3 Chapter

# DESIGN AND OPTIMIZATION

## 3.0 GENERAL

Many researchers had suggested different procedures of blower design considering fundamental principles of fluid flow and continuity equation on which casing is design. Austin church has brought a revolution in establishing the design methodology for Blowers. He has considered in his design the compressibility effect of a fluid. This shows the change in volume flow rate of a flowing fluid due to the density change. W. C. Osborne [19] has made very good attempt to embrace the simple physical principles necessary for the proper application, control and design of blowers. Several other design methodologies are also proposed by various authors like Stepanoff, F Bleier, W Cory, O E Balje etc. These design methodologies are mostly based on their own experiences and empirical data rather than theoretical equations only. Proper sizing of blowers involves determining many system factors and prioritizing them into requirements versus preferences. Some of these factors are the static pressure the blower must overcome, the average air flow volume required, the shape and direction of the desired air flow, space limitations, audible noise allowances, available power, efficiency, air density, and cost. The first two of these, air flow and static pressure, along with available power considerations are generally the most critical for system designers. These three addresses the fundamental questions how much air do I need and what is it going to cost in system power to get it? Defining these three parameters is generally the first step a system designer must complete towards Sizing a blower, finally on the design of volute casing. For design of casing first design of impeller is required and finally design of casing is done. Uniform design methodology is adopted for casing and optimized design is evolved by considering various constraints. Optimization tools have increase the efficiency of the casing and made selection of casing much easier with various conditions as stated above. An experimental reading is to be with minimum errors so uncertainty analysis is required, which gives much accurate results and increase the confidence level of the experiments. The fundamental blower performance is primarily determined by the impeller of the blower

## 3.1 IMPELLER DESIGN

The first step in the impeller design is to select relative speed as per the requirement of head and flow rate conditions. This establishes the specific speed or type of the impeller. Selection of the speed is governed by a number of considerations:

1. Type of driver contemplated for the unit.
2. Higher specific speed results in a smaller blower and cheaper drivers.
3. Optimum hydraulic (and total) efficiency possible with each type varies with the specific speed. This relationship has not been established accurately for blower. However, in the centrifugal pump field (including axial flow pump) the optimum efficiency possible with each specific speed is definitely known to the whole industry. Figure: 3.1 show a typical curve representing such relationships.

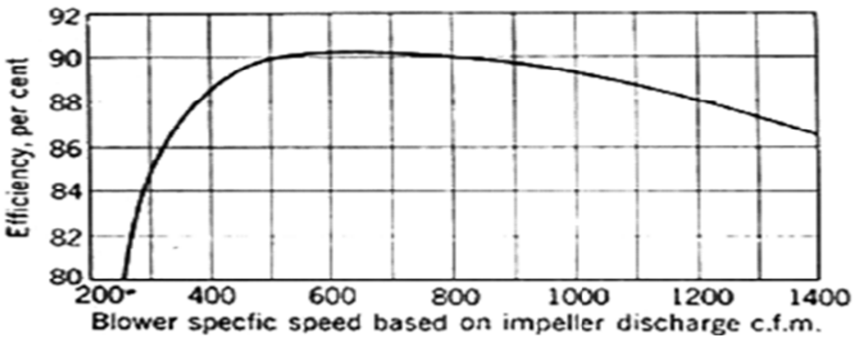


Figure: 3.1 Specific speed versus efficiency.

4. If the total required head cannot be produced in one stage, owing to limitation of maximum possible impeller peripheral

velocity, it is divided between two or more stages. The head per stage also affects the final specific speed and hence, the expected efficiency of the blower.

Having established the specific speed of the proposed impeller, the designer looks for a suitable “model” from existing impeller of the same specific speed which has satisfactory hydraulic performance, i.e. suitable slope of the head capacity curve and acceptable efficiency.

In present case, the considered speeds are 1450 RPM and 2850 RPM. Because the available induction motors have these speeds generally.

### 3.1.1 SELECTION OF IMPELLER TYPE AS PER SPECIFIC SPEED

In the present design the flow rate and static pressure is taken as an input data based on the industrial requirement of machine. The selected input data are:

- **Discharge Q = 0.5 $m^3/s$ = 0.5 × 35.31 × 60 = 1059.3 cfm**
- **Static pressure $P_S$ = 981.25 Pascal = 3.938 inch of WC**

**1. Specific Speed for N = 2850 RPM for available motor**

$$N_S = \frac{RPM * (Q)^{1/2}}{(P_S)^{3/4}} \qquad ------------------(3.1)$$

$$= \frac{2850 \times (0.5)^{\frac{1}{2}}}{(981.25)^{\frac{3}{4}}}$$

$$= 11.49 \ (RPM, \frac{m^3}{s}, Pascal)$$

$$= 33{,}181 \ (RPM, cfm, inch\ of\ WC)$$

Where

$N_S$ = specific speed
RPM = fan speed in RPM
Q = volume flow rate, in $m^3/s$ or cfm
$P_S$ = static pressure, in Pascal or inch of WC.
From Table: 3.1, it is observed that, the backward curved (BC) type impeller has $N_S$ = 30,000 in middle range. That’s why BC impeller is chosen.

**2. Specific Speed For N = 1450 RPM**

$$N_S = \frac{RPM * (Q)^{1/2}}{(P_S)^{3/4}}$$

$$= \frac{1450 \times (0.5)^{\frac{1}{2}}}{(981.26)^{\frac{3}{4}}}$$

$$= 5.84 \ (RPM, \frac{m^3}{s}, Pascal)$$

$$= 16{,}881 \ (RPM, cfm, inch\ of\ WC)$$

$R_{req} = N_s/RPM$ this value is help full in selection of range.

From Table 3.1, it is observed that $N_S$ = 16,881 does not fit in our criteria. 1450 RPM blower will have bigger outer diameter $d_2$ (or exit blade angle $\beta_2$) to produce same static pressure $P_S$. This will increase manufacturing cost. That's why 1450 RPM speed is not selected.

The speeds between (1500-2800 RPM) are also not considered as these speeds require special motor or variable speed drive unit or belt drive or gear system. Special motor or variable speed drive unit will increase initial cost; while belt drive or gear system will induce mechanical losses plus maintenance problems.

| Type of fan | Specific speed range (rpm) | Air volume (cfm) | Static pressure | Fan efficiency | Manufacturing cost | Size and weight |
|---|---|---|---|---|---|---|
| Propeller | 500,000–100,000 | Large | Low | Low | Low | Small |
| Tubeaxial | 300,000–60,000 | Large | Low | Medium | Low | Small |
| VAF, one stage | 130,000–50,000 | Large | Medium | High | Medium | Small |
| VAF, two stages | 90,000–35,000 | Large | Medium | High | Medium | Medium |
| Mixed flow | 80,000–10,000 | Medium | Medium | Low | High | Medium |
| Wide FC | 70,000–25,000 | Large | High | Medium | Medium | Large |
| AF, BC, BI | 70,000–20,000 | Medium | Medium | High | Medium | Medium |
| Narrow FC | 40,000–10,000 | Medium | High | Medium | Medium | Medium |
| Radial tip | 65,000–25,000 | Medium | Medium | Medium | Medium | Medium |
| Radial blades | 25,000–10,000 | Low | Medium | Low | Medium | Medium |
| Turbo blower | 30,000–5,000 | Small | High | Medium | Medium | Medium |
| Multistage turbo blower | 8,000–1,000 | Small | High | Low | High | Large |

Table:3.1 shows specific speed ranges and other features of various types of fans [22]

### 3.1.2 OPTIMUM $D_1$ AND BLADE ANGLE AT ENTRY $\beta_1$

In the impeller design ratio $d_1/d_2$ plays an important role as impeller losses depend upon this ratio. With decreasing ratio $d_1/d_2$ the blade passage becomes narrower and longer. The maximum velocity-which generally appears in the impeller, is the entry relative velocity $W_1$. This also causes impeller losses and hence we must pay due attention to its magnitude. The simplest assumption to be made is that the smallest possible entry velocity $W_1$ is required, so that the losses can be reduced. If the following conditions are satisfied, then objective is achieved. [5]

$$\beta_1 \leqq 35.26^\circ \qquad (3.2)$$

$$\frac{d_1}{d_2} \geqq 1.194 \sqrt[6]{\varphi} \qquad (3.3)$$

### 3.1.3 OPTIMUM ENTRY BREADTH $b_1$

The factors involved in determining the size of the axial breadth $b_1$ of a blade entry can be readily obtained. Before the introduction of air into the impeller, the air must be turned through an angle of 90° (approximately) from the axis of the suction of intake duct. This is analogous to a change of direction occurring at a bend. The inside radius of curvature, however, is not always adequate in fans because of the insufficient room. In order to save expense in manufacture it will often be found that the rounding-off of the front shroud is omitted.

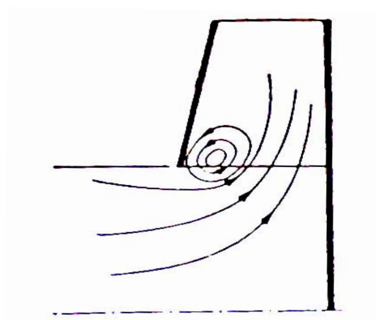

Figure: 3.2 Back flow at inlet of impeller due to large inlet breadth $b_1$

Even due to a small radius a separation of flow occurs and it will lead to the pressure loss which will indirectly influence the impeller performance. Figure: 3.2 shows the results obtained by Bruno Eck during experiments conducted with an impeller that was too wide. The experimental results indicate the flow-separation zone quite clearly and show that it the entry breadth is only partially filled by the active stream which contributes towards a significant increase in the meridional velocity at this point. Due to this highly undesirable shock occurs with normal volume flow. A secondary effect is a “back flow” which could arise in the remaining portion of the impeller entrance. The air flowing back into the suction or intake is often responsible for large losses of energy.

To avoid this detrimental influence upon the impeller, separation of flow at the turn must be prevented. The most effective measure to combat separation at this point is to accelerate the main stream. The impeller shapes which yield such acceleration (20% acceleration) are schematically illustrated in Figure 3.3 and equation 3.6.[5]

$$A_1 = A_s \quad (3.4)$$

$A_s$ = Area of shroud

$A_1$ = Impeller inlet area

$$\therefore b_1 = 0.221 \times d_1 \text{ (for no acceleration)} \quad (3.5)$$

$$A_s = 1.2A_1 \quad (3.6)$$

$$\therefore b_1 = 0.21 \times d_1 \text{ (for 20\% acceleration)} \quad (3.7)$$

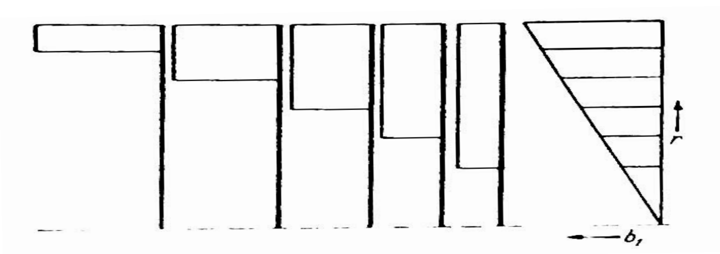


Figure: 3.3 possible shapes of impellers at 20% inlet acceleration

Recent investigations have shown, as opposed to earlier opinion, that the acceleration is not necessary. In fact, under certain circumstances, a considerable deceleration can be possible and even desirable.

The value of blade width will evidently affect the air volume because with the increase in the blade width the air volume also increases up to a certain point. When this point reaches, a further increase in $b_1$ will not increase the air volume because the constant inlet cone inside diameter will act as a choke. It simply will not allow any more air to pass through. The question is how large can we make $b_1$ for optimal performance, i.e. for maximum air volume, without impairing the efficiency. The answer to this question is $b_{1max}$ should be such that:

$$A_s = 2.1A_1 \quad \text{------------} \quad (3.8)$$

$$\therefore b_{1max} = 0.46 \times d_1 \text{ (maximum width)} \quad (3.9)$$

Equation:3.8 is an empirical formula. It is the result of experimentation. It makes allowance for the inlet cone inside diameter being somewhat smaller than the shroud inside diameter.[22]

Thus, it can be summarized that the 90° change of direction of flow cannot be adequately supported by the simple analogy of “flow at a bend” because the energy is being transferred from the blades to the fluid and in certain cases a phenomenon occurs similar to boundary-layer suction.[2]

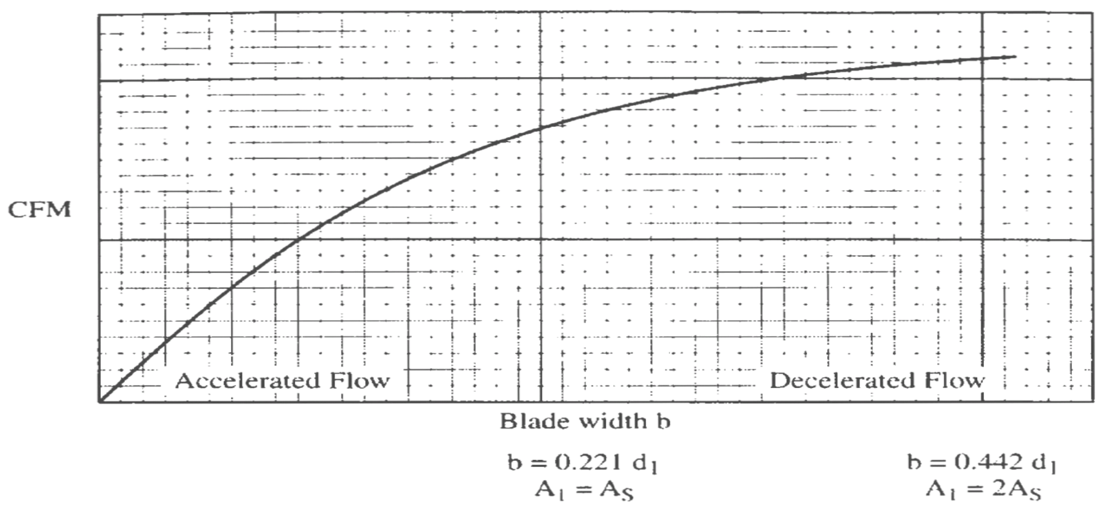


Figure: 3.4 Significance of impeller eye area and blade inlet area

### 3.1.4 IMPELLER VANE DISCHARGE ANGLE $\beta_2$

This is the most important single design element. It has been shown that theoretical characteristics are determined by the vane angle alone. In actual turbo machinery $\beta_2$ is still the deciding factor in design. All the design constants and proportions depend on the value of $\beta_2$. Therefore a choice of $\beta_2$ is the main step in selecting impeller design constants. This selection is based on consideration of the desired steepness of the head- capacity curve, operating range and whether or not a maximum output is desired from the impeller for a given diameter at the selected speed.[10]

Both head and capacity increase with the angle $\beta_2$. For that reason, for aircraft use where size and weight are highly evaluated, centrifugal superchargers use $\beta_2 = 90°$. This also permits a maximum peripheral speed not possible with lower values of $\beta_2$ where mechanical strength is a factor. However, reduction in size of blowers is accomplished at the expense of efficiency.

Both impeller and casing efficiency falls off appreciably with increasing values of $\beta_2$. The difference in the blower efficiency may be of the order of five to ten percent for $\beta_2 = 25°$ and $\beta_2 = 90°$, or about one percent for each 5°. This can be easily understood in view of the fact that the flow through the impeller and casing is decelerating and higher impeller discharge angles lead to a rapidly divergent channel, requiring a great number of vanes. Velocities leaving the impeller are higher for higher values of $\beta_2$ (higher percentage of kinetic energy) and conversion of velocity into pressure in the casing incurs losses which are difficult to control.

In the centrifugal pump industry, it has been well established that the optimum performance is realized with impeller angles of about 25° for all specific speeds. Experience with blowers follows the same trend. Whenever possible, lower values of $\beta_2$ are chosen, and higher values are employed only to obtain maximum head for a given maximum selected value of the peripheral velocity as determined from considerations of strength.

### 3.1.5 NUMBER OF BLADES Z

The number of blades in a centrifugal blower can vary from 2 to 64 depending upon the application, type and size. Too few blades are unable to fully impose their geometry on the flow, whereas too many of them restrict the flow passage and lead to higher losses. Most of the efforts to determine the optimum number of blades have resulted in only empirical relations given below.[2]

$$Z = 8.5 \times \frac{\sin\beta_2}{1 - \frac{d_1}{d_2}} \qquad (3.10)$$

$$Z = \frac{\beta_2}{3} \qquad (3.11)$$

$$Z = \frac{6.5(d2 + d1)\sin\frac{1}{2}(\beta_1 + \beta_2)}{(d2 - d1)} \qquad (3.12)$$

The first equation given above is recommended by Bruno Eck, the second one is suggested by Pfleirderer and the third one is experimentally derived by Stepanoff.

### 3.1.6 THE HEAD OR PRESSURE COEFFICIENT ψ

It is a dimensionless performance parameter that expresses the actual head in a fraction of the maximum theoretical head at zero flow for a given impeller tip speed and geometry with radial inlet. Relative to actual head the head coefficient is a cumulative representation blade angle slip and efficiency. The head coefficient ψ is defined as:

$$\psi = \frac{H}{U_2^2/g} \text{ or } H = \psi\frac{U_2^2}{g} \qquad \text{---------} \qquad (3.13)$$

Where $H$ = equivalent head for given $P_S$
$U_2$ = peripheral velocity at impeller exit.

### 3.1.7 THE FLOW OR CAPACITY COEFFICIENT φ

Flow coefficient is a dimensionless number which shows the flow handling capability of a blower. It gives general idea about impeller geometry and efficiencies anticipated for a calculated flow coefficient. [10] It is defined as:

$$\phi = \frac{V_{m2}}{U_2} \qquad \text{-----------} \qquad (3.14)$$

Where $V_{m2}$ = meridional velocity at impeller discharge
$U_2$ = peripheral velocity at impeller exit

For the design point, based on the net discharge area (excluding vanes) and ignoring the leakage, after the pressure coefficient is selected and $U_2$ is established, $V_{m2}$ can be calculated. The capacity coefficient increases for higher specific speed at constant values of $\beta_2$.

### 3.1.8 BLADE INSIDE DIAMETER $d_1$

The minimum value of the blade inside diameter $d_1$ can be calculated using following formula:

$$d_{1min} = \sqrt[3]{\frac{Q}{N}} \times 10 \qquad \text{---------------} \qquad (3.15)$$

From the above equation we can say that the blade inside diameter depends on the speed and air flow rate but not on static pressure. The value of the speed and flow rate is taken as per the customer requirements. If the volume flow rate is taken larger at lower speed the inside diameter will increase. The static pressure will be produced after the blade inside diameter $d_1$ has been passed, by blades which start at $d_1$ and extend to $d_2$.[22]

### 3.1.9 IMPELLER OUTSIDE DIAMETER $d_2$

Impeller outside diameter can be found out using following equation.

$$d_2 = 18000 \times \frac{\sqrt{P_s}}{RPM} \qquad \text{-----------} \qquad (3.16)$$

From the equation we can see that the blade outside diameter depends on speed and static pressure but not on air volume. Larger static pressure and smaller speed will result in a larger blade outside diameter. Larger diameter will allow more air volume to enter but it will be taken care by the blade inlet and outlet diameter.[22]

### 3.1.10 SHROUD DIAMETER $D_S$

The shroud diameter is very near to the inside diameter $d_1$. Its difference $d_1$ - $d_S$ is small, just large enough to allow for the curved portion of the shroud $d_S$.[22] It is taken as:

$$d_s = 0.94 \times d_1 \qquad (3.17)$$

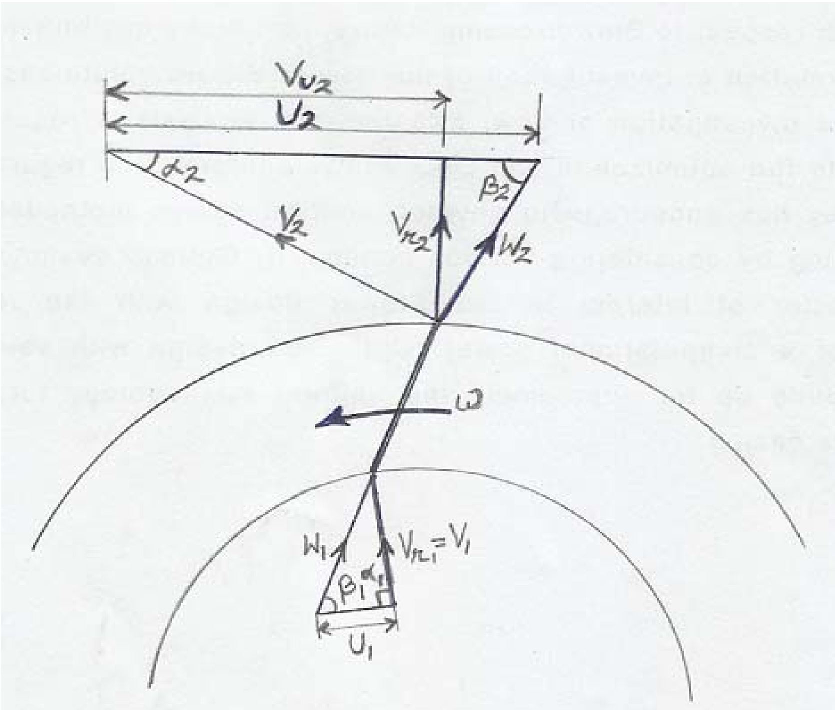


Figure: 3.5 Inlet and Outlet Velocity Diagram for impeller.

As per the figure: 3.5, we can find various parameters to get the detail theoretical condition at inlet and outlet of impeller. At the outlet of impeller flow with these conditions will enter the casing. To get this condition at inlet and outlet of impeller some steps are to be followed.

### 3.1.11 DESIGN STEPS

Following are steps for the design of impeller of blower. This design steps are to be required to satisfy all the limitations and get flow condition as per theoretical design. This will give base for further design for casing. Where this velocity transformation will takes place in pressure rise.

$$d_{1min} = \sqrt[3]{\frac{Q}{N}} \times 10$$

$$d_2 = 18000 \times \frac{\sqrt{P_s}}{RPM}$$

$$d_s = 0.94 \times d_1$$

$d_S$ = Eye or shroud

$$A_s = \pi \times \left(\frac{d_s}{24}\right)^2$$

$A_1 = A_s$

$$b_1 = 0.221 \times d_1$$

$$A_1 = \pi \times d_1 \times \frac{b_1}{144} \text{sq.ft}$$

$$U_1 = \frac{\pi d_1 N}{12} \text{fpm}$$

**1**

**1**

$$V_1 = \frac{Q}{A_1}$$

$$\beta_1 = \tan^{-1}\left(\frac{V_1}{U_1}\right)$$

$$W_1 = \frac{V_1}{\sin \beta 1} \text{fpm}$$

$$b_2 = b_1$$

$$A_2 = \frac{\pi d_2 b_2}{144}$$

$$U_2 = \frac{\pi d_2 N}{12} \text{fpm}$$

$$V_{r2} = \frac{Q}{A_2} \text{fpm}$$

Taking $\beta_2 = 45^\circ$

**2**

**2**

$$V_{u2} = U_2 - \frac{V_{r2}}{\tan\beta_2}\ \text{fpm}$$

$$V_2 = \sqrt[2]{V_{u2}^2 + V_{r2}^2}$$

$$W_2 = \frac{V_{r2}}{\sin\beta 2}$$

$$\epsilon = \frac{d_2}{d_1}$$

Є = Diameter

Taking $Z = 14$

$$\epsilon_{limit} = e^{8.16 \times \frac{\sin(\beta_2)}{Z}}$$

$Є_{limit}$ = Limiting diameter

**IF $Є_{limit} < Є$**

$$\mu = \frac{V'_{u2}}{V_{u2}} = \left(1 - \frac{\sqrt{\sin\beta_2}}{Z^{0.7}}\right)$$

$\mu$ = Slip factor

**3**

**3**

**IF $Є_{limit} > Є$**

$$\mu = \frac{V'_{u2}}{V_{u2}} = \left(1 - \frac{\sqrt{\sin\beta_2}}{Z^{0.7}}\right)\left(1 - \frac{\left(\frac{1}{\epsilon}\right) \quad \left(\frac{1}{\epsilon_{limit}}\right)}{1 - \left(\frac{1}{\epsilon_{limit}}\right)}\right)^3$$

$$\mu = \frac{V'_{u2}}{V_{u2}}$$

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

**Here, $Є_{limit} < Є$**

$$V'_{u2} = \mu \times V_{u2}$$

$$V'_2 = \sqrt[2]{V'^{2}_{u2} + V_{r2}^2}$$

$$W'_2 = \sqrt[2]{(U_2 - V'_{u2})^2 + V_{r2}^2}$$

**4**

**4**

$$\beta_2' = \sin^{-1}\left(\frac{V_{r2}}{W_2'}\right)$$

$\beta_2'$ = Actual blade exit

$$\alpha_2' = \sin^{-1}\left(\frac{V_{r2}}{V_2'}\right)$$

$\alpha_2'$ = Actual air exit angle

$$V_{eye} = Q \times \frac{4}{\pi \times d_s^2}$$

$$\Delta P_{entry} = K_i \frac{1}{2} \rho V_{eye}^2 ,$$

$K_i$=0.5, $\rho$=1.23

$K_i$ = Loss factor $\approx$ 0.5 to 0.8 $\approx$ 0.15 to 0.25

**5**

**5**

$$\Delta P_{Passage} = K_{ii} \frac{1}{2} \rho \left(W_1 - W_2'\right)^2$$

$K_{ii}$=0.2, $\rho$=1.23

$$\psi = \frac{V_{u2}'}{U_2}$$

$$\phi = \frac{V_{r2}}{U_2}$$

**6**

## 3.2 VOLUTE CASING DESIGN

Volute casing is generally suitable for single stage blowers. Its main advantage is its small size and low cost. The selection of design elements of volute casing like volute area, volute angle, volute width etc. are governed by theoretical considerations discussed below but their actual values have been established experimentally for the best performance. Many researchers had developed systematic design of centrifugal blower although each has a slightly different method of calculation. However, the principles for all are comparable. Casing is design on two methods constant velocity and constant Angular momentum methods.

### 3.2.1 DESIGN ON ASSUMPTION OF CONSTANT VELOCITY DISTRIBUTION

It has been established experimentally that volutes with a constant average velocity for all sections result in the best efficiency. In this, the volute is designed on the basis of constant mean velocity throughout the volute. For obtaining high efficiency, it is found, from the experience, that it is necessary to maintain constant velocity of the fluid in the volute passage at design point. This would also give uniform static pressure distribution around the impeller. This method is shown in the following steps for designing a volute.[22],[9]

**6**

$V_3 = R \times V_2'$ , R=0.6

$$H = U_2 \times \frac{V_{u2}'}{9.81}$$

H = Actual head

$$\text{Pressure head} = \text{Actual head} - \left(\frac{V_3^2}{2g}\right)$$

$$P_r = 1 - \left(\text{pressure head} \times \frac{\gamma - 1}{(\gamma \times R \times T)}\right)^{\frac{\gamma}{\gamma-1}}$$

$P_r$ = Pressure ratio

**7**

**7**

$$Q_v = Q \times \left(\frac{1}{P_r}\right)^{\frac{1}{1.4}}$$

$Q_v$ = Volute exit flow rate

$$A_v = \frac{Q_v}{V_3}$$

$A_v$ = Area of volute throat

$$d_3 = 1.1 \times d_2$$

$$r_3 = \frac{d_3}{2}$$

$$r_\theta = r_3 + \frac{\theta \cdot Q_v}{2 \cdot b_2 \cdot 360 \cdot V_3} \qquad (\because \text{considering } h_3 = 2h_2)$$

**8**

Volute widths $b_3$ are typically taken equal to $2b_2$ or $1.25b_2$. The volute widths do not affect performance appreciably. Rather, smaller volute width yields bigger volutes. [9] Thus, in present work, volute width $b_3 = 2b_2$ is preferred in order to achieve a smaller design.

### 3.2.2 DESIGN ON ASSUMPTION OF CONSTANT ANGULAR MOMENTUM

If ideal flow is considered the law of constant angular momentum will apply and the flow will leave the impeller on a logarithmic spiral path. The flow for a free vortex flow follows the equation, $r*V_u$ = constant, where r is the radius of a point lying outside the impeller and $V_u$ is the tangential component of velocity at that point. If the flow through the volute is incompressible and volute has constant width, then the direction of streamline remains constant.[9] This method of designing a volute is used where friction is ignored.

**For free vortex flow:**

$$r * V_u = r_1 * V_{u2} = \text{Constant}(C) \quad \text{------------------------------------} \quad (3.18)$$

$$V_u = {}^{C}\!/_{r}$$

As the cross section of volute selected is rectangular, the direction of streamlines remains constant.

$$\tan\alpha = \frac{V_r}{V_u} = \text{constant}$$

The total volume (Q) of the flow supplied by the impeller is uniformly divided at the volute base circle. Therefore, the flow rate at a section of the volute passage θ degrees away from the section at $\theta = 0^o$ is

$$Q_\theta = \left(\frac{\theta}{360}\right) * Q$$

Q=Area*Velocity

$Q_\theta$ = volume at volute passage angle

Consider an infinitesimal section of cross section $(b * dr)$ then the flow rate through it is,

$$dQ_\theta = V(b * dr) = V_u(b_3 * dr) \qquad \text{(As } b_3=b_4\text{)}$$

$$dQ_\theta = C * b_3 * (dr/r)$$

For full cross-section of volute passage,

$$Q_\theta = C * b_3 * \ln\left(\frac{r_4}{r_3}\right)$$

$$Q_\theta = \left(\frac{\theta}{360}\right) * Q$$

$$\therefore \ln\left(\frac{r_4}{r_3}\right) = \frac{\left\{Q * \left(\frac{\theta}{360}\right)\right\}}{(C * b_3)}$$

For rectangular cross section, the radius $r_4$ of the volute boundary from $\theta=0^\circ$ to $\theta=360^\circ$.

$$r_4 = r_\theta = r_3 * \exp\left(\frac{\theta}{360} * \frac{Q}{cb_4}\right) \qquad \text{----------------------------------} \qquad (3.19)$$

After designing the volute by both the methods, it is found that the radius of casing by constant velocity method is smaller than that of constant angular momentum method. So, design of volute as per constant velocity method is more compact and economical and hence it is chosen in the present work.

**8**

$$\Delta P_{casing} = K_{III}\frac{1}{2}\rho\,(V_2{}' - V_3)^2, K_{III} = 0.4$$

$\Delta P_{casing}$ = Pressure loss in casing due to turbulence and friction

$\Delta P_{entry}$ = Pressure loss at impeller entry

$$\Delta P_{total} = \Delta P_{casing} + \Delta P_{entry} + \Delta P_{Passage}$$

$$\text{New } P_s = P_s - \Delta P_{total}$$

$$Q_L = \pi\, C_D\, \delta\, d_1 \sqrt{\frac{\frac{4}{3} \times \text{new } P_s}{\rho}}$$

$Q_L$ = Leakage across impeller inlet and casing

$\delta$ = Radial clearance between impeller inlet and casing

**9**

**9**

$$\text{New discharge } Q = Q - Q_L$$

$$\tau = \frac{f\,\rho\,\omega^2\,r_2^5}{5}, f = 0.005, \omega = \frac{U_2}{r_2}$$

$\tau$ = Torque due to disk friction

$\omega$ = Rotational velocity

$\rho$ = Air density

$$P_{disc} = \tau \times \omega$$

$$\eta_h = \frac{P_s}{P_s + \Delta P_{casing} + \Delta P_{entry} + \Delta P_{Passage}} \times 100$$

$\Delta P_{passage}$ = Pressure loss in impeller blade passages

**10**

**10**

$$\eta_v = \frac{Q}{Q + Q_L} \times 100$$

$$\eta_t = \eta_h \times \eta_v \times 100$$

$\eta_t$ = Total efficiency

$\eta_v$ = Volumetric efficiency

$\eta_h$ = Hydraulic efficiency

$$P = \frac{\left(P_s + \Delta P_{casing} + \Delta P_{entry} + \Delta P_{Passage}\right) \times (Q + Q_L)}{\eta_t} + P_{disc}$$

$P_{disc}$ = Power loss due to disc friction

P = Power required to run the blower

$$\epsilon = \frac{H_{act}}{H_{th}}$$

$\epsilon$ =Volute efficiency or effectiveness

$H_{act}$= Actual head develop in volute casing

$H_{th}$ = Theoretical head develop in volute casing

Considering all above steps from inlet to outlet of the blower we can design the complete system as per requirement. To get the complete value of parameters some calculation be made.

## 3.3 CALCULATED PARAMETERS

By considering discharge, static pressure and specific speed further calculation are summarized in the table.

| SR. NO. | DESIGN PARAMETERS | NOMENCLATURE | VALUES |
|---|---|---|---|
| 1 | Minimum impeller inlet diameter | $d_{1min}$ | 0.183 m |
| 2 | Impeller outside diameter | $d_2$ | 0.318 m |
| 3 | Eye or shroud diameter | $d_S$ | 0.172 m |
| 4 | Area of shroud | $A_s$ | 0.023 $m^2$ |
| 5 | Impeller inlet blade width | $b_1$ | 0.040 m |
| 6 | Impeller inlet area | $A_1$ | 0.023 $m^2$ |

| | | | |
|---|---|---|---|
| 7 | Blade peripheral velocity at inlet | $U_1$ | 27.252 m/s |
| 8 | Absolute velocity at impeller inlet | $V_1$ | 21.611 m/s |
| 9 | Blade angle at impeller inlet | $\beta_1$ | 38.435$^o$ |
| 10 | Relative velocity at impeller inlet | $W_1$ | 34.782 m/s |
| 11 | Impeller outlet blade width | $b_2$ | 0.040 m |
| 12 | Impeller outlet area | $A_2$ | 0.0404 $m^2$ |
| 13 | Outlet blade velocity | $U_2$ | 47.504 m/s |
| 14 | Radial component of outlet velocity | $V_{r2}$ | 12.388 m/s |
| 15 | Blade exit angle | $\beta_2$ | 45$^o$ |
| 16 | Tangential component of outlet velocity | $V_{u2}$ | 35.12 m/s |
| 17 | Absolute velocity at impeller outlet | $V_2$ | 37.24 m/s |
| 18 | Relative velocity at impeller outlet | $W_2$ | 17.52 m/s |
| 19 | Diameter ratio | $\epsilon$ | 1.743 |
| 20 | Number of blades | Z | 14 |
| 21 | Limiting diameter ratio | $\epsilon_{limit}$ | 1.510 |
| 22 | Slip factor | $\mu$ | 0.867 |
| 23 | Actual exit velocity peripheral component due to slip | $V_{u2}$’ | 30.46 m/s |
| 24 | Actual absolute exit velocity | $V_2'$ | 32.878 m/s |
| 25 | Actual relative velocity | $W_2'$ | 21.07 m/s |
| 26 | Actual blade exit angle | $\beta_2'$ | 36.01$^o$ |
| 27 | Actual air exit angle | $\alpha_2'$ | 22.14$^o$ |
| 28 | Air velocity at impeller eye | $V_{eye}$ | 21.61 m/s |
| 29 | Pressure loss at impeller entry | $\Delta P_{entry}$ | 143.648 Pa |
| 30 | Loss factor | $K_i$ | 0.5 |

| | | | |
|---|---|---|---|
| 31 | Air density | $\rho$ | 1.23 |
| 32 | Pressure loss in impeller blade passages | $\Delta P_{passage}$ | 23.069 Pa |
| 33 | Loss factor | $K_{ii}$ | 0.2 |
| 34 | Pressure / Head coefficient | $\psi$ | 0.641 |
| 35 | Flow coefficient | $\varphi$ | 0.261 |
| 36 | Average volute velocity | $V_3$ | 19.727 m/s |
| 37 | Actual head generated by impeller | H | 147.486 m of air |
| 38 | Pressure head in volute | $H_v$ | 127.652 m of air |
| 39 | Pressure ratio | $P_r$ | 1.001 |
| 40 | Volute exit flow rate | $Q_v$ | 0.499 $m^3/s$ |
| 41 | Area of volute throat | $A_v$ | 0.025 $m^2$ |
| 42 | Volute tongue diameter | $d_3$ | 0.350 m |
| 43 | Volute tongue radius | $r_3$ | 0.175 m |
| 44 | Volute radius at various angles from tongue | $r_\theta$ | 0.21452 m |
| 45 | Volute width | $b_3$ | 0.08 m |
| 46 | Pressure loss in casing due to turbulence and friction | $\Delta P_{casing}$ | 42.5477 Pa |
| 47 | Loss factor | $K_{iii}$ | 0.4 |
| 48 | Total pressure loss | $\Delta P_{total}$ | 207.265 Pa |
| 49 | New static pressure | New $P_s$ | 773.98 Pa |
| 50 | Static pressure | $P_s$ | 981.25 Pa |
| 51 | Leakage across impeller inlet and casing | $Q_L$ | 0.0166 $m^3/s$ |
| 52 | Radial clearance between impeller inlet and casing | $\delta$ | 0.002 |
| 53 | Discharge coefficient | $C_D$ | 0.5 |
| 54 | Volume flow rate | Q | $0.5\frac{m^3}{s}$ |
| 55 | New discharge | New Q | $0.4834\frac{m^3}{s}$ |

| | | | |
|---|---|---|---|
| 56 | Torque due to disc friction | $\tau$ | 0.0091 N-m |
| 57 | Power loss due to disc friction | $P_{disc}$ | 2.715 Watts |
| 58 | Hydraulic efficiency | $\eta_h$ | 82.56 % |
| 59 | Volumetric efficiency | $\eta_v$ | 96.78 % |
| 60 | Total efficiency | $\eta_t$ | 79.90 % |
| 61 | Power required to run the blower | P | 771.14 Watts |

All the above values give the design conditions of the blower system. But to get at off-design conditions or with change in the system one should know the effect of flow parameters and limitations of various parameters at outlet and inlet of the system. So further mismatching of impeller in the casing, which is designed with some another conditions, this is needed to be studied.

## 3.4 CONCEPT OF MISMATCHING FOR IMPELLER

As per design we generally, in blowers, impeller flow is collected directly by the volute. The volute passage is designed to operate with some degree of diffusion i.e. velocity is reduced and Kinetic energy is converted to pressure. For designing a volute, a base circle of radius ($r_3$) is selected which is little larger than impeller radius. With this radius as a base circle, volute is designed as per given below.

1) Constant means velocity design.

Now, as per in designing of volute casing for forward impeller using constant velocity method:

Absolute velocity of flow (V) has two components, $V_u$ and $V_r$ as increase in $V_u$ then average volute velocity. As $V_u$ increases then area of volute casing will decrease and when $V_u$ decrease area of the volute casing increased.

In a constant velocity method design, the boundary radius ($r_4$) is determined using the expression

$$r_4=((\theta/360)*(Q_v/(V_3*b_3)))+r_3 \quad \text{---------------------------(A)}$$

Where θ is angular position, Q is volute discharge and $V_3$ is average volute velocity for constant velocity method, width $b_3$ is constant. For given values of $r_3$, Q and $b_3$, $r_4$ depends on the values of $V_3$. The reduction is as shown diagrammatically in figure: 3.6.

Now consider three impellers, Backward ($\beta_2$<90), Radial ($\beta_2$=90) and Forward ($\beta_2$>90) having same discharge Q, same outer radius ($r_2$) and operating at same rotational speed (N). Velocity diagrams for three impellers are as under:

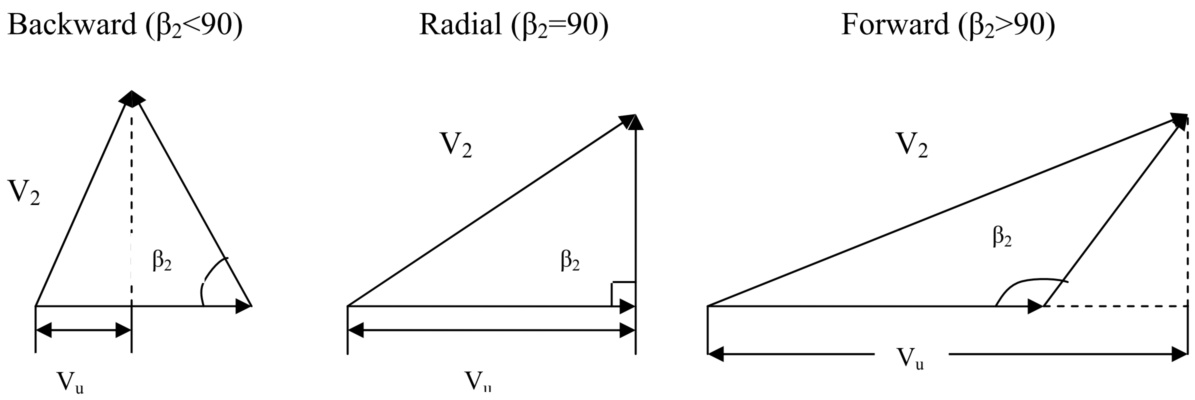


Figure: 3.6 Outlet velocity triangles

Comparison of their velocity diagrams shows that:

1) $V_u$ is highest for forward and hence the value of $V_3$ is also highest for the forward i.e. the value of $V_3$ decreases with decrease in $\beta_2$.

2) The absolute velocity of flow, $V_2$, is highest for the forward and lowest for the backward i.e. the value of $V_2$ decreases with decrease in $\beta_2$.

Therefore, to achieve greater reduction in velocity (higher diffusion), the size of the volute should increase with increase in $\beta_2$. But the application of equation (A) results into decrease in size of volute with increase in angle $\beta_2$.i.e smaller size diffuser for forward compared to radial as shown in figure:3.6.

Here as shown in the table: 3.2 & 3.3 for the different values of outlet angle for impeller, volute casing design parameters are given. From this table we can observe that, from backward to forward direction the area of the volute will decrease as shown in figure:3.7. In table: 3.4 show that volute angle decreases with increase in blade angle. We have adopted constant velocity method for volute casing design.

## TABLE:3.2 FOR CALCULATION OF VOLUTE CASING AS PER CONSTANT VELOCITY METHOD

| SR. No | Outlet Blade Angle $\beta_2$ in degree | $Vu_2$ in m/s | Act Head H m of air | Head co-efficient $\psi$ | Capacity co-efficient $\varphi$ | $\alpha'_2$ in degree | Act Velocity $V_2'$ in m/s | $Rc_3$ | Avg.Volute Velocity $v_3$ in m/s | Pressure Head in volute Hv in m. | Pressure Ratio | Qv volute flow $m^3/s$ | Area Av in $cm^2$ |
|---|---|---|---|---|---|---|---|---|---|---|---|---|---|
| **1** | 30 | 35.797 | 206.142 | 0.486 | 0.256 | 27.873 | 35.462 | 0.64 | 22.696 | 179.888 | 0.997 | 2.727 | 1201.869 |
| **2** | 48 | 49.579 | 271.852 | 0.641 | 0.256 | 21.851 | 44.543 | 0.64 | 28.507 | 230.430 | 0.997 | 2.726 | 956.444 |
| **3** | 70 | 58.469 | 301.887 | 0.711 | 0.256 | 19.855 | 48.813 | 0.64 | 31.240 | 252.143 | 0.997 | 2.726 | 872.627 |
| **4** | 90 | 64.500 | 326.587 | 0.770 | 0.256 | 18.459 | 52.362 | 0.64 | 33.511 | 269.347 | 0.996 | 2.725 | 813.365 |
| **5** | 100 | 67.422 | 343.069 | 0.808 | 0.256 | 17.628 | 54.746 | 0.64 | 35.037 | 280.499 | 0.996 | 2.725 | 777.879 |
| **6** | 110 | 70.532 | 364.171 | 0.858 | 0.256 | 16.665 | 57.813 | 0.64 | 37.000 | 294.393 | 0.996 | 2.725 | 736.528 |
| **7** | 120 | 74.068 | 391.519 | 0.923 | 0.256 | 15.559 | 61.809 | 0.64 | 39.558 | 311.762 | 0.996 | 2.724 | 688.807 |
| **8** | 130 | 78.405 | 427.149 | 1.007 | 0.256 | 14.317 | 67.045 | 0.64 | 42.909 | 333.305 | 0.996 | 2.724 | 634.898 |
| **9** | 140 | 84.249 | 473.636 | 1.116 | 0.256 | 12.961 | 73.917 | 0.64 | 47.307 | 359.570 | 0.995 | 2.723 | 575.751 |
| **10** | 150 | 93.203 | 536.715 | 1.265 | 0.256 | 11.481 | 83.294 | 0.64 | 53.308 | 391.873 | 0.995 | 2.722 | 510.796 |

## TABLE:3.3 FOR AREA OF VOLUTE CASING AT DIFFERENT OUTLET ANGLE

| θ in degree | $R_\theta$ at 30deg. in cm | $R_\theta$ at 48deg. in cm | $R_\theta$ at 70deg. in cm | $R_\theta$ at 90deg. in cm | $R_\theta$ at 100deg. in cm | $R_\theta$ at 110deg. in cm | $R_\theta$ at 120deg.in cm | $R_\theta$ at 130deg.in cm | $R_\theta$ at 140deg.in cm | $R_\theta$ at 150deg. in cm |
|---|---|---|---|---|---|---|---|---|---|---|
| **0** | 23.375 | 23.375 | 23.375 | 23.375 | 23.375 | 23.375 | 23.375 | 23.375 | 23.375 | 23.375 |
| **45** | 31.749 | 28.122 | 27.570 | 27.225 | 27.063 | 26.763 | 26.452 | 26.132 | 25.807 | 25.475 |
| **90** | 40.123 | 32.868 | 31.766 | 31.076 | 30.751 | 30.151 | 29.529 | 28.888 | 28.239 | 27.576 |
| **135** | 48.497 | 37.615 | 35.961 | 34.926 | 34.439 | 33.539 | 32.606 | 31.645 | 30.670 | 29.676 |
| **180** | 56.871 | 42.361 | 40.156 | 38.776 | 38.127 | 36.927 | 35.683 | 34.402 | 33.102 | 31.777 |
| **225** | 65.245 | 47.108 | 44.352 | 42.626 | 41.815 | 40.315 | 38.760 | 37.158 | 35.534 | 33.877 |
| **270** | 73.619 | 51.854 | 48.547 | 46.477 | 45.503 | 43.703 | 41.837 | 39.915 | 37.966 | 35.977 |
| **315** | 81.993 | 56.601 | 52.742 | 50.327 | 49.191 | 47.091 | 44.914 | 42.672 | 40.398 | 38.078 |
| **360** | 90.367 | 61.347 | 56.938 | 54.177 | 52.879 | 50.479 | 47.991 | 45.428 | 42.829 | 40.178 |

## TABLE:3.4 FOR VOLUTE ANGLE AND LENGTH OF THROAT FOR DIFFERENT OUTLET ANGLE

| Outlet Blade Angle $\beta_2$ in degree | Length Of Throat L In cm | Volute Angle $\alpha_v$ |
|---|---|---|
| | | |
| **30** | 66.992 | 33.393 |
| **48** | 37.972 | 21.852 |
| **70** | 33.563 | 19.856 |
| **90** | 30.802 | 18.459 |
| **100** | 29.504 | 17.629 |
| **110** | 27.104 | 16.665 |
| **120** | 24.616 | 15.560 |
| **130** | 22.053 | 14.317 |
| **140** | 19.454 | 12.962 |
| **150** | 16.803 | 11.481 |

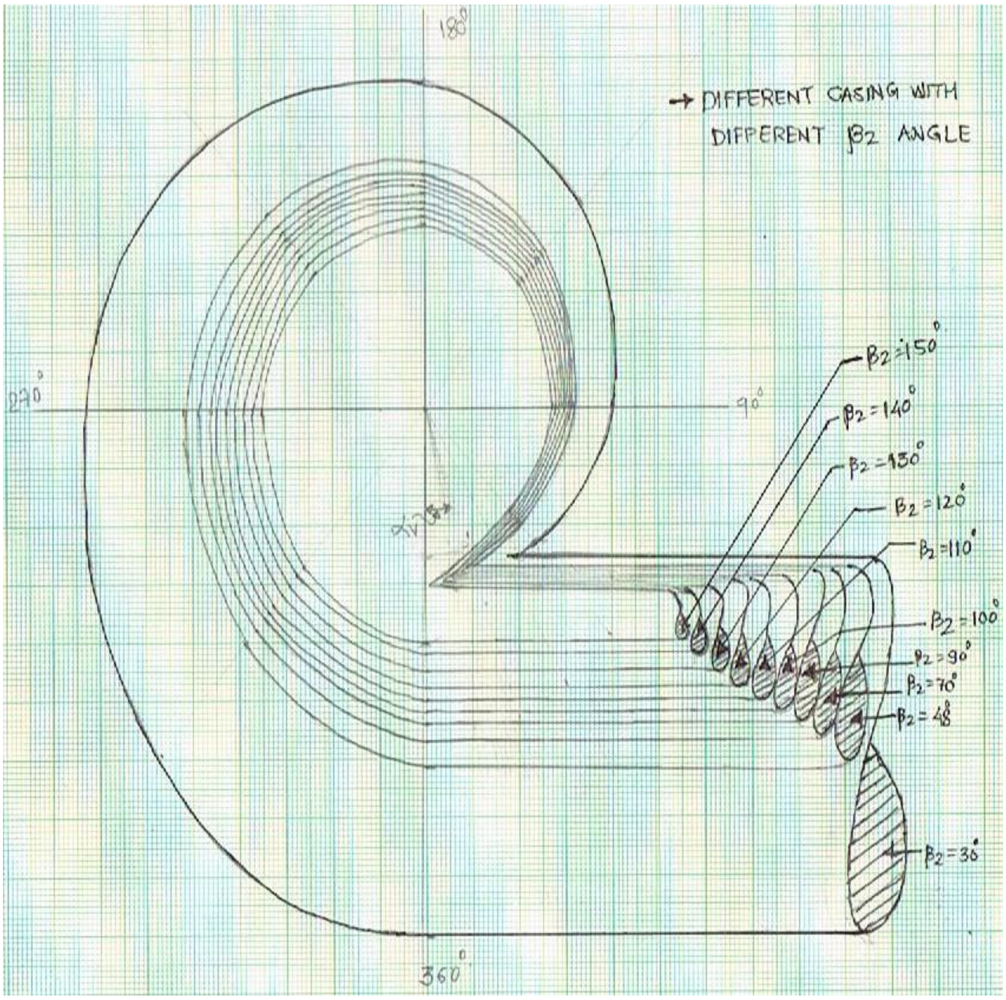


Figure: 3.7 Size of Volute with different outlet angle

In the design of volute casing, volute should serve its function of pressure recovery. This will help in improving the efficiency of the system. Increasing demands for electrical efficiency has renewed interest in improving the efficiency of the system. After designing the casing of blower, it should be optimized design in respect to get maximum efficiency with respect to air-flow, static pressure and available power.

### 3.5 OPTIMIZATION:

Efficiency (η) is the main objective of the optimization. All independent variables are varied to check whether they increase η or not. The combinations which will give higher η are selected.

$d_2$ is the secondary objective of optimization process. The higher values of $d_2$ increases size of machine and consequently its cost increases.

Heuristic and Genetic Algorithm both method were applied to get optimal solutions by considering the overall design of a blower. All independent variables are identified and varied to check whether they increase or which combination will give higher efficiency. Finally, Casing has been designed as per the requirement of the system.

### 3.5.1 INDEPENDENT VARIABLES:

- $\beta_2$, Z, $d_1$, $b_1$ and $b_2$.
- $\beta_2$ is primary independent variable. Choosing $\beta_2$ is the key step in design [22]. It is varied through whole span. $\beta_2$ = 15° to 65° as this is the BC type impeller.
- Z is varied corresponding to $\beta_2$. The equation for number of blades is given in Eck Bruno [22].
- $d_1$ is secondary independent variable. $d_1$ is varied to satisfy diameter ratio condition for centrifugal type impellers that is $d_1/d_2$ = 0.5 to 0.8. After deciding $d_1$ (for which highest $\eta$ is obtained) next variable i.e. $b_1$ is varied to check next.
- $b_1$ is varied from $0.208d_1$ to $0.46d_1$ [22].
- $b_2$ is varied corresponding to $b_1$, such that the angle of divergence is not greater than 12° [22].

### 3.5.2 CONSTRAINTS:

- $V_{eye} < 22$ m/s; as the higher velocities in inlet duct induce more losses, higher value of $V_{eye}$ should be avoided [5].
- $d_1 / d_2$ = 0.5 to 0.8, this is a typical value of centrifugal impellers.[22]
- $(d_1/d_2) / (1.194*\varphi^{1/3})$ = 1 to 1.1, this condition is derived in Eck Bruno [22]

- $b_1/d_1$ = 0.208 to 0.46. The values greater than 0.221 decelerate the flow while the values less than 0.221 accelerate the flow. There is difference of opinion for these conditions in various references. That's why both conditions are checked. [5]

- $b_2 < b_1$ and divergence angle < 12°. A larger value of divergence angle will induce a greater loss that's why it should be avoided[2][22].

- $\beta_1$ = 15° to 35°. The higher values of $\beta_1$ induce more losses due to shock and separation. Higher values should be avoided [22].

- $V_{m1}/V_{m2}$ = 1.25 to 1.6. The meridional (radial) velocity ratio should be obeyed in order to achieve a good design [10].

- $W_1/W_2$ > 1.05. Typically, relative velocity at inlet is higher than relative velocity at exit [10].

- $b_3$ are typically taken equal to $2b_2$ or $1.25b_2$

Above all conditions give the optimum result of design of the casing. But cost reduction with material used for fabrication is available. So this analysis is required. As per the application and pressure build up in the casing, we can decide the selection of material for casing.

### 3.6 MATERIAL SELECTION FOR VOLUTE CASING

In volute type of design it is found that the pressure distribution is approximately constant at the design flow, but as the flow departs from designed conditions the distribution changes as shown in figure: 3.8

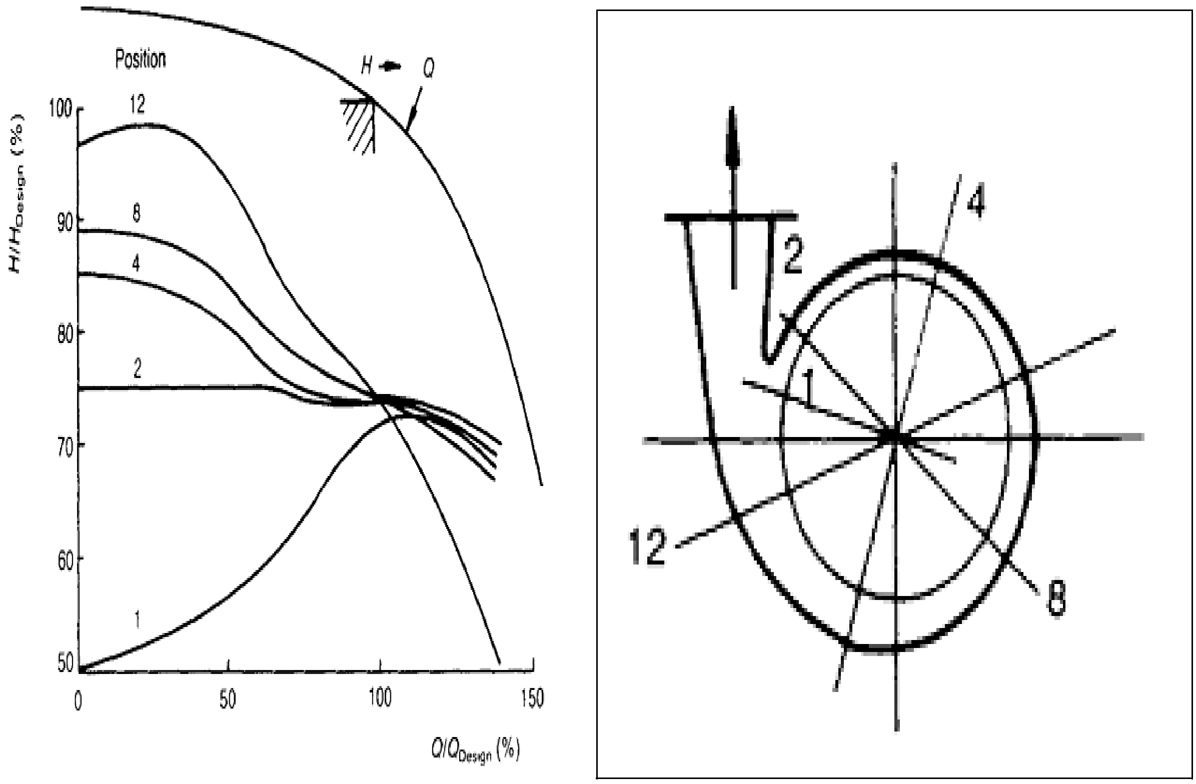


Figure: 3.8 Variation of pressure round an impeller.

The radial thrust force is calculated by

$$F_r = K\rho g H D_2 B_2$$

Where K is an empirical constant defined as

$$K = 0.36[1 - (Q/Q_{design})^2]$$

All indicate that the resulting radial loads to be absorbed by the bearing supporting the shaft tend to be large and cannot be neglected. The formulae are used to size shafts and bearing. One device for reducing the radial loads is the double volute. The penalty is increased friction loss which may result in probably a 2% reduction in efficiency. Axial thrust is the summation of unbalanced forces acting on the impeller in the axial direction. Reliable thrust bearings are available so that this does not present problems except in large machines. The forces arise due to the distribution of pressure in the space between the impeller and the casing. The pressure expansion is to be calculated for casing size and wall thickness as the equation given below.

$$\Delta Pressure = \left(\frac{1}{2*E_t} - \frac{V_{min}}{E_h}\right) * \left(\frac{P*r}{t}\right)$$

$\Delta Pressure$ = $Pressure\ expansion, Pa$

$E_t$ = $Axial\ Tensile\ modulus, Pa$

$V_{min}$ = $Minor\ poisson's\ ratio$

$E_h$ = $Hoop\ tensile\ modulus, Pa$

$P$ = $Internal\ pressure, Pa$

$r$ = $outside\ radius, m$

$t$ = $wall\ thickness, m$

The force magnitude depends on size, outlet pressure, and rotational speed. Pressure distribution in the casing, from the tongue region to the exit, is a deciding factor in determining strength of the material of the casing and support required to hold the casing. If pressure distribution is not given due consideration then it may result into collapse of the casing or damage at the joints. Generally, materials such as MS, casting, sheet metal or FRP are used in fabrication of casing. Total assembly requires perfect support and foundation; otherwise it will lead to failure. To get the detail flow analysis in the volute casing for determining the various flow parameters, experimental measurement is required. To get the confidence level of the experimental value uncertainty analysis is required.

### 3.7 SUMMARY

Proper design and optimization of blowers is determined as per system factors and prioritizing them into requirements versus preferences. Some of these factors are the static pressure the blower must overcome, the average air flow volume required, the shape and direction of the desired air flow, space limitations, mismatching of impeller, available power, efficiency, material selection and cost. These all factors are used to determine critical parameters when selecting a blower. The blower curves enable accurate blower selections based on static pressure and flow rate. Design of the Blower is divided in two steps, first impeller design and second casing design. Effect of the optimization is analyzed by considering two methods such as Genetic Algorithm method and combined heuristic method. Considering the various Independent variables such as $\beta_2$, Z, $d_1$, $b_1$ and $b_2$ and satisfying the constraints such as $V_{eye} < 22$ m/s, $d_1 / d_2 = 0.5$ to 0.8, $(d_1/d_2) / (1.194*\varphi^{1/3}) = 1$ to 1.1, $b_1/d_1 = 0.208$ to 0.46, $b_2 < b_1$ and divergence angle < 12°, $\beta_1 = 15°$ to 35°, $V_{m1}/V_{m2} = 1.25$ to 1.6, $W_1/W_2 > 1.05$. These all above conditions will give an optimized designed.

Once the Blower casing is designed, to validate its rated performance experimental analysis is required. To get the detail flow parameter inside the volute casing and region of interaction between casing and impeller, a flow measuring setup is required. This study outlines a general procedure that other researchers may use to determine three-hole and five-hole probe for flow measurement with the help of water tube manometer, result uncertainty and provides guidance to improve measurement technique.