Design and Analysis of Roll Cage for an Electric Hybrid Tricycle

Siddharth Aphale[#1], Pradnesh Lachake[#2]

[#]Department of Mechanical Engineering
K.K. Wagh College of Engineering Education & Research, Amrutdham, Nashik 422 003
Savitribai Phule University, Pune, Maharashtra, India

Abstract**—** *With the growing concern of steep dip in the fuel reserves worldwide and alarming rates of pollution, green technological trends are being progressively adopted to promote eco-friendly spirits. Efficycle is an electrically assisted, dual human pedal powered three wheeled vehicle, designed to facilitate daily mobility needs.*

The vehicle frame was to be ergonomically designed, engineered for performance and safety. Enhancing the structural integrity and overall aesthetics were the focal points in the design and analysis of the roll cage. This paper deals with the roll cage material selection process and finite element static structural analysis of the roll cage under predetermined conditions in FEA software Ansys 15.0 to determine its structural strength. The roll cage material selection was carried out with an aim to optimize strength, weight and cost. The key parameters taken under consideration were safety driver ergonomics, weight reduction and cost of manufacturing the chassis.

Keywords — *Efficycle, FEA, 3D Modelling, Roll cage, Static Analysis*

I. INTRODUCTION

The primary function of the roll cage is to ensure driver safety in case of a crash or roll over. The secondary objective in chassis design is to provide mountings for all the components, keeping in mind a low centre of gravity while doing so. Moreover the driver comfort and ergonomics should be taken into consideration while designing of the frame.

These objectives are met by proper material selection, designing a low weight reliable frame and carrying out extensive finite element analysis of roll cage against various modes of failure to verify its safety. Based on the results the roll cage is modified accordingly. After finalizing the roll cage design it is fabricated.

The CAD model of the chassis was prepared in PTC Creo 3.0 and finite element analysis was performed in Ansys Workbench 15.

II. DESIGN METHODOLOGY

The design procedure of roll cage is a manifold process. It involves material selection, frame design and material cross section determination. The detailed procedure is explained below.

A. Material Selection

Roll Cage material is one of the key aspects in design which greatly affects the safety, reliability and performance. The roll cage material should possess high strength to weight ratio along with low carbon content and good weldability.

The cost is an important deciding factor in the material selection process. Thus a material selection decision matrix was formulated to aid the process.

Table 1: Material Selection decision matrix

Property	AISI 1018	HSLA 340	Chromoly 4130
Yield Strength	3	2	1
UTS	3	2	1
Carbon%	2	1	3
Weldability	1	2	3
Cost	1	2	3
Total	10	9	11

Hence, we selected HSLA 340 micro alloy steel for our chassis because of its high weldability, strength & low cost.

The selected material specifications:

Table 2: HSLA 340 properties

Property	Value
Material	HSLA 340
Carbon%	0.080%*
Density	7.87 gm/cm^3
Poisson ratio	0.3
Elastic Modulus	210 GPa
Yield Strength	571.92 MPa*
UTS	651.95 MPa*

*practically tested from an NABL accredited lab

B. Frame Design

In the initial frame design period, the transmission of vehicle, driver's ergonomics and placement, suspension and manufacturing methods were set. There was a requirement to keep a minimum clearance of 3 inches between the drivers and the roll cage members. Also keeping a low centre of gravity is also crucial to prevent toppling of vehicle. This was achieved by placing the heavy components such as motor, battery and drivers' position directly on the chassis[1].

Thus, a CAD assembly with all component mountings was created to decide the approximate dimensions of the chassis.

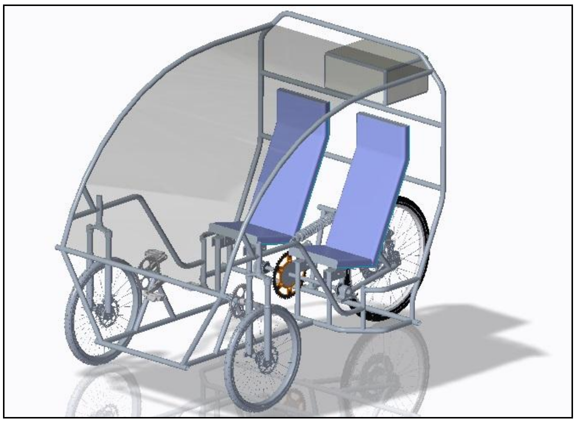

Fig 1: Isometric view of the vehicle CAD model

After the preliminary design of the vehicle, a prototype with PVC pipes was developed to check the functionality of the frame.

Fig 2: PVC pipe roll cage prototype

The design of chassis was finalised for CAE analysis after few design modifications and iterative changes for the component mountings and C.G. adjustment. Then the wireframe model of the roll cage was generated to be imported into Ansys for finite element analysis.

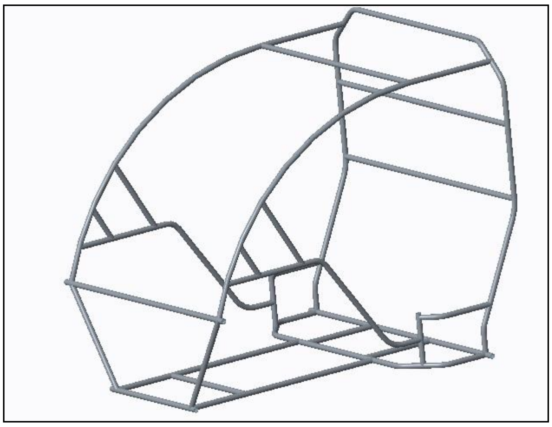

Fig 3: Isometric view of finalised chassis model

C. *Pipe Cross Section Determination*

While deciding the cross section, bending strength, bending stiffness and ease in fabrication processes are taken into consideration. As per the material requirements specified in rulebook[2], bending strength and ending stiffness of chosen cross section should be greater than or equal to that of plain carbon steel pipe of 0.18% Carbon of Outer diameter 25.4 mm and 2 mm thickness. Also there are fabrication limitations regarding welding and bending processes. Welding becomes difficult for thickness less than 1mm. After considering all these factors, cross section of Outer diameter 25.4 mm and thickness 2 mm is selected.

Calculations of Bending Strength and Bending Stiffness[3] for HSLA 340:

Outer Diameter = 25.4 mm
Inner Diameter = 21.4 mm
Yield Strength (S_y) = 571.92 N/mm^2
Distance from center axis to outer fiber (C)
= (25.4/2)
= 12.7 mm

Polar Sectional Modulus I,

$I = \pi \left(D^4_{outer} - D^4_{inner} \right)/32$
I = 10136 mm^4

Bending Strength M,

$M = (S_y \times I)/C$
$M_{HSLA\ 340} = 271.377 \times 10^3$ N-mm

Bending Stiffness σ,

$\sigma = E \times I$
$\sigma_{HSLA\ 340} = 2.128 \times 10^6$ kN-mm^2

Similarly the bending strength and bending stiffness values of 0.18% Carbon steel were calculated and a table was formed to compare values.

Bending Strength, $M_{0.18\%\ C\ steel} = 291.31 \times 10^3$ Nmm
Bending Stiffness, $\sigma_{0.18\%\ C\ steel} = 2077.8 \times 10^6$ kN-mm^2

Table 3: Material Property comparison chart

Property	0.18%C Steel	HSLA 340
Bending Strength	291.31 N-m	456.45 N-m
Bending Stiffness	2077.8 N-m^2	2128.5 N-m^2

Thus, as the bending strength and bending stiffness of HSLA 340 exceed that of the 0.18% plain carbon steel of same cross section, we finalized HSLA steel pipe of Outer diameter of 25.4 mm and thickness 2mm for the chassis.

III. FINITE ELEMENT ANALYSIS

After the completion of CAD model of chassis along with the material selection it is necessary to test the impact and rollover safety of the vehicle. The frame should be able to withstand impact, torsion and rollover conditions to provide driver safety. Crash pulse scenario standard set by industries is 0.15 to 0.2sec. We assumed it to be 0.2 seconds in our analysis[4].

Assumptions and Considerations:

1. The Roll cage material is isotropic.
2. All the roll cage members have uniform cross section.
3. Roll cage is stationary, i.e. we are considering the situation when the roll cage is stationary and someone impacts from front/side.
4. Time of impact is assumed to be 0.2 seconds
5. Speed of the object impacting on the roll cage is considered in terms of G-force.
6. Force impact location considered at the first roll cage members in contact with the collision.

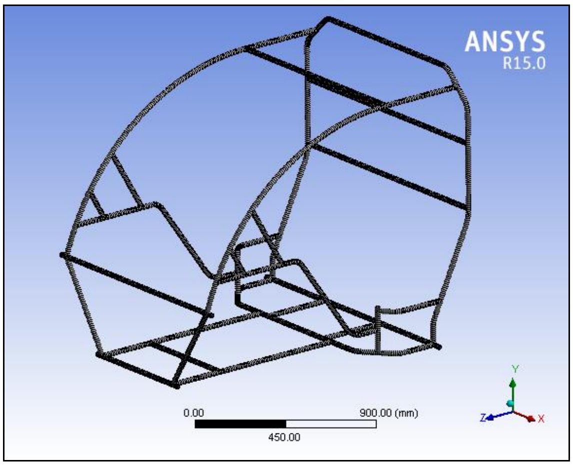

Fig 4: 1D Meshed model of roll cage

The chassis CAD model was imported to Ansys and meshed using 1D mesh with element size 10mm forming a total of 5165 nodes and 2562 elements.

D. Front Impact Analysis

This test is performed to analyze the deformation of the roll cage under the condition of collision from the front.

Calculation of Impact forces:

Assuming 5G force for a vehicle/driver mass of 240 kg,

$F = 5 \times m \times g$

$F = 5 \times 230 \times 10$

$F = 11500$ N

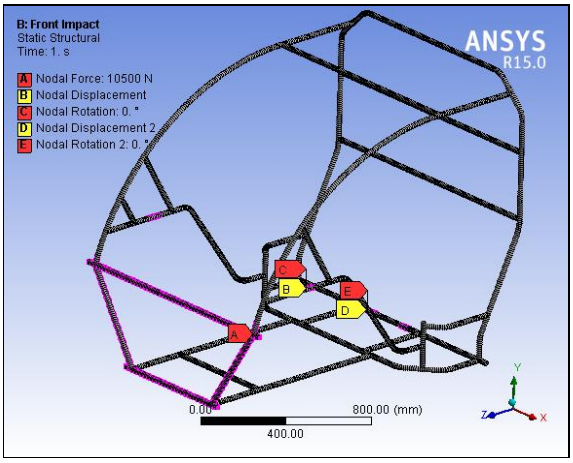

Fig 5: Front Impact Analysis Constrained model

We applied the calculated 5G force of 10500N to the front impact members of chassis while applying the Boundary conditions to the chassis. We constrained the motion of front suspension in the z axis direction. The motion of rear suspension was constrained in all directions. The rotation of all suspension mounting points along all axes is locked.

Analysis Result:

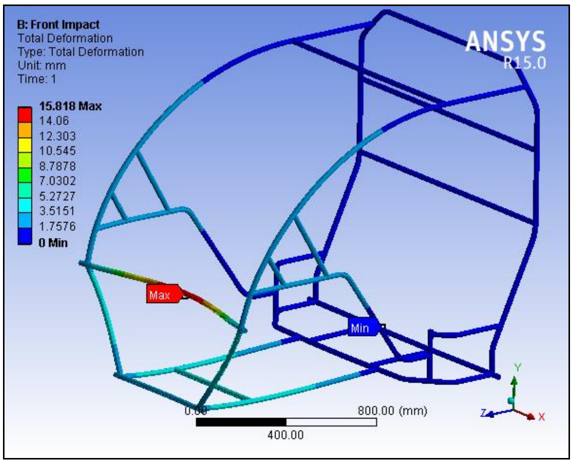

Fig 6: Front Impact Analysis Total Deformation

Maximum deformation= 15.818 mm

According to analysis, deformation at the time of collision does not affect the driver safety.

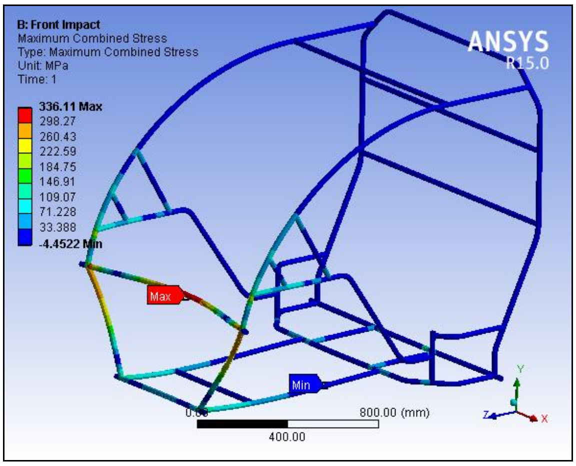

Fig 7: Front Impact Analysis Maximum Combined Stress

Maximum Combined Stress= 336.11 MPa

Incorporated Factor of Safety

$= \sigma_{yt}/$ Maximum Combined Stress

$=571.92/336.11$

$=1.701$

As the FOS is greater than 1.2, the design is safe against specified stress for front impact.

E. Side Impact Analysis

This test is performed to see the behaviour of the roll cage in the condition of collision from side and thus check drivers' safety in the condition of a side impact.

Calculation of Impact forces:

Assuming 3G force for a vehicle/driver mass of 240 kg,

$F = 3\times m\times g$

$F = 3\times 230\times 10$

$F = 6900$ N

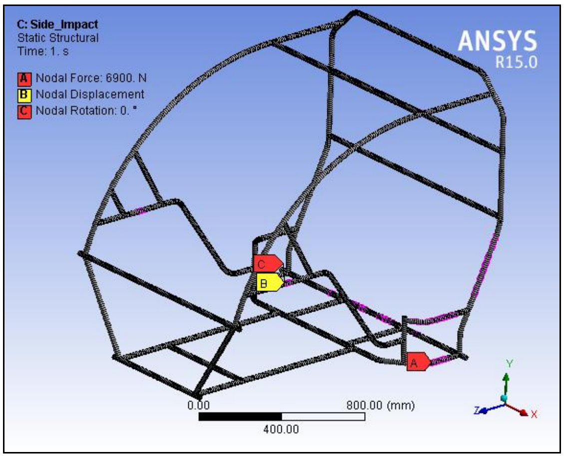

Fig 8: Side Impact Analysis Constrained model

We applied the calculated 3G force of 6900N to the side impact protection members of the chassis while applying the Boundary conditions. The translation and rotation of all suspension mounts is locked.

Analysis Result:

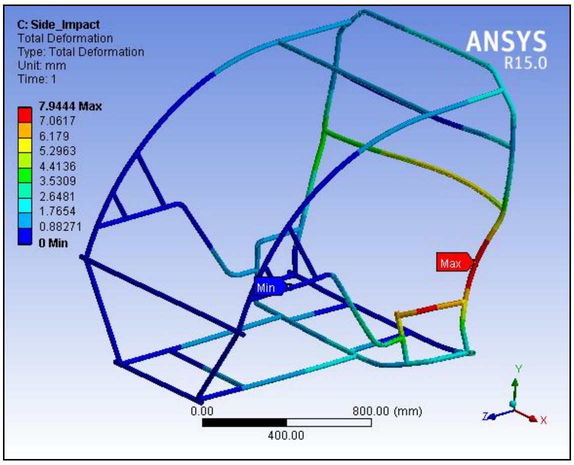

Fig 9: Side Impact Analysis Total Deformation

Maximum deformation= 7.94 mm

According to analysis, deformation at the time of collision does not affect the driver safety.

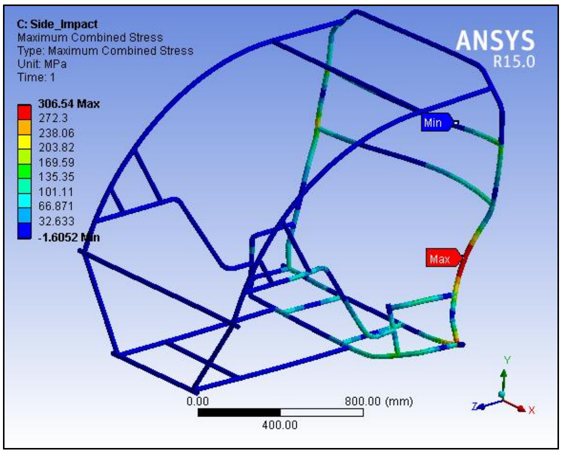

Fig 10: Side Impact Analysis Maximum Combined Stress

Maximum Combined Stress= 306.54 MPa

Incorporated Factor of Safety

$= \sigma_{yt}/$ Maximum Combined Stress

$=571.92/306.54$

$=1.86$

As the FOS is greater than 1.2, the design is safe against specified stress.

F. Roll Over Analysis

This analysis is performed to analyse the behaviour of roll cage in the condition of vehicle toppling.

Calculation of Impact forces:

Assuming 3G force for a vehicle/driver mass of 240 kg,

$F = 3\times m\times g$

$F = 3\times 230\times 10$

$F = 6900$ N

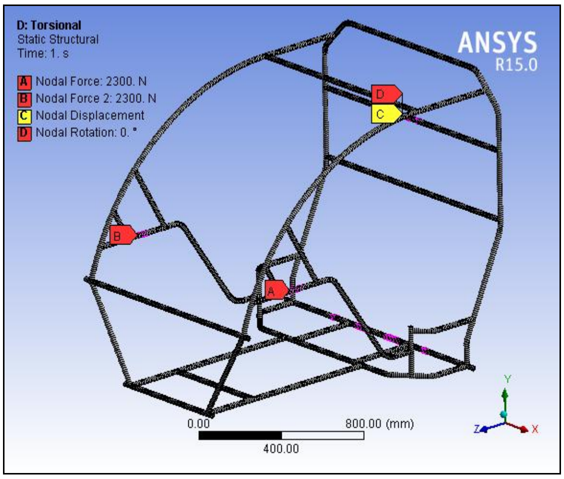

Fig 11: Roll Over Analysis Constrained model

We applied the calculated 3G force of 6900N to the members of chassis which would be first in contact with the surface of road in case of rollover. The force was applied perpendicular to the curved members the chassis. The translation and rotation of all suspension mounts is locked.

Analysis Result:

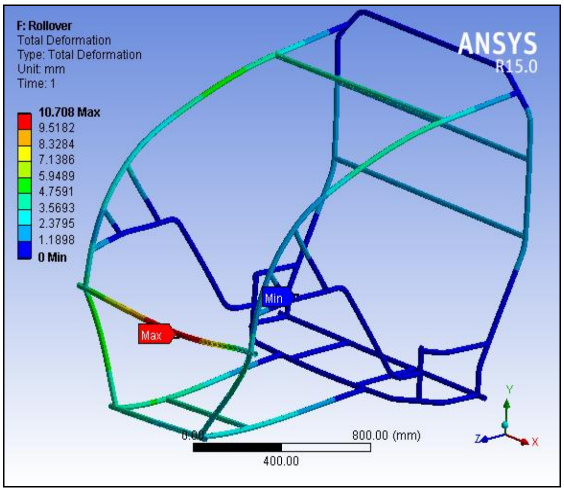

Fig 12: Roll Over Analysis Total Deformation

Maximum deformation= 10.708 mm

According to analysis, deformation at the time of collision does not affect the driver safety.

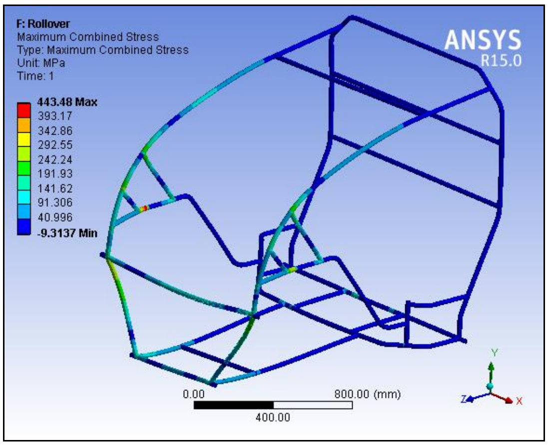

Fig 13: Roll Over Analysis Maximum Combined Stress

Maximum Combined Stress= 293.98 MPa

Incorporated Factor of Safety

$= \sigma_{yt}/$ Maximum Combined Stress

$=571.92/443.48$

$=1.28$

As the FOS is greater than 1.2, the design is safe against specified stress.

G. Torsional Analysis

This test is performed to examine the structure under twisting loads. This occurs when one of the front wheel pass over a road hump.

Calculation of Impact forces:

Assuming 2G force for a total vehicle/driver mass of 240 kg,

$F = 2\times m\times g$

$F = 2\times 230\times 10$

$F = 4600$ N

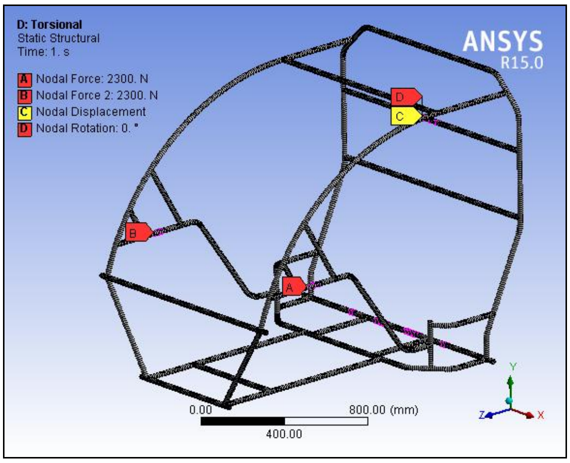

Fig 14: Torsional Analysis Constrained model

We applied the calculated 2G force of 4600N to the front suspension mounting points of the chassis (2300N separately to each suspension mount). We constrained all degrees of freedom of the rear suspension.

Analysis result:

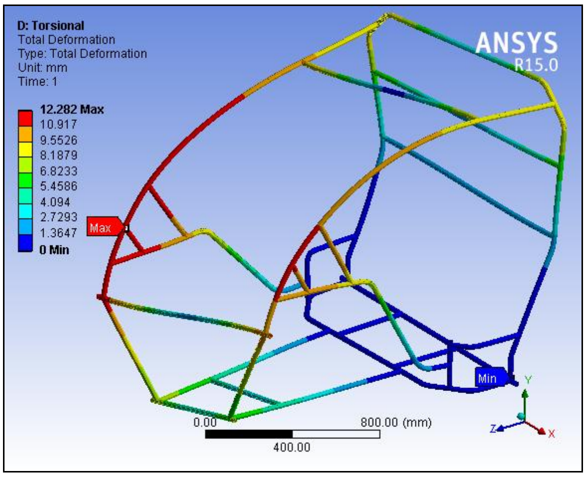

Fig 15: Torsional Analysis Total Deformation

Maximum deformation= 12.28 mm

According to analysis, deformation at the time of collision does not affect the driver safety.

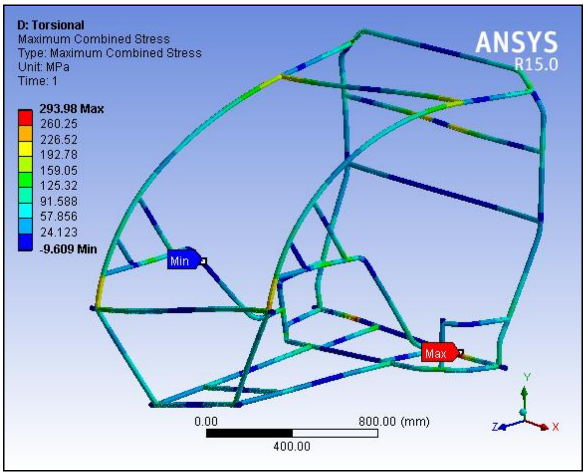

Fig 16: Torsional Analysis Maximum Combined Stress

Maximum Combined Stress= 293.98 MPa

Incorporated Factor of Safety

$= \sigma_{yt}$/ Maximum Combined Stress

=571.92/293.98

=1.94

As the FOS is greater than 1.2, the design is safe against specified stress.

IV. Conclusion

We have successfully analysed the roll cage for its strength for its safety against collision from front and side and under rollover condition. The roll cage was deemed safe for front impact load of 5G, side impact load of 3G, rollover loading of 3G and torsion of magnitude 2G. The deformation & stresses are under limit. Hence this roll cage was finalised for manufacturing and fabrication of the vehicle.

References

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[4] Linder, Astrid; Avery, Matthew. “Change of Velocity and Pulse Characteristics in Rear Impacts: Real World and Vehicle Tests Data,” The Motor Insurance Repair and Research Centre. Thatcham, United Kingdom.'