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A COST EFFECTIVE FEA-BASED APPROACH FOR DEVELOPING COMPLIANT AND SAFE AUTOMOBILE

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ABSTRACT

Safety is becoming one of the most important aspects of modern automotive industry to reduce the development and testing costs of new vehicles, it is advisable to use computational crash simulations for early evaluation of behavior of car under testing, that way the severity of impact parameters can be estimated well in advance of a real crash test and possible design changes can be easily evaluated. The objective of the study is to demonstrate the front crash, rear crash and side crash simulation of the vehicle against a rigid wall to examine injury risk and potential of safety. In this paper, simulations were performed with the explicit finite element Hypermesh software running on a multiprocessor computational platform to numerically simulate the crash test of the roll cage with a rigid wall. The acceleration, distortion and deformed energy at the frontal, rear, and side region of the vehicle are traced. A very good agreement of simulation and real crash tests results was observed, which in turn justifies the use of computer simulations in the process of development of an optimized structure of the vehicle. The design procedure follows all the rules laid down by FSAE rulebook for formula type cars. All the analysis were carried out in Hypermesh 13.0.

KEYWORDS: FSAERoll Cage; Impact Attenuator; Finite Element Analysis; Torsion stiffness; Crash Analysis.

I. INTRODUCTION

For a proper working of a formula student car, it is important that all the components work in the desired manner. As roll cage being the important part of the vehicle, which absorbs all the static and dynamic loads experienced under normal driving conditions. the structure must be such that it should sustain the stresses generated without any deformation. because of the failure of structural members which leads to accidents. The structure of a roll cage must be light and rigid. The conventional roll cage looks like a truss and is welded at joints. This frame protects its occupants from an accident in case of a rollover and impact.

There are a lot of forces acting on the vehicle when the vehicle is in static as well as in the dynamic condition. These forces can cause deformation resulting into stress generation in various parts of the roll cage. These forces are generally occurring during braking, acceleration, cornering, impact or combination of above. The stiffness of the roll cage must be such that it must be able to resist these forces. A roll cage which is torsional stiff enables a desirable roll moment distribution to be achieved for good handling balance.

Racing cars are designed to be driven near the limit of adhesion always and are therefore prone to be involved in accidents, the most likely scenarios being a frontal, side impact and rear. there is more chance of front impact. A roll cage which can absorb high energy impacts whilst controlling the rate of deceleration will increase the likelihood of drivers surviving a crash without injury. To achieve this honeycomb-like structure is provided in the frontal area of roll cage which known as impact attenuator

II. FINAL CHASSIS DESIGN

After few iterations, the final chassis designed is as shown:

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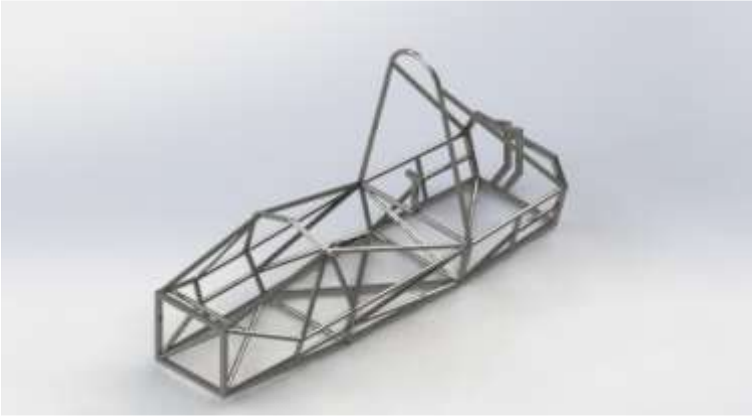
Figure:



3D Isometric view of chassis.

III. MATERIAL SELECTION

For chassis

The material selected for chassis design is AISI 1020. The various cross sections selected steel members are:

Table 1. Mechanical and metallurgical properties of AISI 1020.

PARAMETERS	VALUE
Initial Density	7.9x10-9 Ton/mm^3
Young's Modulus	210000 MPa
Poisson's Ratio	0.3
Yield Stress	350 MPa
Hardening Parameter	450 MPa
Hardening Exponent	0.2
Maximum Stress	465 a

For impact attenuator

For material selection, the factors to be considered are cost, weight, reliability, availability. Rohacell 71 IG foam is used.

Table 2. Mechanical and metallurgical properties of Rohacell 71 IG foam.

PROPERTY	VALUE
Density	75 Kg/m^3
Compressive Strength	1.5 MPa
Tensile Strength	2.8 MPa
Shear Strength	1.3 MPa
Elastic Modulus	92 MPa
Shear Modulus	29 MPa

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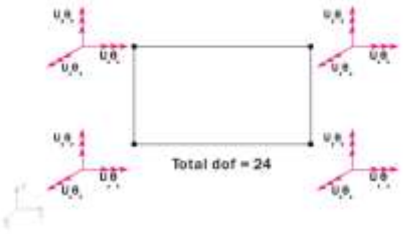
IV. MESHING

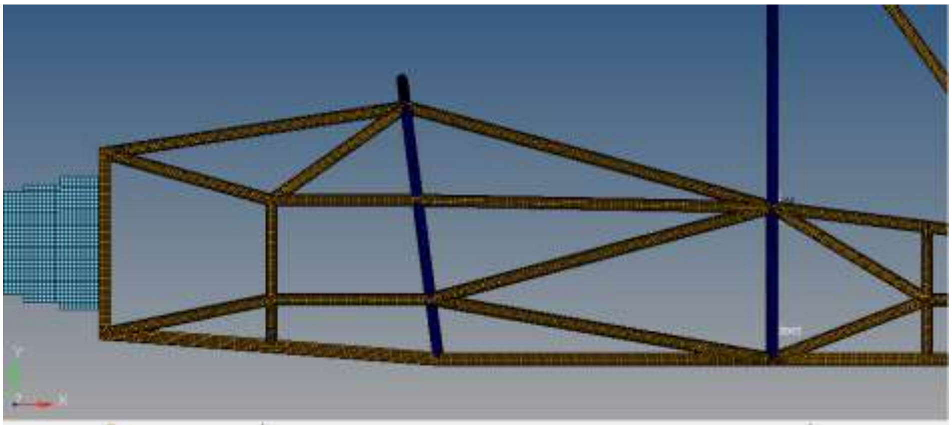
Roll cage
Roll cage meshed with 2-D Shell elements which are the most general type of element. DOFs:6Dof/Node (U_x, U_y, U_z, θ_x, θ_y, θ_z)

Figure:



Possible DOFs of a 4 nodes element.

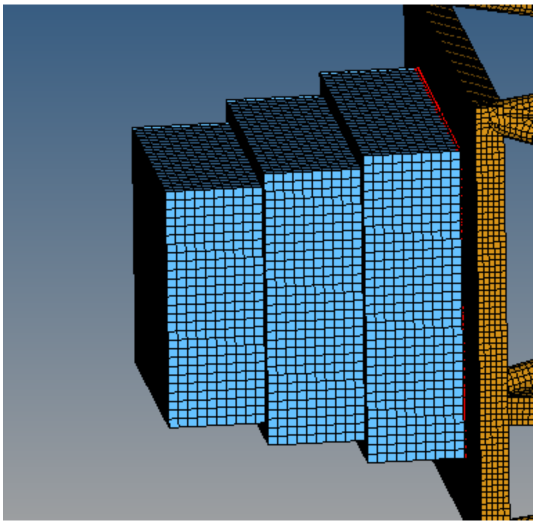
Figure:



Front Impact arrangement

Impact attenuator
3D meshing is done on impact attenuator with Brick elements.

Figure:



Impact Attenuator

V. CHASSIS ANALYSIS

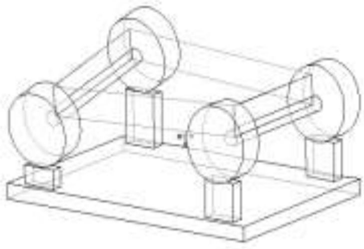
Frame stiffness

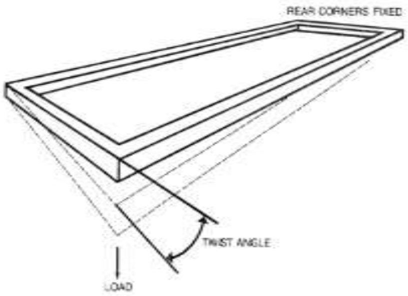
Frame is not only the foundation to assembly of other systems and the bearing structure of the whole vehicle, but also influence overall control stability. In general, here are the principles for designing frame:

(1) To have adequate strength to prevent damage from happening in the worst cases.
(2) To have adequate stiffness to ensure stability of handling and other systems.
(3) To develop a chassis with a high value of roll stiffness to counter forces applied by the suspension during cornering while keeping the weight as low as possible.

The main deformation modes considered for chassis are:

Longitudinal Torsion
Figure:





By expanding on the principles of solid mechanics and making some simplifications a method can be developed to give an approximate value for the chassis. If the applied torque (T) is related to the angle of twist of a chassis (φ) through the following equation:

$$T = \frac{GJ\varnothing}{L} = K_T\varnothing \quad (1)$$

Where:
J= Polar moment of inertia
G= Material shear modulus of elasticity
L= Characteristic length of cross section
Ø=angular displacement
K_T=Torsional rigidity

And using simulation. FEA is performed on the chassis, equal and opposite loads are applied at the front suspension mounting locations while the rear mounting locations remain fixed. The equations used to determine the torsion stiffness is based on the total deflection of the mounting locations. The torsion stiffness is calculated using the following equations

$$K_T = \frac{T}{\varphi} = \frac{FB}{(\varphi_d + \varphi_p)} \quad (2)$$

Where

$$\varphi_d = \tan^{-1}\left(\frac{V_d}{B/2}\right) \tag{3}$$

$$\varphi_p = \tan^{-1}\left(\frac{V_d}{B/2}\right) \tag{4}$$

In the above equations, the torque, T, is represented by the vertical force applied at the mounting locations, F, and the track width of the vehicle, B. The angular deflections φ_d and φ_p are based on the vertical deflections for the driver V_d and passenger V_p sides of the vehicle, as well as the track width. The angular deflections should be similar but are not necessarily exactly equal due to small differences in the geometry of the vehicle as well as small differences in where the loads are applied on the vehicle mesh. For formula car, taking B as the average width of front bulkhead.

Stress and displacement analysis in torsion and calculation of torsional rigidity.

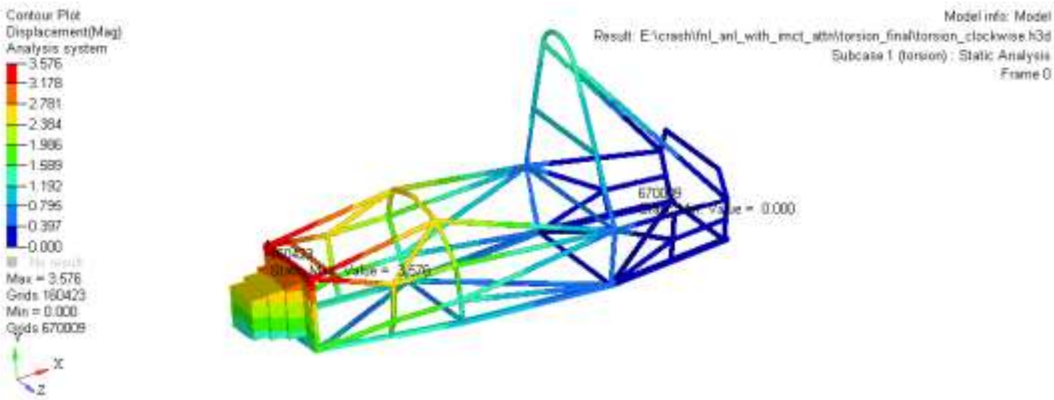
The frame is modelled in Solidworks and 2-D analysis is performed in Altair Hyperworks using Optistruct as a solver. Opposite forces are applied at suspension mountings to simulate the torque of a bump impact. After several iterations performed to optimize torsional rigidity with minimum possible weight, some of the frame members were removed and some new were added and the Torsional Rigidity obtained was 4915.33 Nm/deg.

Calculations

***Table 3**. Table showing various parameters calculated.*

Input parameters	**MAGNITUDE**
Force	6000 (assuming 2g force)
B	0.580 m
Ø	0.70deg
Torsional rigidity	4915.33 Nm/deg

Figure:



Torsion displacement plot

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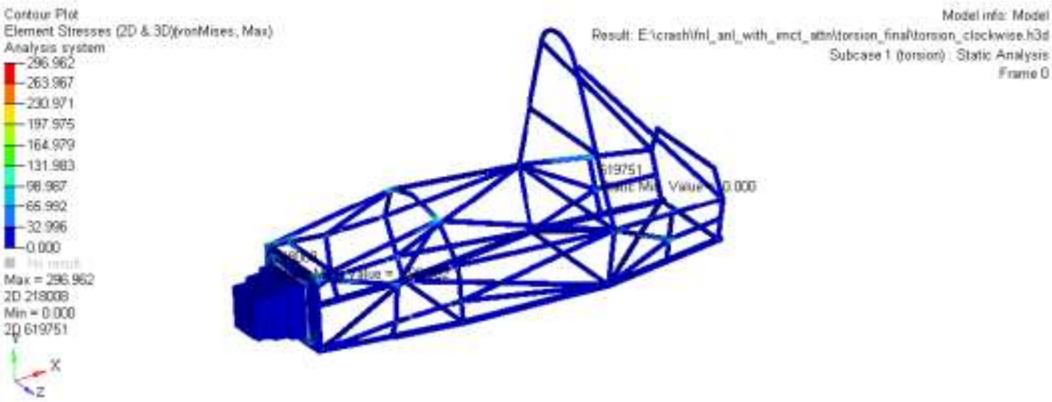
Figure:

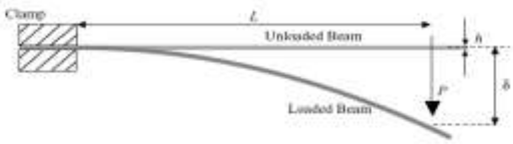


Torsion stress plot

Result

The maximum stress is **296.962 MPa** and is less than yield strength of AISI 1020**.**

Stress and displacement analysis in bending and bump and calculation of bending stiffness



$$\delta = \frac{PL^2}{2EI} \tag{5}$$

$$K = \frac{P}{\delta} = \frac{2EI}{L^2} \tag{6}$$

Where,

P= total applied force
E= modulus of elasticity
I= moment of inertia
δ = linear displacement
K=Bending stiffness

And using simulation. FEA is performed on the chassis, equal loads are applied at the front suspension mounting locations while the rear mounting locations remain fixed. The equations used to determine the Bending stiffness is based on the total deflection of the mounting locations. The bending stiffness is calculated using the following equations.

$$K= \frac{loading}{deflection} = \frac{P}{\delta} \tag{7}$$

In the above equation M is the bending force at the point of applied force and δ is the linear deflection in the direction of applied force.

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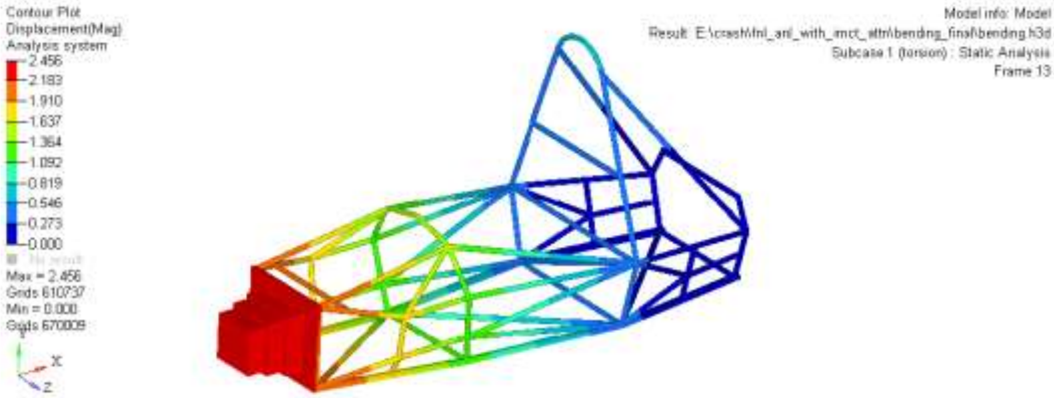
Analysis for bending stiffness

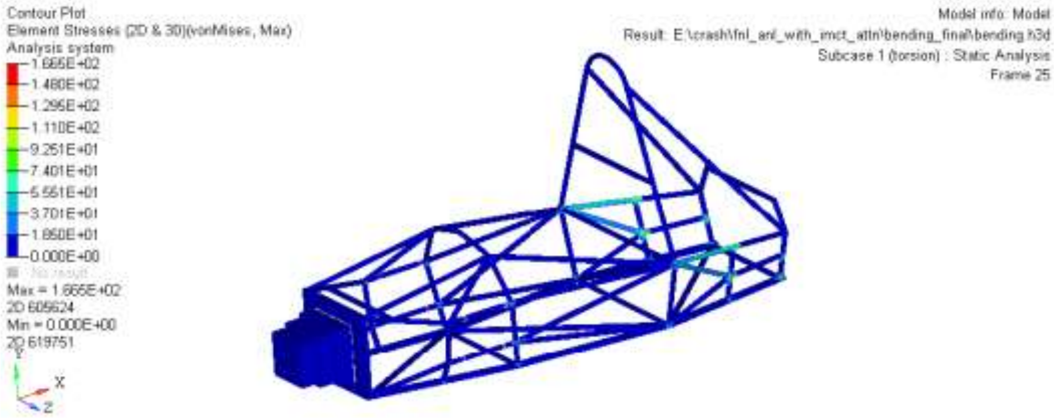
Figure:



Bending displacement plot

Considering total weight to be the criteria of deciding bending stiffness, static 2D analysis using OPTISTRUCT has been performed. With a number of iterations amendments were made in the chassis. The bending stiffness achieved is **4013.5 N/mm.**

Figure:

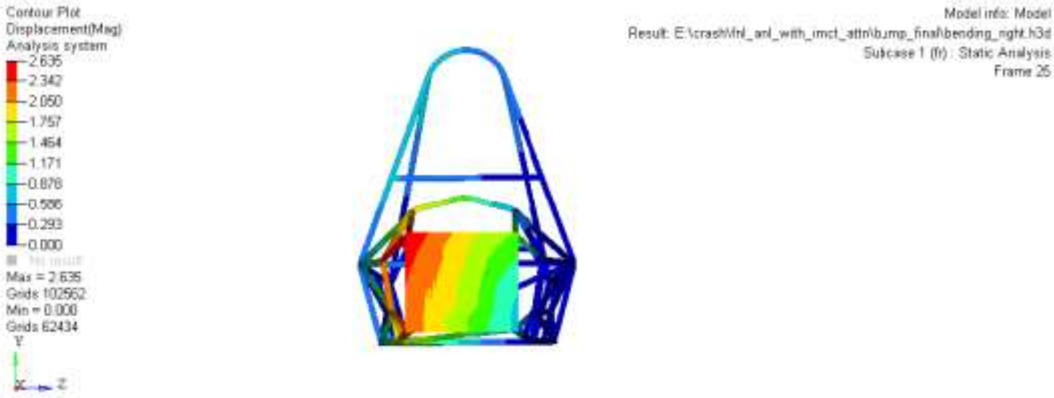


Bending stress plot

Result

The maximum stress is **166.5 MPa** and is less than yield strength of AISI 1020**.**

Figure:



Bump displacement plot

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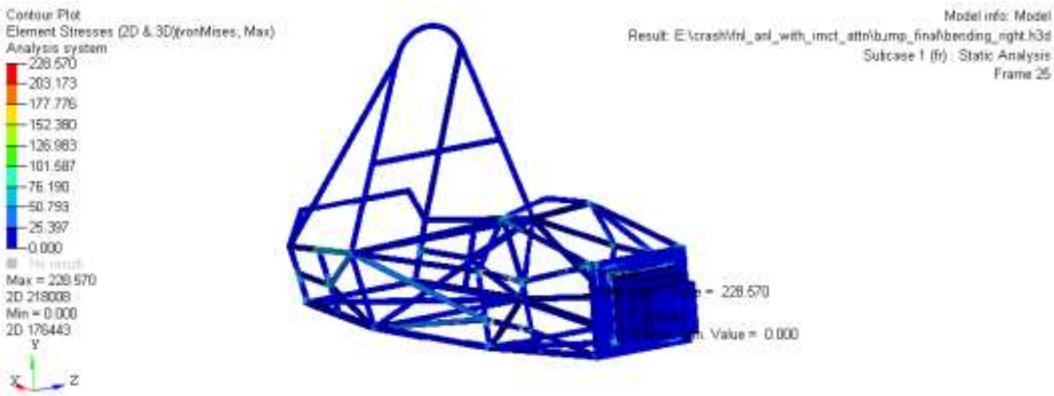
Figure:



Bump stress plot

Result

The maximum stress is **228.570 MPa** and is less than yield strength of AISI 1020**.**

VI. CRASH ANALYSIS

Why crash analysis? Racing cars are designed to be driven near the limit of adhesion at all times and are therefore prone to be involved in accidents, the most likely scenarios being a frontal side impact and rear. In order to validate the overall performance of the vehicle, all of these scenarios were simulated using the non-linear finite element analysis package in Altair Hyperworks using Radioss.

Front crash

The competition rules have a comprehensive section covering safety parameters, which specify the minimum requirements for a spaceframe chassis. Due to the composite construction of the Leeds chassis, energy absorbent structures are used to provide equivalent protection. These structures are the nosecone for frontal impacts

The roll cage attached with the impact attenuator is crashed against a rigid wall at an initial velocity of 7 m/s.

The aim is to ensure the safety of driver's feet in case of deformation of front bulkhead and to verify the maximum energy absorption capacity of impact attenuator.

Table 4. Table showing parameters considered for crash.

INPUT PARAMETERS	DETAILS
Mass	320 kg
Initial velocity	7 m/s
Impact wall	Rigid and frictionless

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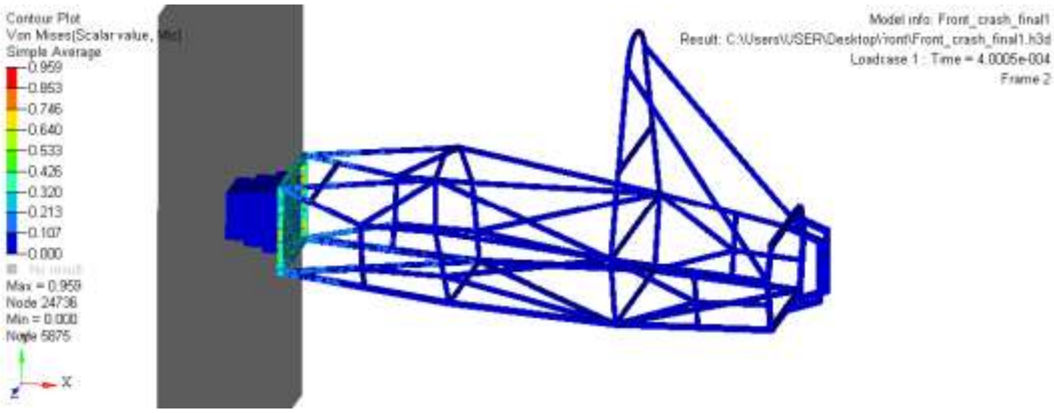
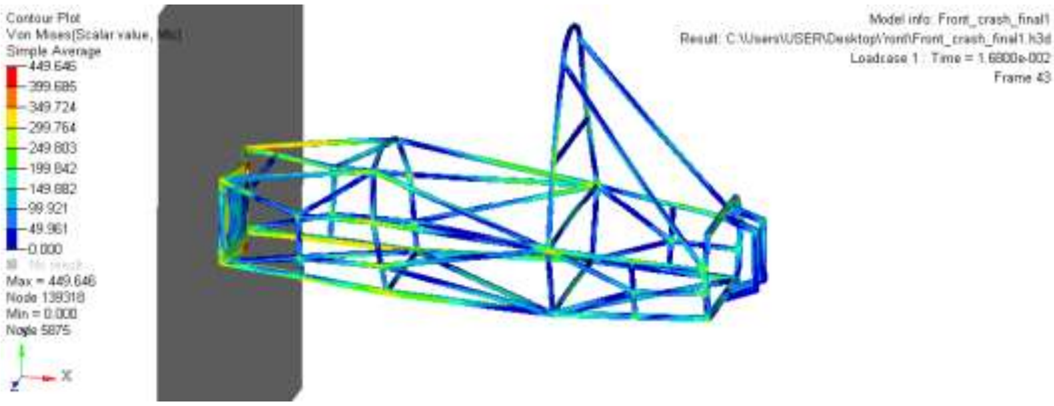
Output plots

Figure:



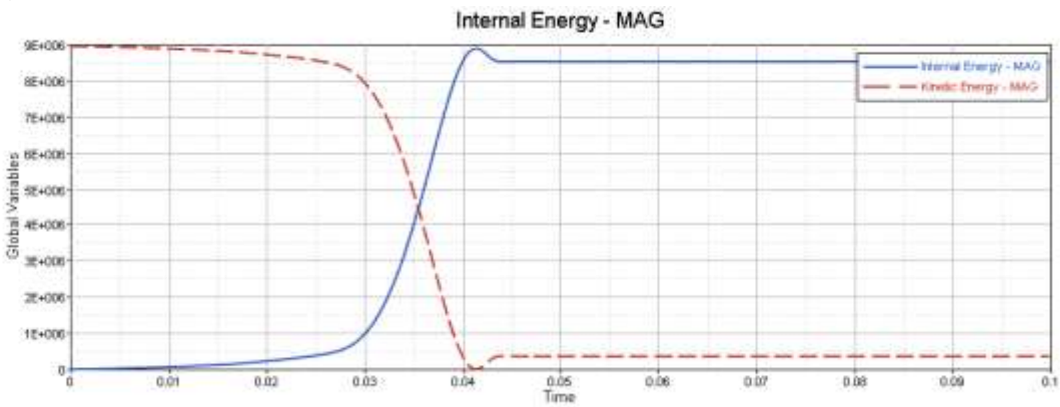
Stress plot at the start of collision

Figure:



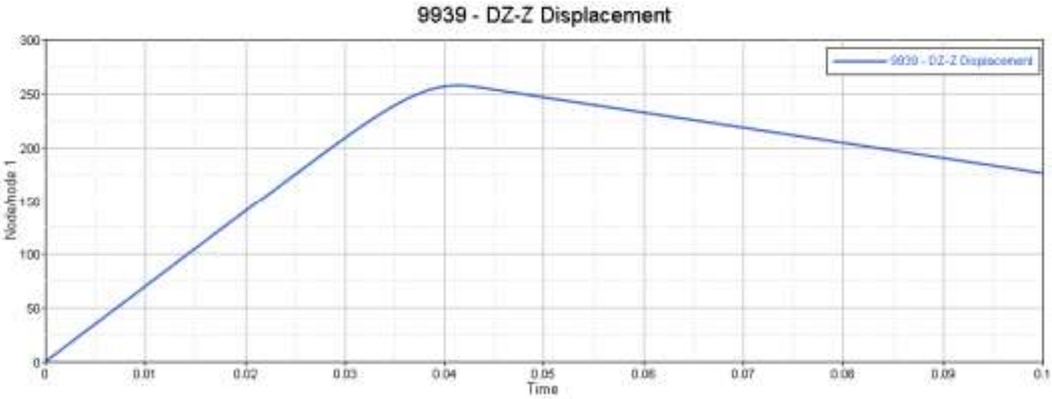
Stress plot at the end of collision

Figure:



Energy plot of collision

Figure:



Displacement plot

Results and discussion

Stress and displacement

The maximum stress induced is 449.6 MPa and crossing the ultimate tensile strength of roll cage material. This indicates that fractures will occur in these regions. But the maximum displacement is less than the length of the impact attenuator. Hence the driver is safe.

Table 5. Maximum displacement and stress induced.

Output parameter	Magnitude
Maximum displacement	225 mm
Maximum stress	449.6 MPa

Energy

From the energy plot it can be concluded that nearly all the kinetic energy of the roll cage is converted into internal energy of the impact attenuator within fraction of seconds. The remaining kinetic energy represents the rebound velocity and after the collision both energies remains constant.

Impact attenuator

The energy absorbed by the impact attenuator for mass=320 kg, velocity = 7 m/s

Energy =kinetic energy of structure

$$= \frac{1}{2} \times mass \times velocity^2 \qquad (8)$$

$$= 0.5 \times 320 \times 7^2$$

= 7840 J > 7350 J, as specified in the rules

Rear crash

The roll cage1 is crashed against a rigid wall from the rear side at an initial velocity of 7 m/s. it is to ensure minimum deformation and hence damage to the engine compartment.

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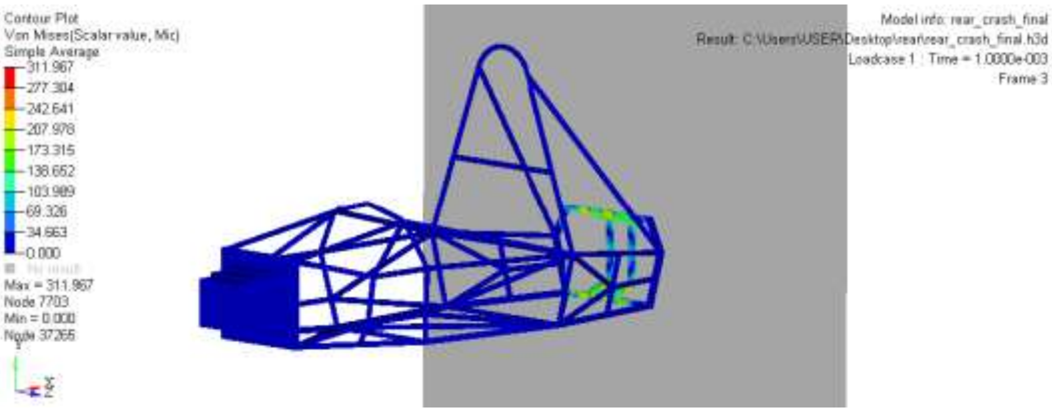
Table 4. Table showing parameters considered for crash.

INPUT PARAMETERS	DETAILS
Mass	320kg
Initial velocity	7 m/s
Impact wall	Rigid and frictionless

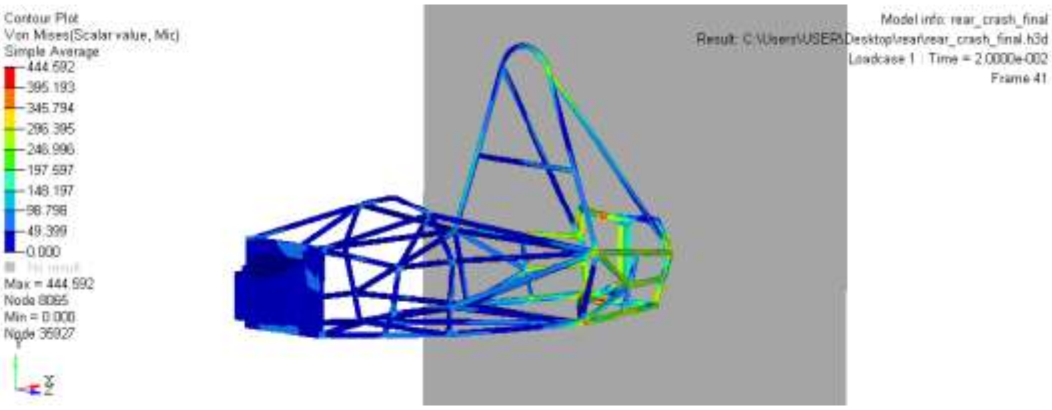
Output plots

Figure:



Stress plot at the start of collision

Figure:



Stress plot at the end of collision

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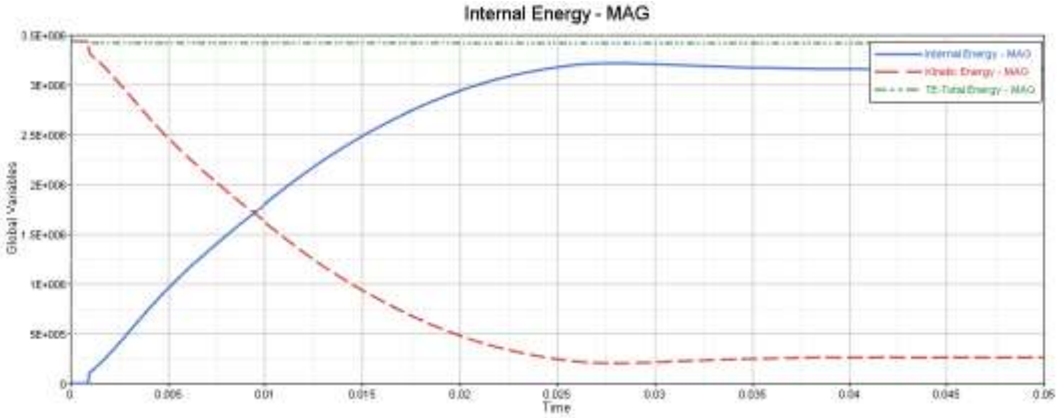
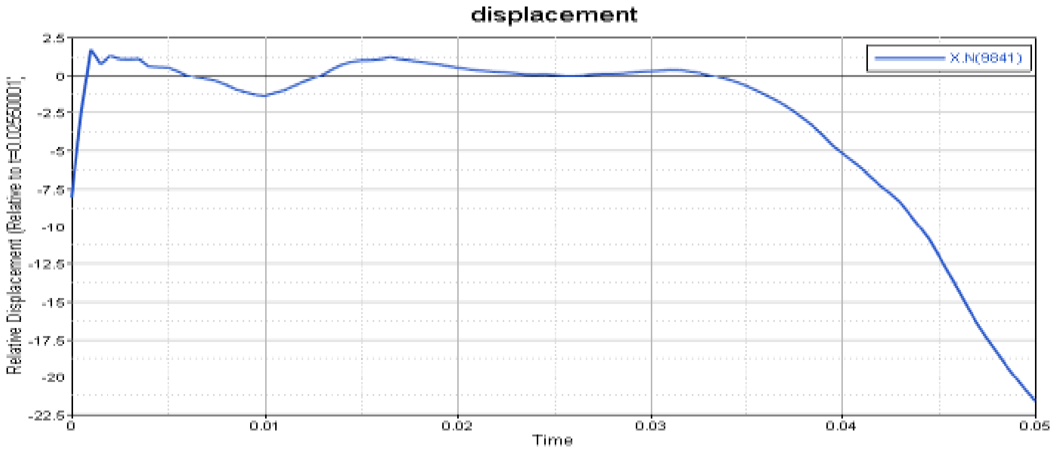
Figure:



Energy Plot

Figure:



Displacement plot

Results and Discussion

The energy plot explains about the energy transformations during impact. The kinetic energy drops suddenly and there is corresponding rise in the internal energy. After this both the energies remains constant. The remaining KE represents the rebound after the impact.

In the stress plot, the dynamic maximum stress is crossing the ultimate tensile strength of AISI 1010. This results in local fracture of the members. But the displacement of the node of maximum deformation is in the range of 22.5-25 mm. Also, there are no components of engine compartment in this range of deformation. Therefore, the considered crash is safe.

Table 5. Maximum displacement and stress induced.

Output parameter	Magnitude
Maximum displacement	22.5-25 mm
Maximum stress	444.59 MPa

Side crash

In a side impact collision, the main consideration, with respect to driver injury, is cockpit intrusion. The main method of energy dissipation in this scenario is the deformation of the side members. The simulation results showed that the cockpit suffers no intrusions due to the deformation of the side members which absorbed the impact energy and the stiffness of the cockpit side.

This assumes the side impact in case of skidding etc. the roll cage is crashed sidewise against a rigid wall at a speed of 7 m/s. this analysis is to ensure the driver's safety in case of side impact and hence shows the effective strength of the side impact members.

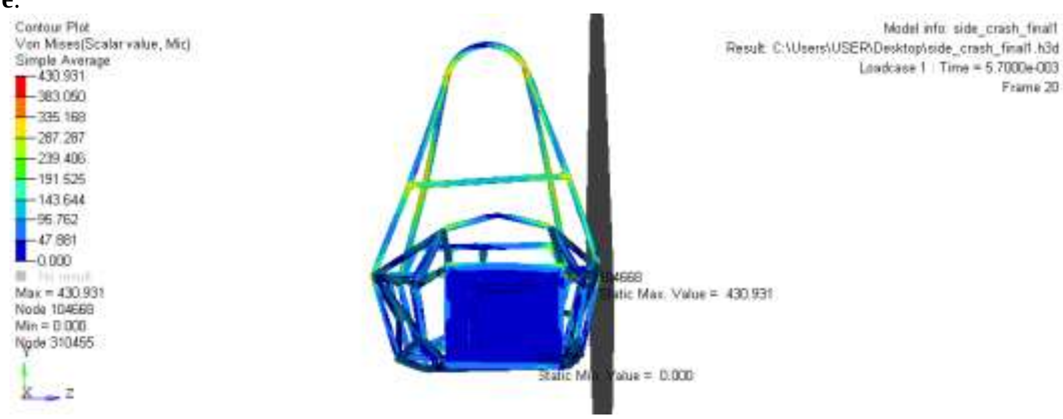
Table 4. Table showing parameters considered for crash.

Input parameters	Details
Mass	320 kg
Initial velocity	7 m/s
Impact wall	Rigid and frictionless

Figure:

Stress plot at the start of collision

Figure:



Stress plot at the end of collision

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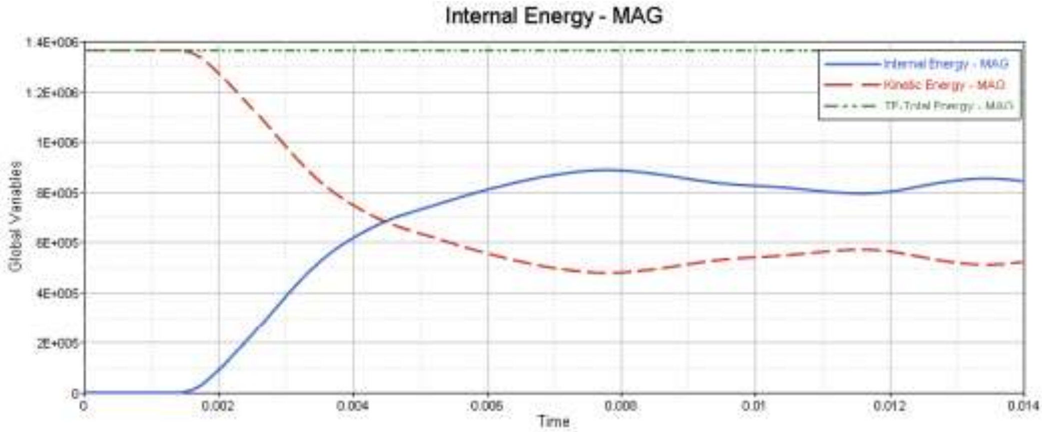
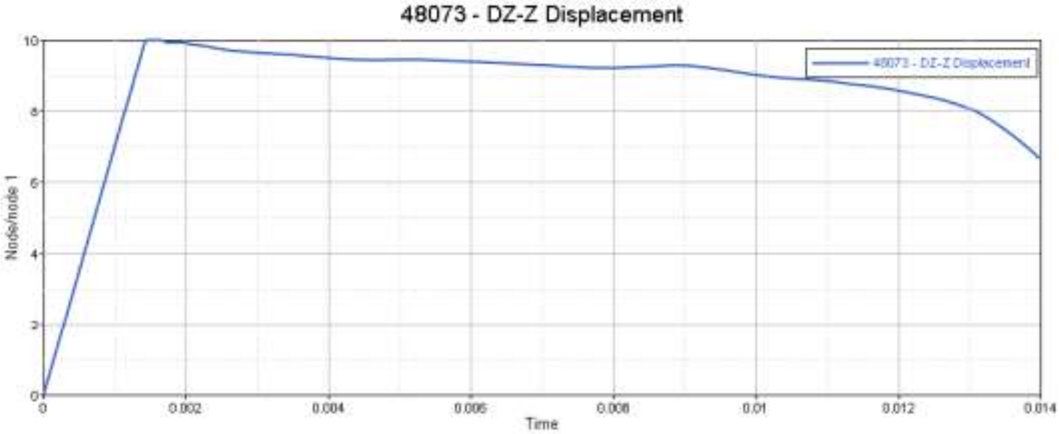
Figure:



Energy plot

Figure:



Displacement plot

Results and Discussion

The energy curves depict the partial conversion of kinetic energy in internal energy or the deformation. The remaining KE shows the considerable rebound velocity.

The dynamic stress values are reaching 430.93 MPa and hence exceeding tensile strength of steel used and therefore results in fractures. But the maximum displacement of the node after deformation is 10 mm and this is less than the side clearance provided. Therefore, the structure is safe in case of side impact.

Table 5. Maximum displacement and stress induced.

Output parameter	Magnitude
Maximum displacement	10 mm
Maximum stress	430.93 Pa

VII. Conclusion

This paper has dealt with a variety of issues related to roll cage design and to evaluate the driver's risk of injury during a frontal, rear and side impact of a racing car. By viewing these images and graphs there are many things that can be seen which would include stress developed in various locations and displacement of various members in the design, the way the chassis distorts showing driver safety potential from a controlled folding of the chassis. This controlled folding is due to the triangular orientation of the members within the chassis. And the triangular segment that is located under the floor pan of the car. These members are crucial to see how they deform under load and how they distribute stress throughout the system for understanding their ability to provide driver safety, which is critical when the potential loss of human life is involved in an engineer's product. After seeing stress, displacement plot, energy and displacement curve we can conclude that when the car is running with very high velocity and it gets collided with other car or any other structure like wall then there is a chance that the stress induced in the structure are greater than the ultimate stress of that material because of very high reaction forces due to sudden impact which may cause failure of structural member like bending or crushing failure member may enter to the driver area and it may cause injury to the driver which is not acceptable.so to study how our driver is safe in this type of scenario We have performed dynamic analysis of roll cage with impact attenuator attached to front bulkhead with velocity of 7m/s and get impact with rigid wall in front, side and rear. And study the total amount of energy absorbed before get rebound and maximum displacement of impact point and with this study examined the driver's risk of injury. This study would help the designers to develop more competitive roll cage designs.

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