6 Chapter

# RESULT AND DISCUSSION

## 6.0 GENERAL

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, available power, efficiency, modification in casing, change of impeller, 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. Considering all above factors analysis is done in casing and result are categorized in three parts:
1) Experimental results 2) CFD results and 3) Optimization results.

## 6.1 EXPERIMENTAL RESULTS

The experiment is carried out to know the flow condition within the volute based on Free Vortex design, Constant Mean Velocity, Modification in volute casing based on constant mean velocity, Mismatching of impeller in volute casing and optimized design of casing. Using a calibrated probe, observations were taken at different planes in the casing and observed flow parameters.

### 6.1.1 DESIGN BASED ON FREE VORTEX METHOD:

Using a Five hole probe, observations were taken at different planes as shown in figure: 6.1A. The flow measurement is carried out at seven different radial-axial planes, 1-7. All these radial planes and also the planes A, B, C & D are 30° apart from each other. The readings were taken at five different radial positions from near to impeller to outer wall; i.e., 0, I, II, III, IV. They are equally placed in the radial direction. For each hole; i.e., for ‘0’, 31 readings are taken axially; each at 5mm distance from both the walls for first few readings and in between at a distance of 10mm, starting from motor side wall.

### 6.1.2 FLOW DIRECTION WITHIN THE VOLUTE

The flow directions were determined with tangential directions. The magnitude and velocity direction at various axial positions in radial tangential plane is represented at each location from 1 to 6 as shown in figure: 6.2. These figures show that the flow is more or less tangential over the impeller and it is more sort of a random in nature, in the overhung portion. As majority of the mixing takes place in this region.

Design of the volute is based on free vortex method, to verify the free vortex condition within the volute, the product of radius and tangential component of velocity are plotted, as shown in figure: 6.3. The constant values are plotted at some of the positions along the width of the impeller radially. As indicated by the figure the values are varying. This means that the flow is not obeying the free vortex condition. The variation of the value from the average is approximately ±20%.

### 6.1.3 FLOW AT OUTLET OF IMPELLER:

The flow after leaving the impeller enters the volute casing. To determine the transformation of kinetic energy and subsequent loss that may takes place, the flow parameters were determined at the exit of the impeller. At the impeller exit flow was surveyed at planes A, B, C, D and 1.0, 2.0, 3.0, 4.0, 5.0, 6.0, 7.0 locations. The flow condition all around the impeller with variation in stagnation pressure across the width of the impeller, just at the exit of it and along the whole axial length as shown in figure: 6.4 & 6.5. It has observed that impeller delivers flow with large variation in stagnation pressure at different angular positions. The humps and valley indicates this trend. This means that the flow is not delivered by constant stagnation pressure by impeller. The stagnation pressure is very low in the overhung portion particularly at tongue. This may be because of effect of tongue on the flow parameter. To determine the lost co-efficient the area average value of stagnation pressure was determined. The average value obtained 189.7054 mm of water gauge ($P_{0av}$).

### 6.1.4 FLOW CONDITIONS IN VOLUTE CASING:

The flow condition at various planes is presented in the form of contours of stagnation pressure, static pressure and flow velocity. The contours of stagnation pressure are presented in figure: 6.6(A-G). The plot shows that stagnation pressure varies across the volutes casing in all directions axial, radial and stream wise. The values do not remain constant as delivered by the impeller. The stagnation pressure gradient over the impeller region marked by region ‘A’ is relatively less compared to overhung portion of the volute. Minimum stagnation pressure regions visible in the overhung portions of the volute as marked by ‘B’, this region of low pressure gradient grows as flow moves towards exit of the volute, that means there is subsequent growth of low pressure region. This low pressure region is roughly about 60% to 65% away from the wall of the impeller tip. It occupies considerable space along radial direction. Almost at all planes relatively higher value of stagnation pressure is observed at the wall of volute. The static pressure plots as shown in figure: 6.7(A-G), that relatively there is lower pressure at the exit of the impeller, gradually grows to a higher value towards the periphery. This is because of subsequent conversion of kinetic energy into pressure. Nearly higher values of pressure are found at the region near to the periphery. The gradient near to the volute wall particularly at plane 7 is relatively less. The pressure and velocity plots revels that there is relatively reduction in the value of velocity. This is because of conversion of kinetic energy to pressure. The contours of flow velocity are depicted in figure:6.8(A-G). It shows that there is no uniformity of flow velocity across the volute. At plane-1 it forms nearly 3 pockets of different velocities as marked by ‘A’, ‘B’, ‘C’. at the exit of the impeller as marked by regain ‘A’ the velocity is relatively higher. The reduction in velocity takes place along radial direction at all planes. As the flow moves towards discharge and the flow velocity decreases at plane 5, 6 and 7. The region of low velocity is quite high except in the vicinity of impeller. Very low velocity region is noticed at plane 3 and 4.

### 6.1.5 TRANSFORMATION OF ENERGY:

The conversion of kinetic energy into static pressure is expressed in non-dimensional form. The gain in pressure is non-dimensionlised by dynamic head at exit of the impeller. The dynamic head at exit of impeller is given by ½ $\rho C_2$ and this is equal to ($P_0$ – $P_s$). Since the value of $P_0$ and $P_s$ both are varying, their average values are taken for determination of average value of dynamic head. Average value of $P_0$ i.e. $P_{0av}$ at exit of impeller is 180.7054 mm of water gauge and average value of static pressure is 113.8885 mm of water gauge. The values of $C_p$ calculated throughout volute casing are plotted in figure:6.9(A-G). The value of $C_p$ is low near to the impeller tip radial position. It increases along the radial position. The value of $C_p$ is higher near the peripheral wall of volute casing. Relatively there is lower variation in $C_p$ at plane 1. At plane 6 and 7 the value of $C_p$ is almost same in axial direction within the volute region. As the values are varying locally the average value of $C_p$ was determined for all the planes applying area average technique. As observed above there is rise in static pressure as the flow moves from tongue to discharge of the volute casing. The volute as such forms a diffusing passage has the viscous effect, causes losses in the flow. The losses occurring through the volute casing are expressed in terms of loss co-efficient the loss co-efficient is defined as $w = \frac{P_{0av} - P_0}{P_{0av} - P_{av}}$ where $P_0$ is local stagnation pressure and $P_{0av}$ and $P_{av}$ are as mentioned above. The loss co-efficient is plotted in terms of contours at various angular planes such as 1, 2, 3, 4, 5, 6 and 7. They are presented in figure: 6.10(A-G). It is notice that the loss coefficient is having positive values indicating loss in stagnation pressure. Some pockets also indicate negative loss co-efficient indicating higher value of stagnation pressure than the average delivered by the impeller. Over hung portion causes more losses compared to the region above the impeller along the radial direction. The overall average value of the lost co-efficient of the volute was also determined and its value is 0.0502.

### 6.1.6 DESIGN BASED ON CONSTANT VELOCITY METHOD

Using a Five hole probe, observations were taken at different planes as shown in figure: 6.1B. The flow measurement is carried out at seven different radial-axial planes1to7 and also the planes A, B, C & D are 30° apart from each other. The readings were taken at five different radial positions from near to impeller to outer wall; i.e., 0, I, II, III, IV. They are equally placed in the radial direction. For each hole; i.e., for ‘0’, 31 readings are taken axially; each at 5mm distance from both the walls for first few readings and in between at a distance of 10mm, starting from motor side wall.

### 6.1.7 STAGNATION PRESSURE VARIATION IN THE CASING

The average value of stagnation pressure at different angular position of holes near the impeller as shown in figure: 6.11. The value of stagnation pressure increases very gradually from suction to exit of the volute casing. The flow condition around the impeller is presented in figure: 6.12 from which we can see that impeller delivers flow with large variation in stagnation pressure at angular position from a to 1 compared to 2 to 7. Also at overhung portion the stagnation pressure is very low and doping particularly near tongue region. Total pressure variation at holes near impeller is more and it is also observed that very less variation along the radial direction, as the values are almost constant as shown in figure:6.13. The total pressure in front of the impeller is highest at the section–7 and the lowest at section-1. In radial direction, the variation in the values of stagnation pressure is lower across the casing width for all section except at section–1. The stagnation pressure gradient decreases, as the flow moves along the radial outward directions.

### 6.1.8 STATIC PRESSURE VARIATION IN THE VOLUTE CASING

The pressure rise is higher in the region in the vicinity of the impeller in radial direction, where it is lower near the outer wall of the volute casing as shown in figure: 6.14. Static pressure increases as we move from suction side to exit side, which indicate pressure rise in volute casing. It is observed that, near to the volute wall rise in static pressure is very less, whereas higher near the region in the vicinity of the impeller exit. High-pressure flow occupies almost all the region in the vicinity of the volute wall. The variations in the static pressure were negligible at the outward radial position.

### 6.1.9 VELOCITY VARIATION IN THE VOLUTE CASING

Velocity variations across the casing were shown by contour graph in figure: 6.15. As per the trend Value of velocity near impeller is higher. There is lower velocity region occur near the end of impeller width at holes near the impeller. The reduction in velocity is clearly indicated in the contour plot of velocity at section-7. The variations in the velocity are very less for holes position at the outward in radial direction.

### 6.1.10 RECOVERY OF KINETIC ENERGY INTO STATIC PRESSURE

The dynamic head is 59.745 mm of water, which was utilized for non-dimensional rise in pressure and loss. The average values were used to determine the dynamic head, which were obtained by area average technique and mentioned in table: 6-1. The rise in pressure is defined as pressure recovery coefficient and expressed as,

$$C_p = \frac{P - P_{av}}{P_{0av} - P_{av}}$$

The loss is defined as loss-coefficient and defined as,

$$W = \frac{P_{oav} - P_o}{P_{0av} - P_{av}}$$

| Values of $P_{oavg}$. & $P_{avg}$. near impeller holes along Impeller width | | | | | |
|---|---|---|---|---|---|
| | Section-1 | Section-3 | Section-5 | Section-7 | Average |
| $P_{oavg}$ | 157.8 | 161.3 | 161.4 | 174.8 | 163.825 |
| $P_{avg}$. | 84.3 | 102.5 | 113.8 | 123.7 | 106.08 |

Table: 6 - 1

The values of $C_p$ calculated throughout volute casing. Their variations are presented in figure:6.16. From all the graphs, it is clear that at section 1 value of $C_p$ is lowest and at section 7 it is highest, this is due to conversion of velocity into pressure. Also as we move radially outward the value of $C_p$ increases. $C_p$ comes out to be negative at section 1 at holes I, II & III and at section 3 which are near to impeller, which indicate that static pressure delivered by impeller is lower than average static pressure of volute casing. At overhung portion near casing wall, the variation of $C_p$ is very large, which may be due to wall effect in the flow.

The variation of loss coefficient (W) is represented in figure: 6.17. At 15.4% of impeller width the value of W is negative at hole near impeller for all sections, which indicate that stagnation pressure delivered by impeller is higher than the average stagnation pressure. As we move radially outward the value of W is decreasing. On the casing wall side of the overhung portion, the variations in loss coefficient were less compare to overhung portion of motor side & impeller width. From contours, it is clear that the flow near impeller width delivered at higher stagnation pressure than its average value, therefore this region having lower value of loss coefficient. From middle portion to end of impeller the local stagnation pressure is lower at position near impeller at section 1 & 3 and as we move towards exit of the casing loss coefficient is decreasing.

### 6.1.11 PITCH ANGLE VARIATION IN THE VOLUTE CASING

The local flow direction with axial direction of the blower casing is expressed in terms of pitch angle, for this purpose the probe was calibrated from +25 to –25 degree. The flow tilted along the axial direction, tilting of the flow with r-$\theta$ plane towards casing wall end side is negative and towards motor end side is positive.

Figure: 6.18 indicate that pitch angle varies at impeller exit between -5 to 20 degrees that means the flow has got small axial component towards motor end. The variation in pitch angle became very large from positive to negative in the vicinity of the tongue. Both the curves indicate large variations with pitch angle. As there is no definite trend of variation of pitch angle at hole near tongue we may conclude that the flow may get very much complicated near the tongue region.

The pitch angle variations at location designated at all sections that there is large variation in flow direction in terms of pitch angle and at position near the impeller the flow had positive value of pitch angle i.e. flow direction towards motor end side throughout the axial width and as we move radially outward, the flow direction tilted towards casing wall side.

### 6.1.12 FLOW DIRECTION WITHIN THE VOLUTE

The flow velocity and magnitude at various axial locations of volute casing along the width, from tongue to exit of volute casing is shown in figure: 6.19. The Vector plot of flow indicates that the velocity of flow is decreasing from tongue to exit of volute casing. At 92.3% of impeller width the flow velocity is very less as compared to other position of axial width. Flow gets very much diverted from its tangential direction at overhung portion of motor side i.e. –23% of impeller width.

## 6.2 COMPARISON OF FLOW PARAMETERS

To get the evaluation of flow, effect of both design are compared. As size of casing is different in both designs, the flow parameters will also differ.

### 6.2.1 BASED ON STAGNATION PRESSURE

More stagnation pressure gradients is found in the volute casing based on “Free vortex” principle i.e. flow became little smooth in the volute casing based on “Constant velocity Method”. At section-1 as referred in figure: 6.20 and 6.13 the high pressure region at overhung portions near casing wall side, is disappear in the volute casing based on “Constant velocity method”. The high pressure region near the impeller hub region had reduced in volute casing based on “Constant velocity method” compared to “Free vortex “designed volute casing. At section-5 there is large variation occur in the volute design based on “Free vortex method” and flow became quite homogeneous except near impeller in the volute casing design based on “Constant velocity method”.

### 6.2.2 BASED ON STATIC PRESSURE

It has observed lower gradient of Static pressure variation in the casing designed on “Constant mean velocity” compared with ”Free vortex method” as shown in figure: 6.21 & 6.14. In the case of “Constant mean velocity” casing, as we move radially outward, the variation in the static pressure became negligible and relatively larger portion is occupied near wall by high pressures region as compared to “Free vortex method”. Figure: 6.21-A shows the variation in static pressure before and after the modification in the suction side of the volute casing. The modification is shown in figure: 6.1C.

### 6.2.3 BASED ON VELOCITY

The gradient of velocity variations were more at section 1 & 3 compared to section 5 & 7(figure: 6.22 & 6.15) in the casing based on

“Free vortex method”, while in “Constant velocity method” velocity variation gradient is same for all the sections. Also at overhung portion value of velocity is higher in the volute casing designed by “Free vortex method” which turns to be moderate values in case of “Constant velocity method”. There is a reduction in the area of high velocity, at region near the impeller in case of “Constant velocity” designed volute compare to “Free Vortex” designed volute. In case of volute designed based on “Constant velocity method” flow became homogeneous except near impeller.

### 6.2.4 BASED ON PRESSURE RECOVERY COEFFICIENT

The region of high value of pressure recovery coefficient is observed in the “Constant velocity method” as compare to “Free vortex” designed volute casing. Better value of pressure recovery coefficient and smooth variation found in case of “Constant velocity method” (figure: 6.23 & 6.16). At section-5 more gradients were found in case of volute designed on “Free vortex method”, while improved flow conditions is observed in case of “Constant velocity method”.

### 6.2.5 BASED ON FLOW DIRECTION

In case of “Free Vortex Method” less deviation occur in the direction of flow, from its tangential direction, as compared to volute casing based on “Constant velocity method”. Reverse flow is obtained in the case of “Free vortex method” in the overhung portion of casing wall side for the region positioned near to volute exit, while in case of volute designed on “Constant velocity method” reversible flow not found.

Results of design conditions are plotted with various flow parameters. At off-design or mismatching of impeller with volute casing will give the flow behavior as per selection of casing and impeller.

## 6.3 MISMATCHING OF IMPELLER WITH VOLUTE CASING:

To analyze the flow in constant mean velocity designed based volute, which is designed considering backward type impeller having outlet blade angle of impeller $48^0$. Mismatching is done by fitting impeller of outlet blade angle $120^0$. To get the complete effect of this mismatching, the variable speed drive is connected with the motor. Ultimately, this mismatching will affect the load on the motor and flow in the volute casing. Performance of this modification is presented in the International Conference. [45]

### 6.3.1 FLOW VARIATIONS IN THE VOLUTE CASING

Stagnation and static pressure at different axial positions in the volute casing is shown in figure: 6.24. The stagnation pressure in front of impeller is highest at all section for axial distance 92% and +154% of impeller width, but it is decreases towards outer wall of volute casing at section 7. At overhung portion stagnation pressure is found to be highest at section 1&3. The variation is not much between section 3 and section 5. In Radial direction, stagnation pressure is lower at section 7 compared to other section across the casing width. Total pressure in front of the impeller is highest at section 7 and the lowest at section 1. Figure: 6.24 shows that the rate of rise in static pressure is higher in the region, towards the wall of the impeller in radial direction, where it is slightly lower near the outer wall of the volute casing. Static pressure increases, as we move from suction side to exit side, which indicates pressure rise in volute casing. When the flow comes angularly outward the static pressure is increases but slight variation is seen at section 5 and section 7.

Experimental analysis result will give the actual flow condition inside the casing. As per prediction of experimental flow, this will be guideline for theoretical analysis. CFD analysis can be effective tool after experimental verification.

## 6.4 CFD ANALYSIS

Unsteady flow simulations in the volute casing which is based on “free vortex method” i.e. “Constant angular momentum” having width of casing twice than impeller width of the blower have been carried out using the CFX- Tascflow code and the results are obtained. Discussion of the CFD and experimental analysis was presented in National and International Conferences. [43][44]

### 6.4.1 STATIC PRESSURE VARIATION IN THE VOLUTE CASING

Figure: 6.25 show various Chart of the static pressure at different axial positions of the volute casing. When the flow comes radially outward, the static pressure increases; these trends are observed at all angular positions. The variation in the value of static pressure for all angular position is higher near impeller and steadily increases near volute wall. At section 3 the low pressure region is created so that the static pressure at this section is lowest compared to the section 1.

Figure:6.25 shows the contour for static pressure variation at different axial positions. As we observed in the overhung portion, pressure conversion is very less. Plane at +92% to +177% of impeller width lower pressure region is progressively increasing.

### 6.4.2 STAGNATION PRESSURE VARIATION IN THE VOLUTE CASING

The variation in the stagnation pressure at different axial positions of the volute casing is shown in figure:6.26. The stagnation pressure for the section 7 is highest near the volute wall. We observed in the overhung portion that the stagnation pressure variation is increased. As the flow moves from tongue region to exit, the stagnation pressure is observed more or less constant in the plane -23% to +92% of impeller width. But in overhung portion stagnation pressure at plane +154% to +177% of impeller width progressively increases towards exit.

### 6.4.3 VELOCITY VARIATION IN THE VOLUTE CASING:

Figure:6.27 show the velocity variation at different axial positions. Obviously the velocity near impeller is higher and decreases as the flow proceed toward exit. In overhung portion velocity goes on decreasing.

### 6.4.4 NEAR THE TONGUE REGION

Near the tongue region high pressure region is observed at the plane -23% to +92.3% of impeller width. Stagnation pressure near the tongue region is observed maximum at the plane -23% to +54% of impeller width. Velocity near the tongue region is observed higher at all planes. In the overhung portion near the tongue region velocity is maximum at the plane +154% of impeller width. In vector plot as mixing of flow is observed and it goes on increasing in overhung portion.

Figure:6.27 indicate that the velocity of flow is decreasing from tongue to exit end of volute casing. In the overhung portion flow is non uniform where the magnitude of velocity is very low. Some disturb region is observed in front of the impeller. It is also observed that mixing of the flow occurs at the tongue region. Flow circulation is observed in between plane -3 & 5 and also at tongue region.

### 6.4.5 MISMATCHING ANALYSIS

Mismatching results of static pressure and velocity vector variation is shown in figure: 6.28 & 6.29 at 31% of impeller width. Variation in pressure is uneven with drastic drop. Velocity vector near the tongue region high and upto to $90^0$ of casing, further there is a drop in velocity upto the exit of the casing. Static pressure and velocity vector variation shown in volute casing design with forward type impeller having outlet blade angle $128^0$ is shown in figure: 6.30 & 6.31. As casing shows, both pressure and velocity increases gradually from tongue to exit.

Result of experimental analysis are done by using instruments as stated above, to check whether this instruments are set as per standard or having some errors. So to prediction of uncertainty analysis is required.

## 6.5 UNCERTAINTY ANALYSIS

All major factors affecting the measurement accuracy are considered in this section with possible remedial steps to minimize the measurement uncertainty. When the precision of measuring devices for each parameter, e.g., static pressure, differential pressure, temperature, etc., is known the flow measurement uncertainty associated with each parameter can be predicted. This prediction method is based on the theoretical relationship of each parameter to the flow. For measurement of pressure in the casing, inclined water tube manometer and for the measurement of angle of the flow, circular disc is used. Experimental uncertainty is calculated as shown in figure: 6.32. this is found to maximum 2.98801 and minimum 0.047782.

Experimental and CFD analysis is done on casing which is fabricated as per design conditions. Further, to get the effect of optimization volute casing design is optimized.

## 6.6 OPTIMIZATION

Optimization is the act of obtaining the best result under given circumstances. In design, construction and maintenance of Blower system, engineers have to take many technological and managerial decisions at several stages. The main aim of the optimization is to select the values for decision variables such that there will be optimum efficiency and product cost. The efficiency of the Blower casing can be increased mainly by optimization of Blower system. We have selected two methods for optimization 1) Heuristic method and 2) Genetic Algorithm. Optimization analysis was discuss and published in International conference. [47][49]

### 6.6.1 COMBINED HEURISTIC METHOD OF OPTIMIZATION

We have solved first by "one variable at a time" approach, the effects of variable on blower design are identified. From that approach "how these independent variables affect efficiency and how to vary them to satisfy individual constraints" is recognized. However, through these individual variations some constraints are satisfied while some are not. No design was found which satisfies all constraints and give highly efficient performance. In order to get acceptable design, we have tried to satisfy all the constraints. That's why approach illustrated in Tables A-1 to A-7 is employed.

Exit blade angle $\beta_2$ was discretized in different ranges. For individual $\beta_2$ firstly blade number Z was varied. Then inlet diameter $d_1$ was varied to get optimum efficiency for respective $\beta_2$. Then that $d_1$ was kept fixed while varying $b_1$ and then $b_2$. In each Table 2 to 3 designs were found which satisfy all the constraints and gives efficient, small sized and reliable design.

In Table A-8. the most efficient results from Tables A-1 to A-7 are compared. Most efficient result is for $\beta_2$ = 15°. But it does not satisfy the constraint $W_1/W_2' > 1.05$. Thus it should be omitted.

### 6.6.2 FINAL DESIGN FOR ORIGINAL REQUIREMENTS:

Remark the values of new Q and new $P_S$ in Table A-9. They are far different from original customer requirements i.e. Q = 0.5 $m^3/s$ and $P_S$ = 981.25 Pascal. That's why these results are iterated in order to "goal seek" original requirements. All parameters were modified up to 5% to achieve desired results and considering that the constraints are not violated.

Results of the goal seek iterations are shown in Table A-10. Mark that original requirements are achieved with all the constraints satisfied (except for $\beta_2$ = 15°). The efficiency is also preserved. The design with $\beta_2$ = 20° gives highest efficiency 89.09%. Efficiency reduces with increases in $\beta_2$. For $\beta_2$ = 65° achievable efficiency is 83.27%. However, this efficiency is quite high for 65° blower. This shows the significance of optimization done. The efficiency is increased almost 10% after optimization. As we see design graph, the original design efficiency was near 80%. Figure: 6.33 shows that entry and casing losses remain nearly constant for different $\beta_2$. While passage losses increase with increase in $\beta_2$. Figure: 6.34 show that leakage loss increase with increase in inlet diameter $d_1$. Figure: 6.35 shows disk friction torque and power loss (3-4 watts) increase with increase in outlet diameter $d_2$. Though it is only a small fraction of total power required to run the blower (680-880 watts). Figure: 6.36 shows that highest efficiency is achieved for $\beta_2$ = 15°. But it is omitted because constraint $W_1/W_{2'}$ > 1.05 is not satisfied. $\eta_v$ remains constant over entire range. $\eta_h$ decreases almost 2% with 10° increase in $\beta_2$.

### 6.6.3 GENETIC ALGORITHM

Optimization is carried out with Genetic Algorithm (GA), initially two independent variables outlet blade angle $\beta_2$ and number of blades Z. $\beta_2$ is the most important parameter in designing the blower because all the design constants and proportions depend on the value of $\beta_2$. Here, it is varied in the range of $15^0$ to $65^0$ for backward curved blades. The reason behind choosing Z as an independent variable is that it has a great impact on the flow and it is varied from 4 to 34 numbers. The optimization is carried out using GA tool in MATLAB as shown in figure: 6.37. The population size is taken as 20. Crossover and mutation probabilities are taken as 0.8 and 0.01 respectively. The program runs for 51 generations and after that it gives the optimum efficiency of about 82.67%. The second optimization is carried out with five independent variables. They are blade width at inlet ($b_1$), blade width at outlet ($b_2$), inlet diameter of an impeller ($d_1$), outlet blade angle ($\beta_2$) and number of blades (z). $b_1$ and $b_2$ are varied between 1.304 to 4.612 (0.208 $d_1$- 0.46 $d_1$). $d_1$ is varied between 6.267- 10.0264 (0.5 $d_2$ – 0.8 $d_2$). $\beta_2$ is varied between $15^0$ to $65^0$ and Z is varied between 4-34 numbers. As shown in figure: 6.38.

As shown in figure: 6.39 we have observed hydraulic efficiency is maximum about 85.5% and is almost constant with increase in blade number. Volumetric efficiency does not vary much with the increase in number of blades. It varies between 96.7% to 96.8%. Total efficiency is maximum about 89.7% and is constant with increase in number of blades. But as the losses are very high for higher number of blades it is preferable to keep the number of blades between 5 to 10. Leakage losses are maximum for $\beta_2$ value of 0.38 radians ($21.8^0$). It decreases gradually and the leakage losses are constant with increase in number of blades. Generally, the entry losses in the blower occur when there is a sudden turn of fluid at the eye of an impeller. It includes separation losses. It has found that entry losses are independent of the outlet blade angle $\beta_2$. Passage losses occur in the blade passage as fluid moves through the impeller. It has found that, passage losses are minimum for the value of $\beta_2$ 0.4 radians ($23^0$). Higher outlet blade angle ($\beta_2$) increases the losses in blade passage. This is because, as the outlet blade angle increases, relative velocity decreases and the passage losses depend on the difference of relative velocities, which increases with the increase in outlet blade angle. Casing losses are minimum for the $\beta_2$ 0.3 radians ($17.2^0$). Casing losses depend on the actual absolute velocity at the outlet of the impeller, which increases with increase in $\beta_2$ and results in higher pressure losses in casing.

As shown in figure: 6.40, Total losses vary with respect to outlet blade angle $\beta_2$. It has minima at 0.38 radians ($21.8^0$). Variation of hydraulic efficiency with change in $\beta_2$ and It is maximum of about 85.5% at $\beta_2$ 0.38 radians ($21.8^0$). It decreases to 81% for higher value of $\beta_2$. Hydraulic efficiency depends on pressure losses at different sections of blower. It is observed that pressure losses are minimum for $\beta_2$ between 0.3 to 0.4 radians and hence, hydraulic efficiency is higher in that range of $\beta_2$. Volumetric efficiency has very less effect of change in $\beta_2$. It increases gradually above $\beta_2$ value of $21.8^0$. It has observed that total efficiency is maximum of about 82.7% at $\beta_2$ value $21.8^0$. It decreases gradually to value 78.5% with increase in $\beta_2$, because pressure losses are increasing. It has been also found that entry losses are constant irrespective of the change in number of blades. Pressure losses in blade passage, casing losses and the total pressure losses increase with increase in number of blades.

The same optimization is carried out using Genetic Algorithm keeping same above mentioned five variables. The population size is taken as 20 and crossover and mutation probabilities are taken as 0.8 and 0.01 respectively. The final results are obtained after 51 generations. The efficiency obtained with genetic algorithm is about 90.5% which is higher than efficiency obtained with two variables and Combined heuristic. GA has given very good selection with respect to $R_{req}$ $\beta_2$ , $N_t$ and $N_s$ as shown in figure: 6.41 & 6.42. Selection of volute size can directly obtain from the figure : 6.43. As per optimum value of $d_2$ we have to add to the graph value. Volute angle can be obtained from the figure : 6.44 for the particular $\beta_2$. From GA we have obtained the corrected volute angle value with respect to $\beta_2$. Verification of this optimized design experimental analysis is done.

## 6.7 EXPERIMENTAL ANALYSIS OF OPTIMIZED DESIGN

We have design volute casing for exhaust system, efficiency is the main objective of the optimization. All independent variables are varied to check whether they increase efficiency (η) or not. In optimization, combinations which give higher η are selected. Size is the secondary objective of optimization process. The higher values of outer diameter of machine ($d_2$) increase the size of machine and consequently its cost also. The specific speed $N_S$ is a characteristic number from which centrifugal machine range can be calculated. Purpose of these casing was to design for exhaust air handling machine for small range. Results of these optimize design machine were presented in International Journal. [46][48]

- **Dimension of Impeller:**

  Inner Diameter ($d_1$) = 0.10 meter

  Outer Diameter ($d_2$) = 0.26 meter

Speed of Impeller (N) = 800 -1200 rpm

$\eta_{total}$ = 80.78%

Impeller of this unit having following specification:

Type: Backward inclined blade type

Inlet Blade Angle: 30.41 deg. with tangential direction,

Outlet Blade Angle:70.62 deg. with tangential direction.

Thickness of the blade: 2mm

Inner diameter: 10 cm

Outer diameter: 26 cm

No of blades=10 nos.

The designed volute casing is mild steel sheet of 2 mm thickness. It is designed on the basis of "Constant Mean Velocity". In this the casing wall at casing wall side overhung portion, designed parallel to impeller taper shape as shown in figure: 6.45. To investigate the flow through volute casing, impeller is run by a motor having (Rated input: 63 Watt, Voltage: 220/230 A.C) capacity and variable rotational speed of 800-1200rpm.

We have observe, flow angle of air in volute casing is near about constant. It slightly fluctuate between at an angle 270° to 360° as shown in figure: 6.46. At exit of the impeller flow angle is fluctuating from $100^0$ to $350^0$. In this case, the flow angle is progressively increasing up to 270° and then decreasing as shown in figure: 6.47. So it is observed that the flow angle is more stable at casing wall than impeller periphery. It has found that volute effectiveness $\epsilon$ = 79%.

### 6.7.1 PRESSURE VARIATION IN VOLUTE CASING

The stagnation pressure, dynamic pressure and static pressure are shown in figure: 6.48 & 6.49. It is observed that, static pressure is maximum at about 315˚.it is lower at 360˚. The static pressure and dynamic pressure are not constant from entry to exit of volute due to uneven behaviors of velocity of flow. In this case, from 45˚ to 90˚ the static pressure is suddenly increased. After that it is decreased up to

225°and again pressure is recovered highest level at an angle 270°. It is observed in both case, Static pressure is higher near the casing periphery than at impeller exit because conversion of kinetic energy into pressure energy. Static pressure is low near the tongue. From the tongue portion it is progressively increased up to angular position 270° of casing and again it falls during rest of casing. In both cases, it is observed that static pressure is higher near an angle 270°.

### 6.7.2 FLOW VISUALIZATION

As casing is design for exhaust system, so smoke pattern is visualize at suction of casing without and with nozzle as shown in figure: 6.50. As it is observed that flow is uniformly entering in the casing with nozzle, this will helpful in getting more height to underside of smoke layer and height of enclosure containing the smoke reservoir.

Effect of pressure and velocity over the volute casing of the blower can give the selection of material for casing. As vibration is generated, support is required to hold the casing. This can be evaluated by static and dynamic characteristics.

## 6.8 Static and Dynamic Analysis:

Casing is a stationary part of blower, but it experiences static and dynamic pressure. It is also affected by vibration from the motor side. So, selection of material for fabrication of casing plays very important role. As shown in figure: 6.51& 6.52 region of deformation as per the default thickness of the material 5mm. Figure: 6.53 shows equivalent stress region with maximum and minimum stress. Material selection is as per 1998 ASME BPV code, section8, geometry is much simple and load is applied as per standard conditions. Only the analysis can vary with the mesh quality, figure: 6.54 & 6.56 shows the mesh statistics and mesh over the blower, which was taken for this analysis. For the verification of mesh sensitivity, further increase in mesh quality shows no effect in results as shown in figure:6.55 & 6.57. Suction pipe length can create effect over the casing, Figure:

6.58 shows static analysis considering different length of suction pipe. Figure: 6.59 shows the result after CFD analysis such that output of CFD analysis will be input to static analysis. As per these effect of pressure variation in the casing, we get the result with reduction in material thickness. Conveying blowers are equipped with special impeller geometry to allow different materials to be conveyed directly and reliably. so it can helpful to choose between aluminum or steel casing. The casing can made in different materials as per requirement and applications and are predestined for customization as shown in figure: 6.60.

## 6.9 Summary

The problems for the investigation of flow in volute casing of Blower is formulated and it is found that the results and its interactions are projected into three main dimensions of the analysis like experimental, CFD and optimization. Experimental results are plotted as per the variation of flow properties such as static and stagnation pressure and velocity variations inside the volute casing of different type as per stated earlier. Further, CFD analysis is carried out with the volute casing on which the experiment is performed. To validate the experimental results uncertainty analysis is carried out. And finally optimization of the design is done and results are discussed.

As per the results obtained from experimental and CFD analysis at various boundary conditions are quite satisfactory. CFD analysis gives the clear picture of analysis of flow in volute casing with various conditions. Static analysis is very helpful as per the consideration of pressure distribution by CFD analysis. Finally, conclusion can be done from the results obtain from all the analysis. This will be helpful in determine the future scope.