

Fatigue Life Evaluation of an Automobile Frontal Axle

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ABSTRACT: The Front axle apart from vehicle load is critically subjected to cyclic and shock loads. The stress which is built up due to these loads plays a prominent role where the component concedes before the desired design life. Modern computers have made it possible to conduct sensitivity analysis and design optimization through computational techniques blending classical theories with numerical techniques. This has reduced the cost on experimentation and repetition of experiments for validation. Front axle are subjected to carry 60% of continuous loads in the automobile vehicles. This loads are from engine and the vehicle load both during static and dynamic condition of the vehicle. Hence the automobile industries call for optimum front axle. An optimum front axle was defined, as an axle with functional geometry which when loaded yields minimum stress along with less weight. In the present work approach has been done for design optimization of the axle with holes, to ensure stress is uniformly distributed along the filleted circular holes of the axle, which otherwise leads to crack initiation and propagation and hence axle lands in failure. This may further lead to load shift and effect on stage and overall vehicle failure over a period of cycles. Hence, special attention is made on front axle with holes which fails and to guarantee for safe and reliable operation under all working conditions. Commercial FE software is used to determine the structural integrity of Frontal axle. The axle is suitably designed using Campbell diagram by ensuring the working frequency is well with 6e excitations for the specified operating speeds. Hence increasing the life of the axle. Further, the work is focused on structural evaluation of front axle using FEA approach with detailed design considerations which, includes Gross weight of the vehicle, inertial loads, dynamic loads and rolling resistance. And the theoretical basis of scientific designers to improve design quality and shorten the design cycle and reduce design costs.

KEYWORDS: Front axle, Static analysis, Campbell diagram, modal analysis, FEM.

I. INTRODUCTION

The axle is connecting rod for the rotating wheel. The wheels are entrenched to the axle, with the bearings or bushings provided at the mounting points where the axle is supported. The positions of the wheels relative to each other are well maintained with the help of axle.

Basically there are two types of axle's Front axle and rear axle. In this paper we concentrate on the study of front axle which carries the weight of front part of the vehicle, it facilitates steering, it absorbs the shocks transmitted from road surface irregularities and it also absorbs the torque applied on it due to the barking of the vehicle [1] Spindle

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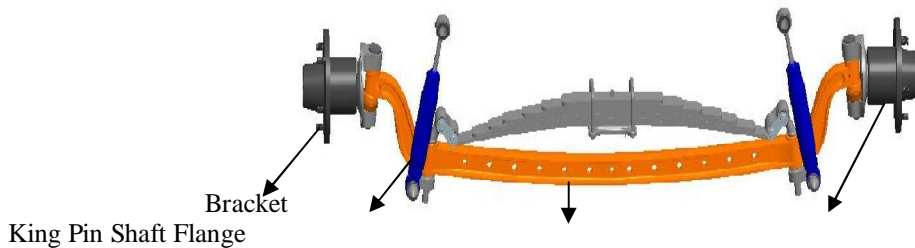


Fig 1.0 Dead Front Axle Mechanism of Commercial Truck

Power is transmitted from clutch to gearbox than to drive shafts which in turn transmits the power to wheels. Shaft and Flanges are bolted on either side of the couplings and the drive is transmitted through flexible rubber. Stub axle are connected to front axle by king pin and they turn on kingpins. Both ends of front axle has circular cross-section so that it can withstand the bending loads due to weight of the vehicle and torsional load due to the braking torque applied. Wheel is mounted on the minor axis.

In operation axle is subjected to torsional stress and bending stress due to self-weight or the weight of the components. Thus these rotating parts are susceptible to fatigue by the nature of their operation and the fatigue failures are generally of the torsional, rotating-bending, and reversed (two-way) bending type.

It is observed that the maximum stress has reduced by introducing coaxial holes which is a good reduction from the initial value of plate with a single hole. Hence, the same concept of the plate with hole model is used when doing the sub modeling of the components also, combining with the neutral fiber concept to picture the optimum solution of the system without affecting the structural integrity.

Fatigue failures start at the most vulnerable point in a dynamically stressed area particularly where there is a stress raiser [3]. Though we all are aware of the front axle in four wheeler trucks, Here an approach has been made to introduce the holes in front axle neutral axis as shown in fig 1. and run the axle against the fatigue failure and the concept of sub modelling to study the stress concentration is customized using the simple front axle geometry.

II. LOADS ON FRONT AXLE

In a vehicle 55% to 65% curb weight acting front axle and impact loads due to dynamic conditions acts directly on front axle. Different loads acting on front axle are as shown in fig.2.0, 3.0

- **Engine Load** - acting vertically downwards as a dynamic load when vehicle is moving on a bumpy road even at constant velocity or as static load when the vehicle is at rest or moving at constant velocity on even road [2] as shown in Fig 2



Fig 2.0: Engine load on dead axle

- **Reaction Load** – When vehicle traverse on uneven road front and rear axle experience a moment. The torque is applied on front axle and it is reacted by rear axle i.e., clockwise torque on front axle here is balanced by anticlockwise torque on the rear axle thus resulting in torsion moment [5]. As we all know torque is always accompanied by bending due to gravity as shown in Fig 3.

W1 - Load on front wheels

P- Force due to gyroscopic effect of wheel

F- Force due to gyroscopic effect of engine

Q- Force due to centrifugal effect

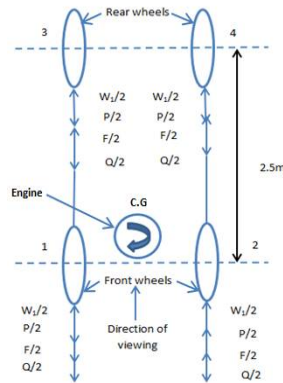


Fig 3.0 Reaction loads

$$\text{Load acting on front wheel 1} = \frac{w_1}{2} - \frac{Q}{2} - \frac{P}{2} - \frac{F}{2}$$

$$\text{Load acting on front wheel 2} = \frac{w_1}{2} + \frac{Q}{2} + \frac{P}{2} - \frac{F}{2}$$

$$\text{Torque acting on the axle} = \frac{BP \times 60}{2\pi N} \text{ N-m}$$

Where,

BP=Brake Power in KW

N=Speed in rpm

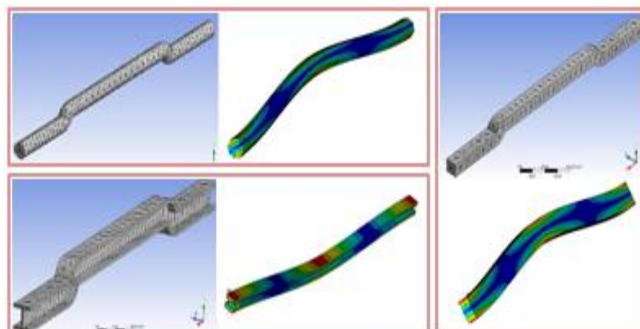
III. DESIGN OPTIMISATION OF FRONT AXLE

Optimization helps to achieve extreme Excellency in designing, simulation and fabrication. Hence optimization is considered to get most out of the system.

Considering the engine load is acting on the beam for circular, rectangular and I- Section in order to investigate the effects of the engine load, area of cross section is assumed to be constant(2600mm²). Bending stress is obtained and compiled.

Tabular column 2.0

Section	Continuum method (MPa)	Finite element method (MPa)	Error (%)
Circular	13.89	13.89	0
Rectangular	8.28	8.22	7
I section	7.514	7.53	2



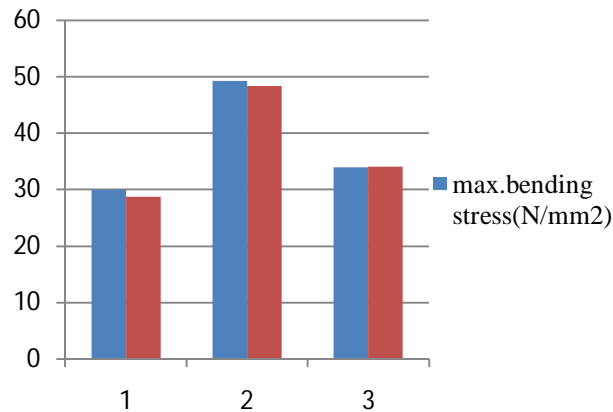


Fig 4.0 Response chart for stress obtained and comparison of FEM results with theoretical results

From the above compilation of results and graph it is clear that Square section has got least bending stress, I – section is favorable, Since most of the area is distributed away from centroid axis hence the moment of inertia is greater than circular cross section and less material required for manufacturing.

IV. STRUCTURAL ANALYSIS

DESIGN EXPLORATION:

The below fig 5.0 shows the design of dead front axle. The 3D CAD model of the frontal dead axle of a four wheeler truck is modeled using modeling software. The idea was to exploit the symmetry of the model hence only half model is created by creating the base half I-section. The end of the I-section is modeled with the circular cross section on both the ends. A spline is created from the cylinder edge which follows the I-section, as the suspension pick up points are major consideration which can pull off or push off the axle in the lateral direction during corner entry [4].



Fig 5.0 Dead Front Axle

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Loading Conditions considered for analysis

First and foremost step for the design of the axle is to find the boundary condition and the placing of the components

Test load conditions

Sl.No.	Loading Value	Remarks
1	Load calculated due to gyroscopic effect under maximum vehicle load in four wheeled truck $W_1=5252.957N, W_2=10791.84N$	These loads are calculated considering vehicle taking a corner entry and loads will be acting vertically downwards from the wheels to perform Linear static analysis and fatigue to know the safe life of the vehicle
2	Dead weight =50000N	This load is the average front axle load acting on the front axle which is slightly higher than the above considered load

V. AXLE MATERIAL

The axle material is assumed to be Structural steel and its mechanical properties are as below:

	Specification	Value
	Density	8.03 gm/cc
	Poisson's ratio	0.30
	Elastic Modulus	210 MPa
	Tensile Strength	744.6 MPa
	Yield Strength	472.3 MPa

VI. ANALYSIS

All the design requirements are met only when the system is analyzed holistically. To start with the Design, the static structural analysis is done in order to capture the weak zones. From the basic knowledge of the strength of material, the maximum stresses are expected to arrive at the inner and outer transitional zone of the frontal axle due to sudden change in the geometry. Also, the type of load acting calls for the fatigue analysis [6].

The deflections are expected to be high due to the engine weight at the margin somewhere in between 40% to 60% of the overall length which leads to sub modeling of the region where maximum deformation occurs.[7]

Dynamic analysis always plays a major role in any kind of analysis. The system is analyzed to obtain natural frequency from the engine vibrations and wheel undulations which affect the system. By considering this the stimulus and the stresses are plotted in the Campbell diagram to validate the results.

VII. MESH GENERATION FOR 3D MODEL

The Model of the front axle is imported to ANSYS and ideal model is generated using Finite element modeling technique of ANSYS Software. The geometry imported is a 3D model which is meshed considering the SOLID 186 which is dominant. Wherever 186 is not possible SOLID 187 is considered, htype convergence is considered

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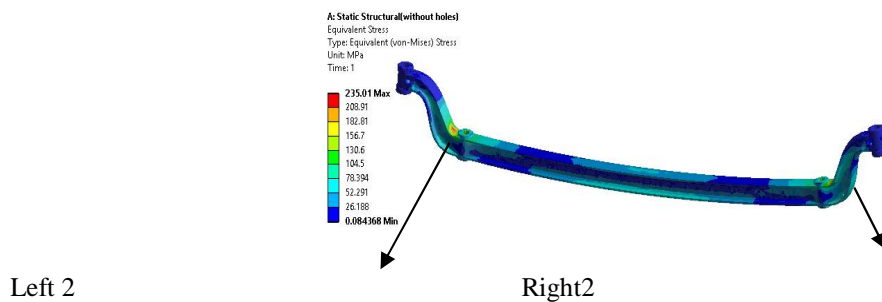
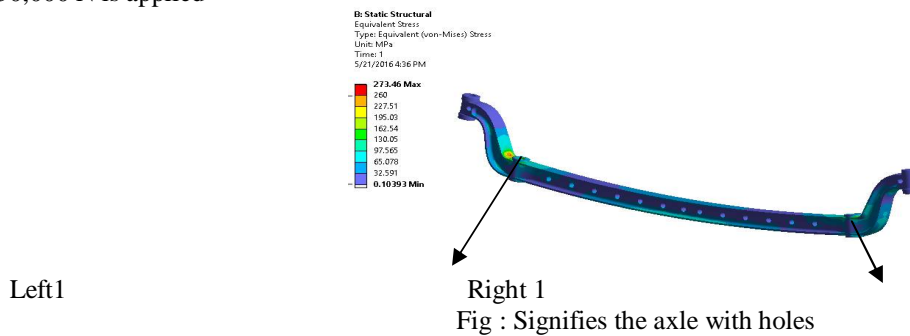
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VIII. RESULTS AND DISCUSSION

ANALYSIS FOR FOUR WHEELED VEHICLE:

The analysis is carried out by loading the engine weight to the axle with pertaining hole and one without holes, Model is fixed at all degrees of freedom at the two circular cross sections in both the cases and a uniformly distributed load of 50,000 N is applied



In the above fig 6.0 Maximum equivalent stress obtained is 273.46MPa in axle with holes and 235.01MPa in axle without holes. It is noticed that the maximum deformation is happening at the center of the beam. The stresses captured are as follows;

Position	Outside stress(MPa)	Inside stress(MPa)
Right	8.0937	39.884
Left	15.339	66.081
Right 2	245.27	70.104
Left 2	192.78	67.649

It can be seen that the maximum equivalent stress are maximum at the Right 2 of axle with holes. But the maximum stress is well within the elastic limit and hence is safe for the further consideration.

Compilation of Results for Dead weight (20kN)

Sl. No.	Parameter	Value (Max)	Value (Max) without holes
1.	Von-mises Stress	273.46MPa	235.01MPa
2.	Maximum Principal stress	269.13MPa	143.12MPa
3.	Minimum Principal stress	38.579MPa	20.932MPa
4.	Maximum elastic strain	0.0012941	0.0012032

In this case equivalent stress plays a major role and acts as the decision maker. Equivalent stress is the deviator

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stress of all the combined effect of stress; derived from the strain energy theory it is calculated and found to be well within the range.

As the maximum principal stress increases beyond the yield limit in the axle without hole, the crack initiation begin to happen as the maximum principal stress acts as tension in the material and shear stresses are completely absent in the material.

Minimum principal stress always tends towards zero and doesn't create any serious deformations. Middle principal stress is compressive in nature. It tries to close the crack. If the middle principal stress is 68% of the yield, then the tensile force cannot open the crack up and the failure is 4 in million as per six sigma standard.

IX. FATIGUE

The main aim of fatigue analysis is to characterize the capability of the axle material to survive many cycles during its life time. Stress life approach was posited as it is concerned with total life and does not distinguish between initiation and propagation like strain life. Stress life addresses high cycle fatigue usually more than 1, 00,000 cycles. [8]

Here the analysis is carried out considering the dead weight of the engine 50KN, Fatigue for fluctuating stress and strain of specified nature that material will sustain before failure occurs was calculated for both the axles and juxtaposed using stress life approach (Powers law) and the simulation obtained.

Most widely fatigue life model is expressed by 'Coffin Manson' equation for stress through 'power law' [10]

Coffin Manson relation between fatigue life and stress life is as follows:

$$\sigma_{ar} = \sigma_{fr} [2Nf]^b$$

Where,

σ_{ar} = average of maximum principal stress and minimum principal stress.

σ_{fr} = Fatigue strength coefficient

N = denotes fatigue life cycles

b = Fatigue strength exponent

The Stress life parameters considered for structural steel are:

Fatigue strength coefficient, $\sigma_{fr} = 920\text{MPa}$

Fatigue strength exponent, $b = -0.106$



fig 7.0 Shows axle with holes fatigue life



The above fig 7.0 shows fatigue life of the component exhibiting maximum design life

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Sl.no.	Axle	Stress approach(cycles) 10^5	Simulation (cycles) 10^5
1.	Axle with holes	4.43	1.069
2.	Axle without holes	13.42	1.5976

From the above tabulation, Minimum life of 1.069×10^5 cycles at the right corner of the stub for high cycle fatigue for axle with holes and 1.597×10^5 cycles at the right corner of the stub for high cycle fatigue for axle without holes was observed which was approximately equal to the theoretical values obtained from Power's law Stress life theory . Thus we have obtained 9% more cycles in the axle with holes when compared to the base model (axle without holes) which was about 13.42×10^5 cycles.

X. SUB MODELLING OF THE AXLE: (DFEM Approach)

Sub modeling is a way of obtaining more accurate results by considering the domain of interest from the model. As the deflection is very high at the center, the center section is considered as the sub- model [9]

This is method of Sub modeling is adapted to the axle subjected to engine load to obtain the stress concentration

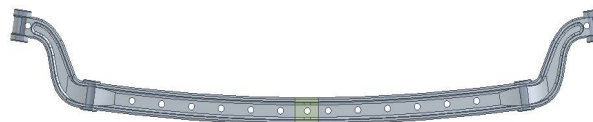


Fig 8.0: sub modeling of axle

The above fig 8.1 shows the sub modeling section at the center of the axle where the engine load is acting. It is very interesting to know that, sub- Modeling reduces the computational time. The mesh can be refined for any complex geometry without taking time to solve compared to parent model.

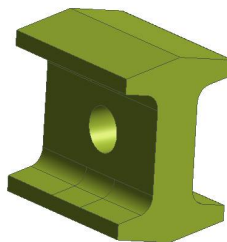


Fig 8.1 Model and meshing of sub modeled element

XI. STRESS CONCENTRATION FACTOR

Axle will be analyzed for stress concentration considering it as a plate with the hole problem. To account for the peak in stress near a stress raiser, the stress concentration factor or theoretical stress concentration factor is defined as the ratio of the calculated peak stress to the nominal stress that would exist in the member if the distribution of stress remained uniform[11]; that is,

$$K_t = \frac{\sigma_{\max}}{\sigma_{\text{nom}}} = \frac{472.3}{132.81} = 3.5$$

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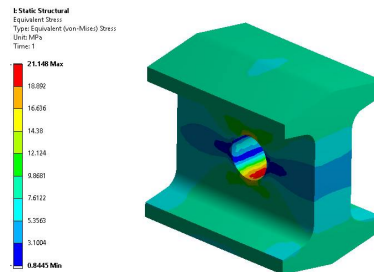


Fig 9.0: Von-mises stress at the sub model

Conclusion:

From the above calculation it is delineated that the value of stress raiser is 3.5 .Here the stress at the edge of transverse crack is larger which is 472.3MPa hence the stress concentration factor is also high.

XII. SEPARATION MARGIN EVALUATION USING CAMPBELL DIAGRAM

Whenever front axle is considered, there will bewide range of reasonsthrough which it shows failure. The source of excitation are chassis, engine, wheel, suspension, knuckle, random vibrations etc., are prominent.[12] But Engine vibration and the Wheel excitation can cause a serious damage to the system. Hence Campbell diagram is plotted to know the possible resonance in the system.Considering the operating speed as 6000rpm.

Table 8.1: Compilation of Results in Modal Analysis

Mode	Deformation	Frequency
1	15.989mm	74.293
2	15.265mm	204.93
3	17.254mm	291.61
4	18.965mm	336.74
5	13.011mm	359.46
6	21.819mm	521.39

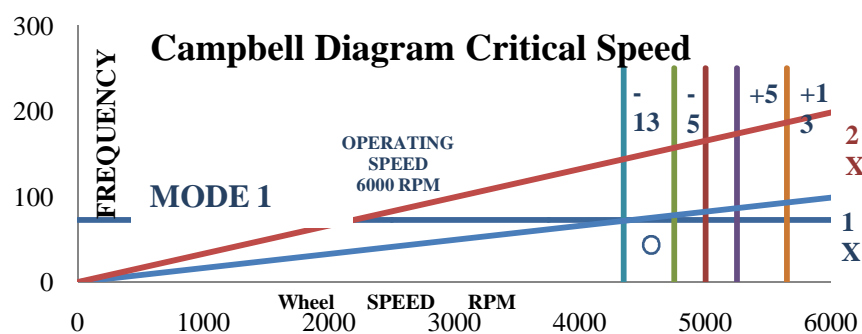


Fig 8.1 Campbell diagram

Engine frequency is far apart from the axle frequency, hence no chances of resonance with respect to engine. Rotating wheel can act like a external excitation system which is considered in this example. The Campbell diagram shows the resonance of the first mode with first wheel harmonics is happening not less than 10% and hence the probability of resonance of first mode is almost zero. First mode and second mode constitute more mass participation factor .If these two modes are not resonating, than the probability of other modes being excited is less than one percent. The 2X harmonic is far away from the natural frequency and hence there is no excitation in the system and the design is safe.

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XIII. CONCLUSION

The Major Conclusions drawn from the structural analysis of frontal axle is as follows:

- Dynamic analysis is carried out on front of an automobile and verified for assumed preliminary design considerations.
- The results of the analysis showed that the reaction load conditions, stress is distributed evenly between the cut-out sections, so the selected axis is a "uniform intensity beam."
- The analysis results shows that the equivalent stress(Gross yielding stress) is not crossing the yield strength of the material even for the maximum loaded conditions of the vehicle and hence axle is considered to be robust in design with infinity life. If it is so, then the material can be taken with higher yield point.
- The analysis of axle matched with sub modeling results hence it is customized methodology to do the analysis. And can be inferred that the results obtained are correlating with each other.
- Sub modeling sectional stresses was well within the yield range. But it is greater than the without holes. Also, the strength to weight ratio is large in the with hole axle by 15 percent.

REFERENCES

1. Lalit Bhardwaj, et al., "Design and static analysis of rear axle", www.academia.edu
2. Kalpan Desai, et al., "Bending stress analysis of rear axle of Maruti-800car", International Journal of engineering research and technology ISSN: 2278-0181, volume1, Issue6, August 2012.
3. AnupamSinghal, et al., "Failure analysis of automotive FWD flexible driveshaft– A review", International journal of engineering research and applications, ISSN:2248-9622, volume3, Issue1, Jan – Feb 2013, pp.577-580.
4. C Kendall Clarke, et al., "Failure analysis of induction hardened automotive axles", ASM International, 16 May 2008.
5. Ji-xin Wang, et al., "Static and dynamic strength analysis on rear axle of small payload off – highway dump trucks", Jilin University, Changchun 130025, PR China.
6. Lawrence Kashar, "Effect of strain rate on the failure mode of a rear axle", ASM International, volume1, pp. 74-78, 1992.
7. Mulani, et al., "FEA analysis of bullock cart axle under static and dynamic condition", International journal of engineering research and technology ISSN: 2321-5747, volume1, Issue2, 2013.
8. Javad Tarighi, et al., "Static and dynamic analysis of front axle housing of tractor using finite element methods", Australian journal of agricultural engineering, volume2, pp.45-49, 2011.
9. G Rajesh Babu, et al., "Static and modal analysis of rear axle housing of a truck", International journal of mathematical sciences, technology and humanities, volume7, pp.69-76, 2011.
10. Osman Asi, "Fatigue failure of a rear axle shaft of an automobile", Engineering failure analysis 13 (2006) 1293 – 1302.
11. Meng Qinghua, "Fatigue failure fault prediction of truck rear axle housing excited by random road roughness", International journal of the physical sciences, vol.6 (7), pp.1563-1568, 4 April, 2011.
12. Happy Bansal, et al., "Weight reduction and analysis of trolley axle using ANSYS", International journal of engineering and management research, vol-2, Issue-6, December 2012.