To Design and Analyze a Formula Vehicle Based on the FSAE International Rules

Final Project Report

Submitted by:

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Submitted to:

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In partial fulfillment for the award of the degree

Of

BACHELOR OF TECHNOLOGY

IN MECHANICAL ENGINEERING

Declaration

'To Design and Analyze a Formula Vehicle Based on the FSAE International Rules.'
The Project Dissertation is submitted in partial fulfillment of academic requirements for Bachelors of Technology in Mechanical. This dissertation is a result of our own investigation. All sections of the text and results, which has been obtained from other sources, are fully referenced.

We understand that cheating and plagiarism constitute a breach of University regulations and will be dealt with accordingly.

Signature

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Acknowledgement

The progress of any project work is the endeavor of all the individuals that supports, inculcate and foster the much needed enthusiasm and confidence to the doer of the project work without which the whole task prove to be an impossible mission.

We would like to extend our gratitude to our project supervisor, Dr. Ashwini Sharma, for the immense support he extended towards our project. It would not have been possible to conceive this idea without him.

Abstract

The objective of the project was to design a formula vehicle according to the rules specified by FSAE. The vehicle has to be designed on 3D modeling software like Solidworks etc. The challenge faced by the team is to develop a vehicle that can successfully compete in all the events and test runs described by FSAE.

The vehicle conforms to all the guidelines in the rulebook of the event FSAE, which is organized by SAE International. The chassis was designed in SolidWorks 2012. Along with designing of the vehicle chassis, various test were performed using the same software.

Various parameters were set before designing the chassis, for example, the weight was set at an upper limit of 300 Kilograms. The front wheel track width was set at 52 in and the rear wheel track width at 51 in.

The tests performed on the chassis were front impact test, rear impact test, side impact test, roll over test and modal analysis. These tests establish the maximum displacement that the vehicle chassis may undergo in the event of a collision (frontal, rear, side or roll over).

An airfoil was designed as well. This is to increase the downforce at speeds over 60 Km/Hr. This will give the vehicle stability and maneuverability at higher speeds. The airfoil used is NACA 2412. Various tests were performed on the airfoil, which established the drag coefficient and downforce at various speeds (keeping the angle of attack at 13°).

Table of Contents

List of Tables

Table **Page Nos.**

List of Figures

Figure **Page Nos.**

Chapter 1
Literature Review

"The Formula SAE ® Series competitions challenge teams of university undergraduate and graduate students to conceive, design, fabricate and compete with small, formula style, autocross racing cars. Over the course of three days, the cars are judged in a series of static and dynamic events including: technical inspection, cost, presentation, and engineering design, solo performance trials, and high performance track endurance."

The vehicle designed by Cornell University is being discussed here briefly [6]. Their project had a varied scope. It is given as follows:

The determination of most efficient suspension configuration and geometry was their first scope. Then they determined the requirements of the spring and damper systems. Along with this they determined the anti-dive/anti-squat requirements. Following this, they went for the determination of optimal values for camber, caster, and kingpin angles as well as scrub radius. After the steering geometry was determined, the attachment points at wheel, brake, steering rack, axle, and chassis interfaces was determined. Their design based off of existing wheel sand tires and they carried out design synthesis and real-time simulation of complete and functional suspension system. They gave an output of a working, useable suspension system for the 2010-2011 US FSAE car. They maintained a high level of easy adjustability for further tuning of the suspension system.

The choice of their suspension style was a unequal double A-arms. Their justification for using this kind of suspension was that it is the most commonly used kind of suspension, which is almost exclusively used for FSAE vehicles. Secondly, this is the kind of suspension most suited for stiff racing independent suspension and lastly, it is reliable with predictable and calculable motions and forces throughout travel. Their design incorporated adjustable anti dive, anti squat, camber and A-arm lengths. This gave them freedom to change the maneuverability and responsiveness of the vehicle depending on the track conditions.

Their force analysis on the final design centered on maximum cornering and braking forces estimated during competition. This team decided on a goal of structural integrity through a 5g vertical impact. Their estimated braking and turning values surpassed the benchmarked 1.4g.

Chapter 2
Introduction

2.1 Introduction

The approach for this project is to bring out a successful design for a formula vehicle with some new innovations, which would help in increasing the performance of the car in the endurance test.

Before beginning to design such a car, we need to be aware of the requirements of a formula racecar. The guidelines provided by SAE International have been strictly followed.

2.2 Requirements of a Formula Student Vehicle

The requirements for a good formula vehicle are as follows:

i. Less Mass and Overall Dimensions
ii. Low center of gravity
iii. Speed and Acceleration
iv. Maneuverability
v. Safety of the driver
vi. Good Ergonomics
vii. Aesthetics
viii. Ease of parts availability for maintenance
ix. Overall cost of the vehicle

2.3 Targets for the Vehicle

The following are the targets for the vehicle, which is to be fabricated. This is to ensure maximum performance at minimum cost.

i. Restrict the weight to 300kg.
ii. 0-60 Km/hr in around 4 seconds.

iii. Max. Speed up to 110 Km/hr.
iv. Wheelbase of 1600 mm (63 in).
v. Track width of 1400 mm (55.1 in).

Further, as designing is based on prevention of failure, the condition of failure for each system of the vehicle is as follows:

i. The frame is designed in such a way that the relative position of various mountings should be unaffected due to any kind of loading.
ii. The brakes are designed in such a way that the all four wheels will lock simultaneously.
iii. The tires are chosen in such a way that it may have maximum traction with the track surface.
iv. The suspensions are designed in such a way that they may completely isolate the driver from the road shocks at the same time maintaining their stiffness to ensure maneuverability.
v. The steering is designed in such a way that the driver may apply the least effort to steer the vehicle.

Hence, the designing process incorporates all these parameters to ensure the proper working of this vehicle.

2.4 Vehicle Design Objectives

The following are the objectives that were followed while designing the formula vehicle:

i. For the purpose of the Formula SAE competition, teams are to assume that they work for a design firm that is designing, fabricating, testing and demonstrating a prototype vehicle for the non- professional, weekend, competition market.
ii. The vehicle should have very high performance in terms of acceleration, braking and handling and be sufficiently durable to successfully complete all

the events described in the Formula SAE Rules and held at the Formula SAE competitions.

iii. The vehicle must accommodate drivers whose stature ranges from 5^{th} percentile female to 95^{th} percentile male and must satisfy the requirements of the Formula SAE Rules.

iv. Additional design factors to be considered include: aesthetics, cost, ergonomics, maintainability, manufacturability, and reliability.

Once the vehicle has been completed and tested, your design firm will attempt to “sell” the design to a “corporation” that is considering the production of a competition vehicle. The challenge put forward to us is to develop a prototype car that best meets the FSAE vehicle design goals and which can be profitably marketed.

2.5 Methodology

The following method was followed by us while executing the project:

i. A detailed study the FSAE rulebook was carried out.

ii. The vehicle parameters were assumed.

iii. The kinematic suspension model was designed.

iv. Preliminary modeling of chassis according to assumed parameters in was done in Solidworks 2013.

v. Manual analysis of chassis was carried out.

vi. The dimensions were fine-tuned according to preliminary analysis to bring them within the assumed factor of safety.

vii. The analysis of the chassis according to the redefined dimensions was carried out in Solidworks 2013.

viii. The powertrain was selected and all the other components were designed according to the rulebook guidelines. Also, selecting the optimum kinematic suspension model and steering geometry for the car according to track and wheelbase of the racecar was carried out.

ix. Placement of components in the vehicle to balance its weight was accurately done.

x. Designing of outer body to increase aesthetic appeal and aerodynamics was carried out.

xi. The aerodynamic devices were designed to increase downforce and reduce lift while reducing drag as well.

xii. The entire vehicle was analyzed along with the aerodynamic devices attached to the vehicle.

Chapter 3
Problem Definition

3.1 Introduction

This chapter deals with the definitions of various terms that have been used in this report along with listing out the assumptions that have been considered for the purpose of designing the vehicle.

It also describes in depth, the kinematic suspension model, and its importance in a race-ready vehicle. Anti-squat and anti-dive have been introduced here. In the end, the technical specifications of the engine to be used have been tabulated.

3.2 Assumptions

The following are the data that have been assumed for the purpose of designing the chassis and the kinematic suspension model.

The following assumptions are in accordance with the FSAE guidelines 2013[1]:

i. Wheelbase: 1600 mm (63 in)
ii. Front Track: 1320.8 mm (52 in)
iii. Rear Track: 1295.4 mm (51 in)

The following assumptions are based on the literature on vehicle dynamics and the kinematic suspension model of a vehicle. This is used for the designing of the vehicle chassis:

iv. Static Camber: (Front & Rear) -1°
v. Caster: (Fr) 5°; (Rr) 0°

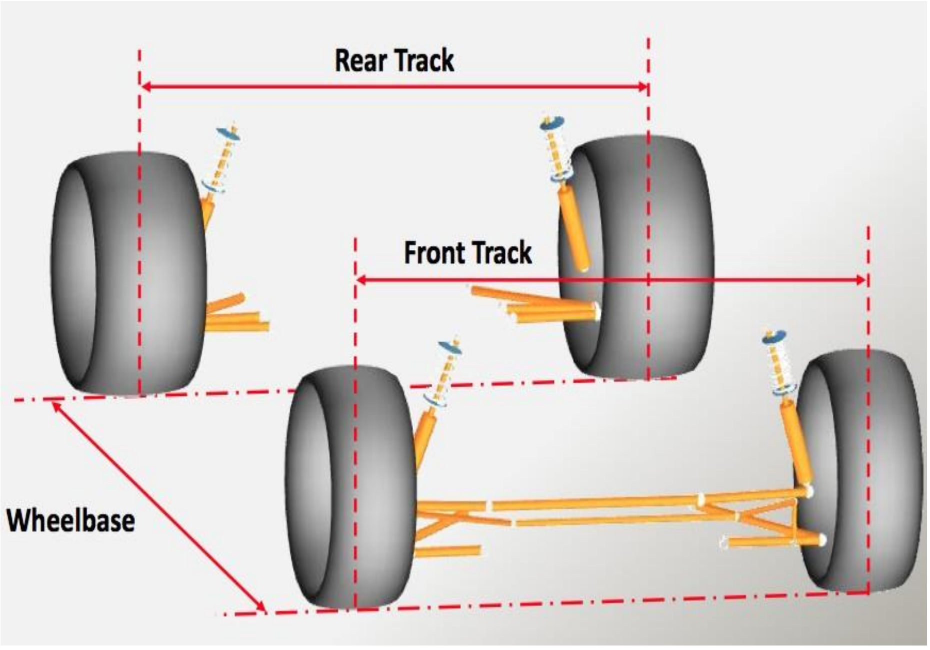


Fig 1. Wheelbase, front track, rear track

vi. KPI Angle: (Fr) 3° from Camber; (Rr) 1° from Camber
vii. Ground Clearance: (Fr) 3 in; (Rr) 6.5 in
viii. Front to Main roll hoop distance: 850 mm (33.46 in)
ix. Front Mount point difference: 304.8 mm (12 in)
x. Rear Mount point difference: 254 mm (10 in)
xi. Horizontal distance from chassis to wheel pivot (front): 465.4 mm (18.3 in)
xii. Horizontal distance from chassis to wheel pivot (rear): 495.3 mm (19.5 in)
xiii. Main hoop center height: 1025.4 mm (40.4 in)
xiv. Front hoop height: 625.4 mm (24.6 in)
xv. Overall length: 2245.4 mm (88.4 in)

3.3 Kinematic Suspension Design

Definition

Camber angle: is the angle between the wheel axis and vertical axis as viewed from the front or the back of the car. Maximum cornering force is achieved when the camber of the outside wheels relative to the ground is about -1°. A slight negative camber in a turn maximizes the tire contact patch due to the way the tire deforms under lateral load. Hence, it is good to have some negative camber to increase cornering force.[2]

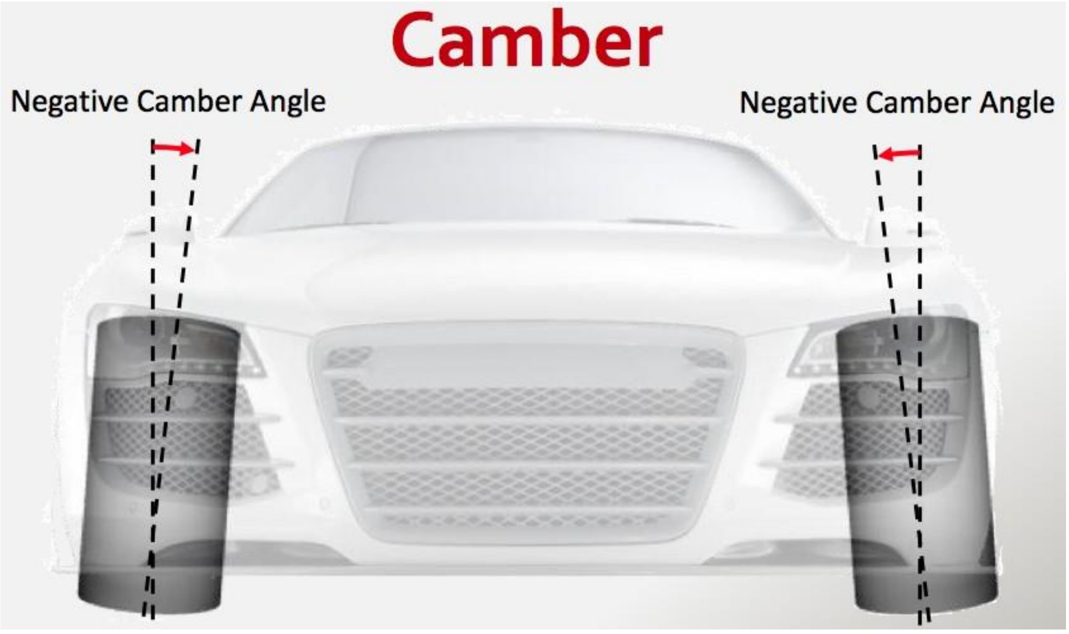


Fig 2. Camber

Another reason why it is helpful to align your suspension with a slight negative camber is that camber will change with suspension travel and body roll. Most suspension systems are designed so that camber increases with more suspension travel.[2]

Caster angle: is the angular displacement from the vertical axis of the suspension of a steered wheel in a car, bicycle or other vehicle, measured in the longitudinal direction.

The pivot points of the steering are angled such that a line drawn through them intersects the road surface slightly ahead of the contact point of the wheel. The purpose of this is to provide a degree of self-centering for the steering - the wheel casters around so as to trail behind the axis of steering. This makes a car easier to drive and improves its directional stability (reducing its tendency to wander). Excessive caster angle will make the steering heavier and less responsive, although, in racing, large caster angles are used to improve camber gain in cornering. Caster angles over 4° with radial tires are common. Power steering is usually necessary to overcome the jacking effect from the high caster angle.[3]

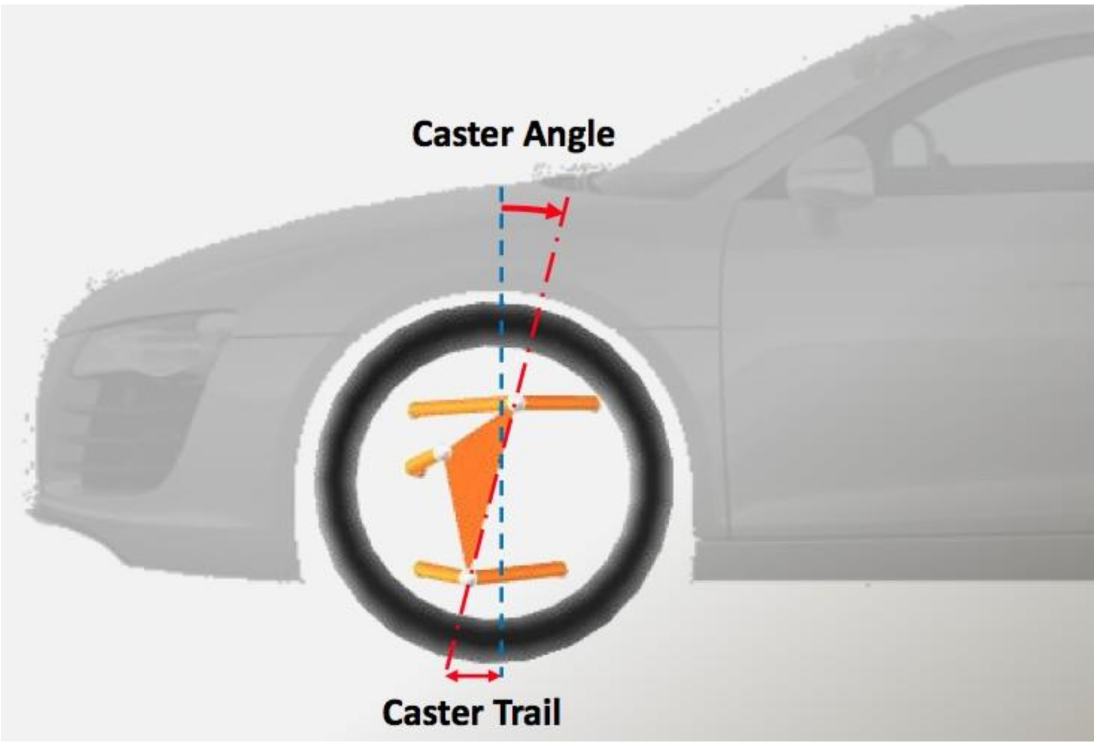


Fig 3. Caster angle and caster trail

KPI angle: it stands for King Pin Inclination angle.

It is the transverse angle of the swivel axis of the front wheel and its stub axle. The effect of the inclination is usually discussed in terms of the king pin offset which determines the self-centering torque when the steering is turned for cornering. Although many cars have a positive value of offset which tends to return the wheel to the straight ahead position, some modern cars have a negative offset to improve stability when the tire blows or the brake fails on one front wheel.

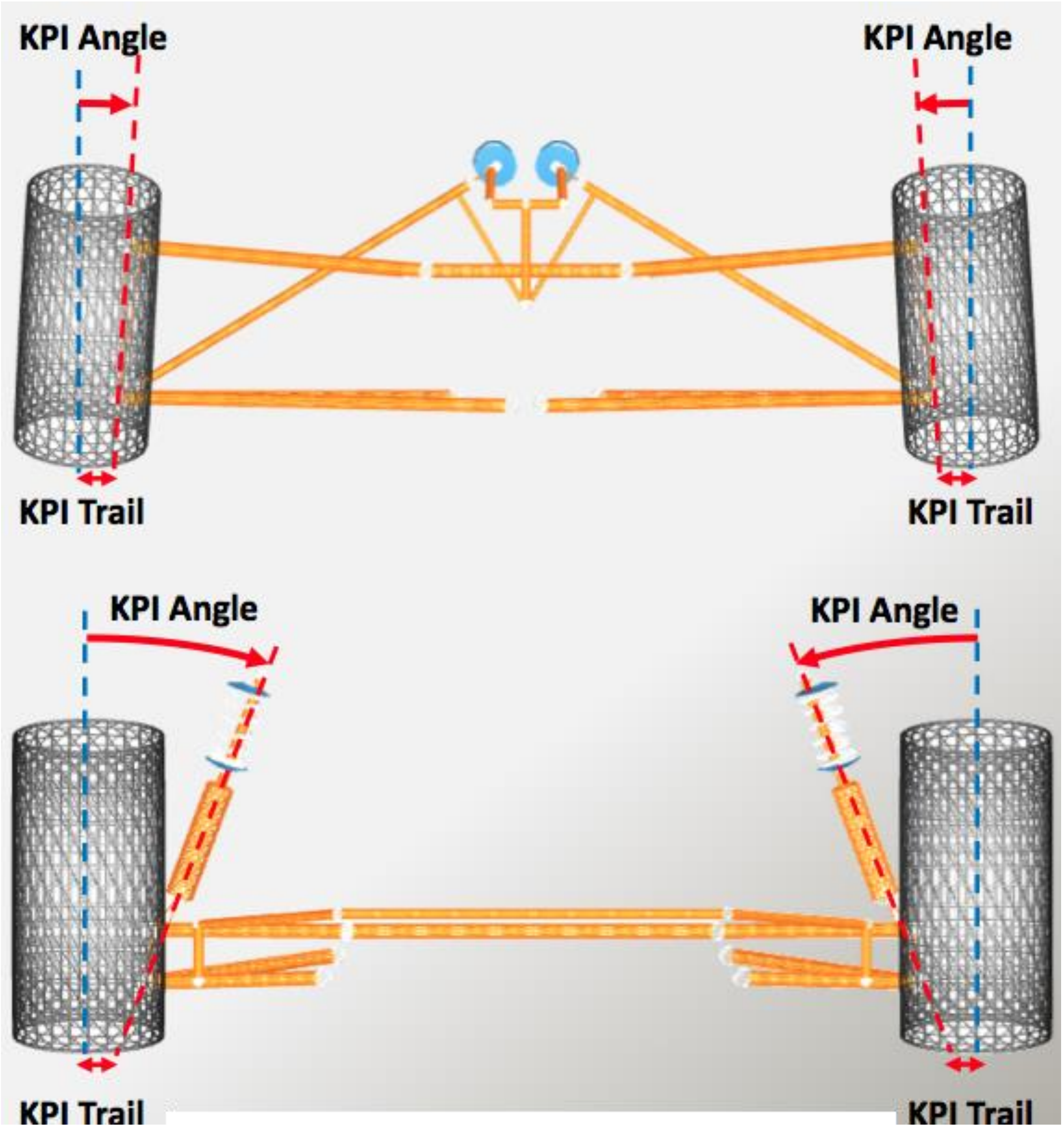


Fig 4. KPI angle and KPI Trail

Instant Center: The "hinge" around which one corner of the wheel rotates.

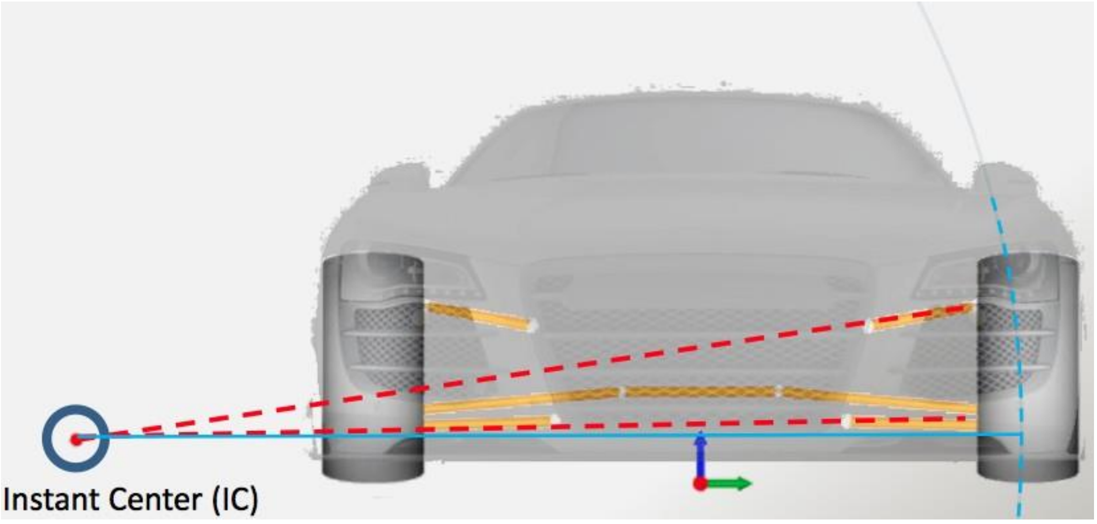


Fig 5. Instant Center

Roll Center: SAE defines roll center as: The point in the transverse vertical plane through any pair of wheel centers at which lateral forces may be applied to the sprung mass without producing suspension roll.

We can think of it as the "hinge" around which the chassis rotates. This is a simplification of reality, but is still valid for basic calculations.

Roll center/Instant centers determine how much of the weight transfer is reacted through the springs.

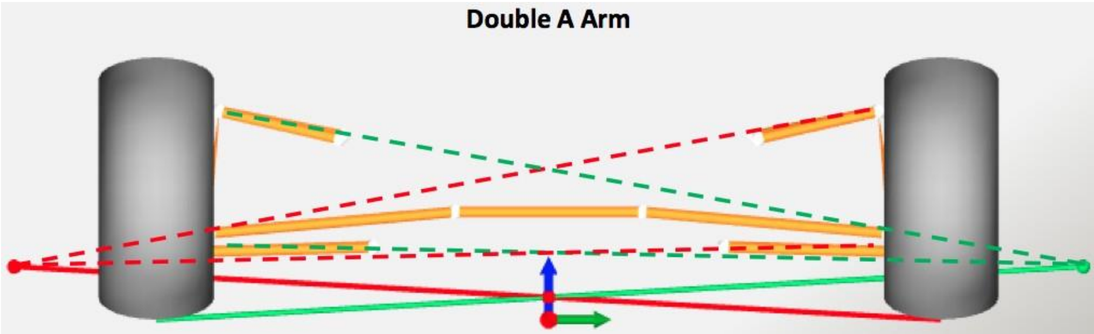
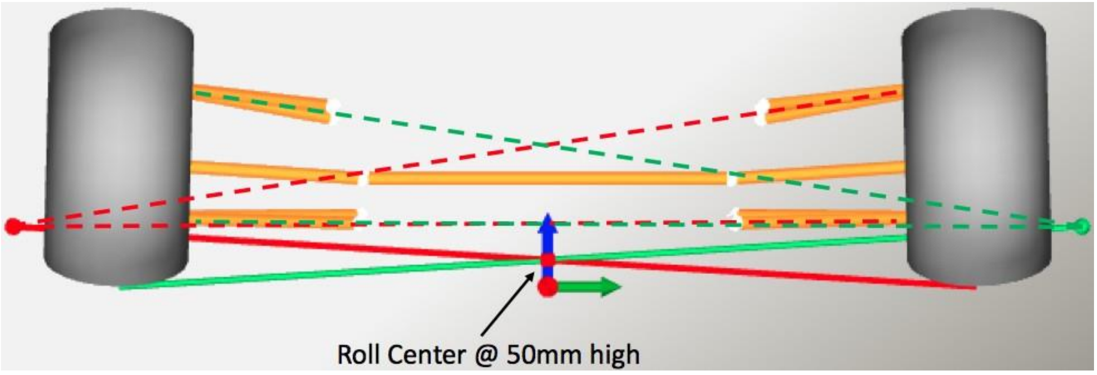


Fig 6. Roll Center - Double A Arm

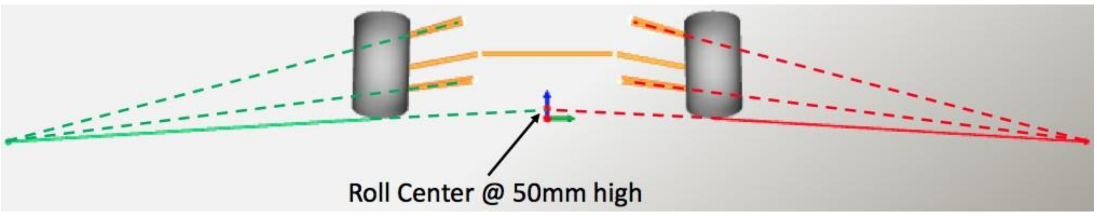
There are two ways to have the roll center at the same height:

i. Wheel instant center inboard of wheels

ii. Wheel instant center outboard of wheels



(a) Wheel instant center inboard of wheels



(b) Wheel instant center outboard of wheels

Fig 7. Same roll center height

Anti Dive: Inline Weight Transfer - Forces are applied at the front tire contact patch center if brakes are outboard (in the wheel/upright).

Anti-dive is a percentage and refers to the front diving under braking. It can be thought of as the counterpart for braking as jacking forces are to cornering.

To determine the percentage of front suspension braking anti-dive for outboard brakes, it is first necessary to determine the tangent of the angle between a line drawn, in side view, through the front tire patch and the front suspension instant center, and the horizontal. In addition, the percentage of braking effort at the front wheels must be known. Then, multiply the tangent by the front wheel braking effort percentage and divide by the ratio of the center of gravity height to the wheelbase. A value of 50% would mean that half of the weight transfer to the front wheels, during braking, is being transmitted through the front suspension linkage and half is being transmitted through the front suspension springs.

For inboard brakes, the same procedure is followed but using the wheel center instead of contact patch center.

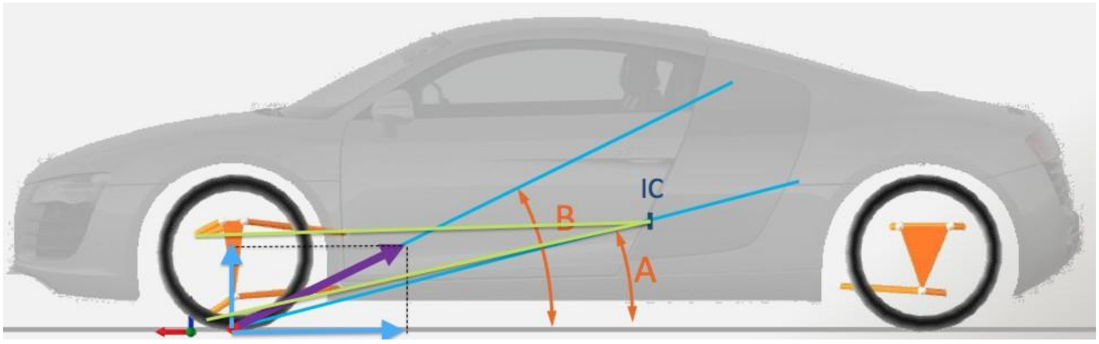


Fig 8. Anti Dive

$$\text{Anti-Dive (\%)} = \frac{\%Front\ Brake\ Force \times TanA}{TanB}$$

Where,

$$TanA = \frac{Side\ view\ front\ vertical\ VSAL}{Side\ view\ front\ longitudinal\ VSAL}$$

$$TanB = \frac{CG_{Height}}{Wheelbase}$$

(VSAL: Virtual Swing Arm Length)

Anti Squat: Anti-squat is a percentage and refers to the rear squatting under acceleration. It can be thought of as the counterpart for acceleration as jacking forces are to cornering.

Forward acceleration anti-squat is calculated in a similar manner and with the same relationship between percentage and weight transfer. Anti-squat values of 100% and more are commonly used in drag racing, but values of 50% or less are more common in cars which have to undergo severe braking. Higher values of anti-squat commonly cause wheel hop during braking. It is important to note that, while the value of 100%, in either case means that all of the weight transfer is being carried through the suspension linkage, this does not mean that the suspension is incapable of carrying additional loads (aerodynamic, cornering, etc.) during an episode of braking or forward acceleration. In other words, no "binding" of the suspension is to be implied.[4]

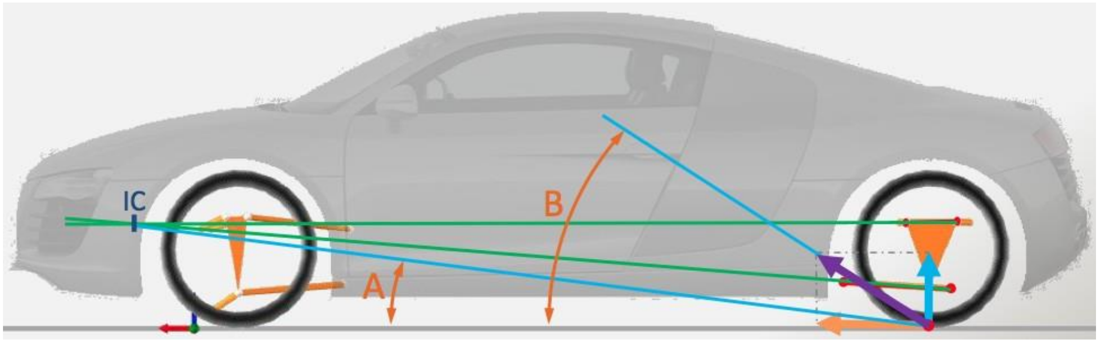


Fig 9. Anti Squat

$$AntiSquat\,[\%] = \frac{100 \times TanA}{TanB}$$

Tan A = Fn. (Instant suspension geometry, wheel radius)

Tan B = Fn. (Inline Weight Transfer)

Forces would be applied at the tire contact patch center in case of a solid axle.

3.4 Engine Specifications

The following are the specifications of the internal combustion engine that is to be used in the FSAE vehicle: Royal Enfield Thunderbird 500 Engine

Table1. Royal Enfield Thunderbird 500 Engine Specifications

PARAMETERS	DETAILS
Type	Single cylinder, 4-Stroke, Twin

	Spark, Air cooled
Displacement	499 cc
Bore x Stroke	84 mm x 90 mm
Compression Ratio	8.5 : 1
Max. Power	32 bHP @ 5350 rpm
Max. Torque	41.3 Nm @ 4300 rpm
Ignition	Digital Electronic Ignition
Clutch	Wet Plate Multiclutch
Gearbox	5 Speed Constant Mesh
Lubrication	Wet sump
Engine Oil	15W50 API SL Grade JASO MA
Fuel Supply	Keihin Electronic Fuel Injector
Air Cleaner	Paper Element
Engine Start	Kick/Electronic

Chapter 4

Design, Calculation and Analysis

Introduction

This chapter gives the idea of design and calculation of vehicle including kinematic suspension model, chassis design and analysis.

The chapter is divided into two sections viz. Kinematic suspension design & chassis design and analysis.

Section 4.1 - Chassis and Suspension Design

4.1.1. Kinematic Suspension Design

This is one of the most crucial steps in designing a race car. It involves the optimization of the vehicle suspension parameters like camber, caster, KPI etc.

The following are the calculation for the kinematic suspension model.

Kinematic Suspension Model Calculations:

$$\%AntiDive = FrontBrakeBias \times TanA \times \frac{Wheelbase}{CG} \times 100$$

a = $\tan^{-1}$ (0.07733)

a = 4.42°

$$\%AntiSquat = \frac{\tan\varphi_R}{l/h} \times 100$$

$\phi = \tan^{-1}$ (0.05)

ϕ = 2.86°

Front Suspension:

Table 2. Front Suspension

Parameters	Values
Static Camber	-1°
Camber gain in Jounce	-.35°/1 in
Camber Gain in Rebound	.34°/1 in
Caster	5°
KPI	3°
Scrub Radius	3 in
Toe In	-1°
Ground Clearance	2 in
Static Roll Center	2.531 in
Front Track Width	52 in

Rear Suspension:

Table 3. Rear Suspension

Parameters	Values
Static Camber	-1°
Camber gain in Jounce	-.58°/1 in
Camber Gain in Rebound	.61°/1 in
Caster	0°
KPI	3°
Scrub Radius	6.5 in
Toe In	0°
Ground Clearance	2 in
Static Roll Center	2.18 in
Rear Track Width	51 in

Anti Dive

As discussed in the previous chapter, the vehicle suspension has been designed to absorb the 40% of braking force acting on it. It is has been designed as follows:

Fig 10. Anti Dive - Final Design

Here the angle A is 4.76°. The caster angle is 5°. The wheel diameter has been depicted as 20 in (508 mm) and the inner diameter of the wheel is 13 in (330.2 mm).

Anti Squat

As discussed in the previous chapter, the vehicle suspension has been designed to absorb the 40% of acceleration force acting on it. It is has been designed as follows:

Fig 11. Anti Squat - Final Design

Here the angle A is 2.86° and the caster angle is 0°. As it is evident, the outside wheel diameter is smaller that the front wheel diameter, at 19.50 in (495.3 mm). However, the inside wheel diameter is the same as the front wheel inside diameter at 13.0 in (330.2 mm).

Front Suspension

As discussed in the previous chapter, the vehicle front suspension has been designed considering all the parameters mentioned there.

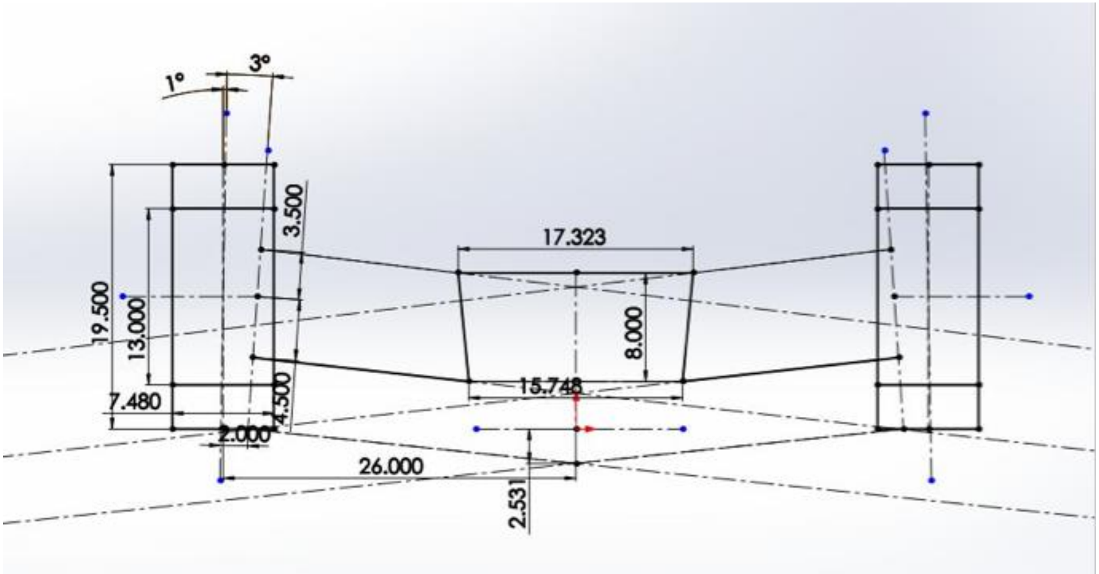


Fig 12. Front Suspension - Final Design

(kindly refer to table 2, page 20 for details)

The roll center for the front suspension is 2.531 inches above the ground. The camber is negative at -1° and the KPI angle is -3° from the camber.

Rear Suspension

As discussed in the previous chapter, the vehicle rear suspension has been designed considering all the parameters mentioned there.

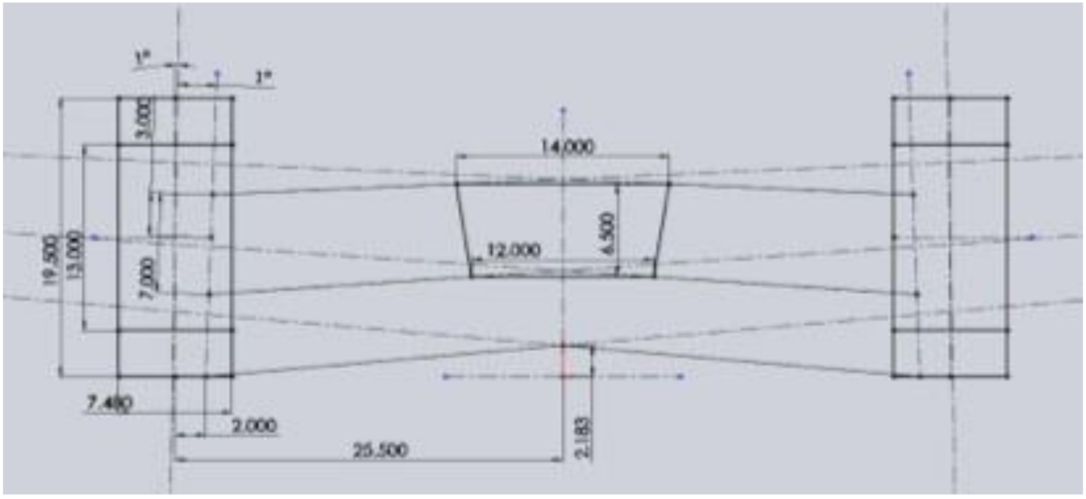


Fig 13. Rear Suspension - Final Design

(kindly refer to table 3, page 20 for details)

The roll center for the rear suspension is 2.183 inches above the ground. The camber is negative at -1° and the KPI angle is -1° from the camber.

4.1.2. Chassis Design

The chassis design has been shown below. It has been designed in the software SolidWorks by Dassault Systems.

The chassis conforms to (but not limited to conforming to the following guidelines. It conforms to all the guidelines in the FSAE rulebook 2013. The ones listed here are the most important) the following FSAE 2013 guidelines:

i. T3.4 Minimum material requirement guidelines
ii. T3.10 Main and front roll hoops - general guidelines.
iii. T3.10.4 95th Percentile Male seating guidelines
iv. T3.25 Side Impact structure for tube frame cars.

The following is the front view for the vehicle chassis:

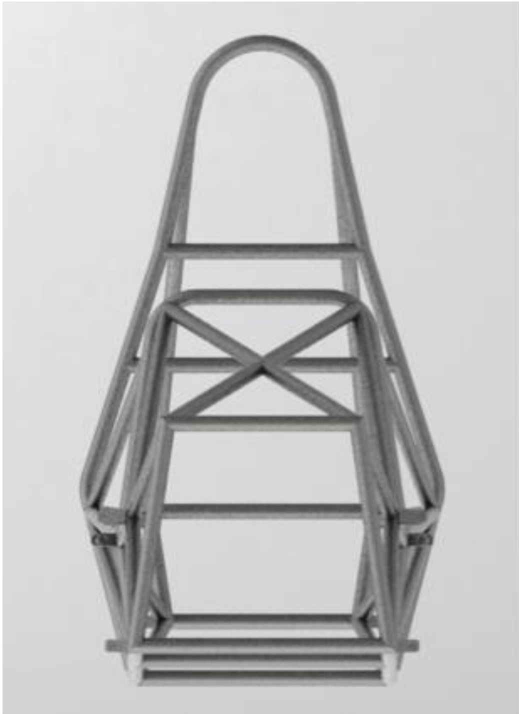


Fig. 14. Chassis Front View

The following is the side view for the vehicle chassis:

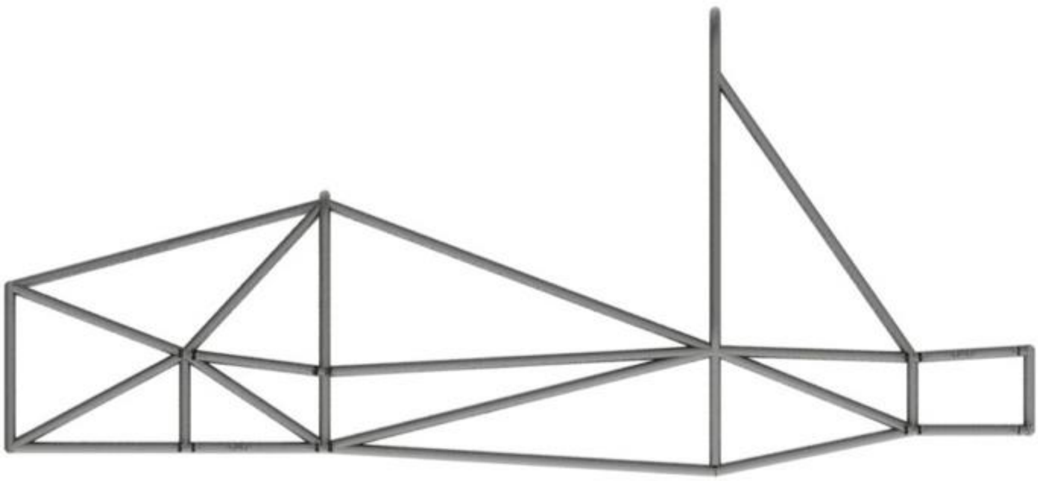


Fig. 15. Chassis Side View

The following is the top view for the vehicle chassis:

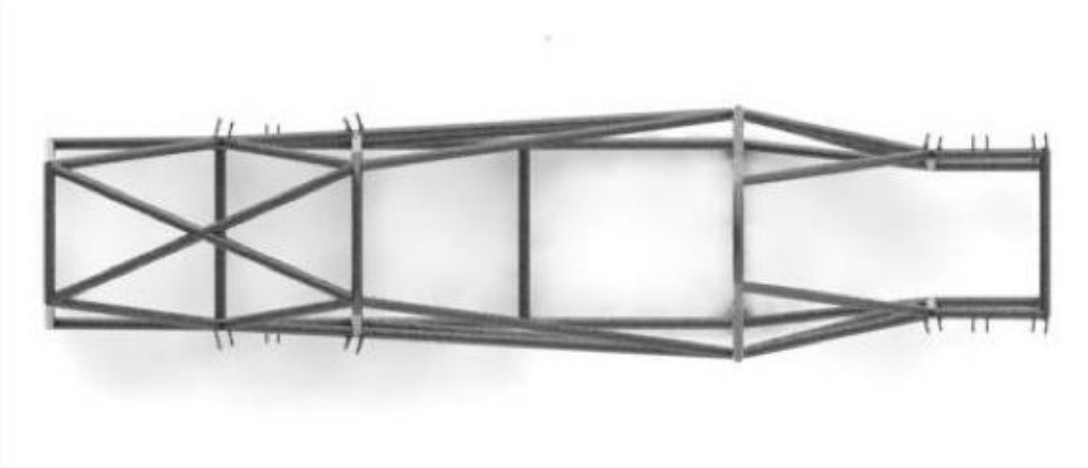


Fig. 16. Chassis Top View

$$\%AntiDive = FrontBrakeBias \times TanA \times \frac{Wheelbase}{CG} \times 100$$

This is the isometric view of the chassis:

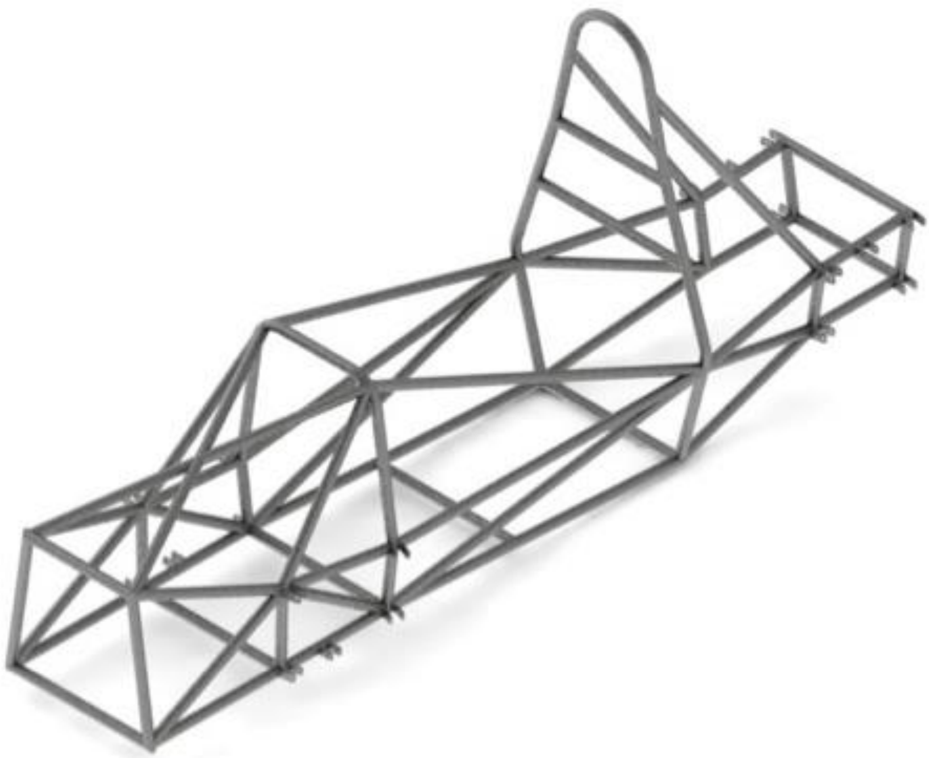


Fig. 17. Chassis Isometric View

As is evident from the figures, the chassis has been properly triangulated. The triangulation is node to node.

Section 4.2 - Frame Analysis and Validation

SolidWorks finite element analysis package was utilized to design and optimize the chassis. Many different chassis designs were analyzed using the front, side, rear impact tests and the roll over test, in order to minimize the amount of frame members used while still maintaining proper stiffness.

4.2.1. Front Impact Test

The following are the results for the front impact test performed in the SolidWorks finite element analysis package.

In frontal impact the two bodies are assumed to collide head on. It was calculated that the force of impact would be 18300 N or 18.3 kN. The maximum stress induced was calculated at 184 MPa.

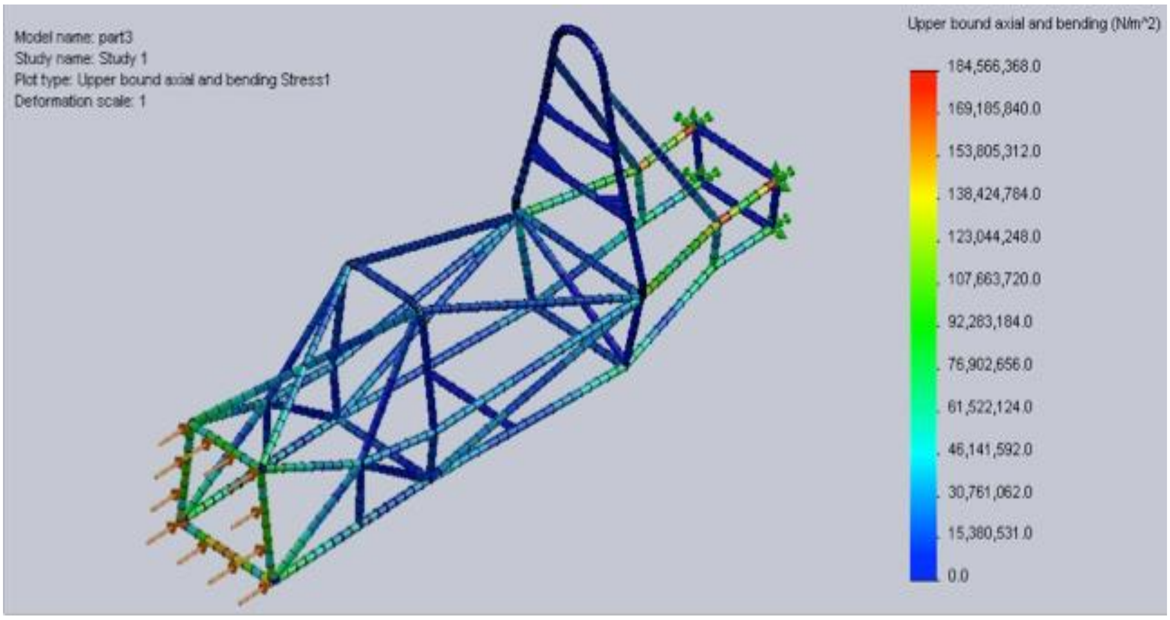


Fig 18. Stress analysis - Frontal Impact (Type Automatic)

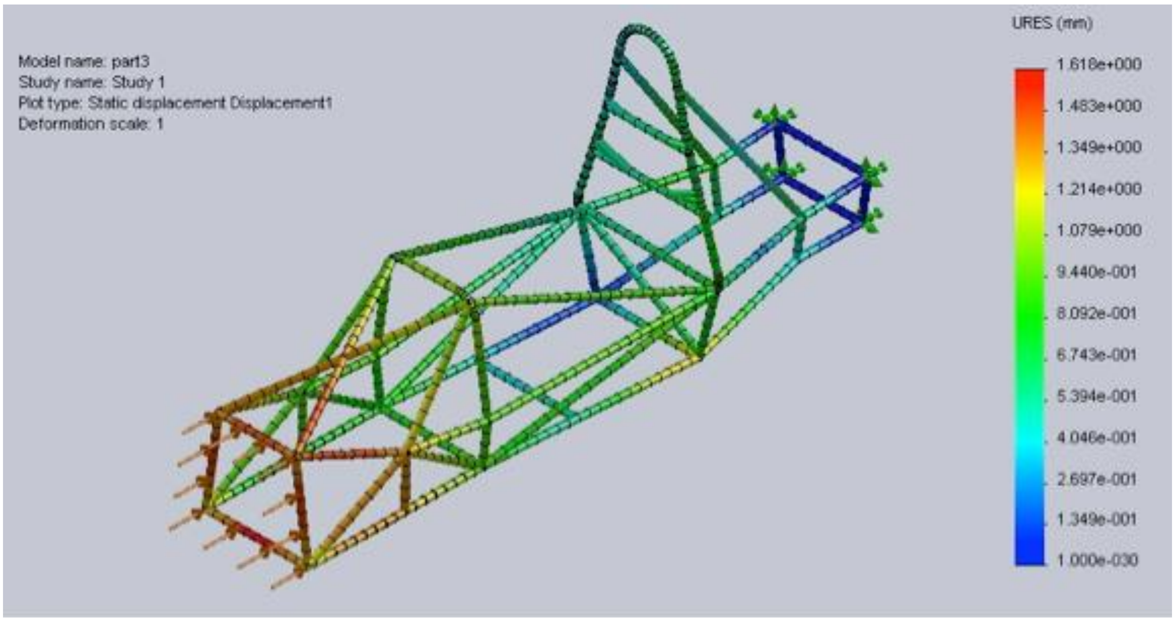


Fig 19. Displacement analysis- Frontal Impact

It is seen from the figure that the displacement produced in the chassis with an impact load of 18.3 kN is 1.618 mm. The maximum allowable displacement is 25 mm. It is therefore well within the requirement.

4.2.2. Rear Impact Test

The following are the results for the rear impact test performed in the SolidWorks finite element analysis package.

In this type of impact, it is assumed that the vehicle in the rear will collide with the the vehicle in the front, impacting its rear bulkhead. The force of the impact was calculated at 9150 N or 9.15 kN. The max stress induced was found out to be 816 MPa.

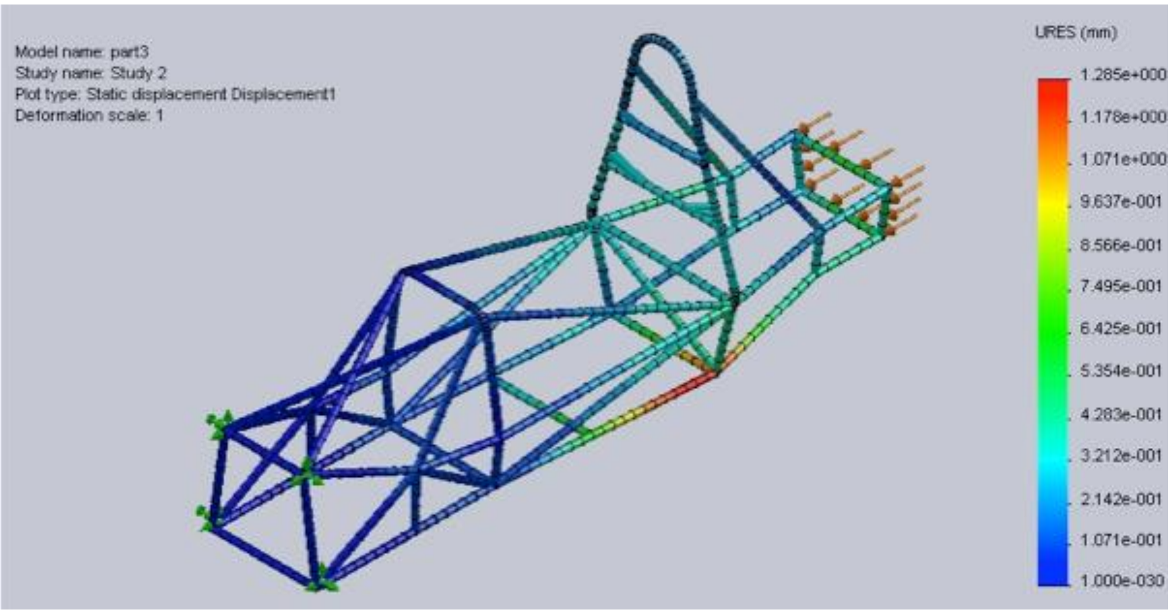


Fig 20. Rear Impact Displacement Analysis (Type Automatic)

According to the figure, the maximum displacement induced at a maximum impact load of 9.15 kN is 1.285 mm, which is the well within the maximum allowable limit of 25 mm.

4.2.3. Side Impact Test

The following are the results for the side impact test performed in the SolidWorks finite element analysis package.

In this test it is assumed that the colliding vehicle will collide on the side of the chassis, impacting the side structural members. The force of impact was calculated at 9150 N or 9.15 kN and the maximum stress induced was found out to be 816 MPa.

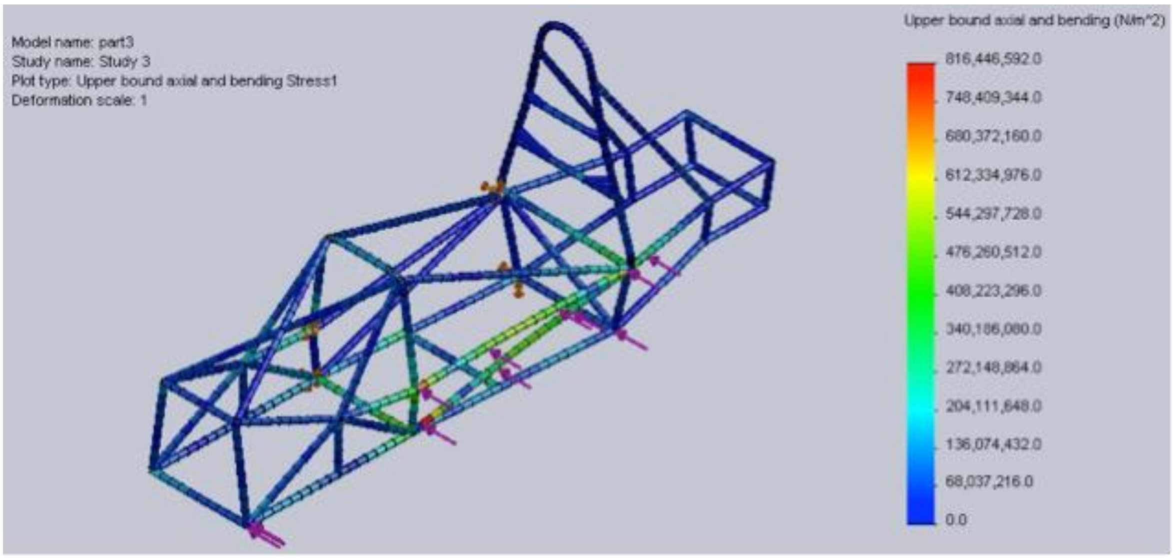


Fig 21. Side Impact Stress Analysis (Type Automatic)

The following is the displacement analysis:

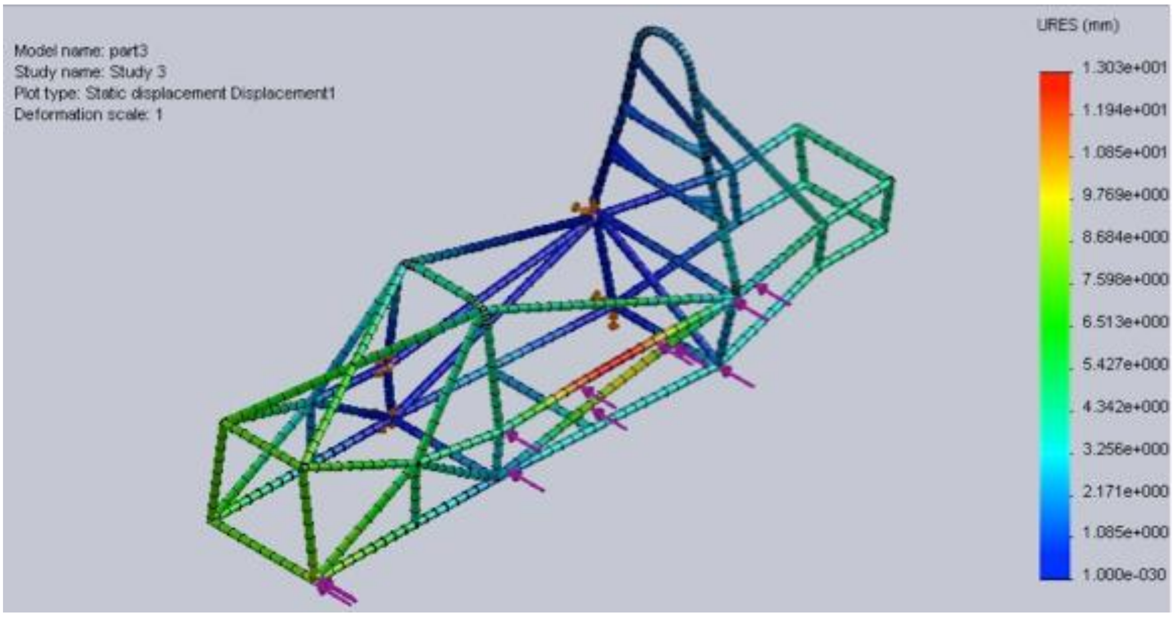


Fig 22. Side Impact Displacement Analysis (Type Automatic)

Keeping the force of impact at 9.15 kN it was found out that the displacement induced at maximum stress of 816 MPa is 1.3003 mm.

4.2.4. Roll Over Test

The following are the results for the front impact test performed in the SolidWorks finite element analysis package.

In the roll over test the force of impact was calculated according to the ‘3g’ rule (i.e., the force acting will be three times the force of gravity). The force calculated according to this rule on the front hoop is 5055.15 N or 5.055 kN and on the main hoop is 8829 N or 8.829 kN. According to this the maximum stress induced was found out to be 479 MPa.

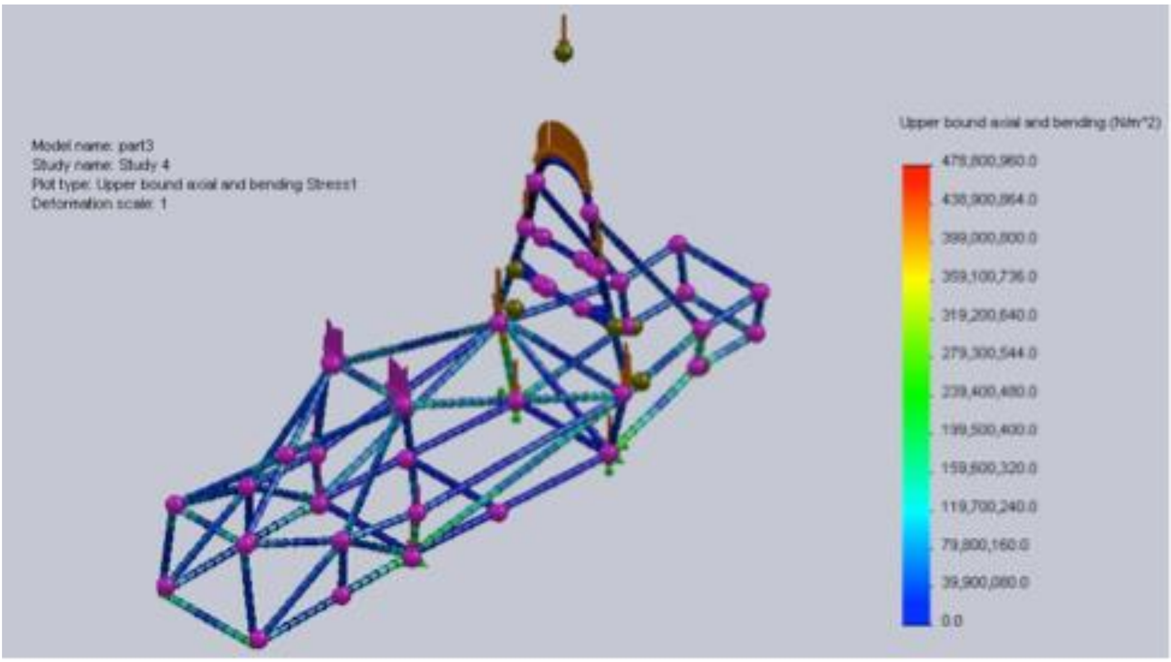


Fig 23. Roll Over Impact Stress Analysis (Type Automatic)

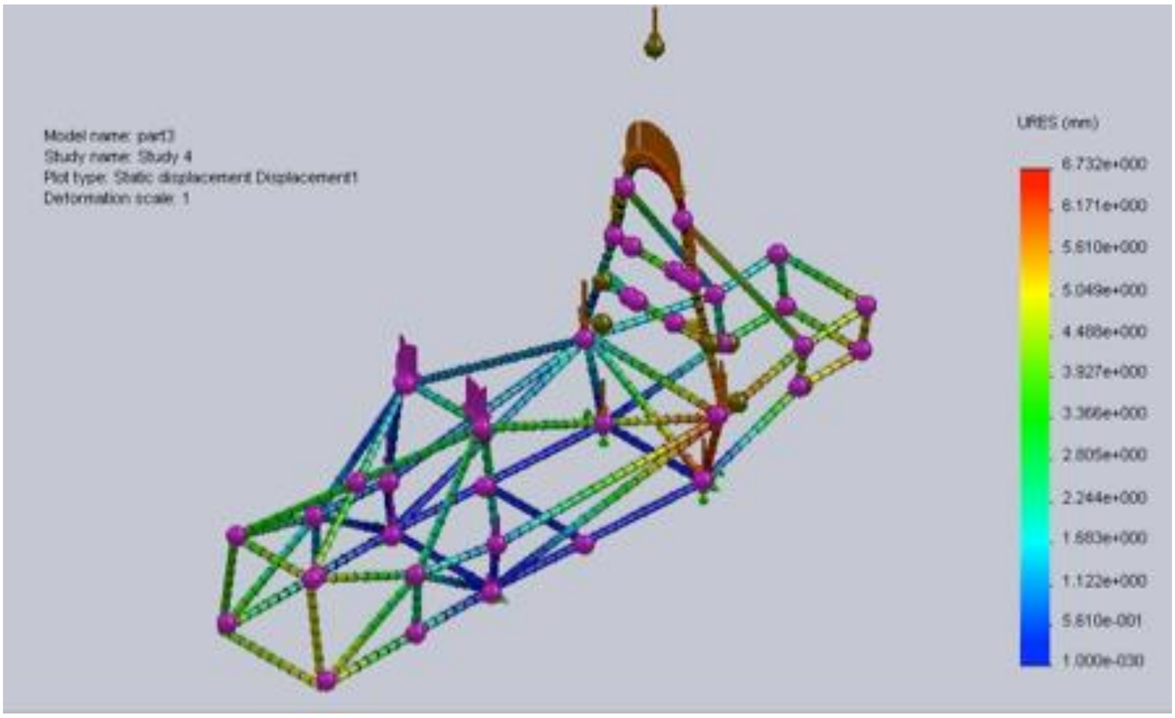


Fig 24. Roll Over Impact Displacement Analysis (Type Automatic)

On calculating the displacement, it was found out to be 6.732 mm. Again this is well within the maximum displacement limit of 25 mm.

4.2.5. Modal Analysis

In this analysis the natural frequency of the chassis is calculated without constraining any part of the chassis. The chassis is allowed to vibrate in any direction. According to this the natural frequencies of six degrees of freedom has been calculated. The result has been given in the following table.

Table 4. Modal Analysis Test Result

Frequency Number	Rad/sec	Hertz	Seconds
1	0.0027935	0.0004446	2249.2
2	0.0018198	0.00028963	3452.6
3	0.0010526	0.00016753	5969
4	0.0015956	0.00025394	3937.9
5	0.002225	0.00035412	2823.9
6	0.0036423	0.00057968	1725.1
7	255.88	40.725	0.024555
8	316.12	50.63	0.019751

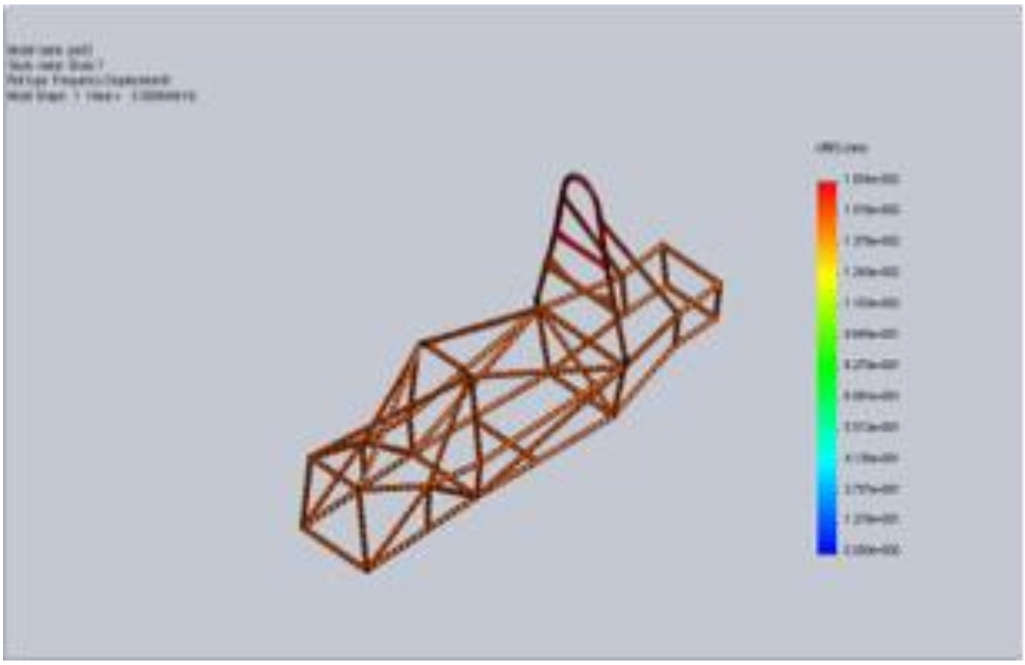


Fig 25. Modal Analysis Displacement (Mode Shape 1)

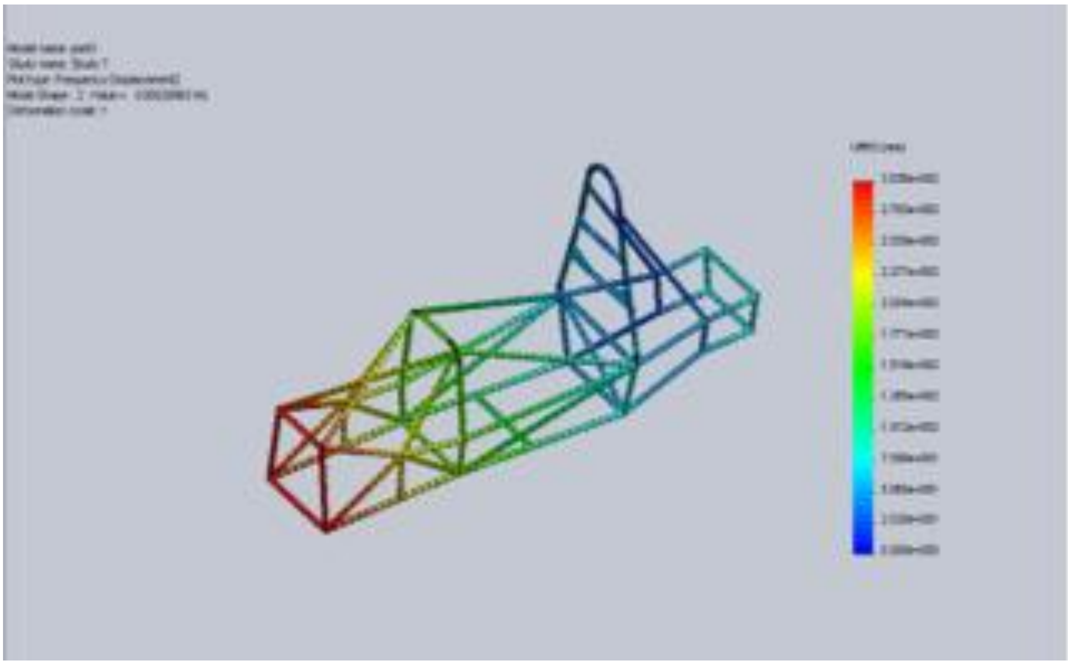


Fig 26. Modal Analysis Displacement (Mode Shape 2)

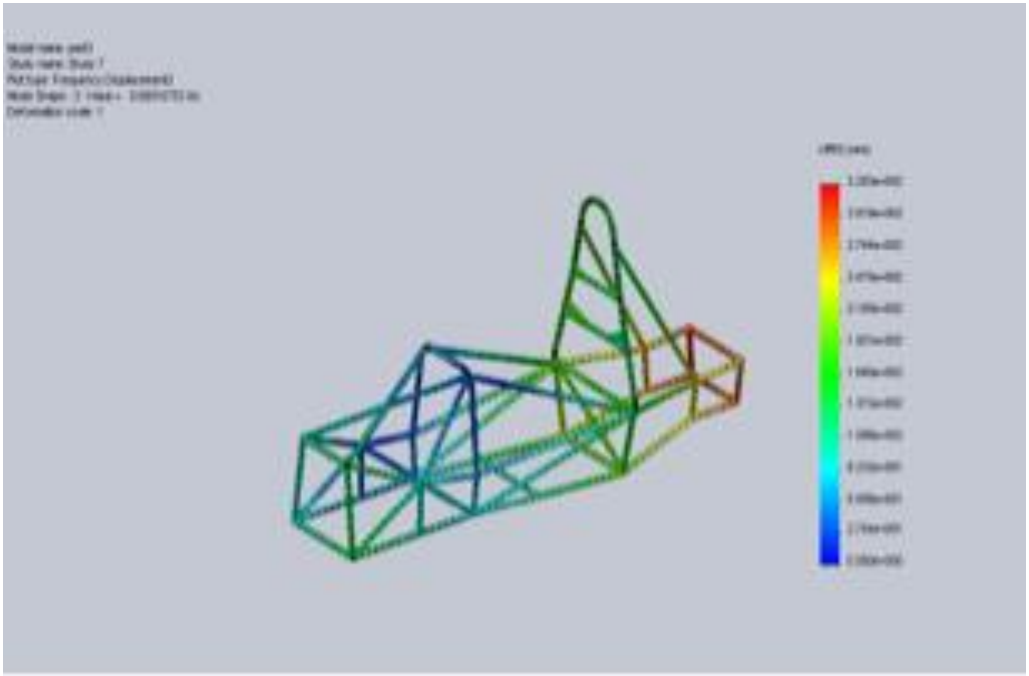


Fig 27. Modal Analysis Displacement (Mode Shape 3)

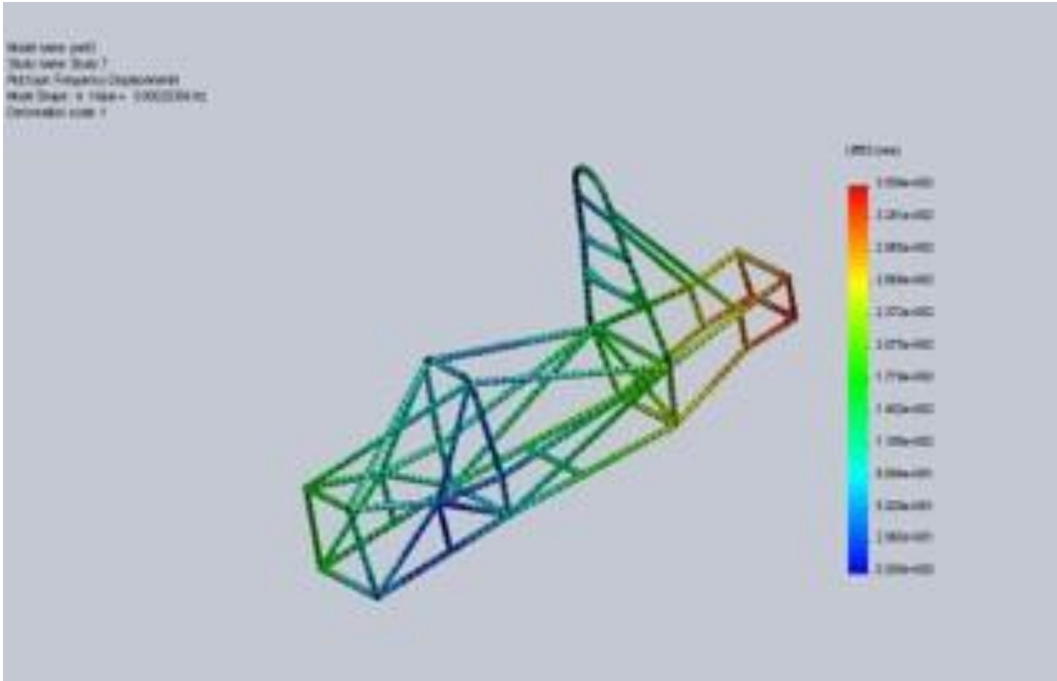


Fig 28. Modal Analysis Displacement (Mode Shape 4)

Section 4.3 Component Design and Validation

4.3.1. Rack and Pinion Steering System

Due to the lightweight and simplicity of rack and pinion, we will use custom made steering assemblies designed specifically for these types of race cars.

The specifications of the steering wheel to be used are:

No. of teeth on pinion = 9

No. of teeth on rack = 16

Rack length = 457.2 mm (18 in)

Rack Gain = 64.516 mm/turn (2.54 in/turn)

Steering ratio = 6.03

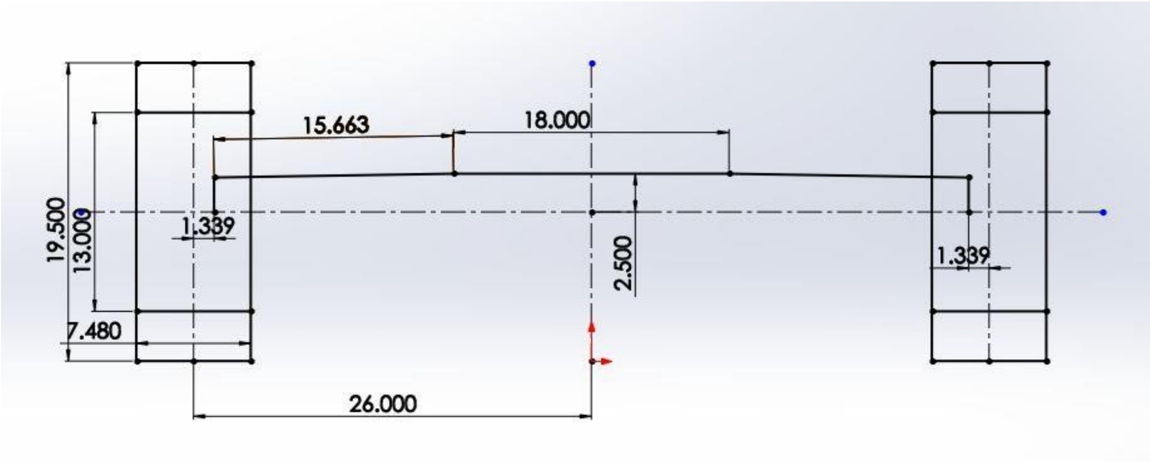


Fig 29. Steering Validation

4.3.2. Brakes

The vehicle has inboard brakes in the rear and disc brakes in the front on each wheel. The specification of the brakes are given below:

Table 5. Brake Specifications

Parameter	Front	Rear
Rotor, outer radii	114.3mm	118mm
Inner radii	79mm	83mm
Placement	Outboard	Inboard
Caliper	2	1
Bore Diameter of Piston	16mm	20mm
Friction radius	101.6mm	100.51mm

The load calculated on the brakes is as follows:

Under static condition:

Front axle – 135 Kgs
Rear axle – 165 Kgs

Under dynamic condition:
Front axle – 190.91 Kgs
Rear axle – 109.08 Kgs

For successful locking all four wheels on braking, the brake torque requirement is:
For front tires = 332.98 N-m
For rear tires = 190.276 N-m

The brake torque developed was calculated as:
Torque developed at front = 693.3 N-m
Torque developed at rear = 266.56 N-m

Since the braking torque developed well exceeds the required braking torque, this will ensure the successful locking of all four wheels.

4.3.3. Wheel Upright

An upright/ spindle is a part of the suspension system that carries the hub for the wheel and attaches to the upper and lower control arms. While braking, due to weight transfer, a torque is applied on spindle.

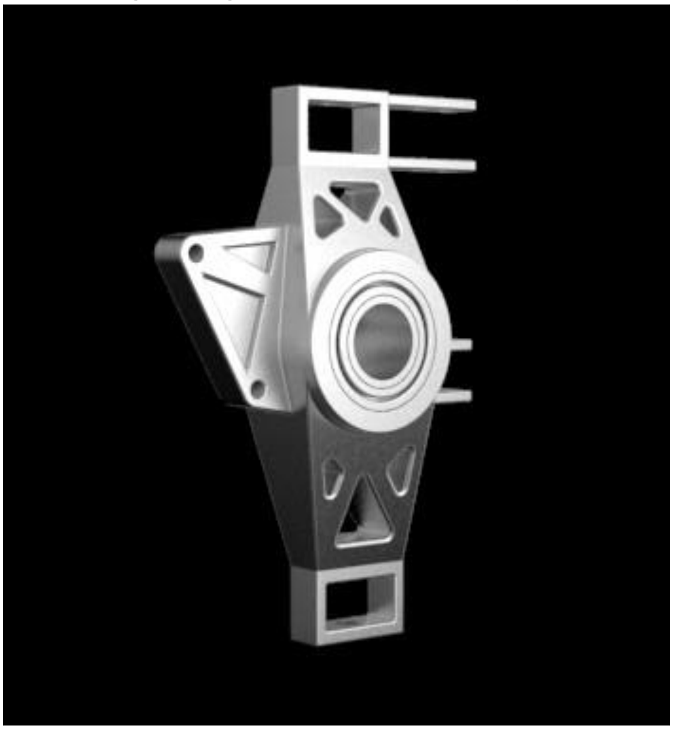


Fig 30. Wheel Upright

The stress analysis was conducted on the wheel upright. It was calculated that the maximum torque acting on the upright would be 693.3 N-m. According to this the maximum stress induced was found out to be 8.4 MPa.

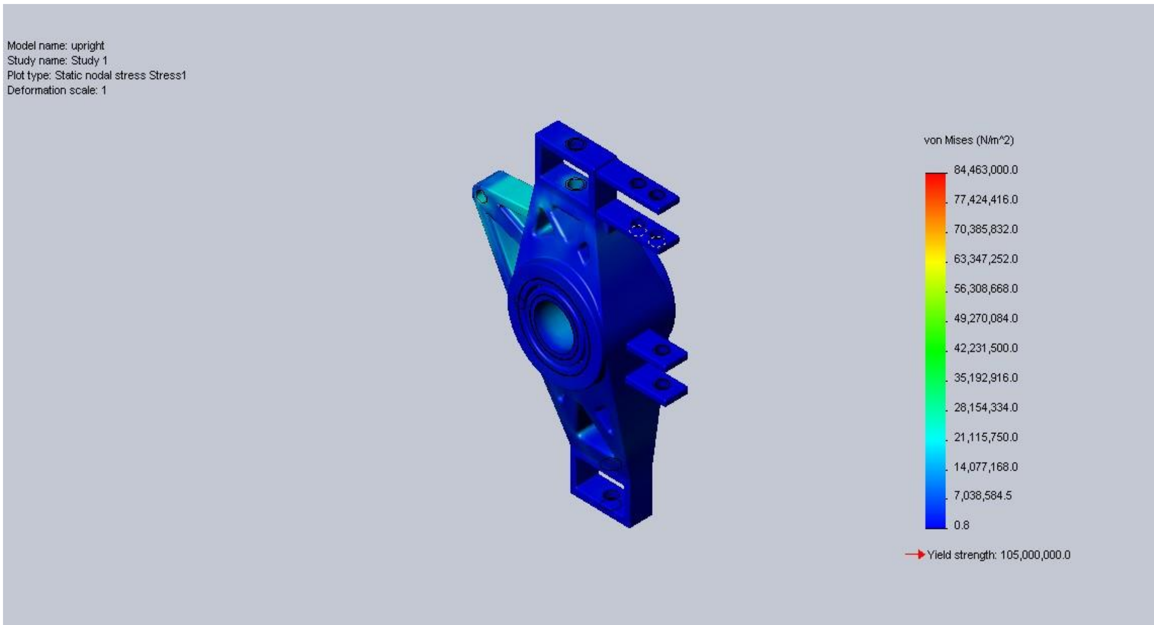


Fig 31. Wheel Upright Stress Analysis

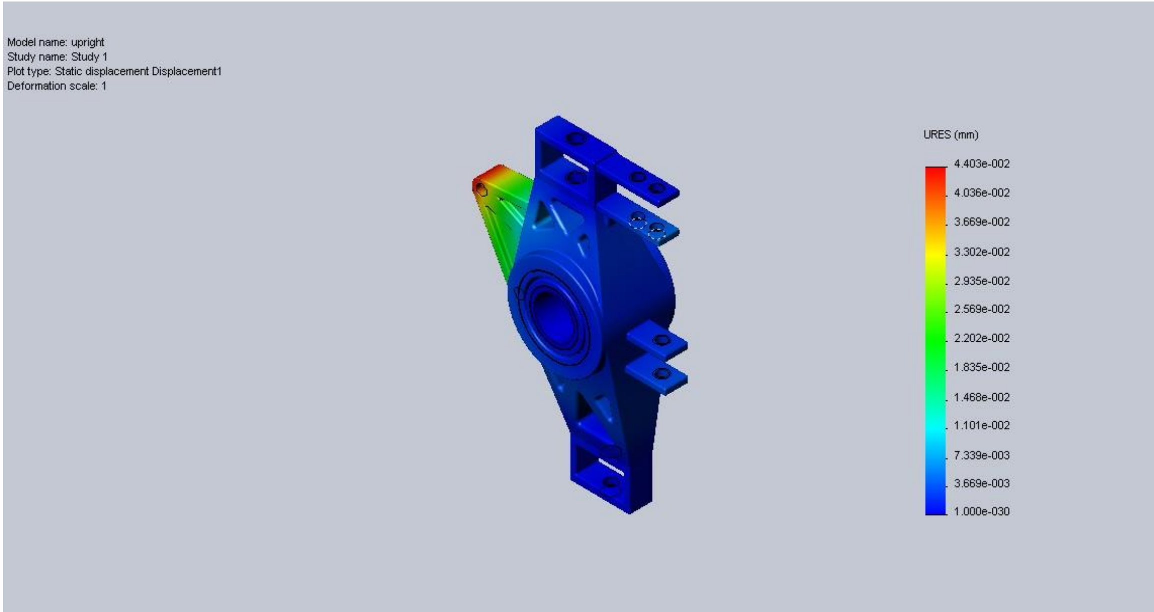


Fig 32. Wheel Upright Stress Analysis

Keeping the torque applied constant, it was found out that the maximum displacement of the upright was just 0.044 mm.

Chapter 5

Vehicle 3D Modeling and Detailed View

Vehicle 3D Modeling

The following images show the final outer body design of the vehicle after it has been fabricated. The design is subject to change according to the circumstances that may be encountered during the process of fabrication.

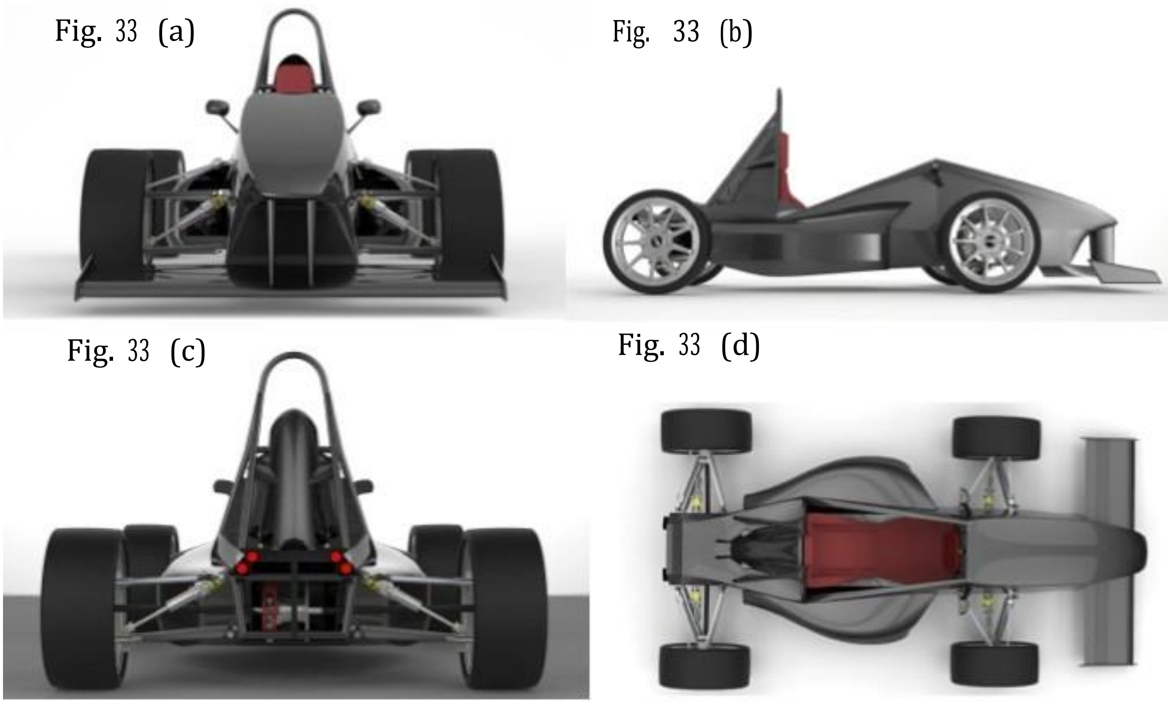


Fig 33. 3D Modeling View; (a) Front View; (b) Side View; (c) Rear View; (d) Top View

The isometric view of the vehicle after 3D modeling has been shown in the next page. The outer body is made of FRP, which is an abbreviation of Fiber Reinforced Plastic. This material has been chosen due to its light-weight, easy mouldability and cost.

Fig. 34. Isometric View

Detailed View

The detailed view is a particular view of the entire car in such a way that all the components are visible. All the components are well labeled facilitating the ease of recognition of the various components that have been mentioned in the text.

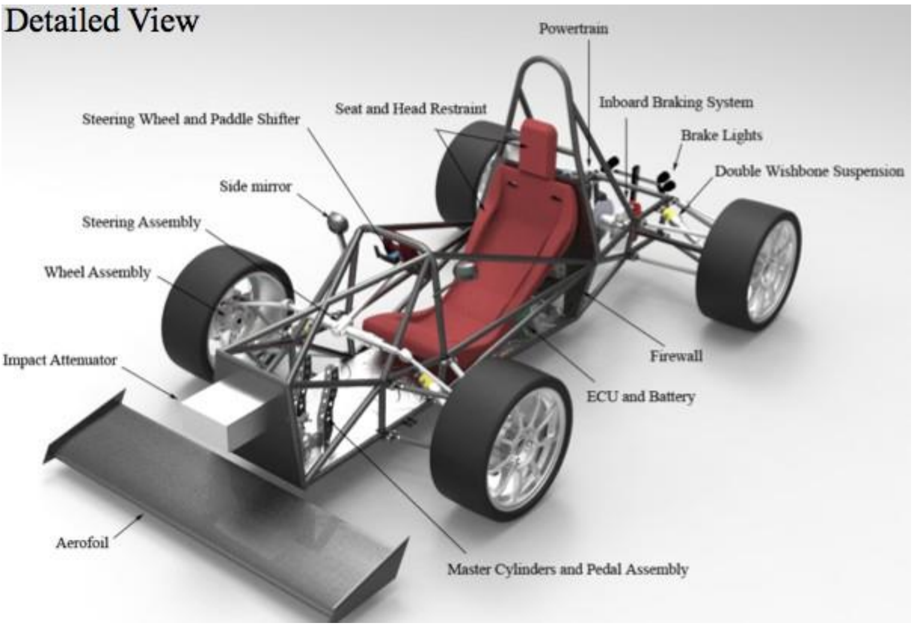


Fig. 35. Detailed View

Chapter 6

Estimation and Cost Analysis

This chapter deals with the tentative cost that may be incurred right from fabrication of the vehicle to cartage to the race venue. It should be understood that the costs are only tentative. This may change according to the market situation at the time of fabrication. These figures are conservative. The cost may increase during fabrication.

Table 6. Tentative Cost Analysis

Parts	Quantity	Total Cost (INR)
Engine	1	50,000.00
ECU	1	10,000.00
Engine Modification	-	50,000.00
Suspension Dampers	4	1,00,000.00
Steel Tubes	200 ft.	25,000.00
Steering system	1	20,000.00
Fuel Tank	1	2,500.00
Tires	4	40,000.00
Rims (Alloys)	4	20,000.00
Body Fiber Work	-	30,000.00
Pedal Assembly	1	22,000.00
Firewall	1	2,000.00
Battery	1	2,500.00
Attenuator	1	20,000.00
Aluminium block	20 Kgs	10,000.00
Rod ends	18	11,000.00
Brake assembly	3	15,000.00

Seat	1	10,000.00
5 – point Seat harness	1	10,000.00
Miscellaneous Expenses	-	1,50,000.00
Grand Total		**6,00,000.00**

Chapter 7
Conclusion

The vehicle has been designed according to the rules and parameters set forth by SAE International. The parameters assumed are optimized according to the requirements of this particular vehicle and also according to the undue stresses and force that it may be subjected to during the endurance.

The chassis has been designed and validated through frame analysis on the software SolidWorks by Dassault Systems.

This vehicle has passed all the dynamic and static tests performed on the chassis. In the front impact test the displacement of the chassis was 1.618 mm. The rear impact test saw a maximum displacement of 1.285 mm and the side impact test witnessed a maximum displacement of 1.3003 mm. The maximum allowed displacement in all the cases is 25 mm.

The roll over test saw maximum forces of 5.055 kN on the front hoop and 8.829 kN on the main hoop. The maximum stress induced due to these forces was 479 MPa. The forces were calculated according to the 3g rule.

The wheel upright being used was also designed in SolidWorks 2013. The displacement analysis revealed a maximum displacement of 0.044 mm whereas the maximum allowed is 25 mm.

As is evident, all the components used are well within the specified safety limits. This ensures that the vehicle will be completely safe and will ensure that the driver will escape unhurt in any eventuality.

References

[1], FSAE Rulebook 2012 (http://students.sae.org/competitions/formulaseries/rules/2012fsaerules.pdf), Page 21

[2], http://www.240edge.com/performance/tuning-camber.html

[3], http://en.wikipedia.org/wiki/Caster_angle

[4], Anti Dive and Anti Squat: http://en.wikipedia.org/wiki/Suspension_(vehicle)#Anti-dive_and_anti-squat

[5], http://students.sae.org/competitions/formulaseries/west/eventguide.pdf

[6], Cornell University; Presentation given by Seth Beckley, Kevin Gygrynuk, Josh Hilferty, Mike Teri

[7], Race Car Vehicle Dynamics, William F. Milliken & Douglas L. Milliken, ISBN 1-56091-526-9

[8], Tune to Win, Carrol Smith, ISBN 0-87938-071-3

Appendix B

Weight Distribution

Total weight of car – 300 Kgs

Weight of Driver – 80 Kgs

Parameters	Weight (in Kgs)
Powertrain	70
Rear Unsprung Mass	34
Front Unsprung Mass	37
Frame	34
Braking System	5
Steering System	5
Attenuator	3
Fuel Tank with Fuel	10
Fire Wall	1.5
Other Items contributing to the weight	17.5

Weight at the rear = powertrain + rear unsprung mass + fire wall + 50% of driver + fuel tank with fuel + 40% of frame weight + miscellaneous

= 70 + 34 + 1.5 + 40 +10 + 13.5 = 165 Kg

Weight at the front = total vehicle weight – weight at the rear

= 300 – 165 = 135 Kg

Weight distribution = Weight at front:Weight at rear

= 45:55

Appendix C
Calculation

C.1. Front Impact Calculation

$m_1 = m_2 = m = 300$ Kg

$u_1 = 30.55$ m/s

$u_2 = 30.55$ m/s

According to conservation of momentum

Total momentum before collision = Total momentum after collision

$$m_1 u_1 + m_2 u_2 = m_1 v_1 + m_2$$

$$v_2 \quad m(30.55 - 30.55) = m(v_1 +$$

$$v_2) \quad v_1 + v_2 = 0 \quad (1)$$

Assuming inelastic collision

Coefficient of restitution, e = 0

$$e = \frac{velocity \ of \ separation}{velocity \ of \ approach}$$

$$e = \frac{v_2 - v_1}{u_2 - u_1}$$

$$v_2 - v_1 = 0$$

$$v_1 = v_2 = v$$

From (1) eq. v=0

$$a = \frac{v_1 - u_1}{0.5}$$

Force of impact $= ma = 300 \times 61 = 18300$

C.2. Rear Impact Calculation

$m_1 = m_2 = m = 300$ Kg

$u_1 = 30.55$ m/s

$u_2 = 30.55$ m/s

According to conservation of momentum

Total momentum before collision = Total momentum after collision

$m_1 u_1 + m_2 u_2 = m_1 v_1 + m_2$

v_2 $m(30.55 - 30.55) = m(v_1 +$

$v_2)$ $v_1 + v_2 = 0$ (1)

Assuming inelastic collision

Coefficient of restitution, e = 0

$$e = \frac{velocity\ of\ separation}{velocity\ of\ approach}$$

$$e = \frac{v_2 - v_1}{u_2 - u_1}$$

$$v_2 - v_1 = 0$$

$$v_1 = v_2 = v$$

From equation (1) v=0

$$a = \frac{v_1 - u_1}{0.5}$$

Force of impact $= ma = 300 \times 61 = 18300$ N

C.3. Side Impact Calculation

$m_1 = m_2 = m = 300$ Kg

$u_1 = 0$ m/s

$u_2 = 30.55$ m/s

According to conservation of momentum

Total momentum before collision = Total momentum after collision

$$m_1 u_1 + m_2 u_2 = m_1 v_1 + m_2 v_2$$

$$m(0 - 30.55) = m_1 v_1 + m_2 v_2$$

$$v_1 + v_2 = 30.55 \quad (1)$$

Assuming inelastic collision

Coefficient of restitution, e = 0

$$e = \frac{velocity\ of\ separation}{velocity\ of\ approach}$$

$$e = \frac{v_2 - v_1}{30.55}$$

$$v_2 - v_1 = 0$$
$$v_1 = v_2 = v$$

From equation (1)

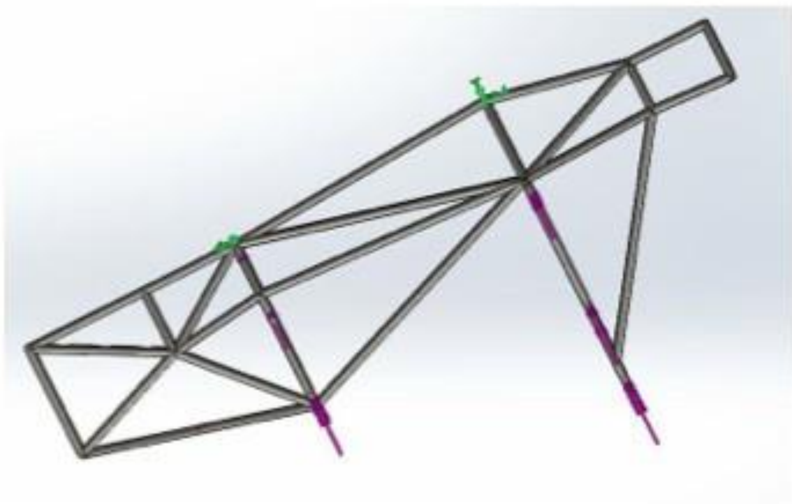
$$v_1 = v_2 = \frac{30.55}{2} = 15.275 \text{ m/s}$$

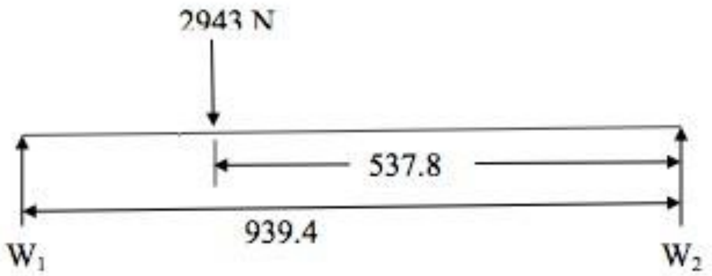
$$a = \frac{v_1 - u_1}{0.5}$$

Force of impact $= ma = 300 \times 30.55 = 9150$ N

C.4. Roll Over Calculation

Rollover impact





W_1 x 939.4 =2943 x 537.8

W_1 = 1685.05 N

W_2 = 2943 – 1685.05 = 1257.94 N

Due to first coming to contact of main hoop with the ground during rolling, complete body weight is acted on main hoop, so force on main hoop = 2943 N

As impact time is very less , so we have consider 3 g effect

Force on front hoop = 3 x W_1 = 5055.15 N

Force on main hoop = 3 x W = 8829 N

C.5. Brake Calculation

<u>Under dynamic load conditions</u>

Rear axle load = $(m.g.x_{cg} - m.a.y_{cg})/l$ = 109.08 Kg

Front axle load = mg – rear axle load = 190.91 Kg

Where m = mass of vehicle

g = acc. due to gravity

x_{cg} = Distance of CG from front Axle

a = deceleration during braking

y_{cg} = CG Height

l = wheelbase

<u>Brake torque requirement</u>

Torque = $\mu.N.R$

where R = rolling radius

tyre specification: 20" x 7"/ R13

rolling radius = 254mm

For front tires = 332.98 N-m

For rear tires = 190.276 N-m

<u>Braking Torque developed</u>

Line pressure developed

Force applied by driver on pedal, F_{pedal} = 200N

Pedal Ratio, PR = 6:1

Force on master cylinder input shaft = F_{pedal} x PR = 1200N

Master cylinder piston diameter

$D_{mc\ f}$ = 0.625 in = 15.875mm

$D_{mc\ r}$ = 0.75 in = 19.05mm

Pressure developed at front,

$P_f = F_{mc}/ (3.14 \times (D_{mc}\ f/2)^2 = 6062.66 \times 10^3$ N/m^2

Similarly at rear,

$P_r = 4210.187 \times 10^3$ N/m^2

Brake torque developed at front

Radius of rotor = 101.6 mm

$\mu_{pad} = .35$

Caliper piston dia.= 32mm

No. of piston per caliper = 2

No. of calipers = 2

A_{cp} = Total caliper piston area = $3.14 \times (D_{cp}/2)^2 \times Np \times Nc$

$= 3.216 \times 10^{-3}$ m^2

$F_{clamp\ f} = P_f \times A_{cp} = 19497.5$ N

Brake torque developed at rear

Radius of rotor = 100.5 mm

$\mu_{pad} = .35$

Caliper piston dia.= 48mm

No. of piston per caliper = 1

No. of calipers = 1

A_{cp} = Total caliper piston area = $3.14 \times (D_{cp}/2)^2 \times Np \times Nc$

$= 1.809 \times 10^{-3}$ m^2

$F_{clamp\ r} = P_r \times A_{cp} = 9616.215$ N

Torque developed at front = .35 x 19497.5 x .100 = 693.3 N-m

Torque developed at rear = .35 x 9616.215 x .100 = 266.56 N-m

Torque developed by the brakes is more than the required therefore this results in locking of all wheels.

Appendix D
FSAE Rule Book

FORMULA SAE SAE International

2010 FORMULA SAE RULES
PART B – TECHNICAL REQUIREMENTS

ARTICLE 1: VEHICLE REQUIREMENTS & RESTRICTIONS

B1.1 Technical Inspection

The following requirements and restrictions will be enforced through technical inspection. Noncompliance must be corrected and the car re-inspected before the car is allowed to operate under power.

B1.2 Modifications and Repairs

B1.2.1 Once the vehicle has been presented for judging in the Cost or Design Events, or submitted for Technical Inspection, and until the vehicle is approved to compete in the dynamic events, i.e. all the inspection stickers are awarded, the only modifications permitted to the vehicle are those directed by the Inspector(s) and noted on the Inspection Form.

B1.2.2 Once the vehicle is approved to compete in the dynamic events, the ONLY modifications permitted to the vehicle are those listed below. They are also referenced in Part C of the Formula SAE Rules – Static Event Regulations.

a. Adjustment of belts, chains and clutches
b. Adjustment of brake bias
c. Adjustment of the driver restraint system, head restraint, seat and pedal assembly
d. Substitution of the head restraint or seat insert for different drivers
e. Adjustment to engine operating parameters, e.g. fuel mixture and ignition timing
f. Adjustment of mirrors
g. Adjustment of the suspension where no part substitution is required, (except that springs, sway bars and shims may be changed)
h. Adjustment of tire pressure
i. Adjustment of wing angle, but not the location
j. Replenishment of fluids
k. Replacement of worn tires or brake pads
l. The changing of wheels and tires for "wet" or "damp" conditions as allowed in Part D of the FSAE Rules – Dynamic Event Regulations.

B1.2.3 The vehicle must maintain all required specifications, e.g. ride height, suspension travel, braking capacity, sound level and wing location throughout the competition.

B1.2.4 Once the vehicle is approved for competition, any damage to the vehicle that requires repair, e.g. crash damage, electrical or mechanical damage will void the Inspection Approval. Upon the completion of the repair and before re-entering into any dynamic competition, the vehicle MUST be re-submitted to Technical Inspection for re-approval.

ARTICLE 2: GENERAL DESIGN REQUIREMENTS

B2.1 Vehicle Configuration

The vehicle must be open-wheeled and open-cockpit (a formula style body) with four (4) wheels that are not in a straight line.