OPEN ACCESS

International
Journal
Of Modern Engineering Research (IJMER)

Simulation and Static Analysis of an Off-Road Vehicle Roll Cage

R. Bhandari[1], P. Birajdar[2], A. Dafedar[3], S. Bammani[4], A. Pereira[5]
[1,2,3,4,5] Department of Mechanical Engineering, N. K. Orchid College of Engineering and Technology, Gat No.16, Solapur-Tuljapur Road, Tale-Hipparaga, Solapur 413002, India

Abstract: *The SAE-BAJA competition is arranged every year with a purpose to have teams of engineering students design, build and race a prototype of a four-wheel, one passenger, off-road vehicle. The most important aspect of the vehicle design is the frame. The frame contains the operator, engine, brake system, fuel system and steering mechanism, it must be of adequate strength to protect the operator in the event of a rollover or impact. The roll cage must be constructed of steel tubing, with minimum dimensional and strength requirements dictated by Society of Automotive Engineers (SAE). Increased concern about the roll cage has created the importance of simulation and analysis thereby predicting failure modes of the frame. In the present paper, we have used ANSYS to investigate the response of the frame under various impacts. We considered a direct frontal impact and side impact that results in a 4g horizontal loading, a rollover impact of 3g deceleration value, bump impact and front torsional impact analysis with 3g deceleration value. The impact loading is simulated by restricting displacements at certain locations, and applying discrete forces at various points on the frame where the weight is concentrated. Throughout the analysis of roll cage more emphasis was given on obtaining a allowable factor of safety and designed according to it.*

Keywords: *roll cage; frontal impact; side impact; rollover impact; ANSYS.*

I. Introduction

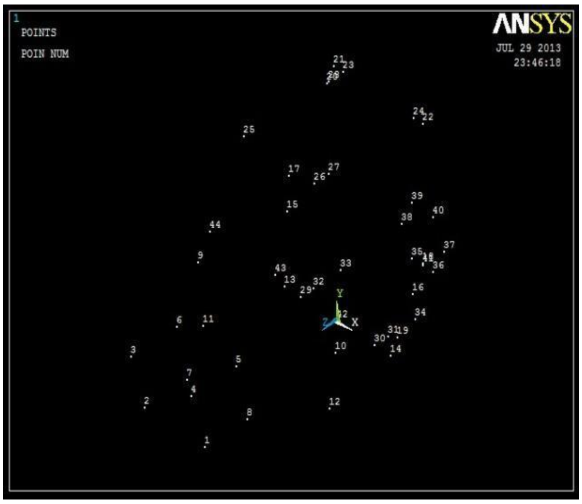
A frame of a vehicle plays the most important role in safety of the passenger. The frame contains the operator, engine, brake system, fuel system, and steering mechanism, and must be of adequate strength to protect the operator in the event of a rollover or impact. The passenger cabin must have the capacity to resist all the forces exerted upon it. This can be achieved either by using high strength material or better cross sections against the applied load. But the most feasible way to balance the dry mass of roll-cage with the optimum number of members is done by triangulation method. The roll cage must be constructed of steel tubing, with minimum dimensional and strength requirements dictated by SAE. The SAE BAJA vehicle development manual also restricts us about the vehicle weight, shape and size, and dimensions [1-3].

Circular cross-section is employed for the roll cage development as it helps to overcome difficulties like increment in dimension, rise in the overall weight and decrease in fuel efficiency. It's always a perfect one to resist the twisting and the rolling effects, therefore is preferred for torsional rigidity.

a) Design objective of roll cage are:

1) Provide full safety to the driver, by obtaining required strength and torsional rigidity, while reducing weight through diligent tubing selection.
2) Design for manufacturability, as well as cost reduction, to ensure both material and manufacturing costs are competitive with other SAE vehicles.
3) Improve driver comfort by providing more lateral space and leg room in the driver compartment.
4) Maintain ease of serviceability by ensuring that roll cage members do not interfere with other subsystems [4-6].

This roll cage is developed in ANSYS Multiphysics Menu by plotting the keypoints, lines and arcs. The element type selected for it is PIPE 16, a uniaxial element with tension, 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. The real constants involved in the pre-processing of PIPE 16 element are its outer diameter and thickness value. The material used for the roll cage is AISI 1018 with Young's Modulus 210 GPa; yield strength is 365.5 MPa and Poisson's ratio 0.29. The density of material is 8000 kg/m^3 with hardness (Brinell) of 126 HB [7].



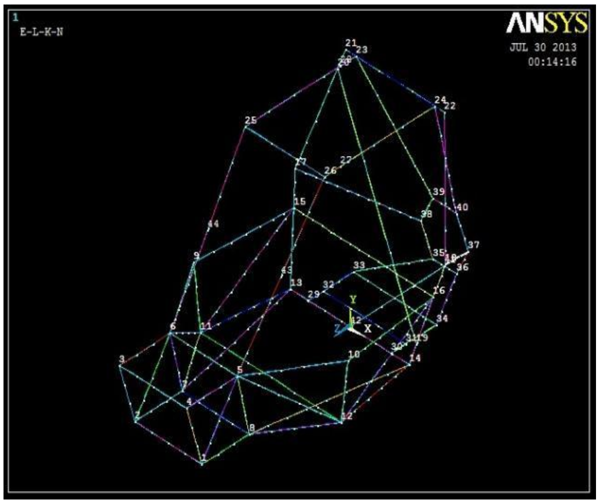


Figure 1. Modelling of keypoints

Figure 2. Modelling of lines

II. Meshing And Analytical Calculations

a) Meshing constraints and calculations

As the roll cage was developed by plotting keypoints, lines and splines, so every member of the roll cage is considered to be properly constrained at every joint. For boundary conditions for frontal impact test, the roll cage is to be fixed from the rear side and the front member will come across the applied load. In the similar way, for side impact test, one side of the roll cage elements are fixed while the other side will be applied with load. For rollover impact test, the lower elements of the roll cage are fixed. For bump impact test and torsional impact test, the roll cage is to be fixed from the rear side. The load will be distributed among the number of joints framed by front members in the opposite direction to the frame, i.e. in X axis.

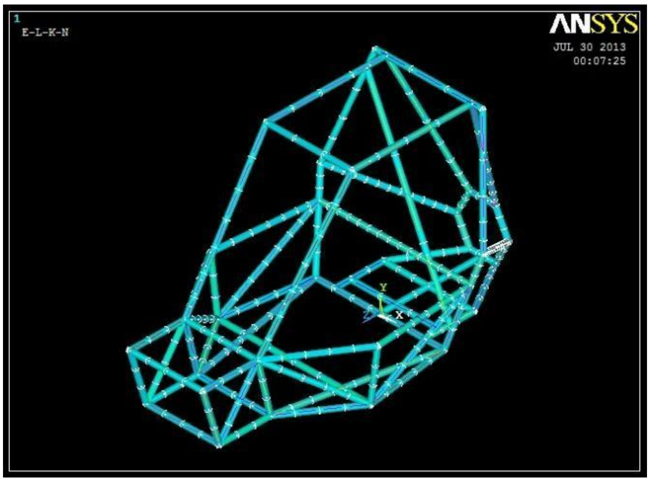


Figure 3. Meshed model with element shape

b) Analytical calculation for determining impact on roll cage

To properly analyze the impact force, we need to find the deceleration of the vehicle after impact. To approximate the worst case scenario that the vehicle will undergo, momentum equations were used to determine the deceleration of the vehicle. The vehicle was considered to be at maximum speed of 60 km/hr having total weight of 400 kg and according to different scenarios the conditions of head on impacts, oblique collisions, and inelastic or partially elastic collisions were employed with a crash pulse consideration of 0.1s.

The forces which were impacted on the roll cage were decelerations of 4g & 3g and it is calculated as follows:-

Assume gravitational force = 9.8 m/s² ≈ 10 m/s²

g = mass of the vehicle × gravitational force acting on the vehicle
  = 400 × 10
  = 4000 N
Therefore, 4g = 16000 N & 3g = 12000 N.

III. Impact Analysis Using Ansys

a) Frontal Impact Analysis

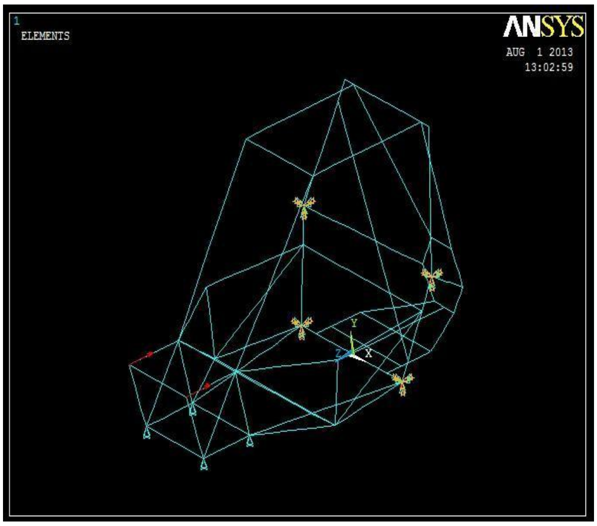


Figure 4. Frontal impact load application

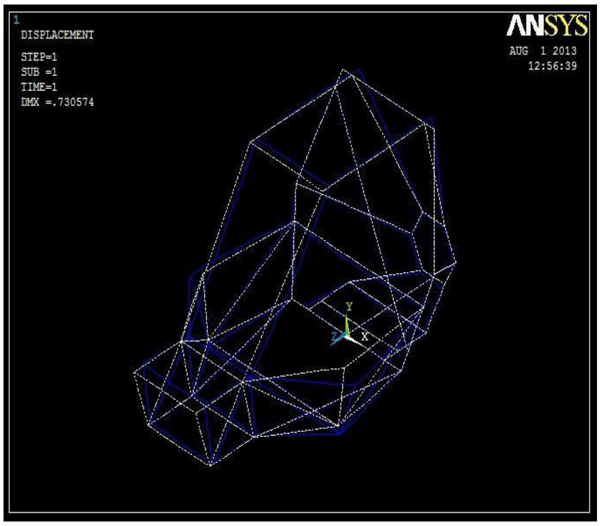


Figure 5. Deformation plot

It is the impact wherein there is a possibility of vehicle crashing into another vehicle head on during the race (Sharma and Purohit, 2012).The deceleration value for frontal impact is 4g. This is equivalent to a loading force of 16000 N. The load is applied on two nodes at front as depicted by red arrows in (Fig. 4). Thus, the force gets divided into two parts i.e. 8000 N on each node. The value recorded for the deformed shape is 0.73 mm (Fig. 6) which abides by the safety regulations and standards of SAE BAJA competition. (Fig. 7) shows the Von Mises stress plot, where the maximum stress is observed at the front members (68.673 MPa) where the load is applied. The driver cabin members are shown in yellow and green colors which clearly depicts the safety of driver cabin even when loaded with such high force.

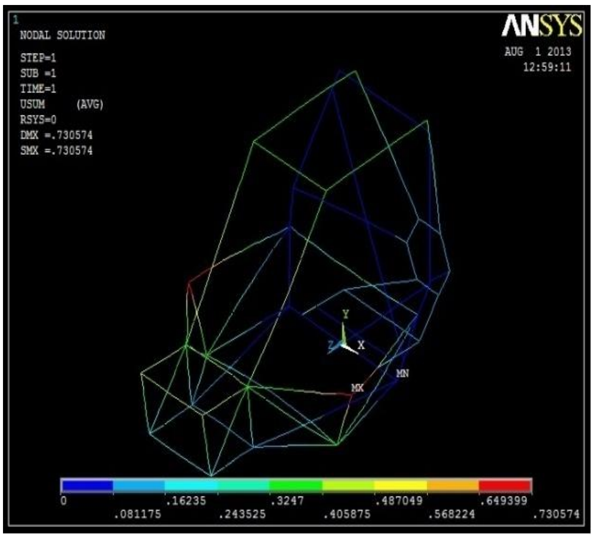


Figure 6. Nodal solution plot

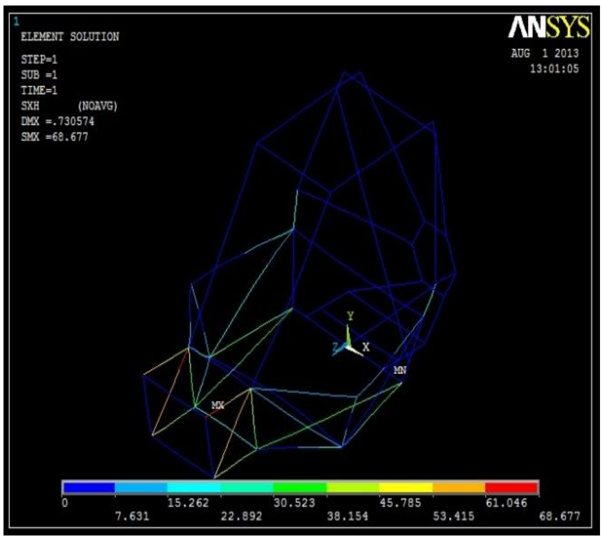


Figure 7. Von Mises stress plot.

b) Side Impact Analysis

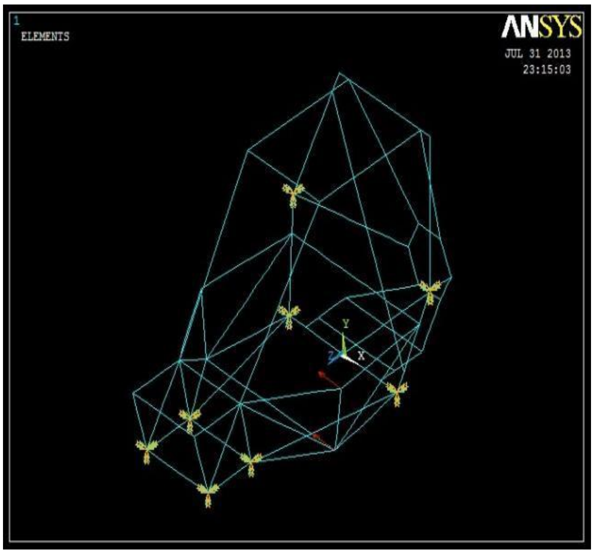


Figure 8. Side impact load application

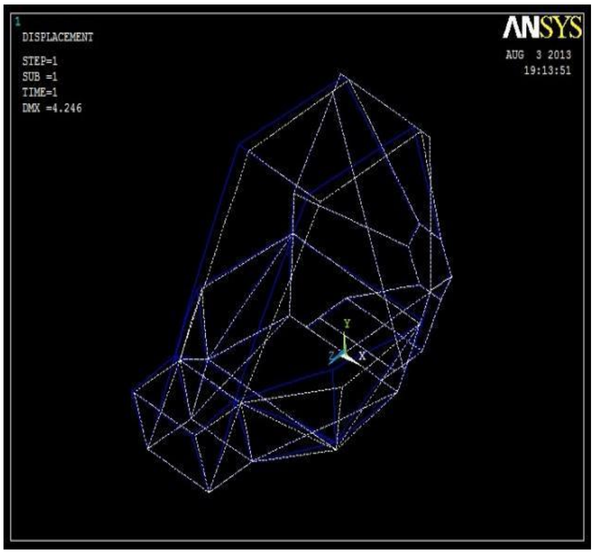


Figure 9. Deformation plot

The side impact analysis is carried out as there is a possibility of collision with another vehicle from either direction. Thus, the stresses acting on the side members of the roll cage are analyzed. The deceleration value for side impact is 4g. This is equivalent to a loading force of 16000 N. The load is applied on two nodes as shown in (Fig. 8). Thus, the force on each node is 8000N. The nodal solution shows a deformation of 4.246 mm in colored contour same as the deformation plot as depicted in (Fig. 10). The Von Mises stresses came out to be 99.453 MPa which is inside the permissible range of the material (Fig. 11). The driver cabin members are shown with green colours which reflect the safety of driver cabin even when such a high load is introduced.

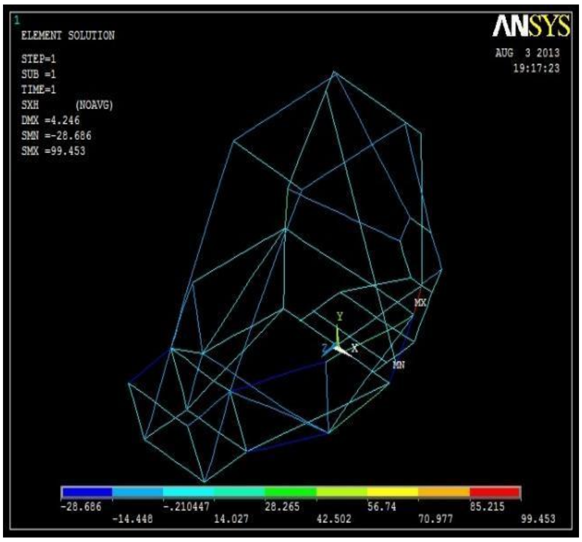


Figure 10. Nodal Displacement plot

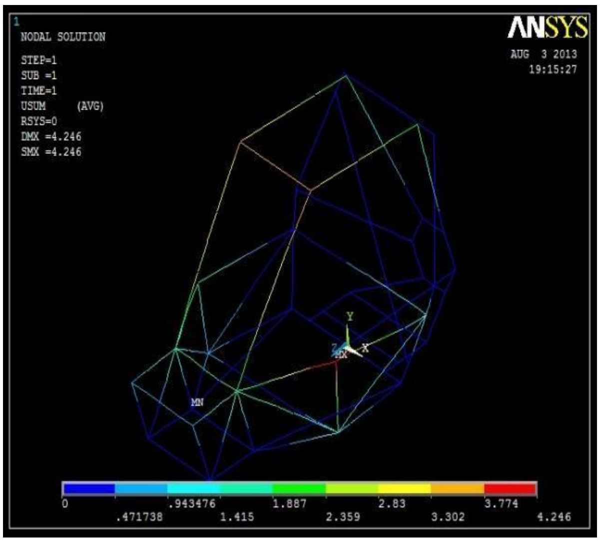
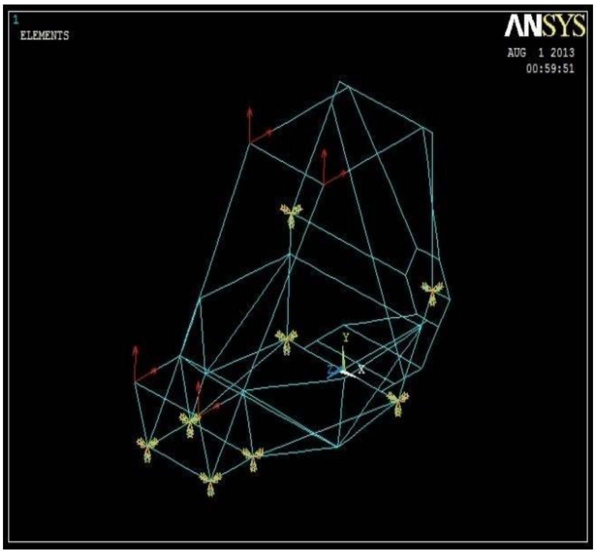


Figure 11. Von Mises stress plot

c) Rollover Impact Analysis



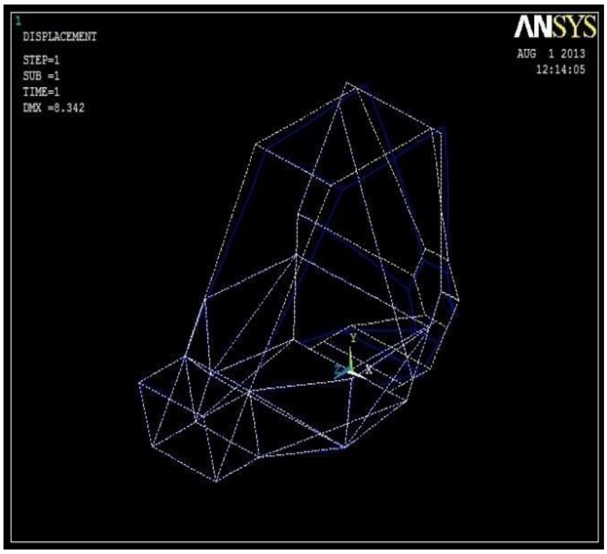
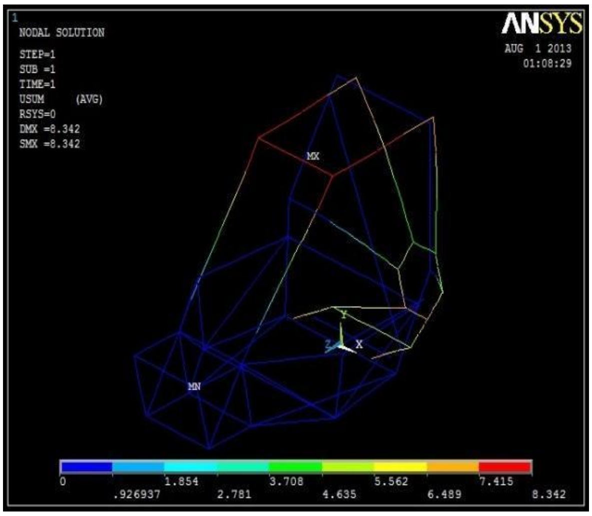


Figure 12. Rollover impact load application *Figure 13. Deformation plot*

The rollover impact analysis is carried out by considering the stresses induced on the members of the roll cage when the vehicle topples down from a slope with an angle of 45°. In this impact, the upper and rear members of the vehicle will bear the force. The deceleration value for rollover impact is 3g. This is equivalent to a loading force of 12000 N. The number of nodes on which the load is applied is 4 (Fig. 12). Thus 3000 N was applied on each node. The maximum deformation is 8.342 mm in the members of the vehicle as shown in (Fig. 14). The Von Mises stress induced on the members is shown in (Fig. 15). The maximum stress i.e. 139.69 MPa was observed at the upper members of the vehicle which is well below the permissible range.



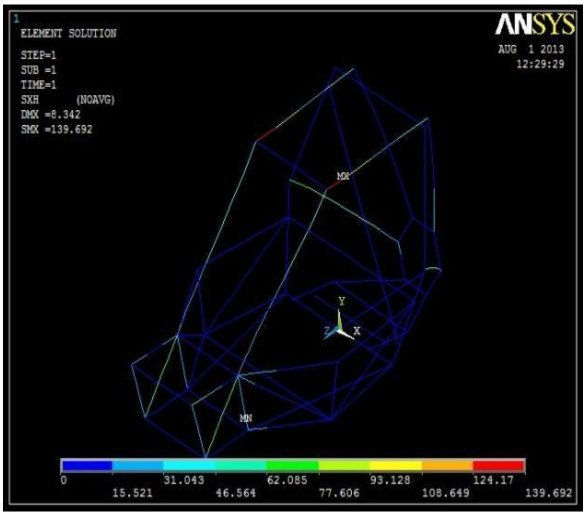


Figure 14. Nodal solution plot *Figure 15. Von Mises stress plot*

d) Two Wheel Bump Impact Analysis

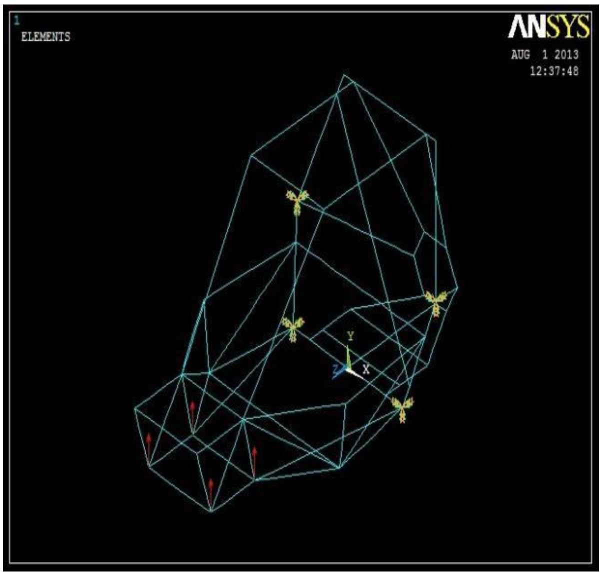


Figure 16. Bump impact load application

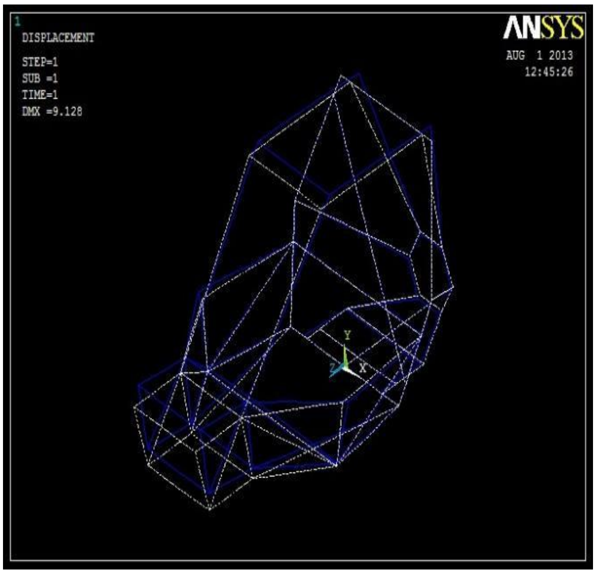


Figure 17. Deformation plot

During the SAE-BAJA competition, the vehicle has to travel on uneven tracks. There are times when the vehicle moving along an upward slope travels about a curved projectile in air before landing on its wheels. The lower frontal part of the vehicle is the initial member which faces this impact. Once the front tyres touch the surface, the suspension system absorbs the initial forces exerted on it. A time comes when the suspension system are compressed to its maximum extent and act like solid member of the vehicle. The rest of the load is transferred to the roll cage members of the vehicle [9]. In order to ensure the safety of the driver, we determine this impact force using ANSYS.

The deceleration value for bump analysis was taken as 3g i.e. 12000N.The force was applied on the frontal four suspension pick up points (Fig. 16). The load applied on each node was 3000 N. The deformation value is 9.128 mm which abides by the safety regulations and standards of SAE BAJA competition (Fig. 18). The Von Mises stress is plotted in (Fig. 19) showing the individual stresses in the members. The maximum stress as observed was found out to be 190.79 MPa which is within the permissible limit of the materials yield strength.

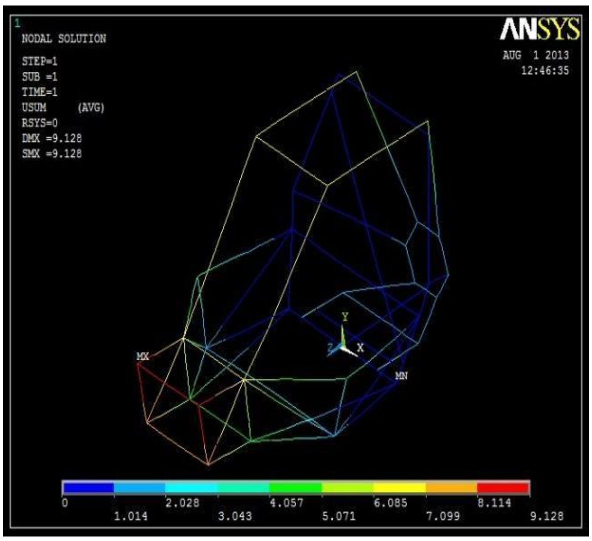


Figure 18. Nodal solution plot

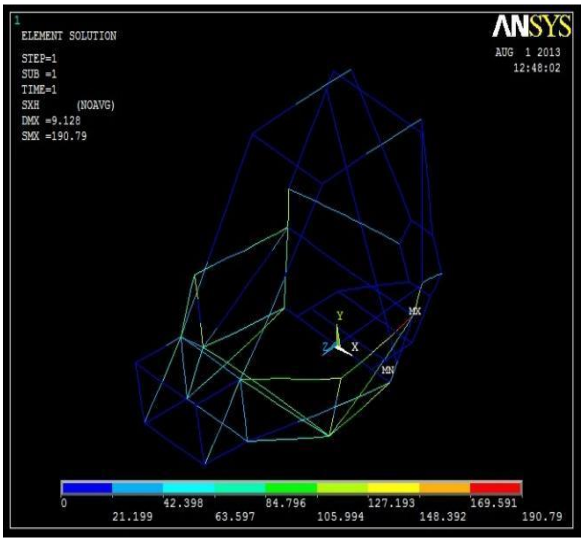
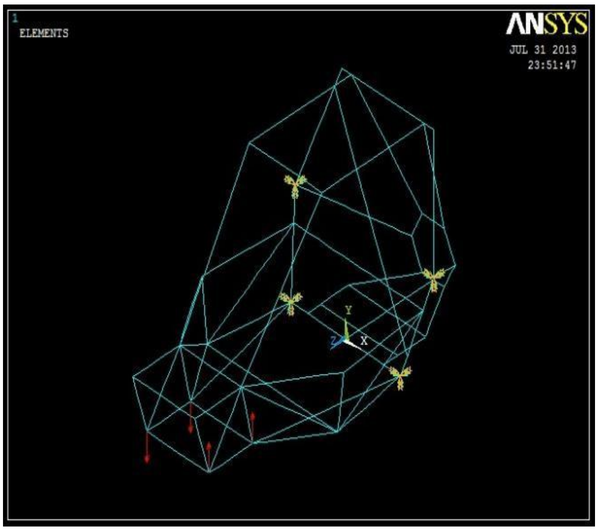


Figure 19. Von Mises stress plot

e) Front Torsional Impact Analysis

This impact is analyzed taking into consideration the torsional forces acting on the frontal elements of the vehicle. This type of force is exerted on the vehicle when it traverses on an uneven road. The two tyres on the front axle experience a moment. The torque is applied to one tyre and reacted by the other one (Fig. 20). These forces are equal and opposite. The deceleration value for this impact is 3g i.e. 12000 N. But as these act in opposite direction, the number of nodes for application of force is 2. The amount of force per node is 6000 N.

The deformed value after analysis is 9.476 mm as shown in (Fig. 22). The maximum Von Mises stress in this analysis came out to be 90.117 MPa which lies within the permissible range of the material (Fig. 23).



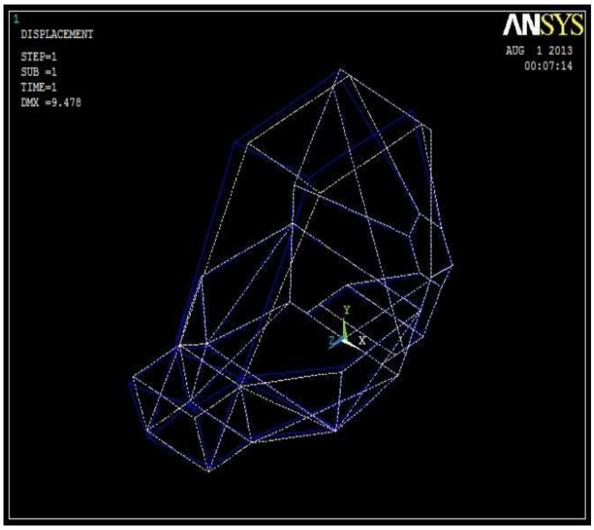
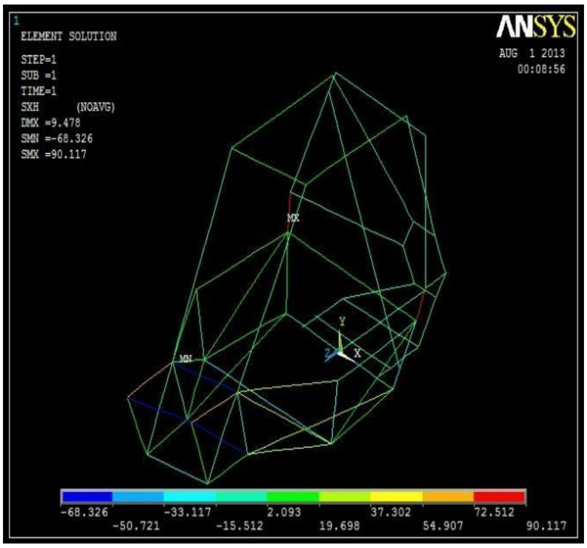


Figure 20. Torsional impact load

Figure 21. Deformation plot application



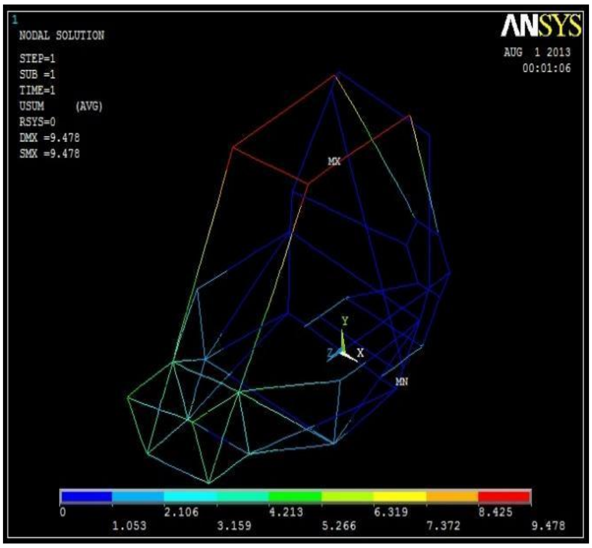


Figure 22. Nodal solution plot

Figure 23. Von Mises stress plot

IV. Results And Discussion

Stress plots and deformations of critical elements undergoing different loads during the impact tests were analyzed using ANSYS. This project helps us to understand the vital components of designing. As mentioned above the yield strength of the material which we are using is 365.5 MPa. The maximum stress values of various impact tests have been determined and we can easily find the factor of safety of the vehicle. Safety is of utmost concern in every respect; for the driver, crew & environment. Considerable factor of safety

(FOS) or design factors is applied to the roll cage design to minimize the risk of failure & possible resulting injury. This FOS value implies the safe value of applied loads and deformations. The following table shows the various loading conditions, deformations, maximum stress values and factor of safety for various test conditions.

Table 1. Analysis results of impact tests.

Factor of Safety (FOS)	Von Mises stress (MPa)	Maximum Deformation (mm)	Number of Nodes	Loading force (N)	Type of impact test
5.322	68.673	0.73	2	16,000	Front
3.675	99.453	4.246	2	16,000	Side
2.616	139.69	8.342	4	12,000	Roll-over
1.915	190.79	9.128	4	12,000	Bump
4.157	90.117	9.476	2	12,000	Torsional

V. Conclusion

The use of finite element analysis was invaluable to the design and analysis of the frame for SAE-BAJA off-road vehicle. The analysis was helpful in finding out the maximum deformation, Von Mises stress and the factor of safety for five different impact tests namely frontal impact, side impact, rollover impact, bump impact and torsional impact. The findings from the finite element analysis and the actual failure will allow future designers to integrate a solution to this problem into their design from the beginning.

REFERENCES

[1]. BAJA SAE INDIA Rulebook. 2013. http://www.bajasaeindia.org/downloads.html.

[2]. Milliken, D., Kasprak, E., Metz, L. and Milliken W. (2003). *Race Car Vehicle Dynamics: Problems, Answers and Experiments*. Society of Automotive Engineers, Inc.

[3]. Gillespie, T., (2001). *Fundamentals of Vehicle Dynamics*, Society of Automotive Engineers, Inc.

[4]. Fenton, J., (1996). *Handbook of Vehicle Design Analysis*, Society of Automotive Engineers, Inc.

[5]. Erjavec, J., (2009). *Automotive Technology: A Systems Approach.* Delmar Cengage Learning.

[6]. *Huang, M., (2002). Vehicle crash mechanics. CRC press.*

[7]. Dave, H.K., Desai, K.P. and Raval, H.K. 2008. Investigations on prediction of MRR and surface roughness on electro-discharge machine using regression analysis and artificial neural network programming. Proceedings of the World Congress on Engineering and Computer Science, pp. 123-128.

[8]. Ezugwu, E.O. 2007. Improvements in the machining of aero-engine alloys using self-propelled rotary tooling technique. Journal of Materials Processing Technology, 185: 60-71.

[9]. Hascalik, A., Caydas, U. and Gurun, H. 2007. Effect of traverse speed on abrasive waterjet machining of Ti–6Al–4V alloy. Materials and Design, 28: 1953–1957.

[10]. Kansal, H.K., Singh, S. and Kumar, P. 2005. Parametric optimization of powder mixed electrical discharge machining by response surface methodology. Journal of Materials Processing Technology, 169: 427–436.

[11]. Sharma, V. and Purohit, D. 2012. Simulation of an Off-road Vehicle Roll Cage: A Static Analysis. International Journal of Engineering Research and Applications, 4(2): 126-128.

[12]. Kao, J.Y. and Tarng, Y.S. 1997. A neutral-network approach for the on-line monitoring of the electrical discharge machining process. Journal of Materials Processing Technology, 69: 112-119.

[13]. Khan, M.A.R., Rahman, M.M., Kadirgama, K. and Bakar, R.A. 2012. Artificial neural network model for material removal rate of Ti-15-3 in electrical discharge machining. Energy Education Science and Technology Part A: Energy Science and Research, 29(2): 1025-1038.

[14]. Rahman, M., Wong, Y.S. and Zareena, A.R. 2003. Machinability of titanium alloys. JSME International Journal Series C: Mechanical Systems, Machine Elements and Manufacturing, 46: 107-115.

[15]. Rahman, M.M., Khan, M.A.R., Kadirgama, K., Maleque, M.A. and Bakar, R.A. 2011a. Parametric optimization in EDM of Ti-6Al-4V using copper tungsten electrode and positive polarity: A statistical approach. Mathematical Methods and Techniques in Engineering and Environmental Science, pp. 23-29.

[16]. Rahman, M.M., Khan, M.A.R., Kadirgama, K., Noor, M.M. and Bakar, R.A. 2011b. Optimization of machining parameters on tool wear rate of Ti-6Al-4V through EDM using copper tungsten electrode: A statistical approach. Advanced Materials Research, 152-153: 1595-1602.

[17]. Razak, N.H., Rahman, M.M. and Kadirgama, K. 2012. Investigation of machined surface in end-milling operation of Hastelloy C-2000 using coated-carbide insert. Advanced Science Letters, 13: 300-305

[18]. Wu, C.F.J. 2002. Experiments planning, analysis and parameter design optimization. 2nd ed. New York: John Wiley.