Design And Structural Analysis Of Heavy Duty Industrial Blower Casing Using FEA

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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. 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 casing. 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. First design calculation for blower casing are done and then geometry editing is performed on given model. After that meshing is done and frictional contacts are given to the model.

In actual analysis stage modal analysis, shock analysis and fatigue analysis are performed in this project with weld material qualification and parent material qualification. In total 13 cases of analysis [modal analysis (1 case), shock analysis (6 cases), fatigue analysis (6 cases)] are performed on the model and results are studied for this 13 cases. Campbell diagram and Goodman diagram are drawn based on results with calculation of endurance strength. Safe design of blower casing is concluded after studying all the results.

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.

Keywords- Centrifugal blower, Structural analysis, Modal analysis, Shock analysis, Fatigue analysis and Optimization.

I. INTRODUCTION

1.1 Introduction to Blower

Blower is a very important fluid machine. It which has characteristics of energy transfer between continuous stream of fluid and rotating element about a axis. Blower is a head generating machine which employs the dynamic action of a rotating elements. Blowers and Compressors are pressure-increasing machines.[3] In all these, the fluid enters axially and is discharged by the rotor into a static collector system casing and then into a discharge pipe. Main components of Blower are impeller which is having rotary motion, where energy is transfer and followed by stationary part casing, in which energy transformation takes place. Casing decides the size and pressure rise in the system.

Blowers are used where large volumes of gas at low pressure are required. They generally operate at low speeds & pressure ratios. In pumps & fans the fluid is considered incompressible while in the compressor and blower there is usually a considerable density change. Blower efficiencies differ from designs of each other. Blower has 2 categories : centrifugal flow blower and axial flow blower. In centrifugal flow, airflow change direction 2 times - first when entering and second when leaving. They are also used to produce negative pressures for vacuum systems used in industries.[6] Centrifugal blowers look more like centrifugal pumps than fans. In multi-stage blowers, air is accelerated when it passes through each impeller. In single-stage blower, air does not take many turns, and hence it is more efficient. Air flow in the blowers usually is subject matter of research as blowers command greater share of applications in various sectors of industries. With relatively poor energy scene in developing economy, greater emphasis is usually made on power intake requirements, efficiency etc. These parameters have direct influence on air flow in the blower system both in the rotor and the volute casing. It is due to this reason that, what follows is a brief description of air flow in various parts of a Centrifugal Blower.

1.2 Introduction to FEA

Definition of FEM is hidden in its words itself. Basic theme is to make calculations at only limited (Finite) number of points and then interpolate the results for entire domain (surface or volume).

Finite - Any continuous object has infinite degrees of freedom & it's just not possible to solve the problem in this format. Finite Element Method reduces degrees of freedom from Infinite to Finite with the help of discretization i.e. meshing (nodes & elements).

Element - Calculations are made at limited number of points known as nodes. Entity joining nodes and forming a specific shape such as quadrilateral or triangular etc. is known as Element. To get value of variable (Say displacement) anywhere in between the calculation points, interpolation function (as per the shape of element) is used.

Method-There are 3 methods to solve any engineering problem. Finite element analysis belongs to numerical method category.

FEM is the most popular numerical method.

Applications - Linear, Nonlinear, Buckling, Thermal, Dynamic & Fatigue analysis. Finite Element Method (FEM) and finite Element Analysis (FEA) both are one & the same.

II. PROBLEM STATEMENT AND OBJECTIVE

2.1 Problem Statement

To design heavy duty industrial blower and analyzing it by using FEA for safe design

In **Modal**, **Shock** and **Fatigue criteria**.

2.2 Objectives

1. Design of blower casing
2. Modal analysis (1 case)
3. Steady state structural analysis (non linear)
4. Shock analysis \ Proof test (6 cases)
5. parent material qualification
6. welding qualification
7. Fatigue analysis (6 cases)
    a. parent material qualification
    b. Welding qualification
8. To draw camber diagram
9. To draw Goodman Diagram
10. Optimization

3. DESIGN PROCEDURE

3.1. Input Parameters for Centrifugal Blower Design

The input parameters for the design of centrifugal blower casing are described below:

Flow Discharge Q = $6m^3/s$
Static Suction Pressure = -196.4 N/m2
Static Delivery Pressure = 784.8 N/m2
Static Pressure Gradient ΔPs = 7611.261 Pa
Speed of impeller rotation N = 3105 rpm
Air Density = 1.165 kg/m3
Optimized number of blades z = 11
Outlet Blade Angle β2 = 90°
Suction Temperature Ts = 30 ^{0}C = 303 K
Atmospheric Pressure Patm= 1.01325 x 10^5 Pa
Atmospheric Temperature Tatm= 30^0C = 303K

3.2. 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.

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

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.

$$\therefore V_{eye} = V_1$$

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

$$\therefore \quad U_1 = 1.1V_1$$

Discharge, $Q = \frac{\pi}{4} * (D_1)^2 * V_1$

$V_1 = \frac{4Q}{\pi * D1^2}$

$U_1 = \frac{\pi D1N}{60} = 1.1V_1$

$\frac{\pi D1N}{60} = 1.1\frac{4Q}{\pi * D1^2}$

Here Q=6 m3/s and speed of impeller rotation N=3105 rpm,

∴Impeller Inlet Diameter

D1=0.370 m = Deye

∴Peripheral speed at inlet

$U_1 = \frac{\pi D1N}{60} = 60.15$ m/s = $1.1V_1$

$V_1 = 54.68$ m/s

$\therefore D_{duct} = 1.1$ $\mathbf{D_1 = 0.407}$ **m**

3.3. Impeller width at inlet

Here Z=11 and assumed blade thickness t = 9 mm

$Q=[\pi D_1 - z\ t] * b_1 * V_1$

$6 = [\pi\ 0.370 - 11*0.009] * b_1 * 54.68$

$\mathbf{b_1 = 0.103}$ **m = 103 mm**

3.4. Impeller outlet parameters

The Fan Power =ΔP * Q = 7611.261 * 6 = 45667.56 W
Considering 10% extra to accommodate flow recirculation and impeller exit hydraulic losses.

So, 1.1* fan power = 1.1 * 45667.56 = 49221.66 W

Power, P = $\dot{m}$ * Ws

Specific Work Done, Ws = $\frac{49221.66}{1.165*6}$ = 7186.19 W/(kg/s)

Euler Power = $\dot{m} Vu_2 U_2$

$Vu_2 = 0.8U_2$(Assuming slip factor = 0.8 for radial blades)

$\therefore 49221.66 = 1.165 * 6 * 0.8U_2 * U_2$

$U_2 = 64.78$ m/s

$U_2 = \frac{\pi D2N}{60}$

∴Impeller outlet diameter, $\mathbf{D_2 = 0.583}$ **m = 583 mm**

3.5. Design of Volute Casing

Analyzing steady flow energy equation at inlet and exit:

$$\frac{P1}{\rho 1} + 0.5\,(V_1)^2 + gZ_1 + Ws = \frac{P2}{\rho 2} + 0.5\,(V_2)^2 + gZ_2$$

Neglecting potential difference,

$$V_4^2 = \frac{-2*\Delta P}{\rho f} + V_1^2 + 2Ws$$

$$= \frac{-2*7611.261}{1.165} + 54.68^2 + 2(7186.19)$$

Hence casing outlet velocity, $V_4 = 65.546$ m/s

Let, $Q = A_v * V_4$

Where Av is Exit area of volute casing = $A_v = b_v * (r_4 - r_3)$

Width of volute casing (bv) is normally 2 to 3 times b1

Let us take it 3 times. Hence

bv= 3*b₁= 3*103 = 309 mm

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

$r_3 = r_2 + 5 = 297$ mm

Scroll inlet diameter, $\mathbf{D_3 = 594}$ **mm**

$\therefore Q = A_v * V_4$

$\therefore 6 = 0.309*(r_4 - 0.297)*65.546$

$r_4 = 0.593$ m = 593 mm

Scroll outlet diameter, $\mathbf{D_4 = 1186}$ **mm**

IV. MODAL ANALYSIS

Modal analysis is a technique to study the dynamic characteristics of a structure under vibrational excitation. Natural frequencies, mode shapes and mode vectors of a structure can be determined using modal analysis. Modal analysis allows the design to avoid resonant vibrations or to vibrate at a specified frequency and gives engineers an idea of how the design will respond to different types of dynamic loads. The crankshaft of an engine is one such structure whose dynamic characteristics can be better studied by modal analysis.

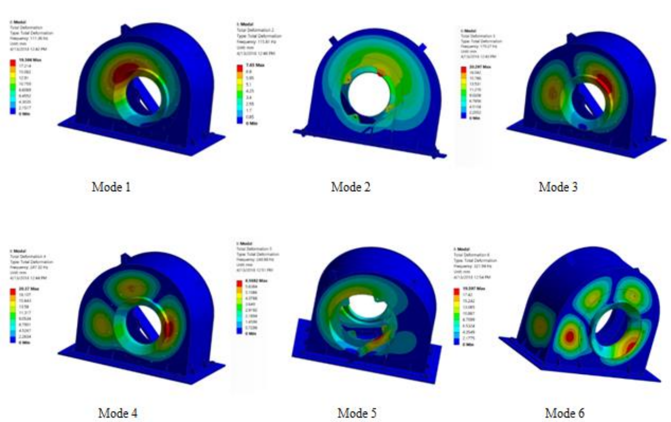
4.1 Methodology For Modal Analysis and Campbell Diagram

1) CAD geometry is imported in **Ansys 16.0** software and edited in design modeler as per the requirement of good meshing.(**Static structural** module is used).
2) Dead weight of Motor is added as point load in the blower model. **Dead weight of motor is 110.78 kg.**
3) Contacts are given between welded and bolted region.
   a) For **welded** region, contact type is given as **Bonded**.
   b) For **bolted** region,
      1] Between **Nut and Bolt**, Contact type is given as **Bonded**.
      2] Between **Nut and Plate**, between **Bolt and Plate** and between **Bolt and Plates,** Contact typeis given as **Frictional**.
4) Model is meshed by using **Hexa and Tetra** elements. Number of Elements are**9,00,949**. Then Quality criteria is checked for elements for getting good results.
   a) **Aspect Ratio** is kept as **40**.
   b) **Jacobian Ratio** is kept as **40**.
   c) **Skewness** is kept as **0.99**.
5) Boundary condition is given for the model. **Fixed support** is applied for **Base plate.**
6) Model is solved.
7) Possible Frequencies for vibration are displayed on the screen

Mode	Frequency [Hz]
1.	111.36
2.	115.81
3.	179.27
4.	247.32
5.	248.88
6.	321.94

Table 4.1. Frequencies

8) Results are evaluated for these frequencies and deformation is seen for each frequency



4.2 Deformation for Mode Shapes

9) These frequencies should not lie below 55 Hz to avoid resonance. Then Campbell diagram is plotted.

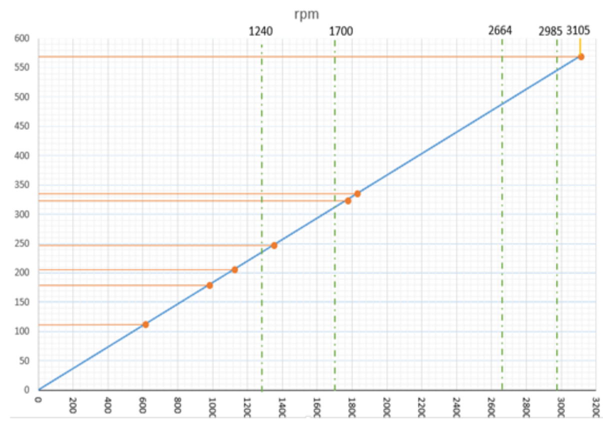


Fig. 4.3. Campbell diagram

4.2. Results and conclusion:-

a) The frequencies for modal analysis are **111.36, 115.81, 179.27, 247.32, 248.88** and **321.94 Hz.** These frequencies are above the motor frequency i.e. **55 Hz**. Therefore, resonance will not occur and **design is safe for modal analysis**.
b) The intersection points for horizontal lines with inclined line are observed. These points should not coincide with intersection points of vertical lines with inclined line for safer design. As per the results the points are not coinciding. Therefore, **design is safe**.

V. SHOCK ANALYSIS

Mechanical shocks are a very common occurrence in everyday life; examples of shockloading include explosions, car hitting a pothole, reciprocating engine fuel explosions inside cylinders, airplane landing, bolted joints suddenly opening and closing with an impact, high winds can be a source of step loading, earthquakes, drop impact during handling, g forces generated for internal components, high speed fluid entry, etc. When the load increases to its maximum value over five or six natural periods, it is a quasi-static load. When it does so over a fraction of the period, it is a shock loading. Broader definitions exist, where anything with up to two periods duration is a shock loading .

5.1 Parent material qualification:-

5.1.1 Loading Conditions

a) Loading condition in X-direction, **acceleration** is given in X-direction as **1.5g.** i.e. **14,709 N**.

b) Loading condition in Y-direction, **acceleration** is given in Y-direction as **1.8g.** i.e. **17,650 N**.

c) Loading condition in Z-direction, **acceleration** is given in Z-direction as **1.5g.** i.e. **29,418 N**.

d) Boundary condition is given for the model. **Fixed support** is applied for **Base plate.**

e) **Pressure** is applied on inner side of blower casing as **6.384e-3 Mpa.**

f) Model is solved for **Von-mises stress.**

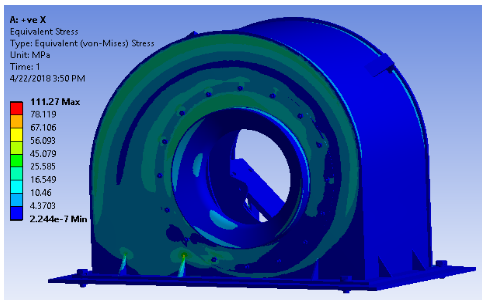


Fig 5.1 Equivalent stress in X-Direction

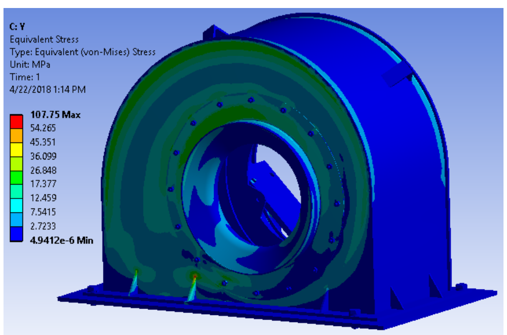


Fig 5.2 Equivalent Stress in Y-Direction

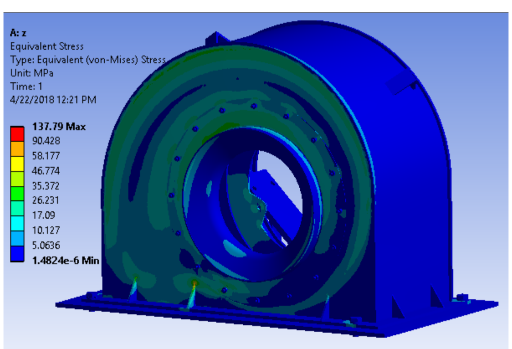


Fig 5.3 Equivalent Stress in Z-Direction

5.1.2 Results and conclusion :-

1] Maximum equivalent von mises stress for parent material in **X direction** is **111.27 Mpa.**

2] Maximum equivalent von mises stress for parent material in **-X direction** is **105.83 Mpa.**

3] Maximum equivalent von mises stress for parent material in **Y direction** is **109.35 Mpa.**

4] Maximum equivalent von mises stress for parent material in **-Y direction** is **107.75 Mpa.**

5] Maximum equivalent von mises stress for parent material in **Z direction** is **137.79 Mpa.**

6] Maximum equivalent von mises stress for parent material in **-Z direction**is **111Mpa.**

7] As all results have values **below 150 Mpa**. So FOS for parent material is greater than 2. Which is good, so **design is safefor parent material qualification in shock analysis.**

5.2 Weld material qualification:-

5.2.1 Methodology and Procedure :-

1. Lines are drawn at **0.8 times** the **miniumum thickness of two welding components**. This line is known as **1 toe weld length** line. This line is used for stress evaluation in weld material qualification in shock analysis.
2. Contacts are given between welded and bolted region.

a) For **welded** region, contact type is given as **Bonded**.

b) For **bolted** region,

  1] Between **Nut and Bolt**, Contact type is given as **Bonded**.

  2] Between **Nut and Plate**, between **Bolt and Plate** and between **Bolt and Plates,** Contact typeis given as **Frictional**.

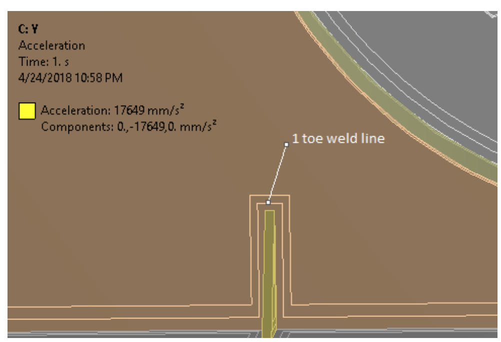


Fig. 5.4 . One toe weld line

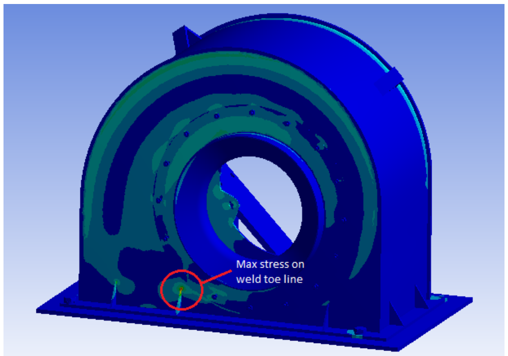


Fig. 5.5. Maximum equivalent von mises stress on 1 toe weld line

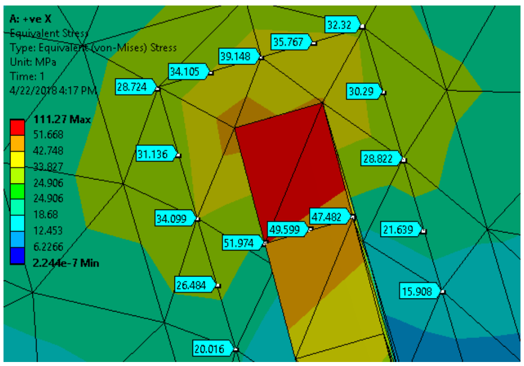


Fig 5.6 Stress on 1 toe weld line for X-Direction

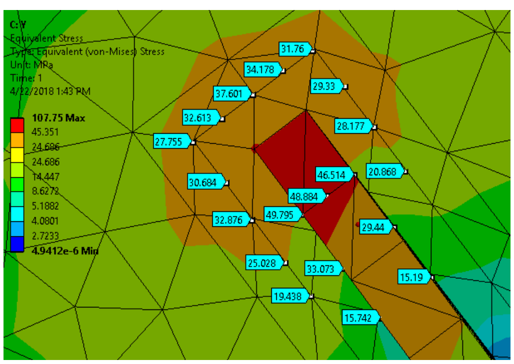


Fig 5.7 Stress on 1 toe weld line for Y-Direction

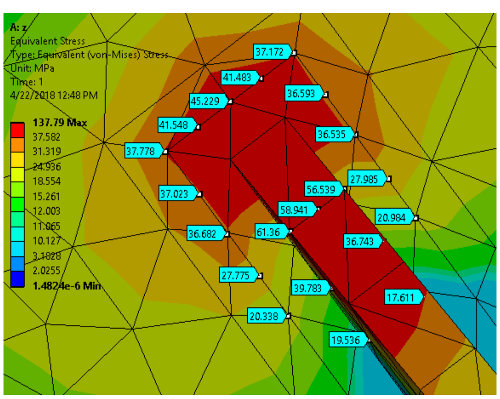


Fig 5.8 Stress on 1 toe weld line for Z-Direction

5.2.2 Results and conclusion:-

1) Maximum equivalent von mises stress on 1 toe weld line for all cases given above are **51.974 Mpa,50.373 Mpa,49.795 Mpa,50.812 Mpa,61.36 Mpa,41.151 Mpa.**
2) As maximum equivalent von mises stress on 1 toe weld line is below **50% of tensile yield strength** of the parent material. (0.5*300) i.e. below **150 Mpa**. So **design is safe for weld material qualification in shock analysis.**

VI. FATIGUE ANALYSIS

While many parts may work well initially, they often fail in service due to fatigue failure caused by repeated cyclic loading. Characterizing the capability of a material to survive the many cycles a component may experience during its lifetime is the aim of fatigue analysis. In a general sense, Fatigue Analysis has three main methods, Strain Life, Stress Life, and Fracture Mechanics; the first two being available within the ANSYS Fatigue Module

There are 5 common input decision topics upon which your fatigue results are dependent upon.

These fatigue decisions are grouped into the types listed below:

- Fatigue Analysis Type
- Loading Type
- Mean Stress Effects
- Multi axial Stress Correction
- Fatigue Modification Factor

6.1. Parent material qualification:-

6.1.1. Methodology

1) CAD geometry is imported in **Ansys 16.0** software and edited in design modeler as per the requirement of good meshing. (**Static Structural** module is selected).
2) Dead weight of Motor is added as point load in the blower model. **Dead weight of motor is 110.78 kg.**
3) Contacts are given between welded and bolted region.
a) For **welded** region, contact type is given as **Bonded**.
b) For **bolted** region,

1] Between **Nut and Bolt**, Contact type is given as **Bonded**.
2] Between **Nut and Plate**, between **Bolt and Plate** and between **Bolt and Plates,**Contact typeis given as **Frictional**.

4) Model is meshed by using **Hexa and Tetra** elements. Number of Elements are **5,74,630**. Then Quality criteria is checked for elements for getting good results.

6.1.2 Loading Conditions :-

a) Loading condition in X-direction, **acceleration** is given in X-direction as **0.5g.**

i.e. **4903 N**.

b) Loading condition in Y-direction, **acceleration** is given in Y-direction as **0.5g.**

i.e. **4903 N**.

c) Loading condition in Z-direction, **acceleration** is given in Z-direction as **1.5g.**

i.e. **4903 N**.

d) Boundary condition is given for the model. **Fixed support** is applied for **Base plate.**

e) **Pressure** is applied on inner side of blower casing as **6.384e-3 Mpa.**

Model is solved for **Von-mises stress**

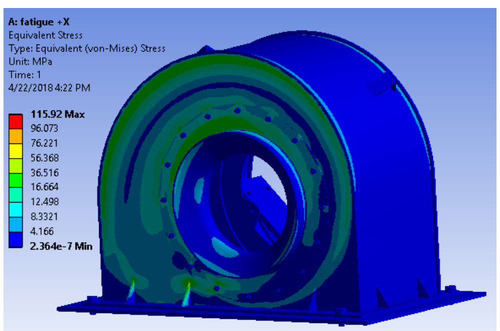


Fig 6.1 Fatigue stress in X-Direction

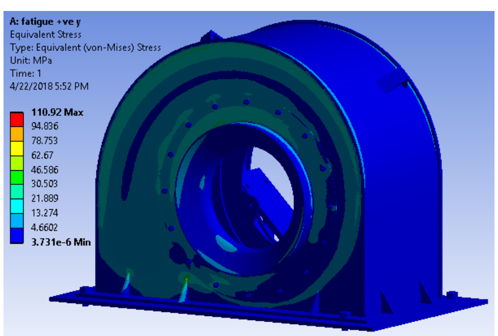


Fig 6.2 Fatigue stress in Y-Direction

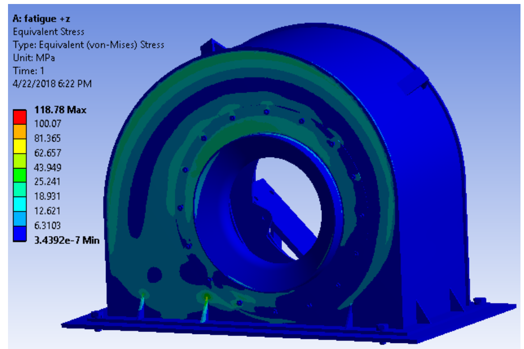


Fig 6.3 Fatigue stress in Z-Direction

6.1.3 Procedure for calculating mean and alternating stress:-

a) For finding mean and alternating stress, some values were found out from the software.

b) For X, find **maximum principle** stress in **+ve X** direction and **mark the node for that maximum value.**

c) Now select same node in **–ve X** direction model and find the **minimum principle stress on that node.**

Mean stress = $\frac{max.principle\ stress(p1) + min.princple\ stress(p3)}{2}$

Alternating stress = $\frac{max.principle\ stress(p1) - min.princple\ stress(p3)}{2}$

Ex. For +ve X direction (sample calculation),

Mean stress = $\frac{151.73 + 15.098}{2}$ **= 83.414 Mpa.**

Alternating stress = $\frac{151.73 - 15.098}{2}$ **= 68.316 Mpa.**

d) Follow above procedure for all the remaining cases

like**–ve X, +ve Y, -ve Y, +ve Z, -ve Z.**

Table 6.1. Values for +ve X and –ve X

Node A	P1 (+X) Mpa	P3 (-X) Mpa	MEAN	ALTERNATING
	151.73	15.098	83.414	68.316
Node B	P3 (+X)	P1 (-X)		
	-77.98	-6.6015	42.29	-35.6892
Node C	P1 (-X)	P3 (+X)		
	145.81	16.105	80.9575	64.8525
Node D	P3 (-X)	P1 (+X)		
	-76.691	-5.6813	41.186	-35.5048

Table 6.2. Values for +ve Y and –ve Y

Node A	P1 (+Y) Mpa	P3 (-Y) Mpa	MEAN	ALTERNATING
	147.31	15.186	81.248	66.062
Node B	P3 (+Y)	P1 (-Y)		
	-77.194	-6.5639	41.87	-35.315
Node C	P1 (-Y)	P3 (+Y)		
	146.72	15.248	80.984	65.736
Node D	P3 (-Y)	P1 (+Y)		
	-76.945	-6.6211	41.7830	-35.1619

Table 6.3. Values for +ve Z and –ve Z

Node A	P1 (+Z) Mpa	P3 (-Z) Mpa	MEAN	ALTERNATING
	155.5	14.731	85.1155	70.3845
Node B	P3 (+Z)	P1 (-Z)		
	-80.357	-6.3123	43.3346	-37.0223
Node C	P1 (-Z)	P3 (+Z)		
	142.15	16.497	79.3235	62.8265
Node D	P3 (-Z)	P1 (+Z)		
	-74.331	-5.9386	40.1348	-34.1962

6.1.4Goodman Diagram:-

A straight line joining endurance strength (S_e) on ordinate axis and ultimate tensile strength (S_{ut}) on abscissa is called as goodman diagram.

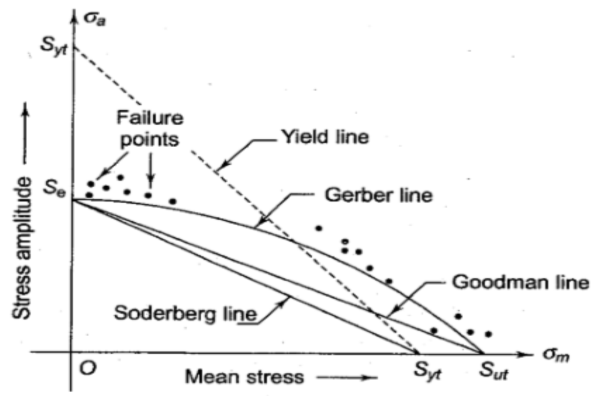


Fig 6.4 Goodman Diagram

a) **Fatigue life** is minimum for shaft, so find Endurance limit stress for shaft and check whether design is safe or not, by plotting **goodman diagram**.
b) Diameter of shaft (d) = 15 mm.

Surface finish factor (K_a) = 0.82
Size factor (K_b) = 0.85
Reliability factor (K_c) = 0.879 (90% reliability)
Stress concentration factor (K_d) = $\frac{1}{kf}$ = 1.
S_e' = endurance limit stress of a rotating beam specimen subjected to reversed bending stress (N/mm^2).
$S_e' = 0.5S_{ut} = 0.5 * 400 = 200\ N/mm^2$.
$S_e = K_a . K_b . K_c . K_d . S_e'$
$= (0.82)(0.85)(0.879)(1)(200)$
S_e= 122.53 N/mm².

d) drawgoodman diagram by taking **mean stress as abscissa** and **alternating stress as** ordinate.

e)mark**'Se' on ordinate** and '**S_{ut}'on abscissa** and join these two points.

f) Plot points by using mean and alternating stresses on goodman diagram.

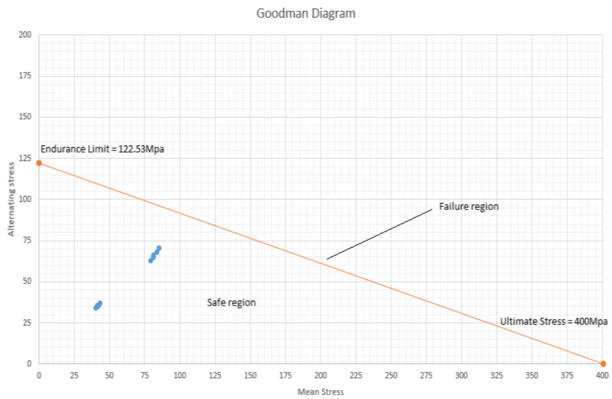


Fig 6.5 Goodman Diagram For Blower Casing

6.1.5 Results and Conclusion:-

a) All plotted points lie in the region, below the line joined by $S_e$andS_{ut}.
b) As all the points are in **safe region**, so **design is safe for fatigue analysis**.

6.2. Weld material qualification:-

6.2.1 Methodology

Weld Material Qualification for Fatigue Analysis is similar as Shock Analysis. We get results shown below –

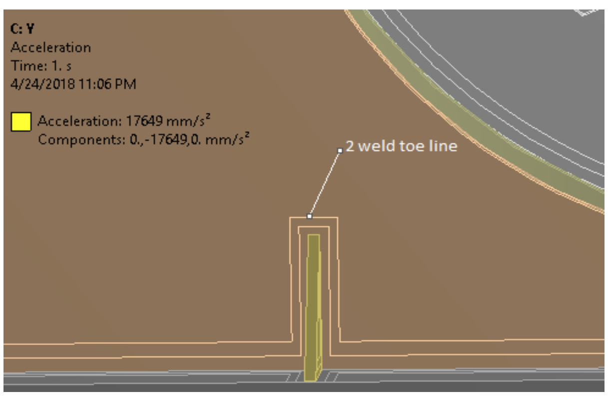


Fig. 6.6 2 weld toe line for Fatigue

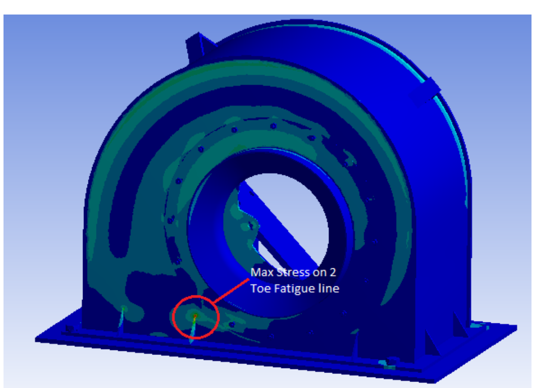


Fig. 6.7. Max stress location on 2 toe weld line

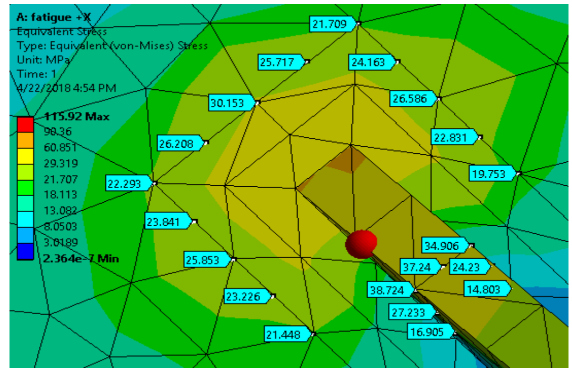


Fig 6.8 Stresses on 2 toe weld fatigue line X-Direction

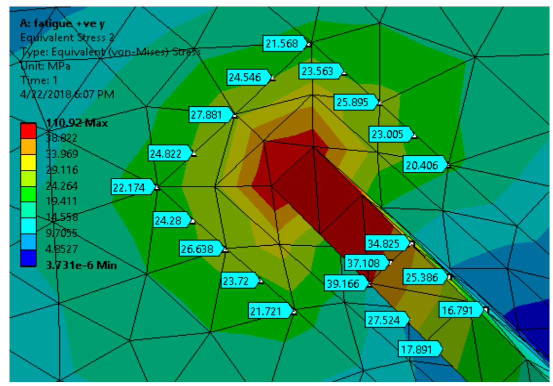


Fig 6.9 Stresses on 2 toe weld fatigue line for Y-Direction

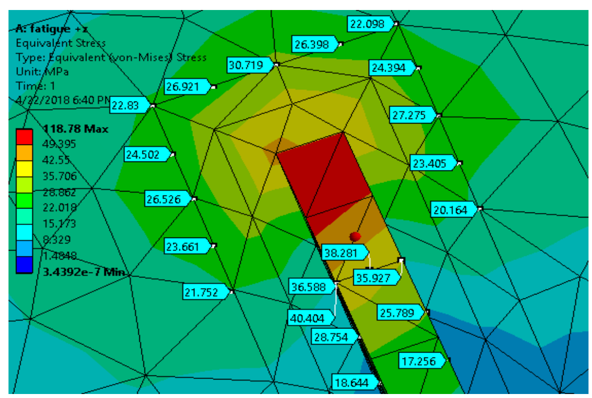


Fig 6.10 Stresses on 2 toe weld fatigue line For Z-Direction

6.2.2 Results and conclusion:-

1) Maximum equivalent von mises stress for parent material in **X-direction** is **38.72Mpa.**
2) Maximum equivalent von mises stress for parent material in **-X-direction** is **39.19Mpa.**
3) Maximum equivalent von mises stress for parent material in **Y-direction** is **39.16Mpa.**
4) Maximum equivalent von mises stress for parent material in **-Y-direction** is **39.61Mpa.**
5) Maximum equivalent von mises stress for parent material in **Z-direction** is **40.40Mpa.**

6) Maximum equivalent von mises stress for parent material in **-Z-direction**is **38.34Mpa.**
The stresses on fatigue weld line are less than **allowable stress 50 Mpa**. Hence the Blower Casing is safe in fatigue weld criteria.

VII. RESULTS AND CONCLUSION

(1) Three types of analysis are done. These are **Modal analysis, Shock analysis** and **Fatigue analysis**.
(2) For **modal analysis**, possible frequencies of vibration are found out for different modes.

a) The frequencies for modal analysis are **111.36, 115.81, 179.27, 247.32, 248.88** and **321.94 Hz.** These frequencies are above the motor frequency i.e. **55 Hz**.Therefore, resonance will not occur and **design is safe for modal analysis**.
b) By observing Campbell diagram, The intersection points for horizontal lines with inclined line are observed. These points should not coincide with intersection points of vertical lines with inclined line for safer design. As per the results the points are not coinciding. Therefore, **design is safe**.

(3) For **shock analysis**, maximum equivalent stress for parent and weld material are extracted.

a) Maximum equivalent von mises stress for parent material are **111.27 Mpa, 105.83 Mpa, 109.35 Mpa, 107.75 Mpa, 137.79 Mpa, 111Mpa.**
b) As all results have values **below 150 Mpa**. So **FOS** for parent material is **greater than 2**. Which is good, so **design is safefor parent material qualification in shock analysis.**
c) Maximum equivalent von mises stress on **1 toe weld line** for all cases given above are **51.974 Mpa,50.373 Mpa,49.795 Mpa,50.812 Mpa,61.36 Mpa,41.151 Mpa.**
d) As maximum equivalent von mises stress on **1 toe weld line** is below **50% of tensile yield strength** of the parent material. (0.5*400) i.e. below **200 Mpa**. So **design is safe for weld material qualification in shock analysis.**

(4) For **Fatigue analysis**, mean stress and alternating stress are found out for parent material. Also, Equivalent stress are extracted for weld material.

a) All plotted points lie in the region, below the line joined by $S_e$andS_{ut}in goodman diagram**.**
b) As all the points are in **safe region**, so **design is safe for fatigue analysis for parent material in fatigue analysis**.
c) Maximum equivalent von mises stress on **2 toe weld line** are **38.724 Mpa, 39.199 Mpa, 39.166 Mpa, 39.613 Mpa, 40.404 Mpa, 38.347 Mpa.**
c) all the values of stresses are **below 50 Mpa**, **Design is safe** for **weld in fatigue analysis**.

VIII. FUTURE SCOPE

a) The above results show that **FOS** for the blower casing is **greater than 2.** This shows that **blower casing is overdesigned.**
b) Future scope of the project is to remove the material from blower casing in efficient way using FEA.
c) So that weight of the casing is reduced without compromising safety.
d) **Optimization** in design.

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