Published by :
http://www.ijert.org

International Journal of Engineering Research & Technology (IJERT)
ISSN: 2278-0181
Vol. 11 Issue 02, February-2022

# Design and Finite Element Analysis of Four-Wheel Drive Roll Cage

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***Abstract*: Roll cage design has been implemented and enhanced over past decade all over the world for all terrain performance. Four-wheel drive-in BAJA SAE is a newly started concept for which this paper discusses design and analysis of roll cage for off roading conditions. A detailed process of 3D sketching along with Finite element study is discussed and reported. The roll cage for the usage of four-wheel drive layout is designed considering safety, aesthetics, driver comfort without reducing the strength of the chassis.**

***Keywords: Roll cage, design, four-wheel drive, ATV, finite element analysis***

## 1. INTRODUCTION

When it comes to motorsports, especially all-terrain vehicles, the roll cage plays an important role in the safety. The prevention of deaths or fatal injuries in the situations like sudden impact or vehicle crash. Apart from forming a rigid structure against loads it also protects all the automotive components along with serving as a base for its mounting. The ergonomics and aesthetics for the overall car is majorly designed based on the design of roll cage. Keeping all the mechanical design and safety aspects, the weight reduction in the chassis is very important to have dynamically stable vehicle with Center of gravity as low as possible. Playing such an important role the roll cage needed to be designed with variety of aspects in consideration. There has to be an optimum design with balances both weight and strength of the roll cage. The design investigated in this paper is an All-Terrain Vehicle (ATV) Roll cage based on SAE Standards. The newly introduced concept of four-wheel drive for BAJA SAE is implemented for the design of roll-cage along with the static and dynamic analysis. The detailed 3D Sketching and weldments design is also elaborated the paper.

## 2. LITERATURE REVIEW

The literature study gives insights with ATV chassis design with implementing of creating a new design without altering the actual performance of the current design using CATIA and ANSYS by Aakash t al.[1] The study gave an analysis with varying materials and giving extensive detailed results on various static analysis. The investigators have studied the flaws in the previous year's designs and have made an attempt to fix the issues in the design making it optimal in terms of weight and ergonomics. Such a Finite element analysis is reported and given a rigid structure design for the roll cage by S. Jacob et al. [2] Similar numerical investigation with strength optimization is carried by Srivastava et al. [3] with a primary design of Go-Kart roll cage. The reference from the paper is to have a basic idea of how varying material can affect the various typed of impact load on the chassis. The paper gives a strength-based analysis with three types of steel variety being tested in the investigation. Furthermore, Murthy et al. [4] studies BAJA roll cage design with single material usage and testing it on various typed of static loading conditions. The paper by Dubey et al.[5] materialistic behavior chassis with respect to mechanical strength, durability and fatigue. Some papers also coved the vibration characteristics and vibrational loads acting on the chassis and their effect on the mechanical properties by L. Dai and J. Wu. [6] The design along with the fabrication technique is studied and explained by Chawla et al. [7] The chassis investigated has optimum values in safety, comfort and low cost in terms of manufacturing. Effects of different loading conditions on roll cage members is investigated and lead to safer and lighter roll cage as compared to conventional design by Gautam et al. [8] K. Sandeep [9] studied roll cage design with material details and its response to various loading conditions. This study prioritized the driver in terms of safety and overall chassis design consideration. The final iteration of the roll cage design along with the analysis results are reported by Angadi et al. [10] giving a strong comment on durability of the chassis. S. Venkatesh [11] investigated different designs of the roll cage in consideration to the deformational behavior, maximum stress and safety factor on the roll cage. The paper suggests ASTM 181 is the best suiting material for the roll cage and this can be further used for the manufacturing. Similar study is explored by Harshit Raj [12] reporting analysis results and deformational behaviors of the chassis. A novel work with bending and compression tests along with roll cage analysis has been investigated by Garg et al. [13] Denish Mevawala [14] reported a study with all-terrain vehicle chassis defined by ANSI standards based on varying loading conditions and its effects.

## 3. OBJECTIVES

The purpose of roll cage is to maintain a limited circumferential space around the driver and to sustain the load of other subsystems such as powertrain, suspension, brakes and steering. The roll cage must be reliable enough to endure all the loads from the various sub-systems yet maintain weight effectiveness and provide effective driver safety with comfort

## 4. DESIGN CONSIDERATIONS

The roll cage is primarily designed keep driver safe during sudden impacts or un avoidable crashes. The design of the chassis is thus majorly focused on energy absorbing ability. The secondary requirement is a structure and spacing to hold all the automotive components like engine, transmission, differential, fuel tank, etc. The four-wheel drive layout is a recent development by the BAJA SAE and addition of new components in the driveline is considered while initializing the

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ISSN: 2278-0181
Vol. 11 Issue 02, February-2022

compartment spacing in the cockpit. Along with this weight reduction is very crucial along with appropriate stiffness when it comes to cornering ability and bumps impacts.

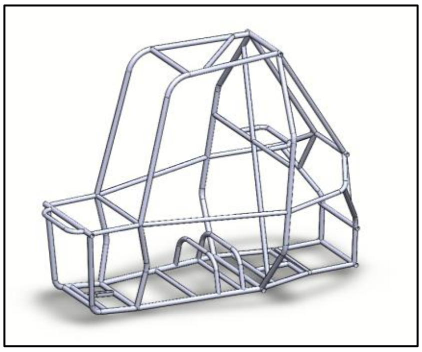

*Fig. 4.1. Roll cage design*

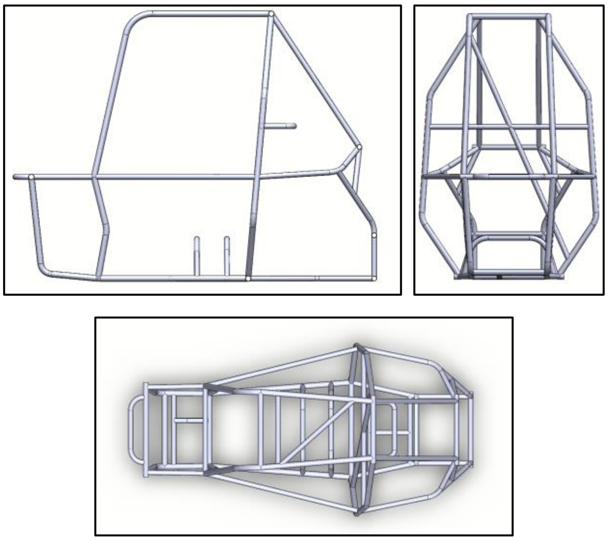

*Fig. 4.2. Roll cage from different views (side, front and top* view)

*Table 4.1. Design Parameters*

| **Parameter** | ***ALC length*** | ***RRH inclination*** | ***LDB Inclination*** | ***Distance between $S_L$ and $S_R$*** | ***FBM Inclination*** |
|---|---|---|---|---|---|
| Rule Book Threshold | ≥ 18 | ≤ 20° | ≥ 20° | ≥ 32 | ≤ 45° |
| Present design | 22.4 | 5° | 55.58° | 32.82 | 12.6° |

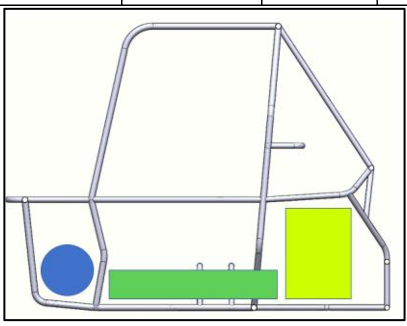

*Fig. 4.3. Additional spacing for four wheel drive chassis*

*Table 4.2*

| ***Color*** | ***Compartment*** |
|---|---|
| Blue | Differential |
| Green | Drive line |
| Yellow | Transmission and Engine |

## 5. DESIGN PARAMETERS

The design parameters are considered with respect to driver ergonomics and overall dimensions of the driver in driving position. The values are used with solid agreement with the Rule of BAJA SAE roll cage specifications and are shown in the table 3.1. The dimensions for the inclinations and lateral crosses are used to ensure optimum cockpit space, improving driver comfort and to have an appropriate envelope for powertrain and components. Besides this a wide space for driver quick egress is also considered.

### *5.1. Design Constraints*

The defining pillar of overall chassis design is the fixing track width and wheel base obtained from vehicle dynamics and steering calculations. This is the most important constraint giving a deciding factor in packaging of the vehicle. Apart from this hardpoints from suspension and spring calculations becomes the secondary constraints. This is also accompanied by transmission constraints like CVT angle, engine location, gearbox location and its inclination, drive shaft angle and differential location.

### *5.2. 3D sketching and CAD Modelling*

The cad design starts with set of hard points and geometrical constraints set by vehicle dynamics and transmission domain of the vehicle design.

A-section is then Sketched using front suspension hard points and 3D sketching as shown in figure 4.2.1. While designing the end points the additional constraints are given to fix the 3D sketch, in order to maintain the accuracy and symmetricity up to 6 decimal places of coordinate values.

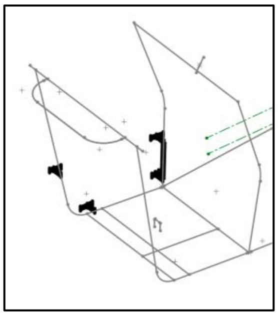

*Fig. 5.2.1. A-section 3D sketch*

The cockpit of the driver is sketched after fixing the A-section. The vertical space is kept appropriately taking consideration of drive line space and angle of the driveshaft. Furthermore, the FBM of the B-section is also sketched and iterated to vary the angle considering the thresholds from the rule book.

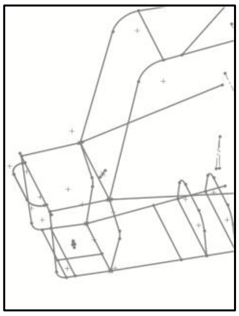

Fig 5.2.2. Cockpit 3D sketch

The rear roll hoop is then sketched with ends of Side impact members, Lower side members and rear hoop overhead and keeping all major angular and dimensional thresholds of the rule book. The 3D sketch is terminated with iterating the design of C-section. This includes multiple domains undertaken for spacing and contains like engine spacing, transmission component assembly, fuel tank and space for splash shield. Vehicle packaging for four-wheel drive requires more vertical extensions in 3D sketches as compared to conventional rear wheel drive ATV chassis. When the 3D sketches are fully defined after iterative changes in various domains of vehicle design, the full 3D sketch checked against the rulebook template to ensure satisfaction of all rules.

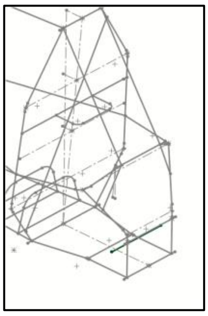

Fig. 5.2.3. C-section 3D sketch

The fully defined sketch is modelled using weldments tool with appropriate inner and outer diameters for the circular cross section pipes. The primary and secondary pipes are modelled separately and added. The weldments are then trimmed and extended to make perfect end profiles at welding areas. The additional parts of extensions are removed using delete options in the weldments tool.

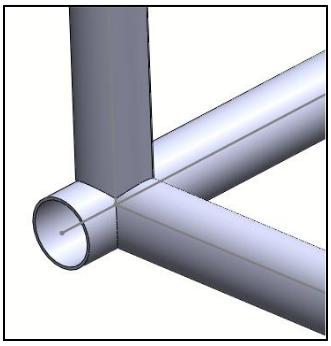

Fig. 5.2.4. Trimmed Weldment

### 5.3. Four-wheel drive

The four-wheel drive system requires higher amount of spacing below the cock pit area. Thus, there has been a novel design of an additional U-Shaped member on the Lower side members to allow drive line to be spaced and mounted below the seat keeping a safe separation between driver and the moving components of transmission. The base plate can be attached to this member from RRH to front leg space area.

### 5.4. Material selection

The material chosen for the roll cage plays a key role in stiffness and weight saving. After considering material properties, cost effectiveness and market surveying, AISI 4130 was chosen for its higher Yield Tensile strength and Ultimate tensile strength helping in making the roll cage lighter and safer without adding excessive members as in the previous design. Diameter of the tube was increased to achieve higher bending stiffness values. For both primary and secondary member of roll cage the material assigned is alloy steel containing chromium and molybdenum as strengthening agents.

Table 5.4.1 Material properties

| ***Parameters*** | ***Specification*** |
|---|---|
| Material | AISI 4130 |
| Density | 7850 Kg/m3 |
| Ultimate tensile strength | 560 MPa |
| Yield Tensile strength | 460 MPa |
| Modulus of elasticity | 210 GPa |
| Bulk modulus | 140 GPa |
| Shear modulus | 80 GPa |
| Poisson's ratio | 0.3 |

### 5.5. Pipe cross section

Due to availability and high strength-to-cost ratio a 1.65mm thickness primary member and 1mm thick secondary member was chosen to maintain sufficient bending strength making it resilient enough to bear up all of the loads. The considerations for cross sections inner and outer diameters are taken with respect to Rule book of BAJA SAE India 2021.

## 6. FINITE ELEMENT ANALYSIS

Finite Element Analysis (FEA) is a simulation technique which uses discretization of the domain into smaller divisions called as elements and calculate all the parameters for the study of each end point of a particular element known as nodes. This allows to study behavior and pathway to optimize and design a product.

### 6.1. Static Analysis

#### 6.1.1. Methodology

Roll cage after designing in Solid works is then imported to Hyper works. Solids, points and lines were deleted and only surfaces were retained to avoid unnecessary errors. Mid-surfaces of whole geometry were extracted to do 2D analysis. The uneven surfaces and joints were trimmed and extended to generate a continuous surface. After this, secondary and primary members were grouped separately. The Mid-surfaces were meshed and mesh quality was checked and failing elements were corrected. Materials were then assigned.

International Journal of Engineering Research & Technology (IJERT)
ISSN: 2278-0181
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Properties were assigned as PSHELL and thickness were assigned to primary and secondary individually. Forces were calculated taking worst case scenarios into account and Mass of car was taken 250 Kg. Load steps and load collectors were created for all the analysis intended.

#### *6.1.2. Meshing*

*<u>Mesh Type</u>*: 2D meshing (mid-surface) This method is taken because the thickness is very less compared to the diameter and length of members.

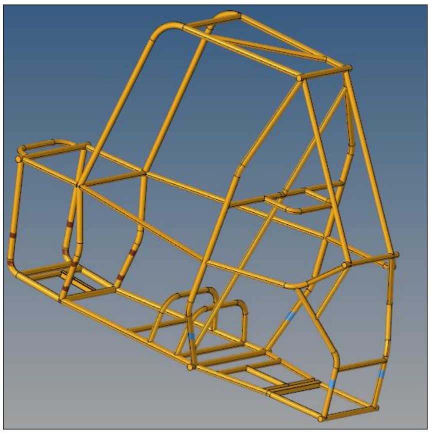

*Fig. 6.1. Meshed mid surface of the roll cage pipes*

*<u>Element Type:</u>* Mixed (quad and Trias) taken so that the profile is perfectly captured and the stress flow is maintained smoothly. Less Trias are used because Trias introduces increased stiffness and hence accuracy is affected.

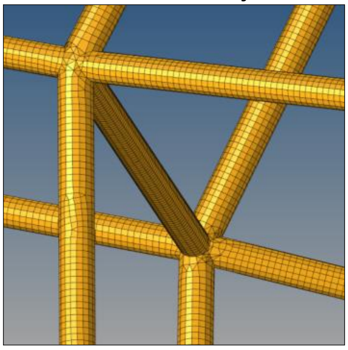

*Fig. 6.2. Quad and tria mesh*

*<u>Mesh Size:</u>* 5mm (to match with industry standards) and further decreasing of element size did not provide considerable change in results.

#### *6.1.3. Quality Check Parameters and Values*

Element quality was checked as per the following parameters. These were chosen to get best quality elements providing the most precise and highly accurate real-life results.

*Table 6.1. Quality parameters*

| ***Quality Parameter*** | ***Value*** |
|---|---|
| Warpage | 5 |
| Aspect ratio | 5 |
| Skew angle | 50 ° |
| Jacobian | 0.05 |

### *6.2. Simulation and modelling results*

#### *6.2.1. Front Impact Test*

The frontal impact simulation is carried with applied force of 9323N applied on the frontal hitch member and constraining all the bracket points. The force is applied parallel to the motion of the vehicle giving maximum amount of force component to act on the chassis.

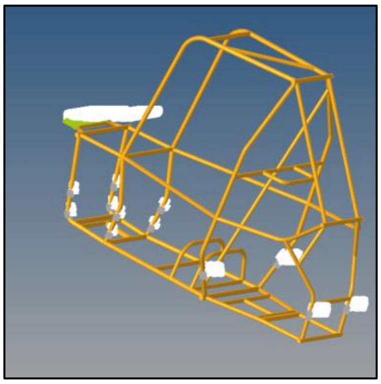

*Fig. 6.1 Constraints*

The maximum displacement is found to be 0.677 mm with maximum von mises stress of 97.6 MPa. The factor of safety for this test is reported to be 4.71 giving a good agreement with respect to strength of the roll cage for front crash condition.

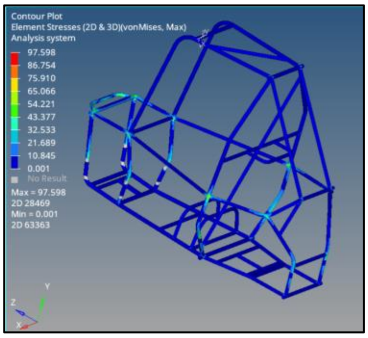


*Fig. 6.2. Von Mises Stress Contour*

#### *6.2.2. Rear Impact Test*

The front suspension hard points are constrained and rear C-section members are given distributed force of 4666 N with displacement found to be 0.2 mm and maximum von mises stress of 35.4 MPa.

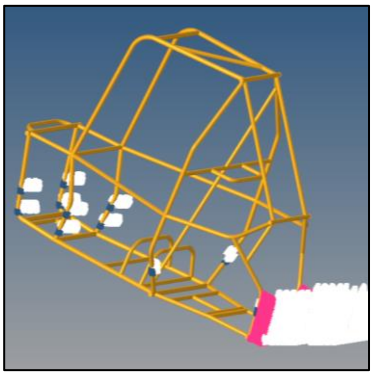

*Fig. 6.3. Constraints*

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The factor of safety for rear impact is 12.97 which states that the rear design can handle enormous amount of force at rear crash conditions.

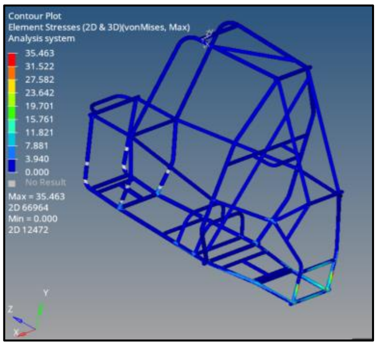


*Fig. 6.4. Von Mises Stress Contour*

### *6.2.3. Side Impact test*

All the suspension hard points are constrained Side Impact members are given distributed force of 4666.6 N with displacement found to be 0.345 mm and maximum von mises stress of 75.1 MPa.

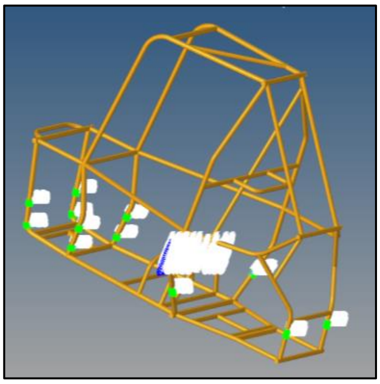

*Fig. 6.5. Constraints*

The factor of safety for rear impact is 6.1 giving a good safety with side crash conditions.

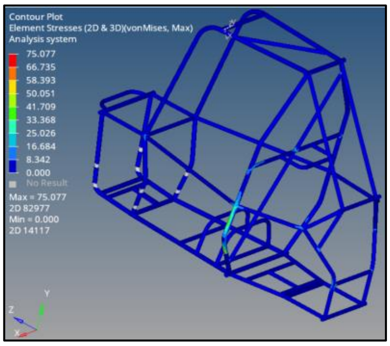


*Fig. 6.6. Von Mises Stress Contour*

### *6.2.4. Torsional test*

The rear suspension hard points are constrained and front suspension hardpoints are givens are given distributed force of 1125 N in coupled manner. The displacement found to be 0.529 mm and maximum von mises stress of 43.5 MPa. This test gives an insight of the ability of chassis to handle loads in twisting and torque conditions.

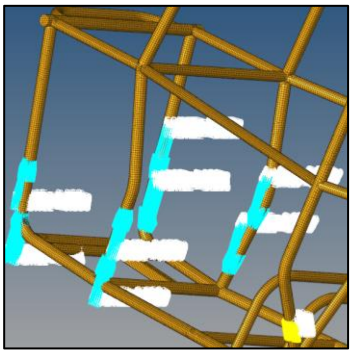

*Fig. 6.7. Constraints*

The factor of safety for rear impact is 10.57. The torsional effect can be caused when on side of the tire gets sudden impact generating an equal and opposite side force on the other tire generating a couple.

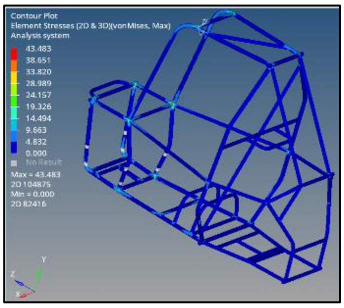


*Fig. 6.8. Von Mises Stress Contour*

### *6.2.5. Rear Bump Impact test*

The front suspension hard points are constrained and rear suspension hardpoints are given distributed force of 2566 N with displacement found to be 5.2 mm and maximum von mises stress of 229.6 MPa.

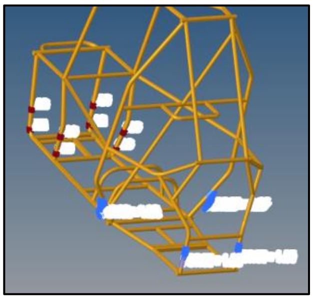

*Fig. 6.9. Constraints*

The factor of safety for rear impact is 2.0 stating a safer design in case of sudden bump on the rear side of the roll cage.

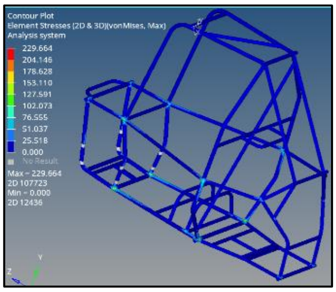


Fig. 6.10. Von Mises Stress Contour

### *6.2.6. Front Bump Impact test*

The rear suspension hard points are constrained and front suspension hardpoints are given distributed force of 2100 N with displacement found to be 3.01 mm and maximum von mises stress of 116.9 MPa.

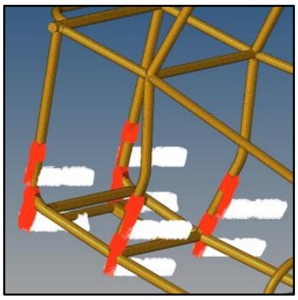

Fig. 6.11. Constraints

The factor of safety for rear impact is 3.9 stating a safer design in case of sudden bump on the front side of the roll cage.

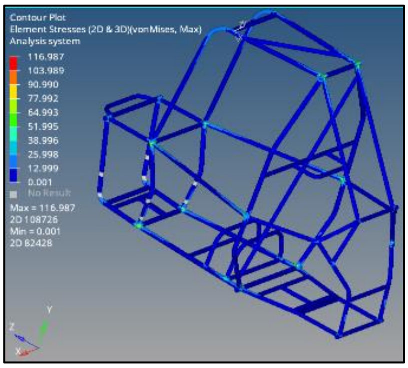


*Fig. 6.12. Von Mises Stress Contour*

### *6.2.7. Roll over test*

All the suspension hard points are constrained and upper bend of the front bracing member is given distributed force of 4516 N with displacement found to be 1.02 mm and maximum von mises stress of 130.6 MPa.

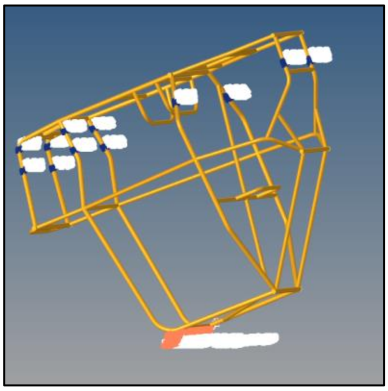

*Fig. 6.13. Constraints*

The factor of safety for rear impact is 3.5. This test gives an idea condition for chassis design in case of vehicle getting an impact after a roll over.

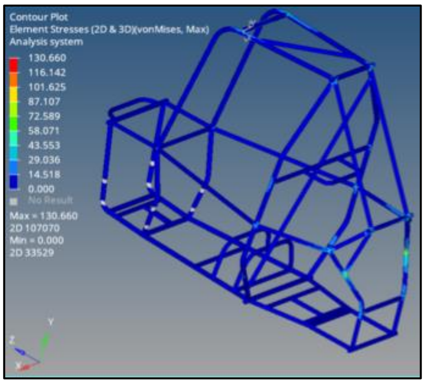


*Fig. 6.14. Von Mises Stress Contour*

### *6.2.8. Drop test*

The front suspension hard points are constrained and rear C-section members are given distributed force of 9450 N with displacement found to be 0.606 mm and maximum von mises stress of 71.7 MPa.

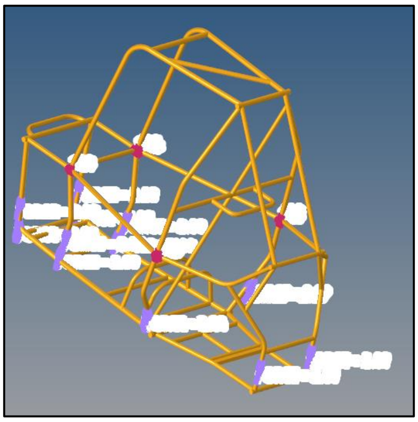

*Fig. 6.15. Constraints*

The factor of safety for rear impact is 6.4 This test is to study the effect of forces at suspension points when the vehicle falls from a height of 2 meters and lands straight on wheels.

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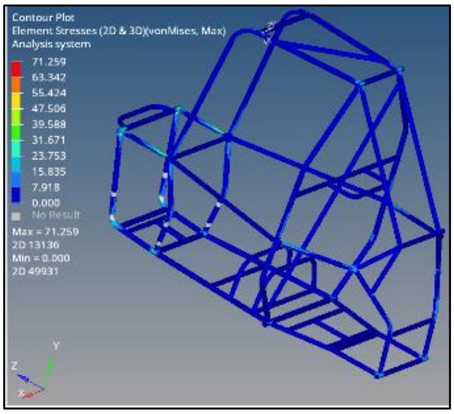


*Fig. 6.16. Von Mises Stress Contour*

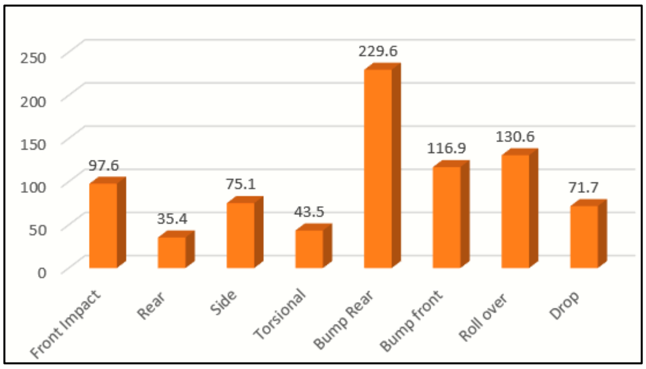


*Chart 5.1. Von Mises stress (MPa)*

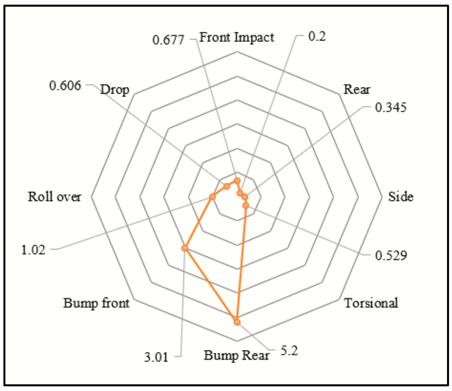


*Chart 5.2. Maximum displacement (mm)*

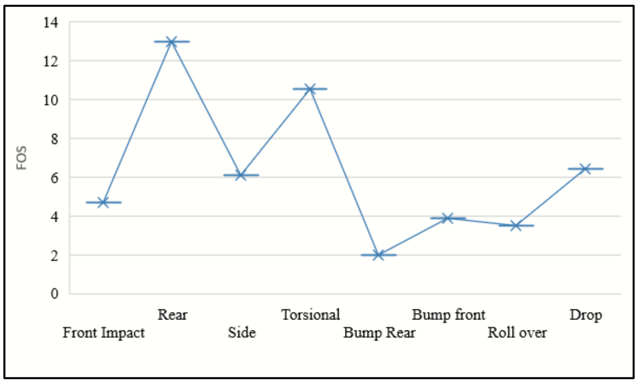


*Chart 5.3. Factor of safety*

## *6.3. Dynamic Analysis*

Dynamic crash analysis is used to study a real time simulation of the vehicle crashing against a wall, tree, or a solid barrier at the maximum speed.

### *6.3.1. Methodology*

Mid-surface was meshed for dynamic analysis as explained previously. Static element quality check parameters also maintained here to generate high quality mesh and get most accurate results. Material and properties were assigned. Interfaces and contacts were generated in the Hyper works. Maximum Theoretical speed of the car was set to 16.66 m/s. Rigid wall was created at the front to simulate dynamic front crash scenario and Output blocks were assigned to keep track of solver and processing.

### *6.3.2. Meshing*

Mesh Type: 2D meshing(mid-surface)
Element Types: Mixed (quad and Trias)
Mesh Size: 5mm
Order: First Order

### *6.3.3. Rigid Wall*

An infinite rigid wall is created against which the collision takes place. The wall was made tough enough to not allow the roll cage to pass through.

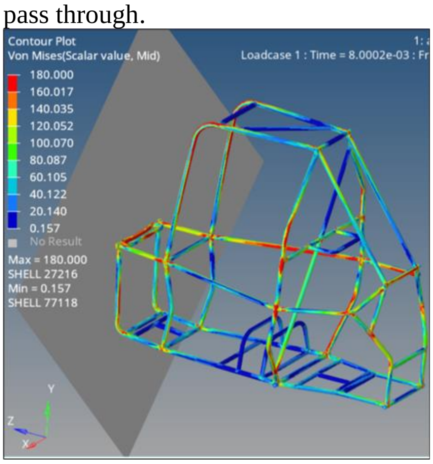


*Fig. 7.1. Dynamic Crash analysis contour*

The maximum von mises stress obtained was 180 MPa with maximum displacement of 13 mm. the plastic train reported to be 1.2 and Factor of safety of 2.5. The impact time was 0.009 seconds given for the whole dynamic crash test.

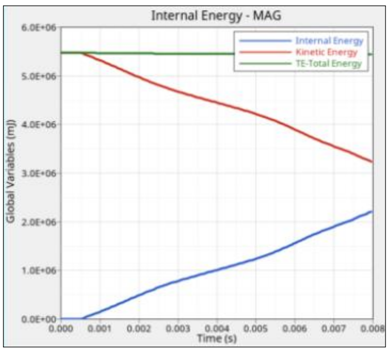


*Fig. 7.2. Internal Energy plot*

## 7. CONSLUSION

A novel design to implement four-wheel drive in the BAJA ATV is investigated. The study reveals behavior and effect of different types of loads acting on the vehicle chassis. The roll cage is strong enough to withstand rollover, torsional, bump, front, side & rear impacts. Furthermore, the chassis also can withstand dynamic crash situation. There is an additional space below the driver cockpit to provide safe and appropriate spacing for the driveline to be placed. Driver's foot is safe from the crumble in the nose section of the roll cage. Driver head and shoulders are safe in case of roll over or side impacts. Powertrain and electrical components are safe as the external loads are taken by the roll cage with minimum amount of structural displacement possible.

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