BAJA SAE, IITK MOTORSPORTS
INDIAN INSTITUTE OF TECHNOLOGY, KANPUR

Chassis
2015-2016

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1. ABSTRACT

SAE (Society of Automotive Engineers) International organizes a number of automotive events round the year. BAJA (pronounced as ba-ha) International is one of these events. Organized primarily in seven countries, *BAJA SAE* is a competition where groups of engineering students design and build an All-Terrain Vehicle. *BAJA student India* is an Indian version of this competition. This event is not associated with *SAE International* and is organized by *Delta Inc.* BAJA Student India' 2016 is going to be held from 23rd January to 26th January 2016.

IITK Motorsports is a hobby group started back in 2009, by group of automotive enthusiasts who came together to study different aspects of a cars and automobiles. Over time, this group has grown out and now we have about 150 members in our club. We are divided in four teams. One for marketing and three technical teams, each focused on a different competition. The BAJA team, which consists around 30 members is currently in its second season.

In the first season, the older team built a vehicle for competing in BAJA Student India' 2015. The event took place in January' 2015 in Jamshedpur. In the competition, we stood at fourth position in Design and acceleration event. We were also awarded the best Rookie team award. Overall, after a penalty of 200 points due to late submission of running car video, we stood at 24th position in the event. The first season left the team with a lot experience and knowledge.

2. INTRODUCTION AND DESIGN OBJECTIVE

Chassis of any vehicle is the basic structural element over which all the other components of the vehicle are attached. In our case, chassis as defined by rulebook, must be a space frame made by tubular steel. The main purpose of the roll cage is to maintain a minimum space surrounding the driver. It is also responsible for maintaining the integrity of the vehicle.

The Design Objective of this season's car is to be lightweight, ergonomically comfortable, aesthetically pleasing, and endurable. The Roll cage of the car plays an important role in weight, ergonomics, aesthetics and strength of vehicle. Hence these factors have been kept in mind while designing the roll cage.

Also, the geometry of the roll cage and that of suspension are largely dependent. As this vehicle is expected to be running on the roughest patch of land, constraining the geometry of suspension wouldn't be a good choice. The roll cage has been built keeping the suspension geometry and the rule book in mind.

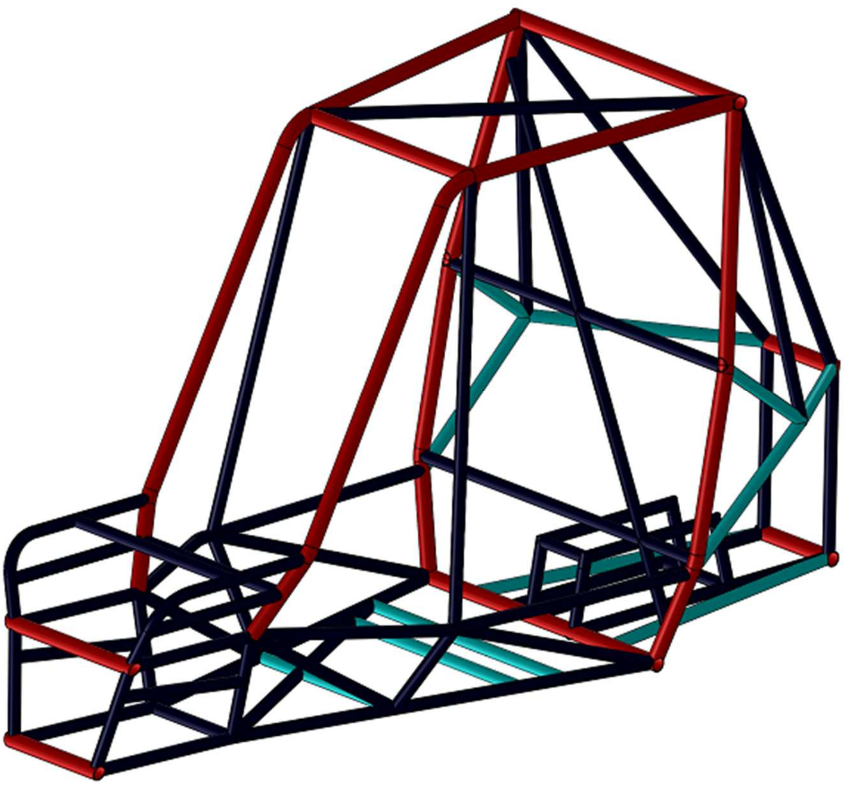


Figure 1. CAD Model of B15's Roll cage

3. DISCUSSIONS

3.1 Inferences from Last season's car

As evident from the picture of last season's CAD, the roll cage was overdesigned. It weighed approximately 31 Kgs in the CAD itself. Added the weight of tabs and filler material, it ended up being above 35 Kgs.

The basic approach in designing it was to make it as strong and rigid as possible. The Forces applied during simulation were far greater than expected in the field. A lot of unnecessary bracing was added. Although, the structure was strong, given the small leg space, it was highly uncomfortable to be seated in. Added the position of gear shifter, it was a nightmare for our driver to sit in that position for 4 hours.

So, the basic inferences were:

- To not to overdesign the roll cage.
- To use tubes efficiently.
- To keep ergonomics in mind while designing.

Also, the choice between fore/aft bracing was criticized. The rear structure of our roll cage with slight modification would have completed the aft bracing, but still the front bracing was used, which resulted in significant increase in weight. We decided that a discussion should be done about it.

3.2 Selection of Material

A large number of grades of steel are available in the market. Upon studying a lot of design reports of other teams, it was concluded that AISI 1018 and AISI 4130 are the general choice of majority of teams. The general reason behind these choices were their availability, tensile strength, affordable cost and availability of variety of dimensions.

So, we narrowed down our choices to these. Further, the choice was made upon analyzing a lot of factors.

Factors		AISI 1018	AISI 4130
Availability		Easily Available	Less Available
Density	(in g/cc)	7.87	7.85
Young's Modulus	(in GPa)	205	205
Yield Strength	(in MPa)	370	435
Ultimate Tensile Strength	(in MPa)	440	670
Carbon Content	(in %age)	0.18	0.30
Cost(31.75 OD*1.6mm*3 m)	(in INR)	825	2400
Fatigue strength	(relatively)	High	Low

Table 1. Comparison between materials

The major differences as observed in these two were that of yield strength, ultimate tensile strength, carbon content and the cost. The density of these two are almost equal.

Carbon content doesn't affects the strength directly. It is yield strength and the ultimate tensile strength that matter the most. Given the two fold increase in price for using AISI 4130, but a relatively minor difference between the yield strength/ultimate tensile strength, AISI 4130 seemed like a very costly option. Although AISI 4130 is stronger than AISI 1018, but using AISI 4130 will cost us around INR 45000 more than AISI 1018. Also, since our team is in its initial seasons, we felt like we must improve our design before using such costly material.

Hence, AISI 1018 was selected for this season!

3.3 Dimensions of Cross Section of Primary and Secondary Members

The rule book says a lot about this section. The roll cage members are basically divided in two sections.

- Primary Members

This include: Rear Roll Hoop (RRH), Roll Hoop Overhead members (RHO), Front Bracing Members (FBM), Lateral Cross Members (LC), Front Lateral Cross (FLC) and Lower Frame Side members (LFS).

- Secondary Members

This include: Lateral Diagonal Bracing (LBD), Side Impact Member (SIM), Fore/Aft Bracing (FAB), Under Seat Member (USM), All Other Required Cross Members, Any tube that is used to mount the safety belts.

All these members have been defined separately in section B.8 of the rule book.

- **Primary Members**

Regulations set up by Rule Book *(B8.3.12)*:
The material used for the Primary Roll Cage Members must be:

A. Circular steel tubing with an outside diameter of 25mm (1 in) and a wall thickness of 3 mm (0.120 in) and a carbon content of at least 0.18%.

OR

B. A steel shape with bending stiffness and bending strength exceeding that of circular steel tubing with an outside diameter of 25mm (1 in.) and a wall thickness of 3 mm (0.120 in.) and a carbon content of 0.18%. The wall thickness must be at least 1.57 mm (0.062 in.), regardless of material or section size. Documentation of the equivalency must include:

1. Calculations must be presented at Technical Inspection which proves sufficient bending stiffness and bending strength. All calculations must be in SI units, to three significant figures to the nominal tube sizes as specified by the invoice.
2. Invoices of the roll cage materials.
3. Material tests or certifications, which specify the carbon content and yield strength.

C. The bending stiffness and bending strength must be calculated about a neutral axis that gives the minimum values. Bending stiffness is considered to be proportional to the product *EI* where:

E Modulus of elasticity (205 GPa for all steels)

I Second moment of area for the structural cross section

$$I = \frac{\pi({r_o}^4 - {r_i}^4)}{2}$$

r_o Outer Radius of the tube

r_i Inner Radius of the tube

Bending strength is given by: $\frac{S_y I}{c}$

Where:

S_y Yield strength (365 MPa for 1018 steel)

c Distance from neutral axis to extreme fiber

For steel tubing with an outside diameter of 25mm (1 in) and a wall thickness of 3 mm (0.120 in) and a carbon content of at least 0.18%, Bending Stiffness comes out to be 5526.2 Nm^2 while Bending Strength is 774.8 Nm.

The Major factors which further effect the choice of dimension are:

- Bending Strength

- Bending Stiffness
- Weight per unit length

Given that the young's modulus and Yield strength do not change, ensuring that I and the ratio I/c for the chosen dimension is greater than that for outer diameter of 25.4mm and thickness 3mm, and by minimizing the weight per unit length would do our job!

Weight per unit length, W is directly proportional to area of cross section. So minimizing the cross section area would be the same. Plot of Cross sectional area vs Thickness & outer diameter and plot of bending strength vs cross sectional area are as follows.

Outer Diameter which are integral multiple of 1/8th of an inch are available in the market. So, they were taken into account. Also, thickness was constrained from 1.57 mm to 3 mm.

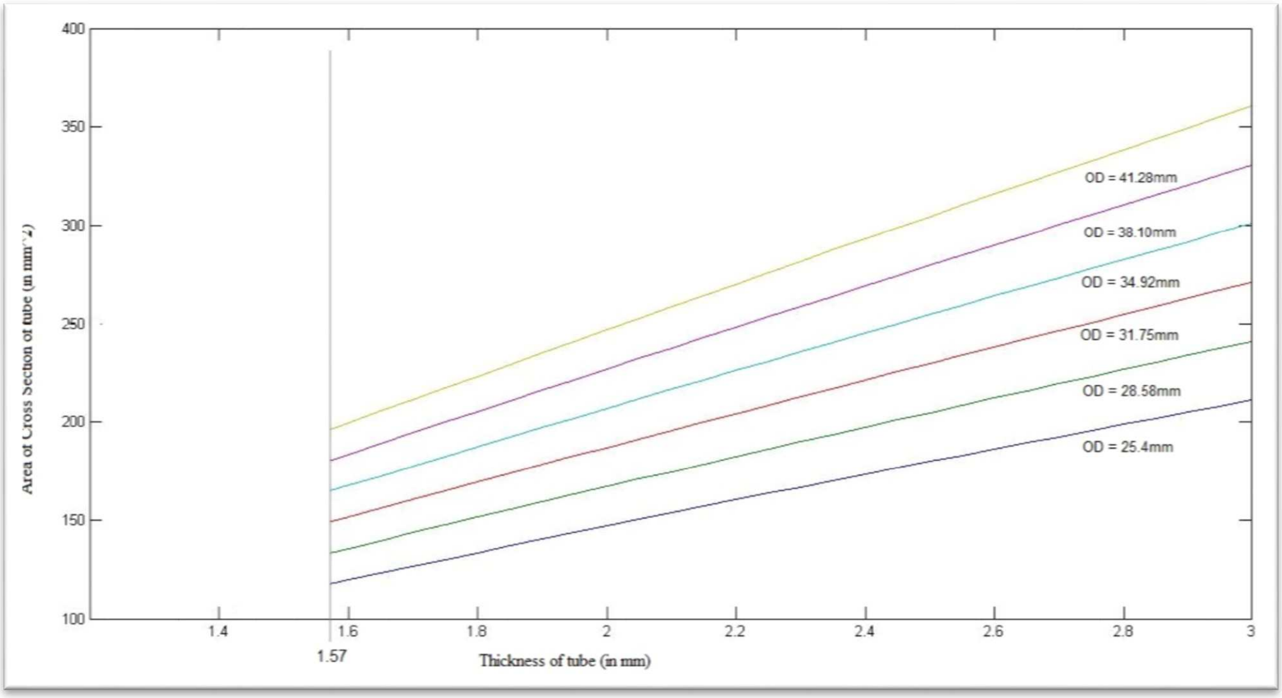


Figure 2. Variation of Cross Sectional area vs. Thickness and Outer Diameter

The Variation of cross sectional area vs thickness of tube is almost linear. However this does not gives us any results.

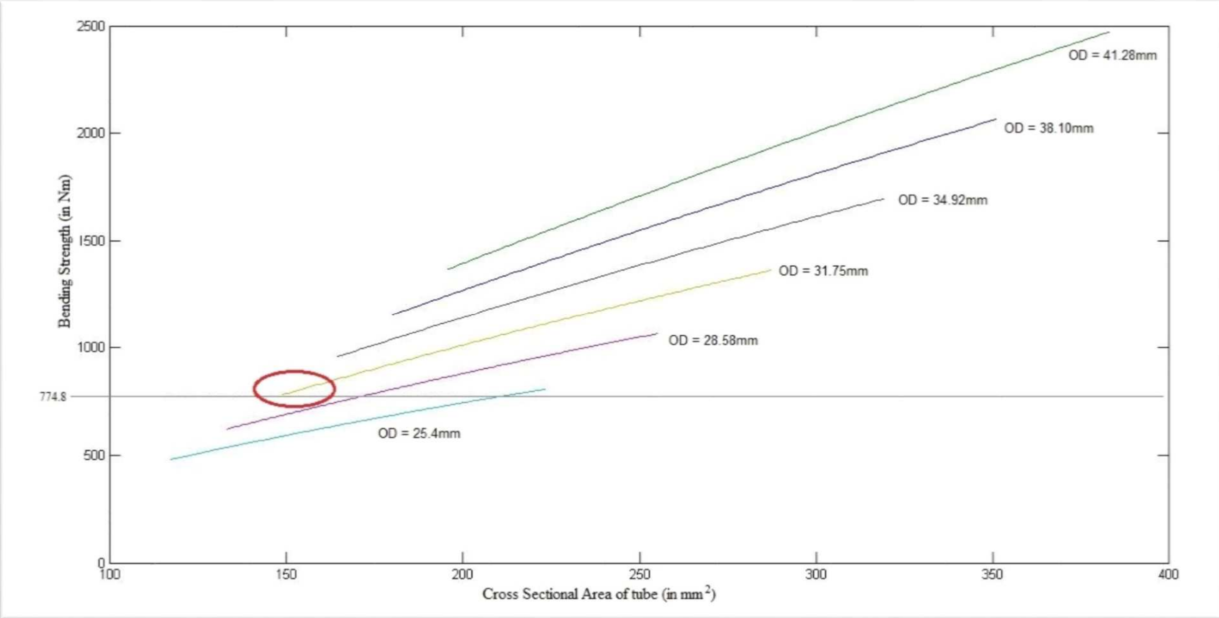


Figure 3. Variation of Bending Strength vs Cross sectional area for different ODs

As seen from the graph, the plot marked with red circle has the required bending strength as well as minimum cross sectional area. Hence, for primary tubing, Outer diameter of 31.75mm (1.6 inches) and a thickness of 1.6mm was selected.

The bending stiffness of our tube is 7080.2 Nm^2 and bending strength is 794.1 Nm with a cross sectional area of 151.5 mm^2. This calculation has been shown in section 7.1.

- Secondary Members.

Regulations set up by Rule book *(B8.3.1):*

Secondary members must be steel tubes having a minimum wall thickness of 0.89 mm (.035 in) and a minimum outside diameter of 25.4 mm (1.0 in).

To minimize cross sectional area, it was suggested that we start with minimum dimensions and increase them if required.

3.4 Tube Bending Vs. Welding

At many joints in the roll cage, we had an option between using a single bent tube and two tubes welded on each other. Pros and con of each of the method was listed out, which are as follows:

i. Tube Bending

Pros:

- Less length of tubing required
- The stress get distributed quite effectively

Cons:

- Possibility of Buckling
- Facility not available in the campus
- CNC Bending not available

ii. Tube Welding

Pros:

- Stronger, since a gusset has to be used
- Facility available in campus
- Higher Accuracy

Cons:

- Heavier, added weight of gusset, weld material and tubing
- Concentration of stress in an area

We decided over using tube bending wherever possible. Also, gusseting a bend was also kept as an option

3.5 Choosing between Fore/Aft Bracing

Fore/Aft Bracing, each of these has their own pros and cons. fore bracing where adds a lot of strength to our roll cage, aft bracing is light. Even if we fore braced our roll cage, the rear structure, whatever configuration we could think of, with a slight modification always would result in a rear braced structure. Having both fore and aft bracing would result in a heavier model. Also, it would be a step towards over-designing the vehicle. Aft Bracing was the common consent of the team for this season’s vehicle.

3.6 Bend Radius Selection

Not much choice was available in this regard. As rule B8.3.1 states:

Roll cage members which are not straight must not extend longer than 711 mm (28 in.) between supports. Small bend radii (<152 mm, 6 in.) at a supported end of a member are expected, and are not considered to make a member not straight. The minor angle between the two ends of a not-straight tube must not exceed 30°.

The available dies (bending profiles) were of 2 in., 3 in. and 6 in. bend radius and 1 in. and 1.25 in. tube diameter. So, we were pretty constrained in choosing the bend radius. However the availability of bending profile for a tube with diameter of 1.25 in (31.75mm) turned out to be in our favor.

All the bends in our current CAD model are with a 3 in. bend radii.

3.7 3-Dimensional Bending

Another Constrain which we had to face was non-availability of CNC bending. CNC bending helps us to make multiple bends in a single tube all of which could be in a single or different planes. But we had to use a mechanical bender. Mechanical bender is quite inaccurate. Also, it doesn't has any facility of multiple bending in different planes. So, we had to limit our number of bends in each tube to one. However, if multiple bends in a tube lie in a single plane, it can be manufactured. In such cases, multiple bends were possible.

4. Roll Cage Structure: The Design Phase

The designing process has been described as follows:

4.1 Rule Book and Safety Regulations

The main feature of chassis is to protect the driver. Rule book lays a lot of regulations to ensure the safety of the driver. The rule book of BSI' 16 can be found at this link. The sections B4, B8 and B10 compromise of the rules related to roll cage.

All the rules have to be kept in mind while designing each part of roll cage.

4.2 The First CAD

Our first CAD looked like following.

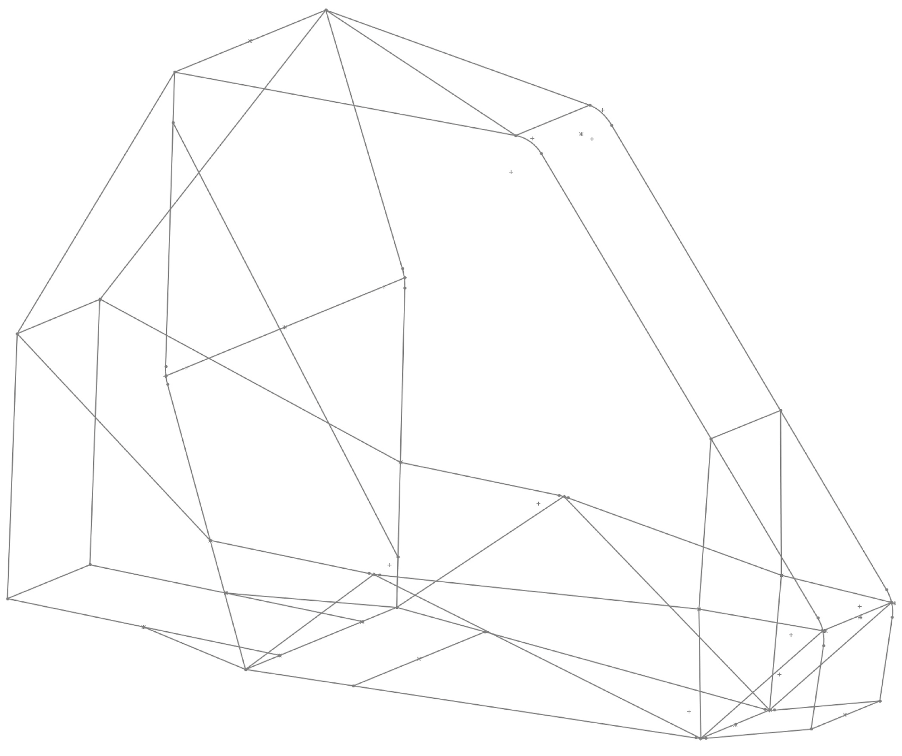


Figure 4.Our Initial CAD

It is basically an image of what we wanted our roll cage to look like. Ensuring that it satisfied all the rules, we made it keeping the aesthetics and ergonomics in mind. Some members were later added to increase its strength.

4.3 Moving Further

Adding the structural members, and making some aesthetical changes…

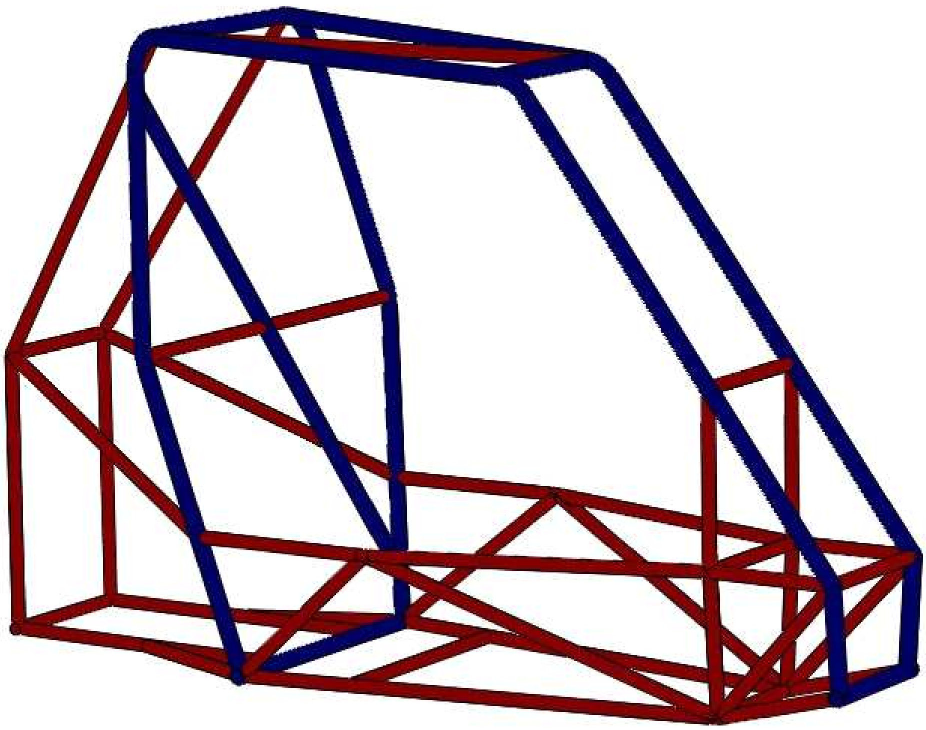


Figure 5-A. First Chassis with Structural members

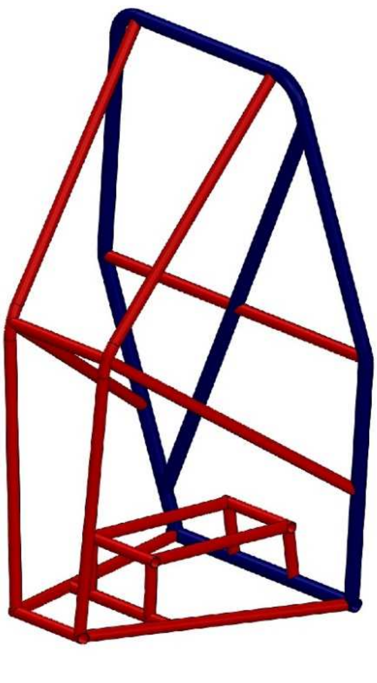


Figure 6. Plan for Aft Bracing

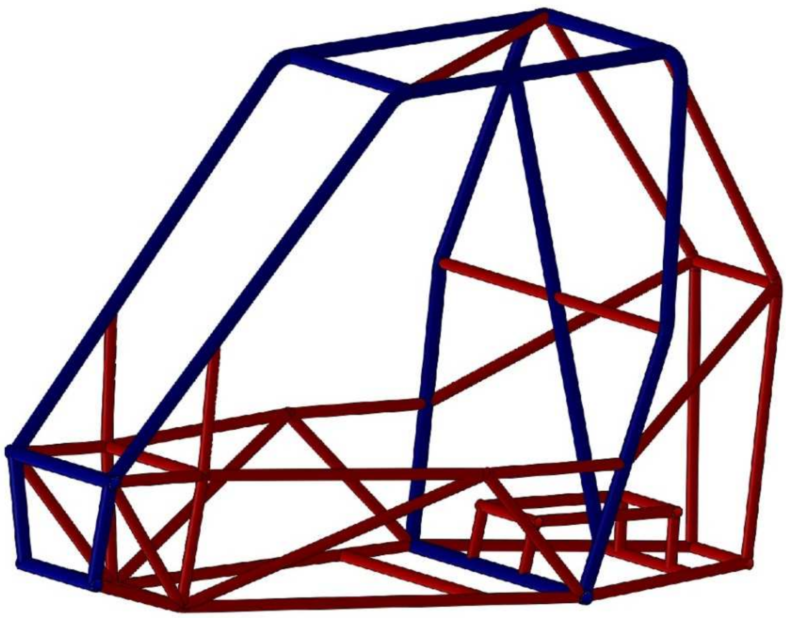


Figure 7. Cad of Chassis6 A

Then a PVC model was made so that we can verify the rules and also check the compatibility of driver with the car and make adjustments in the chassis if required.

Figure 8. Our first PVC Model

After making this, certain changes were made in accordance with other sub-assemblies, rule book and driver comfort:

- The vertical height of RRH was reduced by 1 inch.
- The length of lower lateral cross in the RRH plane was increased to 25 inches from 22 inches.
- Dimensions of front box were changed according to brake sub-assembly.
- Height of SIM members w.r.t. that of LFS members was increased.

4.4 Simulation

Now, the roll cage had to be tested for its structural rigidity. We used Ansys 16.0 for the simulation.

4.4.1 Discussion: Static Structural & Explicit Dynamics

Ansys provides a platform for wide variety of simulations. The most relevant to our simulations were:

i. Static Structural

In this case, a part of body is kept rigid. This part doesn't moves or undergoes any stress. A force is applied at another part of the same body.

ii. Explicit Dynamics

In this case, a moving body is collided with a flexible wall whose other face is kept rigid.

The forces experienced by our vehicle/roll cage would generally be similar to that as in explicit dynamics. But, since explicit dynamics is a very complex case, Ansys takes a lot of time to solve this on our laptops. So, we were forced to carry out most of our simulations in static structural scenario.

4.4.2 Calculations

Since, simulations were being carried out at static structural system, we had to convert our dynamic model to a static model. Some of the assumptions made for these calculations were:

- The vehicle moves at 10 m/sec before collision/impact and comes to complete stop after it.
- The time of impact is 0.25 sec
- The velocity decreases at a constant rate. (Constant deceleration)
- The mass of vehicle & driver is 250 Kgs. (projected weight of vehicle: 180 Kgs)

Please refer to section 7.2 for the calculations.

4.4.3 Front Impact

The roll cage was exposed to a force of 4G on front members (along positive x-axis), while keeping the mounting points of rear suspension fixed. There result was soothing. The maximum von-Mises stress developed in the roll cage was 1.8E08 N/m with safety factor being 2.02.

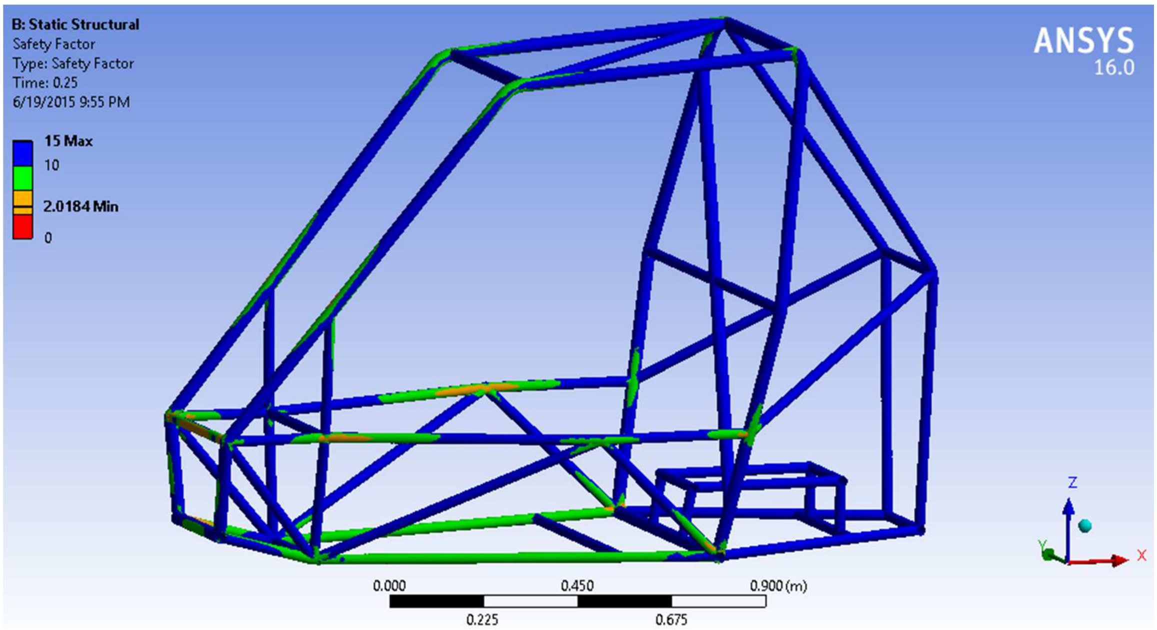


Figure 9. Front Impact Simulation

4.4.4 Rear Impact

Rear impact was carried out at similar terms as that of front impact. An impact of 4G on rear tube structure (along negative x-axis), while keeping the front suspension points fixed. It failed miserably in the first try. The Factor of safety turned out to be 0.41.

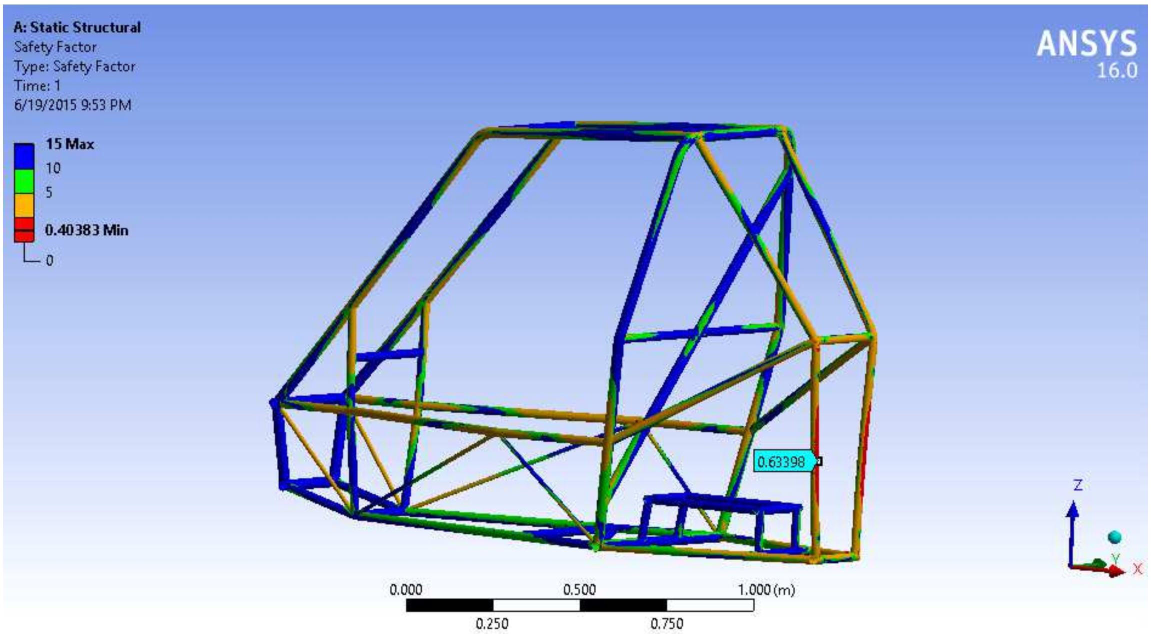


Figure 10. Rear impact simulation - I

Several structural changes were made. The height of third vertex of structural triangle was decreased. A bracing was added to support the rear vertical tube.

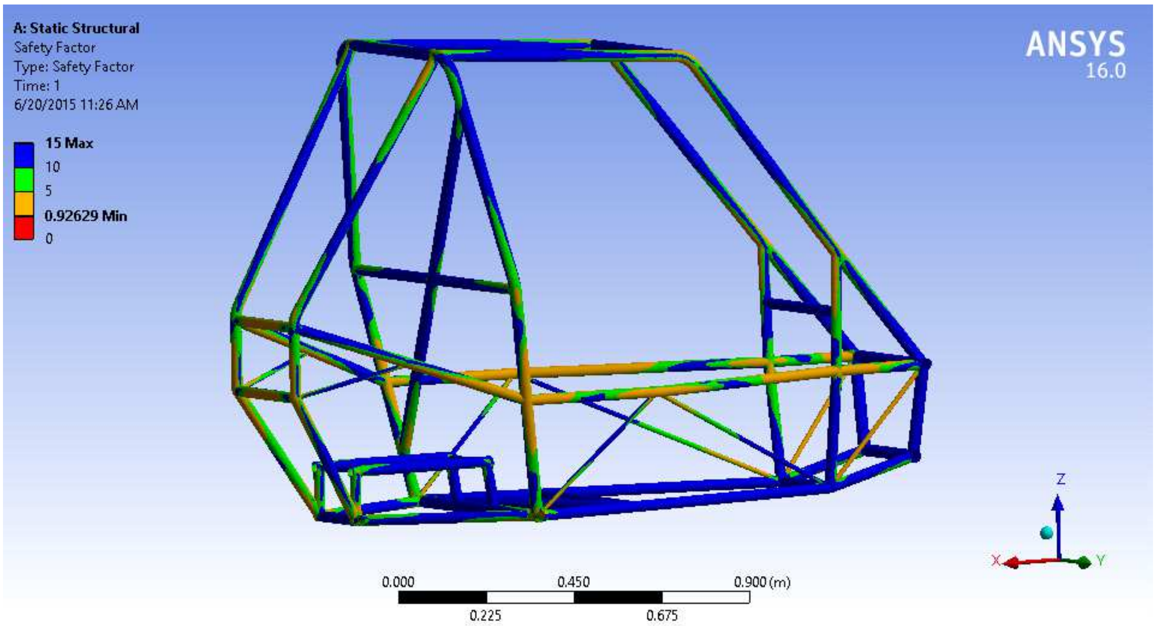


Figure 11. Rear Impact Simulation - II

The maximum Von-Mises stress turned out to be 3.53E8 N/m and the factor of safety for this design was 0.93. But this design was discarded. There is no space for gear box in this geometry. Also the rear suspension design didn't match with it. The final design however could tolerate an impact of 4G.

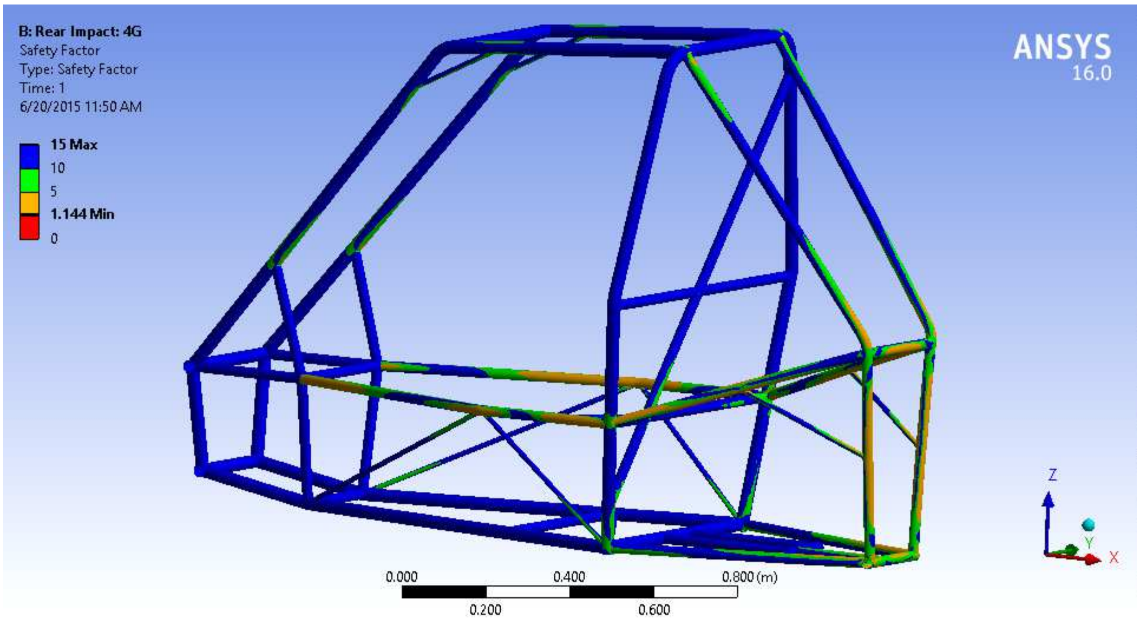


Figure 12. Rear Impact Analysis – III

The maximum von-Mises stress in this structure is 3.23E8 N/m and the factor of safety of this frame (for this impact) is 1.14.

4.4.5 Side Impact

For this simulation, an impact of 2G was applied on one side of roll cage (along y axis) keeping the suspension points of other side fixed. At first, the result was no better than that of rear. The maximum von-Mises stress was and FOS turned out to be 0.70.

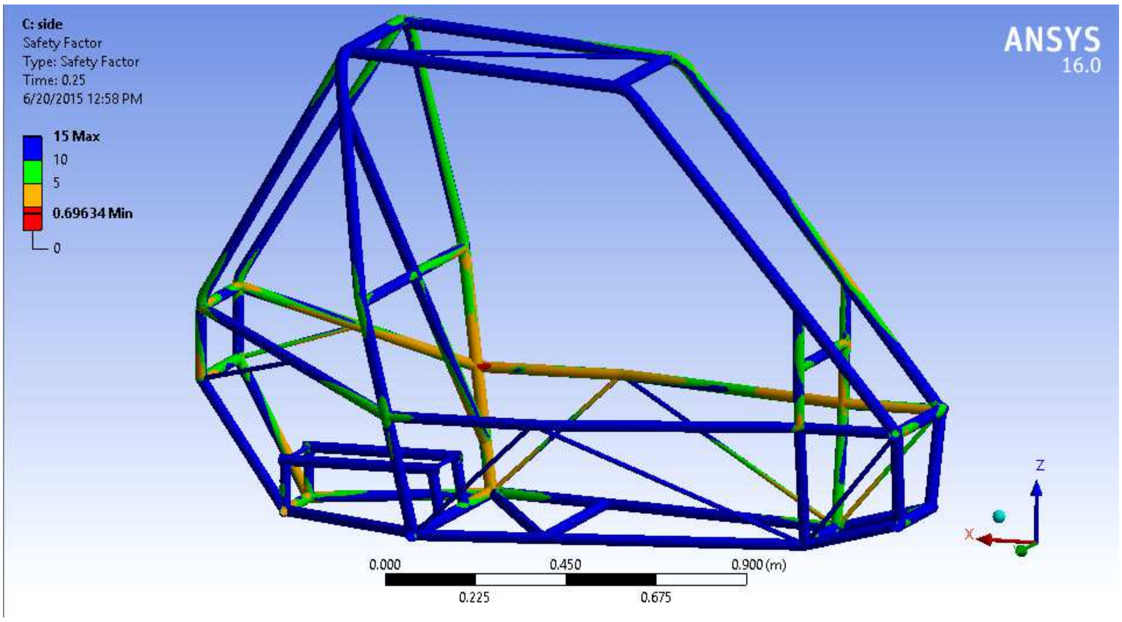


Figure 13. Side Impact Simulation - I

But after adding a LC and some bracing members and modifying some lengths, the following results were obtained.

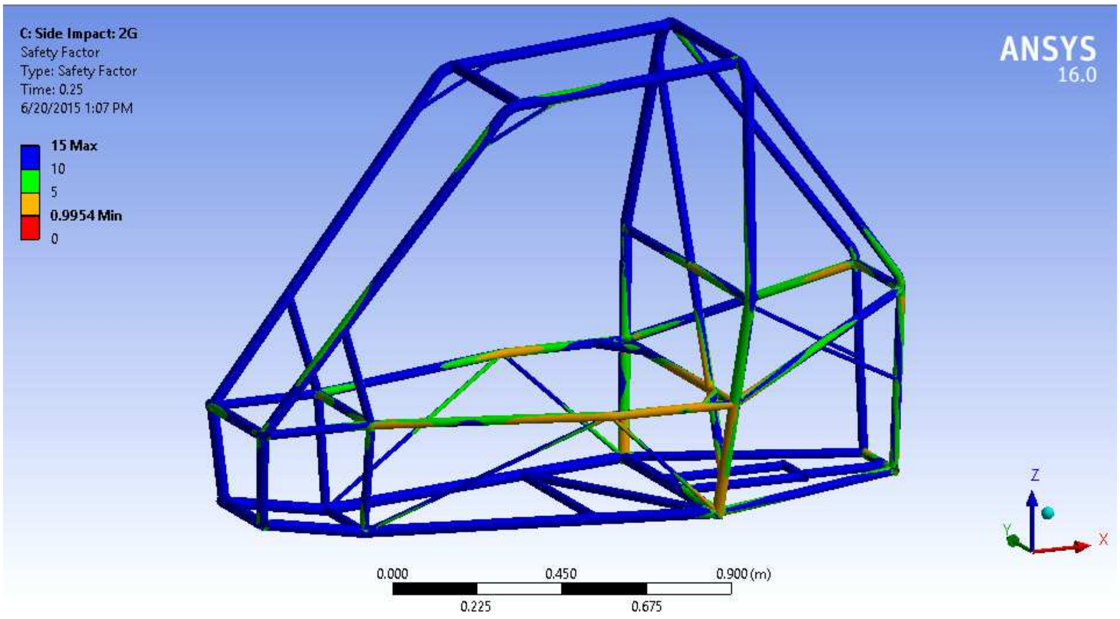


Figure 14. Side Impact Simulation – II

The Maximum von-Mises stress which developed in final model for side impact simulation 3.71E8. The Factor of safety is 0.99.

4.4.6 Roll over (Pitch)

As observed, through the course of various events in previous competitions, a lot of vehicles go through a roll over. I.e. a complete 360 degree turn along y axis of vehicle. Such incidents are to be avoided. We went through a lot of videos of such roll overs. Some common conclusions were

- Such events happen generally during continuous bumps and droops or while a jump. This point has been elaborated in section 4.5.
- The front-most part of roll cage touches the ground and it has to bear the major impact. So, the general assumption that the intersection of RRH tubes and RHO tubes being the point of impact is wrong! So, we could actually treat this similar to front impact.

But for the sake of safety of driver and to ensure the strength of vehicle, we tested our roll cage for such a condition where the point of intersection of RRH and RHO members has to go through a large instantaneous force. A simulation with rear suspension geometry kept fixed and a force of 3G acting on the intersection points was solved. Following are the results:

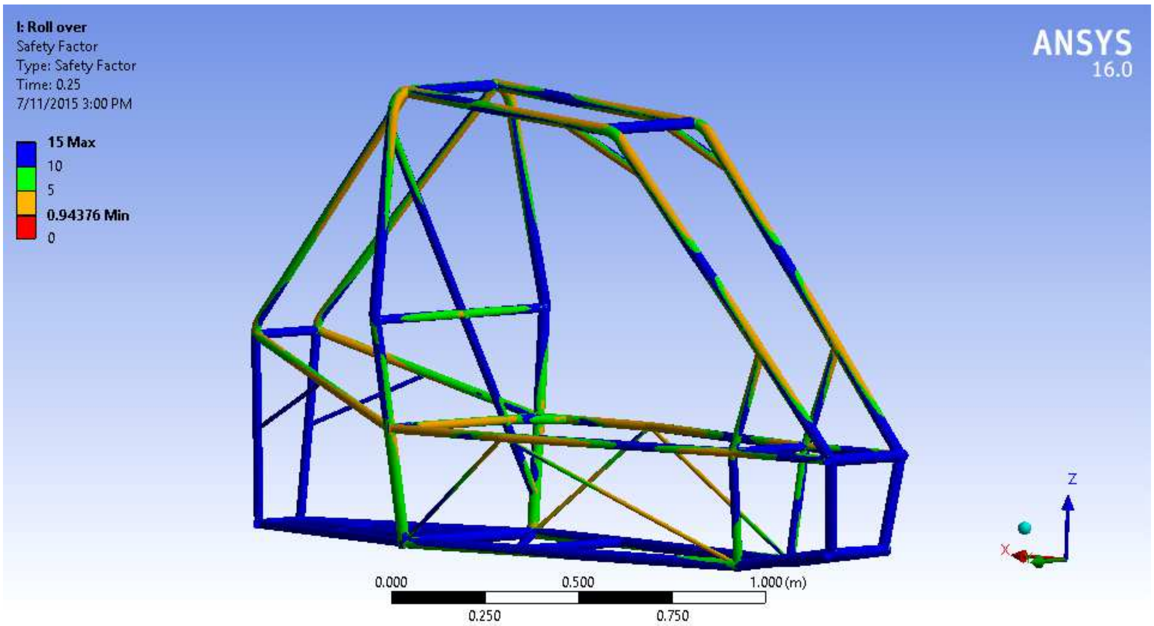


Figure 15. Roll over simulation

The frame could handle this impact to a great extent. The factor of safety turned out to be 0.94, which looked quite safe, given this was only a precautionary simulation.

4.4.7 Forces through Spring & Dampers

Key forces on frame come from damper-spring joint points. So, some mind has been indulged into maximum forces that suspension can transmit to the frame, damping provided by suspension, oscillation frequency of frame.

To simplify the calculations, effect of layback of lower A-arm is neglected as it produces only longitudinal stress in tubes. Spring and Damper Force curves provided by FOX® (suspension FOX Float Evol 3 R) were best interpolated to cubic equations as shown below:

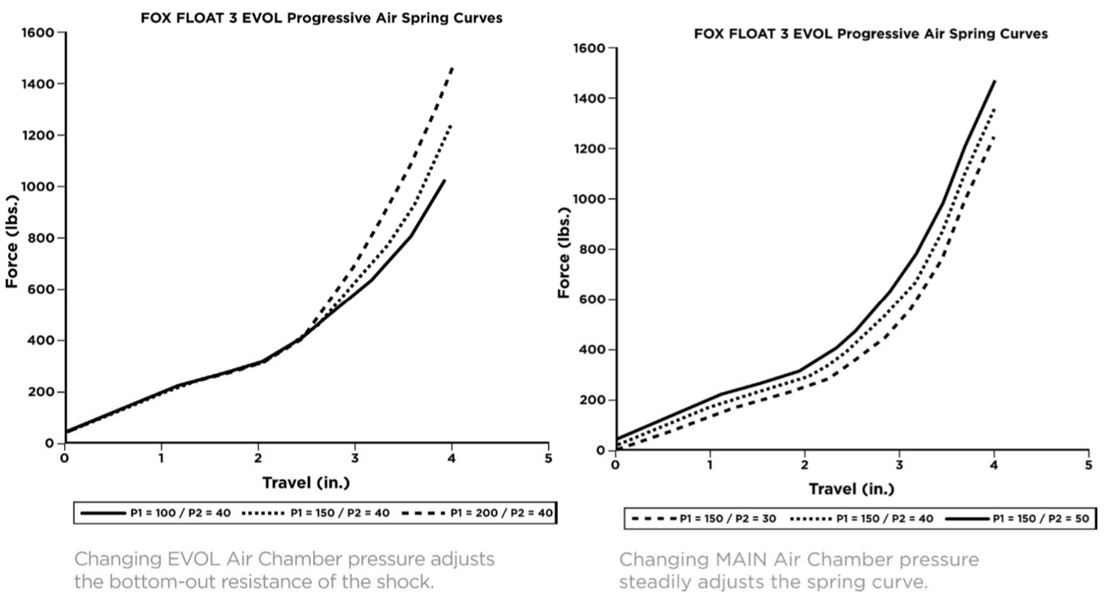


Figure 16. Variation of force vs travel of damper curve with varying Pressure in Main and Evol chambers

$$F_{spring} = -4 + 277x - 133x^2 + 37x^3;_{for\ P_1=150\ and\ P_2=30}$$

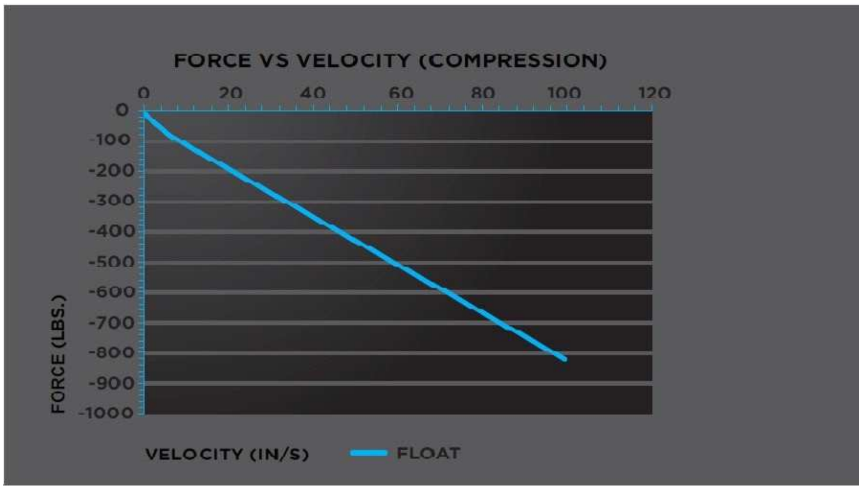


Figure 17. Variation of damping force vs compression velocity of shocks

$$F_{Damping} = 9.5 - 6.4\dot{x} + 0.057\dot{x}^2 - 0.00029\dot{x}^3;$$

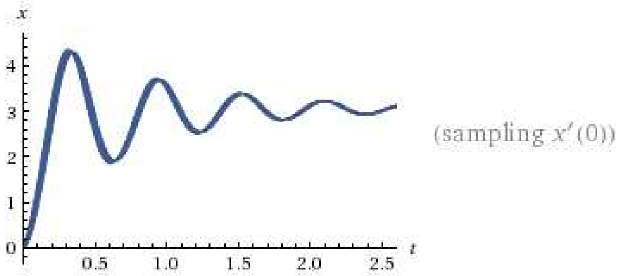
Applying our all-time friend geometry and mathematical manipulations on suspension frame interface reduces whole motion to the following equation:

$$\frac{\left(35 - 2464x + 1183x^2 - 329x^3 + 84 - 57x' + 0.5x'^2 - 0.0026x'^3\right)\left((17.4 - x)^2 - 123\right)}{13.6(17.4 - x)}$$
$$= 0.7x'^2 + 0.7x''(17.4 - x) - 2450$$

Which is a second-order non-linear ordinary differential equation in terms of *x* i.e. compression of spring and *t* i.e. time. But this equation is reduced to first-order non-linear differential equation in terms of compression *x* and rate of compression *y*.

$$\frac{(35.584 - 2464.192x + 1183.168x^2 - 329.152x^3 + 84.512 - 56.9344y + 0.50707y^2 - 0.00258y^3)((18.11 - x)^2 + 11.97}{22.32(18.11 - x)}$$
$$= 0.569y^2 + 0.569(17.359 - x)yy' - 2452.5$$

This equation is solved with Euler method differential equation solver of Wolfram alpha® pro. Following graphs show sample data:



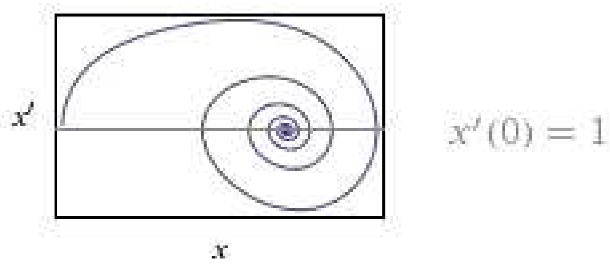


Figure 18. Variation of compression vs time after a bump

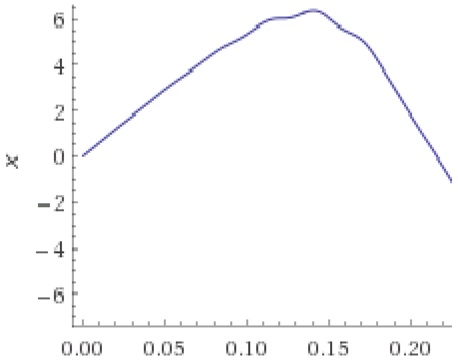
Figure 19. Variation of compression velocity vs compression

A C code is written which calculates forces on frame with a method analogous to Euler's method.

- 5 meter drop test

A thought experiment of free fall of whole vehicle from 5 meter height is done. It is assumed that the chassis fall only on front wheels.

Methods developed to study forces through springs and dampers were implemented to get the following results:



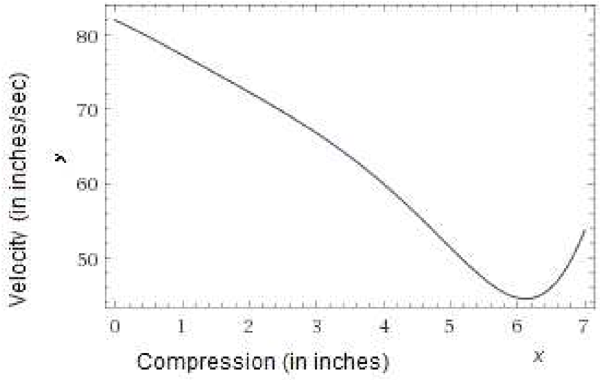


Figure 20. Compression vs time curve

Figure 21. Velocity vs compression curve

Data exaggerated by Calculation:

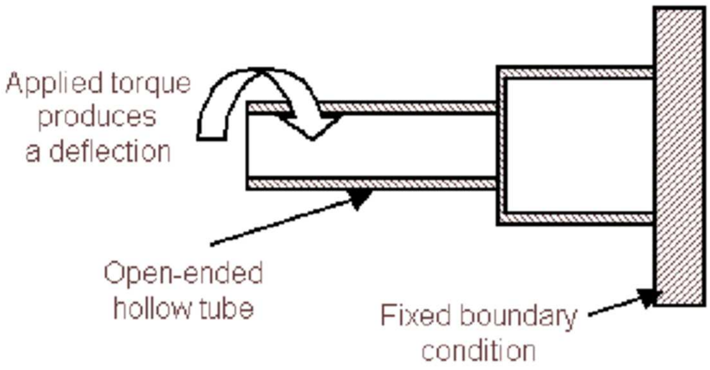
Maximum force exerted by spring:	12545.93 N
Maximum Horizontal force:	8663.34 N
Maximum Vertical force:	10864.78 N

These maximum horizontal and vertical forces were applied on the roll cage, through the spring–damper mounting points individually and together.

4.4.8 Torsional Rigidity

Torsional Rigidity is the resistance of frame against stress which may twist it along any axis.

As shown in the picture, the torque applied on the left tube displaces one end of the tube tangentially along its surface.



Torsional stiffness is a measure of resistance of material against being twisted. It's commonly used unit is N-m/deg.

This analysis is most important of all analyses here due to the reason that our vehicle is in continuous in torsion due to bumps and droops. Hence, the vehicle faces varying torque almost all the time.

We intend to check torsional rigidity of roll cage, which is one of the most imposing and thus most important dynamic phenomenon to study about the vehicle frame. This varying torque is also a reason for fatigue.

The analysis was carried on ANSYS Workbench platform (Static analysis). In each of these cases, force assumed is 2500 N. Arrows represent the direction and point of application of forces, while nodes marked with circles are fixed points.

We are considering longitudinal torsion in all these three cases below. In this type of torsion, torsion is along X-axis i.e. vehicle centerline i.e. orange arrow.

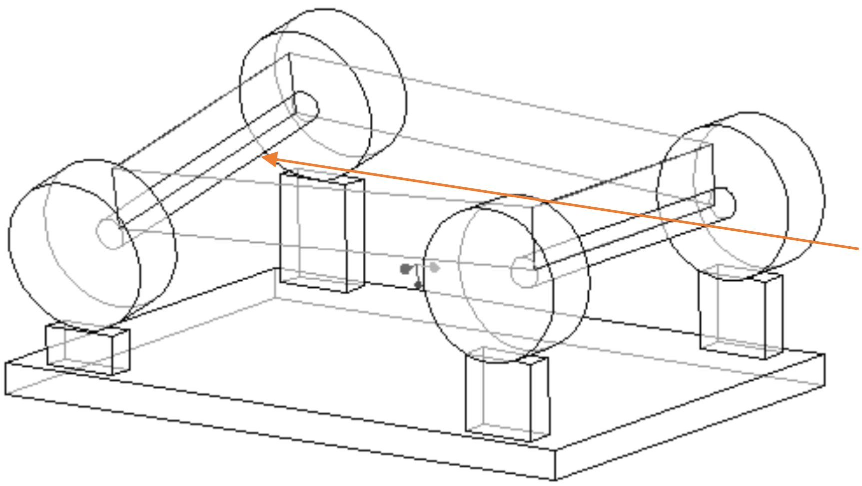


Figure 22. Longitudinal Torsion Deformation Mode

A target for torsional stiffness was set at 2000 N-m/deg.

Since, it is not possible to calculate the actual torsional stiffness, torsional stiffness was checked under following three conditions to get an approximate data

i. Rigid support on rear suspension spring joints & and applying coupled forces on front suspension spring points.

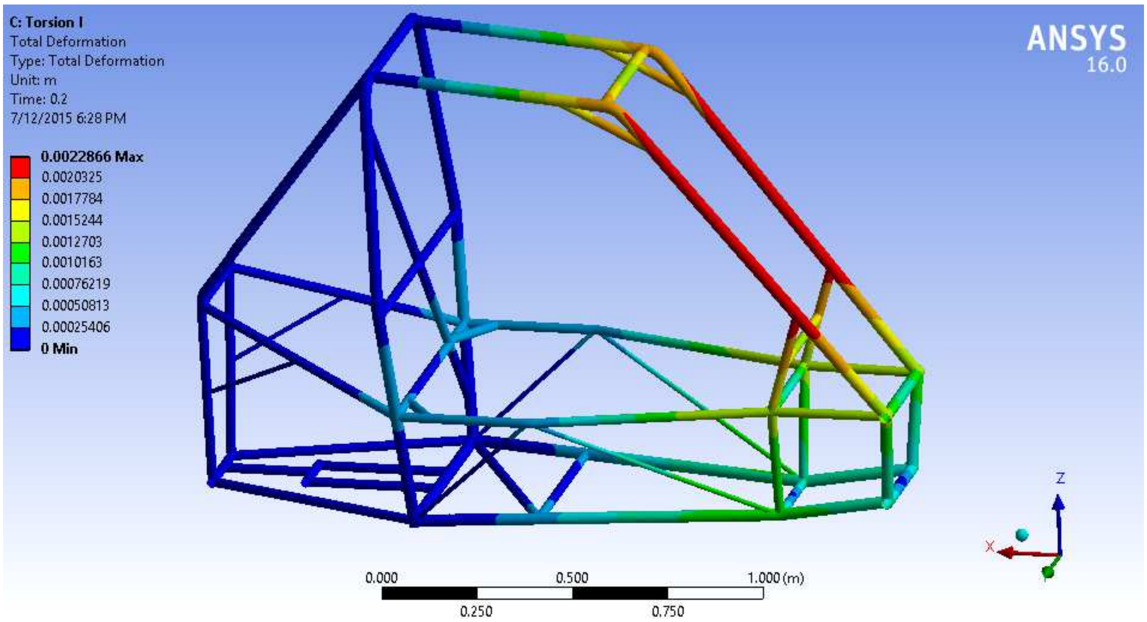


Figure 23. Torsional stress - I

ii. Rigid support on front points & coupled forces on rear force members.

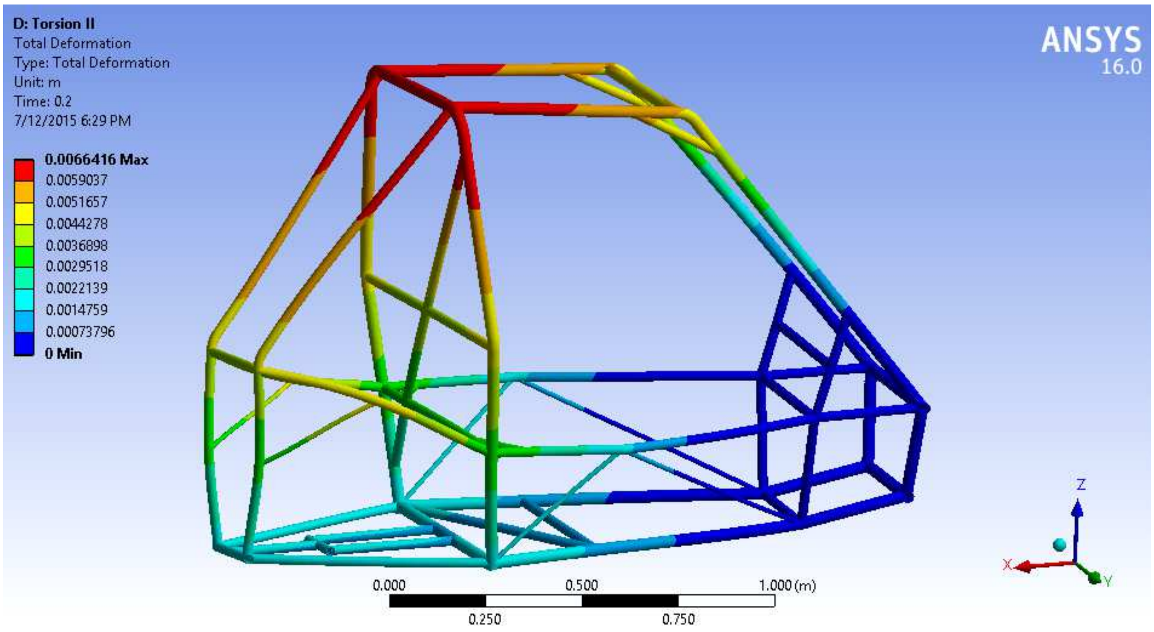


Figure 24. Torsional stress - II

iii. Rigid support on one of the front load point as well as diagonally opposite rear member & upward force on remaining suspension spring joints.

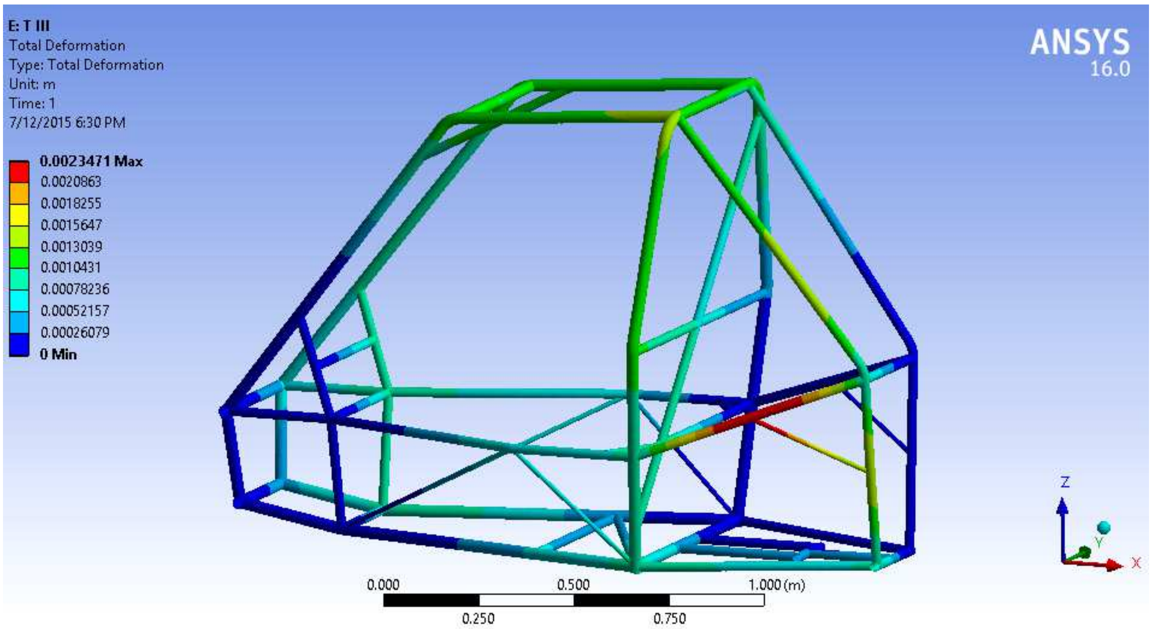


Figure 25. Torsional Stress – III

Calculations and Equations

Torsional rigidity (k) can be derived using the generated data and following equations.

$$k = \frac{\tau}{\theta};$$

$$\vec{\tau} = \sum \vec{l} * \vec{F}; \qquad \theta = tan^{-1}\left(\frac{\Delta y_1 + \Delta y_2}{2l}\right);$$

Where,

$\tau \quad = \quad Torque\ applied\ about\ vehicle\ centerline;$

$\Delta y_i \quad = \quad displacement\ of\ points\ of\ applications\ of\ force;$

$\vec{F} \quad = \quad Net\ Force\ Vector\ on\ each\ member;$

$\vec{l} \quad = \quad Position\ vector\ of\ point\ of\ application\ of\ force\ from\ vehicle\ centerline.$

Please refer to section 7.3 for detailed calculations.

Using above formulae, following results were derived.

	K {torsional stiffness} (in N-m/deg)	*Factor of Safety*
Case I	*1604.69*	*1.94*
Case II	*1257.8*	*1.82*
Case III	*2733.7*	*1.59*
Mean Values	*1865.4 N-m/deg*	*1.78*

Table 2. Torsional stiffness calculated data

As evident from given data, the approximate torsional stiffness resulted out to be 1865.4 N-m/deg, which is pretty close to our target.

4.4.9 Fatigue Analysis

Fatigue analysis is important because being our objective to make an off-road car, it is imperative *that it is sufficiently durable.*

The simplest design rule to prevent fatigue failure is

$$\sigma_{applied} \leq \sigma_{max} < S_e$$

Where S_e is the endurance limit.

This is a valid concept, but not quite so simple in reality. S_e is determined experimentally. Simple approximate S_e formulas exist for steel.

For steel with $S_{ut} \leq 1400$ MPa which is the case with AISI 1018,

$S_e' = 0.5*S_{ut} = 220$ MPa for AISI 1018.

Which comes at about 2 inches of compression of suspension.

Where S_{ut} = ultimate strength and S_e' = unmodified, laboratory determined value.

We have done fatigue analysis in static structural analysis of ANSYS workbench. And fatigue data which we need to give it are as follows:

Cyclic strain hardening exponent	*0.27*
Cyclic strength coefficient	*1259 MPa*
Fatigue strength coefficient	*782 MPa*
Fatigue strength exponent	*-0.11*
Fatigue ductility coefficient	*0.19*
Fatigue ductility exponent	*-0.41*

Table 3, Fatigue data for our AISI 1018 tubes

There are two types of analysis down here for fatigue inspection:

- Stress life based
- Strain life based

Also both fully-reversed and with mean-stress analyses are done so as to appreciate the difference on durability of frame due to weight. When mean stress is taken into account, Gerber mean stress correction theory is applied.

As a matter of fact, strain-life based analysis is more exact for small life (<10000 cycles) thus we have considered it for purpose.

We have considered two cases which are most susceptible to failure for fatigue analysis:

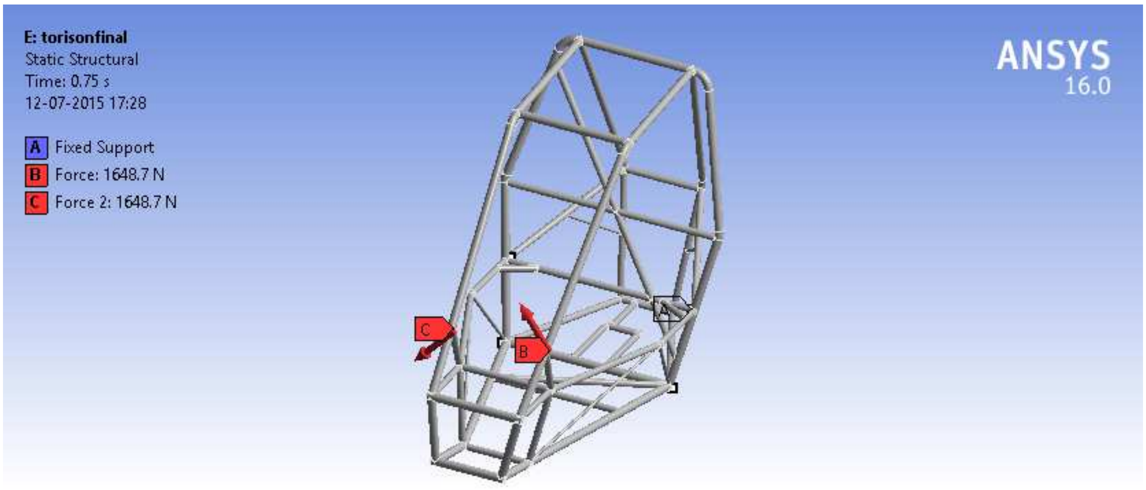


Figure 26. Fatigue Analysis: Longitudinal Torsion

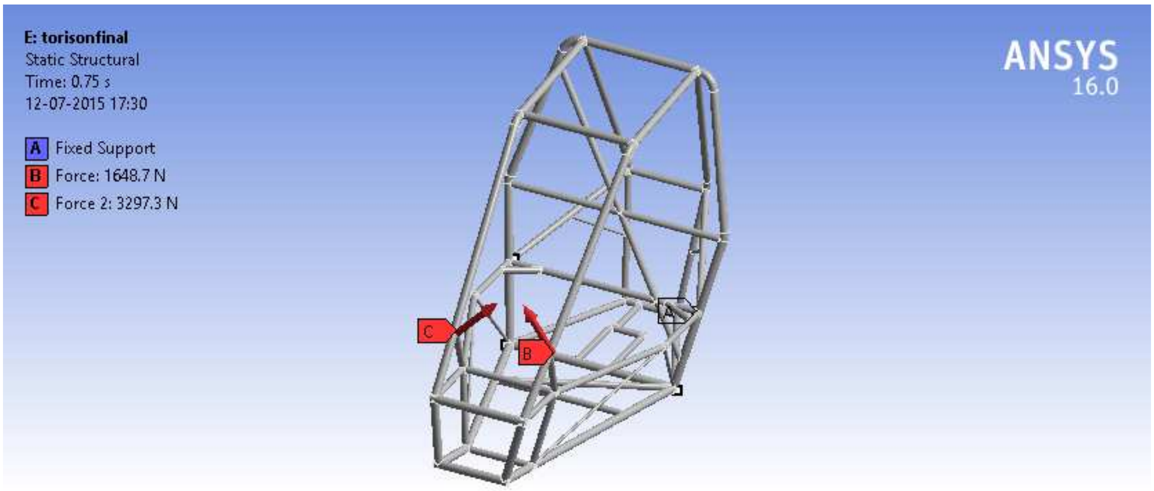


Figure 27. Fatigue Analysis: Vertical Bending

Two types of analyses are done for each case:

- Type I: Fully-reversed strain-life based analysis with maximum forces that can be transmitted through suspension without damaging it (F_h=8600N & F_v=10000N).
- Type II: Fully-reversed stress-life based analysis with quarter of maximum forces i.e. (F_h=2150N & F_v=2500N)

Note that the safety factor is in accordance with infinite life = 10E9.

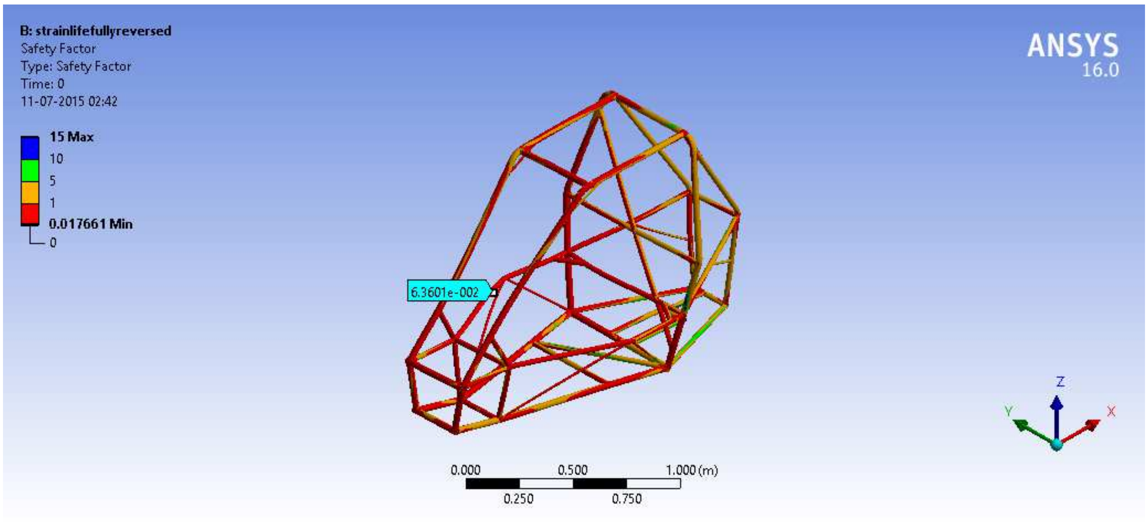


Figure 28. Fatigue Analysis: Type I, Vertical Bending, FOS

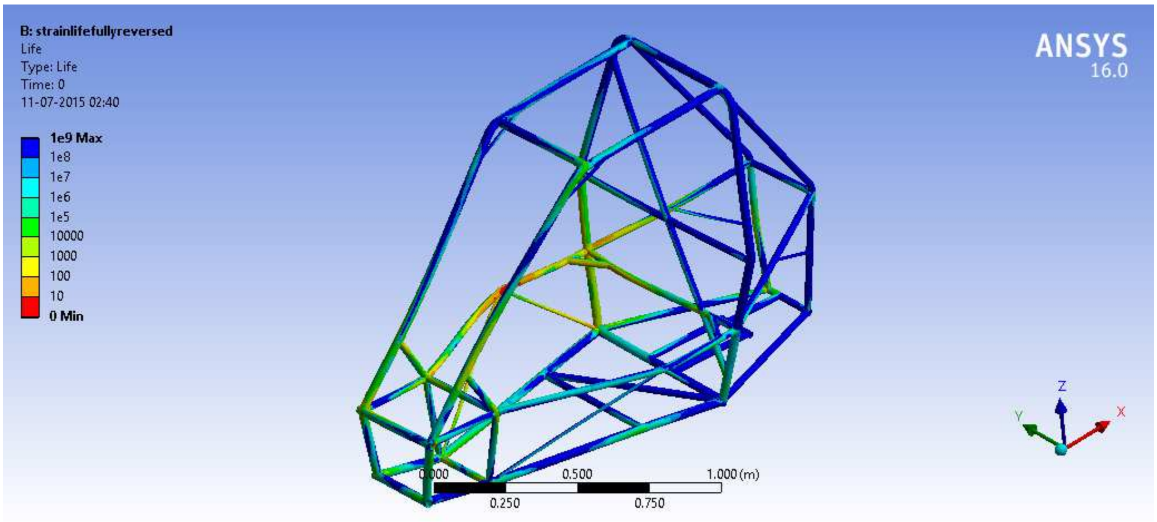


Figure 29. Fatigue Analysis: Type 1, Vertical Bending, Life

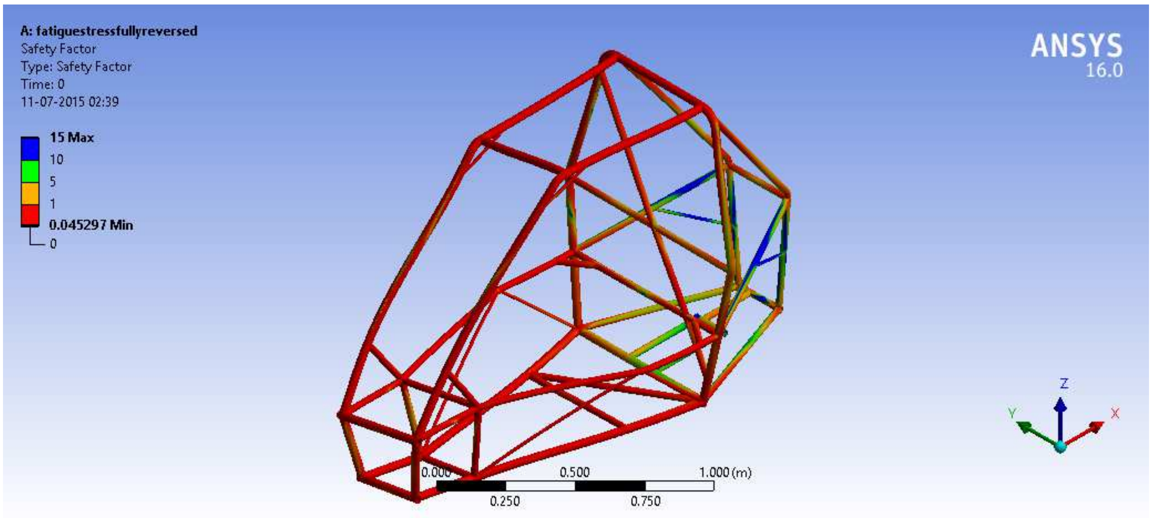


Figure 30. Fatigue Analysis: Type 1, Longitudinal Torsion, FOS

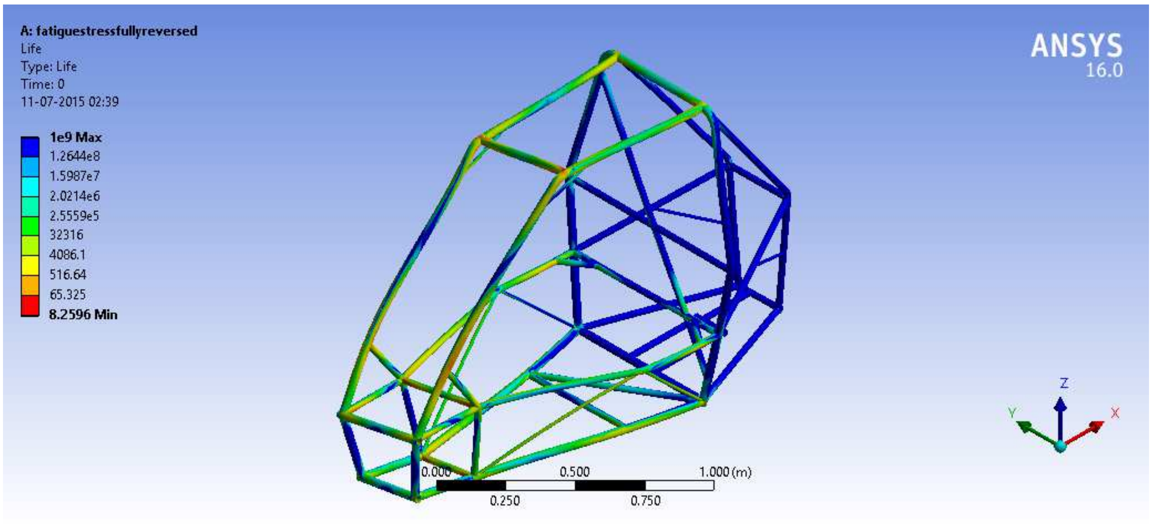


Figure 31. Fatigue Analysis: Type 1, Longitudinal Torsion, Life

FATIGUE TYPE	Life (cycles)	FOS
Longitudinal Torsion	*8*	*0.04*
Vertical Bending	*1*	*0.01*

Table 4. Fatigue Analysis: Type 1, Analysis Results

For Type II (Fully-reversed stress-life based analysis with quarter of maximum forces)

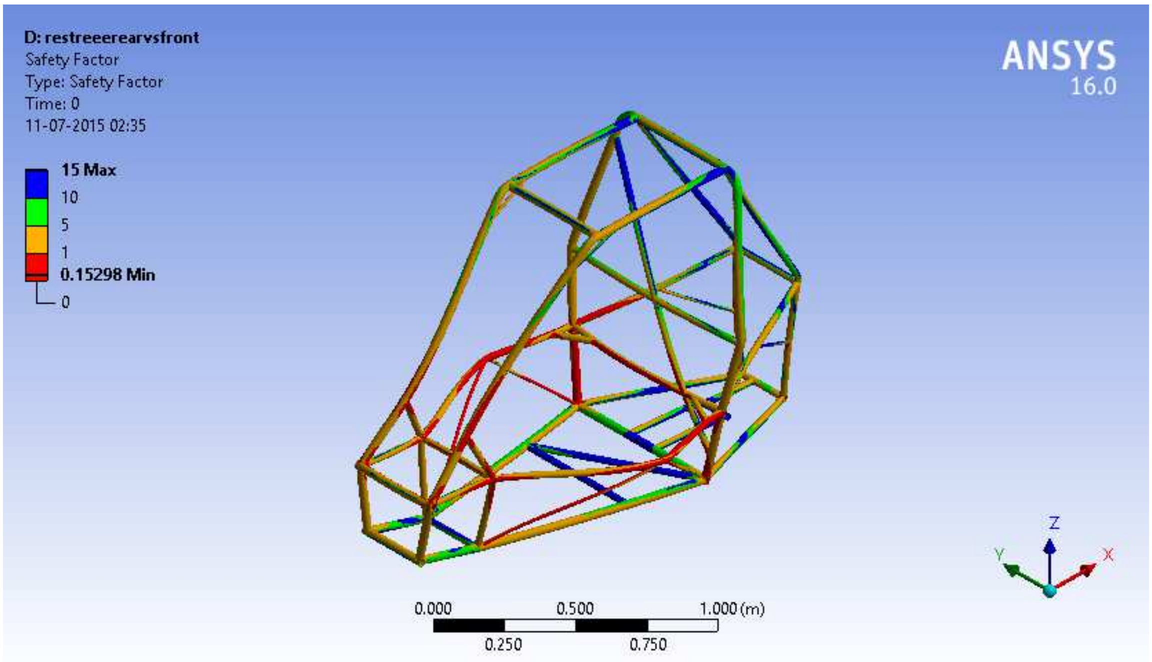


Figure 32. Fatigue Analysis: Type II, Vertical Bending, FOS

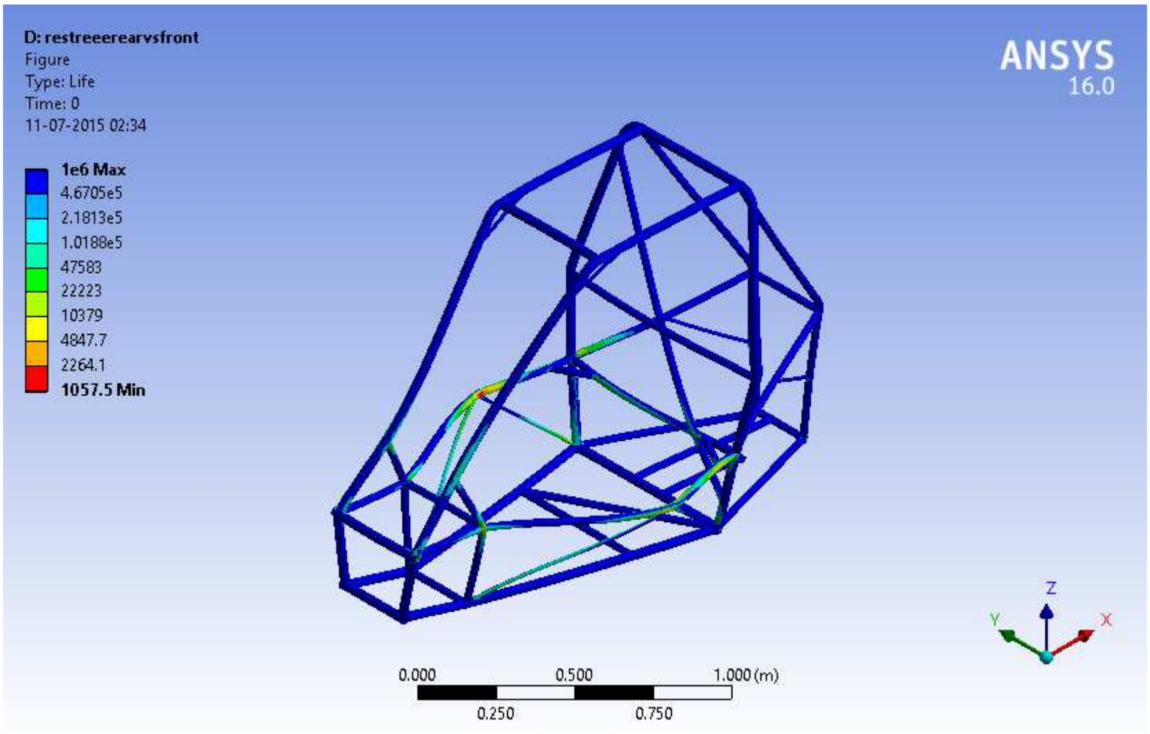


Figure 33. Fatigue Analysis: Type II, Vertical Bending, Life

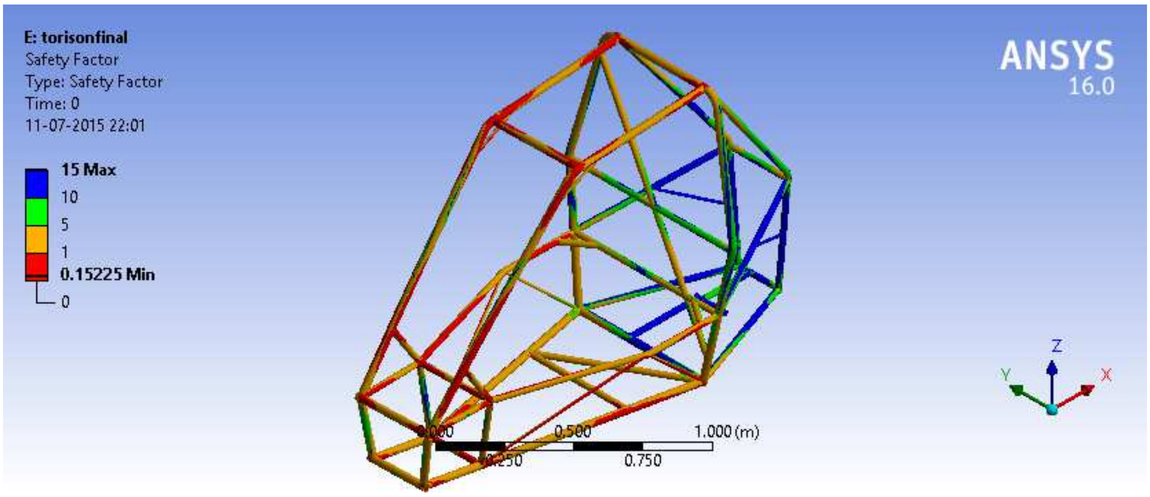


Figure 34. Fatigue Analysis: Type II, Torsion, FOS

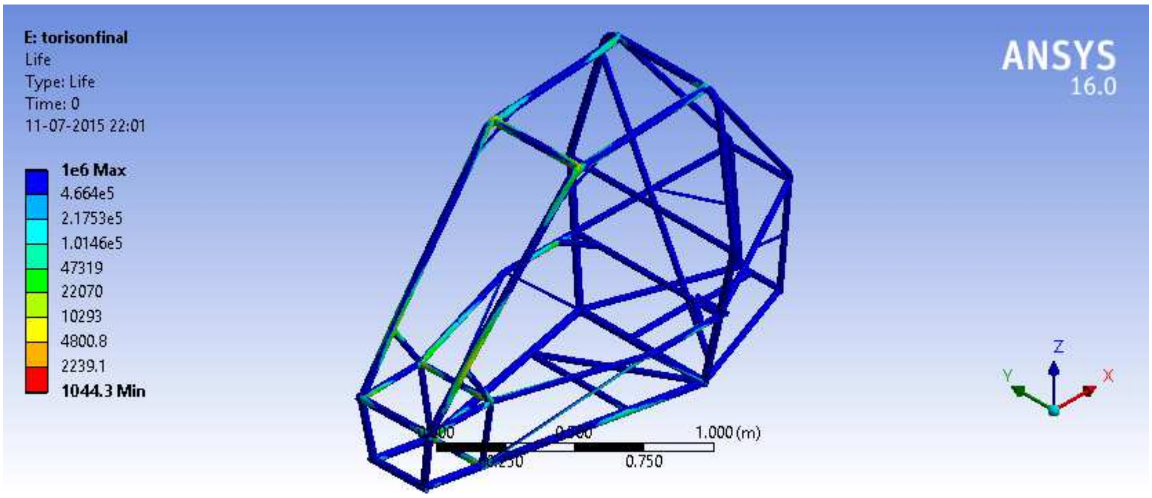


Figure 35. Fatigue Analysis: Type II, Torsion, Life

FATIGUE TYPE	Life (cycles)	FOS
Longitudinal Torsion	*1044*	*0.15*
Vertical Bending	*1057*	*0.15*

Table 5. Fatigue Analysis, Type II Results

In the end, we emphasize the impact of finishing, grinding and welding on endurance with the fact that

$$S_e = k_a * S_e' \quad \textit{(if all other k are 1)}$$

Where S_e is modified endurance limit.

$$K_a = \textit{surface finish factor} = aS_{ut}^{b}$$

Surface finish	a (MPa)	b
Ground	*1.58*	*-0.085*
Machine or cold drawn	*4.51*	*-0.265*
Hot rolled	*57.7*	*-0.718*
As-forged	*272.0*	*-0.995*

Table 6. Impact of Surface Finish on Strength

We have cold-drawn and ground tubes which are undoubtedly the best available to us.

4.4.10 Modal (Vibrational Analysis)

Why modal analysis?

Though the concept in itself being imperative, nevertheless also with experience and study of various materials on vehicle dynamics, we decided on to do modal analysis.

It deals with the natural modes of oscillations of a body under self-weight and their respective frequencies.

We have done the modal analysis in ANSYS Workbench's Modal. Under modal analysis, we studied natural frequency of vehicle frame under various modes (first 5 modes with frequency <100Hz) with base of frame kept fixed.

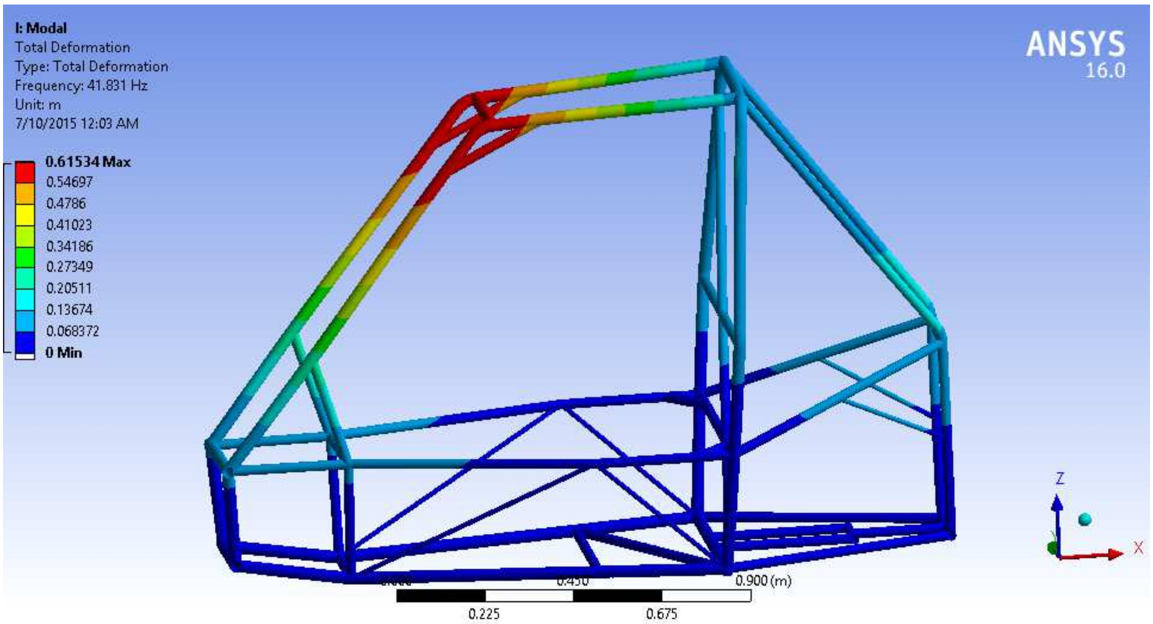


Figure 36. Modal Analysis, Mode 1: Flapping, Frequency: 41.8 Hz

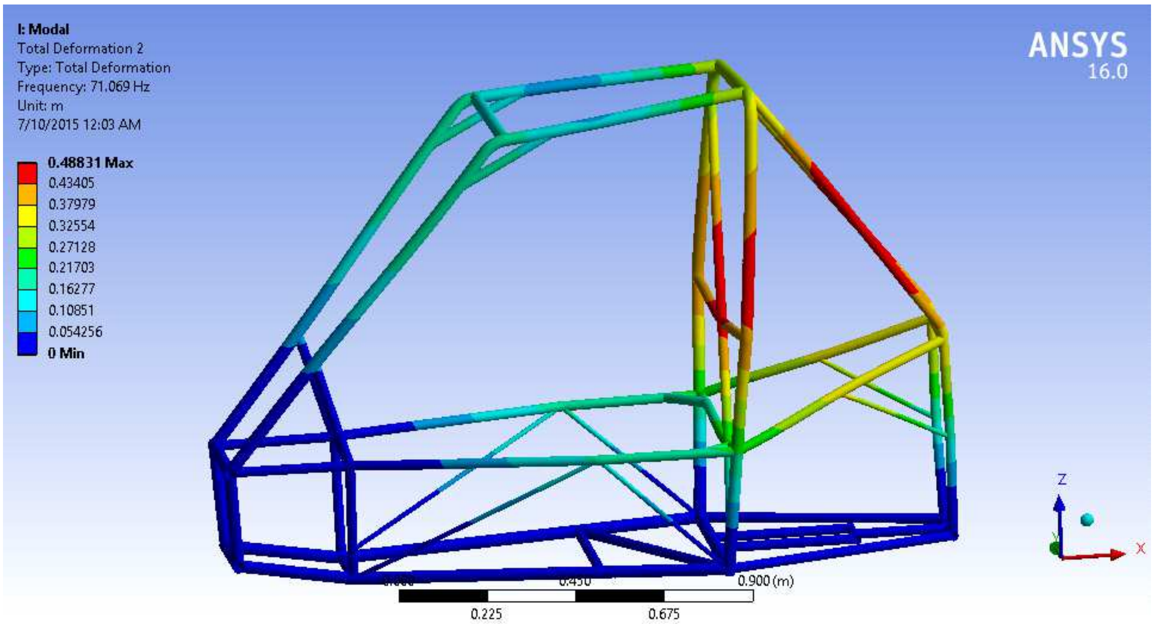


Figure 37. Modal Analysis, Mode 2: Lateral Flapping, Frequency: 71.1 Hz

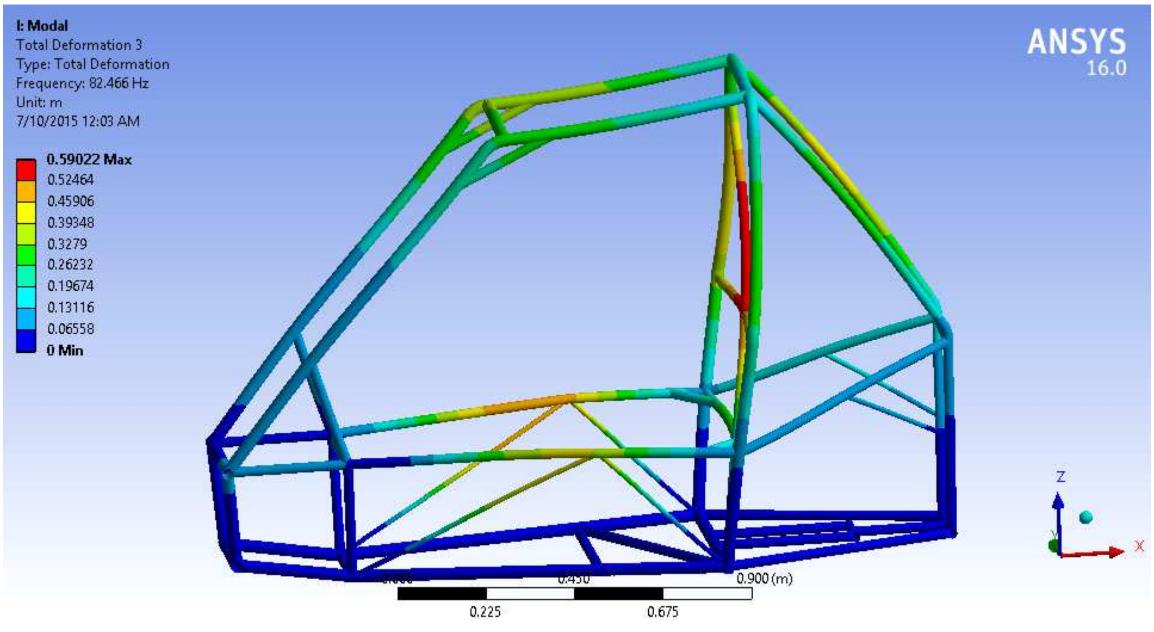


Figure 38. Modal Analysis, Mode 3: Torsional Frequency, Frequency: 82.47 Hz

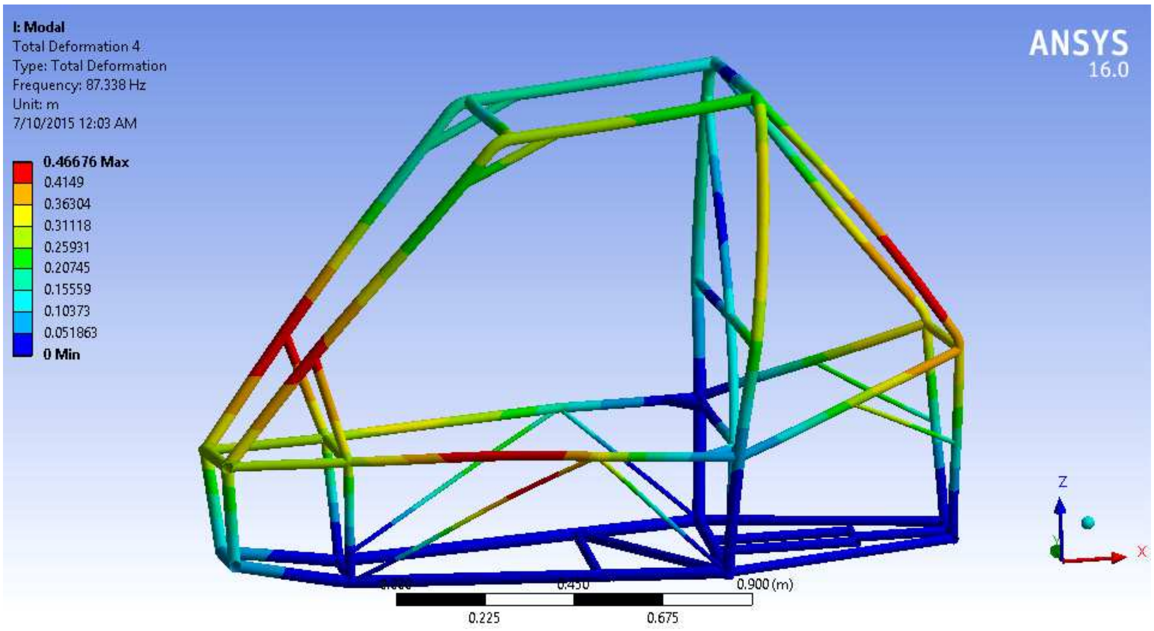


Figure 39. Modal Analysis, Mode 4, Frequency: 87.3 Hz

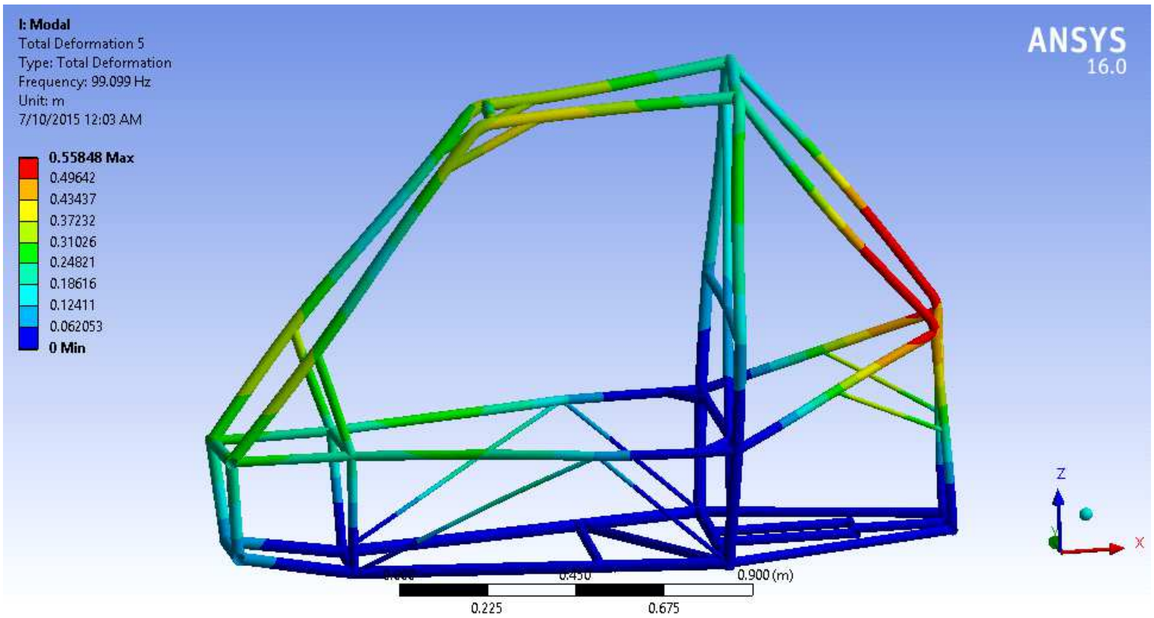


Figure 40. Modal Analysis, Mode 5, Frequency: 99.1 Hz

The sole motivation of doing modal analysis is that under any condition while driving natural frequency of vehicle frame should remain far from frequency of forces which it receives from suspension, otherwise it will undergo resonance. Taking into consideration that point we found two areas of concern:

->stable state: frequency of suspension is frequency of force.

Frequency of suspension=x

So, FOS=71.1/x

->unstable state: frequency of force is frequency of wheel (wobbling)

Frequency of wheel<=y

$$y = \frac{Maximum\ Speed\ of\ vehicle}{Applicable\ radius}$$

Y=51.35 rad/sec

So, FOS=$\frac{71.1}{51.35}$=1.385

Conclusions:

As the FOS for resonance of vehicle frame is high enough we may conclude that vehicle frame members are not prone to high amplitude vibrations which prove to be catastrophic.

4.4.11 Tabs

Tabs are the foremost thing which transfers the forces from dampers/other components to the roll cage. Thus, their analysis must be a priority. We performed these simulations in SolidWorks Simulation wizard. We present here screenshots of analysis for the front damper tabs.

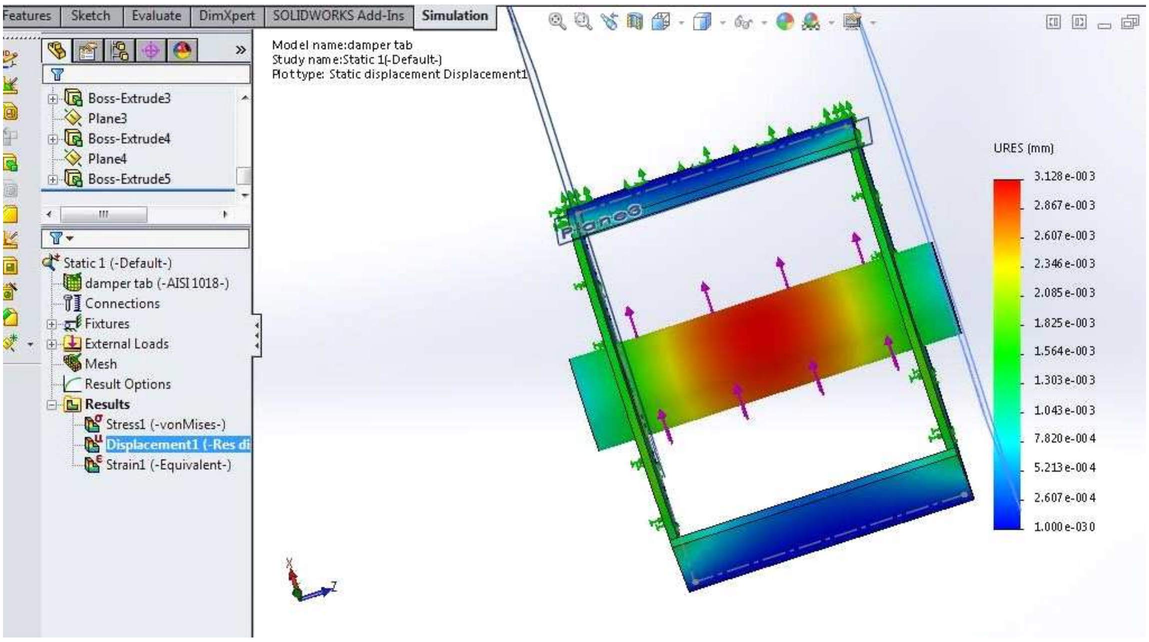


Figure 41. Front Damper Tabs Analysis, Deformation

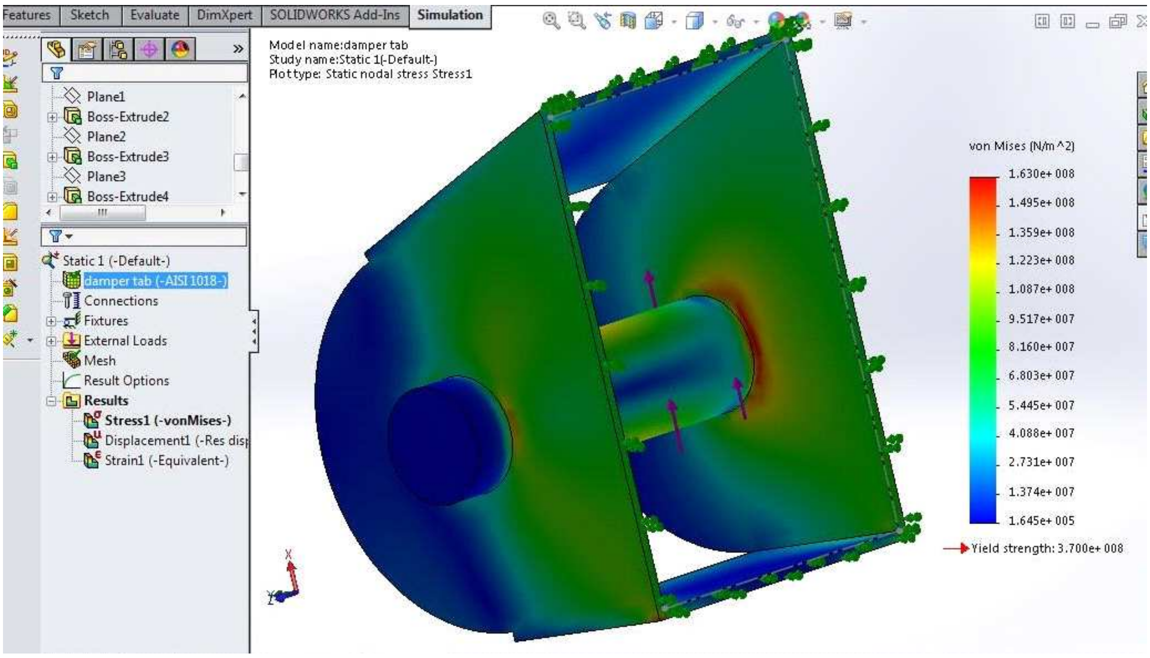


Figure 42. Front Damper Tabs Analysis, Von-Mises Stress

As observed, the peak stress (1.63E008) is much lesser than the Yield strength of the material. Similar results were obtained for the rest of them.

4.5 The Final Roll Cage CAD

The term “Final Roll Cage Cad” is really vague for us. No matter, how many changes we did, some changes (due to rules/driver comfort) were always left. However, the image CAD as of when the report was being drafted was as follows:

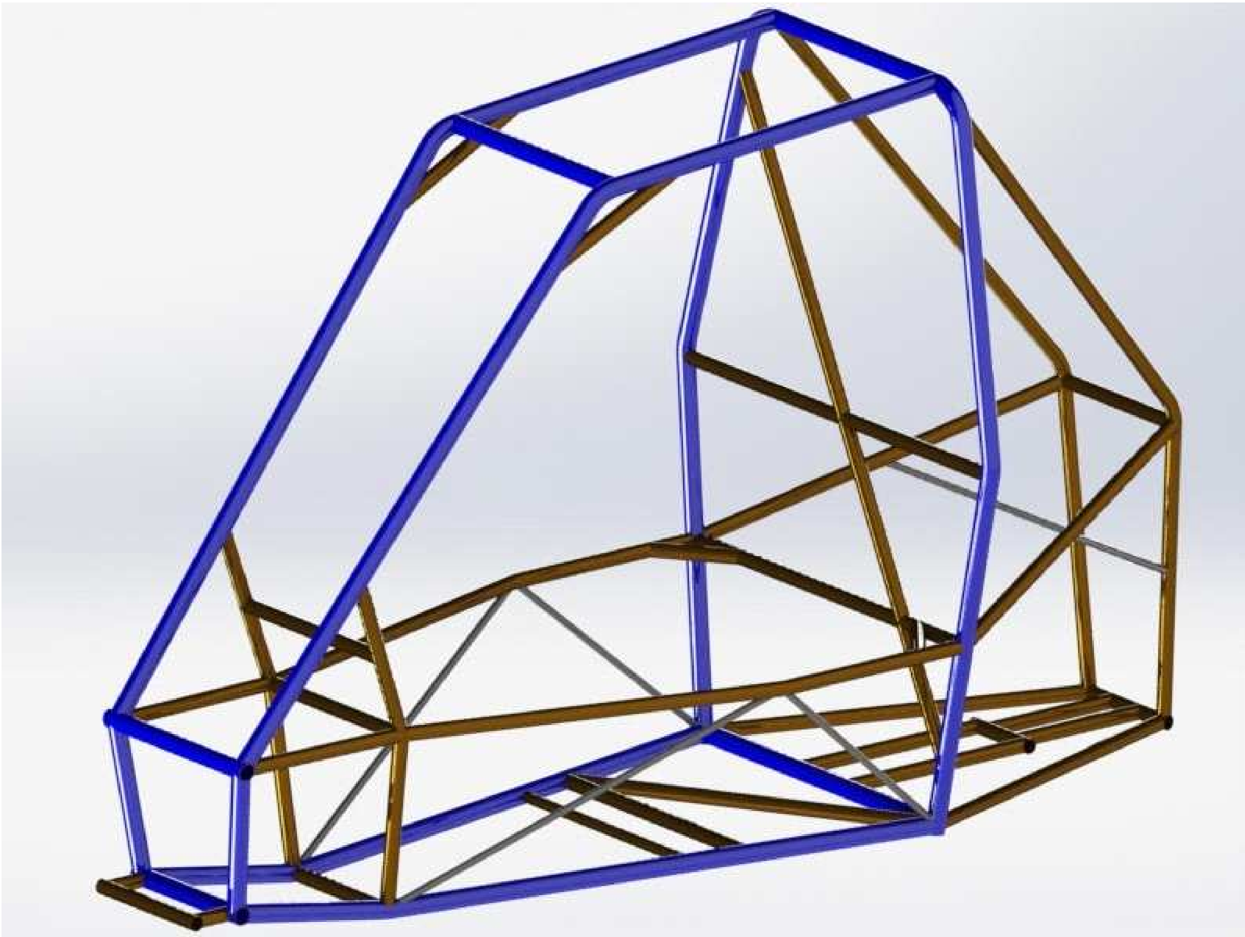


Figure 43. Final CAD Rendering

As according to Solidworks, the Roll cage weighed 24.7 KGs, with the height of its center of mass was at a height of 453 mm from the base of roll cage. The weight of chassis can be estimated to around 27-28 Kgs.

5. Manufacturing

5.1 Notching and Bending Profiles

A quite complex process was opted for notching and bending tubes. But, in the end, our hard work paid off with the accuracy we obtained in bending and notching of the tubes. A sample bending profile (Scaled down) has been attached.

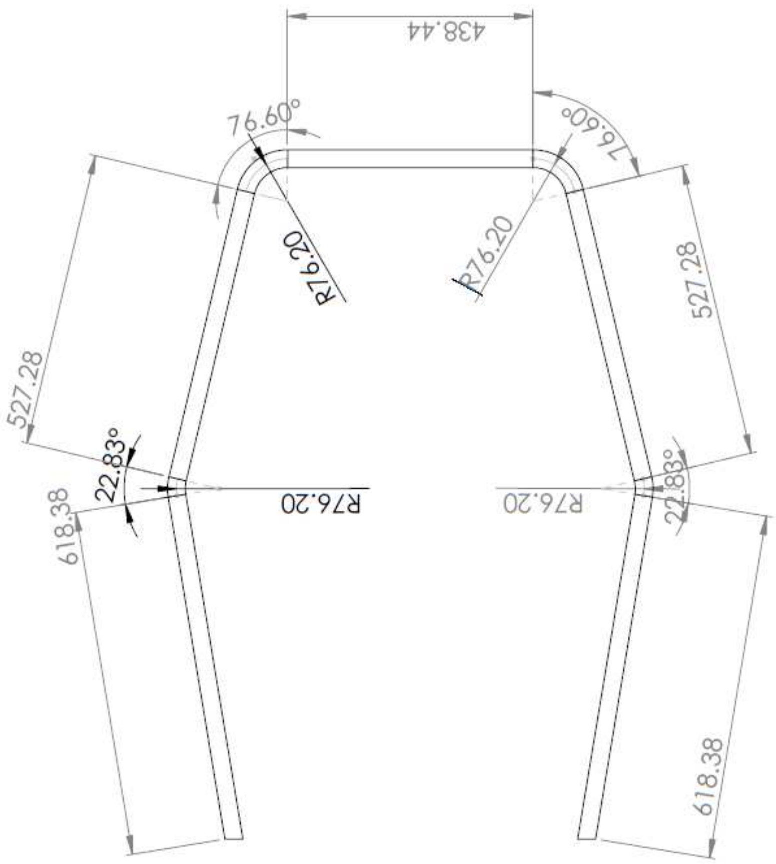


Figure 44. RRH Bending Profile

A Notching Profile is also attached.

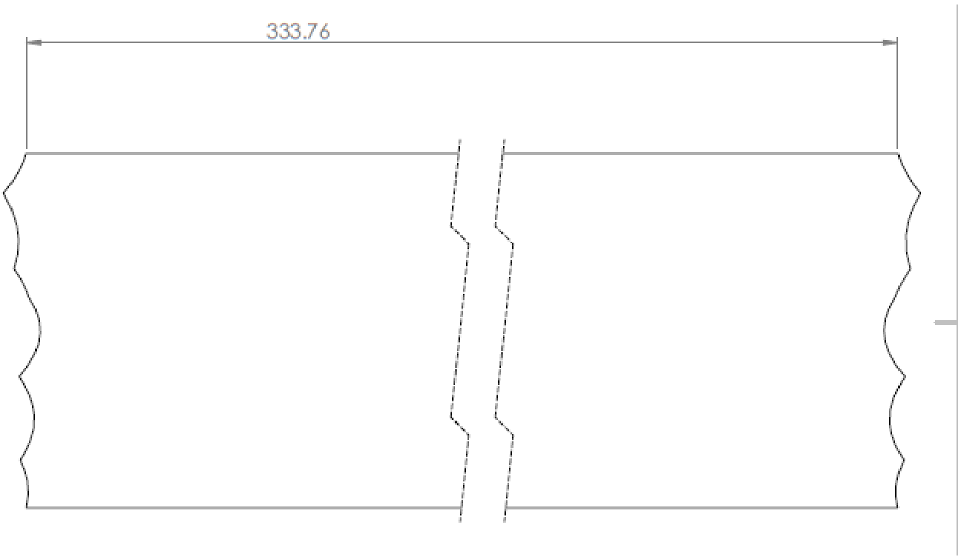


Figure 45. Notching Profile Tube S11

5.2 Jigs, Fixtures and Welding

A fixture is work holding device that holds, supports and locates the workpiece for a specific operation but does not guide the cutting tool. It provides only a reference surface or a device. What makes a fixture unique is that each one is built to fit a particular part or shape. The main purpose of a fixture is to locate and in some cases hold a workpiece during either a machining operation or some other industrial process. A jig differs from a fixture in that a it guides the tool to its correct position in addition to locating and supporting the workpiece. Examples: Vises, chucks.

Some images of fixtures that we made are attached:

Figure 46. Fixtures of Lower members of Aft Bracing

Figure 47. Fixture for Front Lateral Cross

Method that we used for welding our roll cage was TIG.


Figure 48. RRH-RHO-Aft Bracing Intersection

5.3 The Final Structure

6. References, Tools & Software

i. SAE technical paper: Design, Analysis and Testing of a Formula SAE Car Chassis

ii. Wolfram alpha math tool

iii. Fox E vol 3 technical manual

iv. ASM Handbook, Volume 1: Properties and Selection: Irons, Steels, and High-Performance Alloys

v. Solidworks 2015 Student Edition

vi. Ansys 16.0: Workbench

vii. Matlab 2015

7. Appendix