1 Chapter

INTRODUCTION

1.0 GENERAL

Blower is an important class of fluid machine, which has characteristics of transfer of energy between continuous stream of fluid & an element rotating about a fixed axis. Blower is a head generating machine which employs the dynamic action of a rotating element “the rotor” the action of rotor changes the energy level of the continuously flowing fluid. Blowers & Compressors are pressure-increasing machines. In all these, the fluid enters axially and is discharged by the rotor into a static collector system casing and then into a discharge pipe. Main components of Blower are impeller which is having rotary motion, where energy is transfer and followed by stationary part casing, in which energy transformation takes place. Casing decides the size and pressure rise in the system.

Blowers are used where large volumes of gas at low pressure are required. They generally operate at low speeds & pressure ratios. In pumps & fans the fluid is considered incompressible while in the compressor & blower there is usually a considerable density change. Fans and blowers provide air for ventilation and industrial process requirements. Fans generate a pressure to move air (or gases) against a resistance caused by ducts, dampers or other components in a fan system. [1]

Blower selection depends on the volume flow rate, pressure, type of material handled, space limitations, and efficiency. Blower efficiencies differ from design to design and also by types of impeller. Blower falls into two general categories: centrifugal flow and axial flow. In centrifugal flow, airflow changes direction twice - once when entering and second when leaving. They are also used to produce negative pressures for industrial vacuum systems. Major types are: centrifugal blower and positive-displacement blower.

Centrifugal blowers look more like centrifugal pumps than fans. In multi-stage blowers, air is accelerated as it passes through each impeller. In single-stage blower, air does not take many turns, and hence it is more efficient.

Air flow in the blowers usually is subject matter of research as blowers command greater share of applications in various sectors of industries. With relatively poor energy scene in developing economy, greater emphasis is usually made on power intake requirements, efficiency etc. These parameters have direct influence on air flow in the blower system both in the rotor and the volute casing. It is due to this reason that, what follows is a brief description of air flow in various parts of a Centrifugal Blower.

1.1 CENTRIFUGAL BLOWER

Blower is power consuming machine, where large volumes of gas or air at low pressure are required. According to the "Compressed Air Institute", it is a machine to compress air or gas by centrifugal force to a final pressure not exceeding 2.4 bar. It is not water cooled, as the added expense of the cooling system is not justified in view of the relatively slight gain at this pressure. Centrifugal blowers are generally used for large air supply systems for reduced noise and maintenance. Here are few applications of blowers enumerated as High pressure air blower, Sewage aeration blower, Scavenging two cycle diesels blower, Cupola blowers, Blast furnace gas blowers, Water gas blowers, Municipal gas plant blowers, Cock plant exhausters and blowers, Airplane superchargers and Circulators.

1.1.1 WORKING PRINCIPLE

Centrifugal blower consists of an impeller which has blade fixed between the inner and outer diameters. It can be mounted either directly on shaft extension of the prime mover or separately on a shaft supported between two additional bearing. The latter arrangement is applied for large blower in which impeller is driven by flexible couplings. Air or gases enters the impeller axially through inlet duct or nozzle. In impeller the rotating vanes are imparted kinetic & potential energy to the fluid.

As the fluid leaves the impeller at high velocity and pressure, it is collected either by a volute or scroll casing or series of diffusing passages which converts kinetic energy into pressure and increases static pressure of the fluid before deliver the fluid from the exit of the blower. The outlet passage after the scroll can also take the form of a conical diffuser. The centrifugal blower consists of a rotor or impeller which rotates causing air-flow by centrifugal action. The air usually enters the impeller at the axis and leaves at the tip in a direction determined by the angle of the impeller blades. Upon leaving the tip, the air flows through a volute chamber, sometimes provided with a vaned diffuser casing. The diffuser casing utilizes part of the kinetic energy of the out flowing fluid and raises its static pressure. The volute chamber collects all the fluid at constant velocity and leads it to a diverging discharge pipe which may again provide more of diffusive action.

1.1.2 IMPELLER

According to rotating blade type the impeller are classified as radial, forward and backward type. The blade exit angle decides the type of impeller. The pressure rise and flow rate in blower depends on the peripheral speed of the impeller and blade angle.

In backward impeller the exit blade angle is less than 90^0. The channel of blade is gradually expanding, so that the relative airflow will decelerate gradually, while passing through the channel of blade.

Compared to other two types of impeller the backward curved impeller has maximum efficiency at design condition [2] and at the design point the energy coefficient which is the measure of pressure rise in the blower is less in case of the impeller compared to other two.

In radial impeller the exit blade angle is equal to 90^0. At the maximum efficiency condition the flow coefficient is higher for this type of impeller.

In forward impeller the blade tips incline towards the direction of rotation and exit angle is greater than 90^0, which is a very large blade angle. For the same size & speed these types of impellers have higher flow rate compared to others. As tangential velocity is very large at the exit, this gives higher stage pressure rise compared to other two and it has maximum energy coefficient. As the selection of the impeller is preliminary factor for Blower system.

1.1.3 OUTLET SYSTEM

Fluids leaves the impeller at an higher absolute velocity as compare to that in the discharge pipe, therefore this fluid is collected by outlet system without affecting it's performance. Outlet system reduces the flow velocity, by improving the outlet pressure. These outlet systems have an annular space outside the impeller before the volute or diffuser ring. This annular vaneless space decreases the non-uniformities and turbulence of flow entering the volute as well as the noise level. There are three types of outlet system 1) Vaneless Diffuser 2) Vaned Diffuser 3) Volute or Spiral Casing.

1.1.3.1 VANELESS DIFFUSER

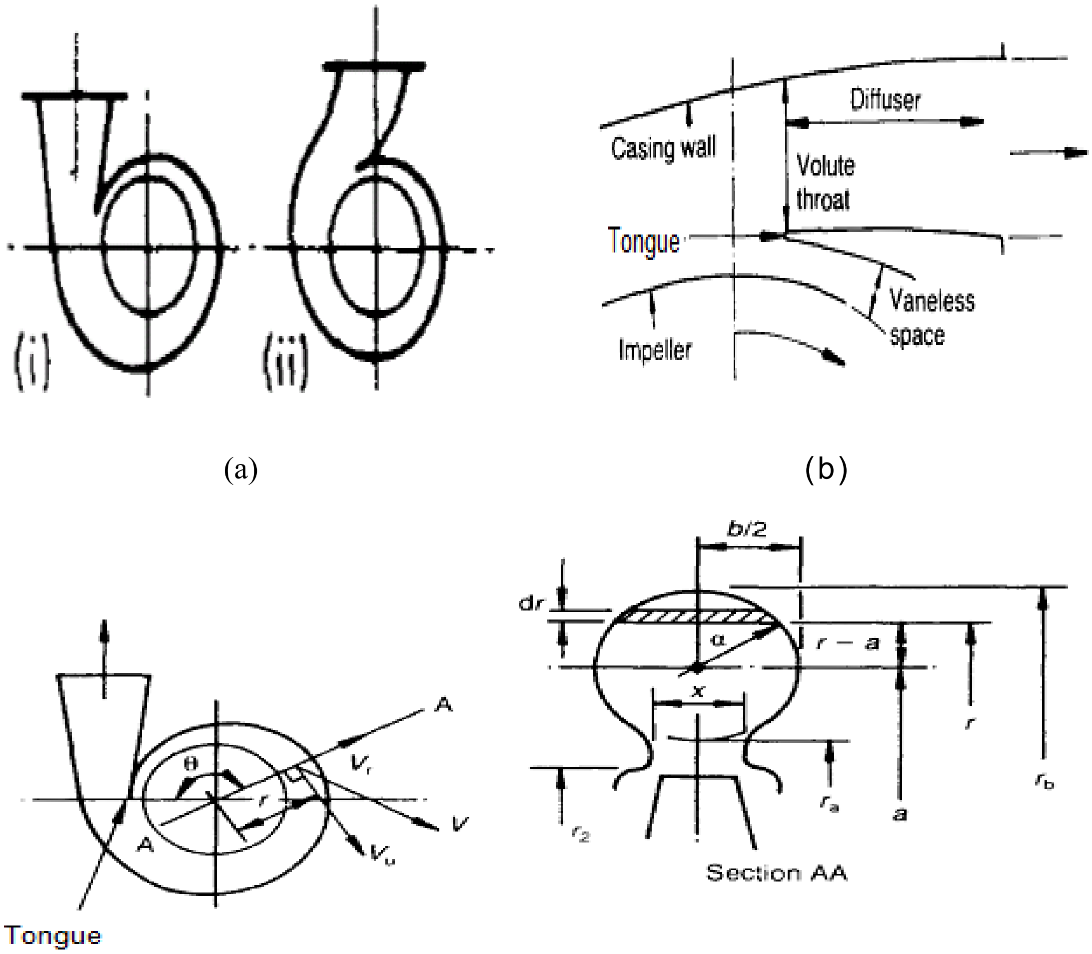
In this fluid is diffused in the vaneless space around the impeller before it leaves the stage through a volute. Diffusion occurs from smaller diameter to larger diameter and gain in static pressure occurs. As diffusion is directly proportional to diameter ratio it gives relatively large size diffuser, which is the limitation of this outlet system. Besides this it has a lower efficiency. This type of application generally used for large size compressor, as it does not suffer from blade stalling and shockwaves.

1.1.3.2 VANED DIFFUSER

For high pressures centrifugal blowers the fluid from the impeller is discharged through a vane diffuser. In this diffusion achieved by means of diffuser vanes, which results smaller size diffuser. Also the vanes provide greater guidance to the flow in the diffusing passages. The provision of diffuser in a blower can give a slightly higher efficiency (4-5%) than a blower with only a volute casing. Every diffuser blade ring is designed for given flow condition at the entry where optimum performance is obtained. Therefore at off- design condition the diffuser will give poor performance on account of mismatching the flow.

1.1.3.3 VOLUTE CASING

The purpose of the blower casing is to guide the fluid from the impeller or diffuser and convert into pressure. The flow with high kinetic energy is discharge from impeller and leads gas or air away. The Volute surrounds the impeller and whose cross-sectional area increases from a minimum at the tongue or cutwater to the throat. The tongue represents the nearest part of the casing to the impeller and is aligned into the general direction of the flow leaving the impeller as shown in figure: 1.1. [3] The casing plays an important part in locating best efficiency point by virtue of its hydraulic loses and its ability to restrict the flow against given head without incurring any additional losses. Flow leaving out of the casing can tangential or radial as shown in figure: 1.1a. Finally flow leaving out of the casing is through the volute throat as shown in figure: 1.1b. Tongue and throat position is very important in the design of volute casing. Velocity components in the volute is shown in 1.1c and change in cross-section area of volute at different radius. As cross-section area of volute casing goes on increasing from tongue region to throat.



(c)

Figure: 1.1 Tangential discharge and radial discharge volute casing

1.2 ARRANGEMENT OF VOLUTE CASING

Volute casing are classified on the basis of Cross-section of Volute passage and splitting of outlet passage for blower.

1.2.1 PARALLEL SIDE WALLS

In this type of volute casing the sides of the casing are parallel as shown in figure: 1.2. The width of the casing is nearly kept equal to the width of the impeller. With most ventilation blowers this is the normal form of design. In this the width of casing is wider than the impeller width as shown in figure: 1.3.

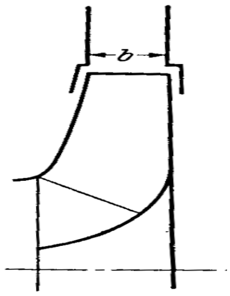


Figure: 1.2 Parallel Side Walls

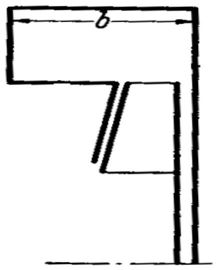


Figure: 1.3 Parallel Side Walls Wider Than impeller

1.2.2 RECTANGULAR CROSS SECTION

This type of volute casing has geometrically similar section as shown in figure: 1.4. The individual section whose basic shape is rectangular should be altered radially and axially so that it results in similar areas. These construction naturally yield no body of rotation with reference to the sidewalls, so that deviations from the law “r x C_θ = constant” are to be expected.

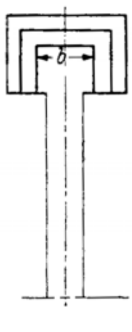


Figure 1.4 Rectangular Cross Section

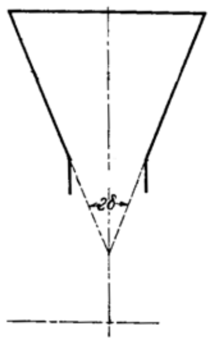


Figure1.5 Tapering side wall type

1.2.3 TRAPEZOIDAL CROSS SECTION

In this type of casing the two sides of casing are tapered which is obtained by an angle 2δ between sides as shown in figure: 1.5 and is usually about 60°.The larger 2δ results turbulence and inefficiency while smaller 2δ increased casing diameter and weight of the blower. The selection of delta is dependent upon separation of flow.

1.2.4 CIRCULAR CROSS SECTION

If the diameter‘d’ of the volute section is not too great in relation to the total casing diameter then the hyperbolic path over the circular section is very flat. In this type of volute one should first of all selects ‘d’ approximately, mean while one calculates the cross-section which gives a constant velocity ‘c’ in the complete volute. In this case, the section increases proportionally to the admission arc as shown in figure: 1.6.

1.2.5 INNER VOLUTE

In this type of volute the air flows around the impeller and will be divided on one or both sides of impeller. According to the law of rotation the peripheral component will be subsequently larger. This gives rise to a significantly smaller volute as shown in figure: 1.7. The impeller friction falls off because the air flows along the disc with a larger velocity. The friction that in the volute is significantly shorter than in case of normal construction. On such type of casing the discharge velocity are high so that a diffuser must be fitted to the outlet.

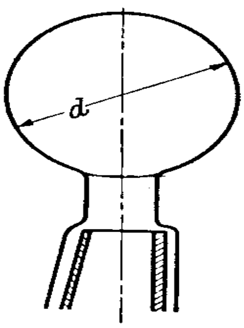


Figure: 1.6 Circular Cross-section

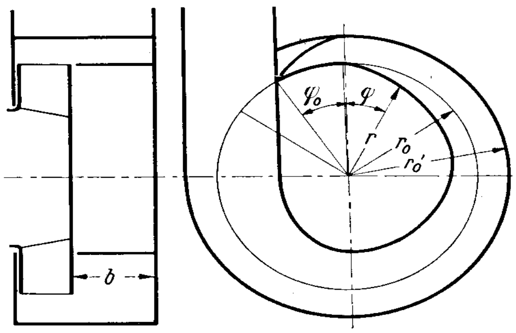


Figure: 1.7 Inner Volute

1.2.6 AXIAL VOLUTE

This type of volute has the great advantage of arising from cylindrical surfaces. In this the “volute” is developed between two co-axial cylinders in the axial direction as shown in figure: 1.8. In order to achieve the outside diameter of the entry to the volute the air already divides before the tongue so that the volume which flows in at 0° goes through the complete casing. On account narrow rectangular section of these volutes essential losses arise through secondary flows.

1.2.7 HELIX VOLUTE

These types of volutes are especially suitable for axial flow fan in which the velocity behind the impeller is neither changed in magnitude nor in direction as shown in figure:1.9. In this the conversion of the high velocity of discharge into static pressure, and then carried out in a connected straight diffuser.

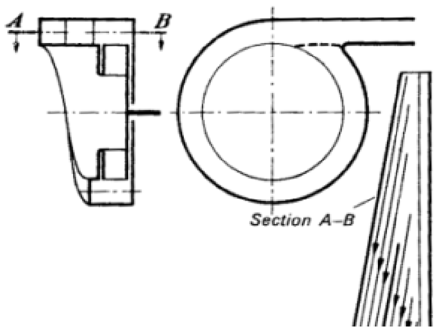


Figure 1.8 Axial volute

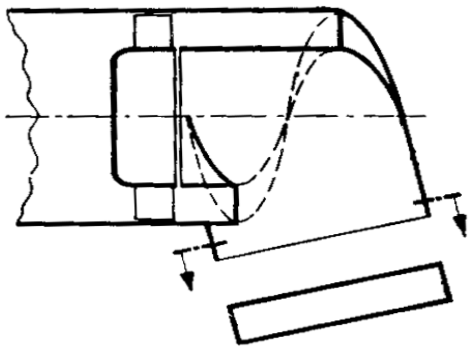


Figure: 1.9 Helix volute

1.2.8 HELIX-FORMED CONSTRUCTION OF VOLUTE CASING

The volute casing is the best and simplest guide or diffusion arrangement for radial-flow fans which is known in fan engineering as shown in figure: 1.10. This is not so for axial-flow fans. This is all the more regrettable as the use of volute casing with axial flow fans, produces a large unit, and in addition to this numerous experiments have shown it can cause undue loses.

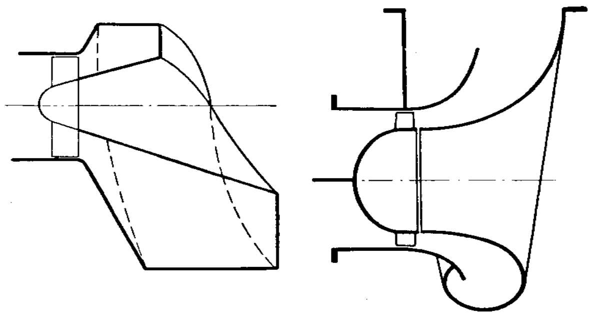


Figure: 1.10 Helix-formed construction of volute casing

1.2.9 SUBDIVIDED VOLUTE CASING

With large radial extent of the volute casing undue disturbances arise through secondary streams. Depending upon circumstances, this influence can be modified if; one resolves an average tongue into two or more spirals. Such type of volute casing is also known as double volute casing as shown in figure: 1.11. In this type of volute casing radial thrust can be reduced considerably. If one requires decreasing the discharge velocity at the same time in the shortest possible space, then to get this one can subdivide the discharge section with a number of guide blades as shown in figure: 1.11.

For large volumes one can go a step further. By sub-division and partial profiling of the guide blades, air is discharged at a width which is more than double the diameter of impeller. It is used with impellers having high volume co-efficient, i.e. multivaned impellers.

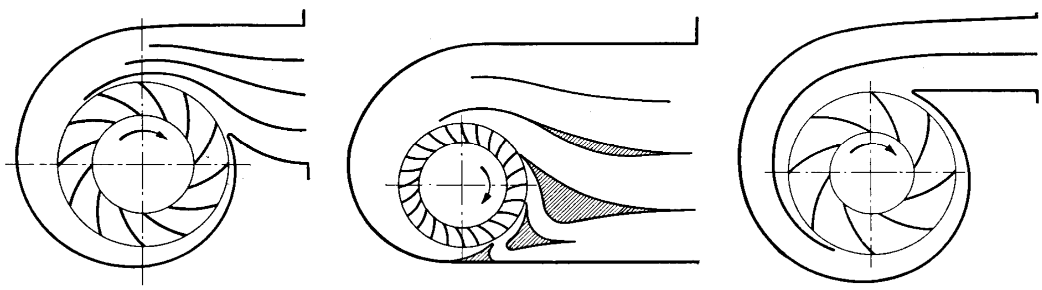


Figure 1.11 subdivided volute casings

1.2.10 VOLUTE CASING WITH AN ADJUSTABLE TONGUE

The requirements often demand that a volute casing must satisfy different volumes in order to avoid too large a loss when the duty falls below normal design conditions. Particularly in mine ventilation, for example, the conditions are continuously changing so that this aspect is extremely important. The adjustable tongue ends are available for different positions the effect of such measures is limited as shown in figure: 1.12.

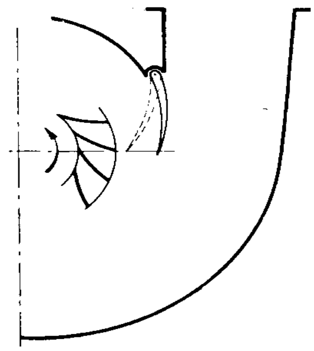


Figure: 1.12 Volute casing with an adjustable tongue

1.3 ROLE OF IMPELLER

A centrifugal blower impeller have backward swept blades, radial tipped blades or forward swept blades as shown in figure: 1.13 below. The inlet and outlet velocity triangles are also shown accordingly in the figure. Under ideal conditions, the directions of the relative velocity vectors V_{r1} and V_{r2} are same as the blade angles at the entry and the exit. A zero whirl at the inlet is assumed which results in a zero angular momentum at the inlet. The backward swept blades are employed for lower pressure and lower flow rates. The radial tipped blades are employed for handling dust-laden air or gas because they are less prone to blockage, dust erosion and failure. The radial-tipped blades in practice are of forward swept type at the inlet. The forward-swept blades are widely used in practice. On account of the forward-swept blade tips at the exit, the whirl component of exit velocity V_{w2} is large which results in a higher stage pressure rise [4].

$V_{w2} < U_2$, if $\beta_2 < 90^0$, backward swept blades

$V_{w2} = U_2$, if $\beta_2 = 90^0$, radial blades

$V_{w2} > U_2$, if $\beta_2 > 90^0$, forward swept blades.

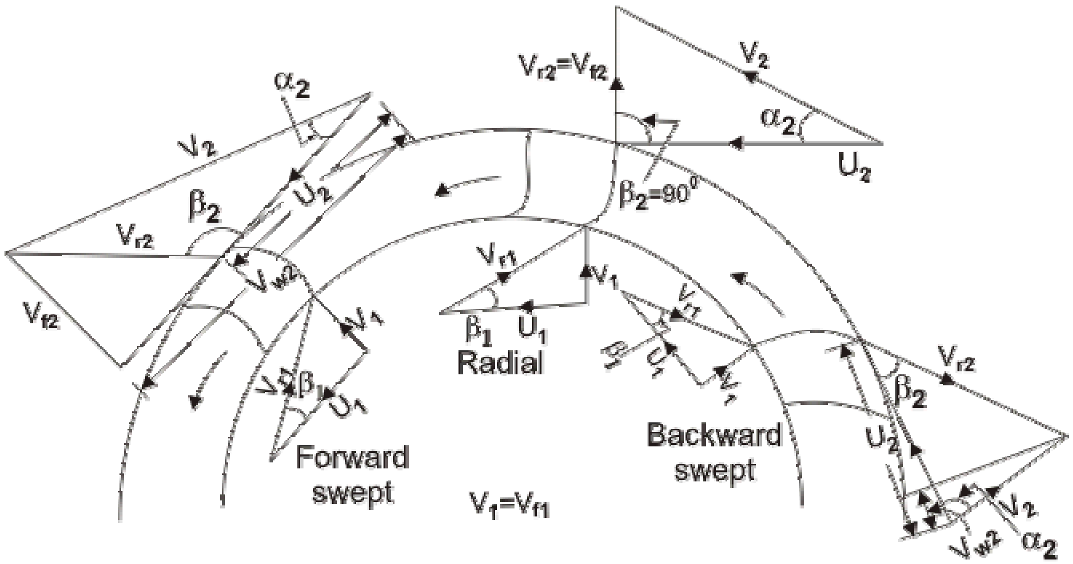


Figure: 1.13 Velocity triangles at inlet and outlet of different types of blades of an impeller of a centrifugal blower.

1.3.1 THEORETICAL PRESSURE DEVELOPED BY A CENTRIFUGAL IMPELLER

The theoretical pressure developed by the impeller, is achieved by using Euler's Equation. In fluid dynamics, the Euler equations are a set of equations governing in viscid flow. They are named after Leonard Euler. The equations represent conservation of mass (continuity), momentum and energy, corresponding to the Navier–Stokes equations with zero viscosity and heat conduction terms. Historically, only the continuity and momentum equations have been derived by Euler.

The static pressure rise through the impeller is due to the change in centrifugal energy and the diffusion of relative velocity component. Therefore, it can be written as

$$p_2 - p_1 = (\Delta p) = 1/2*\rho\,(U_2^2 - U_1^2) + 1/2*\rho\,(Vr_1^2 - Vr_2^2) \qquad (1.1)$$

The stagnation pressure rise through the stage can also be obtained as

$$(\Delta p_0)=1/2*\rho\ (U_2^2-U_1^2) + 1/2*\rho\ (Vr_1^2-Vr_2^2) + 1/2*\rho\ (V_2^2-V_1^2) \quad (1.2)$$

Where, U_1 and U_2 are peripheral velocities and V_1 and $V_2$are absolute velocities at inlet and outlet

From equations (1.1) and (1.2) we get

$$(\Delta p_0) = (\Delta p) + 1/2*\rho\ (V_2^2-V_1^2) \quad (1.3)$$

1.3.2 SLIP FACTOR

The slip factor is a vital piece of information needed for the designers (also by designers of radial turbines) as its accurate estimation enables the correct value of the energy transfer between impeller and the fluid. Various attempts to determine the values of slip factor have been made and numerous research papers concerned solely with this topic have been published. Wiesner (1967) has given an extensive review of the various expressions used for determining slip factors. Most of the expressions derived relate to radially vaned impellers or to mixed flow designs, but some are given for backward swept vane designs. All of these expressions are derived from inviscid flow theory even though the real flow is far from ideal. However, despite this lack of realism in the flow modeling, the fact remains that good results are still obtained with the various theories.

Under the ideal (frictionless) conditions the relative flow leaving the impeller of a blower will receive less than perfect guidance from the vanes and the flow is said to slip. If the impeller could be imagined as being made with an infinite number of infinitesimally thin vanes, then an ideal flow would be perfectly guided by the vanes and would leave the impeller at the vane angle. But, the relative eddy causes the flow in the impeller passages to deviate from the blade angle β_2 at the exit to an angle β'_2. This deviation is being larger for a larger blade pitch or for smaller number of impeller blades. On account of this effects the apex of the actual velocity triangle at the impeller exit is shifted away (opposite to the direction of the rotation) from the apex of the ideal velocity triangle. This phenomenon is known as slip and the shift of the apex is the slip velocity. It may be seen that on account of the slip the whirl component is reduced which in turn decreases the energy transfer and the pressure developed [1]. Figure: 1.14 compares the relative flow angle β_2, obtained with a finite number of vanes, with the vane angle, β'_2. A slip factor may be defined as ratio of actual and ideal values of the whirl components at the exit.

$$\mu = Vu_2' / Vu_2 \qquad (1.4)$$

Where μ = slip factor

V_{u2} = ideal exit velocity peripheral component

V_{u2}' = actual exit velocity peripheral component due to slip

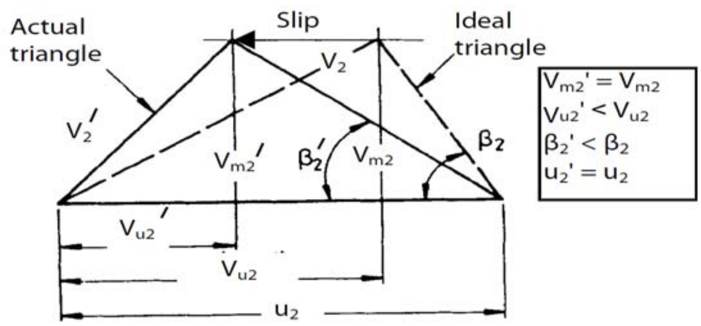


Figure: 1.14 Slip Factor

V = absolute velocity of fluid

V_m = meridional velocity component

V_u = peripheral velocity component

U = peripheral velocity of impeller

W = relative velocity

β = blade angle

1.4 FLOW PHENOMENON IN BLOWER CASING

Flow generated in the casing is quite complicated with three-dimensionality and unsteadiness. The fundamental blower performance is primarily determined by the impeller and is mainly affected by the casing. However, the theoretical estimation of the effect of the casing on the performance has not been well established. It is assumed that flow inside the casing should follow streamline at an angle of flow angle. But there are many factors which results in deviation of the flow in casing. This leads to loss and eddy formation, which leads to affect the performance. In order to estimate the casing effect on blower performance, detailed three-dimensional (3D) flow analysis in the casing is necessary. Three hole-probe, five-hole probe and Stereoscopic PIV is one of the useful techniques for experimental analysis of flow fields. There are some difficulties in practical application for flow analysis in fluid machinery with complicated geometry, but the results obtained provide useful information for understanding the three dimensional flow field. The detail analysis at various angular positions inside the casing and behavior of the flow as per different turning angle of the casing effects the performance of the Blower. The efficiency and pressure for the design flow condition can be enhanced, by reducing the losses inside the casing. This behavior of the flow in the casing can be also carried out with the help of CFD software. The overall performance characteristics of a Blower can be analyzed by experimental measurements and numerical simulation. The performance characteristic of a blower is a function of the shape and width of the volute casing. The volute performance is dependent on the quality of flow passed on to it from the impeller or diffuser, the performance of impeller or the diffuser also depends on the environment created by the volute around them. The non-uniform pressure distribution around the impeller provided by its volute gives rise to the undesirable radial thrust and bearing pressures.

1.5 PERFORMANCE CHARACTERISTICS OF A BLOWER

The actual shape of blower performance curves is determined by the combined effect of the hydraulic losses of the impeller and casing. Mechanical losses including disk friction remains the same for all capacities and leakage loss is small and varies slightly with the impeller head near the shaft. Usually for the design, comparison and critical assessment of blower the dimensionless coefficients are adopted. This is done so as to arrive at the performance values which are independent of actual increase in pressure, flow rate, physical and other properties. The user on the other hand is concerned with the head increase in the blower and the volume flow rate, so investigation of flow in casing helps in minimizing loses and selection of appropriate casing for the impeller.

The performance coefficients of blower are pressure coefficient ψ, flow coefficient φ, diameter coefficient δ and speed coefficient σ. Pressure coefficient or the pressure rise coefficient is defined as the ratio of the difference between the static pressure at the exit flange of the blower and the total pressure at the inlet flange of the blower to the dynamic head corresponding to the peripheral speed at the exit of the rotor of the blower.[5] Therefore it important to investigate the flow inside the casing where there is transformation of energy to pressure rise.

$$\Psi = \Delta p / (1/2 * \rho U_2^2) \tag{1.5}$$

Flow coefficient or volume coefficient is defined as the ratio of the volume flowing through the blower to the theoretical volume calculated on the basis of peripheral speed at the exit of the rotor of the blower and the passage area at the exit of the rotor of the blower. In other words it the ratio of the radial or the meridional velocity at the exit of the rotor to the peripheral speed at the exit of the rotor of the blower. As peripheral speed is depending upon the diameter, speed and width of impeller, so width of casing can be deciding parameter for design of casing.

$$\Phi = Vr_2 / U_2 \tag{1.6}$$

This is obtained from

$$\Phi = Q / \pi * D_2 * b_2 * U_2 \tag{1.7}$$

The speed coefficient σ is also known as shape factor as it gives the overall shape of the machine based on its duty, i.e. the pressure head and flow rate. It is also called specific speed and written as N_s. The speed coefficient or the specific speed is given as

$$N_s = \Phi^{0.5} / \psi^{0.75} \tag{1.8}$$

The diameter coefficient δ is given by

$$\delta = \psi^{0.25} / \Phi^{0.5} \tag{1.9}$$

The relation between the flow parameters is

$$\psi = (\Phi\, \delta^{2}) \tag{1.10}$$

The performance of a blower is usually presented in term of variation of pressure coefficient with flow coefficient.

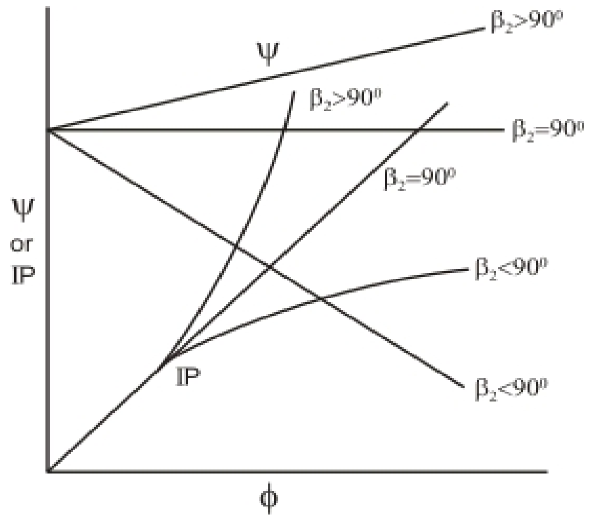


Figure: 1.15 Theoretical performance curves of a blower

The characteristics in figure: 1.15 depict the following

(i) Forward curved fans $\beta_2 > 90^0$ develop the highest pressure for a given impeller diameter and speed.

(ii) Power requirement of a forward curved fan increases steeply for a small change in flow rate.

(iii) Pressure developed decreases fast with increasing flow rate in a backward curved fan.

In conclusion, the forward curved fans have large volume discharge and pressure rise but they demand higher power. However, forward curved fans are unstable for off-design operating conditions.

Backward curved fans are very efficient and the drooping power characteristic makes them suitable for a better off-design performance.

Radial curved fans are preferred for dust-laden fluids. Due to their shape, the solid particles are not stuck and deposited on the blade surface. Specific speed is basic for selection of design of casing. Size of casing with rise in pressure differs with the outlet blade angle. So flow behavior changes drastically in the casing from backward to forward impeller. So one should have an idea regarding fitting of impeller in casing, which might be design for different blade angle. What should be the range for selecting the blade for designing the casing? The effect of disturbance of flow in casing will be mainly at off-design condition.

1.5.1 ACTUAL CHARACTERISTICS OF A CENTRIFUGAL IMPELLER

The actual characteristics of the centrifugal impeller are obtained by deducting the stage losses from the theoretical head or pressure coefficient. Therefore the nature of the actual characteristic depends on the manner in which the stage losses vary with the operating parameters. Friction and shock losses affect the performance significantly.

In all cases, friction and shock losses produce pressure-volume curves that tend toward zero pressure when the machine runs on open circuit, that is, with no external resistance. Figure: 1.17 shows a typical pressure-volume characteristic curve for a backward blade.

Frictional losses occur due to the viscous drag of the fluid on the faces of the vanes. A diffuser effect occurs in the diverging area available for flow as the fluid moves through the impeller. This results in a further loss of available energy. In order to transmit mechanical work, the pressure on the front face of a vane is necessarily greater than that on the back side. A result of this is that the fluid velocity close to the trailing face is higher than that near the front face. These effects result in an asymmetric distribution of fluid velocity between two successive vanes at any given radius and produce an eddy loss. The transmission of power is not uniform along the length of the blade. [6]

The separation losses occur particularly at inlet and reflect the sudden turn of near 90° as the fluid enters the eye of the impeller. In practice, wall effects impart a vortex to the fluid as it approaches the inlet. By a suitable choice of inlet blade angle β_1, the shock losses will not be in lower range of blower and may be small at the optimum design flow. An inlet cone at the eye of the impeller or fixed inlet and outlet guide vanes can be fitted to reduce shock losses. In the development of the theoretical pressure and power characteristics, we assumed radial inlet conditions. When the fluid has some degree of pre-rotation, the flow is no longer radial at the inlet to the impeller. The second term in Euler's equation takes a finite value and again, results in a reduced blower pressure at any given speed of rotation.

The combined effect of these losses on the three types of centrifugal impeller is to produce the characteristic curves shown on figure: 1.16. The non-overloading power characteristic, together with the steepness of the pressure curve at the higher flows, are major factors in preferring the backward impeller for large installations as shown in figure: 1.17[7]. All the above losses will be there in the blower but losses in casing due outlet blade angle, shape of the casing and angle of the tongue plays major role in the actual performance of the blower. For determining the major losses inside the casing can be achieved by the detail analysis of flow at various positions inside the casing.

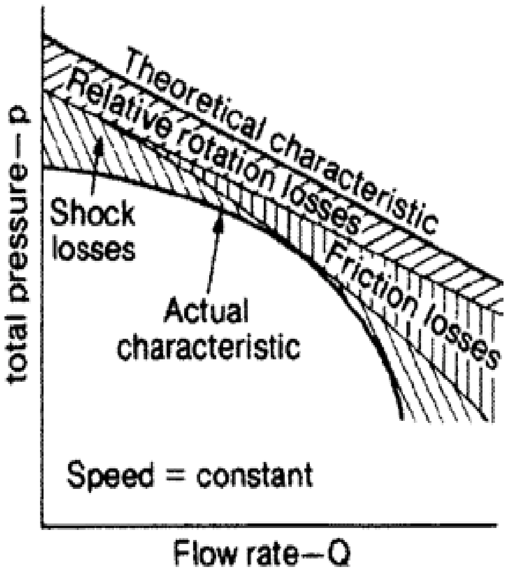


Figure: 1.16 Effect of losses on the pressure-volume characteristic of a backward bladed.

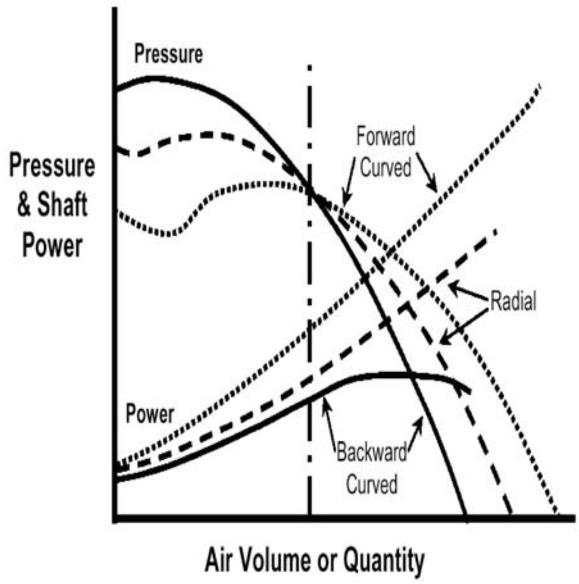


Figure: 1.17 Actual pressure and shaft power characteristics for centrifugal impellers.

1.5.2 EFFICIENCY OF BLOWER:

The blower efficiency is defined as a ratio of the energy output to the energy input. The true efficiency of blower is a definite physical quantity depending upon the degree of perfection of hydraulic and mechanical design as shown in figure: 1.18 [8]. In actual practice, it is very difficult to determine a true head from inlet and outlet pressures and energy input.

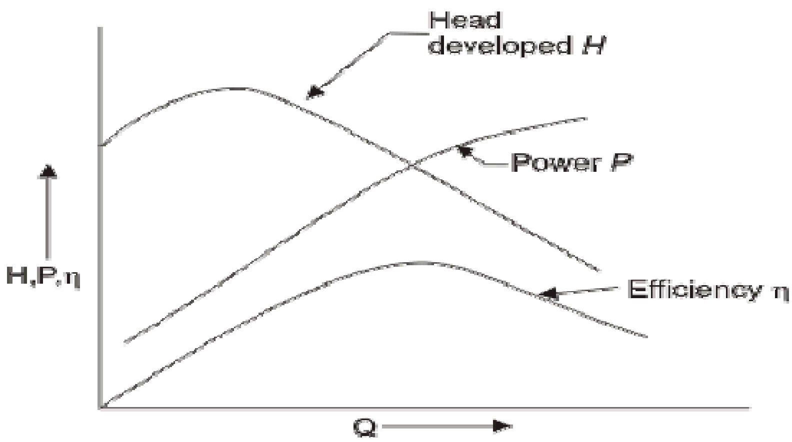


Figure: 1.18 Performance characteristic curves of a centrifugal blower.

The ’total’ or ‘gross’ blower efficiency is consisting of several partial efficiencies. They are:

Mechanical efficiency $\eta_m = \frac{\text{Brake horse power} - \text{Mechanical losses}}{\text{Brake horse power}}$ (1.11)

Where, mechanical losses include bearing, mechanical seal and impeller disk friction losses.

Volumetric efficiency $\eta_v = \frac{Q}{Q + Q_L}$ (1.12)

Where, Q is the measured volume of flow and Q_L is the leakage through the sealing rings which is bypassed back to the impeller inlet.

Hydraulic efficiency $\eta_h = \frac{H}{H + h_L}$ (1.13)

Where H is the head in feet available at blower discharge and hL represents hydraulic losses through blower passages, including skin friction and eddy losses.

It can be noted that all the above three partial efficiencies account for losses of volume (or weight of flow), head and power.

These partial efficiencies are connected to the total efficiency as follows:

$$\eta_T = \eta_h * \eta_v * \eta_m \quad (1.14)$$

In determining the total efficiency of blower, Mechanical efficiency will be fixed and limited as per the bearing and seals provided. Major role is of Volumetric and Hydraulic efficiency, which depends upon the casing flow. In investigating the flow inside the casing helps in determining the eddy losses and passage losses of the Blower.

1.6 SUMMARY

Blower system consists of impeller and casing, in which energy transfer and energy transformation takes place. For the energy transformation casing are of different type and design. In investigating the flow inside the casing helps in determining the eddy losses and passage losses of the Blower. As per the geometry of the impeller flow is directed in the stationary part such as diffuser or casing. A characteristic gives the limitation and operating range of the Blower. Air flow in the blowers casing usually is a subject matter of research as blowers command greater share of applications in various sectors of industries. Behavior of the flow from the impeller to casing is quite complicated and to understand the detail characteristics of flow many researchers have worked. So survey of Literature review can give the detail information regarding the flow investigation and design developed of blower casing.