4.1 Introduction

Power absorbing turbomachines used to handle compressible fluids like air, gases etc, can be broadly classified into: (i) Fans (ii) Blowers and (iii) Compressors. These machines produce the head (pressure) in the expense of mechanical energy input. The pressure rise in centrifugal type machines are purely due to the centrifugal effects.

A fan usually consists of a single rotor with or without a stator. It causes only a small pressure rise as low as a few centimeters of water column. Generally it rises the pressure up to a maximum of 0.07 bar (70 cm WG). In the analysis of the fan, the fluid will be treated as incompressible as the density change is very small due to small pressure rise. Fans are used for air circulation in buildings, for ventilation, in automobiles in front of engine for cooling purposes etc.

Blower may consists of one or more stages of compression with its rotors mounted on a common shaft. The air is compressed in a series of successive stages and is passed through a diffuser located near the exit to recover the pressure energy from the large kinetic energy. The overall pressure rise may be in the range of 1.5 to 2.5 bars. Blowers are used in ventilation, power station, workshops etc.

Compressor is used to produce large pressure rise ranging from 2.5 to 10 bar or more. A single stage compressor can generally produce a pressure rise up to 4 bar. Since the velocities of air flow are quite high, the Mach number and compressibility effects may have to be taken into account in evaluating the stage performance of a compressor.

In general the centrifugal compressor may be known as a fan, blower, supercharger etc, depending on the need to be served. Broadly speaking, fans are the low-pressure compressors; blowers are the medium pressure compressors. It is therefore the analysis of one, say centrifugal compressor, will also holds good to the other machines like blower, fans.

4.2 Important Elements of a Centrifugal Compressor

Fig.4.1 shows the essential parts of a typical centrifugal compressor. It mainly consists of (i) inlet casing with the converging nozzle (ii) the impeller (iii) the diffuser and (iv) the outlet casing.

The function of the inlet casing with the conversant nozzle is to accelerate the entering fluid to the impeller inlet. The inlet nozzle accelerate the fluid from the initial condition (state 0) to the entry of the Inlet Guide Vanes (IGV) which direct the flow in the desired direction at the inlet of the impeller (state 1).

The impeller convert the supplied mechanical energy into fluid energy whereby the fluid kinetic energy and the static pressure rises. An impeller is made of radial blades which are brazed to the shroud. It can be made from a single piece consisting of both the inducer and a largely radial portion. The inducer receives the flow between the hub and tip diameters (dh and dt) of the impeller eye and passes on to the radial portion of the impeller blades. The flow approaching the impeller may be with or without swirl. The inlet diameter
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Fig.4.1 Important elements of a centrifugal compressor

Fig.4.2 shows the general schematic diagram of a centrifugal compressor. The impeller may be of single-sided or double-sided as shown in Fig.4.3 (a) and (b). Double-sided impeller may be used where for the given size the compressor has to handle more flow.
4.3 Variations of Pressure and Velocity

Fig. 4.4 shows the variation of the pressure and the velocity across the impeller and the diffuser. As the fluid approaches the impeller, it is subjected to centrifugal effect thereby the kinetic energy (velocity) and the pressure of the fluid both increases along the radial direction. When the impeller discharges the fluid into the diffuser, the static pressure of the fluid rises due to the deceleration of the flow. Therefore the velocity reduces and the pressure still increases as shown in Fig. 4.4. This is mainly due to the conversion of kinetic energy into pressure energy of the fluid.

4.4 Principle of Operation

Fig. 4.5 shows the enthalpy-entropy diagram for a centrifugal compressor. Air enters the impeller eye through an accelerating nozzle. As the fluid velocity increases in the nozzle, there will be a pressures drop between the nozzle exit and the impeller inlet, and is represented by the process 00-1. The air is then enters the impeller with a static pressure and temperature \( p_1 \) and \( T_1 \)
respectively. Even though there is increase of entropy due losses and the pressure drop in the accelerating nozzle and the IGV, the stagnation enthalpy at the inlet of the nozzle and the impeller inlet remains same (h00=h01) because of no work transfer during this process. The energy transfer occurs only in the impeller blades. The process 1-2 shows the actual compression process in the impeller where the pressure of air increases from p1 to p2 due to the centrifugal effect. The process 1-2’ is the isentropic compression. The stagnation pressure corresponding to the exit state of the impeller is p02.

The process 2-3 is the actual diffusion process in the diffusor where the large kinetic energy of the fluid is converted into pressure energy, thereby the static pressure rises further from p2 to p3. The diffusion process would have been taken place isentropically then the process becomes 2-3’. The stagnation pressure at the exit of the diffuser is p03. The stagnation enthalpy remains constant from state 2 to 3 (i.e, h02=h03) even though the stagnation pressure decreases progressively (i.e, p02 > p03). This mainly due to the diffusion process is incomplete and as well as irreversible. If the isentropic compression would have been taken place from pressure p1 to delivery pressure of the stage p3, the process would be 1-3’ and in terms of stagnation states the process is 01-03’.

4.5 Entrance Velocity Triangle

Fig.4.6 shows the velocity triangle at the inlet of the impeller. It can be seen that the fluid enters the inducer section axially with no whirl velocity when there is no IGV, i.e., V1=Vf1, Vu1=0, a1=900. This is the general case at the inlet for the maximum energy transfer condition.

Fig.4.7 shows the flow through axially straight inducer section in the presence of IGV’s. Due to the presence the inlet guide vanes the fluid enters the inducer with a1 so that it has some swirl velocity Vu1 but the straight inducer blades made the relative velocity axial, i.e, b1=900.
4.6 Optimum Inlet Velocity at the Impeller Eye

The magnitude of relative velocity at the inlet of the impeller eye is very important as the relative Mach number at the inlet is mainly depend on this velocity only. We know that more the Mach number value more will be the compressibility effect and hence it reduces the compressor efficiency. It is therefore necessary to keep the relative velocity value as low as possible. There is a value of eye tip speed which will give minimum relative velocity as can be seen from Fig.4.8.
The eye root diameter can be as small as possible which will be decided by the size of shaft and bearing arrangement. Then for the given flow Q the area of the eye flow may be large, giving a low inlet velocity V1 and a high eye tip speed U1 or it may be small, giving a large V1 and small U1. In these two extreme cases the relative velocity Vr1 is high and hence the minimum value is exist in between these. An expression for Vr1 can be obtained in terms of eye tip diameter d_t using inlet velocity triangle as follows.

$$V_{r1}^2 = U_1^2 + V_1^2$$
$$V_{r1} = \left[ \left( \frac{\pi d_t N}{60} \right)^2 + \left( \frac{4Q/\pi V_1^2}{d_t^2 - d_r^2} \right) \right]^{1/2}$$

With the d_i, Q and N being fixed, differentiate the eqn.(4.1) with respect to d_t and equate to zero for getting the value of d_t for minimum value of Vr1. Fig. 4.9 shows the variation of relative velocity, hence the relative mach number, with the eye tip diameter. If there is no possibility to reduce the relative velocity further for a given machine, the relative Mach number can be reduced further using the pre-whirl at the inlet with the use of inlet guide vanes but the penalty will be the reduction in energy transfer in the impeller. Thus this technique is usually used only in high pressure ratio compressor, where the inlet relative Mach number exceeds unity and shock waves reduce the impeller efficiency.