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Design and Development for Roll Cage of All-Terrain Vehicle

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Abstract - **BAJA is a collegiate competition sponsored by the Society of Automotive Engineers, India (SAEINDIA®). Its aim is to build an All-Terrain Vehicle (ATV) as per the constraints given by the organizers. The growing popularity of the competition coupled with the need to design safe and dynamically balanced ATV has led to the origin of the idea of this paper. The process of Finite Element Analysis (FEA) is expensive and time consuming as well as simulating the problem statement is unnecessarily tedious at an initial design stage. Therefore, it is always advantageous to do extensive research on the basic requirements of roll cage design. To understand the changes that need to be incorporated in the design, perform the static stress analysis first. This will offer a simplistic simulation criterion of the problem statement and requires a lower computational time; and then dynamic analysis to validate the safety of the preliminary design. The paper aims to give an introduction to the material selection procedure, pipe size selection and various tests that need to be done before finalizing the design, using ANSYS® WORKBENCH 14.0. In this present work, various factors such as impact force determination, loading points, the mesh size dependence of generated stress, Von-Misses Stress, Deformation and Factor of Safety (FOS) are studied.**

Keywords - Roll Cage, Analysis, All-terrain vehicle, Factor of Safety, Von-Misses Stress, Finite-Element Analysis

1. INTRODUCTION

The Society of Automotive Engineers (SAE) organizes design competitions to indulge students in the general practices of mobility Engineering. One such student design activity is the SAEINDIA® BAJA event held each year in Pithampur (India). The objective for the team of students is to develop a dynamically balanced vehicle to withstand all kind of terrain during its mobility. The SAE BAJA vehicle development manual restricts the vehicle weight, shape and size, and dimensions. The objective of SAE BAJA competition is to simulate real world engineering design projects and their related challenges. Moreover, it develops the best performing vehicle with the rugged and economical vehicle frame that satisfies all the SAE BAJA design requirements [Rulebook BAJA SAEINDIA® 2014]. During actual road performance, any vehicle is subjected to loads that cause stresses, vibrations and noise in the different components of its structure. This requires appropriate strength, stiffness and fatigue properties of the components to be able to stand these loads. On top of that, quality of a vehicle, as a system, which includes efficient energy consumption, safety, riding dampness and provision of comfort to the driver is highly desired. Roll Cage is the structural basis of an All-terrain vehicle. The components of a vehicle like a Power Source, Transmission system, Axles, Wheels and Tyres, Suspension System, Controlling systems like Braking , Steering etc. , and also electrical system parts are mounted on the chassis frame. It is the main mounting for all the components including the body. So it is also called as a carrying unit [Nagarjuna et al.,2013] Extensive Research has been conducted in order to design each main component of the vehicle. It is considered that each component is significant, and thereby designed the vehicle as a whole trying to optimize each component while constantly considering how other components would be affected. This forced to think outside the box, research more thoroughly, and redesign components along the way in order to have a successful design. The necessary parameters were used to create a Qualitative Function Diagram (QFD) to determine which parameters were the most critical. These key parameters ranging from most critical to least critical are safety, reliability, low cost, ease of operation and maintenance, and overall performance. Considering the functional objectives and the rules laid by the SAE Baja, a preliminary design of the Roll Cage structure was developed in a 3D environment using SOLIDWORKS® 2013. As weight is critical in a vehicle powered by a small engine, a balance must be found between the strength and weight of the design. To best optimize this balance the use of solid modeling and finite element analysis (FEA) software is extremely useful in addition to conventional analysis. The Centre of Gravity was tried to keep in middle of the vehicle & closest to the ground for optimum stability. The length of the vehicle was kept small so as to reduce weight and maintain a desired center of gravity. Later the design is tested against all modes of failure by conducting various simulations and stress analysis with the aid of ANSYS® 14.0 Software. Finite Element Analysis (FEA) is carried out on 3D model of roll cage in cases of front impact, rear impact, side impact, Roll-over, Torsional Rigidity, front bump & Rear bump, Modal analysis and Drop Test in ANSYS®. Based on the result obtained from the testing the design is modified accordingly. The analysis showcases the distribution of Von Misses stresses and the deformation of the frame members, when subjected to the applied loads. If the stress generated in the chassis members was found to be above the yield limit of the material, the existing frame was modified for a safe design. The new design was again subjected to the same analysis, and the iterations continued till the stress and deformation was within the desired limit. After successfully designing the roll cage, it is ready to be fabricated.

This paper has been divided in various sections viz; Section 2 provides Design and Development for Roll Cage of ATV, Section 3 presents the details of FEA, Section 4 gives the descriptions of Results and finally conclusions from present work are drawn in section 5.

2. DESIGN AND DEVELOPMENT

The design and development process of the roll cage involves various factors; namely material selection, pipe size selection, frames design and finite element analysis. The details of each step are given below.

2.1 Material Selection

One of the key design decisions that greatly increase the safety, reliability and performance in any automobile design is material selection. As per the rule book constraint there should be at least 0.18% of carbon content in metal [Rulebook BAJA SAEINDIA® 2014]. Our initial step was to conduct a market survey to have an idea of the availability of the material. Based on market survey we have chosen following material namely: - AISI 1018 steel, 1020 DOM, and AISI 4130 steel. Table below gives a side by side comparison of these materials [Nayak et al., 2012].

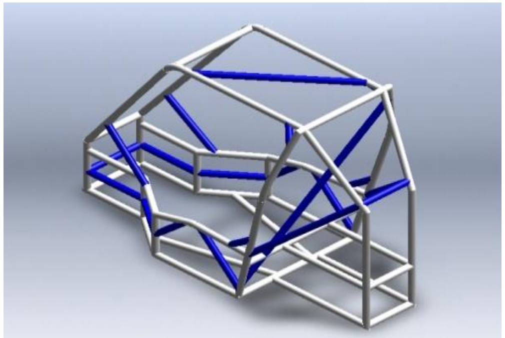
Table 1: Comparison of Material Properties

Property	Material	1018 Steel	1020 DOM	4130 Steel
Physical Properties	Density (kg/m^3)	7870	7865	7850
	Yield Strength (MPa)	370	480	460
	Tensile Strength (MPa)	440	550	731
Chemical Properties	Carbon, C	0.14-0.20	0.05-0.26	0.28-0.33
	Iron, Fe	98.81-99.26	99.08-99.53	97.3-98.2
	Manganese, Mn	0.60-0.90	0.3-0.6	0.4-0.6
	Sulphur, S	$\leq$0.50	$\leq$0.05	$\leq$0.04
	Phosphorus, P	$\leq$0.40	$\leq$0.04	$\leq$0.035
	Chromium, Cr	-	-	0.80-0.95

To ensure that the optimal material is chosen, extensive research was carried out. The key Categories for comparison were strength, weight, and cost. The 4130 is selected due to its optimum strength-to-weight ratio. Further, the properties such as high corrosion resistance and the availability of the metal enabled the choice of AISI 4130 steel in the making of the roll cage [Kumar et al.,2013]. The roll cage is virtually designed as a tubular space frame.

2.2 Pipe Size Selection

As per guidelines, the primary members must be Circular steel tubing with an outside diameter of 25.4 mm (1 inch) and a wall thickness of 3 mm (0.120 in) [Rulebook BAJA SAEINDIA® 2014]. Also, we were having a constraint in selection of size for secondary members of the frame that the minimum wall thickness should be 0.89 mm (0.035 in) and min outside diameter should be 25.4 mm (25.4 in). The bending strength and ease of fabrication is also taken into consideration while deciding the cross section of pipe [Sharma et al. 2012]. To keep the weight of roll cage as low as possible, we used the circular tubes with following dimensions:

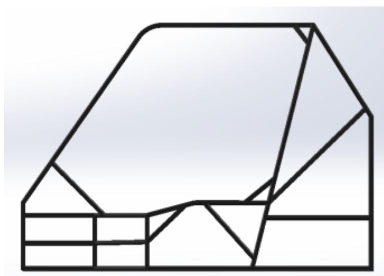
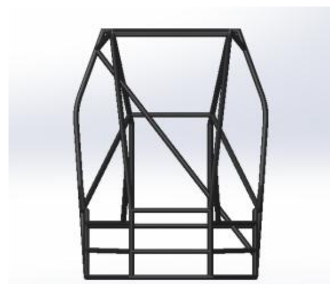


Primary Members: 25.4 x 3 mm

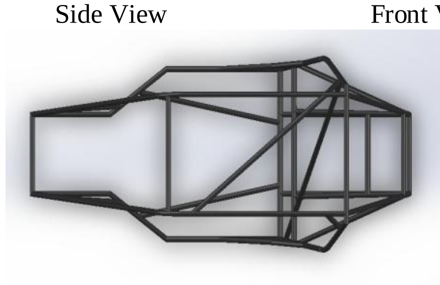
Secondary Members: 25.4 x 0.89 mm

Figure 1: Isometric View of frame

2.3 Roll Cage 3D Views:

Side View Front View



Top View

Figure 2: Orthogonal Views of Frame

3. FINITE ELEMENT ANALYSIS

3.1 Element Type:

The meshing has always been the key of the finite element model and for the exact solution of any object; it should be properly meshed with relevant element shape and size. Generally for the solid bodies the **tetra mesh** is preferred, so as we did [Matthias Goelke, 2014].

3.2 Assumptions:

a) The chassis material is considered to be isotropic and homogenous.
b) Chassis tube joints are considered to be perfect joints.
c) The 'Crumple zone' phenomenon is not considered.

3.3 Mesh Size:

Meshing of roll-cage was carried out in ANSYS® WORKBENCH 14.0. In finite element analysis, the degrees of freedom is reduced from infinite to finite with the help of discretization or meshing (nodes and elements). Each small volume is called an element. Each element has a set of points called nodal points or nodes [Matthias Goelke, 2014]. Nodes are usually located at the endpoints of elements.

The analysis was carried out using progressively reducing elemental sizes. The elemental size having consecutive stress error less than 5% is generally considered as the optimum size of mesh. It means that any further decrease in size will only negligibly increase the accuracy of the results. The stress values for different mesh sizes are as tabulated in Table 2.

Table 2: Comparison of Mesh Sizes

Parameter	Case I	Case II	Case III	Case IV
Size of Mesh (Units)	40	20	10	8
No. of nodes	106684	219584	434629	485534
No. of Elements	74562	133446	236481	253079
Max value of Von-Misses Stress (MPa)	98.230	205.62	265.89	270.6
%age Error	-	52.27	29.31	1.8

3.4 Calculations:

i) Impact Force Determination by Speed Limit:

According to the constraints in the rulebook, the maximum speed of the car is assumed to be 60km/hr. or roughly around 16.66 m/s.

For a perfectly inelastic collision, the impact force is as calculated from Eqn. (1).

$$W_{net} = \left(\frac{m\times v^2}{2}\right)_{final} - \left(\frac{m\times v^2}{2}\right)_{initial} \qquad \dots (1)$$

Where, W_{net} is net work done on account of an inelastic collision.

$$W_{net} = -\left(\frac{m\times v^2}{2}\right)_{initial} \qquad \dots (2)$$

But W_{net} = Impact force x d … (3)

Where d is the distance travelled during impact.

It is considered that for static analysis, the vehicle comes to rest in 0.1 seconds after impact. Therefore, for a vehicle which moves at 16.66 m/s (or 60 km/hr), the travel of the vehicle after impact is 1.66 m. From Eqn. (1), (2) and (3), we get:

$$\text{Impact force} = \frac{W_{net}}{d}$$

$$\text{Impact Force} = \left(\frac{m\times v^2}{2}\right)_{initial} \times \frac{1}{d}$$

$$\text{Impact Force} = \left(\frac{310\times 16.66^2}{2}\right)_{initial} \times \frac{1}{1.66}$$

Impact Force (F1) = 25916 N

Therefore, Impact Force by Speed Limit (F1) ≈ 26,000 N

ii) Impact Force Determination by Acceleration Limit:

The ‘Motor Insurance Repair Centre’ has analyzed that the Baja car will see a maximum of 7.9 G’s of force during impact.

F = m x a … (5)

Where, m = 310 kg, a = 7.9 x 9.8 m/s^2

And Force = 24000.2 N

Therefore, Impact Force by Acceleration Limit (F2) ≈ 24000 N

These two values of F1 (from Eqn.4) and F2 (from Eq. No.5) are practically comparable.

iii) Impact Force Determination for Worst case Scenario:

According to research, a human body will pass out at forces much higher than 7.9 G’s [Oturkar et al., 2013]. Therefore, a value of 10 G’s was considered for an extreme worst case collision. Therefore for static frontal impact analysis, the load on the vehicle is calculated from Eqn. (6).

F = m x a … (6)

Where, m = 310 kg, a = 10 x 9.8 m/s^2

And Force = 30380 N

F3 ≈ 30000 N

Thus a force of **30000 N or 10G** is applied in the frontal impact analysis. The impact forces, as defined by automobiles industry, for other tests are given in result table.

3.5 Loading Points and Constraints:

The forces are applied on the impacted part of the Roll Cage as it is the first point of contact in case of a collision eg. In case of front Collision, forces are applied at front part and during side impact, forces are applied at Side Impact Members (SIM) of Roll Cage. For Roll-over analysis, forces are applied at an angle of 45° to the impacted member. The Suspension mounting Points on roll cage are fixed for all the static tests. For Front Bump, Forces are applied at the Front Suspension Points and rear suspension points are fixed to analyze the situation; and vice-versa for rear bump. A lot of failures brought about by resonance and excessive vibration of components and systems [Rajput et al., 2013]. Therefore, Modal Analysis is done, in which the frame is fixed at the lower base of the vehicle frame to know the different mode shapes of the upper substructure.

3.6 Von-Misses Stress:

Failure of mechanical components subjected to bi-axial or tri-axial stresses occurs when the strain energy of distortion per unit volume at any point in the component, becomes equal to the strain energy of distortion per unit volume in a standard tension test specimen during yielding. According to this theory, the yield strength in shear is 0.577 times the yield strength in tension [Dr. Sadhu Singh, “Strength of Materials”, Khanna Publishers India]. Experiments have shown that the distortion energy theory or Von-Misses Theory is better in agreement for predicting the failure of ductile components than any other theory of failure.

4. RESULTS

The following results are obtained during different tests of Roll cage

Table 3: Result of Impact Test for Front, Rear and Side

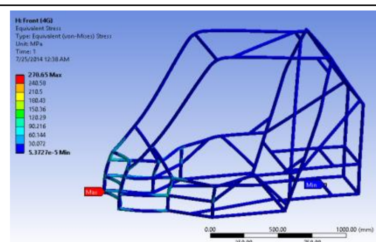
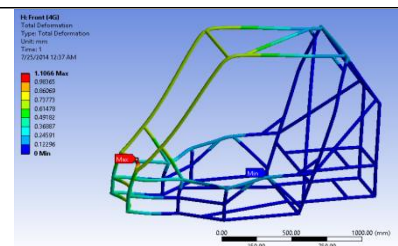
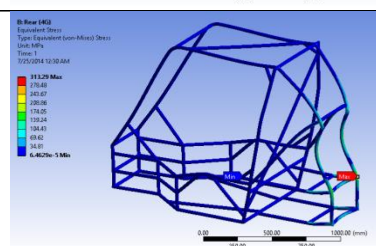
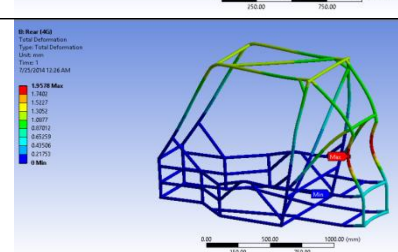
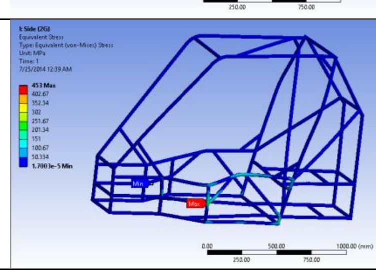
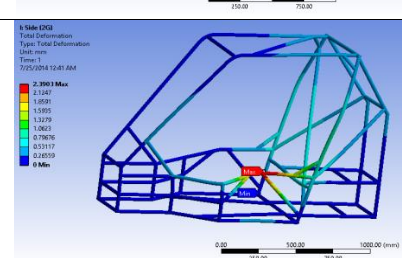
	Parameters	Values	Stress Analysis	Deformation
Front	Load Applied	10G		
	Max. Stress	270.65 MPa		
	Max. Deformation	1.11 mm		
	FOS	1.69		
Rear	Load Applied	5G		
	Max. Stress	313.29 MPa		
	Max. Deformation	1.96 mm		
	FOS	1.46		
Side	Load Applied	2G		
	Max. Stress	453 MPa		
	Max. Deformation	2.39 mm		
	FOS	1.01		

Table 4: Result of Bump Test for Front and Rear

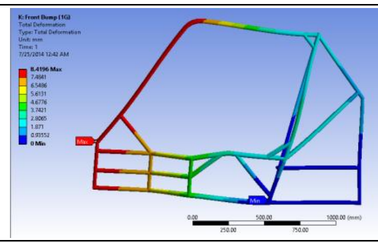
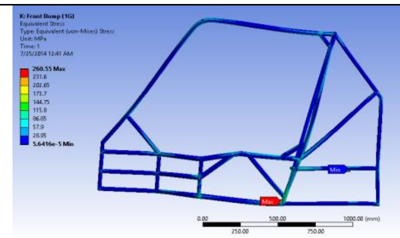
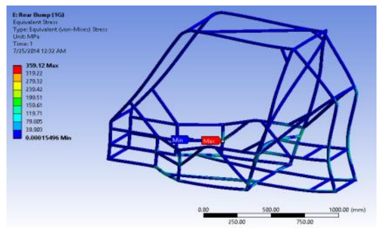
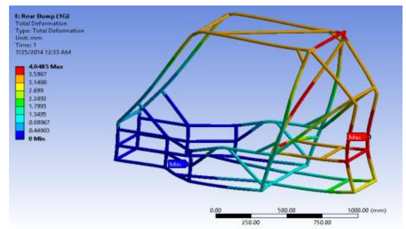
	Parameters	Values	Stress Analysis	Deformation
Front	Load Applied	1G		
	Max. Stress	260.55MPa		
	Max. Deformation	8.42 mm		
	FOS	1.76		
Rear	Load Applied	1G		
	Max. Stress	356.12MPa		
	Max. Deformation	4.05 mm		
	FOS	1.29		

Table 5: Result of Rollover Test for Front, Rear and Side

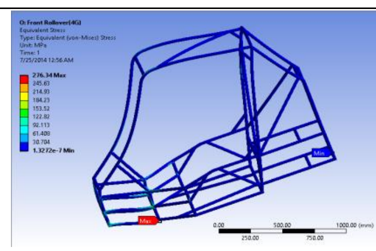
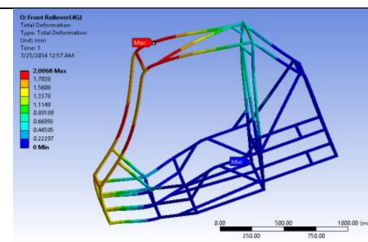
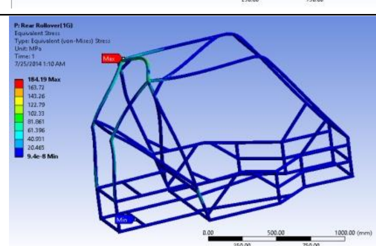
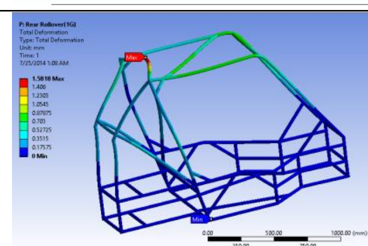
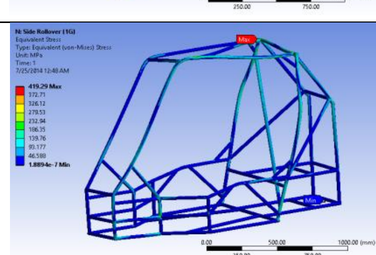
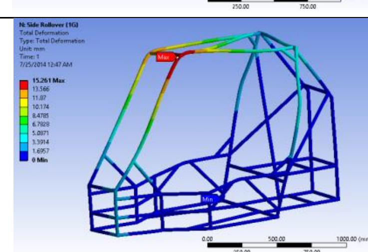
	Parameters	Values	Stress Analysis	Deformation
Front	Load Applied	1G		
	Max. Stress	276.34MPa		
	Max. Deformation	2.00 mm		
	FOS	1.66		
Rear	Load Applied	1G		
	Max. Stress	184.19Mpa		
	Max. Deformation	1.58 mm		
	FOS	2.49		
Side	Load Applied	1G		
	Max. Stress	184.19Mpa		
	Max. Deformation	1.58 mm		
	FOS	1.09		

Table 6: Result of Torsional Rigidity

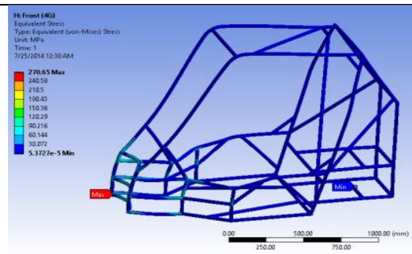
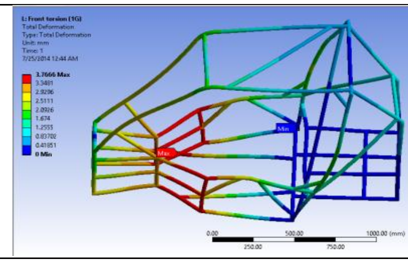
Parameters	Values	Stress Analysis	Deformation
Load Applied	1G		
Max. Stress	296.89MPa		
Max. Deformation	3.77 mm		
FOS	1.54		

Table 7: Result of Drop Test

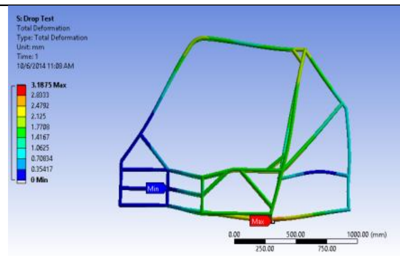
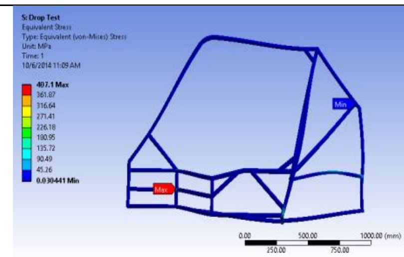
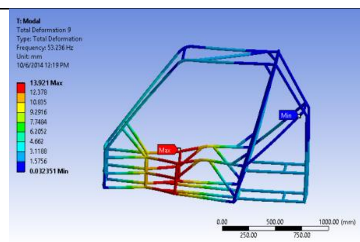
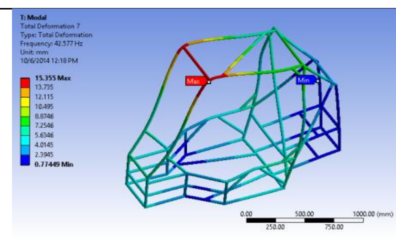
Parameters	Values	Stress Analysis	Deformation
Drop Height	4 meters		
Max. Stress	407.1MPa		
Max. Deformation	3.19 mm		
FOS	1.13		

Table 8: Result of Modal Analysis

Mode No.	Frequency (Hz)		
7	42.577		
8	45.203		
9	53.236		

5. CONCLUSION

Safety is of utmost concern in every respect; for the driver, crew & environment. A considerable Factor of Safety (FOS) or design factor is applied to the roll cage design to minimize the risk of failure and possible resulting injury. FOS value implies the safe value of applied loads and deformations. This clearly reaffirms the vehicle's ability to withstand extreme conditions. The usage of finite element analysis was invaluable to the design and analysis of the frame for All Terrain Vehicle. The designing and analysis is a difficult part to carry on as so many tests are needed to be conducted with lot of constraints. The chassis was designed so as the vehicle can withstand all kinds of loads and is capable of moving on terrains like hilly areas, Rocky Mountains etc. This paper thoroughly dealt with various load analysis on Roll Cage. The chosen design was the safest & the most reliable for any long terrain.

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