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Design Report on ATV

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Abstract:-This paper aims at studying the standard vehicle system and modifying it according to the constraints provided by the Rulebook of Baja SAEINDIA 2016 and to be used as All-Terrain Vehicle (ATV). It includes selection and development customized components for ATV for fine performance and greater safety of driver in endurance race.

The team's primary objective is to design a safe and functional vehicle based on a rigid and torsion-free roll cage and chassis, well mounted powertrain, and dynamically tested steering and suspension systems. The secondary objective was to enhance performance and maneuverability of the vehicle.

The team was divided into core groups responsible for the design and optimization of major sub-systems which were later integrated into the final blueprint. Current CAD modeling and FEA approaches were used.

I. INTRODUCTION

The ATV KARNIK V1.0 is modeled in CATIA V5R21 and SOLIDWORKS 2014 because of their greater flexibility and productivity for modeling different components.

To test and validate the different models the ANSYS Workbench 16.0 is used. The results were studied and remodeled and retested. Such iteration is repeated unless and until maximum possible weight reduction is achieved without any loss of structural stability and driver safety.

Dynamics analysis was done in Lotus suspension analysis software. The aim was to optimize suspension variables to improve maneuverability. Theoretical calculations of performance characteristics were also done.

Extensive weight reduction techniques were followed at every stage of the design to improve performance without sacrificing structural integrity.

We have designed the roll cage keeping in view the safety and aesthetics. These are the two factors which matters us the most, therefore they are given utmost consideration. All work was performed in accordance with the SAE Baja guidelines to maintain the car's eligibility for the competition.

II. DESIGN OF MAJOR SYSTEMS

The major systems involved in making an ATV are as follows:

- Roll Cage Design
- Suspension System
- Powertrain
- Steering and Brakes

A. Roll Cage Design

The material chosen for the roll cage is AISI 1018 with an outer diameter of 1.25 inches and a wall thickness of 1.6 mm. All the members of the frame i.e. primary and secondary members are of the same cross-section.

Joining method used will be Shield Metal Arc Welding (SMAW). The earlier frame design had the outer diameter of 1 inch and wall thickness of 3 mm. The Finite Element Analysis was done on the frame and the result were found to be quite satisfactory but the frame had too much weight and also when the entire powertrain was modeled, the engine bay area was found to be insufficient. So we resized the engine bay and remodeled our roll cage design.

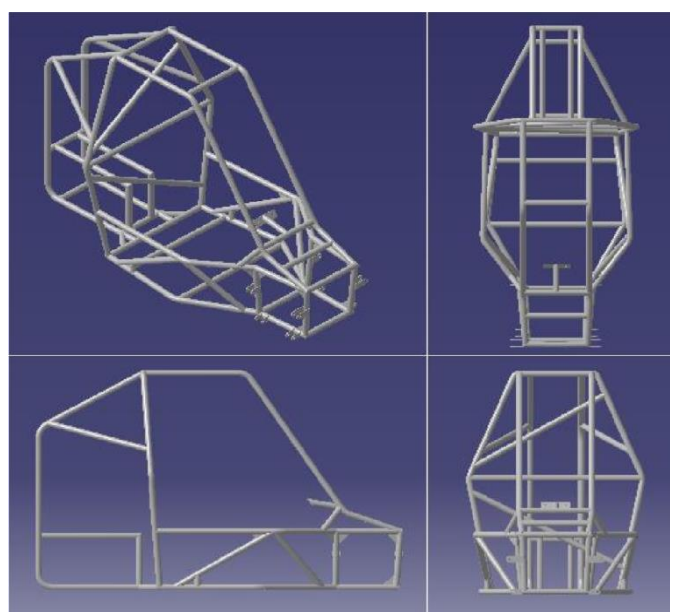


Figure 1.Final Roll Cage Design

a). Front Impact

In front impact the total force equivalent to 4 times the gross weight of the vehicle i.e. 4x300x10 (or 9.81) = 12000 N which is said to be a 4G (4xGross Weight) force can be applied. Therefore, the force of 3000 N was applied at 4 points on the frame and the back of the frame was completely constrained.

The deformation and stresses are shown below. The permissible stress for AISI 1018 is 365MPa. Hence, for a stress of 311.42MPa, the FOS obtained was 1.17.

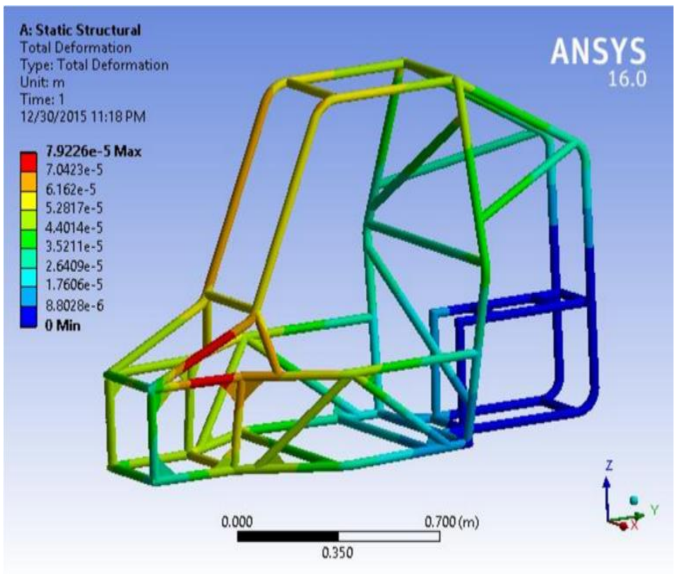


Figure 2.Total Deformation in Front Impact

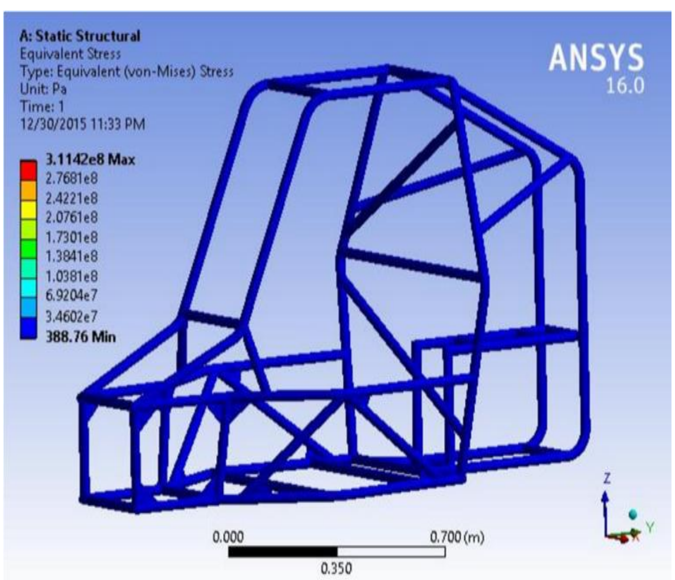


Figure 3.Stress Distribution in Front Impact

b). Rear Impact

In the rear impact a total force equivalent to the 2.5G i.e. 7500 N can be applied. Therefore, force of 1875 N was applied at 4 points on the frame. The nose of the frame was fully constrained at its four corners.

The deformation and stresses are shown below. For a stress of 295.64MPa, the FOS obtained was 1.23.

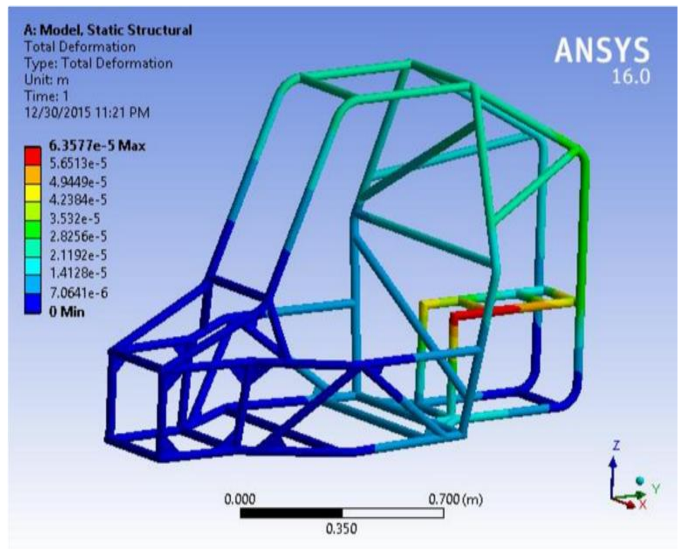


Figure 4.Total Deformation in Rear Impact

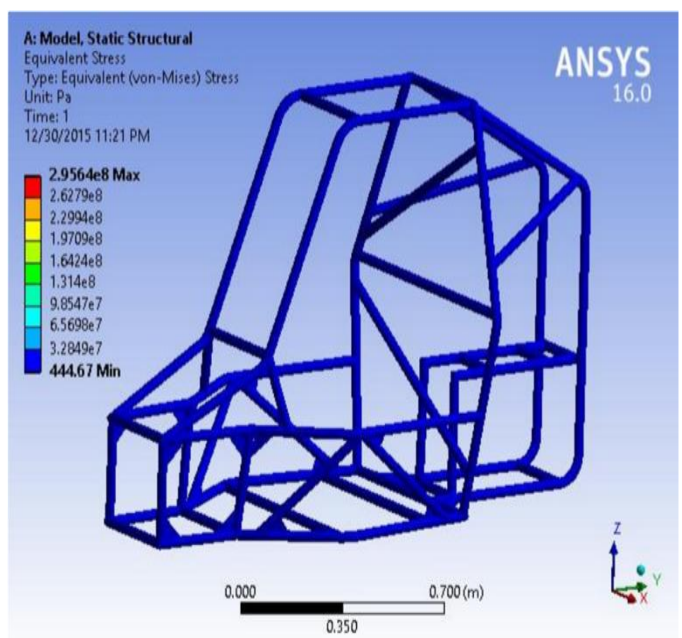


Figure 5.Stress Distribution in Rear Impact

c). Side Impact

Also in the side impact a total force of 3G i.e. 9000 N can be applied. Therefore, force of 2250N N was applied at 4 points on the frame. The other side impact member was fully constrained at four points.

The deformation and stresses are shown below. For a stress of 140.28MPa, the FOS obtained was 2.602

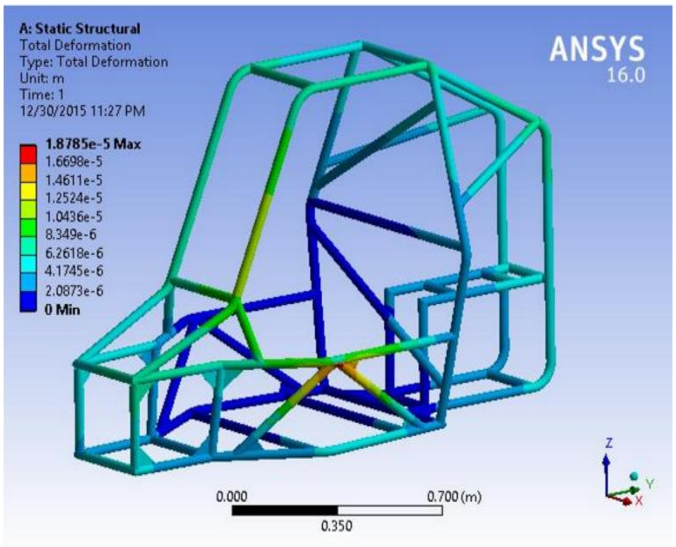


Figure 6.Total Deformation in Side Impact

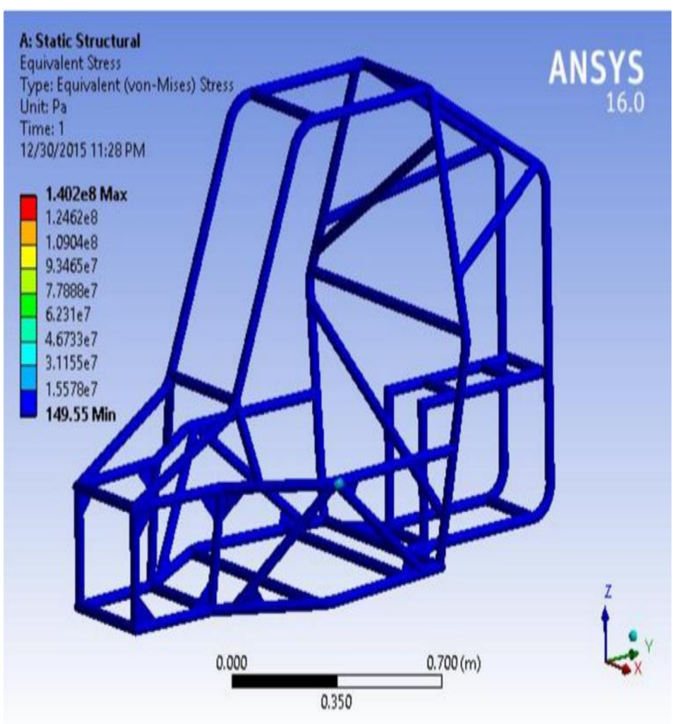


Figure 7.Stress Distribution in Side Impact

d). Rollover Test

In the rollover test the force equivalent to the 3G i.e. 9000 N was applied at four points (2250 N at each point) of the upper body members and the lower side members were fully constrained at six points.

The deformation and stresses are shown below. For a stress of 149.21MPa, the FOS obtained was 2.44

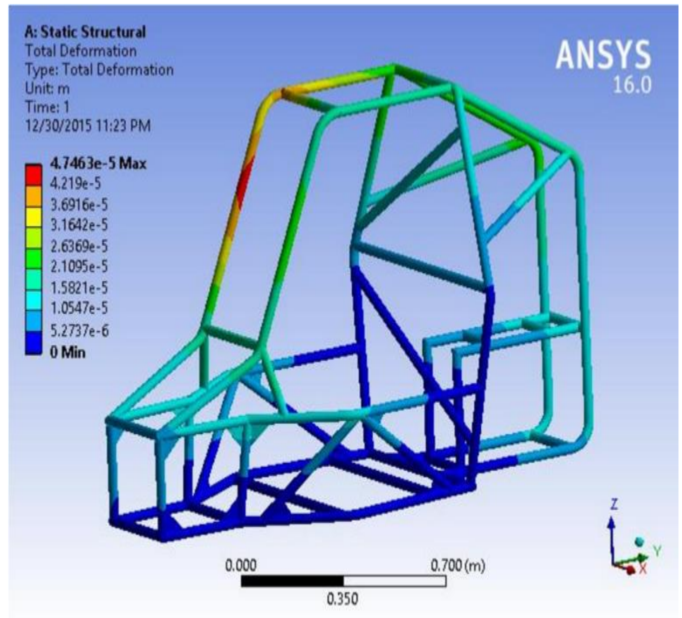


Figure 8.Total Deformation in Rollover Test

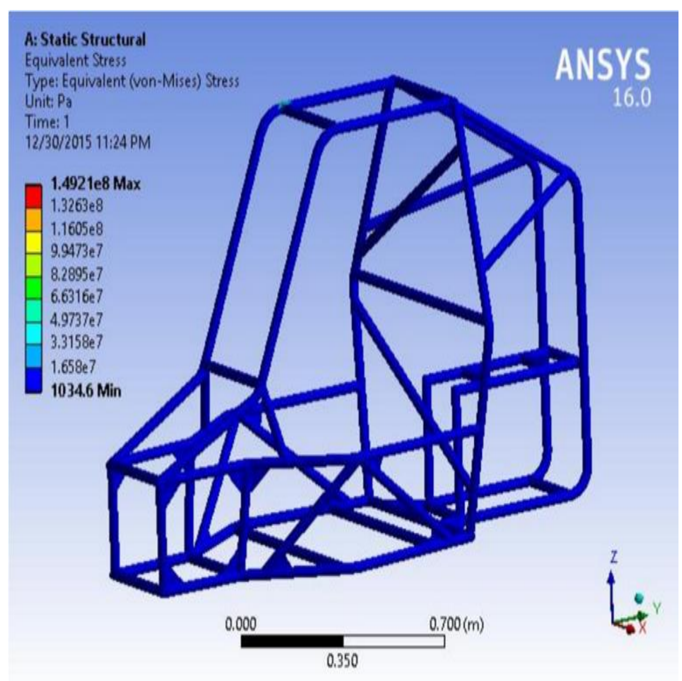


Figure 9.Stress Distribution in Rollover Test

B. Suspension Design

The main objective of the suspension system is to provide greater travel which allows better absorption of the shocks during the changes in ground conditions. To start the suspension design, firstly the vehicle parameters such as wheel track, wheel base were defined according to the rules specified by SAE INDIA BAJA.

A double wishbone suspension setup was chosen for the front as it is lightweight, independent and prevents deflection during hard cornering which ensures that the steering and wheel alignment stay constant. For rear suspension the trailing arm with camber links is used which is also called as three link trailing arm suspension it consists of normal trailing arm and also contains two links in lateral direction which is used to carry lateral load and also controls camber through suspension travel.

a). Wishbones

Material used for wishbones is AISI 4130 with diameter of 1 in and thickness of 3 mm. In the analysis a 1500 N force on the ball joint and shock absorber mounting was applied and the max stress obtained was 391.09 MPa, which gives a FOS of 1.18.

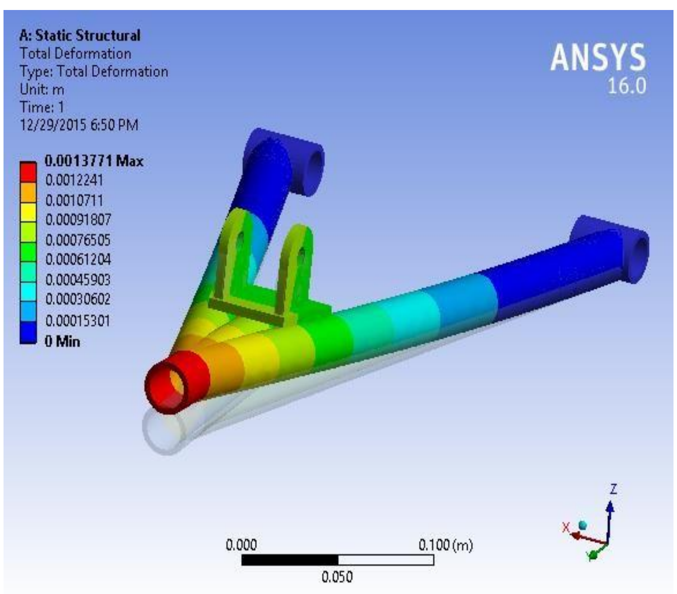


Figure 10.Total Deformation in Wishbone Drop Test

b). Trailing Arm

For the trailing arm, the material used is Mild Steel. As seen below, for an 1800 N force on the hub, the maximum stress obtained is 156Mpa, which gives a FOS of 1.6.

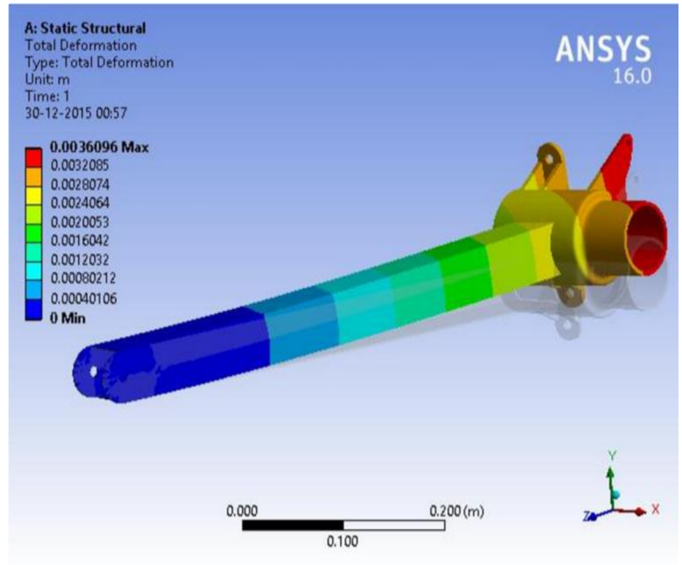


Figure 11.Total Deformation in Trailing Arm Bump Test

c). Hubs

We have used Maruti Suzuki Alto's hub in the rear for power transmission to the wheels with a customized disc spacer for maintaining the distance between rim and disc so as to place the brake caliper. The material used for the rear hub disc spacer and front hub is also the Mild Steel. On the rear wheels, there we have provided the rear hub disc spacer as shown below.

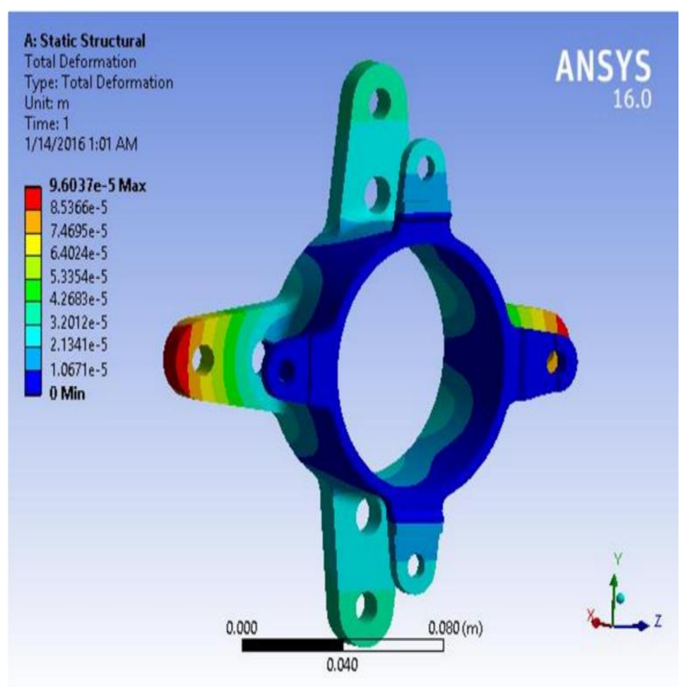


Figure 12.Total Deformation in Rear Hub Disc Spacer Bump Test

On the front wheels we have used the customized front hub on which knuckle of the premier Padmini NE 118 is to accommodate along with the two taper roller bearings and also brake disc seating knuckle has two ends for ball joints for upper and lower wishbones, material used for front hub is mild steel and the results as shown below.

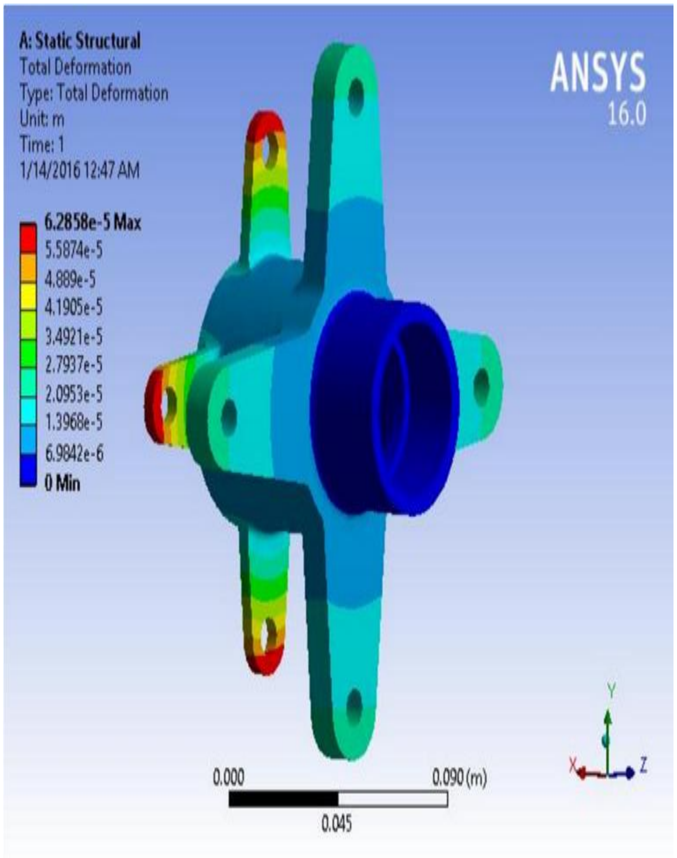


Figure 13.Total Deformation in Front Hub Braking Torque Test

d). Shock Absorbers

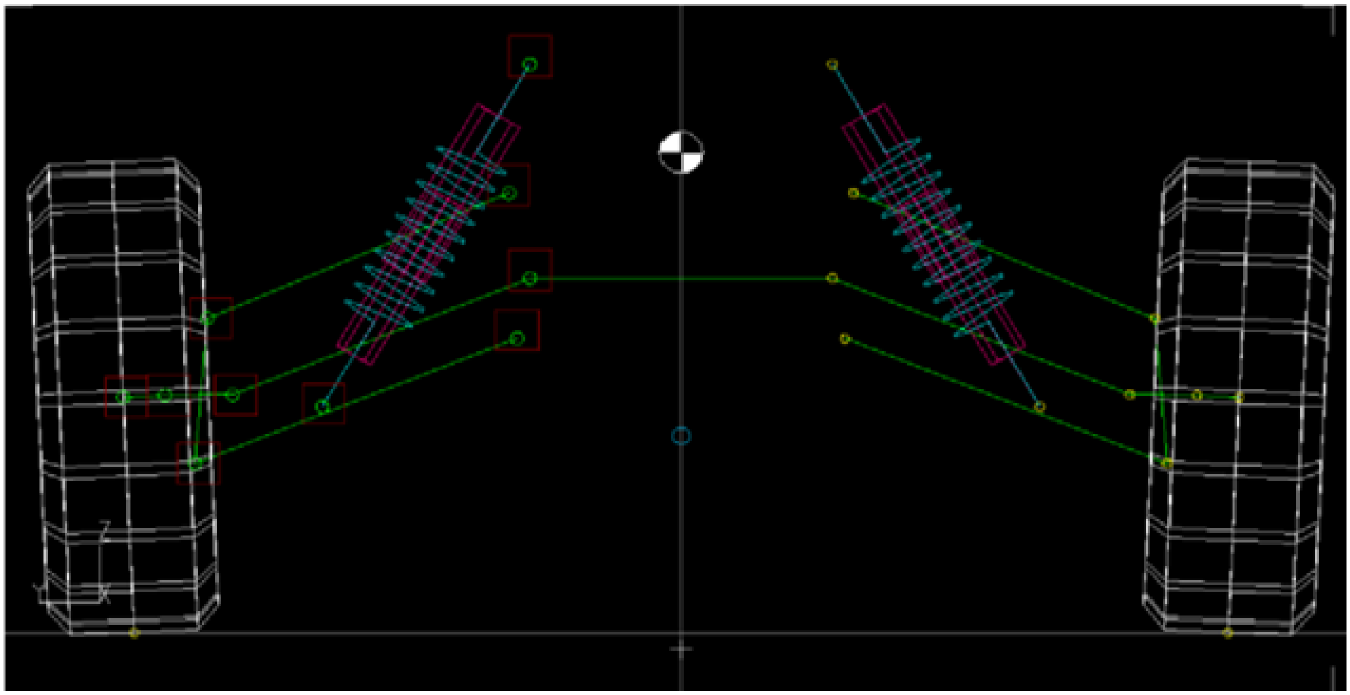
On the front wheels we are using the Coil over spring and damper that is customized by us. Material used for the spring is the music wire ASTM A228. With the designed spring rate for this dampers or coil over springs is 17.75 N/mm and designed wire diameter 10mm and 12 inch free length. Mounting for the lower and upper wishbones are also designed with dynamic analysis which can sustain the bump forces.

e). Dynamic Analysis

During wishbone design it was found that parallel arrangement of the upper and lower wishbones provides with recessional travel and allows the wheels to lift in vertical direction straight Dynamic analysis was done on the front suspension setup to check the response of the vehicle for bump, in roll and while steering. Key points were obtained from the CAD model. Variables were tuned to reduce bump steer, camber angles and wayward movement of roll center.

The dynamic analysis of the suspension system of the front and rear wheels is carried out on the LOTUS SUSPENSION ANALYSER. And results are as follows and as shown in the graphs and the photographs, the forces and the actions considered here for dynamic testing are Bump, roll, and steer. The suspension geometry parameters incremental values accordingly are shown in the graphs below. As in the front wheels we are having shock absorber that we have customized and it is simple coil spring shocker, on the other hand we are having Fox Float 2 Shockers. And so in such a case our rear trailing arm geometry helps in more wheel travel. On the case of front portion parallel upper and lower wishbones helps in stability and also having straight vertical wheel travel.

f). Bump



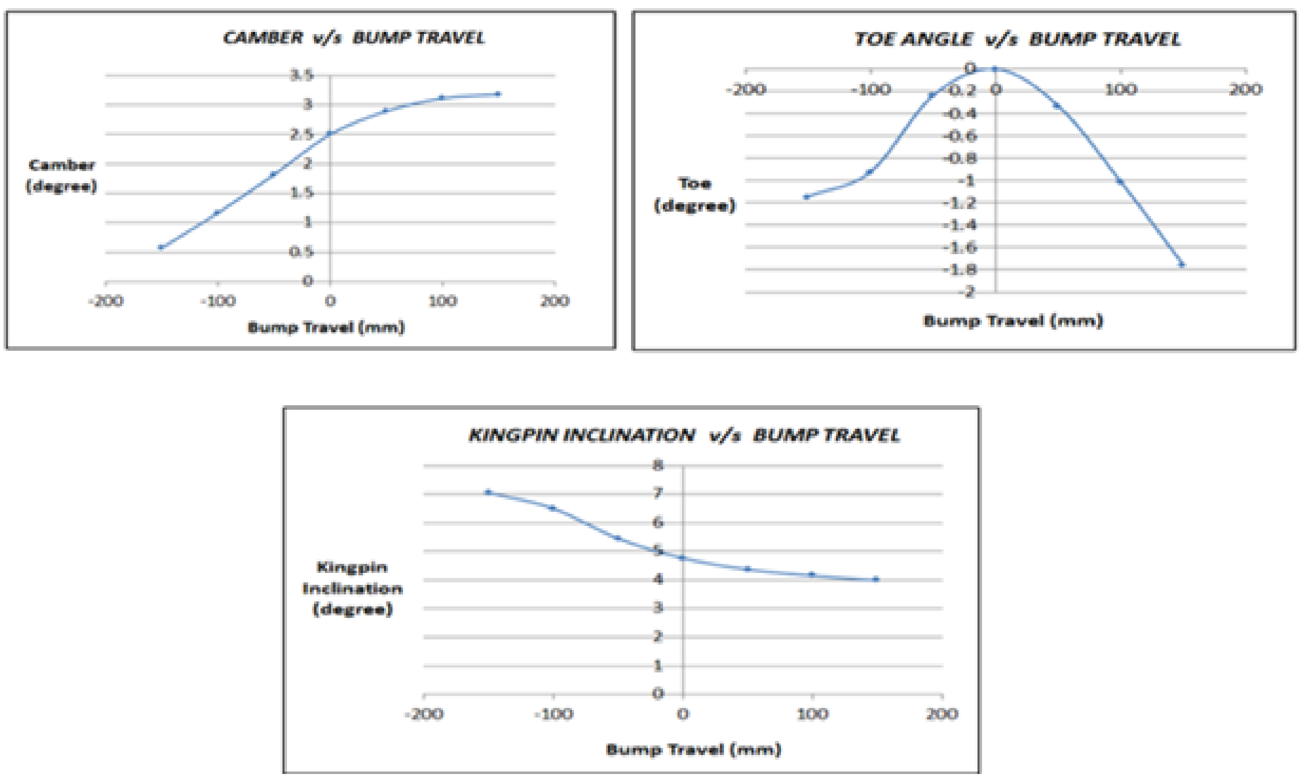


Figure 14.Graphical Results Obtained in the Bump Test

Above are the graphs for bump (mm) (x-axis) versus toe, camber and castor angles. For a bump and rebound of 150 mm each the camber was restricted within 0.5 deg and toe within 1.1 deg. This minimizes the forces on the knuckle ball joints during bumps.

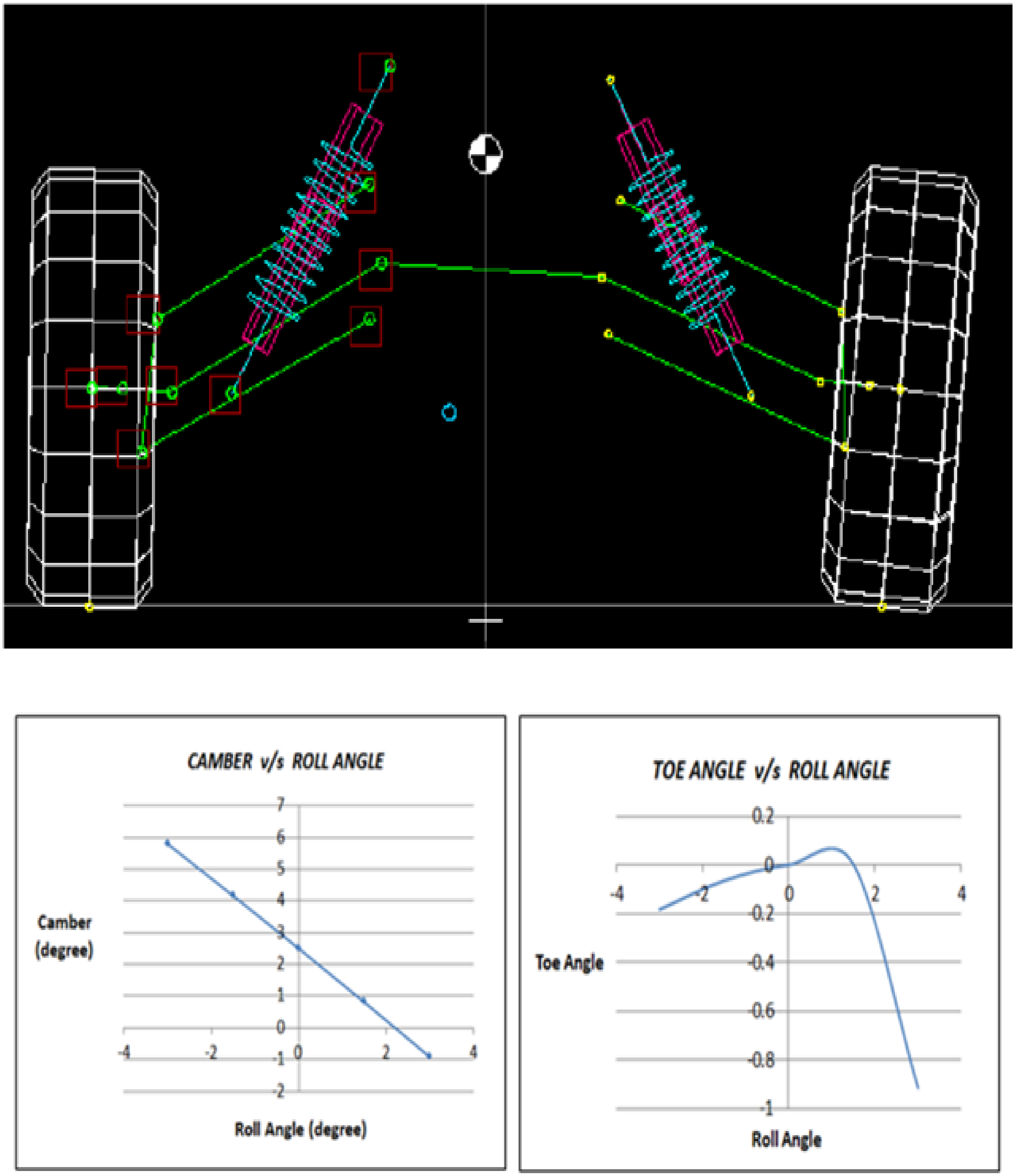


Figure 15.Graphical Results Obtained in the Roll Test

Values of toe angle, camber angle and roll center height versus roll angle (in degree) (x-axis) indicate that driver will experience good control over the vehicle while cornering.

g). Steer

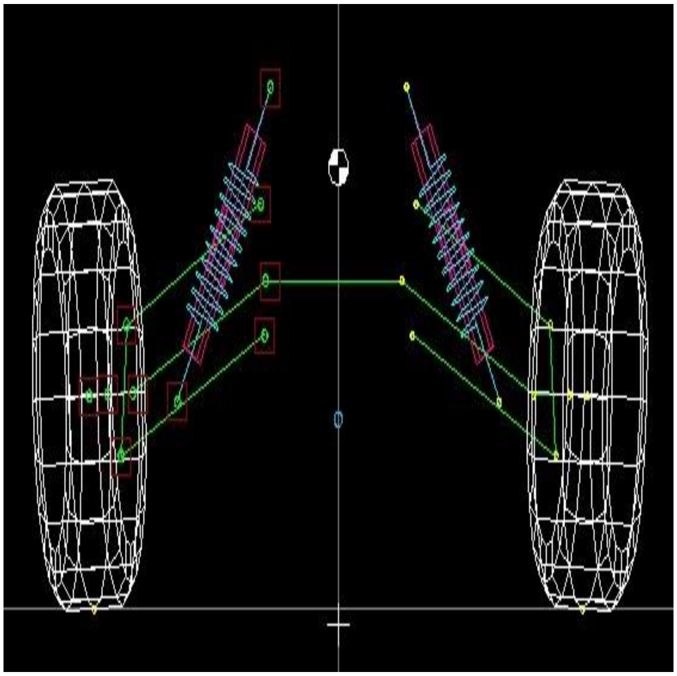


Figure 16.Dynamic Analysis Result Obtained in Steer Test

The performance characteristics of the whole suspension system are as follows:

Parameters	Front	Rear
Shock-absorbers	Coil over spring (customized)	Fox Float 2
Suspension System type	Double wishbone type	Trailing arm with camber links
Motion ratio	0.67	0.87
Static camber	2.5 deg	1.5 deg
Scrub Radius	55.18 mm	-
Ground clearance	12.5 Inch (317.5) mm	

Table 1. Performance Characteristics of the Suspension and Wheels

C. Powertrain Design

A house fabricated trance axle gearbox will be used with a Continuous Variable Transmission (CVT). Gearratios are 10.32 as per our calculations. Engine is mounted on rubber bushings to reduce NVH characteristics.

Using a CVT also enables the increase in the performance of the vehicle.

a). Design Calculations

While designing the whole powertrain, following calculations were carried out: *Drag Force*

Power is needed to counter-act the resistance created by the vehicle moving through the air. The air resistance opposing force is directly proportional to the square of the vehicle's speed. Therefore the drag force can calculated as:

$$Fd=\frac{1}{2}\rho AC_D V - V_w{}^2$$

$$= \frac{1}{2} \times 1.2 \times 0.8 \times 1.2 \times (14.72-3)^2 = 79.1$$

Therefore, Drag Force = 79.11N

b). Tractive Force

The tractive force can be given as,

$$\frac{Te \times G\eta \times Gr}{Rr} = \frac{40.70 \times 0.70 \times 10.32 \times 3.08}{0.2921}$$

Therefore, Tractive Force = 3100

c). Roll Resistance

It is the force resisting the motion when a body (tire) rolls on a surface (road). It is given as:
R_r= (0.015 + 0.00016 x v) W

Where, v = Velocity of the vehicle
W = Weight of the vehicle

Hence, R_r = (0.015 + 0.00016 x 14.72) x (280 x 9.81)

R_r = 47.26

The performance characteristics of the whole powertrain are as follows:

Drag Force		79.11 N
Tractive Force		3100 N
Roll Resistance		47.26 N
Overall Transmission Ratio	Initial	31.78
	Final	3.508
Speed	Initial	4.36 kmph
	Final	53 kmph
Acceleration	Initial	10.66 m/s^2
	Final	1.17 m/s^2
Traction	Initial	3099 N
	Final	342 N
Axle Torque	Initial	1293 Nm
	Final	143 Nm
Gradeability		13 %

Table 2.Performance Characteristics of the Powertrain

D. *Steering & Brakes*

A centered rack & pinion steering assembly is been selected for the buggy. According to the geometry, the steering ratio of about 7:1 was obtained. The rack is placed behind the front wheel's center axis. Taking into consideration, the tangential force applied by the driver, the steering effort is 7.62 Nm.

The Turning Radius Can Be Calculated As:

$$R = \frac{Track\ Width}{2} + \frac{Wheel\ Base}{\sin(avg.\ of\ the\ wheel\ angles)}$$

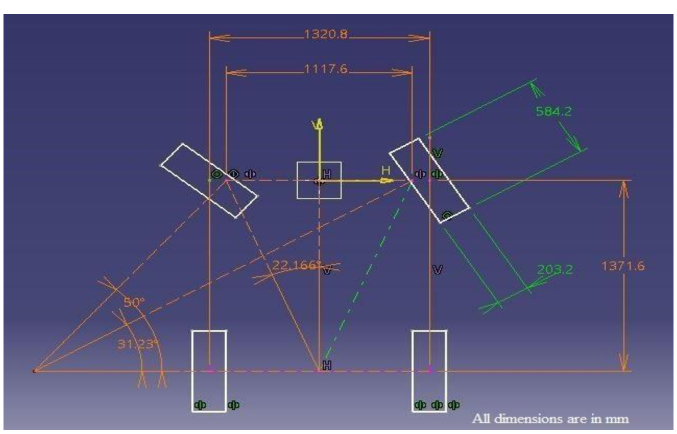


Figure 17.Steering Geometry

In the above diagram we can see the turning angles of the wheels as well as the Ackerman angle of the steering geometry which is 22.16 degrees.

Lock to lock turns	1.2
Steering wheel diameter	12 in
Ackerman percentage	90.10%
Outside wheel angle	31.23^0
Inside wheel angle	50^0
King pin centre to centre distance	44 in
Turning radius	2.767 m

Table 3.Design Parameters of the Steering System

In vehicle hydraulic brake circuit is installed. Brake force is distributed via a TMC. In order to obtain the locking of all four wheels in all road condition independent X split hydraulic brake circuit is applied. Brakes are foot operated. Brake rotor and calipers are the part of un-sprung mass. So it is desirable to reduce the total weight of wheel assembly. We find the greater scope for mass reduction in rotor along with desired structural and thermal characteristics.

a). *Design and Analysis of Brakes*

The changes in axle load during braking depends upon the static laden conditions and deceleration.

b). *Required braking torque*

Let, w1 = Static load on the front wheel = 112 kg

w2 = Static load on the rear wheel = 168 kg

α = retardation of the vehicle = 1 g

W = Weight of the vehicle = 280 kg

h = Distance of C.G from ground = 23.34 in =0.5979 m

L = Wheel Base = 55 in = 1.397 m.

Radius of tyre, R_t = 23 in = 0.5842 m

Dynamic weight transfer,
$Md = (\alpha/g) \times W \times (h/L)$

Front axle dynamic load is given as,
$W_f = w1 + M_d$

Rear axle dynamic load is given as,
$W_r = w2 – M_d$

From analytical calculations we find that dynamic weight transfer is 118.82kg. Dynamic load on front and rare axel is 230.82kg and 49.18kg respectively.

Required braking torque at front wheels,
$Tf=Wf \times (\alpha /g) \times Rt= 134.84$ Nm

Required braking torque at rare wheels,
$T_r =W_r \times (\alpha/g) \times R_t=35.01$ Nm

From above required braking torque is derived.

c). Applied braking torque

Let, Pedal ratio = 1:3
Force applied on TMC = F = 206.01 N

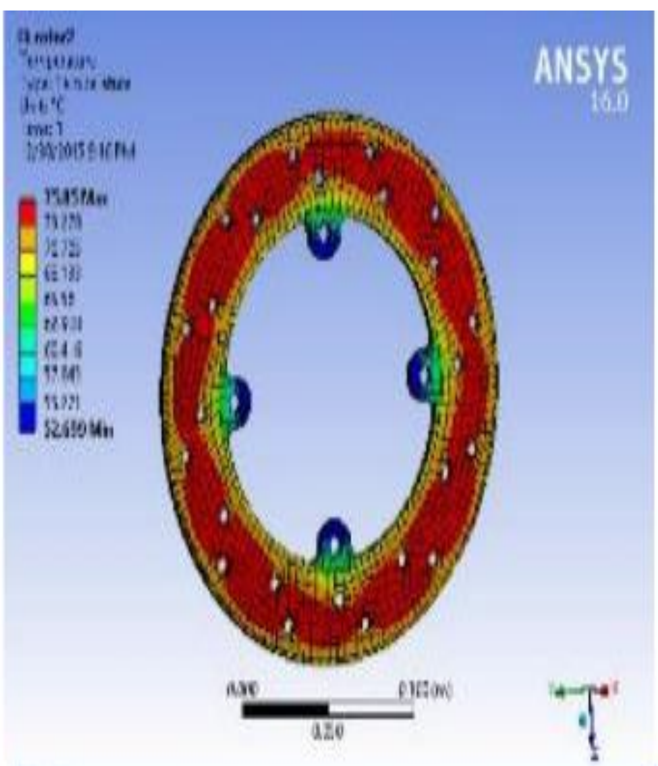


Figure 18.Temperature Distribution In the Brake Disc

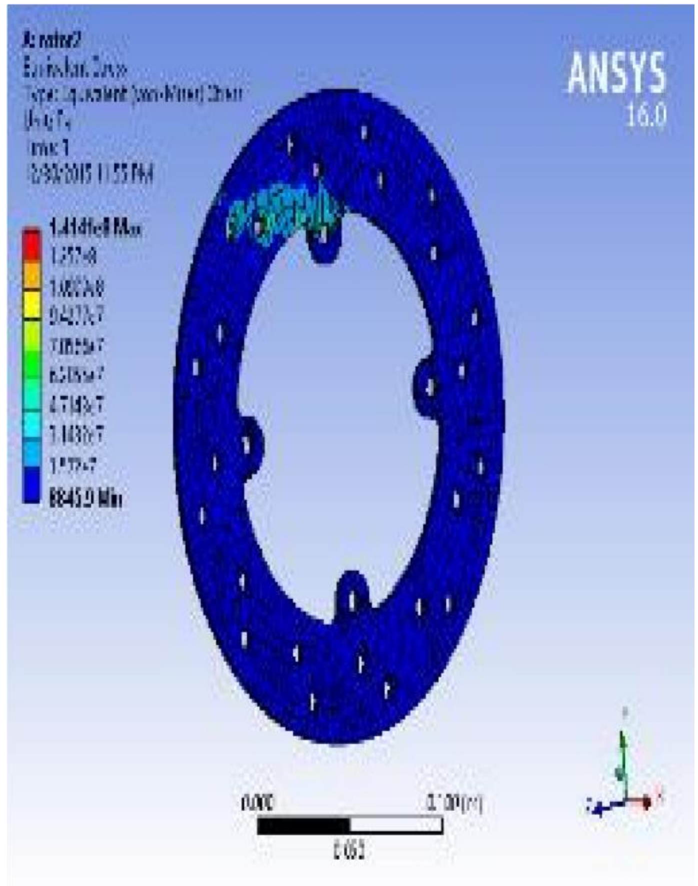


Figure 19.Von Mises Stresses in the Brake Disc

III. VEHICLE TECHNICAL SPECIFICATIONS

Overall Length	77 in
Overall Width	60 in
Overall Height	61 in
Track Width	52 in
Wheel Base	54 in
Curb Weight	225 kg
Front & Rear Wheels	23 x 8 x 12 in
FAW to RAW ratio	0.68

Table 4. Performance Characteristics of the Suspension and Wheels

Bore diameter of Tandem Master Cylinder =0.01905
Pressure generated in TMC = P = 0.689Mpa
Area on caliper = A =1.4135e-3
Coefficient of friction between Rotor and Brake liners = μ = 0.4

The applied torque to stop the vehicle can be calculated as,

$T_B = T = 2\times\mu\times (P\times A) \times R\times$ Number of disc brakes = 686.42Nm

Effective radius = $R_e = 1/3[(D^3-d^3)/(D^2-d^2)\}]$

Where, D = outer diameter of rotor

d = inner diameter of rotor

From above we find the effective diameter of brake rotor 0.1829m. Outer diameter of rotor is decided to be 0.220m. Following results were obtained in the thermal and static structural analysis of the brake disc.

A. 3D View of the Vehicle

Figure 20. 3D View of the Vehicle Model

IV. CONCLUSIONS

In above discussed paper designing of ATV is completed. The design is made to meet the requirements of Driver comfort keeping it as simple and safe. Suspension are designed so as to give greater ride comfort. Maximum possible weight reduction is achieved in the designing of brake rotor for lesser unsprung weight. An overall vehicle is made to sustain the requirement of all possible terrines. Maximum weight reduction is attain for above described configuration of sub systems.

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