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DESIGN, ANALYSIS & OPTIMIZATION OF FRONT AXLE IN ELECTRIC VEHICLES

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ABSTRACT

Front axle (FA) is the most important part especially in a load carrying vehicle. The failure of front axle is a serious concern in commercial vehicles. So, it is necessary to analyse the front axle that able to withstand at severe load conditions. To this kingpin stub axle plays a major role in direction control of a vehicle. This is to analyse the front axle at different loads. Kingpin stubaxle linked with other linkages and supports vertical weight of the vehicle. This mainly deals about the improvement of strength of front axle, material selection. For finding the stresses on the beam using solid works software for static analysis and finding the loads at which axle is going to be deformed. The analysis is carried out to the vertical loads where total weight is carried out by the vehicle [7]. As using the kingpin stub axle assembly for front axle the cornering loads are more for kingpin. As the vertical loads are applied on the PAD spring which gives major support for front axle. Further the objective of analysis is to improve product quality, developing time, material and manufacturing costs and maintaining stress levels. This can be achieved by performing detailed load analysis.

KEYWORDS: FA, kingpin stub axle, FEA, commercial vehicles, static analysis.

I. INTRODUCTION

An axle is a centre shaft for a rotating wheel or gear. On wheeled vehicles, the axle may be fixed with the wheels, rotating with them or fixed to the vehicle with the wheels rotating around the axle. The 30-40% of vehicle weight is carried by the front axle. Bearings or bushings are provided at the mounting points where the axle is supported[4]. In the present analysis the front axle (FA) is designed and analysed using softwares like Solidworks, hyperworks capabilities, Ansys analysis software[7]. It is further optimised for reduction in mass. Static analysis of FA is carried out to withstand all the forces coming in the working conditions of the vehicles. Off road conditions like uneven surfaces and bumpy roads on which the vehicle has to operate. These ground irregularities leads to unexpected loads coming on to the body parts. Ackerman geometry plays a important role in turning radius. Kingpin stubaxle assembly is main load carrying member for front wheels and helps in steering of the vehicle[7]. Stubaxle takes the load coming from the front wheels and transfer it into supports. Front axle is like a simply supported beam. Under dynamic conditions, vertical bending moment is increased due to road roughness. Thus it is very difficult to find the crack propagation in short time. During the operation of the vehicle, road surface irregularity causes cyclic fluctuations of stresses on the axle, which is the main load carrying member.

II. MATERIALS AND METHODS

2.1 Front axle beam:

Front portion of the car is carried out by a beam. So, in the front axle there are two types

(i) Dead axle
(ii) Line axle

Dead axle are those which do not rotate. These axles have sufficient rigidity and strength to take the weight. The ends are suitably designed to accommodate stub axles. Line axles are used to transmit power from gear box to front wheels. Kingpin on one end is connected to stub axle and other end is connected to front axle via stud and bushes.

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Figure 2.1.1 kingpin along with stub axle

At static condition the axle may be considered as beam supported vertical upwards at the ends i.e. at the centre of the wheels and loaded vertically downwards at the centre of the spring pads[5]. The vertical bending moment thus caused is zero at point of support and rises linearly to maximum at the point of loading remains constant.

Figure 2.1.2 Live axle of the vehicle

On wheeled vehicles, the axles could be mounted to the wheels, also can maintain the position of the wheels relative to each other and to the vehicle body. Figure 2.1 below shows the front axle beam along with hub which we seen in commercial vehicles which are heavy weight. To optimize such solution to design a new shape of front axle support this reduces the weight by changing material of the axle and to increase the performance of the vehicle.

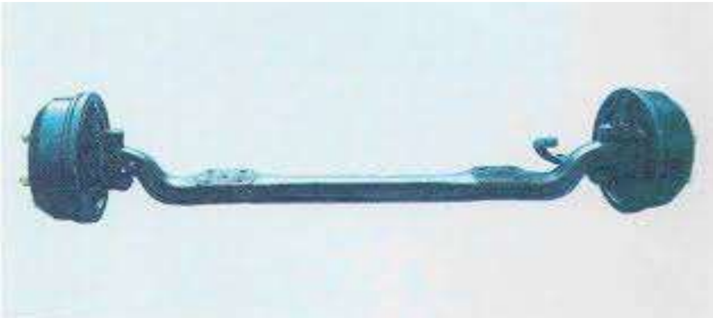

Figure 2.1.3 Front dead axle along with hub

FA is analysed to find out factor of safety for critically stressed region due to vertical load, brake load and cornering load under running conditions.

2.2 Vehicle specification:

Motor	Brushless direct current motor (BLDC) of 1.5 KW, 48 V, 32 amp
Maximum power	3000 rpm of 1.5 kw
Maximum torque	18.8 N-m
Transmission	One speed synchromesh gear box
Rear axle	Differential of two half axles
Frame	(AISI 1018) seamless mild steel material of hollow circular section pipe of 1 inch dia, 2mm thickness
Suspension	Semi- elliptical laminated of single leaf with centre bolt PAD.
Steering	Rack and pinion
Brakes	Hydraullic disc brakes for only rear wheels.
Battery	Exide C10 batteries of rating 48V, 65 Ah.
Tyres	Front – 90/90 r12 of MRF nylon Rear - 90/90 r12 of MRF nylon

Table – 1: Specifications of the vehicle[5]

2.3 Dimensions:

Wheel base	1802 mm
Overall length	2900 mm
Overall width	1450 mm
Front track width	1360 mm
Rear track width	1360 mm
Min ground clearance	200 mm
Max. Speed	45 Kmph

Table – 2: Vehicle dimensions[5]

III. EXPERIMENTATION

3.1 Methodology:

(a) Select the part which is to be improved.
(b) Create a 3-D parametric model in the software.
(c) For this we are using solid works software for stress analysis.
(d) For mesh using static analysis in the software
(e) Apply material for the part using to be tested.
(f) Apply fixed geometry for the product and apply some force and gravity to the part.
(g) The obtained FEA results are verifies with the set of experimental test procedures.

3.2 Data required from vehicle:

For analysis of front axle we are using square pipe of 2 inch diameter of 4 mm thickness of electric car.

Parametre	Value
Total weight of the vehicle (kg)	350
Laden weight on front axle (kg)	400
Laden weight on rear axle (kg)	610
Wheel base in mm	1801
KP eye distance to KP eye distance in mm	1160
PAD to PAD distance in mm	900
Spring pad width in mm	80
Spring pad length in mm	80
Velocity of vehicle (km/hr)	22

Bore diameter KBP (Di)	10
KBP outer diameter (Do)	12

Table - 3: Input data from the vehicle under analysis

3.3 Material description of FAB:

Specification – Stainless steel of grade (SAE 304)

Chemical composition - Austenite 70%
Carbon 0.15%
Chromium 16%

Density – 7.75 (g/cm3)
Poisson's ratio – 0.275
Hardness – 1700 – 2100 Mpa
Bulk modulus – 134 – 151 Gpa
Fracture toughness – 119-228 Mpa. M(1/2)
Modulus of elasticity - 193-200 Gpa
Tensile strength,yield- 215 Mpa

3.4 Modeling:

The result of the analysis of our front axle (FA) is to be done using solid works where the more load is acting on the beam and on which area the load is actually concentrating is to be known using finite element analysis(FEA) using analysis software. By using this we can easily know the weak areas on the beam.

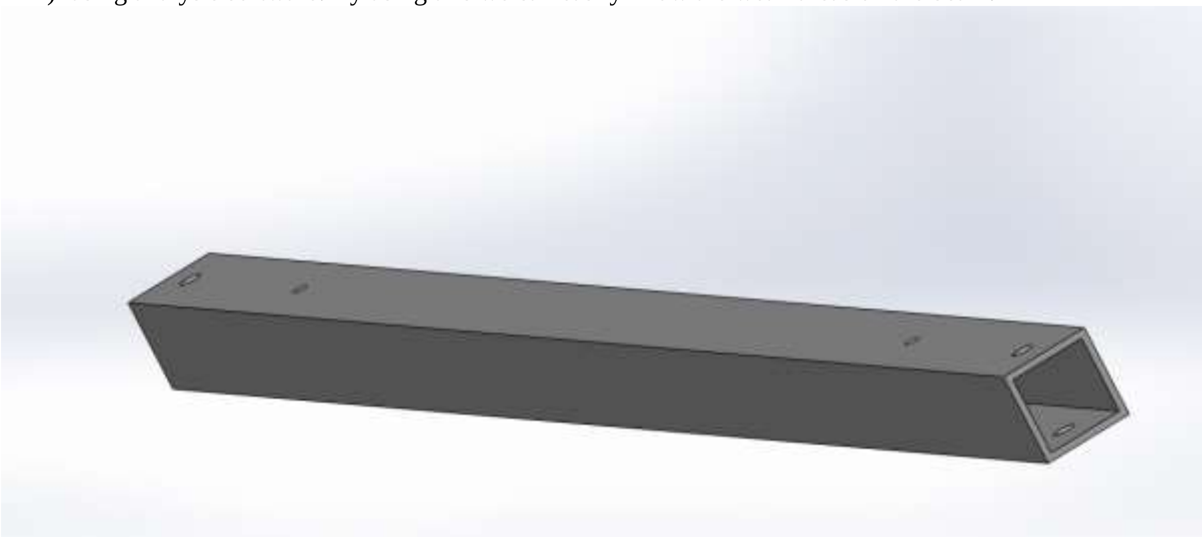

Figure 3.4 3D Modelling of Front axle

3.5 Analytical Load calculations:

Total load on FAB = 400 kg
Load on one side PAD = 400/2 = 200 kg
Taking factor for impact conditions = 2
Load to be applied on one side PAD (p) = 200 * 2
= 400 kg
= 3924 N

Kingpin area = 3.142/4*($Do^2 – Di^2$)
= 34.562 mm2

Pressure on each KBP = 3924/34.562
= 113.535 N/mm2

IV. RESULTS AND DISCUSSION

4.1 Loads and boundary conditions:

We calculate loads in terms of pressure, which directly applied on KPB surface. Similarly the PAD is fixed to the axle. After solving the problem where the more stress is applied is found.

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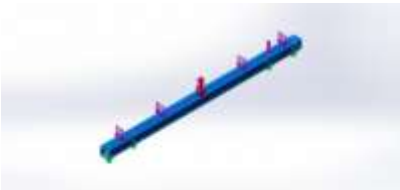
4.2 Finite element analysis of front axle:

4.2.1 Material Properties

Model Reference	Properties	Components
	**Name:** AISI 304 **Model type:** Linear Elastic Isotropic **Default failure criterion:** Max von Mises Stress **Yield strength:** 2.06807e+008 N/m^2 **Tensile strength:** 5.17017e+008 N/m^2 **Elastic modulus:** 1.9e+011 N/m^2 **Poisson's ratio:** 0.29 **Mass density:** 8000 kg/m^3 **Shear modulus:** 7.5e+010 N/m^2 **Thermal expansion coefficient:** 1.8e-005 /Kelvin	SolidBody 1(Cut-Extrude1)(front ax)
Curve Data:N/A		

4.2.2Boundaries and Fixtures

Fixture name	Fixture Image	Fixture Details
Fixed-1		**Entities:** 4 edge(s) **Type:** Fixed Geometry

Resultant Forces

Components	X	Y	Z	Resultant
Reaction force(N)	0.0956688	1571.72	0.0888062	1571.72
Reaction Moment(N·m)	0	0	0	0

4.2.3 Meshing:

Front axle designed using Solidworks 17.0 in which after the CAD modeling completed it undergoes meshing. For meshing we applied a fine standard mesh is used for the hollow axle to get the accurate deformation results. As the fine mesh is used so that it applies load on each and every corner of the axle to obtain feasible results.

Mesh Information

Mesh type	Solid Mesh
Mesher Used:	Standard mesh
Automatic Transition:	Off
Include Mesh Auto Loops:	Off
Jacobian points	4 Points
Element Size	13.5898 mm
Tolerance	0.679491 mm
Mesh Quality	High

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Mesh Information - Details

Total Nodes	17102
Total Elements	8470
Maximum Aspect Ratio	10.351
% of elements with Aspect Ratio < 3	47.3
% of elements with Aspect Ratio > 10	0.0826
% of distorted elements(Jacobian)	0
Time to complete mesh(hh;mm;ss):	00:00:11
Computer name:	GANESH-PC

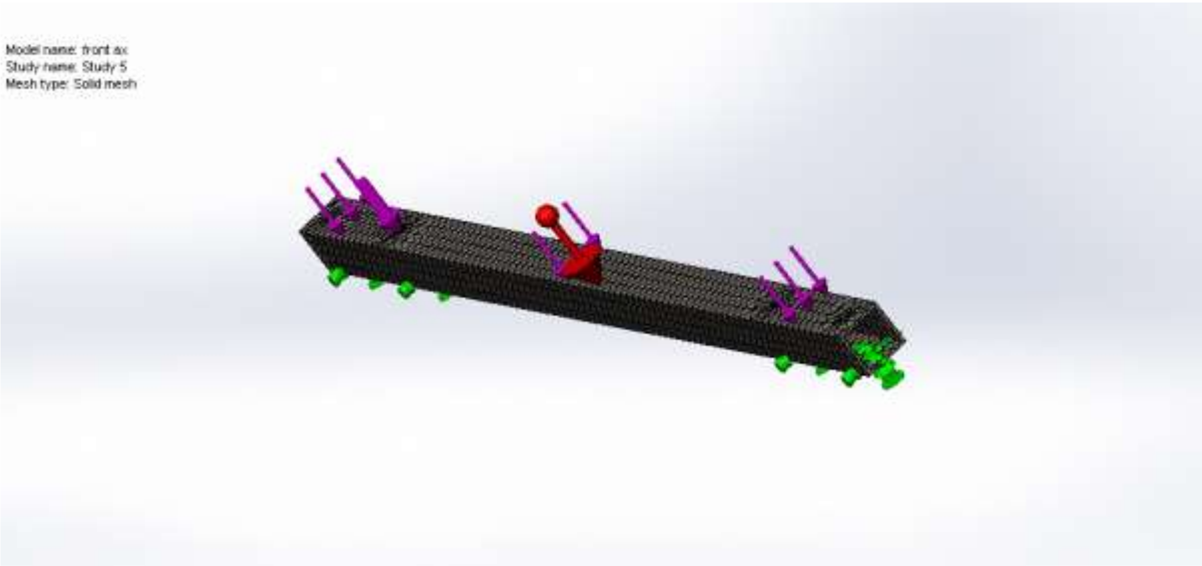

Figure – 4.2.3: Meshing of axle.

4.2.4 Resultant Forces

Reaction Forces

Selection set	Units	Sum X	Sum Y	Sum Z	Resultant
Entire Model	N	0.0956688	1571.72	0.0888062	1571.72

Reaction Moments

Selection set	Units	Sum X	Sum Y	Sum Z	Resultant
Entire Model	N·m	0	0	0	0

4.2.5 Results

Name	Type	Min	Max
Stress1	VON: von Mises Stress	56046 N/m^2 Node: 16208	8.75237e+007 N/m^2 Node: 16726

Optimization of front axle support is the best optimize design due to the lowest occurred stress and mass. By the finite element analysis method and the assistance of solidworks software, it is able to analyze stress and strain. In analytical we made maximum stress up to 215Mpa. Shape optimization were analyzed to the front axle support are according to the results. Figure 4.2.3, 4.2.4, 4.2.5 shown below are the design of front axle support, factor of safety and deflection produced after analysis.

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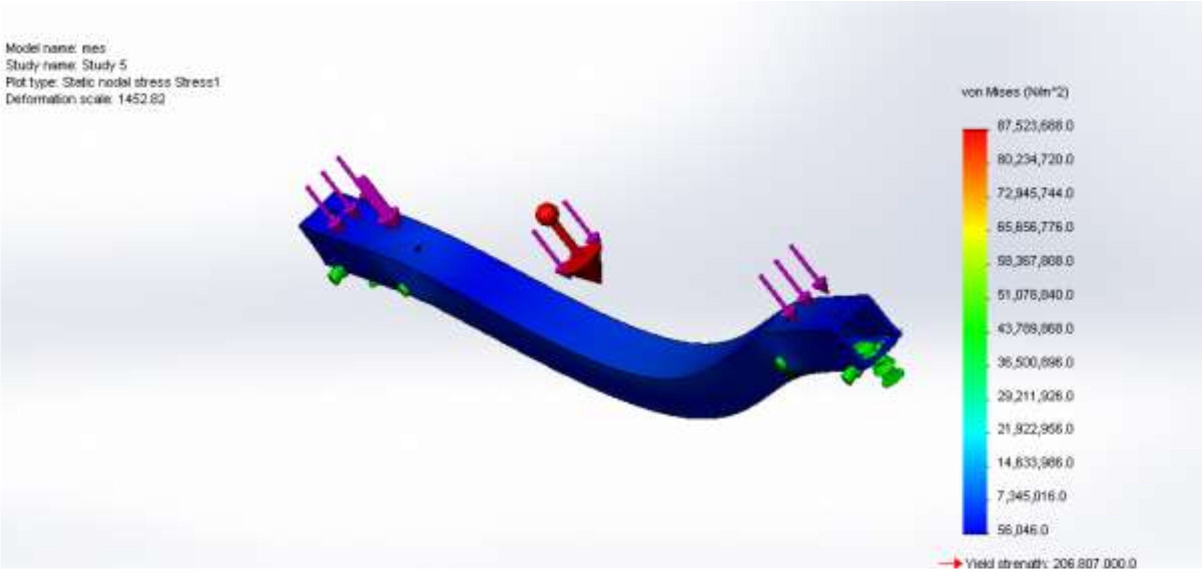

Figure – 4.2.2: Von mises stress in axle for SAE 304 material.

Figure 4.2.3 shows the chosen material SAE 304 shows the yield strength of 206807.000 in the analysis scale. As the gravity 9.81 m/s2 is also applied on the centre of the front axle.

Name	Type	Min	Max
Displacement1	URES: Resultant Displacement	0 mm Node: 15	0.0846632 mm Node: 4442

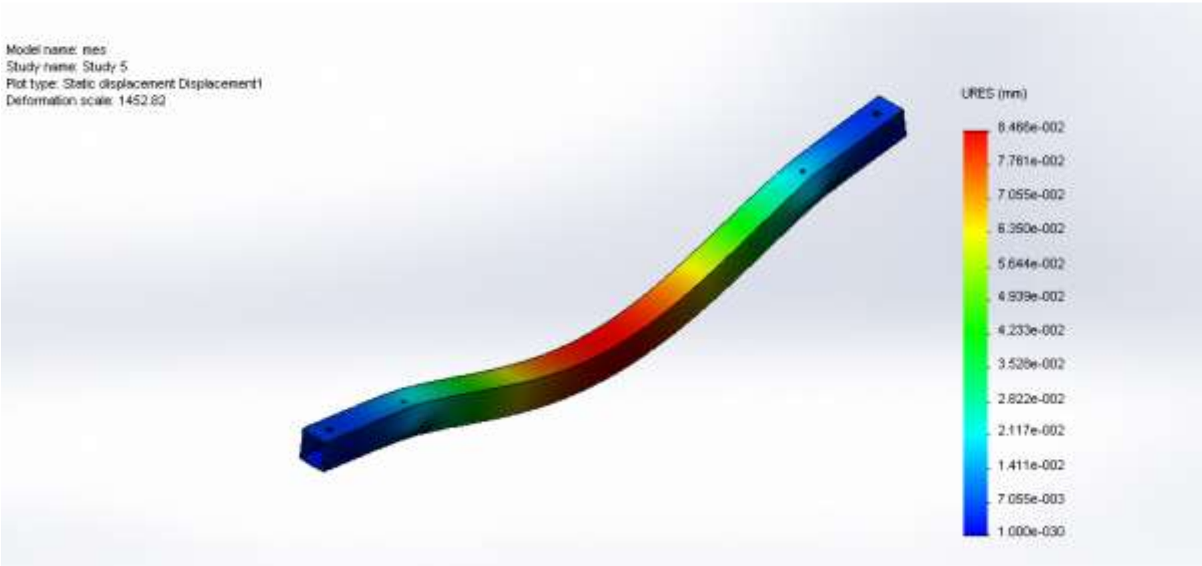

Figure – 4.2.4: Deflection of axle for SAE 304 material.

The above schematic shows the different levels of deformation under the load applied on the front axle support which gives the optimized solution under different stress and strain. As per analysis scale shown in the figure 4.2.4.

Name	Type	Min	Max
Strain1	ESTRN: Equivalent Strain	2.0367e-007 Element: 3276	0.000216966 Element: 4759

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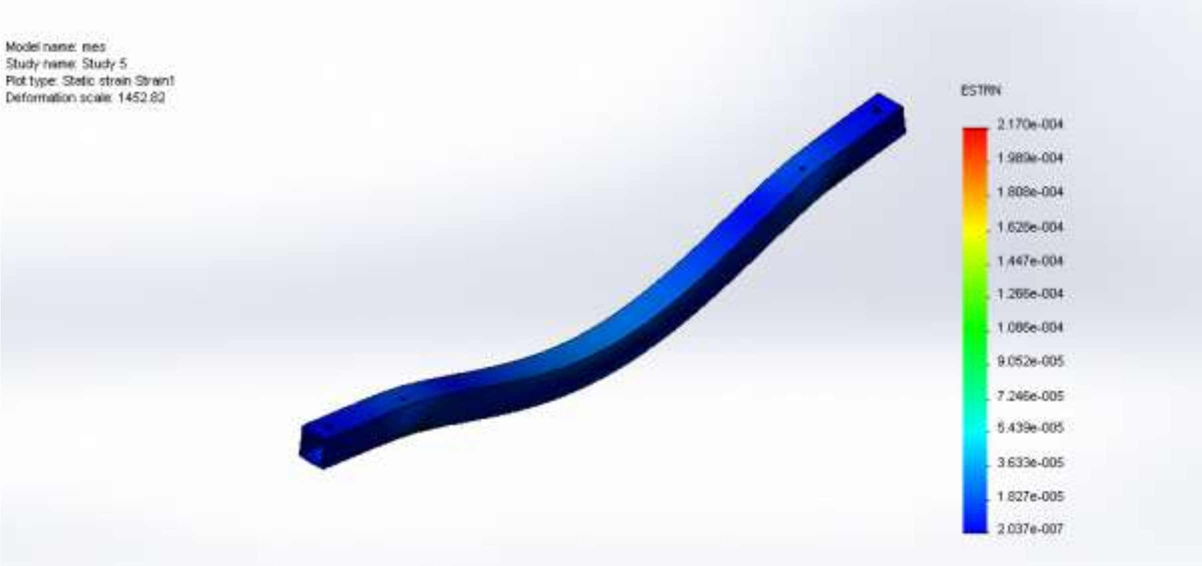
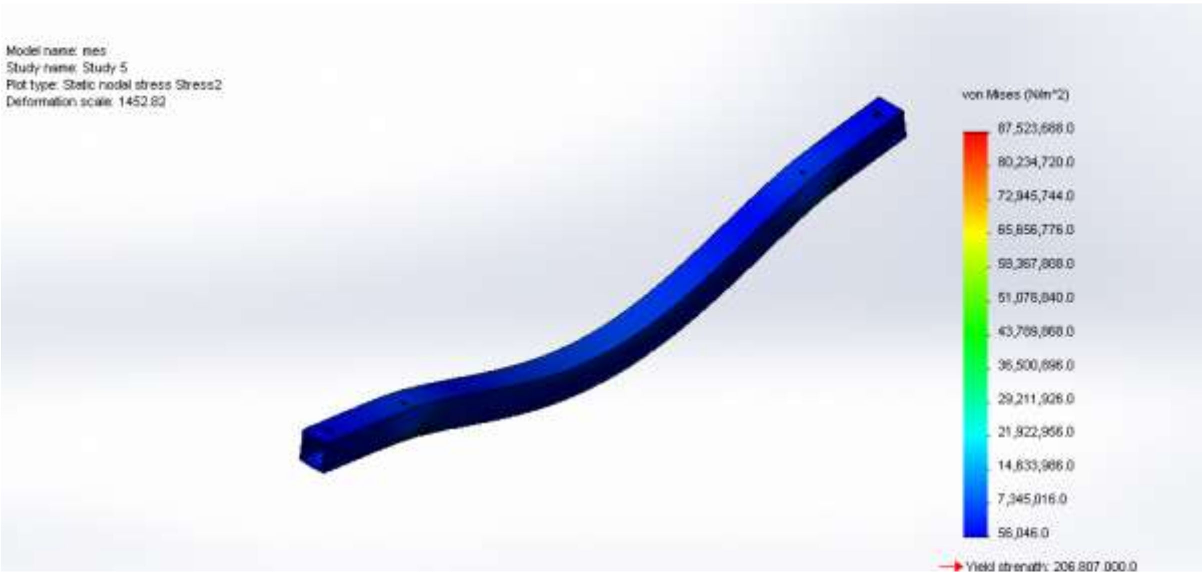


Name	Type	Min	Max
Stress2	VON: von Mises Stress	56046 N/m^2 Node: 16208	8.75237e+007 N/m^2 Node: 16726



Name	Type	Min	Max
Strain2	ESTRN: Equivalent Strain	2.53682e-007 Node: 16208	0.00039616 Node: 16726

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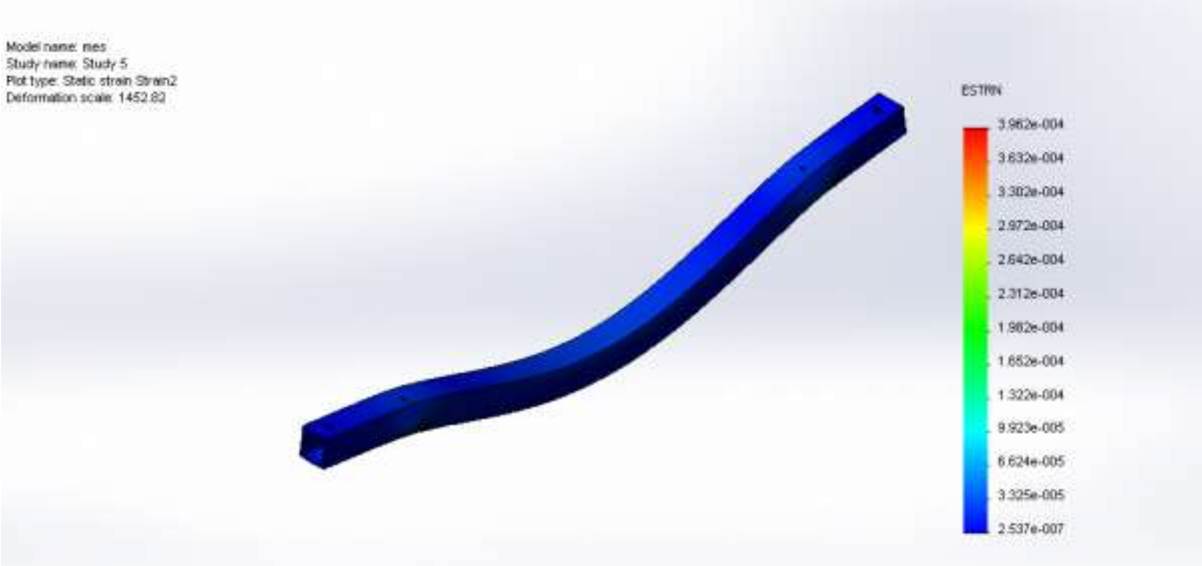


4.2.6 Factor of safety:

From this analysis at this stress the obtained factor of safety is Min FOS = 2.4. From this FEA (Finite element analysis) we obtain a better result which is preferable for fabrication of the front axle.

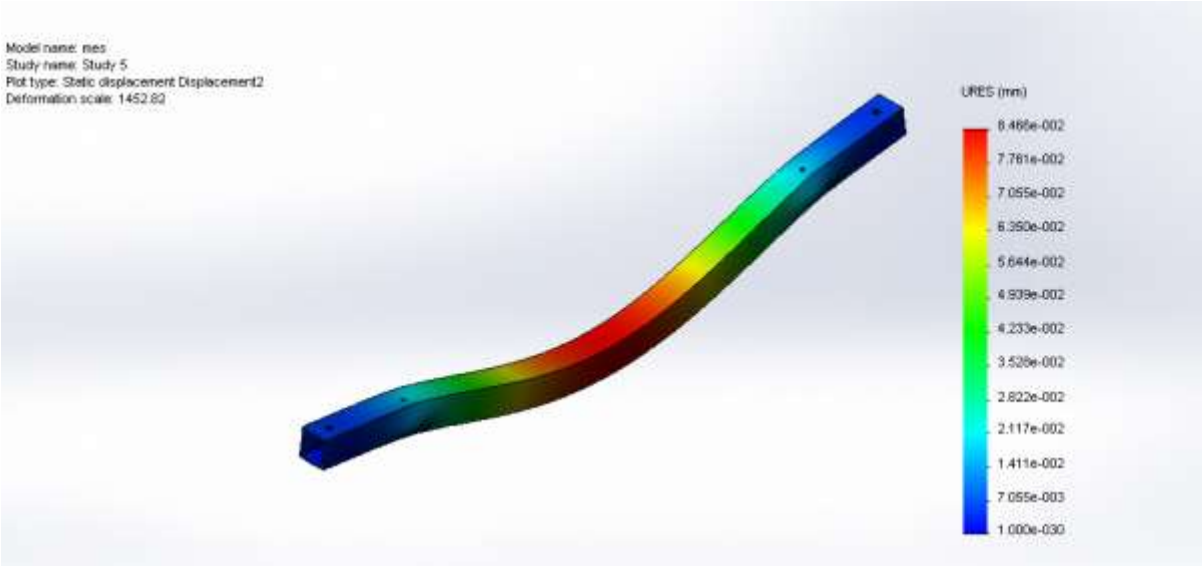
Name	Type	Min	Max
Factor of Safety1	Automatic	2.36287 Node: 16726	3689.95 Node: 16208

Figure - 4.2.6: Factor of safety of axle for SAE 304 material.

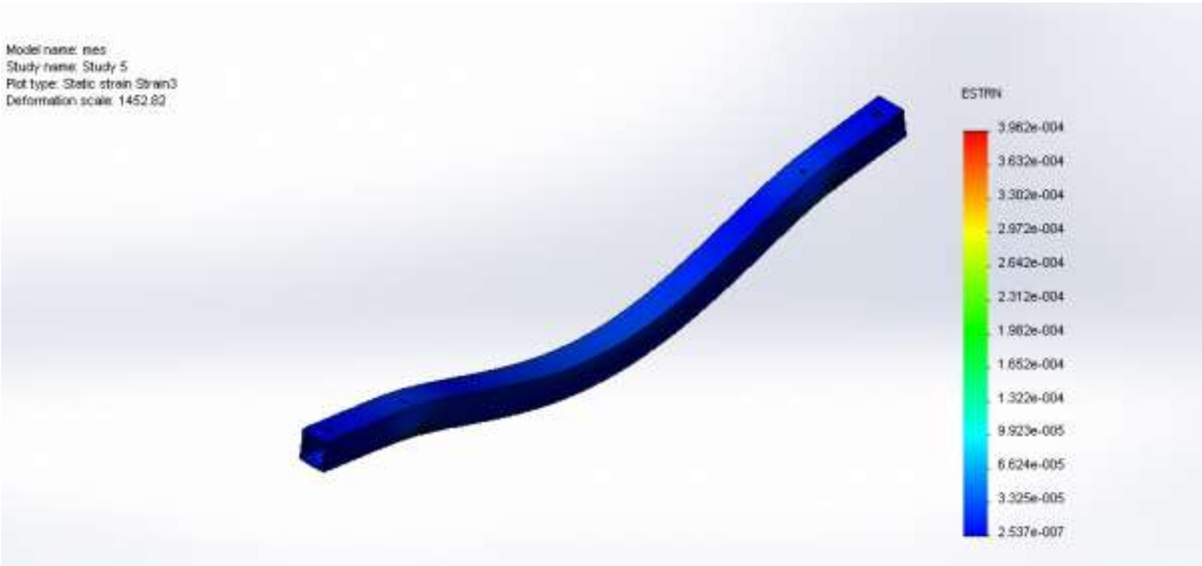
Name	Type	Min	Max
Displacement2	URES: Resultant Displacement	0 mm Node: 15	0.0846632 mm Node: 4442

4.3 Optimization approach:

The objective of optimization technique is to minimize the mass of the front axle support and reduce the cost of production. The front axle support is subjected to tensile load which gives the factor of safety up to 2.4. The implementation of these optimizations is to find out the best design and shape of the front axle support to improve the performance and strength especially in critical positions.



Name	Type	Min	Max
Strain3	ESTRN: Equivalent Strain	2.53682e-007 Node: 16208	0.00039616 Node: 16726



4.4 Comparision of results:

Comparision between analytical and FEA results.

S.no	Material	Parameter	Analytical Result	FEA result	% of error
1.	SAE 304	stress	215	1452	0.14

Table – 4 Results comparison

RESULTS: Hence correct design of front axle (FA) is very critical. The approach in the report has been divided into two steps. In first step analytical method used for design of front axle. In second step is to analyse the front axle (FA) using solidworks software. The finite element analysis (FEA) result is to be compared with analytical method.

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V. CONCLUSION

From above results it is clearly shown, AISI 304 material is suitable for our vehicle. Also in the present we have satisfactory co-relation between hand calculations done analytically FEA results. The deflection in FEA gave confidence that the boundary condition for the axle is correctly simulated. Meshing and modelling used for the component is well defined.

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