International Journal of Engineering Sciences &Research Technology

(A Peer Reviewed Online Journal)
Impact Factor: 5.164

IJESRT

ISSN: 2277-9655
Impact Factor: 5.164
CODEN: IJESS7

IJESRT
INTERNATIONAL JOURNAL OF ENGINEERING SCIENCES & RESEARCH TECHNOLOGY

DESIGNING OF ELECTRIC ATV

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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 fallow.
- 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.

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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 2758Nm^2 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.

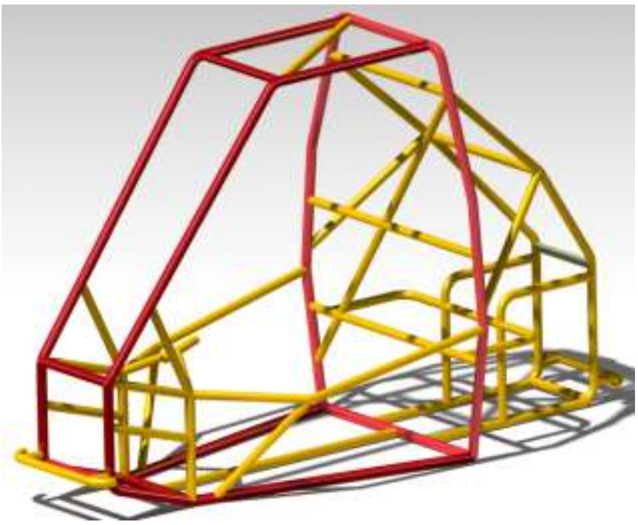

Fig. 2.1. CAD model primary & sec. members.

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/mm^2

I_X=moment of inertia=13478.6378mm^4

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Outer Diameter (do) = 25.4mm
Inner Diameter (di) = 19.4mm
Centre Distance (c) = 12.7mm
Modulus of Elasticity (Ex) = $205*10^3 N/mm^2$
Bending strength=$365 * \frac{\pi}{64} * (d_2^4 - d_1^4)*12.5$
=387Nm
Bending stiffness (Ex*Ix)=$2758Nm^2$

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	Y	Bending Strength	Youngs Modulus	Stiffness	Density	Mass	Material
25.4	3	[illegible]	0.365	12.7	[illegible]	0.205	[illegible]	7850	[illegible]	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 0^0 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 30^0.

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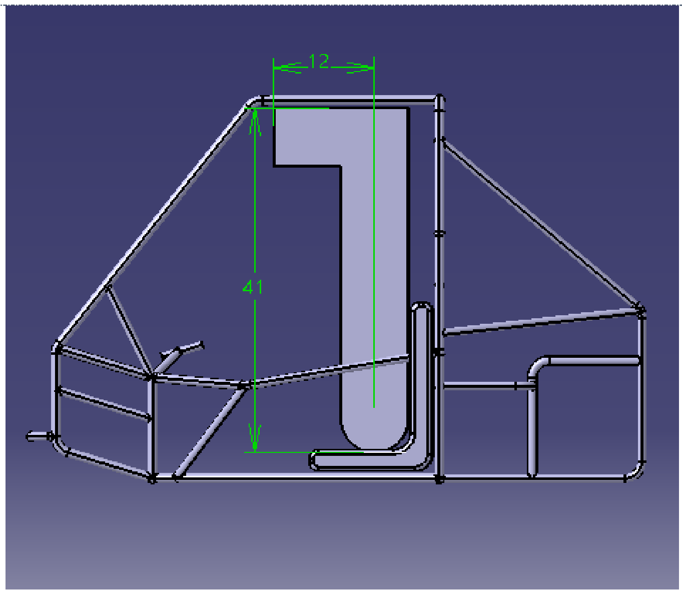

Fig. 2.2 CAD model with template.

2.8 Aerodynamics:
This time we have designed our roll cage without nose and given the caster angle of 17^0 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
$F = \{M \times (V_i - V_f)\} / t$
$= 9.81*\{300 \times (40-36.15)\} / 1$
$= 11330.55N$
Force applied:11.33KN
For FEA the vehicle was consider as running at 40 Kmph and conducted static structural test.

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2.9.1 Front Impact

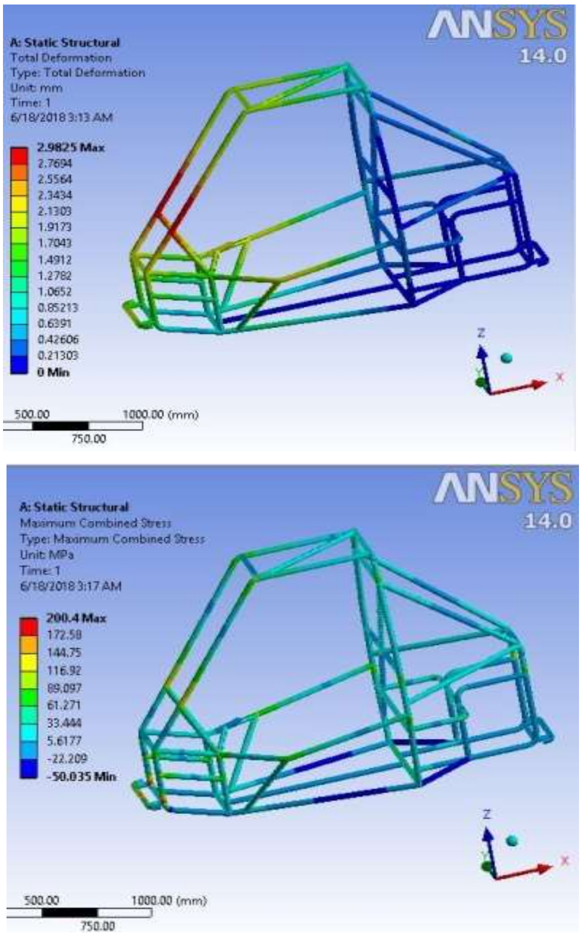

Fig. 2.4. Front Impact Analysis

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

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[Pradeep * *et al.*, 8(3): March, 2019]
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2.9.2 Rear Impact Analysis
2.9.3

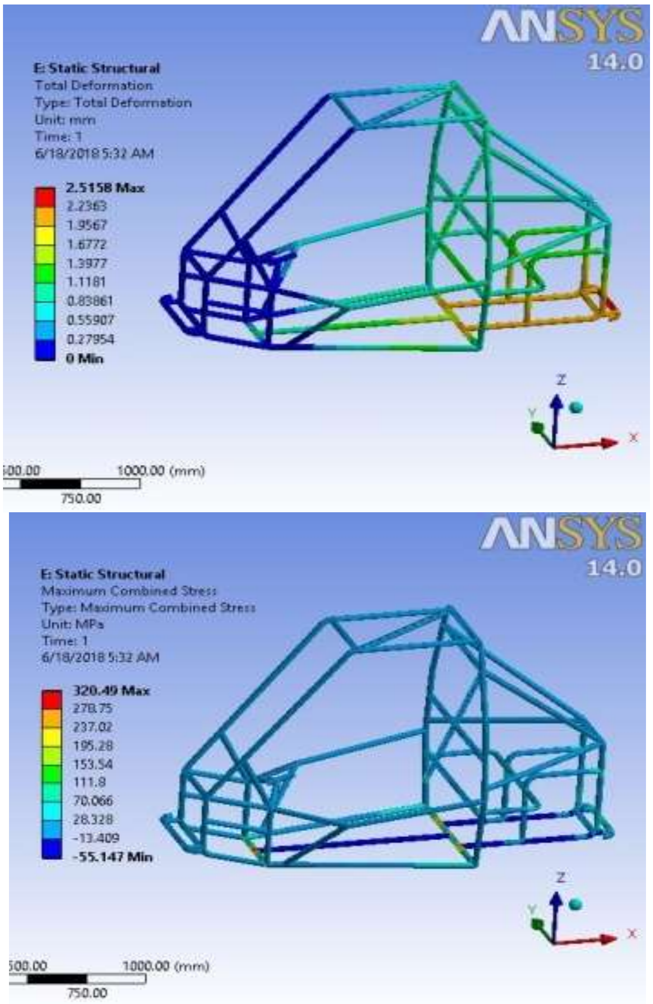

Fig. 2.5. Rear Impact Analysis

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

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2.9.3 Side Impact Analysis

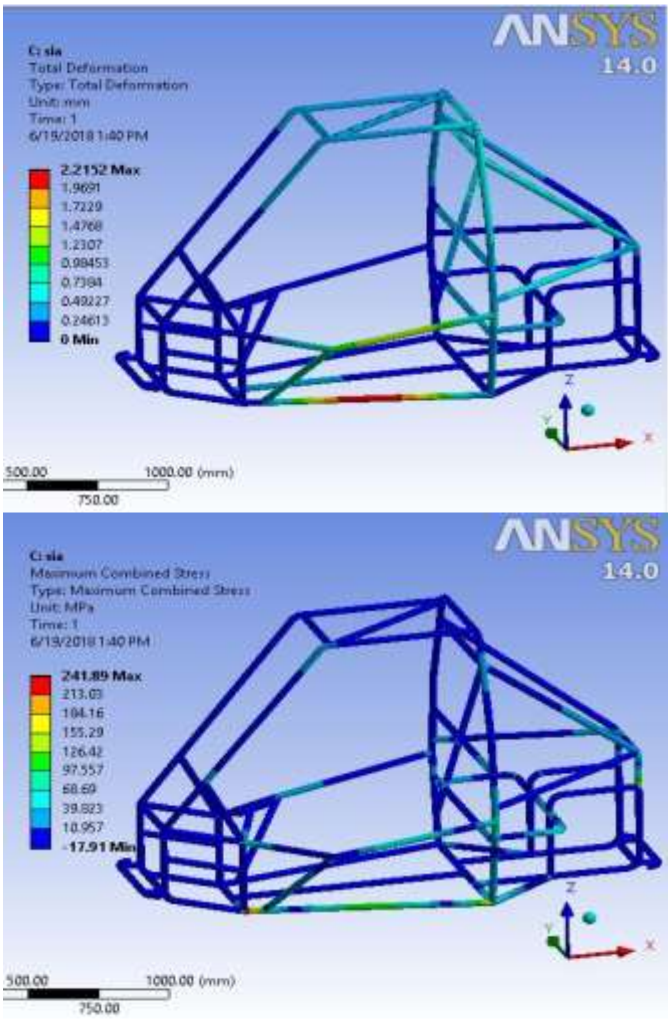

Fig.2.6 Rear Impact Analysis

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

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2.9.4 Torsional Analysis

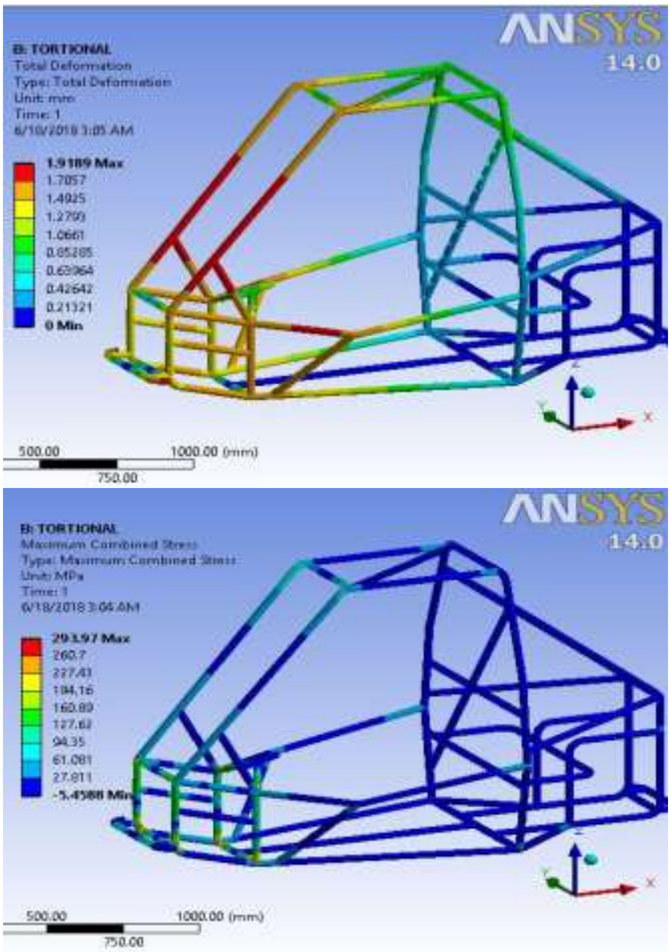

Fig.2.7 Torsional Analysis

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.

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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.

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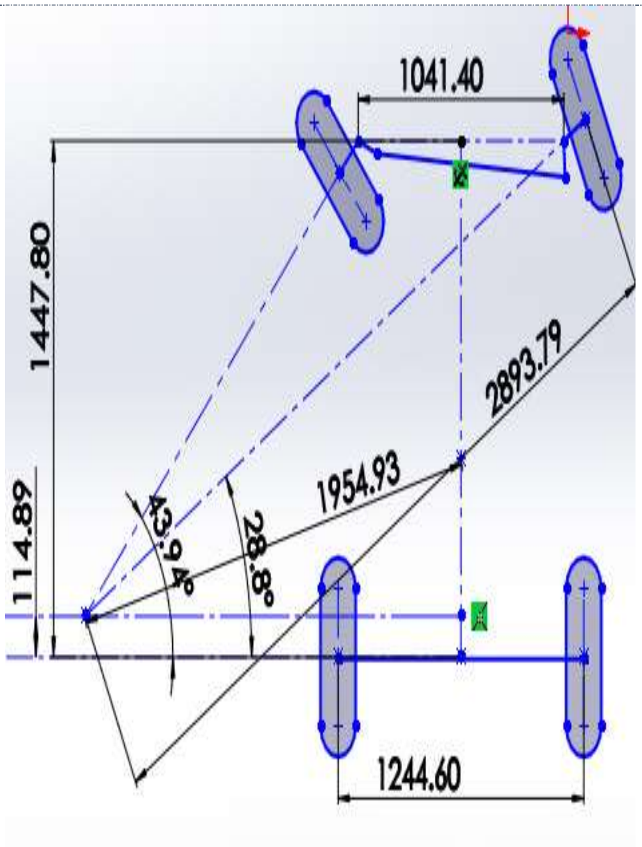

Fig. 3.1. Steering Geometry

Best optimal results from steering geometry are as:

Table 3.2. Steering Geometry Parameter

<table>
<tr><td>Inside Wheel angle</td><td colspan="2">43.66^0</td></tr>
<tr><td>Outside wheel angle</td><td colspan="2">27.34^0</td></tr>
<tr><td>Steering arm length</td><td colspan="2">3.96”</td></tr>
<tr><td>Ackermann angle</td><td colspan="2">28^0</td></tr>
<tr><td rowspan="2">Turning circle radius</td><td>C.G.</td><td>Front outer</td></tr>
<tr><td>1.99 m</td><td>2.94 m</td></tr>
<tr><td>Ackermann %</td><td colspan="2">88.01</td></tr>
</table>

3.2.2 Design Calculation:

Rack Travel (centre to lock) = rθ

$$= 100.68 \times 43.66 \times \frac{\Pi}{180}$$

$$= 76.71 \text{ mm}$$

Assume, Lock to lock turns: 1.5 Turns

∴ Steering Wheel angle (lock to lock)

$$= 1.5 \times 360 = 540^0$$

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

But rack travel per revolution = 2 ×Π×R =102.29 mm

where R= Radius of pinion

∴ R= 16.28 mm; ∴ D= 32.56 mm

Assume no of teeth on pinion, T = 18

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$\therefore$ Module , m = $\frac{D}{T} = \frac{32.56}{18}$ = 1.8
by selecting next series module, m = 2
$\therefore$ Diameter of pinion, D = 36 mm

$$\text{Steering Ratio} = \frac{\text{Steering wheel angle}}{\text{Road wheel angle}}$$

$$= \frac{270}{\frac{43.66+27.34}{2}} = 7.6:1$$

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
   $\therefore F_t = 684.57$ N

$\therefore$ Torque acting on pinion = $F_t \times$ Radius of pinion
= 684.57 × 18
=12322.26 N.mm
= 12.32 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 =steering effort × 110
$\therefore$ Steering effort = 112.02 N
So, in order to decrease the steering effort we have selected the steering wheel Diameter as 260 mm.

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

3.3 CAD models

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

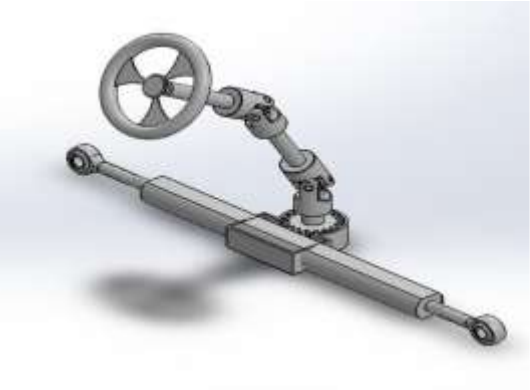

Fig3.2 . Steering assembly

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.

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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/m3
Ultimate tensile strength	541 Mpa
Yield strength	468 Mpa
Modulus of elasticity	71.7 Gpa
Poisson's ratio	0.33

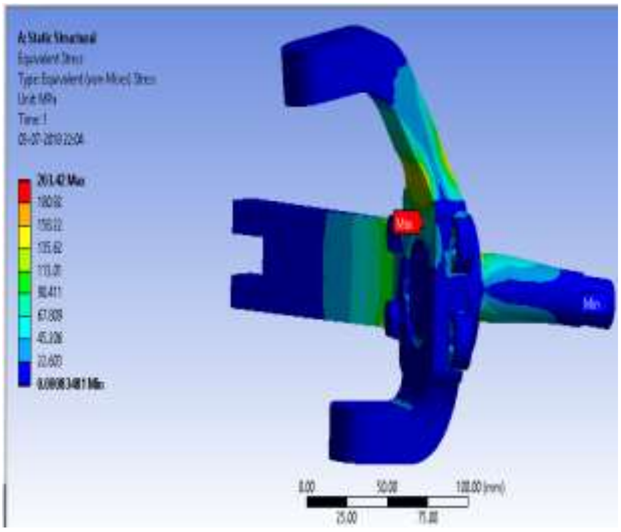

fig 3.3. FEA of Knuckle

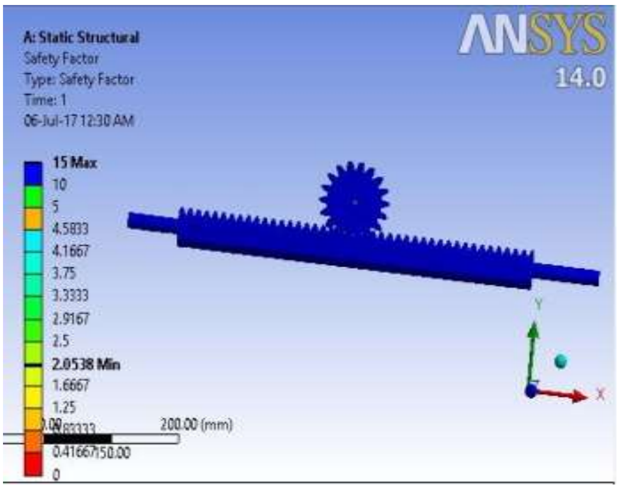

fig 3.4. FEA of Rack & Pinion

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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.

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4.3.1. Static Structural Analysis

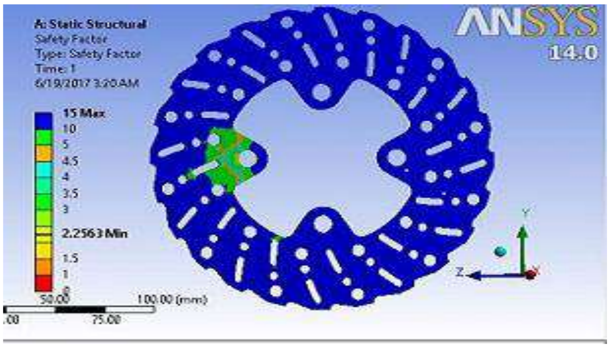

Fig 4.1 Static Structural Analysis of front

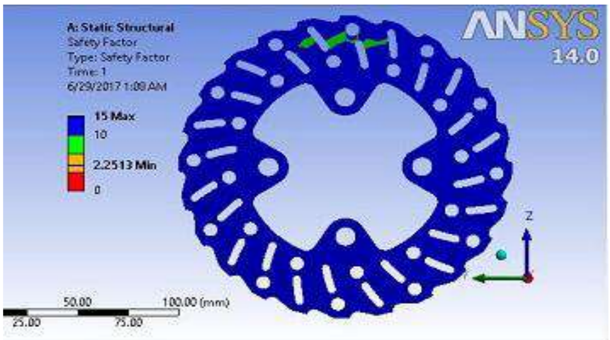

Fig 4.2 Static Structural Analysis of rear

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

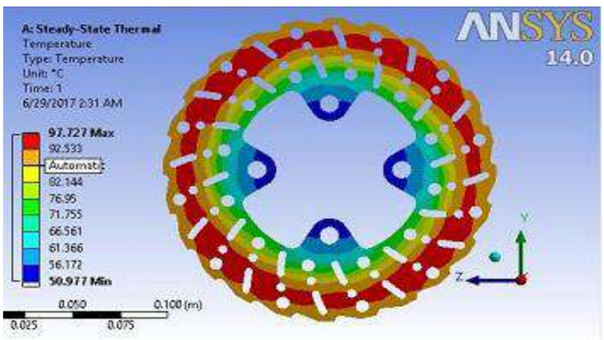

Fig 4.3 Thermal Analysis of Front rotor

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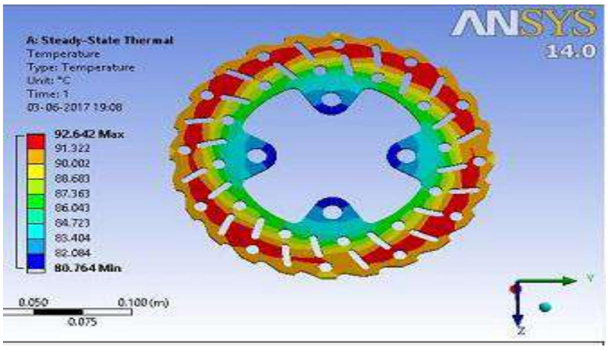

Fig 4.4 Thermal Analysis of Rear rotor

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
  μ_{road}=0.6 μ_{pad}=0.4
  CG Height =571.5 mm
  Pedal Ratio= 4.5:1 Pedal Effort= 200 N
- Weight transfer $=\frac{(\mu road*mass*c.g.hight)}{wheel\ base}$ = 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

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- Braking Force at Rear(BFr)= FOS × μ_{road} × Dynamic Load Rear×g = 1.5×0.6×105×9.81
  = 927.04 N
- Deceleration(a)= $\frac{(\text{BF at Front+BF at Rear})}{\text{m}}$
  = (1721.65+927.04)/300
  = 8.82 m/s^2
- Deceleration Time= $\frac{v}{a} = \frac{12.5}{8.82}$ = 1.38 s
- Stopping Distance= $\frac{v^2}{2a} = \frac{12.5^2}{2\times 8.82}$ = 8.2 m
- Effective Radius of Disc at Front(Rf)=
  $\left(\frac{1}{3}\right) \times \frac{[(\text{Front OD}^3)-(\text{Front ID}^3)]}{[(\text{Front OD}^2-(\text{Front ID}^2)]}$ = 70.45 mm
- Effective Radius of Disc at Rear (Rr)= 67.578 mm
- Torque acting on front wheel: -
  = $\frac{\text{BFf}*\text{radius of tire}}{2}$= 251.44Nm
- Torque acting on rear wheel: -
  = $\frac{\text{BFr}*\text{radius of tire}}{2}$= 135.39Nm
- Required Clamping force on front=
  $\frac{\text{Braking torque on front}}{\mu\text{pad}*\text{Re front}*\text{n}}$=7813.45 N
- Required Clamping force on Rear=
  $\frac{\text{Braking torque on Reart}}{\mu\text{pad}*\text{Re rear}*\text{n}}$=4363.47
- Pressure acting in master cylinder: -
  =$\frac{\text{Pedal effort}\times\text{pedal ratio}}{\text{piston area of master cylinder}}$= 3.1576 mpa
- Actual Clamping Force on rotors(Fc):-
  = 2*Pressure×Area of caliper piston
  = 3.1576×2×($\frac{\pi}{4}$)×30^2
  =8927.9 N
- Torque acting on front rotor:
  = μ_{pad} * front effective radius*Fc
  = 251.58 Nm
- Torque acting on front rotor :
  = μ_{pad} * rear effective radius*Fc
  = 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

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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.

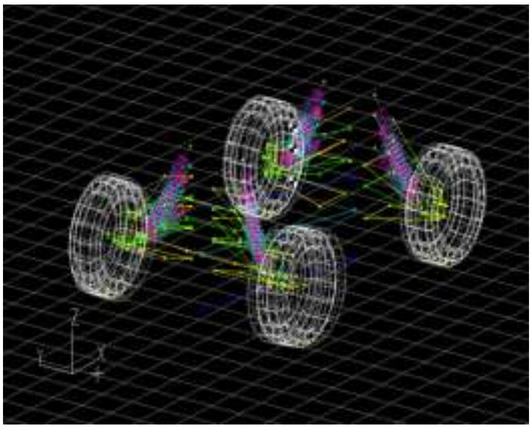

Fig.5.1 Simulation through software

Table 5.2 Static suspension parameters:

CG height	571.4mm
Camber at kerb wt.	-0.5^0
Castor	10^0
Static Toe	0.7^0
KPI	8^0
Scrub radius	79.44mm
Camber gain	2^0
Roll gradient	2.29^0/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

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

G=80000 Mpa

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

  =1545.07 N

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

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

- $T_{max.\ (design)} = \frac{8*Fmax*c\ \ *Kw}{\pi*d^2}$

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

Hence K_w = 1.162
$T_{max.\ (design)}$= 507.03
As $T_{max(design)} < T_{max(material)}$
So design is Safe.

- Spring Rate(K) = $\frac{Fmax}{\partial max}$
  K=10.862N/mm
- Number of Turns(N)=$\frac{G*d}{8*C^3*K}$

N=12 turns
For square and grounded end
N`=N+2
N` =14 turns

- Solid Length(L_s) = N` x d= 130mm
- Free length= L_s+ (1.15 x ∂_{max})
  L_f= 293.57mm
- Pitch= $\frac{Lf-2*d}{N}$
  p =22.63 mm
- Natural Frequency(Fn)= $\frac{wn}{2*\pi}$

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

- $F_n = \frac{14.38}{2*\pi}$
  = 2.28Hz

Spring calculation for rear spring
Material for spring: Chrome-Silicon
$T_{max(material)}$ =729N/mm^2

- $F_{max} = \frac{m*W.d*g*L}{2}$
  =3090.15 N
- Spring travel(∂_{max}) = w.t x M.R.
  =142.24mm

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

- $T_{max.\ (design)} = \frac{8*Fmax*c*Kw}{\pi*d^2}$

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

  Hence K_w= 1.184
  $T_{max.(design)}$= 615.91
  As, $T_{max(design)} < T_{max(material)}$
  So design is Safe.

- Spring Rate(K)=$\frac{Fmax}{\partial max}$
  =21.72 N/mm
- Number of Turns(N)= $\frac{G*d}{8*C^3*K}$

N = 10 turns

For square and grounded end

$$N` = N+2$$

$$N` = 12 \text{ turns}$$

- Solid Length(L_s)= N`* d= 132mm
- Free length(L_f)= L_s+ (1.15 x ∂_{max})
- L_f= 295.57mm
- Pitch= $\frac{Lf-2*d}{N}$

  p=27.35mm
- Natural Frequency(Fn)= $\frac{wn}{2*\pi}$

  $w_n = \sqrt{\frac{K}{m}}$=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

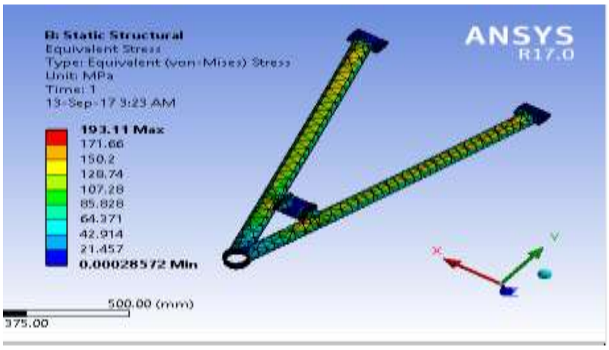

Fig.5.2 FEA of front arm

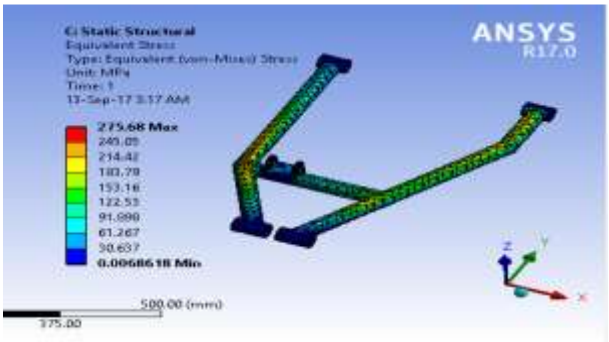

Fig.5.3 FEA of Rear arm

Sr no.	Parameter	Front	rear
1	Analysis type	Static structural	
2	No. of nodes	7063	7039
3	No. of elements	3355	3323

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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

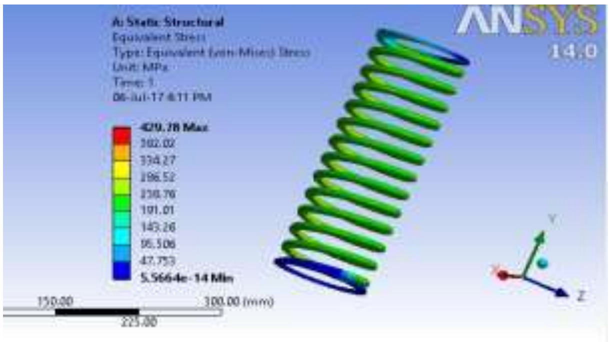

Fig.5.4 Front Spring

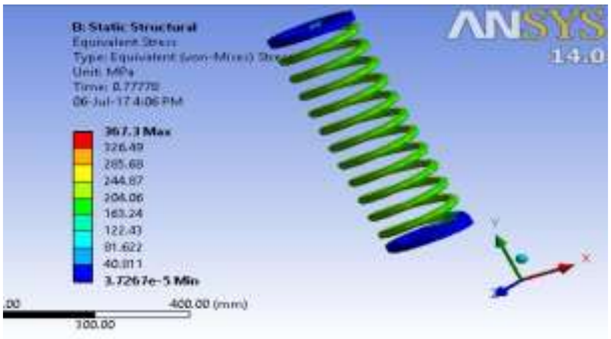

Fig.5.5 Rear Spring

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

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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 Kmph = 6.94 m/s

Air Drag Coefficient (Cd) = 0.4
Max. Motor RPM (Nmax) =5000 rpm
Motor Torque (M_t) =25N.m
Tractive Effort = RR + GR + AR
= µ.m.g.cosϴ+m.g.sinϴ+ ½ ρAV2Cd
=1128.185N

Acceleration=$\frac{Mt*GR*\eta gearbox}{Rwheel*m}$
= (25*13.5*0.92)/ (0.2921*300)
=3.54 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.

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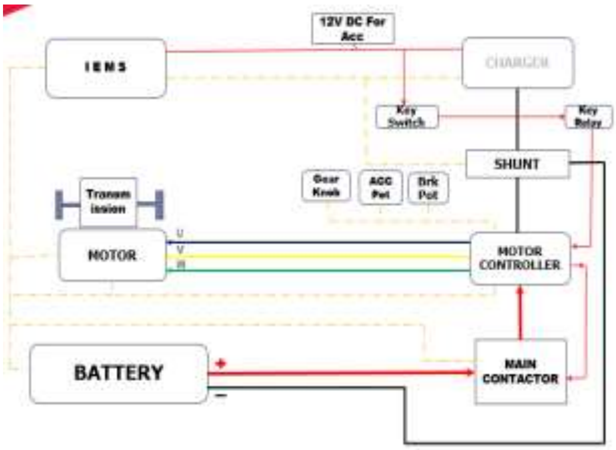
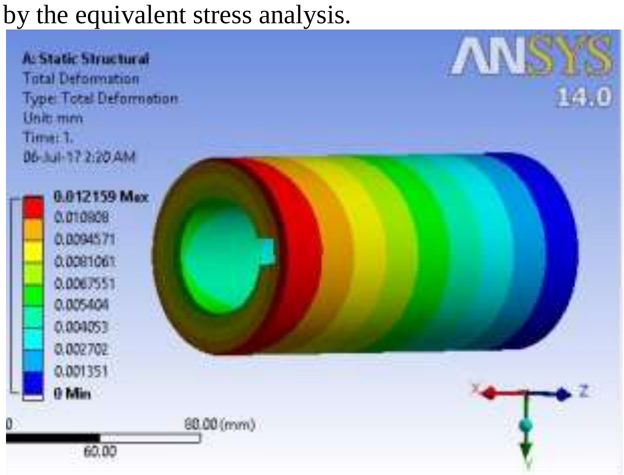


Fig.6.1 Electrical Transmission power flow

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.



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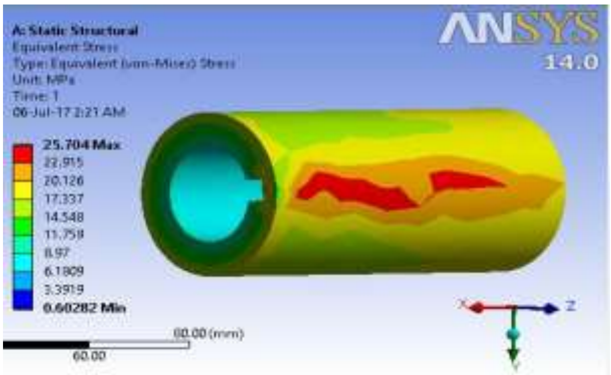

Fig. 6.2 FEA of muff coupling

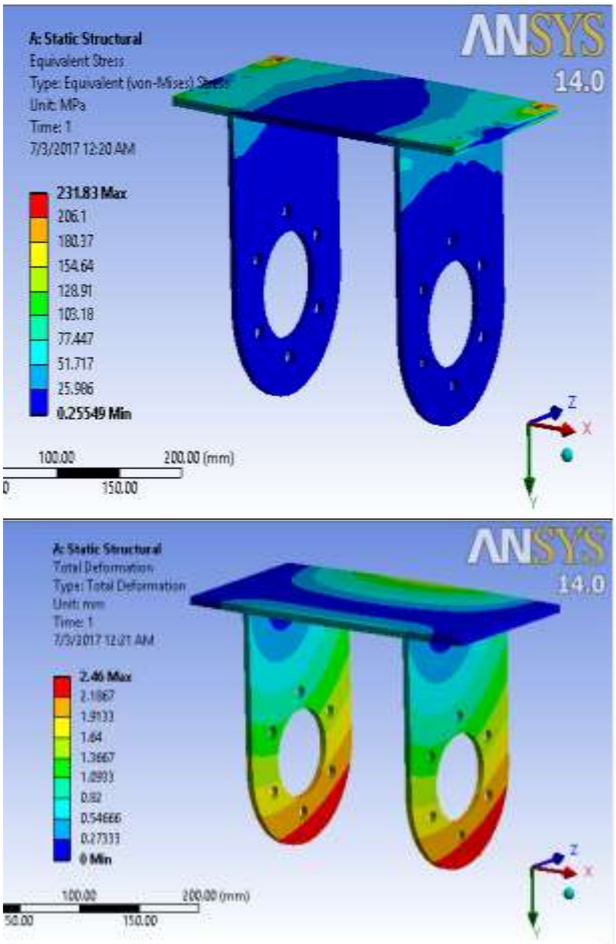

Fig. 6.3 FEA of transmission mounting

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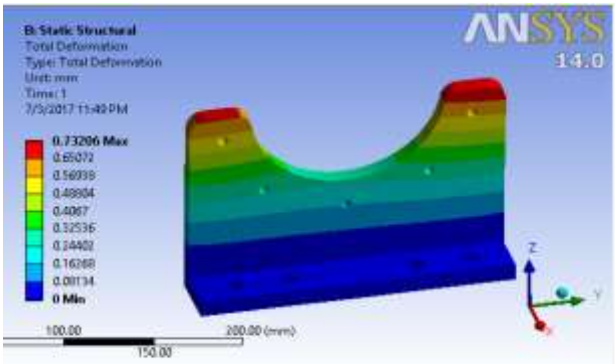

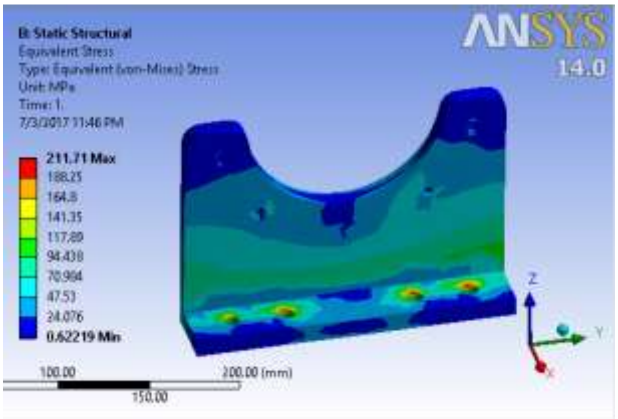

Fig.6.3 FEA of motor mounting

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

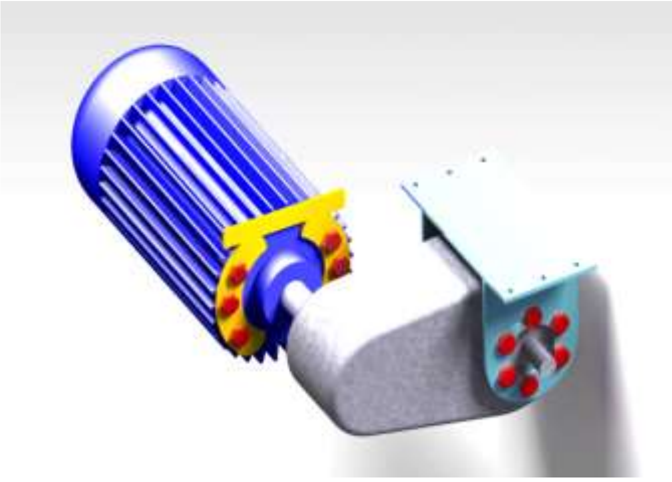

Fig. 6.5 Powertrain Assembly

7. CAD MODELS

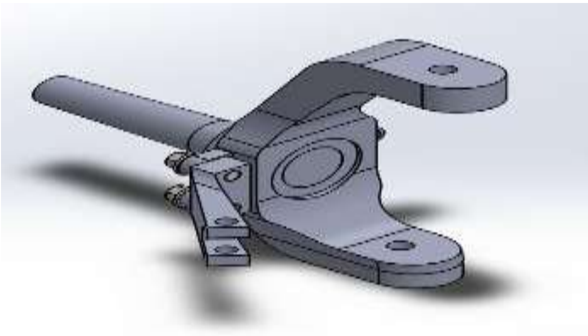

Fig. 7.1 Front Knuckle

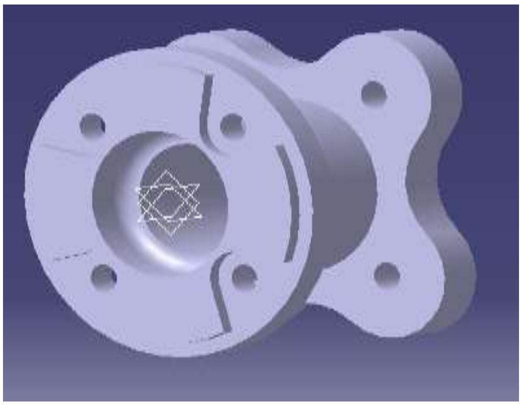

Fig. 7.2 Front Hub

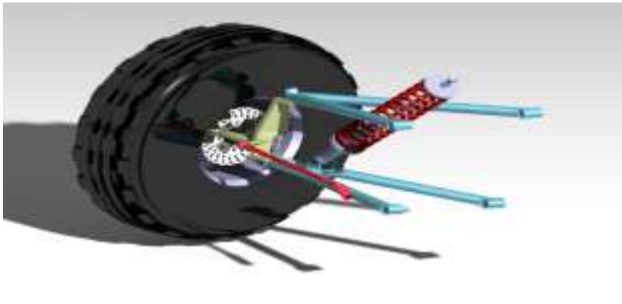

Fig. 7.3 Front wheel assembly

Fig. 7.4 Rear wheel Assembly

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Fig. 8. Isometric View of complete Vehicle