Optimization of Front Axle for Heavy Commercial Vehicle by Analytical and FEA Method

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Abstract - *An axle beam is a central shaft for rotating wheel. Front axle carries the weight of the front part of automobile as well facilitates steering and absorb shock due to road surface variations. So, proper design and optimization of front axle is extremely crucial to improve strength to weight ratio. The paper focuses on design, analysis and optimization of front axle. The approach in this research paper has been divided into two steps. The First step involves design of front axle by Analytical method. For this, the vehicle specification gross weight and payload capacity is used to determine the stress and deflection in the beam. Second step involved further modeling of front axle using CATIA-V5 and ANSYS software. For model optimization, FEA results were compared with analytical design.*

Key Words: ***Front axle beam, Analytical Design, Modeling Catia, Analysis and optimization ANSYS.***

Abbreviations: Y_{max} -Maximum deflection, W-Front axle weight, a-Length (Kingpin center to spring pad center hole), L-Front track, I-Moment of inertia, σ-Principle stress.

1. INTRODUCTION

The Automotive industry is one of the fastest growing sectors not only in India but all over the world. This industry includes automobiles, auto component sectors, commercial vehicles, multi -utility vehicles, passenger cars, two-wheelers and auto related parts. The front axle beam is one of the main parts of vehicle suspension system as it houses the steering assembly as well. Nearly 30 to 40% of the total vehicle weight is taken up by front axle. The front axle experiences load conditions such as static and dynamic loads due to irregularities of road, mostly during its travel on and off road. Front axles are subjected to both bending and shear stresses. In the static condition, the axle might be considered as beam supported vertically upward at the ends. Under the dynamic conditions, vertical bending moment is increased due to road roughness. Therefore axle must be resistant to tolerate additional stress and loads. The axles must additionally bear the weight of the vehicle plus any cargo. A misaligned front axle may result in improper turning of tires, reduced tire life, difficulty in driving and unsafe vehicle. The present research work deals with design of optimized front axle for heavy commercial vehicles. The cross-section areas and materials in different models are varied by adopting the analytical method. An existing front axle is modified for the given load condition. Further, actual deflection occurring in existing axle is studied and modified front axle with different materials and cross sectional areas is designed accordingly. The main objective of present research is to provide safe working conditions, effective stress concentration, optimum weight and cost reduction for front axle in heavy commercial vehicles.

2. CONSTRUCTION AND OPERATION

2.1 Front Axle

The front axle is designed to transmit the weight of automobile from the leaf springs to 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 center of gravity of the road vehicles, the entire center portion of the axle is dropped.

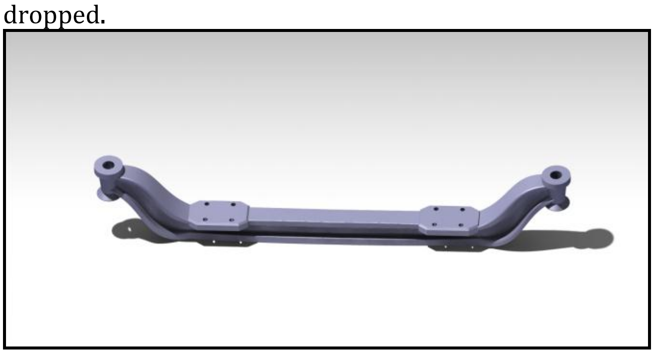


Fig 1: Front axle Beam

Front axle includes the axle-beam, stub-axles with swivel pin brake assemblies, track rod and stub-axle arm as shown in Figure-1.

2.1.1 Function of Front Axle

Front wheels of the vehicles are mounted on front axles. Functions of front axle are listed below:

a) It supports the weight of front part of the vehicle.
b) It facilitates steering.
c) It absorbs shocks which are transmitted due to road surface irregularities.
d) It absorbs torque applied on it due to braking of vehicle.

2.1.2 Types of front axles

a) Dead front axle
b) Live front axle

- **Dead front axle :**

Dead axle is those axles, which don't rotate. These axles have sufficient rigidity and strength to take the weight. The ends of front axle are suitably designed to accommodate stub axles.

- **Live front Axle:**

Live axle is used to transmit power from gear box to front wheels. Live front axle although resemble rear axles but they are different at the ends where wheels are mounted. Maruti-800 has live front axle.

2.2 Construction and Assembly

The front axle beam will have I cross section in the middle and circular or elliptical cross sections at the ends. An axle is usually a forged component for which a higher strength to weight ratio is desirable. The I-cross section has lower section modulus and hence gives better performance with lower weight. This type of construction produces an axle that is lightweight and yet has great strength. The I-beam axle is shaped so that the center part is several inches below the ends. This permits the body of the vehicle to be mounted lower than it could be if the axle were straight. A vehicle body that is closer to the road has a lower center of gravity and holds the road better. On the top of the axle, the springs are mounted on flat, smooth surfaces or pads. The mounting surfaces are called spring seats and usually have five holes. The four holes on the outer edge of the mounting surface are for the U bolts which hold the spring and axle together. The center hole provides an anchor point for the center bolt of the spring. The head of the center bolt, seated in the center hole in the mounting surface, ensures proper alignment of the axle with the vehicle frame. A hole is located in each end of the I-beam section. It is bored at a slight angle and provides a mounting point for the steering knuckle or kingpin. A small hole is drilled from front to rear at a right angle to the steering knuckle pinhole. It enters the larger kingpin hole very slightly. The kingpin retaining bolt is located in this hole and holds the kingpin in place in the axle. The steering knuckle is made with a yoke at one end and a spindle at the opposite end. Bronze bushings are pressed into the upper and lower arms of the yoke, through which the kingpin passes.

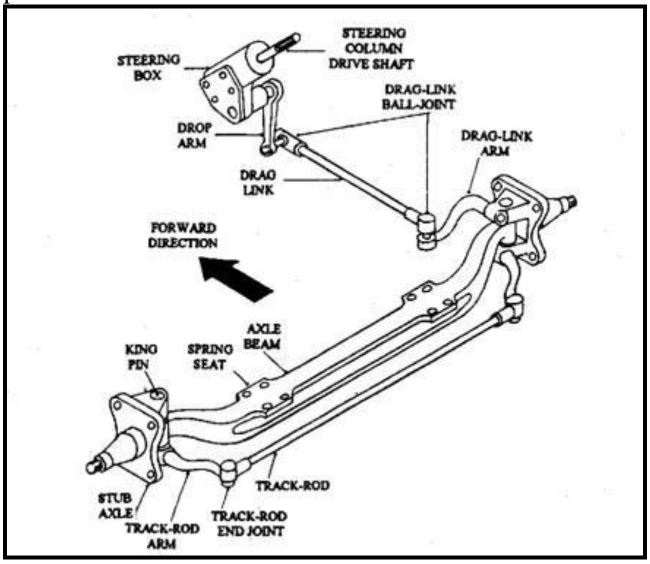


Fig 2: Front axle beam linkage assembly with steering system

These bushings provide replaceable bearing surfaces. A lubrication fitting and a drilled passage provide a method of forcing grease onto the bearing surfaces of the bronze bushings. The spindle is a highly machined, tapered, round shaft that has mounting surfaces for the inner and outer wheel bearings. The outer end of the spindle is threaded. These threads are used for installing a nut to secure the wheel bearings in position. A flange is located between the spindle and yoke. It has drilled holes around its outer edge. This flange provides a mounting surface for the brake drum backing plate and brake components. The kingpin acts like the pin of a door hinge as it connects the steering knuckles to the ends of the axle I-beam. The kingpin passes through the upper arm of the knuckle yoke, through the end of the I-beam and a thrust bearing, and then through the lower arm of the knuckle yoke. The kingpin retaining bolt locks the pin in position. The ball-type thrust bearing is installed between the I-beam and lower arm of the knuckle yoke so that the end of the I-beam rests upon the bearing. This provides a ball bearing for the knuckle to pivot on as it supports the vehicle's weight. When the vehicle is not in motion, the only job that the axle has to do is hold the wheels in proper alignment and support part of the weight. When the vehicle goes into motion, the axle receives the twisting stresses of driving and braking. When the vehicle operator applies the brakes, the brake shoes are pressed against the moving wheel drum. When the brakes are applied suddenly, the axle twists against the springs and actually twists out of its normal upright position. In addition to twisting during braking, the front axle also moves up and down as the wheels move over rough surfaces. Steering controls and linkages provide the means of turning the steering knuckles to steer the vehicle. As the vehicle makes a turn while moving, a side thrust is received at the wheels and transferred to the axle and springs. These forces act on the axle from many different directions. You can see, therefore, that the axle has to be quite rugged to keep all parts in proper alignment.

3. ANALYTICAL DESIGN

Analytical design need to find end solutions with helping system parameter inputs. The moment at section, bending stresses and principle stress is directly related to the strength. Now the above stresses are calculated by using following system input's and formula.

3.1 Vehicle Specification

Table -1: Vehicles parameters

Parameters	Unit	Value
Front axle weight (FAW)	Kg	4750
Gross vehicle Weight (GVW)	Kg	12900
C.G. Height (H)	mm	1350
Dynamic Radius (R)	mm	452
Front Track (L_t)	mm	1860
Distance of section (L_a)	mm	400
Wheel Base (L)	mm	4200
Deceleration during braking	--	0.8
Weight of front spring (W_s)	Kg	65.2
Unsprung mass of Front Axle (M_{fa})	Kg	420
Unsprung mass of Rear Axle (M_{ra})	Kg	730
Total unsprung mass (M_a)	Kg	1150
Braking torque ratio	--	0.23

3.2 I-Section Design

Initially consider four models of cross-section of front axle beam for calculating bending stress, tensional stress and Principal stress by taking I section dimensions. As sample I section shown below such as b1, b2, b3 are width of I section and h1, h2, h3 are the height of I section. And according to width and height four cross-section models is taken as below.

- **Sample model of I section**

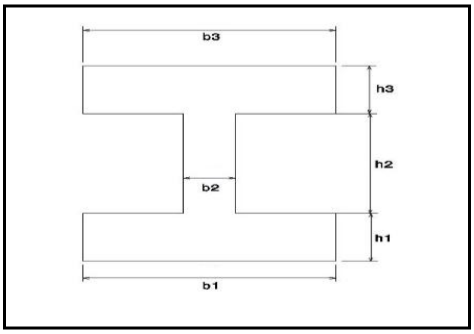
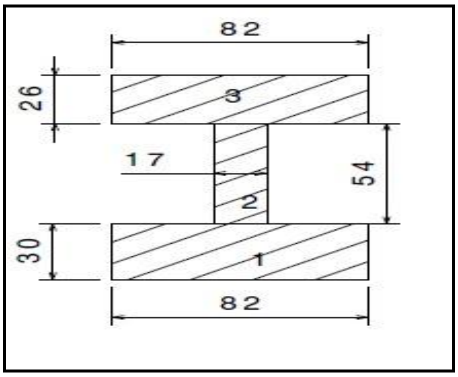


Fig 3: Axle I-section

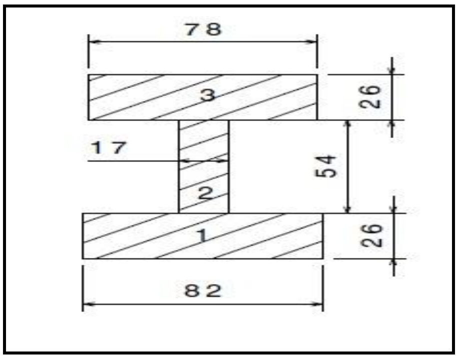
- **Model no. 1**

b_1=82mm, b_2=17mm, b_3=82mm
h_1=30mm, h_2=54mm, h_3=26mm



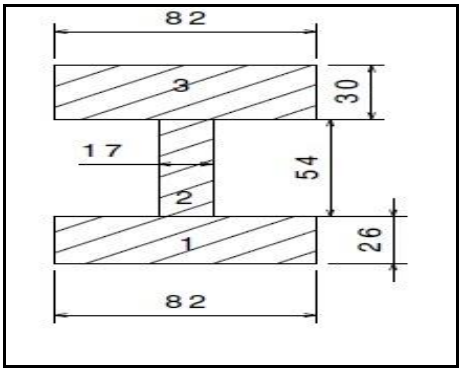
- **Model no.2**

b_1=82mm, b_2=17mm, b_3=78mm
h_1=26mm, h_2=54mm, h_3=26mm



- **Model no.3**

b_1=82mm, b_2=17mm, b_3=82mm
h_1=26mm, h_2=54mm, h_3=30mm



- **Model no.4**

b_1=82mm, b_2=17mm, b_3=82mm
h_1=26mm, h_2=54mm, h_3=26mm

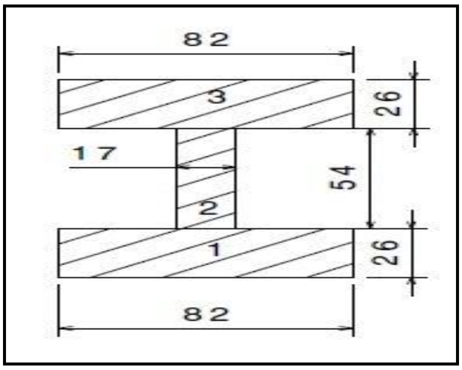


Table 2: Analytical Calculation and Result

Model no.	Principle Stresses $(\frac{kg}{mm^2})$	Max Deflection (mm)	Max Bending Stresses $(\frac{kg}{mm^2})$
1	21.46	4.84	17.21
2	21.56	6.61	19.8
3	23.74	5.88	18.52
4	25	5.46	19.5

From the above results, maximum deflection and principle stress for model no.1 is less and within yield limit as compared to all other models.

3.3 Deflection calculation along X direction:

Deflection at center of beam (max deflection)

$$Ymax = \frac{Wa(3L^2-4a^2)}{24EI}$$

Table 3: Analytical Calculation of Deflection for Different materials

Materials	Deflection (mm)	Stresses $(\frac{kg}{mm^2})$
SAE 1018	4.84	21.46
AISI 1020	5.085	21.46
Ductile cast-iron	9.246	21.46
SAE 4310	5.353	21.46

In proposed design work, by comparing selection of above material, SAE-AISI 1018 is better for manufacturing of Front axle beam. SAE-AISI 1018 has good forging, toughness, excellent weldability and load sustaining property.

3.4 Analytical Result

The value of principle stress of model no. 1 is 21.46 Kg/mm^2, but material Yield Limit for SAE-AISI 1018 is 40.77 Kg/mm^2. From the above Analytical calculation it has been observed that the SAE-AISI 1018 material is within material yield limit for model no. 1 for carrying forward the design, manufacturing and scope for weight optimization.

4. ANALYSIS OF FRONT AXLE BEAM

ANSYS mechanical software is a finite element analysis tool for structural analysis, including linear, nonlinear and dynamic studies. This computer simulation product provides finite elements to model behavior and supports material models and equation solvers for a wide range of mechanical design problems.

4.1 Material Description (SAE-AISI1018)

- **Mechanical Properties**

1) Poisson's Ratio: 0.29
2) Young's Modulus: 210 GPa
3) Yield Stress: 400 MPa
4) Elongation: 16% min
5) Tensile Ultimate Strength: 470 MPa
6) Density: 7900 kg/m^3

- **Chemical Compositions**

1) C: 0.15 - 0.20%
2) Fe: 98.8 – 99.25%
3) Mn: 0.6 - 0.9%
4) P: 0 - 0.040%
5) S: 0 - 0.050%

4.2 Finite Element Analysis

After application of boundary condition, the model no. 1 is solved. The deformation and stress has been plotted in ANSYS software module. The model of axle is simulated for four materials namely as SAE-AISI 1018, AISI 1020, Cast iron & SAE 4310.In FEA analysis,4mm deflection is considered for assembly of stub axle, knuckle pin and other parts.

- **For Material SAE-AISI 1018**

Max principle stress

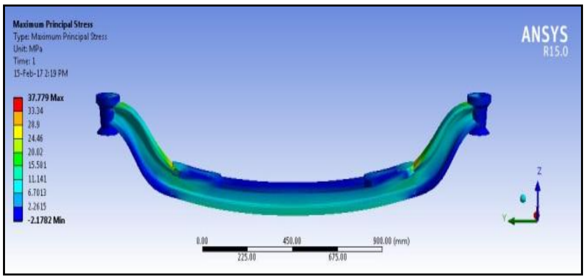


Fig 4: Max. Principle stress result for SAE1018

Total deflection

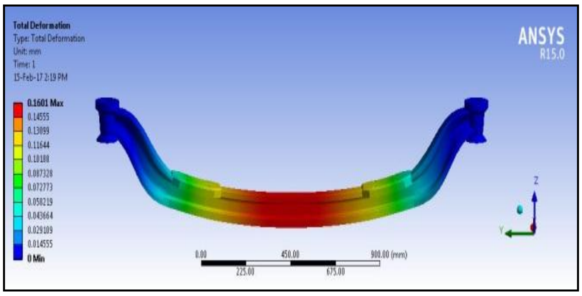


Fig 5: Deflection of axle for SAE1018

- **For Material AISI 1020**

Max principle stress

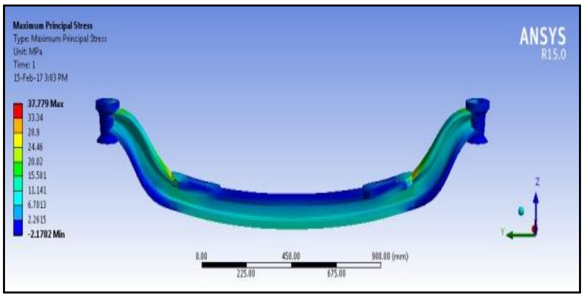


Fig 6: Max. Principle stress result for AISI1020

Total deflection

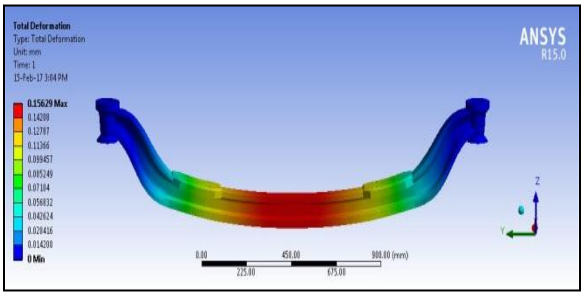


Fig 7: Deflection of axle for AISI1020

- **For Material CAST IRON**

Max principle stress

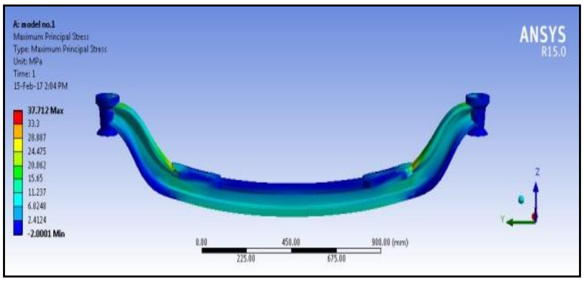


Fig 8: Max. Principle stress result for cast iron

Total deflection

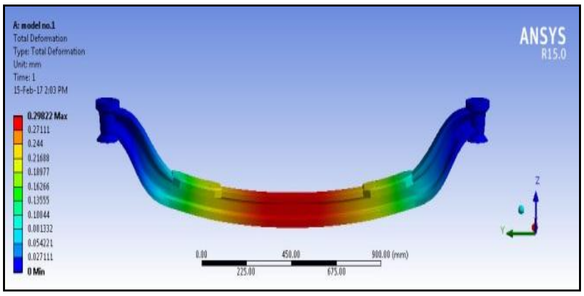


Fig 9: Deflection of axle for ductile cast iron

- **For Material SAE 4130**

Max principle stress

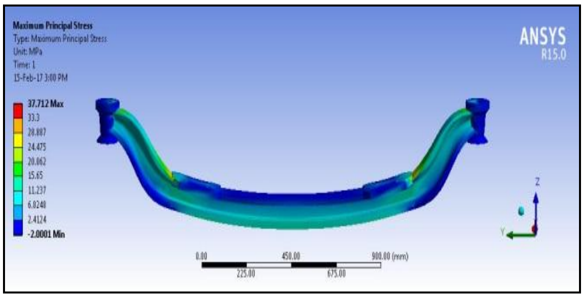


Fig 10: Max. Principle stress result for SAE4130

Total deflection

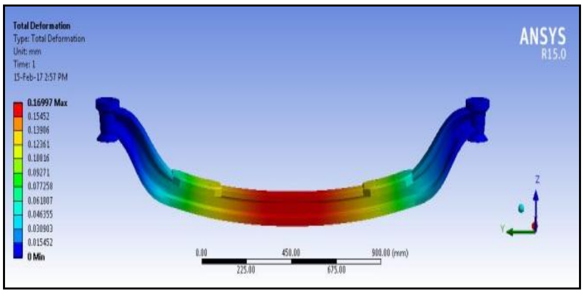


Fig 11: Deflection of axle for SAE4130

Table 4: Analysis overview table

Analysis type	Static Structural
No. of nodes	17137
No. of elements	9668
Type	3D

4.3. Comparison of Result

The table 5 shows the comparison between Analytical and FEA results. Deflection from FEA results are in good relation with Analytical results.

Table 5: Comparison of analytical calculation and FEA results

Materials	Parameters	Analytical Result	FEA Result
SAE1018	Y_{max}(mm)	4.84	4.16
	$\sigma\left(\frac{N}{mm^2}\right)$	21.46	37.78
AISI1020	Y_{max}(mm)	5.085	4.171S
	$\sigma\left(\frac{N}{mm^2}\right)$	21.46	37.79
Cast iron	Y_{max}(mm)	9.246	4.29
	$\sigma\left(\frac{N}{mm^2}\right)$	21.46	37.12
SAE4130	Y_{max}(mm)	5.353	4.175
	$\sigma\left(\frac{N}{mm^2}\right)$	21.46	37.712

From the above table 5, it is clear that maximum deflection in axle for SAE-AISI 1018 is 4.84mm and is less as compared to other materials. The maximum stress distribution is low and within stress yield limit. So SAE-AISI 1018 material is better for manufacturing of axle beam than other materials.

5. CONCLUSION

The static strength and dynamic characteristics are analyzed by ANSYS software. Analytical results confirm the modified design. By comparing analytical with FEA result, combine use of model no.1 and SAE1018 material gives best result and helps to avoid expensive and time consuming development loops. Moreover, it also allows the number of high-cost test carriers to be substantially reduced. This results in shortening of design period. Finally a safe and validate design has been developed to fulfill the requirement.

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