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Fatigue life evaluvation of an automobile Front Axle

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Abstract: An axle is central shaft for a rotating wheel or gear. On wheeled vehicles, the axle may be fixed to the wheels, rotating with them or fixed to its surroundings, the wheels rotating around the axle. Is one of the most important parts of a vehicle. Present study is to focus on mechanical integrity and life evolution of front axle using FEA approach, blending the classical approach for preliminary design considerations and loading conditions, including Gross weight of the vehicle, Inertial loads Gradient resistance and rolling resistance. Consider the selected front axle is proved to be beam of uniform strength. And Customized methodology of analysis through sub modelling technique, dynamic characteristics subjected to cyclic loadinggood man diagram are utilised to find life evaluation,

Index Terms— Frontal axle, Bilinear Kinematic hardening, , Good man diagram

1 INTRODUCTION

A typical automobile front axle consists of main beam, stub axle, and swivel pin or kingpin. The wheels are mounted on stub axles. The front axle beam is subjected to bending loads due to vertical forces, while driving a truck around a corner results in multiple forces such as twisting forces on kingpin or steering knuckle, axial forces between Pad and spring interface and along the length of the beam unsymmetrical vertical loads owing to centrifugal action. Worst situation arises while a cornering truck is braked to stop, turning moment on pad and retarding force acting on the surface of the pad the study involves on the numerical simulation towards the front axle to understand comprehensively, the stress distribution in automobile front axle's , strain distribution and the vibration frequency under different varieties of conditions, and this provides the scientific theoretical base for the designer, to improves design quality, shortens design cycle and reduces cost. After years of steady, predictable model changes, the automobile industry is in the midst of the most intense product changeover in history. To accomplish the need to design a moderate automobile, the engineer essential to use imaginative concepts. Demands on the automobile designer is increasing and changing rapidly,

Mathematical modelling is logical avenue to explore. Most recently, the finite element method, a computer dependent numerical technique, has opened up a new approach to vehicle design through sensitivity analysis

Structural analysis is the determination of the effects of loads on physical structures and Structures subject to different types loads and it should withstand, such as buildings, bridges, vehicles, machinery, furniture, attire, soil strata, prostheses and biological tissue. The analysis incorporates the fields of applied mechanics, materials science and applied mathematics to compute a structure's deformations and also internal forces, stresses, support reactions, accelerations, and stability. The analysis required to verify a structure's fitness, it is necessary for engineering design of structures

Fig.1 Front Axle

LalitBhardwaj, et al., [1] has carried out static analysis of rear dead axle, The modelled the front axle and discretized into small elements and solved . Finally Von Misses stress distributions and displacement contours on rear axle are calculated for various loading conditions, ranging from 3KN to 120 KN and determine the plastic and elastic failure of the axle and observed that elastic failure of rear axle occurred at 60 KN and plastic failure happened at 120 KN. Hence it is concluded that by considering factor of safety as 3, the safe design load for rear axle is 20 KN at static condition

Kalpan Desai, [2] considered rear dead axle (Make: Maruthi 800) of a car, load distribution and stability. modelled rear dead axle considering total weight of vehicle loaded and the value of maximum bending stress calculate, for three different cross sections of axle Circular, Square and I – section and the results compared to find out the most desirable cross section. It found that maximum bending stress obtained in case of circular cross section (48.76 MPa) as compared to that of square (29.2 MPa) and I – section(33.83 MPa)., hence the value of bending stress is minimum for square cross section, concluded that I – section axle is most desirable ,since average bending stresses were evident which is slightly higher than square cross section and much lesser than circular cross section and reduction in material ,compared to square section due to less cross sectional area.

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2 Design Considerations

2.1 Combined load

Under dynamic conditions, the vertical bending moment is increased due to road roughness. front axle also experiences a horizontal bending moment because of resistance to motion The resistance to motion also causes a torque in the case of drop type front axle as shown in Fig.3, Thus the portions projected after the spring pads are subjected to combined bending and torsion

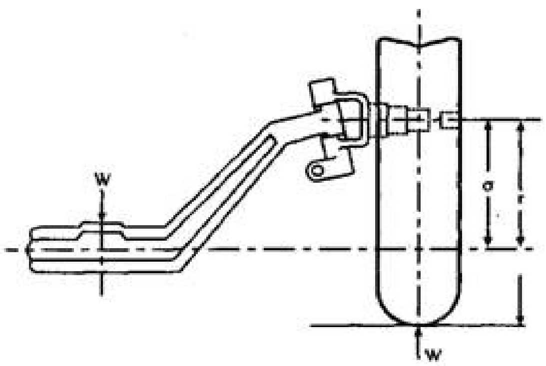

Fig.2 Combined Load acting on Front Axle

The magnitude of the

Torque = Rδ Nm

R = Resistance to motion (in Newton)

δ = Drop from spindle axis to the centre of the section (in meter)

2.2 Engine Load

It acting vertically downwards as a concentrated load. Engine load taken as reaction load in dynamic condition.

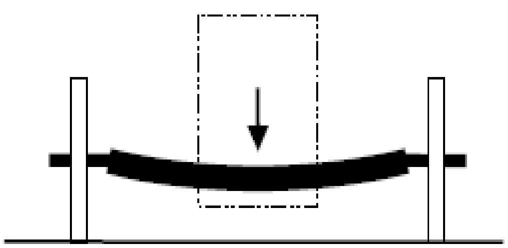

Fig.3: Load on Engine

2.3 Tension Load

Load is due to the effect of steering. Tension acts at both ends of axle where it is fixed to wheel hub.

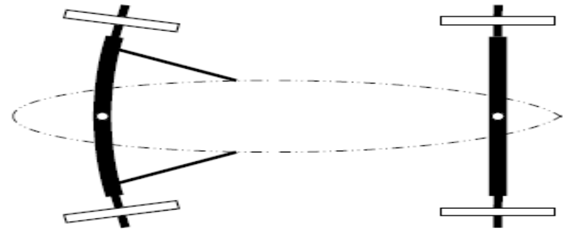

Fig.4 Tension load

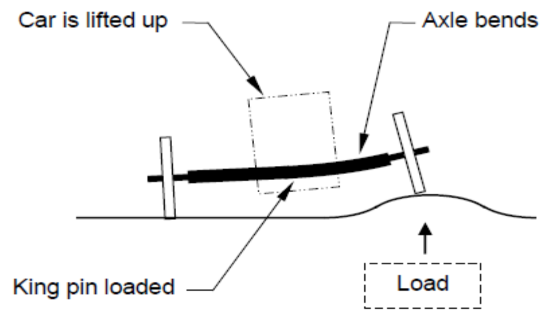

Fig.5 Impact Load

Torque

The torque acting on the axle is given by the relation

$$T = (BPx60)/2\pi N \qquad \text{N-m}$$

Where,

BP = Brake Power (in KW)

n = Engine speed (in RPM)

2.4 Reaction Load

These are the loads acting on the wheel due to traction force.

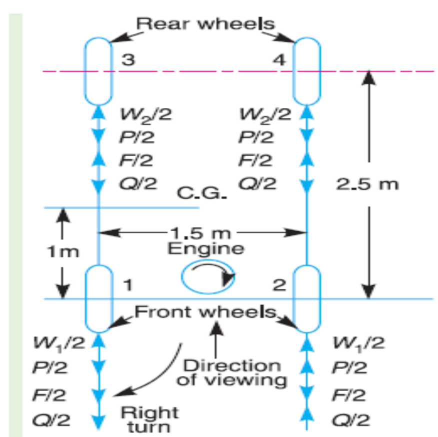

Fig.6 Reaction Load

process overcomes the restriction of using small lengths of Then, Load on Front wheel 1 and Front wheel 2

$$= \frac{W_1}{2} - \frac{P}{2} - \frac{F}{2} - \frac{Q}{2}$$

$$= \frac{W_1}{2} + \frac{P}{2} - \frac{F}{2} + \frac{Q}{2}$$

Where,
W1, W2 = Load on front wheels
W3, W4 = Load on rear wheels
P = Force due to gyroscopic effect of wheel
F = Force due to gyroscopic effect of engine
Q = Force due to centrifugal effect

3 Objective

1. Effect of loads acting on the axle
2. Design verification through classical approach and failure criteria for frontal axle design
3. FE approach for study the stress / strain and deformations
4. Customized methodology for mechanical integrity.
5. Strength evaluation of dynamic characteristics of vehicle axle subjected to cyclic loading

4 Methodology

Linear finite element analysis is an essential component of structural design. Testing of prototypes is increasingly being replaced by finite element modelling for dynamic analysis of automobile components which provides more rapid and less expensive way to evaluate design concepts and design details. Commercial FE packages have made life easier for analyst to take up most complex simulation problem.

5 Preliminary Design Considerations

5.1 Axle Material

The axle material considered for structural analysis is ANSI 4340 steel and following are its important mechanical properties

Table 1: Axle Material properties

Sl. NO	Specification	Value
1	Density	7.7 - 8.03 gm/cc
2	Poisson's ratio	0.27-0.30
3	Elastic Modulus	190-210 GPa
4	Tensile Strength	744.6 MPa
5	Yield Strength	472.3 MPa
6	Elongation	22.0 %
7	Reduction in Area	49.9 %
8	Hardness	217 HB
9	Impact Strength (Izod)	51.1 J

The axle considered for structural analysis is a frontal dead axle of a commercial pickup truck and following are its specifications:

Table 2: Vehicle specification

Sl. NO	Parameter	Specification	Value
1	Engine	Volume	3700 CC
		Power	302 HP @ 6500 RPM
		Torque	278 lb.- ft @ 4000 RPM
2	Axle	Wheel Base	3200 mm
		Track width (Front)	1700 mm
		Axle ratio	3.55
3	Weight	Base curb weight	4685 Pounds
		Payload capacity	1710 Pounds
		Gross vehicle weight	6450 Pounds
		Maximum Loaded Trailer Weight	7900 Pounds

5.2 Shear Force and Bending Moment

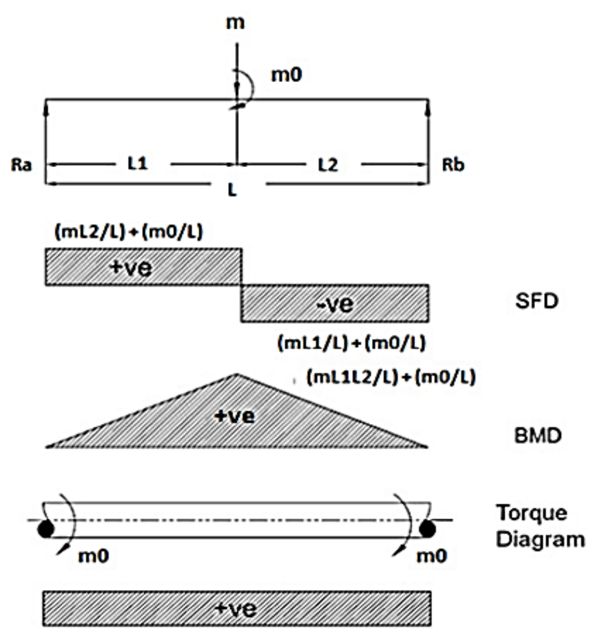

Fig.7 SFD, BMD and Torque Diagrams

5.3 Distribution Concept

Shear Force and bending moment diagrams are analytical tools used in conjunction with structural analysis to help perform structural design diagrams can be used to easily determine the type, size, and material of a member in a structure so that a given set of loads can be supported without structural failure. Another application of shear and moment diagrams is that the deflection of a beam can be easily determined using either the moment area method or the conjugate beam method In transient analysis, the dead front axle is considered to be an I – section beam which is simply supported at both the ends by wheels. The engine and vehicle weight is considered to act at the center of axle.

5.4 Major Design Assumptions

Following are the important design assumptions adopted in this structural analysis of frontal dead axle of a commercial pickup truck

5.5 Finite element approaches:

Linear Static Analysis

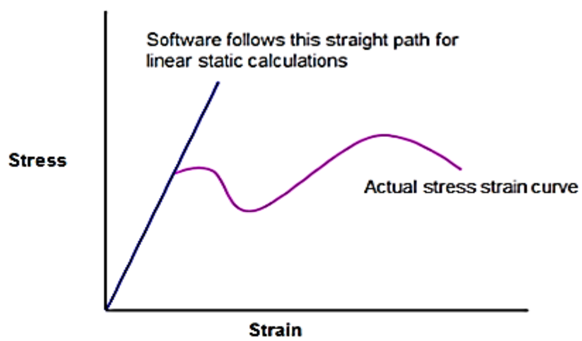

Fig.8 Linear Static Approach

Simplest and most commonly used type of analysis, Linear means straight line. σ = €E is the equation of straight line (y = mx+c) passing through origin. 'E', Young's modulus is the slope of curve and is a constant. In real life, after passing yield point material follows nonlinear curve but software follows same straight line. Component break into two separate pieces after crossing ultimate stress but software based analysis never show failure in this fashion. It single unbroken part only with red color zone at the location of failure. Analyst has to conclude whether the component is safe or fail by comparing maximum stress value with the yield or ultimate stress.

There are two conditions for static analysis:
No variation of force with respect to time (Dead weight)
Equilibrium condition - ∑ Force = 0 and ∑Moments = 0.

Bilinear Kinematic Hardening:

The back stress tensor for bilinear kinematic hardening evolves so that the effective stress versus effective strain curve is bilinear. The initial slope of the curve is the elastic modulus of the material and beyond the user specified initial yield stress σ0, plastic strain develops and the back stress evolves so that stress versus total strain continues along a line with slope defined by the user specified tangent modulus ET. This tangent modulus cannot be less than zero or greater than the elastic modulus.

For uniaxial tension followed by uniaxial compression, the magnitude of the compressive yield stress decreases as the tensile yield stress increases so that the magnitude of the elastic range is always 2σ0,

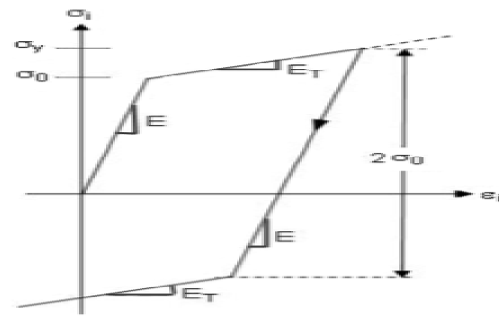

Fig.9 Stress v/s Total Strain for Bilinear

Kinematic Hardening

In this method yield stress, tangent modulus and young's modulus for the dead frontal axle model will be provided through commercial FE package to find out the true stress and true strain. The true strain obtained by this method will be a combination of elastic strain and plastic strain

Modal analysis:

An analysis of measured data is a process in which the measured frequency response functions are analyzed in order to find a theoretical model that most closely resembles the dynamic behaviour of the structure under test

Goodman Equation:

Alternating stress plotted on to the Goodman diagram, shows the variation of the limiting range of stress (σmax – σmin) on mean stress. As the mean stress becomes more tensile the allowable range of stress is reduced.

The Goodman relation can be represented mathematically as:

$$\sigma_a = \sigma_{fat} \times (1-\sigma_m/\sigma_{ts})$$

Where
σ_a = Alternating stress
σ_m = Mean stress,
σ_{fat} = Fatigue limit
σ_{ts} = ultimate tensile stress of the material

Goodman relation is an equation used to quantify the interaction of mean and alternating stresses on the fatigue life of a material. Goodman diagram, sometimes called a Haighdiagraor a Haigh-Soderberg diagram, is a graph of (linear) mean stress vs. (linear) alternating stress, showing when the material fails at some given number of cycles.

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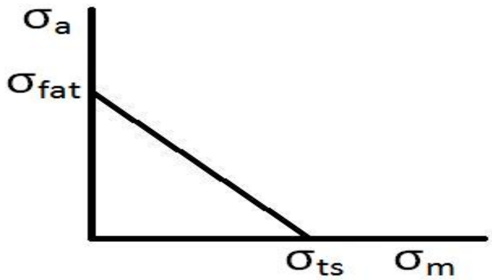

Fig.10

Goodman diagram

All this classical approaches interpretation done with FEM. The object being analysed can have arbitrary shape and the results of the calculations are acceptable. Fatigue analysis of front axle performed using FEM and final result compare with classical approaches and deriving custom made methodology in design of front axle of an automobile for dynamic loads.

Modelling

By solid modelling technique of CATIA. The shape of the front axle beam is based on an I-beam and as it is forged

Fig.11 Solid Front Axle Beam

6 Results and Discussion

6.1 Determination of stress concentration factor

Stress concentration factor = (Maximum Principal Stress) / (Average stress)

= (123.72 / 70.8) = 1.74

It can be observed that selected frontal axle with circular cut out sections is having a stress concentration factor of 1.74 which is less than 3.

Observation

Von – Mises stress of 58.77 MPa is induced in the sub model, achieve highly accurate results

Linear Analysis

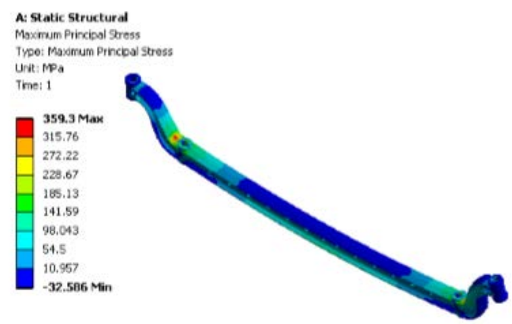

Fig. 12 Von –Maximum Principal Stres for a load of 40 KN

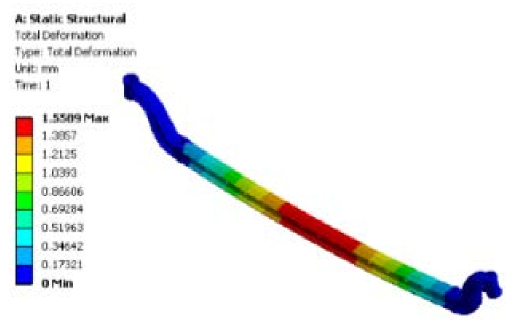

Fig.13 Total Deformation for a load of 40 KN

6.2 Good man diagram:

Mean stress from area weighted approach: σmean= Mb / Z
Z= section modulus = (BH2 /6) - (bh3 /6H) , B= 4t, H= 4t, h= 2t, b= 2t and t= 20mm
Mb = Bending moment, σ_{mean} = Mean stress ,
σ_{mean} =1800000/77280, σ_{mean} = 233Mp

Observations

Plotting mean stress on good man diagram for 1E7 cycles of fatigue loading alternating stress limit obtained: 165Mpa

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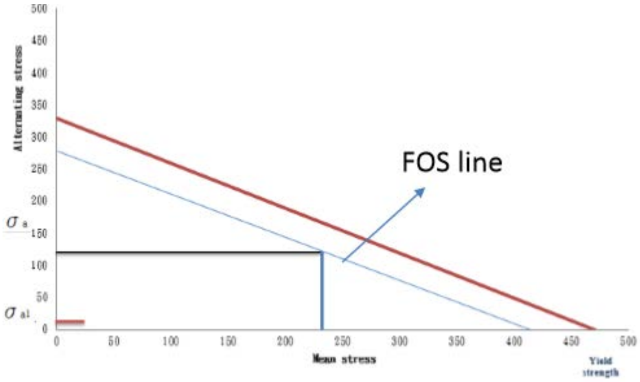

Fig, 14 Good man diagram

7 Conclusion

Reaction force load condition exhibited stress is uniformly distributed among the cut out sections and hence the selected axle is beam of uniform strength, The stress concentration factor is determined and found to be 1.74 which is less than 3

Area weighted approach technique is followed to determine the average stress and it is notice that mean stress is obtained for the worst load condition of dynamic analysis

Maximum principle stress under low cycle fatigue is found, tensile in nature and it is less than yield strength of the material Hence considered front axle obligate infinite life cycles with a reserve factor 21%.

Different mode shapes are determined at the respective alternating frequencies. and Good man diagram plotted using alternating stress of 3.85Mpa as obtained from mode 1 and noticed that alternating stress with fatigue limit is 165Mpa with a reserve factor of 4.2, hence this can be adopted as a customized methodology for stress analysis there by reducing the overall design period

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