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DESIGN OPTIMIZATION AND EXPERIMENTAL STUDY ON THE BLOWER FOR FLUFFS COLLECTION SYSTEM

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Abstract

Centrifugal fans play an important role in the fluffs collection system for industrial cleaner. Therefore it has become necessary to study on the parameters which influences the performance of the blower. Parameters chosen for optimization are - fan outer diameter, number of blades and fan blade angle. Taguchi's orthogonal array method helps to find out the optimum number of cases and the modelling has been carried out using SOLIDWORKS. ICEM CFD is used for meshing the blowers and analysed using FLUENT. In this study, analytical results are compared with experimental values. ANOVA is used to find out the percentage contribution of parameters on the output. Using Minitab software the optimum combination is identified. The result shows that the optimum combinations are 190 mm outer diameter, 80° blade angle and 8 numbers of blades.

Keywords: Centrifugal blowers, CFD, Taguchi, ANOVA, Experimental data.

1. Introduction

Travelling cleaner is a machine which is mounted on overhead rails and used to collect the cotton fluffs around the machines by moving on the rail. The cleaner blows the air around the machines and sucks unwanted fluffs from the floor in order to maintain the quality of the end 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 are fixed at both sides of the machines and gets filtered at the top of the 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 was

Nomenclatures

D	Fan diameter, mm
MRF	Moving reference frame
mmwc	Millimeters of water column, mm
Q	Discharge, m^3 / hr

Greek Symbols

β_α	Inner blade angle, deg.
β_β	Outer blade angle, deg.
ρ	Blade Profile in radius, mm

Abbreviations

ANOVA	Analysis of Variance
CFD	Computational Fluid Dynamics
DOE	Design of Experiments
OA	Orthogonal Array

removed using an automatic centralized waste collection system. The present automatic centralized waste collection system consists of a pipeline connection and a separate blower system with 5 HP motor, fitted at one end of the plant. This pipe line connection is used to transport the waste from the machine to centralized blower.

Centrifugal blower has three important parts namely, fan, volute casing and inlet duct. When the fan rotates, creates low pressure zone at the inlet thus intakes the air from atmosphere. The air goes radially and then circulated in the volute casing and comes out orthogonally.

In this present study, analysis the performance of the centrifugal blower will replace the centralized dust collection system. The pulley which is rotating idle serves as a rotating unit for the blower. The centralized dust collection system collects the flying fluffs from four parallel travelling cleaner.

Garron and Bruce [1] optimised the blower housing using a Design of Experiments (DoE) technique where the geometry of housing was varied in a structured manner to capture expected second order behaviour. Jayapragasan et al. [2] analysed the importantce of centrifugal fans role in the proper functioning of any travelling cleaner. The blades of the fan were fixed between the inner and outer diameters. Because of its high static pressure and capability of handling air streams containing a high level of particulate, radial blade is suitable for the application of dust laden. Keyur and Prajesh [3] introduced the CFD part for improving the results of static pressure and efficiency generated at the entry of the impeller. The MRF (moving reference frame) applied in the CFD analysis of centrifugal impeller as a rotating region around the impeller and component of the impeller stationary. Comparisons are conducted between the original impeller and two larger impellers with the increments in impeller outlet diameter of 5% and 10% respectively in the numerical and experimental investigations. Chunxi et al. [4] analysed and showed that the flow rate, total pressure rise, shaft power and sound pressure level have increased, while the efficiency have decreased when the impeller operates with larger impeller. The impeller with radial tipped blades showed a weak dependency on tip clearance. The inlet radius had a major impact on the flow rate of the centrifugal blower. Too small or too large of the inlet radius will result in a noticeable loss in the flow rate.

Pham et al. [5] analysed the effect of bell mouth radius which has a moderate impact on performance of the centrifugal blower. Too small radius will have negative impact on flow rate. Anyway, it's best to have the bell mouth radius ratio at the value around 9%. 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.

Ragoth and Nataraj [6] conducted experiments using taguchi orthogonal array (OA) based design of experiments (DoE) technique and determined the required experimental trials. The experimental results were justified by using Analysis of Variance (ANOVA). The open wheel or paddle wheel is the most common of the radial blade impellers [7]. Air wheel and radial tip impeller are the other variants of radial impellers which are ideal for contaminated air streams but both were not intended for bulk material handling.

Vijaypratap et al. [8] studied the critical parameters of the centrifugal impeller and found that the outlet width, outlet diameter, blade thickness, impeller blade outlet angle, and number of blades of the impeller which will highly affect the performance. Fernandez and Nirschl [9] analysed the effect of the distance of the blade with top of the casing and have been deducted that the impeller with backward-curved blades was very sensitive, whereas the other two types were not.

Xiaojin et al. [10] analysed a 250 mm backward curved blower impeller through modelled in two CFD scenarios: one which resolves the impeller and shroud geometry and one which only the shroud geometry. Church [11] compared between the test data MRF models and lumped compact models using the P-Q curves and found that the flow fields was much better than the lumped compact models.

The radial blades are employed for handling dust-laden air because they are less prone to blockage and dust erosion. In the present study, the radial blade profile has been analysed for the required application to find the optimal combination of the blower. In a few research paper of the radial fan blower performance analysis, it was observed that the width of the impeller alone had been considered whereas the operating parameters such as outer diameter, width and number of blades of the fan had not been taken into consideration. Even though little scientific documentation related to blower performance was observed, no one has attempted to formulate an experimental design to find the optimal combination of blower operating parameters. In this context, our laboratory has focused to investigate the above said parameters to find the optimal condition of blower for centralized collection system.

2. Methods

The methods comprises of following steps:

1. Taguchi's orthogonal array method.
2. Modelling and meshing of blowers.
3. CFD analysis.
4. Experimental set up.

2.1. Taguchi's orthogonal array method

The parameters that influence the pressure and power are outer diameter, outer blade width, speed, fan blade angle, and number of blades of the fan. In the above mentioned parameters, speed of the fan is fixed at 3660 RPM, and fan outer blade width is dependent on fan outer diameter so, it cannot consider these parameters for optimization. Therefore the parameters selected are outer diameter, blade angle, and number of blades of the fan is 6-12.

Three parameters have been selected each having four levels for optimization. For full factorial method, (4^3 = 64 experiments) it is supposed to do more experiments. Therefore the Taguchi's orthogonal array method was used for the same number of parameters and levels by which is reduced to 16 experiments (L16). The models and test conditions are shown in Table 1.

The orthogonal array of four models fan which was experimentally observed is shown in Table 2.

Table 1. Parameters with their levels.

Parameters	Level 1	Level 2	Level 3	Level 4
A. Fan outer diameter (mm)	170	180	190	200
B. Fan blade angle (deg.)	60	70	80	90
C. Number of blades	6	8	10	12

Table 2. L16 Orthogonal array.

Case No.	Fan outer diameter (mm)	Fan blade angle (deg.)	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

2.2. Modeling and meshing of blowers

The 3D modelling of the blowers are created using SOLIDWORKS. The major parts of the blowers are fan and volute casing. Figure 1 shows the assembly of fan and volute casing. Figures 2 and 3 show the geometry configuration of blade profile and 3D model of the blower assembly.

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. The calculation of blade profile is given by Eq. (1)

$$P = \left(R_b^2 - R_a^2\right)/2\left(R_b \cos\beta_b - R_a \cos\beta_a\right) \tag{1}$$

where (R_b) and (R_a) are the outer and inner radius of the fan respectively and β_b and β_a is the outer and inner radius of the blade angle.

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

$P = (85^2 - 40^2)/2(85 \cos 90 - 40 \cos 60)$, $P = 66.18$ mm

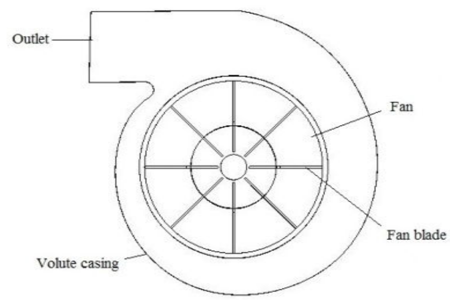

Fig. 1. Front view of centrifugal blower.

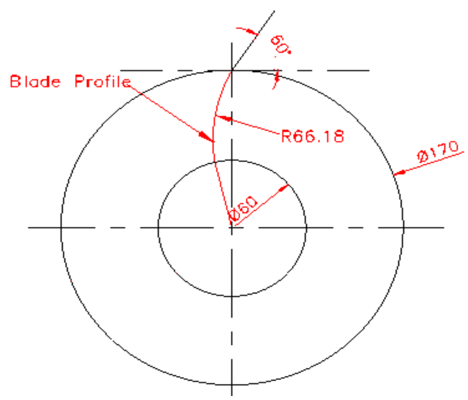

Fig. 2. Configuration of blade profile.

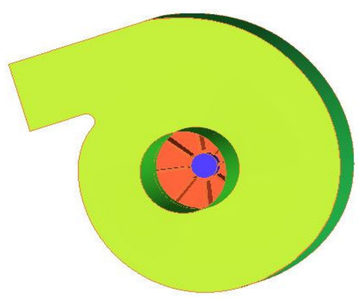

Fig. 3. 3D Model of blower.

The volute casing was kept constant and the fan alone had been changed during modelling. Jayapragasan et al. [12] an analytical performance of 16 cases of the blower were carried out and got the optimum combination of parameters. These analytical values were taken to compare with the experimental results. The few modelling of the designed fans are shown in Figs. 4(a) and (b).

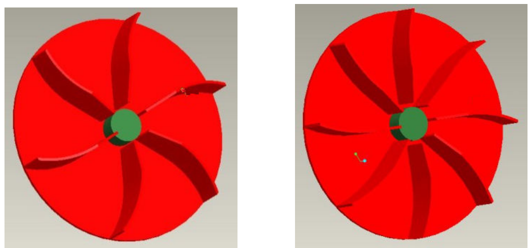

Fig. 4(a). 190 mm diameter fan with 6 and 8 number of blades.

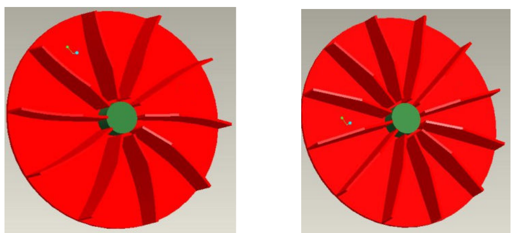

Fig. 4 (b). 190 mm diameter fan with 10 and 12 number of blades.

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. The mesh element size for volute casing was kept constant throughout the analysis. After grid generation, the mesh independency tests were carried out and the results were shown in Table 3.

Table 3 indicates, it is clear that mesh size 2, mesh size 1.5, and mesh size 1 are converged. Hence mesh size 2 is chosen to reduce the computational time. The 3D meshed model of the blower is shown in Fig. 5.

Table 3. Mesh independency test.

Inlet fan element size	Number of elements	Static pressure (mmwc)
4	697307	64.07626
3.5	1030738	67.06006
3	1411083	68.39047
2.5	1801259	69.21327
2	2339349	69.29532
1.5	2804579	69.29566
1	3321614	69.29577

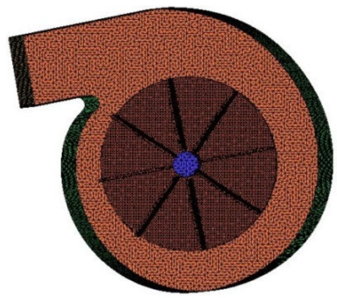

Fig. 5. 3D meshed model of blower.

2.3. 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 study, CFD FLUENT was 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,

1. Steady state air flow.
2. Implicit solver.
3. Standard wall function.
4. Second order upwind scheme.

Table 4 represents the boundary conditions of the blower for analysis.

Table 4. Boundary conditions.

Boundary Condition	Value
Material	Air
Density	1.2 kg/m^3
Viscosity	1.789e-05 kg/m-s
Turbulence model	K-ω SST model
MRF	Fan
Inlet condition	Pressure inlet
Hydraulic diameter at inlet	0.080 m
Hydraulic diameter at outlet	0.075 m
Fan speed	383.274 rad/s

Once the solution is converged the results are post processed using CFD POST. Figures 6 to 13 represent the various results of static pressure and velocity contour for a few cases. Figures 7, 11 and 13 show the velocity contours and it was found that the low velocity zones for curved blades are moving away from the fan centre which is due to the circularity effect of the fluid. Figure 9 shows that the low velocity zone stays near the fan centre.

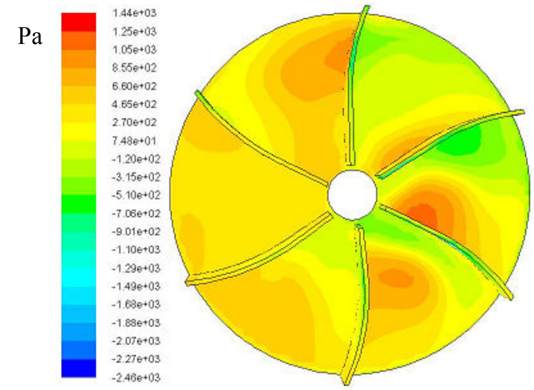

Fig. 6. Static pressure contours of case 6 blower.

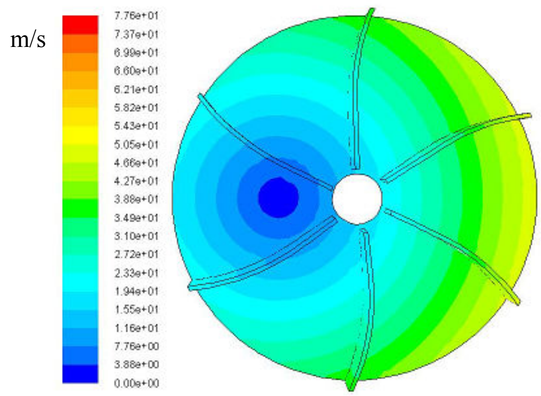

Fig. 7. Velocity contours of case 6 blower.

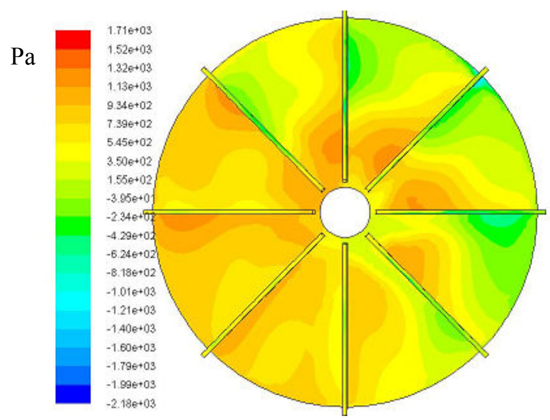

Fig. 8. Static pressure contours of case 12 blower.

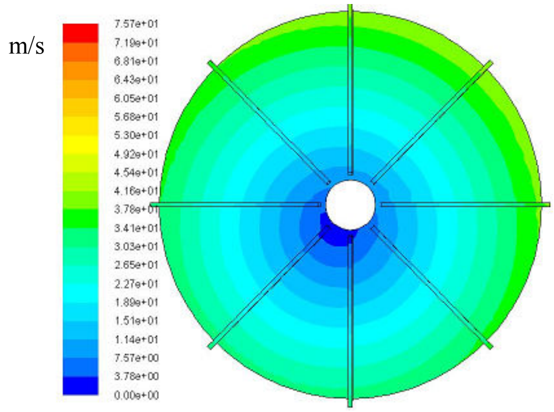

Fig. 9. Velocity contours of case 12 blower.

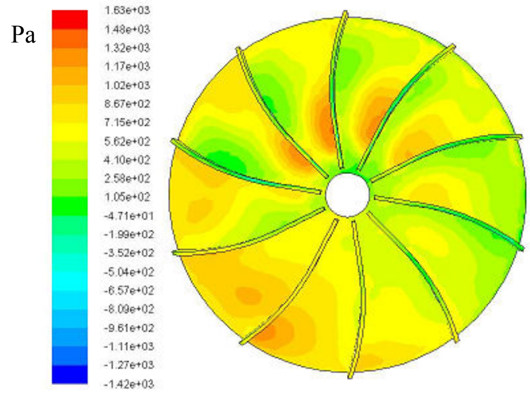

Fig. 10. Static pressure contours of case 15 blower.

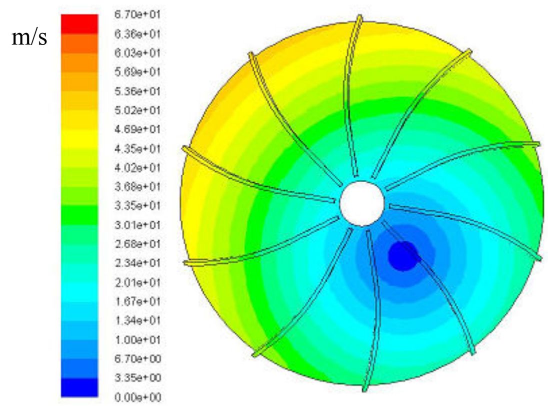

Fig. 11. Velocity contours of case 15 blower.

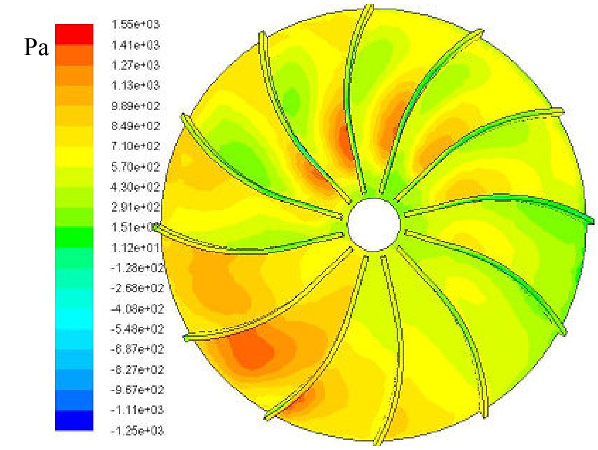

Fig. 12. Static pressure contours of case 13 blower.

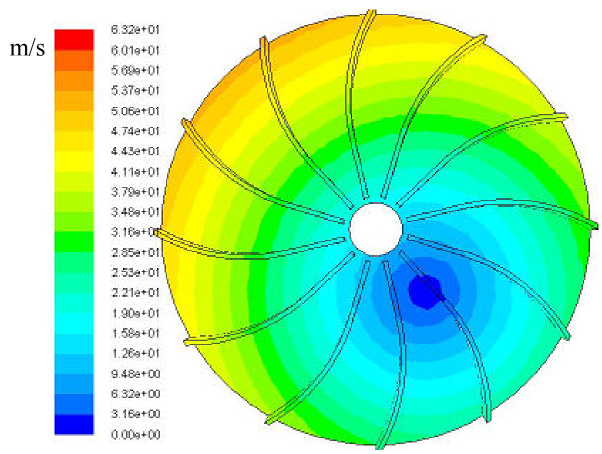

Fig. 13. Velocity contours of case 13 blower.

2.4. Experimental Setup

As per the Standard Procedure the Experimental set up was developed and determined the air delivery and pressure of the fan. The fan inlets shall be attached with a parallel duct having the same cross section as the fan outlet and length equal to twice its diameter. The four side tappings at plane shall be equally spaced at 90 deg. on the cylindrical duct. The four side tappings shall be connected to the manometer, each connection being of the same length, bore and arrangement of tubing to minimize the effect of flow due to difference of pressure at the tappings. The other limb of the manometer shall be opened to the ambient atmosphere and the manometer reading shall be taken as equal to the average static pressure in the airway. A resistance comprising a screen having evenly spaced aperture of uniform size, not exceeding D/20 should be filled at a distance D from the commencement of the cylindrical portion of the inlet. The screen may be composed of one or more layers of even wire or fabric supported by a wire guard. Figures 14 and 15 show the assembly parts of the blower and schematic diagram of standard experimental setup.

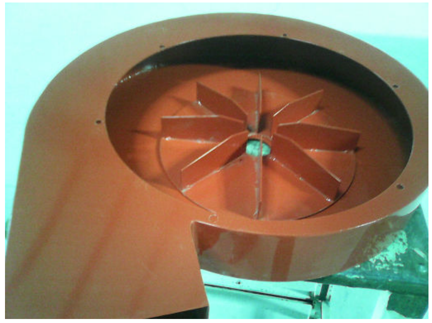

Fig. 14. Experimental model of blower and fan assembly.

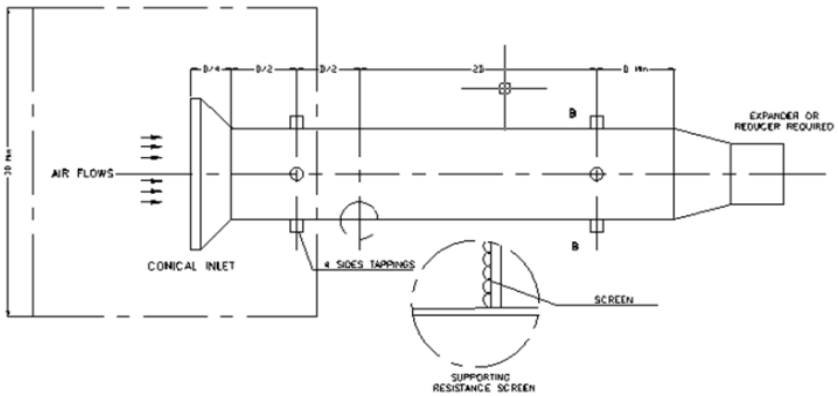

Fig. 15. Schematic diagram of standard experimental set-up.

The sources of error in the experiment are needed to define certain terms related to measurement uncertainty. In this experimental set up, instrumental error such as co-efficient of discharge (C_d) of the conical inlet was observed and the error was nullified through proper calibration with air at ambient pressure, controlled inlet temperature 25±1°C and relative humidity 55±5%. The overall uncertainty in measurement of actual flow rate is calculated using the formula Eq. (2)

$$E_s(q_a) = \pm\sqrt{E_a(q_c)^2 + E_s(p_c{}^2) + E_s(p_t{}^2)} \qquad (2)$$

$$E_s(q_a) = \pm\sqrt{(0.02)^2 + (0.037^2) + (0.03^2) + (0.01^2)}$$

$$E_s(q_a) = \pm 0.99\,\%$$

The configurations of different types of fans are shown in Figs. 16(a) to (h) for experimental study.

(a) 6 Blade radial straight.

(b) 6 Blade radial curved.

(c) 8 Blade radial straight.

(d) 8 Blade radial curved.

(e) 10 Blade radial straight.

(f) 10 Blade radial curved.

(g) 12 Blade radial straight.

(h) 12 Blade radial curved.

Fig. 16. Configuration of fans in experimentation.

3. Results and Discussion

The CFD and experimental results of different type of fans are discussed in detail in the following subheadings.

3.1. ANOVA results

Analysis of variance was used to find out the percentile contribution of each parameter on pressure value of the blowers. ANOVA table for CFD results and experimental results is mentioned in the below Tables 5 and 6.

Table 5. ANOVA table for analytical results.

Factors	DOF	Sum of squares	Mean squares	Variance	% Contribution
Fan outer diameter	3	836.830	278.943	69.767	80.619
Fan blade angle	3	102.031	34.010	8.506	9.828
Number of blades	3	99.271	33.090	8.276	9.562
Total	9	1038.133	-	-	-
Error	6	23.989	3.998	-	-

Table 6. ANOVA table for experimental results.

Factors	DOF	Sum of squares	Mean squares	Variance	% Contribution
Fan outer diameter	3	1028.385	342.795	35.445	75.018
Fan blade angle	3	179.101	59.700	6.173	13.065
Number of blades	3	163.351	54.450	5.630	11.916
Total	9	1370.852	-	-	-
Error	6	58.031	9.671	-	-

3.2. MINITAB results - Analytical

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 blades on static pressure to get the optimum combination of these parameters. The CFD and experimental results of mean effects plot for means and for SN ratio is shown in Figs. 17 and 18. In CFD results, the fan outer diameter influences the most on static pressure while the number of blades influences the least. Since the aim is get the nominal static pressure and power value, nominal is better has chosen. Table 7 represents the maximum affecting parameters from CFD. The optimum combination of the fan from CFD is shown in Table 8.

Table 7. CFD maximum affecting parameters.

Levels	Fan outer diameter (mm)	Fan blade angle (deg.)	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

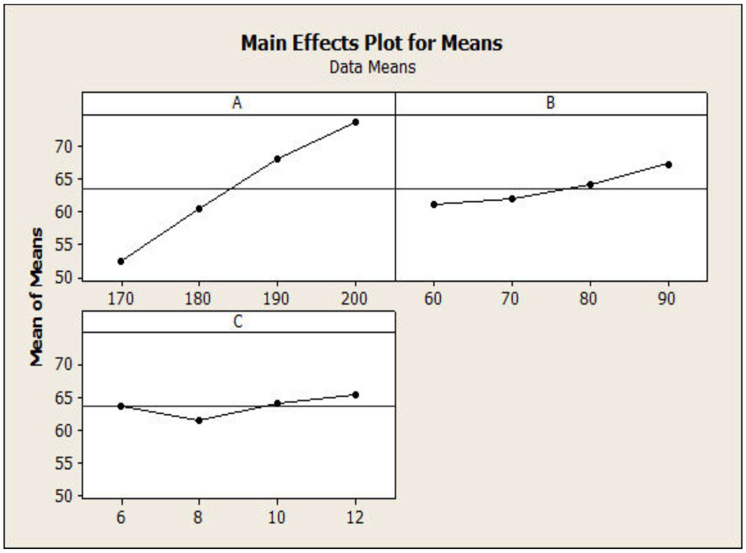

Fig. 17. CFD - Responses of parameters by mean effects plot for means.

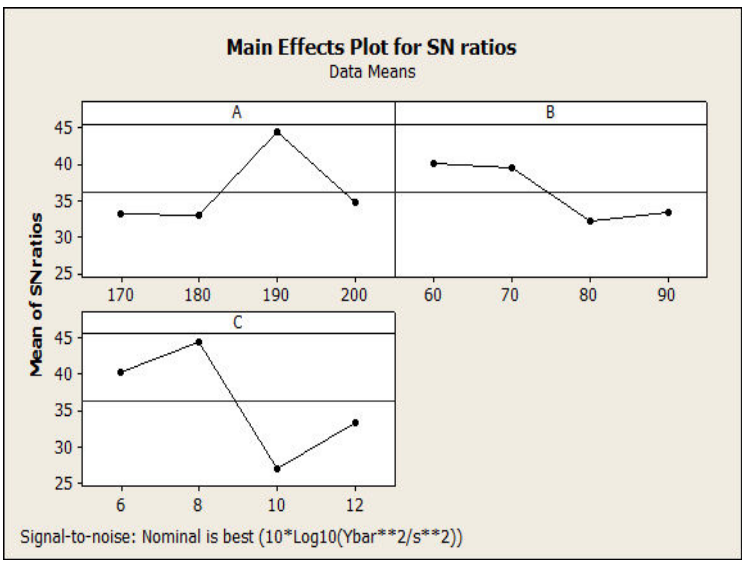

Fig. 18. CFD - Responses of parameters by mean effects plot for SN ratios.

Table 8. CFD optimum combination.

Fan outer diameter (mm)	Fan blade Outlet angle (deg.)	Number of blades
190	60	8

3.3. MINITAB results - Experimental

In experimental results also the fan outer diameter influences the most on static pressure while the number of blades influences the least. Since the aim is get the nominal static pressure and power value, nominal is better has chosen. The maximum affecting parameters from experimental are shown in Table 9.

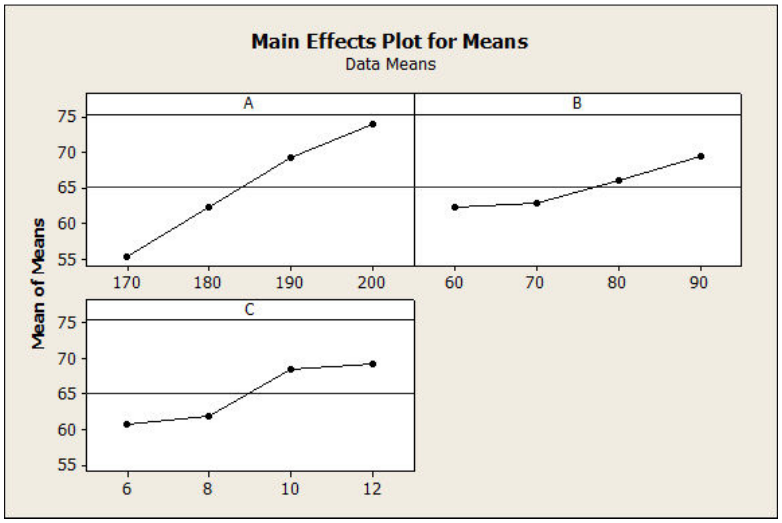

Fig. 19. Experimental - Responses of parameters by mean effects plot.

Table 9. Experimental maximum affecting parameters.

Levels	Fan outer diameter (mm)	Fan blade angle (deg)	Number of blades
1	55.204	63.954	61.872
2	63.751	64.183	62.983
3	69.894	66.381	68.473
4	74.526	71.586	69.483
Δ	19.322	7.632	7.611
Rank	1	2	3

The outcome of the experimental results, simulation and predicted results are represents in Table 10. The deviation was found to be less than 5% when compared to the experimental and predicted value. The results show the good agreement between the predicted and experimental value for pressure has been observed. Figure 20 shows the plots of the pressure value and the case numbers for analytical and experimental value. The graph shows that the experimental value of the system is high for the fan diameter 190 mm with 80 degree angle.

Table 10. Comparison of experimental results with analytical results.

Case No.	Analytical Pressure value (mmwc)	Experimental pressure value (mmwc)	Predicted value (mmwc)	Deviation (%)
1	47.60	42.68	41.76	2.15
2	50.57	53.44	53.56	0.23
3	57.77	62.71	62.59	0.19
4	59.38	67.86	68.85	1.45
5	55.88	57.94	58.92	1.69
6	57.73	53.27	54.92	3.10
7	65.52	68.92	67.71	1.75
8	66.90	72.16	70.53	2.26
9	67.09	72.67	71.69	1.34
10	68.05	70.25	71.46	1.72
11	68.62	66.95	65.30	2.47
12	69.29	70.24	71.87	2.32
13	75.06	79.51	80.09	0.73
14	72.85	76.18	74.21	2.58
15	68.35	69.84	71.81	2.82
16	75.85	73.53	72.88	0.88

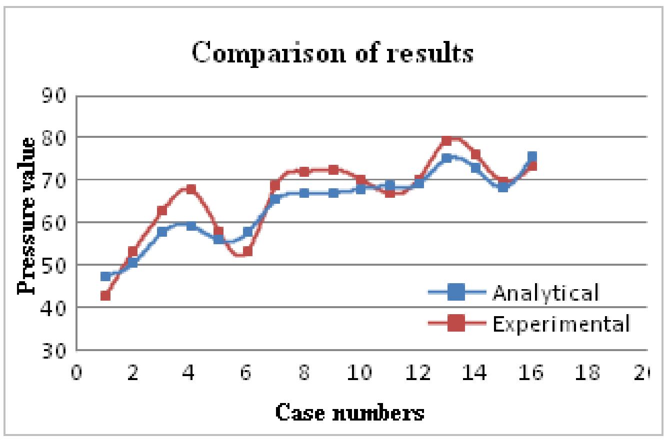

Fig. 20. Comparison of Analytical and Experimental results.

From the regression analysis, it was found that the second-order polynomial equation explained the regression coefficients of pressure and is given by Eq. (3)

$$Y = -554.902A + 3.902A + 3.021B + 20.05C - 0.0041A^2$$
$$+ 0.009B^2 - 0.19C^2 - 0.019AB - 0.05AC - 0.07BC \qquad (3)$$

where Y is the response value. In this experiment, Y value is the Pressure value (mmwc). A, B and C represent the parameters outer diameter, blade angle and number of blade respectively. From the above regression equation, it was clearly observed that the square terms and interaction terms has less influence in the response. Hence the linear term was taken in to the consideration and is given by Eq. (4)

$$Y = -554.902A + 3.902A + 3.021B + 20.05C \tag{4}$$

Results of categoric regression analysis which was applied for determining effects of parameters that effective on identified characteristic are

R-Sq = 98.18% and adjusted R-Sq = 95.45%

The regression coefficient of pressure results clearly shows that the R-Square and the adjusted R-Square values are more than 95%. This shows that the variance between experimental results and analytical results are minimum. The S/N ratio for each parameter is shown in Fig. 21. The level of each parameter that provides the optimum level that is listed in Table 11.

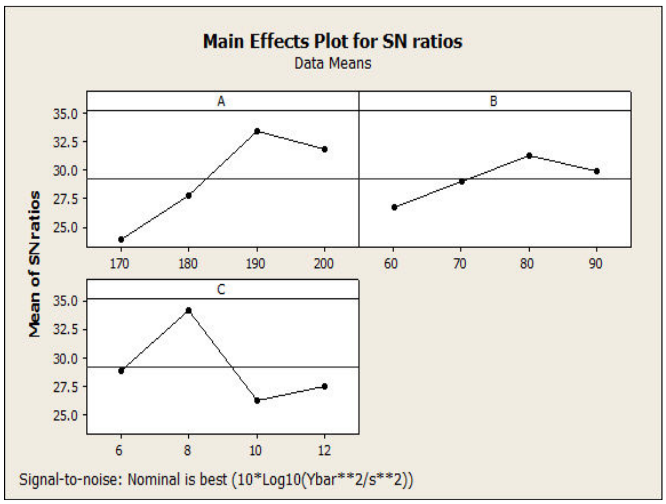

Fig. 21. Experimental - Responses of parameters by mean effects plot for SN ratios.

Table 11. Experimental optimum combination

Fan outer diameter (mm)	Fan blade outlet angle (deg)	Number of blades
190	80	8

4. Conclusions

In this study, the major parameters like the outer diameter, blade angle and number of blades of the fan were considered and analysed. The CFD and experimental approach helps to improve the results and the following conclusions were made.

- The rotational frequency of fan was kept constant and the parameters such as outer diameter, width and number of blade of the fan were considered for the analysis. The analytical performance works were carried out to get the optimum combination of blower parameters. These analytical values were taken to compare the experimental results.

- Analysis of variance (ANOVA) for analytical results shows that the static pressure is influenced by fan outer diameter (80.619%) followed by fan blade angle (9.828%) and number of blades (9.562%).
- The experimental work has been carried out and the analysis of variance results shows that the static pressure is influenced by fan outer diameter (75.018%) followed by fan blade angle (13.065%) and number of blades (11.916%).
- Taguchi's orthogonal array method thus has helped to reduce the number of trials and save the time. The optimum combinations for analytical results are 190 mm outer diameter, 60^0 blade angle and 8 numbers of blades but the optimum combinations for experimental results are 190 mm outer diameter, 80^0 blade angle and 8 numbers of blades. This is due to the assumptions made in CFD like steady state air flow, implicit solver, constant temperature, constant air density and viscosity.
- The deviation was found to be less than 5% when compared to the experimental and predicted value. The results show the good agreement between the predicted and experimental value for pressure that has been observed.
- From the above results, it is concluded that the combination obtained using experimental results (190 mm, 80^0 and 8 number of blades) has been chosen as an optimized alternative blower for fluff collection system of travelling (Industrial) cleaner.

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