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Design Analysis of the Chassis for the Go-Kart

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Abstract: *The chassis is the skeleton of any vehicle. Also it acts like a shell surrounding the occupants which protect the occupants in case of impact. It also adds to aesthetics of the vehicle.*

The primary objective of this project is to design a chassis for a go-kart that meets the international standards and is also cost effective at the same time. It is designed to incorporate all the features required for design of a student race car. We have focused on every point of the chassis to improve the performance of the vehicle without its failure.

Keeping in view the aim of the project, extensive research is carried out on the chassis building for the go– kart so as to build the chassis according to the required design and minimum weight. The installation of the sub-systems with the chassis is also considered. The software model is prepared in the solidWorks software designed through finite element modelling techniques. Computational analysis has been carried out for the selection of material for building the frame, chassis and frame design, cross section determination, stress analysis, simulation to test the chassis failures including both static and dynamic test. Factor of safety of the driver cockpit in case of impact is also analyzed.

Keywords: *chassis, solid works, frame design, cross-section, stress analysis, chassis failure*

1. INTRODUCTION

A frame of a vehicle plays the most important role in safety of the passenger. The frame contains the operator, engine, brake system, fuel system, and steering mechanism, and must be of adequate strength to protect the operator in the event of a rollover or impact. The passenger cabin must have the capacity to resist all the forces exerted upon it. This can be achieved either by using high strength material or better cross sections against the applied load. But the most feasible way to balance the dry mass of a chassis with the optimum number of members is done by triangulation method [1].

The front and the rear can be failed during the various testing but the passenger cabin must be safe to withstand load. It could be achieved either by using the material of high strength or of better cross section against the applied load. Material is also a limitation, increment in dimension raising overall weight, thereby lowering the fuel efficiency, so in order to overcome all this, circular cross-section is employed for the roll cage development. And circular section is always a perfect one to resist the twisting and the rolling effects. Circular section is preferred for torsional rigidity [2].

Chassis of a go-kart plays a significant role in the jacking of the kart while the kart is cornering. In the absence of a differential in a kart, chassis frame plays following pivotal roles in the performance:

- It allows for lifting of the rear inside wheel of the kart while cornering by the virtue of its flexibility and relatively low torsional stiffness. This can cause the kart to turn very smoothly even without a differential
- It acts as a spring to absorb various shocks and vibrations from the road to provide maximum comfort to the driver

2. DESIGN OBJECTIVES AND METHODOLOGY

The frame of the kart chassis was designed with following aims:

- To have minimum wheelbase and track-width as permitted ergonomic norms to improve cornering performance
- To weigh less than 15 kg.
- To be flexible enough to allow rear 'jacking' effect and absorb road shocks
- To protect the driver in front and side crash events
- To provide comfortable posture to a large range of driver statures
- To be easy to fabricate
- To have an open airflow over the engine compartment for cooling.

Keeping the above mentioned objectives in view, a *tubular double rail chassis* was used in the front part to facilitate an open ergonomically suitable compartment. Fig 1 shows the design workflow that was adopted for this project [3].

Conceptual design was initially agreed upon keeping all the initial design parameters in view.

Thereafter, virtual modelling was done on Solid Works 2012 for frame which was then analysed structurally by two CAE packages COSMOS 13 (SW Simulation) and ANSYS 15. Using two softwares simultaneously greatly reduced the chances of error which easily creep up in FEA (Finite Element Analysis).

The model was further analysed in dynamic loading conditions in ANSYS for cornering performance etc. and ergonomic analysis was done in CATIA v5 R20. Changes were made in the design to satisfy all conditions necessary. Multi-body modelling was done to accommodate all the auxiliary components on the frame [4].

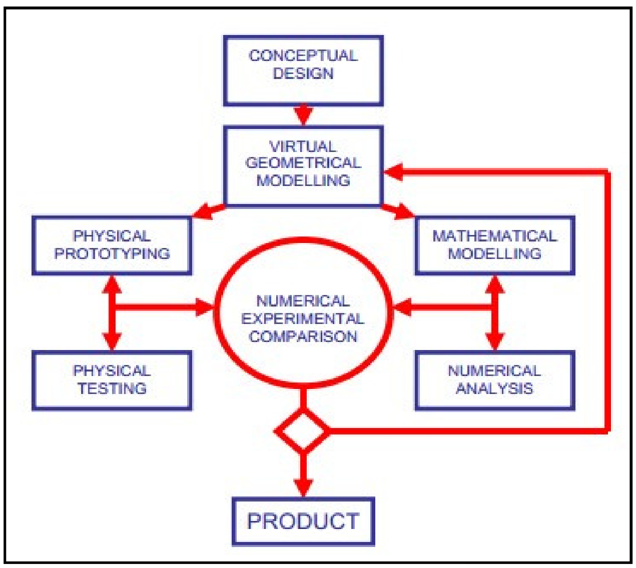

Fig 1. Design Methodology

3. CHASSIS SPECIFICATIONS

A. Material Selection: Various parameters were kept in view while deciding the frame material which included availability, cost, machinability and tensile strength. Fig. 2 shows a plot of statistical data compiled for various materials that were analysed for kart frame. IS 2062: E250 was selected to be the frame material for

- Lowest cost
- Highest weldability
- Highest availability
- Moderate Strength

B. Cross Section: The cross section for the members was chosen as circular tubular for its higher torsional stiffness for a given area of cross section compared to square and other sections. The standard cross section determined after market research was 25mm OD and 1.6mm ID.

Material Properties of IS 2062 E250 steel are given in the table below:

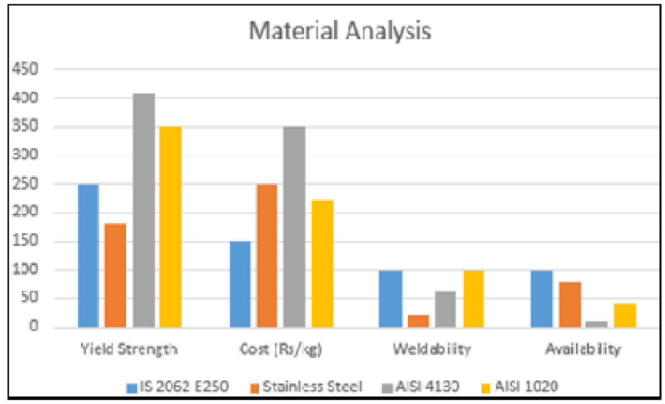

Fig. 2. Material Analysis for kart strength

TABLE 1: MATERIAL PROPERTIES

Property	Value
Density	$7850\ kg/m^3$
Elastic Modulus	205 GPa
Poisson’s Ratio	0.285
Yield Strength	250 MPa
Ultimate Tensile Strength	A

Final Chassis Layout: The three normal views of frame are shown in following figures:

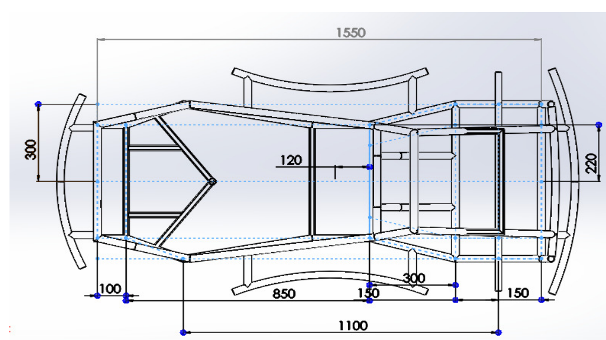

Fig. 3. Top View

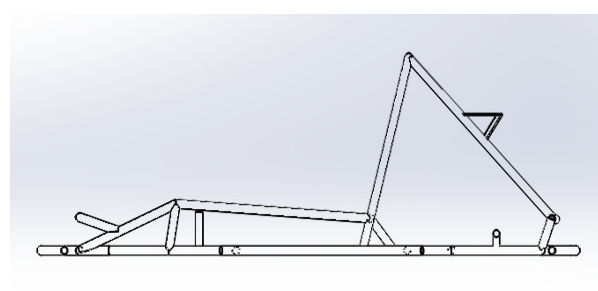

Fig. 4.Side View

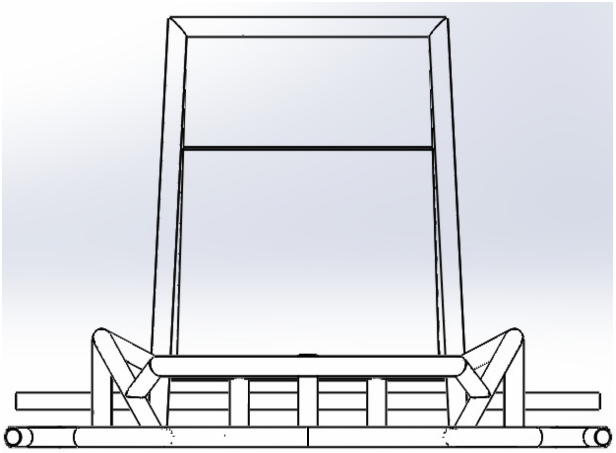

Fig. 5. Front View

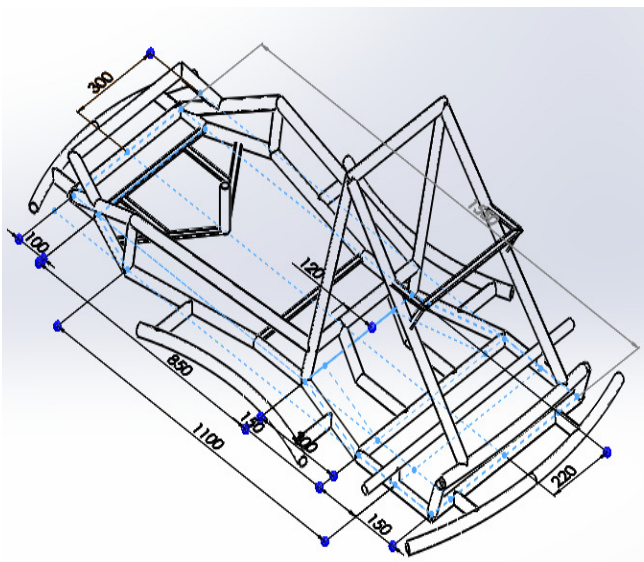

Fig. 6. Isometric View

Final Geometrical Parameters: Major dimensions which associated with the frame have been tabulated in Table 2.

TABLE 2: FRAME PARAMETERS

Parameter	Value (mm)
Wheelbase	1100
Front Track	850
Rear Track	850
Total Length	1800
Total Height	600
Total Width	740
Cross Sectional Data	
Type	Tubular
Outer Diameter	25 mm
Thickness	mm

4. FINITE ELEMENT ANALYSIS

The finite element theory was employed for predicting the behaviour of chassis under the methods proposed in the Fig. 7.

FEA is a method in which the model is discretized into small elements, properties of which are then evaluated using general equations of motion and boundary conditions specified during a test. This involves solution of the equation:

$$[Reaction] = [Stiffness] * [Displacement] + [Load]$$

The non-structural elements such as driver and engine were modelled as remote mass acting on their respective mounting positions. Also, theanalysis was done in both COSMOS and ANSYS to get better validated results under same loaded conditions.

Grid Characteristics*:* The frame was meshed from beam elements for analysis.In COSMOS, the model was generated automatically from beam elements with 6 degrees of freedom for every element.In ANSYS, the element was generated as the BEAM188 element which is a 2 node 3D finite strain beam (6 degrees of freedom).

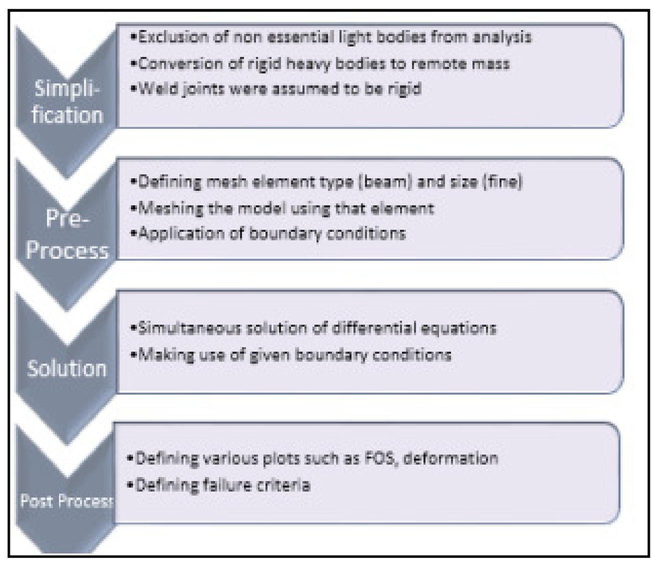

Fig. 7. FEA Methodology

TABLE 3: ELEMENTS

Platform	Element	Size	Number of elements
COSMOS	Beam	20.1 mm	747
ANSYS	Beam-188	25.7 mm	580

The members which were predicted to be the heaviest loaded were applied ‘fine’ mesh control to gain better accuracy. The final mesh for COSMOS and ANSYS are shown in Figs. 8 and 9.

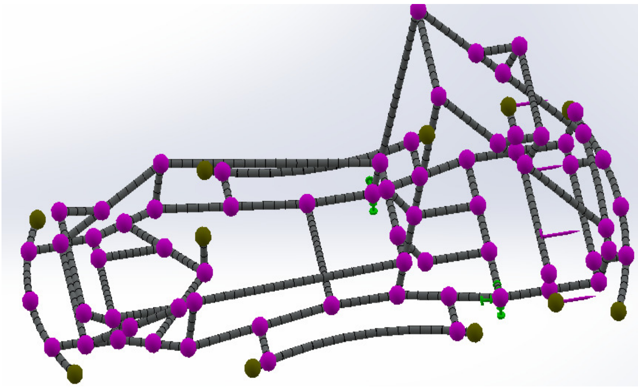

Fig. 8. Beam mesh in COSMOS

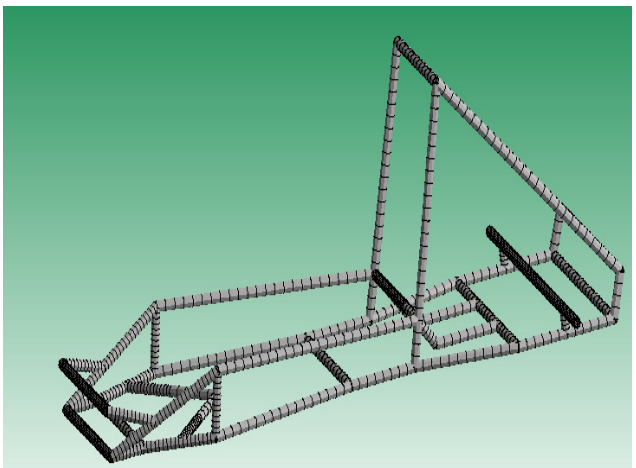

Fig. 9. Beam mesh in ANSYS with mesh control applied

After setting the mesh, 5 static studies were performed on the model:

Static Bending Test

d) Front Impact Test

Torsional Stiffness Test

e) Side Impact Test

Lateral Bending Test

These models have been discussed in detail in the upcoming section.

Static Bending Test: In this test, various stresses developed in a fully loaded chassis were analysed.

TABLE 4: LOADING DIAGRAM

Loads	Driver, Engine gravity loads at mounting position
Constraints	Wheel Hub mounting positions
Gravity	On

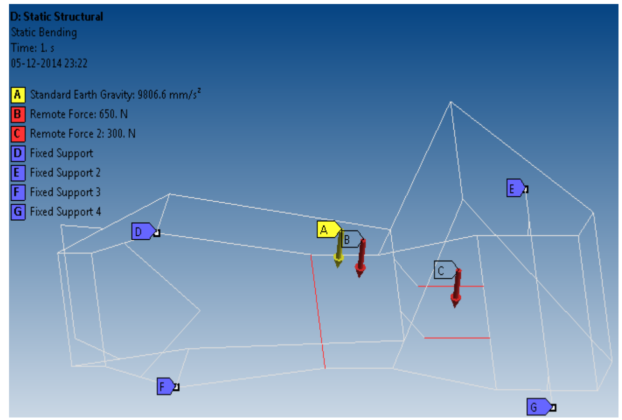

Fig. 10 Loading Diagram

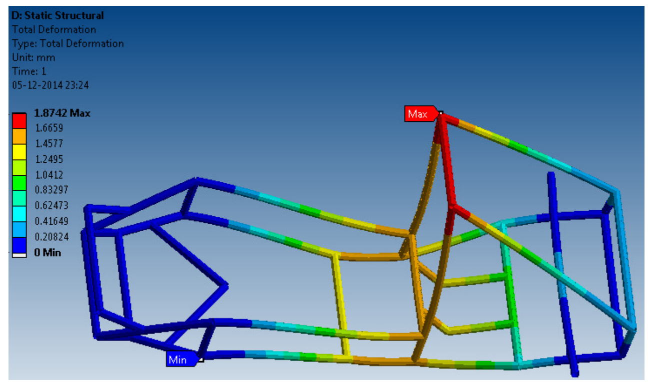

Fig.11. Deformation Plot

As can be seen from the chart, maximum deformation was 1.8 mm at driver seat during sagging which is acceptable. The highest combined stress was encountered at the rear wheel hub mounting positions and its value was 83.4 MPa.

The yield stress of E250 is 250 MPa. So,

$$FOS = \frac{Max\ combined\ stress}{Yield\ Stress} = 3.2$$

Torsional Stiffness test: This test determines the resistance offered by the chassis frame against a twist which is normally developed during cornering, or when the vehicle encounters a bump in the road.

TABLE 5: LOADING DIAGRAM

Loads	Driver, Engine gravity loads at mounting position A moment of 1000 Nm applied on the axle of the chassis
Constraints	Front Wheel Hub mounting positions
Gravity	On

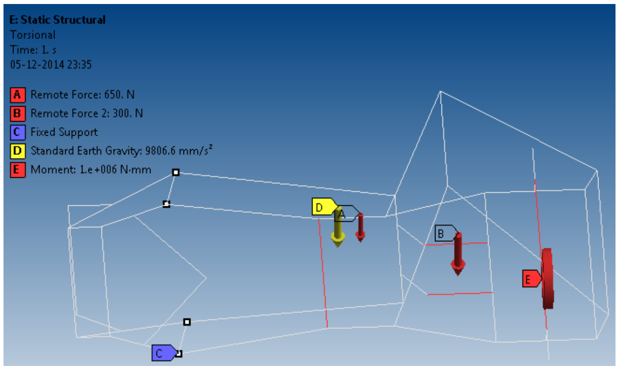

Fig. 12. Loading Diagram

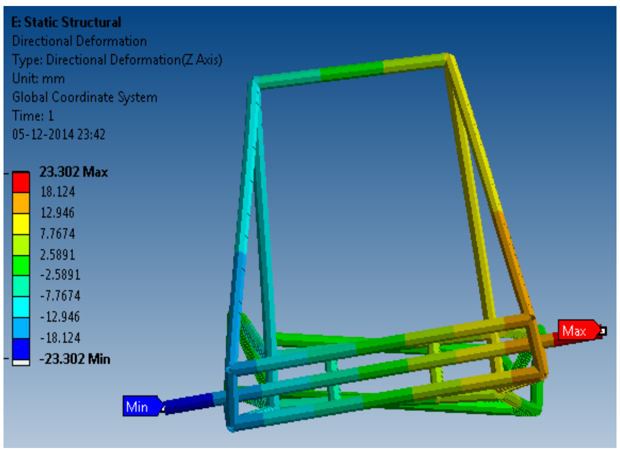

Fig. 13. Deformation Plot

Directional Deformations at the ends of the axle are 23.1mm and 23.6mm.

$$Twist = \tan^{-1}\frac{D_1 + D_2}{L} = \tan^{-1}\frac{46.7}{850} = 3.144\ degrees$$

$$Torque = 1000Nm$$

$$Torsional\ Rigidity = \frac{Torque}{Twist} = \frac{1000}{3.144} = \frac{318\ Nm}{deg}$$

A similar setup in COSMOS yielded a rigidity value of 305 Nm/deg. Since this value lies within the standards adopted, it is acceptable.

TABLE 6: LOADING DIAGRAM

Constraints	All Wheel Hub mounting positions
Gravity	On
LOADS	A 3g acceleration applied on the kart. Corresponding forces are applied on remote mass CGs

Bending test: In this test, a lateral acceleration is applied on the kart to simulate cornering forces. The acceleration applied corresponds to a 2g turn. The stresses developed in various members is then analysed.In a race, a 2g turn can be easily visualized as doing a turn of 2m radius at a speed of 9m/s.

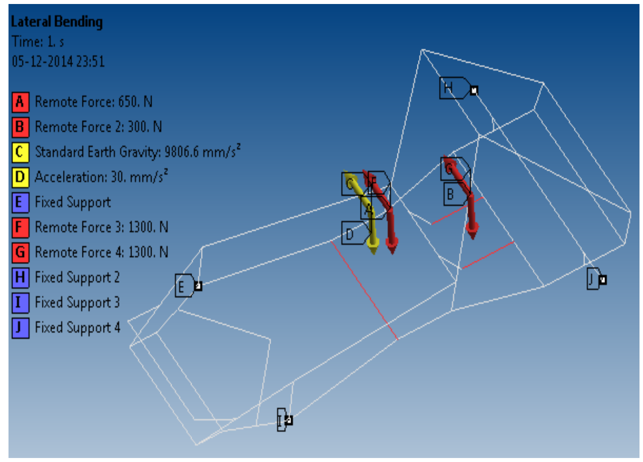

Fig. 14. Loading Diagram

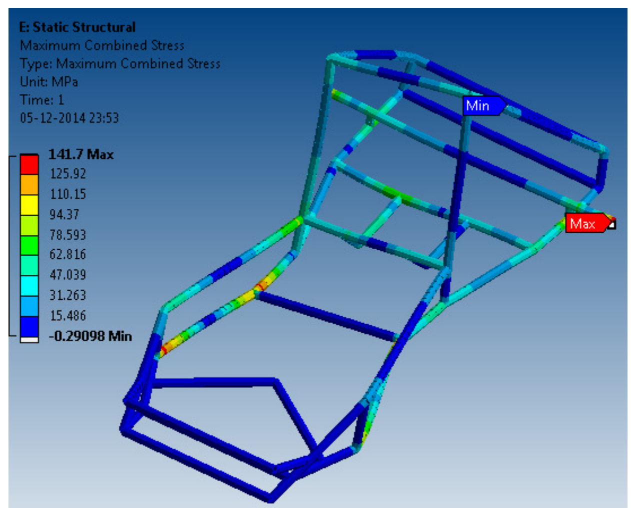

Fig. 15. Maximum Bending Stress Plot

The maximum bending stress induced at such a turn is 141.7 MPa which falls well short of the yield stress giving a FOS of 1.6. Hence the kart frame should have no structural trouble to do 2g turns.

Front Impacttest: This test determines the effect of a crash on the chassis at speeds up to 55 km/h (determined to be the maximum speed when brakes are applied at least for 0.5 seconds before crash).The collision time in a chassis without a crumple zone is statically averaged to 100ms. But the chassis of this go kart has an aluminium bumper and a thin deformable tube which can act as a crumple zone and increase the collision time to 150ms.

$$F.t\ (Impulse) = m * \Delta v \quad \Rightarrow \quad F = m * \left(\frac{\Delta v}{\Delta t}\right)$$

$$= 150 * \frac{55 - 0}{0.150} * \frac{5}{18}$$

$$\Rightarrow F = 11.2\ kN$$

TABLE 7: LOADING DIAGRAM

Loads	**11.2kN applied on front cross members**
Constraints	**Rear cross members**
Gravity	**On**

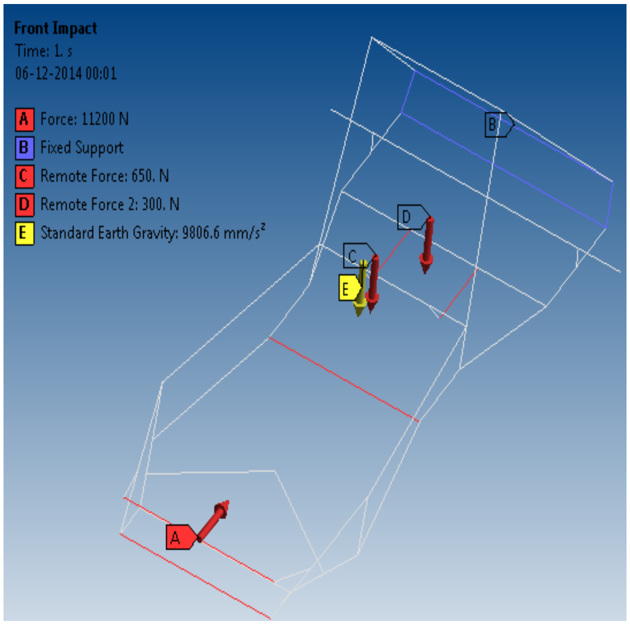

Fig. 16. Loading Diagram

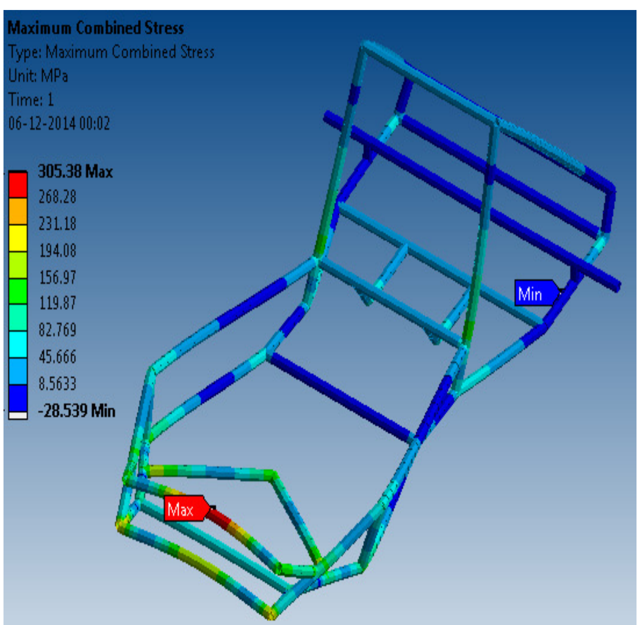

Fig. 17. Maximum Bending Stress Plot

The FOS of the front cross members amount to 0.8 which indicates that they will fail during collision. However the FOS in the cockpit has a minimum value of 1.8 which is safe enough.

A similar study done in COSMOS yielded a minimum FOS of 1.6 in the cockpit region. Hence the lower of thetwo values is greater than the chassis standard of 1.5 and our model can be said to be validated against front impact test at a speed of up to 55 km/h.

Side Impact test: This test determines the effect of a crash on the chassis when another kart collides with it on the side members at an angle of 45 degrees. The maximum speed difference between thekarts in such a collision is taken to be 25 km/hr. The collision can be modelled as two component forces acting on the side members with a resultant equal to total force applied which are calculated below.

Suppose the test chassis is at rest and another chassis collides into it at a relative speed of 25 km/h at 45 degrees. We apply the momentum theory to this situation considering

e=0.5.

$$v_1 - v_2 = 0.5 * 25 * \frac{5}{18} = 3.47 m/s and, v_1 + v_2 = 25 * \frac{5}{18} = 6.94 m/s$$

$$Then\ v_1 = \frac{6.94 + 3.47}{2} = 5.22 m/s$$

$$F.t\ (Impulse) = m * \Delta v \Rightarrow F = m * \left(\frac{\Delta v}{\Delta t}\right) = 150 * \frac{5.22-0}{0.150}$$

$$\Rightarrow F = 5.22\ kN\ at\ 45\ degrees$$

TABLE 8: LOADING DIAGRAM

Loads	5.2kN applied on side pod members
Constraints	Front and Rear wheel mounts on opposite side
Gravity	On

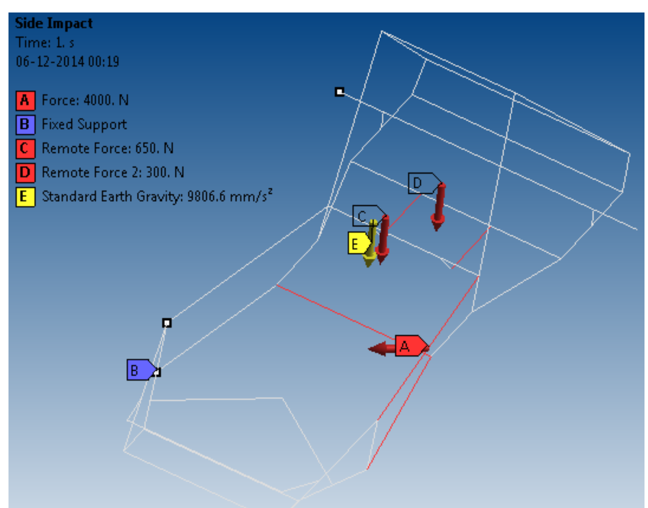

Fig. 18. Loading Diagram

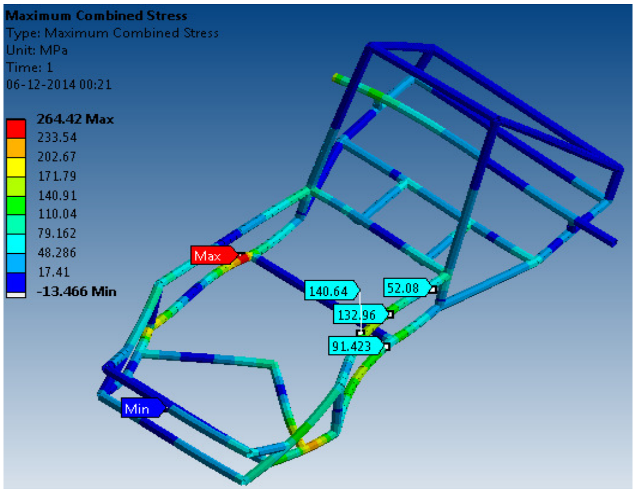

Fig. 19. Maximum Bending Stress Plot

The FOS of the cockpit amounts to 1.3 which indicates that they will not fail during collision. However the FOS in the cockpit has a minimum value of 3 which is safe for the driver asimilar study done in COSMOS yielded a minimum FOS of 1.4 in the cockpit region.Hence the lower of the two values is greater than the chassis standard of 1.5 and our model can be said to be validated against side impact test at a speed difference of up to 25 km/h.

5. CONCLUSIONS

This project helps us to understand the vital components of designing. As mentioned above the yield strength of the material which we are using is 250 MPa. The maximum value of stress generated while side impact on chassis excluding the bumpers is 141.7 MPa which is well within the limits. And therefore, the factor of safety of our chassis is 1.76. Safety is of utmost concern in every respect; for the driver, crew & environment. Considerable factor of safety (FOS) or design factors is applied to the chassis design to minimize the risk of failure & possible resulting injury. This FOS value implies the safe value of applied loads and deformations.

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