ISSN: 2455-2631 © March 2020 IJSDR | Volume 5, Issue 3

Design and Fatigue Analysis of Yaw Gear box Shaft (17CrNiMo6) Subjected to Pure Torsion

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***Abstract*: In material engineering it is important to determine the cause of the failure & prevention of the failure. In present day the failure of the machine component is about 90% of the failure is because of the fatigue. In present study the failure of the shaft in the yaw gear box is analyzed. The shaft is rotating part of the yaw drive is used to keep the rotor facing into the wind as the wind direction changes. It holds the maximum stress. In this case shaft of heat treated component with ultimate tensile strength Su= 1295Mpa is Designed in UG NX-10 and Analysis done with ANSYS18.1 (FEA) software & compared with the theoretical calculation. There are different methods which are used to predict fatigue life include stress life(S-N), strain Life (E-N) and Linear Elastic Fracture Mechanics (LEFM). In this project study, S-N approach is used to predict fatigue life for yaw gear box shaft.**

***Index Terms*: Shaft, UG-NX10, FEA-ANSYS (18.2), SN-curve, Fatigue –life, Fatigue factor of safety, Fatigue damage.**

I. INTRODUCTION

A shaft has a circular cross section & it is a rotating part of the yaw drive is used to keep the rotor facing into the wind as the wind direction changes. A shaft usually not a uniform cross section because it is mounted by the bearing, fly wheels, clutches & other machine elements are mounted on the shaft .In the present shaft of heat treated material (17CrNiMo6) is mounted by spur gear with pressure angle of 20^0 & supported by the two bearing (Roller bearing).

Stress Analysis

Stress in the Shaft In actual practice there are three kinds of stress are induced in it. a.) Shear stress by the transmission of the torque. b.) Bending stress by the force acting upon machine element, and weight of shaft itself. c.) Stress from both combined tensional and bending loads.

Fig 1: Detailed view of yaw drive

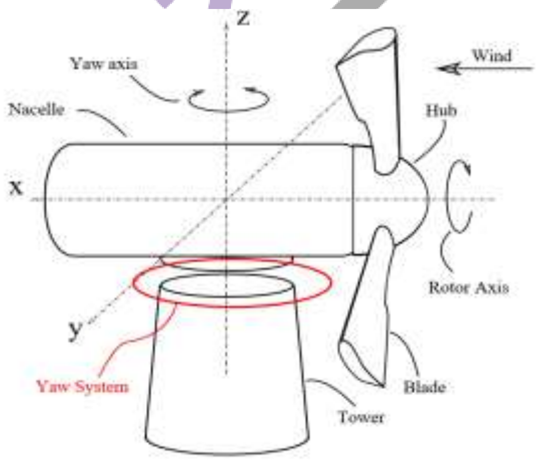


Fig 2: view of yaw drive system

II. SHAFT MATERIAL COMPOSITION AND MECHANICAL PROPERTIES

Material is used in the shaft is Chrome–Nickel- Moly carburizing steel (17CrNiM06)

Table 1 Chemical composition of 17CrNiM06

composition	percentage
Carbon	0.18
Silicon	0.20
Nickel	1.55
Manganese	0.70
Chromium	1.65

Mechanical properties of 17CrNiMo6

Tensile yield strength Sy =1050Mpa

Ultimate tensile strength Su =1295Mpa
Young's modulus of material E = 207.Gpa
Passion's ratio = 0.3
Density δ=7800 Kg/m3

III. DESIGN AND CALCULATIONS

Fatigue stress concentration factor

When there is an existence of the irregularities or discontinuities, such as holes, grooves, or notches, in a part increases the theoretical stresses significantly in the immediate vicinity of the discontinuity defined as a stress concentration factor Kt or Kts.
Effective maximum stress is

$\sigma_{max}= K_f\ \sigma_{nominal}$, $\tau max=K_{fs}\ \tau_{nominal}$ (1)

Where Kf is a reduced value of Kt and is the nominal stress $\sigma 0$. The factor Kf is commonly referred as fatigue stress-concentration factor and hence the subscript f. The resulting factor is defined by the equation

$$Kf = \frac{\text{maximum stress in notched specimen}en}{\text{Stress concentration notch free specimen.}} \quad (2)$$

$Kf=1+q\ (Kt-1)$ or $Kfs=1+q_{shear}\ (Kts-1)$ (3)

Where q = Notch sensitivity factor

$$q=\frac{1}{1+\frac{\sqrt{a}}{\sqrt{r}}} \quad (4)$$

Where $\sqrt{a}$= Neuber constant
r = Fillet radius

Neuber constant is given by the equation

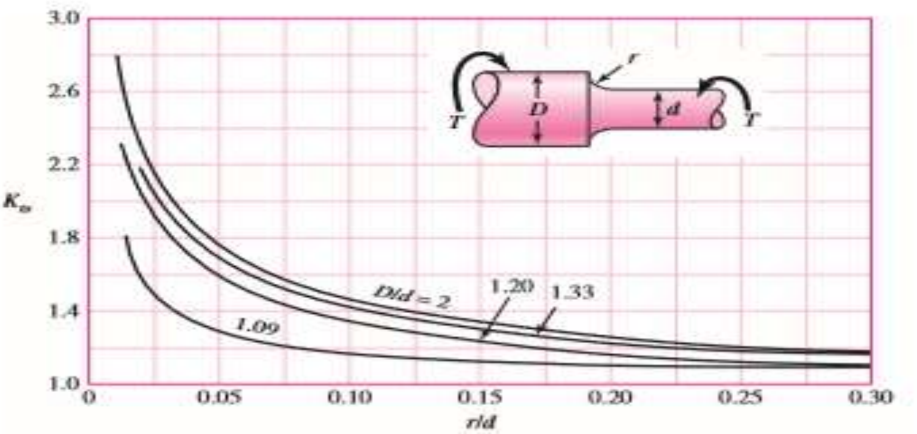
a. Bending: $\sqrt{a}$ =0.246−3.08 (10^{-3}) Sut+1.51(10^{-5}) (S_{ut}) ^2−2.67(10^{-8}) (Sut)^3 (5)

b. Torsion: $\sqrt{a}$ =0.190−2.51 (10−3) Sut +1.35(10−5) (S_{ut}) ^2−2.67 (10^{-8}) (Sut) ^3 (6)

Applying S_{ut} = 1295Mpa

$$q=\frac{1}{1+\frac{\sqrt{.0765}}{\sqrt{1.414}}}=.9486$$

Where Kt = Stress concentration factor for torsion.



Stress concentration factor Graph for torsion

Where r = Fillet radius
d = Diameter of smaller shaft
D = Diameter of larger shaft
r/d = 2/85 =.0235
D/d =100/85=1.1764
From the graph & question (3) Kt =1.85
K fs =1+0.9486(1.85-1)
=1.80631

Generation of S-N curve

Fatigue strength (Endurance strength S_n) of the shaft material was calculated as

Sn = 0.5* UTS = 0.5* 1295 = 647.5Mpa

.Considering the corrections factors for endurance limit

Ka = surface finish factor = 0.5

Kb = size factor = d>50 mm = 0.75

Kc = Reliability factor(for 90% Reliability) = 0.897

Kd = Load factor = 1

Endurance limit Se = 0.5 Sn*Ka*Kb*Kc*Kd

= 0.5 *1295*0.5*0.75*0.897*1

= 217.81 Mpa

S-N Curve

Tensile yield strength Sy =1050Mpa

Ultimate tensile strength Su =1295Mpa

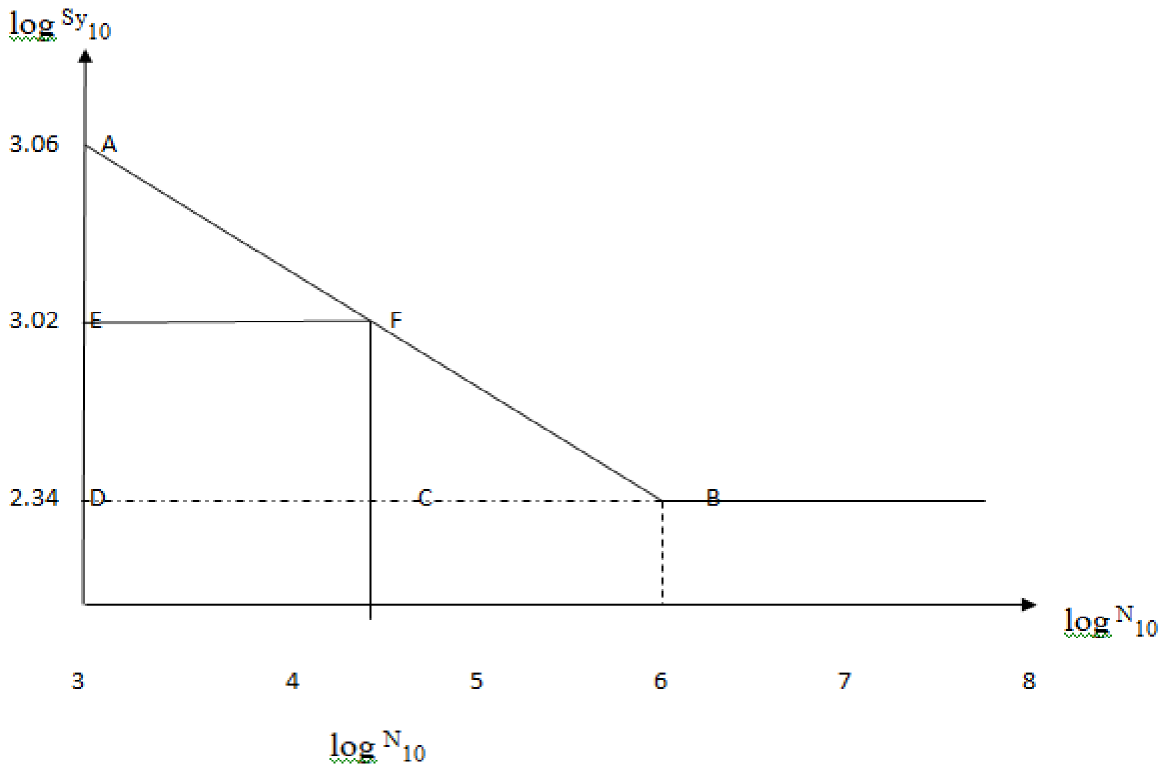
Reliability = 90%

0.9 Su = 0.9 *1295 = 1165.5Mpa

$\log\ (_{10}{}^{0.9\,Su})$ = 3.0665

$\log\ (_{10}{}^{0.9\,Se})$ =2.34

$\log\ (_{10}{}^{0.9\,Sf})$ =3.02



S- N CURVE

$$\overline{EF} = \frac{\overline{DB} \times \overline{AE}}{\overline{AD}} = (6-3)*(3.06-3.02) / (3.06-2.34) = 0.18$$

$\log {}^{N}{}_{10} = 3 + \overline{EF} = 3+0.18=3.18$

N = 1513.56 Cycles

Shaft Shear stresses

Yaw gear box shaft is subjected to a torque of 50000 N-m .The maximum shear stress is shown in which appears in the cross-section of the shaft τmax with the fatigue stress concentration k fs = 1.8

$$\tau max = \frac{16T * 1.8}{\Pi d^3}$$

$$= \frac{16*50000000*1.8}{\Pi 85^3}$$

$$= 746.37 \text{ Mpa}$$

Equivalent alternating stress or Equivalent von-misses stress

In a fatigue Life Stress analysis, one always needs to be query of an SN curve to relate the fatigue life to the stress state. Thus of the “equivalent alternating stress” is the stress used to query the fatigue SN curve after accounting for fatigue loading type, mean stress effects, multi axial effects and any other factors in the fatigue analysis. Thus in a fatigue analysis, the equivalent alternating stress can be thought of as the fast calculated quantity before determining the fatigue life. The maximum value of the equivalent stress is 1527.4Mpa, which takes place in the cross-section of the shaft where the Roller -bearing was located.

Equivalent von-mises stress or alternating stress σv

$$\sigma v = \sqrt{3 * \tau max^2}$$

$$= 1292.75 \text{ Mpa}$$

Factor of safety of fatigue life

$$\text{Factor of safety of fatigue life} = \frac{\text{Endurance stress}}{\text{Equivalent alternating stress}}$$

$$= \frac{217.8}{1292.75}$$

$$= 0.17$$

Fatigue damage

Fatigue damage is defined as the design life divided by the available life

$$\text{Fatigue damage} = \frac{\text{Design life}}{\text{Availble life}}$$

$$= \frac{10^6}{1513.56}$$

$$= 660.6$$

IV. DESIGN OF YAW GEAR BOX SHAFT IN UG NX 10

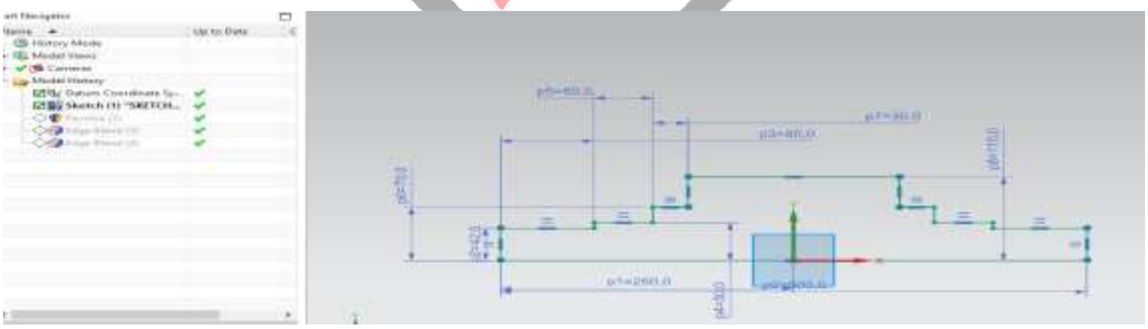


Fig 3: Sketch of yaw gear box shaft

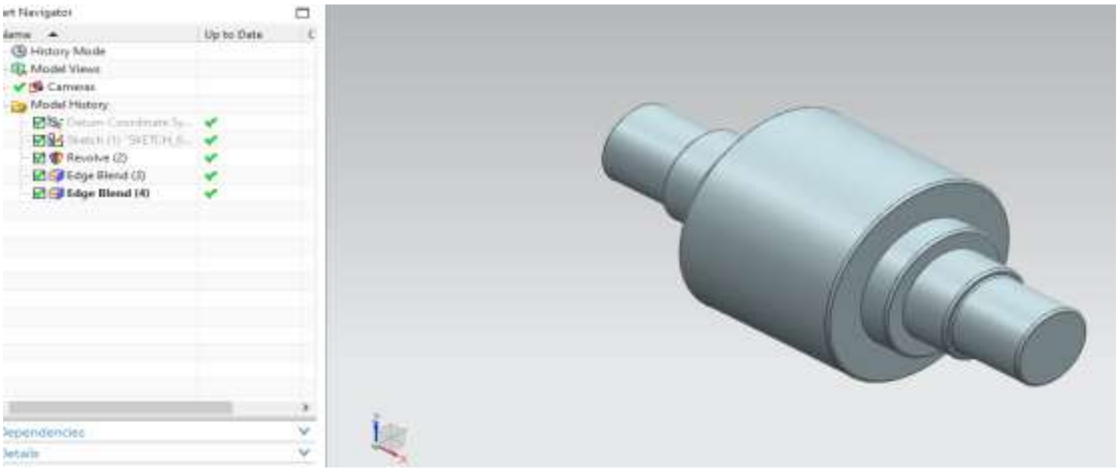


Fig 4: Modeling of yaw gear box shaft

V.ANALYSIS OF YAW GEAR BOX SHAFT

Meshing of shaft

Basic idea of FEM is to perform calculations at limited number of points called nodes and interpolate the results for entire domain using interpolation functions. Any continuous object has infinite degrees of freedom and such problems cannot be solved using this method. So this method reduces the degrees of freedom from infinite to finite by making assumptions and by discretization/meshing in other terms creating nodes and elements. Hexagonal mesh is used, relevance center is fine & elements size is 3.14mm created the nodes 10483 and elements 5839.

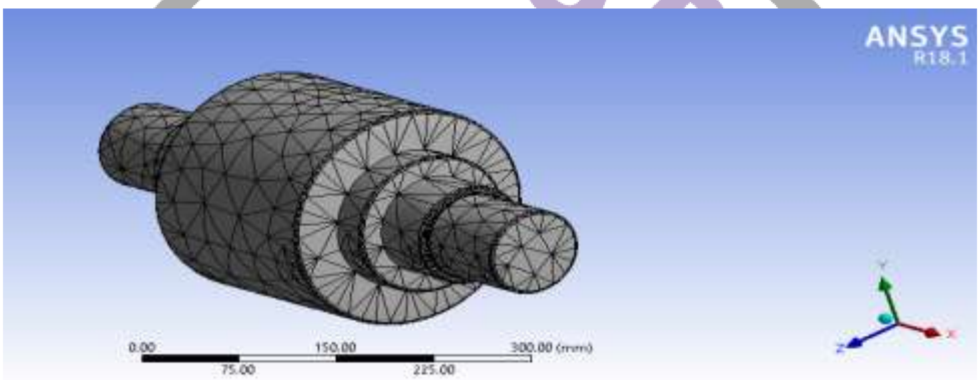


Fig 5: Meshing of shaft

Loads acting on shaft

Stepped shaft is subjected to a torque of 50000N-m at the right end of shaft.

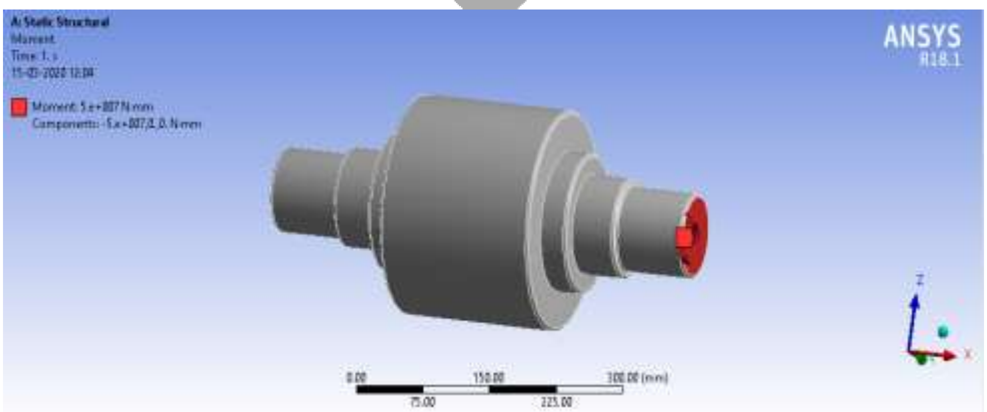


Fig 6: Load acting on the shaft

Shaft Shear stresses

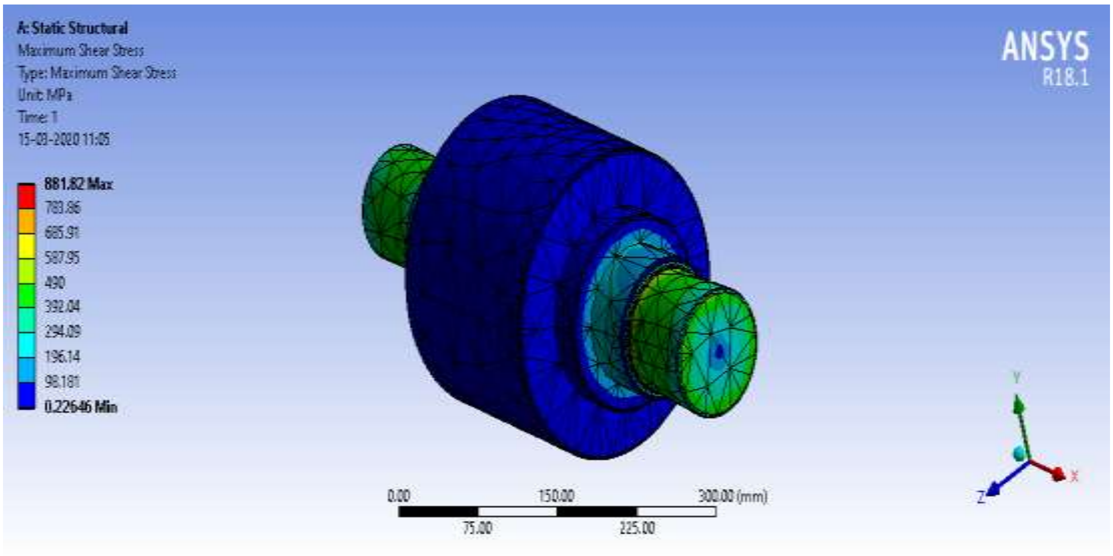


Figure 7: Shaft Shear stress in Mpa

Equivalent alternating stress or Equivalent von-misses stress

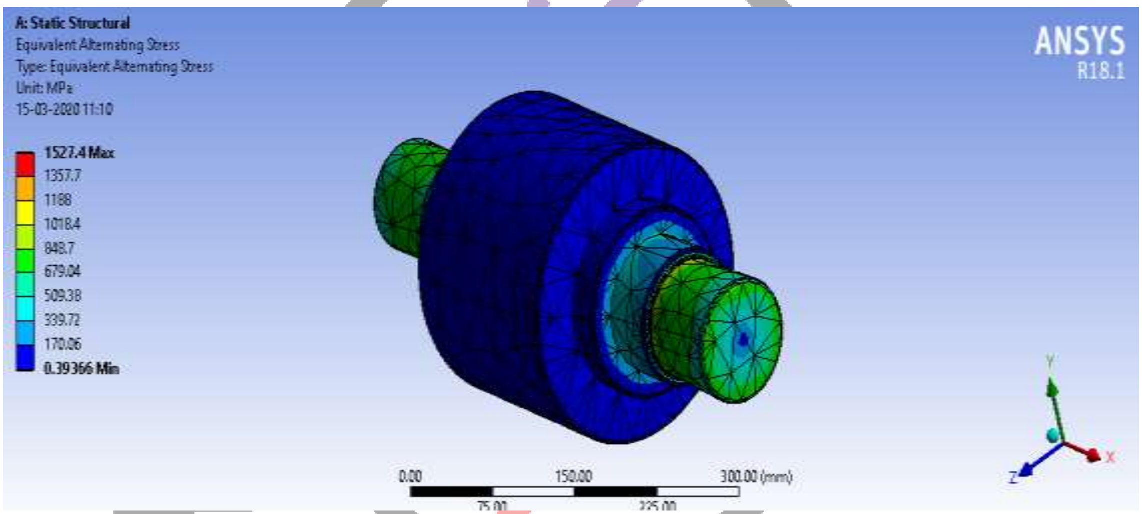


Figure 8: Equivalent alternating stress in Mpa

Factor of safety of fatigue life

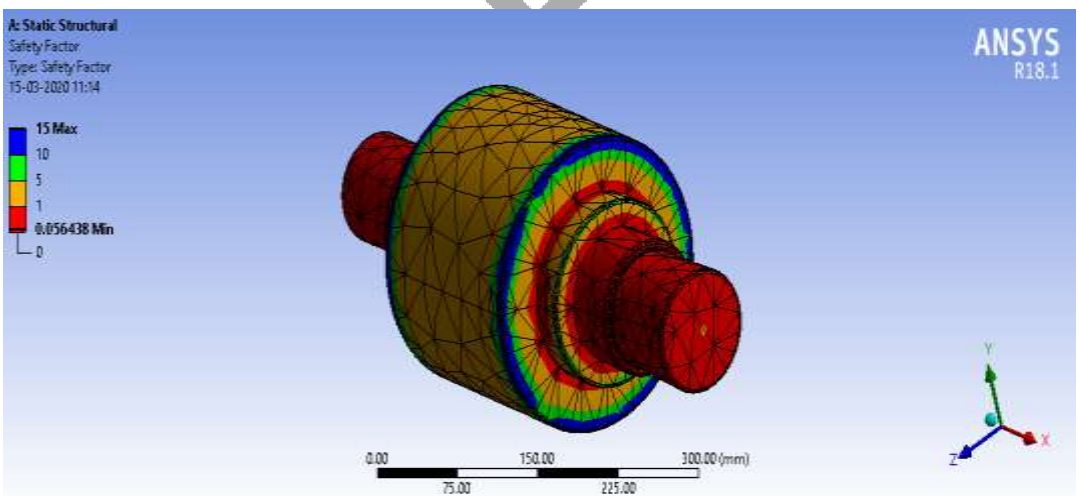


Figure 9: Factor of safety

Number of cycles

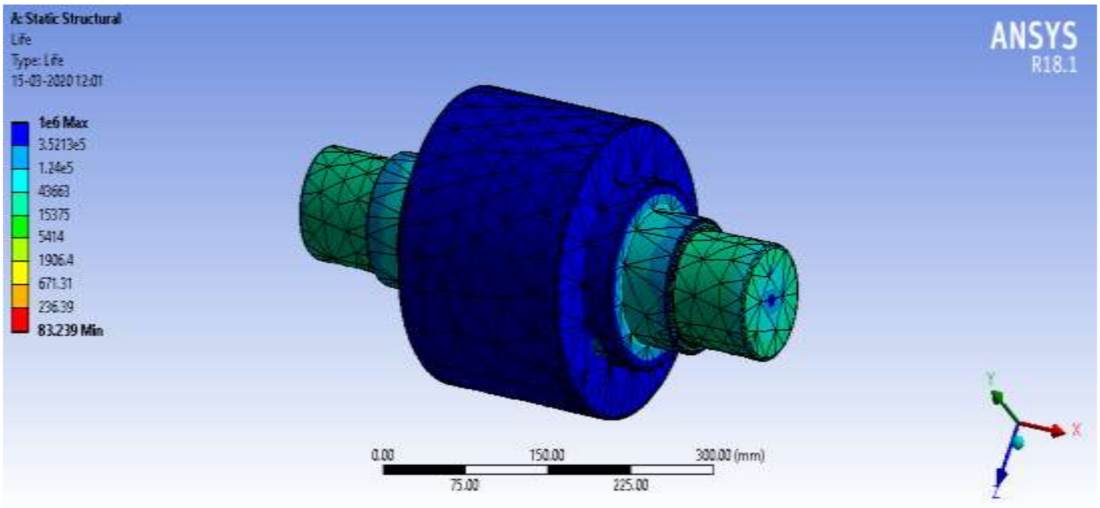


Figure 10: Showing the fatigue life

Fatigue damage

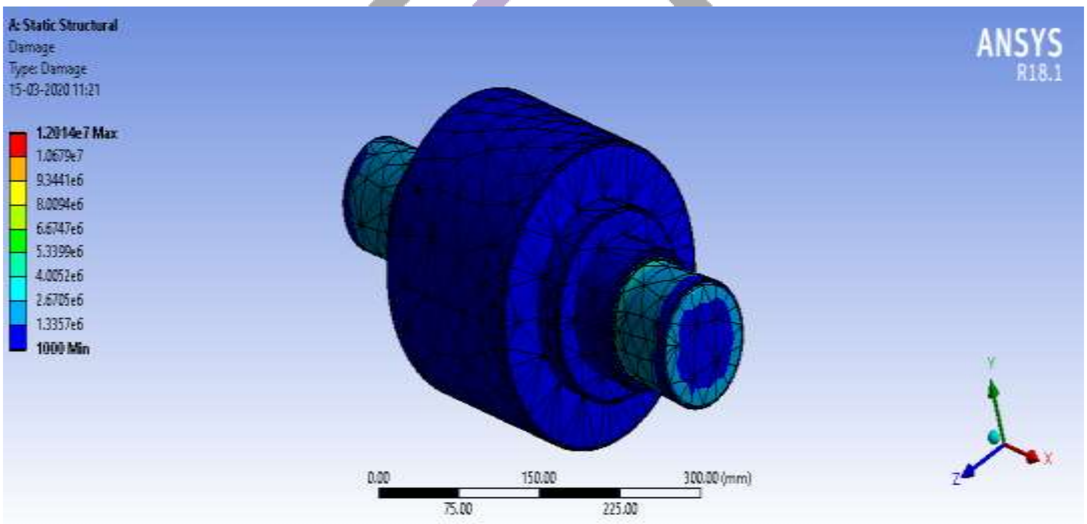


Figure 11:Fatigue damage

V.RESULTS AND DISCUSSION

Table 2 Result of Comparison

Parameters	Analytical results	FEM results
1. Shear stress in Mpa	746.37	881.82
2. Equivalent alternating stress in Mpa	1292.75	1527.4
3. Factor of safety	0.17	0.06
4.Fatigue life cycles	1513.56	832.29
5. Fatigue damage	660.6	1000

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VI.CONCLUSIONS

Fatigue Failure analysis of the shaft is investigated in detail. Force acting on the bearing due to the torque is determined. Endurance limit & fatigue factor of safety is calculated. Fatigue life of the shaft is estimated & fatigue damage is calculated. Forces and stresses are calculated by using an analytical approach and ANSYS software. Both methods show that the stresses and fatigue life are nearly same and in the admissible range.

1. From the static analysis it was observed that maximum stress is located at the change in the cross-section area of the shaft it is found 1527 Mpa.
2. Fatigue life is found to be 832 cycles.
3. Factor of safety is 0.06<1 which means the design is not safety.

4. Fatigue damage at the given life is 1000
5. Stress concentration is more where there is a change in the cross-section of the shaft.

REFERENCES

1. Failure Analysis and Fatigue Life Estimation of a Shaft of a Rotary Draw Bending Machine B. Engel, Sara Salman Hassan Al-Maeeni Vol:11, No:11, 2017
2. S. Gujaran, and S. Gholap, "Fatigue analysis of drive shaft," International Journal of Research in Aeronautical and Mechanical Engineering, Vol. 2, Issue. 10, P. 22-28, 2014.
3. Finite Element Structural and Fatigue Analysis of Single Cylinder Engine Crank Shaft" Bhumesh J. Bagde Laukik P. Raut ISSN: 2278-0181 Vol. 2 Issue 7, July – 2013
4.. R.A. Gujar, Shaft design under fatigue loading, IJERA, Volume 3, Issue 4, Jul-Aug 2013.
5.Pushpendra Kumar, Fatigue Analysis of connecting rod using FEA, IJARSE, 2012, Vol. No.1, September.
6. Prediction of Fatigue Life of Crank Shaft using S-N Approach Mahesh L. Raotole Prof. D. B. Sadaphale , Prof. J. R.Chaudhari (ISSN 2250-2459, ISO 9001:2008)