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Design and Optimisation of Sae Mini Baja Chassis

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ABSTRACT
The objective is to design and develop the roll cage for All - Terrain Vehicle accordance with the rulebook of BAJA 2014 given by SAE. The frame of the SAE Baja vehicle needs to be lightweight and structurally sound to be competitive but still protect the driver. The vehicle needs to traverse all types of off-road conditions including large rocks, downed logs, mud holes, steep inclines, jumps and off camber turns. During the competition events there is significant risk of rollovers, falling from steep ledges, collisions with stationary objects, or impacts from other vehicles. Material for the roll cage is selected based on strength and availability. A software model is prepared in Pro-engineer. Later the design is tested against all modes of failure by conducting various simulations and stress analysis with the aid of ANSYS 13. Based on the result obtained from these tests the design is modified accordingly. A target of 2 is set for Yield Factor of Safety.
***Keywords* -** SAE Baja vehicle, Factor of Safety, All Terrain Vehicle, Roll cage, Chasis

I. INTRODUCTION

A chassis consists of an internal framework that supports a man-made object in its construction and use. It is analogous to an animal's skeleton. If the running gear such as wheels and transmission, and sometimes even the driver's seat, are included then the assembly is described as a rolling chassis.

The Mini-Baja Vehicle is an off-road race vehicle powered by a small gasoline engine. As is such the combination frame and roll cage must be equally strong and light. In an effort to fulfill the rules set down by the governing body and ensure proper integration, strength, and weight minimization; it is imperative to properly analyze the material properties and geometry as well as the overall design geometry.

Types of Impact Tests: Front collision test, rear impact test, side impact test and roll over impact test . The vehicle in the track could hit a stationary object travelling at a speed of 30-40mph.The model is analyzed by applying the loads. The front collision test simulates the vehicle hitting a solid, immovable object at a speed of 35 mph . This is the maximum top speed the vehicle is expected to reach. The rear impact test simulates the vehicle being rear-ended by another 500 lb Baja vehicle, again at a speed of 35 mph. To make this test as hard as possible, the front of the vehicle is resting against a solid wall. The side impact test is identical to the rear impact, but the vehicle is oriented sideways relative to the motion of the incoming 500lb vehicle. Roll over impact simulates the vehicle rolled on its side.

II. DESIGN AND DEVELOPMENT

2.1 Material Selection

As per the constraints given in the rulebook[1], the roll cage material must have at least 0.18% carbon content. The following materials which are commercially available and are currently being used for the roll cage of an ATV are shortlisted. A comparative study of these shortlisted materials is done on the basis of strength, availability and cost. The shortlisted materials are as follows.

Table 2.1: Material Properties

	AISI 1018 steel	AISI 1026 steel	AISI 4130 alloy steel
Density (g/cc)	7.87	7.85	7.85
Poisson's ratio	0.29	0.27-0.30	0.27-0.30
YoungsModulus (GPa)	205	190-210	190-210
Carbon content (%)	0.14-0.2	0.22-0.28	0.28-0.33
Tensile strength Yield (MPa)	370	415	460

2.2 Material Requirements

The materials used in the cage must meet certain requirements of geometry as set by SAE, and other limitations. As the frame is used in a racing vehicle, weight is a crucial factor and must be considered. The proper balance of fulfilling the design requirements and minimizing the weight is crucial to a successful design.

The rules define the cage to be made with materials equivalent to the following specification Steel members with at least equal bending stiffness and bending strength to 1018 steel having a circular cross section having a 25.4 mm (1 inch) OD and a wall thickness of 3 mm (0.120 in.)[1]A key factor of this statement is those only steel members are allowed for the frames construction. However the alloy of the steel is definable by the competitor as long as it meets the equivalency requirements. These values are required to be calculated about the axis that gives the lowest value. Calculating the strength and stiffness this way ensures that tubes with a non-circular cross-section will be equivalent even in a worst case loading situation. The rules go on further to define bending strength and stiffness by: Bending stiffness is proportional to the EI product and bending strength is given by the value of $S_y I/c$, (for 1018 steel the values are; Sy= 370Mpa (53.7ksi) E=205GPa (29,700 ksi).

E = the modulus of elasticity

I = the second moment of area for the cross section about the (inch^4)

axis giving the lowest value

I = $\pi(D_o^4 - D_i^4)/64$

Sy = the yield strength of material (psi)

c = the distance from the neutral axis to the extreme fiber

Table 2.2: Bending stiffness Vs Wall thickness

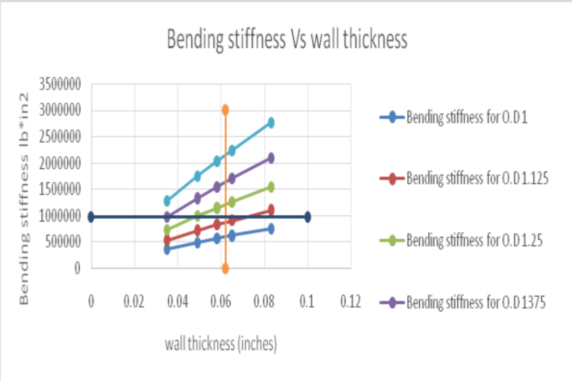
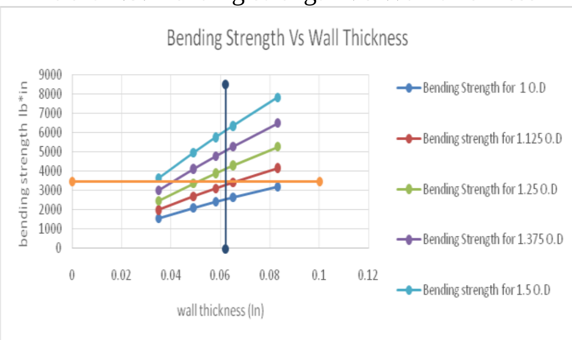


Table 2.3: Bending strength Vs Wall thickness



After reviewing each of these analyses it is evident that the best choice would be use 4130 Chromoly tubing with a 1.125 inch diameter and a 0.083 inch wall thickness.

2.3 GEOMETRY CREATION

The design was made using the Pro-engineer software package. The model was made fully parametric. This means the features of the model are based upon those preceding it, and will change according to any modifications to the parent features. The usage of parametric design was extremely important with this design. As so many factors interact in the design of the frame, the parametric properties allowed the change of a single part to automatically change the design of all parts interacting with it.

III. ANALYSIS

The next stage in the design process is to analyze the frame and add features accordingly. There were a few features of the design that might need some additional strengthening. For these reasons it was deemed that there should be an analysis of front impact, side impact, rollover impact, and the loading on the frame from the front shocks. However before these analyses are performed an examination of the loading forces exerted on the vehicle must be completed. The finite element analysis software program used for this project was ANSYS.

PIPE16 is a uniaxial element with tension-compression, torsion, and bending capabilities. The element has six degrees of freedom at two nodes: translations in the nodal x, y, and z directions and rotations about the nodal x, y, and z axes. This element is based on the three-dimensional beam element (BEAM4), and includes simplifications due to its symmetry and standard pipe geometry.

Total number of elements = 6305

Total number of Nodes = 6276

3.1 FRONTAL IMPACT

The first analysis to be completed was that of a front collision with a stationary object. In this case a deceleration of 10 G's was the assumed loading. The model is supposed to make contact at its front junctions where FBM (Front Bracing Members) SIM(Side Impact Members) and LFS(Lower Frame Side)members join. So the loads act horizontally in positive X direction on this points.

3.1.2 Boundary conditions

Calculation of frontal impact force = M * 10*9.8

Mass = 320 kg (combined weight of vehicle and occupant)

Load=31360 N

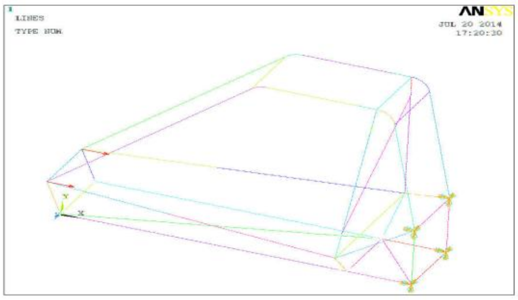


Fig 3.1 :Loading conidtions for Frontal impact

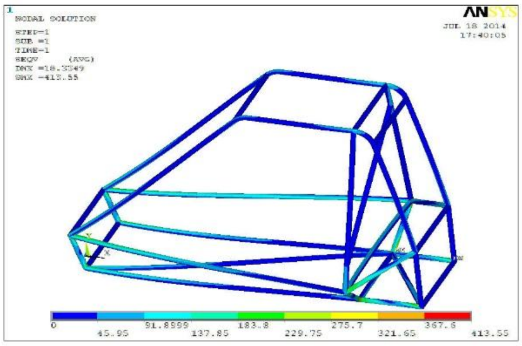


Fig 3.2: Overall Von-Mises stress view (model 1)

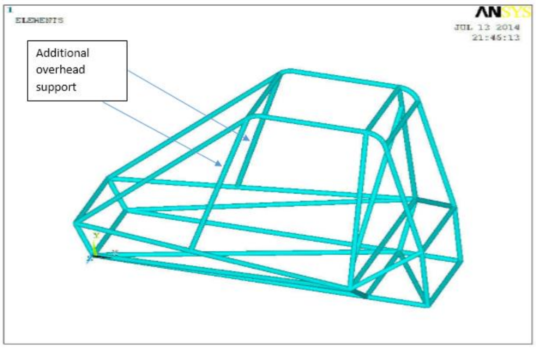


Fig 3.3: with additional bracings (Model3)

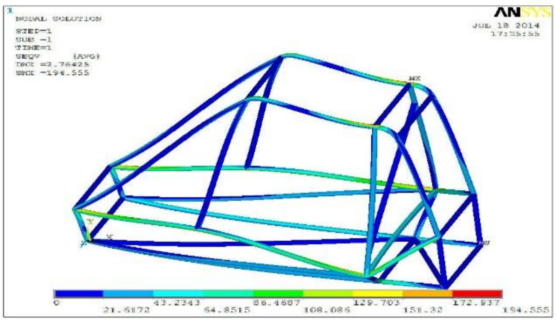


Fig 3.4 : Overall Von-Mises stress view(model 3)

Table 3.1: Results for Frontal impact analysis

Frontal impact analysis Outer Dia-28.575mm;Wall Thickness – 2.1082mm					
	Max Von mises stressMPa	Max Displacement mm	Weightkgs	Factor of safety	status
Model 1	413.55	18.3349	30.97	1.112	FAIL
Model 2	236.07	5.37043	35.5	1.948	FAIL
Model 3	194.555	2.76425	37.99	2.364	PASS

3.2 SIDE IMPACT

The next step in the analysis was to analyze a side impact with a 5 G load. The model is impacted on its side. This is equivalent to a loading force of 16KN. The point of application of this force is shown in Figure3.5. The Detailed view of the resulting stress is shown in Figure 3.6.

3.2.1 Boundary conditions

Calculation of frontal impact force = M * 5*9.8
Mass = 320 kg (combined weight of vehicle and occupant)
Load= 15680 N (a load of 16000N is applied)

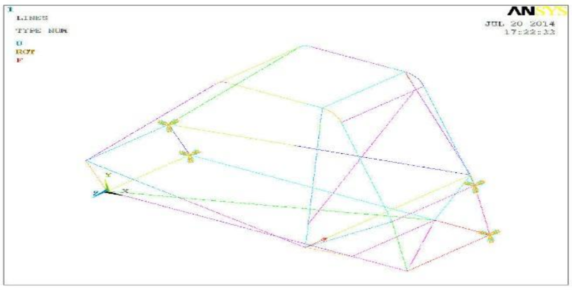


Fig 3.5: Loading conditions for Side Impact

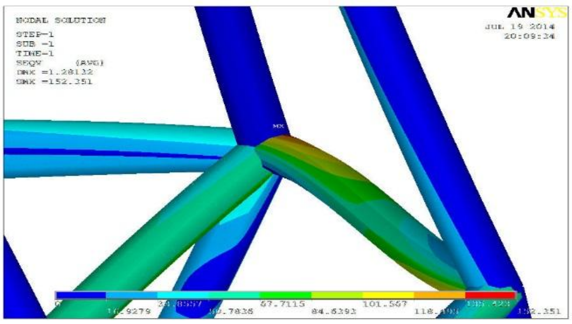


Fig 3.6: Detailed Von-Mises stress view(Model 3)

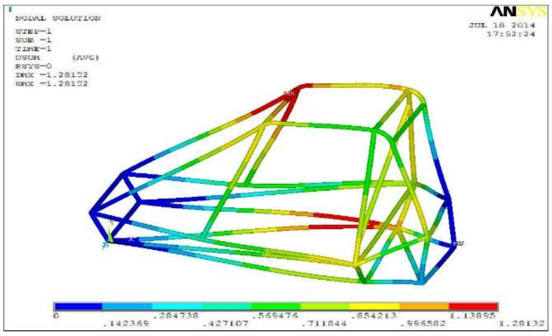


Fig 3.7: Max displacement(Model 3)

Table 3.2: Results for Side impact analysis

SIDE impact analysis comparision(Model 3)				
Max Von mises stress	Max Displacement	Weight (kgs)	Factor of Safety	Status
152.351	1.2813	37.99	3.019	PASS

3.3 REAR IMPACT ANALYSIS

The next step is to analyse the model for Rear impact with 5g load which is equivalent to 16KN.The point of application is shown in figure 3.8 below.

3.3.1 Boundary Conditions

Calculation of frontal impact force = M * 5*9.8
Mass = 320 kg (combined weight of vehicle and occupant)
Load= 15680 N (a load of 16000N is applied)

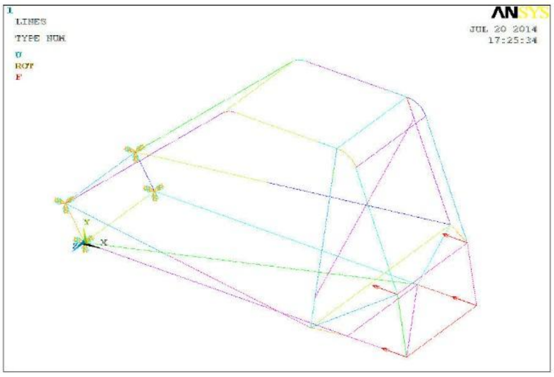


Fig 3.8 : Loading conditions for Rear Impact

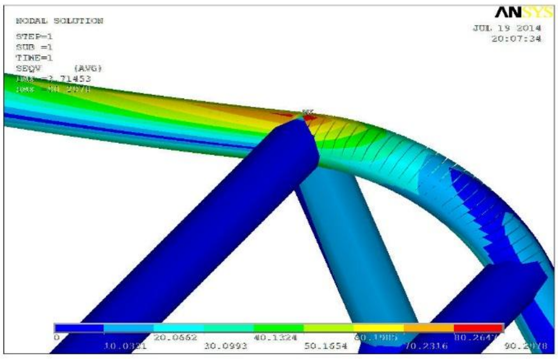


Fig 3.9: Detailed Von-Mises stress view(model 3)

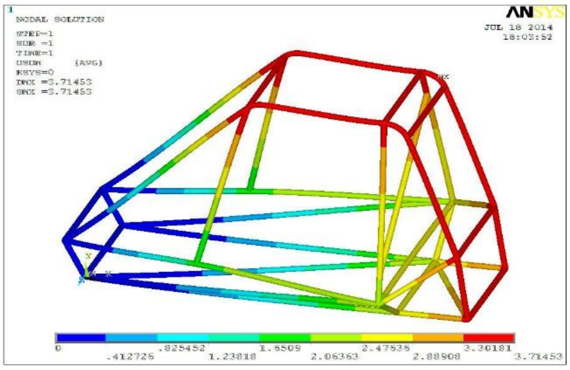


Fig 3.10: Max displacement(Model 3)

Table 3.3 : Results for Rear impact analysis

REAR impact analysis comparision(Model 3)				
Max Von mises stress	Max Displacement	Weight kgs	Factor of Safety	status
90.2978	3.7145	37.99	5.094	pass

3.4 ROLL-OVER IMPACT ANALYSIS

The Final step in the analysis was to analyze the stress on the roll cage caused by a rollover with a 2.5 G load on the cage. This is equivalent to a loading force of 8KN. The Loading was applied to the upper forward corners of the perimeter hoop with a combination vector sideways and downward. Figure 3.11 shows the point of application for the loading on the roll cage.

3.4.1 Boundary Conditions

Calculation of frontal impact force = M * 2.5*9.8
Mass = 320 kg (combined weight of vehicle and occupant)
Load= 7840 N (a load of 8000N is applied) With combination of horizantal and vertical force components.

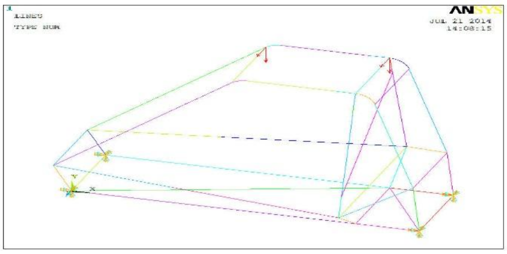


Fig 3.11: Loading conditions for roll over impact

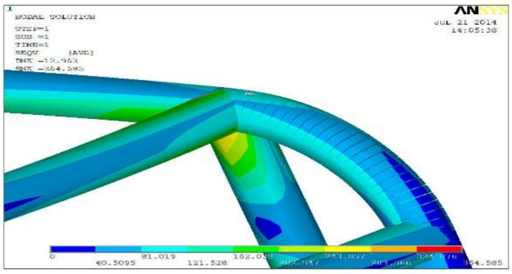


Fig 3.12: Detailed Von-Mises stress view(model 3)

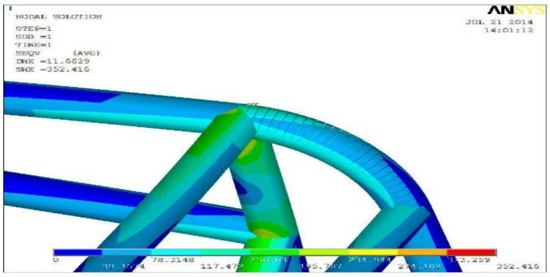


Fig 3.13: Detailed Von mises stress view (model 4)

Table 3.4: Results for Roll-Over impact analysis

Type of Impact	Initial model's factor of safety	Optimised model's Factor of safety	Status
Frontal	0.832	**2.322**	PASS
Side Impact	2.003	**2.247**	PASS
Rear mpact	1.878	**4.336**	PASS
Roll-Over	0.762	**1.8517**	FAIL

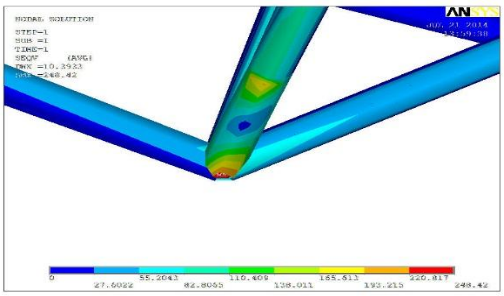


Fig 3.14: Detailed Von-mises stress view (model 5)

IV. RESULTS AND DISCUSSIONS

The modifications approved as per Roll over impact would also affect for Frontal,Side as well as Rear Impact. The final model is thus carried out for analysis of frontal, side and rear impacts to determine their respective vonmises stress and maximum displacements. The final results obtained are show in the Table 3.5.

Table 3.5 :Final Results

V. Conclusion

The usage of finite element analysis was invaluable to the design and analysis of the frame. The analysis allowed the addition of important and key structural components to help the vehicle with stand front, side impacts as well as the rear impacts. While a viable solution to the stresses seen in a rollover type impact could not be found due to the set design constraints, the finite element analysis gave a very accurate prediction of where failure would occur in this situation.

ROLL impact analysis comparision					
	Max Von mises stress	Max Displacement	Weight kgs	Factor of Safety	status
Model 3	364.585	12.962	37.99	1.2617	Fail
Model 4	352.416	11.6629	39.6	1.3052	Fail
Model 5	248.42	10.3933	39.99	1.8517	Fail

REFERENCES

[1] Rulebook BAJA SAE INTERNATIONAL 2014.

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