LECTURE NOTES ON

POWER PLANT ENGINEERING

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MODULE – III

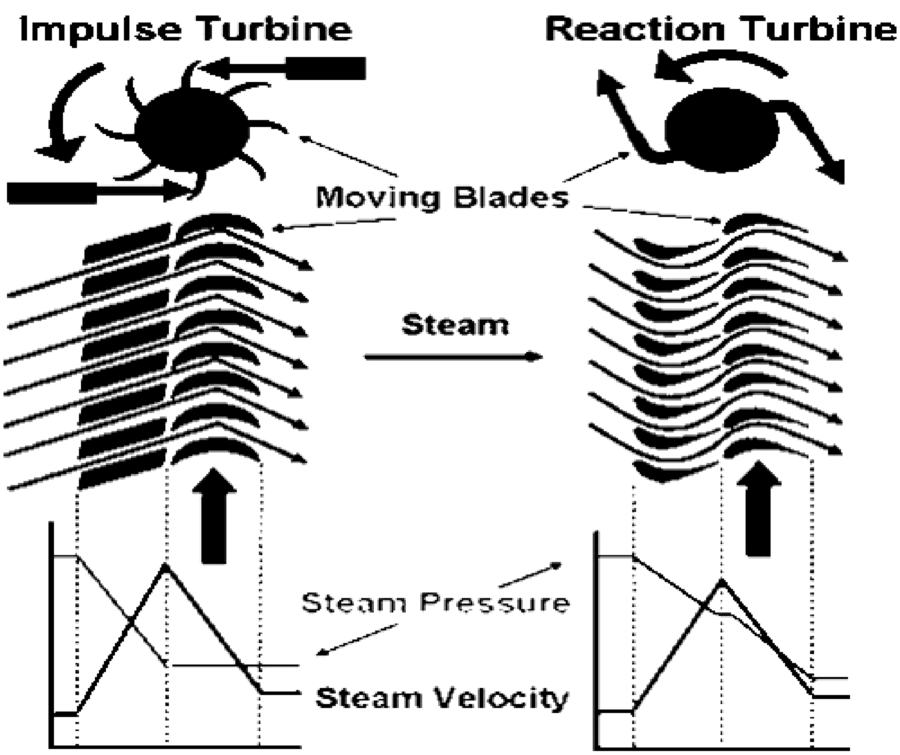
TURBINES

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

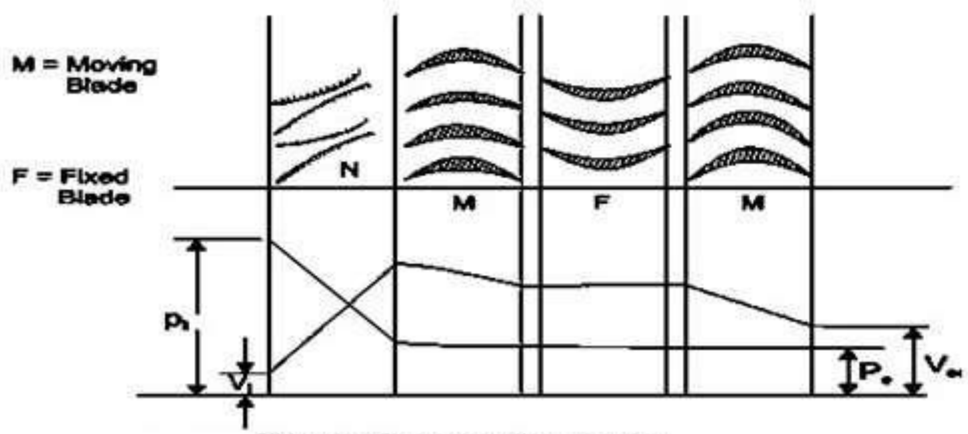
3.5.2 Reaction Turbines

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:



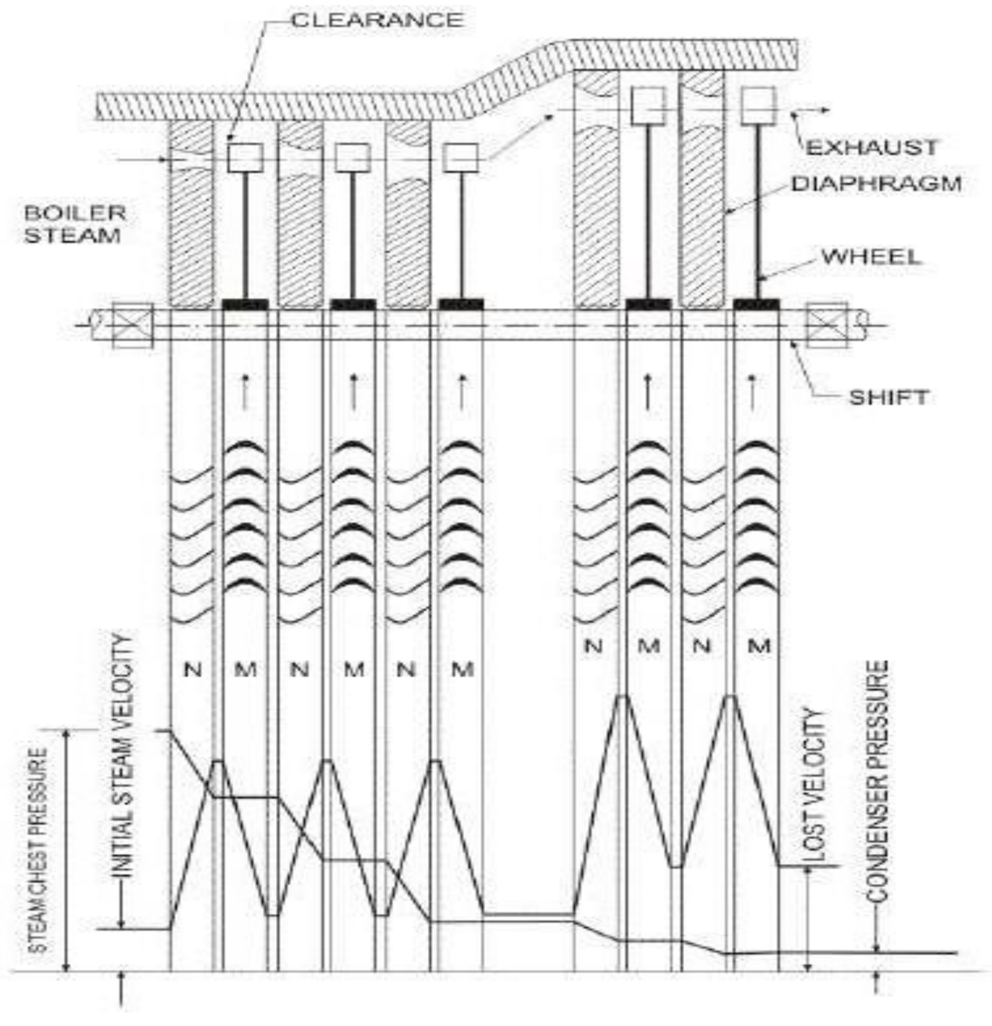
Velocity Compounding diagram

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.

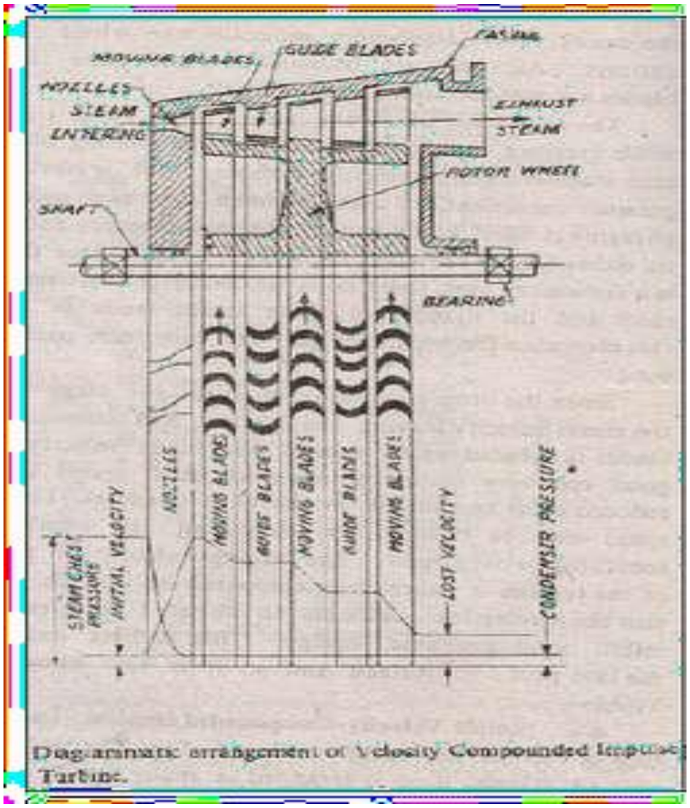
Pressure Compounding:

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

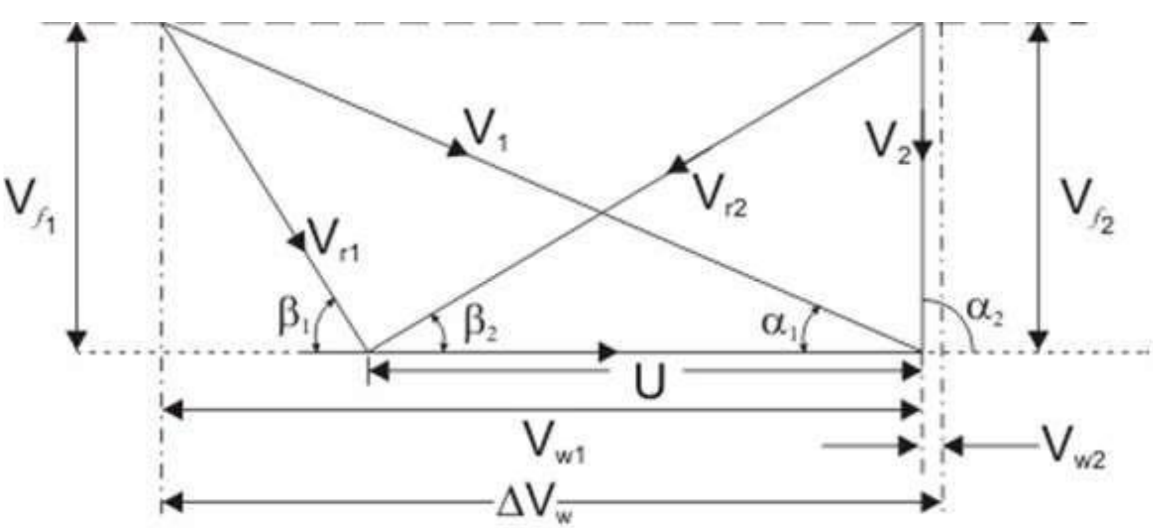
This is a combination of pressure and velocity compounding. A two-row velocity compounded turbine is found to be more efficient than the three-row type. In a two-step pressure velocity compounded turbine, the first pressure drop occurs in the first set of nozzles, the resulting gain in the kinetic energy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

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**

V_{w_1} and V_{w_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

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

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

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

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

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

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

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

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

α_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.

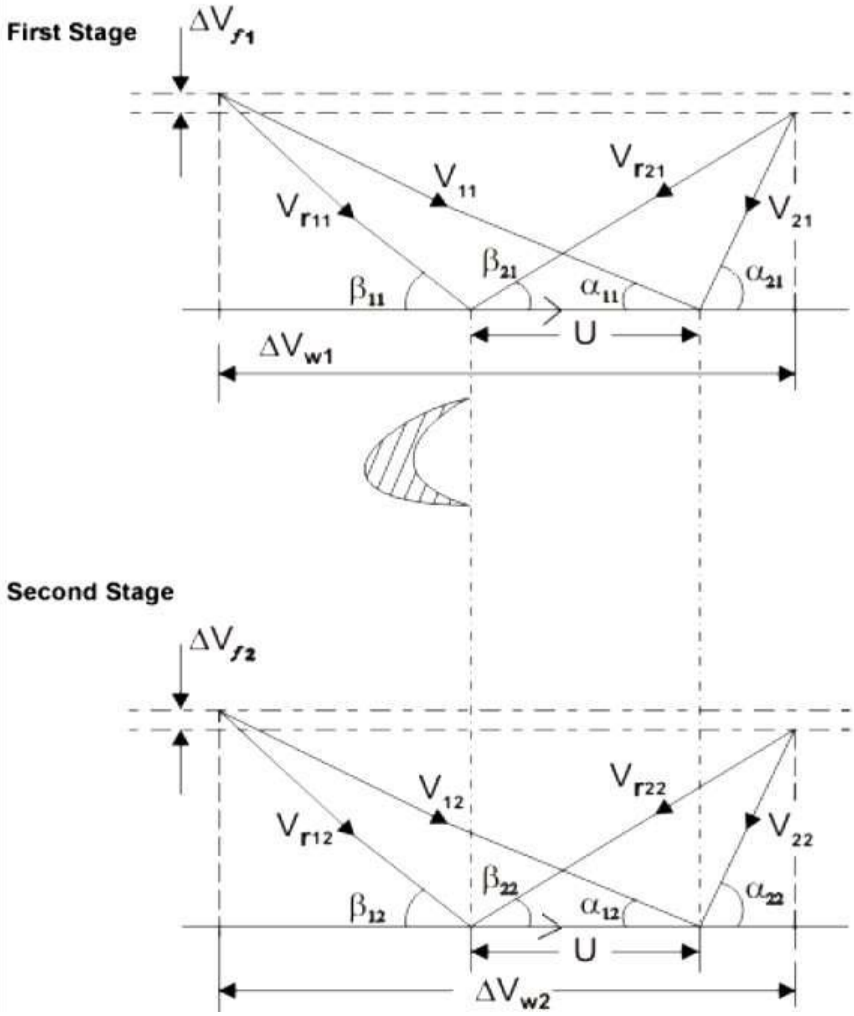
K, the blade velocity coefficient may be different in each row of blades

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

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

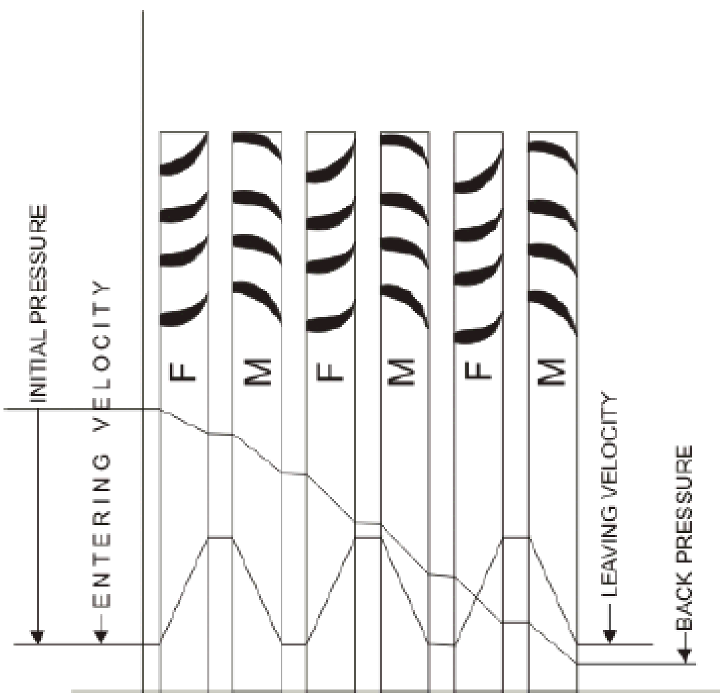
The optimum velocity ratio will depend on number of stages and is given by $P_{opt} = \frac{\cos\alpha_{11}}{2n}$

Velocity diagram of the velocity compounded turbines:



Reaction Turbine:

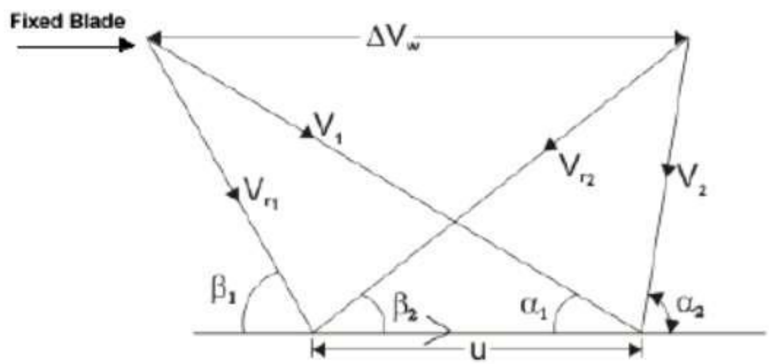
A *reaction turbine*, therefore, is one that is constructed of rows of fixed and rows of moving blades. The fixed blades act as nozzles. The moving blades move as a result of the impulse of steam received (caused by change in momentum) and also as a result of expansion and acceleration of the steam relative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

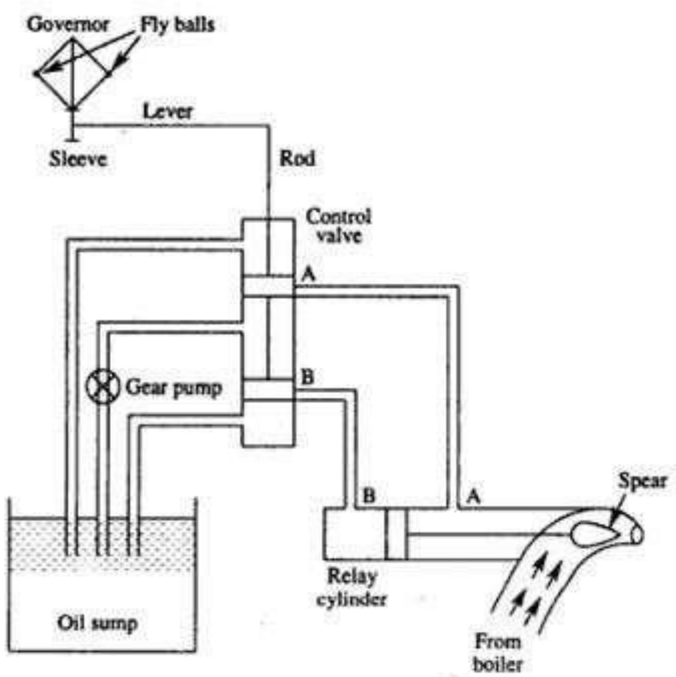


Fig 3.14 Throttle Governing

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:

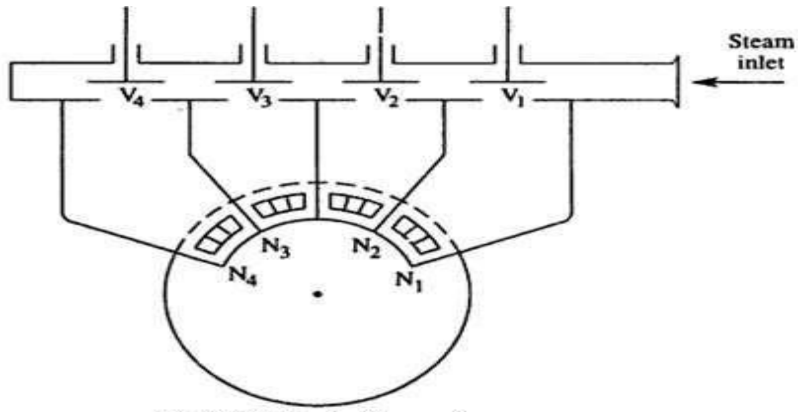


Fig 3.15 Nozzle Governing

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

$s_1 = s_{f1} + x_1 \text{ x } s_{fg}$

= 2.159 +0.95 x 4.407 = 6.34565 kJ/kgk

$h_1 = h_{f1} + x_1 \text{ x } h_{fg1}$

= 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

$s_t = s_{ft} + x_t \text{ x } s_{fgt}$

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

η_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)

Θ=43°

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

from triangle ABD,

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

Φ=35°39·

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

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

=195.28 m/sec

$v_f = v_2 = 195.28 m/sec$

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

= 1482.18kW.

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 25^0C 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 Δ 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 Δ ACE
$\tan\theta = V_{f1} / \underline{V_{w1}}- V_b = 253.57/ [543.79- 250]$

$$v_{r1} = \sqrt{[253.57^{\,2}} + (543.79 - 250)^2$$
=388.09 m/ sec
$v_{r2} = 0.9 \times v_{r1} = 0.9 \times 388.09 = 349.28$ m/sec
From Δ 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 $W = m(v_{w1}+v_{w2})\ v_b$

$W = 1(543.79+52.49)\times250 = 149.07$ KW/kg.

Diagram efficiency $\eta_D = \dfrac{(vw1+vw2)\ vb}{mv_1^2/2}$
$= \dfrac{149.07\times1000}{1\times0.5\times600^2} = 82.82\%$

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 ×d

Solution ;

According to sine rule

ΔABC

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

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

= 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,

$V_{sup} = 0.73$ m³/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 h²

h²=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)×60 = 214650W

=214.65 kW.

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

= 214.65/0.85 =252.52 kJ/sec.