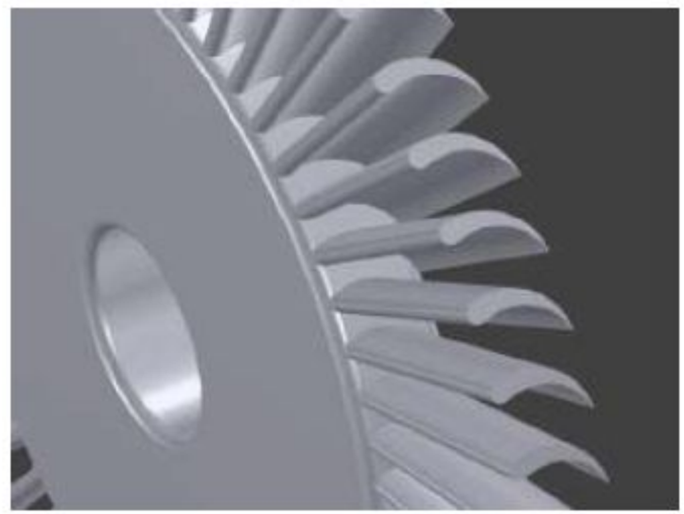
PEMP
RMD 2501

Impulse Turbine



Process of complete expansion of steam takes place in stationary nozzle and the velocity energy is converted into mechanical work on the turbine blades.

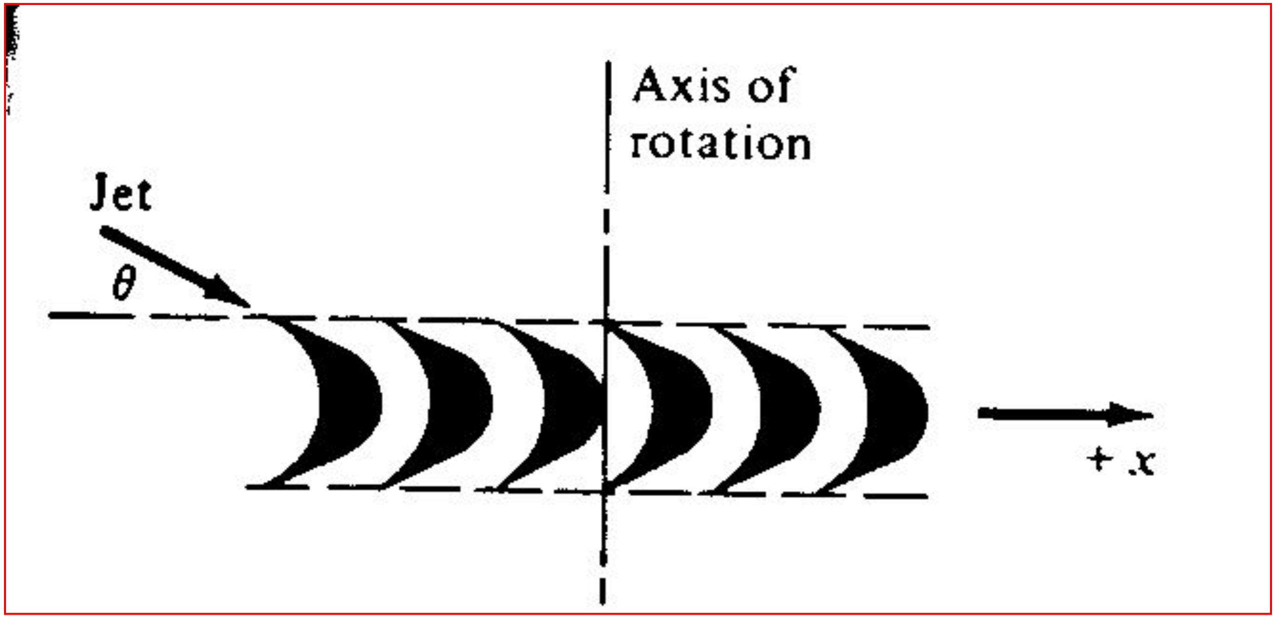
Reaction Turbine



Pressure drop with expansion and generation of kinetic energy takes place in the moving blades.

Impulse Principle

It is impossible to have 180^0 curved blade in actual application jet exit will impinging on the back of next blade . Blade entrance angle and blade exit angle cannot be zero, as shown in the figure below

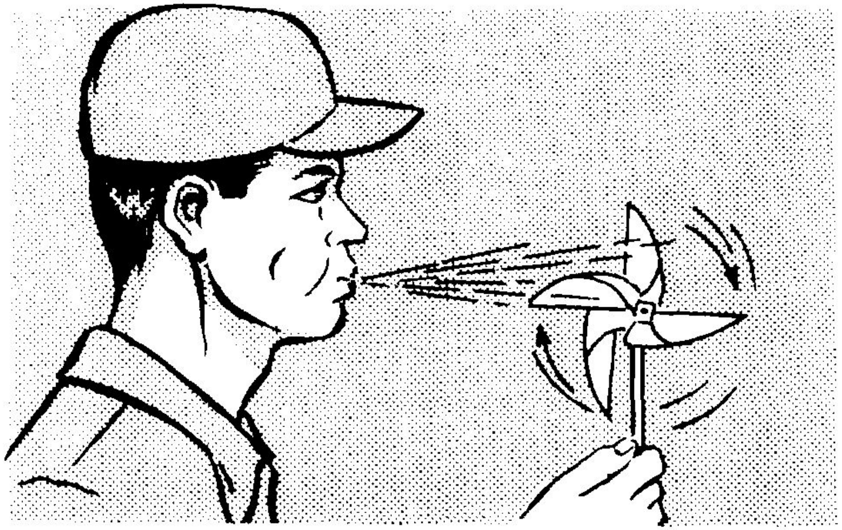


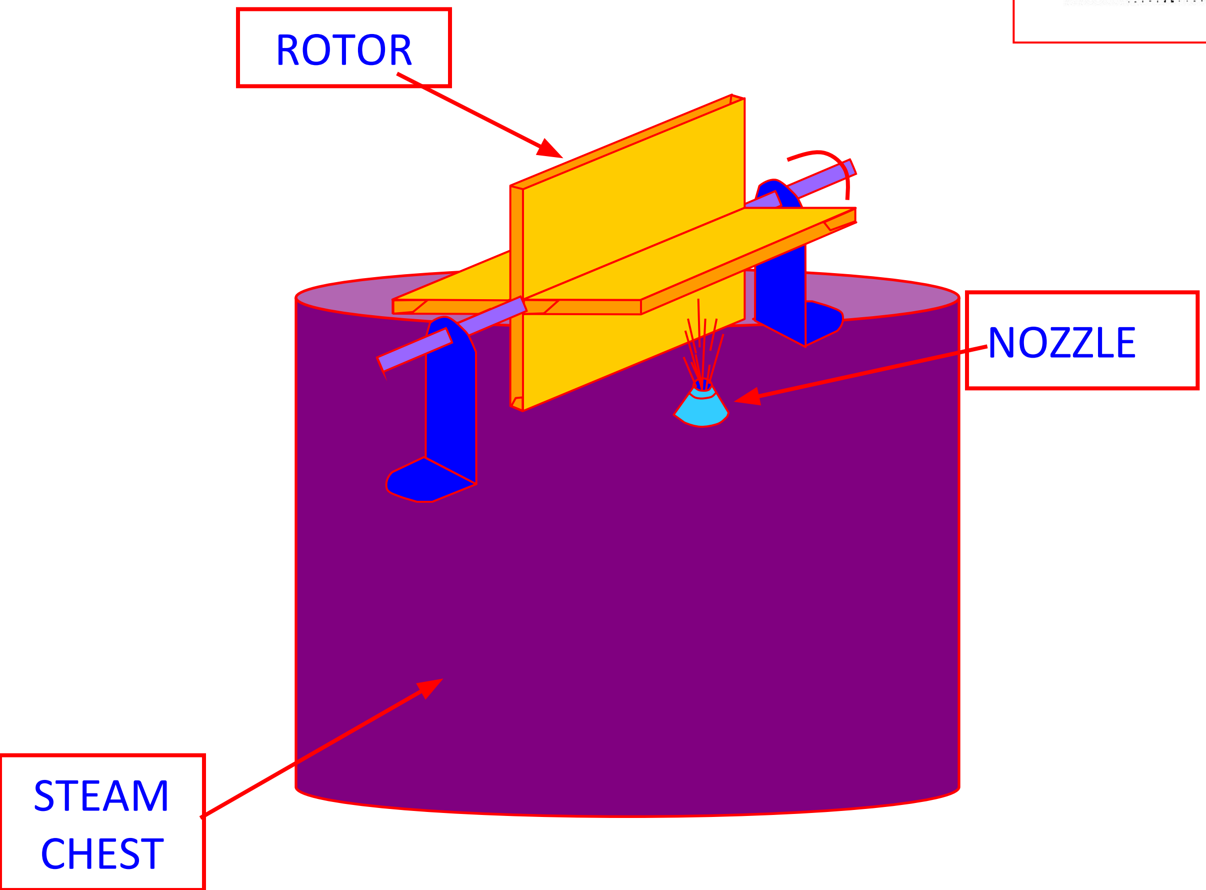
TYPES OF IMPULSE TURBINE

Simple Impulse Turbine.

Compounding of Impulse Turbine.

IMPULSE TURBINE PRINCIPLE





Simple Impulse turbine

- In Impulse turbine, the enthalpy drop (pressure drop) completely occurs in the nozzle itself and when the fluid pass over the moving blades it will not suffer pressure drop again.
- Hence pressure remain constant when the fluid pass over the rotor blades. Fig. shows the schematic diagram of Impulse turbine.

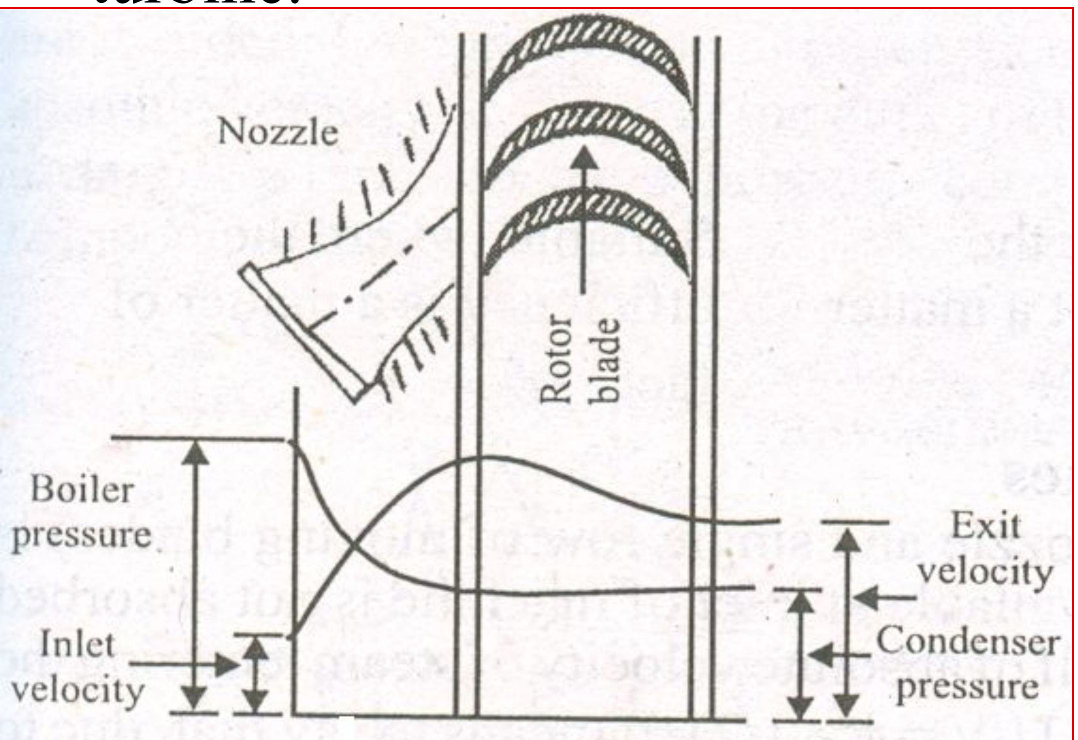
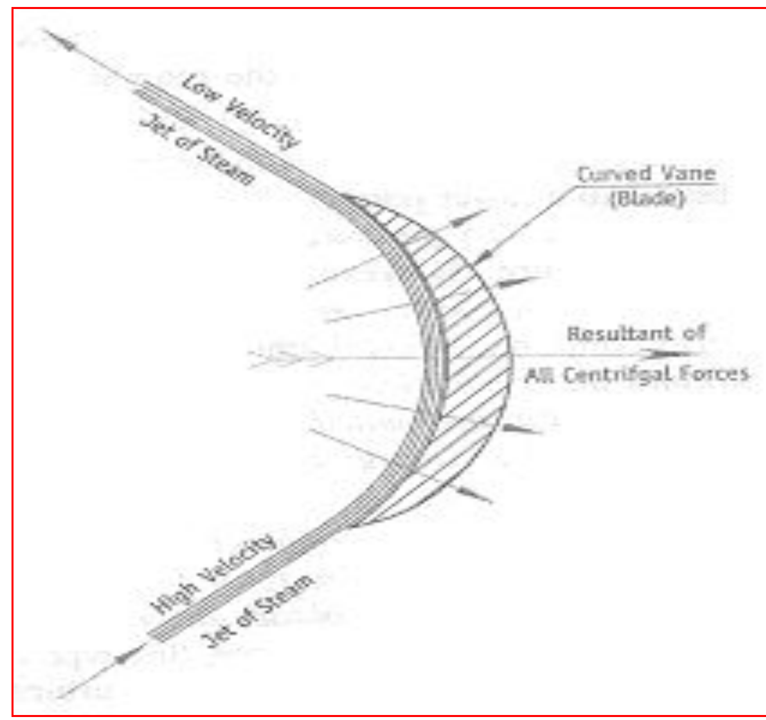
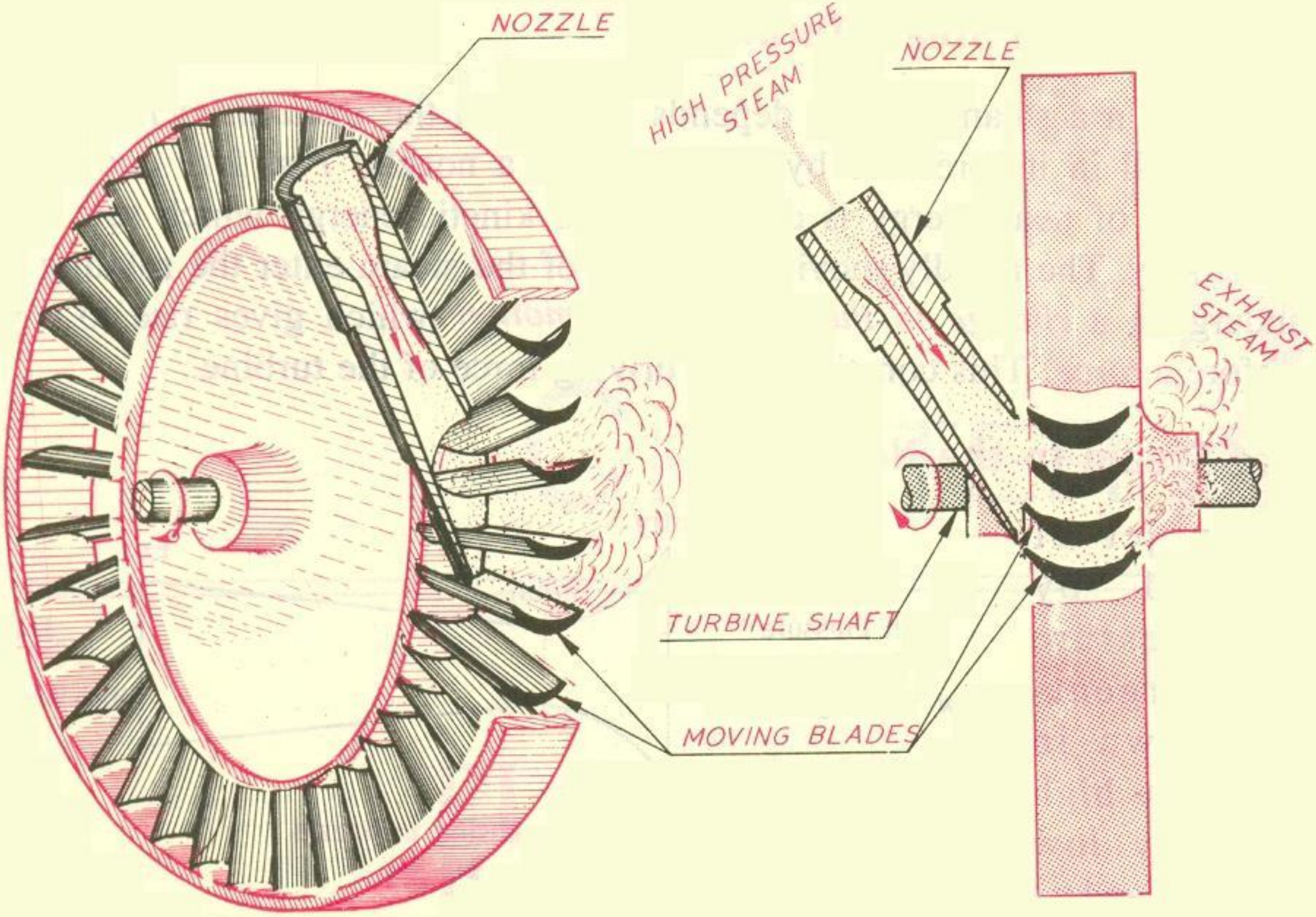


Fig. Impulse turbine

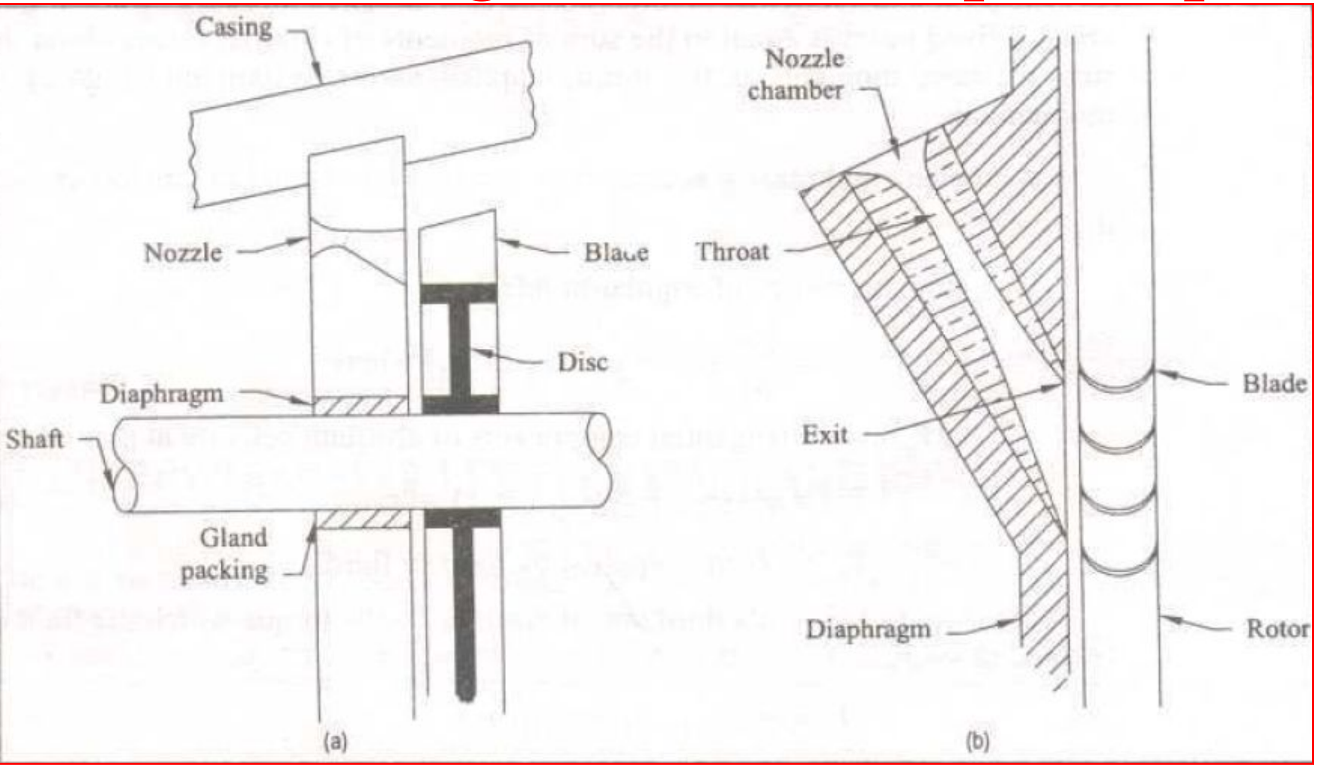


- The basic idea of an impulse turbine is that a jet of steam from a fixed nozzle pushes against the rotor blades and impels them forward. The velocity of steam **is twice as fast** as the velocity of blade.
- Blade is usually symmetrical.
- Entrance angle (φ) and exit angle (γ) are around 20°.
- Enthalpy drop and pressure drop occur in the nozzle.
- It means that **the pressure remain constant** through turbine.
- Single stage impulse turbine is called as De laval Turbine
- A De-lavel turbine named after Swedish Engineer De-lavel is the simplest impulse turbine and is commonly used.



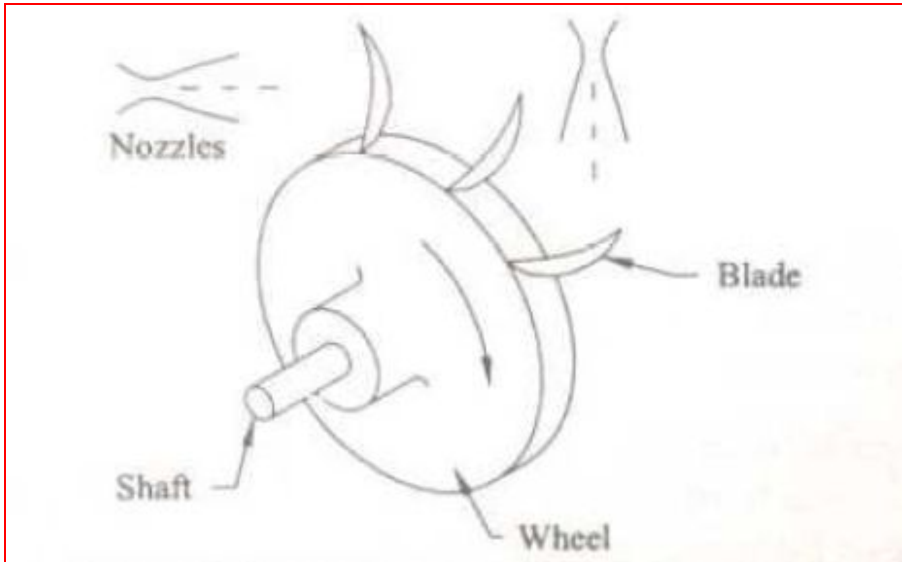
IMPULSE TURBINE

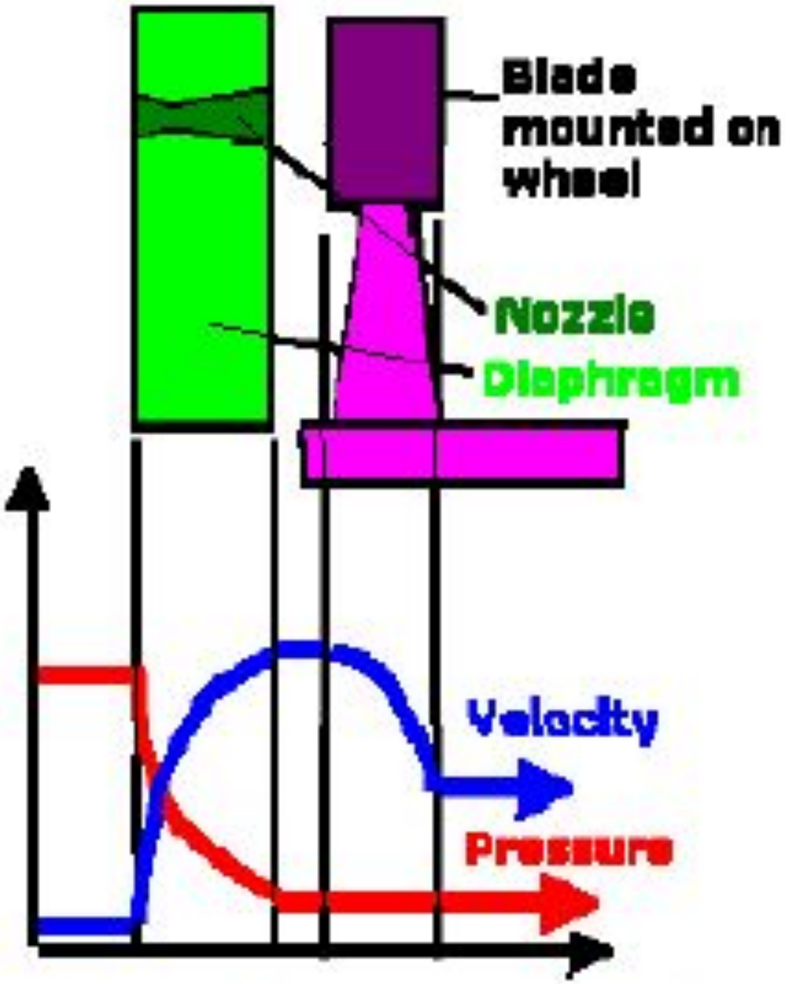
Arrangement of Simple Impulse Turbine



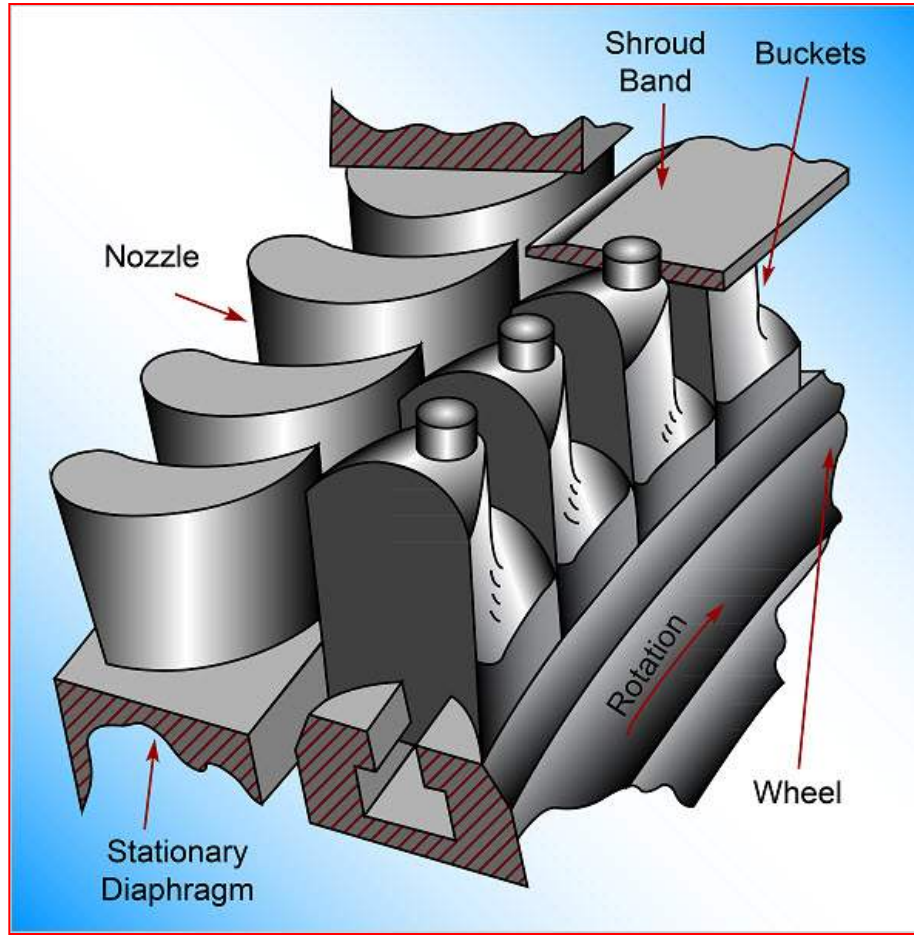
The essential parts of an impulse turbine are –

nozzles
blades
casing.





De Laval Impulse Turbine.



COMPOUNDING OF IMPULSE TURBINE

- Compounding is a method for reducing the rotational speed of the impulse turbine to practical limits.
- If the high velocity of stream is allowed to flow through one row of moving blades, it produces a rotor speed of about **30,000 r.p.m**. which is too high for practical use.
- Leaving loss is also very high. It is therefore essential to incorporate some improvement in the simple impulse turbine for possible by making use also to achieve high performance.

- This is possible by making use of more than one set of nozzles, blades, rotors, in a series keyed to a common shaft, so that either the steam pressure or the jet velocity is absorbed by the turbine in stages. The leaving loss also will than be less.

  This process is called compounding of steam turbine.

Compounding is employed for reducing the rotational speed of the impulse turbine to practical limits.

WHY COMPOUNDING IS REQUIRED?

- The steam produced in the boiler has got very high enthalpy. In all turbines the blade velocity is directly proportional to the velocity of the steam passing over the blade.

- Now, if the entire energy of the steam is extracted in one stage, i.e. if the steam is expanded from the boiler pressure to the condenser pressure in a single stage, then its velocity will be very high.

Hence the velocity of the rotor (to which the blades are keyed) can reach to about 30,000 rpm, which is pretty high for practical uses.

Moreover at such high speeds the centrifugal forces are immense, which can damage the structure. Hence, compounding is needed.

TYPE OF COMPOUNDED IMPULSE TURBINE

Pressure Compounded Impulse Turbine

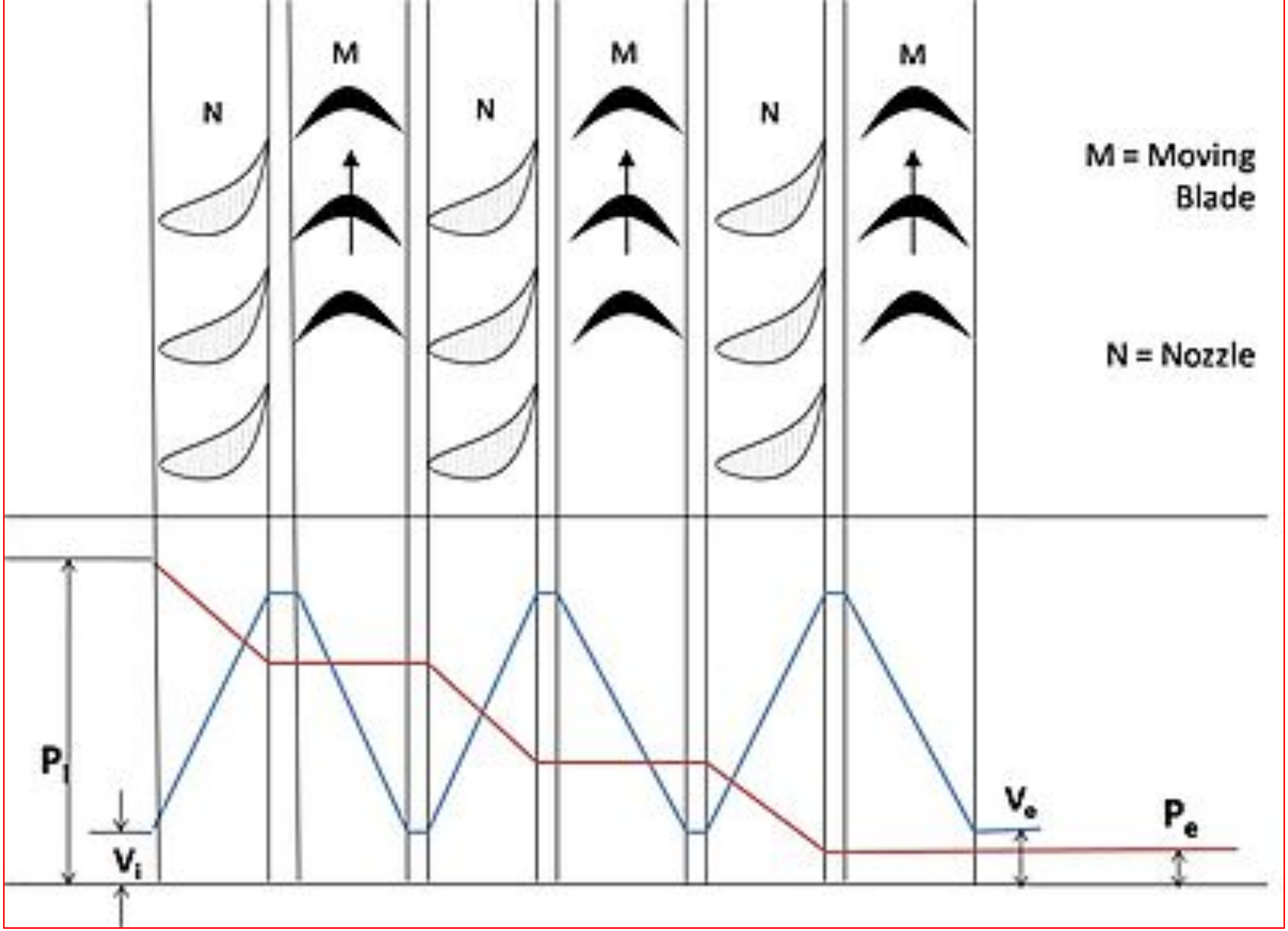
Velocity Compounded Impulse Turbine

Pressure and Velocity Compounded Impulse Turbine

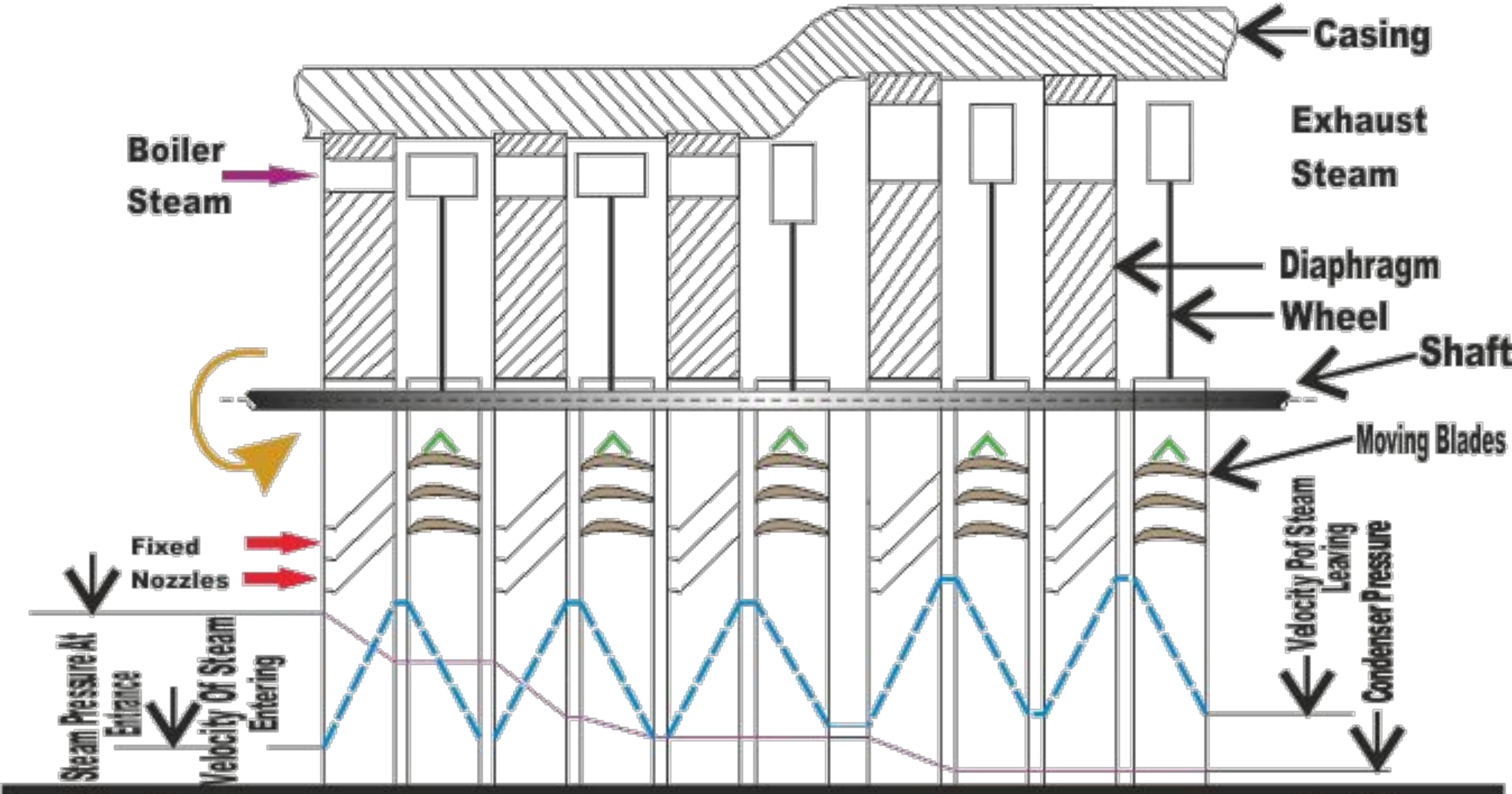
PRESSURE COMPOUNDED IMPULSE TURBINE

- In this type of turbine, the compounding is done for pressure of steam only i.e. to reduce the high rotational speed of turbine the whole expansion of steam is arranged in a number of steps by employing a number of ***simple turbine*** in a series keyed on the same shaft.
- Each of these simple impulse turbine consisting of one set of ***nozzles and 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 the whole pressure*** drop from the ***steam chest pressure to the condenser pressure*** into a series of smaller pressure drop across several stages of impulse turbine and hence this turbine is called Pressure – compound impulse turbine
- **THIS IS ALSO KNOWN AS RATEAU TURBINE**

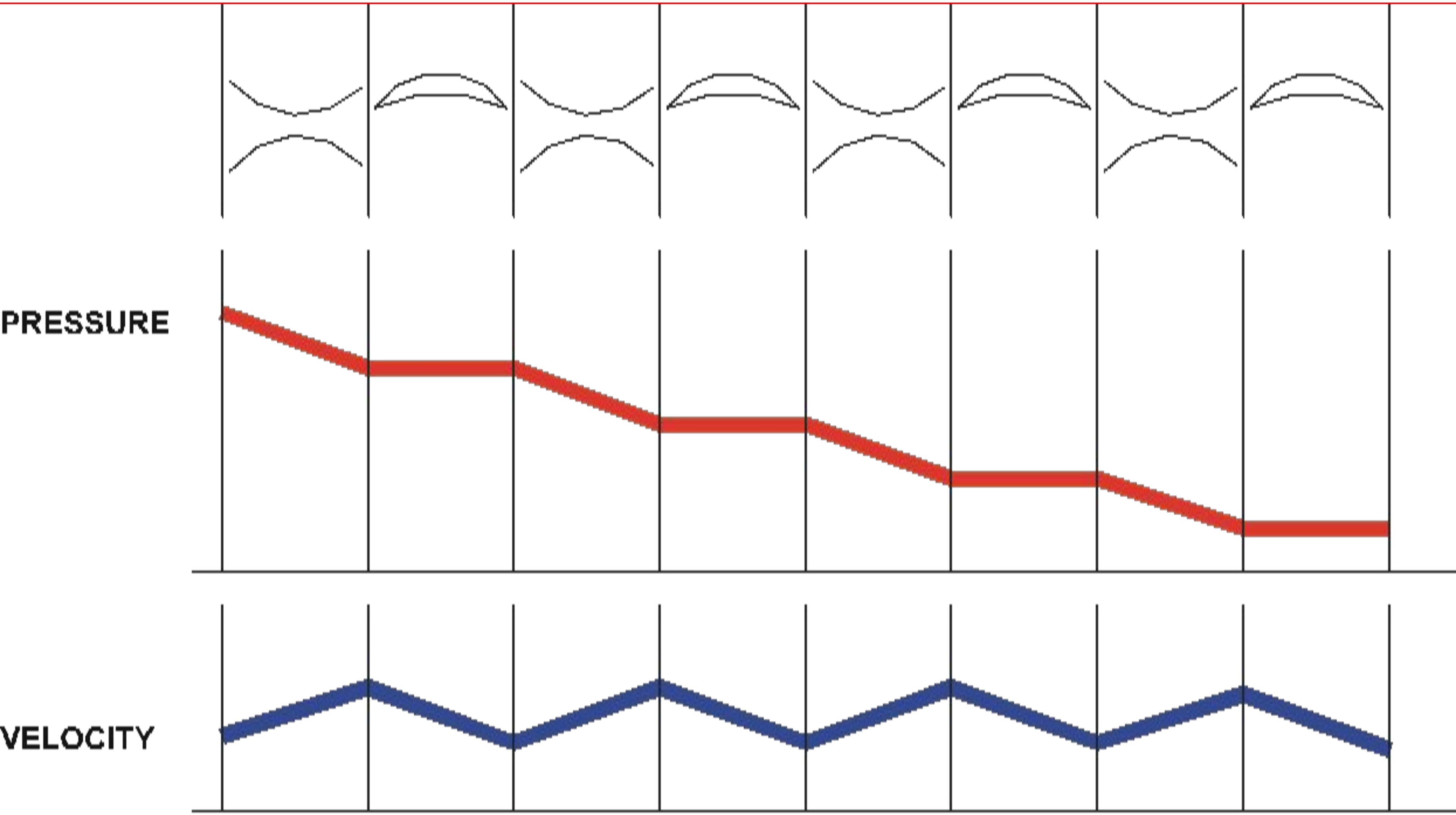


Pressure Compounded Impulse Turbine



Pressure Compounding

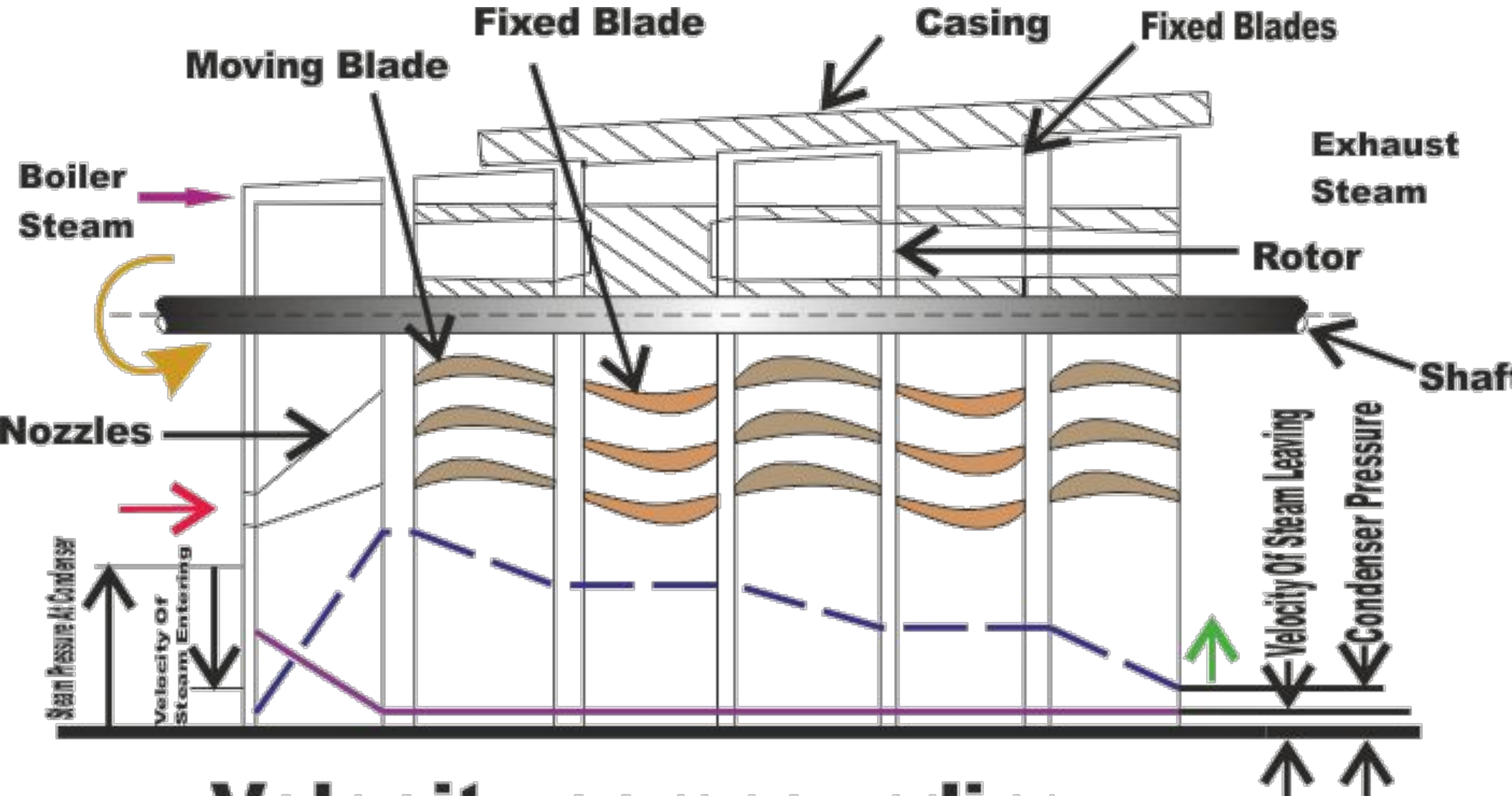
PRESSURE COMPOUNDED TURBINE



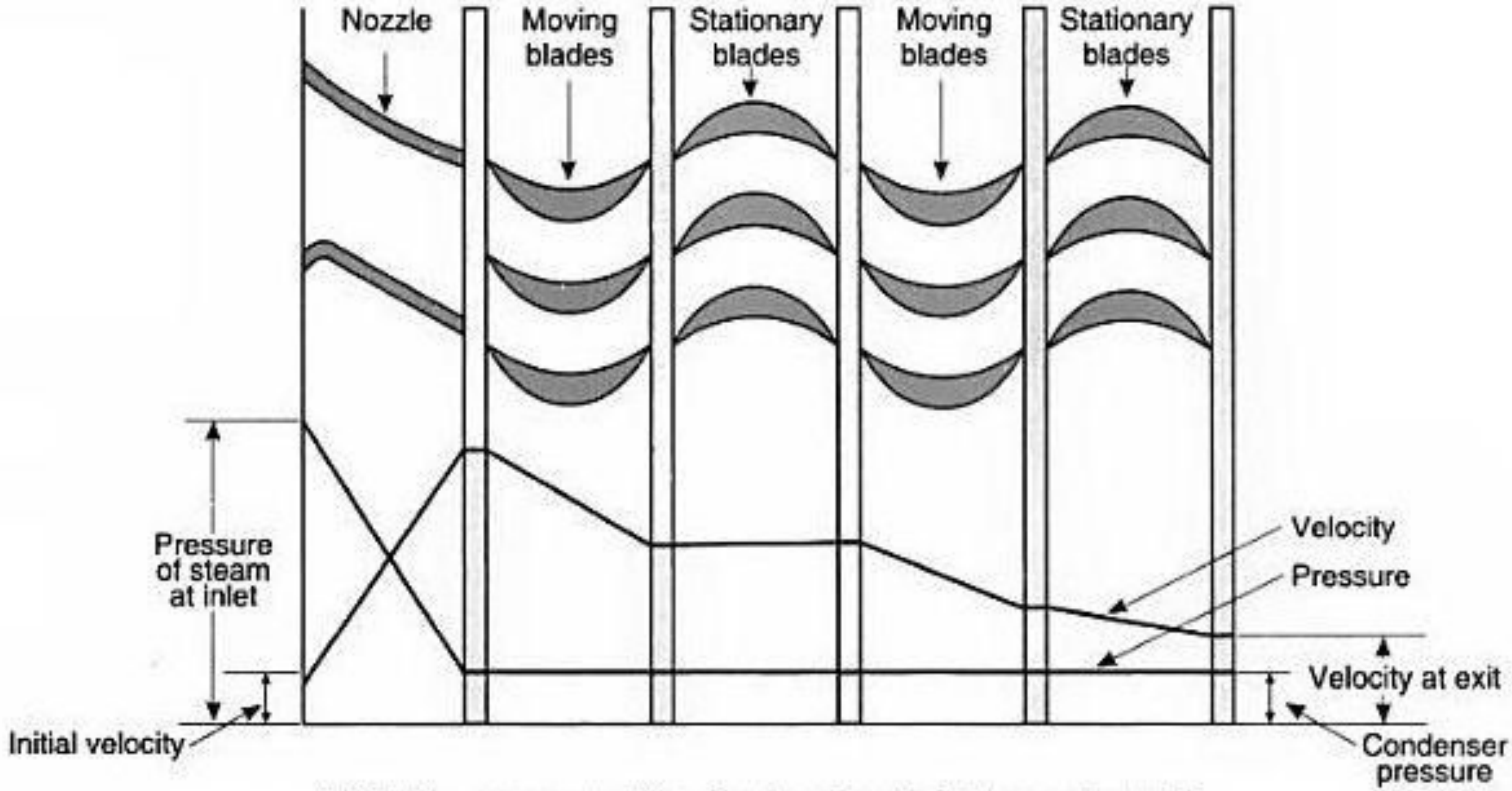
VISUALIZATION OF A PRESSURE COMPOUNDED TURBINE

VELOCITY COMPOUNDED IMPULSE TURBINE

- In this type of turbine, the compounding is done for velocity of steam only i.e. drop in velocity is arranged in many small drops through ***many moving row of blades instead of a single row of moving blades***.
- In this turbine, moving and fixed blades (***guide blades***) are placed alternately.
- ***Moving blades are fitted with the wheel*** while ***the fixed blades are fitted with the casing***.
- The whole expansion of steam from the steam chest pressure down to the exhaust pressure take place in the nozzles only.
- Therefore pressure remain constant after nozzle. **IT IS ALSO KNOWN AS CURTIS TURBINE.**

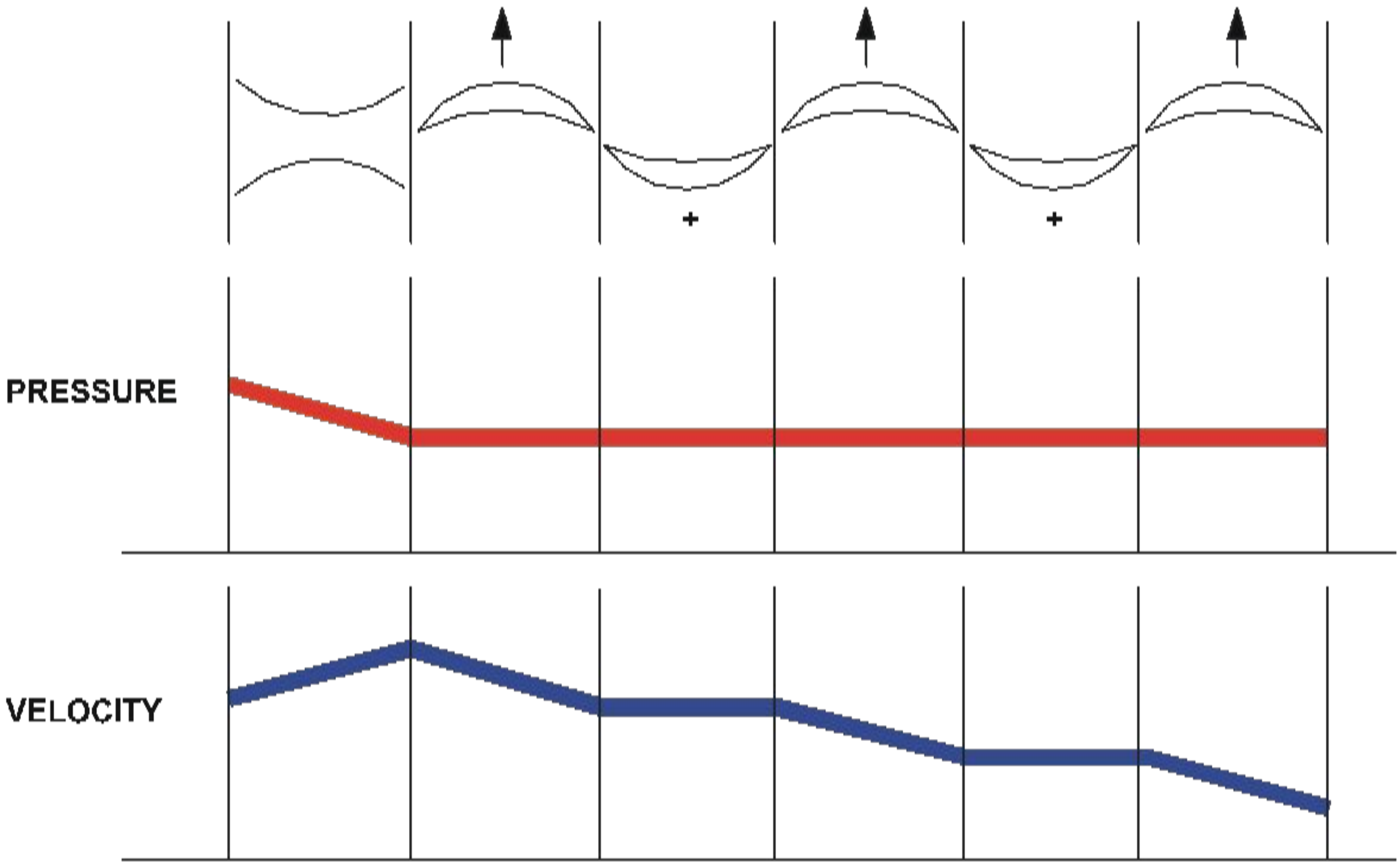


Velocity compounding



Velocity compounding in impulse turbine: principal.

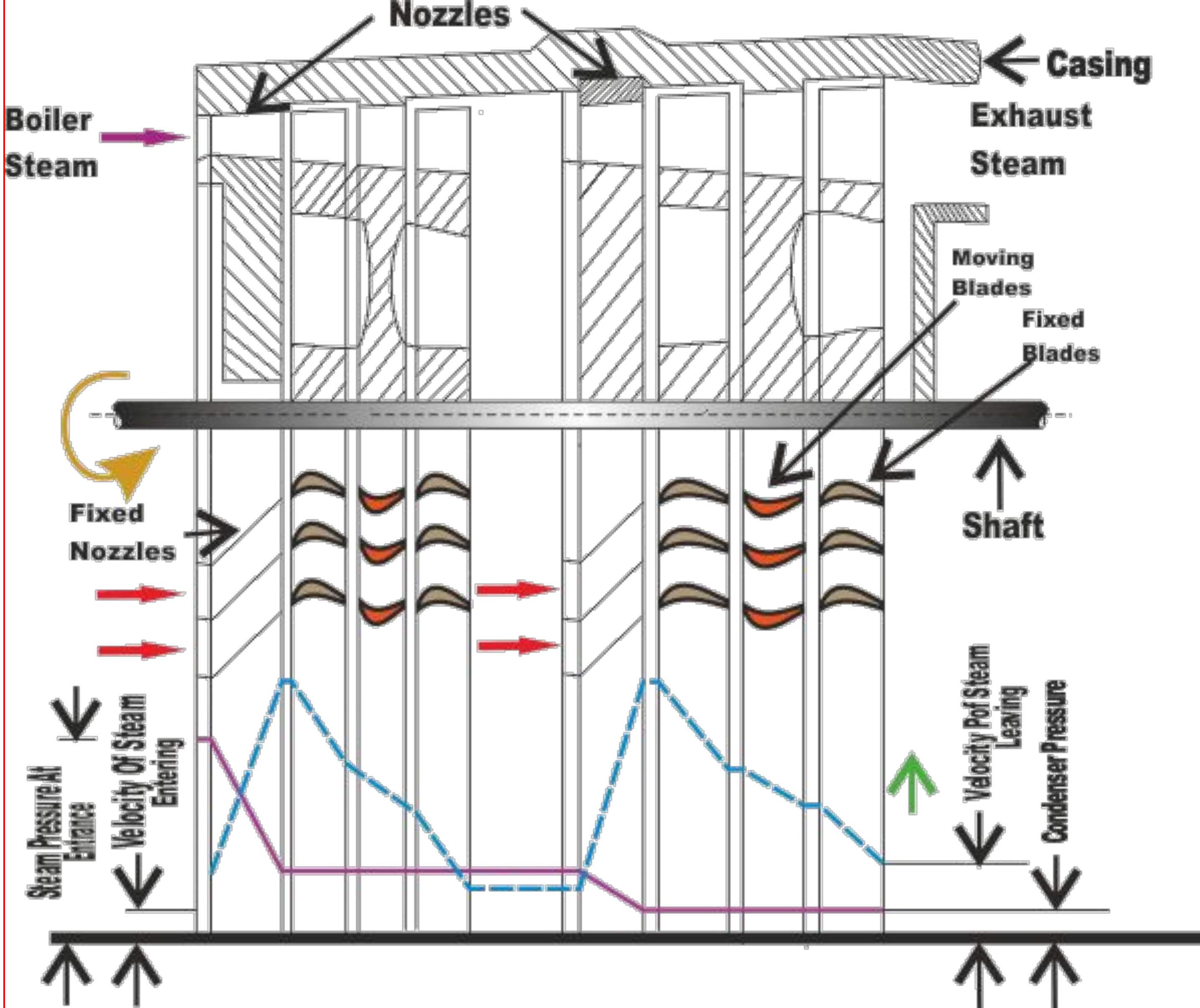
VELOCITY COMPOUNDED TURBINE



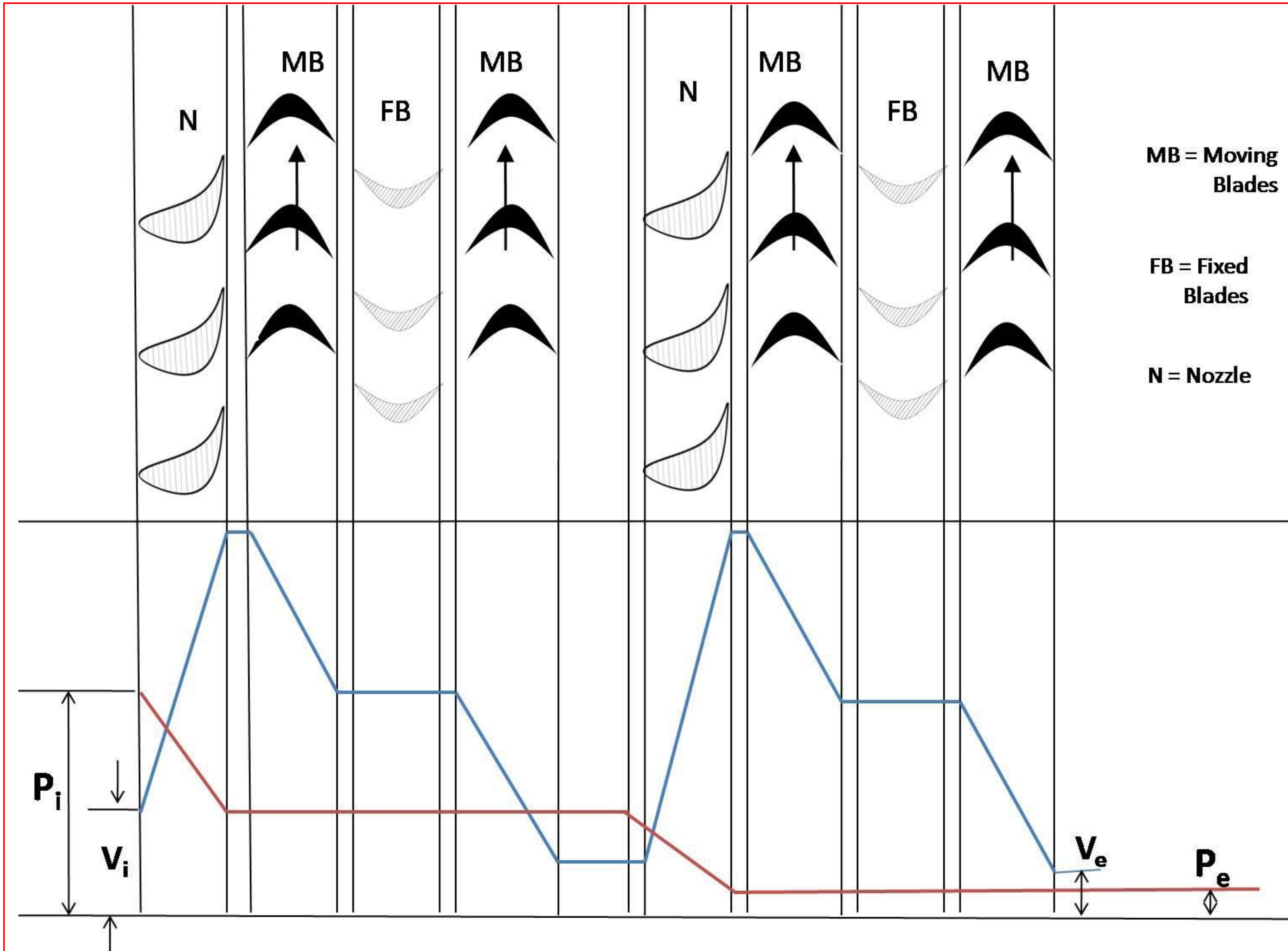
VISUALIZATION OF A VELOCITY COMPOUNDED TURBINE

PRESSURE-VELOCITYCOMPOUNDED IMPULSE TURBINE

- Here we consider ***pressure and velocity simultaneously***, hence it is named as pressure velocity compounded impulse turbine.
- Here we have arrangement of ***two rotors*** or wheels in each rotars we have installation of ***many row of moving blades, hence we get a decrease in velocity***.
- Also we install ***two nozzles*** in whole arrangement, which helps in ***splitting pressure***.
- Here we use two row wheels as we consider ***two row wheels arrangement more efficient than three row wheels arrangement***.

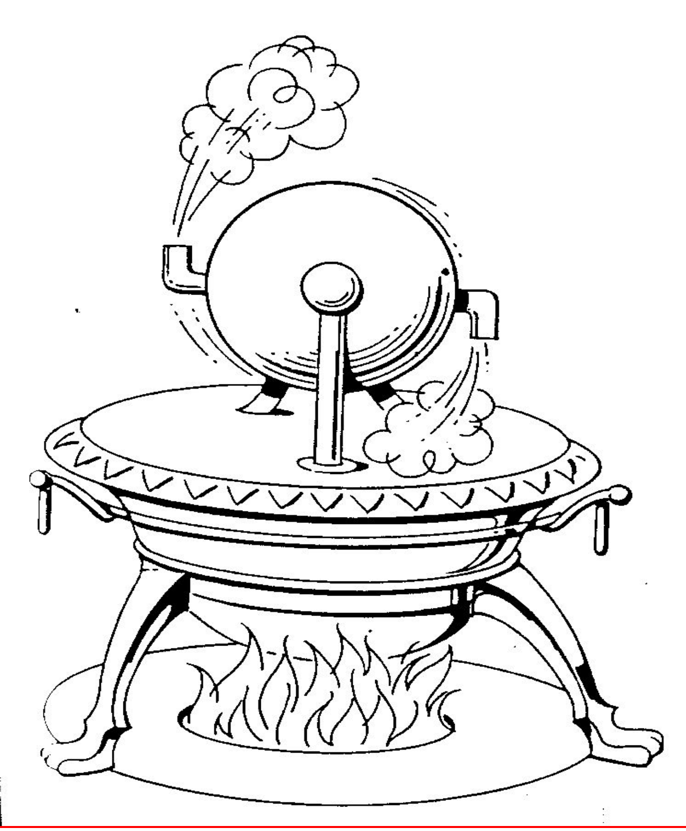


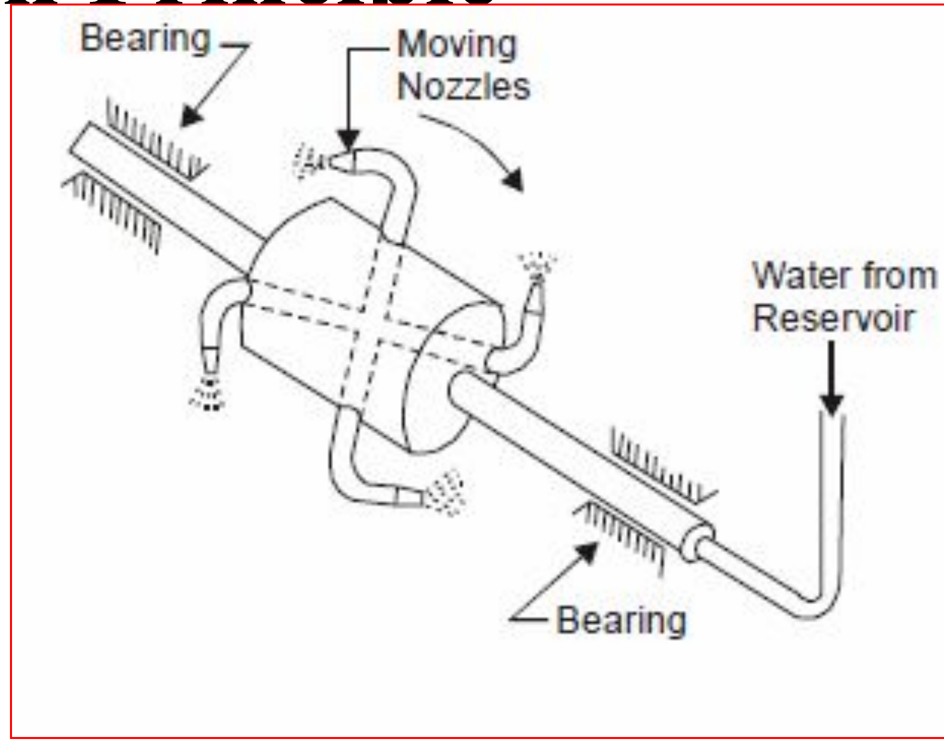
Pressure Velocity Compounding

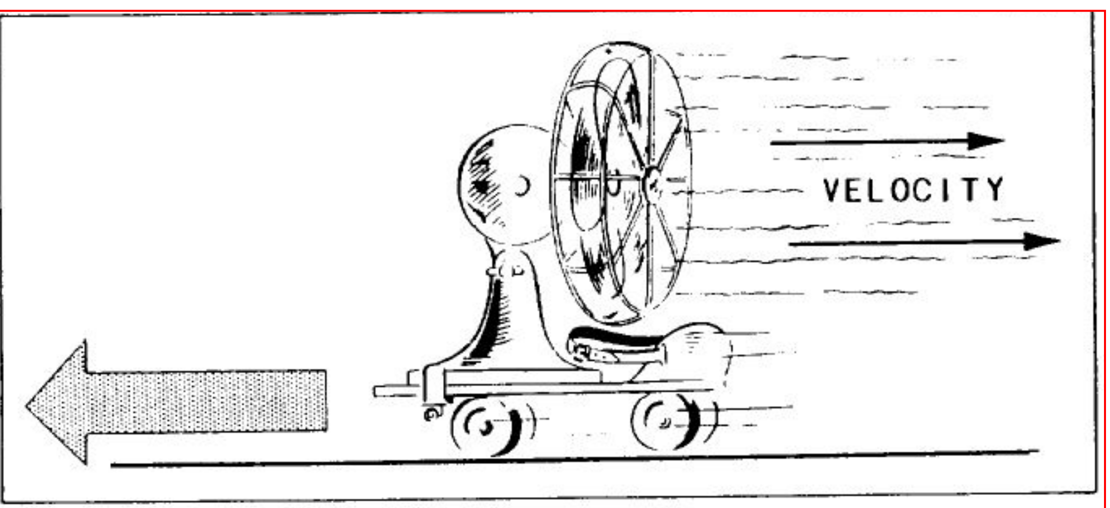


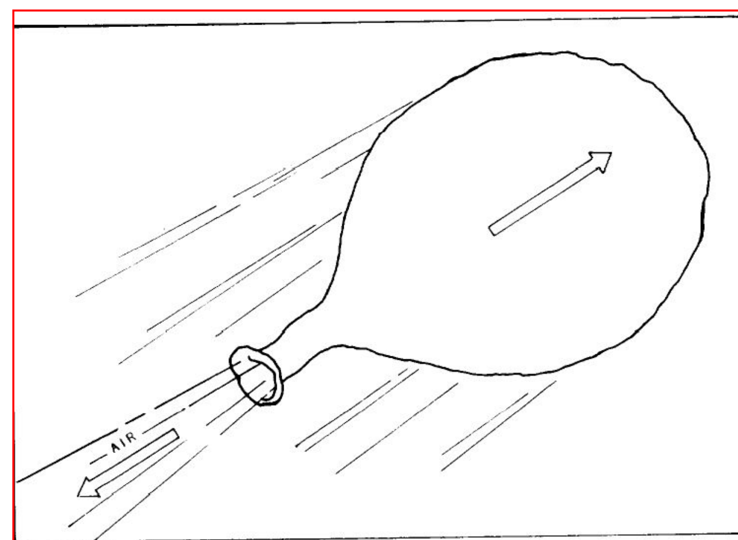
N
MB
FB
MB
N
MB
FB
MB
MB = Moving Blades
FB = Fixed Blades
N = Nozzle
P_i
V_i
V_e
P_e

Reaction Principle

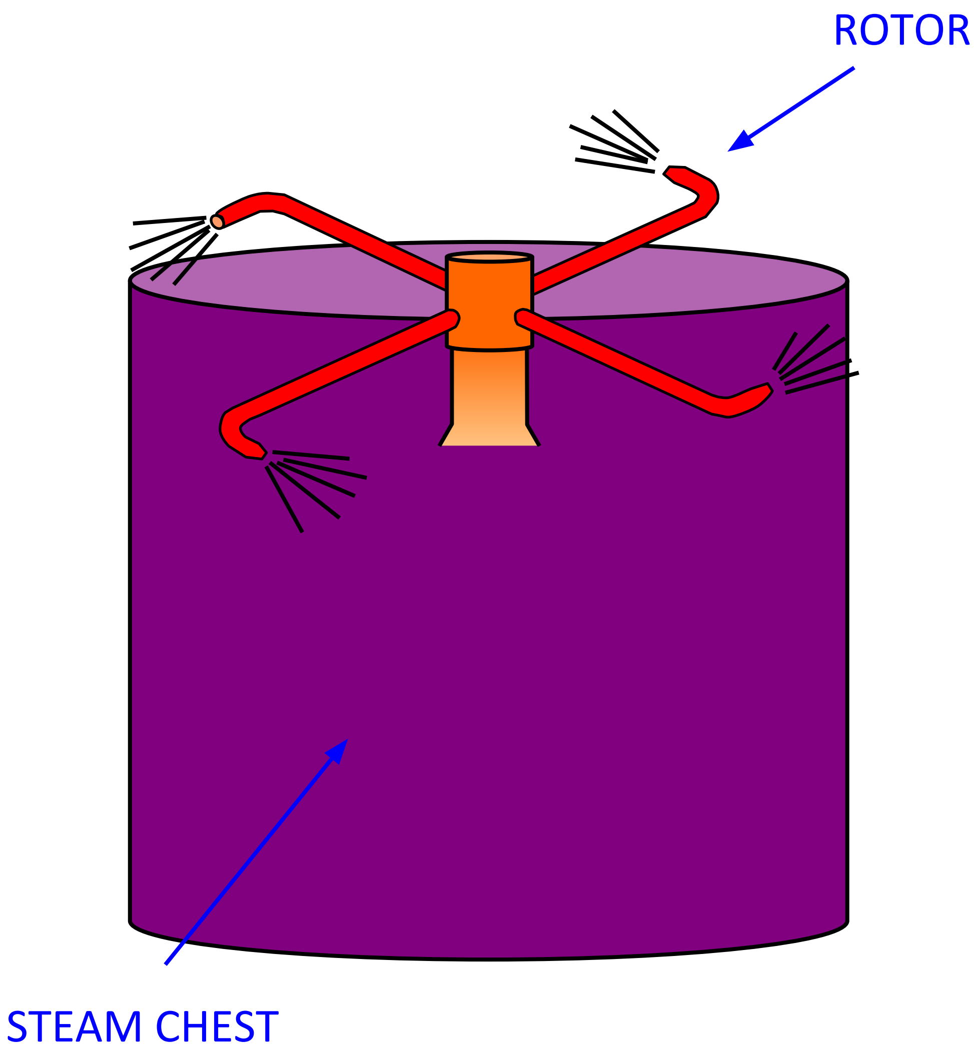


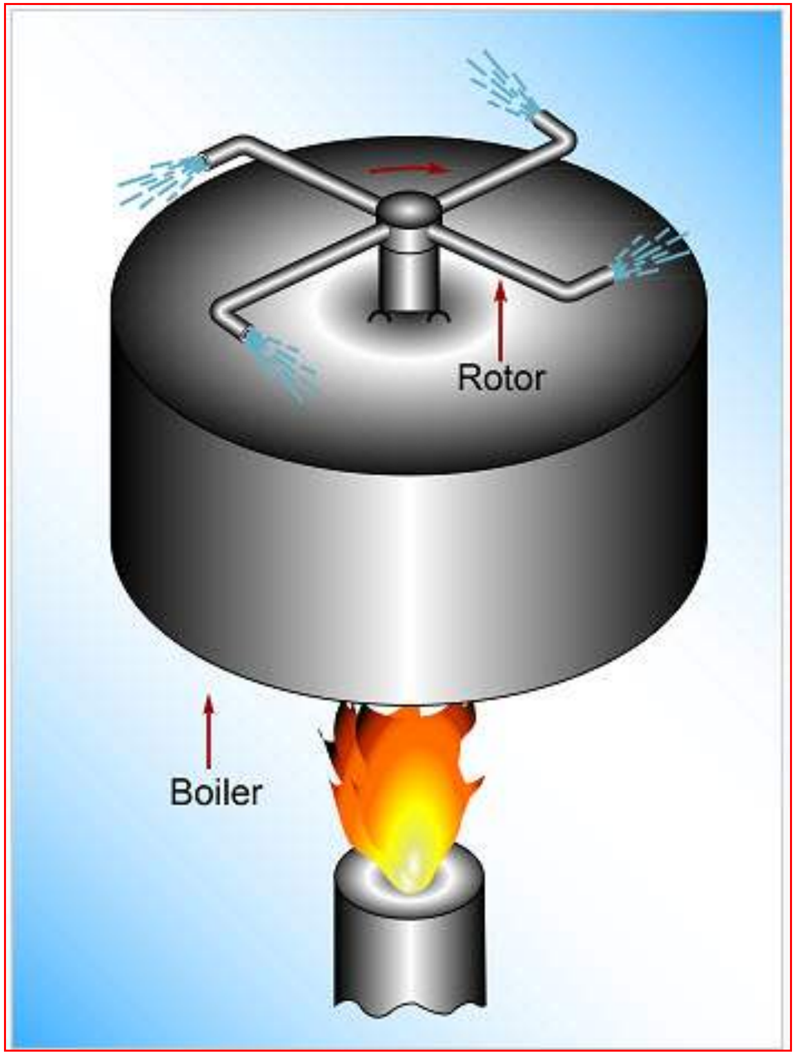






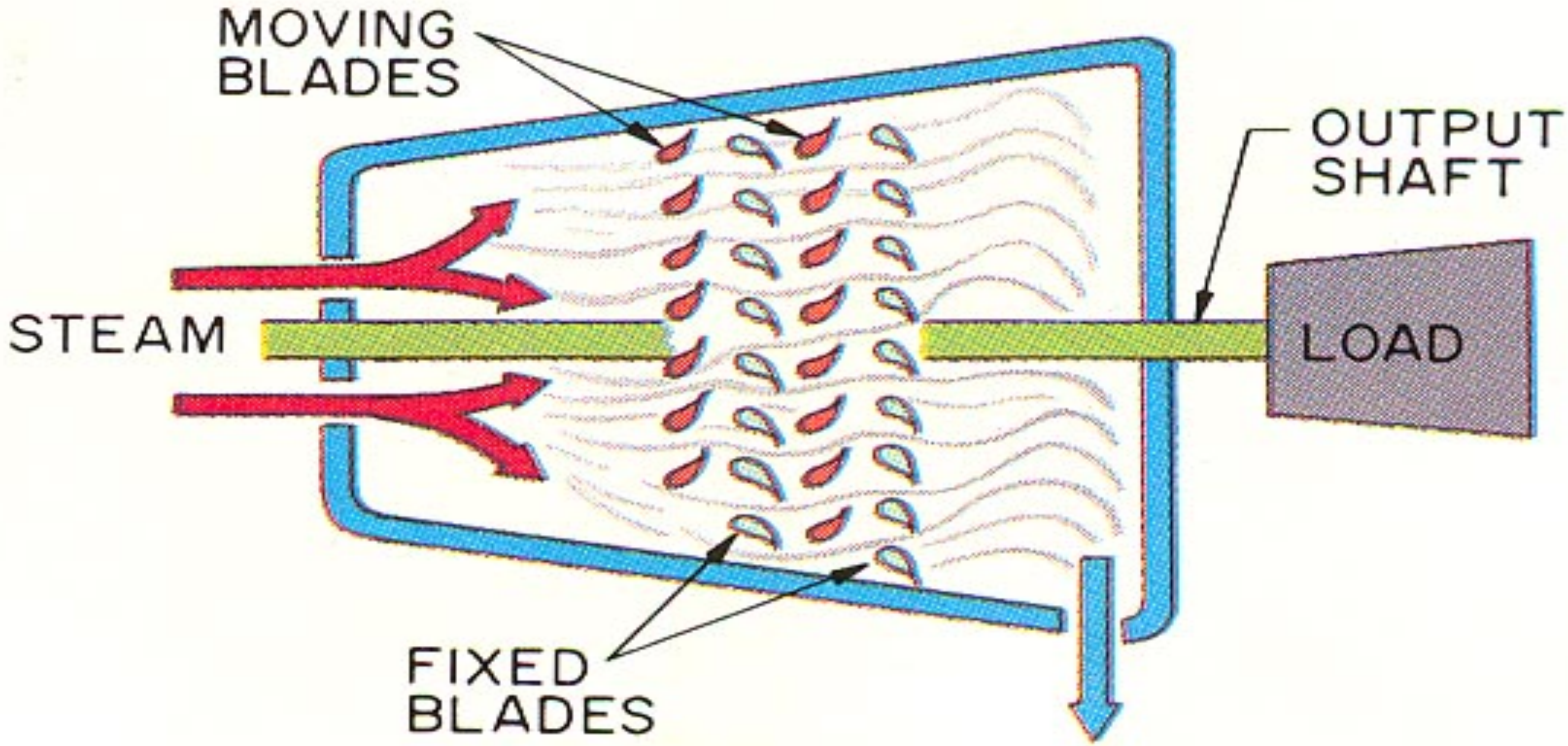
REACTION TURBINE PRINCIPLE





REACTION TURBINE PRINCIPLE

In a reaction turbine, the steam increases in speed as it passes through the moving blades. The fixed blades direct the steam to other moving blades.



Reaction Turbine

In this type of turbine, there is a gradual pressure drop takes place continuously over the fixed and moving blades.

The rotation of the shaft and drum, which carrying the blades is the result of both impulse and reactive force in the steam.

The reaction turbine consist of a row of stationary blades and the following row of moving blades .

The fixed blades act as a nozzle which are attached inside the cylinder and the moving blades are fixed with the rotor as shown in figure

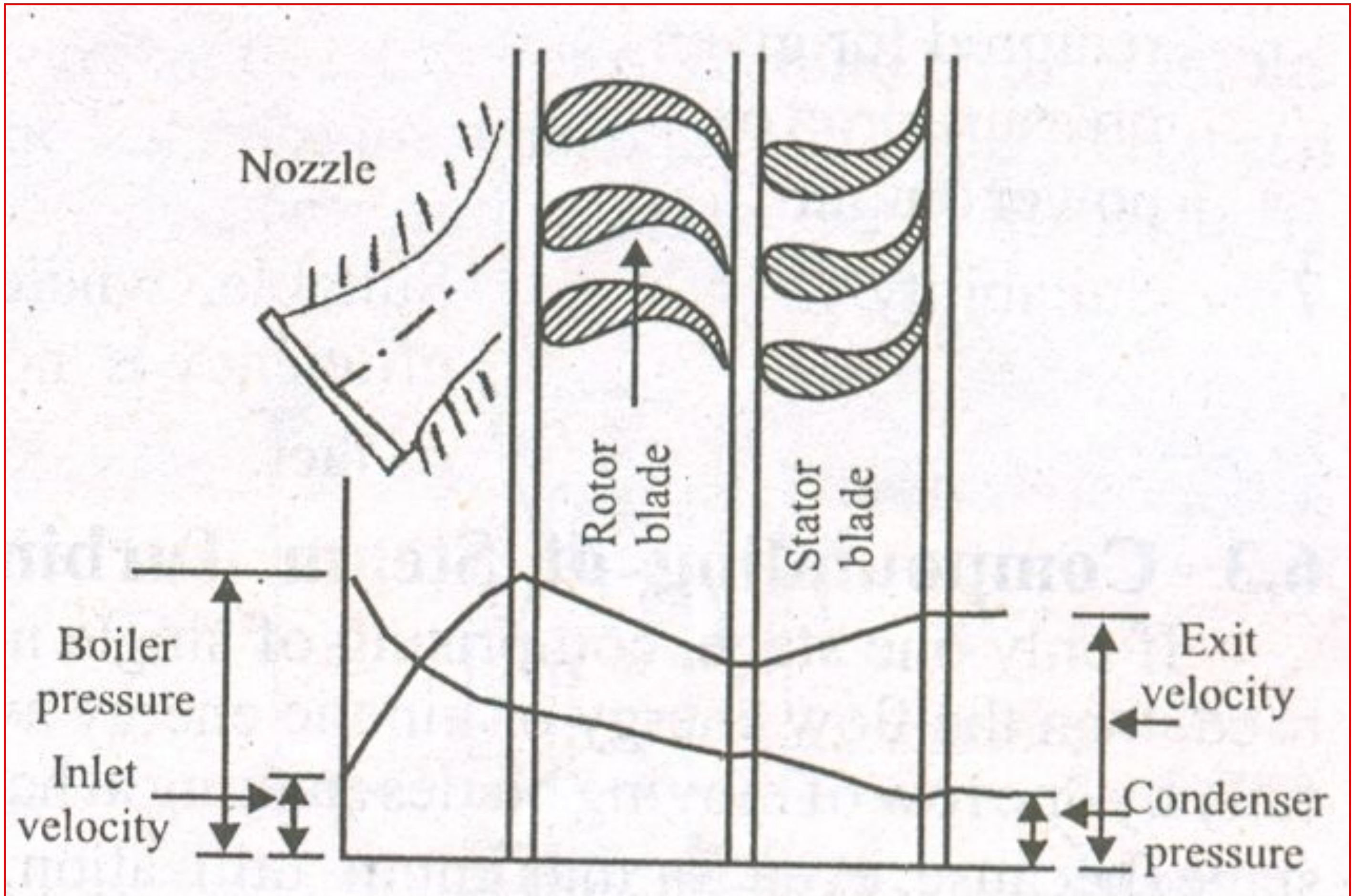
When the steam expands over the blades there is gradual increase in volume and decrease in pressure. But the velocity decrease in the moving blades and increases in fixed blades with change of direction.

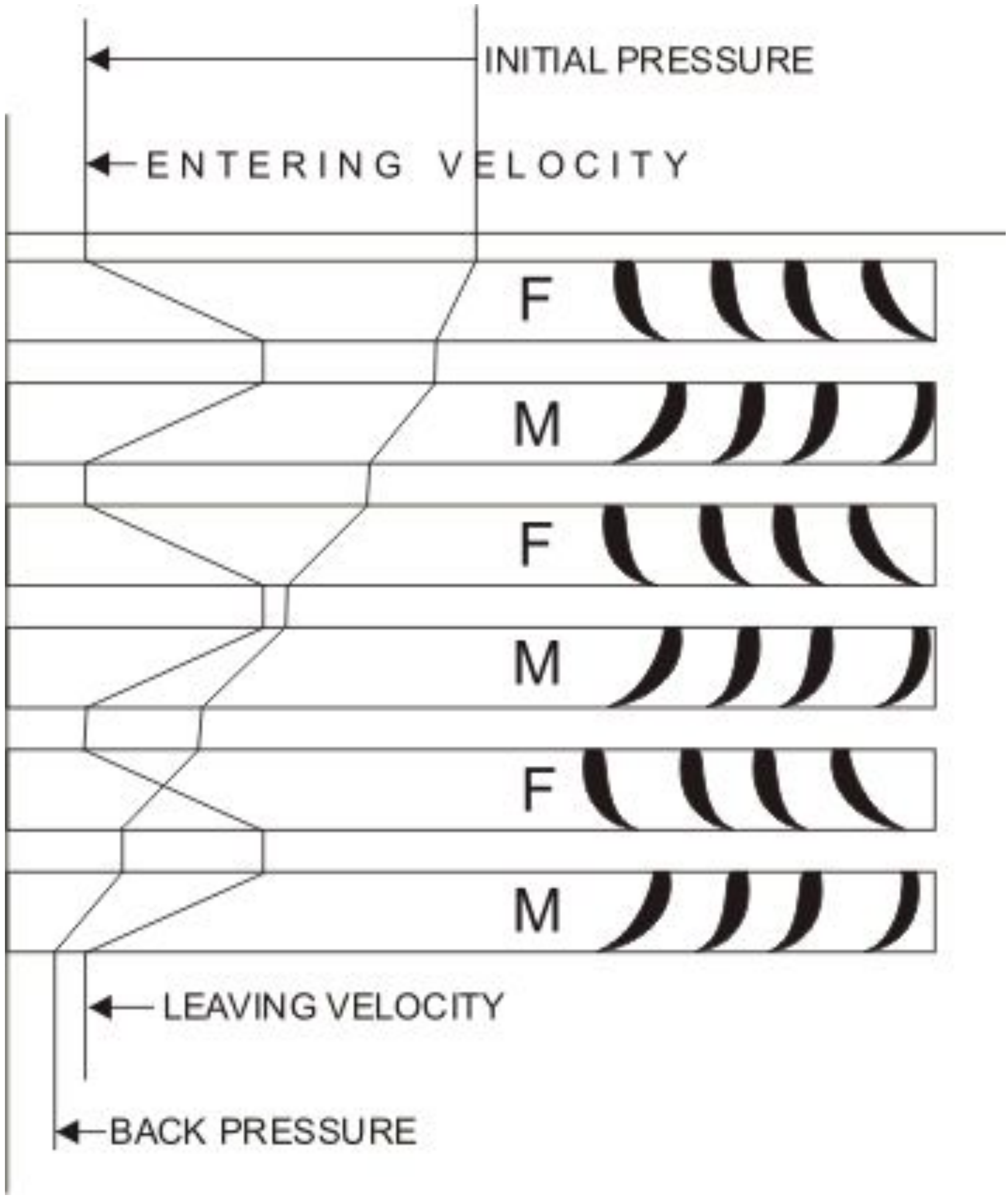
Because of the pressure drops in each stage, the number of stages required in a reaction turbine is much greater than in a impulse turbine of same capacity.

It also concluded that as the volume of steam increases at lower pressures therefore the diameter of the turbine must increase after each group of blade rings.

Reaction turbine.

- In Reaction turbines, addition to the pressure drop occurs in the nozzle there will also be pressure drop occur when the fluid passes over the rotor blades. Fig. shows the Reaction turbine.
- Most of the steam turbine are of axial flow type devices except Ljungstrom turbine which is a radial type.

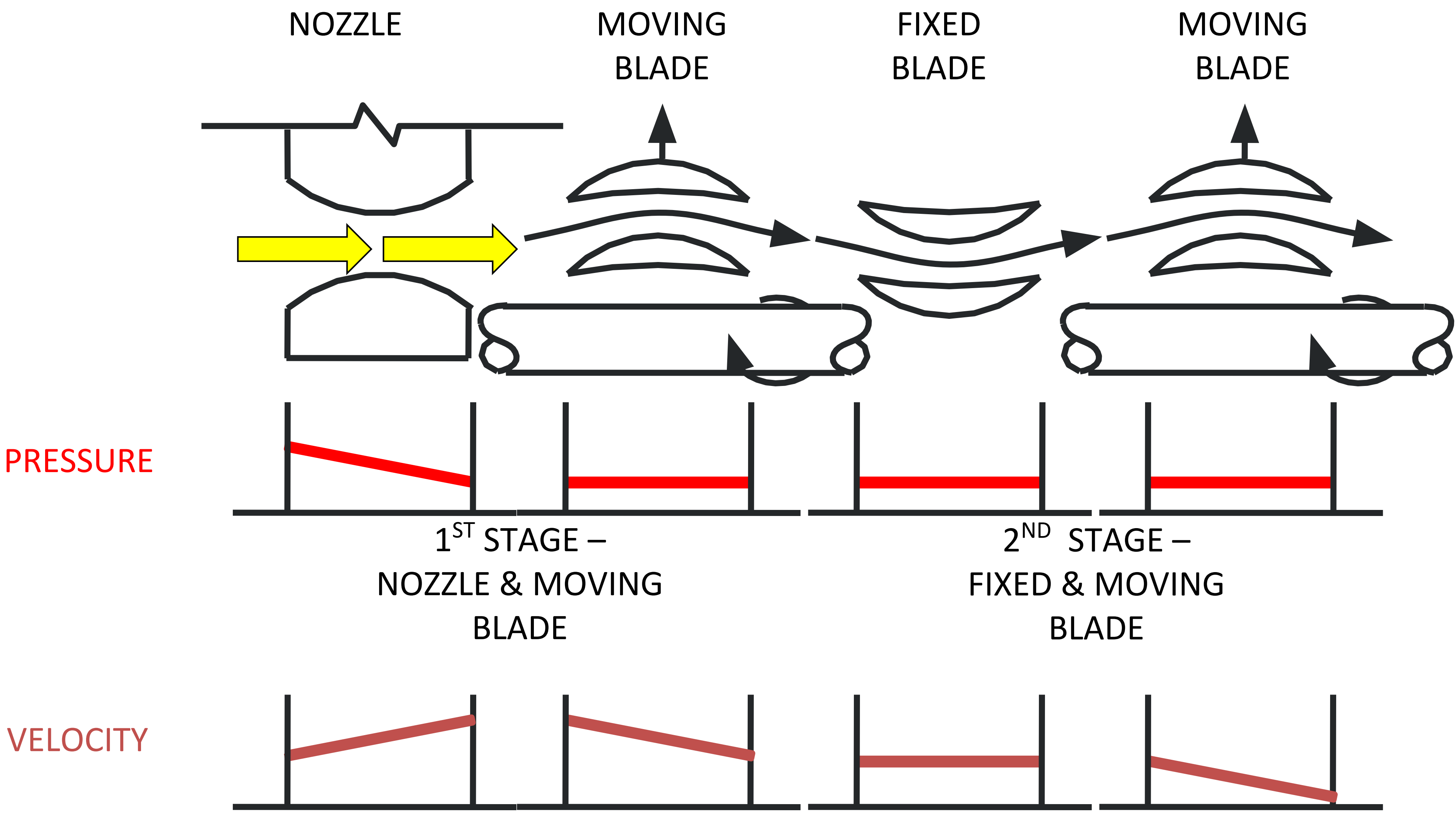



INITIAL PRESSURE
ENTERING VELOCITY
F
M
F
M
F
M
LEAVING VELOCITY
BACK PRESSURE

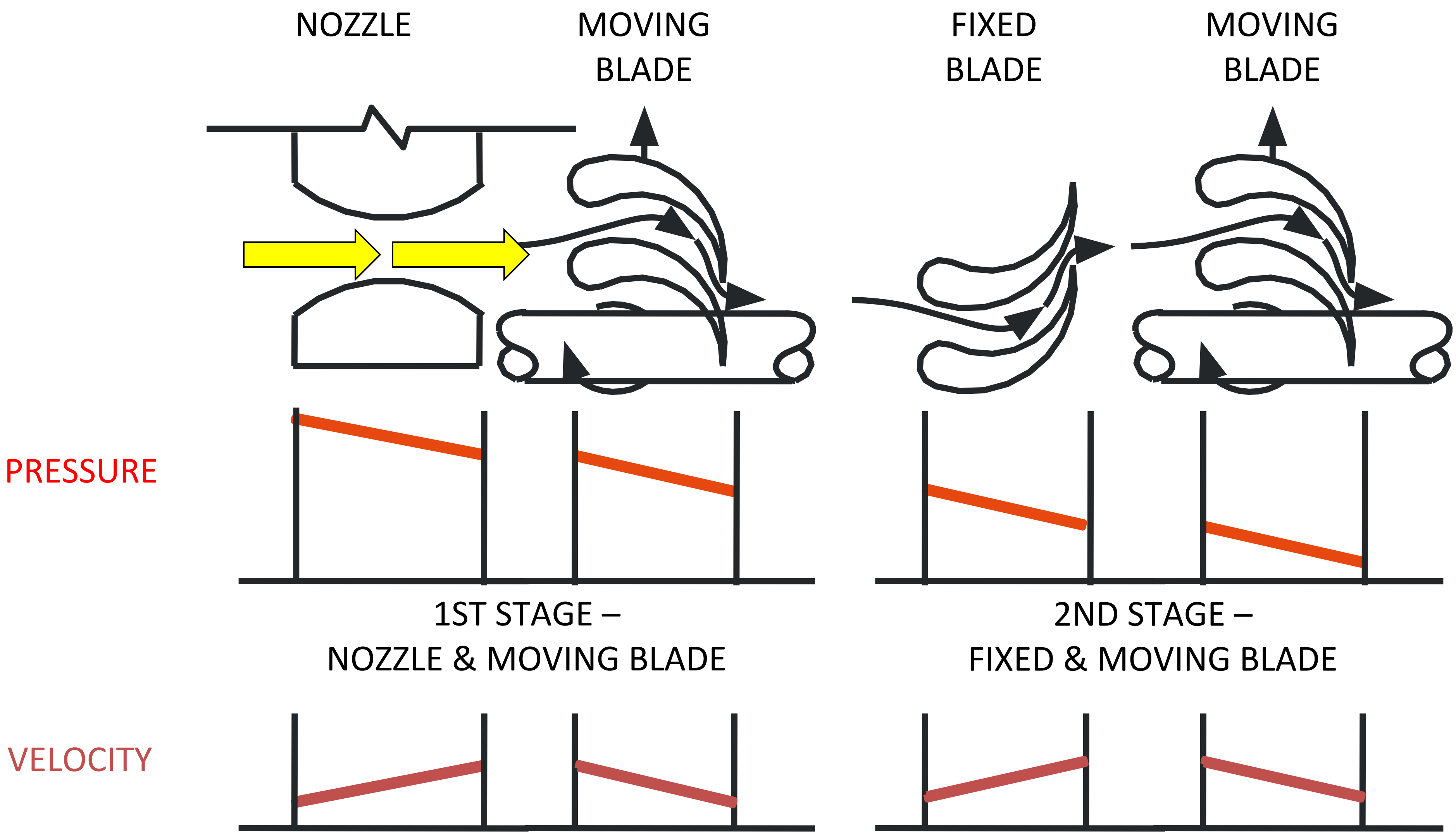
- A ***reaction turbine***, therefore, is one that is constructed of rows of fixed and rows of moving blades.
- The fixed blades act as nozzles. The moving blades move as a result of the impulse of steam received (caused by change in momentum) and also as a result of expansion and acceleration of the steam relative to them. In other words, they also act as nozzles.
- The moving blades of a reaction turbine are easily distinguishable from those of an impulse turbine in that they are not symmetrical and, because they act partly as nozzles, have a shape similar to that of the fixed blades, although curved in the opposite direction.
- The schematic pressure line (Fig.) shows that pressure continuously drops through all rows of blades, fixed and moving. The absolute steam velocity changes within each stage as shown and repeats from stage to stage.

- The enthalpy drop per stage of one row fixed and one row moving blades is divided among them, often equally.
- Thus a blade with a 50 percent degree of reaction, or a 50 percent reaction stage, is one in which half the enthalpy drop of the stage occurs in the fixed blades and half in the moving blades. The pressure drops will not be equal, however. They are greater for the fixed blades and greater for the high-pressure than the low-pressure stages.

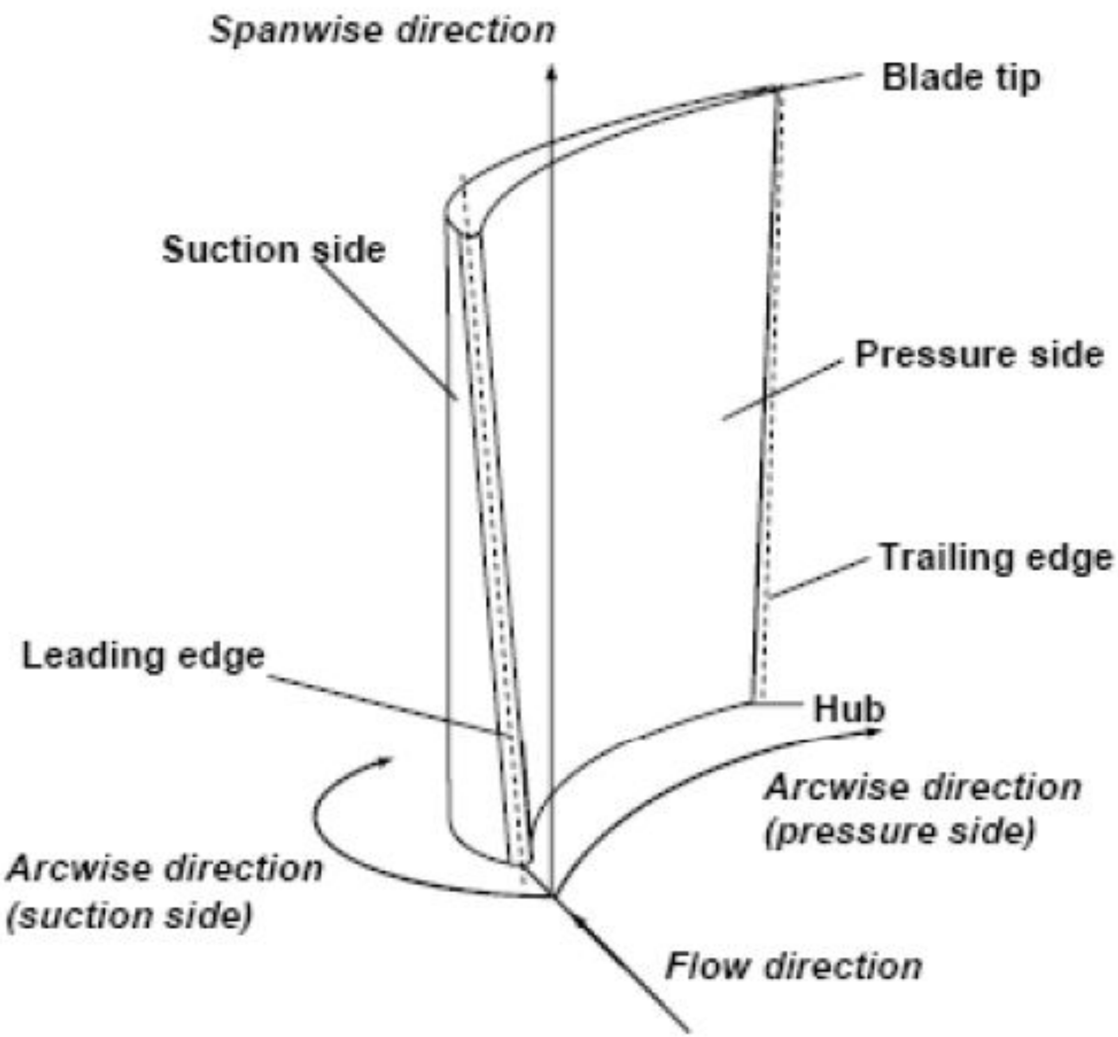
IMPULSE TURBINE STAGING



REACTION TURBINE STAGING



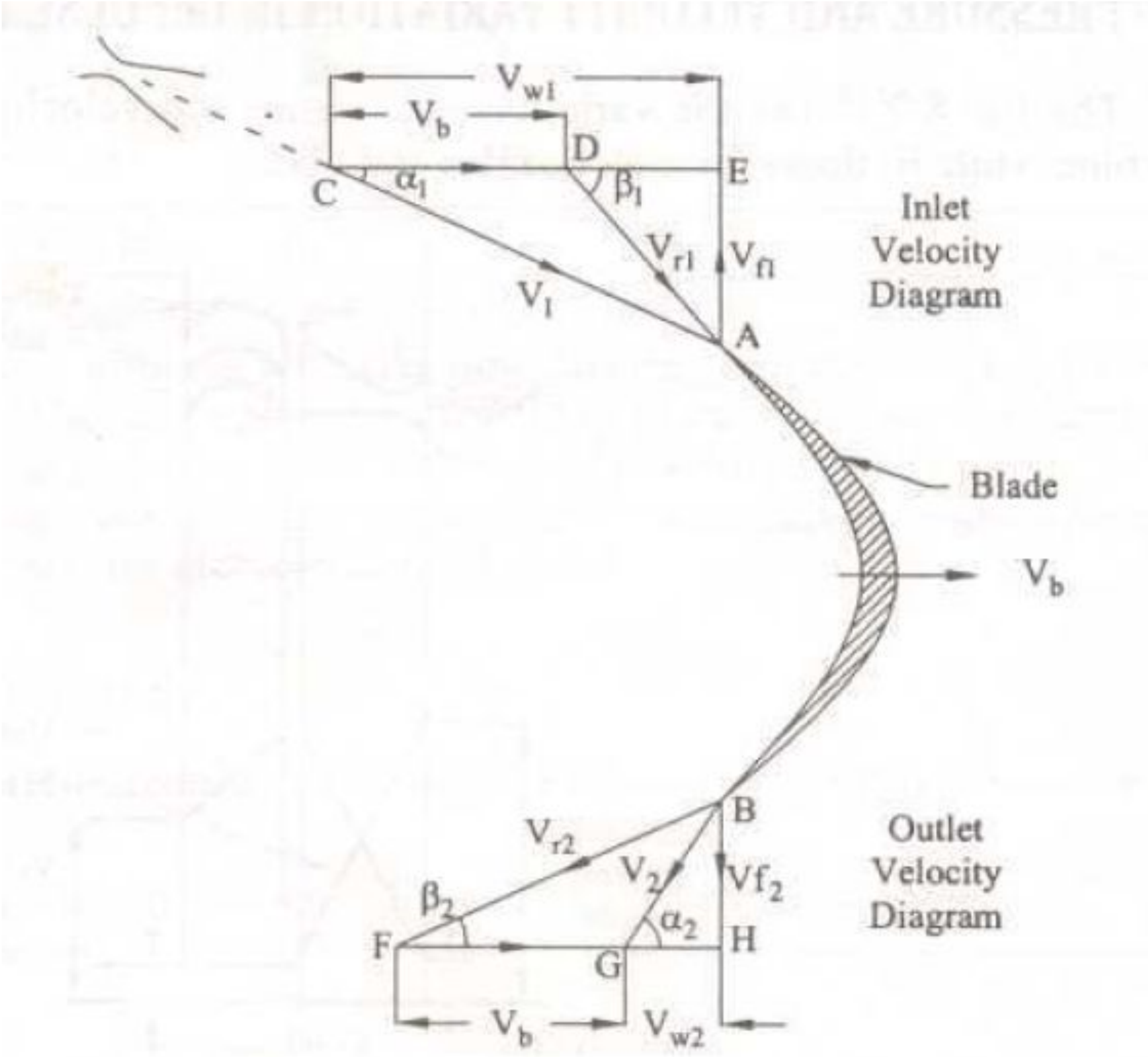
Steam Turbine Blade Terminology



Velocity Triangles

- The three velocity vectors namely blade speed, absolute velocity and relative velocity in relation to the rotor are used to form a triangle called as velocity triangle.
- Velocity triangles are used to illustrate the flow in the blading of turbo machinery.
- Changes in the flow direction and velocity are easy to understand with the help of the velocity triangles.
- Note that the velocity triangles are drawn for the inlet and outlet of the rotor at certain radii.

VELOCITY DIAGRAMS FOR AN IMPULSE TURBINE



Notation

Let

V_b = Linear velocity of moving blade. C_b

V_1 = Absolute velocity of steam at inlet to moving blade i.e., exit velocity of nozzle. C_1

V_{w1} = Tangential component of entering steam.

V_{w1} Also known as velocity of whirl at entrance. C_{w1}

V_{r1} = Relative velocity of steam with respect to tip of blade at inlet.

It is the vectorial difference between V_b and V_1 C_{r1}

V_{f1} = Velocity of flow

= Axial velocity at entrance to moving blades.

It is the vertical component of V_1 C_{f1}

α_1 = Angle of nozzle

= Angle which the entering steam makes with the moving blade at entrance - with the tangent of the wheel at entrance.

β_1 = Angle which the relative velocity makes with the tangent of the wheel direction of motion of blade. It is also known as blade angle at inlet.

The above notations stand for inlet triangle.

V_2, V_{w2}, V_{f2}, V_{r2}, α_2, β_2 are the corresponding values at the exit of the moving blades.

They stand for outlet triangle.

Notation for steam turbine

Velocity notation	Hydraulic turbine	Steam Turbine
u	u_1 ,u_2 u1=u2	Cb
V	V1, V2	C1, C2
Vr	Vr1 ,Vr2	Cr1, cr2
Vw	Vw1, Vw2	Cw1 ,Cw2
α		α1
β		β1
θ		α2
ϕ		β2

Working of impulse steam Turbine

The steam jet with absolute velocity V1 Or C1 impinges on the blade at an angle of $\alpha 1$ to the tangent of the blade.

The absolute velocity V1 Or C1 can be considered as having two components.

The tangential component called whirl component $Vw1 = V1\cos\alpha 1$ parallel to direction of rotation of blades

and

The axial or flow component $Vf1 = V\sin \alpha 1$ perpendicular to the direction of rotation of blades.

The tangential component of the steam jet does work on the blade because it is in the same direction as the motion of the blade.

The axial component doesn't work on the blades because it is perpendicular to the direction of motion of blade.

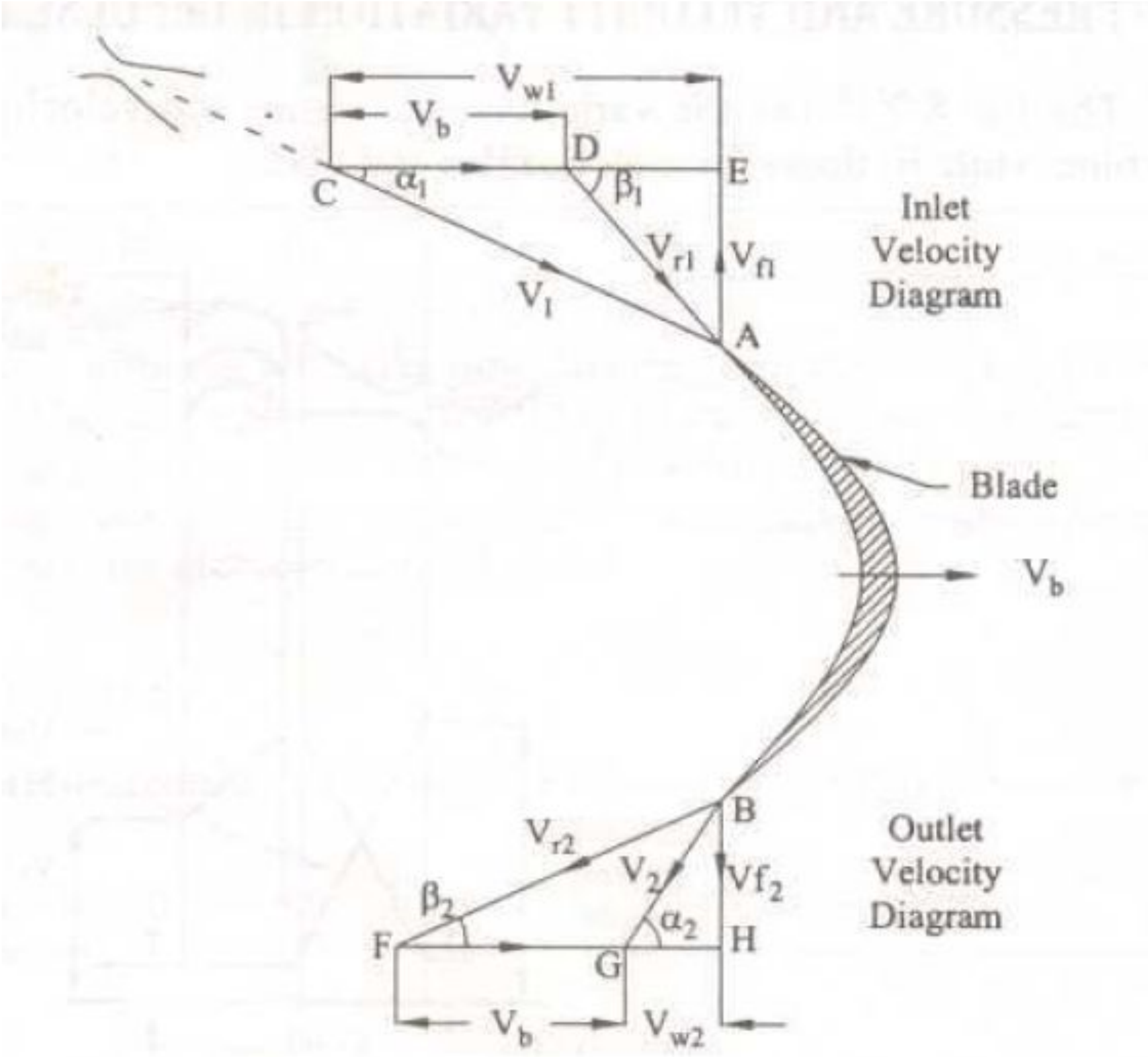
It is responsible for the flow of steam through the turbine. Change of velocity in this component causes an axial thrust on the rotor.

As the blade moves with a tangential velocity in peripheral direction, the entering steam jet will have relative velocity to the blades.

If there is no friction loss at the blade, relative velocity at inlet is equal to relative velocity at outlet i.e., $V_{r1} = V_{r2}$.

As the steam glides over the blades without shock, the surface of the blade at inlet must be parallel to relative velocity V_{r1}. So, the moving blade at inlet must be inclined to the tangent of the blade at an angle β_1. In other words, to avoid shock at entrance, vector V_{r1} must be tangential to the blade tip at entry i.e, β_1 must be equal to angle of blade at entrance. The blade is designed on this principle.

VELOCITY DIAGRAMS FOR AN IMPULSE TURBINE



From the above analysis, following points are to be noted.

1. No expansion of steam takes place in the moving blades. The blades only deflect steam. This causes change in momentum and consequently force.
2. If the steam has to enter and leave the blades without shock, angle $\beta 1$, should be angle of blade at inlet and angle $\beta 2$ should be angle of blade at outlet. This is an essential condition.
3. Since there is no pressure drop in the moving blades, the pressure on the two sides of the blades is equal.
4. $\alpha 1$ is the outlet angle of nozzle. If steam has to enter the next nozzle ring without shock, its inlet angle must be equal to $\alpha 2$.
5. In a simple impulse turbine, the loss at exit is the whirl component at outlet $- V2 \cos \alpha 2$.

   For minimum loss, this quantity should be minimum, i.e., $\alpha 2$ should be equal to 90°.

In that case the turbine discharges axially and it is called axial turbine.

Construction of combined velocity diagram :

1. First, draw a horizontal line and cut off AB equal to velocity of blade to some suitable scale.
2. From B, draw a line BC at an angle $\alpha 1$, with AB as base. Cut off BC equal to V1 to scale.
3. Join AC. It represents Vr1.
4. From A; draw a line AD at an angle $\beta 2$ with AB as base . With A as centre and radius equal to AC, draw an arc that meets the line through A at D such that AC = AD. Or Vr1 = Vr2.

5. Join BD. It represents absolute velocity at exit to scale.
6. From C and D draw perpendiculars to meet the line AB produced at E and F.
7. Now; to scale,

EB = velocity of whirl at entrance.

BF = velocity of whirl at exit.

CE = velocity of flow at inlet.

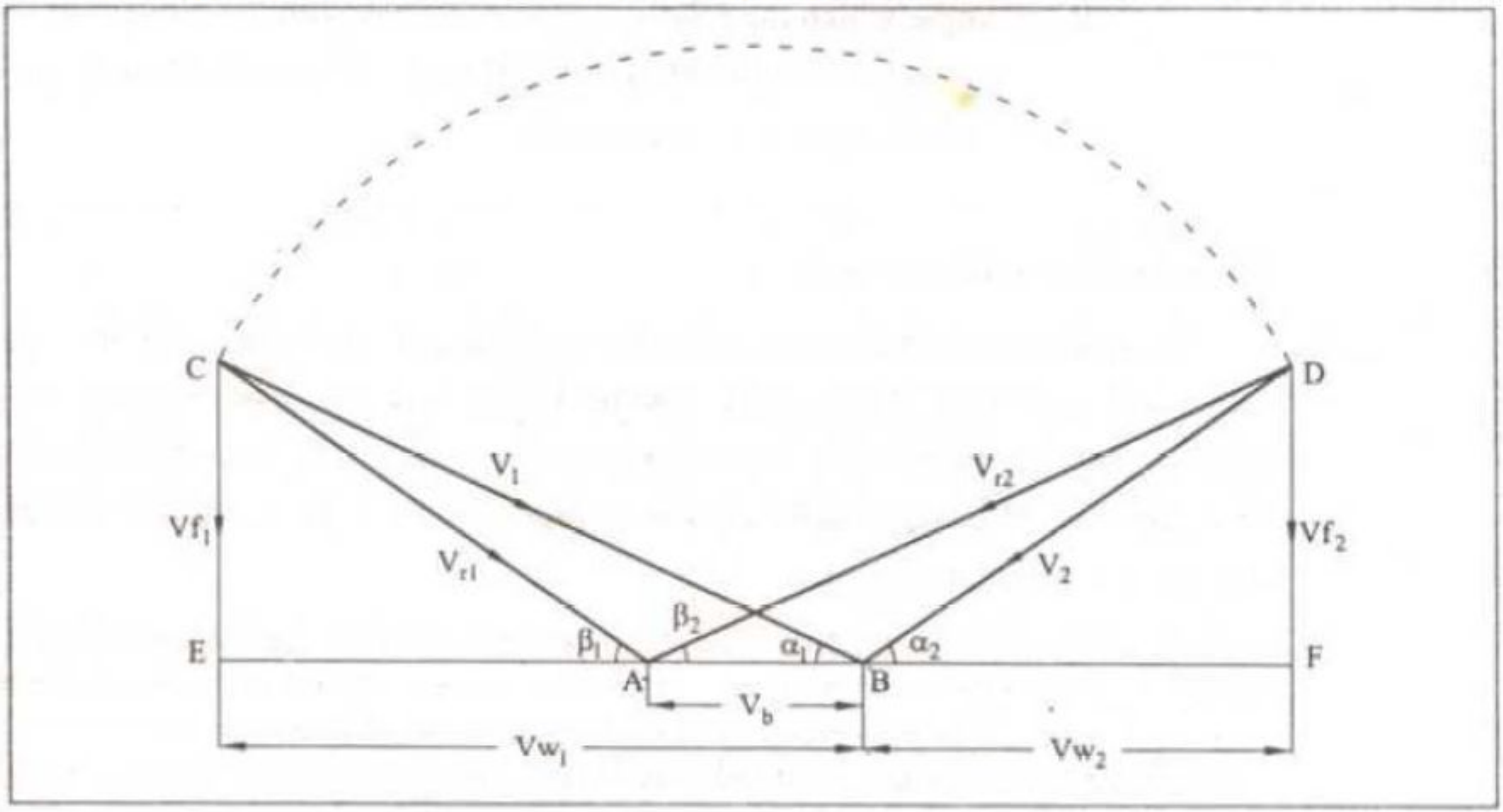
DF = velocity of flow at outlet.

When friction is neglected, there will be no fall in steam pressure as it flows over the blades and $V_{r1} = V_{r2}$.

Also, when friction is absent, $\beta_1 = \beta_2$ and $V_{f1} = V_{f2}$

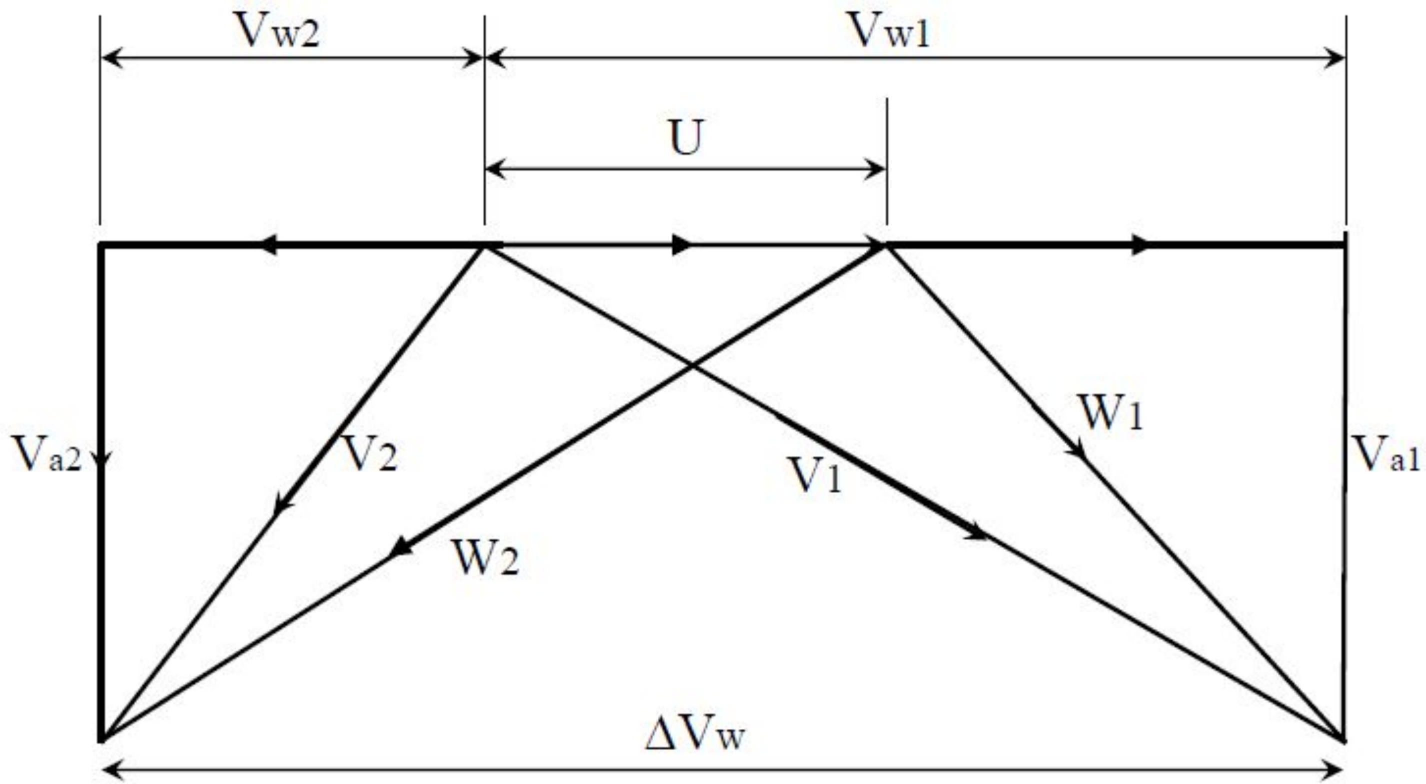
COMBINED VELOCITY DIAGRAM

To solve problems on turbines conveniently, it is common practice to combine both the inlet and outlet velocity diagrams on a common base which represents the blade velocity.



COMBINED VELOCITY DIAGRAM FOR AN IMPULSE TURBINE

Combined Velocity Triangles



Effect of Blade Friction

In an impulse turbine, the relative velocity remains same as steam passes over the blades if friction is neglected.

In actual practice, the flow of steam the blades is resisted by friction. The effect of this friction is to reduce the relative velocity of steam while passing over the blades-

Generally, there is a loss of 10-15% in relative velocity. Owing to friction in blades. Vr2 is less than Vr1 and we may write

$$V_{r2} = K \cdot V_{r1} \qquad K = \frac{V_{r2}}{V_{r1}}$$

The ratio of Vr2 to Vr1 is called blade velocity coefficient or coefficient of velocity friction factor K.

Depending upon the shape of the blades, value of K varies from 0.75 to 0.85.

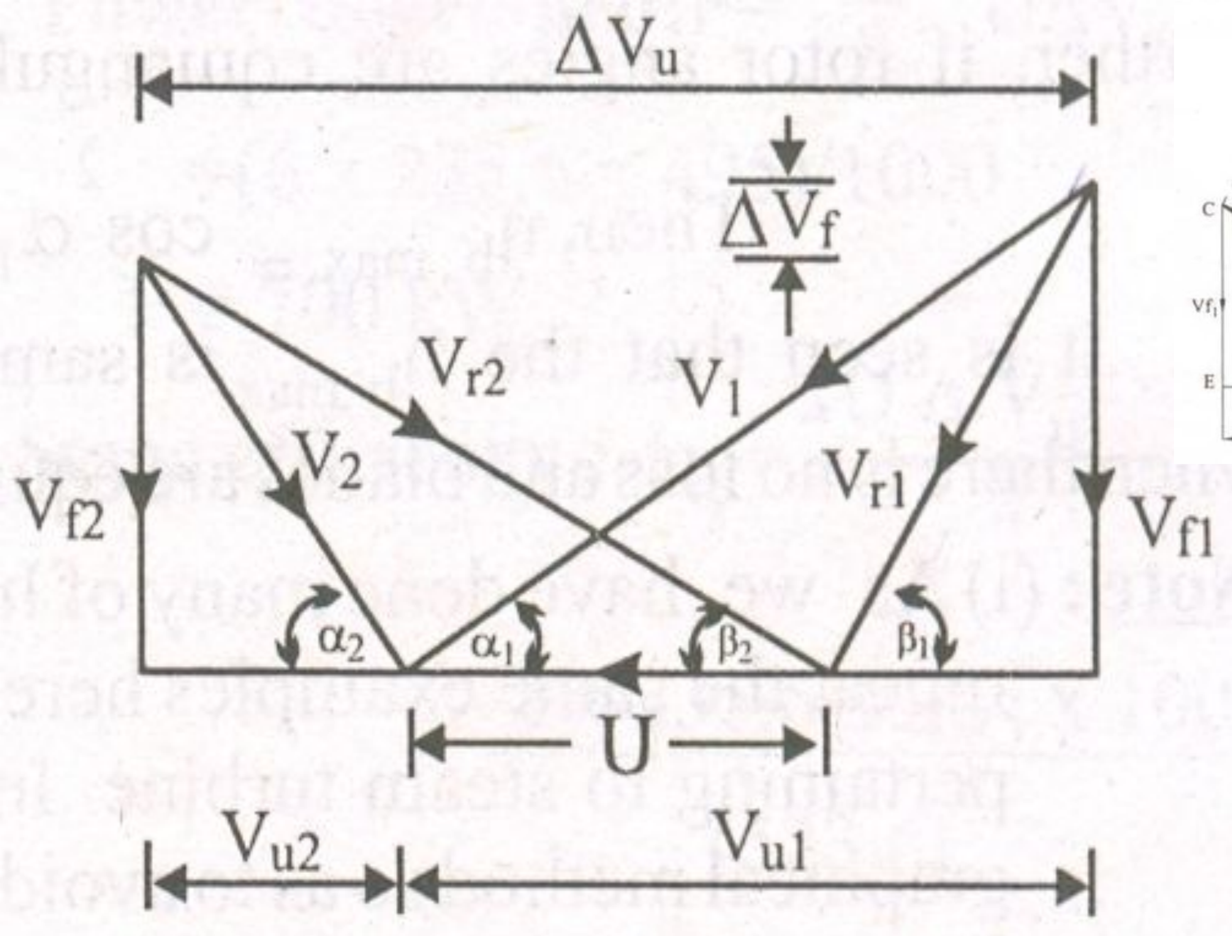
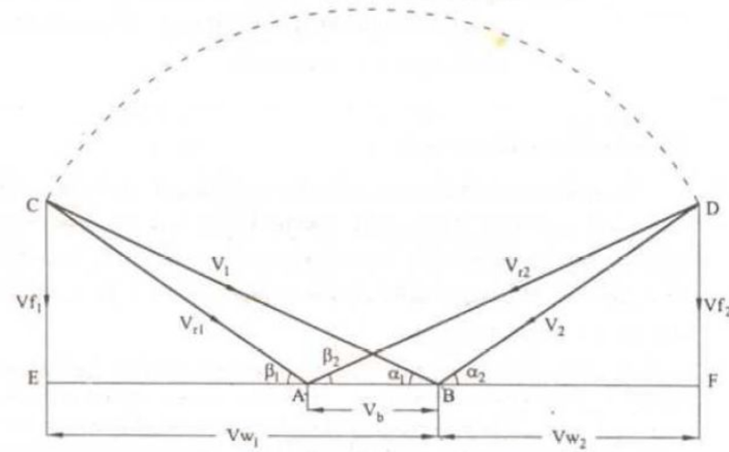


Fig. b Effect of blade friction



Due to the effect of blade friction loss, the relative velocity at outlet is reduced than the relative velocity at inlet.

Therefore $V_{r2} = K\, V_{r1}$. Corresponding velocity triangle shown if fig (b)

FORCES ON BLADE

Force on Rotor: According to Newton's second law of motion,

Tangential force on rotor = mass x tangential acceleration.

= m(Vw1 –(- Vw2)) = m(Vw1 + Vw2)

where m = Mass flow rate of steam - kg/sec.

Vw2 is negative as the steam is discharged in opposite direction to blade motion.

So, Vw1 and Vw2 are added together.

Generally, Positive sign is to be used when Vw2 and , Vw1 are in opposite direction as shown above and negative sign is to be used when Vw2 and , Vw1 are in same direction.

$$F_t = \dot{m}\,(V_{\omega 1} \pm V_{\omega 2})\ \text{Newton.}$$

Power Developed by the Turbine:

Power = Rate of doing work

$$= m \cdot (V_{\omega 1} \pm V_{\omega 2}) \cdot V_b \text{ watts.}$$

(1 watt= 1 N-m/sec)

This power is known as Rim power or diagram power to distinguish it from shaft power.

Axial Thrust on Rotor:

Axial force Fa= Mass x Axial acceleration

= Mass x change in velocity of flow.

$$= m \cdot (V_{f1} - V_{f2}) \text{ Newtons.}$$

This axial force must be balanced or must be taken by a thrust bearing.

Efficiencies of Turbine

Blade or Diagram Efficiency

It is defined as the ratio of work done on blades to energy supplied to blades. This is also called diagram efficiency

Let V1or C1 = Absolute velocity of steam at inlet —m/sec

m = Mass of steam supplied — kgs/sec.

Energy of steam supplied to blade = $1/2 * m * V_1^2$

Work done on blade = m . (Vw1 ± Vw2) . V_b J/sec

Diagram or blade efficiency

$$\eta_{bl} = \frac{\text{Work done on blade}}{\text{Energy supplied blade}}$$

$$= \frac{m \,.\, (V_{\omega 1} \pm V_{\omega 2} \cdot V_b}{\frac{1}{2} m V_1^{\,2}}$$

$$= \frac{2 V_b \cdot (V_{\omega 1} \pm V_{\omega 2})}{V_1^{\,2}}$$

Gross or Stage Efficiency

stage consists of a set of nozzles and a row of moving blades and so, stage efficiency includes the performance of nozzles also.

Stage efficiency is defined as the ratio of work done on blades per kg of steam to total energy supplied per stage per kg of steam.

If h1 and h2 represent energy before and after expansion of steam through the nozzles, then the enthalpy drop

(h1- h2) is the enthalpy drop through a stage, i.e., the heat energy (h1 - h2) is the energy supplied per stage per kg of steam.

$$\text{Stage efficiency} = \frac{\text{Work done on blade}/kg\text{ of steam}}{\text{Total energy supplied}/\text{stage}/kg\text{ of steam}}$$

$$\eta_{\text{stage}} = \frac{(V_{\omega 1} \pm V_{\omega 2} \cdot V_b}{h_1 - h_2}$$

Nozzle Efficiency

- It is defined as the ratio of energy supplied to blades per kg of steam to total energy supplied per stage per kg of steam.
- Energy supplied to blades per kg of steam = $1/2*m*V_1^2$
- Total energy supplied per stage per kg of steam = (h1-h2)

$$\text{Nozzle efficiency} = \frac{\text{Energy availbale at entrance}/kg}{\text{Enthalpy drop through a stage}/kg\text{ of steam}}$$

$$\eta_{nozzle} = \frac{\frac{1}{2} V_1^{\,2}}{(h_1 - h_2)}$$

$$= \frac{V_1^{\,2}}{2\,(h_1 - h_2)}$$

- Stage efficiency = blade efficiency x nozzle efficiency.
- Energy converted to heat by blade friction= Loss of kinetic energy during flow over the blades

$$= \frac{1}{2} m \cdot \left(V_{r1}^{\,2} - V_{r2}^{\,2}\right) J$$

Forces acting on a reaction blade

Reaction force: It is due to the change in momentum of relative velocity of the steam while passing over the blade passage.

Centrifugal force: It is the force acting on the blade due to change in radius of steam entering and leaving the turbine.

Resultant force:

It is the resultant of

Reaction force

and Centrifugal force.

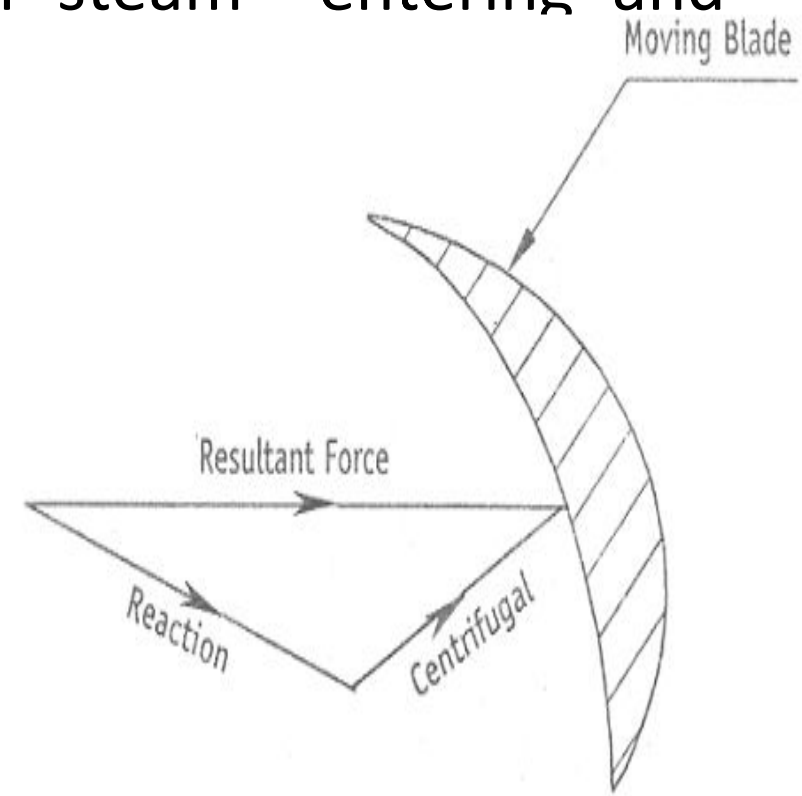


Fig. Force Diagram

Reaction turbines:

These are of axial type. But pure reaction turbine are not in general use, only impulse-reaction turbines are used.

1. The velocity triangle for general case:

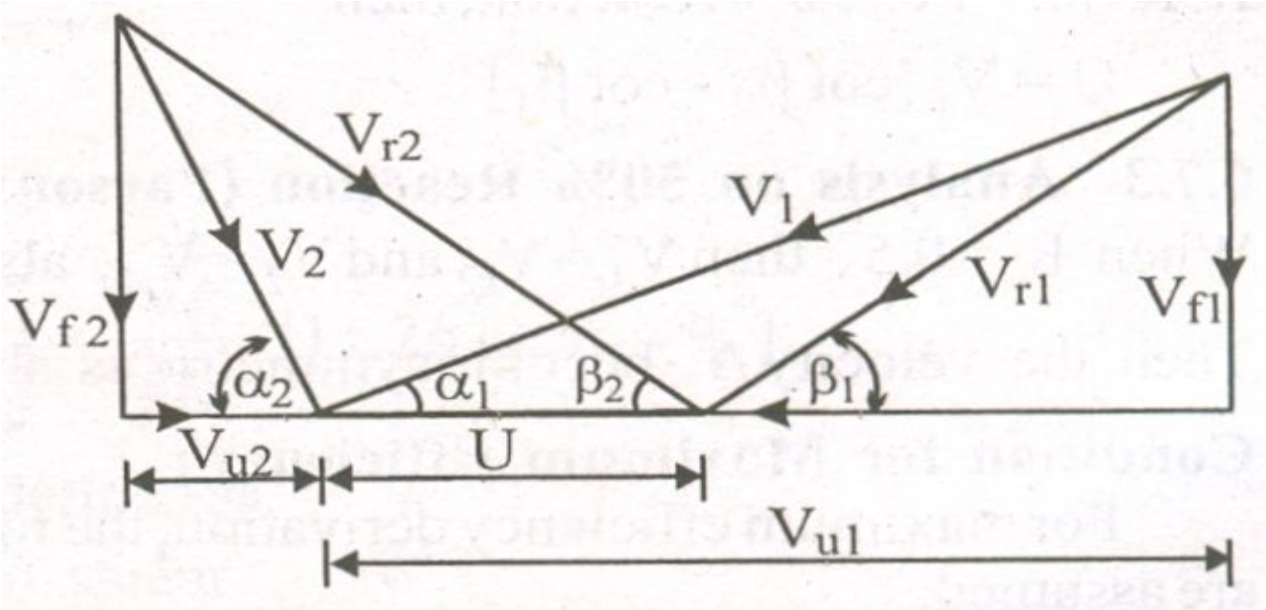


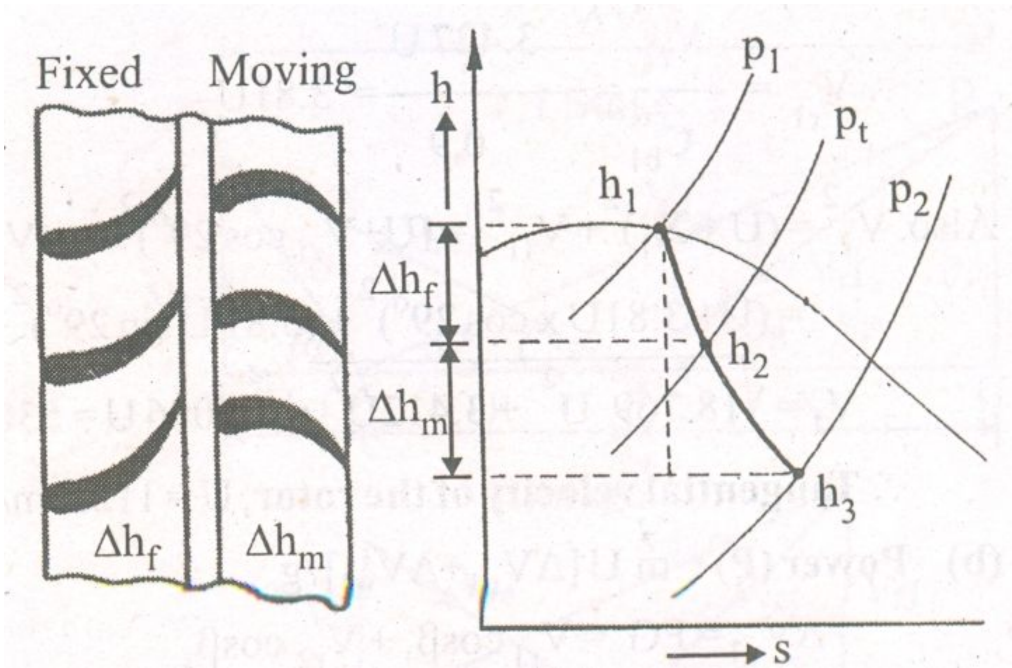
Fig. .Impulse Reaction turbine stages.

In reaction turbine steam continuously expands as it flows over the blades thereby increases the relative velocity of steam, i.e., **Vr2 > Vr1**

Degree of reaction

- **Degree of reaction is a parameter that describes the relation between the** energy transfer due to the static pressure change and the energy transfer due to dynamic pressure change.
- The degree of reaction for reaction turbine stage is defined as the ratio of enthalpy drop in the moving blades to the total enthalpy drop in fixed and moving blades (i.e., static enthalpy drop to total enthalpy drop), as shown in fig.

$$\Lambda = \frac{\text{The enthalpy drop in the moving blades}}{\text{The enthalpy drop in the stage}}$$



Degree of Reaction (Rd)

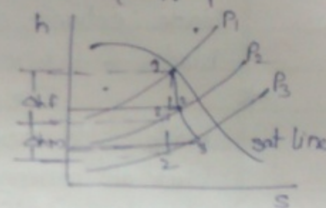
The degree of Reaction of Reaction turbine is defined as the ratio of heat drop over moving blades to the total heat drop in the stage

$$R_d = \frac{\text{Heat drop in moving blade}}{\text{Heat drop in the stage}}$$

Δh_m = heat drop in moving blade

Δh_f = heat drop in fixed blade

$$R_d = \frac{\Delta h_m}{\Delta h_f + \Delta h_m}$$



(Enthalpy) heat drop in moving blade is equal to increase in relative velocity of steam passing through blade

$$\Delta h_m = \frac{V_{r_2}^2 - V_{r_1}^2}{2} \quad \text{where } h_m \text{ in J/kg}$$

Total heat drop in a stage is equal to work done by a steam in the stage

$$\Delta h_f + \Delta h_m = u[V_{w_1} + V_{w_2}]$$

$$\boxed{R_d = \frac{V_{r_2}^2 - V_{r_1}^2}{2u(C_{w_1} + C_{w_2})}}$$

From fig, R_d : $V_{r_2} = V_{f_2} \operatorname{cosec} \beta_2$, $V_{r_1} = V_{f_1} \operatorname{cosec} \beta_1$

$V_{w_1} + V_{w_2} = V_{f_1} \operatorname{cosec} \beta_1 + V_{f_2} \operatorname{cosec} \beta_2$

$V_{f_1} = V_{f_2} = V_f$

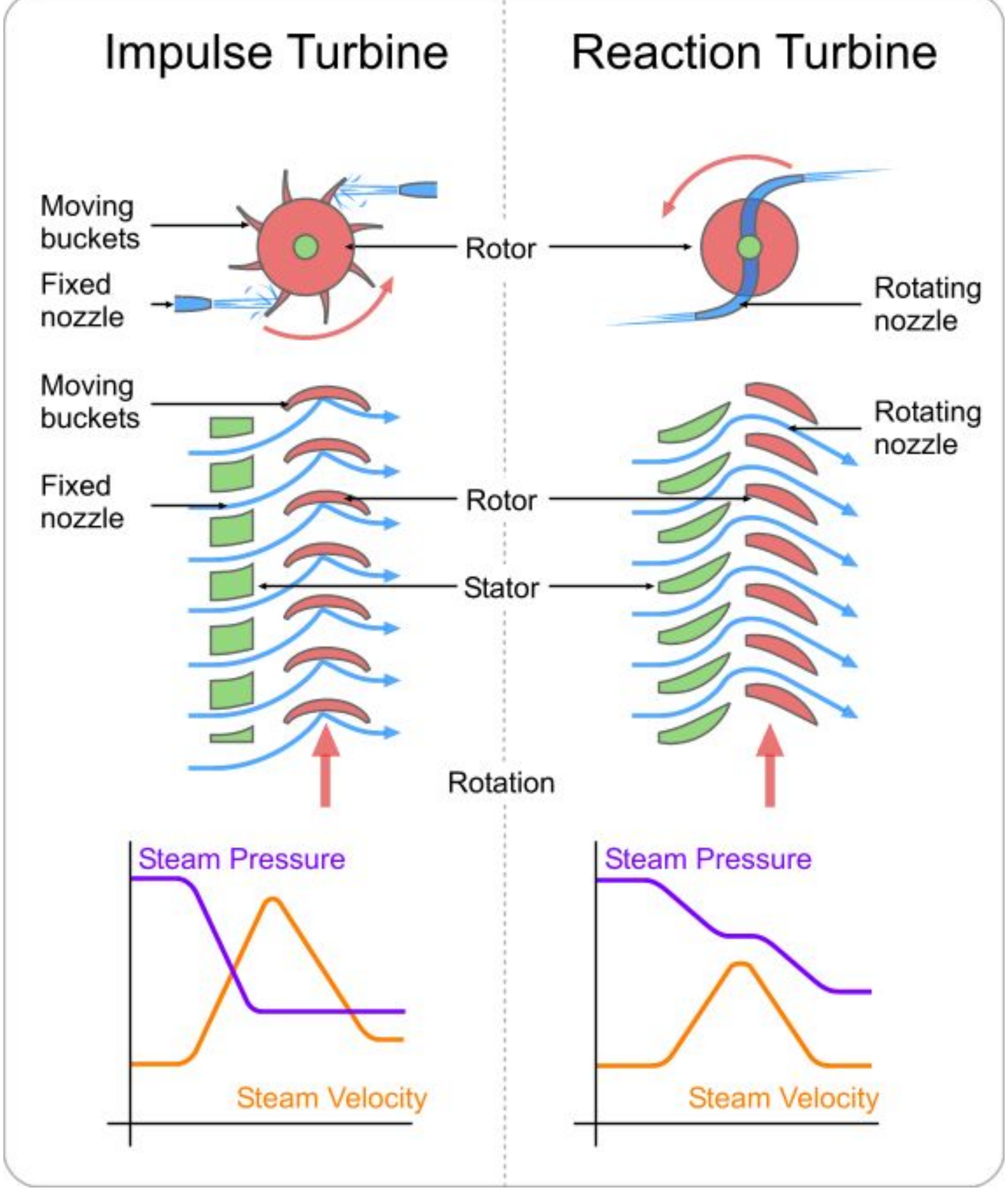
$$\boxed{R_d = \frac{V_f^2 [\operatorname{cosec}^2 \beta_2 - \operatorname{cosec}^2 \beta_1]}{2u V_f [\cot \beta_2 + \cot \beta_1]}}$$

Comparison between impulse and reaction Turbine

	Impulse Turbine		Reaction Turbine
1.	During flow of steam, pressure drops only in the nozzle.	1.	During steam flow, pressure drops over fixed nozzle and moving blades.
2.	The blades of impulse turbine are of profile types.	2.	The blades of reaction turbine are of aerofoil type.
3.	The power developed is comparatively less.	3.	The power developed is more.
4.	The blades are symmetrical.	4.	The blades are not symmetrical.
5.	The number of stages required for a given power is less.	5.	The number of stages required for a given power is more.
6.	Efficiency of impulse turbine is less.	6.	Efficiency of reaction turbine is more.
7.	Steam velocity is more.	7.	Steam velocity is less.
8.	It is suitable for small power requirement.	8.	It is suitable for medium and high power requiremen
9.	The pressure of steam while flowing over moving blades remains constant.	9.	The pressure of steam while flowing over the moving blades reduces.
10.	Manufacturing of blades is easy and hence inexpensive.	10.	Manufacturing of blades is difficult and hence costlier.

Impulse Turbine	Reaction Turbine
1) In impulse Turbine, only impulsive force strikes to the blades fixed to the rotor	1) In reaction turbine, vector sum of impulsive and reactive force strikes the blades fixed to the rotor.
2) Steam expands completely when it passes through the nozzles and its pressure remains constant.	2) pressure can't expand fully. It partially expands when it pass through the nozzles and rest on the rotor blades.
3) Blades are symmetrical shape.	3) Blades are asymmetrical shape.
4) Since the velocity of steam is high, speed is high in impulse turbine.	4) But reaction turbine speed is much lower than impulse turbine because steam velocity is lower in reaction turbine as compared to impulse turbine.
5) For producing same power, the number of stages required are much less.	5) It require more stages to develop same power.
6) The blade efficiency curve is high.	6) The blade efficiency curve is lower than impulse turbine.

Sr no	Particulars	Impulse Turbine	Reaction Turbine
1.	Pressure drop	Only in nozzles and not in moving blades.	In fixed blades(nozzles) as well as in moving blades.
2.	Area of blade channels	Constant.	Varying.
3.	Blades	Profile type.	Aerofoil type.
4.	Admission of steam	Not all round.	All round.
5.	Nozzles	Diaphragm contains the nozzles.	Fixed blades attached to the casing.
6.	Power	Small power capacities.	Much power can be developed.
7.	Space	Less space for same power.	More space required.
8.	Efficiency	Low	High
9.	Suitability	Small power requirements.	For medium and high power requirements.
10.	Blade manufacture	Easy	Difficult

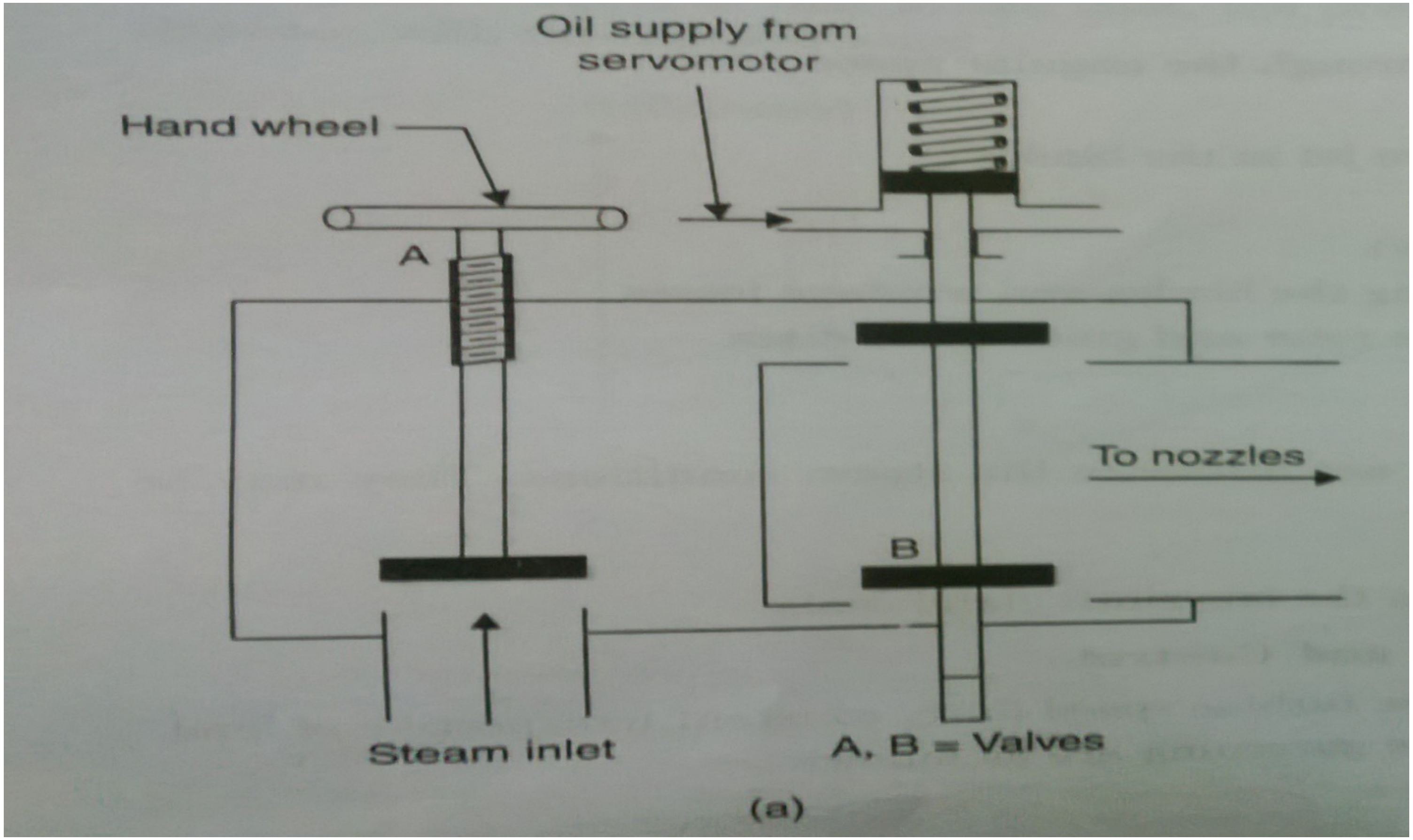

Impulse Turbine
Reaction Turbine
Moving buckets
Fixed nozzle
Rotor
Rotating nozzle
Moving buckets
Fixed nozzle
Rotor
Rotating nozzle
Stator
Rotation
Steam Pressure
Steam Velocity
Steam Pressure
Steam Velocity

*

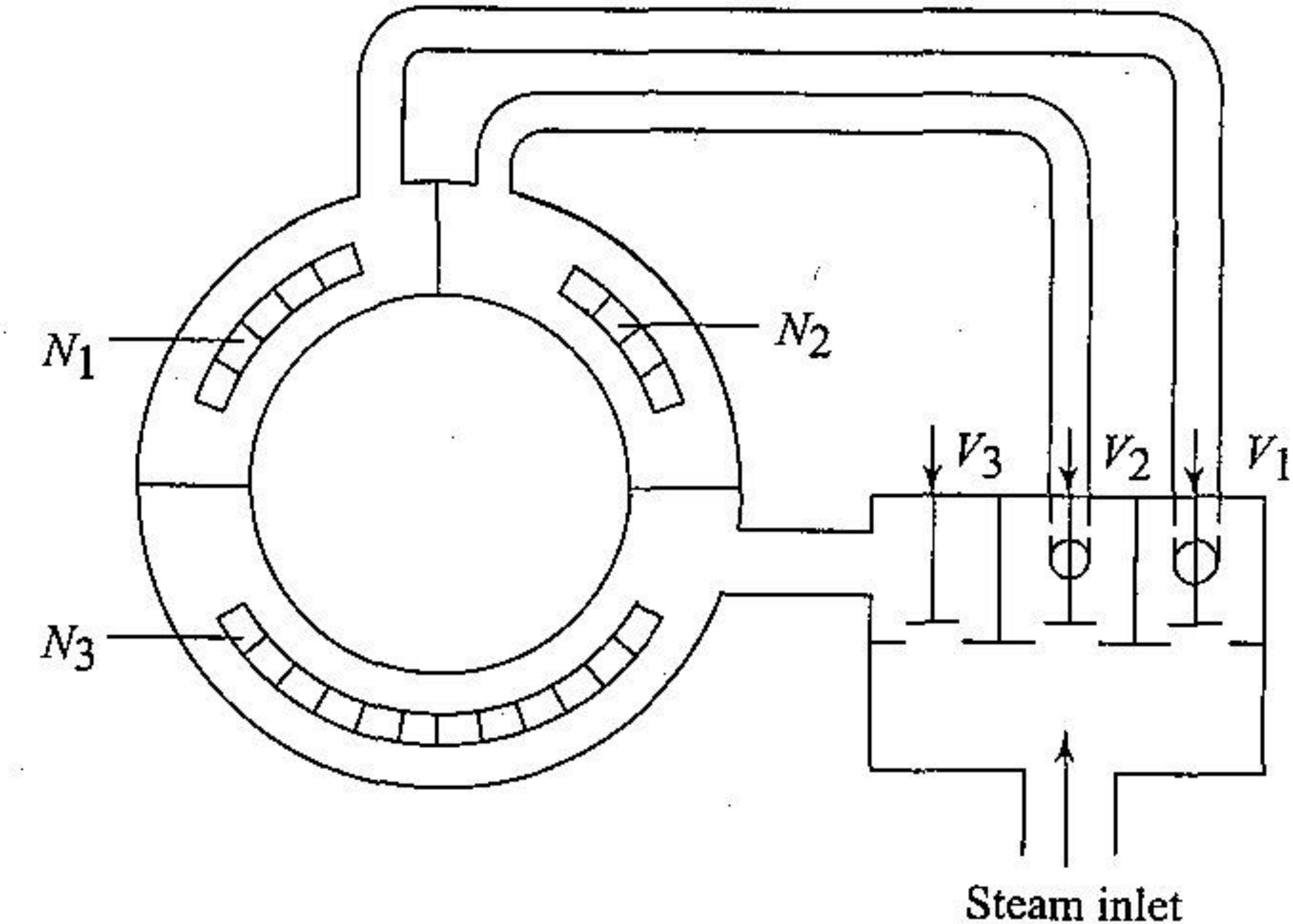
Steam turbine governing and control

Objective is to keep turbine speed fairly constant irrespective of load . Different methods are

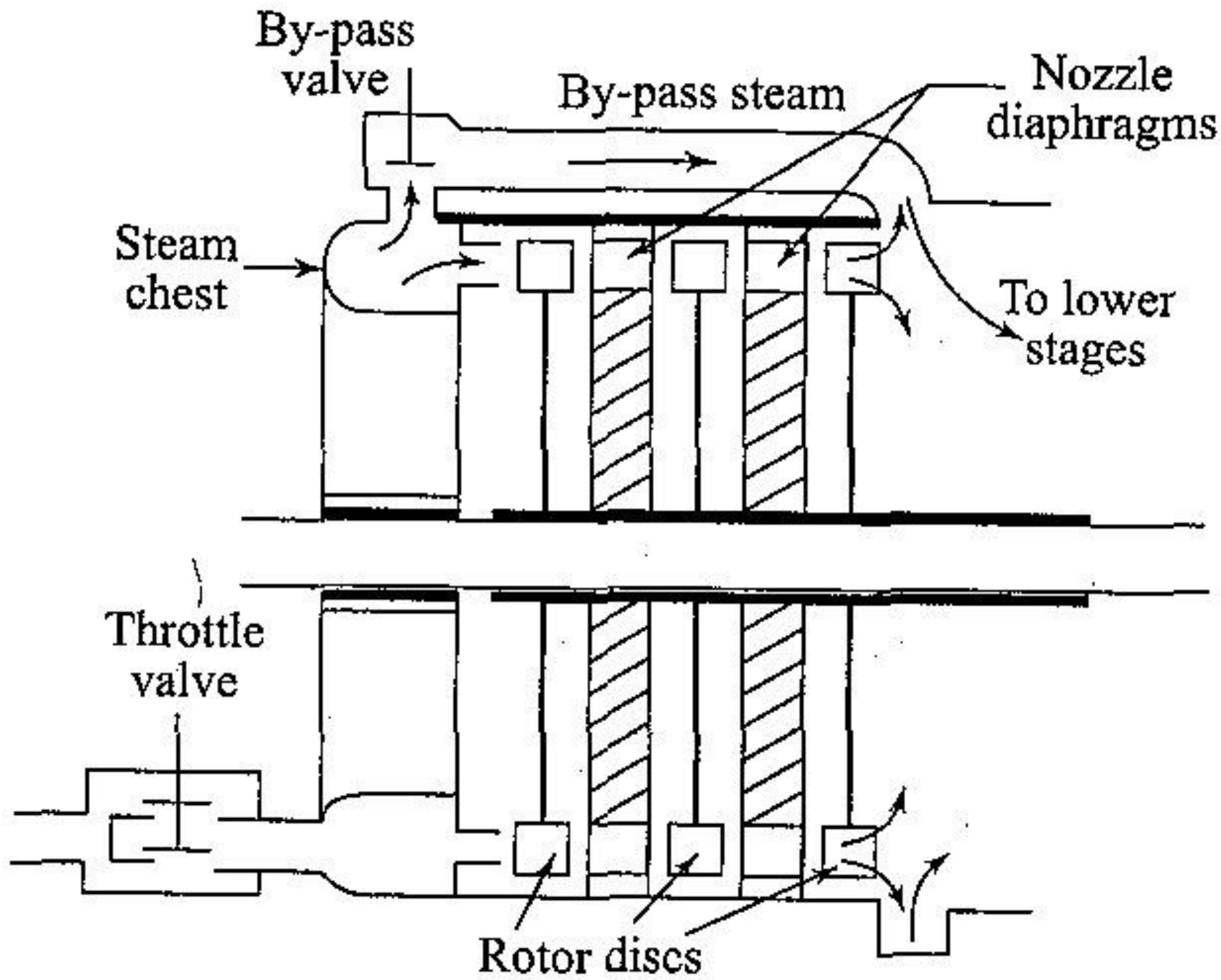
- **Throttle governing**



- Nozzle Governing



Bypass governing



Loss steam turbine

1. Nozzle Friction Loss :-

It is a very important loss for Impulse Turbine. When steam passes through the nozzles, friction loss occurs and the formation of eddies. Friction occurs in the nozzle due to the factor of nozzle efficiency and it is the ratio of actual enthalpy drop to isentropic enthalpy drop.

2. Blade Friction Loss :-

This loss is important for both Impulse and Reaction turbine . Blade friction loss is due to the steam's gliding over the blades and friction of the surface of the blades. The effect of turbine blades is considered as a blade velocity coefficient. The relative velocity of steam is reduced for this loss.

3. Wheel Friction Loss :-

When steam passes through the rotating turbine wheel, it produces some resistance on the turbine wheel. As a result, it rotates in lower speed from its original speed. It is the loss in both Impulse and Reaction turbine. The total frictional loss is about 10% of total turbine loss.

4. Losses due to mechanical friction :-

This loss is for turbine's bearing. Mechanical friction loss is due to the friction between the shaft and wheel bearing and also the regulating valve of the turbine. This loss may be reduced by proper lubrication of the moving parts of the turbine. This loss occurs both Impulse and Reaction turbine.

5. Losses due to leakage :-

Leakage loss is different in both Impulse and reaction turbine. In Impulse turbine, leakage loss occurs between the shaft, bearings, nozzles and stationary diaphragms. For Reaction turbine, it may occur at the blade tips. This loss is due to the leakage of steam on each stage of the turbine. Total leakage loss is about 1 to 2% of total turbine loss.

6. Residual Velocity loss :-

When kinetic energy of steam leaves from the turbine wheel, it happens. Actually, steam leaves from the turbine with some certain absolute velocity. That's why steam looses some kinetic energy. Residual velocity loss can be reduced by multistage turbines. This loss is about 10 to 12% in a single stage turbine.

7. Loss in regulating valves :-

Before entering the steam to the turbine, it passes through the boilers stop and regulating valve. Steam gets throttled in these regulating valves and as a result steam pressure will be less than the boiler pressure at the entry of turbine.

8. Loss due to wetness of steam :-

This loss is due to the moisture present in the turbine. When steam passes through the lower stage of the turbine, it becomes wet. At the lower stage, the velocity of water and steam are different and will not form a homogeneous mixture. That's why the velocity of water particle is less than that of steam and water particle has to be dragged with the steam and some part of kinetic energy of steam is lost. This loss occurs both impulse and reaction turbine.

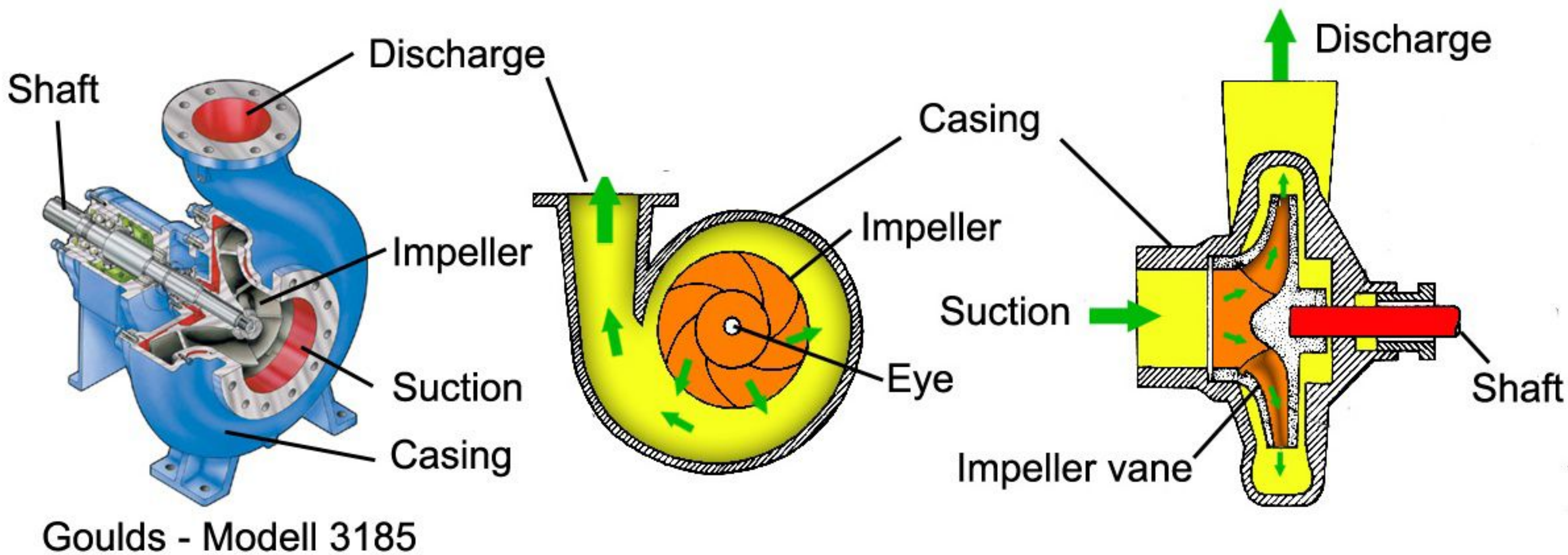
9. Radiation Loss :-

This loss mainly takes place due to the temperature difference between turbine casing and its surrounding atmosphere. This loss can be minimized by the heavily insulated turbine. This loss is in also both impulse and reaction turbine.

10. Governing loss :-

It is the loss in both impulse and reaction turbine and this loss is due to the throttling of the steam at the main stop valve of the governor.

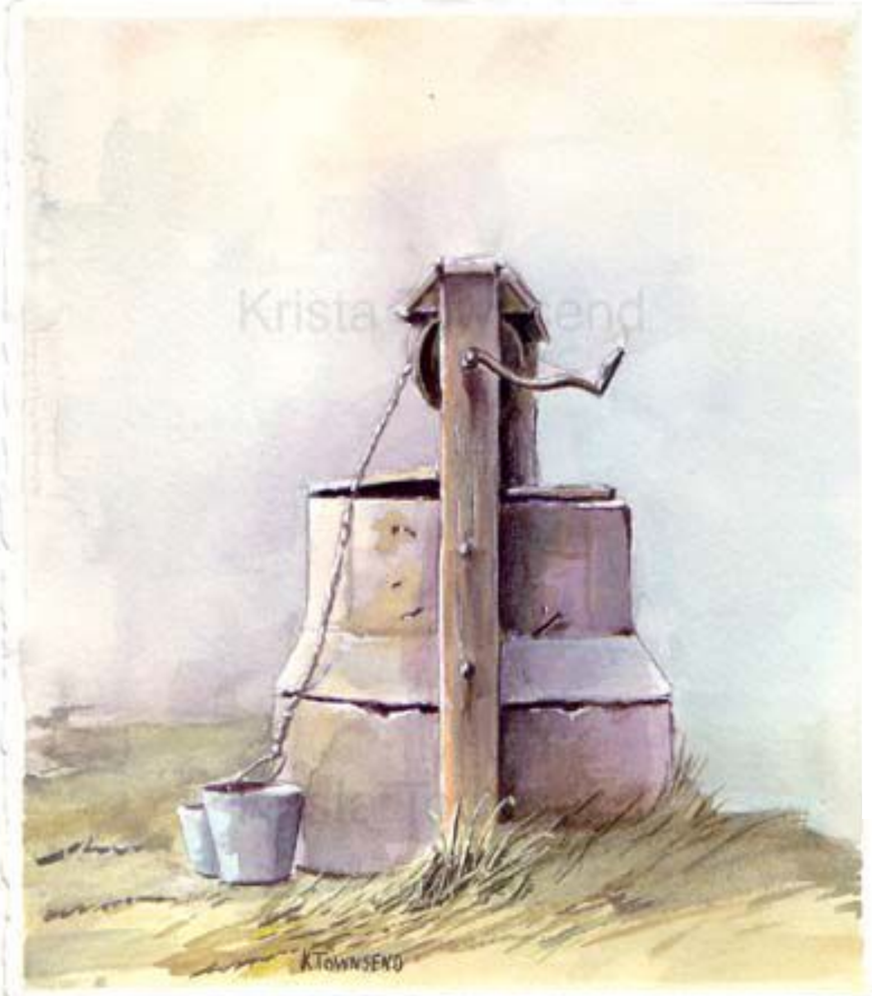
UNIT III
CENTRIFUGAL PUMP

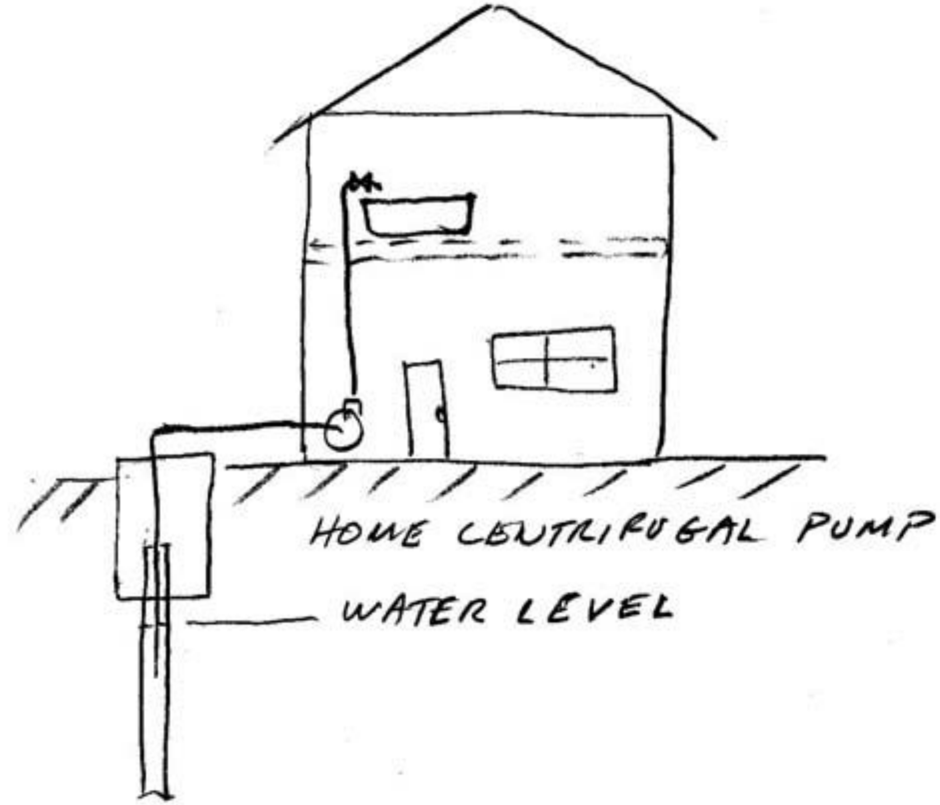


SYALLBUS

- Introduction & classification of rotodynamic Pumps
- Main Components of Centrifugal Pump
- Construction and Working of Centrifugal Pump
- Types of heads
- Velocity triangles and their analysis
- Effect of outlet blade angle
- Work done and Efficiency
- Series and parallel operation of pumps
- Priming of pumps
- specific speed

- **Study of Cavitation, NPSH, Thoma’s cavitation factor, maximum suction lift.**
- **Study and Trial on Centrifugal Pump and plotting the operating characteristics**





The old days.

Typical residential pump system

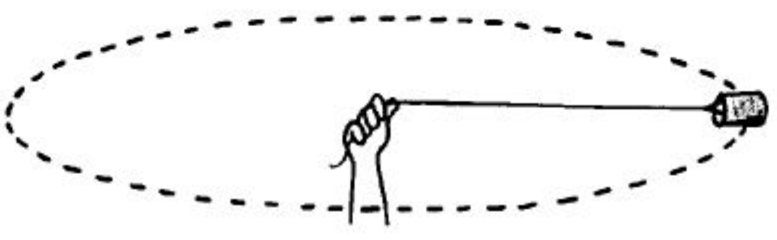
Definition

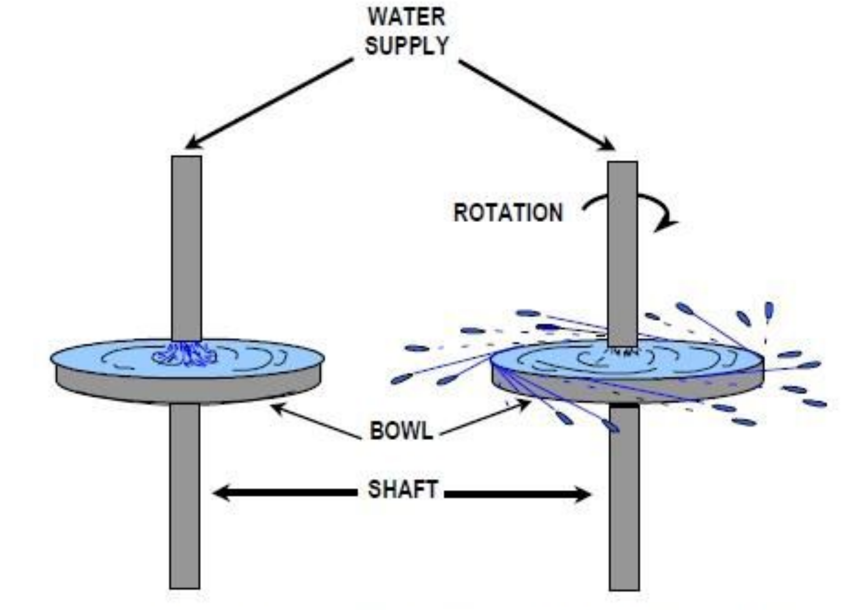
- The pumps are defined as the **hydraulic machines** which **convert the mechanical energy** into **hydraulic energy** which is mainly in the form of **pressure energy.**

- If the **mechanical energy** is converted into **hydraulic energy**, by means **of centrifugal force** acting on the fluid, the pump is known as **centrifugal pump.**

CENTRIFUGAL FORCE

- The word, ' centrifugal ' is derived from the latin language
- It is formed from two words
- Centri---- 'centre' and
- fugal ---- 'to fly away from'.
- Centrifugal - 'to fly away from the centre'.





IN THIS EXAMPLE, WATER IS FLOWING INTO A BOWL MOUNTED ON A SHAFT. WHEN THE ASSEMBLY IS ROTATED AT SPEED, CENTRIFUGAL FORCE CAUSES THE WATER TO FLY OFF THE EDGE OF THE RIM OF THE BOWL

Centrifugal Pump Principle:

When a certain mass of fluid is rotated by an external source, it is thrown away from the central axis of rotation and a centrifugal head is impressed which enables it to rise to a higher level.

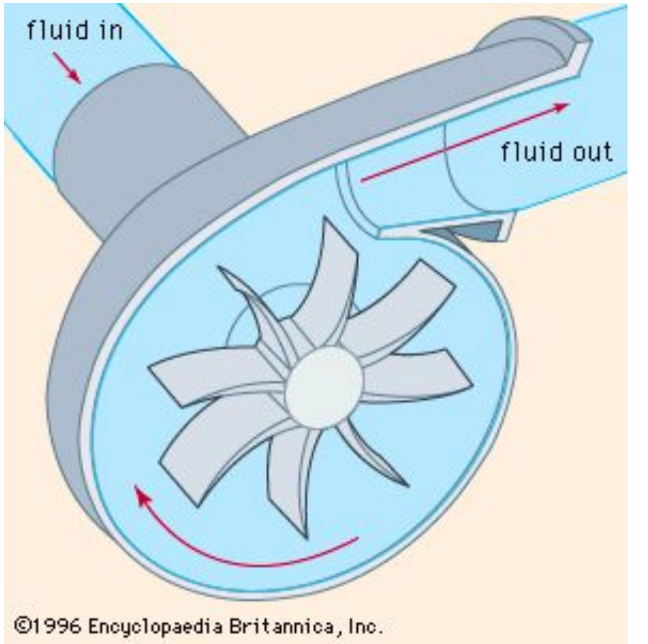
When a certain mass of fluid is rotated by an external source, the rise in pressure head of the rotating liquid take place Rise in pressure head is directly proportional to Square of tang velocity of liquid at that point

Rise in pressure head α v^2 /2g

At the outlet of impeller, where radius is more rise in pressure head will be more and liquid is discharged at high pressure head

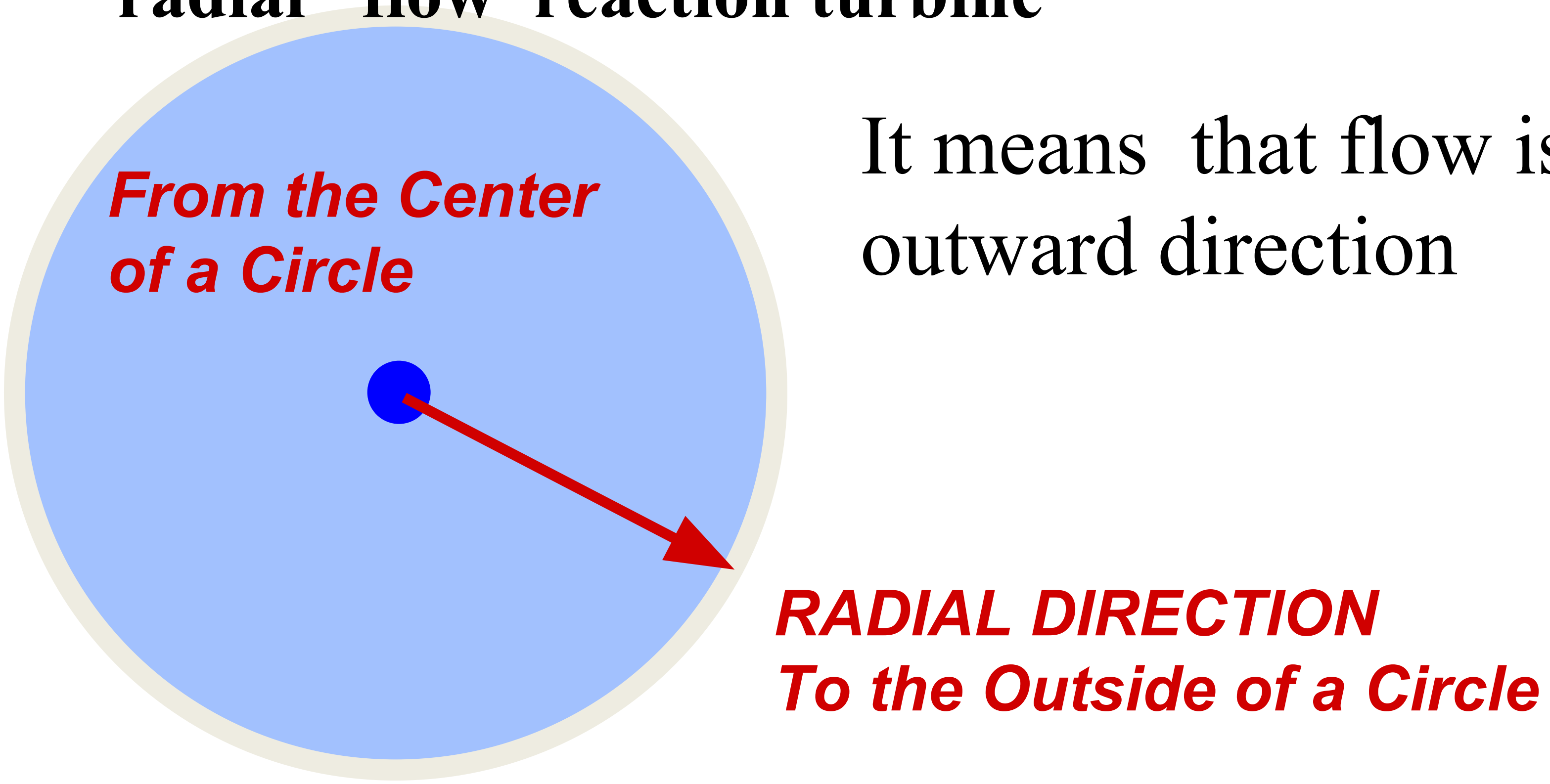
CENTRIFUGAL PUMP- Introduction

- Those pumps which have **a rotating element** are known as rotodynamic pumps.
- In these pumps, the liquid passes through the rotating element and its **angular momentum changes**.
- Due to the change of angular momentum, the pressure energy of the liquid increases.
- A centrifugal pump is a **rotodynamic machine** that converts **rotary motion into pressure energy**
- ROTO = SPIN
- DYNAMIC = CHANGE
- Flow is normally from high pressure to low pressure



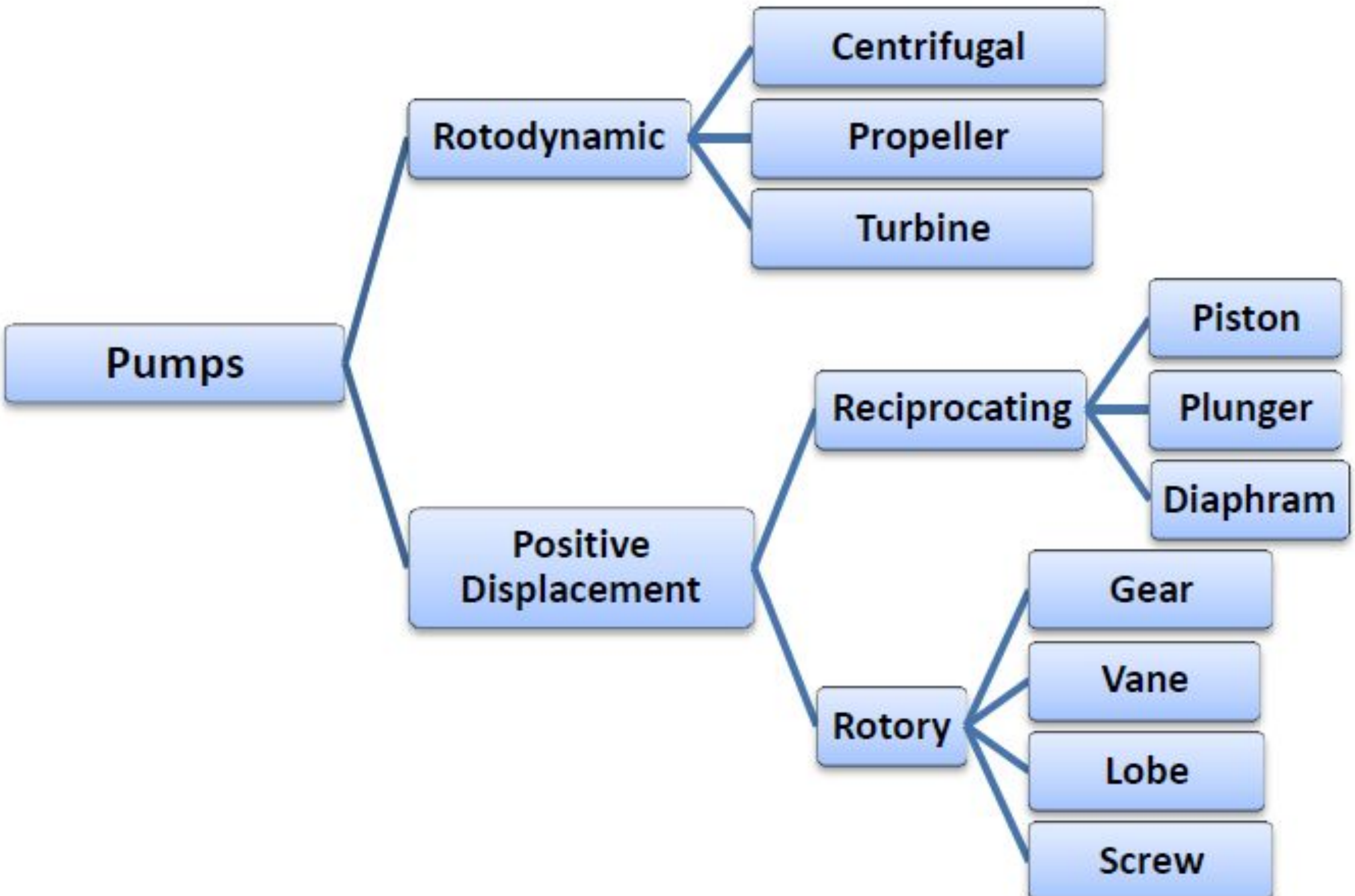
Centrifugal Pumps Theory

- A machine for moving fluid by accelerating the fluid *RADIALLY* outward.
- **Centrifugal pump** **Reverse** **inward radial flow reaction turbine**



It means that flow is in outward direction

Classification of Pump



TYPES OF CENTRIFUGAL PUMPS

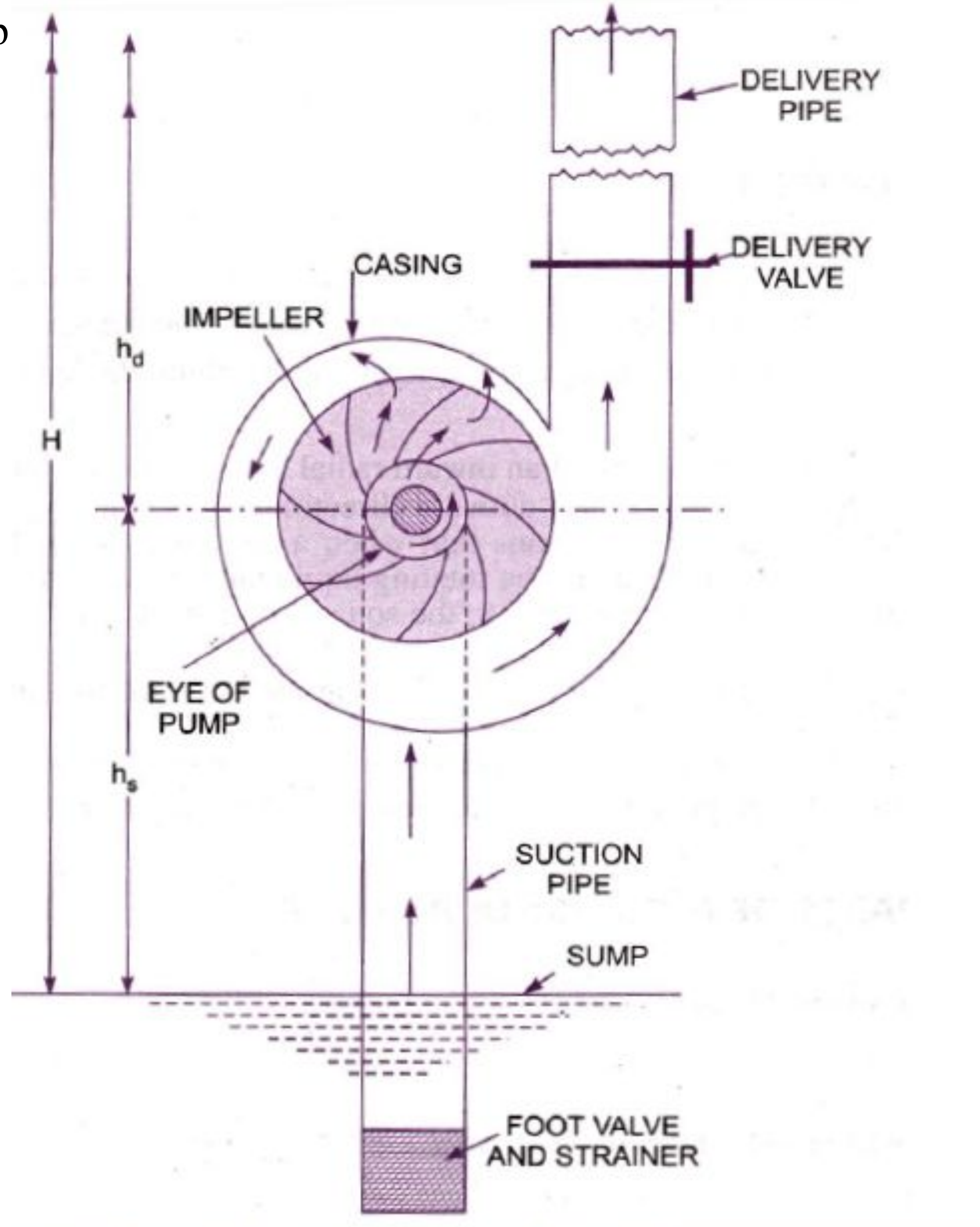
- According to the type of casing
  - **Volute casing**
  - **Vortex casing**
  - **Diffuser casing**
- According to the number of stages
  - **Single stage**
  - **Multi stage**
- According to the types of impellers
  - **Single suction impeller**
  - **Double suction impeller**
- According to the shape of the vanes (blades) of the impeller
  - **Radial flow impeller**
  - **Axial flow impeller**
  - **Mixed flow impeller**

Construction & Working of Centrifugal Pump

Main Components of a Centrifugal Pump

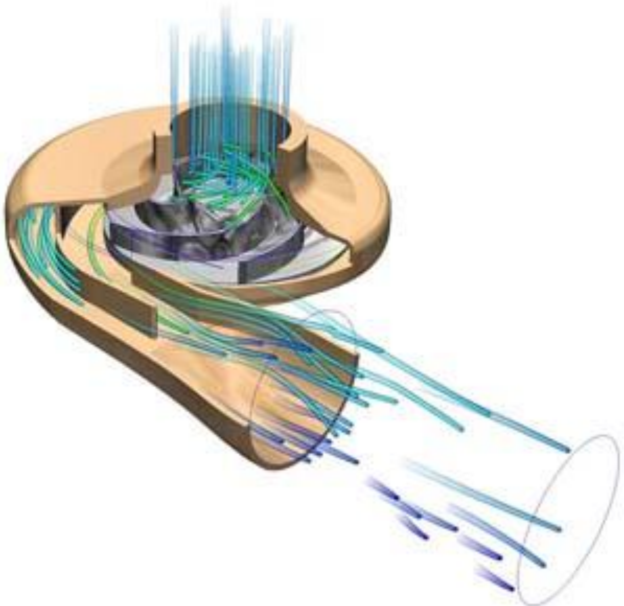
1. Impeller
2. Casing
3. Suction Pipe with strainer and foot valve
4. Delivery Pipe
5. shaft

Main Components of a Centrifugal Pump



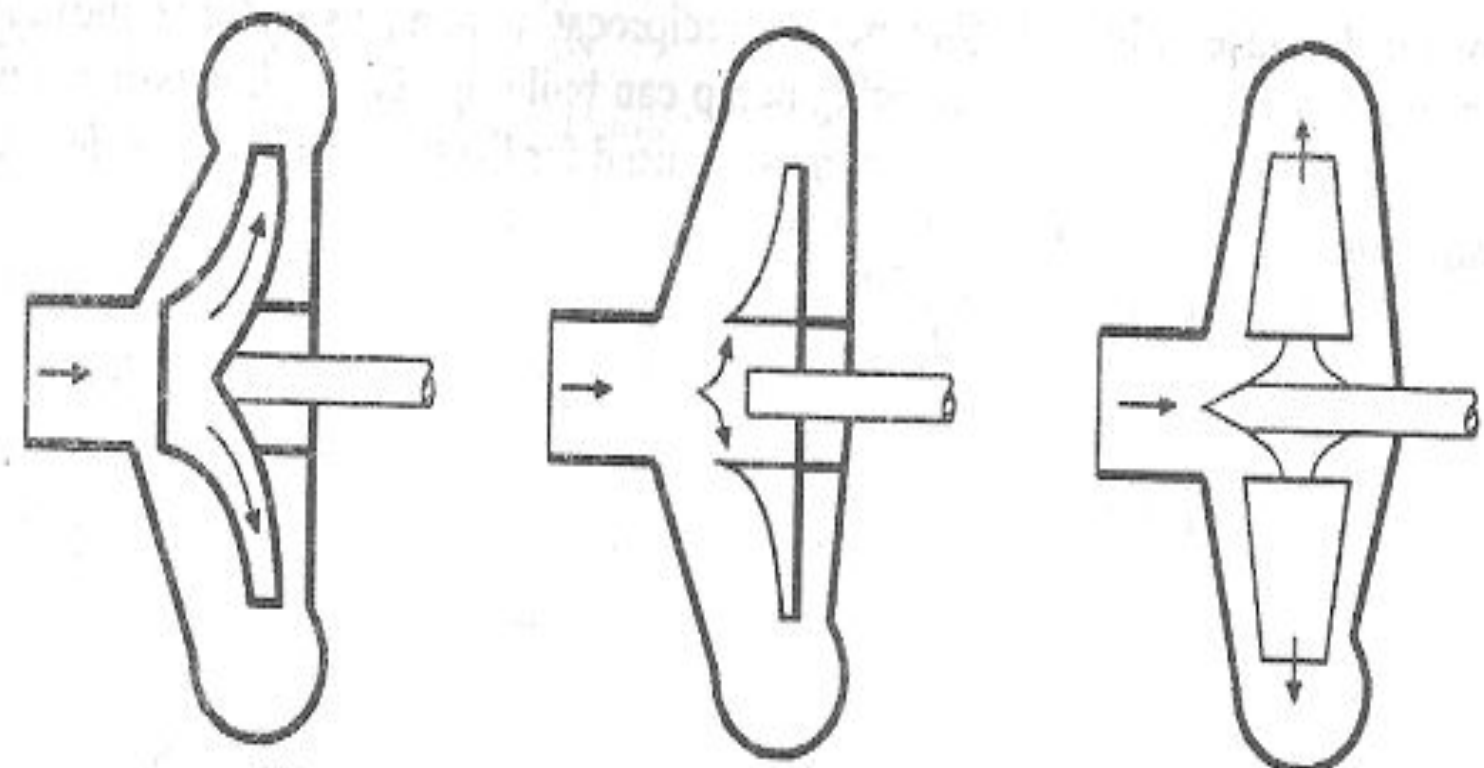
Impeller

- Centrifugal pump is **hydraulic machine** with a **rotating part** called **Impeller.**
- The rotating part of a centrifugal pump is called ‘impeller’.
- It consists of **a series of backward curved vanes or blades**. The impeller is mounted on a shaft which is connected to the shaft of an electric motor. The impellers may be classified as



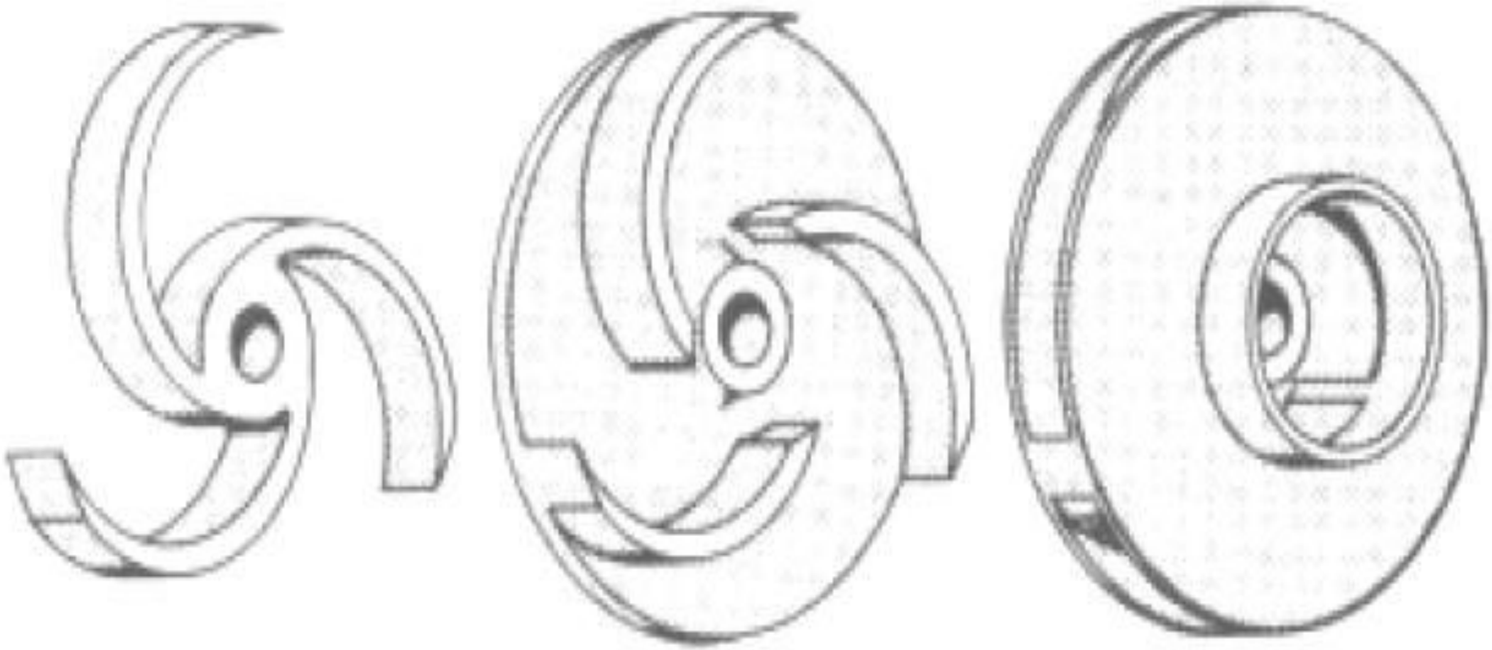
The impeller is driven by a shaft which is connected to the shaft of an electric motor

]***Closed Impeller (a)Semi open Impeller(b)***
Open Impeller (c)



(a) (b) (c)

Open Impeller Semi - Open Closed



Curved Vanes , Single Shroud and Double Shroud

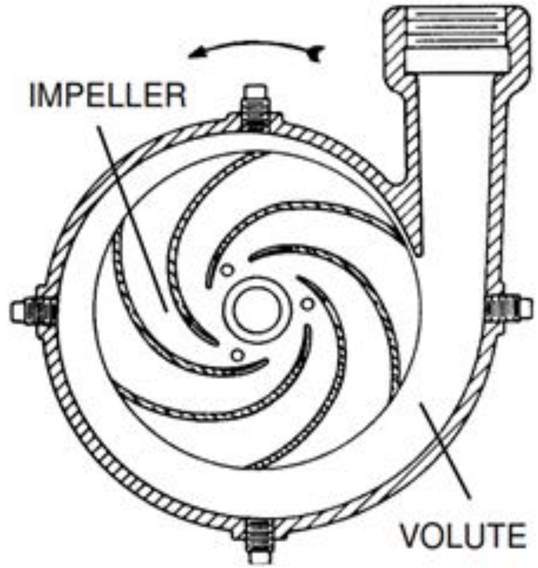
Casing

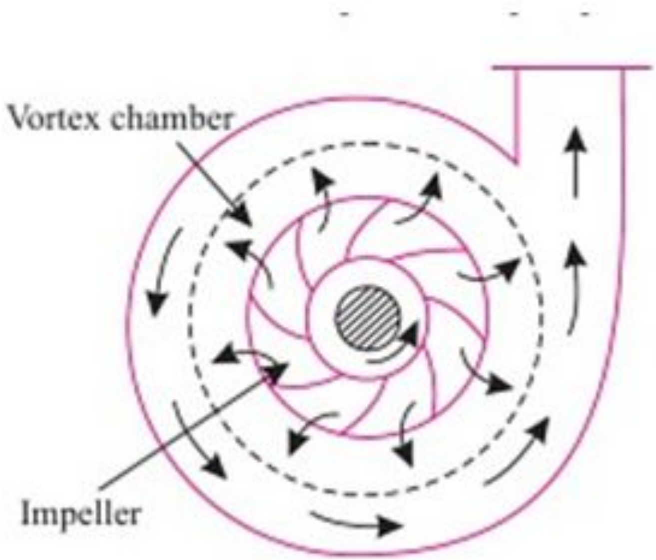
- The casing of a centrifugal pump is **similar** to the casing of a reaction turbine.
- It is an air-tight passage surrounding the impeller
- It surrounding the impeller and is designed in such a way that the kinetic energy of the water discharged at the outlet of the impeller is converted into pressure energy before the water leaves the casing and enters the delivery pipe.
- It is designed to direct the liquid to the impeller and lead it away
- **Types of casing**

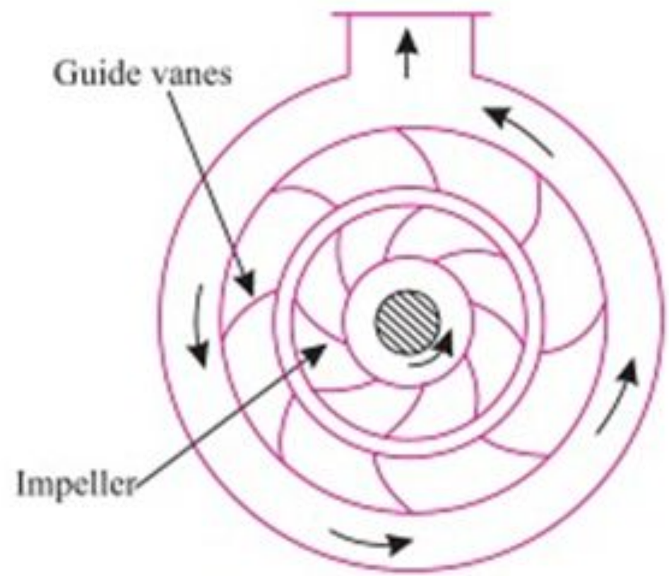
1. Volute casing
2. Vortex casing
3. Diffuser casing or Casing with guide blades

Types of casings

Volute Casing







Vortex Casing

Casing with guide blades

a) *Volute Casing:* Volute = spiral

- Figure Shows the volute casing, which surrounds the impeller.
- It is of **spiral type** in which area of flow increases gradually.
- **The increase in area of flow decreases the velocity of flow. The decrease in velocity increases the pressure of the water flowing through the casing.**
- It Captures the liquid exiting the impeller
- It Converts **kinetic energy (velocity) into potential energy (pressure)**

b) Vortex Casing:

- **A circular chamber** is introduced between the **volute casing** and the impeller as shown in Figure known as **Vortex Casing** or whirlpool chamber.
- The liquid leaving the impeller blades at a high pressure moves freely in this vortex chamber. Its velocity head is gradually transformed into pressure head.
- By introducing the circular chamber, the loss of energy due to the formation of eddies is reduced to a considerable extent.

(c) ***Casing with Guide Blades***:

- This casing is shown in Figure, in which the impeller is surrounded by a concentric casing with fixed guide vanes. **The ring of fixed guide vanes is known as diffuser.**
- **The guide vanes are designed in which a way that the water from the impeller inters the guide vanes without shock.**

- Also the area of the guide vanes increases, thus reducing the velocity of flow through guide vanes and consequently increasing the pressure of water. The water from the guide vanes then passes through the surrounding casing, which is in most of the cases concentric with the impeller as shown in Figure

Suction Pipe with a foot-valve and a strainer

- Suction pipe is connected at its **upper end** to the **inlet of the pump** at the **center of the impeller**, i.e **at the suction eye of the Impeller.**
- The **lower end** of the suction pipe **dips into liquid in a sump** from which the liquid is to be lifted up.
- The liquid from the **sump** enters the **strainer.** The strainer filters the impurities and the liquid then passes through foot valve to enter the suction pipe.
- Foot valve is a **non return valve**. **i.e one way valve**. It **opens** only in the **upward direction**. Therefore, the liquid will pass through the foot valve upwards only. It will not allow the liquid to flow downwards back to the pump.

Delivery Pipe:

- A pipe whose **one end** is connected to **the outlet of the pump** and **other end** delivers the **water at a required height** is known as **delivery pipe.**
- Just near the outlet of the pump on the delivery pipe, a delivery valve is provided.
- It controls the flow of fluid from the pump into the delivery pipe.

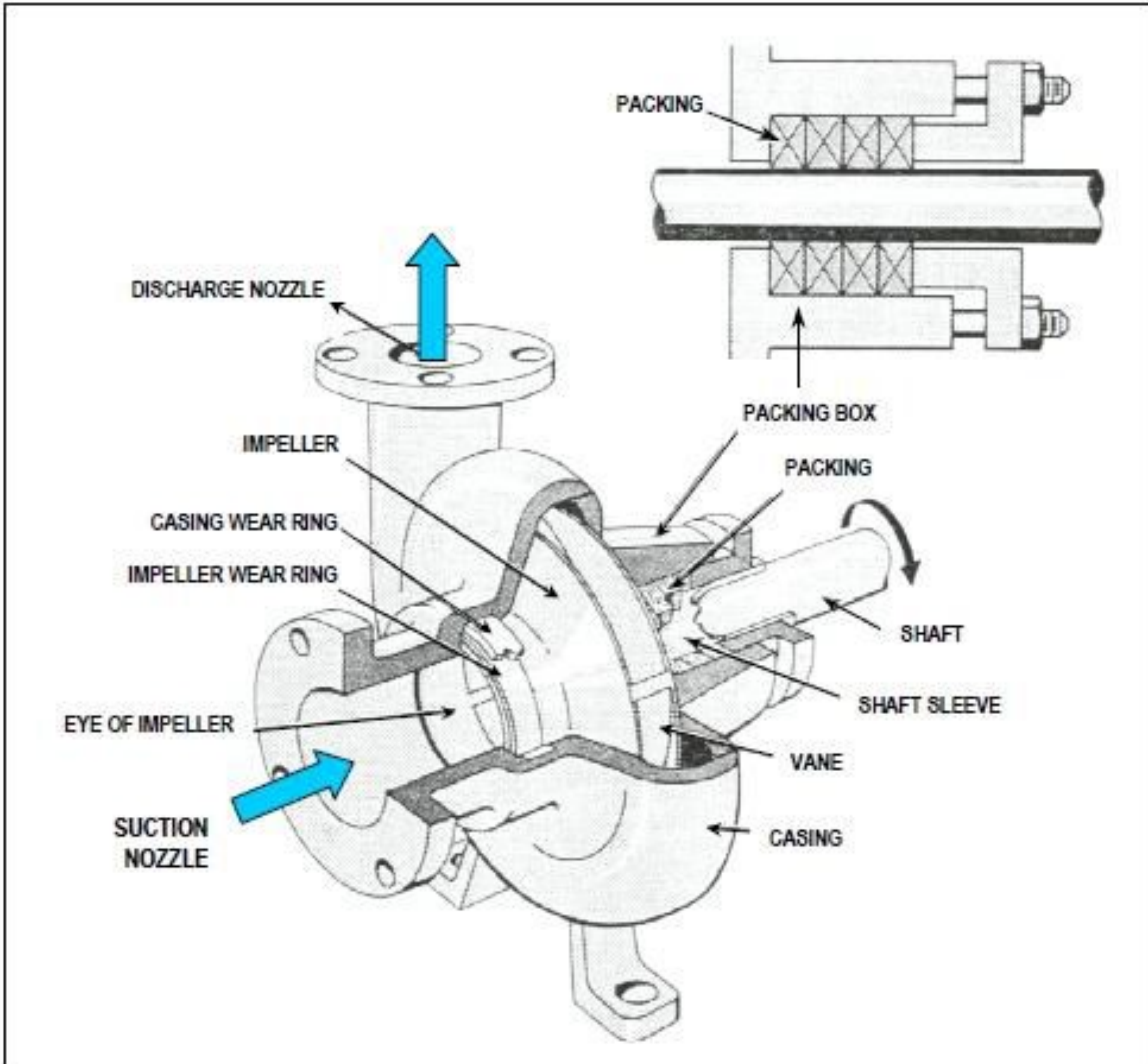
Shaft:

- Shaft is coupled to motor. It transfers the torque from motor to impeller.

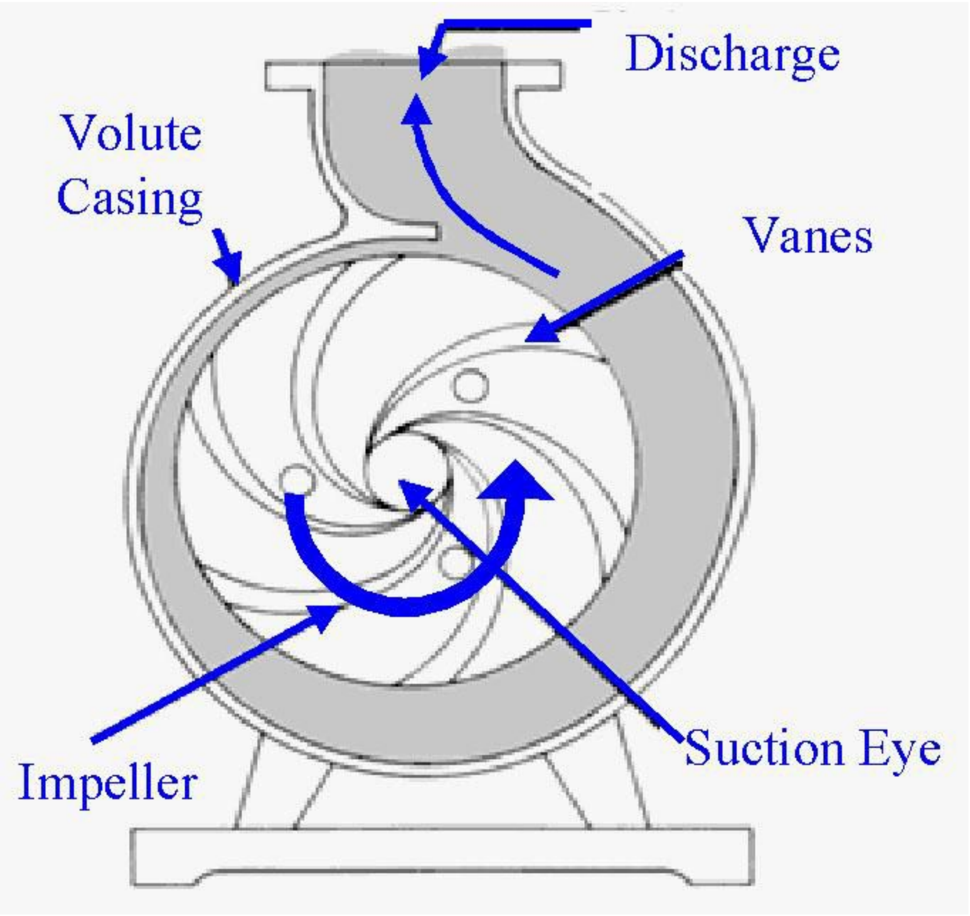
Foot Valve and Strainer

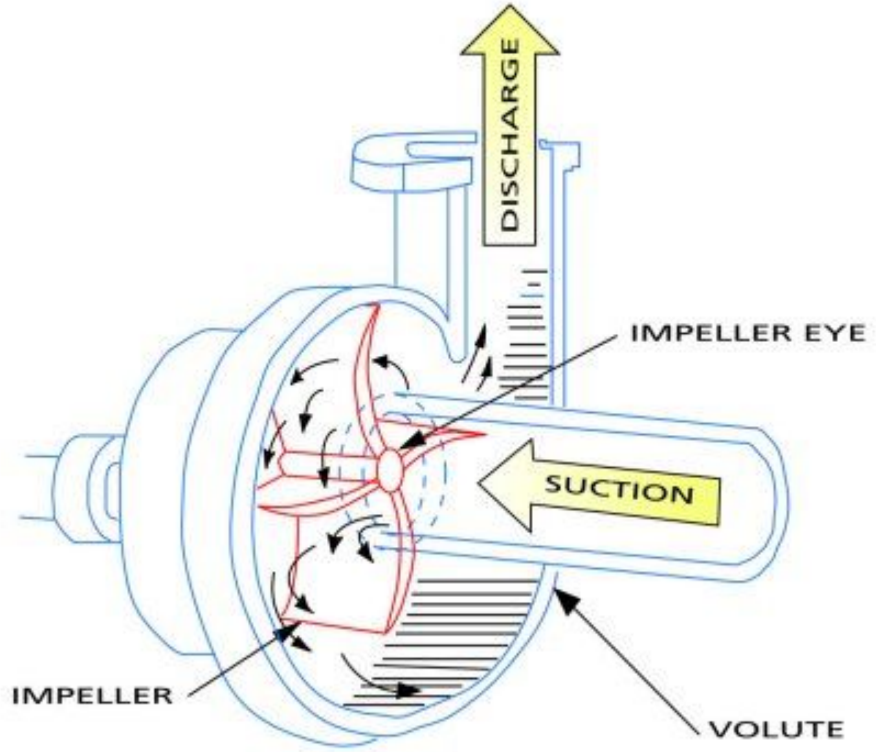
Strainer

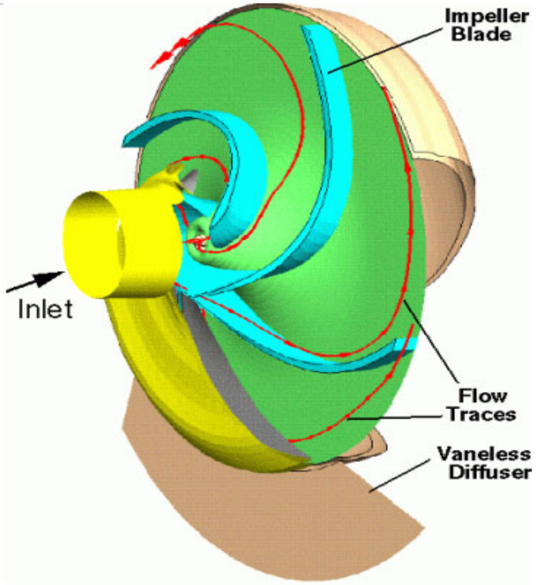
Foot Valve


PACKING
DISCHARGE NOZZLE
PACKING BOX
IMPELLER
PACKING
CASING WEAR RING
IMPELLER WEAR RING
SHAFT
SHAFT SLEEVE
EYE OF IMPELLER
VANE
SUCTION NOZZLE
CASING

Principle of operation







Working Principle

- The **delivery valve is closed** and the pump is **primed** that is, suction pipe, casing and portion of the delivery pipe up to the delivery valve are completely filled with the liquid (to be pumped) so that no air pocket is left.
- Keeping the **delivery valve still closed the electric motor is started** to rotate the impeller. The rotation of the impeller causes strong suction or vacuum just at the eye of the casing.
- The speed of the impeller is gradually increased till the impeller rotates at its normal speed and develops normal energy required for pumping the liquid.

Working Principle

- After the **impeller attains the normal speed** the **delivery valve is opened** when the liquid is continuously sucked (from sump or well) up to the suction pipe, it passes through the eye of casing and enters the impeller at its center or it enters the impeller vanes at their inlet tips.
- This **liquid is impelled out by the rotating vanes** and it becomes out at **the outlet tips of the vanes into the casing.**
- Due to impeller action the **pressure head as well as velocity head of the liquid are increased** (some of this velocity head is converted into pressure head in the casing and in the diffuser blades / vanes if they are also provided.

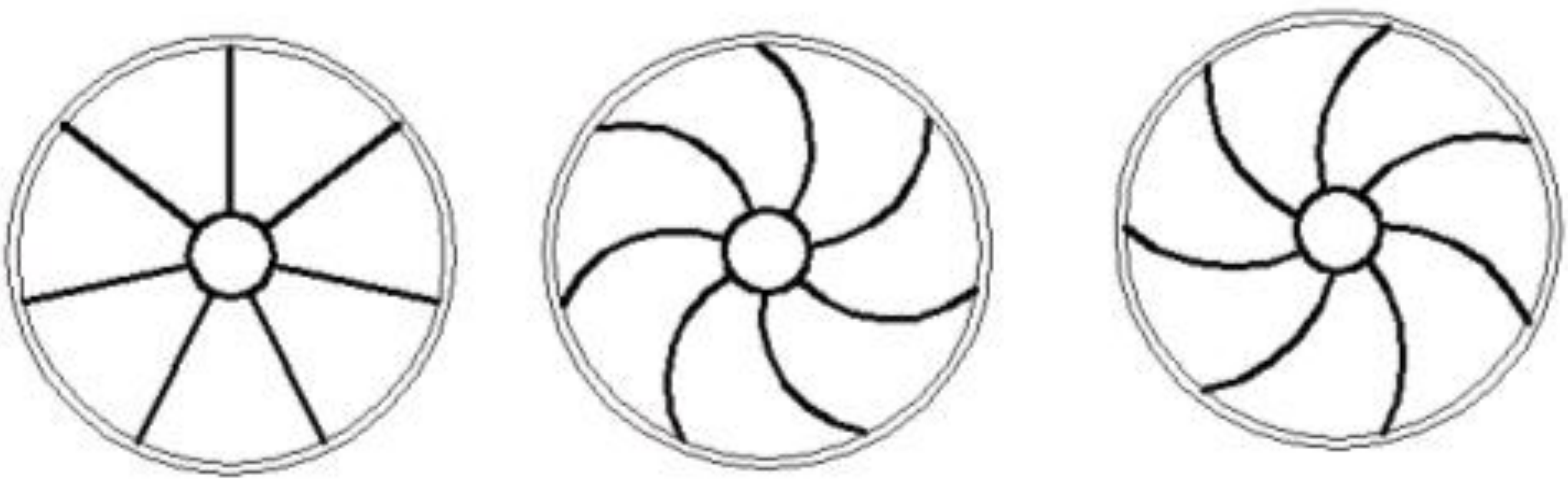
Working Principle

- From **casing, the liquid passes into pipe and is lifted to the required height** (and discharged from the outlet or upper end of the delivery pipe).
- So long as motion is given to the impeller and there is supply of liquid to be lifted the process of lifting the liquid to the required height remains continuous.
- When the **pump is to be stopped the delivery valve should be first closed, otherwise there may be some back flow from the reservoir**.

Type of Impeller & Velocity Triangle

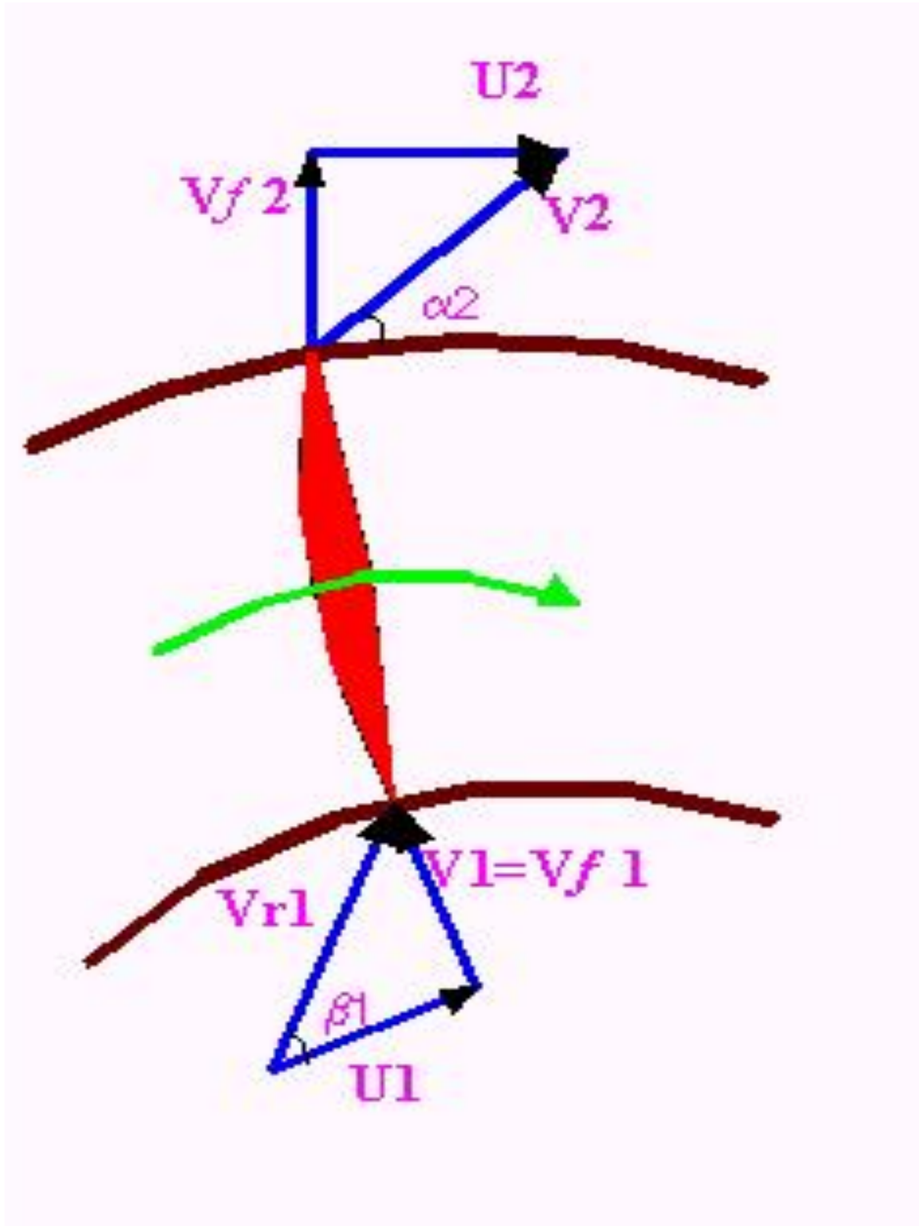
- There are three main categories of impeller due type of impeller's vane, which are used in the centrifugal pumps as

▪ Radial vanes, Fig. (a).
▪ Backward vanes, Fig. (b).
▪ Forward vanes, Fig. (c).

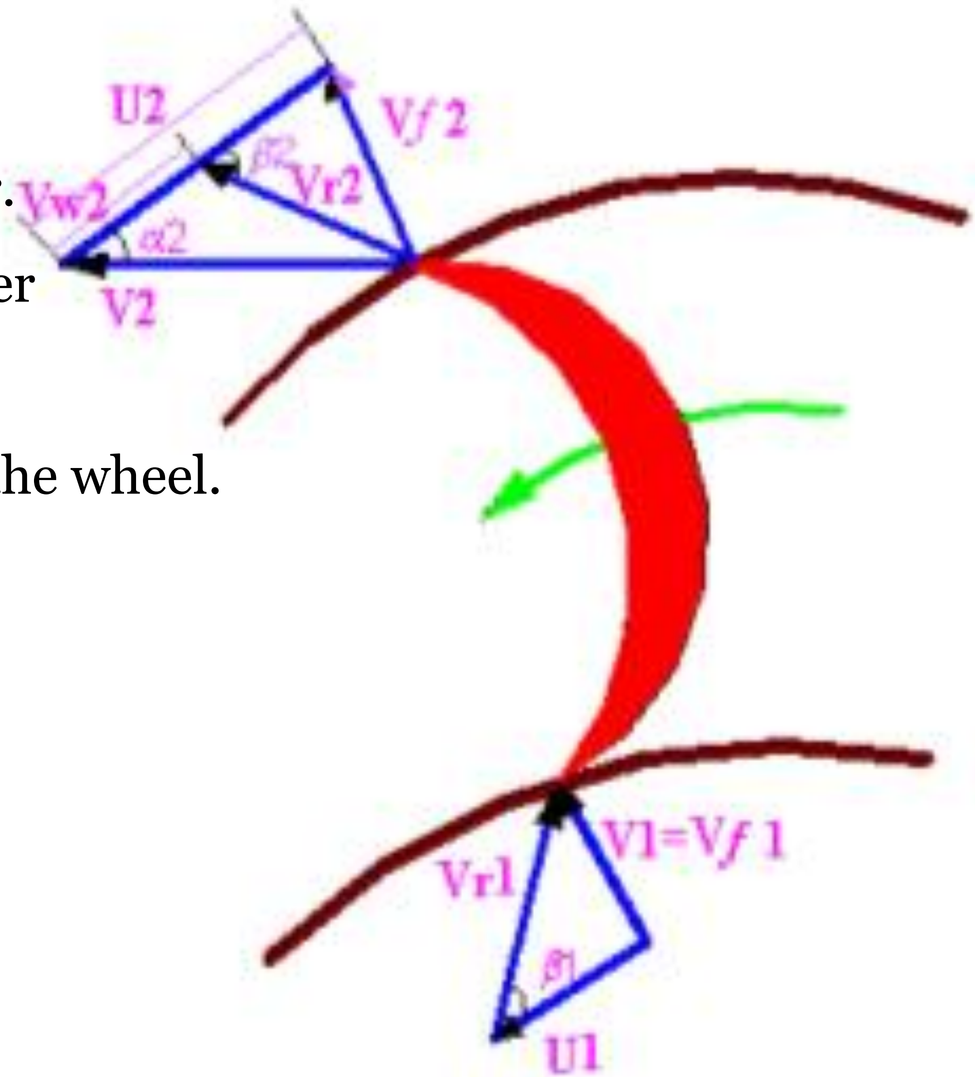


(a) (b) (c)

b) when $\beta_2 = 90^o$, the radial curved vanes of the impeller.



- when $\beta_2 > 90^o$, the Forwards curved vanes of the impeller.



where :

V = absolute velocity of the water.

U = Tangential velocity of impeller

(peripheral velocity).

$\boldsymbol{V_r}$ = relative velocity of water to the wheel.

$\boldsymbol{V_f}$ = velocity flow.

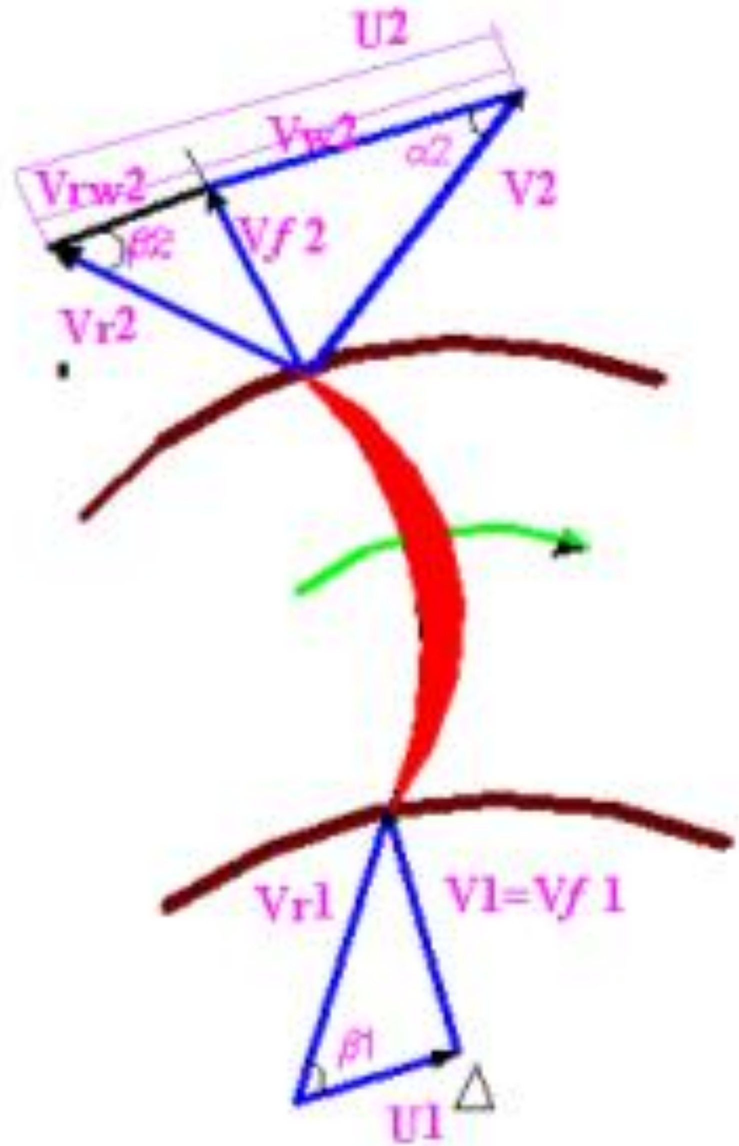
N = Speed of impeller in (rpm).

α = Vane angle

B = angle at which water leaves

Vw = Velocity of whirl

when $\beta_2 < 90^o$, the Backwards curved vanes of the impeller.



Velocity triangle for centrifugal pump

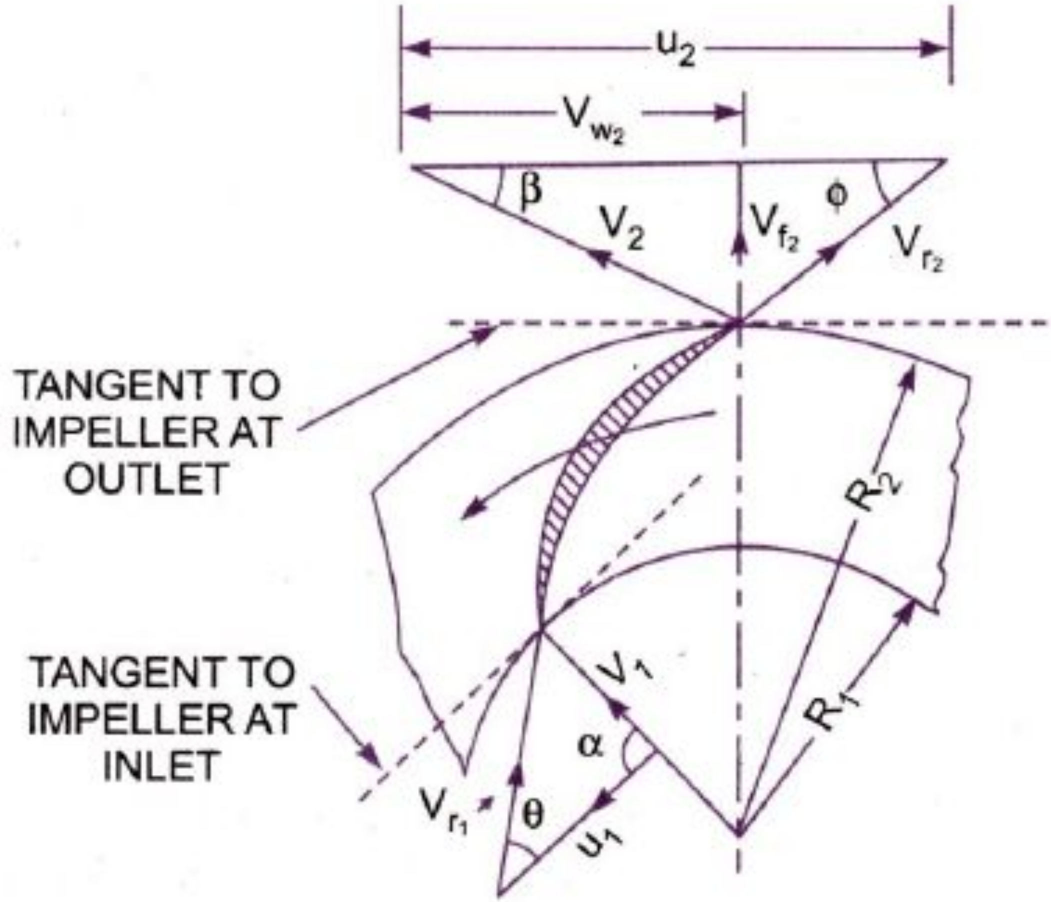


Fig. 4.3 Velocity triangles of centrifugal Pump

- In case of the centrifugal pump, work is done by the impeller on the water. The expression for the work done by the impeller on the liquid is obtained by drawing velocity triangles at the inlet and outlet of the impeller in the same way as for a turbine.
- Fig. 4.3 shows the vane of impeller and velocity triangles at the inlet and outlet of the impeller.

- The water enters the impeller radially at inlet for the best efficiency of the pump, which means the absolute velocity of water at inlet makes an angle of 90° with the direction of motion of the impeller at inlet. Hence $\alpha = 90° \ and \ V_{w1} = 0$.

- Let,

  N = Speed of the impeller in rpm.

  D_1 = Diameter of impeller at the inlet

  D_2 = Diameter of impeller at the outlet

  u_1 = Tangential velocity of impeller at the inlet $= \frac{\pi D_1 N}{60}$

  u_2 = Tangential velocity of impeller at the outlet $= \frac{\pi D_2 N}{60}$

  V_1 = Absolute velocity of water at the inlet

  V_{r1} = Relative velocity of water at the inlet

  α = Angle made by absolute velocity at inlet with the direction of motion of vane

  θ = Angle made by relative velocity at inlet with the direction of motion of vane and

  V_2, V_{r2}, β *and* φ are corresponding values at outlet.

A centrifugal pump is the reverse of a radially inward flow reaction turbine. But in case of a radially inward flow reaction turbine, the work done by the water on the runner per sec per unit weight is given by,

$$= \frac{1}{g}(V_{w1}u_1 - V_{w2}u_2)$$

Therefore, work done by the impeller on the water per sec per unit weight,

$$= -[Work\ done\ in\ case\ of\ turbine]$$

$$= -\frac{1}{g}(V_{w1}u_1 - V_{w2}u_2)$$

$$= \frac{1}{g}(V_{w2}u_2 - V_{w1}u_1)$$

$$= \frac{1}{g}(V_{w2}u_2) - - - - - - - - -(4.1) \qquad (\because V_{w1} = 0)$$

Work done by the impeller on water per sec,

$$= \dot{m}(V_{w2}u_2)$$

$$= \rho Q(V_{w2}u_2) - - - - - - - - -(4.2)$$

Discharge,

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

Where,

B_1 *and* B_2 are the width of the impeller at the inlet and outlet respectively.

Equation (4.1) gives the head imparted to the water by the impeller ***or*** *energy given by impeller to water per sec per unit weight.*

PUMP HEAD

The term “head” likely derives from the elevation difference available to power a water wheel

Head and pressure are interchangeable terms provided that they are expressed in their correct units.

Major Heads in a Centrifugal Pump

- **Suction Head**
- **Delivery Head**
- **Static Head**
- **Manometric Head**

Pump Head

- The head of a pump can be expressed in metric units as:

$$\boldsymbol{head = (p_2 - p_1)/(\rho g) + (v_2^2 - v_1^2)/(2g) + (z_2 - z_1)}$$

- *where*
- *h = total head developed (m)*
- *p_2 = pressure at outlet (N/m^2)*
- *p_1 = pressure at inlet (N/m^2)*
- *ρ = density of liquid (kg/m3)*
- *g = acceleration of gravity (9.81) m/s^2*
- *v_2 = velocity at the outlet (m/s)*

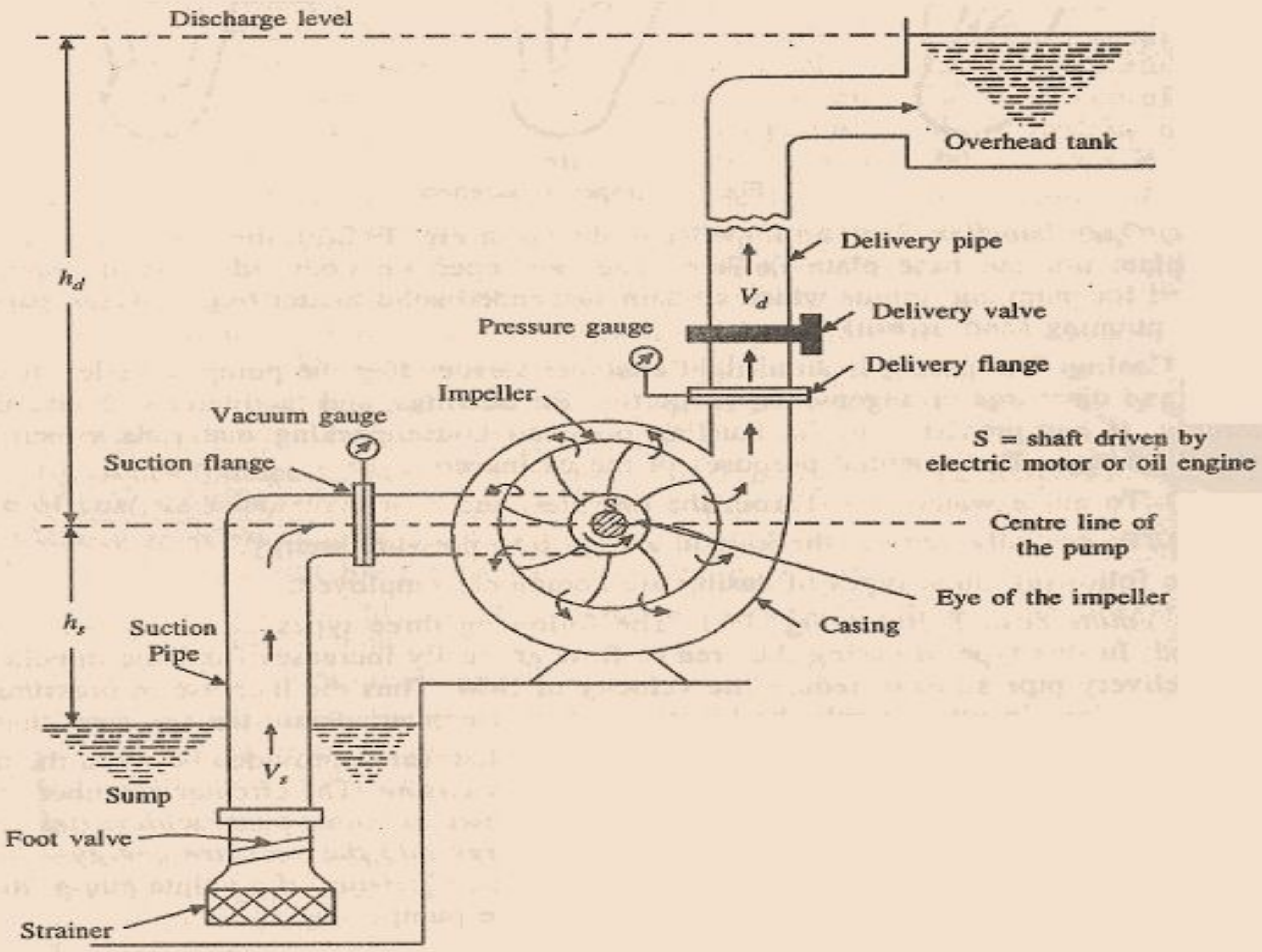
1) Suction Head: (hs)

The distance to which the pump has to lift water. In short it is the distance from surface of water to be lifted and centre of pump.

Suction head is the head at the pump inlet (suction location)

2) Delivery Head: (hd)

The distance to which the pump has to deliver water. In short it is the distance from surface of water delivered and centre of pump.


Discharge level
Overhead tank
Delivery pipe
V_d
Delivery valve
Pressure gauge
Delivery flange
Impeller
Vacuum gauge
S = shaft driven by electric motor or oil engine
Suction flange
S
Centre line of the pump
h_d
h_s
Eye of the impeller
Casing
Suction Pipe
V_s
Sump
Foot valve
Strainer

3) Static Head:

Sum of Suction and Delivery heads

Static head = hs + hd

4) Manometric Head

Head against which centrifugal pump has to work.

$$H_m = H + h_f + \frac{V_d^2}{2g}$$

H_m = Manometric Head

H = sum of actual lifts (hs +hd)

hf = frictional losses in actual heads (hfs + hfd)

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

H_m = Head imparted by impeller to water – losses of head in pump

= $V_{w2}u_2/2g$ - losses of head in pump

- Total inlet head to the pump = $(p_1 + \rho g) + (V_1^2 / 2g) + z_1$
- Total outlet head of the pump = $(p_2 + \rho g) + (V_2^2 / 2g) + z_2$

- Hm = **Total outlet head of the pump - Total inlet head to the pump**

Efficiencies

Efficiencies:

In case of a centrifugal pump, the power is transmitted from the shaft of the electric motor to the shaft of the pump and then to the impeller. From the impeller, the power is given to the water.

The followings are the important efficiencies of a centrifugal pump:

1. Manometric Efficiency
2. Mechanical Efficiency and
3. Overall Efficiency

1. **Manometric Efficiency (η_{man})**

- It is defined as the ratio of the manometric head developed by the pump to the head imparted by the impeller to the liquid.

$$\therefore \eta_{man} = \frac{Manometric\ head}{Head\ imparted\ by\ impeller\ to\ liquid}$$

$$\therefore \eta_{man} = \frac{H_m}{\left(\frac{V_{w2}u_2}{g}\right)} = \frac{gH_m}{V_{w2}u_2} ---------(4.8)$$

- The power at the impeller of the pump is more than that the power given to the liquid at outlet of the pump.

$$\boldsymbol{Power\ given\ to\ water\ at\ outlet\ of\ the\ pump = \frac{WH_m}{1000} = \frac{\rho g Q H_m}{1000}\ k}$$

$$\boldsymbol{Power\ at\ the\ impeller = \frac{WD\ by\ the\ impeller\ per\ sec}{1000} = \frac{\rho Q(V_{w2}u_2)}{1000}\ kW}$$

Mechanical Efficiency $(\boldsymbol{\eta_m})$

It is defined as the ratio of the power actually delivered by the impeller to the power at the shaft of the centrifugal pump.

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

$$\therefore \eta_m = \frac{\dot{m}(V_{w2}u_2)/1000}{S.P.\, in\, kW} ---------(4.9)$$

Overall Efficiency $(\boldsymbol{\eta_o})$

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

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

$$Power\ input\ to\ the\ pump = Power\ supplied\ by\ the\ electric\ motor$$
$$= Shaft\ power\ of\ the\ pump$$

$$\therefore \eta_o = \frac{\left(\frac{WH_m}{1000}\right)}{S.P.} ---------(4.10)$$

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

Minimum starting speed

When a pump is started, water will not flow until the pressure developed by the impeller is sufficient to overcome the manometric head.

The water will start flowing only if the centrifugal head or pressure head raised by the impeller be at least equals to or more than manometric head.

Centrifugal head or head raised by the impeller,

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

Where,

u_1 = Tangential velocity of impeller at inlet

u_2 = Tangential velocity of impeller at outlet

The flow of water commence only if,

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

For minimum speed,

$$\frac{u_2^2}{2g} - \frac{u_1^2}{2g} = H_m \text{ - - - - - - - - -}(4.17)$$

But,

$$\eta_{man} = \frac{gH_m}{V_{w2}u_2}$$

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

Substitute these values of H_m in equation (4.17), we get,

$$\frac{u_2^2}{2g} - \frac{u_1^2}{2g} = \eta_{man} \times \frac{V_{w2}u_2}{g}$$

But,

$$u_1 = \frac{\pi D_1 N}{60} \ and \ u_2 = \frac{\pi D_2 N}{60}$$

$$\therefore \frac{1}{2g}\left(\frac{\pi D_2 N}{60}\right)^2 - \frac{1}{2g}\left(\frac{\pi D_1 N}{60}\right)^2 = \eta_{man} \times \frac{V_{w2}}{g} \times \frac{\pi D_2 N}{60}$$

By dividing with $\frac{\pi N}{60g}$, we get,

$$\frac{\pi N D_2^2}{120} - \frac{\pi N D_1^2}{120} = \eta_{man} \times V_{w2} D_2$$

$$\therefore \frac{\pi N}{120}[D_2^2 - D_1^2] = \eta_{man} \times V_{w2} D_2$$

$$\therefore \boldsymbol{N = \frac{120 \times \eta_{man} \times V_{w2} D_2}{\pi[D_2^2 - D_1^2]}} - - - - - - - - - (\mathbf{4.18})$$

Equation (4.18) gives minimum starting speed of a centrifugal pump.

Maximum Suction lift

Maximum Suction Lift or Suction Height

Fig. 4.4 shows a centrifugal pump that lifts a liquid from a sump. The free surface of the liquid is at a depth of h_s below the pump axis. The liquid is flowing with a velocity of V_s in the suction pipe.

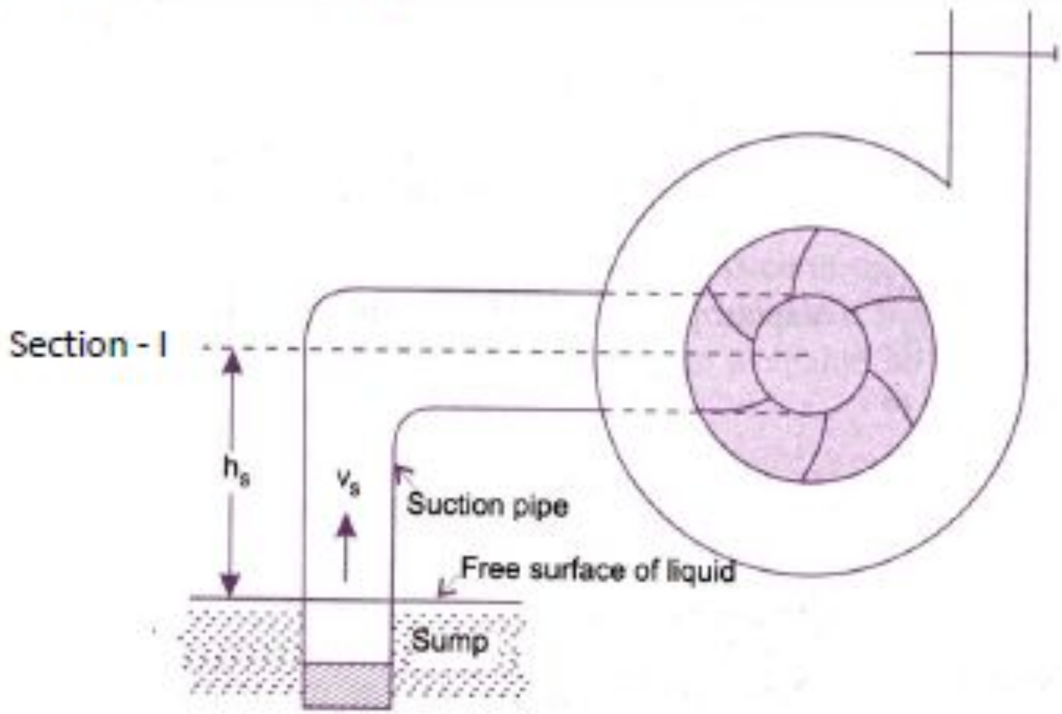


Fig. 4.4 Suction lift

Let, h_s is the suction height or suction lift.

Applying Bernoulli's equation at the free surface of liquid in the sump and section - I in the suction pipe just at the inlet of the pump.

Take free surface of liquid as datum line, we get,

$$\frac{P_a}{\rho g} + \frac{V_a^2}{2g} + Z_a = \frac{P_1}{\rho g} + \frac{V_1^2}{2g} + Z_1 + h_L$$

Where,

P_a = Atmospheric pressure on the free surface of liquid

V_a = Velocity of liquid at the free surface $\cong 0$

Z_a = Height of free surface from datum line = 0

P_1 = Absolute pressure at the inlet of the pump

V_1 = Velocity of liquid through suction pipe $= V_s$

Z_1 = Height of inlet of pump from datum line $= h_s$

h_L = Loss of head in foot valve, strainer and suction pipe $= h_{fs}$

$$\therefore \frac{P_a}{\rho g} + 0 + 0 = \frac{P_1}{\rho g} + \frac{V_s^2}{2g} + h_s + h_{fs}$$

$$\therefore \frac{P_a}{\rho g} = \frac{P_1}{\rho g} + \frac{V_s^2}{2g} + h_s + h_{fs}$$

$$\therefore \frac{P_1}{\rho g} = \frac{P_a}{\rho g} - \left(\frac{V_s^2}{2g} + h_s + h_{fs}\right) - - - - - - - - -(4.19)$$

For finding the maximum suction lift the pressure at the inlet of the pump should not be less than vapor pressure of the liquid.

Hence taking minimum pressure at the inlet of the pump equal to vapor pressure of the liquid.

We get,

$$P_1 = P_v$$

Where,

$P_v =$ Vapor pressure of the liquid in absolute unit.

$$\therefore \frac{P_v}{\rho g} = \frac{P_a}{\rho g} - \left(\frac{V_s^2}{2g} + h_s + h_{fs}\right)$$

$$\therefore h_s = \frac{P_a}{\rho g} - \frac{P_v}{\rho g} - \frac{V_s^2}{2g} - h_{fs}$$

Now taking,

$\frac{P_a}{\rho g} = H_a =$ Atmospheric pressure head

$\frac{P_v}{\rho g} = H_v =$ Vapor pressure head

$$\therefore h_s = H_a - H_v - \frac{V_s^2}{2g} - h_{fs} - - - - - - - - - - (4.20)$$

Equation 4.20 gives value of maximum suction lift (or suction height) for a centrifugal pump.

Hence if the suction height of the pump is more, then vaporization of liquid at the inlet of pump will take place and there will be a possibility of Cavitation.

Net Positive Suction Head

"It is defined as the total head developed at the pump inlet above the vapor pressure of the liquid."

It is also defined as the absolute pressure head at the inlet to the pump minus the vapor pressure head plus the velocity head. Thus,

$$NPSH = \frac{P_1}{\rho g} - \frac{P_v}{\rho g} + \frac{V_s^2}{2g}$$

Introducing the value of $\frac{P_1}{\rho g}$ from equation 4.19 in the above expression, we get,

$$NPSH = \frac{P_a}{\rho g} - \left(\frac{V_s^2}{2g} + h_s + h_{fs}\right) - \frac{P_v}{\rho g} + \frac{V_s^2}{2g}$$

$$\therefore NPSH = \frac{P_a}{\rho g} - \frac{P_v}{\rho g} - h_s - h_{fs}$$

$$\therefore NPSH = \left(H_a - h_s - h_{fs}\right) - H_v - - - - - - - - - (4.21)$$

In other words, NPSH may also be defined as the total head required to make the liquid to flow through the suction pipe to the impeller.

For any pump installation a distinction is made between the required NPSH and the available NPSH.

Required NPSH

The value of required NPSH is given by the pump manufacturer.

The value of required NPSH varies with the pump design, the speed of the pump, and the capacity of the pump.

The value of required NPSH can be calculated experimentally. For determining its value, the pump is tested with different suction lifts and minimum value of h_s is obtained at which the pump gives maximum efficiency without any objectional noise (i.e. Cavitation free).

Available NPSH

When the pump is installed the available NPSH can be determined from the equation 4.21.

In order to have Cavitation free operation of centrifugal pump, the available NPSH should be greater than the required NPSH.

[**Note:** NPSH is a measure of how much spare pull you have before the bubbles form]

NPSH NET POSITIVE SUCTION HEAD

- NPSH CURVE **= NOT PUMPING SO HOT?**

- To avoid cavitation in centrifugal pumps, the pressure of the fluid at all points within the pump must remain above saturation pressure.
- The quantity used to determine if the pressure of the liquid being pumped is adequate to avoid cavitation is the net positive suction head (NPSH).
- Divided into two parts- NPSH Available and NPSH Required
- NPSH Available :

The absolute pressure at the suction port of the pump.

The absolute dynamic head at the pump inlet (suction) in excess of the vapour pressure.

- $NPSH_A = H_A - (Hv + Hs + Hvs + Hfs)$

- Hv – Pressure head
- Hs – Suction head
- Hvs – velocity head suction
- Hfs – Frictional loses in suction pipe
- **$NPSH_A$ > $NPSH_R$** to avoid cavitation
- IT'S THE POSITIVE PRESSURE REQUIRED AT THE PUMP INLET
- NEED TO COMPARE AVAILABLE TO REQUIRED NPSH

NPSH Required

- The minimum pressure required at the suction port of the pump to keep the pump free from cavitating.
- The minimum value of $NPSH_A$ that is needed to prevent cavitation in the pump, i.e., the value of $NPSH_A$ that causes p_{min} to equal p_{vap}.
- $NPSH_R$ is determined experimentally by pump manufacturers and reported as a function of pump flow rate (usually called ‘capacity’).
- To avoid cavitation, always operate with $NPSH_A \geq NPSH_R$.

NET POSITIVE SUCTION HEAD REQUIRED

- The pump manufacturer's specified margin of suction pressure above the boiling point of the liquid being pumped, is required to prevent cavitation. This pressure is called the 'Net Positive Suction Head' pressure (NPSH).
- In order to ensure that a NPSH pressure is maintained, the Available NPSH should be higher than that required. The NPSH depends on the height and density of the liquid and the pressure above it.

Specified Speed

Specific Speed

"The specific speed of a centrifugal pump is defined as the speed of a geometrically similar pump which delivers unit quantity against a unit head."

It is used to compare the performance of different pumps.

For a centrifugal pump,

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

$$\therefore Q = \pi DB \times V_f$$

$$\therefore Q \propto DBV_f ---------(4.11)$$

Where,

$D =$ Diameter of the impeller of the pump

$B =$ Width of the impeller

We know that,

$$B \propto D$$

$$\therefore Q \propto D^2 V_f --------(4.12)$$

Tangential velocity is given by,

$$u = \frac{\pi DN}{60} \propto DN --------(4.13)$$

Now tangential velocity (u) and velocity of flow (V_f) are related to the manometric head (H_m) as,

$$u \propto V_f \propto \sqrt{H_m} --------(4.14)$$

Substituting value of u in equation (4.13), we get,

$$\sqrt{H_m} \propto DN$$

$$\therefore D \propto \frac{\sqrt{H_m}}{N}$$

Substituting value of D in equation (4.12), we get,

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

$$\therefore Q \propto \frac{H_m}{N^2} \sqrt{H_m}$$

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

$$\therefore Q = K \frac{{H_m}^{3/2}}{N^2} --------(4.15)$$

Where, K = Constant of proportionality.

By definition, if $H = 1m$ and $Q = 1\, m^3/sec$, N becomes N_s

Substituting these values in equation (4.15), we get,

$$1 = K \times \frac{1}{N_s^2}$$

$$\therefore K = N_s^2$$

Substituting value of K in equation (4.15), we get,

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

$$\therefore N_s^2 = \frac{QN^2}{{H_m}^{3/2}}$$

$$\therefore \boldsymbol{N_s = \frac{N\sqrt{Q}}{{H_m}^{3/4}}} ----------(\mathbf{4.16})$$

Multiple-Pump Operation

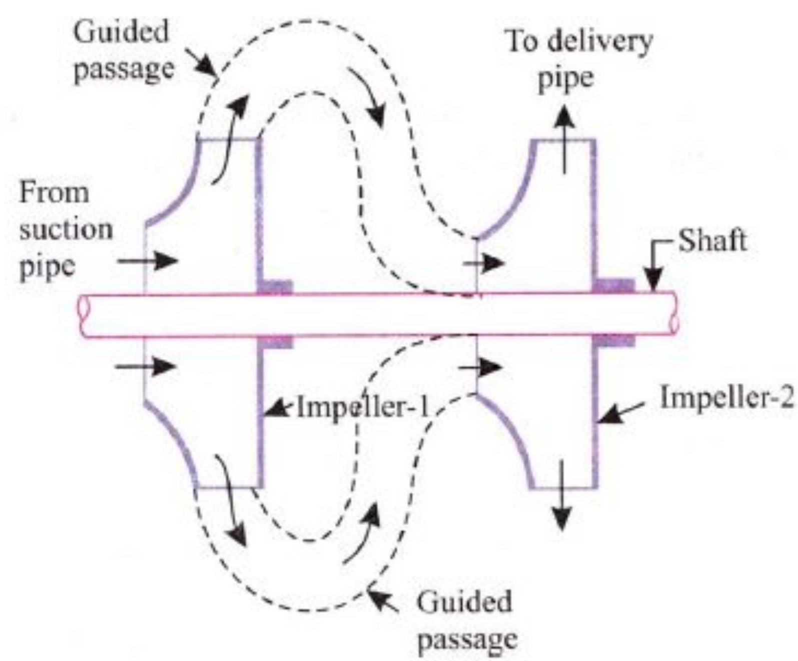
- To install a pumping station that can be effectively operated over a large range of fluctuations in both discharge and pressure head, it may be advantageous to install several identical pumps at the station.

Pumps in Parallel

Pumps in Series

4.10 Multi-stage Centrifugal Pump

- If a centrifugal pump consists of two or more impellers, the pump is called a multi-stage centrifugal pump.
- The impellers may be mounted on the same shaft or different shaft.
- A multi-stage pump is having the two important functions:
    I. To produce a high head and
    II. To discharge a large quantity of water.
- For **high head**, impellers are connected in series (on same shaft) as shown in Fig.4.5.



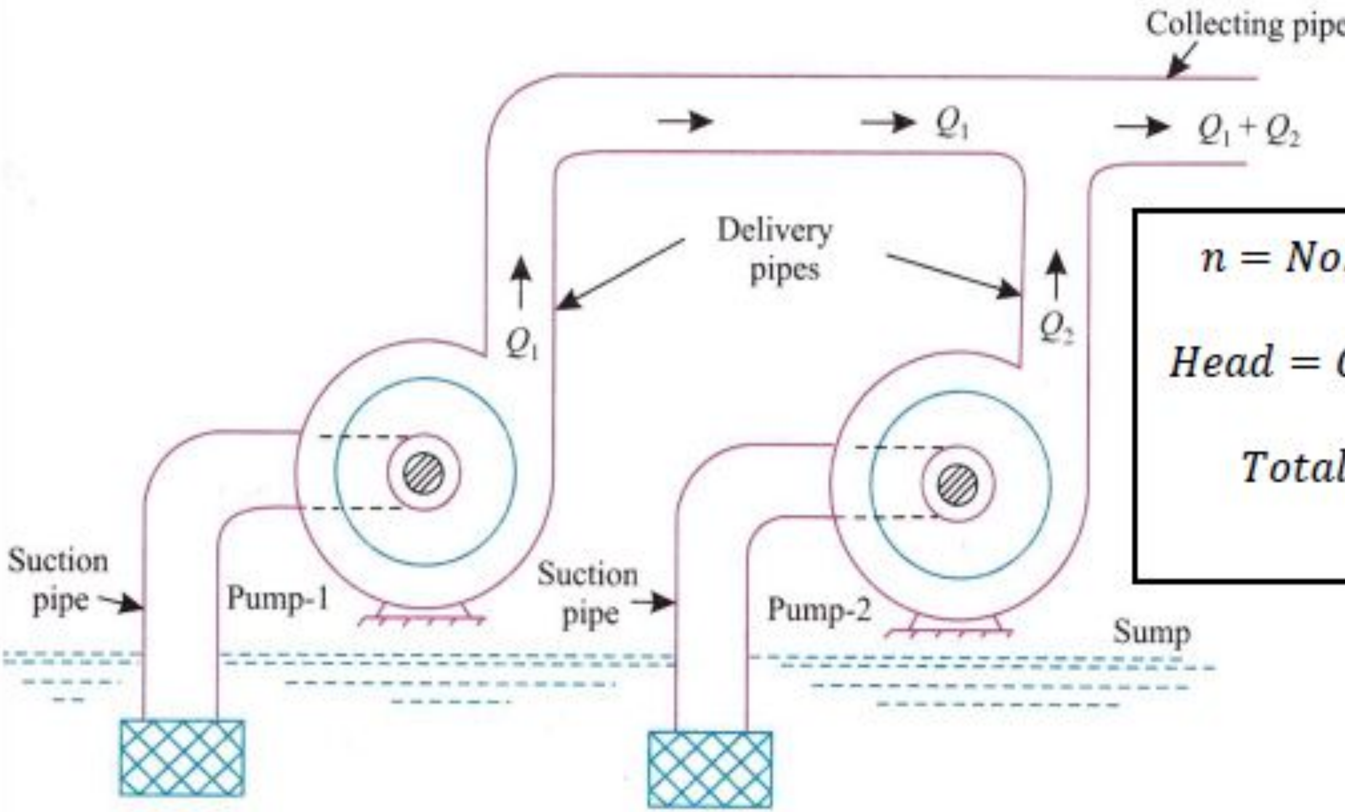
$n = Number\ of\ impellers$

$Q = Discharge = Constant$

$Total\ Head = n \times H_m$

Fig. 4.5 Pump in series

– For **high discharge**, the impellers are connected in parallel as shown in Fig. 4.6.



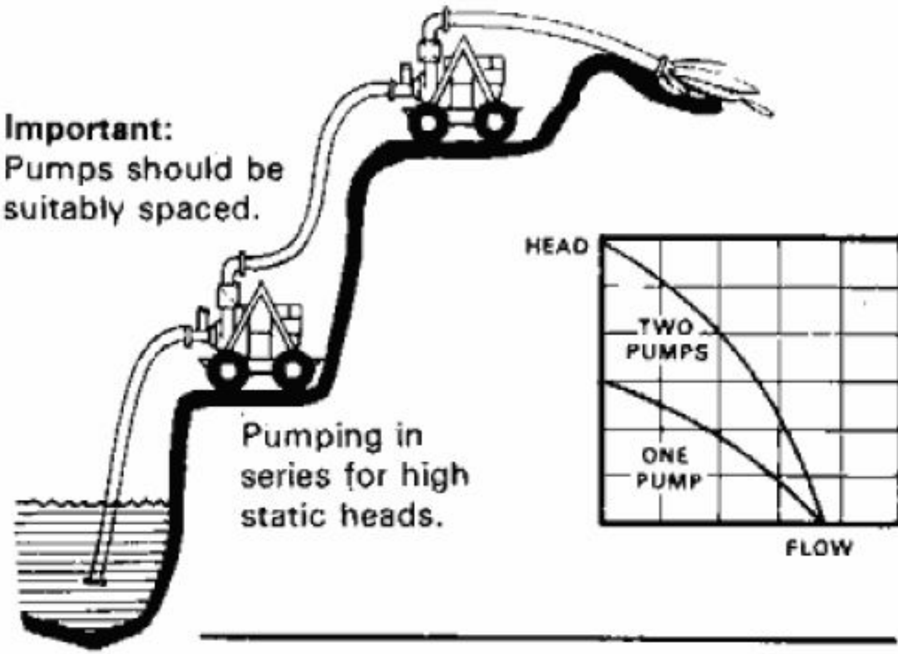
$$n = No.\,of\ pumps\ in\ parallel$$

$$Head = Constant\ for\ all\ pumps$$

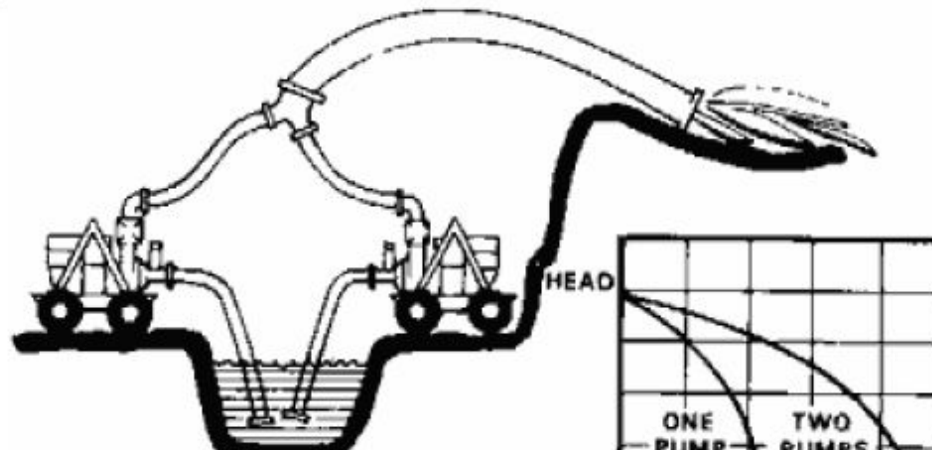
$$Total\ Discharge = n \times Q$$

Fig. 4.6 Pumps in parallel

PUMPING IN SERIES



PUMPING IN PARALLEL



The pipe must be big enough for the combined flow.

Priming of Centrifugal Pump

Before starting a centrifugal pump, the suction pipe, casing and portion of the delivery pipe up to delivery valve is completely filled with water by external source of water to remove the air from the suction pipe and casing. This is known as priming of a pump.

The work done by the impeller per unit weight of liquid per sec is known as the head developed by an impeller.

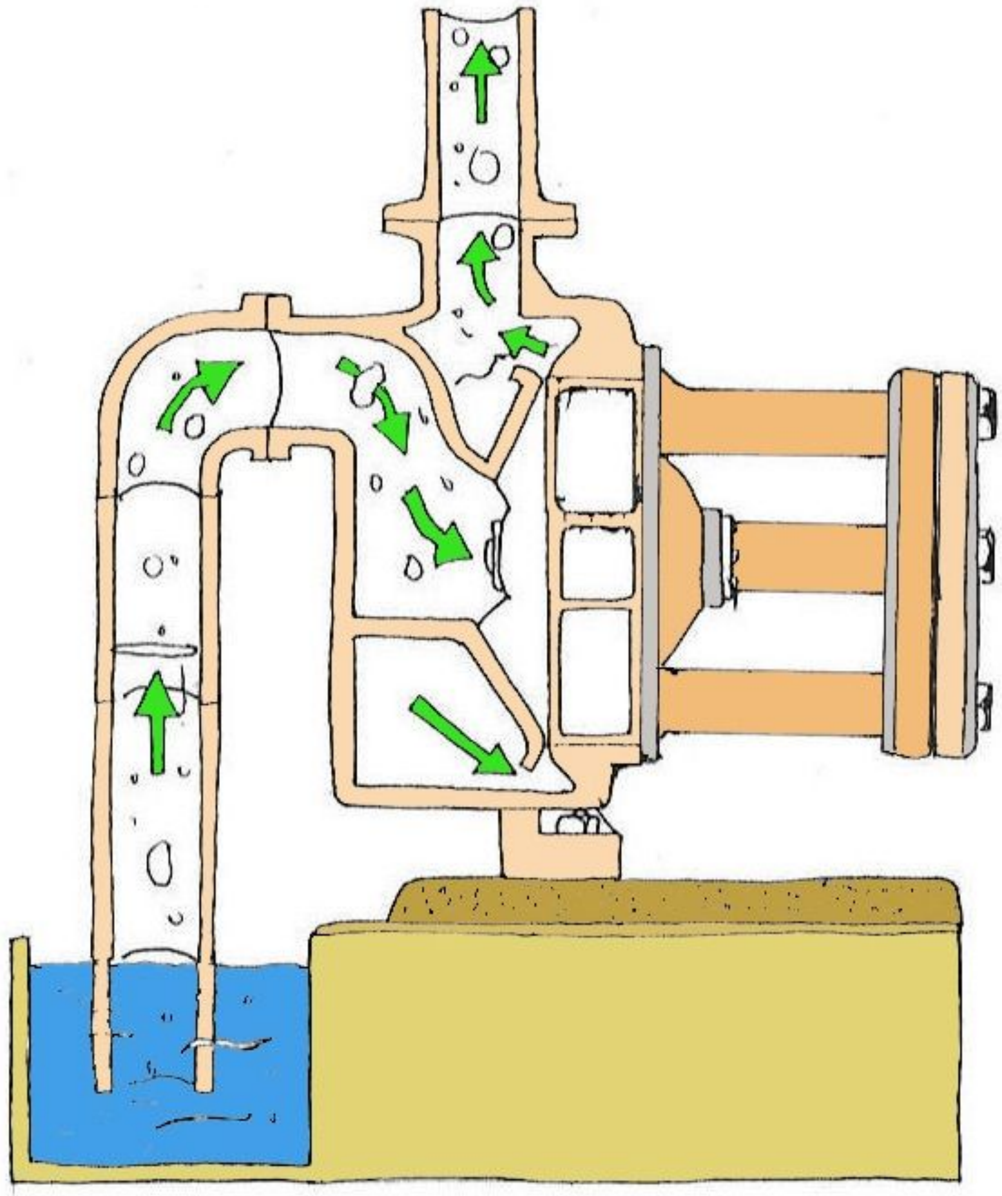
Head developed by the impeller is given by $\frac{u_2 V_{w2}}{g}$ meter. Since this equation is independent of the density of the liquid, the head developed will be in terms of meters of air when pump is running in the air.

If the pump is primed with water, the head generated is same meter of water. But as the density of air is very low, the generated head of air is negligible compared to meter of water head. Hence the water may not be sucked from the pump. To avoid this difficulty, priming is necessary.

PRIMING:

- The first step in the operation of a centrifugal pump is priming.
- Priming means removal of air from the pump casing and suction line.
- If an impeller is made to rotate in the presence of even a small air packet in any portion of the pump, only a negligible pressure would be produced.
- The result is that no liquid will be lifted by the pump.
- If vapours of the liquid being pumped are present on the suction side of the pump, this results in Cavitation , which can cause loss of prime or even serious damage to the pump.

PUMP PRIMING



Cavitations

- Cavitation is the formation of vapor bubbles in any flow that is subjected to an ambient pressure equal to or less than the vapor pressure of the liquid being pumped.
- Cavitation damage is the loss of material produced by the collapse of the vapor bubbles against the surfaces of the impeller or casing.
- Cavitation may be present in combination with erosion and corrosion – especially in wastewater
- Cavitation – Causes
- 1 – Insufficient NPSH available Occurs on the low-pressure, or visible, surface of the impeller vane
- 2 – Recirculation – partial reversal of flow through the impeller Occurs on the high-pressure, or invisible, surface of the impeller vane

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 collapse of these vapor bubbles in region of higher pressures.

- When the vapor bubbles collapse, a very high pressure is created. The metallic surfaces, above which these vapor 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.

- Cavitation includes formation of vapor bubbles of the flowing lquid and collapsing of the vapor bubbles. Formation of vapor bubbles of the flowing liquid takes place only whenever the pressure in any region falls below vapor pressure.
- When the pressure of the flowing liquid is less than its vapor pressure, the liquid starts boiling and vapor bubbles are formed.
- These vapor bubbles are carried along with the flowing liquid to higher pressure zones where these vapors condense and bubbles collapse.
- Due to sudden collapsing of the bubbles on the metallic surface, high pressure is produced and metallic surfaces are subjected to high local stresses. Thus the surfaces are damaged.

- When the pressure falls below the vapour pressure of the liquid at a given temperature, boiling occurs and small bubbles of vapour are formed. These bubbles will grow in the low-pressure area and implode when they are transported to an area of pressure above vapour pressure. The term given to this local vaporisation of the fluid is Cavitation.

- The collapsing of the bubbles is the area of Cavitation we are concerned with, as extremely high pressures are produced, which causes noise and erosion of the metal surface.

Effect of cavitations

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

Effects of Cavitation: The following are the effects of cavitation:

1) The metallic surfaces are damaged and cavities are formed on the surfaces.

2) Due to sudden collapse of vapor bubble, considerable noise and vibrations are produced.

3) 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 blade decreases. Hence the work done by water or output horse power becomes less and thus efficiency decreases.

Thoma's cavitation constant

The cavitation constant: is the ratio of *(NPSH)*$_R$ to the total dynamic head (H_t) is known as the Thoma's cavitation constant (σ)

$$\boxed{\sigma = \frac{(NPSH)_R}{H_t}}$$

Note: If the cavitation constant is given, we can find the maximum allowable elevation of the pump inlet (eye) above the surface of the supply (suction) reservoir.

Cavitation of Pump & Turbine

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 bubble collapse, a very high pressure is created. The metallic surfaces, above which these vapor 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.

Cavitation includes formation of vapor bubbles of the flowing liquid and collapsing of the vapor bubbles.

Precaution against Cavitation: The following precautions should be taken against Cavitation:

1) The pressure of the flowing liquid in any part of the hydraulic system should not be allowed to fall below its vapor pressure.

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

Cavitation in Centrifugal Pumps:

In centrifugal pumps the cavitation may occur at the inlet of the impeller of the pump, or at the suction side of the pumps, where the pressure is considerably reduced.

Hence if the pressure at the suction side of the pump drops below the vapor pressure of the liquid then the cavitation may occur.

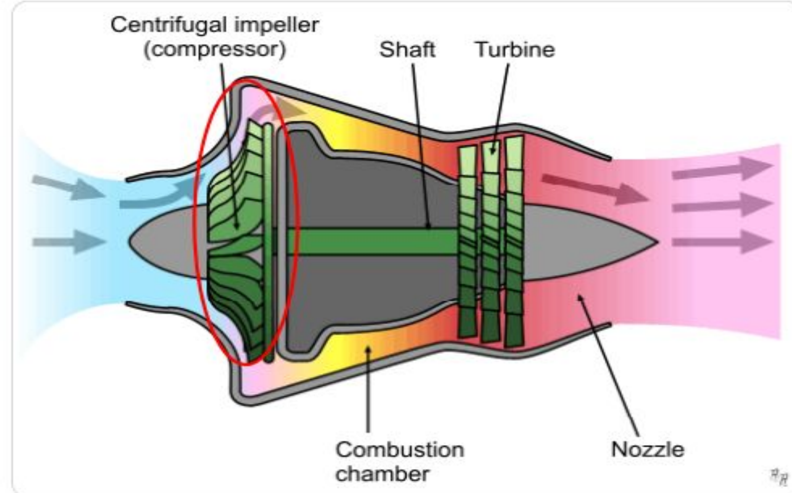
The cavitation in a pump can be noted bay a sudden drop in efficiency and head.

In order to determine whether cavitation will occur in any portion of the suction side of the pump, the critical value of Thoma's cavitation factor (σ) is calculated (Equation 4.23).

$$\sigma = \frac{H_b - H_s - h_{fs}}{H_{net}} = \frac{(H_{atm} - H_v) - H_s - h_{fs}}{H_{net}} \text{---------}(4.23)$$

If the value of Thoma's cavitation factor (σ) is greater than critical cavitation factor (σ_c), the cavitation will not occur in that turbine or pump. The critical cavitation factor (σ_c) may be obtained from tables or empirical relationships.

Centrifugal Compressor & Axial Compressor



Centrifugal Compressors:

Classification of Centrifugal Compressor, construction and working, velocity diagram, flow process on T-S Diagram, Euler's work, actual work input, various losses in Centrifugal Compressor

Axial flow compressors:

Construction and working, stage velocity triangle and it's analysis, enthalpy entropy diagram, stage losses and various efficiencies of axial flow compressors, [No numerical]

Compressors - What is a compressor?

Machine **to raise pressure** of a fluid

Uses several energy transformations

1. **Input energy** converted to **rotating** mechanical energy
2. **Rotating impeller** increases fluid's **kinetic** energy (velocity)
3. **Decrease in kinetic energy due to flow area expansion &increase in pressure energy**

Energy inputs: electricity, high pressure steam, fuel oil, compressed air, etc.

Introduction

- In the **seventeenth century** it was discovered that **air had weight** and was compressible. Its practical use in a compressed form started at that time.
- Early documented used of **compressed air** was for the reed type of **musical wind instrument**, which later became the pipe organ.
- **The first compressor constructed in the United States was in 1865**.

Turbo machines employing **centrifugal effects for increasing fluid** pressure have been in use for more than a century.

The earliest machines using this method were hydraulic pumps followed later by ventilating fans and blowers.

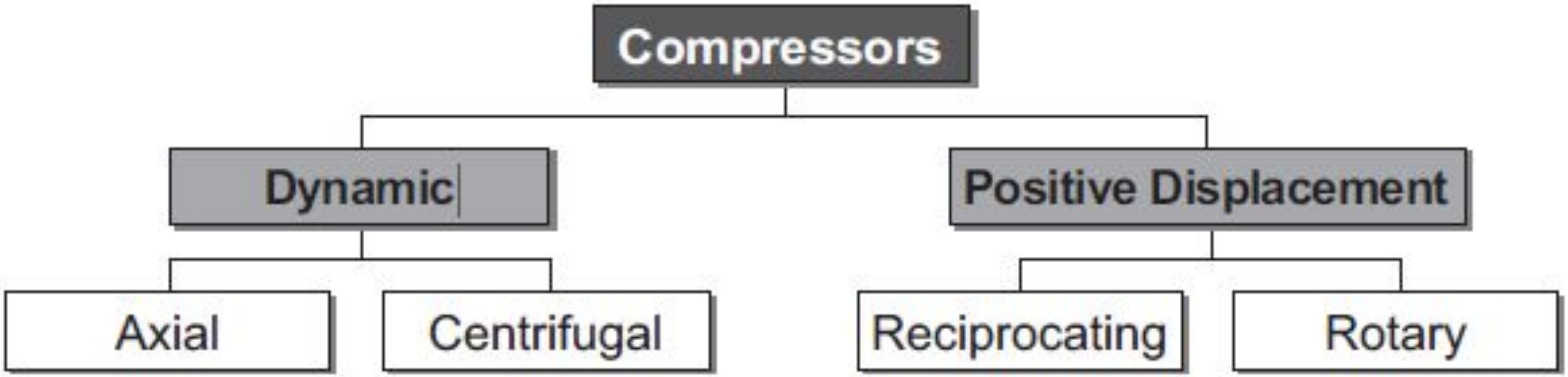
A centrifugal compressor was incorporated in the Whittle turbojet engine.

Axial flow compressors are more suitable for larger engines in terms of smaller frontal area (and drag) and 3-4% higher efficiency for the same duty than centrifugal compressors.

But for very small compressors with low flow rates, the efficiency of axial compressors drops sharply, blading is small and difficult to make accurately, and the centrifugal compressor is again preferable .

Many applications are found in small gas turbines for road vehicles and commercial helicopters as well as bigger applications, e.g., diesel engine turbochargers, chemical plant processes, factory workshop air supplies,large-scale air-conditioning plant, etc

Classification of Compressor



Dynamics compressors

Dynamics compressors use **rotating elements to accelerate the gas by diffusing action, velocity is converted to static pressure.**

Total energy in A flowing gas stream is constant. Entering an enlarged section, flow speed is reduced and some of the velocity energy turns into pressure energy. Thus, static pressure is higher in the enlarged section.

A compressor **using a rotating mechanism to add velocity and pressure to gas.**

Two types of dynamic compressors:

1. **Centrifugal**
2. **Axial**

Centrifugal compressor

- Centrifugal compressors are turbo-machines **employing centrifugal effects** to **increase the pressure of fluid.**
- In centrifugal compressors energy is transferred by **dynamic means from a rotating impeller** to the **continuously flowing fluid**.
- The **main f**eature of the centrifugal compressors is that the **angular momentum of the fluid** flowing through the impeller is **increased partly by virtue of the impeller outlet diameter being significantly larger than its inlet diameter.**
- **A pressure ratio** of the order of **4:1** can be obtained from a single stage compressor manufactured using conventional materials.

Characteristics Features of Centrifugal Compressors

- occupies a **smaller length** than an equivalent axial flow compressor and it has been superseded by the axial flow compressor in jet air crafts engines, where a short overall engine length is required.
- It has **better resistance to foreign object damage**.
- Because of the **relatively short passage length**, **loss of performance** due to build-up deposits on blade surfaces will not be **as great as** the axial flow compressors.
- It can **work reasonably well** in a contaminated atmosphere as compared to axial flow compressor.
- It has **ability to operate over a wide range** of mass flow rate at any particular rotational speed.

- Its **efficiency** under the most favorable circumstances, **are less than those of axial compressors** designed for the same duty, by as much as 3 or 4 %. However, at very low mass flows the axial flow compressor efficiency drops, blading is small and difficult to make accurately, and the advantage appear to lie with the centrifugal compressor in its relative simplicity and cost.

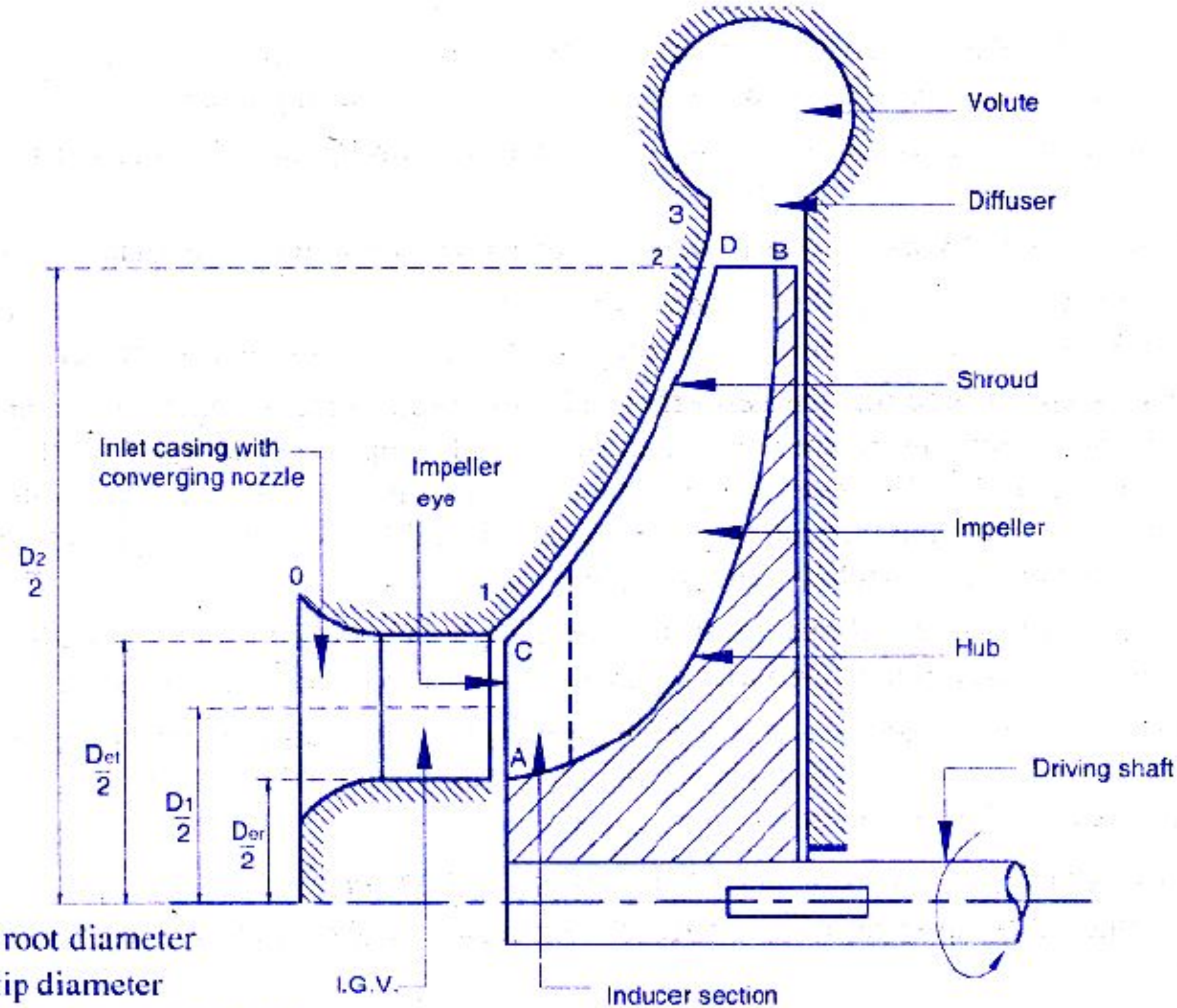
- The advent of titanium alloys, permitting much higher tip speeds, combined with advances in aerodynamics now permit **pressure ratios of greater than 8:1 to be achieved in a single stage.**

- It is widely used in gas pumping in long distance pipe line, petro-chemical industries, large scale refrigeration plants, big central air conditioning plants, fertilizer industry, supercharging of I. C. Engines etc.

Essential Parts of a Centrifugal Compressor

The principal components of a centrifugal compressor are shown in Fig. Detail of each part is given below.

- **Inlet casing with accelerating (converging) nozzle**
- **Inlet guide vanes (IGV)**
- **Impeller**
- **Diffuser**
- **Scroll or volute**
- **Inducer section**



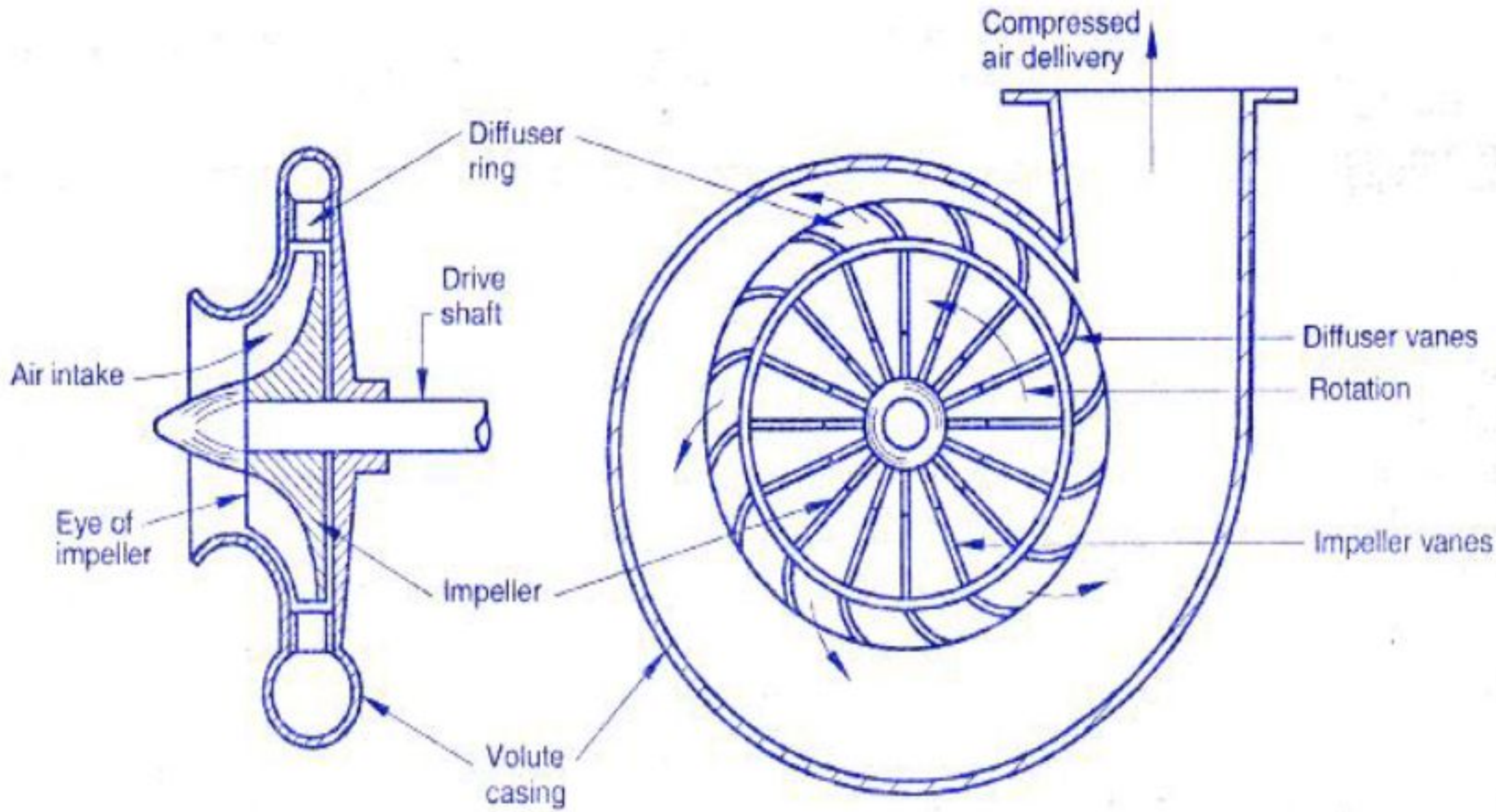
D_{er} = Eye root diameter

D_{et} = Eyetip diameter

D_1 = Impeller diameter at inlet

D_2 = Impeller diameter at outlet

8.2 Essential components of centrifugal compressor stage



Compressed
air dellivery
Diffuser
ring
Drive
shaft
Air intake
Eye of
impeller
Impeller
Volute
casing
Diffuser vanes
Rotation
Impeller vanes

- **Inlet casing with accelerating nozzle**

The function of inlet casing is to **accelerate the fluid from its initial condition to the entry of inlet guide vanes** and to provide **uniform velocity** at the eye.

The inlet flange is axisymmetric and the **inlet duct** takes the form of a **simple converging nozzle.** The **outlet** of the inlet casing is known as the **eye.**

- **Inlet guide vanes**

  The function of inlet guide vanes is to **direct the flow in the desired direction** at the entry of the impeller.

  The inlet guide vanes should be chosen so as to obtain a **minimum relative Mach number at the eye tip.**

Impeller

- The **function** of the impeller is to **increase the energy level of fluid by whirling it outwards by increasing the angular momentum** of the fluid.
- Both **static pressure and velocity of fluid are increased** in the impeller. The various impeller components are shown in Fig.
- The impeller vanes help to **transfer the energy from the impeller to the fluid.**

- The hub is the curved surface of revolution of the impeller A-B. The shroud is the curved surface C-D forming the outer boundary to the flow of fluid.
- Impellers may be enclosed by having the shroud attached to the vane ends (called shrouded impellers) or unenclosed with small clearance gap between the vane ends and the stationary wall.

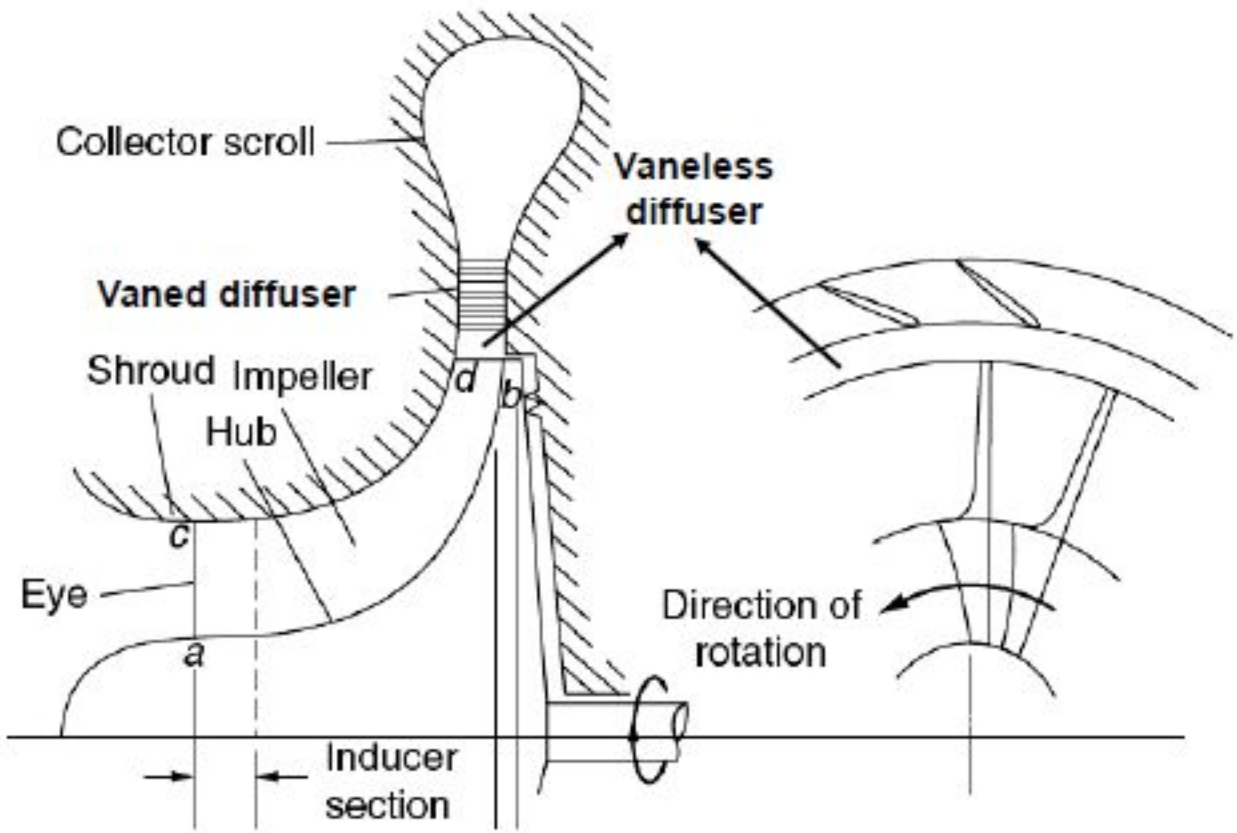
  Whether or not the impeller is enclosed the surface C-D is generally called the shroud. Shrouding an impeller has the merit of eliminating tip leakage losses but at the same time increases friction losses

Diffuser

- The function of the diffuser is to convert the **high kinetic energy of the fluid at impeller outlet into static pressure.**
- Impeller imparts energy to the air by increasing its velocity.
- The **diffuser converts this imparted kinetic energy into static pressure rise.** Diffuser is housed in a radial portion of the casing.
- It may be **vaneless or vaned diffuser**.
- The fluid flows radially outwards from the impeller, through a vaneless region and then through a vaned diffuser.

Diffuser

- In a centrifugal compressor, the flow leaving the impeller, passes through diffuser.
- The diffuser can be vaneless space, vaned or a combination of both.
- The function of the diffuser is to convert the exit kinetic energy into pressure.
- Diffuser being a static part the total conditions (pressure and temperature) do not change across it. But the static pressure and temperature increase with consequent decrease in absolute velocity.
- In high stage pressure ratio compressors, the diffuser leading edge region is critical because of high Mach numbers giving rise to shocks and shock losses.



Scroll or volute

The air leaving the **diffuser is collected in a spiral passage known as volute or scroll.** The volute discharges the air through delivery pipe.

Inducer section

Inducer is the impeller entrance section where the **tangential motion of the fluid is changed in the radial direction.**

This may occur with a little or no acceleration. Inducer ensures that the flow enters the impeller smoothly. Without inducers, the rotor operation would suffer from flow separation and high noise.

Inducer section

At entry to the impeller the relative flow has a velocity Cr_1, at angle α_1 to the axis of rotation.

This relative flow is turned into the axial direction by the inducer section or rotating guide vanes.

The inducer starts at the eye and usually finishes in the region where the flow is beginning to turn into the radial direction.

Some compressors of advanced design extend the inducer well into the radial flow region apparently to reduce the amount of relative diffusion.

Working

- When the impeller rotates at high speed, **suction is created at the impeller eye** and the **air is drawn** in through an accelerating nozzle.
- Due to flow acceleration in the compressor inlet part, the velocity of air is **increased from C_0 to C_1**, and thus pressure and temperature decrease.
- This acceleration is not isentropic but accompanied with friction. Thus P_1, and T_1, are the pressure and temperature at the inlet of the impeller.
- Due to energy supplied on the compressor shaft, the impeller is rotated at speeds of **20000 to 30000 rpm** and thus each particle of air passing through the impeller is accelerated i.e. **the kinetic energy of fluid is increased.**

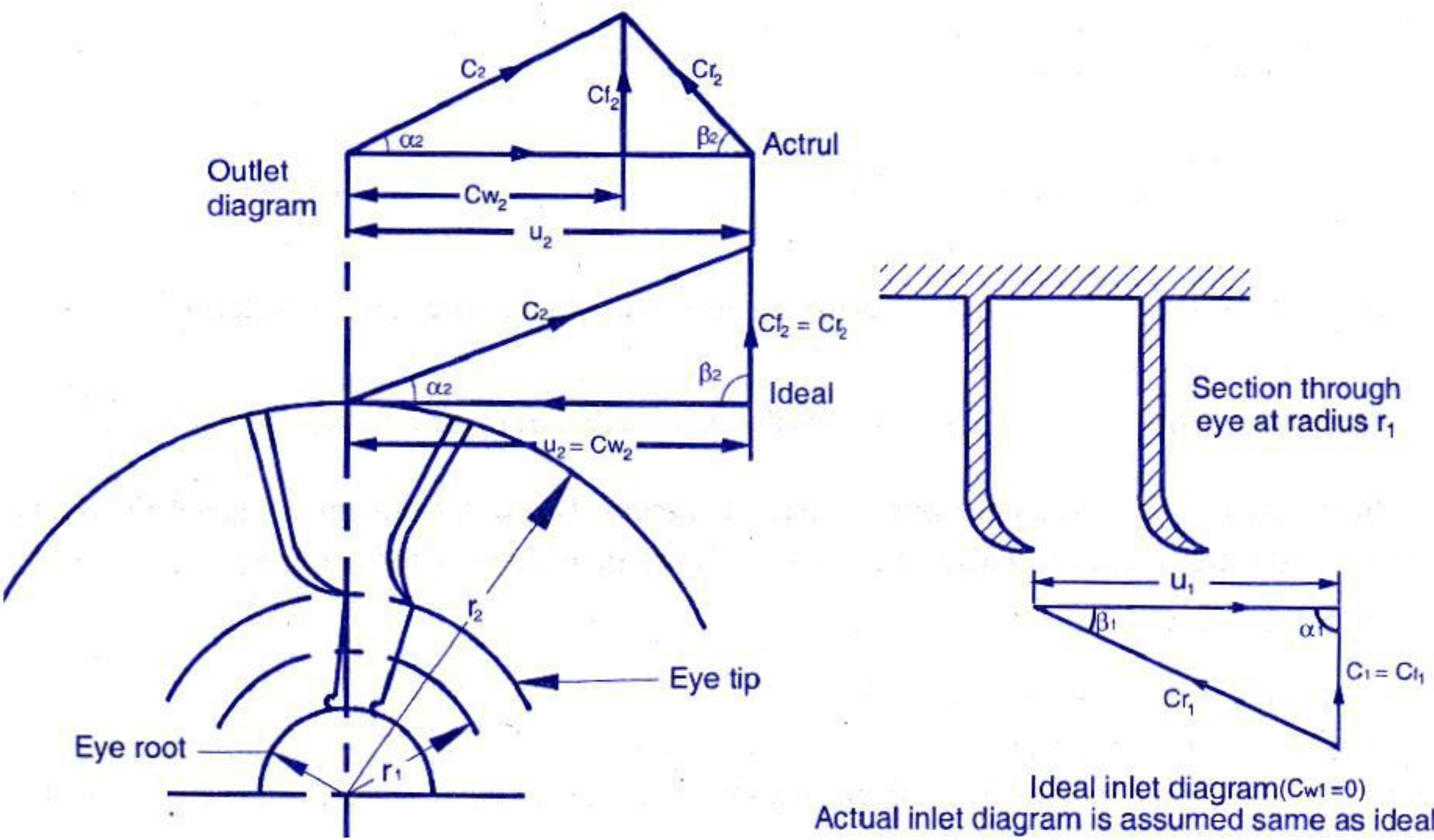
Working

- The impeller vanes are such that the **cross-sectional area between two vanes increases from inlet to outlet of the impeller.**
- This gives rise to **diffusion action** $(Cr_2^{\,2} - Cr_1^{\,2})/2$ addition to this air enters the impeller at smaller diameter and comes out at larger diameter.
- This gives rise to ***centrifugal action,*** $u_2^{\,2} - u_1^{\,2}/2$ Thus due to diffusion and centrifugal action, a part of the kinetic energy imparted to the air is converted into static pressure and temperature rise.

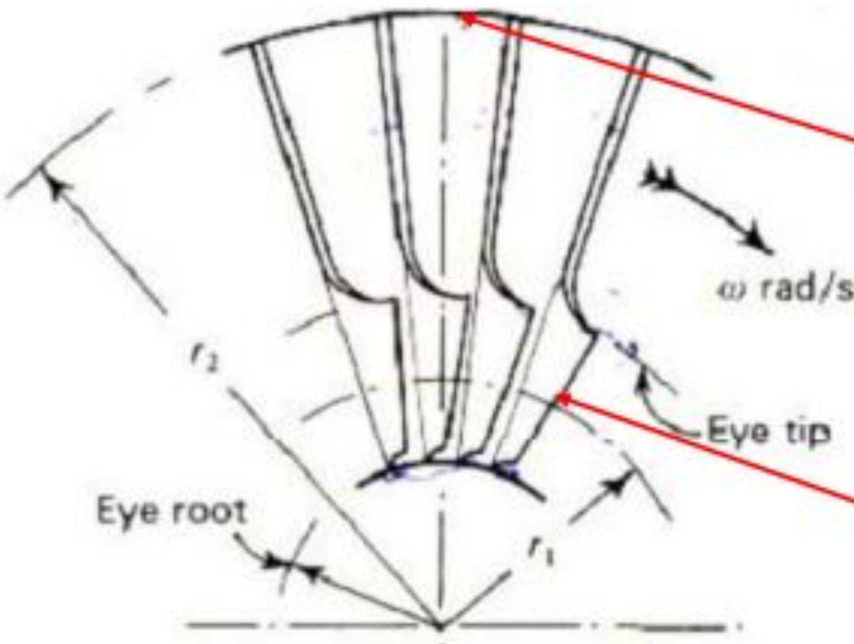
Working

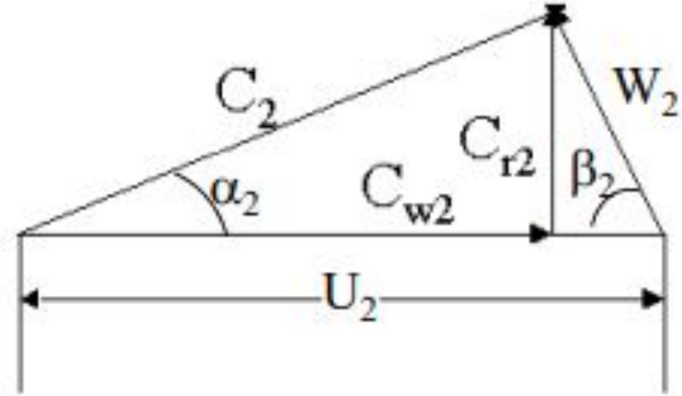
- The absolute velocity C_2 of air at the impeller outlet is very high and it has to be converted into pressure energy. This conversion is achieved in the vaneless and vaned diffusers.
- The vaneless diffuser converts some part of kinetic energy into pressure energy and velocity reduces from C_2 to C_3. The vaneless diffuser also stabilizes the flow coming out from the impeller so that the entry to the vaned diffuser is without shock.
- The rest of the kinetic energy is converted into pressure energy in the vaned diffuser and the velocity reduces from C_3 to C_4. The air leaving the vaned diffuser is collected in spiral passage (scroll or volute) from which it is discharged from the compressor.

Velocity triangles for a centrifugal compressor.

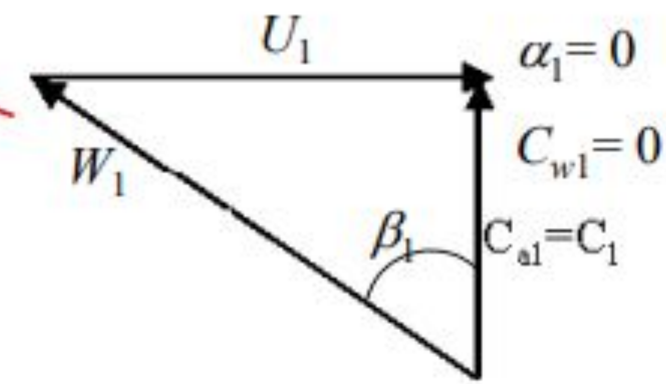


Velocity Triangles





Outlet velocity triangle



Inlet velocity triangle

Velocity triangles for a centrifugal compressor

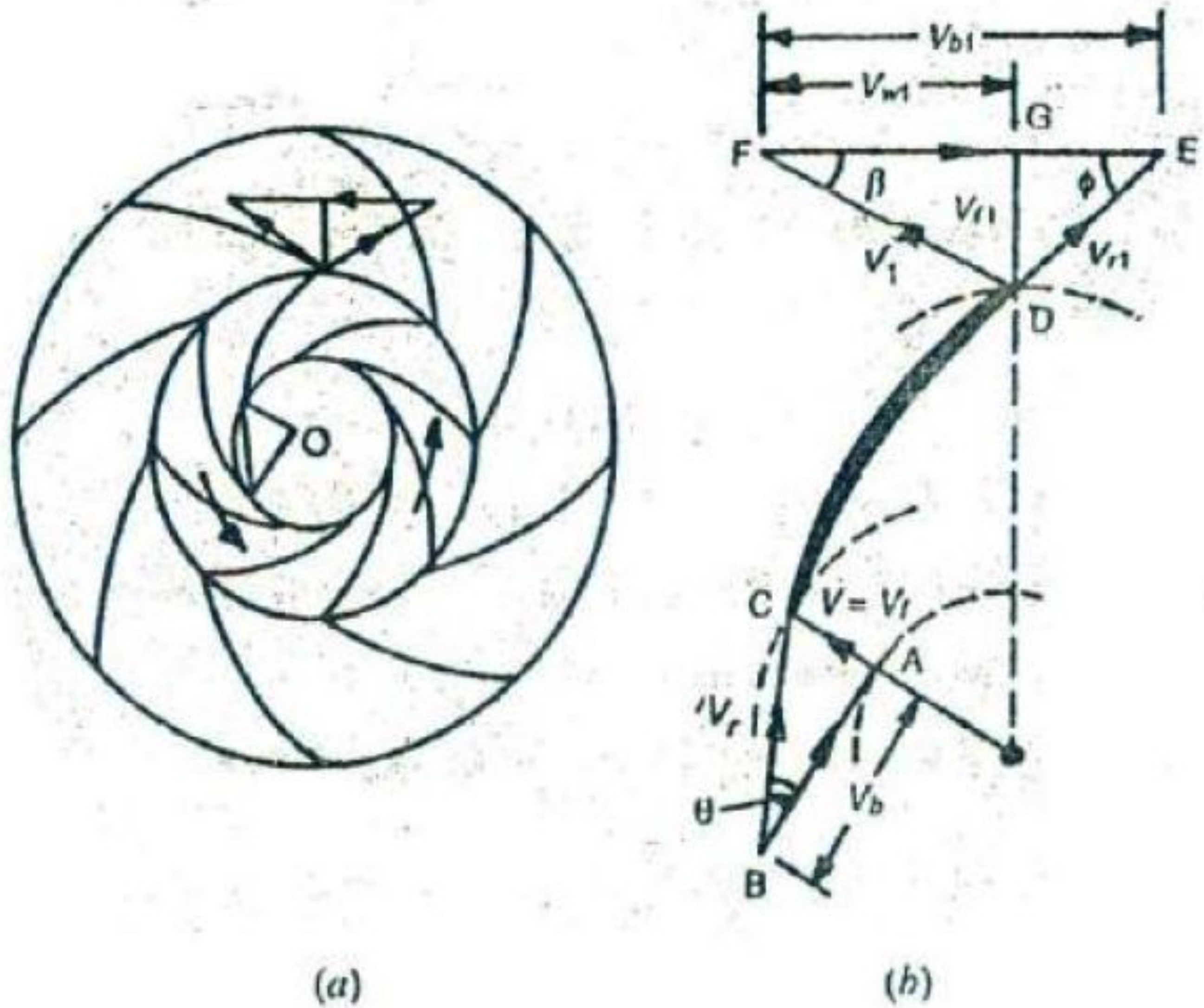


Fig. 29.5. Velocity triangles for a centrifugal compressor.

centrifugal compressor, the air enters **radially** and **leaves axially**. Moreover, the **blades and diffuser** are so designed that the **air enters and leaves them tangentially** for the shockless entry and exit.

Now let us draw the velocity triangles at the inlet and outlet tips of the blades as shown in Fig. *(a) and (b)*.

Let

V_b *or* C_{b1} *or* u_1 = *Linear velocity of the moving blade at inlet (BA),*

V_1 or C_1 = Absolute velocity of the air entering the blade (AC),

Vr_1 or C_{r1} = Relative velocity of air to the moving blade at inlet (BC). It is vectorial difference between V_b *and* V,

Vf 1 or C_{f1} = *Velocity of flow at inlet,*

θ = Angle which the relative velocity of steam (Vr or C_{r1}) makes with the direction of motion of the blade

α = Angle which the velocity of steam (V or C_1) makes with the direction of motion of the blade,

Similarly V_b *or* C_{b2} *or* u_2 , V_2 or C_2 Vr_2 or C_{r2} V_{f2} *or* C_{f2}

The following are the notations used in the analysis of a centrifugal compressor.

$\boldsymbol{\alpha}$ = Exit angle from the guide vanes at entrance = absolute angle at inlet

$\boldsymbol{\theta}$ = Inlet angle to the rotor or impeller

$\boldsymbol{\beta}$ = Outlet angle from the rotor or impeller

$\boldsymbol{\Phi}$ = Inlet angle to the diffuser or the stator

$\boldsymbol{Cb_1}$ = Mean blade velocity at inlet

$\boldsymbol{Cb_2}$ = Mean blade velocity at exit

$\boldsymbol{C_1}$ = Absolute velocity of air at inlet to the rotor

$\boldsymbol{C_2}$ = Absolute velocity of air at exit to the rotor

$\boldsymbol{C_{r1}}$ = Relative velocity of air at inlet to the rotor blade

$\boldsymbol{C_{r2}}$ = Relative velocity of air at exit to the rotor blade

C_{w1} = Velocity of whirl at inlet (tangential component of absolute velocity C_1)

$\boldsymbol{C_{w2}}$ = Velocity of whirl at exit (tangential component of absolute velocity C_2)

$\boldsymbol{C_{f1}}$ = Velocity of flow at inlet (Component of C_1 perpendicular to the plane of rotation)

$\boldsymbol{C_{f2}}$ = Velocity of flow at exit (Component of C_2 perpendicular to the plane of rotation)

m = Mass flow rate, kg/sec

Let $\qquad m$ = Mass of air compressed by the compressor in kg/s.

We know that according to Newton's second law of motion, force in the direction of motion of blades (in newtons),

$$F = \text{Mass of air flowing in kg/s} \times \text{Change in the velocity of whirl in m/s}$$

$$= m\,(V_w + V_{w1}) = m\,V_{w1} \qquad \ldots (\because V_w = 0)$$

and work done in the direction of motion of the blades,

$$W = \text{Force} \times \text{Distance} = m\,V_{w1}\,V_{b1}\ \text{N-m/s or J/s}$$

Now power required to drive the compressor may be found out, as usual, by the relation,

$$P = \text{Work done in J/s} = m\,V_{w1}\,V_{b1}\ \text{watts} \qquad \ldots (1\ \text{J/s} = 1\ \text{watt})$$

Notes : 1. The blade velocity at inlet or outlet (V_b or V_{b1}) may be found out by the relation,

$$V_b = \frac{\pi DN}{60} \quad \text{and} \quad V_{b1} = \frac{\pi D_1 N}{60}$$

where D and D_1 are the internal and external diameters of the impeller.

2. Under ideal conditions (or in other words for maximum work) $V_{w1} = V_{b1}$.

Therefore ideal work done

$$= m\,(V_{w1})^2 = m\,(V_{b1})^2\ \text{J/s}$$

Work Done Equations

(1) Euler's Work done Equation for Impeller

According to the Newton's second law of motion, force applied is equal to the rate of change of momentum

Tangential force = mass of air x Rate of change of tang. velocity

$$= m\ (C_{w2} - C_{w1})/t\ = m'\ (C_{w2} - C_{w1})$$

Torque T = Rate of change of angular momentum

$$= \boldsymbol{m'}\ (C_{w2} r_2 - C_{w1} r_1)$$

Where r_1 and r_2 are impeller radius at inlet and outlet Theoretical work done or energy transfer,

$W = E = m'\ (C_{w2}\, r_2\ - C_{w1} r_1)\omega$

But $w\, r_1 = u_1$ and $wr_2 = u_2$ then,

$$\boldsymbol{W = E = m'\ (C_{w2}\, u_2 - C_{w1} u_1)}$$

In a centrifugal compressor it is assumed that air enters the impeller eye in an axial direction

$a_1 = 90^0$, and $C_{w1} = 0$, $C_{1} = 0$, $C_{f1} = C_1$

Then,

$$\boldsymbol{W = E = m'\ C_{w2}\, u_2}$$

Work done per kg = $\boldsymbol{Cw2\ u2}$

Ideal Energy Transfer or Maximum Work Done Equation for Impeller or Ideal power

For an ideal case it is assumed that:

The impeller is radial (ideal case) vaned ($\beta_2 = 90^0$) i.e. the air leaves the impeller with a tangential velocity $\boldsymbol{C_{w\,2} = u_2}$ and $\boldsymbol{C_{f\,2} = C_{r\,2}}$

Thus, work done or energy transfer is,

$$W = E = m\, C_{w\,2}\, u_2$$

$$W = m u_2\, u_2$$

$$W = m\, u_2^{\,2}$$

The ideal work is called Euler work

Since the air cannot leave the impeller at a velocity greater than the impeller tip velocity, the above equation gives the maximum work capacity of the impeller.

Width of Impeller Blades

The width of impeller blades at inlet or outlet of a rotary air compressor is found out from the fact that mass of air flowing through the blades at inlet and outlet is constant.

Now consider a rotary air compressor compressing the air, whose blade widths at inlet or outlet is required to be found Out

Let b = Width of the impeller blades at inlet.

D = Diameter of impeller at inlet,

V_f = Velocity of flow at inlet,

v_s = Specific volume of air at inlet,

b_1, D_1, V_{f1}, v_{s1} = Corresponding values at outlet, and

m = Mass of the air flowing through the impeller.

We know that the mass of air flowing through the impeller at inlet,

$$m = \frac{\pi D b V_f}{v_s}$$

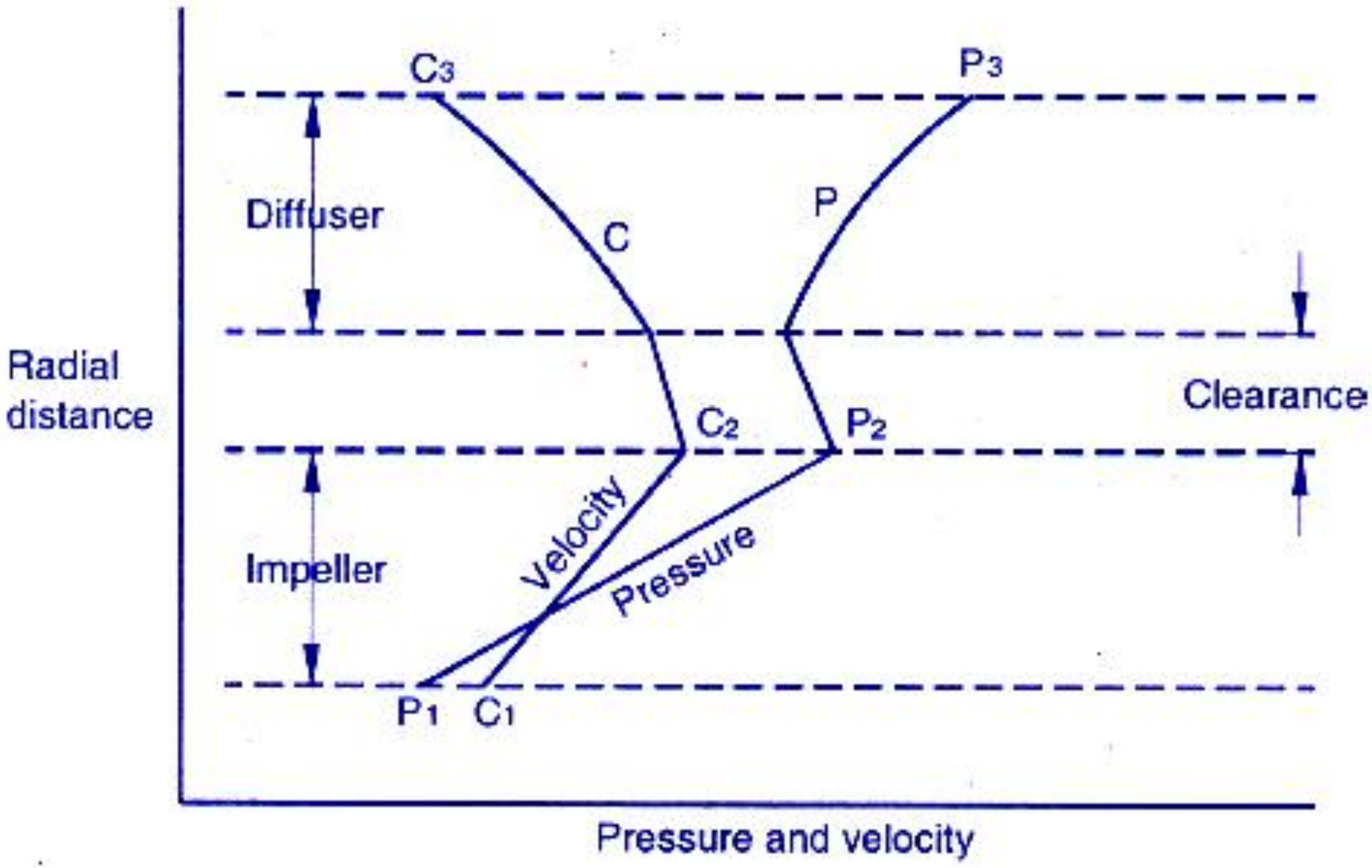
Similarly, mass of air flowing through the impeller at outlet,

$$m = \frac{\pi D_1 b_1 V_{f1}}{v_{s1}}$$

Since the mass of air flowing through the impeller is constant, therefore,

$$\frac{\pi D b V_{f1}}{v_s} = \frac{\pi D_1 b_1 V_{f1}}{v_{s1}}$$

Pressure and velocity variation across centrifugal compressor



- Above figures shows the pressure and velocity variation across a centrifugal compressor.
- Air enters the compressor at mean radius with a low velocity C_1, and atmospheric pressure P_1. It is then accelerated to a high velocity C_2, and pressure P_2, depending upon the centrifugal action of the impeller. The air now enters the diffuser where its velocity is reduced to some value C_3, and pressure increases to P_3.
- In practice, about half of the total pressure rise per stage is achieved in the impeller and the remaining half in the diffuser.

Static and Total Head Properties

- In rotary compressors high fluid velocities are encountered and therefore total head quantities which take into account the kinetic energy have to be considered.
- **If the moving air is brought to rest isoentropically without external work transfer then resulting state is known as total head or stagnation state and corresponding values of the properties describing this state are called stagnation properties.**
- Consider a horizontal passage of varying area with no external heat transfer and work transfer.

Static and Stagnation Temperature (T_0)

"It is the actual temperature of the air that would be registered by a thermometer moving with air with the same speed of the air is called *static temperature (T)*."

"If the moving air is brought to rest isentropically without external work transfer then the kinetic energy of the air is converted in to heat energy increasing the temperature of the air, the resulting temperature of the air is called *stagnation temperature* (T_0)."

Apply steady flow energy equation for 1 kg of air flow diverging passage (Fig) with no heat and work transfer is given by,

$$u_1 + P_1V_1 + \frac{C_1^2}{2} = u_2 + P_2V_2 + \frac{C_2^2}{2}$$

But $u + PV = h$

$$h_1 + \frac{C_1^2}{2} = h_2 + \frac{C_2^2}{2}$$

For specific heat $h = C_pT$

$$C_pT_1 + \frac{C_1^2}{2} = C_pT_2 + \frac{C_2^2}{2}$$

$$C_pT + \frac{C^2}{2} = C_pT_0 = h_0 = \text{Constant}$$

Where T_0 is known as stagnation temperature

$$T + \frac{C^2}{2C_p} = T_0$$

$$T - T_0 = \frac{C^2}{2C_p}$$

$$\boxed{T_0 = T + \frac{C^2}{2C_p}}$$

Where T is static temperature and $\frac{C^2}{2C_p}$ is called dynamic temperature.

- **Stagnation Pressure (P0)**
- If the moving air is brought to rest ientropically without external work transfer then the kinetic energy of the air is converted in to pressure of the air, the resulting pressure of the air is called *stagnation pressure (P0*
- Stagnation pressure can be found by using following relation between pressure and temperature is, given by,

$$\boxed{\frac{P_0}{P} = \left(\frac{T_0}{T}\right)^{\frac{\gamma}{\gamma-1}}}$$

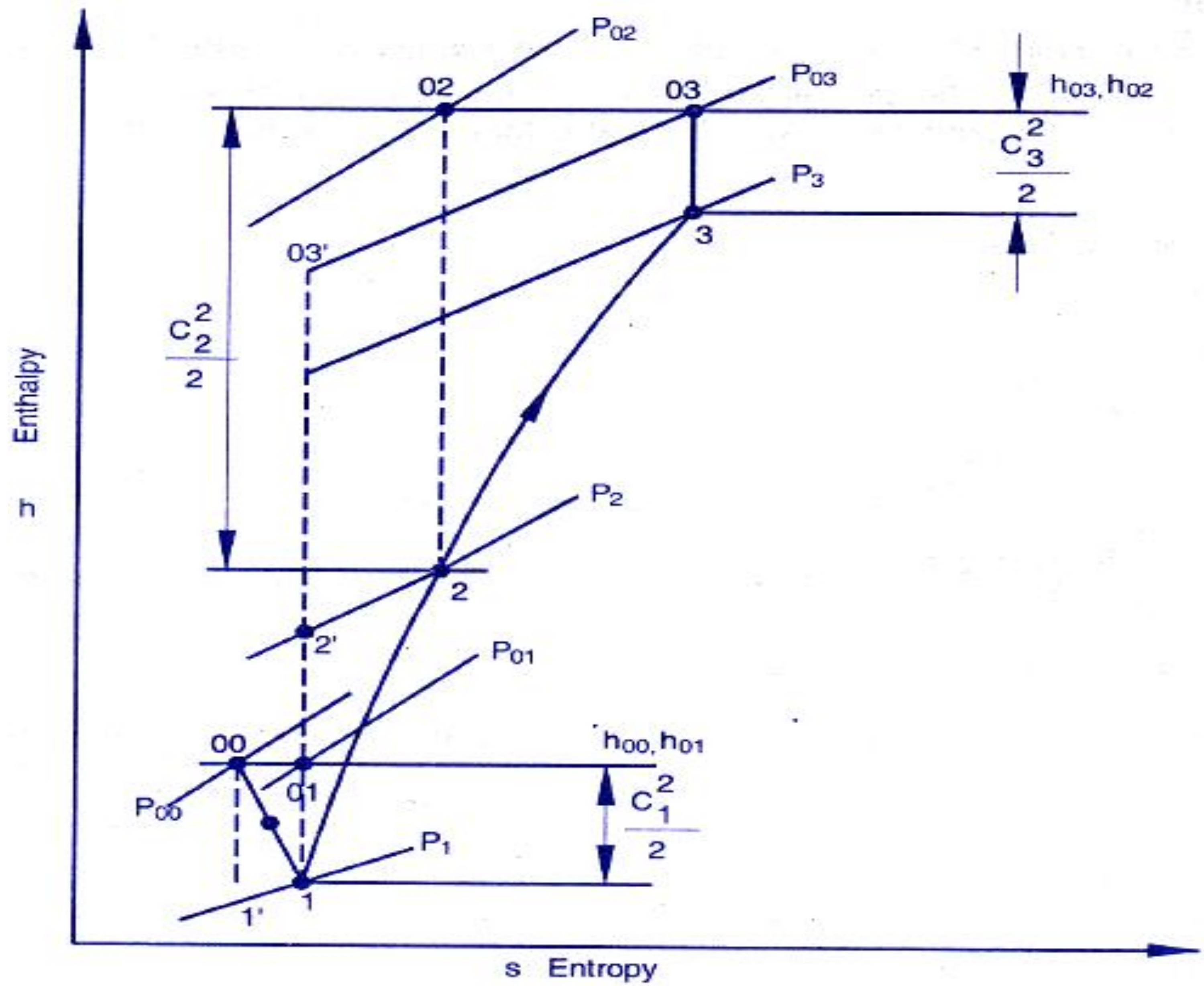
Stagnation Enthalpy (h_0)

"If the moving air is brought to rest isentropically without external work transfer then resulting enthalpy is known as *stagnation enthalpy*." The stagnation enthalpy remains constant in a moving stream in the absence of heat and work transfer.

Stagnation enthalpy can be found by using following relation is given by,

$$\boxed{h_0 = h + \frac{C^2}{2}}$$

H-S Diagram for centrifugal compressor



Centrifugal Compressors - Blade Types

There are three impeller vane types defined according to the exit blade angles (Discharge Vane Angles)

Impellers with exit blade angle equal to 90 degrees are **radial vanes**

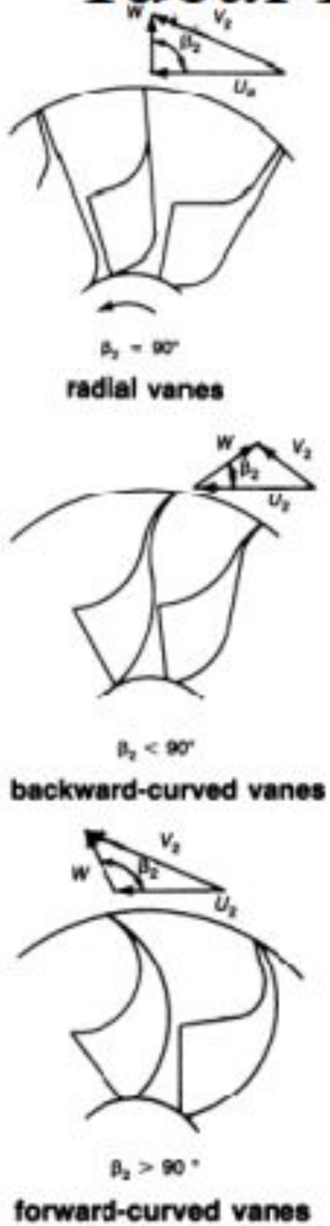
Impellers with exit blade angle less than 90 degrees are **backward-curved or backward swept**

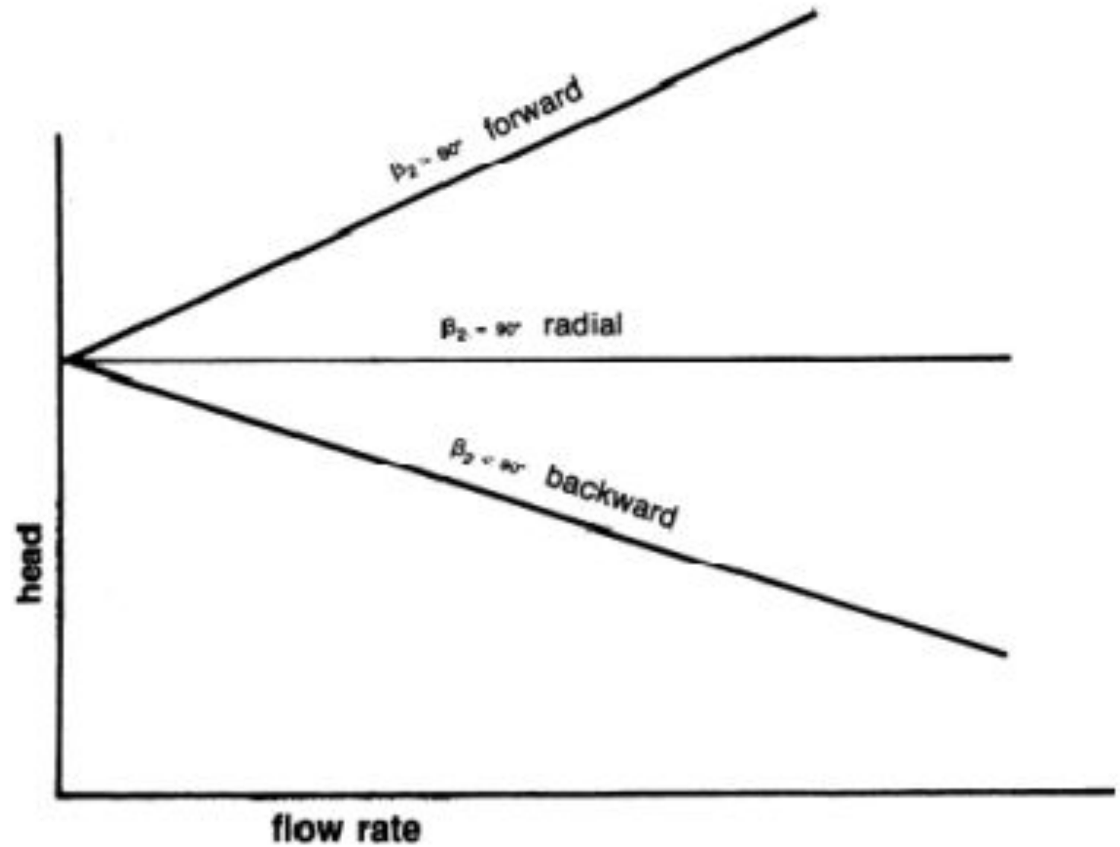
Vanes with exit blade angle greater than 90 degrees are known as **forward swept vanes**

The forward-curved blade has the highest theoretical head.

Radial vanes represent a compromise between max pressure ratio, max efficiency & size

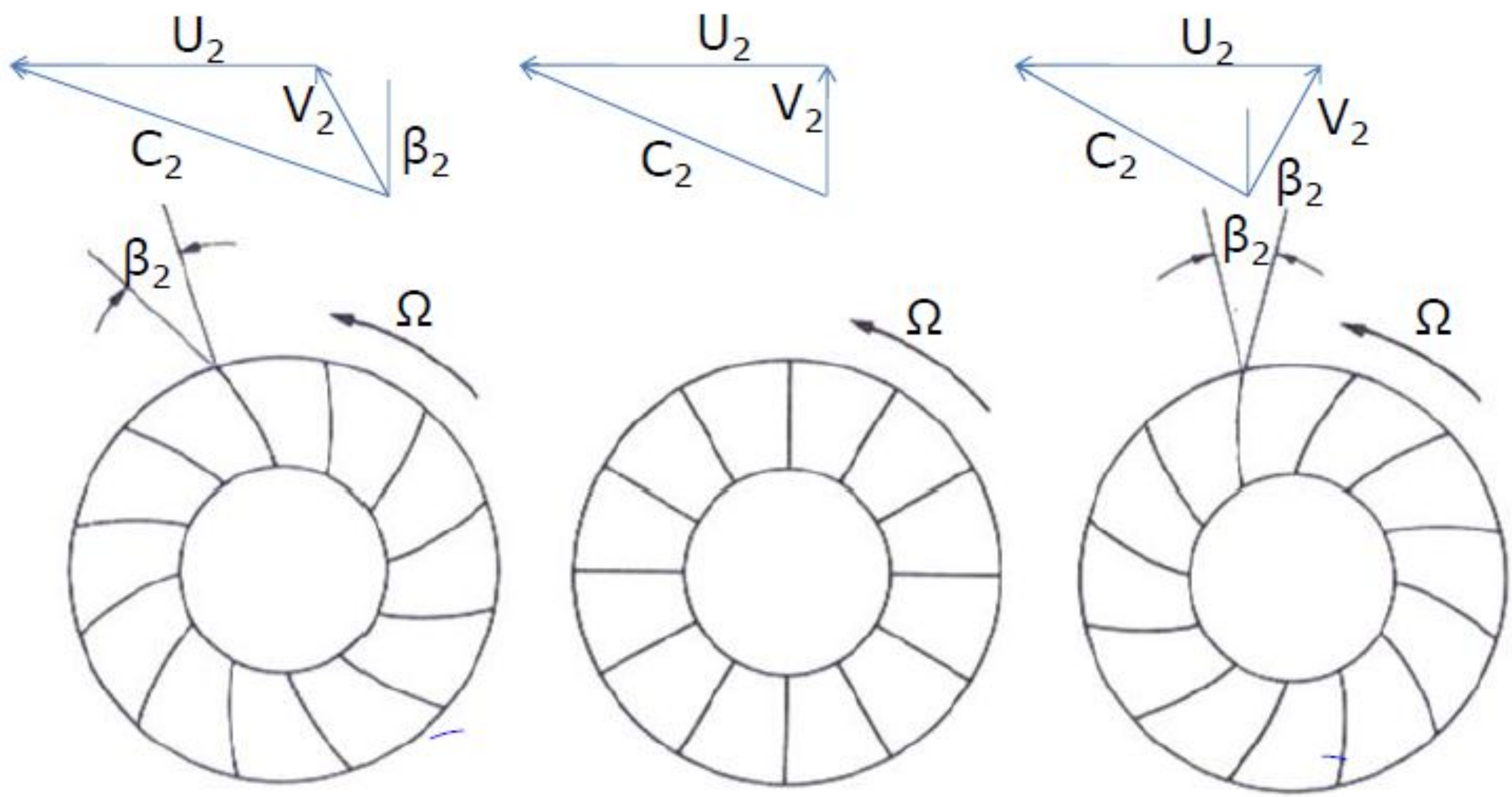
Ideal Performance of Impellers





Head – flow characteristics for various outlet blade angles

Impeller

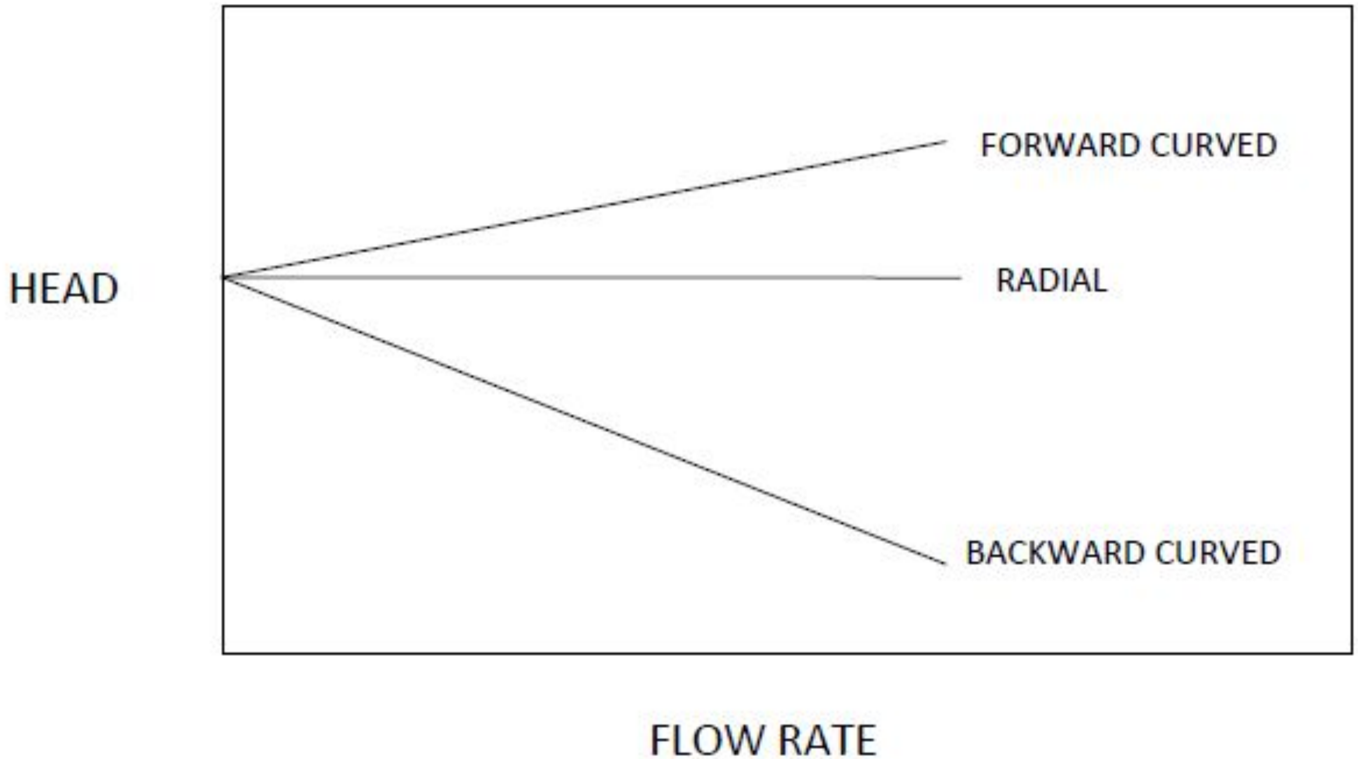


Forward leaning blades (β_2 is negative)

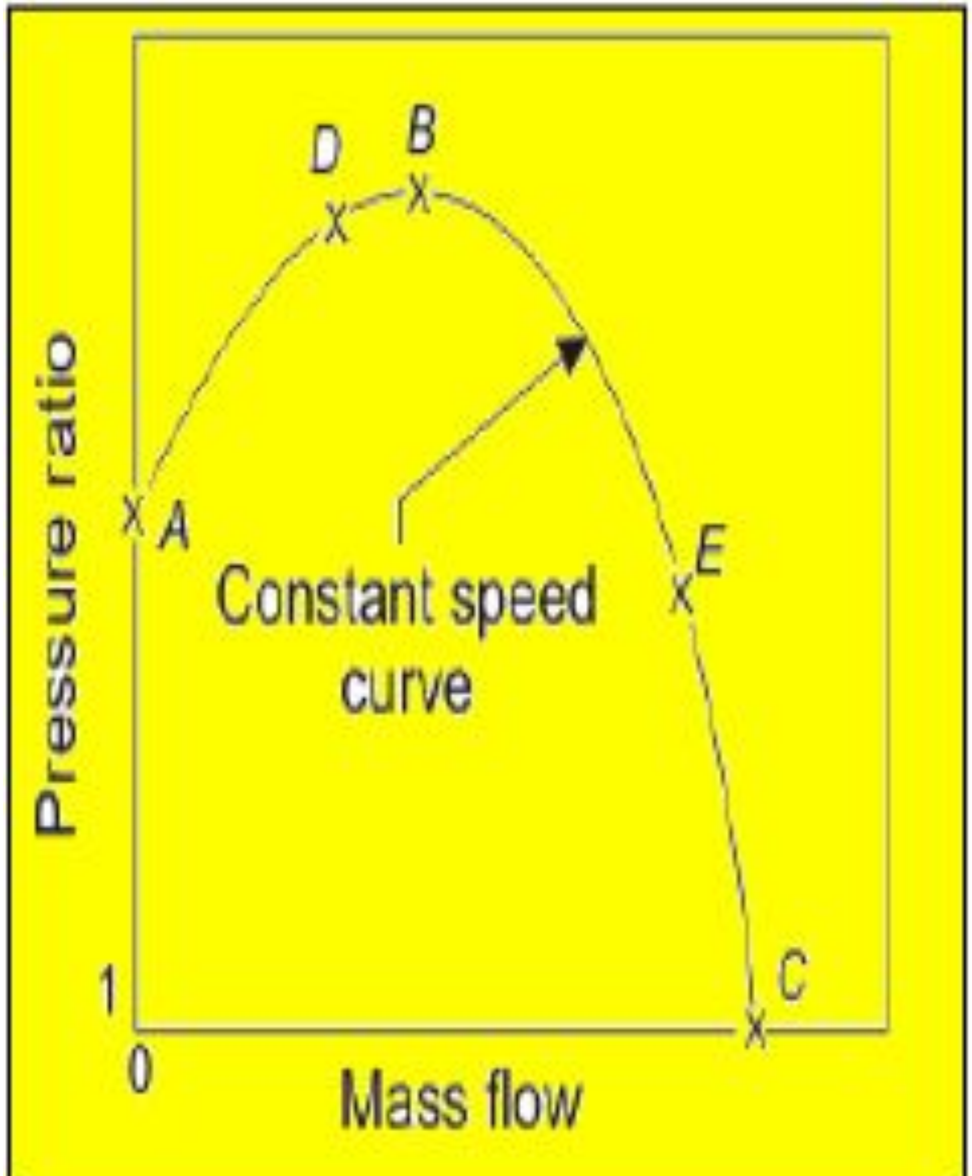
Straight radial

Backward leaning blades (β_2 is positive)

Centrifugal Compressors - Blade Types



Centrifugal Compressor Performance Curve



Lets analyse what will occur when a valve placed in the delivery line of a compressor running at constant speed is slowly opened .The variation in pressure ratio is shown above

Point **A occurs when the valve is shut & mass flow is** zero. It corresponds to centrifugal pressure head produced by action of impeller on the air trapped

between the vanes.

At point **B, efficiency and pressure ratio approach** maximum value. Further increase in mass flow will result in fall of pressure ratio. For mass flows greatly in excess of design mass flow, air angles will be widely different from vane angles leading to breakaway of air & fall in efficiency.

The pressure ratio drops to unity at **'C' , when the** valve is fully open and all the power is absorbed in overcoming internal frictional resistances

Centrifugal Compressor Performance Curve - Surging

between 'A' and 'B' could not be obtained due to Surging.

• Surging is associated with sudden drop in delivery pressure & with violent aerodynamic pulsation which is transmitted throughout the machine

• For any operating point **D on the part of characteristics curve** having a positive slope, a decrease in mass flow will be accompanied by a fall in delivery pressure.

• If the pressure of the air downstream of the compressor does not fall quickly enough, the air will tend to reverse its direction and will flow back in the direction of the resulting pressure gradient.

• When this occurs, the pressure ratio drops rapidly causing a further drop in mass flow until the point 'A' is reached, where the mass flow is zero.

Surging starts to occur in the diffuser passages where flow is retarded by frictional forces near the vanes

• Tendency to surge increases with number of diffuser vanes

• Several diffuser channels to every impeller channel – tendency for air to flow up one channel & down another (conditions conducive to surging)

• Only in one pair of channels the delivery pressure will fall& increase likelihood of surging Thus number of diffuser vanes is less than no. of impeller vanes Surging is then not likely to occur

Performance Curve- Rotating Stall

It is another important cause of instability & poor performance which can exist in the nominally stable operating range.

- A,B & C are three consecutive flow channels
- When there is non-uniformity in flow or geometry of channels between vanes or blades, breakdown can occur in one channel (say channel B)
- Air deflects in such a way that C receives fluid at reduced incidence & A at increased incidence
- Channel A stalls which reduces incidence in B enabling flow in that channel to recover
- Rotating stall may lead to aerodynamically induced vibrations leading to fatigue failures in other parts

Centrifugal Compressors

Stall

Defined as the (aerodynamic stall) or the break-away of the flow from the suction side of the blades.

A multi-staged compressor may operate safely with one or more stages stalled and the rest of the stages unstalled . but performance is not optimum. Due to higher losses when the stall is formed.

Surge

Is a special fluctuation of mass flow rate in and out of the engine. No running under this condition.

Surge is associated with a sudden drop in delivery pressure and with violent aerodynamic pulsation which is transmitted throughout the whole machine.

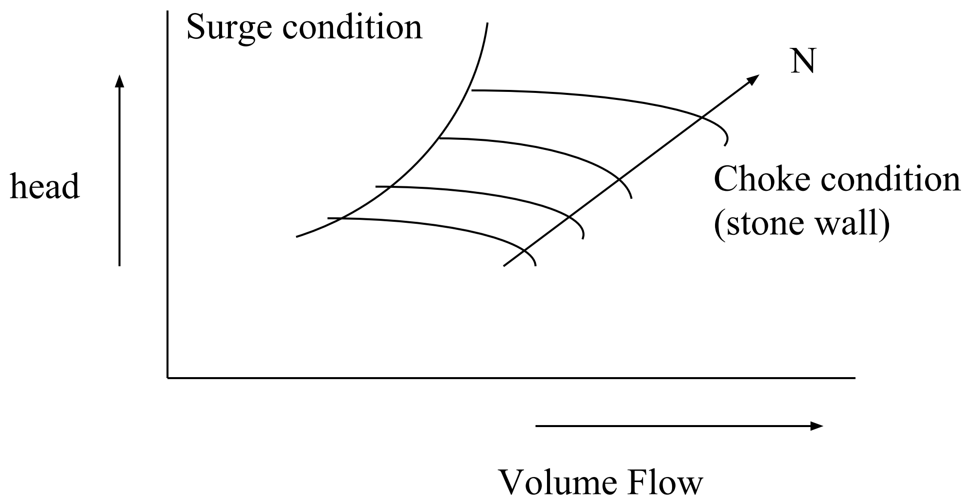
Centrifugal Compressor Performance Curve

There is an additional limitation to the operating range, between 'B' and 'C'. As the mass flow increases and the pressure decreases, the density is reduced and the radial component of velocity must increase.

- At constant rotational speed this means an increase in resultant velocity and hence an angle of incidence at the diffuser vane leading edge.
- At some point say **'E', the position is reached where no** further increase in mass flow can be obtained no matter how wide open the control valve is – **CHOKING**
- This point represents the maximum delivery obtainable at the particular rotational speed for which the curve is drawn.

Surge Point

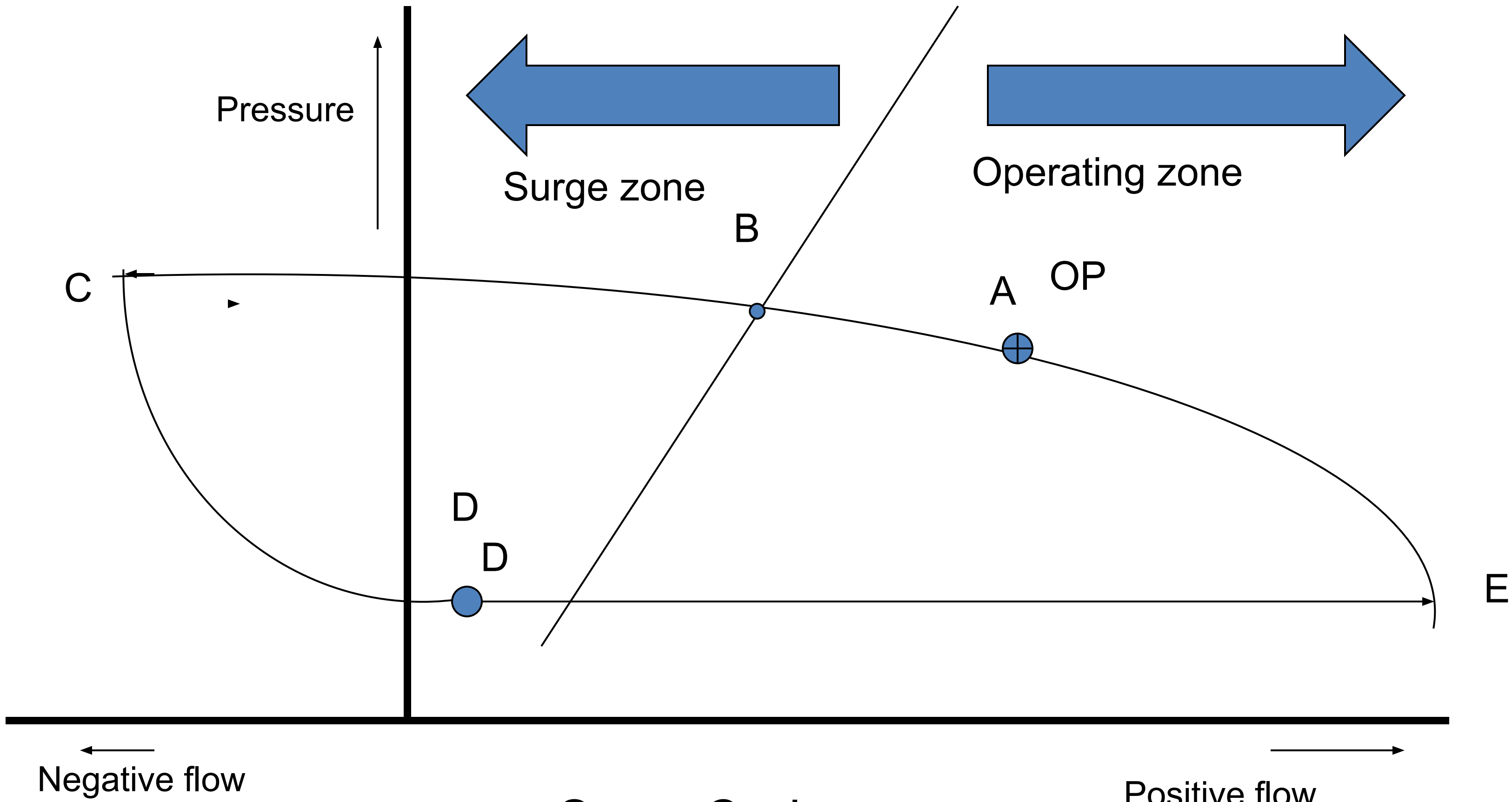
- There is for every speed and pressure of a centrifugal compressor a certain minimum volume below which the machine does not operate properly. This volume is called the surge point
- Below it delivery of gas becomes irregular, reversing itself at frequent intervals with a characteristic noise known as surging.



Compressor Performance Curve

Dynamic Compressor - Centrifugal

SURGE



Surge Cycle

Surge (con't)

Refer to - Surge cycle figure

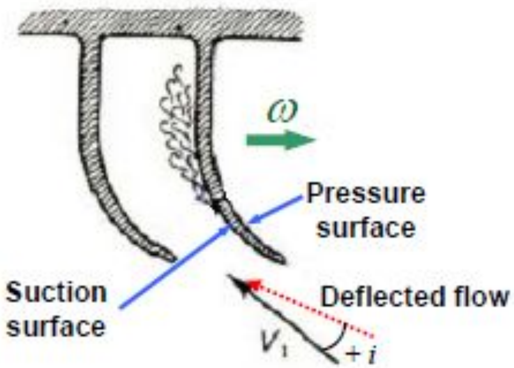
Consider a compressor operating in steady state at point A. If the load is reduced, the OP (operating point) must move toward B. the SURGE POINT. At B the compressor is producing more flow than the load can absorb. This fluid is temporarily stored in the discharge volume, but the discharge pressure cannot rise above B. The only relief for these conditions is for the OPERATING POINT to jump to point C. This is the flow reversal often observed during surge.

With negative flow the discharge pressure drops (traject C-D). At point D we find that the flow is insufficient to build up the pressure necessary to reach B, so the OPERATING POINT jumps to E. Now the flow is in excess of the load and the OPERATING POINT will move up the curve to reach B again. This completes one SURGE CYCLE. The typical duration of one SURGE CYCLE is 0.5 to 2.0 seconds.

The consequences of surge are severe. Besides process disturbance and the eventual process trips and disruption, surge can damage the compressor:

- Damage to seals and bearings is common.
- Internal clearance are altered, leading to internal recycle
- Lowering of compressor efficiency

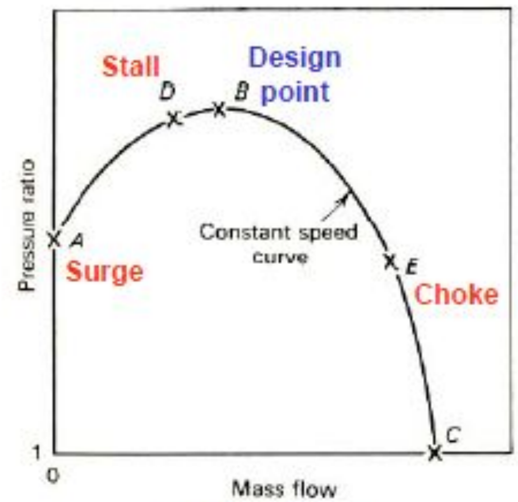
Stall, Surge and Choke



Stall and Surge

At low flow rates at a given speed, the reduction in axial velocity causes the flow to enter the inducer at large positive incipience resulting in flow separation on the suction surface leading to the phenomenon of rotating stall.

Stall can also initiate at the diffuser due to large positive incidence at reduced flow rates.



Choking

At high flow rates at a given speed, the pressure and density reduce, causing an increase in radial velocity (continuity equation). The relative velocity also become high with negative incidence at inducer and diffuser leading ends.

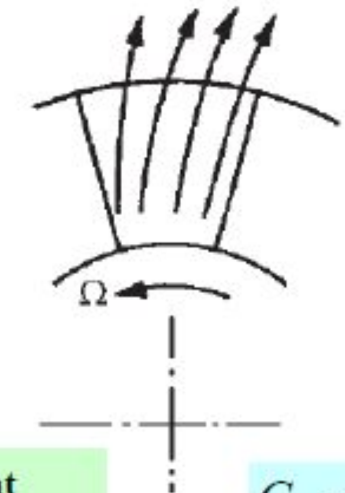
Finally, choking may occur owing to large flow blockage due to separation on the pressure surface or due to formation of shocks in the inducer / diffuser passages.

Slip

Even under ideal (frictionless) conditions the relative flow leaving the impeller does not receive perfect guidance from the vanes and the flow is said to *slip*. Hence, $\beta_2 > \beta_2'$.

average relative flow angle

vane angle



The *slip velocity is* defined as

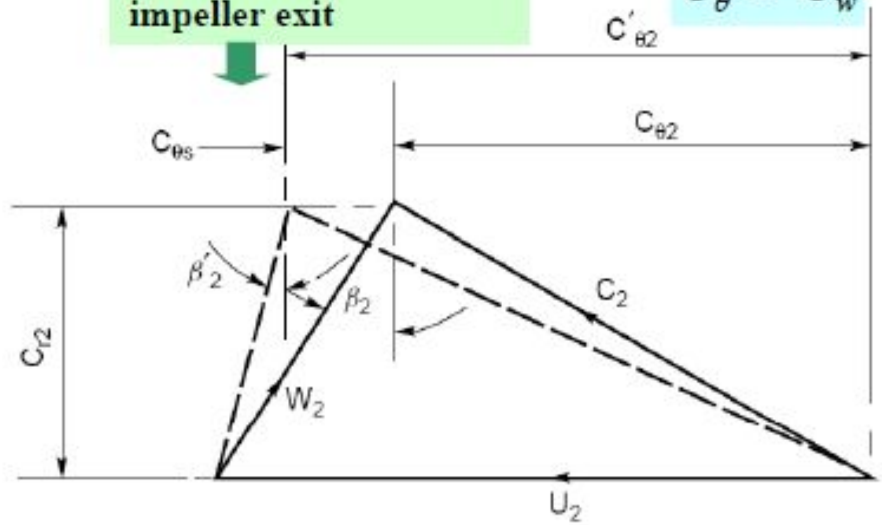
$$C_{ws} = C_{w2}^{'} - C_{w2}$$

and Slip Factor, $\boxed{\sigma = \frac{C_{w2}}{C_{w2}^{'}} < 1}$

For radial impeller, $\sigma = \frac{C_{w2}}{U_2}$

Velocity triangle at impeller exit

$C_\theta \equiv C_w$



Pre whirl

It has been observed that tangential velocity of the inlet impeller end is very high due to its exceedingly high revolutions per minute (sometimes, as high as 20 9(X) r.p rn). At this point, there

is always a tendency for the air stream to break away from the trailing face of the curved part of the impeller vane. This phenomenon, under certain **sel of condltions*causes the shock waves to form.**

The shock waves increase the loss of energy.

In order to eliminate (or reduce) the shock waves, the air is made to rotate before it enters the

impeller blades. This process, which causes the air to enter the impeller blades at a reduced velocity

(Without effecting the mass of air to flow and velocity of flow), is known as pre-rotation or *prewhirl.*

Losses in Compressors

Frictional Losses

Major portion of the losses is due to fluid friction in stationary and rotating blade passages

Flow in impeller and diffuser is decelerating in nature

Frictional losses are due to both skin friction and boundary layer separation

Depend on the friction factor, length of the flow passage and square of the fluid velocity

Incidence Losses

During the off-design conditions, the direction of relative velocity of fluid at inlet does not match with the inlet blade angle

Hence, fluid cannot enter the blade passage smoothly by gliding along the blade surface

The loss in energy that takes place because of this is known as incidence loss

This is sometimes referred to as shock losses. However, the word shock in this context should not be confused with the aerodynamic sense of shock

Clearance and leakage losses

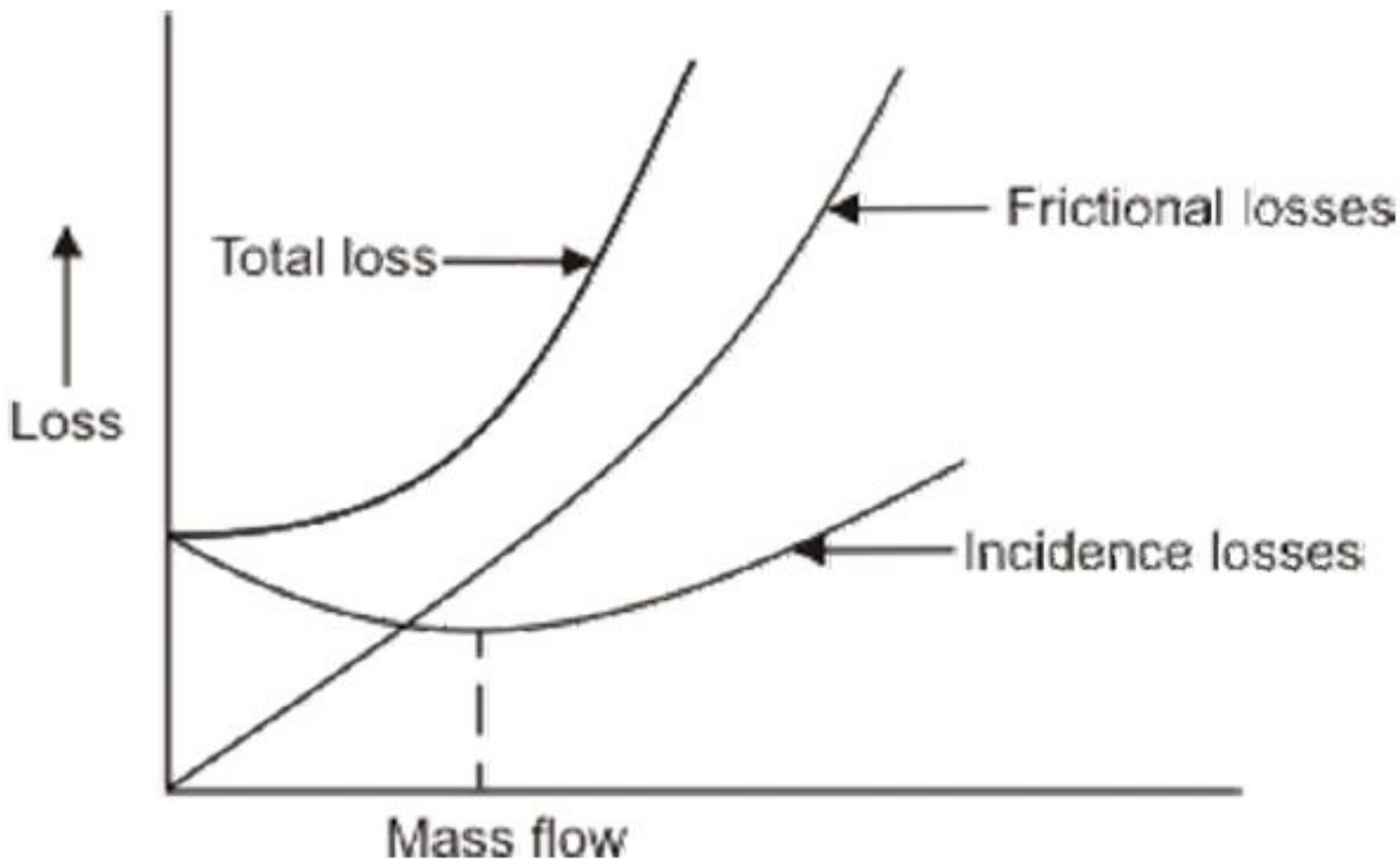
Certain minimum clearances are necessary between the impeller shaft and the casing and between the outlet periphery of the impeller eye and the casing

The leakage of gas through the shaft clearance is minimized by employing glands.

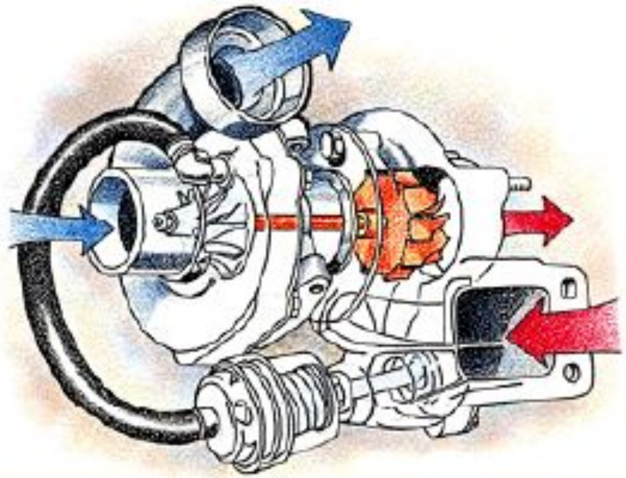
The clearance losses depend upon the impeller diameter and the static pressure at the impeller tip.

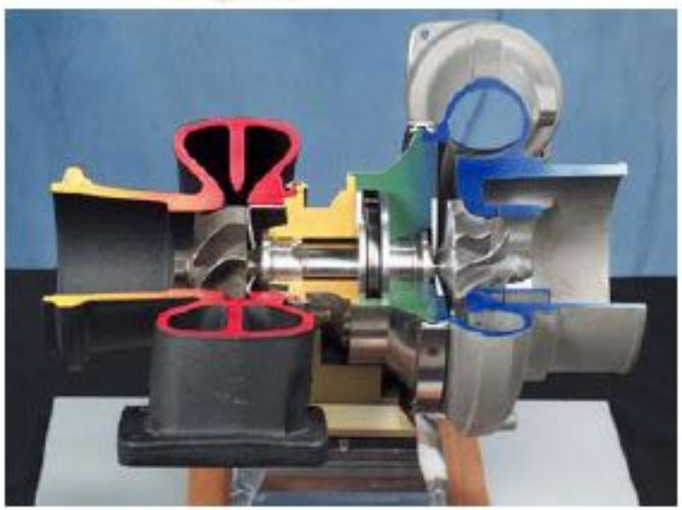
A larger diameter of impeller is necessary for a higher peripheral speed and it is very difficult in the situation to provide sealing between the casing and the impeller eye tip.

Losses in Compressors –Dependence of various losses with mass flow



Applications of Centrifugal Compressors





Use of centrifugal compressor in turbocharger

Rolls Royce Goblin II engine using centrifugal compressor