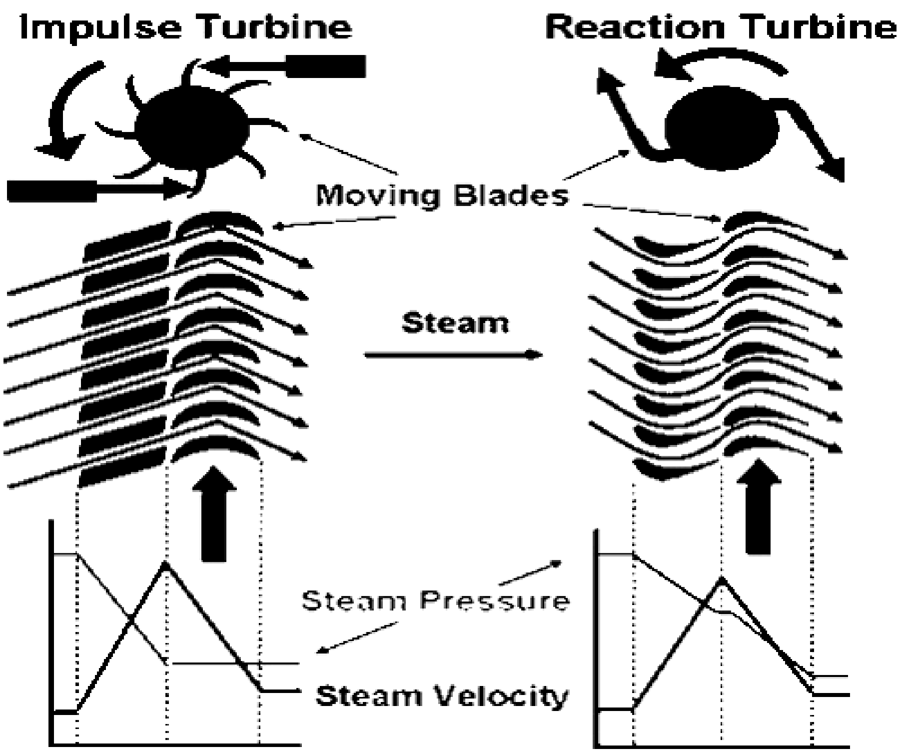
UNIT 03

# STEAM TURBINE

Normally the turbines are classified into types,

1. Impulse Turbine
2. Reaction Turbine

**Impulse and Reaction Turbines:**




impulse turbine and reaction turbine pressure and velocity diagram

### 3.5.1 Impulse Turbines:

The steam jets are directed at the turbines rotor blades where the pressure exerted by the jets causes the rotor to rotate and the velocity of the steam to reduce as it imparts its kinetic energy to the blades. The blades in turn change the direction of flow of the steam however its pressure remains constant as it passes through the rotor blades since the cross section of the chamber between the blades is constant. Impulse turbines are therefore also known as constant pressure turbines. The next series of fixed blades reverses the direction of the steam before it passes to the second row of moving blades

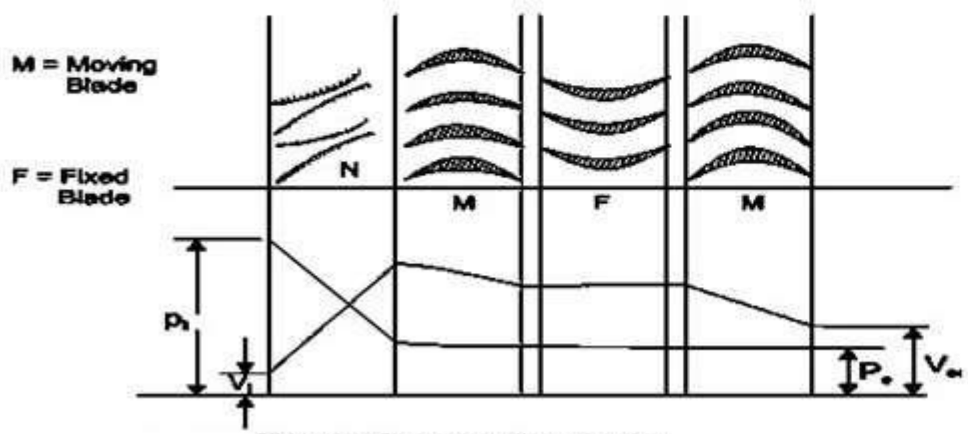
### 3.5.2 Reaction Turbines

The rotor blades of the reaction turbine are shaped more like aero foils, arranged such that the cross section of the chambers formed between the fixed blades diminishes from the inlet side towards the exhaust side of the blades. The chambers between the rotor blades essentially form nozzles so that as the steam progresses through the chambers its velocity increases while at the same time its pressure decreases, just as in the nozzles formed by the fixed blades. Thus the pressure decreases in both the fixed and moving blades. As the steam emerges in a jet from between the rotor blades, it creates a reactive force on the blades which in turn creates the turning moment on the turbine rotor, just as in Hero's steam engine. (Newton's Third Law – For every action there is an equal and opposite reaction).

**Compounding of impulse turbine:**

This is done to reduce the rotational speed of the impulse turbine to practical limits. (A rotor speed of 30,000 rpm is possible, which is pretty high for practical uses.) - Compounding is achieved by using 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. - Three main types of compounded impulse turbines are: a) Pressure compounded, b) velocity compounded and c) pressure and velocity compounded impulse turbines.

**Velocity Compounding:**



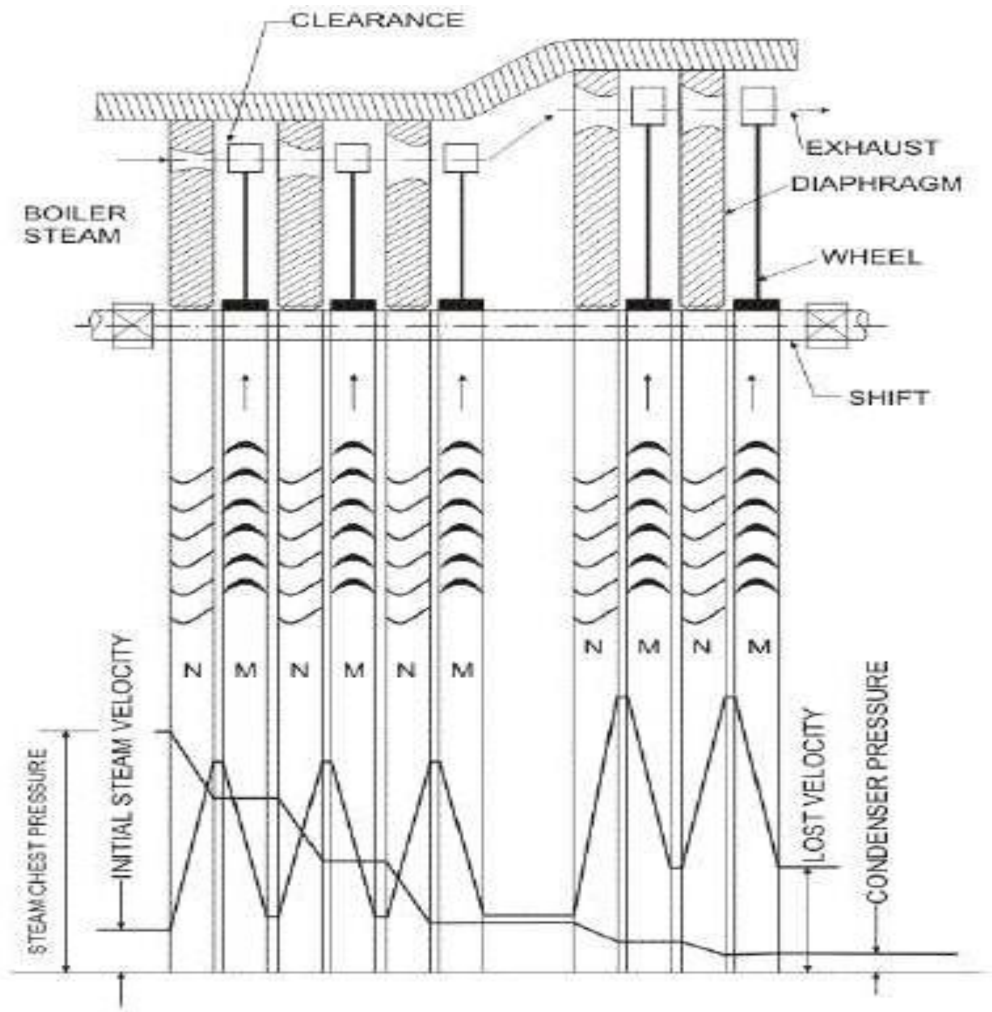
Velocity Compounding diagram

$P_i$ = Inlet Pressure, Pe= Exit Pressure, $V_i$ =Inlet Velocity, $V_e$=Exit Velocity.

The velocity-compounded impulse turbine was first proposed by C.G. Curtis to solve the problems of a single-stage impulse turbine for use with high pressure and temperature steam. The Curtis stage turbine, as it came to be called, is composed of one stage of nozzles as the single-stage turbine, followed by two rows of moving blades instead of one. These two rows are separated by one row of fixed blades attached to the turbine stator, which has the function of redirecting the steam leaving the first row of moving blades to the second row of moving blades. A Curtis stage impulse turbine is shown in Fig. with schematic pressure and absolute steam- velocity changes through the stage. In the Curtis stage, the total enthalpy drop and hence pressure drop occur in the nozzles so that the pressure remains constant in all three rows of blades.

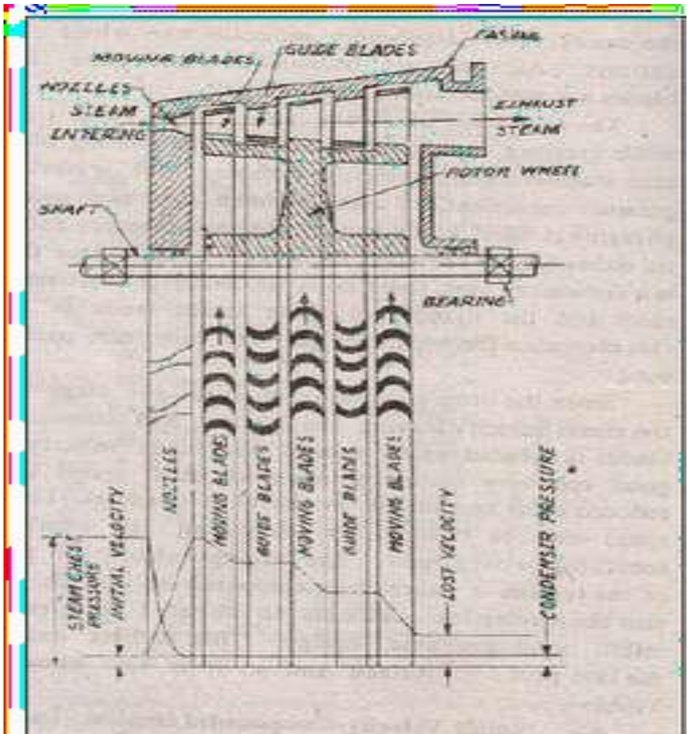
**Pressure Compounding:**

This involves splitting up of the whole pressure drop from the steam chest pressure to the condenser pressure into a series of smaller pressure drops across several stages of impulse turbine. -The nozzles are fitted into a diaphragm locked in the casing. This diaphragm separates one wheel chamber from another. All rotors are mounted on the same shaft and the blades are attached on the rotor.




**Pressure-Velocity Compounding**

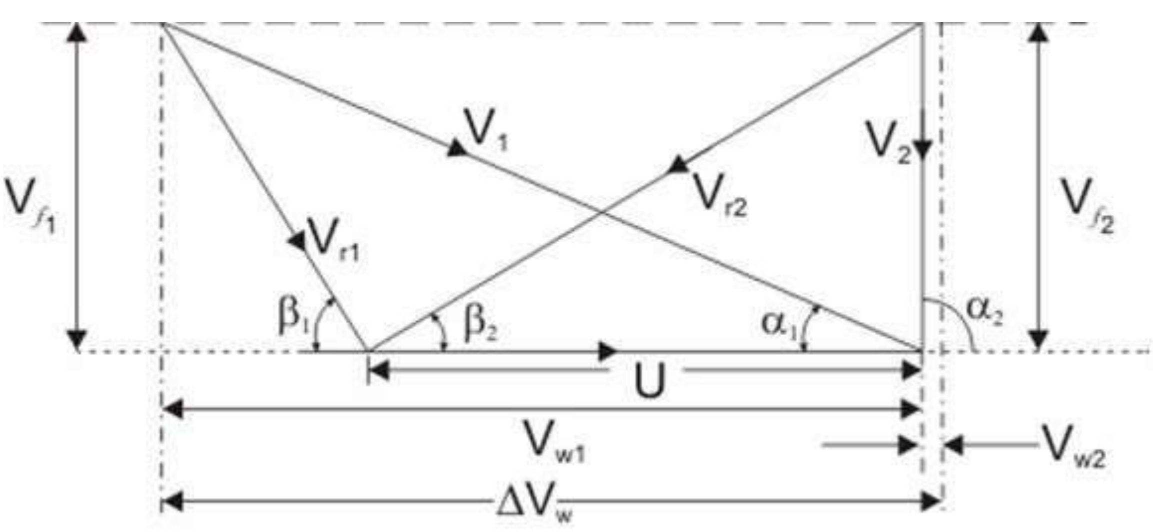
This is a combination of pressure and velocity compounding. A two-row velocity compounded turbine is found to be more efficient than the three-row type. In a two-step pressure velocity compounded turbine, the first pressure drop occurs in the first set of nozzles, the resulting gain in the kinetic energy is absorbed successively in two rows of moving blades before the second pressure drop occurs in the second set of nozzles. Since the kinetic energy gained in each step is absorbed completely before the next pressure drop, the turbine is pressure compounded and as well as velocity compounded. The kinetic energy gained due to the second pressure drop in the second set of nozzles is absorbed successively in the two rows of moving blades.



Diagrammatic arrangement of Velocity Compounded Impulse Turbine.

The pressure velocity compounded steam turbine is comparatively simple in construction and is much more compact than the pressure compounded turbine.

**Velocity diagram of an impulse turbine:**



Velocity diagram of an impulse turbine

$V_1$ and $V_2$ = Inlet and outlet absolute velocity

$V_{r_1}$ and $V_{r_2}$ = Inlet and outlet relative velocity (Velocity relative to the rotor blades.)

U = mean blade speed

$\alpha_1$ = nozzle angle, $\alpha_2$ = absolute fluid angle at outlet

It is to be mentioned that all angles are with respect to the tangential velocity ( in the direction of U )

$\beta_1$ and $\beta_2$ = Inlet and outlet **blade angles**

$V_{w1}$ and $V_{w2}$ = Tangential or whirl component of absolute velocity at inlet and outlet

$V_{f1}$ and $V_{f2}$ = Axial component of velocity at inlet and outlet

Tangential force on a blade,

$$F_u = \dot{m}(V_{w1} - Vw2)$$

(mass flow rate X change in velocity in tangential direction)

or,

$$F_u = \dot{m}\Delta V_w$$

**Power developed =** $\dot{m} U \Delta V_w$

Blade efficiency or Diagram efficiency or Utilization factor is given by

$$\eta_b = \frac{\dot{m} \cdot U \cdot \Delta V_w}{m(V_1^2/2)} = \frac{\text{Workdone}}{\text{K.E supplied}}$$

Or,

$$\eta_b = \frac{2U\Delta V_w}{V_1^2}$$

**stage efficiency** $= \eta_s = \dfrac{\textit{Work done by the rotor}}{\textit{Isentropic enthalpy drop}}$

$$\eta_s = \frac{\dot{m}U\Delta V_w}{\dot{m}(\Delta H)_{isen}} = \frac{\dot{m}U\Delta V_w}{\dot{m}\left(\frac{V_1^2}{2}\right)} \cdot \frac{\dot{m}(V_1^2/2)}{\dot{m}(\Delta H)_{isen}}$$

**or,**

**or,** $\eta_s = \eta_b \times \eta_n \qquad [\eta_n = \textit{Nozzle efficiency}]$

Optimum blade speed of a single stage turbine

$$\begin{aligned}\Delta V_w &= V_{r1}\cos\beta_1 + V_{r2}\cos\beta_2 \\ &= V_{r1}\cos\beta_1 + \left(1 + \frac{V_{r2}}{V_{r1}} \cdot \frac{\cos\beta_2}{\cos\beta_1}\right) \\ &= (V_1\cos\alpha_1 - U) + (1 + kc)\end{aligned}$$

where, $k = (V_{r2}/V_{r1})$ = friction coefficient

$$c = (\cos\beta_2/\cos\beta_1)$$

$$\eta_b = \frac{2U\,\Delta V_w}{V_1^2} = 2\frac{U}{V_1}\left(\cos\alpha_1 - \frac{U}{V_1}\right)(1 + kc)$$

$$\rho = \frac{U}{V_1} = \frac{\text{Blade speed}}{\text{Fluid velocity at the blade inlet}} = \text{Blade speed ratio}$$

$\eta_b$ is maximum when $\frac{d\eta_b}{d\rho} = 0$ also $\frac{d^2\eta_b}{d\rho} = -4(1+kc)$

$$\text{or,} \quad \frac{d}{d\rho}\{2(\rho\cos\alpha_1 - \rho^2)(1+kc)\} = 0$$

$$\text{or,} \quad \rho = \frac{\cos\alpha_1}{2}$$

$\alpha_1$ is of the order of $18^0$ to $22^0$

Now, $(\rho)_{opt} = \left(\frac{U}{V_1}\right)_{opt} = \frac{\cos\alpha_1}{2}$ (For single stage impulse turbine)

∴ The maximum value of blade efficiency

$$(\eta_b)_{max} = 2(\rho\cos\alpha_1 - \rho^2)(1+kc)$$

$$= \frac{\cos^2\alpha_1}{2}(1+kc)$$

For equiangular blades,

$$(\eta_b)_{max} = \frac{\cos^2\alpha_1}{2}(1+k)$$

If the friction over blade surface is neglected

$$(\eta_b)_{max} = \cos^2\alpha_1$$

The fixed blades are used to guide the outlet steam/gas from the previous stage in such a manner so as to smooth entry at the next stage is ensured.

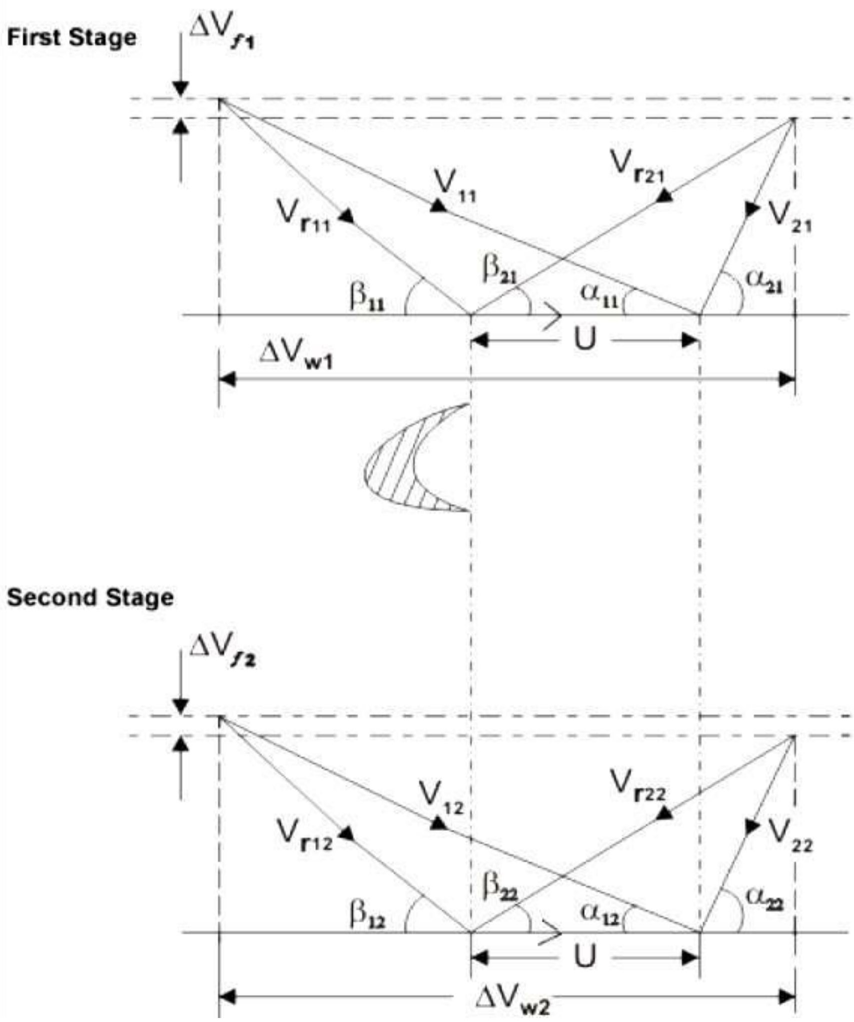
K, the blade velocity coefficient may be different in each row of blades

**Work done =** $\dot{m}.U(\Delta V_{w1} + \Delta V_{w2})$

$$\textbf{End thrust} = \dot{m}(\Delta V_{f1} + \Delta V_{f2})$$

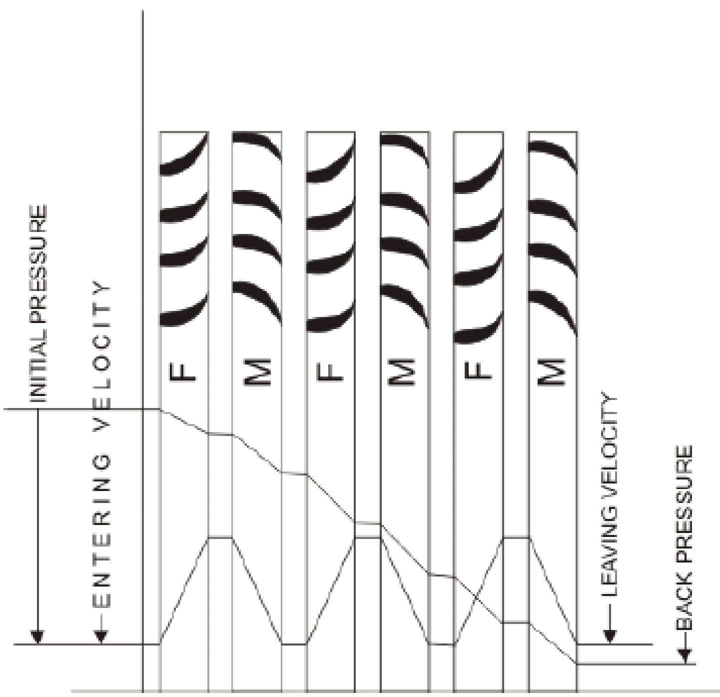
The optimum velocity ratio will depend on number of stages and is given by $P_{opt} = \frac{\cos \alpha_{11}}{2n}$

**Velocity diagram of the velocity compounded turbines:**



**Reaction Turbine:**

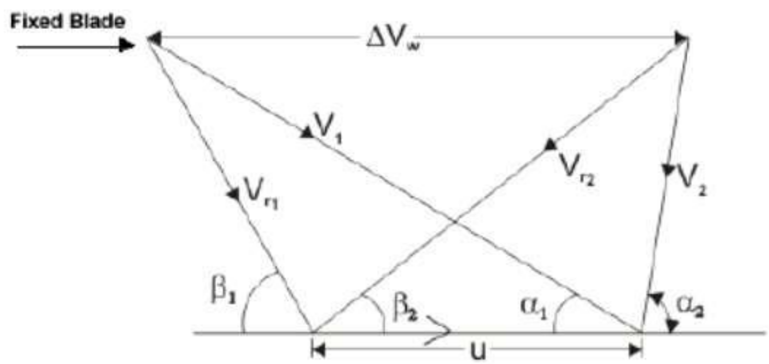
A ***reaction turbine***, therefore, is one that is constructed of rows of fixed and rows of moving blades. The fixed blades act as nozzles. The moving blades move as a result of the impulse of steam received (caused by change in momentum) and also as a result of expansion and acceleration of the steam relative to them. In other words, they also act as nozzles. The 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. 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 in figure 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 second figure shows a typical velocity diagram for the reaction stage.




Pressure and enthalpy drop both in the fixed blade or stator and in the moving blade or **Rotor**

**Degree of Reaction =** $\dfrac{\textit{Enthalpy drop in Rotor}}{\textit{Enthalpy drop in Stage}}$

**or,** $R = \dfrac{h_1 - h_2}{h_0 - h_1}$




A very widely used design has half degree of reaction or 50% reaction and this is known as Parson's Turbine. This consists of symmetrical stator and rotor blades.

The velocity triangles are symmetrical and we have

$$\alpha_1 = \beta_2 \quad . \quad \beta_1 = \alpha_2$$

$$V_1 = V_{r2} \quad . \quad V_{r1} = V_2$$

Energy input per stage (unit mass flow per second)

$$E = \frac{V_1^2}{2} + \frac{V_{r2}^2 - V_{r1}^2}{2}$$

$$E = V_1^2 - \frac{V_{r1}^2}{2}$$

$$E = V_1^2 - \frac{V_1^2}{2} - \frac{U^2}{2} + \frac{2V_1 U \cos\alpha_1}{2}$$

$$E = (V_1^2 - U^2 + 2V_1 U \cos\alpha_1)/2$$

From the inlet velocity triangle we have,

$$V_{r1}^2 = V_1^2 - U^2 - 2V_1 U \cos\alpha_1$$

Work done (for unit mass flow per second) $= W = U\Delta V_W$

$$= U(2V_1 \cos\alpha_1 - U)$$

Therefore, the Blade efficiency

$$= \eta_b = \frac{2U(2V_1 \cos\alpha_1 - U)}{V_1^2 - U^2 + 2V_1 U \cos\alpha_1}$$

**Governing of Steam Turbine:** The method of maintaining the turbine speed constant irrespective of the load is known as governing of tubines.The device used for governing of turbines is called Governor. There are 3 types of governors in steam turbine,

Throttle governing
Nozzle governing
By-pass governing

**1. Throttle Governing:**

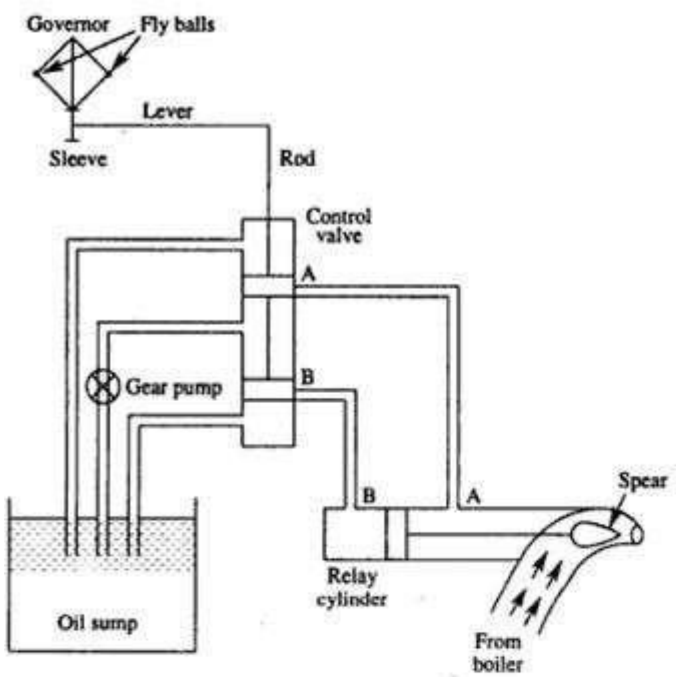



Fig 3.14 Throttle Governing

Let us consider an instant when the load on the turbine increases, as a result the speed of the turbine decreases. The fly balls of the governor will come down. The fly balls bring down the sleeve. The downward movement of the sleeve will raise the control valve rod. The mouth of the pipe AA will open. Now the oil under pressure will rush from the control valve to right side of piston in the rely cylinder through the pipe AA. This will move the piston and spear towards the left which will open more area of nozzle. As a result steam flow rate into the turbine increases, which in turn brings the speed of the turbine to the normal range.

## 2.Nozzle Governing:

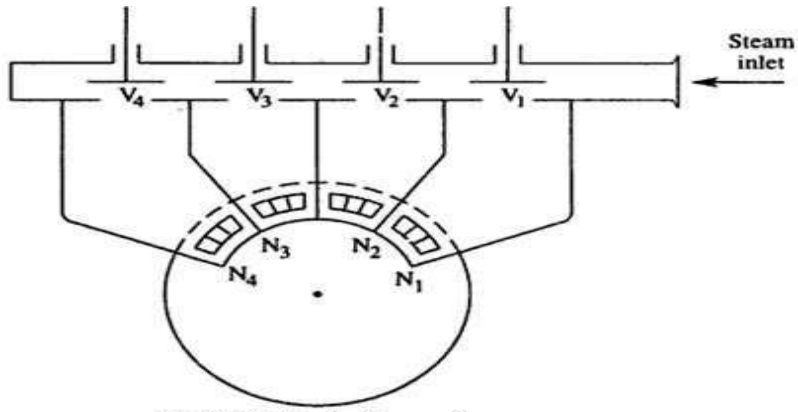



Fig 3.15 Nozzle Governing

A dynamic arrangement of nozzle control governing is shown in fig. In this nozzles are grouped in 3 to 5 or more groups and each group of nozzle is supplied steam controlled by valves. The arc of admission is limited to 180º or less. The nozzle controlled governing is restricted to the first stage of the turbine, the nozzle area in other stages remaining constant. It is suitable for the simple turbine and for larger units which have an impulse stage followed by an impulse reaction turbine.

## Practice Problems

**1. Steam at 10.5 bar and 0.95 dryness is expanded through a convergent divergent nozzle. The pressure of steam leaving the nozzle is 0.85 bar. Find i) velocity of steam at throat for maximum discharge, ii) the area at exit iii) steam discharge if the throat area is 1.2cm2. assume the flow is isentropic and there are no friction losses. Take n= 1.135.**

**Given data:**

$P_1$ = 10.5 bar

$P_2$ = 0.85 bar

**Solution:**

**Area at throat** $A_t$ = 1.2 cm2

$x_1$ = 0.95

n = 1.135

solution:

we know that, for n = 1.135

Throat pressure Pt = 0.577 x $P_1$ = 0.577 x10.5 = 6.06 bar

properties of steam from steam tables:

**$P_1$ = 10.5 bar** $h_f$

= 772 KJ /kg $s_f$

=2.159 KJ/kg

$h_{fg}$ = 2006 KJ/kg

$s_{fg}$ = 4.407 KJ/kg

**$P_t$ = 6.09 bar**

$h_f$ = 673.25 KJ/kg

$s_f$ = 1.9375 KJ/kg

$h_{fg}$ = 2082.95 KJ/kg

$s_{fg}$ = 4.815 KJ/kg

$v_f$ = 0.01101 m3/kg

$v_g$ = 0.31556 m3 /kg

**P2 = 0.85 bar**

$h_f$ = 398.6 kJ/kg

$h_{fg}$ = 2269.8 kJ/kg

$s_f$ = 1.252 kJ/kgk

$s_{fg}$ = 6.163 kJ/kgk

$v_f$ = 0.001040 m3/kg

$v_g$ = 1.9721 m3/kg

$s_1 = s_{f1} + x_1 \times s_{fg}$

= 2.159 +0.95 x 4.407 = 6.34565 kJ/kgk

$h_1 = h_{f1} + x_1 \times h_{fg1}$

= 772 +0.95 x 4.407 = 6.34565 KJ/Kgk

1-t isentropic expansion between inlet and throat

$s_1 = s_f$ = 6.34564 kJ/kg

$s_t = s_{ft} + x_t \times s_{fgt}$

6.34565 = 1.9375 + $x_t$ x 4.815

$x_t = 0.915$
$h_t = h_{ft} + x_t \times h_{fgt}$
$= 673.25 + 0.915 \times 2082.95$
$= 2579.15$ kJ/kg

**Velocity of steam at throat:**
$V_t = \sqrt{2000(h1 - ht)} = \sqrt{2000\,(2677.7 - 2579.15)}$
$= 443.96$ m/s
$v_t = x_t \times v_{gt}$
$= 0.915 \times 0.31156 = 0.2887\ m^3 / kg$

**Mass of steam discharged**:
$m = A_t \times V_t / v_t = 1.2 \times 10^{-4} \times 443.96/0.28874$
$= 0.1845$ kg/s
t-2 isentropic expansion between throat and exit
$s_t = s_2 = 6.34565$ kJ/kgk

$6.34565 = 1.252 + x_2 * 6.162$
$x_2 = 0.83$
$v_2 = 0.83 \times 1.9721 = 1.637\ m^3 / kg$

$h_2 = h_{f2} + x_2 \times h_{fg2} = 398.6 + 0.83 \times 2269.8$
$= 2282.534$ KJ/K

**Velocity of steam at exit**
$V_2 = \sqrt{2000(h_1 - h_2)}$
$= \sqrt{200(2677.7 - 2282.534)}$
$= 889$ m/sec
Accoding to mass balance , steam flow rate of throat is equal to flow rate at exit
$m_t = m_2 = A_2 V_2 / v_2$
$A2 = 3.397 \times 10^{-4} m^2$

**2. Dry saturated steam at 2.8 bar is expanded through a convergent nozzle to 1 .7 bar. The exit area is 3 cm$^2$. Calculate the exit velocity and mass flow rate for, i) isentropic expansion ii) supersaturated flow.**

**Given Data :**
$P_1 = 2.8$ bar
$P_2 = 1.7$ bar
$A_2 = 3\ cm^2 = 3 \times 10^{-4} m^2$

**Solution :**
Properties of steam table
**$P_1$ = 2.8 bar**

$h_1 = 2721.5$ KJ/kg
$s_1 = 7.014$ KJ/kgK
$v_1 = 0.64600\ m^3/kg$
**$P_2$ = 1.7 bar**
$h_f = 483.2$ KJ/kg
$h_{fg} = 2215.6$ KJ/kg , $s_f = 1.475$ KJ/kgK
$s_{fg} = 5.706$ KJ/kgK
$v_f = 0.001056\ m^3/kg$
$v_g = 1.0309\ m^3/kg$
For isentropic flow
$s_1 = s_2 = 7.014$ J/kgK
$s_2 = s_{f2} + x_2 \times s_{fg2}$
$7.014 = 1.475 + x_2 \times 5.706$
$x_2 = 0.97$
$h_2 = h_{f2} + x_2 \times h_{fg2}$
$= 483.2 + 0.97 \times 2215.6$
$h_2 = 2634.152$ KJ/kg
$v_2 = x_2 \times v_{g2}$
$= 0.97 \times 1.0309 = 1.00\ m^3/kg$

**Velocity of steam at exit**
$V_2 = \sqrt{2000(h_1 - h_2)}$
$= \sqrt{200(2721.5 - 2631.15)}$
$\underline{V_2 = 418 m/sec}$

**Mass flow rate at exit**
$$m_2 = \frac{A_2 \times v_2}{v_2}$$
$$= \frac{3 \times 10^{-4} \times 418}{1.00}$$
$= 0.1257\ m^3/kg$

**For super saturated flow**
$$V_2 = \sqrt{\frac{2n}{n-1} \times p_1 \times v_1 \left[1 - \left(1 - \frac{p_2}{p_1}\right)^{\frac{n-1}{n}}\right]}$$
$$V_2 = \sqrt{\frac{2 \times 1.3}{1.3-1} \times 2.8 \times 10^5 \times 0.6460 \left[1 - \left(1 - \frac{1.7}{2.8}\right)^{\frac{1.3-1}{1.3}}\right]}$$
$\underline{V2 = 413 m/sec}$

**Mass flow rate at exit**
$$m2 = \frac{A_2 \times v_2}{v_2} = \frac{3 \times 10^{-4} \times 413}{0.94827}$$
$= 0.1306$ kg/sec.

**3. Dry saturated steam at a pressure of 8 bar enters a C-D nozzle and leaves it a pressure of 1.5 bar. If the steam flow process is isentropic and if the corresponding expanding index is 1.135, Find the ratio of cross sectional area at exit and throat for maximum discharge.**

**Given Data:**

$P_1 = 2.8$ bar
$P_2 = 1.7$ bar
$n = 1.135$

**Solution:**
We know that $n = 1.135$
**Throat pressure** $p_t = 0.577 \times p_1 = 0.577 \times 8 = 4.62$ bar
Properties of steam at steam table
**At 8 bar**
$h_1 = 2769.1$ KJ/kg
$s_1 = 6.6628$ KJ/kgK
$v_1 = 0.2404$ $m^3/kg$

**At 4.62 bar**
$h_f = 626.7$ KJ/kg
$h_{fg} = 2117.2$ KJ/kg
$s_f = 1.829$ KJ/kgK
$s_{fg} = 5.018$ KJ/kgK
$v_f = 0.001090$ $m^3/kg$
$v_g = 0.40526$ $m^3/kg$

**At 1.5 bar**
$h_f = 467.11$ KJ/kg
$h_{fg} = 2226.5$ KJ/kg
$s_f = 1.4336$ KJ/kgK
$s_{fg} = 5.7897$ KJ/kgK
$v_f = 0.001053$ $m^3/kg$
$= 626.7 + 0.963 \times 2117.2$
$h_t = 2666.18$ KJ/kg
$v_t = x_t \times v_{gt}$
$= 0.963 \times 040526 = 0.39$ $m^3/kg$

**Velocity of steam at throat**

$V_t = \sqrt{2000}(h_1 - h_t)$

$= \sqrt{200}(2769.1 - 2666.18)$

$= 477.749$ m/sec

t-2 isentropic expansion

$s_t = s_2 = 6.6628$ KJ/kgK

$s_2 = s_{f2} + x_2 \times s_{fg2}$

$6.6628 = 1.4336 + x_2 \times 5.7897$

$x_2 = 0.903$

$v_2 = x_2 \times v_{g2}$

$= 0.903 \times 1.1593 = 1.04695\ m^3/kg$

$h_2 = h_{f2} + x_2 \times h_{fg2}$

$= 467.11 + 0.903 \times 2226.5$

$h_2 = 2477.6395$ KJ/kg

**Velocity of steam at exit**

$V_2 = \sqrt{2000}(h_1 - h_2)$

$= \sqrt{200}(2769.1 - 2477.639)$

$= 763.5$ m/sec

**According to mass balance**

Mass flow rate of steam at throat = Mass flow rate of steam at exit

$$m_t = m_2$$

$$\frac{A_t \times V_t}{v_t} = \frac{A_2 \times V_2}{v_2}$$

$$\frac{A_2}{A_t} = \frac{1.04695 \times 477.749}{763.5 \times 0.39} = 1.68$$

4.**The following data refer to a single stage impulse turbine. Isentropic nozzle entropy drop=200kj/kg Nozzle efficiency=90% Nozzle angle=Ratio of blade speed to whirl component of steam speed=0.5. blade coefficient =0.9. the velocity of steam entering the nozzle 30m/s. find (1).blade angles at the inlet and outlet if the steam enters the blade without shock and leaves the blade in the axial direction.(2). Blade efficiency (3).power developed (4).axial thrust if the steam flow rate is 10kg/s.**

**Given data:**

$h_{t=}h_e$=200kJ/kg

$\eta_N$=90%

$\alpha=25^0$

$\frac{v_b}{v_{w1}}=0.5$

$\frac{vr2}{vr1}=0.9$

$v_i$=30m/s

$v_2 = v_{f2}$

$v_{w2}=0$

$\beta=90^0$ for axial discharge

**Solution:**

Actual enthalpy drop

$h_i - h_e = (h_i - h_e)\eta_N$

$h_i - h_e = 200 \times 0.9$

=180 KJ/kg

$V_e=\sqrt{2(h_i\text{-}h_e)+v_i^2} = \sqrt{2(1000\text{-}180)+30^2}$

= 600.75 m/sec.

Inlet velocity of steam to the turbine

$v_1 = v_1 = 600\frac{m}{sec}$

From triangle ABC , $v_{\omega 1} = v_1 \cos 25°$

$= 600.75 \cos 25°$

= 544.46 m/sec

$v_{f1} = v_1 \sin 25°$

$=60.75 \times \sin 25° = 253.89$ m/sec

$v_b / v_{\omega 1}=0.5$

$v_b = 0.5 \times 544.46 = 272.3$ m/sec

from triangle ACE $v_{r1} = \sqrt{[{v_{f1}}^2 + (v_{\omega 1} - v_b)^2]}$

$= \sqrt{[253.89^2 + (544.76 - 272.23)^2]}$

$=372.25$ m/sec

$\tan \frac{v_{f1}}{v_{\omega 1} - v_b}$

$=253.89/(544.46-272.23)$

$\Theta=43°$

$v_{r2} = 0.9 \times v_{r1} = 0.9 \times 372.25 = 335.03$ m/sec

from triangle ABD,

$\cos \varphi = \frac{AB}{AD} = \frac{v_b}{v_{r2}} = \frac{272.23}{335.03}$

$\Phi=35°39'$

$v_2 = \sqrt{}(v_{r2} - v_b)^2$

$v_2 = \sqrt{}(335.03^2 - 272.03^2)^2$

$=195.28$ m/sec

$v_f = v_2 = 195.28\text{m/sec}$

**Power developed** $P = m(V_{w1} + V_{w2}) \times V_b = 10(544.46 + 0) \times 272.23$

$= 1482.18\text{kW}$.

**Blade efficiency:**

$\eta_b = m(V_{w1} + V_{w2}) \times V_b / (1/20(600.75)2 = 82.14\%$

Axial thrust

$Fy = m(V_{f1} - V_{f2}) = 10(253.89 - 175.28)$

$Fy = 586.1N$.

**5.Steam enters the blade row of an impulse turbine with a velocity of 600m/s at an angle of $25^0$C to the plane of rotation of the blades. The mean blade speed is 250m/s. the plant angle at the exit side is $30^0$. The blades friction less is 10%. Determine**

i) **The blades angle at inlet**
ii) **The workdone per kg of steam**
iii) **The diagram efficiency**
iv) **The axial thrust per kg of steam per sec.**

**Given data:**

$V_1 = 600$m/s

$\alpha = 25^0$

$V_b = 250$ m/s

$\emptyset = 30^0$

$V_{r2}/V_{r1} = 0.9$

Solution :

From $\Delta$ BCE,

$V_{w1} = V_1 \cos\alpha = 600 \cos 25^0 = 543.79$m/s

$V_{f1} = V_1 \sin\alpha = 600\sin 25^0 = 253.57$ m/s

From $\Delta$ ACE

$\tan\theta = V_{f1} / V_{w1} - V_b = 253.57/ [543.79 - 250]$

$$v_{r1} = \sqrt{[253.57^2 + (543.79 - 250)^2}$$

$=388.09$ m/ sec

$v_{r2} = 0.9 \times v_{r1} = 0.9 \times 388.09 = 349.28$ m/sec

From $\Delta$ ADF

$V_b + v_{w2} = v_{r2} \cos 30°$

$250 + v_{w2} = 349.28 \cos 30°$

$v_{w2} = 52.49$ m/sec

$V_{f2} = Vr_2 \sin 30° = 349.28° \sin 30^0 = 174.64$ m/s

**Work done** $W = m(v_{w1}+v_{w2})\, v_b$

$W = 1(543.79+52.49)\times 250 = 149.07$ KW/kg.

**Diagram efficiency** $\eta_D = \dfrac{(vw1+vw2)\, vb}{mv_1^2/2}$

$= \dfrac{149.07\times 1000}{1\times 0.5\times 600^2} = 82.82\%$

**Axial thrust** $F_y = m\,(v_{f1} - v_{f2})$

$= 1(253.57-174.64)$

$=79.73$ N/kg-sec

**6.At a particular stage of a reaction turbine , the mean blade speed is 60 m/sec and the steam pressure is 3.5 bar with atemperature of 175℃ . The identical fixed and moving blades have inlet angles 30° and outlet angle of 20°. Determine (i) The blade height if it is 1/10 of the blade ring diameter for a flow rate of 13.5 kg/sec.**

**(ii) The power developed by a pair**

**(iii) the specific enthalpy drop if the stage efficiency is 85%.**

**Given Data :**

Mean blade speed vb= 60 m/sec

Steam pressure = 3.5 bar

Temperature = 175°C

For identical fixed and moving blade,

$\Theta=\beta=30°$, $\alpha=\varphi=20°$.

m=13.5 kg/sec.

$h=1/10 \times d$

**Solution ;**

According to sine rule

$\Delta ABC$

$$\frac{v_1}{\sin 150} = \frac{v_{r1}}{\sin 20} = \frac{60}{\sin 10}$$

$V_{r1} = \frac{60}{\sin 10} \times \sin 20.$

$= 118.2$ m/sec

$v_{f1} = v_{r1} \times \sin 30° = 118.2 \times \sin 30°$

$= 59.1$ m/sec.

$FA = v_{r1} \times \cos 30° = 118.2 \times \cos 30°$

=102.4 m/sec.

$v_{w1}+v_{w2}$ = EA+AB+BF=102.4+60+102.4

= 264.8 m/sec.

**Velocity flow at exit** , $v_{f1} = 60$ m/sec.

Pressure of 3.5 bar and 175 °C.

From steam table,

$V_{sup} = 0.73$ $m^3$/kg.

**Mass of steam flow** (m)

$$13.5 = \frac{\pi(d+h)hV_{f1}}{V_{sup}} = \frac{\pi(10h+h)h \times 60}{0.73}$$

13.5= 2838 $h^2$

$h^2$=13.5/2838

h=0.068m=68mm.

**The power developed,**

By a pair of fixed and moving blade rings

$P = m(v_{w1}+v_{w2})\, v_b$

$= 13.5\,(264.8) \times 60 = 214650$W

=214.65 kW.

Heat Drop required for the efficiency of 85% Heat drop required

= 214.65/0.85 =252.52 kJ/sec.

**Steam Turbines**

A steam turbine is a key unit in a steam power plant from which we get power. A steam turbine is a turbo machine and a prime mover in which energy of stem is transformed into kinetic energy and this kinetic energy is then transformed into mechanical energy of rotation of shaft of turbine

The modern steam turbine was invented in 1884 by Sir Charles Parsons, whose first model was connected to a dynamo that generated 7.5 kW (10 hp) of electricity. The Parsons turbine also turned out to be easy to scale up. Parsons had the satisfaction of seeing his invention adopted for all major world power stations, and the size of generators had increased from his first 7.5 kW set up to units of 500MW capacity.

**Classification of steam turbine**

Classification of steam turbines may be done as following:

1. According to action of steam

(a) Impulse turbine

(b) Reaction turbine

(c) Combination of both

2. According to direction of flow:

(a) Axial flow turbine

(b) Radial flow turbine

3. According to number of stages

(a)Single stage turbine

(b)Multi stage turbine

4 .According to steam pressure at inlet of Turbine:

(a) Low pressure turbine

(b) Medium pressure turbine.

(c) High pressure turbine

(d) Super critical pressure turbine.

5. According to method of governing:

(a) Throttle governing turbine.

(b) Nozzle governing turbine.

(c) By pass governing turbine.

6. According to usage in industry:

(a) Stationary turbine with constant speed.

(b) Stationary turbine with variable speed.

(c) Non stationary turbines.

**Description of common types of Turbines**

The common types of steam turbine are

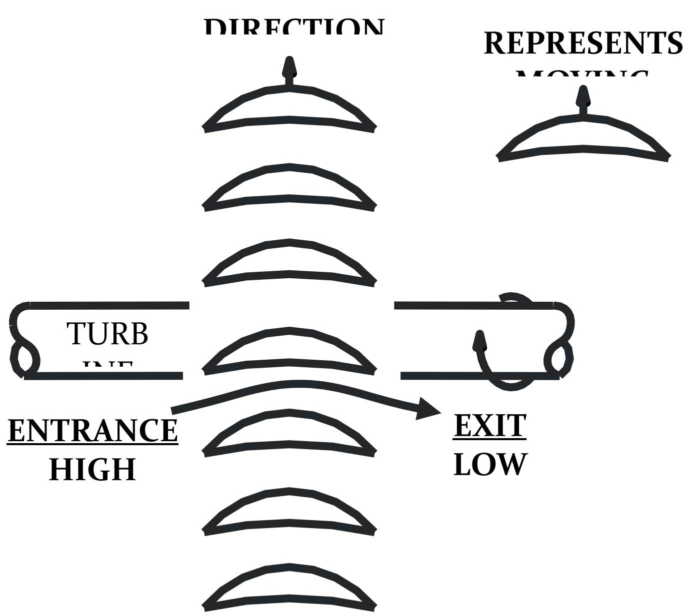
1. Impulse Turbine.

2. Reaction Turbine.

The main difference between these two turbines lies in the way of expanding the steam while it moves through them.

**Simple impulse Turbine**

In the impulse turbine, the steam is expanded within the nozzle and there is no any change in the steam pressure as it passes over the blades.



**Compounding in Steam Turbine**

The compounding is the way of reducing the wheel or rotor speed of the turbine to optimum value. It may be defined as the process of arranging the expansion of steam or the utilization of kinetic energy or both in several rings.

There are several methods of reducing the speed of rotor to lower value. All these methods utilize a multiple system of rotors in series keyed on a common shaft, and the steam pressure or jet velocity is absorbed in stages as the steam flow over the blades.

Different methods of compounding are:

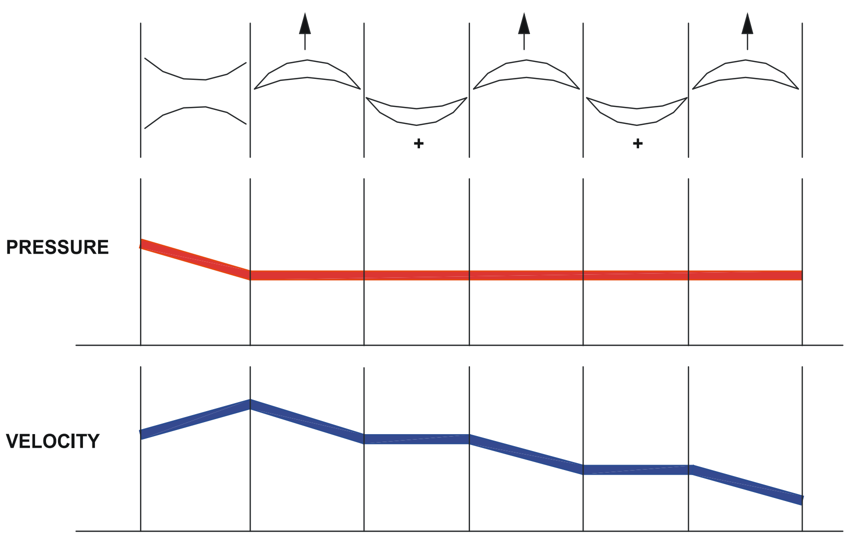
1. Velocity Compounding
2. Pressure Compounding
3. Pressure Velocity Compounding

**Velocity Compounding**

There are number of moving blades separated by rings of fixed blades. All the moving blades are keyed on a common shaft. When the steam passed through the nozzles where it is expanded to condenser pressure. Its Velocity becomes very high. This high velocity steam then passes through a series of moving and fixed blades.

When the steam passes over the moving blades it's velocity decreases. The function of the fixed blades is to re-direct the steam flow without altering it's velocity to the following next row moving blades where a work is done on them and steam leaves the turbine with allow velocity as shown in diagram.

**Velocity compounded turbine**




VISUALIZATION OF A VELOCITY COMPOUNDED TURBINE

**Pressure Compounding**

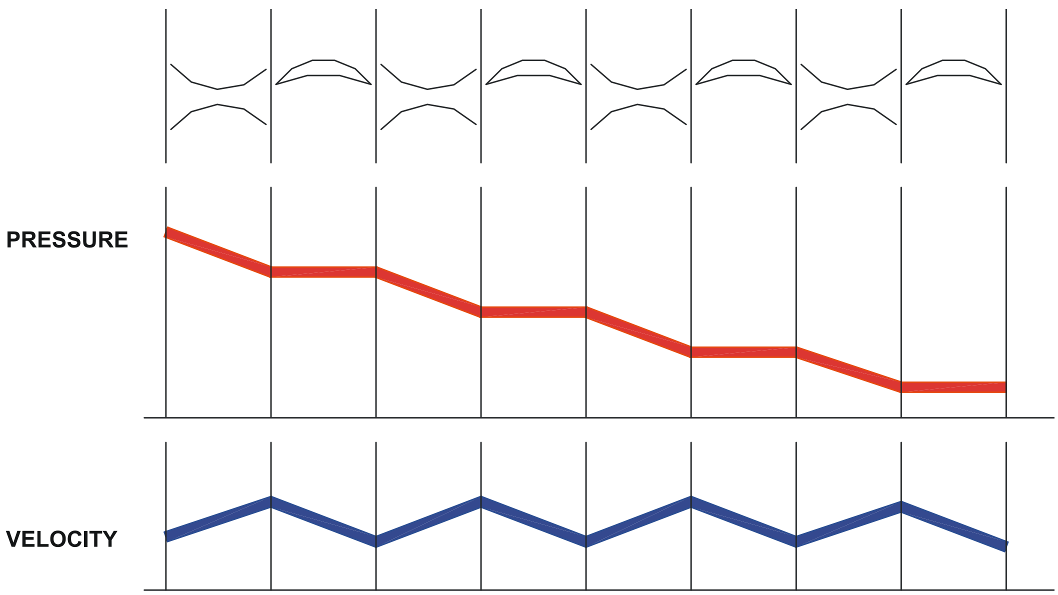
There are the rings of moving blades which are keyed on a same shaft in series, are separated by the rings of fixed nozzles.

The steam at boiler pressure enters the first set of nozzles and expanded partially. The kinetic energy of the steam thus obtained is absorbed by moving blades.

The steam is then expanded partially in second set of nozzles where it's pressure again falls and the velocity increase the kinetic energy so obtained is absorbed by second ring of moving blades.

This process repeats again and again and at last, steam leaves the turbine at low velocity and pressure. During entire process, the pressure decrease continuously but the velocity fluctuate as shown in diagram.

**Pressure compounded turbine**



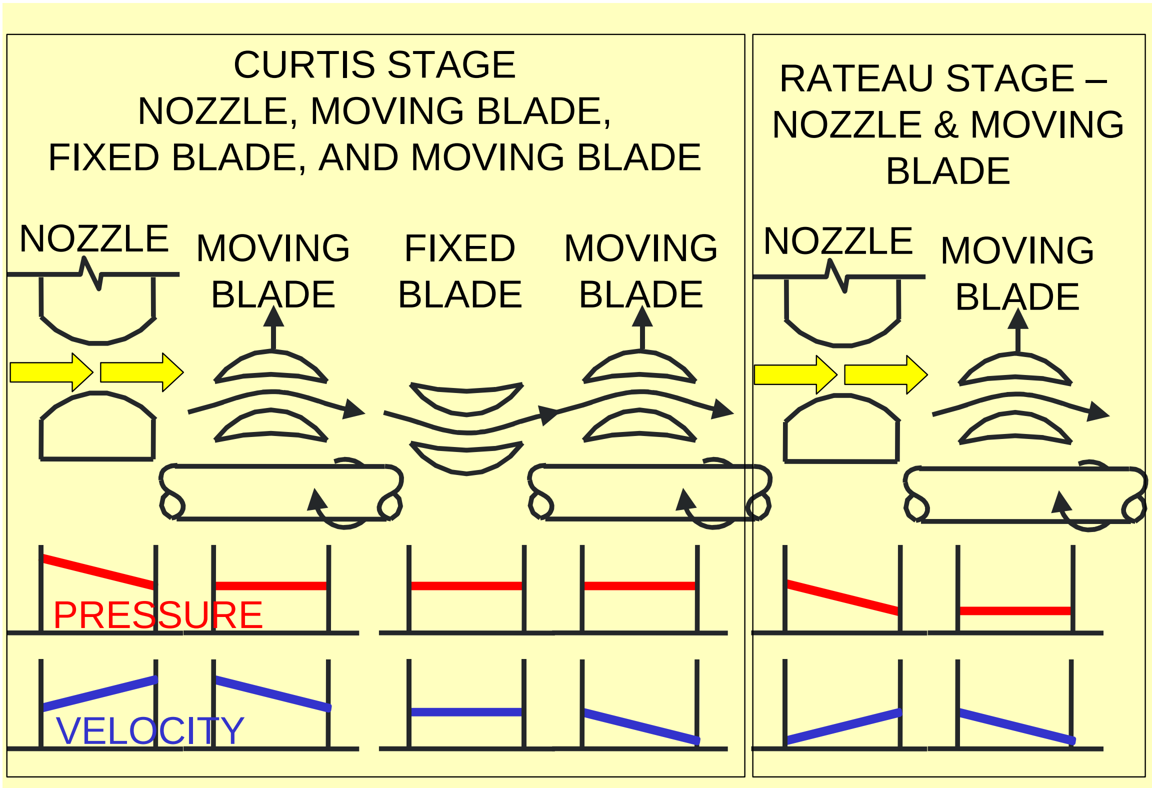

VISUALIZATION OF A PRESSURE COMPOUNDED TURBINE

**Pressure velocity compounding**

This method of compounding is the combination of two previously discussed methods. The total drop in steam pressure is divided into stages and the velocity obtained in each stage is also compounded. The rings of nozzles are fixed at the beginning of each stage and pressure remains constant during each stage as shown in figure.

The turbine employing this method of compounding may be said to combine many of the advantages of both pressure and velocity staging By allowing a bigger pressure drop in each stage, less number stages are necessary and hence a shorter turbine will be obtained for a given pressure drop.

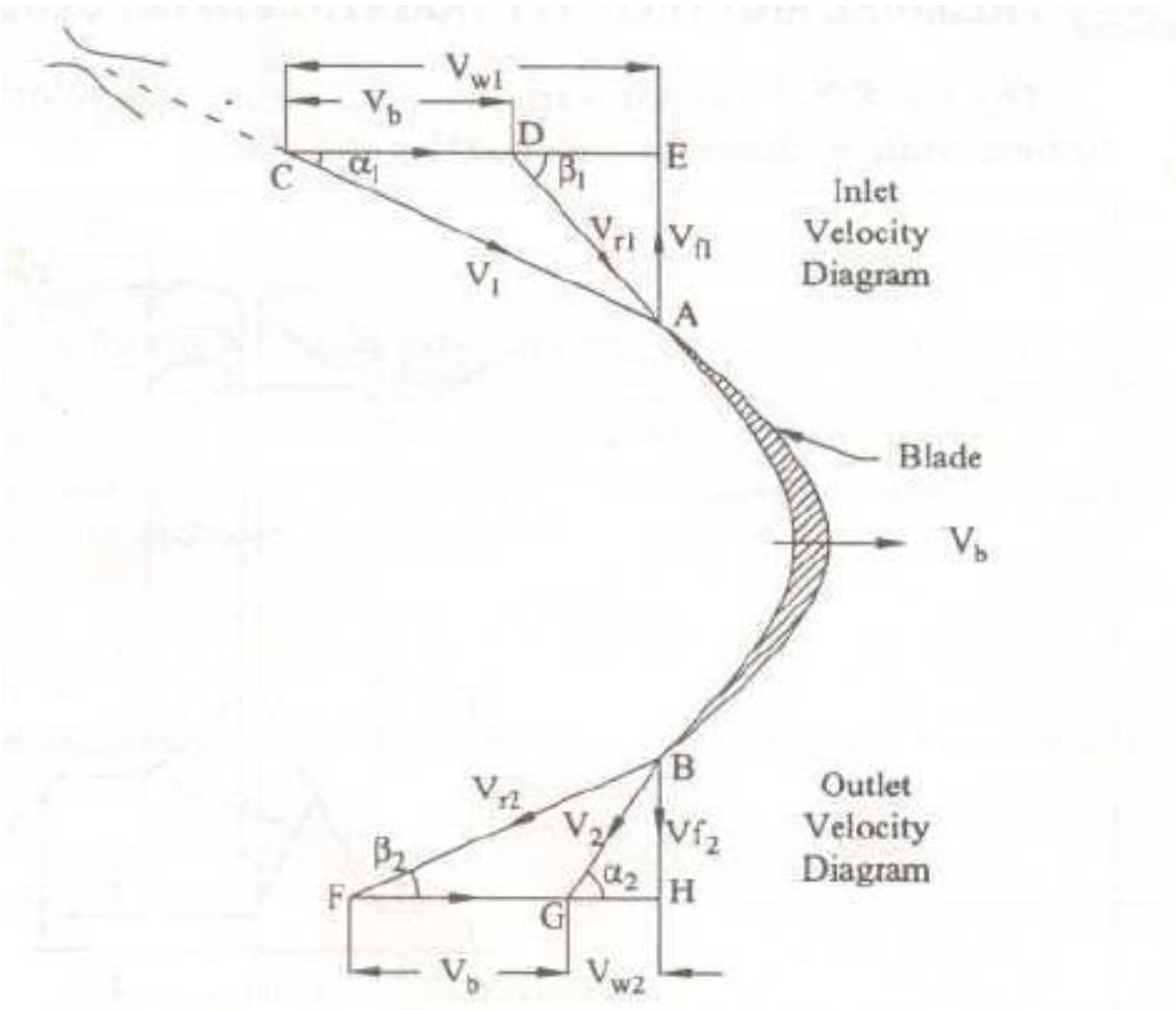
**Pressure-velocity compounded impulse turbine**




**Velocity Diagram and Analysis of Impulse and Reaction Turbines**

We should be able to calculate the propelling force applied to the turbine rotor. We can estimate work done and hence power.

Since the force is due to change of momentum mainly caused by change in direction of flow of steam, it is essential to draw velocity diagram that shows how velocity of the steam varies during its passage through the blades. Velocity is vector quantity as it has magnitude and direction. So we can represent velocity by a straight line and its length indicates its magnitude and direction is indicated by direction of line with reference to some fixed direction.




Velocity diagram for Impulse Turbine

Let:- Vb = linear velocity of moving blade

V1= absolute velocity of steam at inlet to moving blade; i. e. exit velocity of nozzle

Vw1= Tangential component of entering steam, also known as velocity of whirl at entrance

Vr1= relative velocity of steam wrt tip of blade at inlet,

It is the vectorial difference between Vb and V1

Vf1= velocity of flow = axial velocity at entrance to moving blades.

It is the vertical component of V1

α1= angle of nozzle= angle which the entering steam makes with the moving blade at entrance with tangent to the wheel at entrance

$\beta 1$= angle which the relative velocity makes with tangent of the wheel direction of motion of blade. Also known as blade angle at inlet.

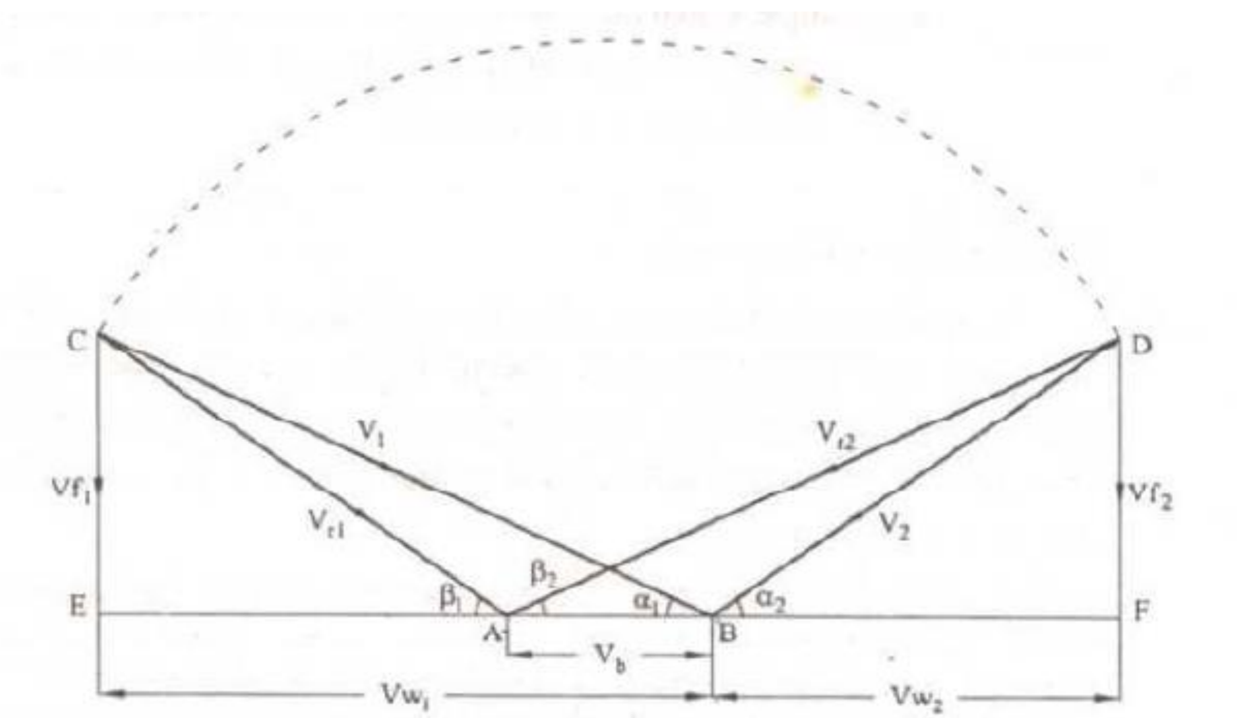
V2, Vw2, Vf2, Vr2, $\alpha 2$, $\beta 2$ are corresponding values at exit of the moving blade. They stand outlet velocity triangle.

The absolute velocity V2 can be considered as two components.

The tangential component called whirl component Vw1= V1 cos $\alpha 1$ is parallel to direction of rotation of blades and axial or flow component Vf1= V1 sin $\alpha 1$ is perpendicular to the direction of rotation of blades.

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

**Combined Velocity diagram for Impulse Turbine**



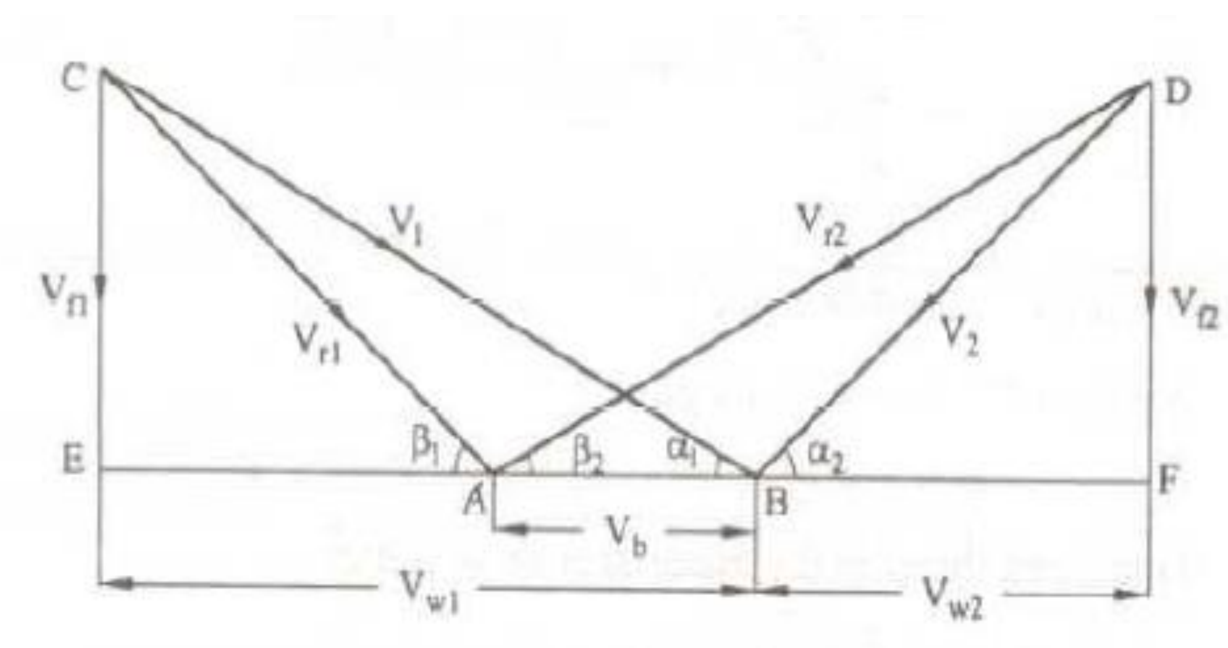
1) Draw horizontal line and cut off AB equal to velocity of blade
2) Draw line BC at an angle $\alpha 1$, with AB. Cut off BC equal to V1
3) Join AC. It represents Vr1
4) From A, draw line AD at an angle $\beta 2$ with AB. With A as centre and radius equal to AC , draw an arc that meets the line through A at D such that AC=AD or V1= Vr2
5) Join BD. It represents absolute velocity at exit.
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,

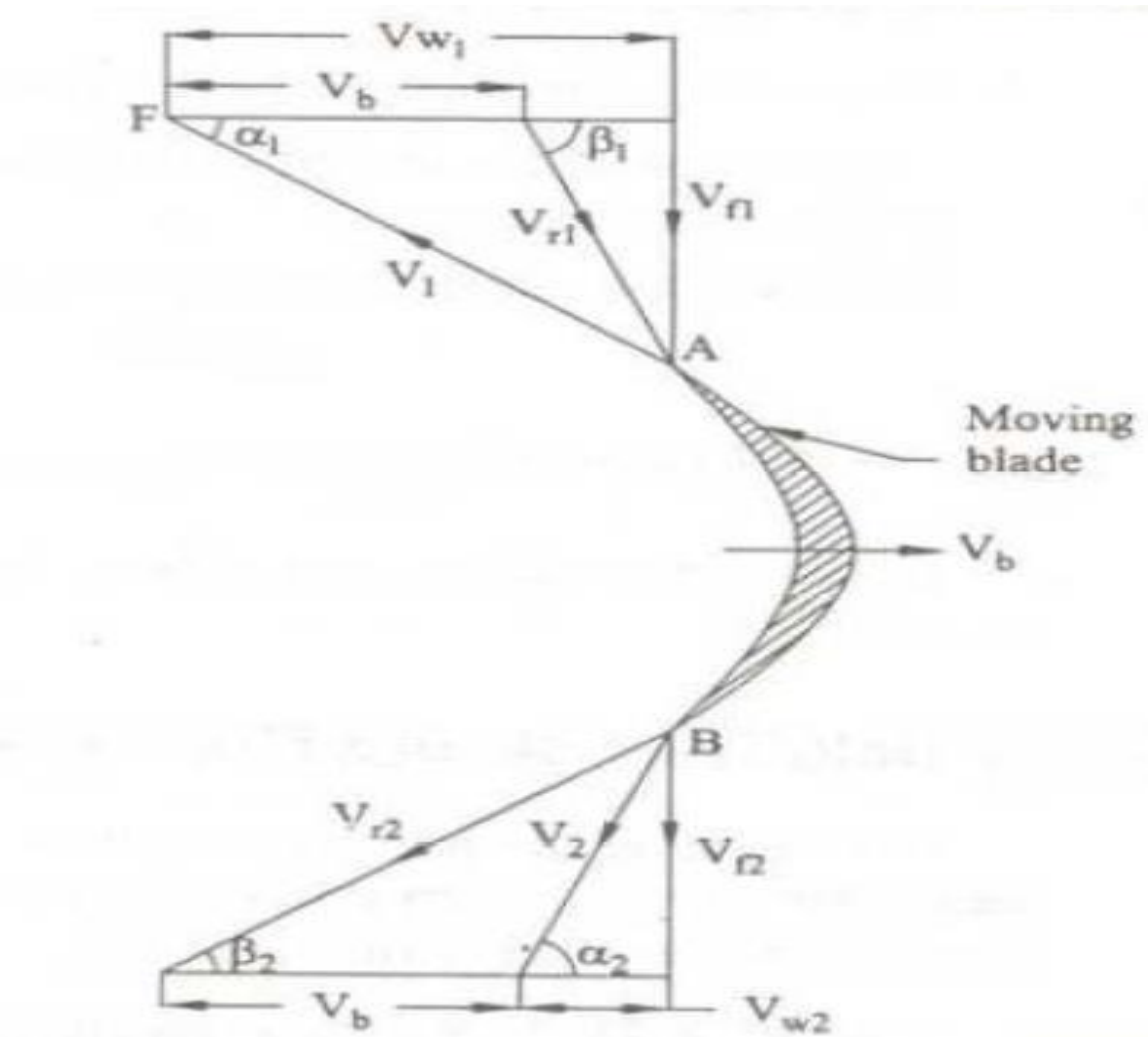
   BF→ velocity of whirl at exit, CE--> velocity of flow at inlet,

   DF→ velocity of flow at outlet.

When friction is neglected:- Vr1= Vr2, Vf1= Vf2 and $\beta 1= \beta 2$

**Velocity diagram for Reaction Turbine**






In Parsons Reaction Turbine, both the fixed and moving blades are made identical.

Hence, $\alpha1= \beta2$ and $\alpha2= \beta1$ .

Hence, velocity diagram for parsons reaction turbine will be symmetrical about vertical center line and

Vf1= Vf2 ; V1= Vr2;

V2= vr1

**Forces on blade and WD by the blades**

**1. Force on rotor= mass × tangential acceleration**

$$= m \times (V_{\omega 1} - V_{\omega 2})$$

Where m = mass flow rate of steam in kg/sec

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

Hence, Vw1 and Vw2 are added together. Generally,

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

2. Work done on blade = force ×distance

= tangential force × distance moved in unit

time in the direction of force

$$= F_t \,.\, V_b\, N - m/sec$$

$$= m\,.\,(V_{\omega 1} \pm V_{\omega 2})\,.\,V_b\, N - m/sec$$

**3. Power developed by the turbine = rate of doing work**

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

**4. Axial thrust on the rotor = Fa= mass × axial acceleration**

**= mass × change in velocity of flow**

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

**Efficiencies**:- Following efficiencies are common to both impulse and reaction turbines:-

1. Blading or diagram efficiency
2. Gross or stage efficiency
3. Nozzle efficiency

**1. Diagram efficiency or blading efficiency**

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

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

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

This is called diagram efficiency because the quantities involved in it are obtained from velocity diagram.

**2. Gross or Stage Efficiency**

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

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

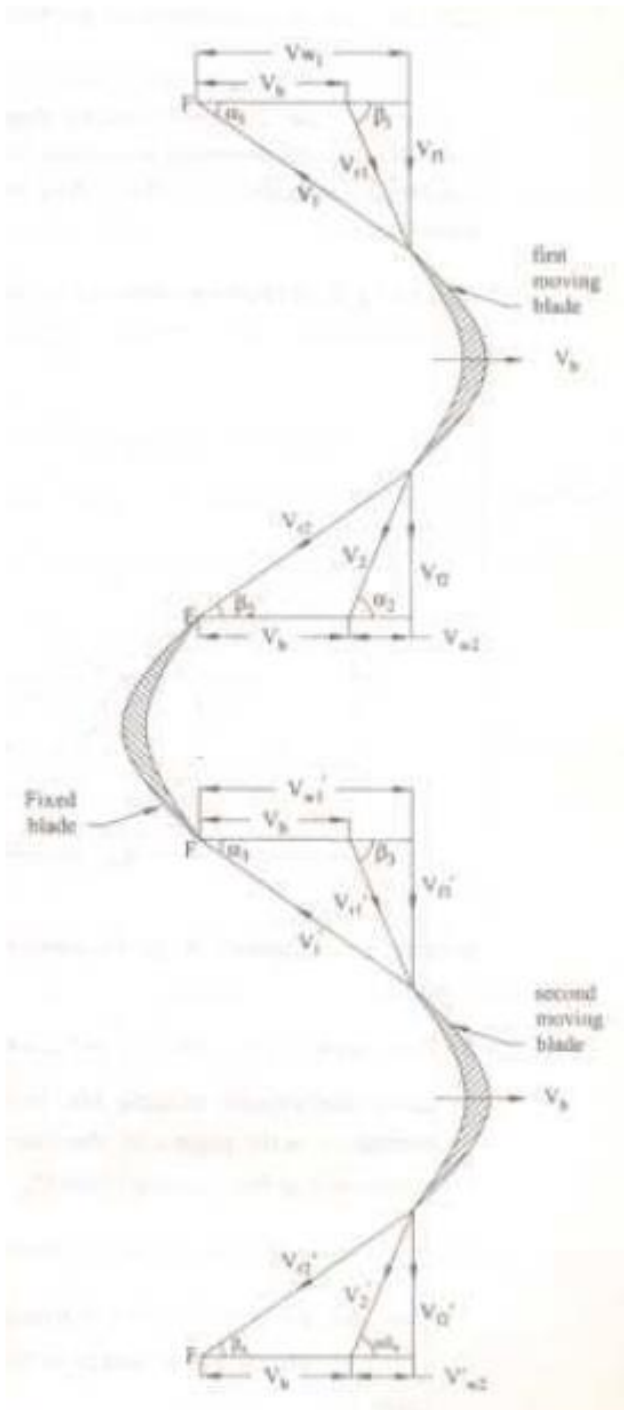
**3. Nozzle Efficiency**

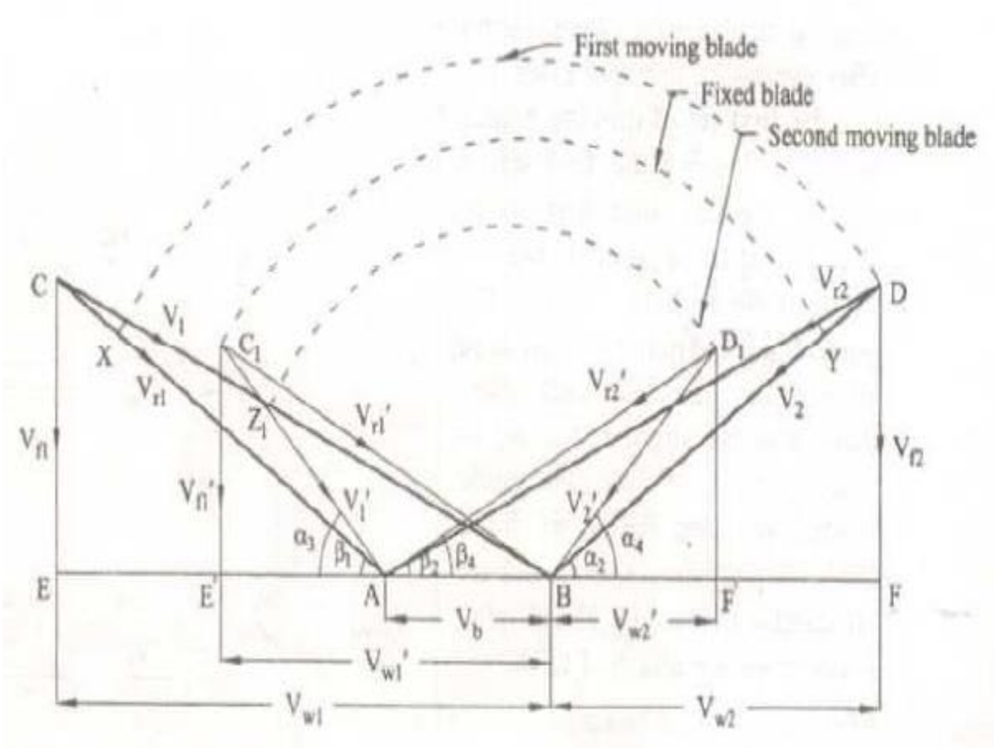
$$\text{Nozzle efficiency} = \frac{\text{Energy availbale at entrance}/kg}{\text{Enthalpy drop through a stage}/kg \text{ of steam}}$$

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

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

**Multi Stage Turbines**






**Calculations**

**1.WD per kg of steam passing through both the stages (Wt)**

**Wt = WD in first moving blade stage + WD in second moving blade stage**

$$= 2\,V_b \,.\, (V_1 \cos\alpha_1 - V_b) + 2\,V_b\,(V_1 \,.\, \cos\alpha_1 - 3\,V_b).$$

$$= 4\,V_b\,(V_1 \,.\, \cos\alpha_1 - 2\,V_b).$$

**2. Blading or Diagram Efficiency for two stage turbine**

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

$$= \frac{\omega_t}{\frac{1}{2} \,.\, m \,.\, {V_1}^2} = \frac{\omega_t}{\frac{1}{2}{V_1}^2} \qquad [m = 1\ kg]$$

$$= \frac{4 V_b \cdot (V_1 \cos \alpha_1 - 2 V_b) \cdot 2}{V_1^{\ 2}}$$

$$= \frac{8V_b}{V_1^{\ 2}} (V_1 \cos \alpha_1 - 2 V_b)$$

$$= 8 \cdot \frac{V_b}{V_1} \left( \cos \alpha_1 - 2 \cdot \frac{V_b}{V_1} \right)$$

$$= 8 \rho (\cos \alpha_1 - 2 \rho) \text{ where}$$

$$\rho = \frac{V_b}{V_1} = \text{blade speed ratio.}$$

For maximum efficiency, $\rho_{opt} = \dfrac{\cos \alpha_1}{4}$

Maximum efficiency $\eta_{bl(max)} = \cos^2 \alpha_1$.

**3. Maximum work done**

$$\omega_t = 4 V_b \cdot (V_1 \cos \alpha_1 - 2 V_b)$$

$$\text{Optimum value of } \rho = \frac{\cos \alpha_1}{4}$$

$$\frac{V_b}{V_1} = \frac{\cos \alpha_1}{4}$$

$$V_1 = \frac{4V_b}{\cos \alpha_1}$$

$$\omega_{t(max)} = 4V_b\left(\frac{4V_b}{\cos\alpha_1}.\cos\alpha_1 - 2V_b\right)$$

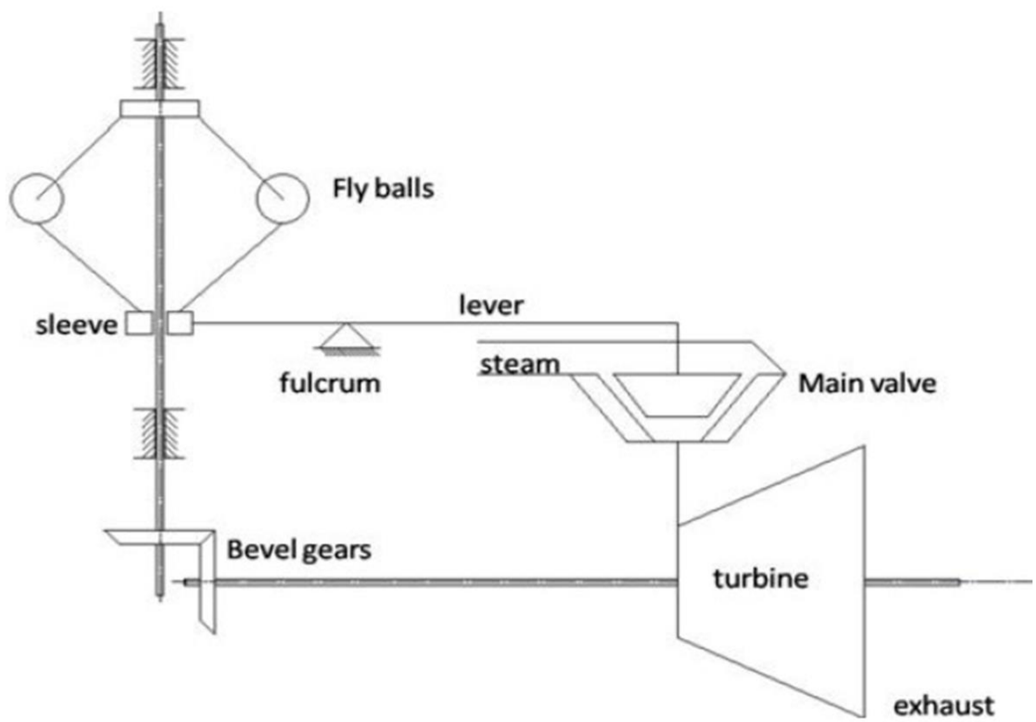
$$= 8V_b^{\ 2}$$

**Governing of Steam Turbines**

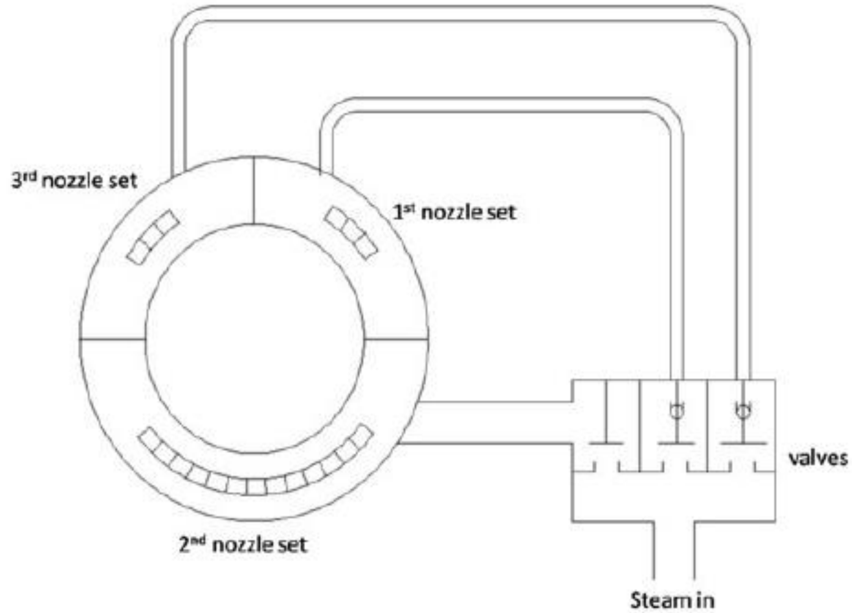
The main function of the governing is to maintain the speed constant irrespective of load on the turbine. The different methods which are commonly used for governing the steam turbines are listed below:-

1) Throttle governing
2) Nozzle control governing
3) By-pass governing

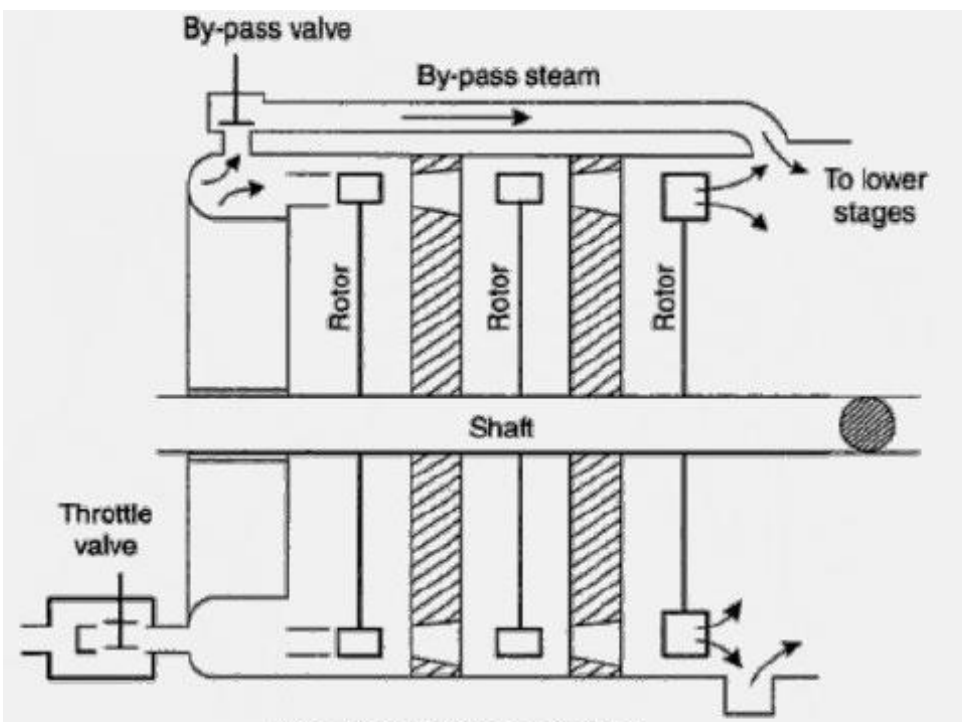
**Throttle Governing**



**Nozzle Control Governing**




**Bypass Governing**




*By-Pass Governing*

## UNIT III

## STEAM TURBINES PROBLEMS

**A steam at 4.9 bar and 160$^0$C is supplied to the single stage impulse turbine at a mass flow rate of 30 kg/min, from where it is exhausted to a condenser at a pressure of 19.6 kpa. the blade speed is 300 m/s. The nozzles are inclined as 25$^0$ to the plane of wheel and the outlet blade angle is 35$^{0.}$**

**Neglecting friction losses, determine i) theoretical power developed by the turbine. ii) diagram efficiency, and iii) stage efficiency.**

We know that for dry saturated steam ( or when n= 1.135), critical pressure ratio,

$$\frac{p_2}{p_1} = 0.577$$

$\therefore$ $p_2$=0.577p1=0.577x 8 = 4.616 bar

Now complete the Mollier diagram for the expansion of steam through the nozzle.

From Mollier diagram, we find that

$h_1$ = 2775kJ/kg; $h_2$ = 2650kJ/kg; $h_3$ = 2465 kJ/kg; $x_2$ = 0.965; and $x_3$ = 0.902

From steam tables, we also find that the specific volume of steam at throat corresponding to 4.616 bar,

$$\upsilon_{g^2} = 0.405 m^3 / kg$$

and specific volume of steam at exit corresponding to 1.5 bar,

$$\upsilon_{g^3} = 1.159 m^3 / kg$$

**. In a stage of impulse reaction turbine operating with 50% degree of reaction, the blades are identical in shape. The outlet angle of the moving blade is 19$^0$ and the absolute discharge velocity of steam is 100m/s in the direction70$^0$Cto the motion of blades. If the rate of flow through the turbine is 15000 kg/hr, calculate the power developed by the turbine**

We know that heat drop lost in friction

= 10% = 01.

$\therefore$ Nozzle coefficient or nozzle efficiency

K = 1 – 01 = 0.9

We know that final velocity of the steam,

$$V_2 = 44.72\ \sqrt{Kh_d} = 44.72\sqrt{0.9 \times 96.5} = 416.8 m/s$$

$\therefore$ Percentage reduction in final velocity

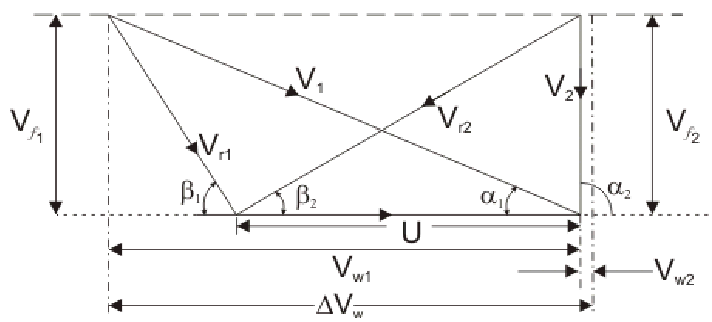
$$= \frac{439.3 - 416.8}{439.3} = 0.051 \ or \ 5.1\%$$

**The Velocity of steam, leaving the nozzles of an impulse turbine, is 1200m/s and the nozzle angle is 20°. The blade velocity is 375 m/s and the blade velocity coefficient is 0.75. Assuming no loss due to shock at inlet, calculate for a mass flow of 0.5kg/s and symmetrical blading: (a) blade inlet angle; (b) driving force on the wheel; (c) axial thrust on the wheel; and (d) power developed by the turbine.**

***Given :***

***V=1200m/s;*** $\alpha = 20^o; V_b = 375 m/s; K = V_{r1}/V_r = 0.75; m = 0.5 kg/s;$

$\theta = \phi$, for symmetrical blading.




***Solution:***

Now draw the combined velocity triangle, as shown in fig.22.7, as discussed below:

1. First of all, draw a horizontal line, and cut *AB* equal to 375 m/s to some suitable scale representing the velocity of blade ($V_b$).
2. Now at *B*, draw a line *BC* at an angle of 20° (Nozzle angle, $\alpha$ ) and cut off *BC* equal to 1200m/s to the scale to represent the velocity of stream jet entering the blade ($V_b$).
3. Join *CA*, which represents the relative velocity at inlet ($V_r$). By measurements, we find that *CA* = $V_r$ = 860 m/s. Now cut off *AX* equal to 860 x 0.75 = 645 m/s to the scale to represent the relative velocity at exit ($V_{r1}$).
4. At *A*, draw a line *AD* at an angle equal to the angle $\phi$ equal to the $\theta$, for symmetrical blading. Now with *A* as centre, and radius equal to *AX*, draw an are meeting the line through *A* at *D*, such that *AD* = $V_{r1}$.
5. Join *BD*, which represents the velocity of steam jet at exit ($V_1$).
6. From *C* and *D*, draw perpendiculars meeting the line *AB* produced at *E* and *F* respectively. CE and DF represents the velocity of flow at inlet ($V_f$) and outlet ($V_{f1}$) respectively.

   The following values are *measured from the velocity diagram :

   $\theta = 29^o; V_w = BE = 1130 m/s; V_{w1} = BF = 190 m/s$

   $V_f = CE = 410 m/s$ and $V_f = DF = 310$ m/s

***(a) Blade inlet angle***

By measurement from the velocity diagram, we find that the blade angle at inlet,

$\theta = 29^0$ **Ans.**

***(b) During force on the wheel***

We know that driving force on the wheel,

$F_x = m(V_w + V_{w1}) = 0.5\ (1130 + 190) = 660$ N **Ans.**

***(c) Arial thrust on the wheel***

We know that axial thrust on the wheel,

$F_\gamma = m(V_f - V_{f1}) = 0.5(410 - 310) = 50 N$ **Ans.**

***(d) Power development by the turbine***

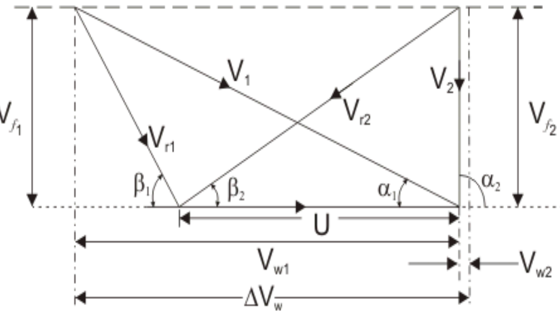
We know that power development by the turbine,

$P = m(V_w + V_{w1})\ V_b$

$= 0.5(1130 + 190)\ 375 = 247\ 500\ W$

$= 247.5\ kW$ **Ans.**

**The steam supply to an impulse turbine with a single row of moving blades is 2 kg/s. The turbine develops 130 kW, the blade velocity being 175 m/s. The steam flows from a nozzle with a velocity of 400 m/s and the velocity coefficient of blades is 0.9. Find the nozzle angle blade angle at entry and exit, if the steam flows axially after passing over the blades.**




***Given :***

***$m=2$ kg/s; $P=130kW = 130 \times 10^3$V***
***$V_b = 175$ m/s; $V=400$ m/s; $K=0.9$***

***Solution**:*

Let

We know that power development (*P*),

$130 \times 10^3 = m \times V_w \times V_b = 2 \times V_w \times 175 = 350$ V

$\therefore$ $V_w = 371.4$m/s

By measurement, we find that $\alpha = 19^0$; $\theta = 33^o$ and **$\phi$= 36º Ans.**

**Explain the procedure to draw Velocity Diagram for Two Stage Impulse Turbine.**

First of all, draw a horizontal line and cut off AB equal blade velocity ($V_h$) to

1. Now draw the inlet velocity triangle ABC for the first moving ring on the base AB with the help of nozzle angle of the first moving ring ($\alpha$) and verily of steam entering the turbine (v)
2. Now cut off CX equal to the friction of the bladed on the first moving ring. The length AX will give the value of relative velocity at exit of the first moving ring (V1)
3. Now draw the outlet velocity triangle ABD for first moving ring on the same base AB with the help of exit blade angle for the first moving ring and relative velocity at the first ring (V1)
4. Now cut off DY equal to the friction of the blades of the fixed ring. The length BY will give the exit velocity of steam from the fixed ring. It will also be equal to the velocity of steam entering the second moving ring (V1)
5. Now draw the inlet velocity triangle ABC 'for the second moving ring on the same base AB with the help of nozzle angle of the second moving ring ($\alpha$ ) and velocity of steam entering the second moving ring (V')
6. Now cut off C'Z equal to the friction of blades on the second moving ring. The length AZ will give the value of relative velocity at exit of the seconding moving ring (V'r).
7. Now draw the outlet velocity triangle ABC' for the second moving ring on the same base AB with the help of exit blade angle for the seconding moving ring( $\Phi$ ) and exit velocity of the second moving ring (V2).

We know that power developed by a two stage impulse turbine,

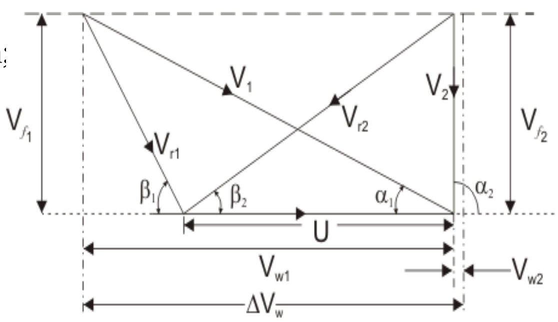
$P=m(EF+E'F)V_h$ Watts.

Where m is the mass of steam supplied in kg/s.

**In a reaction turbine, the blade tips are inclined at 35º and 20º in direction of motion. The guide blades are of the same shape as the moving blades, but reversed in direction. At a certain place in the turbine, the drum diameter is I meter and the blades are 100 mm high. At this place, steam has a pressure of 1.7 bar and dryness 0.935. If the speed of the turbine is 250 r.p.m and the steam passes through the blades without shock, find the mass of steam flow and the power developed in the ring of the moving blades.**

***Given** :*

$\theta = \beta = 35^o$ ; $\phi = \alpha = 20^o$; d = 1m; h=100mm = 0.1m;
p = 1.7 bar; $x$ = 0.935; N = 250 r.p.m




***Solution:***

We know that blade speed,

$$V_b = \frac{\pi(d+h)N}{60} = \frac{\pi(1+0.1)250}{60} = 14.4 m/s$$

Now let us draw the combined velocity triangle, as shown in Fig.23.9, as discusses below;

1. First of all, draw a horizontal line and cut off AB equal to 14.4 m/s to some suitable scale to represent velocity of blade ($V_b$)

2. Now draw velocity triangle ABC on the same base AB with $\alpha = 20^o$ and $\theta = 35^o$.

3. Similarly draw outlet velocity triangle ABD on the same base AB with ; $\phi = 20^o$ and $\beta = 35^o$.

4. From C and D draw perpendiculars to meet the line AB produced at E and F.

By measurement from velocity triangle, we find that

Change in the velocity of whirl, $(V_w + V_{w1})$ = EF = 42.5 m/s

and velocity of flow at outlet, $V_{f1}$ = DF = 10 m/s

**Mass of steam flow**

From steam tables, corresponding to a pressure of 1.7 bar, we find that the specific volume of steam, $\upsilon = 1.031 m^3/kg$.

We know that mass of steam flow,

$$m = \frac{\pi(d+h)hV_{f1}}{x\upsilon_g} = \frac{\pi(1+0.1)0.1\times 10}{0.935\times 1.031} = 3.58 kg/s \qquad \textbf{Ans.}$$

**Power developed in the ring of the moving blades**

We know that power developed in the ring of the moving blades,

$P = m (V_w + V_{w1})V_b$ = 3.58 x 42.5 x 14.4 = 2191 W

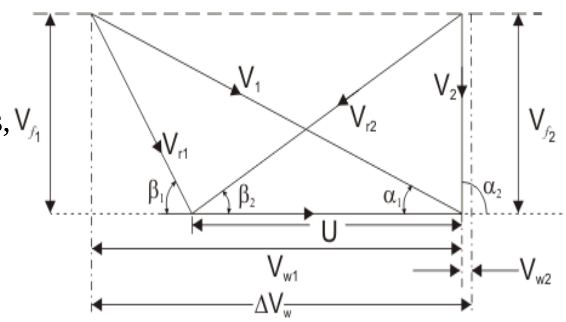
**= 2.191 kW** **Ans.**

**A reaction turbine runs at 300 r.p.m. and its steam consumption is 15 400 kg/h. The pressure of steam at a certain pair is 1.9 bar; its dryness 0.93 and power developed by the pairs is 3.5 kW. The discharging blade tip angle is 20° for both fixed and moving blades and the axial velocity of flow is 0.72 of the blade velocity. Find the drum diameter and blade height. Take the tip leakage steam as 8%, but neglect blade thickness**

***Given :***

N = 300 r.p.m. ; $m_1$ = 15 400 kg/h = 4.28 kg/s ;
p = 1.9bar ; $x$ = 0.93;
P=3.5kW = 3.5 x $10^3$W; $\alpha = \phi = 20°$ ; $V_f = 0.72\ V_b$

***Solution :***




Since the tip leakage steam is 8 %,
therefore actual mass of steam flowing over the blades,

m = 4.28 – (4.28 x 0.08) = 3.94 kg/s

***Blade height***

h = Blade height, and

$d_m$ = Mean

We know that blade velocity,

$$V_b = \frac{\pi d_m N}{60} = \frac{\pi d_m \times 300}{60} = 15.71 d_m m/s$$

∴ $V_f = 0.72 \times 15.71\ d_m = 11.3 d_m$ $11.3 d_m$ m/s

Now let us draw the combined velocity triangle, as shown in Fig.23.10, as discussed below

1. First of all, draw a horizontal line and cut off AB equal to 15.71 $d_m$ to some suitable scale to represent the blade velocity ($V_b$)

2. Now draw velocity triangle ABC on the base AB with $\alpha = 20°$ and BC = $V_f/\sin 20° = 11.3 d_m / 0.342 = 33 d_m$ to the scale.

3. Similarly draw outlet velocity triangle on the same base AB with ; $\phi = 20°$ and Vr1 = $V_{f1}/\sin 20° = 11.3 d_m/0.342 = 33 d_m$, to the scale.

4. From C and D draw perpendiculars to meet the line AB produced at E and F.

By measurement from velocity triangle, we find that change in the velocity of whirl,

$(V_w + W_{w1}) = 46 d_m$ m/s

We know that power developed ($P$),

$3.5 \times 10^3 = m(V_w + V_{w1})V_b = 3.94 \times 46\ d_m \times 15.71 d_m = 2845\ d_m^2$

∴ $d_m^2 = 1.23$ *or* $d_m = 1.11$ m

and $V_{f1} = V_f = 11.3\ d_m = 11.3 \times 1.11 = 12.54$ m/s

From steam tables, corresponding to a pressure of 1.9 bar, we find specific volume of steam,

$\upsilon_g = 0.929 m^3/kg$

We know that mass of steam flow ($m$),

$$3.94 = \frac{\pi d_m h V_{f1}}{x \upsilon_g} = \frac{\pi \times 1.11 \times h \times 12.54}{0.9. \times 0.929} = 50.6h$$

$\therefore$ $h = 0.078$ m = 78 mm Ans.

***Drum dialeter***

We know that drum diameter,

$d = d_m - h = 1.11 - 0.78 = \mathbf{1.032\ m}$ **Ans.**

**17. Derive the equation for critical pressure ratio for steam nozzles.**

$W_{c,\text{with CV}}$ = Area 1234

$$= \left(\frac{n}{n-1}\right)(p_1V_1)\left[\left(\frac{p_2}{p_1}\right)^{\left(\frac{n-1}{n}\right)} - 1\right] - \left(\frac{n}{n-1}\right)(p_4V_4)\left[\left(\frac{p_3}{p_4}\right)^{\frac{n-1}{n}} - 1\right]$$

Here $P_1 = P_4$, $P_2 = P_3$

$$= \left(\frac{n}{n-1}\right)(p_1V_1)\left[\left(\frac{p_2}{p_1}\right)^{\left(\frac{n-1}{n}\right)} - 1\right] - \left(\frac{n}{n-1}\right)(p_1V_4)\left[\left(\frac{p_2}{p_1}\right)^{\frac{n-1}{n}} - 1\right]$$

$$= \left(\frac{n}{n-1}\right)(p_1)\left[\left(\frac{p_2}{p_1}\right)^{\left(\frac{n-1}{n}\right)} - 1\right](V_1 - V_4)$$

**18. Ina single stage impulse turbine the blade angles are equal and nozzles angle is $20^0$. The velocity coefficient for the blade is 0.83. Find the maximum blades efficiency possible. If the actual blade efficiency is 90% of maximum blade efficiency, find the possible ratio of blade speed to steam speed.**

**(e) Blade inlet angle**

By measurement from the velocity diagram, we find that the blade angle at inlet,

$\theta = 29^0$

**(f) During force on the wheel**

We know that driving force on the wheel,

$F_x = m(V_w + V_{w1}) = 0.5\ (1130 + 190) = 660$ N

**(g) Arial thrust on the wheel**

We know that axial thrust on the wheel,

$$F_\gamma = m\left(V_f - V_{f1}\right) = 0.5(410 - 310) = 50N$$

**19. A 50 % reaction turbine(with symmetrical velocity triangles running at 400 rpm has the exit angle of the blades as $20^0$ and the velocity of steam relative to the blades at the exit is 1.35 times the mean blade speed. The steam flow rate is 8.33 kg/ s and at a particular stage the specific volume is 1.381 $m^3$/kg. Calculate for this stage: a suitable blade height , assuming the rotor mean diameter to be 12 times the blade height.**

**Blade inlet angle**

By measurement from the velocity diagram, we find that the blade angle at inlet,

$\theta = 29^0$

**During force on the wheel**

We know that driving force on the wheel,

$F_x = m(V_w + V_{w1}) = 0.5\ (1130 + 190) = 660$ N

**Arial thrust on the wheel**

We know that axial thrust on the wheel,

$$F_\gamma = m(V_f - V_{f1}) = 0.5(410 - 310) = 50N$$

**Power development by the turbine**

We know that power development by the turbine,

$P = m(V_w + V_{w1})\ V_b$

$= 0.5(1130 + 190)\ 375 = 247\ 500$ W

$= 247.5$ kW

**EXPECTED QUESTIONS**

**1. In a test on a steam nozzle the issuing steam jet impinges on stationary flat plate which is perpendicular to the direction of flow and the force on the plate is measured. With convergent divergent nozzle supplied with steam at 10 bar dry saturated and discharging at 1 bar: the force is experimentally measured to be 600N. the area of the nozzle at throat measures 5cm$^2$ and the exit area is such that complete expansion is achieved under these conditions. Determine the flow rate of the steam and the efficiency of the nozzle assuming that all losses occur after the throat. Assume n = 1.135 for isentropic expansion.**

Heat drop between entrance and exit,

$$h_{d3} = h_1 - h_3 = 2775 - 2465 = 310 \text{kJ/kg}$$

∴ Velocity of steam at throat,

$$V_3 = 44.72\sqrt{h_{d3}} = 44.72\sqrt{310} = 787.4 m/s$$

and $$m = \frac{A_2 V_2}{x_2 \upsilon_{g2}}$$

or $$A_3 = \frac{m x_3 \upsilon_{g3}}{V_3} = \frac{m \times 0.902 \times 1.159}{787.4} = 0.001.33m$$

∴ Ratio of cross-sectional area at exit and throat,

$$\frac{A_3}{A_2} = \frac{0.00133m}{0.000786m} = 1.7$$

**2. A 50 % reaction turbine(with symmetrical velocity triangles running at 400 rpm has the exit angle of the blades as 20$^0$ and the velocity of steam relative to the blades at the exit is 1.35 times the mean blade speed. The steam flow rate is 8.33 kg/ s and at a particular stage the specific volume is 1.381 m$^3$/kg. Calculate for this stage: a suitable blade height , assuming the rotor mean diameter to be 12 times the blade height.**

**Blade inlet angle**

By measurement from the velocity diagram, we find that the blade angle at inlet,

$\theta = 29^0$

**During force on the wheel**

We know that driving force on the wheel,

$F_x = m(V_w + V_{w1}) = 0.5\ (1130 + 190) = 660$ N

**Arial thrust on the wheel**

We know that axial thrust on the wheel,

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**Power development by the turbine**

We know that power development by the turbine,

$P = m(V_w + V_{w1})\ V_b$

$= 0.5(1130 + 190)\ 375 = 247\ 500$ W

$= 247.5$ kW

**3. A convergent nozzle required to discharge 2kg of steam per second. The nozzle is supplied with steam at 7 bar and 180$^0$C and discharge takes place againsta back pressure of 1 bar. The expansion up to throat is isentropic and the frictional resistance between the throat and exit is equivalent to 63 kJ/kg of steam.Take approach velocity of 75 m/s and throat pressure 4 bar, estimate**

**i) suitable areas for the throat and exit,and**

**ii) Overall efficiency of the nozzle based on enthalpy drop between the actual inlet pressure and temperature and the exit pressure.**

***Final velocity of the steam***

From steam tables, corresponding to a pressure of 15 bar, we find that enthalpy of dry saturated steam,

$h_1 = 2789.9$ kJ/Kg

and corresponding to a pressure of 1.5 bar, enthalpy of dry saturated steam,

$h_2 = 2693.4$ kJ/Kg

$\therefore$ Heat drop, $h_d = h_1 - h_2 = 2789.9 - 2693.4 = 96.5$ kJ/Kg

We know that final velocity of the steam,

$$V_2 = 44.72\ \sqrt{h_d} = 44.72\sqrt{96.5} = 439.3 m/s$$

**4. In a stage of impulse reaction turbine operating with 50% degree of reaction, the blades are identical in shape. The outlet angle of the moving blade is 19$^0$ and the absolute discharge velocity of steam is 100m/s in the direction70$^0$Cto the motion of blades. If the rate of flow through the turbine is 15000 kg/hr, calculate the power developed by the turbine.**

We know that heat drop lost in friction

$= 10\% = 01.$

$\therefore$ Nozzle coefficient or nozzle efficiency

$K = 1 - 01 = 0.9$

We know that final velocity of the steam,

$$V_2 = 44.72\ \sqrt{Kh_d} = 44.72\sqrt{0.9 \times 96.5} = 416.8 m/s$$

$\therefore$ Percentage reduction in final velocity

$$= \frac{439.3 - 416.8}{439.3} = 0.051 \; or \; 5.1\%$$

**27. In a De Lavel turbine steam issues from the nozzle with a velocity of 1200m/s. The nozzle angle is $20^0$, the mean velocity is 400m/s and the inlet and outlet angles of blades are equal. The mass of steamflowing through the turbine per hour is 1000kg. Calculate:**

**i) Blade angles**

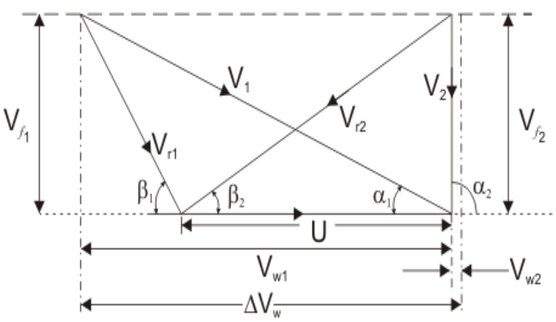
**ii) Relative velocity of steam entering the blades**

**iii) Tangential force on the blades**

**iv) Power developed**

**v) Blade efficiency. Take blade velocity coefficient as 0.8**

***Solution :***




Since the tip leakage steam is 8 %,
therefore actual mass of steam flowing over the blades,

m = 4.28 – (4.28 x 0.08) = 3.94 kg/s

***Blade height***

h = Blade height, and

$d_m$ = Mean

We know that blade velocity,

$$V_b = \frac{\pi d_m N}{60} = \frac{\pi d_m \times 300}{60} = 15.71 d_m m/s$$

$\therefore$ $V_f$ = 0.72 x 15.71 $d_m$ = 11.3$d_m$ 11.3$d_m$ m/s

**Brayton Cycle**

The Brayton cycle (first proposed by George Brayton, 1870) is the air standard cycle for gas turbine plant.

The various operations are as follows:

**Operation 1-2.** The air is compressed isentropically from the lower pressure $p_1$ to the upper pressure $p_2$, the temperature rising from $T_1$ to $T_2$. No heat flow occurs.

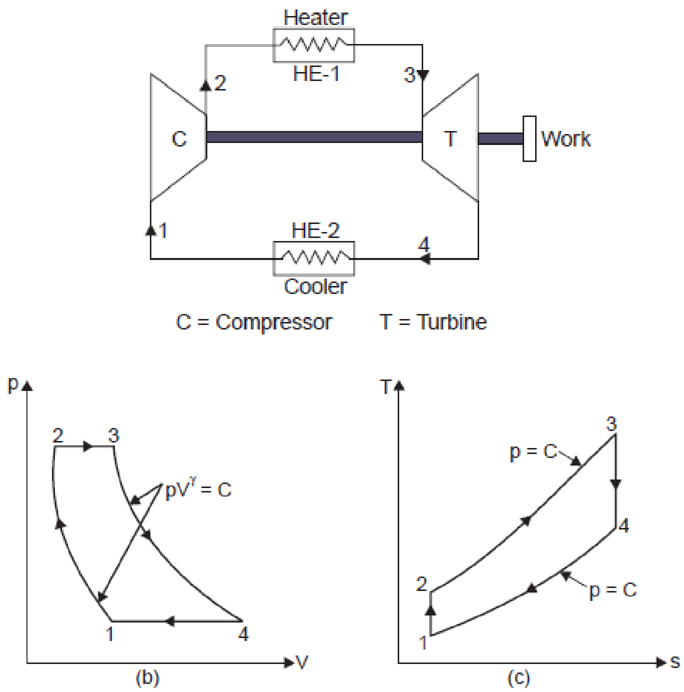
**Operation 2-3.** Heat flows into the system increasing the volume from $V_2$ to $V_3$ and temperature from $T_2$ to $T_3$ whilst the pressure remains constant at $p_2$.

Heat received = $mc_p$ $(T_3 - T_2)$.

**Operation 3-4.** The air is expanded isentropically from $p_2$ to $p_1$, the temperature falling from $T_3$ to $T_4$. No heat flow occurs.

**Operation 4-1.** Heat is rejected from the system as the volume decreases from $V_4$ to $V_1$ and the temperature from $T_4$ to $T_1$ whilst the pressure remains constant at $p_1$.

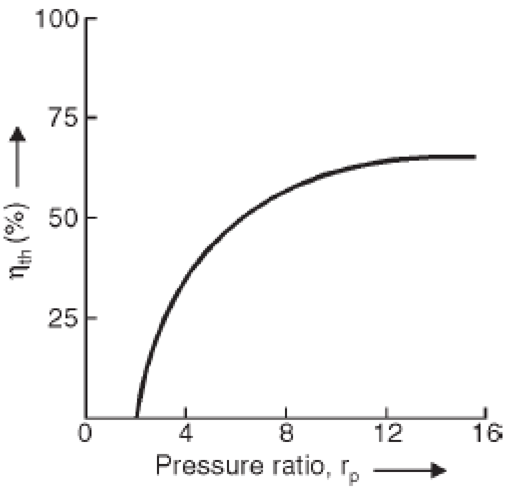
Heat rejected = $mc_p(T_4 - T_1)$.




The efficiency of Brayton cycle

$$\eta_B = \frac{W_{net}}{Q_s} = \frac{Q_s - Q_r}{Q_s} = 1 - \frac{Q_r}{Q_s} = 1 - \frac{mc_p(T_4 - T_1)}{mc_p(T_3 - T_2)} = 1 - \frac{(T_4 - T_1))}{(T_3 - T_2))}$$

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




The efficiency of the ideal joule cycle increases with the pressure ratio. The absolute limit of upper pressure is determined by the limiting temperature of the material of the turbine at the point at which this temperature is reached by the compression process alone, no further heating of the gas in the combustion chamber would be permissible and the work of expansion would ideally just balance the work of compression so that no excess work would be available for external use.

**Work output during the Brayton cycle**

W = Heat received/cycle – heat rejected/cycle

$$W = mc_p (T_3 - T_2) - mc_p (T_4 - T_1)$$

$$W = mc_p (T_3 - T_4) - mc_p (T_2 - T_1)$$

$$W = mc_p\left[T_3\left(1 - \frac{T_4}{T_3}\right) - T_1\left(\frac{T_2}{T_1} - 1\right)\right]$$

For isentropic processes, $\dfrac{T_3}{T_4} = \dfrac{T_2}{T_1} = r_p^{\left(\frac{\gamma-1}{\gamma}\right)} = r_p^z$

where $z = \dfrac{\gamma - 1}{\gamma}$

$$W = mc_p\left[T_3\left(1 - \frac{1}{r_p^z}\right) - T_1\left(r_p^z - 1\right)\right]$$

For maximum work output, $\dfrac{dW}{dr_p} = 0;$ gives $\dfrac{T_3}{T_1} = r_p^{2z} = r_p^{2\left(\frac{\gamma-1}{\gamma}\right)}$, $r_p = \left(\dfrac{T_3}{T_1}\right)^{\frac{\gamma}{2(\gamma-1)}}$

The work output per unit mass of gas

$$W = c_p (T_3 - T_4) - c_p (T_2 - T_1) = c_p (T_3 - T_4 - T_2 + T_1)$$

$$W = c_p\left(T_3 - \frac{T_1T_3}{T_2} - T_2 + T_1\right)$$

For maximum work output $\frac{dW}{dT_2} = 0$, gives $T_2 = \sqrt{T_1T_3}$

Therefore $W_{max} = c_p\left(\sqrt{T_3} - \sqrt{T_1}\right)^2$

Maximum efficiency of Brayton cycle $\eta_{max} = 1 - \frac{1}{\left(r_p\right)^{\frac{\gamma-1}{\gamma}}} = 1 - \sqrt{\frac{T_1}{T_3}}$

**Effect of friction in turbine and compressor on Brayton cycle**

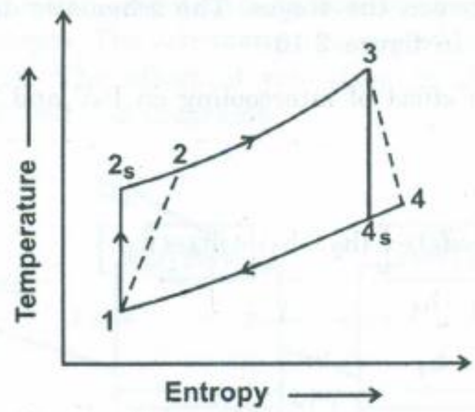



Fig. 2.15: Effect of turbine and compressor efficiency on Brayton cycle

Heat supplied $Qs = mc_p(T_3 - T_2)$

Heat rejected $Qr = mc_p(T_4 - T_1)$

Turbine work $W_T = mc_p(T_3 - T_4)$

Compressor work $W_C = mc_p(T_2 - T_1)$

Efficiency of the cycle $\eta = \frac{W_T - W_C}{Qs}$

Turbine efficiency

$$\eta_T = \frac{W_{act}}{W_{isen}} = \left(\frac{T_3 - T_4}{T_3 - T_{4s}}\right)$$

Compressor efficiency

$$\eta c = \frac{W_{isen}}{W_{act}} = \frac{T_{2s} - T_1}{T_2 - T_1}$$

**Brayton cycle with intercooling**

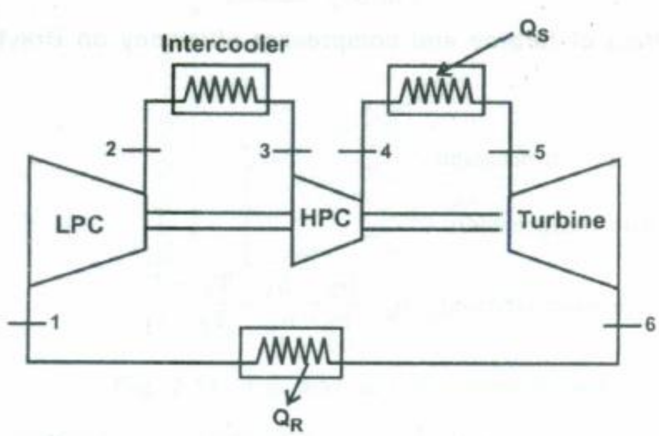



Fig. 2.16: Brayton cycle with intercooling.

Heat supplied $Qs = mc_p(T_5 - T_4)$ Heat rejected $Qr = mc_p(T_6 - T_1)$

Turbine work $W_T = mc_p(T_5 - T_6)$ Compressor work $W_C = mc_p[(T_2 - T_1) + (T_4 - T_3)]$

Efficiency of the cycle $\eta = \dfrac{W_T - W_C}{Qs}$

**Brayton cycle with reheating**

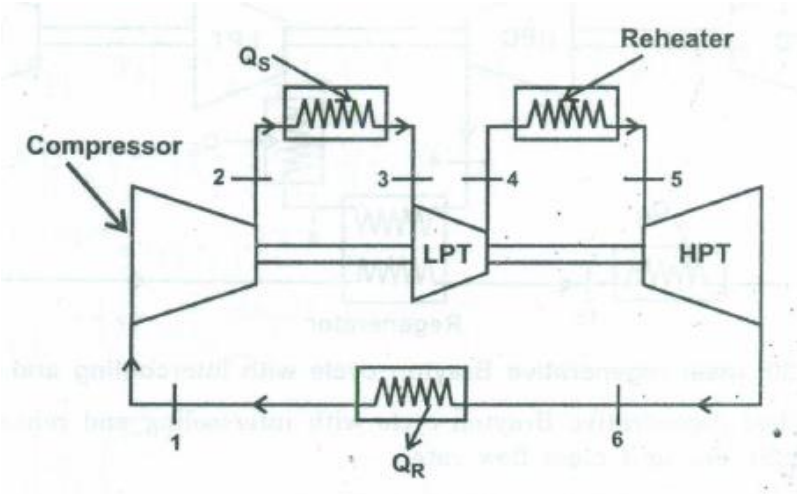



Fig. 2.18: Brayton cycle with reheating.

Heat supplied $Qs = mc_p[(T_3 - T_2) + (T_5 - T_4)]$ Heat rejected $Qr = mc_p(T_6 - T_1)$

Turbine work $W_T = mc_p[(T_3 - T_4) + (T_5 - T_6)]$ Compressor work $W_C = mc_p(T_2 - T_1)$

Efficiency of the cycle $\eta = \dfrac{W_T - W_C}{Qs}$

**Brayton cycle with regeneration**

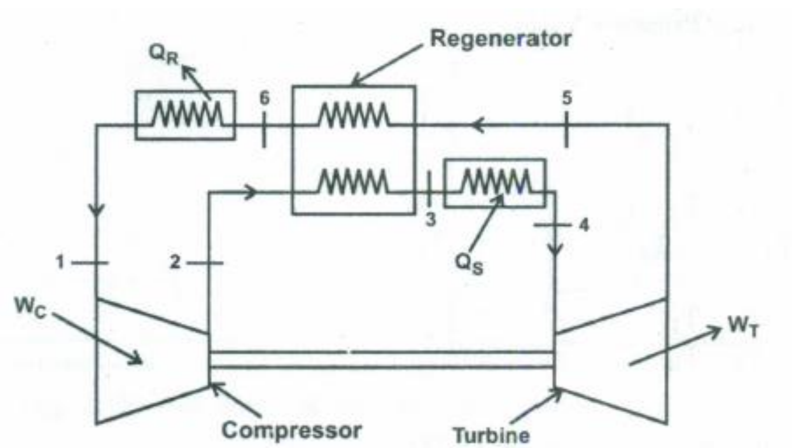



Fig. 2.13: Brayton cycle with regeneration

Heat supplied $Qs = mc_p(T_4 - T_3)$ Heat rejected $Qr = mc_p(T_6 - T_1)$

Turbine work $W_T = mc_p(T_4 - T_5)$ Compressor work $W_C = mc_p(T_2 - T_1)$

Efficiency of the cycle $\eta = \dfrac{W_T - W_C}{Qs}$

**1. What is compounding?**

Compounding is the method in which multiple system or rotors are keyed to common shaft in series and the steam pressure or jet velocity is absorbed in stages as it flows over the rotor blades.

**2.Explain the purpose of compounding?**

Reduction of pressure (from boiler pressure to condenser pressure) in single results in the very high velocity entering the turbine blades. Therefore, the turbine rotor will run at a high speed about 30,000 rpm which is not useful for practical purpose. In order to reduce the rotor speed up to about 400 m/sec, compounding of steam turbine is necessary.

**3. What are the types of compounding in steam turbines?**

Velocity compounding, Pressure compounding and Pressure velocity compounding.

**4. What are the advantages of velocity compounded turbines?**

**Advantages:**

- The cost of turbine is less because less number of stages.
- It occupies less area.
- The system is reliable and easy to operate.
- Turbine casing is very simple and need not be very strong.

**5. What are the disadvantages of velocity compounded turbines?**

**Disadvantages:**

- The friction losses are large due to very high steam velocity in the nozzle.
- Low efficiency because blade speed ratio is less than the optimum value.
- The power developed in the later rows is only a fraction of power developed in the first row.

**6. What is governing of steam turbine and state the various methods of governing?**

Governing is the method of maintaining the constant speed of the turbine irrespective of load variation by varying the flow rate. The various methods of governing in steam turbines are Throttle governing, Nozzle control governing, By pass governing, Combinations of throttle and nozzle governing and Combinations of throttle and by pass governing.

**7. What is the fundamental difference between the operation of impulse and reaction steam turbines?**

Ø In impulse turbine, the steam completely expands in the nozzle and its pressure remains constant during its flow through the rotor blades.

Ø In reaction turbine, the steam expands partially in the nozzle and remaining in rotor blades.

**8. Differentiate Impulse and Reaction Turbine.**

1. An impulse turbine works due to change in kinetic energy of the fluid flowing through the rotor when the pressure remains constant. Reaction turbine is one in which the decreases gradually & Kinetic energy is increased, when the steam flows over a set of fixed and moving blades
2. Less number of stages are required for same output power. More number of stages are required for same output power.
3. It is suitable for small power requirements It is suitable for medium and high power requirements.
4. Blade passage is of constant cross sectional area. Blade passage is of converging type.
5. Blades shapes are profile type. Blade shapes are aerofoil type.
6. Steam is admitted over the part of then circumference of the wheel. Steam is admitted over the entire circumference of the wheel.
7. Blade manufacture is easy Blade manufacture is difficult.

8. Steam fully expands in nozzle. Steam expands in both nozzle and moving blades.
9. Flow can be regulated without loss Flow cannot be regulated without loss

**9. Differentiate between nozzle governing and throttle governing.**

| Sl. No. Throttle governing | Nozzle governing |
|---|---|
| 1. More throttling losses occur. | Throttling losses are negligible |
| 2. Partial admission losses are low | Partial admission losses are high |
| 3. Less heat drop is available for work | More heat drop is available for work |
| 4. Employed for both impulse and reaction turbines | Employed only in impulse turbines |
| 5. Less efficient method | More efficient method |
| 6. Suitable for small turbines | Suitable for medium and large turbines |

**10.Enumerate the energy loses in steam turbines.**

*losses in regulating valves
*losses due to steam friction
*losses due to mechanical friction
*losses due to leakage
*Residual velocity losses

**11.List out some internal losses in steam turbines.**

*Losses due to friction between fluid layers
*losses due to entry and exit of flow at the turbines
*losses due to rotation of fluid particles
*losses due to blade surface roughness.

**12.What is blading efficiency?**

Blade efficiency is defined as the ratio between workdone on the blade and enery supplied to the blade

$$n_b = \frac{\text{workdone on the blade}}{\text{Energy supplied to the blade}}$$

**13. Define the term stage efficiency in case of reaction turbines.**

It is the ratio of work done on blades per kg of steam to the enthalpy drop in the nozzle.

**14.Write down the expression for velocity at exit from steam nozzle.**

Exit velocity$V_2^2$=2000($h_1$-$h_2$)

**15. What is a steam turbine?**

Steam turbine is device which is used to convert kinetic energy of steam into mechanical energy

**16. How does impulse work?**

The high velocity jet of steam which is obtained from the nozzle impinges on the blades fixed on a rotor.The blades change the direction of the steam flow without changing its pressure,This causes change in momentum and the force developed drive the turbine rotor.

**17. State the function of fixed blades.**

The function of fixed blades is that they guide the steam as well as allow it to expand a larger velocity

**18.State the function of moving blades.**

*It converts the kinetic energy of the steam into usefull mechanical energy

*The velocity of steam decreases as the kinetic energy of the steam absorbed.

**19.What are the different methods of governing steam turbines?**

*Throttle governing
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*By-pass governing

**20.How is throttle governing done?**

Steam pressure at inlet to a steam turbine is reduced by throttling process to a maintain the speed of the speed of the turbine constant at part is load

**21.Where nozzle control governing is used?**

It is used in large power steam turbines to which very high pressure steam is supplied.

**22.Where by-pass governing more suitable?**

It is most suitable for reaction turbine and a single by-pass valve

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**23.What are the basic requirements to be considered for the selection gas turbine working fluid?**

a.Environmental sustainability
b.ozone depletion potential
c.global warming potential
d.non flammable
e.non toxic
f.non corrosive
g.thermal stability
h.critical temperature

**24.Why Brayton cycle is more suitable than otto cycle for gas turbine plants?**

Brayton cycles operates based on pressure ratio.Otto cycle operates based on compression ratio.so brayton cycle is more suitable than Otto cycle for gas turbine plants.

**25.What fuel does a gas turbine use?**

The following fuels are used in gas turbines

i.petrol
ii.Diesel
iii.kirosene
iv.Natural gas.

**26.What is the effect of reheat on the brayton cycle efficiency.why?**

The thermal efficiency of brayton cycle is increased.Because when reheating the cycle pressure again increased significantly.So it will increase the efficiency.

**27.How are gas turbine classified?**

1.Based on cycle (Brayton.Erickson)
2.based on Fuel (petrol,Diesel,natural gas)
3.based on application(Air craft,Marine,locomotive)

**28.Depict the influence of pressure ratio on the efficiency of a Brayton cycle.**

$n_{brayton}=1-1/r_p{}^{\gamma-1/\gamma}$

if pressure ratio increases in the above eqn,the efficiency of the gas turbinr also inceases.

**29. Name the various vapour power cycle.**

Carnot cycle and Rankine cycle.

**30. Define efficiency ratio.**

The ratio of actual cycle efficiency to that of the ideal cycle efficiency is termed as efficiency ratio.

**31. Define overall efficiency.**

It is the ratio of the mechanical work to the energy supplied in the fuel. It is also defined as the product of combustion efficiency and the cycle efficiency.

**32. What are the effects of condenser pressure on the Rankine Cycle?**

By lowering the condenser pressure, we can increase the cycle efficiency. The main disadvantage is lowering the back pressure in rease the wetness of steam. Isentropic compression of a very wet vapour is very difficult.

**34. Mention the improvements made to increase the ideal efficiency of Rankine cycle.**

1. Lowering the condenser pressure.
2. Superheated steam is supplied to the turbine.
3. Increasing the boiler pressure to certain limit.
4. Implementing reheat and regeneration in the cycle.

**35. Why reheat cycle is not used for low boiler pressure?**

At the low reheat pressure the heat cycle efficiency may be less than the Rankine cycle efficiency. Since the average temperature during heating will then be low.

**36. What are the disadvantages of reheating?**

Reheating increases the condenser capacity due to increased dryness fraction, increases the cost of the plant due to the reheats and its very long connections.

**37. What are the advantages of reheat cycle?**

1. It increases the turbine work.
2. It increases the heat supply.
3. It increases the efficiency of the plant.
4. It reduces the wear on the blade because of low moisture content in LP state of the turbine.

**38.Define Relative efficiency.**

It is defined as the ratio of actual efficiency to the air standard efficiency

**39. Define degree of super saturation.**

The ratio of super saturation pressures corresponding to the temperature between super saturated region is known as the degree of super saturation.

**40. Define blade efficiency or utilization factor.**

It is the ratio of rotor blade work to energy supplied to the rotor.

**41. Define degree of reaction.**

It is defined as the ratio of the actual isentropic heat drop to the total heat drop in the entire stage.

**19. What is compounding and explain the purpose of compounding?**

Compounding is the method in which multiple system or rotors are keyed to common shaft in series and the steam pressure or jet velocity is absorbed in stages as it flows over the rotor blades.

**Purpose of compounding:** Reduction of pressure (from boiler pressure to condenser pressure) in single results in the very high velocity entering the turbine blades. Therefore, the turbine rotor will run at a high speed about 30,000 rpm which is not useful for practical purpose. In order to reduce the rotor speed up to about 400 m/sec, compounding of steam turbine is necessary.

**20. What are the types of compounding in steam turbines?**

Velocity compounding, Pressure compounding and Pressure velocity compounding.

**21. What are the advantages and disadvantages of velocity compounded turbines?**

**Advantages:**

Ø The cost of turbine is less because less number of stages.
Ø It occupies less area.
Ø The system is reliable and easy to operate.
Ø Turbine casing is very simple and need not be very strong.

**Disadvantages:**

Ø The friction losses are large due to very high steam velocity in the nozzle.
Ø Low efficiency because blade speed ratio is less than the optimum value.
Ø The power developed in the later rows is only a fraction of power developed in the first row.

**22. What is governing of steam turbine and state the various methods of governing?**

Governing is the method of maintaining the constant speed of the turbine irrespective of load variation by varying the flow rate. The various methods of governing in steam turbines are Throttle governing, Nozzle control governing, By pass governing, Combinations of throttle and nozzle governing and Combinations of throttle and by pass governing.

**23. What is the fundamental difference between the operation of impulse and reaction steam turbines?**

Ø In impulse turbine, the steam completely expands in the nozzle and its pressure remains constant during its flow through the rotor blades.
Ø In reaction turbine, the steam expands partially in the nozzle and remaining in rotor blades.

**24. Differentiate Impulse and Reaction Turbine. Sl. No. Impulse Turbine Reaction Turbine**

1. An impulse turbine works due to change in kinetic energy of the fluid flowing through the rotor when the pressure remains constant. Reaction turbine is one in which the decreases gradually & Kinetic energy is increased, when the steam flows over a set of fixed and moving blades
2. Less number of stages are required for same output power. More number of stages are required for same output power.
3. It is suitable for small power requirements It is suitable for medium and high power requirements.
4. Blade passage is of constant cross sectional area. Blade passage is of converging type.
5. Blades shapes are profile type. Blade shapes are aerofoil type.
6. Steam is admitted over the part of then circumference of the wheel. Steam is admitted over the entire circumference of the wheel.
7. Blade manufacture is easy Blade manufacture is difficult.
8. Steam fully expands in nozzle. Steam expands in both nozzle and moving blades.
9. Flow can be regulated without loss Flow cannot be regulated without loss

**25. Differentiate between nozzle governing and throttle governing.**

| Sl. No. Throttle governing | Nozzle governing |
|---|---|
| 1. More throttling losses occur. | Throttling losses are negligible |
| 2. Partial admission losses are low | Partial admission losses are high |
| 3. Less heat drop is available for work | More heat drop is available for work |
| 4. Employed for both impulse and reaction turbines | Employed only in impulse turbines |
| 5. Less efficient method | More efficient method |
| 6. Suitable for small turbines | Suitable for medium and large turbines |

**26.Enumerate the energy loses in steam turbines.**

*losses in regulating valves
*losses due to steam friction
*losses due to mechanical friction
*losses due to leakage
*Residual velocity losses

**27.List out some internal losses in steam turbines.**

*Losses due to friction between fluid layers
*losses due to entry and exit of flow at the turbines
*losses due to rotation of fluid particles
*losses due to blade surface roughness.

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**30.Write down the expression for velocity at exit from steam nozzle.**

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**31.What is the critical pressure value for dry saturated steam?**

P2/p1=0.577

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$[p+(a/V2)](V-b) = RT$

**50.What are the assumptions made to derive ideal gas equation analytically using the kinetic theory of gases?**

The ideal gas equation $pV=mRT$ has two important assumptions,
1. There is little or no attraction between the molecules of the gas.
2. That the volume occupied by the molecules themselves is negligibly small compared to the volume of the gas. This equation holds good for low pressure and high temperature ranges as the intermolecular attraction and the volume of the molecules are not of much significance.

**43. Name the various gas power cycles**

Carnot cycle, Otto cycle, Diesel cycle, Dual cycle, Brayton cycle, Atkinson cycle, Stirling cycle,

**43. What are the effects of introducing regenerator in the basic gas turbine cycle?**

a. The fuel economy is improved. The quality of fuel required per unit mass of air is less
b. The work output from turbine, the work required to the compressor will not change.
c. Pressure drop will occur during regeneration
d. It increased thermal efficiency when the turbine operates at low-pressure ratio.

**44. When the reheater is employed in the gas turbine cycle?**

When air fuel ratio is high, the combustion products after expansion in the highpressure turbine contain more oxygen. This can be utilised in the reheater and the gas is further expanded in the low-pressure turbine.

**45. Compare the Diesel and Brayton cycles**

| **Diesel cycle** | **Brayton cycle** |
|---|---|
| 1. It consist of two isentropic, one constant volume and one constant pressure processes | 1. It consist of two isentropic, two constant pressure processes |
| 2. Heat is rejected at constant volume | 2. Heat is rejected at constant pressure |
| 3. Used in Diesel engines | 3. Used in gas turbines |

**46. Why Brayton cycle is used in gas turbine?**

Inside the turbine the gas is continuously flowing in the processes are flow processes. Since all the processes involved in Brayton cycle is flow process, it has been used as the cycle for gas turbine.

**47.Why intercooler is provided between two stage of compressor?**

Intercooler is provided to reduce the exit air temperature of first stage .Thereby the specific volume of air is reduced and work required to compress the air is reduced.

**48.For perfect intercooler ,what is the condition of intercooler outlet temperature?**

Intercooler outlet temperature must be equal to inlet temperature of fresh air which is fed into the first stage of compression.

**49.What is meant by regeneration?**

The temperature of exhaust gases of the turbine is higher than the temperature of the air after compression.If the heat energy is used to heat the air after compression in the heat exchanger called regeneration.It will reduce the energy requirement from the fuel therby increasing the efficiency of the cycle.

50 What are the effects of reheat cycle?

(i)Thermal efficiency is less .since the heat supplied is more
(ii)Turbine output is increased for the same expansion ratio.