UNIT – 7 PART-I

(Reciprocating Compressors)

Contents

Important Repeated Questions:

1. **What is the need/Justify the need for multistaging in reciprocating compressor?**
   (W25 - Q4a, 03 marks) (W22 - Q3a, 03 marks) (S23 - Q1c, 07 marks)
2. **Derive the equation of work done on air for multi-stage reciprocating air compressor. / Show that for a two-stage reciprocating air compressor with complete intercooling the total work of compression becomes minimum when the pressure ratio in each stage is equal.**
   (S25 - Q2 OR c, 07 marks) (S23 - Q1c, 07 marks) (S22 - Q5c, 07 marks)
3. **What is the effect of clearance on the performance of air compressor?**
   (W22 - Q5b, 04 marks) (W24 - Q5a)

Legends: W- Winter, S- Summer, Q- Question and 03/04/07- Marks of Question

7.1 Introduction

- A **compressor** is a power absorbing machine used to increase the pressure of fluid (i.e air, gas or vapour) above that at which it is available.
- Compressor may be of the reciprocating piston-cylinder type or of the rotary type. Since the process of compressing fluid requires work should be done on it, thus compressor has to be driven by a prime mover, such as electric motor or engine.
- “A machine which takes in air during suction stroke at low pressure and compresses it to high pressure in a piston cylinder arrangement and then delivers it to some storage vessel (receiver) is known as Reciprocating air compressor”.
- The high pressure air from the receiver may be supplied by a pipe line to wherever it is required.

7.2 Classification of Compressors

According to design and principle of operation, compressors can be classified in following two categories:

a) Positive displacement type compressors
b) Roto dynamic type compressors

(a) Positive Displacement Type Compressors

- “In positive displacement type compressors, the pressure of air is increased by decreasing its volume.”
- They are further divided in two types as below:
  - In positive displacement ***Reciprocating compressors***; the air is compressed due to the action of reciprocating piston moving axially in the cylinder.
  - In positive displacement ***Rotary compressors***; the air is trapped in between two sets of engaging surfaces and the pressure rise is either by the back flow of air (i.e Roots blower) or by both squeezing action and back flow of air (i.e. Vane blower).
  - In our scope of syllabus, we will discuss only reciprocating type compressors.

(b) Roto Dynamic Compressors

- In Roto-dynamic compressors the air is not trapped in specified boundaries but it flows continuously and steadily through the machine and the kinetic energy imparted to the air by the rotor is changed into pressure energy partly in the rotor and the rest in the diffuser”.
- The rise in pressure is carried by the **dynamic action** of air due to change in angular momentum of air passing through the rotor.
- They further divided in two types as below:
  - In ***centrifugal compressors****;* the flow of air is more or less radial. The kinetic energy imparted to the air is partly changed into pressure energy due to centrifugal action in the rotor and due to diffusion action in the diffuser.
  - In ***axial flow compressors*** the flow of air is axial and there is no centrifugal action.
- Roto-dynamic compressors are characterized by *large volumetric capacity* and relatively low pressures.

7.3 Construction and Working of Single Stage Single Acting Reciprocating Compressor

A single stage, single acting reciprocating compressor is shown in Fig. 7.1.

Construction

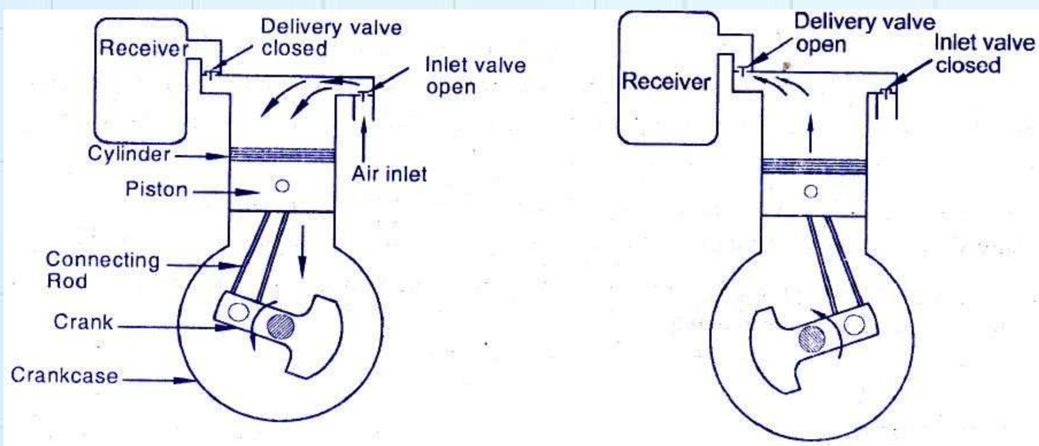
- It consists of a piston cylinder assembly fitted with inlet and delivery valves.
- The piston reciprocates in a cylinder and derives its motion through a connecting rod and crank mounted in a crank case.
- The inlet and delivery valves are provided in the head of the cylinder. The valves are mostly of thin steel plates with light springs for reducing interia. The valves operate because of pressure difference between them.

Working

There are mainly two strokes in reciprocating compressor as explain below:

(a) Forward Stroke (or Suction Stroke)

- In this stroke; when the piston is moving from TDC towards BDC in downward direction; the inlet valve will open due to difference of pressure and fresh air will entering into the cylinder as shown in Fig. 7.1 (a).



(a) *(b)*

Fig.7.1 – Single stage single acting reciprocating compressor

- During suction stroke; the delivery valve remains closed because the receiver pressure on the outside of this valve is much higher than the suction pressure.
- In actual practice; Inlet valve and Delivery valve will never open suddenly. Due to valve fluttering & valve inertia; it has required some time to open; thus both the valves will open in advanced.

(b) Reversed Stroke (Compression & Delivery Stroke)

- This stroke will divide in two processes, namely compression & delivery process.
- During compression; when the piston is moving from BDC towards TDC in upward direction; the air will begins to compressed & slight increase in pressure will close the inlet valve.
- Since both the inlet and delivery valves are closed during compression and the pressure of air will rise at the expense of its volume.

- Finally, when a pressure will be reach slightly more that compressed air pressure on the outside of the delivery valve then the delivery valve will open as as shown in Fig. 7.1 (b).
- The compressed air is now delivered from the cylinder to the receiver until the piston reaches the end of its upward stroke.
- At the end of delivery stroke piston once again begin to move downwards, the delivery valve closes, the inlet valve opens and the cycle is repeated.

7.4 Work Done Equation For Single Stage Reciprocating Compressor Neglecting Clearance Volume

Fig. 7.2 shows a theoretical P-V and T-S diagrams for a reciprocating air compressor neglecting clearance volume.

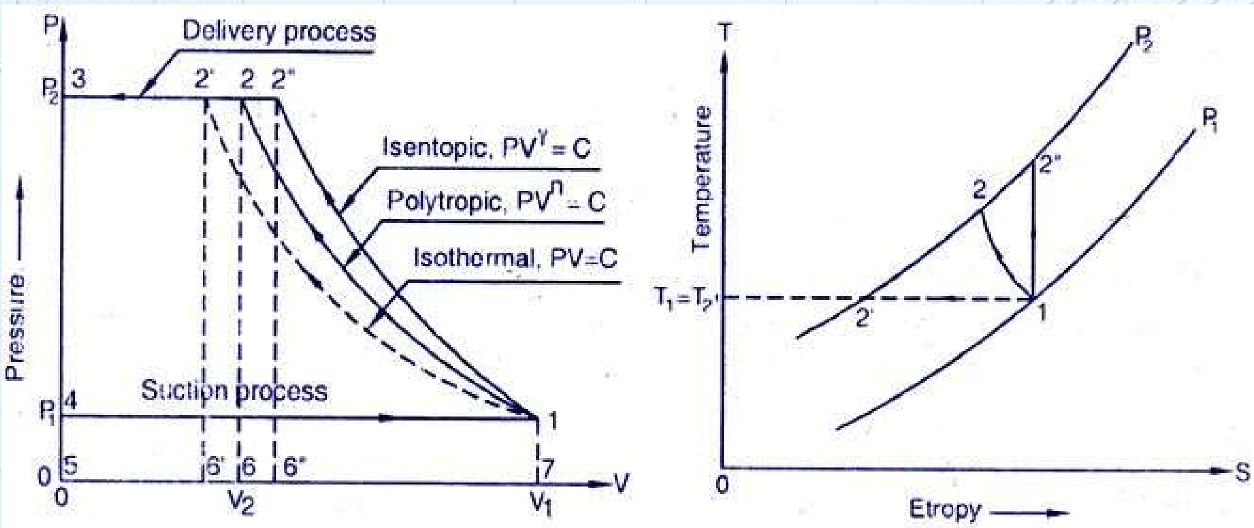


Fig.7.2 – P-V and T-S diagrams for a reciprocating air compressor neglecting clearance volume

The sequences of operations as represented on the diagrams are as follows:

Suction Process 4-1:

- In suction process, the volume in the cylinder increases from zero at 4 to the required volume V1 to fill the cylinder at 1 and air is inducted into the cylinder at constant pressure P1, and temperature T1.

Compression Process 1-2:

The air can be compressed as per following three cases:

a) Polytropic compression

- If the air is compressed according to the law of ***PV^n = constant*** (polytropic compression) from pressure P_1 to P_2, volume decreases from V_1 to V_2 and temperature increases from T_1 to T_2; then this process represented by ***process 1-2'*** on P-V diagram.

b) Isentropic (or Reversible adiabatic) compression

- In the absence of heat transfer, if the air is compressed according to the law of ***PV^γ = constant*** (Isentropic compression); then this process represented by ***process 1-2"*** on P-V diagram.

c) Isothermal compression

- If the heat transfer during compression is controlled in such a way that temperature during compression process remains constant and air is compressed according to the law of ***PV = constant***. This process represented by ***process 1-2*** on P-V diagram.

ln actual practice the compression process is ***neither isentropic nor isothermal***'; But it lies between isentropic and isothermal processes and regarded as polytropic process with equation PV^n = constant."

Delivery Process 2-3:

- In delivery process, the compressed air at volume V_2, pressure P_2, and temperature T_2 is delivered from the cylinder to the receiver.

Work done equation for each of these three cases are discused in detail as below:

7.4.1 Case: (A) Work Done Equation For Polytropic Compression ($PV^n = C$)

- The net work done in the cycle is given by the area of the P-V diagram and is the work done on the air as shown in Fig. 7.2.
- Let, P_1 = Pressure of the air at the beginning of the compression, N/m^2

  V_1 = Volume of the air at the beginning of the compression, m^3

  T_1 = Absolute temperature of the air at the beginning of the compression, K P_2, V_2,

  T_2 = Corresponding values at the end of the compression
- Indicated work done on the air per cycle is given by,

$$W = \text{Area 1-2-3-4-1}$$
$$= \text{Area 2-3-5-6-2 + Area 1-2-6-7-1 – Area 1-4-5-7-1}$$
$$= \text{Area under 2-3 + Area under 1-2 – Area under 1-4}$$
$$= P_2V_2 + \frac{P_2V_2 - P_1V_1}{n-1} - P_1V_1$$
$$= \left(P_2V_2 - P_1V_1\right) + \frac{P_2V_2 - P_1V_1}{n-1}$$
$$= \left(P_2V_2 - P_1V_1\right)\left[1 + \frac{1}{n-1}\right]$$
$$= \frac{n}{n-1}\left(P_2V_2 - P_1V_1\right)$$
$$= \frac{n}{n-1} P_1V_1 \left[\frac{P_2V_2}{P_1V_1} - 1\right]$$

- Now for polytropic compression process 1-2, we can write

$$P_1V_1^n = P_2V_2^n$$
$$\therefore \frac{V_2}{V_1} = \left(\frac{P_1}{P_2}\right)^{1/n} = \left(\frac{P_2}{P_1}\right)^{-1/n}$$

- Substituting the value of $\frac{V_2}{V_1}$ in equation, we get

$$W = \frac{n}{n-1} P_1V_1 \left[\left(\frac{P_2}{P_1}\right)\left(\frac{P_1}{P_2}\right)^{1/n} - 1 \right]$$

$$\boldsymbol{W = \frac{n}{n-1} P_1V_1 \left[\left(\frac{P_2}{P_1}\right)^{\frac{n-1}{n}} - 1\right], J/cycle} \qquad \textit{Eq. (7.1)}$$

- But for polytropic process, $\left(\frac{P_2}{P_1}\right)^{\frac{n-1}{n}} = \frac{T_2}{T_1}$ and substitute $PV_1 = mRT_1$ in above eq. (6.1),

$$W = \frac{n}{n-1} mRT_1 \left[\left(\frac{T_2}{T_1}\right) - 1 \right]$$

$$W = \frac{n}{n-1} mR(T_2 - T_1) \text{ J/cycle}$$

Where 'm' is the mass of air inducted and delivered per cycle.

- Work input per kg of air is given by,

$$\boldsymbol{W = \frac{n}{n-1} RT_1 \left[\left(\frac{P_2}{P_1}\right)^{\frac{n-1}{n}} - 1\right], J/kg} \qquad \textit{Eq. (7.2)}$$

7.4.2 Case: (B) Work Done Equation For Isentropic Compression (PV^{Υ} = Constant)

- The work required per cycle when the air is compressed isentropically is obtained by substituting 'Υ' in place of 'n' in above equations.

$$\boldsymbol{W = \frac{\gamma}{\gamma-1} P_1V_1 \left[\left(\frac{P_2}{P_1}\right)^{\frac{\gamma-1}{\gamma}} - 1\right], J/cycle} \qquad \textit{Eq. (7.3)}$$

- Also, for isentropic process, $\left(\frac{P_2}{P_1}\right)^{\frac{\gamma-1}{\gamma}} = \frac{T_2}{T_1}$ and substitute $P_1V_1 = mRT_1$

$$W = \frac{\gamma}{\gamma-1} mRT_1 \left[\left(\frac{P_2}{P_1}\right)^{\frac{\gamma-1}{\gamma}} - 1 \right]$$

$$W = \frac{\gamma}{\gamma-1} mR(T_2 - T_1) \text{ J/cycle}$$

- But the ratio of specific heats, $\gamma = \frac{C_P}{C_V}$ and $C_P - C_V = R$; Substitute these values in above equation we get,

$$W = \frac{\frac{C_P}{C_V}}{\frac{C_P}{C_V} - 1} m (C_P - C_V)(T_2 - T_1)$$

$$\boldsymbol{W = mC_p(T_2 - T_1), J/cycle} \qquad \textit{Eq. (7.4)}$$

7.4.3 Case: (C) Work Done Equation For Isothermal Compression (PV = Constant)

- The isothermal compression follows the law of PV = constant; and the temperature remains constant.
- Thus by the law of conservation of energy the entire work of compression is discharged to the cooling media, thus ***no energy is wasted*** in heating the air or increasing the internal energy.
- Thus less work will be required to be done per kg of air to raise its pressure from P_1 to P_2.
- Work done on air per cycle,

$$W = \text{area } 1\text{-}2'\text{-}3\text{-}4\text{-}1$$

$$= \text{area } 2'\text{-}3\text{-}5\text{-}6'\text{-}2' + \text{area } 1\text{-}2'\text{-}6'\text{-}7\text{-}1 - \text{area } 1\text{-}4\text{-}5\text{-}7\text{-}1$$

$$= \text{area under } 2\text{-}3 + \text{area under } 1\text{-}2 - \text{area under } 4\text{-}1$$

$$= P_2V_2 + P_2V_2 \ln\frac{V_1}{V_2} - P_1V_1$$

But for isothermal process, by substituting $P_1V_1 = P_2V_2$ and $\frac{V_1}{V_2} = \frac{P_2}{P_1}$ in above equation,

$$W = P_2V_2 \ln\frac{V_1}{V_2} = P_1V_1 \ln\frac{V_1}{V_2} = P_1V_1 \ln\frac{P_2}{P_1}$$

$$W = mRT_1 \ln\frac{P_2}{P_1}, J/cycle \qquad Eq.\ (7.5)$$

Work input per kg of air is,

$$\boldsymbol{W = RT_1 \ln\frac{P_2}{P_1}, J/kg} \qquad Eq.\ (7.6)$$

7.5 Construction and Working of Two Stage Reciprocating Air Compressor With Intercooler

Fig. 7.3 shows the schematic diagram of a two stage reciprocating compressor and Fig. 7.4 shows P-V and T-S diagrams of the compression cycle.

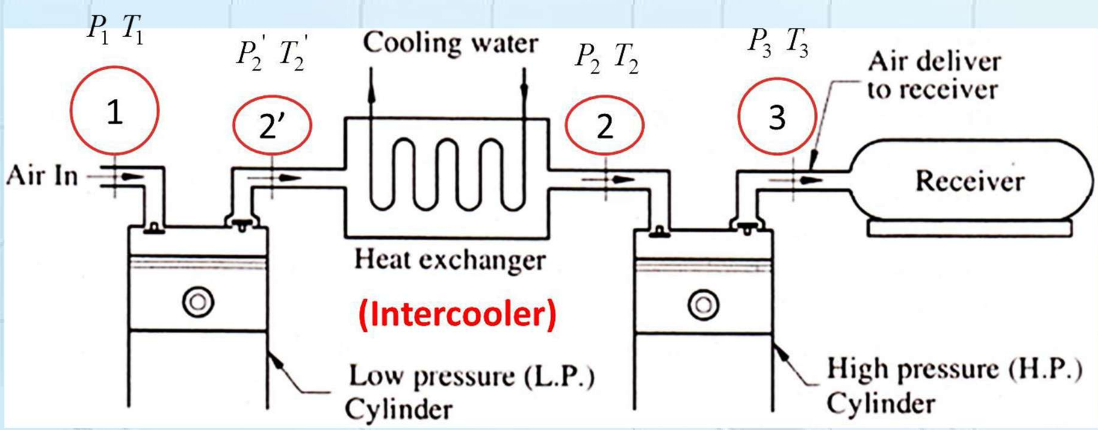


Fig.7.3 Schematic of two stage compressor with intercooler

First Stage Compression (L.P. Stage)

- In the two stage compressor with intercooler, the air is first drawn into the low pressure (L.P) cylinder at point 1 with pressure P_1 and volume V_1.

- The air is compressed polytropically to some intermediate pressure P_2, as shown by process 1-2'.
- The compressed air at pressure P_2 and temperature T_2 is then cooled at constant pressure P_2 by passing the air through **intercooler**. This is represented by a constant pressure process 2'-2 on P- V diagram.

Second Stage Compression (H.P. Stage)

- In 2nd stage compression; the air is enters in the high pressure (H.P) cylinder, where it is further compressed polytropically along process 2 – 3 as shown in P-V diagram and the pressure of air increases from P_2 to P_3, and finally the air is discharged to the receiver at final required delivery pressure P_3. (Refer Fig. 7.4)
- Thus in each stage pressure of the air is successively increased and the initial temperature is maintained at the end.
- During the whole process the compression is approximated isothermal process.
- If the compression had taken place in a **single stage**, the compression curve would have followed the **polytropic curve 1 – 3"**.

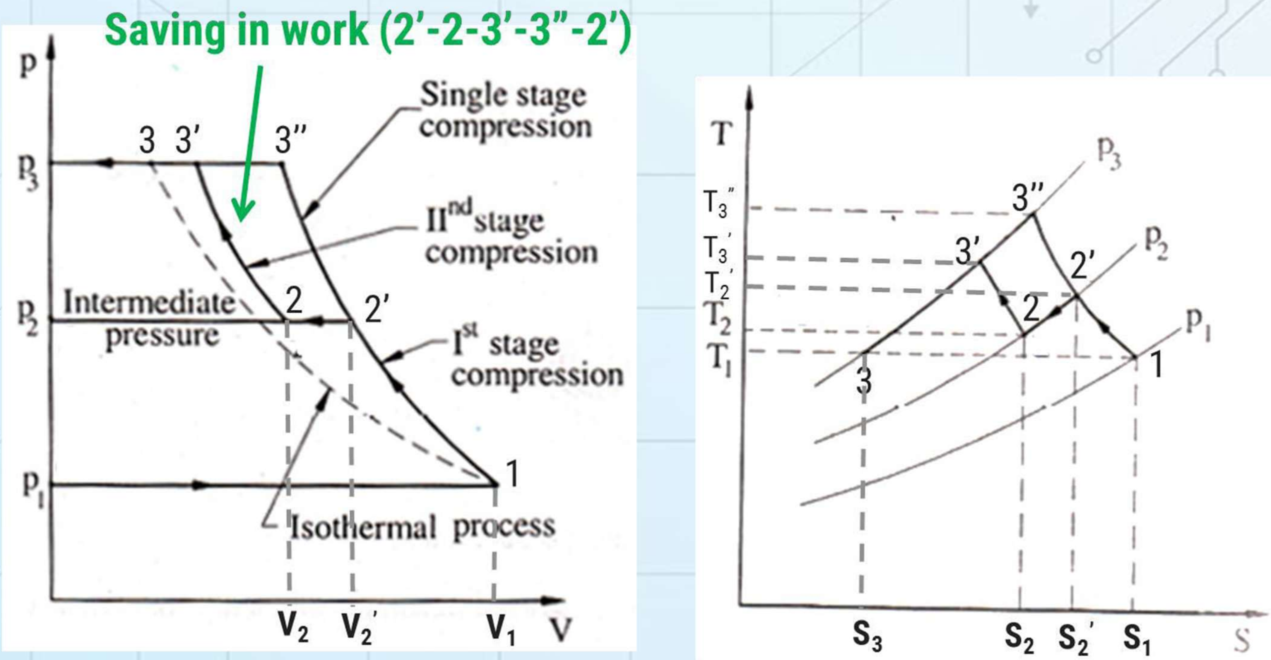


Fig.7.4 – P-V and T-S diagram of two stage compressor with intercooler

- The ***saving in work input*** by the use of multi stage compression with intercooling is shown by shaded area ***2' - 2 – 3' - 3'' – 2'***.
- Both the L.P and H.P cylinders are mounted on the same crank shaft and are driven by an electric motor or prime mover.

7.6 Work Done Equation For Two Stage Reciprocating Air Compressor With Intercooler

For calculating the workdone in a multistage compressor the following assumptions are made:

1. The effect of clearance is neglected.
2. The index 'n' in the polytropic compression law $PV^n = C$ is same for each cylinder.

3. The intercooling in each stage is at constant pressure and there is no pressure drop between two stages (i.e. delivery pressure of one stage equals the suction pressure of the next stage.)
4. The mass of air handled by the L.P and H.P cylinders is the same.
5. Suction and delivery pressures remain constant during each stage.

- Consider a two stage reciprocating air compressor with intercooler. Then,
- Let P_1 = pressure of air entering the L.P. cylinder
  V_1 = volume of L.P. cylinder = stroke volume of L.P. cylinder
  P_2 = Pressure of air leaving the L.P. cylinder or entering the H.P. cylinder V_2 = Volume of the H.P. cylinder = stroke volume of the H.P. cylinder.
  P_3 = Pressure of air leaving the H.P. cylinder

7.6.1 Case-I When The Intercooling Is Imperfect Or Incomplete

- If the temperature of air leaving the intercooler, T_2 is greater than the original atmospheric air temperature T_1 then the intercooling is called imperfect or incomplete cooling as shown in Fig. 7.5.

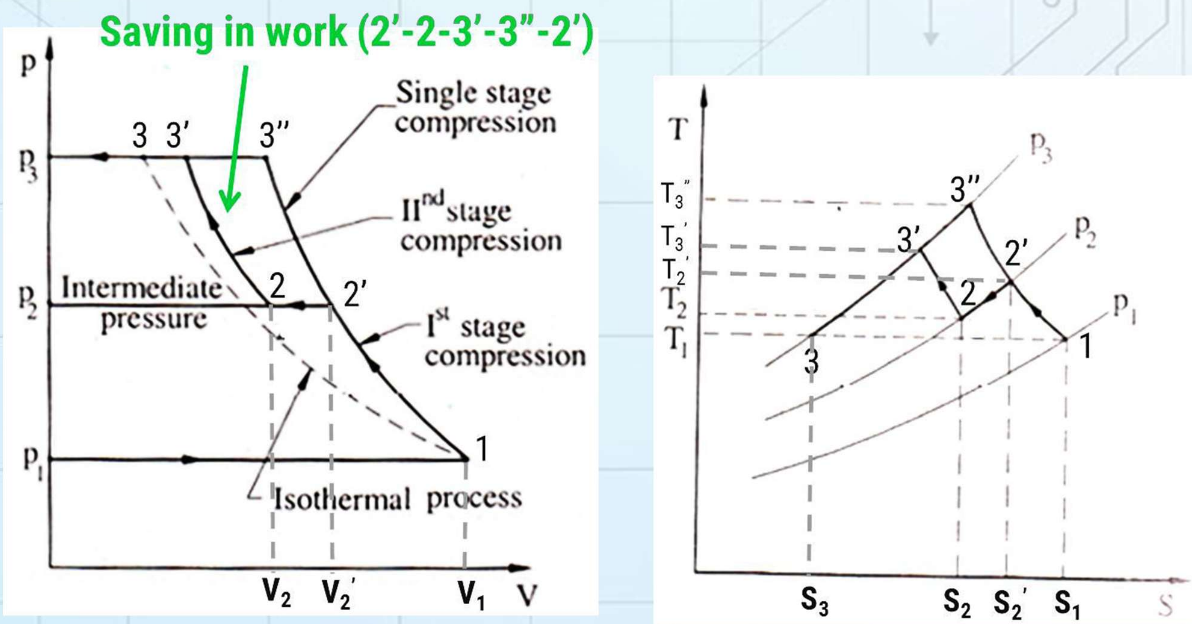


Fig. 7.5 – P-V and T-S diagram with incomplete intercooling of air

- The ***work saved*** due to imperfect (incomplete) intercooling is shown by the shaded area ***2'-2-3'-3''-2'.***
- In this case point '2' lies on the right side of isothermal curve as shown in Fig. 7.5.
- Work done required per cycle in the L.P cylinder,

$$W_{LP} = \frac{n}{n-1} P_1 V_1 \left[\left(\frac{P_2}{P_1} \right)^{\frac{n-1}{n}} - 1 \right]$$

Work done required per cycle in the H.P cylinder,

$$W_{HP} = \frac{n}{n-1} P_2 V_2 \left[\left(\frac{P_3}{P_2} \right)^{\frac{n-1}{n}} - 1 \right]$$

Total work done required per cycle in both the cylinders is,

$$W = W_{LP} + W_{HP}$$

$$W_{2-stage} = \frac{n}{n-1} P_1 V_1 \left[\left(\frac{P_2}{P_1}\right)^{\frac{n-1}{n}} - 1\right] + \frac{n}{n-1} P_2 V_2 \left[\left(\frac{P_3}{P_2}\right)^{\frac{n-1}{n}} - 1\right] \qquad \text{Eq. (7.7)}$$

7.6.2 Case-II When The Intercooling Is Perfect Or Complete

- If the temperature of air leaving the intercooler T_2 is equal to the oiginal atmospheric air temperature T_1, then the intercooling is called complete or perfect and in this case point 2 lies on isothermal curves shown in Fig. 7.6.
- The ***work saved*** due to perfect (complete) intercooling is shown by the shaded area ***2'-2-3'-3''-2'*** in both the cases and the amount of work saved with incomplete intercooling is less than that in case of complete intercooling.

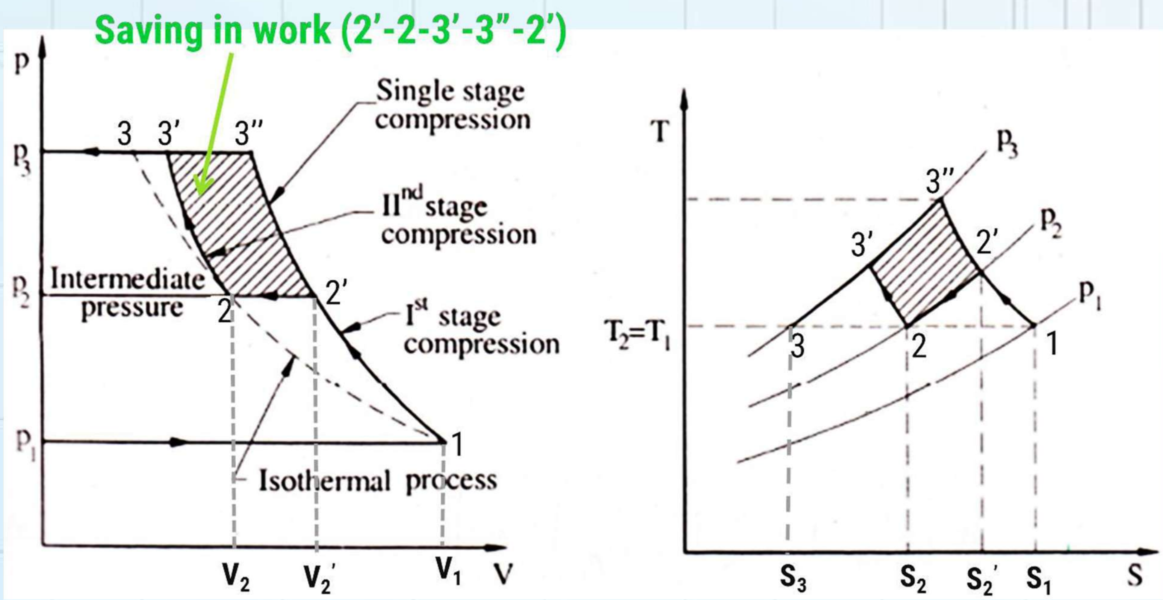


Fig.7.6 – P-V and T-S diagram with complete intercooling of air

- Total work done per cycle in both the cylinders is,

$$W = \frac{n}{n-1} P_1 V_1 \left[\left(\frac{P_2}{P_1}\right)^{\frac{n-1}{n}} - 1\right] + \frac{n}{n-1} P_2 V_2 \left[\left(\frac{P_3}{P_2}\right)^{\frac{n-1}{n}} - 1\right]$$

But for perfect intercooling , $P_1V_1 = P_2V_2$

By substituting $P_1V_1 = P_2V_2$ in above equation,

$$W = \frac{n}{n-1} P_1 V_1 \left[\left(\frac{P_2}{P_1}\right)^{\frac{n-1}{n}} + \left(\frac{P_3}{P_2}\right)^{\frac{n-1}{n}} - 2\right] \qquad \text{Eq. (7.8)}$$

7.7 Condition For Minimum Work Or Maximum Efficiency For Two Stage Reciprocating Compressor

- The total work required per cycle in two stage reciprocating compressor with complete intercooling is given by,

$$W = \frac{n}{n-1} P_1 V_1 \left[\left(\frac{P_2}{P_1}\right)^{\frac{n-1}{n}} + \left(\frac{P_3}{P_2}\right)^{\frac{n-1}{n}} - 2 \right]$$

- Let, $\frac{n-1}{n} = y$

$$W = \text{Constant} \left[\left(\frac{P_2}{P_1}\right)^{y} + \left(\frac{P_3}{P_2}\right)^{y} - 2 \right]$$

$$= \text{Constant} \left[P_2^{y} \times \frac{1}{P_1^{y}} + P_3^{y} \times P_2^{-y} - 2 \right]$$

- For minimum work,

$$\frac{dW}{dP_2} = \text{Constant} \left[\frac{y P_2^{y-1}}{P_1^{y}} - y P_3^{y} P_2^{-y-1} - 0 \right] = 0$$

$$\therefore \frac{y P_2^{y-1}}{P_1^{y}} = \frac{y P_3^{y}}{P_2^{y+1}}$$

$$P_2^{y-1} \cdot P_2^{y+1} = P_1^{y} P_3^{y}$$

$$P_2^{2y} = \left(P_1 P_3\right)^{y}$$

$$P_2 = \sqrt{P_1 P_3} \quad \text{or} \quad \frac{P_2}{P_1} = \frac{P_3}{P_2} \qquad \textit{Eq. (7.9)}$$

- Above equation shows that for minimum work required, the intercooler pressure is geometric mean of the initial ***and final pressures*** or pressure ratio in each stage is the same.

- Substitute the $\frac{P_2}{P_1}$ for $\frac{P_3}{P_2}$ in equation of work done,

Total minimum work required per cycle,

$$W = \frac{n}{n-1} P_1 V_1 \left[\left(\frac{P_2}{P_1}\right)^{\frac{n-1}{n}} + \left(\frac{P_2}{P_1}\right)^{\frac{n-1}{n}} - 2 \right]$$

$$W = \frac{2n}{n-1} P_1 V_1 \left[\left(\frac{P_2}{P_1}\right)^{\frac{n-1}{n}} - 1 \right] \qquad \textit{Eq. (7.10)}$$

Substitute the $\frac{P_2}{P_1} = \left(\frac{P_3}{P_1}\right)^{1/2}$ in above equation of work done,

$$W_{\min} = \frac{2n}{n-1} P_1 V_1 \left[\left(\frac{P_3}{P_1}\right)^{\frac{n-1}{2n}} - 1 \right]$$

- For $'x'$ number of stages, the pressure ratio is,

$$\frac{P_2}{P_1} = \frac{P_3}{P_2} = \frac{P_4}{P_3} \cdots\cdots\cdots\cdots\cdots\cdots \frac{P_{x+1}}{P_x} = \left(\frac{P_{x+1}}{P_1}\right)^{1/x} = \left(\frac{\text{Final pressure}}{\text{Initial pressure}}\right)^{\frac{1}{\text{No. of stages}}}$$

- Minimum work for X number of stages with perfect intercooling at all stages is given by,

$$W = \frac{xn}{n-1} P_1 V_1 \left[\left(\frac{P_{x+1}}{P_1}\right)^{\frac{n-1}{xn}} - 1 \right] \qquad \textit{Eq. (7.11)}$$

Where $'x' =$ number of stages; P_1 = initial or suction pressure; P_{x+1} = final delivery pressure and $'n' =$ polytropic index

- **Conditions for the minimum work required are as follow:**
  - The air is cooled to the initial temperature after each stage of compression.
  - The pressure ratio in each stage is the same.
  - Work done in all stages is equal.
  - The temperature ratios and maximum temperature are same in each stage.

7.8 Terminology of Compressors

Following terms are worthwhile to discuss for compressors:

1. **Single Acting Compressor:** In single acting compressor; suction, compression and delivery of air takes place on one side of the piston and there is one delivery stroke per rev. of crank shaft.
2. **Double Acting Compressor:** In double acting compressor; suction, compression and delivery of air takes place on one both the sides of the piston and there are *two delivery strokes* per revolution of the crank shaft.
3. **Single Stage Compressor:** In single stage compressor compression of air from intake pressure to final pressure delivery pressure takes place in one cylinder.
4. **Multi Stage Compressor:** In multi stage compressor compression of air from intake pressure to final pressure delivery pressure is carried out in more than one cylinder.
5. **Compression ratio (Pressure ratio):** It is the ratio of absolute discharge pressure to the absolute inlet pressure.
6. **Swept Volume:** It is the volume swept in the cylinder; when piston moving from TDC to BDC during one stroke. For a single acting compressor; swept volume is given by, $V_s = \frac{\pi}{4} D^2 \times L$ and For a double acting compressor; it is the volume wept by both the sides of the piston and it is given by, $V_s = 2 \times \frac{\pi}{4} D^2 \times L$

7. **Free Air Delivered (FAD):** It is the actual volume of air delivered by a compressor reduced to the surrounding atmospheric pressure (i.e 1.01325 bar abs.) and temperature (i.e 15^0C)
8. **Capacity:** It is the quantity of free air actually delivered by the compressor in cubic meter per minute.
9. **Indicated Power:** It is the power required to compress air from intake conditions to discharge conditions and work done per cycle = area of the indicator diagram. Indicated power is determined by taking indicator diagram with the instrument called indicator.
10. **Brake Power:** It is the power delivered to the compressor shaft or the power required to drive the compressor.
11. **Mechanical Efficiency:** It is defined as the ratio of the indicated power to the brake power. It usually varies from 85% to 95%.
12. **Adiabatic Efficiency:** It is defined as the ratio of the polytropic (i.e actual) work input to the adiabatic work input.
13. **Isothermal Efficiency:** It is defined as the ratio of the isothermal work input to the polytropic (i.e actual) work input.
14. **Overall Isothermal Efficiency:** It is defined as the ratio of isothermal power to the shaft power (i.e brake power) of the prime mover required to drive the compressor.

7.9 Control of Compressors

Generally, compressors are not required to run continuously at their maximum rated capacity but also required to suply at a fraction of their rated or maximum capacity according to the requirement.

As requirement may continously vary, it is necessory to incoporate some methods to control the amount of air delivered.

Capacity control comprises the adjustment for the quantity of air to be supplied by reciprocating compressor to meet the fluctuating demand, so that the discharge pressure varies within acceptable limits.

There are two main methods of control:

(i) Variable speed and (ii) Constant speed

7.9.1 Variable Speed

- It is simplest method of controlling the amount of air delivered.
- It is best suited for the compressors that are driven by steam engine, internal combustion engines and electric commutator motor.
- But, these are rarely used due to their high price and sensitivity.

7.9.2 Capacity Control At Constant Speed

- It is suitable for the compressors driven by electric synchronous motor and capacity control by speed variation is not possible.
- The different methods are as mention below:
    1) Throttling the suction line
    2) Increasing the clearnace volume by Clearance pockets
    3) Blowing off to waste
    4) Cylinder ports
    5) By passing air from delivery to suction inlet
    6) Temporary unloading of suction valve

7) Automatic stop-start control
8) Cylinder cut-out

- Each of the above method discussed in detail as below:

7.9.2.1 *Throttling The Suction Line*

- When the demand is less, the pressure in the receiver increases and high pressure air from the reciever led to cylinder. The movement of the piston is resisted by the spring, with excessive pressure the piston depresses the spring, thus closing partly the suction valve, till the supply and demand are balanced. The reverse action takes place when the pressure in the receiver falls due to increased demand.

7.9.2.2 *Increasing The Clearnace Volume By Clearance Pockets*

- The clerance volume can be increased or reduced by providing clerance pockets.
- The volumetric efficiency is reduced in proportion to control the output.
- When the receiver pressure exceeds the desired the desired amount, a dead weight is raised by diaparagm releases air to pressure chamber. Thus, the spring lifts the valve and clearance space is put in communication with the cylinder. If the pressure still continue to rise, a scond valve puts another pocket of clarnce space in communication till the supply and demand are balanced.
- The use of clearance pockets results in a reduction of power in proportion to load reduction, because the work is recovered by re-expansion.

7.9.2.3 *Blowing Off To Waste*

- In this method, there is a by pass valve from the high pressure cylinder delivering air direct to atmoshphere when the receiver pressure exceeds a predetermined value due to decrease in demand.
- The high pressure air froces the relay upwards against the resistance of a dead weight until the port through which it is connected is uncovered, thus the high pressure air forces the piston downward and the air escaping to atmoshphere.
- When the pressure in the receiver falls, the relay piston moves down by the dead weight and closes the communication between relay cylinder and cylinder B.

7.9.2.4 *Cylinder Ports*

- The stroke of compressor decreases by use of ports in the cylinder. The ports connected back to the compressor intake and air allowed to escape from the cylinder at beginning of compression stroke.
- There is loss of power by use of ports. The ports may be controlled manually or automatically.

7.9.2.5 *By Passing Air From Delivery To Suction Inlet*

- Th output is controlled by passing the cooled air from delivery line to suction line.
- The stop valve on he by pass line is actuated by an automatic air governor which is connected to the receiver. In some cases the delivery line is connected by a throttle valve to the suction line.

7.9.2.6 *Temporary Unloading Of Suction Valve*

- It is most efficient method because the passage of air in or out of the cylinder through the suction valve without compression involve less loss.

- It also permits the unloading of individual cylinder.

7.9.2.7 *Automatic Stop-Start Control*

- This control is advantageous when the compressor is driven by electric motor and demand for compressed air fluctuates widely.
- When the receiver pressure reaches certain specified value the power supply to drive motor is switched off by a pressure switch connected to the receiver. When the pressure in the receiver falls to pre-set value the power supply to the motor is restored.

7.9.2.8 *Cylinder Cut-Out*

- In multi cylinder compressors, one or more cylinders by by-passing the air from discharge to intake of compressor.
- The check valve in the cylinder discharge by-pass connection to separate the inactive cylinders from active cylinders.
- There is no decrease of power in proportional to output because some power is required by inactive cylinder to make the flow of air through valves, cylinders and connections.

7.10 Referances

1. Engineering Thermodynamics by P.K. Nag, McGraw-Hill Education.
2. Turbines, Compressors and Fans by S.M. Yahya., TMH Publishers.
3. Fluid Power Engineering by V.L. Patel, Dr. R.N. Patel, Mahajan Publication House.

UNIT – 7 PART - II

(Centrifugal Compressors)

Contents

Important Repeated Questions:

1. **Explain the phenomenon of surging and choking/stalling in a centrifugal/axial flow compressor.**
(S25 - Q5 OR b, 04 marks) (W25 - Q5b, 04 marks) (S24 - Q5a) (W23 - Q5a, 03 marks) (W22 - Q3b)
2. **Differentiate between centrifugal and axial flow compressor.**
(S25 - Q3b, 04 marks) (S23 - Q4a, 03 marks) (W22 - Q3 OR a, 03 marks) (W23 - Q2 OR c, 07 marks)
3. **Explain various losses in centrifugal compressor.**
(W25 - Q4 OR c, 07 marks) (S23 - Q4 OR b, 04 marks)
4. **Explain effect of pre-whirl in centrifugal compressor.**
(W25 - Q5a, 03 marks) (S22 - Q5a, 03 marks)

Legends: W- Winter, S- Summer, Q- Question and 03/04/07- Marks of Question

7.1 Introduction

Centrifugal compressors are turbo-machines employing centrifugal effect to increase the pressure of fluid.

In centrifugal compressors energy is transferred by dynamic means from a rotating impeller to the continuously flowing fluid.

The main feature of the centrifugal compressors is that the angular momentum of the fluid flowing through the impeller is increased partly by virtue of the impeller outlet diameter being significantly larger than its inlet diameter.

A pressure ratio in the order of 4:1 can be obtained from a single stage compressor manufactured using conventional materials.

7.1.1 Characteristics Features of Centrifugal Compressors

- It occupies a smaller length than an equivalent axial flow compressor.
- It has better resistance to foreign object damage.
- Because of the relatively short passage length, loss of performance due to build-up deposits on blade surfaces will not be as great as the axial flow compressors.
- It can work reasonably well in a contaminated atmosphere as compared to axial flow compressor.
- It has ability to operate over a wide range of mass flow rate at any particular rotational speed.
- Its efficiency under the most favourable circumstances, are less than those of axial compressors designed for the same duty, by as much as 3 or 4 %.
- However, at very low mass flow the axial flow compressor efficiency drops, blading is small and the advantage appears to lie with the centrifugal compressor in its relative simplicity and cost.
- The advent of titanium alloys, permitting much higher tip speeds, combined with advances in aerodynamics now permit pressure ratios of greater than 8:1 to be achieved in a single stage.

7.1.2 Applications of Centrifugal Compressors

- Gas pumping in long distance pipe line and in petro-chemical industries.
- Large scale refrigeration plants and big central air conditioning plants.
- Fertilizer industry.
- Supercharging of I. C. Engines.

7.2 Essential Parts of Centrifugal Compressor

The principal components of a centrifugal compressor are shown in Fig. 7.1 and detail of each part is given below.

1. Inlet casing with accelerating (converging) nozzle
2. Inlet guide vanes (IGV)
3. Impeller
4. Diffuser
5. Scroll or volute
6. Inducer section

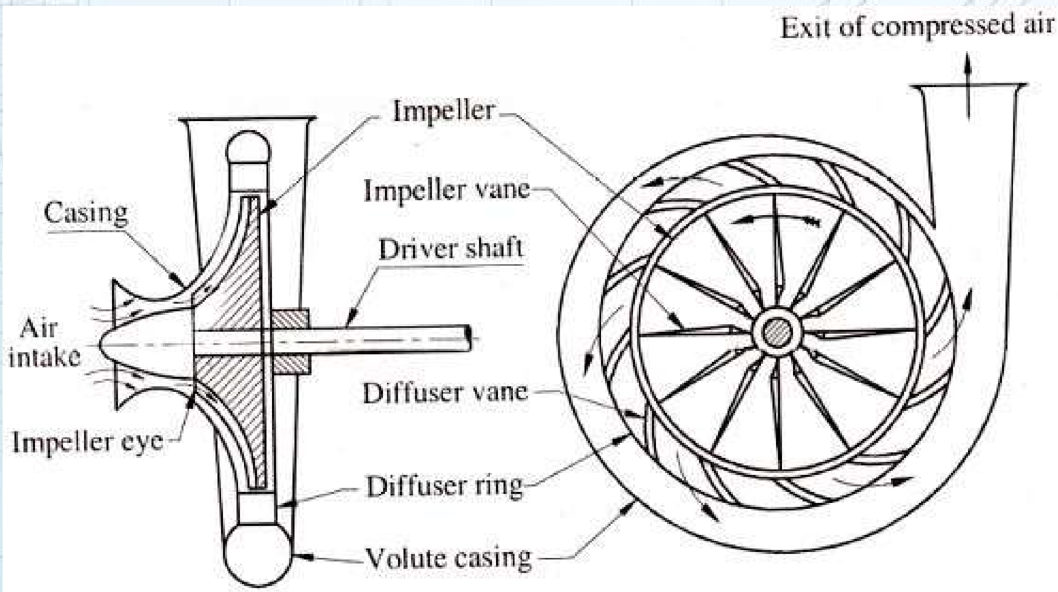


Fig.7.1 – Centrifugal compressor

1) Inlet Casing With Accelerating Nozzle

- The function of inlet casing is to accelerate the fluid from its initial condition to the entry of inlet guide vanes and to provide uniform velocity at the eye.
- The inlet flange is axisymmetric and the inlet duct takes the form of a simple converging nozzle.
- The outlet of the inlet casing is known as the ***impeller eye***.

2) Inlet Guide Vanes (IGV)

- Its function is to direct the flow in the desired direction at the entry of the impeller.
- The inlet guide vanes should be chosen so as to obtain a minimum relative Mach number at the eye tip.

3) Inducer Section

- At entry to the impeller the relative flow has a velocity Vr_1, at angle α_1 to the axis of rotation.
- This relative flow is turned into the axial direction by the inducer section or rotating guide vanes.
- The inducer starts at the eye and usually finishes in the region where the flow is beginning to turn into the radial direction.

4) Impeller

- The function of the impeller is to increase the energy level of fluid by whirling it outwards by increasing the **angular momentum** of the fluid.
- Both static pressure and velocity of fluid are increased in the impeller. The impeller vanes help to transfer the energy from the impeller to the fluid.
- The **hub** is the curved surface of revolution of the impeller. The ***shroud*** is the curved surface forming the outer boundary to the flow of fluid. (*Shrouding an impeller eliminate tip leakage losses but at the same time increases friction losses*.)
- Impellers may be enclosed by having the shroud attached to the vane ends (called shrouded impellers) or unenclosed with small clearance gap between the vane ends and the stationary wall.

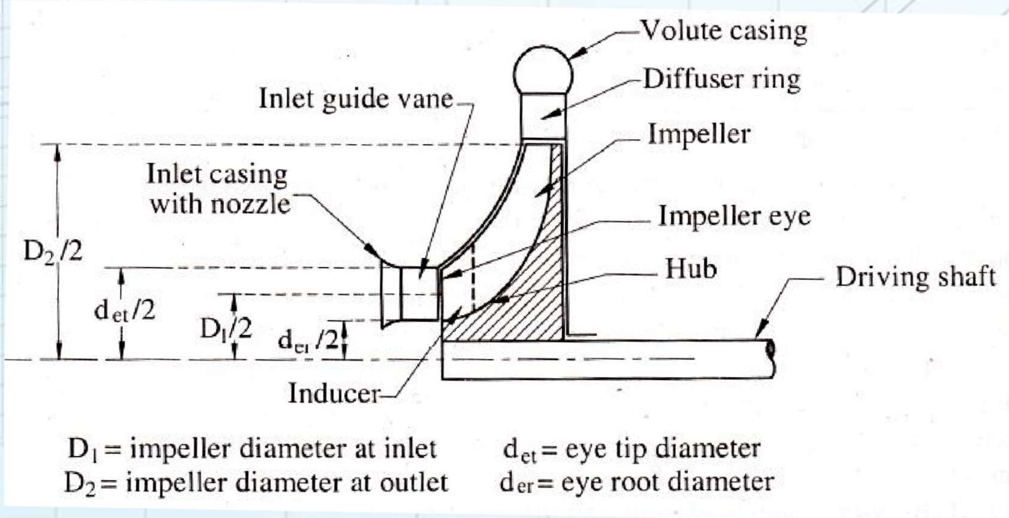


Fig.7.2 – Centrifugal compressor stage

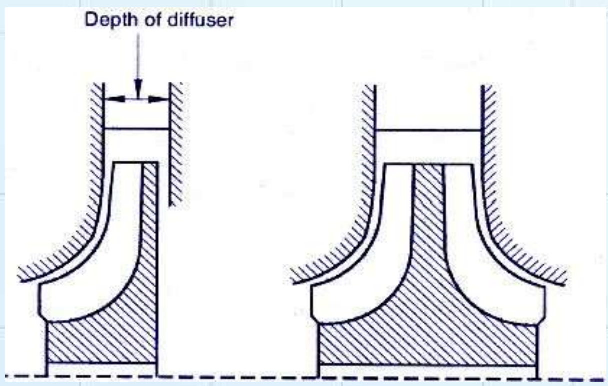


Fig.7.3 – Single sided and double sided impeller

- The impeller may be a single sided or double sided as shown in Fig. 7.3.
- In a single sided impeller, air enters in to the compressors from one side only. In double sided impeller, there is an eye on either side of the impeller and air enters from both the sides and the advantage is the impeller is subjected to approximately **equal stresses** in the axial direction.

5) Diffuser

- The function of the diffuser is to convert the high kinetic energy of the fluid at impeller outlet into static pressure.

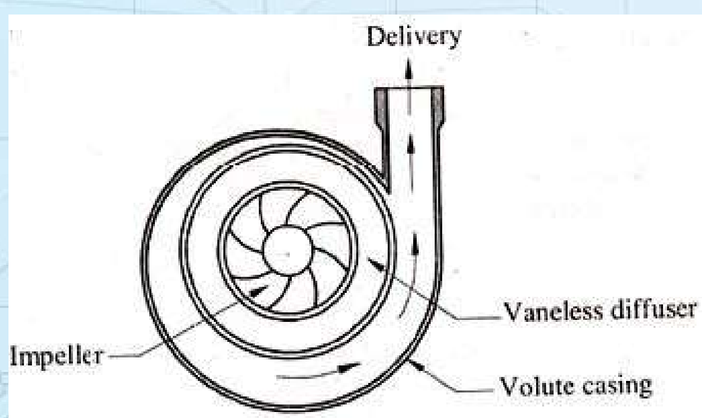


Fig.7.4 – Vaneless diffuser

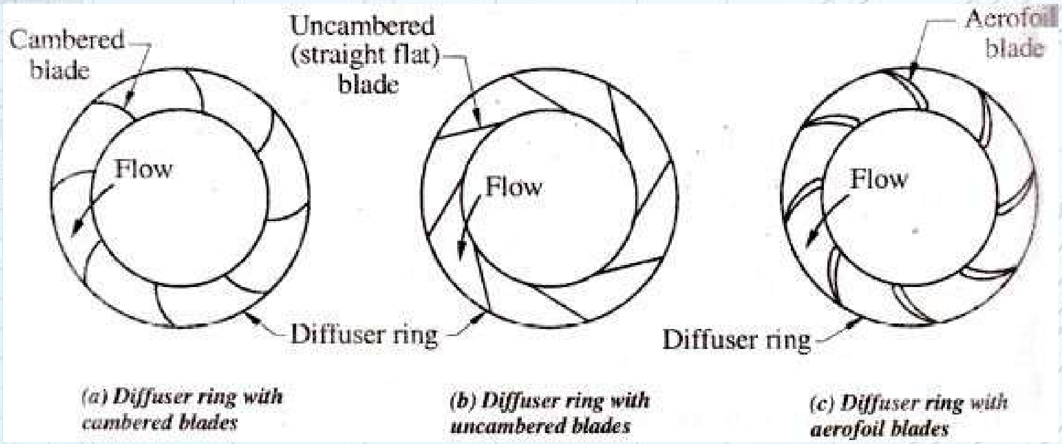


Fig.7.5 – Vanned diffuser

- As the impeller imparts energy to the air by increasing its velocity; and the diffuser converts this imparted kinetic energy into static pressure rise.
- Diffuser is housed in a radial portion of the casing and it may be vaneless or vaned diffuser as shown in Fig. 7.4 and 7.5.

6) Scroll or Volute

- The air leaving the diffuser is collected in a spiral passage known as ***volute*** or scroll and the volute discharges the air through delivery pipe.
- Different cross sections are employed for the volute passage are rectangular, circular and trapezoidal.

7.3 Principle Of Operation Of Centrifugal Compressor

- When the impeller rotates at high speed, suction is created at the impeller eye and the air is drawn in through an accelerating nozzle.
- Due to flow acceleration in the compressor inlet part, the velocity of air is increased from V_0 to V_1, and thus pressure and temperature decrease.
- This acceleration is not isentropic but accompanied with friction. Thus P_1, and T_1, are the pressure and temperature at the inlet of the impeller.
- Due to energy supplied on the compressor shaft, the impeller is rotated at speeds of *20000 to 30000 rpm* and thus each particle of air passing through the impeller is accelerated i.e. the kinetic energy of fluid is increased.
- There are two actions takes place: (i) Diffusion action and (ii) Centrifugal action as following:

Diffusion Action

- The impeller vanes are such that the cross-sectional area between two vanes increases from inlet to outlet of the impeller and this gives rise to *diffusion action,* $\frac{Vr_1^2 - Vr_2^2}{2}$.

Centrifugal Action

- The air enters the impeller at smaller diameter and comes out at larger diameter and this gives rise to *centrifugal action,* $\frac{u_2^2 - u_1^2}{2}$.

- Thus due to diffusion and centrifugal action, a part of the kinetic energy imparted to the air is converted into static pressure and temperature rise.
- The absolute velocity V_2 of air at the impeller outlet is very high and it has to be converted into pressure energy and this conversion is achieved in the **vaneless diffuser** and **vaned diffusers**.
- The vaneless diffuser converts some part of kinetic energy into pressure energy and velocity reduces from V_2 to V_3 and it also stabilizes the flow coming out from the impeller so that the entry to the vaned diffuser is without shock.
- The rest of the kinetic energy is converted into pressure energy in the vaned diffuser and the velocity reduces from V_3 to V_4.
- The air leaving the vaned diffuser is collected in spiral passage (scroll or volute) from which it is discharged from the compressor.

7.3.1 Pressure And Velocity Variation Across A Centrifugal Compressor

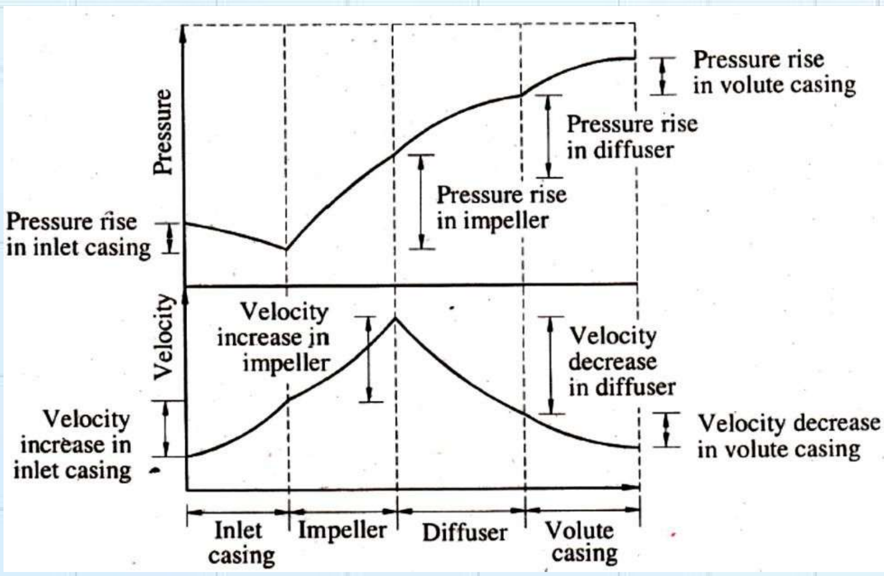


Fig.7.6 – Pressure and velocity variation across centrifugal compressor

- Air enters the compressor at mean radius with a low velocity V_1, and atmospheric pressure P_1 as shown in Fig. 7.6.
- It is then accelerated to a high velocity V_2, and pressure P_2, depending upon the centrifugal action of the impeller.
- The air now enters the diffuser where its velocity is reduced to some value V_3, and pressure increases to P_3.
- In practice, about half of the total pressure rise per stage is achieved in the impeller and the remaining half in the diffuser.

7.4 Static and Total Head Properties

In rotary compressors *high fluid velocities* are encountered and therefore total head quantities which take into account the kinetic energy have to be considered.

If the moving air is brought to rest **isentropically** without external work transfer then resulting state is known as ***total head or stagnation state*** and corresponding values of the properties describing this state are called *stagnation properties.*

- Following are the important properties to be considered:
  1) Stagnation Temperature (T_0)
  2) Stagnation Pressure (P_0)
  3) Stagnation Enthalpy (h_0)
  4) Stagnation Density (ρ_0)
- Consider a horizontal passage of varying area with no external heat transfer and work transfer.

7.4.1 Static and Stagnation Temperature (T_0)

- "It is the actual temperature of the air that would be registered by a thermometer moving with air with the same speed of the air is called ***static temperature (T)***."
- "If the moving air is brought to rest ***isentropically*** without external work transfer then the kinetic energy of the air is converted in to heat energy increasing the temperature of the air, the resulting temperature of the air is called ***stagnation temperature (T_0)***."
- Apply steady flow energy equation for 1 kg of air flow diverging passage with no heat and work transfer is given by,

$$u_1 + P_1 v_1 + \frac{V_1^2}{2} = u_2 + P_2 v_2 + \frac{V_2^2}{2}$$

But $u + Pv = h$

$$h_1 + \frac{V_1^2}{2} = h_2 + \frac{V_2^2}{2}$$

For specific heat $h = C_p T$

$$C_p T_1 + \frac{V_1^2}{2} = C_p T_2 + \frac{V_2^2}{2}$$

$$C_p T + \frac{V^2}{2} = C_p T_0 = h_0 = \text{Constant}$$

Where T_0 is known as stagnation temperature

$$T + \frac{V^2}{2C_p} = T_0$$

$$T = T_0 - \frac{V^2}{2C_p}$$

$$T_0 = T + \frac{V^2}{2C_p} \qquad \textit{Eq. (7.1)}$$

Where T is static temperature and $\frac{V^2}{2C_p}$ is called *dynamic temperature*

7.4.2 Stagnation Pressure (P_0)

- "If the moving air is brought to rest isentropically without external work transfer then the kinetic energy of the air is converted in to pressure of the air, the resulting pressure of the air is called *stagnation pressure (P_0)*."

- Stagnation pressure can be found by using following relation between pressure and temperature,

$$\frac{P_0}{P} = \left(\frac{T_0}{T}\right)^{\frac{\gamma}{\gamma-1}} \qquad \text{Eq. (7.2)}$$

7.4.3 Stagnation Enthalpy (h_0)

- "If the moving air is brought to rest isentropically without external work transfer then resulting enthalpy is known as *stagnation enthalpy*."
- The stagnation enthalpy remains constant in a moving stream in the absence of heat and work transfer.
- Stagnation enthalpy can be found by using following relation is given by,

$$h_0 = h + \frac{V^2}{2} \qquad \text{Eq. (7.3)}$$

7.4.4 Stagnation Density (ρ_0)

- "If the moving air is brought to rest isentropically without external work transfer then resulting density is known as *stagnation density*."
- Stagnation density can be found by using following perfect gas relation is given by,

$$\rho_0 = \frac{P_0}{RT_0} \qquad \text{Eq. (7.4)}$$

Also for isentropic process $\dfrac{\rho_0}{\rho} = \left(\dfrac{T_0}{T}\right)^{\frac{1}{\gamma-1}}$

7.5 Velocity Diagram Of Centrifugal Compressor

- The velocity diagrams at inlet and outlet of the impeller of a centrifugal compressor (assuming no pre-whirl and no slip) is shown in Fig. 7.7 and 7.8.
- In the analysis of centrifugal compressor the following assumptions are made:
  (i) The flow phenomenon is steady and uniform throughout.
  (ii) There is no separation of flow.
  (iii) The flow through the impeller is frictionless.
  (iv) There are no shock waves occurring any where.
- The following are the notations used in the analysis of a centrifugal compressor. Let's take,
  α_1 = Exit angle from the guide vanes at entrance = absolute angle at inlet
  β_1 = Inlet angle to the rotor or impeller
  β_2 = Outlet angle from the rotor or impeller
  α_2 = Inlet angle to the diffuser or the stator
  u_1 = Mean blade velocity at inlet
  u_2 = Mean blade velocity at exit
  V_1 = Absolute velocity of air at inlet to the rotor

V_2 = Absolute velocity of air at exit to the rotor

V_{r1} = Relative velocity of air at inlet to the rotor blade

V_{r2} = Relative velocity of air at exit to the rotor blade

V_{w1} = Velocity of whirl at inlet (tangential component of absolute velocity V_1)

V_{w2} = Velocity of whirl at exit (tangential component of absolute velocity V_2)

V_{f1} = Velocity of flow at inlet (Component of V_1 perpendicular to the plane of rotation) V_{f2} = Velocity of flow at exit (Component of V_2 perpendicular to the plane of Rotation) m = Mass flow rate, kg/sec

- If no pre-whirl, the air enters the impeller eye in an axial direction, $\alpha_1 = 90^0, V_{f1} = V_1, V_{w1} = 0$ and air will be leaving the impeller in radial direction $\beta_2 = 90^0, V_{r2} = V_{f2}$ and $V_{w2} = V_{u2}$.

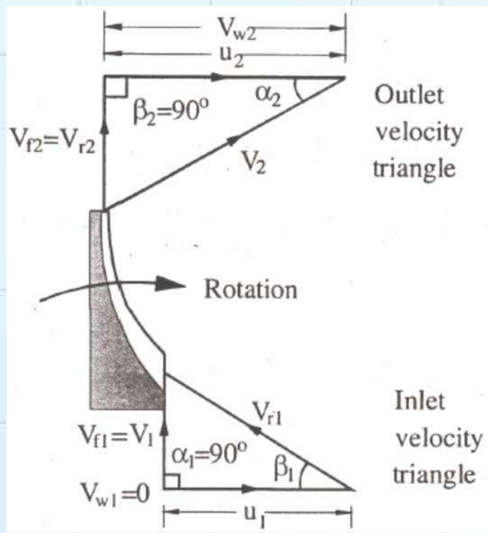


Fig.7.7 – Velocity diagrams of centrifugal compressor when $\alpha_1 = 90^0$ and $\beta_2 = 90^0$

- If the air enters the impeller eye in an axial direction $\alpha_1 = 90^0$ but air will not leaving the impeller in radial direction $\beta_2 < 90^0, V_{r2} \neq V_{f2}$ and $V_{w2} \neq V_{u2}$

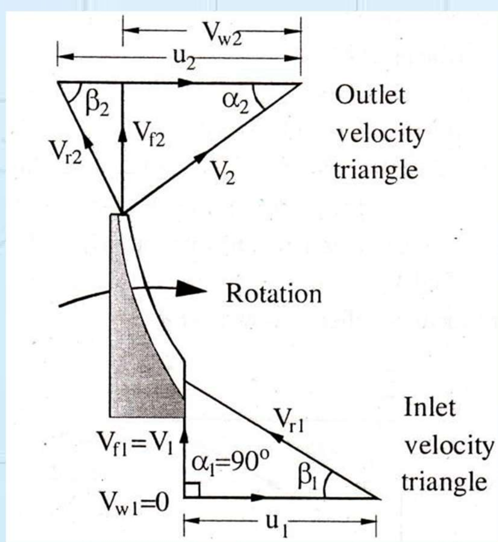


Fig.7.8 Velocity diagrams of centrifugal compressor when $\alpha_1 = 90^0$ and $\beta_2 < 90^0$

7.6 Surging And Choking in Centrifugal Compressors

7.6.1 Surging

- It is the phenomenon which is caused due to unsteady, periodic and reversal of low through the compressor when compressor has to operate at less mass flow rate than a predetermined value (a value corresponding to maximum pressure ratio).
- The theoretical pressure ratio versus mass flow rate for centrifugal compressor shown in the Fig. 7.9 which delivering air through a flow control value situated after the diffuser.
- The mass flow rate is zero when the discharge valve is fully closed and the pressure ratio will have some value represented by point 'A'.
- The pressure at this condition is equal to the centrifugal pressure head produced by the action of the impeller on the air trapped between the vanes.
- As the discharge valve is open slightly, mass flow commences and the diffuser becomes effective in increasing the static pressure and the pressure ratio increases as shown at 'B'.
- With further opening of the discharge valve, the pressure ratio goes on increasing till the maximum pressure rise is attained represented by the point 'C'; at this point maximum efficiency achieved for the given speed, inlet pressure and temperature.
- With further opening of the discharge valve, the mass flow rate increases beyond the point 'C', the efficiency of the compressor decreases with a corresponding decrease in pressure ratio.
- For mass flows greatly in excess of the designed mass flow, the air angles become widely different from the vane angles, causing break away of the air (flow separation and shock occur) and the efficiency decreases rapidly.
- Finally when the discharge valve is fully opened, the pressure ratio approaches unity, the mass flow rate would be maximum but the compression efficiency will be zero. *(All the power is absorbed in overcoming internal frictional resistance).*

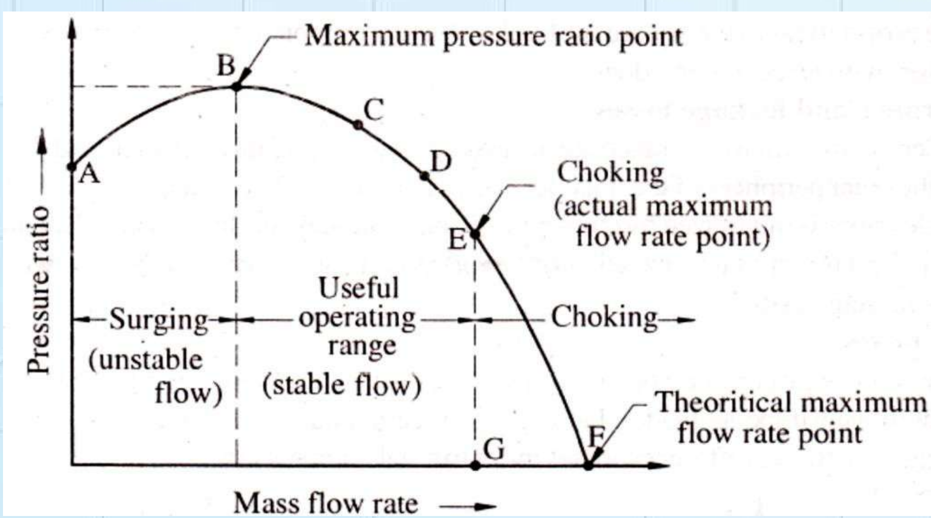


Fig.7.9 – Surging and chocking in centrifugal compressor

- Let the compressor operate to the left of the point 'C', at some point 'B' and at this point decrease in mass flow rate is accompanied by decrease in pressure developed by the compressor.
- If the static pressure in the discharge line does not decrease as rapidly as the developed pressure, there is a natural tendency for the air to flow back into the compressor in the direction of the pressure gradient.

- This will result momentarily in the ***reversal of flow***; and after a lapse of time the fluid from the delivery pipe would leave, the pressure downstream of the compressor falls, the pressure gradient is reversed and again the air flow is back to its normal direction.
- The pressure therefore surges back and forth, if the downstream conditions are unchanged; then this phenomenon is known as *surging* or pumping.
- Thus, when, the compressor has to operate at **reduced mass flow rates**, the air surges and pulsates throughout the compressor and the compressor does not give a steady flow of air.
- The region between the points A and C is known as ***surge region*** and it is objectionable to operate the compressor in this region.
- Surging does not take place in the region CF as the reduction in mass flow is accompanied with increase in pressure and of tire flow is not possible and the stability of the operation is maintained.

Effects of Surging

- Surging could lead to failure of the compressor parts.
- Surging causes noise, vibration, overheating and stress reversal in the vanes and may damage the compressor.

Remedies of Surging

- Surging can be reduced by making the number of diffuser vanes an odd-number multiple of the impeller vanes.
- Thus, a pair of diffuser passages will be supplied with air from an odd number of vanes and pressure fluctuations are evened out around the circumference than if exact multiple of diffuser vanes are employed.

7.6.2 Chocking

- "The maximum mass flow rate possible in a compressor is known as ***choking* flow**" OR the fixed mass flow rate regardless of the pressure ratio.
- At constant rotor speed, the tangential velocity component at the impeller remains constant.
- With the increase in mass flow rate, at the right of maximum pressure point, the pressure ratio decreases and hence the density is decrease.
- These effects result in a considerable increased redial velocity which increases the absolute velocity and the incidence angle at the diffuser vane tip and there is a rapid progression towards a choking state.
- The slope of the characteristic steepens and finally vertical (i.e. the mass flow cannot be increased further); thus the phenomenon of choking puts an upper limit on the mass flow.
- The point 'E' on the characteristic curve is called the ***choking point*** as shown in Fig. 7.9.
- Choking indicates that at some point within the compressor **sonic conditions** have been reached causing; the limiting **maximum mass flow rate** to be set as in the case of compressible flow through a converging – diverging nozzle.

7.7 Losses in Centrifugal Compressor

The total losses in a centrifugal compressor may be divided into:

1. Frictional losses
2. Inlet losses
3. Incident losses
4. Clearance and leakage losses

The power supplied to the centrifugal compressor stage is the power input at the coupling less the mechanical losses on account of the bearing, seal and disc friction as shown in Fig. 7.10.

These losses result from fluid friction, separation, circulatory motion and shock wave formation and losses lead to an increase in enthalpy and decrease in stagnation pressure.

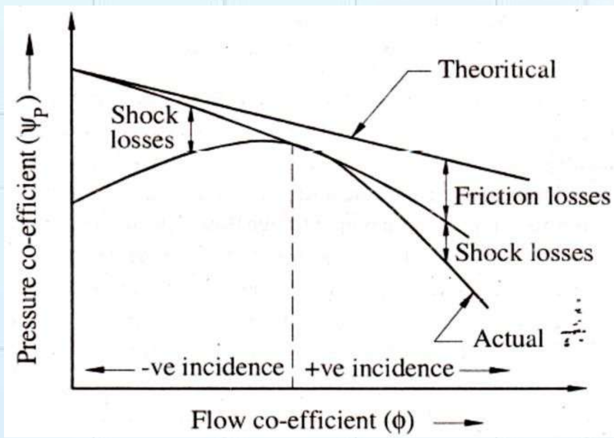


Fig.7.10 – Losses in centrifugal compressor

7.7.1 Frictional Losses

- The major portion of the losses is due to fluid friction in stationary and rotating blade passages.
- Losses due to friction depend on friction factor, passage length and the square of fluid velocity (V^2) and hence proportional to m_2.
- Therefore the stage with relatively larger impeller, diffuser and volute passages and higher fluid velocity will have poor performance.

7.7.2 Inlet Losses

- In centrifugal compressors fluid enters axially and turns radially in the vaneless space before entering the impeller blades.
- In this process the fluid suffers losses and these losses depend on velocities V_i and V_1.
- Inlet losses increase due to change in mass flow designed conditions. When mass flow changes the ratio of axial velocity of flow to blade velocity also changes.
- The aerodynamic losses occurring in the stage during the flow processes from its entry to exit are taken into account by the stage efficiency.
- The actual temperature of air coming out from the compressor is higher than the temperature of air compressed isentropically.
- The flow, except in accelerating nozzle and inlet guide vanes is throughout decelerating; thus the thickening boundary layer separates; where the adverse pressure gradient is too steep and leads to additional losses on account of stalling and wasteful expenditure of energy in vortices.

7.7.3 Incident Losses

- During the off design conditions the flow at the entry of the impeller and diffuser blades approaches them with some degree of incidence as shown in Fig. 7.11.

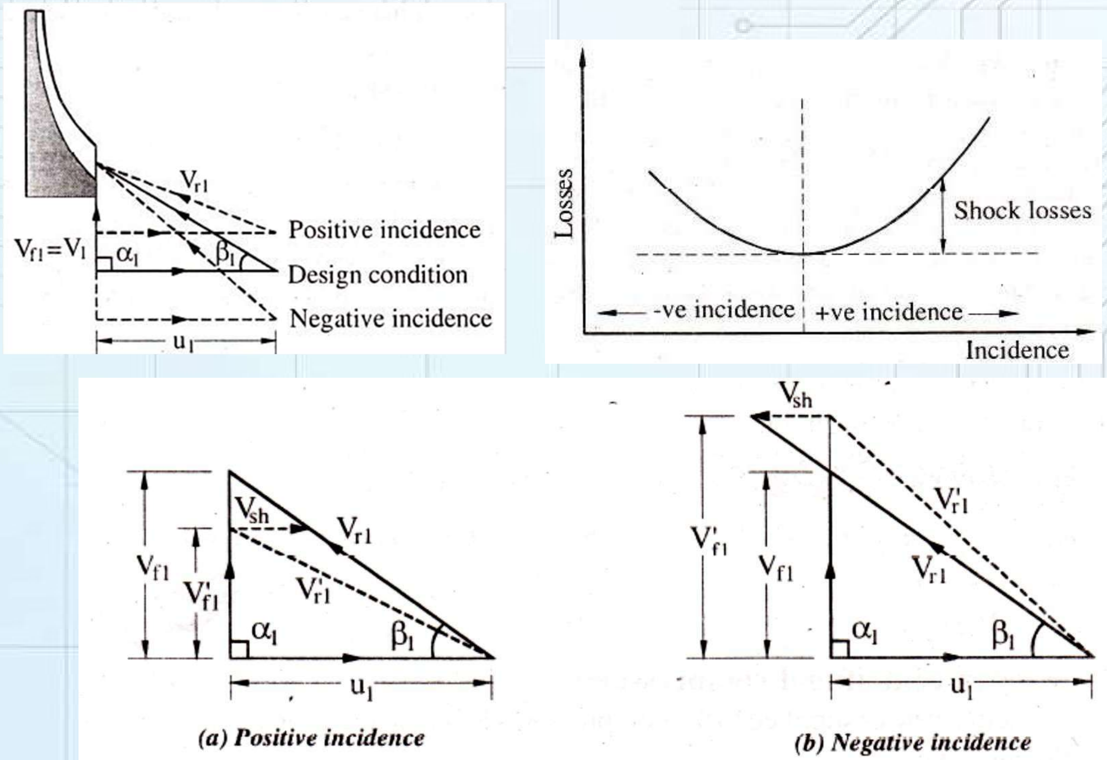


Fig.7.11 – Effect of incidence on velocity diagrams and variation of shock losses

- At the same rotational speed the reduced flow rate introduces **positive incidence** whereas **negative incidence** results from increased flow rate; shock losses are increase rapidly at large value of incidence.
- Large positive incidences lead to flow separation, stalling and surge. Incidence losses in terms of drag coefficient C_D are proportional to C_D V_2.
- Shock losses also occur in the diffuser and volute.

7.8 Comparison Between Reciprocating And Centrifugal Compressors

Sr. No.	Parameters	Reciprocating	Centrifugal
1.	Balancing	Poorly balanced and vibration problem occurs.	Better balanced; because no reciprocating part.
2.	Mechanical efficiency	Less efficiency due to more sliding and bearing members	More due to less bearing members
3.	Pressure ratio	Pressure ratio per stage is high about 5 to 8.	Pressure ratio per stage is high about 3 to 4.5.
4.	Initial cost	High	Less

	Delivery pressure	Capable to deliver high pressure.	Capable to deliver medium pressure.
	Capacity	Handles small volume	Handles large volume
	Flexibility	More flexible with capacity and pressure range.	No flexibility in capacity and pressure range.
	Compression efficiency	Higher at compression ratio above 2.	Higher at compression ratio below 2.
	Speed	Adaptability to low speed drive	Adaptability to high speed low maintenance cost driver such as turbine.
	Maintenance cost	High	Low
	Suitability	For low, medium and high pressure and low and medium gas volumes.	For low and medium pressure and large gas volumes.

7.9 Referances

1. Engineering Thermodynamics by P.K. Nag, McGraw-Hill Education.
2. Turbines, Compressors and Fans by S.M. Yahya., TMH Publishers.
3. Fluid Power Engineering by V.L. Patel, Dr. R.N. Patel, Mahajan Publication House.

UNIT – 7 PART - III

(Axial Flow Compressors)

Contents

Important Repeated Questions:

1. **Define/Explain: Mach number, Mach angle, Mach cone, zone of action, zone of silence.** (S25 - Q3 OR a, 03 marks) (W23 - Q1b/Q2b, 04 marks) (S23 - Q4 OR a, 03 marks)
2. **Explain Fundamental equations for compressible flow.** (S25 - Q5c, 07 marks) (W23 - Q2c, 07 marks)
3. Derive the following from one dimensional steady flow energy equation and also explain various regions of flow based on it: $\frac{a^2}{\gamma-1}+\frac{V^2}{2}=\frac{a_0^2}{\gamma-1}=\frac{V_{max}^2}{2}=\frac{a^{*2}\gamma+1}{2\gamma-1}=h_0$ (S22 - Q2 OR c)
4. What is the effect of compressibility on Mach number? Prove for $\gamma=1.4\,\frac{P_0-P}{\frac{1}{2}\rho V^2}=1+\frac{M^2}{4}+\frac{M^4}{40}+\cdots$ ... ... (S22 - Q2c)
5. What are the stagnation Properties? Derive an equation for Stagnation Pressure and Stagnation Density. (W22 - Q5 OR c, 07 marks)
6. With a suitable sketch explain the working principle of an axial flow compressor. What is meant by a stage and explain the stage velocity triangles. (W23 - Q5b, 07 marks)
7. Explain the use of aerofoil blading in axial flow compressor. (W25 - Q5 OR c, 07 marks)
8. What do you mean by lift and drag? (S22 - Q5 OR a, 03 marks)
9. Define flow co-efficient and work co-efficient with reference to axial flow compressor (S24 - Q5 OR a)
10. An axial flow compressor stage has mean diameter of 60cm and runs at 15000rpm . If the actual temperature rise and pressure ratio developed are 30°C and 1.35 respectively. Determine: (1) power required to drive the compressor while delivering 57kg/s of air, if mechanical efficiency is 86% and inlet temperature 35°C (2) the stage loading co-efficient (3) the stage efficiency (4) the degree of reaction if the temperature at the rotor exit is 55°C . (S24 - Q5 OR c)

Legends: W- Winter, S- Summer, Q- Question and 03/04/07- Marks of Question

7.1 Introduction

Axial flow compressors are positive displacement type of compressor.

Axial flow compressors are turbo machines that increase the pressure of air or gas flowing continuously in the axial direction.

Earlier Axial flow compressor had pressure ratio of 5:1 in 10 stages & 40% efficiency.

Due to continue aerodynamic development resulted in increased in stage pressure ratio & number of stage for given overall pressure ratio has been greatly reduced.

Today Axial flow compressors reported efficiency up to 90% at pressure ratio of 8:1.

7.1.1 Advantages Of Axial Flow Compressors

1) It has higher efficiency than centrifugal compressor.
2) It is more suitable for multi staging & increase in pressure with negligible losses.
3) Pressure ratio of 8:1 or even higher can be achieved using multi stage axial flow compressor.
4) It can handle large amount of air inspite of small frontal area.
5) It has high thrust per unit frontal area.

7.1.2 Disadvantages Of Axial Flow Compressors

1) The performance is very sensitive to its mass flow rate at the design point and any deviation from the design condition causes the efficiency to drop off drastically.
2) It has more complexity and cost.

7.1.3 Applications Of Axial Flow Compressors

- Constant load applications such as in aircraft gas turbine engines.
- Fossil fuel power stations; where gas turbines are used for topping up the station output when normal peak loads are exceeded.
- Large marine gas turbine plant.

7.2 Construction And Working Of An Axial Flow Compressor

Detail construction and working of axial flow compresor is explain below:

Construction

- An axial flow compressor consists of fixed and moving sets of blades in alternating sequence as shown in Fig. 7.1.
- The sets of ***moving blades*** are attached to periphery of a rotor hub and the sets of ***fixed blades*** are attached to the walls of the outer stationary casing called the stator.
- At the inlet of the compressor, an extra row of fixed vanes called ***inlet guide vanes*** are fitted; which guide the air at the correct angle onto the first row of moving blades.
- The rotor and stator blade banks must be as close as possible for smooth and efficient flow.
- The radius of the rotor hub and the length of the blades are designed so that there is only a very *small tip clearance* at the end of the rotor and stator blades.
- One set of the rotor blades and one set of the stator blades constitute a ***stage***.

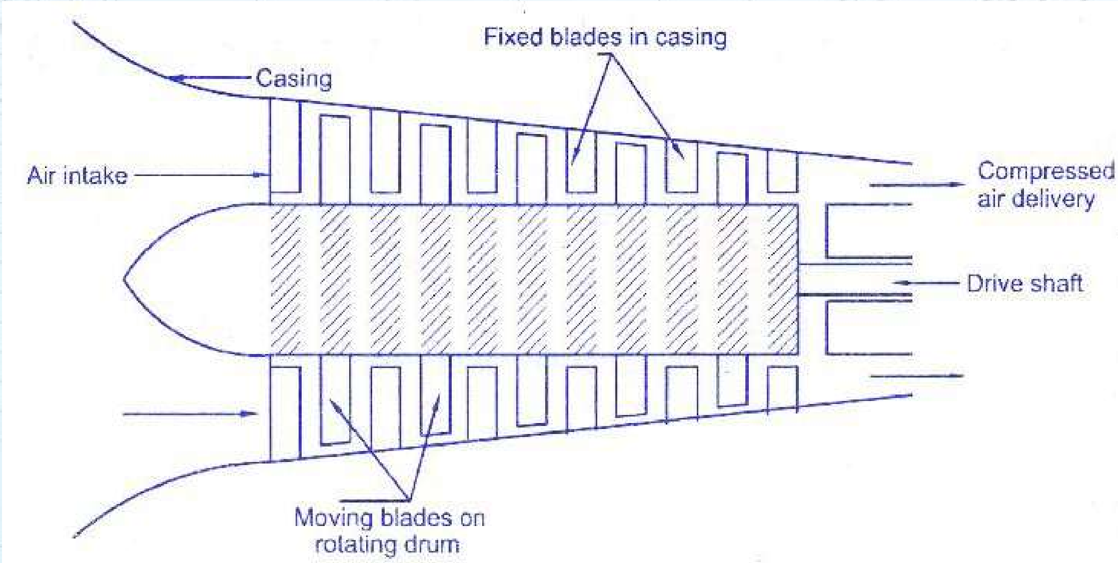


Fig.7.1 – Axial flow compressor

- The height of successive sets of blades is reduced to compensate for the reduction in volume resulting from increasing pressure from stage to stage; thus the ***axial flow velocity constant*** through the compressor.
- The rotor and stator blades are so arranged that the spaces between them form diverging passages; hence the velocity of the air is decreased as it passes through them, and rise in pressure takes place.

Working

- As the air passes through the moving blades, the kinetic energy added to the air and pressure rises at the expense of a ***reduction in the relative velocity*** of the air; by providing diffuser passages between blades.

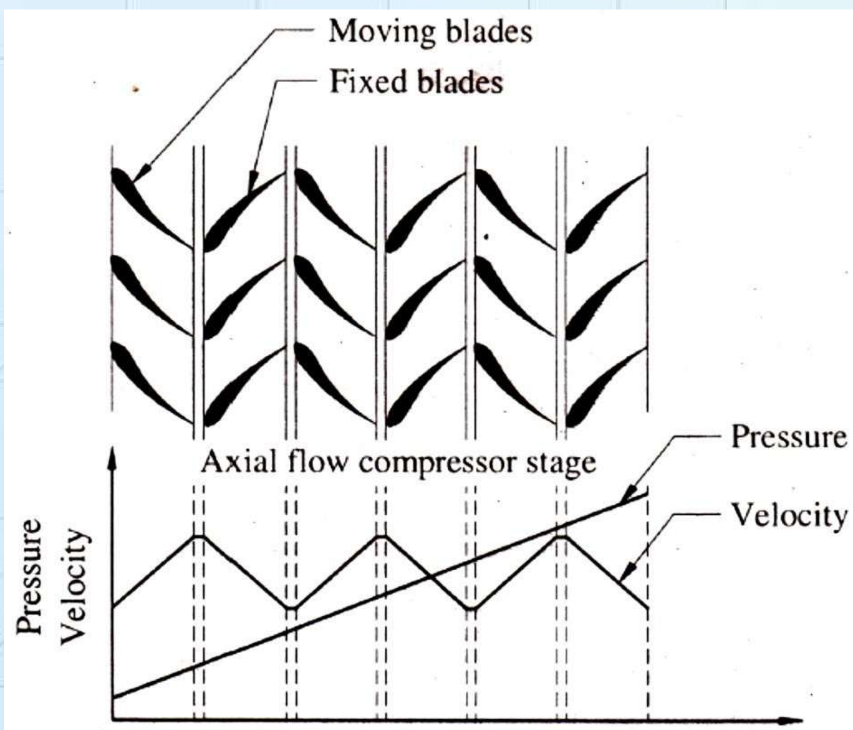


Fig.7.2 – Velocity and pressure variation in axial flow compressor

- The *absolute velocity of air increases* due to work input to the moving blades through the rotor shaft. (Fig. 7.2)
- Now, the air is then discharged at the proper angle to the first row of fixed blades; where the pressure is further increased by diffusion.
- Then, the air is directed to the second row of moving blades and the same process is repeated through the remaining compressor stages.
- It is usually arranged to have an ***equal temperature rise*** in the moving and the fixed blades; to keep the axial velocity of the air constant throughout the compressor.
- Thus each stage of the compression is exactly similar with regard to air velocity and blade inlet and outlet angles.

7.3 Aerofoil Blading

- "An aerofoil blade may be defined as a streamlined body bounded principally by two flattened curves and whose length and width are very large in comparison with the thickness."
- Axial flow compressors fitted with aerofoil blading efficiencies as high as 90 %.
- It has a thick rounded leading edge and a thin (sometimes sharp) trailing edge and its maximum thickness occurs somewhere near the midpoint of the chord a shown in Fig. 7.3.
- The back bone line lying midway between the upper and lower surfaces is known as the camber line.
- When such a blade is suitably shaped and properly oriented in the flow, the force acting on it normal to the direction of flow is considerably larger than the force resisting its motion.
- Aerofoil shapes are used for aircraft wing sections and the blades of various turbo machines.
- The system of blades used in centrifugal compressor is known as momentum blading because the force on the blades is calculated from the rate of change of momentum.
- Axial flow compressors fitted with momentum blading are not efficient on account of the occurrence of boundary layer separation from the blade surfaces. This does not occur usually on radial flow blading owing to the increase in pressure due to centrifugal head.
- Boundary layer separation causes serious loss pressure due to violent eddies and distortion of flow.
- Aerofoil blades give less restricted passage of air compared with momentum blading.

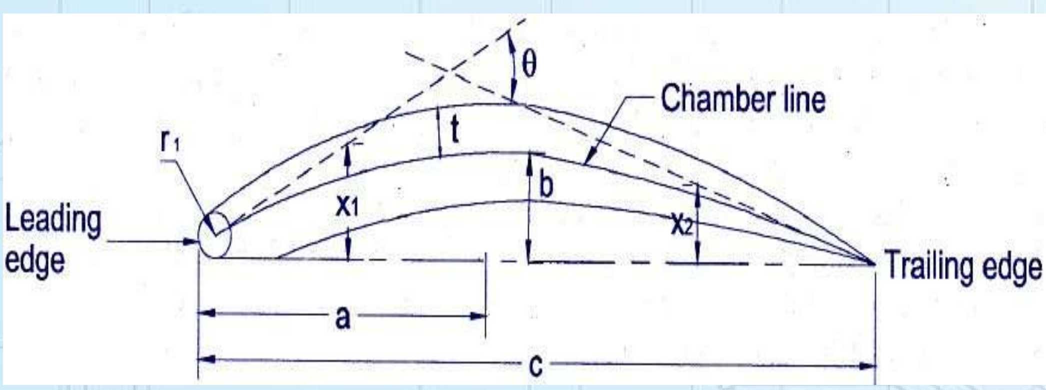


Fig.7.3 – Aerofoil blade section

7.3.1 Notations Of An Aerofoil Blade:

1) **Camber line:** It is the line representing the locus of all points midway between the upper and lower mean surfaces of an aerofoil.
2) **Leading edge:** It is a circular arc blended into the main profile and specified by its radius r_1 as a percentage of the maximum thickness.
3) **Maximum thickness 't':** It is useful parameter for describing an aero-foil and it is expressed as a percentage of blade length.
4) **Trailing edge:** It is ideally sharp i.e. of zero radius, but this is impossible from strength consideration. It is also a circular arc specified as a percentage of the maximum thickness.
5) **Camber angle:** The tangents to the camber line at the entry and exit make the camber angles with the axial direction.

x_1 = Camber inlet angle

x_2 = Camber outlet angle

$\square$ = Camber angle = $x_1 \square x_2$

b = Maximum Camber

a = Distance of the point of maximum camber from the leading edge.

c = Chord, the straight line form the leading to the trailing edge i.e. the distance between the blade leading and trailing edges.

7.3.2 Symmetrical (Un Cambered) Aerofoil

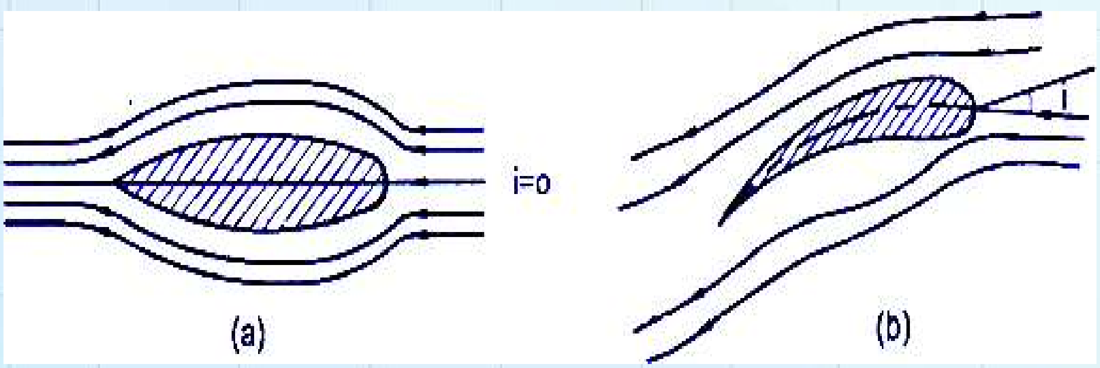


Fig.7.4 – Flow pattern around un-cambered and cambered aerofoil

- "The aerofoil whose axis of symmetry is parallel to the direction of undisturbed velocity of approach is called symmetrical aerofoil."
- The flow pattern around a symmetrical aerofoil placed in a stream of air is shown in Fig. 7.4 (a).
- The air divides around aerofoil at the leading edge and then re-joins at the trailing edge.
- Though there is some local is no permanent deflection of the main stream and the only significant forces exerted on the aerofoil and due to friction and the local disturbance.

7.3.3 Non Symmetrical Aerofoil

- Fig. 7.4 (b) shows a non-symmetrical aerofoil inclined with the direction of the undisturbed approaching flow at an angle i. This angle is known as the "angle of attack" or incidence angle.

- Unlike the symmetrical aerofoil, there is a pronounced disturbance which results in greater local deflection of flow.
- The local deflection of air stream can only be created (by Newton's law) if the aerofoil exerts a force on the air and hence the reaction of the air must produce an equal and opposite force on the aerofoil.
- The presence of the aerofoil has changed the local pressure distribution and hence by the Bernoulli equation the local velocity.

7.4 Lift And Drag

- The centrifugal force on the fluid particles on the upper convex side tries to move them away from the surface; this reduces the static pressure on this side below free, stream pressure. On account of this suction effect, the convex surface on the blade is known as the *suction side*.
- But the centrifugal force on the lower concave side presses the fluid harder on the blade surface, thus increasing the static pressure above that of the free stream. Therefore, the concave side of the blade is known as *pressure side*.
- The static pressure distribution around a cambered aerofoil with an angle of attack 'i' is shown in Fig. 7.5.

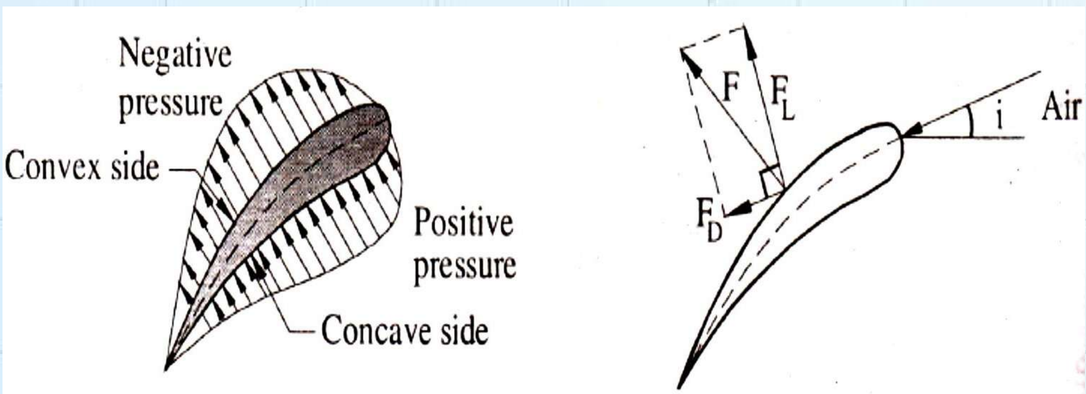


Fig.7.5 – Pressure distribution around a cambered aerofoil blade and lift & drag forces

- Due to this phenomenon, the flow on the suction side begins accelerating along the blade chord accompanied by a deceleration on the pressure side.
- The vertical sum of these pressures will produce some resultant force R acting on the blade normal to the chord tine.
- This resultant aerodynamic force R can be resolved into two components:

  (i) The component normal to direction of undisturbed air stream called the ***lift (L)***

  (ii) The component parallel to the direction of undisturbed air stream is called the ***drag (D)***.
- The lift is due to an unbalanced pressure distribution over the aerofoil surface and it is the basic force causing the aeroplane to maintain its lift.
- On aircraft wing there is a large area available for the production of lift force; therefore only a small pressure difference over its aerofoil wing section will provide the required lift.
- The drag force is made up of a friction drag, due to the pure skin friction effects and a pressure drag, due to unbalanced pressure distribution around the blade.
- Lift and drag forces depend only on the density, velocity of the fluid and the blade chord.

$L, D = f(\rho, C, l)$

The projected area per unit lenght of the blade is $A = l \times 1$

where $l =$ chord lenght

- The lift and drag co-efficient based on this area relate the dynamic pressure $\frac{1}{2}\rho C^2$ to the lift and drag forces.

$$L = C_L A \frac{1}{2} \rho C^2$$

$$D = C_D A \frac{1}{2} \rho C^2$$

$$\text{Lift co-efficient}, C_L = \frac{L}{\rho l C^2}$$

$$\text{Drag co-efficient}, C_D = \frac{D}{\rho l C^2}$$

- For a given blade C_L and C_D depend upon the aerofoil shape, the degree of curvature, Reynolds number, Mach number and the angle of incidence.

7.5 Performance Characteritics Of Axial Flow Compressors

The characteristics curves have the following salient points:

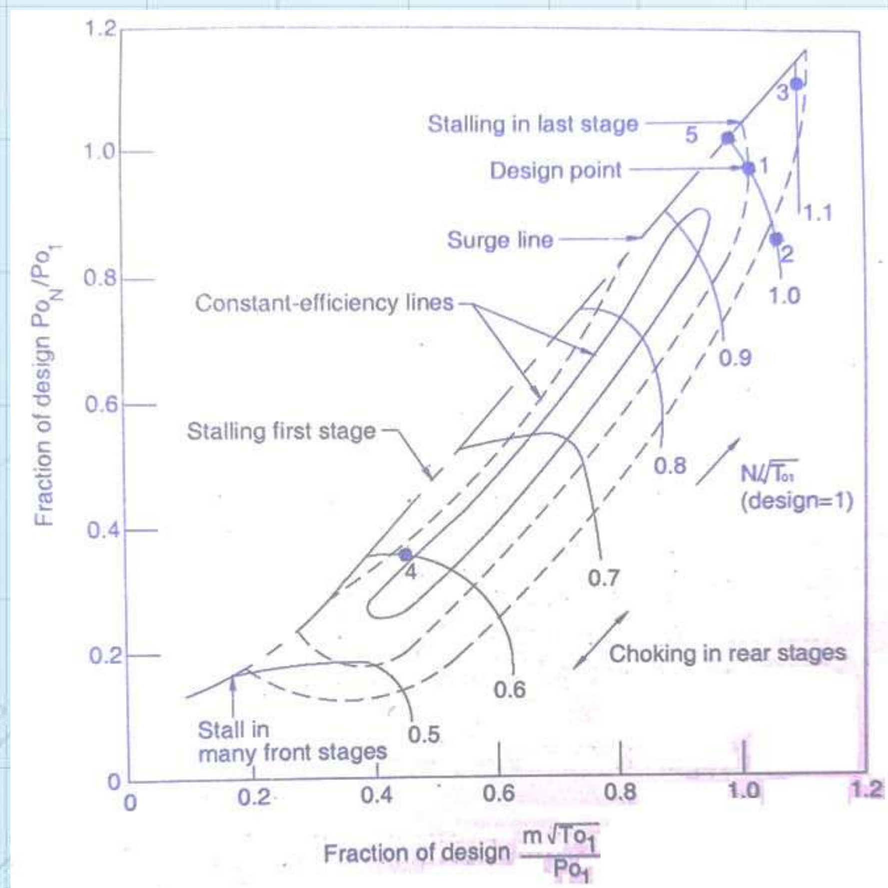


Fig.7.6 – Performance characteritics of axial flow compressor

1. The design mass flow and pressure ration are at point 1, and when operating at this condition, all the air flow angles and velocities throught the compressor are at their correct design values.
   The design point is very close to the surge line (Point-5). If the mass flow is only slightly reduced, the pressure ratio and density in the rear stages will both increase. Since axial velocity will decrease and hence the incidence angle will increase sharpely in the rear stages. This cause stalling in these stages.
2. Conversely, a small increase in mass flow will lead to a sharp drop in pressure ratio to point 2. The density also drops sharpely so that axial velocity increases. This result in the large decrease of the incidence angle in the rear stages, thereby causing stalling in the rear stages with negetive incidence.
3. If the speed of the compressor is reduced so that the operating point moves to 4. The mass flow and therefore axial velocity decrease faster than the blade speed, resulting in an increased incidence angle.
   The further slight reduction in mass flow along the constant speed characteristics will have little effect on the fisrt stage density but will cause a further increase in incidence angle and possible stalling in the first stage. If the mass flow rate is increased at low speed, the possibility of first stage stalling reduces but the density in the rear stages is very low and hence axial velocity increases until sonic conditions and choking of the flow in the rear stages occures.
4. When the design speed is increased to the point 3, the characteristic eventually becomes almost vertical. The increased speed allows more air to be passed at a higher density and pressure ratio. But at the inlet, the mass slow increases faster than density and chocking of the inlet is usually the first to occur.
5. All of the limiting conditions discussed above lead to unstable or inefficient operation and should be avoided at all times.

7.6 Referances

1. Engineering Thermodynamics by P.K. Nag, McGraw-Hill Education.
2. Turbines, Compressors and Fans by S.M. Yahya., TMH Publishers.
3. Fluid Power Engineering by V.L. Patel, Dr. R.N. Patel, Mahajan Publication House.