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Optimizing the Design Parameters of Radial Tip Centrifugal Blower for Dust Test Chamber Application Through Numerical and Statistical Analysis

The tightness of electro-technical components for dust entrapment is an extremely important parameter to be considered for effective functioning. International Electro-technical Commission (IEC) developed standards for testing the parts under severe environments like rain, dust etc. The procedure for developing and conducting the dust proof test is known by IEC as 60068-2-68 standard. The blower is an essential component used in the dust test system in order to create a forced airflow along with dust to simulate a dusty environment. The radial tip centrifugal blower is designed in this work for the application of dust test chamber. The pressure and velocity of airflow at the outlet of the radial tip centrifugal blower is mostly influenced by inlet blade angle (β_1), impeller diameter (D_2) and impeller width (b). The design procedure is elaborated and the above said design parameters are optimized using iterative procedure using MATLAB software. A numerical analysis is carried out using ANSYS Fluent software for the design of Taguchi L9 orthogonal array. In the perspective of further exploring the relationship between design parameters of blower and output parameters like pressure and velocity, a statistical analysis is carried out using Design-Expert software. The optimized design parameters are then used for manufacturing the blower. The experimental test is carried out on the designed blower. The experimental, numerical and statistical results for the optimized design parameters are compared and known to be in good consensus.

Keywords: Dust test, Design of radial tip centrifugal blower, Optimization, Numerical Analysis, Statistical Analysis, Experimental Analysis

1. INTRODUCTION

The entrapment of dust particles into the enclosures of electrical and electronic components leads to malfunction due to the change of contact resistance and faulty contact. It also affects the functioning of mechanical components such as bearing, axle and shaft by impeding the disturbance in motion. The clogging of these dust particles contaminates the lubricant and optical surfaces in the components. The flow of these particles with air along the surface erode the surfaces which in turn promote the surface failure and successive failure of the structure. It becomes inevitable to test the electrical and electronic components for their tightness to the entrapment of the dust particles. In order to ensure the constructional integrity of the component, it is required to simulate the actual dusty environmental condition and observe the functionality of the components after the test. The dust test chamber, which simulates the actual airflow with dust in various service conditions, is designed by following the international standards published by IEC (60068-2-68 standard). The airflow is artificially induced in this system with the help of a blower. The blower used in this system is intended to push the dust particles suspended in the air. As it plays a major role, it is inevitable to pay special attention to the design and manufacturing of blower used in this system.

The blower is designed using different theories proposed by the designers. However, the design parameters of the blower and its components are commonly calculated based on empirical and semi-empirical relations with more assumptions. This was due to the complex nature of three-dimensional flow in the passages. The reduced life-cycle cost and reliability of centrifugal compressors make them preferable than reciprocating compressors. The backward curved impeller is being widely used in most of the applications due to its efficiency comparing to radial and forward curved impeller. The velocity of the fluid at the tip of the vane depends on either the size of the impeller or the speed of the impeller shaft rotates. The pump and blower design methodologies were initially explained by Church [1]. The methodology was further refined and analyzed by many researchers (i.e. Eck [2], Osborne [3], Yan [4] etc). Beena Devendra Baloni et.al [5] have worked on a unified design procedure in order to achieve improved performance parameters for the same pressure head of industrial blower. The flow rate of the boiler designed through their unified theory was increased by 35% and showed minimum losses. Jeyapragasam CN et al [6]

optimized the performance of centrifugal blower against impeller outer diameter, speed of the impeller and fillet radius at entry. Jeyapragasam CN et al also expressed that the impeller outer diameter affects the efficiency of the centrifugal blower than the speed of the impeller and fillet radius. Chen-Kang Huang et al [7] investigated the effect of blade angle and number of blades on the efficiency of backward curved airfoil centrifugal blowers. They have concluded that 49.39º was identified as optimum blade angle and 10 number of blades were noted to be an ideal number of blades. Zhang et al [8] have conducted research on improving the performance of low-speed centrifugal blower by optimizing the blade profile. The multi-point Bezier curve approach was found to be effective in determining the efficient blade profile. The variable thickness blade was further suggested in order to decrease the flow loss and to increase the pressure rise of the blower. Yu-Tai Lee et al [9] evaluated the effect of the width of the impeller on increasing the pressure required for optimum input fan power. The static pressure was linearly improved with increasing width of impeller whereas the input power almost remained constant. The numerical analysis with the help of a software based on the finite volume method is being successfully used by several researchers in assessing the performance and flow characteristics of turbomachines. The static pressure distribution and relative velocity at the interior passage of rotating parts of turbomachines can be easily evaluated using numerical methods [10]. In the application of dust test chamber, the air flow with required velocity needs to be generated along with dust. Wayne T Sproull [11] explained that the viscosity of the air decreased when the dust concentration in air was increased. The predominant design parameters like blade angle (β), impeller diameter (D_2) and impeller width (b) would affect the flow characteristics and efficiency of the blower.

The rotation speed plays a vital role in the outlet properties of fluid from the blower. In this research work, our objective is to optimize the dimensions of the blower. Hence, the design parameters of the blower, that mostly influence the outlet property, are considered by keeping the rotation speed constant. This article discusses the salient features of the design parameters of blower used in the dust test chamber. The main objective of this study was to conduct a numerical analysis based on Taguchi L9 orthogonal array using inlet blade angle ($β_1$), impeller diameter (D_2) and impeller width (b) as affecting factors and to identify the significant design parameter which affected the flow parameters and performance of the blower.

2. DESIGN PROCEDURE

The design procedure mainly depends on the fundamentals of fluid flow that deals with the continuity equation and the energy balance equation. The input parameters required for the design of a centrifugal fan for the purpose of dust chamber given in table 1. These values are identified based on the conditions given in the IEC 60068-2-68 standard and a practical requirement for the dust chamber.

Table 1. Design Parameters

Parameter	Value
Flow Discharge, Q	0.277 m³/s
Static suction pressure	-196.4 Pa
Static delivery pressure	2000 Pa
Static pressure gradient, $ΔP_s$	2196.4 Pa
Speed of the impeller, N	2880 rpm
Air density, $ρ_{air}$	1.165 kg/m³
Suction Temperature, T_s	30 °C = 303 K
Atmospheric Pressure, P_{atm}	1.01325 x 10⁵ Pa
Atmospheric Temperature, T_{atm}	30° C = 303° K
Inlet duct diameter, D_{duct}	0.150 m

The impeller eye diameter is calculated using empirical relation in equation (1) and it is usually a function of duct inlet diameter. The duct size in this work is assumed based on the practical requirement. It is considered that there is no loss occurred while the fluid turns 90º from the suction direction into the impeller inlet ($V_1 = V_{mi} = V_{eye}$). The inlet velocity of the fluid is computed using equation (2) from the known values of discharge and eye diameter. The empirical relation that connects inlet fluid velocity and impeller inlet tip velocity are given in equation (3). The equation (3) is framed based on the consideration 10% deviation between both components of velocities.
impeller eye diameter,

$$D_{duct} = 1.1 D_{eye} = 1.1 D_1 \tag{1}$$

Discharge,

$$Q = \frac{\pi}{4} D_{eye}^2 V_1 \tag{2}$$

The inlet tangential velocity (U_1) must be greater than axial velocity (V_1) to enhance the flow. The deviation of velocity was assumed based on the practical experience. The dimensions obtained using this assumption offered better results in this research work. The impeller inlet tip velocity,

$$U_1 = 1.1 V_1 = \frac{\pi D_1 N}{60} \tag{3}$$

The resultant peripheral velocity at the inlet of impeller is obtained using the relation shown in equation (5). Beena Devendra Baloni et.al [5] suggested that the blade inlet angle should not exceed 35.26° for the effective operation of the impeller. They have also mentioned that the selection of the number of blades should also be taken care as too many numbers of blades would restrict the motion of fluid and very few of them would not guide fluid properly. The number of blades for a specific blower either can be calculated through empirical relation shown in equation (4) or by practical experience.

$$Z = 8.5 \times \frac{\sin β_2}{1 - \dfrac{d_1}{d_2}} \tag{4}$$

The equation (4) was proposed by Bruno Eck [2]. Bruno Eck also expressed in a patent (US3306528A) that there is strong relationship between diameter ratio, and number of blades. The resultant peripheral velicity

at the inlet can be calculated using equation (5). The inlet blade angle (β_1) and impeller width (b) play a vital role in deciding the performance parameters of the centrifugal fan as stated earlier. It is calculated using equation (6) & (7). The width of the impeller blade is assumed to be constant at inlet and outlet of the impeller. The outer diameter of the impeller is calculated using equation (8).

The peripheral resultant velocity at the inlet,

$$W_1 = \sqrt{U_1^2 + V_1^2} \tag{5}$$

impeller inlet blade angle,

$$\tan \beta_1 = \frac{V_1}{U_1} \tag{6}$$

impeller width at the inlet,

$$b_1 = \frac{Q}{(\pi D_1 - zt)V_1} \tag{7}$$

impeller outer diameter,

$$D_2 = \frac{60U_2}{\pi N} \tag{8}$$

The volute dimensions of the blower are designed using a spiral curve equation shown in equation (9). The equation (9) represents logarithmic spiral curve. It is a known fact that the volute design follows logarithmic spiral curve. The equation (9) fit the volute curve within volute exit radius and volute tongue radius. The radius of the volute tongue is mentioned in equation (10). The exit radius of the volute is calculated with the help of equation (11) by the known values of width of the volute and volute outlet area. The width of the volute can be calculated by adding the width of impeller and reasonably assumed values of front and rear clearance. The volute outlet area is calculated with the known duct dimension at the outlet as there is no change in the cross section between volute outlet and duct. The blade profile is modelled using well-known tangent arc method and the radius of the impeller blade is calculated using equation (12)

Volute radius at angle 'θ',

$$r_\theta = r_2 + \frac{\theta}{360} + \Delta r \tag{9}$$

Radius of the volute tongue,

$$r_3 = 1.075 r_2 \tag{10}$$

Exit outlet area

$$A_v = b_v (r_4 - r_3) \tag{11}$$

Radius of the impeller blade,

$$R_b = \frac{r_2^2 - r_1^2}{2(r_1 \cos \beta_1 - r_2 \cos \beta_2)} \tag{12}$$

The initial dimensions of the centrifugal fan are calculated based on the formulas discussed. In order to optimize the design parameters, further iterations are made by considering the leakage loss and pressure loss. This process is carried out until no major difference is observed in impeller diameter (D_2) and all other parameters. The Math lab program is generated for the above calculation as well as the iteration process and the respective algorithm for the iterative process is shown in Fig. 1. The formulas to compute the leakage and various pressure drops at different locations are shown in equations (13-18).

Leakage Loss

$$Q_L = C_d \pi D_1 \delta \sqrt{\frac{2P_S}{\rho}} \tag{13}$$

C_d – Coefficient of discharge - 0.6 to 0.7
D_1 – eye diameter
δ - Clearance between eye inlet and casing – 2 mm
Ps - Static suction pressure, =2/3(ΔP)s
ρ - Density of air

Suction Loss

$$dp_S = \frac{1}{2} k_i \rho V_{eye}^2 \tag{14}$$

k_i – loss factor, 0.5- 0.6
ρ-density of air
V_{eye}-inlet velocity of fluid

Impeller Loss

$$dp_{imp} = \frac{1}{2} k_{ii} \rho (W_1 - W_2')^2 \tag{15}$$

k_{ii}– a factor lies between 0.2-0.5 for sheet metal blades for maximum efficiency
W_1 – inlet resultant velocity
W_2' – outlet resultant velocity

$$W_2' = \sqrt{W_{U2}^2 - W_{m2}^2} \tag{16}$$

Volute Loss

$$dp_v = \frac{1}{2} k_{iii} \rho (V_2' - V_4^2) \tag{17}$$

k_{iii}-volute loss factor,0.4
V_2' - resultant velocity of fluid at outlet
V_4 - velocity of fluid at the outlet of volute

$$V_2' = \sqrt{V_{U2}^2 - V_{m2}^2} \tag{18}$$

The exit velocity V_3 is calculated using Bernoulli's equation as shown in equation (19-20). The specific work done is computed from equation (21). The velocity component at the outlet of the impeller is also calculated using power and mass flow rate relation as shown in equation (21-22). The fluid velocity is assumed to be 80% of impeller outlet velocity

$$\frac{p_1}{\rho_2} + \frac{1}{2} V_1^2 + gz_1 + W_3 = \frac{p_2}{\rho_1} + \frac{1}{2} V_4^2 + gz_2 \tag{19}$$

Exit velocity,

$$V_4^2 = \frac{-2[p_2 - p_1]}{\rho_f} + V_1^2 + 2W_s \tag{20}$$

$$W_s = \frac{Power}{mass\,flow\,rate} \qquad (21)$$

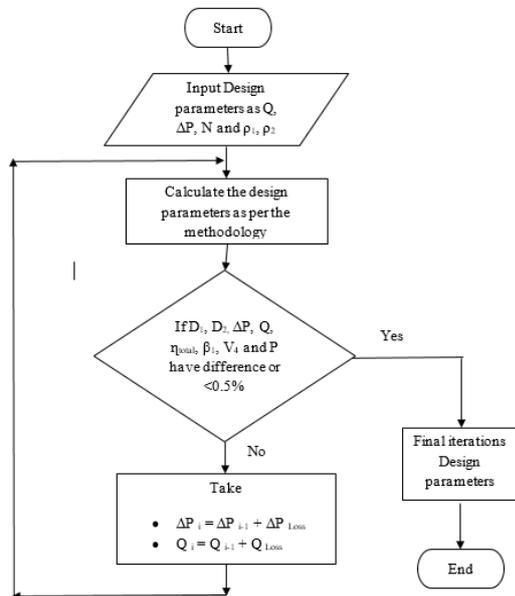$$Power = mass\,flow\,rate \times V_{U2}^2 \times U_2 \qquad (22)$$

Figure 1. Flow chart for design of the impeller

The optimized design parameters were obtained through the iteration process. The solid model of the centrifugal blower was designed using Solid works 2018 software. The solid model for a radial tipped centrifugal blower and its dimensions were shown in Fig. 2, 3, and 4.

3. DESIGN OF THE NUMERICAL EXPERIMENT

It is decided to optimize the design parameter that influences the output performance of radial tip centrifugal blower. Three important design parameters that affect the performance of the centrifugal blower were considered for optimization. The factors which influence the flow, as well as the efficiency of the radial tip centrifugal blower, are impeller inlet blade angle (β_1), impeller outer diameter (D_2), and impeller width (b).

Taguchi method is the most appropriate procedure for solving this kind of problems. This method uses special set of arrays called orthogonal arrays. These standard arrays stipulate the way of conducting the minimal number of experiments which give the full information of all the factors that affect the performance parameter. The crux of the orthogonal arrays method lies in choosing the level

combinations of the input design variables for each experiment. While there are many standard orthogonal arrays available, each of the arrays is meant for a specific number of independent design variables and levels. It is decided to use design of experiment based on Taguchi L9 orthogonal array to optimize the above said design factors.

The L9 orthogonal array is meant for understanding the effect of 3 independent factors each having 3 level values. The pressure and velocity at the outlet of the blower are considered as target or response of the analysis. The factors with three levels (Low, Medium and High) of their values are taken for the design and analysis numerically. The factors and their levels of values are tabulated in Table.2. The design of experiment is shown in table 3. Based on those nine combinations of design parameters, the respective solid models are designed using Solid works software. The numerical analysis is carried out for solid models using ANSYS v.19.2 fluent software. The results of pressure and mean velocity at the outlet of the centrifugal blower is displayed in table 4.

Table 2. Factors and Levels

		Levels		
	Parameters	Low (L)	Medium (M)	High (H)
Factors	Inlet Blade angle (β_1) (degree)	36	42	48
	Impeller diameter (D_2) (mm)	250	300	350
	Impeller width (b) (mm)	45	50	55

4. NUMERICAL ANALYSIS

The mesh of the geometry in the numerical analysis based on finite element technique need to be optimized as it directly affects the output of the results. Mesh optimization is the process which is based on the discretization of the model.

The mesh optimization of each solid model is carried out with the iterative process. The quality of the mesh is also based on some factors that are skewness, orthogonal quality, nodes, and elements. For good quality of the mesh, the skewness should be less than 0.25 and Orthogonal quality and element quality should be greater than 0.85. The boundary condition of the analysis is shown in table 5.

Table.3. Design of Experiment – Taguchi L9 -Numerical Results

EXP. NO.	Inlet Blade angle (β_1) (degree)	Impeller diameter (D_2) (mm)	Impeller width (b) (mm)	Pressure (Pa)	Mean Velocity (m/s)
1	L (36)	L (250)	L (45)	1819.4	32.2
2	L (36)	M (300)	M (50)	1769.7	38.9
3	L (36)	H (350)	H (55)	1999.0	14.0
4	M (42)	L (250)	M (50)	1962.3	23.3
5	M (42)	M (300)	H (55)	1913.7	28.0
6	M (42)	H (350)	L (45)	1969.3	17.6
7	H (48)	L (250)	H (55)	1208.3	71.3
8	H (48)	M (300)	L (45)	1686.2	40.6
9	H (48)	H (350)	M (50)	1945.8	20.7

Table.4. Comparison of results

| S. No.	Inlet Blade angle (β_1) (deg)	Imp. Dia (D_1) (mm)	Imp. Width (b) (mm)	Numerical Results		Statistical Results		Error (%)	
				Pressure (Pa)	Mean Velocity (m/s)	Pressure (Pa)	Mean Velocity (m/s)	Pressure (Pa)	Mean Velocity (m/s)
1	37	270	47	1999.3	12.1	2002.6	10.7	0.17	-13.55
2	39	290	49	1991.8	13.6	1988.6	15.0	-0.16	9.08
3	40	210	51	1999.2	12.3	1999.1	12.3	0.00	0.31
4	43	225	52	1977.0	18.3	1973.3	19.9	-0.19	7.95
5	45	240	54	1956.6	25.6	1960.1	24.0	0.18	-6.44
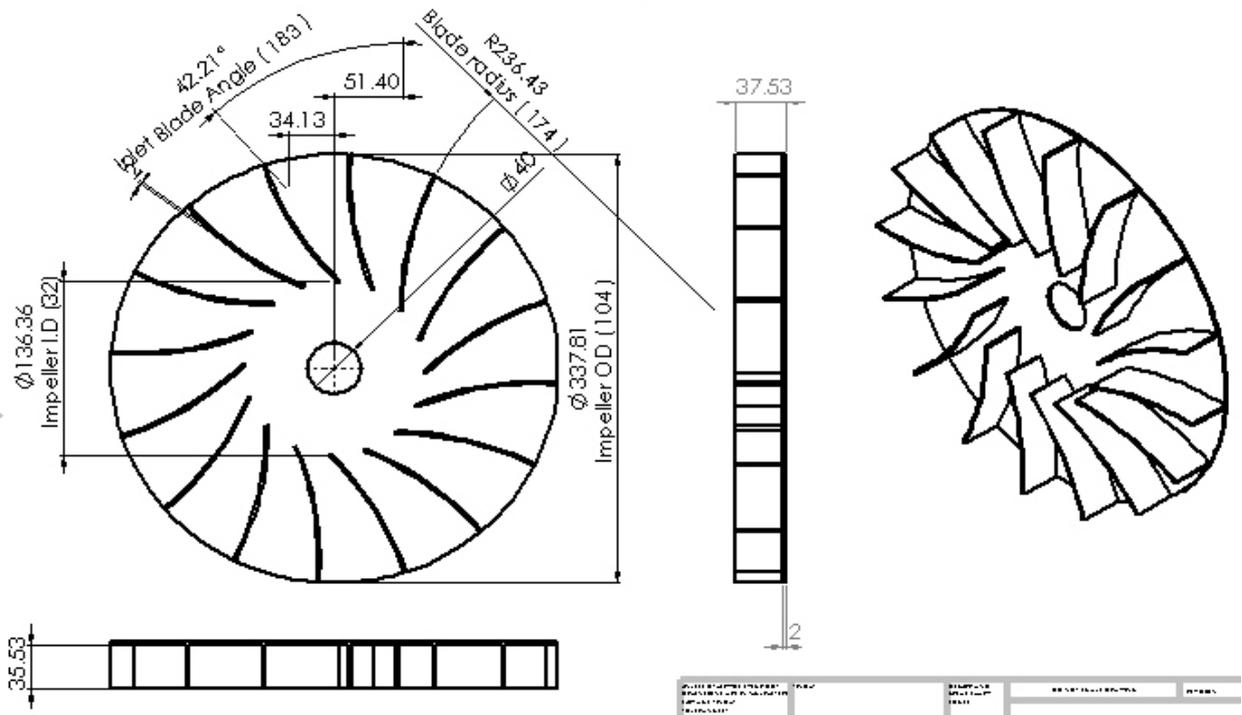
Figure 2. Draft model of an impeller
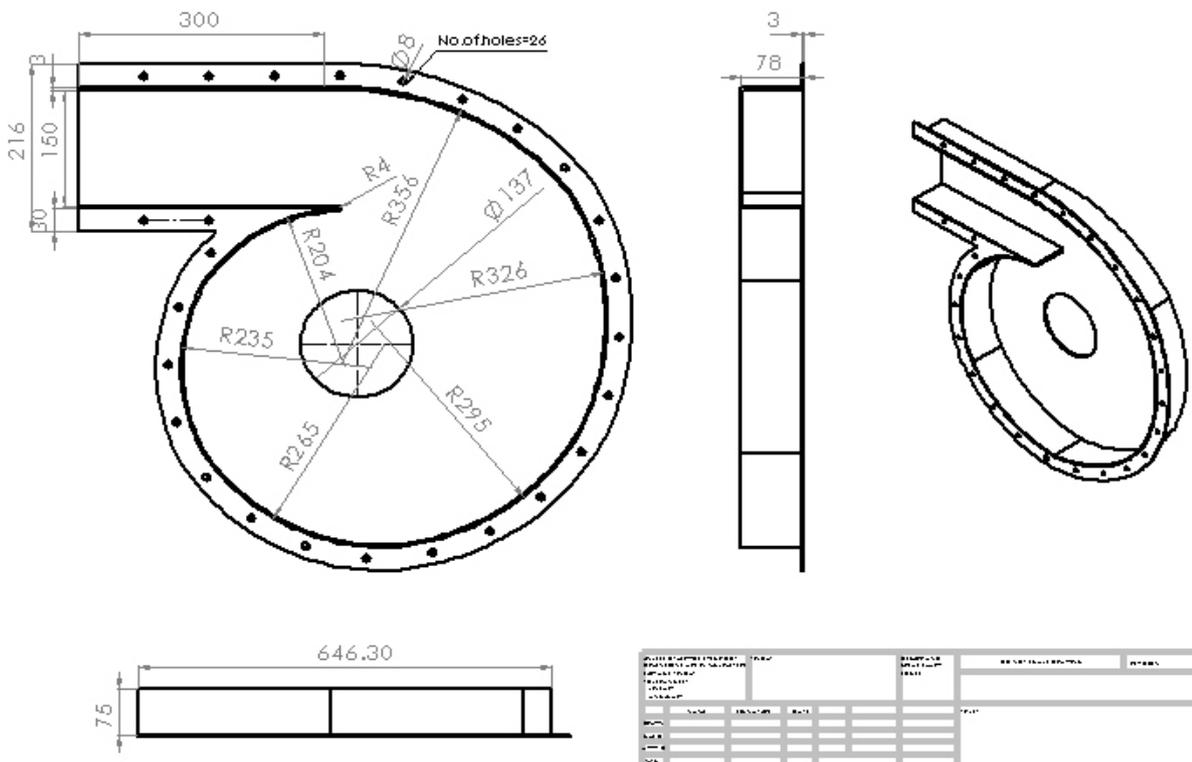
Fig. 3. Draft model of the casing

Table.6. Detail of elements and nodes in optimized solid models

Model No.	1	2	3	4	5	6	7	8	9
Nodes	62278	14896	4488	61406	15112	12236	60255	16171	18434
Elements	338547	73262	18815	333507	75053	55972	327003	81183	90428

Table.5. Boundary condition of the analysis

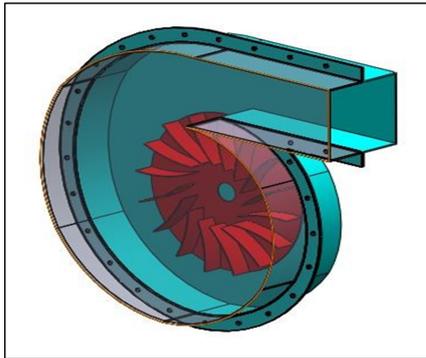| Position	Parameters	Values
Inlet	Pressure (P_1)	-196.4 Pa
	Velocity at the inlet (V_1)	From calculations
Outlet	Pressure (P_2) (Pa)	2000
Solution model: k-epsilon		
Material: Air and Talcum powder		
Impeller condition: Rotating about Z axis at 2880 rpm		

Figure 4. Assembly view of centrifugal blower

The number of cells of converged solid model are shown in table.6. The average values of pressure and velocity at the outlet are considered as there is a possibility for minor changes in the parameters at the outlet section of the centrifugal blower.
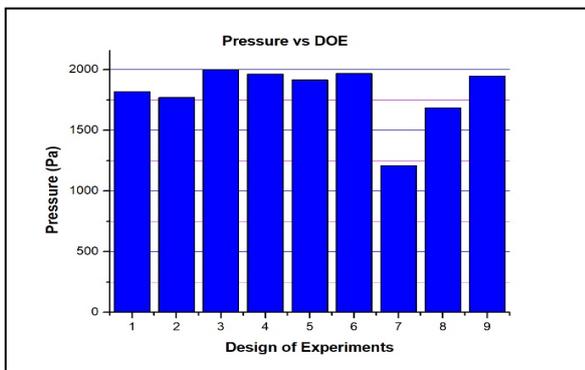

Figure 5. Pressure with experiments

The average pressure and mean velocity values at the outlet for each experiment are shown in figs 5 and 6.
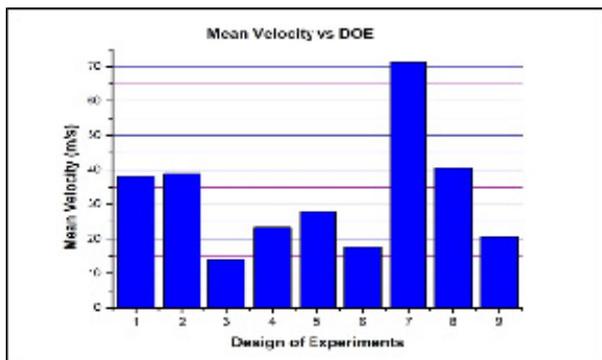

Figure 6. Mean velocity with experiments

It is inferred from fig.5 and 6 that experiment 7 showed the maximum value of mean velocity and low value of static pressure. In experiment 7, the blade angle is kept as 48°, impeller outer diameter is maintained as 250 mm and the width of the blade is modelled as 55 mm. It is understood that the blade angle and width of the blade play a vital role in the design of radial tip centrifugal blower in increasing the velocity of the fluid at the outlet. The numerical analysis results of each experiment are needed further exploration in the statistical point of view in order to interpret the results for the combination of design parameters.

The vector plots are generated and the particles are enlarged. The solutions of such vector plot for all solid models based on the design of the experiment are shown in Fig. 7 (a-i).
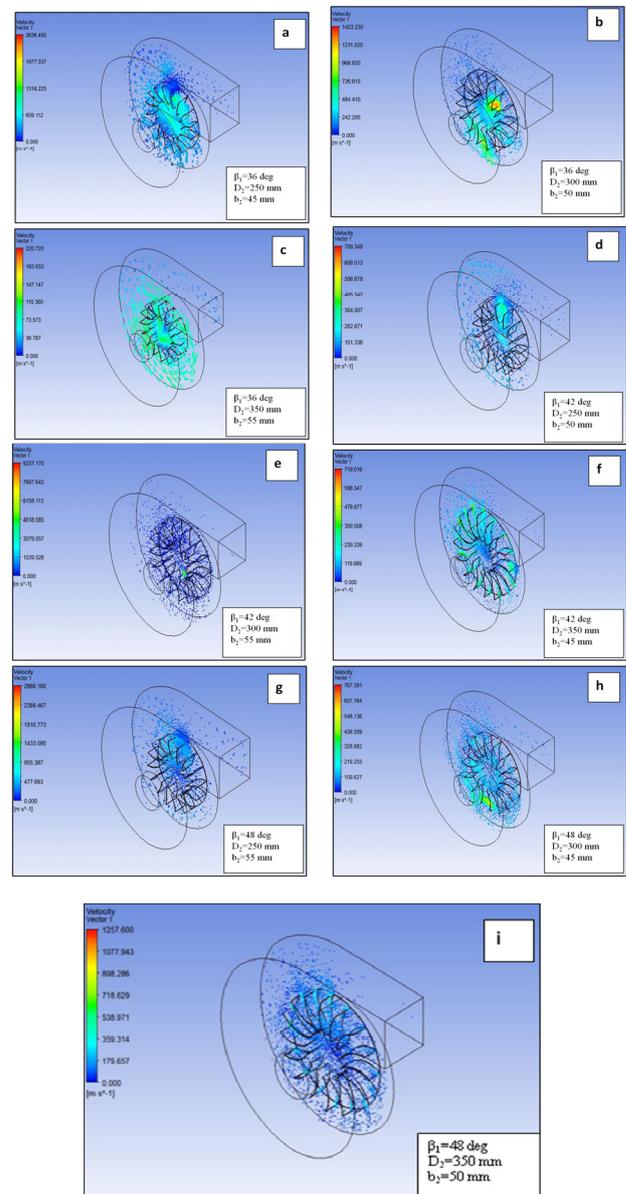

Fig. 7. CFD vector plot of experiments

5. STATISTICAL ANALYSIS

Regression analysis is carried out to examine the relationship between design parameters and responses. The regression analysis is done by using Design expert software. In this regression analysis, the factor values and corresponding responses are given as input. The responses are obtained from numerical analysis.

By selecting the combinations of the factors, the regression equations are formed. The regression equation for pressure and mean velocity at the outlet are mentioned in eqn. (23) and (24).

$$P = -659.57167 + 36.59750\beta_1 - 29.18427D_1 + \\ +241.91237b_2 + 0.72878(\beta_1 \cdot D_2) - 5.52011(\beta_1 \cdot b_2) \quad (23)$$

$$V = 312.53663 - 4.81146\beta_1 + 1.42602D_2 - \\ -15.12488b_2 - 0.037866(\beta_1 \cdot D_2) + 0.34650(\beta_1 \cdot b_2) \quad (24)$$

The respective R-squared value for eqn. (23) and (24) were 0.7554 and 0.7220. In order to check the fitness of the equation, another numerical analysis is planned based on arbitrary values of inlet blade angle (β_1), impller diameter and impeller width (b). The solid models are created for the arbitrary values of design parameters and numerical analysis is carried out. The numerical results are compared with statistical results obtained from equation (23) and (24) for the same input parameters and shown in table.4.

5.1 Grey Relational Analysis

Grey Relation Analysis, in other words, Grey Incidence Analysis, is used to find out the optimum combination of design parameters which influence more than one output response simultaneously. In this research work, the responses are pressure and mean velocity of fluid at the outlet of blower. Hence this method is applied to optimize the input design parameters, such as blade angle (β_1), impeller diameter (D_2) and impeller width (b), for multiple responses (i.e. pressure and mean velocity). In Grey relational analysis, numerical results obtained are first normalized in the range zero to one. This process is known as Grey relational generation. Based on the normalized experimental data, Grey relational coefficient is calculated to represent the correlation between the desired and actual numerical data. The overall Grey relational grade is determined by averaging the Grey relational coefficient corresponding to selected responses. The overall performance characteristic of the multiple response processes depends on the calculated Grey relational grade. This approach converts multiple response optimization problems into a single response optimization situation with the objective function of overall Grey relational grade. The optimal parametric combination is then evaluated which would result in the highest Grey relational grade. The optimal factor setting for maximizing overall Grey relational grade can be performed by the results of the Taguchi method. In Grey relational generation, impeller width (b) and impeller outer diameter (D_2) were corresponding to the smaller-the-better (SB) criterion. impeller inlet blade angle (β_1) should follow the larger-the-better (LB)

criterion. The criteria can be expressed as shown in equation (25).

$$x_i(k) = \frac{\max y_i(k) - y_1(k)}{\max y_i(k) - \min y_i(k)} \quad (25)$$

The equation (25) is used to calculate the normalized value of responses out of experiments. $y_i(k)$ is the outcome of each experiment.

The Grey relational coefficient can be computed using equation (26).

$$\xi_i(k) = \frac{\Delta_{min} - \Psi\Delta_{max}}{\Delta_{oi}(k) + \Psi\Delta_{max}} \quad (26)$$

The equation (26) is used to calculate the grey relational coefficients of normalized responses. Ψ is the differencing coefficient, $0<\Psi<1$. It is included to weaken the influence of Δ_{max}. Δ_{oi} = difference of the absolute value of $x_0(k)$ and $x_i(k)$.
Where, $x_i(k)$ is the value after the Grey relational generation, ψ is the distinguishing coefficient (this case ψ= 0.5), Δ_{oi} = difference of the absolute value of $x_0(k)$ and $x_i(k)$, After averaging the Grey relational coefficients, the Grey relational grade g_i can be computed as:

$$g_i = \frac{1}{n}\sum_{k=1}^{n}\xi_i(k) \quad (27)$$

The equation (27) is used to calculate grey relational grade. This would be calculated for every experiment. 'n' is number of output responses. The average of grey relational coefficient obtained for pressure and mean velocity will be considered as grey relational grade. The maximum value of grade would represent optimum combination of input parameters.
Where 'n' is the number of process responses. The higher value of Grey relational grade corresponds to an intense relational degree between the reference sequence $x_0(k)$ and the given sequence $x_i(k)$. The reference sequence $x_0(k)$ represents the best process sequence. Therefore, higher Grey relational grade means that the corresponding parameter combination is closer to the optimal. The Grey relational grade for the corresponding design was tabulated in Table 7.

Table.7. Grey relational grade of various designs

Exp. No.	Inlet Blade Angle (β_1) (degree)	Impeller Diameter (D_2) (mm)	Impeller Width (b) (mm)	GRA grade	Rank
1	36	250	45	0.778	3
2	36	300	50	0.444	7
3	36	350	55	0.333	9
4	42	250	50	0.667	4
5	42	300	55	0.444	7
6	42	350	45	0.611	5
7	48	250	55	0.778	2
8	48	300	45	0.833	1
9	48	350	50	0.611	5

The combination of design parameters in experiment 8 denotes the optimized model comparing to others. The

optimized model values from the table are, Inlet blade angle (β_1) = 48°, impeller diameter (D_2) = 300 mm and the impeller width (b) = 45 mm.

6. MANUFACTURING

The optimum combination of design parameters are then considered for manufacturing the radial tip centrifugal blower. In case of centrifugal blower, the mild steel plates are used to manufacture the impeller and casing. The blades profiles are cut using laser cutting process. The blades are welded in the impeller plate of 2mm thickness using Tungsten Inert Gas (TIG) welding process. The casing is divided into three parts that are two outer plates and a circumferential cover. These three parts are also joined using welding process. The parts of centrifugal blower are shown in Fig. 8 and 9. The casing and impellers are assembled and mounted on the motor shaft. The speed of the impeller is decided to adjust in the variable speed motor on which the blower is mounted during test process. Finally, the experimental values of pressure and velocity are measured with the help of manometer and anemometer for the corresponding frequency values. The experimental data are shown in table 8.

Figure 8. Casing

Figure 9. Impeller

The manufactured blower was tested in the actual system of dust test chamber developed by following the international standard IEC 60068-2-68. The assembly of blower in dust test chamber as per IEC standard shown in fig.10, 11 & 12 along with CAD model and manufactured system. The rotation speed for comparing the

results for verification was set as 3000 rpm. The comparison of output results at 50 Hz frequency (Pressure and velocity) for the optimum combination of input design parameter (Inlet blade angle (β_1)= 48°, impeller diameter (D_2) = 300 mm and the impeller width (b) = 45 mm) in the different analysis is shown in table 9. Error 1 in table 9 represents the deviation of statistical results with experimental results and error 2 represent the deviation of numerical results with experimental results.
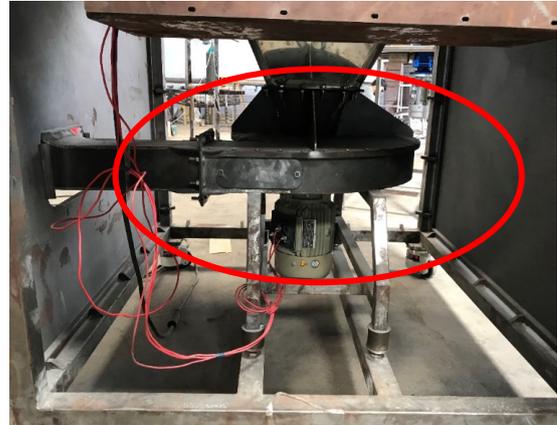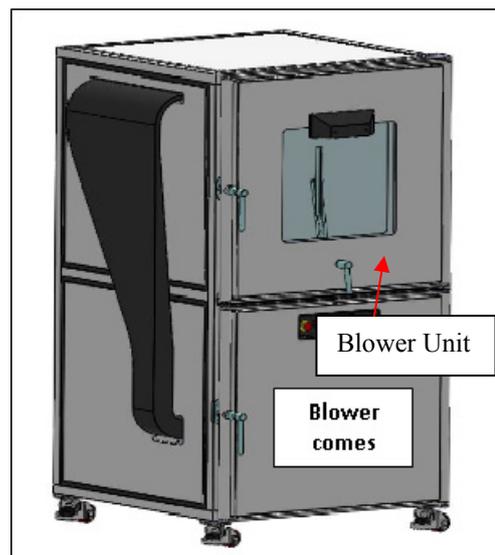

Fig.10. Blower in dust test chamber

Fig. 11. CAD model of dust test chamber

Fig.12. Manufactured dust test chamber

Table.8. Centrifugal blower Experimental test results

S. No.	Frequency (Hz)	Velocity (m/s)		Differential pressure (mm)		Pressure (Pa)	
		Inlet	Outlet	Inlet	Outlet	Inlet	Outlet
1	20.32	3.6	12.7	11	22	107.8	215.8
2	25.41	6.7	16.8	18	32	176.4	313.9
3	30.2	6.8	19.4	25	35	245	343.3
4	35.31	8.9	23.2	29	70	284.2	686.7
5	45.41	9	25.4	69	111	676.2	1432.2
6	50	11.5	26.2	66	146	646.8	1853.4

Table.9. Comparison of results for optimum combination of design paramete

S.No.	Parameter	Numerical Results	Statistical Results	Experimental Results	Error 1 (%)	Error 2 (%)
1	Pressure (Pa)	1956.6	1960.1	1853.4	-5.76	-5.56
2	Velocity (m/s)	25.6	24.0	26.2	8.16	2.22

7. CONCLUSION

The radial tip centrifugal blower is designed using fundamental design concept followed by iterative process. The design parameters (inlet blade angle (β_1), impeller outer diameter and width of the impeller) are optimized using Taguchi L9 orthogonal array with the help of numerical analysis. The regression equations are further developed to establish the relationship between design parameters and outlet responses (pressure and velocity). Gray relational analysis (GRA) is applied on the results of Taguchi's design matrix results and optimum combination of design parameters are identified by ranking the results of GRA. The blower is manufactured finally using optimized design parameters. The experimental test is conducted on the manufactured blower at 50 Hz. The pressure and velocity of the airflow along with dust is measured at the outlet of the blower. The numerical and statistical results of optimized design parameters are compared with experimental results and the results are in good consensus.

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DECLARATION OF CONFLICT OF INTEREST

The Authors declare that there is no conflict of interest.

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ОПТИМИЗАЦИЈА ПАРАМЕТАРА КОНСТ-РУИСАЊА ЦЕНТРИФУГАЛНОГ ВЕНТИЛАТОРА СА РАДИЈАЛНОМ ОШТРИ-ЦОМ ЗА ПРИМЕНУ У КОМОРИ ЗА ИСПИТИВАЊЕ ПРАШИНЕ КОРИШЋЕЊЕМ НУМЕРИЧКЕ И СТАТИСТИЧКЕ АНАЛИЗЕ

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Изузетно значајан параметар ефикасног функционисања електро-техничког уређаја за сакупљање прашине подразумева чврсто спојене компоненте таквог уређаја. Организација ИЕЦ је прописала

стандарде за испитивање делова који раде у тешким условима као што је киша, прашина и др. Поступак за развијање и извођење испитивања сакупљања прашине се врши према стандарду 60068-2-68 прописаном од стране ИЕЦ. Вентилатор је главна компонента која се испитује у циљу изазивања присилног струјања ваздуха у коме има честица прашине и тиме симулира средина са присуством прашине. Центрифугални вентилатор је конструисан за примену у комори за испитивање прашине. Притисак и брзина протока ваздуха на излазу радијалне оштрице центрифугалног вентилатора је највећим делом под утицајем угла излазне лопатице, пречника и ширине радног кола. Детаљно је приказан поступак конструисања и оптимизације наведених параметара применом итерације помоћу MATLAB софтвера. Нумеричка анализа је обављена коришћењем ANSYS Fluent софтвера за израду Тагучијевог плана експеримента. Статистичком анализом, применом Design Expert софтвера, су обухваћени следећи параметри: конструисање вентилатора, излазни притисак и брзина. Оптимизирани параметри конструисања су примењени у производњи вентилатора. Експериментално испитивање је обављено на вентилатору. Експериментални, нумерички и статистички резултати оптимизираних параметара су упоређени и показали су добро слагање.