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# Design Optimization and Parametric Study on the Alternative Blower of Travelling Cleaner

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**Abstract**

Centrifugal blowers are widely used in different industrial applications because of their suitability in any practical circumstance. Therefore it is necessary to find out the parameters that affects the efficiency and power consumption of the blower. In this present study, the blower serves as a fluffs collecting system for travelling cleaner. Taguchi's orthogonal array method has been implemented and the parameters chosen for optimizing the blower are fan outer diameter, blade tip angle, and number of blades. Based on the input data the design calculations has been carried out and modeled using SOLIDWORKS. The cleanup and meshing are carried out in ICEM CFD and the problem is solved using FLUENT V6. The post processing is carried out in CFD POST and the corresponding results are mentioned below. Using Minitab statistical tool the response of parameters and the optimum combinations has been determined.

***Keywords:*** **Blower, Computational fluid dynamics, Taguchi's method, Specifications, Minitab tool.**

## 1. Introduction

Travelling cleaner is a device which is used to clean the machines at working condition by moving on the rail system. The cleaner blows the air around the machines and sucks unwanted fluffs from the floor in order to maintain the quality of the product. The main motor is fixed at one end of the railing with drive pulley and a driven pulley is fixed at other end of the railing system which helps to operate the cleaner using Belt transmission system. The dust and fluffs are collected by the suction pipes which was fixed at both sides of the machines get filtered at the top of the central fan. The filtered air is then discharged at high pressure on the floor near the machine thus keeping the floor also clean. The accumulated fluffs and dust are removed using an automatic centralized waste collection system. The present automatic centralized waste collection system consists of pipeline connection and separate blower system with 5/7.5 HP motor, fitted at one end of the plant. The pipe line connection is used to transport the waste from the machine to centralized blower.

Centrifugal blowers are widely used in different industrial applications because of their suitability in any practical circumstance. The main parts of centrifugal blowers are inlet, fan, volute casing, and outlet. The air comes in radially and goes out tangentially thus provides high static pressure and discharge.

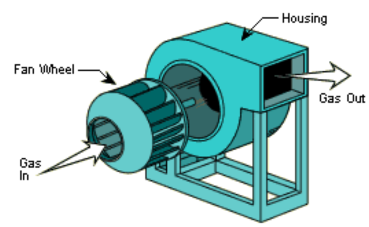



Fig. 1 Parts of centrifugal blower

The fan rotates inside the casing creates low pressure at the inlet thus allows the air to flow inside, which is further passed through the blade passages. Due to centrifugal motion, the air is pushed out of the fan passage and thus flows through the spiral casing. The spiral casing maintains the discharge of the flow and guides out the air to outlet. The geometry of the centrifugal fan plays an important role in their performance. Based on the fan geometry, it is classified as backward curved fan, forward curved fan, and radial blade fan. The forward curved and backward curved fans accumulates the fluffs at center, thus increases the power.

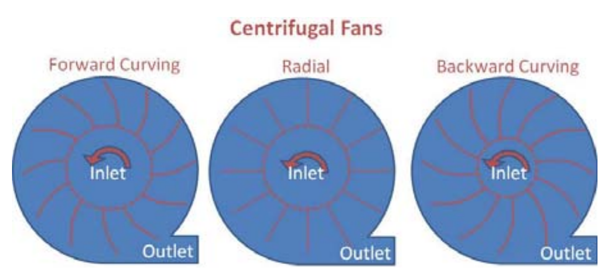



Fig. 2. Types of centrifugal fan

In this study, we used radial blade fan which is most commonly used for dust collection system. The parameters such as volute casing, inlet fan diameter, tip clearance, and fan clearance are kept constant during the analysis. The fan outer diameter, fan blade angle, and number of blades are considered as optimizing parameters which helps to get high static pressure with less power consumption.

# 1. Methodology

The methodology for this present study comprises of following steps:

i) Literature survey
ii) Basic design calculation
iii) Identification and selection of parameters for optimization
iv) Implementation of Taguchi's orthogonal array method
v) Modeling and analysis of blowers
vi) CFD results

## 1.1. Literature survey

### 1.1.1. Phase I – Blower types

The centrifugal fan uses the centrifugal power generated from the rotation of impellers to increase the pressure of air/gases. It may be classified into three basic types according to blade configuration: forward curve, backward inclined, and radial or straight blades [1]. Because of its high static pressure and capability of handling airstreams containing a high level of particulate, radial blade is suitable for the application of dust laden. The open wheel or paddle wheel is the most common of the radial blade impellers. Air wheel and radial tip impeller are the other variants of radial impellers ideal for contaminated airstreams but neither is intended for bulk material handling [2].

### 1.1.2. Phase II – Parametric study on blowers

The critical parameters of the centrifugal fan impeller is found out which highly affect the performance characteristics are impeller outlet width, impeller outlet diameter, blade thickness, impeller blade outlet angle, and number of blades [3]. Comparisons are conducted between the fan with original impeller and two larger impellers with the increments in impeller outlet diameter of 5% and 10% respectively in the numerical and experimental investigations. Experiment results show that the flow rate, total pressure rise, shaft power and sound pressure level have increased, while the efficiency have decreased when the fan operates with larger impeller [4]. The tip clearance (i.e. distance of the blade with top of the casing) has been deduced that the impeller with backward-curved blades was very sensitive, whereas the other two types were not. The impeller with radial tipped blades showed a weak dependency on tip clearance. However, for the case of fully radial blades, it has been observed that the fan is almost insensitive to the tip clearance [5]. The inlet radius has major impact on the flow rate of the centrifugal blower. Too small or too large of a radius will result in a noticeable loss in the flow rate. The bell mouth radius has moderate impact on performance of centrifugal blower. Too small of a radius will have negative impact on flow rate. Anyway, it's best to have the bell mouth radius ratio at the value around 9% [6].

### 1.1.3. Phase III – CFD Analysis

Computational Fluid Dynamics usually abbreviated as CFD, is a branch of fluid mechanics uses numerical methods and algorithms to solve and analyze problems that involve fluid flow [3]. The CFD part is used for improving the results of static pressure generated at the entry to the impeller, static efficiency. The MRF (moving reference frame) applied in the CFD analysis of centrifugal fan as a rotating region around the impeller and component of the impeller stationary [7]. A 250mm backward curved blower impeller in a shroud was tested and modeled in two CFD scenarios: one which resolves the impeller and shroud geometry and one which requires knowledge of the pressure versus flow curve and models only the shroud geometry. Comparing with the test data MRF models predict the flow fields much better than the lumped compact models using the P-Q curves [8].

### 1.1.4. Phase IV – Design of Experiments

Design of experiments (DOE) or experimental design is the design of any information-gathering exercises where variation is present, whether under the full control of the experimenter or not [3]. Taguchi orthogonal array (OA) based design of experiments (DOE) technique determines the required experimental trials. The experimental results are justified by Analysis of Variance (ANOVA) [9]. The blower housing was optimized using a Design of Experiments (DOE) technique where the geometry of housing was varied in a structured manner to capture expected sound order behavior [10].

Table 1: Design Specifications

| *Sr. No.* | *Parameter* | *Values* |
|---|---|---|
| 1 | Fan shaft diameter ,mm | 15 |
| 2 | Fan Speed ,rpm | 3660 |
| 3 | Fan Hub Diameter, mm | 35-55 |
| 4 | Fan Type | Radial |
| 5 | Pressure co-efficient | 0.7 |
| 6 | Discharge ,$m^{3}/$ hr. | 450 |

## 1.2. Basic Design calculation

For the development of the centrifugal blower and fan, the required discharge capacity is 450 $m^3$ / hr. (or) 0.125 $m^3$/ s.

The eye diameter of the fan,

$$D_0 = \sqrt{(4/\pi)(Q_0/V_0) + D_h^2} \quad .....(1)$$

Since the pulley shaft diameter is 15mm, the Hub diameter of the fan should be between 35 - 55 mm, we take it as 45mm and the air velocity as 45m/s.

$D_0$ = 74.577 mm

$D_0$ ~ 75 mm

Inlet diameter $D_1$ should be more than the eye diameter, say $D_1$ = **80mm.** Let N = 3660 rpm, now the inlet tip velocity Ui

$$U_1 = (\pi D_1 N) / 60 \quad .....(2)$$

As inlet is axial, there is no whirl component at inlet, so the triangle formed at inlet as an inlet triangle as right angle triangle,

$$\alpha_1 = 45^0$$

$$\cos \alpha_1 = U_1 / V_1 \quad .....(3)$$

Since flow is radial $V_1 = V_{r1}$

$$\text{Cos } 45 = U_1 / Vr_1$$

The inlet fan leaf area is required to construct the leaf position with required angle.

$$A = 1.03\, Q_0 / V_1 \quad .....(4)$$

During the movement, there is some air will be rotated at bottom of the disc, these losses consider as 3%.

Thickness factor consider as 0.8 because this leaf is transferring with air at particular density,

Inlet Width $b_1 = A_1/\pi D_1 \epsilon$ .....(5)

Flow inside the impeller is an adiabatic compression,

$$T_2/T_1 = (P_2/P_1)^{(k-1)/k} \quad .....(6)$$

Pressure Head

$$H = (RT / 0.285)\,((P_2 / P_1)^{(k-1)/k} - 1) .....(7)$$

Due to circularity, friction, and turbulence losses, 25% of head will be lost.

According to adiabatic compression $Vr_2 < Vr_1$

$$\Delta Vu = (0.6 * U_2 * \pi * \sin\beta_2)/Z \quad ....(8)$$

$$W_2 = \text{sq.rt } (Vr_2^2 + (U_2 - Vu_2)^2) ....(9)$$

$$W_1 = \text{sq.rt } (V_1^2 - U_1^2) \quad ..(10)$$

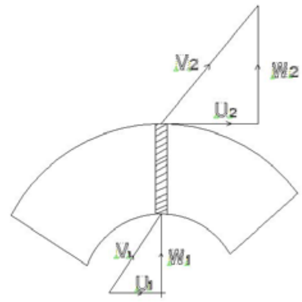



Fig. 3. Velocity diagram of radial fan

Virtual pressure head developed in the fan at the required speed.

$$Hp = 1/2g\,(U_2^2 - U_1^2 + W_1^2 - W_2^2) \quad .....(11)$$

$W_1^2 - W_2^2 / 2g$ and $U_2^2 - U_1^2 / 2g$ are the pressure head which is developed in the fan, the $V_2^2 - V_1^2 / 2g$ is the velocity head developed in the fan and converted into the pressure in the volute, the pressure head developed in the fan,

$$\varepsilon_{P2}^{0.285} - 1 = 0.285\,(H) / (RT_0)$$

The outlet specific weight

$$\gamma_2 = P_2 / RT_2 \quad ....(12)$$

The flow leaving in the fan at outlet area of the fan

$$Q_2 = (1.025 * Wg) / \gamma_2 \quad ....(13)$$

The net radial outlet area should be calculated,

$$A_2 = Q_2 / Vr_2 \quad .....(14)$$

$$B_2 = A_2 / (\pi * D_2 * \epsilon) \quad .....(15)$$

The energy Transfer (E)

$$= \text{Slip factor} * U_2 * Vu_2 \quad .....(16)$$

Total Pressure developed inside the equipment

$$(\Delta P) = (½)\,\rho\,(U_2^2 - U_1^2) + ½\,\rho\,(W_1^2 - W_2^2) + ½\,\rho\,(V_2^2 - V_1^2) \quad .....(17)$$

$$\text{Dynamic Pressure} = \rho\,(V_2^2 - V_1^2) / 2 \quad .. ...(18)$$

Total Pressure = Static Pressure + Dynamic Pressure

The Eulerian Work is equal to stage work

$$W_E = \Delta P / \rho$$

The motor Power is required to drive the fan,

$$P = m * W_E / \eta \quad .....(19)$$

$$m = \rho * Q_1$$

The total air power required for drive the fan for specified discharge of the blower

$$= E * \rho * Q \quad ......(20)$$

Based on the above Engineering expression and theoretical calculation we obtained the following results of the fan.

Table 2: Fan Theoretical calculated Results

| *Sr. No.* | *Parameter* | *Value* |
|---|---|---|
| 1 | Inlet vane diameter, mm | 80 mm |
| 2 | Outlet vane diameter, mm | 170 mm |
| 3 | Vane inlet width, mm | 33 mm |
| 4 | Vane outlet width, mm | 24 mm |
| 5 | Total pressure, mmwc | 76.33 mmwc |
| 6 | Static pressure, mmwc | 52.86 mmwc |
| 7 | Power to drive the fan, watts | 79.235 W |

### 1.2.1 Volute Design Results

Assume volute width 75mm as per the application

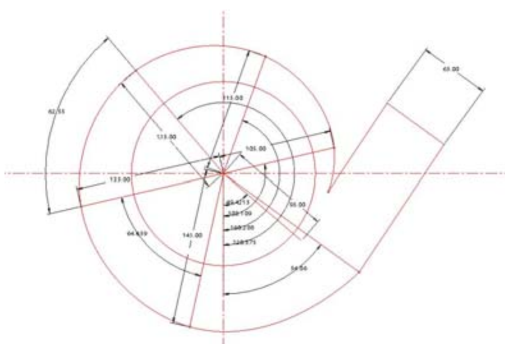


Fig. 4. Volute profile

## 1.3 Identification and selection of paramenters

The parameters that influences the pressure and power are fan outer diameter, fan outer blade width, speed of the fan, fan blade angle, and number of blades. In the mentioned parameters, speed of the fan is fixed one (3660 RPM), and fan outer blade width is dependent to fan outer diameter so, we cannot consider these parameters for optimization. Therefore the parameters selected are fan outer diameter, fan blade angle, and number of blades.

## 1.4 Implementation of Taguchi's orthogonal array method

Three parameters have been selected having four levels each for optimization. If we would go with full factorial method, ($4^3$ = 64 experiments) we supposed to do more experiments. Therefore Taguchi's orthogonal array method has been implemented and for the same number of parameters and levels, it would be reduced to 16 experiments (L16).

Table 3: Parameters with their levels

| | *Parameters* | *Level 1* | *Level 2* | *Level 3* | *Level 4* |
|---|---|---|---|---|---|
| A | Fan outer diameter | 170 | 180 | 190 | 200 |
| B | Fan blade angle | 60 | 70 | 80 | 90 |
| C | Number of blades | 6 | 8 | 10 | 12 |

Table 4 : L16 orthogonal array

| *Experiments* | *Fan outer diameter* | *Fan blade angle* | *Number of blades* |
|---|---|---|---|
| 1 | 170 | 60 | 6 |
| 2 | 170 | 70 | 8 |
| 3 | 170 | 80 | 10 |
| 4 | 170 | 90 | 12 |
| 5 | 180 | 60 | 8 |
| 6 | 180 | 70 | 6 |
| 7 | 180 | 80 | 12 |
| 8 | 180 | 90 | 10 |
| 9 | 190 | 60 | 10 |
| 10 | 190 | 70 | 12 |
| 11 | 190 | 80 | 6 |
| 12 | 190 | 90 | 8 |
| 13 | 200 | 60 | 12 |
| 14 | 200 | 70 | 10 |
| 15 | 200 | 80 | 8 |
| 16 | 200 | 90 | 6 |

### 1.4.1 Mathematical Expression and calculation

**DOE 1**

$D_2 = 170$ mm  $B_2 = 60^0$  $Z = 6$  $\varepsilon = 0.666$

From basic calculation,

$D_1 = 80$mm  $Q_1 = 0.12875\ m^3/s$  $A_1 = 5.9383 \text{ X } 10^{-3}\ m^2$  $A_2 = 9.35627 \text{ X } 10^{-3}\ m^2$

$U_1 = 15.33$ m/s  $V_1 = 21.68$ m/s  $W_1 = 15.33$ m/s

Inlet Width ($b_1$) $= A_1 / \pi D_1 \epsilon$

$= 0.0059383 / \pi * 0.080 * 0.666$

$=$ **36 mm.**

$U_2 = \pi D_2 N / 60$

$U_2 = 3.14*0.170 / 60$

$U_2 = 32.5783$ m/s.

$Vu_2 = U_2 - (Vr_2 / \tan \beta_2)$

$= 32.5783 - (15 / \tan 60^0)$

$= 23.918$ m/s

$\Delta Vu = (0.6*U_2*\pi*\sin \beta_2) / Z$

$= (0.6*32.5783*\pi*\sin 60^0) / 6$

$= 8.863$ m/s

$Vu_2' = Vu_2 - \Delta Vu$

$= 15.054$ m/s

$V_2' = \text{sq.rt}\ (Vr_2^2 + Vu_2'^2)$

$= 21.251$ m/s

$W_2 = \text{sq.rt}\ (Vr_2^2 + (U_2 - Vu_2)^2)$

$= 17.32$ m/s

ISSN 2348 – 7968

$B_2 = A_2 / (\pi * D_2 * \epsilon)$

$= 0.00935627 / (\pi * 0.170 * 0.666)$

$= 26$ mm

$(\Delta P) = (½) \rho (U_2^2 – U_1^2) + ½ \rho (W_1^2 – W_2^2) + ½ \rho (V_2^2 – V_1^2)$

$(\Delta P) = 460.589$ pa

$P_{st} = 471.989$ pa

$= 48.113$ mmwc

The Eulerian Work is equal to stage work

$Wst = \Delta P / \rho$

$= 371.443$ J /kg.

The motor Power is required to drive the fan,

$P = m * Wst / \eta$

$m = \rho * Q_1$

$= 1.24 * 0.125$

$= 0.15965$ kg / s.

Consider the efficiency as 85 %

Power = 0.15965 * 371.443 / 0.85

Power = 69.765 W

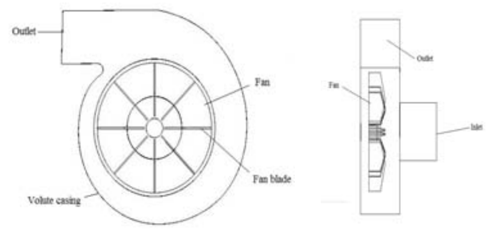
Table 5: L16 DOE calculated Results

| *DOE* | *Fan outer diameter (mm)* | *Fan blade angle* | *Number of blades* | *Inlet width $B_1$ (mm)* | *Outlet width $B_2$ (mm)* | *Theoretical pressure (mm wc)* | *Power (W)* |
|---|---|---|---|---|---|---|---|
| 1 | 170 | 60 | 6 | 36 | 26 | 48.11 | 69.7 |
| 2 | 170 | 70 | 8 | 33 | 24 | 50.97 | 89.9 |
| 3 | 170 | 80 | 10 | 31 | 23 | 52.41 | 108.4 |
| 4 | 170 | 90 | 12 | 29 | 21 | 52.85 | 126.3 |
| 5 | 180 | 60 | 8 | 33 | 23 | 56.23 | 93.72 |
| 6 | 180 | 70 | 6 | 36 | 25 | 59.09 | 98.16 |
| 7 | 180 | 80 | 12 | 29 | 21 | 60.53 | 132.9 |
| 8 | 180 | 90 | 10 | 31 | 22 | 60.97 | 141.1 |
| 9 | 190 | 60 | 10 | 31 | 20 | 64.83 | 117.9 |
| 10 | 190 | 70 | 12 | 29 | 19 | 67.68 | 138.9 |
| 11 | 190 | 80 | 6 | 36 | 24 | 69.12 | 127.2 |
| 12 | 190 | 90 | 8 | 33 | 22 | 69.56 | 153.1 |
| 13 | 200 | 60 | 12 | 29 | 18 | 73.85 | 142.8 |
| 14 | 200 | 70 | 10 | 31 | 19 | 69.13 | 143.5 |
| 15 | 200 | 80 | 8 | 33 | 21 | 70.20 | 148.7 |
| 16 | 200 | 90 | 6 | 36 | 22 | 78.62 | 158.7 |

## 1.5 Modeling and analysis of blowers

### 1.5.1 modeling

The 3D modeling of the blowers are created using SOLIDWORKS. The major parts of the blowers are fan inner diameter, fan outer diameter, blade inlet width, blade outlet width, number of blades, blade thickness, and volute casing. Geometry parameters are changed based on the Taguchi's orthogonal array method which is mentioned in Table 5.




(a) (b)

Fig. 5 (a) & (b) Front and side view of centrifugal blower

The geometry of the blade depends on fan diameters and blade angle. For each number of experiments, the blade profile varies which could be constructed using tangent circular arc method.

$$\rho = (R_b^2 - R_a^2) / 2(R_b \cos \beta_b - R_a \cos \beta_a) \quad \text{…… (21)}$$

$R_b$ = Fan outer radius, $R_a$ = Fan inner radius, $\beta_b$&$\beta_a$ = Outer and inner blade angle

For experiment 1: $R_a = 40$mm $R_b = 85$mm $\beta_a = 90^0$ $\beta_b = 60^0$

$$\rho = (R_b^2 - R_a^2) / 2(R_b \cos \beta_b - R_a \cos \beta_a) \quad \rho = 66.18$$

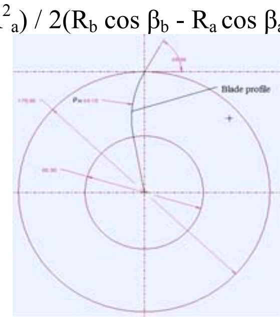



Fig . 6 Blade profile

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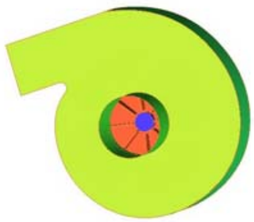

Fig. 7 3D model of blower

During modeling the volute casing is kept constant and the fan alone has been changed. Fig. 8(a) & Fig. 8(b) shows some of the fans model.

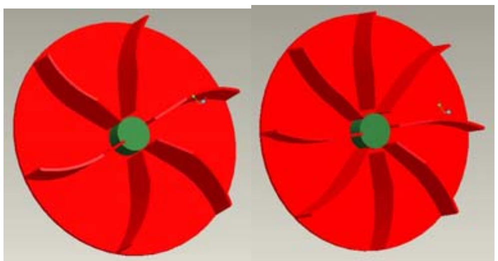

Fig. 8(a). 170mm diameter fan with 6 & 8 number of blades

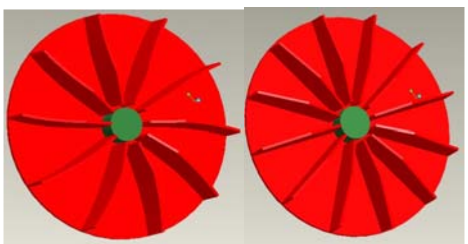

Fig. 8(b). 170mm diameter fan with 10 & 12 number of blades

## 1.5.2 Meshing

After modelling the blowers, the meshing was carried out in ICEM CFD tool. The unwanted curves and surface which could affect the quality of meshing has been removed and some curves and surfaces have been added. Three fluid domains was created first one is at inlet, second is around fan, and third is at volute casing. The blowers are meshed using tetrahedral elements since they maintain good quality for complex shapes. After grid generation, the mesh independency tests were carried out and the results were shown in Table-6.

Table 6: Mesh independency test

| *Mesh* | *No. of elements* | *Static pressure(mmwc)* |
|---|---|---|
| A | 1078803 | 47.523 |
| B | 1813095 | 47.603 |
| C | 2362398 | 47.601 |

From the above table, it is clear that mesh B and mesh C are converged. Hence mesh B is chosen to reduce the computational time.

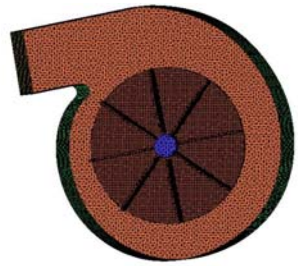

Fig. 9. 3D meshed model of blower

## 1.5.3 CFD Analysis

Computational Fluid Dynamics (CFD) approach is the effective method of solving non-linear partial differential equations that governs fluid flow, heat transfer and turbulence of flow. In this present study, CFD FLUENT is used to solve the flow inside the centrifugal blowers. Moving Reference Frame (MRF) method is adaptable for the condition where fluid rotates. A separate cell zone has to be mentioned for MRF method, the second cell zone condition in our study was considered as MRF zone. The assumptions made for this study are,

i. Steady state air flow
ii. Implicit solver
iii. Standard wall function
iv. Second order upwind scheme

Table 7: Boundary conditions

| *Boundary Condition* | *Value* |
|---|---|
| Material | Air |
| Density | 1.2 kg/m3 |
| Viscosity | 1.789e-05 kg/m-s |
| Turbulence model | K-ω SST model |
| MRF | Impeller |
| Inlet condition | Pressure inlet |
| Outlet condition | Pressure outlet |
| Hydraulic diameter at inlet | 0.080 m |
| Hydraulic diameter at outlet | 0.075 m |
| Impeller speed | 383.274 rad/s |

## 1.6 CFD Results

### 1.6.1 Post processing results

The different plots for various cases are depicted below.

#### 1.6.1.1 Trail DOE 1

For the original fan, Fig. 10 shows the velocity contours, pressure contour plotted on mid-plane respectively.
Once the solution is converged the results are post processed using CFD POST. Various values like static

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pressure contour, velocity contour and velocity vector has been mentioned in the below figures,

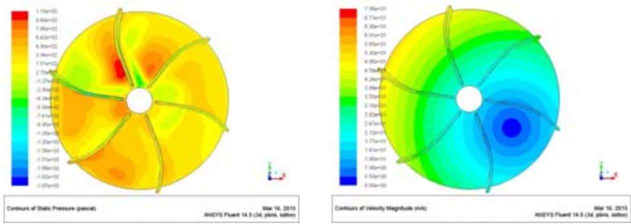

Fig. 10. Static pressure and velocity contours of DOE 1 blower

### 1.6.1.2 Trail DOE 5

The velocity and pressure contours for trail 3 are shown in Fig. 11.

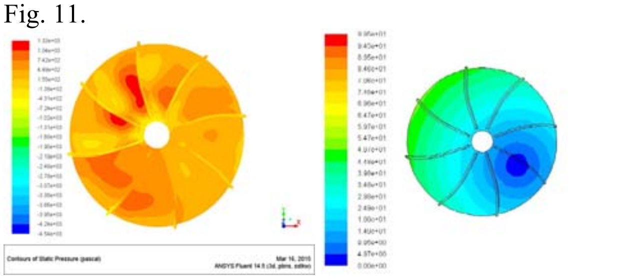

Fig. 11. Static pressure and velocity contours of DOE 5 blower

### 1.6.1.1 Trail DOE 9

The velocity and pressure contours for trail 3 are shown in Fig. 12.

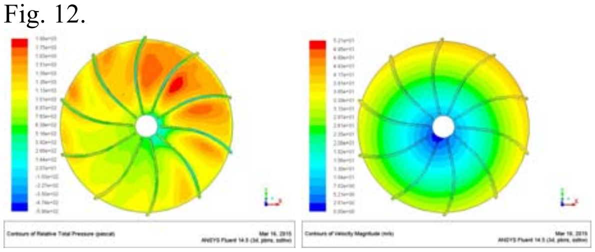

Fig. 12. Static pressure and velocity contours of DOE 9 blower

### 1.6.1.1 Trail DOE 13

The velocity and pressure contours for trail 3 are shown in Fig. 13.

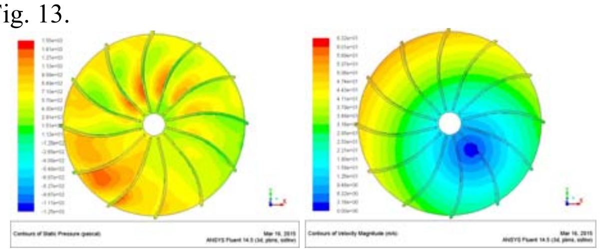

Fig. 13. Static pressure and velocity contours of DOE 13 blower

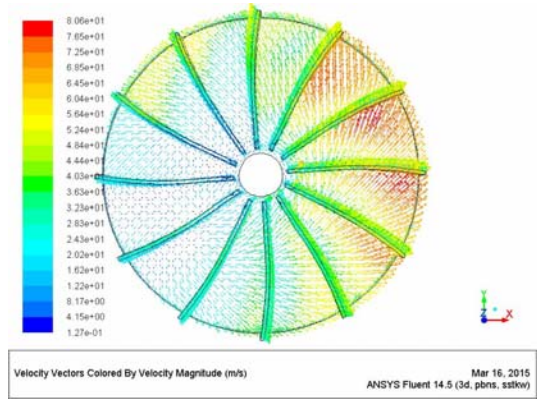


Fig. 13. Velocity vector magnitude of fan

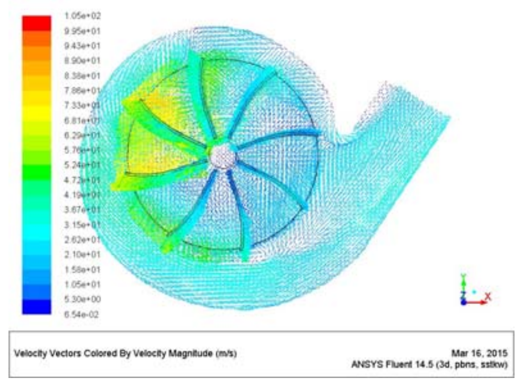


Fig.14. Velocity vector magnitude of blower

## 2. Results and Discussion

The static pressure of the blowers at inlet condition during CFD analysis has been noted down and the value is compared with the theoretical value in the below table.

Table 8: Comparison of theoretical and CFD results

| *Experiments number* | *Theoretical Pressure value (mmwc)* | *CFD Pressure value (mmWc)* | *% deviation* |
|---|---|---|---|
| 1 | 48.11 | 47.60 | 10.60% |
| 2 | 50.97 | 50.57 | 0.78% |
| 3 | 52.41 | 57.77 | 10.24% |
| 4 | 52.85 | 59.38 | 12.35% |
| 5 | 56.23 | 55.88 | 0.62% |
| 6 | 59.09 | 57.73 | 2.30% |
| 7 | 60.53 | 65.52 | 8.24% |
| 8 | 60.97 | 66.90 | 9.72% |
| 9 | 64.83 | 67.09 | 3.48% |
| 10 | 67.68 | 68.05 | 0.54% |
| 11 | 69.12 | 68.62 | 0.72% |

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| | | | |
|---|---|---|---|
| 12 | 69.56 | 69.29 | 0.38% |
| 13 | 73.85 | 75.06 | 1.63% |
| 14 | 69.13 | 72.85 | 5.38% |
| 15 | 70.20 | 68.35 | 2.63% |
| 16 | 78.62 | 75.85 | 3.52% |

## 3. Result table of Taguchi method

Minitab is statistical software which is used to find out the optimum combination from the Taguchi's input. In this study, we used minitab to find out the responses of fan outer diameter, fan blade angle and number of parameters on static pressure and to get the optimum combination of these parameters. The results of mean effects plot for means and for SN ratio is mentioned in the below figures.

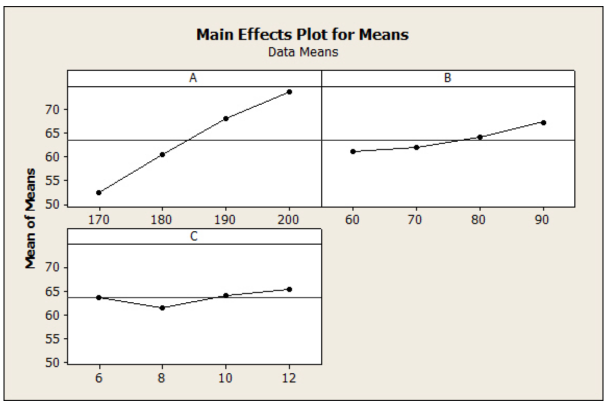



Fig. 15. Responses of parameters by mean effects plot for means

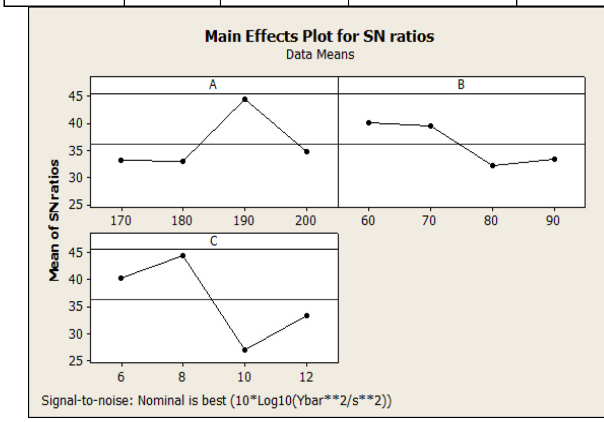
Table 9: Maximum affecting parameters

| *Levels* | *Fan outer diameter* | *Fan blade angle* | *Number of blades* |
|---|---|---|---|
| 1 | 59.389 | 57.942 | 63.224 |
| 2 | 60.277 | 63.311 | 59.097 |
| 3 | 62.389 | 65.783 | 64.449 |
| 4 | 72.974 | 68.204 | 67.305 |
| Δ | 13.585 | 10.261 | 8.208 |
| Rank | 1 | 2 | 3 |

The impeller outer diameter influences the most on static pressure while the number of blades influences the least. Since our aim is get the nominal static pressure and power value, nominal is better has chosen. The optimum combination has been mentioned in Table-10.

Table-10. Optimum combination

| *Fan outer diameter (mm)* | *Fan blade angle* | *Number of blades* | *Theoretical Pressure (mmwc)* | *Power (W)* |
|---|---|---|---|---|
| 190 | 60 | 8 | 64.83 | 112.09 |




## 4. Conclusion

This paper studies in detail about the optimization of centrifugal blower using computational fluid dynamics approach. The major parameters like the fan outer diameter, fan blade angle, and number of blades are considered. The effect of each parameter has been observed on the performance of the fan. Taguchi's orthogonal array method has helped to reduce the number of trials and save computational time with giving the optimum result. The results of this study will surely help to improve the performance of the fan.

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