Failure Analysis And Design of Front Axle Beam of Heavy Commercial Vehicle By FEA Approach

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Abstract- *Front axle beam of heavy duty truck is the important component of vehicle and needs good design under the various loading conditions of the complete vehicle. Over the past century, automotive mass production has increased the demand for forged components. Front axle is the major component in an automotive chassis The Front Axle Beam is most important part in load carrying vehicle. The failure of Front Axle Beam is serious concern to heavy vehicle and thus for human life. So it is necessary to analyse the axle beam's ability to withstand typical service loading which develops stress in the beam resulting into failure. Further the objective of analysis is to improve its product quality. In the global competition, it is very important for the manufacturer to bring new product designs to market at a faster rate. Front axle carries the weight of front part of the automobile, as well as facilitates steering and absorbs shocks due to road surface variations. The front axle is designed to transmit the weight of the Automobile front the spring to the front wheels, turning right and left as required. An axle is a central shaft for a rotating wheel. On wheeled vehicles, the axle may be fixed to the wheels, rotating with them, or fixed to its surroundings, with the wheels rotating around the axle. The axles serve to transmit driving torque to the wheel, as well as to maintain the position of the wheels relative to each other and to the vehicle body. The front axle beam is one of the major parts of vehicle suspension system. It houses the steering assembly as well. Hence proper design of the front axle beam is extremely crucial. Up to 35% of the vehicle load carrying capacity is taken up by the Front axle beam. The front axle requires a properly designed support with high strength and stiffness. The aim is to determine the load capacity of the front axle beam of a Heavy Commercial vehicle and determine its behaviour at static condition. The paper deals with failure analysis and design of front axle beam. In present research work design of the front axle for Ashok Leyland 2516 XL heavy commercial vehicle were done which has a gross vehicle load of around 25 tons. The same analysis with help of FE results were compared with analytical design. For which paper has been divided in to two steps. In the first step front axle was design by analytical method. For this vehicle specification – its gross weight, payload capacity, braking torque used for subject to matter to find the principle stresses & deflection in the beam has been used. In the second step front axle were modelled in CAD software & analysis in ANSYS software The performance parameters such as stress, strain and displacement are measured. Stress analysis of front axle of truck combine under static loading conditions resulted from the applied modifications was performed by using finite element method. The commercial finite element package ANSYS 15.0 was used for the solution of the problem. Baseline analysis is carried out on the complete front axle assembly to extract the stress and displacement in the system. The FE results were compared with analytical design. The objective is to carried out failure analysis and design of front axle beam. The FEA is been carried out for vertical loads due to total weight carried by vehicle. An inertial load due to acceleration and deceleration of vehicle, which is, causes twisting of king pin portion with respect of PAD centreline. The geometry of axle is created in CREO 2.0 software which is imported to ANSYS 15.0. A fine congregate finite element model (meshed) is generated using the software to assess the strength and capability of the product to survive against all forces and vibrations. Compare the modified design with old design for improvement. Comparison study between hand calculations, FEA result.*

Keywords- Front Axle beam, Heavy commercial vehicle, Analytical Design, Modelling, Meshing, Analysis, Comparison of Results.

I. INTRODUCTION

An auto industry is one of the important and key sectors of the Indian economy. The auto industry includes of automobile sector, auto components sectors and includes commercial vehicles, passenger cars, multi-utility vehicles, two wheelers, three wheelers and related auto parts. There are many industrial sectors using this truck for their transportations such as the logistics, agricultures, factories and other industries. In global economic scenario optimum vehicle design & life prediction of different parts of vehicle like front axle, steering knuckle, crankshaft etc. is of major concern and at the same time improve a product quality. Increasing the productivity and accuracy are the two basic aims of mass production. According to the limit of domestic road conditions and traffic regulations, the front axle steering system is used in heavy commercial vehicles to increase carrying capacity and improve handling stability. But this steering system often gives rise to vehicle tire wear problem. Front axle is the one of the most important safety parts, and it is also the biggest and heaviest forging parts in Automobile, which request front axle to higher strength and fatigue strength. The parts are complex shape, symmetrical shapes, but big section fluctuate, especially the steel position and stopper position, its cross-section is not only deep but narrow, and it is one of the most difficulty long shaft type forging. The front axle of a truck is one of the major and very important components and needs to be designed carefully; this part also experiences the worst load condition such as static and dynamic loads due to irregularities of road, mostly during its travel on off road. 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. Front axle is made of I-section in the middle portion and circular or elliptical section or I section at the ends. The front axle beam is subjected to bending loads due to vertical forces due to mass present above in static condition of the vehicle, 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 along the length of the beam and unsymmetrical vertical loads due to centrifugal action. Worst situation arises while a cornering truck is braked to stop giving rise to turning moment on Pad and a retarding force acting on the surface of the Pad in the sense of vehicle motion. A live front axle contains the differential mechanism through which the engine power flows towards the front wheels. The front axles are generally dead axles, which does not transmit power. Axle experiences completely different loads in different direction, primarily bending load or vertical beaming due to curb weight and payload, torsion, due to drive torque, cornering load and braking load. Most of the mass of an axle beam is taken by the I section beam between the two leaf springs. Front axle beam takes about 30-35 % of total vehicle weight. While applying the break about 60% of the vehicle weight is acted on the front axle. Due to this higher loading capacity Solid axles are most commonly used in heavy duty commercial vehicles. The main three reasons of failure of mechanical components are Corrosion, wear and fatigue. Main reason for failure of front axle beam is the fatigue damage due to continuous fluctuating loads which acts from irregular road surface. A typical HCV front axle consists of main beam, stub axle, and swivel pin or kingpin. The wheels are mounted on stub axles. Front axle will experience a 3G load condition when the vehicle goes on the bump. In present research work design of the front axle for Ashok Leyland 2516 XL heavy commercial vehicle were done. The existing geometry of the front axle is modified to the optimum size which suits for functional life requirements. In this analysis, the geometry of the front axle is modified and a new design is proposed. In our project work, analytical analysis approach is used, (By considering change in existing shape and size). An existing front axle is redesigned for the given load condition, then check the actual deflection occurred in existing axle, also different shape axle get design, Select the best axle according to condition. Performing physical test for vertical beaming fatigue load is expensive and time consuming. So there is a necessity for building FE models which may virtually simulate these loads and can predict the behaviour. Even though the FEA produce fairly accurate results, solution accuracy heavily depends on accuracy of input conditions and overall modelling methodology used to represent the actual physics of problem. Therefore, validation of FEA model is of utmost importance. Typically, FEA model is validated by correlating FEA results analytical design. Hence correct design of the front axle beam is very critical. The approach in this paper has been divided into two steps. In the first step analytical method used to design front axle. For this, the vehicle specifications, its gross weight and payload capacity in order to find out the stresses and deflection within the beam has been used. In the second step front axle were modelled in Creo. The cad model was solved in ANSYS software system. The FE results were compared with analytical design.

Figure – Front Axle Beam

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

III. CONSTRUCTION & ASSEMBLY

The front axle is generally 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 centre 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 centre 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 centre hole provides an anchor point for the centre bolt of the spring. The head of the centre bolt, seated in the centre hole in the mounting surface, ensures proper alignment of the axle with the vehicle frame. 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. Therefore, that the axle has to be quite rugged to keep all parts in proper alignment. 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. 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. 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For design purpose the front axle beam of Ashok Leyland 2516 XL truck was chosen. All standard axles have an I cross section in the middle (spring seat to spring seat) and circular or elliptical cross sections at the ends. The front axle beam will have I cross section in the middle and circular 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. 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.

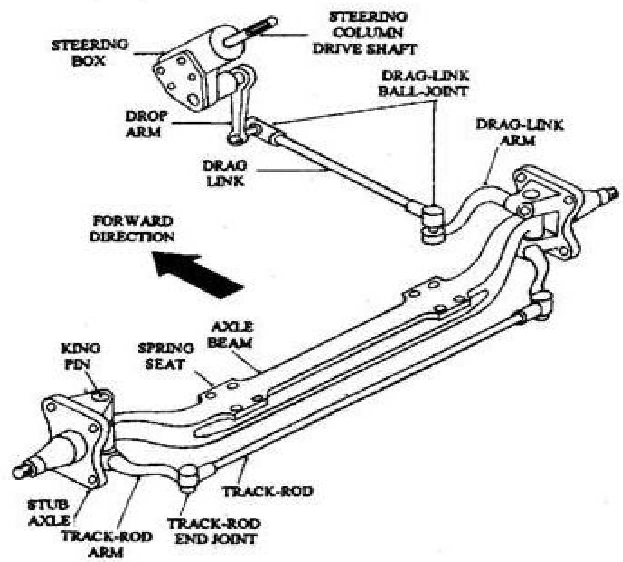

Figure- Front axle beam linkage assembly with steering system

IV. OPERATION

An axle is a central shaft used for rotating wheel. On wheeled vehicles, the axle could be mounted to the wheels, rotating with them, or located to its surroundings, with the wheels rotating around the axle. The axles achieve to transmit driving torque to the wheel. Also it can maintain the position of the wheels relative with each other and to the vehicle body. The axles must additionally bear the weight of the vehicle plus any cargo. The front axle beam is one of the main parts of vehicle suspension system. It houses the steering assembly as well. About thirty 30-40 percentage of the total vehicle weight is taken up by front axle. 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. It consists of main beam, stub axle, and swivel pin, etc. The wheels are mounted on stub axles. 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. 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.

V. COMPONENTS OF FRONT AXLE BEAM

The Front axle is fitted with wheels at its ends using steering knuckle and king pin as shown in figure. The king pin inclination is desirable for alignment of wheels. The part of vehicle weight is transmitted through the wheels through this axle. Front axle beam supports the weight of front parts of the vehicle. It absorbs shocks which are transmitted due to road surface irregularities and gives the cushioning effect and also when brakes are provided at the front wheels, it withstands bending and torsional stresses. All standard front axles have an ‘I’ Cross section between the left spring pad to right spring pad and rectangular or circular 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 higher Section modulus and hence gives better performance with lower weight. This type of construction produces an Axle which is lightweight and has good strength. When the vehicle goes into motion, the axle receives the torsional stresses of driving and braking. When the brakes are applied suddenly, the axle twists against the springs and actually twists about normal upright position. In addition to twisting during braking, the front axle moves up and down as the wheels move on rough surfaces roads and also cornering force applied as vehicle takes turn.

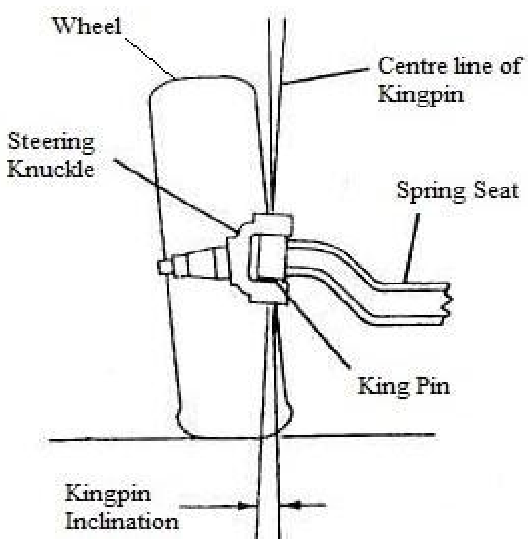

Figure- Components of front axle beam

VI. TYPE OF FRONT AXLE

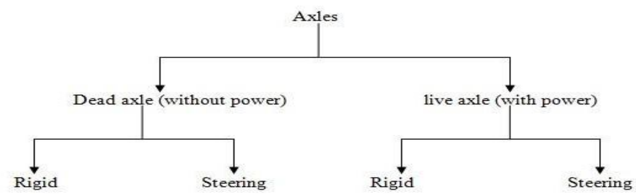

Figure- Type of front axle

VII. METHODOLOGY

1. Data accumulation related to front axle will be done.
2. CAD model generation of front axle beam.
3. Mathematical modeling/analytical design of existing model.
4. During this proposed engineering development process, import the geometry of the 3D CAD model is in the FEA software for displacement, stress and strain analysis.
5. Mesh the product geometry using analysis software.
6. Apply boundary conditions to be analysed.
7. The results of this analysis are then used to formulate new product geometry with the 3D Parametric CAD software, which is analysed until a satisfactory stress and strain map is obtained.
8. Mathematical modeling/analytical design of modified model.
9. Modifications in CAD model.
10. Finite Element Analysis of modified model of front axle beam.
11. Comparison of the existing and modified version.
12. The obtained FEA results are verifies with the mathematical modelling.
13. Conclusion.

VIII. TRUCK MODEL

Ashok Leyland 2516 XL

Figure -Truck model

IX. TRUCK SPECIFICATION

Emission norms	BS-3
Engine cylinder	6
Displacement	5600 CC
Maximum power	160 BHP @ 2400 RPM
Maximum torque	550 N-M @ 1700 RPM
Transmission	Manual
Clutch	15’’ facing diameter clutch
Gear box	6 speed
Fuel tank	330 litre
Gradeability	18%
Turning radius	11000 mm
Max.speed	79 km/hr
Body option	customizable
Chassis type	Chassis with cowl
Cabin type	No cabin
Axle configuration	6*2
Front tyre	10*20 – 16 PR
Wheel base	6200 mm
Overall length	12000 mm
Overall width	2462 mm
Ground clearance	260 mm
Gross vehicle weight	25000 kg
Kerb weight	6640 kg
Payload Brakes	16000 kg Air brakes
Front axle	Heavy duty forged I beam
Front suspension	Semi elliptical laminated multileaf
Rear axle	Single speed Hyphoid gear axle
Rear suspension	Semi elliptical laminated multileaf

Table -Truck specification

X. AXLE DETAILS

- Type: Front Non-Drive Steer Axle (Heavy Duty Truck)
- Axle rating: 8750 kg
- **Material: AISI 1045**

Material property	
Yield Strength	884 MPa
Ultimate Tensile Strength	991 MPa
Poisson ratio	0.3
Modulus of Elasticity	$2.1*10^5$ MPa

Table - Material property

Sr. No.	**Element**	**Weight Ratio**
1.	Carbon, (C)	0.43-0.50 %
2.	Manganese, (Mn)	0.60-0.90 %
3.	Phosphorous, (P)	≤ 0.040 %
4.	Sulphur, (S)	≤ 0.050 %

Table- Chemical composition

XI. INPUT DATA FROM THE VEHICLE UNDER ANALYSIS

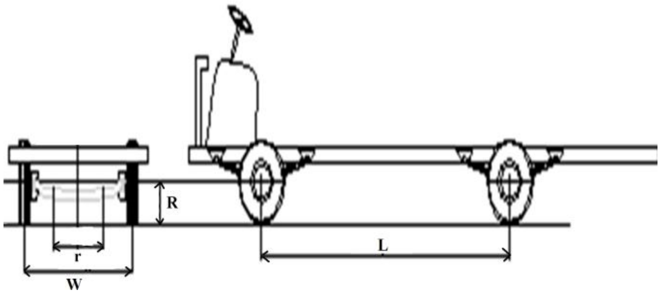

Figure -Position and various parameters of FAB

1. Gross vehicle weight = 25000 kg
2. Payload = 16000 kg
3. Kerb weight = 6640+2360=9000 kg
4. Front axle load = 8750 kg
5. Weight of wheel rim + tyre + tube = 110kg
6. K.P eye to K.P eye distance = 1800 mm
7. Pad to pad distance (r)= 1000 mm
8. Spring pad width = 220 mm
9. Spring pad length = 138 mm
10. Radius of wheel (R) = 535 mm
11. Overall length = 12000 mm
12. Overall width = 2432 mm
13. Front track = 1915 mm
14. Ground clearance = 260 mm
15. Turning radius = 11000 mm
16. Wheel base (L) = 6200mm
17. K.P bore diameter = 500 mm
18. K.P outer diameter = 86 mm
19. K.P height = 88 mm
20. K.P to pad distance = 400 mm

XII. DIFFERENT VIEW OF FRONT AXLE BEAM

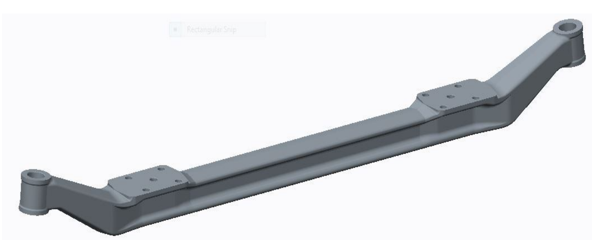
Figure- 3D view


Figure-Front view


Figure-Top view


Figure-R.H.S view

PARAMETERS	EXISTING MODEL	MODIFIED MODEL
Moment of inertia(I)	$4.903*10^6$ mm^4	$7.416*10^6$ mm^4
Max. deflection(y_{max})	18.89 mm	12.49 mm
Max. shear stress(ζmax)	82.72 N/mm^2	69.20 N/mm^2
Bending stress($\sigma b_{tensile}$)	533.27 N/mm^2	397.43 N/mm^2
Bending stress($\sigma b_{compressive}$)	464.58 N/mm^2	331.73 N/mm^2

Table-Result of analytical design

XIII. FEA APPROACH

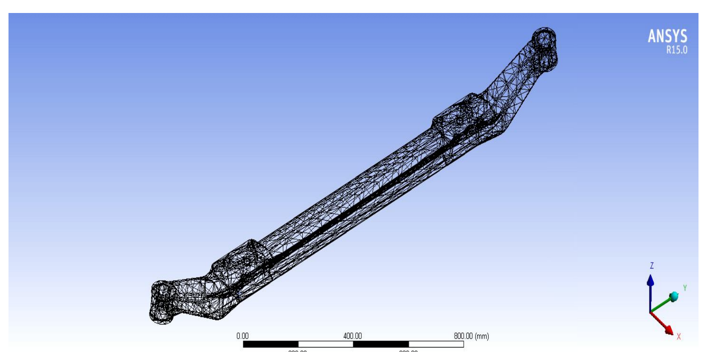

Figure-Meshing

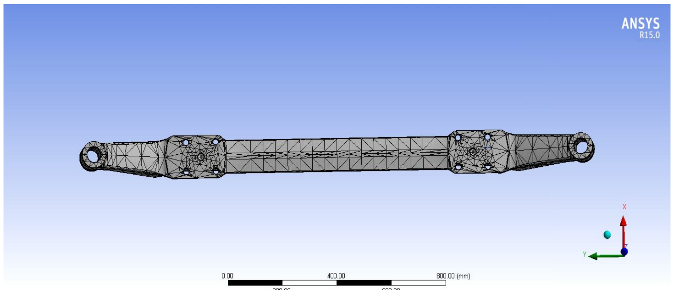

Figure-Top view meshing

Boundary condition

$U_X = 0$

$U_Y = 0$

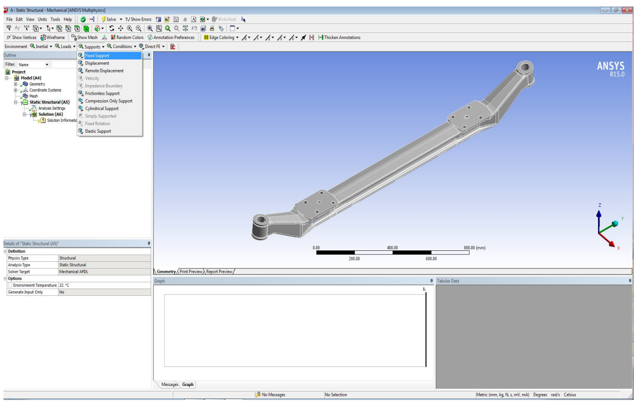

Figure-Boundary condition

Force and Fixed Support

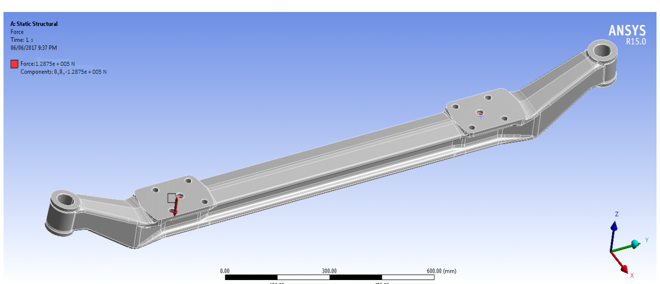

Figure-Force

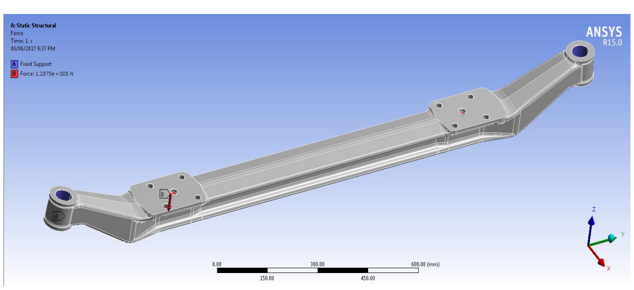

Figure-Fixed support

Solutions of existing model

Maximum deflection

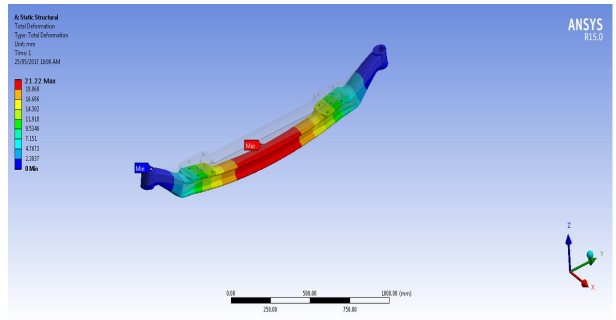

Figure-Maximum deflection = 21.22 mm

Maximum Shear Stress

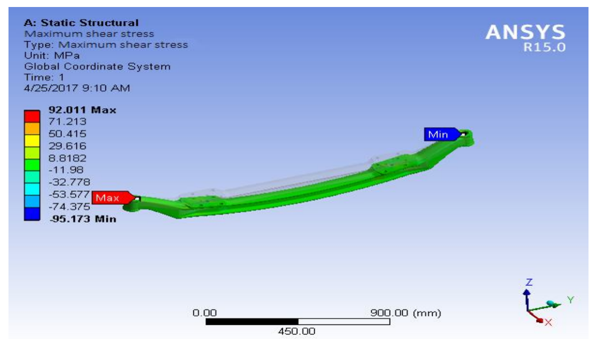

Figure-Maximum shear stress = 92.011 MPa

Equivalent (Von-Mises) Stress

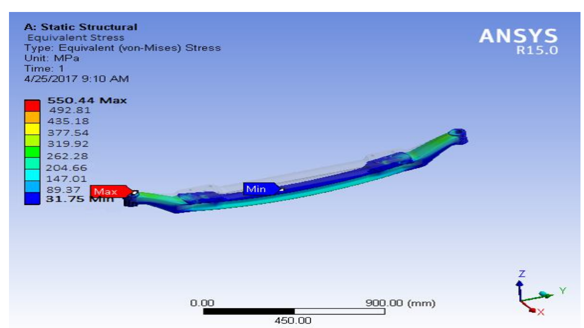

Figure-Equivalent (Von-Mises) stress = 550.44 MPa

Maximum Principal Stress

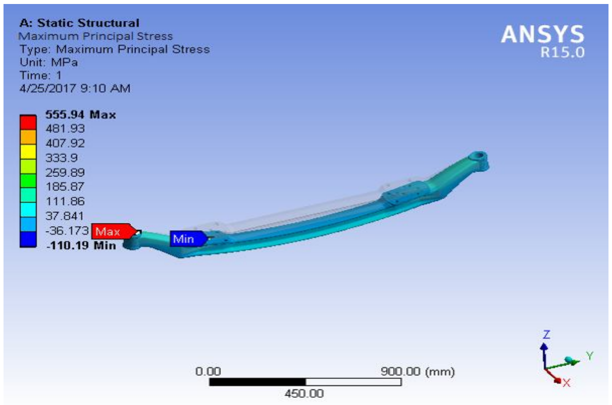

Figure-Maximum principal stress = 555.94 MPa

Middle Principal Stress

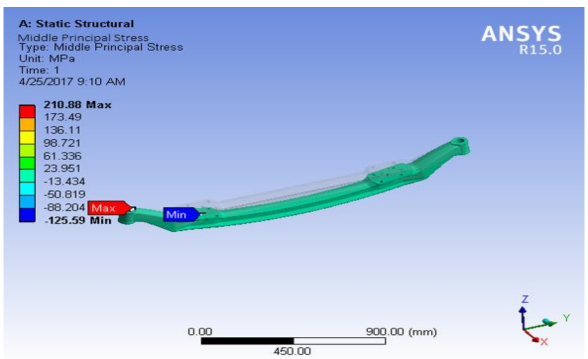

Figure-Middle principal stress = 210.88 MPa

Minimum Principal Stress

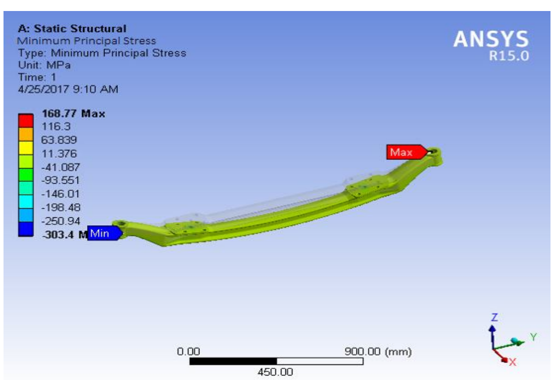

Figure-Minimum principal stress = 168.77 MPa

Normal Stress

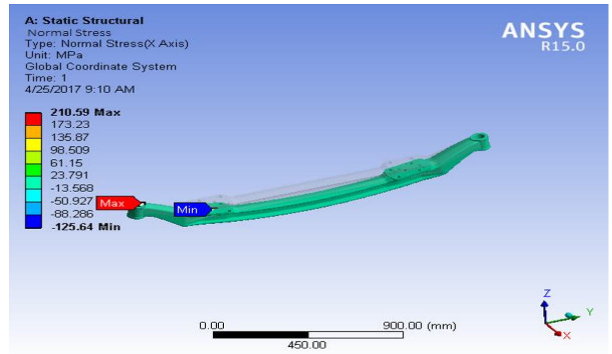

Figure-Normal stress = 210.59 MPa

Equivalent Elastic Strain

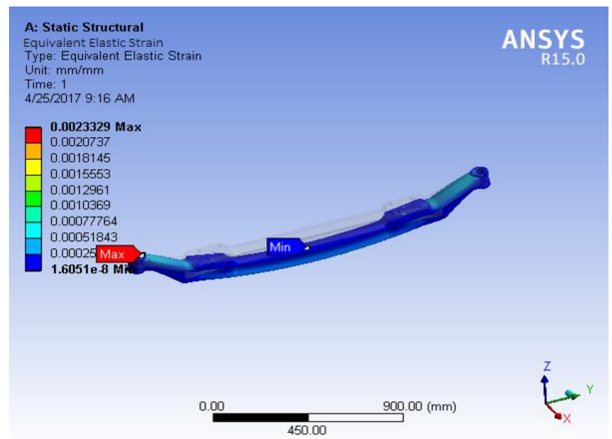

Figure-Equivalent elastic strain = 0.0023329

Maximum Principal Elastic Strain

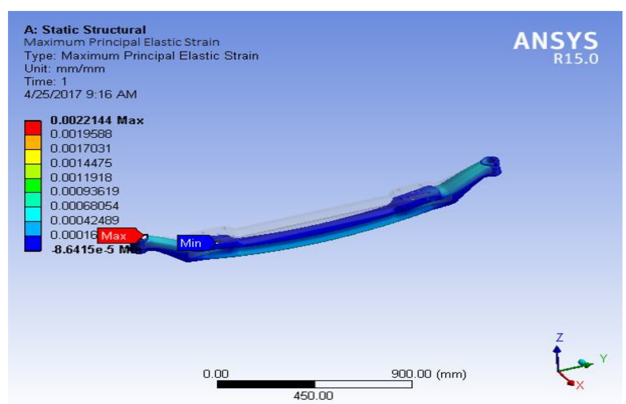

Figure-Maximum principal elastic strain = 0.0022144

Middle Principal Elastic Strain

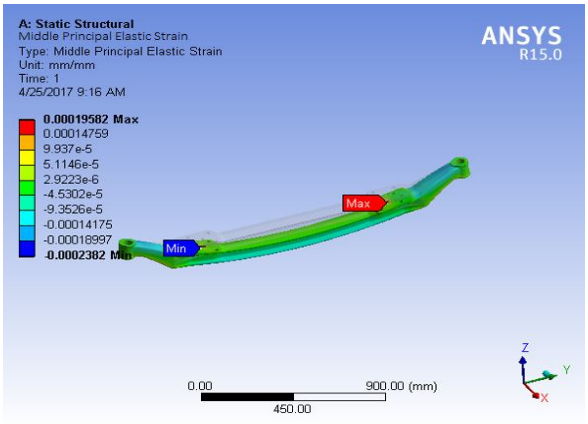

Figure-Middle principal elastic strain = 0.00019582

Minimum Principal Elastic Strain

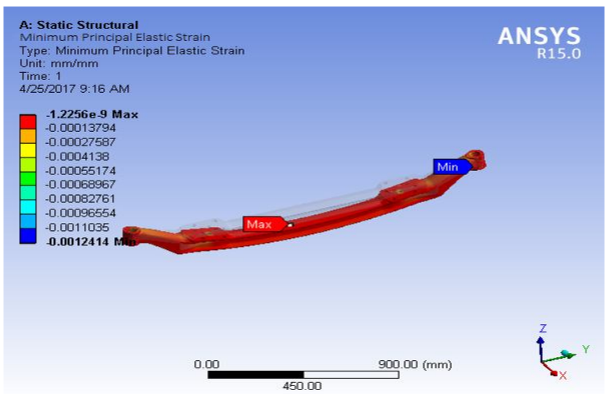

Figure-Minimum principal elastic strain = -1.2256e-9

Normal Elastic Strain

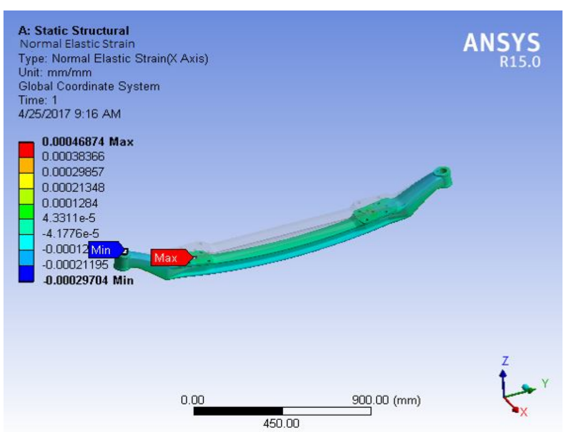

Figure-Normal elastic strain

Result of Different Material

SAE 1020 (Carbon Steel)

$E = 2.5 * 10^5$ MPa

$S_{yt} = 246$ MPa

a) Maximum Deflection

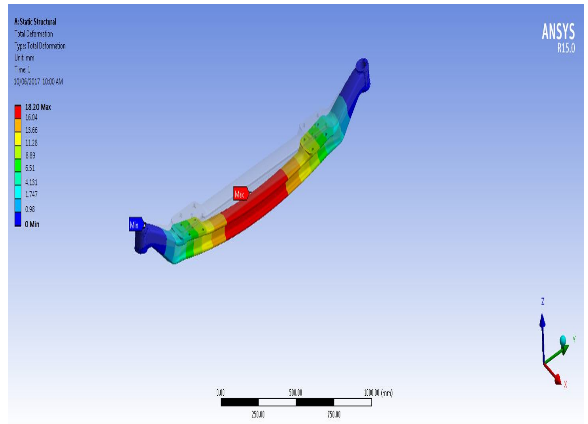

Figure-Maximum deflection = 18.20 mm

1. Here deflection = 18.2 mm

   18.2 mm < 21.22 mm

Deflection is less than existing

2. Von-Misses Stress = 538.25 MPa

   538.25 MPa > 246 MPa

   Von-Misses Stress > yield strength

   Hence, design is not safe

b) Equivalent (Von-Mises) Stress

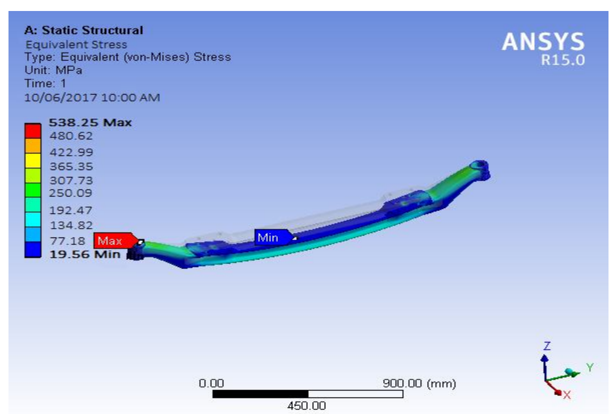

Figure-Equivalent (Von-Mises) stress = 538.25 MPa

SAE 3240 (Nickel Chromium Steel)
E = 2.08 * 10^5 MPa
S_{yt} = 810 MPa

a) **Maximum Deflection**

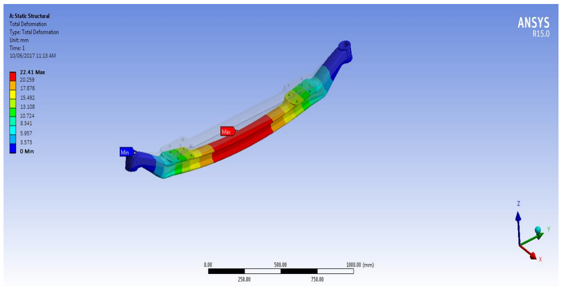

Figure-Maximum deflection = 22.41 mm

b) **Equivalent (Von-Mises) Stress**
c)

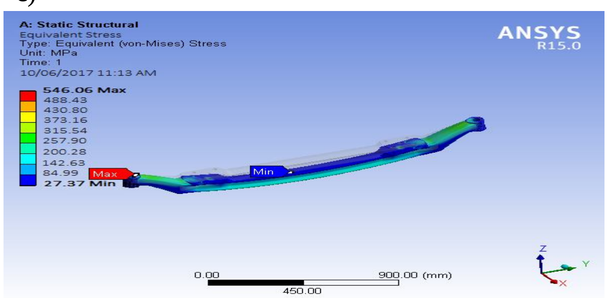

Figure-Equivalent (Von-Mises) stress = 546.06 MPa

1. Here deflection = 22.41 mm
22.41 mm > 21.22 mm

Deflection is greater than existing

2. Von-Misses Stress = 546.06 MPa
546.06 MPa < 810 MPa
Von-Misses Stress < yield strength

Hence, design is safe

AISI 1053
E = 2.1 * 10^5 MPa
S_{yt} = 470 MPa

a) **Maximum Deflection**

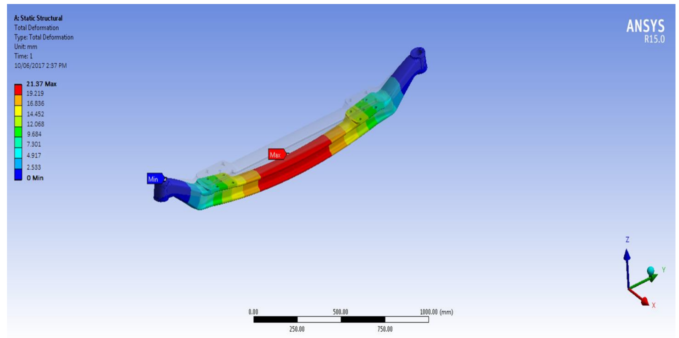

Figure-Maximum deflection = 21.37 mm

b) **Equivalent (Von-Mises) Stress**
c)

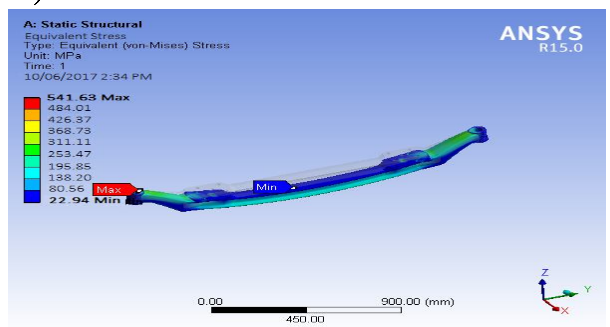

Figure-Equivalent (Von-Mises) stress = 541.63 MPa

1.Here deflection = 21.37 mm
21.37 mm > 21.22 mm

Deflection is greater than existing

2. Von-Misses Stress = 541.63 MPa
541.63 MPa > 470 MPa
Von-Misses Stress > yield strength

Hence, design is not safe

Cast Iron
E = 2 * 10^5 MPa
S_{yt} = 520 MPa
1.Here deflection = 22.17 mm
22.17 mm > 21.22 mm

Deflection is greater than existing

2. Von-Misses Stress = 543.27 MPa
543.27 MPa > 470 MPa
Von-Misses Stress > yield strength

Hence, design is not safe

a) Maximum Deflection

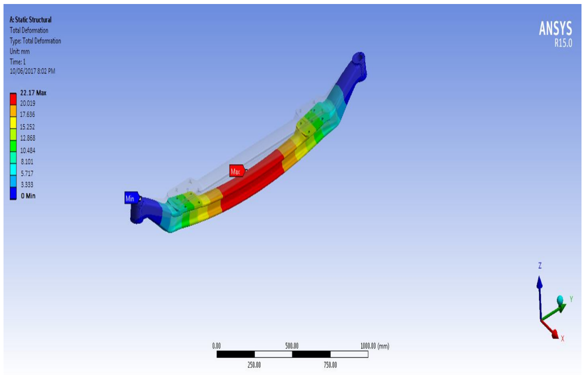

Figure-Maximum deflection = 22.17 mm

a) Equivalent (Von-Mises) Stress

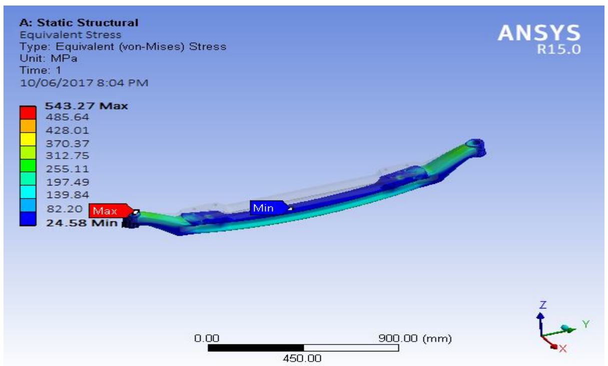

Figure-Equivalent (Von-Mises) stress = 543.27 MPa

Result of Different Cross section (AISI 1045) CIRCULAR

Case 1]

Deflection = 19.12 mm

Von-Mises stress = 582.51 MPa

Mass = 138.41 kg

D = 100 mm

Stress decreases

Deflection decreases

Weight increases

a) Maximum Deflection

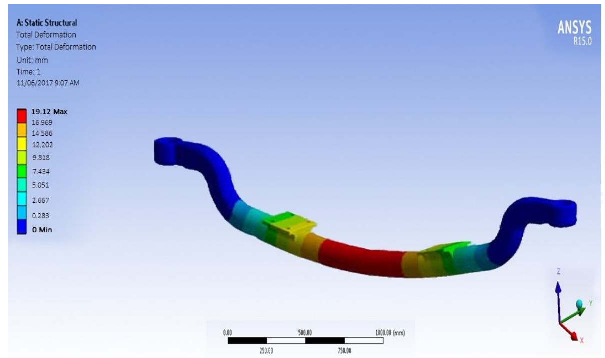

Figure-Maximum deflection = 19.12 mm

b) Equivalent (Von-Mises) Stress

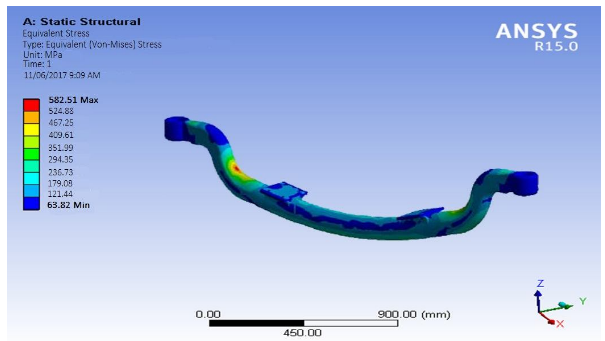

Figure-Equivalent (Von-Mises) stress = 582.51 MPa

Case 2]

Deflection = 29.40 mm

Von-Mises stress = 691.29 MPa

Mass = 89 kg

D = 65 mm

Stress increases

Deflection increases

Weight decreases

a) Maximum Deflection

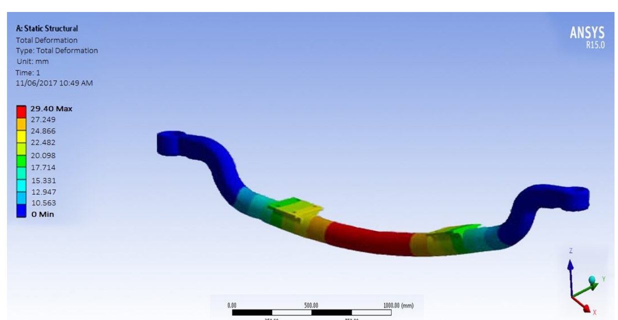

Figure- Maximum deflection = 29.40 mm

b) Equivalent (Von-Mises) Stress

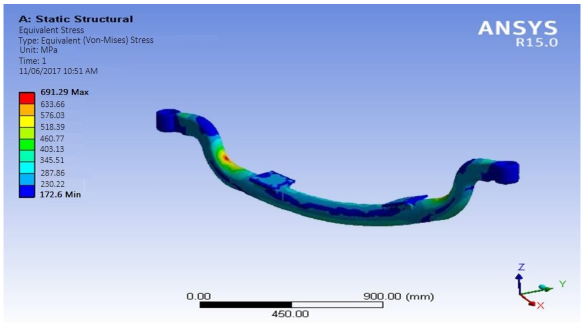

Figure-Equivalent (Von-Mises) stress = 691.29 MPa

SQUARE

Case 1]

Deflection = 15.07 mm

Von-Mises stress = 412.35 MPa

Mass = 141.80 kg

90 mm * 90 mm

Stress decreases

Deflection decreases

Weight increases

a) Maximum Deflection

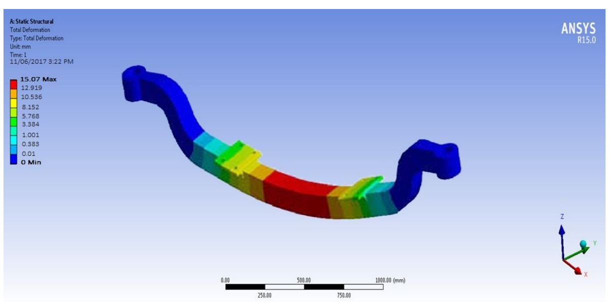

Figure-Maximum deflection = 15.07 mm

b) Equivalent (Von-Mises) Stress

c)

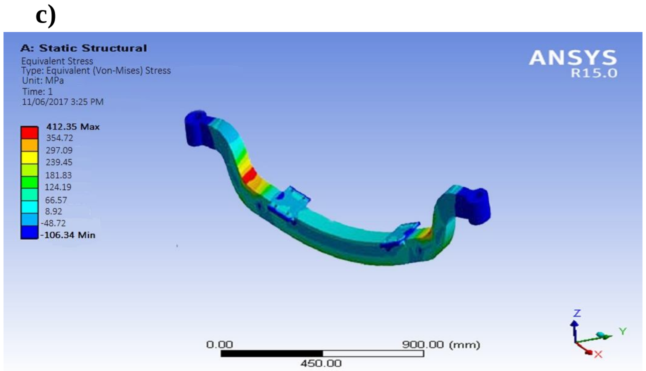

Figure-Equivalent (Von-Mises) stress = 412.35 MPa

Case 2]

Deflection = 25.35 mm

Von-Mises stress = 521.13 MPa

Mass = 89 kg

57 mm * 57 mm

Stress increases

Deflection increases

Weight decreases

a) Maximum Deflection

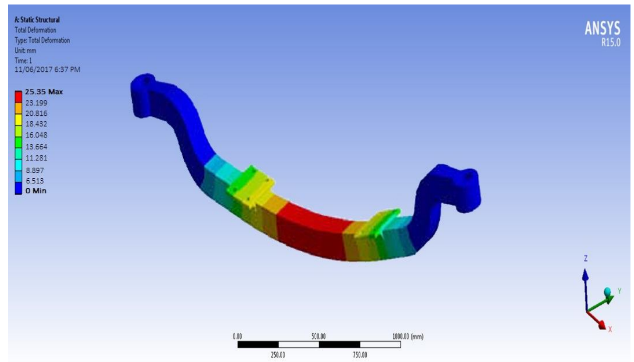

Figure-Maximum deflection = 25.35 mm

b) Equivalent (Von-Mises) Stress

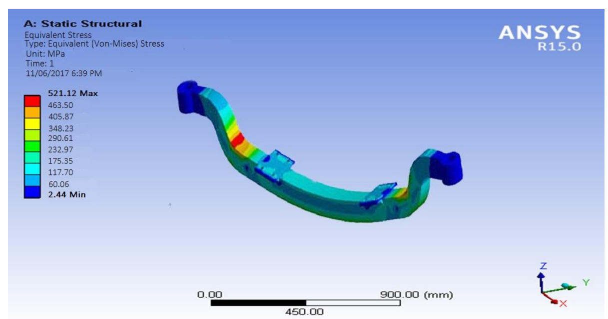

Figure-Equivalent (Von-Mises) stress = 521.12 MPa

Solutions of Modified Model

Maximum Deflection

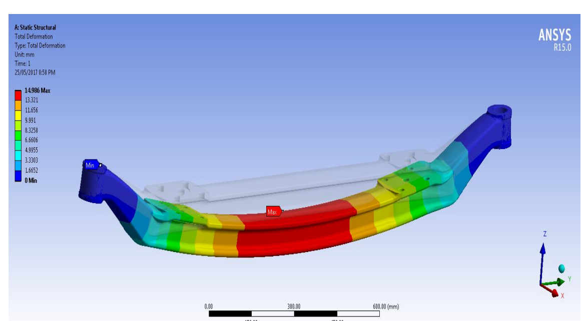

Figure-Maximum deflection = 14.986 mm

Maximum Shear Stress

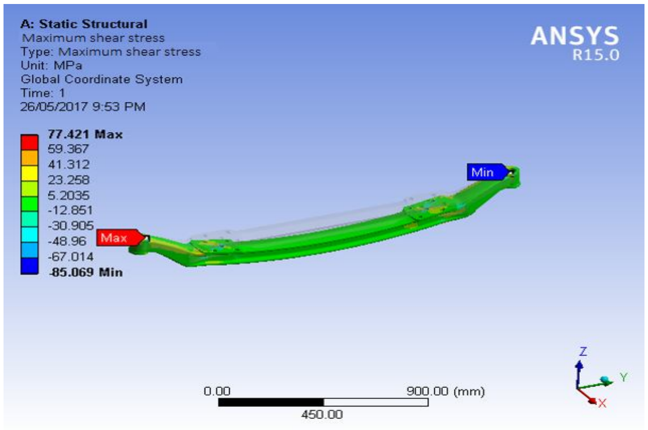

Figure-Maximum shear stress = 77.421 MPa

Equivalent (Von-Misses) Stress

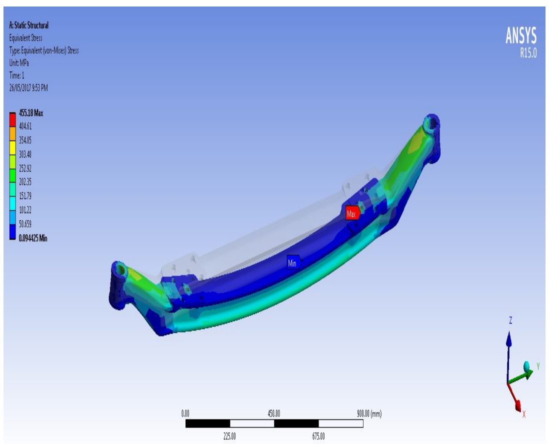

Figure-Equivalent (Von-Mises) stress = 455.18 MPa

Maximum Principal Stress

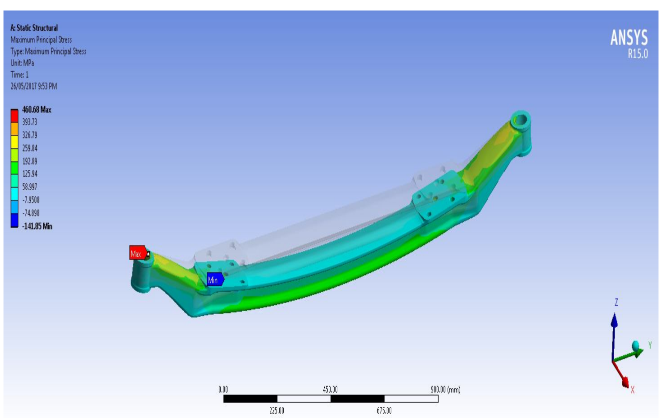

Figure-Maximum principal stress = 460.68 MPa

Middle Principal Stress

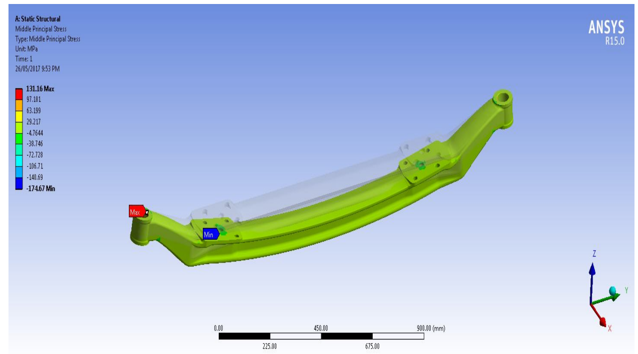

Figure-Middle principal stress = 131.16 MPa

Minimum Principal Stress

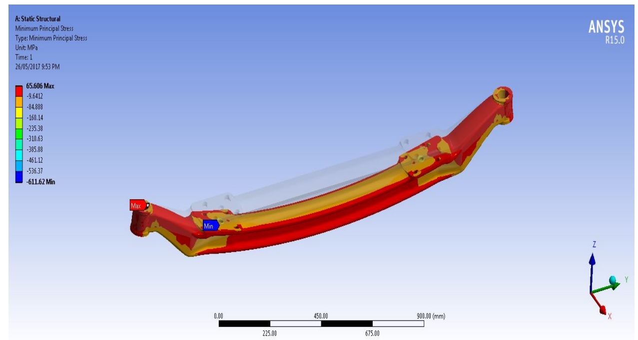

Figure-Minimum principal stress = 65.606 MPa

Normal Stress

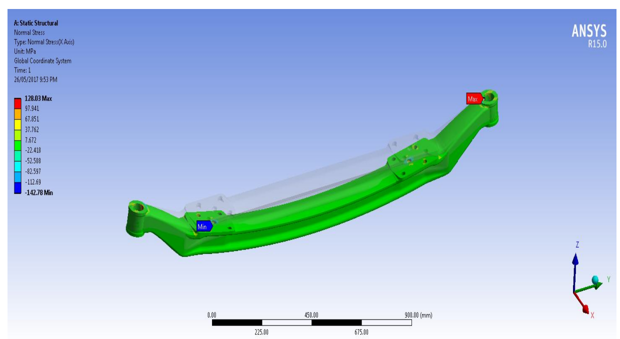

Figure-Normal stress = 128.03 MPa

Equivalent Elastic Strain

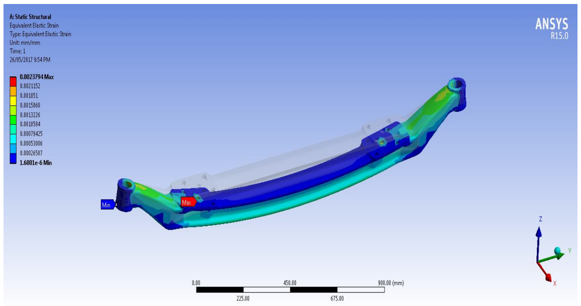

Figure-Equivalent elastic strain = 0.0023794

Maximum Principal Elastic Strain

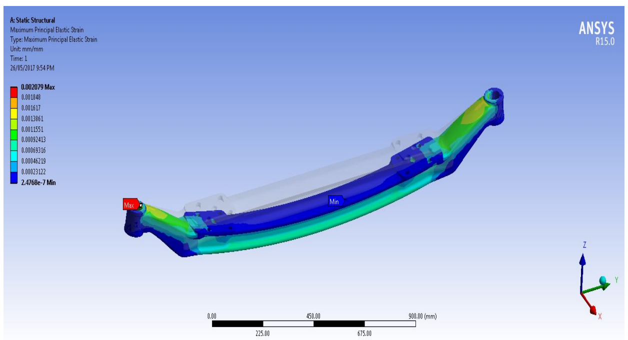

Figure-Maximum principal elastic strain = 0.002079

Middle Principal Elastic Strain

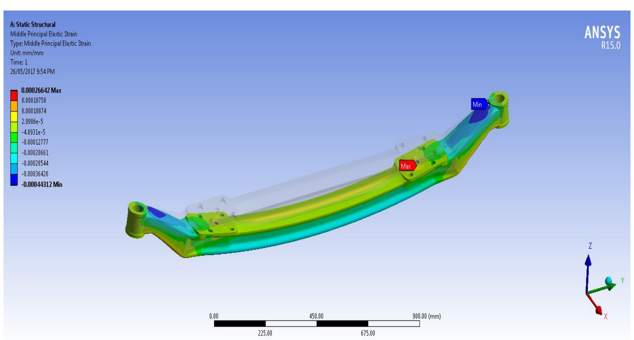

Figure-Middle principal elastic strain = 0.00026642

Minimum Principal Elastic Strain

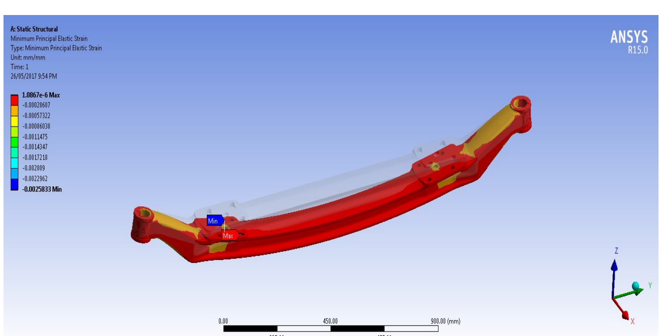

Figure-Minimum principal elastic strain = 1.0867e-6

XIV. EXISTING MODEL RESULTS

PARAMETERS	RESULTS
Maximum deflection	21.22 mm
Maximum shear stress	92.01 MPa
Equivalent(von-misses) stress	550.44 MPa
Maximum principal stress	555.94 MPa
Middle principal stress	210.88 MPa
Minimum principal stress	168.77 MPa
Normal stress	210.59 MPa
Equivalent elastic strain	0.0023
Maximum principal elastic strain	0.0022
Middle principal elastic strain	0.0019
Minimum principal elastic strain	$1.22*10^{-9}$
Safety factor	1.6686
Equivalent(von-misses) stress	726.05 MPa
Maximum principal stress	791.35 MPa
Maximum shear stress	130.97 MPa
Maximum deflection	28.007 mm

Table-Existing model results

XV. RESULT OF DIFFERENT MATERIALS

MATERIAL							
SAE 1020		SAE 3240		AISI 1053		CAST IRON	
Max. deflection (mm)	Von-Mises Stress (MPa)	Max. deflection (mm)	Von-Mises Stress (MPa)	Max. deflection (mm)	Von-Mises Stress (MPa)	Max. deflection (mm)	Von-Mises Stress (MPa)
18.20	538.25	22.41	546.06	21.37	541.63	22.17	543.27

Table-Result of different materials

XVI. RESULT OF DIFFERENT CROSS SECTION

CIRCULAR					
CASE 1			CASE 2		
Mass (kg)	Maximum deflection (mm)	Von-Mises Stress (MPa)	Mass (kg)	Maximum deflection (mm)	Von-Mises Stress (MPa)
138.41	19.12	582.51	89	29.40	691.29

SQUARE					
CASE 1			CASE 2		
Mass (kg)	Maximum deflection (mm)	Von-Mises Stress (MPa)	Mass (kg)	Maximum deflection (mm)	Von-Mises Stress (MPa)
141.80	15.07	412.35	89	25.35	521.13

Table-Result of different cross section

XVII. MODIFIED MODEL RESULTS

PARAMETERS	RESULTS
Maximum deflection	14.98 mm
Maximum shear stress	77.42 MPa
Equivalent(von-misses) stress	455.18 MPa
Maximum principal stress	460.68 MPa
Middle principal stress	131.16 MPa
Minimum principal stress	65.60 MPa
Normal stress	128.03 MPa
Equivalent elastic strain	0.0023
Maximum principal elastic strain	0.0020
Middle principal elastic strain	0.00026
Minimum principal elastic strain	$1.08*10^{-6}$
Safety factor	5.3944
Equivalent(von-misses) stress	572.1 MPa
Maximum principal stress	632.21 MPa
Maximum shear stress	106.24 MPa
Maximum deflection	18.82 mm

Table-Modified model results

XVIII. COMPARISON OF RESULTS

PARAMETERS	EXISTING MODEL	MODIFIED MODEL
Maximum deflection	21.22 mm	14.98 mm
Maximum shear stress	92.01 MPa	77.42 MPa
Equivalent(von-misses) stress	550.44 MPa	455.18 MPa
Maximum principal stress	555.94 MPa	460.68 MPa
Middle principal stress	210.88 MPa	131.16 MPa
Minimum principal stress	168.77 MPa	65.60 MPa
Normal stress	210.59 MPa	128.03 MPa
Equivalent elastic strain	0.0023	0.0023
Maximum principal elastic strain	0.0022	0.0020

Middle principal elastic strain	0.0019	0.00026
Minimum principal elastic strain	$1.22*10^{-9}$	$1.08*10^{-6}$
Safety factor	1.6686	5.3944
Equivalent(von-misses) stress	726.05 MPa	572.1 MPa
Maximum principal stress	791.35 MPa	632.21 MPa
Maximum shear stress	130.97 MPa	106.24 MPa
Maximum deflection	28.007 mm	18.82 mm

Table-Comparison of results

XIX. VALIDATION OF RESULTS

	EXISTING MODEL		MODIFIED MODEL	
PARAMETERS	ANALYTICAL	FEA	ANALYTICAL	FEA
Maximum deflection (mm)	18.89	21.22	12.49	14.98
Maximum shear stress (MPa)	82.72	92.01	69.20	77.42

Table-Validation of results

XX. DISCUSSION

1. Result of existing model are studied such as maximum deflection, maximum shear stress, equivalent (von-mises) stress, maximum principal stress, etc.
2. Result of different materials are solved such as SAE 1020, SAE 3240, AISI 1053, Cast iron. From this, existing material (AISI 1045) have best result which is suitable for requirement.
3. Result of different cross section are analysed such as circular, square, I-section have best result to suit the requirements.
4. By taking I-section and material as AISI 1045 with changes in geometric dimension of I-section, result of modified model are analysed such as maximum deflection, maximum shear stress, equivalent (von-mises) stress, maximum principal stress, etc.
5. In this project, the analysis of front axle beam is done through ANSYS analysis software. The result coming from these are reasonably (85-95%) same as analytical design hence it can conclude that software predict us best information and its result can be trusted.
6. The deflection in the finite element analysis confirms that the boundary condition for front axle beam are correctly simulated. Correlation between stress result from analytical design and from finite element analysis assures that the mesh size and modelling approach used for the component were well defined. Finally, we were able to deliver a safe and validate design to suit the requirements.
7. It is concluded from stress analysis of front axle beam that the induced stress in front axle beam during full load condition are well below to yield stress of front axle beam material, hence front axle beam can sustained all types of loads during running.
8. Geometric modification is done to reduce deflection and stress concentration above analysis is done to validate results. In allusion to the tire wear and other problems are caused by large deflection of front axle beam of heavy commercial vehicles.

XXI. CONCLUSION

As the maximum principle stress increases beyond the yield limit, the crack initiation begin to happen as the maximum principle stress acts as tension in the material and shear stresses are completely absent in the material. Minimum principle stress always tends towards zero and doesn't create any serious deformations. Middle principle stress is compressive in nature; it tries to close the crack. If the middle principle 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. In this case equivalent stress plays a major role and acts as the decision maker. Mostly I-section is used for front axle beam and there will be wide scope in analysing the other cross section beam like I-cross section, circular cross section, square cross section. Stress distribution analysis of front axle beam was investigated by using finite element analysis. The model of the axle was developed by using Creo-2.0 CAD software. The dimension for front axle follow the real dimension based on the data collected. In order to run the simulation in the FE ANSYS -15.0 software, the model must be meshed. By using small values of the element size, the result given will be more accurate. All the parameter and boundary condition for the front axle beam was defined in the finite element analysis before simulation. In the analysis, in which the stress is distributed, stress concentration from the load given to the axle spring pad makes the axle failure. From the analysis, the result for stress distribution for equivalent von-Mises stress, normal stress, maximum principal stress, maximum shear stress and deflection of axle beam was analysed and determined. From the several load given, the maximum load for the axle spring pads can stand was determined by using FE analysis. By analysing the front axle beam of heavy commercial vehicle we will assess the strength and capability of the product to survive against failure.

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