Unit III

STEAM NOZZLES AND TURBINES

Flow of steam through nozzles, shapes of nozzles, effect of friction, critical pressure ratio,supersaturated flow.Impulse and reaction principles, compounding, velocity diagrams for simple and multistage turbines, speed regulations-governors and nozzle governors. Numerical Problems.

STEAM NOZZLES AND TURBINES

TECHNICAL TERMS:

1. Wet steam: The steam which contains some water particles in superposition.

2. Dry steam / dry saturated steam:

When whole mass of steam is converted into steam then it is called as dry steam.

3. Super heated steam: When the dry steam is further heated at constant pressure, the temperature increases the above saturation temperature. The steam has obtained is called super heated steam.

4. Degree of super heat: The difference between the temperature of saturated steam and saturated temperature is called degree of superheat.

5. Nozzle: It is a duct of varying cross sectional area in which the velocity increases with the corresponding drop in pressure.

6. Coefficient of nozzle: It is the ratio of actual enthalpy drop to isentropic enthalpy drop.

7. Critical pressure ratio: There is only one value of ratio (P2/P1) which produces maximum discharge from the nozzle . then the ratio is called critical pressure ratio.

8. Degree of reaction: It is defined as the ratio of isentropic heat drop in the moving blade to isentrpic heat drop in the entire stages of the reaction turbine.

9. Compounding: It is the method of absorbing the jet velocity in stages when the steam flows over moving blades. (i)Velocity compounding (ii)Pressure compounding and (iii) Velocity-pressure compounding

10. Enthalpy: It is the combination of the internal energy and the flow energy.

11. Entropy: It is the function of quantity of heat with respective to the temperature.

12. Convergent nozzle: The crossectional area of the duct decreases from inlet to the outlet side then it is called as convergent nozzle.

13.Divergent nozzle: The crossectional area of the duct increases from inlet to the outlet then it is called as divergent nozzle.

Flow of steam through nozzles:

The flow of steam through nozzles may be regarded as adiabatic expansion. - The steamhas a very high velocity at the end of the expansion, and the enthalpy decreases as expansion takes place. - Friction exists between the steam and the sides of the nozzle; heat is produced as the result of the resistance to the flow. - The phenomenon of super saturation occurs in the flow of steam through nozzles. This is due to the time lag in the condensation of the steam during the expansion.

Continuity and steady flow energy equations

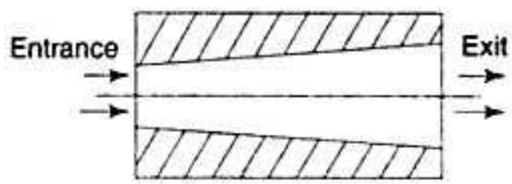
Through a certain section of the nozzle: m.v = A.C m is the mass flow rate, v is the specific volume, A is the cross-sectional area and C is the velocity. For steady flow of steam through a certain apparatus, principle of conservation of energy states:

$$h1 + C_1^2/2 + gz_1 + q = h_2 + C_2^2/2 + gz_2 + w$$

Types of Nozzles:
1. Convergent Nozzle
2. Divergent Nozzle
3. Convergent-Divergent Nozzle

Convergent Nozzle:
A typical convergent nozzle is shown in fig. in a convergent nozzle, the cross sectional area decreases continuously from its entrance to exit. It is used in a case where the back pressure is equal to or greater than the critical pressure ratio.

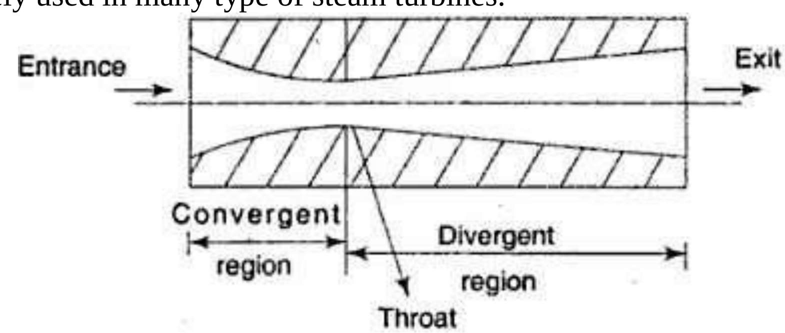
Divergent Nozzle:
The cross sectional area of divergent nozzle increases continuously from its entrance to exit. It is used in a case, where the back pressure is less than the critical pressure ratio.



Divergent Nozzle:

Convergent-Divergent Nozzle:

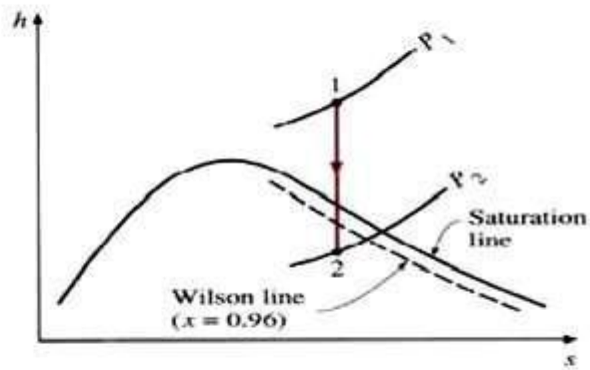
In this case, the cross sectional area first decreases from its entrance to throat, and then increases from throat to exit.it is widely used in many type of steam turbines.



Convergent-Divergent Nozzle

Supersaturated flow or Meta stable flow in Nozzles: As steam expands in the nozzle, its pressure and temperature drop, and it is expected that the steam start condensing when it strikes the saturation line. But this is not always the case. Owing to the high velocities, the residence time of the steam in the nozzle is small, and there may not sufficient time for the necessary heat transfer and the formation of liquid droplets. Consequently, the condensation of steam is delayed for a little while. This phenomenon is known as super saturation, and the steam that exists in the wet region without containing any liquid is known as supersaturated steam.

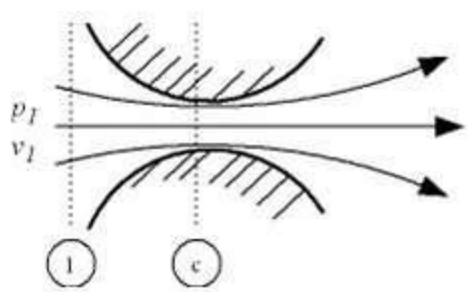
The locus of points where condensation will take place regardless of the initial temperature and pressure at the nozzle entrance is calledthe Wilson line. The Wilson line lies between 4 and 5 percent moisture curves in the saturation region on the h-s diagram for steam, and is often approximated by the 4 percent moisture line. The super saturation phenomenon is shown on the h-s chart below:



The h-s diagram for the isentropic expansion of steam in a nozzle.

Critical Pressure Ratio: The critical pressure ratio is the pressure ratio which will accelerate the flow to a velocity equal to the local velocity of sound in the fluid.

Critical flow nozzles are also called **sonic chokes**. By establishing a shock wave the sonic choke establish a fixed flow rate unaffected by the differential pressure, any fluctuations or changes in downstream pressure. A sonic choke may provide a simple way to regulate a gas flow.



Critical flow nozzles

Effect of Friction on Nozzles:

1) Entropy is increased.
2) Availableenergyisdecreased.
3) Velocity of flow at throat is decreased.
4) Volume of flowing steam is decreased.
5) Throat area necessary to discharge a given mass of steam is increased.

Most of the friction occurs in the diverging part of a convergent-divergent nozzle as the length of the converging part is very small. The effect of friction is to reduce the available enthalpy drop by about

10 to 15%. The velocity of steam will be then

$$V_2 = 44.72\sqrt{K(H_1 - H_2)}$$

Where, k is the co-efficient which allows for friction loss. It is also known as nozzle efficiency.

Velocity of Steam at Nozzle Exit:

$$V_2^2 = 2000(H_1 - H_2) + V_1^2 \quad \therefore \quad V_2 = \sqrt{2000(H_1 - H_2) + V_1^2}$$

As the velocity of steam entering the nozzle is very small, V_1 can be neglected.

$$\therefore \qquad V_2 = \sqrt{2000(H_1 - H_2)} = 44.72\sqrt{(H_1 - H_2)} \text{ m/s}$$

If frictional losses are taken into account then

$$V_2 = 44.72\sqrt{(H_1 - H_2)\eta_n} \text{ m/s}$$

3.5 Mass of steam discharged through nozzle:

$$m = A\sqrt{2000\frac{n}{n-1}\times\frac{P_1}{v_1}\left[\left(\frac{P_2}{P_1}\right)^{\frac{2}{n}} - \left(\frac{P_2}{P_1}\right)^{\frac{n+1}{n}}\right]}$$

Condition for maximum discharge through nozzle: The nozzle is always designed for maximum discharge

$$\frac{m}{A} = \sqrt{2000\frac{n}{n-1}\times\frac{P_1}{v_1}\left[\left(\frac{P_2}{P_1}\right)^{\frac{2}{n}} - \left(\frac{P_2}{P_1}\right)^{\frac{n+1}{n}}\right]}$$

The mass flow per unit area will be maximum at the throat because the throat area is minimum.

It is seen from the above equation that the discharge through a nozzle is a function of $\frac{P_2}{P_1}$ only, as the expansion index is fixed according to the steam supplied to the nozzle.

Therefore, $\frac{m}{A}$ is maximum when

$$\left[\left(\frac{P_2}{P_1}\right)^{\frac{2}{n}} - \left(\frac{P_2}{P_1}\right)^{\frac{n+1}{n}}\right] \text{ is minimum}$$

Values for maximum discharge:

$$m = A\sqrt{2000\frac{n}{n-1}\times\frac{P_1}{v_1}\left[\left(\frac{P_2}{P_1}\right)^{\frac{2}{n}}-\left(\frac{P_2}{P_1}\right)^{\frac{n+1}{n}}\right]}$$

we know $\dfrac{P_2}{P_1}=\left(\dfrac{2}{n+1}\right)^{\frac{n}{n-1}}$

Putting the value of $\dfrac{P_2}{P_1}$ in the above equation

$$m_{\max} = A\sqrt{2000\frac{n}{n-1}\times\frac{P_1}{v_1}\left[\left(\frac{2}{n+1}\right)^{\frac{2}{n-1}}-\left(\frac{2}{n+1}\right)^{\frac{n+1}{n-1}}\right]}$$

$$m_{\max} = A\sqrt{2000\frac{n}{n-1}\times\frac{P_1}{v_1}\times\left(\frac{2}{n+1}\right)^{\frac{n+1}{n-1}}\left[\left(\frac{2}{n+1}\right)^{\frac{2}{n-1}-\frac{n+1}{n-1}}-1\right]}$$

$$= A\sqrt{2000\frac{n}{n-1}\times\frac{P_1}{v_1}\times\left(\frac{2}{n+1}\right)^{\frac{n+1}{n-1}}\left[\left(\frac{2}{n+1}\right)^{\frac{1-n}{n-1}}-1\right]}$$

$$= A\sqrt{2000\frac{n}{n-1}\times\frac{P_1}{v_1}\times\left(\frac{2}{n+1}\right)^{\frac{n+1}{n-1}}\left[\left(\frac{2}{n+1}\right)^{-1}-1\right]}$$

$$= A\sqrt{2000\frac{n}{n-1}\times\frac{P_1}{v_1}\times\left(\frac{2}{n+1}\right)^{\frac{n+1}{n-1}}\left[\frac{n+1}{2}-1\right]}$$

$$= A\sqrt{2000\frac{n}{n-1}\times\frac{P_1}{v_1}\times\left(\frac{2}{n+1}\right)^{\frac{n+1}{n-1}}\left(\frac{n-1}{2}\right)}$$

$$m_{\max} = A\sqrt{1000n\times\frac{P_1}{v_1}\times\left(\frac{2}{n+1}\right)^{\frac{n+1}{n-1}}}$$

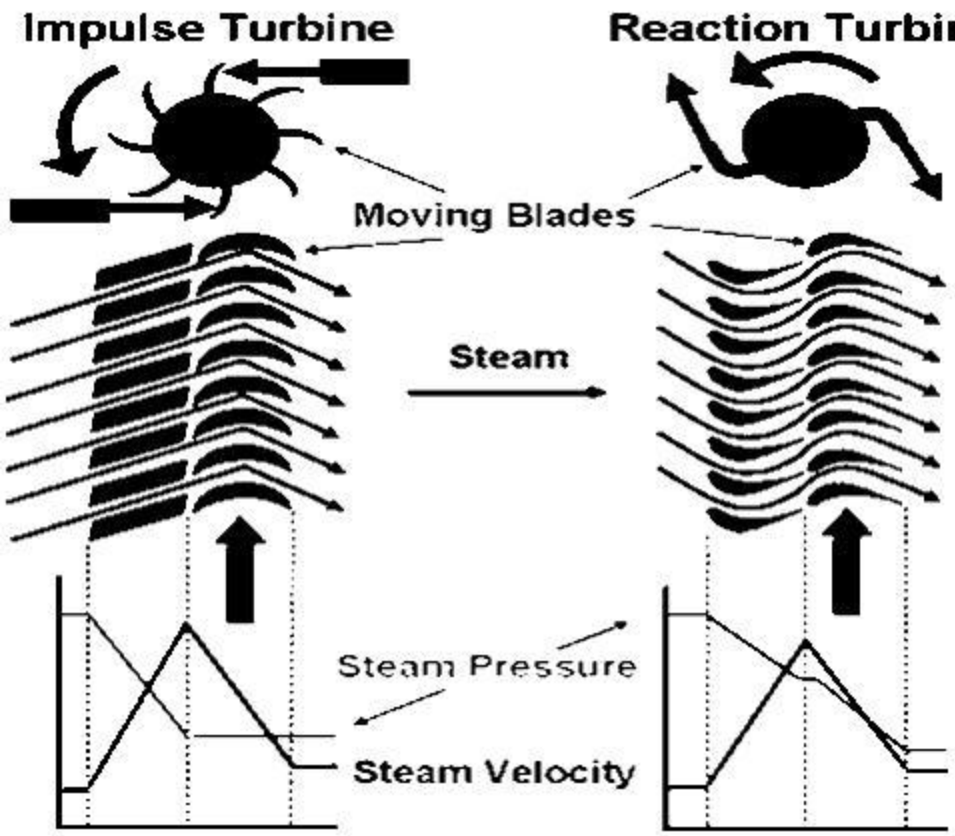
Where P1 is the initial pressure of the steam in kpa and v1 is the specific volume of the steam in m3/kg at the initial pressure.

3.6 STEAM TURBINES: 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

3.6.1 Impulse Turbines:

The steam jets are directed at the turbines bucket shaped 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

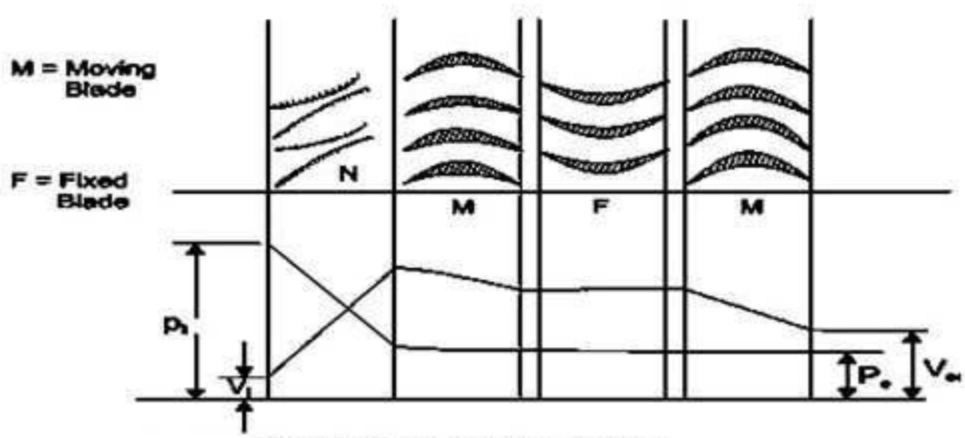
3.6.2 Reaction Turbines

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).

3.7 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.

3.7.1. Velocity Compounding:



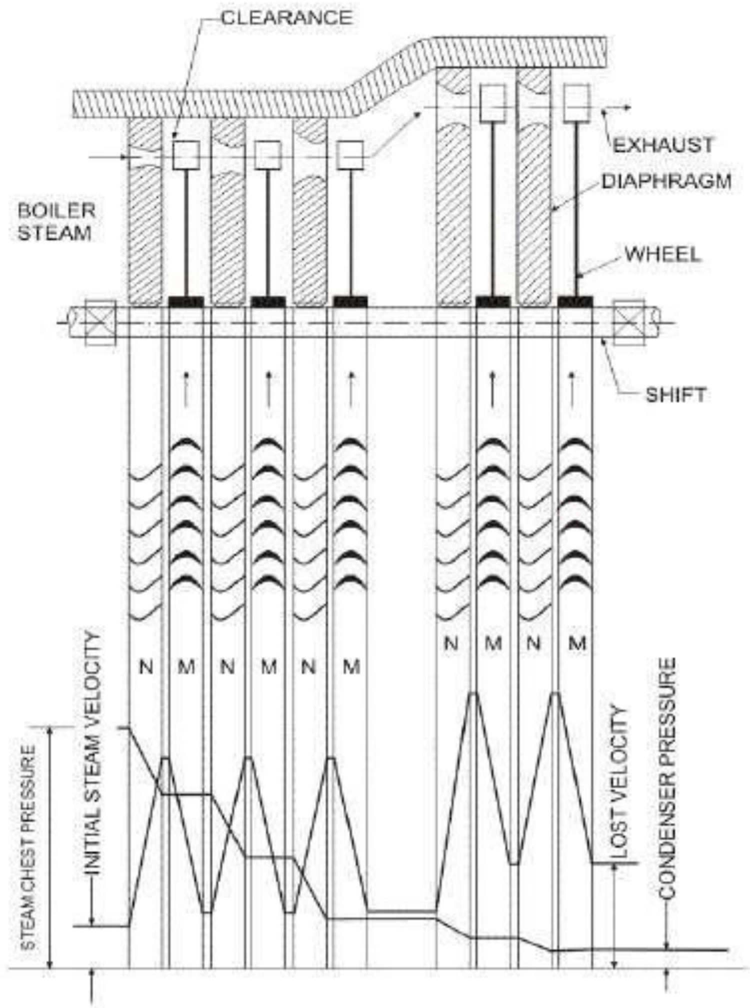
Velocity Compounding diagram

Pi = Inlet Pressure,   Pe= Exit Pressure,   Vi =Inlet Velocity,   Ve=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.

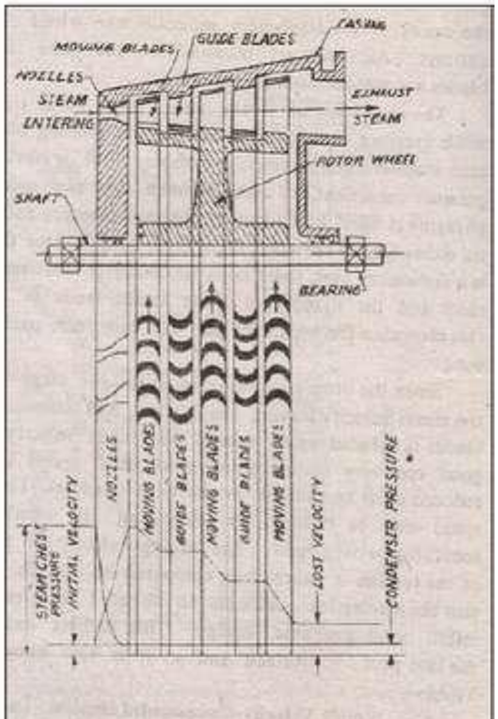
3.7.2. Pressure Compounding:

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.



3.7.3**. Pressure-Velocity Compounding**

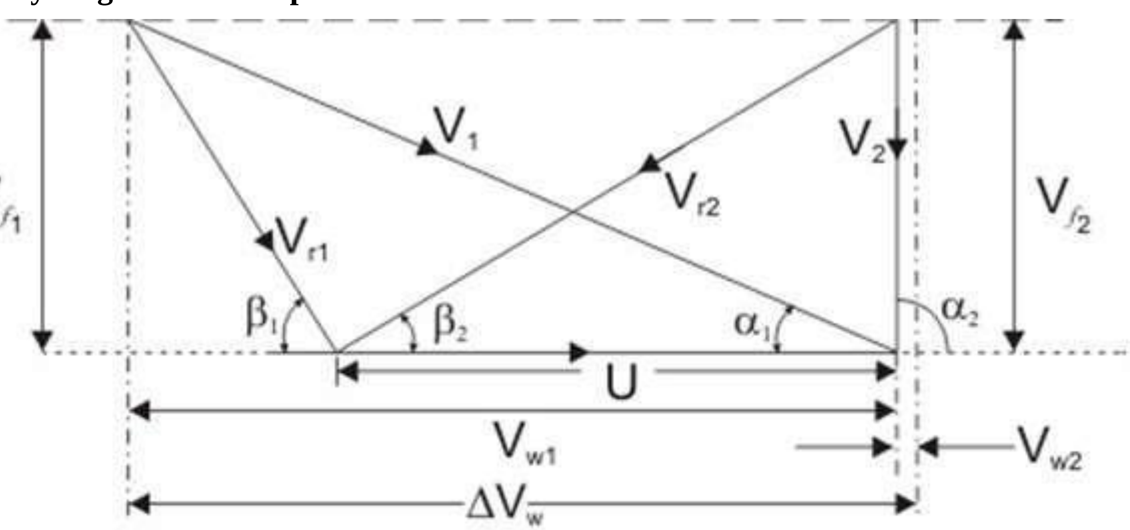
This is a combination of pressure and velocity compounding. A two-row velocity compounded turbine is found to be more efficient than the three-row type. In a two-step pressure velocity compounded turbine, the first pressure drop occurs in the first set of nozzles, the resulting gain in the kinetic energy 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.

3.8 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

α_1 = nozzle angle, α_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)

β_1 and β_2 = Inlet and outlet **blade angles**

Vw_1 and Vw_2 = 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 $= \eta_s = \dfrac{\textit{Work done by the rotor}}{\textit{Isentropic enthalpy drop}}$

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

or,

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

Optimum blade speed of a single stage turbine

$$\Delta V_w = V_{r1}\cos\beta_1 + V_{r2}\cos\beta_2$$

$$= V_{r1}\cos\beta_1 + \left(1 + \frac{V_{r2}}{V_{r1}}\cdot\frac{\cos\beta_2}{\cos\beta_1}\right)$$

$$= (V_1\cos\alpha_1 - U) + (1 + kc)$$

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

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

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

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

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

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

or, $\rho = \dfrac{\cos\alpha_1}{2}$

α_1 is of the order of 18^0 to 22^0

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

$\therefore$ 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.

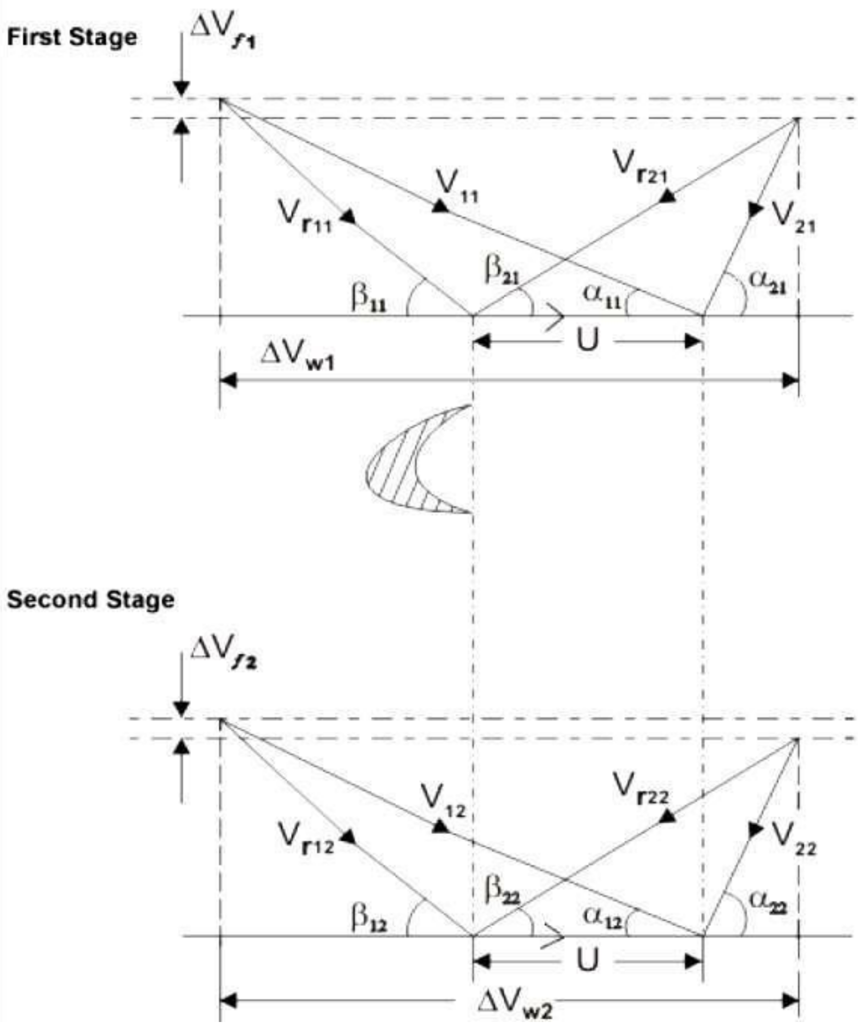
K, the blade velocity coefficient may be different in each row of blades

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

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

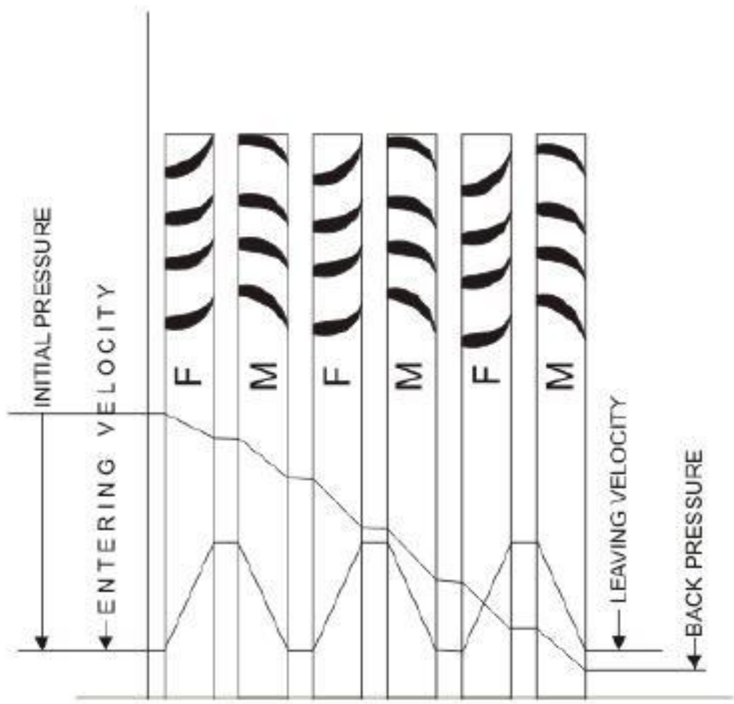
The optimum velocity ratio will depend on number of stages and is given by $P_{opt} = \dfrac{\cos\alpha_{11}}{2n}$

Velocity diagram of the velocity compounded turbines:



Reaction Turbine:

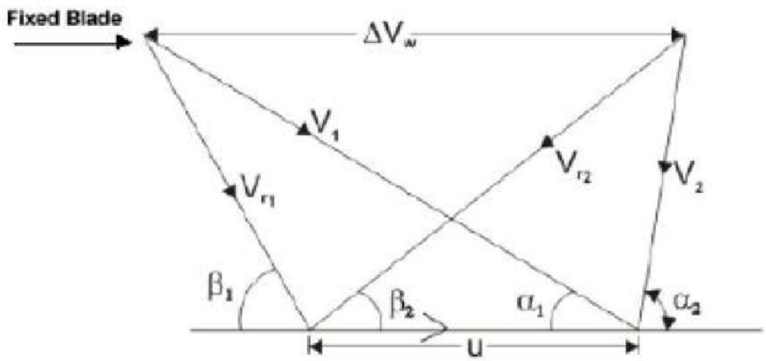
A *reaction turbine*, therefore, is one that is constructed of rows of fixed and rows of moving blades. The fixed blades act as nozzles. The moving blades move as a result of the impulse of steam received (caused by change in momentum) and also as a result of expansion and acceleration of the steam relative 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**

$$\textbf{Degree of Reaction} = \frac{\textit{Enthalpy drop in Rotor}}{\textit{Enthalpy drop in Stage}}$$

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



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,

1. Throttle governing
2. Nozzle governing

3. By-pass governing

3.10.1. Throttle Governing:

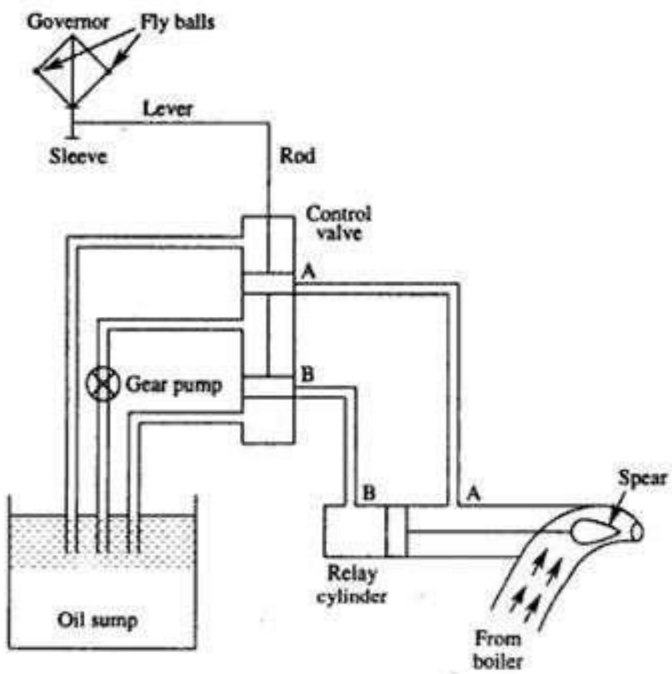


Fig 3.14 Throttle Governing

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.

Nozzle Governing:

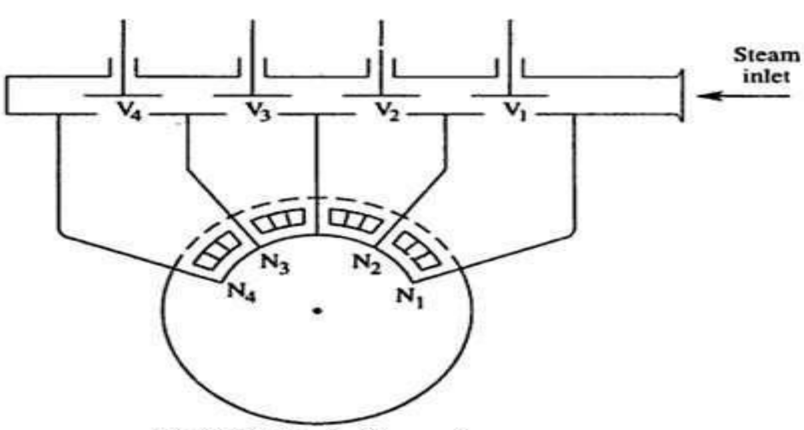


Fig 3.15 Nozzle Governing

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.