Mechanical Engineering

Power Plant Engineering

Short Notes

IMPORTANT FORMULAS TO REMEMBER

CHAPTER 1: RANKINE CYCLE

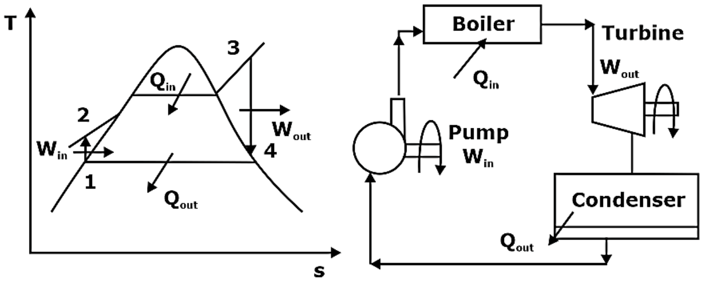
1. RANKINE CYCLE: - Rankine cycle has 4 major components.

(i) Heat addition in boiler,

(ii) Steam expansion in turbine,

(iii) Condensation of saturated steam in condenser,

(iv) Compression of saturated liquid by pump.



The ideal Rankine cycle

Pump	Q=0	$W_{pump,in} = h_2 - h_1$
Boiler	W=0	$Q_{in} = h_3 - h_1$
Turbine	Q=0	$W_{turbine,out} = h_3 - h_4$
Condenser	W=0	$Q_{out} = h_4 - h_1$

- **Efficiency:** $\eta_R = \dfrac{\text{net work output}}{\text{heat supplied to the boiler}} = \dfrac{(h_3 - h_4) - (h_2 - h_1)}{(h_3 - h_2)}$

- **Specific steam consumption (SSC):** It is the mass flow rate of steam, required to produce 1kW of power.

$$\text{ssc} = \frac{\dot{m}_f}{P} = \frac{3600}{W_{net}} \text{ in kg / kWhr} \quad \text{where, } W_{net} = (h_3 - h_4) - (h_2 - h_1) \text{ in kJ / kg.}$$

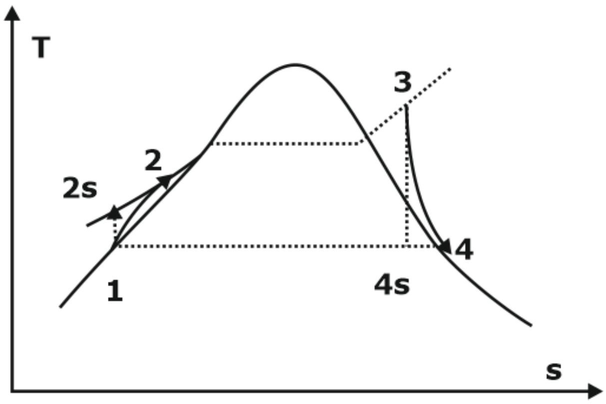
- **Heat rate (H.R.):** It is the amount of heat required to produce 1kW of power output.

$$\text{H.R.} = \frac{\text{heat supplied(kW)}}{\text{power developed(kW)}} = \frac{1}{\eta}$$

- A steam power plant system consists of many components like, boiler, turbine, different auxiliaries, generator etc. So, the overall efficiency of steam power plant can be written as:

$$\eta_{overall} = \eta_{boiler} \times \eta_{cycle} \times \eta_{mechnical} \times \eta_{generator} \times \eta_{auxiliaries}$$

- Due to irreversibility in the turbine and pump, process in them is not isentropic, so to find actual turbine and pump work isentropic efficiency is used.



Isentropic efficiency of pump, $\eta_{pump} = \dfrac{\text{isentropic work}}{\text{actual work}} = \dfrac{(h_{2S} - h_1)}{(h_2 - h_1)}$

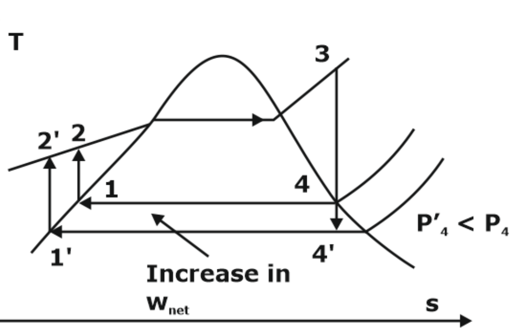
Isentropic efficiency of turbine, $\eta_{turbine} = \dfrac{\text{actual work}}{\text{isentropic work}} = \dfrac{(h_3 - h_4)}{(h_3 - h_{4s})}$

1.1. Methods of increasing the efficiency of Rankine cycle:

1.1.1. Decreasing the Condenser Pressure

By decreasing the condenser pressure, mean temperature of heat rejection in the condenser decreases. Thus, the thermal efficiency of the cycle will be increased.

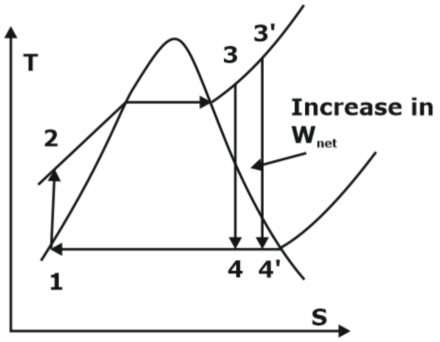
$$\eta \propto 1 - \frac{T_L}{T_H}$$



Effect of lowering condenser pressure on ideal Rankine cycle

1.1.2. Superheating the Steam to High Temperature

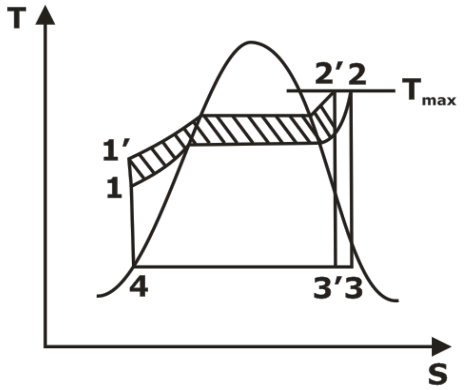
Superheating the steam will increase the network output and the efficiency of the cycle. It also decreases the moisture contents of the steam at the turbine exit.



The effect of superheating on the ideal Rankine cycle.

1.1.3. Increasing the Boiler Pressure

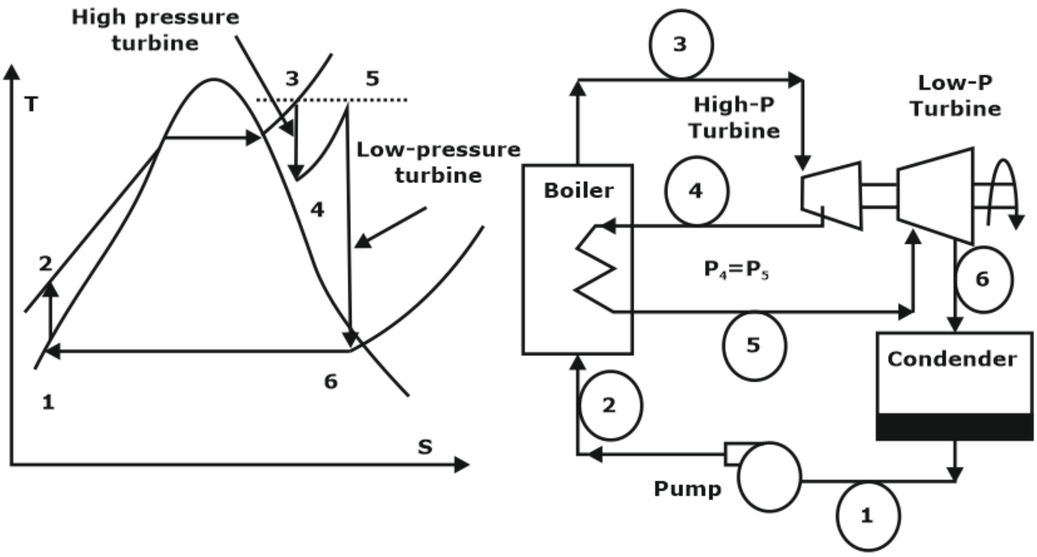
On increasing the boiler pressure, the mean temperature of heat addition increases hence thermal efficiency increases.



Increasing the boiler Pressure

1.2. IDEAL REHEAT RANKINE CYCLE

To take advantage of the increased efficiencies at higher boiler pressure without facing the excessive moisture content at final stages of the turbine, reheating is used. In this process steam is expended in more than one turbine by providing heat between two turbines.



The ideal reheat Rankine cycle

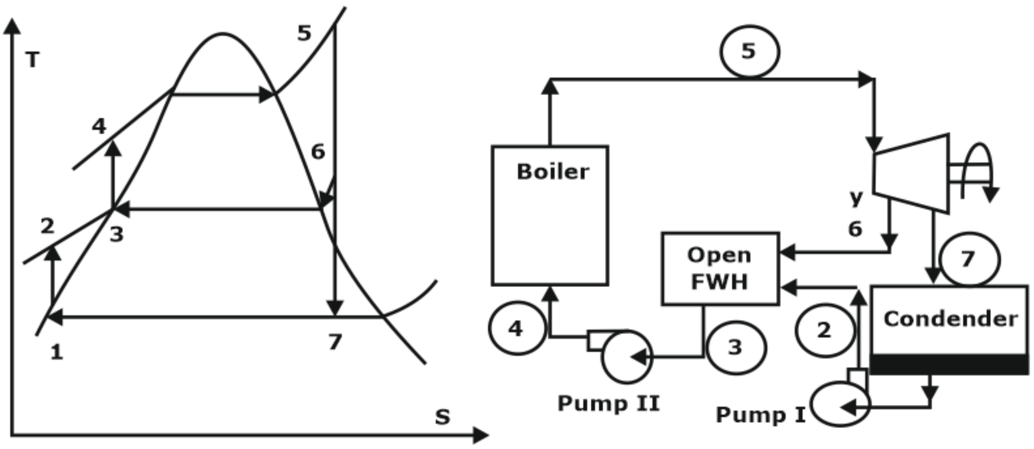
- Optimum reheat pressure for the most of the modern power plant is 0.20 to 0.25 of the initial steam pressure.

$$\text{Heat input} = \text{Primary heat} + \text{Reheat} = (h_3 - h_2) + (h_5 - h_4)$$

$$W_{turbine} = W_{H-P\ turbine} + W_{L-P\ turbine} = (h_3 - h_4) + (h_5 - h_6)$$

1.3. IDEAL REGENERATIVE RANKINE CYCLE

The mean temperature of heat addition can also be increased if heat addition at low temperature is avoided such that feed water enters the boiler at saturated liquid condition. The feed water can be brought to saturated condition by internal heating using extracted steam. This concept is known as regeneration.



The ideal regenerative Rankine cycle with an open FWH

y = amount of bleed from turbine

$Q_{in} = h_5 - h_4$

$Q_{out} = (1-y)(h_7 - h_1)$

$W_{turbine\ out} = (h_5 - h_6) + (1-y)(h_6 - h_7)$

$W_{pump} = (1-y) W_{pump1} + W_{pump2}$

$W_{pump1} = v_1(P_2 - P_1)$ and $W_{pump2} = v_3(P_4 - P_3)$,

(Because change in specific volume of liquid is very small)

Where v is specific volume at pump inlets.

- **Canonization of Rankine Cycle**: If infinite number of feed water heater is used, Rankine cycle can be converted into a Carnot cycle.
- **Law of diminishing returns**: It states that the greatest increment in efficiency is brought by the 1st heater, the increment for each additional heater thereafter successively diminish. This law is used to determine the optimum number of regenerator.

- Regeneration is done with the help of water heaters. Rankine cycle uses mainly two type of water heaters:

(i) Open feed water heater: In this type of water heater direct mixing of two fluids occurs i.e. both fluid come in physical contact with each other. Example : Deaerator .

(ii) Close feed water heater: In this type of water heater no physical mixing of fluid is allowed and heat exchange takes place by the non-contact type heat exchangers.

1.4 Binary fluid cycles:- This type of cycle is used to get the advantage of two fluids in one system. In this cycle heat rejected by one cycle works as source for other cycle.

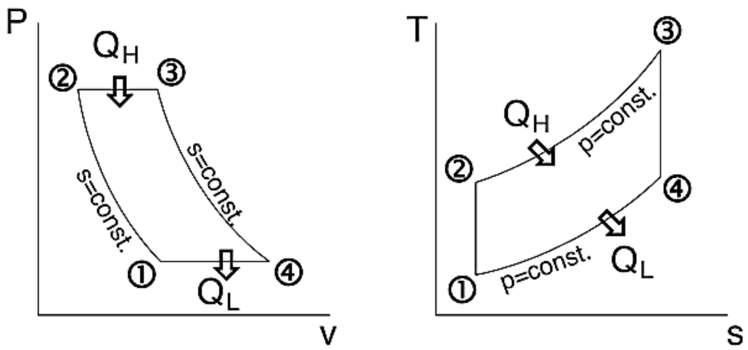
$$\eta_{binary} = 1 - (1 - \eta_{I})(1 - \eta_{II})$$

Where η_I = efficiency of 1st cycle, η_{II} =efficiency of 2nd cycle.

CHAPTER 2: GAS TURBINE

2.1 Brayton Cycle: - It consists of 2 isentropic and 2 isobaric processes. It can be used as open Brayton cycle and close Brayton cycle.

- In open cycle, fuel is burn inside combustion chamber and burnt gases are used as the working fluid.
- In close cycle, fuel burn outside the cycle and heat is exchanged with the help of heat exchanger. They are larger in size because of the use of two heat exchangers.



- Work done by compressor (1-2): $W_c = h_2 - h_1 = C_p(T_2 - T_1)$
- Heat addition (2-3): $Q_{addition} = h_3 - h_2 = C_p(T_3 - T_2)$
- Work done by turbine (3-4): $W_T = h_3 - h_4 = C_p(T_3 - T_4)$
- Heat rejection (4-1): $Q_{rejection} = h_4 - h_1 = C_p(T_4 - T_1)$
- Important relationship: $\frac{T_2}{T_1} = \frac{T_3}{T_4} = r_p^{\frac{\gamma-1}{\gamma}}$
- Efficiency: $\eta_{brayton} = 1 - \frac{1}{r_p^{\frac{\gamma-1}{\gamma}}}$, where $r_p = \frac{P_2}{P_1} = \frac{P_3}{P_4}$ (pressure ratio)
- Efficiency of Brayton cycle increases with increasing the value of pressure ration and γ.
- **Work ratio (r_w):** It is the ratio of net work and turbine work.

$$r_w = \frac{W_{net}}{W_T}$$

- **Back work ratio (r_{bw}):** It is the ratio between compressor work and turbine work.

$$r_{bw} = \frac{W_c}{W_T} \quad , \ r_w + r_{bw} = 1$$

- Back Work Ratio for gas power cycles: 40% - 60%
- **Air rate (A.R.)**: It is the mass flow rate to air used to produce 1kW of net power.

$$A.R. = \frac{\dot{m}_{air}}{P} = \frac{3600}{W_{net}} \text{ in kg/kW-hr}$$, where W_{net} is in kJ/kg.

- **Power (P):** $P = \dot{m}_{air} \times W_{net}$ in kW

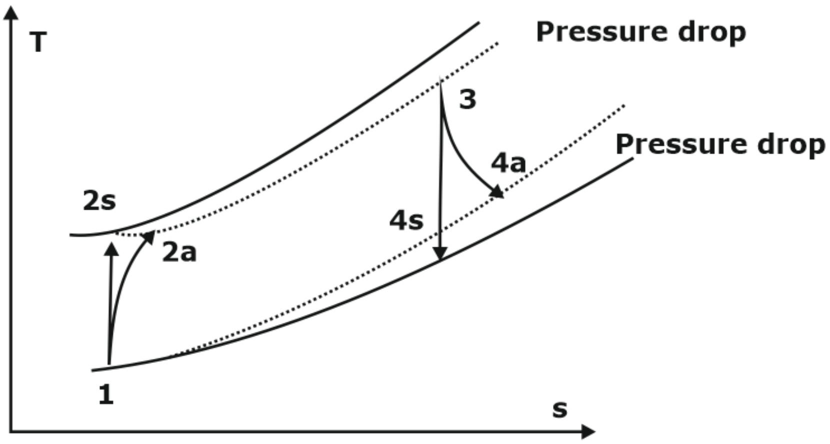
2.1.1. Maximum Pressure Ratio

- $(r_p)_{max} = \left(\frac{T_{max}}{T_{min}}\right)^{\frac{\gamma}{\gamma-1}}$
- At maximum pressure ratio the efficiency of brayton cycle is equal to Carnot cycle operating between same temperature limit.
- At maximum pressure ratio the net-work is zero.

2.1.2. Optimum Pressure Ratio

- $(r_p)_{optimum} = \left(\frac{T_3}{T_1}\right)^{\frac{\gamma}{2(\gamma-1)}} = \sqrt{(r_p)_{max}}$
- Optimum pressure ratio gives maximum net-work.
- In this condition: $T_2 = T_4 = \sqrt{T_1 \times T_3}$ and $(W_{net})_{max} = C_p\left(\sqrt{T_3} - \sqrt{T_1}\right)^2$.
- Efficiency can be written as: $\eta_{optimum} = 1 - \sqrt{\frac{T_1}{T_3}}$

2.1.3. Actual Brayton Cycle: In actual brayton cycle compression and expansion are not isentropic and pressure drop occurs due to friction in the heat exchangers /combustion chamber.

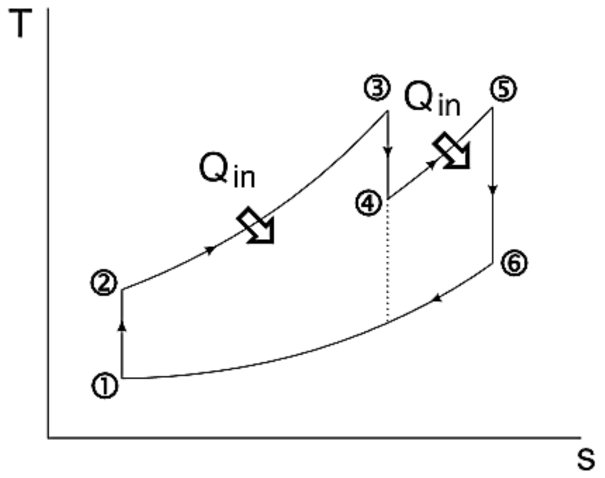


Isentropic efficiency of compressor, $\eta_{compressor} = \frac{\text{isentropic work}}{\text{actual work}} = \frac{(h_{2S} - h_1)}{(h_{2a} - h_1)}$

Isentropic efficiency of turbine, $\eta_{turbine} = \frac{\text{actual work}}{\text{isentropic work}} = \frac{(h_3 - h_{4a})}{(h_3 - h_{4s})}$

2.1.4. Brayton Cycle with Reheat

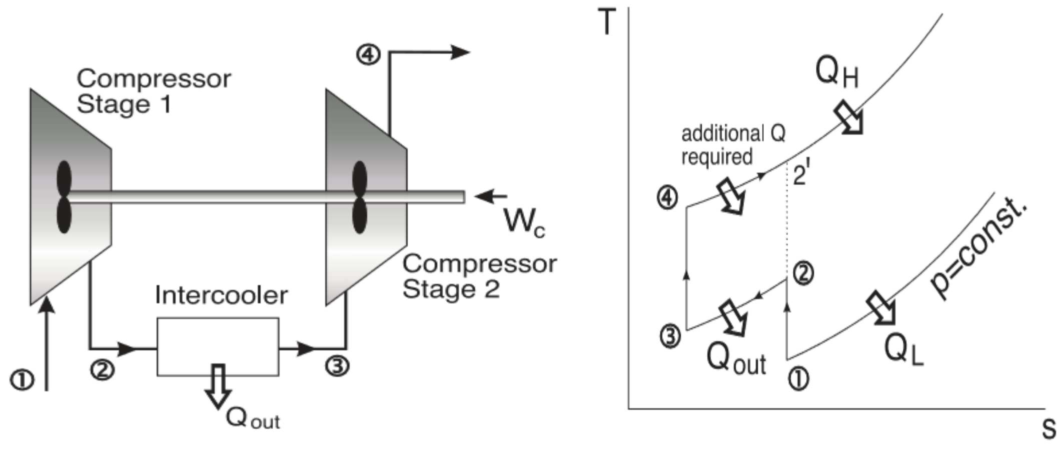
- This uses more than one gas turbine that results in increase in net work output. But the efficiency decrease.



- For perfect reheating:
  - $T_3 = T_5$
  - Intermediate pressure will be, $P_i = \sqrt{P_1 \times P_2}$
  - Work done by both the turbine will be same, $W_{T_1} = W_{T_2}$

2.1.5. Brayton Cycle with Intercooling

- It uses more than one compressor to compresses the air. By this modification, compressor work reduces and network done increases. But due to this efficiency of cycle decreases.

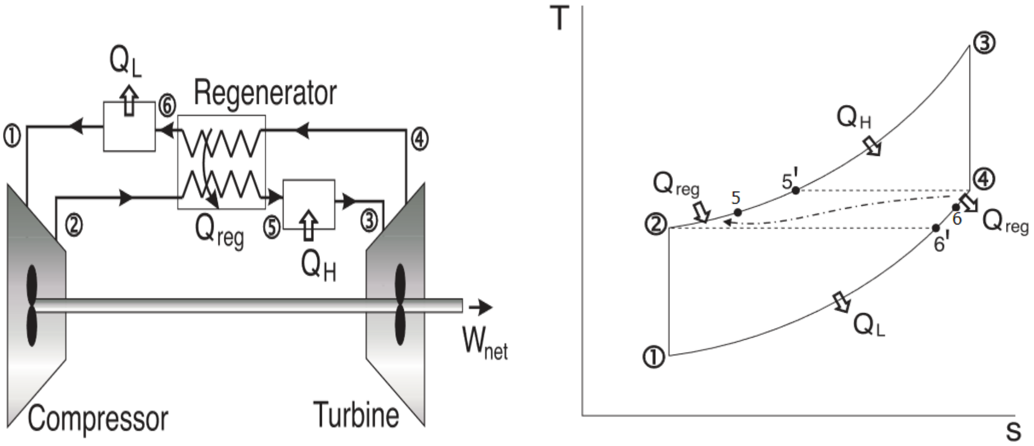


- For perfect intercooling:
  - $T_3 = T_1$
  - Intermediate pressure will be: $P_i = \sqrt{P_1 \times P_4}$
  - Work done of both compressor will be same: $W_{c_1} = W_{c_2}$

2.1.6. Brayton Cycle with Regeneration

- In this exhaust gas from turbine is used to exchange heat with the air coming to the combustion chamber.

- This will increase the mean heat addition temperature and decrease the mean heat rejection temperature hence increases the efficiency of cycle.



- Effectiveness in this type of system can be defined as the ratio of actual temperature rise and maximum possible rise in temperature due to regeneration.

$$Q_{regeneration,actual} = h_5 - h_2,\ Q_{regeneration,max} = h_4 - h_2$$

$$\text{Effectivenss}(\varepsilon) = \frac{Q_{regeneration,actual}}{Q_{regeneration,max}} = \frac{h_5 - h_2}{h_4 - h_2} = \frac{T_5 - T_2}{T_4 - T_2}$$

- For ideal regeneration:
  - $T_5 = T_4 = T_{5'}$ and $T_2 = T_6 = T_{6'}$
  - Efficiency: $\eta_{ideal\ regenerative} = 1 - \frac{T_1}{T_3}\left(r_p^{\frac{\gamma-1}{\gamma}}\right)$
  - For ideal regenerative cycle efficiency will decrease on increasing the pressure ratio and γ.

2.1.7. COMBUSTION CHAMBER:

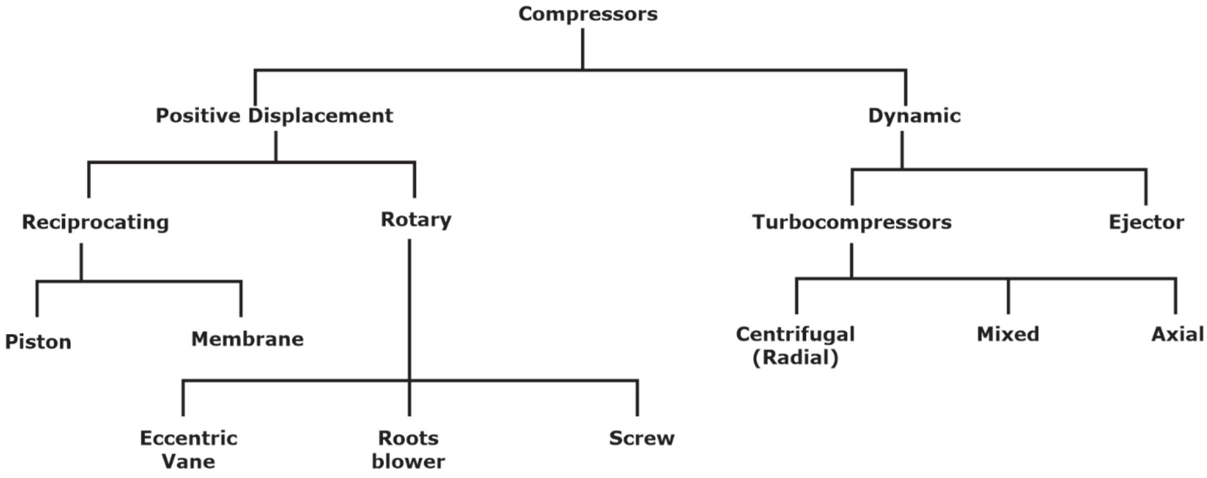
Combustion chamber is used in open brayton cycle. Generally, A very lean mixture is used in brayton cycle i.e. A/F=50:1 to 80:1. For open cycle equation of combustion chamber can be written as:

$$\dot{m}_f \times (CV)_{fuel} \times \eta_{combustion} = (\dot{m}_f + \dot{m}_a) \times C_{p_{gas}} \times T_3 - \dot{m}_a \times C_{p_{air}} \times T_2$$

Where $\dot{m}_f$ = mass flow rate of fuel, $\dot{m}_a$ =mass flow rate of air, $\eta_{combustion}$ = combustion efficiency, $C_{p_{gas}}$ = specific heat of burnt gases, $C_{p_{air}}$ =specific hear of air, T_2, T_3 are temperature at the inlet and exit of combustion chamber.

CHAPTER 3: COMPRESSOR

Compressor is a device which is used to compress the gas or vapour from lower to higher pressure and for that work input is required from outside. In process of compressing, temperature of fluid also increases.



3.1. Reciprocating compressor

3.1.1. Work input required (without clearance volume)

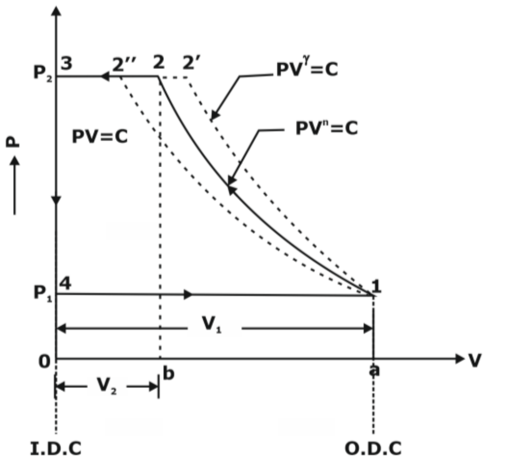
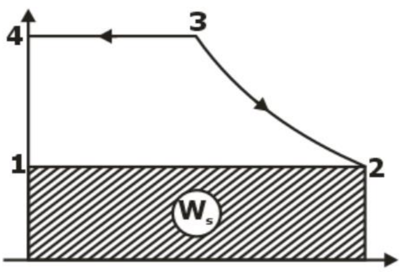
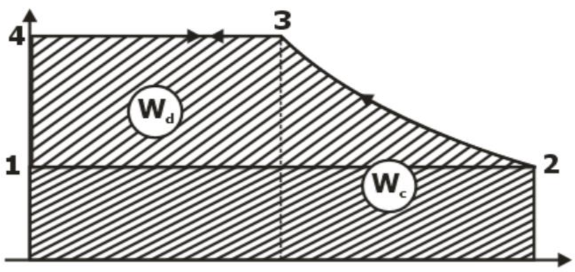


Fig.: Representation of compression Process on P-V diagram.





$$W / cycle = \left(\frac{n}{n-1}\right)\left[p_2 v_2 - p_1 v_1\right]$$

$$W / cycle = \left(\frac{n}{n-1}\right) p_1 v_1 \left[\left(\frac{p_2}{p_1}\right)^{\frac{n-1}{n}} - 1\right]$$

3.1.2. Compressor work input (with clearance volume)

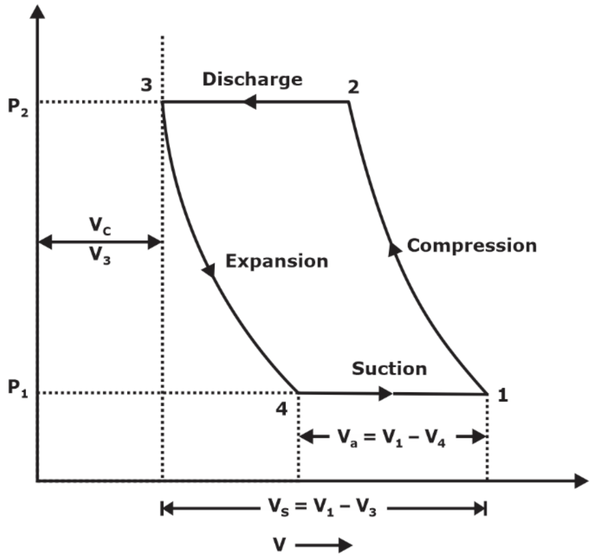


Fig.: P-V diagram with clearance

$$W / cycle = \frac{n}{n-1} p_1 \left[v_1 - v_4\right]\left[\left(\frac{p_2}{p_1}\right)^{\frac{n-1}{n}} - 1\right]$$

- For compressor minimum work input is required if the compression is isothermal. So isothermal efficiency is used for reciprocating compressor.
- Isothermal efficiency is the ratio of isothermal work and actual work of compressor. Isothermal work to compress air from pressure P_1 to P_2 is

$$W_{isothermal} = P_1 V_1 \times \ln\left(\frac{P_2}{P_1}\right)$$

$$\eta_{isothermal} = \frac{\text{isothermal work}}{\text{actual work}}$$

3.1.3. Volumetric efficiency: It is ratio of actual volume intake and swept volume.

$$\eta_{vol} = \frac{\text{Actual volume}}{\text{Swept volume}}$$

$$\eta_v = 1 + C - C\left(\frac{p_2}{p_1}\right)^{1/n}$$

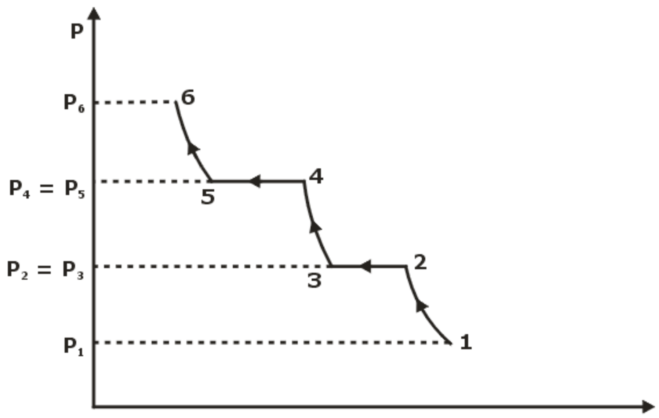
Where C= clearance ratio.

- Volumetric efficiency decreases with increase to pressure ratio. So to get higher pressure ratio multi-stage compression is used.
- Volumetric efficiency becomes zero at pressure ratio of:

$$\frac{P_2}{P_1} = \left(\frac{1}{C} + 1\right)^n$$

- FAD (free air delivered) is the volume of air compressed at atmospheric conditions i.e. P= 1 bar, and T=288 K. FAD is used to compare different compressors capacity.

3.1.4. Multi-stage compression:



P-V diagram of Multiple stage compression

- The compression is done with multi-stage with intercooling between stages which results in less total compressor work.
- The optimum pressure ratio per stage in case of n stage compression with intercooling

$$(r_P)_{opt} = \left[\frac{P_F}{P_i}\right]^{1/n}$$

- The work done in each stage is same.
- For same efficiency and perfect intercooling, dimensions of cylinder can be related as:

$$P_1D_1^2 = P_2D_2^2 = \text{constant}$$

4. CENTRIFUGAL COMPRESSOR:-

- The principal components of centrifugal compressor are impeller and diffuser.
- Theoretical Power consumed by compressor is:

$$P_{theoritical} = \dot{m} \times \left(V_{w_2}u_2 - V_{w_1}u_1\right)$$

Where $\dot{m}=$ mass flow rate of air, V_{w_2} , V_{w_2} are the whirl velocity at exit and entry respectively, u_2 , u_1 are the rotor velocity at exit and inlet respectively.

4.1 FOR RADIAL COMPRESSOR:-

- Actual power produced by turbine (for radial blade with slipping):

  $P_{actual} = \dot{m} \times \phi_s \times \phi_w \times u_2^2$

  Where, ϕ_s = slip factor ($\phi_s < 1$), ϕ_w = work factor ($\phi_w > 1$).

- In terms of enthalpy and stagnation temperature actual power consumed will be:

  $P_{actual} = \dot{m} \times \left(h_{o_2} - h_{o_1}\right) = \dot{m} \times C_p \left(T_{o_2} - T_{o_1}\right)$

  If the velocity at inlet and exit to be same. i.e. $V_1 = V_2$

  $P_{actual} = \dot{m} \times C_p \left(T_2 - T_1\right)$

  Where, h_{o_2}, h_{o_1} are the stagnation enthalpy at entry and exit of compressor,

  T_{o_2}, T_{o_1} are the stagnation temperature at entry and exit of compressor,

  T_2, T_1 are the temperature at the entry and exit of compressor.

4.2 FOR AXIAL COMPRESSOR:

- Power consumed by the compressor is:

  $$W_{net} / \text{stage} = \dot{m} \times u \times v_f \times \left(\tan\beta - \tan\alpha\right) = \dot{m} \times u \times v_f \times \left(\tan\theta - \tan\phi\right)$$

  Where u= velocity of blade at mean diameter, v_f=flow velocity of fluid,

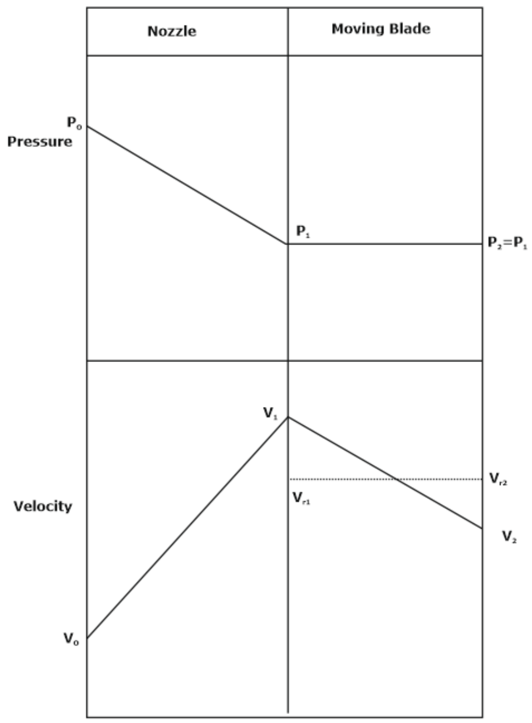
  α, β are the flow angles at entry and exit, θ, ϕ are the blade angle at entry and exit

- Forward curved blade consumes maximum power.
- Backward curved blade gives best efficiency and are stable for wide operating range.
- **Surging** is the complete breakdown of steady flow through compressor, due to periodic flow reversal. This reversal of flow causes abnormal sound, vibration, decrease in efficiency, and increase in temperature and if the intensity is more it will lead to mechanical damage.
- The maximum mass flow rate possible through compressor is termed as **choking**. It occurs when the Mach number corresponding to relative velocity at inlet become sonic. Choking means fixed mass flow rate irrespective of pressure ratio.
- **Stalling** is an aerodynamic flow separation form the blade surface due to improper design of blade. It is a local phenomenon and chances of flow separation are more at low mass flow rate, non-uniform surface, improper design of blades and higher no. of diffuser blades.

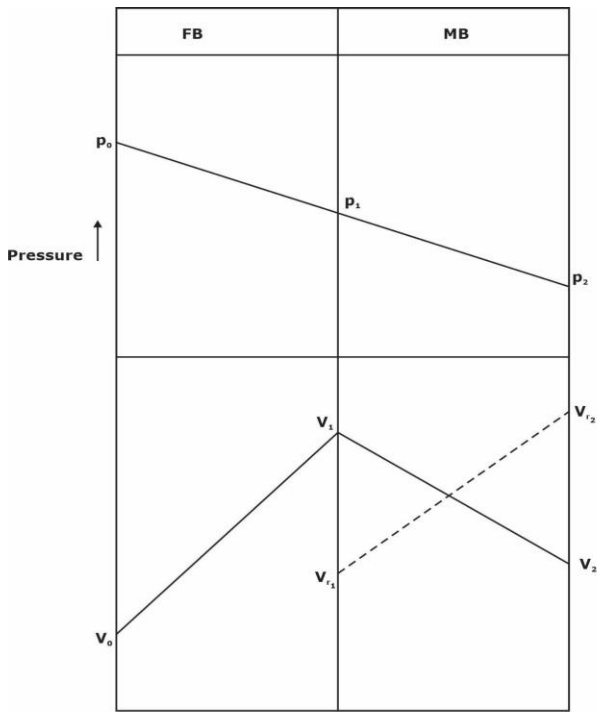
CHAPTER 4: STEAM TURBINE

A steam turbine converts the energy of high-pressure, high temperature steam produced by a steam generator into shaft work.

- On the basis of working principal steam turbine can be classified into Impulse type turbine and reaction type turbine.
- In **impulse turbine** pressure drop take place only in the nozzle and there is no drop of pressure in moving blade. This turbine converts the only kinetic energy into electrical energy.
- In **reaction turbine** pressure drop takes place both in fixed as well as moving blade. This turbine converts pressure and kinetic energy into electric energy.



Impulse turbine



Reaction turbine

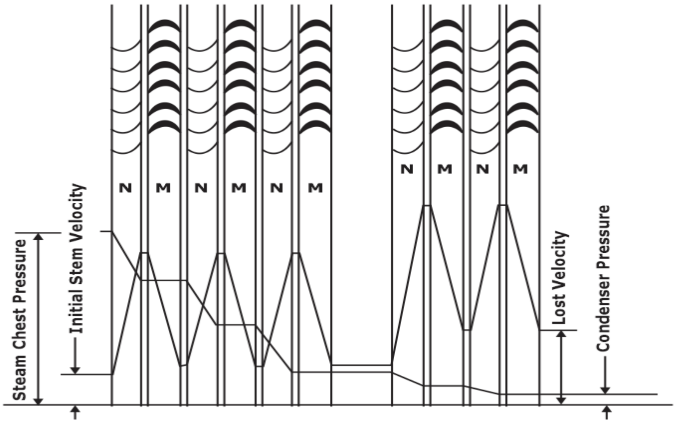
- **Degree of reaction:** This can be defined as the ratio of enthalpy drop in the rotor to the total enthalpy drop (in stator +rotor blade).

$$R = \frac{\text{enthalpy drop in rotor}}{\text{enthalpy drop in stator+enthalpy drop in rotor}} = \frac{\Delta h_{rotor}}{\Delta h_{stator} + \Delta h_{rotor}}$$

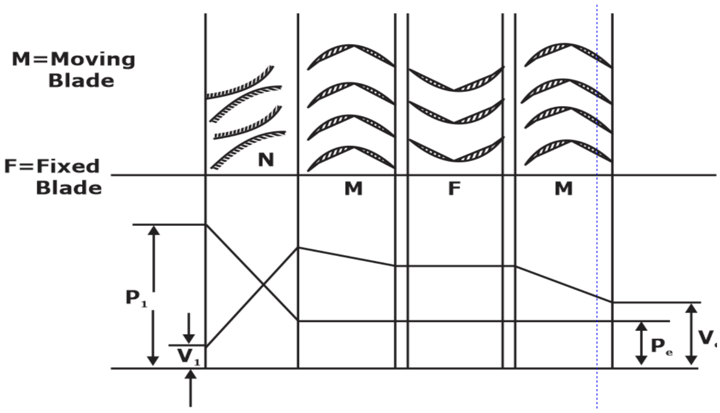
  Degree of reaction for impulse turbine is always zero.

- In Impulse steam turbine, two type of compounding is possible.

- **Pressure compounding:** In this pressure drop takes place in more than one row of nozzles and the increase in kinetic energy after each nozzle is held within limits. Example: Rateau turbine.



- **Velocity compounding:** In this type of compounding first all the pressure energy is converted into kinetic energy with the use of single nozzle, then this kinetic energy of steam is extracted by the row of moving and fixed blades. Example: Curtis turbine.



DIFFERENT STEAM TURBINES:-

1. **De Laval turbine:-** A single-stage impulse turbine is called the de Laval turbine. It consists of a single rotor with impulse blades attached to it. In this turbine all the pressure energy converted into kinetic energy with the help of nozzle and then kinetic energy is converted into shaft work with the help of impulse blade attached to rotor.

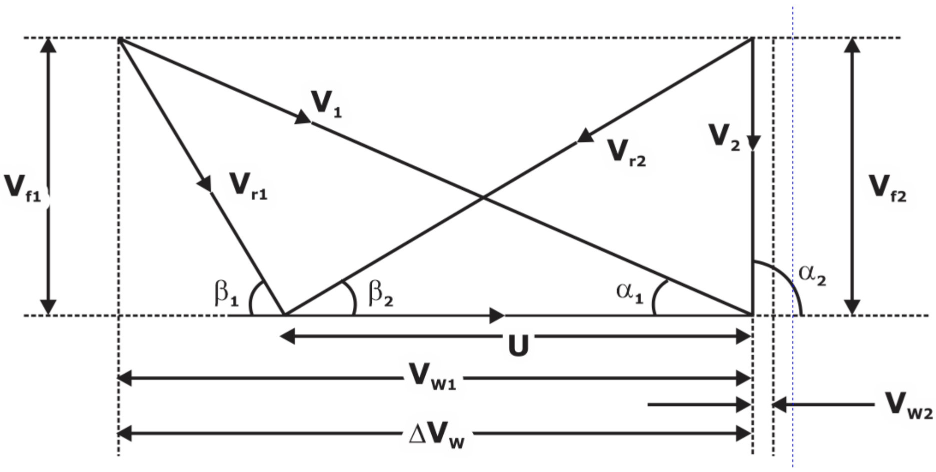


Fig.: Velocity Triangle of Impulse Turbine

Power developed: $P = \dot{m} \times u \times \left(V_{w_2} - V_{w_1}\right)$

Where, V_{w_2}, V_{w_1} are the whirl velocity at inlet and exit respectively, u= velocity of blade/rotor, $\dot{m}$ = mass flow rate of steam.

Nozzle efficiency: It is defined as the ratio of kinetic energy at nozzle exit and isentropic enthalpy change within nozzle.

$$\eta_{\text{nozzle}} = \frac{\text{Kinetic energy at exit of nozzel}}{\text{isentropic enthalpy change in nozzle}} = \frac{\frac{1}{2}\dot{m} \times V_1^2}{\dot{m} \times \Delta h_{\text{isentropic}}}$$

Blade efficiency: It is the ratio of power developed and the kinetic energy provided by the nozzle.

$$\eta_{\text{blade}} = \frac{\text{work by turbine blade}}{\text{Kinetic energy supplied by nozzel}} = \frac{\dot{m} \times u \times \left(V_{w_2} - V_{w_1}\right)}{\frac{1}{2}\dot{m} \times V_1^2}$$

It can be written as:

$$\left(\eta_b\right) = 2\left(\rho\cos\alpha - \rho^2\right)\left(1 + k_b\right)$$

Where, $\rho = \dfrac{V_b}{V_1}$

For maximum efficiency,

$$\rho_{\text{optimum}} = \frac{V_b}{V_1} = \frac{\cos\alpha}{2}$$

$$(\eta_b)_{\max} = \left(\frac{1 + k_b}{2}\right)\cos^2\alpha$$

If friction is neglects (k_b=1), maximum efficiency will be,

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

Where, V_b is the blade velocity, V_1= fluid velocity at inlet, α =nozzle angle at inlet,

k_b=constant for friction loss.

2. **Rateau turbine :-**To resolve the problem of very high speed of rotor in case of single-stage impulse turbine, the total enthalpy is divided among many single-stage impulse turbine in series. And such a turbine is known as Rateau turbine. In this turbine :

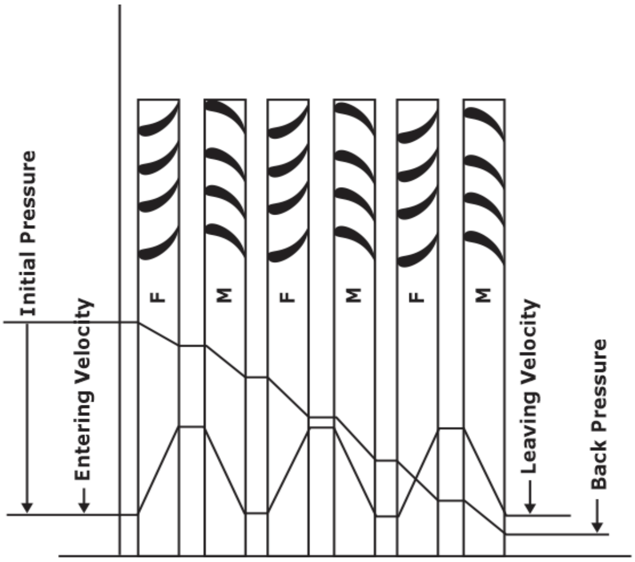
$$(\Delta h)_{stage} = \frac{\Delta h_{total}}{n}$$

3. **Curtis turbine: -** This is an impulse turbine with velocity compounding. In this turbine pressure and enthalpy drop of steam take place in single stage/row of nozzle and the kinetic energy of steam is absorb by the number of moving blades. For n-row Curtis turbine condition of maximum efficiency will be (for no friction, symmetric blades and axial discharge):

$$\rho = \frac{V_b}{V_1} = \frac{\cos \alpha}{2n}$$

NOTE: Pearson turbine is a reaction turbine with degree of reaction 0.5. Hero's turbine is a reaction turbine with degree of reaction 1.

4. **Reaction turbine:**
   - A reaction turbine 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.
   - 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.

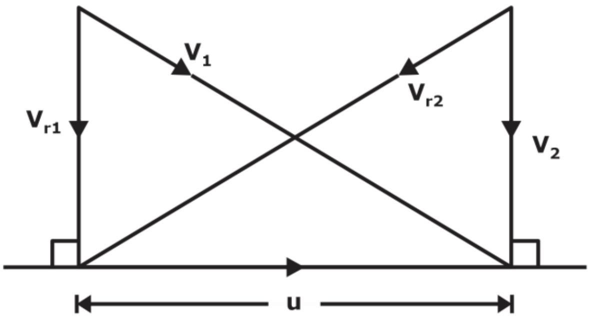


Reaction turbine indicating pressure and velocity distribution

- Condition for maximum efficiency in reaction turbine:

$$\rho_{optimum} = \frac{V_b}{V_1} = \cos\alpha$$

And maximum efficiency will be: $\eta_{max} = \dfrac{2\cos^2\alpha}{1+\cos^2\alpha}$



Velocity diagram for maximum efficiency

Generally $\eta_{reaction} > \eta_{rateau} > \eta_{curtis}$

Reheat factor and Stage efficiency: - The Thermodynamic effect on the turbine efficiency can be best understood by considering a number of stages between two stages, 1 and 2 as shown in Figure. (4 stages in between).

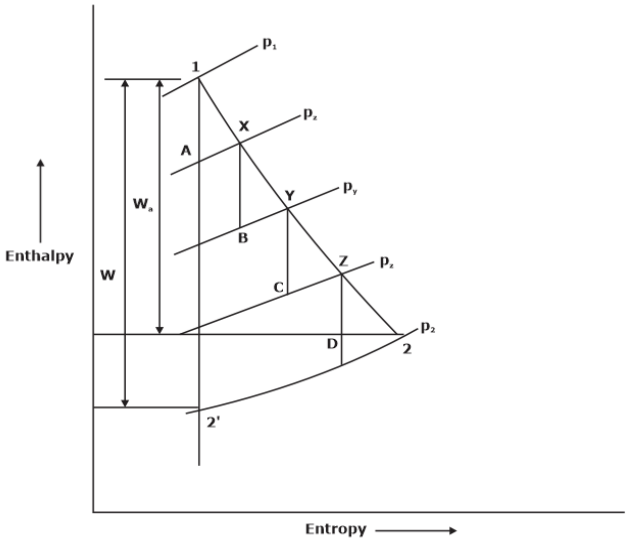


Fig: Different stage of a steam turbine

The total expansion is divided into four stages of the same efficiency (η_s) and pressure ratio.

$$\frac{P_1}{P_x} = \frac{P_x}{P_y} = \frac{P_y}{P_z} = \frac{P_z}{P_2}$$

$$\text{So, Overall efficiency}\left(\eta_0\right) = \frac{W_a}{W} = \frac{\text{actual enthalpy drop}(1-2)}{\text{isentropic heat drop}(1-2')}$$

Reheat factor is defined as:

$$\text{Reheat factor(RF)} = \frac{\text{Cumulative enthalpy drop (isentropic)}}{\text{Isentropic enthalpy drop}(\text{overall})} = \frac{\Delta h_{1A} + \Delta h_{XB} + \Delta h_{YC} + \Delta h_{ZD}}{\Delta h_{12}}$$

Generally R.F is varied from 1.03 to 1.04

Stage efficiency $\left(\eta_s\right)$ is defined as the ratio of actual enthalpy drop and isentropic enthalpy drop for a particular stage. If η_s remains same for all the stages or η_s is the mean stage efficiency.

$$\eta_s = \frac{\Delta h_{1x}}{\Delta h_{1A}} = \frac{\Delta h_{xy}}{\Delta h_{xB}} = \frac{\Delta h_{yz}}{\Delta h_{yc}} = \frac{\Delta h_{z2}}{\Delta h_{zD}}$$

So, overall efficiency can be written as

$$\eta_o = \eta_s \times \text{R.F.}$$

This makes the overall efficiency of the turbine greater than the individual stage efficiency.

Steam Turbine Governing and Control:

The objective of governing is to keep the turbine speed fairly constant irrespective of load. The principal methods of steam turbine governing arc as follows:

(i). Throttle governing

(ii). Nozzle governing

(iii). By-pass governing

Comparison of Throttle Governing and Nozzle Control Governing

S. No.	Aspects	Throttle Control	Nozzle Control
1.	Throttling losses	Severe	No throttling losses (Actually there are a little throttling losses in nozzles valves which are partially open).
2.	Partial admission losses	Low	High
3.	Heat drop available	Lesser	Larger
4.	Use	Used in impulse and reaction turbines both.	Used in impulse and also in reaction (if initial stage impulse) turbines.
5.	Suitability	Small turbines	Medium and larger turbines

CHAPTER 5: COMPRESSIBLE FLOW

A flow is said to be compressible flow when there is considerable change in the density of fluid during process. Mach number is an important non-dimensional number used in study of compressible flow.

- Mach number is the ratio of velocity of object and velocity of sound in fluid.

  $$M = \frac{\text{velocity of object}}{\text{velocity of sound}} = \frac{v}{c}$$

  For M<1 : flow is called **subsonic**.

  For M=1 : flow is called **sonic.**

  For M>1 : flow is called **supersonic.**

  For M>5 : flow is called **hypersonic.**

- **Stagnation state:** when a fluid is decelerated isentropically to a zero velocity state than the final zero velocity state is known as stagnation state. All properties corresponding to the stagnation state is known as stagnation properties. Example: stagnation enthalpy, stagnation temperature, stagnation pressure etc.

- For compressible flow stagnation temperature and pressure can be written as:

$$T_o = T \times \left(1 + \left(\frac{\gamma - 1}{2}\right) \times M^2\right)$$

$$\frac{P_o}{P} = \left(\frac{T_o}{T}\right)^{\frac{\gamma}{\gamma-1}}$$

For isentropic process: stagnation pressure and stagnation temperature remains constant. i.e. $T_{o_1} = T_{o_2}$ and $P_{o_1} = P_{o_2}$.

For adiabatic process: only stagnation temperature remains constant. i.e. $T_{o_1} = T_{o_2}$ but $P_{o_1} \neq P_{o_2}$.

- A state is said to be critical state when Mach number becomes one. i.e. M=1 (v=c). And the properties at critical state are known as **critical properties**.
- Relation between critical and stagnation properties:

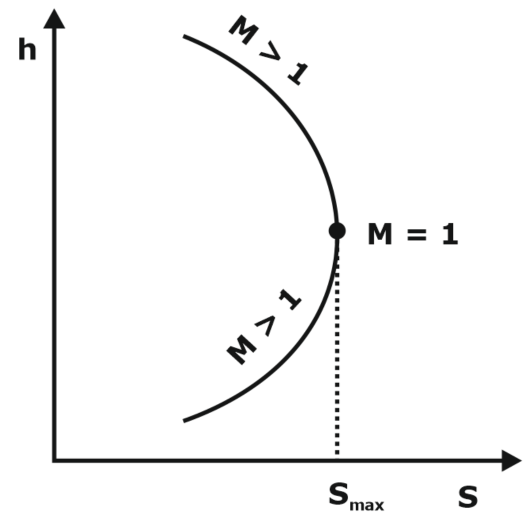
$$\frac{T^*}{T_o} = \left(\frac{2}{\gamma + 1}\right) \text{ and } \frac{P^*}{P_o} = \left(\frac{2}{\gamma + 1}\right)^{\frac{\gamma}{\gamma-1}}$$

- Stagnation velocity of sound: $c_o = \sqrt{\gamma R T_o}$
- Relation between maximum velocity and stagnation velocity of sound:

$$\frac{v_{max}}{c_o} = \sqrt{\frac{2}{\gamma - 1}} \text{ , where } v_{max} = \sqrt{2 C_p T_o}$$

- **Fanno flow**: Under idealized condition this type of flow may be assumed in gas duct of aircraft, air conditioning system etc. where there is friction but no heat transfer. Fanno flow is the adiabatic flow through a constant area duct where the effect of friction is taken into consideration. The Fanno line defines the possible states for a gas when the mass flow rate and total enthalpy are held constant, but the momentum varies.

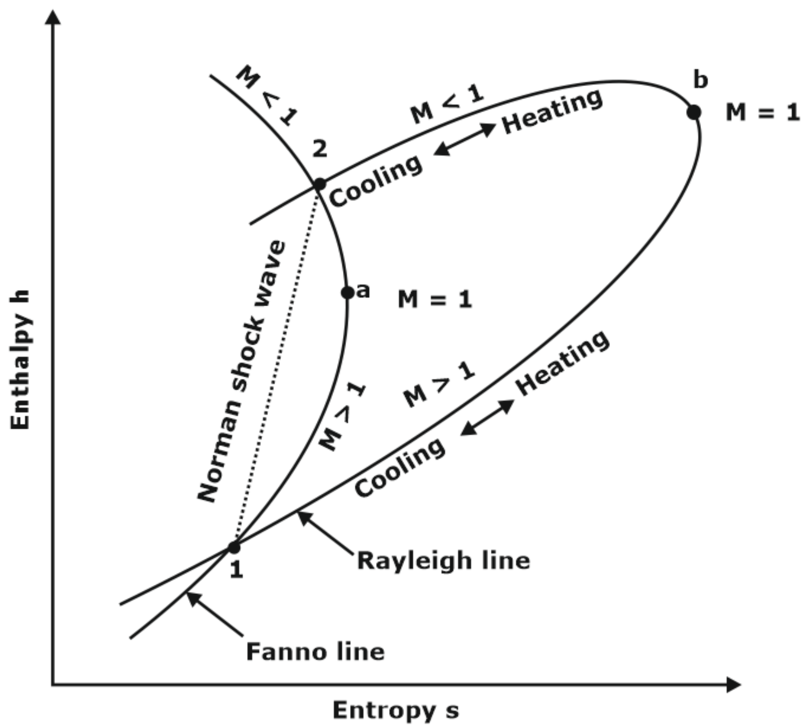
  The Fanno flow model is often used in the design and analysis of nozzles.



- **Rayleigh flow**: Flow in a constant area duct with heat transfer and without friction can be assumed as Rayleigh flow.

  The Mach number at maximum temperature is:

$$M = \frac{1}{\sqrt{\gamma}}$$

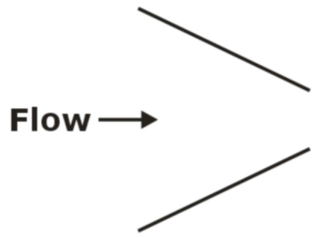


- **Nozzle:** nozzle is a device used to increase the velocity of fluid and decrease the pressure of fluid and nozzle efficiency can be define as:

$$\text{nozzle efficiency}(\eta_n) = \frac{\text{actual enthaply drop}}{\text{ideal enthalpy drop}}$$

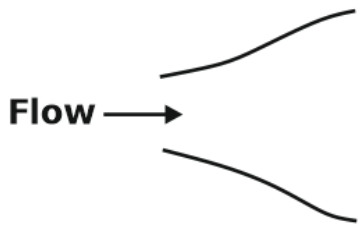
Types of Nozzles:

(i) Convergent Nozzle:



Cross section of the nozzle decreases continuously from entrance to exit

(ii) Divergent Nozzle:



Cross section of the nozzle increases continuously Cross section of the nozzle increases continuously from entrance to exit

(iii) Convergent –Divergent Nozzle

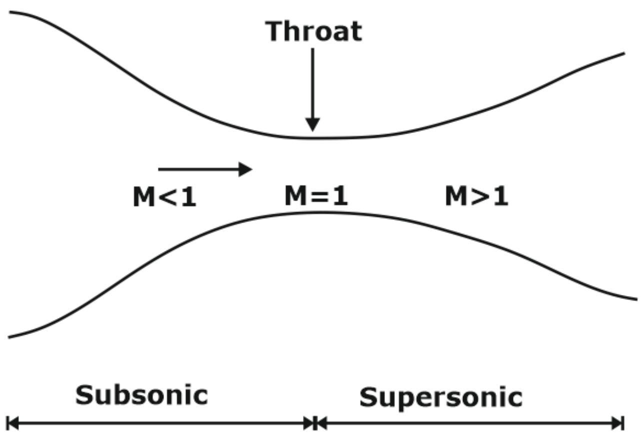


Fig.: Cross section of the nozzle first decreases and then increases

- **Diffuser:** Diffuser is a device used to increase the pressure of fluid and decrease the velocity of fluid and diffuser efficiency can be define as:

$$\text{diffuser efficiency}(\eta_d) = \frac{\text{ideal enthaply rise}}{\text{actual enthalpy rise}}$$

- For nozzle relation between area and velocity:

$$\frac{dA}{A} = \frac{dV}{V}(M^2 - 1)$$

Where A= Areas, V= Velocity of fluid.

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