Structural Analysis and Weight optimization of Blower Housing

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Abstract—A centrifugal blower is a mechanical device for moving air or other gases. Centrifugal fans are one of the most widely used pieces of rotating mechanical equipment found today. Fans are used in offices,homes,automobiles industry and many other applications. The most common use of centrifugal fans is to move air and gases for air handling or ventilation systems. Centrifugal fans utilize centrifugal force (thus their name) to increase the velocity of the air/gas it passes between the fan wheel blades and exits at the tip of the fan wheel. In this project work is focused on blower housing. In this practical application is considered for working conditions, various loads acting on it. Based on the study, structural analysis carried out to know for stresses and deformation. This has given the area for low and high stressed region. Another area of work is optimization. Material can be removed from low stress region to optimize for weight. Or strength can be increased by making structural changes.

Index Terms- Centrifugal blower, Analysis and Optimization.

I. INTRODUCTION

A centrifugal blower is a mechanical turbo device for moving air or other gases. The terms blower and squirrel cage fan are used as synonyms. Those fan increase the speed of air stream with the rotating impellers. They use the kinetic energy of impellers or the rotating blade to increase the pressure of the air/gas stream which in turn moves them against the resistance caused by blower ducts, dampers and other components. Centrifugal blowers accelerate air thorough radially, changing the direction (typically by 90 Degree) of the airflow. They are quiet,sturdy, reliable, and competent of working over a wide range of conditions. These blowers are constant CFM devices or constant volume devices, meaning that, at a fan speed constant, a centrifugal fan pump a fix volume of air rather than a constant mass. This means that the air velocity in Centrifugal Blower is fixed even though mass flow rate through the fan is not. The centrifugal blower is one of the most extensively used fans. Centrifugal blowers are by far the most prevalent type of fan used in the Heating,Ventilation and air Conditioning industry today. They are usually cheaper than axial fans and simpler in construction. Centrifugal blower used in transporting gases or any materials and in ventilation system for buildings. They are also used commonly in central cooling and heating systems. They are also well-suited for industrial processes and air contamination control systems. It has a blower rotating wheel composed of a number of blades/ ribs, mounted around a hub, hub turns on a driveshaft that passes through the blower housing. The gas enters from the side wall of the fan wheel, turns 90^0 and accelerates due to centrifugal force as it flows over the fan blades and finally exits the blower housing.

Main parts of a centrifugal blower are:

1. Blower Housing
2. Impeller or Fan Wheel
3. Inlet and outlet ducts
4. Drive Shaft
5. Drive mechanism

Fig I. IMAGE OF A BLOWER

II. PRINCIPLE OF WORKING:

The centrifugal fan uses the centrifugal force generated from the rotation of impellers to increase the pressure of air or gases. When the impellers rotate, the gas near the impellers is get away from the impellers due to the centrifugal force and then moves into the casing. Due to this gas pressure in the fan casing is increased. Then gas is guided to the exit via exit ducts. After the gas is thrown off, the gas pressure in the middle region of the impellers decreases. The gas from the impeller eye makes rush to normalize this pressure. This cycle repeats and therefore the gas can be continuously transferred.

III. CAD MODEL GENERATION

A lot of research has gone into more and more systematic design of centrifugal blower. Everyauthorhas been suggested different procedures, even if each has a slightly different method of calculation, the wide underling principles all are similar. Some of the factors needed for selection of blower are the static pressure the blower must overcome, the normal air flow volume required, the shape and direction of the desired air flow, audible noise allowances,space limitations,efficiency,available power, air density, and cost. The first two of these, air flow and static pressure, along with available power considerations are generally the most critical for system designers. For design of casing first off all design of impeller is required and finally design of casing is done.

Selection of Centrifugal Blowers is based on following:

1. Air flow rate required by the process in m^3/hr. orcubic feet/min.
2. Static pressure as offered by the ducting. In inch/ mm of wc. (needs to be calculated based on ducting design)
3. Temperature at which Air enters into the blower.
4. Application for which centrifugal fans & blower is required

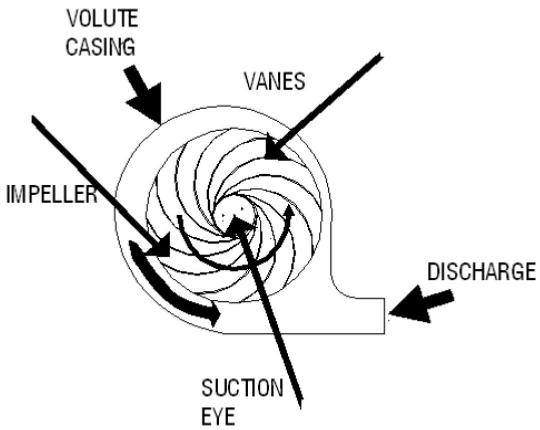

FIG II.: SECTION DRAWING OF A BLOWER

Figure illustrates a cross section of the blower designed. Fluid enters at the inlet port which is center of the rotating impeller;we can call it as suction eye. As the impeller spins in aanti-clockwise direction, it thrusts the fluid outward radially, cause centrifugal acceleration. As it does this, it creates a vacuum in its wake, drawing even more fluid into the inlet. Centrifugal acceleration built energy proportional to the speed of the impeller. The faster the rotation of impeller,the faster the fluid movement and the stronger its force. Impellers are the rotating blades/Ribs that actually move the fluid. They are coupled to the drive shaft that rotates within the blower casing. The impeller is designed to impart a whirling or motion to the air in the blower.

IV. PROJECT OBJECTIVE

- To propose suitable material to reduce the weight of blower.
- Experimental validation of simulated results through fabrication and testing.

V. SELECTION OF BLOWER FOR APPLICATION

DIRECT/ COUPLING DRIVE - STANDARD "EMINENT SYSTEMS" BLOWER CATALOGUE						
Model No	Volume Flow		Static Pressure		Fan Speed	Motor Rating
r-1-	cfm	m3ph	in. of wc	mm of wc	rpm	HP
0005	500	850	2	51	1440	0.5
0010	1500	2549	2	51	1440	0.75
0015	2000	3398	2	51	1440	1
0020	3000	5098	2	51	1440	1.5
0025	4000	6797	2	51	1440	2
0030	5000	8496	2	51	1440	3
0035	8000	13594	2	51	1440	5
0040	11500	19541	2	51	1440	7.5
0045	500	850	3	76	1440	0.5
0050	900	1529	3	76	1440	0.75
0055	1200	2039	3	76	1440	1
0060	1900	3229	3	76	1440	1.5
0065	2500	4248	3	76	1440	2
0070	3750	6372	3	76	1440	3
0075	6000	10195	3	76	1440	5
0080	8000	13594	3	76	1440	7.5
0085	9500	16143	3	76	1440	10
0090	13000	22090	3	76	1440	12.5
0095	16000	27188	3	76	1440	15
0500	400	680	4	102	2850	0.5
0100	750	1274	4	102	1440	0.75
0505	750	1274	4	102	2850	0.75
0105	1000	1699	4	102	1440	1
0510	1000	1699	4	102	2850	1
0110	1500	2549	4	102	1440	1.5

Flow Discharge Q = 0.5 m3/s
Static Suction Pressure = -196.4 N/m2
Static Delivery Pressure = 784.8 N/m2
Static Pressure Gradient ΔPs = 981.2 Pa
Speed of impeller rotation N = 2800 rpm
Air Density = 1.165 kg/m3
Optimized number of blades z = 16
Outlet Blade Angle $\beta 2 = 90°$
Suction Temperature Ts = 30ºC = 303º K
Atmospheric Pressure Patm = 1.01325 x 105 Pa
Atmospheric Temperature Tatm = 30ºC = 303ºK

DESIGN OF IMPELLER:

- **Impeller eye and inlet duct size:**

Let inlet duct size be 10% higher than impeller eye size or impeller inlet diameter. This will make conical insertion of inlet duct and flow acceleration at impeller eye or inlet.

$$\therefore D_{duct} = 1.1D_{eye} = 1.1D_1$$

Assuming no loss during 90º turning from eye inlet to impeller inlet, the eye inlet velocity vector will remain same as absolute velocity vector at the entry of impeller.

$$i.e. V_{eye} = V1 = V_{ml}$$

Further let tangential velocity component be 10% higher than axial velocity component for better induction of flow. So, Inlet tips velocity.

$$U1 = 1.1V_1 = 1.1V_{ml}$$

$$Discharge, Q = \frac{\pi}{4} D_{eye}^2 X V_1$$

$$Q = \frac{\pi}{4} X (D_1)^2 X V_1$$

$$V_1 = \frac{4Q}{\pi X D_1^2}$$

$$U_1 = 1.1V_1$$

$$\therefore \frac{\pi DN}{60} = 1.1 \frac{4Q}{\pi D_1^2}$$

Here Q=0.5 m^3/S and speed of impeller rotation N=2800 rpm,

∴ Impeller Inlet Diameter

D_1=0.168 m = D_{eye}

∴ Peripheral speed at inlet

$U_1 = \frac{\pi D_1 N}{60} = 24.63 m/s$

$V_1 = 22.45\, m/s = V_{m1} = V_{eye}$

$D_{duct} = 1.1D_1 = 1.1X0.168 = 0.185m = 185mm$

$W_1 = \sqrt{U_1^2 + V_1^2} = \sqrt{24.63^2 + 22.45^2}$ = 33.37m/s

- Impeller inlet blade angle

$$\tan \beta_1 = \frac{V_1}{U_1} = \frac{22.45}{24.63}$$

$$\beta_1 = 42.35^o$$

- Impeller width at inlet

Here Z=16 and assumed blade thickness

t = 2 mm

- Impeller outlet parameters:

$$TheFanPower = \Delta PXQ = 981.2\, X\, 0.5 = 490.6\, W$$

Considering 10% extra to accommodate flow recirculation and impeller exit Hydraulic losses.

So 1.1 X the fan power = 1.1X490.6 =539.66W

Power, P = $\dot{m}XW_s$

$$\therefore \text{Specificworkdone, Ws} = \frac{539.66}{1.165X0.5}$$

$$= 926.45 \text{ W/(kg/s)}$$

$$\text{Eulerpower} = \text{m}\dot{\text{V}}_{\text{u2}}\text{U}_2$$

$TakingV_{u2}$

$= 0.8U_2$ $(Assumingslipfactor0.8\, forradialblades)$

$$539.66 = 1.165\, X\, 0.5\, X\, 0.8U_2 X U_2$$

$$\text{U}_2 = 34.03 \text{ m/s}$$

$$\text{V}_{\text{u2}} = 0.8\text{X}34.03 = 27.22 \text{ m/s}$$

$$\text{and}\, \text{U}_2 = \frac{\pi \text{D}_2 \text{N}}{60}$$

$$\therefore \text{D}_2 = 0.232\text{m}$$

Taking width of blade at inlet = outlet blade width

$$\therefore \text{b}_1 = \text{b}_2$$

$$\text{Q} = [\pi \text{D}_2 - \text{zt}]\text{ X b}_2 \text{ X V}_{\text{m2}}$$

$$0.5 = [(\pi \text{ X } 0.232) - (16 \text{ X } 2 \text{ X } 10^{-3})] \text{ X } 0.045 \text{ X V}_{\text{m2}}$$

$$\text{V}_{\text{m2}} = 16\text{m/s}$$

$$\text{W}_{\text{U2}} = 0.2\text{U}_2 = 6.81\text{m/s}$$

$$\text{W}_2' = \sqrt{\text{W}_{\text{U2}}^2 + \text{Vm}_2^2} = \sqrt{(0.2\text{U}_2)^2 + \text{Vm}_2^2}$$

$$\text{W}_2' = \sqrt{6.81^2 + 16^2} = 17.39 \text{ m/s}$$

$$\text{V}_2' = \sqrt{\text{V}_{\text{U2}}^2 + \text{Vm}_2^2}$$

$$\text{V}_2' = \sqrt{27.22^2 + 16^2} = 31.58 \text{ m/s}$$

$$\tan \alpha_2' = \frac{Vm_2}{Vu_2} = \frac{16}{27.22} = 0.59$$

$$\alpha_2' = 30.45^o$$

DESIGN OF VOLUTE CASING

Analyzing steady flow energy equation at inlet and exit:

$$\frac{P_1}{\rho_1} + \frac{1}{2}V_1^2 + gz_1 + W_s = \frac{P_2}{\rho_1} + \frac{1}{2}V_4^2 + gz_2$$

Neglecting potential difference,

$$V_4^2 = -\frac{2[P_2 - P_1]}{\rho} + V_1^2 + 2W_s$$

$$= \frac{-2(981.2)}{1.165} + 22.45^2 + 2(926.45)$$

Hence casing outlet velocity, V4 = 25.93 m/s

$$Q = A_v V_4$$

Where Av is Exit area of volute casing

= Av=bv (r4 – r3)

Allowing for 5 mm radial clearance between impeller and volute tongue,

$$r_3 = \frac{D_2}{2} + 5 = \frac{232}{2} + 5 = 121mm$$

$$D_3 = 2 \text{ X } 121 = 242mm$$

Width of volute casing (b_v) is normally 2 to 3 times b_1

Let us take it 2.5 times, Hence

$$b_v = 2.5 \text{ X } b_2 = 2.5 \text{ X } 45 = 112.5 \text{ mm}$$

$$Q = A_v V_4$$

$$0.5 = b_v(r_4 - r_3) \text{ X } 25.93$$

$$0.5 = 0.1125(r_4 - 0.121) \text{ X } 25.93$$

$$r_4 = 292mm$$

Now incremental volute angle with respect to increase in radius of casing

$$r_\theta = r_3 + \frac{\theta}{360} \text{ X } \Delta r$$

$$\Delta r = (r_{at360^o} - r_{at0^o})$$

$$r_\theta = 0.121 + \frac{\theta}{360}(0.292 - 0.121)$$

The calculated volute radiuses at different volute angles are given in below table:

θ in Degree	Volute Radius r in m
0	0.121
60	0.150
120	0.178
180	0.207
240	0.235
300	0.264
360	0.292

Radius of Volute Tongue

r_t = = 1.075r2 = 1.075 X 0.116 = 0.125m

Angle of Volute Tongue

$$\theta_t = \frac{132 \log_{10}\left(\frac{r_t}{r_2}\right)}{\tan \alpha_2}$$

$$\theta_t = \frac{132 \; X \; 0.0325}{0.59} = 7.26^o$$

So angle of volute tongue = 7.26^o

VOLUTE CASING:

TableI. – Parameter of Casing

Parameters	**Symbol**	**Unit**	**Value**
Width Of Casing	b_v	mm	112.5
Outlet Velocity Of Casing	V_4	m/s	25.93
Scroll Radius at Inlet	r_3	mm	121

Scroll Radius at Outlet	r_4	mm	292
Scroll Height	Hs	mm	172
Volute Tongue Angle	θ_t	Degree	7
Radius of Tongue	r_t	mm	125

Existing Blower analysis

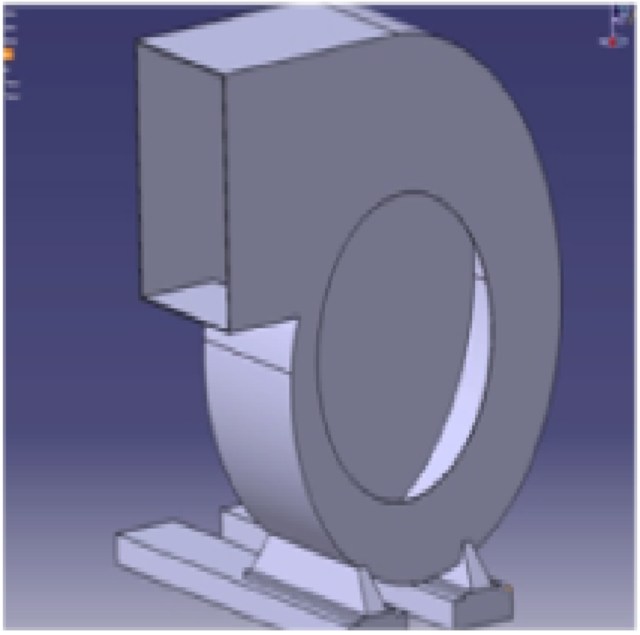

Fig III. : Blower housing CAD model view

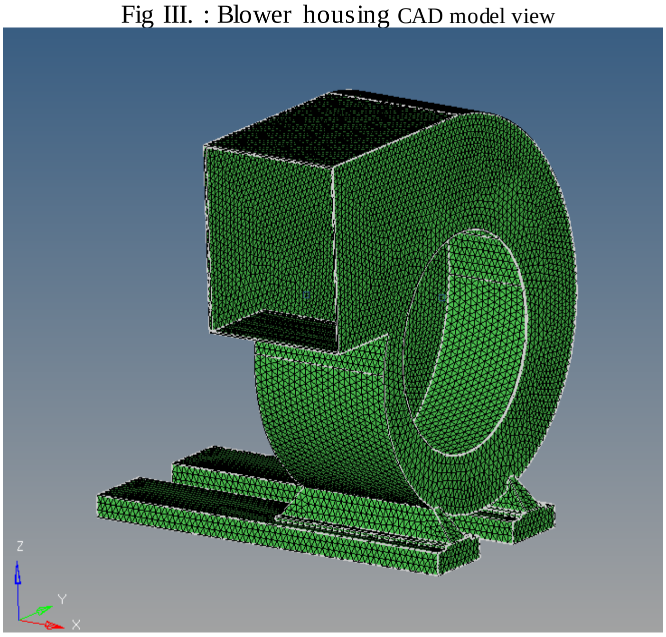

Fig IV. : Meshed model of blower housing

No. of nodes- 21822
No. of Elements- 69766
Size of Element- 4mm

Structural Properties (MS):

Table II. – Material properties of steel

Property	Value
Young's Modulus, E	2.1×10^5 MPa
Poisson's Ratio ,ν	0.3
Density, ρ	7850 kg/m^3
Yield Stress, σ_{yield}	250 MPa
Ultimate Tensile Stress, σ_{uts}	390 MPa

The advantages of using steel are-

- Best steel alloys are very strong.
- Best stiffness overall.
- Long-lasting.
- Air-hardened alloys make ultra-high strength affordable.

The limitations of steel are-

- Can be heavy - not the materials for big, light frames.
- Rust-prone.

- **BOUNDARY CONDITIONS:**

After meshing is completed we apply boundary conditions. These boundary conditions are the reference points for calculating the results of analysis. In short we here go for the preparation of deck. Here we apply define and apply various loads. Different load steps are created which are to be applied during analysis. Here surrounding effect is been taken into consideration while applying loads. Elements are defined by their properties. Material properties such as density, modulus of elasticity, Poisson's ratio etc. is assigned to the elements. Here proper arrangements are made so that we can run the analysis in solver software. After the completion of process model is exported to the solver.

- Motor weight = 2Kg
- Blower weight = 20N
- Motor torque =Motor torque can be calculated as:
- Power = 2 π n T/60
- P = 981 W
- N = 2800 rpm
- ∴ T = 3.340 N/m

Following are the results displayed for stress and deformation:

Von-mises stress for blower housing:

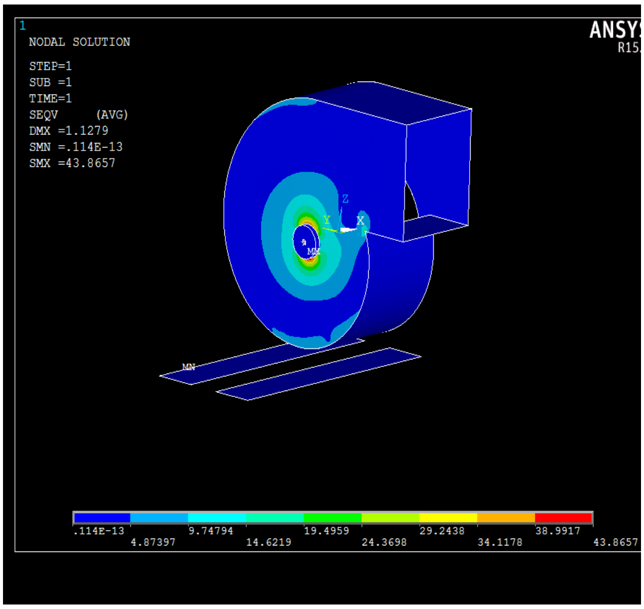

Fig V.: von-mises stress for blower housing

Stress value for blower housing is 43.86 N/mm^2 which is well below the critical value. Hence, design is safe.

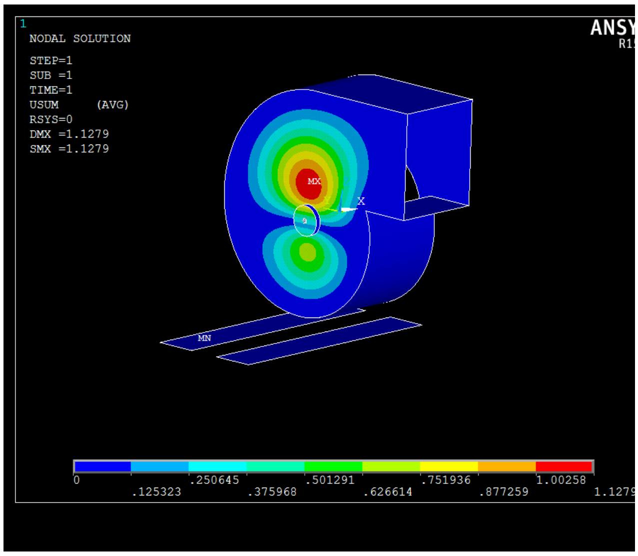

Fig VI. : Displacement result for blower housing

From fig, deformation for blower housing is 1.12 mm.

VI. ANALYSIS WITH COMPOSITE MATERIAL

Material properties for glass fiber:

Table III. :Material properties of Glass fibers

Property	Value
E_1	40 GPa
E_2	6 GPa
E_3	40 GPa
Poisson's Ratio ,ν	0.24
G_{xy}	15 GPa
G_{yz}	2.3 GPa
G_{zx}	15 GPa
Density, ρ	2000 kg/m^3

Following are the results displayed for stress and deformation:

Von-mises stress for blower housing:

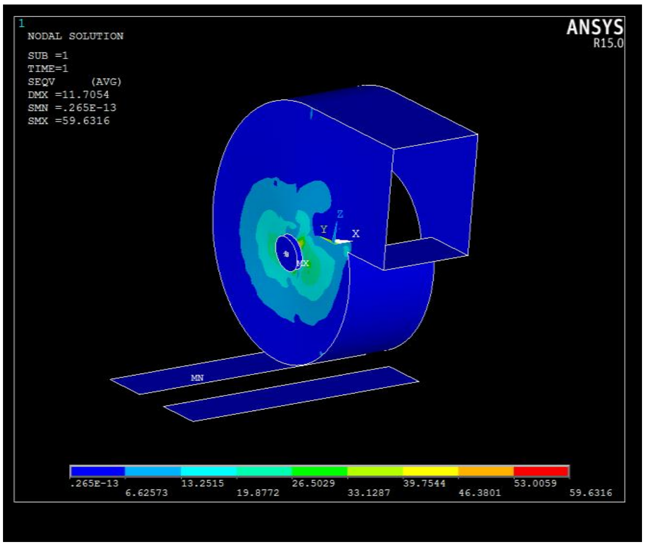

Fig. VII: von-mises stress for blower housing (Glass fiber)

Stress value for blower housing is 59.63 N/mm^2 which is well below the critical value. Hence, design is safe.

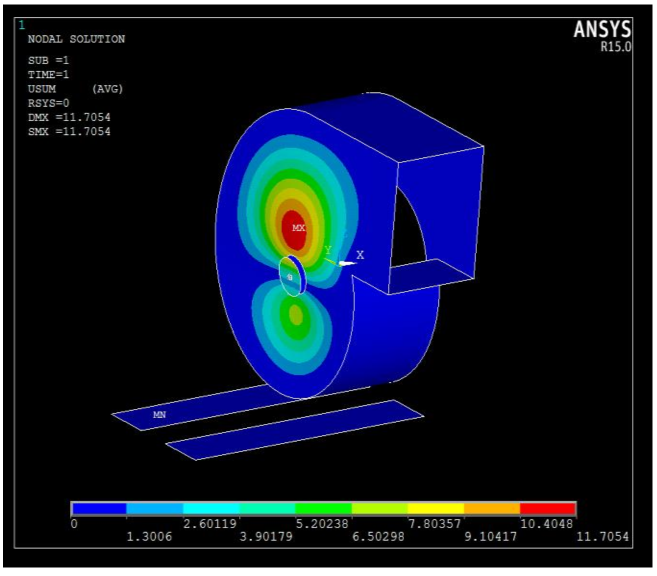

Fig. VIII: Displacement result for blower housing (Glass fiber)

From fig, deformation for blower housing is 11.7 mm.

VII. ANALYSIS WITH ALUMINUM MATERIAL

Material properties for Aluminum:

Table IV. :Material properties of **Aluminum:**

Property	**Value**
E_1	69GPa
Poisson's Ratio ,ν	0.32
Density, ρ	2000 kg/m^3

Von-mises stress for blower housing:

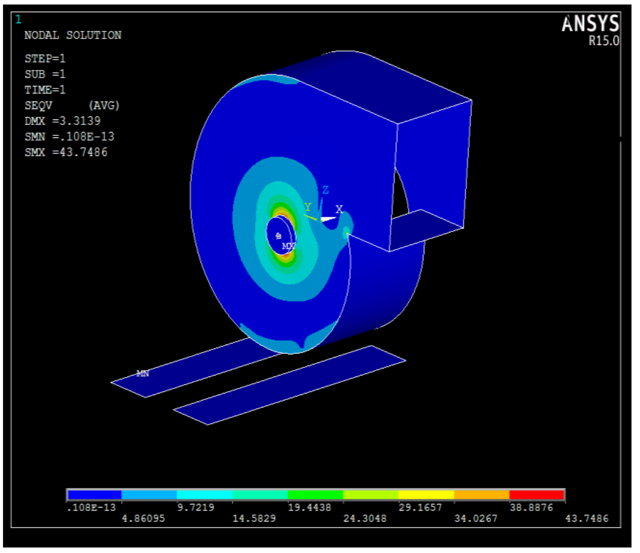

Fig IX: von-mises stress for blower housing (Aluminum)

Stress value for blower housing is 43.74 N/mm^2 which is well below the critical value. Hence, design is safe.

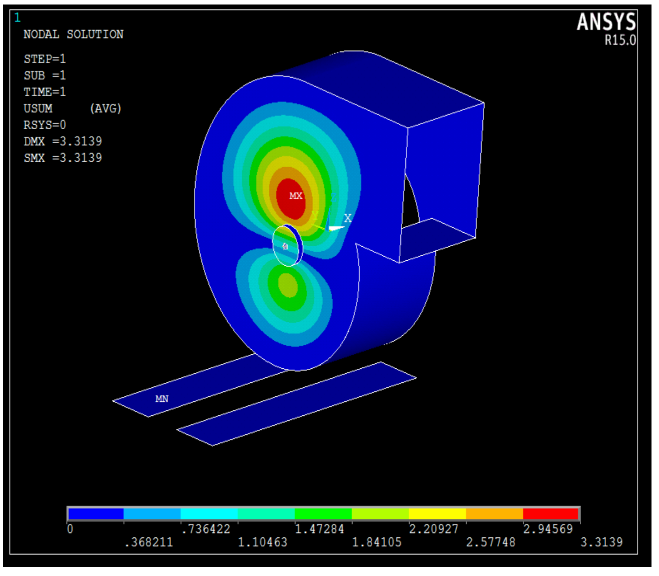

Fig.X: Displacement result for blower housing (Aluminum)

From fig, deformation for blower housing is 3.31 mm.

Comparison:

The analysis of blower housing has been done for all the three materials viz. steel, aluminum alloy 6063, glass fiber. The comparison of properties and analysis results is shown in table A and B respectively.

Table V – Comparison of material properties

S.No.	Material	Young's Modulus E	Poisson's Ratio υ	Density ρ	Yield Strength	Ultimate Tensile Stress σ
1.	**Steel**	210 GPa	0.3	7850 kg/m^3	290 MPa	390 MPa
2.	**Glass Fiber**	E1-40 GPa E2-6 Gpa	0.24	2000 kg/m^3	1850 MPa	2000 MPa
3.	**Aluminum**	69 Gpa	0.32	2800 kg/m^3	270 Mps	310 Mpa

Table VI – Comparison of analysis results

Sr.No	Material	Max. Stress	Max. Displacement
1.	**Steel**	43.86 MPa	1.12 mm
2.	**Glass Fiber**	59.63 MPa	11.7 mm
3.	**Aluminum**	43.74 Mpa	3.13 mm

It is also observed that all the analysis have stress values less than their respective permissible yield stress values. So the design is safe.From analysis results and comparison of properties of all the materials, it is found that glass fiber is the material which is having the least density; also it is easily available and cheap as compared to other alternate materials. Also machining cost for glass fiber is less. Hence it is the best suited alternate material for blower housing and is expected to perform better with satisfying amount of weight reduction.

Closure

For the dissertation work this chapter discusses about finite element based weight optimization of blower housing and thus helped in finding out the appropriate alternate material to steel through which a prototype can be fabricated. This chapter includes the study of Glass fiber as alternative materials, its properties, advantages and limitations. These materials have been compared with conventional steel and the analysis is done. From the analysis results glass fiber is found out to be the most suited alternate material for blower housing,this result we are validating by Experimental validation of simulated results through fabrication and testing.

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