Design and Analysis of BAJA Rollcage

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Abstract - *This report deals with the Design and analysis of Roll Cage for the ATV Car. In a ATV Car the roll cage is one of the main components. It is the main subsystem of ATV on which the other subsystems like Engine, Steering, and Transmission are mounted. Roll Cage comes under the sprung mass of the vehicle. The forces acting on vehicle from various directions are responsible for crack initiation and deformation in the vehicle. Therefore stresses are generated. In this paper an attempt is made to find out these areas by carrying out FEA of the roll cage. We have carried out crash analysis (front, rear, side impact, rollover) and torsional analysis. All these analysis have been carried out in ANSYS APDL 19.2. The design procedure follows all the rules laid down by SAE rule book for m-baja cars.*

Key Words: Rollcage, Finite Element Analysis, Strength, Factor of safety, sprung mass

1. INTRODUCTION:

For a better performance of a ATV car it is important to make sure that all components work as per proposed design calculations. As roll cage chassis being the important system, which absorbs all the static and dynamic loads the structure must be such that it will sustain the stresses generated without deformation.

Various forces act's on the vehicle when the vehicle is in static as well as in the dynamic condition. These forces cause deformation which resulted in stress generation at different areas of the roll cage. These forces generally occur during the event performing various rounds such as breaking, acceleration, maneuverability and endurance. The stiffness factor of the roll cage must be able to resist these forces acting.

The main function of roll cage is to absorb all the loads from the suspension with minimum deformation. The second basic function is that it should act as a mounting member for all the other components on the vehicle. The third and most important function is that it must have more continuous members to distribute the stresses as and high torsional stiffness to sustain the forces during event.

There are two types of masses inside the car – Sprung and the Un-sprung mass. All the mass of other subsystem components that is damped by the spring is called as the sprung mass. Generally the sprung mass must be great than that of the un-sprung mass. Roll cage comes under the category of the sprung mass of the car.

Moreover, the rollcage is made by joining seamless tubes by means of welding. First a proper design of the rollcage is to be made by taking desired dimensions from suspension and power subsystems. The pipes are made into desired dimensions by cutting and bending. Then notching is done of these pipes. These pipes are then joined by welding.

2. MATERIALS AND METHODS:

Material of higher yield strength should be selected to maintain the stress level in the roll cage up to a desired limit. Here the strength of the material especially the ultimate yield strength plays a very important role. The factors affecting the selection of the material are

1. The stress generated and the factor of safety of the roll cage.
2. The conditions of the land or track on which the ATV is required to perform.
3. The allotments given for other subsystem components to be mounted on the roll cage.

The factor of safety (FOS) plays a major role. The factor of safety is defined as the ratio of the ultimate yield strength to that of the stress generated in the roll cage. This value for the roll cage of a ATV car is always required to be within 1-3. If the value is less than 1, leads to huge deformation before reaching the maximum stress value. If the value is greater than 3, then the weight of rollcage will be too heavy.

The vehicle is required to be performed in sunny, muddy and off road terrain conditions. The parts on rollcage need to be either welded or nut bolted. Thus the materials selected must be weldable. Hence we require a material which has high strength, durability and also which is weldable. By considering these requirements the material had been selected as AISI 4130

AISI 4130

Table 1: Material properties:

Property	Metric
Tensile strength	750 MPa
Yield strength	650 MPa
Elongation	25.5%
Modulus of elasticity	205 GPa
Density	750 Kg/m^3
Percentage of carbon	0.28%

3. FRAME DESIGN:

The Rollcage is designed considering many factors such as cross sectional area, the front impact force, the side impact force, the endurance, the vibrations generated by the engine and the damping capacity of the wheel while running

The cross section selected as circular tube since it was mentioned in rulebook. The primary and secondary cross sections are as follows Primary diameter 29.2mm with thickness of 1.65mm and secondary as 25.4mm diameter with 1.2mm thickness. These dimensions are selected by calculating the bending stress and bending stiffness.

3.1. AREA MOMENT OF INERTIA(I) :

$I = (\pi/64)*(D^4 - d^4)$

$= 1.3598 \times 10^{-8}\ m^4$

3.2. BENDING STRENGTH:

$= Sy*I/y$; $Y = D/2$

$= 405.3320505\ N/m^2$

D= distance from neutral axis

3.3. BENDING STIFFNESS:

$= E*I$

$= 2787.59\ N/m$

The Rollcage must be designed in such a way that it should withstand all the forces from all directions, the stress must be distributed along the pipes and must also allow this impact to flow from the front part to the rear part of vehicle, so that the stresses are distributed all over the rollcage.

The Rollcage is designed in CAD software- CREO 3.0. The proposed design of roll cage is shown below.

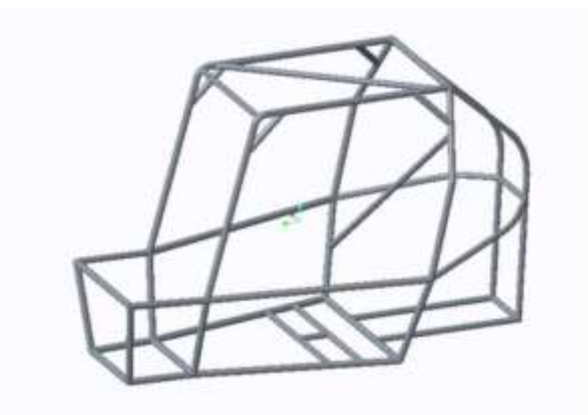

Fig -1: Rollcage

4. FINITE ELEMENT ANALYSIS (FEA):

This FEA software is used to predict the failure and stress concentration in the design before going into manufacturing and also shows whether a product will break, wear out, or work the way it was designed. . Therefore the cost of manufacturing will be optimized. Here depending on the element size the rollcage is divided into small elements to form a perfect mesh so that the results obtained will be more accurate. The computer analyses and solve by the computational method provided.

To finalize the material and structure of roll cage FEA was performed on it.

4.1. Choosing the type of analysis in preferences (static analysis):

- Type of element Pipe 288, it has 2 nodes in 3D space having 6 degrees of freedom at each node.
- Given input on material properties such as density, young's modulus and Poisson's ratio.
- Inputs on section properties are outer diameter, inner diameter, wall thickness.
- Creating nodes in working plane.(58 nodes)
- Joining nodes with elements.(47 elements)
- Meshing attributes are selected as element shape tetrahedral and size 6.
- Applying loading and boundary conditions as per respective impact tests.

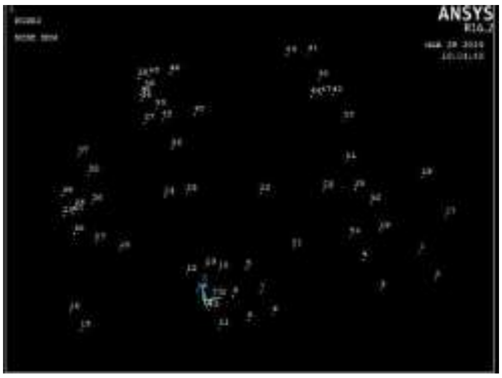

Fig -2: Nodes

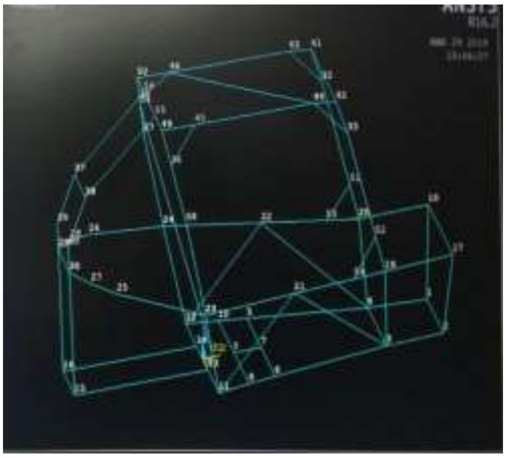

Fig -2.1: Elements

Following tests were performed on the vehicle

1. Front impact.
2. Rear impact.
3. Side impact.
4. Rollover
5. Torsional analysis.

4.1. FRONT IMPACT:

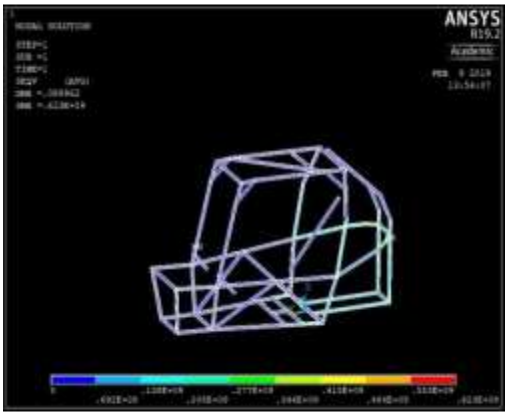

Fig -3: Equivalent von-misses

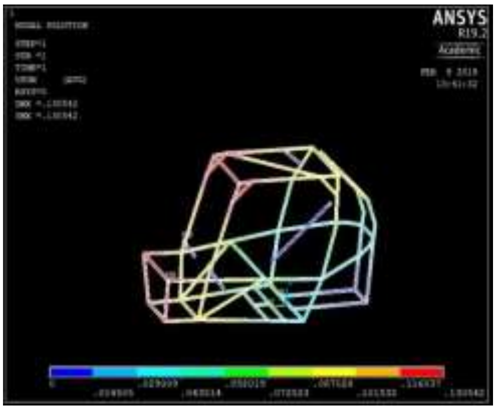

Fig -3.1: Deformation

The front impact analysis is done in a condition that when the impact occurred at the front part of the Atv, as the stresses are generated at the front part so that deformation and factor of safety are observed. By those results it is easy to judge that whether this rollcage can withstand to that impact or not. The crash analysis has been performed in Ansys APDL19.2. This analysis determines the safety of the driver.

4.1.1. Boundary conditions:

1. Total vehicle weight is assumed as 250 Kg in order to analyze the vehicle at max G load the vehicle weight is assumed high. since the weight for proposed design is only 170 kgs
2. Symmetry along (plane z-x)
3. Rare suspension points are fixed leaving the desired degree of freedom at the mounting points and a 7G

   u(initial velocity)=16.5m/s

   (Targeted max. speed);

   t (impact time) =0.25;

   Deceleration suffered by the vehicle

   =u/t =16.5/0.25

   =66 m/s2

   = 66/g

   =6.7 approx 7G equally distributed load is applied on the front bumper or suspension pick up points.

4.1.2. Results:

1. Maximum von-misses stress = 623 MPa
2. Factor of safety = 650/623

   = 1.04
3. Deformation = 13.5cm

Here the design is safe.

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4.2. REAR IMPACT

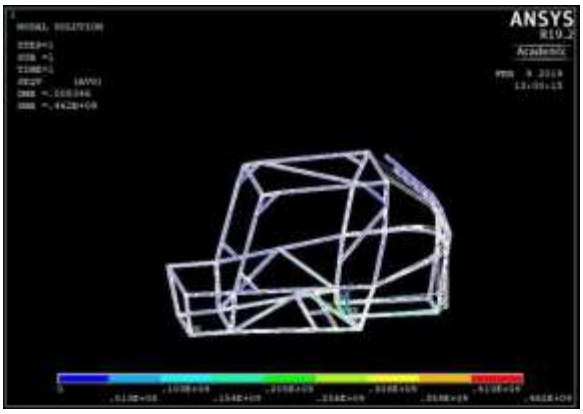

Fig -4: Equivalent von-misses stress

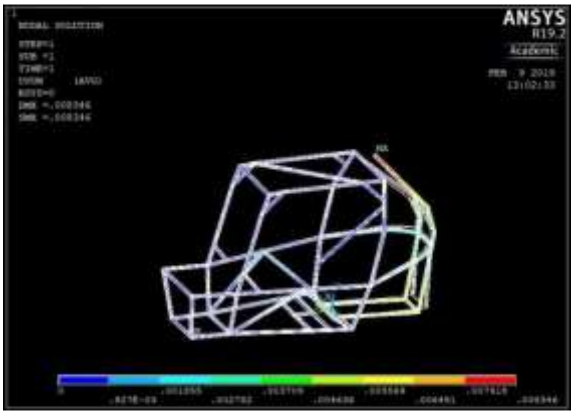

Fig -4.1: Deformation

The rear impact analysis is done in a condition that when the impact occurred at the rear part of the ATV, as the stresses are generated at the rear part so that deformation and factor of safety are observed. By those results it is easy to judge that whether this rollcage can withstand to that impact or not. The crash analysis has been performed in Ansys APDL19.2. This analysis determines the safety of the driver.

4.2.1. Boundary conditions:

1. Total vehicle weight is assumed as 250 Kg in order to analyze the vehicle at max G load the vehicle weight is assumed high. since the weight for proposed design is only 170 kgs

2. Symmetry along (plane z-x)

3. Front suspension points are fixed leaving the desired degree of freedom at the mounting points and a 7G equally distributed load is applied on the rear bumper or suspension pick up points.

   u(initial velocity)=16.5m/s (Targeted max. speed);

   t (impact time)=0.25;

   Deceleration suffered by the vehicle

   =u/t

   =16.5/0.25

   =66 m/s2

   = 66/g

   =6.7approx 7G

4.2.2. Results

1. Maximum von-misses stress = 462 MPa

2. Factor of safety = 650/462

   = 1.4

3. Deformation = 8.3mm

Here the design is safe.

4.3. SIDE IMPACT:

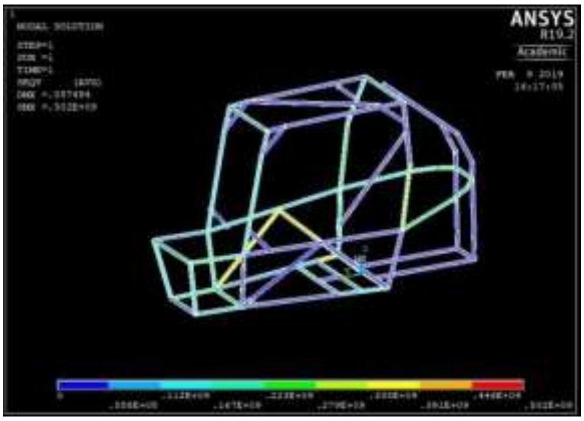

Fig -5: Equivalent von-misses

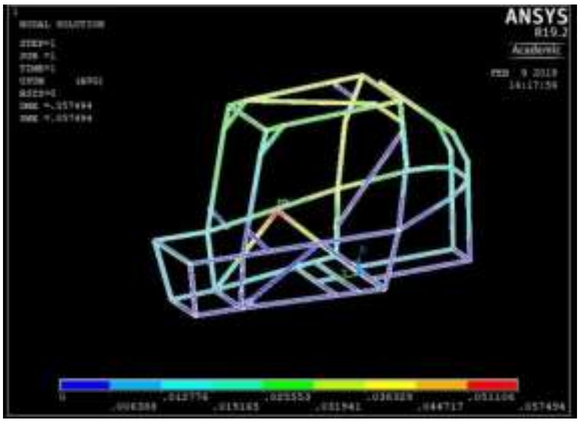

Fig -5.1: Deformation

The side impact analysis is done in a condition that when the impact occurred at the side part of the ATV, as the stresses are generated at the side part so that deformation and factor of safety are observed. By those results it is easy to judge that whether this rollcage can withstand to that impact or not. The crash analysis has been performed in Ansys APDL19.2. This analysis determines the safety of the driver.

4.3.1. Boundary conditions:

1. Total vehicle weight is assumed as 250 Kg in order to analyse the vehicle at max Gload the vehicle weight is assumed high. since the weight for proposed design is only 170 kgs.
2. Symmetry along (plane z-x)
3. Suspension points on the opposite side of the interested side are fixed leaving the desired degree of freedom at the mounting points and a 5G equally distributed load is applied on the joints in the side members which are prone to side impact.

   u(initial velocity)=16.5m/s (Targeted max. speed);

   t (impact time)=0.33;

   Deceleration suffered by the vehicle

   =u/t

   =16.5/0.33

   =50 m/s2

   = 50/g

   =5G loading applied

4.3.2. Results:

1. Maximum von-misses stress = 502 MPa
2. Factor of safety = 650/502 = 1.29
3. Deformation = 5.7mm

Here the design is safe.

4.4. ROLLOVER ANALSYS:

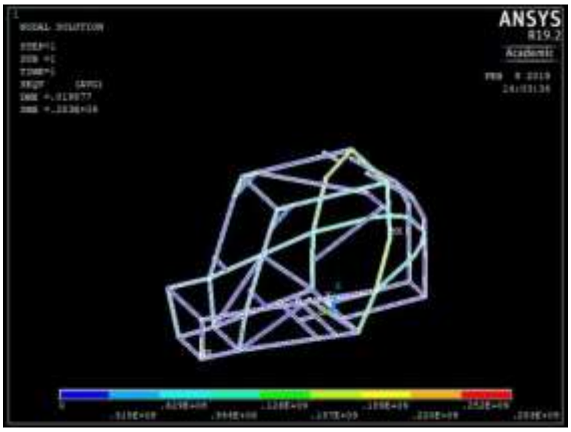

Fig -6: Equivalent von-misses stress

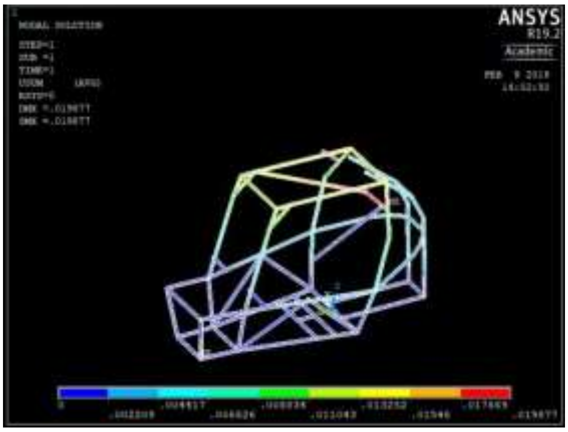

Fig -6.1: Deformation

In roll over impact test it is observed that all the members associated with the RHO which is gussets and diagonal supporting member took the load along with the RHO. This analysis was done in Ansys APDL 16.2. This analysis determines the safety of the driver.

4.4.1. Boundary conditions:

1. Total vehicle weight is assumed as 250 Kg in order to analyze the vehicle at max G load the vehicle weight is assumed high. since the weight for proposed design is only 170 kgs

Here we assumed as rollcage is falling upside down from a height of 5 meters by taking an impact time of 0.33sec

By using

S=ut+1/2at^2

S= height of body from ground; taken as 5 meters

U= velocity of body falling from height in m/s

5=u*(0.33)+1/2*9.8*0.33^2

U=13.5m/s

4.4.2. G loading calculation:

u(initial velocity)=13.5m/s (Targeted max. speed);

t (impact time)=0.33;

Deceleration suffered by the vehicle

=u/t

=13.5/0.33

=40 m/s2

= 40/g

=4G

4.4.3. Results

Maximum vonmisses stress= 650/283

Factor of safety= 2.29

Displacement= 1.9mm

4.5. TORSIONAL ANALYSIS:

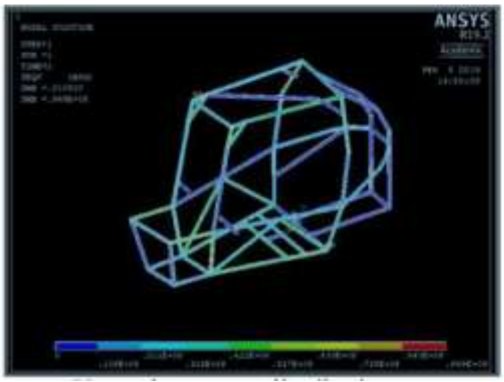

Fig -7: Equivalent von-misses

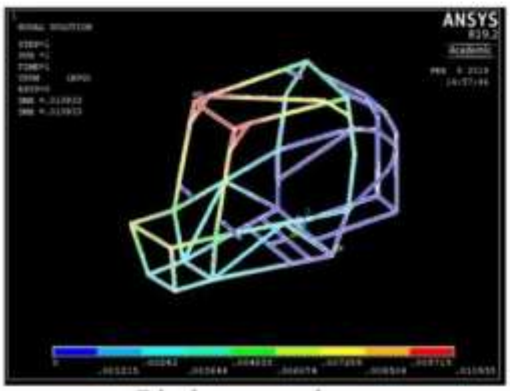

Fig -7.1: Deformation

Torsional analysis is used to determine the torsional stiffness of the structure i.e., how much the structure can resist the twisting. It is very important parameter of any vehicle to resist torsional stress and deflection during the turning, drifting, cornering and undulating road surface. Since the ATV has high torsional stiffness so it could withstand bending in any direction.

4.5.1. Boundary conditions:

1. Rear suspension points were constrained
2. Equal and opposite forces in z direction were applied on the front suspension points
3. Track width = 1320.8 mm
4. Standard load = 1G
5. Weight of vehicle = 250 Kg

4.5.2. Results:

1. Vertical deflection = 10.9 mm
2. Max von-misses stress = 249 Mpa
3. Torsional stiffness = 1704.87 Nm/degree
4. Factor of safety = 650/249 = 2.61

4.5.3. Calculations and formulae:

Calculation for torsional stiffness

Roll cage weight = 27 Kg

Car weight = 250 Kg

Force (F) = 1G

= 1×9.81×250

= 2452.5 N

Torque = F×$\left(\frac{1}{2}\right)$×track width

= 2452.5×$\left(\frac{1}{2}\right)$×1.3208

= 1619.6310 Nm

θ = Angle of deflection = $\tan^{-1}\left(\frac{vertical\ displacement}{\left(\frac{1}{2}\right)\times track\ width}\right)$

= $\tan^{-1}\left(\frac{10.9}{\left(\frac{1}{2}\right)\times 1320.8}\right)$

= $\tan^{-1}(0.0165)$

= 0.95°

Torsional stiffness (k) = $\frac{torque}{angle\ of\ deflection}$

$= \frac{1619.63}{0.95}$

= 1704.88 Nm/degree

5. RESULTS AND CONLUSION:

TEST NAME	EQUIVALENT VON MISSES(Mpa)	DEFORMATION (mm)	F.O.S
FRONT (7G)	623	13.5	1.04
REAR (7G)	462	8.3	1.4
SIDE (5G)	502	5.7	1.29
ROLLOVER (4G)	283	1.9	2.29
TORSIONAL (1G)	249	10.9	2.61

The design is said to be safe by considering above obtained values.

6. REFERENCES:

1. Baja tutor
2. Baja Rule book 2019
3. Creo 3.0 and Ansys APDL 16.2 FEA tests
4. Vehicle dynamics by THOMAS GILLESPE
5. Crash tests of different cars like NANO, HYUNDAI I10