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STATIC AND DYNAMIC ANALYSIS OF SHAFT (EN24) OF FOOT MOUNTING MOTOR USING FEA

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ABSTRACT-
This paper is about static and dynamic analysis of foot mounted motor which is also known as (EN24)type motor by using FEA software ANSYS 14.5 and modeling is done with help of CATIA V5 R20. This is the special problem which is observed by Laxmi Hydraulics Pvt. Ltd Solapur, Maharashtra, India. They have reported frequent failures of this type of Shaft. .

KEY WORDS: Static analysis, Foot Mounting Motor, CATIA V5, FEA

1. INTRODUCTION

In industry there are various requirement of motion for which we are using prime movers like Motor or Engines. Motors are widely used as prime movers as it gives uniform motion and control of speed and direction is easy as compared to engines. There are few types of motors which are classified on basis of mounting methods. These are as follows:

Types of motor
B3- Foot mounting motor
B 5- Flange mounting
B 14-Face mounting
B 34-Foot cum face mounting
B 35-Foot cum flange mounting

B3 type is a motor in which foot is bolted to wall on which whole motor is mounted. So that entire weight of a motor is going to act on foot. This is the case which is somewhat similar with cantilever beam of motor shaft. This type of motor is used at elevation places like overhead cranes, overhead machines etc. the R & D department of Laxmi Hydraulics departments has given the dimensions of foot of B3 motor.

Figure 1:Foot mounting B3 type motor

2. FEA ANALYSIS OF FOOT (B3) TYPE MOTOR

Analysis can be defined as the process of breaking a complex topic into finite i.e. smaller parts for better understanding of it. The technique has been applied in the study of mathematics and logic since before Aristotle (384–322 B.C.), though analysis as a formal concept is a relatively recent development.

Basically in mechanical engineering the word analysis which comprises two phases out of which one is modeling and another is analysis. In this paper the analysis is carried out with help of ANSYS 15 and Modeling is done with CATIA V5 R20.

2.1 MODELING OF B5 FOOT

Actual dimensions obtained by the firm are as follows:

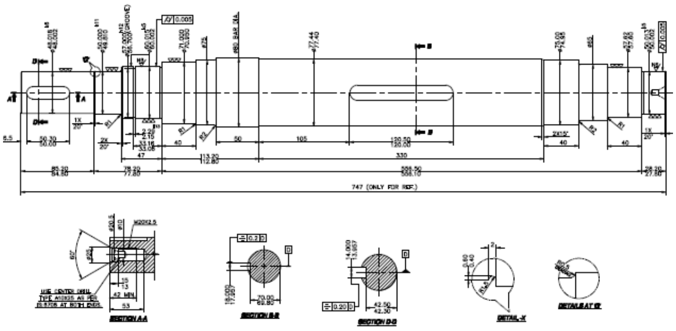

Figure 2:Actual dimensions of shaft of foot type motor

The modeling is done with CATIA V5 R20. All the dimensions are taken as per given by the firm.

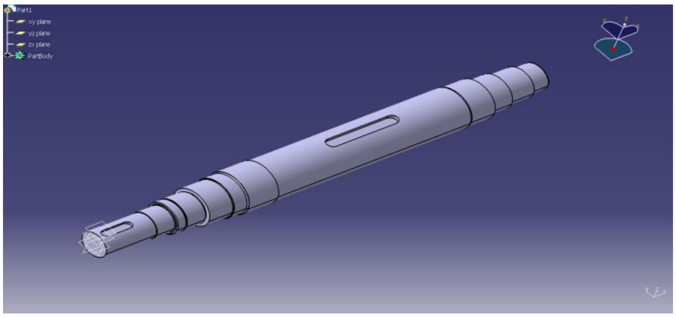

Figure 3: CATIA model of shaft of foot type motor

For analysis though the analysis tool is available in CATIA but ANSYS software gives more accurate and correct prediction about the failure of flange. The first phase is modeling the joint using CAD software.
The model geometry was generated using CATIA software and then imported as a neutral file in ANSYS

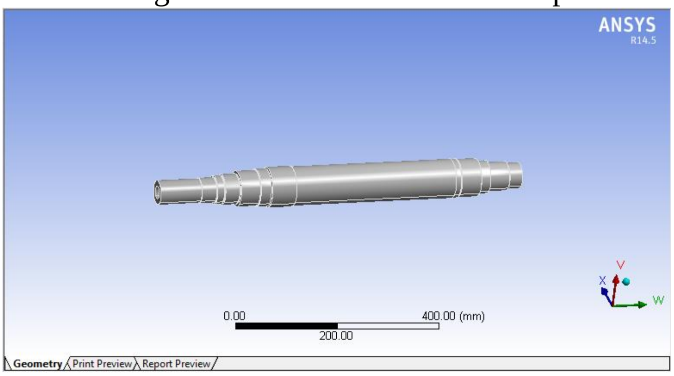

Figure 4: CATIA model of shaft of foot type motorin ANSYS

2.2 MESHING

Next, the prepared geometric structure is reproduced by finite elements. The finite elements are connected by nodes that make up the complete finite element mesh.
Each element type contains information on its degree of freedom set (e.g. translational, rotational, and thermal), its material properties and its spatial orientation 3D-element type's solid brick 8 node. The mesh was controlled in order to obtain a fine and good quality mapped mesh. The shaft has36426nodes and 21642 elements.

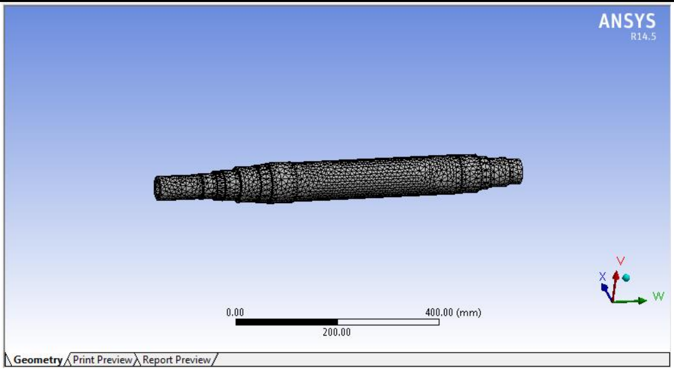

Figure 5: Meshing

2.2.1STATIC ANALYSIS

Static analysis deals with the conditions of equilibrium of the bodies acted upon by forces. Astatic analysis can be either linear or non-linear. All types of non-linearity are allowed such as larged formations, plasticity, creep, stress stiffening,contact elements etc. A static analysis calculates the effects of steady loading conditions on a structure, while ignoring inertia and damping effects, such as those carried by time varying loads. A static analysis can however include steady inertia loads such as gravity, spinning and time varying loads conditions are assumed, that is the loads and the structure responses are assumed to vary slowly with respect to time. The kinds of loading thatcan be applied in static analysis includes, Externally applied forces, moments and pressures, Steady state inertial forces such as gravity and spinning, imposed non-zero displacements.

2.3 LOAD APPLICATION

In order to solve the resulting system equation, boundary and loaded conditions are specified to make the equation solvable. In our model, shaft one end fixed and other end has different axial loads were applied. The Torque variation in range of 285 N-m to 287 N-m on user end of the shaft is applied.

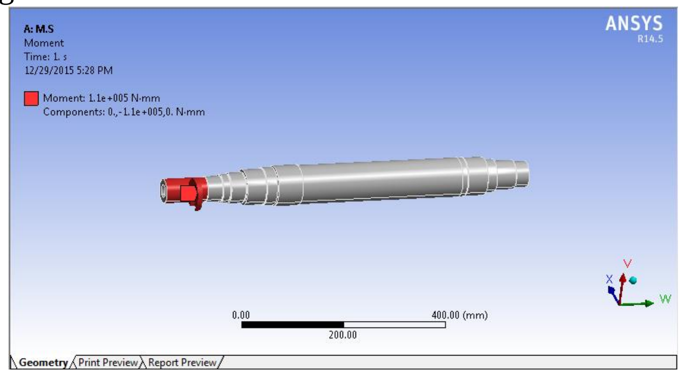

Figure 6: Load application (Torque)

2.3.1 Analytical solution

Most of the transmission shafts supporting gears and pulleys are subjected to a combined load of bending and torsional moments. The shaft materials are ductile and principal shear stress theory of failure is used to determine the shaft diameter. When the shaft is subjected to Bending moment Mb and Torsional moment Mt, the Bending stress 6b and Torsional shear stress T an given by

$$6a = \frac{Mby}{I} = \frac{32\ Mb}{d^3} \qquad 01$$

$$T = \frac{Mtr}{J} = \frac{16\ Mb}{\pi d^3} \qquad 02$$

The max shear stress in the shaft can be determined by constructing a Mohr’s circle and is given by

$$T\ max = \sqrt{\left(\frac{6b}{2}\right)(T)^2}$$

Substituting equation 1) & 2) in above expression

$$T\ max = \frac{16}{\pi d^3}\sqrt{Mb^2 + Mt^2} \qquad 03$$

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One important approach of designing a transmission shaft is to use the ASME code. According to this code, the permissible shear stress Ta for shaft without keyways is taken as 30% of yield strength in tension or 18% of the ultimate tensile strength of material whichever is minimum, therefore.

$$\left.\begin{aligned} Td &= 0.30\ Syt \\ &= 0.18\ Sut \end{aligned}\right\}\ \text{Whichever is minimum}$$

Syt = Yield Strength in Tension
Sut = Ultimate Tensile Strength

If keyways are present these values are to be reduced by 25%. According to ASME code, the bending and torsional moments are to be multiplied by factors kb and kt respectively, to account for shock and fatigue in operating conditions. Thus eqn 3) is modified and rewritten as

$$Tmax = \frac{16}{\pi d^3}\sqrt{(KbMb)^2 + (KtMt)^2} \quad 04$$

Calculation for shaft diameter.

The material for given shaft is 40 c 8
Hence Syt = 380 Mpa (N/mm^2)
Sut = 650 Mpa
For max stress

$$\begin{aligned} Ta &= 0.3\ Syt \\ &= 0.3 \times 380 \\ &= 114\ N/mm^2 \end{aligned}$$

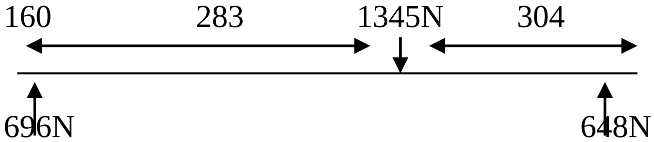
$$\begin{aligned} \text{Similarly } Ta &= 0.18\ Sut \\ &= 0.18 \times 650 = 117\ N/mm^2 \end{aligned}$$

The lower of the two values is 114 N/mm^2 and there are keyways on shaft, therefore
Td = 0.75 x 114 = 85.5 N/mm^2
(Permissible stress)

The shaft is supported with two bearings and a rotor is mounted on shaft.

The self weight of shaft and rotor are calculated from Ansys, similarly the force reactions at bearings locations is also calculated from Ansys.

Thus,



The total self weight of shaft and rotor is approximately 1345 N and its location (center of gravity) is 304 mm from right side end of bearing.

Thus maximum bending stress observed in shaft is at C.G. and it is.

Mb = 1345 x 304
Mb = 408880 N mm

Also the torque developed by motor is 287 NM (according to the value given by LHP as well as from design data book)

Mt = 287000 N mm.

Thus more stress developed in shaft is

$$Tmax = \frac{16}{\pi d^3}\sqrt{(KbMb)^2 + (KtMt)^2}$$

From ASME code Kb = 1.5, Kt = 1
Substituting all the values for permissible Stress i.e.
Ta = 85.5 N/mm2

$$d3 = \frac{16}{\pi td}\sqrt{(1.5 \times 408880)^2 + (287000 \times 1)^2}$$

d = 34 mm

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so minimum shaft diameter for permissible stresses is approximately 34 mm. where as lowest diameter in given shaft is 45mm (Groove dia). hence shaft is safe and it will not fail.

The actual stresses developed in shaft for given loading condition is

$$T\ max = \frac{16}{\pi(45)^3}\sqrt{(1.5 \times 408880)^2 + (287000 \times 1)^2}$$

$$= 37.84$$

$$= 38\ MPa$$

Thus the actual stresses in shaft are less than the permissible stress, hence shaft is safe for current design.

3. RESULTS

As analysis finishes, the result as shown in fig is found

- **3.1 Static Analysis For Equivalent (Von-Misses) Stress:**

Static analysis of Shaft is done by using FEA. Analysis is done for the material EN24 in order to check Equivalent stresses and its corresponding deformations induced in the Shaft.

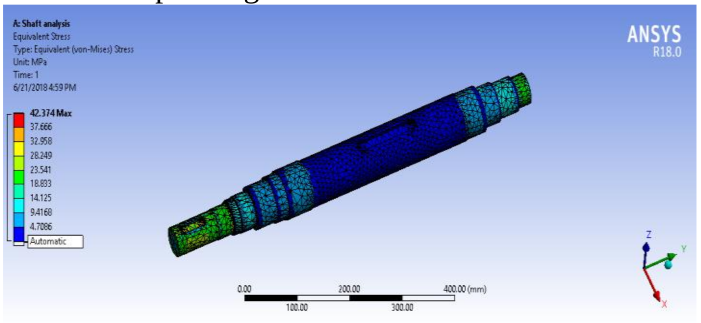


Fig (3.1.1): Equivalent Stress of Shaft, Mpa

- **3.1.2 Maximum Shear Stress:**

The Maximum shear stresses are.

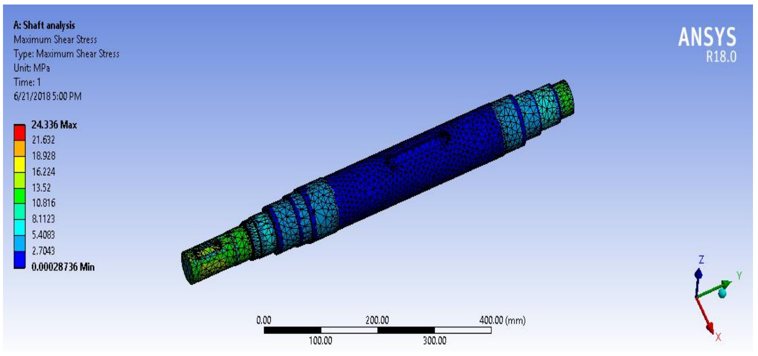


Fig (3.1.2): Shear Stress of Shaft, MPa

- **3.1.3 Static Analysis for Total Deformation:**

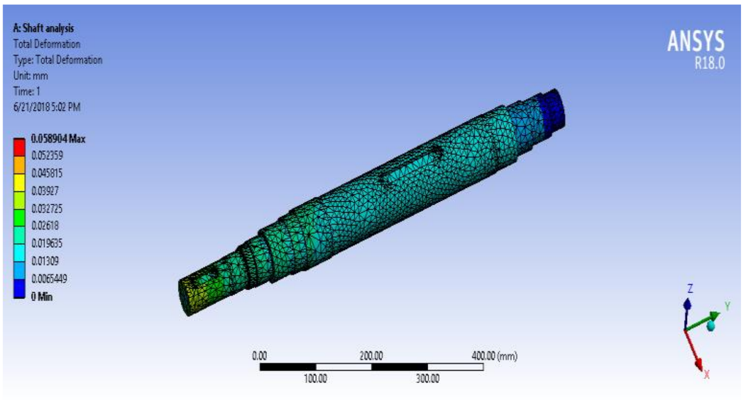


Fig (3.1.3): Total Deformation of EN24Shaft mm6.9.4

- **3.1.4 Fatigue Life**:

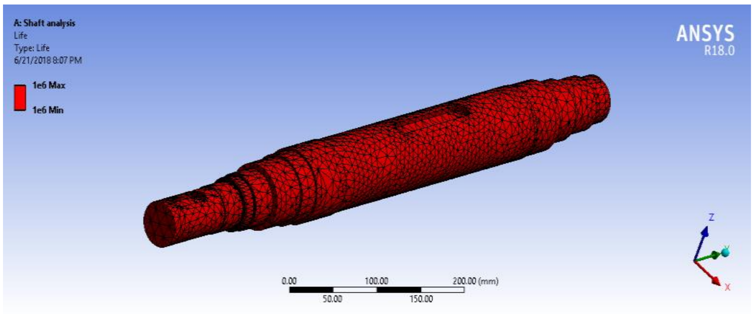


Fig (3.1.4): Fatigue Life

- **3.1.5 Fatigue Factor of Safety:**

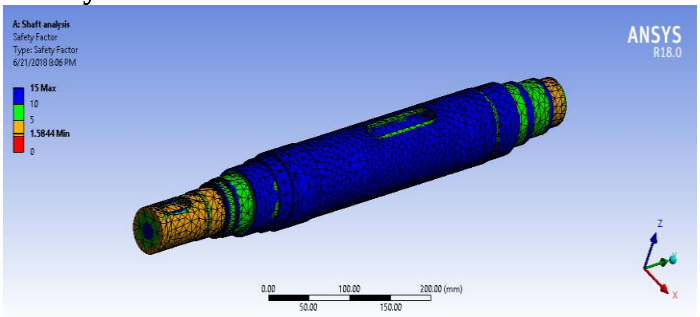


Fig (3.1.5): Fatigue Factor of Safety

- **3.1.6 Fatigue Sensitivity:**

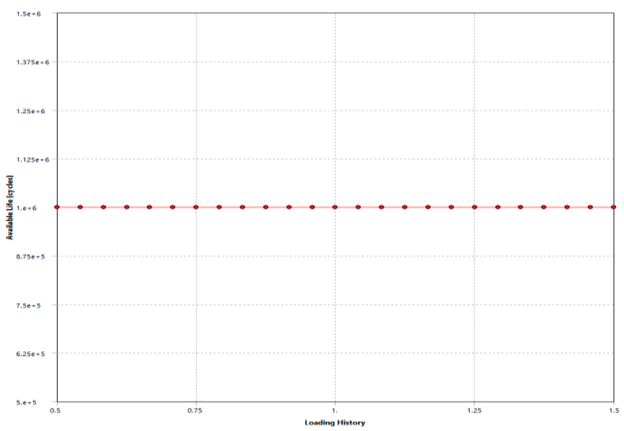


Fig (3.1.6): Fatigue Sensitivity

- **3.1.7 S N Curve**

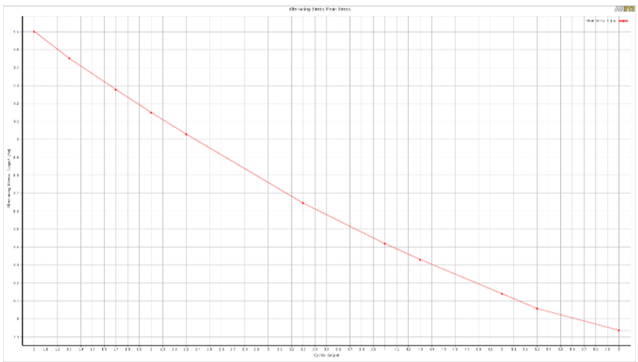


Fig (3.1.7): S-N Curve

➢ 3.2 Results & Discussions:

Sr. No.	Material	Type	Equivalent Stress(MPa)	Shear Stress(MPa)
1	40C8	Analytical	38MPa	22.12
		Ansys	42MPa	24.33

The maximum deflection induced in shaft CI material is 15 mm and corresponding Equivalent and shear stress induced in Shaft is 14.39 & 7.39 respectively. Weight of the existing Shaft is 13.10 kg.

➢ 3.3 Improvement (Design):

As we can see maximum stresses are produces on the shaft as compared to analytical solution. So stress can be reduced by changing the material properties. Material change from 40C8 to 45C5.

➢ 3.3.1 Equivalent Stress improved

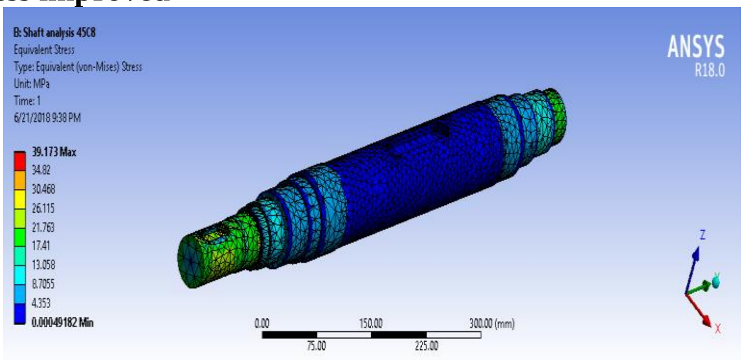


Fig (3.3.1) Equivalent stress

➢ 3.3.2 Static Analysis for 45C8 Total Deformation improved:

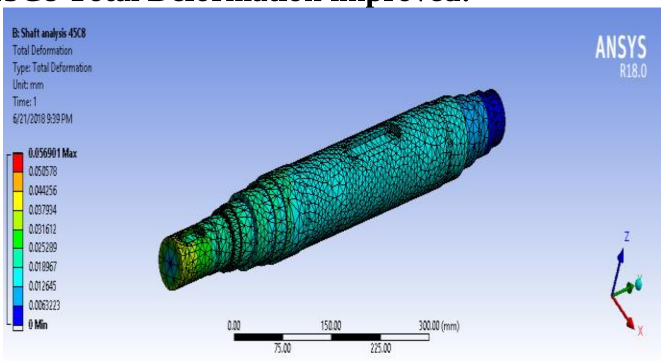


Fig (3.3.2): Total deformation

➢ 3.3.3 Fatigue Alternative Stress Improved:

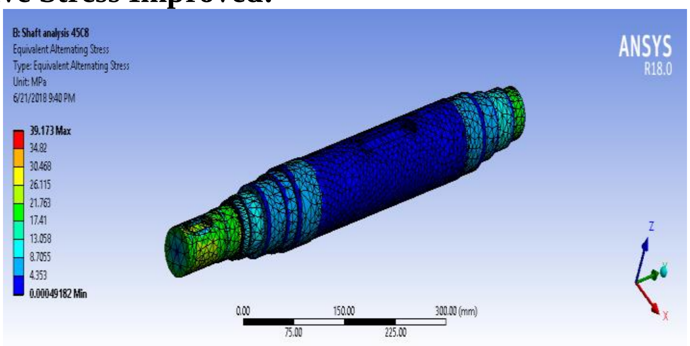


Fig (3.3.3) Fatigue alternative stress

- **3.3.4 Fatigue Factor of Safety Improved:**

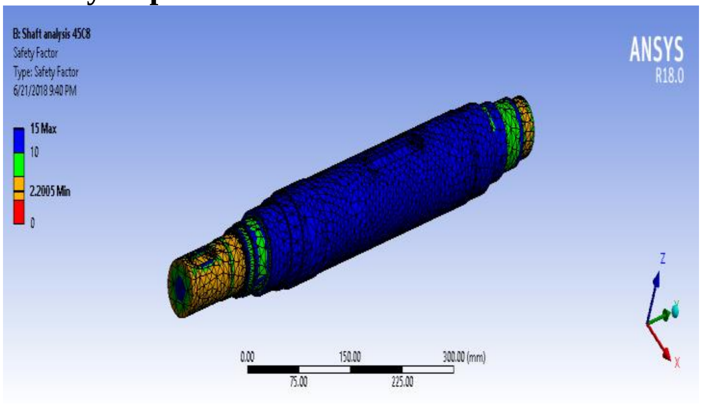


Fig (3.3.4): Fatigue factor of safety

- **3.3.5 Fatigue Life Improved**

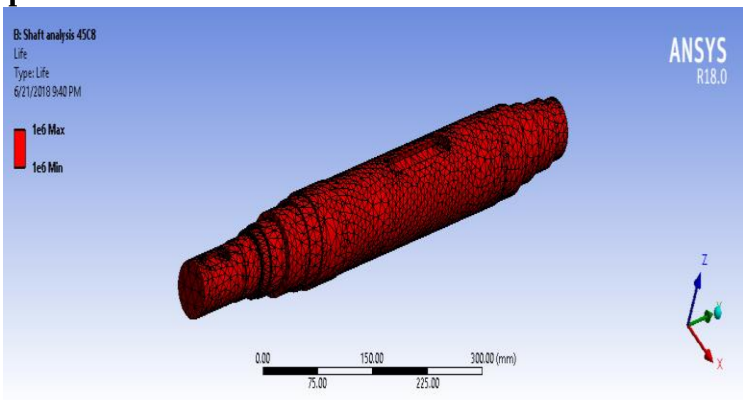


Fig (3.3.5): Fatigue Life Improved

- **3.3.6 Maximum sheer stress improved:**

Now maximum sheer stress is achieved

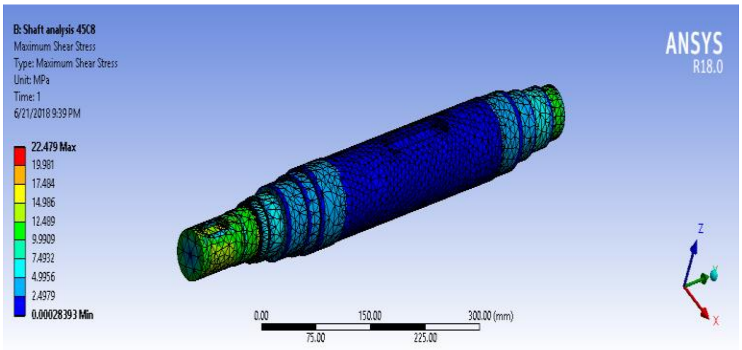


Fig (3.3.6) Maximum Sheer Stress Improved

- **3.4 Results & Discussions:**

Sr. No.	Material	Equivalent Stress(MPa)
1	40C8	38
2	45C8	39.173

4. MODAL ANALYSIS

Any system occurring in the universe has a property of vibrating. The frequency at which natural vibration occurs can be determined by using the mode shapes. This mode shapes can be determined analytically using the modal analysis. Modal Analysis is a field of science which mainly deals with the study of the structures or components dynamic properties during the vibration excitation. It is mainly done in order to determine the natural frequency of the system which helps in determining the speed at which component can satisfactorily work without compromising with the stability of the vehicle. In modal analysis coarse mesh can be used because stresses are not taken into consideration. Natural frequency and different mode shapes are found by modal analysis. Natural frequency is a function of stiffness and mass.

In order to check the vibrations, a modal analysis is done. For which a theoretical value of Natural frequency is required. It is calculated by,

- **4.1 Theoretic calculations of Natural frequency:**

As there are no various of diameters on various length on the shaft, so converting it into equivalent length by following formulae.

- **4.1.1 Equivalent length:**

$l = l_1+l_2 (d_1/d_2)^4+l_3(d_i/d_3)^4$----$l_{11}(d_1/d_{11})^4$

Putting values of Diameters d in above equationl

$=42+68(42/44.6)^4+21(42/45)^4+70(42/50)^4+140(42/52.3)^4+15(42/56)^4+92(42/50.15)^4+31(42/45)^4+26.9(42/40)^4+8(42/36)^4+41(42/48)^4$

=42+52.47+15.93+34.85+58.22+4.74+45.25+23.52+32.69+14.82+207.56

Equivalent length= 532.05 mm

- **7.1.2 Equivalent Diameter:**

D1 = 42.25 mm.

$\delta_1 = W_1 a_1 b_1 / 3 EIL$

Where,

E = Young's Modulus of the Material, N/mm2

L= Effective length of the shaft, mm

I= Moment of Inertia of the shaft, mm4

I= π/4× (d2)

I= π/4× (422)

I= 86.59 mm4.

Putting the value of Iin Natural frequency formulae.

δ1= 33.12×1922×1522/ 3× (114×103) ×86.59×532.05

δ1= 35.79mm.

Similarly

$\delta_2 = W_2 a_2 b_2 / 3 EIL$

δ2= 5296×4232×792 / 3×(114×103)×86.59×532.05

δ2= 375.34 mm

Natural frequency is given by,

$Fn= 0.4985/\sqrt{\delta_1+\delta_2}$

$Fn= 0.4985/\sqrt{30.32+ 316.15}$

Fn= 0.024 Hz

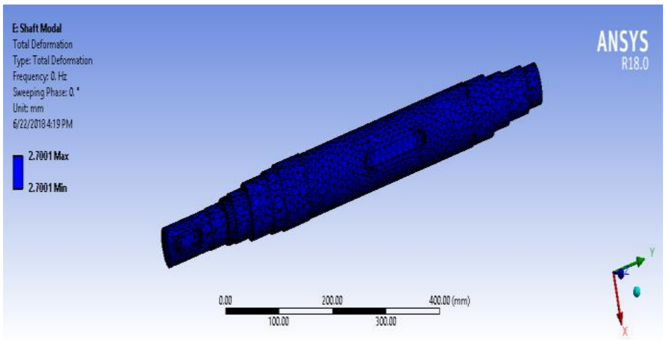
7.2Validation of Natural frequency by Dynamic Analysis:

In Dynamic Analysis range Following details was give

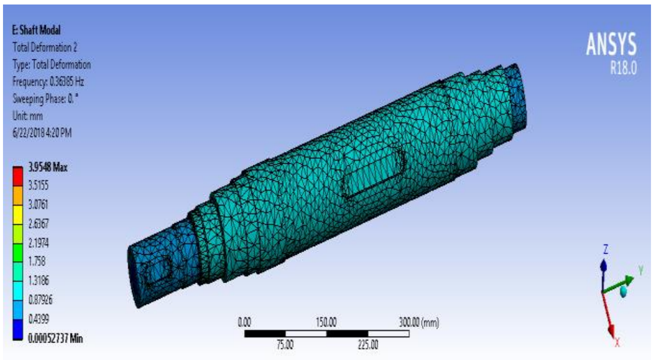
Range of Natural Frequency = 0 to 2546.7 Hz.

No of Modes = 10

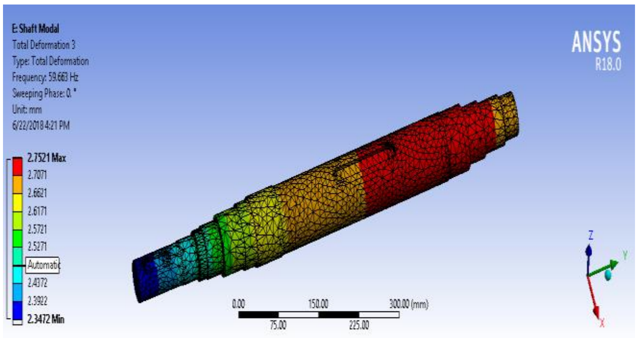
7.2.1 Results :



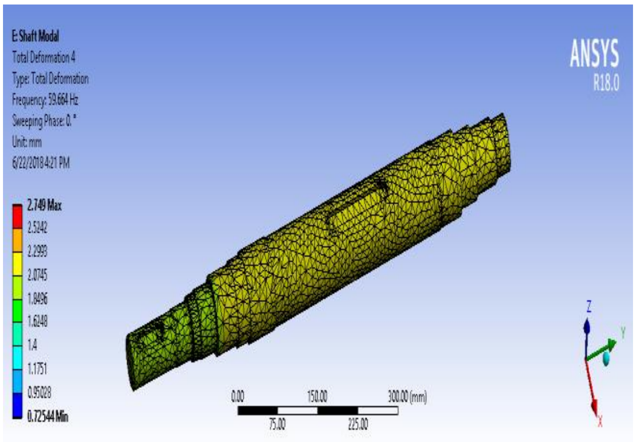
TotalDeformation1



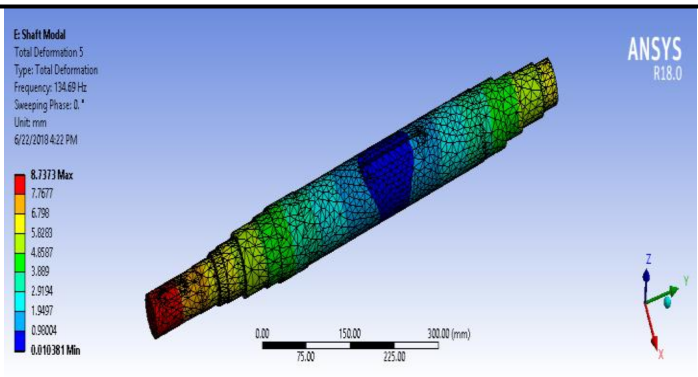
TotalDeformation2

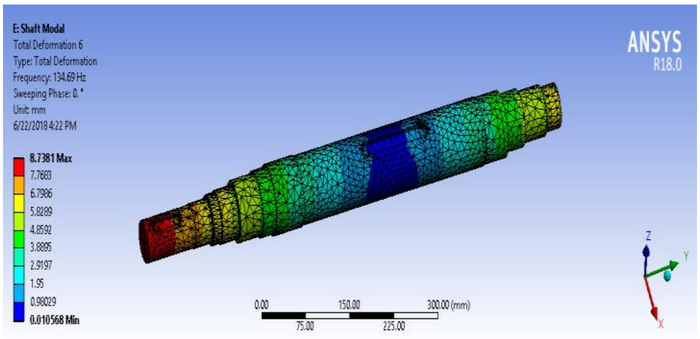


TotalDeformation3

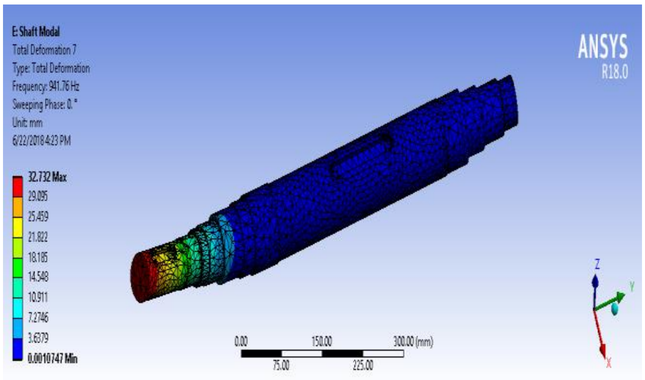


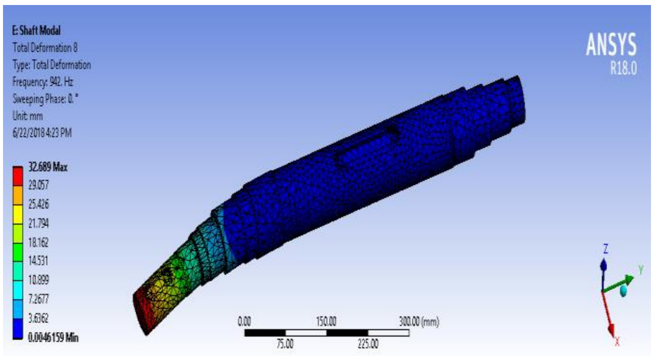
TotalDeformation4 TotalDeformation5



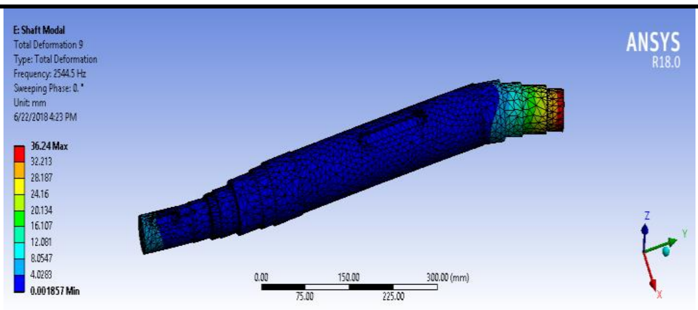


TotalDeformation6

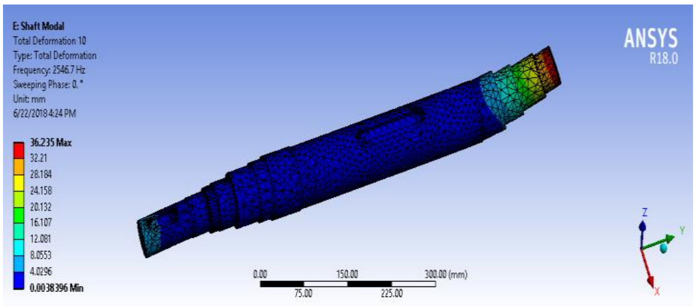




TotalDeformation7 TotalDeformation8



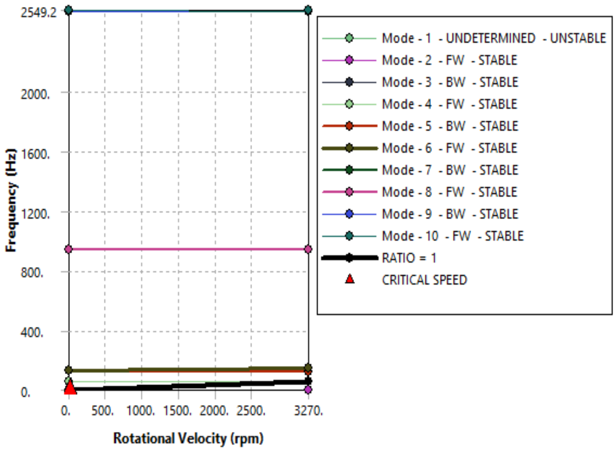
TotalDeformation9



Total Deformation 10

➢ 4.3 Campbell Diagram

Model (E4) > Modal (E5) > Solution (E6) > Campbell Diagram



Model (E4) > Modal (E5) > Solution (E6) > Campbell Diagram

Table 4.1

Mode	**Whirl Directic**	**Mode Stabil**	**Critical Sp**	**0. rpm**	**3270. rpm**
1.	**UNDETERMIN**	**UNSTABLE**	**NONE**	**0. Hz**	**0. Hz**
2.	FW	STABLE	21.831 rpm	0.36385 Hz	0.36385 Hz
3.	BW	STABLE	NONE	59.663 Hz	59.661 Hz
4.	FW	STABLE	NONE	59.664 Hz	59.666 Hz
5.	BW	STABLE	NONE	134.69 Hz	123.48 Hz
6.	FW	STABLE	NONE	134.69 Hz	146.89 Hz
7.	BW	STABLE	NONE	941.76 Hz	939.11 Hz
8.	FW	STABLE	NONE	942. Hz	944.76 Hz
9.	BW	STABLE	NONE	2544.5 Hz	2542.1 Hz

Critical speed for given shaft is 21.83 rpm at 0.3638 Hz for given speed and shaft is stable for all other frequency.

➢ 4.4 COMPARISON TABLE (DESIGN):

Table: 4.4Comparison Table (Design)

Sr. No.	Parameters	Analytical	Ansys
1	Static	38 MPa	39.173 MPa
2	Dynamic	0.024 Hz	0.024 Hz

5.CONCLUSION

Stresses developed in shaft are within permissible limit. But there was deviation in Theoretical and actual stresses. To minimize the actual stresses in the material of shaft is changed from 40C8 to 45C8 and thus the stress developed in shaft found close to the theoretical calculation.

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