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ROLL-CAGE FRAME DESIGN FOR AN ALL-TERRAIN VEHICLE

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Abstract: *All terrain vehicles (ATV) are designed to run and perform on any surface irrespective of the ground conditions. The design structure of an ATV solely depends upon it roll-cage. As an ATV is an experimental vehicle the design procedure of its roll-cage plays a critical role to give maximum possible strength while keeping the weight minimum. Roll cage of an ATV also absorbs various impacts during an event or in test runs. To verify the design finite element analysis can be done for static loads with defined impact times and modal analysis for calculating natural frequencies of different modes. This paper will cover design analysis and procedure of roll-cage frame to ensure the vehicles safety in all possible conditions.*

1. Introduction

All-terrain vehicles are popularly known for their versatile performance over both road and mountain terrain (including loose gravel) were around early 70s in USA and since then with each preceding year there have been significant changes in its designs. A competition organized by the Society of Automotive Engineers (SAE), SAE-BAJA indulges students to design, analyse and manufacture their own all-terrain vehicle. The speeds of these vehicles vary with the specified engine displacement and by applied transmission mechanism. As the roll-cage models are iterated in the design software it is also very important to check all failure modes for each successful iteration or the final model to ensure the durability of the body members and for the safety of the drivers. As per the analysis of the impacts done by Aru et al., 2014 [1], Noorbhasha 2010 [2] and Raina et al 2015 [3] of the roll-cage predicting the every failure modes. Impacts test validation of the static analysis results were done using varying mesh grid size. This research will further discuss about the static analysis of every possible case at competition site. For the static analysis the values of the forces which are going to be applied are calculated theoretically using the work-energy theorem and by assuming the impact times. To validate the static analysis the dynamic impact analysis is also being done with calculated forces and the on-site possible cases. By that we can confirm the safety of our design and manufacture it.

2. Dimensions

The design of the roll cage in any condition must always ensure enough clearances from members for safety and free movement of the driver. In order to reduce the weight members with diameters were used with different cross sections area to construct the roll-cage. The diameter of the primary member is 29.2 (mm) with thickness of 1.65 (mm) and that of the secondary members is 25 (mm) with thickness of 1.00 (mm). The cross-section of the members is circular as it would bear more strength than beam section with ability to withstand torsional stresses also.

3. Material Properties

AISI-4130 steel pipes were used as material for the roll-cage fabrication. It was selected because of its high tensile and yield strength. In addition to these its weld ability risks are low. We also calculated forces on different materials like AISI-1020, AISI-1050 and AISI-4120 but found it best for the AISI-4130 and to validate it we used pipes of same cross-section as primary member defined in BAJA-Rulebook [4] to verify the materials feasibility.

The vehicle dimension and material specification are given in table 1 and table 2 respectively also the composition of the selected material is given in table 3.

Table 1- Roll cage vehicle and Dimensions

Attributes	Values
Length x Width x Height	1831.9 mm x793.7 mm x1276.3 mm
Weight of the vehicle (including the driver)	220 kg
Weight of the roll-cage	19.5 kg

Wheelbase	1345.9 mm
Track-width	F- 1219.2 mm R- 1117.6 mm
Ground Clearance	304.8 mm

Table 2- Material Properties

S. no	Property	value
1.	Composition	Fe, Cr, Mn, C, Si, Mo, S & P
2.	Density(g/cm^3)	7.8
3.	Yield strength, σ_{yt} (MPa)	620
4.	Ultimate tensile, σ_{ut} strength (MPa)	750
5.	Modulus of elasticity (GPa)	190

Table 3- Material Composition

S. no	Element	% weight
1	Iron (Fe)	98
2	Chromium (Cr)	0.855
3	Manganese (Mn)	0.52
4	Carbon (C)	0.30
5	Silicon (Si)	0.15
6	Molybdenum (Mo)	0.15
7	Sulphur(S)	0.025

4. 2D and 3D Views of Roll cage

Figure 1 - 3D view of Vehicle

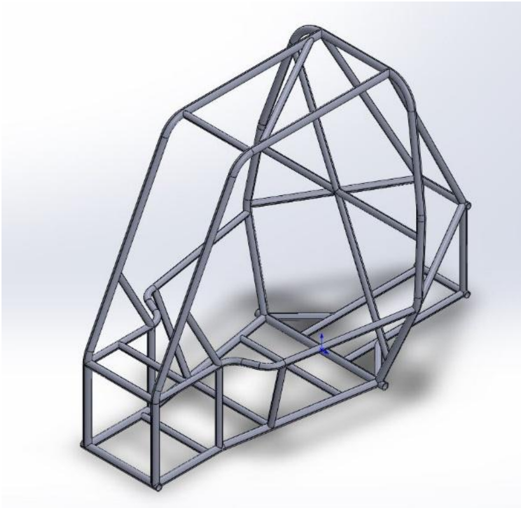

Figure 2- 3D view of Roll-cage Frame

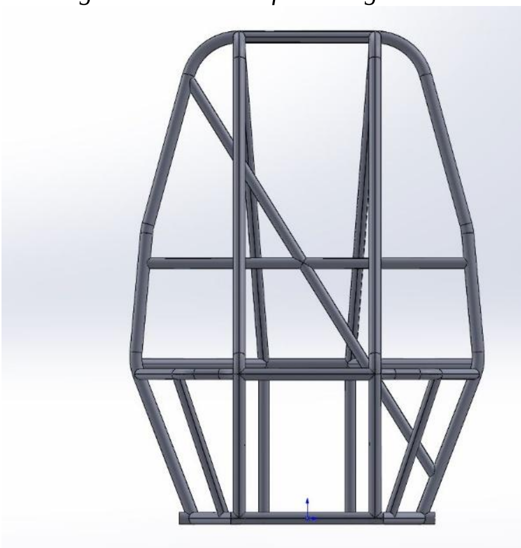

Figure 3 – Front View of Roll-cage Frame

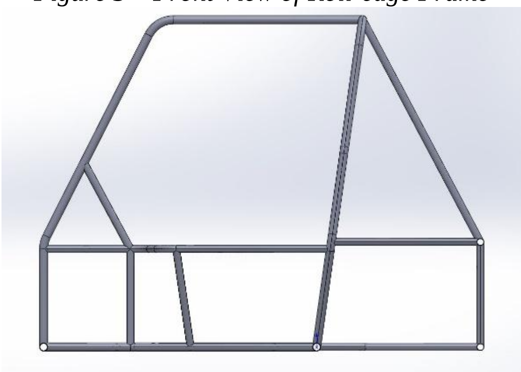

Figure 4 – Side View of Roll-cage Frame

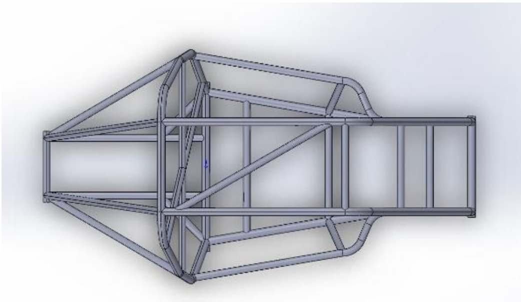

Figure 5 – Top View of Roll-cage Frame

5. Finite Element Analysis

Using solid modelling in PTC Creo 4.0 and Finite Element Analysis (FEA) in ANSYS R18.2 the roll-cage that is designed and optimized to achieve maximum strength and minimum weight while carrying the safety factors for the driver. The static analysis of the roll-cage for the impacts like front, rear, side and roll over were done using static structural workbench and for dynamic analysis of the same was accomplished with explicit dynamic workbench in which the head-on collision was also done.

5.1 Mesh Size Selection

Mesh size is one of the important factors that are to be considered while doing FEA as the mesh size defines the size of the smallest uniform element which is to be analysed and therefore plays a critical role in any analysis procedure**.** In the analysis our roll cage the optimum size of mesh was found out to be 4mm by trial and error method because after 4mm mesh size didn't affect the result as it was doing from 15mm to 3mm mesh size.

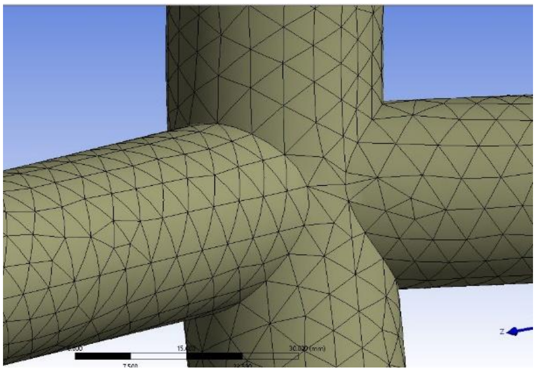

Figure 6 – Roll-cage Meshing (4mm)

5.2 Static Analysis

In static analysis vehicle is to be considered in static state and maximum possible forces are to be applied to roll cage body with constraints as per different conditions. Static analysis is done in ANSYS Static Structural solver module. As per points noted in studies [1] impact time of roll cage body in cases of impact with rigid bodies (wall, floor etc.) is taken as 0.13 sec, while in case of impact with a deformable object (vehicle, flexible walls, etc) impact time is taken as 0.30 sec. All the captured images of deformation and stresses caused by the impact are also shown with respective force analysis and analysis conditions are also defined.

5.3 Modal Analysis

Modal analysis is done to calculate natural frequencies of the first 6 vibrational modes of body that are calculated under few constraints and are analysed in ANSYS Modal Analysis solver module. Instead of forces only constraints are defined which would be impacted and damped.

6. Impact Force Calculation & Analysis for Static Impact Case

6.1 Front Impact

During front impact, the vehicle may hit another vehicle or a wall. Time of impact is greater for deformable bodies as compare to that of non-deformable (or rigid) bodies so impact force in the case of wall will be more than that in case of another vehicle. Impact time in case of wall is taken as 0.13 seconds as per earlier studies [1]. For analysis, vehicle is in a static state and forces corresponding to the maximum velocity of 60 km/h with impact time 0.13 seconds is applied to front part of the roll cage keeping rear suspension members fixed constraints. The analysis conditions and results are shown in Fig (7-9).

Calculations:

Overall Weight of the ATV, M =220 kg

Initial velocity before impact, $v_{initial}$ = 16.67 m/s

Final velocity after impact, v_{final} = 0 m/s, as vehicle would be halted

Impact time = 0.13 seconds

Work done = change in kinetic energies

As per earlier studies [3]

From work energy principal,

Work done = change in kinetic energies

$W = (0.5 \times M \times v_{final}^2 - 0.5 \times M \times v_{initial}^2)$

$|W| = |-0.5 \times M \times (V_{initial})^2|$

$= |-0.5 \times 220 \times 16.672 \times 16.672|$

$= 30575.11$ Nm

Now,
Work done = force × displacement= F × S

$S = \text{impact time} \times V_{maximum}$

$= 0.13 \times 16.67$

$= 2.1671$ m

So, from (1) we get,

$F = W/s$

$= 30575.114/2.1671$

$= 14028$ N (6.5G's)

1G = 220 × 9.81 = 2158.2 N

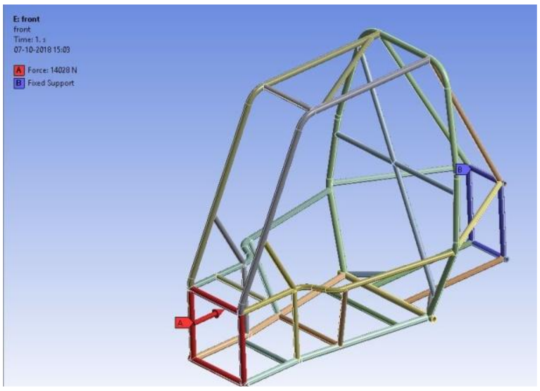

Figure 7 – Analysis Conditions for front impact

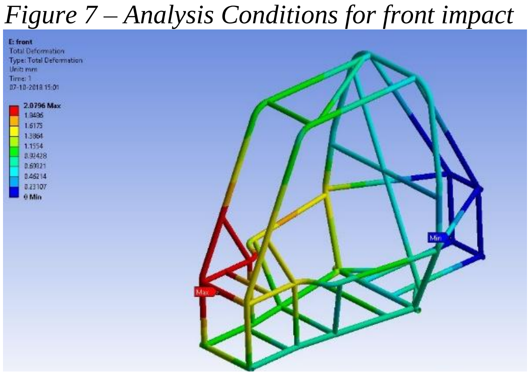

Figure 8 – Deformation due to front Impact

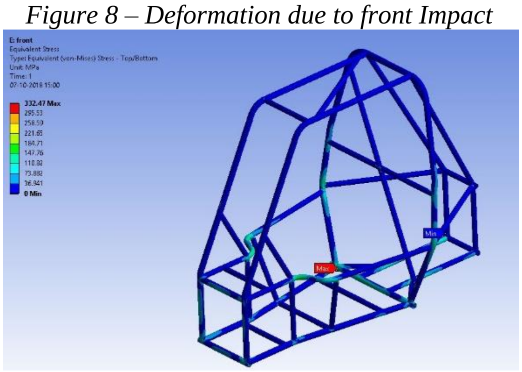

Figure 9 – Stresses due to front Impact

6.2 Rear Impact

In real conditions during rear impact, another vehicle is going to hit vehicle on its rear part. As the vehicle is a deformable body so the impact time would be taken as 0.30 seconds. For analysis, vehicle is in static state and force corresponding to maximum velocity of 60 km/h with impact time 0.30 seconds are applied to rear part of the roll cage of vehicle keeping front suspension members constrained and fixed. The analysis conditions and results are shown in Fig (10-12).

Calculations:

From work energy principal,

Work done = change in kinetic energies

$$W = (0.5 \times M \times v_{final}^{2} - 0.5 \times M \times v_{initial}^{2})$$

$$|W| = |- 0.5 \times M \times (V_{initial})^{2}|$$

$$= |- 0.5 \times 220 \times 16.672 \times 16.672|$$

$$= 30575.11 \text{ Nm}$$

Now,

Work done = force × displacement= F × s

S = impact time × $V_{maximum}$

$= 0.3 \times 16.67$

$= 5.001$ m

So, from (1) we get,

$F = W/ s = 30575.11 / 5.001$

$= 6112.3 \approx 6500$ N (3G's)

6.3 Side Impact

In real conditions during rear impact, another vehicle is going to hit vehicle on its side members. As the vehicle is a deformable body so the impact time would be taken as 0.30 seconds. For analysis, vehicle is in static state and force corresponding to maximum velocity of 60 km/h with impact time 0.30 seconds are applied to rear part of the roll cage of vehicle keeping front and rear suspension members constrained and fixed. The Side impact analysis is only done for one side and other side would be having same results. The analysis conditions and results are shown in Fig (13-15).

Calculations:

From work energy principal,

Work done = change in kinetic energies

$$W = (0.5 \times M \times v_{final}^{2} - 0.5 \times M \times v_{initial}^{2})$$

$$|W| = |- 0.5 \times M \times (V_{initial})^{2}|$$

$$= |- 0.5 \times 220 \times 16.672 \times 16.672|$$

$$= 30575.11 \text{ Nm}$$

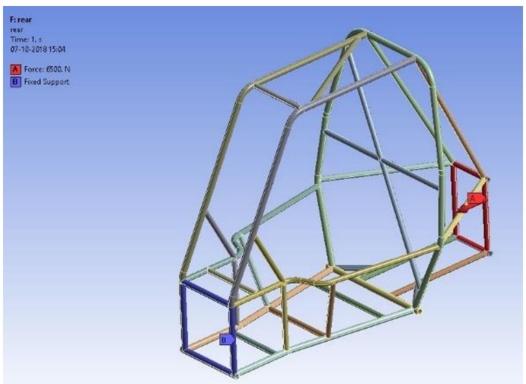

Figure 10 – Analysis Conditions for rear Impact

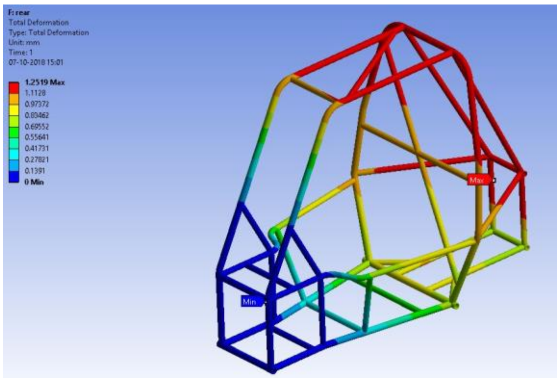

Figure 11 – Deformation due to Rear Impact

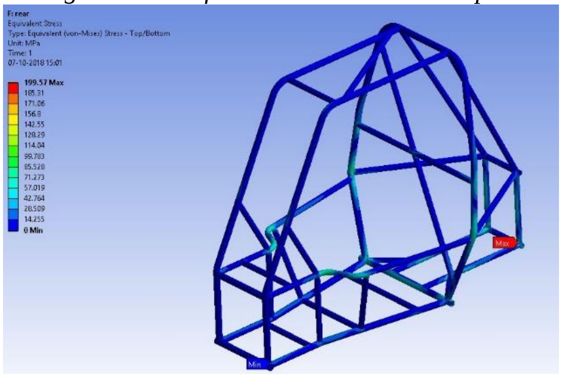

Figure 12 – Stresses due to Rear Impact

Now,

Work done = force × displacement= F × s

S = impact time × $V_{maximum}$

= 0.3 ×16.67

= 5.001 m

So, from (1) we get,

F = W/ s = 30575.11 / 5.001

= 6112.3 ≈ 6500 N (3G's)

6.4 Roll-Over Impact

In roll over impact, vehicle is dropped on its roof on road or ground from a height of 10 feet. A 10 feet (or 3.048 m) drop height is selected because it is sufficiently greater than any other drops that could occur at the event site. Since road and ground are non-deformable bodies, the impact time is taken as 0.13 seconds, and for analysis, vehicle is in a static state and force corresponding to the calculated velocity of 27.83 km/h for 10 feet with impact time of 0.13 seconds is applied to top of the roll cage of vehicle keeping bottom members constrained and fixed. The analysis conditions and results are shown in Fig (16-18).

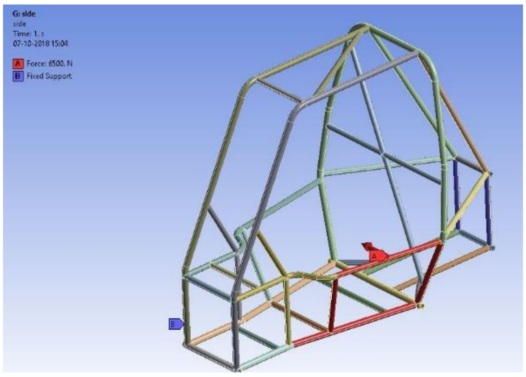

Figure 13 – Analysis Conditions for side impact

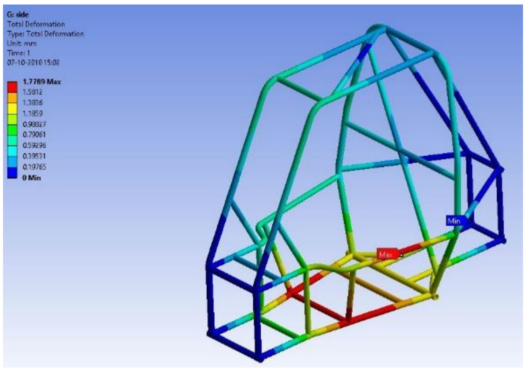

Figure 14 – Deformation due to Side Impact

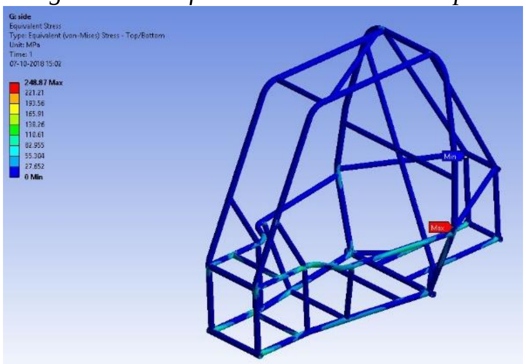

Figure 15 – Stresses due to Side Impact

Calculations:

Impact time = 0.13 s

During the fall, the whole potential changes into kinetic energy,

$$\mathbf{M \times g \times h = 0.5 \times M \times V^2}$$

$$\mathbf{= \sqrt{(2 \times g \times h)}}$$

$$= \sqrt{(2 \times 9.81 \times 3.048)}$$

$$= 7.73 \text{ m/sec (or 27.83 km/h)}$$

Now from work energy theorem

Work done = change in kinetic energies

$$|W| = |-0.5 \times M \times v_{initial}^2|$$

$$= |-0.5 \times 220 \times 7.732 \times 7.732|$$

$$= 6576.22 \text{ Nm}$$

Now,

Work done = force × displacement= F × s

$$S = \text{impact time} \times V_{maximum}$$

$$= 0.13 \times 7.732$$

$$= 1.005 \text{ m}$$

So, from (1) we get,

$$F = W/s = 6576.22/1.005$$

$$= 6543.28 \text{ N} \approx 6500 \text{ N (3G's)}$$

7. Modal Analysis

In modal analysis free natural vibration frequencies of roll cage frame was found out for the first 6 modes keeping the suspension members and the attached member of the mounts constrained. All the modes that were found out were rigid body mode. As the reciprocating combustion engine runs at vibrations ranging from (approximately)15-30 Hz which if are near the natural frequency of modes could cause serious damage to the roll cage frame as resonance phenomenon would tend to occur. Vibrations can also be induced due to the bumps or road but as their intensity would be low it is not considered. Although the bumps and vibration due to reciprocating engine could add up to a greater value but it would be less than the calculated frequency of first mode of vibration which is 43.5 Hz and graph of all is shown from figure (19-20).

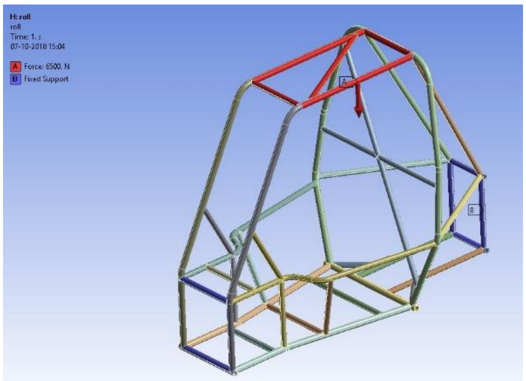

Figure 16 – Analysis Conditions for roll-over impact

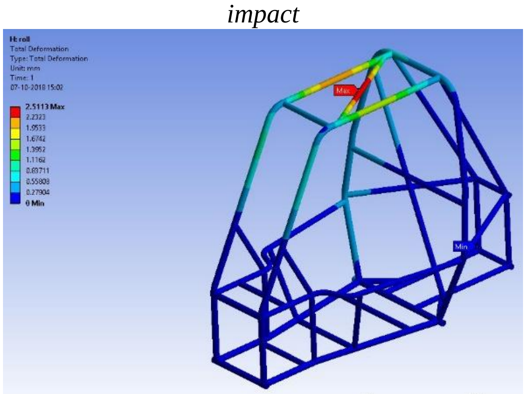

Figure 17 – Deformation due to roll-over Impact

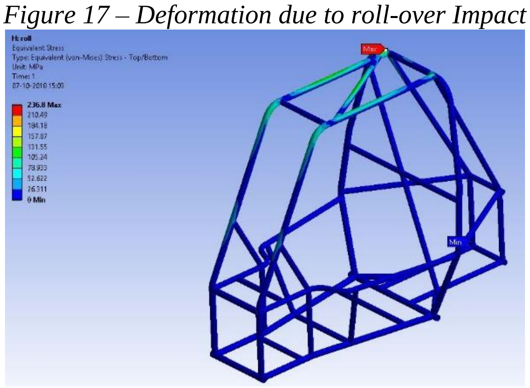

Figure 18 – Stresses due to roll-over Impact

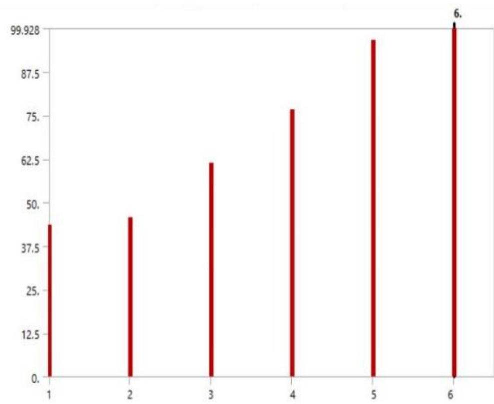

Figure 19 – Frequencies of Modes of Vibrations

Table 4 Frequency of modes tabular form

Mode	Frequency [Hz]
1	43.594
2	45.541
3	61.437
4	76.743
5	96.568
6	99.928

8. Summary of results

Table 5 Result of Analysis

	Force applied (N)	Maximum stress (MPa)	Maximum Deformation (mm)	Factor of safety
Front	14028	332.47	2.0796	1.86
Rear	6500	199.57	1.2519	3.10
Side	6500	248.87	1.7789	2.49
Roll over	6500	236.8	2.5113	2.61

9. Conclusion

The study helps to understand the concepts of static analysis, modal analysis and mesh size selection in finite element analysis. The main objective was to obtain optimum factor of safety in all impact cases which has been achieved. During the study roll cage frame was analysed & optimized for many iterations and optimum factor of safety was obtained in all possible impact cases at event site in static which ensures that the roll cage of ATV will be safe in all conditions. The min and maximum frequencies of which will be faced are 15-30 Hz and as the first mode frequency if 43.594 Hz the vehicle is safe for the operations.

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