

# Design and Analysis of Driven Shaft for Automobile

G.Raghavendra Rao<sup>1</sup>, G.Maruthi Prasad Yadav<sup>2</sup>

M.Tech (Cad/Cam), Chadalawada Ramanamma Engineering College, Tirupati.

Principal & Professor, Chadalawada Venkata Subbaiah Engineering College,  
Department Of Mechanical Engineering Tirupati.

## Abstract

Drive shafts in commercial heavy duty vehicle are subjected to cyclic loads due to variation in the torque demanded by the varying road loads. Cases of failure of the drive shaft of heavy duty trucks have been reported, which provides an opportunity to investigate the issue.

In this project an attempt has been made on prediction of fatigue life of the drive shaft using FEA technique. The load spectrum acting on the drive shaft is calculated for specified terrain. Structural analysis was carried out on the FE model of drive shaft, and potential areas of high stress concentration are obtained. Von Mises stress, fatigue damage, fatigue life, factor of safety and total deformation are the results of fatigue analysis.

It was observed that from the static analysis that the roots of the splines are the areas of high stress concentration. The fatigue analysis reveals that the drive shaft fails in the region of high stress concentration as expected. The fatigue life of the component is found to be infinite for design pay load. The life of the drive shaft for 40% and 50% over load is 69806 kilometers and 20564 kilometers respectively. The present work can be used in automobile industry as a reference for predicting life of any component, in the drive line of the automobile subjected to non constant amplitude loading.

**Keywords:** Fatigue, Cumulative Damage, Stress-life, Duty cycle, Numerical Fatigue Analysis, Palmgren-Miner Linear damage rule, Rain-flow cycle counting.

## 1. INTRODUCTION

The focus of this project is to predict the fatigue life of drive shaft of a heavy duty truck that fails during the service period. Fatigue, one out of many, is the cause of failure. Fatigue life of the drive shaft is predicted for the design load and for other cases of 10, 20, 30 40 and 50 percent overload are also calculated.

## 2. PROBLEM STATEMENT

Predicting the fatigue life of Drive Shaft of a heavy duty truck from a duty cycle calculated for two kilometers. Several cases of drive shaft failure of TATA 1613 truck have been reported in the service station during a span of 2-years. Figure 1 shows the failed drive shaft of the truck and Table 1. shows date and kilometers, the vehicle had done when cases were reported. Visual examination of the component reveals the presence of symptoms, “beach marks” or “striations”, of fatigue failure. Of all the components inspected some failed due to over load which can be made out from the brittle fracture of the shaft.

Kilometre reading and date on which failure have been reported

S No	Kilometer Reading	Complaint Date	Sale Date
1	30,384	6/5/2007	4/04/2006
2	75006	9/6/2007	29/05/2006
3	10010	16/05/2007	24/02/2007
4	44843	29/05/2007	20/03/2006
5	44001	27/09/2006	22/12/2005
6	9012	19/12/2006	11/8/2006

## 3. OBJECTIVES

The objectives of the project are

- To review the literature on fatigue analysis and to set a stage for problem formulation
- To study relevant analytical models for fatigue of automotive drive shafts that are available in the literature
- To create geometric and FE model of the drive shaft for fatigue analysis
- To perform static analysis and to identify the stress concentration area of the drive shaft

- To predict the fatigue life of the drive shaft subjected to various load from the road load data

#### 4. METHODOLOGY

Methodology adapted to achieve the objectives

- Literature review for fatigue failure analysis is carried out by referring journals, books, manuals, technical papers and related documents
- Relevant analytical model for fatigue analysis are studied
- The geometric model of the drive shaft is created by using CATIA
- Duty cycle is calculated from road topology
- Stress analysis of the drive shaft is carried out using ANSYS Workbench
- Fatigue analysis of the drive shaft is carried out using ANSYS Workbench
- Fatigue life of the drive shaft is predicted using ANSYS Workbench

#### 5. SOFTWARES USED

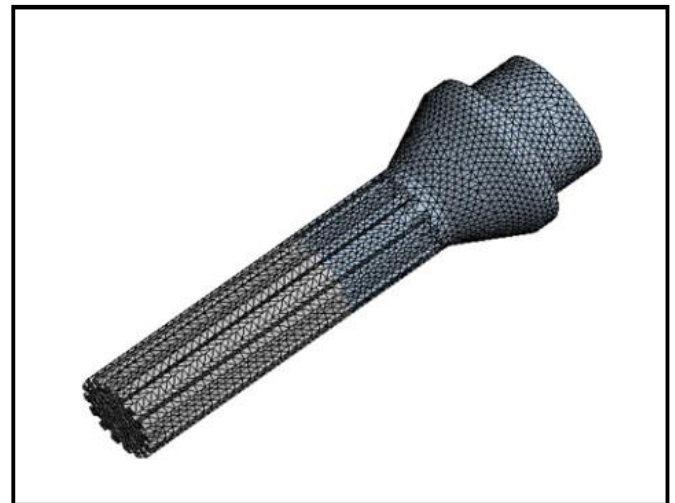
- CATIA
- ANSYS Workbench

##### Technical specifications of the vehicle

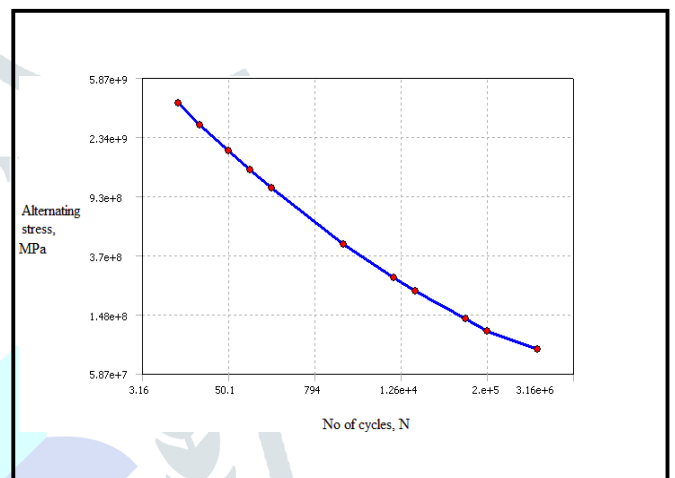
<b>ENGINE</b>	
Model	697 NA
Type	Water cooled, direct injection diesel engine
No. of Cylinders	6 inline
Bore / Stroke	97mm x 128mm
Capacity	5675 cc
Max. Engine Output (as per DIN 70020)	130PS at 2800 rpm
Max torque	37mkg.(363 Nm)at 1800-2000rpm
<b>GEAR BOX</b>	
Model	TATA GBS-40
Type	Synchromesh on all forward gears
No. of Gears	5 forward and one reverse
Gear ratios	1st - 8.98
	2nd - 3.99
	3rd - 2.50
	4th - 1.51
	5th - 1.00
	Reverse - 6.93
Ratio(final drive ratio)	41/6

##### Mesh details.

Element Type / Shape	Solid 45 / Tetrahedrons
Elements Size	5.e-003 m
No. of Elements	33158
No. of Nodes	57743



Finite Element model of drive shaft



S-N curve of structural steel

##### Material properties of structural steel.

<b>Structural Properties</b>	
Material	Structural Steel
Volume	5.9968e-004 m <sup>3</sup>
Mass	4.7075 kg
Young's Modulus	2.e+011 Pa
Poisson's Ratio	0.3
Density	7850. kg/m <sup>3</sup>
Thermal Expansion	1.2e-005 1/°C
Tensile Yield Strength	2.5e+008 Pa
Compressive Yield Strength	2.5e+008 Pa
Tensile Ultimate Strength	4.6e+008 Pa

##### Calculation of duty cycle

The longitudinal section of a road profile measuring 2 kilometres, between Gaddankerri and Semekerri, location of which is shown in the Figure 15 is obtained [14] for the calculation of duty cycle. The velocity of the truck is calculated by equating rolling resistance, air resistance and gradient resistance of the road selected to the rated power of engine with a transmission efficiency of 0.98.

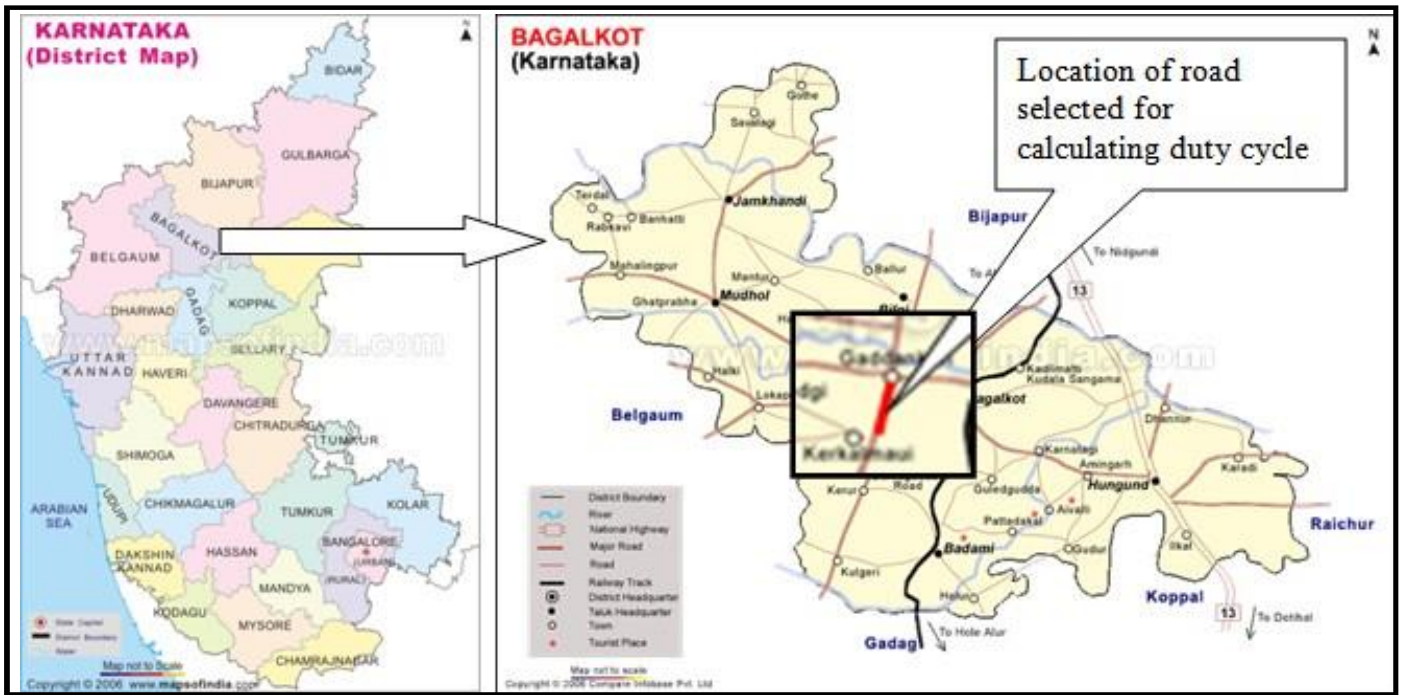


Figure 1: Location of road selected for calculating duty cycle.

Figure 15 shows the elevation (longitudinal section) of road with horizontal distance from a reference point. Details of

longitudinal section of road are added at the end of the report.

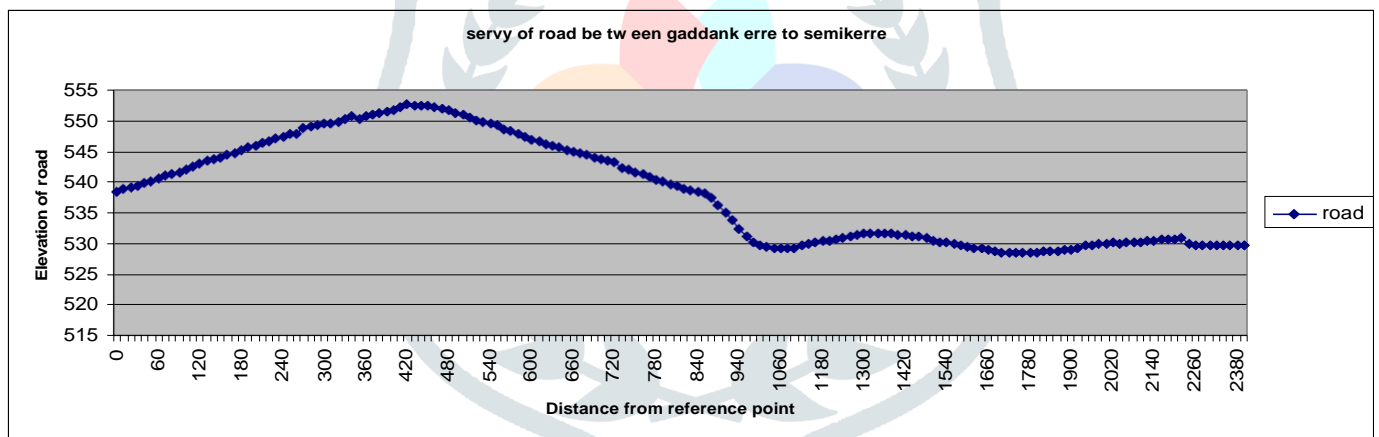


Figure 2: Longitudinal section of the road

**Following are the assumption made during calculations**

The frontal projected area of the truck is  $4 m^2$

- The rolling resistance is constant
- The positive gradient (i.e. when the truck is ascending a hill) is added to the total resistance and negative gradient (i.e. when the truck is descending a hill) is subtracted from the total resistance
- The truck is always supplying rated power all the time i.e. maximum power of the engine.
- A representative value of 0.015 is used for the constant of rolling resistance.
- A value of 0.045 is used for the co-efficient of air resistance for trucks.
- Transmission efficiency is 0.98%

- When full power of engine is supplied to the vehicle down the slope, the velocity of the vehicle exceeds its maximum (78 kmph) as calculated. During such cases the vehicle velocity is assumed to be 60 kmph.

**Calculations of vehicle velocity**

The calculation of vehicle velocity, sited is for first span of 10 meters starting from the reference point, with a vertical gradient of 0.335 meter. On the same guidelines velocity is calculated for whole road.

$$Power = Velocity \times totalresistance$$

$$total\ resistance = R_a + R_g + R_r$$

Rolling resistance,  $R_r = (a + Vb)W$   
 $= (0.015 + 0.00016V)16200 \times 9.81$   
 $= \frac{2354.4 \times 1000}{3600} + \frac{25.1136V \times 1000}{3600}$   
 $= 662.175 + 7.0632V$

The value of  $a = 0.015$  a representative value and  $b = 0.00016$  are considered [13]

Air resistance,  $R_a = K_a AV^2$   
 $= 0.045 \times 4 \times V^2$   
 $= 2.32 \times V^2$

$K_a = 0.045$  is used for trucks and lorries.

Gradient resistance,  $R_g = m \times g \times \sin \theta$   
 $= 16200 \times 9.81 \times 0.93$   
 $= 149336.9608 \text{ N}$

$95 = V \times (149336.96 + 2.32V^2 + 662.175 + 7.0632V)$

The roots for the cubic equation are solved in hand calculation in MS Excel. The imaginary roots are neglected and integer values are taken into consideration. Hence, for the above equation the roots are  $-8.62 + 52.43i$ ,  $-8.62 - 52.43i$ ,  $14.12$ . Similarly, the roots for all the equations are solved. Appendix shows values of all the roots for design and overload cases. Once the value of vehicle velocity is calculated, the linear velocity of vehicle is co-related to wheel speed using the relation

$V = r \omega$

where,  $\omega = \frac{2 \times \pi \times N}{60}$

From the above equation the speed with which the axle shaft is calculated

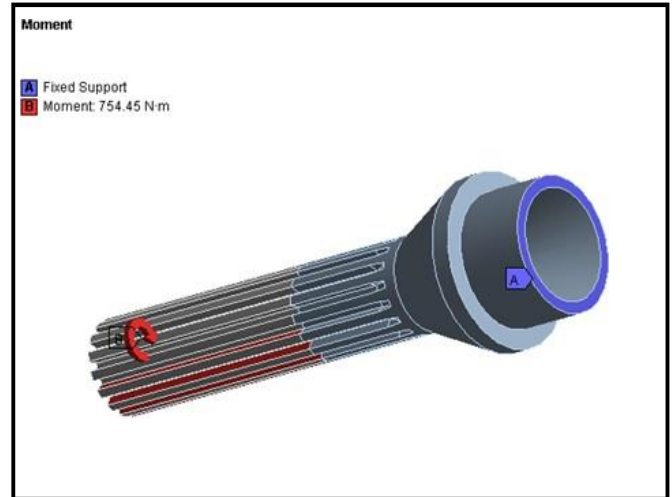
$N = \frac{V \times 60}{2 \times \pi \times r} = \frac{14.4611 \times 60}{2 \times \pi \times 0.55} = 251.05 \text{ r.p.m.}$

Since, it is assumed that the transmission efficiency of the driveline is 0.98, out of 95Kw power available at the engine, 93Kw is transmitted through drive shaft. Hence, torque transmitted by drive shaft is calculated by the relation

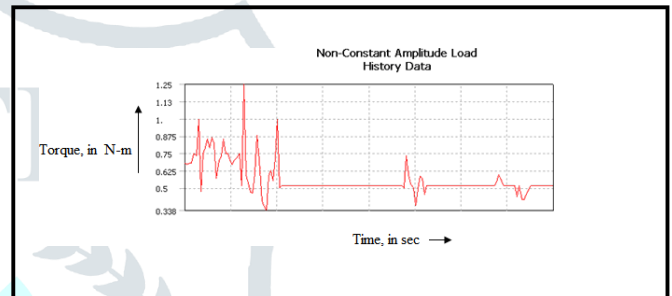
$T = \frac{P \times 60}{2 \times \pi \times N} = \frac{93000 \times 60}{2 \times 3.142 \times 251.05} = 520.16 \text{ N-m}$

**Details of boundary conditions**

There are 16 teeth on the drive shaft milling. By visual inspection it is evident that there is uniform wear on all teeth. Hence, the torque is transmitted by all the teeth. The calculated torque is applied on all the teeth, i.e. along side of teeth face. Since, the drive shaft tube is fastened to the milling all degrees of freedom are constrained on this face. Figure 16 shows the boundary conditions applied on the FE model.



Boundary conditions imposed

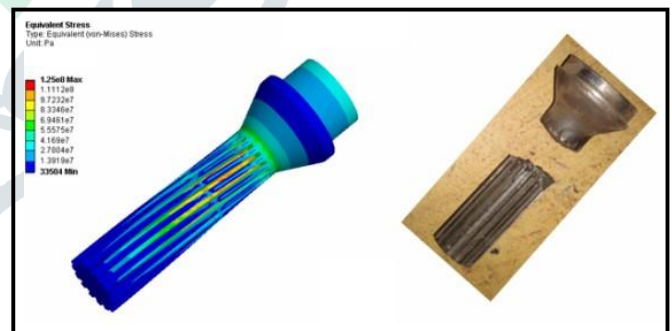


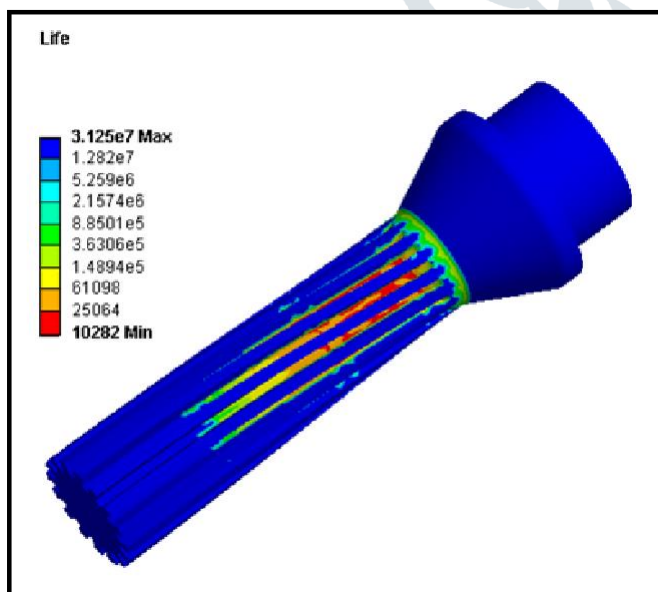
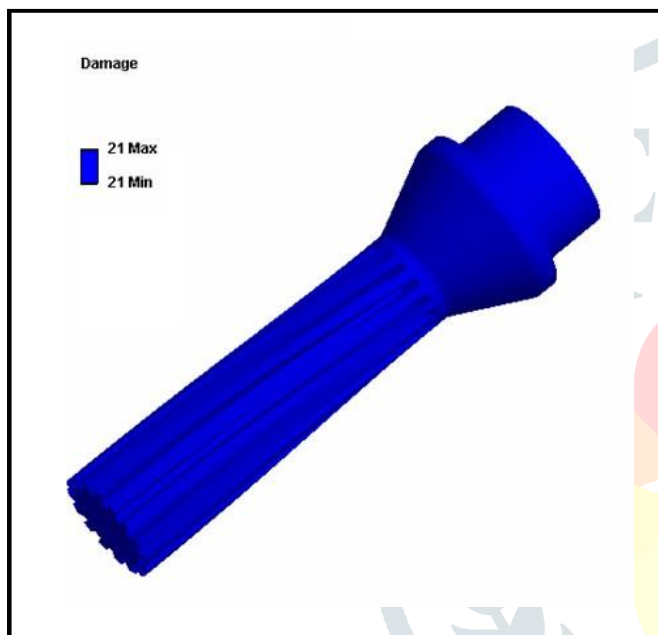
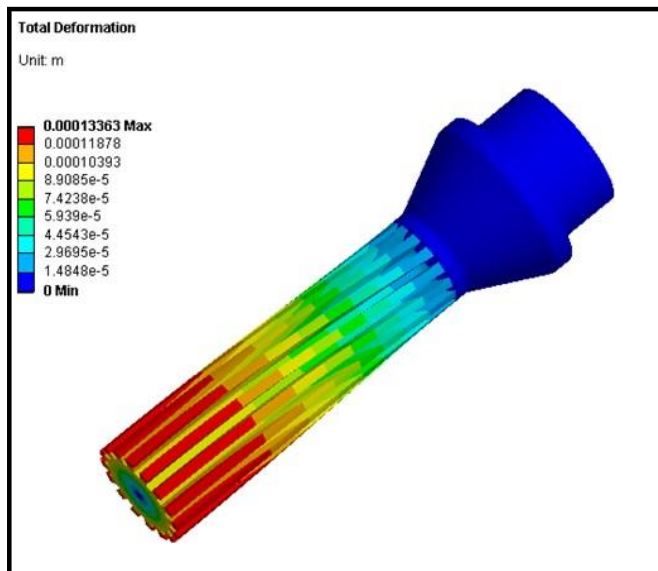
Load-time history

**6. RESULT**

The following results are discussed

- Fatigue life
- Fatigue damage at a specified design life
- Fatigue factor of safety at a specified design life
- Rain flow matrix output





## CONCLUSION

The present work has resulted in various conclusions and also introduced scope for future work. The various conclusions that can be drawn from the project are listed below.

- Accurate representation of dimensions of the component/s under analysis by adopting the reverse engineering technique like co-ordinate measuring machine or laser scanner
- Minimizing discretization error by remeshing of fillets
- Realistic representation of the load time history experienced by the component
- Static stress analysis reveals that the roots of the splines of the drive shaft are the area of maximum stress concentration. The maximum stress experienced by the drive shaft for the specified terrain and for the design load is 125 MPa
- The total deformation of the drive shaft is 0.13 mm
- The factor of safety for the drive shaft is more than 1. Hence, the drive shaft is safe.
- For design load, life of the drive shaft is infinite For 10, 20 and 30 % overloading the life of drive shaft is infinite. For 40% overloading the life of drive shaft is finite and 69806 kilometers. For 50% overloading the life of drive shaft is finite and 20564 kilometers
- The life of the drive shaft is different for different duty cycle.
- Over all the cause for fatigue failure of the drive shaft is due to overloading.

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