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Design Optimization of Aerofoil Bladed Radial Fan

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Abstract: **The aim of the project work is design optimization of Aerofoil bladed radial fans through finite element analysis. The cost of the present impeller is very high due to its massive weight. The design and analysis is carried out for the present impeller of Aerofoil bladed radial fan. Both Static and modal analysis is performed to determine the stress distribution, maximum stress induced, maximum deflection and the nature frequency. Design and Analysis is done by commercial software Ansys.**

Key words: Radial fan, Ansys.

II. INTRODUCTION

Most manufacturing concerns spend more than a 60% of their money for materials, i.e., material soak up substantial portion of the capital invested in an industrial concern. This emphasizes the need for adequate material management because even a small saving in the material can reduce the production cost to a fair extent and thus add to the profits. The analysis is carried out in **ANSYS.** The analysis was carried out in two phases comprising of static and modal analysis. Static analysis consists of finding the deformation under loading, the strains and the working stresses. The modal analysis consists of predicting the natural frequency of vibration of the rotating impeller.

The objective of this study was to optimize the weight of the radial fan by using commercial software Ansys.

III. DESCRIPTION OF FANS

Fans are one of the turbo machines used for energy transfer. The main aim of a fan is to move a gas without an appreciable increase in its pressure. The total pressure developed by fans is of the order of a few millimeters of water gauge (W.G.). The flow is continuous because the flow at entry and exit and also through the impeller is steady.

Classification of Fans:

The fans are classified as according to the direction of the flow of the fluid with respect to the fan axis. The schematic representation of different kinds of fans are shown in Figure1.

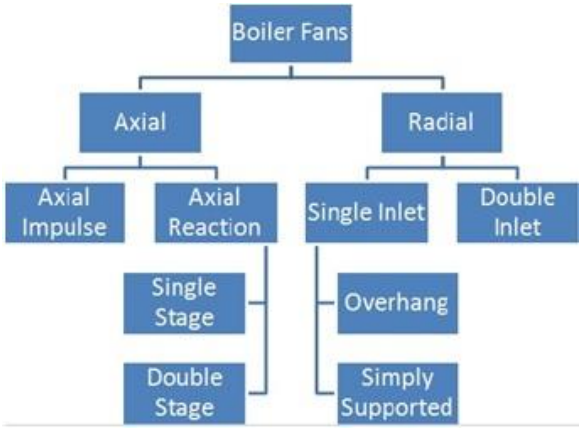

Figure1: Schematic representation of different kinds of fans

Axial flow fan: The flow is parallel to the axis of rotation of the fan both at entry and exit. Axial fans may be classified further into impulse type and reaction type fans.

Radial fan: The fluid enters into the rotating element (rotor) perpendicular to the axis of rotation. The schematic representation of radial fan shown in Fig2.

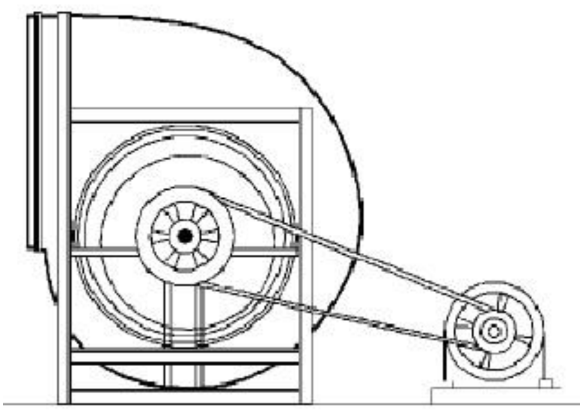

Figure 2: Radial fan

IV. SELECTION OF FANS

The fans are commonly utilizing in power plants, ventilation systems and cooling of electric motors and generators [3].

The guide lines for selection of fans are as follows.

1. The operating point should lie in the peak efficiency zone at the highest possible vane or blade opening.
2. The maximum operating speed should be chosen so that the unit size is small.
3. The operating parameters selected should not lie in the unstable zone or even close to the 'stall line'. It should be ensured that the system resistance curve does not cut the stall line and enter the unstable zone.

From the above guide lines we are chosen radial fan.

V. SELECTION OF MATERIAL

The material chosen for above mentioned radial fan is NAXTRA 70. The properties of above material are Poisson's ratio is 0.3, density is 8.002E-10kg/mm^3 and young's modulus is 21000kgf/mm^2[1]. The impeller is rotated at 1000 rpm. Geometrical parameters of the existing and modified impeller are shown in Table1 and Table 2 and aerodynamic parameters are shown in Table 3.

Table 1: Geometrical parameters of existing model

S.NO.	GEOMETRICAL PARAMETERS FOR PRESENT MODEL	VALUES
1	Inner diameter of Bottom Back Plate	300
2	Outer diameter of Bottom Back Plate	650
3	Thickness of Bottom Back Plate	20
4	Inner diameter of Top Back Plate	650
5	Outer diameter of Top Back Plate	1000
6	Thickness of Top Back Plate	10
7	Inner diameter of Cover Plate	650
8	Outer diameter of Cover Plate	1000
9	Thickness of Cover Plate	8
10	Diameter of Rim	650
11	Thickness of Rim	30
12	Diameter of Flange	300
13	Thickness of Flange	50
14	Arc length of Blade	290.2
15	Thickness of Bottom Part of Blade	12
16	Thickness of Top Part of Blade	3.15

Table 2: Geometrical parameters of Modified Model

S.NO.	GEOMETRICAL PARAMETERS FOR MODIFIED MODEL	VALUES
1	Inner diameter of Bottom Back Plate	300
2	Outer diameter of Bottom Back Plate	650
3	Thickness of Bottom Back Plate	16
4	Inner diameter of Top Back Plate	650
5	Outer diameter of Top Back Plate	1000
6	Thickness of Top Back Plate	8
7	Inner diameter of Cover Plate	650
8	Outer diameter of Cover Plate	1000
9	Thickness of Cover Plate	6.4
10	Diameter of Rim	650
11	Thickness of Rim	24
12	Diameter of Flange	300
13	Thickness of Flange	40
14	Arc length of Blade	290.2
15	Thickness of Bottom Part of Blade	9.6
16	Thickness of Top Part of Blade	2.52

Table 3: Aerodynamic parameters

S.NO	PARAMETER	VALUE	UNIT
1.	Coal flow rate	30	m^3/sec
2.	Pressure	1261	mm of WC
3.	Air Temperature	280	^{0}C
4.	Temperature Difference	6 to 10	^{0}C
5.	Motor Capacity	446	kW
6.	Motor Speed	1000	rpm

VI. NUMERICAL SIMULATION

For numerical simulation the above mentioned radial fan is selected. The operating speed is taken as 1000 rpm and angular velocity is 104.67 rad/sec[2]. and an iterative solver is chosen for simulation. The thickness of the model is reduced up to 20% of the existing design. The mesh generation and design of the mode accomplished by commercial software Ansys[2], this uses finite element method. The schematic representation of model diagram shown in figure 3.

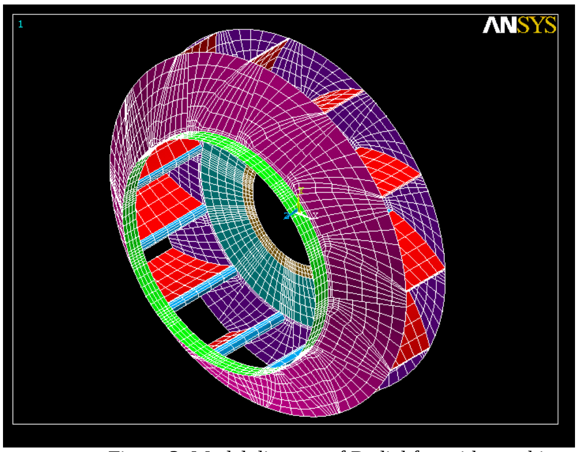

Figure 3: Model diagram of Radial fan with meshing

VII. VALIDATION

The variables stress, deflection, mass and cost are validated with existing design, the variations are shown in below Table. The variation in mass and cost of the impeller shown in figure 8 and 9.

	Stress (Kgf/mm^2)	Deflection(mm)	Mass(Kg)	Cost (Rs)
Existing Model	10.7	0.23	8000	28,00,000
Modified model	11.7	0.27	6400	22,40,000
Maximum limit	12.5	0.3	-	-
% Reduction	-	-	20	20

VIII. RESULTS AND DISCUSSIONS

Thus we have done the iterative analysis on the impeller by reducing the thickness of all parts of impeller in steps 2% and finally we end up with a maximum reduction of 20% thickness with all constraints satisfied. The stress distribution on the modified impeller and cover plate as shown in figures 4 and 5. The maximum stress induced and also the maximum deflection is examined. Actually the stress induced in Cover Plate ($11.7 kgf/mm^2$) is maximum of all and it is within the maximum allowable stress $12.5 kgf/mm^2$. The maximum deflection occurs in Cover Plate (0.27mm) which is also within the allowable limit of 0.3mm. Thus the constraints are satisfied and the design is safe. The natural frequency of the modified model is reduced than existing model shown in Figure 6 and 7. From figures 8 and 9 mass and cost of the modified model decreasing rather than existing model.

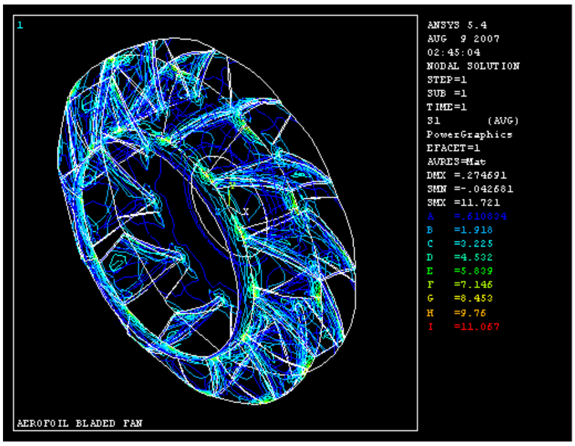

Figure 4: Stress distribution in the impeller

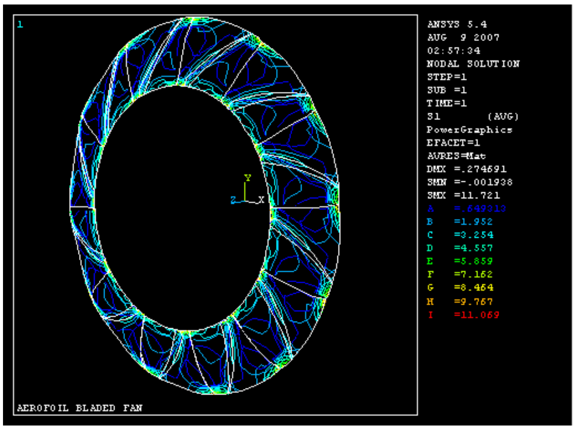

Figure 5: Stress distribution in the cover plate

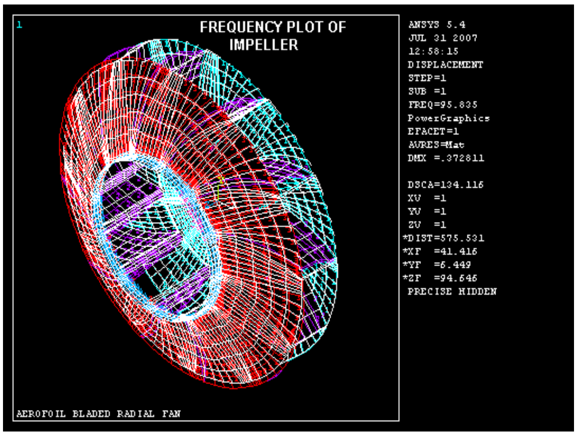

Figure6: Natural frequency of existing model

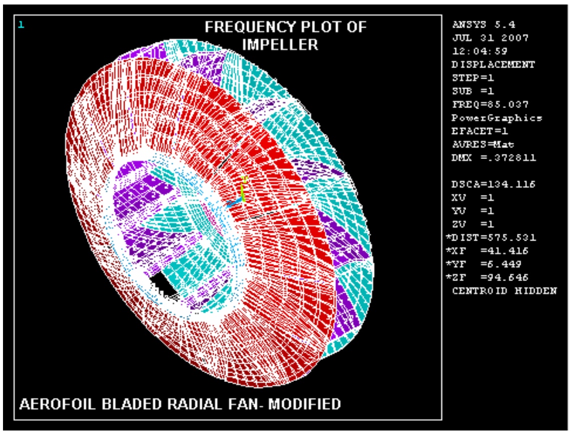

Figure 7: Natural frequency of modified model

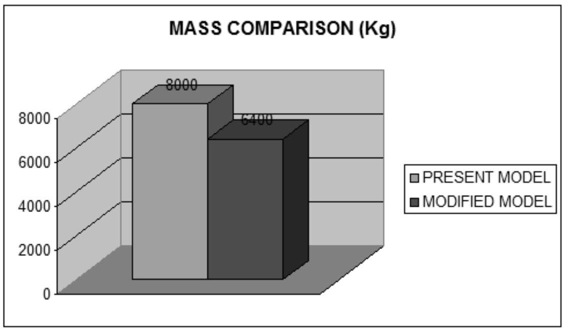

Figure 8: Mass comparison

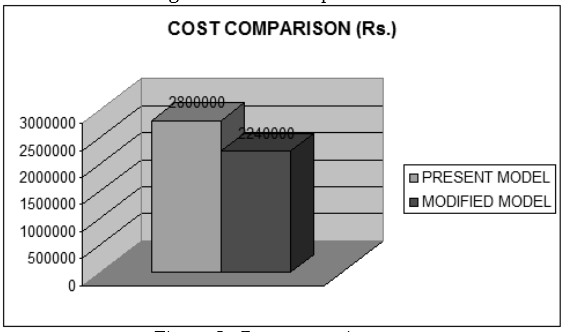

Figure 9: Cost comparison

IX CONCLUSIONS

1. The value of the product can be increased either by increasing its utility with the same cost or by decreasing its cost for the same function
2. The cost and weight of the radial fan is decreases without changing its usefulness.
3. From static and modal analysis when decreasing the thickness of the radial fan the weight and cost both are reducing and at the same time frequency is decreasing, stress and deflections are increasing.

X. REFERENCES

(1) P.S.G data Handbook, Design data., PSG college of Technology.
(2) Zienkiewicz. O. C., Irons, B. M. scott, F.C., and Campbell, J. S., Three dimensional stress analysis, “Proceedings, Symposium on high speed computing of elastic structures”, University of Liege, Belgum.
(3) Engineering cook book, A hand book for the Mechanical designer, Loren cook company, Springe field, MO.
(4) www. Ansys.com