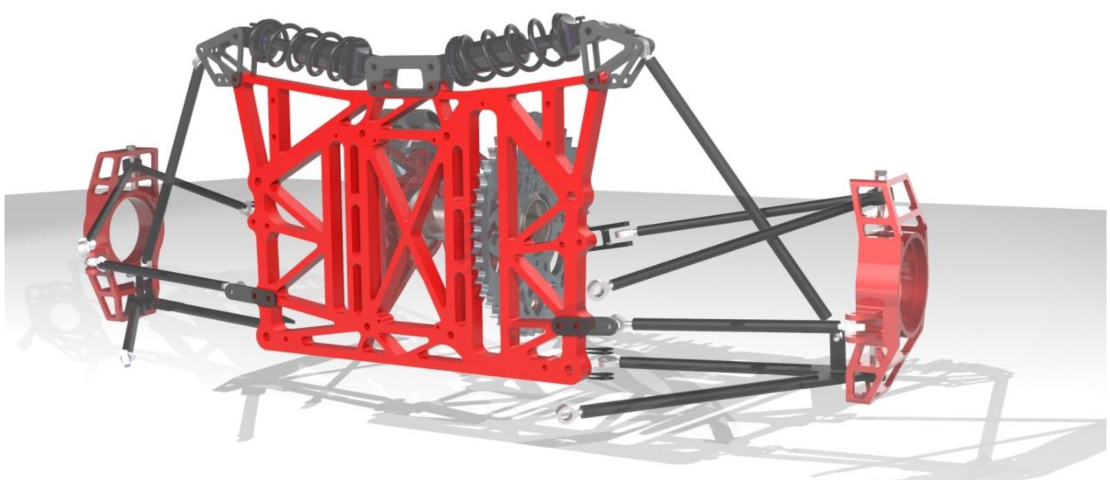
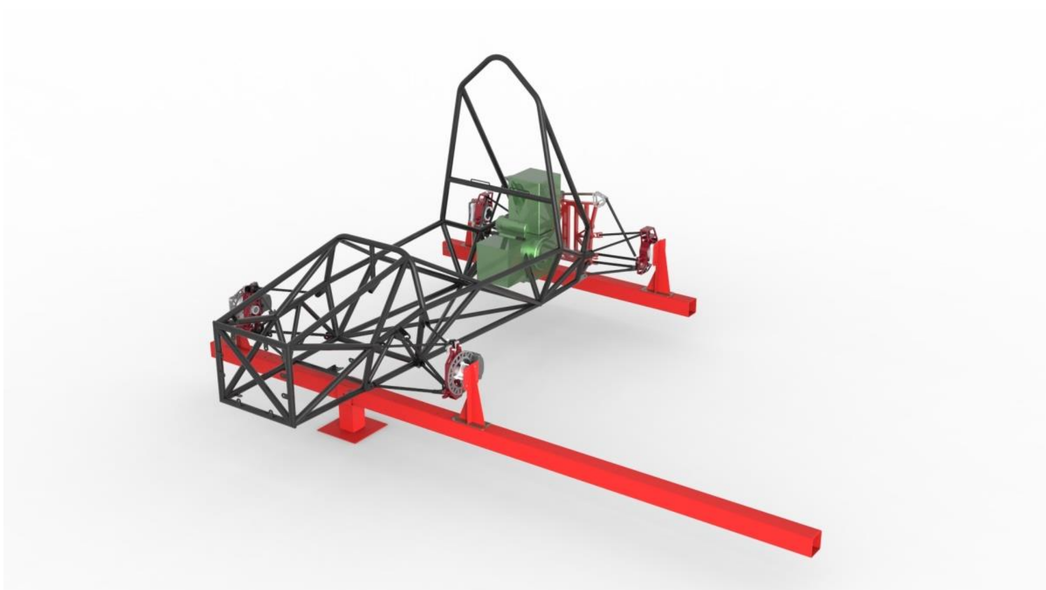
SPCE RACING

KLEIN 5.0

Chassis Documentation





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DESIGN

A vehicle frame, also known as its chassis, is the main supporting structure of a motor vehicle, to which all other components are attached.

Departmental Goals

- Lower the weight of the chassis
- Solve the problems faced by the previous car and incorporate the required changes in the current design.
- Increase the compactness without compromising the accessibility.
- Increased driver safety giving utmost importance to driver ergonomics.

Design Goals

- Weight reduction
- Increasing torsional rigidity/stiffness
- Lowering CG
- Getting suspension points at nodes of chassis for optimum force transfer
- Increasing driver comfort and visibility
- Increasing production accuracy

Design considerations

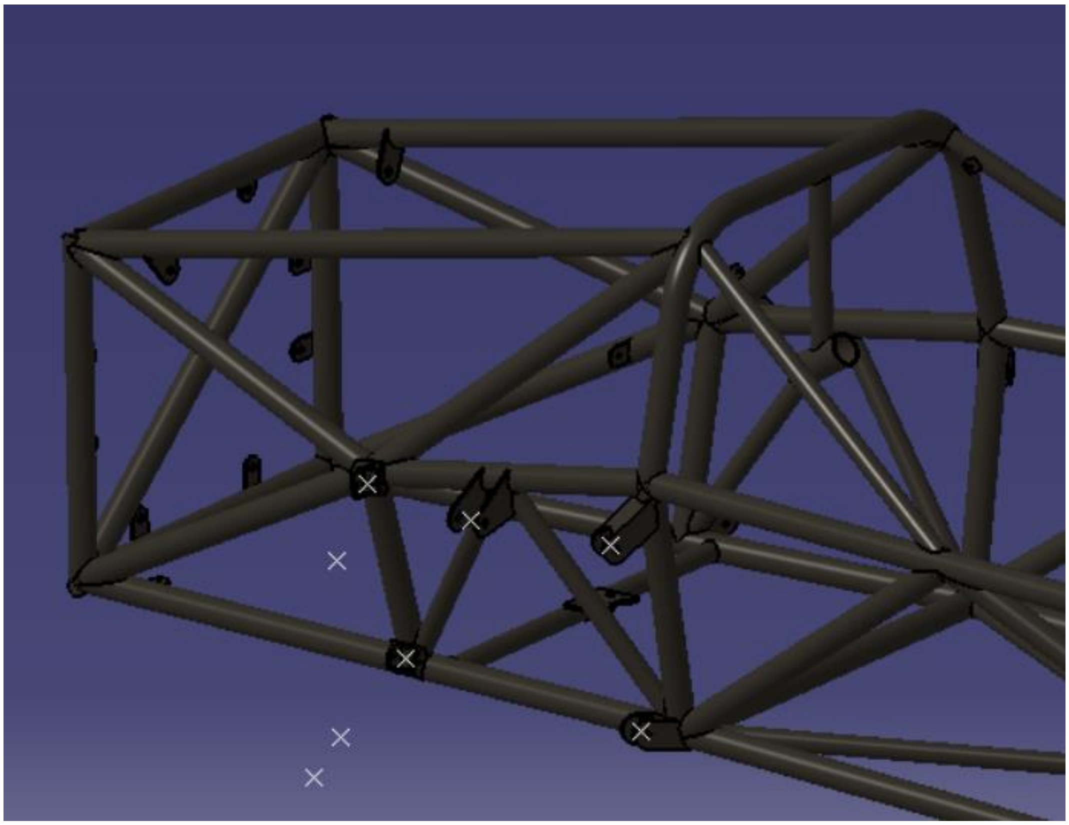
- **Suspensions points**

The first step of designing was the plotting of suspension points, the aim being to get the suspension mounts at the nodes of the chassis. This year we used direct suspension for the front as compared to previous year where push rod suspension was used .There were as many as 10+ iterations. The final suspension points are (concerned).

FRONT SUSPENSION:

X (mm)	Y (mm)	Z (mm)		
-1147.56	-221.24	129.00	POINT:1	Lower wishbone front pivot
-874.00	-245.32	129.00	POINT:2	Lower wishbone rear pivot
-1019.49	-521.93	151.00	POINT:3	Lower wishbone outer ball joint
-1141.00	-280.00	317.00	POINT:4	Upper wishbone front pivot
-877.15	-280.00	295.00	POINT:5	Upper wishbone rear pivot
-1015.14	-495.14	347.36	POINT:6	Upper wishbone outer ball joint
-1019.49	-480.00	175.00	POINT:7	Damper wishbone end
-1019.49	-313.25	285.00	POINT:8	Damper body end

With the help of front suspension points the chassis nodes of respective suspension mount was decided as follows. Considering a distance of 1.5 inch from suspension coordinate to the centre of the node. The lower member extending from the front bulkhead to the front hoop is horizontal for ease of manufacturing which contains nodes for lower wishbone pivot points. The upper wishbone rear pivot is slightly deviated from the node as there was restriction in height of upper side impact member to be within 240mm to 320mm. Hence adjustment was made in the position of the mount. With the help of front suspension points the position of the Front Hoop was decided.



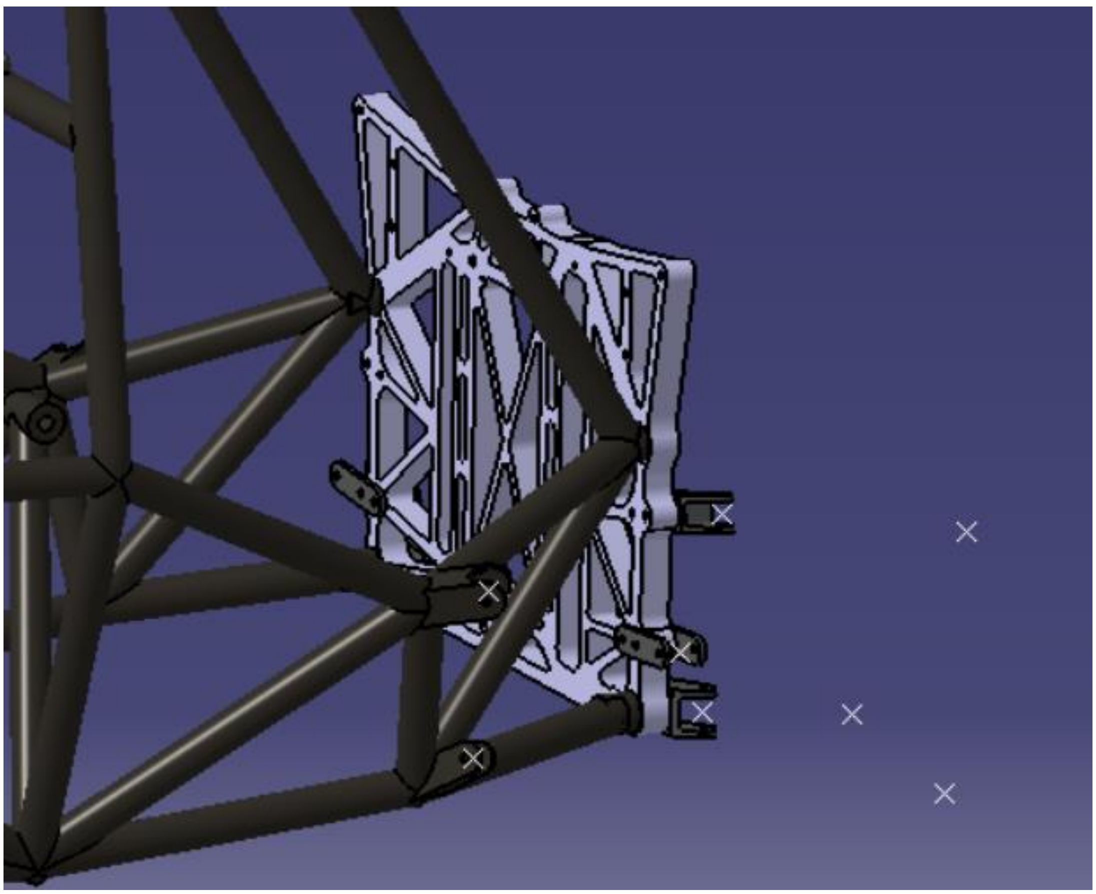
REAR SUSPENSION:

X (mm)	Y (mm)	Z (mm)		
276.82	-270.00	135.00	POINT:1	Lower wishbone front pivot
461.64	-215.00	135.00	POINT:2	Lower wishbone rear pivot
511.00	-479.00	140.72	POINT:3	Lower wishbone outer ball joint
266.99	-310.00	253.18	POINT:4	Upper wishbone front pivot
461.64	-237.30	268.63	POINT:5	Upper wishbone rear pivot
520.00	-483.47	310.73	POINT:6	Upper wishbone outer ball joint
492.98	-460.14	15 8.97	POINT:7	Push rod wishbone end
432.00	-273.99	446.50	POINT:8	Push rod rocker end
440.00	-484.06	199.82	POINT:9	Outer track rod ball joint

439.64	-220.08	181.78	POINT:10	Inner track rod ball joint
432.00	-26.40	425.00	POINT:11	Damper to body point
432.00	-223.08	461.27	POINT:12	Damper to rocker point

Similar to the front suspension the nodes for the suspension points were decided considering the length of mount required (i.e. 1.5in). For increasing the accuracy of suspension points as well as the rear components, Aluminium rear bulkhead was used.
The lower members were decided keeping in mind the position of engine so that it comes within the limits of the frame. After the engine was positioned, the mounting points of engine were fixed.

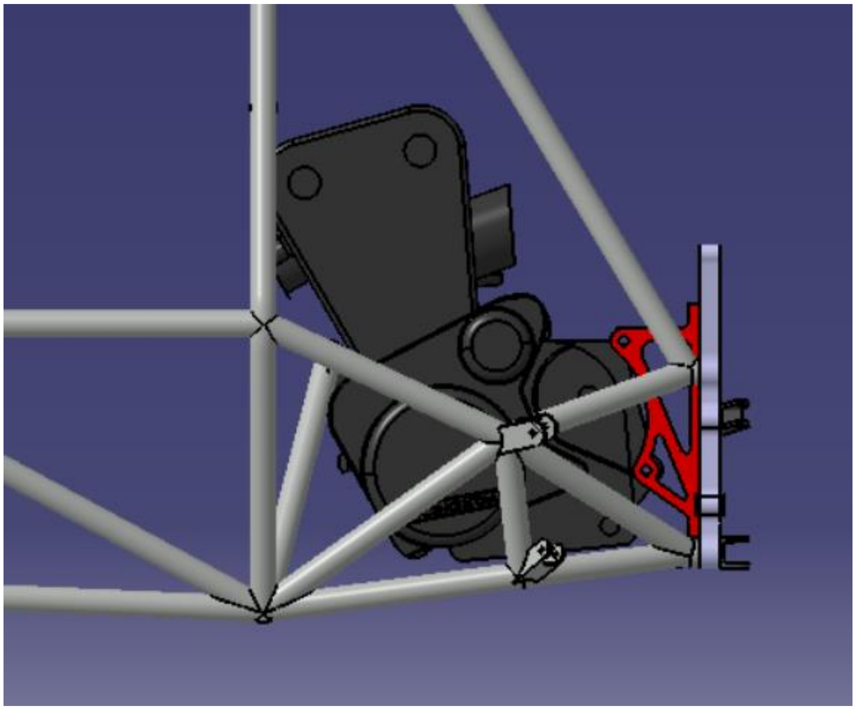
The position of the Main Hoop was decided while simulating the suspension model on LOTUS software. The plane in which the main hoop lies contains the origin of the model.



- **Engine dimensions and positioning**

Engine dimensions: KTM 390 360mm X 450mm X 290mm

Keeping in mind the position of driver seat, the engine was positioned within the limits of the chassis i.e.to lower the CG the engine was placed as low as possible within the chassis limits. After the engine was placed the engine mount position was fixed.



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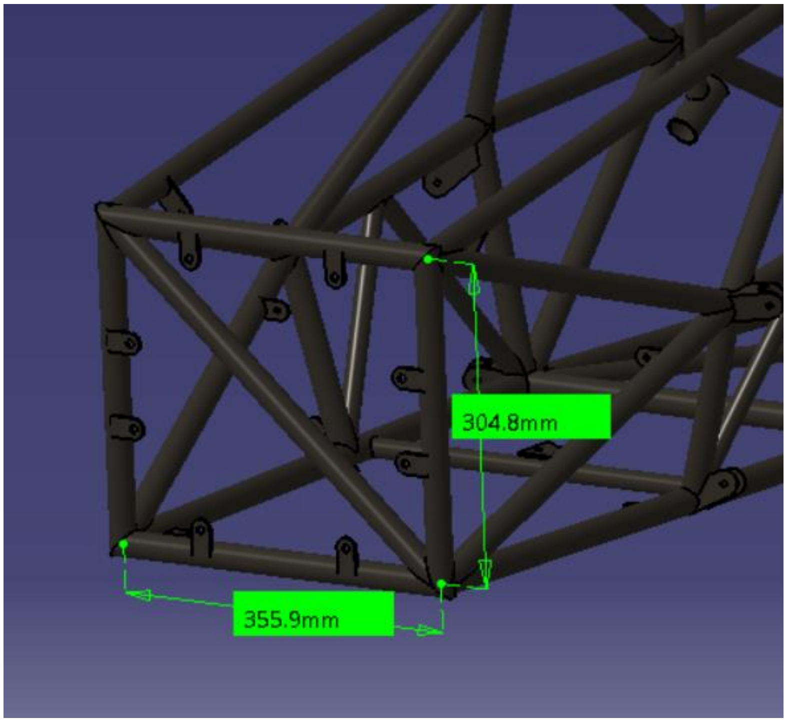
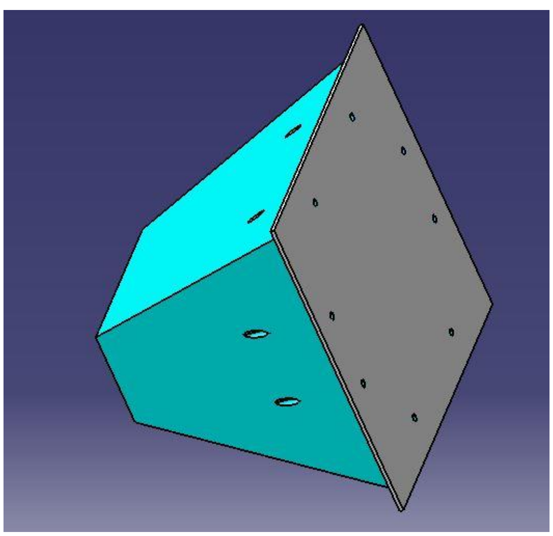
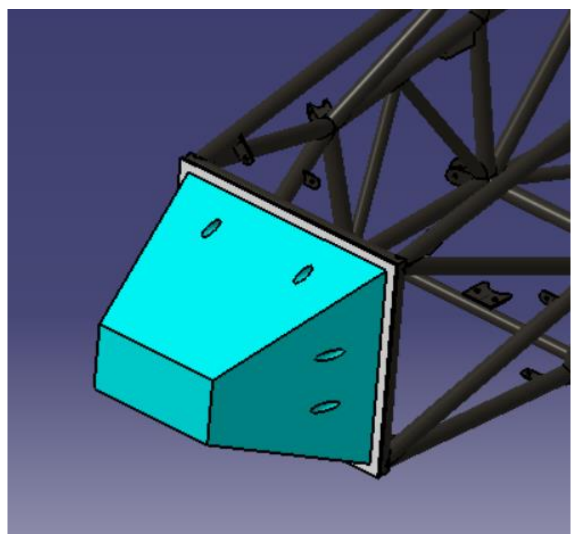
- **Impact attenuator dimensions**

The dimensions of IA are 350mm X 300mm X 260mm. The IA used was “Standard IA design” of material IMPAXX 700 Foam.

The dimensions of the AIP are similar to the previous year. So the dimensions of the Front bulkhead were also kept same







- **Driver seating position**
  The driver seating position is kept reclined similar to last year giving importance to driver ergonomics. The reclined position of driver lowered the height of CG which is one of our design goals. The 95th percentile male or Percy placement rule is shown below.

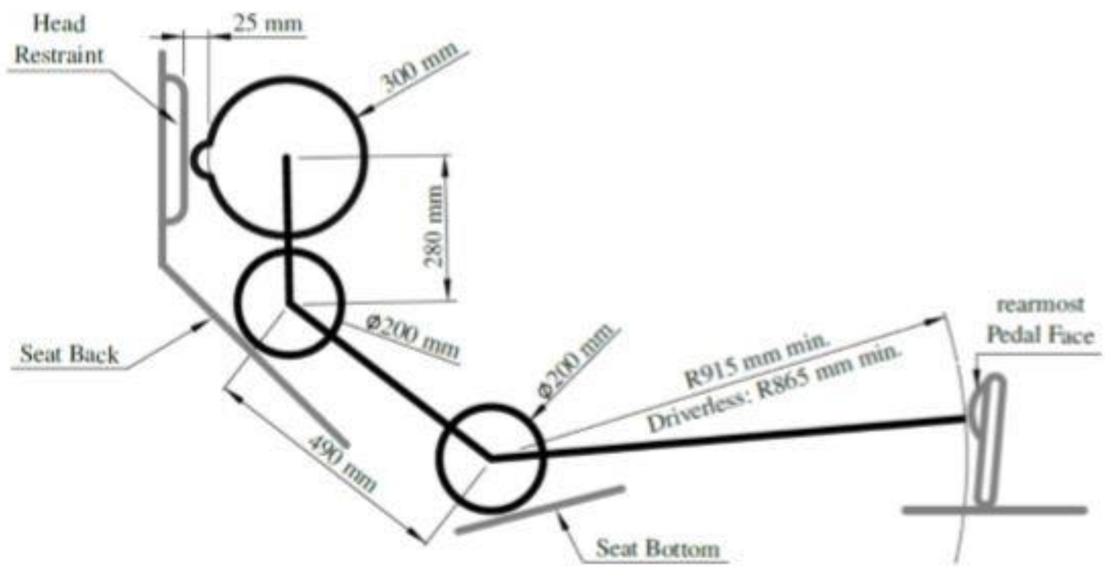
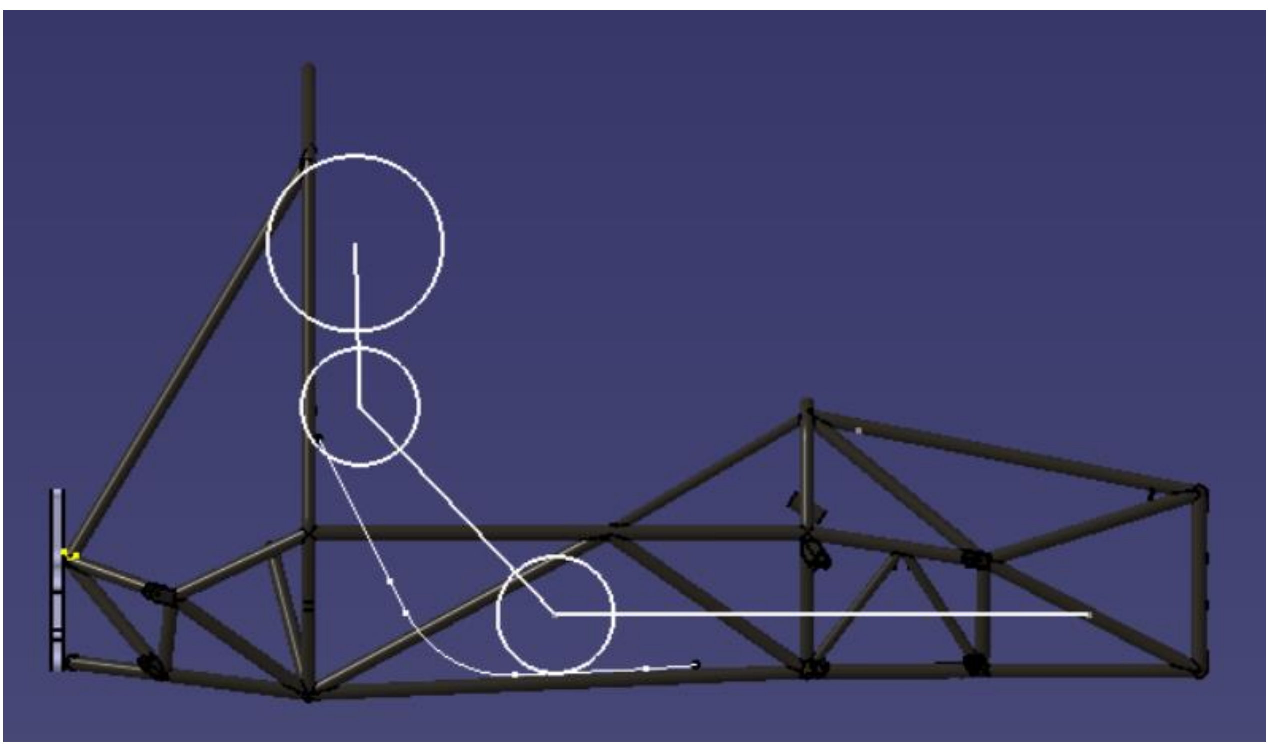
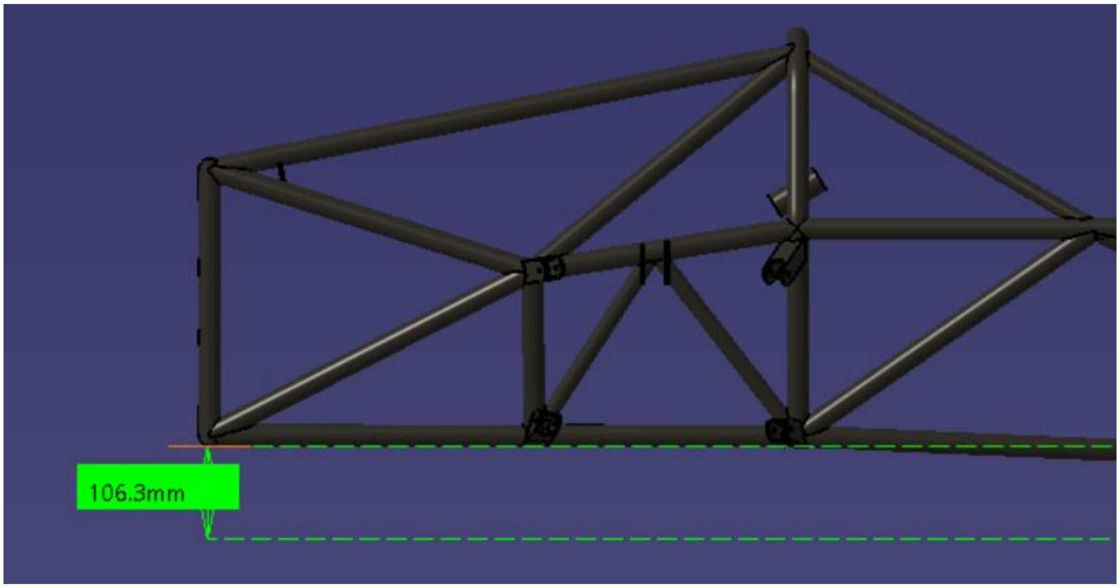


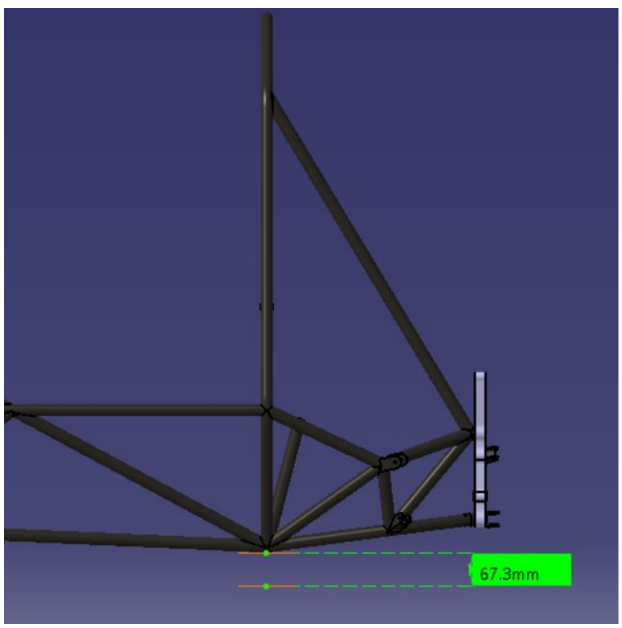
Figure 11: Percy placement



- **Ground clearance**
  1. Ground clearance at front bulkhead –
     Ground clearance at the front bulkhead was kept 106.3mm

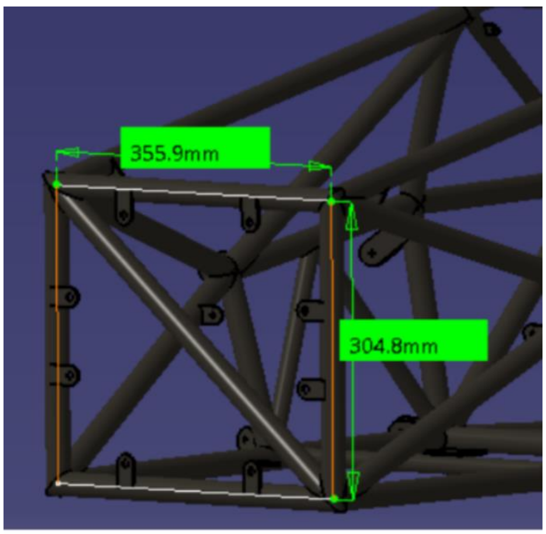


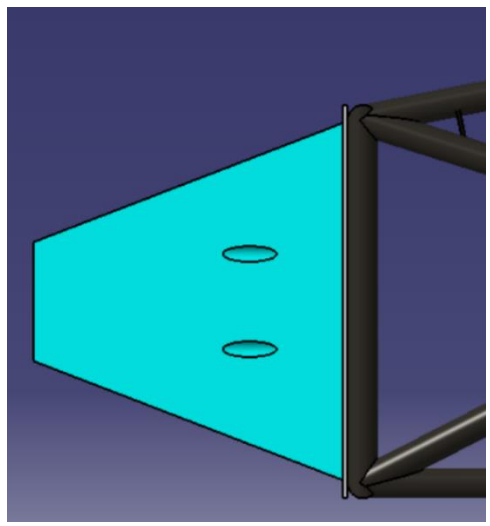
  1. Ground clearance at main hoop-
     The minimum Ground clearance to be maintained is 30mm (including the aerodynamic package). Main hoop is the lowest point on the chassis. So with a view to accommodate the aerodynamic package the ground clearance was increased from 57.3mm to 67.3mm.



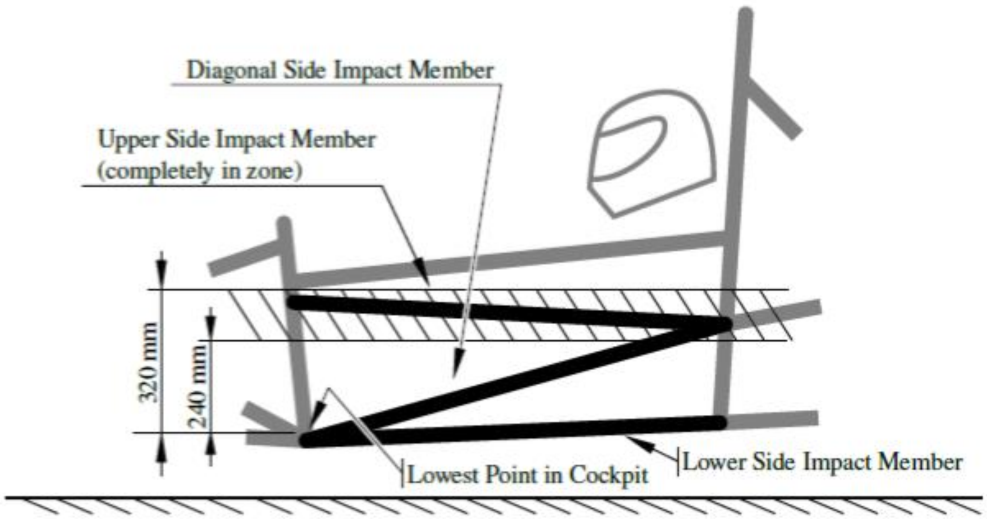
- Rule book compliance-

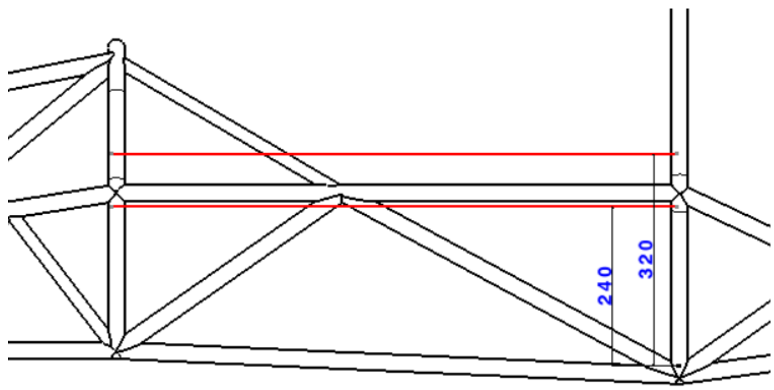
1. <u>Front bulkhead dimensions</u>- The dimensions of the front bulkhead were decided according to the dimensions of the Impact attenuator (IA) and Anti intrusion plate (AIP). As the IA used was Standard IA of material IMPAXX 700 Foam so the only change we could make was reducing the dimensions of the AIP. According to rule T3.17.3 the AIP must not extend past the outside edges of the front bulkhead. So according to the dimensions of the AIP dimensions of FBH were decided.



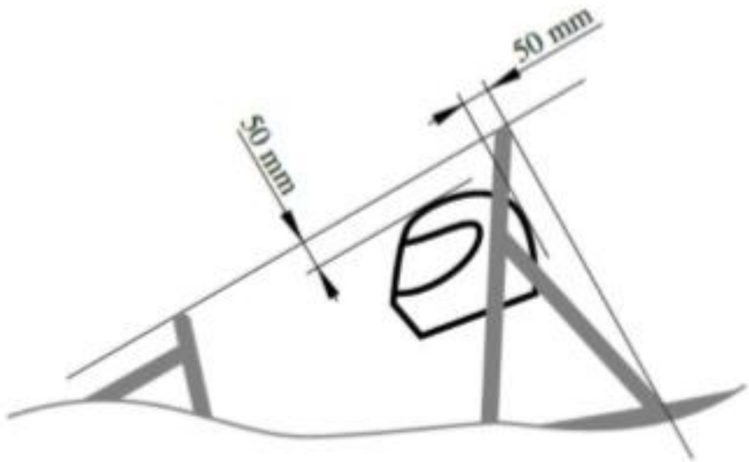


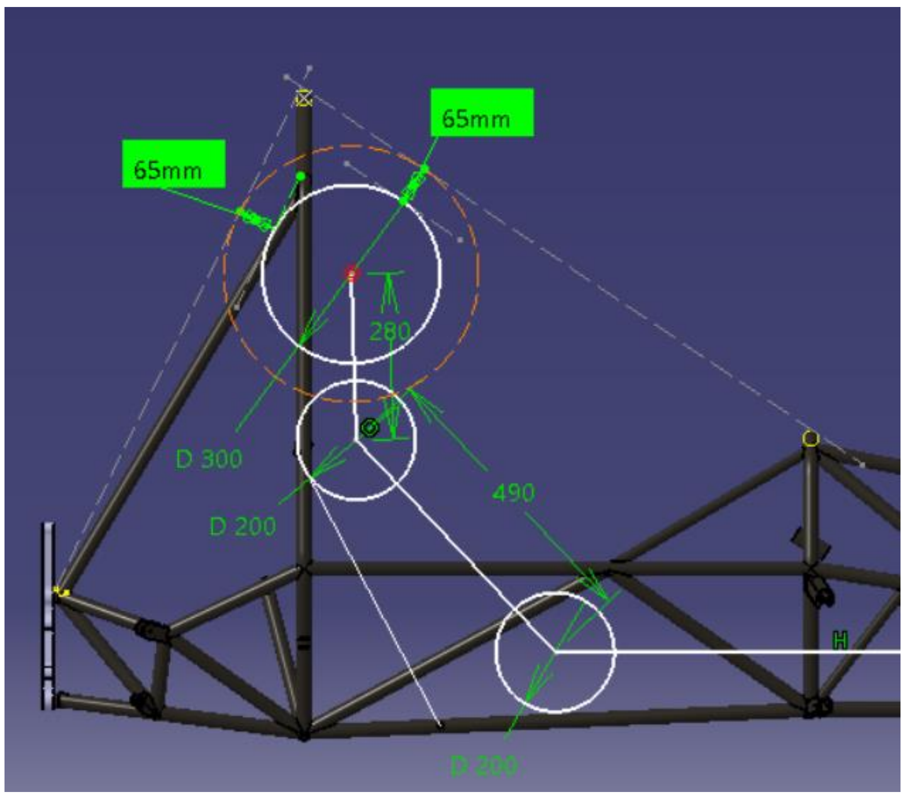
2. <u>Side Impact Sructure</u>- According to rule, the side impact structure must consist of at least three steel tubes.
    - The upper member must connect the main hoop and the front hoop. It must be at a height between 240 mm and 320 mm above the lowest inside chassis point between the front and main hoop.
    - The lower member must connect the bottom of the main hoop and the bottom of the front hoop.
    - The diagonal member must triangulate the upper and lower member between the roll hoops node-to-node.





3. <u>Main hoop height</u>- The main hoop height was decided placing the Percy and in accordance with the rule T3.3.1
   When seated normally and restrained by the driver's restraint system, the helmet of a 95th percentile male and all of the team's drivers must-
   - Be a minimum of 50 mm away from the straight line drawn from the top of the main hoop to the top of the front hoop.
   - Be a minimum of 50 mm away from the straight line drawn from the top of the main hoop to the lower end of the main hoop bracing if the bracing extends rearwards.





After the height of main hoop was decided the MH bracing position was decided in accordance with rule T2.11

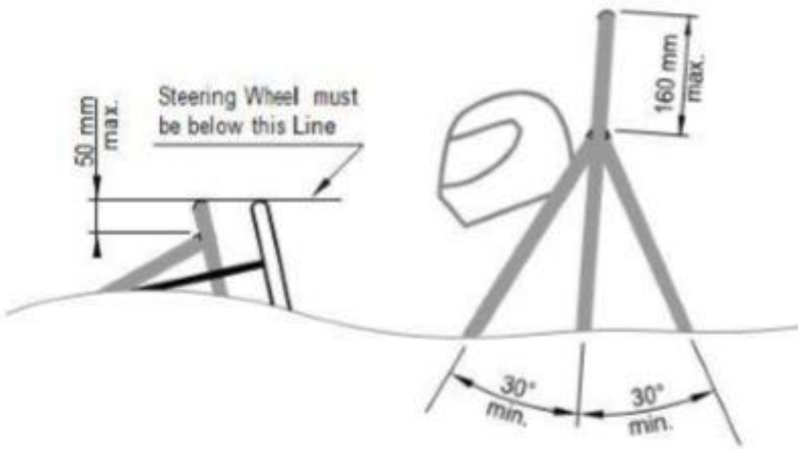


Figure 3: Front Hoop Bracing, Main Hoop Bracing and Steering Wheel Requirements

- Thickness of pipes

  We used chassis members of different thickness keeping in mind the rules explained in rulebook as shown,

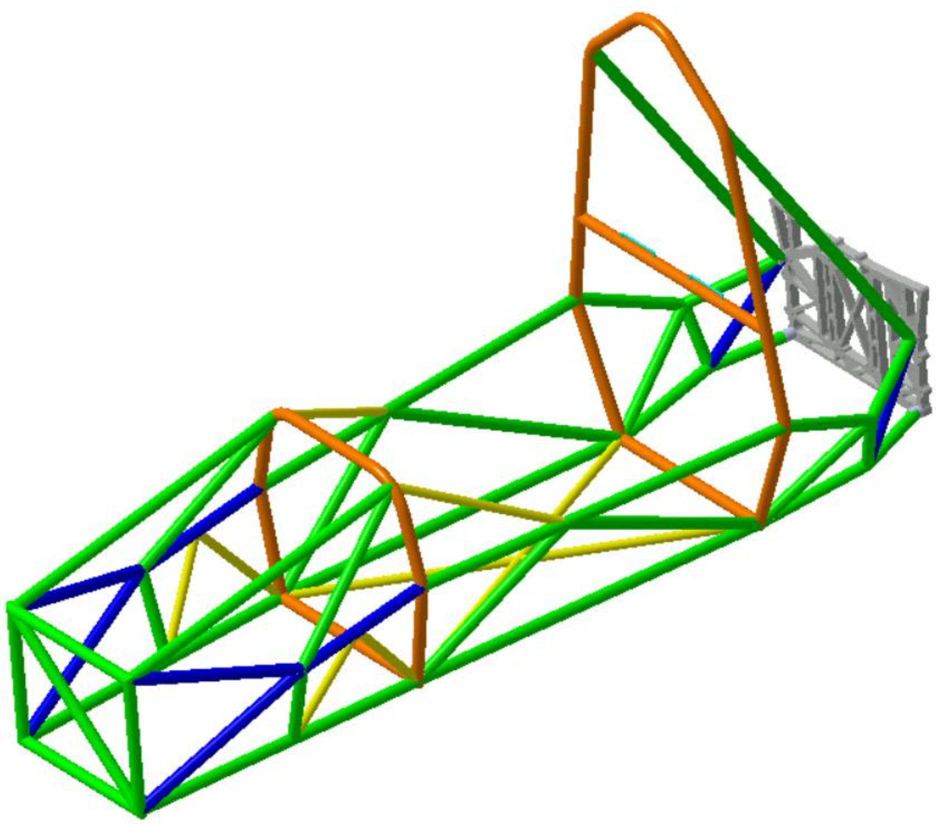
Item or Application	Minimum Wall Thickness	Minimum area moment of inertia
Main and front hoops, shoulder harness mounting bar	2.0 mm	11320 mm^4
Side impact structure, front bulkhead, roll hoop bracing, driver's restraint harness attachment (except as noted above) EV: Accumulator protection structure	1.2 mm	8509 mm^4
Front bulkhead support, main hoop bracing supports EV: Tractive system components	1.2 mm	6695 mm^4

Table 4: Minimum Material Requirements

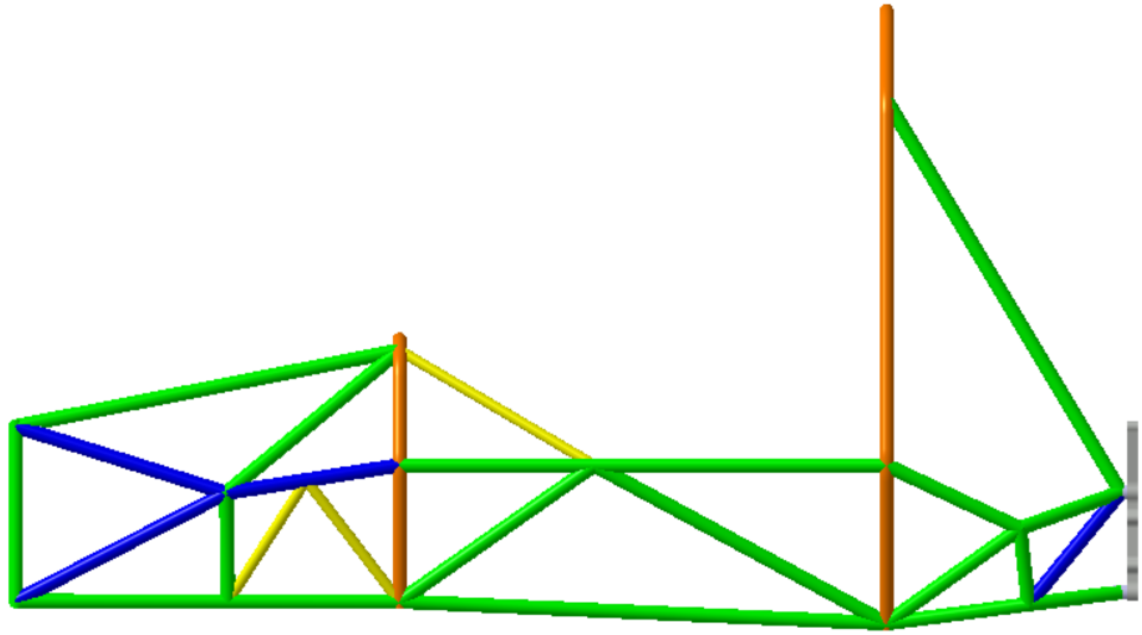
Hence, chassis members selected are as below,

	Orange	Steel Tubing 25.4x2.5mm
	Green	Steel Tubing 25.4x1.65mm
	Blue	Steel Tubing 25.4x1.25mm
	Yellow	Steel Tubing 19x2mm
LEGEND		

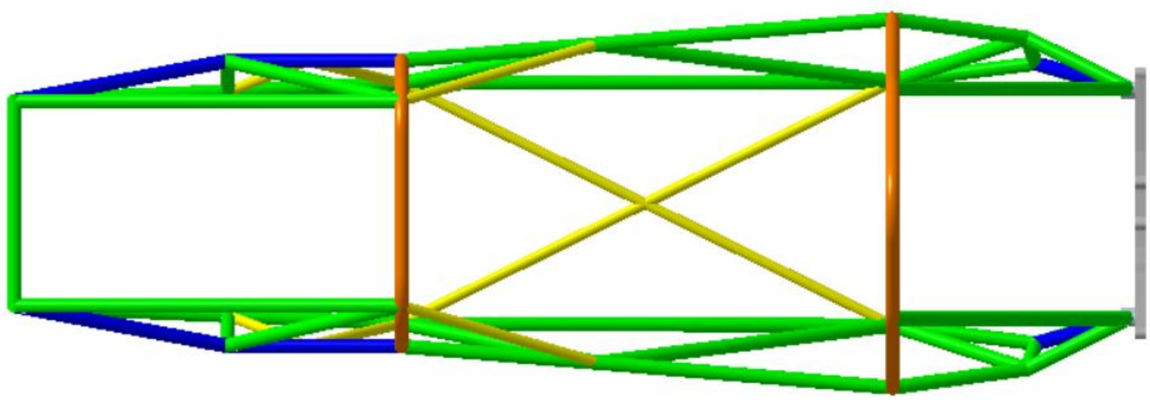
And following are different views showing all members with their respective thickness,



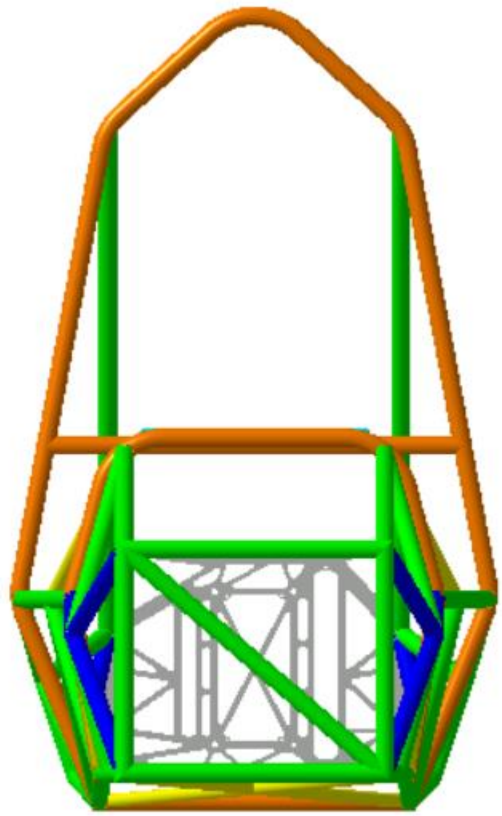
Isometric view



Side view



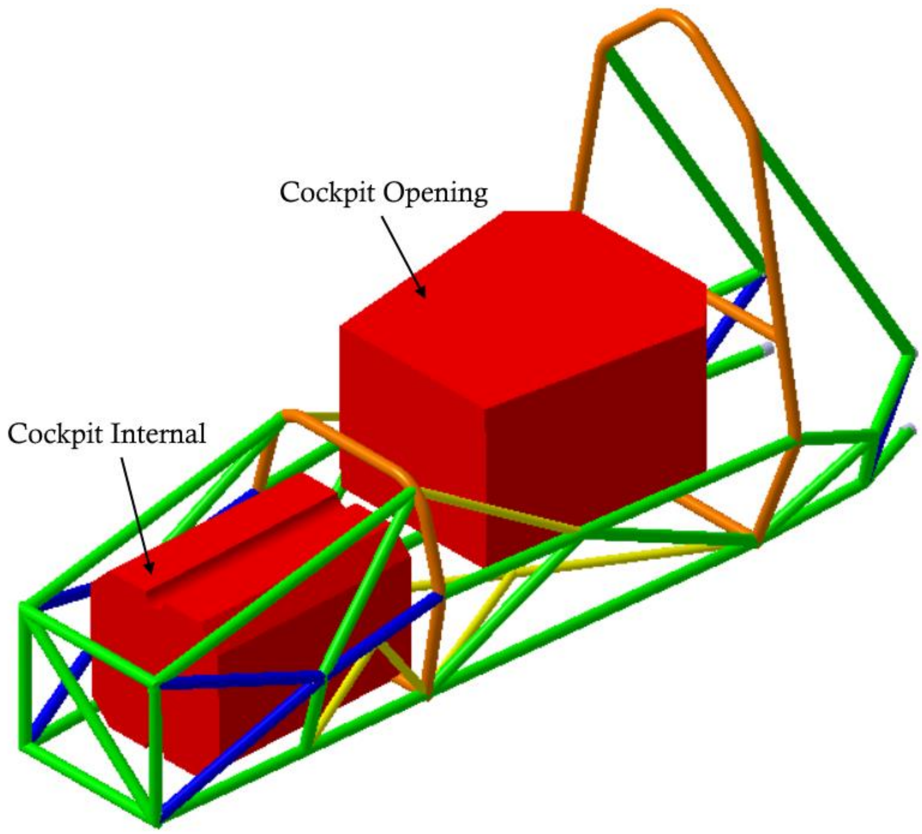
Top view



Front view

- Template compliance check

  The figure below shows template of internal cross section and cockpit opening which had to be considered while designing the extent dimension of the Front Hoop and distance between the two roll hoops



RBH

Rear Bulk Head: Rear part frame or Rear Bulkhead is the rearmost chassis structure that usually supports the drivetrain assembly and other connected components.

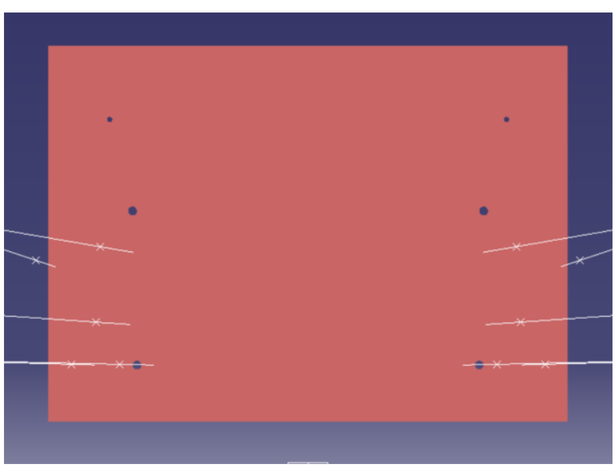
For FB19 we used a steel tube/ pipe structure for the rear part of the frame which supported components like spool mount, 4 rear suspension hard points (UCAR, toe) mount, spring / damper mount, rear engine mount.
There is a possibility of error in manufacturing due to multiple welded tubes. Compact and complex fixture requirement and difficulty in achieving accuracy in manufacturing and mounting of each of the components. Misalignment of spool mounting possible due to chain tightening uneven possibility.

For FB20 we opted for a detachable aluminium rear bulkhead having mountings of rear engine mount, spool mount, 6 rear suspension (UCAR, LCAR, Toe), rocker mount and spring/damper mount. Complete structure was CNC milled and attached to the chassis with use of four M12 bolts. CNC machining results in maximum accuracy in manufacturing. Easily mounted to the rear bushing which is welded to Chassis.

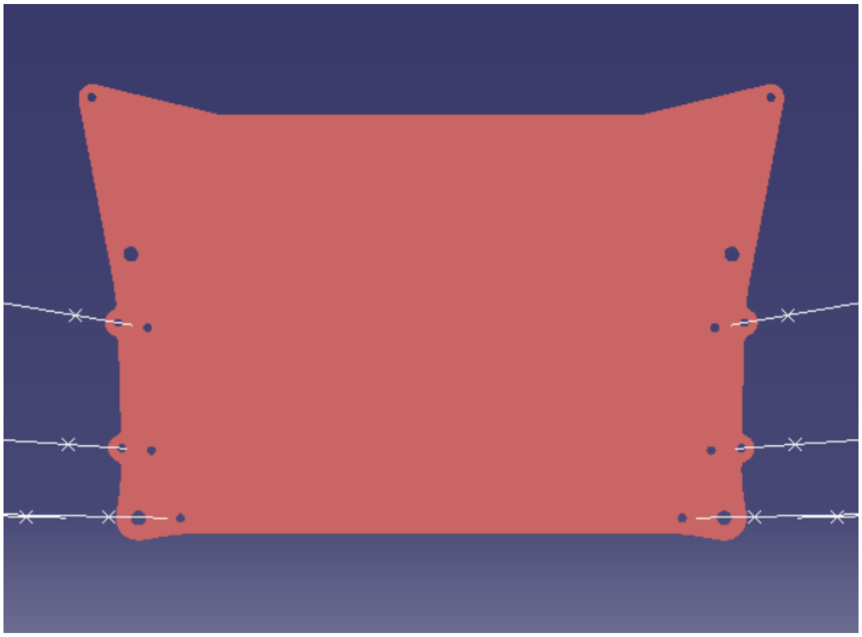
Aluminium RBH- mostly planar aluminium structure rigidly attached to the rear of the chassis frame with the help of bolts, used to mount many of the rear components like drive train assembly, suspension hard points, and suspension components, etc.

Design considerations- chassis mounting points, spool mount mounting points, engine rear mounting position, rocker axis position, spring/damper mounting points, ARB mount, sprocket and brake disc position.

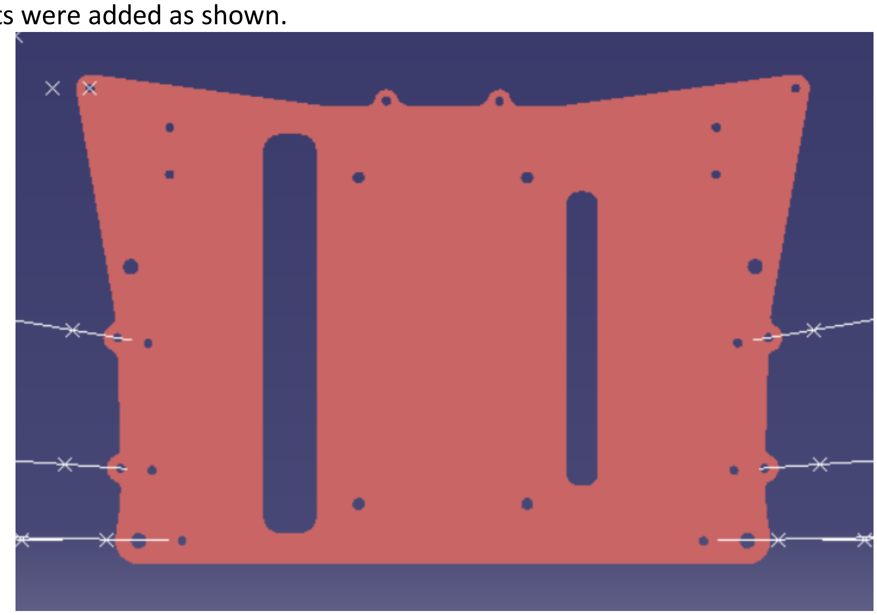
Design process-
Suspension points were plotted and the position of the wishbones (A-arms) were drawn. Chassis points were plotted, and with the help of rear suspension the position of lower chassis mounting points were fixed and the upper chassis mounting points were fixed by ensuring that the minimum angle of bracing with respect to main hoop i.e. 30 degrees is maintained. The rocker axis was plotted and the rear plane and was decided based on position of the rear suspension and drivetrain assembly. These points were located on an initial slab of 20mm aluminium, as shown in the figure below.



Based on the location of the suspension hard point locations, the mounting holes for suspension mounts were positioned. The toe mount mounting holes were common with lower chassis mounting holes. The direction of mounts was in line with the direction of control arms for efficient force transfer. The outer profile was trimmed as shown in figure.



The position of spool mount mounting points was finalised such that the it was common to the secondary engine mounts. The engine was positioned exactly in the centre line of the chassis and the slots for the Driven sprocket and brake disc were measured and positioned accordingly keeping enough clearance with side wall of RBH. The ARB points were also added after various position considerations. Damper mounts were added as shown.



Now pockets were added considering force direction and position of fixed point from the mounts which bear high loads for the purpose of weight reduction.

FEA

RBH-

Forces are going to get transmitted on Aluminium Rear Bulkhead (i.e. RBH) through the components attached on it. These forces come both in running i.e. dynamic conditions as well as static conditions. For all these conditions, we considered that, the components which transfer forces to RBH are fully rigid, i.e. they don't undergo deformation and all forces are transmitted to RBH fully as these components act only as transfer medium. Because we have designed all these components already and analysed them so that to get them safe in all conditions. Hence we assumed them to be of infinite strength so as to get worst condition forces on RBH and make sure that RBH remains safe.

Following are the possible forces discussed-

1. **Rocker & ARB -**

   It is transmitting force coming through push rod to damper and its weight is going to get applied on RBH.

2. **Toe Mount -**

   During dynamic conditions, while turning, centrifugal force acts. Also due to acceleration, bump, brakes compressive as well as tensile forces get transmitted from tiers to suspension geometry. These forces come to chassis through RBH, as all rear suspension geometry is attached to RBH through mounts. These force values and directions we determine for worst possible conditions which we got from Vehicle Dynamics (VD) department.

   These forces can act on left as well as right side of RBH at a time, hence 2 forces.

3. **UCA -**

   As discussed above, on Upper Control Arm (UCA) forces can act on left as well as right side of RBH at a time, hence 2 forces.

4. **LCA -**

   As discussed above, on Lower Control Arm (LCA) forces can act on left as well as right side of RBH at a time, hence 2 forces.

5. **Damper Spring -**

   Tensile or compressive forces are transmitted from tiers to pushrod to rocker to damper. Hence through damper, forces are transmitted to RBH. These forces also can come from left as well as right side.

6. **Forces through Spool Mount -**

   All forces coming through drive train components are transmitted to RBH through Spool Mount. Following are the forces that act on RBH through Spool Mount –

   I. *Weight of drive train components -*

      When car is not on wheels, all drive train components are suspended in air on Spool Mount, which is fixed to RBH. Hence, all the weight is carried by RBH. This is in static condition, when car is not moving. As in moving conditions, weight is carried by tiers. This force will never act simultaneously with other forces and acts individually.

   II. *Brake force -*

Brake Calliper is mounted on Spool Mount. When brakes are applied, callipers press the disc and stops it. Hence, brake torque is converted into heat by callipers. Hence, force gets applied on calliper mount which is on spool mount. This force tries to pull out the calliper which is restricted by bolts on Spool Mount which is restricted by RBH. Hence, force directly acts on the RBH. As we considered Spool Mount as a rigid component as discussed above.

III. *Load Transfer -*

At the time of jacking, or also at time of acceleration or breaking load transfers. Hence, this force needs to be applied on RBH as all these forces can be transferred to the RBH. Sin component of weight of all components behind RBH (in rear direction) will get applied on RBH which are in air. But we ignored that, as these forces are very less compared to other forces.

IV. *Impact Force -*

When we jump on jacking rod, impact force gets on the jacking rod which is attached to the Spool Mount which transfers this force to the RBH.

V. *Torque Tension in Chain -*

Due to torque on engine sprocket, tension force acts on the engine. Hence engine gets pulled to rear side. This motion is restricted by engine mounts which are held by RBH on rear side. Hence, equivalent force acts on engine mount, which we consider rigid here, and hence transfer these forces to RBH in rear side. But these resistance is given by front engine mounts also.

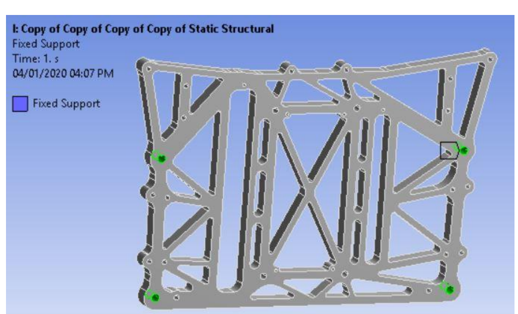
Also big sprocket is also getting pulled in front direction due to tension on chain. Due to this, sprocket, hence spool insert, hence spool, hence Spool Mount gets pulled in front direction which is restricted by RBH again. Hence we need to apply this force also on RBH considering all other components rigid.

VI. *Forces on Engine-*

Constant weight of engine and centrifugal force on engine are transmitted to RBH through rear Engine Mount.

Fixed Support -

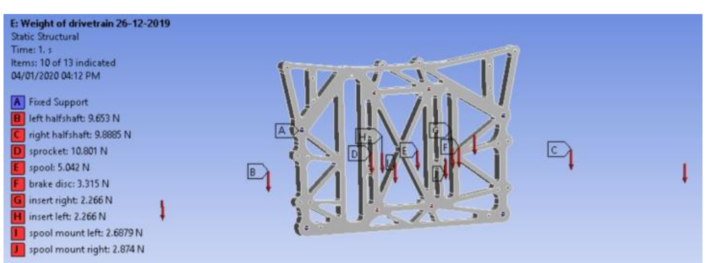
1. Four pipes coming from main hoop are welded with 20mm bushes with internal threads in it.
2. RBH is fitted at the end of these bushes by 10mm bolts.
3. These four holes drilled on RBH act as a fixed support, as it is holding the RBH in fixed plane.
4. Hence, in all conditions, for analysis these four holes act as fixed support.

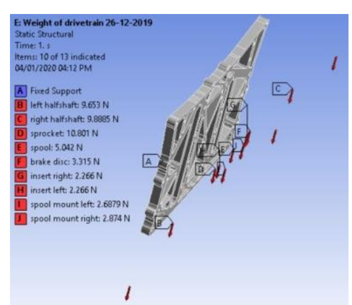


Weight of Drive Train components -

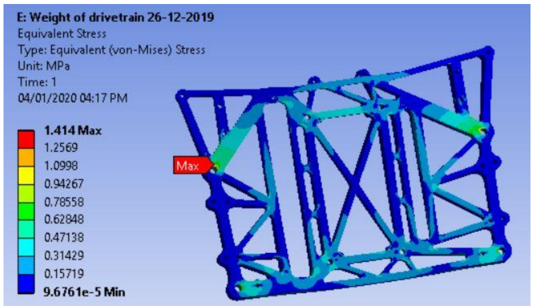
1) We considered drive train assembly as a beam of different cross sections and of different materials. This assembly consists of Half Shafts (2), Sprocket (1), Spool (1), Brake Disc (1), Inserts (2), Spool Mounts (2), Knuckle (2), Rear Calliper (1), Suspension Geometry etc.
2) These all weights are held by Spool Mounts
3) As there shape and materials (of all these components) are different, it is kind of UVL and not UDL (uniformly varying load and not uniformly distributing load. Rather sometimes it is non-uniformly varying load)
4) We calculated weight of each component and assumed it is acting at C.G.
5) All C.G. must be lying on central axis as all are approximately round parts fixed over this axis and from these weights will be acting downwards.
6) These spool mounts will cause moments as well as bending at Spool Mount and as Those are like fixed support here, reactions will act at both Spool Mounts.
7) Hence, we applied Remote Forces which will be acting at the holes on the RBH where we connected Spool Mount and location of these forces will be at the C.G.
8) Though, these forces are extremely small, it may affect the RBH and will keep RBH bending gradually in fatigue. Because, this weight is not acted directly on RBH but moment is also acting due to it, as it is away from the RBH. Hence, for safety we checked that.

SR. NO.	Component	Centre of Gravity (C.G.) (mm)			Force (N)
		X	Y	Z	
1	Left Half Shaft	511.254	-303.272	245.969	9.65304
2	Right Half Shaft	511.254	297.679	247.145	9.88848
3	Sprocket	510	-97.205	265.6	10.80081
4	Spool	510	-5.85	265.6	5.04234
5	Brake Disc	510	76.146	265.6	3.31578
6	Insert Right	510	63.971	265.6	2.26611
7	Insert Left	510	-75.679	265.6	2.26611
8	Spool Mount Left	518	-54.834	246.6	2.68794
9	Spool Mount Right	513	47.854	246.6	2.87433
10	Knuckle Right	519.3	518.22	209.98	6.77871
11	Knuckle Left	519.3	-518.22	209.98	6.77871
12	Rear Calliper	561	82.343	291.072	6.52365

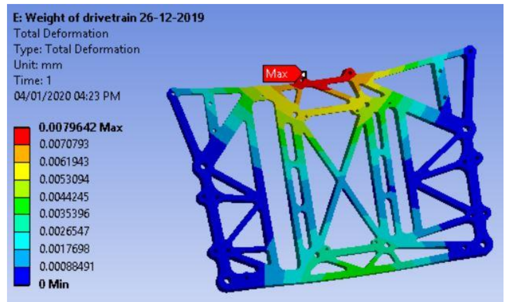




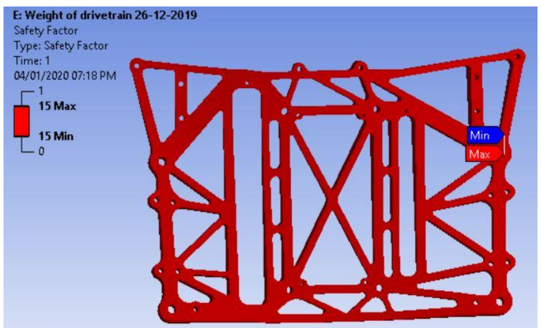
Results -



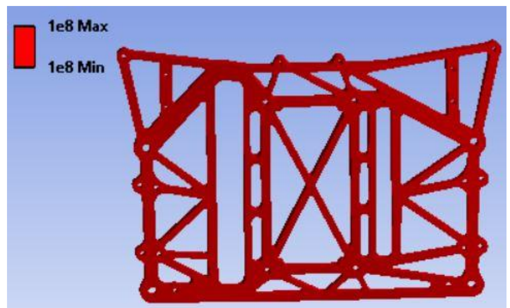
Equivalent Stress



Total Deformation



Factor of Safety



Fatigue Life

SR	**Quantity**	**Value**
1	Maximum equivalent stress	1.414 MPa
2	Maximum total deformation	0.00796 mm
3	Minimum safety factor	15
4	Minimum fatigue life	1e8

Impact Force-

1. When we jump on jacking rod, we assumed that person jumps from certain height and he has some potential energy on that height.
2. This is converted into spring energy having three springs as two of damper and one RBH itself.
3. From that we calculated equivalent force as,

$K = 52\ N/mm \rightarrow$ stiffness of spring (damper)

$x = 25\ mm$

$$mgh = \frac{1}{2}K'\delta^2 + \frac{1}{2}Kx^2 + \frac{1}{2}Kx^2$$

K' = stiffness of the RBH.

F (N)	δ (mm)	$\therefore K' = F/\delta$ (N/mm)
1) 50,000 N	5.952	8400.537634
2) 1 N	1.175×10^{-4}	8510.638298
3) 100 N	1.1901×10^{-2}	8402.655239

$\therefore K'_{average} = 8437.943724\ N/mm$

as, $K' = \frac{F}{\delta}$ $\therefore \delta = \frac{F}{K'}$

$$\therefore mgh = \frac{1}{2}K' \times \left(\frac{F}{K'}\right)^2 + \frac{1}{2}Kx^2 + \frac{1}{2}Kx^2$$

$$\therefore mgh = \frac{1}{2}\frac{F^2}{K'} + \frac{1}{2}Kx^2 + \frac{1}{2}Kx^2$$

$$\therefore mgh = \frac{1}{2}\frac{F^2}{K'} + Kx^2$$

$$\therefore \frac{1}{2}\frac{F^2}{K'} = mgh - Kx^2$$

$$\therefore F^2 = 2K'(mgh - Kx^2)$$

$$= 2 \times 8437.943724 \times \frac{N}{mm} \times \left[100\ kg \times 9.81 \times 70\ mm - 52\frac{N}{mm} \times (25\ mm)^2\right]$$

$$\therefore F^2 = 2 \times 8437.943724 \times \frac{N}{mm} \times \left[100\ kg \times 9.81\frac{m}{s^2} \times 70\ mm - 52\frac{N}{mm} \times (25\ mm)^2\right]$$

$$= \cancel{2\times}\ 16875.88745\frac{N}{mm} \times \left[981\ N \times 70\ mm - 52 \times (25)^2\ Nmm\right]$$

$$= 16875.88745\frac{N}{mm}\left[68670\ N.mm - 32500\ N.mm\right]$$

4. We assumed that this force acts on the centre of jacking rod, which will be transmitted to RBH through Spool Mount.

$\therefore F^2 = 16875.88745 \frac{N}{mm} \times 36170 \ N \cdot mm$

$\therefore F^2 = 610400849.1 \ N^2$

$\therefore F = 24706.29169 \ N$

But, for any stiffness of spring, after full compression of spring, then only RBH is goint to deform.

We considered, that in any case RBH is move downward at least 30mm (BCm) for worst least stiffness.

Hence,

F (N)	δ (mm)	δ + 30mm	$\therefore K' = F/\delta$ (N/mm)
50,000N	5.952	35.952	1390.7432

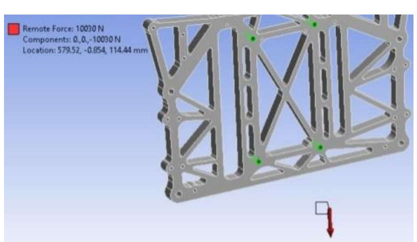
$\therefore F^2 = 1390.7432 \frac{N}{mm} \times 36170 \times 2$

$\therefore F^2 = 100606363.088 \ N^2$

$\therefore \boxed{F = 10030.2723 \ N}$

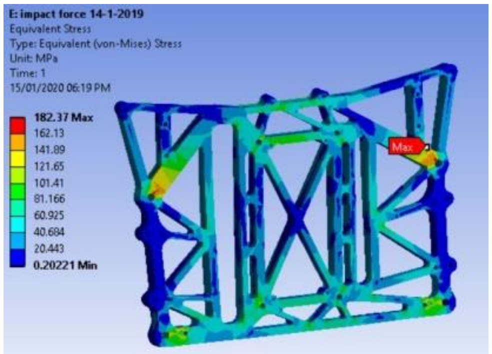
Location $\rightarrow$ (579.52, −0.854, 114.435)

5. Hence, we applied remote force on the Spool Mount holes of a RBH, at the centre of a jacking rod and in the downward direction.

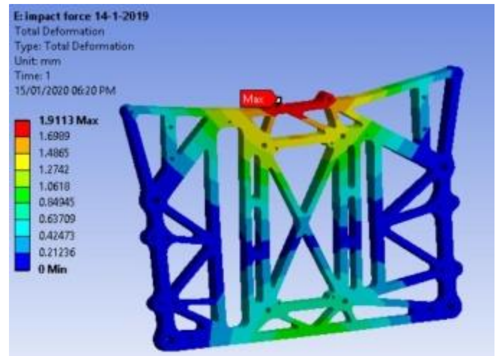


X Coordinate	579.52 mm
Y Coordinate	-0.854 mm
Z Coordinate	114.44 mm
Location	Click to Change
Definition	
Type	Remote Force
Define By	Components
X Component	0. N (ramped)
Y Component	0. N (ramped)
Z Component	-10030 N (ramped)

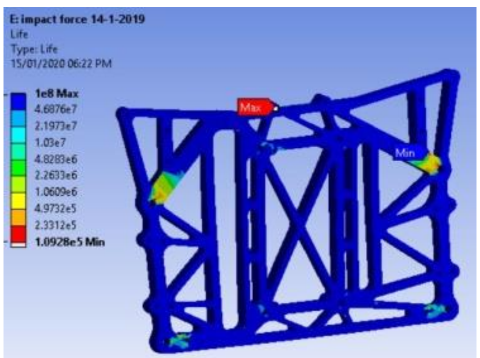
Results-



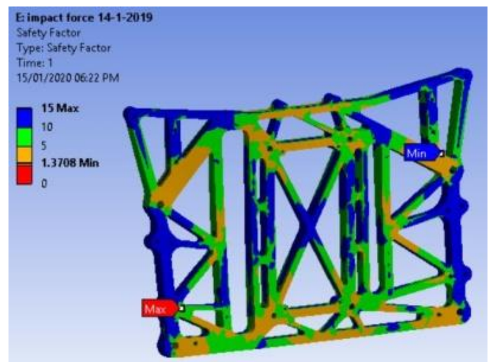
Equivalent Stress



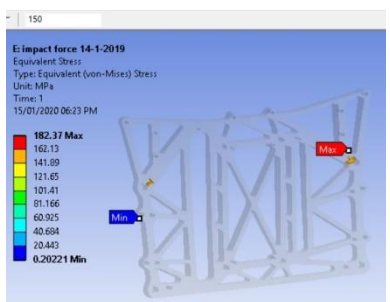
Total deformation



Factor of Safety



Fatigue Life



Singularity

SR	Quantity	Value
1	Maximum equivalent stress	182.37 MPa
2	Maximum total deformation	1.9113 mm
3	Minimum safety factor	1.3708
4	Minimum fatigue life	1.0928e5

Forces on Suspension geometry -

Abbreviation	Full form
TR	Tie Rod
LCAF	Lower Control Arm Front
LCAR	Lower Control Arm Rear
UCAF	Upper Control Arm Front
UCAR	Upper Control Arm Rear
PR	Push Rod

Unit vector for each of the front suspension members

	Tie Rod	LCA (F)	LCA (R)	UCA (F)	UCA (R)	Push Rod
Unit Vector(X)	-0.00025	0.38239	-0.47508	0.5103	-0.5347	0
Unit Vector(Y)	-0.9997	-0.9239	-0.87957	-0.8576	-0.8409	-0.79814
Unit Vector(Z)	0.02149	-0.0031	0.0256	0.0640	0.08239	-0.60247

Unit vector for each of the rear suspension members.

	Tie Rod	LCA(F)	LCA (R)	UCA (F)	UCA (R)	Push Rod
Unit Vector(X)	0.0329	0.737	0.165	0.807	0.216	0.174
Unit Vector(Y)	-0.997	-0.675	-0.986	-0.560	-0.962	-0.528
Unit Vector(Z)	0.0696	0.0184	0.021	0.185	0.164	-0.831

Front Forces

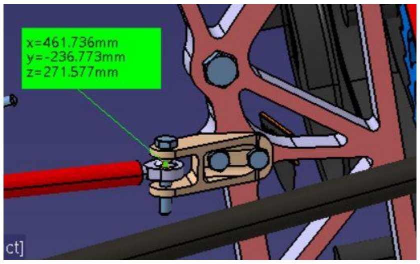
Tie Rod Force (kg)	1065.5593
LCA [F] Force (kg)	1938.2667
LCA [R] Force (kg)	-414.2302
UCA [F] Force (kg)	-738.2682
UCA [R] Force (kg)	-555.5835
Push Rod Force (kg)	-736.7957

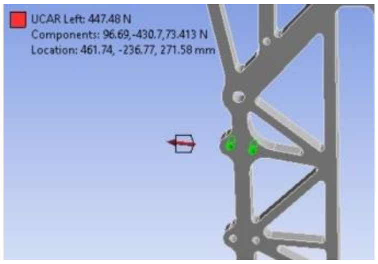
Rear Forces

Tie Rod Force (kg)	493.9619
LCA [F] Force (kg)	1851.9416
LCA [R] Force (kg)	-1798.1337
UCA [F] Force (kg)	-784.5920
UCA [R] Force (kg)	447.6402
Push Rod Force (kg)	-1395.7982

UCAR -

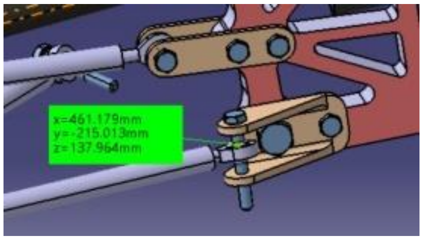
1. The force coming from UCAR is transmitted to RBH through the mount of same.
2. Hence, we applied remote force applied on holes of the RBH, location is connection of UCAR and its mount (considering mount as fully rigid) and its magnitude is governed by above table as we have unit vector and maximum force coming on it.

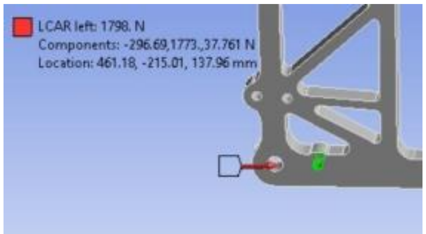




LCAR-

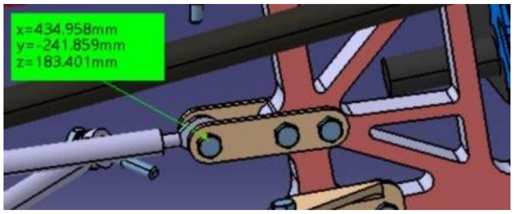
1. The force coming from LCAR is transmitted to RBH through the mount of same.
2. Hence, we applied remote force applied on holes of the RBH, location is connection of LCAR and its mount (considering mount as fully rigid) and its magnitude is governed by above table as we have unit vector and maximum force coming on it.

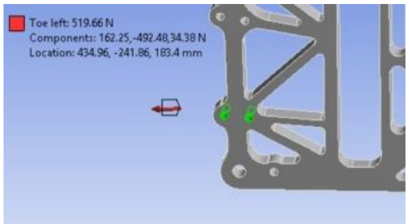




Toe-

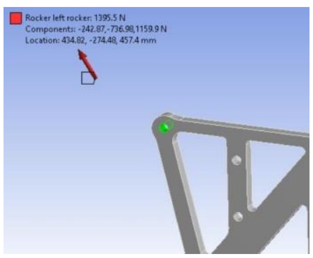
1. The force coming from Tie Rod is transmitted to RBH through the mount of same.
2. Hence, we applied remote force applied on holes of the RBH, location is connection of Tie Rod and its mount (considering mount as fully rigid) and its magnitude is governed by above table as we have unit vector and maximum force coming on it.

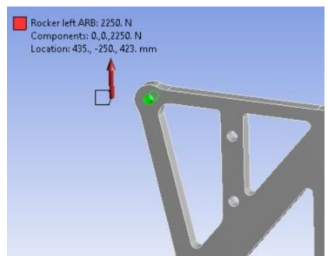


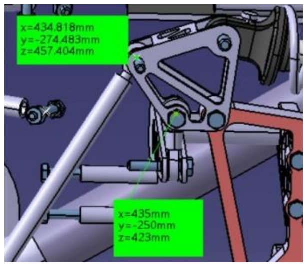


Rocker & ARB-

1. During bump, force comes on the rocker through pushrod, which tries to pull out rocker in the direction of pushrod.
2. From VD department, using tire data, we got the force coming from pushrod. We got magnitude as well as its unit vector. Hence, we got the components of forces in x, y, & z directions.
3. Considering pushrod as fully rigid, these forces will act on the rocker directly.
4. For worst condition, though rocker is not fixed in all directions and is free to move in direction of damper, we considered it to be fix and this condition is partially possible, when spring is made very stiff. (Note that this worst condition is for designing of rocker)
5. For RBH analysis, we considered rocker to be completely rigid.
6. Hence, force as well as moment will be acting due to distance equal to dimension of the rocker.
7. Hence, we applied remote force, which will act on the hole of the rocker where it is connected to pushrod, its magnitude is governed by tire data and its location of acting will be on RBH hole where rocker pivots about bolt.
8. We have the unit vector of pushrod and magnitude of force coming from pushrod by VD department. Hence, we got components of forces acting as well as location
9. Also, when bump comes on another side rear tire, due to Anti Roll Bar (ARB) force gets transformed on opposite rocker.
10. This force directly acts in vertical (z) direction on rocker, hence again considering rocker as a rigid body, remote force is applied on hole of the rocker where it is connected to ARB, its magnitude is 2500N (We got this from VD department) and its location of acting will be on RBH hole where rocker pivots about bolt.
11. Note that, these forces won't be acting simultaneously as bump force cannot come at a time on both rear tires. Hence each force will act in different situations.
12. Also constant weight of rocker will be acted on RBH but as this force will be very less and it won't be causing any moment on RBH, hence we ignored this force.

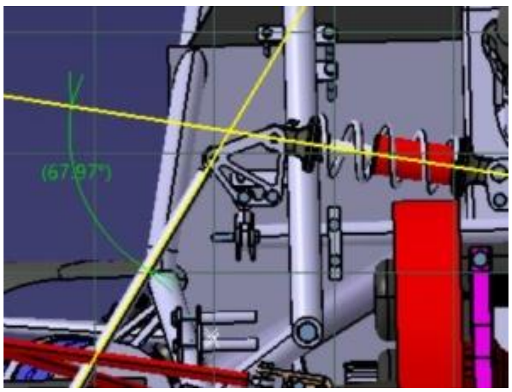


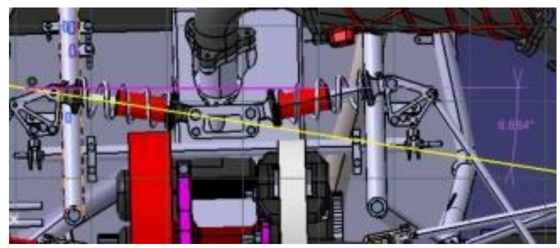




Damper-

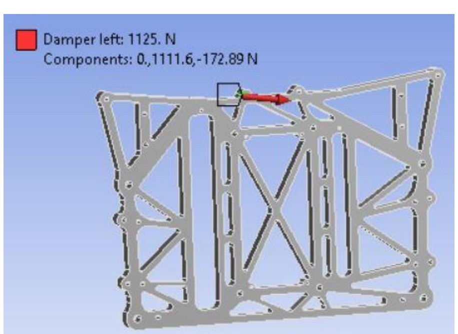
1. Due to force coming from pushrod during bump, rocker rotates and transmits the force to damper spring.
2. This force compresses the spring and hence RBH. While retraction of the spring, tensile force acts on the spring and hence RBH.
3. We know,
   Stiffness of spring as 45N/mm and maximum deformation as 25 mm.
   Force = 45*25 = 1125 N.
   Hence we found its component along the spring as,





4. Its component in +ve Y direction is 1125 * cos(8.884) = 1111.6365 N

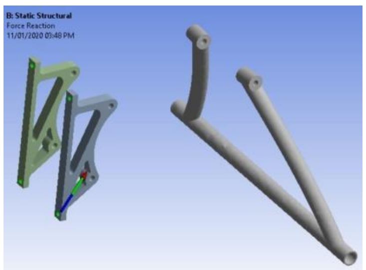
   Its component in –ve Z direction is 1125 * sin(8.884) = 172.8851 N
5. As Spring (MS) and RBH (Aluminium) are made up of ductile materials and as they are weak in compression than in tension, hence we considered forces on RBH while compression of the spring.
   Therefore we applied forces on RBH as,



X Component	0. N (ramped)
Y Component	1111.6 N (ramped)
Z Component	-172.89 N (ramped)

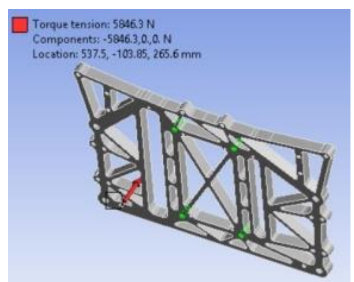
Tension in Chain -

1. Torque on drive train sprocket is 680 Nm and pitch circle radius of sprocket is 0.1163135 m (dia- 232.627mm).
2. Hence, T1 and T2 tension acts in the chain on tight and slack side respectively of a chain, hence engine is pulled in rear direction by equivalent of (T1 – T2) force.
3. Neglecting the slack side (T2) tension, force acting on chain comes out to be 5846.2689 N in rear direction which will try to pull engine in rear direction.
4. This is restricted by engine mounts both in front (AISI 1018) and rear (Aluminium 6082), hence reactions will act due to force and it causing moments at fixed supports.
5. Rear engine mounts (Aluminium 6082) are fixed to RBH, hence the reaction forces that are induced will act on RBH in opposite directions of reactions.



Maximum Value Over Time	
X Axis	-3162.4 N
Y Axis	-2202.1 N
Z Axis	799.04 N
Total	3935.6 N

6. Hence, force was applied on engine mount holes on RBH in rear direction.



7. Also, rear sprocket is pulled by same force (T1 - T2) in front direction. Though this force is pulling through area in contact of chain by UVL but we considered to be acting as a point load, from rearmost point on the rear sprocket i.e. at middle of area of contact of chain and sprocket.
8. Hence, it will try to pull sprocket in front direction, along with drive train assembly, hence spool mount.
9. But again it is restricted by bolts on RBH. Hence, we applied remote force on the holes of attachment of spool mount on RBH, its location is rear most point of sprocket and direction will be in front direction.
10. Hence these two forces, one from engine mount and other from spool mount will try to compress RBH but one being greater force and there location of action being different they don’t cancel each other.

Forces on Engine –

1. Constant weight of engine is held by engine mounts. Hence 1g force in downward direction.
2. Also, we assumed bump force on engine in worst condition as 3g. Hence total 4g force acts in the downward direction.
3. Centrifugal force acts in the lateral direction which assumed as 2g.
4. These forces act directly on front (MS) and rear (Aluminium) engine mounts.
5. Hence, we found the reaction coming on Aluminium mounts due to these forces. As it is possible to have bump force coming while turning.
6. Hence,

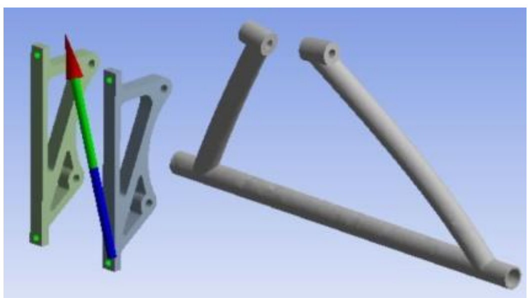
   4g = 4*40*9.81

   4g = 1569.6N

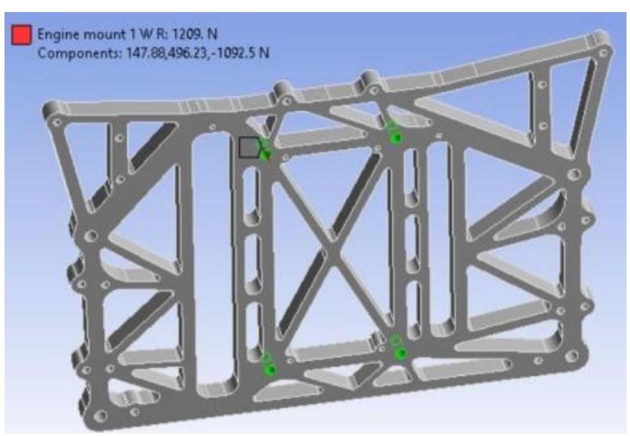
   This acts in negative Z direction.
7. 2g = 2*40*9.81

   2g = 784.8 N

   This acts in Y direction depending on left or right turn.

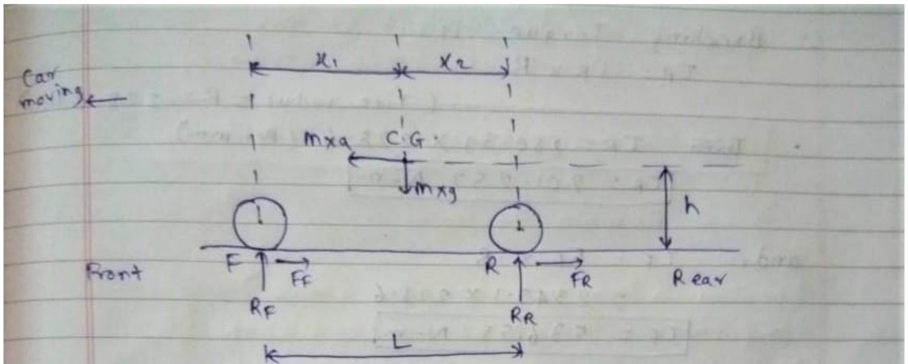


X Axis	-147.88 N
Y Axis	-496.23 N
Z Axis	1092.5 N
Total	1209. N



Brake Force -

1) We know the maximum torque needed to apply on the rear brake disc, to stop it.
2) It is stopped by rear calliper, by pressing piston against the rotating disc. Calliper is mounted in such a way that piston and disc has common tangent as shown.
3) So piston needs to apply equivalent braking torque in opposite direction. Due to this, force acts on the piston. But, this acts at the centre of piston and not at the tip of the disc, even though there is not much distance between centre of piston and tip of radius.
4) But, for getting more worst and realistic condition, we calculate force at centre of the piston as shown.



Where, for our vehicle,

$x_1 = 795.08$ mm
$x_2 = 733.92$ mm
$L = 1529$ mm
$h = 270$ mm
$a = 1.4g$
$\mu = 1.4$

Taking $\Sigma F_y = 0$

$\therefore R_F + R_R = m \times g$

$R_F + R_R = 235 \times 9.81 = 2305.85$ N ... (1)

Taking $\Sigma M_F = 0$

$\therefore m \times a \times h - m \times g \times x_1 + R_R \times L = 0$

$\therefore 235 \times 1.4 \times 9.81 \times 270 - 235 \times 9.81 \times 795.08 + R_R \times 1529 = 0$

$\therefore R_R = 628.85$ N ... (2)

$\therefore$ From eq^n (1) & (2),

$R_F = 1676.5$ N

$\therefore F_F = \mu \times R_F = 1.4 \times 1676.5 = 2347.1$ N

$F_R = \mu \times R_R = 1.4 \times 628.85 = 880.39$ N

∴ Breaking Torque is

$\tau_R = F_R \times R$

... (Tire radius = R = 228.6 mm)

∴ ~~Brea~~ $\tau_R = 880.39 \times 228.6$ (N-mm)

∴ $\boxed{\tau_R = 201.257 \text{ N-m}}$

and, $\tau_F = F_F \times R$

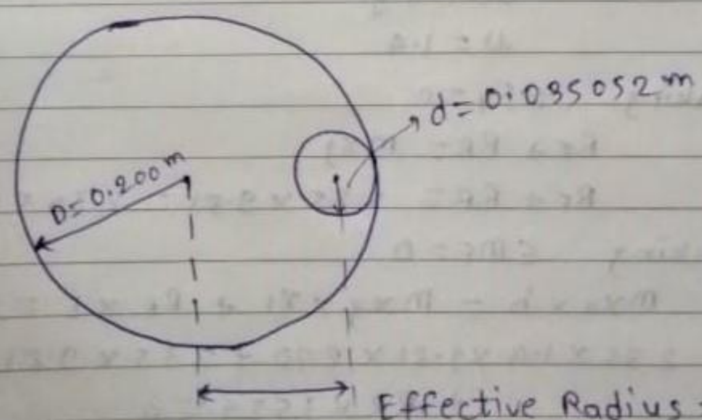
$= 2347.1 \times 228.6$

∴ $\boxed{\tau_F = 536.55 \text{ N-m}}$

∴ Front Breaking Torque = 536.55 N-m
Rear Breaking Torque = 201.257 N-m

For, worst condition we took

Rear Breaking Torque = 250 N-m



Effective Radius $= \frac{D}{2} - \frac{d}{2}$

$= \frac{0.200}{2} - \frac{0.035052}{2}$

= 82.474 mm

$\tau_R = F \times$ effective radius

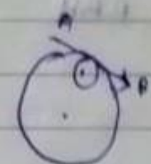
$$F = \frac{\tau_R}{\text{effective radius}} = \frac{250}{0.082474}$$

$\therefore$ $\boxed{F = 3031.29834\ N}$

As, we have points A & B,

$A(x_1, y_1, z_1)$ & $B(x_2, y_2, z_2)$

$\therefore \overline{AB} = \overline{b} - \overline{a}$



$\therefore$ For $\overline{AB}$ direction is as shown, in our case

Let, $A \equiv (612.532, 91.149, 314.49) \equiv (x_1, y_1, z_1)$

$B \equiv (573.512, 91.149, 353.509) \equiv (x_2, y_2, z_2)$

$\therefore \overline{AB} = \overline{b} - \overline{a}$

$= (x_2 - x_1), (y_2 - y_1), (z_2 - z_1)$

$\therefore \overline{AB} = (39.02, 0, 39.02) \rightarrow 39.02\hat{i} + 0\hat{j} + 39.02\hat{k}$

Magnitude $= \sqrt{39.02^2 + 0^2 + 39.02^2} = 55.1826$

$\therefore$ Unit vector $= 0.7071\hat{i} + 0\hat{j} + 0.7071\hat{k}$

Final force $\rightarrow$

∴ Final force = |F| × unit vector
$= 3031.25834 \times (0.7071\,\hat{i} + 0\,\hat{j} + 0.7071\,\hat{k})$

∴ $\boxed{\text{Final force} = 2143.403\,\hat{i} + 0\,\hat{j} + 2143.403\,\hat{k}}$

∴ Final Remote Force :-

Location → (590.711, 91.149, 336.311)
Components → 2143.403N, 0, 2143.403N
Application → Spool mount surface (only right spool mount on RBH

5) As, we know radius of disc and piston, we calculate effective radius as,

Effective Radius = D/2 – d/2
= 0.200/2 – 0.035052/2
= 82.474 mm …(D = Diameter of brake disc
d = Diameter of piston)

6) Rear Brake Torque = Force * Effective Radius
As we know, rear brake torque is 250 Nm, we calculated the force acting on piston.
Force = 3031.25834 N
We found the magnitude and we want to find direction as well, because it won’t be in Z direction only, as it depends on the mounting position of calliper, hence piston. Because it can be anywhere in X-Z direction.

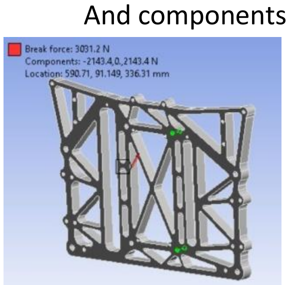
7) It will act in tangential direction at the point of contact between piston and disc at point O as shown,

8) We found the direction of tangent at point O to the disc by finding unit vector of AB as shown,
Coordinates are A = (612.532, 91.149, 314.49)
B = (573.512, 91.149, 353.509)
O = (590.711, 91.149, 336.311)

9) Hence, unit vector = -0.7071 i + 0 j + 0.7071 k
Final force = -2143.403 i + 0 j + 2143.403 k

10) This force tries to pull out calliper in front direction, which is restricted by bolts on Spool Mount where it is mounted and motion of Spool Mount is restricted by bolts on RBH.

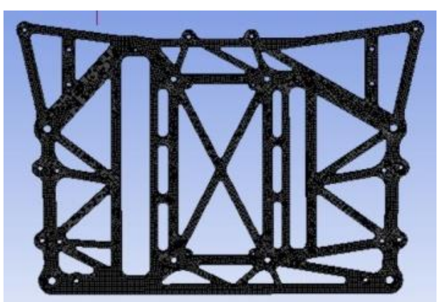
11) Hence, remote force is applied at the holes of mounting of right Spool Mount on RBH,
Its location will be at point O = (590.711, 91.149, 336.311)
And components are x=-2572.0833 N, y=0 N, z=2572.0833 N

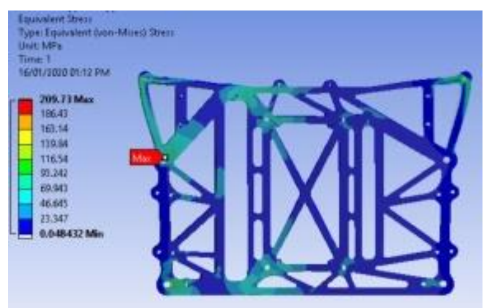


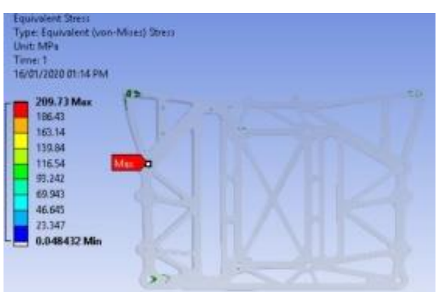
Coordinate System	Global Coordinate System
X Coordinate	590.71 mm
Y Coordinate	91.149 mm
Z Coordinate	336.31 mm
Location	Click to Change
Definition	
Type	Remote Force
Define By	Components
X Component	-2143.4 N (ramped)
Y Component	0. N (ramped)
Z Component	2143.4 N (ramped)

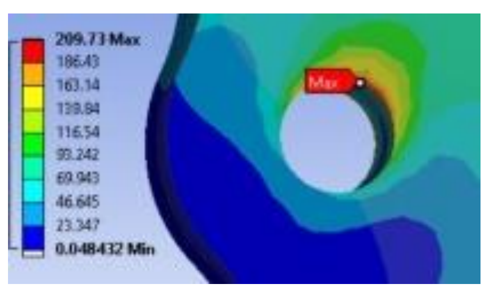
Important Note –

1. From above mentioned forces, weight of drive train, load transfer while jacking and impact force act individually. They do not act with other forces.
2. Forces through Rocker & ARB, UCA, LCA, Toe, Damper, Torque tension, Break force can act simultaneously in dynamic conditions in worst condition.
3. But, when torque tension acts, brake force doesn’t and vice versa. Though we know that torque tension force is greater than brake force, we still had to do it in different cases as there application on RBH and directions are different. Hence accordingly, we had to do pocketing. That’s the reason we have asymmetric pocketing on the RBH on either side.
4. Also, due to meshing technique in simulation software (Ansys), in some cases stress was concentrated at mesh node close to fix support holes.
5. As, (stress = Resistive force / Area) Hence, as we go on making dense mesh stress concentration goes on increasing as effective area is decreased. This is called as SINGULARITY.
6. Because all other region was completely safe and was below yield stress of the material.
7. This is because of holes on the RBH but in actual practise there are going to bolts i.e. material is there, hence we ignored such stresses.
8. Also, after testing of a car all materials are found to be safe.
9. Below pictures show that, when we keep on making dense mesh, though stress increases, but it goes on concentrating at the fixed point hole. Other area is quite safe, hence we ignored such stresses as a singularity. Even though still we kept the max stress below yield strength of the material.



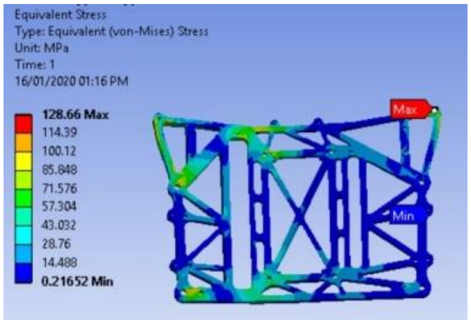




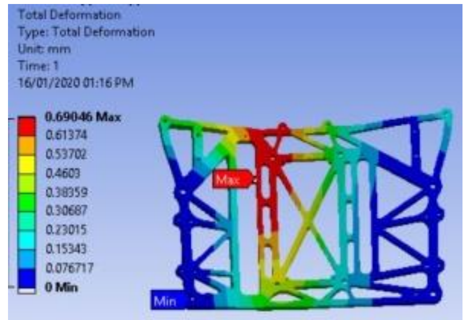


Above images are stress results without brake force.

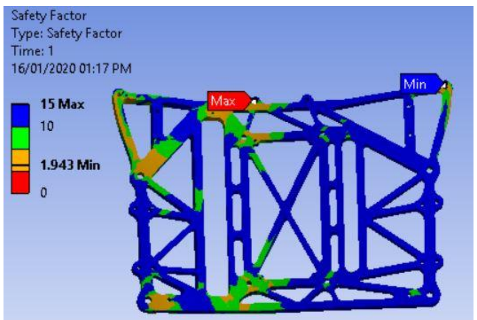
Results Without Brake Force-



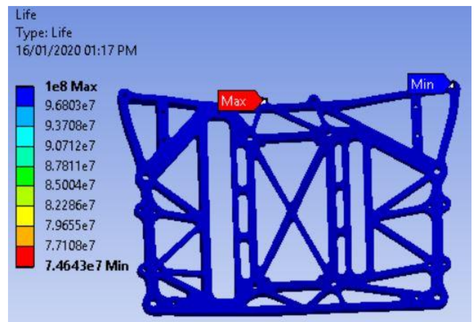
Equivalent Stress



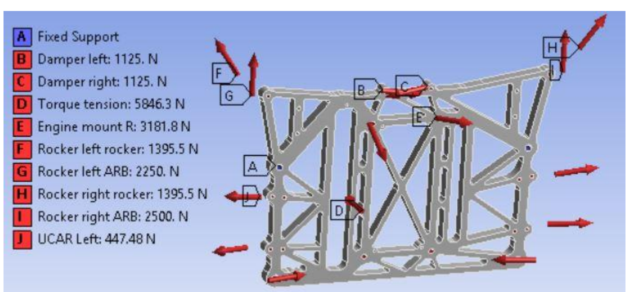
Total Deformation



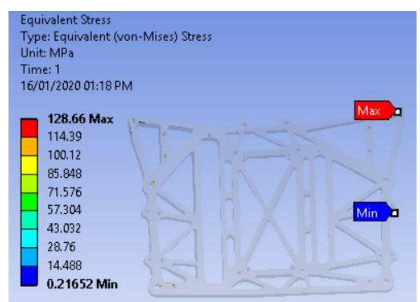
Factor of Safety



Fatigue Life



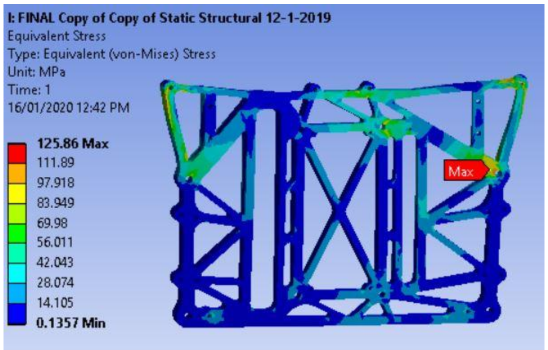
Forces



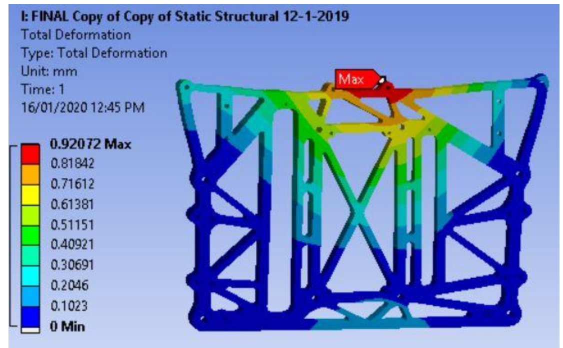
Singularity

SR	Quantity	Value
1	Maximum equivalent stress	128.66 MPa
2	Maximum total deformation	0.69046 mm
3	Minimum safety factor	1.943
4	Minimum fatigue life	7.4643e7

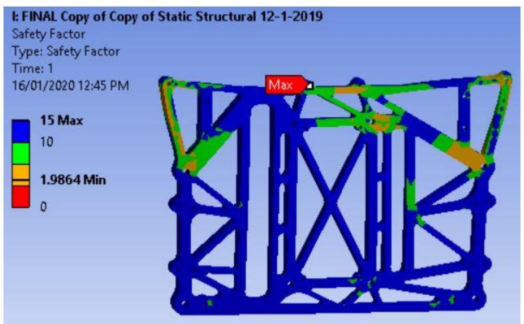
Results with Brake Force-



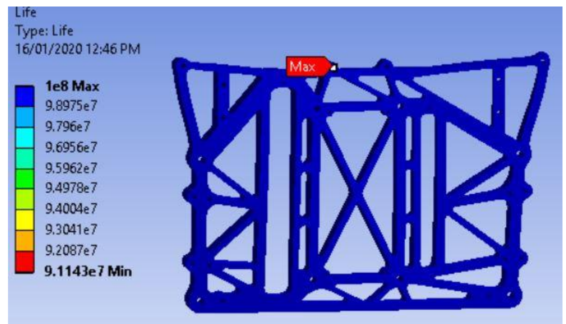
Equivalent Stress



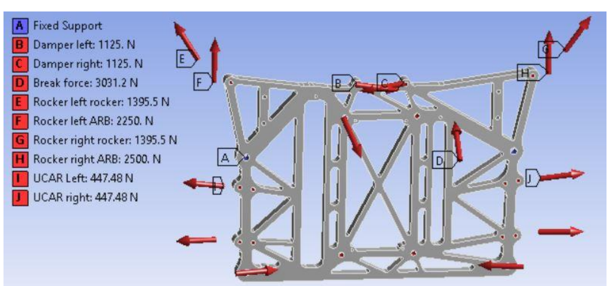
Total Deformation



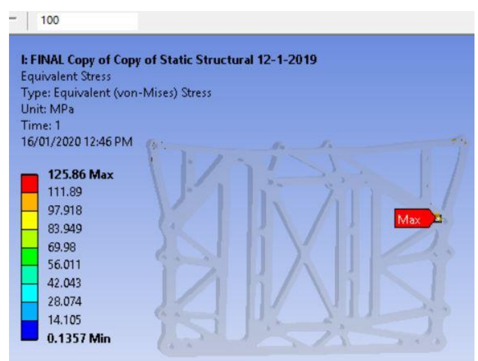
Factor of Safety



Fatigue Life



Forces

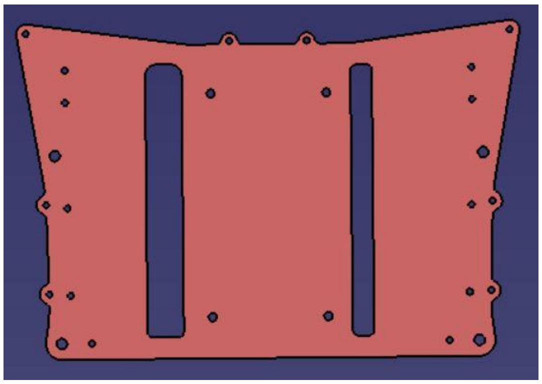


Singularity

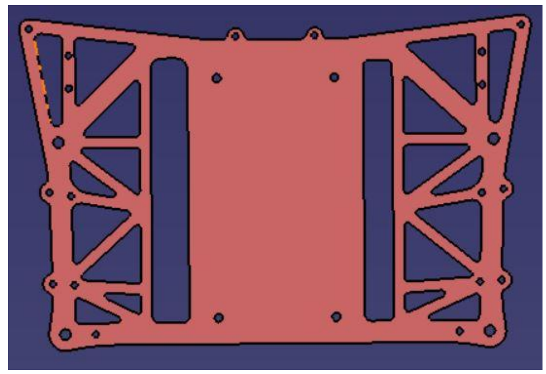
SR	Quantity	Value
1	Maximum equivalent stress	125.86 MPa
2	Maximum total deformation	0.92072 mm
3	Minimum safety factor	1.9864
4	Minimum fatigue life	9.1143e7

Iterations-

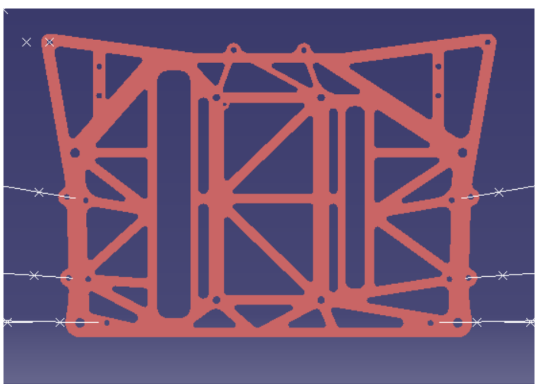
1. Based on the stress concentration and requirement for sprocket & brake disc, we made pockets accordingly.
2. We tried to reduce weight as much as we can but at the same tried we ensured that there is sufficient space for bolt heads.



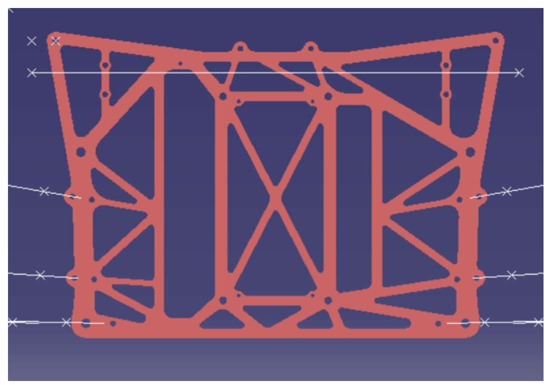
Iteration 1



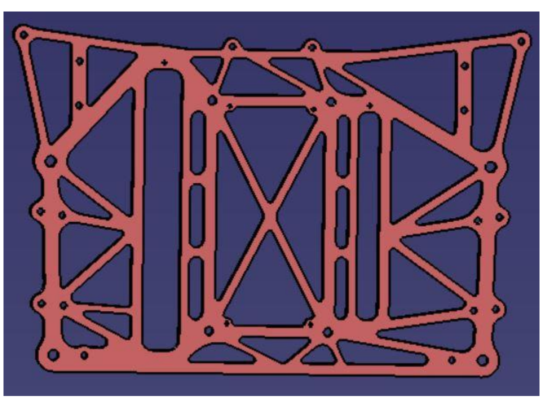
Iteration 2



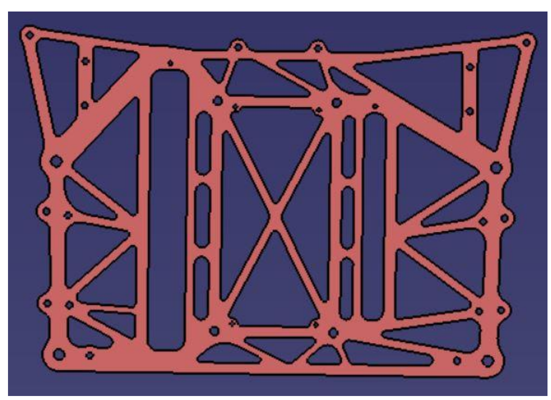
Iteration 3



Iteration 4



Iteration 4

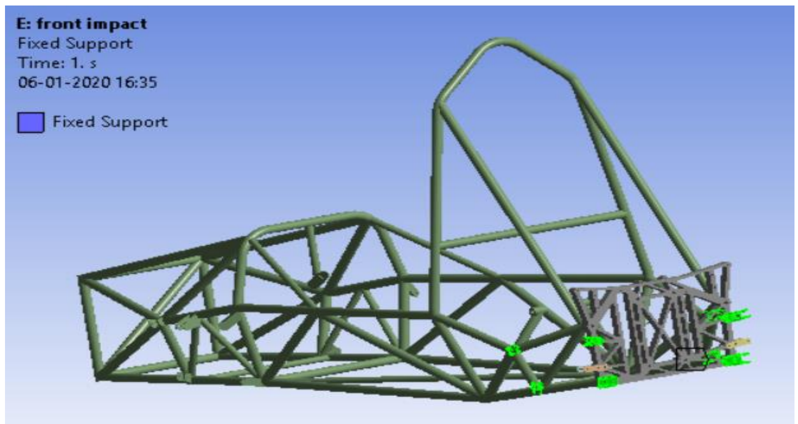


Final Iteration

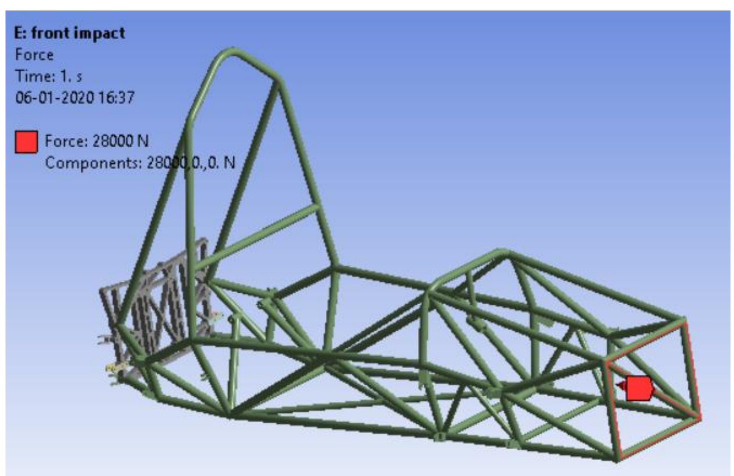
Front Impact Test

Front impact analysis is carried out to check whether the chassis would withstand the hit from the front with any other car, barrier or wall from the front side or any other car hit our stationary car from the front.

i. **Fix Support -** Rear suspension mounts are selected as fixed support.
It is assumed that the car is hit from the front side, the rear tires would try to hold the car stationary.



ii. **Load Points -** The load of 28,000 N is applied on front bulkhead as it is assumed that the car hit a stationary wall with the speed of 40 kmph and the complete momentum of car is transferred to wall so that the final velocity of car become zero. The speed is consider as 40 kmph as it is the average speed.



iii. **Calculations -** Force is rate of change of momentum

F = m (v – u) / t

(where, F = Force acting on the car after impact
m = mass of car i.e. 250kg
v = final velocity of car after impact i.e. 0

u = initial velocity of car before impact i.e. 40kmph

t = time of impact i.e. for worst case 0.1sec)

Therefore, F= 250 (40 kmph) / 0.1

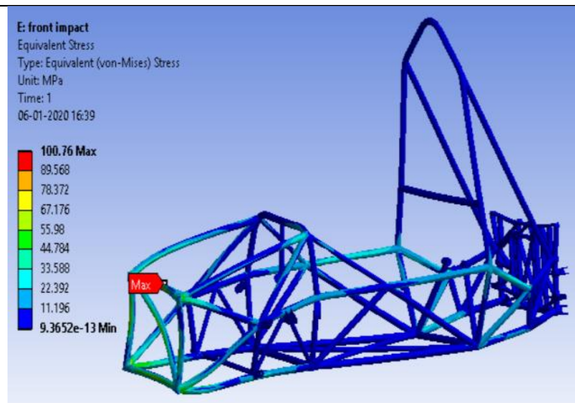
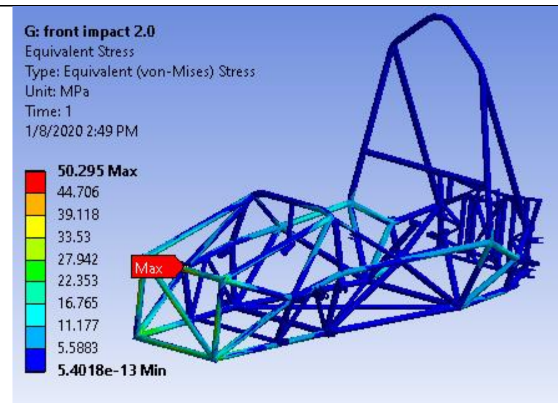
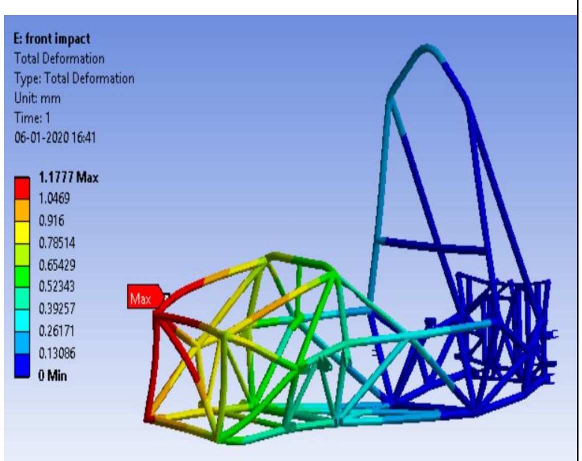
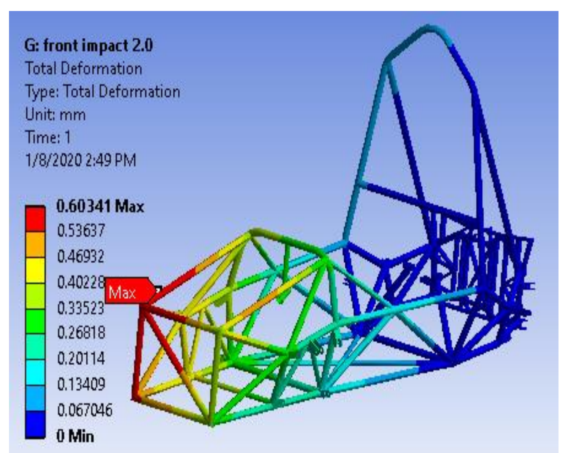
= 250 [(40*5) / 18mps] / 0.1

= 27,777.7778 N

For safe side, force is taken as **28,000 N** in positive X direction

No force in Y and Z direction.

For impact time of 0.2 sec, F = 13,888.8889 N which we take as 14,000 N.

Sr No.	Quantity	t = 0.1 sec	t = 0.2 sec
1	Maximum equivalent stress		
2	Maximum total deformation		

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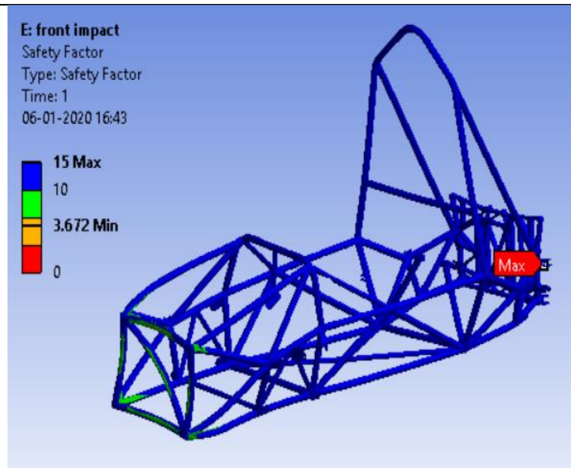
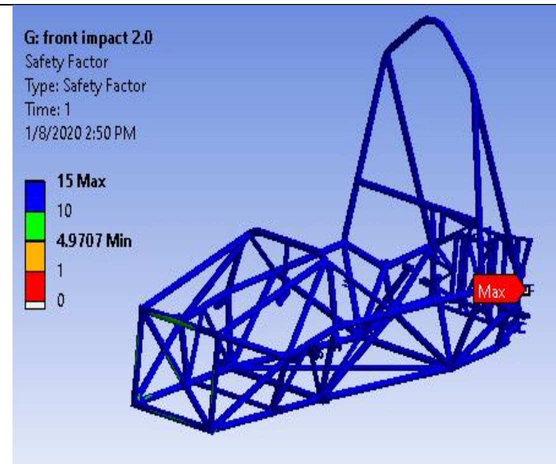
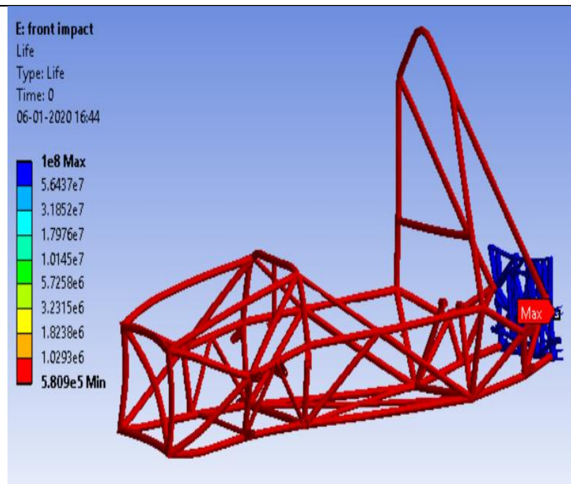
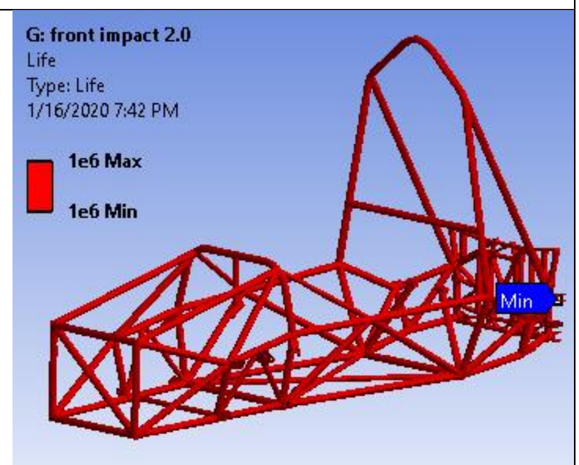
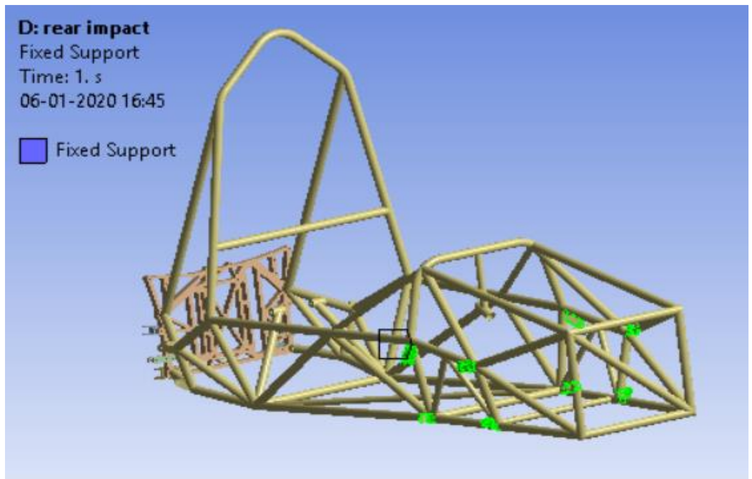
safety factor</td><td></td><td></td></tr>
<tr><td>4</td><td>Minimum
fatigue life</td><td></td><td></td></tr>
</table>

SR	**Quantity**	**t=0.1sec**	**t=0.2sec**
1	Maximum equivalent stress	100.76 MPa	50.295 MPa
2	Maximum total deformation	1.7778mm	0.6034mm
3	Minimum safety factor	3.672	4.9707
4	Minimum fatigue life	5.809e5	1e6

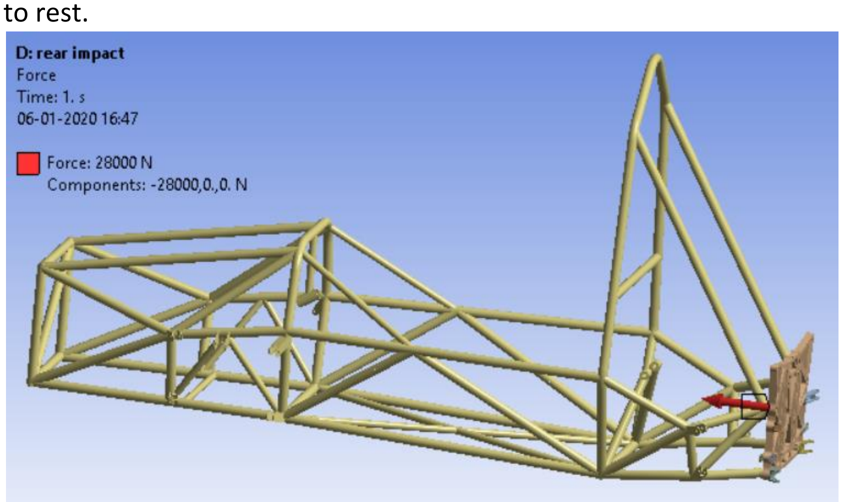
Rear Impact Test

Rear impact analysis is carried out to check whether the chassis would withstand the hit from the rear with any other car from the rear side.

i. **Fix Support -** Front suspension mounts are selected as fixed support as if the car is hit from the rear side, the tires would try to hold the car stationary.



ii. **Load Points -** The load of 28,000 N is applied on the chassis where it is attached to rear bulkhead. The rear bulkhead is only used to connect the suspension mounts with chassis which is used as fixed support. Similar to front impact test, it is assumed that our stationary car is hit by a car moving with 40 kmph and transfer complete momentum to come to rest.



iii. Calculations - Force is rate of change of momentum

F = m (v – u) / t

(Where, F = Force acting on the car after impact

m = mass of car i.e. 250kg

v = final velocity of car after impact i.e. 40

u = initial velocity of car before impact i.e. 0kmph

t = time of impact i.e. for worst case 0.1sec)

Therefore, F= 250 (40 kmph) / 0.1

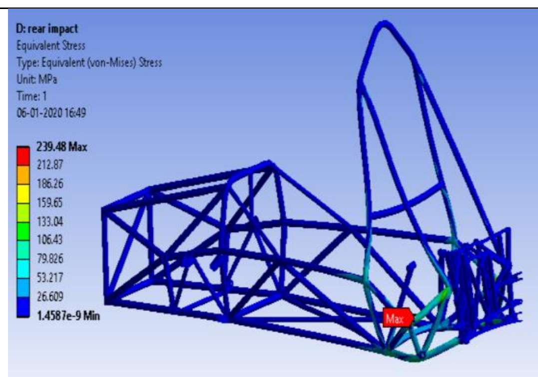
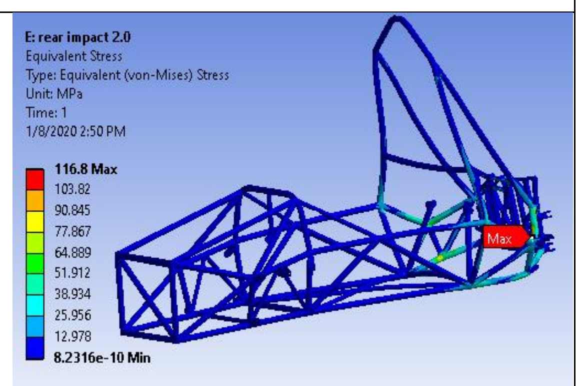
= 250 [(40*5) / 18mps] / 0.1

= 27,777.7778 N

Hence for safer side, force is taken as **28,000 N** in negative X direction

No force in Y and Z direction.

For impact time of 0.2 sec, F = 13,888.8889 N which we take as 14,000 N.

<table>
<tr><th>Sr No.</th><th>Quantity</th><th>t = 0.1 sec</th><th>t = 0.2 sec</th></tr>
<tr><td>1</td><td>Maximum equivalent stress</td><td>
</td><td>
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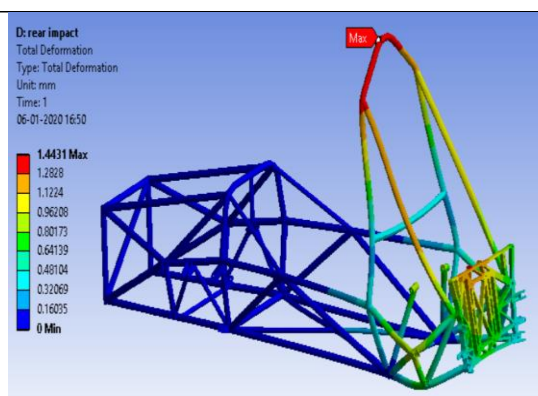
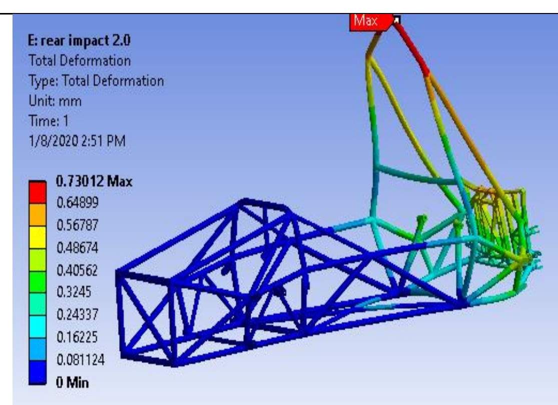
<tr><td>2</td><td>Maximum total deformation</td><td>
</td><td>
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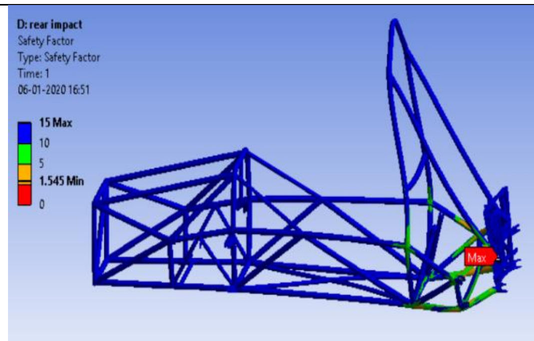
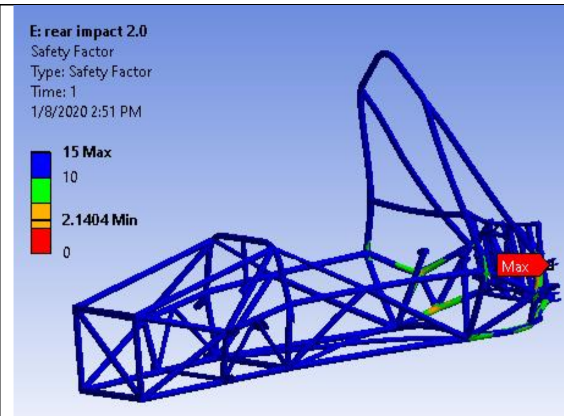
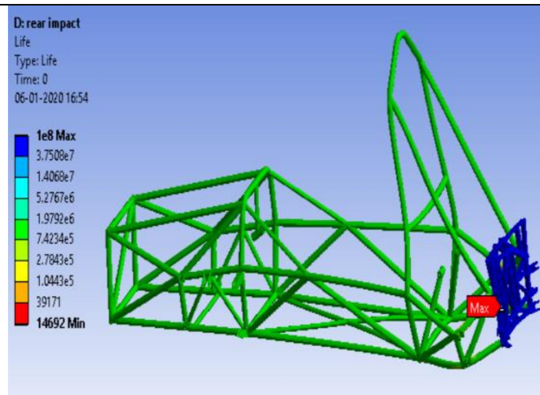
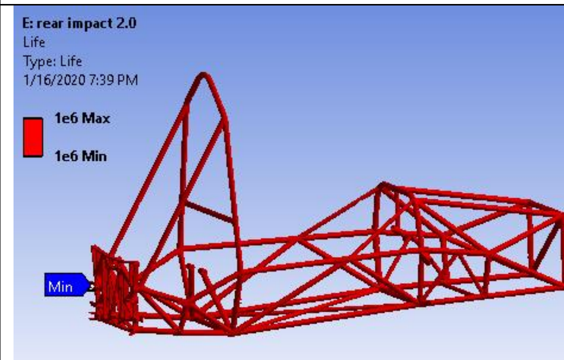
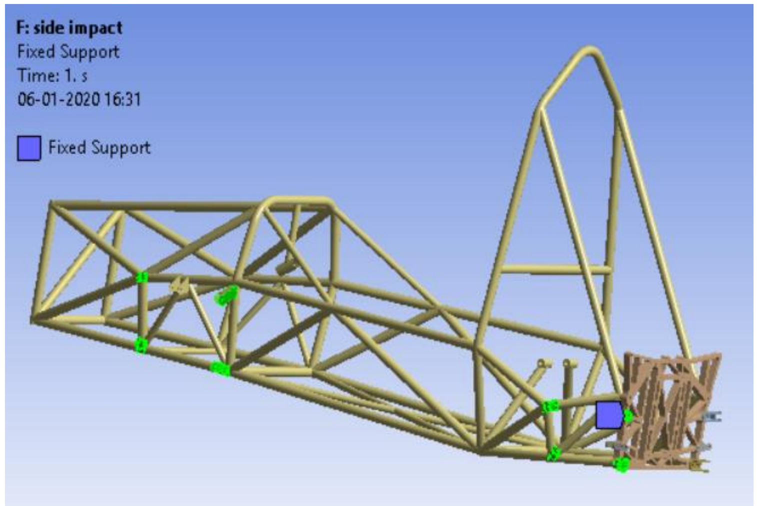
3	Minimum safety factor		
4	Minimum fatigue life		

SR	Quantity	t=0.1sec	t=0.2sec
1	Maximum equivalent stress	239.48 MPa	116.8 MPa
2	Maximum total deformation	1.4431mm	0.7301mm
3	Minimum safety factor	1.545	2.1404
4	Minimum fatigue life	14692	1e6

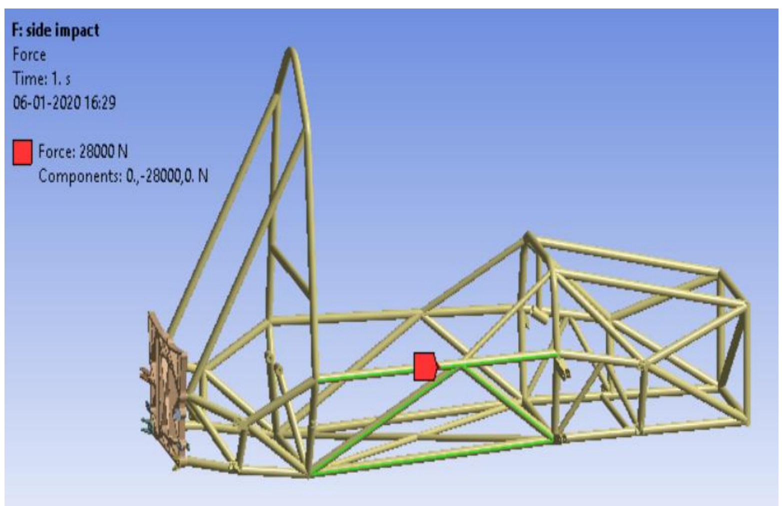
Side Impact Test

In side impact test it is assume that the car is hit from one side and the other side resists to move. It is carried out to check weather our car is capable to withstand that impact.

i. **Fixed Support -** The suspension mounts of one side of the car is consider as fixed support as when the car would be hit from one side the tiers of other side would try to be fixed. The tiers are connected to suspension mounts through A-arms and so the suspension mounts of one side are consider as fixed support.



ii. **Loading Points -** The load of 28,000 N is applied on the side impact structure of the opposite side of the suspension mounts which are used as fixed. Here it is assume that our car is stationary and another car with 40 kmph hit our car and transfer complete momentum to come to rest.



iii. **Calculations -** Force is rate of change of momentum

F = m (v – u) / t

(Where, F = Force acting on the car after impact

m = mass of car i.e. 250kg

v = final velocity of car after impact i.e. 40

u = initial velocity of car before impact i.e. 0kmph

t = time of impact i.e. for worst case 0.1sec)

Therefore, F= 250 (40 kmph) / 0.1

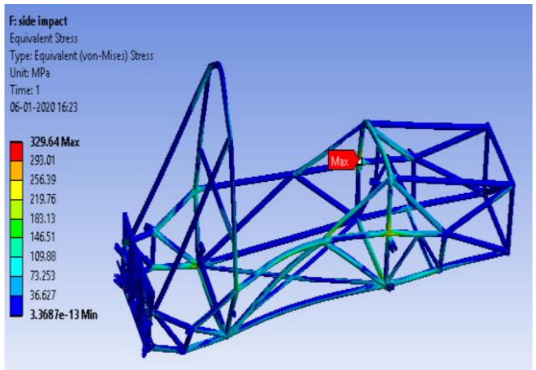
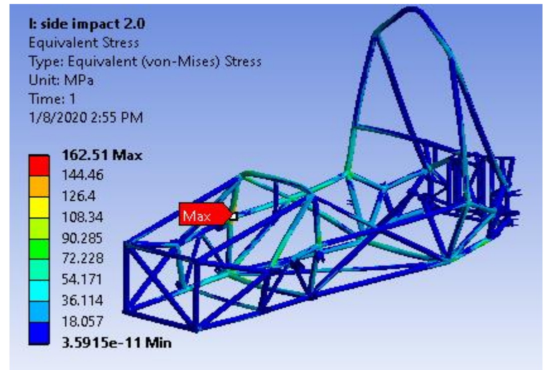
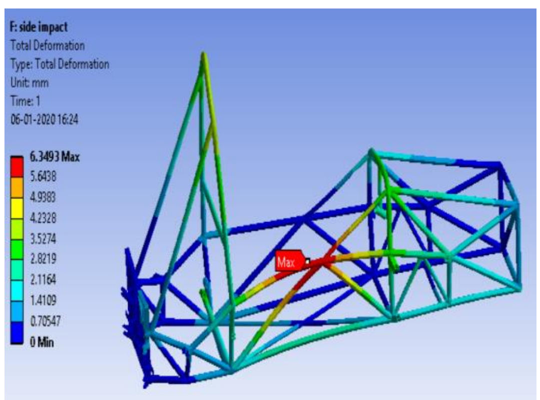
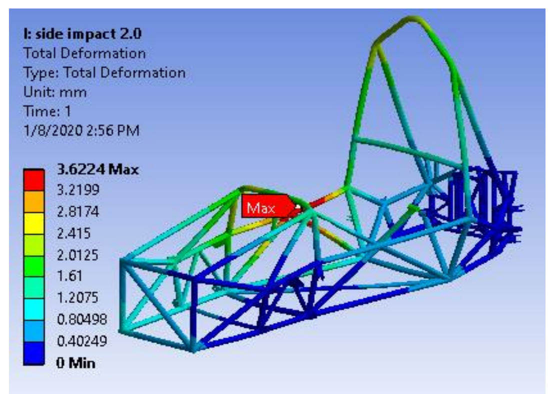
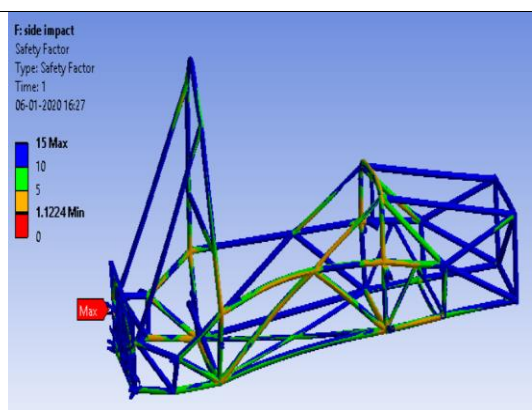
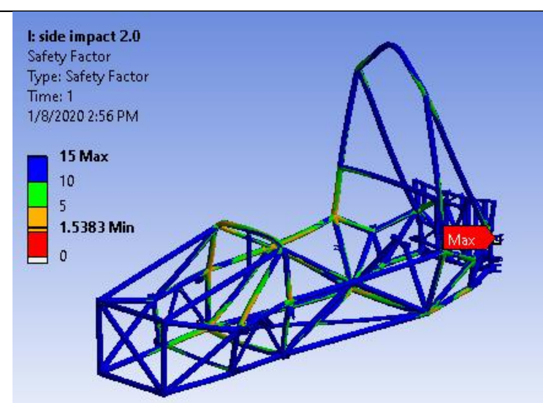
= 250 [(40*5) / 18mps] / 0.1

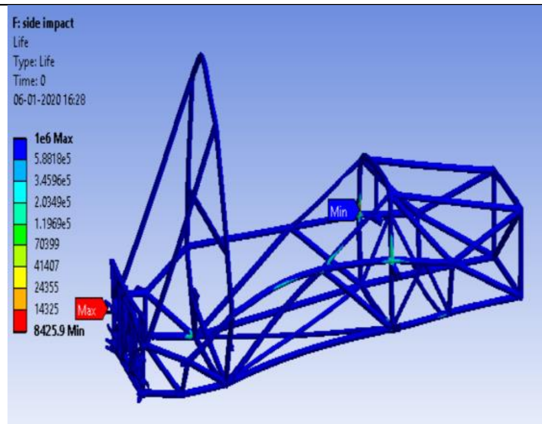
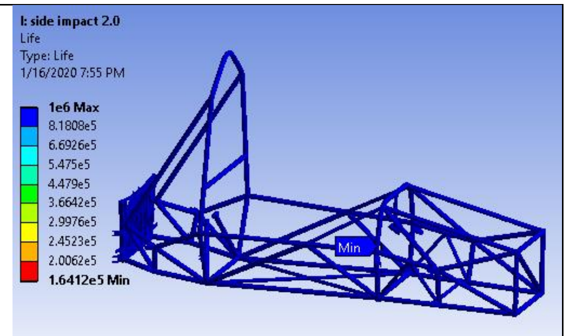
= 27,777.7778 N

Hence for safer side, force is taken as **28,000 N** in positive Y direction

No force in X and Z direction.

For impact time of 0.2 sec, F = 13,888.8889 N which we take as 14,000 N.

Sr No.	Quantity	t = 0.1 sec	t = 0.2 sec
1	Maximum equivalent stress		
2	Maximum total deformation		
3	Minimum safety factor		

4	Minimum fatigue life		

SR	Quantity	t=0.1sec	t=0.2sec
1	Maximum equivalent stress	329.64 MPa	162.51 MPa
2	Maximum total deformation	6.3493mm	3.6224mm
3	Minimum safety factor	1.1224	1.5383
4	Minimum fatigue life	8425.9	1.6412e5

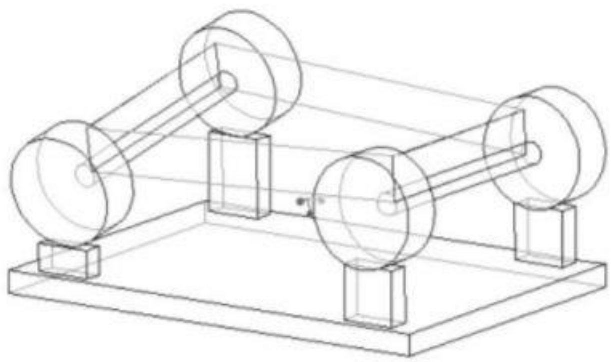
Torsional Rigidity Analysis

What is torsional rigidity?

Torsional rigidity is the amount of resistance a cross section has against torsional deformation. The higher the rigidity, the more resistant is the cross section to torsional deformation. While calculating wheel loading under various driving conditions, it is important to recognize that all the calculations of suspension parameters are based on the assumption that the chassis of the car is rigid and has infinite stiffness, that is, it does not bend or twist under any driving condition. But in actual the chassis isn't infinitely stiff and does bend and twist under loading.

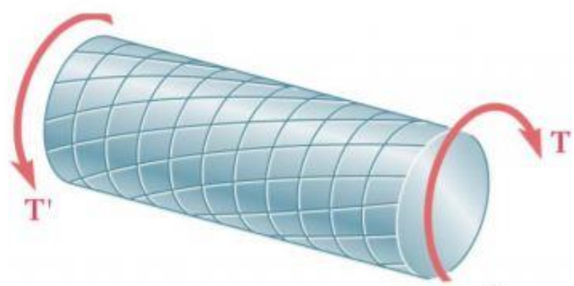
A performance vehicle must have adequate chassis torsional stiffness around the x-axis, that is, longitudinal axis along the length of the vehicle. Thinking of the chassis as a large spring connecting the front and rear suspensions, if the chassis torsional spring is weak, attempts to control the lateral load transfer distribution will be confusing at best and impossible at worst. This is because a flexible spring adds another spring to an already complex system. Predictable handling can best be achieved if the chassis is stiff enough to be safely ignored. Hence it is important to make sure the chassis is stiff enough. Now the question that follows is 'how stiff is stiff enough?' We'll come back to this question in later part of this document.

Torsional deflections of a car chassis result from forces induced in the frame by the suspension. If the chassis is not stiff enough, the suspension loads create a tendency for the frame to twist about the longitudinal axis running down its length. Excessive deflections make control of lateral load transfer, one of the primary functions of a car's suspension, difficult. Therefore, the torsional rigidity of the frame is critical to the dynamic performance of the car. The figure below shows the twisting of a chassis.



The first step in any engineering problem is to break it down into its simplest form and fit it to a theoretical model. Torsion, in its simplest form, is defined on a solid circular member. From Mechanics of Materials (or Solid Mechanics), the applied torque couple, T, is proportional to the angular deflection θ. The constant of proportionality is known as the torsional rigidity and is related to geometric and material properties of the member in torsion.

$$T = k * \theta$$



Where T is the applied torque, θ is the angular deflection, and k is torsional rigidity.

$$k = J * G / L$$

Where J is the second moment of area, G is the modulus of rigidity (or shear modulus), and L is the length of the member. For a simple beam, such as the one in Figure, 'k' can be arithmetically determined because 'J', 'G', and 'L' are all easily known. However, for a more complicated beam shape, such as an FSAE space frame, the second moment of area is more challenging to calculate.

Instead, an experiment can be run in which different torques are used to measure angular deflections. Theory says that these values plotted against each other should be linear as long as the experiment remained within the elastic range of the material. A linear trend analysis is used to determine torsional rigidity, which is no longer constant because 'J' becomes a function of position for irregular beams.

The biggest effect of torsional deflection has been found to be the effect it has on lateral load transfer distribution between the front and rear axle. During a steady state turn an infinitely rigid chassis will cause the front and rear roll angle to be identical, as is assumed when suspension design calculations are performed. Allowing for twist in the chassis will redistribute some amount of weight transfer between the front and rear tires, causing these values to deviate from their designed values. Because of these effects, the vehicle will handle more predictably if the chassis is stiff enough, relative to the suspension roll stiffness, that the twist can be ignored. To achieve this goal, most racecars design to be greater than the suspension roll stiffness. Finding a standard value against which a chassis can be compared would be helpful in the design of vehicles, but as of current, no standard criterion exists.

As well as the effects above, twist in the chassis has also been theorized to have negative effects on the fatigue of the vehicle, suspension characteristic changes caused by relative hard point displacement, and transient response time of the vehicle as a whole to driver inputs. All of these effects can be easily mitigated by increasing the stiffness of the vehicle, but this has the direct response of increasing vehicle weight. For this reason, it is desired to find the lowest value of torsional stiffness that will not decrease the performance of the car. The target value that was set for the torsional rigidity was 1400 Nm/degree. This value was decided as the target value based on various documents online and seminars attended.

Why do we need torsional rigidity?

During cornering or a sudden bump on any wheel of your car, your car goes under torsion. Here, torsional rigidity plays an important role giving you an idea whether your car can withstand this torque. It also helps to decide a design for a particular component. Whenever a car goes under torsion, the majority torque acting is on the chassis of your car.

How to calculate torsional rigidity?

There are two methods for calculating torsional rigidity of your chassis:

- Simulation Method
- Experimental Method

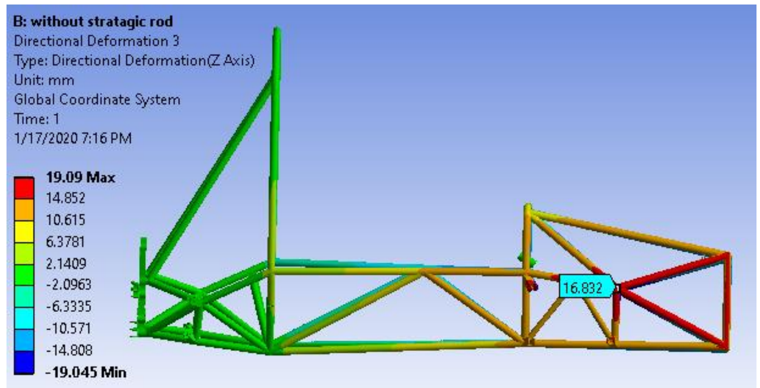
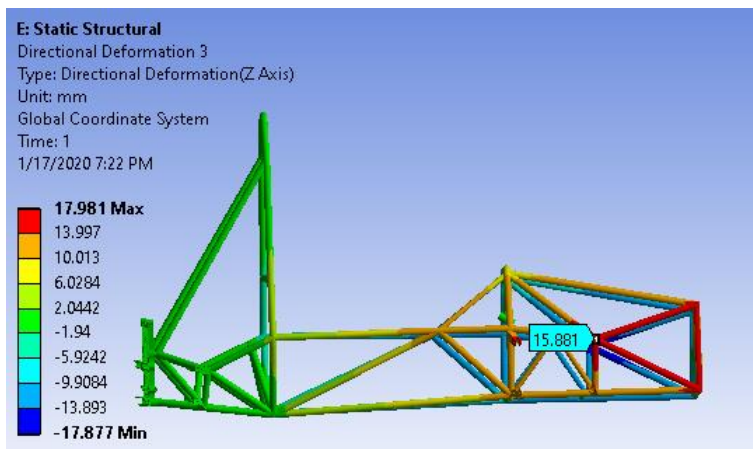
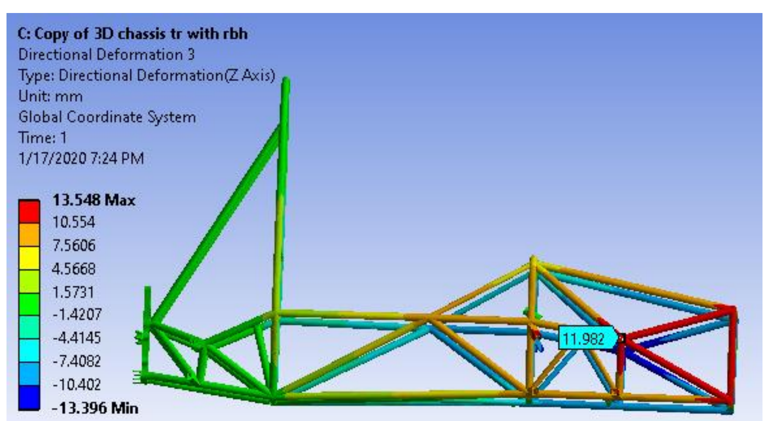
<u>SIMULATION METHOD</u>

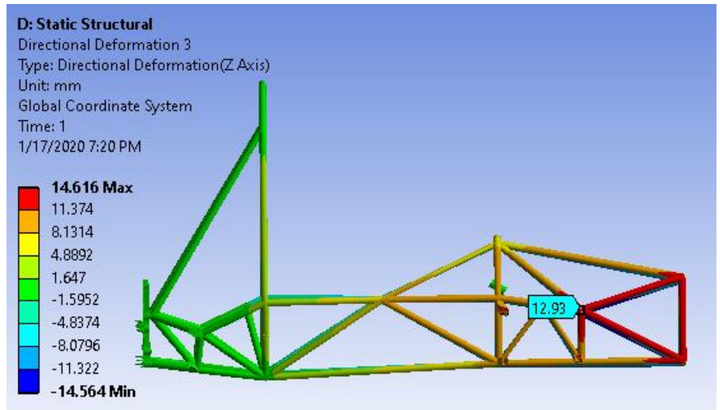
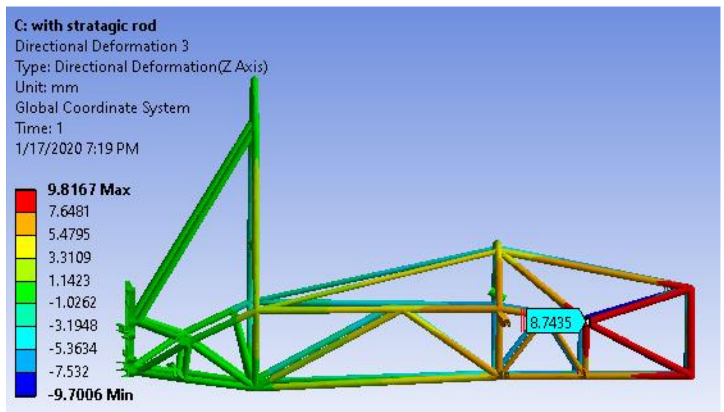
The simulation of the chassis is done on the ANSYS software.

Before simulation, the magnitudes and direction of the forces to be applied are calculated by basic formula and values of other parameters.

One way is the rear wheel hubs or real wheel hub mounts are fixed and couple forces are made to act on the front wheel hubs or their mounts (based on the CAT part and assembly). The software simulates the chassis CAT part and gives corresponding deformation or twist due to couple forces assuming complete ideal conditions. The same procedure is followed another way. The only difference is the fixed positions and positions of the couple forces are interchanged i.e. front is made fixed and couple forces are applied on the rear part of the chassis. The reason for the fixed positions and forces being applied at the hubs is they are the first member to experience force after wheels. Also, couple forces are used for simulation rather than torque. This is because chassis is irregular and has no fixed radial length. Hence a fixed value of torque is difficult to calculate.

The position of node of Front Hoop Bracing and Side Impact Structure was chosen such as the torsional stiffness at the position is optimum among the points on the member of side impact structure.

Sr. No.	Ratio of (Length of front to node/Length of Side impact member)	Deformation
1.	0	
2.	0.3	
3.	0.4	

4.	0.5	
5	1	

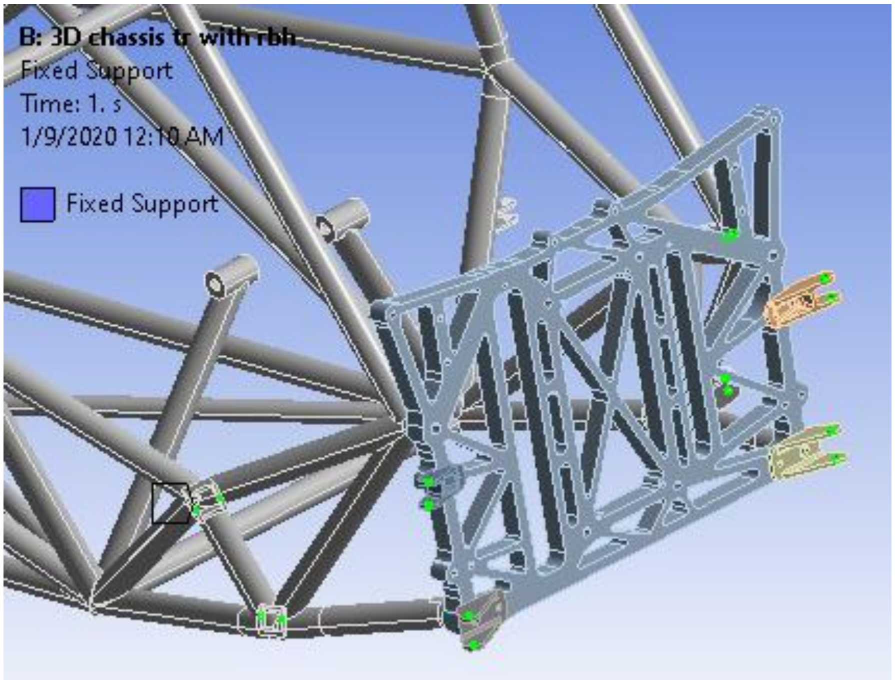
Sr. No.	Ratio of distance of front hoop bracing lower attachment	Maximum Deformation
1.	0	19.09 mm
2.	0.3	17.981 mm
3	0.4	13.548 mm
4.	0.5	14.616 mm
5.	1	9.8167 mm

As the ratio of distance of front hoop bracing attachment increases, we observed, the deformation due to torsion decreases till 0.4 and then increases eventually.

We chose the ratio to be 0.4 for maximum stiffness and minimum deformation.

1. Fixed Supports-

Rear suspension points as shown,

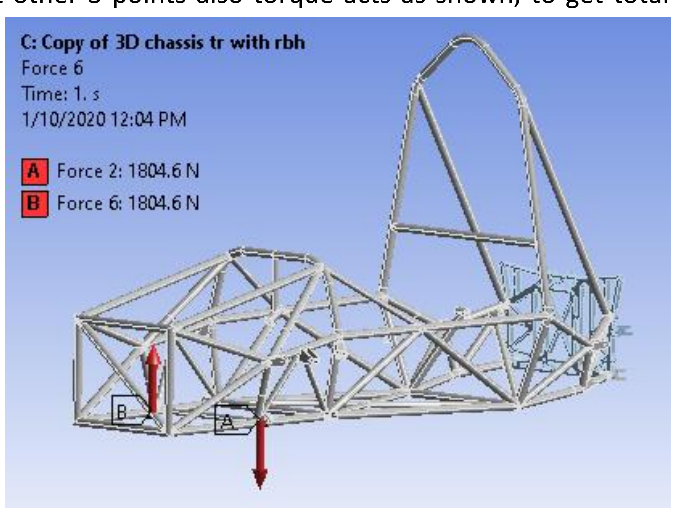


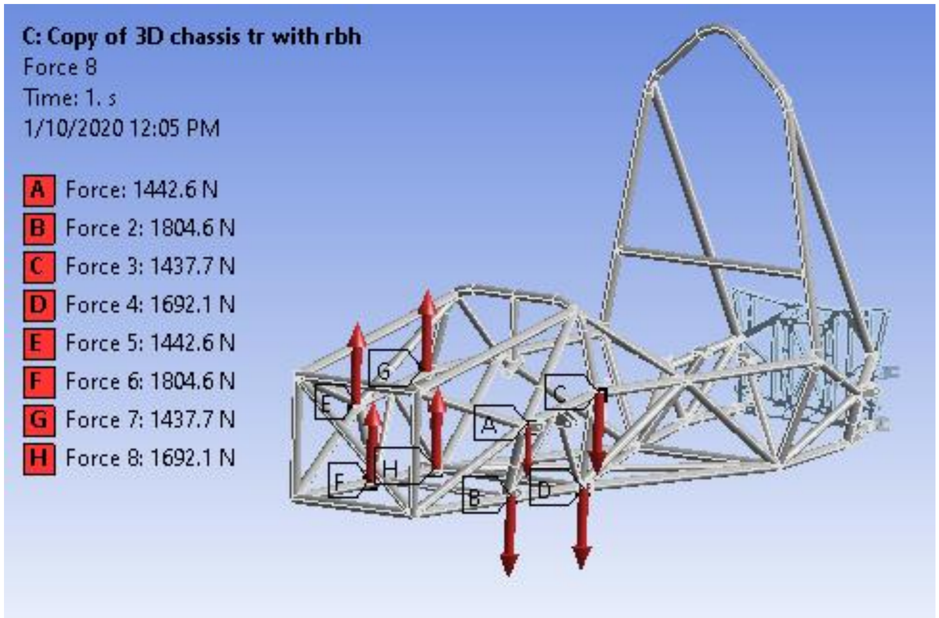
2. Load Points-

- First, torque due to bump force on tires is taken as 3000Nm.
- This torque is applied through forces acting on 4 front suspension points, coming through tires and then through A arms.
- These 4 front suspension points are say upper front (UF) 1, upper rear (UR) 2, lower front (LF) 3, lower rear (LR) 4.
- Hence, at each front suspension point equivalent amount of force acts which lead to total of 3000Nm torque.

At each suspension point force acts in upward direction and equivalent force acts in opposite direction at corresponding point on other side as shown,

- Similarly, at other 3 points also torque acts as shown, to get total amount of 3000Nm torque.



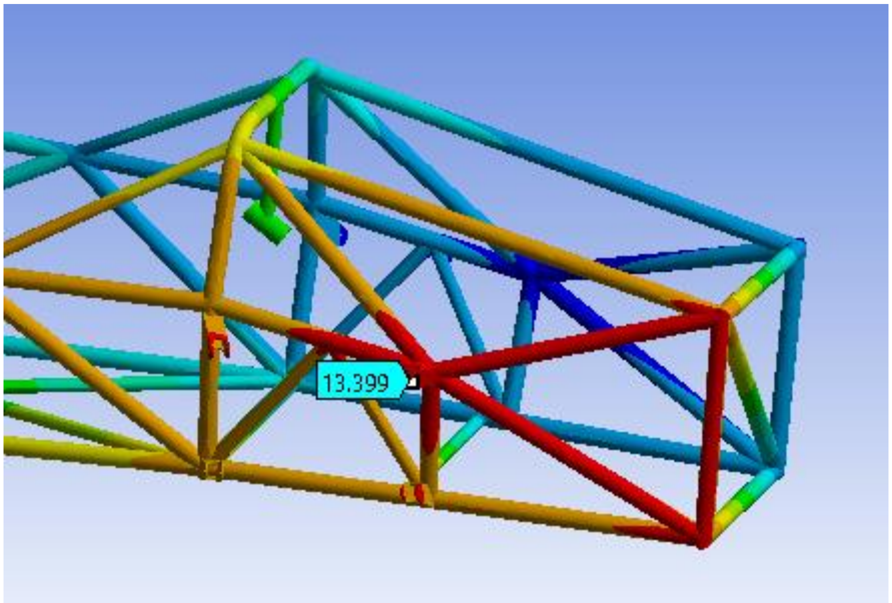


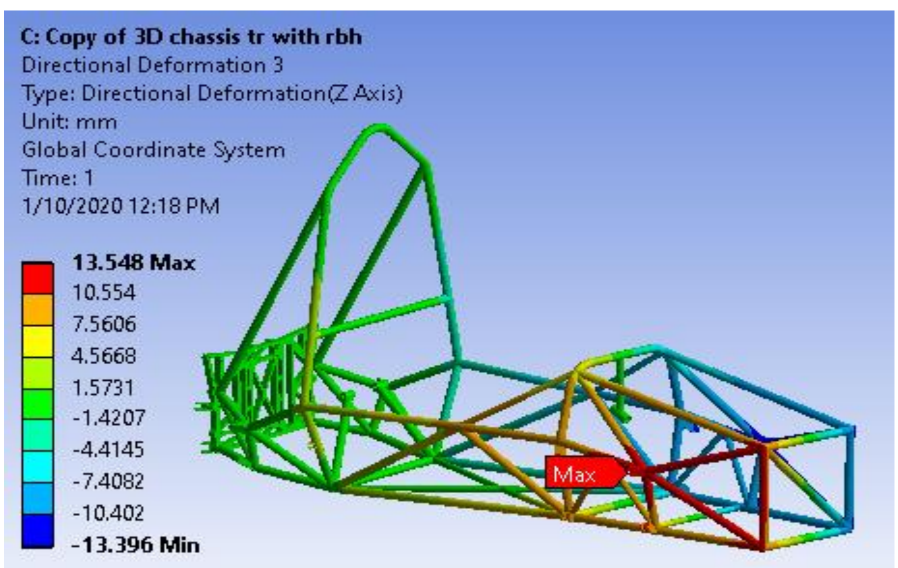
- Distances between corresponding points are
  UF = 519.884mm = d1
  UR = 521.674mm = d2
  LF = 415.6mm = d3
  LR = 443.228mm = d4

- These forces are calculated as follow,
  Total torque = torque at UF + torque at UR + torque at LF + torque at LR
  3000 = F1*d1 + F2*d2 + F3*d3 + F4*d4
  Assume,
  F1*d1 = F2*d2 = F3*d3 = F4*d4 = 3000/4 = 750 Nm
  Hence, F1*d1 = 750
  F1 = 750/d1
  = 750/0.519884m
  F1 = 1442.62951N
  Similarly,
  F2 = 750/d2=1473.67947N
  F3 = 750/d3=1804.619827N
  F4 = 750/d4=1692.131364N

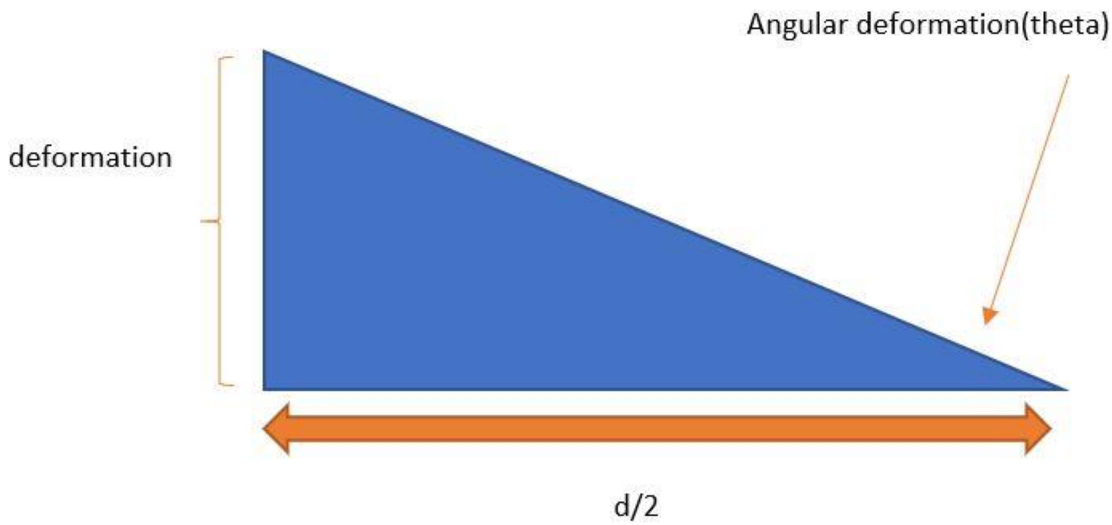
3. Results

After performing FEA, we got directional deformations in Z direction at each suspension point as shown,





- UF=11.782mm
  UR=9.6258mm
  LF=9.518mm
  LR=8.1312mm



- Hence angular deformation is found at each point as

$$\tan^{-1} \quad deformation/distance$$

- And, torsional rigidity= (Total torque)/angular deformation
  1. UF-
     Theta=$\tan^{-1} \quad 11.782 * 2/519.884$ = 2.5952 degree
     T.R. = 3000/2.5952 = 1155.98027 N.m/degree

  2. UR-
     Theta=$\tan^{-1} \quad 9.6258 * 2/521.674$= 2.11345 degree
     T.R. = 3000/2.11345 = 1419.48 N.m/degree

  3. LF-
     Theta=$\tan^{-1} \quad 9.518 * 2/415.6$ = 2.6225 degree
     T.R. = 3000/2.6225 = 1142.94662 N.m/degree

  4. LR-
     Theta=$\tan^{-1} \quad 3.1318 * 2/443.228$= 2.10129degree
     T.R. = 3000/2.10129 = 1427.69442 N.m/degree

     Average (Final) T.R. = $\frac{1155.98027+1419.48+1142.94662+1427.69442}{4}$

     T.R = 1286.52533 Nm/degree

- Average T.R. = 1286.52533 N.m/degree
- Average Theta = 2.35811 degree

4. Similarly,

We calculated T.R. and Theta average for different torques from 1000 Nm to 10000 Nm as shown in following calculations,

1) Torque (total) = 1000 Nm

Hence, F1d1 = F2d2 = F3d3 = F4d4 = 1000/4 = 250 Nm

SR .No.	Dist.	Forces (N)	Deform. (mm)	θ (degree)	T.R. (Nm/degree)
1	d1	480.8765	3.9287	0.8659	1154.8678
2	d2	479.2265	3.2086	0.7048	1418.8422
3	d3	601.5399	3.1732	0.8746	143.3798
4	d4	564.0428	2.7091	0.7004	1427.7556

Average (Final) T.R. = 1286.2113 Nm/degree

Average θ = 0.7864 degree

2) Torque (total) = 2000 Nm

Hence, F1d1 = F2d2 = F3d3 = F4d4 = 2000/4 = 500 Nm

SR .No.	Dist.	Forces (N)	Deform. (mm)	θ (degree)	T.R. (Nm/degree)
1	d1	961.753	7.8527	1.7303	1155.8689
2	d2	958.543	6.4159	1.4090	1419.4464
3	d3	1203.0799	6.3458	1.7492	1243.3798
4	d4	1128.2113	5.4202	1.4011	1427.4499

Average (Final) T.R. = 1286.5363 Nm/degree

Average θ = 1.5724 degree3. Torque (total) = 4000 Nm

Hence, F1d1 = F2d2 = F3d3 = F4d4 = 4000/4 = 1000 Nm

SR .No.	Dist.	Forces (N)	Deform. (mm)	θ (degree)	T.R. (Nm/degree)
1	d1	1923.5060	15.722	3.4612	1155.6685
2	d2	1916.9059	12.834	2.8169	1420.0002
3	d3	2406.1598	12.678	3.4913	1145.7050
4	d4	2256.1752	10.840	2.8003	1428.4184

Average (Final) T.R. = 1287.4484 Nm/degree

Average θ = 3.1444 degree

3) Torque (total) = 5000 Nm
Hence, F1d1 = F2d2 = F3d3 = F4d4 = 5000/4 = 1250 Nm

SR .No.	Dist.	Forces (N)	Deform. (mm)	θ (degree)	T.R. (Nm/degree)
1	d1	2404.3825	19.617	4.3157	1158.5606
2	d2	2396.1324	16.044	3.5198	1420.5353
3	d3	3007.6997	15.866	4.3662	1145.1605
4	d4	2820.289	13.56	3.5014	1428.0002

Average (Final) T.R. = 1288.4484 Nm/degree

Average θ = 3.9258 degree

4) Torque (total) = 6000 Nm
Hence, F1d1 = F2d2 = F3d3 = F4d4 = 6000/4 = 1500 Nm

SR .No.	Dist.	Forces (N)	Deform. (mm)	θ (degree)	T.R. (Nm/degree)
1	d1	2885.2590	23.561	5.1791	1158.5024
2	d2	2875.3589	19.253	4.2215	1421.290
3	d3	3609.2369	19.031	5.2327	1146.6356
4	d4	3384.2627	16.264	4.1926	1429.4904

Average (Final) T.R. = 1288.981 Nm/degree

Average θ = 4.7076 degree

5) Torque (total) = 7000 Nm
Hence, F1d1 = F2d2 = F3d3 = F4d4 = 7000/4 = 1750 Nm

SR .No.	Dist.	Forces (N)	Deform. (mm)	θ (degree)	T.R. (Nm/degree)
1	d1	3366.1355	27.469	6.0323	1160.497
2	d2	3354.5854	22.46	4.9214	1422.3595
3	d3	4210.7796	22.207	6.0999	1147.5598
4	d4	3948.3065	18.971	4.8928	1430.6444

Average (Final) T.R. = 1290.2458 Nm/degree

Average θ = 5.4866 degree

6) Torque (total) = 8000 Nm
Hence, F1d1 = F2d2 = F3d3 = F4d4 = 8000/4 = 2000 Nm

SR .No.	Dist.	Forces (N)	Deform. (mm)	θ (degree)	T.R. (Nm/degree)
1	d1	3487.012	31.405	6.8888	1161.3053
2	d2	3833.812	25.665	5.6195	1423.6142
3	d3	4812.3195	25.378	6.9629	1148.9466
4	d4	4512.3503	21.682	5.5878	1431.6945

Average (Final) T.R. = 1291.4126 Nm/degree

Average θ = 6.2647 degree

7) Torque (total) = 9000 Nm
Hence, F1d1 = F2d2 = F3d3 = F4d4 = 9000/4 = 2250 Nm

SR .No.	Dist.	Forces (N)	Deform. (mm)	θ (degree)	T.R. (Nm/degree)
1	d1	4327.8885	35.317	7.7371	1163.2265
2	d2	4313.0384	28.876	6.3172	1424.818
3	d3	5413.8595	48.545	7.8216	1150.6597
4	d4	5076.3941	24.342	6.281	1432.8928

Average (Final) T.R. = 1292.8652 Nm/degree

Average θ = 7.0392 degree

8) Torque (total) = 10000 Nm
Hence, F1d1 = F2d2 = F3d3 = F4d4 = 10000/4 = 2500 Nm

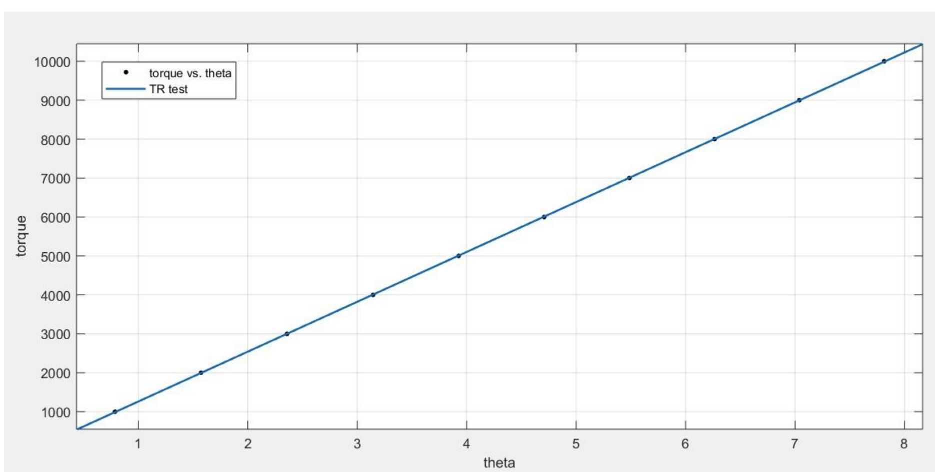
SR .No.	Dist.	Forces (N)	Deform. (mm)	θ (degree)	T.R. (Nm/degree)
1	d1	4808.7650	39.261	8.5889	1164.2934
2	d2	4792.2649	32.081	7.0117	1426.1876
3	d3	6015.3994	31.725	8.6804	1152.0206
4	d4	5640.4379	27.116	6.9758	1435.5853

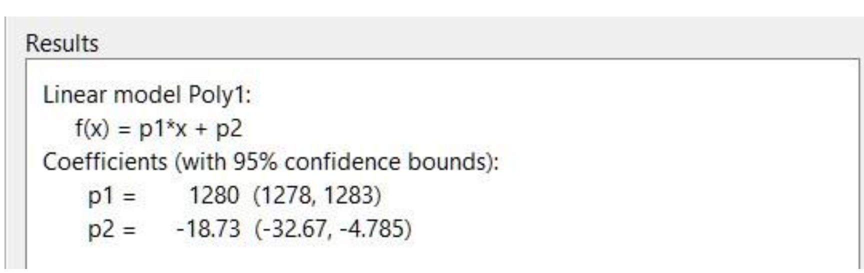
Average (Final) T.R. = 1294.5217 Nm/degree

Average θ = 7.8142 degree

- We observe that all T.R. values are nearly constant.
- Average of all T.R. values is 1289.3035 Nm/degree

5. After plotting best curve fit using MATLAB, we found its slope which is our final T.R. value



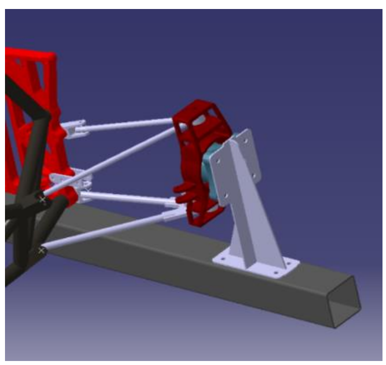


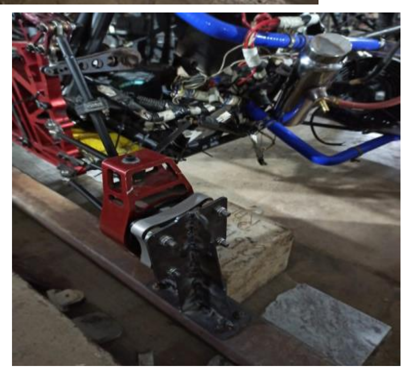
Hence, slope of above curve fit is 1280.

Therefore, **final simulated torsional stiffness is 1280 Nm/degree** BY FEA. Then we found T.R. practically by RIG and cross checked the results.

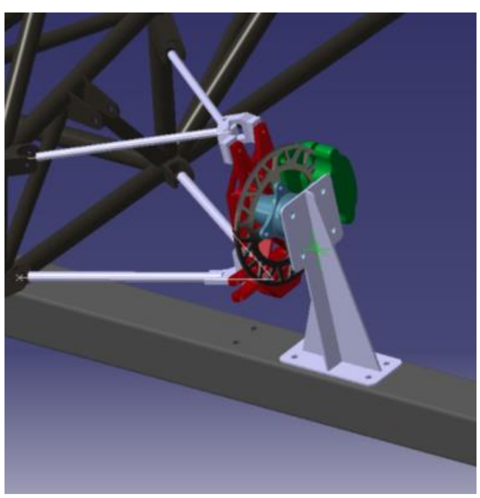
EXPERIMENTAL METHOD

The experimental setup is as shown below:

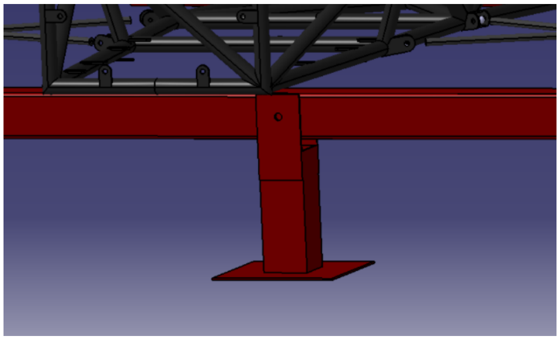




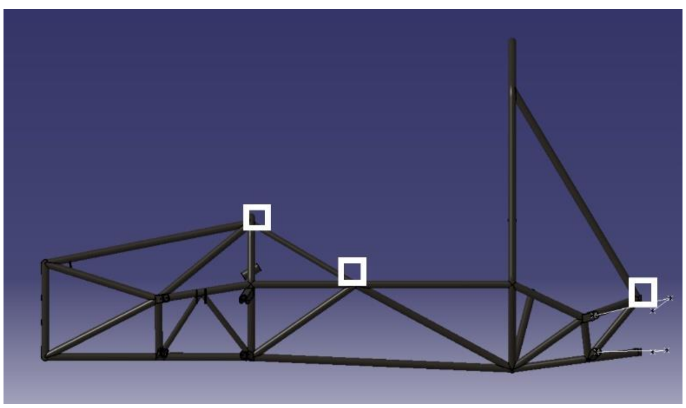
The whole chassis is bolted through wheel hubs on two beams parallel to the track width. The rear beam is fixed and restricts rolling and yawing of the chassis. Pitching is possible at rear wheel hubs which is restricted by the front beam. The front beam is supported on a pivot which is fixed to the ground.



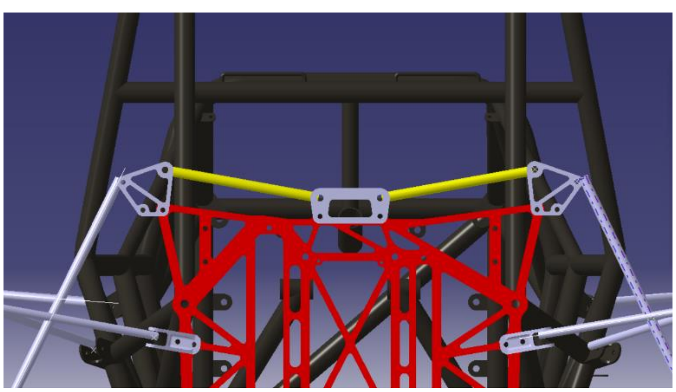
The front pivot was made of same cross section square pipe. An Allen bolt was used as the pivot to ensure a smooth rotation and adequate clearance was kept between the beam and the pivot. The setup allows rolling at the front part of the chassis on the pivot thereby restricting at the rear and creates a torsion on the chassis.

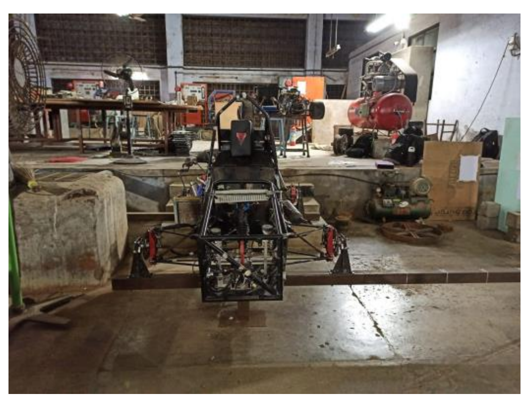


For calculating the twist angle of the chassis, lasers are mounted at 3 places - front suspension point, middle and rear bulkhead .The couple force is applied on the front beam by putting weights on the cantilever part of the beam. The twist angle produced in the chassis is calculated through laser displacement on the screen and distance of screen form the rolling axis of the chassis (or pivot point).



In the setup, dampers were replaced by rigid rods to eliminate any error that might occur due to spring action. Also the tension produced in the springs might damage the dampers. To minimize errors, firstly the weight of the cantilever beam was calculated and was used in observations and calculations.





The accuracy of the experiment can be increased by increasing the distance of the sheet from the rolling axis so as to get considerable displacement of laser on screen, taking more number of observations, and other parameters like straightness of the setup components and sufficient rigidity.

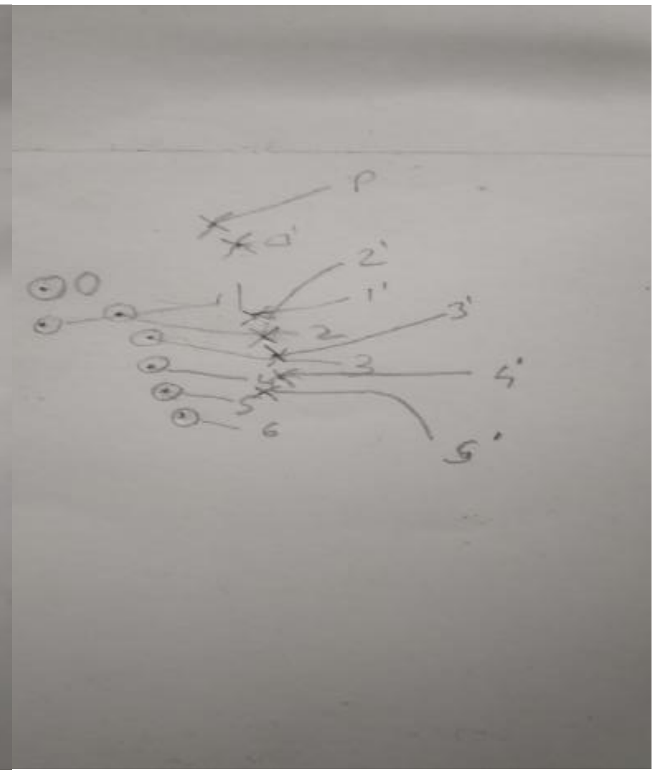
Experimentation

The white board was aligned in a plane parallel to the longitudinal plane passing through the pivot and center of the chassis. This board was at a distance of 3m front of the central plane. The board was kept in position as shown in the image above with the help of bricks. This board was kept vertical.

After the board and vehicle were placed, the lasers were first positioned at the desired locations. Their height was measured from the ground. After the lasers were put on, the laser was further adjusted to get a starting point on the screen that is at the same height as that of the laser so that any deflection further can be calculated with respect to this position simply be forming a right angle triangle and using geometry.

At this point the setup was done, and we started the experiment by applying a torque that was enough to eliminate the plays in the systems. The preloaded torque was the weight of the extended part of the cantilever beam and the weight added at a particular distance from the pivot. The position of the laser at this point was taken as the first reading. The deflection of laser at the screen was very minimal and hence we still could consider a right angled triangle for calculation, as also the deflection on screen was negligible with respect to the distance of the screen from the pivot. Further on, weights were added to apply torque in steps and readings were noted for loading condition. Weights were then removed in steps and readings were noted for unloading condition. The deflection measured on the screen are shown in the images below.

Front laser during front twisting



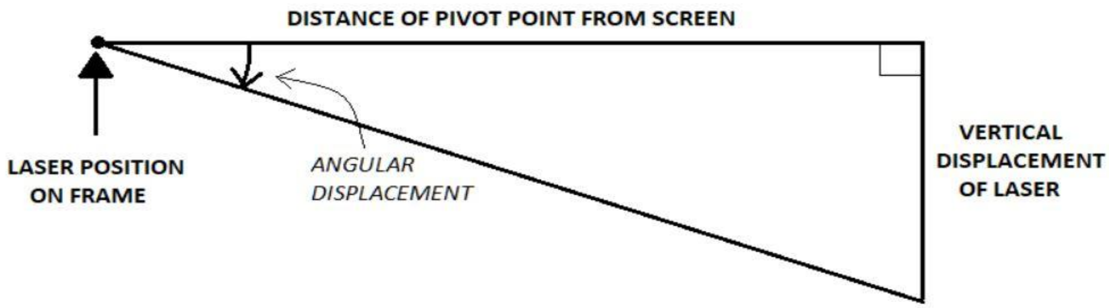
Rear laser during front twist

In the images above, during the first test the loading was marked by circle ‘O’ and unloading was marked by cross ‘X’. The reason for mentioning the rear laser is the displacement of the laser is subtracted from the corresponding front laser reading to get a relative displacement with respect to zero displacement of the laser at the rear.

Data collection and calculation

The measured data included the vertical displacement of the lasers, various geometries of the system, and the force applied to the lever arm. Using these values, the angular deflection was calculated using basic trigonometry. The triangle used to calculate the angular deflection is shown in the figure below.

Angular Deflection = θ = $\tan^{-1}$ (Vertical displacement of laser / Distance from laser to board)



After calculating the angular deflection, these points were then plotted against the torque applied the frame. The torque applied to the frame can be calculated from below equation.

$$Torque = T = moment\ arm * force\ applied$$

In this case, the moment arm is taken to be the distance between the applied load and the pivot point.

Data Presentation

The results of first torsional stiffness test are below. For the first torsional test three lasers were used and their positions were as shown in previous Image 6. We considered the following data which is tabulated below. This table contains the torque and angular deflection for the lasers mounted at the front bulk head (denoted as laser A) and at the front axle (denoted as laser B). The laser B is in more of our interest as it is in the plane where the suspension forces actually acts in and where the torsional stiffness is calculated from. For calculating the torsional stiffness, we used the best curve fit cut method (regression). The mathematical model for the experiment is slightly different from the analytical model in a sense that preload wasn't considered in the above model. The equation used for the experimental data is in consideration with the preloaded torque.

$$T = k * \theta + T_0$$

Here, T_0 = Preload torque i.e. the torque applied for which angular displacement is zero (i.e. θ=0)

k = torsional rigidity.

Observations

TR Rig Validation (FRONT twisting)

Preloaded mass - 6.2 kg
Distance from pivot - 2.247 m
Preloaded torque - 138.7485 N.m
Distance of screen - 2.91 m

Dead mass - 6.6 kg
Distance from pivot - 1.75 m
Dead torque - 113.3056 N.m
Equivalent dead torque - 252.0541 N.m

Loading

SR. NO.	Mass (kg)	Dist. From Pivot (m)	Torque (N-m)	Laser F (mm)	θ_F (Deg)	Laser M (mm)	θ_M (mm)	Laser R (mm)	θ_R (Deg)	θ_F-θ_R (Deg)
1	+9.4	2	436.482	18.72	0.3686	18.1	0.2973	4.65	0.0915	0.2771
2	+9.4	2	620.91	33.34	0.6564	28.24	0.5678	3.57	0.0703	0.5861
3	+9.4	2	805.388	47.97	0.9444	41.19	0.8109	6.54	0.1288	0.8156
4	+9.4	2	989.766	62.46	1.2296	54.09	1.0649	10.53	0.2073	1.0223
5	+9.4	2	1274.194	76.1	1.498	66.74	1.3138	14.11	0.2778	1.2202
6	+18.1	1	1451.755	89.82	1.7679	78.79	1.5509	17.54	0.3436	1.4243

Unloading

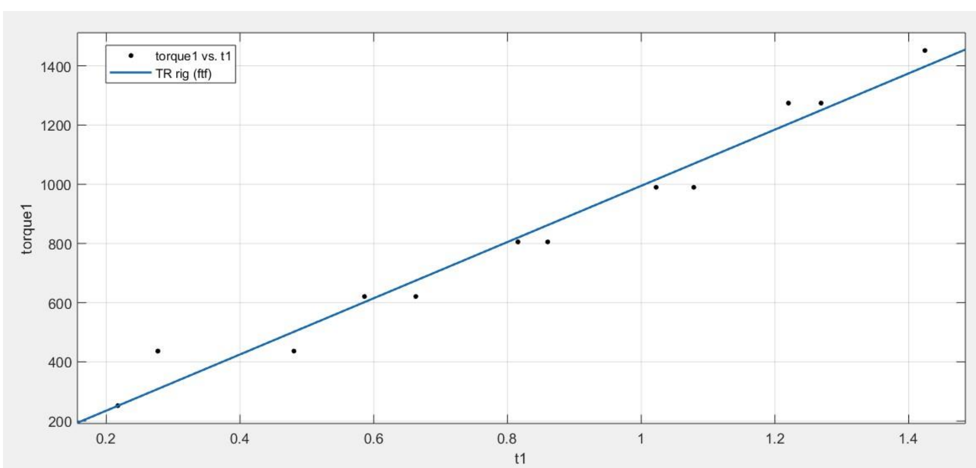
SR. NO.	Mass (kg)	Dist. From Pivot (m)	Torque (N-m)	Laser F (mm)	θ_F (Deg)	Laser M (mm)	θ_M (mm)	Laser R (mm)	θ_R (Deg)	θ_F-θ_R (Deg)
7	-18.1	1	1274.194	78.55	1.5462	68.27	1.3439	14.08	0.2772	1.269
8	-9.4	2	989.766	67.12	1.3213	58.08	1.1434	12.33	0.2427	1.0786
9	-9.4	2	805.338	53.33	1.0499	46.3	0.9115	9.64	0.1898	0.8601
10	-9.4	2	620.91	40.65	0.8003	33.18	0.6533	6.98	0.1374	0.6629
11	-9.4	2	436.428	28.15	0.5542	21.88	0.4308	3.64	0.0726	0.4806

By best curve fit method, slope of torque vs theta line was found as follow,

Y - torque

X - theta

The figure below shows the plot of torque versus angular deflection.



Results

Linear model Poly1:
f(x) = p1*x + p2
Coefficients (with 95% confidence bounds):
p1 = 949.3 (837.2, 1062)
p2 = 45.46 (-56.33, 147.2)

TR Rig Validation (REAR twisting)

Preloaded mass - 5.2 kg
Distance from pivot - 2.083 m
Preloaded torque - 116.8178 N.m

Dead mass - 6.6 kg
Distance from pivot - 1.75 m
Dead torque - 113.3056 N.m

Distance of screen - 2.91 m

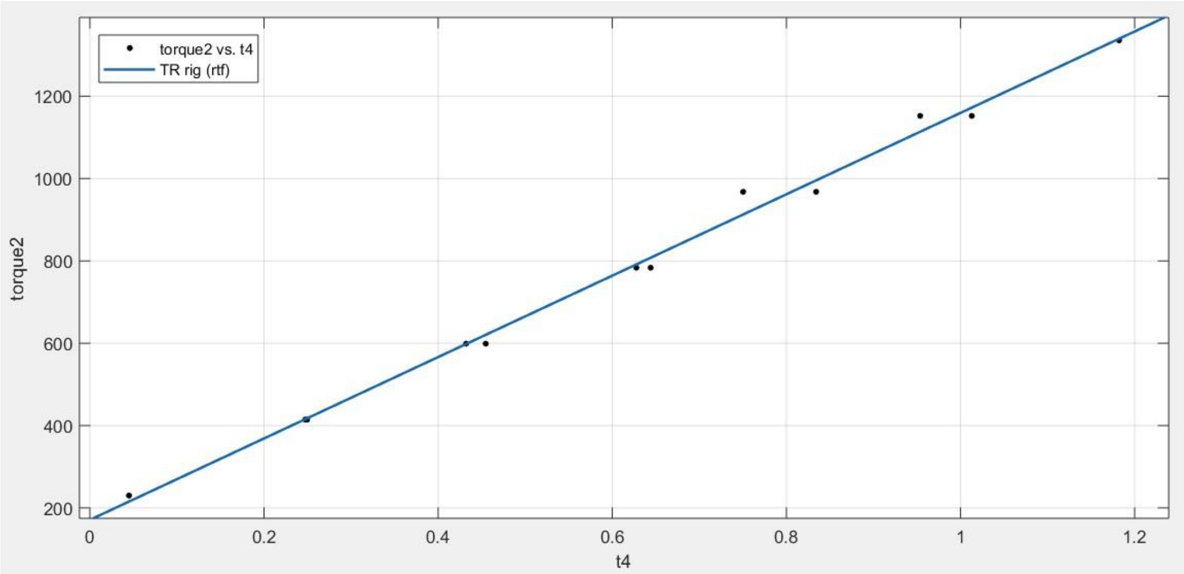
Equivalent dead torque - 230.1234 N.m

Loading

SR. NO.	Mass (kg)	Dist. From Pivot (m)	Torque (N-m)	Laser F (mm)	θ_F (Deg)	Laser M (mm)	θ_M (mm)	Laser R (mm)	θ_R (Deg)	θ_F-θ_R (Deg)
1	+9.4	2	414.5513	18	0.3544	6.25	0.1231	5.44	0.1071	1.269
2	+9.4	2	598.9793	34.16	0.6725	11.12	0.2189	11.07	0.2179	1.0786
3	+9.4	2	783.4073	46.29	0.9113	15.58	0.3067	14.41	0.2837	0.8601
4	+9.4	2	967.8352	61.91	1.2188	22.57	0.4444	19.54	0.3847	0.6629
5	+9.4	2	1152.2632	74.84	1.4732	26.54	0.5225	23.38	0.4603	0.4806
6	+18.1	1	1335.8242	86.68	1.7062	31.05	0.6113	26.62	0.5241	0.2172

Unloading

SR. NO.	Mass (kg)	Dist. From Pivot (m)	Torque (N-m)	Laser F (mm)	θ_F (Deg)	Laser M (mm)	θ_M (mm)	Laser R (mm)	θ_R (Deg)	θ_F-θ_R (Deg)
7	-18.1	1	1152.2632	75.07	1.4777	26.96	0.5308	26.62	0.5241	0.9536
8	-9.4	2	967.8352	61.49	1.2105	23.04	0.4536	23.38	0.4603	0.7502
9	-9.4	2	783.4073	48.65	0.9578	17.9	0.3524	15.94	0.3138	0.6440
10	-9.4	2	598.9793	33.57	0.6609	12.18	0.2398	11.62	0.2288	0.4321
11	-9.4	2	414.5513	19.08	0.3756	7.37	0.1451	6.41	0.1262	0.2494



```
Results
Linear model Poly1:
    f(x) = p1*x + p2
Coefficients (with 95% confidence bounds):
     p1 =       988.5  (936.8, 1040)
     p2 =       171.1  (134.8, 207.3)
```

Torsional Stiffness for front twisting is 949.3Nm/degree and rear twisting is 988.5Nm\degree

- Final experimental torsional stiffness by averaging is 968.9 Nm/degree.

  So, the final simulated torsional stiffness is 1280 Nm/degree and final experimental torsional stiffness is 968.9 Nm/degree. This gave an absolute relative error of 24.3%.

  The error in stimulated value and experimental value is because of the error in the manufacturing process, difference in the pure and impure material properties and error in experimental setup.

Conclusion

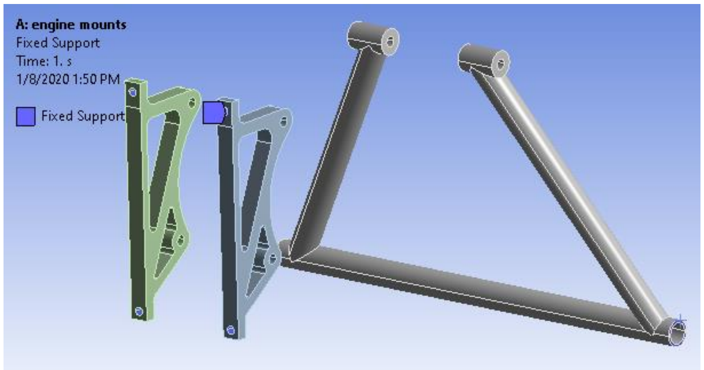
As the chassis gets less stiff the vehicle behaves in an increasing undesirable manner to driver inputs, and becomes increasing hard to drive. Vehicles should be designed for the lightest chassis possible that has a torsional stiffness greater than a target value, 1300 Nm/deg in our case, with further research required on the targeted value. Stiffness greater than this value result in added weight for diminishing returns, while stiffness less than this value put the car in danger of entering a regime in which the stiffness has significant effect on vehicle behavior.

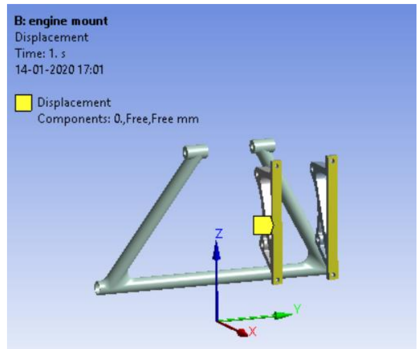
Engine Mounts

i. **Fixed Supports**

Member of the chassis in contact with the engine mounts are chosen as fixed supports. It was chosen as fixed support because,

- mounts are directly welded to the member
- members are least affected by various forces

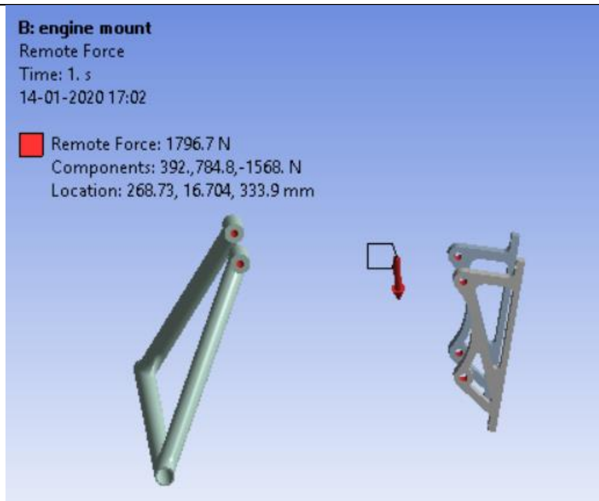
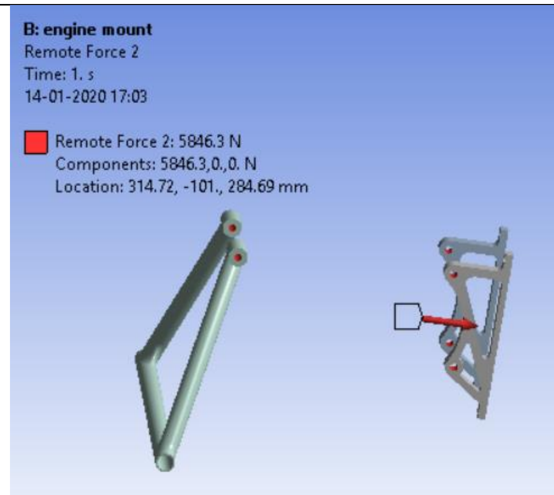




ii. Load Points

Inner surface where engine bolt is fitted is chosen as load point.
They were chosen as load points because,

- They are attached to the engine
- Forces on engine will be distributed on the mounts through those points

<table>
<tr><td>
</td><td>
</td></tr>
<tr><td>Remote force 1</td><td>Remote force 2</td></tr>
</table>

Type	Remote Force
Define By	Components
X Component	392. N (ramped)
Y Component	784.8 N (ramped)
Z Component	-1568. N (ramped)

Magnitude of remote force 1

Type	Remote Force
Define By	Components
X Component	5846.3 N (ramped)
Y Component	0. N (ramped)
Z Component	0. N (ramped)

Magnitude of remote force 2

iii. Functions

- Holding engine
- Aluminum RBH cannot be connected to MS mounts very efficiently, thus aluminum mounts are used.
- In order to reduce the vibrations caused by engine, degree of freedom is reduced, by mounting the engine by all the points.

iv. Forces

- ***Bump force***
  3g in positive z direction
  = 3*40*9.81=1177.2 N,
  as weight of engine is 40kg.
- ***Centrifugal force***
  2g is acting on engine while turning in (lateral) y direction-
  2g = 2*40*9.81=784.8N
- ***Inertia force***
  1g in negative x direction due to applying breaks in your maximum velocity
  = 1*40*9.81=392.4N
- ***Gravitational force***
  Weight due to gravity= 1g
- ***Tension forc1`es***
  (Torque/Radius) acting on chain of sprocket in positive x direction-

It is maximum at the time of launching, pitch circle radius is 0.1163135 m and torque acting at the time of launching is 680 Nm.

Torque = T1 * R

… (Where T1= Tension on tight side of sprocket chain

R = Pitch circle radius)

680Nm = T1 * 0. 1163135 m

680000Nmm = T1 *116.3135mm (4.579in=.1163066m)

T1 = 5846.2689N

- ***Final Direction of forces-***

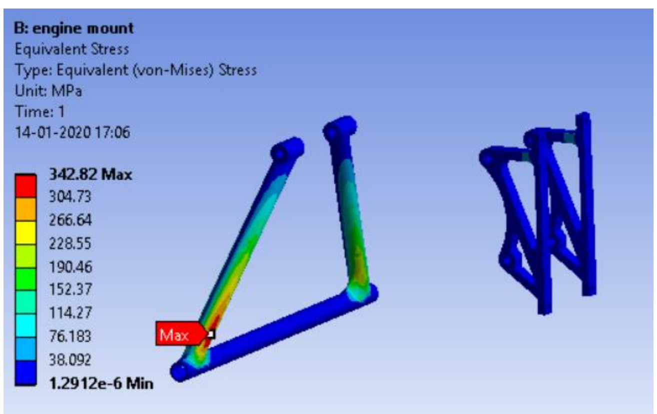
  -In positive x direction, T1 + 1g =5846.2689+392.4= 6238.6686N

  -In positive y direction, 2g=784.8N

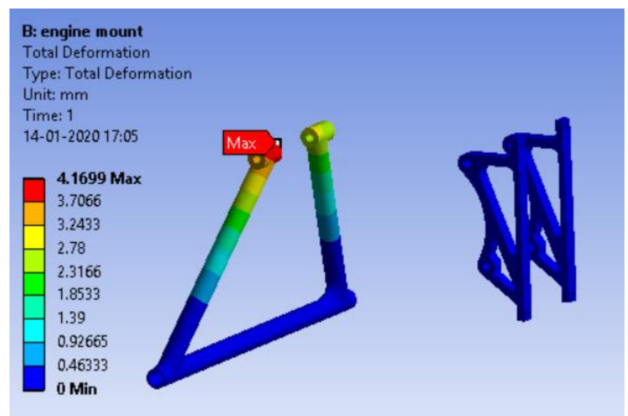
  -In negative z direction, 3g+1g= 4g= 1569.6N

v. Result

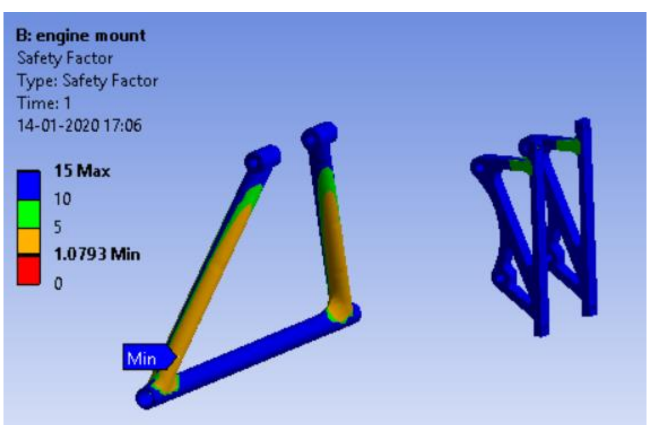
- Maximum equivalent stress is 342.82 MPa
- Maximum deformation is 4.1699 mm
- Minimum safety factor is 1.0793
- Minimum fatigue life is 7436



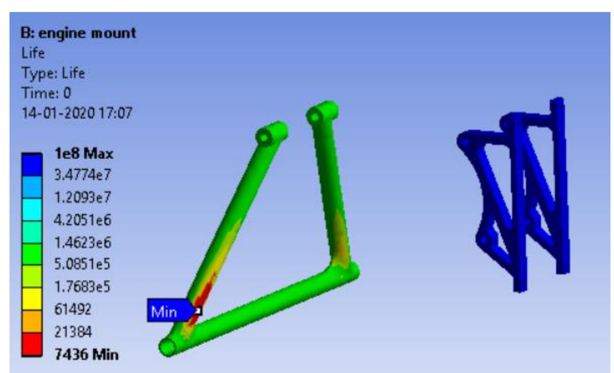
Equivalent Stress



Total Deformation



Factor of Safety



Life

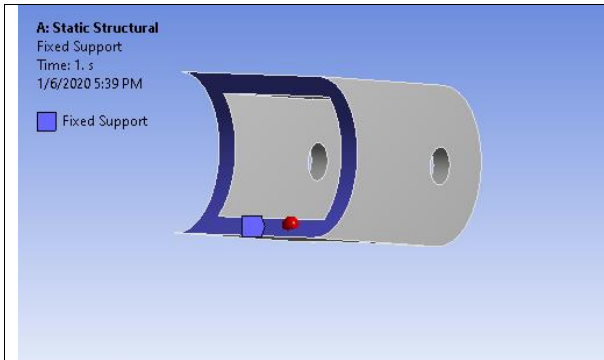
Conclusion

Engine mount is safe in all worst conditions.

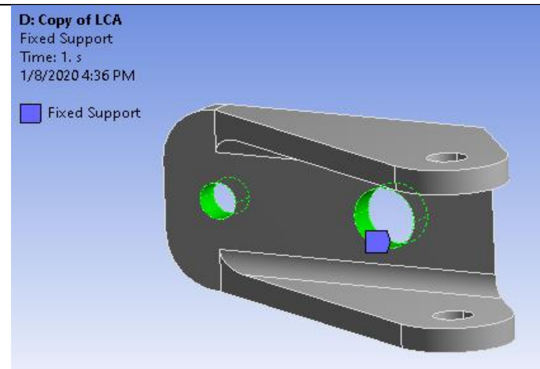
<u>Suspension Mounts</u>

1. **Fixed Support** –

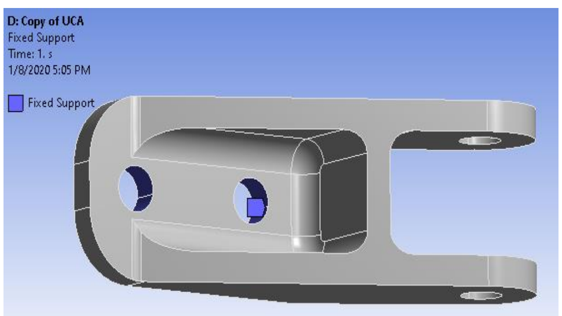
Notch of the mounts are taken as fixed support. It was chosen as fixed support because, they are welded to chassis and considered as fixed support as shown,



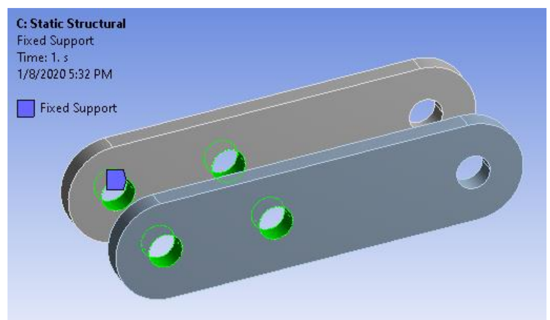
Suspension Mount



LCA



UCA

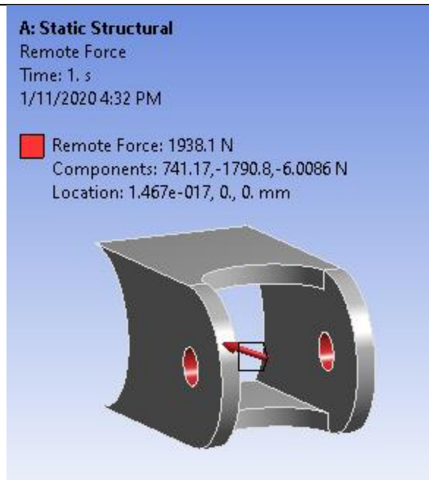


Toe mount

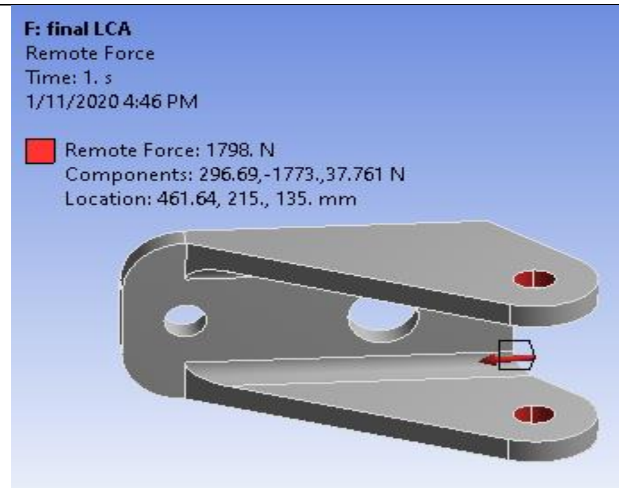
i. Load Points

Force applied at the location of center of rose joint is chosen as load point.
They were chosen as load points because

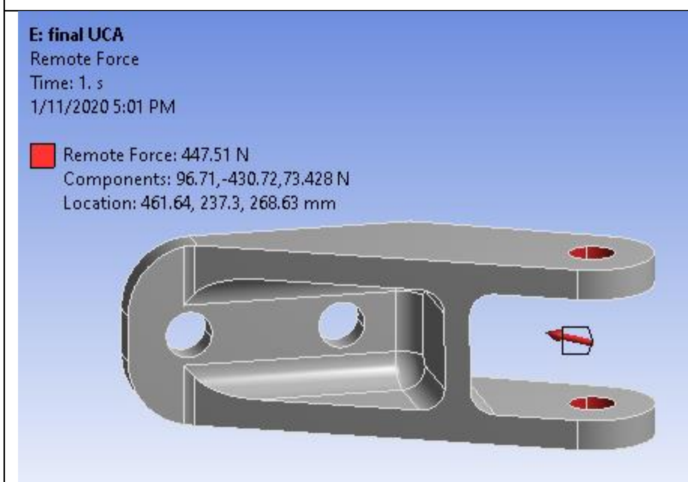
1. Forces that will act where rose joint is there between suspension mount.



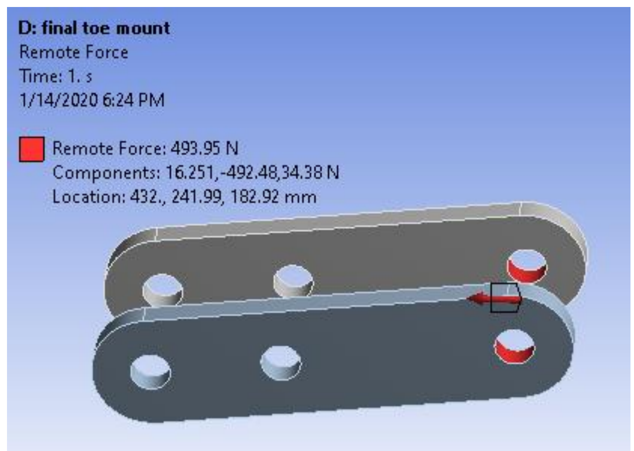
Suspension Mount



LCA



UCA



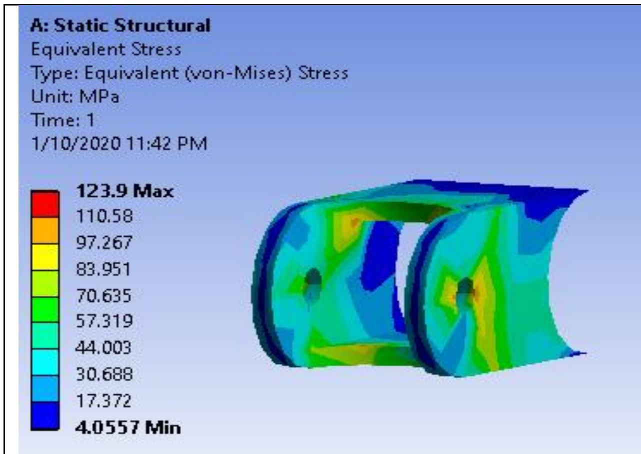
Toe mount

2. Front suspension mounts are made of MS
3. Due to Aluminum Rear Bulk Head, rear suspension mounts are of aluminum.
4. Thus, two different materials are used for suspension mounts.

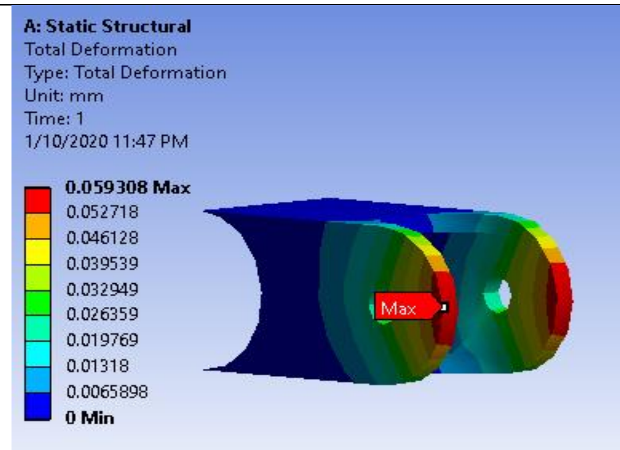
ii. Forces

- Various forces act at the wheel center, like Bump force, Braking force, Inertia force while acceleration and centrifugal forces while turning.
- It leads to act moments along with forces at suspension points.
- These are transmitted towards suspension mounts through A Arms, Push Rod, and Tie Rod etc.
- Analytical calculations of these forces was done by VD department and there magnitude along with unit vectors were found as below,

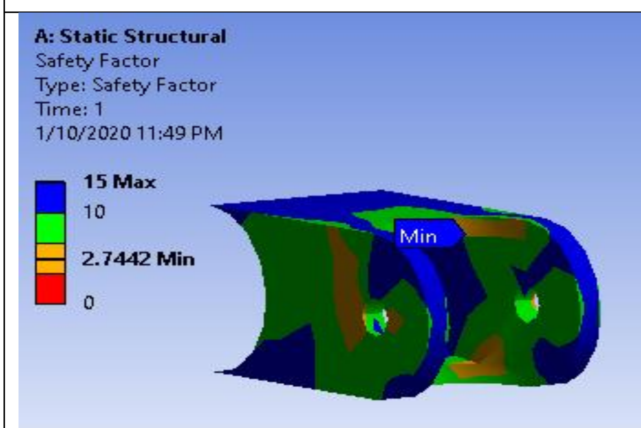
Suspension Mounts



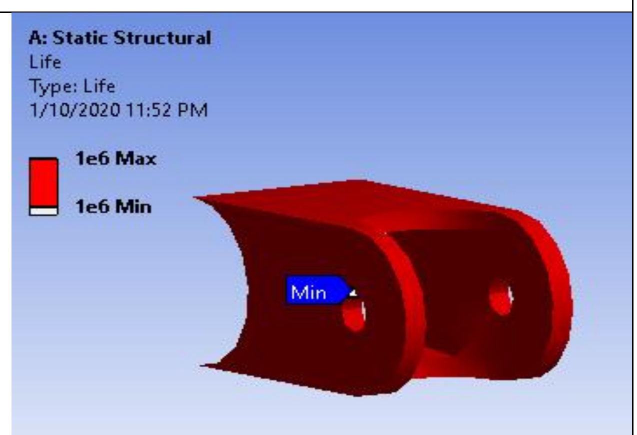
Equivalent Stress



Total Deformation



Safety Factor



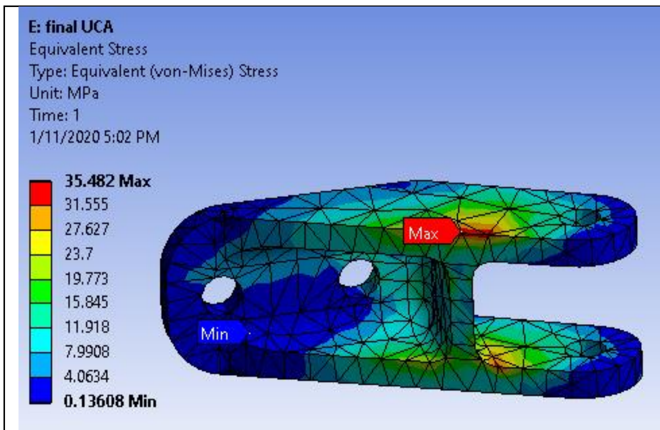
Fatigue Life

SR	Quantity	Values
1	Maximum equivalent stress	123.9 MPa
2	Maximum total deformation	0.0593 mm
3	Minimum safety factor	2.7442
4	Minimum fatigue life	1e6

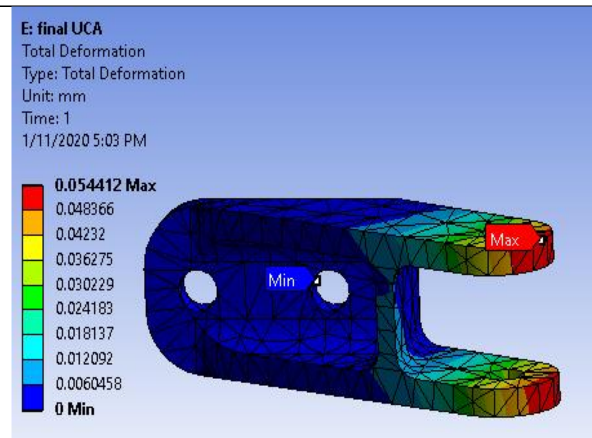
Aluminium Suspension Mounts

UCA(R)

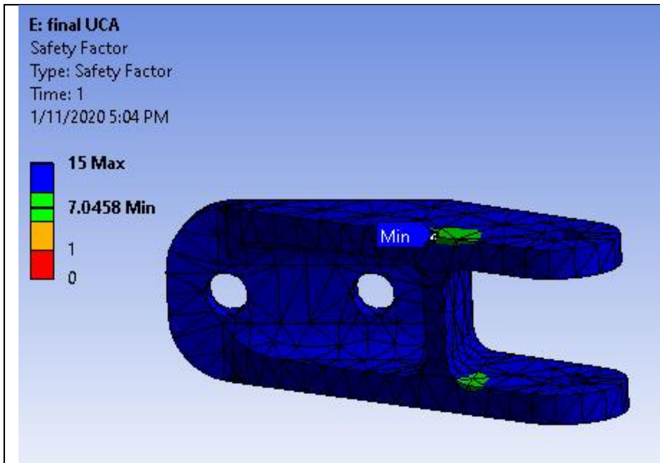
Direction	Magnitude	Unit Vector	Component Force
X	447.7304	0.216	966.7097
Y	447.7304	-0.982	-430.7166
Z	447.7304	0.164	73.4278



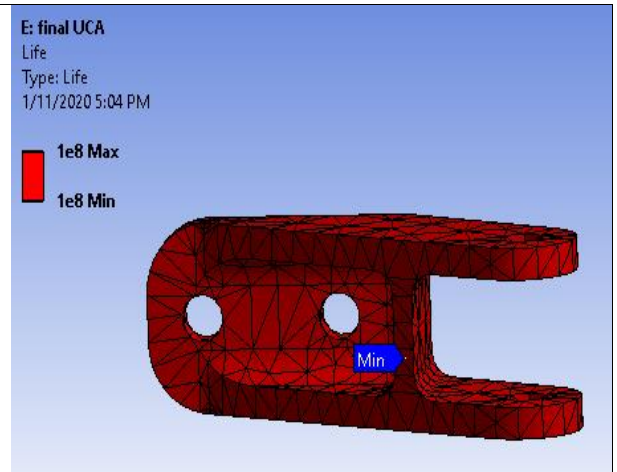
Equivalent Stress



Total Deformation



Safety Factor

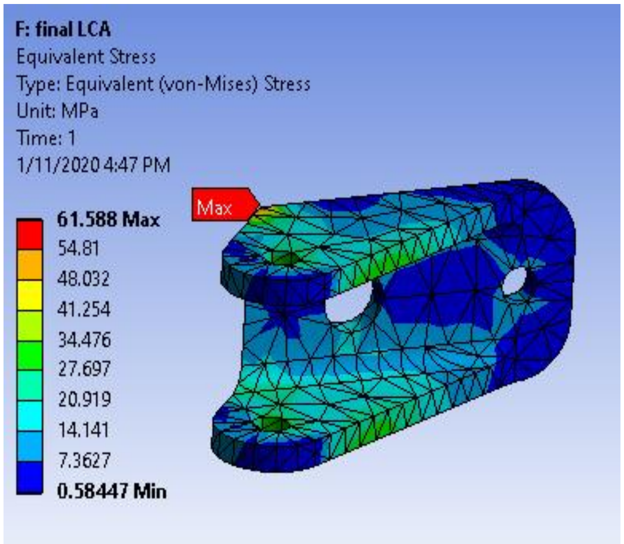


Fatigue Life

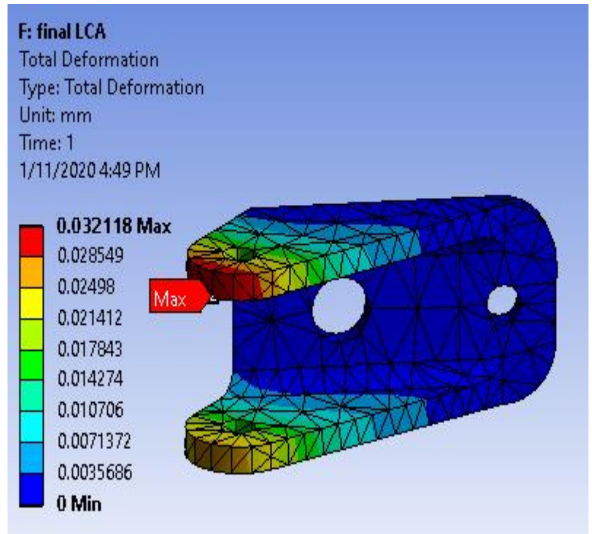
SR	Quantity	Values
1	Maximum equivalent stress	35.482 MPa
2	Maximum total deformation	0.05441 mm
3	Minimum safety factor	7.0458
4	Minimum fatigue life	1e8

LCA(R)

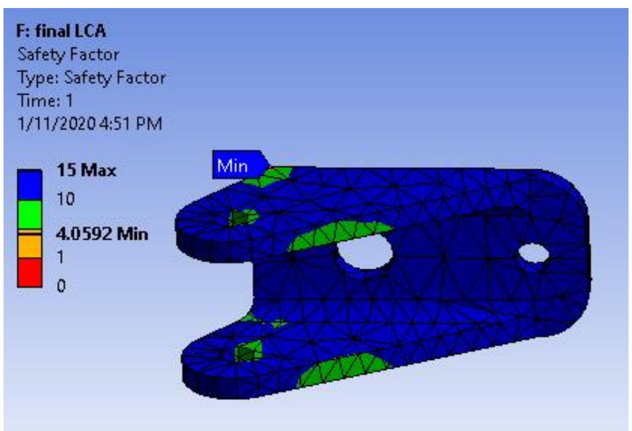
Direction	Magnitude	Unit Vector	Component Force
X	-1798.1337	0.165	-296.6921
Y	-1798.1337	-0.986	1772.9598
Z	-1798.1337	0.021164	-37.76081



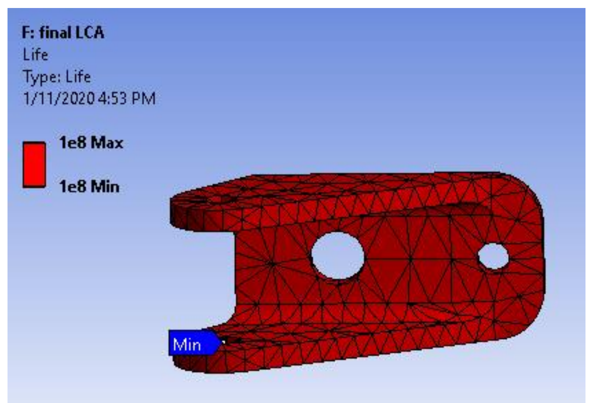
Equivalent Stress



Total Deformation



Safety Factor



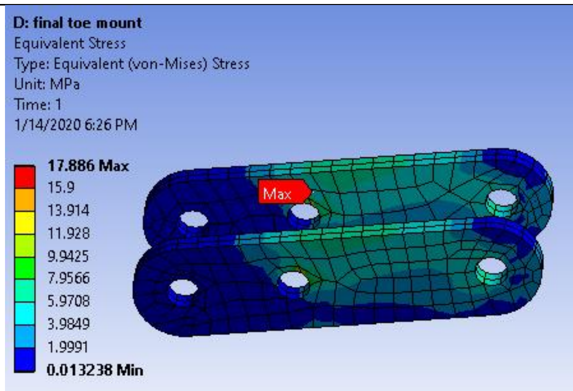
Fatigue Life

iii. Results

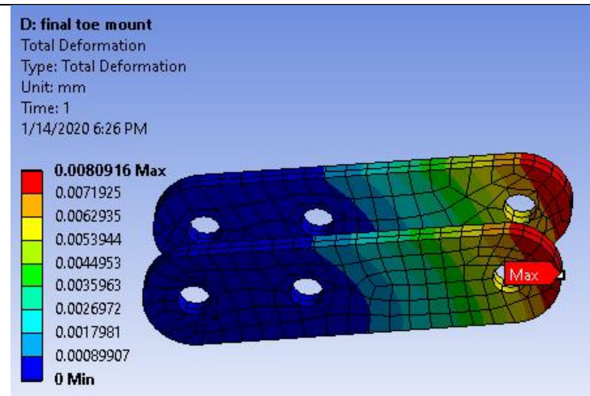
SR	Quantity	Values
1	Maximum equivalent stress	61.588
2	Maximum total deformation	0.032118
3	Minimum safety factor	4.0592
4	Minimum fatigue life	10^8

Toe mount

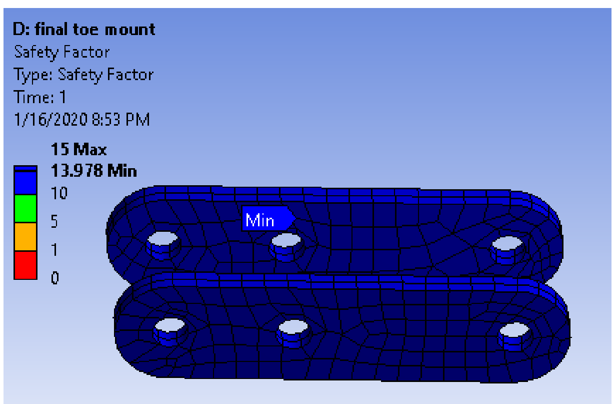
Direction	Magnitude	Unit Vector	Component Force
X	493.9619	0.0329	16.2513
Y	493.9619	-0.997	-492.4800
Z	493.9619	0.0696	34.3797



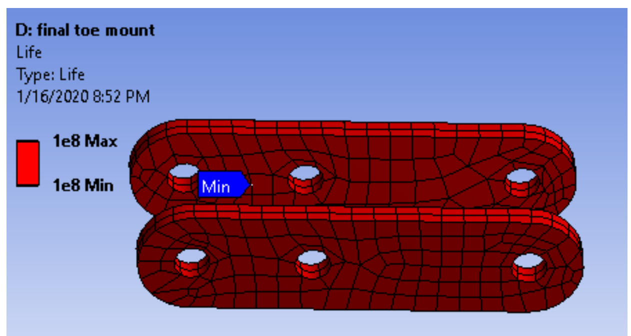
Equivalent Stress



Total Deformation



Safety Factor



Fatigue Life

SR	**Quantity**	**Values**
1	Maximum equivalent stress	17.886
2	Maximum total deformation	0.00809
3	Minimum safety factor	13.978
4	Minimum fatigue life	1e8

iv. Conclusion-

All results are within permissible values, hence all suspension mounts are safe to use.

<u>Head Restraint Pipes Analysis</u>-

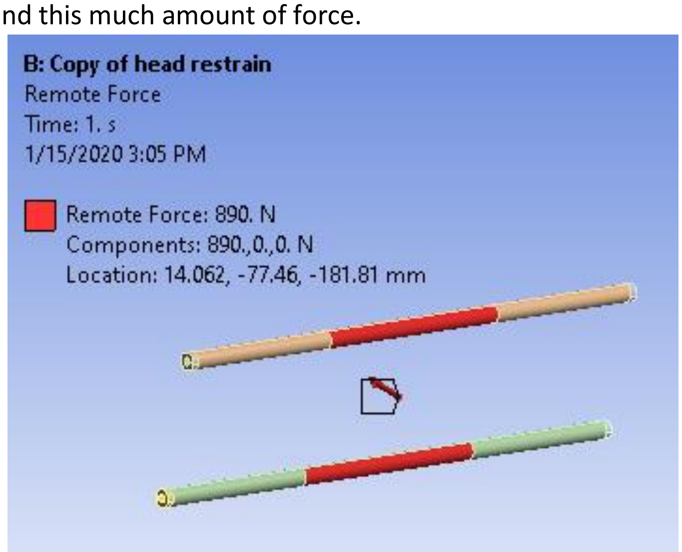
Boundary Condition-

1. **Fixed Support-**

The pipes are bolted to the chassis at the end, so the end of the pipes are consider as fixed support.

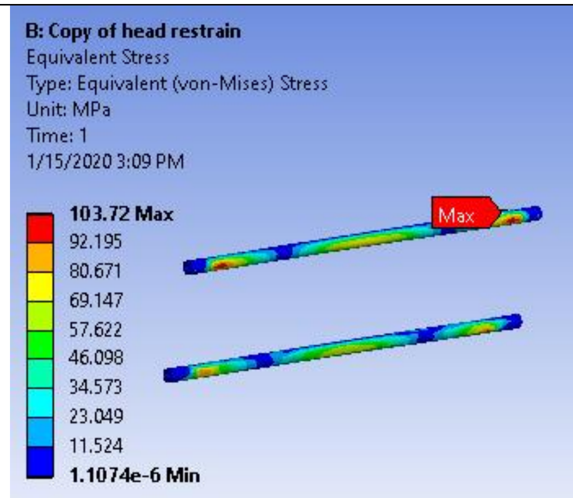
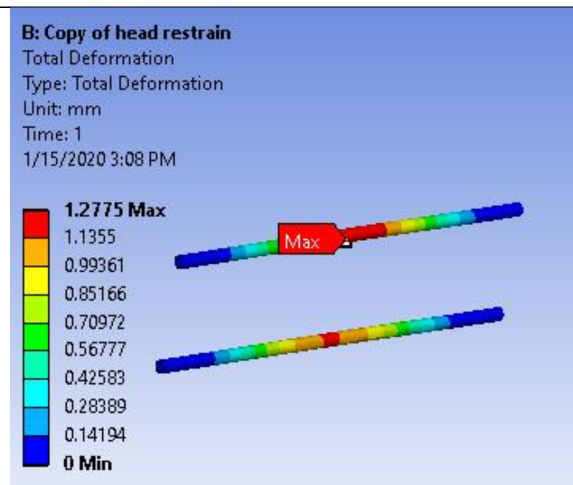
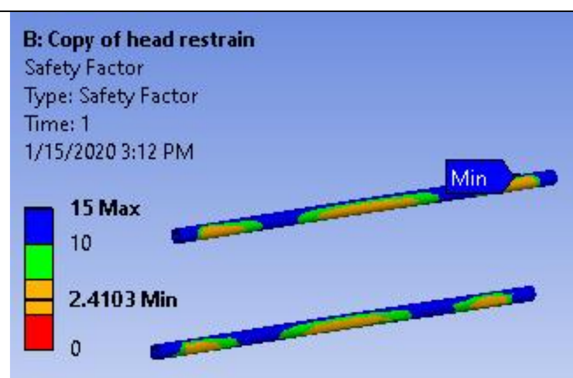
2. **Force-**

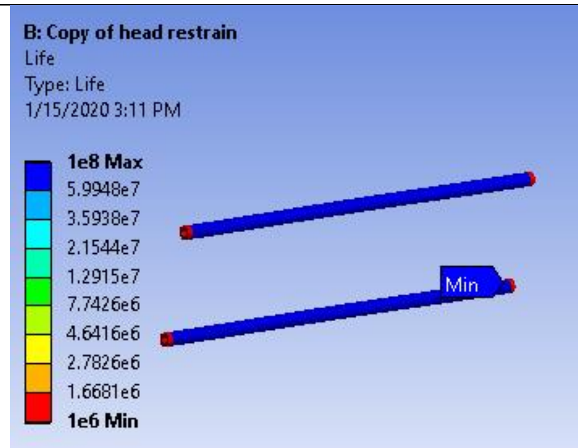
Force is transfer from the head restraint to the pipe where they are in contact with it. Remote force is assumed to act at center position of plate as it will be contact point between driver's head and plate. So the remote force of 890N is applied as mentioned in the rulebook to withstand this much amount of force.



Direction of this force will be normal to plate that is only in negative X direction as the impact will be only in negative X direction.

3. Results

Sr. No.	Quantity	
1	Maximum equivalent stress	
2	Maximum total deformation	
3	Minimum safety factor	

4	Minimum fatigue life	

SR	Quantity	Values
1	Maximum equivalent stress	103.72
2	Maximum total deformation	1.2775
3	Minimum safety factor	2.4103
4	Minimum fatigue life	10^6

Vibrational analysis of Engine Mount

Member of the chassis in contact with the engine mounts are chosen as fixed supports. It was chosen as fixed support because,

- mounts are directly welded to the member
- members are less affected by various forces than the mounts.

i. **Theory**-

a) When we start engine, it vibrates with its *IDLING FREQUENCY* which is 3200 rpm and with MAXIMUM FREQUENCY which is 9500 rpm for our engine.
b) Design of engine mounts should be such that natural frequency of mounts should not resonate with engine's frequency at any point, otherwise it can damage the mounts and/or chassis.
c) Thus natural frequency range of engine mounts is kept outside the range of frequency limit of engine.
d) To reduce the vibrations, degree of freedom has to be reduced. It is done by mounting engine at all the points.

ii. CALCULATION FOR RANGE OF ENGINE FREQUENCY-

f = $\frac{N*c}{60*n}$

… (f = Frequency of vibration in Hz

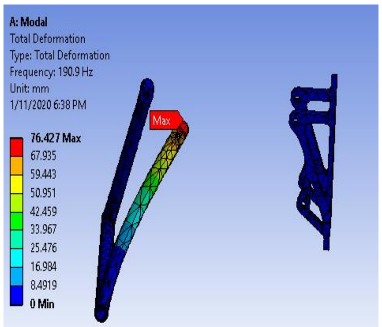
N = Engine rpm [3200 rpm and 9500 rpm]

c = Number of engine cylinders [1]

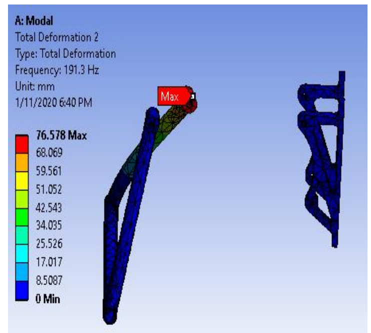
n = Number of crank rotations per cycle [2])

$$f = \frac{3200*1}{60*2} = 26.6667 \text{ Hz and } \quad f = \frac{9500*1}{60*2} = 79.16667$$

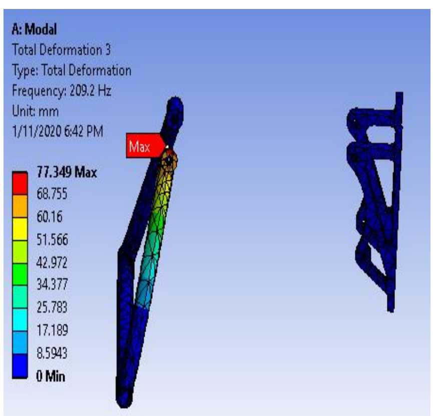
Therefore, frequency range of engine is 26.6667 Hz to 79.16667 Hz.

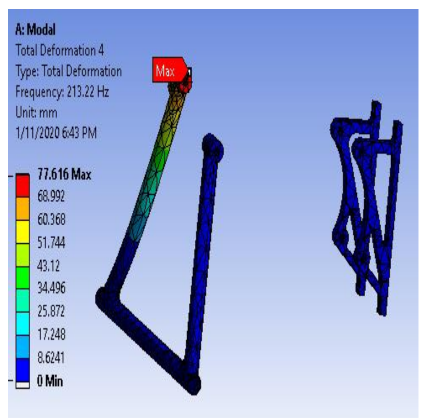


Mode 1

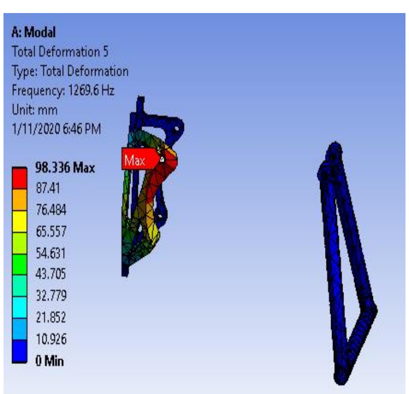


Mode 2

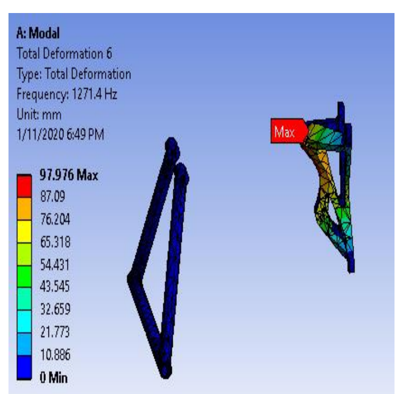




Mode 4



Mode 5

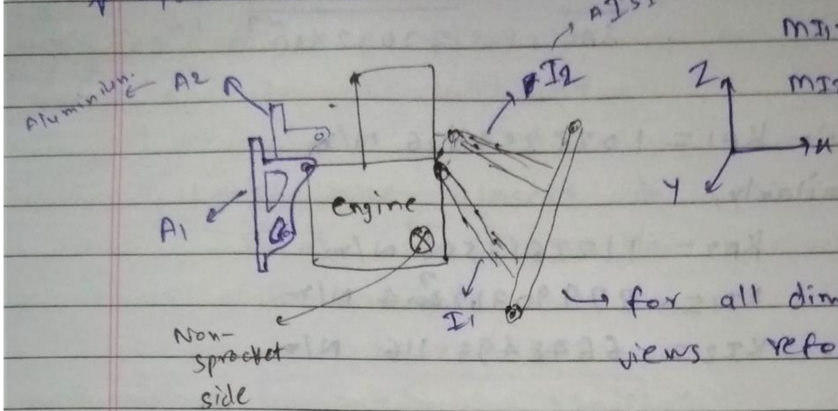


Mode 6

* Natural Vibrations :-

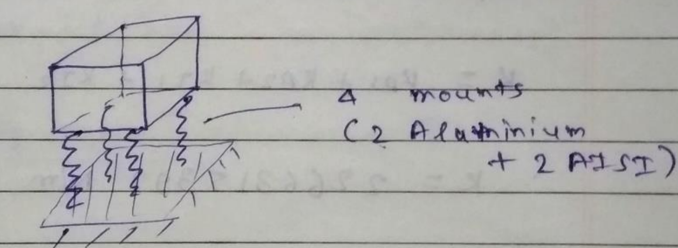
$M_{A2} = 0.309\,kg$
$M_{A1} = 0.309\,kg$
$M_{I1} = 0.415756\,kg$
$M_{I2} = 0.415756\,kg$

* System is →



↳ for all dimensions & views refer drafted pdf.

It is like



∴ Here it is like having 4 springs in parallel, having perticular stiffness in z-direction. (single degree of freedom) ↳ assume.

* Mass of engine is 40 kg (∴ 40 × 10 = 400N)
From ansys, deformation in z-direction after applying weight force only at C.G. by remote force is,

$\delta_{A1} = 3.7097 \times 10^{-7}\,m$
$\delta_{A2} = 3.3967 \times 10^{-7}\,m$
$\delta_{I1} = 1.0529 \times 10^{-4}\,m$
$\delta_{I2} = 6.0164 \times 10^{-5}\,m$

$$\therefore K_{A1} = \frac{F}{\delta_{A1}} = \frac{400}{3.7097 \times 10^{-7}} \text{ N/m}$$

$$\therefore K_{A1} = 1078254306 \text{ N/m}$$

Similarly,

$K_{A2} = 1177613566$ N/m

$K_{I1} = 379903127.4$ N/m

$K_{I2} = 6648494.116$ N/m

$\therefore$ As these are in parallel, equivalent spring stiffness K is

$$K = K_{A1} + K_{A2} + K_{I1} + K_{I2}$$

$$\therefore K = 2266315397 \text{ N/m}$$

* $\therefore$ Natural frequency of system is,

$$\omega_n = \sqrt{\frac{K}{m}}$$

$$= \sqrt{\frac{2266315397}{40}}$$

$$\therefore \omega_n = 7527.1432 \text{ rad/s}$$

$\therefore$ Wrong → because in this moment due (mg) not considered.
Have to do it by D'Alembert's principle.

* From ansys, (modal analysis) :-

For only mounts (without chassis)

one half wave — 1^{st} Mode → 149.8 Hz

2 half waves — 2^{nd} Mode → 153.21 Hz

3 half waves — 3^{rd} Mode → 164.61 Hz

∴ Let's take closest to our range.

∴ Final f_n = 149.8 Hz

$$\therefore \omega_n = 2\pi f_n$$

$$= 2 \times \pi \times 149.8$$

$$\therefore \boxed{\omega_n = 941.2211 \text{ rad/s}}$$

* Let's first try to get Critically damped vibrations i.e. $\xi = 1$

$$\text{as, } \xi = \frac{c}{2 m \omega_n}$$

$$\therefore c = 2 m \omega_n \xi$$

$$= 2 \times 40 \times 941.2211 \times 1$$

$$\therefore c = 75297.680 \text{ N-s/m}$$

* Frequency of damped vibrations is

$$\omega_d = \left(\sqrt{1 - \xi^2}\right) \omega_n$$

$$= \left(\sqrt{1 - 1}\right) \omega_n$$

$$\therefore \omega_d = 0 \text{ rad/s}$$

↳ obvious for $\xi = 1$

Forced Damped Vibrations:-

→ ~~As~~ Let's assume that only unbalanced force causing vibrations is due to reciprocating motion of piston, hence rotation of crank.

→ Though ~~it~~ ~~can~~ they are comming through unbalanced and frequently comming forces through tires in dynamic conditions, drivetrain assemblies etc.

→ We are concerned only about ~~engine~~ vibrations due to engine.

→ Another assumption is made that force is getting applied more only during power / expansion stroke. So let's calculate frequency of this unbalanced force.

$$\text{frequency } (f) = \frac{(\text{engine rpm}) \times (\text{no. of cylinders})}{60 \times (\text{no. of strokes/cycle of crank})}$$

~~Let~~ Say, if our idling to max rpm range of our KTM 390 Duke engine is 4000 rpm to 9500 rpm,

then $f = \dfrac{4000 \times 1}{60 \times 2} = 33.34 \text{ Hz}$

and, $f = \dfrac{9500 \times 1}{60 \times 2} = 79.167 \text{ Hz}$

This is frequency range of unbalanced force getting applied.

→ Logic behind the formula is, that, engine rpm that we have is of crank.

→ Say x rpm. (∴ in 1 minuite x-revolutions of crank)

→ ∴ $= \frac{x}{60}$ r.p.s. (Hz) (revolutions in one second)

→ $= \frac{x}{60} \times c$ (for c no. of cylinders)

→ $= \frac{x}{60} \times \frac{c}{2}$ (for 4-stroke engine for 2 revolutions of crank, only single time force is getting applied at power stroke)

ω range is

$\omega_{min} = 2\pi \, 33.34 = 209.4813$ rad/s

$\omega_{max} = 2\pi \, 79.167 = 497.421$ rad/s

→ Disturbing / Unbalanced force is

$f_0 = m_p r \omega^2$

m_p: mass of reciprocating part (piston)

r: (length of stroke)/2

ω: [according to khurmi it is engine rpm. But I think may be it should be force frequency]

↳ that can be confirmed by how formule is used i.e. $f_0 = m_p r \omega^2$

∴ ~~According to force frequency,~~

~~f_{0min}~~ Now we need engine specification of KTM 390 duke, which we got from fmsci

* Engine Specifications:-
1) Weigh' of piston = 316 gram
2) Piston pin weight = 67 grams.
But by searching online, (not much confirmed) pin floats, i.e. it don't move.
So weight is taken of piston only.
$\therefore$ Weight of piston = m_p = 316 grams = 0.316 kg

2) Stroke length (l) = 60 mm
$\therefore \; r = \frac{l}{2} = 30 \text{ mm} = 0.03 \text{ m}$

* $\therefore$ Unbalanced / disturbing force is,
$\Rightarrow$ According to frequency of force,
$f_{omin} = 0.316 \text{ kg} \times 0.03 \text{ m} \times (\omega_{min})^2$

$= 0.316 \times 0.03 \times (209.4813)^2$
$\therefore f_{omin} = 416 \text{ N}$
~~Sim~~ Similarly,
$f_{omax} = 0.316 \times 0.03 \times (\omega_{max})^2$
$= 0.316 \times 0.03 \times (497.421)^2$

$\therefore f_{omax} = 2345.614 \text{ N}$

4000 rpm to 9500 rpm
$\Rightarrow$ According to engine rpm. frequency,
$f_{omin} = 0.316 \times 0.03 \times \left(\frac{4000}{60} \times 2 \times \pi \right)^2$

$\therefore f_{omin} = 1663.3573 \text{ N}$
$f_{omax} = 0.316 \times 0.03 \times \left(\frac{9500}{60} \times 2 \times \pi \right)^2$

$f_{omax} = 9382.3749 \text{ N}$

n

$\omega = 2n\pi$

for no damping

$c=0 \therefore \xi=0 \to$ & then find that, $A\uparrow$, $mf\uparrow$, $\varepsilon\uparrow$

* Calculations for ω_{min} & hence $f\omega_{min}$:-

1) $$A = \frac{f_0/s}{\sqrt{\left[1-\left(\frac{\omega}{\omega_n}\right)^2\right]^2 + \left(\frac{2\xi\omega}{\omega_n}\right)^2}}$$

$$= \frac{416\,N/2266315397\,N/m}{\sqrt{\left[1-\left(\frac{209.4813}{941.2211}\right)^2\right]^2 + \left(\frac{2\times1\times209.4813}{941.2211}\right)^2}}$$

$\therefore A = 1.7127 \times 10^{-7}\,m$

$\boxed{\text{For } \xi=0,\; A = 1.9312\times10^{-7}\,m}$

2) $$mf = \frac{A}{f_0/s}$$

$$= \frac{1.7127\times10^{-7}}{416/2266315397} = \frac{1.7127\times10^{-7}\times2266315397}{416}$$

$\therefore mf = 0.933$

$\boxed{\text{For } \xi=0,\; mf = 1.0521159}$

3) $$\varepsilon = \frac{\sqrt{1+\left(\frac{2\xi\omega}{\omega_n}\right)^2}}{\sqrt{\left[1-\left(\frac{\omega}{\omega_n}\right)^2\right]^2 + \left(\frac{2\xi\omega}{\omega_n}\right)^2}}$$

$$= \frac{\sqrt{1+\left(\frac{2\times1\times209.4813}{941.2211}\right)^2}}{\sqrt{\left[1-\left(\frac{209.4813}{941.2211}\right)^2\right]^2 + \left(\frac{2\times1\times209.4813}{941.2211}\right)^2}}$$

$\therefore \varepsilon = 1.02125$

$\boxed{\text{For } \xi=0,\; 1.0521159}$

* Calculations for ω_{max} & hence f_{omax} :-

1) $$A = \frac{f_0/s}{\sqrt{\left[1-\left(\frac{\omega}{\omega_n}\right)^2\right]^2 + \left(\frac{2\xi\omega}{\omega_n}\right)^2}}$$

$$= \frac{2345.614/2266315397}{\sqrt{\left[1-\left(\frac{497.421}{941.2211}\right)^2\right]^2 + \left(\frac{2 \times 1 \times 497.421}{941.2211}\right)^2}}$$

$\therefore A = 8.0903 \times 10^{-7}$ m

For $\xi = 0$
$A = 1.43608 \times 10^{-6}$ m

2) $$M_f = \frac{A}{f_0/s}$$

$$= \frac{8.0903 \times 10^{-7}}{2345.614} \times 2266315397$$

$\therefore M_f = 0.7816798$

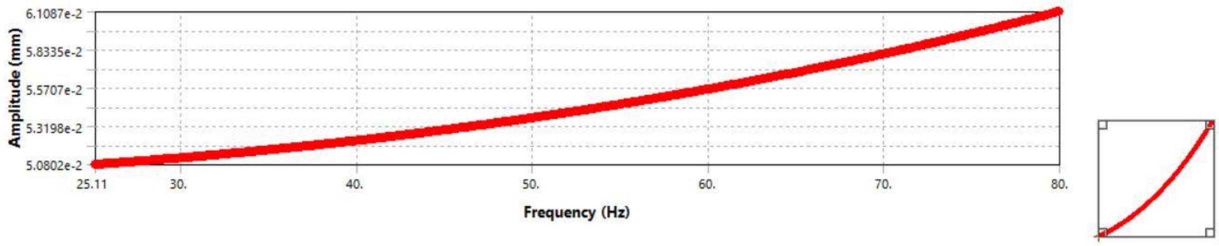
For $\xi = 0$
$M_f = 1.38823$

3) $$\varepsilon = \frac{\sqrt{1 + \left(\frac{2\xi\omega}{\omega_n}\right)^2}}{\sqrt{\left[1-\left(\frac{\omega}{\omega_n}\right)^2\right]^2 + \left(\frac{2\xi\omega}{\omega_n}\right)^2}}$$

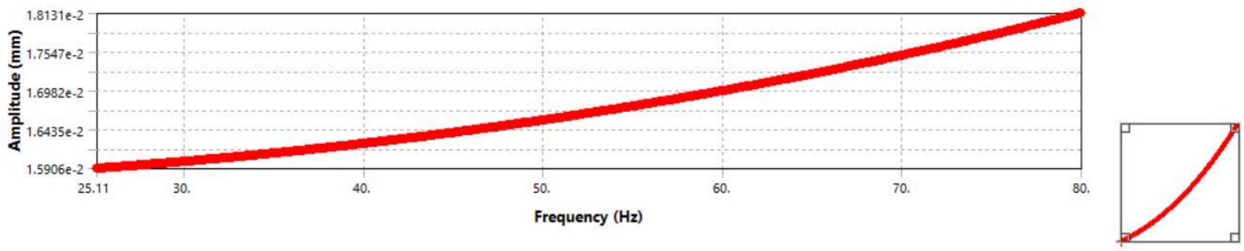
$$= \sqrt{1+\left(\frac{2\xi\omega}{\omega_n}\right)^2} \times \underbrace{\left(\frac{A}{f_0/s}\right)}_{M_f}$$

$$= 0.7816798 \times \sqrt{1 + \left(\frac{2 \times 1 \times 497.421}{941.2211}\right)^2}$$

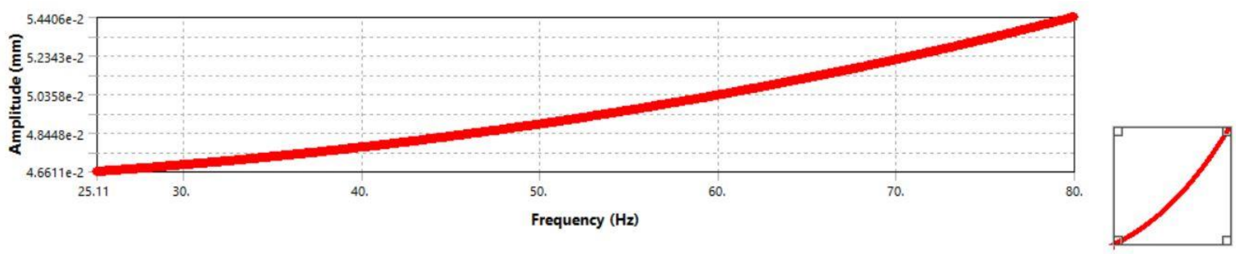
$\therefore \boxed{\varepsilon = 1.1373}$ For $\xi = 0$ $\varepsilon = 1.387532$



X Amplitude



Y Amplitude



Z Amplitude

	Frequency [Hz]	Amplitude [mm]
74	33.14	4.7158e-002
75	33.25	4.7166e-002
76	33.36	4.7175e-002
77	33.47	4.7184e-002
78	33.58	4.7192e-002
79	33.69	4.7201e-002
80	33.8	4.721e-002

X

	Frequency [Hz]	Amplitude [mm]
74	33.14	1.6066e-002
75	33.25	1.6069e-002
76	33.36	1.6071e-002
77	33.47	1.6074e-002
78	33.58	1.6076e-002
79	33.69	1.6079e-002
80	33.8	1.6082e-002

Y

	Frequency [Hz]	Amplitude [mm]
74	33.14	4.7158e-002
75	33.25	4.7166e-002
76	33.36	4.7175e-002
77	33.47	4.7184e-002
78	33.58	4.7192e-002
79	33.69	4.7201e-002
80	33.8	4.721e-002

Z

Conclusion-

a) Engine mount vibrates with very less amplitude.
b) For our working engine frequency range, i.e. 33.3334 Hz to 79.16667 Hz maximum amplitude in various directions in mm are as follow,

SR.NO.	Direction	Amplitude (mm)
1	X	6.8 e^-2
2	Y	1.81 e^-2
3	Z	5.44 e^-2

c) As per our calculation, we validated the amplitude results from ansys.

Hence, we can say that our engine mounts are safe enough to withstand against vibrations occurring due to engine

MANUFACTURING

1. Material selection

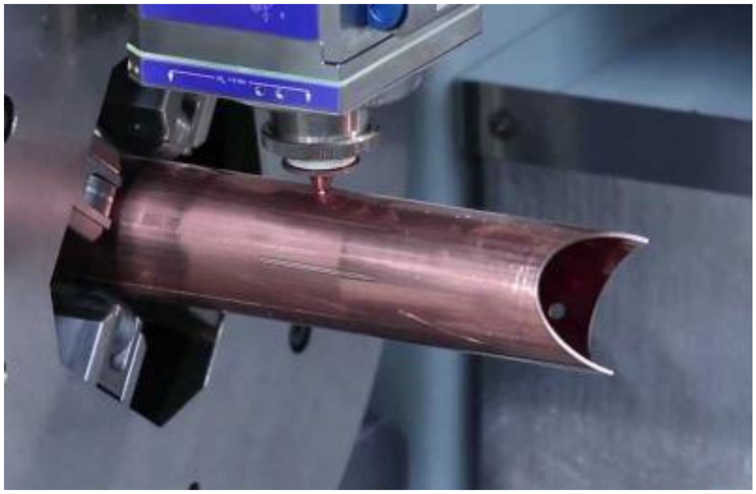
Properties	AISI 1018	AISI 4130
UTS	440 MPa	670 MPa
YS	370 MPa	435 MPa
Density	7.87 g/cc	7.85 g/cc
Carbon %	0.14-0.20	0.28-0.33

We had two options, AISI 1018 and AISI 4130, as these are the most preferred materials for roll cages. AISI 4130 has greater strength as seen in the table above. Both materials have almost 98% same composition, but 4130 has 2 extra elements namely molybdenum and chromium, which are the main reason for its greater strength. About AISI 4130, it has a carbon percentage between 0.28 and 0.33, while that of AISI 1018 is between 0.14 and 0.20 i.e. it has higher carbon percentage than 1018 due to which it has a greater hardness. But 4130 requires heat treatment post welding. If not treated post weld then the extra strength benefits come to nil. It's a waste of money if not heat treated. 4130 is very much more expensive than 1018. It has higher strength given in the table above only if heat treated "properly".
Why heat treat? Because during welding all heat put in changes atomic structure of the metal and the weld joints. The heat affected zone and /or weld itself can become really brittle and weak, especially in case of 4130.

On the other hand, 1018 has excellent weldability and good balance of toughness, strength and ductility. 1018 just serves the purpose at sufficient strength, lower cost and no post weld heat treatment process requirement.

2. Pipe cutting method

Similar to previous year this year also we used pipe laser cutting which saved time and highest level of accuracy was achieved.

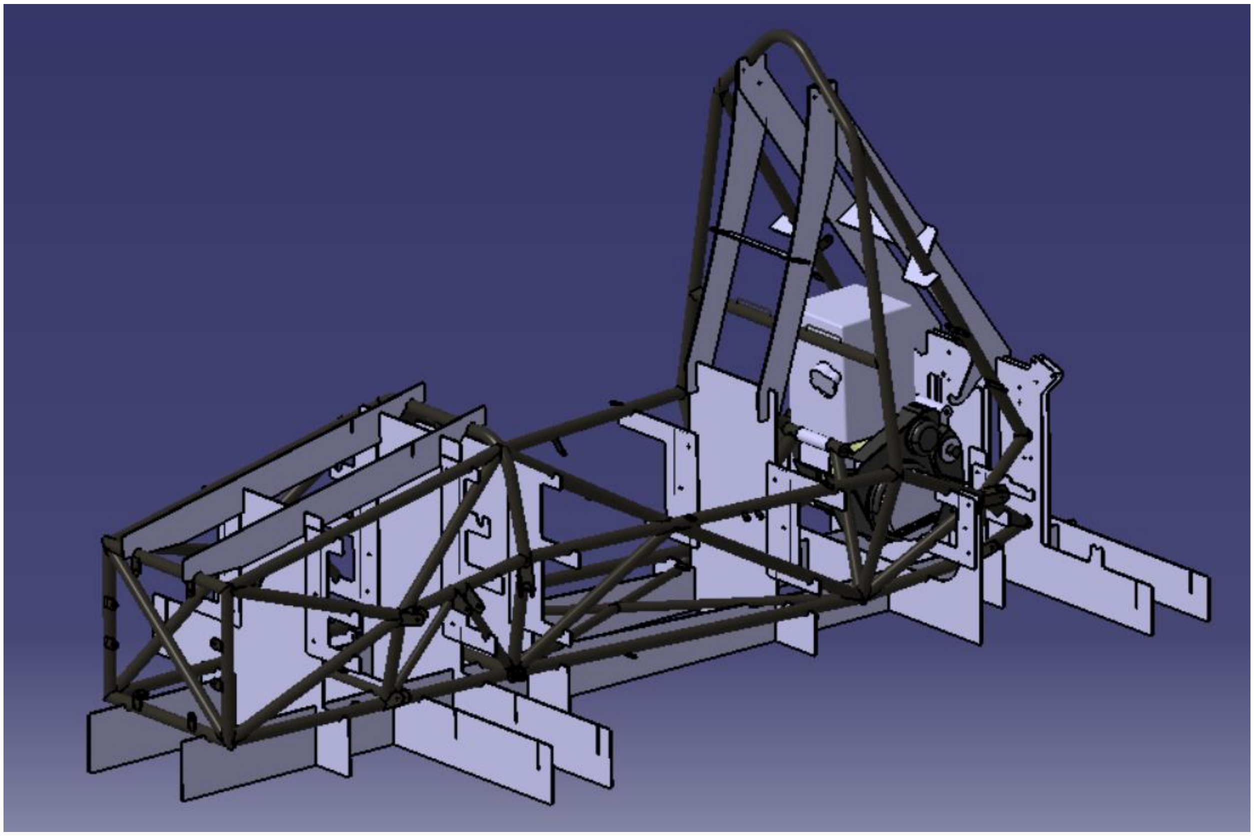


3. CNC bending-

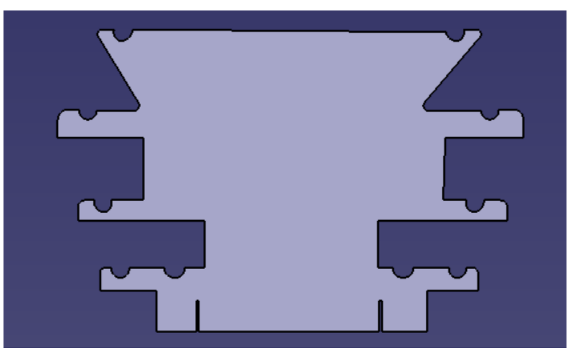
For the bends in the main and front hoop we opted for CNC bending instead of manual, which increased the accuracy and saved time.

4. Fixtures

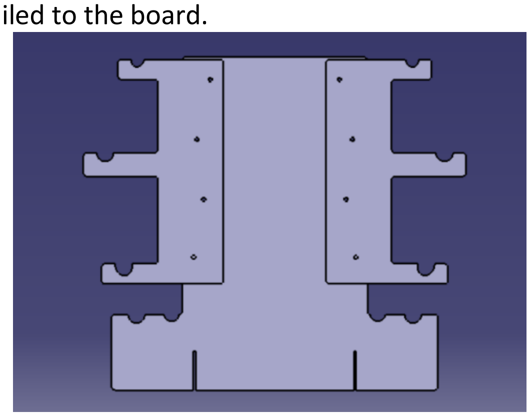
The welding fixtures used this year were different to the previous year. For the ease of manufacturing of chassis we used removable fixtures. We made use of laser cut plates, which consisted of 2 plates going lengthwise and a number of lateral plates which fit into each other like a jigsaw puzzle. Since these plates fit into each other there was no need for welding, which eliminated inaccuracies like warping of plates, etc.



Fixtures were designed such that all the plates would fit into each other to form interlocking fixtures. The design was based on the idea that all the plates would be assembled on 2 long horizontal plates with the help of slits on both the interlocking sheets, then the whole assembly was fixed on a straight board with the help of wooden blocks which were nailed to the board.



Previous year's fixture



This year's fixture

Previously the fixtures were made such that while removing it after the manufacturing of chassis we had to cut it which was a tedious job. So this year we decided to design and manufacture removable fixtures. The removable plates which had the proflie for holding the pipes were bolted to the main plate which was fixed on the table. So once the chassis is welded we had to remove only the plates which were bolted which saved our time.

- **Impact Attenuator (IA) and Anti Intrusion Plate (AIP)**

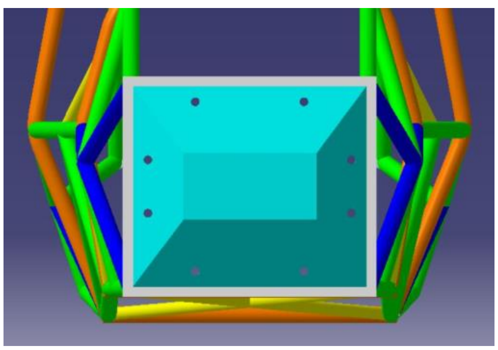
The dimensions of IA are 350mm X 300mm X 260mm. The IA used was “Standard IA design” of material IMPAXX 700 Foam.

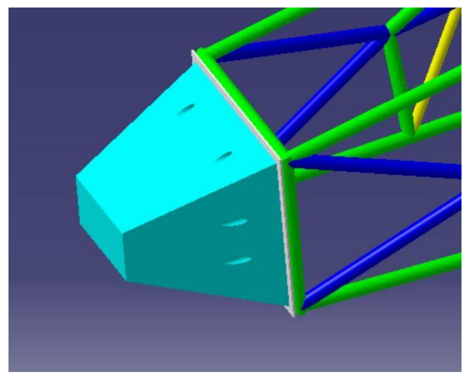
Material(s) Used	IMPAXX 700 Foam
Description of form/shape	Strong,low density, closed cell foam/ square frustum
IA to Anti-Intrusion Plate mounting method	Bonded (Araldite)
Anti-Intrusion Plate to Front Bulkhead mounting method	Bolted; 8 bolts, M8, 12.9 Grade

Length (fore/aft direction) (mm)	260	**Width (lateral direction) (mm)**	350 at base	**Height (vertical direction) (mm)**	300 at base

Bolting method:8 M8 holes through AIP

M13 holes in the IA for the bolt head





- **Head Restraint-**

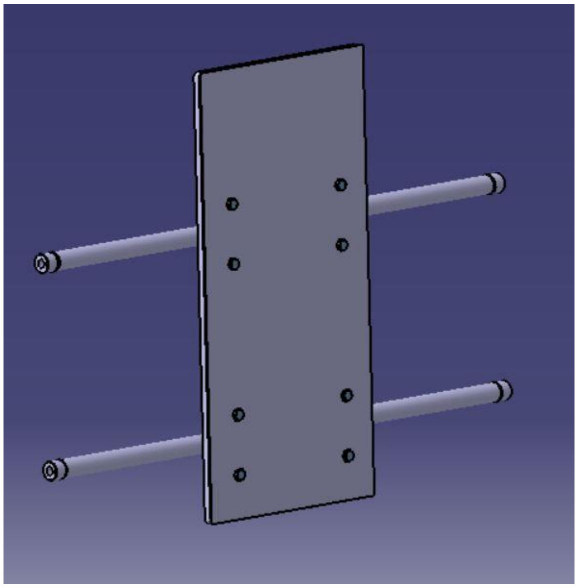
The mounting of head restraint is similar to last year as it is quite simple and light weight which uses 2 aluminium pipes across the bracing and bolted to the chassis through MS inserts.

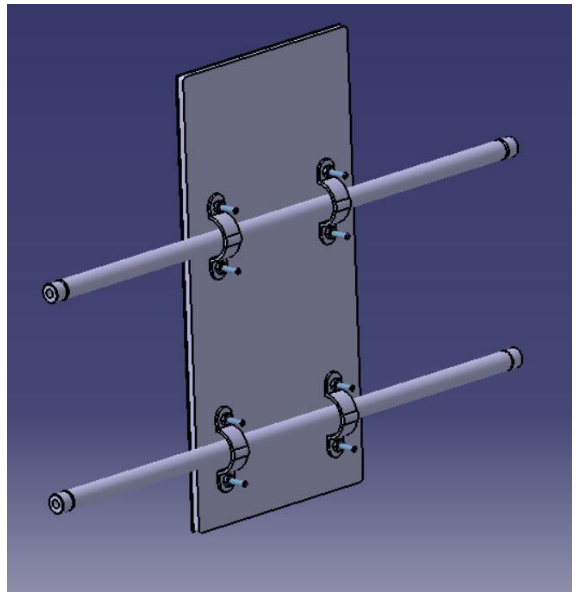
Dimensions of aluminium pipe- 13mm X 2mm

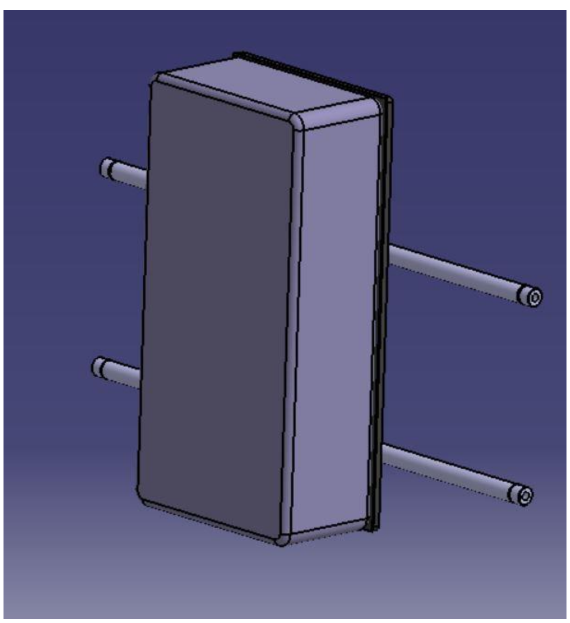
The plate on which the ethafoam is bonded is of carbon fibre and a huge weight reduction in the plate is achieved with this plate. The plate dimension is same as that of the ethafoam.

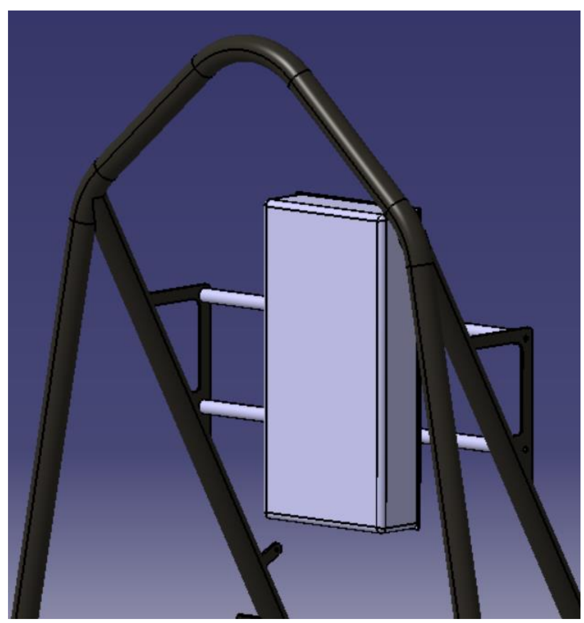
Dimensions of plate- 300mm X150mm X3mm

Dimensions of foam- 300mm X150mm X50mm

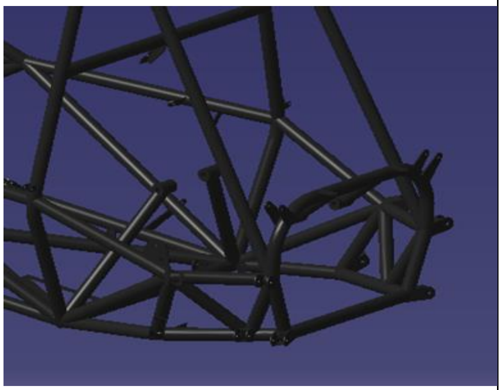


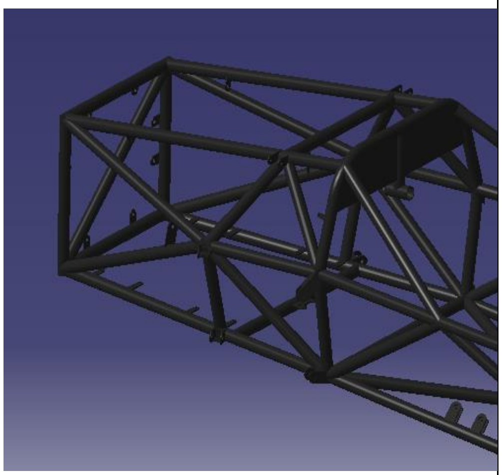
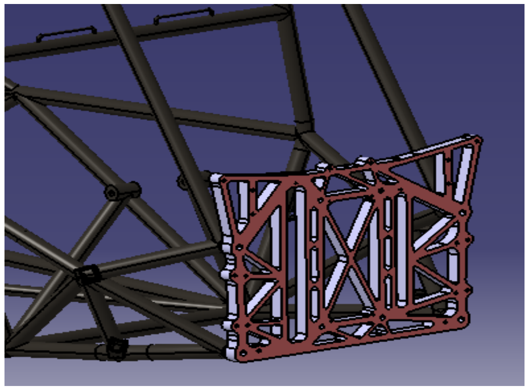


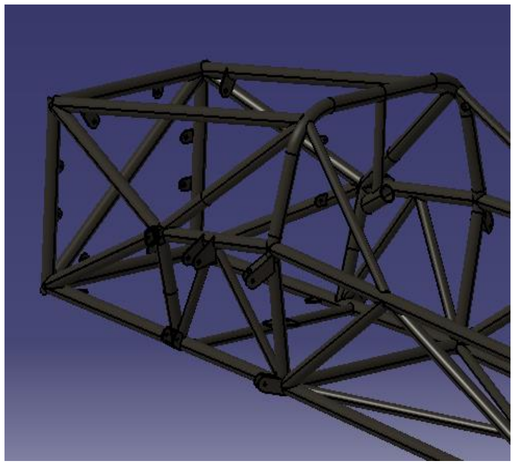


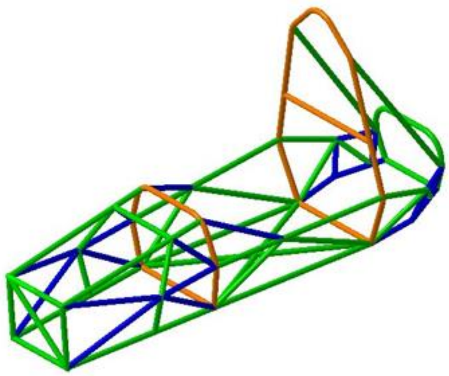
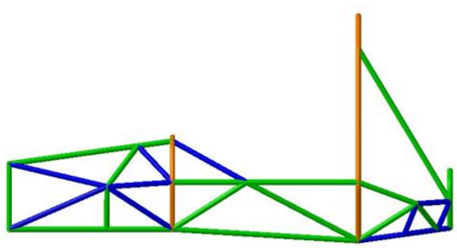
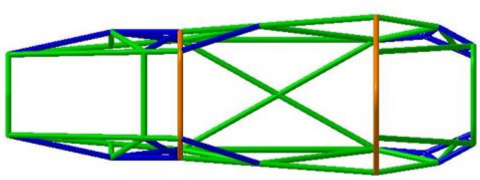
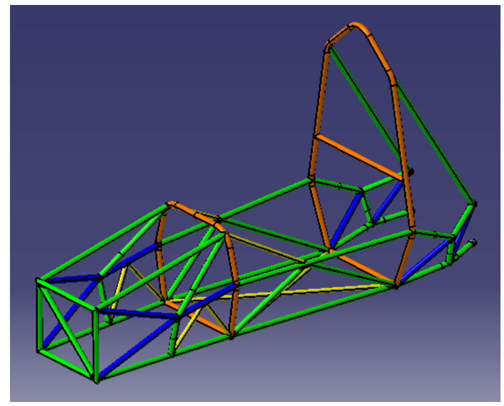
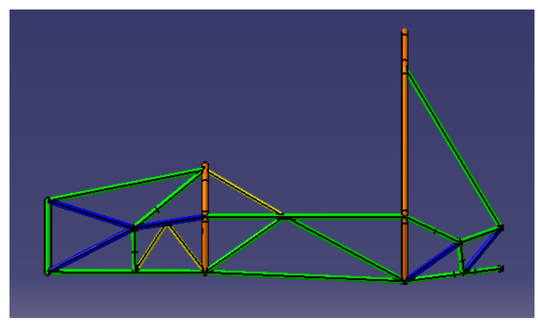
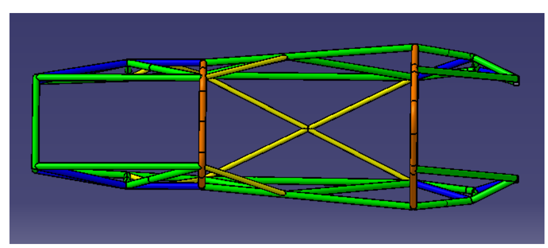


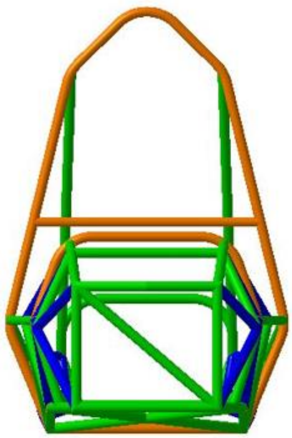
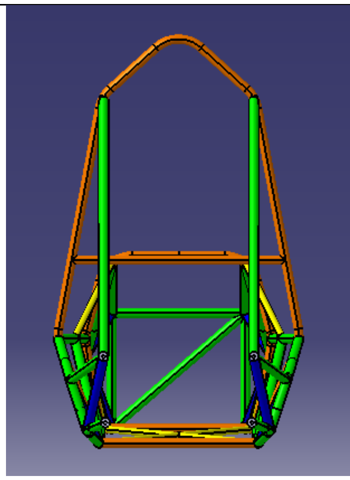
<u>Comparision with Previous Year</u>

<table>
<tr><td></td><td>FSB 19</td><td>FSB 20</td></tr>
<tr><td>WEIGHT</td><td>Rear bulkhead

The weight of last year rear hoop was 1.75kg.

Suspension

Pushrod suspension was used</td><td>Rear Bulkhead

Weight increased due to aluminum rear bulkhead which is used for accurate suspension points, this year the weight is 3.089kg.

Suspension

Direct suspension eliminated the rod on which the dampers were mounted thereby reducing the weight</td></tr>
</table>

	Thickness of Pipe  Isometric view  Side view  Top view	Thickness of Pipe  Isometric view  Side view  Top view

	 Front view	 Front view Compare to last year we use 19x2mm pipes wherever possible in order to reduce weight
CENTRE OF GRAVITY	320.755 mm	280
GROUND CLEARANCE (considerin g rigid links)	57.3 mm	67.3 mm
MAIN HOOP HEIGHT	1118.28 mm	1068.694 mm
FRONT HOOP HEIGHT	552.8 mm	457.064 mm
FRONT BULKHEAD HEIGHT	428.32 mm	304.8 mm
DIST. BETWEEN 1) FBH TO FH	759.948 mm	674.304 mm
2) FH TO MH	876.062 mm	855.696 mm
TOTAL LENGTH	2059.664 mm	1975.653 mm
WHEELBAS E	1539 mm	1529 mm
FRONT TRACK WIDTH	1219.2 mm	1150 mm

REAR TRACK WIDTH	1211.8 mm	1120 mm
DRIVER SITTING	Reclined	Reclined
PIPE CUTTING METHOD	Leaser cutting was used for accurate and fast cutting of pipe	Leaser cutting