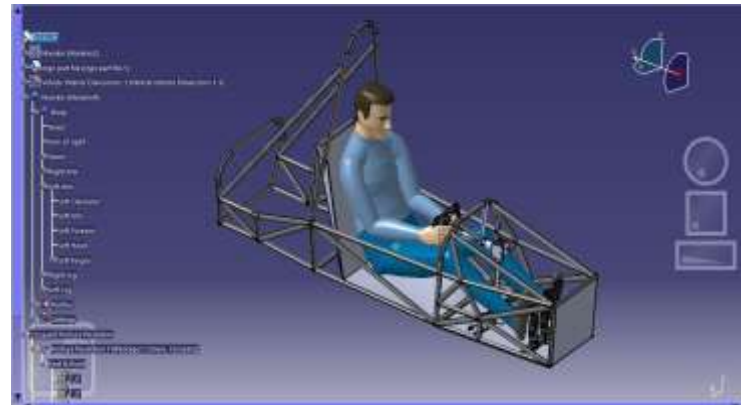# Design and Finite Element Analysis(FEA) of Formula Student Chassis

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Abstract – *This paper is a part of Design Report of **Shaurya Racing**. This Report and Analysis was presented during **Formula Bharat 2019**. Finite element stress analysis of chassis plays an important role during design stages. This paper focuses on Stress, Deformation analysis of the **FSAE** chassis using **Altair Hypermesh.** 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 of a chassis for **driver's safety** is always been a concern, especially for a race car. In this paper, few techniques are mentioned on how to analyze a formula student race car chassis in order to ensure its structural stability for driver's safety. Aim of this paper is to produce a clear idea about the **types of analysis** to be done on a formula student chassis with the amount of load or **G forces** to be applied on it using **Altair Hypermesh** software in order to make sure that the driver is safe inside the cockpit. The modelling of Chassis was done in **Solidworks 2018.**

Key Words: FSAE, Altair Hypermesh, FEA, Solidworks, 1-D Analysis, Ansys Workbench, G force.

INTRODUCTION

FSAE Chassis is a **Single Bracket** which hold all the essential assembly like **Paddle Assembly, Wheel assembly, Engine assembly, Steering assembly,** etc. and it has some cockpit space for diver. The chassis should be designed in order to withstand high amount of **G force**. As chassis being important system absorbing all the static and dynamic loads, the structure must withstand the stresses generated without much deformation.

The modeling of **FSAE Chassis** was done in **Solidworks.** The Chassis was designed in **3-D**; weldment feature was used to provide proper tube thickness and provide material properties. All the important rules were kept in mind that was important to manufacture chassis and to clear all the rules that were mentioned in **Rulebook- Formula Bharat 2K19.** The node to node triangular was provide in order to distribute the tensile and compressive force during loading.

To check the driver comfortness **RULA** and **REBA** Analysis was performed using **CATIA V6** to get the proper score so that we can assure that our chassis can accommodate 95th percentile driver.

Overall RULA score is 6 which means Driver is at medium Risk.

The chassis was designed in such a way that driver has **maximum comfort with proper view angle, lap angle, knee angle and seat angle**.

It was kept in mind that chassis is more "**TRIANGULATED**", So that Only **tensile and compressive** forces act and forces are distributed by node to node.

The chassis is designed in such a way that from side view only triangles are visible with no unsymmetrical member to ensures **maximum stability and proper load transfer**.

The tube material was chosen as **AISI1020**.

1. Inexpensive

2. Easy to work with no post weld heat treatment needed.

3. In case of **CrMo**, there was need to normalizing the structure after the   welding to prevent the formation of

**HAZ's** (Heat effected zone), which leads to complexity, higher cost and time consumption.

4.Easy weldability of **AISI 1020** then compare to CrMo.

Chassis Analysis

For the **FEA** Altair Hyperworks to perform various analysis. The **1-D** mesh of chassis skeleton with proper tube thickness and properties were selected to perform **1D Beam** Analysis.

As the **L>>>D** so, we have used 1D meshing to discretized model into the small cylindrical shape element to perform Finite Element Analysis.
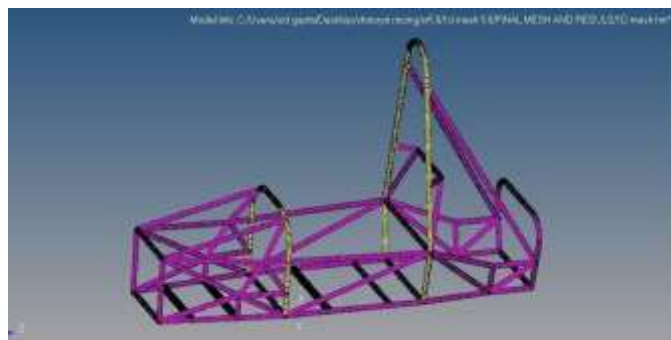
Finite element model information

Number of nodes:	**29930**
Number of elements:	**30310**

Analysis Performed:

1.Front Impact Analysis

2.Side Impact Analysis

3.Rear Impact Analysis

4.Bump Analysis

5.Front Axial Analysis

6.Rear Axial Analysis

7.Front Torsional Analysis

8.Rear Torsional Analysis

9.Lateral Bending Analysis

10.Roll over Analysis

11. Validation of Torsional Rigidity of chassis
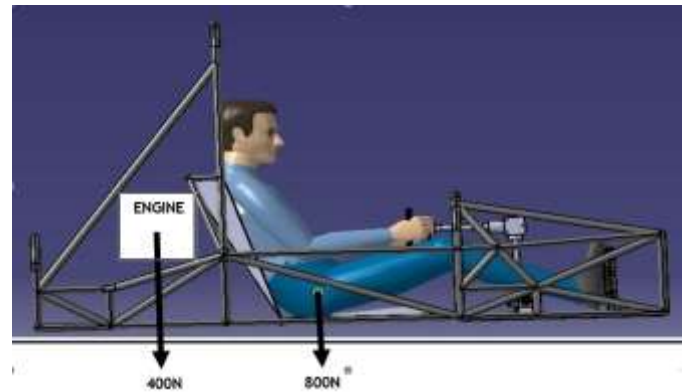
The two major heavy thing in car is engine(**40kg**) and driver(**80kg**). For all analysis we have pre applied the weight of driver and the engine.

Front Impact Analysis

Let the mass of car be **300KG.**

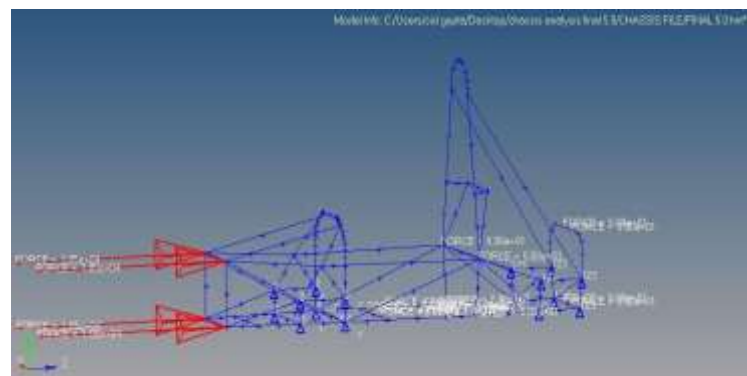Initial velocity be **28m/s**

Final velocity at impact be **0m/s**

So using Newton law of Motion

V=U +a*t (t=0.4s)

A=70m/s^2

Force =Mass*Acceleration Force=21000N
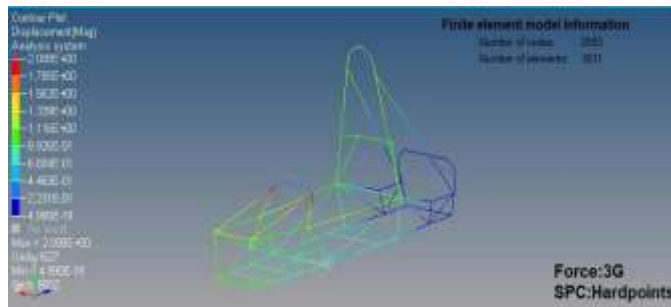
So for sudden impact Stress doubles 2F=42000N

Load and SPC

Force:42000N(2F)
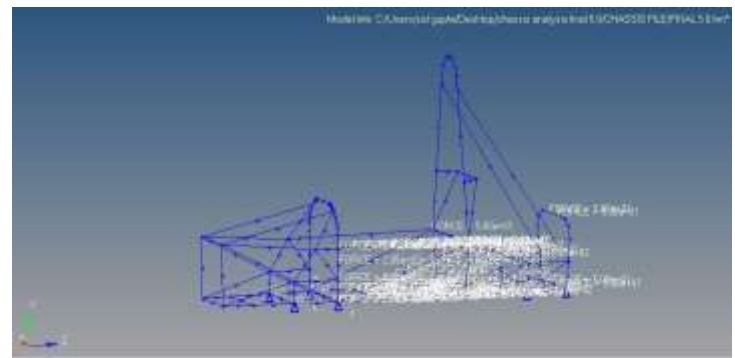
Single Point Constraint:

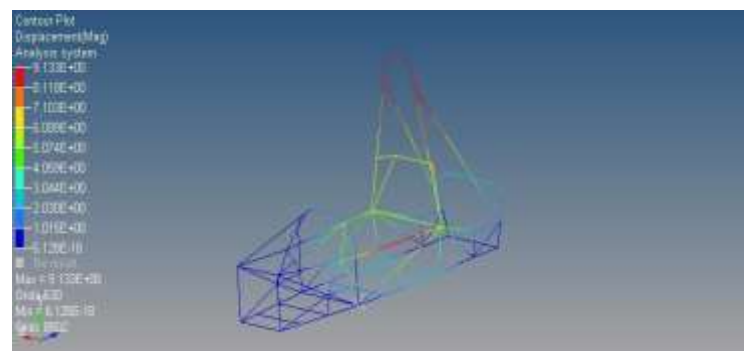Rear Suspension point (123)

Front Suspension point (2)

Displacement

Stress

Factor of Safety was Found to be 2.

Maximum Displacement was 2 mm.

Maximum Stress was 146 MPa.

Side Impact Analysis

Let the mass of car be **300KG.**

Initial velocity be **28m/s**

Final velocity at impact be **0m/s**

So using Newton law of Motion

V=U +a*t (t=0.4s)

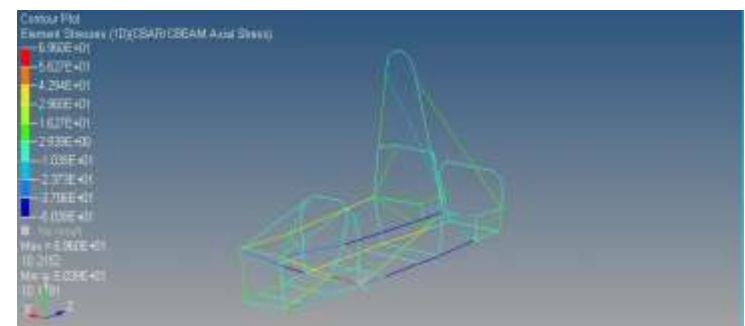**A=70m/s^2**

Force =Mass*Acceleration Force=21000N

So for sudden impact Stress doubles 2F=42000N

A load of 42000N was applied uniformly on the side impact structure

Force:42000N(2F)

SPC: Rear Suspension point (2)

Front Suspension point (123)

Load and SPC

Displacement

Stress

Factor of Safety was Found to be 4.34.

Maximum Displacement was 9 mm.

Maximum Stress was 69.6 MPa.

Rear Impact Analysis

Let the mass of car be **300KG.**

Initial velocity be **28m/s**

Final velocity at impact be **0m/s**

So using Newton law of Motion

V=U +a*t (t=0.4s)

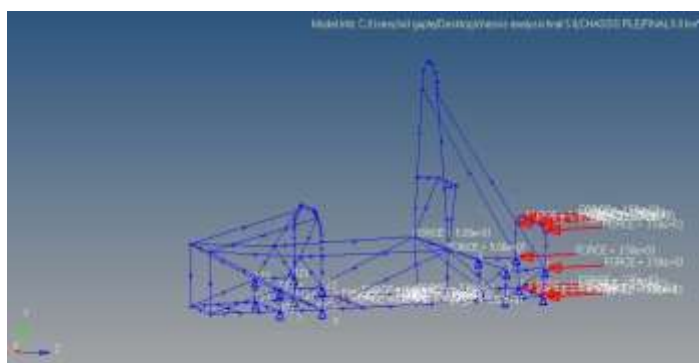**A=70m/s^2**

Force =Mass*Acceleration Force=21000N

So for sudden impact Stress doubles 2F=42000N

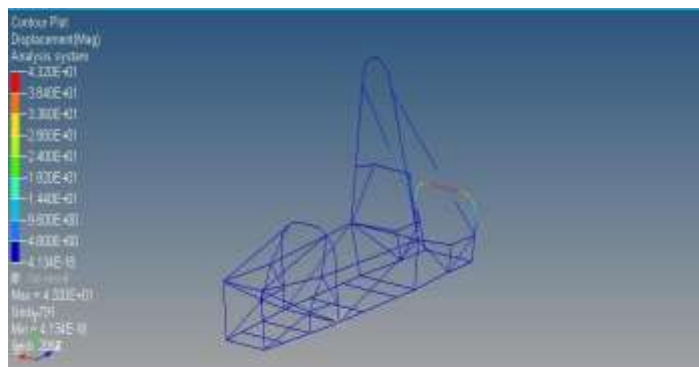**A load of 42000N was applied point uniformly distributed load on the rear bulkhead of the chassis.**
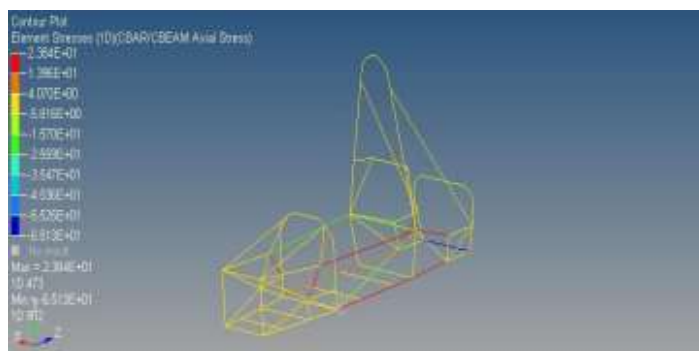
Force:42000N(2F)

SPC: Rear Suspension point (2)

   Front Suspension point (123)

Load and SPC

Displacement

Stress

Factor of Safety was Found to be 12.
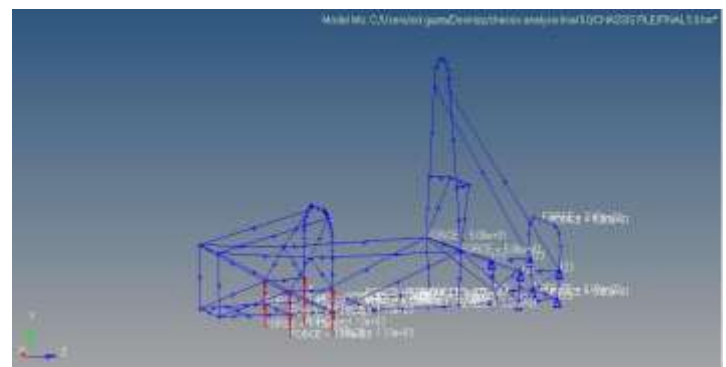
Maximum Displacement was 43 mm.

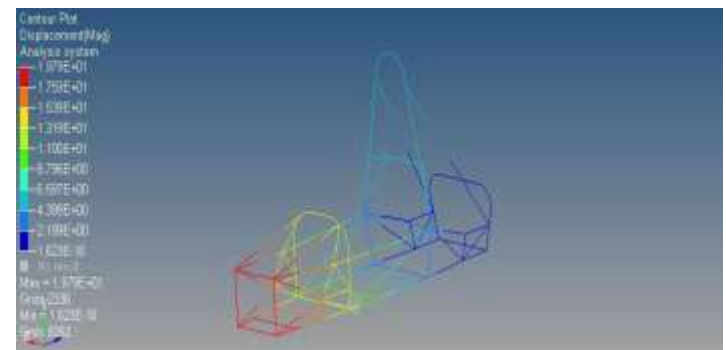**Maximum Stress was 23.8 MPa.**

Bump Analysis

The forces exerted on the suspension mounting point due to bump on track so total of 3G load was applied on the Front Suspension point. So total there are 8 Suspension mounting point so each node was applied with 1130N
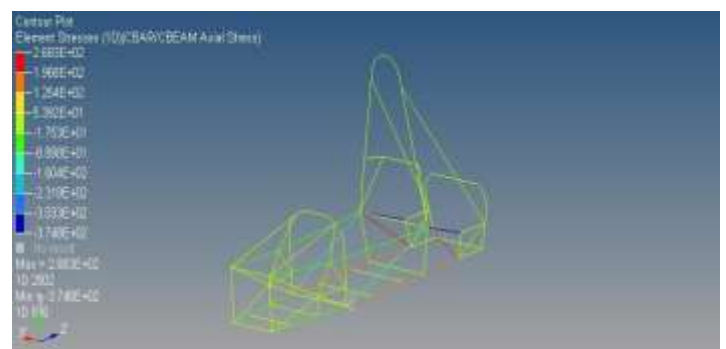
   Force:4520N in opp. direction

   SPC: Rear Suspension point (123)

Load and SPC

Displacement

Stress

Factor of Safety was Found to be 1.19.

Maximum Displacement was 19.79 mm.

Maximum Stress was 268.3 MPa.

Front Torsional Analysis

For the front torsional impact analysis, the 3G was used

1G=300*10=**3000N**

So **3G=9000N**

For extra Safety **10000N**

The rear hard point was made fix and then the opposite force was applied on the front left and right hard point

Each hard point was applied with the force of 2500 Newton each.

SPC: Rear Suspension point (123)

Factor of Safety was Found to be 1.2.

Maximum Displacement was 47.5 mm.
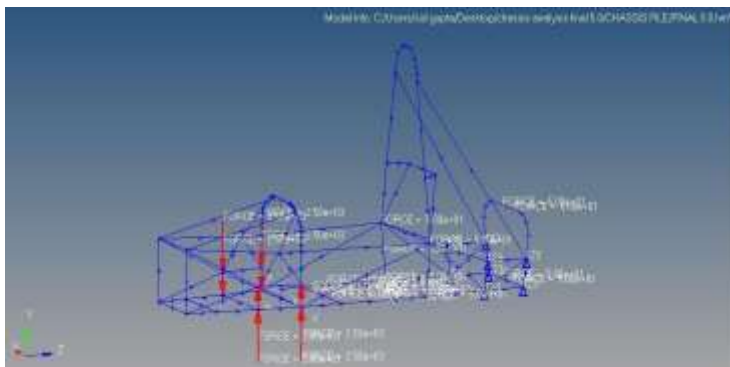
Maximum Stress was 250.6MPa.

Rear Torsional Analysis

For the Rear torsional impact analysis, the 3G was used

1G=300*10=3000N

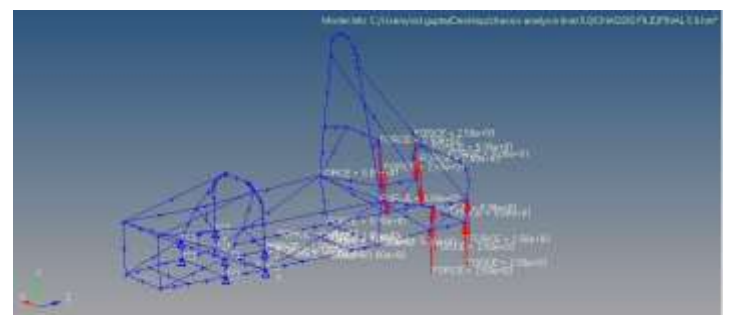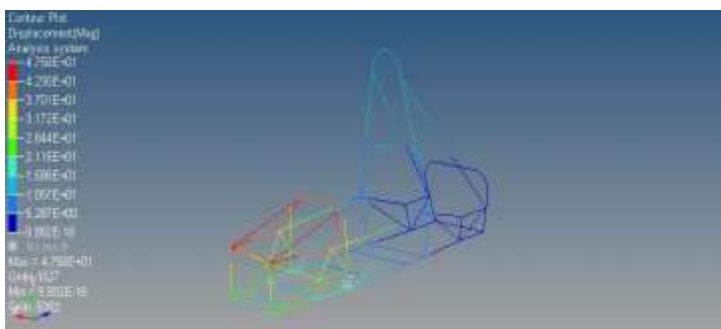So 3G=9000N

For extra Safety 10000N

The front hard point was made fix and then the opposite force was applied on the front left and right hard point. Each hard point was applied with the force of 2500 Newton each.
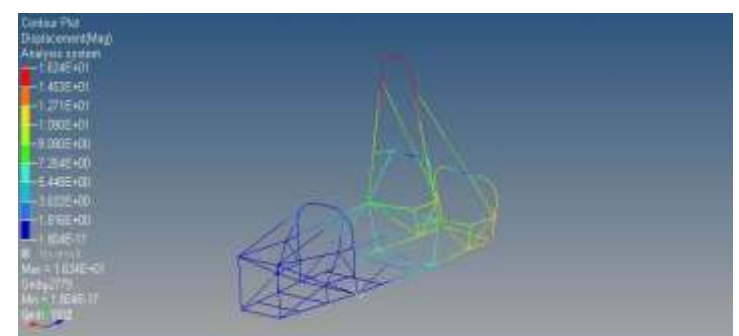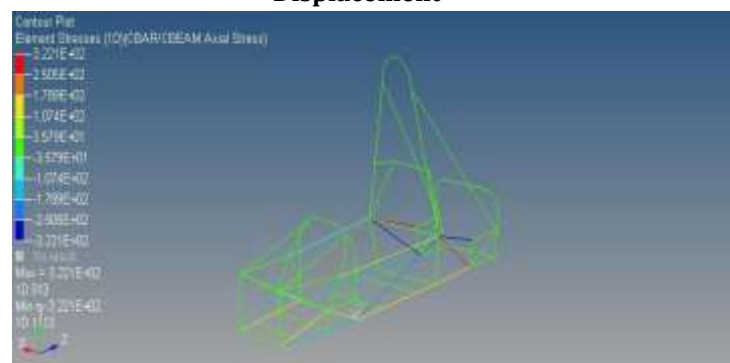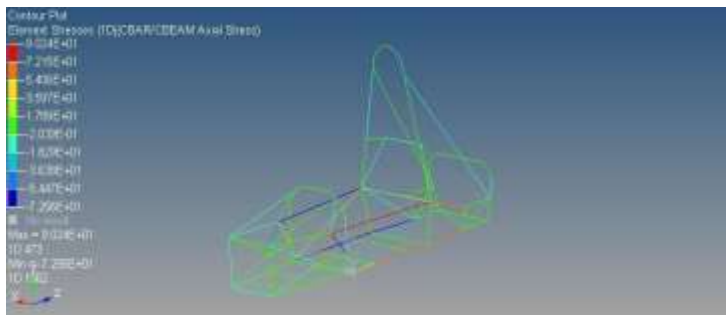

Load and SPC

Displacement

Stress

Force:10kN (one side)

Load and SPC

Displacement

Stress

Force:10kN (one side)

SPC: Front Suspension point (123)

Factor of Safety was Found to be 1.21.

Maximum Displacement was 16.3 mm.

Maximum Stress was 251.4MPa.

Lateral Bending

The Force is given by mv^2/r
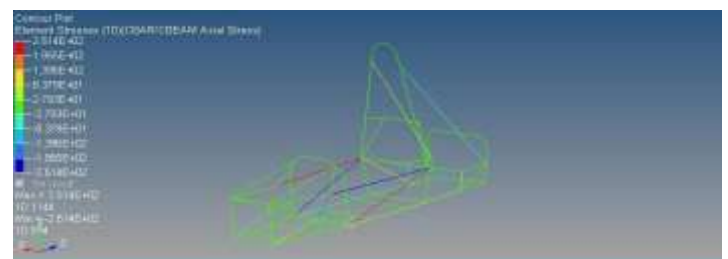
Let v= 50km/hr.=14m/s

Mass =300kg

Critical Radius of track is 15.25 m

So on solving we get **F=3855.75N**

 Factor of Safety=max stress/working stress

So max stress=16000N (Taking **FOS=4**)

The hard points were made fixed then the force was applied on the **side impact structure** uniformly.

Load and SPC

Displacement

Stress

 Force:16kN

 SPC: Front Suspension point (123)

 Rear Suspension point (123)

 Factor of Safety was Found to be 3.324.

 Maximum Displacement was 8.6mm.
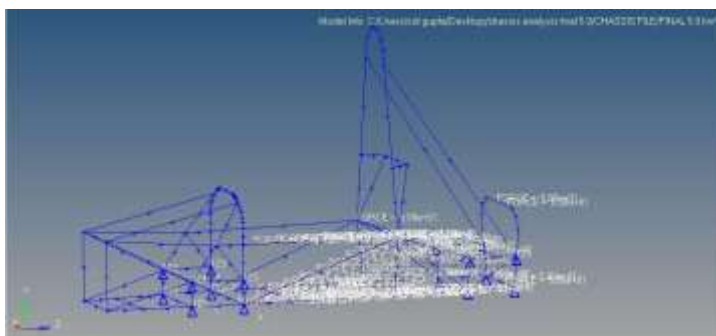
 Maximum Stress was 90.24MPa.

Roll Over Analysis

This case occurs when the is the unbalance between the CG and Track.

In this case a local coordinate system was created w.r.t. the origin then it was rotated 45^0 to apply the load.
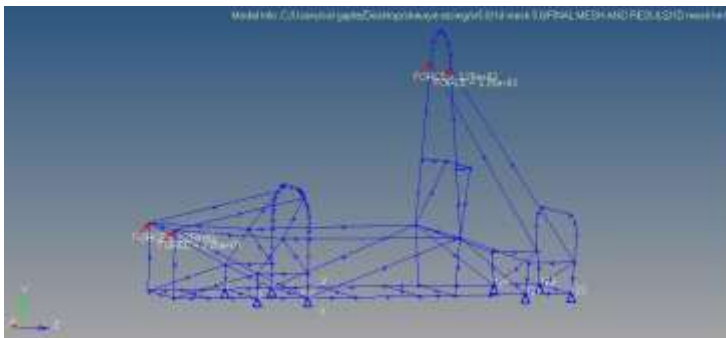
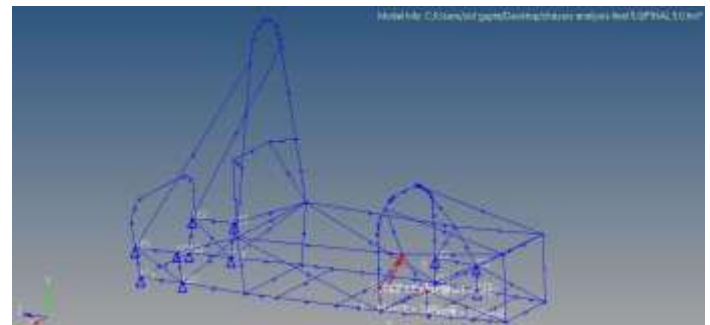A total of **3.5G** force was applied on the front bulkhead and the main hoop.

 Force:10kN

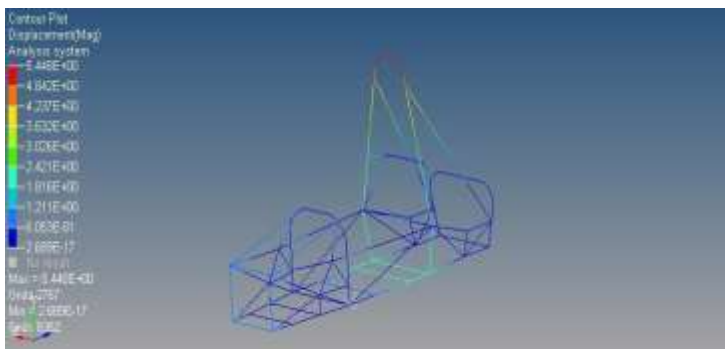SPC: Front Suspension point (2) Rear Suspension point (123)

Load and SPC

Load and SPC

Displacement
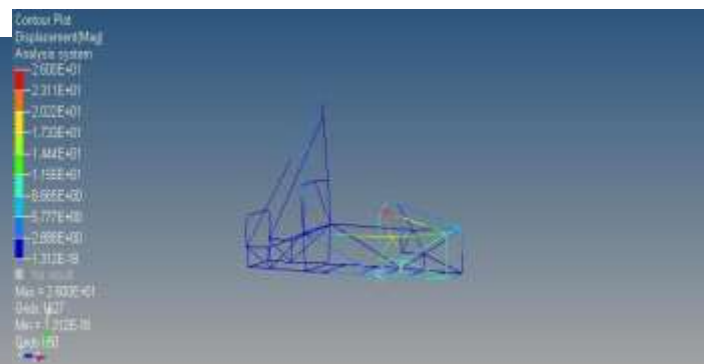
Displacement

Stress
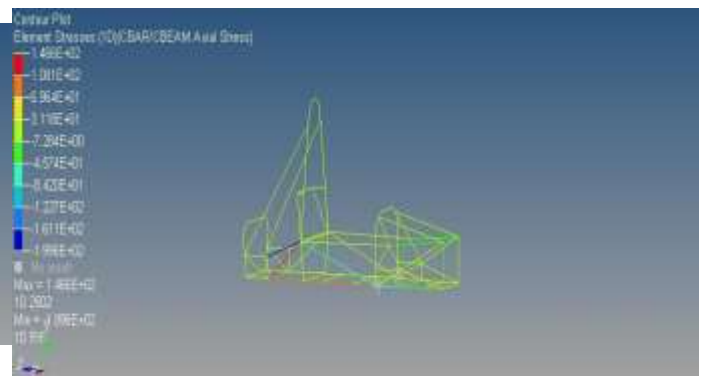
Factor of Safety was Found to be 5.57.

Maximum Displacement was 5.44mm.

Maximum Stress was 53.8MPa.

Front Damper Analysis

For the Front Damper Analysis, the exact dimension along with angles were used to applied force in the same direction as the actual car damper has.

The local coordinate system was used to apply the force. The rear hard point of the chassis was made fixed.

Front A Arms 5800N

Front Spring 4900N

Stress

Factor of Safety was Found to be 2.1.

Maximum Displacement was 26.7mm.

Maximum Stress was 146MPa.

SPC: Front Suspension point (123)

Rear Suspension point (123)

Rear Axial A-Arms Analysis

For the Front Damper Analysis, the exact dimension along with angles were used to applied force in the same direction as the actual car damper has.

The local coordinate system was used to apply the force. The front hard point of the chassis

Rear A Arms 1575N

Rear Spring 8660N

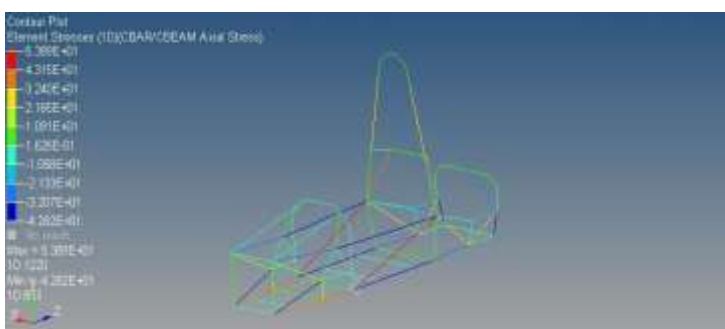**Load and SPC**

Displacement

Stress

Factor of Safety was Found to be 12.6.

Maximum Displacement was 3.48mm.

Maximum Stress was 23.8MPa.

SPC: Front Suspension point (123)

   Rear Suspension point (123)

Validation of Torsional Rigidity of chassis

Setup
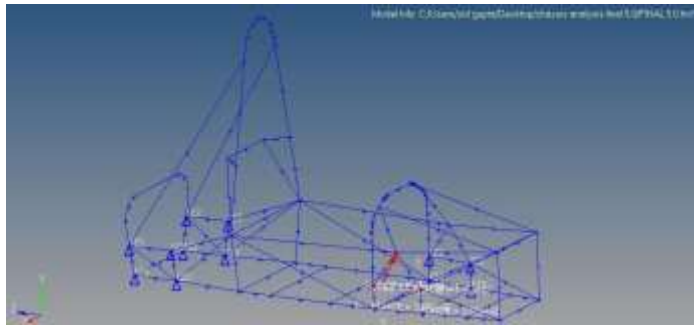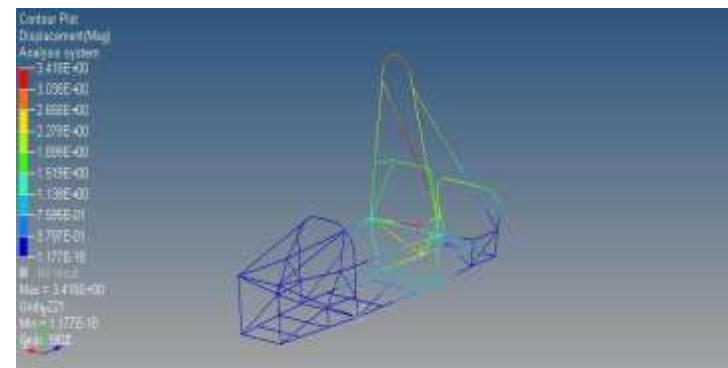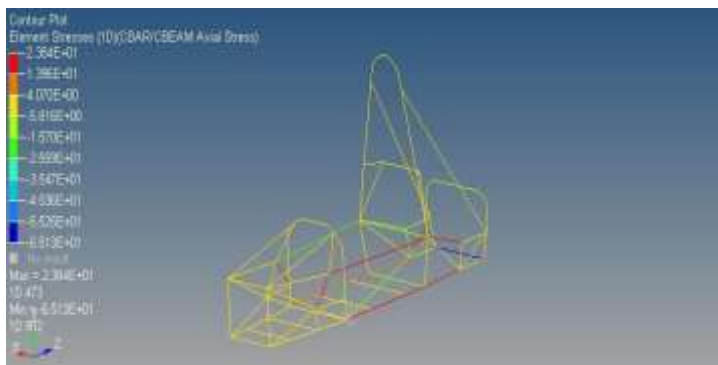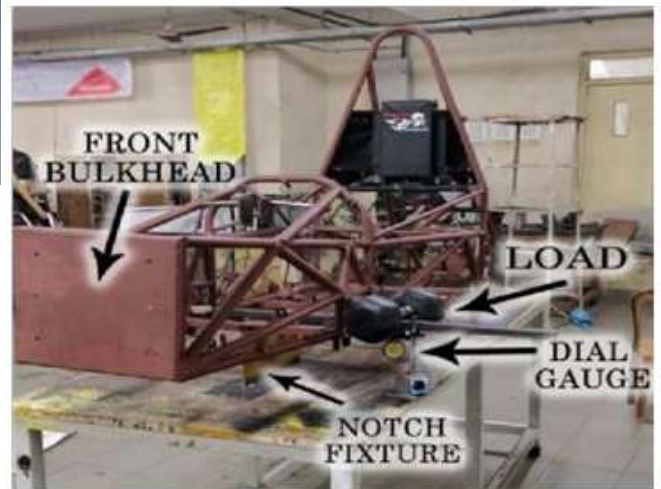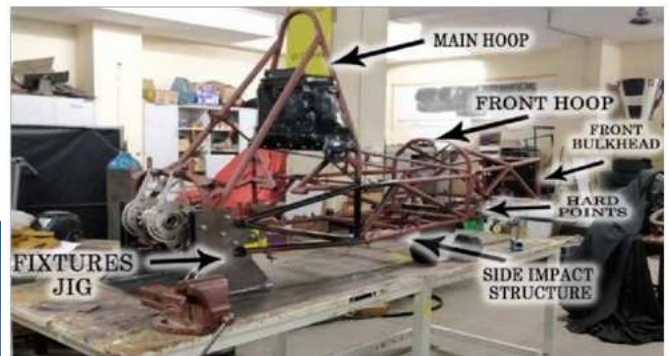
Results

Table 1: Deflection is Measured at a Point Which is 310mm from the Front Bulkhead

Load (kg)	Deflection (mm)	Angle of Twist Taking L=355mm (degrees)	Toque Applied (Nm)	Torsional Rigidity (Nm/deg)	Average Torsional Rigidity (Nm/mm)	FEA Angle of Twist	FEA Torsional Rigidity	FEA Avg Torsional Rigidity
12.5	0.34	0.058151	43.53188	748.6013		0.0393	1107.681	
15.5	0.36	0.061572	53.97953	876.6952		0.04873	1107.727	
18.5	0.51	0.087226	64.42718	738.6198	758.6025	0.05816	1107.757	1107.748
21.5	0.62	0.10604	74.87483	706.0997		0.06759	1107.78	
24.5	0.69	0.118012	85.32248	722.9965		0.07702	1107.796	

According to Table 1, the value of torsional rigidity is 31.5% less than that found out by FEA. This is because our notch fixture which was resisting the bending of the chassis only between the front hoop (Point where notch fixture is attached to the chassis) and the rear hard points. The chassis is bending between the front hoop and front bulkhead because of its own weight.

Table 2: Deflection is Measured at a Point Which is between Front Lower Hard points

Load (kg)	Deflection (mm)	Angle of Twist Taking L~355mm (degrees)	Toque Applied (Nm)	Torsional Rigidity (Nm/deg)	Average Torsional Rigidity (Nm/mm)	FEA Angle of Twist	FEA Torsional Rigidity	FEA Avg Torsional Rigidity
12.5	0.31	0.05302	43.53188	821.0466		0.03989	1091.298	
15.5	0.39	0.066703	53.97953	809.2571		0.04946	1091.377	
18.5	0.45	0.076965	64.42718	837.1025	840.2017	0.05904	1091.246	1091.318
21.5	0.51	0.087226	74.87483	858.3959		0.06861	1091.311	
24.5	0.57	0.097488	85.32248	875.2065		0.07818	1091.359	

According to Table 2, the value of torsional rigidity is 23% less than that found out by FEA. This is because placing the notch fixture with 100% accuracy and a perfectly rigid fixing of the rear is not possible in real life.

Table 3: Deflection is Measured at a Point Which is in Between of the Cockpit

Load (kg)	Deflection (mm)	Angle of Twist Taking L~355mm (degrees)	Toque Applied (Nm)	Torsional Rigidity (Nm/deg)	Average Torsional Rigidity (Nm/mm)	FEA Angle of Twist	FEA Torsional Rigidity	FEA Avg Torsional Rigidity
12.5	0.09	0.015393	43.53188	2828.05		0.01285	3387.695	
15.5	0.11	0.018814	53.97953	2869.185		0.01593	3388.545	
18.5	0.13	0.022234	64.42718	2897.663	2709.13	0.01901	3389.12	3388.373
21.5	0.17	0.029075	74.87483	2575.189		0.0221	3388.001	
24.5	0.21	0.035917	85.32248	2375.562		0.02518	3388.502	

According to Table 3, the value of torsional rigidity is 20% less than that found out by FEA. This experimental value deviates minimum from the FEA value because the point at which the measurement was taken is farthest from our jig fixture and notch fixture so errors are less.

Table 4: Deflection is Measured is Below the Main Hoop

Load (kg)	Deflection (mm)	Angle of Twist Taking L~355mm (degrees)	Toque Applied (Nm)	Torsional Rigidity (Nm/deg)	Average Torsional Rigidity (Nm/mm)	FEA Angle of Twist	FEA Torsional Rigidity	FEA Avg Torsional Rigidity
12.5	0.05	0.008552	43.53188	5090.489		0.003894	11179.22	
15.5	0.07	0.011972	53.97953	4508.719		0.004828	11180.51	
18.5	0.09	0.015393	64.42718	4185.514	4265.58	0.005763	11179.45	11180.11
21.5	0.11	0.018814	74.87483	3979.837		0.006697	11180.35	
24.5	0.14	0.023945	85.32248	3563.343		0.007631	11181.03	

According to Table 4, the value of torsional rigidity is 61.8% less than that found out by FEA. This experimental value deviates most from the FEA value because in FEA we had fixed the rear hard points but in real life the rear hard points will still have some rotational movement because of torsion and as there will be small gaps between the plates of the jig fixtures, deflection would be more and thus torsional rigidity will be less.

CONCLUSIONS

So, from all the analysis we can conclude that the result generated from stress data. All the values were below the Yield strength of the material i.e. **310Mpa**. So we can conclude that the will not be plastic deformation for the material. The Material and the Design of the chassis is safe.

REFERENCES

[1] VALIDATION OF TORSIONAL RIGIDITY OF FSAE CHASSIS DEVESH ANAND, ANUJ GUPTA & JEYANTHI SUBRAMANIAN.
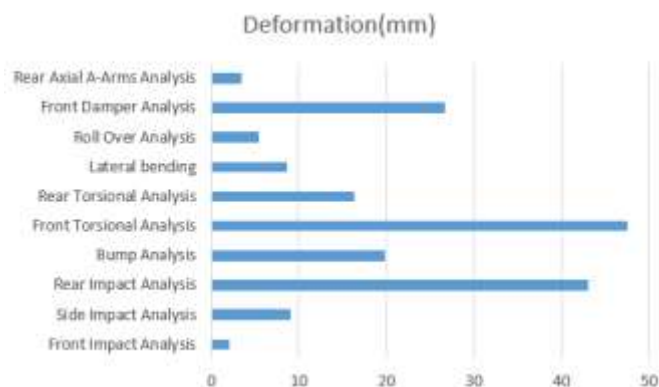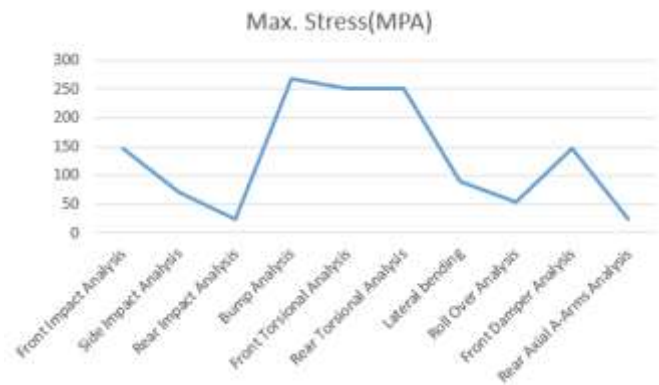
[2] Cost Effective Manufacturing and Optimization of the Formula Student Nose Cone Antim

[3] Solidworks

[4] Altair Hypermesh

[5] https://www.youtube.com/watch?v=Vb0i8VbhYaY

[6] https://www.youtube.com/watch?v=0cZO7nly_E4

[7]   https://www.youtube.com/watch?v=X8qE8FxuxyU

BIOGRAPHIES

Antim Gupta

The author is an undergraduate student of Mechanical Engineering at VIT-Chennai and Senior Designer Engineer at **Shaurya Racing**, Chennai, India. He is Solidworks certified at a level of **Expert** by **Dassault Systemes.**