21028_ INVINCIBLES 5.0 _CAE Report

Author's Name: Syed Md. Yasin

Co-Author's Name: Sharan .CH

1.INTRODUCTION

Computer aided engineering (CAE) is the process of solving engineering problems through the use of sophisticated, interactive graphical software. Our main aim is to design the vehicle ergonomically and ensure that it provides safety. To achieve the required safety level, we have followed an iterative approach to designing the vehicle. A large amount of effort has been made to keep the weight of the vehicle as low as possible.

The CAD software used to design and develop the 3D model was SolidWorks 2021 and the CAE software used to analyze the design was ANSYS 2021 R2.

2.FRAME MATERIAL DETAILS

Material-1 (AISI 4130)
Cross-Section Type; 25.4mm x 21.9 mm x 1.75mm

PARAMETER	AISI 4130
Density (kg/m²)	7850
Ultimate tensile strength (MPa)	600
Yield strength	460
Young modulus (GPa)	215
Poisson's ratio	0.29

Material-2 (AISI 1018)
Cross-Section Type; 25.4mm x 21.4mm x 2mm

PARAMETER	AISI 1018
Density (kg/m²)	7870
Ultimate tensile strength (MPa)	440
Yield strength	365
Young modulus (GPa)	205
Poisson's ratio	0.29

3.CALCULATION OF BENDING STRENGTH AND BENDING STIFFNESS

		Bending Strength	Bending Stiffness
	Unit	N-mm²	N-mm⁻¹
Material-1	AISI 4130		
Cross Section	A= 25.4mm B= 21.9mm T =1.75mm	**662.14**	**4142.41x10³**
Material-2	AISI 1018		
Cross Section	A=25.4mm B=21.4mm T=2mm	**582.68**	**4106.17x10³**

Material 1: AISI 4130
Outer Diameter (d_2) = **25.4 mm**
Inner Diameter (d_1) = **22.4 mm**
Yield Strength (S_y) = **460MPa**
C = Distance from neutral axis to extreme fiber = **12.7 mm**

1) Second Moment of Area (I)

$$I = \left(\frac{\pi}{32}\right) \times (d_{24} - d_{14})$$

$$= \frac{\pi \times (25.4^4 - 21.90^4)}{32}$$

$$\boldsymbol{I = 18,281 \times 10^{-12}\ N\ mm^{-2}}$$

2) Bending Strength (M)

$$M = \frac{S_y \times I}{C}$$

$$= \frac{(460 \times 10^6) \times (18281 \times 10^{-12})}{12.7 \times 10}$$

$$\boldsymbol{M = 662.14\ N\ mm^2}$$

3) Bending Stiffness (E)

$$Bending\ stiffness = E \times I$$

(We assume proportionality constant to be one for comparative purpose)

$$= (215 \times 10^9) \times (18281 \times 10^{-12})$$

$$\boldsymbol{Bending\ stiffness = 4142.41 \times 10^3\ N\ mm^{-1}}$$

Material 2: AISI 1018
Outer Diameter (d_2) = **25.4 mm**
Inner Diameter (d_1) = **21.4 mm**
Yield Strength (S_y) = **365MPa**
C = Distance from neutral axis to extreme fiber = **12.7 mm**

1) Second Moment of Area (I)

$$I = \left(\frac{\pi}{32}\right) \times (d_{24} - d_{14})$$

$$= \frac{\pi \times (25.4^4 - 21.4^{4)}}{32}$$

$$\boldsymbol{I = 20,274 \times 10^{-12}\ N\ mm^{-2}}$$

2) Bending Strength (M)

$$M = \frac{S_y \times I}{C}$$

$$= \frac{(365 \times 10^6) \times (20274 \times 10^{-12})}{12.7 \times 10}$$

$$\boldsymbol{M = 582.68\ N\ mm^2}$$

3) Bending Stiffness (E)

$$Bending\ stiffness = E \times I$$

(We assume proportionality
constant to be one for comparative purpose)

$$= (205 \times 10^9) \times (20274 \times 10^{-12})$$

$$\boldsymbol{Bending\ stiffness = 4106.17 \times 10^3\ N\ mm^{-1}}$$

4. CAE ANALYSIS OF VEHICLE/FRAME

CAE Analysis on the frame was performed to evaluate the safety offered by the frame to drivers in impact conditions including Frontal impact, Side Impact, Roll over along with the analysis in case of bending load case, torsional load case and loading on hard-points on frame. The whole vehicle model is subjected to this safety analysis through CAE. A vehicle frame is the main supporting structure of a motor vehicle to which all other components are attached/ mounted. The main functions of a frame in a vehicle are
1. To support the vehicle's mechanical components and riders.
2. To deal with static and dynamic loads, without undue deflection or distortion,
For example:
• Vertical and torsional twisting transmitted by going over uneven surfaces.
• Transverse lateral forces caused by road conditions, side wind, and steering the vehicle.
• Torque from the engine and transmission.
• Longitudinal tensile forces while acceleration, as well as compression at the time of braking.
• Sudden impacts from collisions. Thus, it is required to Analyse the frame structure when acted upon by these loads to offer a safe drive. Finite Element Analysis is a mathematical modelling technique used to determine the response of real structures to external and internal loads. CAE Analysis on the frame is performed to evaluate the safety offered by the frame to drivers in case of any accident including Frontal Impact, Side Impact and Rollover.

We have developed a graph between the mesh size and maximum stress to check the optimal mesh size to be taken

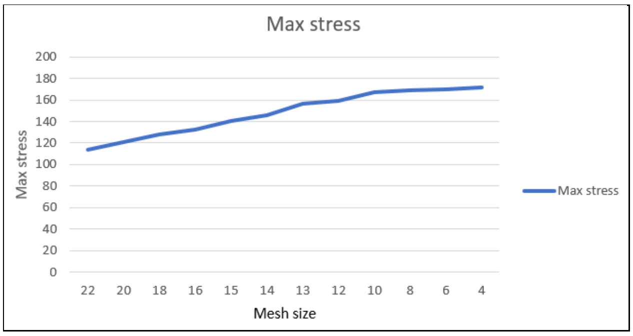


Fig 1- Mesh size vs Max stress

From the above graph, it is evident that the max stress increases considerably between the mesh size of 22mm to 10mm. It is also noticed that the maximum stress remains about the same after mesh size of 10mm and mesh size does not play a major role in the result.

For the above reason, we decided to take a mesh size of 8mm, which was a good compromise between obtaining accurate results and computing time.

4.1. FRONTAL IMPACT ANALYSIS

a) **Assumption & Considerations**:

For the boundary conditions in front impact tests, the suspension mounting faces on the chassis are fixed and load/force is applied on front face and back face members of chassis.

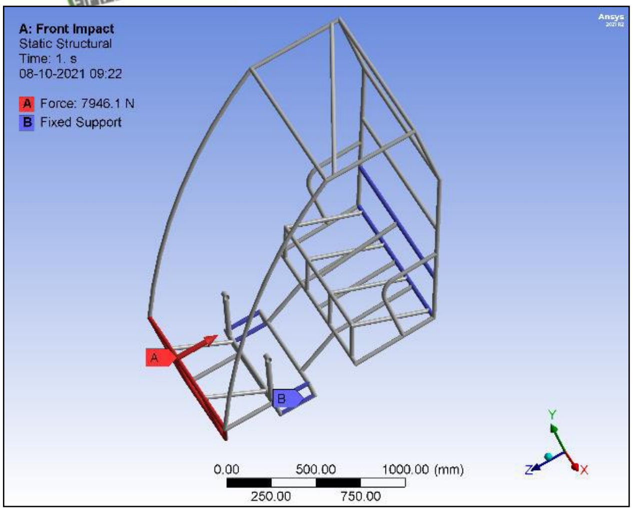


Fig 1- Boundary conditions

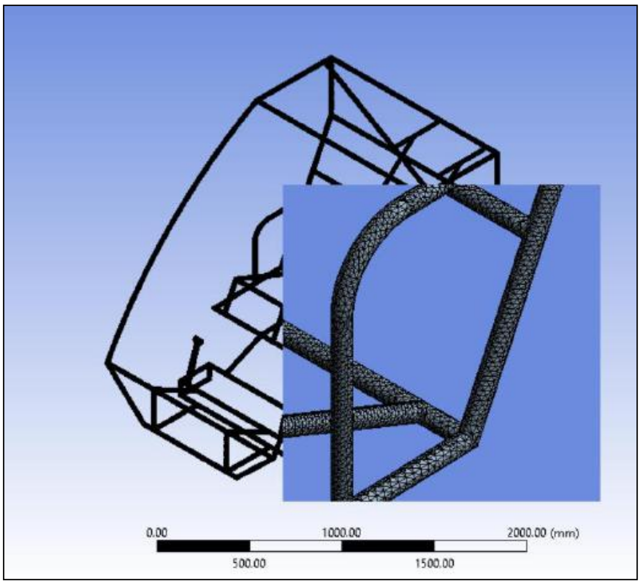
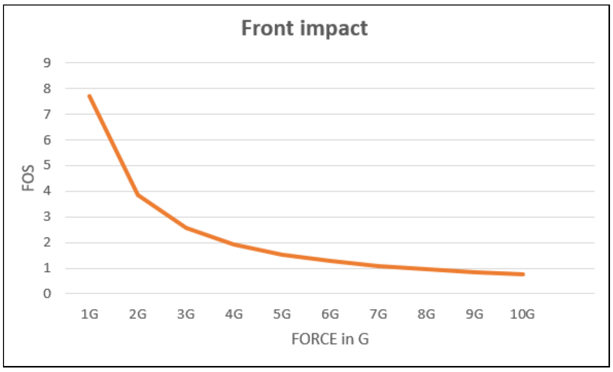


Fig 2- Meshed view

b) Calculation of Impact Forces:



Graph is drawn between FOS and Force(G)

From the graph, at FOS 1.5 the value of force is 5G

The value of force is newtons is 13,243.5 N

Calculation of velocity of impact:

$$V = G \times g \times t$$

Where V= Velocity of impact

G= Acceleration multiplier

g= Acceleration due to gravity

t= Time of impact

$$V = 5 \times 9.81 \times 0.3$$

$$V = 14.715\ ms^{-1} = 52.974\ kmph$$

The vehicle is found to be safe if it collides head on with a stationary wall at 52kmph.

c)Analysis Results:

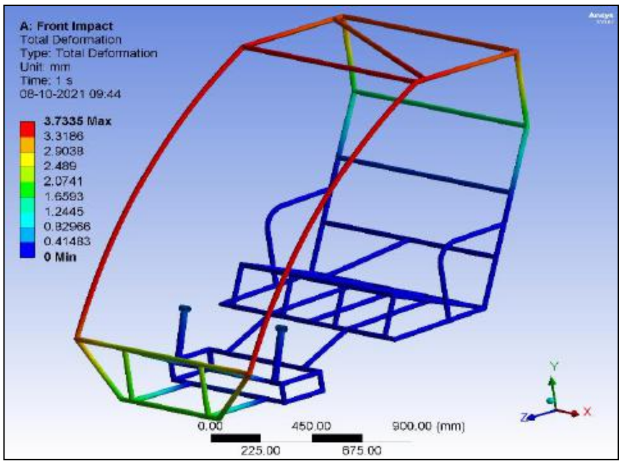


Fig 3- Total deformation

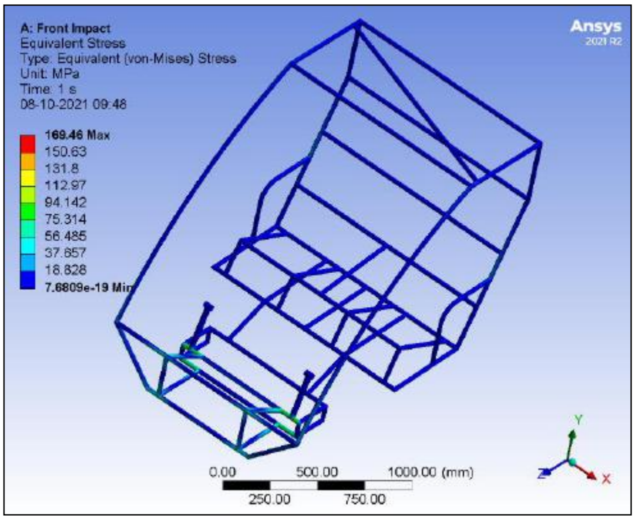


Fig 4- Equivalent stress

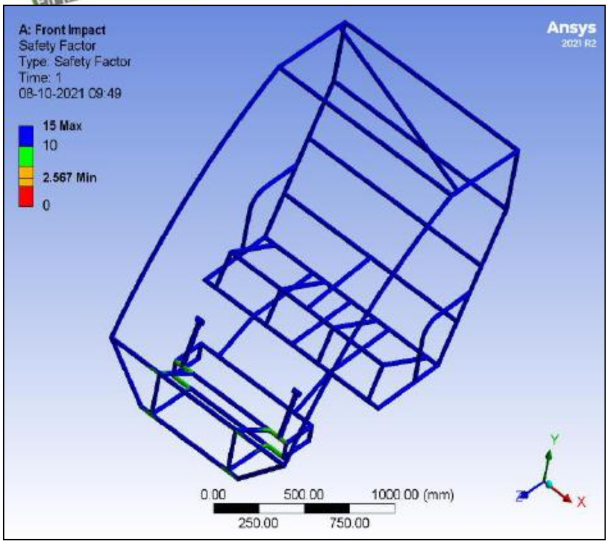


Fig 5 - Factor of safety

Maximum deformation	**3.73mm**
Maximum stress	**169.46MPa**
Factor of safety	**2.56**

d)Optimizations:

The FOS is good enough, thus no further optimization was required.
This was the 4th iteration of the frame.

4.2. SIDE IMPACT ANALYSIS

a) Assumption & Considerations:

For boundary conditions in side impact the frame is fixed either from left/right and other side members will come across the applied load. Suspension mounting faces on the chassis are fixed. Weight of the vehicle is considered 270kg including driver's weight.

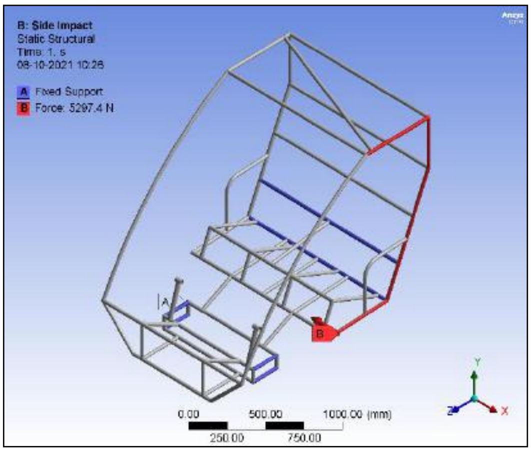


Fig 1- Boundary conditions

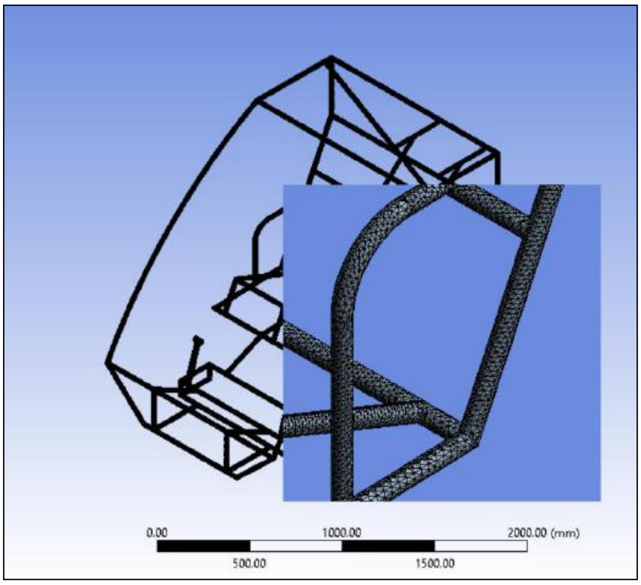
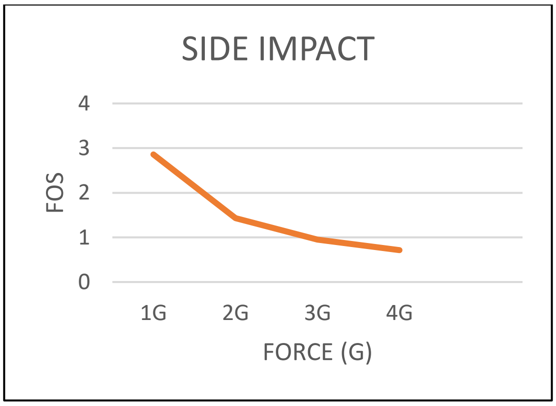


Fig 2- Meshed view

b) Calculation of Impact Forces:



Graph is drawn between FOS and Force(G)

From the graph, at FOS 1.5 (approx) the value of force is 2G (approx)

The safe value of force in newtons is 5297.4 N

$$\boldsymbol{V = G \times g \times t}$$

$$\boldsymbol{V = 2 \times 9.81 \times 0.3}$$

$$\boldsymbol{V = 5.886\ ms^{-1} = 21.1896\ kmph}$$

The vehicle is found to be safe if it collides sideways with a velocity of 21 kmph.

c) Analysis Results:

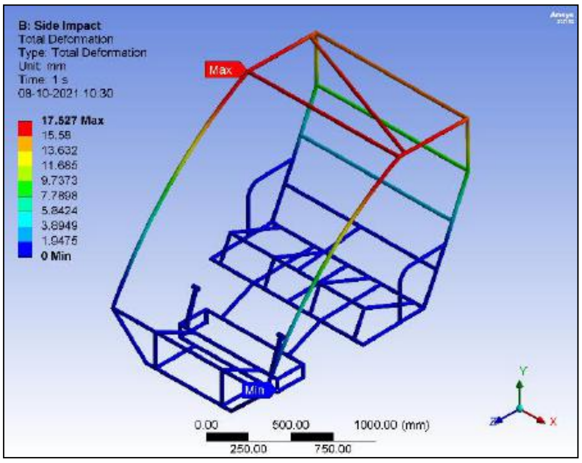


Fig 3- Total deformation

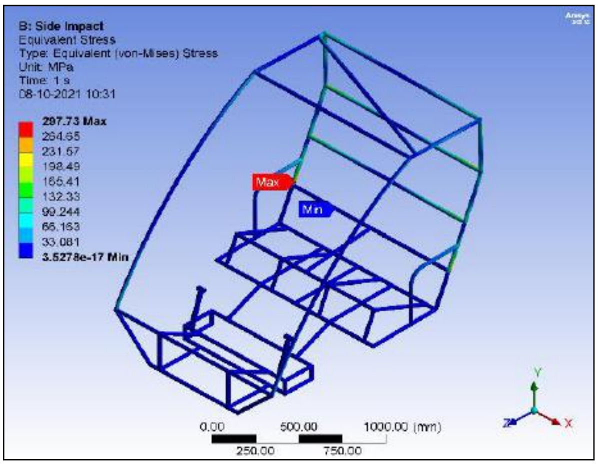


Fig 4- Equivalent stress

Maximum deformation	**17.52**
Maximum stress	**297.73MPa**
Factor of safety	**1.461**

d) Optimizations:

The FOS is good enough, thus no optimization was required.
This was the 4th iteration of the frame.

4.3. ROLLOVER ANALYSIS

4.3.1 FRONT ROLL OVER ANALYSIS

a) Assumption & Considerations:

For boundary conditions for roll over impact test, the frame is fixed on the bottom side and the top members will come across the applied load. For the calculation of the impact caused due to a rollover, the vehicle of mass 270kg. (Taking worst case scenario into consideration).

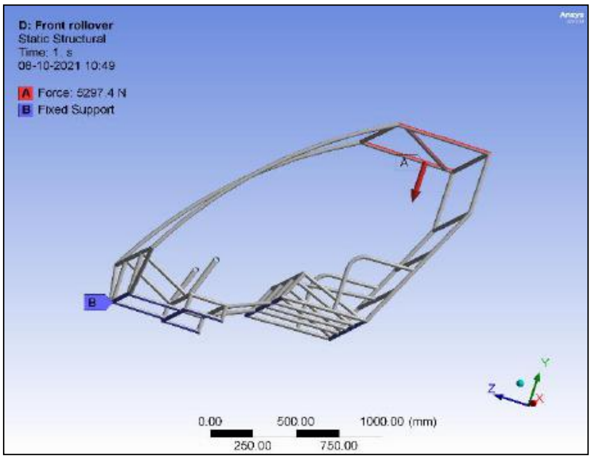


Fig 1- Total deformation

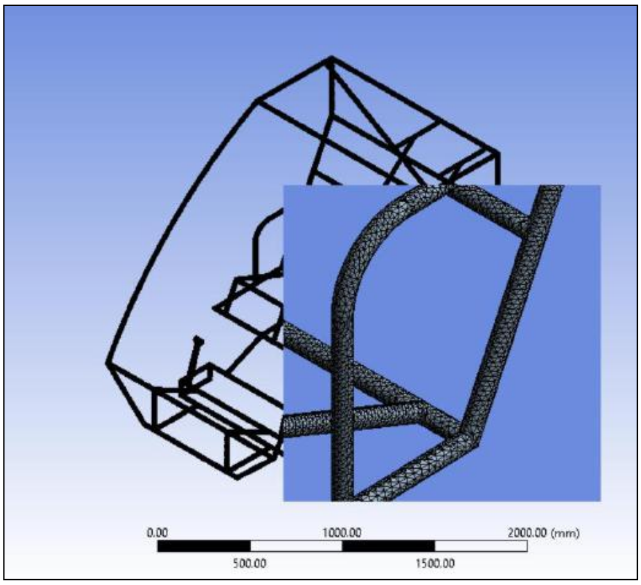
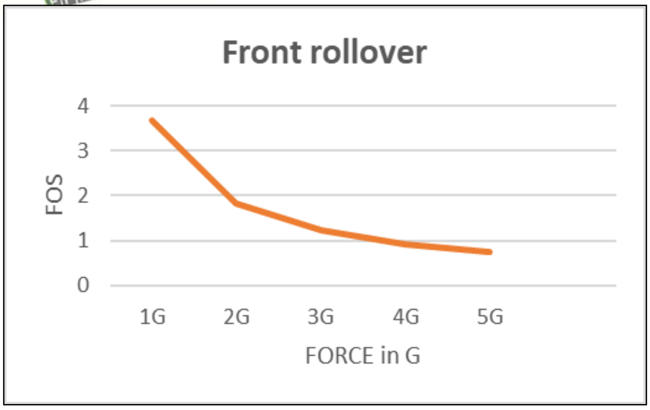


Fig 2- Meshed view

b) Calculation of Impact Forces:



Graph is drawn between FOS and Force(G)

From the graph , at FOS 1.5 (approx) the value of force is 2.5G.

The value of force under which vehicle is deemed to be safe is 6,621.75 N.

c) Analysis Results:

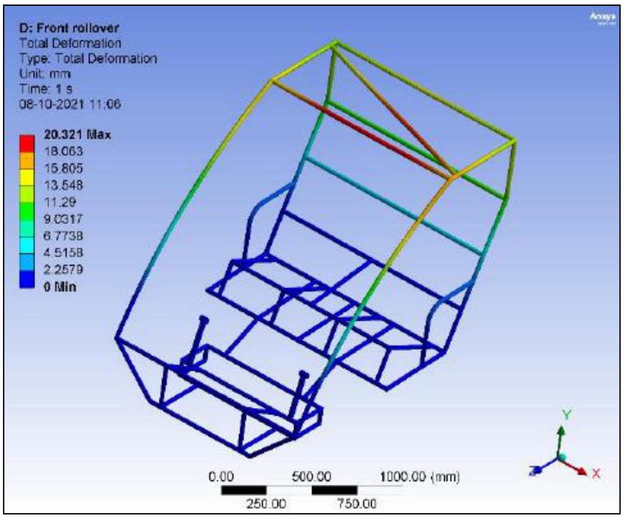


Fig 3- Total deformation

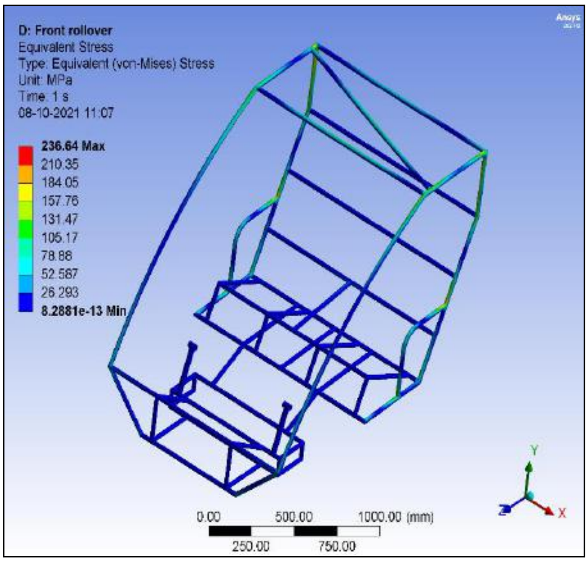


Fig 4- Equivalent stress

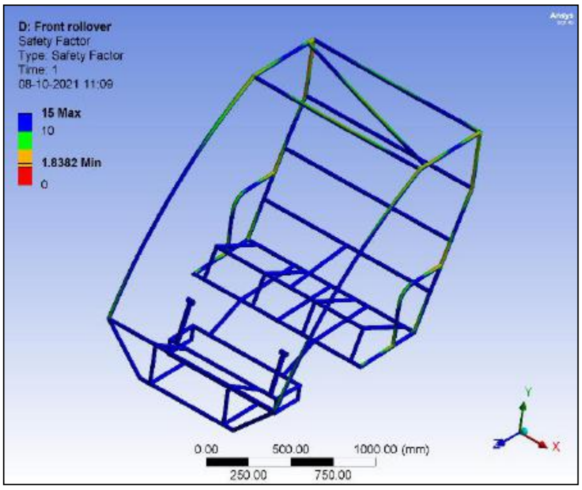


Fig 5- Factor of safety

Maximum deformation	**20.23mm**
Maximum stress	**236.64MPa**
Factor of safety	**1.83**

d) Optimizations:

The FOS is good enough, thus no optimization was required.

This was the 4th iteration of the frame.

4.3.2 SIDE ROLL OVER ANALYSIS

a) Assumption & Considerations:

This impact test is done to simulate the condition when the vehicle rolls sideways, due to excessive cornering force. Total weight of vehicle including drivers is taken as 270kg.
For the boundary conditions for the side roll analysis, the base plane members are fixed.

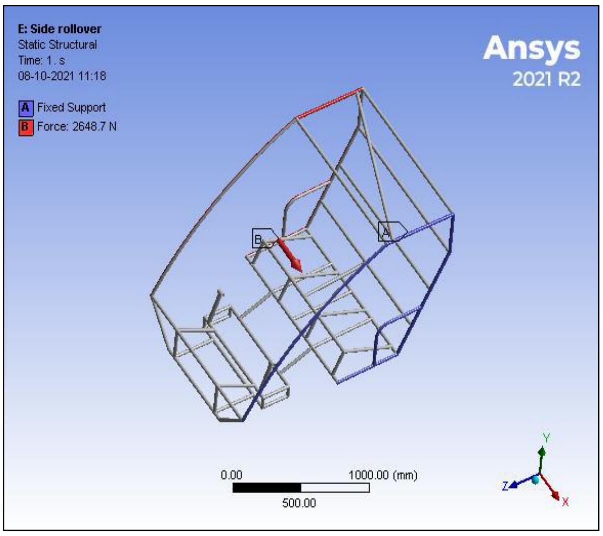


Fig 1- Boundary conditions

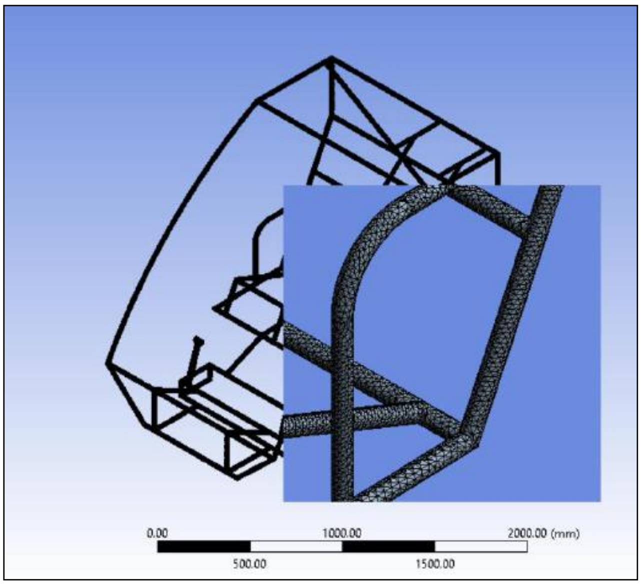
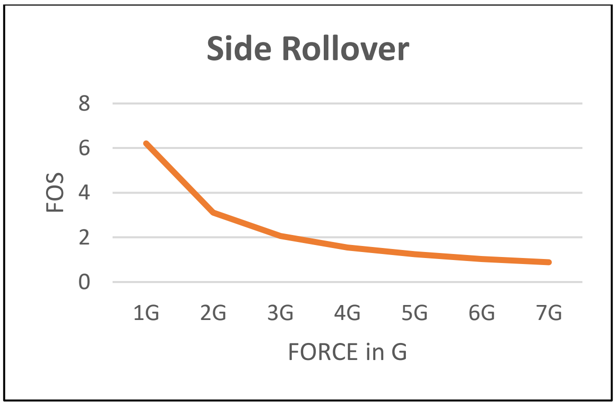


Fig 2- Meshed view

b) Calculation of Forces:



Graph is drawn between FOS and Force(G)

From the graph , at FOS 1.5, the value of force is 4.5 G (approx)

The value of force under which vehicle is deemed to be safe is 11,919.15 N.

c) Analysis Results:

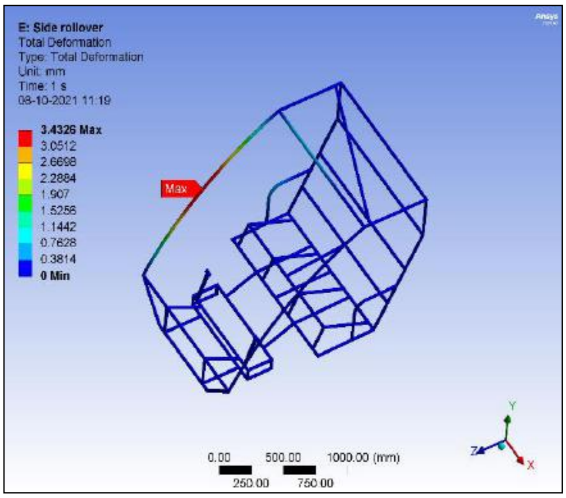


Fig 3- Total deformation

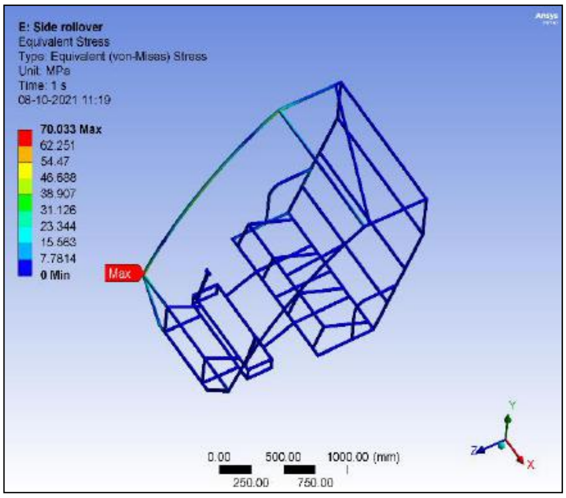


Fig 4- Equivalent stress

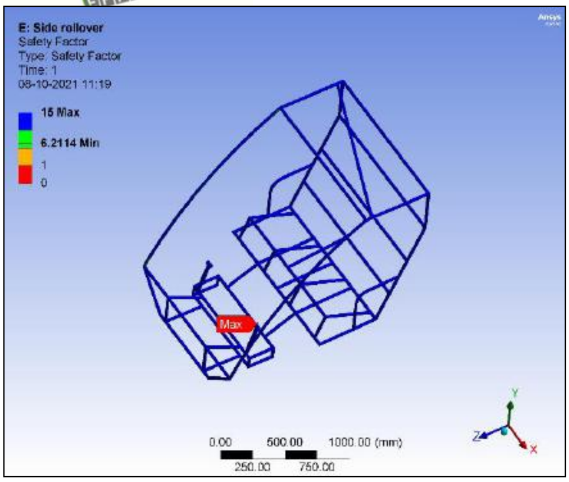


Fig 5- Factor of safety

Maximum deformation	**3.43 mm**
Maximum stress	**70.03 MPa**
Factor of safety	**6.21**

d) Optimizations:

The FOS is good enough, thus no optimization was required.
This is the 4th iteration of the frame.

4.4. BENDING ANALYSIS

4.4.1 PAY LOAD BOX MOUNTINGS

a) Assumption & Considerations:

The hardpoint members of chassis are fixed and the faces of rods which are in contact with utility box are given a load of **500N** in vertical-direction.

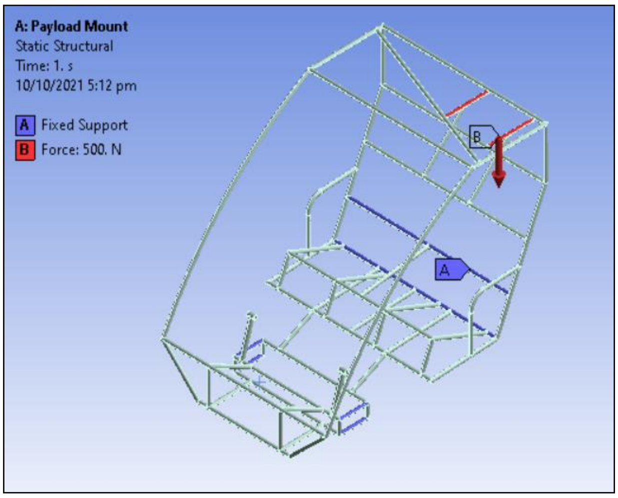


Fig 1- Boundary conditions

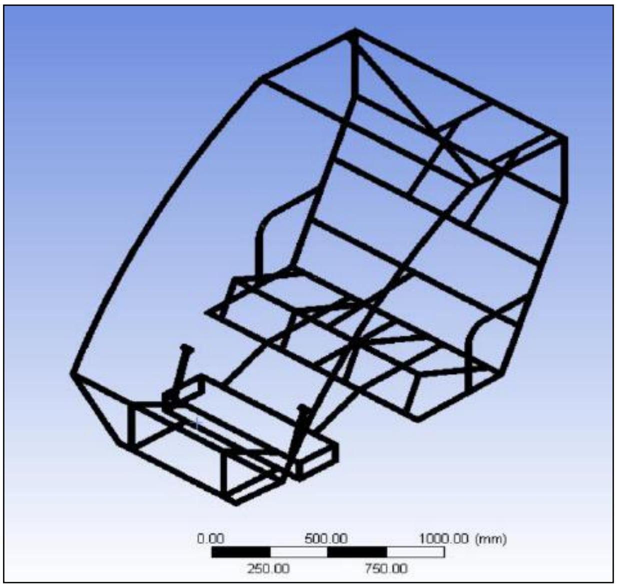


Fig 2- Meshed view

b) Calculation of Forces:

Weight in utility box = 20kg
Weight of utility box = 3kg
Total weight = 23kg
F = ma = 20 x 9.8 = **225.63N**
But a force of 500N was taken to endure safety under extreme conditions

c) Analysis Results:

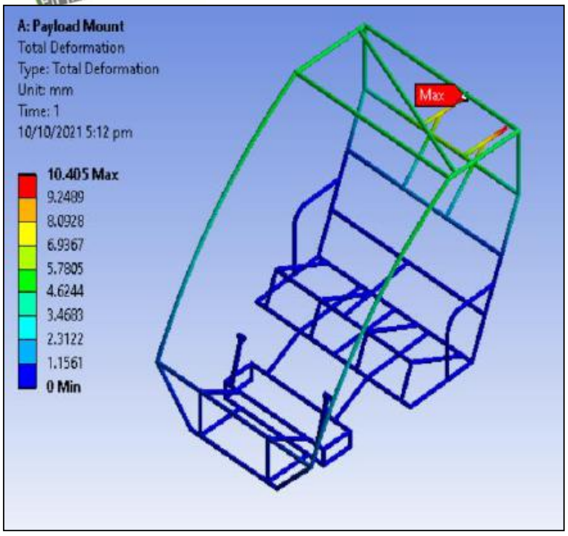


Fig 3- Total deformation

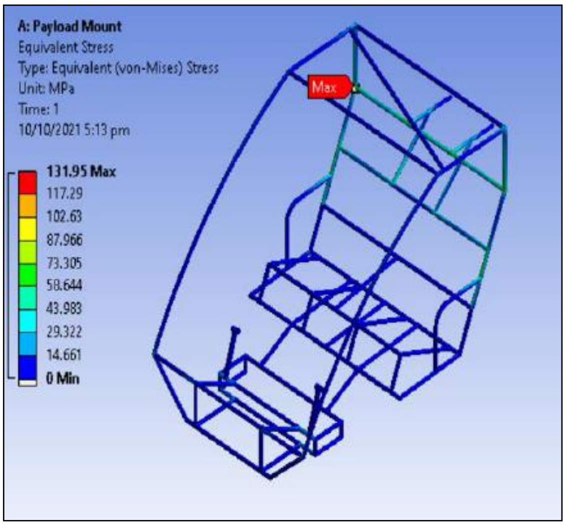


Fig 4- Equivalent stress

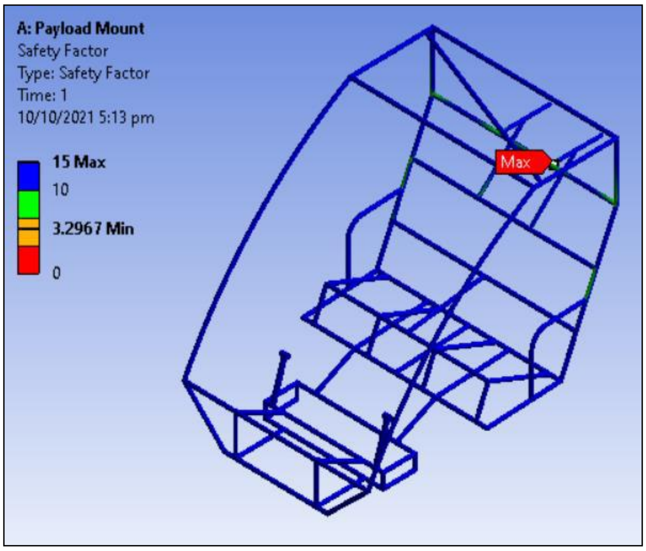


Fig 5- Factor of safety

Maximum deformation	**10.40mm**
Maximum stress	**131.95 MPa**
Factor of safety	**3.29**

d) Optimizations:

The FOS is good enough, thus no optimization was required.

4.4.2 HAND SUPPORTS

a) Assumption & Considerations:

The two side faces of the chassis are fixed. It is assumed that a driver of 75 kg is hanging with on driver support rod and reaction force of 300N.

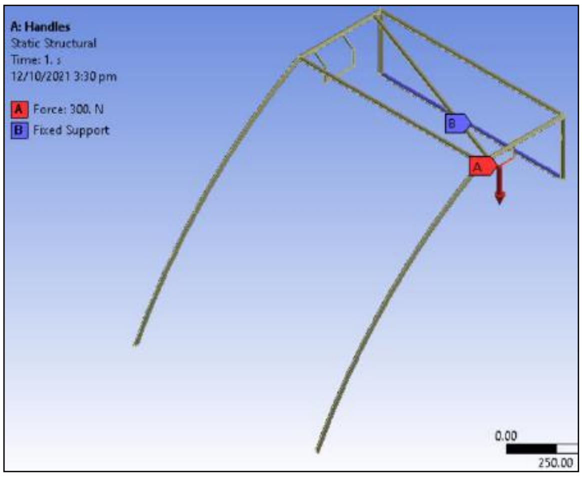


Fig 1- Boundary conditions

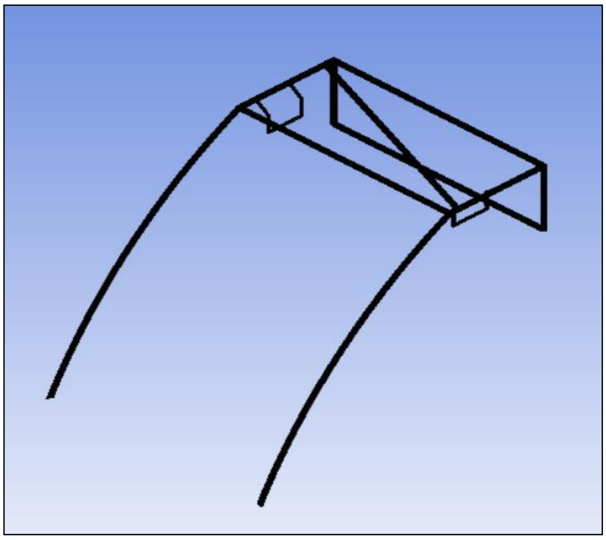


Fig 2- Meshed view

b) Calculation of Forces:

Approximate of 300N of reaction force of hand is considered when a 75Kg person taking support of hand supports.

c) Analysis Results:

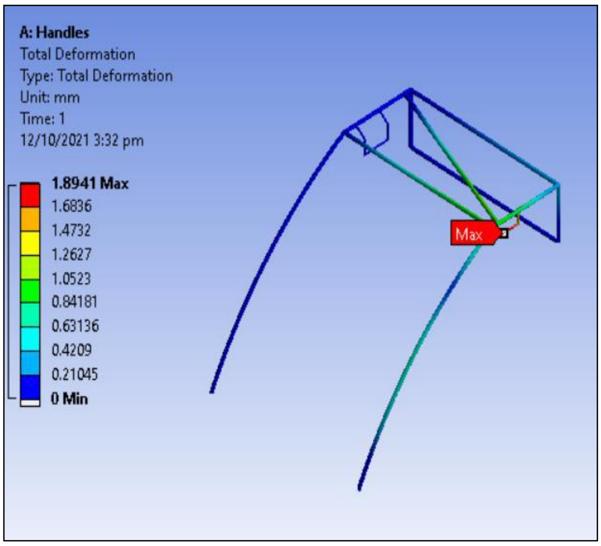


Fig 3-Total deformation

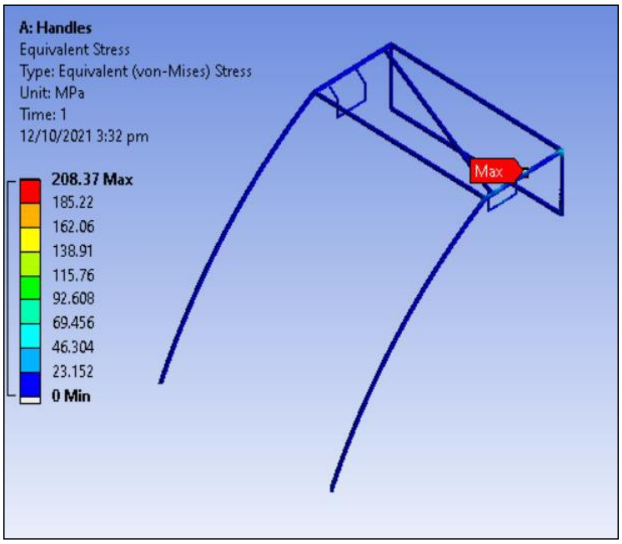


Fig 4- Equivalent stress

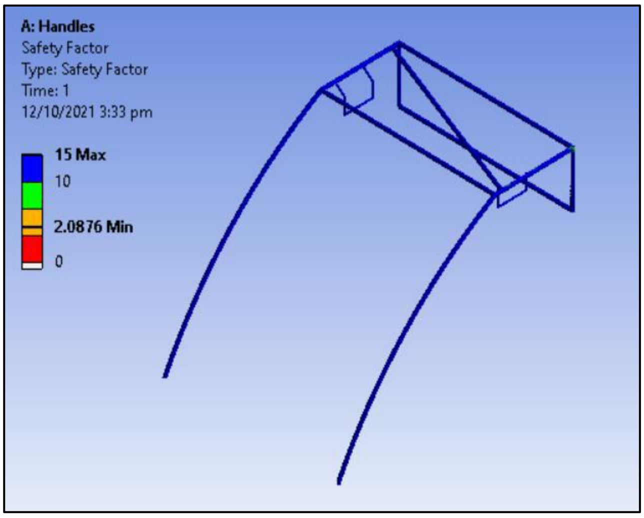


Fig 5- Factor of safety

Maximum deformation	**1.89mm**
Maximum stress	**208.37 MPa**
Factor of safety	**2.08**

d) Optimizations:

The FOS is good enough, thus no optimization was required.

4.4.3 BATTERY MOUNTING

a) Assumption & Considerations:

weight of the battery is 12kg so the reaction force is 120N is applied of battery mount.

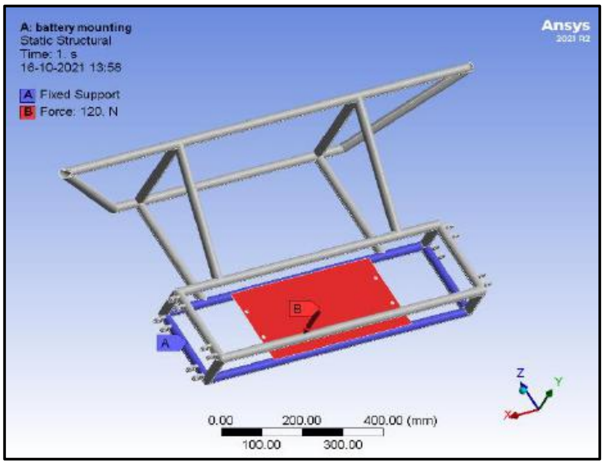


Fig 1- Boundary condition

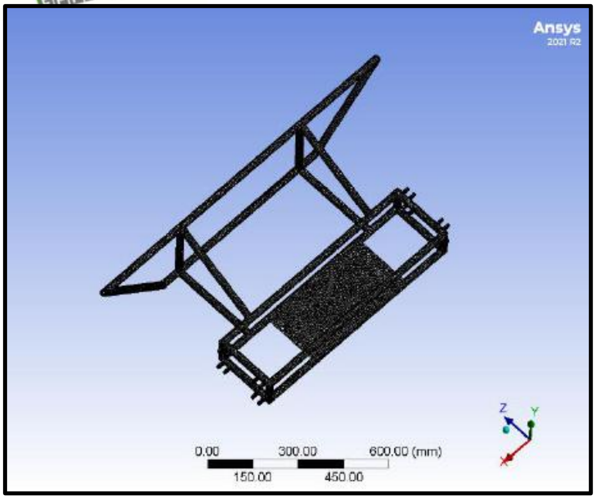


Fig 2- meshed view

b) Calculation of Forces:

weight of battery=12kg
Reaction force = 12x10=120N

c) Analysis Results:

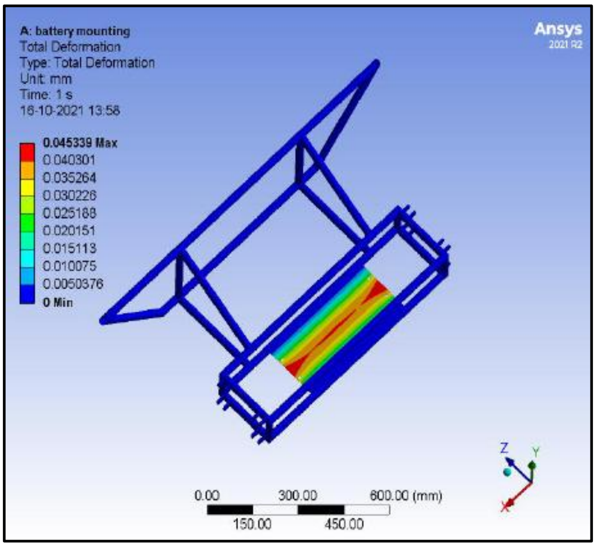


Fig 3- Total deformation

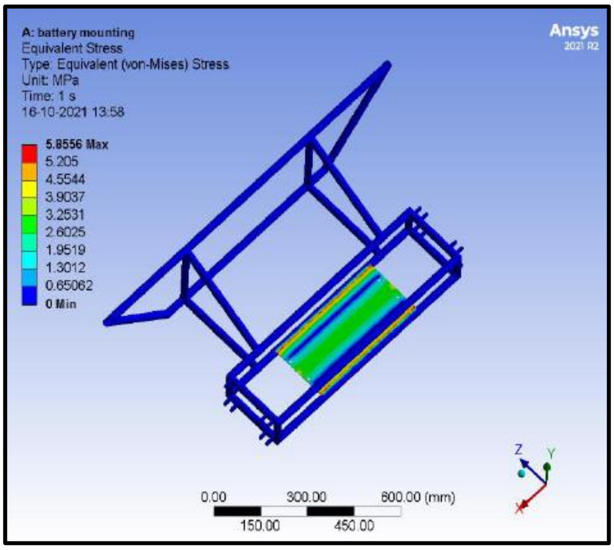


Fig 4- Equivalent stress

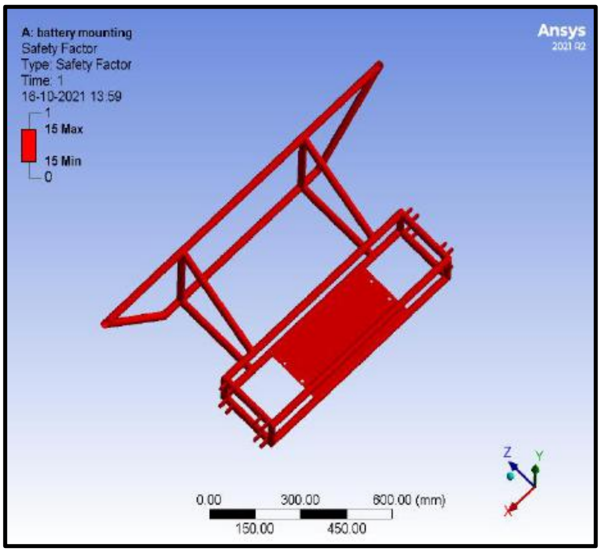


Fig 5- Factor of safety

Maximum deformation	**0.045mm**
Maximum stress	**5.85 MPa**

d) Optimizations:

The FOS is good enough, thus no optimization was required.

4.4.4. PEDAL MOUNTINGS

a) Assumption & Considerations:

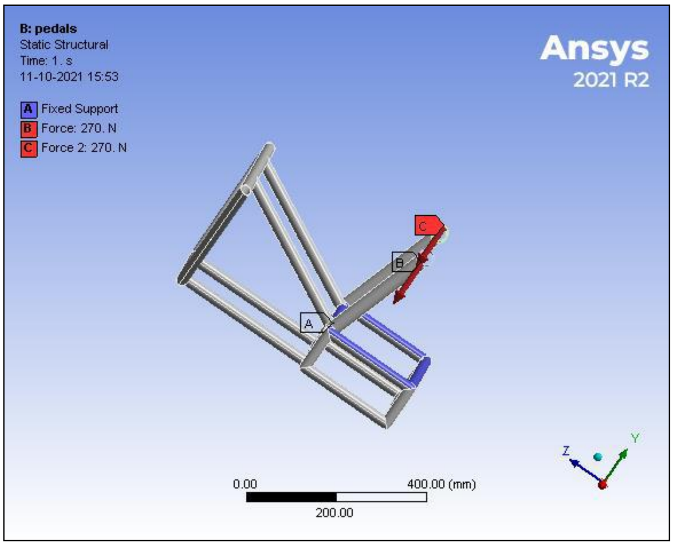


Fig (1) Boundary conditions

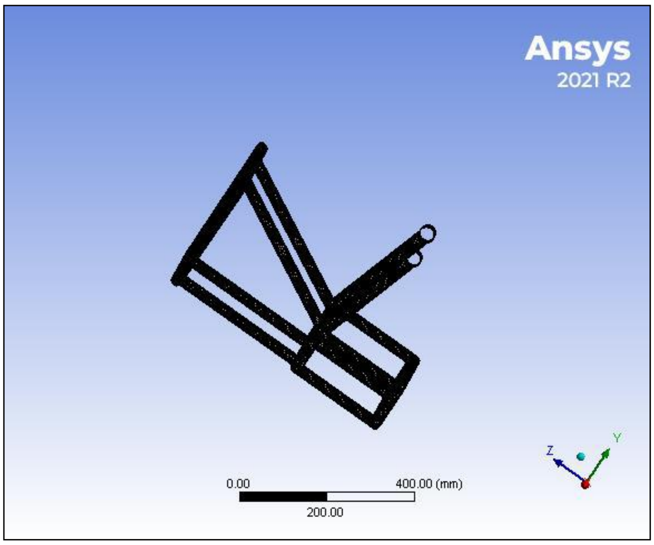


Fig 2- Meshed view

b) Calculation of Forces:

Human average pedal force =270N

c) Analysis Results:

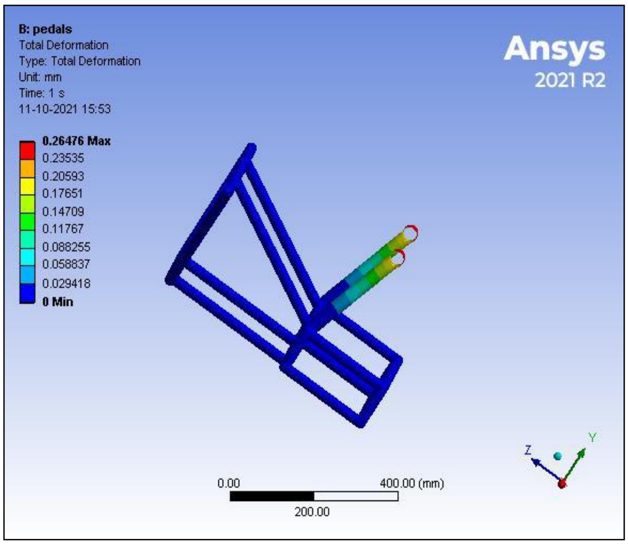


Fig 3- Total deformation

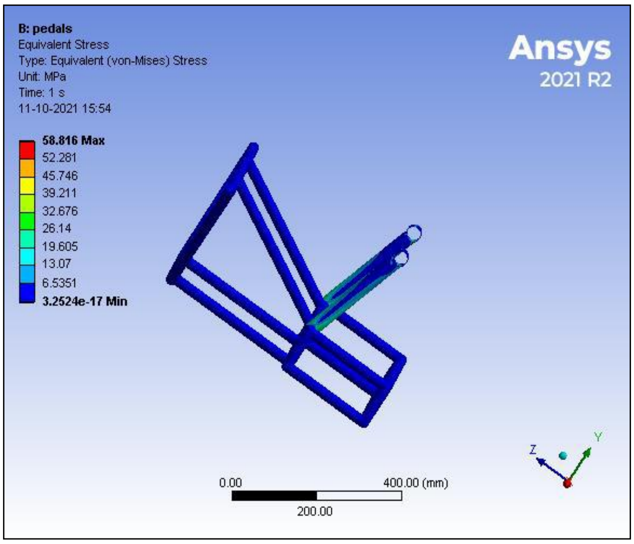


Fig 4- Equivalent stress

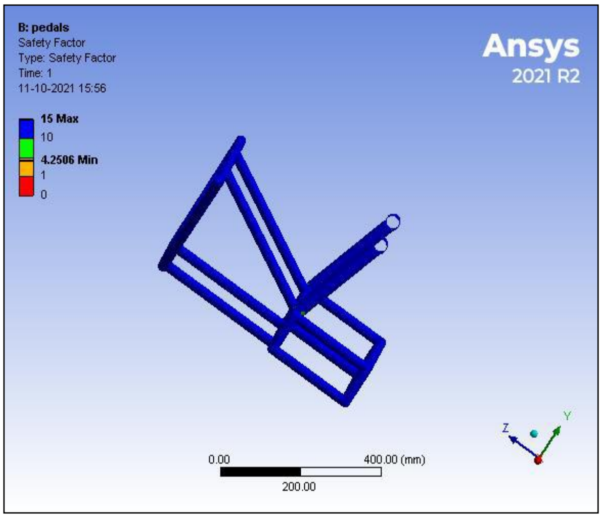


Fig 5- Factor of safety

Maximum deformation	**0.26mm**
Maximum stress	**58.81MPa**
Factor of safety	**4.25**

d) Optimizations:

The FOS is good enough, thus no optimization was required.

4.5 TORSIONAL ANALYSIS

4.5.1 BUMP TEST ON SINGLE WHEEL

a) Assumption & Considerations:

weight of the vehicle is 270kg including driver's weights.

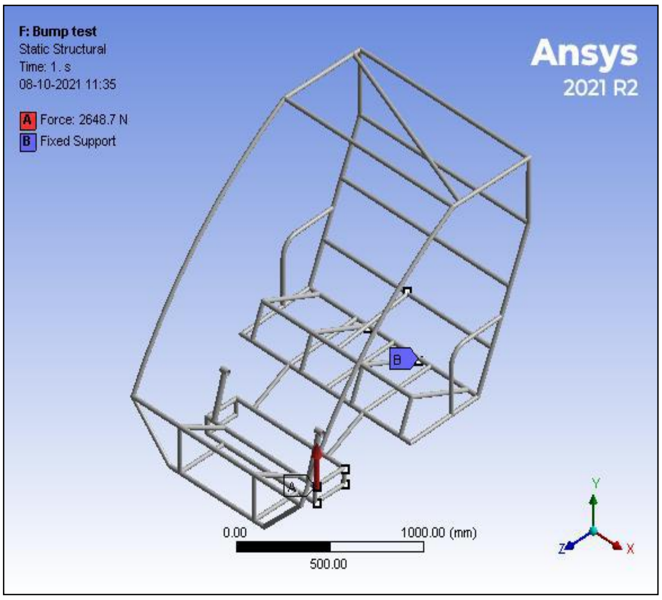


Fig 1- Boundary conditions

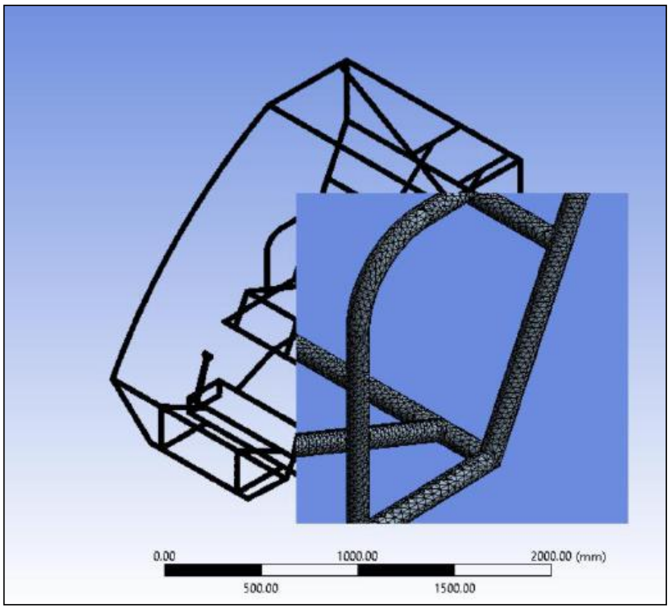
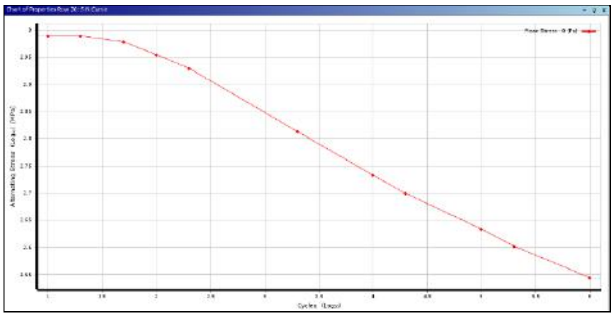


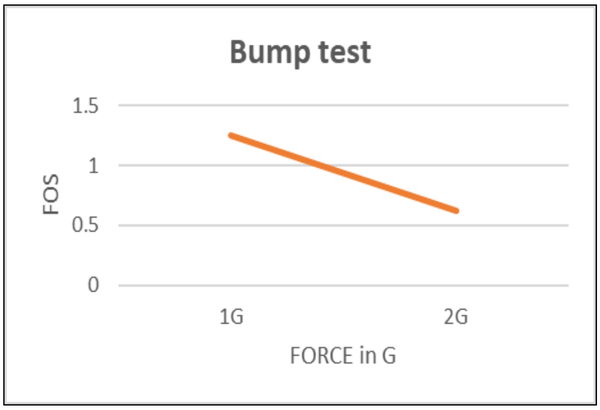
Fig 2- Meshed view

SN curve of AISI 4130:

	B	C
1	Cycles	Alternating Stress (MPa)
2	10	975
3	20	975
4	50	950
5	100	900
6	200	850
7	2000	650
8	10000	540
9	20000	500
10	1E+05	430
11	2E+05	400
12	1E+06	350



b) Calculation of Forces:



Graph is drawn between FOS and Force(G)

From the graph , at FOS 1.5(approx) the value of force is 1G (approx)

The value of force is newtons is approx 2648.7 N

c) Analysis Results:

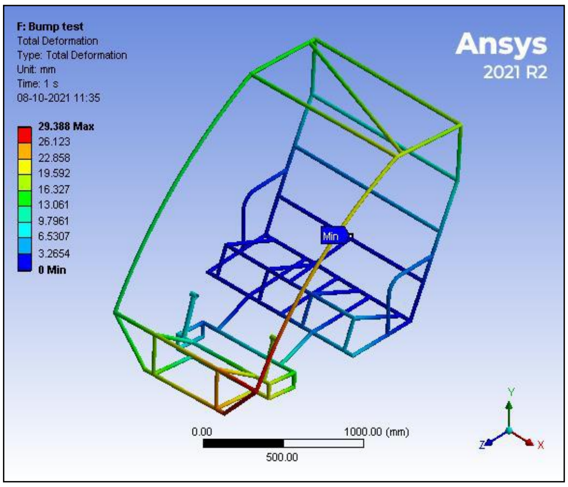


Fig 3- Total deformation

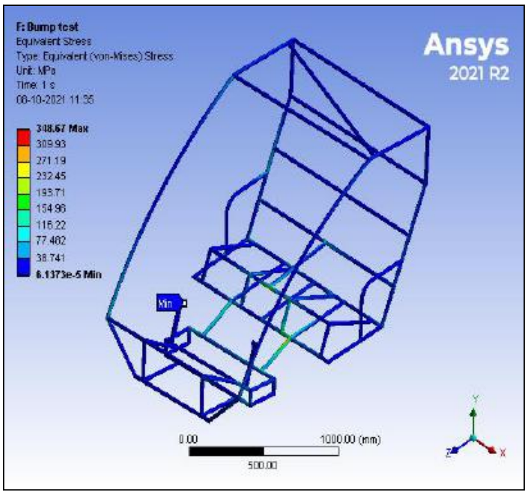


Fig 4- Equivalent stress

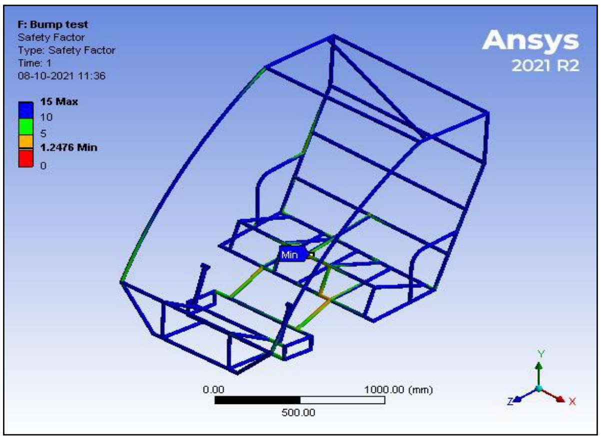


Fig 5- FoS (static)

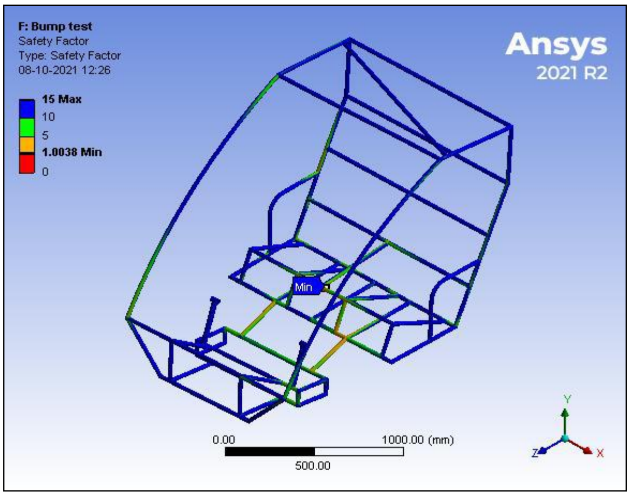


Fig 6- FOS (Fatigue)

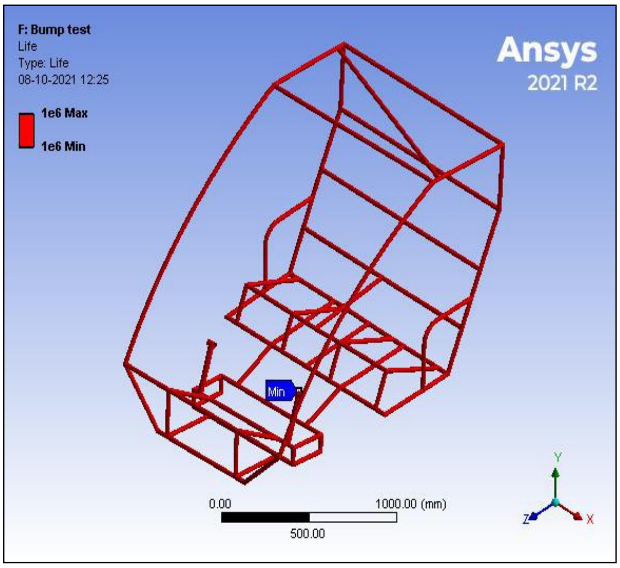


Fig 7- Minimum fatigue life

Maximum deformation	**29.38mm**
Maximum stress	**348.67MPa**
Factor of safety(static)	**1.24**
Factor of safety (fatigue)	**1.003**
Minimum fatigue life	**Infinite life**

d) Optimizations:

The FOS is good enough, thus no optimization was required.

4.5.2 TORSIONAL

a) Assumption & Considerations:

Two forces are applied on frame in opposite directions considering the weight of vehicle including drivers is 270kg

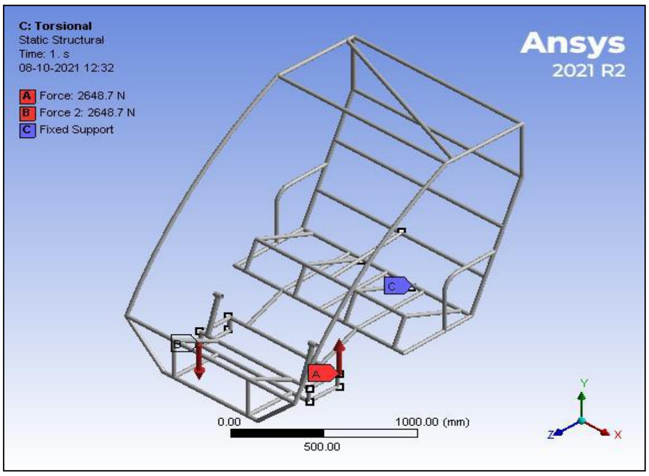


Fig 1- Boundary conditions

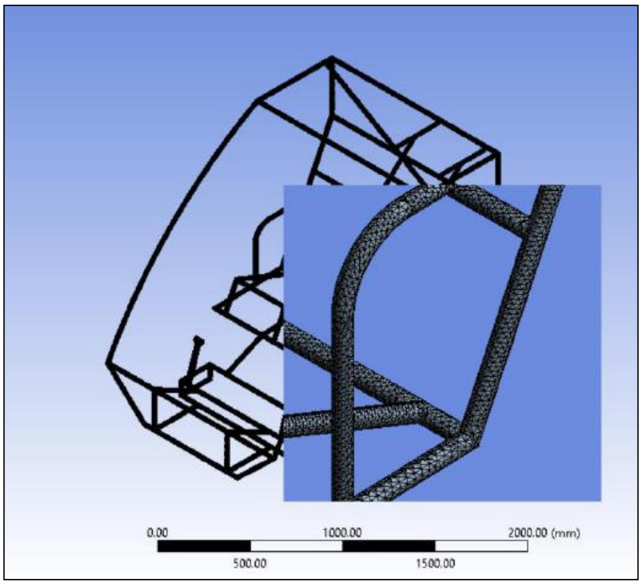
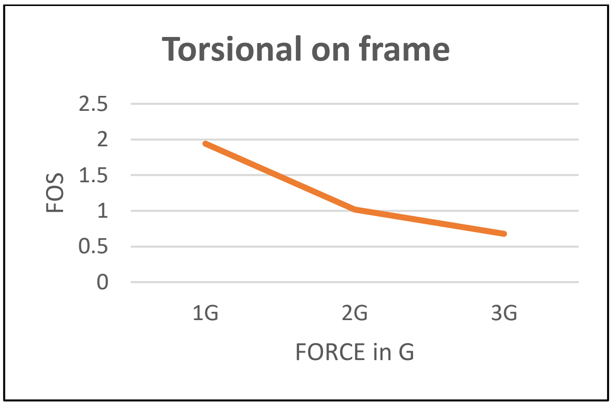


Fig 2- Meshed view

b) Calculation of Forces:



Graph is drawn between FOS and Force(G)

From the graph , at FOS 1.5(approx) the value of force is 1G (approx)

The value of force is newtons is approx 2648.7 N

c) Analysis Results:

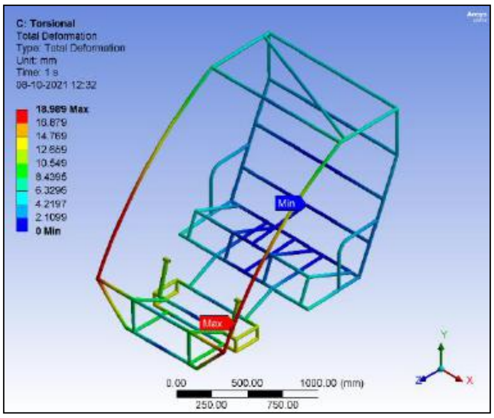


Fig 3- Total deformation

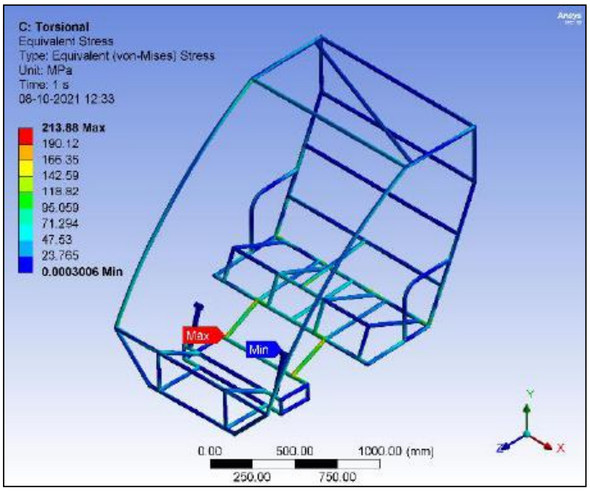


Fig 4- Equivalent stress

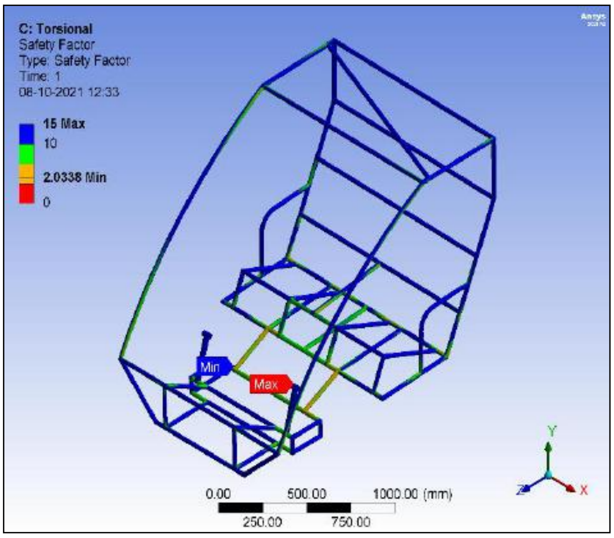


Fig 5- FoS(static)

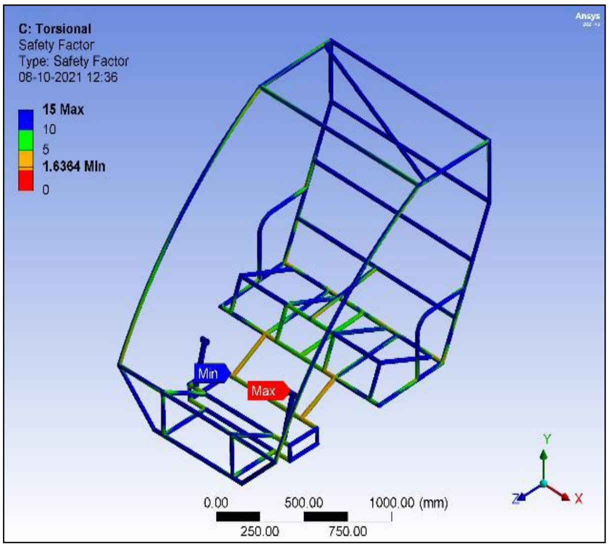


Fig 6- FOS (fatigue)

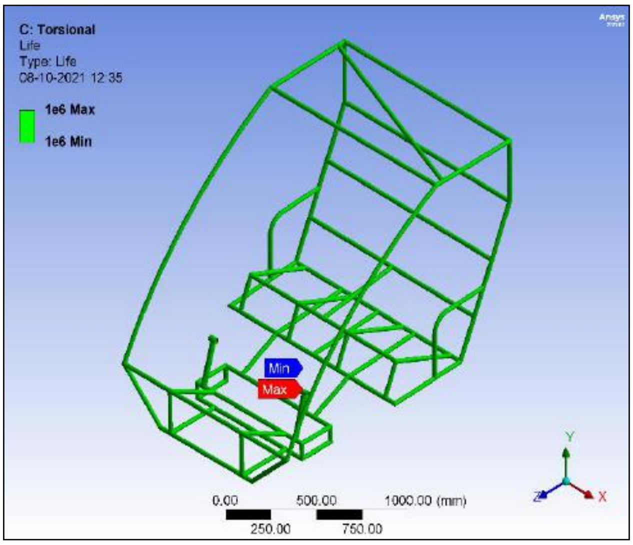


Fig 7- Minimum fatigue life

Maximum deformation	**18.98mm**
Maximum stress	**213.88MPa**
Factor of safety(static)	**2.03**
Factor of safety (fatigue)	**1.63**
Minimum fatigue life	**Infinite life**

d) Optimizations:

The FOS is good enough, thus no optimization was required.

4.5.3 BUMP TEST ON BOTH WHEELS

a) Assumption & Considerations:

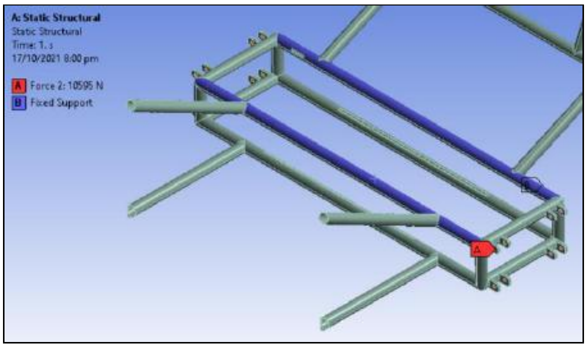


Fig 1- Boundary conditions

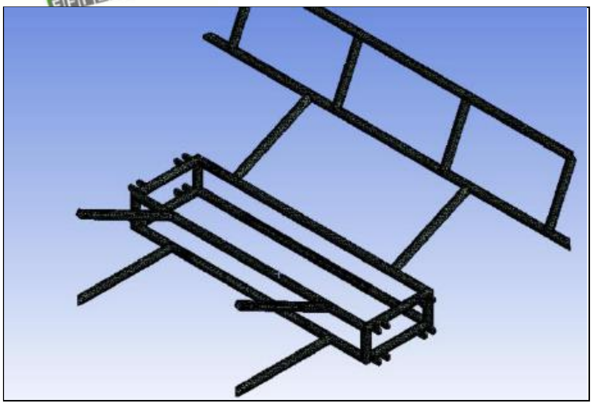


Fig 2- Meshed view

b) Calculation of Forces:

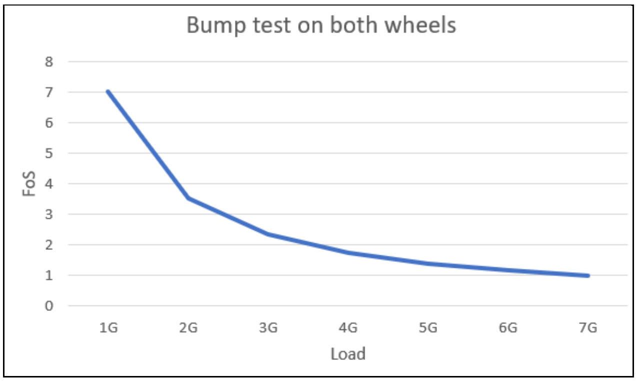


Fig 3- Load vs FoS

Graph is drawn between FOS and Force(G)

From the graph, at FOS 1.5 (approx) the value of force is 4.5G.

The value of force under which vehicle is deemed to be safe is 11,919.15 N.

c) Analysis results:

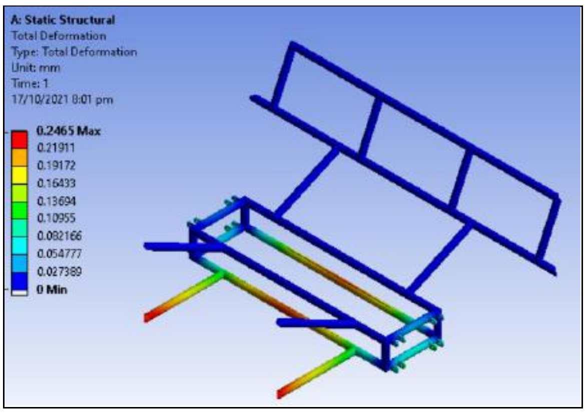


Fig 4- Deformation

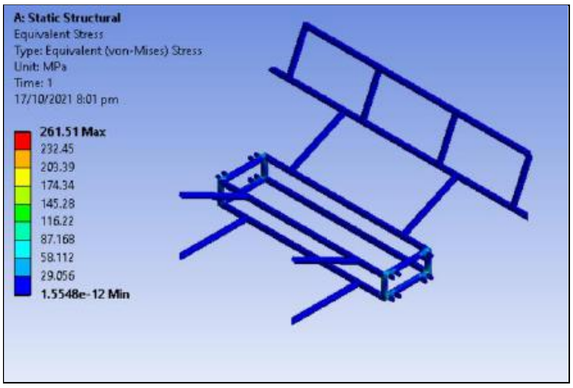


Fig 5- Stress

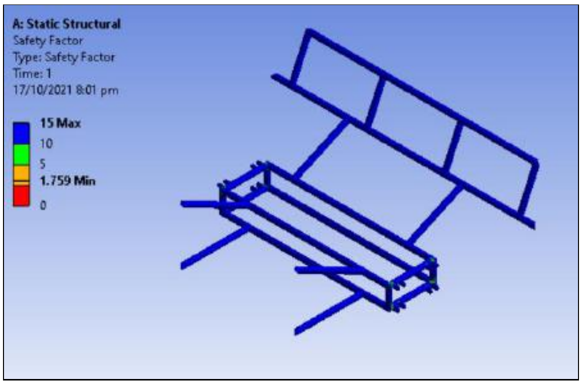


Fig 6- FoS

Maximum deformation	**0.465 mm**
Maximum stress	**264.51 MPa**
Factor of safety (fatigue)	**1.759**

d) Optimizations:

The FOS is good enough, thus no optimization was required.

4.5.4 TRANSMISSION SHAFT

a) Assumption & Considerations:

The end faces of transmission shaft are fixed, weights of the sprockets and shaft are considered as Loads in vertical-direction and given at their respective positions. Torque produced at sprockets is considered and given at their respective positions of sprockets. CAE analysis was done on whole shaft assemble (which include sprockets, ball bearings, grub screws and shaft).

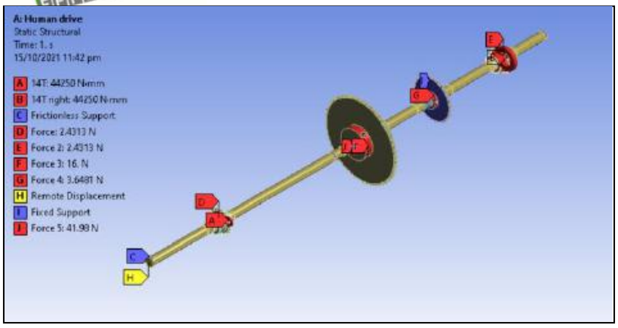


Fig 1- Boundary conditions(human drive)

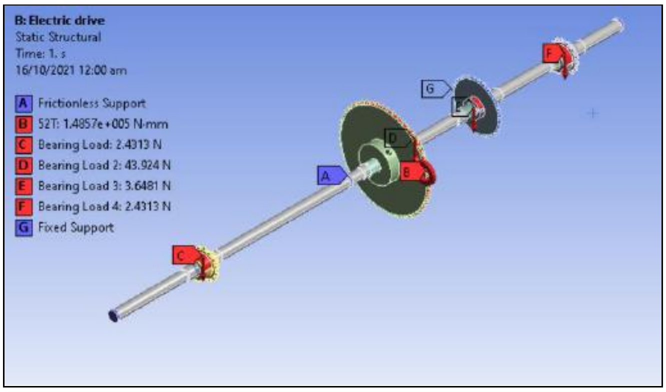


Fig 2- Boundary conditions (electric drive)

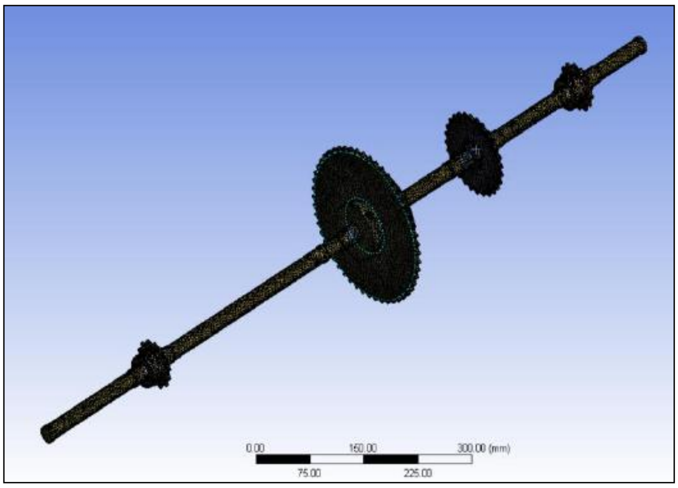


Fig 3- Meshed view

b) Calculation of Forces:

. Consider pedal force = 750N

Torque at crankset = 750 x 0.177

= 132.75 N m

Gear ratio = $\frac{42}{14}$ =3

$$\frac{T1}{T2} = \frac{\mathrm{T1}}{\mathrm{T2}}$$

42/14 = 132.75/ T_2

T_2 = 44.25 Nm

T_{per} = Syt/ 2fos = 415/2(6.5)

$T/J = T_{per}/r$

$J = \pi d^4/ 32$

$T /(\pi d^4/ 32) = T_{per}/(d/2)$

$$d^3 = \frac{2\ x\ T\ \ x\ 32}{\pi\ x\ \mathrm{Tper}}$$

$$d = \sqrt[3]{\frac{2\ x\ T\ \ x\ 32}{\pi\ x\ \mathrm{Tper}}}$$

$$d = \sqrt[3]{\frac{1000\ x\ T\ x2\ x\ 6.5x\ 32}{3.14\ x\ 415\ x\ 2}}$$

d = 19.18

d≈ 20 mm

c) Analysis Results:

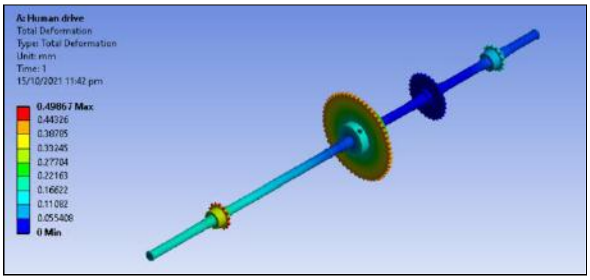


Fig 4- Total deformation (human drive)

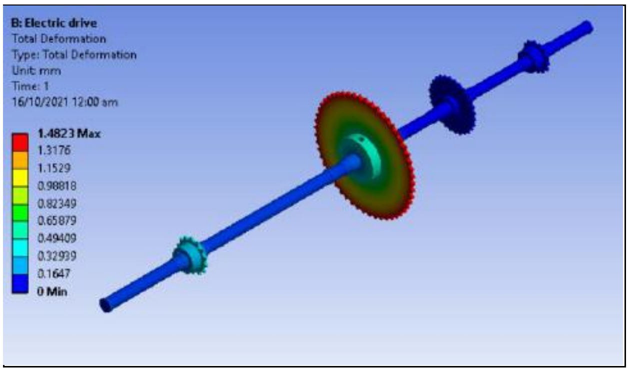


Fig 5- Total deformation (electric drive)

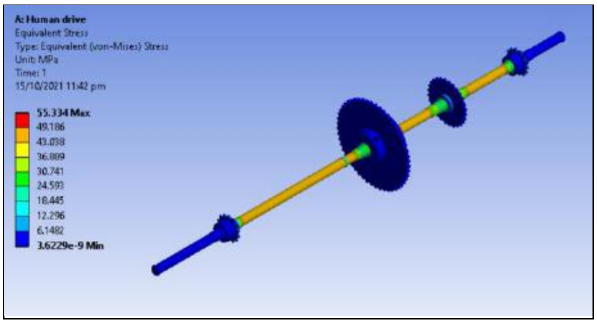


Fig 6- Equivalent stress (human drive)

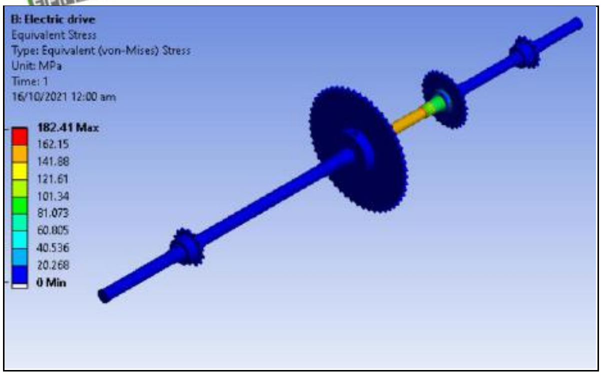


Fig 7- Equivalent stress (electric drive)

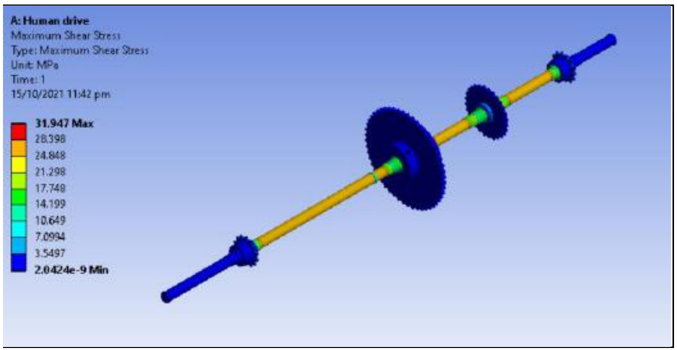


Fig 8- Shear stress (human drive)

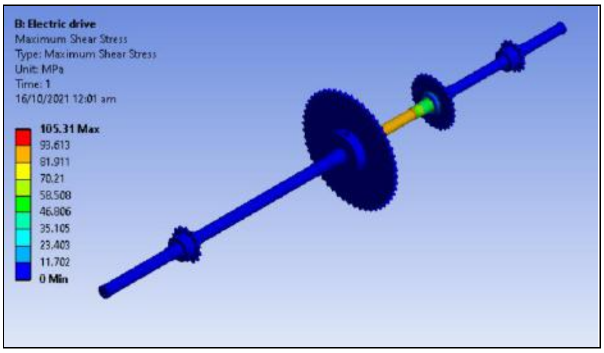


Fig 9- Shear stress (electric drive)

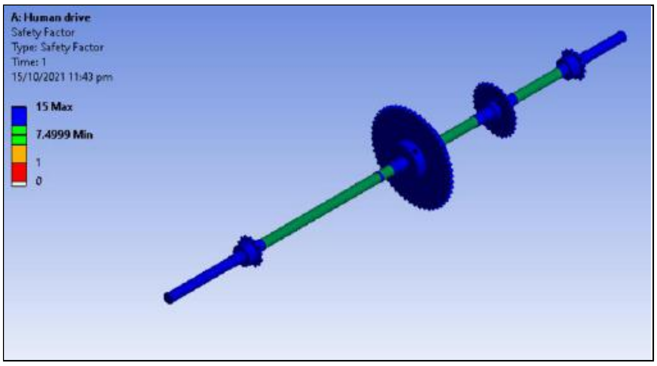


Fig 10- Static FOS (human drive)

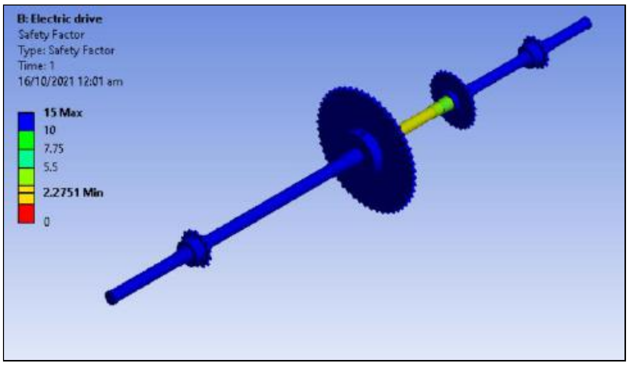


Fig 11- Static FOS (electric drive)

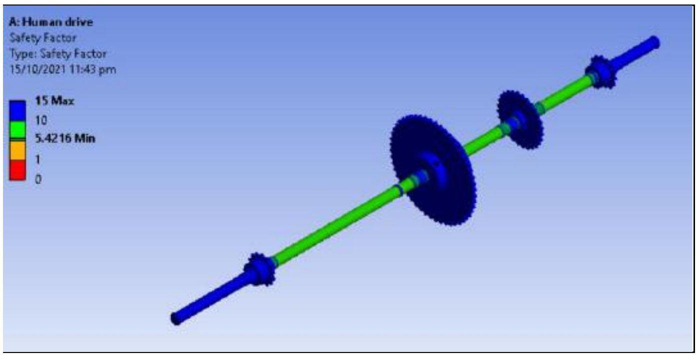


Fig 12- Fatigue FOS (human drive)

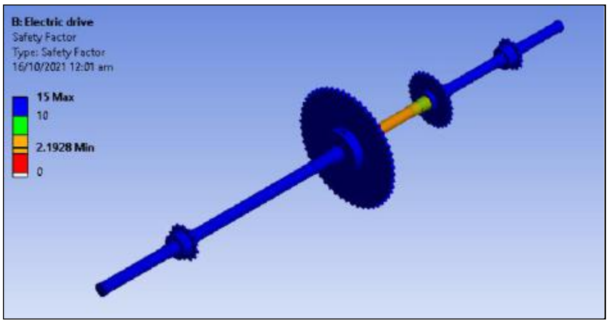


Fig 13- Fatigue, FOS (electric drive)

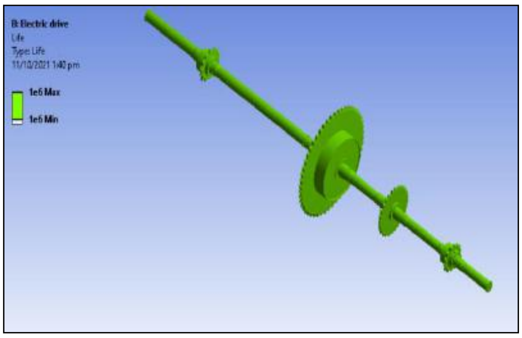


Fig 14- Minimum fatigue life

HUMAN DRIVE

Maximum deformation	**1.125mm**
Maximum stress	**125MPa**
Factor of safety(static)	**3.19**
Factor of safety (fatigue)	**4.43**
Minimum fatigue life	**Infinite life**

Maximum shear stress	**72.20MPa**

ELECTRIC DRIVE

Maximum deformation	**0.99mm**
Maximum stress	**122.68MPa**
Factor of safety(static)	**4.52**
Factor of safety (fatigue)	**3.26**
Minimum fatigue life	**Infinite life**

Maximum shear stress	**70.82MPa**

d)Optimizations:

The FOS is good enough, thus no optimization was required.

4.6 HARD POINT ANALYSIS

4.6.1 SEAT SUPPORT MEMBERS

a) Assumption & Considerations:

Base of the chassis is fixed, its assumed that weight of each driver is 115kg and load is applied on contact of seat and the chassis. But we performed analysis under more load.

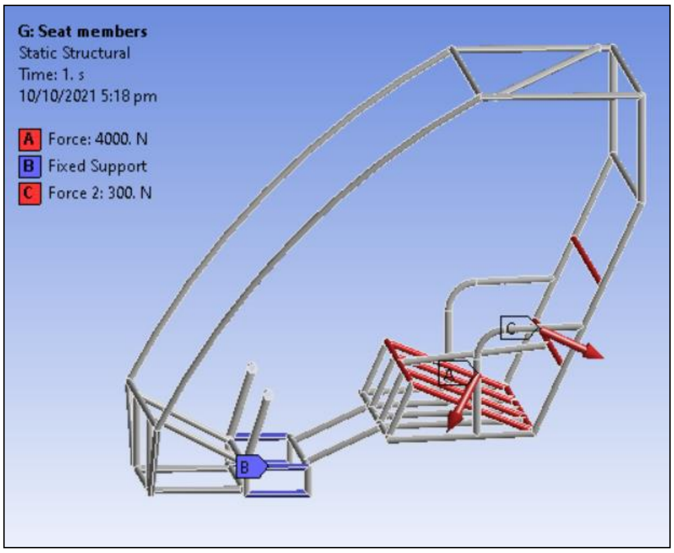


Fig 1- Boundary conditions

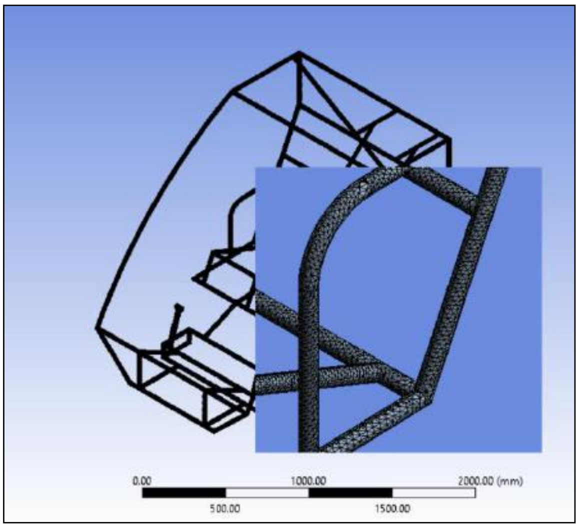


Fig 2- Meshed view

b) Calculation of Forces:

Weight of each driver is considered as 115kg and reaction force of 4000N is considered and 300N.

Reaction force = 230x10=2300N

And another force of 300N is considered when drivers are in rest position.

c) Analysis Results:

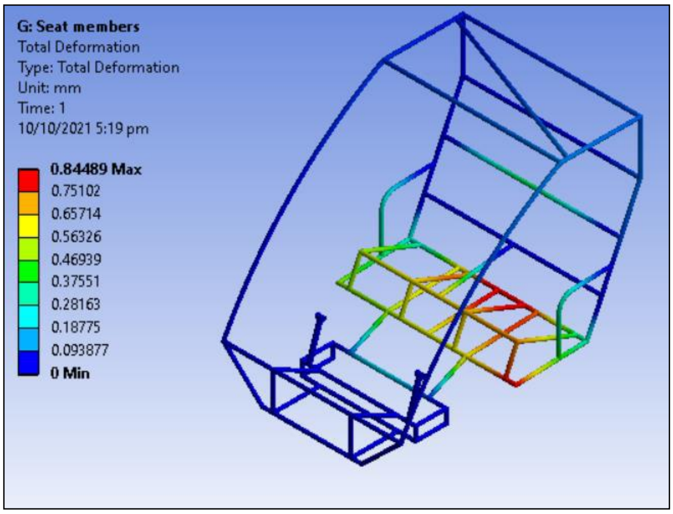


Fig 3- Total deformation

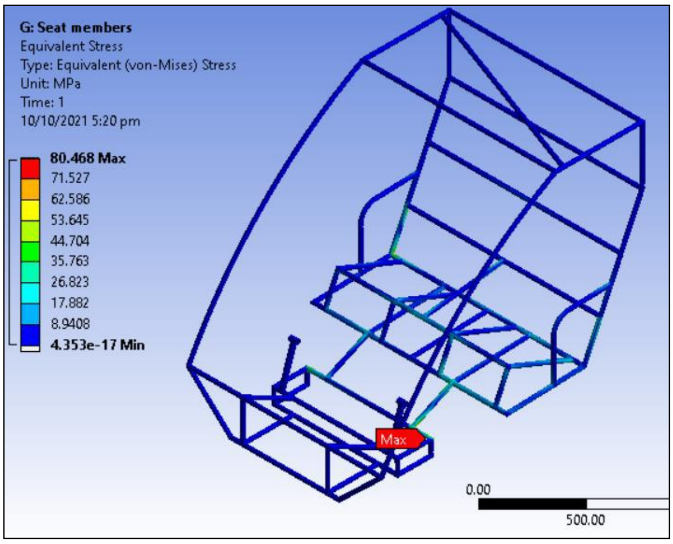


Fig 4- Equivalent stress

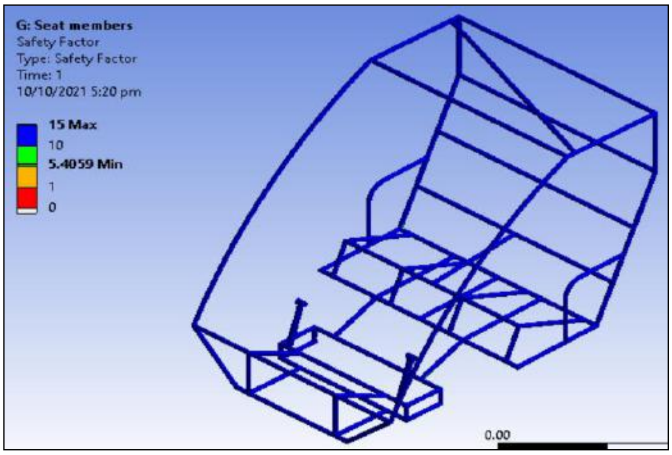


Fig 5- FOS (Static)

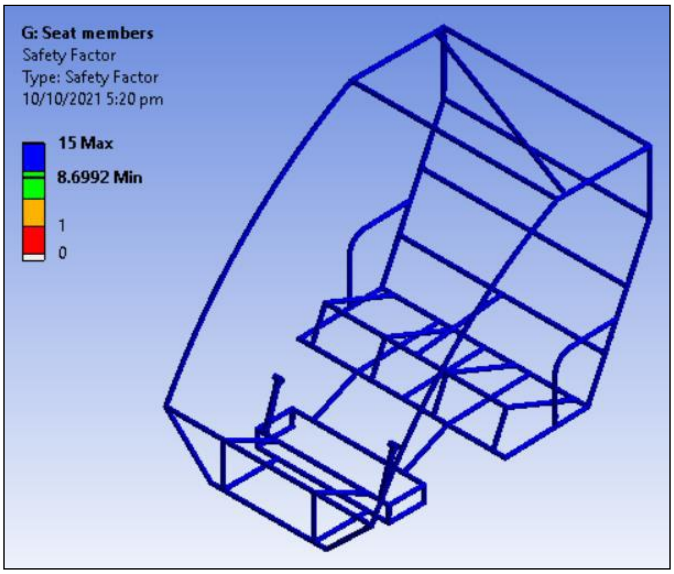


Fig 6- FOS (fatigue)

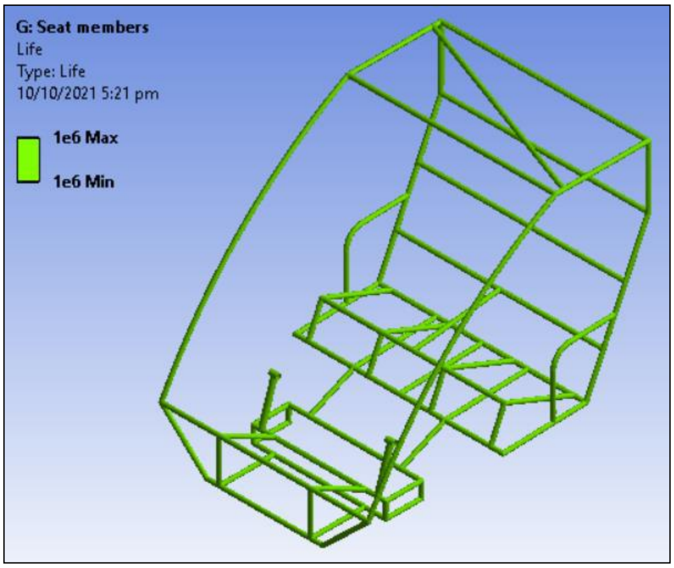


Fig 7- Minimum fatigue life

Maximum deformation	**0.84mm**
Maximum stress	**80.46MPa**
Factor of safety(static)	**8**
Factor of safety (fatigue)	**5.4**
Minimum fatigue life	**Infinite life**

d) Optimizations:

The FOS is good enough, thus no optimization was required.

4.6.2. FRONT SUSPENSION

a) Assumption & Considerations:

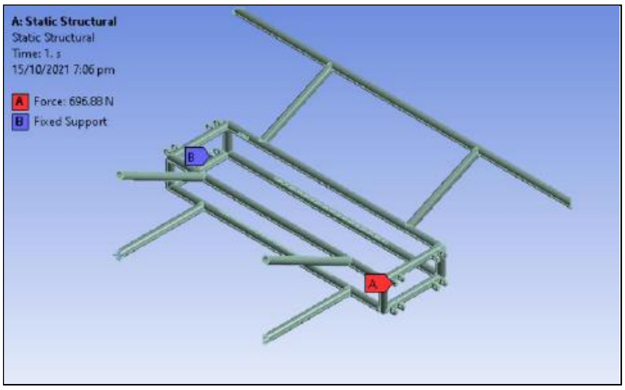


Fig 1- Boundary conditions

b) Calculation of forces

The weight of the vehicle along with the passengers is assumed to be 270 kgs. According to our weight distribution of 46.6 : 53.4, the weight on the front axle is 125.82 kgs. This is divided into tw parts because of two wheels. So, the weight on one wheel is 62.91 kgs.

Reaction force on each wheel is 617.14 N.

c) Analysis Results:

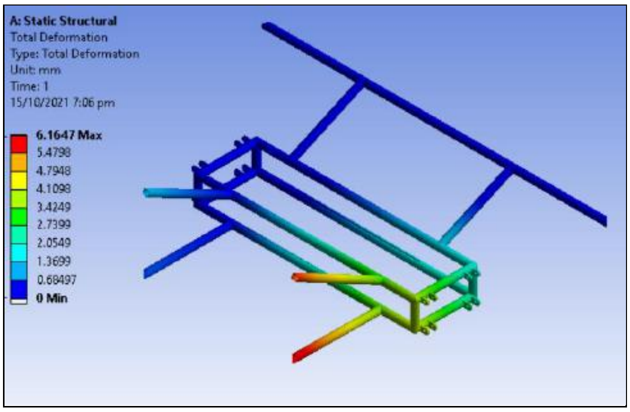


Fig 2- Deformation

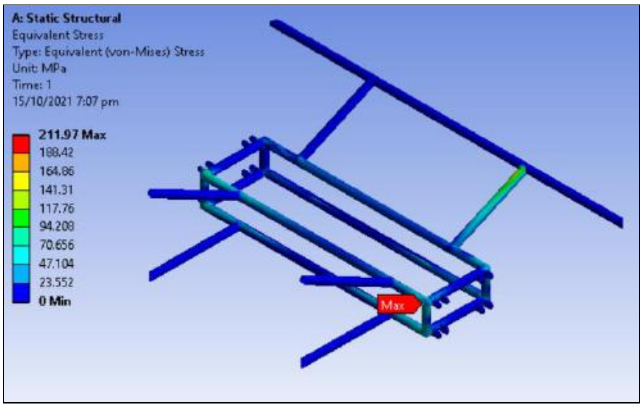


Fig 3- Maximum stress

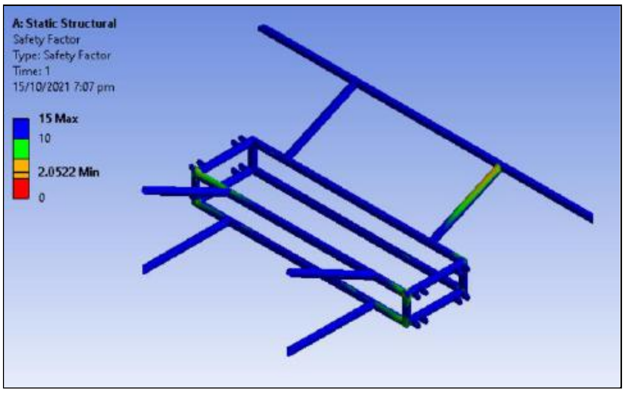


Fig 4- FOS

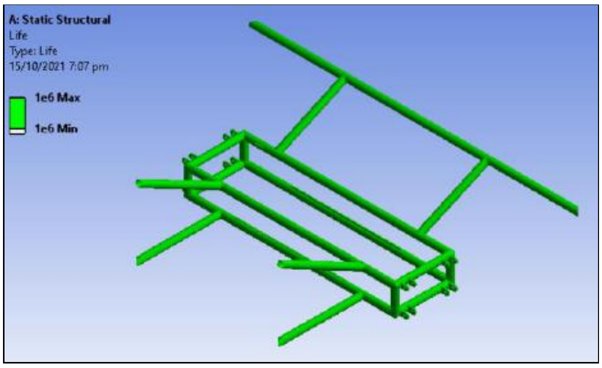


Fig 5- Fatigue life

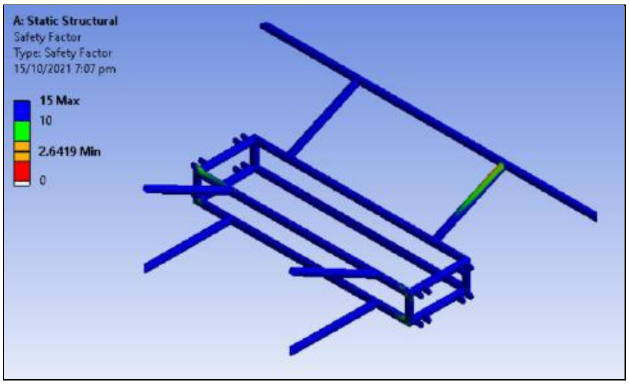


Fig 5- Fatigue FOS

4.6.3. MOUNTING TABS

a) Assumption & Considerations:

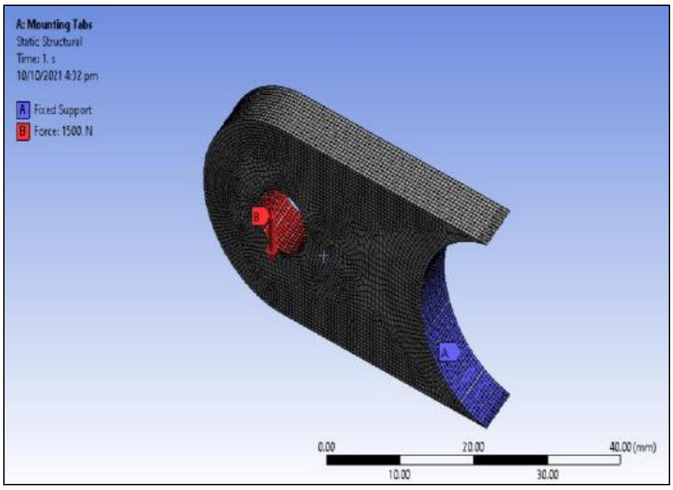


Fig 1- Boundary conditions and Meshed view

b) Calculation of Forces:

weight distribution of vehicle is 46.6:53.4

under critical conditions we performed analysis

weight of vehicle including drivers=340

The reaction force of 1500 is considered.

Reaction force = 340x (46.4/2) x100x10

~ 800N

c) Analysis Results:

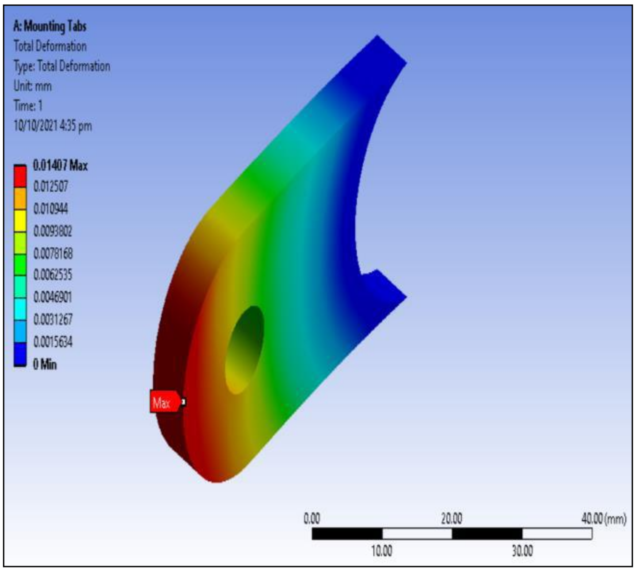


Fig 2- Total deformation

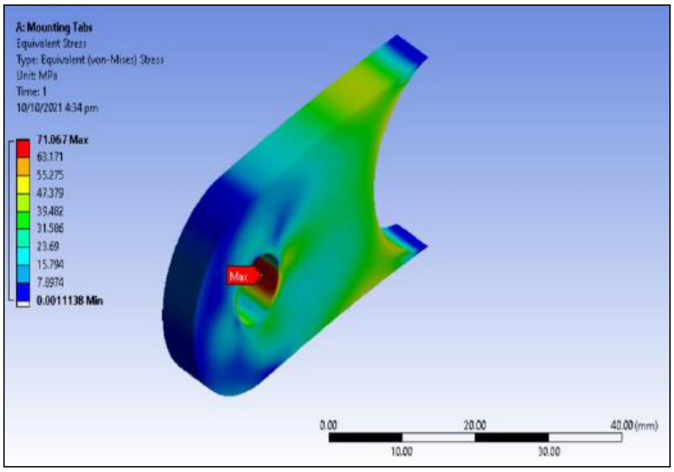


Fig 3- Equivalent stress

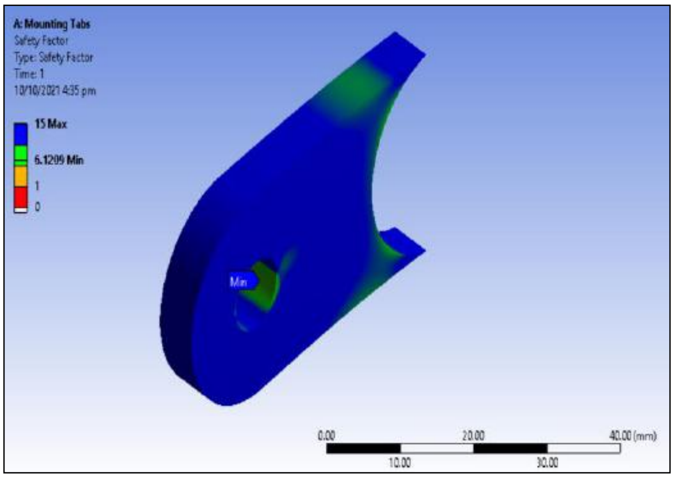


Fig 4- FOS(static)

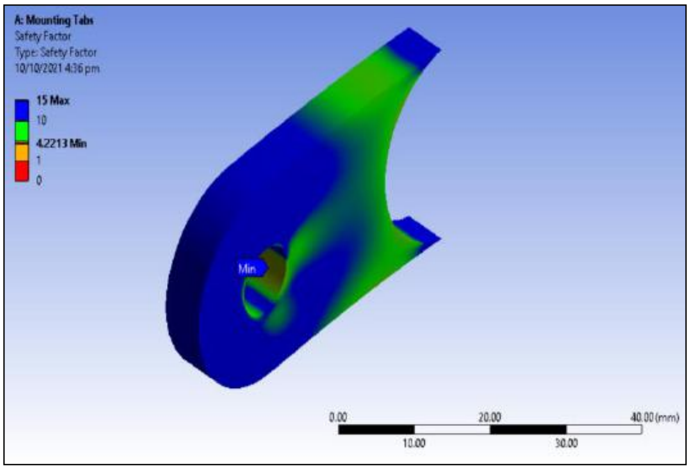


Fig 5- FOS (fatigue)

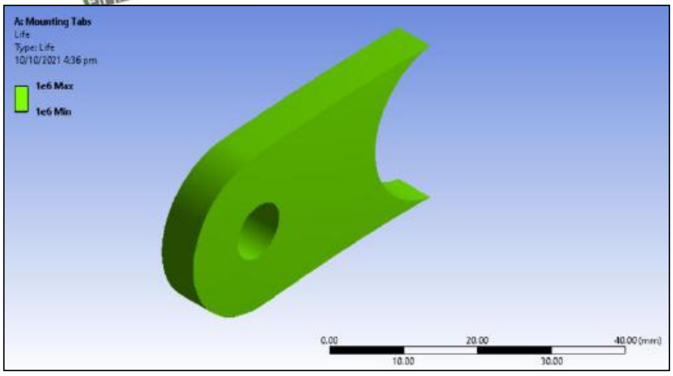


Fig 6- Minimum fatigue Life

Maximum deformation	**0.014mm**
Maximum stress	**71.06MPa**
Factor of safety(static)	**6.12**
Factor of safety (fatigue)	**4.22**
Minimum fatigue life	**Infinite life**

d)Optimizations:

The FOS is good enough, thus no optimization was required.

4.6.4 REAR SUSPENSION BRACING

a) Assumption & Considerations:

The reaction force of 8000N is considered on rear suspension bracing.

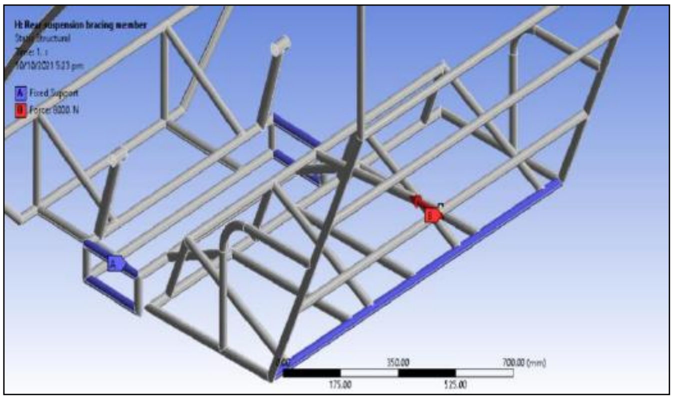


Fig 1- Boundary conditions

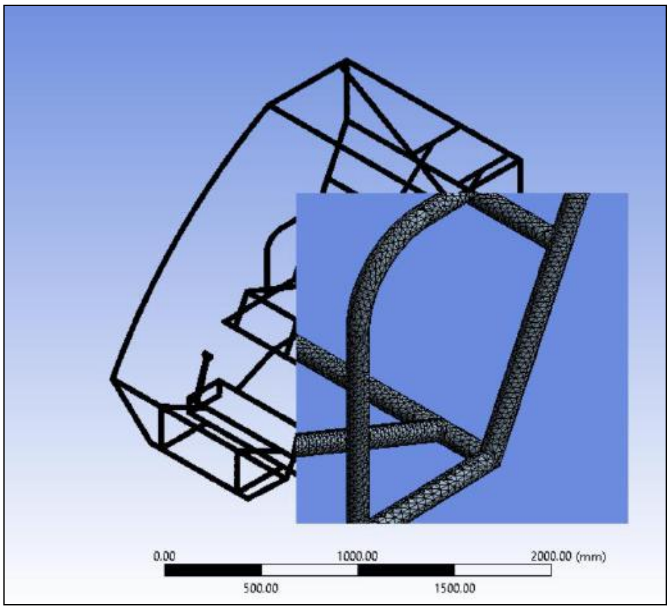


Fig 2- Meshed view

b) Calculation of Forces:

The reaction force of 8000N is considered on rear suspension bracing. (under the critical conditions)

Considering two people weighing 115 kgs to be sitting on the seat, weighing 3 kgs each and a motor with its mounting to be 10 kgs, we get a total weight of 246kgs. Which is 2413.26 N of force.

But we have taken a maximum force of 8000N to be acting on the seat members for extreme conditions.

c) Analysis Results:

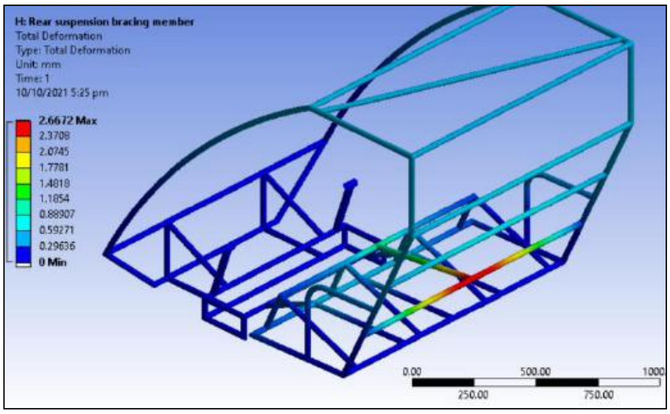


Fig 3- Total deformation

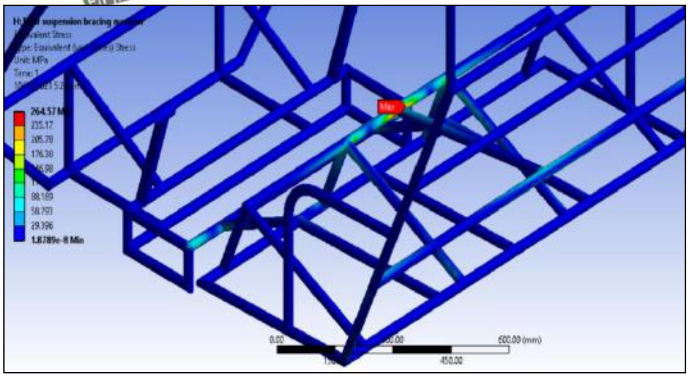


Fig 4- Equivalent stress

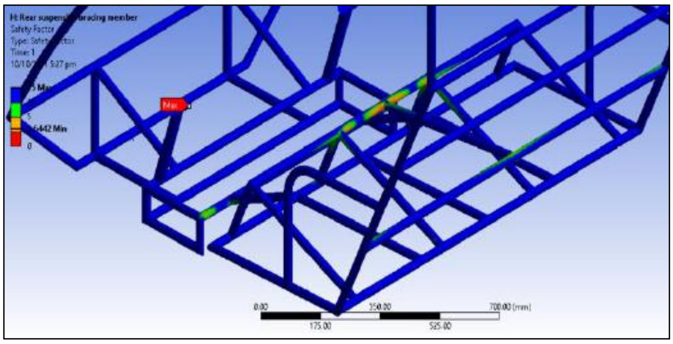


Fig 5- FOS (static)

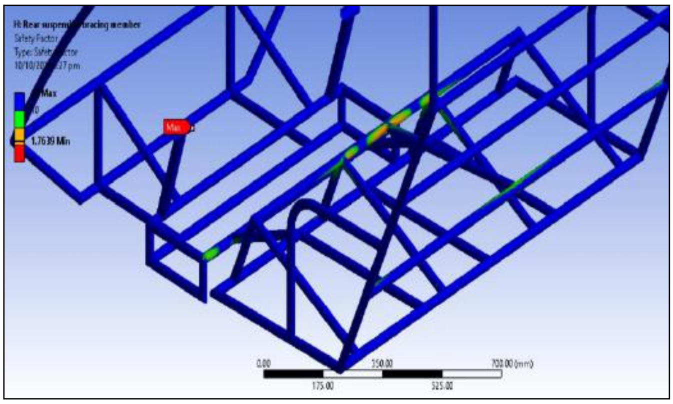


Fig 6- FOS (fatigue)

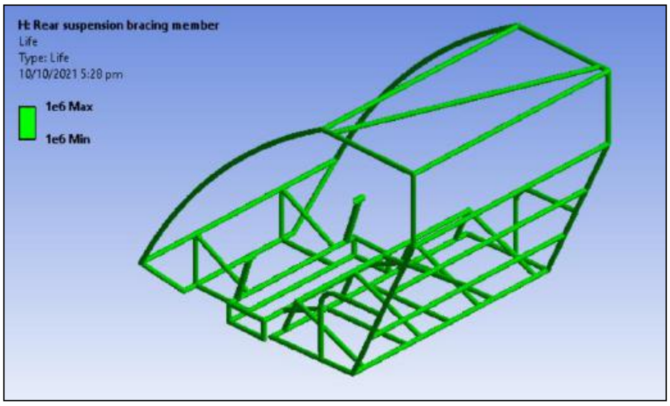


Fig 7- Minimum fatigue life

Maximum deformation	**2.66mm**
Maximum stress	**264.57MPa**
Factor of safety(static)	**1.64**
Factor of safety (fatigue)	**1.76**
Minimum fatigue life	**Infinite life**

d) Optimizations:

The FOS is good enough, thus no optimization was required.

5. CAE ANALYSIS OF OTHER PARTS

5.1 CONTROL ARMS

a) Assumption & Considerations:

The faces mounting to the chassis fixed, a force of 629.1 N is applied on eyeball joint contact face, a reaction force of 1385.99N is also considered and was applied on spring mounting tabs.

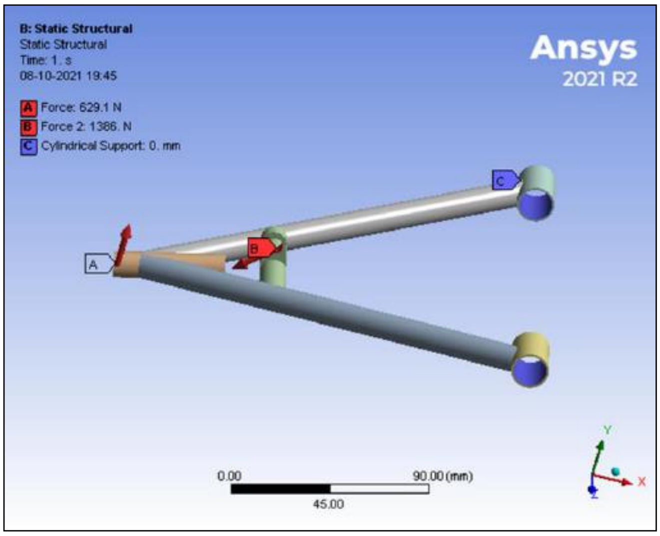


Fig 1- Boundary conditions

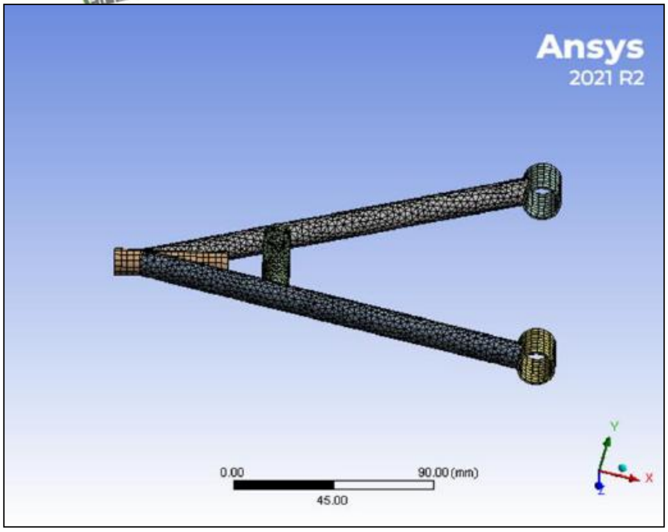
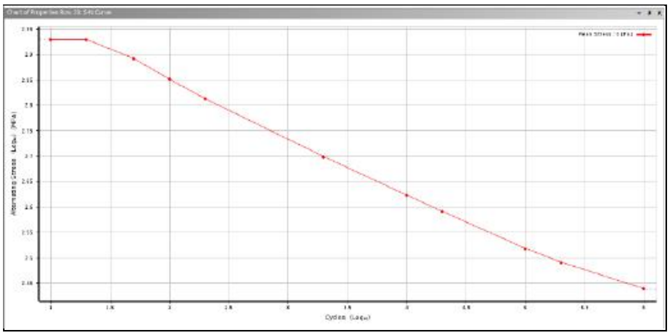


Fig 2- Meshed view

SN Curve for AISI 1018:

	B	C
1	Cycles	Alternating Stress (MPa)
2	10	850
3	20	850
4	50	780
5	100	710
6	200	650
7	2000	500
8	10000	420
9	20000	390
10	1E+05	330
11	2E+05	310
12	1E+06	275



b) Calculation of Forces:

The faces mounting to the chassis fixed, a force of 629.1 N is applied on eyeball joint contact face, a reaction force of 1385.99N is also considered and was applied on spring mounting tabs. Load distribution of our vehicle is 46.6:53.4. So, on front wheels a net load of 125.82 kg is found so on each front wheel net load is 62.91 kgs. Therefore, the reaction force on the ground on the wheel =629.1N=R_N

The force on the spring be F_s

Now taking the moments R_N X Arm length = F_{SV} x distance of the spring from the chassis.

[F_{SV} = vertical component of spring]

629.1 x 22.19 = F_{SV} x sin θ x 15.0114

F_S = 1385.99N

c) Analysis Results:

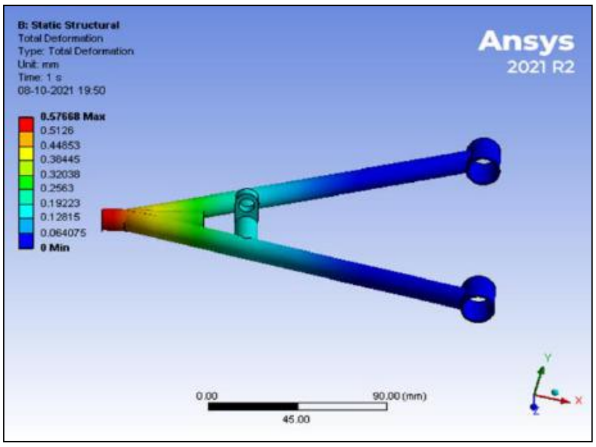


Fig 3- Total deformation

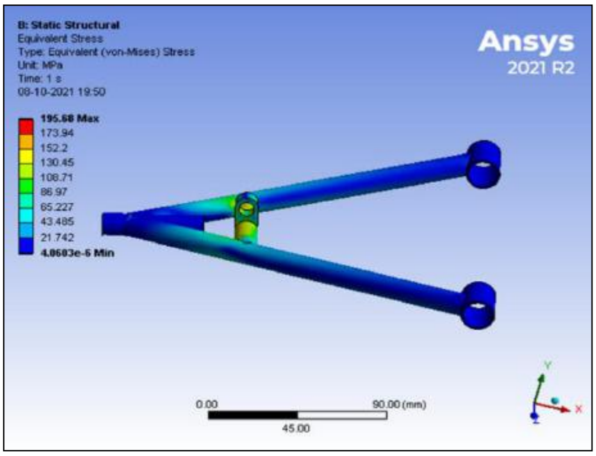


Fig 4- Equivalent stress

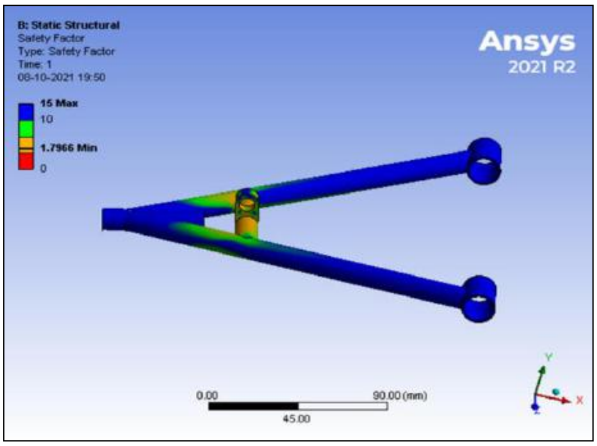


Fig 5- FOS (static)

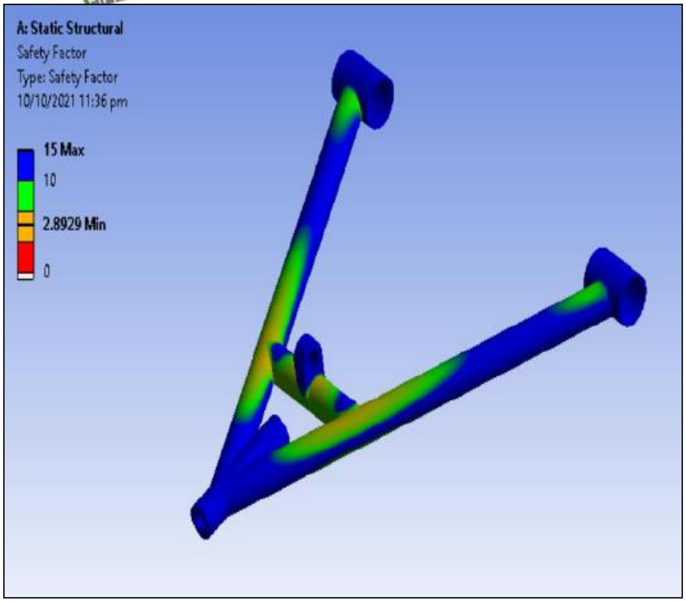


Fig 6- FOS (fatigue)

Maximum deformation	**0.57mm**
Maximum stress	**195.68MPa**
Factor of safety(static)	**1.79**
Factor of safety (fatigue)	**2.89**

d) Optimizations:

The FOS is good enough, thus no optimization was required.

5.2 PULLOUT FORCE OF EYE BALL JOINT

a) Assumption & Considerations:

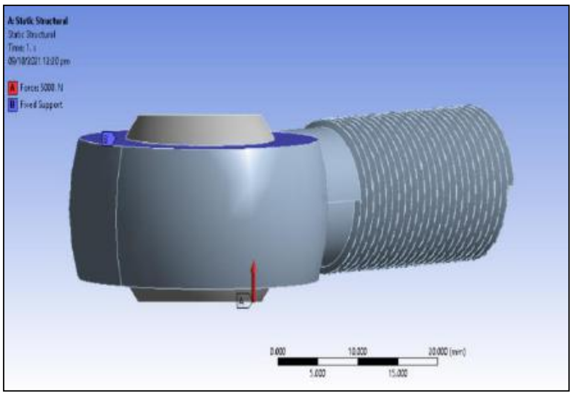


Fig 1- Boundary conditions

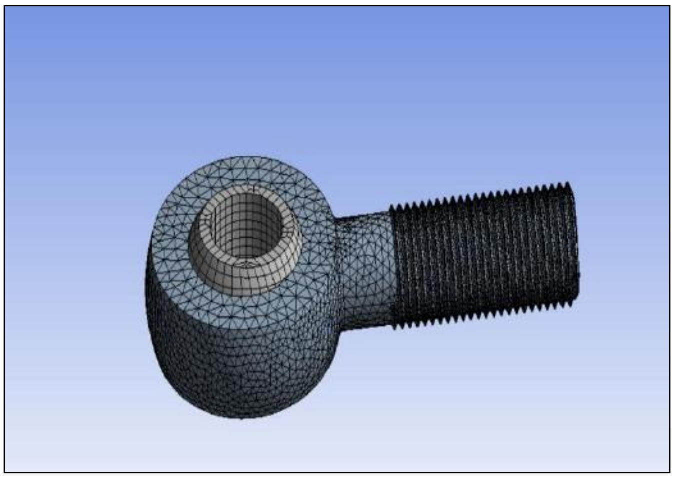


Fig 2- Meshed view

b) Calculation of Forces:

The upward reaction force of 5000N is considered (analysis performed under worst circumstances). 5400N was supposed to be the maximum loading under dynamic condition according to the manufacturer recommendation.

c) Analysis Results:

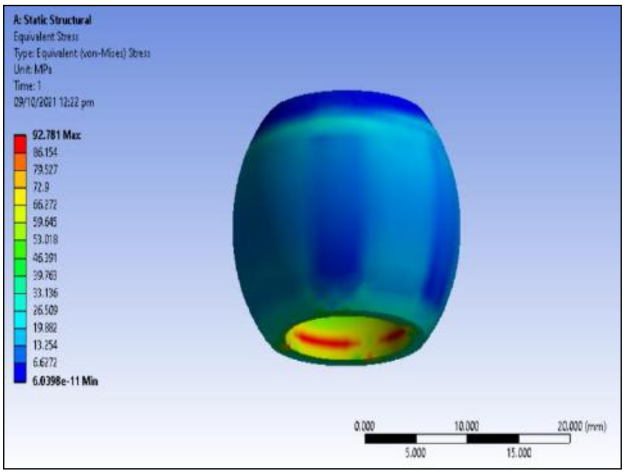


Fig 3- Equivalent stress

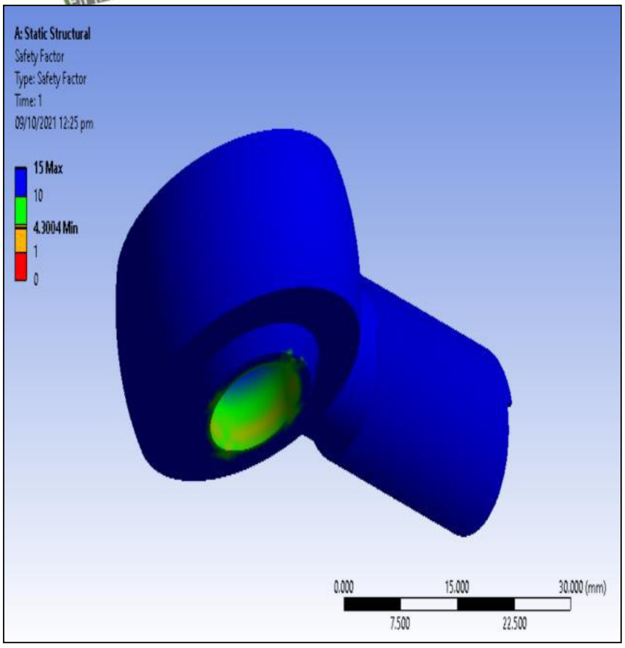


Fig 4- FOS (static)

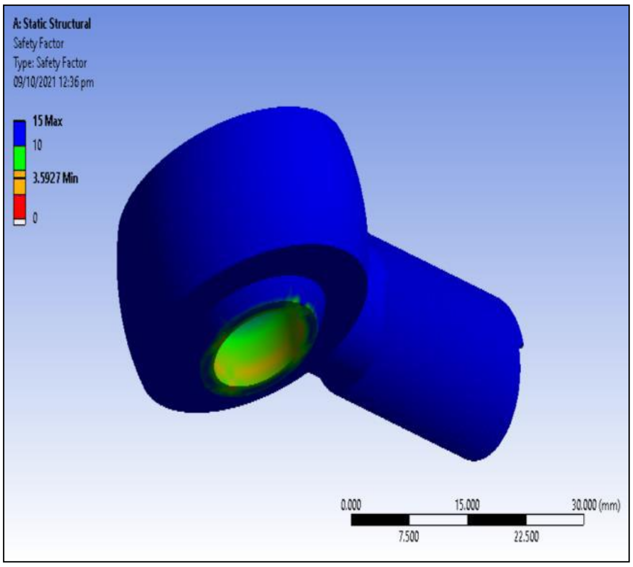


Fig 5- FOS (fatigue)

Maximum stress	**92.78MPa**
Factor of safety (static)	**4.3**
Factor of safety (fatigue)	**3.59**

d) Optimizations:

The FOS is good enough, thus no optimization was required.

5.3 BRAKE DISC (STATIC AND THERMAL ANALYSIS)

a) Assumption & Considerations:

The mounting points on brake disc are fixed, a moment of 118.56Nm is given to the disc, and force of 1482N is given on contact face between brake shoe and disc.

A temperature of 80 ^{0}C is given on the contact face between brake disc and brake shoe.

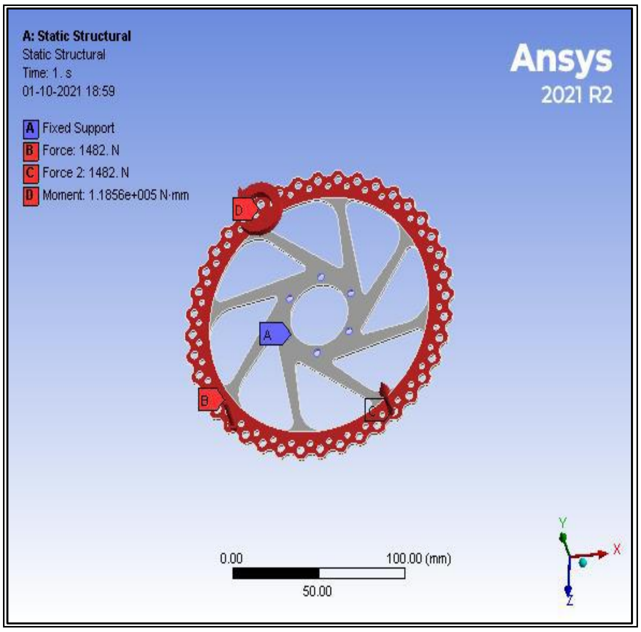


Fig 1- Boundary conditions (Static)

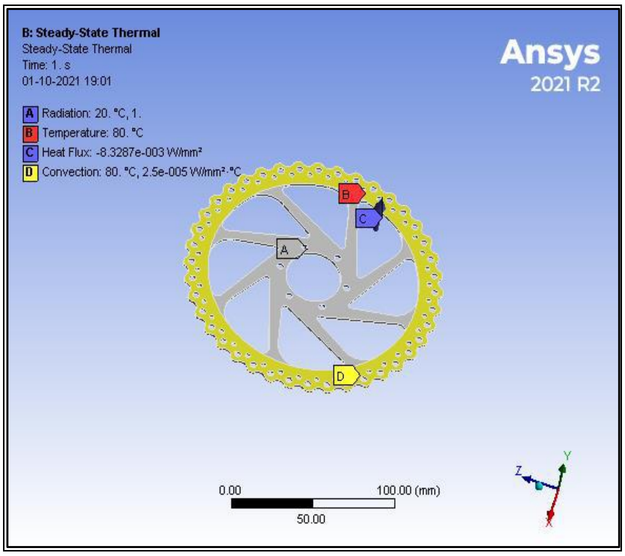


Fig 2- Boundary conditions (thermal)

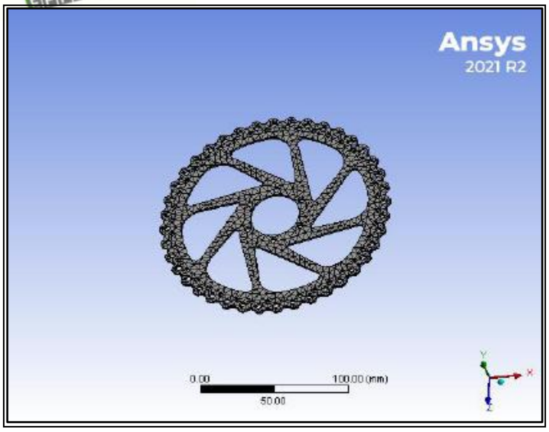


Fig 3- Meshed view

b) Calculation of Forces:

Thermal conductivity of brake disc = **30W/mK** Thermal conductivity of air = 50W/mK
Film coefficient of brake disc = 1/ (1/thermal coefficient of air) + (thickness of brake disc/thermal coefficient of brake disc) + (1/thermal coefficient of air) = 1/ (1/50) + $(2\times10^{-3}/30)$ + (1/50)
= 25W/m^{2}K
Heat flux = film coefficient * change in temperature
Change of temperature = 80°-20° = 60°C **= 333.15K**
Therefore, heat flux = 25x333.15**= 8328.75W/m**
Force exerted by brake pads on the rotor**1482N**
Torque on brake disc = $n\mu F_{cr}$ = **118.56Nm**

c) Analysis Results:

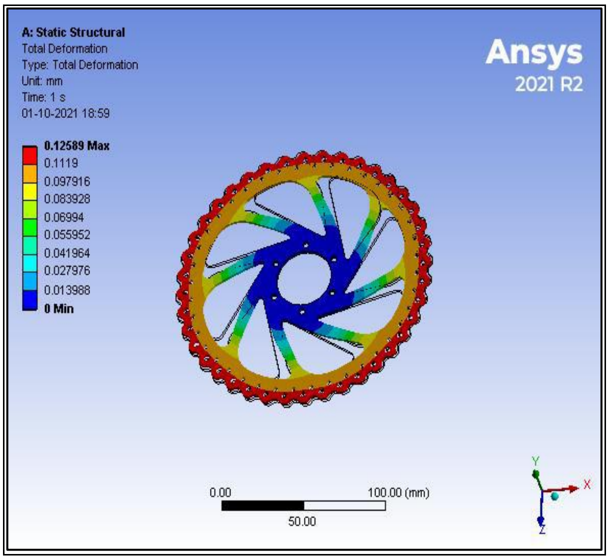


Fig 4- Total deformation

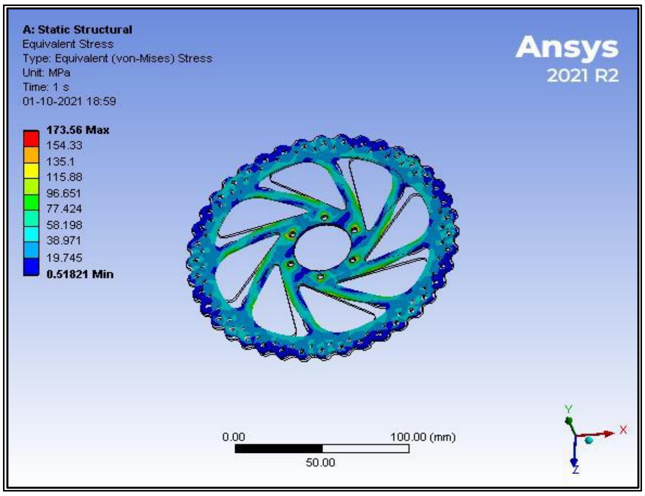


Fig 5- Equivalent stress

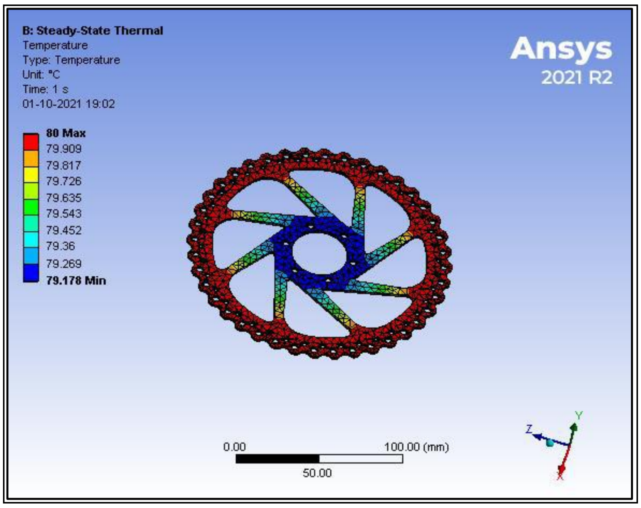


Fig 6- Temperature

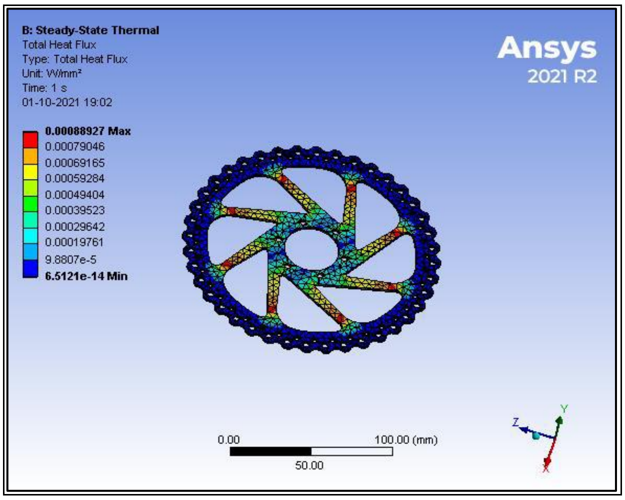


Fig 7- heat flux

Maximum deformation	**0.12mm**
Maximum stress	**173MPa**
Temperature	**80^{0}c**
Maximum heat flux	**0.0008 w/m**

d) Optimizations:

The mounting points on brake disc are fixed, a moment of 118.56Nm is given to the disc, and force of 1482N is given on contact face between brake shoe and disc.

Force exerted by brake pads on the rotor = **1482N**

Torque on brake disc = $N\mu F_{cr}$

= **118.56Nm**

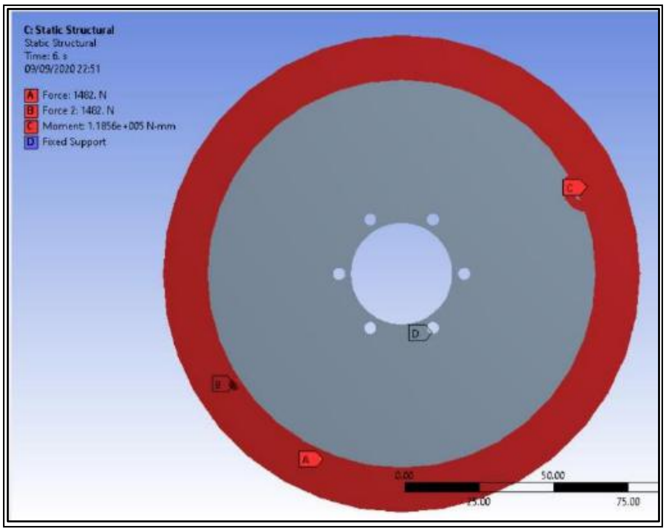


Fig 8- Boundary conditions

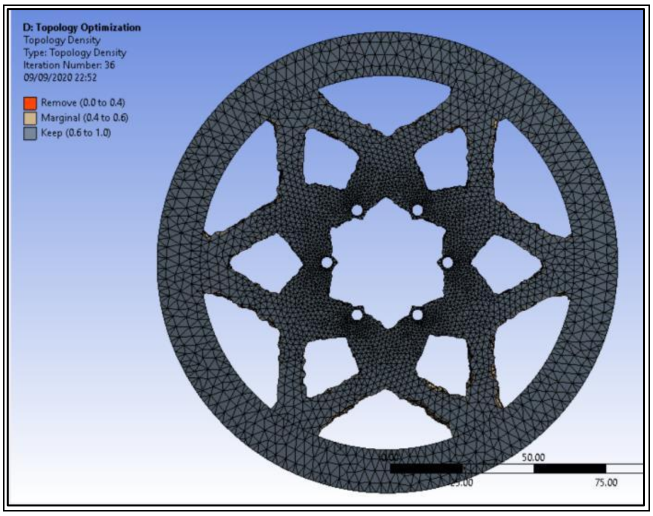


Fig 9- Topology density

To reduce the weight and increase rate of cooling, holes are added in the brake disc.

5.4 REAR TRIANGLE

a) Assumption & Considerations:

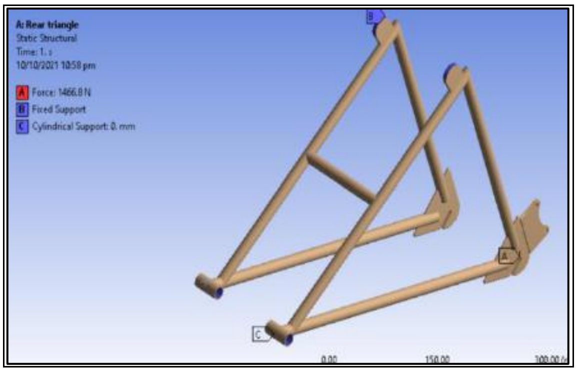


Fig 1- Boundary conditions

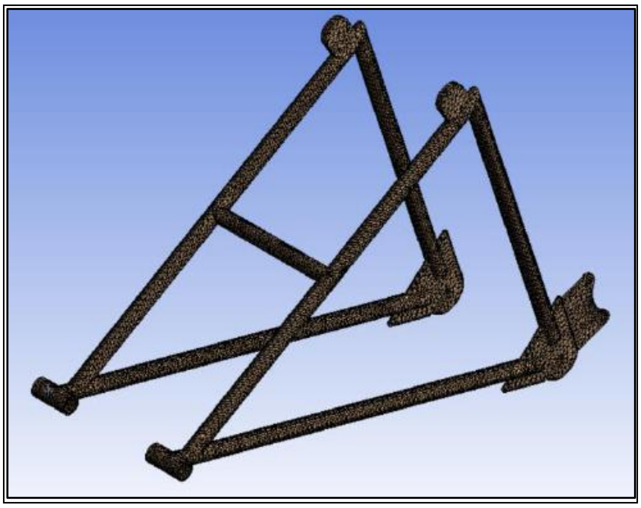


Fig 2- Meshed view

b) Calculation of Forces:

Weight distribution of vehicle 43.6:56.4

On rear triangle the reaction force is taken as 1466.8

Weight of the vehicle including drivers = 270

The reaction force = 270x (56.4/100) x 10

= **1466.8 N**

c) Analysis Results:

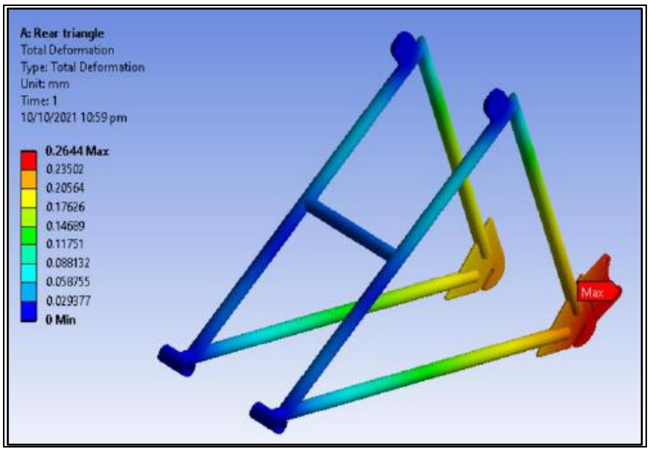


Fig 3- Total deformation

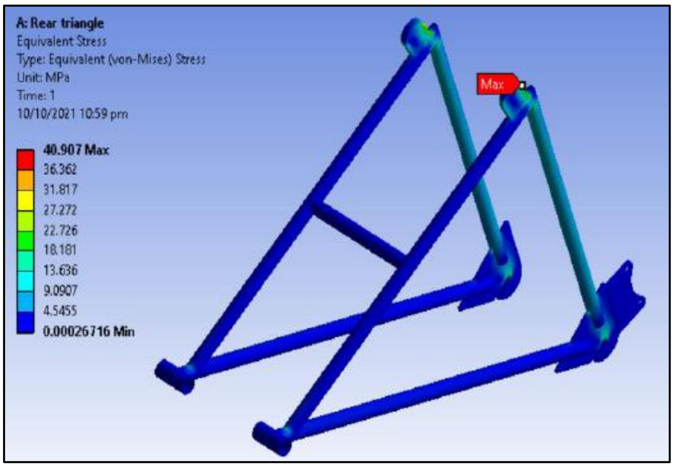


Fig 4- Equivalent stress

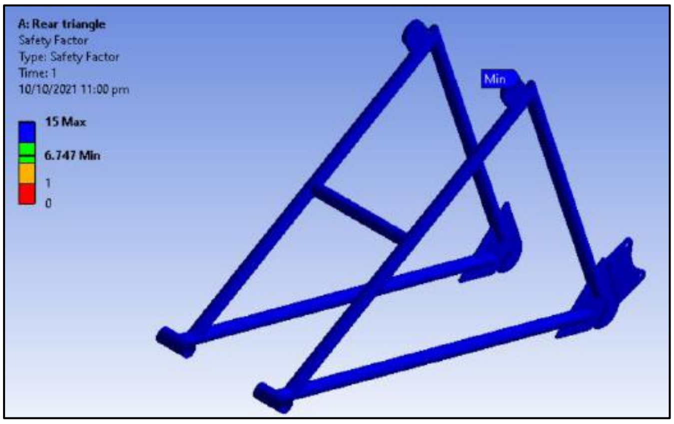


Fig 5- FOS (static)

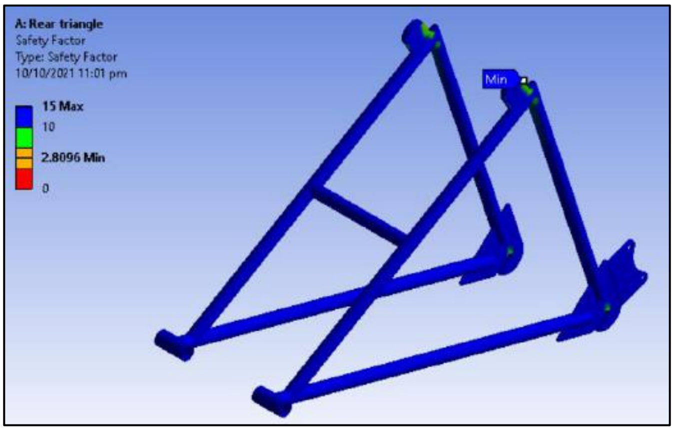


Fig 6- FOS (fatigue)

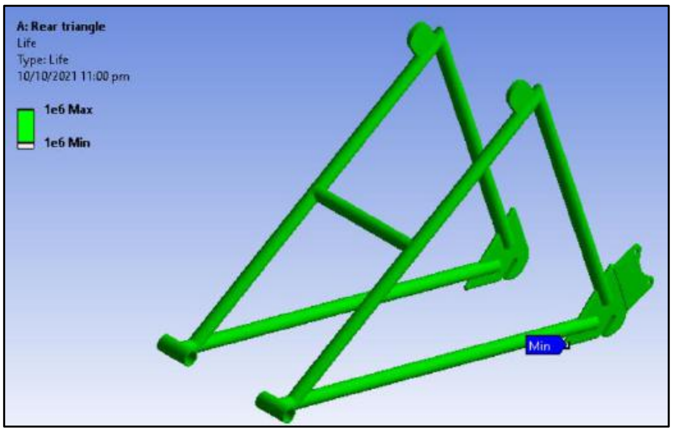


Fig 7- Minimum fatigue life

Maximum deformation	**0.26mm**
Maximum stress	**40.90MPa**
Factor of safety(static)	**2.8**
Factor of safety (fatigue)	**6.7**
Minimum fatigue life	**Infinite life**

d) Optimizations:

The FOS is good enough, thus no optimization was required.

5.4. STEERING COULUMN

a) Assumption & Considerations:

Bearing load of 120N is considered. Length of steering column is 0.049m.

Force is calculated by taking torque into consideration.

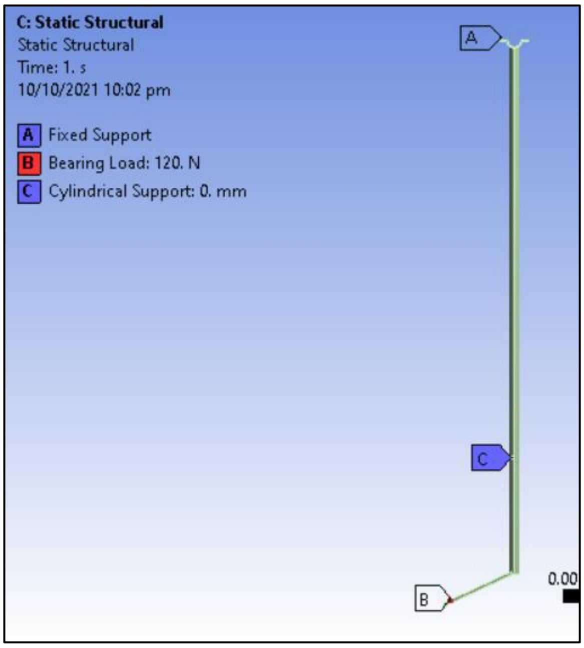


Fig 1- Boundary conditions

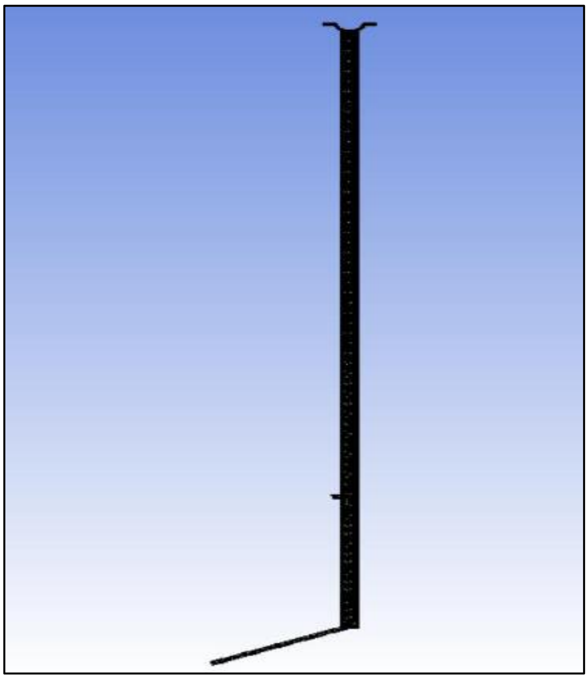


Fig 2- Meshed view

b) Calculation of Forces:

As the tie rods pushing the tire from one side and pulling it from another side

Total force = 322.8941+210.352=533.2461 N

Length of the steering column plate = 1.9645 in= 0.04989 m

Torque on steering column =Total force * perpendicular distance of the link between steering column and tie rod from the line joining kingpin center.

= 533.2461*1.9645*0.0254 Nm

=26.608 Nm

Torque on steering handle = torque on steering column =26.608 N-m

Length of steering handle (half length) = 9 in = 0.2205 m

Force applied by the driver to steer the wheels = Torque on steering handle/length of steering handle = 26.608/0.2205 = 120 N

After we get, Force applied by the driver to steer the wheels = 120 N

c) Analysis Results:

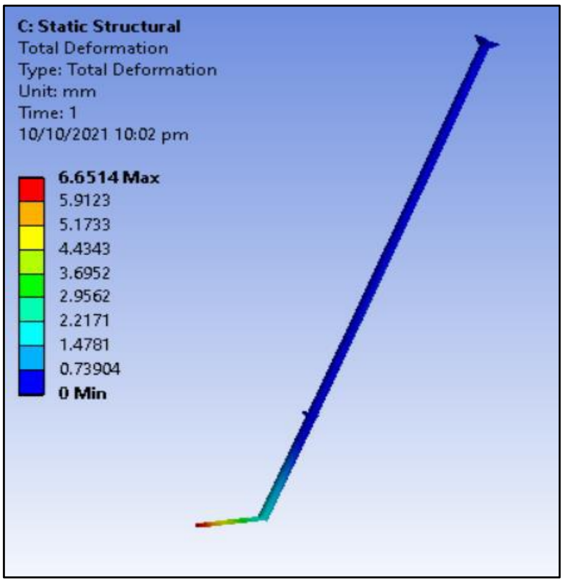


Fig 3- Total deformation

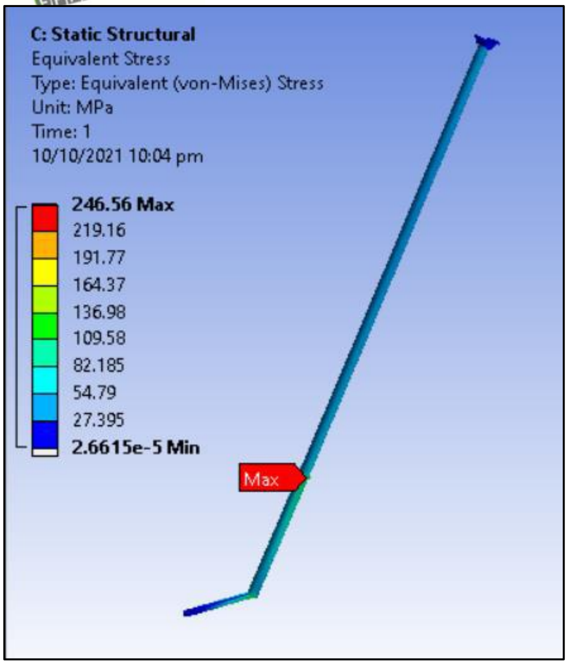


Fig 4- Equivalent stress

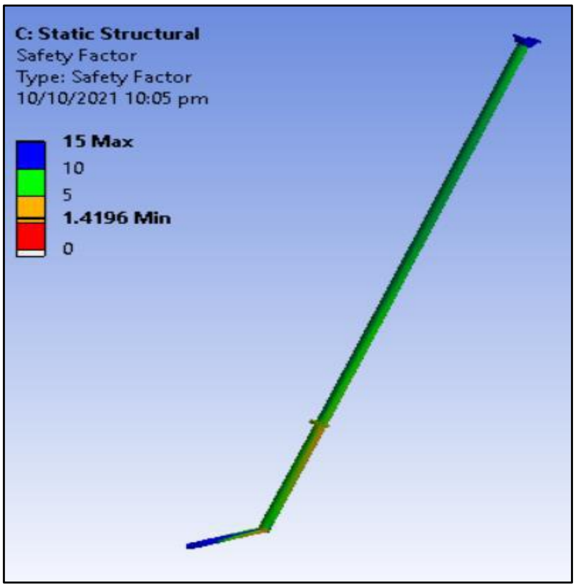


Fig 6- Fatigue

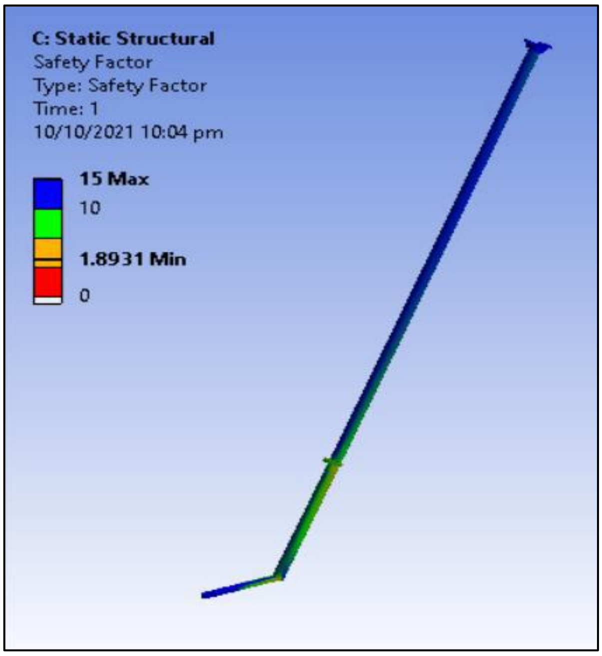


Fig 5- FOS (static)

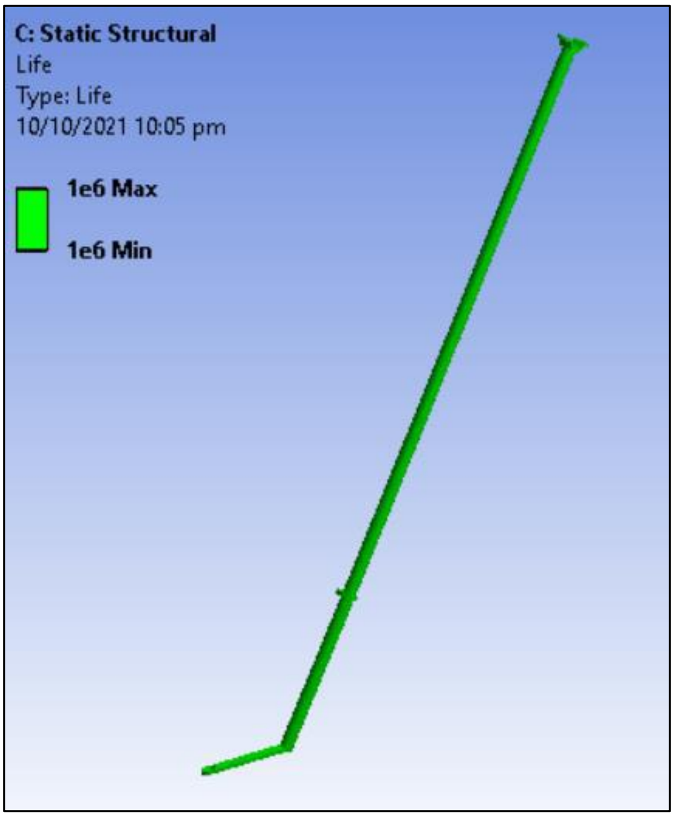


Fig 7- Minimum fatigue life

Maximum deformation	**6.6mm**
Maximum stress	**246.56MPa**
Factor of safety(static)	**1.83**
Factor of safety (fatigue)	**1.41**
Minimum fatigue life	**Infinite life**

d) **Optimizations:**

The FOS is good enough, thus no optimization was required

5.5 REAR TIE ROD

a) **Assumption & Considerations**:

The reaction force of 322.84 N is considered.

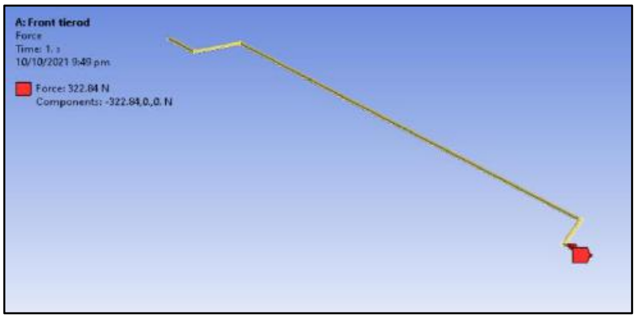


Fig 1- Boundary conditions

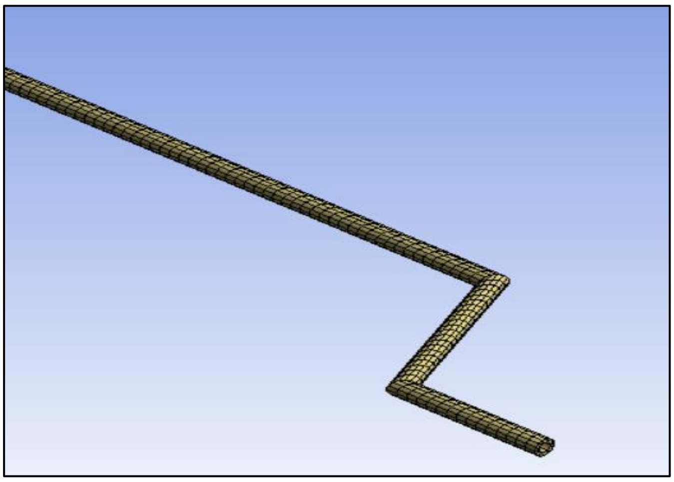


Fig 2- Meshed view

b) **Calculation of Forces:**

weight distribution of vehicle is 46.6: 53.4

the reaction force =322N

c) **Analysis Results:**

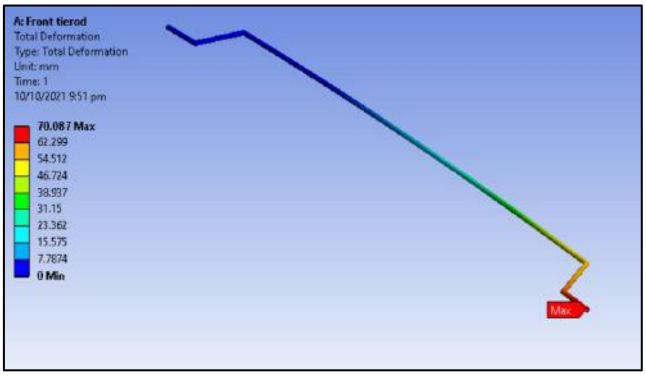


Fig 3- Total deformation

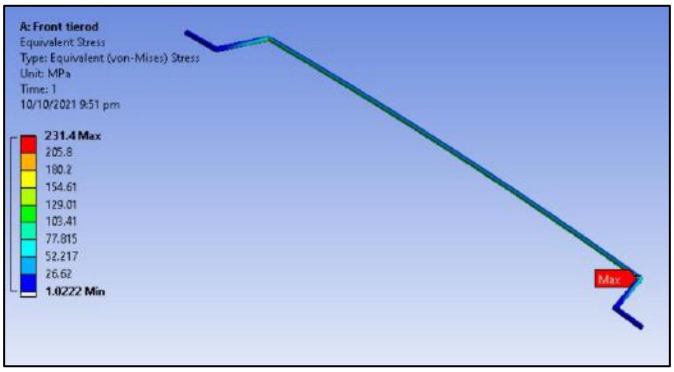


Fig 4- Equivalent stress

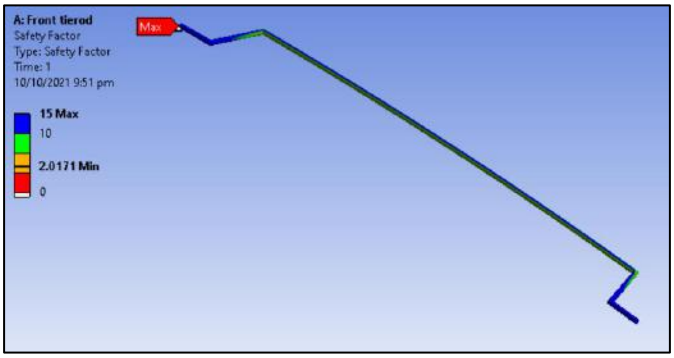


Fig 5- FOS (static)

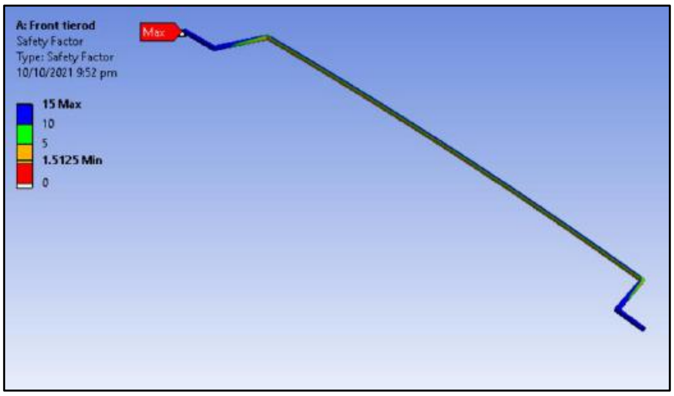


Fig 6- FOS (fatigue)

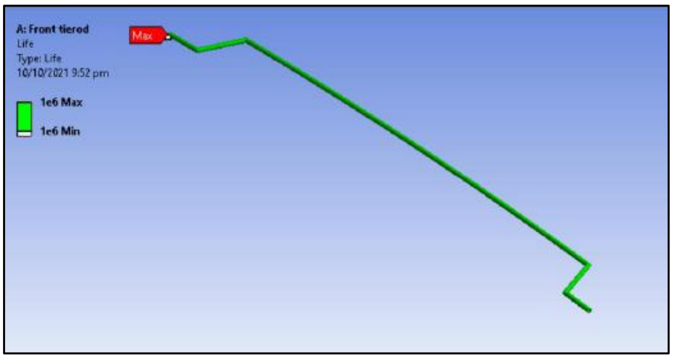


Fig 7- Minimum fatigue life

Maximum deformation	**70.08mm**
Maximum stress	**231.4MPa**
Factor of safety(static)	**1.51**
Factor of safety (fatigue)	**2.01**
Minimum fatigue life	**Infinite life**

d) Optimizations:

The FOS is good enough, thus no optimization was required

5.6 FRONT TIE ROD

a) Assumption & Considerations:

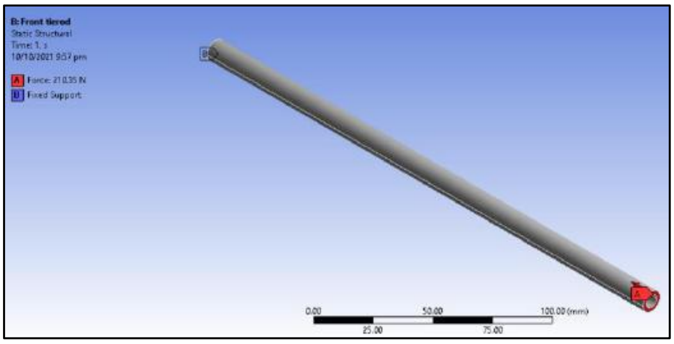


Fig 1- Boundary conditions

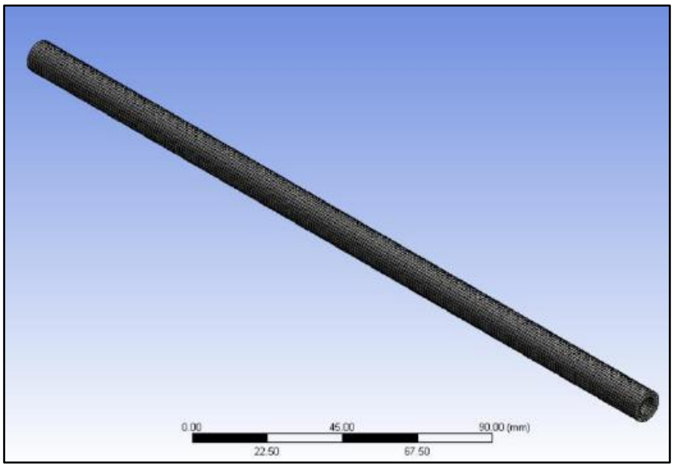


Fig 2- Meshed view

b) Calculation of Forces:

weight distribution of vehicle is 46.6: 53.4

the reaction force =210N

c) Analysis Results:

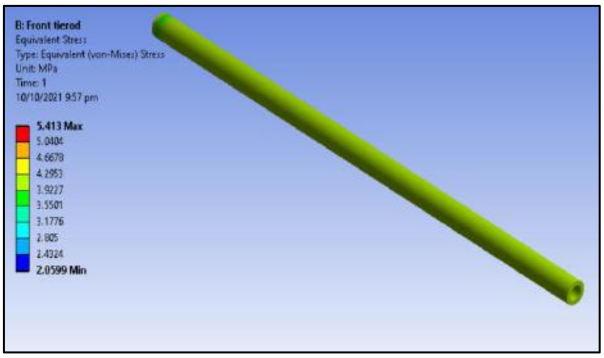


Fig 3- Equivalent stress

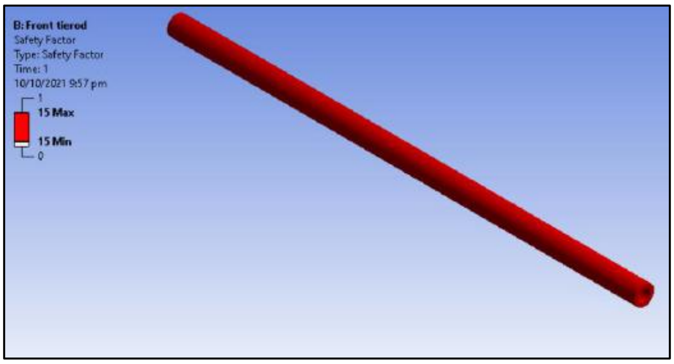


Fig 4- FOS (static)

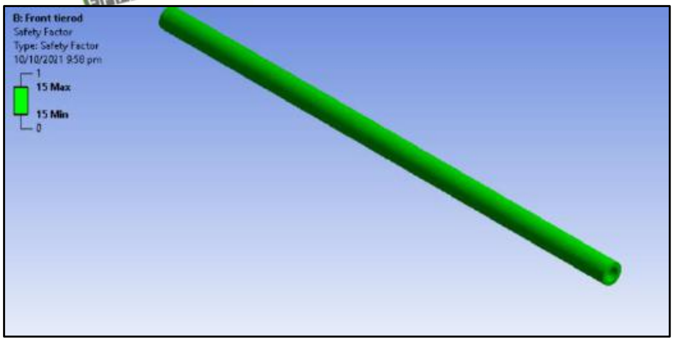


Fig 5- FOS (fatigue)

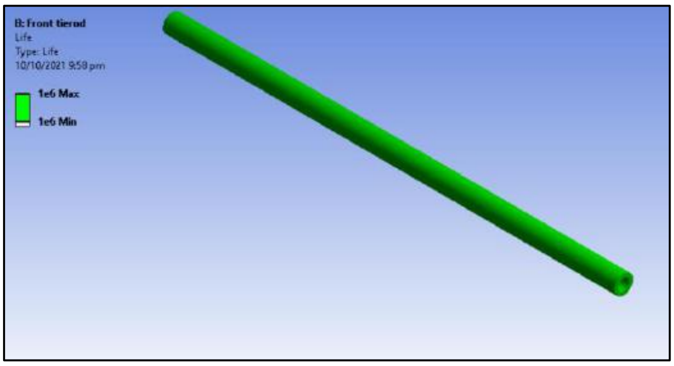


Fig 6- Minimum fatigue life

Maximum stress:**5.413MPa**

Minimum fatigue life: **Infinite life**

Maximum stress	**5.41MPa**
Minimum fatigue life	**Infinite life**

d) Optimizations:

The FOS is good enough, thus no optimization was required.

5.7 M8 BOLTS GRD 8.8

a) Assumption & Considerations:

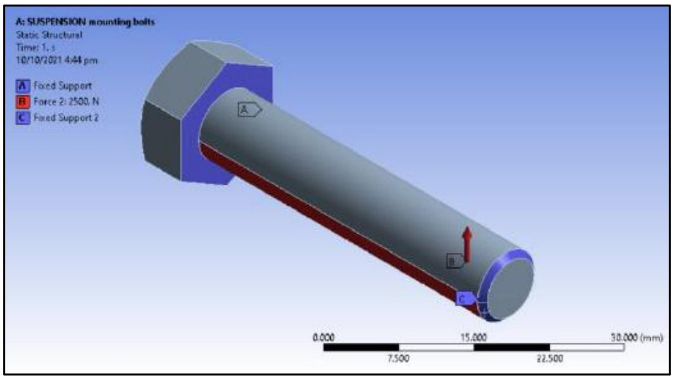


Fig 1- Boundary conditions

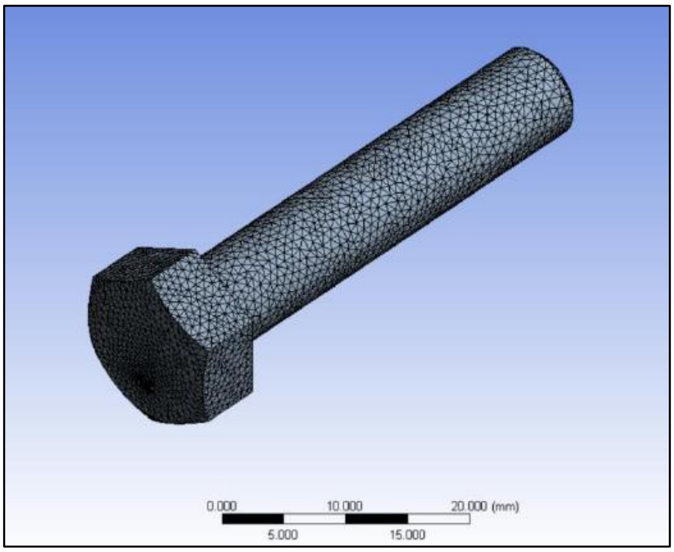


Fig 2- Meshed view

b) Calculation of Forces:

2500N force is considered under the conditions

The vehicle at approx. 1G force

c) Analysis Results:

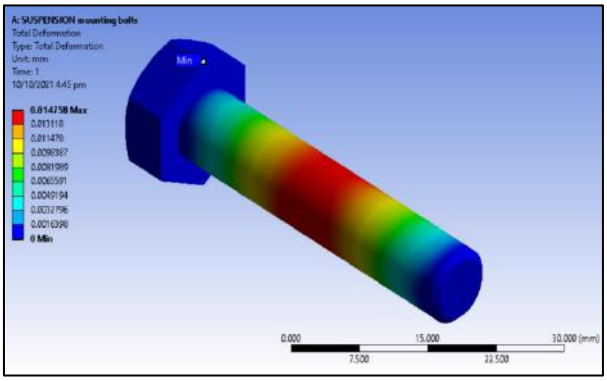


Fig 3- Total deformation

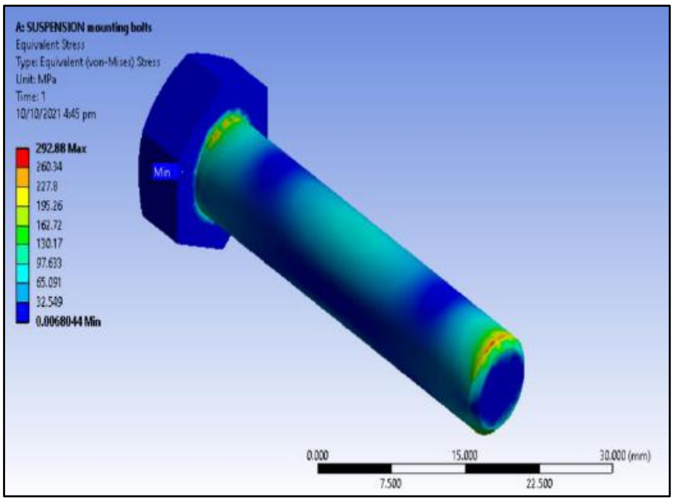


Fig 4- Equivalent stress

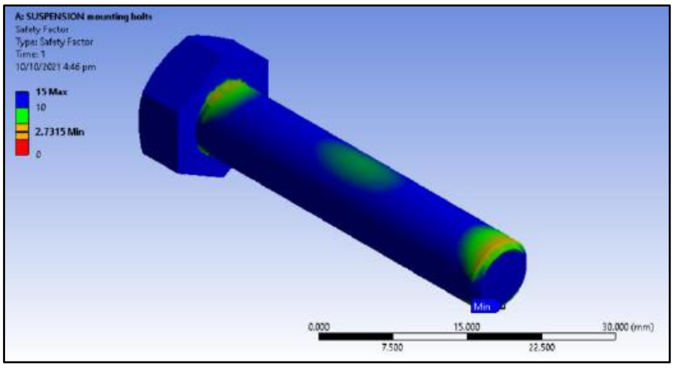


Fig 5- FOS (static)

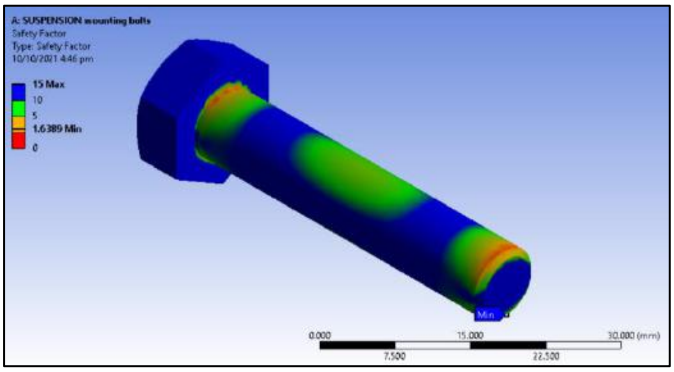


Fig 6- FOS(Fatigue)

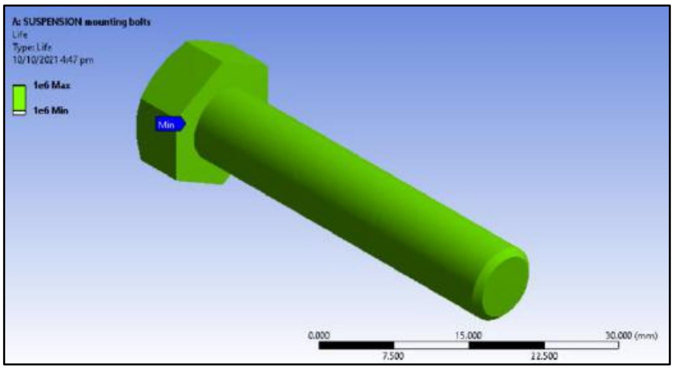


Fig 7- Minimum fatigue life

Maximum deformation:**0.014mm**

Maximum stress:**292.88MPa**

FOS (static):**2.43**

FOS(Fatigue):**1.63**

Minimum fatigue life: Infinite life

Maximum deformation	**0.014mm**
Maximum stress	**292.88MPa**
Factor of safety(static)	**2.43**
Factor of safety (fatigue)	**1.63**
Minimum fatigue life	**Infinite life**

d) Optimizations:

The FOS is good enough, thus no optimization was required.

5.7 KNUCKLE

a) Assumption & Considerations:

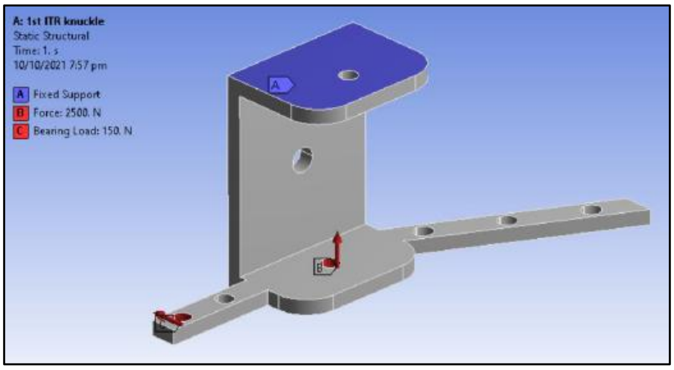


Fig 1- Boundary conditions

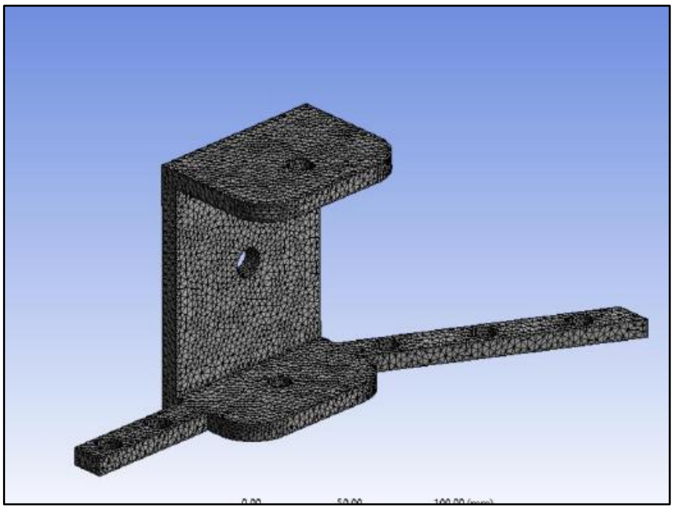


Fig 2- Meshed view

b) Calculation of Forces:

The reaction force of 2500N is considered under the condition the vehicle at 1G force.

c) Analysis Results:

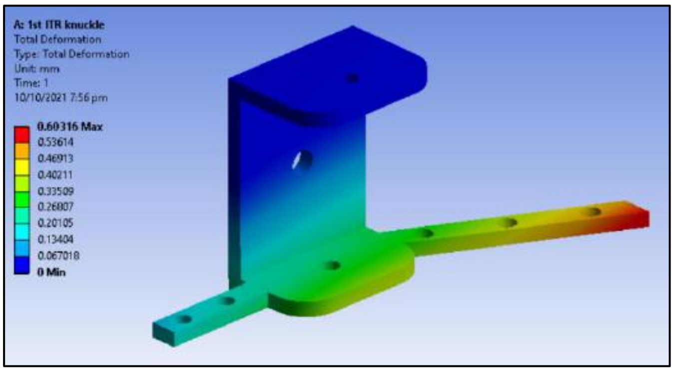


Fig 3- Total deformation

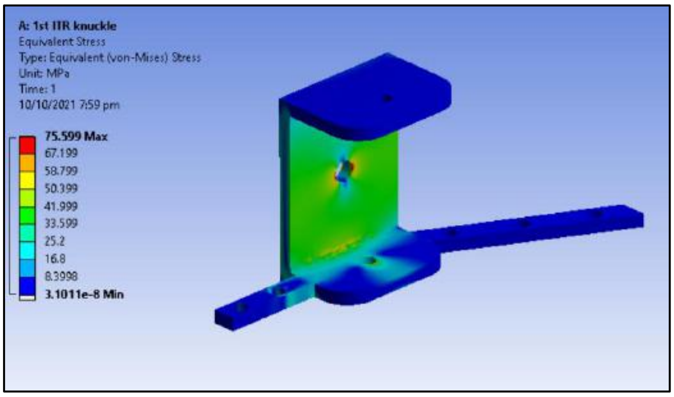


Fig 4- Equivalent stress

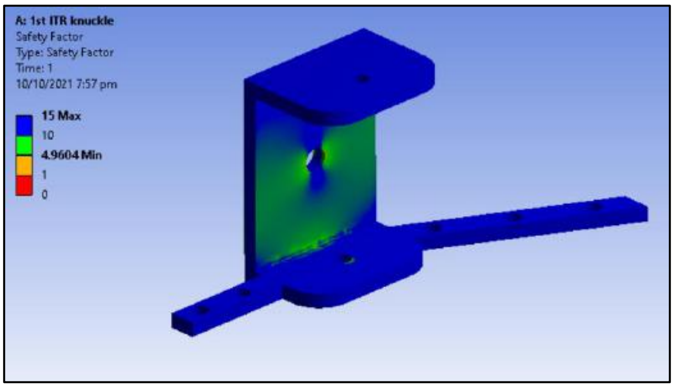


Fig 5- FOS (static)

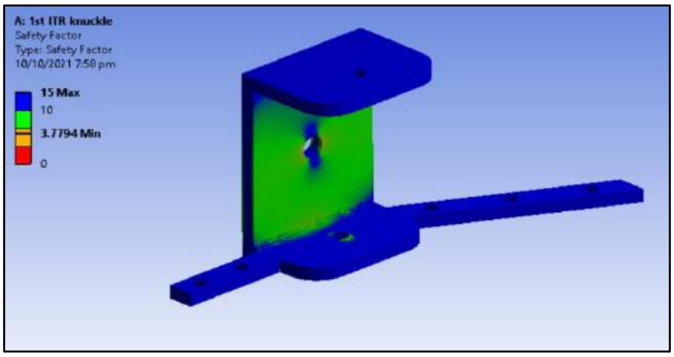


Fig 6- FOS (fatigue)

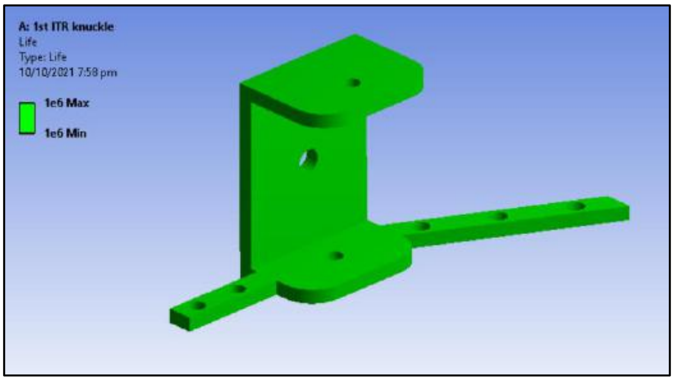


Fig 7- Minimum fatigue life

Maximum deformation	**0.60mm**
Maximum stress	**75.99MPa**
Factor of safety(static)	**4.96**
Factor of safety (fatigue)	**3.77**
Minimum fatigue life	**Infinite life**

d) Optimizations:

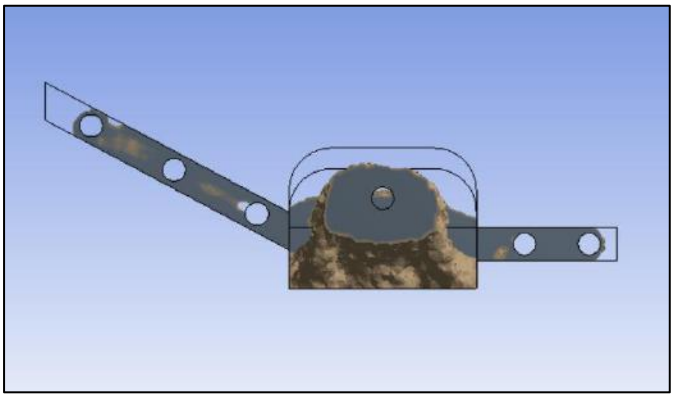


Fig 8- Top view

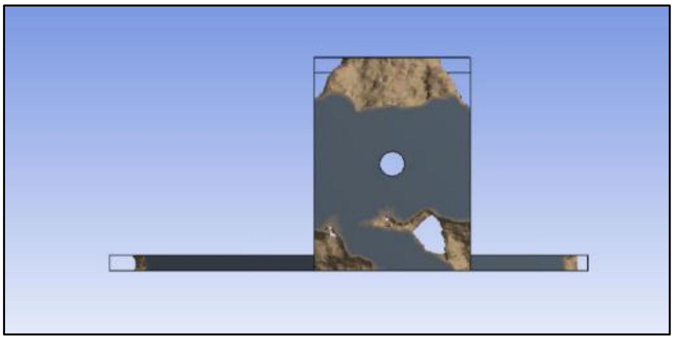


Fig 9- Front view

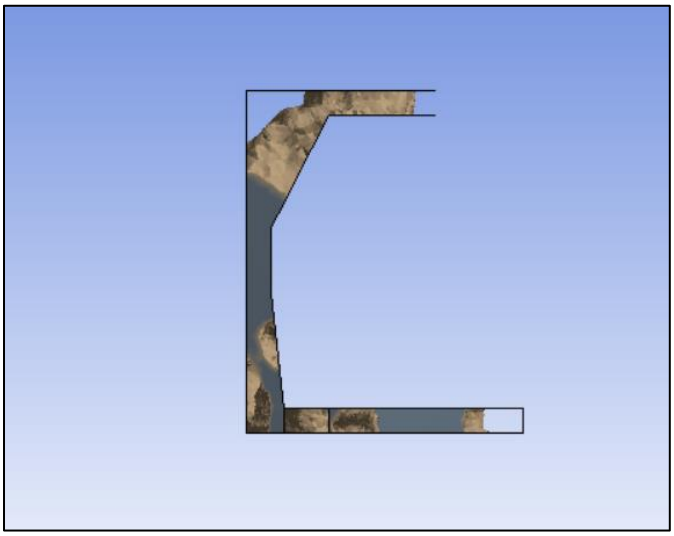


Fig 10- Side view

5.7.1 OPTIMISED KNUCKLE

a) Assumption & Considerations

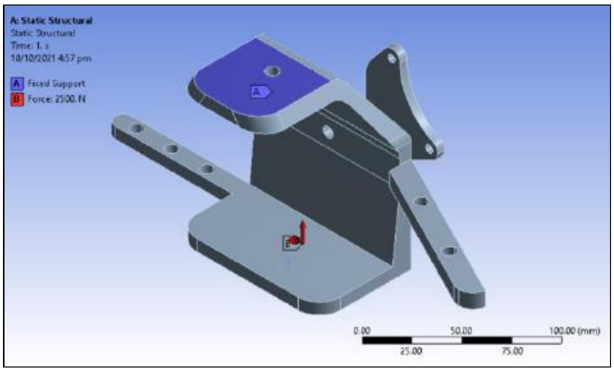


Fig 1- Boundary conditions

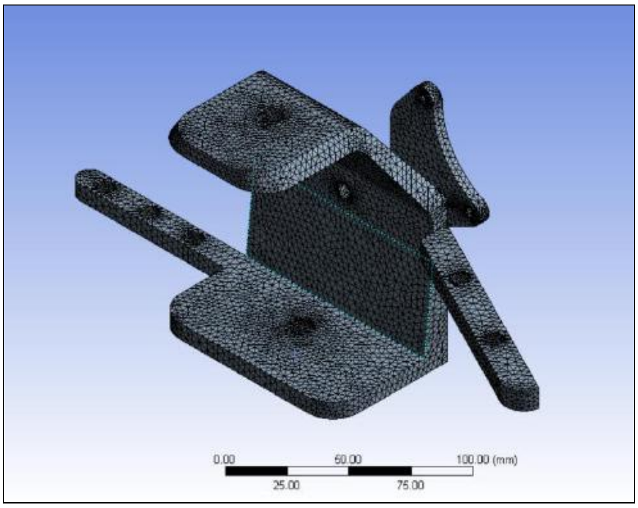


Fig 2- Meshed view

b) Calculation of Forces:

The reaction force of 2500N is considered.

c) Analysis Results:

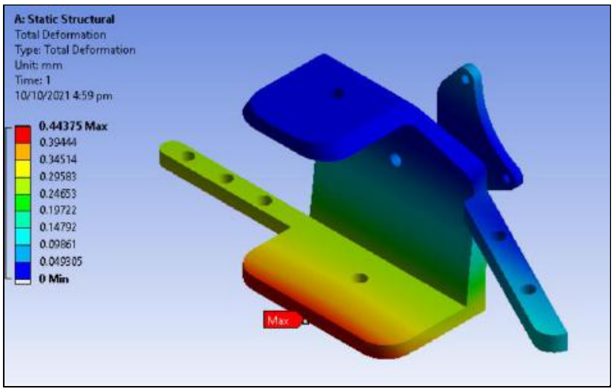


Fig 3- Total deformation

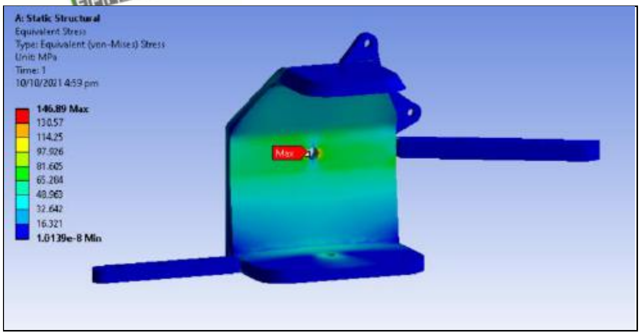


Fig 4- Equivalent stress

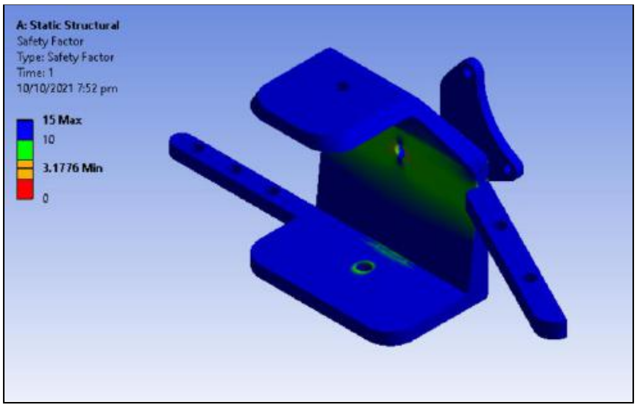


Fig 5- FOS (static)

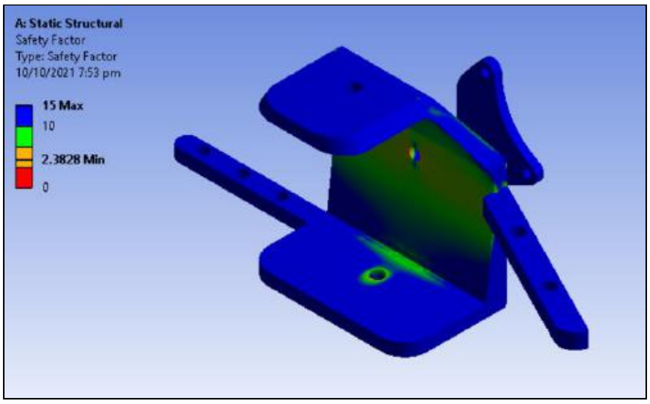


Fig 6- FOS (fatigue)

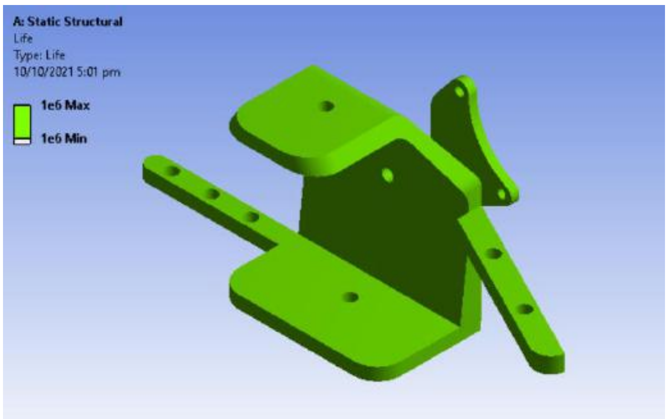


Fig 7- Minimum fatigue life

Maximum deformation:**0.443mm**

Maximum stress:**146.89MPa**

Factor of safety(static):**3.1776**

Factor of safety(fatigue):**2.382**

Minimum fatigue life **Infinite life**:

Maximum deformation	**0.44mm**
Maximum stress	**146.89MPa**
Factor of safety(static)	**3.177**
Factor of safety (fatigue)	**2.38**
Minimum fatigue life	**Infinite life**

5.8 WHEELS

a) Assumption & Considerations:

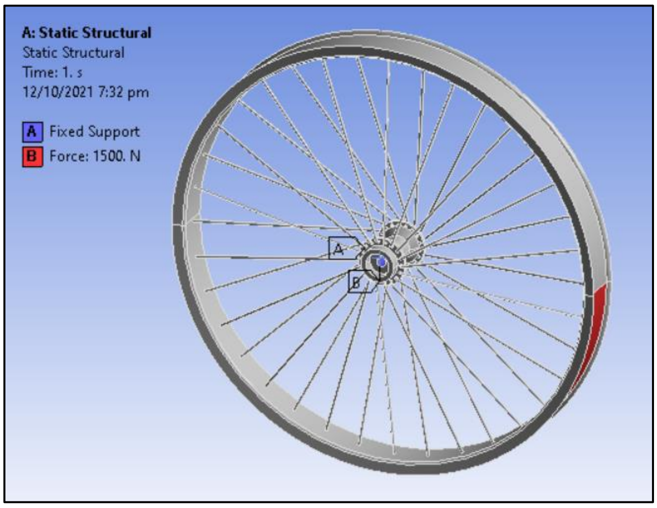


Fig 1- Boundary conditions

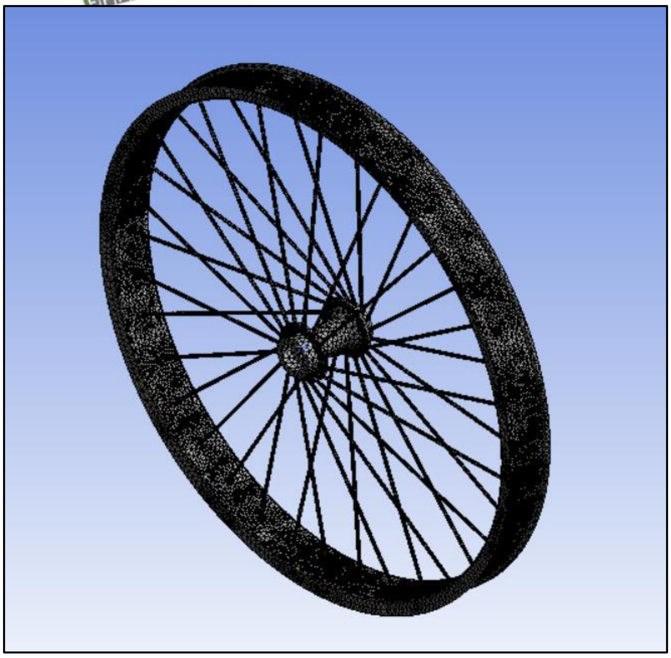


Fig 2- Meshed view

b) Calculation of Forces:

If we consider the back tire

Weight distribution of vehicle is 46.6:53.4

Weight of vehicle including drivers' weight =270kg

So, the net reaction force on back wheel is =

270 x (53.4/100) (x10= approx. 1500N

c) Analysis Results:

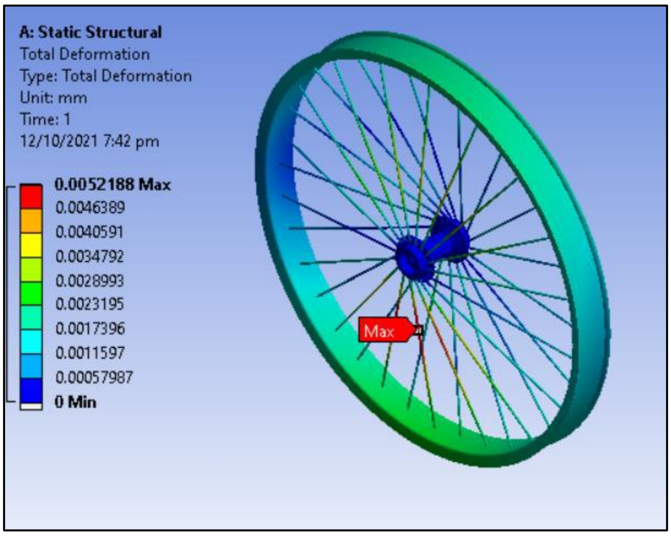


Fig 3- Total deformation

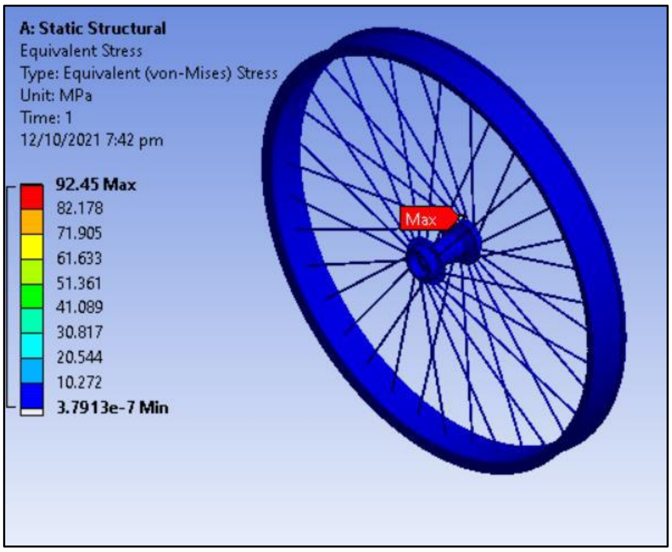


Fig 4- Equivalent stress

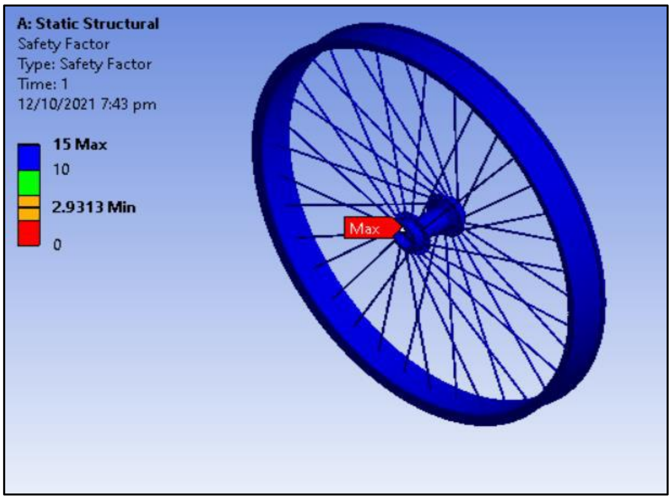


Fig 5- FOS

Maximum deformation	**0.0052mm**
Maximum stress	**92.45MPa**
Factor of safety(static)	**2.93**

d) Optimizations:

The FOS is good enough, thus no optimization was required.

5.9 FRONT FAIRING

a) Assumption & Considerations:

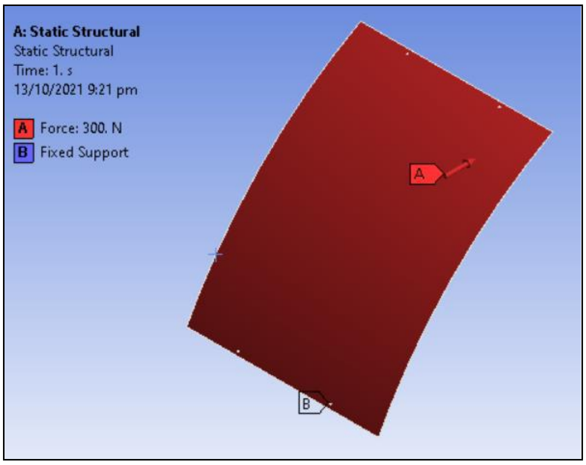


Fig 1- Boundary conditions

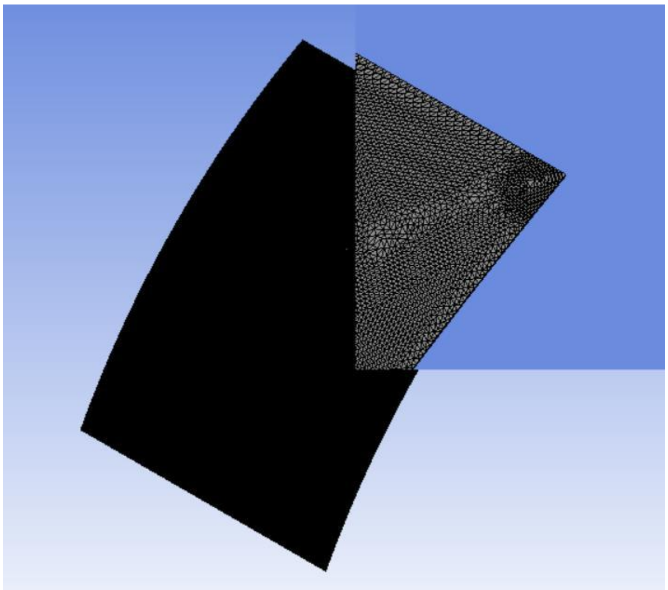


Fig 2- Meshed view

b) Calculation of Forces:

The drag force was found to be about 28N.

c) Analysis Results:

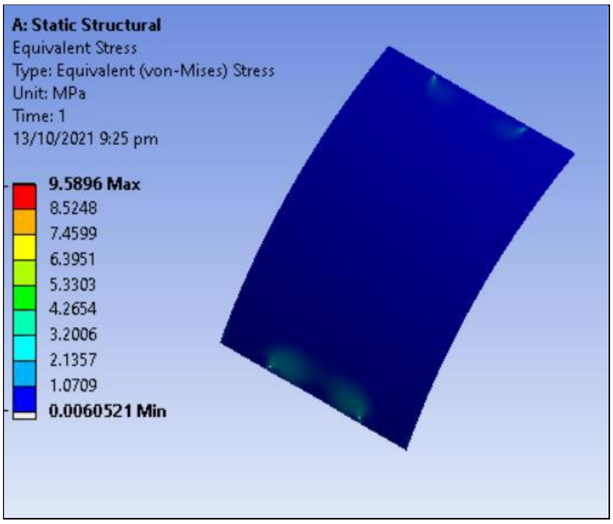


Fig 3- Equivalent stress

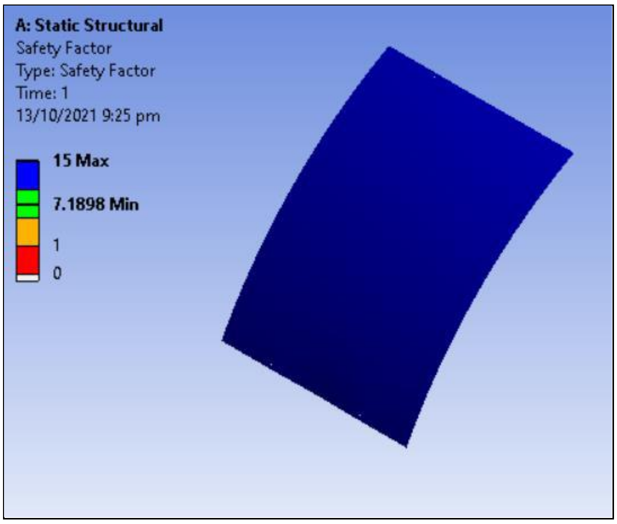


Fig 4- FoS

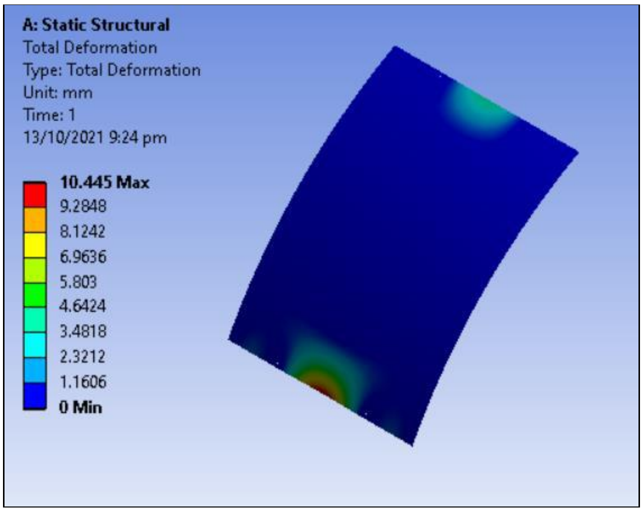


Fig 5- Total Deformation

Maximum deformation	**10.45mm**
Maximum stress	**9.58MPa**
Factor of safety(static)	**7.18**

d) Optimizations:

The FOS is good enough, thus no optimization was required.

5.10 SPRING ANALYSIS

a) Assumption & Considerations:

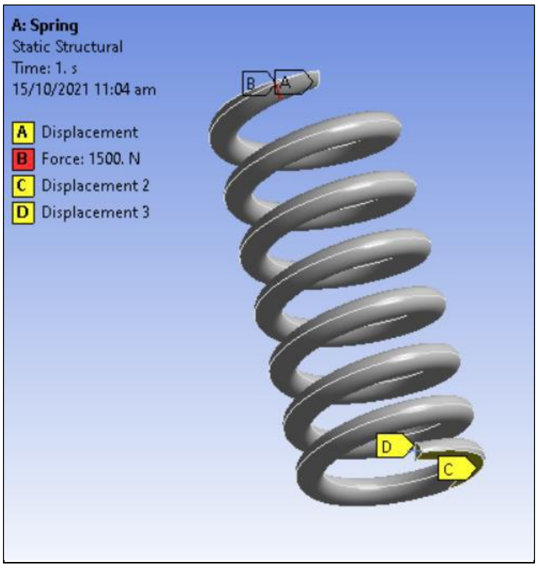


Fig 1- Boundary conditions

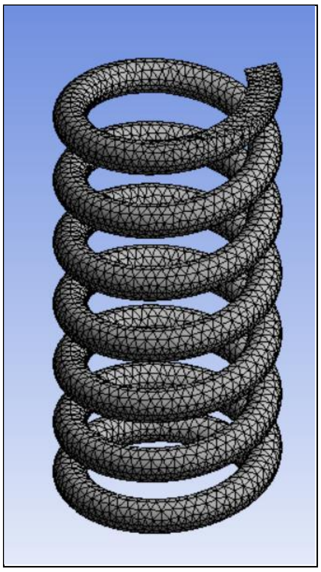


Fig 2- Mesh

b) Calculation of forces:

Force on rear spring is found from weight distribution
The weight distribution about the axles is 46.6:53.4
The weight is assumed to be 270kgs so force is
F=ma= 270*9.81 = 2,648.7 N
The force acting on rear axle is 1,324.35 N
There are two springs on the rear suspension therefore the force on each spring is 662.175 N.
But considering the worst-case scenario, the force was taken as 1500N.

c) Analysis results:

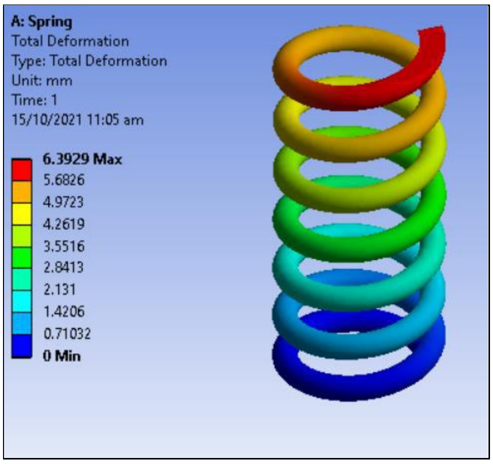


Fig 3- Maximum Deformation

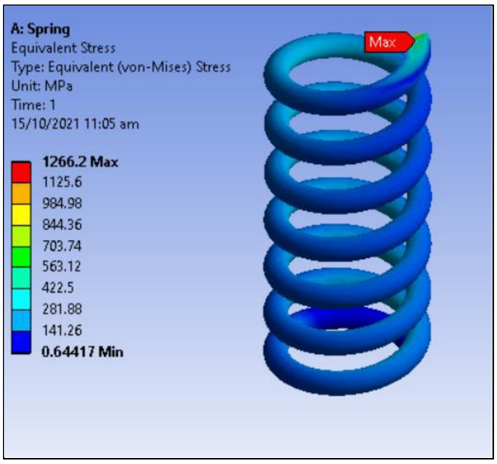


Fig 4- Maximum stress

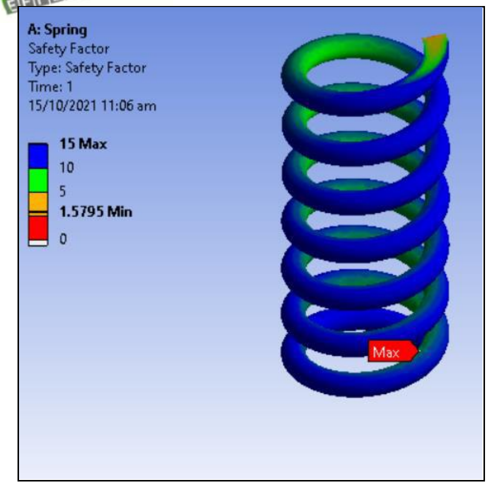


Fig 5- FOS (static)

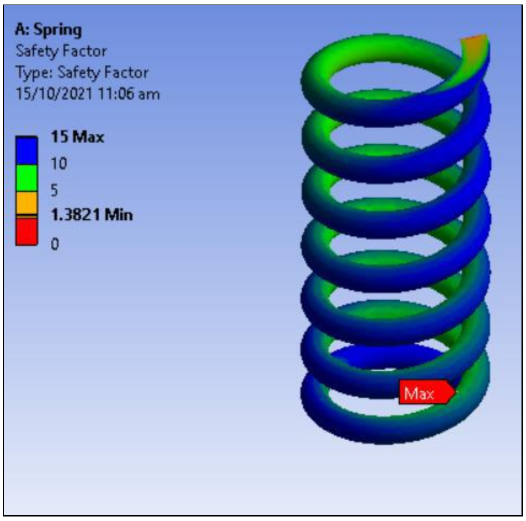


Fig 6- FOS (Fatigue)

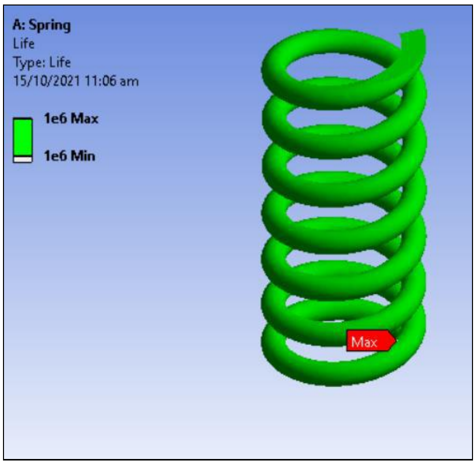


Fig 7- Minimum fatigue life

d) Optimizations:

No optimizations are necessary as this is an OEM part and will be purchased based on specifications

5.11 SERVO MOTOR HOUSING

a) Assumptions and considerations:

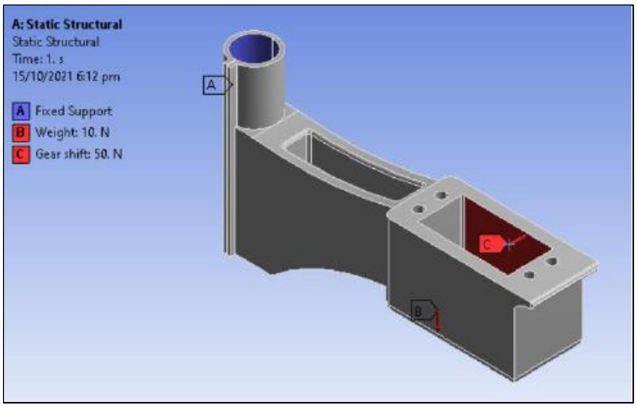


Fig 1- Boundary conditions

b) Calculation of forces:

The forces acting on the servo motor housing are the weight of the servo motor and the force applied by motor
The force applied by motor is taken as 50N in the worst-case scenario.

c) Analysis results

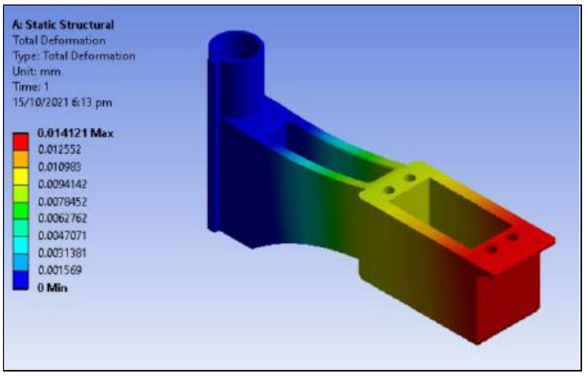


Fig 2- Deformation

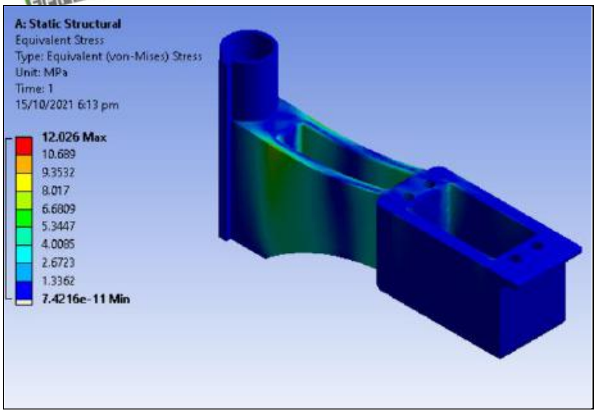


Fig 3- Equivalent Stress

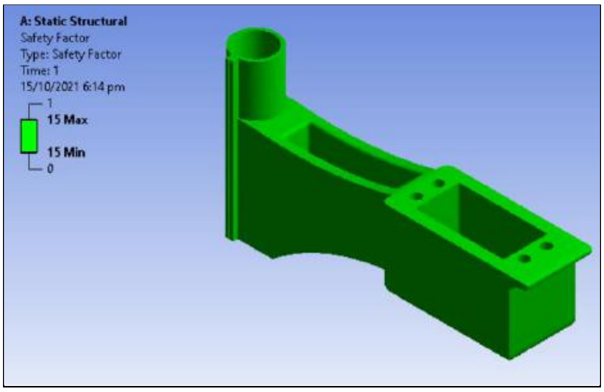


Fig 4- FOS

d) Optimizations:

No optimizations were needed.

5.12 CFD ANALYSIS ON FRAME

a) Assumption & Considerations:

The enclosed frame was taken as the geometry and the effects of airflow were seen around it.

b) Calculation of forces:

The headwind velocity was taken as 39 Kmph which is the maximum speed of the vehicle.

c) Analysis results:

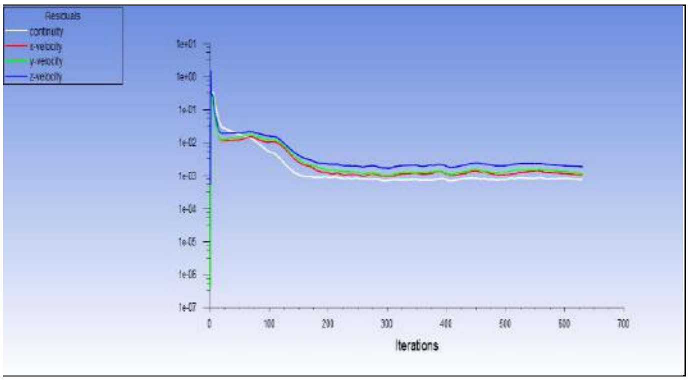


Fig 1- Convergence plot

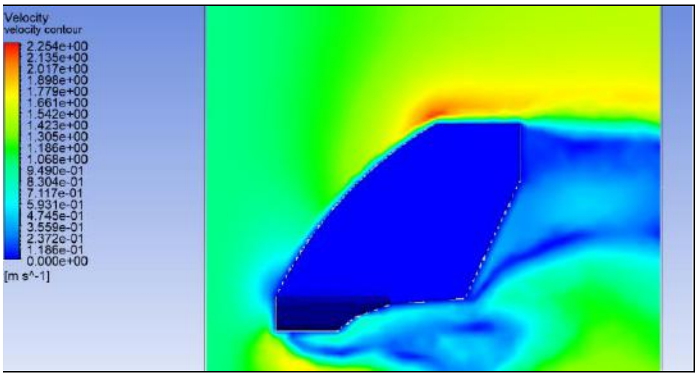


Fig 2- Velocity contour

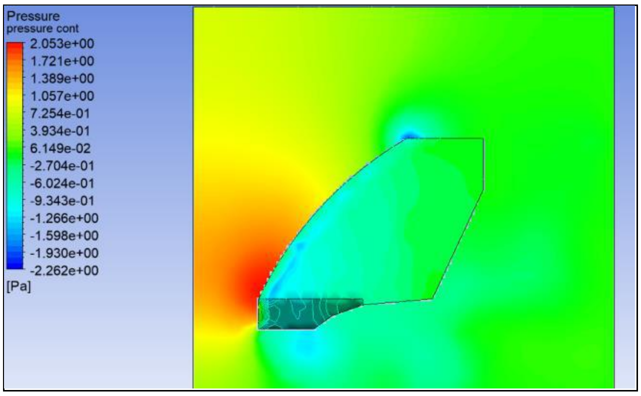


Fig 3- Pressure contour

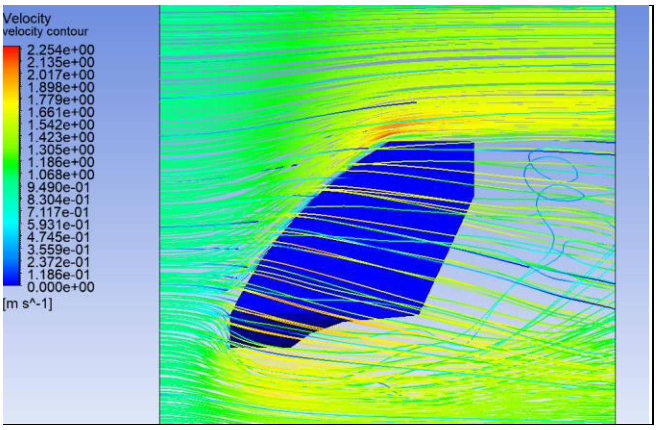


Fig 4- Particle trace

```
 iter  continuity  x-velocity  y-velocity  z-velocity          cd        lift        drag     time/iter


reversed flow in 188 faces on pressure-outlet 6.
  570  7.9665e-04  1.2725e-03  1.4495e-03  2.1943e-03  2.7590e-02 -1.9265e+00  2.0863e+00  0:00:49   8(
```

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1. VEHICLE VIEWS

APPENDIX-1: Vehicle View

Figure-1 (Isometric View of Vehicle)

Figure-2 (Front View of Vehicle)

Figure-3 (Rear Isometric View of Vehicle)

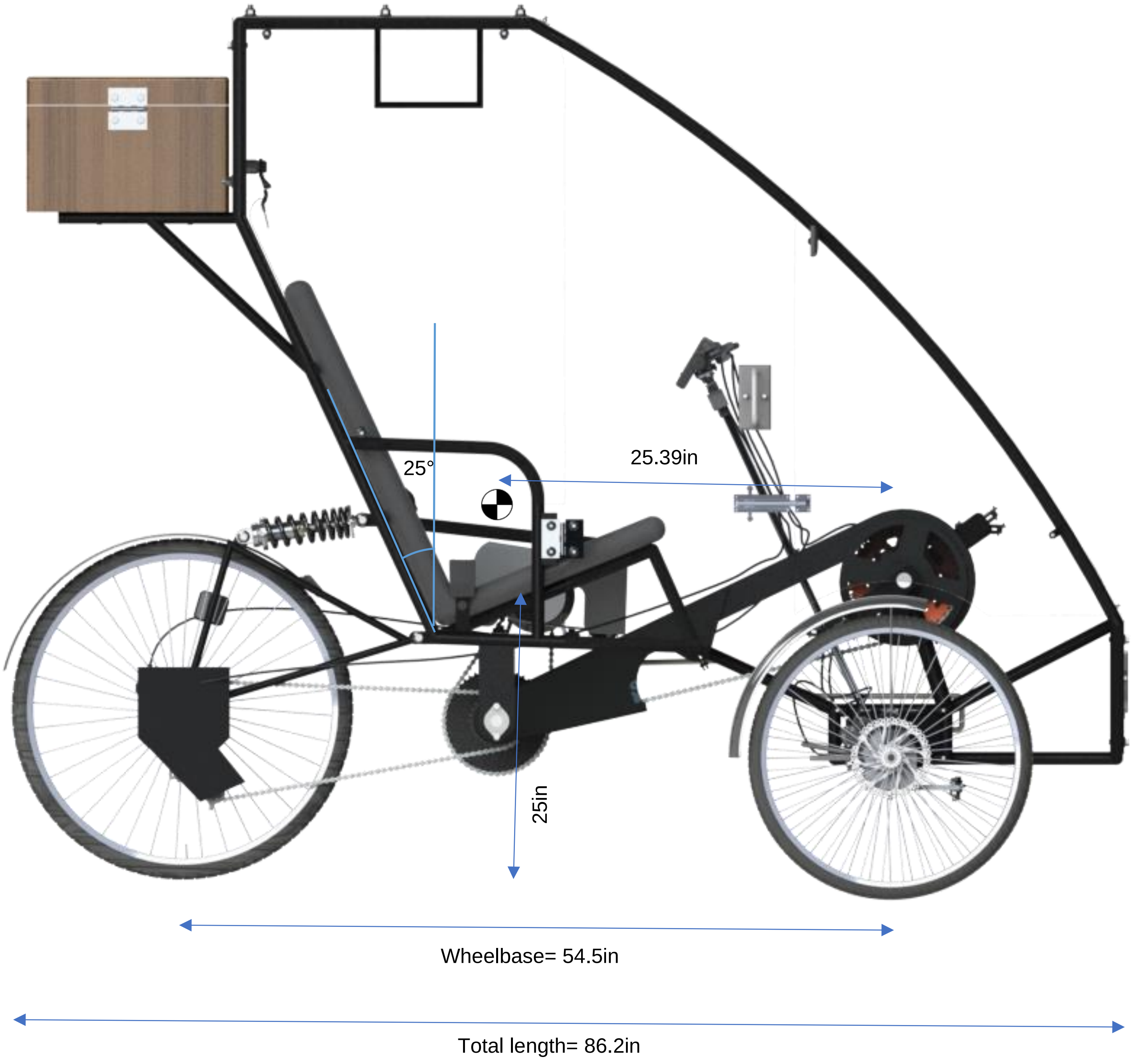

Figure-4 (Side View of Vehicle)

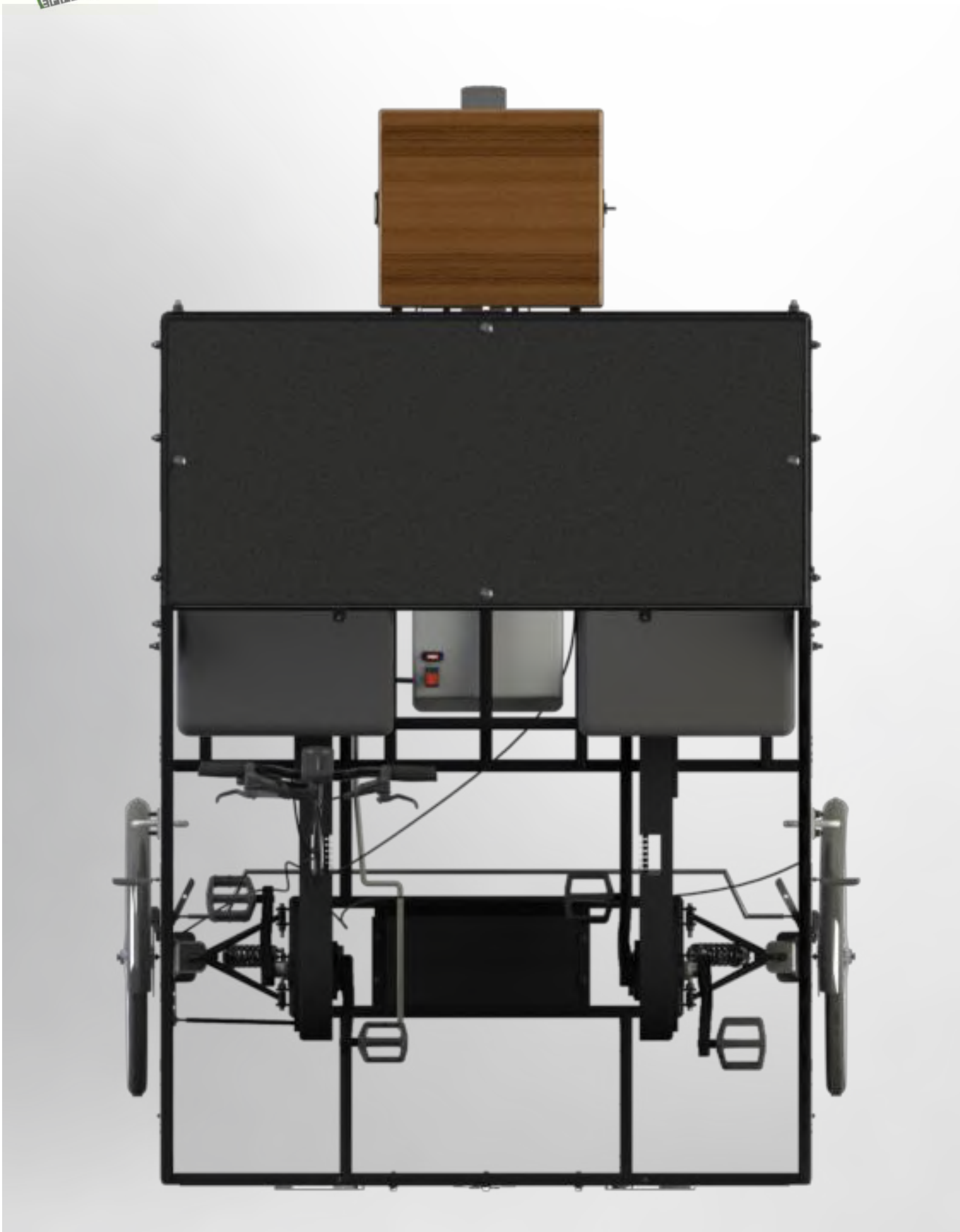

Figure-5 (Top View of Vehicle)

Figure-6 (Rear View of Vehicle)

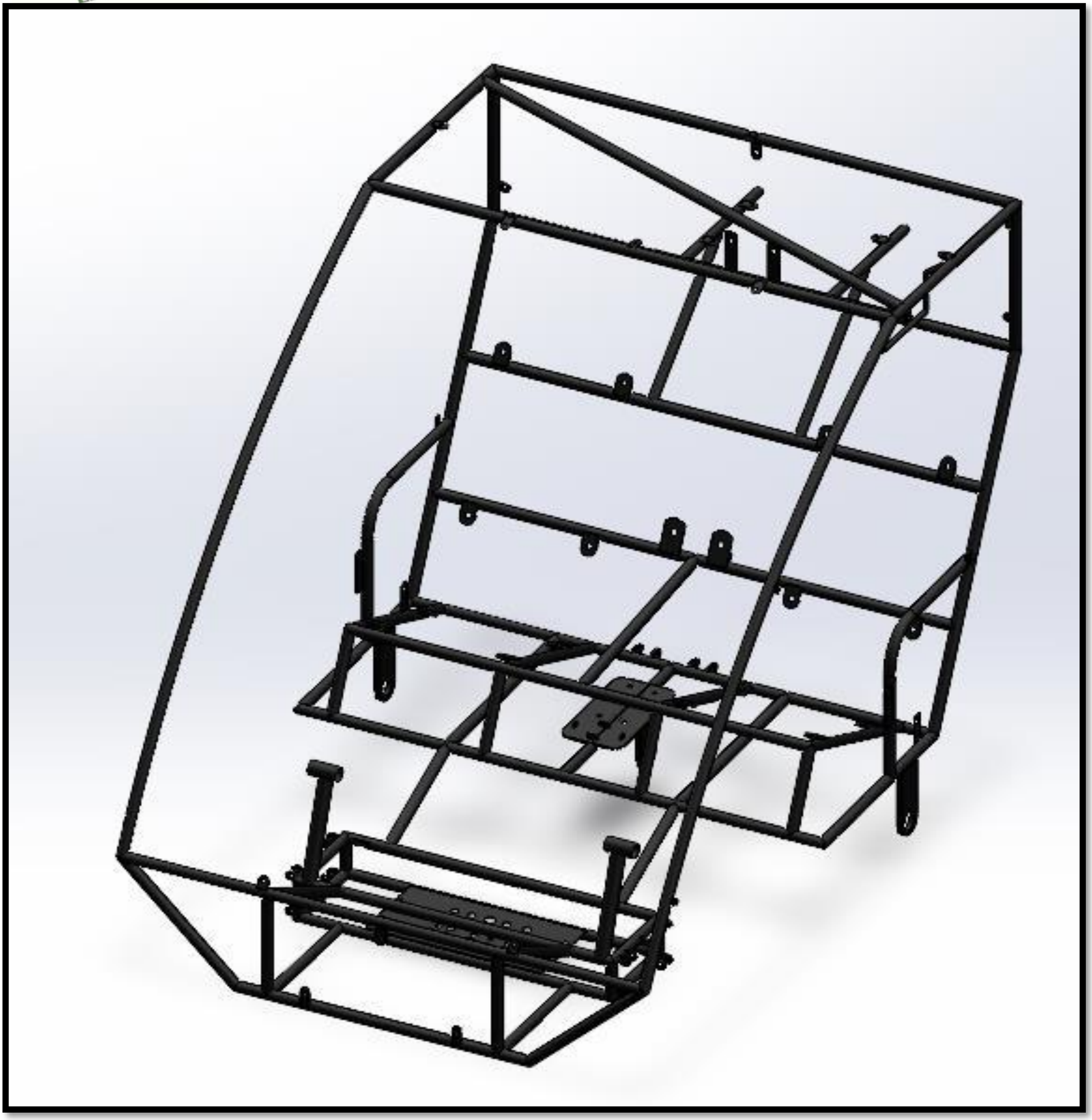

Figure-7 (Isometric View of Frame)

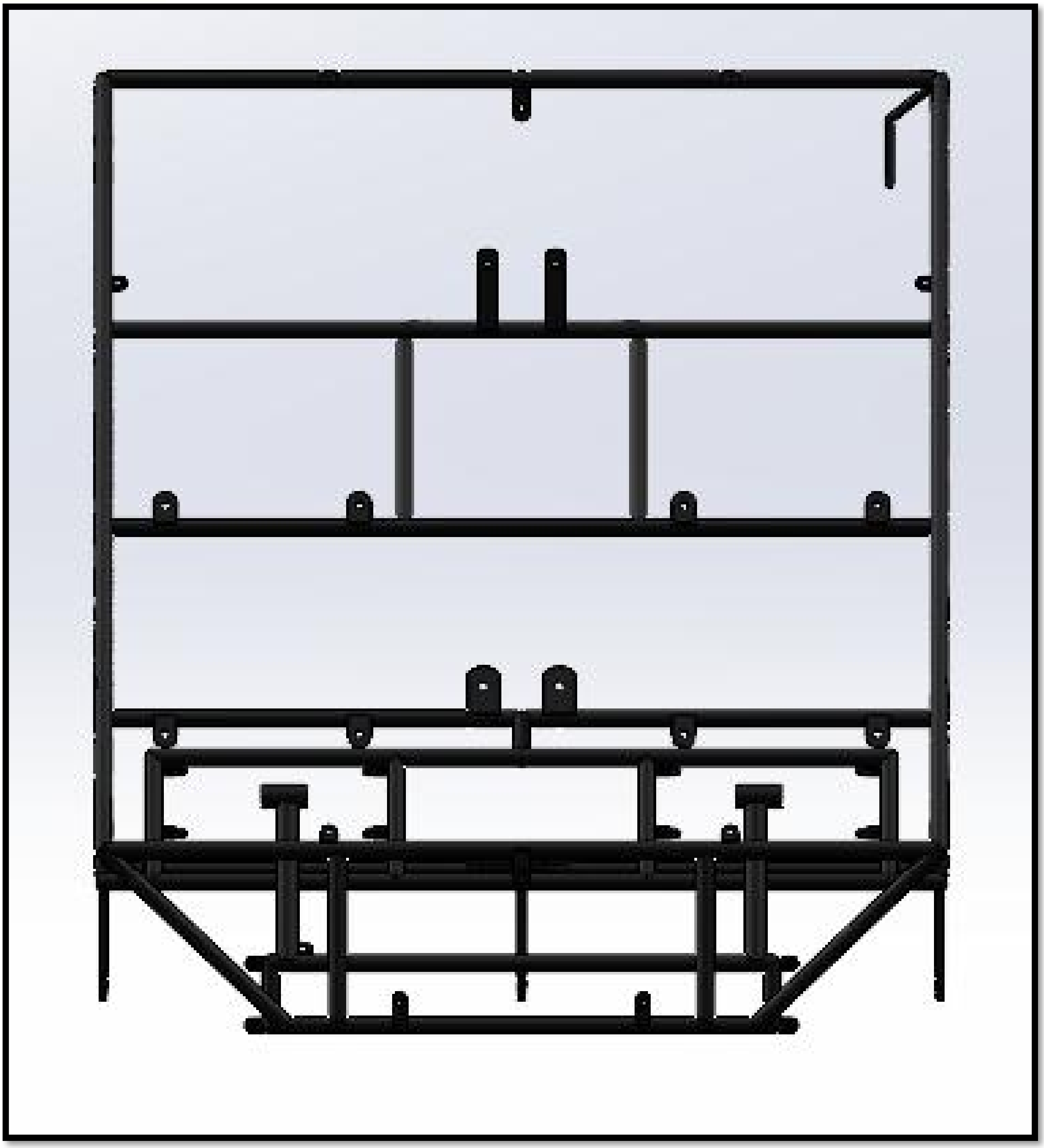

Figure-8 (Front View of Frame)

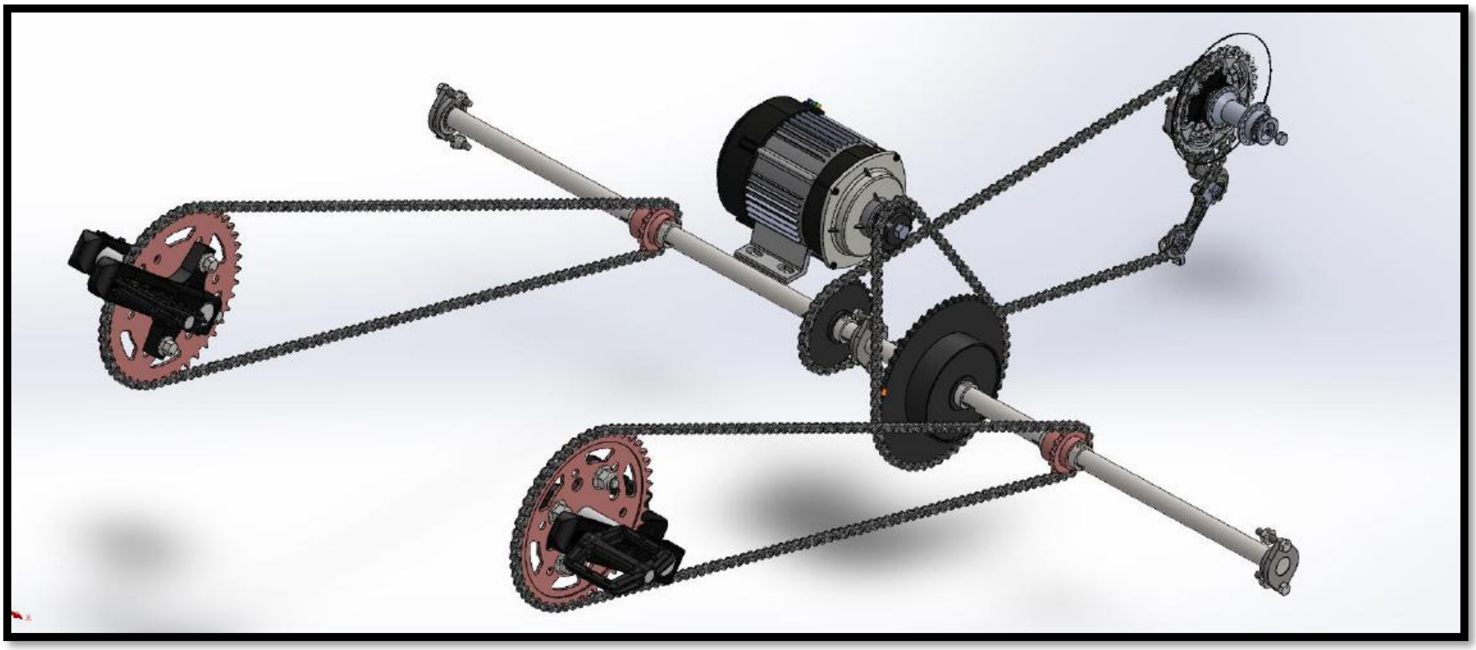

Figure-9 (Isometric View of Drivetrain)

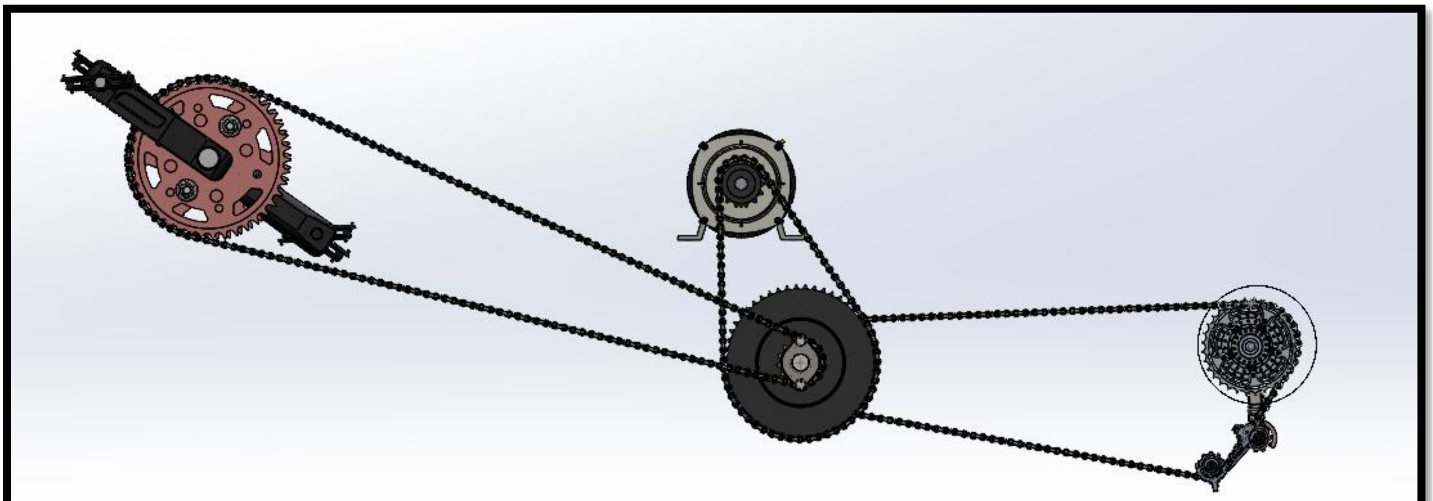

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Figure-10 (Side View of Drivetrain)

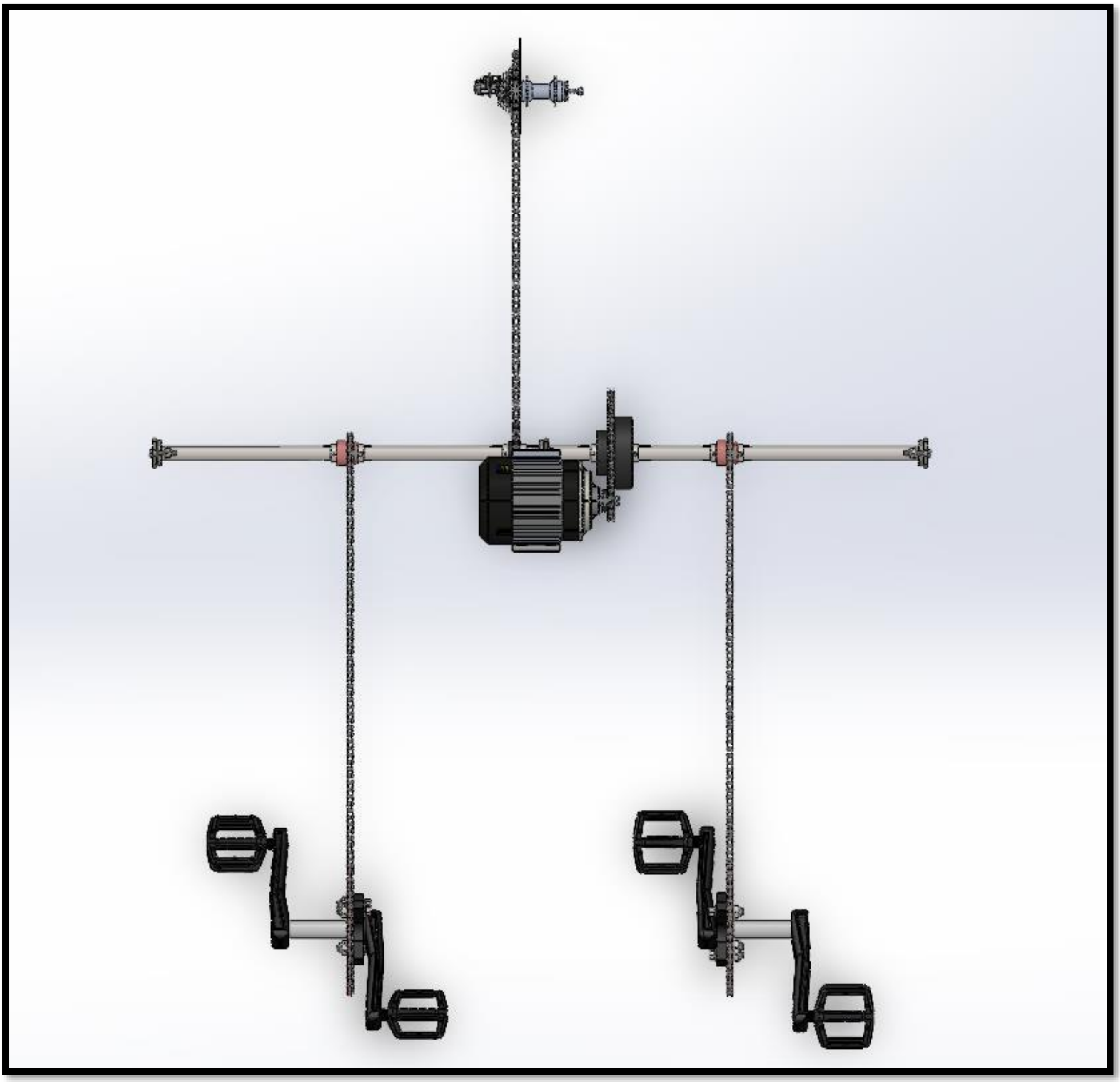

Figure-11 (Top View of Drivetrain)

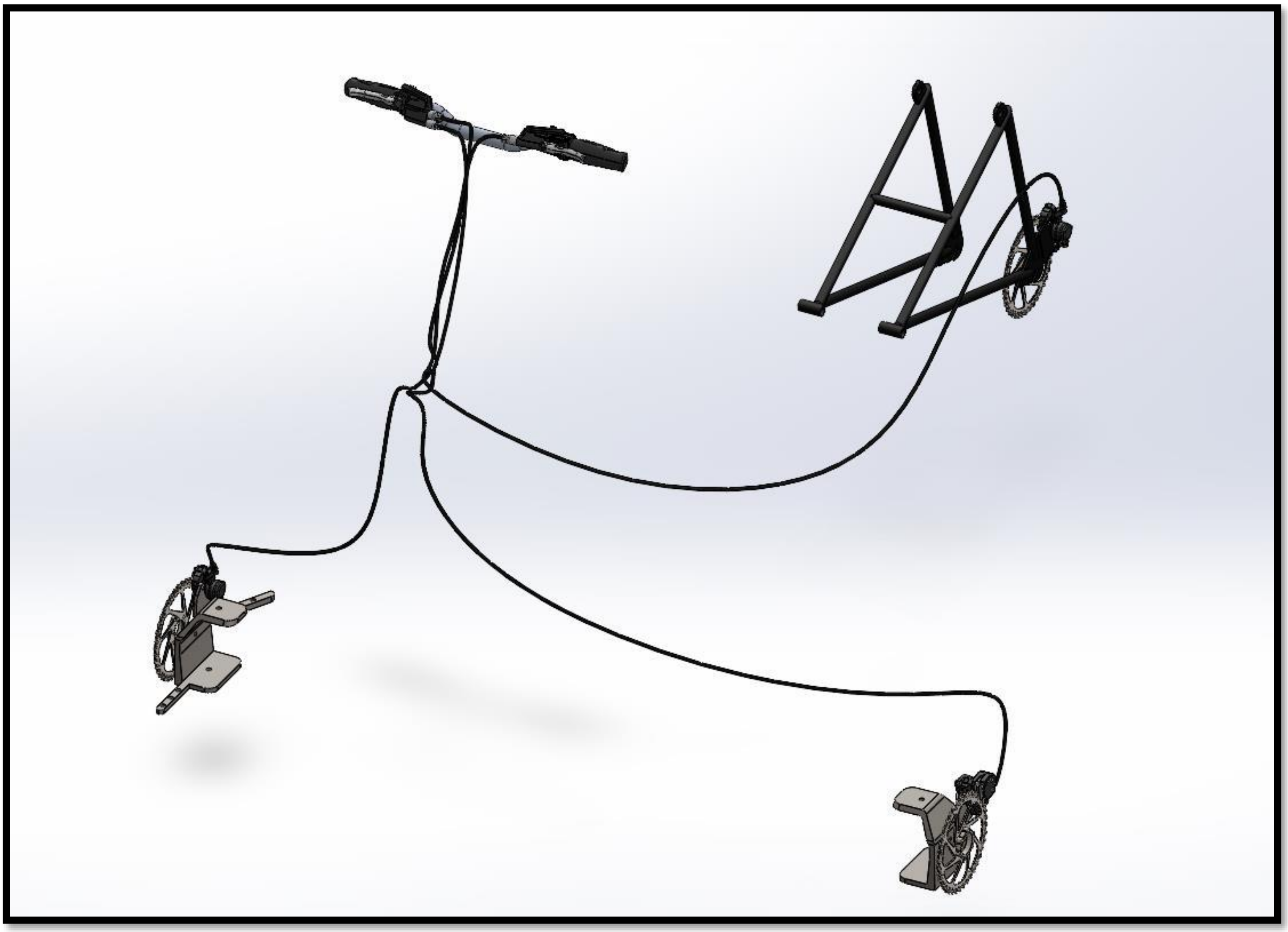

Figure-12 (Isometric View of Braking system)

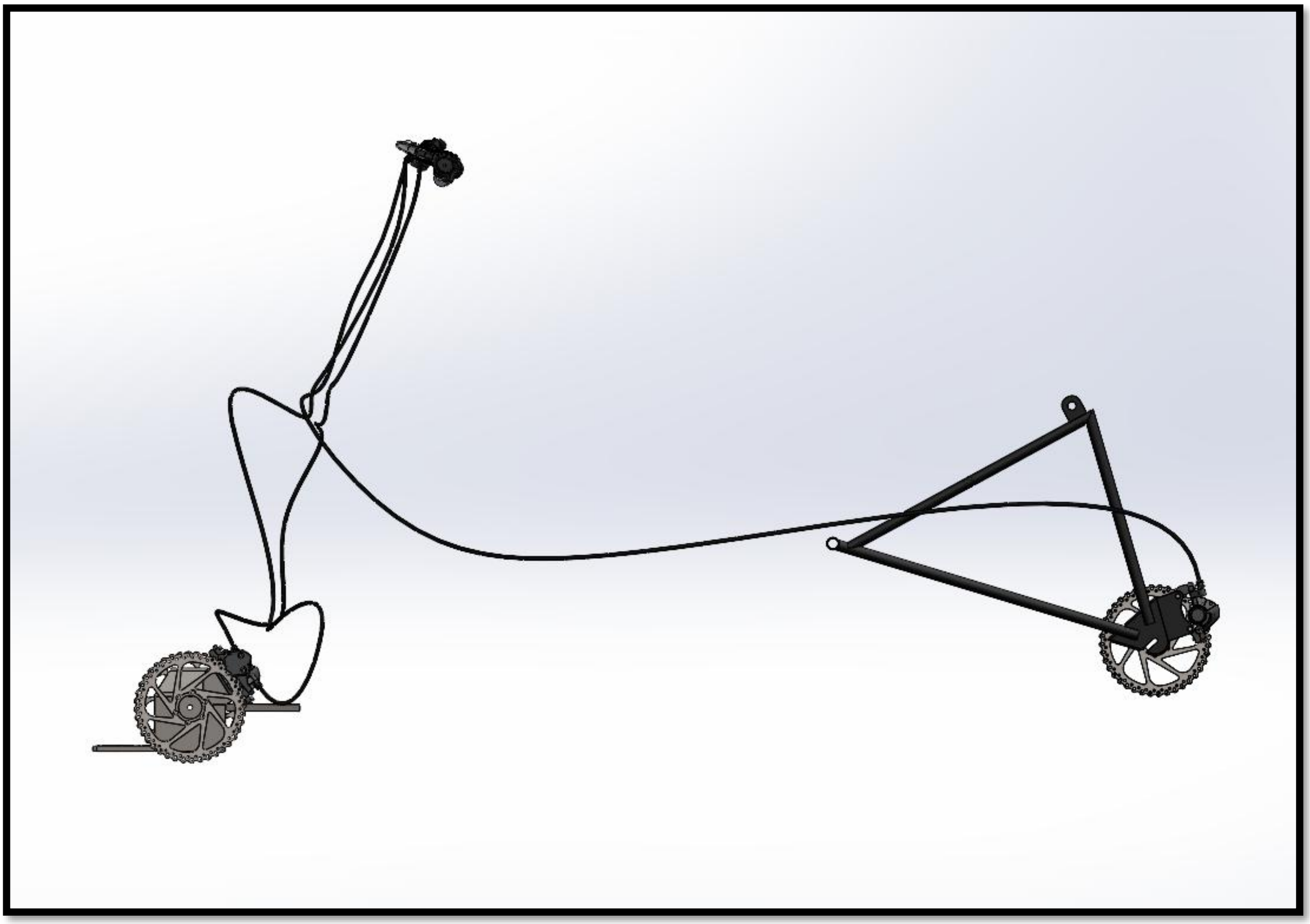

Figure-13 (Side View of Braking system)

Figure-14 (Isometric View of Steering system)

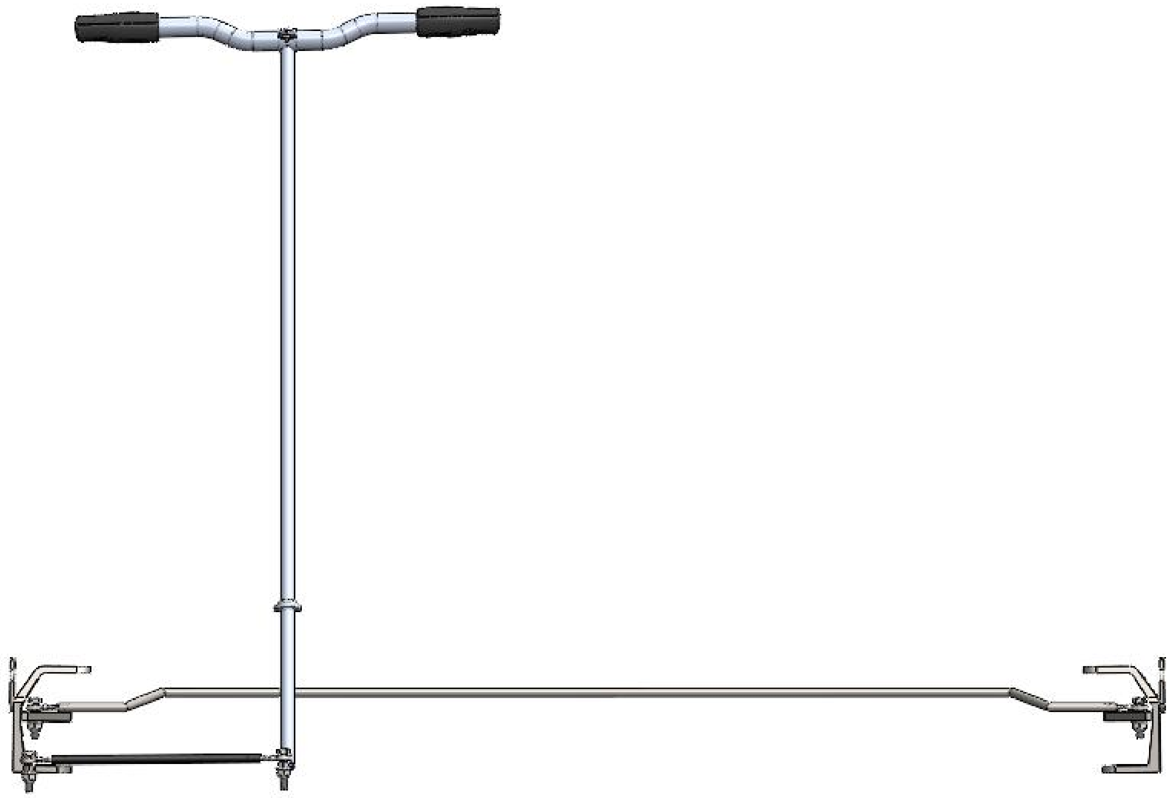

Figure-15 (Front View of Steering system)

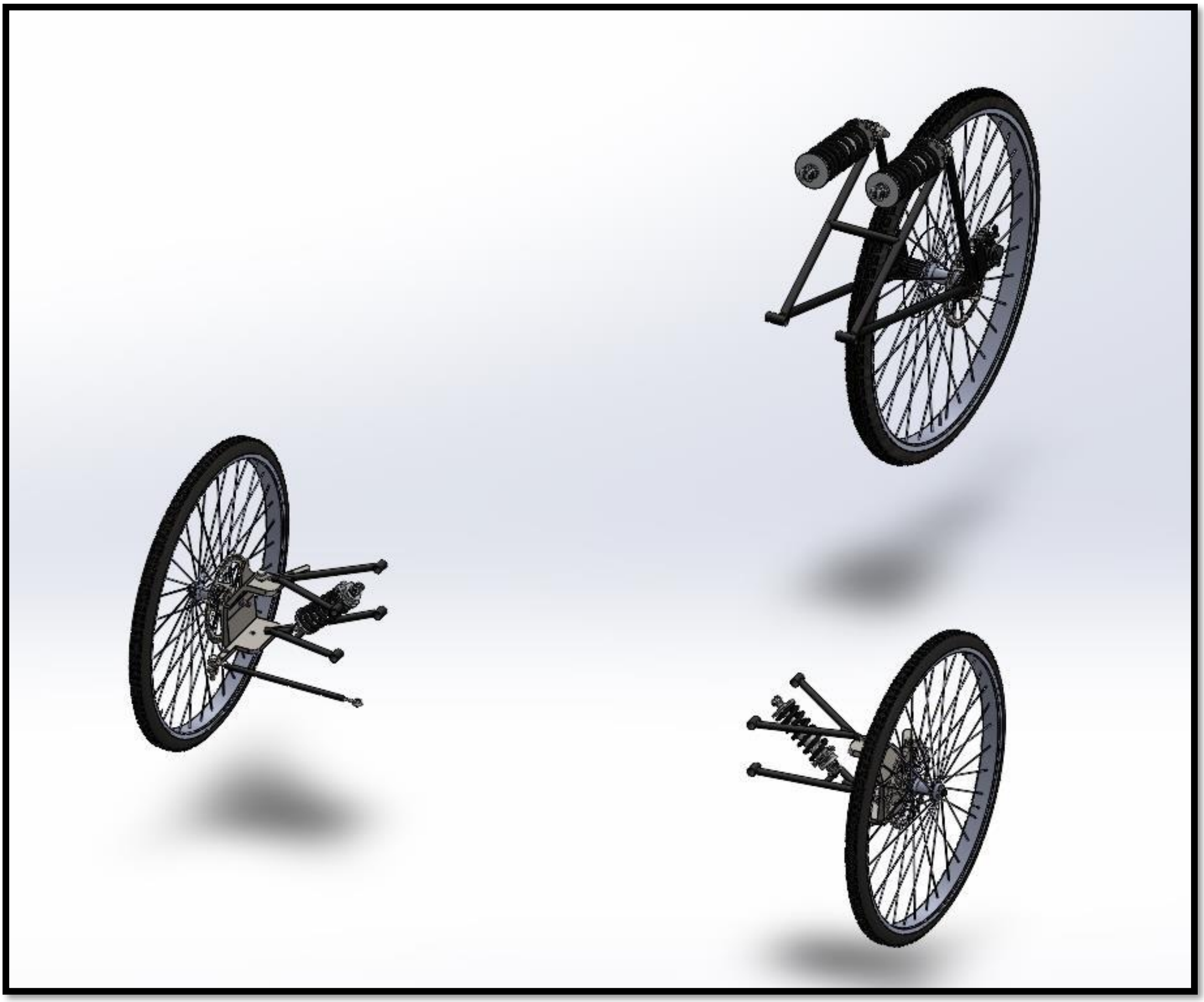

Figure-16 (Isometric View of Suspension system)

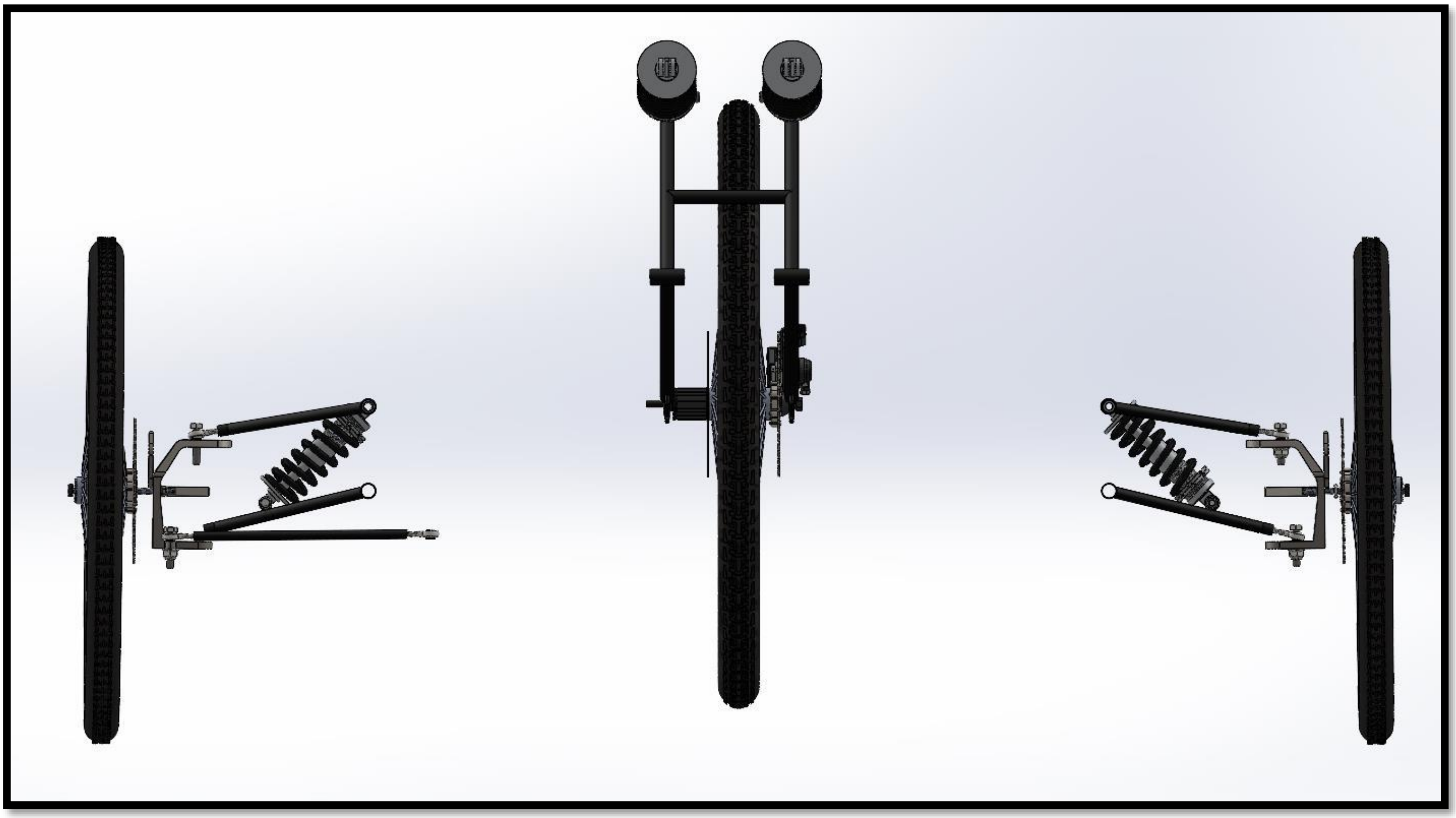

Figure-17 (Front View of Suspension system)

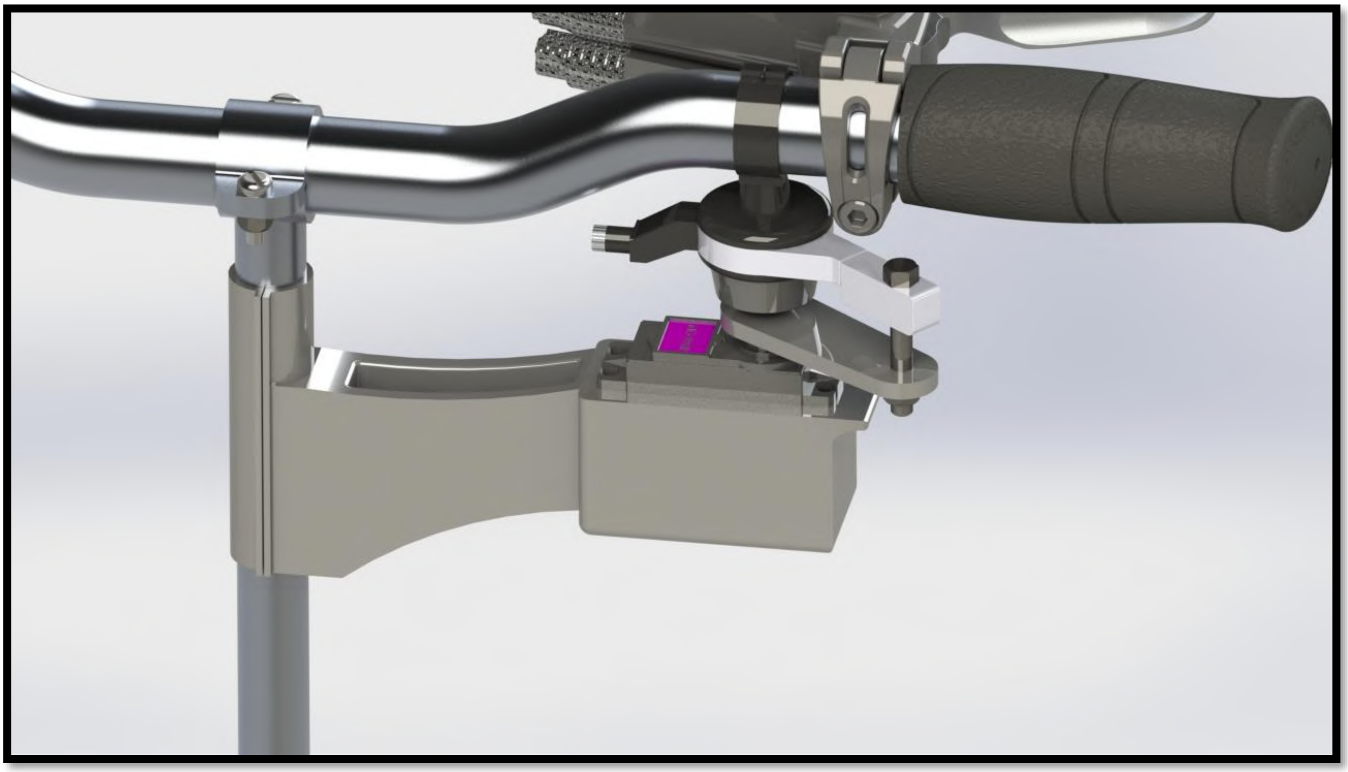

Figure-18 (View of the Automatic Gear System)

APPENDIX-2: CAE INPUT PARAMETERS

1. MATERIAL PROPERTIES

	Unit	AISI 1018	AISI 4130	Al 6061-T6
Density	Kg/m^3	7870	7850	2700
Young's Modulus	GPa	205	215	68
Yield Strength	MPa	365	460	276
Ultimate Strength	MPa	440	600	310
Poison's Ratio		0.29	0.29	0.33

2. MESH DETAILS

	Frontal Impact	Side Impact	Rollover	Bending Analysis	Torsional Analysis	Hard Point Analysis
No of Elements	79124	79124	79124	79124	79124	79124
Type of Elements	Tetrahedral	Tetrahedral	Tetrahedral	Tetrahedral	Tetrahedral	Tetrahedral
No of Nodes	166203	166203	166203	166203	166203	166203
Minimum Element Length	9.58×10^{-6}	9.58×10^{-6}	9.58×10^{-6}	9.58×10^{-6}	9.58×10^{-6}	9.58×10^{-6}
Maximum Element Length	0.0136mm	0.0136mm	0.0136mm	0.0136mm	0.0136mm	0.0136mm