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Weight Optimization and Load Path visualization using FEA for ATV Roll Cage

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ABSTRACT
The objective of this paper is to demonstrate the design procedure and weight optimization techniques for a Roll cage. The front and Rear of the frame are designed around the Roll hoop which is base of the frame. The rear frame provides mountings for the powertrain components while the front part encloses driver & driving controls. The roll hoop must be strong as front & rear part welded to it and it also ensures driver safety.The roll cage is designed keeping in mind front impact, rear impact, rollover, side impact, front wheel landing & rear wheel landing to check whether the frame can endure to all kind of possible scenarios.

Frame is that component of the vehicle to which everything is attached. So, the frame has to be analysed for various situations in order to predict whether the frame will survive or fail. The frame is analysed for worst case scenarios which the vehicle may face. The frame must be sufficiently light in weight, without any compromise with driver safety.

Keywords
Space Frame, Roll Cage, ATV, BAJA, Load Paths SOLIDWORKS, ANSYS, Finite Element Method.

1. INTRODUCTION
The BAJA SAE is an intercollegiate engineering design competition for undergraduate engineering students to design & fabricate a reliable, durable, ergonomic and economical All-Terrain Vehicle. The event originated in the name of Mini - BAJA, in the year 1976 at University of Carolina. Since then, the event has spanned across six countries – USA, Mexico, South Africa, Korea, Brazil and India. Each year this 3-4 days long event tests the vehicle in all types of off-road conditions including rocks, wooden logs, concrete slabs, mud holes, steep inclines, jumps and sharp corners. There is a high risk of frame failure due to collisions with stationary objects, or front & rear impacts from other vehicles and rollovers.

The 3-4 days long events comprise of Acceleration test, Braking test, Suspension & Traction test, Manoeuvrability Test, Hill climb test and 4 hour endurance run. The main aim of the team is to sustain the rugged tracks while providing good performance in terms of speed & handling.

2. MATERIAL SELECTION
The roll cage is designed using beams of circular cross section of 1 inch outer diameter and thickness ranging from 1mm to 3mm. The material for these beams was selected as AISI 4130 Chromoly on the basis of following criteria.

Table1: Comparison of Mechanical Properties

Properties	AISI 1018	AISI 4130	Al- 6061-T6
Yield Strength	351 MPa	480	270
Tensile Strength	450 MPa	560	310
Mass Density	7900	7850	2700
Poisson Ratio	0.285	0.29	0.33

3. DESIGN METHODOLOGY
3.1 Defining Cross-Section
The rulebook for BAJASAE series 2015-16 categorizes the beams in the rollcage as primary and secondary members with defined minimum thickness as 3mm and 0.9mm respectively. For further weight optimization and structural rigidity, the secondary beams have been further divided into two different cross- sections that are 1mm and 1.5mm. The beams colored red are primary members and beams with color grey and blue have pipe thickness as 1.5mm and 1mm respectively.

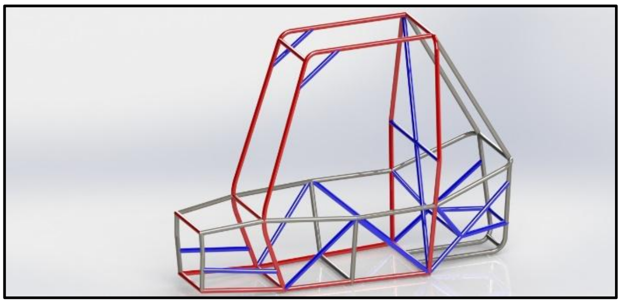

Fig 1: Pipes of different cross sections used in the frame.

3.2 Ergonomics
After the basic dimensions of the beams and roll cage structure have been finalized, the design if the roll cage was modelled in wireframe using CATIA V5. The ergonomics of the chassis were carefully analyzed using RULA analysis by placing a dummy model inside the frame to check for clearances and intrusions if any. The angle between arms and

shoulders and between back and torso were examined to ensure comfortable driving position.

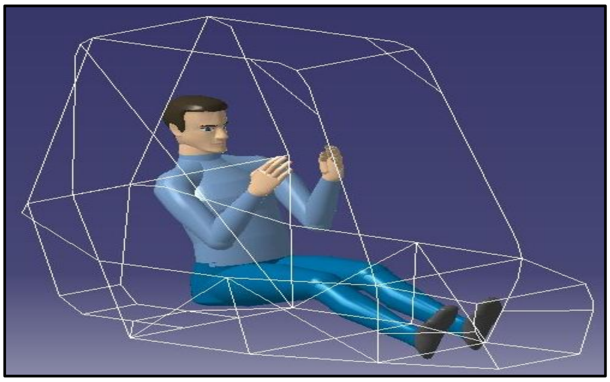

Fig 2: RULA analysis for ergonomics in CATIA V5.

3.3 Weight Optimization of the Roll-Cage

1) The cylindrical beams used as the structural beams for the roll cage have very high strength and stiffness against the axial loads as compared to bending loads acting perpendicular to the axis of the pipe.Members which were only under axial tension and compression were significantly reduced in their cross section because of their high stiffness and strength against axial forces.
2) This property was extensively used to determine and control load paths within the frame using these cylindrical beams of less thickness (1mm) as carriers. As a result of this, number of beams were significantly reduced in the roll cage leading to overall weight reduction and greater strength.
3) Less number of welded joints were employed within the roll cage and most of the members were made out of single tubes using bends and curves. This lead to overall increase in the structural rigidity of the frame and reduced the loss of strength due to welded joints extensively.
4) Instead of designing a roll cage to absorb the impact forces, beam positions were carefully optimized to allow the flow of forces within the frame structure between the suspension points which act as the reaction points for the car during operation.

3.4 Load Path Visualization

The load paths in various loading cases are shown below. Beams in yellow are the immediate load bearers and beams in red and green are load carriers.

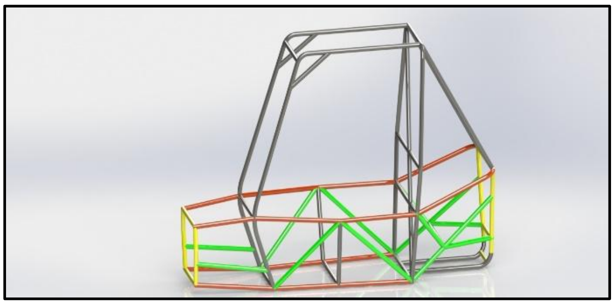

Fig 3: Load paths in case of Front and Rear Impact.

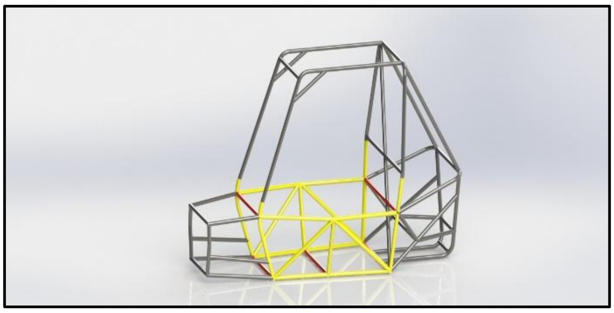

Fig 4: Load paths in case of Side Impact.

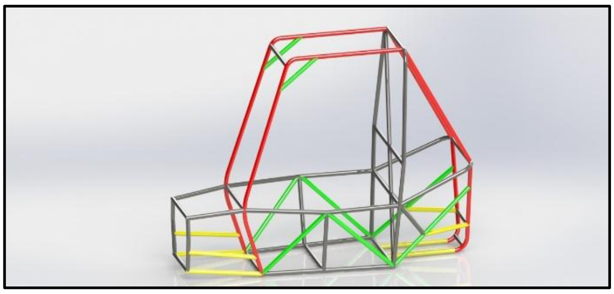

Fig 5: Load paths in case of Front and Rear Landing.

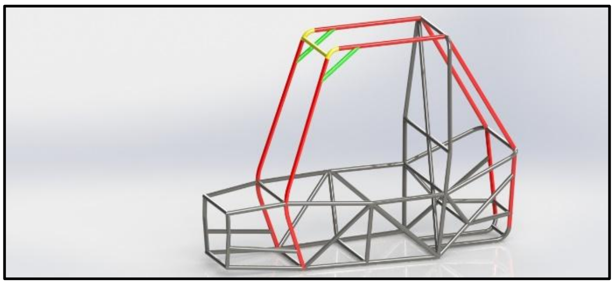

Fig 6: Load paths in case of Roll over Impact.

4. FINITE ELEMENT ANALYSIS

The roll cage designed was first modelled in CATIA V5 as wireframe which acts as the input file for 1D meshing in ANSYS Workbench. Pipe elements with suitable cross section as input were used to mesh the frame in ANSYS and carry out the FEA simulating real time off-road scenarios. The FEA results with suitable Factor of Safety allowed effective weight optimizations in the frame and suggested alterations where necessary.

4.1 Front Impact

The vehicle with total mass of 250 kg including driver goes through a head on collision with another vehicle of identical mass or a wall causes one of the most severe impact loads on the frame. Assuming the vehicle velocity at impact to be 40kmph, the impact force is calculated around 5g for front impact.

The deformation and stresses induced are shown below. Maximum combined stress induced is 168.66MPa which provides FOS of 2.07. Maximum deformation obtained was 2.48mm.

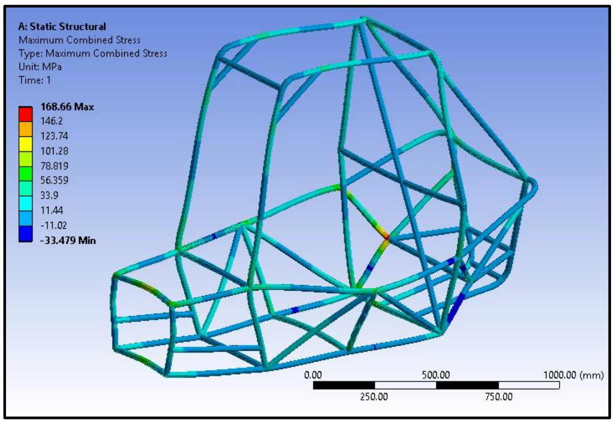

Fig 7: Max. Stress in front impact using ANSYS WB.

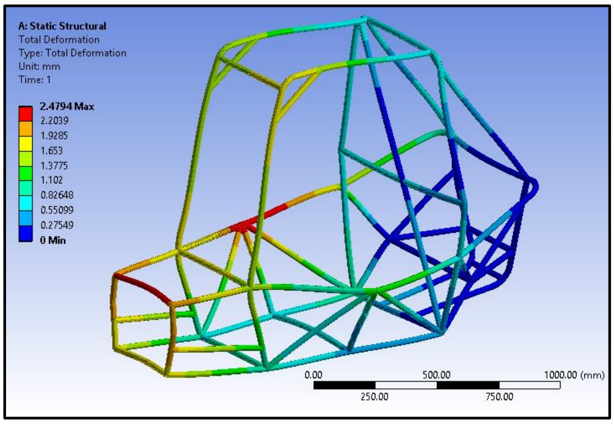

Fig 8: Max. Deformation in front impact using ANSYS WB.

4.2 Rear Impact

Similarly, vehicle approaching from rear can also hit the vehicle at similar speeds. Rear members of the frame transmit the impact load through the main roll hoop towards the front hard points thus providing a smooth flow for the impact load. Load of equivalence of 5g was applied onto the roll cage rear protruding members keeping the front hardpoints as fixed supports.

The deformation and stresses induced are shown below. Maximum combined stress induced is 179.61MPa which provides FOS of 1.94 Maximum deformation obtained was 6.45mm.

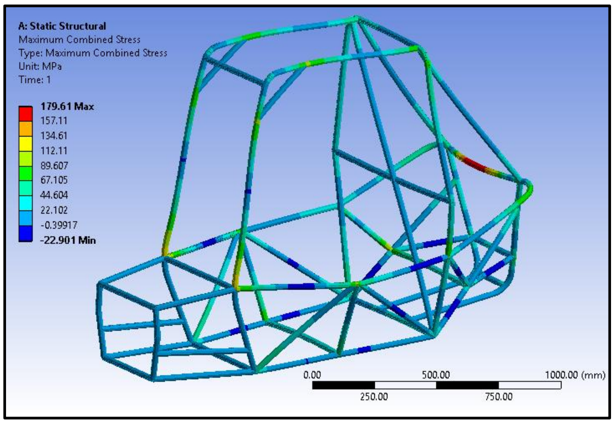

Fig 9: Max. Stress in rear impact using ANSYS WB.

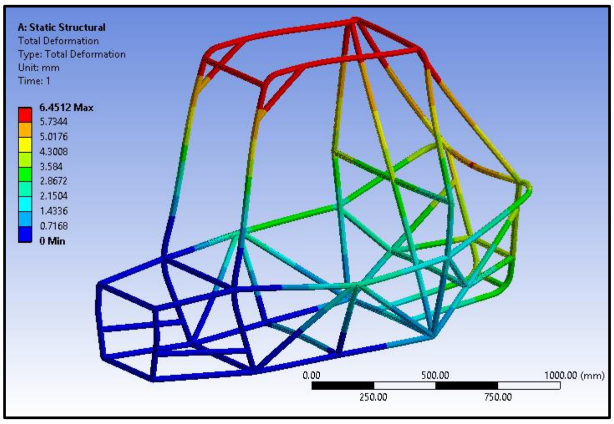

Fig 10: Max. Deformation in rear impact using ANSYS WB.

4.3 Side Impact

Side impact test for the roll cage simulates a vehicle hitting the frame from either of the two sides. Impact load of 4-5g was applied to the side impact members and the lower frame side members and constraining the hard points of the opposite side as fixed supports.

The deformation and stresses induced are shown below. Maximum combined stress induced is 167.8MPa which provides FOS of 2.17. Maximum deformation obtained was 5.84mm.

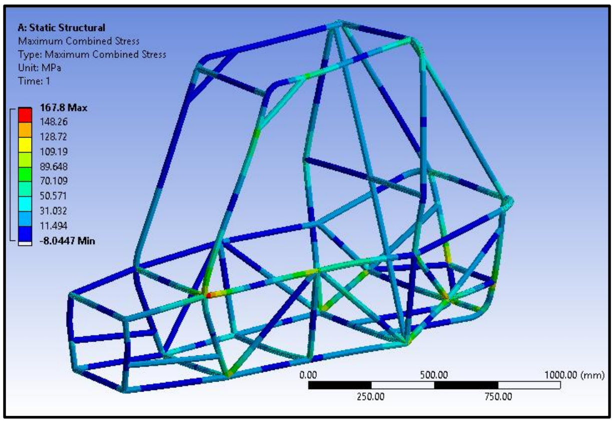

Fig 11: Max. Stress in side impact using ANSYS WB.

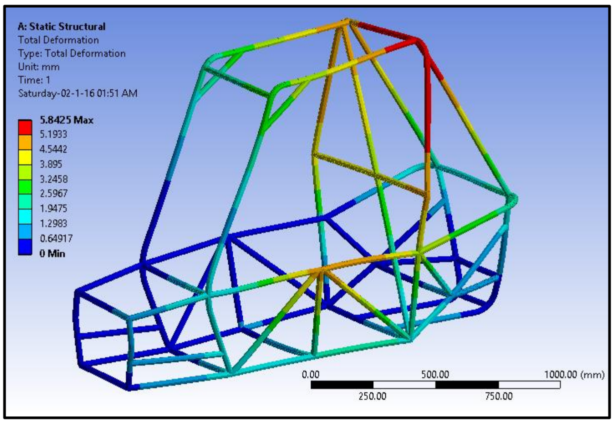

Fig 12: Max. Deformation in side impact using ANSYS WB.

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4.4 Roll over Analysis

During the off-road racing, the vehicle tends to roll over towards the front or rearwards either due to sudden depressions or bumps. If the vehicle hits a rock or log of wood at high speed, the momentum of vehicle cause it to roll over attaining a projectile motion. Assuming maximum height during such motion as 6ft i.e. 1.83m the impact force is calculated equivalent to 3g. This force was applied to the FBM and RHO joint which is intentionally kept as curved fillet of radius 3inches to allow greater strength and continuous load transfer.

The deformation and stresses induced are shown below. Maximum combined stress induced is 197.41MPa which provides FOS of 1.77. Maximum deformation obtained was 8 mm.

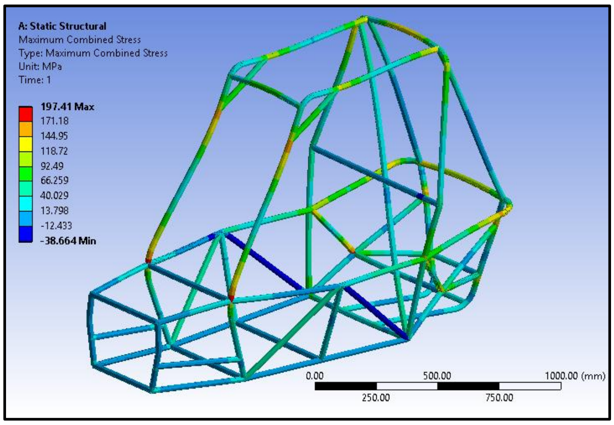

Fig 13: Max. Stress in roll over using ANSYS WB.

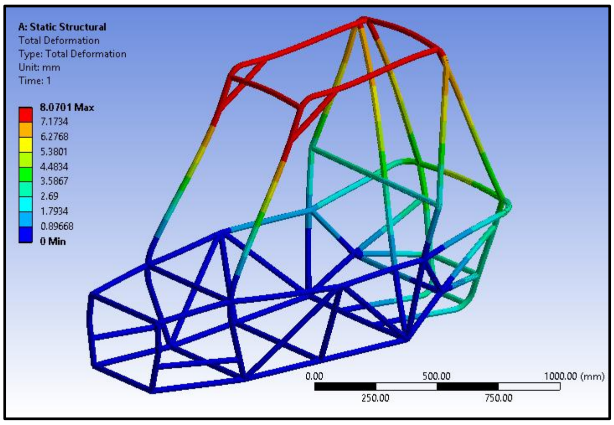

Fig 14: Max. Deformation in roll over using ANSYS WB.

4.5 Front Landing

The BAJA vehicle often undergoes jumps of high altitude with maximum height attaining upto 6ft i.e. 1.83m. With this height the vehicle hitting the ground with only front two wheels landing undergoes the impact force equivalent to 3g acting only on the front part of the roll cage where the front wishbones are to be attached and mainly concentrated around the upper mounting point for the shock absorber.

The deformation and stresses induced are shown below. Maximum combined stress induced is 192.25MPa which provides FOS of 1.82. Maximum deformation obtained was 7.13mm.

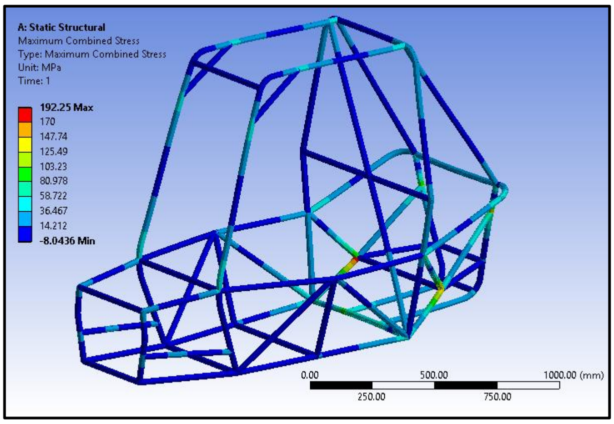

Fig 15: Max. Stress in front landing using ANSYS WB.

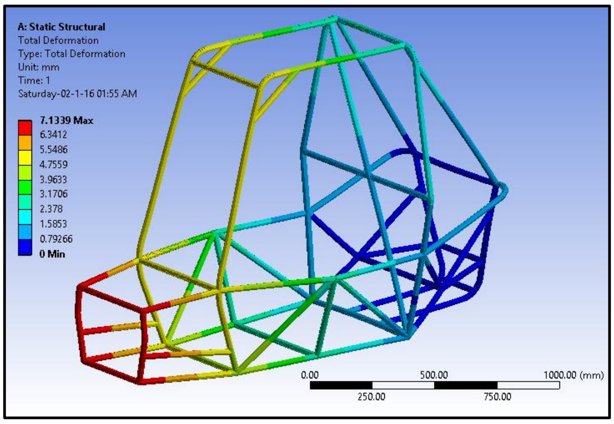

Fig 16: Max. Deformation in front landing using ANSYS WB.

4.6 Rear Landing

Similarly, vehicle hitting the road with only 2 rear wheels undergoes same amount of 3g force acting upon the rear frame and upper mounting point for the shock absorber. The FAB members behind the main roll hoop are subjected to impact load of 3g in the upward direction keeping the front hard points fixed.

The deformation and stresses induced are shown below. Maximum combined stress induced is 175.77MPa which provides FOS of 1.99 Maximum deformation obtained was 2.82mm.

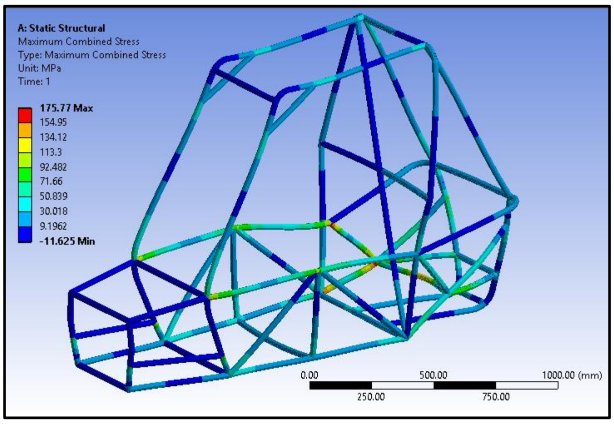

Fig 17: Max. Stress in rear landing using ANSYS WB.

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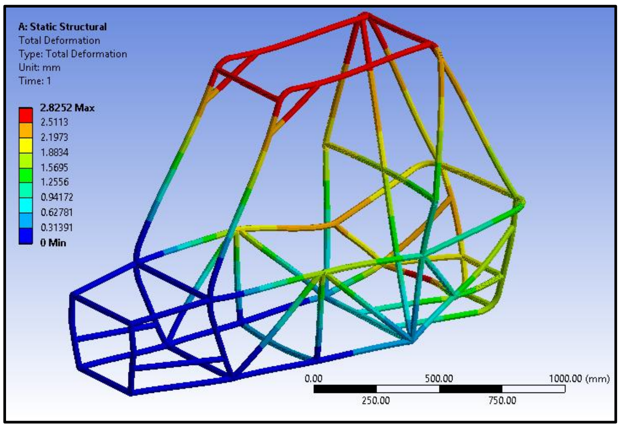

Fig 18: Max. Deformation in rear landing using ANSYS WB.

5. CONCLUSION

Table 2: Results

Case	**Maximum Stress (MPa)**	**Maximum Deformation (mm)**	**FOS**
Front Impact	168.66	2.47	2.07
Rear Impact	179.61	6.45	1.94
Side Impact	167.8	5.84	2.17
Roll Over	197.41	8.07	1.77
Front Landing	192.25	7.13	1.82
Rear Landing	175.77	2.82	1.99

Since the stresses induced in each of the tests performed is less than the yield strength of the material it can be considered that the designed frame is safe for the driver.

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