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DESIGN ANALYSIS AND OPTIMIZATION OF FRONT AXLE FOR COMMERCIAL VEHICLE USING CAE

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ABSTRACT

In the global competition, it is very important for the manufacturer to bring new product designs to market at a faster rate and at reduced cost. Up to 40% of the vehicle load carrying capacity is taken up by the Front axle beam. Therefore, optimization of front axle beam is necessary to improve strength to weight ratio for a given factor of safety without altering any assembly parameters. This is being achieved by using high-end optimization tool hyper mesh optistruct. Shape optimization tool in hyper mesh optistruct is used for analysis. In these work Design parameters such as Wheel track, King pin center, spring center, Weight of the axle and Rated load of axle was analyzed. The performance parameters such as stress, strain and displacement are measured by applying the vertical load, vertical and braking combined load and vertical and cornering combined load. The life cycle and strain value of the axle beam is analyzed.

Keywords: wheel track points, rolling radius, vertical load, braking load and cornering load.

1. INTRODUCTION

Front axle carries the weight of the front part of the automobile as well as facilitates steering and absorbs shocks due to road surface variations. The front axle must be rigid and robust in construction. It is usually steel drop forging having 0.4 % carbon steel or 1 to 3% nickel steel. The front axles are generally dead axles, but are live axles in small cars of compact designs and also in case of four-wheel drive. Front axles can be live axles and dead axles. A live front axle contains the differential mechanism through which the engine power flows towards the front wheels. For steering the front wheels, constant velocity joints are contained in the axle half shafts. Without affecting the power flow through the half shafts, these joints help in turning the stub axles around the king-pin. The front axles are generally dead axles, which does not transmit power. The front wheel hubs rotate on antifriction bearings of tapered-roller type on the steering spindles, which are an integral part of steering knuckles. To permit the wheels to be turned by the steering gear, the steering spindle and steering knuckle assemblies are hinged on the end of axle. The pin that forms the pivot of this hinge is known as king pin or steering knuckle pin. Generally dead front axles are three types. In the Elliot type front axles the yoke for king spindle is located on the ends of I-beam. The axle ends are forked to hold the steering knuckle extension between them. The reverse Elliot front axles have hinged spindle yoke on spindle itself instead of on the axle. The forked portion is integral with the steering knuckle. This type is commonly used as this facilitates the mounting of brake backing plate on the forged legs of the steering knuckle. In the Lemoine type front axle, instead of a yoke type hinge, an L-shaped spindle is used which is attached to the end of the axle by means of a pivot. It is normally used in tractors. To act as an axle beam upon the ends of which the road wheels can revolve and through which the weight of the body and load can be transmitted via the spring and the road wheels to the ground. The dead front axle has sufficiently rigidly and strength to transmit the weight of the vehicle from springs to the front wheels. The ends of the axle beam are shaped suitably to assemble the stub axle. The ends of the axle beam are usually shaped either as yoke or plain surface with drilled hole for connecting the stub axle assembly. The downward sweep is given to the axle beam at the central portion to keep the low chassis. These types of axles are made of I-sections in centre portions while the ends are made either circular or elliptical. In this construction it takes bending loads due to the load of the vehicle and also torque due to braking of the wheels.

1.1 Front axle

The front axle is designed to transmit the weight of the automobile from the springs to the front wheels, turning right or left as required. To prevent interference due to front engine location, and for providing greater stability and safety at high speeds by lowering the centre of gravity of the road vehicles, the entire centre portion of the axle is dropped. Front axle includes the axle-beam, stub-axles with swivel pin brake assemblies, track rod and stub-axle arm as shown in Figure-1.

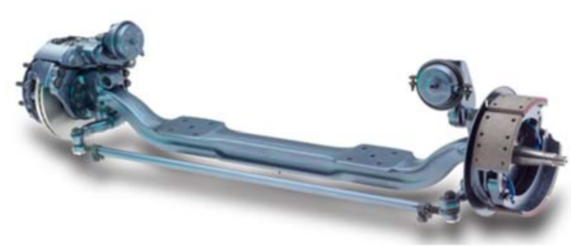

Figure-1. Front axle.

1.1.1 Functions of front axle

A front axle is a rotating shaft at the front of a vehicle that turns the front wheels. Front wheels of the vehicle are mounted on front axles. It supports the weight of front part of the vehicle and facilitates steering of the vehicle to turn left or right. It absorbs shocks, torque and

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horizontal bending moment which are transmitted due to road surface irregularities, braking of vehicle and resistance to motion.

1.1.2 Construction and operation

Front axle is made of I-section in the middle portion and circular or elliptical section at the ends. The special x-section of the axle makes it able to withstand bending loads due to weight of the vehicle and torque applied due to braking. On kind of front axle which consists of main beam, stub axle, and swivel pin, etc. The wheels are mounted on stub axles. The front axle has sufficiently rigidly and strength to transmit the weight of the vehicle from springs to the front wheels. The ends of the axle beam are shaped suitably assemble the stub axle. The ends of the beam are usually shaped either as yoke or plain surface with drilled hole in order to accommodate a swivel pin for connecting the stub axle assembly.

A front axle beam is a suspension system, also called a solid axle, in which one set of wheels is connected laterally by a single beam or shaft. A front axle beam that does not transmit power is sometimes called a dead axle. Front axles are typically suspended either by leaf springs or coil springs. To keep the low chassis height its center portion is given a downward sweep.

1.2 Design optimization and need for front axle beam

The optimization process involves selection of variables that describe the design alternatives, selection of objective functions to be minimized or maximized, establishment of restrictions, expressed in terms of design variables, which must be satisfied by any acceptable design. Shape optimization is to be carried out for all three load cases with constraining the maximum stress level as in the previous analysis run for respective cases, with objective to minimize mass while maintaining the same/lower stress levels as for the existing front axle beam. The front axle is needed to improve turning circle radius, optimize load carrying capacity, optimize king pin center to achieve max stability & better center of gravity. Also it will optimize king pin angle for added maneuverability and optimize the axle beam weight and stress levels of the material

1.3 Design parameters to be considered

- Vertical Load
- Combined Vertical and Cornering Load
- Combined Vertical and Braking Load
- Wheel Track
- King pin Centre
- Spring Centre
- Weight of Axle
- Axle Rated Load

2. PROCESS METHODOLOGY

The process flow chart shows that the sequences of process for the design optimization of front axle beam which is shown in below flow chart.

Process flow chart

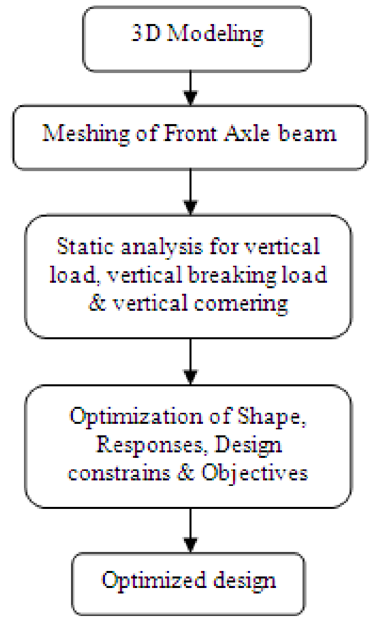

Figure-2.

2.1 Concepts and optimum design

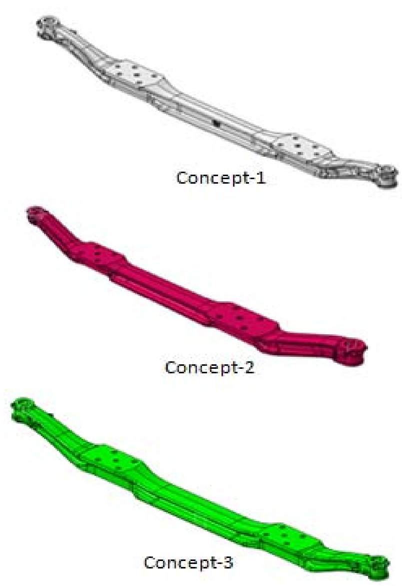

Figure-3. Concept designs.

The below Table-3.4 shows that the compilation of all the concepts design values and the stress values, in which concept - 2 highlighted in yellow is the lowest values when comparing concept 1 and 2.so concept - 2 is considered as optimized design and recommended for CAE analysis.

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Table-1. Combined stress values of all concepts.

Section	Half Axle load in Kgf			Dist. from track in mm			Stress (kg/mm^2)		
	Conpt 1	Conpt 2	Conpt 3	Conpt 1	Conpt 2	Conpt 3	Conpt 1	Conpt 2	Conpt 3
Beam Mid-Section	2450	2400	2400	944.48	944.48	917.78	21.10	20.61	21.47
Spring Pad-Section	2450	2400	2400	544.48	544.48	517.78	17.27	16.64	15.82
Goose Neck Section - 4	2450	2400	2400	404.48	404.48	377.78	15.90	15.31	14.34
Goose Neck Section - 3	2450	2400	2400	350.48	350.48	323.78	21.19	21.48	21.94
Goose Neck Section - 2	2450	2400	2400	255.48	255.48	228.78	17.34	17.03	17.47
Goose Neck Section - 1	2450	2400	2400	166.48	166.48	139.78	13.11	12.84	11.07
King Pin Boss-Section	2450	2400	2400	104.48	104.48	77.78	3.16	3.09	2.30

Based on the stress value, axle load carrying capacity and weight of the axle, concept-2 is considered as optimum design. The figure 4 shows the optimum design 3D model of axle beam.

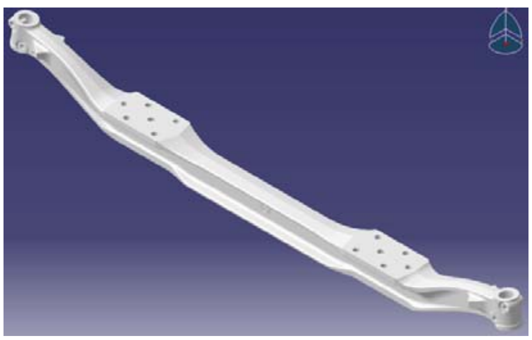

Figure-4. Optimum Design 3D Model.

3. STATIC ANALYSIS OF OPTIMIZED DESIGN

First of all the 3D CAD model was made using Catia, followed by tetrahedral mesh using Hyper mesh, Nastran, Optistruct tool CAE analysis carried out.

Load cases for Fatigue analysis

- Vertical fatigue test - Vertical force: 0.2 to 3.0g
- Vertical and Braking test - Vertical force: 1 to 2.8G (Tension) at track point and braking force 0 to 2.0g at tire rolling radius in the rearward direction only.
- Vertical and cornering test - Vertical force 0.5 to 1.5g and Cornering force 0.25 to 0.75g1g = 4800 Kg axle rated load. i.e. 2400 Kg per wheel

3.1 Loading and boundary conditions

Boundary condition defined in Symmetrical Half model. Static analysis is done for three cases viz. vertical load, Vertical + Braking load and Vertical + Cornering load with respective boundary conditions to find out maximum stress level for each case.

3.2 Vertical load

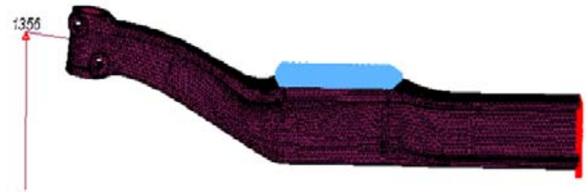

Figure-5. Loading and boundary condition-Vertical 3g.

This condition simulates the static loading of vehicle. For this vertical test, the model has been restrained at the wheel center point and 3g load (3x2400=7200 Kgf) has been applied at the spring pad as shown in Fig-6, where the spring will be mounted in the vehicle. The vertical load will be applied on both spring pads simultaneously and the amount of stress level and displacement of the axle beam will be measured.

3.3 Vertical and braking load

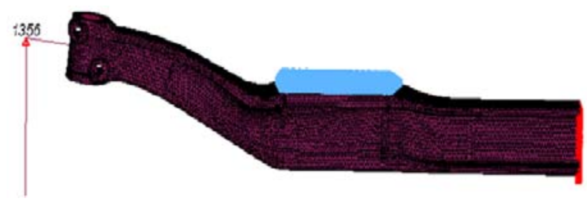

Figure-6. Loading and boundary condition-Vertical 3g.

To simulate the condition of subjecting the vehicle to loading and braking, the above condition is applied. For braking load condition, the model has been restrained at the spring bolt holes and at the spring pad top face in the vertical direction alone. The vertical 2.8g load (2.8x2400=6720 Kgf) has been applied at the wheel center point and braking load 2g (2x2400=4800 Kgf) at the tire rolling radius as shown in Figure-7. The combined vertical and braking loads are phased as follows:

- When the vertical force is at the minimum load, the braking force is to be at its minimum load.
- When the vertical force is at maximum load, the braking force is at maximum in the rearward direction

The amount of stress level and displacement of the axle beam will be measured in the above loading conditions.

3.4 Vertical + cornering load

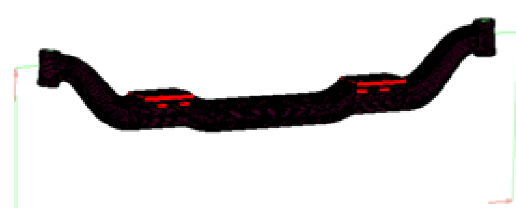

Figure-7. Loading and boundary condition - Vertical 1.5g and cornering 0.75g.

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When the vehicle is taking a turn right or left due to the weight of the vehicle and payload, vertical load will act on the axle at the same time due to the turning effect cornering load also applied horizontally from side of the axle For cornering load condition, the model has been restrained at the spring bolt holes and at the spring pad top face in the vertical direction alone. The vertical 1.5g load (1.5x2400=3600) has been applied at the wheel center point and cornering 0.75g (0.75x2400=1800) load at tire rolling radius as shown in Figure-7. The combined vertical and Cornering loads are phased as follows:

- When the vertical force is at the minimum load, the cornering force is to be at its minimum load.
- When the vertical force is at maximum load, the Cornering force is at maximum in the inward direction

The amount of stress level and displacement of the axle beam will be measured in the above loading conditions.

3.5 Various loading points

The Figure-8 shows that the loading point on the axle beam as vertical load on the spring pad combined vertical load at wheel track point and braking load at tyre rolling radius which is perpendicular to the axle beam axis. Also combined vertical load at wheel track point and cornering load at tyre rolling radius which is in line to the axle beam axis

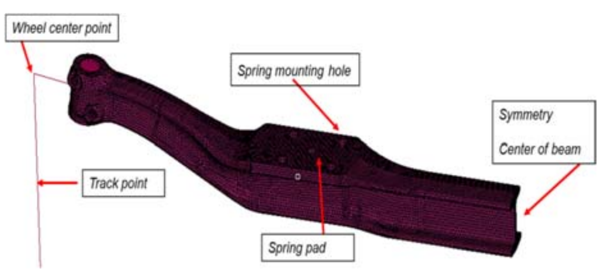

Figure-8. Various loading locations.

The below Table-2 contains the details of static analysis results in which displacement and stress values are tabulated for each load case that is vertical, vertical + braking and vertical + cornering. In which the proposal - 1 show that the stress value and displacement value in each load case is lower than the proposal 2 and 3. From this result the design of proposal - 1 is considered as it is within the permissible limit and safe.

Table-2. Static analysis result.

Load case	Stress in Mpa	Displacement in mm
Vertical 3g	617	21.61
2.8g Vertical & 2g Braking	1640	9.78
1.5g Vertical & 0.75g Cornering	593	3.99

3.6 Vertical load – stress & displacement plot

The Figure-9 shows that the analysis result of vertical 3g Load at spring pad. The observed maximum stress and deflection is 618.5 Mpaand 21.2 mm respectively.

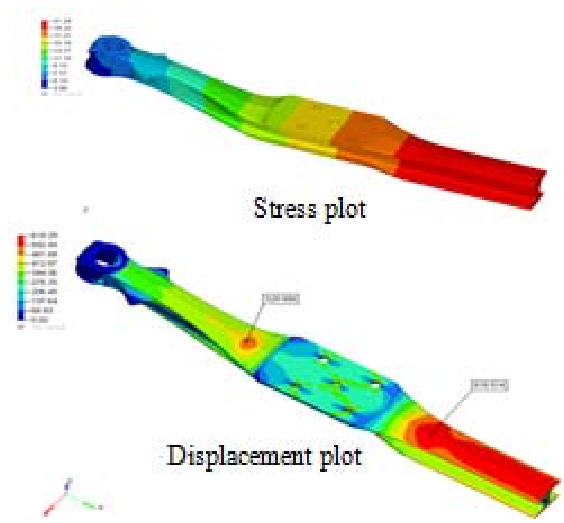

Figure-9. Vertical Stress and Displacement plot.

3.7 Vertical and braking load - stress and displacement plot

The Figure-10 shows that the analysis result of vertical 2.8g and Braking 2g load. The observed maximum stress and deflection is 1434.4 Mpa and 8.6 mm respectively.

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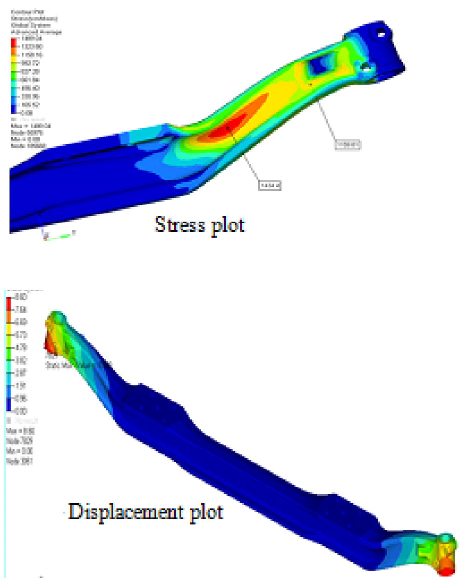

Figure-10. Vertical and Braking Stress and Displacement plot.

3.8 Vertical and cornering load - stress and displacement plot

The Figure-11 shows that the analysis result of vertical 1.5g and Cornering 0.75g load. The observed maximum stress and deflection is 487 Mpa and 3.7 mm respectively.

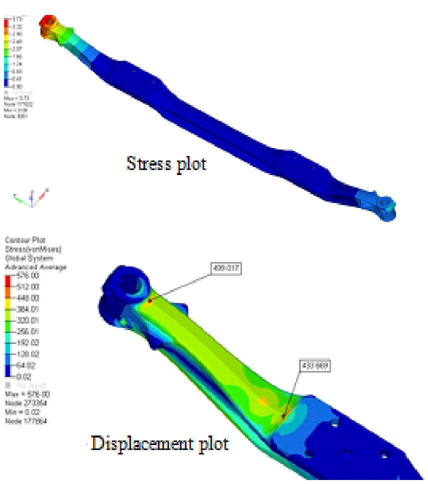

Figure-11. Vertical and Cornering Stress and Displacement plot.

3.9 Fatigue analysis of optimized design Vertical fatigue load cycle - stress plot

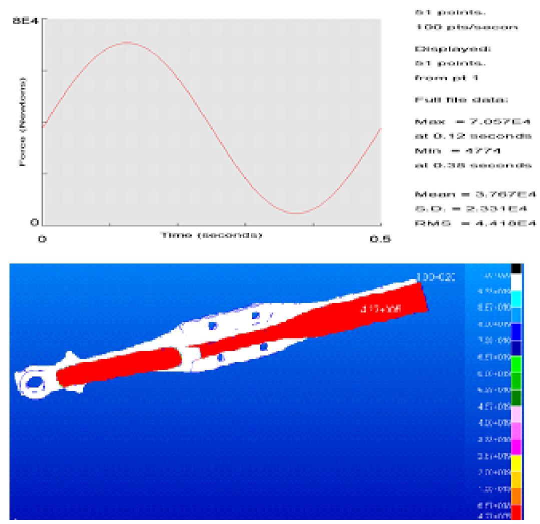

Figure-12. Life and Stress plot for Vertical Fatigue.

Figure-12 shows stress plot of Vertical fatigue analysis. The loading cycle for the vertical load is 427000 cycles as against the target life of 100000 cycles. Hence the design is within the safe.

3.10 Combined vertical and braking fatigue load cycle - stress plot

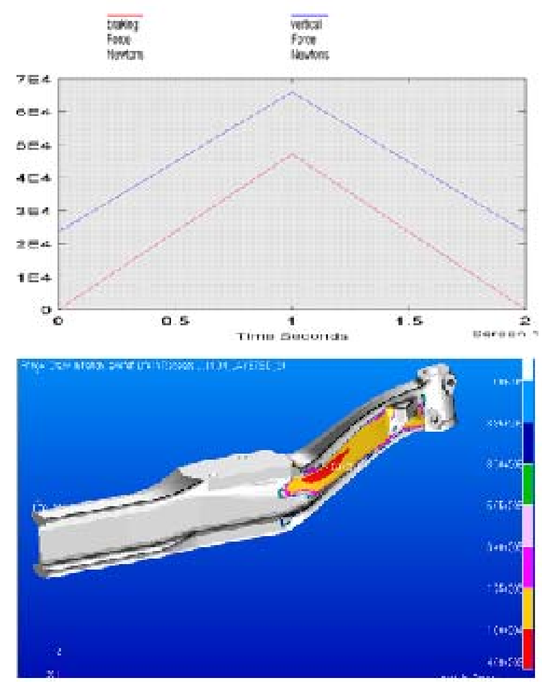

Figure-13. Life and Stress plot for Vertical + braking fatigue.

Figure-13 shows life and stress plot of Combined Vertical and Braking fatigue analysis. The life of axle beam in the combined vertical and braking is 4090 as

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against 2000 cycles. Hence the design is within the safe. The table-3 shows fatigue life from the analysis.

Table-3. Fatigue Life.

Load case	Fatigue life cycles	Target life cycles
Vertical Fatigue	427000	100000
Vertical and Braking combined Fatigue	4090	2000

4. CONCLUSIONS

The proposed engineering development process proved to be useful in reducing the development time and costs by the introduction of the highly sophisticated CAD/CAE tools as Hyper works 7.0 - Hyper mesh and Optistruct. Also the rework at the later stage avoided. The result clearly shows improving strength to weight ratio. The fatigue life result shows the highest reliability.

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