Anti-Dive and Anti-Squat Suspension Geometry

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Abstract - *This paper presents a review of recent research that has been carried out on the Suspension system of a Formula Student car.* In any formula student race car, racing cars, baja cars, or any given car suspension plays an important role. Suspension is primarily used to prevent the driver from sudden road shocks. It's also used to control the longitudinal load transfer, lateral load transfer and giving the exact amount of feedback needed by the driver. While designing a suspension system few factors need to be considered such as C.G, Roll-centre, camber, toe-in, toe-out. Handling characteristics is also improved by optimizing suspension system. The modeling of suspension geometry can be carried out in Lotus shark software to carry out validation purpose and to check whether the desired suspension geometry is achieved or not.

Key Words: Suspension, Lateral load transfer, C.G, Roll-centre, Handling

1. INTRODUCTION

Formula SAE is a student design competition organized by SAE International (also known as Society of Automotive Engineer, SAE). This competition challenges students to conceive, design, fabricate the cars and compete with other student formula teams. The competition has dynamic events like brake test, skidpad, endurance. The suspension is arguably the most important part of a vehicle, this is especially true for a racecar. Without a good suspension design the performance of a racecar will suffer dramatically. The goal this time while designing our car is to have **Anti-dive** and **Ani-squat** geometry. This anti-geometry is referred to in the form of geometry at the front and the geometry at the rear of the car. The basic fundamental of this geometry is to reduce the pitching moment of the car by altering the loading conditions. When the car is in steady state acceleration there is longitudinal weight transfer at the rear (squat moment) and when the car is decelerating or braking there is longitudinal weight transfer at the front (dive moment).

In this geometry all the forces occurring due to braking and deceleration acts through the centre of gravity i.e., the rotation of the car will be around COG. It affects the suspension deflection when the brakes are applied, therefore causing the lowering of the front end of the car.

2. PROBLEM STATMENT

In Formula SAE competition, along-with performance design itself also plays a huge role. The concept of Anti-dive and Anti-squat is to enhance the performance of the car by altering the forces acting on the car during the competition. The design itself doesn't created a huge difference, when comparing the weight of the system to the simple suspension system, so this will be important in aspect of judging and scoring. Suspension system is considered as the most crucial part in vehicle, optimizing the suspension to alter the loading forces as required will benefit in overall improved performance of the race car which will eventually lead to a competition winning car. The main purpose of this design is to manufacture a car which will be better in handling and stability which will certainly improve the performance of the car, as well as this design process will help future team members to reiterate and optimized the design.

- Anti-dive & anti squat geometry is used to control the amount of load going through the springs and the pitch attitude of the car.
- Anti-geometry is an understanding of how a car moves and rotates under acceleration or braking conditions must be achieved.
- All forces acting upon a car act through the center of gravity.
- The center of gravity is therefore the center of rotation for any acceleration or braking inputs.

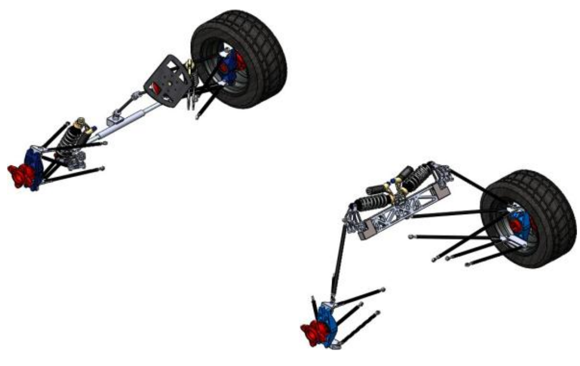


FIG: CAD OF SUSPENSION SYSTEM

3. Design Objectives

The points we keep in mind while designing the suspension system are:

1. Cost efficient
2. Double Wishbone Adjustable Anti-dive and Anti-squat suspension geometry
3. Adjustable camber angle with Camber plates
4. Less weight
5. High strength
6. Adjustable stiffness Anti-Roll bar
7. Pull and Push suspension

and specifically designed to meet the operating conditions of FSAE Race Car.

4. SELECTION OF DIFFERENT PARTS

4.1. TIRE SELECTION

Tire selection plays a most important role in vehicle. After selecting tire, bearing selection, hub design, suspension points, upright design are made. What should be kept in mind while selecting tire? 1. Contact patch should be optimum, so to get more traction 2. Diameter of the rim 3. PCD (Pitch Circle Diameter) 4. Width of the rim 5. Tire life.

We are using HOOSIER R – 10 tires. R – 10 (rim has diameter of 10'') is used instead of R – 13 because,

1. Due to compact structure, the force transmission from the center can easily transmitted to the A-Arms.
2. It keeps the roll center and center of gravity close to the ground as compared to R-13
3. Load on the engine will be less as the diameter is small
4. Vehicle is close to the ground
5. Weight of R-10 is less as compared to R-13
6. Weight of every component will be less
7. Fuel efficiency will be more as compared to R-13

4.2. SUSPENSION POINT SELECTION

The first point to find in suspension is Lower Ball joint (LBJ). Let consider X-axis w.r.t solid works first there is hub, then rotor and caliper by considering all their thickness & bearing thickness and placing the point as close to the centre as possible, as all forces pass through the centre. We fixed the X-axis coordinate of LBJ. Now let us consider Y-axis w.r.t solid works first we take inner diameter of rim, outer diameter of bearing, diameter of rotor and then very small distance. Placing the LBJ close to the centre we fixed the point of LBJ on Y-axis.

Finally considering Z-axis w.r.t solid works we are placing the Z-axis point at the center line of the tire as all forces passes through the center. We fixed all the 3 points on different axis. This is done by making the design of hub, taking information and CAD of rotor and caliper and selecting bearing.

Similarly, we have to find UBJ. For finding the points on chassis, we have to first need the line diagram of front hoop, after that we have assumed the lower A-Arm to be parallel to the ground and for upper A-Arm we have done iteration on the angle views from the front view. The lower A-Arm is kept parallel because: -

1. To ease the design
2. To bring C.G down
3. To keep Instantaneous Centre (I.C) moderate
4. To lower the vehicle and bring it close to ground
5. To bring Roll center down

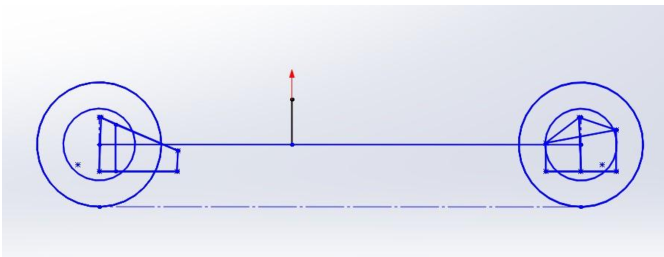


FIG: SIDE VIEW

By this view, we can see that the rear A-Arms has more angle made with the horizontal than the front. This angle is decided by using Anti-dive and Anti-squat suspension geometry. This angle helps to control the pitch movement. With this angle the pitch center is coming out to be close to the center of gravity

4.3 BEARING SELECTION

By selecting the bearing, we are getting the outer diameter of the hub and inner diameter of the upright.

FOR FRONT

Radial load only:

Fe = V*R

V = 1 inner ring

N = 100*10^6

Reliability = 90%

Kr = 1 & Ka = 1

Rx = 2613.741 + 27.74= 2641.481N

Ry = 1136.409 N

R= $(Rx^2 + Ry^2)^{1/2}$ = $(2641.481^2 + 1136.409^2)^{1/2}$ = 2875.560343 N

Fe = V x R = 1 x 2875.560343 = 2875.560 N
L = 100

	a	Creq = $Fe(L)^{1/a}$ in N
Ball Bearing	3	13347.16878
Roller Bearing	10/3	11447.81192

From above values,
We selected SKF 7007 angular contact ball bearing with
OD = 62 mm, ID = 35 mm and width =14mm.

For Rear

Radial load only

Fe = V*R

V = 1 inner ring

N = 100*10^6

Reliability = 90%

Kr = 1 & Ka = 1

Rx = 3263.09 – 21.281 = 3241.728 N

Ry = 1418.7 N

R = $(Rx^2 + Ry^2)^{1/2}$ = $(3241.728^2 + 1418.7^2)^{1/2}$ = 3538.57458 N

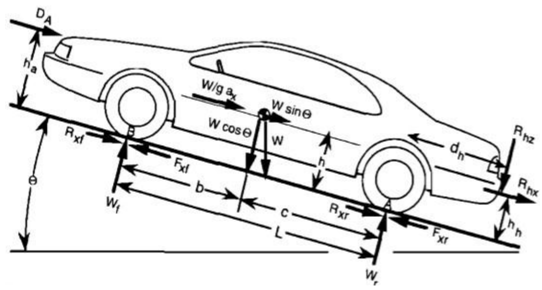
Fe = V x R = 1 x 3538.5745 = 3538.5745 N

L = 100

	a	$C_{req} = Fe(L)^{1/a}$ in N
Ball Bearing	3	1664.355804
Roller Bearing	10/3	1427.51147

From above values
We selected SKF 71910 angular contact ball bearing with OD = 72 mm, ID = 50 mm and width = 12mm

5. FORCES ACTING ON TIRES



Wheelbase (L)	65”	1651 mm
Trackwidth (t)	47”	1193.8 mm
CG height	13.8”	350 mm
Total weight (m)	595.248 lb	270 kg
% front weight	40	40
% rear weight	60	60
B	39”	990.6 mm
C	26”	660.4 mm
Wheel radius (r)	3.1496”	80 mm
μ	2.3	2.3
Longitudinal	2648.7 N	2648.7 N
Lateral	4820.634 N	4820.634 N
Braking	16.677 N	16.677 N
Roll angle	3^0	3^0
Front weight (mf)	238.099 lb	108 kg
Rear weight (mr)	357.1489 lb	162 kg
Grade angle	0.04	0.04
Rolling coefficient (fr)	0.015	0.015
Longitudinal acceleration	9.81 m/s^2	9.81 m/s^2
Lateral acceleration	17.8542 m/s^2	17.8542 m/s^2

All the forces calculated are the maximum forces on the tires at different condition.

Static

$$Fz = \frac{W \times b}{L}$$

Front right	N
Fx	0
Fy	0
Fz	-541.53

Front left	N
Fx	0
Fy	0
Fz	-541.53

Rear right	N
Fx	0
Fy	0
Fz	-806.39

Rear left	N
Fx	0
Fy	0
Fz	-806.39

Linear acceleration and tractive force on rear wheels only with no lateral force.

$$Fx = \frac{(W \times b \times \mu)}{(L - (h \times \mu))}$$

$Fy = 0$

$$Fz = Wfs \pm \frac{W \times a_x \times h}{L}$$

Front left	N
Fx	0
Fy	0
Fz	-246.7555

Front right	N
Fx	0
Fy	0
Fz	-246.7555

Rear left	N
Fx	-2496.215
Fy	0
Fz	-836.30

Rear right	N
Fx	-2496.215
Fy	0
Fz	-836.30

Steady State Cornering

$M_{\phi f} = \underline{W \times h1 \times a_y} + Wf \times h_f \times a_y =$

$\underline{(270 \times 9.81 \times 0.27785 \times 1.82)}$ + 252.6976 ($K_{\phi f} + K_{\phi r}$) – (W x h1) [41329.93306 – (270 x 9.81 x 0.22785)] = **252.696 Nm**

$M_{\phi r} = \underline{W \times h1 \times a_y} + Wr \times h_r \times a_y =$

$\underline{(270 \times 9.81 \times 0.27785 \times 1.82)}$ + 379.04645 ($K_{\phi f} + K_{\phi r}$) – (W x h1) [41329.93306 – (270 x 9.81 x 0.22785)] = **379.04645 Nm**

Corning while accelerating

Fx	N
Fxfl	1054.389
Fxfr	80.684
Fxrl	3263.009
Fxrr	1802.348

Fy	N
Fyfl	834.3426
Fyfr	63.8456
Fyrl	2582.034
Fyrr	1426.205

Fz	N
Fzfl	458.43
Fzfr	35.08
Fzrl	1418.7
Fzrr	783.6295

Rolling Resistance

While accelerating:

Fx	N
Fxf	7.40
Fxr	33.0349

Corning while accelerating:

Fx	N
Fxfl	6.876
Fxfr	0.526
Fxrl	21.281
Fxrr	11.75

6. MATERIAL SELECTION

The material we are using for hub, upright, clevis, bell crank is selected as Aluminium 7075 T6.

The material of the A-Arms, pull rod in front, push rod in rear, tie rod, third link are selected by using matrix calculation.

Then we choose AISI 1018 material of hollow tubes with OD = 14 mm and ID = 10 mm of thickness.

Sr no.	Mild steel AISI 1018	Aluminium 7075 T6
1	Heavy material	Light material
2	High strength	Comparatively low strength for same dimension
3	Easily available	Available in big market
4	Low cost	Very high cost
5	Machining cost is less	Machining cost is high

7.1 ANTI-ROLL BAR (ARB)

In our vehicle we are using U type anti-roll bar. It consists of a tube and its mounts. we are using the ARB in front.

ARB stiffness front

$K\phi br = K\phi - K\phi f - K\phi br = 314.575 - 83.58558 - 120.48 = 110.50942$ lb-ft/deg

Rear roll stiffness

$K\phi r = \frac{K\phi f - K\phi sf}{MR^2} = \frac{83.58558 - 69.0765}{0.3} = 161.212$ lb-ft/deg

ARB stiffness rear

$K\phi br = 14.50$ lb-ft/deg

$\phi = \underline{-W \times hs \times ay} = \underline{-(79.366 + 158.18) \times 15.935 \times 0.046228} = 0.049^{c} = 2.8075^{0} = 3^{0}$

$K\phi f + K\phi r - (Ws \times hs)(83.58 + 161.212) - (79.366 + 158.18 \times 15.9)$

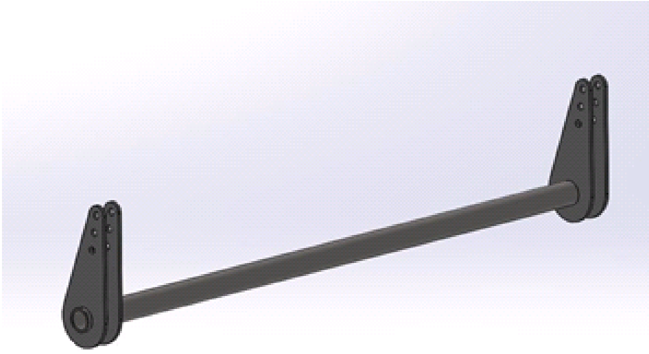


FIG: - ANTI-ROLL BAR (ARB)

7. SHOCKS SELECTION

The shocks selection is done on the basis of the design requirement and analysis and on the performance, cost and on the market availability.

Sr No.	Local Shocks	Fox DHX RC 4	DNM Burner RCP 2
1	Low build quality	High built quality	High build quality
2	Leakage issues	No leakage issues	No leakage issues
3	No compression and rebound adjustments	Compression and rebound adjustments	Compression and rebound adjustments
4	Easily available at local dealers	Needs to import	Needs to import
5	Low cost	High cost	Average cost
6	less durable	More durable	More durable

By comparing all the parameters and also by using calculation DNM Burner RCP 2 was finalized. Calculation is given below.

FIG: DNM BURNER RCP 2

7.1 RIDE RATE & ROLL RATE

RIDE RATE
The sprung mass resting on the suspension and tire springs is capable of motion in the vertical direction. The effective stiffness of the suspension and tire springs in series is called the "ride rate" determined as follows:

$$RR = \frac{Ks\ Kt}{Ks + Kt}$$

where:

RR = Ride rate

Ks = Suspension stiffness

Kt = Tire stiffness

The ride rate for our vehicle is fined using the following calculation:

Unsprung mass

Component	Front (kg)	Rear (kg)
Upright	0.45 x 2	0.5 x 2
Wheel assembly	5.55 x 2	5.55 x 2
A-arm	0.3 x 2	0.3 x 2
Disc	0.5 x 2	0.5 x 2
Hub	0.45 x 2	0.5 x 2
Upright bearing	0.25 x 2	0.3 x 2
Clevis	0.2 x 2	0.3 x 2
Caliper	0.9 x 2	0.8 x 2
A-arm	0.25 x 2	0.25 x 2
Total	180	185

Front

Total Front unsprung mass = 36.5 kg

Front weight =108 kg

Front unsprung mass =36 kg

Front sprung weight/ mass =72 kg

Front corner sprung mass (Wsf)=36 kg

Wsf = 36 kg = 79.366 lb

By performing different iteration, we finalized the following result:
$w_n = 2.3$ Hz

$$\text{Front ride rate, } Krf = \frac{4\pi^2 w_n^2 Wsf}{386.4} = 42.896 \text{ lb/in}$$

Kt =1198 lb/in

$$\text{Wheel centre rate, } Kwf = \frac{Krf \times Kt}{Kt - Krf} = 44.488 \text{ lb/in}$$

Damping ratio

Range – 0.2, 0.3 & 0.4

0.2

$$Ksf = \frac{Kwf}{LR^2} = \frac{44.488}{0.2^2} = 1112.2 \text{ lb/in}$$

0.3

$$Ksf = \frac{Kwf}{LR^2} = \frac{44.488}{0.3^2} = 494.3 \text{ lb/in}$$

0.4

$$Ksf = \frac{Kwf}{LR_f^2} = \frac{44.488}{0.4^2} = 278.05 \text{ lb/in}$$

Rear

Rear weight = 162 kg

Rear unsprung mass = 18.5 kg

Rear sprung mass = 143.5 kg

Rear corner sprung mass (Wsr) = 71.75 kg = 158.18 lb

By performing different iteration, we finalized the following result:
$w_n = 2.15$ Hz

$$\text{Rear ride rate, } Krr = \frac{4\pi^2 w_n^2 Wsr}{386.4} = 47.846 \text{ lb/in}$$

Kt =1198 lb/in

$$\text{Wheel centre rate, } Kwr = \frac{Krr \times Kt}{Kt - Krr} = 49.8365 \text{ lb/in}$$

Damping ratio

Range – 0.2, 0.3 & 0.4

0.2

$$K_{sr} = \frac{K_{wr}}{LR^2} = \frac{49.8365}{0.2^2} = 1245.913 \text{ lb/in}$$

0.3

$$K_{sr} = \frac{K_{wr}}{LR^2} = \frac{49.8365}{0.3^2} = 553.739 \text{ lb/in}$$

0.4

$$K_{sr} = \frac{K_{wr}}{LR^2} = \frac{49.8365}{0.4^2} = 311.4782 \text{ lb/in}$$

For above all cases and calculations we decided to choose DNM Burner RCP 2 for both front and rear with a Stiffness which has spring rate, Ks = 550 lb/in by giving,

	Front	rear
Natural undamped frequency (w_n) in Hz	2.3	2.15
Left ride rate (Kr) in lb/in	42.896	75
Left tire spring rate (Kt) in lb/in	1198	1198
Left wheel centre rate (Kw) in lb/in	44.488	80
Damping ratio (LR)	0.3	0.4
Left spring rate (Ks) in lb/in	494.3	497.42
Total ride rate in lb/in	85.792	150

8. ASSEMBLY OF SUSPENSION SYSTEM

The parts are:

1. Upright
2. Hub
3. Wheel / Rim
4. Clevis
5. Shim
6. Bearings
7. A-Arm end
8. A-Arm tube
9. Nut
10. Rod end
11. Billet
12. Shock Absorber
13. Rear Upper Bulkhead
14. Bell crank / Rocker
15. Rear Upper Bulkhead mounts

The parts we are using with Aluminium 7075 T6 are:

1. Hub
2. Upright
3. Bell crank
4. Clevis

The parts we are using with Mild Steel AISI 1018 are:

1. A-Arm end
2. A-Arm tube
3. ARB mounts
4. ARB tube
5. Billet

The parts we are using with Stainless Steel 316 is Shims

9. ANALYSIS OF SUSPENSION COMPONENTS

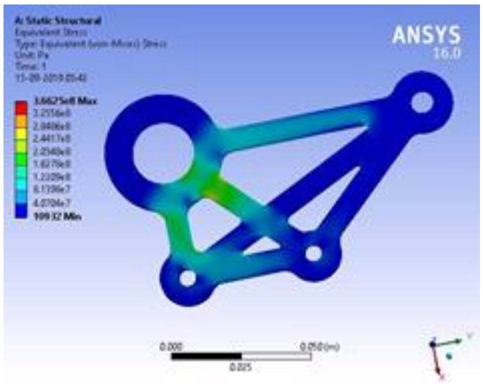


FIG: STRESS ANALYSIS OF BELL CRANK

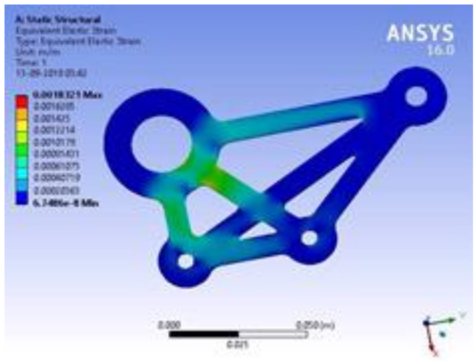


FIG: STRAIN ANALYSIS OF BELL CRANK

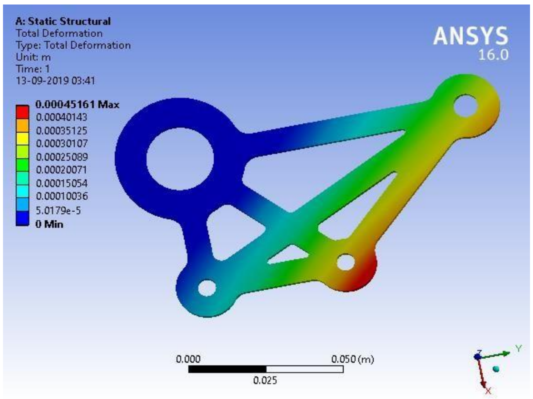


FIG: DISPLACEMENT OF FRONT BELL CRANK

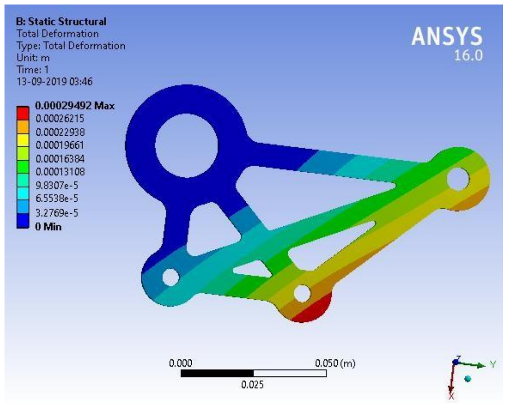


FIG: DISPLACEMENT OF REAR BELL CRANK

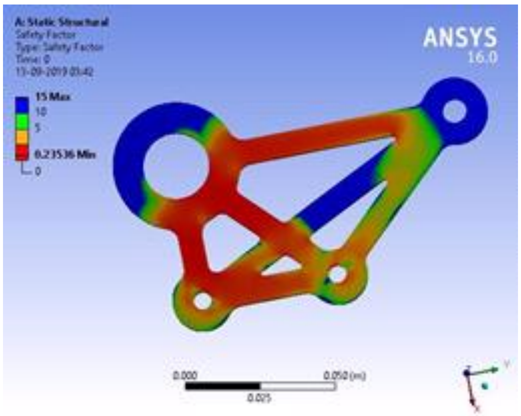
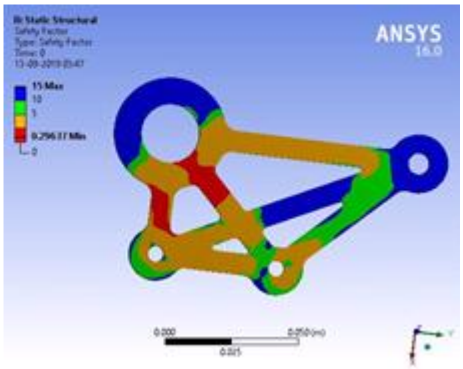


FIG: FOS OF FRONT BELL CRANK



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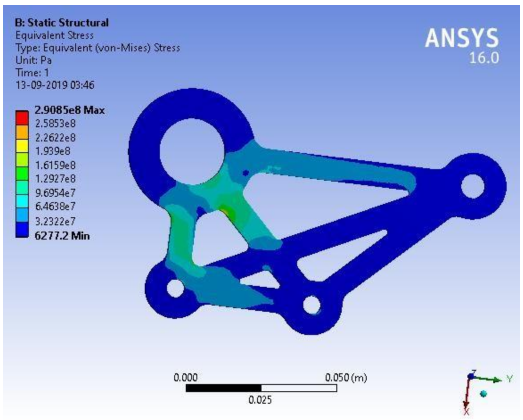

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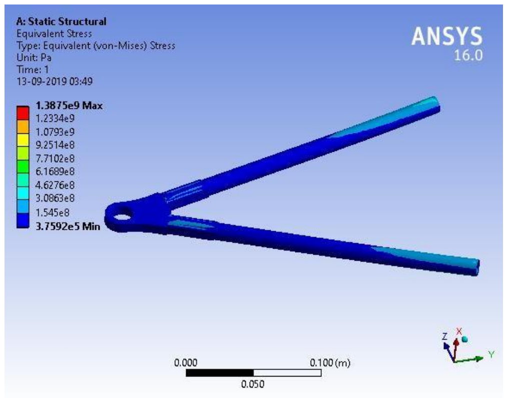
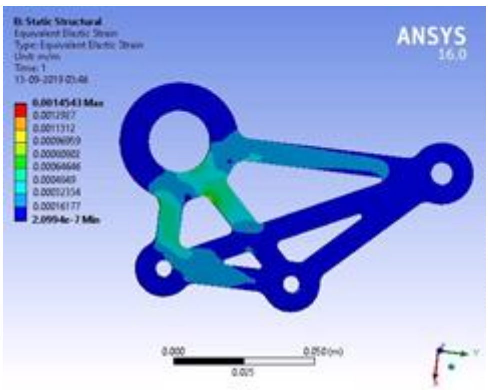


FIG: STRESS ANALYSIS OF FRONT A-ARM



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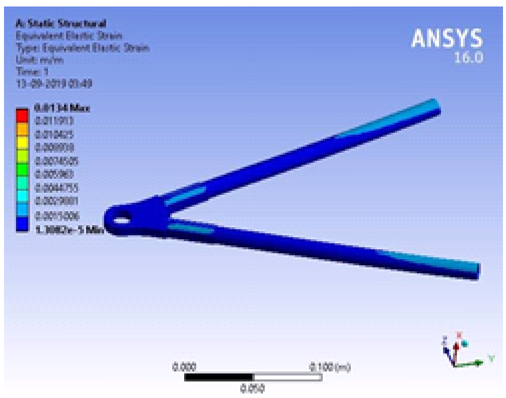

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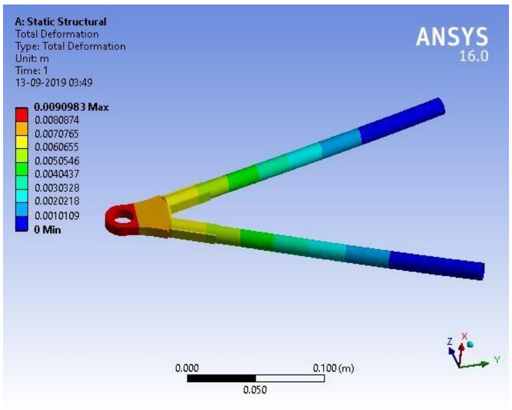


FIG: DISPLACEMENTOF FRONT A-ARM

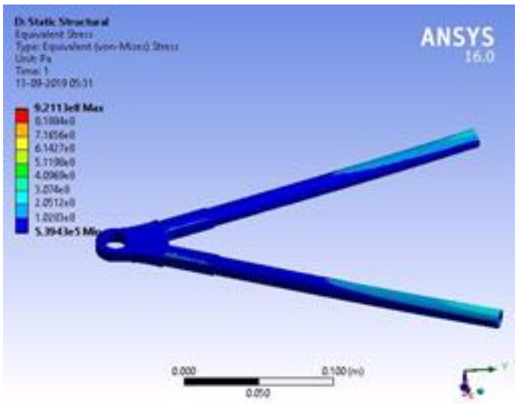
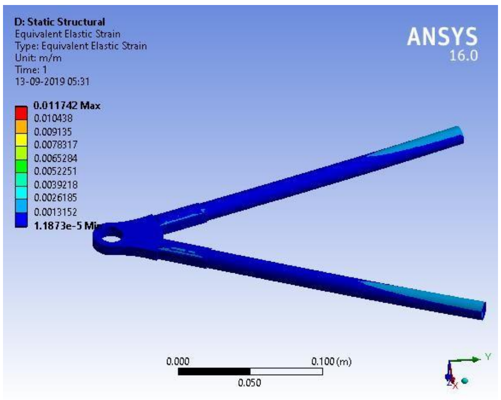


FIG: STRESS ANALYSIS OF REAR A-ARM



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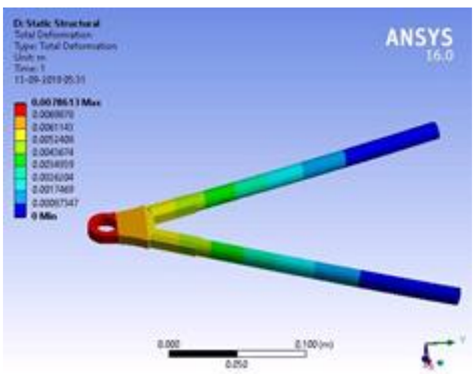

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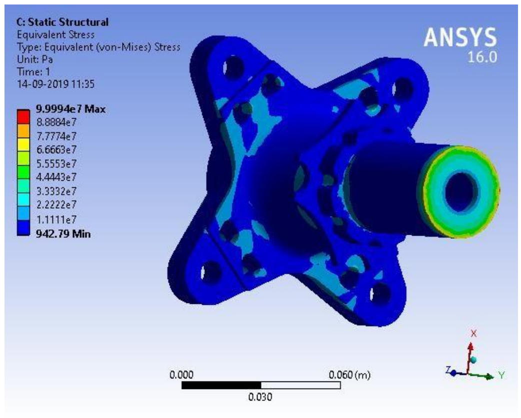


FIG: STRESS ANALYSIS OF FRONT HUB

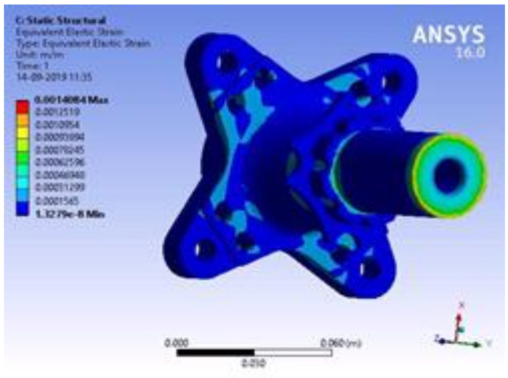
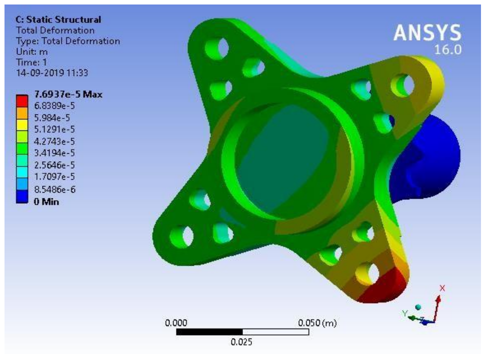


FIG: STRAIN ANALYSIS OF FRONT HUB



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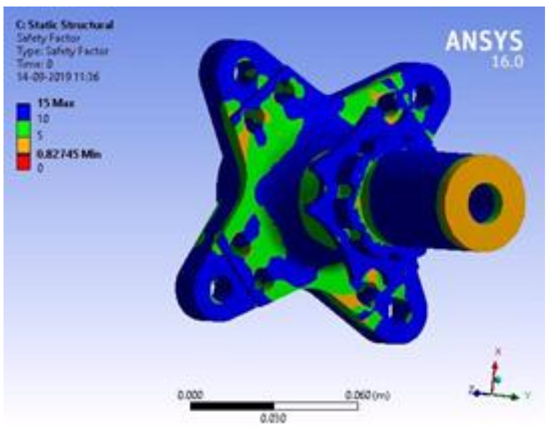


FIG: FOS OF FRONT HUB

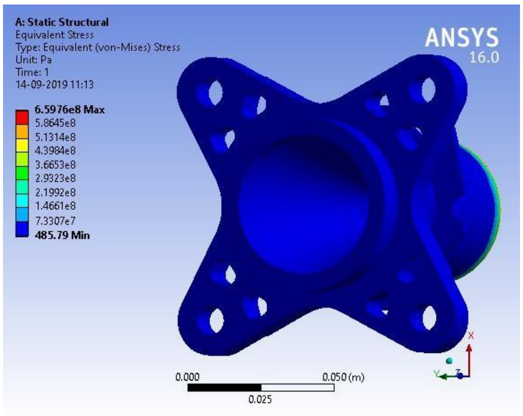
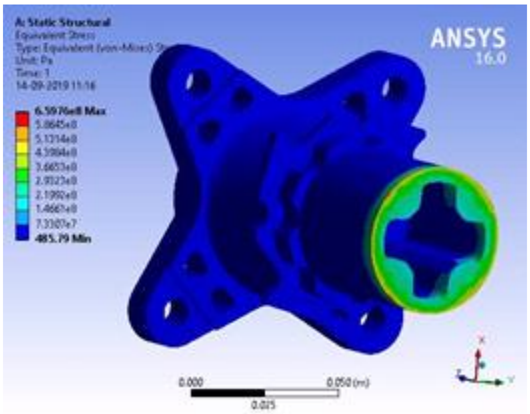


FIG: STRESS ANALYSIS OF REAR HUB



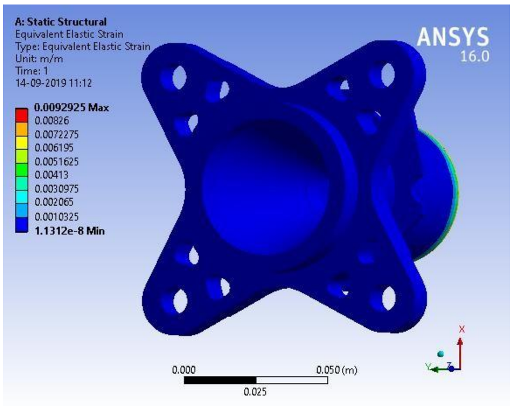
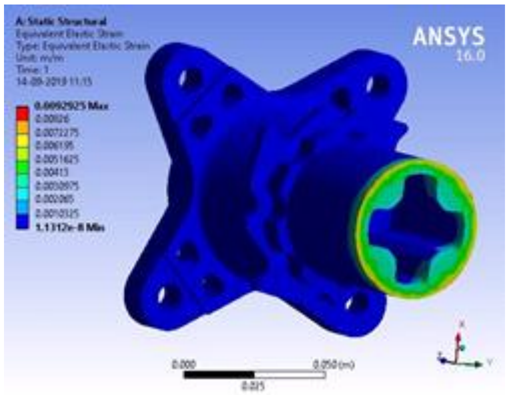


FIG: STRAIN ANALYSIS OF REAR HUB



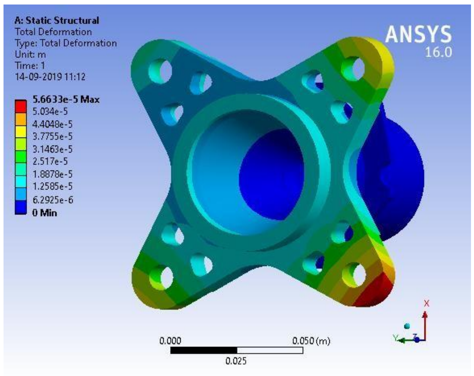
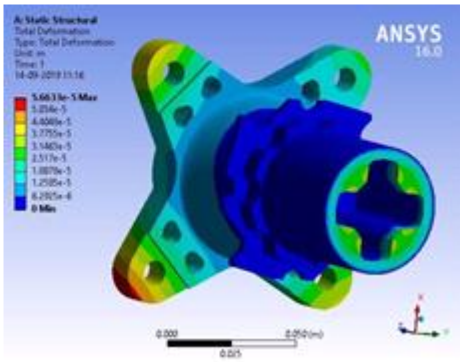


FIG: DISPLACEMENT OF REAR HUB



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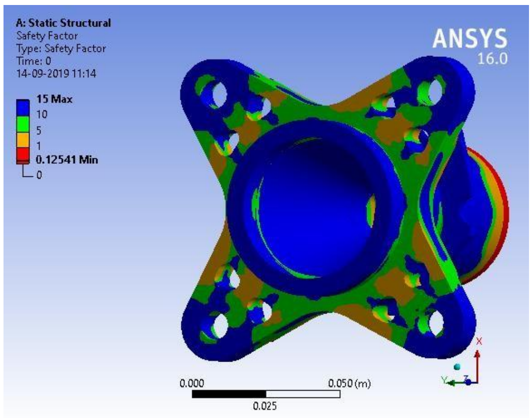
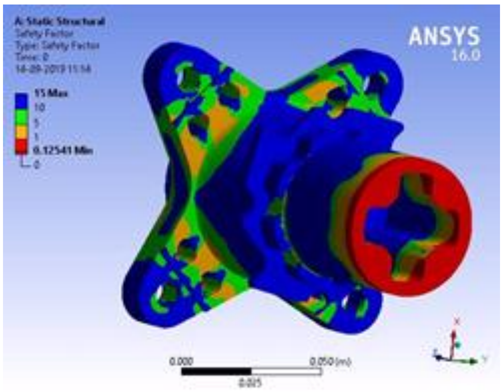


FIG: FOS OF REAR HUB



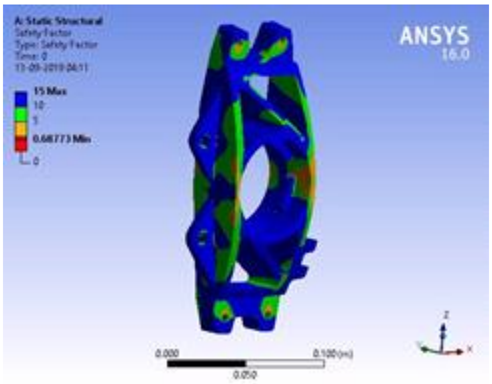


FIG: FOS OF FRONT UPRIGHT

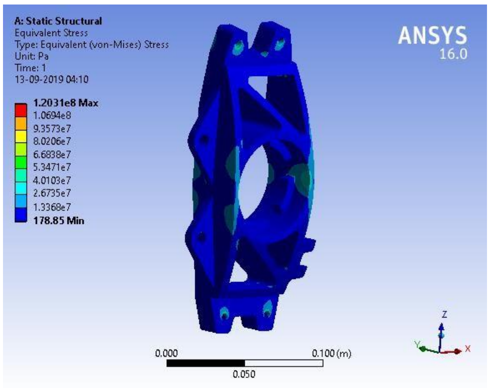


FIG: STRESS ANALYSIS OF FRONT UPRIGHT

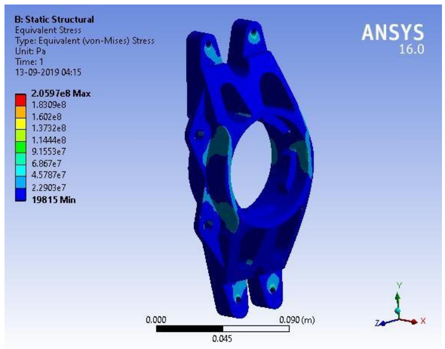


FIG: STRESS ANALYSIS OF REAR UPRIGHT

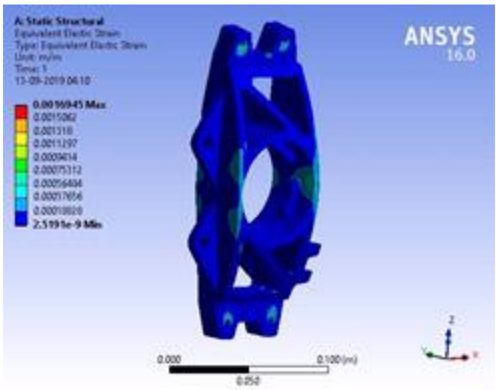
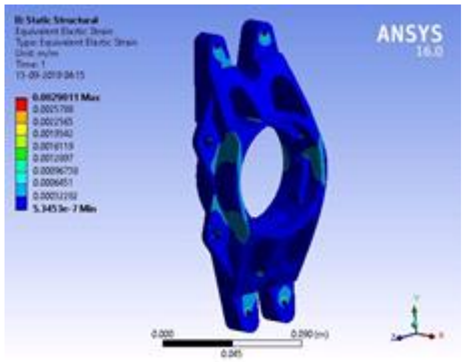


FIG: STRAIN ANALYSIS OF FRONT UPRIGHT



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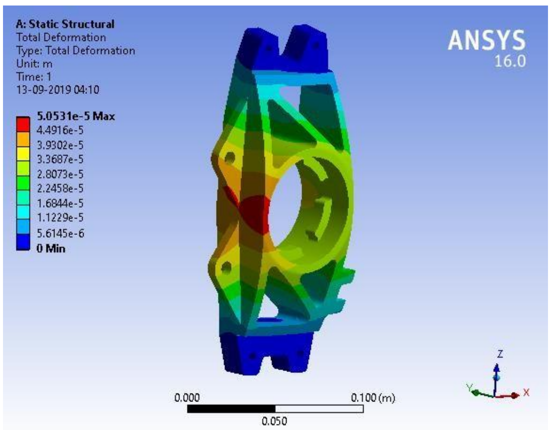
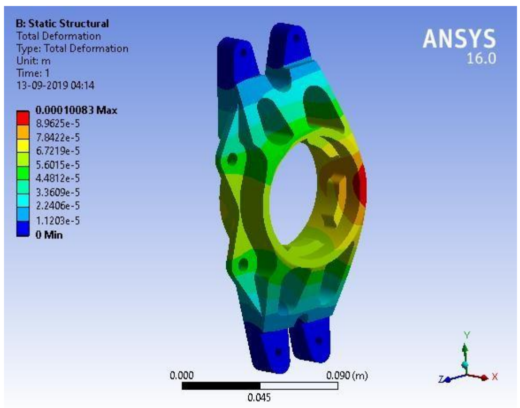


FIG: DISPLACEMENT OF FRONT UPRIGHT



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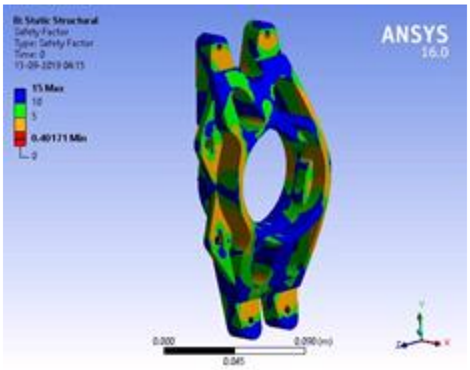

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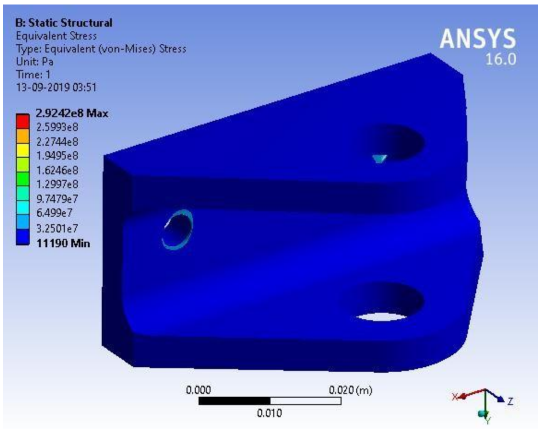
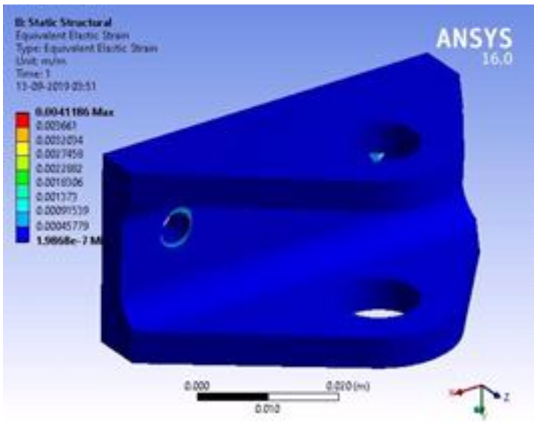


FIG: STRESS ANALYSIS OF FRONT CLEVIS



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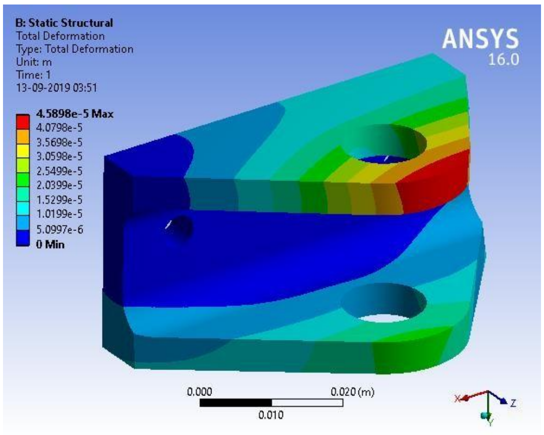

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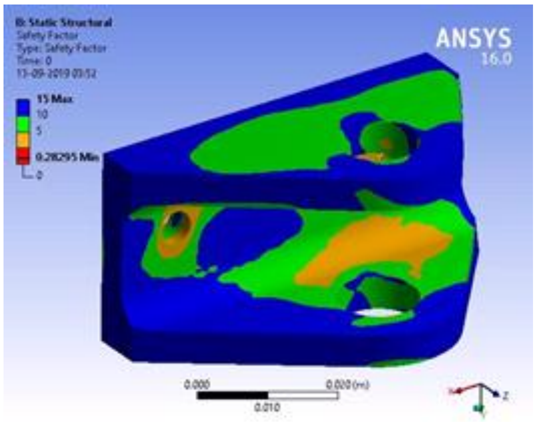

FIG: FOS OF FRONT CLEVIS

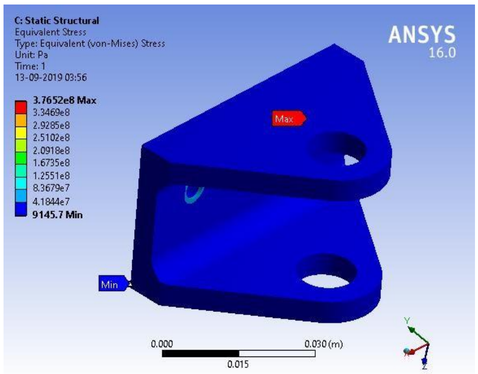

FIG: STRESS ANALYSIS REAR CLEVIS

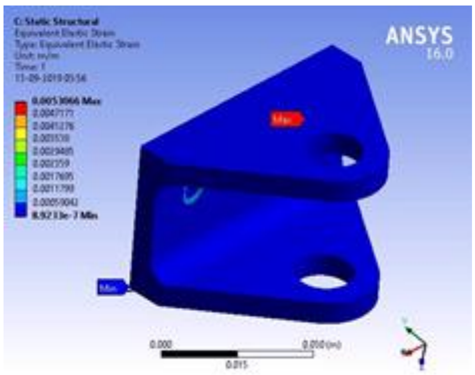

FIG: STRAIN ANALYSIS OF REAR CLEVIS

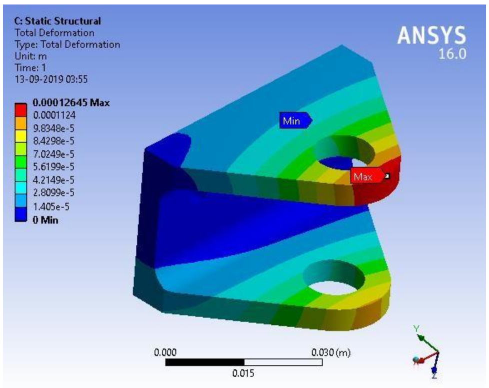

FIG: DISPLACEMENT OF REAR CLEVIS

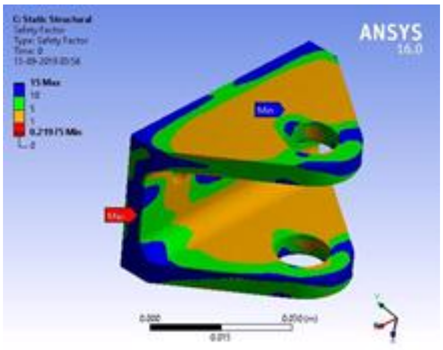

FIG: FOS OF REAR CLEVIS

10. CONCLUSION

Lots of conclusions can be made to this study. Increasing vehicle speed over a barrier caused, greater inertial imbalance, thus reducing the effect of anti-dive and anti-squat features which are designed essentially for normal pitch plane dynamics with smaller suspension vertical travel. This has been shown in the results of negotiating bumps at progressively higher forward speeds.

ACKNOWLEDGEMENT

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