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Buggy Role Cage – Analysis and Design

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ABSTRACT

Design and analysis of Buggy roll cage commonly used as a recreational vehicle on off road terrains have been studied. These vehicles are usually modified from their existing design to provide performance and safety. In this research paper, an attempt has been made to design a roll cage for a buggy considering different members of the roll cage. In this context roll cage has been drawn on solid works CAD software and has been analyzed using finite analysis by applying the different boundary conditions. The roll cage has been designed considering the AISI 4130 steel and carbon fiber considering five cases such as front impact, side impact, rear impact, drop test and roll over test to ensure safety of the operator to survive the impact scenario. The main significance of this study is to analyze buggy role cage from safety point of view of operator. From these results it can be concluded that carbon fiber roll cage is also one of the promising alternatives for roll cage design and can be recommended from safety point of view.

1. INTRODUCTION

A buggy or all-terrain vehicle is a recreational vehicle consisting of large wheels, wide tires, designed for use on sand dunes or beaches. The design typically incorporates a modified vehicle and engine mounted on an open chassis frame [1, 2]. The modifications usually attempt to increase the power to weight ratio by either lightening the vehicle and/or increasing engine power [3]. Dune buggies designed specifically for operation on open sandy terrain are called sand rails. In this context Gupta et al. [4] have studied design and manufacturing of all-terrain vehicle considering static, dynamic analysis, and mathematical modelling. There main objective is to design high performance, easy maintenance and safety of an ATV with reasonable prices by implementing simplicity in design. During the design process they considered consumer interest though some innovative and effective methods. They concluded that the design of vehicle has good ergonomic, aerodynamic and highly engineered and can be easily manufactured. Raina et al. [5] have studied roll cage of all-terrain vehicles for a BAJA competition. Their main aim is to design a high terrain vehicle as per the needs of the competition. They considered the material selection, pipe selection and tested using ANSYS workbench considering safety working of the vehicle. They concluded that the designed vehicle is capable to withstand various load under extreme conditions. Eswara and Ramana [6] have studied roll cage for all-terrain vehicle considering static and dynamic analysis. They analyzed all-terrain vehicle considering different materials. They concluded by giving different analysis report of the chassis design for different types of materials. Gautam et al. [7] have studied design optimization of a roll-cage using finite element analysis. They designed and optimized roll cage considering AISI-1020 for maximum load bearing capacity and maneuverability. They concluded by using this AISI-1020, it is possible to develop a lighter roll cage as compared to conventional design. Qamar and Hasan [8] have studied performance of all-terrain roll cage vehicle during crash in the real-life scenarios. During the investigation they considered mass of all the elements including engine, steering assembly etc. They concluded that roll-cage stress and deflection was within the safety limits. Aru et al. [9] have studied roll-cage for autonomous vehicle. They considered materials, forces, impacts for safety of the roll-cage design. They concluded that the designed roll-cage is stiffer and stronger. Safiuddeen et al. [10] have studied roll cage vehicles for automobile applications. They studied using finite element analysis the structure of a cage roll for crash worthiness during collision. They concluded by testing front, side and roll over test by considering ASTM A36 steel properties by comparing ANSYS results with LS dyna. Abhinav [11] have studied roll-cage of an ATV considering inertia. He studied chassis of an automobile to study the crash and the effect considering safety aspects using transient multibody analysis. He concluded that this method can be used to analyze inertia forces in case of ATV. Shrivastava [12] have studied Quad bike for an off-road vehicle and analyzed chassis, suspension drive train etc. He considered the design process considering consumer interest as a primary goal. He concluded that their design improves the durability of various systems used in the car. Shiva et al. [13] have studied chassis which is an important part of SAE-BAJA competition. They studied the material selection and cross section to analyze the frame for various loading conditions. They concluded that frame designed is safe during the worst-case crash scenario. Riley and George [14] have studied key concepts considering frame design analytically and experimentally. They studied the torsional stiffness and developed a simple spring model for frame and overall stiffness of a chassis. They concluded by giving more emphasis on experimental method in analyzing the stiffness. Naiju et al. [15] have studied roll-cage of a vehicle using finite element analysis. The drawings of a roll-cage have been done using CAD package considering different condition to study the structural members. They concluded by analyzing the bending moment using two different materials. The main highlight of this roll cage design is overall length of the vehicle has been reduced to ensure good performance during turning and handling of the vehicle. Also, strength to weight ratio have been reduced to ensure safety and efficiency of the vehicle. From this literature review it is observed that many authors have designed roll cage considering various CAD packages. The main difference between the roll cage designed by the authors and other literature review is that it is having less strength to weight ratio.

2. METHODOLOGY

The methodology chart is as shown in Figure 1. The overall length of the vehicle considered is restricted to be less than 108 inches so that during the turning, performance and the handling of the vehicle will be easy. For easy turning, compact design, the overall vehicle width has been restricted to 64 inches. To lower the center of mass of the vehicle, the overall height of the vehicle has been lowered. A generic high strength carbon fiber was used for performing analysis, to simulate the stresses and deformations if the chassis was produced using lightweight high strength carbon fiber, which is prevalent in high performance cars and monocoques for Formula racing vehicles and is compared with AISI 4130 steel. Carbon Fiber in general has extremely high strength to mass ratio, along with high rigidity and bending strength. However, these advantages of the material are offset by the high cost and labor intensive and difficult manufacturability. The material considered for the analysis is AISI 4130 steel and carbon fiber. The mechanical characteristics of AISI 4130 steel and carbon fiber is as shown in Table 1 and the chemical properties of AISI 4130 steel is as shown in Table 2.

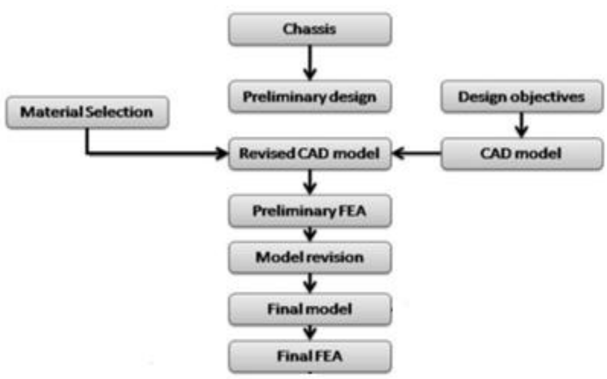

Figure 1. Methodology chart

Table 1. Mechanical properties of AISI 4130 steel and carbon fiber

Material	AISI 4130 Steel	Carbon Fiber
Density (kg/m^3)	7850	2000
Modulus of Elasticity (GPa)	205	228
Yield Strength (MPa)	460	725
Ultimate Tensile Strength (MPa)	731	1400
Poisson's Ratio	0.285	0.2

Table 2. Chemical properties of AISI 4130 steel

Element	Content (%)
Iron, Fe	97.03-98.22
Chromium, Cr	0.80-1.10
Manganese, Mn	0.40-0.60
Carbon, C	0.28-0.33
Silicon, Si	0.15-0.30
Molybdenum, Mo	0.15-0.25
Sulfur, S	0.04
Phosphorous, P	0.035

3. ROLL CAGE TUBE AND SIZE SELECTION

The most generally available tubes shapes are round, square, and rectangle. The round shape is commonly used for the space frames. Round tubing is superior to square in compression and torsional strength for a given weight. In terms of tensile strength and shear strength, they are similar. Bending strength is a function of the force vector. Circular tubing is much stronger than square due to the even weight distribution. Square has weak points, such as the edges and corners, where stress concentration occurs.

The selection of cross section geometry is restricted by standards followed in the industry. To keep costs low, it is necessary to employ widely available tubing sizes and materials. Tubing is available in standard fractional sizes to the 1/8th of an inch: 1, 1.125, 1.25, 1.375, and 1.5. The thickness of the tube wall is limited to the common tubing gauges.

It is optimum to choose the lease possible diameter for the tubes, as a smaller diameter translates into less weight, and therefore, improved performance. Large diameter tubes with relatively thinner walls will have an appropriate balance between strength and weight. The aspect of tube size selection in the design process needs careful consideration. When comparing a square and circular tube of the same dimensions, the circular tube will have more desirable properties in all aspects other than bending. However, when compared from a mass standpoint, the square tube will have thinner walls, and thus will not necessarily have better bending properties than circular tubing. Another benefit of round tubing lies in the improves bucking resistance per unit weight.

The frame is designed for a dune buggy, and hence mass of the space frame is a critical factor and must be considered. For a good design, an appropriate balance must be struck between fulfilling the design targets and minimizing mass of the roll cage. For SAE BAJA, the specifications specify the roll cage to be produced with materials equivalent to circular steel tubing with an outside diameter of 1 inch (25.4 mm). The wall thickness of the tubing must be 0.120 in (3mm) and a carbon content of at least 0.18%.

3.1 Determination of bending strength & bending stiffness

Bending stiffness, kb, is given by $kb = E.I$ (1)

where, E -Modulus of elasticity, I –Area moment of inertia
Bending strength, Sb, is given by

$$Sb = SyI/C \tag{2}$$

where, Sy - Yield strength, I - Area moment of inertia, c - Distance from neutral axis to extreme fiber.

From tubing geometry:

E = 205 GPa for AISI 4130 Steel, E = 228 GPa for Thornel Mat Carbon Fiber, Do = 25.4mm Thickness t = 3mm, Inner diameter Di =19.4mm

Area Moment of Inertia:

$$Ix = [\pi x (Do4 - Di4)]/64c \tag{3}$$

Ix = π x (25.44 - 19.44) / 64 = 13478.63 mm4

Material: AISI 4130 Steel

Yield strength, Sy = 460 MPa Distance to the extreme fiber, c = Do/2 = 25.4/2 = 12.7mm (4)

Bending Stiffness, kb = E I = 205 x 103 x 13478.6 = 2763.11 Nm^2.

Bending Strength, Sb = Sy I/C = (460 x 13478.63)/12.7 = 488.2 N m.

3.2 Material: Carbon fiber

Yield strength, Sy = 725 MPa, Distance to the extreme fiber, c = Do/2 = 25.4/2 = 12.7mm.

Bending Stiffness, kb = E I = 228 x 103 x 13478.63 = 3073.12 Nm^2.

Bending Strength, Sb = Sy I/C = (725 x 13478.63)/12.7 = 769.4 Nm.

3.3 Design of roll cage

The roll cage is designed with multiple tubes which form a tubular space frame. This helps to distribute loads and stresses throughout the frame and forms a rigid structural body. It serves the driver with a safe enclosed 3-D structure and creates structural support for all the necessary systems. To ensure a strong frame without compromising on weight, it is always beneficial to create a frame of interconnecting triangular sections, as this allows for a coherent load path for the roll cage as shown in Figure 2. The load path is the route that the forces and subsequent stresses follow through a structure. When a coherent load path does not exist inside the structure, the forces and stresses react in a single tube, generating a high stress gradient in the area and significantly increasing the probability of failure. A good load path will disperse the loads and stresses across interconnected members throughout the roll cage.

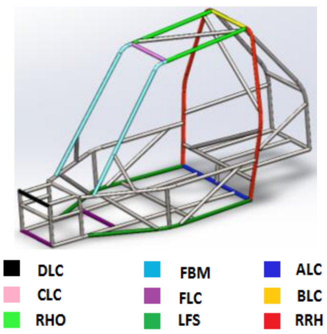

Figure 2. Primary members of the roll cage

3.4 Primary members of the roll cage

The Rear Roll Hoop, also known as firewall, is a panel positioned behind the driver and the rear side of the roll cage. It defines the boundary between the front half and the rear half of the roll cage and separates the engine from the driver. The rear roll hoop has been designed as a single continuous bent tube, with no breaks, to give it greater strength. It is substantially vertical, inclined at an angle of 5 degrees from vertical. It consists of two Lateral Cross members (Overhead lateral cross member and Aft lateral cross members) at the vertical extremes, joining the vertical bent members.

The Roll Hoop Overhead members define the topmost horizontal plane of the vehicle. They connect the firewall to the front bracing members and must be designed to ensure adequate headspace of the occupant.

Lower Frame Side members distinguish the lower left and right edges of the frame of the vehicle. They connect the rear roll hoop and the front lateral cross member. Side Impact members consist of two members which lie on a plane horizontal above the base plane. These members absorb any impacts from either side of the vehicle. They are connected to the rear roll hoop and extend horizontally towards the front.

Front Bracing Members are continuous members which connect the roll hoop overhead and the side impact members. They are crucial to the rigidity and structural integrity of the frame.

3.5 Frame specifications

The center of gravity of roll cage have been analyzed and is as shown in Figure 3. The vertical position of the center of mass is found to be 35.6 cm. If a vehicle has a low center of mass, it always minimizes the probability of the vehicle toppling or rolling over. The specifications of the frame considered is shown in Table 3 and the corresponding front, side, isometric and top views is as shown in Figure 4.

Design and manufacturing of dune buggies require distinctive approaches as compared to traditional passenger vehicles due to complexity of loading conditions and impact occurring in the chassis at critical points. Apart from this for off-road vehicles like buggies loads due to longitudinal acceleration and steady state turning exits in the rigid body. However, these loads are smaller in magnitude as compared to forces on the roll cage due to impact scenarios. This assumption makes the design process easier as it ensures roll cage is robust enough to withstand the rigid body loads if it can withstand the various impact loads. So, in this context the design of space frame for off road racing needs to be taken into consideration which otherwise will cause damage and failure of the roll cage and suspension.

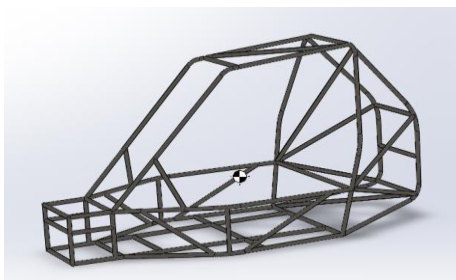

Figure 3. Centre of gravity of the roll cage

Table 3. Specifications of the frame

Specifications	Dimensions
Length	215.12 cm
Width	98.7 cm
Height	120 cm
Mass (AISI 4130)	59.5 kg
Mass (Carbon Fiber)	15.1 kg
Centre of Gravity Coordinates	X: 107.29 cm, Y: 35.62 cm, Z: -0.5 cm

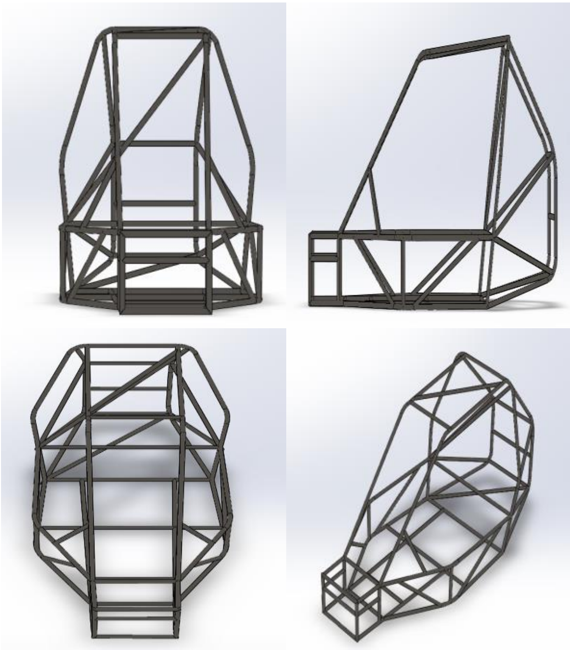

Figure 4. Front, side, isometric, and top views of the roll cage

For the purposes of numerical analysis, some of the assumptions have been considered. The mass of the drivetrain is of 45 kg. This includes the engine, transmission box, and connecting chains. The weight of the suspension system and tires is accounted to be a total of 70 kg. This includes the suspension arms, shocks, and tabs. The mass of the driver is 100 kg. Mass of the roll cage was estimated to be 60 kilograms for AISI 4130 steel by using the SolidWorks mass evaluation tool. The materials considered in the analysis were AISI 4130 steel and Carbon Fiber with a 25.4 mm outer diameter and 3mm wall thickness tubes. The force equations stated in the test descriptions were applied to each load to simulate the acceleration experienced during the impact.

The space frame is constructed of tubular members and the members are interconnected in the form of a 3-D truss, where they experience tensile or compressive forces. The structural strength and integrity of the roll cage needs to be high to ensure that failure does not occur during impact scenarios. The greatest impact loads that a vehicle can experience can be that from frontal impact. The load experienced by the roll cage for frontal impact accident scenario needs to be calculated. It is obtained by studying the force acting on the roll cage if it undergoes a head on collision with a rigid body, such as a wall. The gross mass of the buggy with the driver is 275 kg. The front impact load is calculated using basic mechanics and the corresponding front impact scenarios is as shown in Figure 5 and the corresponding loading and boundary conditions is as shown in Figure 6.

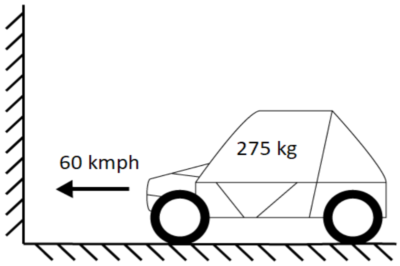

Figure 5. Front impact scenario

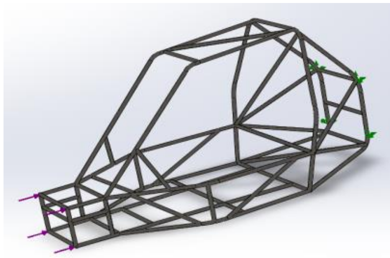

Figure 6. Boundary conditions and loading for the frontal impact test

Theoretical calculations for the roll cage are as follows. Gross Mass of the dune buggy, m = 275 kg (Gross weight), Velocity of the buggy, v = 60 kmph = 16.66 m/s (maximum velocity), Time of impact, t = 0.15s (Time from max speed to full stop).

Acceleration, $a = (v - u)/\,t$ [20] = (16.66-0)/0.15s = 111 m/s^2.

From Newton's second law, Force, F = m x a = 275 x 111 = 30525 N.

The geometry of the frame is imported on SolidWorks Simulation static stress analysis. A point load of 30.5 KN is divided and applied to the four joints located at the front most members of the frame i.e. the nose. The displacement of 4 points of the rear most end of the frame is fixed for all degrees of freedom. The calculated force for front impact test is roughly equal to a force of 11 Gs. This deceleration in the front impact test is taken as a worst-case for human body. This conforms with the upper limit of deceleration a human body can endure before passing out i.e. 9 G.

3.6 Side, rear and drop impact analysis

The analysis for the side impact testing of the buggy roll and the corresponding side impact scenario considering loading, boundary conditions is as shown in Figure 7 and Figure 8. The side impact testing analysis of the buggy roll cage is performed to ensure that the roll cage does not fail during a collision impacting the side members of the frame. The chassis is oriented normal and laterally, relative to the oncoming vehicle for the side impact test. A fraction of the energy resulting from the impact is always transferred to the suspensions and the wheels and is not considered in the analysis.

The buggy roll cage is oriented sideways relative to the motion of the incoming 275 kg vehicle. The load required for the side impact is achieved by creating a situation in such a way as it is moving towards the other vehicle's side at a speed of 65 kmph. The mass of the car including the driver is assumed to be 280 kg. The loads to be applied to the chassis are calculated based on the equations used in the front impact analysis. To perform this analysis, the geometry of the frame is imported on solid works simulation static stress analysis. Then the force is applied to the outermost points of one side of the roll cage and the corresponding opposite points of the roll cage are constrained in all DOF.

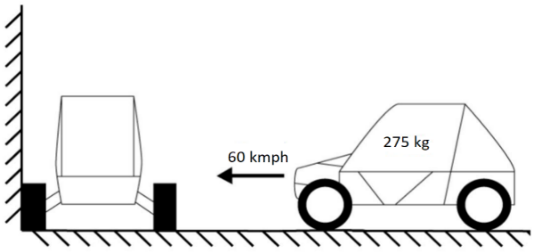

Figure 7. Side impact scenario

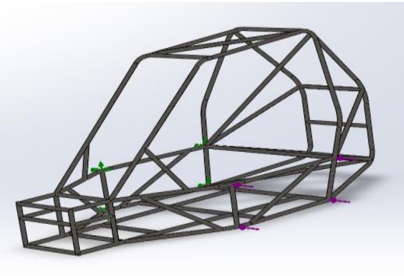

Figure 8. Boundary conditions and loading for the side impact test

The rear impact analysis has been performed by considering another 275 kg vehicle moving at a speed of 60 kmph and is colliding with this vehicle from the backwards. The forces are calculated in such a way that the load is applied to the rear most ends of the roll cage of the buggy and the front ends of the roll cage are constrained. The rear impact scenario and the corresponding loading and boundary conditions is as shown in Figure 9 and Figure 10. In this analysis it was observed that the vehicle has experienced upwards forces of 11 G's in worst case scenario.

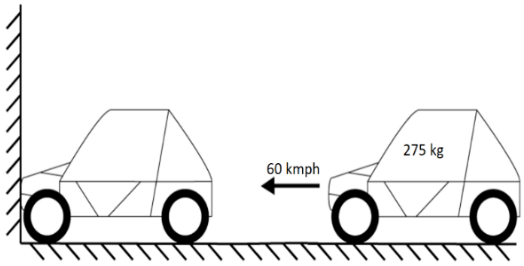

Figure 9. Rear impact scenario

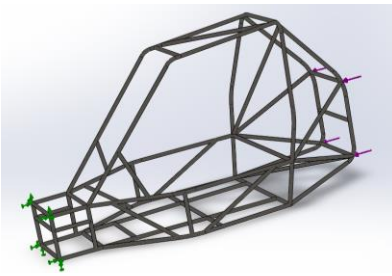

Figure 10. Boundary conditions and loading for rear impact test

The drop impact test simulates the forces acting on the vehicle when it rolls over and drops from a height. It is presumed that the vehicle falls from a height of 3m following a roll over. The weight of the vehicle considered is 275 kg and the impact time is 0.15 seconds. The corresponding drop impact scenario and loading and boundary conditions for drop impact test is as shown in Figure 11 and Figure 12. The drop impact test boundary conditions are set by applying the calculated load to the roof of the roll cage of the buggy. The bottom of the chassis is fixed and constrained in all degrees of freedom. The potential energy of the vehicle is converted into kinetic energy when dropped and reaches the ground with impact velocity. To analyze the frame in a drop scenario, the following equations is used to determine the force of impact.

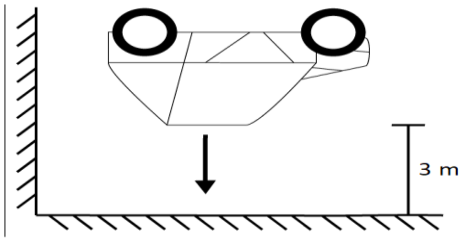

Figure 11. Drop impact scenario

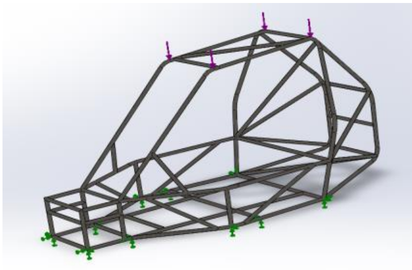

Figure 12. Boundary conditions and loading for the drop impact test

Potential Energy, P.E = m x g x h; Drop Height, h = 3m, Impact velocity, v = $\sqrt{2xgxh} = \sqrt{2x9.81x3} = 7.672$ m/s, Impact time, t = 0.15s, Force, F = $m \cdot \sqrt{2gh}/t = 275*7.672/0.15 = 14065.38$ N.

3.7 Roll over analysis

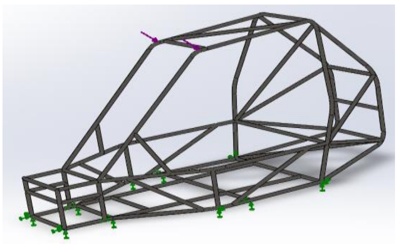

Figure 13. Boundary conditions and loading for the roll over test

In the roll over test, the behavior of the roll cage when a load of 5 Gs is applied in a roll over scenario is simulated. The load acts on the upper top corners of the roof (roll hoop overhead) members at an angle of 45 degrees. Force, F = 5 G = 5 x 9.81 x 275 = 13488.75 N. The roll over analysis is accomplished by applying the computed force to the roof members of the roll cage. The force is applied at an angle of 45 degrees relative to the base plane. The bottom of the chassis is fixed and constricted in all degrees of freedom. The corresponding loading and boundary conditions is as shown in Figure 13.

4. RESULTS AND DISCUSSIONS

The 3D buggy roll cage model was created using SolidWorks and the finite element analysis was conducted using the Solid works static analysis. The analysis was run by meshing the roll cage into a beam mesh. This mesh converts the body in annular partitions and solves for the results. The results of finite element analysis were performed for 5 accident scenarios considering AISI 4130 steel and carbon fiber material. The corresponding mesh details, analysis properties and result type are as shown in Table 4. The mechanical characterization of both the material considered for analysis is as shown in Table 5.

Table 4. Analysis characteristics

Mesh Details	
Mesh type	Beam mesh
Total Nodes	965
Total Elements	883
Analysis Properties	
Model type	Linear Elastic Isotropic
Failure Criterion	Maximum Von-Mises stress
Failure Criterion	Maximum von-Mises stress
Result Type	
Deformation	URES displacement
Stress	Upper bound axial and bending stress (von-Mises stress)

Table 5. Mechanical characteristics of AISI 4130 and carbon fiber

Material	AISI 4130 Steel	Carbon Fiber
Density (kg/m^3)	7850	2000
Modulus of Elasticity (GPa)	205	228
Yield Strength (MPa)	460	725
Ultimate Tensile Strength (MPa)	731	1400
Poisson's Ratio	0.285	0.2

4.1 Frontal impact analysis

The geometry of the frame is imported on SolidWorks Simulation stress analysis. A uniformly distributed load of 30525 N is applied to the four joints located at the front most members of the frame i.e. the nose. The displacement of 4 points of the rear most end of the frame is constrained in all degrees of freedom. The corresponding FEA comparison results of AISI 4130 steel and carbon fiber material is as shown in Table 6 and the frontal impact stresses and displacement comparison for AISI 4130 steel and carbon fiber is as shown in Figure 14 and Figure 15. From the analysis results obtained it is observed that the factor of safety is 2.06 and 3.73 for the load of 30525 N is observed for both AISI 4130 steel and Carbon fiber correspondingly. Hence, from the point of consideration of factor of safety the design is considered safe both the material and factor of safety is more in case of carbon fiber material as compared to AISI 4130. From Figure 14 it is observed that the maximum stress is 223.1 MPa and 194.55 MPa for AISI 4130 steel and carbon fiber. From this it is seen that the maximum stress is more in case of AISI 4130 steel as compared to carbon fiber. From Figure 15, it is also observed that the maximum displacement is 3.86 mm and 2.10 mm in case of AISI 4130 steel and carbon fiber respectively and from this it can be concluded that the maximum displacement is less in case of carbon fiber as compared to AISI 4130 steel.

Table 6. Comparison of FEA results of frontal impact scenario

Front Impact Analysis Results	AISI 4130	Carbon Fiber
Load (N)	30525	30525
Max Stress (MPa)	223.1	194.55
Max Displacement (mm)	3.86	2.10
Factor of Safety	2.06	3.73

4.2 Side impact analysis

The buggy roll cage is oriented sideways relative to the motion of the incoming 275 kg vehicle. The load required for the side impact is achieved by creating a situation where one car is moving towards the other vehicle's side at a speed of 65 kmph. The mass of the car including the driver is assumed to be 280 kg. Table 7 shows the comparison of FEA results for AISI 4130 steel and carbon fiber. From this table it is observed that factor of safety is 2.75 and 4.65 for AISI 4130 steel and carbon fiber. From this it can be concluded that both the material is safe from factor of safety point of View. Figure 16 shows the comparison of maximum stress for both AISI 4130 steel and carbon fiber and from this Figure 16 it is observed that the maximum stress is 155.96 MPa and 155.92 MPa for both AISI 4130 steel and carbon fiber. From this it is observed that the maximum stress approximately remains same in both the material considered. Figure 17 shows the comparison of displacement for both AISI 4130 steel and carbon fiber. From this it is observed that maximum displacement is 4.75 mm and 5.66 mm in case of AISI 4130 steel and carbon fiber and maximum displacement is more at carbon fiber as compared to AISI 4130 steel. This occurs due to the difference in hardness between the carbon fiber and steel. In case of carbon fiber, the hardness of the material will be less, and tensile stresses will be higher which in turn have a maximum amplitude to absorb the shocks. Also, it takes the maximum load and corresponding maximum displacement will be more in case of carbon fiber as compared to steel. But the carbon fiber material will behave just like spring back effect and it will come back to initial state of stress.

Table 7. Comparison of FEA results of side impact scenario

Side Impact Analysis Results	AISI 4130	Carbon Fiber
Load (N)	30525	30525
Max Stress (MPa)	155.96	155.92
Max Displacement (mm)	4.75	5.66
Factor of Safety	2.75	4.65

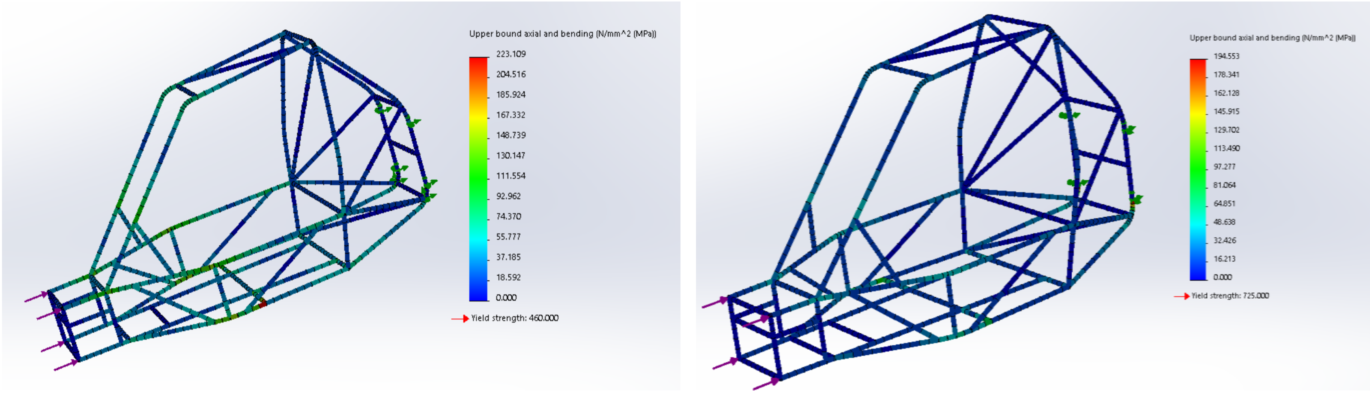

Figure 14. Comparison of frontal impact FEA stress for AISI 4130 and carbon fiber

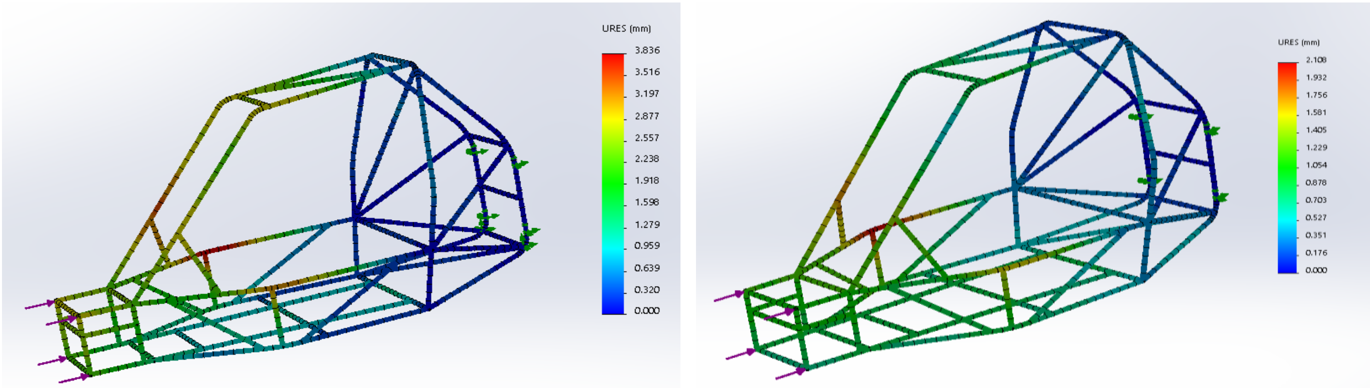

Figure 15. Comparison of frontal impact FEA displacement for AISI 4130 and carbon fiber

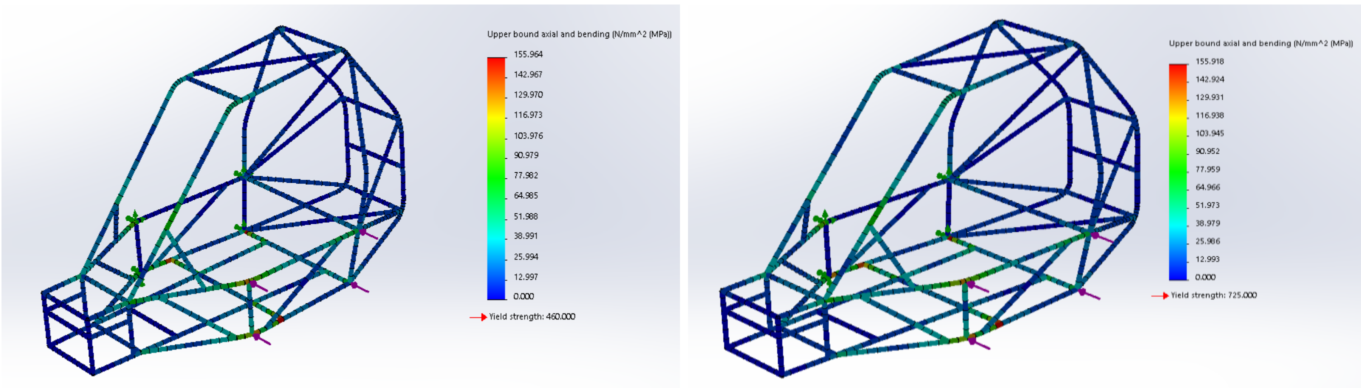

Figure 16. Comparison of side impact FEA stress for AISI 4130 and carbon fiber

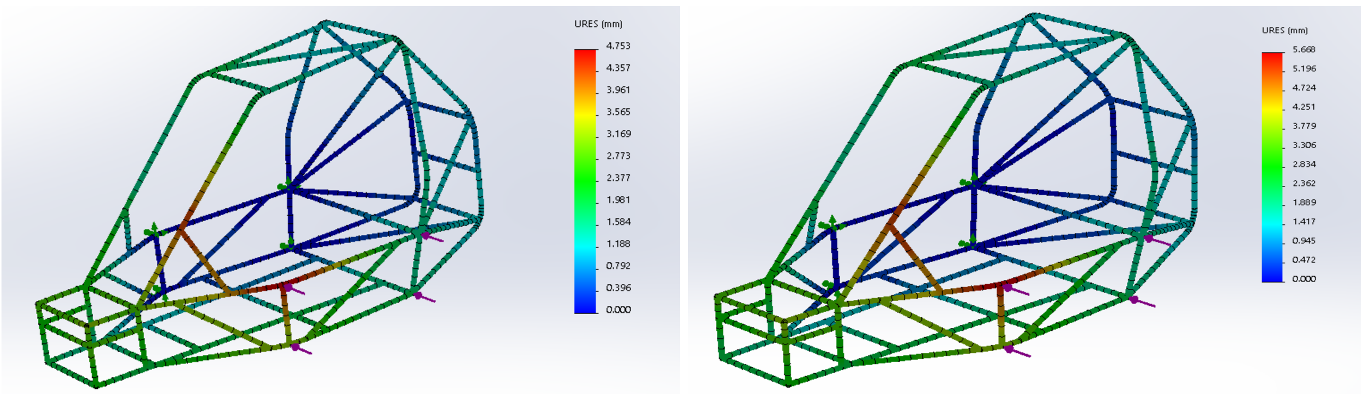

Figure 17. Comparison of side impact FEA displacement for AISI 4130 and carbon fiber

4.3 Rear impact analysis

The rear impact analysis tests the event of the vehicle being rear-ended by another 275 kg vehicle, again at a speed of 60 kmph. The calculated forces are applied to the rear most ends of the roll cage of the buggy i.e. the engine bay, and the front ends of the roll cage are constrained. Table 8 shows the comparison of side impact scenario for both AISI 4130 steel and carbon fiber. From this Table 8 it is observed that the factor of safety is 3.53 and 5.92 for AISI 4130 steel and carbon fiber and hence it can be concluded that both the materials are safe from factor of safety point of view. Figure 18 shows the comparison of maximum stress for both AISI 4130 steel and carbon fiber. From this it is observed that maximum stress is 130.12 MPa and 122.44 MPa for AISI 4130 steel and carbon fiber. From this Figure 18 it is observed that the maximum stress occurs at carbon fiber as compared to AISI 4130 steel and occurs due to the hardness of the material. The hardness of the material in case of carbon fiber is less as compared to steel and hence the stresses are more in case of carbon fiber as compared to steel. Figure 19 shows the comparison of maximum displacement in case of AISI 4130 steel and carbon fiber. From this Figure 19 it is observed that maximum displacement is 9.02 mm and 10.32 mm in case of steel and carbon fiber. In this case also the maximum displacement is higher s in case of carbon fiber as compared to steel and behaves in the same way as discussed in side impact analysis.

4.4 Drop impact analysis

The drop impact test simulates the forces acting on the vehicle when is rolls over and drops from a height. The geometry of the frame is imported on SolidWorks Simulation static stress analysis. The drop impact test boundary conditions are set by applying the calculated load to the roof of the roll cage of the buggy. The bottom of the chassis is fixed and constrained in all degrees of freedom. Table 9 shows the comparison of FEA results for drop impact scenario for both AISI 4130 and carbon fiber. From this Table 9 it is observed that maximum factor of safety is 2.34 and 4.36 for both AISI 4130 steel and carbon fiber. From this it can be concluded both the material are safe from factor of safety point of view. Figure 20 shows the comparison of maximum stress for AISI 4130 steel and carbon fiber and from this Figure 20, it is observed that maximum stress is 195.85 MPa and 166.37 MPa for AISI 4130 steel and carbon fiber. In this case the maximum stress is higher in case of AISI 4130 steel as compared to carbon fiber. Figure 21 shows the comparison of maximum displacement for AISI 4130 steel and carbon fiber. From this figure it is observed that the maximum displacement is 3.65 mm and 3.72 mm for AISI 4130 steel and carbon fiber. From this it can be concluded that maximum displacement remains same in case of both AISI 4130 steel and carbon fiber.

Table 8. Comparison of FEA results of side impact scenario

Rear Impact Analysis Results	AISI 4130	Carbon Fiber
Load (N)	30525	30525
Max Stress (MPa)	130.12	122.44
Max Displacement (mm)	9.02	10.32
Factor of Safety	3.53	5.92

Table 9. Comparison of FEA results of drop impact scenario

Rear Impact Analysis Results	AISI 4130	Carbon Fiber
Load (N)	14065	14065
Max Stress (MPa)	195.85	166.37
Max Displacement (mm)	3.65	3.72
Factor of Safety	2.34	4.36

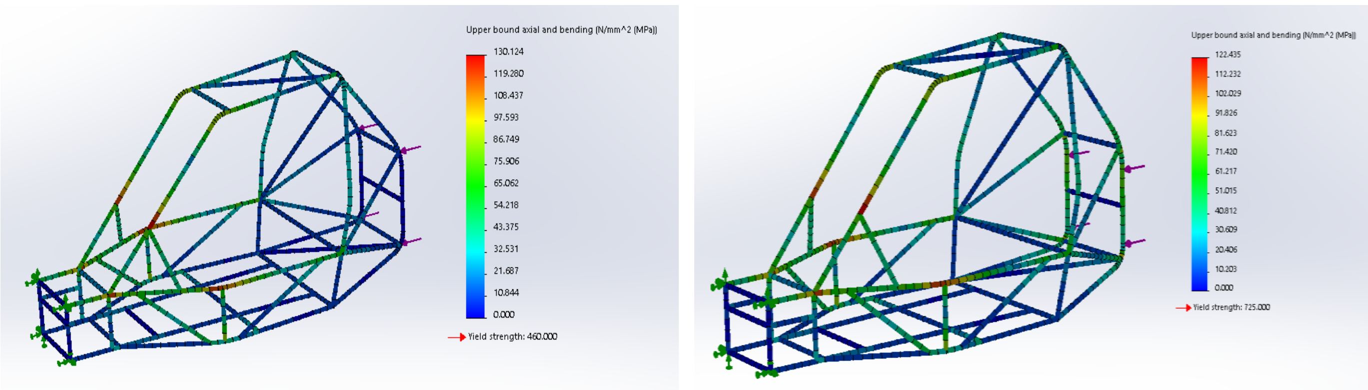

Figure 18. Comparison of rear impact FEA stress for AISI 4130 and carbon fiber

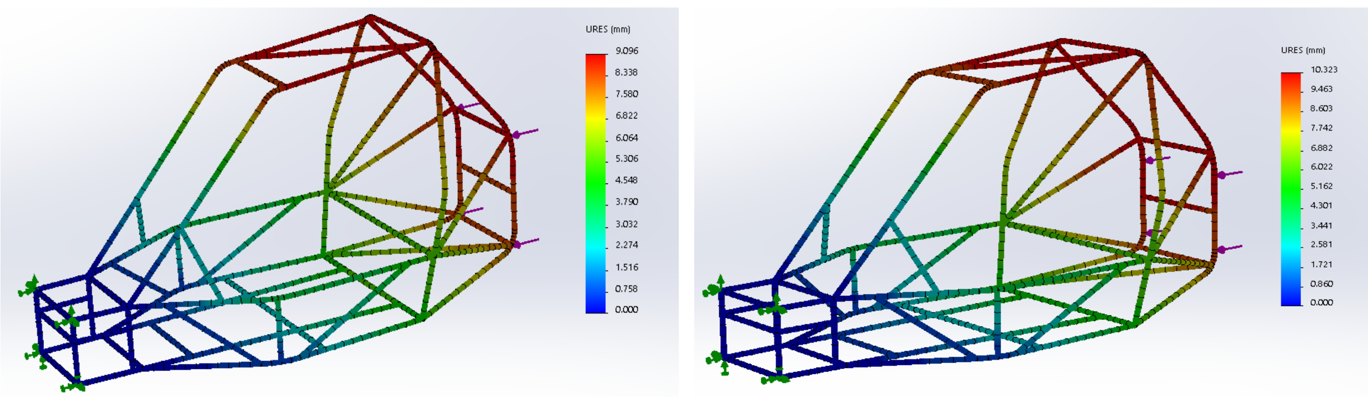

Figure 19. Comparison of side impact FEA displacement for AISI 4130 and carbon fiber

4.5 Roll over analysis

In the roll over test, the behavior of the roll cage when a load of 5 Gs is applied in a roll over scenario is simulated. The load acts on the upper top corners of the roof (roll hoop overhead) members at an angle of 45 degrees. Table 10 shows the comparison of FEA results of roll over analysis for both AISI 4130 steel and carbon fiber. From this analysis it is observed that factor of safety is 2.69 and 5 for both AISI 4130 steel and carbon fiber. From this Table 10, it is observed is safe from factor of safety point of view. Figure 22 shows the comparison of roll over analysis for AISI 4130 steel and carbon fiber. From this figure, it is observed that the maximum stress is 170.48 MPa and 144.92 MPa for AISI 4130 steel and carbon fiber. From this maximum stress analysis, the maximum stress is observed in case of AISI 4130 steel as compared to carbon fiber. Figure 23 shows the comparison of maximum displacement for AISI 4130 steel and carbon fiber and it is observed that maximum displacement is 3.77 mm and 3.85 mm. From this it is observed that maximum displacement is more in case of carbon fiber as compared to AISI 4130 steel. Table 11 shows the analysis results of various impact tests done for comparison purposes for both AISI 4130 steel and carbon fiber for various loading conditions and the corresponding maximum stress, maximum displacement, factor of safety is also tabulated. Table 12 shows the bending stress, bending strength and minimum factor of safety required for both AISI 4130 steel and carbon fiber. From this table it is observed that maximum bending strength is 488.2 MPa and 769.4 MPa for AISI 4130 steel and carbon fiber and is more in case of carbon fiber as compared to AISI 4130 steel. Also, from this table it is observed that bending stiffness is 2763.11 and 3073.12 for AISI 4130 steel and carbon fiber and from this it can be concluded that bending stiffness is more in case of carbon fiber as compared to AISI 4130 steel.

From the results of the analysis of five impact tests conducted, the chassis satisfies the engineering design targets. For building a buggy that demonstrates high factor of safety, low deformation, and low cost, AISI 4130 steel is the perfect material. It has high yield strength, low in cost, and high strength to weight ratio compared to other metals. However, with a weight of 60 kgs, there is a compromise in the performance aspect, as the high weight requires more power to drive the vehicle, hence lowering engine and fuel efficiency. Overall, it has a good balance between cost and strength. The carbon fiber roll cage has a total weight of only 15.1 kg, due to its substantially lower density of 2000 kg/m^3, compared to 7850 kg/m^3 of AISI 4130 steel. While optimization of the roll cage design can lower the weight of the steel welded roll cage, it can never match the extremely high strength to weight ratio of the carbon fiber molded roll cage.

Table 10. Comparison of FEA results of roll over analysis

Rear Impact Analysis Results	AISI 4130	Carbon Fiber
Load (N)	13488.75	13488.75
Max Stress (MPa)	170.48	144.92
Max Displacement (mm)	3.77	3.85
Factor of Safety	2.69	5.0

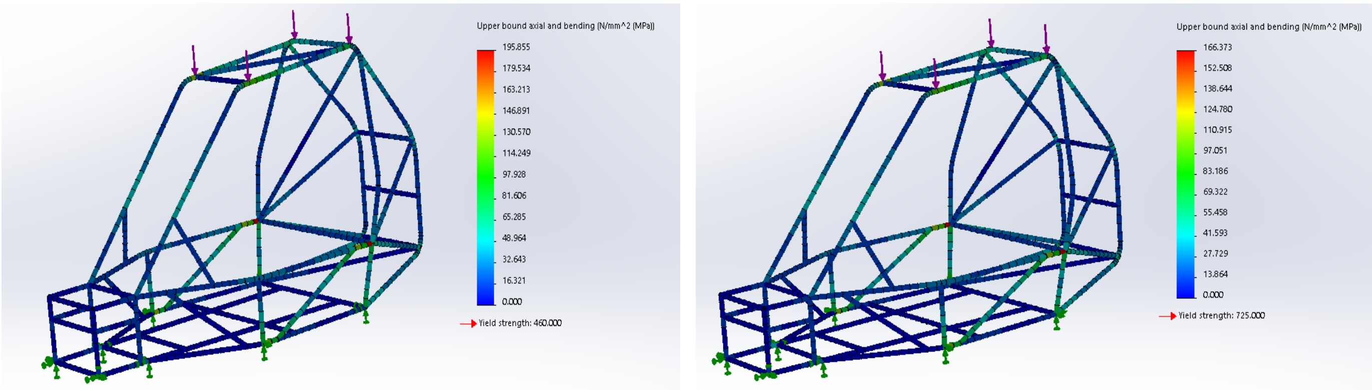

Figure 20. Comparison of drop impact FEA stress for AISI 4130 and carbon fiber

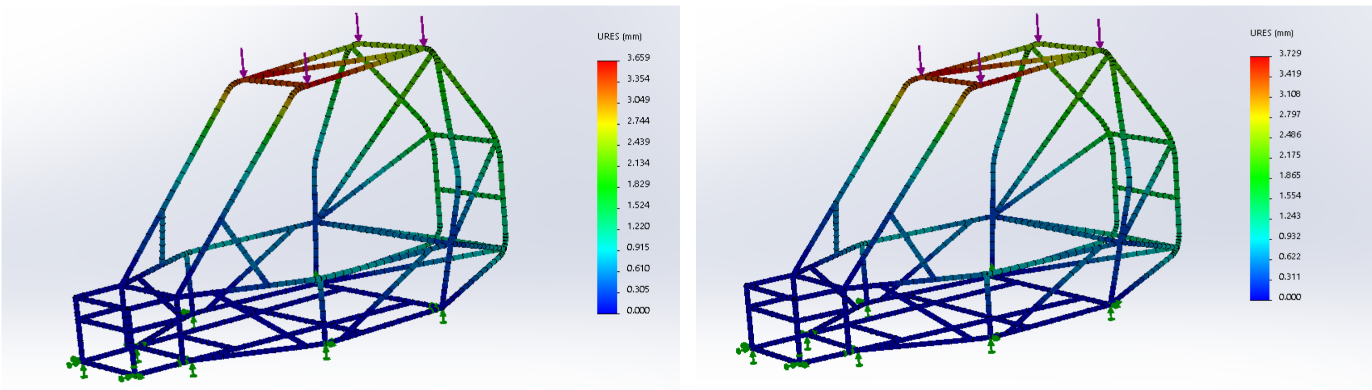

Figure 21. Comparison of drop impact FEA displacement for AISI 4130 and carbon fiber

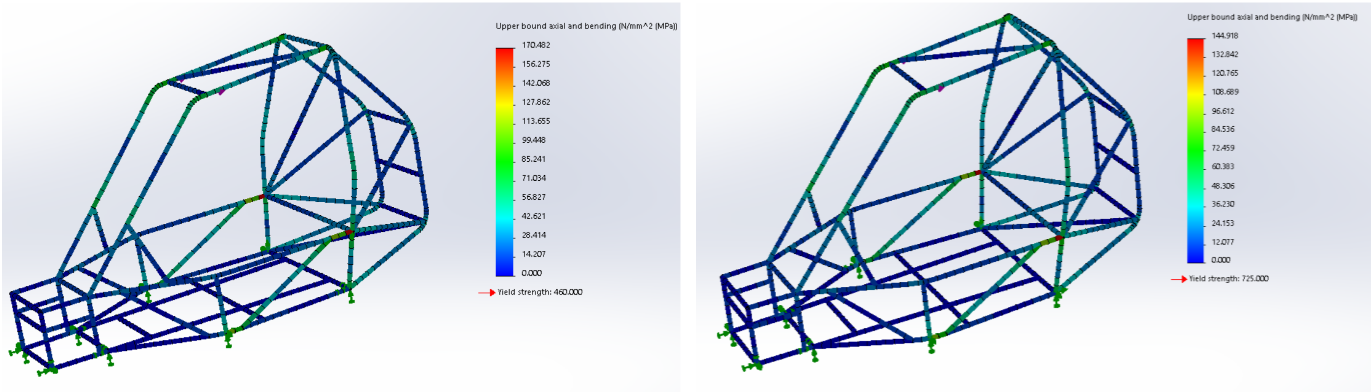

Figure 22. Comparison of roll over analysis for AISI 4130 and carbon fiber

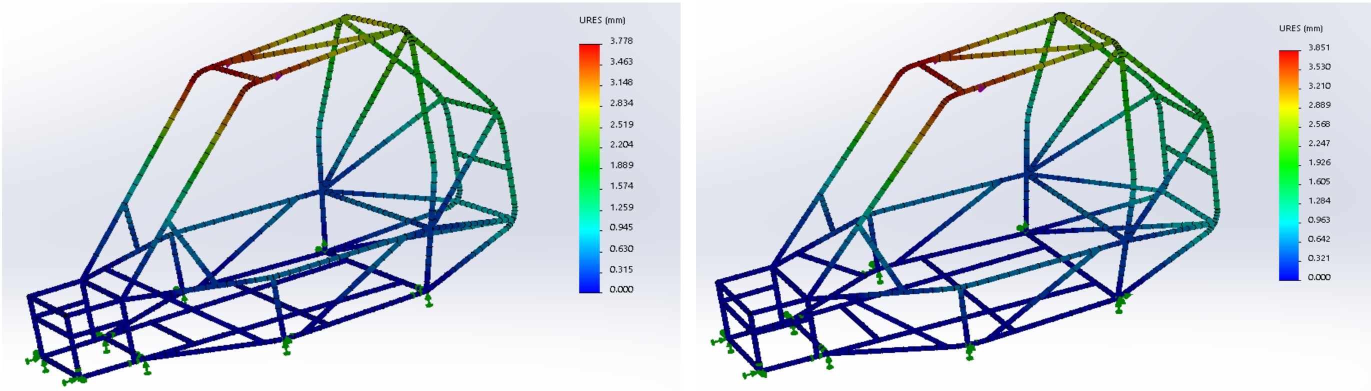

Figure 23. Comparison of roll over analysis for AISI 4130 and carbon fiber

Table 11. Analysis results of AISI 4130 and carbon fiber for various impact tests

Result / Tests	Loading (N)	AISI 4130			Carbon Fiber		
		Max Stress (MPa)	Factor of Safety	Displacement (mm)	Max Stress (MPa)	Factor of Safety	Displacement (mm)
Front Impact Test	30525	223.1	2.06	3.86	194.55	3.73	2.10
Side Impact Test	30525	155.96	2.75	4.75	155.92	4.65	5.66
Rear Impact Test	30525	130.12	3.53	9.02	122.44	5.92	10.32
Drop Test	14065	195.85	2.34	3.65	166.37	4.36	3.72
Roll Over Test	13488	170.48	2.69	3.77	144.92	5.0	3.85

Table 12. Engineering design targets compared to actual values

Requirement	Target	Actual	
		AISI 4130	Carbon Fiber
Length (cm)	< 270	215.12	215.12
Width (cm)	< 162	98.7	98.7
Bending Strength (Nm)	395	488.2	769.4
Bending Stiffness (Nm^2)	2763	2763.11	3073.12
Min. Factor of Safety	1.25	2.06	3.76

The carbon fiber roll cage exhibits very similar stress and deformation results compared to those of AISI 4130 steel. It, however, has much higher factor of safety due to its yield strength being 157.6% higher than AISI 4130 steel (725 MPa (carbon fiber) compared to 460 MPa (AISI 4130 steel)). The weight of the carbon fiber roll cage is 74.62% lower than that of AISI 4130 steel.

5. CONCLUSIONS

Design and analysis of Buggy roll cage commonly used as a recreational vehicle on off road terrains have been studied. Using solid works simulation the roll cage analysis has been done using Finite element analysis for five different impact scenarios considering front impact, side impact, rear impact, drop test and roll over test have been studied for both AISI 4130 steel and carbon fiber. From this analysis it can be concluded that

(1) High factor of safety is achieved, and the design is safe for both AISI 4130 steel and carbon fiber.

(2) Lower deformation is observed in case of both AISI 4130 steel and carbon fiber.

(3) Steel is a material which can be welded quite easily and is the better option in terms of manufacturability and costs. Steel tubes are readily available in a variety of gauges and shapes, and the technology and equipment requisite for manufacturing is widely accessible. Carbon Fiber requires skilled labor and special equipment to manufacture. The fibers need to be impregnated with epoxy resin and curing agents before it can be laid in the mold, which must be manufactured for the specific shape of the application. The carbon fiber must be laid in multiple layers to provide strength in all directions, increasing man hours. Therefore, carbon fiber is unviable in terms of both manufacturability and cost.

(4) The carbon fiber roll cage exhibits very similar stress and deformation results compared to those of AISI 4130 steel. The minimum factor of safety in carbon fiber roll cage is 3.76, compared to 2.06 in AISI 4130 steel roll cage. The carbon fiber roll cage has a total weight of only 15.1 kg, which is 74.62% lower than the mass of AISI 4130 steel roll cage.

Rollover accidents are among the most dangerous type of vehicular crashes. They account for the highest fatality rate with more than 10,000 people killed every year in a rollover accident. The Carbon Fiber roll cage displays a very high factor of safety of 5.0 in the rollover test, while the AISI 4130 steel has a good 2.69 safety factor. The Carbon Fiber roll cage is one of the alternate materials which can be recommended to ensure the maximum safety of the driver in impact scenarios.

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NOMENCLATURE

DLC	SIM lateral cross member
FBM	Front Bracing member
ALC	Aft lateral cross member
CLC	Upper lateral cross member
FLC	Front lateral cross member
BLC	Overhead lateral cross member
RHO	Roll hoop overhead members
LFS	Lower frame side members
RRH	Real roll hoop