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Design and Analysis of Formula SAE Chassis

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Abstract—*The design of formula chassis [9] involves optimization between more performance parameters than other automobile chassis types. Besides achieving high torsional stiffness and strength, an efficient design should accommodate weight reduction and ease of manufacturing. This paper introduces a torsionally, laterally and longitudinally stiff chassis design, which has been drafted ergonomically to accommodate anthropomorphic models of the tallest (95th percentile male) and the smallest (5th percentile female) driver. Binocular visions for the entire range of drivers were simulated to ensure sufficiently large domain of vision for them. A variety of materials were considered and a comprehensive comparison was drawn amongst the material properties to select the material which could settle with the structural requirements. Hence the design was thoroughly analyzed, through simulation for the various load distributions that a Formula SAE car may encounter and subsequent deformations. This paper also introduces the design of a crumble zone, and an equally comprehensive structure and material selection for impact attenuator design.*

The torsional rigidity calculated for the model was 4026.785 Nm/deg and the minimum factor of safety obtained amongst all load distributions was 9.16 for a weight of 55.78kg. In this context, it is obligatory to mention that though the chassis design developed corresponds with the rules of Formula SAE, the engineering aspects of the conditions specified in the rules have also been thoroughly explored to induce maximum augmentation in structural strength and rigidity.

1. INTRODUCTION

Generally formula one cars are designed to withstand 3.5 g bump, 1.5 g braking and 1.5 g lateral forces [3].

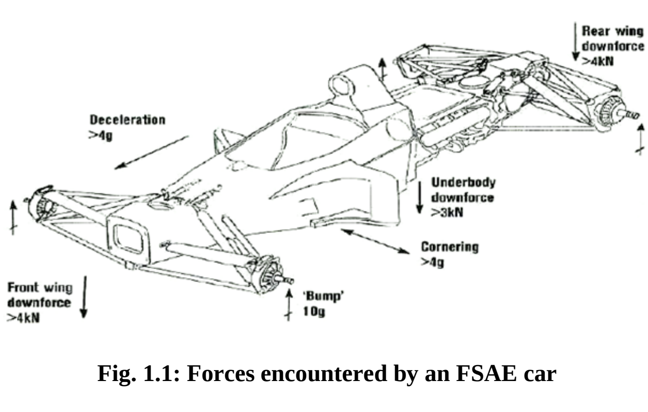

Fig. 1.1: Forces encountered by an FSAE car

1.1 Structural requirements of Formula chassis

- Should be strong enough to protect the driver from external intrusion.
- Torsional stiffness should be enough to avoid angular flexing.
- Chassis should be rigid enough to avoid longitudinal and lateral flexing [2].
- Rigidity is also important to maintain precise control over suspension geometry, i.e. to maintain contact between the wheels and race road surface.
- Light Weight
- Exhibiting proper safety factors

1.2 Crumple Zone

The crumple zone [8] (also called crush space) is a structural feature mainly used in automobiles and recently incorporated into railcars. Crumple zones are designed to absorb the energy from an impact by deforming. They are usually placed at the front and rear of a car as these are the locations of most impacts.

The ability of a crumple zone to collapse when a force is applied to it helps to increase the time taken for the vehicle to come to a complete stop. Since acceleration is inversely proportional to time, increasing the time taken causes the magnitude of deceleration of the vehicle to be reduced. The crumple zone is designed to reduce the magnitude of deceleration of a vehicle, so that the force exerted on the vehicle is also reduced.

2. THEORETICAL MODELLING AND SIMULATION

2.1 Selection of Space Frame Chassis

For the following reasons, space frame chassis was selected –

- Ladder frame structure could not be selected because of lack of diagonal bracing, because of which it can easily be twisted along its length. For making the chassis more stiff and more rigid, extra members have to be added which in turn increases the weight. Moreover it takes time to align and weld all the individual members together.

- Backbone chassis was rejected because it does not offer effective protection from side impact and offset crash. As the chassis rails are closer together to fit between the seats i.e. the loss of stiffness. Also its manufacturing is costly and complicated.
- Carbon fibre monocoque structure was excluded from the design, because once it is cast, the design freezes and cannot be adjusted to accommodate the changes in mounting points. The thinner material used in monocoque designs compared to tubular-based structures is easier to buckle. Also carbon fibre monocoque is neither economical nor easy to repair.

2.2 Selection of Material for Tubular Space Frame

The material which is used for chassis construction should possess the following material characteristics -

- Light Weightiness – this enables the formula car to extract maximum acceleration out of the limited engine power allowed.
- Economic Effectiveness
- Crashworthiness- It should be capable of withstanding the loads and absorbing the corresponding energy that are likely to appear in Formula 1 cars, in order to keep the driver safe.

Thus a material with the highest possible tensile strength, yield strength and shear modulus was required, which at the same time was economical and not bulky. Such a combination of properties is not possible, hence an optimization was sought.

A vast variety of materials ranging from Aluminium and its alloys to steel and its alloys, which are industrially employed in the manufacture of various kinds of chassis structures were studied. Aluminium alloys and alloy steels were further studied in detail to draw a comprehensive comparison between desired properties.

Upon comparing various alloy steels and grades of stainless steels, it was observed that stainless steels in general have less tensile strength and yield strength as compared to alloy steels. Next, the properties of alloy steels and Aluminium alloys were studied and compared. A broad comparison between AISI 304, AISI 4130, AISI 4140, AISI 440C, Aluminium 2024-T3 and Aluminium 7075-T6 has been shown in table 2.1.

Aluminium offers the best solution to the problem of weight reduction, however it is not as rigid as steel. Also the ratio of stiffness to weight is the same for both, which implies that to attain the stiffness and rigidity offered by steel frame, an Aluminium frame must weigh the same as the steel frame. It is also more expensive as compared to steel.

Upon further comparing costs, AISI 4130 i.e. Chromoly was chosen for Chassis design because of its high hardness and an optimization between yield strength, elongation, ultimate tensile strength, shear modulus, specific strength and cost. Moreover forming can be performed in annealed condition, and all the commercial welding techniques can be used to weld it. It can be forged between 954^0C-1204^0C, which is a feasible temperature, hot working and cold working can be performed through conventional methods. Thus for a DFM Structure, Chromoly appears to be the best alternative.

Table 2.1: Comparison in material properties

Property	AISI 4130	AISI 4140	AISI 440C	AISI 304	AA 7075 - T6	AA 2024 - T3
Density (lb/in^3)	0.284	0.284	0.275	0.29	0.101	0.101
UTS* (ksi)	81.2	95.0	110-286	85	76	70
0.2% YS (ksi)	66.7	60.2	65-280	35	67	50
Elongation (%)	21.50	25.70	14	55	11	18
Hardness (Brinell)	217	197	97	123	150	120
Shear Modulus (ksi)	11600	11600	11603	12500	10600	10400
Poisson's ratio	0.27-0.30	0.27-0.30	0.29	0.29	0.33	0.33

*UTS: - Ultimate Tensile Strength
*YS: - Yield Strength

2.3 Selection of Material for Impact Attenuator

The factors that these were measured on were:

- Cost: with any engineering project, budget constraints cannot be over looked.
- Weight: overall design is meant to keep the car lightweight to not take away from its speed.
- Reliability: this is essential to how the design will perform.
- Safety: the attenuator will be taking high impact forces and should pose no threat to the driver.
- Feasibility: how likely the concept could be implemented into the attenuator.

Numerous types of crumble zones were further compared comprehensively, to obtain the most suitable design for impact attenuator.

- **Airbag Design**

Although utilizing an airbag would be lightweight and reliable, the cost could not be overlooked. It would cost a good deal of money to obtain the necessary parts while part replacement would be expensive. This idea was rejected after doing further detailed analysis of the pressure requirements of the impact and comparing these to the limitations found in common airbags.

- **Crimped Metal Lattice Design**

Next, a crimped metal lattice design was investigated. This concept involved separating several rows of metal plates with crimped metal that would absorb much of the force by crushing upon impact. A main concern of the design was the ability to be rebuilt after impact due to the intensive time involvement in fabrication.

- **High Impact Foam Design**

Compared to other materials, foam provides a lightweight and cost friendly material for impact absorption. The foam was evaluated as being one of the two safest concepts and the ability to easily manipulate it in any size or shape necessary made the use of the foam very feasible.

- **Honeycomb Design**

Its light weight and low price make the material appealing in that it has the ability to be created relatively cheap and also easily replaced. Honeycomb also has its ability to reduce impact will not vary a great deal due to design and construction variance hence provide safety and reliability.

- **Rubber Bumper Design**

Rubber favored in terms of weight, cost, and feasibility as it is a common material that can be made to fit into the proper dimensions. Problems were foreseen with this material because its elasticity is small compared to the other designs. Thus, it would transfer too great an amount of force to the body of the car rather than absorbing it on its own.

- **Final Selections**

After careful consideration of all previously mentioned concept designs, the group decided to look further into two of the preliminary models. The two models being the honeycomb and foam ideas. Both of the selections scored the highest overall on our decision matrix chart. Each has great energy absorbing properties while being cost effective and lightweight.

For better results we considered Aluminum Honeycomb as material.

3. RESULTS AND DISCUSSION

3.1 Chassis Design

Impact attenuator is depicted in grey, while AISI 4130 is depicted in red. Considering the origin at the bottommost, rightmost point on the chassis, when it is viewed from the front of front bulkhead, the following calculations were made:-

- Centre of gravity :- Gx= -79.193 mm, Gy=-2.619 mm, Gz= 118.788 mm
- Principal Moments/G :- M1=10.779 $kgxm^2$, M2=49.44 $kgxm^2$, M3= 50.917 $kgxm^2$
- Mass of chassis :- 55.785 kg
- Calculation of torsional rigidity: - Torque = 360 Nm, displacement ($\Delta z1=\Delta z2$)= 0.395 mm, Length L = 253.1485 mm. By

$$K = \frac{FL}{\tan^{-1}\frac{\Delta z1 + \Delta z2}{2L}}$$

Torsional Stiffness (K) = 4026.7853 Nm/deg.

The driver's compartment has been ergonomically [13] designed; Fig. 3.1, which is a CAD drawing of the driver's compartment, illustrates the position of the head, torso and pelvis in this chassis design. It corresponds to a comfortable driving posture.

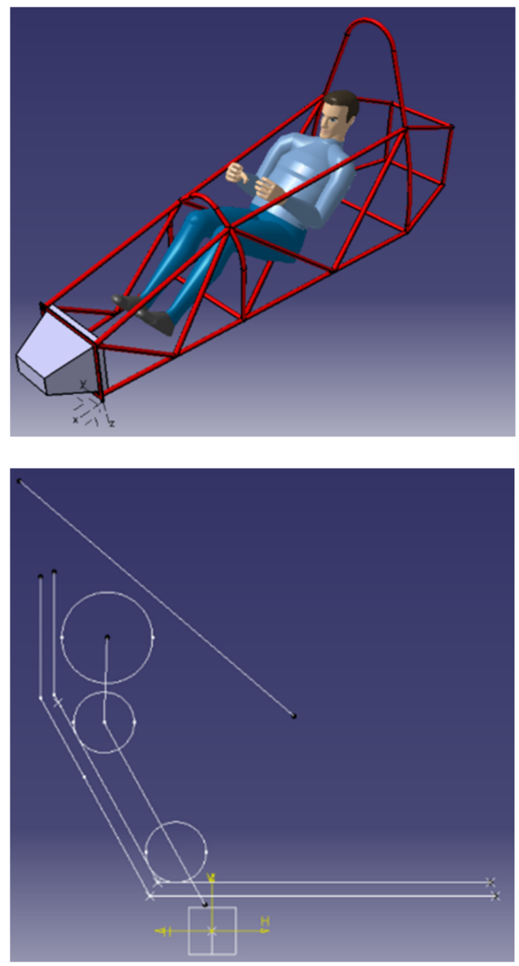

Fig. 3.1: Chassis design (Left) and sketch of driver's compartment which shows an ergonomically comfortable body posture (Right)

3.2 Simulation of Driver's Position

Chassis has been ergonomically designed using anthropomorphic models, to accommodate drivers having statures from 5th percentile female to 95th percentile [12] male. Figures 3.2 compares the binocular vision which is received by the maximum (95th percentile male) and minimum (5th percentile female) statures which can be accommodated. As can be observed from the simulation, both and hence the entire range of statures in between receives sufficient domain of vision.

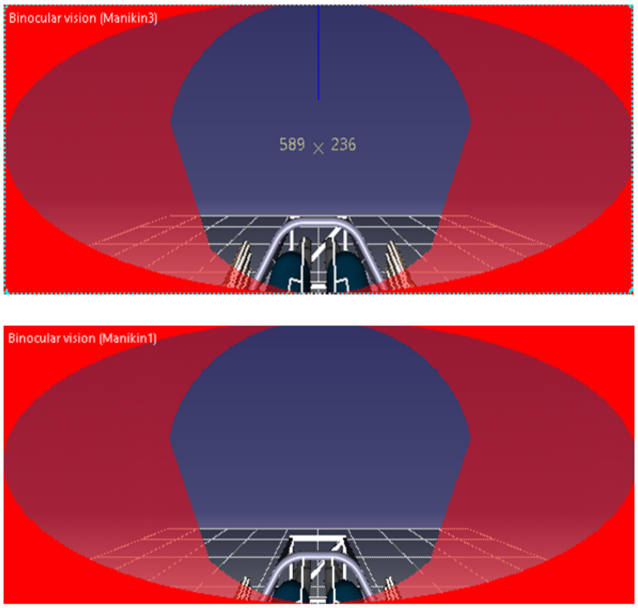

Fig. 3.2: Binocular vision for 5th percentile female (Left) and 95th percentile male (Right).

3.3 Simulation and Analysis

Linear tetrahedral elements were used for meshing [11] in CATIA V5, which was followed by application of static loads which correspond to real life forces which appear on formula frames [7]. The chassis was also tested for a frequency excitation which appears from the engine and transmission.

- **Bump Test**

For a bump test, the chassis is assumed to behave like a cantilever beam, it was clamped at the location of rear suspension mounts and a distributed force of 5390 N was applied at the base between the front bulkhead and the front hoop as has been shown in Fig. 3.3.

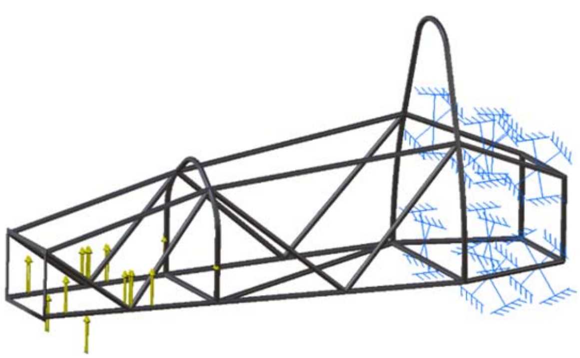

Fig. 3.3: Boundary conditions (Bump test)

Blue color depicts clamping and distributed force is shown in yellow. This was followed by simulation, after which a Von-Mises stress profile and a displacement graph was generated.

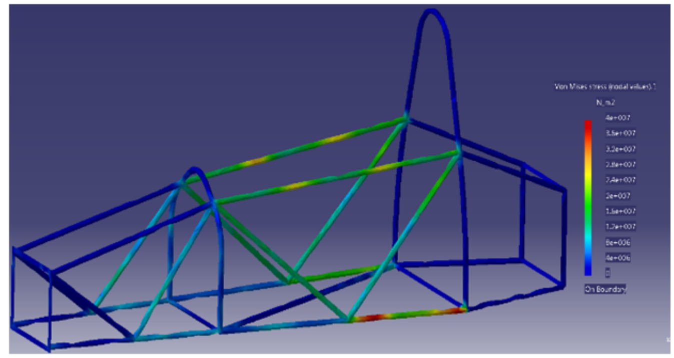

Fig. 3.4: Von-Mises stress profile (Bump test)

Table 3.1: Analysis results (Bump test)

Maximum Von Mises Stress	4 x 107 N/m2
Maximum displacement	0.679 mm
Strain Energy	1.329e+000 J

- **Crash Test**

To simulate a crash, the chassis was clamped at the calculated front suspension mounting points. Assuming the mass of the car to be 350 kg (inclusive of the weight of driver), and a hypothetical acceleration of 20m/s^2, the force which should appear in a crash was calculated to be 7,000 N. A distributed force of 16,000 N which is 2.3 times the force which appears in a crash was applied at the front bulkhead.

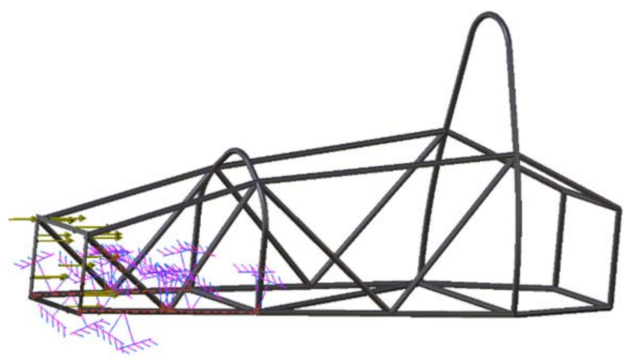

Fig. 3.5: Boundary conditions (Crash test)

Purple color shows clamping while yellow arrows represent the distributed force. This was followed by simulation, which created a Von-Mises stress profile.

Table 3.2: Analysis results (Crash test)

Maximum Von Mises Stress	5.02 x 107 N/m2
Maximum displacement	0.279 mm
Strain Energy	5.576e-001 J

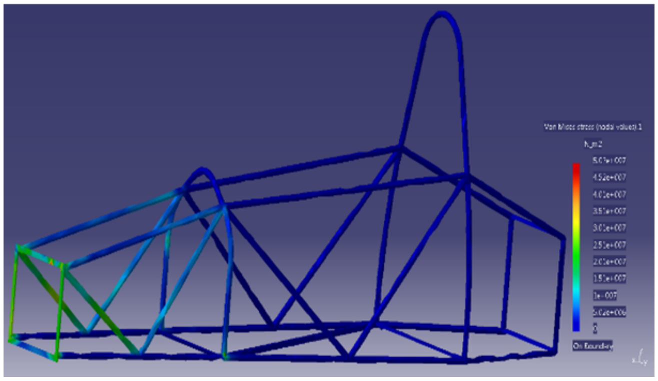

Fig. 3.6: Von-Mises stress profile (Crash test)

- **Torsion Test**

For torsion [1], the chassis was clamped at the rear suspension mount like a cantilever beam, and a clockwise bending moment of 360 N-m was applied along the X axis. Upon subsequent simulation, the Von-Mises stress graph and displacement graph were generated.

Table 3.3: Analysis results (Torsion test)

Maximum Von Mises Stress	2.66 x 107 N/m2
Maximum displacement	0.347 mm
Strain Energy	1.675e-001 J

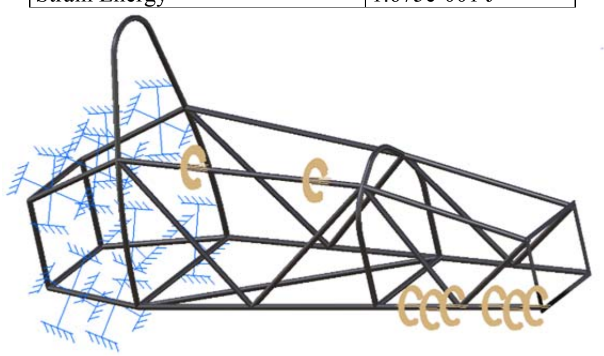

Fig. 3.7: Boundary conditions (Torsion test)

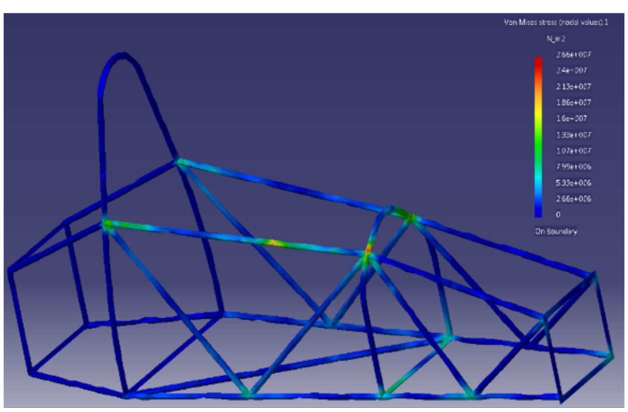

Fig. 3.8: Von-Mises stress profile (Torsion test)

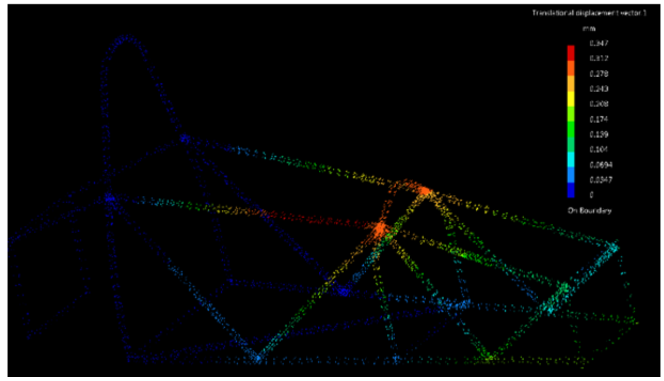

Fig. 3.9: Translational displacement graph (Torsion test)

- **Lateral Bending Test**

Upon cornering and in cases of a side crash, a vehicle experiences centrifugal force [4], which might lead to destructive lateral flexing. To test lateral chassis stiffness, the chassis was clamped at the front and rear suspension mounts and a distributed force of 2500 N was applied laterally across the driver's compartment.

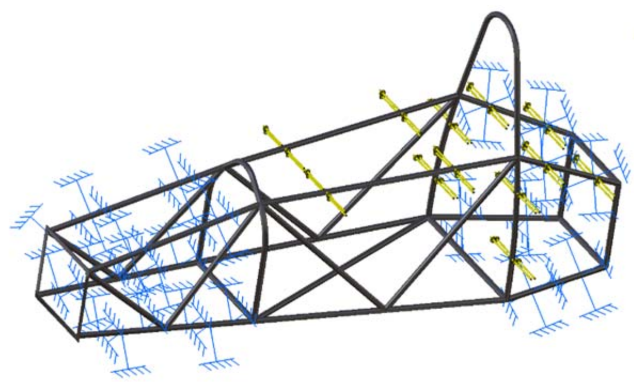

Fig. 3.10: Boundary conditions (Lateral bending)

When simulation was performed, a Von-Mises Stress profile and a displacement graph were generated. A very large factor of safety of 42.59 was detected, which shows the extremely high lateral stiffness that this chassis design cherishes.

Table 3.4: Analysis results (Lateral bending test)

Maximum Von Mises Stress	1.08 x 107 N/m2
Maximum displacement	0.0889 mm
Strain Energy	2.224e-002 J

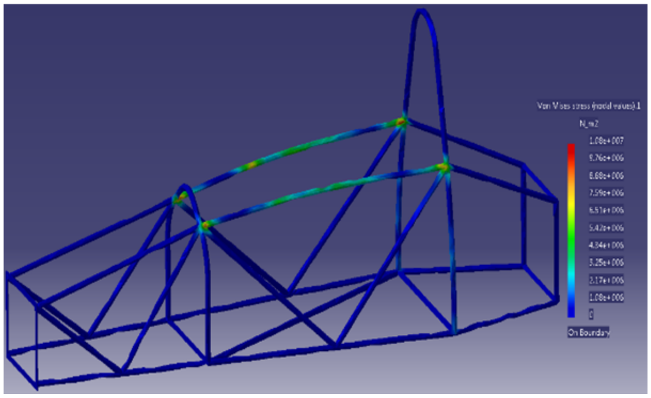

Fig. 3.11: Von-Mises stress profile (Lateral bending)

- **Frequency Test**

Engine and transmission system, generally induce vibration in a vehicle [5]. Any correspondence of induced vibrations with the set of natural frequencies of chassis might lead to deformation. After clamping the chassis at front and rear suspension mounting points, the chassis was automatically tested for its natural frequency values of 165.818Hz, 168.244Hz, 204.547Hz, 270.649Hz, 275.891Hz, 299.705Hz, 308.061Hz, 314.178Hz, 323.133Hz, and 392.914Hz. The chassis survived with a high factor of safety for the lower frequency ranges (less than 170 Hz), which are of consequence to us. The maximum stress was obtained at 270.649Hz, where the stress exceeded the permissible value, however, such ranges are not important, because they are not generated. Mostly, the side impact member and the main hoop were affected.

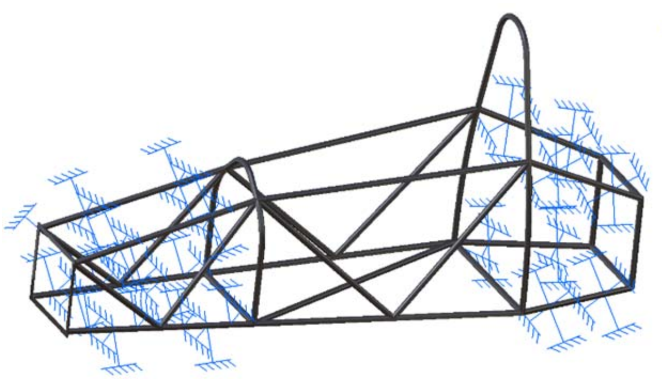

Fig. 3.12: Boundary conditions (Frequency test)

- **Vertical Bending Test**

For vertical bending analysis, the chassis was clamped at the front and rear suspension mounts, and a vertically downward distributed force of 2100 N was applied at the base.

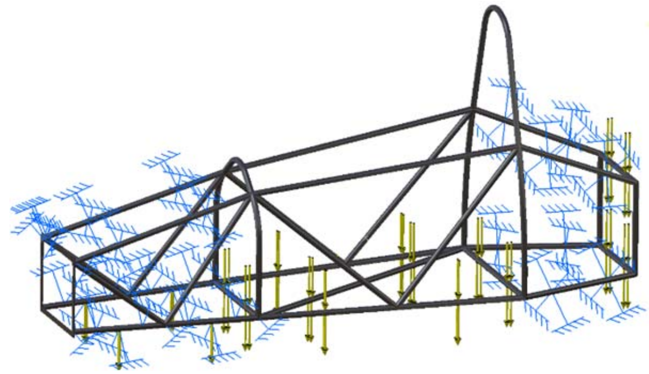

Fig. 3.13: Boundary conditions (Vertical bending)

Upon simulation, the Von-Mises stress graph was generated. The Maximum Stress has been shown in red.

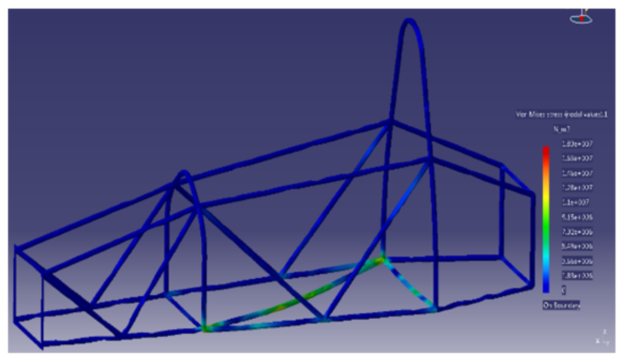

Fig. 3.14: Von-Mises stress profile (Vertical bending)

Table 3.5: Analysis results (Vertical bending test)

Maximum Von Mises Stress	1.83 x 107 N/m2
Maximum displacement	0.153 mm
Strain Energy	5.535e-002 J

4. CONCLUSION

The factor of safety [10] as calculated for the various tests has been mentioned below:-

For yield stress = 4.6 x 10^8 N/m^2,

Table 4.1: Factor of safety for various simulations.

Test	Maximum occurring stress	Factor of Safety
Torsion Test	3.03 x 107 N/m2	15.18
Bump Test	4 x 107 N/m2	11.5
Crash Test	5.02 x 107 N/m2	9.16
Vertical Bending Test	1.83 x 107 N/m2	25.13
Lateral Bending Test	1.08 x 107 N/m2	42.59

The table clearly depicts the extremely high factors of safety obtained during various analysis. Where light-weightiness was slightly compromised with, a resilient structure, which could be safely taken to very high velocities was created. Very high torsional, longitudinal and lateral stiffness can be clearly seen in the chassis design.

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