Static and Modal Analysis of All Terrian Vehicle Roll-Cage

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***Abstract*—This paper studies the structural analysis of All terrain vehicle (ATV) roll-cage for all possible cases at the event site. The three-dimensional solid model of a roll cage is created using Solidworks 2018 and static structural analysis of the roll cage is done using Ansys 18.1 for different collisions like front, rear, side, roll over, torsional, bump and drop. Also, modal analysis of roll cage is carried out in order to avoid resonance at harsh conditions. The main objective of the analysis is to obtain optimum factory of safety which ensures that the roll cage of ATV will be safe in all conditions.**

Keywords—Roll-cage, Static analysis, Modal analysis, Baja, All terrian vehicle,

I. INTRODUCTION

The roll-cage of an all-terrain vehicle (ATV) is a skeleton structure which not only protect the driver also support systems such as suspension, steering, and the powertrain. The design of the roll-cage is crucial aspect on which the success of the ATV is dependent, if the roll-cage fails that puts the driver at tremendous risk. Therefore, roll-cage of an all-terrain is design by considering the factors such as safety of the driver, easy of manufacturing, durability, compactness, light weight and ergonomics design. The roll cage is made of thin seamless pipe which are welded together to make rigid structure which can withstands vehicle flips upside downside and harsh conditions at event site. The roll cage is design under the guidelines of SAE (Society of Automotive Engineers) Baja rulebook ensuring that the chassis must also be resilient enough to endure all the loads imposed upon it still maintaining a lightweight.

One of the major design considerations while designing the roll-cage was driver ergonomics. Ergonomics is a study of how-to layout and design the driver control in order to increase the efficiency and comfort in the working environment. ATV cockpit was designed in such a way that it carries a person 190 cm (75 inches) tall weighing 113 kg (250 lbs.) [1]. Some of the major factors that were taken into account were the seat location and inclination, location of the steering wheel, design of the foot box area so that the driver will be able to properly operate the vehicle in all driving scenarios. To enable for the proper movement of the driver's feet to control the acceleration and brake pedal, the footbox was designed as small as possible assisting the driver to operate the control safely. Another point which was taken into account while designing the steering system is that the driver's wrists are attached to the steering wheel with safety straps. This has a major influence on the type of mechanism that is used to relay driver input to the steering by limiting the angle that the wheel can be turned. If the wheel must be turned more than 90 degrees to reach the lock on the steering mechanism, and the driver is unable to adjust their hands, than this will create an extremely uncomfortable angle as well as it will increase the possibility of injury that could can happen at full lock. To meet the goal of performing well and completing the endurance race in good standing, then ergonomics must be considered. Good ergonomics will allow the driver to drive quickly and comfortably avoiding unnecessary stoppage throughout the endurance race.

II. DESIGN CONSIDERATION

The design process of the roll-cage involves various factors such as material selection, cross section selection, frame design and finite element analysis. One of the key design decisions that tremendously increase the safety, reliability and performance in any vehicle structure is material selection. As per the SAE Baja rulebook 2018 [1], there should be at least 0.18% of carbon content in a circular tube, also it has to satisfy the bending stiffness and bending strength criteria mentioned. Our underlying advance was to direct a market overview to have a thought of the accessibility of the material. Based on the market overview we have selected AISI 4130 as material in order to have high strength to weight ratio.

A. *Material Selection*

AISI 4130 is a low-alloy steel containing chromium and molybdenum as strengthening agents. The steel has good strength, toughness, weldability and machinability. AISI 4130 is corrosion resistance and has a reasonable strength, also it has high strength to weight ratio. It easily available in the market. In light of the over the favorable circumstances referenced we haven chosen AISI 4130 as a material.

Sr. No.	Property	Value
1	Density	7850 kg/m^3
2	Yield strength	460 MPa
3	Ultimate tensile strength	560 MPa
4	Modulus of elasticity	205 GPa
5	Poisson's ratio	0.29

B. *Cross Section Selection*

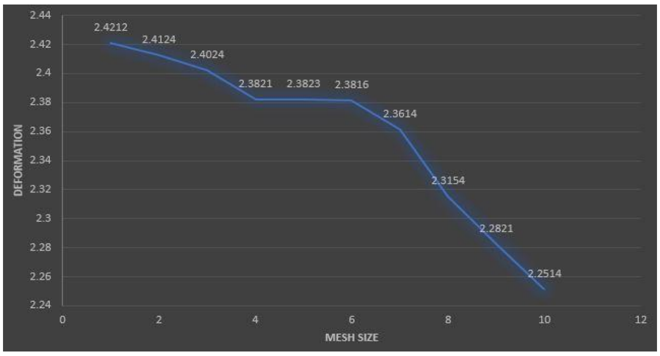
As discussed above the criteria mentioned in the SAE Baja rulebook 2018 [1], the material which we have selected i.e. AISI 4130 satisfies the carbon content criteria. In order to reduce the weight of the roll cage we have used different diameter of pipes. The diameter of the primary member is 29.2 mm and thickness of 1.65 mm and that of the secondary member is 25.4 mm and thickness of 1.25 mm. As the primary member also satisfies the bending stiffness and bending strength criteria which comes out to be 2789.13 Nm^2 and 428.52 Nm.

III. FINITE ELEMENT ANALYSIS

Using solid modelling tool in Solidworks and finite element analysis in Ansys roll cage is designed and optimized for high strength to weight ratio. Static analysis is carried out for all cases along with modal analysis. We incorporate work energy principle and impulse momentum equation to find out the forces which are used as input parameters in static analysis with all suitable constraints and variable conditions for validation of analysis.

A. Mesh Size Selection For Analysis

Mesh size is calculated by plotting a graph between deformation and mesh size from 1mm to 15mm for various iterations. Hence a mesh size of 5mm is selected as the deformation becomes constant. It means that there will be negligible changes in the accuracy of the results on the further reduction in the mesh size.



B. Type of Element

We are selecting 2-d element as the element type as the length (x) and the outer diameter (y) of the pipe is greater than the thickness (t) of the pipe (x, y >t).

C. Assumptions

As per earlier studies [2],

- The roll cage acts as a deformable body.
- Impact time in case of a deformable object is taken as 0.30 seconds.
- Impact time in the case of rigid body is taken as 0.13 seconds.

IV. IMPACT FORCE CALCULATION AND BOUNDARY CONDITIONS

A. Front impact

During front impact, the ATV may hit to another ATV at the event site. As the ATV is the deformable body hence, the impact time is assumed to be 0.3 seconds. For analysis, ATV is considered to be in static state and force corresponding to velocity 60 Km/hr. with the impact time 0.3 seconds is applied to the front part of the roll cage of ATV keeping rear suspension mounting to be fixed.

Calculations:

Weight of the ATV (m) = 250 kg

Initial velocity ($V_{initial}$) = 16.67m/s (60 Km/hr.)

Final Velocity (V_{final}) = 0 m/s.

Impact time (t) = 0.30 seconds.

As per earlier studies [3]

From work energy principle,

Work done = Change in K.E,

$|W| = |1/2 \times m \times (V_{final})^2 - 1/2 \times m \times (V_{inital})^2|$

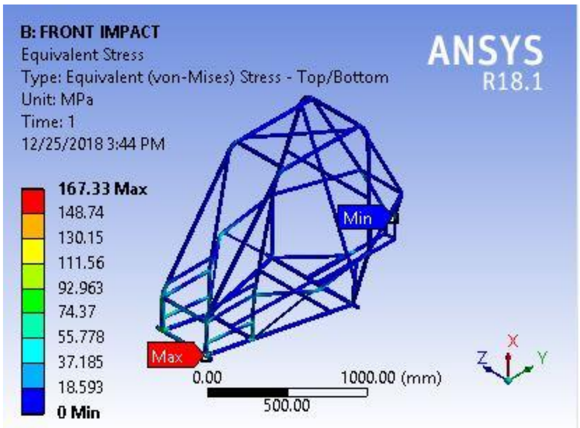
$|W| = |1/2 \times m \times (V_{inital})^2| = 34736.11$ Nm

Displacement (s) = Velocity(V) x Time (t)

$s = V_{final} \times t = 16.67 \times 0.3 = 5.001$ m

W=Force(F) x Displacement (s)

$F = W/s = 34736.11/5.001 = 6945.83\ N \approx 7000\ N$



B. Rear impact

During rear impact, another ATV may hit to ATV at the rear part at the event site. As the ATV is the deformable body hence, the impact time is assumed to be 0.3 seconds. For analysis, ATV is considered to be in static state and force corresponding to velocity 60 Km/hr. with the impact time 0.3 seconds is applied to the rear part of the roll cage of ATV keeping front suspension mounting to be fixed.

Calculations:

Weight of the ATV (m) = 250 kg

Initial velocity ($V_{initial}$) = 16.67m/s (60 Km/hr.)

Final Velocity (V_{final}) = 0 m/s.

Impact time (t) = 0.30 seconds.

From work energy principle,

Work done = Change in K.E,

$|W| = |1/2 \times m \times (V_{final})^2 - 1/2 \times m \times (V_{inital})^2|$

$|W| = |1/2 \times m \times (V_{inital})^2| = 34736.11$ Nm

Displacement (s) = Velocity(V) x Time (t)

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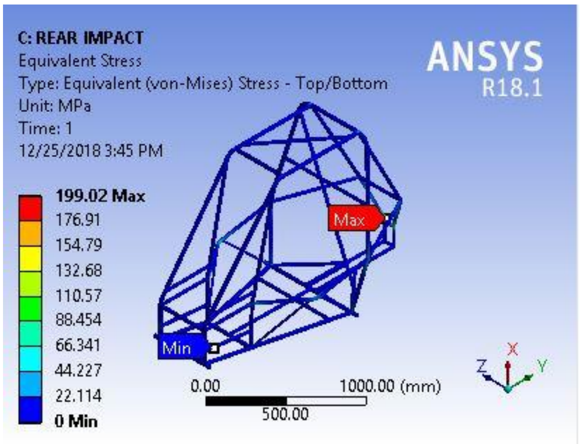
W=Force(F) x Displacement (s)

F = W/s = 34736.11/5.001 = 6945.83 N ≈ 7000 N



C. Side impact

During side impact, another ATV may hit to ATV at the side part at the event site. As the ATV is the deformable body hence, the impact time is assumed to be 0.3 seconds. For analysis, ATV is considered to be in static state and force corresponding to velocity 60 Km/hr. with the impact time 0.3 seconds is applied to the side part of the roll cage of ATV keeping front and rear suspension mounting to be fixed.

Calculations:

Weight of the ATV (m) = 250 kg

Initial velocity ($V_{initial}$) = 16.67m/s (60 Km/hr.)

Final Velocity (V_{final}) = 0 m/s.

Impact time (t) = 0.30 seconds.

From work energy principle,

Work done = Change in K.E,

$|W| = |1/2 \times m \times (V_{final})^2 - 1/2 \times m \times (V_{initial})^2|$

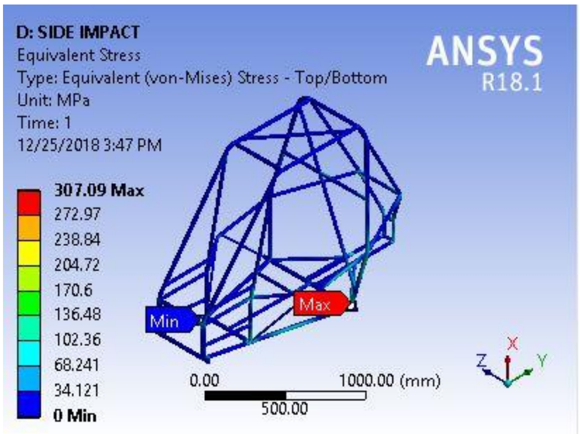
$|W| = |1/2 \times m \times (V_{inital})^2| = 34736.11$ Nm

Displacement (s) = Velocity(V) x Time (t)

$s = V_{final} \times t = 16.67 \times 0.3 = 5.001$ m

W=Force(F) x Displacement (s)

F = W/s = 34736.11/5.001 = 6945.83 N ≈ 7000 N



D. Roll over impact

In the roll over impact, ATV is considered to drop on its roll over hoop members on the road or ground from a height of 10 feet the drop height is considered as 10 feet because it is greater anything expected height at the event site. Since, the road and ground are non-deformable bodies, therefore impact time is taken as 0.13 seconds. For analysis, ATV is considered to be in static state the force corresponding to the calculated velocity of 27.612 Km/hr. for the corresponding height with the impact time of 0.13 seconds is applied to the top of the roll cage of the ATV keeping front and rear suspension mounting to be fixed.

Calculations:

Weight of the ATV (m) = 250 kg

Impact time (t) = 0.13 seconds.

Height (h) = 3 m

Potential Energy = Kinetic Energy

$m \times g \times h = 1/2 \times m \times V^2$

$V = \sqrt{2gh} = 7.67$ m/s

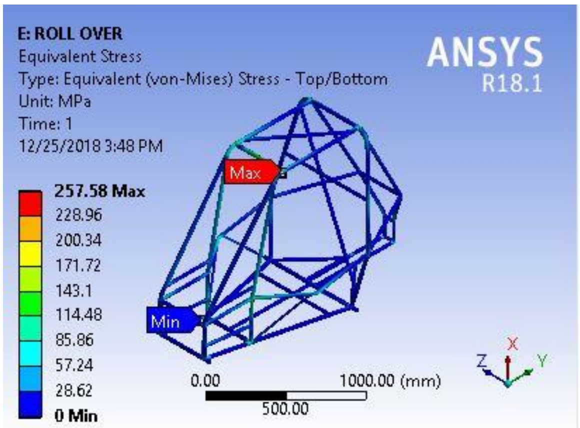
$|W| = |1/2 \times m \times V^2| = 7353.61$ Nm

Displacement (s) = Velocity(V) x Time (t)

$s = V_{final} \times t = 7.67 \times 0.13 = 1$ m

W=Force(F) x Displacement (s)

F = W/s = 7353.61/1 = 7353.61 N ≈ 7500 N



E. Drop impact

Let us considered that our ATV is drop on the road or ground from a height of 1 meter. Since, the road and ground are non-deformable bodies, therefore impact time is taken as 0.13 seconds. For analysis, ATV is considered to be in static state the force corresponding to the calculated velocity of 15.912 Km/hr. for the corresponding height with the impact time of 0.13 seconds is applied on each damper mounting of the ATV and fixing lower base of the ATV where the occupant is seated.

Calculations:

Weight of the ATV (m) = 250 kg

Impact time (t) = 0.13 seconds.

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Height (h) = 1 m

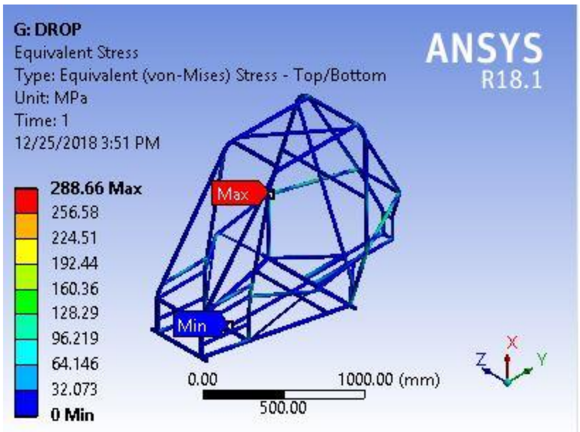
V = √2gh = 4.42 m/s

In front, m = 0.4 x 250 = 100 kg

F = m × V /t = 100×4.42/0.13 = 3400 N

In Rear, m = 0.6 x 250 = 150 kg

F = m× V /t = 150×4.42/0.13 = 5100 N



F. Torsional impact

Torsional Analysis of the roll cage is carried out in order to determine the torsional stiffness during the cross bump at the front and rear which occurs at event site. The main purpose of torsional analysis is to find greater roll cage stiffness to withstand dynamic suspension loads.

Calculations:

The weight distribution of our ATV is 60:40 (Rear: Front). So, the force transfer from rear to front after the applying the brake at the bump 60% of the total weight of the ATV.

Weight on front axle = 0.6 x 250 x 9.81 = 1471.5 N

A couple is generated which tries to twist the roll cage so the force 1471.5 N is applied on the four mounting points of the suspension and fixing the rear suspension mounting, in case of front torsional analysis.

Similar, calculation is carried out in case of rear torsional analysis.

Force on each mounting = 367.875 N

Length(L) = 16inch = 406.4mm

UDL load (W) = 0.46 kg

Deflection (D) = WL/2 = 0.46 × 0.4064 /2 = 0.093 m

tan ϴ = D / (mean nose length) = 0.093/ (1/2 ×0.4064)

tan ϴ = 0.4576

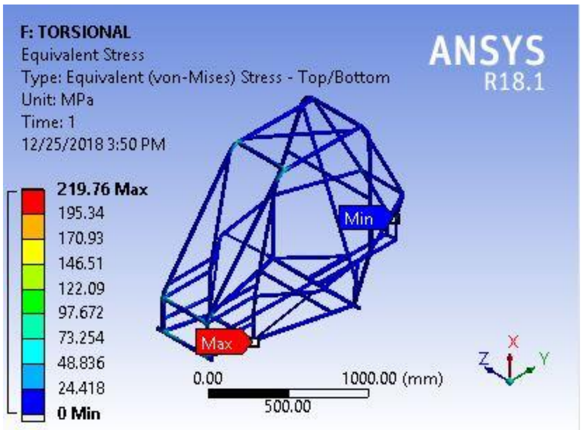
ϴ = 24.59^0 = 0.429 rad.

Torque (T) acting on the members

(T) = (left side force + right side force /2) × Track width

(T) = (1471.5 /2) ×1.2954 = 953.1 Nm

Stiffness (K) = T/ϴ = 953.1/0.429 = 2221.67 Nm/rad.



G. Bump impact

During bump analysis, let us consider an ATV undergoes a bump. According to the Indian Road Congress code [4] suggested that speed breaker is formed basically by providing a rounded radius of 17-meter, hump of 3.7-meter width and 0.10-meter height for the preferred crossing speed of 25 Km/hr.

Calculations:

Radius = 17 m

Hump Width = 3.7 m

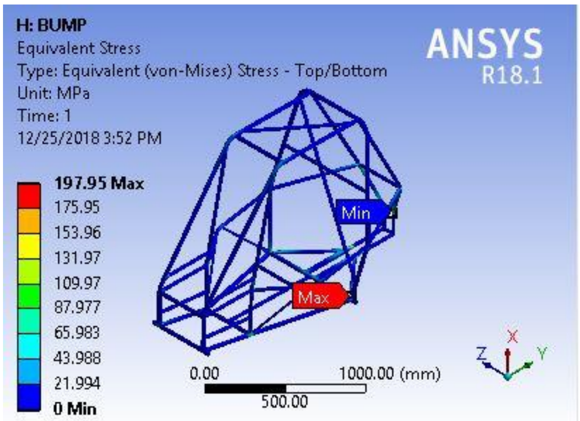
Height = 0.1 m

Speed = 25 Km/hr. = 6.94 m/s

Mass of passenger = $m_{passenger}$ = 75 kg

Bump Force (F) = $(m \times v^2)/r + mg$

F = $(75 \times 6.94^2)/17 + 75 \times 9.81$ = 948.24 N ≈950 N



H. Modal Analysis

The Baja ATV, being an off-road racing vehicle experiences severe uneven loading. When the natural frequency of vibration of frame equals to the excitation frequency of forced vibrations there occurs a phenomenon of resonance which cause a large deflection of the structure. This excessive vibration and resonance results failure of frame of ATV due to harsh conditions in which vehicle is driven. Therefore, we are finding the natural frequency of ATV roll cage using modal analysis in order to avoid such harsh situation of resonance.

International Journal of Engineering Research & Technology (IJERT)
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The natural frequency of ATV roll cage can be calculated using the following equation:

$W_n = \sqrt{k/m}$

where k = stiffness of the roll cage

m = mass of the roll cage

The major cause of forced vibration in an ATV is the engine which is mounted at the rear end of the vehicle. The range of vibrating frequency of an engine is 15Hz-25Hz in most cases of single cylinder 305 cc,10Hp engine. Therefore, we have to analyze our roll cage such that its natural frequency at various mode shape is well above this range.

Boundary Conditions:

The chassis frame is fixed at the suspension points in order to know about the various mode shapes of the upper body structure. Fixed supports are given at suspension points since the wheels and suspension are mounted to the axle thereby restricting the degree of freedom of lower base to zero.

Various mode shape values are tabulated below:

Sr. No.	Frequencies	Values (Hz)
1	Frequency 1	38.167
2	Frequency 2	45.661
3	Frequency 3	68.079
4	Frequency 4	85.001
5	Frequency 5	94.207
6	Frequency 6	109.13

V. RESULTS

Results of all the analysis are tabulated below:

Test	Forces (N)	Stress (N/mm^2)	Deformation (mm)	Factory of Safety
Front	7000	167.33	2.2417	2.75
Rear	7000	199.02	1.1048	2.3113
Side	7000	307.09	3.4994	1.498
Roll Over	7500	257.58	2.6724	1.7858
Torsional	1500 (Equal and Opposite)	219.76	1.4927	2.0932
Bump	950	197.95	5.3162	2.3238
Drop	3400(Front) 5100(Rear)	288.66	2.8003	1.5936

VI. CONCLUSION

Safety of the driver is the first and the foremost priority. Therefore, considerable factory of safety is applied to roll cage of an ATV to reduce the risk of failures and possible injuries. Larger factory of safety implies large ability of an ATV to withstand all kind of loads and capable of moving on various terrains. This paper has illustrated the entire design methodology of roll cage and understanding the critical aspects of the design, also static analysis in finite element analysis along with modal analysis to avoid the phenomenon of resonance.

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