ISSN No:-2456-2165

Formula SAE Chassis System

Design, Optimization & Fabrication of FSAE Spaceframe Chassis

Rishi Desai.
Student - Mechanical Engineering
C.S.P.I.T – Charusat University
Anand, Gujarat – India.

Abstract:- Chassis is a major part of any automotive design. It is responsible for supporting all functional systems of a vehicle and also accommodates the driver in the cockpit. Designing a chassis for driver's safety is always been a concern, especially for a race car. In this report, few techniques are mentioned on how to analyze a formula student race car chassis to ensure its structural stability for the driver's safety.

This report aims to produce a clear idea about the types of analysis to be run on a student formula chassis with the amount of load or G forces to be applied to it using Solid works software, to make sure that the driver is safe inside the cockpit.

The overall scope of this project can be broken down into two objectives. The first objective of this report was to design, manufacture, and test a Formula SAE racecar chassis for use in the 2020 Formula Bharat & SAE SUPRA. Several factors will be taken into account, including vehicle dynamics, chassis rigidity, component packaging, and overall manufacturing and performance. The major objectives of Team Ojaswat while designing this chassis are listed below –

- **Design and optimize the chassis system considering aesthetics ergonomics and giving utmost priority to the driver's safety. For the design procedure, we have taken references for various SAE research papers.**
- **The CAD file is entirely developed on Solid works 2018-19. Also, we have tried to use Ansys 18.2 2D structural analysis. For performing dynamic suspension simulations, we have used Lotus shark and Raven. The mathematical truss model was developed in MathWorks – R2020.**
- **The fabrication is done in house using Jigs & Fixture table. We have used the TIG and Arc welding machine for welding purposes. The material used in overall frame design is AISI 4130 chromium-molybdenum steel alloy for maximum strength to weight ratio. And in addition to that, it has great weldability.**
- **Fabrication of the 2019-2020 model is brought out in a very unique way. We have used the weldments feature of solid works in a very unique way to profile and notch the tubes to obtain great accuracy.**
- **The base sketch was also developed uniquely by printing the top view of the chassis and developing laser-cut jigs and fixtures for maximum accuracy.**
- **For final validation, the COG of the cad file and the prototypes were compared from a moment formula obtained from William & Douglas Vehicle dynamics. And finally, the results were verified using destructive testing performed on the torsional rig.**

Keywords:- *FSAE Chassis, Chassis Torsional Rigidity, Bending Stiffness, Simulations, Suspension, Vehicle Dynamics.*

I. INTRODUCTION

A. Formula Student: The Challenge

Team Ojaswat is a formula student racing team consisting of students, from the Charotar University of Science & Technology. Each year the team designs, builds, tests, and eventually races their car against other university teams from all over the world in the Formula Student competition.

The students are to assume that a manufacturing firm has engaged them to produce a prototype car for evaluation. The intended sales market is the nonprofessional weekend auto crosser sprint race and the firm is planning to produce 1,000 cars per year at a cost below 10 lakhs.

The car must be low in cost, easy to maintain, and reliable, with high performance in terms of its acceleration, braking, and handling qualities. Watched closely by industry specialists who volunteer their time each team will go through the following rigorous testing process of their car:

Static events: Design, Cost, and Presentation Judging − Technical and Safety Scrutineering − Tilt Test to prevent cars from rolling over − Brake and Noise Test.

Dynamic Events: Skid Pad − Acceleration − Sprint/qualification − Endurance and Fuel Economy – Autocross.

B. Problem Definition

A typical open-wheeled single-seater chassis in the Formula Student competition consists of several parts: − a lightweight structural and protective driver compartment or cockpit − a lightweight structural engine compartment − esthetic and aerodynamic exterior − crash impact attenuators. So far Team Ojaswat has been building a tubular space frame model.

However, to use them correctly in a race car is very difficult because they offer very little design freedom. Problems are met when trying to attach the advanced

ISSN No:-2456-2165

suspension system to the structural cockpit. Additional material is required to meet stiffness and strength demands which partly cancels the advantage of the lightweight panels. The necessary additional material increases the material cost and the increase in vehicle mass and center of gravity height reduces performance in handling.

The main challenge for our team was to shift from 13-inch rims to 10-inch alloy wheels with a heavy engine of 600 ccs. And maintain the total weight of the vehicle to 250 kg for best performance. For that purpose, we had to come up with a new design without any references. We performed several iterations to reach a final design for fabrication.

Even after performing several simulations on advanced software like Solid works, Annsys, Lotus, and many more, we had no assurance the chassis would last in real space and time scenario. Therefore, this encouraged us to proceed forward with Destructive testing and obtain experimental value on the torsional Rig apparatus.

C. Design constraints

Considering Formula Bharat 2020 rule book which is affiliated with FSG (Formula student Germany) following were main constraints considering chassis design and the rest are attached in the Appendix.

T 3 GENERAL CHASSIS DESIGN

T 3.1 General Requirements

T 3.1.1 Among other requirements, the vehicle's structure must include:

- Two roll hoops that are braced
- A front bulkhead with support system and IA
- Side impact structures

T 3.2 Minimum Material Requirements

T 3.2.1 Table 4 shows the minimum requirements for the members of the primary structure if made from steel tubing.

Item or application	Minimum wall thickness	Minimum cross sectional area	Minimum area moment of inertia
Main and front hoops, shoulder harness mounting bar	2.0 mm	175 mm^2	11 320 mm^4
Side impact structure, front bulkhead, roll hoop bracing, driver's restraint harness attachment (except as noted above)	1.2 mm	119 mm^2	8509 mm^4
Front bulkhead support, main hoop bracing supports	1.2 mm	91 mm^2	6695 mm^4

Table 4: Minimum Material Requirements

Fig. – I.C.1 (General Chassis Constraints)

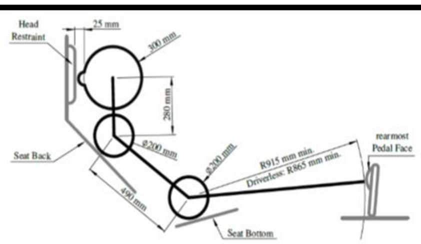


Figure 13: Percy placement

Fig. – I.C.2 (Percy Templet)

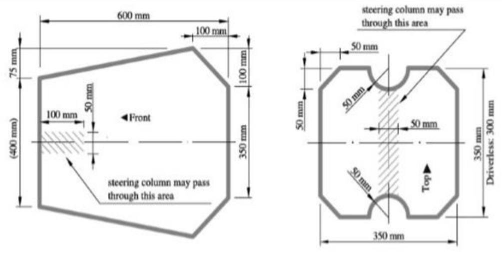


Figure 11: Cockpit opening template (left) and cockpit internal cross section template (right)

Fig. – I.C.3 (Cockpit Templets)

D. Concept Generation

➢ *General procedure –*

To construct the chassis, the design team took a “bottom-up” approach. This approach allows for flexibility in the final design. the initial plan is to design a space frame car with the standard FSAE tubing rules, minimum wheelbase (1600mm), wide impact attenuator (standard – 300x200x200 mm), and constructed from Chromoly steel (AISI 4130). The team created possible concepts in SolidWorks and used finite model analysis (FEA) to accurately assess the design's stiffness, weight, etc. This allowed the team to easily compare different iterations for positive and negative metric gains.

➢ *Space-Frame vs. Monocoque –*

Any FSAE team stands with 3 options, Spaceframe, monocoque, and hybrid frame. Out of which Team Ojaswat 2020 decided to use a tubular spaceframe to reduce complexities. Also, the tubular spaceframe has greater strength, stiffness, weldability machinability and above all easy to fabricate using jigs and fixtures.

➢ *Standard vs. Alternate Frame Design –*

The alternate design allows for much more flexibility with the cost of more engineering analysis on the overall design. The group would like to focus on the overall design and ensuring all components of the car are compatible with the chassis design instead of focusing on structural equivalence analysis to comply with the FSAE rules. Thus the group has selected to not use any alternate frame rules to simplify the workload, and allow for a greater depth of engineering to be spent on functionality.

E. Design Development

The purpose of the frame is to rigidly connect the front and rear suspension while providing attachment points for the different systems of the car. Relative motion between the front and rear suspension attachment points can cause inconsistent handling. The frame must also provide attachment points that will not yield within the car’s performance envelope.

There are many different styles of frames; space frame, monocoque, and ladder are examples of race car frames. The most popular style for SUPRA SAEINDIA/FSAE is the tubular space frame. Space frames are a series of tubes that are joined together to form a structure that connects all of the necessary components.

However, most of the concepts and theories can be applied to other chassis designs.

A Space frame chassis was chosen over a monocoque despite being heavy, as its manufacturing is cost-effective, requires simple tools, and damages to the chassis can be easily rectified. The chassis design started with the fixing of suspension mounting coordinates and engine hardpoints.

F. Material selection

There are different materials for car chassis which include alloys of aluminum, steel, carbon fiber, etc. Carbon fiber is very lightweight and strong but making chassis from carbon fiber is not an economical decision. Now, there are two materials which meet requirements.

Those materials are SAE AISI 1018 steel and Chromoly AISI 4130 steel. Since AISI 4130 has a better strength to weight ratio, it was finalized. All the tubes that were used to develop the spaceframe were tested. And the hardness, tensile strength & chemical test reports are attached in **Appendix 1.**

PROPERTIES [2],[3]	SAE AISI 1018	Chromoly 4130 Steel
Density (g/cc)	7.8	7.8
Young's Modulus (GPa)	210	210
Elongation at break (%)	19	19
Brinell Hardness	120	200
Strength to weight ratio at Yield (kN-m/kg)	38	100
Yield Strength (MPa)	360	480
Ultimate Strength (MPa)	420	590
Thermal Conductivity: Ambient (W-m/K)	50	42
Thermal Expansion: 20C to 100C (µm/m-K)	11	12
Specific Heat Capacity Conventional (J/kg-K)	370	370

Fig. – I.F.1 (Material Comparision)

G. Design Matrix

Sr.no	Metric	W/C	Units	Target	Accept-able
1	Torsional Rigidity	Stiffness	ft-lb/deg	>1750	>1600
2	Bending Stiffness	Stiffness	kg/m	>45	>42
3	Front Impact	Force	N	<14000	<12000
4	Rear Impact	Force	N	<10000	<8000
5	Side Impact	Force	N	<10000	<8000
6	Freq-uency	Hertz	Hz	0.089	0.067
7	Fatigue	Cycles	Cycles	10 x e6	10 x e6
8	Longitu-dinal bending	Young's Modulus	N/m^2	1.6x10^8	9.2x10^7
9	Lateral bend	Young's Modulus	N/m^2	-	-
10	Weight	Light Weight	kg	<39	<45
11	Weight Distribu-tion	Control/Handling	%	40F 60R	45F 55R
12	Vertical Location of CG	Control/Handling	m	<0.27	<0.35
13	Total Cost	Manufactur-ability	₹	<50000	<65000
14	Ease of Egress	Cockpit Constraint	sec	<3.0	<5.0

Table 1

II. TERMONOLOGIES / LOADS

A. Definitions

- Chassis – The fabricated structural assembly that supports all functional vehicle systems. This assembly may be a single welded structure, multiple welded structures, or a combination of composite and welded structures.
- Chassis member - A minimum representative single piece of uncut, continuous tubing, or equivalent structure.
- Tube frame - A chassis made of metal tubes.
- Monocoque - A chassis made of composite material.
- Main hoop - A roll bar located alongside or just behind the driver's torso.
- Front hoop - A roll bar located above the driver's legs, in proximity to the steering wheel.
- Roll hoops - Both the front hoop and the main hoop are classified as "roll hoops"
- Roll hoop bracing - The structure from a roll hoop to the roll hoop bracing support.
- Roll hoop bracing supports - The structure from the lower end of the roll hoop bracing back to the roll hoop(s).
- Front bulkhead - A planar structure that defines the forward plane of the chassis and provides protection for the driver's feet.
- Impact Attenuator (IA) - A deformable, energy-absorbing device located forward of the front bulkhead.
- Side impact structure - The area of the side of the chassis between the front hoop and the main hoop and from the chassis floor to the height as required in T2.16 above the lowest inside chassis point between the front hoop and main hoop.
- Primary structure - The primary structure is comprised of the following components:
- Main hoop • Front hoop • Roll hoop braces and supports • Side impact structure • Front bulkhead • Front bulkhead support system • All chassis members, guides and supports that transfer load from the driver's restraint system into the above-mentioned components of the primary structure.
- Rollover protection envelope - Envelope of the primary structure and any additional structures fixed to the primary structure which meet the minimum specification defined in T2.3 or equivalent.
- Node-to-node triangulation - An arrangement of chassis members projected onto a plane, where a co-planar load applied in any direction, at any node, results in only tensile or compressive forces in the chassis members as below.

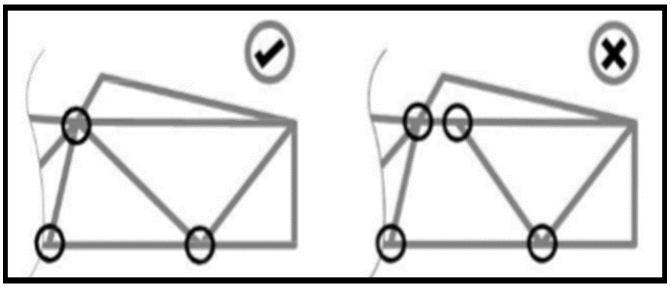


Fig. – II.A.1 (Triangulation Rules)

B. Load transfers in chassis

- *Bending –*

Dynamic loading – Inertia of the structure contributes to total loading and it is always higher than static loading. The road vehicles are 2.5 to 3 times static loads and off-road vehicles are 4 times static loads

Example:

Static loads - Vehicle at rest, moving at a constant velocity on an even road, Can be solved using static equilibrium balance. Results in the set of algebraic equations.

Dynamic loads -Vehicle moving on a bumpy road even at a constant velocity, Can be solved using dynamic equilibrium balance. Generally results in differential equations.

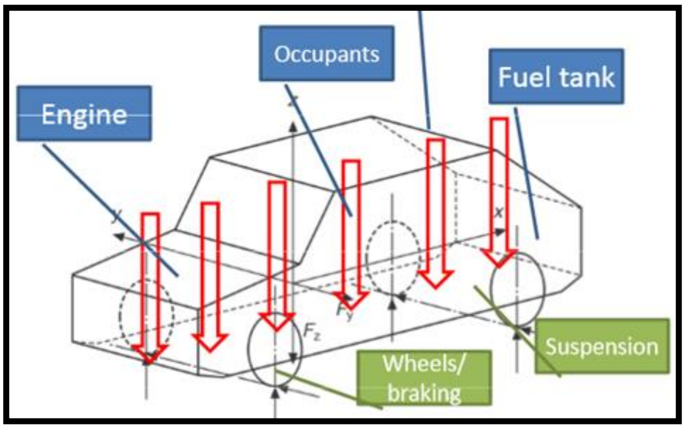


Fig. – II.B.1 (Bending)

- *Torsion –*

When vehicles traverse on an uneven road. Front and rear axles experience a moment. That is Pure simple torsion (Front axle Rear axle).

Torque is applied to one axle and reacted by another axle. –Front axle: anti clockwise torque (front view) –Rear axle: balances with clockwise torque –Resultsinatorsionmoment Results in a torsion moment about the x-axis.

In reality, torsion is always accompanied by bending due to gravity.

ISSN No:-2456-2165

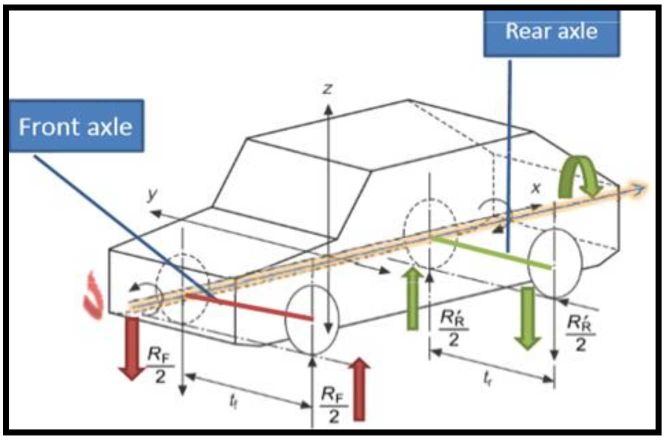


Fig. – II.B.2 (Torsion)

➢ *Combined bending and torsion -*

Bending and torsional loads are superimposed and are assumed to be linear. One wheel of the lightly loaded axle is raised on a bump result in the other wheel go off the ground.

All loads of lighter axle is applied to one wheel. Due to the nature of the resulting loads, the loading symmetry with-z plane is lost. can be determined from moment balance g balance. RR stabilizes the structure by increasing the reaction force on the side where the wheel is off the ground.

The marked – Side is off the ground –Side takes all load of front axle –Side's reaction force increases –Side's reaction force decreases to balance the moment.

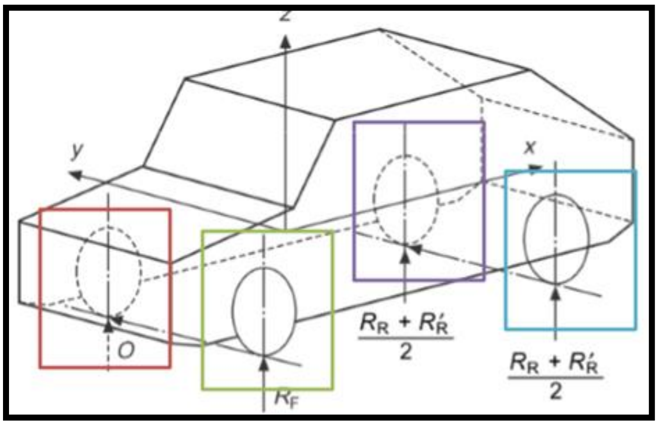


Fig. – II.B.3 (Combined bending and Torsion)

➢ *Lateral loading –*

Due to corning generated attire to ground contact patch, loads are balanced by centrifugal forces. When the inside wheel reaction becomes zero the vehicle rollovers.

Subjected to bending in the X-Y plane, centrifugal acceleration V^2/R =gt/2h. Taking moment at CG during rollover can be given by (MV^2)/R = (Mgt)/2h in both front and rear. Kerb bumping causes high loads and results in the rollover.

Width of car and reinforcements provides sufficient bending stiffness to withstand lateral forces. Lateral shock loads assumed to be twice the static vertical loads on wheels.

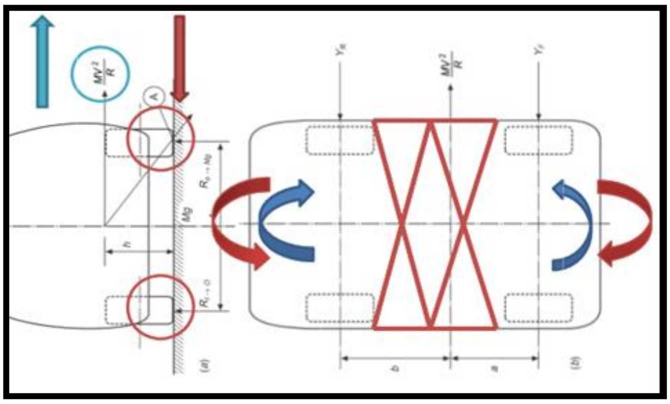


Fig. – II.B.4 (Lateral Loading)

➢ *Longitudinal loading –*

When the vehicle accelerates and decelerates inertia forces are generated.

Acceleration – Weight transferred from front and back. Reaction forces on the rear wheel are given by taking moment about Rr. Rr = [Mg(l-a) – Mh(dV/dt)] / L.

Declaration - Weight transferred from back to front. Reaction forces on front-wheel are given by taking moment about Rf. Rf = [Mg(l-a) – Mh(dV/dt)] / L.

Limiting tractive and g braking forces are decided by a coefficient of friction b/w tires and friction b/w tires and road surfaces.

Tractive and braking forces add bending through suspension. And inertia forces add additional bending.

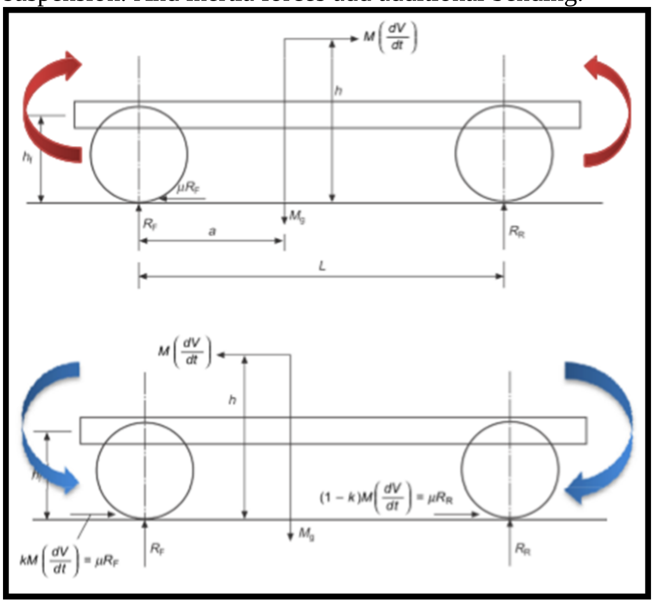


Fig. – II.B.5 (Longitudinal loading)

➢ *Asymmetric loading –*

Results when one wheel strikes a raised object or drops into a pit. It can be resolved as vertical and horizontal loads. Total loading is the superposition of all four loads.

The magnitude of the force depends on – (Speed of vehicle –Suspension stiffness-Wheel mass-Body mass).

The applied load is a shock wave.- (Which has very less time duration-Hence there is no change in vehicle speed-Acts through the center of the wheel).

The resolved vertical force causes: – (Additional axel load, vertical inertia load through CG, Torsion moment) to maintain dynamic equilibrium.

The resolved horizontal force causes- (Bending in X-Z plane, Horizontal inertia load through CG, Moment about Z-axis) to maintain dynamic equilibrium.

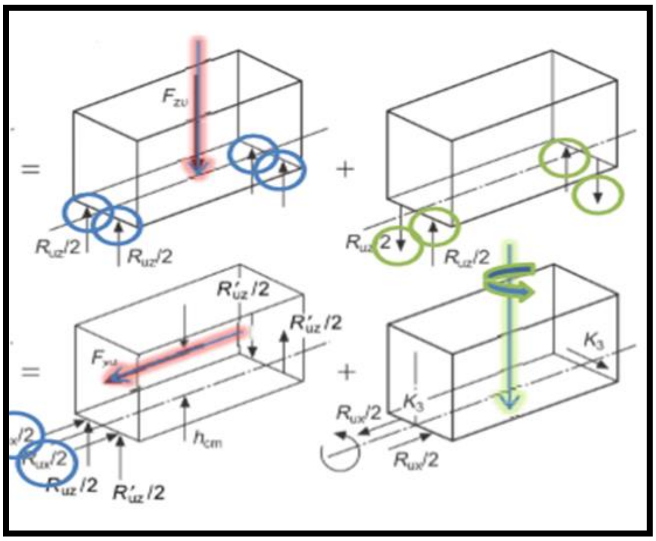


Fig – II.B.6 (Asymmetric loading)

➢ *Allowable stress –*

The nominal allowable stress [σ] is taken to mean the magnitude of stress used for determining the design thickness of the tube wall based on the adopted initial data and the steel grade.

The vehicle structure is not fully rigid. Internal resistance or stress is induced to balance external forces. Stress should be kept to acceptable limits. Stress due to static load X dynamic factor ≤ yield stress.

It should not exceed 67% of yield stress. The safety factor against the yield is 1.5. Fatigue analysis is needed (At places of stress concentration). Eg. Suspension mounting points, seat mounting points).

The allowable stress or allowable strength is the maximum stress (tensile, compressive, or bending) that is allowed to be applied to a structural material. The allowable stresses are generally defined by building codes, and for steel, and aluminum is a fraction of their yield stress (strength):

fa=fy/fs

In the above equation, fa is the allowable stress, fy is the yield stress, and fs is the factor of safety or safety factor. This factor is generally defined by the building codes based on particular conditions under consideration.

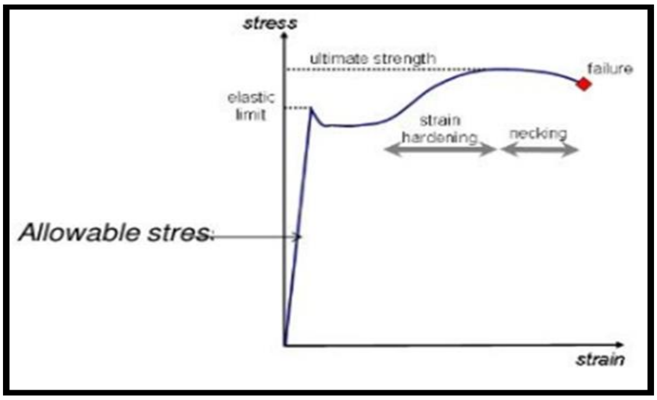


Fig – II.B.7 (Allowable Stress)

➢ *Bending stiffness –*

Bending stress is the normal stress that is induced at a point in a body subjected to loads that cause it to bend. When a load is applied perpendicular to the length of a beam (with two supports on each end), bending moments are induced in the beam. Normal Stress.

It is important in structural stiffness. Sometimes stiffness is more important than strength. Determined by acceptable limits of deflection of the side frame door mechanisms.

Local stiffness of floor is important –Stiffened by swages pressed into panels. The second moment of the area should be increased.

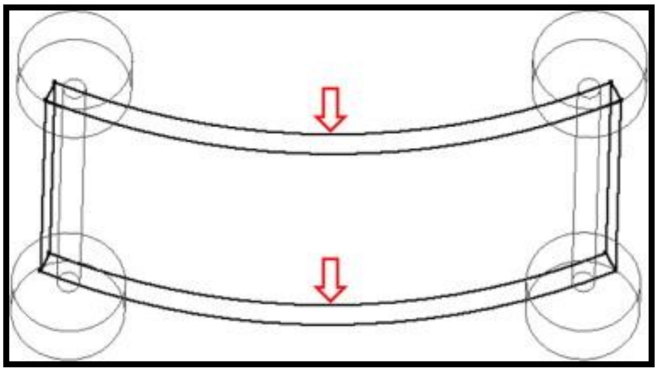


Fig. – II.B.8 (Bending Stiffness)

➢ *Torsional stiffness –*

Torsional stiffness is the characteristic property of a material that signifies how rigid is that material i.e, how much resistance it offers per degree change in its angle when twisted. More torsional stiffness/ rigidity, more load(torque) it can bear within allowable distortion.

Allowable torsion for an FSAE car: 1700 to 2000 N/m /deg. Measured over the wheelbase. Handling becomes very difficult when torsional stiffness is low. When torsion stiffness is low the structure move-up and down and/or whip. When parked on uneven ground doors fail to close.

Torsion stiffness is influenced by the nose. TS reduces by 40% when the nose is removed. Open top cars have poor torsional stiffness

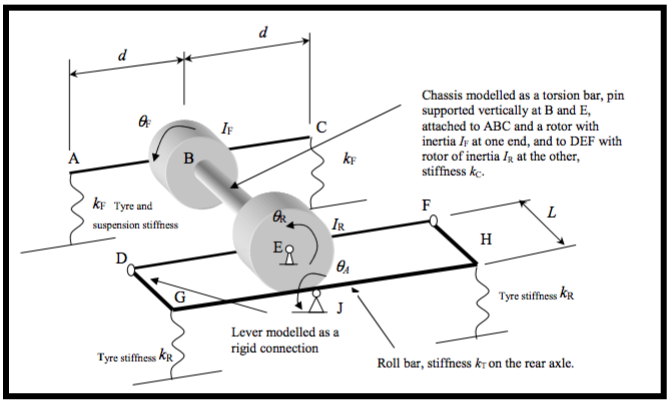


Fig. – II.B.9 (Torsional Stiffness)

C. Development of the mathematical model

The deflection that occurs at the end of the assembly has a component from each of the tubes. The stiffness, then, is also a function of the stiffness of each tube. If we use d to represent the flexibility of each tube then the flexibility of the system is just d(total). The stiffness is the inverse of the flexibility, which for the entire two-tubes system can be found from –

$$\frac{1}{K_{total}} = \frac{1}{K_1} + \frac{1}{K_2} \quad ; \qquad d\ total = d1 + d2$$

Which is the generic equation of stiffness for springs in series? If we had additional springs they would simply be taken into account by another term at the end of the equation. Another useful expression to model suspension effects will be to find the equivalent torsional stiffness for a liner spring at the end of a bar.

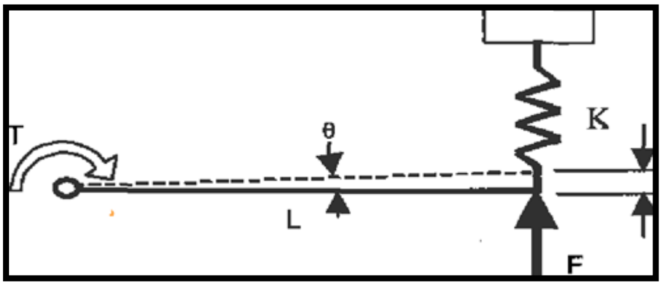


Fig. – II.C.1 (Liner to Torsion Spring)

The diagram depicts a bar, pinned at one end, and connected to a linear spring at the other. The spring is fixed to the ground at one end. From this information, we wish to find the equivalent torsional spring constant for the system. For this calculation, we need to find the torque the liner force is producing about the joint, and the angel the bar is moved through. While the diagram shows the force, F, and the displacement, d, we, know the spring constant, KL. Knowing either KL or F and d the other quantities can be calculated.

If we express KT, the torsional spring stiffness, in units of the in-lbs/radian then the equivalent liner spring stiffness, expressed in lbs/in and approximated using the small-angle approximation is :
KL = L2 . K L

It is also possible to convert from torsional to linear spring stiffness in a similar manner. Performing the analysis we would find the general equation is

$$KL \cong \frac{KT}{L2}$$

Now that we can model both torsion and linear springs in the same system, it is possible to build a model of all the complaint members in an automotive chassis. Depending on the desired complexity, different elements can be included or ignored in the model.

The simplest model we will consider is to calculate the chassis stiffness for a rigid frame and complaint springs. In this model, we assume the frame and suspension members are all infinitely stiff, and only the actual suspension springs themselves allow for any deflection.

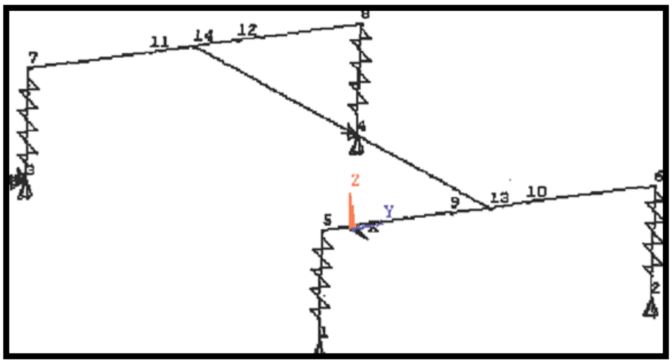


Fig. – II.C.2 (Vehicle Stick Model - Compliant Springs)

The load is applied at the front left wheel (positive x and y-direction). The other wheels are all constrained from motion in the vertical direction. We are neglecting forces and movement other than in the vertical direction, through the actual constraints are shown above.

If we draw a free-body diagram of the model and solve using the sum of forces and moments we can determine that the changes in forces at all four wheels are equal. The back right wheel force is of the same direction as the applied load, while the other two wheels have their forces acting in the opposite direction, or trying to hold the car down.

If we apply a force greater than the weight on those two wheels we would lift our car frame off the ground. For this example, and in real-world testing, we can assume that we have added weight to those corners to limit wheel lift.

ISSN No:-2456-2165

(The forces and deflections we are considering are all differences from the pre-existing forces/deflections that result from the car supporting its weight).

Since the force applied at each wheel is equal, call if F, the deflection of the spring at the wheel can be calculated if we know the spring constant, by the simple expression F=Kx. If we assume that each spring has the same rate, then the defections of each spring will be equal. (If the springs have different rates, front /rear, or even side-to-side, the method will still yield accurate results, but the relative motion of the nodes will change.

We constrained vertically three nodes, 1, 3, and 4. The four springs representing the suspension at the four corners of the car are all acting in series to resist the motion of the left wheel, reacting against some applied load can be found by the following expression:

1/K(total) = 1/K1 + 1/K2 + 1/K3 + 1/K4

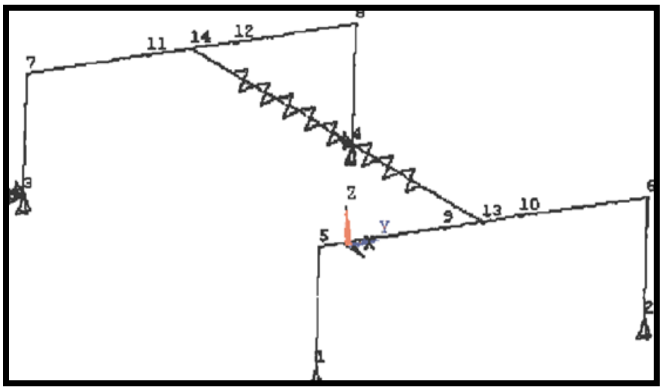


Fig – II.C.3 (Vehicle Stick Model – Complaint Frame)

In the above model a force applied at node 2, the contact patch, causes a torsional deflection in the frame. Since the other suspension element is fixed, no other deflections occur. All other nodes remain at their initial position. Node 6 moves through a vertical deflection corresponding to the equivalent liner rate of the frame torsion spring.

If the frame stiffness measure in ft-lbs/degree is equivalent to 100 lbs/in, then from a 100lb load node 2 deflects 1”. It should be noted that the angle of the bar connecting nodes 5 and 6 will change during this considering only vertical deflections at this time.

Now we can use the principle of superposition to show that considering deflections from both the translational suspension spring and the frame torsion spring produces deflection that is the sum of deflections occurring in each element.

1/K.total = 1/K1 + 1/K2 + 1/K3 + 1/K4 + 1/K5

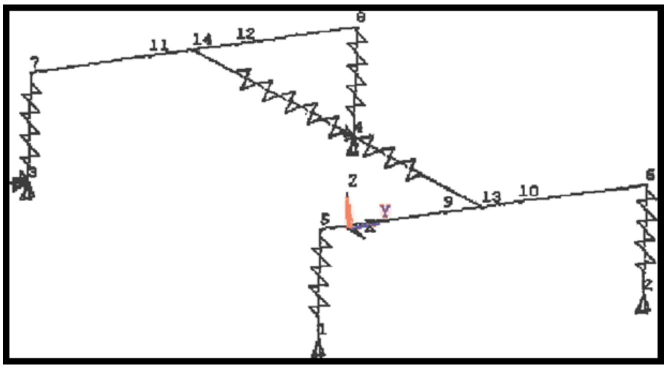


Fig. - II.C.4 (Vehide Stick Model — Compliant Springs and Frame)

Note that Ks is simply the spring constant of the torsion springs. To use this equation we must use consistent values of spring constants – either all translational spring value or all torsion spring values. We can convert back and forth by knowing the track and using the expression developed earlier in this section.

The suspension members, such as wishbones and rockers, also contribute compliance to the overall chassis system. This could be shown graphically as another torsion spring in series with the frame and can be included in our whole-car stiffness equation.

Also, note that we need to use the installed spring rate for each suspension spring rate divided by the motion ratio squared. The squared term arises because the motion ration affects both the force transmitted and the displacement the spring moves through. (Conservation of energy is one way to show the motion ratio must be squared.) A mathematical description of variable names is given below:

$$\frac{1}{K_{chassis}} = \frac{1}{K_{frame}} + \frac{1}{K_{suspension}} + \frac{r_1^{2}}{K_{spring\ 1}} + \frac{r_2^{2}}{K_{spring\ 2}} + \frac{r_3^{2}}{K_{spring\ 3}} + \frac{r_4^{2}}{K_{spring\ 4}}$$

Fig. – II.C.5 (Equivalent linear + torsional torsional stiffness)

The variable r in the above expression is the motion ratio of the corresponding spring. Again, the units of spring stiffness must be consistently measured in equivalent stiffness for a linear spring or rotary spring.

III. CAD DESIGN

A. Starting with 2D tire model

- Selecting sufficient tire data from Hoosier tire data book led us to select 18 inches Hoosier soft compound tires. Consequently led in selecting 10-inch aluminum-alloy wheels from Kizer (3 pieces). Therefore wheels and tires were datum features to the 2D Tire model.
- Approximate wheelbase – 1600mm, track front – 1200 & track rear – 1100mm are decided in the first iteration

considering paddle assembly & cockpit packaging in the front & engine packaging in the rear section.

- ➢ The next step is to define the important parameters of the model like Scrub radii – 60mm, KPI length – 173mm, KPI angle – 2 degrees, Static camber – negative 1 degree, FVSA – approx. 1600mm, Roll center - 30mm, etc considering Front model.
- ➢ A new sketch is started at the distance of the wheelbase on a new plane parallel to the previous one. This is the Rear tire model sketch. To eliminate complexities at the beginning similar parameters were used in the rear 2D tire model except for FVSA reduced to 1500mm, Static camber of 0 degrees to achieve maximum traction.
- ➢ Now, considering side view geometry, firstly we consider a caster angle of 2 degrees in the front to enhance steering effort and 0 degrees in the rear section since we have differential to control rear steering.
- ➢ The next step for side view geometry is specifying SVSA length for both the front and rear models to achieve desirable anti-dive and anti-squat percentages. 10 – 20 % anti-dive and 0 – 10% anti-squat is fine for FSAE cars.

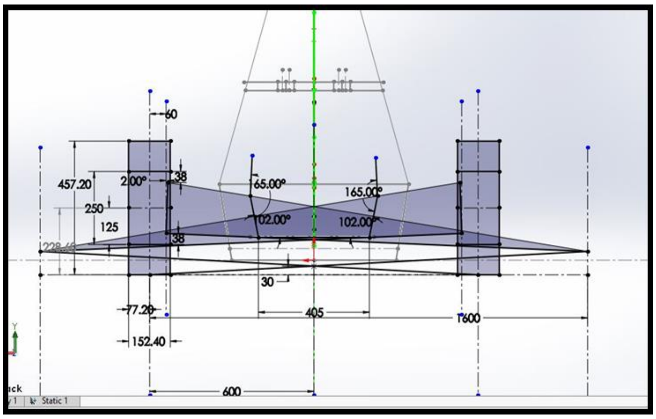


Fig. - III.A.1 (2D tire model)

B. 3D Driver sketch and ergonomics

- ➢ The next stage was to design a cockpit considering driver ergonomics, safety in the racing environment along with concepts of vehicle dynamics.
- ➢ It began with drawing a driver sketch considering the average of the tallest and shortest driver to assume the cockpit packaging space.
- ➢ After understanding each chassis rule precisely and considering all the constraints 3D sketches were made to develop a basic wireframe model leaving adequate tolerances.
- ➢ And lastly, once the model was developed the suspension co-ordinated were exported to Lotus Shark software to perform dynamic simulations.
- ➢ The refined data obtained from Lotus was used to alter the 3D sketch in Solid works for the next iteration.

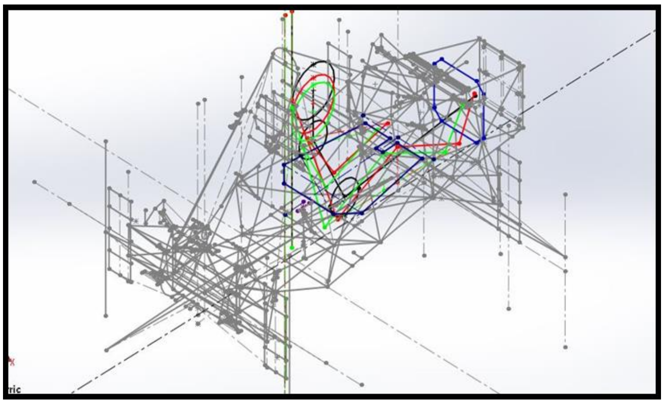


Fig. – III.B.1 (3D sketch for ergonomics)

C. Weldments feature

- ➢ Once the detailed sketch is complete we can use weldments to allot respective members considering the baseline tube rules. Every tube used in chassis is chosen carefully keeping in mind the baseline rules, market availability, and the strength it will impart considering the worst crash scenario.
- ➢ After every group that had been allocated weldments, the trim feature was used to avoid unnecessary interference among the intersections
- ➢ Some complex geometries cannot be made using weldments, hence we had to use other additive features as boss extrude, sweep extrude & revolve.
- ➢ After being completed with piping, several other mountings are added using additive features such as suspension pickup, harness mounts, and other miscellaneous mountings.

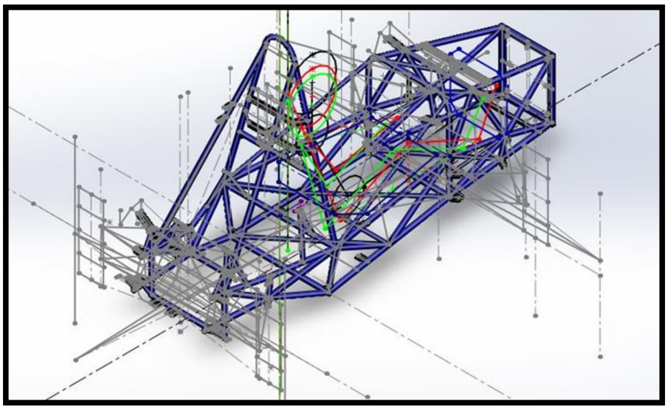


Fig. – III.C.1 (Weldments)

D. Assemble-Disassemble-Simulate-Optimize

- ➢ Once the frame was ready, we tried to assemble all the components. Especially the Steering system in front and drive train in the rear to make sufficient changes in front and rear geometry and improve packaging space.
- ➢ Lastly, after the detailed assembly, the Interference feature is used to run diagnostics against the assembly to check any kind of interference.
- ➢ Only after the CAD file was fully ready with zero interference detection we proceeded with production.

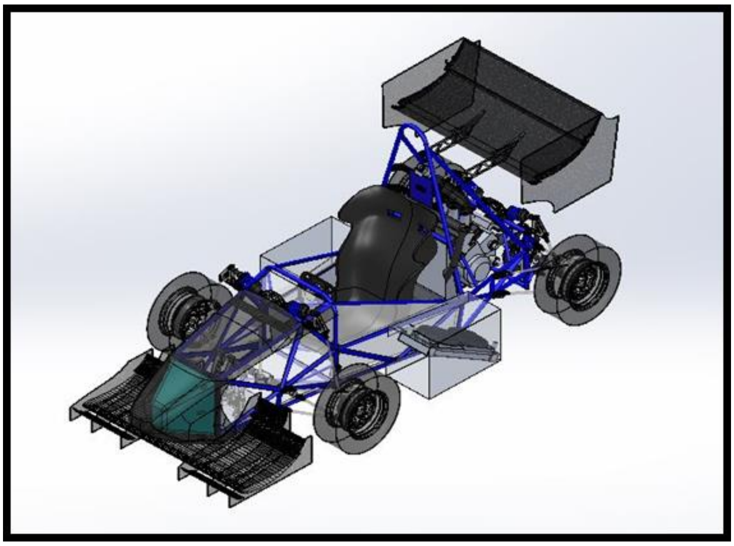
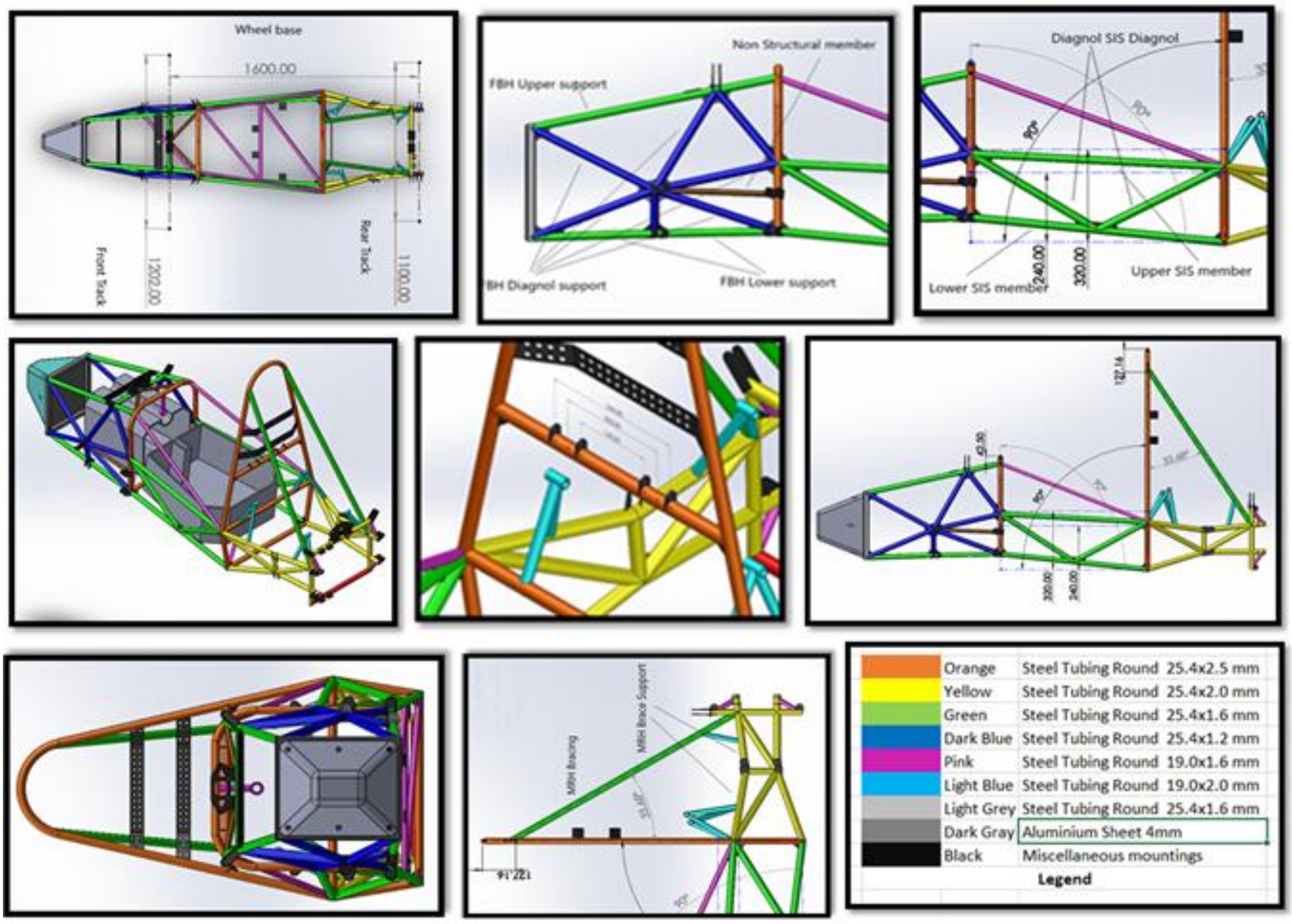


Fig. – III.D.1 (Final Assembly)

E. Chassis Layout (Figures of various sections)



ISSN No:-2456-2165

F. Basic design considerations

FORMULA BHARAT					DESIGN SPECIFICATIONS SHEET		
CAR NO & TEAM NAME		2018 - 58 Team Ojaswat					
Dimensions	**Units**						
Overall Dimensions	mm	Length:	2650	Width:	1375	Height:	1200
Wheelbase & Track	mm	Wheelbase:	1573	Front Track:	1194	Rear Track:	1143
Center of Gravity Design Height	mm	CG Height:	230	Confirmed Via:	N/A		
Mass without driver	kg	Front:	105	Rear:	130	Total:	235
Weight Distribution with 68kg driver		% Front:	45	% Left:	50		
Suspension Parameters	**Units**	**Front**			**Rear**		
Tire Size, Compound and Make		18.0x7.0-10, Hoosier			18.0x7.0-10, Hoosier		
Wheels (width, construction)		6.0x10.0			6.0x10.0		
Suspension Type		Double wishbone			Double Wishbone		
Suspension design travel	mm	Jounce (col D): Rebound (col E):	25	25	Jounce (col G): Rebound (col H):	25	25
Wheel rate (chassis to wheel center)	N/mm	16			16.7		
Roll rate (chassis to wheel center)	Nm/deg	177			163		
Sprung mass natural frequency	Hz	2.62			2.36		
Jounce Damping	% critical	70	at __ mm/sec:	10	70	at __ mm/sec:	10
Rebound Damping	% critical	70	at __ mm/sec:	10	70	at __ mm/sec:	10
Motion ratio	_:_:1	0.8:1	Type:	linear	0.8:1	Type:	linear
Ride Camber (Rate of Camber Change)	deg/m	28			30		
Roll Camber	deg/deg	0.66			0.7		
Static Toe (- out, + in)	deg	1			0		
Static camber	deg	-2			0		
Static camber adjustment method		shims, spherical bearings			shims, spherical bearings		
Anti dive / Anti Squat	%	45			25		
Roll center height above ground, static	mm	45			60		
Roll center position at 1g lateral acc	mm	Height (col D): Lateral (col E):	43	125	Height (col G): Lateral (col H):	59	81
Front Caster, Trail, and Scrub Radius		Caster (deg):	3	Kin Trail (mm):	12	Scrub Rad (mm)	61
Front Kingpin Axis		Inclination (deg):	5.88	Offset (mm):	25.4		
Static Ackermann	%	85-90	Adjustable?	no			
Suspension Adjustment Methods		by adjusting spherical bearing rod ends for castor and trail					
Steer Ratio, C-Factor, Steer Arm Length		Steer Ratio (x:1)	for inner wheel 3.16:1 outer wheel 3.67:1	c-factor (mm)	106.8	Steer Arm Length	58mm
Brake System / Hub & Axle	**Units**	**Front**			**Rear**		
Rotors		Fixed,diameter 150mm, thickness 4.5 mm,			Fixed,diameter 150mm, thickness 4.5 mm,		
Master Cylinder	mm	KIT tandem type, bore dia 19.05mm			N/A		
Calipers		bajaj pulsar 220F @ 2 piston, fixed			bajaj pulsar 220F @ 2 piston, fixed		
Brake Pad/Lining Material		asbestos with polymer metrix			asbestos with polymer metrix		
Force and Pressures @ 1g Deceleration		Front Pres. (bar):	44.1	Rear Pres. (bar):	44.1	Pedal Force (kN)	7
Upright Assembly		material Aluminium 7050 T6. transition fit of upright			material Aluminium 7050 T6. transition fit of upright		
Hub Bearings		2 opp facing single row tapered bearing			2 opp facing single row tapered bearing		
Axle type, size, and material		Fixed half axle, 24mm od, AISI 4340 steel			rotating axle, 24mm od, AISI 4340 steel		
Ergonomics	**Units**						
Driver Size Adjustments		N/A					
Seat (materials, padding/damping)		fiber reinforced plastic sheet with rexin, carboxyl (fire resisting) cover and sponge padding					
Steering Wheel (dia, construction)		Diamter (mm)	250	Construction	student built, ovel shape, aluminium plate		
Shift Actuator (type, location)		cable					
Clutch Actuator (type, location)		cable					
Instrumentation		brake light					
Optional: Driver Safety Systems?		N/A					

Fig. – III.F.1 (Basic design considerations 1)

Electrical	Units						
Power Management / Control		Three main power switches, i.e kill switch, Ignition switch and Emergency switch. Auto switch-off if					
Wiring / Loom / ECM mounting		OEM Wires, Wiring loom, circuit. Stranded copper wires with rubber insulation. Additional plastic t					
Battery / Charging System		Amaron Lead acid. 9AH, 12V, Vibration resistant battery. Charged by Engine built-in Alternator. Al					
Grounding		OEM provided Grounding on Enginebodyitself. Grounding to various equipments/devices(if applicable)					
Driver Assist Systems		OEM provided display on dashboard. Real Time acceleration display.					
Logging / Telemetry		NA					
Special Sensing Technology		NA					

Frame	Units						
Frame Construction		Tubular Steel Spaceframe					
Material		AISI 4130					
Joining method and material		MIG welding, ER 70S-6 0.8 mm wire spool					
Bare frame mass with brackets & paint	kg	Target:	40	Physical Test:	39		
Torsional stiffness	N-m/deg	Target:	5000	Simulated:	6400	Physical Test:	N/A
Torsional stiffness validation method		N/A					
Impact Attenuator configuration		Material used is IMPAXX 700 Energy Absorbing foam and it is Standard Impact Attenuator					
Impact Attenuator dimensions	mm	Width:	305	Height:	250	Depth:	355
Impact Attenuator energy capacity	kJ	Energy:	7450	Method:	Drop Test		

Powertrain	Units						
Manufacturer / Model		KTM RC 390					
Cylinders & Fuel			Cylinders:	1		Fuel Type:	Unleaded Gaso
Displacement & Compression			Displacement (cc):	373		Compression (_:1):	12.8:1
Bore & Stroke	mm		Bore:	89		Stroke:	60
Engine Output		Peak Power (kW)	32	PeakTorque (Nm)	36		
Design Speeds	rpm	Max Power:	8500	Max Torque:	6250	80% Torque:	1700
Induction (natural or forced, intercooled)		natural					
Throttle Body / Mechanism		Santro/Butterfly valve, throttle valve cable mechanism					
Fuel Injection System (manf'r, and type)		KTM RC 390 stock engine, port injection					
Fuel System Sensors (for fuel mapping)		It is a Float Type System					
Fuel Pressure	bar	3					
Injector location		135 mm before the intake valve and directed towards the intake gvalve opening					
Intake Plenum		Volume (cc):	2500		Runner length (mm):	380	
Exhaust Header Configuration		1.0-1.0		Effective Runner Length (mm):	310	Variation (mm):	0
Exhaust Header Diameters		Primary (mm):	35	Collector (mm):	35		
Ignition System		Contactless controlled fully electric ignition with digital ignitition adjustments					
Ignition Timing		22deg BTDC@3500rpm					
Oiling System (wet/dry sump, mods)		Wet sump					
Engine Lubricants / Friction Treatment		Pressure circulation lubrication with two rotary pump					
Coolant System and Radiator location		side mounted core 35*26cm radiator , 720 cfm fan mounted to radiator					
Fuel Tank Location, Type		aluminium 6063, inside cockpit behind driverseat				Capacity(L):	3.5
Muffler		free flow straight through absorbtive type muffler for sound reduction and minimum back pressure					
Other significant engine modifications		N/A					

Drivetrain	Units						
Drive Type		NO.51 Chain Sprocket					
Differential System		Chain Drive Open Differential					
Final Drive Ratio	_:1	3.4:1					
Vehicle Speed @ max power (design) rpm	kph	1st gear:	28.5	2nd gear:	40.9	3rd gear:	53.5
Vehicle Speed @ max power (design) rpm	kph	4th gear:	66.5	5th gear:	79.5	6th gear:	90.5
Half shaft size and material		24mm od , solid, AISI 4130					
Axle Joint type and grease used		Constant Velocity Joint and Axle Grease					

Aerodynamics (if applicable)	Units						
Type / Configuration		N/A					
Forces (at 80 kph, ρ= 1.162 kg/m^3)		Downforce (N):	N/A	% Front:	N/A	Drag (N):	N/A
Coefficients & Reference Area		Cl:	N/A	Reference Area (m^2):	N/A	Cd:	N/A
Noteable Features (active, etc)		N/A					

Other Information	Units	
Body Work (material, process)		fiber reinforced plastic, manual layup method used in adhesive
Optional Information		N/A

Fig. - III.F.2 (Basic design considerations 2

IV. SIMULATIONS - CAE

A. Calculations

➢ Front Impact –
u = 75 km/hr. = 20.833 m/s, v = 0 m/s, t = 0.5 s
using a = (v-u)/t, s = ut+1/2at^2, v^2-u^2 = 2as, F = ma
Front Impact force = 14583.33 N ~ 14500 N

➢ Rear Impact –
u = 50 km/hr. = 13.44 m/s, v = 0 m/s, t = 0.5 s
using a = (v-u)/t, s = ut+1/2at^2, v^2-u^2 = 2as, F = ma
Rear Impact force = 9408 N ~ 9500 N

➢ Side Impact –
u = 50 km/hr. = 13.44 m/s, v = 0 m/s, t = 0.5 s
using a = (v-u)/t, s = ut+1/2at^2, v^2-u^2 = 2as, F = ma

• Rolling over –
Normal reaction force = 3500 N vertical + Horizontal force =1500 N

➢ Torsional Rigidity –
It should be greater than in 1750 (lbs-ft/degree) for FSAE cars (by max. research papers).
$K = [4(F * d1/2) + 4(F * d2/2)]/\theta$
$= 2F (d1 + d2)/\theta$
K = Torsional Rigidity (lb*ft/deg),
F = Force (lb),
d1, d2 = Chassis width (ft),
θ = Chassis rotation (deg).

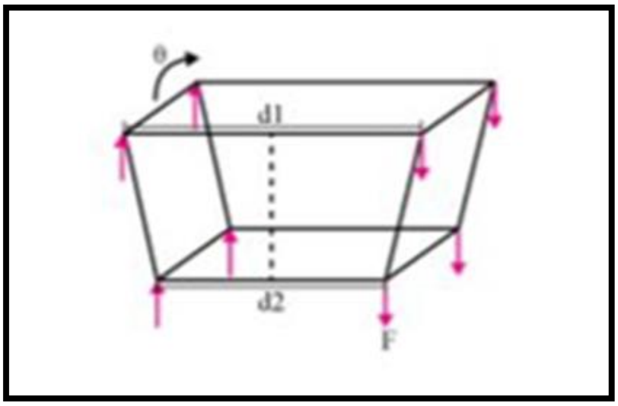


Fig. – IV.A.1 (Torsional Rigidity)

➢ Bending stiffness –
If a chassis satisfies criteria of torsional rigidity, then it has adequate bending stiffness.
$Kb = \Sigma F/\delta$
Kb = Torsional Rigidity (lb/in),
F = Force (lb),
δ = Vertical displacement (in)
In cockpit – F = drivers weight,
In rear section F =engines weight.

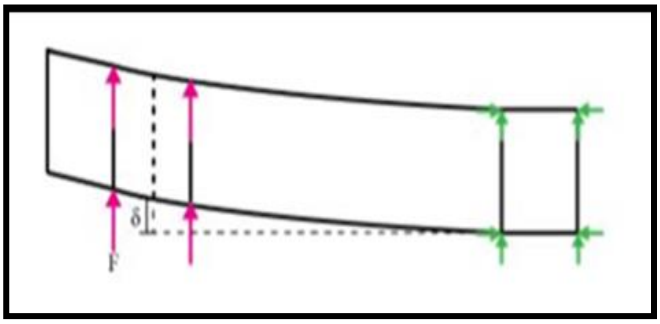


Fig. – IV.A.2 (Bending Stiffness)

➢ Longitudinal bending –
Front section – Pedal + Steering assembly weight
Middle section – Drivers weight (70 kg)
Rear section – Engine weight (65 kg)

➢ Lateral Bending –
Centrifugal force on CG at the fastest corner.
F = (m x v^2)/r = 5401 N ~ 5500

➢ Frequency Analysis –
Total number of frequencies – 5 to 10
The result – To check that the natural frequency of the chassis shouldn't resonate with engine frequency.

➢ Fatigue Analysis –
S/N cycle - 1000000 cycles
Point of application – Suspension hardpoints mountings.

➢ Harness bar simulation
Force – 3000N according to FB rulebook 2020

➢ Drop test-
Gravity – 9.8 m/s^2
Height of drop - 7m
Impact time - 0.5 seconds

➢ Fixtures – Note that in almost all the simulations 16 suspension pickup points are used as fixtures as the hardpoints are the only nodes that are indirectly in contact with Road (Loading conditions).

B. Development of mathematical Truss Spaceframe Model (Matlab R2020a)

➢ *What is a Truss element?*
A truss is a structure that consists of members organized into connected triangles so that the overall assembly behaves as a single object. Trusses are most commonly used in bridges, roofs, towers, and chassis.

The different types of trusses are as follows - Warren Truss, Pratt Truss, K Truss, Fink Truss, Gambrel Truss, Howe Truss. However, we have used simple truss to develop this model.

A simple truss is a planar truss which begins with a triangular element and can be expanded by adding two members and a joint. For these trusses, the number of members (M) and the number of joints (J) are related by the equation M = 2 J – 3.

➢ *Direct Stiffness Method.*

The image below illustrated a simple 2D truss model with 3 nodes, 3 truss elements, and 4 D.O.F. The 4x4 stiffness matrix represents 4 D.O.F of each element with the values of Elasticity Modulus, Area of Cross-section & Length of the truss. Note that the I.D matrix is 2 column matrix where the number of lines represents the number of trusses.

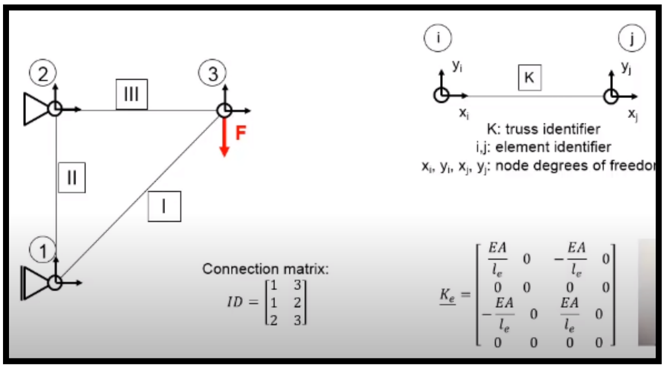


Fig. – IV.B.2.1 (Direct Stiffness Method 1)

To create a system matrix firstly we need to apply a rotational matrix o the truss that is at an angle. This means transferring local coordinates to global coordinates. The local element stiffness matrix is substituted to the Global stiffness matrix via the ID matrix. The image below clearly illustrates the procedure.

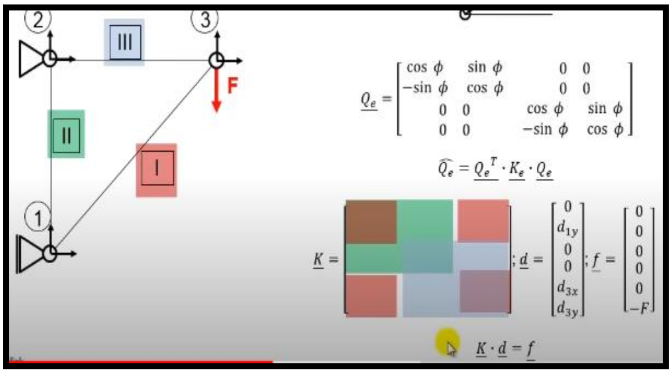


Fig. – IV.B.2.2 (Direct Stiffness Method 2)

➢ *Exporting Point Cloud data from CAD.*

To develop stiffness code/script in the command window, we need exact node coordinates from the CAD file. This can be done by simply exporting the points to MS excel.

The latest 3D sketch from the CAD file is pasted into a new part and saved in.IGES file format which is later converted to.TXT formatted and edited in MS excel.

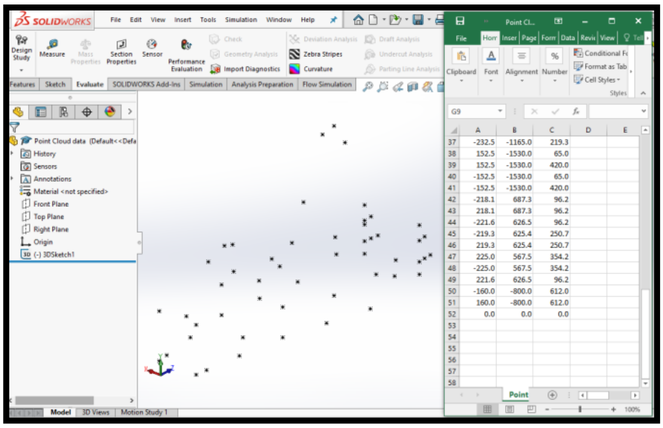


Fig. – IV.B.3 (Exporting Point Cloud data to Excel)

➢ *Developing Code to measure chassis stiffness.*

Solving the Truss framework model is the most basic form of simulation. It helps us understand the right approach behind applying fixtures, loading conditions, and meshes considering advance simulations in Ansys & Solidworks.

A simple approach using a Direct stiffness method can be applied to determine chassis stiffness. The basic procedure of coding involves specifying the number of node matrix (n), establishing assembly matrix relations between 2 nodes (m), and specifying Forces matrix (F). And ultimately solving the Global Stiffness matrix.

The syntax of the Matlab script is available in Matlab racing Lounge (file exchange) named Larry's toolbox which can be modified as per our requirmen.

```
% class design project example
%
% all of the members are quenched steel
% k = 2000 k-lb, Pmax = 1500 k-lb (A = 100 sq-inch)
%
% Referring to the notepad doc USM16(3)
clear
clc
close all
clear all
n = 54; m = 106;
LOADZ = 20000; LOADY = 1000/2; A = 1000;
joint = [
164.82,-1440.06,68.23; -164.82,-1440.06,68.23;
0.00,-982.50,541.50; 225.00,567.50,6.19;
-225.00,567.50,6.19; 225.00,567.50,96.19;
-225.00,567.50,96.19; 225.00,567.50,244.19;
-225.00,567.50,244.19; 250.39,307.50,215.00;
-250.39,307.50,215.00; 240.00,307.50,96.19;
-240.00 307.50 96.19; 0.00,0.00,1200.00;
93.70,0.00,1110.58; -93.70,0.00,1110.58;
250.98,0.00,580.00; -250.98,0.00,580.00;
300.00,0.00,0.00; -300.00,0.00,0.00;
345.85,0.00,260.00; -345.85,0.00,260.00;
275.00,-400.00,42.50; -275.00,-400.00,42.50;
250.00,-800.00,300.00; -250.00,-800.00,300.00;
250.00,-800.00,85.00; -250.00,-800.00,85.00;
```

```
227.08,-800.00,582.00; -227.08,-800.00,582.00;
208.44,-982.50,541.50; -208.44,-982.50,541.50;
202.50,-1165.00,78.10; 232.51,-1165.00,219.31;
-202.50,-1165.00,78.10; -232.51,-1165.00,219.31;
152.50,-1530.00,65.00; 152.50,-1530.00,420.00;
-152.50,-1530.00,65.00; -152.50,-1530.00,420.00;
-218.09,687.30,96.19; 218.09,687.30,96.19;
-221.60,626.49,96.19; -219.34,625.41,250.69;
219.34,625.41,250.69; 225.00,567.50,354.19;
-225.00,567.50,354.19; 221.60,626.49,96.19;
-160.00,-800.00,612.00; 160.00,-800.00,612.00;
];
assembly = [
16,18; 16,11; 17,13; 17,19; 18,15;
19,15; 20,22; 22,28; 28,24; 21,23;23,29; 25,29;
20,21; 28,29; 1,5; 2,5; 2,4; 3,6; 6,4; 3,1; 5,6;
7,8;9,10; 10,8; 1,7; 3,9; 2,8; 4,10; 2,7; 4,9;
11,12; 13,14; 14,12; 7,11; 9,13; 8,53; 53,12;
11,53; 7,53; 9,54; 13,54; 10,54; 54,14; 7,18;
9,19; 1,18; 3,19; 12,20; 16,22; 14,21; 23,17;
30,31; 32,33; 36,37; 33,50; 50,37; 32,52; 52,36;
34,36; 24,30; 25,32; 20,30; 22,30; 21,32; 23,32;
20,31; 21,33; 20,40; 21,41; 40,42; 42,41; 40,43;
41,44; 43,38; 44,39; 39,38; 45,31; 46,37; 46,33;
45,47; 45,31; 46,37; 46,47; 45,47; 30,48; 34,48;
36,48; 32,48; 33,52; 36,50; 31,51; 34,49 24,26;
25,26; 22,18; 23,19; 22,12; 23,14; 42,47;35,34;
35,37; 35,31; 34,30; 35,45; 22,38; 23,39
];
forceJ =  [
 3,1,1,1; 3,1,1,1; 3,1,1,1; 3,1,1,1; -1,0,0,0;
-1,0,0,0; -1,0,0,0; -1,0,0,0; -1,0,0,0; -1,0,0,0;
-1,0,0,0; -1,0,0,0; -1,0,0,0; -1,0,0,0; -1,0,0,0;
-1,0,0,0; -1,0,LOADZ,LOADZ; -1,0,0,0; -1,0,0,0;
-1,0,0,0; -1,0,0,0; -1,0,0,0; -1,0,0,0; -1,0,0,0;
-1,0,0,0; -1,0,0,0; -1,0,0,0; -1,0,0,0; -1,0,0,0;
-1,0,0,0; -1,0,0,0; -1,0,0,0; -1,0,0,0; 3,1,1,1;
3,1,1,1; 3,1,1,1; 3,1,1,1; -1,0,0,0; -1,0,0,0;
-1,0,0,0; -1,0,0,0; -1,0,0,0; -1,0,0,0; -1,0,0,0;
-1,0,0,0; -1,0,0,0; -1,0,0,0; -1,0,0,0; -1,0,0,0;
-1,0,0,0; -1,0,0,0; -1,0,0,0; -1,0,0,0 -1,0,0,0
];
for i = 1:m; stretch(i) = 2000*1000; end; stretch(1);
%
index = 1;
[Jforce,Mforce,Jdispl,Mdispl] =
truss3(n,m,joint,assembly,forceJ,stretch,index);
%
peak_klb = 18*A
maxMforce_klb = max(abs(Mforce/1000))
maxJdispl = max(abs(Jdispl*12));
maxDX_in = maxJdispl(2),maxDY_in =
maxJdispl(3),maxDZ_in = maxJdispl(4)
```

After running the code we can witness the results in the form of a graph depicting deflection. The image below illustrates the result.

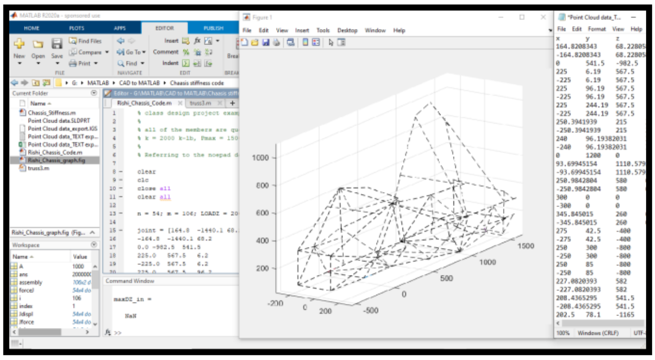


Fig. – IV.B.4 (Running codes + Results)

C. 2D – Static Simulation (Ansys 18.2)

After the development of the mathematical model, it is essential to simulate it with the most accurate solver available (Ansys 18.2) for greater accuracy. The steps to simulate the chassis model are listed below.

- Import the CAD file geometry of chassis from Solid works to Ansys using a file format of Para-solid(*x_t) to ensure that all the solid members of the chassis are imported and not just surfaces.
- The imported geometry is then edited in the space claim window. The editing involves extracting beams from solid members to develop a wireframe model for analysis. The wire model is used for analysis as it consumes less computation time and generates accurate results.
- Then using text(.txt) format suspension co-ordinates (z, x, y) are imported in space claim. Beams are generated using 'create' command from beams to complete the wireframe model.
- After inserting various components into 'New part' the chassis body, A-arms, and the upright wireframe model are 'Shared' separately in workbench.

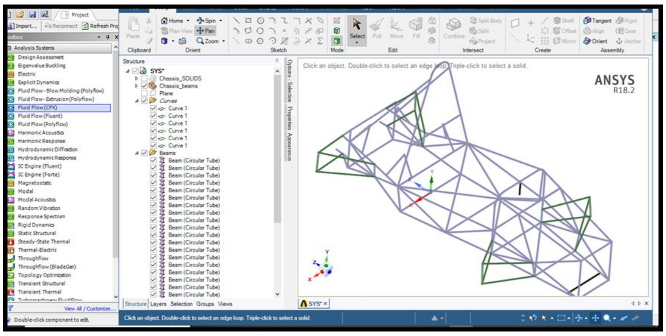


Fig. – IV.C.1 (Editing geometry in Space claim)

- The imported model is now ready to establish connections. Use the 'Name selection' feature to replicate similar kinds of joints. Joints between wishbones and chassis are spherical and between uprights and wishbone are revolute.
- Next, the springs are connected between the upright center and frame members with a stiffness (k = 32 N/m) to the model.
- And then after 'Body sizing" the model is ready to have meshed. Since it is a wireframe model the mesh size, quality, and element are kept default to avoid complexities and larger computation time.

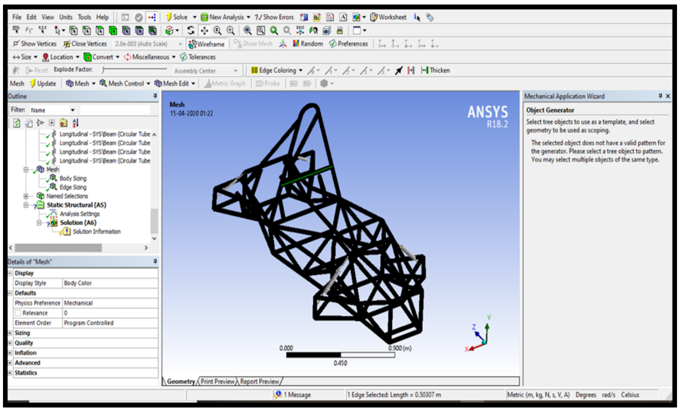


Fig. – IV.C.2 (Body sizing & Meshing)

- Boundary conditions for Torsional test – Scope (y coordinate = 0, which indicates the wheels are in contact with the ground) & Definition (Remote force of 1500N in +y direction on lower points of front uprights). Also Simply supported fixture on 4 nodes of the rear bulkhead.
- Boundary conditions for Cornering + Aerodynamic force test – Point mass of driver(70kg) and engine are added to the model in the respective position. Acceleration of 3g -x-direction, and gravity in –y-direction. And Fixed support as a fixture at every upright's center.
- Boundary conditions for Front Impact - Point mass of driver(70kg) and engine are added to the model in the respective position. And 15000N force on 4 nodes of the front bulkhead. Also Simply supported fixture on 4 nodes of the rear bulkhead.
- Both studies are solved and results are obtained results in terms of Total deformation, Direct stress, and maximum & minimum Combined stress.

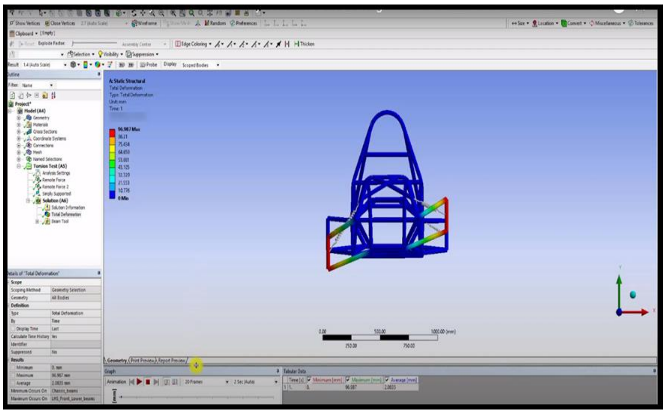


Fig. – IV.C.3 (Torsional Test Results)

- Torsional test – Total Deformation (min – 0.00mm, max – 0.89mm & avg. – 0.73mm), Direct Stress (min - -5.40 Mpa, max - 96.84 Mpa, avg – 1.8 Mpa).

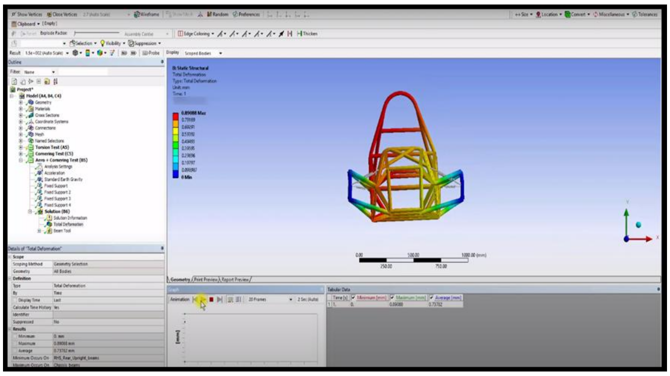


Fig. – IV.C.4 (Cornering + Aero Test Results)

- Cornering + Aerodynamic test – Total Deformation (min – 0.00mm, max – 0.89mm & avg. – 0.73mm), Direct Stress (min - -5.40 Mpa, max - 96.84 Mpa, avg – 1.8 Mpa).
- Front Impact test - Total Deformation (min – 0.00mm, max – 10.52mm & avg. – 4.70mm), Direct Stress (min - -33.80 Mpa, max – 315.34 Mpa, avg – 4.16Mpa).

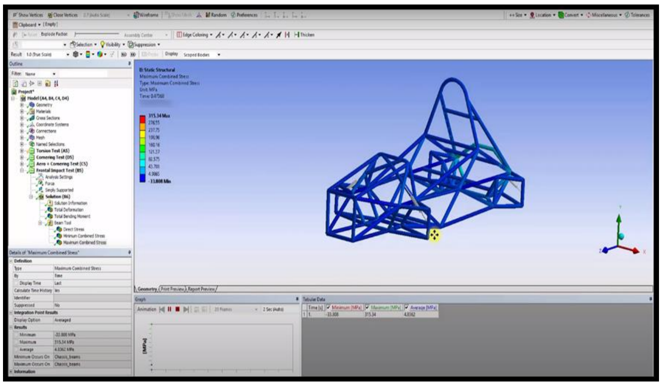


Fig. – IV.C.5 (Front Impact Test Results)

- Ansys 18.2 has one of the most accurate solvers but involves a lot of memory and processing time. Therefore, the most important simulations such as the torsional stiffness test. The torsional test is the most important static structural test because the chassis will always remain under torsional loads. Whereas chances of impact are very less in student formula competition.
- And so, the other simulations are carried out in Solid works which has lesser accurate solver but saves an adequate amount of time.

D. Solid works simulation e – report (detailed)

The report involves, details of simulations, ie: iterations, contact sets, matrices, mesh parameters, sensors, etc.

- *Description –*

This report is entirely based on the design & optimization of the FSAE (Formula racing vehicle) chassis system. The report includes the following simulations

- Front Impact
- Rear impact simulations
- Side impact simulations
- Rollover simulations

- Torsional stiffness (front) simulations
- Torsional stiffness (rear) simulations
- Bending stiffness.
- Longitudinal bending
- Lateral bending
- Miscellaneous simulations
- Drop test
- Fatigue Test
- Frequency analysis

➢ *Assumptions –*

Following are the assumptions considered during designing –

- The geometry is symmetrical
- The global friction coefficient is 0.05
- Ambient conditions are considered during simulations
- Several parameters are assumed or directly adopted from research papers.

➢ *Study Properties -*

- Analysis type - Static
- Mesh type - Mixed Mesh
- Thermal Effect: - On
- Thermal option -Include temperature loads
- Zero strain temperature-298 Kelvin
- Include fluid pressure effects from SOLIDWORKS Flow Simulation - Off
- Solver type - Automatic
- In-plane Effect - Off
- Soft Spring - On
- Inertial Relief - Off
- Incompatible bonding options - More accurate (slower)
- Large displacement - Off
- Compute free body forces -On
- Friction - On
- Friction Coefficient - 5.000000e-02
- Use Adaptive Method: - Off

➢ *Unit system - SI (MKS)*

- Length/Displacement - mm
- Temperature - Kelvin
- Angular velocity - Rad/sec
- Pressure/Stress - N/m^2

Fig. - IV.D.1 (3D mesh)

➢ *Mesh information –*

- Mesh type - Mixed Mesh
- Mesher Used - Curvature-based mesh
- Jacobian points - 16 Points
- Jacobian check for shell - On
- Maximum element size - 11.8191 mm.
- Minimum element size - 0.590954 mm
- Mesh Quality Plot - High
- Mesh information - Details
- Total Nodes - 69076
- Total Elements – 30268
- Time to complete mesh(hh;mm;ss):
- 00:00:01

➢ *Fixture Type Used –*

- Fixed geometry
- Application –

➢ *Load Type Used –*

- Force
- Torque

E. 3D – Static Simulation (Solid works 2018)

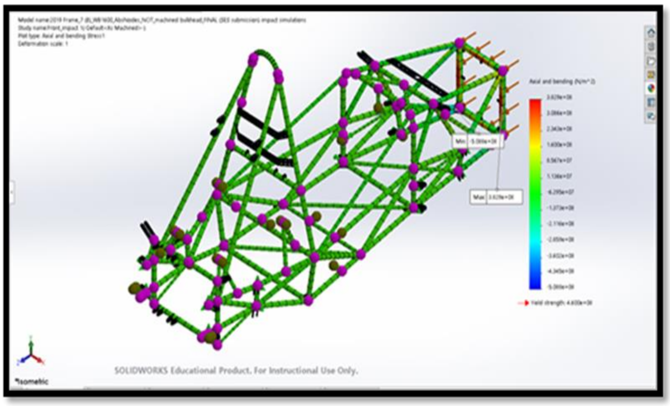


Fig. – IV.E.1a (Front Impact – stress)

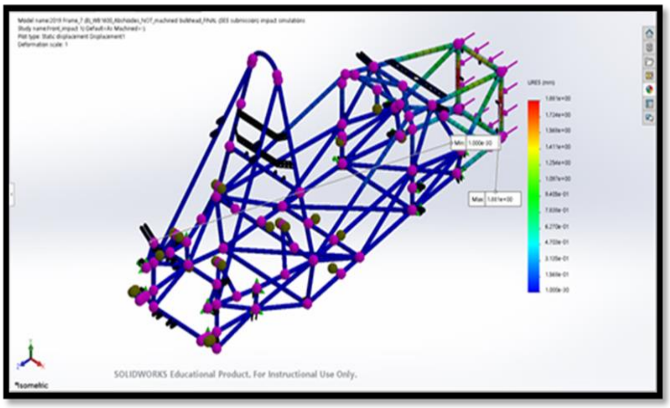


Fig. – IV.E.1b (Front-impact –displacement)

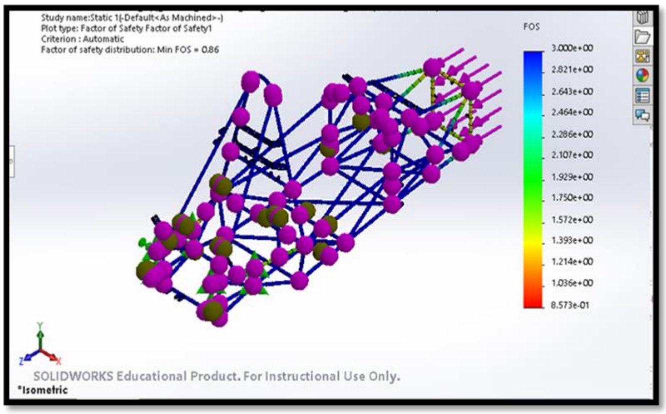


Fig. – IV.E.1c (Front-impact – FOS)

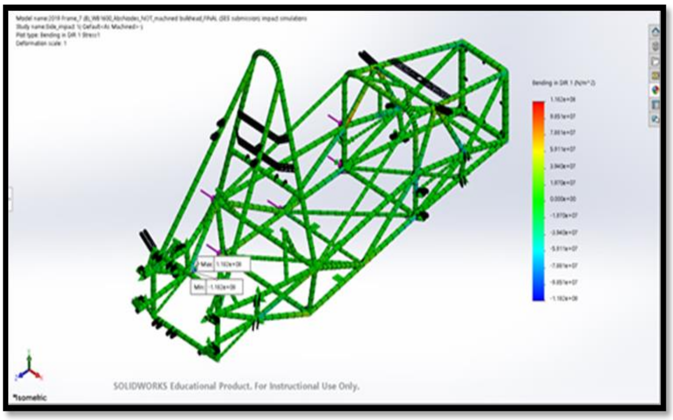


Fig - IV.E.3a – (Side impact 1 –stress)

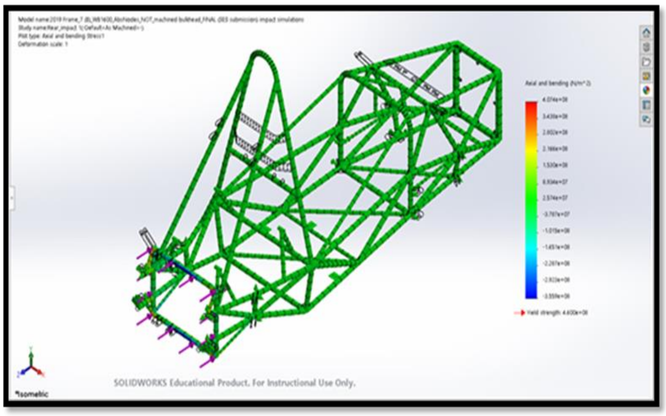


Fig. - IV.E.2a (Rear impact –stress)

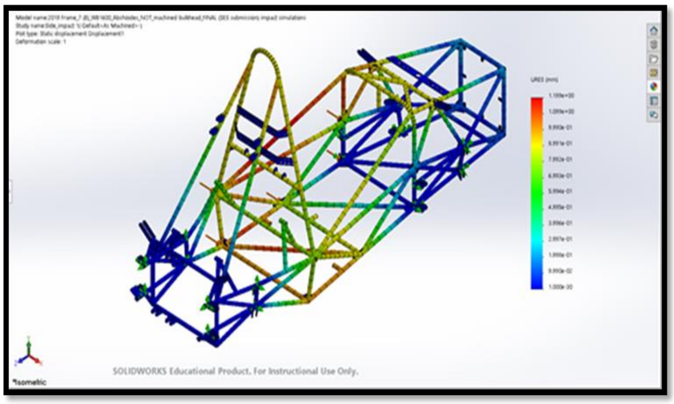


Fig - IV.D.3b (Side impact 1 – displacement)

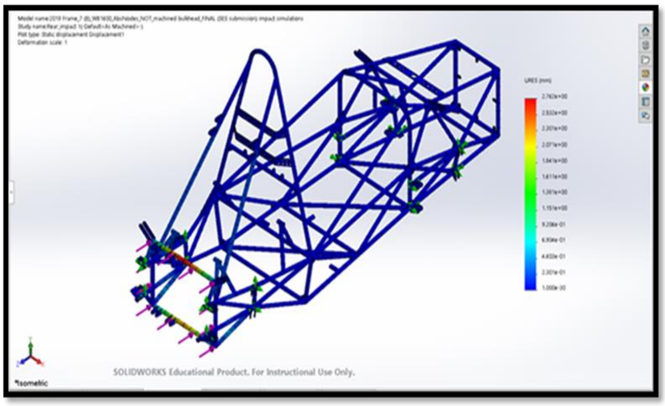


Fig - IV.E.2b (Rear impact – Displacement)

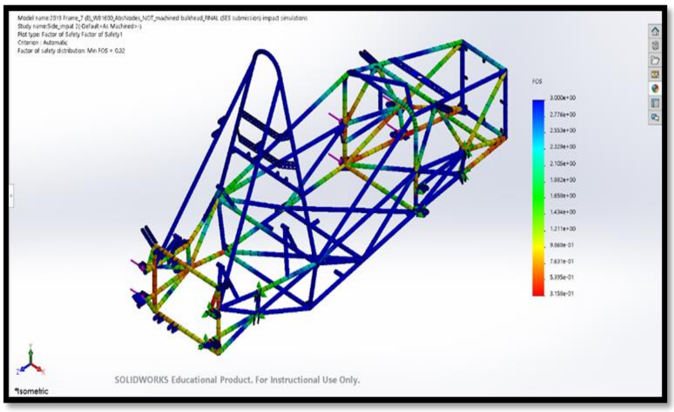


Fig - IV.E.3c (Side impact 1 – FOS)

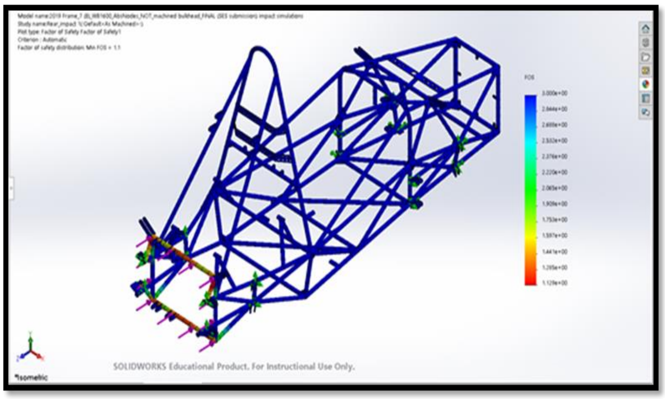


Fig. - IV.E.2c (Rear Impact – FOS)

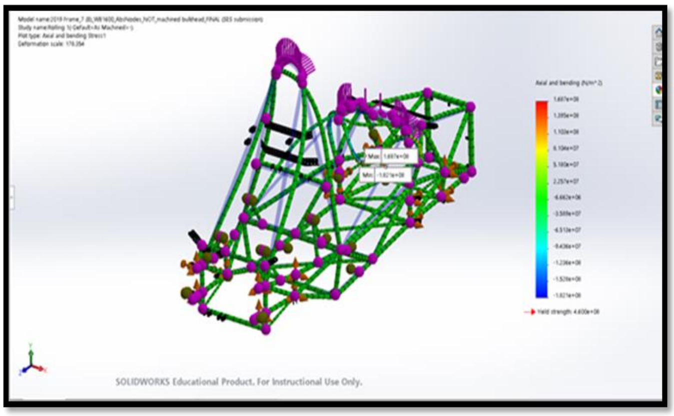


Fig - IV.E.4a (Rolling – stress)

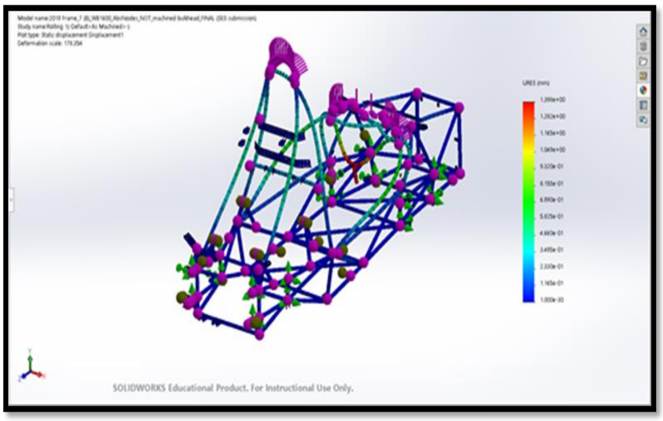


Fig - IV.E.4b (Rolling – displacement)

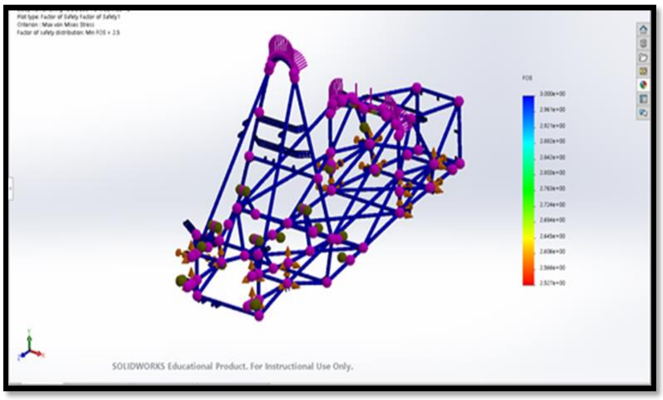


Fig - IV.E.4c (Rolling – FOS)

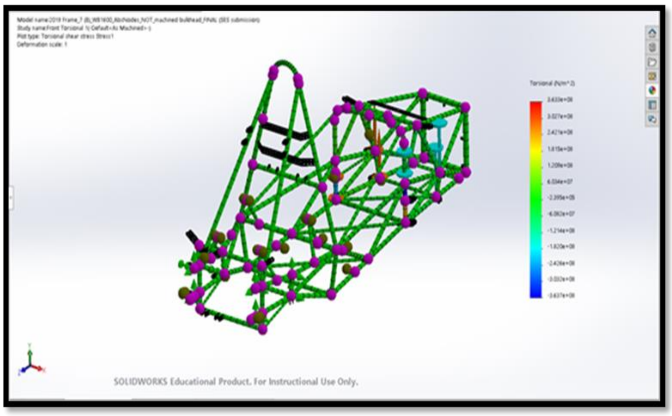


Fig - IV.E.5a (Front torsional – Stress)

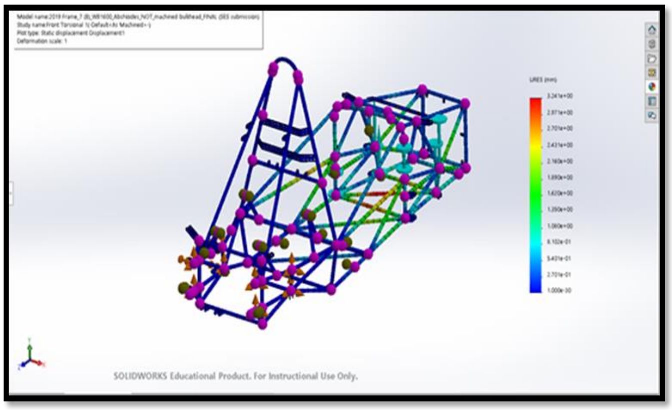


Fig - IV.E.5b (Front torsional – Displacement)

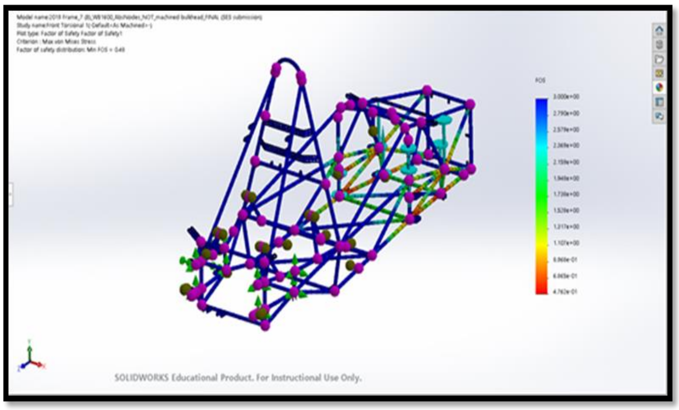


Fig - IV.E.5c (Front torsional – FOS)

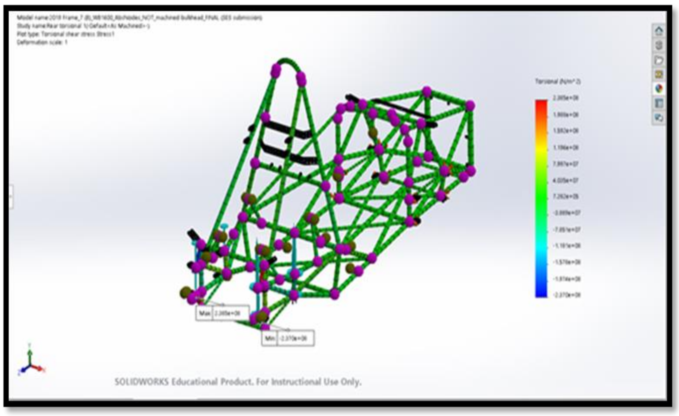


Fig - IV.D.6a (Rear torsional-stress)

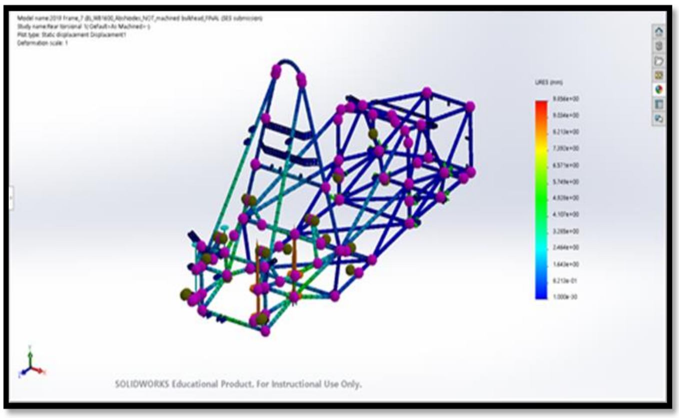


Fig - IV.E.6b (Rear torsional-displacement)

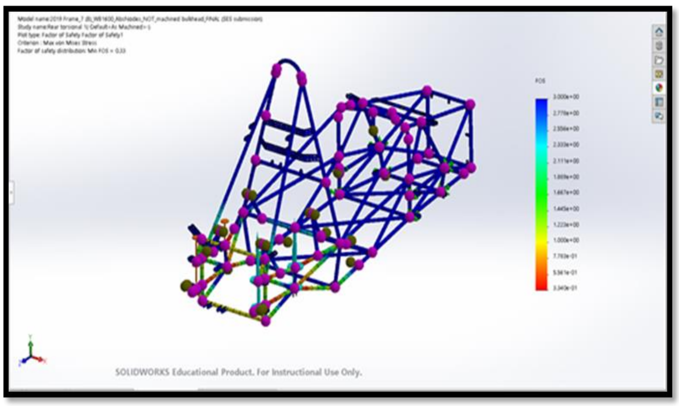


Fig - IV.E.6c (Rear torsional-FOS)

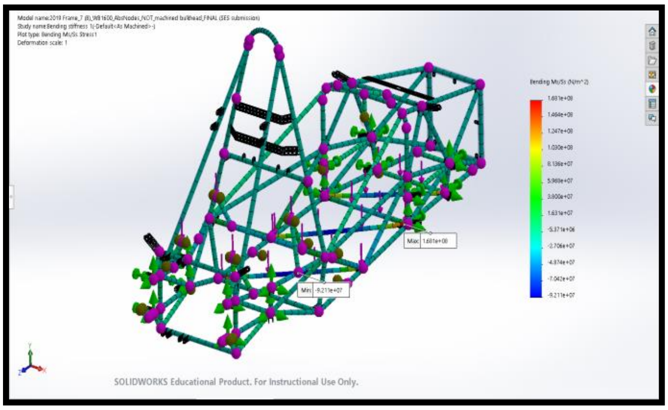


Fig - IV.E.7a (Bending stiffness –Stress)

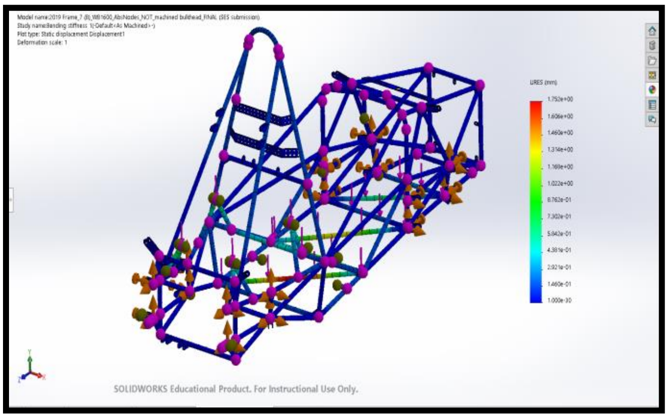


Fig - IV.E.7b (Bending stiffness –Displacement)

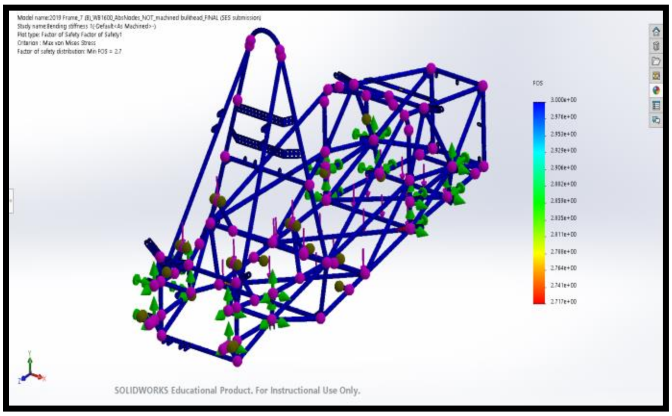


Fig - IV.E.7c (Bending stiffness –FOS)

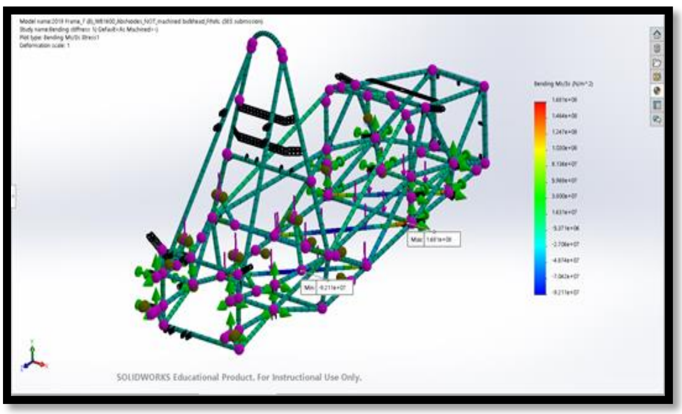


Fig - IV.E.8a – (Longitudinal bending - Stress)

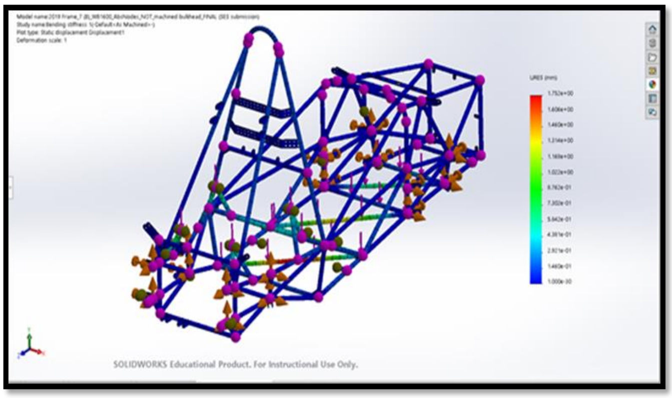


Fig - IV.E.8b– (Longitudinal bending – Displacement)

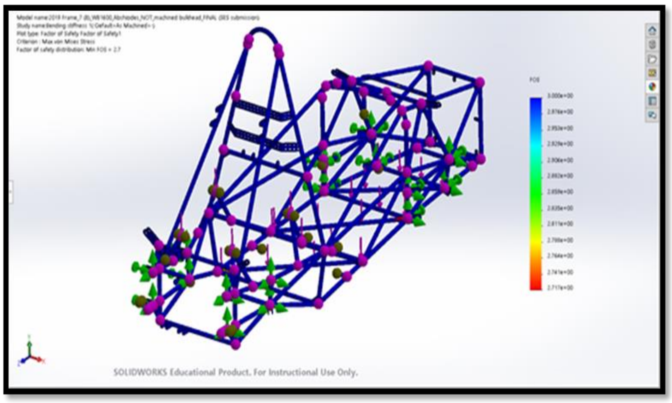


Fig - IV.D.8c - (Longitudinal bending – FOS)

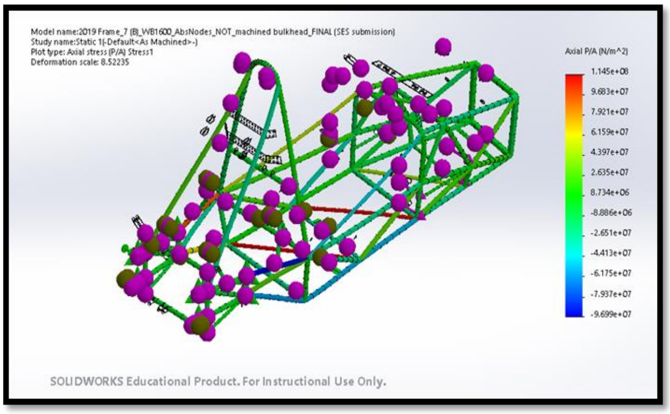


Fig – IV.E.9a – (Lateral bending – Stress)

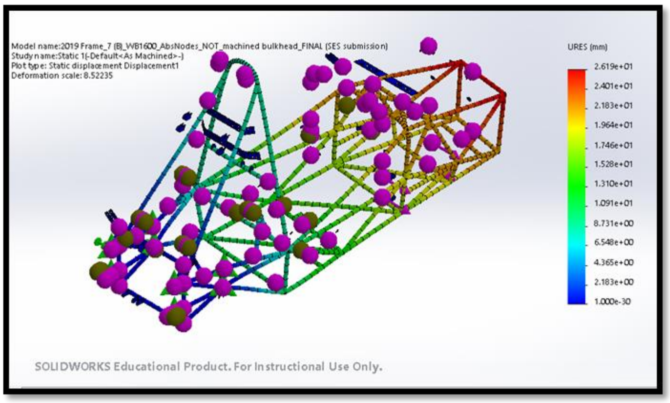


Fig – IV.E.9b – (Lateral bending - Displacement)

ISSN No:-2456-2165

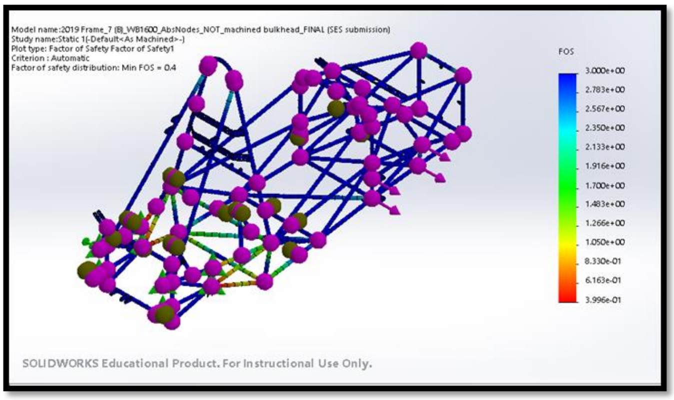


Fig – IV.E.9c – (Lateral bending - FOS)

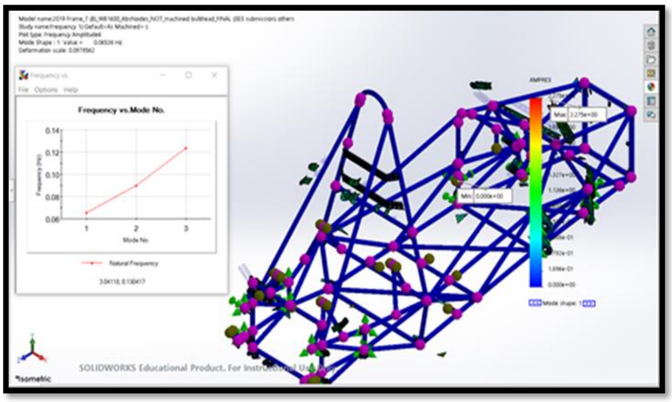


Fig - IV.E.10a (Frequency analysis – Fq vs Amplitude)

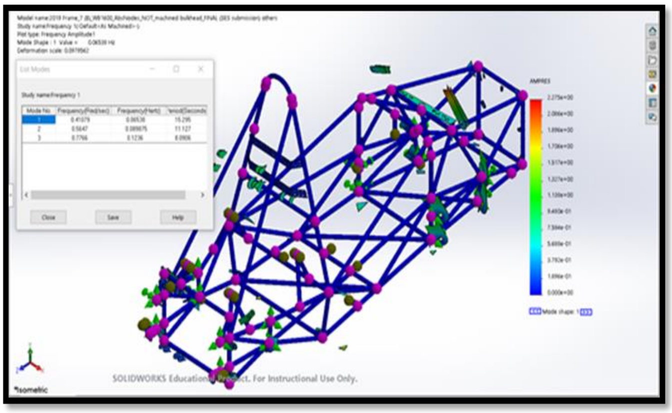


Fig - IV.E.10b (Frequency analysis – Resonant frequency)

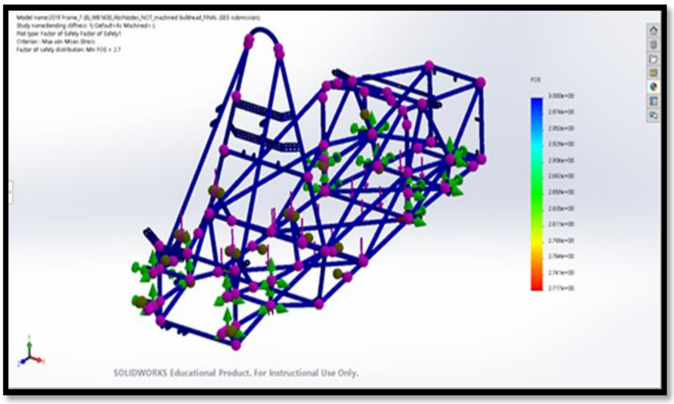


Fig – IV.E.10c (Fatigue analysis)

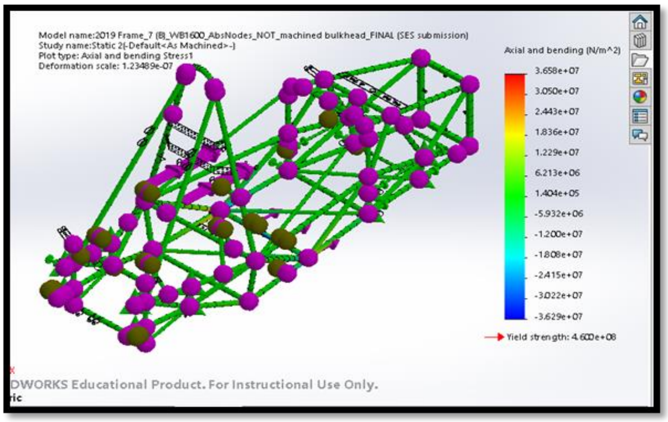


Fig - IV.E.11a (Harness simulation – Stress)

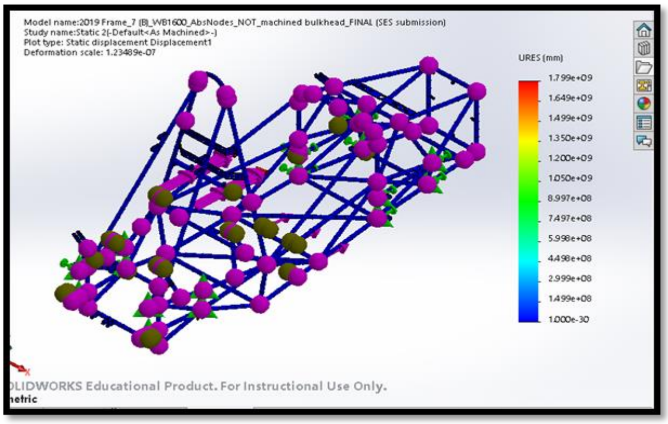


Fig – IV.D.11b (Harness simulation – Displacement)

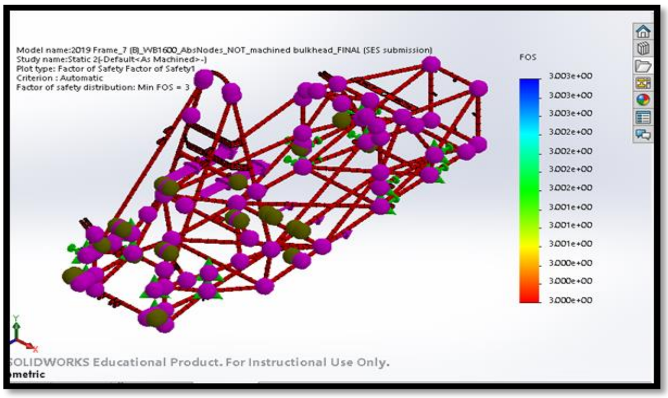


Fig – IV.E.11c (Harness simulation – FOS)

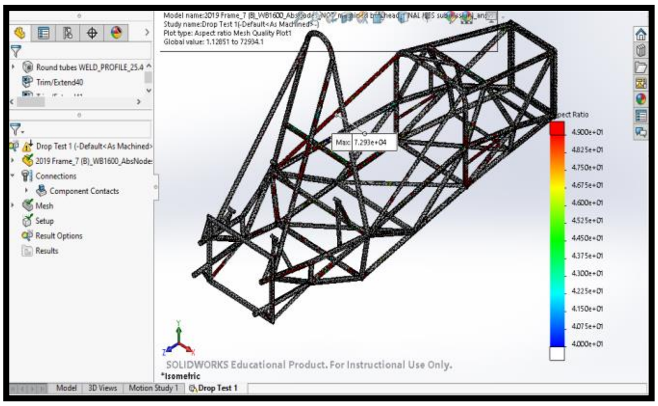


Fig – IV.E.12a (Drop Test) – Dynamic Simulation

ISSN No:-2456-2165

F. Tabulated Results of Static Simulations

Simulation type	Subcategory	Resultant value
Front Impact	Stress (N/mm^2) Y.S – 4.60e+08	Max- +3.82e+08 Min- -5.08e+08
	Displacement (mm)	Max- 1.88e+00 Min– 1.00e-30
	F.O.S -Working/Yield	Max- 3.00e+00 Min– 8.57e-01
Rear Impact	Stress (N/mm^2) Y.S - 4.60e+08	Max- +4.07e+08 Min- -3.55e+08
	Displacement (mm)	Max- 2.76e+00 Min– 1.00e-30
	F.O.S -Working/Yield	Max- 3.00e+00 Min– 1.12e+00
Side Impact	Stress (N/mm^2) Y.S - 4.60e+08	Max- +1.18e+08 Min- -1.18e+08
	Displacement (mm)	Max- 1.17e+01 Min- 1.00e-30
	F.O.S -Working/Yield	Max- 3.00e+00 Min- 3.15e-01
Rolling	Stress (N/mm^2) Y.S - 4.60e+08	Max- +1.68e+08 Min- -1.82e+08
	Displacement (mm)	Max- 1.39e+08 Min- 1.00e-30
	F.O.S -Working/Yield	Max- 3.00e+00 Min- 2.25e+00
Front Torsion	Stress (N/mm^2) Y.S - 4.60e+08	Max- +3.63e+08 Min- -3.63e+08
	Displacement (mm)	Max- 1.00e-30 Min- 3.24e+00
	F.O.S -Working/Yield	Max- 3.00e+00 Min- 4.76e-01
Rear Torsion	Stress (N/mm^2) Y.S - 4.60e+08	Max- +2.38e+08 Min- -2.37e+08
	Displacement (mm)	Max- 9.85e+00 Min – 1.00e-30
	F.O.S -Working/Yield	Max- 3.00e+00 Min– 3.34e-01
Bending stiffness	Stress (N/mm^2) Y.S - 4.60e+08	Max- +1.68e+08 Min- -9.21e+07
	Displacement (mm)	Max- 1.75e+00 Min - 1.00e+00
	F.O.S -Working/Yield	Max- 3.00e+00 Min– 2.71e+00
Longitudinal Bending	Stress (N/mm^2) Y.S - 4.60e+08	Max- +1.68e+08 Min - -9.21e+07
	Displacement (mm)	Max- 1.75e+00 Min- 1.00e+00
	F.O.S -Working/Yield	Max- 3.00e+00 Min– 2.71e+00
Lateral Bending	Stress (N/mm^2) Y.S - 4.60e+08	Max- +1.04e+08 Min - -9.69e+07
	Displacement (mm)	Max- 2.61e+00 Min– 1.00e-30
	F.O.S -Working/Yield	Max- 3.00e+00 Min– 3.99e-01
Harness	Stress (N/mm^2) Y.S - 4.60e+08	Max- +3.65e+07 Min- -3.64e+07
	Displacement (mm)	Max- 1.79e+09

ISSN No:-2456-2165

		Min- 1.00e-30
	F.O.S -Working/Yield	Max- 3.00e+00 Min- 2.00e+00
Frequency analysis	Rad/sec – 529.52 Rad/sec – 740.63	Seconds-0.011866 Seconds-0.008483
	Rad/sec – 796.42 Rad/sec – 852.66	Seconds-0.007889 Seconds-0.007368
Model analysis	Hertz - 84.276 Hertz - 117.87	Seconds-0.011866 Seconds-0.008483
	Hertz –126.75 Hertz - 135.71	Seconds-0.007889 Seconds-0.007368
Drop Test	Stress (N/mm^2) Y.S - 4.60e+08	Max- +3.82e+08 Min- -5.08e+08
	Displacement (mm)	Max- 1.88e+00 Min– 1.00e-30
	F.O.S -Working/Yield	Max- 3.00e+00 Min– 8.57e-01
Fatigue Analysis	Cycles – 200000 Testing fatigue in suspension pickup.	Safe design – (under Soderberg curve)

Table 2

G. Dynamic simulations in Matlab – R2020a

Static simulations are not enough considering the actual racing environment. Just for example Front Impact Static simulation in a real crash scenario is Rear Impact Dynamic simulation.

Elaborating the above statement as in front impact simulation we keep the chassis fix at the rear and apply force on the front bulkhead but under the dynamic crash condition, the front bulkhead comes to rest (fixture), and the momentum transfers from rear to front (force).

Therefore, we had to perform dynamic simulations to make sure that the chassis would sustain all the loads in real space and time. One of which was performed in Solidworks (Drop test). And another dynamic simulation was performed in Lotus and Matlab.

Initially, suspension dynamic simulations were performed in Lotus Shark & Raven software. Once the various suspension related graphs were satisfactory we proceeded with Stiffness dynamic simulations.

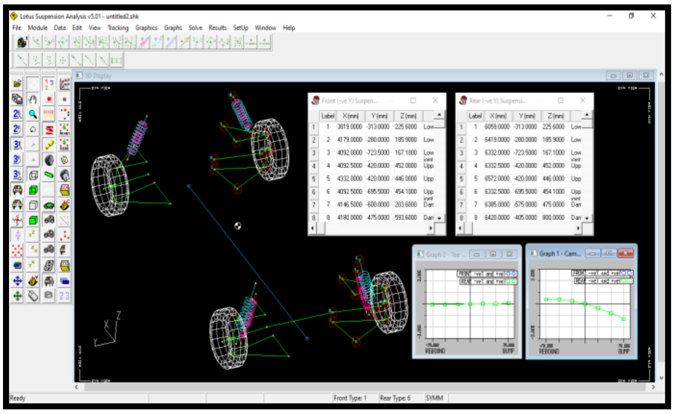
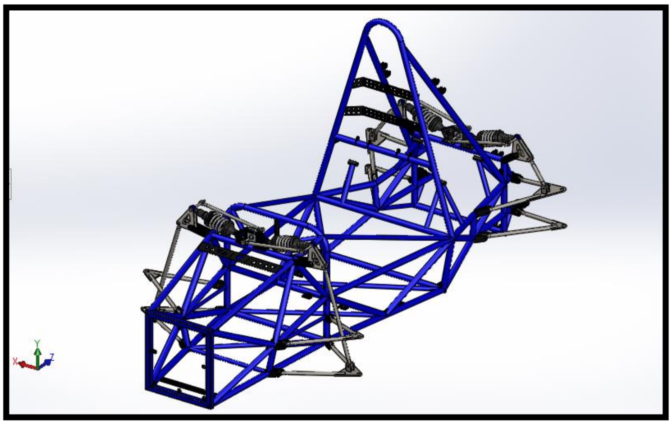


Fig. – IV.G.1 (Lotus suspension analysis)

To import the CAD chassis & suspension assembly to Matlab firstly Simscape multibody feature from Add-ins is used to convert '.sldasm' file to '.XML' file format so it can be imported in Matlab.



And to run the '.XML' file 'smlink_linksw' function is used in a command window followed by file name. On running the file we get the entire Mathematical model in Simulink. We then performed simulations on the model.

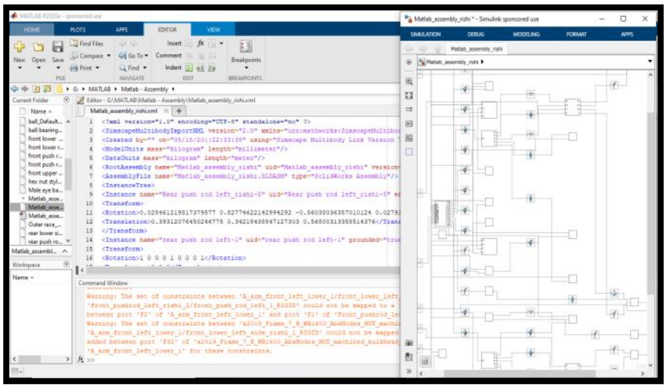


Fig. – IV.G.2 (Mathematical model – Simulink)

ISSN No:-2456-2165

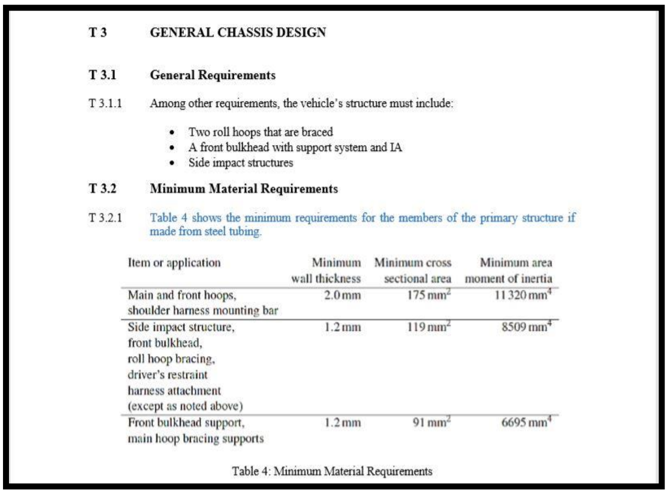
V. FABRICATION

A. *Material Constraints*

Formula Bharat and SAE supra have imposed certain restrictions on material strengths. Also, minimum wall thickness, tube diameter, cross-section area, and area moment of inertia are predefined in the rule book.

Therefore considering the baseline we used 25.45mm, 19.05mm & 14.00mm AISI 4130 Chromoly steel tubing in the entire structure. Datasheets attached in Appendix 1.

25.40mm x 2.50mm – Front & Main roll hoops
25.40mm x 2.00mm – Main hoop bracing support system
25.40mm x 1.65mm – Bulkheads, Side Impact Structures
25.40mm x 1.65mm – Roll hoop bracings, Harness bars
25.40mm x 1.20mm – Front bulkhead support system
19.05mm x 2.00mm – Torsion bars & Supports
14.00mm x 2.00mm – Nonstructural members.



T 3 GENERAL CHASSIS DESIGN

T 3.1 General Requirements

T 3.1.1 Among other requirements, the vehicle's structure must include:

- Two roll hoops that are braced
- A front bulkhead with support system and IA
- Side impact structures

T 3.2 Minimum Material Requirements

T 3.2.1 Table 4 shows the minimum requirements for the members of the primary structure if made from steel tubing.

Item or application	Minimum wall thickness	Minimum cross sectional area	Minimum area moment of inertia
Main and front hoops, shoulder harness mounting bar	2.0 mm	175 mm^2	11 320 mm^4
Side impact structure, front bulkhead, roll hoop bracing, driver's restraint harness attachment (except as noted above)	1.2 mm	119 mm^2	8509 mm^4
Front bulkhead support, main hoop bracing supports	1.2 mm	91 mm^2	6695 mm^4

Table 4: Minimum Material Requirements

Fig. – V.A.1 (Material Constraints)

B. *e - Drawings (1:1 Scale printouts)*

To attain maximum accuracy during production we printed 2D drawings of respective parts. This involved exporting the part file to Solidworks drawing templet and printing on a scale of 1:1.

The fabrication procedure began with production on A-arms. This is because the chassis should always be manufactured according to 16 suspension hardpoints and not the other way round to maintain suspension geometry.

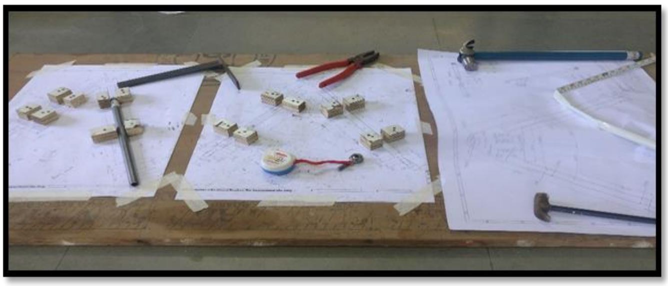


Fig. – V.B.1 (Drawing Prints – A-arms fixture)

C. *Roll hoops production*

The next task was the production of Front, Rear roll hoops, and Front, Rear bulkheads. It is because these 4 components were designed to be perpendicular to the fixture table whereas the other tubing was scattered in 3D space.

To achieve maximum accuracy, the roll hoops were sent for CNC bending and later analyzed in a fixture to remove residual stresses by giving heat treatment. Examples of the roll hoop sketches are attached in Appendix 3.

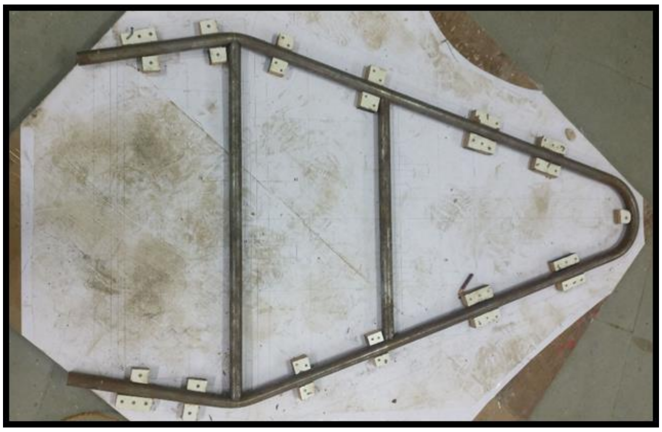


Fig. – V.C.1 (Main Roll hoop Fixture)

D. *Base fixture – laser cut Jigs*

The base fixtures and Jigs had to be as accurate as possible to maintain weight balance and suspension geometry according to the CAD design. Therefore we decided to go to metallic Jigs instead of wooden. The example of the drawings is attached in Appendix 3.

The top view of the chassis was printed on the A0 size sheet and stick on the fixture table to attain maximum accuracy. And the jigs of the base of the chassis were sent for laser cutting to achieve maximum accuracy in Z-axis and later welded to the fixture table, following the sketch outlines.

Fig. – V.D.1 (Laser-cut Jigs on Metallic fixture table)

E. Profile cutting and grinding

To achieve maximum accuracy during profiling individual tubes were imported from the solid works part file by breaking the reference into the new part by using Insert into the new part feature. Then they exported to sheet metal and flattened using insert bent feature.

The drawing of the ends of the pipe was printed and stuck tubes to obtain the most ideal length and profile. The example is illustrated below and an example of a profile cut is given in Appendix 3.

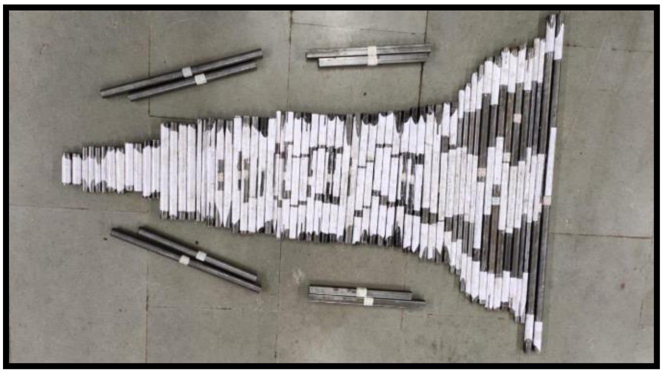


Fig. – V.E.1 (Profile cutting drawings)

F. Welding procedures

Firstly, the base was welded with the jigs exactly perpendicular to the base using arc welding to save time. Later, the base of the chassis was placed in the jigs and tacked to avoid them from lifting due to residual stresses generated during full welding.

Tig welding was used to weld the frame to abolish flux and maintain aesthetics. The entire chassis was welded in house. The welding filler data sheets are attached in Appendix 2.

Fig. – V.F.1 (In house Tig & Arc Welding)

G. Defining Hardpoints Locations

To determine the exact location of the 16 hardpoints, 4 prototype uprights were created using exact dimensions from metallic sheets. The A-arms were used to project the points on node points. On these points, the suspension mountings were welded with great precision.

Later all the other mountings were also welded according to CAD with great precision.

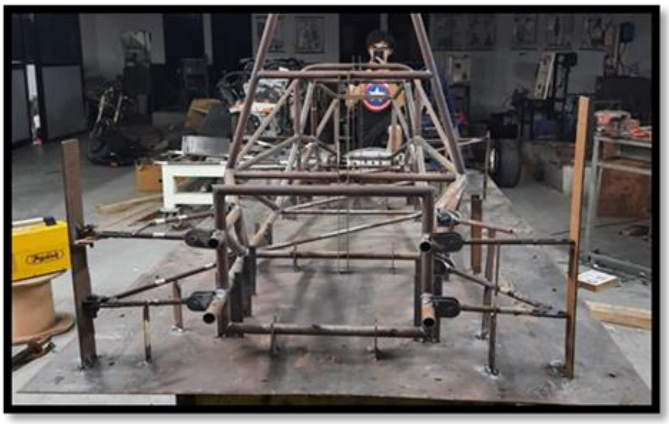


Fig. – V.G.1 (Prototype uprights as jigs)

H. Final set up

One all the tubes and mountings were welded the final set up would look like something illustrated in the figure below.

Later, all the paper was scraped off the tubes and the frame was lifted off from the fixture table by grinding off the tacks that were made to prevent deflection due to residual stresses.

Further, the frame was taken for validation testing like Comparing C.O.G with CAD file, destructive testing on the torsional rig.

And lastly for power coating to bring of aesthetical looks from the rusty frame.

Fig. – V.H.1 (Final Set up on fixture table)

ISSN No:-2456-2165

VI. VALIDATION TESTING

A. Comparison with the 2016 model

Chassis 2016	Chassis 2019
The frame was designed for a 13inch steel wheel.	This frame is designed for 10-inch aluminum-alloy wheels.
The overall weight of the chassis was 37 kg excluding all the mountings and including all the mountings it was around 50 kg.	The overall weight with mountings is just 39 kg including all the mountings.
The torsion bar was used in the rear section to add torsional stiffness in the rear section.	The torsion bar is eliminated to reduce weight and it served no requirement as the engine itself sustains torsional loads.
The front bulkhead involved a cross member sine they were using smaller Impact attenuators.	We eliminated the member s our car complied with the rule.
The suspension hardpoints were not node to node triangulated.	The suspension hardpoints are perfectly triangulated
The chassis has a low weight to strength ratio.	This model has much higher stiffness and weight to strength ratio.
They had used wooden jigs and fixtures for the production of chassis 2016 that resulted in lesser accuracy.	This model is developed with metal jigs and fixture with laser cutting to obtain maximum accuracy.
The 2016 chassis model much deviated from baseline dimensions hence their car was too heavy.	2019 is very close to the baseline and optimized in the best way possible to reduce weight and increase performance.
The overall weight of the 2016 car is 307 kg.	The overall weight of the 2019 model will be 2650-260 kg.
The C.G of 2016 model was not balanced in the XYZ axis.	The C.G of 2019 model is well balanced in the XYZ axis.

Table 3

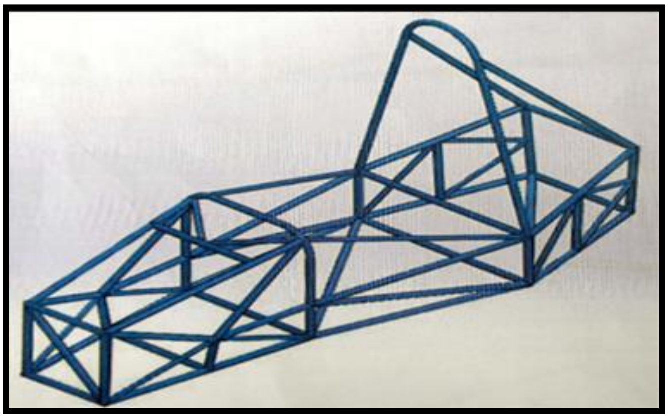


Fig. – VI.A.1 (2016 CAD model)

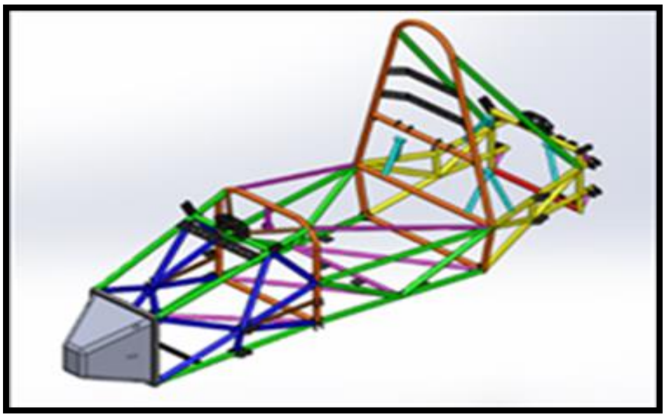


Fig. – VI.A.2 (2019 CAD model)

B. Comparing the Centre of Gravity of CAD file and Prototype

➢ *Total vehicle Horizontal (x & y) location of C.G from the figure VI.B.1.*

(Note that the figure below denotes a method to determine vehicle's C.G but it can also be used to determine chassis C.G only by replacing 4 wheels to 4 extreme lowest hardpoints).

- W – total weight of chassis
- l = Wheelbase (1.60m)
- d = (Tf – Tr)/2
- Tf = Track front (1.20m)
- Tr = Track rear (1.10m)
- X-X axis = Centre line of chassis (x direction)
- X1-X1 axis = Centerline of rear wheel.

✓ Taking the weight of chassis using 4 weighing machines placed under 4 extreme points (suspension hard points front & rear).

- W1 + W2 + W3 + W4 = W (total weight of chassis)
- 12.20 + 11.80 + 10.30 + 10.70 = 45 kg.

✓ Taking moment about Rear axle. (C.G in X-axis is)

- b = (Wf x l)/W
- b = (24.00 x 1.60)/45
- b = 0.8533 m (Distance of C.G from rear track)
- a = l –b
- a = 1.60 – 0.8533
- a = 0.7466 m (Distance of C.G from front track)

- ✓ Now, taking moment about the X1-X1 axis (parallel to the centerline of the car (chassis) through the center of left rear tires).
- $d = (Tf – Tr)/2$
- $d = (1.20 – 1.10)/2$
- $d = 0.05$ m
- $y' = \{W2 \times (Tf – d)\}/W – \{W1 \times (d)\}/W + \{W4 \times (Tr)\}/W$
- $y' = \{12.30 \times (1.20 – 0.05)\}/45 – \{11.70 \times (0.05)\}/45 + \{10.30 \times (1.10)\}/45$
- $y' = 0.552$
- ✓ Now to find y" (shift in m from C.G) we have to use the formula [$y'' = y' –(Tr/2)$] to give lateral shift of C.G from X-axis (centerline).
- $y'' = y' – (Tr/2)$ or
- $y'' = \{W2 \times (Tf – d)\}/W – \{W1 \times (d)\}/W + \{W4 \times (Tr)\}/W – Tr/2$
- $y'' = 0.552 – 1.10/2$
- $y'' = -0.002$ m (shift in C.G y-axis)

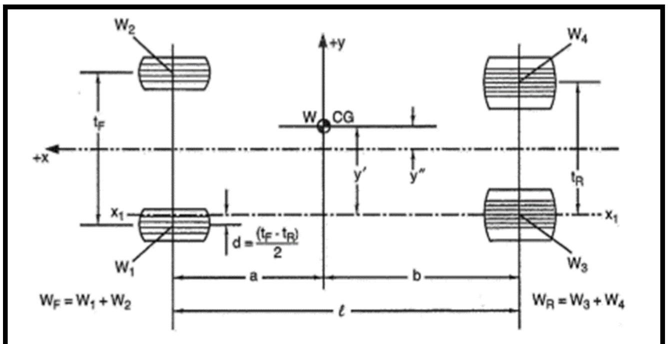


Figure 18.1 Horizontal location of total vehicle center of gravity.

Fig. – VI.B.1 (Horizontal CG of chassis)

(Positive & Negative values of y" describe the shift of C.G in the left or right direction from centerline).

- ➢ *Total vehicle Vertical Location of C.G from figure VI.B.2.*

(Note that the figure below denotes a method to determine vehicle's C.G but it can also be used to determine chassis C.G only by replacing 4 wheels to 4 extreme lowest hardpoints).

- ǿ = 11◦ (angle of the inclined plane)
- W = Total weight of chassis in kg.
- Wf = weight of front axle
- b = horizontal distance from rear axle
- l = wheelbase (1.60m)
- T.l.f = Loaded thickness of front axle (height from ground to suspension pickup centre in front).
- T.l.r = Loaded thickness of rear axle (height from ground to suspension pickup centre in rear).
- ✓ Taking moment about point O & the trigonometric step functions are as follows.
- $L1 = l. \times \cos ǿ$
- $b1= (Wf/W) \times (l \times \cos ǿ)$
- $c = \{(Wf/W) \times l\}–b$
- $Wf \times l = W \times b1$
- ✓ Note that (h1) is the height of C. G above the line connecting front & rear pickup centres, which is at a height of (T.lf).
- $(b1)/ (b + c) = \cos ǿ$
- $(c/h1) = \tan ǿ$
- $h1 = \{(Wf \times l) – W \times b\}/ W \times \tan ǿ$
- $h = Tl + h1$
- ✓ Now if (t) is different for front & rear (ie; both hard point centres have different heights from the ground) then C.G is found by the following formula –
- $T.l.cg = T.l.f \times (b/l) + T.l.r \times (a/l)$
- $h = T.l.cg + h1$
- $h1 = \{25.50(1.6) – 45(0.8533)\}/ 45 \times (\tan 11◦)$
- $h1 = (40.80 – 38.39)/ (45 \times 0.194)$
- $h1 = 0.276$ m
- $T.l.cg = (0.276) \times (0.8533/1.60) + (0.043) \times (0.7466/1.60)$
- $h = 0.276 + 0.102$
- **h = 0.378 m (C.G in z-axis is)**
- ✓ Note that the above method is purely used to calculate the C.G of the vehicle with wheels so it won't give accurate results while measuring the C.G of chassis. But the study gives us the rough idea of the prototype chassis.

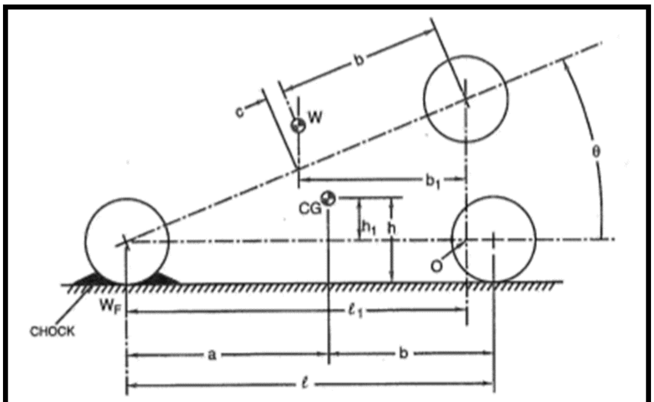


Figure 18.2 Vertical location of the center of gravity.

Fig. – VI.B.2 (Vertical CG of chassis)

- In the figures below is the illustration of the comparison between CAD file chassis and the prototype. MS Paint has been used to illustrate the rough position of CG in the prototype model.
- ✓ The image below shows the centre of mass (purple) of the chassis. The position of the centre of mass (C.O.M) and centre of gravity (C.O.G) are the same in software but changes in real space and time.
- ✓ The C.O.G is measured from (blue coordinate system symbol) origin in software. This is X = -2.30mm, Y= 292.81mm & Z=-343.26mm which is highlighted by purple colour Coordinate system symbol.

ISSN No:-2456-2165

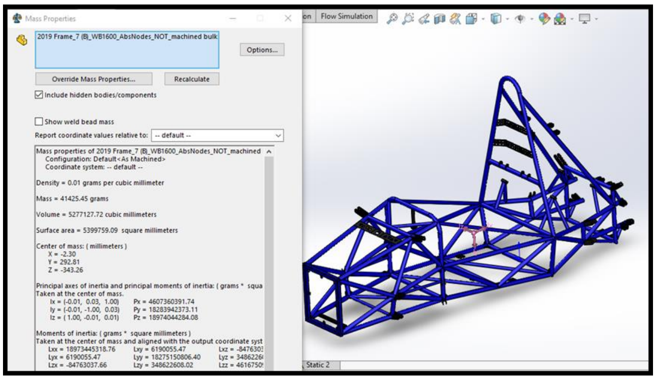


Fig. – VI.B.3 (CG measurement in Solid works software)

- ✓ The image below shows the particle center of gravity of the chassis calculated by the moment formula. Practically the C.G is not measured from the origin.
- ✓ In X-axis it is measured from the front or rear bulkhead, in Y-axis it is measured from the centerline (red), and in Z-axis from the ground.

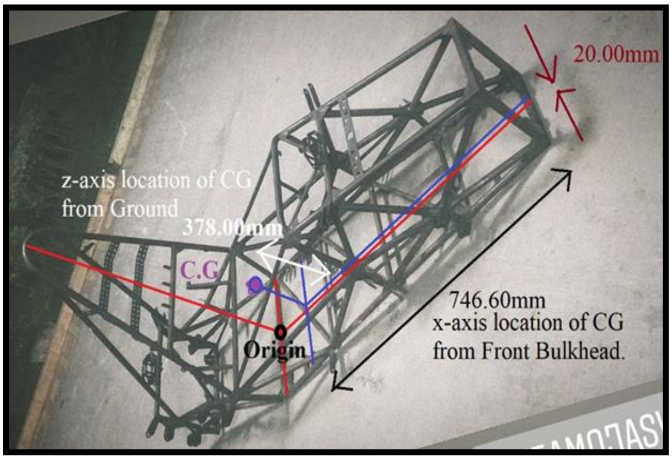


Fig. – VI.B.4 (Location of calculated CG of the prototype)

VII. DESTRUCTIVE TESTING

A. Introduction

- ➢ According to several research papers, FSAE chassis torsional stiffness should be under 1750 lbs-ft/degree, ie: 2372.68 N-m/degree. The 2019 model was designed to achieve 2000 N-m/degree of torsional stiffness under simulation but the destructive is generally performed at a lower scale to prevent the damage of the chassis. So, 1500 N-m/degree was selected as a threshold for experimental testing.
- ➢ The results of FEA simulations are 100% accurate because there are several changes in geometry and structure to manufacturing errors and residual stresses due to welding, therefore we perform destructive testing on the Torsional Rig apparatus.
- ➢ When the load is applied on one side of the chassis, then the side of load application deflects downwards and the other side deflects upwards, the deflection is measured by a dial gauge at varying loads that is varying torque and many readings are taken at a single point to eliminate errors in the experiment.
- ➢ If the chassis is not stiff enough it will bend along Z-axis and the torsional stiffness will cause will affect the suspension system and affect the vehicle dynamics of the car.

B. Methodology

- ➢ A jig was used to fix the hardpoints and torque was applied on front hardpoints. A dial gauge is used to measure the deflection. The jigs are designed in a way that does not leave any gap between the chassis tubes and the jig plates.
- ➢ The height of the jig was decided considering the height of the dial gauge so that the dial gauge can be easily kept below the chassis.
- ➢ The number of bolts is kept more than required as the rear of the chassis should not move in the jig when the load is applied if there is any deflection in any axis in the rear part because of the load the values in the dial gauge will not be correct.
- ➢ The plates are strongly bolted on chassis and plates welded to the base table.
- ➢ A T-shaped structure is made using a square tube and the trunk of the T passes through the hardpoints. A square tube is used, as a round tube will roll when the load is applied and square tubes have higher bending stiffness.
- ➢ A rectangular wooden block is kept between the square tube and the vertical tube connecting 2 hardpoints so that there is no space for the square tube to slide when the load is applied.

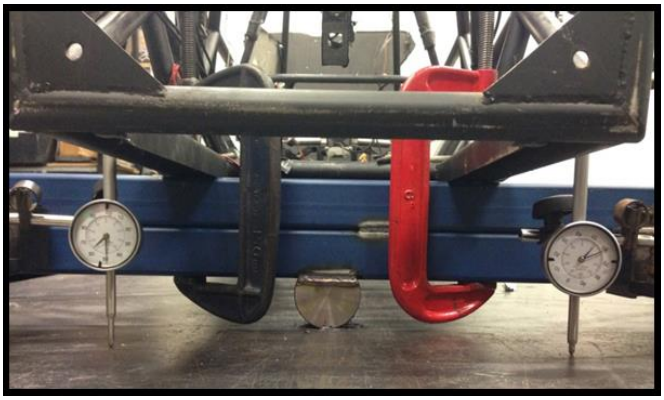


Fig. – VII.B.2 (Torsional Rig Apparatus)

C. Calculation

- ➢ Angle of twist (φ) = sin-1(d/L)
- ➢ d = deflection, L = distance of load application point from th e center of the chassis.
- ➢ Torque = m.g.L
- ➢ Torsional rigidity = T/ǿ
- ➢ Average torsional stiffness = 1/k = 1/k(front) + 1/k(cockpit) + 1/k(rear)
- ➢ a = sin-1(d/D), D is distance from center line to point of application of Load.

D. Tabulated results

- ➢ The deflection measured at a point that is 300 mm from the front bulkhead.

SR. NO.	1	2	3	4	5
Load on Chassis (kg)	12.5	15.5	18.5	21.5	24.5
Deflection (mm)	0.34	0.36	0.51	0.62	0.69
Angle Twist L = 335mm	0.0581	0.0615	0.0872	0.1060	0.1180
Torque (N-m)	43.531	53.979	64.427	74.874	85.322
Torsional Rigidity (N-m/degree)	748.60	876.69	738.61	706.09	722.99
Avg. Torsional Rigidity (Nm/deg.)	758.60				
Simulation angle of twist (degrees)	0.039	0.048	0.058	0.067	0.077
FEA Torsional Rigidity (Nm/deg.)	1107.68	1107.72	1107.75	1107.78	1107.79
FEA Average Torsional Rigidity	1107.74				

Table 4

➢ Deflection measured at a point that is between lower Hardpoints.

SR. NO.	1	2	3	4	5
Load on Chassis (kg)	12.5	15.5	18.5	21.5	24.5
Deflection (mm)	0.31	0.39	0.45	0.51	0.57
Angle Twist L = 335mm	0.0530	0.0667	0.0769	0.0872	0.0974
Torque (N-m)	43.531	53.979	64.427	74.874	85.322
Torsional Rigidity (N-m/degree)	821.04	809.29	837.10	858.39	875.20
Avg. Torsional Rigidity (Nm/deg.)	840.20				
Simulation angle of twist (degrees)	0.039	0.049	0.059	0.068	0.078
FEA Torsional Rigidity (Nm/deg.)	1091.29	1091.37	1091.24	1091.31	1091.35
FEA Average Torsional Rigidity	1091.31				

Table 5

➢ Deflection measured at a point between the chassis cockpit.

SR. NO.	**1**	**2**	**3**	**4**	**5**
Load on Chassis (kg)	12.50	15.50	18.50	21.50	24.50
Deflection (mm)	0.09	0.11	0.13	0.17	0.21
Angle Twist L = 335mm	0.015	0.018	0.022	0.029	0.035
Torque (N-m)	43.53	53.97	64.42	74.87	85.32
Torsional Rigidity (N-m/degree)	2828.05	2869.18	2897.66	2575.18	2375.56
Avg. Torsional Rigidity (Nm/deg.)	2709.13				
Simulation angle of twist (degrees)	0.012	0.015	0.019	0.022	0.025
FEA Torsional Rigidity (Nm/deg.)	3387.69	3388.54	3389.66	3388.18	3388.56
FEA Average Torsional Rigidity	3388.37				

Table 6

➢ Deflection measured at a point below Main roll hoop.

SR. NO.	**1**	**2**	**3**	**4**	**5**
Load on Chassis (kg)	12.50	15.50	18.50	21.50	24.50
Deflection (mm)	0.05	0.07	0.09	0.11	0.15
Angle Twist L = 335mm	0.008	0.011	0.015	0.018	0.023
Torque (N-m)	43.53	53.97	64.42	74.87	85.32
Torsional Rigidity (N-m/degree)	5090.48	4508.71	4185.51	3979.83	3563.34
Avg. Torsional Rigidity (Nm/deg.)	4265.58				
Simulation angle of twist (degrees)	0.003	0.004	0.005	0.006	0.007
FEA Torsional Rigidity (Nm/deg.)	11179.22	11180.51	11179.45	11180.35	11181.03
FEA Average Torsional Rigidity	11180.11				

Table 7

VIII. CONCLUSION

Witness the data illustrated in the above chapters, model 2019-20 is the lightest in the history of FSAE team Ojaswat with a weight of 44.90kg with adequate torsional stiffness and endurance against Front, Rear, Side & Roll impacts. Miscellaneous simulations such as Frequency, Fatigue, and Drop tests were carried out successfully and have contributed to overall data. Also from the above verification, we can conclude that the center of gravity of chassis is nearly matching that of CAD file and well balanced with high Strength to Weight Ratio. This project has further helped us learn -

- Vehicle dynamics – Basis concepts of vehicle dynamics, tire dynamics, suspension geometry, and spaceframe design procedures.
- Chassis and suspension system. – Conceptual knowledge in the field of chassis and suspension systems.
- Formula racing vehicle – Apart from chassis and vehicle dynamics, the project has helped us boost our knowledge in the areas of Wet & Dry Powertrain, Steering systems, Electrical systems & Aerodynamics.
- CAD software like Solid works & Fusion – A good practice with CAD features like industrial drawings, weldments, sheet metals, surface modeling, and many more.
- CAE software like Ansys, Lotus shark & Adams – Apart from Solid works 3D simulation, we have used Ansys 2D wireframe simulation to achieve great accuracy. Also, Lotus Shark has been very useful to us in generating various graphs related to suspension calculations.
- Developing software like MATLAB, Turbo C – Matlab & turbo C+ has been very useful to develop codes. These codes were used to perform several iterations in calculating spring stiffness and other suspension parameters.
- Manufacturing techniques like welding, profiling, etc. – Within this course of 4 years, we acquired great manufacturing skills such as TIG, MIG & Arc welding, profiling, cutting, grinding, drilling, and many more.
- Since these FSAE competitions (SAE Supra, FMAE FFS & FIA Formula Bharat) take place on National & International levels, we had an exposure to interact with great teams, expert judges like Pat Clarke & Claude Rouelle and famous industrialists.
- Personally, as a team captain of Team Ojaswat, this project has helped me develop several important valves such as teamwork, punctuality, responsibility, time management, and many more.
- The figures below describe the transition of the 2019 model of team Ojaswat from CAD file to prototype chassis and later the final assembly.

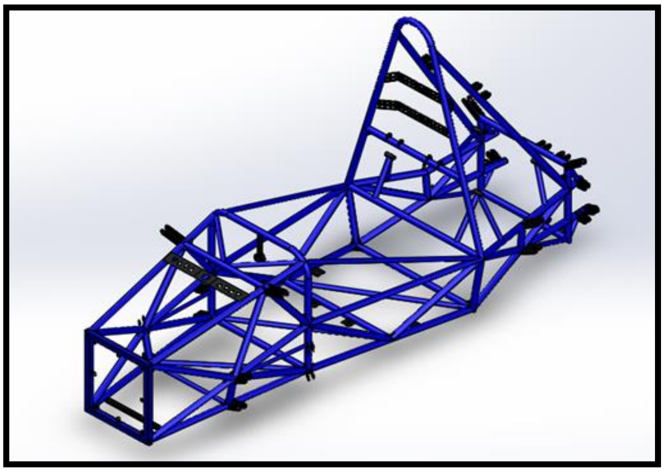


Fig. - VIII.A.1 (CAD file)

Fig. – VIII.B.1 (Prototype)

Fig. – VIII.C.1 (Vehicle 2019-20)

IX. ACKNOWLEDGMENT

I express my cavernous sense of obligation and gratitude to my guide Mr. Anand Patel for his genuine guidance and constant encouragement throughout this project work. I am highly obliged as my honorable guide has devoted his valuable time and shared his expert knowledge.

I pay my profound gratefulness to the Mechanical engineering Department of C.S.P.I.T - Chanrusat University – Changa for allowing me to carry out the project work. I am also thankful to all teammates of Team Ojaswat. who helped me directly or indirectly for the completion of my work.

I extend my sincere thanks to HOD – Dr. Vijay Chaudhary, Department of Mechanical Engineering, Principal – Dr. A. D. Patel of C.S.P.I.T and Dean, Faculty of Technology & Engineering, CHARUSAT for providing me such an opportunity to do my project work.

I also wish to express my heartfelt appreciation to my friends, colleagues, and many who have rendered their support for the successful completion of the project, both explicitly and implicitly.

REFERENCES

➢ *Books –*

[1]. William and Douglas Milliken.
[2]. Richard Stone vehicle dynamics.
[3]. Carrol Smith – Tune to win, Engineer to win, Design to win.
[4]. David A Corolla – Automobile engineering.
[5]. Formula Bharat 2019 rulebook.
[6]. SAE Supra 2020 rulebook.
[7]. The Automotive Chassis - Genta-Morello.
[8]. The Automotive Chassis-Reimpell-Stol-Betzler.
[9]. Richard Stone, Jeffrey K. Ball-Automotive Engineering Fundamentals-SAE International (2004).

➢ *Research Papers –*

[10]. University of Delaware – FSAE chassis and suspension design.
[11]. Ferrington report
[12]. MEEG 402-010 Chassis Design Report.
[13]. Design of a Formula SAE Racecar Chassis: Composite Analysis Utilizing Altair Engineering OptiStruct Software - Margaret Lafreniere.
[14]. FSAE 2015 Chassis and Suspension Final Report – University of Florida
[15]. Design of a Formula Student race car chassis - Jannis D.G.van Kerkhoven, 516303.
[16]. Design of formula student race car chassis - Alperen Kale.
[17]. The dynamic design of automotive systems: Engine mounts and structural joints - R SINGH.
[18]. Design, Analysis, and Testing of a Formula SAE Car Chassis - William B. Riley and Albert R. George Cornell University.
[19]. Design and Optimization of an FSAE Frame, Suspension, and Business Portfolio - Worcester Polytechnic Institute.
[20]. [20] A study of vibration isolation of engine mounts – Ruiping Wang.
[21]. Miscellaneous SAE reports.

➢ *Web lectures –*

[22]. NPTEL vehicle dynamics
[23]. NPTEL tire modeling
[24]. Sims scale Static structural

➢ *Websites –*

[25]. https://www.sae.org/publications/technical-papers
[26]. https://www.solidworks.com/
[27]. https://www.mscsoftware.com/product/adams
[28]. https://www.simscale.com/
[29]. https://in.mathworks.com/
[30]. https://www.ansys.com/en-in
[31]. https://swayam.gov.in/nc_details/NPTEL
[32]. https://www.youtube.com/
[33]. https://www.google.co.in/

➢ *Forums –*

[34]. SAE forums
[35]. Solid works forums
[36]. FSAE forums.
[37]. Ansys forums
[38]. Math works forums
[39]. Sims scale forums

➢ *Resources –*

[40]. Solid works 2019
[41]. Ansys 18.2
[42]. Simsscale 2020
[43]. Lotus Shark & raven
[44]. Matlab R2020a
[45]. MSC Adams
[46]. Hyperworks
[47]. MS office

APPENDIX – 1 (SUSPENSION CALCULATION)

➢ Longitudinal Acceleration:

(Acceleration Track)
For straights, considering 60m
Straight traveling distance, d=60m
Time of traveling, t=4 sec
Velocity of car, v = d/t
= 60/4 = 15 m/sec
Longitudinal acceleration – a = v/t
= 15/4 = 3.7 m/sec^2
Taking F.O.S. = 2.50
a = 1.2g

➢ Lateral Acceleration:

(Skid-pad Track)
Skid pad track diameter, Ø=15.25m
r = 9.125m
Width of track = 3m
Now, Traveling distance d = 2.pi. r
= 2*3.14*9.125= 57.33 m
For above d and t = 5.5 sec. (For complete lap)
a= v^2/r
= ((〖57.33/5.5)〗^2)/9.125
= 11.90 m/s^2 = 1.213g
So for safety we have a = 1.5g

ISSN No:-2456-2165

➢ Roll Chamber:
Chassis Roll angle = 1.6
Roll chamber = (Wheel chamber angle)/ (Chassis roll angle)
= (-1)/1.6 = 0.625

➢ Track Width:
= 1200 (front)
Front view swing arm length:
FVSA= (t/2)/ (1-Roll camber)
= 600/ (1-0.625) = 1600 mm

➢ Weight of the whole car with driver
= 310kg
Suspension arm's length was found from front and side view geometry.
C.G. height = 280 mm
Mass of car= 310kg

➢ Roll Centre Height for front:
= RCH= 30 mm

➢ Roll Centre Height for the rear:
= RCH= 90 mm

➢ Height:
H= h- (yrf + a/l (yrr - yrf))
= 280-(30+ 930/1600 (90 – 30))
= 215.125 mm

➢ Roll Stiffness:
kΦ = (m*H)/Ө
= (310*0.215*180)/ (1.6*3.14)
= 2387.93 (kg m)/rad

➢ Roll Stiffness Distribution:
(At rear, lateral weight distribution is larger. Hence, the roll stiffness distribution is biased 52% in front and 48% in rear)
kΦ(front)
= kΦ* 0.52 = 0.52 * 2387.93
= 1241.72 (kg m)/rad
kΦ(rear)
= kΦ* 0.48 = 0.48 * 2387.93
= 1146.2 (kg m)/rad

➢ Weight Transfer due to lateral acceleration
Weight transfer at front,
ΔWyf = Ay * m/t_F * [(H* 〖kΦ〗 _F)/kΦ + (b* Y_rf)/l]
= 1.5 * 330/1.2 * [(0.215*1241.72)/2387.93 + (670* 0.030)/1600]
= 48.19 kg.
ΔWyr = Ay * m/t_F * [(H* 〖kΦ〗 _r)/kΦ + (b* Y_rr)/l]
= 1.5 * 310/1.2 * [(0.215*1146.2)/2387.93 + (930* 0.090)/1550]
= 60.26 kg.

➢ Ride Rate:
Force needed per unit of vertical displacement of the tire contact patch

KRF = 〖2*KΦ〗 _FS/t^2 = (2*1241.12)/ 〖1.2 〗^2 = 1723.77 kg/m
KRR = 〖2*KΦ〗 _RS/t^2 = (2*1146.2)/ 〖1.1〗^2 = 1894.54 kg/m

➢ Wheel Rate:
Vertical force per unit displacement of the wheel.
KWF = (vertical tyre rate* K_RF)/ (vertical tyre Rate-K_(RF))
= (18000*1865.75)/ ((18000-1865.75))
= 15 N/mm
KWR = (vertical tyre rate* K_RR)/ (vertical tyre Rate-K_(RR))
= (18000*1722.23)/ (18000-1722.23)
= 21.36 N/mm

➢ Installation ratio:
It relates the displacement of spring to the vertical displacement of the wheel

It will reduce both displacement and force at the wheel relative to spring.
IRF =√ (K_WF/K_S) = √ (15/36.36) = 0.65

➢ Motion ratio:
MR = a/b sin Ө
=110/70 sin 35.5
= 0.85

➢ Rocker arm design:
Considering the external weight on spring weight = 70 kg
Now let W is the load through pushrod and P is transferred to spring
W*x= P*y
1300*70= 110 * P
P= 827.27
RF = √ (w^2+P^2)
=√ (〖1300〗^2+ 〖827.27〗^2)
= 1540 N

➢ Design of fulcrum:
d= diameter of fulcrum pin
L= length of fulcrum pin = 1.25d
l=1.25d
RF = d*l*pb
1540=d*1.25d*10
(assume pb = 10N/ 〖mm〗^2)
d= 11.02 ≈ 12 mm
RF = 2*3.14/4 * d2 * Ʈ
1540 = 2*3.14/4 * 122 * Ʈ
Ʈ= 6.8 MPa
Now external diameter of boss= D= 2d= 2*12 = 24 mm
Bronze bush of 2 mm thick.
Internal diameter of hole in lever,
dh = d+ 2*t
= 12+ 2*2= 16 mm
Bending moment at boss hole= W* x
=1300*70 = 91000 N mm
Section modulus Z=
(1/12*b[D^3- 〖d_h〗^3])/12

= (1/12*22[〖24〗 ^3- 〖16〗 ^3])/12
= 1486.22 mm3
Induced bearing stress σb = M/Z
= 91000/1486.22 = 61.22 MPa

- Design of forked end:

Diameter of bolt = d1
Length of bolt =l1= 1.25 d1
W= d1 * l1 * Pb
1300= d1 * 1.25 d1*10
d1= 10 mm l1= 12.5mm
Now
W= 2*3.14/4* 〖d_1〗 ^2 *Ƭ_1
1300= 2*3.14/4* 〖10〗 ^2 *Ƭ_1
Ƭ1= 8.27 MPa
Thickness of each eye
t1= l_1/2 = 6.25mm

- Maximum bending M

= 1300/2 (12.5/2+ 6.25/3) -(1300/2*6.25/4)
= 5416.66 – 1015.62
= 4401.03 N.mm
Z= 3.14/32 * d13 = 98.17 MPa
Bending stress induced = σb
= 4410.03/98.17 = 44.82 MPa
Over all diameter of eye D1 =2*d1 = 2*10 = 20mm
Outer diameter of roller is taken 2mm more.
Clearance of 1.5 mm
l2= l1+ 2*t_1/2+ 2 *1.5
= 12.5+ 2*6.25/2+ 2 *1.5
=21.75 mm
Thickness of lever arm = t
Depth or width = B
M=W (80 - 30/2)
=1300(80 - 30/2)
=84500 Nmm
Z= 1/6*t*B^2
= 1/6*t* 〖30〗 ^2
=150*t

- Bending Stress σb=M/Z

70=84500/(150*t)
t=8mm
Now Wheel Rate = 15 N/mm
f=1/(2*3.14) √((W.R)/(S.W)
=2.83 Hz
Now S. R
= ((W.R))/ (〖 (M.R) 〗 ^2*0.66)
S. R= 32 N/mm (Front)
K = Spring Stiffness
D = mean dia of the spring
d = dia of the wire
G = Shear Modulus
n = no of active coil turns
n' = no of total coil turn
Considering d= 8mm and D=60mm
k= (Gd^4)/ (8*D^3 n)
32= (84* 〖10〗 ^3* 〖 (8) 〗 ^4)/ (8* 〖 (52) 〗 ^3*n)
n= 10 turns

- Spring Data:

S.R. =32 N/mm
D=60 – 8 = 52mm
d=8mm
n=10 turns (Active turns)
L=150mm

- Given below the graphs generated from MSC Adams software of suspension analysis. A similar kind of simulation was also performed in Lotus Shark & Raven but Adams turns out to be more accuruate.

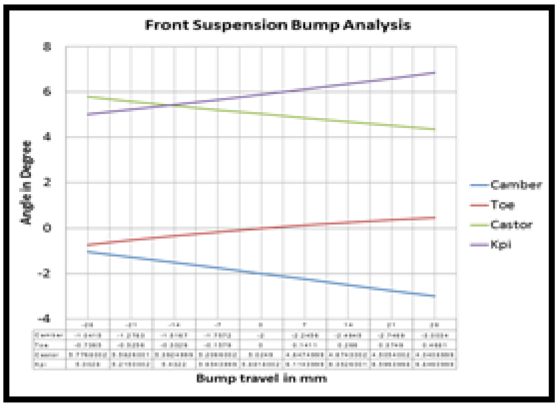


Fig. – A.1.1 (Front Suspension Roll Analysis)

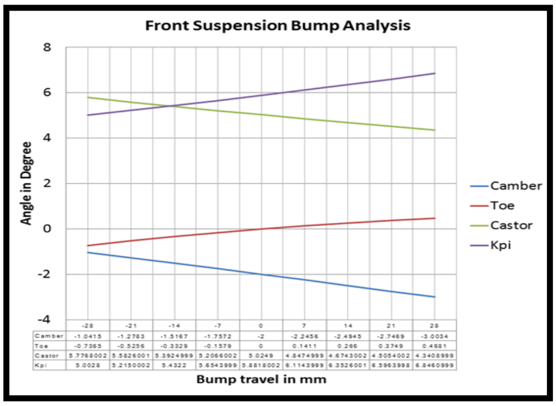


Fig. – A.1.23 (Front Suspension Roll Analysis)

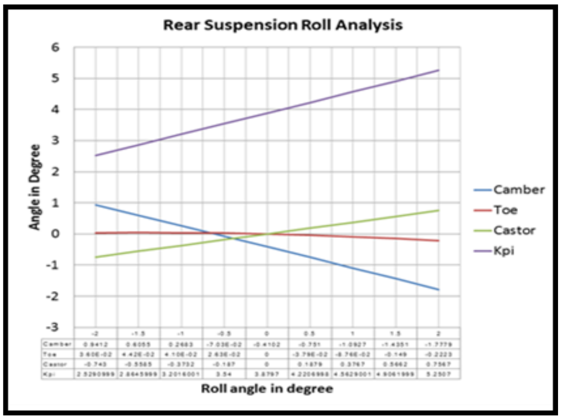


Fig. – A.1.3 (Rear Suspension Roll Analysis)

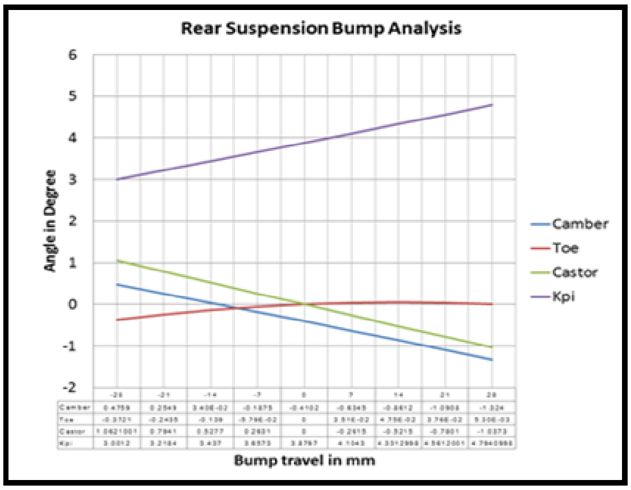
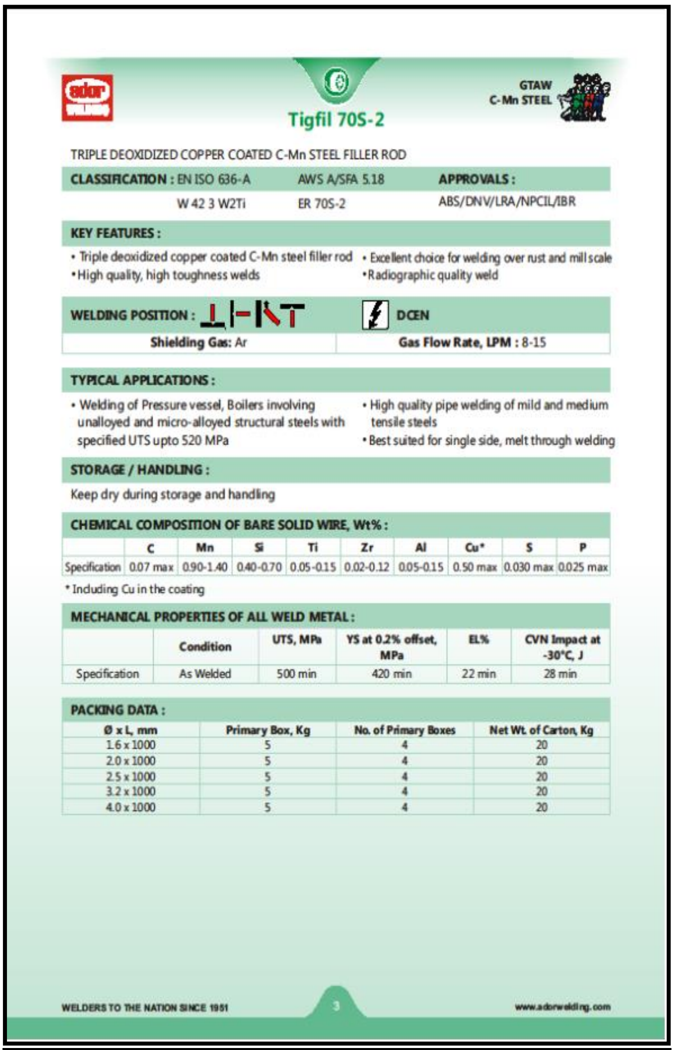


Fig. – A.1.4 (Rear Suspension Bump Analysis)

APPENDIX -2 (WELDING FILLER/MATERIAL DATASHEETS)



ador WELDING

Tigfil 70S-2

GTAW C-Mn STEEL

TRIPLE DEOXIDIZED COPPER COATED C-Mn STEEL FILLER ROD

CLASSIFICATION : EN ISO 636-A W 42 3 W2Ti | AWS A/SFA 5.18 ER 70S-2 | **APPROVALS :** ABS/DNV/LRA/NPCIL/IBR

KEY FEATURES :
- Triple deoxidized copper coated C-Mn steel filler rod
- High quality, high toughness welds
- Excellent choice for welding over rust and mill scale
- Radiographic quality weld

WELDING POSITION : DCEN

Shielding Gas: Ar | **Gas Flow Rate, LPM :** 8-15

TYPICAL APPLICATIONS :
- Welding of Pressure vessel, Boilers involving unalloyed and micro-alloyed structural steels with specified UTS upto 520 MPa
- High quality pipe welding of mild and medium tensile steels
- Best suited for single side, melt through welding

STORAGE / HANDLING :
Keep dry during storage and handling

CHEMICAL COMPOSITION OF BARE SOLID WIRE, Wt% :

	C	Mn	Si	Ti	Zr	Al	Cu*	S	P
Specification	0.07 max	0.90-1.40	0.40-0.70	0.05-0.15	0.02-0.12	0.05-0.15	0.50 max	0.030 max	0.025 max

*Including Cu in the coating

MECHANICAL PROPERTIES OF ALL WELD METAL :

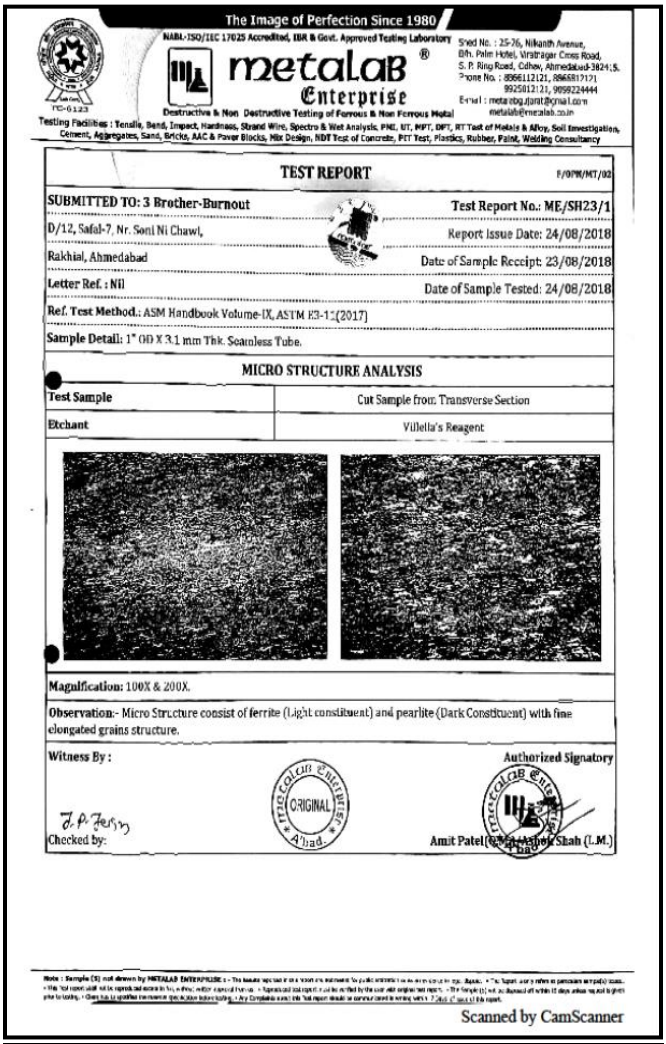
	Condition	UTS, MPa	YS at 0.2% offset, MPa	EL%	CVN Impact at -30°C, J
Specification	As Welded	500 min	420 min	22 min	28 min

PACKING DATA :

Ø x L, mm	Primary Box, Kg	No. of Primary Boxes	Net Wt. of Carton, Kg
1.6 x 1000	5	4	20
2.0 x 1000	5	4	20
2.5 x 1000	5	4	20
3.2 x 1000	5	4	20
4.0 x 1000	5	4	20

WELDERS TO THE NATION SINCE 1951 3 www.adorwelding.com

Fig. - A2.1 (TIG welding filler rod Datasheet)



The Image of Perfection Since 1980

NABL-ISO/IEC 17025 Accredited, IBR & Govt. Approved Testing Laboratory

metalaB® Enterprise

Destructive & Non Destructive Testing of Ferrous & Non Ferrous Metal

Shed No. : 25-26, Nilkanth Avenue, B/h. Palm Hotel, Viratnagar Cross Road, S. P. Ring Road, Odhav, Ahmedabad-382415.
Phone No. : 8866112121, 8866812121 9925012121, 9099224444
Email : metalabgujarat@gmail.com metalab@metalab.co.in

TC-6123

Testing Facilities : Tensile, Bend, Impact, Hardness, Strand Wire, Spectro & Wet Analysis, PMI, UT, MPT, DPT, RT Test of Metals & Alloy, Soil Investigation, Cement, Aggregates, Sand, Bricks, AAC & Paver Blocks, Mix Design, NDT Test of Concrete, PIT Test, Plastics, Rubber, Paint, Welding Consultancy

TEST REPORT F/OPM/MT/02

SUBMITTED TO: 3 Brother-Burnout | Test Report No.: ME/SH23/1
D/12, Safal-7, Nr. Soni Ni Chawl, | Report Issue Date: 24/08/2018
Rakhial, Ahmedabad | Date of Sample Receipt: 23/08/2018
Letter Ref. : Nil | Date of Sample Tested: 24/08/2018

Ref. Test Method.: ASM Handbook Volume-IX, ASTM E3-11(2017)

Sample Detail: 1" OD X 3.1 mm Thk. Seamless Tube.

MICRO STRUCTURE ANALYSIS

Test Sample	Cut Sample from Transverse Section
Etchant	Villella's Reagent

Magnification: 100X & 200X.

Observation:- Micro Structure consist of ferrite (Light constituent) and pearlite (Dark Constituent) with fine elongated grains structure.

Witness By : Authorized Signatory

ORIGINAL

Checked by: Amit Patel(Q.M.) Ashok Shah (L.M.)

Scanned by CamScanner

Fig. – A2.2 (AISI 4130 Microstructure test)

NABL Accredited Testing Lab for Chemical & Mechanical as Per ISO / IEC 17025 : 2005

Certificate No. TC-0342

ACCURATE METAL TEST SERVICES

Testing of Soil and Building Material

(Testing of Metals by PMI, Spectro, Wet analysis, Mechanical, Impact, Micro & IGC Practice)

140, Tejendra Ind. Estate, Opp. C.M.C., Nr. Soni Ni Chal Cross Road, Odhav Road, Odhav, Ahmedabad-382415.
Ph.: (079) 22971510 (M) 9099924208, 9824523645, 9723336390
Web: www.accuratemetaltesting.com E-mail: accuratemetaltest@gmail.com, info@accuratemetaltesting.com

TEST REPORT

AMTS/TE/UF/0979/18-19
Your Ref: -
Condition of Sample/s: - OK
M/s 3 Brothers, Vidit Adani.
Sample Details:- 1" X 2mm Pipe

Date of Issue: - 01-09-2018
Date of Sample Receipt:- 31-08-2018
Sample Tested on Date:- 01-09-2018

PHYSICAL ANALYSIS

Test Carried Out		Tensile
Material		AISI 4130
Test Method		ASTM, A370-17
Outer Diameter	[mm]	25.40
Thickness Of Pipe	[mm]	1.80
Gauge Length	[mm]	50.00
Yield Proof Load KN		88.29
Ultimate Load KN		111.06
Final Gauge Length	[mm]	57.67
Area mm²		133.38
Yield Stress N/mm²		661.94
U.T.S. N/mm²		832.66
Elongation %		15.34
Hardness	[HRC]	25/26/27
Location of Fracture		WGL

Note:- This report refers only to sample(s) submitted for test.

Remarks:-

ORIGINAL ACCURATE METAL TEST SERVICES

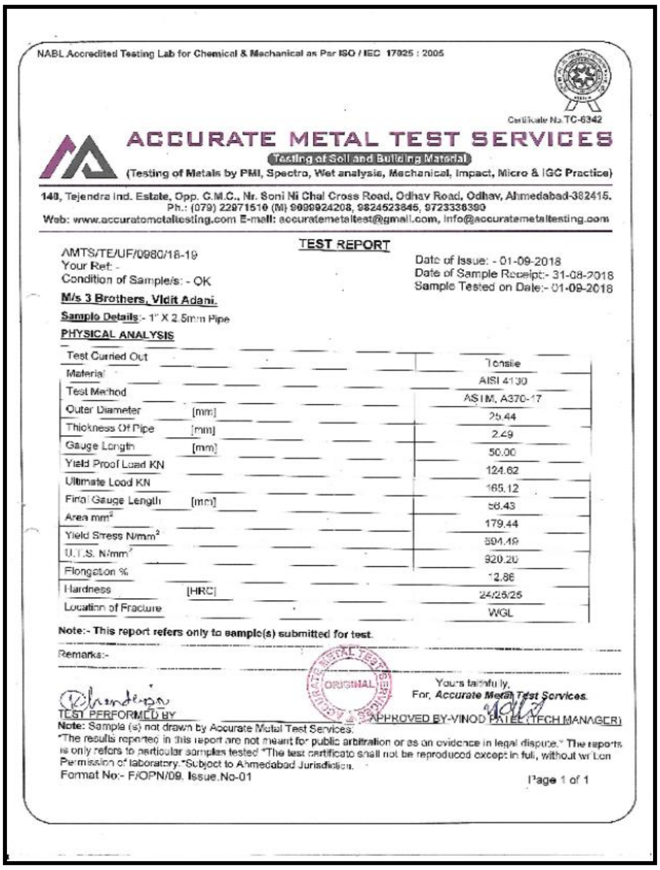
Yours faithfully,
For, Accurate Metal Test Services.

TEST PERFORMED BY APPROVED BY-VINOD PATEL (TECH MANAGER)

Note: Sample (s) not drawn by Accurate Metal Test Services.
"The results reported in this report are not meant for public arbitration or as an evidence in legal dispute." The reports is only refers to particular samples tested "The test certificate shall not be reproduced except in full, without written Permission of laboratory."Subject to Ahmedabad Jurisdiction.
Format No:- F/OPN/09, Issue.No-01

Page 1 of 1

Fig. – A2.3 (AISI 4130 – 25.40x2.00mm tube test report)



NABL Accredited Testing Lab for Chemical & Mechanical as Per ISO / IEC 17025 : 2005

Certificate No. TC-6342

ACCURATE METAL TEST SERVICES

Testing of Soil and Building Material

(Testing of Metals by PMI, Spectro, Wet analysis, Mechanical, Impact, Micro & IGC Practice)

140, Tejendra Ind. Estate, Opp. C.M.C., Nr. Soni Ni Chal Cross Road, Odhav Road, Odhav, Ahmedabad-382415.
Ph.: (079) 22971510 (M) 9099924208, 9824523645, 9723336390
Web: www.accuratemetaltesting.com E-mail: accuratemetaltest@gmail.com, info@accuratemetaltesting.com

TEST REPORT

AMTS/TE/UF/0980/18-19
Your Ref: -
Condition of Sample/s: - OK
M/s 3 Brothers, Vidit Adani.
Sample Details:- 1" X 2.5mm Pipe

Date of Issue: - 01-09-2018
Date of Sample Receipt:- 31-08-2018
Sample Tested on Date:- 01-09-2018

PHYSICAL ANALYSIS

Test Carried Out		Tensile
Material		AISI 4130
Test Method		ASTM, A370-17
Outer Diameter	[mm]	25.44
Thickness Of Pipe	[mm]	2.49
Gauge Length	[mm]	50.00
Yield Proof Load KN		124.62
Ultimate Load KN		165.12
Final Gauge Length	[mm]	56.43
Area mm²		179.44
Yield Stress N/mm²		694.49
U.T.S. N/mm²		920.20
Elongation %		12.86
Hardness	[HRC]	24/25/25
Location of Fracture		WGL

Note:- This report refers only to sample(s) submitted for test.

Remarks:-

ORIGINAL ACCURATE METAL TEST SERVICES

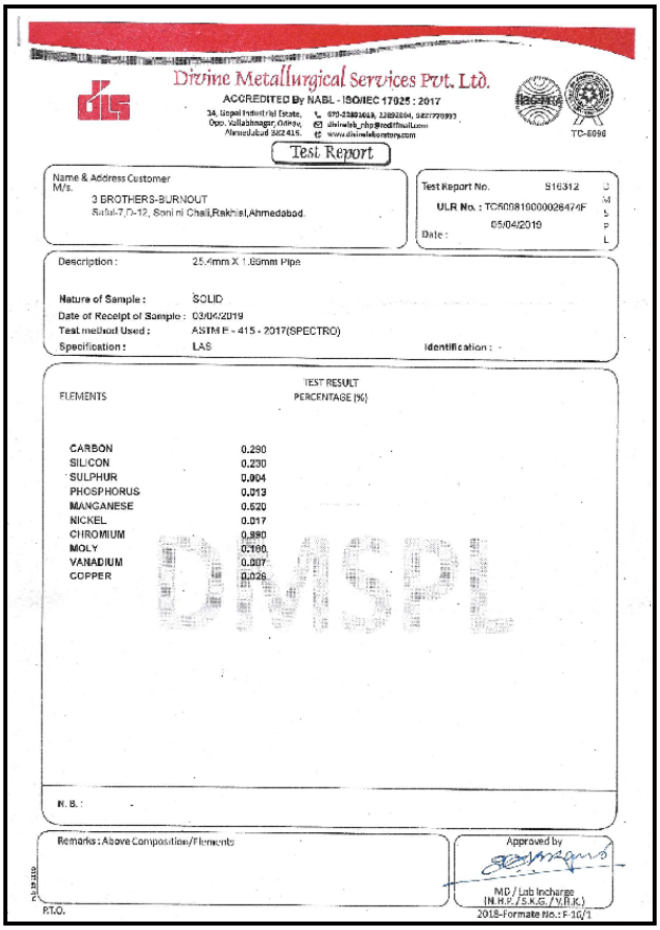
Yours faithfully,
For, Accurate Metal Test Services.

TEST PERFORMED BY APPROVED BY-VINOD PATEL (TECH MANAGER)

Note: Sample (s) not drawn by Accurate Metal Test Services.
"The results reported in this report are not meant for public arbitration or as an evidence in legal dispute." The reports is only refers to particular samples tested "The test certificate shall not be reproduced except in full, without written Permission of laboratory."Subject to Ahmedabad Jurisdiction.
Format No:- F/OPN/09, Issue.No-01

Page 1 of 1

Fig. – A2.4 (AISI 4130 – 25.40x2.50mm tube test report)



Divine Metallurgical Services Pvt. Ltd.

ACCREDITED By NABL - ISO/IEC 17025 : 2017

TC-6090

Test Report

Name & Address Customer
M/s.
3 BROTHERS-BURNOUT
Safal-7,D-12, Soni ni Chali,Rakhial,Ahmedabad.

Test Report No. S16312
ULR No. : TC500819000026474F
Date : 05/04/2019

Description : 25.4mm X 1.65mm Pipe

Nature of Sample : SOLID
Date of Receipt of Sample : 03/04/2019
Test method Used : ASTM E - 415 - 2017(SPECTRO)
Specification : LAS
Identification : -

ELEMENTS	TEST RESULT PERCENTAGE (%)
CARBON	0.290
SILICON	0.230
SULPHUR	0.004
PHOSPHORUS	0.013
MANGANESE	0.520
NICKEL	0.017
CHROMIUM	0.990
MOLY	0.160
VANADIUM	0.007
COPPER	0.026

DMSPL

N. B. : -

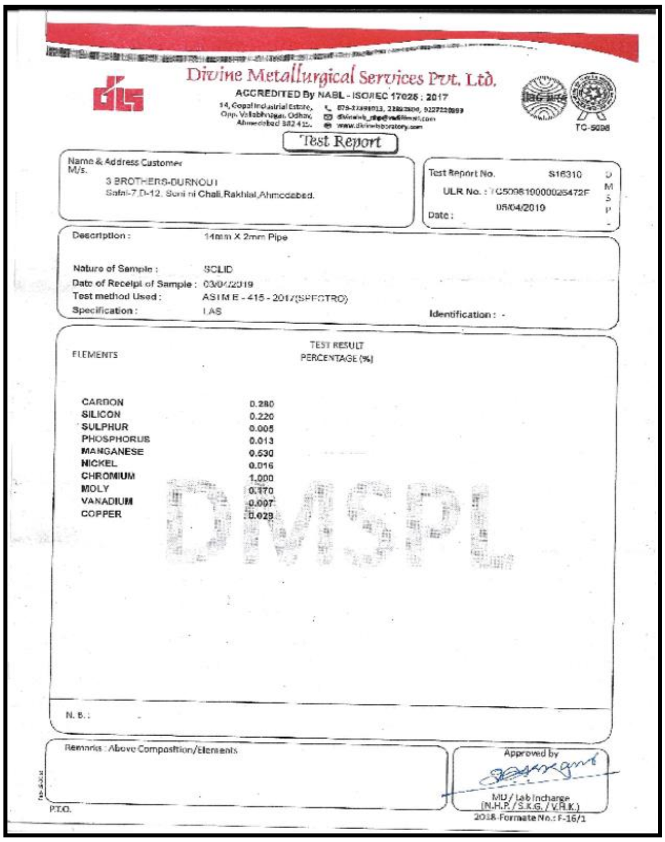
Remarks : Above Composition/Elements

Approved by
MD / Lab Incharge
(N.H.P. / S.K.G. / V.R.K.)

P.T.O.

2018-Formate No.: F-16/1

Fig. – A2.5 (AISI 4130 – 25.40x1.65mm tube test report)



Divine Metallurgical Services Pvt. Ltd.

ACCREDITED By NABL - ISO/IEC 17025 : 2017

TC-5098

Test Report

Name & Address Customer
M/s.
3 BROTHERS-BURNOUT
Safal-7,D-12, Soni ni Chali,Rakhial,Ahmedabad.

Test Report No. S16310
ULR No. : TC509810000025472F
Date : 05/04/2019

Description : 14mm X 2mm Pipe

Nature of Sample : SOLID
Date of Receipt of Sample : 03/04/2019
Test method Used : ASTM E - 415 - 2017(SPECTRO)
Specification : LAS
Identification : -

ELEMENTS	TEST RESULT PERCENTAGE (%)
CARBON	0.280
SILICON	0.220
SULPHUR	0.005
PHOSPHORUS	0.013
MANGANESE	0.530
NICKEL	0.016
CHROMIUM	1.000
MOLY	0.170
VANADIUM	0.007
COPPER	0.029

DMSPL

N. B. : -

Remarks : Above Composition/Elements

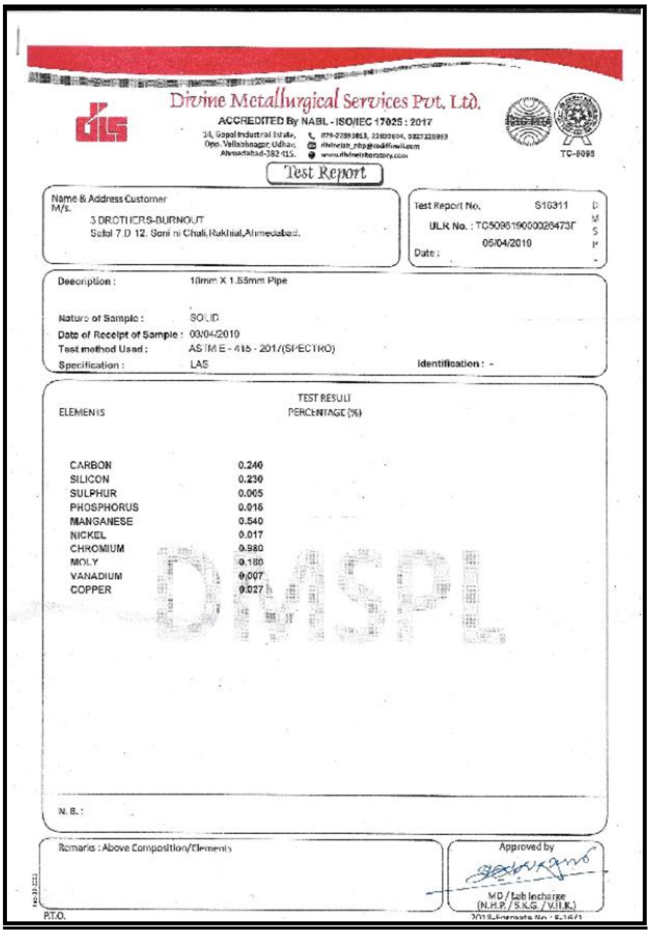
Approved by
MD / Lab Incharge
(N.H.P. / S.K.G. / V.R.K.)

P.T.O.

2018-Formate No.: F-16/1

Fig. – A2.6 (AISI 4130 – 14.00x2.00mm tube test report)

ISSN No:-2456-2165



Divine Metallurgical Services Pvt. Ltd.

ACCREDITED By NABL - ISO/IEC 17025 : 2017

34, Gopal Industrial Estate, Opp. Vallabhnagar, Udhav, Ahmedabad-382 415.

TC-5098

Test Report

Name & Address Customer
M/s.
3 BROTHERS-BURNOUT
Safal 7 D 12, Soni ni Chali, Rakhial, Ahmedabad.

Test Report No. S16311

ULR No. : TC509819000026473F

Date : 05/04/2019

Description : 10mm X 1.55mm Pipe

Nature of Sample : SOLID

Date of Receipt of Sample : 03/04/2019

Test method Used : ASTM E - 415 - 2017(SPECTRO)

Specification : LAS

Identification : -

TEST RESULT

ELEMENTS	PERCENTAGE (%)
CARBON	0.240
SILICON	0.230
SULPHUR	0.005
PHOSPHORUS	0.016
MANGANESE	0.540
NICKEL	0.017
CHROMIUM	0.980
MOLY	0.180
VANADIUM	0.007
COPPER	0.027

DMSPL

N. B. :

Remarks : Above Composition/Elements

Approved by

MD / Lab Incharge
(N.H.P. / S.K.G. / V.H.K.)

P.T.O.

Fig. – A2.6 (AISI 4130 – 19.05x2.00mm tube test report)

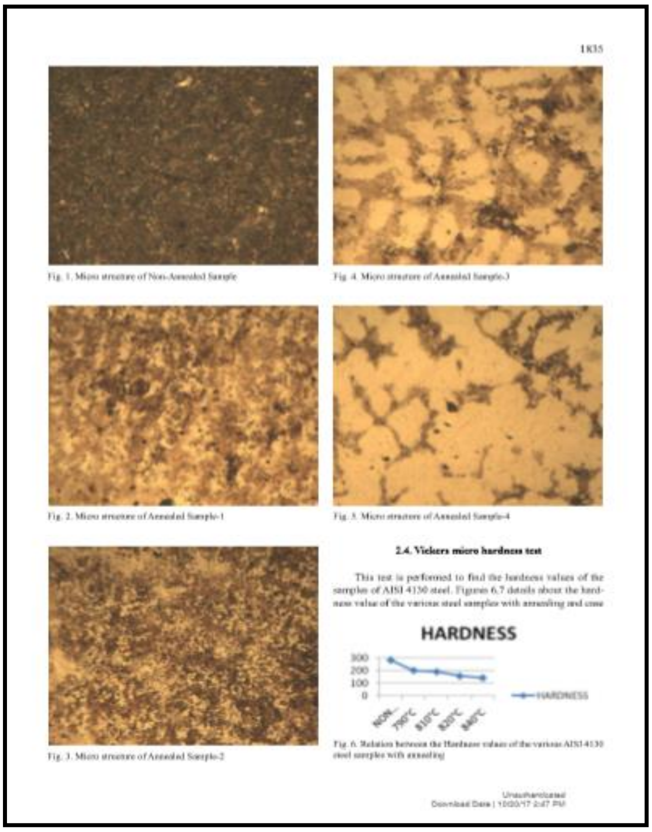


Fig. – A2.8 (AISI 4130 Hardening test)

APPENDIX – 3 (E – DRAWINGS)

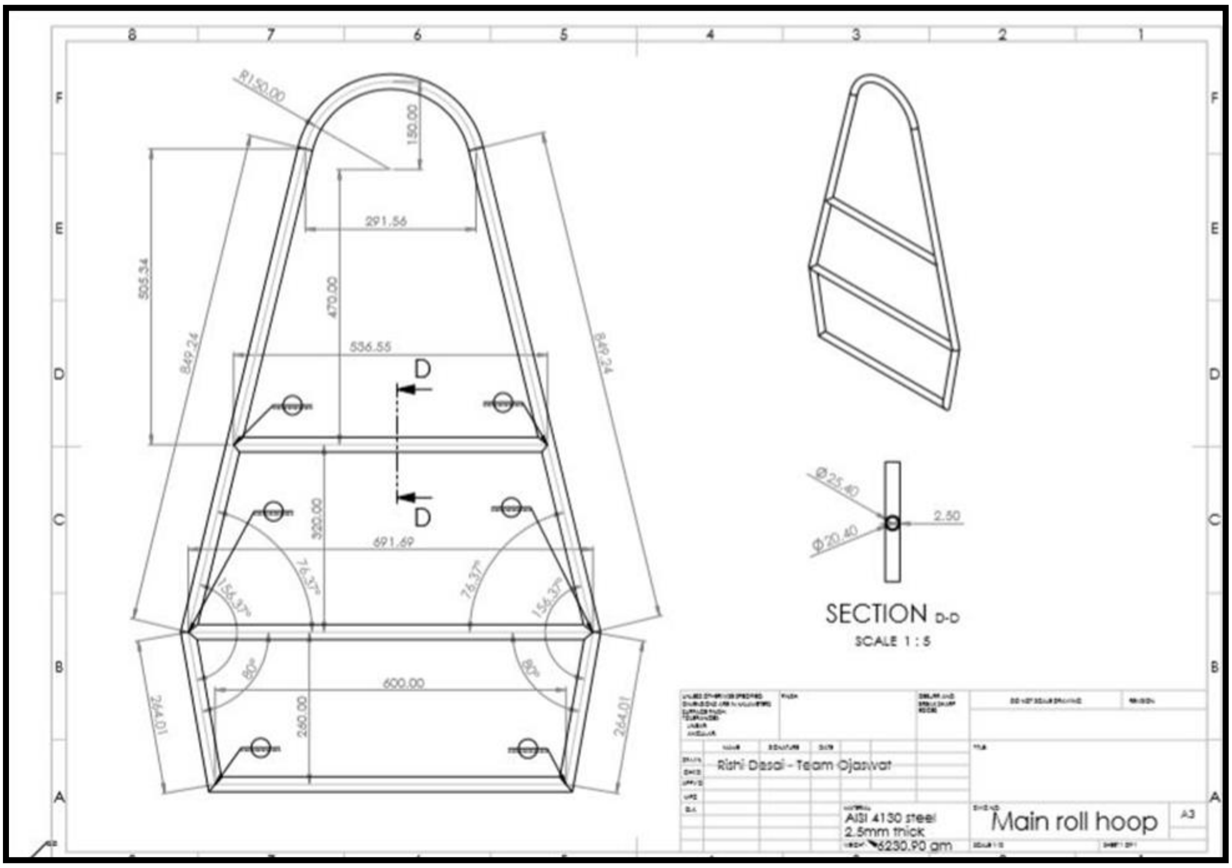


Fig – A3.1 (Main roll hoop drawing, 1:1 scale, Similar was Front hoop Fixture)

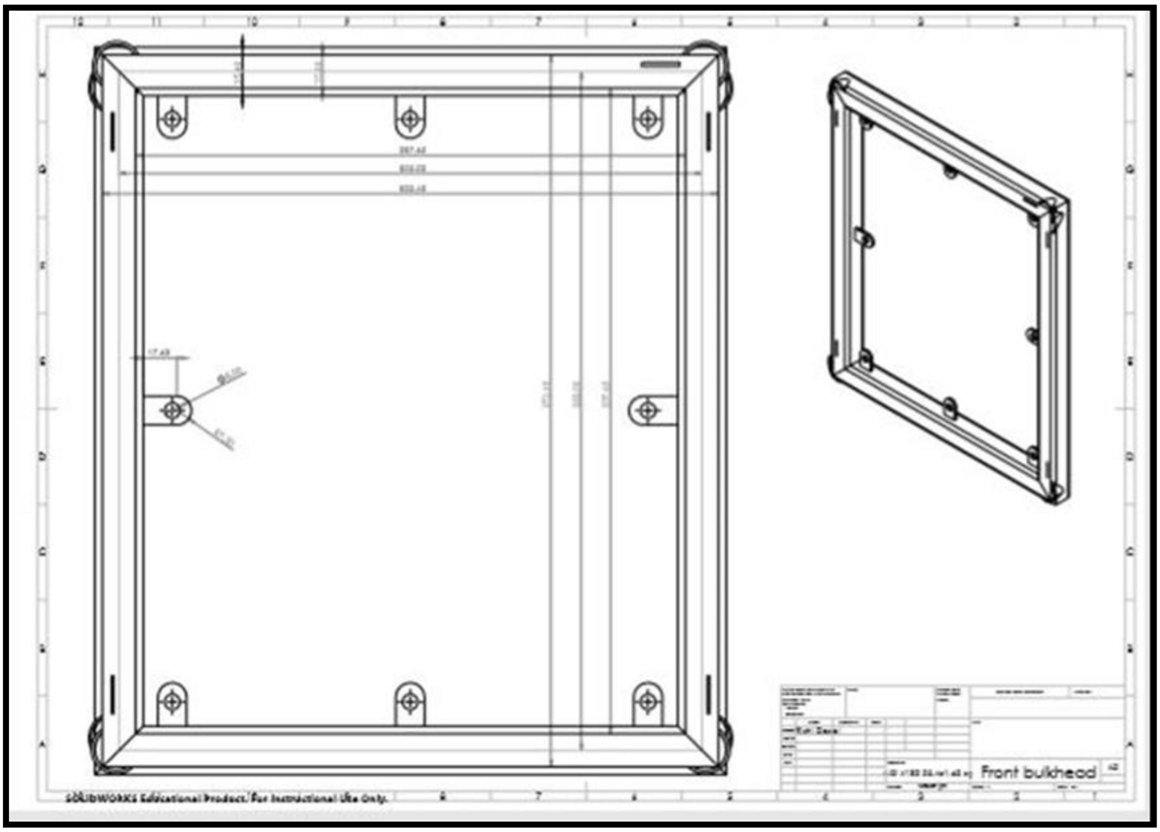


Fig –A3.2 (Front Bulkhead drawing, 1:1 scale, Similar was Rear bulkhead Fixture)

ISSN No:-2456-2165

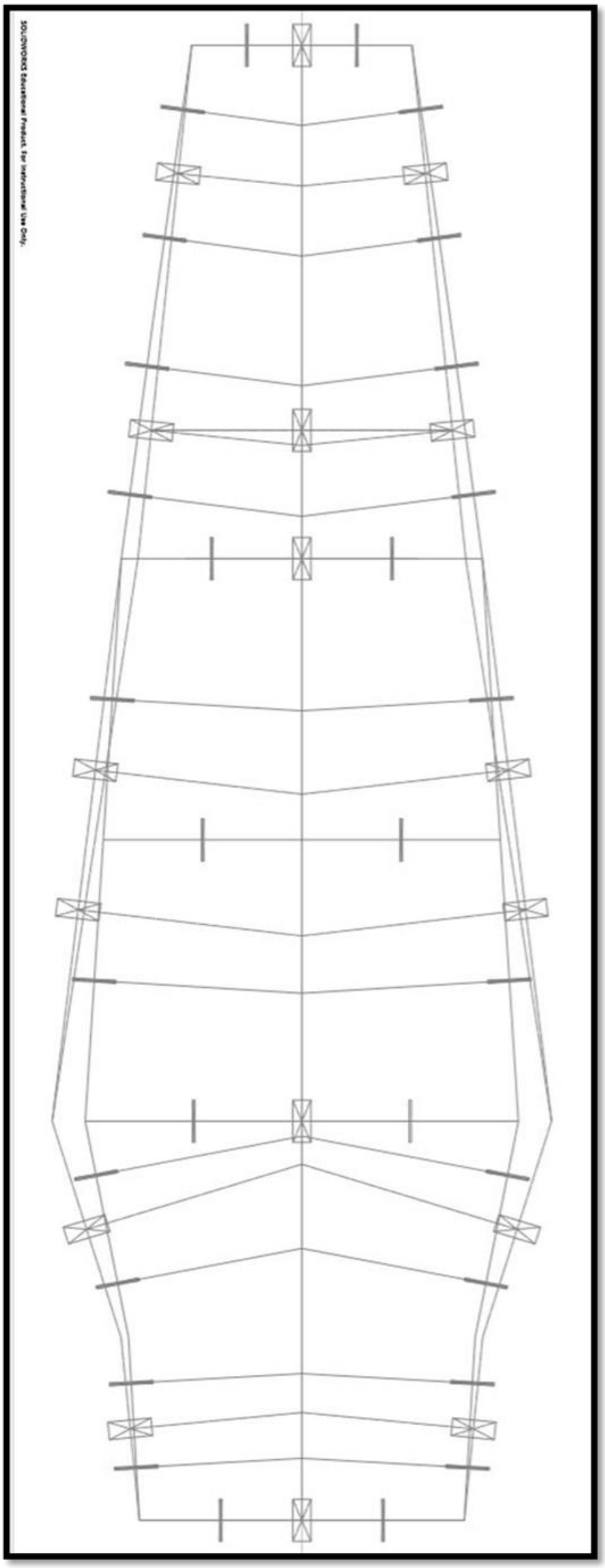


Fig – A3.3 (Chassis sketch Top view sketch for fixture table, 1:1 Scale)

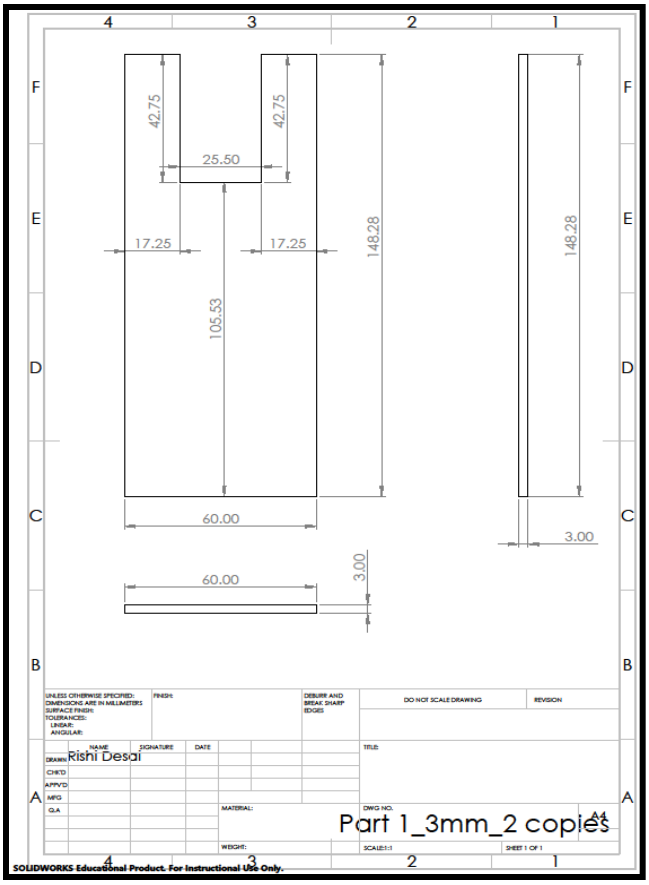


Fig – A3.4 (Chassis jigs drawing, laser-cut, 1:1 Scale)

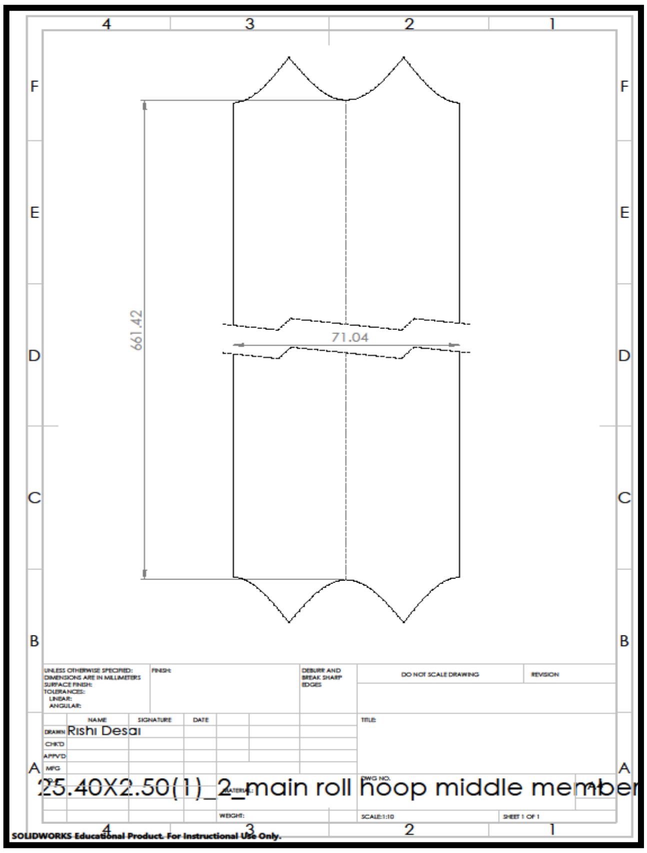


Fig – A3.5 (Tube profile flattened for tube notching, 1:1 scale)