Analysis of Speed Reduction Assembly Output Shaft

Gurudatta N Vernekar[1], V N Sanagoudar[2*], Sharanagouda A Biradar[3], Bharatkumar A Biradar[4]

Research Scholars[1], Associate Professor[2], Research Scholars[3], Research Scholars[4]
School of Mechanical Engineering
KLE Technology University, Hubballi, Karnataka

Abstract: **Shaft is a power transmission element used in automotive, machine tool, heavy industrial applications. Usually shafts used for automotive transmission application will have stepped geometry with splines, groves machined on its surface to accommodate various mechanical components. In this project an output shaft which is used for speed reduction assembly for electric vehicle transmission is analyzed. The purpose of output shaft in speed reduction assembly is to transfer torque from input shaft to next stage of speed reduction system. The output shaft operates when there is demand of torque for speed reduction. Shear force diagram (SFD) and Bending moment diagram (BMD) is drawn to determine maximum bending moments and shear forces acting on the shaft. Based on this calculation equivalent von Mises stress and maximum shear stress calculations are done. Fatigue life, natural frequency, critical speed of the shaft is determined using analytical calculations. Static structural, Modal, Fatigue analysis is carried out using ANSYS and both analytical and software results are compared.**

Index Terms: Output shaft, Speed reduction assembly, SFD, BMD, von Mises stress, shear stress, Fatigue life, natural frequency, critical speed, ANSYS.

1. INTRODUCTION

Shafts form the important machine element, usually of circular cross section used to transmit power or motion. It provides the axis of rotation, for elements such as gears, pulleys, flywheels, cranks, sprockets etc and controls the geometry of their motion Shafts are subjected to torque due to power transmission and bending moment due to reactions from the members on which it is supported. Shafts are distinguished from axles because they support rotating members but do not transfer power. Axles are subjected to only bending and not to the torque.M.D. Shrotriya *et.al* [1] has carried out Design and analysis of mechanical component (shaft) in the alternator. The shaft which is used in the alternator is designed on the basis of static and dynamic loading Macaulay's method was used to determine shaft deflection and Dun Kerley's method was used to find the critical frequency of the shaft. [2] carried out design of shaft with groove under different loading conditions based on the equations of stress concentration factor and conventional Peterson's curves and Roark's curves the design of shaft at critical location Shafts having varying cross sections causes stress concentration at shoulders keyways causes fatigue failure of the shaft**.** B. Engel, *et.al* [3] in this paper a fatigue failure analysis was carried out on a shaft of rotary bending machine**.** stress and deflection analysis of the shaft subjected to combined torsion and bending loads are carried out by an analytical method and compared with a finite element analysis method.

2. PROBLEM FORMULATION

An output shaft is a part of speed reduction assembly which is used to transfer torque from engine to next stage of transmission. The shaft is having complex geometry with multiple steps and splines. The shaft rotates at peak speed of 8500 rpm transferring a torque of 200 N-m. The shaft is supported by a bearing at the rear end and a bushing at the front. Components such as gerotor pump, dog cone, lockup collar rest on the shafts outer diameter. The shaft is subjected to combined bending and torsional loading. As Shaft is a important component of transmission system it is necessary to analyze the critical section to prevent failure.

3. STATIC STRUCTURAL ANALYSIS

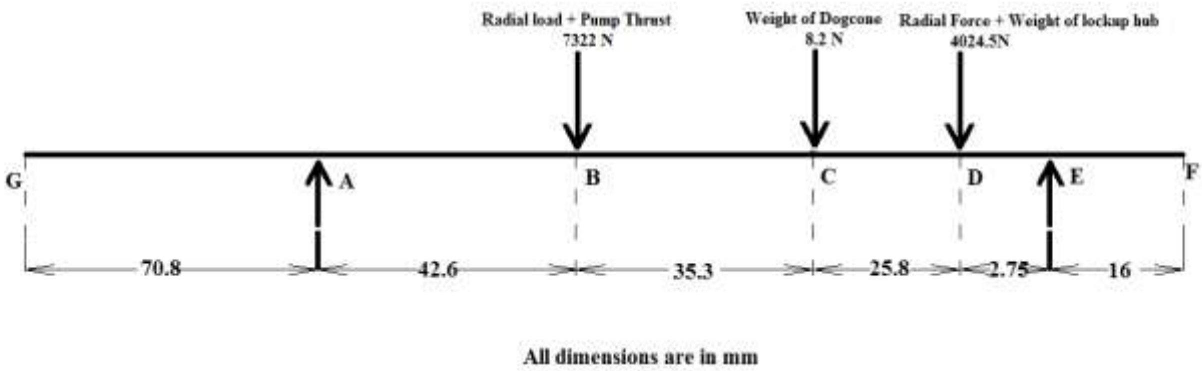

Fig 1: FBD of shaft

Static structural analysis determines the displacements, stresses, strains, and forces in structures or components caused by loads that do not induce significant inertia and damping effects. Steady loading and response conditions are assumed; that is, the loads and the structure's response are assumed to vary slowly with respect to time.The shaft is supported by bearings at A and by bushing at E Three point loads act on shaft at point B,C,D as shown in Fig 1.Considering the shaft as simply supported beam with point loads acting between the supports Shear force and bending moment diagram is drawn to to find the critical location of the shaft. The static deflection of the shaft can be determined using Macaulay's method [1]

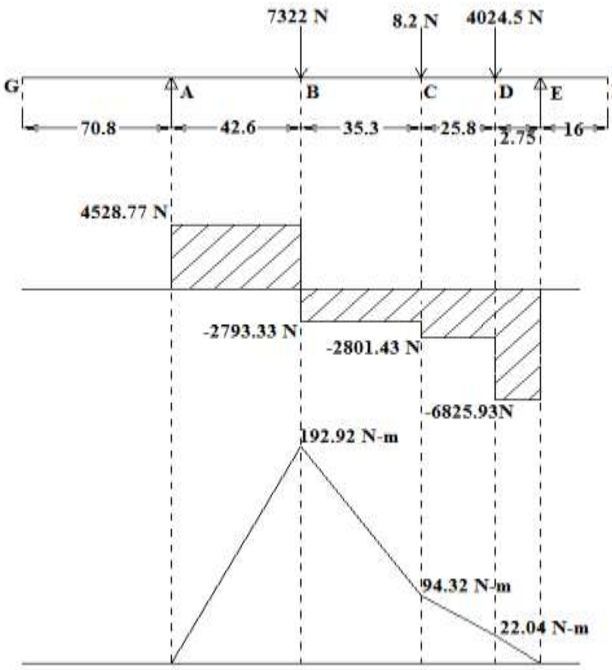

Fig 2: SFD and BMD

3.1Deflection of Shaft using Macaulay's Method

Macaulay's Method is a means to find the equation that describes the deflected shape of a beam. From this equation, any deflection of interest can be found. Macaulay's Method enables us to write a single equation for bending moment for the full length of the beam. When coupled with the Euler-Bernoulli theory, we can then integrate the expression for bending moment to find the equation for deflection.

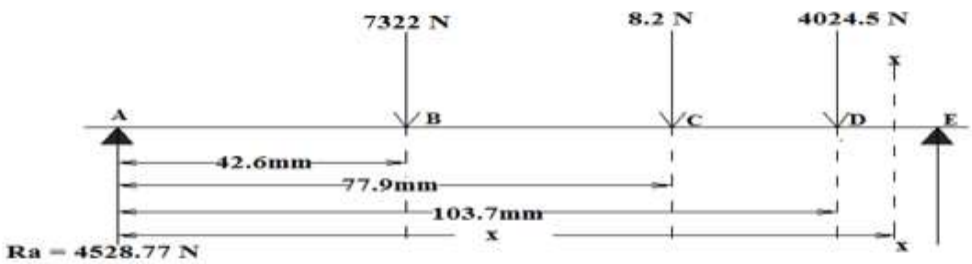

Fig 3: FBD of Beam

Using Macaulay's method for beam deflection

$$EI\frac{d^2y}{dx^2} = S_1(x) - w_1(x-L_1) - w_2(x-L_2) - w_3(x-L_3) \qquad (1)$$

$$EI\frac{d^2y}{dx^2} = 4528(x) - 7322(x-42.6) - 8.2(x-77.9) - 4024.5(x-103.7) \qquad (2)$$

Integrating (2)

$$EI\frac{dy}{dx} = \frac{4528x^2}{2} - \frac{7322(x-42.6)^2}{2} - \frac{8.2(x-77.9)^2}{2} - \frac{4024.5(x-103.7)^2}{2} + C1 \qquad (3)$$

Integrating (3)

$$EI\,y = \frac{4528x^3}{6} - \frac{7322(x-42.6)^3}{6} - \frac{8.2(x-77.9)^3}{6} - \frac{4024.5(x-103.7)^3}{6} + C1x + C2$$

Applying Boundary Conditions

1. at x=0 y=0
2. at x=106.45mm y=0

Applying 1st boundary condition we get

$C_2 = 0$

Applying 2nd boundary condition we get

$C_1 = -5.568\times10^6$

To find maximum deflection

$\frac{dy}{dx} = 0$

$\frac{4528x^2}{2} - \frac{7322(x-42.6)^2}{2} - \frac{8.2(x-77.9)^2}{2} - \frac{4024.5(x-103.7)^2}{2} - 5.568\times10^6 = 0$

x = 68.09 mm

Sub x = 68.09 mm in equation 3 and neglecting negative terms in bracket

EI $y_{max} = -371\times10^6$

Therefore ,

$y_{max} = \frac{-371\times10^6}{20510^3\times\frac{\pi}{64}\times(56.3^4-35.3^4)}$

$y_{max} = -4.34\times10^{-3}$ mm

The maximum deflection is found to be – 0.00434 mm

3.2 Torsional Analysis

$\frac{T}{J} = \frac{G\theta}{L} = \frac{\tau}{R}$

$\tau_{nom} = \frac{T}{J} R$ (4)

Where

T= Torque in N-m

J = Polar moment of inertia in mm^4

R = Radius of the shaft in meter

$\tau_{nom} = \frac{200}{\frac{\pi}{32}(0.0515^4-0.0353^4)} \times 0.02575$

$\tau_{nom} = 9.569\ MPa$

$\tau_{max} = Kt \times \tau_{nom}$

τ_{nom} = nominal shear stress MPa

Where,

Kt = stress concentration factor for torsion

Kb = stress concentration factor in bending.

From Standard Design Data Handbook page, no 158 [4]

For $\frac{r}{d} = 0.03$

$\frac{D}{d} = 1.15$

$K_b = 2.8$

Kt = 2

$\tau_{max} = 2 * 9.569$ MPa

$\tau_{max} = 19.13$ MPa

Angle of twist

$\theta = \frac{TL}{GJ} = \frac{200\times0.193}{81\times10^9\times8.130\times10^{-7}}$

$\theta = 5.86\times10^{-4}$ degree

3.3 Equivalent von Mises stress

$\sigma_{max} = K_b \times \sigma_{nom}$

σ_{max} = Maximum bending stress (MPa)

$\sigma_{nom} = \frac{32M}{\pi(do^3-di^3)}$ (5)

$\sigma_{max} = 2.8 \times \frac{32\times192.92}{\pi(0.056^3-0.0353^3)} = 41.7$ MPa

$\tau_{max} = K_t \times \frac{16T}{\pi(do^3-di^3)}$ (6)

τ_{max} = Maximum shear stress (MPa)

$\tau_{max} = 2 \times \frac{16*200}{\pi(0.056^3-0.0353^3)} = 15.15$MPa

$\sigma_1 = \frac{\sigma_{max}}{2} + \sqrt{\frac{1}{2}(\sigma_{max})^2 + 4(\tau_{max})^2}$ (7)

$\sigma_2 = \frac{\sigma_{max}}{2} - \sqrt{\frac{1}{2}(\sigma_{max})^2 + 4(\tau_{max})^2}$ (8)

Where,

σ_1, σ_2 = Principal stresses MPa

$\sigma_1 = \frac{41.7}{2} + \sqrt{\frac{1}{2}(41.7)^2 + 4(15.15)^2} = 62.73$ MPa

$\sigma_2 = \frac{41.7}{2} - \sqrt{\frac{1}{2}(41.7)^2 + 4(15.15)^2}$ = - 21.43 MPa

According to von Mises criteria for ductile material

$\sigma_{eq} = \sqrt{\sigma_1{}^2 + \sigma_2{}^2 - \sigma_1\sigma_2}$

σ_{eq} = Equivalent von Mises stress (MPa)

$\sigma_{eq} = \sqrt{62.73^2 + (-21.43)^2 - (62.73 \times -21.43)}$

= 75.75 MPa $< \sigma_{all}$

$\sigma_{all} = \frac{S_y}{FOS}$ (9)

σ_{all} = Allowable stress (MPa)

Sy = yield strength for SAE 8620 = 390 MPa

Assuming FOS = 2

$\sigma_{all} = \frac{390}{2}$ = 195 MPa

Equivalent von Mises stress is less than allowable stresses therefore design is safe.

4. FATIGUE ANALYSIS

For SAE 8620[5]

Ultimate tensile stress,

Sut = 659 MPa

Yield stress,

Syt = 390 MPa

Endurance limit at critical section of the shaft

$S_e = k_a \times k_b \times k_c \times k_d \times k_e \times S_e'$ (10)

Fatigue strength modification factors,

k_a = Surface Condition modification factor

k_b = Size modification factor

k_c = load modification factor

k_d = Temperature modification factor

k_e = Miscellaneous factor

S_e' = Rotary beam test specimen endurance limit

S_e' = 0.5 Sut = 329.5 MPa

k_a = a Sut ^{b} = 0.6863

From "Standard Handbook of Machine Design" [6] pg 13.12

For machined surface

a = 4.45

b = - 0.265

$k_b = 1.24\, de^{-0.107}$ = 0.8318

k_c = 1

k_d =1

k_e =1

Substituting all the values

Se = 114.37 MPa

From BMD

M_{bmax} = 192.92 N-m

M_{bmin} = 22.04 N-m

$M_{bmean} = \frac{Mbmax+Mbmin}{2}$ = 104.705 N-m (11)

$M_{b\ altenating} = \frac{Mbmax-Mbmin}{2}$ = 85.895 N-m (12)

M_{tmax} =200 N-m

M_{tmin} = 25 N-m

$\sigma_{xm} = \frac{32\,(Mb)m}{\pi*(do^3 - di^3)}$ = 11.52 MPa

$\sigma_{xa} = \frac{32\,(Mb)a}{\pi*(do^3 - di^3)}$ = 9.44 MPa

$\tau_{x\,m} = \frac{16\,(Mt)m}{\pi*(do^3 - di^3)}$ = 11.48 MPa

$\tau_{x\,a} = \frac{16\,(Mt)a}{\pi*(do^3 - di^3)}$ = 1.436 MPa

Equivalent Stresses

Stress concentration for bending= Kb = 2.8

Kt = 2

$\sigma_m = \sqrt{(\sigma xm \times Kb)^2 + (\tau xm \times Kt)^2}$ = 51.20 MPa

$\sigma_a = \sqrt{(\sigma xa \times Kb)^2 + (\tau xa \times Kt)^2} = 26.89$ MPa

$\tan\theta = \frac{\sigma m}{\sigma a} = \frac{Sa}{Sm} = 0.525$

According to modified Goodman equation

$$\frac{Sa}{Se} + \frac{Sm}{Syt} = 1 \qquad (13)$$

Substituting Sa= 0.525×Sm

Sm = 139.77 MPa

Sa = 73.37 MPa

$FOS = \frac{Sa}{\sigma a} = 2.72$

$$Sf = \frac{\sigma a \times Sut}{Sut - \sigma m} = 29.15 \text{ MPa} \qquad (14)$$

$$\log_{10}N = \frac{(6-3)\times[\log_{10}(0.9\times Sut) - \log_{10}(Sf)]}{[\log_{10}(0.9\times Sut) - \log_{10}(Se)]} + 3 \qquad (15)$$

$N = 2.87 \times 10^8$ Cycles > 10^6 Cycles

4.1 Shaft Diameter

According to ASME code for Shaft design [7]

$$d = \left\{\frac{16n}{\pi} \times \left[\frac{2\times K_f \times M}{Se} + \frac{[3(K_{fs}\times T)^2]^{\frac{1}{2}}}{Sut}\right]\right\}^{\frac{1}{3}} \qquad (16)$$

Where

n = Factor of safety

K_f = stress concentration factor in bending

K_{fs} = stress concentration factor in torsion

From table 7.1 page 361" Mechanical Engineering Design "[7]

For r/d = 0.02 K_f = 2.8 and K_{fs} = 2.2

$$d = \left\{\frac{16\times 2}{\pi} \times \left[\frac{2\times 2.8\times 192.92}{114.3\times 10^6} + \frac{[3(2.2\times 200)^2]^{\frac{1}{2}}}{659\times 10^6}\right]\right\}^{\frac{1}{3}}$$

d = 43.27 mm

Minimum diameter required for given loading condition = 43.27 mm

Actual diameter of the shaft = 56.3 mm

$d_{actual} > d_{min}$ therefore design is safe.

5. MODAL ANALYSIS

Free-Free modal Analysis

For free-free condition Natural frequency is given by

$$\omega^2 = (\beta l)^2 \sqrt{\frac{EI}{\rho A l}} \qquad (17)$$

Where,

βl = Constant

From Textbook Mechanical Vibrations by S.S Rao[8] Pg. 726 Fig 8.15

$\beta l = 4.73004$

$A = \frac{\pi}{4} \times d^2 = 0.002463 m^2$

ρ = Density of material in Kg/m^3

$I = \frac{\pi}{64} \times d^2 = 4.83 \times 10^{-7}$

Substituting all the values in above equation

ω = 42690.39 rad/s

f = 6793 Hz

5.1 Critical speed of the shaft

Influence coefficients

An influence coefficient is the transverse deflection at location i on a shaft due to a unit load at location j on the shaft.

$$\delta_{ij} = \begin{cases} \frac{b_j x_i}{6\,EIl}\left(l^2 - b_j^2 - x_i^2\right) & x_i \le a_i \\ \frac{a_j(l-x_i)}{6\,EIl}\left(2lx_i - a_j^2 - x_i^2\right) & x_i > a_i \end{cases} \qquad (18)$$

$I = \frac{\pi}{64} \times d^2 = \frac{\pi}{64} \times (56.3^4 - 35.3^4) = 416.95\times 10^3 \text{ mm}^4$

$6EIL = 6\times 205\times 10^3 \times 416.95\times 10^3 \times 0.1935 = 5.459\times 10^{13}$

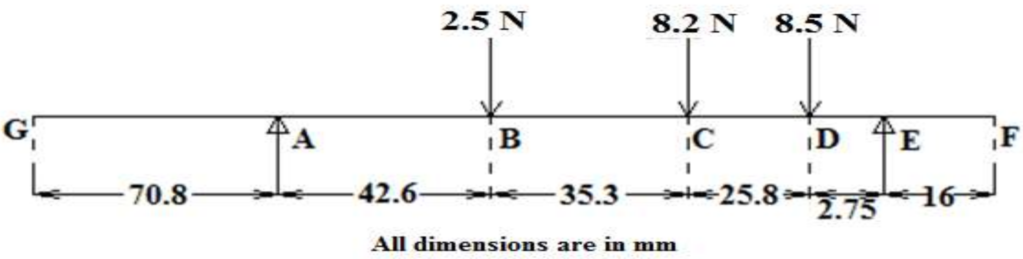

Fig 4: FBD of mass of rotors acting on the shaft

A excel program was written to find the influence coefficients.

δ_{11} = 2.71×10^{-7} m

δ_{22} = 1.81×10^{-7} m

δ_{33} = 2.98 ×10^{-9} m

δ_{12} = 1.94 ×10^{-7} m = δ_{21}

δ_{13} = 2.04 ×10^{-8} m = δ_{31}

δ_{23} = 2.06 ×10^{-8} m = δ_{32}

$$\delta_{ij} = \begin{bmatrix} 2.71 \times 10^{-7} & 1.94 \times 10^{-7} & 2.04 \times 10^{-8} \\ 1.94 \times 10^{-7} & 1.81 \times 10^{-7} & 2.06 \times 10^{-8} \\ 2.04 \times 10^{-8} & 2.06 \times 10^{-8} & 2.98 \times 10^{-9} \end{bmatrix}$$

According to Dun Kerley's method

$$\frac{1}{\omega^2} = \frac{W1}{g1} \times \delta_{11} + \frac{W2}{g2} \times \delta_{12} + \frac{W3}{g3} \times \delta_{13} \qquad (19)$$

Substituting all the values

ω = 2004.52 rad/s

$\omega = \frac{2\pi N_c}{60}$

N_c = 19148.84 rpm

The operating speed of the shaft is 8500 rpm. We can operate the shaft is up to 19148 rpm.

6. FINITE ELEMENT ANALYSIS

6.1 Steps in FEA

1. Pre-processing

Pre-processing includes

- CAD data
- Meshing
- Boundary conditions

2. Processing

Processing is a solution step. The computer solves complex mathematical equation to give user defined solutions. Internally software carries out functions such as matrix formation, inversion, multiplication etc. to give solution

3. Post-processing

Post-processing is viewing results, verifications, conclusions and thinking what steps to be taken to improve the design

6.2 Geometry

Fig 5: Shaft Geometry

The output shaft of speed reduction assembly is multiple stepped shaft with splines to transmit torque. Shaft is checked for errors before importing in to ANSYS. The output shaft geometry is imported in to ANSYS workbench in step format.

6.3 Material Properties

SAE 8620 steel is a low alloy nickel, chromium, molybdenum case hardening steel, SAE steel 8620 offers high external strength and good internal strength, making it highly wear resistant. Chemical composition of SAE 8620 is Carbon 0.20%, Silicon 0.25%,

Manganese 0.80%, Chromium 0.50%, Nickel 0.55%, Molybdenum 0.20%. Typical applications: Arbors, pinions, bushes, camshafts, kingpins, ratchets, gears, splined shafts etc.

Table 1 Material Properties of SAE 8620[9]

Sl.No	Properties	Values
1.	Density	7872 kg/m^3
2.	Young's Modulus	205 GPa
3.	Poisson's ratio	0.3
4.	Tensile yield strength	357 MPa
5.	Ultimate tensile strength	659 MPa

6.4Meshing

Fig 6:Meshed model

Auto mesh is used for meshing. This approach is used for simple geometries and the pre-requisites is a error free CAD model. The user just has to select the volume and the software automatically carries out the meshing as per the specified element length, quality criteria etc. The component is meshed with tetra elements of size 3mm

No. of Elements = 114469

No. of nodes = 193521

6.5 Boundary Conditions

Fig 7: Boundary Conditions

The shaft is supported by ball bearings at the rear end and bushing at the front. The bearings and bushings are represented by cylindrical support which is axially fixed and radially free. A torque of 200 N-m which is transmitted from input shaft acts at point B Weight of dog cone i.e. 8.2 N acts on the shaft at point C.

6.6 Total Deformation

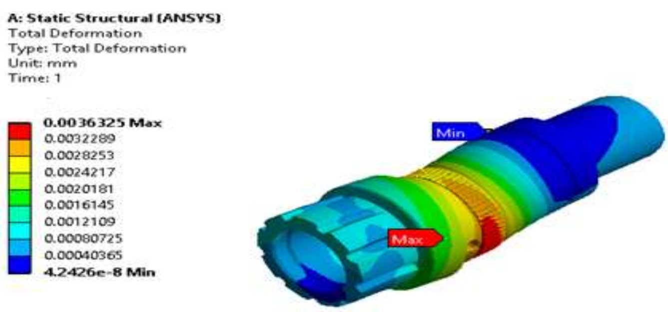

Fig 8: Total Deformation

Maximum deflection of 3.63×10^{-3} mm is observed near the splines. It is evident from Bending Moment diagram that maximum moment takes place at same point.

6.7 Equivalent von Mises stress

Fig 9: Equivalent von Mises Stress

Maximum equivalent von Mises stress is observed near the stepped region of the shaft. Maximum equivalent von Mises stress is 76.48 MPa and the allowable stress is 195 MPa therefore the design is safe.

6.8 Maximum Shear Stress

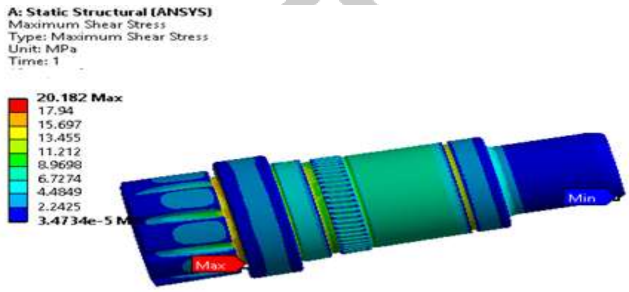

Fig 10: Maximum Shear Stress

Maximum shear stress of 20.182 MPa is found near groove of the shaft. Maximum shear stress is less than allowable shear stress i.e 101.74 MPa therefore design is safe.

6.9 Life

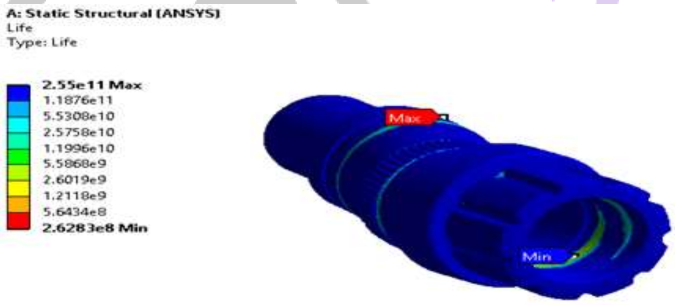

Fig 11: Fatigue Life

It is the available life for given for given fatigue analysis. Fatigue life can be over whole model or parts, surface, edges. The life of the shaft is found to be 2.6283×10^{8} cycles therefore the shaft will sustain infinite life cycles. Minimum life is observed near the stepped region on inner diameter as shown in figure above. The shaft will fail at that point after 2.6283×10^{8} cycles. In order to increase the life sharp step should be avoided.

6.10 Safety factor

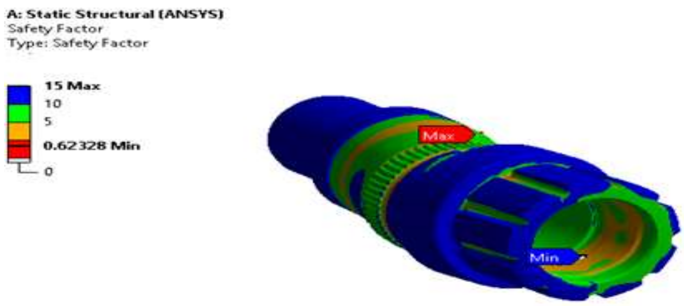

Fig 12: Safety Factor

Safety factor is the factor of safety with respect to a fatigue failure at a given design life. Safety factor may define as the ratio of yield stress to the working stress. Minimum safety factor of 0.623 is observed near stepped regions should be increased at that point to avoid failure and increase the life.

6.11 Free-Free Modal Analysis

For free free modal analysis, the natural frequency is calculated from equation

$$m\ddot{x} + Kx = 0$$

No external force is applied to the system. In free free analysis the first six frequencies will be almost zero from seventh onwards natural frequency value will be obtained

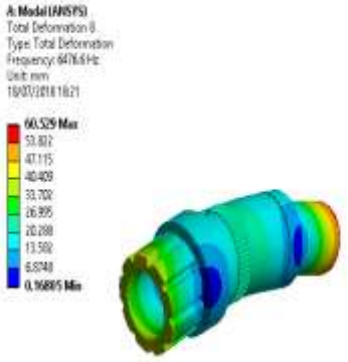

Fig 13: 1st mode

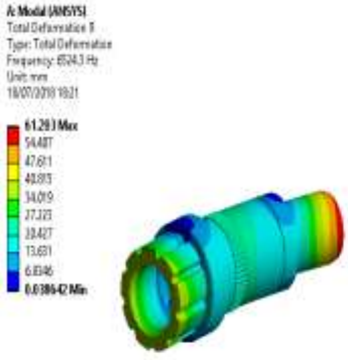

Fig 14: 2nd mode

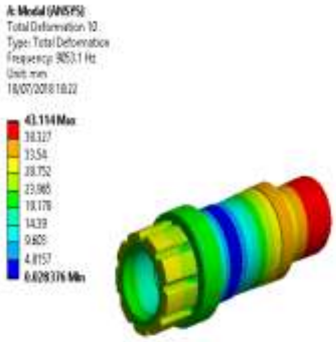

Fig 15: 3rd mode

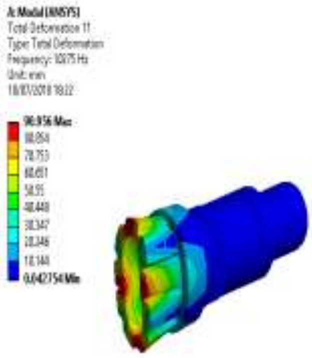

Fig 16: 4th mode

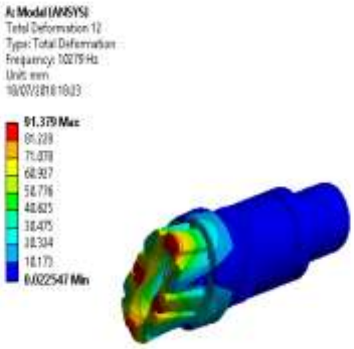

Fig 17: 5th mode

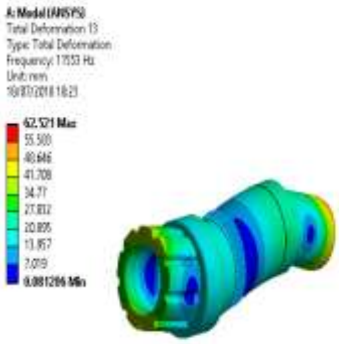

Fig 18: 6th mode

6.12 Forced Modal Analysis

In forced modal analysis the effect of force acting on the system is` considered. The governing equation for forced vibration is

$$m\ddot{x} + c\dot{x} + Kx = F(t)$$

Loading and boundary condition used for Forced modal analysis is same as static structural analysis.

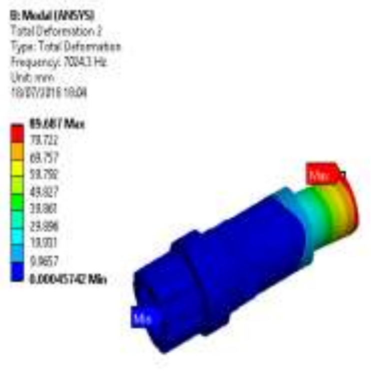

Fig 19: 1st mode

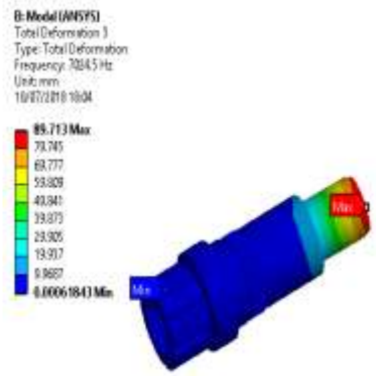

Fig 20: 2nd mode

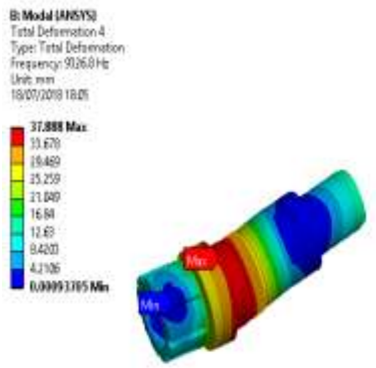

Fig 21: 3rd mode

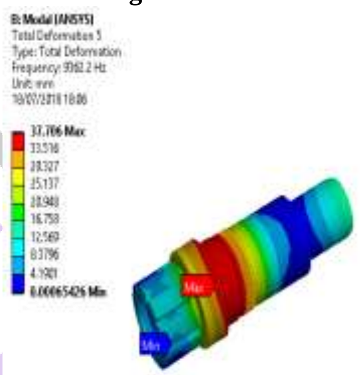

Fig 22: 4th mode

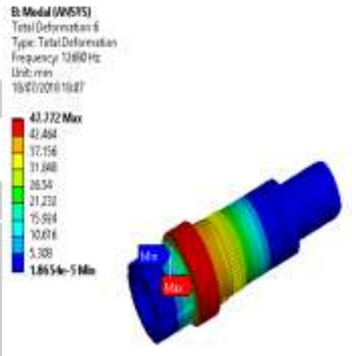

Fig 23: 5th mode

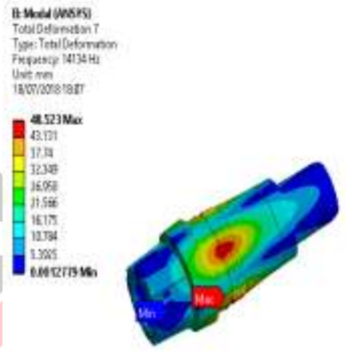

Fig 24: 6th mode

7. RESULTS AND DISCUSSION

Table 2: Analytical and ANSYS Results

	Analytical	ANSYS
Equivalent von Mises stress	75.75 MPa	75.87 MPa
Deflection	4.34×10^{-3} mm	3.59×10^{-3} mm
Maximum Shear stress	19.13 MPa	20.182 MPa
Fatigue life	2.87×10^{8} cycles	2.62×10^{8} cycles
Natural frequency (Free-Free Modal Analysis)	6793 Hz	6476Hz

- Maximum equivalent von Mises stress is 75.87 MPa and the allowable stress is 195 MPa therefore the design is safe.
- Maximum shear stress i.e. 19.13 MPa is less than allowable shear stress i.e. 101.74 MPa therefore design is safe.
- The angle of twist is found to be 5.86×10^{-4} degree.
- The minimum safest diameter for given loading condition is 43.27 mm and the actual diameter is 56.3 mm. Therefore, design is safe.
- The natural frequency of forced vibration is found from ANSYS and the value is found to be 7615.5 Hz
- The Critical speed of the shaft is determined by using Dun Kerley's method and the value is 19148.84 rpm. The operating speed of the shaft is 8500 rpm.

7.1Torque v/s Speed

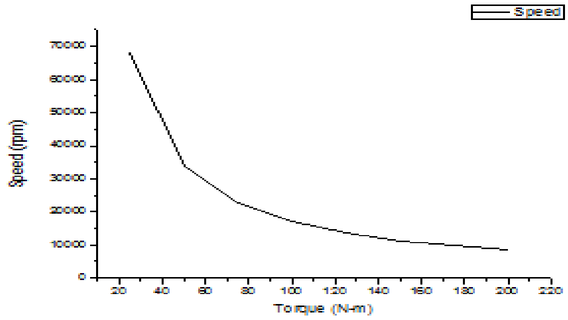

Fig 25: Torque vs Speed

The above plot shows the variation of torque v/s speed. Torque varies inversely wrt speed. Speed reduction is mainly required for higher torque demands such as off-roading or driving the vehicle uphill.

7.2 Torque v/s Equivalent von Mises stress

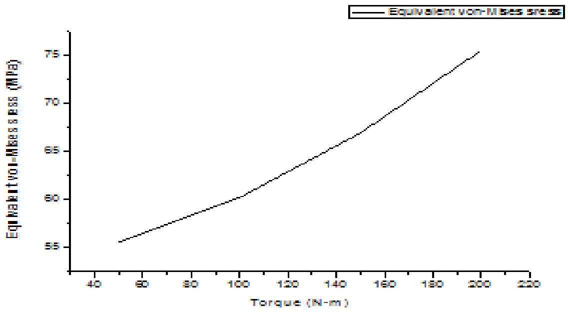

Fig 26: Torque v/s Equivalent von Mises stress

The above plot shows the variation of torque v/s Equivalent von Mises stress. The curve varies linearly. The maximum Equivalent von Mises stress was observed is 75.87 MPa.

7.3Torque v/s Maximum shear stress

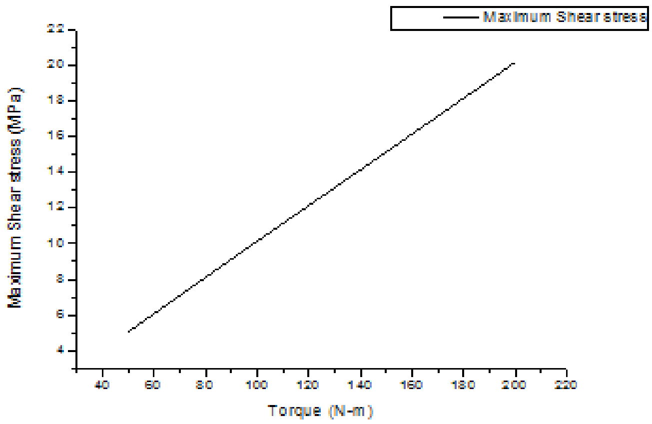

Fig 27: Torque vs Maximum Shear stress

The above plot shows the variation of torque v/s Maximum shear stress. The variation of torque w.r.t Shear stress is linear. The maximum shear stress is found to be 20.182 MPa

7.4 Modes v/s Frequency for free-free Modal analysis

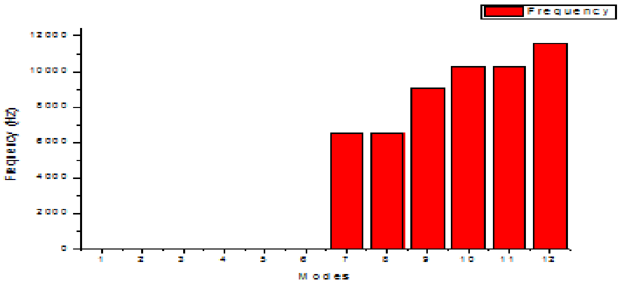

Fig 28: Modes vs Frequency for free free modal analysis

The above plot shows the variation of modes v/s natural frequency for free -free modal analysis. The natural frequency for first six mode is 0 it is also called rigid mode or mechanism mode. After 7th mode onwards we get deformable modes. The natural frequency at 7th mode is observed as 6476 HZ.

7.5 Modes v/s Frequency for Forced Modal analysis

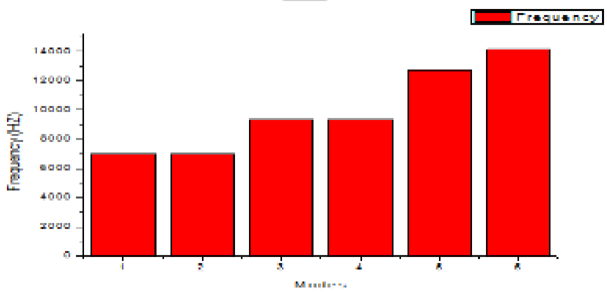

Fig 29: Modes vs Frequency for forced modal analysis

The above plot shows the variation of modes v/s natural frequency for forced modal analysis. Operating frequency is ($\omega = 2\pi N/60 = 141.6$ Hz), when operating frequency matches with natural frequency resonance occurs. The frequency in this case is higher than operating frequency therefore design is safe.

8. CONCLUSIONS

1. Equations are presented for shaft with combined loading conditions
2. Static structural, Fatigue, Modal, Torsional analysis are carried out using ANSYS 18.1
3. Equivalent von Mises stress, deflection, Maximum Shear stress, Angle of twist, Fatigue life, Natural frequency, Critical speed is calculated using analytical method and the results are compared with ANSYS results.
4. Stresses calculated at critical sections are within safe limits.
5. Fatigue analysis shows that the shaft can sustain for infinite life cycles.
6. From Fatigue analysis we can observe that the shaft will fail near the stepped region on inner diameter of splines. Hence suitable corrective action should be taken to increase the life of the shaft.
7. Critical frequency calculations show that the shaft can be used up to 19148 rpm.
8. From analysis and analytical calculation, it is observed that the von Mises stress is far less than the material yield stress, therefore optimization can be carried out.

REFERENCES

[1]. M.D.Shrotriya, P. J. Awasare, R.P.Hirurkar "Design and Analysis of a Mechanical Component (Shaft) in the Alternator" International Engineering Research Journal pp 826-830
[2]. Sudhanshu Mishra , Shani Kumar, Sangam Kumar,Sumit Verma "Designing and Groove Shaft Analysis under Different Loading Condition" International Journal for Research in Applied Science & Engineering Technology(IJRASET) Volume 5 May 2017, Issue V, ISSN: 2321-9653
[3]. B. Engel, Sara Salman Hassan Al-Maeeni Failure Analysis and Fatigue Life Estimation of a Shaft of a Rotary Draw Bending Machine" Vol11, 2017
[4]. Machine Design Handbook McGraw Hill Publications.
[5]. R. A. Gujar, S. V. Bhaskar "Shaft Design under Fatigue Loading By Using Modified Goodman Method" International Journal of Engineering Research and Applications" Vol. 3, Jul-Aug 2013 ISSN: 2248-9622
[6]. Joseph.E.Shigley. and CharlesR Mischke. "Standard Handbook of Machine Design", second edition (Tata McGraw Hill, New Delhi, India).
[7]. Richard G. Budynas, J. Keith Nisbett "Shigley's Mechanical Engineering Design", ninth edition (Tata McGraw Hill, New Delhi, India).
[8]. Singiresu S. Rao "Mechanical Vibrations" fifth edition (Prentice Hall)
[9]. Results and discussions shodhganga. Inflibnet.ac.in >bitstream