

ELECTRIC ATV DESIGN: CONCEPTS AND ENGINEERING CONSIDERATIONS

Joshua Jenkins¹, Isabella Young², Ryan Parker³, and Grace Taylor^{*3}

^{*1}, School of Biomedical Sciences, University of Melbourne, Australia

² Department of Environmental Science, University of California, Berkeley, USA

³ Department of Chemistry, University of Manchester, UK'

ABSTRACT

This design report elaborates the design details taken into consideration and depicts the analysis of the process and sound engineering practices executed in the developing of the Electric ATV vehicle aimed at being very simple yet profound. The Roll Cage is designed for safety in various impacts and with good aesthetics, Steering aimed at minimizing turning radius, Braking to minimize the braking distance and achieving four brake lock simultaneously, Transmission focusing to obtain intermediate gear ratio change between final drive and motor, maximum gradability, maximum acceleration and reliability, Suspension striving for maximum stability of the vehicle. Ergonomic design is the vital element of vehicle and hence the cost effective design is aimed and driver safety is to be the prime objective

KEYWORDS: Rollcage, Ergonomics, suspension, Transmission.

1. INTRODUCTION

This report focus on the design objectives, design calculations, simulation and analysis. To finalize the design, three simple steps were applied to every component of the car- weight reduction, manufacturing feasibility and ease of assembly. Entire vehicle was modeled in ProE, CATIA and SolidWorks. Analysis and optimization of components was done by using ANSYS 14.0. Components such as gearbox, drive shafts, brake calipers, master cylinder, rotors, wheels, tires, and shock absorbers were subjected to design proposal so that each component would satisfy the design goals set by the team as well as rules set by the governing body.

2. Roll Cage

2.1 Design Consideration

We have started our roll cage design by considering some parameters and rules they are listed below.

- Driver ergonomics
- Safety
- Aerodynamics
- Ease of manufacturing
- Cost effectiveness
- Aesthetics

2.2 Objectives of the Roll Cage

We have set some objectives behind our design team to construct optimized roll cage, they are as follow.

- To construct a 3-D space around the driver such that it can accommodate the tallest and healthiest Driver..
- To prevent any failure of the cage's integrity during any dynamic event.
- To support all the sub-systems.
- Light in weight and steady structure.

2.3 Material Properties Selection Criteria

availability, cost, mechanical properties and manufacturing conditions were considered while selecting the material for the Roll Cage.

we must have to select a primary tube material having bending strength and bending stiffness equal to or exceeding that of circular steel of reference material AISI 1018 having circular cross section 25.4mm outer diameter and wall thickness of 3mm containing carbon content of 0.18%, According to these restraints and calculated values, the minimum bending stiffness of 2758Nm² was needed.

From the table, it has been proved that AISI4130 has more bending strength than the required value and easily available in market, hence it is selected for the roll cage for cost effective design. Also for secondary members 4130 is used for high strength design.

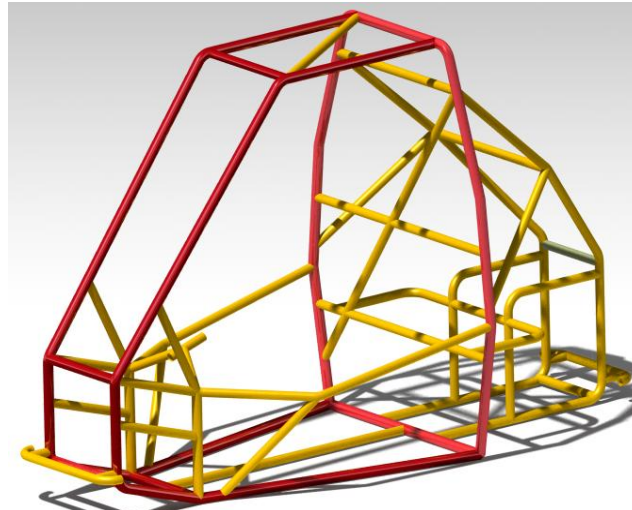


Fig. 2.1. CAD model primary & sec. members.

Table 2.1: Comparison between short-listed materials

| Parameter | AISI 1018 | AISI 1040 | AISI 4130 | AISI 1020 |
|---------------------------|-----------|-----------|-----------|-----------|
| Density Kg/m ³ | 7700 | 7845 | 7700 | 7700 |
| Ultimate Strength MPa | 365 | 518 | 560 | 394.7 |
| Yield strength MPa | 290 | 350 | 460 | 294.8 |
| Young's Modulus GPa | 205 | 210 | 210 | 205 |
| Hardness BHN | 197 | 149 | 156 | 111 |

2.4 Calculation:

For reference material AISI 1018

1) Bending strength(X)=(Sy*I)/C

2) Bending stiffness= Ex*Ix

Where, Thickness = 3mm

Yield Tensile Strength (Sx) = 365 N/mm²

I_x=moment of inertia=13478.6378mm⁴

Outer Diameter (do) = 25.4mm

Inner Diameter (di) = 19.4mm

Centre Distance (c) = 12.7mm

Modulus of Elasticity (Ex) = 205*10³N/mm²

$$\text{Bending strength} = 365 * \frac{\pi}{64} * (d_2^4 - d_1^4) * 12.5 = 387 \text{ Nm}$$

Bending stiffness (Ex*I_x)=2758Nm²

We have done various calculations for comparing stiffness, bending strength and weight for AISI4130 material of available dimensions in market.

Table 2.2:-Calculations of selected material for various dimensions.

| D | T | M.M.I | Yeild strength | γ | Bending Strength | Youngs Modulus | Stiffness | Density | Mass | Material |
|-------|------|------------|----------------|-------|------------------|----------------|-----------|---------|--------|----------|
| 25.4 | 3 | 13478.6388 | 0.365 | 12.7 | 387.3782 | 0.205 | 2763.12 | 7850 | 1.6636 | 1018 |
| 25.4 | 0.89 | 5152.9043 | 0.46 | 12.7 | 186.6497 | 0.21 | 1082.11 | 7850 | 0.538 | 4130 |
| 25.4 | 1 | 5723.6514 | 0.46 | 12.7 | 207.313 | 0.21 | 1201.97 | 7850 | 0.6017 | |
| 25.4 | 1.2 | 6695.0493 | 0.46 | 12.7 | 242.193 | 0.21 | 1405.96 | 7850 | 0.7162 | |
| 25.4 | 1.6 | 8508.8152 | 0.46 | 12.7 | 308.193 | 0.21 | 1786.22 | 7850 | 0.9391 | |
| 25.4 | 2 | 10136.7445 | 0.46 | 12.7 | 367.157 | 0.21 | 2128.72 | 7850 | 1.1542 | |
| 25.4 | 2.5 | 11930.31 | 0.46 | 12.7 | 432.121 | 0.21 | 2505.37 | 7850 | 1.4119 | |
| 29.21 | 0.89 | 7201.937 | 0.46 | 14.6 | 226.91 | 0.21 | 1512.41 | 7850 | 0.8623 | |
| 29.21 | 1 | 8827.0285 | 0.46 | 14.6 | 278.11 | 0.21 | 1853.68 | 7850 | 0.978 | |
| 29.21 | 1.6 | 13268.9 | 0.46 | 14.6 | 417.918 | 0.21 | 2786.47 | 7850 | 1.091 | |
| 29.21 | 2 | 15907.9974 | 0.46 | 14.6 | 501.21 | 0.21 | 3340.68 | 7850 | 1.3441 | |
| 29.21 | 2.25 | 17434.78 | 0.46 | 14.6 | 549.314 | 0.21 | 3661.3 | 7850 | 1.456 | |
| 31.75 | 0.89 | 10280.1328 | 0.46 | 15.88 | 297.783 | 0.21 | 2158.83 | 7850 | 0.6773 | |
| 31.75 | 1 | 11430.2124 | 0.46 | 15.88 | 331.101 | 0.21 | 2400.34 | 7850 | 0.7583 | |
| 31.75 | 1.21 | 13556.0623 | 0.46 | 15.88 | 392.68 | 0.21 | 2846.77 | 7850 | 0.9113 | |
| 31.75 | 1.6 | 17268.8399 | 0.46 | 15.88 | 500.23 | 0.21 | 3626.46 | 7850 | 1.1897 | |
| 31.75 | 2 | 20773.8399 | 0.46 | 15.88 | 601.761 | 0.21 | 4362.51 | 7850 | 1.4674 | |

With the above mechanical properties and calculated values, we have come to conclude that AISI4130 having outer diameter of 29.21mm and wall thickness of 1.6mm as a primary member and 25.4mm outer diameter and 1mm wall thickness as secondary members combination for optimized, cost effective and lesser weighing structure.

2.5 Pipe selection

Primary members=AISI 4130 having outer dia. 29.21mm and wall thickness of 1.6mm.

Secondary members= AISI 4130 having outer dia. 25.4mm and wall thickness of 1mm

2.6 Design methodology:

Steering and Motor Accommodation: Front cross members of width 12” were selected for easy accommodation of steering rack, for accommodation of the two pedals and the spacing was comparable to that of the conventional cars. We designed the rear half to fit the motor, batteries, motor controller, IEMS, charger and the rest of the drive train.

2.7 Ergonomics

- Provided firewall angle as 0° to maintain the cg of the vehicle.
- **Comfortable cockpit:** The distance between the SIM has been increased to 32 inches which will tapered towards the foot box, which will result in the driver being more comfortable.
- **Foot box space:** The size of the foot box has been increased to a maximum length and maximum width so that the driver is in a much more comfortable position during driving.
- FBM were tapered for easy and fast escape of driver from the vehicle.
- Provided perfect FBM inclination as 30°.

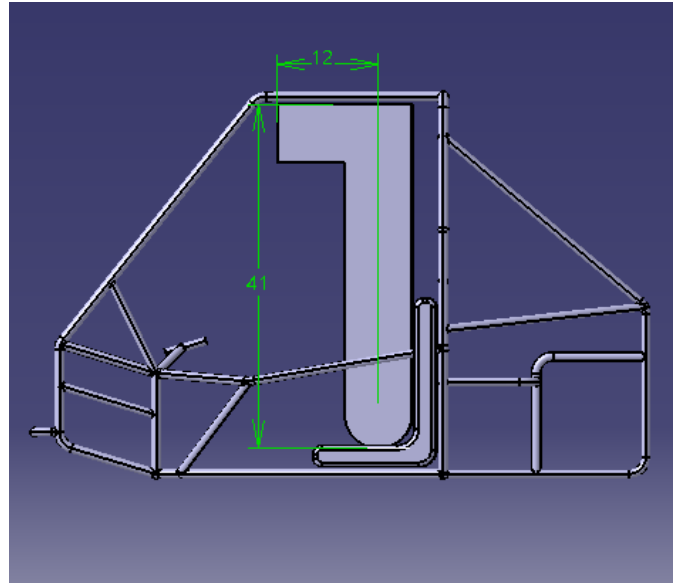


Fig. 2.2 CAD model with template.

2.8 Aerodynamics:

This time we have designed our roll cage without nose and given the caster angle of 17° to get best aerodynamic structure as well as to get best performance of suspension.

2.9 FEA Results

The purpose of analysis is to get stiff roll cage by subjecting it to various conditions and determining the structural stress.

Finite element analysis of the chassis was done in ANSYS14.0 workbench.

The impulse momentum equation was used to calculate force of impact.

$$F \times t = M \times (V_i - V_f)$$

Where:

M - Mass of the vehicle with driver

V_i - Initial velocity of the vehicle

V_f - final velocity of the vehicle

t - Time of impact = 1 sec

F - Force of impact

$$F = \{M \times (V_i - V_f)\} / t$$

$$= 9.81 \times \{300 \times (40 - 36.15)\} / 1$$

$$= 11330.55\text{N}$$

Force applied: 11.33KN

For FEA the vehicle was consider as running at 40 Kmph and conducted static structural test.

2.9.1 Front Impact

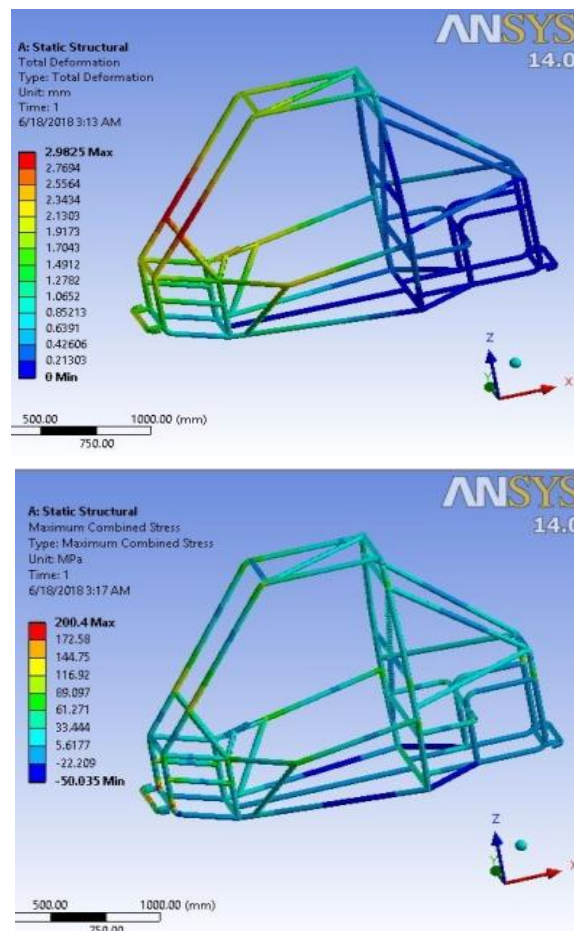


Fig. 2.4. Front Impact Analysis

| | |
|-------------------|-----------------------------|
| Types of analysis | Front impact |
| No. of nodes | 5802 |
| No. of elements | 2918 |
| Element size | 5mm |
| Time of impact | 1 |
| Constraints | Rear suspension pts fixed |
| Max Forces | 4g force on front most pts. |
| Deformation(mm) | 2.9825 |
| Stresses(Mpa) | 200.4 |
| FOS | 2.8 |

2.9.2 Rear Impact Analysis

2.9.3

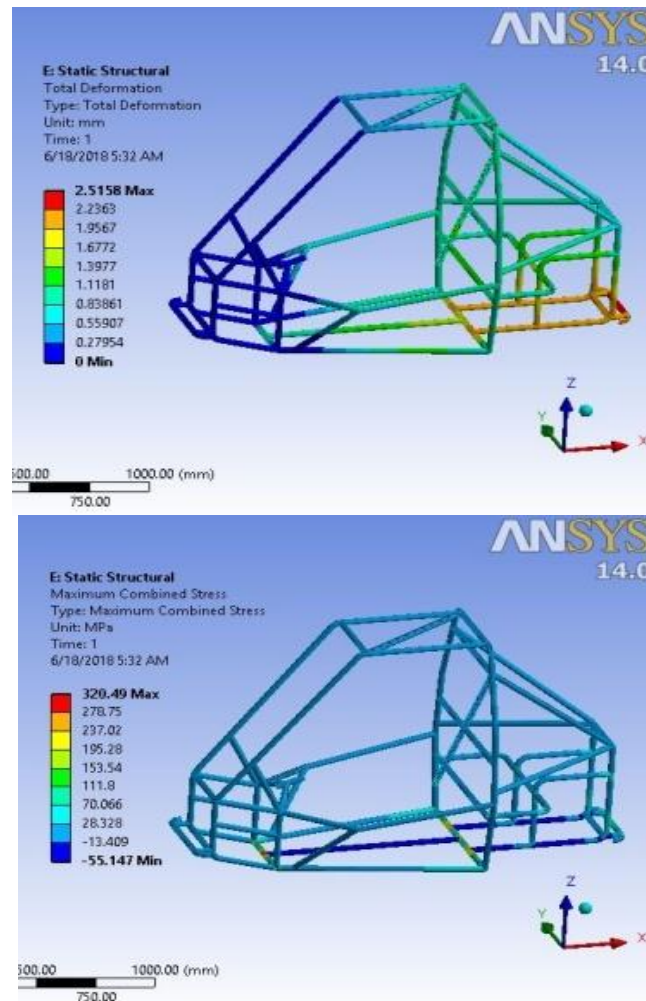


Fig. 2.5. Rear Impact Analysis

| | |
|--------------------|-------------------------------|
| Types of analysis | Rear impact |
| Number of nodes | 5802 |
| Number of elements | 2918 |
| Element size | 10mm |
| Time of impact | 1sec |
| Constraints | Front suspension points fixed |
| Max Forces | 4g force on rear most points |
| Deformation(mm) | 2.5158 |
| Stresses(Mpa) | 320.49 |
| FOS | 1.75 |

2.9.3 Side Impact Analysis

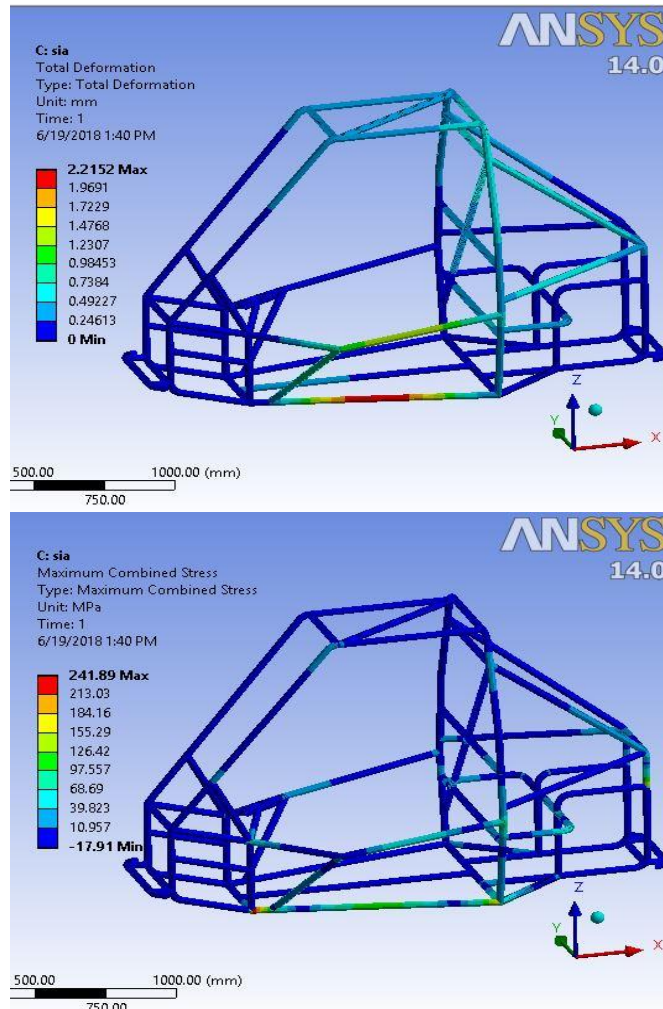


Fig.2.6 Rear Impact Analysis

| | |
|-------------------|-------------------------------------|
| Types of analysis | Side impact |
| No. of nodes | 5802 |
| No. of elements | 2918 |
| Element size(mm) | 10 |
| Time of impact | 1sec |
| Constraints | Side most pts fixed |
| Max Forces | 2g force on opposite side most pts. |
| Deformation(mm) | 2.2152 |
| Stresses(Mpa) | 241.89 |
| FOS | 2.32 |

2.9.4 Torsional Analysis

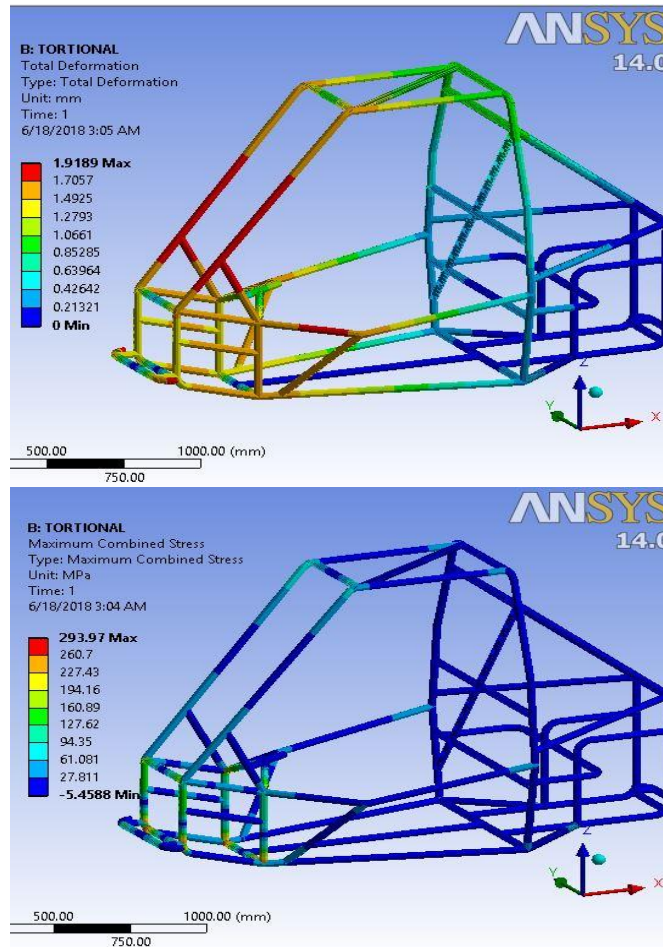


Fig.2.7 Torsional Analysis

| | |
|-------------------|---|
| Types of analysis | Torsional analysis |
| No. of nodes | 5802 |
| No. of elements | 2918 |
| Element size | 10mm |
| Time of impact | 1 sec |
| Constraints | Rear suspension points fixed |
| Max Forces | 4g force on front most either side of suspension points |
| Deformation(mm) | 2.9189 |
| Stresses(Mpa) | 291.97 |
| FOS | 1.91 |

Inference: It can be inferred from the respective tables that the stress values obtained are below the allowable stress values for the material AISI 4130 and the deformation is in the tolerable range.

3. STEERING SYSTEM

Steering system is used for controlling the directional characteristics and the stability of the vehicle. The steering system of an ATV is designed according to the specifications for the worst possible terrain or geographical profile. All the forces and torques encounter during the run are considered in order to design the mechanisms which will sustain these worst case scenarios. We have selected rack and pinion steering system for our ATV. The main objective behind designing and manufacturing rack and pinion steering system for an all-terrain vehicle is to decrease the weight of the system, to make the system efficient in terms of space considerations and to customize the specifications as required.

3.1 Objectives:

- Minimum turning radius
- Minimum steering effort while cornering the vehicle.
- Steering system should be light weight
- To make system efficient in space consideration

3.2. Design methodology:

Table 3.1. Input Parameter

| Sr. No. | Parameter | Value |
|---------|---------------------|-----------------|
| 1. | Wheel Track | 1244.6mm |
| 2. | Wheel Base | 1447.8mm |
| 3. | Castor Angle | 10 ⁰ |
| 4. | Kingpin Inclination | 8 ⁰ |
| 5. | Scrub Radius | 79.44 Mm |

3.2.1 Geometry setup:

It is the most important step of designing because steering geometry actually decides the maneuverability of vehicle during different driving conditions. Initially we obtained various parameters like wheel base, wheel track and pivot to pivot distance from suspension department. We were left with the parameters like steering arm length, Ackermann angle. So we performed multiple iterations to find optimized steering geometry.

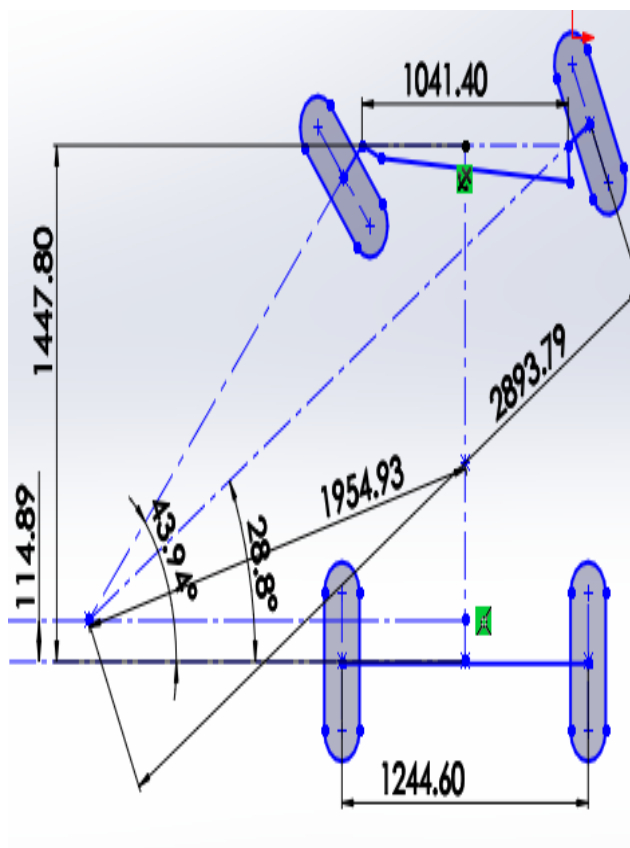


Fig. 3.1. Steering Geometry

Best optimal results from steering geometry are as:

Table 3.2. Steering Geometry Parameter

| | |
|---------------------|--------------------|
| Inside Wheel angle | 43.66 ⁰ |
| Outside wheel angle | 27.34 ⁰ |
| Steering arm length | 3.96'' |

| | | |
|-----------------------|--------|-------------|
| Ackermann angle | 28° | |
| Turning circle radius | C.G. | Front outer |
| | 1.99 m | 2.94 m |
| Ackermann % | 88.01 | |

3.2.2 Design Calculation:

Rack Travel (centre to lock) = $r\theta$
 $= 100.68 \times 43.66 \times \frac{\pi}{180}$
 $= 76.71 \text{ mm}$

Assume, Lock to lock turns: 1.5 Turns

∴ Steering Wheel angle (lock to lock)
 $= 1.5 \times 360 = 540^\circ$

∴ Rack travel per rev = $\frac{76.71}{0.75} = 102.29 \text{ mm}$

But rack travel per revolution = $2 \times \pi \times R = 102.29 \text{ mm}$

where R= Radius of pinion

∴ R= 16.28 mm; ∴ D= 32.56 mm

Assume no of teeth on pinion, T = 18

∴ Module , m = $\frac{D}{T} = \frac{32.56}{18} = 1.8$

by selecting next series module, m = 2

∴ Diameter of pinion, D = 36 mm

Steering Ratio = $\frac{\text{Steering wheel angle}}{\text{Road wheel angle}}$
 $= \frac{270}{\frac{43.66 + 27.34}{2}} = 7.6 : 1$

Total eye to eye length = 14”

Steering system forces and moment:

1. Aligning torque required to overcome the Steering axis inclination (SAI).
2. Aligning torque required to overcome the castor trail.
3. Friction couple generated while cornering
4. Moment due to friction couple
5. Lateral force

By considering all this forces and moment acting on steering we got tangential force acting on rack and pinion

∴ $F_t = 684.57 \text{ N}$

∴ Torque acting on pinion = $F_t \times \text{Radius of pinion}$
 $= 684.57 \times 18$
 $= 12322.26 \text{ N.mm}$
 $= 12.32 \text{ N.m}$

Assume Steering wheel diameter = 220 mm

torque acting on pinion = Torque acting on steering wheel

= Steering effort × radius of steering wheel radius
 $12322.26 = \text{steering effort} \times 110$

∴ Steering effort = 112.02 N

So, in order to decrease the steering effort we have selected the steering wheel Diameter as 260 mm.

∴ Steering effort = $\frac{12322.32}{130} = 94.78 \text{ N}$
 $= 9.66 \text{ Kg.}$

3.3 CAD models

We did virtual prototyping using CAD software Solid works 2016. As per the dimension obtain by precise calculations.

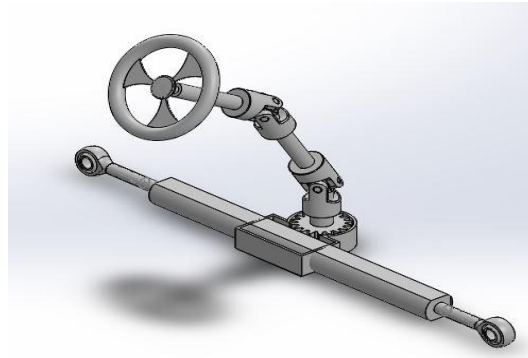


Fig3.2 . Steering assembly

3.4 FEA

Material selection

The materials used in the steering system targets, precise operation and light weight components. Although precision and weight are the top priorities, cost, manufacturability, and reliability were also considered. The use of aluminum offers lesser weight of the assembly.

After considering all these parameters, it was observed that Aluminum 7075 T6 provides a required strength, it is easily available and excellent machinability hence we decided to use it for our purpose.

Table 3.3 Properties of Al 7075 T6

| Parameter | Value |
|---------------------------|------------------------|
| Density | 2810 Kg/m ³ |
| Ultimate tensile strength | 541 Mpa |
| Yield strength | 468 Mpa |
| Modulus of elasticity | 71.7 Gpa |
| Poisson's ratio | 0.33 |

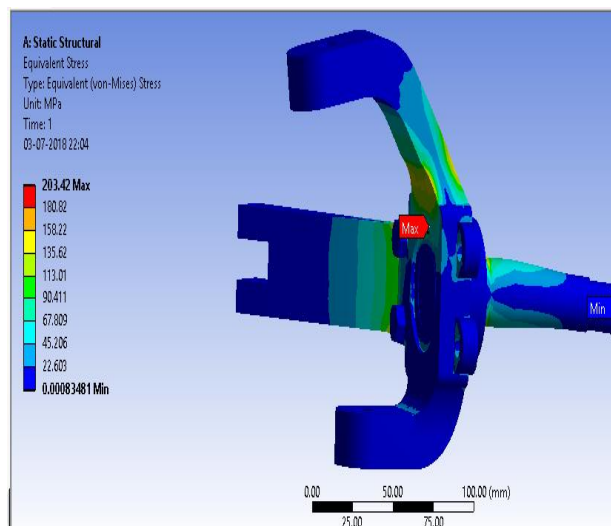


fig 3.3. FEA of Knuckle

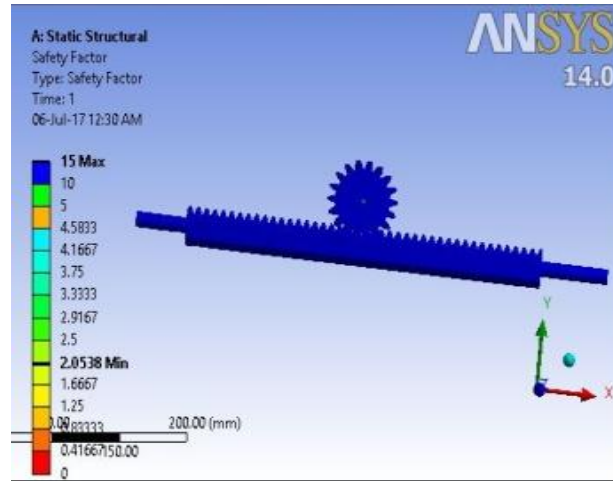


fig 3.4. FEA of Rack & Pinion

| Sr no. | Parameter | Knuckle | Rack & pinion |
|--------|---|-------------------|---------------|
| 1 | Analysis type | Static structural | |
| 2 | No. of nodes | 10579 | 22052 |
| 3 | No. of elements | 5835 | 12126 |
| 4 | Element size | 10 | 5 |
| 5 | Time of impact | 1 | 1 |
| 6 | Maximum force | 4375.26N | 700 N |
| 7 | Max equivalent stress N/mm ² | 293.42 | 84.255 |
| 8 | Max def. (mm) | 1.262mm | 0.062 |
| 9 | F.O.S | 2.48 | 2.0538 |

4. BRAKING SYSTEM

An excellent braking system is the most important safety feature of any land vehicle. Competition regulations require at least two separate hydraulic braking systems, so that in the event of a failure of one, the other will continue to provide adequate braking power to the wheels. The main requirement of the vehicle’s braking system is that it must be capable of locking all four wheels on a dry surface. Ease of manufacturability, performance and simplicity are a few important criteria considered for the selection of the braking system.

4.1. Objectives

The goals for the braking system were:

1. Reduce weight in the overall system.
2. Simultaneous locking of all four wheel.
3. Minimum stopping distance.

4.2. Design Approach

The two main types of braking systems under consideration were Drum and Disc brakes. But in case of drum braking there is a high possibility of heat generation and brake ventilation is poor leads to brake fade. The disc brake is well ventilated and overcomes the drawback of brake fade. So, we have decided to use hydraulic disc brakes in the front and the rear of the ATV. Because it has good heat transfer capacity and high speed to overcome fade associated with high temperature developed. We assume static weight distribution of 40:60. X split hydraulic system is used which is more significant for the event to overcome hydraulic system failure.

4.3. Customized Brake Rotors

Rotors were designed as per design requirement in its shape, size and materials and also which are suitable for the brake caliper.

Taking all the consideration factors, Front rotors were selected as 175 mm OD and 100 mm ID and also Rear rotors were selected as 165mm OD and 100 mm ID respectively. SS410 is selected as rotor material. Rotors were analyzed using Thermal Analysis and Static structural analysis using Ansys Workbench.

4.3.1. Static Structural Analysis

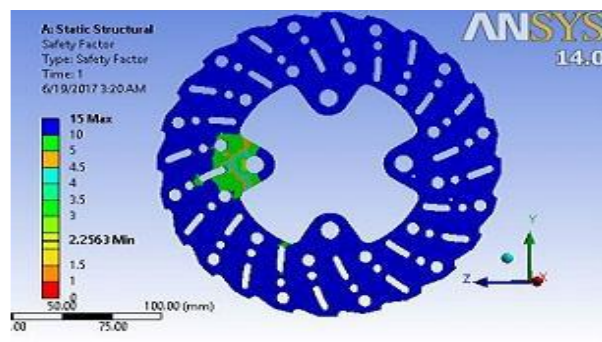


Fig 4.1 Static Structural Analysis of front

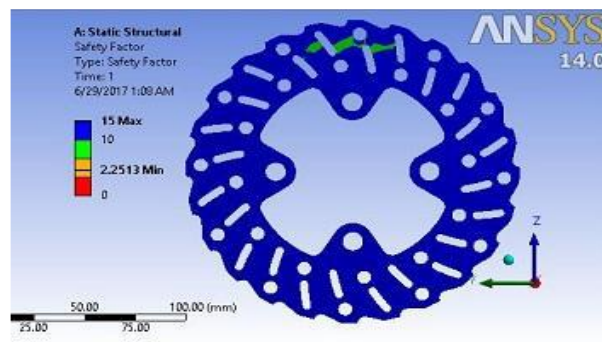


Fig 4.2 Static Structural Analysis of rear

| Sr No. | Parameter | Value | |
|--------|-----------------|--------|--------|
| | | Front | Rear |
| 1 | No. of Nodes | 21552 | 15042 |
| 2 | No. of elements | 8845 | 7097 |
| 3 | Element Size | 5 | 5 |
| 4 | Maximum Force | 8927 | 8927 |
| 5 | Fos | 2.25 | 2.2 |
| 6 | Stress(MPa) | 84.396 | 86.88 |
| 7 | Deformation(mm) | 0.9945 | 0.2307 |

4.3.2. Thermal Analysis

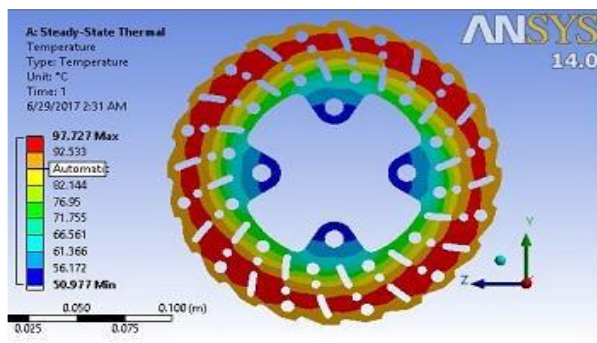


Fig 4.3 Thermal Analysis of Front rotor

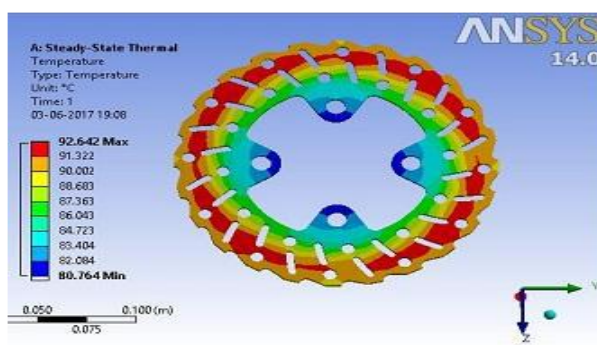


Fig 4.4 Thermal Analysis of Rear rotor

| Sr No. | Parameter | Value | |
|--------|-----------------|-------|-------|
| | | Front | Rear |
| 1 | No. of Nodes | 20447 | 15618 |
| 2 | No. of elements | 9789 | 7427 |
| 3 | Element Size | 5 | 5 |
| 4 | Maximum Force | 8927 | 8927 |
| 5 | Time of impact | 1 | 1 |
| 6 | Maximum Temp. | 97.72 | 92.64 |

Table 4.1 Selection of Braking components:

| Parameter | | Value |
|---|----------------|---------------|
| Brake fluid | | BOSCH DOT-3 |
| Caliper | Apache RTR 160 | Royal Enfield |
| Dia. Of Caliper Piston(mm) | | 30 |
| Tire Outer Diameter (mm) | | 584.2 |
| No. of piston in caliper cyl. | | 2 |
| BMS bore size* stroke (mm) | | 19.5*55 |
| Area of Calliper pad (cm ²) | | 17.84 |
| Tandem Master Cylinder | | Maruti omni |
| Static loading radius of tire(mm) | | 280.41 |

4.4 Braking Calculation:

- **Inputs**
 Mass of vehicle= 300 kg
 $\mu_{road}=0.6$ $\mu_{pad}=0.4$

- CG Height = 571.5 mm
- Pedal Ratio= 4.5:1 Pedal Effort= 200 N
- Weight transfer = $\frac{(\mu_{road} * mass * c.g. height)}{wheel\ base} = 75kg$
- Dynamic Load:
 Front= 0.4 * mass + weight transfer = 195kg
 Rear = 0.6 * mass - weight transfer = 105kg
- Braking Force at Front(BF_f)= FOS × μ_{road} × Dynamic Load Front × g = 1.5*0.6*195*9.81 = 1721.65 N
- Braking Force at Rear(BF_r)= FOS × μ_{road} × Dynamic Load Rear × g = 1.5*0.6*105*9.81 = 927.04 N
- Deceleration(a)= $\frac{(BF\ at\ Front + BF\ at\ Rear)}{m}$
 = (1721.65+927.04)/300
 = 8.82 m/s²
- Deceleration Time= $\frac{v}{a} = \frac{12.5}{8.82} = 1.38\ s$
- Stopping Distance= $\frac{v^2}{2a} = \frac{12.5^2}{2*8.82} = 8.2\ m$
- Effective Radius of Disc at Front(R_f)=
 $\left(\frac{1}{3}\right) \times \frac{[(Front\ OD^3)-(Front\ ID^3)]}{[(Front\ OD^2)-(Front\ ID^2)]} = 70.45\ mm$
- Effective Radius of Disc at Rear (R_r)= 67.578 mm
- Torque acting on front wheel: -
 = $\frac{BF_f * radius\ of\ tire}{2} = 251.44Nm$
- Torque acting on rear wheel: -
 = $\frac{BF_r * radius\ of\ tire}{2} = 135.39Nm$
- Required Clamping force on front=
 $\frac{Braking\ torque\ on\ front}{\mu_{pad} * Re\ front * n} = 7813.45\ N$
- Required Clamping force on Rear=
 $\frac{Braking\ torque\ on\ Rear}{\mu_{pad} * Re\ rear * n} = 4363.47$
- Pressure acting in master cylinder: -
 = $\frac{Pedal\ effort * pedal\ ratio}{piston\ area\ of\ master\ cylinder} = 3.1576\ mpa$
- Actual Clamping Force on rotors(F_c):-
 = 2*Pressure×Area of caliper piston
 = 3.1576×2×($\frac{\pi}{4}$)×30²
 = 8927.9 N
- Torque acting on front rotor:
 = μ_{pad} * front effective radius*F_c
 = 251.58 Nm
- Torque acting on rear rotor :
 = μ_{pad} * rear effective radius*F_c
 = 241.30 Nm

5. SUSPENSION DEPARTMENT

The suspension of an off road vehicle plays a crucial role in maintaining traction between tires and road.

5.1 Design specification

In front, unequal parallel double wishbone type suspension is incorporated which provides flexibility in design for required roll center height, camber gain for efficient cornering and low un-sprung weight. Custom manufactured knuckle and hub are used considering design flexibility, rigidity and weight reduction. Light weight ball joints are selected by considering its maximum angular displacement. In rear, modified Double wishbone with combination of H and A arm type suspension is used. In rear wheel assembly, customized hub and spacer are used to transmit power with less un-sprung weight.

Table 5.1. Suspension parameters

| Parameter | Front | Rear |
|-----------|-------|------|
|-----------|-------|------|

| | | |
|--------------------|------------------------|------------------------------|
| Track width | 52in | 46in |
| Ground clearance | 355.6mm | 355.6mm |
| Arm length | 17.32in | 15.22in |
| Type of suspension | Parallel unequal A-arm | Non Parallel unequal H&A arm |
| Material of arms | AISI 4130 | AISI 4130 |
| Motion ratio | 0.7 | 0.67 |
| Roll centre height | 253.6mm | 284mm |
| Sprung/unsprung | 223.7kg | 76.3kg |
| Spring stiffness | 10.86N/mm | 21.72N/mm |

In order to analyze the kinematics of the suspension, Lotus Shark Software was used. This suspension software allowed us to input suspension points for various geometries. Once the points were given as input, it would display, graphically and visually, how the suspension would move as the vehicle cornered or hit a bump. Suspension point were the outcomes of the iteration done by the Lotus Shark Software and steering system members. In this software we obtained various graph like wheel travel, rolling, steering vs camber change, toe angle, half-track change, roll center variation. This helped us in finding the hard point of the system.

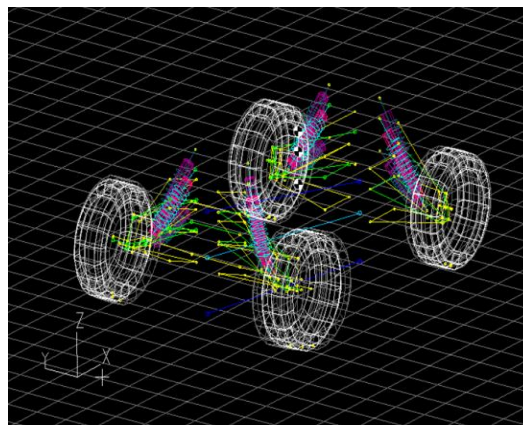


Fig.5.1 Simulation through software

Table 5.2 Static suspension parameters:

| | |
|--------------------|---------|
| CG height | 571.4mm |
| Camber at kerb wt. | -0.5° |
| Castor | 10° |
| Static Toe | 0.7° |
| KPI | 8° |
| Scrub radius | 79.44mm |
| Camber gain | 2° |
| Roll gradient | 2.29°/g |

5.2 Design consideration

Kerb weight(m)= 300 kg
 Wheel travel (w.t)= 8 inch
 Weight distribution (W.d)=65:35
 Dynamic load factor(L)=3

5.3 Design calculation

Spring calculation for front spring
 Material for spring: Chrome-Vanadium

$T_{\max(\text{material})} = 589.5 \text{ N/mm}^2$

$G = 80000 \text{ Mpa}$

- $F_{\max} = \frac{m \cdot W \cdot d \cdot g \cdot L}{2}$

$$=1545.07 \text{ N}$$

- Spring travel(δ_{\max}) = w.t x M.R.
=142.24mm

Assume d= 9, D=81 so C=9;

- $$T_{\max. (design)} = \frac{8 \cdot F_{max} \cdot c \cdot K_w}{\pi \cdot d^2}$$

$$K_w = \frac{4 \cdot C - 1}{4 \cdot C - 4} + \frac{0.615}{C}$$

Hence $K_w = 1.162$

$$T_{\max. (design)} = 507.03$$

As $T_{\max(design)} < T_{\max(material)}$

So design is Safe.

- Spring Rate(K) = $\frac{F_{max}}{\delta_{\max}}$
 $K = 10.862 \text{ N/mm}$

- Number of Turns(N) = $\frac{G \cdot d}{8 \cdot C^3 \cdot K}$

N=12 turns

For square and grounded end

$$N' = N + 2$$

$N' = 14$ turns

- Solid Length(L_s) = $N' \cdot d = 130 \text{ mm}$

- Free length = $L_s + (1.15 \times \delta_{\max})$

$$L_f = 293.57 \text{ mm}$$

- Pitch = $\frac{L_f - 2 \cdot d}{N}$

$$p = 22.63 \text{ mm}$$

- Natural Frequency(F_n) = $\frac{w_n}{2 \cdot \pi}$

$$w_n = \sqrt{\frac{K}{m}} = 14.32 \text{ rad/sec}$$

- $F_n = \frac{14.38}{2 \cdot \pi}$

$$= 2.28 \text{ Hz}$$

Spring calculation for rear spring

Material for spring: Chrome-Silicon

$$T_{\max(material)} = 729 \text{ N/mm}^2$$

- $$F_{\max} = \frac{m \cdot W \cdot d \cdot g \cdot L}{2}$$

 $= 3090.15 \text{ N}$

- Spring travel(δ_{\max}) = w.t x M.R.
=142.24mm

Assume d = 11mm, D= 88mm so C=8;

- $$T_{\max. (design)} = \frac{8 \cdot F_{max} \cdot c \cdot K_w}{\pi \cdot d^2}$$

$$K_w = \frac{4 \cdot C - 1}{4 \cdot C - 4} + \frac{0.615}{C}$$

Hence $K_w = 1.184$

$$T_{\max. (design)} = 615.91$$

As, $T_{\max(design)} < T_{\max(material)}$

So design is Safe.

- Spring Rate(K) = $\frac{F_{max}}{\delta_{\max}}$
 $= 21.72 \text{ N/mm}$

- Number of Turns(N) = $\frac{G \cdot d}{8 \cdot C^3 \cdot K}$

N = 10 turns

For square and grounded end

$$N' = N + 2$$

$N' = 12$ turns

- Solid Length(L_s) = $N' \cdot d = 132 \text{ mm}$

- Free length(L_f) = $L_s + (1.15 \times \delta_{\max})$

- $L_f = 295.57 \text{ mm}$

- $Pitch = \frac{Lf - 2*d}{N}$
 $p = 27.35mm$
- Natural Frequency(Fn) = $\frac{wn}{2*\pi}$
 $wn = \sqrt{\frac{K}{m}} = 14.89rad/sec$
 $F_n = \frac{14.92}{2*\pi}$
 $= 2.37Hz$

5.4 FEA Analysis

Finite element analysis was performed on suspension parts to ensure safety, durability, minimum weight, possibility of failure. Vehicle was analyzed for the worst condition i.e. free fall from 2 m height on one wheel, sudden braking and cornering. Accordingly, forces were calculated and applied on respective parts. Stresses and deformation on the parts were analyzed and required changes were done to get optimum design.

Control arms

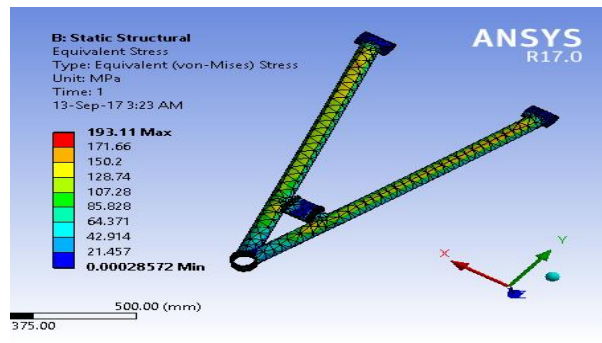


Fig.5.2 FEA of front arm

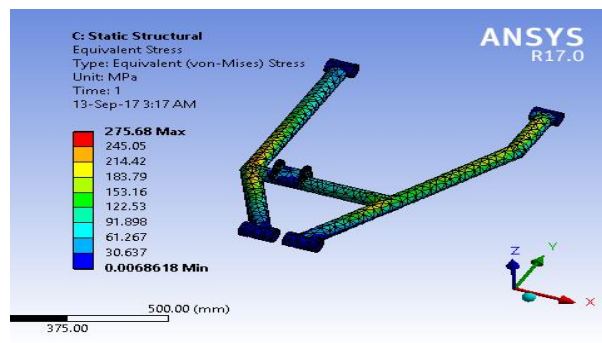


Fig.5.3 FEA of Rear arm

| Sr no. | Parameter | Front | rear |
|--------|-----------------------|-------------------|--------|
| 1 | Analysis type | Static structural | |
| 2 | No. of nodes | 7063 | 7039 |
| 3 | No. of elements | 3355 | 3323 |
| 4 | Element size | 10 mm | 10mm |
| 5 | Time of impact | 1 | 1 |
| 6 | Maximum force | 4G | 4G |
| 7 | Max equivalent stress | 193.11 | 275.68 |
| 8 | Max deformation | 1.9mm | 3mm |
| 9 | F.O.S | 2.5 | 2 |

Springs

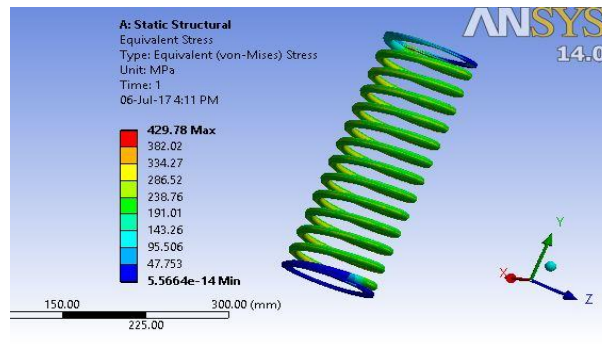


Fig.5.4 Front Spring

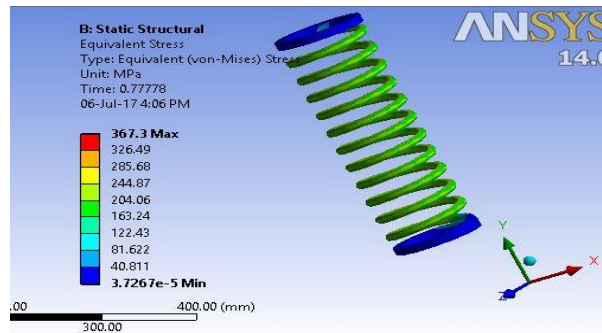


Fig.5.5 Rear Spring

| Sr no | parameter | Front | Rear |
|-------|-----------------------|-------------------|------------|
| 1 | Analysis type | Static structural | |
| 2 | No. of nodes | 19143 | 20344 |
| 3 | No. of elements | 8381 | 8489 |
| 4 | Element size | 10 | 10 |
| 5 | Time of impact | 1 | 1 |
| 6 | Max. force | 1.5G | 1.5G |
| 7 | Max equivalent stress | 429.78 N/mm | 367.3 N/mm |
| 8 | Max deformation | 140mm | 135mm |
| 9 | F.O.S | 1.18 | 1.21 |

6. TRANSMISSION SYSTEM

Transmission act as link between drive motor and the drive wheels to enable vehicle to move off from the rest.

6.1 Selection of gear ratio

Tractive effort required by the vehicle

Rolling Coefficient (μ) = 0.04

Wheel radius (r) = 0.2921 m,

D = 0.5842 m

Gradient (θ) = 20°

Mass of vehicle(m) = 300 kg(approx.)

Air Density (ρ) = 1.125 Kg/m³

Frontal Area (A) = 1.089 m²

Vehicle Velocity (V) = 25 Km/h = 6.94 m/s

Air Drag Coefficient (Cd) = 0.4

Max. Motor RPM (N_{max}) = 5000 rpm

Motor Torque (M_t) = 25N.m
 Tractive Effort = RR + GR + AR
 $= \mu \cdot m \cdot g \cdot \cos\theta + m \cdot g \cdot \sin\theta + \frac{1}{2} \rho A V^2 C_d$
 $= 1128.185N$
 Acceleration = $\frac{M_t \cdot G R \cdot \eta_{\text{gearbox}}}{R_{\text{wheel}} \cdot m}$
 $= (25 \cdot 13.5 \cdot 0.92) / (0.2921 \cdot 300)$
 $= 3.54 \text{ m/sec}^2$

Table 6.1 Selection of gear ratio

| Gear ratio | Speed (km/hr) | Wheel Torque (N.m) | Tractive Effort (N) | Accl. (m/sec ²) |
|-------------|---------------|--------------------|---------------------|-----------------------------|
| 10 | 55.05 | 250 | 855.8 | 2.62 |
| 11 | 50.05 | 275 | 941.45 | 2.88 |
| 12 | 45.88 | 300 | 1027 | 3.14 |
| 13 | 42.35 | 325 | 1112.6 | 3.41 |
| 13.5 | 40.78 | 337.5 | 1155.42 | 3.54 |
| 14 | 39.32 | 350 | 1198.21 | 3.67 |

The 13.5:1 gear ratio is selected because it provides required traction and speed for the vehicle.

6.2 Selection of tyre

Table 6.2 Selection of Tyre

| Tyre size | Speed (km/hr) | Traction (N) | Accel ⁿ m/sec ² |
|------------|---------------|----------------|---------------------------------------|
| 22" | 39.01 | 1207.94 | 3.7 |
| 23" | 40.78 | 1155.42 | 3.54 |
| 24" | 42.55 | 1107.2 | 3.39 |

We have selected the "23*7*10" tire for optimum traction, speed and to achieve the appropriate ground clearance.

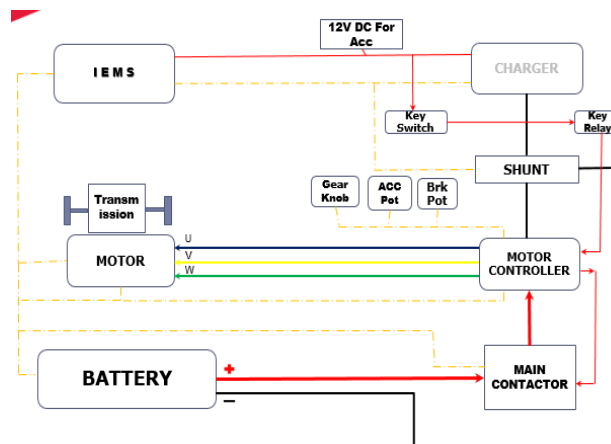


Fig.6.1 Electrical Transmission power flow

| Motor (BLDC) Specification | |
|----------------------------|---------------|
| Max. Power(watts)@rpm | 6661.91 @2892 |

| | |
|-----------------------|-----------------------|
| Max. Torque(N-m)@rpm | 24.112@2388 |
| Max. RPM | 5000 |
| Transmission unit | Electronic controller |
| Li-ion Battery rating | 48V,144AH |
| Gradability (%) | 48.478 |
| Max. Traction (N) | 1155.42 |
| Max. Speed (kmph) | 40.78 |

6.3 FEA analysis

Analysis of the muff coupling, transmission mounting and motor mounting has been done. First analysis is for total deformation followed by the equivalent stress analysis.

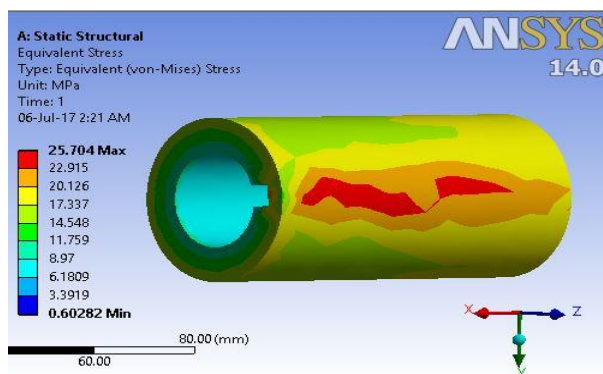
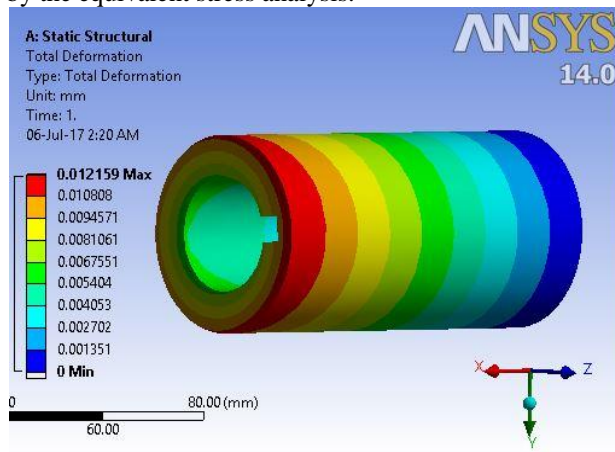


Fig. 6.2 FEA of muff coupling

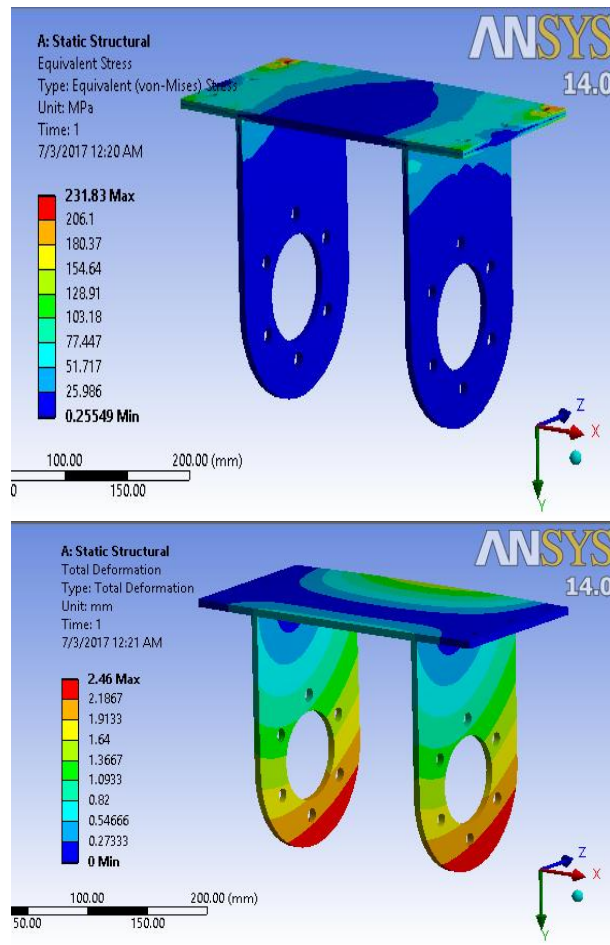
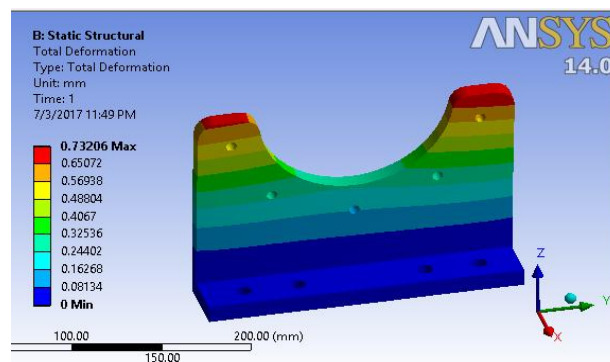


Fig. 6.3 FEA of transmission mounting



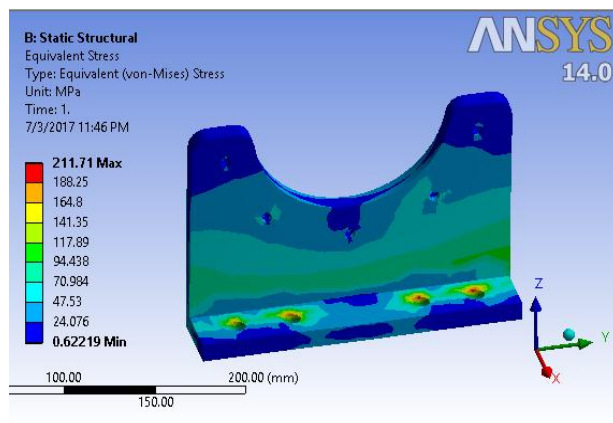


Fig.6.3 FEA of motor mounting

| Parameter | Transmission mounting | Motor mounting | Muff coupling |
|-----------------|-----------------------|----------------|---------------|
| No. of Nodes | 8973 | 5144 | 3296 |
| No. of Element | 4165 | 2636 | 1767 |
| Element size mm | 10 | 10 | 5 |
| Time of Impact | 1sec | 1sec | 1sec |
| Max. Force | 20G | 20G | 4G |
| Max. Eq. stress | 231.83 | 211.71 | 25.704 |
| Max. Def.(mm) | 2.46 | 0.732 | 0.012 |
| FOS | 1.19 | 1.18 | 8.55 |
| Material | 6061 T6 | 6061-T6 | SS904L |

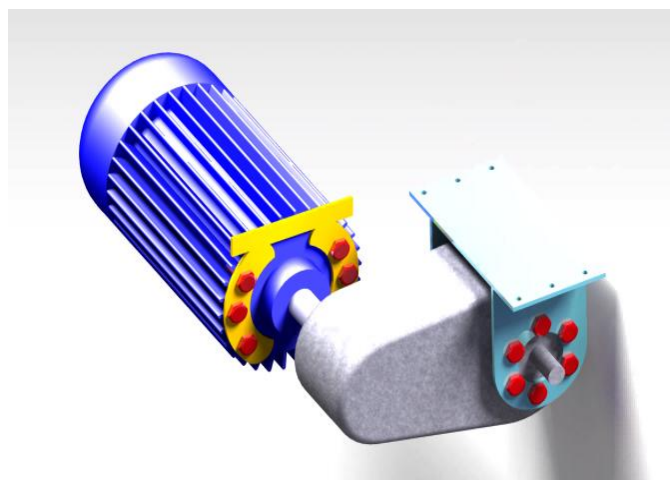


Fig. 6.5 Powertrain Assembly

7. CAD MODELS

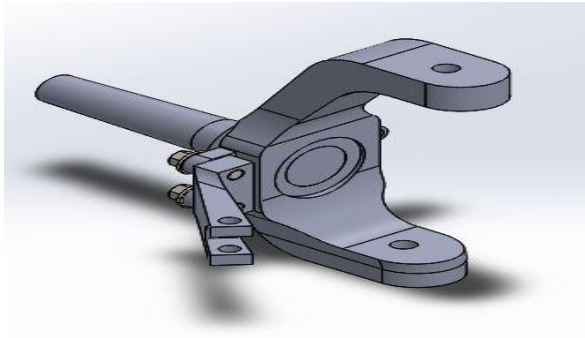


Fig. 7.1 Front Knuckle

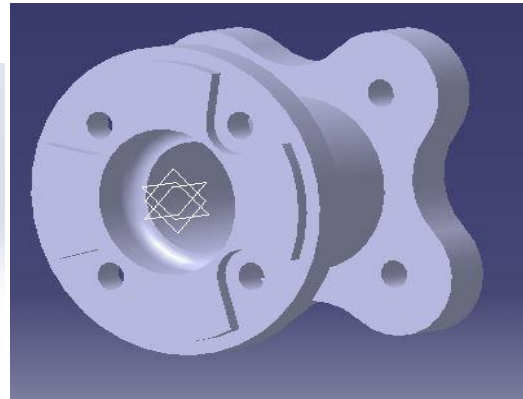


Fig. 7.2 Front Hub

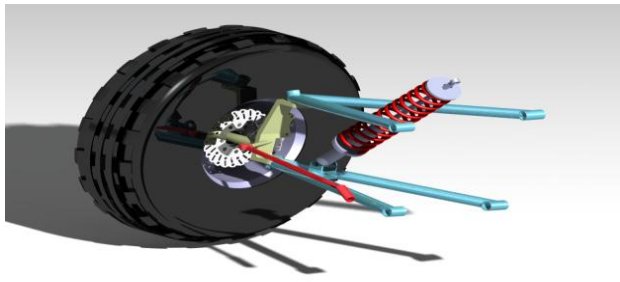


Fig. 7.3 Front wheel assembly

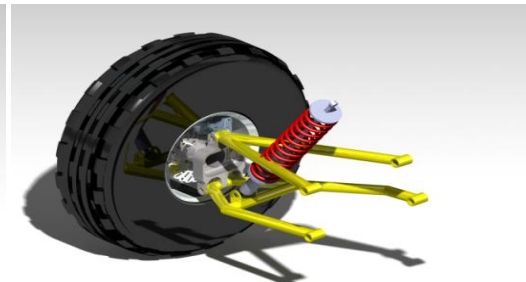


Fig. 7.4 Rear wheel Assembly



Fig. 8. Isometric View of complete Vehicle