

# Simulation of an Off-Road Vehicle Roll Cage: A Static Analysis

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## ABSTRACT

The Baja Collegiate Design Series is a competition run annually under the purview of SAEINDIA for undergraduate students across India. Team Rudraksh from University of Petroleum and Energy Studies, India, has accepted the challenge of competing in the 2015 BAJA event. An aspect of this competition is to analyze your roll cage under various load cases.

## Keywords

failure modes, roll cage, deformation, von-mises stress, Hyper works, BAJA

## 1. INTRODUCTION

The purpose of designing and manufacturing a BAJA car is to create a prototype recreational off-road vehicle that provides a fun, safe and reliable experience. There are many factors to an off-road vehicle, such as the chassis, suspension, steering, drive train and braking, all of which require thorough design considerations. During the entire design, special emphasis was given to the roll cage and its strength so as to safeguard the driver. The roll cage was analyzed under various load cases so as to prevent mishap during event.

## 2. FRAME DESIGN

The goals for the chassis design include:

1. Making a compact design.
2. Reducing the wheel base.
3. Decrease weight and overall length.
4. Increase the ground clearance.
5. Increase driver comfort.
6. Increase packaging for subsystems.

The complete vehicle was modeled and assembled in Solid Works 2013. This was used to ensure the proper fittings and assembly of the Baja car. The Finite Element Analysis (FEA) was done using Hyper works v12 student edition for all the structural parts of the vehicle.

## 3. DESIGN CONSIDERATIONS

Primary members have been selected of 25.4 mm OD and 19.4 mm ID. The frame components include RRH, RHO, FBM, FLC, LBD, LFS, SIM, FAB, USM and the lateral cross members (LC). Clearance guidelines dictate a minimum of 6” vertical distance from the driver's head to the bottom of the

RHO and 3” clearance between the rest of the body and the vehicle envelope. Bends were increased as much as possible in the roll cage design to reduce the number of welds needed for that. The bending members include the Rear roll hoop (RRH) which consists of 6 bends, and Front Bracing Members (FBM).

## 4. MATERIAL SELECTION

The material selection plays a very important role in design of roll cage. The factors considered were strength, weight and cost. We shortlisted four materials for the chassis, 1020 DOM, 1018 steel, 4130 Chromoly, and DIN 2391, and compared their strength, stiffness, weight and cost, as depicted in Table 1. The 4130 Chromoly was found to be expensive as compared to other materials. The challenge was to strike a trade-off between all of these parameters. After an extensive research **AISI1018** was selected for its good strength, weld ability, low cost and overall reduction in the weight of the roll cage. Primary members have been TIG welded to increase the strength of the roll cage.

**Table 1. Comparison of various Tubing**

Material	1018	1020	4130 Chromoly
Yield Strength	365 MPa	350MPa	435MPa
Bending stiffness	1572.3 Nm <sup>2</sup>	1572.3 Nm <sup>2</sup>	1572.3 Nm <sup>2</sup>
Bending Strength	223.96 Nm	214.76 Nm	266.92 Nm
Weight	1.12	0.82	0.82
Cost	\$0.5	\$1.65	\$4.55

## 5. STATIC TESTS

Following tests were performed on the roll cage -

- Front Impact
- Rollover
- Torsion

These tests are used in automotive industry to validate the design and the values used for crash time are based on automotive industry standards (AIS).All the tests performed

are Static analysis although the events are dynamic in nature. To incorporate it, factor of safety is provided. 1D meshing is done on Altair Hypermesh software and optistruct solver is used. 1D meshing will provide almost the identical result as 3d meshing since 1 dimension (length of tube) is large in comparison to the other 2 dimensions (i.e. diameter and thickness of tube).

### 5.1 Front Impact Test

Assuming  $m_1=m_2=350\text{kg}$  and  $u_1=50\text{kmph}$  and  $u_2=0$  (vehicle at rest),

$$W_{net} = \frac{1}{2}mv^2(\text{final}) - \frac{1}{2}mv^2(\text{initial})$$

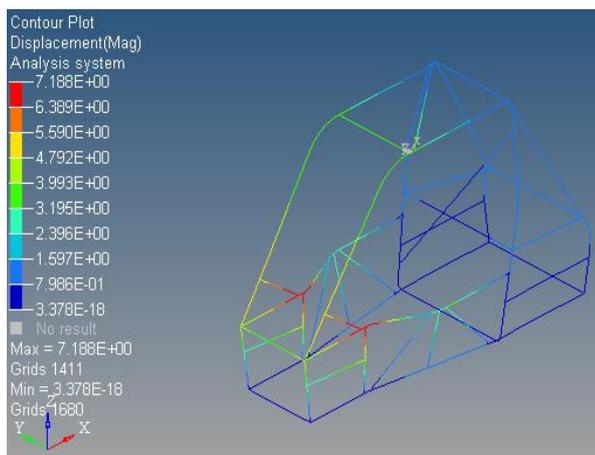
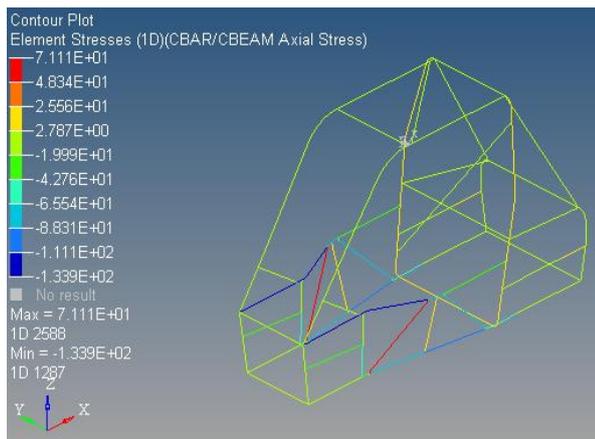
$$W_{net} = \text{force} * \text{displacement}$$

$$f * d = -\frac{1}{2}mv^2$$

It is considered for the static analysis that the vehicle comes to rest 0.1 sec after the impact. For 13.89 mps (about 50 km/hr) speed the travel of the vehicle after the impact is 1.389 m (displacement = velocity \* time)

Hence, a frontal impact force of 24307.5N (about 7g) was applied on the frame. Same force can also be obtained from Newton's second law which gives impact force in terms of momentum change.

The deformation and stresses are shown below. For a maximum stress of 71.11MPa, the FOS obtained was 5.13. Maximum deformation obtained was 7mm at load of 7g.



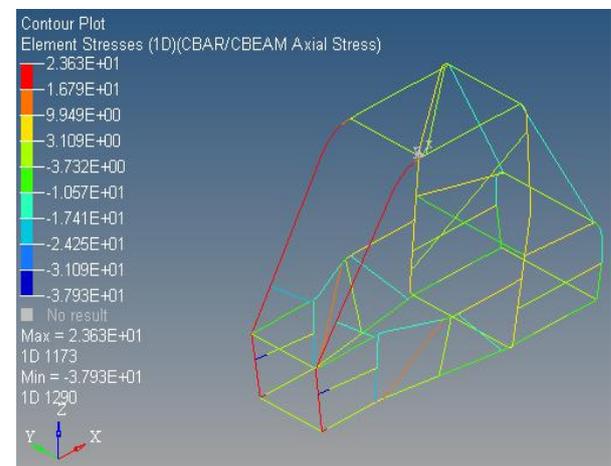
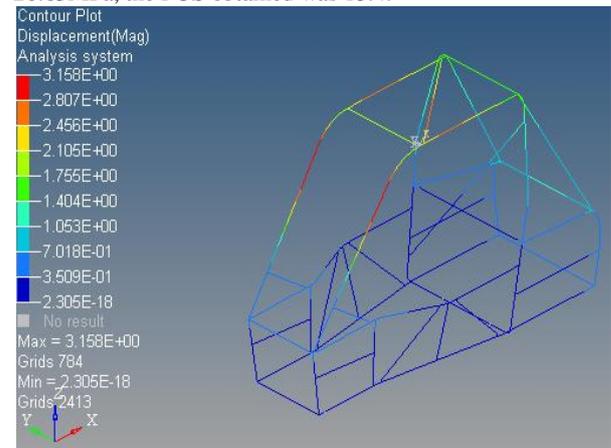
**Fig 1: Stress and Displacement for Frontal Impact**

### 5.2 Rollover

For roll over test, a force equivalent to the 3 times the gross weight of the vehicle (3500N) was applied at the top projection of the frame at angle of 45 degrees while

constraining all the suspension points. The calculations were made for the worst case scenario.

Deformation and stresses were as follows. For a stress of 26.63MPa, the FOS obtained was 15.4.

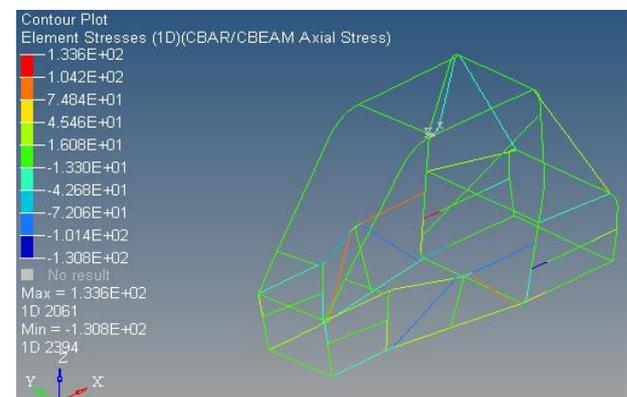


**Fig 2: Stress and Displacement for Rollover Impact**

### 5.3 Torsion

For torsion test, a force equivalent to 3 times the gross weight of the vehicle (3500N) was applied on both the front suspension points in opposite directions while constraining the rear 2.

Deformation and stresses were as follows. For a stress of 133.6MPa, the FOS obtained was 2.73.



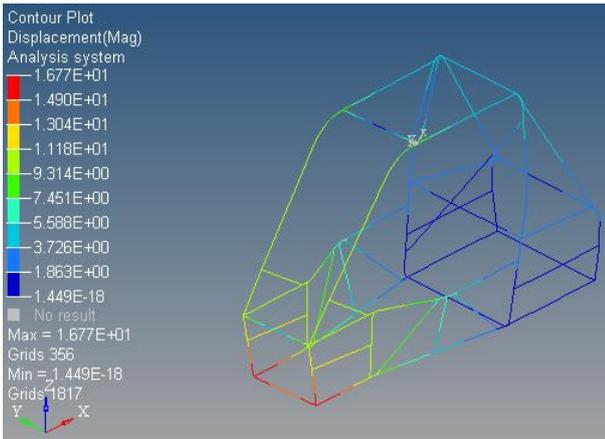


Fig 3: Stress and Displacement for Torsion Impact

## 6. REFERENCES

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