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Design and Analysis of a Formula - Hybrid Vehicle

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Abstract: The paper provides an overall structure to design a formula hybrid vehicle. It combines the attributes of both engine and motor in a specific configuration. The frame is designed keeping in mind the quality to weight ratio. Specific simulations justify the strength of frame, hubs and aerodynamic standards of the vehicle. The steering system has been designed keeping in mind about the racing factor. Proper braking mechanisms and ergonomics are precisely mentioned. Powertrain being the power house of the vehicle is carefully analyzed and chosen and Drivetrain being the muscle of the vehicle is planned. The design methodology, CAD and analysis of specific parts are provided to elaborate the design.
Keywords - Cylindrical Crash Box, Frame Design, Formula Hybrid vehicle; Hybrid Powertrain; Upright;

I. INTRODUCTION

The Race car industry is evolving very fast, it's time to move towards better energy resources and adaptive designs, the aim of this paper is to envelope the modeling, designing concept and calculations involved in making a hybrid race car with the aid of CAD software such as Solidworks 2016 and Analysis on Ansys, to back the claims. Starting with the chassis, material selection and design methodology coupled with suitable simulations to back the designs are provided. The steering being the direct connecting factor between the driver and vehicle, designed for the drivers ease to handle and calculations to justify the claim. The break's geometry with all its key components is elaborately explained with calculations, CAD and analysis to back the claim. Suspension is one of the key components in a race car, its selection and calculations are precisely discussed. Powertrain and electronics being the main attraction of the paper is briefly discussed with reasons for its type and configuration selection. An all around design study of a formula hybrid vehicle has been proposed in this paper.

II. CHASSIS DESIGN AND ANALYSIS

A. Frame Design

The chassis has been delineated by taking factors like dimensional points of confinement (width, height, length and weight), operational impediments, and regulatory issues, legitimately restricting essentials, money related requirements and human ergonomics as a need.

1) Frame being the biggest and bulkiest, the constituent members should be weight optimized.
2) The strength to weight ratio is expected to be high.
3) The weight of the vehicle should be balanced; the COG should lie on the center line of the vehicle towards the rear axle.
4) Front to rear weight ratio should be nearer to 40:60 and left to right to be 50:50
5) The ground clearance should be more than 3 inches.
6) The design should be manufacturing friendly.
7) Node to node triangulation had to be taken seriously.
8) There should be sufficient members to support the suspension, transmission and drivers weight over the frame.
9) The design should be compact and comfortable to enter and exit for the driver.

B. Design Methodology

1) The wheel base and track width were finalized for the vehicle.
a) A rough sketch of the frame was built as shown in Fig.1 with keeping in mind about the dimension limitations for better turning radius and ergonomics.

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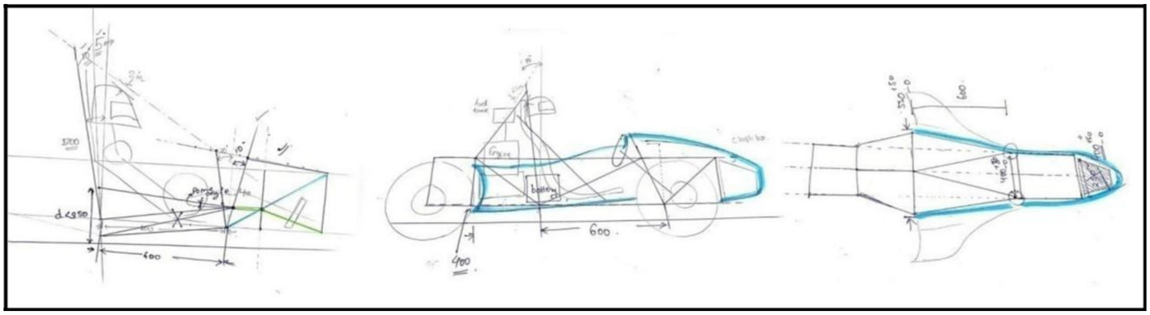
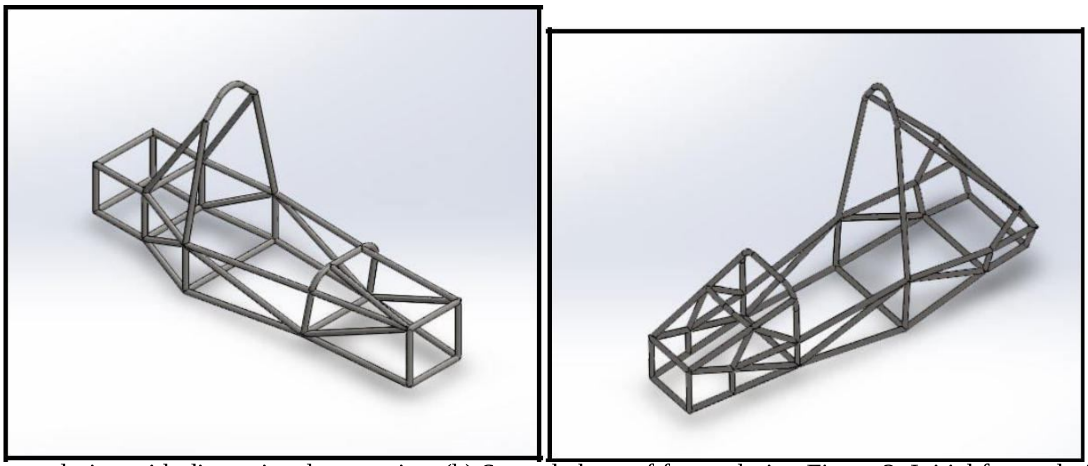


Figure 1: Rough sketches of the frame displaying side and top view

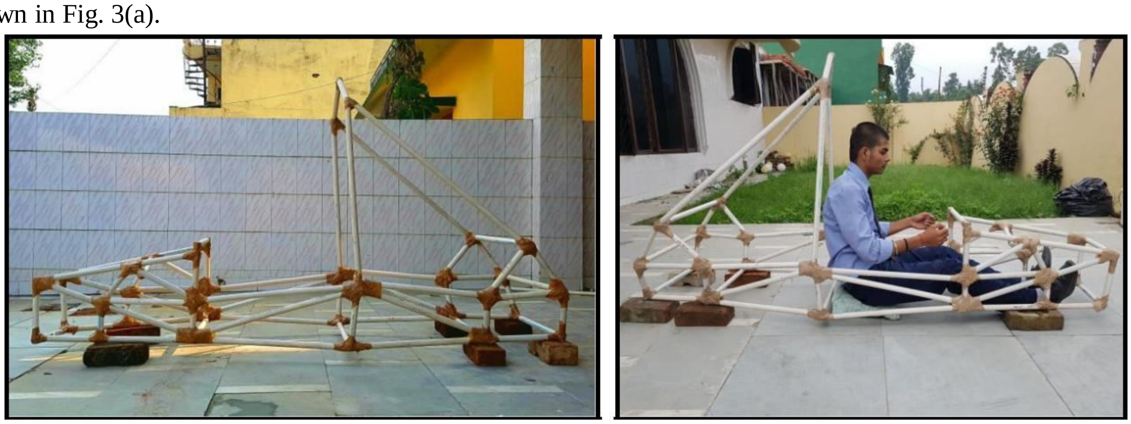
b. The design was improved by changing member positions and adding and removing members. Keeping in mind about the weight factor and the phase 1 frame came to life shown in Fig.2 (a).



(a) First frame design with dimensional constraints (b) Second phase of frame design Figure 2: Initial frame designs

c. After discussions over the triangulation members in the frame and keeping in mind about the manufacturing problems third phase of the frame was build shown in Fig. 2(b).

d. A PVC model was build with proper dimensions to check the ergonomics of the vehicle and to know the changes in the frame shown in Fig. 3(a).



(a) PVC tube model (b) Re-dimensioned PVC tube model Figure 3: PVC tube models of the frame design

e. The third phase of the frame was designed with the desired changes for the ergonomics point of view. The design was rechecked for comfort by adjusting the dimensions in the PVC model as shown in Fig. 3(b).

f. Components were placed in accordance with the weight balance of the vehicle.

g. Ground clearance was taken in account.

h. Engine position, motor position, differential and CVT position was finalized for optimum weight balance.
i. C bracket members were welded in the frame for the seat and brackets were welded for mounting differential, CVT, engine, motor and battery. The fourth phase of the frame was designed as shown in the Fig. 4.

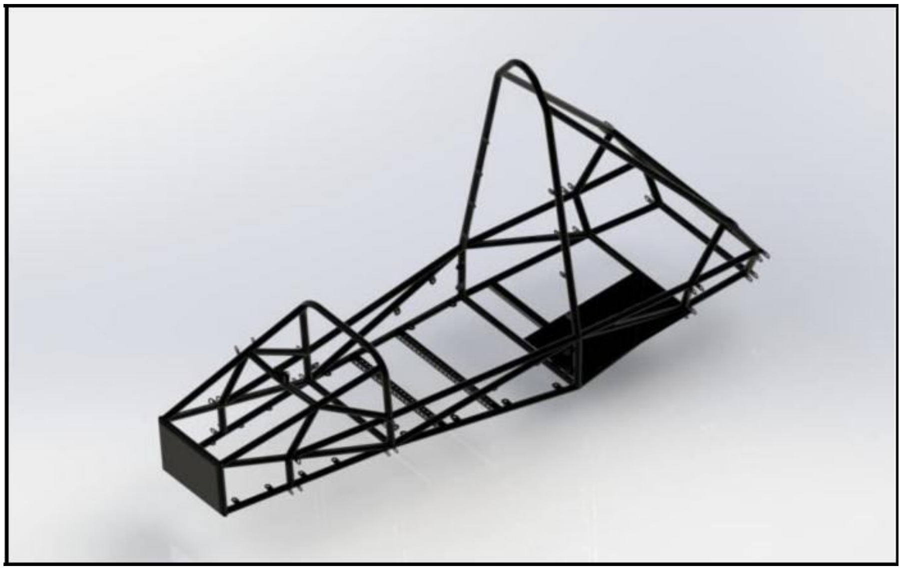


Figure 4: fourth phase of frame with necessary brackets and gusset plates been mounted.

j. After the transmission design the frame was redesigned in its fifth phase, where some dimensions were changed for adjusting the transmission components to fit in the required position.
k. Cylindrical crash box was designed for optimum impact force absorption shown in Fig. 5.

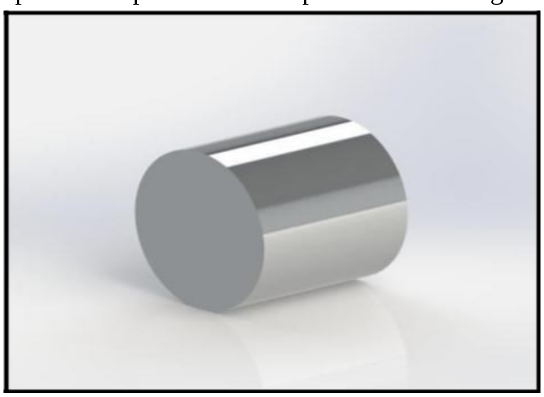
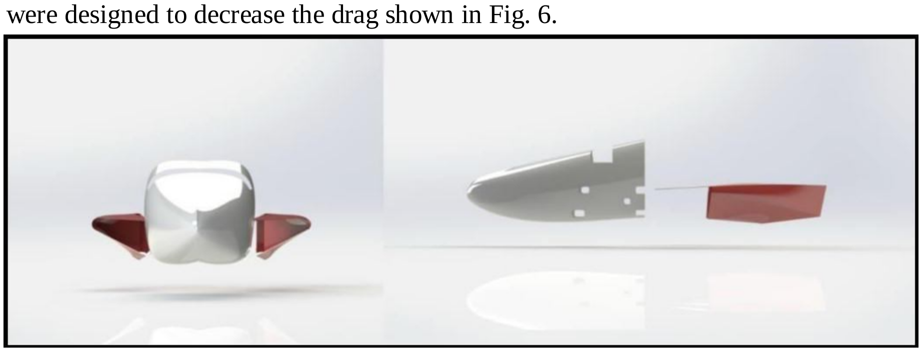


Figure 5: Cylindrical crash box concept

l. After designing the frame, aerodynamics of the vehicle was designed.
m. Nose cone, side pods were designed to decrease the drag shown in Fig. 6.



(a) Front view(b) Side view
Figure 6: Front and side view of nose cone and side pods design

n. Analysis was done to check the impact resistance of the frame and to determine the factor of safety.

o. After approximate weight and acceleration of the kart was known, cross-section of chassis pipes was determined by using bending moment formula.

C. Material Availability

AISI 1018 has excellent weld ability and produces a uniform and harder case and it is considered as the best steel for carburizing parts. The 1018 carbon steel offers a good balance of strengthened ductility and toughness. Considering the above factors we choose AISI 1018 for our chassis material.

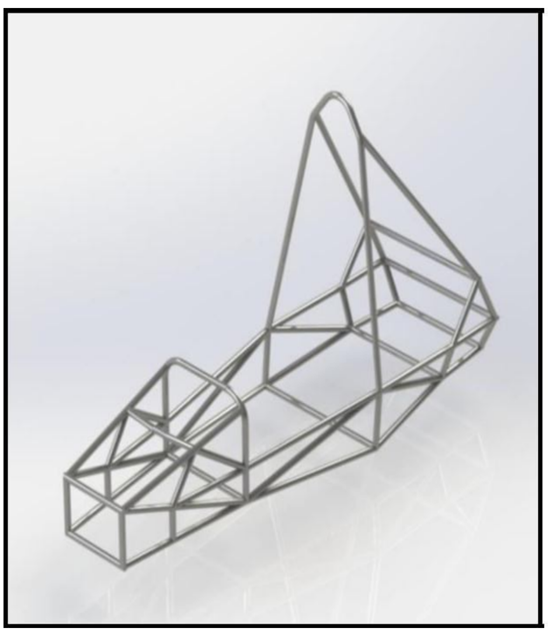
Table 1: Physical Properties of AISI 1018

PROPERTIES	VALUE (Metric)
Density	7.87g/cc
Yield tensile strength	370MPa
Elongation at break (in 50 mm)	15%
Poisons ratio	0.29
Modulus of elasticity	200GPa

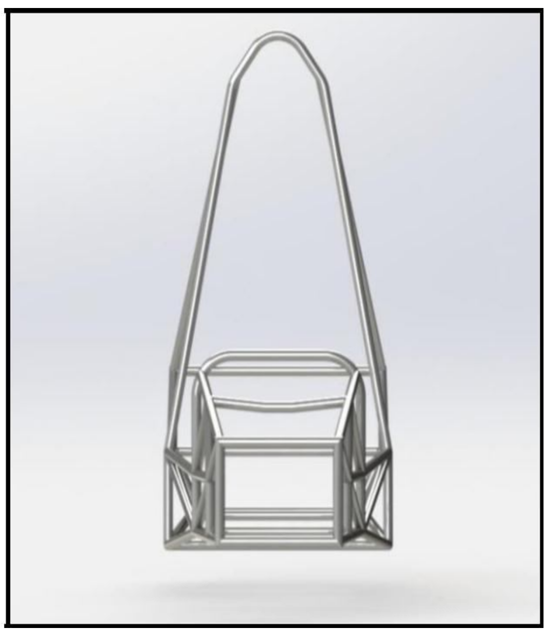
Table 2: Pipe used and Frame

Dimension of pipes	1inch diameter and 1mm thickness for frame.
Mass of frame	14.249Kg
Welding type	Electric arc welding and TIG welding

D. Frame Views



(a) Isometric view



(b) Front view

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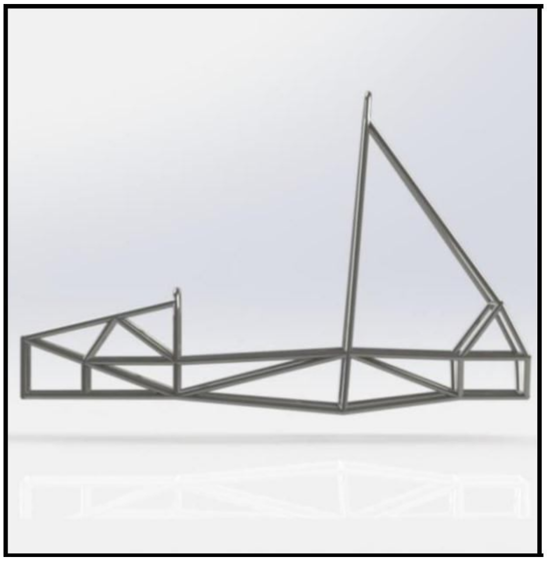
(c) Side view


(d) Top view

Figure 7 Different views of the frame

E. Frame Analysis

For the purpose of analysis, we have conducted certain test on the frame. The following calculations were done to calculate the impact load:

Table 3: Weight distribution

Parameter	Value
Weight of the vehicle	180kg
Weight of driver	60kg
Misc. weight (fuel, fire extinguisher etc.)	10kg
Total weight	250kg

Considering a scenario in which the vehicle hits a stationary object with a velocity of 80 km/hr (22.23 m/s), and let the impact duration be equal to 0.1 sec. Assuming the collision to be elastic in nature, the final velocity of

$$\text{Impact Force} = \text{Mass} \times \text{Impact velocity} \,/\, \times \text{Impact Time} \quad (1)$$

vehicle will be 0 m/s. The impact force obtained is 27,787.5 N ≈ 11.3G.Given by (1).

1) *Front impact test:* The front impact analysis has been carried out on Solidworks 2016; a force of 11.3G was applied to the front ends, constraining the hard points of frame. The results are seen in Fig. 8 and Fig. 9.

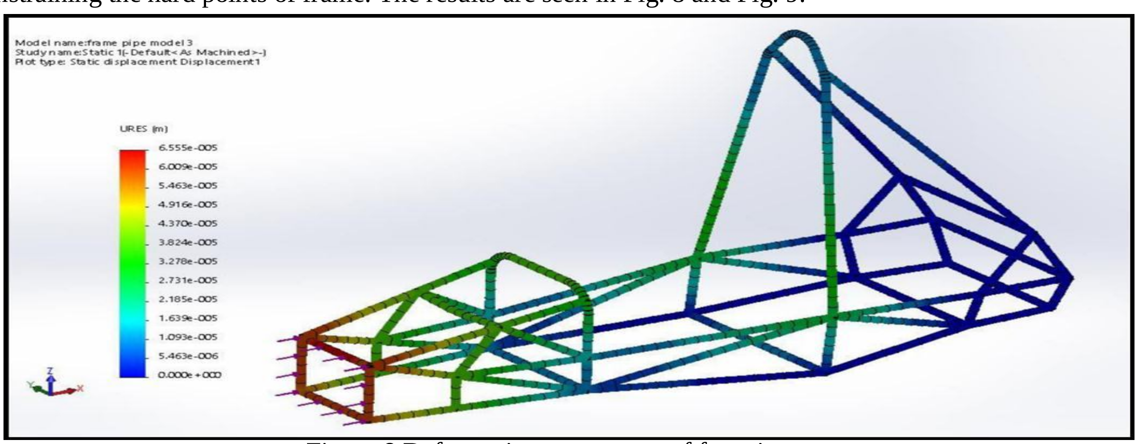

Figure 8 Deformation parameters of front impact

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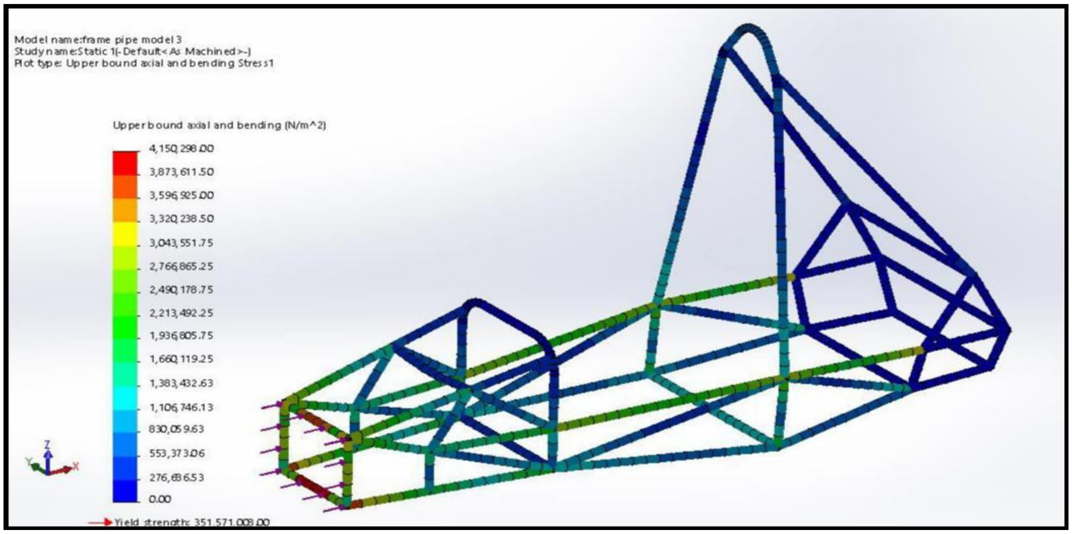

Figure 9 Stress parameters of front impact

The deformation is within the acceptable limits.
FOS = Yield strength of AISI 1018 / Mises Stress
So, FOS = 85

2) *Side impact test:* The side impact analysis has been carried out on Solidworks 2016; a force of 11.3G was applied to the front ends, constraining the hard points of frame. The results are seen in Fig. 10 and Fig. 11.

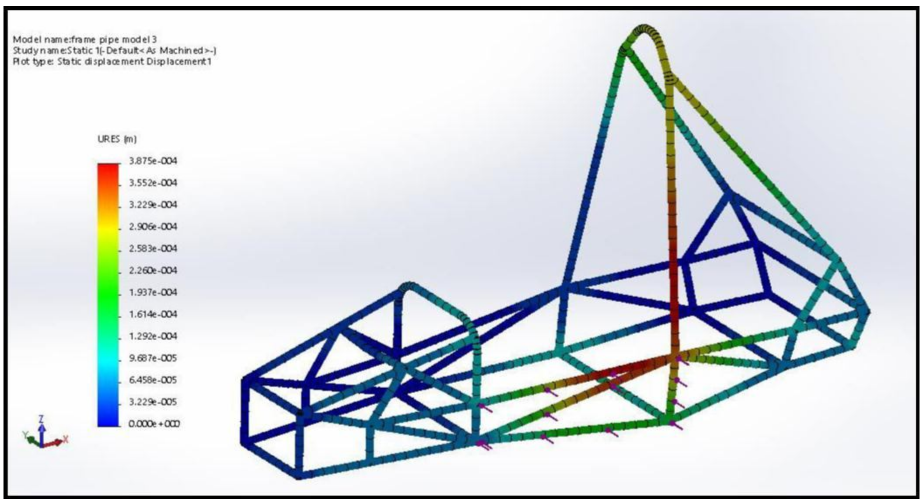

Figure 10 Deformation parameters of side impact

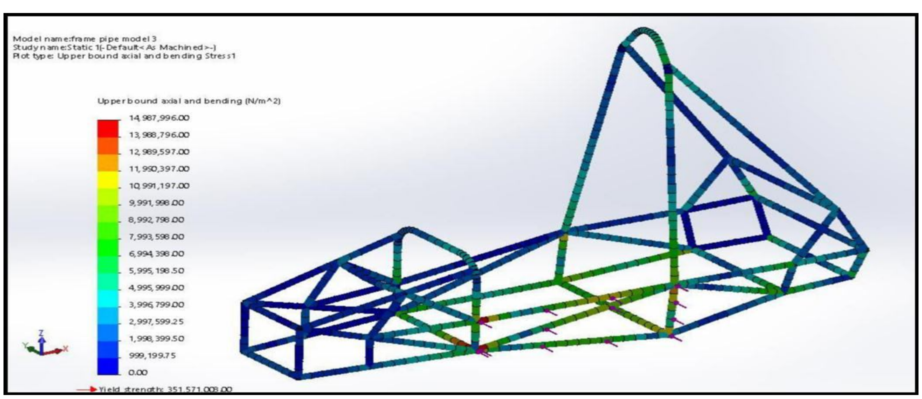

Figure 11 Stress parameters of side impact The deformation is within the acceptable limits.

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FOS = Yield strength of AISI 1018 / Mises Stress So, FOS = 23

3) Rear impact test: The rear impact analysis has been carried out on Solidworks 2016; a force of 11.3G was applied to the front ends, constraining the hard points of frame. The results are seen in Fig. 12 and Fig. 13.

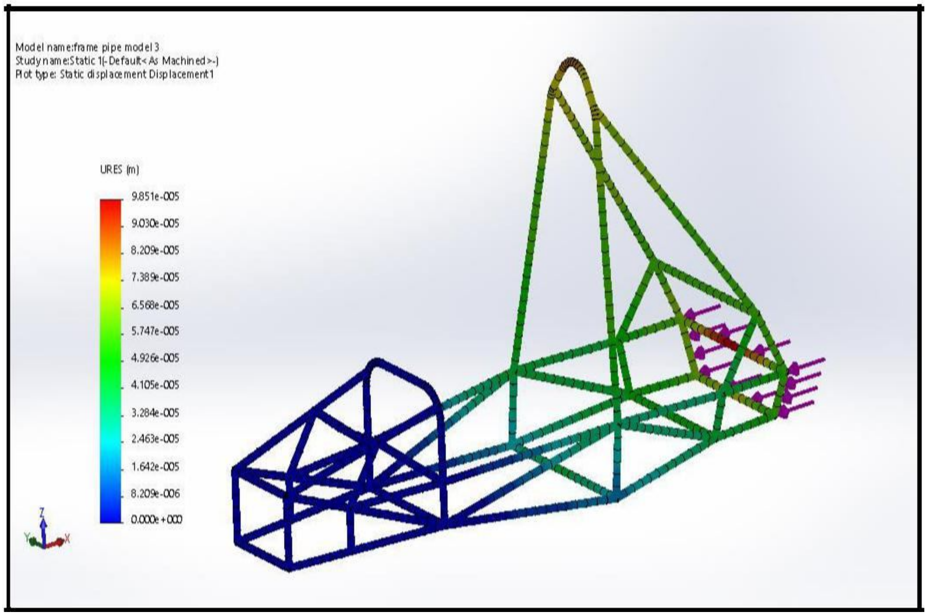


Figure 12 Deformation parameters of rear impact

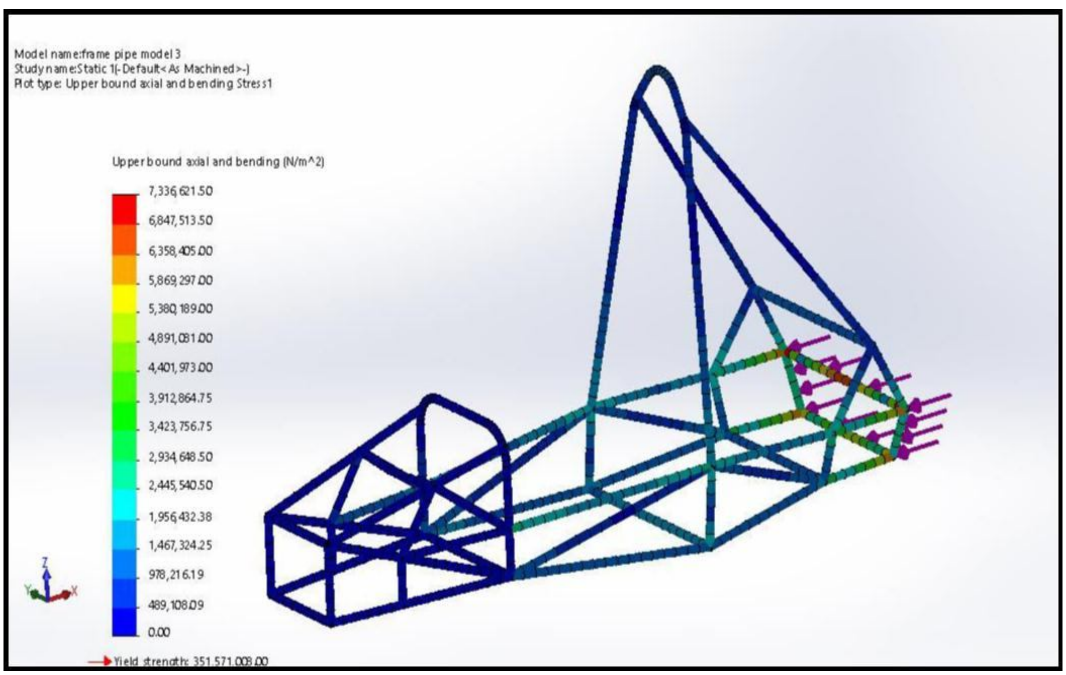


Figure 13 Stress parameters of rear impact The deformation is

within the acceptable limits.

FOS = Yield strength of AISI 1018 / Mises Stress So, FOS = 48

4) Roll over test: The roll over analysis has been carried out on Solidworks 2016; a force of 11.3G was applied to the front ends, constraining the hard points of frame. The results are seen in Fig. 14 and Fig. 15.

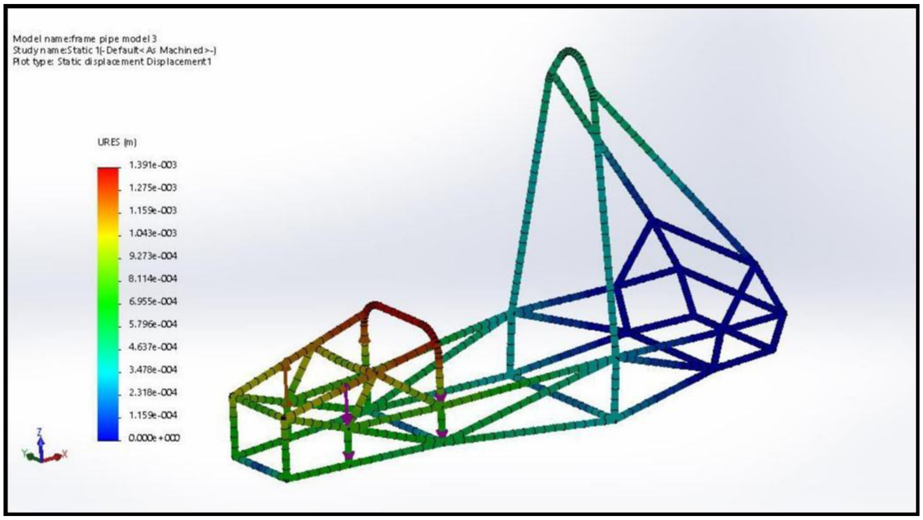


Figure 14 Deformation parameters of roll over

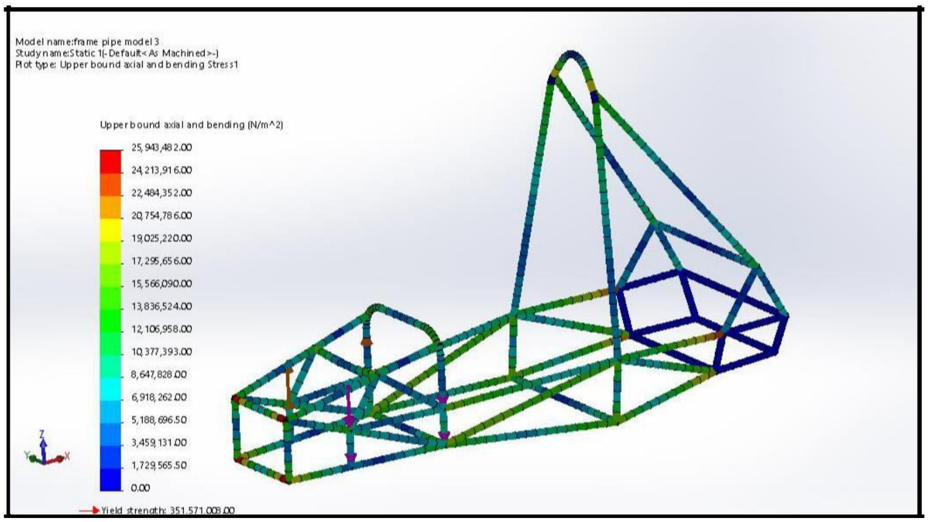


Figure 15 Stress parameters of roll over

The deformation is within the acceptable limits.
FOS = Yield strength of AISI 1018 / Mises Stress
So, FOS = 14

Table 4 summarizing the above discussion

Elements	FOS	Maximum Deformation(mm)	Maximum Stress(MPa)
Front Impact	85	0.065	4
Side Impact	23	0.4	14
Rear Impact	48	1.0	7
Roll Over	14	1.4	25

F. Stability of the Vehicle

In case of a four wheeled vehicle, it is essential that no wheel is lifted off the ground while the vehicle takes a turn. The condition is fulfilled as long as the vertical reaction of the ground on any of the wheels is positive in upward direction.

Mass of vehicle = 250 kg

Weight of the vehicle = 250kg × 9.81m/s^2 = 2452.5 N = 1G

1) Reactions due to weight

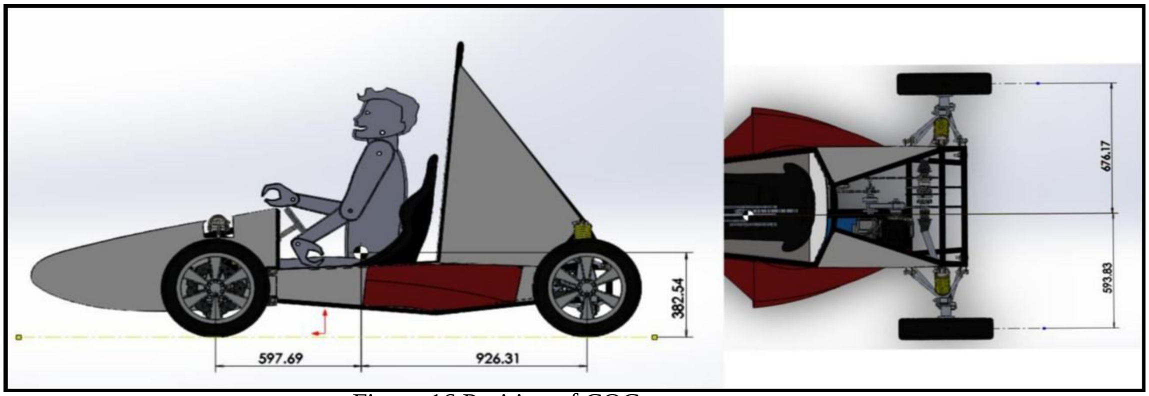


Figure 16 Position of COG

a) Reaction on front wheels due to weight = (597.69 / 1524) × 2452.5 = 961.83N = 0.39G (Upwards)

b) Reaction on rear wheels due to weight = (926.31/ 1524) × 2452.5 = 1490.67N = 0.61G (Upwards) Weight distribution (Front: Rear) = (39:61)

Since, rear inner wheel is most vulnerable to lifting while cornering; we shall consider the reaction of the ground on that wheel only.

c) Reaction on inner rear wheel due to weight = (593.83 / 1270) × 1490.67 = 697.01N = 0.47G (Upwards)

d) Reaction on outer rear wheel due to weight = (676.17 / 1270) × 1490.67 = 793.65N = 0.53G (Upwards)

2) Reaction due to gyroscopic effect on rear wheel: Let us assume radius of turn (R) be 5 m.

a. Radius of wheel (r) =0.22m

b. Track width (w) =1.27 m

c. Gear ratio (G.R) = 0.5

d. Moment of inertia of vehicle (I) =381.21 kgm^2

e. Moment of inertia of engine (I_e) =10.54 kgm^2

f. Gyroscopic couple on rear wheels (C_G) = {(I × v^2) / (r × R)} + G.R × {(I_e× v^2) / (r × R)}

i. {381.21 / (0.22 × 5) + (0.5 × 10.54) / (0.22 × 5)} v^2

ii. 351.32 v^2 Nm

g. Reaction due to couple on each rear wheel = C_G / 2w

i. 351.32 v^2 / (2 × 1.27)

ii. 138.31 v^2 N (Upward)

3) Reaction due to centrifugal force on rear wheels

a. Height of Centre of gravity (h) =0.382 m

b. Couple due to centrifugal force (C_C) = (m × v^2 × h) / R

i. {(250 × 0.382) / 5} × v^2

ii. 19.1 v^2 Nm

c. Force on each rear wheels (F_C) = C_C/2w

i. 7.51 v^2 N (Upward)

4) Maximum velocity attainable at a corner of 5m radius

a) $F_W = F_G+F_C$

b) 745.335 = 138.31v^2+7.51v^2

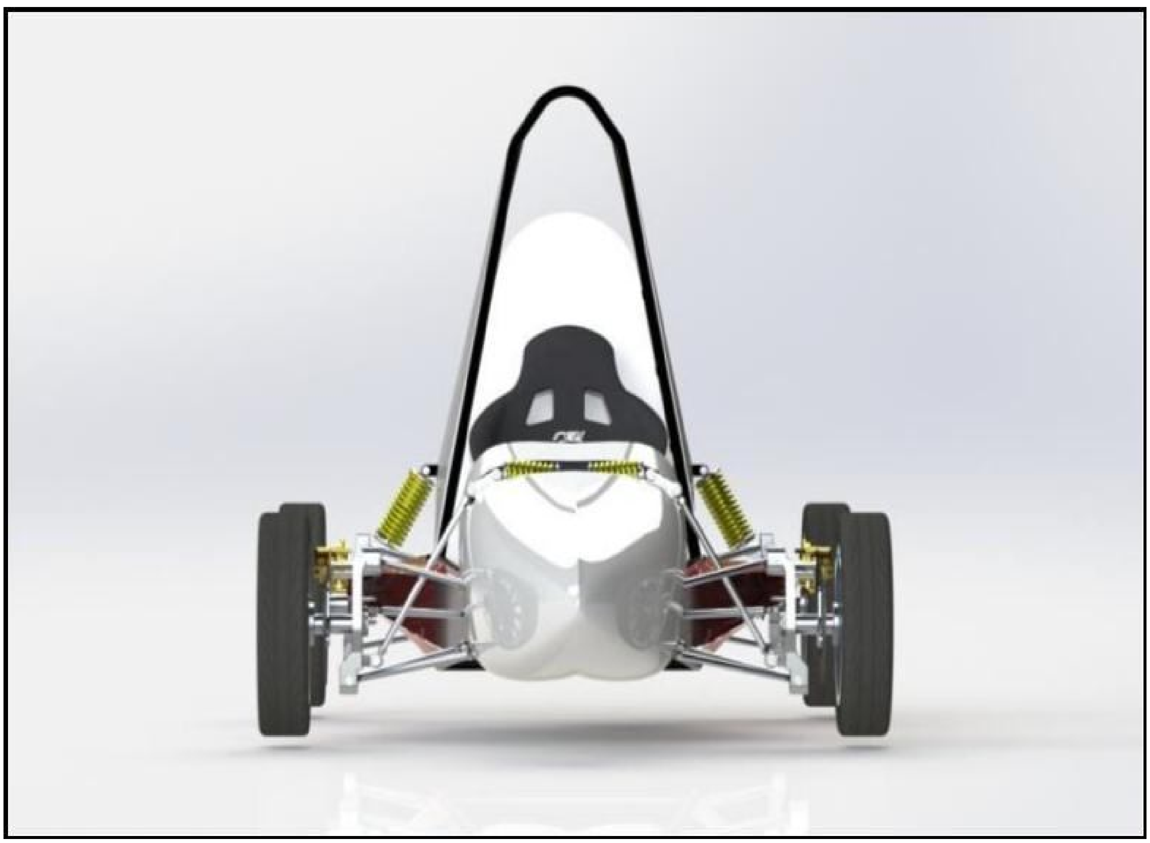
c) v_{max}= 2.26 m/s

Therefore, 2.26 m /s is the maximum speed with which the vehicle can turn without rolling.

G. Different Views of the Vehicle

The CAD model was prepared in Solidworks 2016; all the necessary details were taken care off. The model contains Chassis, Steering System, Braking System, Transmission System and outer body. All the isometric views are shown in Fig. 17.

(a) Isometric View



(b)Front view

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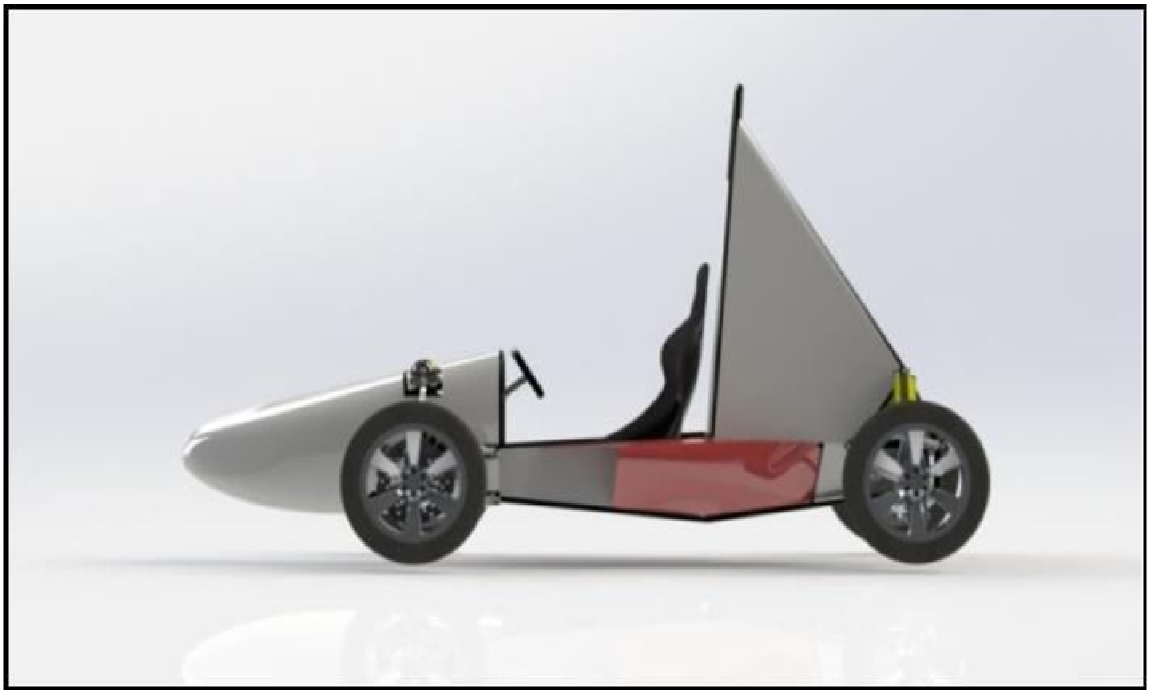
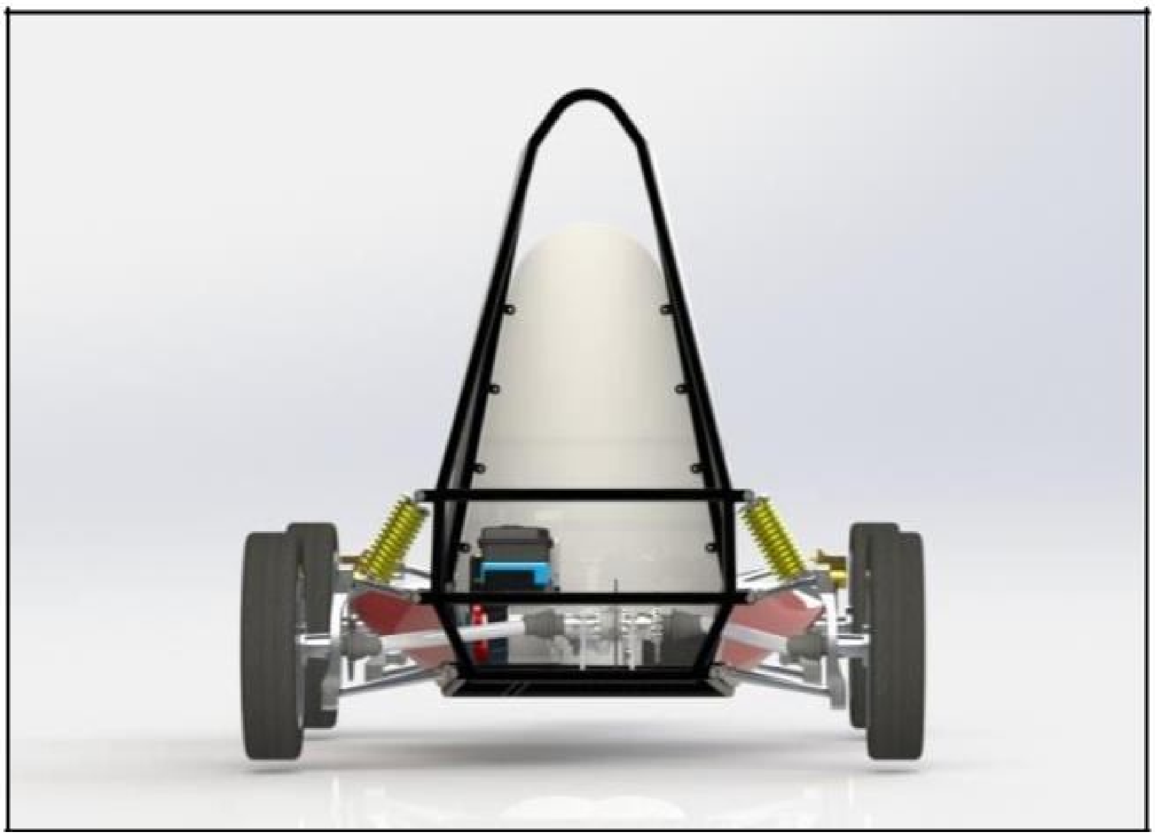


(c) Side view



(d) Rear view

Figure 17 Different views of the vehicle

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III. STEERING SYSTEM

A. *Introduction*

Steering system is a mechanism that provides directional control and stability to the vehicle. It gives complete command on the maneuvering of the vehicle. The system implemented in our project is RACK AND PINION gearbox mechanism.

The design goals behind the mechanism include reducing steering efforts and weight of steering assembly, optimizing the geometry to obtain pure rolling condition.

B. *Design Methodology*

Owing to its simplicity in design and proficiency in obtaining steering ratios, we have chosen RACK AND PINION type steering gear box. It comprises of PINION gear (circular gear) engaging its teeth on the RACK (linear gear), thereby converting rotational motion to linear motion.

In order to achieve perfect steering conditions; steering system according to Ackermann principle was designed. The principle is defined as during turning if I centers of all wheels meet at a point, then the vehicle will take turn about the point which results in pure rolling of the vehicle.

C. *Design Analysis*

A custom rack and pinion of steering ratio 9:1 is used.

1) Wheel base, L=1.524m
2) Wheel track width, W=1.092m
3) Distance between left and right kingpin centre line, B=0.90m The Notations and dimensions are shown in the Fig. 18.

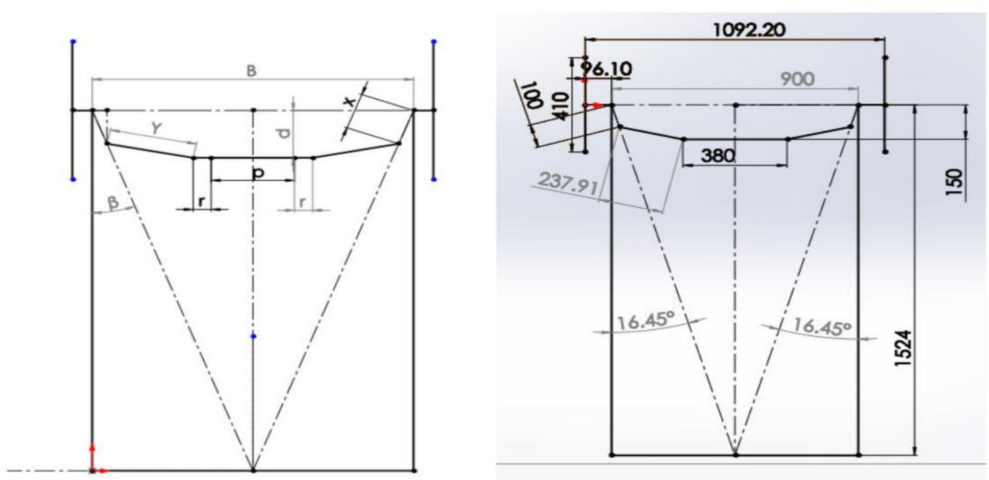


Figure 18 Notations and dimensions of steering system (mm)

After applying Ackerman condition we calculated,

4) Outer wheel angle **δo, Φ** = 28.7°
5) Inner wheel angle **δi , Θ** = 39°

Now using rack and pinion geometry different geometry parameters are calculated (shown in calculations):

6) Difference between front axis and rack centre axis, d = 0.150m
7) Rack ball joint centre to centre length, p+2r=0.380m
8) Ackermann angle, = 16.45°
9) Steering arm length, x= 0.100m
10) Tie rod length, y= 0.23791m
11) Travel of rack, q= 0.0412m
12) x= steering arm length
13) y= tie-rod length (in top view)

14) p= rack casing length
15) p+2r= rack ball joint center to center length
16) q= travel of rack
17) d= distance between front axis and rack center axis
18) β= Ackerman angle

$$- \qquad \theta = / \tag{2}$$

The equations used to derive the following values are as follows:-
According to Ackermann conditions
19)Turning radius inner front = L / **Sin** - (W-B)/2

$$- \qquad = \backslash$$

- Turning radius inner rear = L / T**an** - (W-B)/2

According to rack and pinion steering geometry

(3)

$$y^2 = [B/2 \quad (p + 2r) / 2 - x \sin \ ]^2 + [d - x \cos\boldsymbol{\beta}]^2 \tag{4}$$

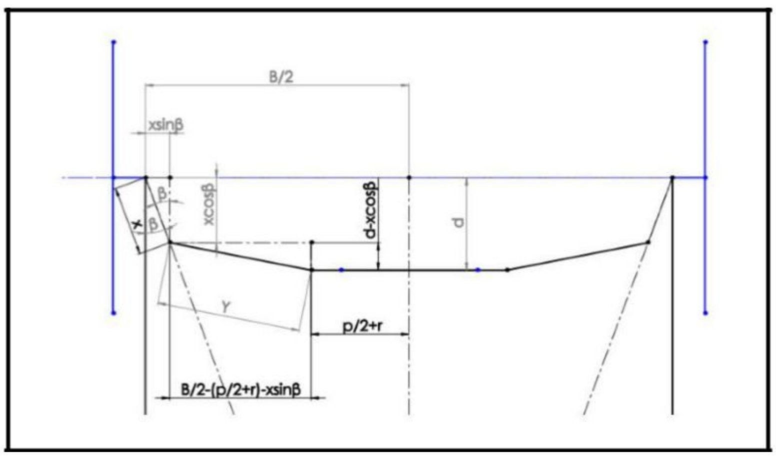


Figure 19 Equation for zero toe condition represented in (4)

$$y^2 = [B/2 - (p/2 + r - q) - x \sin (\boldsymbol{\Theta} + \ )]^2 + [d - x \, \mathbf{cos} \, \mathbf{(} + \ )]^2 \tag{5}$$

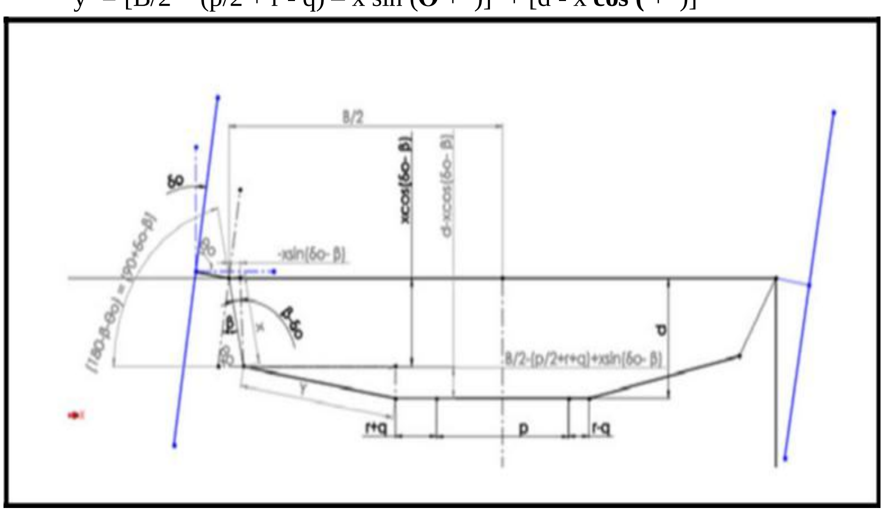


Figure 20 Equation for inner wheel geometry represented in (5)

$$y^2 = [B/2 - (p/2 + r - q) - \mathbf{x \sin (\Phi -} \ )]^2 + [d - \mathbf{x \cos (\Phi -} \ )]^2 \tag{6}$$

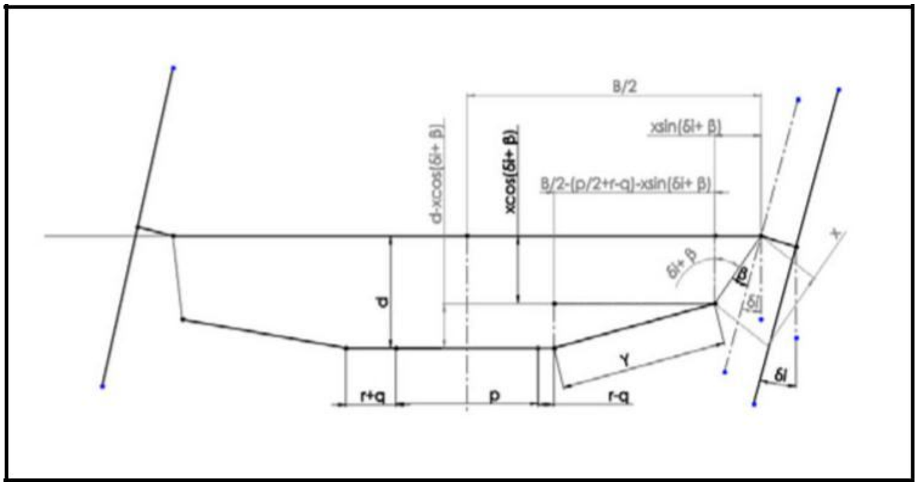


Figure 21 Equation for inner wheel geometry represented in (6)

D. Calculations

From (2)

- CotΦ – CotΘ = B/L
- CotΦ – Cot39 =0.9/1.524
- CotΦ = 1.82
- Φ=28.7°
- Θ=39°

So,

Tan =B\2L

-

= 16.45°

Now, P+2r= 0.38m

Because of restriction due to knuckle design, x = 0.100m

On putting respective values in the Eq. (4), Eq. (5), Eq. (6) we get the values of d, y, q as

- d = 0.150m
- y = 0.23791m
- q = 0.0412m

Now,

Turning radius = (L/ sin) **-** (a-c/2)

= (1524/sin39) – 96.1

= 2.32m

E. Conclusion

The calculated turning radius comes to be in between 2m and 2.5m which is the range for stable steering system; hence we can say that the value of 2.32m will favor stability in accordance with minimum turning radius possible.

F. Steering System Model

Different constituent components of the system are shown in Fig 22

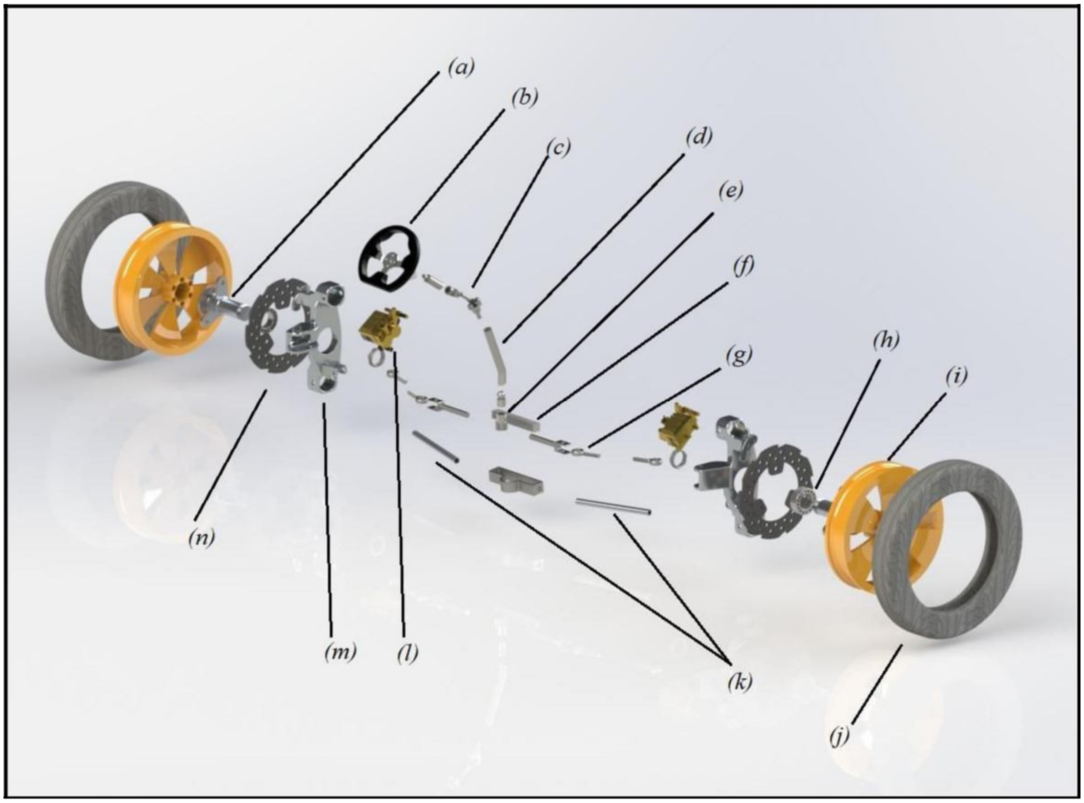


Figure 22 Burst view of the steering and front wheel geometry

Table 5 components in Fig 22

Serial	Part	Material	Weight (Kg)	Quantity
(a)	Front wheel hub	Aluminium 1060-H12	0.62	2
(b)	Steering wheel	Composite	0.18	1
(c)	Universal coupling	Grey cast iron	0.21	1
(d)	Steering column	AISI 1020 Steel	0.29	1
(e)	Pinion	Grey cast iron	0.05	1
(f)	Rack	Grey cast iron	0.43	1
(g)	HEIM joint	AISI 4340 Steel	0.08	4
(h)	Cylindrical roller bearing	Low carbon steel	0.25	2
(i)	Wheel rim	Aluminium 1060-H12	4.54	2
(j)	Tire	Styrene butadiene rubber	4.69	2
(k)	Tie rods	Aluminium 1060-H12	0.06	2
(l)	Dual piston floating caliper	Brass composite	2.13	2
(m)	Front upright	AISI 4130 Steel	4.34	2
(n)	Front wheel rotor	Grey cast iron	1.16	2

1) Front wheel hub

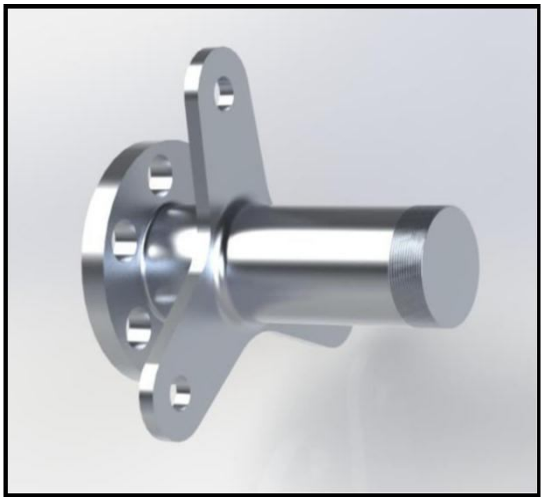
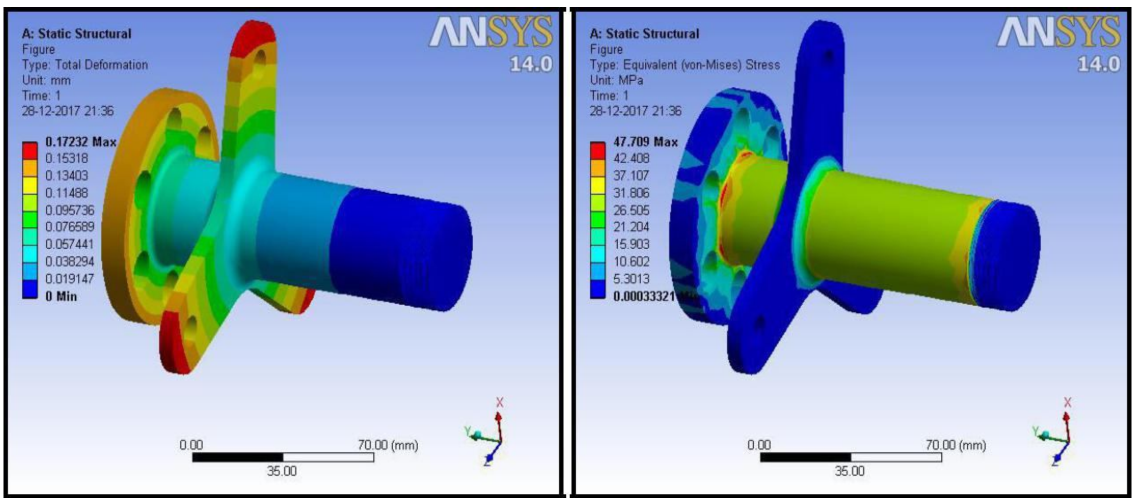


Figure 23 CAD of Front wheel hubs



(a) Static structural Deformation analysis (b) Static structural Stress analysis

Figure 24: Front hub analysis

2) Rear wheel hub

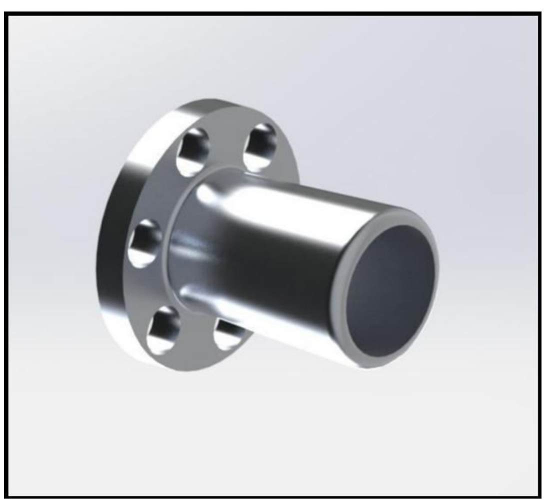
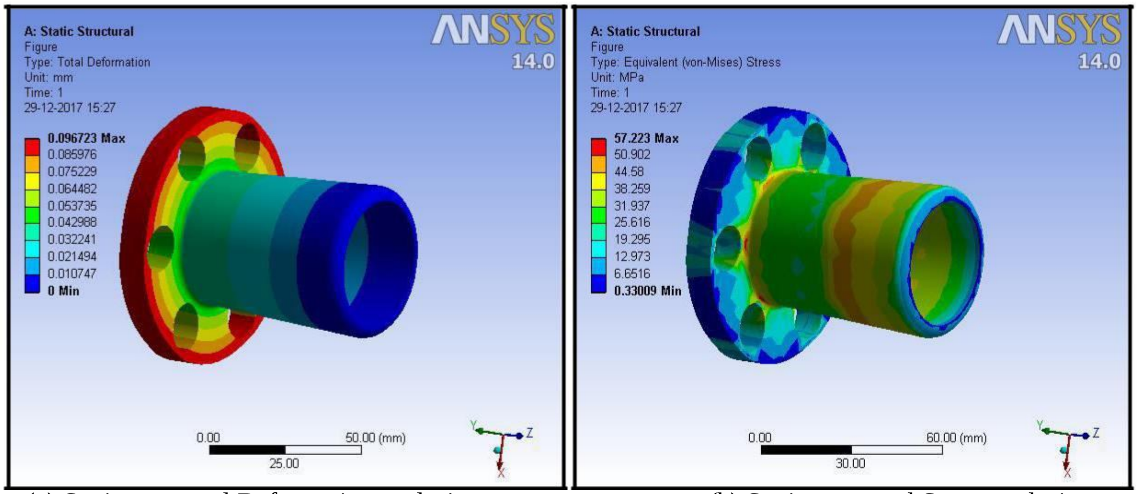


Figure 25 CAD of rear wheel hub



(a) Static structural Deformation analysis (b) Static structural Stress analysis

Figure 26: Rear hub analysis

G. Weight of Steering Assembly

1) Weight of steering column (with steering wheel)- 0.47 Kg
2) Weight of tie rods- 0.06Kg (both)
3) Weight of universal joint- 0.21Kg
4) Weight of steering arms-0.40kg(both)
5) Weight of Rack and Pinion- 0.48Kg
6) HEIM joints (4)- 0.24Kg
7) Extra casing and fixtures- 0.5Kg

Total weight of steering geometry is 2.36Kg (Approx.)

IV. BRAKING SYSTEM

A. Introduction

The brakes are one of the most important safety systems in the vehicle. The car uses brakes to bring the vehicle to a quick and safe stop regardless of weather conditions or topography. The vehicle has Bi- independent hydraulic brake system (i.e.) Disc brakes.
A disc brake is a type of brake that uses calipers to squeeze pairs of pads against a disc in order to create friction that retards the rotation of a shaft, such as a vehicle axle, either to reduce its rotational speed or to hold it stationary.

B. Selection of Disc

After calculating the torque required to stop the vehicle we calculated the diameter of disc (i.e. 240mm for front and 270mm for rear).

C. Calipers

We have selected apache calipers for its better performance after doing market survey with greater piston area to have more frictional force.
In accordance with the rule it has bleeding valve on the top.

D. Brake Fluid

We have decided to use DOT 3 brake fluid. It is inexpensive, and available at most gas stations, department stores, and any auto parts store .It is completely compatible with DOT 3.

E. Design Constraints

1) Area Master Cylinder:-387.09 mm^2=0.00038645 m^2

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2) Caliper Piston Area:-1.023 in^2=0.00065999 m^2
3) Rotor Diameter (rear):- 0.27m
4) Rotor Diameter (front):- 0.24m
5) Brake Disc Type: Full

1) Weight transfer

a) Front Weight (Rest): 941.76 N
b) Rear Weight (Rest): 1412.64 N
c) Total Weight: 2354.4N
d) % Front Weight (Static): 40%
e) CG Height : 0.3825m
f) Wheelbase (inches): 60"

F. Calculations

1) The brake pedal

The brake pedal exists to multiply the force exerted by the driver's foot. From Elementary statics, the force increase will be equal to the driver's applied force multiplied .

By the lever ratio of the brake pedal assembly: = 1750N from (7)

Where, (7)

= the force output of the brake pedal assembly = × /

= the force applied to the pedal pad by the driver

1 = the distance from the brake pedal arm pivot to the output rod clevis.

2 = the distance from the brake pedal arm pivot to the brake pedal pad.

Assuming incompressible liquids and infinitely rigid hydraulic vessels, the pressure generated by the master

cylinder will be equal to 4520912.4 FROM (8)

Where,

= the hydraulic pressure generated by the master cylinder. = / **8**

= the effective area of the master cylinder hydraulic piston.

Brake fluid, brake pipes and hoses assuming no losses along the length of the brake lines, the pressure

transmitted to the calipers will be equal to = = 4520912.4

Where,

= the hydraulic pressure transmitted to the caliper.

2) The caliper, Part 1

The one-sided linear mechanical force generated by the caliper will be equal to = × = 4520912.4 × 0.00066051 = 2986.1

Where,

. = the one-sided linear mechanical force generated by the caliper

. = the effective area of the caliper hydraulic piston(s) found on one half of the caliper body

3) The caliper, Part 2

The clamping force will be equal to, in theory, twice the linear mechanical force as equal to = × 2

= 2986.1 × 2 = 5972.21N

Where,

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= the clamp force generated by the caliper.

4) The brake pads

The clamping force causes friction which acts normal to this force and tangential to the plane of the rotor. The friction force is given by = × = 5972.21 × 0.35 = 2090.27

Where, = the frictional force generated by the brake pads opposing the rotation of the rotor
the coefficient of friction between the brake pad and the rotor.

4.6.6=The rotor

- Rotor Diameter = (0.24m) (FRONT)
- Effective Radius of rotor = 0.0975m

- Torque at rotor = (Frictional Force × Effective radius of rotor) Therefore, torque at rotor is (2090.27 × 97.5) =203.80132N-m.

The torque will be constant throughout the entire rotating assembly as follows: Torque at rotor = Torque at tire

- Effective Radius of tire = 0.1537m.

Force at tire = (Torque at rotor/Effective rolling radius of tire)

Therefore, Total braking force generated at front tires = 1325.97 × 2N

=2651.93N = 1.08G

- Rotor diameter (0.27m) (REAR)
- Effective Radius of rotor=0.1225m

- Torque at rotor = (Frictional Force × Effective radius of rotor) Therefore, torque at rotor is (2090.27 × 122.5) = 256.058075N-m.

The torque will be constant throughout the entire rotating assembly as follows: Torque at rotor = Torque at tire

- Effective Radius of tire=0.1537m.

Force at tire = (Torque at rotor/Effective rolling radius of tire)
Therefore, Total braking force generated at rear tires = 1665.97N = 0.67G
Total force generated at tire to stop the vehicle = 1665.97N + 2651.93N = 4317.9N=1.7G Where,

2

= / = 17.99m/

Where,

= the deceleration of the vehicle
Mv=Vehicle Weight Ftotal=Total braking force
Therefore, Deceleration of the vehicle (av) is 17.99m/sec^2

G. Kinematic Relationships of Vehicles Experiencing Deceleration

For a vehicle experiencing a linear deceleration, the theoretical stopping distance of a vehicle in motion can be calculated as follows: (for velocity (v) = 40kmph) = ^{2}/ × 2 = 3.42m
WHERE:
=Stopping Distance

1) Braking Efficiency

Braking efficiency = (Weight of vehicle × 100) / Vehicle's weight by total brake effort=54.52% Weight of Vehicle=240kg

Vehicle's weight by total brake effort=4317.9/9.81=440.15kg

H. Braking System Model

The brake assembly is shown in Fig 27.

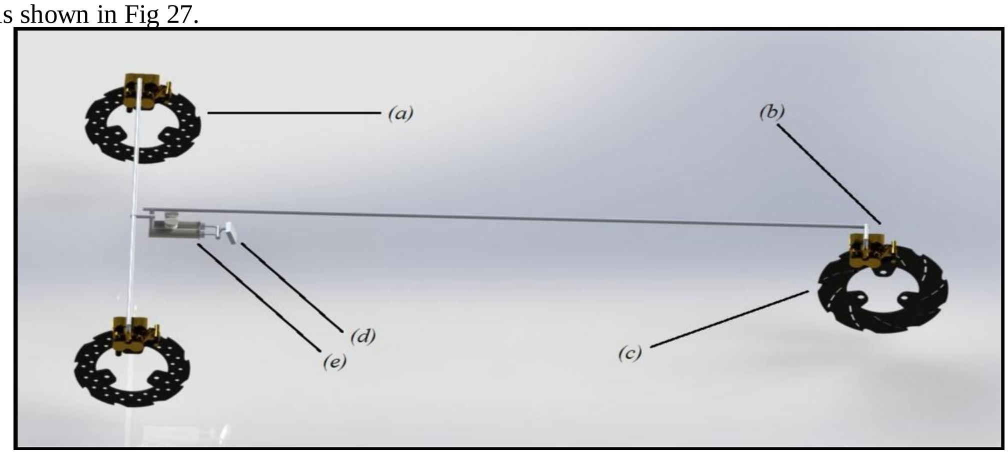


Figure 27 Brake assembly

Table 6 Components in figure 15

Serial	Part	Material	Quantity
(a)	240mm Rotor	Grey cast iron	2
(b)	Dual piston floating caliper	Composite	3
(c)	270mm Rotor	Grey cast iron	1
(d)	Pedal	composite	1
(e)	Master cylinder	Aluminium	1

1) 240mm Rotor

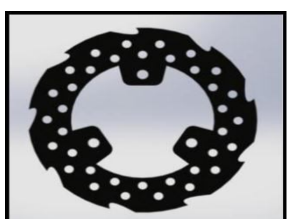
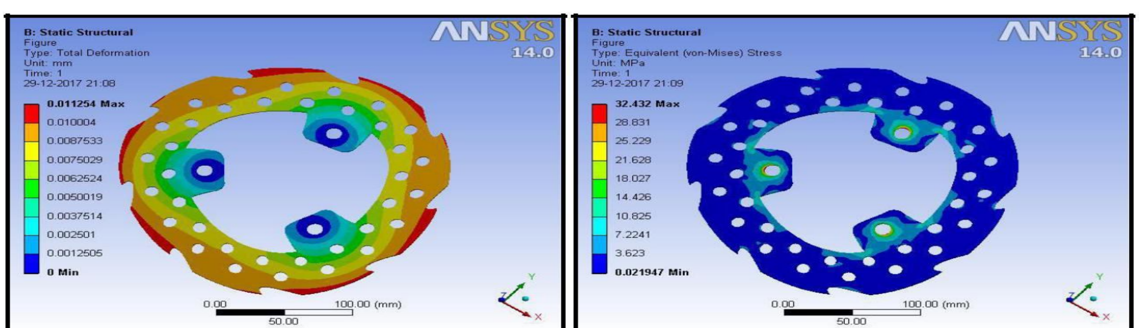
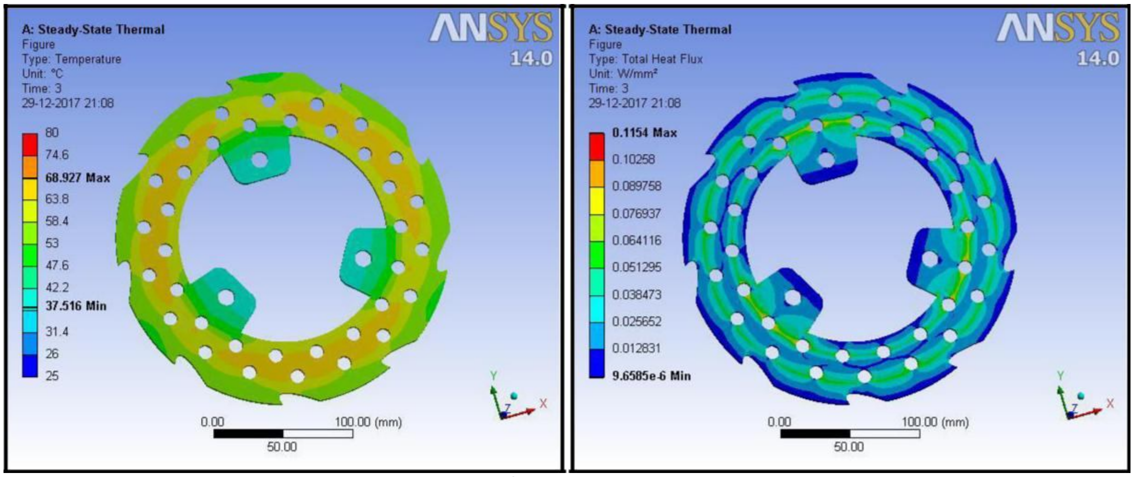


Figure 28: CAD of 240mm Dia. Rotor



(a) Static structural Deformation analysis (b) Static structural Stress analysis



(c) Steady state thermal analysis (d) Steady state heat flux analysis Figure 29: Rotor (240mm) analysis

2) 270mm Rotor

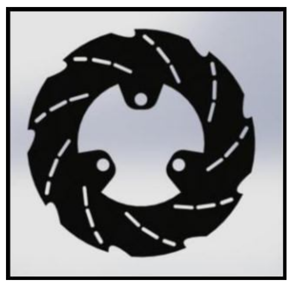
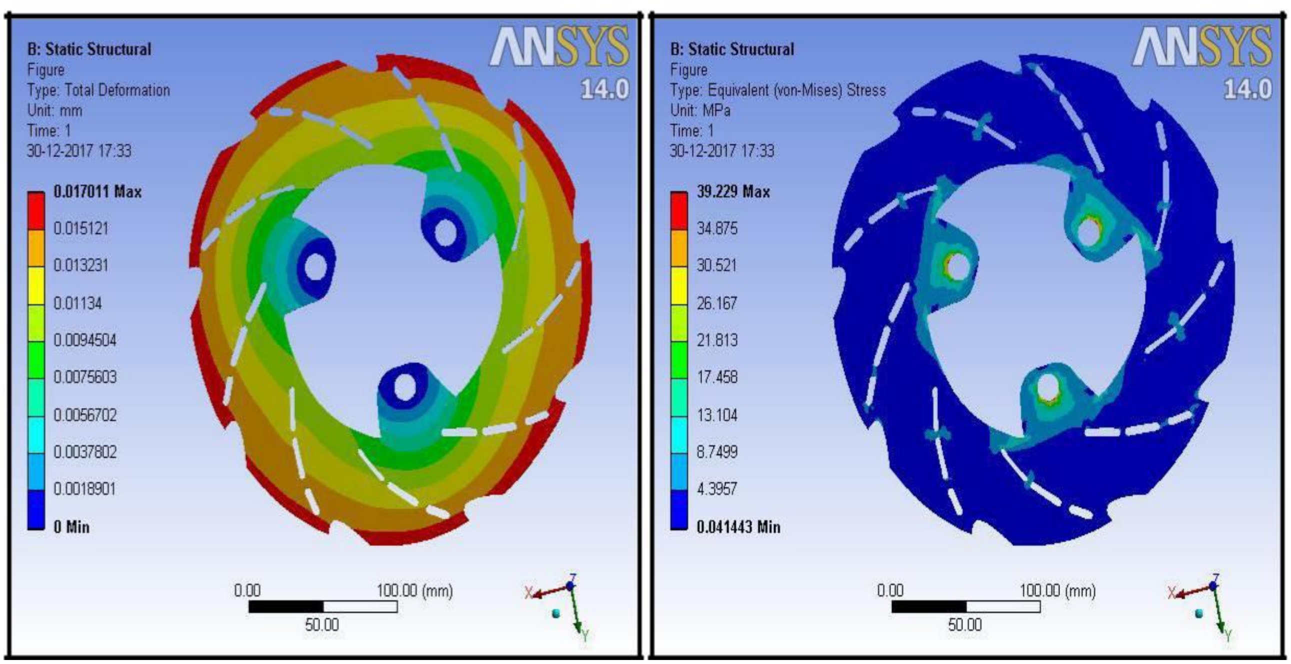
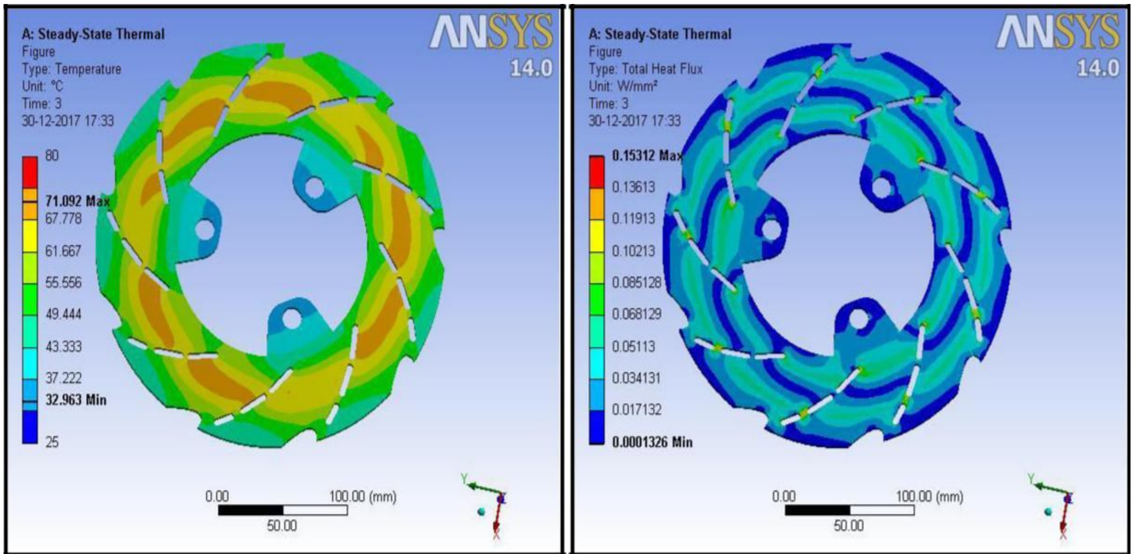


Figure 30 CAD of 270mm Dia. Rotor



(a) Static structural Deformation analysis (b) Static structural Stress analysis



(c) Steady state thermal analysis (d) Steady state heat flux analysis Figure 31: Rotor (270mm) analysis

I.   Conclusion

This braking system provides an effective braking at low and high speed without much of human effort. With an braking efficiency of 54.52%. And a braking distance of 3.42m at 40kmph.

V. SUSPENSION SYSTEM

a)   Front Suspension System

1) Pushrod double wishbone suspension: Push-rod is a rod which pushes up to the rocker & has a strut that runs from the lower wish bone to the upper edge of the chassis.

2)Advantages of pushrod

a) Absence of bulky suspension system.
b) Provide better contact of wheel to the road.
c) Minimize the under-steer & over-steer.
d) Decreases the air drag & turbulence.
e) Motion ratio/spring ratio can be modified with the rocker.

3) Disadvantages of pushrod

Higher CG (center of gravity)

Access to the dampers/springs can be more difficult.

- Another disadvantage is increased friction caused by the increased amount of bearings under high loads.

b) Rear Suspension System

1) *Double wishbone suspension:* This suspension system contains two A-arms, a coil spring mounted between the A-arms.

2) *Advantages*

a) The key reason that we are using the double-wishbone [configuration] is that it gives you a couple of fundamental geometry benefits.
b) The first one probably is that you have got a better opportunity to make kingpin axis more upright than in a MacPherson strut.
c) It is fairly easy to work out the effect of moving each joint, so the kinematics of the suspension can be tuned easily and wheel motion can be optimized.
d) One of its primary benefits is the increase of negative chamber as a result of the vertical suspension movement of the upper and lower arms. This translates to better stability properties for the car as the tires on the outside maintain more contact with the road surface.
e) The double suspension system is much more rigid and stable than other suspension systems, thus you would realize that your steering and wheel alignments are constant even when undergoing high amounts of stress.

3) Disadvantages

a) It is normally bugged by cost issues as it is a more complicated design to produce.

b) This suspension system also proves to be flexible for design engineers, as the arms of the system can be fixed at different angles to the surface, parameters such as camber gain, roll center height and swing arm length can be determined and designed flexibly to suit and road surface in condition.

c) There are many parts to the system, and thus every time any of these malfunction of fail, your whole system fails.

d) Repair, modification and maintenance costs and complexities for double wishbone suspension systems are normally higher due to these reasons.

c) Calculations

1) Total mass of vehicle (excluding the driver) = 200 kg

2) Sprung mass = 170kg

3) Mass distribution = 61:39 (rear : front)

4) Sprung mass per wheel (front) = 34kg; Force=34 × 9.81=333.53N

5) Sprung mass per wheel (rear) = 51kg; Force=51 × 9.81=500.31N

d) Front suspension values

From Fig 32 we can get the below mentioned values.

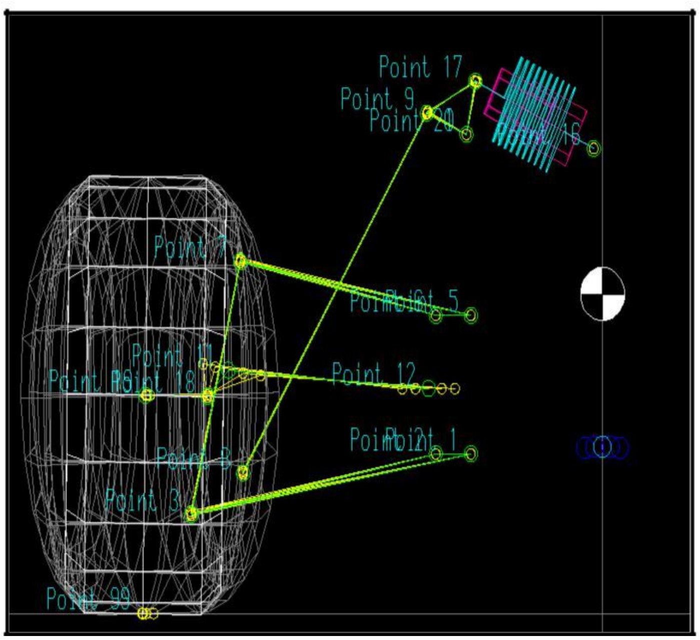


Figure 32 the front suspension geometry from Lotus simulation

1) Motion ratio = [(point 1-point16) / (point 3-point1)] =0.43

2) Angle correction factor = cos(A) =cos 40.82 = 0.75

3) Spring force = 333.53 / 0.43 = 775.65N

4) Adjusted spring force = 775.65 / 0.75 = 1034.2N

5) Spring rate(C) = Adjusted spring force / (Free spring length × 0.60(rear mass)) = (1034.2/(0.18×0.40)) = 14360N/m

6) Wheel rate = Spring rate / MR^2 = 77680N/mm

7) Suspension Frequency = $187.8 \times (WR / \text{Sprung weight})^{0.5}$

$$=187.8 \times (77.68/170)^{0.5}$$
$$=126.94\text{cpm}$$
$$=2.11\text{Hz}$$

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e) Rear suspension values

From Fig 33 we can get the below mentioned values.

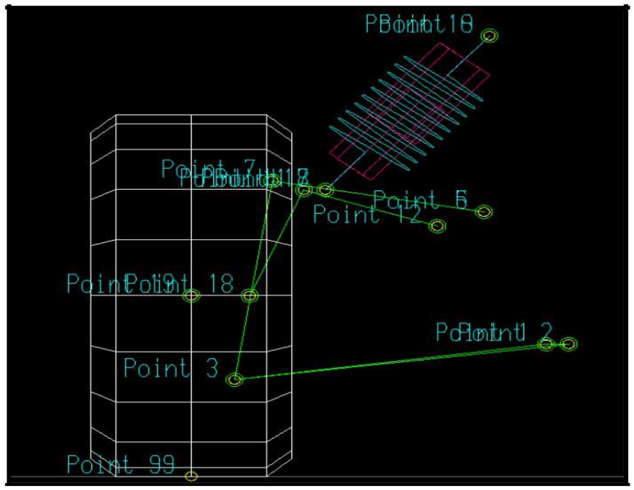

Figure 33 the rear suspension geometry from Lotus simulation

1) Motion ratio = [(point 17 - point1) / (point 19 - point1)] = 0.47
2) Angle correction factor = cos(A) = cos 14.5 = 0.96
3) Spring force = 500.31/ 0.47=1064.4
4) Adjusted spring force = 1064.4/ 0.96=1108.84N
5) Spring rate(C) = Adjusted spring force / (Free spring length × 0.60(rear mass)) = (1108.8 / (0.18 × 0.60)) = 10260N/m
6) Wheel rate = spring rate / MR^2 = 10.26/ 0.22 = 46600N/m
7) Suspension Frequency = 187.8 × $(WR / \text{Sprung weight})^{0.5}$

   $=187.8\times(46.6/170)^{0.5}$

   =98.3cpm

   =1.63Hz

f) Spring calculation

a) Basic value

a) Mass of vehicle: 200Kg

b) Sprung mass:170Kg

c) Material used for spring: Cold Drawn Steel wire

b) Properties of spring

a) Ultimate tensile strength:1050000000 N/m^2

b) Modulus of Rigidity (G): 81370000000N/m^2

c) Spring used: Square ends ground

c) Front spring

a) Mass on front = 200 × 0.4=80

b) Force on single wheel (P) = Mass on front × g/2=392N

c) Motion Ratio: 0.43

d) Taking spring index (C): 7

e) Wahl Factor (K): 1.212 (Formula of K [(4C-1/4C-4) + 0.615/C])

f) According to Indian Standards 4454-1981

Permissible shear stress (S) = 0.6 × UTS=315000000N/m^2

Using Formula S = $[8\times K\times P\times C/ (3.14\times d^2)]$ d=0.00518m

D=0.0363m

g) Rear spring

h) Mass on rear: 200 × 0.6 = 120Kg
i) Force on single wheel (P) = Mass on front × g/2: 588.6N
j) Motion Ratio= 0.47
k) Taking spring index (C): 7
l) Wahl Factor (K): 1.212 (Formula of K =[(4C-1/4C-4) + 0.615/C])
m) According to Indian Standards 4454-1981
Permissible shear stress (S) = 0.6 × UTS=315000000N/m^2
Using formula of S = [8×K×P×C/ (3.14×d^2)] d = 0.00635m
D=7 × 5.004 = 0.0444m

g) Suspension System Model

The suspension assembly is shown in Fig 34.

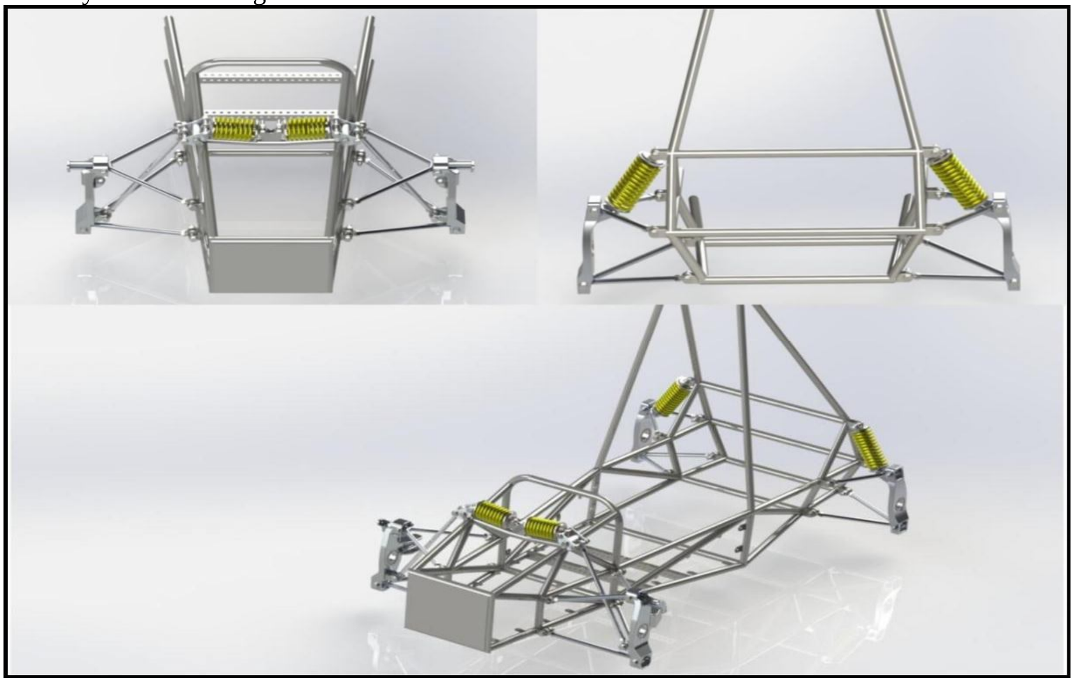

Figure 34 Front and rear suspension geometry

1) Rocker

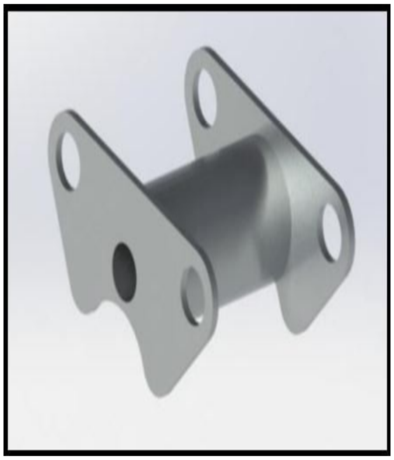

Figure 35 CAD of Rocker

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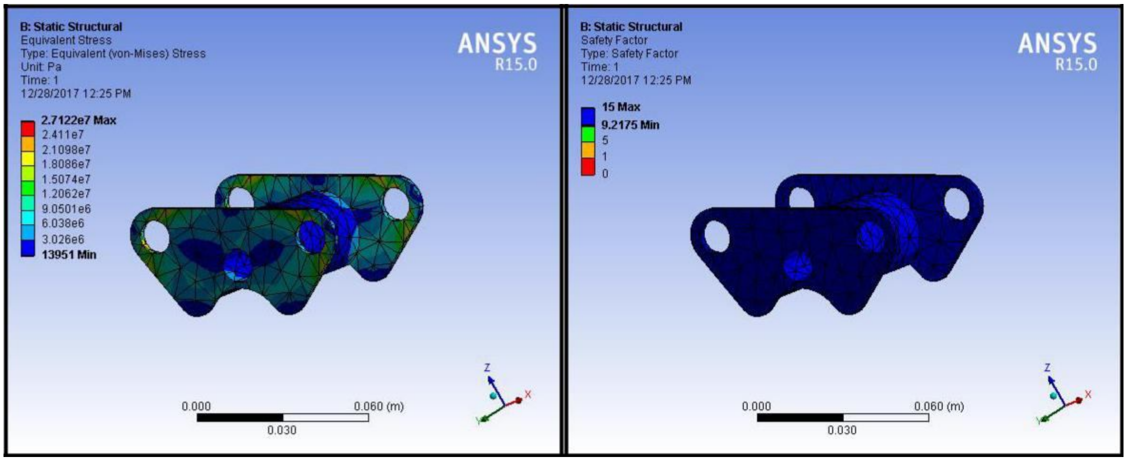
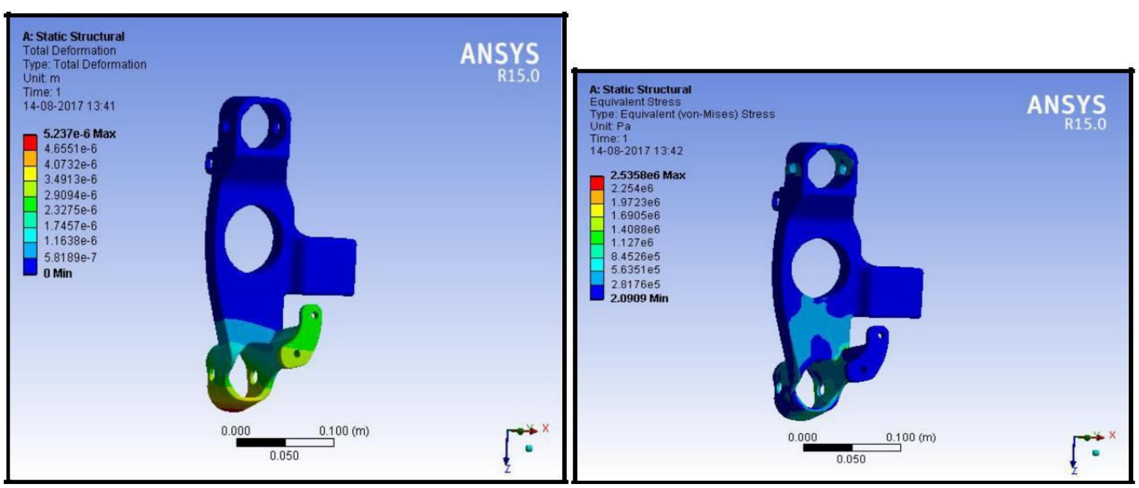


(a) Static structural stress analysis (b) Static structural FOS analysis

Figure 36: Rocker analysis

2) Front upright design

Figure 37 CAD of front upright



(a) Static structural deformation analysis (b) Static structural stress analysis Figure 38: Front upright analysis

3) Rear upright design

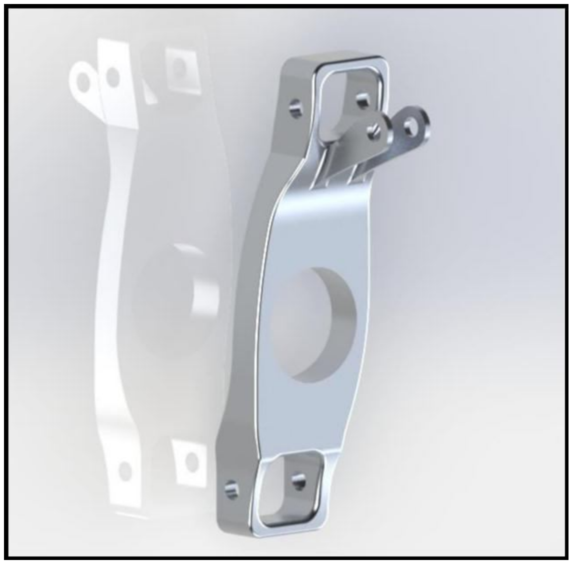
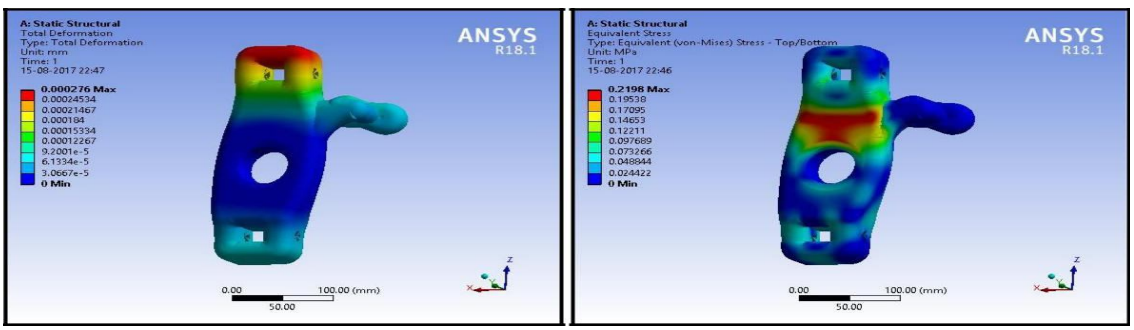


Figure 39 CAD of rear upright



(a) Static structural deformation analysis (b) Static structural stress analysis Figure 40: Rear upright analysis

VI. POWERTRAIN AND ELECTRONICS

A. Powertrain Objective

1) Engine selection & Motor selection.
2) Designing a hybrid system with suitable reductions.

B. Engine Selection

Engine, to be used is Briggs and Stratton 1450 series.

Table 7 Engine Specifications

Displacement	305cc
Cylinder(s)	1
Maximum Power	10hp @ 3600rpm
Maximum Torque	19.65Nm @ 2800rpm
Bore	79.24 mm
Stroke	61.97 mm
Cooling System	Air cooled
Start Type	Electric
Ignition system	Electronic
Fuel Tank	3.78 liters

C. Transmission

CVT (Continuously Variable Transmission) is used to increase the vehicle speed and torque as per the requirements. CVT of the vehicle **"**Honda Activa 125**"** is used because of its variable ratio between 0.6 and 3.1 and the ease of availability.

D. Calculations for Torque Requirement

1) For starting torque

a) Rolling resistance = $(a + b \times V) \times W$
b) Mean value of coefficient a = 0.015
c) Coefficient b = 0.00016
d) Approximate weight of vehicle = 250kg
e) Velocity = 0 m/s … (initial condition)

Therefore,
Rolling resistance = 36.78 N

- Acceleration Force = $m \times a$

Where:
Mass of vehicle = m Acceleration of the vehicle = a

2) For acceleration:

a) $(a / g) = [\mu \times \text{Distance of c.o.g from front axle}] / [L \times (1-\mu \times h/L)]$

Where:
L= Wheelbase
h= distance of cog from ground

b) Therefore, a / g = 0.547
c) Maximum acceleration = 0.547g
d) Acceleration force = 136.75N

Gradient resistance is not considered here because the vehicle will race on a flat track.

e) Total resistance force =Rolling Resistance + Acceleration Force = 173.53 N
f) Torque = Force × radius × μ
g) Radius = 0.21m
h) Coefficient of friction μ = 0.9
i) Starting torque = 32.79 Nm

3) Dynamic resistance

a) Aerodynamic drag= $0.5 \times C_d \times A \times V^2 \times \boldsymbol{\rho}$

Where:
C_d = Drag coefficient
A= Projected area
V= Velocity of vehicle
ρ = Density of fluid (air)

Therefore, Aerodynamic drag = 157.68 N

b) Total resistance = 157.68 + 173.53 = 331.21 N
c) Total torque required= Total resistance × radius × **μ = 62.59 Nm**
d) Power Demand= Force × Velocity = 331.21 × 22.22 = 7359.48W

We are using Parallel hybrid, therefore the power of motor and engine is added which will give the wheels a max power of 9.5kW (7.5kW engine+ 2kW motor). Hence the power requirement is met and the rest is used for acceleration of the vehicle.

E. Design of Hybrid Drivetrain

Design of Hybrid Drivetrain can be approached by two main methods:

a) Series Hybrid
b) Parallel Hybrid

We have selected Parallel Hybrid Drivetrain because of the following reasons:

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c) Both engine and motor supply torque to the driven wheels and since there is no energy form conversion the energy loss may be less

d) It is compact because there is no need for additional generator.

F. Coupling

In order to couple motor and engine, a mechanical coupler is used. For coupling, a chain drive differential is used. A sprocket is attached to the motor shaft and with the help of chain it is connected to the sprocket of differential which is the ring gear. The engine shaft is coupled to CVT and the output shaft from CVT is coupled with pinion gear. Spider gear revolves with ring gear at reduced the motor rpm. Now when the engine shaft, i.e. pinion gear rotates at different rpm, the spider gear rotates around itself as well as revolves and transfers power to the other pinion. Here one thing should be noted that pinion gear and ring gear are not in direct contact. Ring gear is mounted on a bearing on the axle shaft.

The other shaft on which power is received is again connected to a differential through chain and sprocket. Reduction ratios are decided according to the required top speed and required torque.

1:3 reduction ratio is used as first reduction and 1:5 reduction ratio for final drive, that is 42 teeth on the first differential (used for coupling) and 70 teeth on the rear differential.

The rear differential used is Torsen limited slip differential. Powertrain is layout shown in Fig 41 and Table 8.

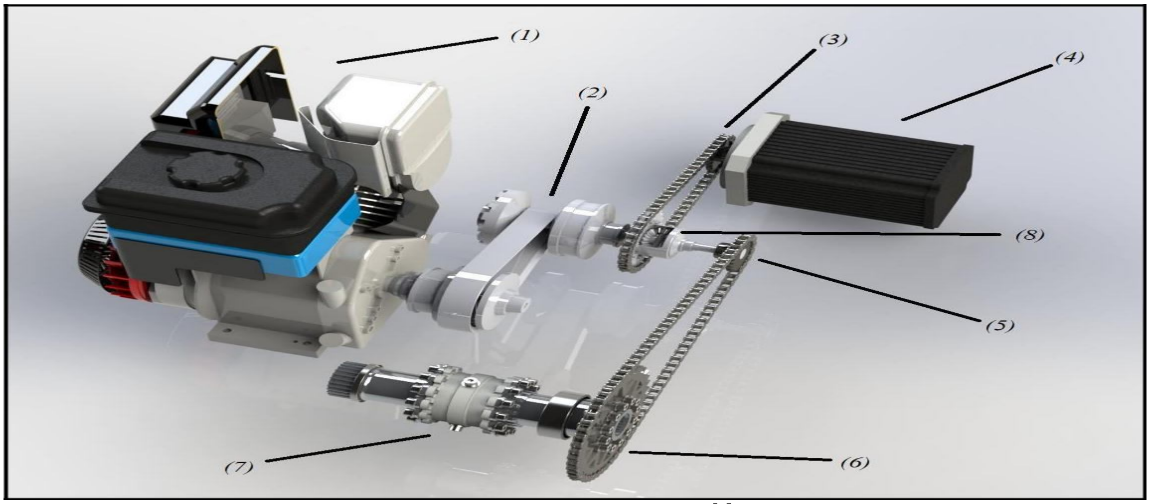

Figure 41 Powertrain assembly

Table 8 Powertrain layout

S.no	Part
1	Briggs and Stratton 1450series 305cc petrol engine.
2	Activa CVT
3	Sprocket-1 (No of Teeth- 14)
4	LBC-05 BLDC Motor 2KW
5	Sprocket-2 (No of Teeth- 14)
6	Sprocket-3 (No of Teeth- 70)
7	Torsen Limited Slip Differential
8	Chain Drive Differential (Wet type)(42 teeth)

1) Half-Shafts

CV joints are used to transmit power from differential to wheels. A CV joint means constant velocity joints, used to drive shaft and to transmit power at a variable angle, at constant rotational speed without much friction.

G. Electrical Objectives

1) Selection of accumulator and motor.

H. Accumulator

The accumulator has been designed in by taking into considerations the endurance, auto cross and acceleration tests the vehicle has to perform for the competition. In the list given below, the main functional requirements have been taken into consideration while designing the battery pack.

1) Functional requirements

a) Supply the maximum allowable power at the desired voltage to the motors
b) Store enough energy to power the car for the entire endurance event
c) Comply with all the safety regulations laid out in the rules
d) Not overheat during any event under reasonable conditions
e) Not add significant weight above that of the bare cells
f) Able to be disassembled quickly and non-destructively

I. Selection of Battery Chemistry

Batteries form the main source of power from the solar panels to run the BLDC motor. When battery is in charging mode electrical energy is converted into chemical energy and while in discharging mode chemical energy is converted into electrical energy.

1) Li-ion over lead acid battery

a) Weight: Lithium-ion batteries are one-third the weight of lead acid batteries.

b) Efficiency: Lithium-ion batteries are nearly 100% efficient in both charge and discharge, allowing for the same amp hours both in and out. Lead acid batteries' inefficiency leads to a loss of 15 amps while charging and rapid discharging drops voltage quickly and reduces the batteries' capacity.

c) Discharge: Lithium-ion batteries are discharged 100% versus less than 80% for lead acid. Most lead acid batteries do not recommend more than 50% depth of discharge.

d) Cycle Life: Rechargeable lithium-ion batteries cycle 5000 times or more compared to just 400-500 cycles in lead acid. Cycle life is greatly affected by higher levels of discharge in lead acid, versus only slightly affected in lithium-ion batteries.

e) Voltage: Lithium-ion batteries maintain their voltage throughout the entire discharge cycle. This allows for greater and longer-lasting efficiency of electrical components. Lead acid voltage drops consistently throughout the discharge cycle.

f) Cost: Despite the higher upfront cost of lithium-ion batteries, the true cost of ownership is far less than lead acid when considering life span and performance.

g) Environmental Impact: Lithium-ion batteries are a much cleaner technology and are safer for the environment. Though they are used to power the same applications that are where the similarity between lithium-ion and lead acid batteries ends. Lithium batteries deliver higher-quality performance in a safer, longer-lasting package.

2) Calculating discharge time

According to engine calculations and reduction calculations we get to know that we will require a battery pack of 85Ah which will be operated at a voltage of 48V. So for calculating discharge time of batteries, Discharge time = (Capacity of battery in Ah × Battery voltage) / (applied load in watt)

Now to calculate the applied load in watt here we have to keep in knowledge that this applied load is actually when the engine is also creating torque and helps to meet the torque requirement of the vehicle.

Therefore, we get the average current requirement in completion of the endurance is of 30A. (Considering the starting current as well as continuous current)

Thus, the applied load on the motor = applied voltage × current = 48 × 30 = 1440W Thus, discharge time = (85 × 48) / 1440 = 2.83hr

Considering efficiency of 90% Discharge time =
2.52hr

J. Motor selection and calculations

The motor used is LBC 05 A motor.

a) Power: 2000W
b) Continuous torque: 8Nm
c) Peak torque: 24Nm
d) DC voltage: 48V

1) Motor Controller

With this we will be using a 35A 36 MOSFET tube BLDC motor controller.

Motor controller is an electronic circuit which controls the speed of the motor by increasing /decreasing the pointer and voltage of potentiometer. Demagnetization of permanent magnet can be prevented by controller by avoiding overloading conditions. Motor controller is purchased with motor matching specifications. Specifications of the motor controller are, 48V 35A (rated) with a peak of 80A.

The placement and positioning of the Powertrain is shown in Fig 42

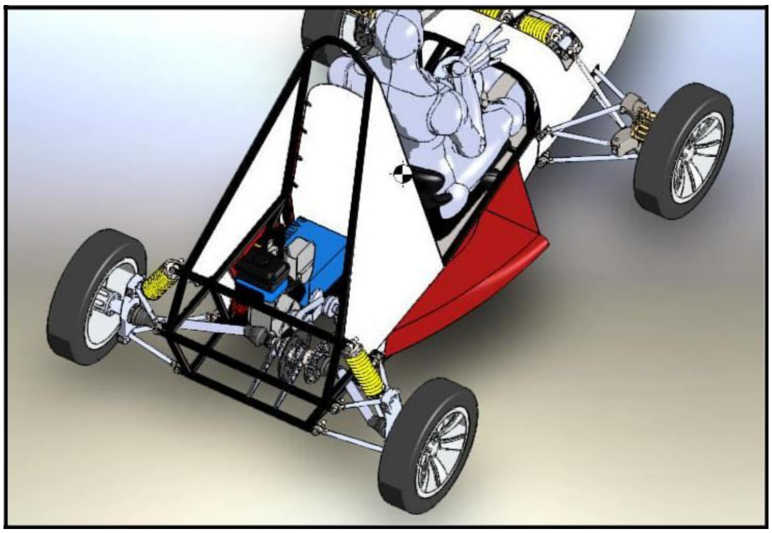

Figure 42 positioning of Powertrain and Electronics

VII. SAFETY AND ERGONOMICS

1) Vehicle dimension are according to Ergonomics, Comfortability and Reachability.
2) Floor Sketches – Hand sketches
3) Ergonomics checked at every design stage

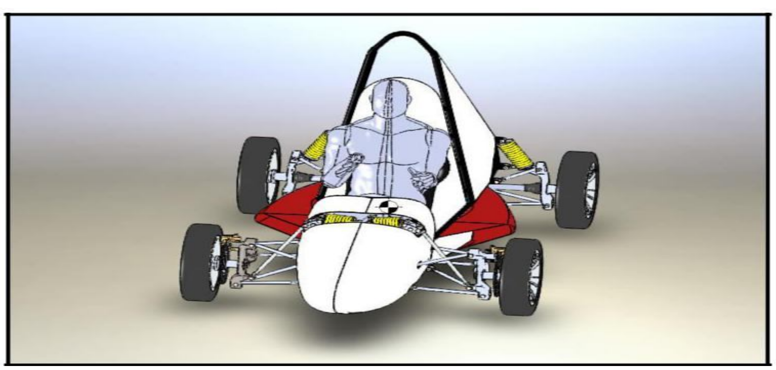

Figure 43 Final Ergonomics model of the vehicle in SolidWorks

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It is very necessary that there is sufficient space between the driver and firewall as shown in figure 44; the driver's feet must reach the pedals comfortably. Also, the engine, fuel tank and battery section should be precisely insulated from firewall.

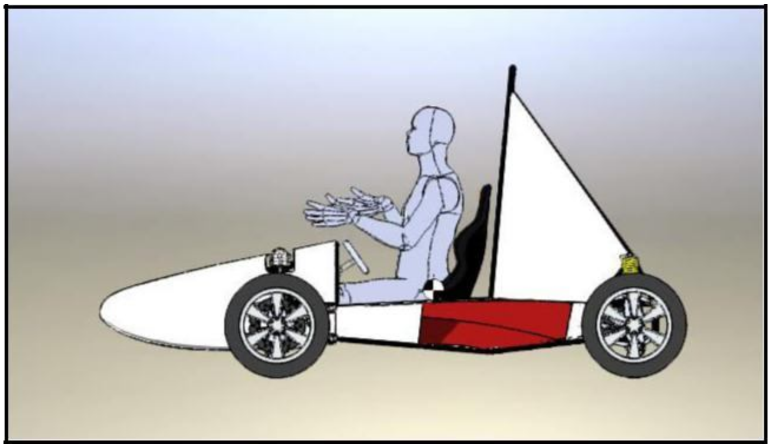
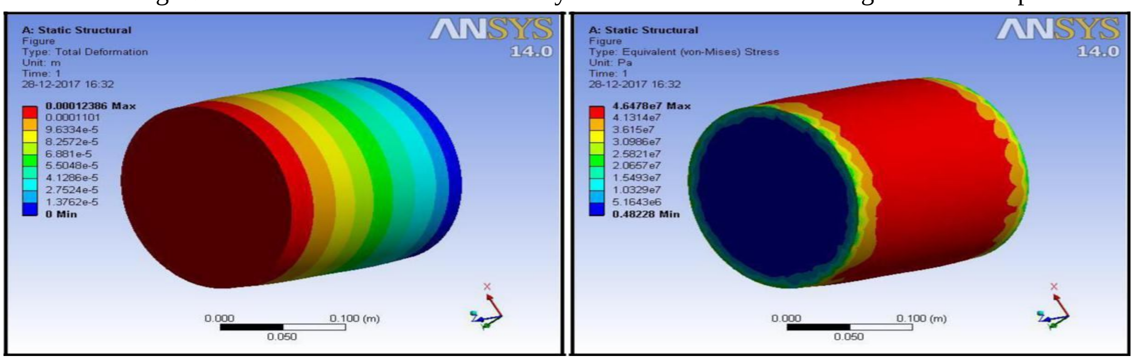


Figure 44 Distance between driver and heat shield

A. Crash box

Specific cylindrical crash box design will best suite the purpose. The design helps in taking more loads over itself as compared to usual cubical designs. The model and deformation analysis has been shown in the figure 20 if a impact load of 11.3G is applied.



(a) Static structural deformation analysis (b) Static structural stress analysis Figure 45: Cylindrical Crash box analysis

VIII. CONCLUSION

In this paper, it was attempted to develop a systematic structure of a race car. All the necessary systems and there key components were elaborately discussed with in line calculations. The CAD was prepared in Solidworks 2016, and the necessary simulations were carried out in Ansys. In an urge to encourage hybrid race cars, the paper clearly shows, hybrid vehicles are ready to take share in race car market. This paper can become a wireframe for new research options and evidence for innovative claims.

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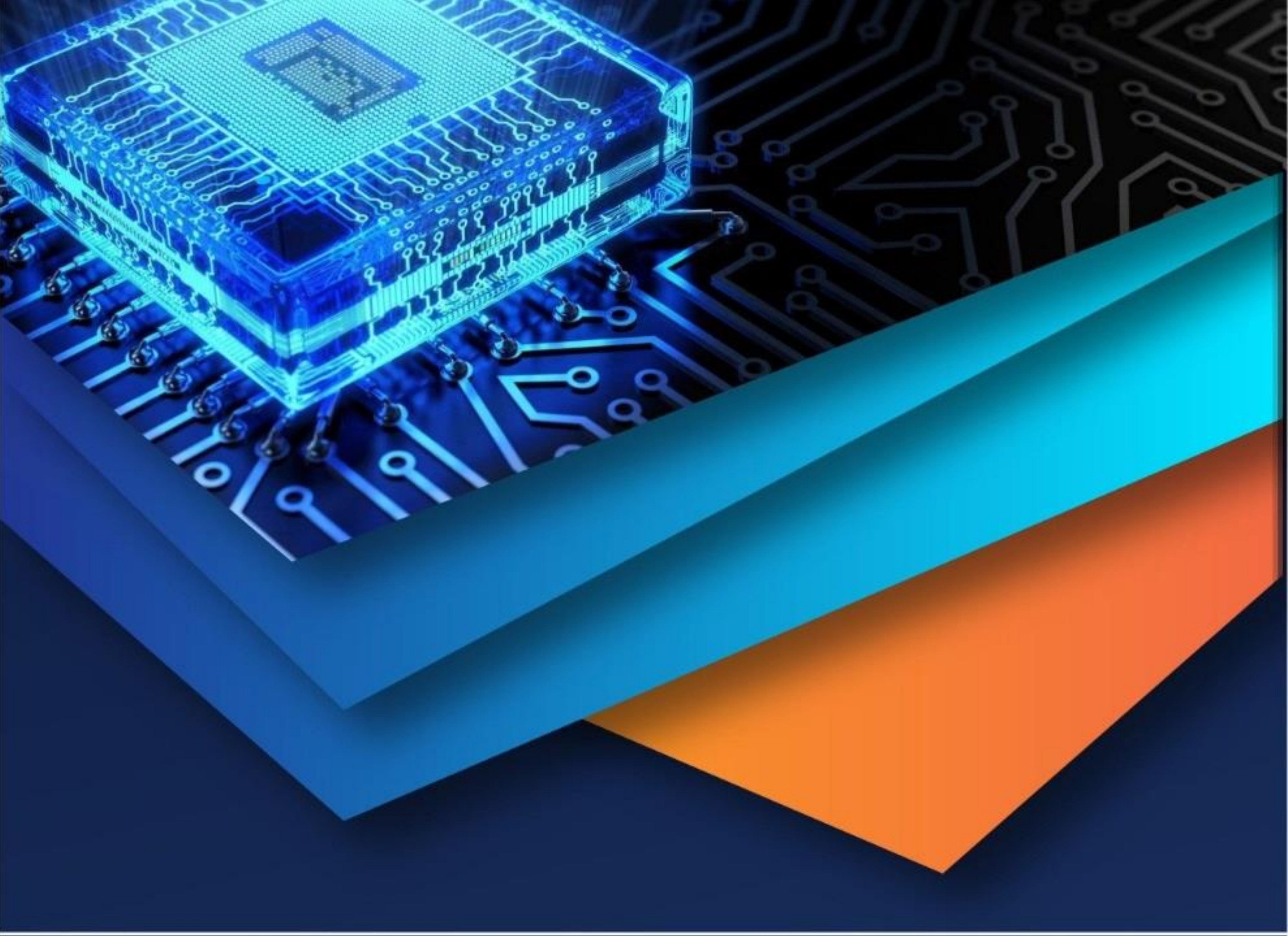

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