ISSN : 2278 - 0181

THESIS

Thesis ID : IJERTTH0016

# Design and Development of A Formula Style Race-Car Prototype

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KIIT Deemed to be University
School of Mechanical Engineering

Published By

International Journal of Engineering Research and Technology
(www.ijert.org)

ISSN: 2278-0181

# DESIGN AND DEVELOPMENT OF A FORMULA STYLE RACE-CAR PROTOTYPE

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International Journal of Engineering Research & Technology (IJERT)
ISSN: 2278-0181

A project report on

# DESIGN AND DEVELOPMENT OF A FORMULA STYLE RACE CAR PROTOTYPE

submitted in complete fulfillment of the requirements for the degree of

Bachelors of Technology

In

Mechanical and Automobile Engineering

By

**ROHAN RAI**
**SHAYAK CHOUDHARY**
**SHARAT ANAND**
**G SASHANK**

# ACKNOWLEDGEMENT

The group members would like to extend their heartiest thanks to all those who were associated with us. We express our immense gratitude to Dr. Isham Panigrahi, Dy. Director, Student Research Wing, KIIT deemed to be University, for being the Torch Bearer of our path of success and giving his constant support and immense motivation all throughout the completion of the project.

We are sincerely thankful to our faculties and staffs of the Central Workshop of School of Mechanical Engineering, KIIT University, for providing us with sufficient space in the department's workshop and allowing us to use the necessary machines for machining our spares.

It was also our privilege to be associated with our astounding and renowned sponsorship partners, without whom the completion of the project would had not been possible.

Lastly, we would like to acknowledge the constant motivation of our parents and friends towards the completion of this project report.

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# ABSTRACT

The objective of this project is to design and develop a versatile, safe, durable, and high-performance, Formula type car for amateur weekend enthusiasts. This Thesis report focuses on explaining the engineering and designing processes implemented behind all the subsystems in the race-car. The design of vehicle and components is within in accordance with the rules laid down by Formula Student Germany and SAE International.

The vehicle manufactured is a Combustion type Formula Student car of Team Hermes Racing which comes under the esteemed banner of SAEINDIA-KIIT. It has been designed to perform effortlessly in all the dynamic conditions and consists of the capability of being tuned manually to enhance the vehicle's performance according to the requirements of a specific dynamic event.

The design process of the vehicle is iterative and is based on various engineering and reverse engineering processes depending on the availability of materials, cost optimization and proper resource planning. So, the design process focuses on: Weight Reduction, Part Optimization, Data Validation, Safety, Serviceability, Cost Efficiency, Standardization, Strength, Ruggedness, Ergonomics and Aesthetics.

We followed the procedure of benchmarking our design against past winning vehicles and then modified it, keeping in view our constraints in processing, materials, and other manufacturing facilities. We kept the design simple and easy for mass production. Also, advance manufacturing technologies such Rapid Prototyping, CNC Machining, Waterjet Cutting, Laser Cutting and Gas Welding were used to manufacture the parts of the vehicle.

The project has bagged continuous Top-4 National Ranks under the category of Cost and Manufacturing Event, at Formula Bharat 2018 and SUPRA SAE India 2018, respectively. Also, it has been rewarded with the Best Student Project, under the category of Innovation, Cost Optimization and Weight Reduction, in Business Conclave 2018 organized by the Aditya Birla Group.

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# LIST OF SYMBOLS / ABBREVIATIONS

| Symbol / abbreviations | Description |
|---|---|
| *IA* | Impact Attenuator |
| *SES* | Structural Equivalency Sheet |
| *EDR* | Engineering Design Report |
| *FB* | Formula Bharat |
| *SAE* | Society Of Automotive Engineers |
| *FV* | Front View |
| *TV* | Top View |
| *SV* | Side View |

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# CHAPTER 1: INTRODUCTION

## 1.1 FORMULA STUDENT EVENT

Formula SAE is the world's largest student engineering design competition whereby university students design, manufacture, test and compete with a single seat racecar. With its origins dating back to 1978 as a humble SAE Mini-Indy competition at the University of Houston, this highly prestigious engineering competition has seen the last 34 years filled with innovative and award-winning designs from universities and colleges across the globe. Now, the competition encompasses 630 registered teams worldwide. The hard work and dedication shown by each team comes to a climax through a number of international competitions based in different regions around the world, which are run every year.

The FSAE competition offers a challenging environment where engineering students practice and develop various engineering skills. Placing well in the competition occurs as a result of two categories; static testing, where design and engineering practice are judged and scored accordingly, and dynamic testing, where the actual performance of the vehicle is judged and scored. Teams that place well attain global recognition for their respective university from both the automotive industry and various even sponsors. Thus, those universities that place well attain an increase in revenues from sponsors/donations and increased relationships with these companies, which lead to an acceleration of the university's programs and an increase in job placement. Thus, the FSAE competition offers a multitude of benefits for both the students that attend and the university as a whole.
In order to place well, as previously mentioned, the vehicle must be designed with good engineering practice and must perform well. Since the event is essentially an autocross event, which favors cornering over top speed, this means having a chassis that is as light a possible while still maintaining required torsional rigidity, and a suspension system that can maximize the performance of the tires in contact with the road.

It is through these competitions, such as SUPRA SAEINDIA Student Formula, that each team may see just how well their cars stack up against their peers. Each car is judged in a number of events, which are separated into two types – static and dynamic. Static events consist of Design, Cost, as well a Business Presentation. These static events allow teams to showcase their financial acumen, marketing strategies and overall design philosophies.

The dynamic events consist of Acceleration, Skid-pad, Autocross, Fuel Economy and Endurance. In these events, teams may directly compare their designs on track, where each team has the opportunity to demonstrate the capabilities of their vehicles.

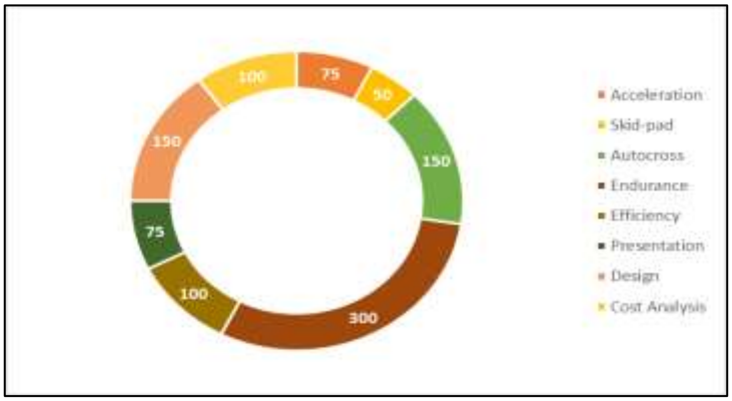


**Figure 1.1: Competition Score Breakdown**

## 1.2 ABOUT THE TEAM

Team Hermes Racing, started back in 2014, is the official formula student team of KIIT-Deemed to be University. For the upcoming season of 2019-20 the team will be developing Formula Student a new age combustion formula student race-car for the University under the aegis of School of Mechanical Engineering.

## 1.3 PROJECT SYNOPSIS

In this report, the overall design process for a Formula SAE vehicle will be explored, as well as the many challenges that must be overcome. This report will be fragmented into different sections: Vehicle Dynamics, Chassis and Ergonomics, Powertrain, Brakes and Safety, Aerodynamics and Data Acquisition. This project will be split into several phases: manufacturing, testing and validation of the 2019 FSAE vehicle and the design, analysis, manufacturing, and finally the test and validation of the 2019 FSAE chassis and suspension. All decisions for design were based on all pros and cons from previous Hermes Racing testing and competition results.

The late 2019 vehicle will be tested to acquire data in order to establish which of the design assumptions and past decisions were valid and which were not. It is possible that after design and testing, the components do not perform as expected. The interaction between the various components on the vehicle could be of greater importance than originally estimated.

During the second phase, design of the vehicle will start with a free body diagram, analyzing all of the forces and moments that the vehicle will be exposed to. It will be shown that race cars are practically built from the tires up, due to the fact that maximizing vehicle performance is directly correlated to maximizing tire performance. From the tire analysis, the suspension pick-up points are determined, and the chassis is built around it.

In the analysis phase, various programs will be used to determine the stresses and the fatiguing of the chassis and suspension. *This will prove that reliability isn't as much of a concern to race cars as it is to production vehicles, and that the vehicles are designed more for material stiffness than strength*. Manufacturing will be done in house, with greater focus on preparation and machining quality than before.

## 1.4 BACKGROUND (PRODUCT SURVEY)

### 1.4.1 PROBLEM STATEMENT

**Design Goals**

The major goals in designing this year's vehicle were as follows:

- Maintain Reliability
- Centralized and lower center of gravity race car
- Improve performance of the power train
- Design for Manufacturing and Assembly
- Better Packaging
- Improve Drivability

**Design Flowchart**

This section will explain the evolution of **HR19** following different phases of design and manufacturing. The vehicle was designed on SolidWorks Student Edition 2018. To validate all the components, FEA for all the components were done on **ANSYS 16.0** and **HYPERWORKS Student Version 17.0**. The hard points for the suspension were designed on **LOTUS SHARK** and better in-depth analysis was done in **DYNATUNE**.


**Figure 1.2: Design Flowchart**

# CHAPTER 2: ERGONOMICS

## 2.1 DESIGN CONSTRAINT

A major rule of the event that constraints the size optimization of the cockpit is to fit the 5 th percentile female up to the 95th percentile male within the vehicle for operation. In addition to this, other rules that constraints the ergonomics design are:

- the cockpit opening to be sufficient enough for the "Cockpit Opening Template" to pass the topmost side impact structure (in the case of space frame) or 320 mm above the lowest chassis member (in the case of alternate frame). Seat, Seat cover, steering wheel etc. may be removed to accommodate the template as mentioned in the rulebook. As stated in rule T3.1 Of Formula Student Germany rulebook.
- For the foot-well template, should cross the leg space without hindrance until 100 mm from the rearmost pedal face.

## 2.2 ERGONOMICS

Ergonomics is the scientific and analytical discipline concerned with the design or arranging workspaces, systems and products, understanding of interactions among humans and other elements of a system so that it fits the humans who use them. It is the profession that applies theory, principles, analytics, data and methodology to design in order to optimize human well-being with respect to interaction with other systems and overall performance.

## 2.3 OBJECTIVE

Problems related to the ergonomics determination were identified. These were: determination of seat inclination, pedal box position, dashboard controls and shifter location in consideration with team's drivers and SAE's body dimension for the 95th percentile male. The conclusion was agreed to be made based on a jigs setup that acts as a mock-up chassis.

### 2.4 DRIVERS' FEEDBACK

A generalized feedback form was given to each driver for its input in the design. Following questionnaire was developed: -

1. Rate the significance of the placement of vehicle controls; state any potential fatigue the driver might experience from the placement.
2. Rate the significance of a seat that can adjustable: can be shifted backward and forwards as desired in the cockpit to adjust to the different heights as per the different drivers on the team.
3. Rate the burden of fatigue of the driver's arms. Suggestions to reduce/remove it.
4. Rate the burden of fatigue to the driver's legs. Suggestions to reduce/remove it.
5. Rate the burden of fatigue to the driver's back. Suggestions to reduce/remove it.

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6. What is the driver’s seating position? (Upright seating posture with legs angled down or reclined back with the legs at an elevated angle? Sample pictures below in fig. 6(a) and 6(b)
7. Describe the steering wheel angle of approach to the driver. Is it parallel the driver’s chest or does it approach at another angle?
8. What will be the position of the driver’s arms while driving? Describe.
9. What will be the position of the driver’s legs while driving? Describe
10. Where is the position of the transmission shifter within the cockpit? Describe

The answers from this form were recorded and the following answers were observed:

**Table 2.1: Questionnaire for the Drivers**

| | |
|---|---|
| **Rate the significance of the placement of vehicle controls; state any potential fatigue the driver might experience from the placement.** | **2.94** |
| **Rate the significance of a seat that can adjustable: can be shifted backward and forwards as desired in the cockpit to adjust to the different heights as per the different drivers on the team.** | 0.76 |
| **Rate the burden of fatigue of the driver’s arms. Suggestions to reduce/remove it** | 3.44<br>Major feedback concluded to maintaining proper arm angle |
| **Rate the burden of fatigue to the driver’s legs. Suggestions to reduce/remove it** | 3.27<br>Keep the legs straight as much as possible |
| **Rate the burden of fatigue to the driver’s back. Suggestions to reduce/remove it** | 4.27<br>Problem experienced during endurance events |
| **What is the driver’s seating position? (Reclined back with the legs at an elevated angle or upright seating posture with legs angled down?) Sample pictures below in figs. 6(a) and 6(b).** | 67.27% for reclined<br>32.77 % for upright |
| **Describe the steering wheel angle of approach to the driver. Is it parallel the driver’s chest or does it approach at another angle?** | Should be at an angle in a way that the steering is raised towards the driver |

| | |
|---|---|
| **What will be the position of the driver's arms while driving? Describe** | The upper arm making an acute angle with the chest plane and an obtuse angle at the elbows |
| **What will be the position of the driver's legs while driving? Describe** | A minimum knee bend is required such that the ankle acts as a pivot point for the proper actuation |

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| | |
|---|---|
| **Where is the position of the gear shifter within the cockpit Describe** | Lever shifter is preferable and its placement on the left is helpful |

## 2.5 DESIGNING OF CHASSIS

A CAD model was drafted for the jigs setup before the start of the actual installation. The design of the jigs for ergonomics was so conceptualized as to achieve the following objectives:

a) Maximum output from the setup

b) Fabrication of jigs from materials in inventory to minimize the budget

c) Provide maximum and easy adjustments in the chassis for better visualization of the chassis and for increasing accuracy in the process.

As a result, angle brackets and ply boards were the selected material. The conceptualized design was implemented into a practical jig as the image below.

**Fig 2.1: Practical Jigs Setup**

The above setup allows flexibility in changing

a) Driver's seat inclination

b) Adjustment of front hoop and main hoop in x-axis

c) Distance of pedal box from the front hoop

## 2.6 OBSERVATIONS

Each driver was asked to sit in the practical jig setup to finalize all the parameters of ergonomics. This was also done to cross verify the line of sight. The theoretical line of sight was also calculated as explained in fig. 2 in addition to the estimation of the nose length. All this achieved by making frequent changes to the jigs. The front hoop was adjusted to ensure the visibility of the shortest driver.

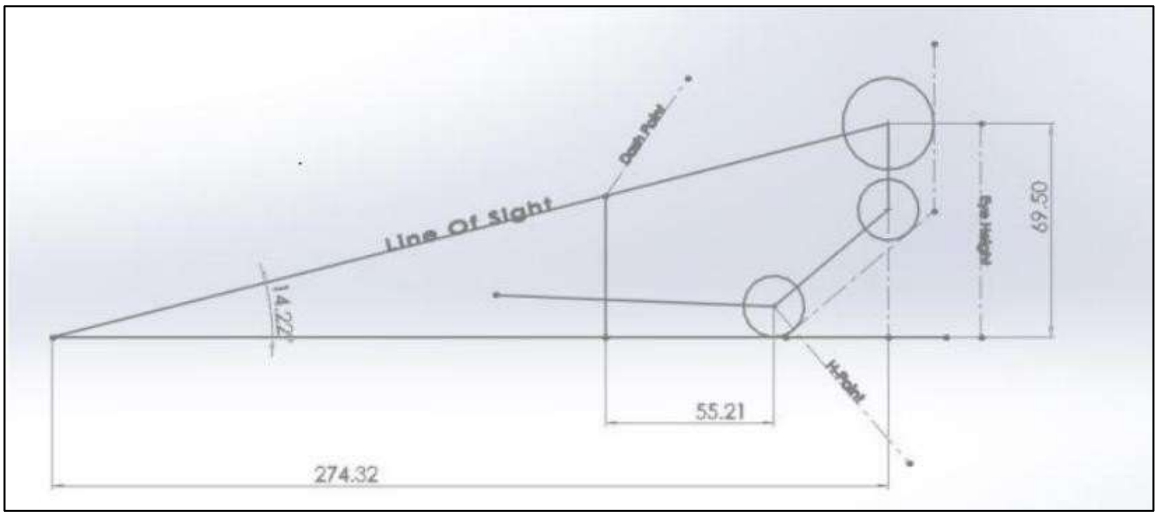


**Figure 2.2: Line of Sight**

Similarly, the height of the main hoop was also adjusted on the basis of the tallest driver, in consideration to the competition rule T3 that states: When seated normally and restrained by the driver's restraint system, the helmet of a 95th percentile male and all of the team's drivers must:

a) Be a minimum of 50mm away from the straight line drawn from the top of the main hoop to the top of the front hoop.

b) Be a minimum of 50mm away from the straight line drawn from the top of the main hoop to the lower end of the main hoop bracing if the bracing extends rearwards.

c) Be no further rearwards than the rear surface of the main hoop if the main hoop bracing extends forwards. The cockpit template is as the following figure:

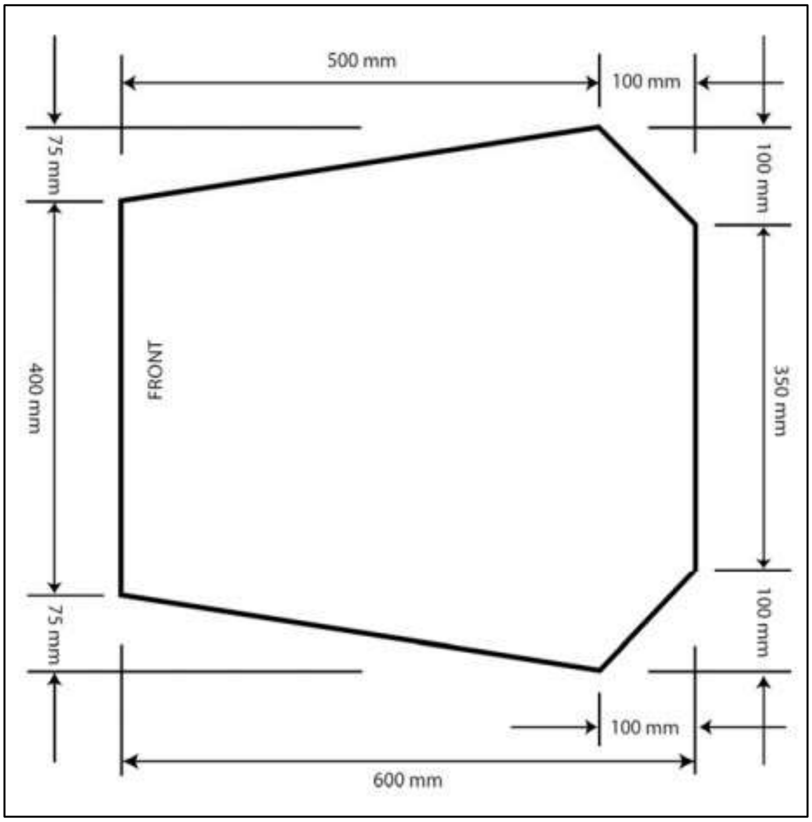


**Figure 2.3: Cockpit Template**

The footwell template is as the following figure:

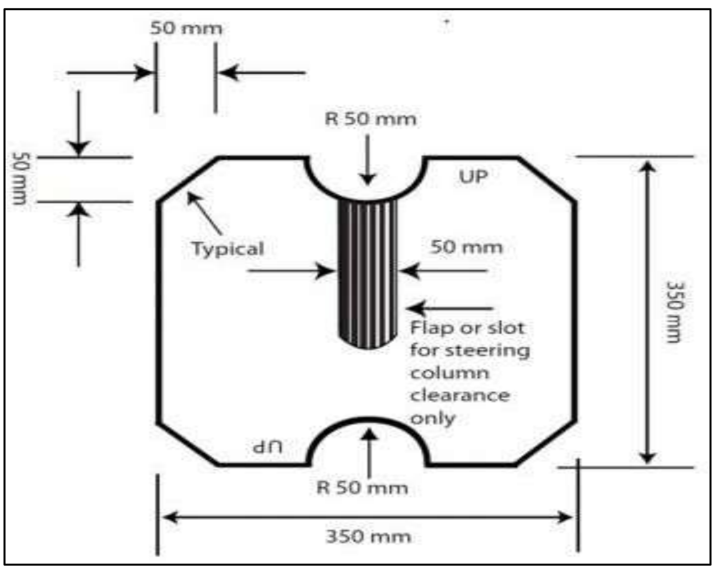


**Figure 2.4: Footwell Template**

**T3.3.2** The 95th percentile male is represented by a two-dimensional figure consisting of two circles of 200mm diameter (one representing the hips and buttocks and one representing the shoulder region) and one circle of 300mm (representing the head with helmet). T3.3.3 The two 200mm circles are connected by a straight-line measuring 490mm. The 300mm circle is connected by a straight-line measuring 280mm with the upper 200mm circle.

The Percy template should be positioned as given in the figure:

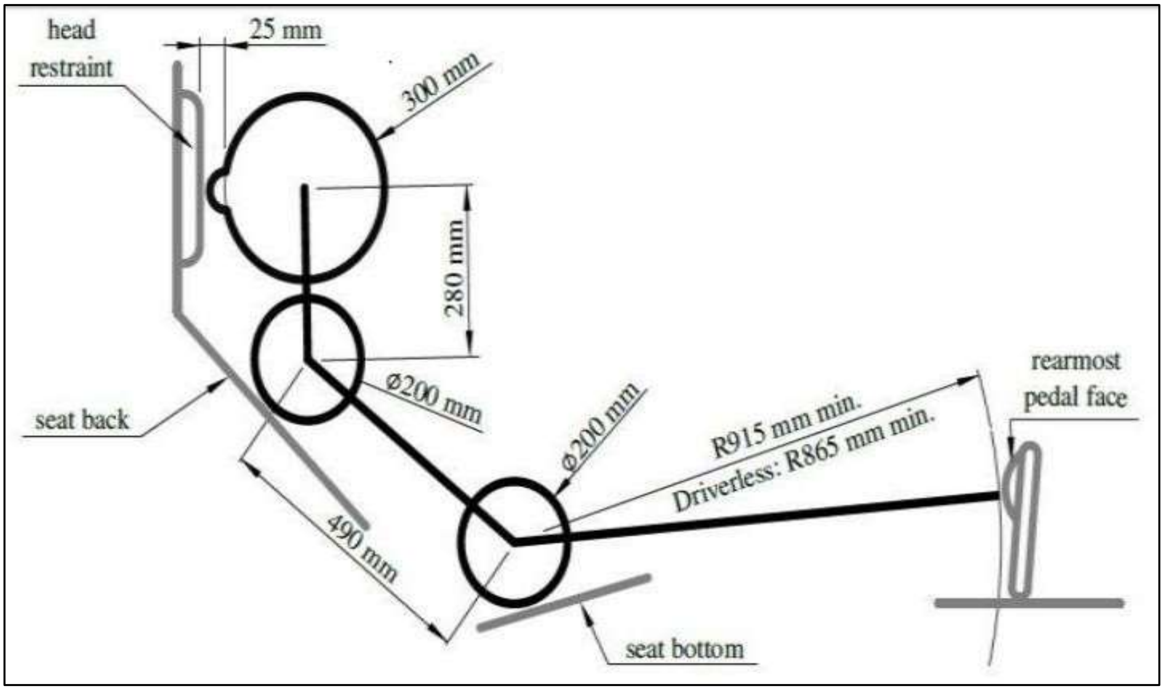


**Figure 2.5: Percy Template**

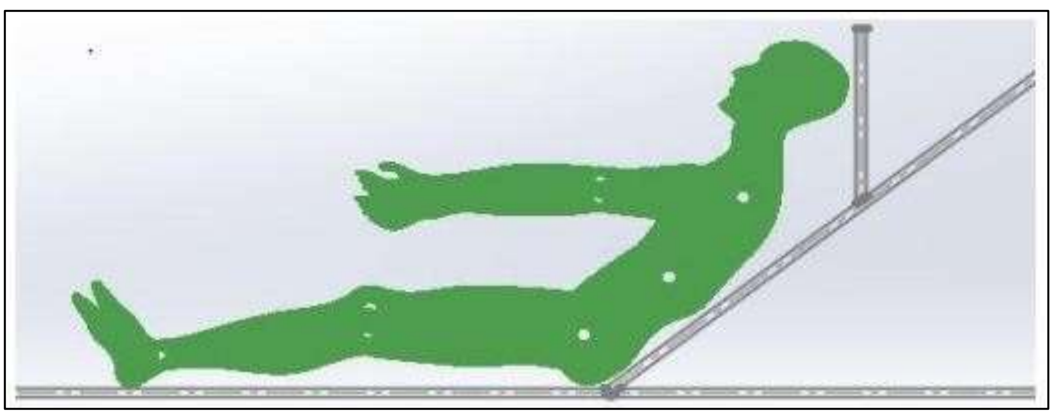

**Figure 2.6: Reclined Seating Positions**

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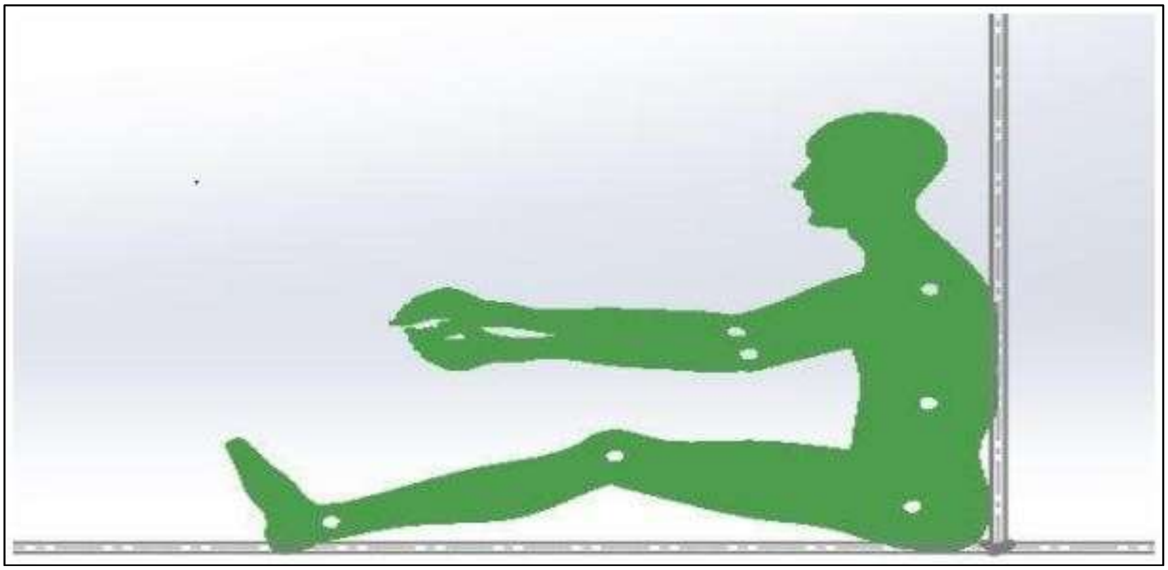

Figure 2.7: Upright Seating Position

Furthermore, knee angles, seat inclination, and elbow angle were also noted down for different drivers. The observation data recorded are mentioned in the tables below:

Table 2.2: Table for Elbow Angle

| DRIVER | ELBOW ANGLE |
|---|---|
| 1 | 107 |
| 2 | 105 |
| 3 | 104 |
| 4 | 94 |
| 5 | 106 |
| 6 | 106 |
| 7 | 111 |
| 8 | 104 |
| 9 | 102 |

**Table 2.3: Table for Knee Angle**

| DRIVER | KNEE ANGLE |
|---|---|
| 1 | 138 |
| 2 | 135 |
| 3 | 135 |
| 4 | 130 |
| 5 | 136 |
| 6 | 136 |
| 7 | 139 |
| 8 | 135 |
| 9 | 134 |

## 2.7 FINALIZED PARAMETER

The desired parameters for the ergonomics were finalized on the basis of driver's feedback and the jigs data. So, from the tables 1, 2 &3, the following were inferred:

**Table 2.4: Finalized Parameters**

| Parameter | Finalized Data |
|---|---|
| **Shifter location** | On the left side of the cockpit |
| **Vehicle controls** | On the right side of the dashboard |
| **Elbow Angle** | $104^0$ |
| **Knee Angle** | $135^0$ |
| **Seat Inclination** | 500 from vertical |
| **Front hoop height** | 46cm |

| | |
|---|---|
| **Main hoop height** | 98cm |
| **Expected nose length** | 114cm |
| **Line of Sight** | 274.32cm |

# CHAPTER 3: SUSPENSION

## 3.1 DESIGN AND PERFORMANCE REQUIREMENTS

- Light, simple and effective system
- Small variations in camber and toe angles during wheel travels using anti-roll bars
- Lighter wheel assembly using optimized designs

## 3.2 MATERIAL SELECTION

Depending upon machinability, strength, cost and market availability two materials considered were:

**TABLE: 3.1**

| **Properties** | **Aluminum T6-7075** | **Aluminum T7-7075** |
|---|---|---|
| **Tensile Strength** | 310 MPa | 510 MPa |
| **Yield Strength** | 276 MPa | 410 MPa |
| **Modulus of Elasticity** | 687.9 GPa | 70 GPa |

## 3.3 THEORY AND CALCULATIONS

The designing phase starts with the selection of tire and rims. The first choice that must be made when choosing tires is what size to use and this dictates the wheel size both diameter and width. 13-inch diameter offers better choice of tires whereas the advantage in choosing a 10-inch diameter is low rotational inertia and better acceleration. Following our design goals, we selected 10-inch diameter tire for lighter wheel assembly.

Value of caster, KPI were decided to be $4^0$ and $8^0$ respectively. These values will be attained by the vehicle in standstill position and 70kg driver seated inside it. Using these values as constrains for rest of the designs, an iterative design procedure was followed.

Keeping in mind that the smaller track should be no less than 75% of the larger track and minimum wheelbase to be 1525mm, the track width was set to 1290mm in front and

1080mm in rear and the wheelbase was decided to be 1550mm.

UBJ and LBJ were determined using SOLIDWORKS. The vertical distance between mounting points of lower A-arms and upper A-arms of 185mm was chosen for the first iteration.

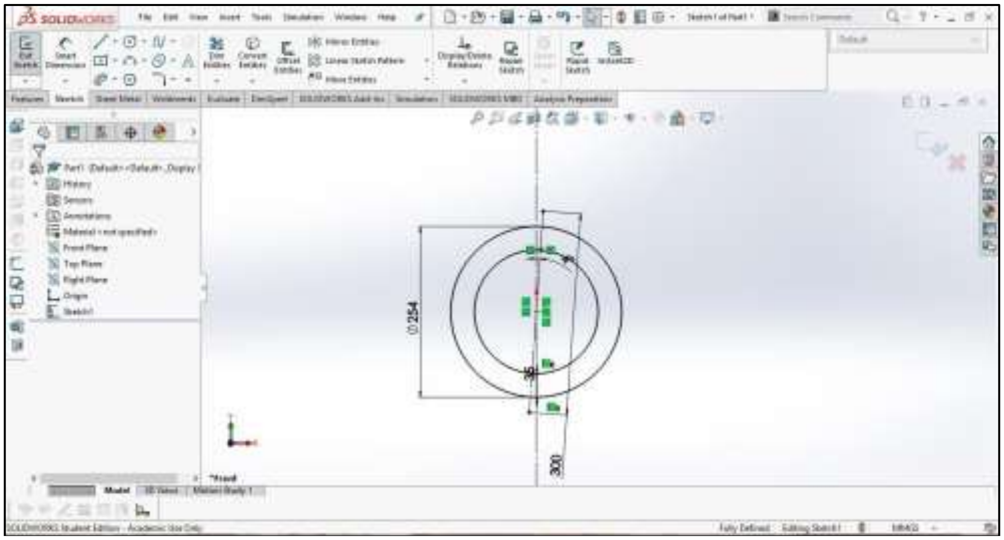


**Figure 3.1: First Iteration**

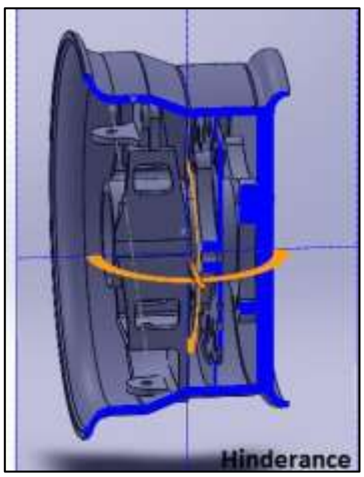


**Figure 3.2: Hinderance**

After completing first iteration, CAD model for upright was designed and assembled. Some hindrance was found and sorted out accordingly

The vertical distance of lower A-arms and upper A-arms was reduced to 170mm and the second iteration was performed. After performing the assembly, the design was finalized. After UBJ and LBJ were finalized, the hard points were found after 4 iterations. The graphs obtained are as follows:

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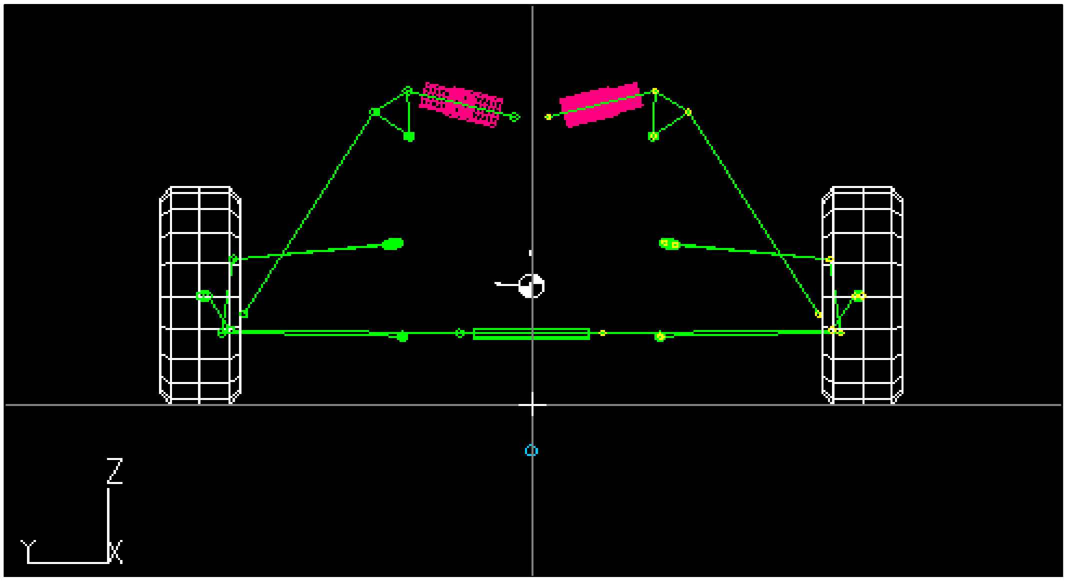


**Figure 3.3: Geometry Simulation on Lotus** 1

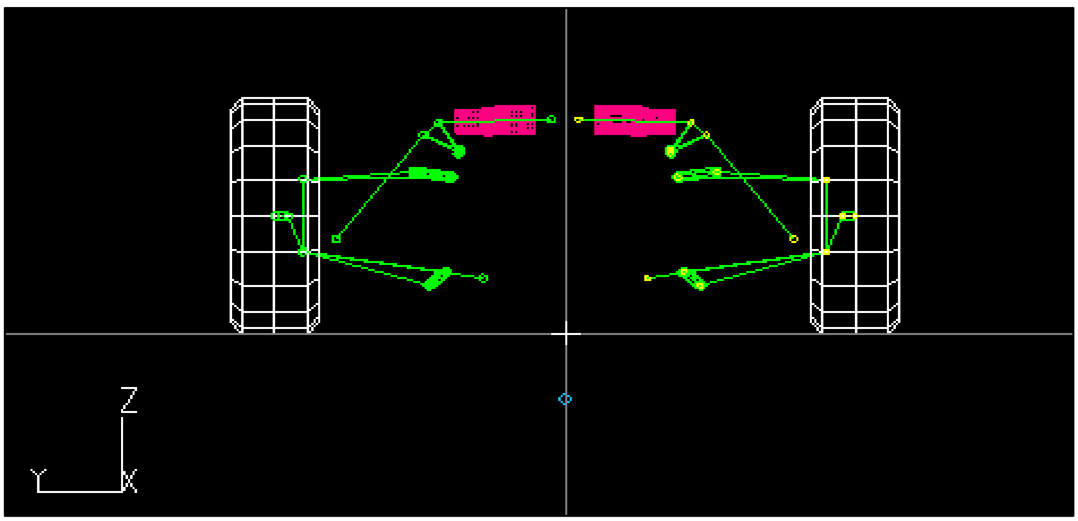


**Figure 3.4: Geometry Simulation on Lotus** 3

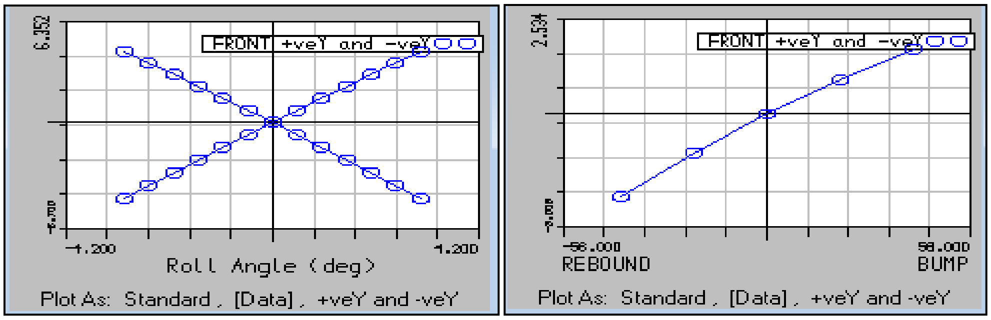


**Figure 3.5: Geometry Simulation on Lotus** 4

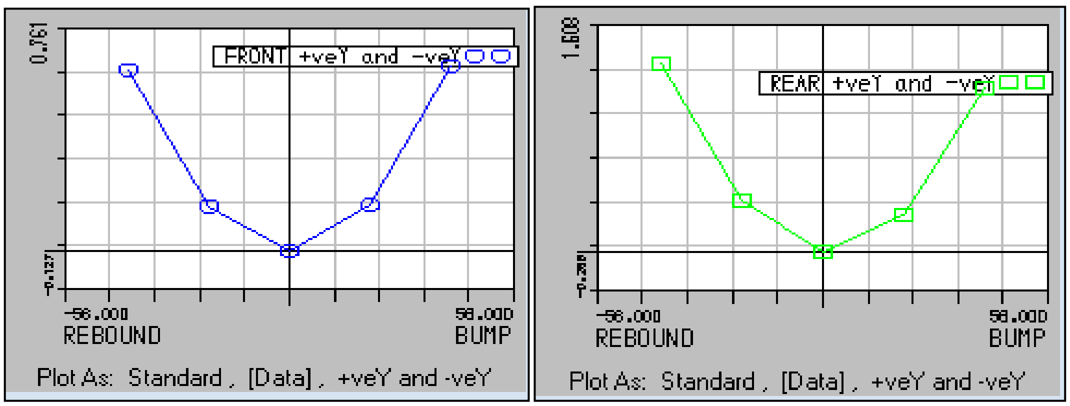


**Figure 3.6: Geometry Simulation on Lotus** 5

## 3.4 <u>ANTI-ROLL BAR CALCULATIONS</u>

1) Angle of twist at maximum wheel travel (Dependent on Motion Ratio)

   $S=r\theta$

   $\Theta=S/r = 63.91/84.255 = 0.758$ rad

2) Dimensions of lever according to packaging which was decided to be 60mm

3) ARB lever ratio

   A=68.65mm

   B=60mm

   Ratio= (60/68.65) = 0.873

4) Roll Stiffness of front of vehicle

   $K_s= 4^* \pi^2{}^*f^2{}^*m_{sm}{}^*MR$ (MR=Wheel Travel/Spring Travel)

   $K_s= 4^* \pi^2{}^*f^2{}^*m_{sm}{}^*MR$
   $K_s=4^* \pi^2{}^*2^2{}^*240^*1.042$

$K_s$=39.491 Nmm
Roll Stiffness= ($K_s$*$S^2$/1375) lb*ft. /deg

Roll Stiffness= (39.491*$2.59^2$/1375) = 0.193*1.36 = 0.263 Nm/deg

5) Motion Ratio of target roll angle set to $3^0$

6) ARB Twist angle

$MR_{roll}$= (ARB twist angle/Target Roll)

$MR_{roll}$= (43.5/3)

$MR_{roll}=14.48^0$

7) Required Roll stiffness of ARB

$K_{arb}$= (momentf/ $MR_{roll}^2$)

$K_{arb}$= ($0.263/14.5^2$)

$K_{arb}$= 1.25 Nmm/deg

## 3.5 RESULTS AND DISCUSSIONS

The final designs and hard points were all validated and tested and the results were matched with the theoretical results.

**Table 3.1: Coordinates Table**

| | **X** | **Y** | **Z** | | **X** | **Y** | **Z** |
|---|---|---|---|---|---|---|---|
| LCA Front Pivot | 511.3 | 252.36 | 143.31 | LCA Front Pivot | 1840.534 | 252.156 | 101.157 |
| LCA Rear Pivot | 747.83 | 251.4 | 143.29 | LCA Rear Pivot | 2067.774 | 220.532 | 129.161 |
| LBJ | 614.59 | 602.94 | 150.69 | LBJ | 2153.89 | 487.03 | 171.74 |
| UCA Front Pivot | 500 | 260 | 340 | UCA Front Pivot | 1803.44 | 282.28 | 339.981 |
| UCA Rear Pivot | 743.77 | 280.327 | 335.37 | UCA Rear Pivot | 2032.416 | 210.374 | 329.014 |
| UBJ | 625.41 | 581.11 | 305.31 | UBJ | 2186.11 | 487.03 | 323.499 |
| Push Rod Body Point | 613.44 | 560 | 190 | Push Rod Body Point | 2110.23 | 425.785 | 198.526 |

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| | | | | | | | |
|---|---|---|---|---|---|---|---|
| Push Rod Rocker | 613.44 | 306.03 | 616.01 | Push Rod Rocker | 2016.81 | 263.34 | 417.71 |

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| | | | | | | | |
|---|---|---|---|---|---|---|---|
| Point | | | | Point | | | |
| Inner Track Ball Joint | 657.9068 | 585.7773 | 156.6 | Inner Track Ball Joint | 2203.89 | 487.03 | 171.125 |
| Outer Track Ball Joint | 590.9184 | 139.7 | 149.5 | Outer Track Ball Joint | 2114.436 | 153.075 | 116.041 |
| Damper Body Point | 613.44 | 32.729 | 605.51 | Damper Body Point | 1997.34 | 25.37 | 451.22 |
| Damper Rocker Point | 613.44 | 241.19 | 660.19 | Damper Rocker Point | 2005.53 | 235.07 | 443.66 |
| Wheel Spindle | 620 | 630 | 228 | Wheel Spindle | 2170 | 516 | 247.05 |
| Wheel Center | 620 | 645 | 228 | Wheel Center | 2170 | 540 | 247.05 |
| Rocker Axis 1 | 603 | 237.38 | 565 | Rocker Axis 1 | 2019.89 | 197.2 | 379.76 |
| Rocker Axis 2 | 623.44 | 237.38 | 565 | Rocker Axis 2 | 2038.4 | 196.74 | 387.31 |

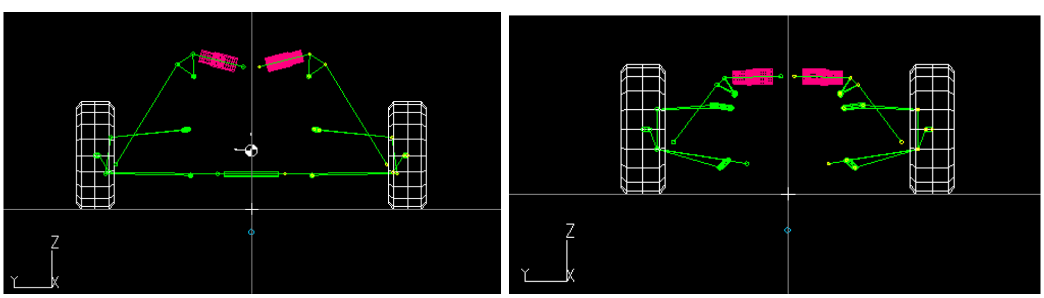

**Figure 3.7: Geometry Simulation on Lotus** 6

Hence after number of iterations, the suspension system was designed and manufactured.

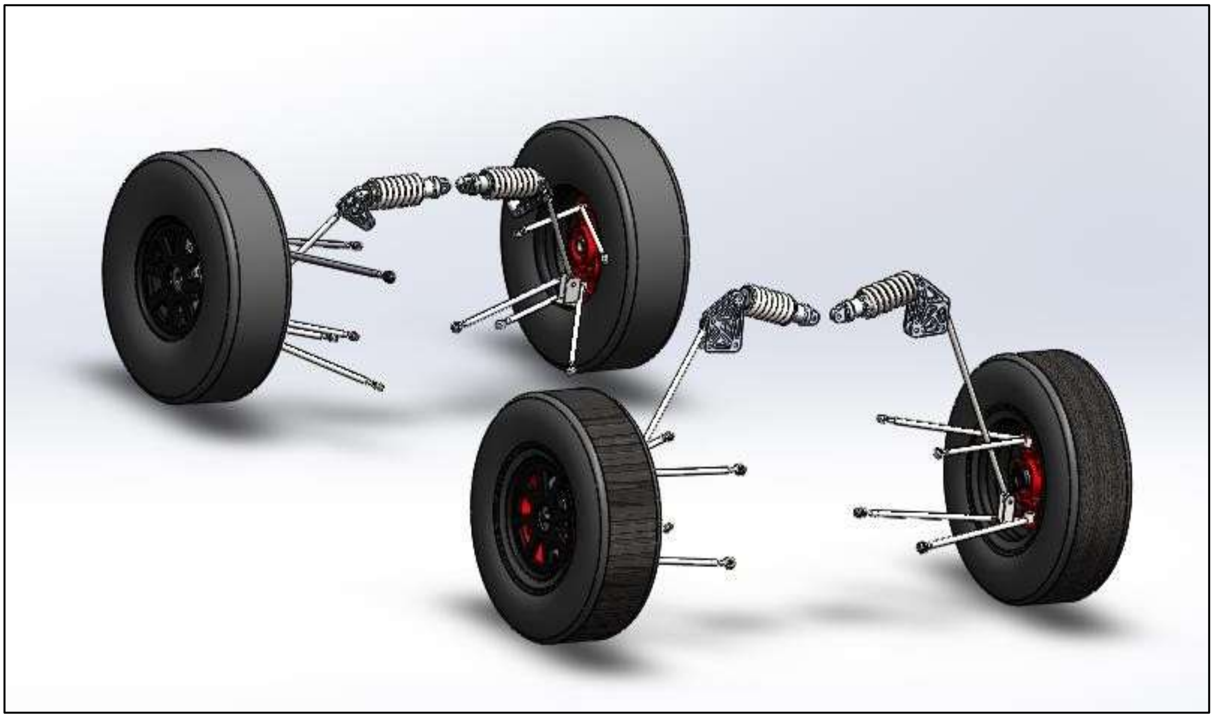

**Figure 3.8: Finalized Suspension Geometry CAD of HR19**

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# CHAPTER 4: STEERING

## 4.1 OBJECTIVE

The main goal for designing the Steering System of HR17 was decided by focusing on the maneuverability of the car thus maximizing the score in the dynamic events. The steering system uses a manual rack-and-pinion mounted to the base of the frame. The rack is bottom-mounted and hence the CG of the rack, pinion, and tie rods are lowered. The aluminum steering wheel provides the benefits of more leg room and better visibility due to the flat top and bottom, all while being lighter than the previous model.

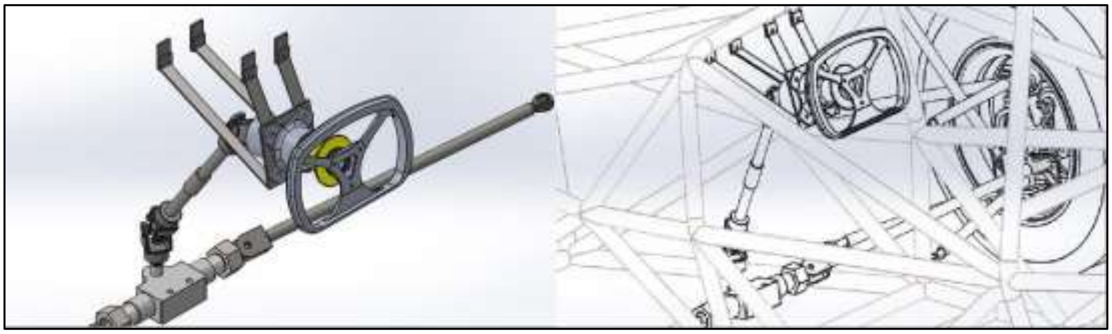

**Figure 4.1: Steering Set-up**

## 4.2 DESIGN GOALS

- To maintain proper roll over stability during steering.
- Minimum turning radius possible with desired stability.
- Minimum Steering Ratio according to driver comfort, thus making the steering system sensitive as sensitive as possible for proper and fast turns during cornering (i.e., for quicker response).
- Avoiding failure within working conditions.

## 4.3 DESIGN AND PERFORMANCE REQUIREMENTS

The steering system is one of the most key designs for overall handling and stability of the car.

Certain parameters like *caster angle, kingpin angle, scrub radius* etc have been kept in mind while deciding the geometry. The following steps involved:

- Geometry Setup
- Geometry validation
- Designing of mechanism
- LOTUS for analysis of linkage geometry
- SOLIDWORKS for CAD modelling
- ANSYS was used for FEA

## 4.4 GEOMETRY SETUP

It is the most important step of designing because steering geometry actually decides the work of vehicle during different driving conditions like Straight line driving, braking, acceleration and cornering etc. Lotus Shark Analysis software was used for simulation.

## 4.5 STEERING RATIO

The steering ratio is the ratio of how much the steering wheel turns in degrees to how much the wheel turns in degrees. After the feedback from performance of the previous year's car and taking into account the driver's feedback, steering ratio is set to be *4.61:1* for easy handling.

## 4.6 STEERING SYSTEM GEOMETRY

The steering system has Ackerman geometry with a maximum inner wheel turn angle of about 38.00°, outer wheel turn angle of 33.30° and the steering rack being placed at a distance of 33.30 mm behind the front wheel centre.

## 4.7 RACK TRAVEL

According to the driver's feedback, the maximum turn required in total on the steering wheel is 113°. Thus, after calculations described afterwards, the required rack travel was set to be 25 mm.

**Table 4.1: Geometrical Values**

| GEOMETRY | VALUE |
|---|---|
| RACK TRAVEL | 25.00 mm |
| TOE ANGLE (Inner Wheel) | 38.00° |
| TOE ANGLE (Outer Wheel) | 33.30° |
| CAMBER ANGLE VARIATION | (-3.47° to +2.96°) |
| ACKERMAN % | 27.10% |
| TURNING CIRCLE RADIUS | 2171.73 mm |
| CASTER ANGLE | 4° |

## 4.8 CONSTRAINTS AND CONSIDERATIONS

The steering system must affect at least two wheels and must have positive steering stops placed either on the uprights or on the rack.

**Constraints:**

- Thus, prevented steering linkages from locking up and the tyres from contacting any other part of the car.
- Allowable steering free play is limited to 7° in total measured at the steering wheel.

**Considerations:**

- Mechanical compliance was taken care of by assuring almost negligible play in all the linkages.
- Simple Design
- Light weight
- High Serviceability
- Responsiveness to Driver Input
- Cost Effectiveness

**Table 4.2: Material Selection**

| Material | Mild Steel | Aluminum T7-7075 | Carbon Fibre |
|---|---|---|---|
| **Density (Kg/m$^3$)** | 7900 | 2830 | 1600 |
| **Specific Modulus (10 $^6$ m$^2$s$^{-2}$)** | 25 | 26 | 113 |
| **Strength to weight ratio (N/m2 per kg)** | 36010.83426 | 166077.73 | 241073.84 |
| **Availability** | Available at nearby stores | Have to be ordered when required | |
| **Cost Effectiveness** | ₹65/Kg | ₹710/Kg | ₹1350/m$^2$ |
| **Maintenance** | Needs to be painted to prevent rusting | No rusting, thus painting is not necessary. Good for parts under static loading. | Easier to maintain in comparison to Mild Steel and Aluminum |
| **Effect on Overall Vehicle Performance** | Not as good as the aluminum | Increases the vehicle performance better | Increases the vehicle |
| | and carbon fibre for overall vehicle performance. | than Mild Steel but not as good as the standard carbon fibre. | performance better than the other two. |

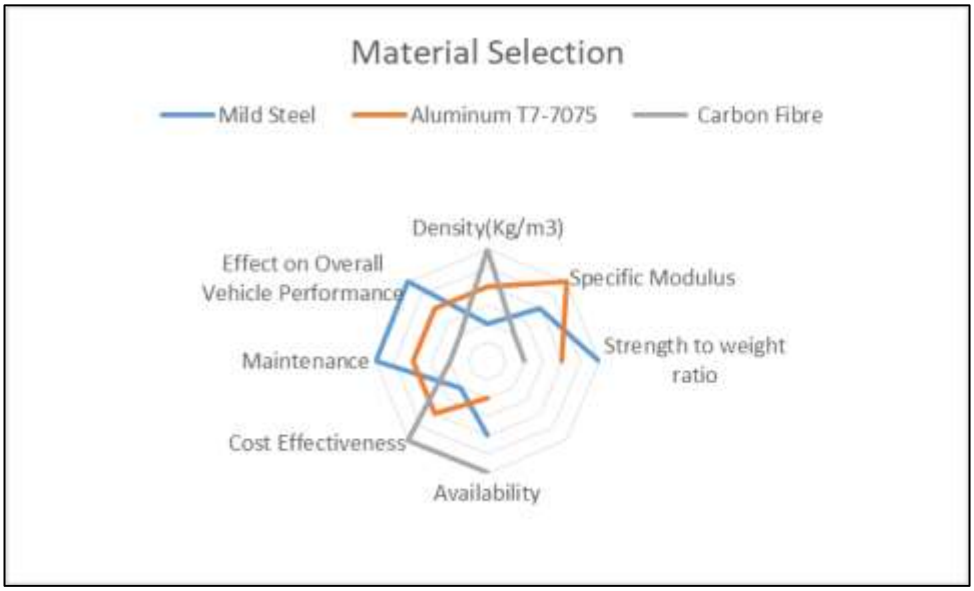


**Figure 4.2: Material Selection Parameters**

This suggests the Mild Steel to be most suitable for the application.

## 4.9 CALCULATION

### Rack travel:

Once the steering ratio has been calculated the rack travel needs to be decided. According to the driver's feedback, the maximum turn required in total on the steering wheel is 113°. Thus, for 90° turn of the steering wheel, rack travel = 19.91 mm. Thus, for 113° turn of the steering wheel, the required rack travel would be = 25 mm.

C-Factor:

C-factor = Travel/360° pinion revolution.
= (19.91 * 4)
= 78.0335 mm.

Steering Ratio:

Steering Ratio = sin-1(C-factor / Steering Arm Length) / 360
= sin-1 (79.64 / 81.4091) / 360
= sin-1 (0.97826) / 360
= 0.21675

Steering Ratio = 4.61: 1

## 4.10 FINAL DESIGN

### Steering Wheel:

**Figure 4.3: Machined Steering Wheel**

**Weight – 270 gm**

**Material – Aluminum 7050 T7451**
**Machining process used – CNC Milling**

### Bearing Housing:

**Figure 4.4: Machined Bearing Housing**

This is used to give support to the Steering Shaft which is connected to the quick disconnect on one side and to a u – joint on the other. The shaft is supported by two bearings.

**Weight – 222 gm**

**Material – Aluminum 7050 T7451**
**Machining process used – CNC Milling**

**Steering Rack:**

**Figure 4.5: Rack and Pinion Set-up**

Steering Rack is used to convert rotational motion of the steering wheel into translational motion which then is used to turn the tires.
The length of the rack was chosen depending on the specifications, availability and mounting spaces available in the space frame.

**Specifications:**

- Rack Pinion Thin Line 11" (Eye to eye)
- 5/8 36 SPLINE SHAFT STILETTO STYLE
- 1 5/16 CENTER TO CENTER OF BOLT HOLE FOR MOUNT
- 3/8’’ BOLT HOLES FOR TIE RODS
- 11 3/8’’ CENTER TO CENTER OF HEIM
- 3 1/2" OF THROW

**Column Attachment:**

**Figure 4.6: Column Attachment**

The column attachment is used to hold the whole steering assembly in proper position with respect to the chassis. It has been designed keeping in mind both static forces in vertically downward direction, i.e., due to weight of the steering column and dynamic forces generated during steering.

**Material – Mild Steel (3mm plate)**
**Weight – 140 gm**
**Machining process used – Conventional machining process (grinding, cutting, MIG welding)**

## 4.11 GEOMETRY CALCULATION

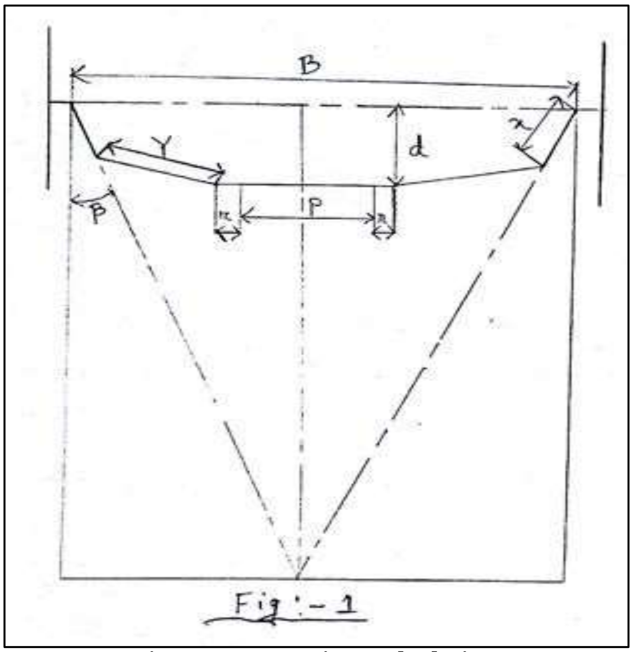


**Figure 4.7: Steering Calculations**

From the figure alongside,

$$y^2 = \left[\frac{B-(P+2r)}{2} - x\sin\beta\right]^2 + [d - x\cos$$

**……. (i)**

Where,
x = Steering Arm length
y = Tie rod length
p = rack casing length
p+2r = rack ball joint center to center length.
q = rack travel
d = distance between front axis and rack center axis.
B = Ackerman Angle.

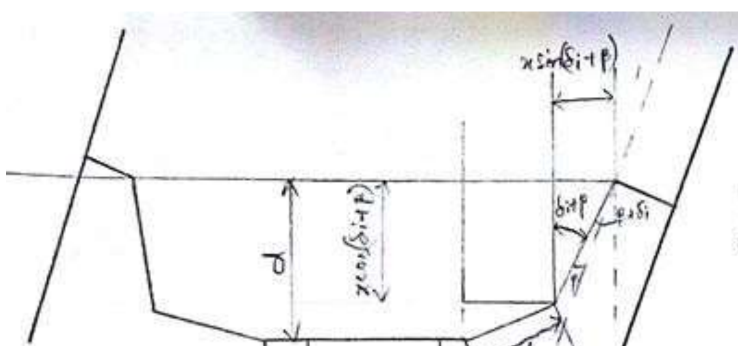


**Figure 4.8: Steering Calculations**

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From this figure it is clear that,

$$y^2 = \left[\frac{B}{2} - \left(\frac{p}{2} + r - q\right) - x\sin(\partial i + \beta)\right]^2 + [d - x\cos(\partial i + \beta)]^2 \ldots\ldots(ii)$$

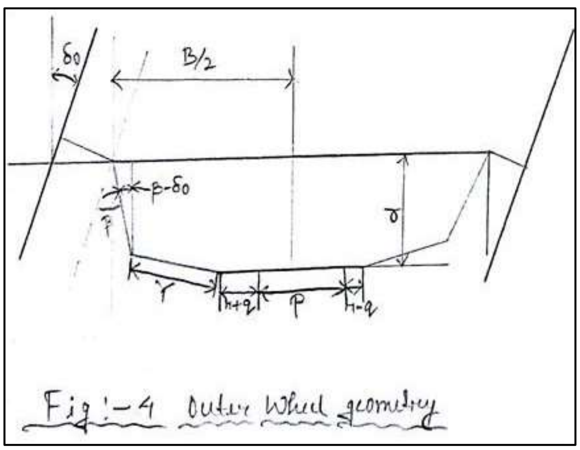


Figure 4.9: Steering Calculations

From figure 4, it is clear that,

$$y^2 = \left[\frac{B}{2} - \left(\frac{p}{2} + r + q\right) + x\sin(\partial o - \beta)\right]^2 + [d - \beta)]^2 \ldots\ldots(iii)$$

## Methodology: -

For the 11-inch Steering rack as ordered from sandparts.com,
p = 88.9 mm
r = 101.6 mm
B = 1290 mm (Track Width of the vehicle)

L = 1550 mm (Wheelbase)

From Lotus analysis software, the values of the following were decided after several iterations: -

$\beta$ = 49.4° (Steering Arm Angle)

$\partial i$ = 37.02° (Maximum Inner Wheel Angle)

$\partial o$ = 31.57° (Maximum Outer Wheel Angle)

q = 25 mm (Rack travel)

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**As stated above, we have the following three equations with us: -**

- $y^2 = \left[\frac{B+(P+2r)}{2} - x\sin\beta\right]^2 + [d - x\cos\beta]^2$
- $y^2 = \left[\frac{B}{2} - \left(\frac{p}{2} + r - q\right) - x\sin(\partial i + \beta)\right]^2 + [d - x\cos(\partial i + \beta)]^2$
- $y^2 = \left[\frac{B}{2} - \left(\frac{p}{2} + r + q\right) + x\sin(\partial o - \beta)\right]^2 + [d - x\cos(\partial i - \beta)]^2$

**From (ii),**

$$y^2 = \left[\frac{B}{2} - \left(\frac{p}{2} + r - q\right) - x\sin(\partial i + \beta)\right]^2 + [d - x\cos(\partial i + \beta)]^2$$

$$y^2 = \left[\frac{1290}{2} - \left(\frac{88.9}{2} + 101.6 - 25\right) - x\sin(37.02° + 49.4°)\right]^2 + [d - x\cos(37.02° + 49.4°)]^2$$

Or, $y^2 = [645 - 121.05 - 0.998x]^2 + [d - 0.0624]^2$

Or, $y^2 = [523.95 - 0.998x]^2 + [d - 0.0624x]^2$……(iv)

**From equation (iii),**

$$y^2 = \left[\frac{B}{2} - \left(\frac{p}{2} + r + q\right) + x\sin(\partial o - \beta)\right]^2 + [d - x\cos(\partial i - \beta)]^2$$

Or, $y^2 = [645 - 171.05 - 0.3062x]^2 + [d - 0.9519x]^2$

Or, $y^2 = [473.95 - 0.3062x]^2 + [d - 0.9519x]^2$…...(v)

**From (i),**

$$\boxed{y^2 = \left[\frac{B+(P+2r)}{2} - x\sin\beta\right]^2 + [d - x\cos\beta]^2}$$

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$$or,\ [(523.95)^2 - 1045.8042x + 0.9960x^2\ ] + [d^2 - 0.1248xd + 3.89376 * 10^{-3}x^2\ ] = [248951.1 - 757.60568x + 0.57638x^2\ ] + [d^2 - 1.3014xd + 0.4234x^2\ ]$$

$$or, \quad [274523.6025 - 1045.8042x + 0.99x^2\ ] + [d^2 - 0.1248xd + 3.89376 * 10^{-3}x^2\ ] = [248951.1 - 738.372x + 4x^2\ ] + [d^2 - 1.3014xd\ ]$$

$$or, \quad [25572.5] = 309.62x - [1.1766xd + 3.8937 * 10^{-3}\ x^2] \dots.(vii)$$

Or, $y^2 = \left[\frac{1290+(88.9+203.2)}{2} - 0.7592x\right]^2 + [d - 0.6507x\ ]^2$

Or, $y^2 = [498.95 - 0.7592x\ ]^2 + [d - 0.6507x\ ]^2$ …(vi)

**Equating Equations (iv) & (vi), we get,**

$$\boxed{or,\ [523.95 - 0.998x\ ]^2 + [d - 0.0624x\ ]^2 = [498.95 - 0.74x\ ]^2 + [d - 0.65x\ ]^2}$$

**Equating Equations (v) and (vi),**

Or,

$$[473.95 - 0.3062x]^2 + [d - 0.9519x]^2 = [498.95 - 0.7592x\ ]^2 + [d - 0.6507x\ ]^2$$

Or,

$$[224628.6025 - 290.2469 + 0.0937x^2] + [d^2 - 1.9038xd + 0.9061x^2] = [248951.1 - 757.6056x + 0.57638x^2\ ] + [d^2 - 1.3042xd + 0.4234x^2\ ]$$

Or, $467.3587x + d^2 - 0.6024xd = 24322.5 \dots\dots.(viii)$

<u>On solving equations (vii) & (viii), we get</u>**,**
x = 91.81 mm
d = 29.0816 mm

<u>On putting the value of x and d on equation (i) we get</u>**,**
y = 451.31 mm.

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Hence, results obtained are as follows,
x = Steering Arm Length = 91.81 mm
d= distance between wheel center and steering rack position = 29.0816 mm
y = Length of tie rod = 451.13 mm.

## Graphs

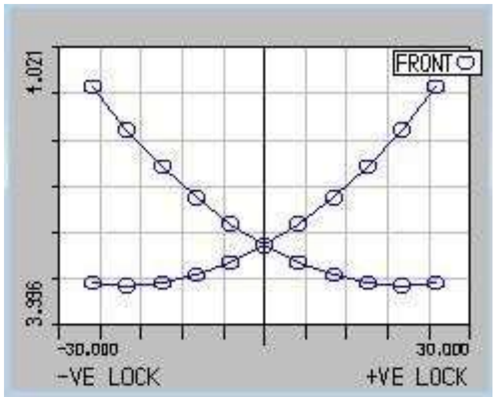


**Figure 4.10: Castor Angle Variations**

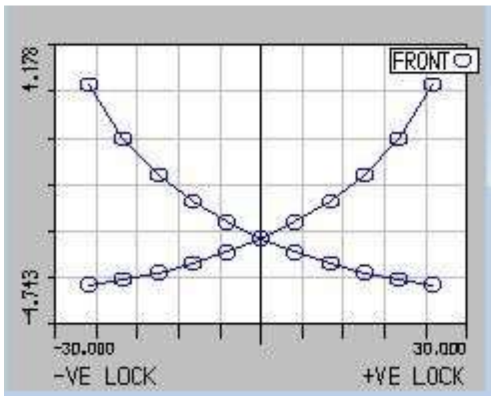


**Figure 4.11: Camber Angle Variation**

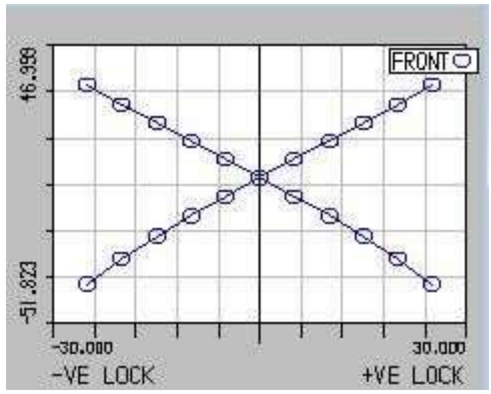


**Figure 4.12: Toe Angle Variation**

## LINKAGE ANALYSIS IN LOTUS

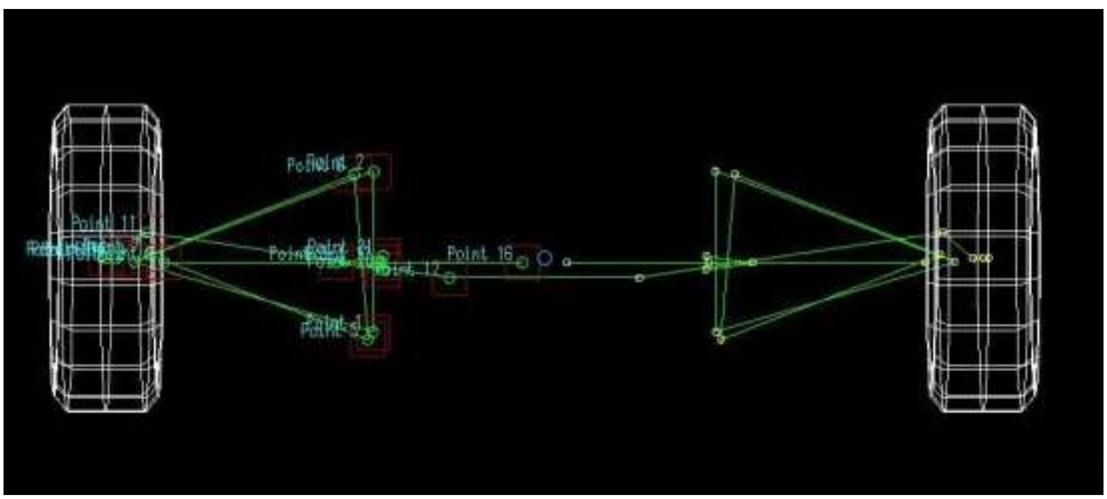


**Figure 4.13: Lotus Analysis of Linkages**

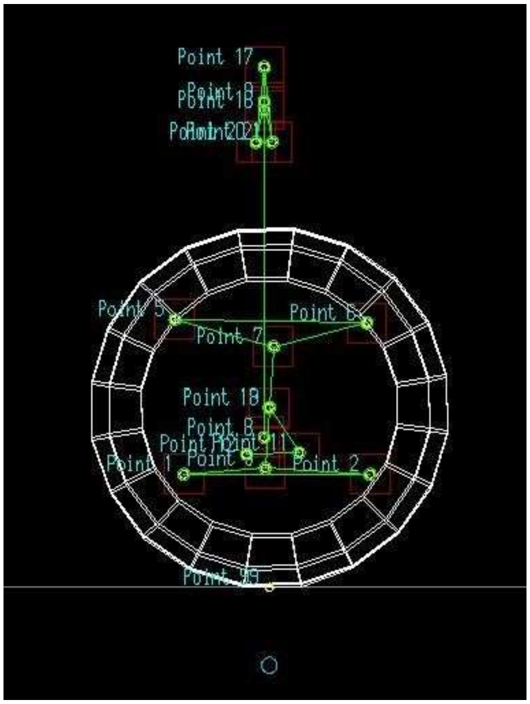

**Figure 4.14: Front View**

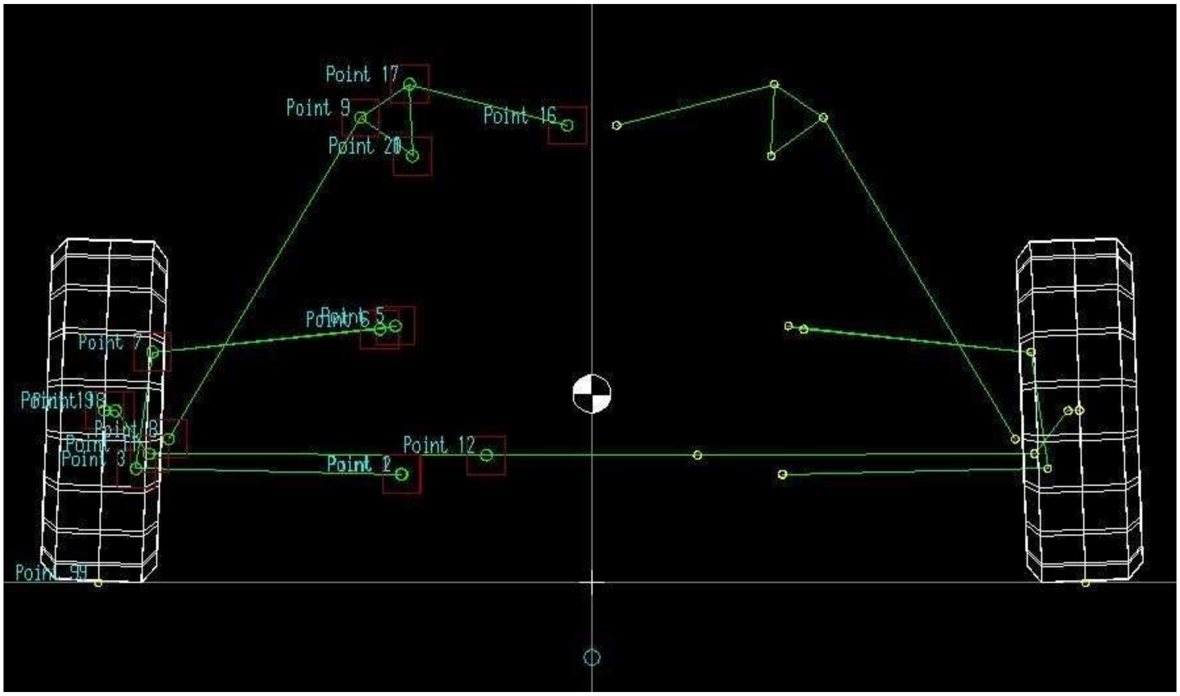

**Figure 4.15: Steering Calculations**

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FRONT SUSPENSION - STEERING TRAVEL

TYPE 14 Double Wishbone, Push Rod to damper

INCREMENTAL GEOMETRY VALUES

| RACK TRAVEL (mm) | TOE ANGLE RHS (deg) | TOE ANGLE LHS (deg) | CAMBER ANGLE RHS (deg) | CAMBER ANGLE LHS (deg) | ACKERMANN (%) | TURNING CIRCLE RADIUS (mm) |
|---|---|---|---|---|---|---|
| -25.00 | -38.00 | 33.30 | 2.96 | -3.47 | 27.10 | 2171.73 |
| -20.00 | -28.84 | 26.33 | 1.21 | -3.32 | 23.35 | 2972.85 |
| -15.00 | -20.87 | 19.62 | 0.05 | -3.09 | 21.18 | 4206.77 |
| -10.00 | -13.57 | 13.05 | -0.80 | -2.80 | 19.91 | 6555.42 |
| -5.00 | -6.66 | 6.54 | -1.47 | -2.44 | 19.24 | 13394.94 |
| 0.00 | 0.00 | 0.00 | -2.00 | -2.00 | 27.10 | 0.00 |
| 5.00 | 6.54 | -6.66 | -2.44 | -1.47 | 19.24 | 13394.94 |
| 10.00 | 13.05 | -13.57 | -2.80 | -0.80 | 19.91 | 6555.42 |
| 15.00 | 19.62 | -20.87 | -3.09 | 0.05 | 21.18 | 4206.77 |
| 20.00 | 26.33 | -28.84 | -3.32 | 1.21 | 23.35 | 2972.85 |
| 25.00 | 33.30 | -38.00 | -3.47 | 2.96 | 27.10 | 2171.73 |

**Figure 4.16: Analysis Values**

# CHAPTER 5: CHASSIS

## 5.1 HR 17 DESIGN GOALS

- Make a lighter chassis by avoiding adding extra members
- Compact packaging of all the components
- Accommodate all components as low as possible
- Lower the driver's seating position
- Improve driver safety and comfort
- Minimize manufacturing error

## 5.2 CONSTRAINTS AND CONSIDERATIONS

**Constraints:**

The constraints that were followed during designing of the chassis are:

1. Envelop of the primary structure and any additional structures fixed to the primary structure must meet the minimum material specification.
2. Node to node triangulation has to be carried out for chassis members projected onto a plane.

**Considerations:**

1. Must meet all Formula SAE requirements, as outlined in the 2018 Formula SAE rulebook.
2. Proper driver ergonomics must be ensured at all the times.
3. Must provide a compact yet not-crammed envelop for the various components of the car
4. Must be able to accommodate all components of the car aiding to its low CG

## 5.3 DESIGN SPECIFICATION

In the formula student competition, performance is not the only important design parameter. Driver's safety and ergonomics also plays an important role. The car has to be built with a relatively small budget, so the chassis must meet the design specifications at all time while remaining simple and straightforward. The main design specifications that were kept in mind during designing the chassis are:

1. **Structural strength**

   The chassis has to withstand the loads from the suspension system and reaction forces from the driver and the engine. No stress peaks should occur due to geometrical inefficiencies.

2. **Low or lightweight**

   Physically lowering the mass (CG) has a much significant effect on the vehicle handling than lowering the mass numerically. Looking at the possibilities, the centre of gravity can be lowered further by reclining the driver. By increasing the angle of the drivers back with the vertical, overall centre of gravity of the car is lowered.

3. **Front impact load**

   A static analysis of a front impact situation is mandatory. The force that acts upon the front bulkhead of the chassis (according to FSG 2017 rule book) is 150 kN. This force is evenly distributed over the outer edge of a front bulkhead going straight into the side walls of the chassis structure.

4. **Ergonomics**

In addition to the structural demands with respect to crash safety, there are some tight geometrical regulations regarding the cockpit to keep the driver safe. This involves mainly dimensional demands for the cockpit to allow for quick driver egress in case of accidents and the roll hoops that surround the driver. These roll hoop regulations are often with respect to the 95$^{th}$ Percentile male.

## 5.4 MATERIAL SELECTION

a. The chassis was built using **AISI 1026** Mild Steel.
b. The reason being its cost effectiveness, availability and ease of manufacturability.
c. This was chosen over **Chromoly 4130** as the difference in overall chassis weight was not significant. Slight compromise was done as far as mechanical properties are concerned in order to be thrifty in over approach.
d. A car chassis is mainly subject to "point" loads whereas Monocoques are best for distributed loads. Therefore, each point load needs a reinforcing patch, which can be a hassle, so it would have been a nightmare to build! It was also difficult to repair depending on construction method. It also had its restrictions around powertrain sections, not really well around the engine because of poor access (to plugs, electrics, oil, etc.) Whereas a space frame steel chassis is well suited to point loads. Manufacturing of the chassis can be done using hacksaw, file, and any type of welding (oxy-acetylene, braze or weld/stick/MIG/TIG), which is why space-frame was used in HR17. These methods can be used for suspension wishbones, miscellaneous brackets. These are easy to repair in steel space-frame. Steel space-frame also provided us with good access to engine, pedals, everything, another benefit of space frame is that they are very simple to analyse accurately with FEA programs, using 2D line elements (beams). Whereas Monocoques, especially composite monocoques, can be extremely difficult to analyse accurately.

As we chose steel spaceframe over monocoque chassis it was time for us to select the steel space frame over which HR 17 chassis would be built. There were three types of material with which our chassis could have been built. Materials like AISI 4130 Alloy (Chromoly), AISI 1020, AISI 1026 was are choice.

Here are the mechanical properties AISI 4130 Alloy (Chromoly), AISI 1020, AISI 1026, which will exactly show why AISI 1026 was excellent choice for manufacturing of our chassis.

## 5.5 MECHANICAL PROPERTIES

The following mechanical properties of cold drawn AISI 4130 Alloy (Chromoly), AISI 1020, AISI 1026 mild steel are displayed in the following table:

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**Table 5.1: Properties of Materials**

| *Mechanical Properties* | | | |
|---|---|---|---|
| **Properties** | **Materials** | | |
| | ***AISI 4130 Alloy Steel(Chromoly)*** | ***AISI 1020 Steel*** | ***AISI 1026 Steel*** |

The above mechanical properties explain why AISI 1026 was the best selection for us over chromoly, AISI 1020.

CDS (Cold Drawn Seamless) tube e.g. AISI 1026 is perfect, inexpensive, easy to work with and needs no post weld heat treatment (something to consider when you have to weld on some afterthought bracket or repair damage), as CDS can also be MIG (Metal Inert Gas) welded. In using Chromoly there was an additional complexity when working with 4130 is the need to normalise the structure after welding. There can be too many catastrophic failures in the HAZ (heat affected zone) of 4130 weldments to know this is necessary, 4130 should also be either TIG (Tungsten Inert Gas) welded or nickel bronze welded, but as we worked with MIG welding AISI 1026 was the best choice for us.

<u>Three types of round tubes thickness were used to build the chassis:</u>

**Table 5.2: Dimensions of Members**

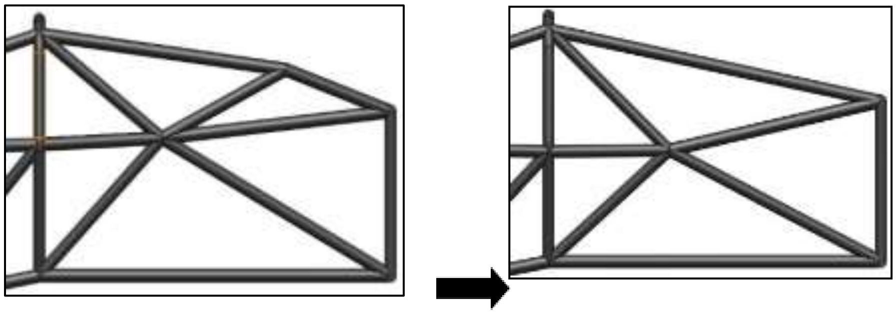
| Section | Tube Type | Dimension |
|---|---|---|
| Main roll hoop | Round Steel tube | 27mm x 3mm |
| Front roll hoop, base ladder and shoulder harness mounting bar | Round Steel tube | 25.4mm x 2mm |
| Real bulkhead | Square tube | 25.4mm x 25.4mm x 2mm |
| The rest of the secondary structure of the chassis | Round Steel tube | 25.4mm x 1.2mm |

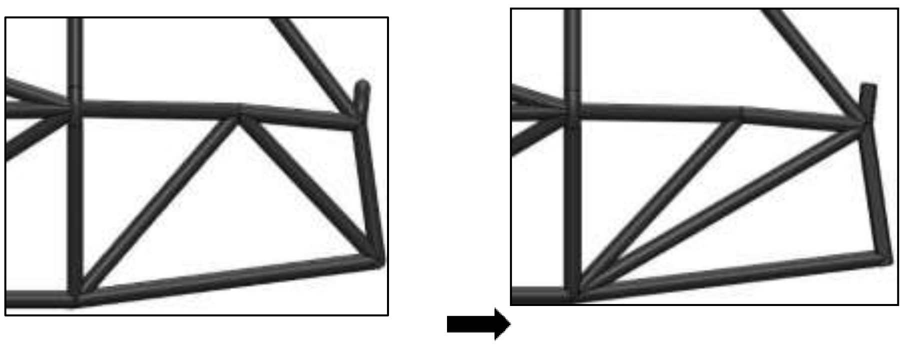
## 5.6 <u>DESIGN ALTERNATIVES</u>

Several iterations were made in the designing process before the construction of the final chassis. Mentioned below are all such iterations:

**1.** The front hoop bracing was made with multiple members to facilitate proper movement of the chassis cross sectional template.

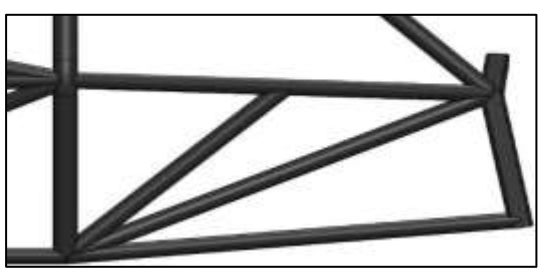
It was later changed to a single member realizing the template would still pass without any hindrance. This resulted in saving a little weight.





The above iteration was made in order to comply with the rule T2.11.5. The rear bulkhead was changed to a square tube structure for easier manufacturability.

The design was further changed. The multiple members connecting the bottom of the main hoop bracing and the upper attachment point of the main hoop with the side impact structure was replaced with a single member to comply with the rule T2.3.5. Rear suspension hard points were slightly altered in order to do so.



## 5.7 DESIGN METHODOLOGY

- **Diver Ergonomics Fixture**

The designing was initialized with the driver ergonomics fixture. Driver was made to sit in the fixture setup to finalize all the parameters of ergonomics, and cross verify the line of sight. In addition to these parameters like the height of front and main hoop, seat inclination, pedal box position, head restraint position, steering wheel position were also finalized. All this was achieved by making frequent changes to the fixture, by pertaining to the rules.

**Figure 5.1: Ergonomics Determination**

- **SolidWorks**

The data acquired from the fixture was used to design the driver cockpit in SolidWorks.

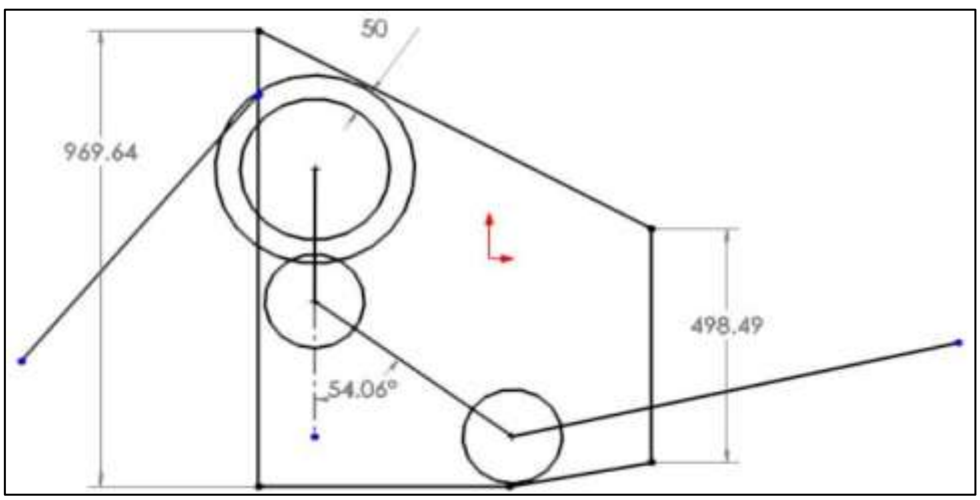



**Figure 5.2: Rules of Templates**

- **Suspension geometry hard-points**

The rest of the chassis was designed in SolidWorks after getting the suspension geometry hard points.

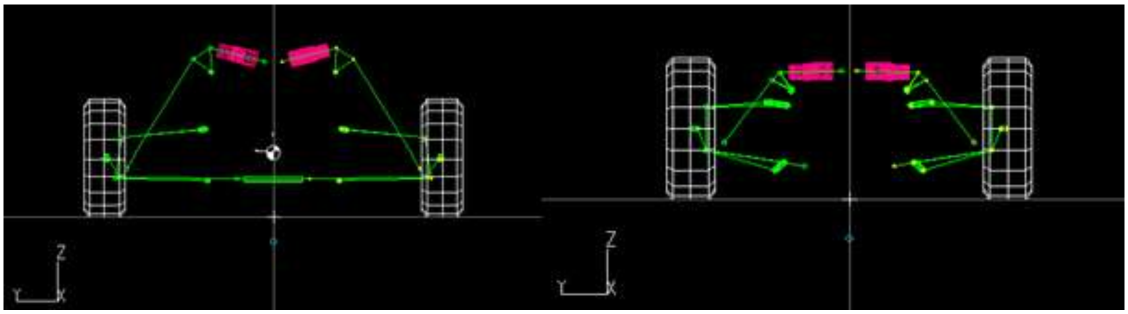


**Figure 5.3: Hard Points Determination in Lotus**

## 5.8 MOCK CHASSIS

PVC pipes were chosen for manufacturing mock chassis. Volume of engine bay was measured to ensure the packaging of different components. Driver was then seated in mock chassis to ensure ergonomic, visibility, safety harness was ensured.

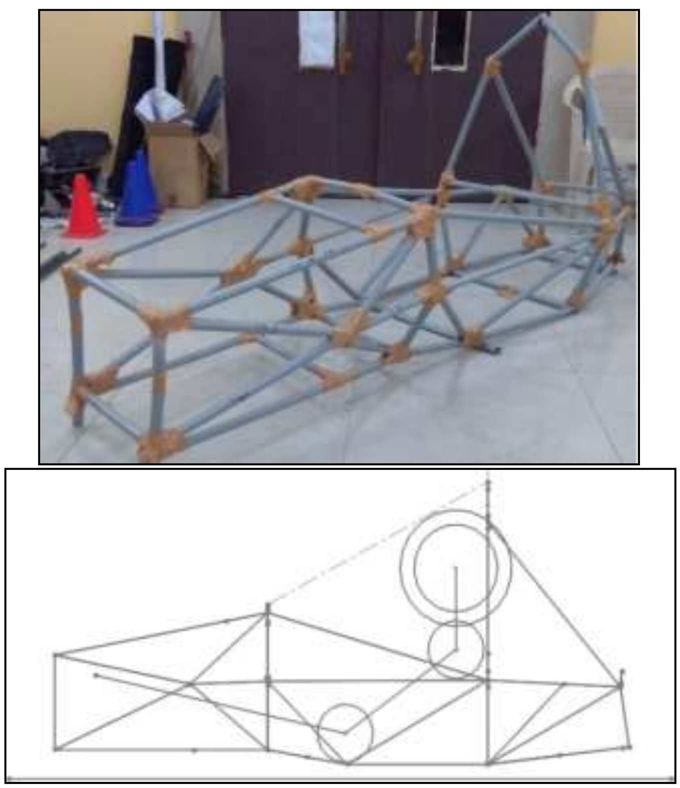


**Figure 5.4: Final Line Sketch**

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## 5.9 MANUFACTURING PROCESS

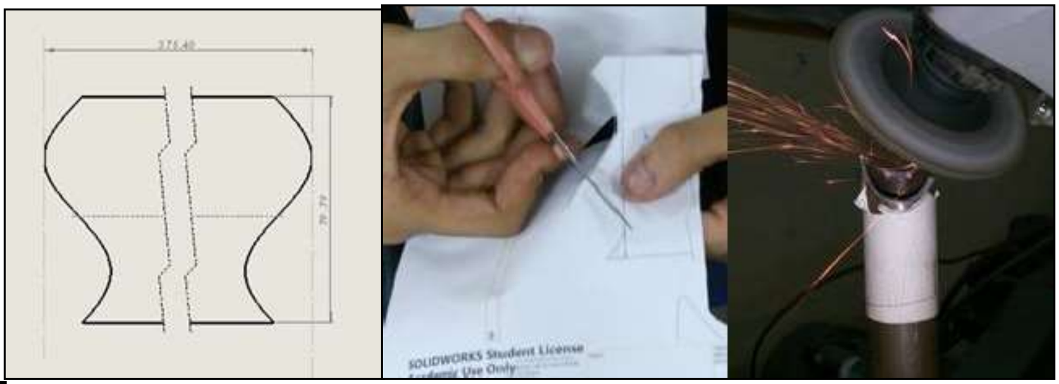
1. Fixtures were machined

**Figure 5.5: Jigs and Fixtures**

2. Main hoop front and base ladder were bent

**Figure 5.6: Front Hoop**

3. Tube profiles were printed from SolidWorks and grinded with pedestal and hand grinder




4. Ladder flex was printed and base ladder was fixed on it.

**Figure 5.7: Base Ladder Fix**

5. Profiled tubes were fixed on the fixtures and after re-confirming the templates, tubes were MIG welded with $CO_2$ as shielding gas.

**Figure 5.8: Welding Front Bulkhead**

6. Clamps were made out rectangular tubes of dimension 41mm x 25mm x 2mm and then profiled to weld on the chassis.

## 5.10 FINAL DESIGN

The final design of the chassis was completed when all the data was collected. Afterwards SolidWorks helped us in assembling the chassis, all the suspension points, brakes, powertrain areas were taken in account. The Formula Bharat Rules were used as a strict guideline to ensure the safety and eligibility of our design. After subjecting the chassis to design selection process i.e. its analysis.

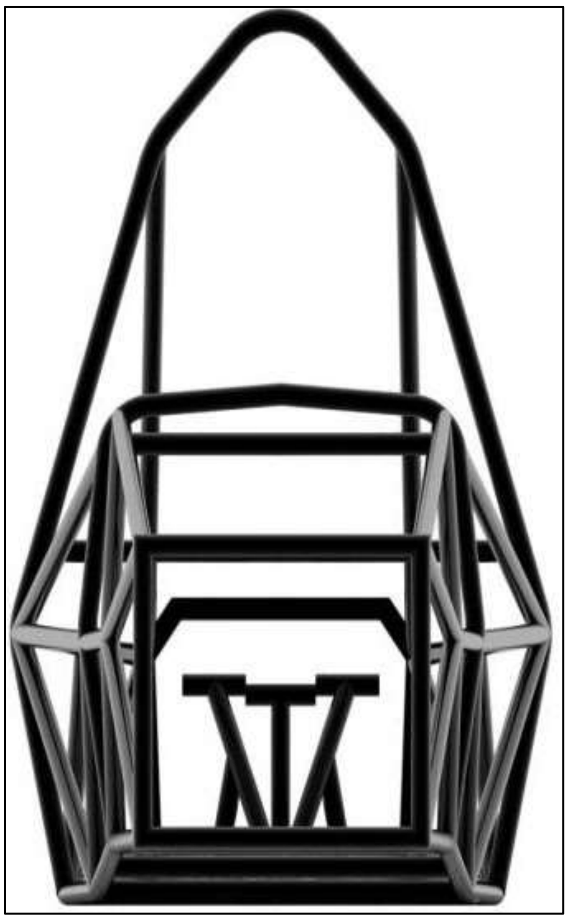


**Figure 5.9: Front View CAD of Final Chassis**

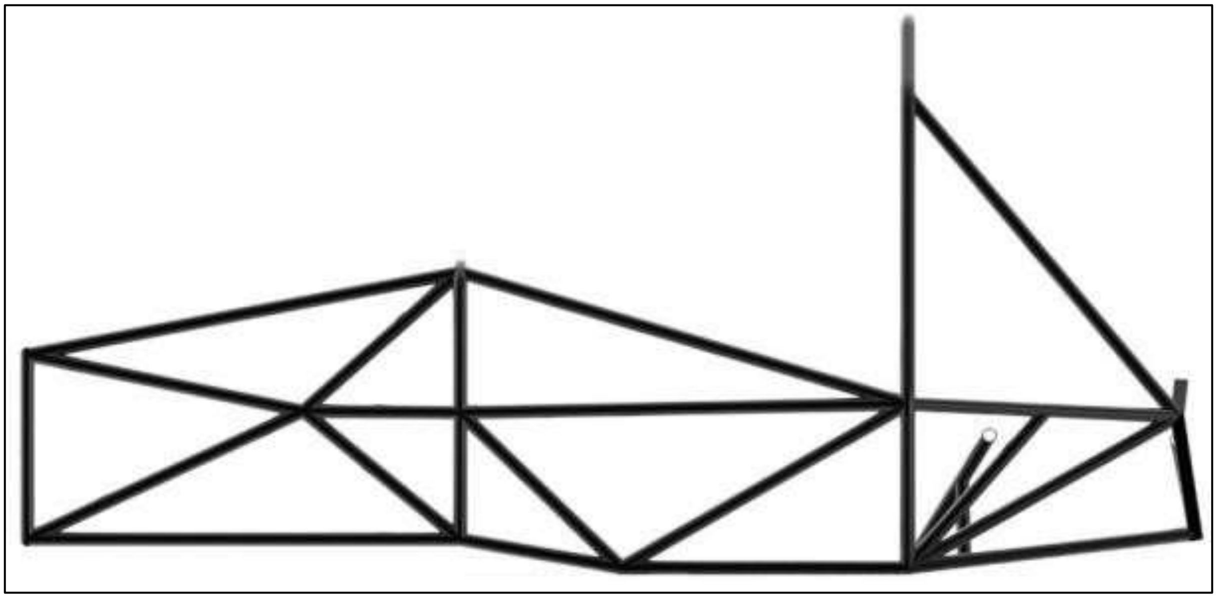


**Figure 5.10: Side View CAD of Final Chassis**

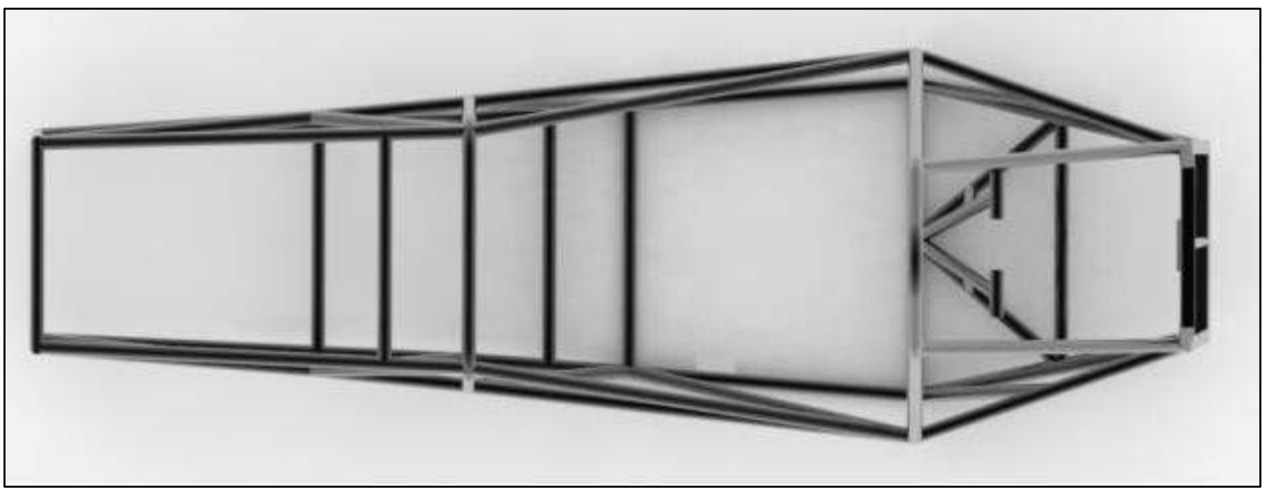


**Figure 5.11: Top View CAD of Final Chassis**

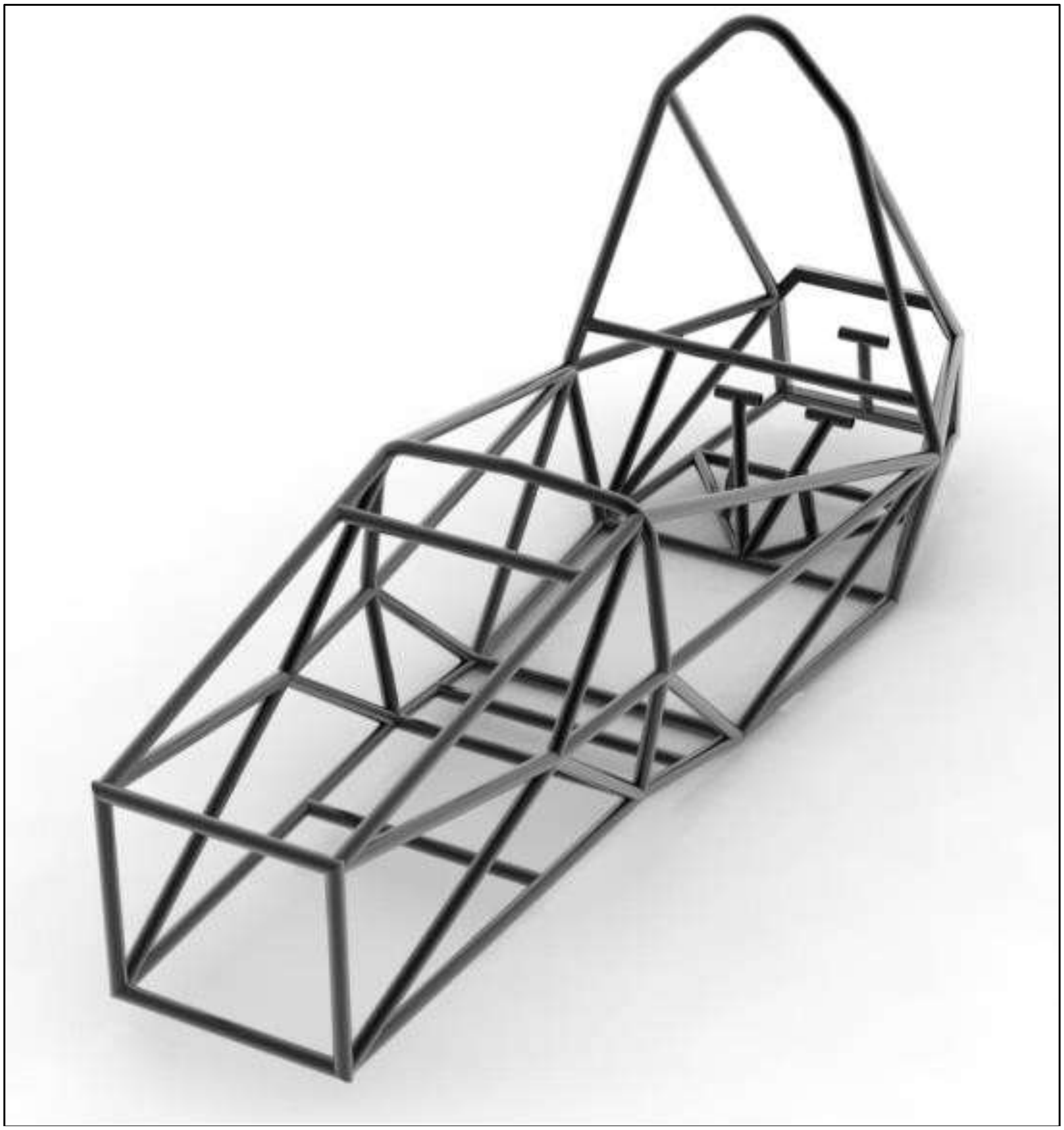


**Figure 5.12: Final Design of Chassis**

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**Engineering Design**

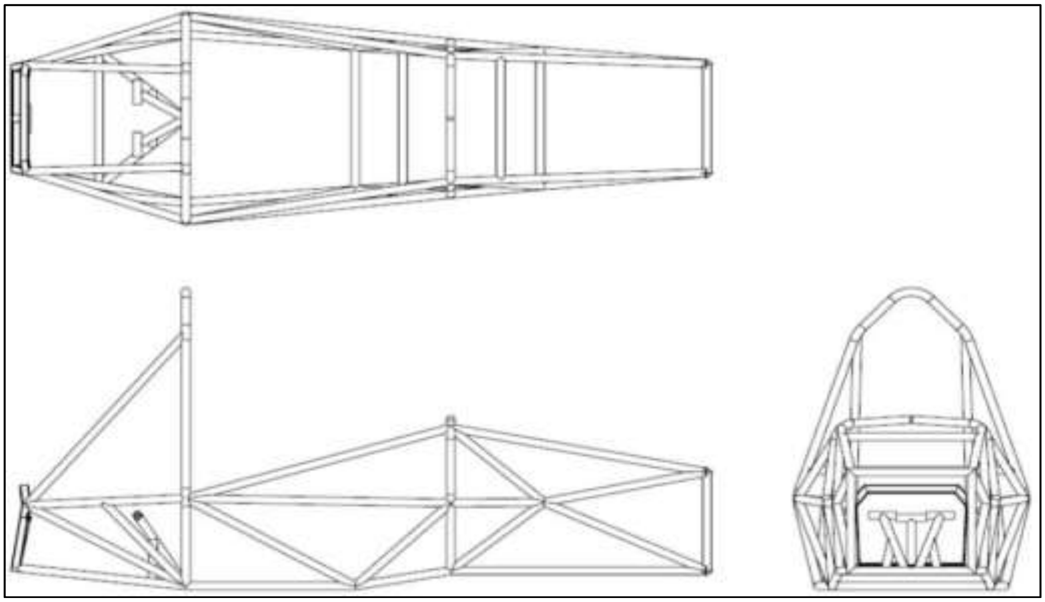

**Figure 5.13: Engineering Design of Chassis**

## 5.11 FINITE ELEMENT ANALYSIS (FEA)

FEA was used in HR 18 to determine the stiffness of our design, the FEA was done for 4 different parameters of chassis i.e. static structural analysis, dynamic analysis, thermal and vibrational analysis.

- Static Structural Analysis**:**

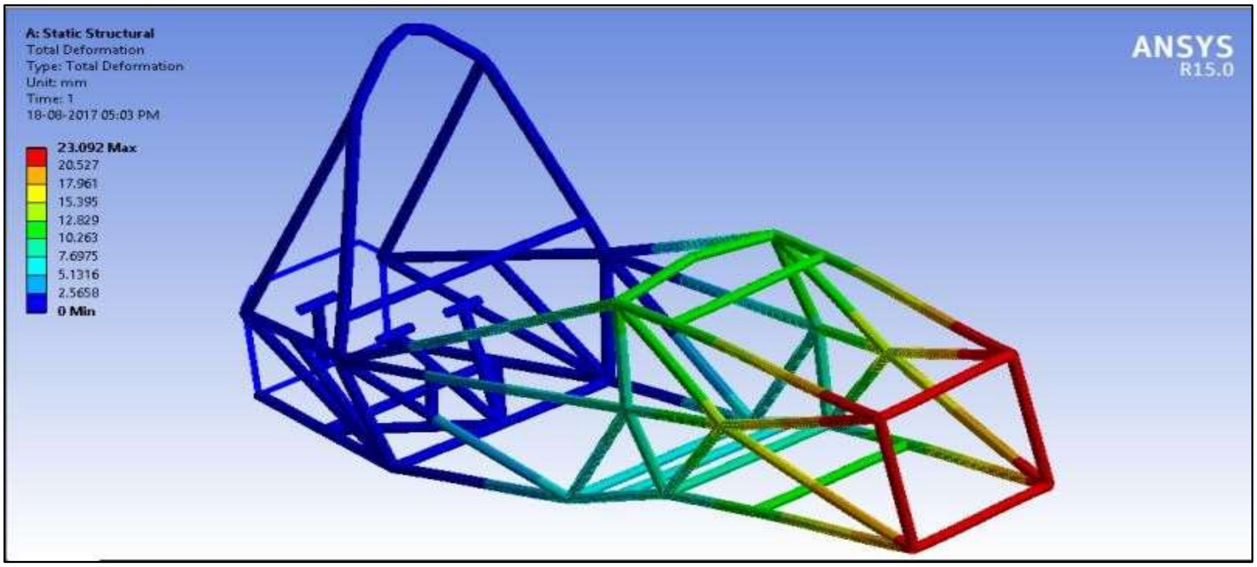


**Figure 5.14: Structural Analysis**

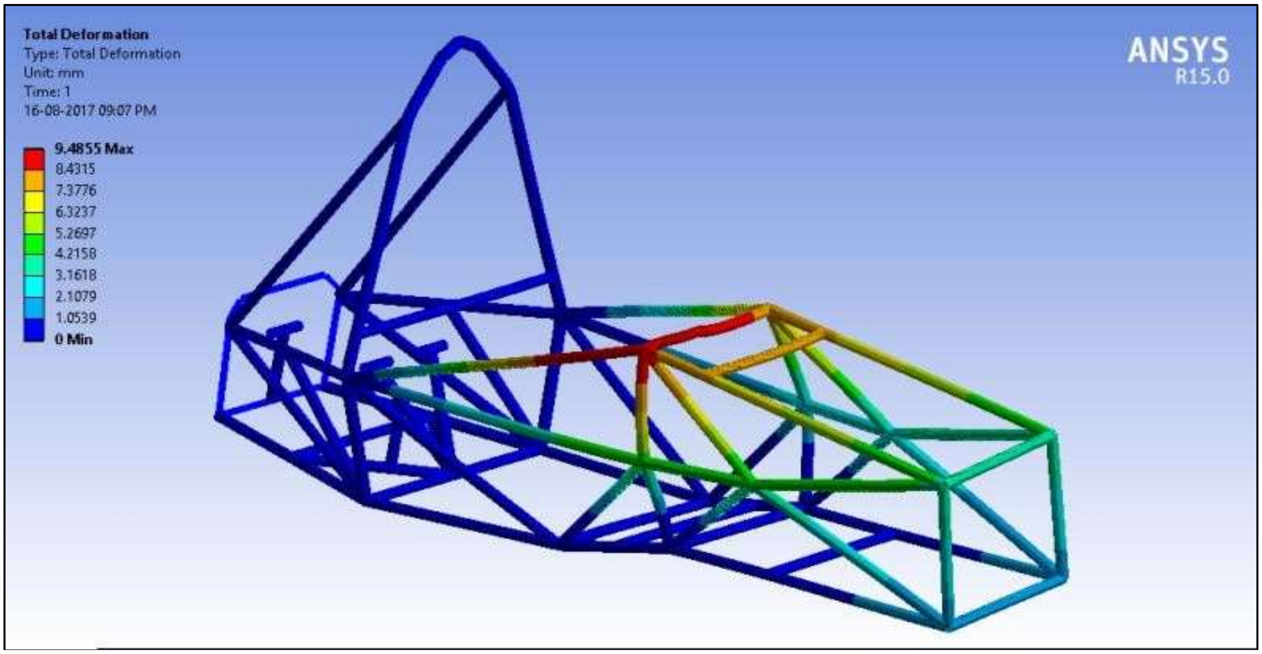


**Figure 5.15: Maximum Deformation= 9.45 mm**

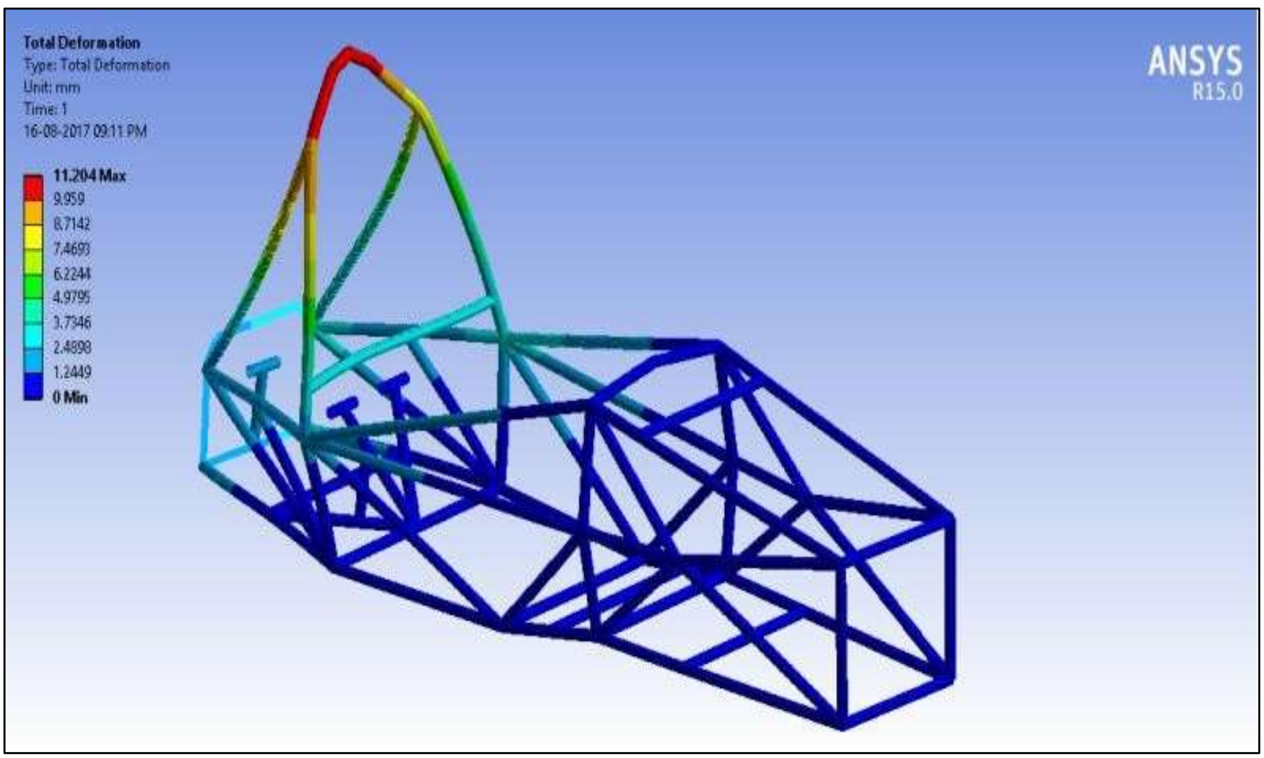


**Figure 5.16: Maximum Deformation= 11.28 mm**

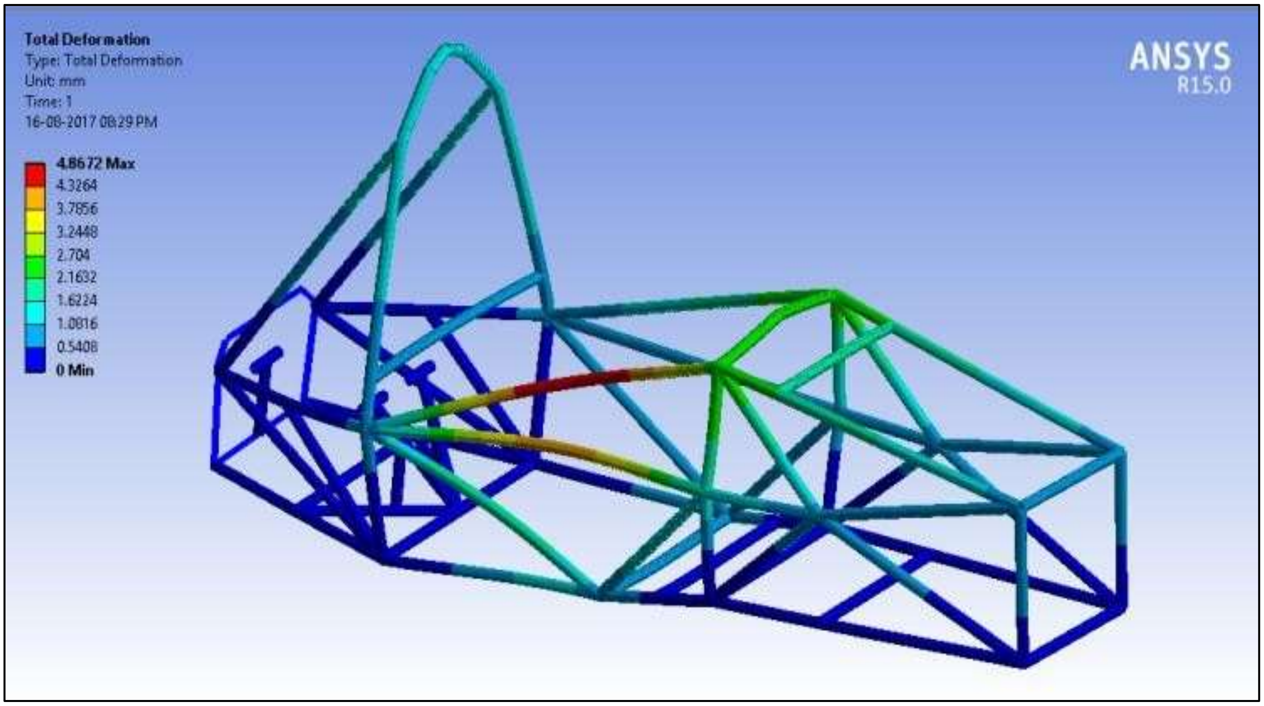


**Figure 5.17: Maximum Deformation= 4.48 mm**

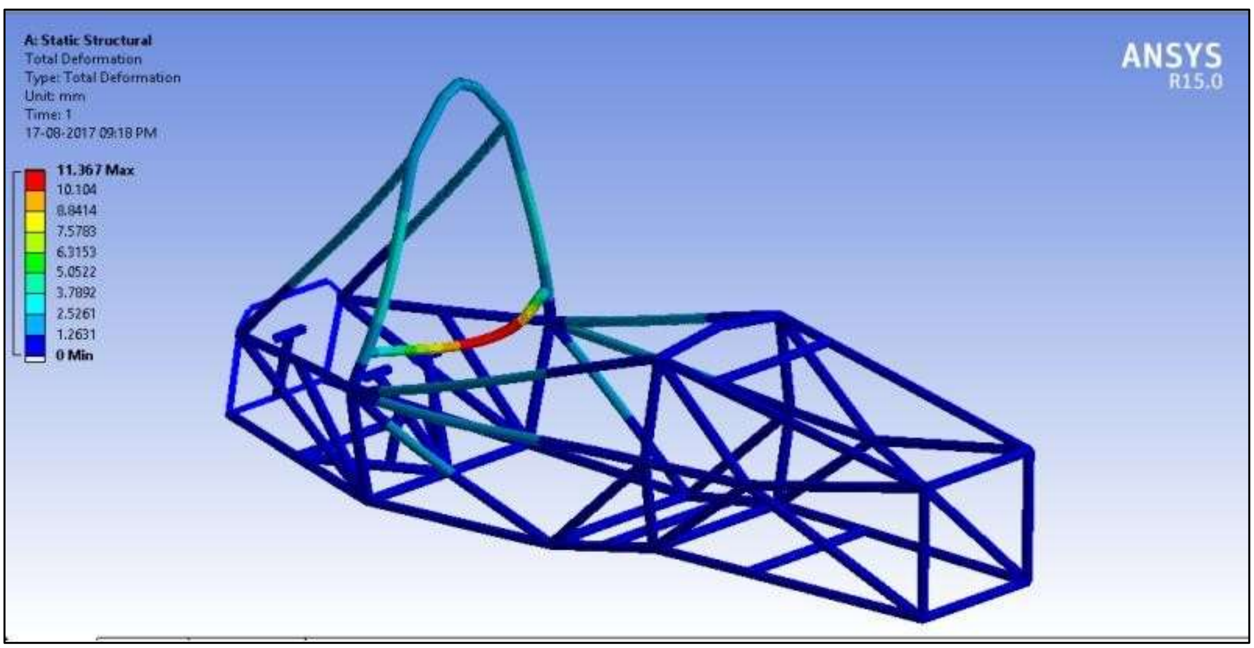


**Figure 5.18: Load on shoulder harness**

## DYNAMIC ANALYSIS:

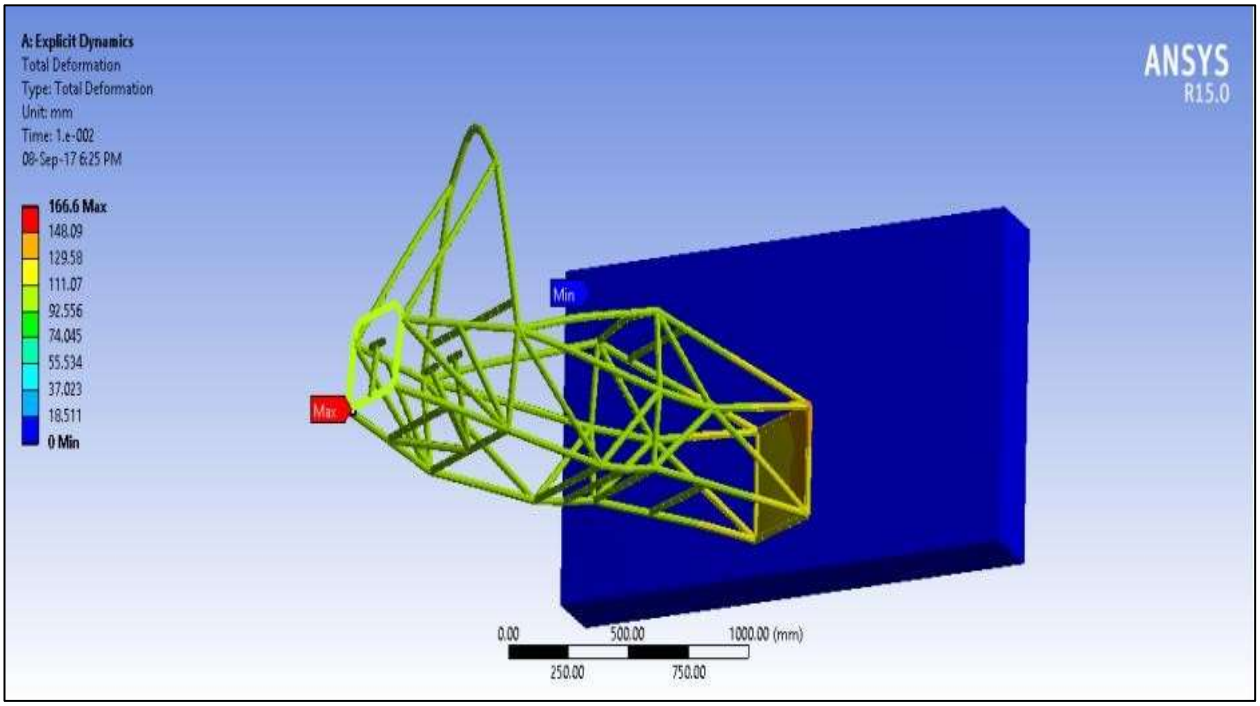


**Figure 5.19: Deformation at 30-degree angle**

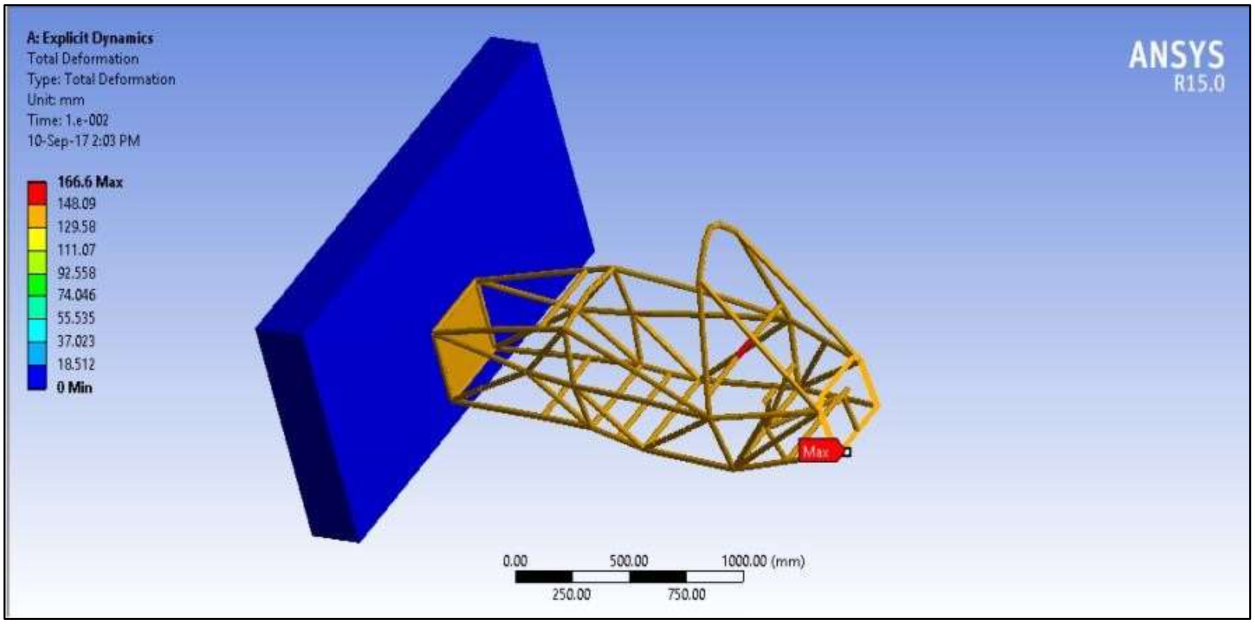


**Figure 5.20: Front Deformation**

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# CHAPTER 6: POWERTRAIN

## 6.1 DESIGN AND PERFORMANCE REQUIREMENTS

The design and performance requirements considered for the vehicle's powertrain are as follows:

- Power unit with high power output and descent torque figures.
- Intake manifold with low pressure drop and flow separation.
- Exhaust manifold compatible with packaging.
- Lightweight tripod housing.
- Sprocket providing high rate of acceleration and decent torque.
- Adjustable differential mount assembly for different chain drive configuration.

## 6.2 LITERATURE SURVEY

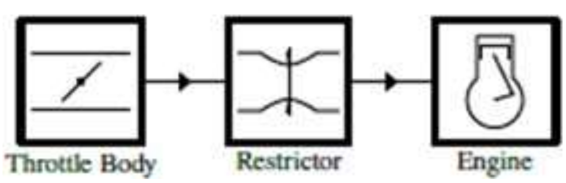
Powertrain consists various parts featuring various concepts and theories;

**Intake manifold-**

The intake manifold of a FSAE vehicle has to comply with certain rules such as;

- The maximum restrictor diameter which must be respected at all times during the competition for Gasoline fueled vehicles is 20mm.

- In order to limit the power capability from the engine, a single circular restrictor must be place in the intake system and all engine airflow must pass through the restrictor. The only allowed sequence of components for naturally aspirated engines are: throttle body, restrictor, and engine.




- The circular restricting cross section may not be movable or flexible in any way.

- All parts of the engine air and fuel control systems (including the throttle and the complete air intake system, including the air filter and any air boxes) must lie within the surface defined by the top of the roll bar and the outside edge of the four tires.

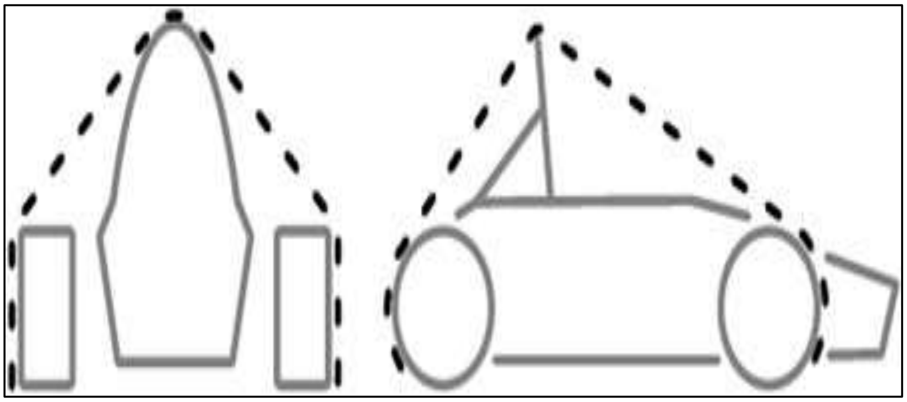


**Fig 6.1: Bracket Limitation of Intake Assembly**

The design of intake manifold is divided into two parts i.e. the restrictor and the plenum. The design of restrictor is critical as it plays a major role in the nature of the flow. A restrictor is used to reduce the airflow to the engine. There are two types of instruments which can be used as a restrictor i.e. the Orifice as well as the Venturi. The Orifice is a simple plate with a hole drilled in it and the Venturi is a tube having a converging and a diverging section with a throat of circular shape connecting the both.

**Table 6.1: Comparison**

| Orifice plate | Venturi tube |
|---|---|
| The coefficient of discharge is between 0.58 to 0.65. | The coefficient of discharge is between 0.95 to 0.975. |
| The pressure loss is medium on a scale of high to low. | The pressure loss is low on a scale of high to low. |
| The cost of manufacturing is cheaper. | The cost of manufacturing is high. |
| The manufacturing is easy as there is just a hole to be drilled on a plate. | The manufacturing is difficult as there is a conical profile to be made. |

From the above comparison Venturi is found to better than orifice in terms of performance requirement. From the above table it is observed that the pressure drops and coefficient of drag characteristics are better in case of Venturi. Designing the restrictor using Venturi fulfils all the necessary design and performance parameters.

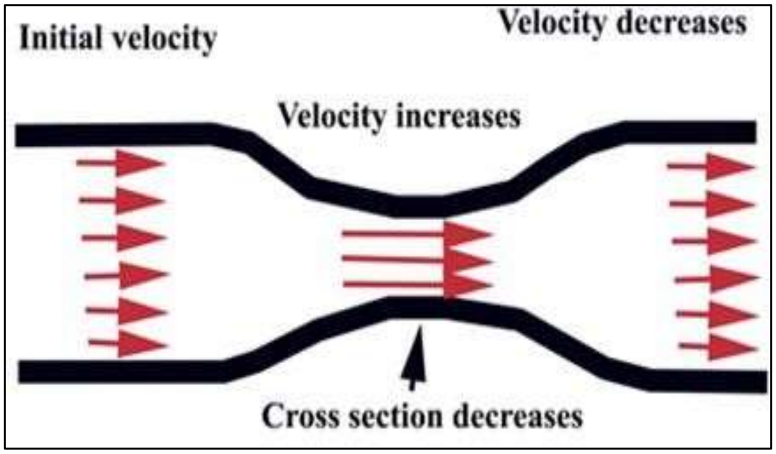



**Fig 6.2: Flow through Venturi**

To further reduce the pressure, drop and flow separation bell-mouth inlet should be provided and the divergence of the outlet should be gradual.

Plenum is the part of the of intake manifold which is connected to the restrictor, it serves the purpose of air environment which stores the surplus air ready to be sucked by the engine. The plenum helps to build the required air pressure which is lost due the presence of restrictor. The volume of the plenum should ideally range between 3 to 10 times of the displacement of the engine and the dimensions of the manifold should be calculated using David Vizard's theory/rule.

## 6.3 SPROCKET

Sprocket interfaces with the differential and the drive chain. Size of the tooth count of the sprocket will influence the acceleration of the car and transitions between the gears of the transmission during dynamic events.

Sprocket takes the linear force from the drive chain and produce rotational energy to rotate the differential as well as the wheels. The diameter of the sprocket helps to produce a moment arm to produce a torque on the differential.

The 45 teeth sprocket is considered because using such a sprocket will result in better start in the acceleration.

Our Main objectives during the designing of a sprocket are to make sprocket lighter and to make sprocket larger for larger moment arm

## 6.4 DIFFERENTIAL MOUNT

The differential mount is designed according to the vehicle's packaging and requirement.

The differential rotates freely inside the differential mount with the help of bearing which is mounted along with the differential.

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## 6.5 HALF SHAFTS

Half shafts are the thing that connect the differential to the hub assembly. The half shafts that we use are OEM shafts with bearing tripods on both ends. These tripods are CV joints that maintain the constant velocity so that the rotation of the hub matches the differential.

The half shafts rotate with the rotation of differential and transfer the rotation to the wheel hub that would turn the rotation into wheel spinning. Moving the differential axis forward could cause to have better angle and a keeping the shorter length of half shafts to make them stronger.

### 6.6 TURN BUCKLES-

Turn buckles are the device for adjusting the tension of the chain as we have differential mount that holds the sprocket along with the differential so to give the chain the desired tension, we use turn buckles the turn buckle we use are made by conventional machining. Among the chain tensioning devices, the turn buckles are more reliable as they are less clunky and have easy machinability. We use the turn buckle to dial in the chain tension.

## 6.7 CONSTRAINTS AND CONSIDERATIONS

### DESIGN CONSTRAINTS

- Intake manifold with 20 mm restrictor.
- Intake manifold of suitable dimension such that it does not exceeds the envelope.
- Injector mount.
- Drive angle of half shafts.

### DESIGN CONSIDERATION

- The restrictor of intake manifold is provided with venturi, and the venturi has a bell-mouth inlet and a gradually diverging outlet to reduce pressure drop.
- The intake manifold’s overall design is made such that it accommodates inside the envelope.
- OEM injector mounted.
- Turnbuckle is provided such that it can be used to adjust the angle of the differential and the other parts of the driveline.

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## MACHINABILITY & COST CONSTRAINTS

Different parts of the powertrain are manufactured using different manufacturing processes. The manufacturing of intake manifold has been done using FDM as this mode of manufacturing is most efficient in terms of product finish and strength. The other modes of manufacturing are expensive compared to FDM and does not satisfies the requirements.
The other parts such as differential mount, and tripod housing is manufactured using CNC as they require high accuracy which is critical. The other parts such as turnbuckle and sprocket are manufactured using conventional machining process as it does not require very high level of accuracy.

## MACHINABILITY & COST CONSIDERATION

The different parts are manufactured using CNC or by conventional machining process according to the requirement. Differential mount and tripod housing are manufactured using CNC turning and milling and sprocket.

# 6.8 ENGINE

**Fig 6.3: KTM RC 390 Engine**

The KTM 390cc engine produces 43hp at 9500 rpm and delivers 35 Nm of torque. It's fast acceleration, low weight and small overall dimensions are the major factors for being chosen as the power unit of the car.

**Table 6.2: Engine Specifications**

| | |
|---|---|
| **DESIGN** | **Single Cylinder, 4 stroke, Spark Ignition, Water Cooled** |
| **DISPLACEMENT** | 375 cc |
| **BORE x STROKE** | 89mm X 60mm |

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| | |
|---|---|
| **COMPRESSION RATIO** | 12.6:1 |
| **MAXIMUM POWER** | 43.29hp @ 9500 rpm |
| **MAXIMUM TORQUE** | 35 Nm @ 7250 rpm |
| **IGNITION SYSTEM** | Digital Integrated Spark/Fuel Ignition |
| **CLUTCH** | Wet Multi-Plate |
| **GEAR BOX** | 6 Speed Constant Mesh |
| **FUEL SYSTEM** | Electric Fuel Injection |
| **LUBRICATION** | Wet Sump |
| **COOLING** | Water Cooled |

## 6.9 INTAKE MANIFOLD

### DESIGN & PERFORMANCE REQUIREMENT

The intake manifold should be designed such that it creates adequate positive air pressure, smooth air flow without much turbulence and minimum pressure drop after restrictor. The manifold should be manufactured such that it operates without getting damaged during and after its operational period. Material to be used should have enough strength for the required purpose.

### CONSTRAINTS & CONSIDERATIONS

- 20mm restrictor complying with rule.
- Overall dimension of manifold should be such that it accommodates inside the envelope according to rule.
- Selecting material with high strength.
- Minimal pressure drops.
- Minimal flow separation.

### MATERIAL SELECTION

Additively Manufactured Carbon Fiber Reinforced Composite (AMCFRC) is selected for the manufacturing of the intake manifold because of its high strength, durability and lightweight.

### CALCULATION

David Vizadr's rule states that "You should begin with a runner length of 17.8 cm for a 10,000-rpm peak torque location, from the intake opening to the plenum chamber. You add 4.3cm to the runner length for every 1000 rpm that you want the peak torque to occur before 10,000 rpm".

*For Peak Torque Location at 5000 rpm,*
Runner Length = 17.8 + (5 x 4.3)
= 39.3 cm = 393 mm

## FINAL DESIGN

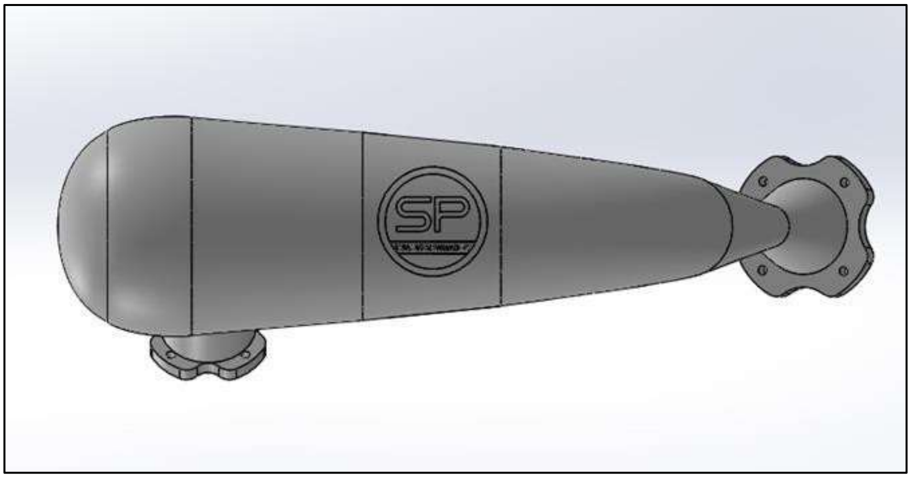


**Fig 6.4: CAD Model of Plenum Chamber**

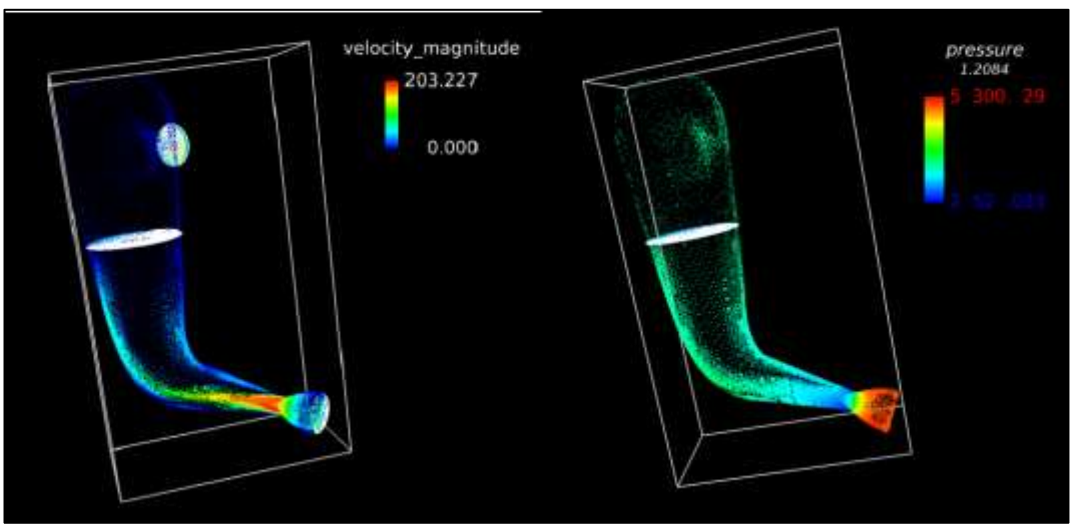
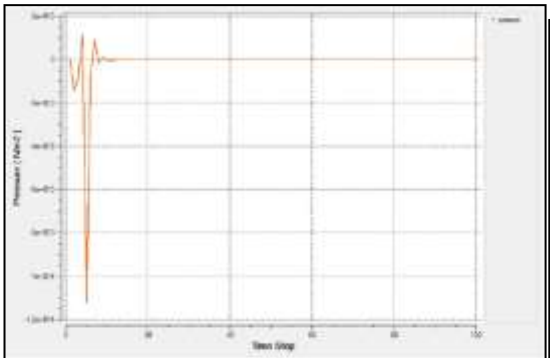



**Fig 6.5: Analysis of Plenum Chamber**


Pressure vs Time Graph of Plenum

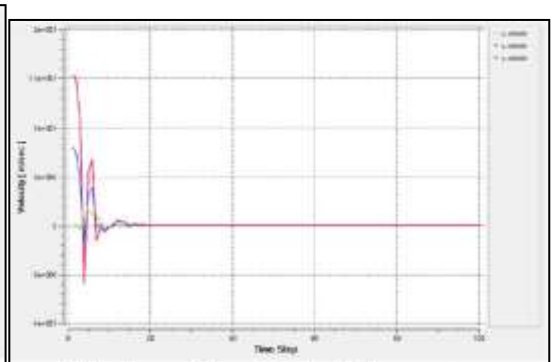

Velocity vs Time Graph of Plenum

Fig 6.6: Pressure and Velocity VS Time Graph

## 6.10 SPROCKET

### DESIGN & PERFORMANCE REQUIREMENT

Sprocket is the final element of the driveline where reduction occurs, so design of the sprocket should be such that the rate of acceleration and output torque should be decent and one should not be compromised for the other.

### CONSTRAINTS & CONSIDERATIONS

- Overall dimension should be compatible with the rear end packaging of the car.
- The number of teeth defines the level of reduction hence deciding the output characteristics.

### MATERIAL SELECTION

The sprocket is manufactured using Aluminum T7. The material is chosen because of its high strength, good machinability & light weight.

### MECHANICAL PROPERTIES

Table 6.3: Material Properties

| PROPERTIES | METRIC |
|---|---|
| TENSILE STRENGTH | 510 MPa |
| YIELD STRENGTH | 410 Mpa |
| MODULUS OF ELASTICITY | 70 Gpa |
| SHEAR MODULUS | 26Gpa |
| POISSON’S RATIO | 0.32 |

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**Calculation**

*Acceleration for sprocket with 45 teeth,*

Given:
Torque = 35 Nm
$1^{st}$ gear ratio = 12/32
Final drive ratio = 2.66x2.66 x (45/14)
No. of teeth in driven sprocket = 45
No. of teeth in driving sprocket = 14
Assumed driveline efficiency = 60% = 0.6
Rolling radius of wheels = 0.228 m
Vehicle gross weight = 220 kg

$$\text{Acceleration Thrust} = \frac{\text{Torque x Final drive ratio x Driveline efficiency}}{\text{Rolling radius of wheels}}$$

$$\text{Acceleration Thrust} = \frac{35 \times 2.66 \times 2.66 \times 3 \times 0.6}{0.228} = 1955.1 \text{ kg.m/s}^2$$

$$\text{Acceleration Rate} = \frac{\text{Acceleration Thrust}}{\text{Vehicle gross weight}} = \frac{19551.1}{220} = 8.88 \text{m/sec}^2$$

## 6.11 DESIGN ALTERNATIVES

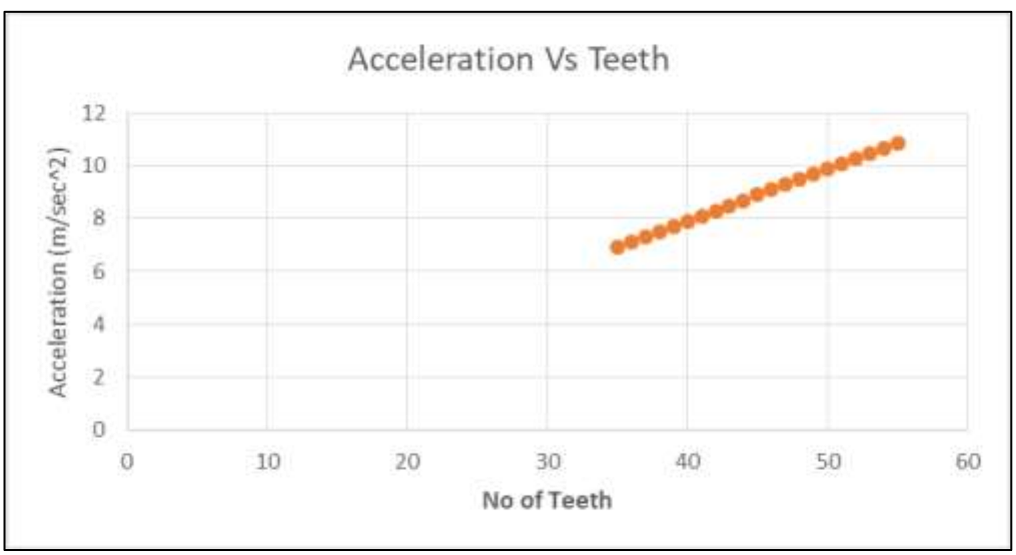


**Fig 6.7: Acceleration vs No of Teeth Graph**

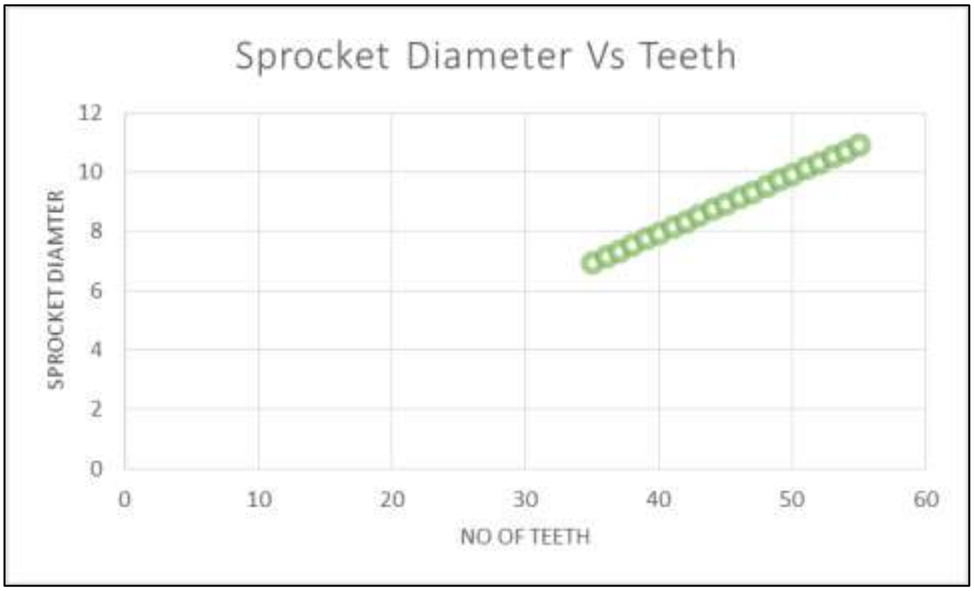


Fig 6.8: Sprocket Dia. vs No of Teeth

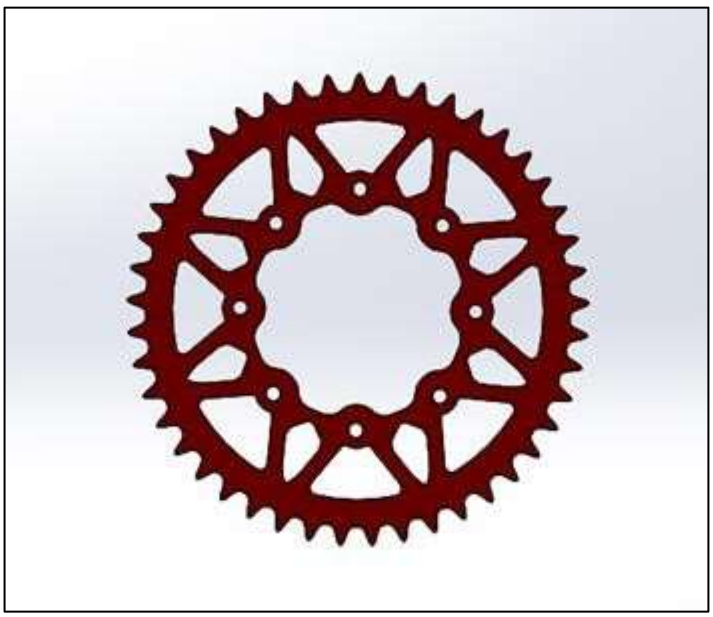

Fig 6.9: Sprocket CAD

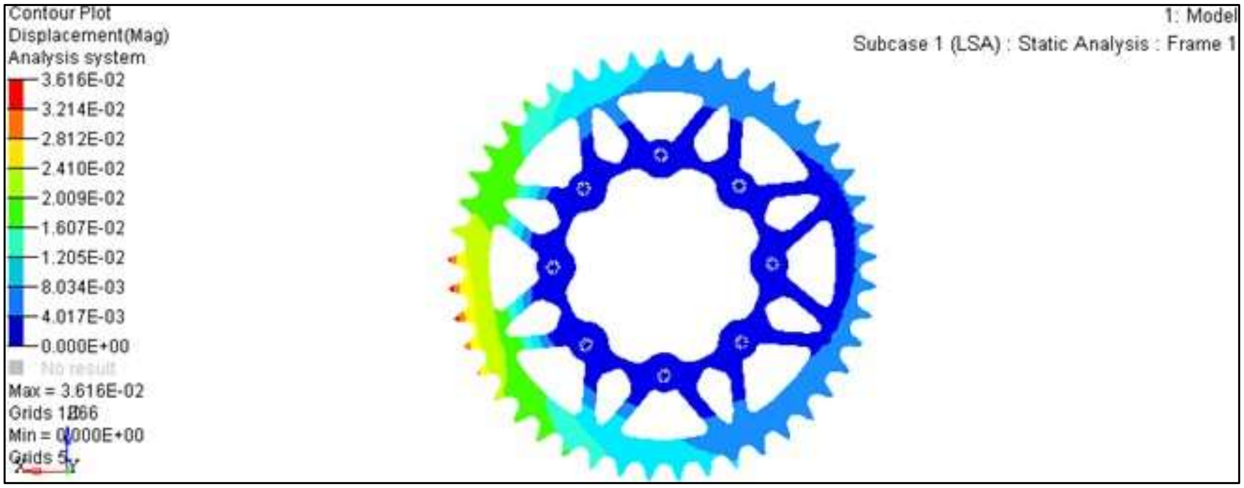


**Fig 6.10: Static Structure Analysis**

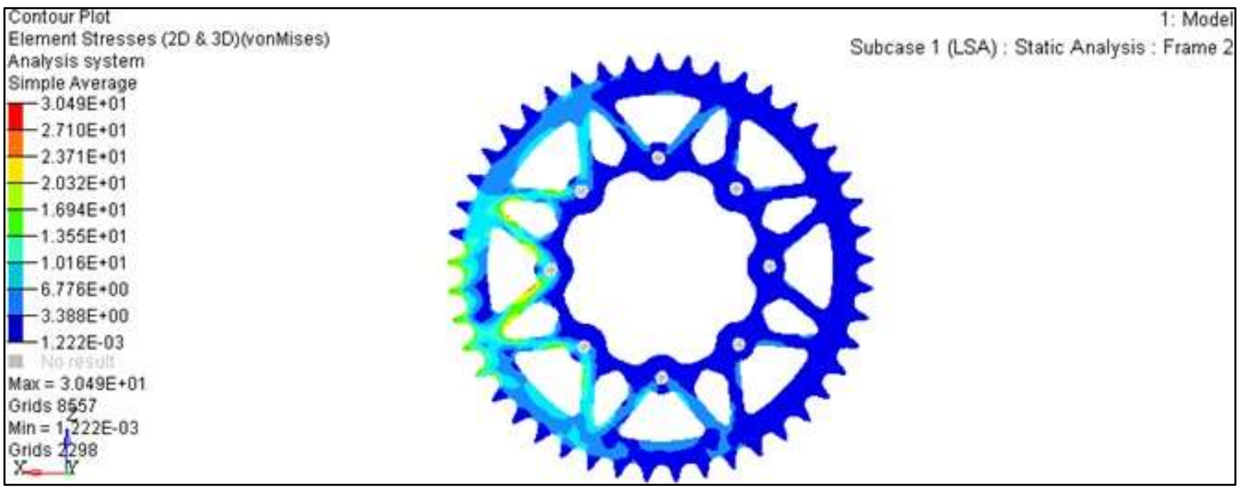


**Fig 6.11: Thermal Analysis**

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## 6.12 DIFFERENTIAL MOUNT

### DESIGN & PERFORMANCE REQUIREMENT

The differential mounts hold the differential in place while leaving it free to rotate. The mount is also adjustable so that the chain tension can be dialed in our main objective during the designing of the differential mount is to simplify design and manufacturing process and it lighter

### CONSTRAINTS & CONSIDERATIONS

Differential mount has two main functions namely:

1. Support for the differential bearings
2. Reduction in cumulative weight of the mount
3. Adjustable differential mount to dial in the chain tension

### MATERIAL SELECTION

The differential mount is manufactured using Aluminum T7. The material is chosen because of its high strength, good machinability & light weight.

### MECHANICAL PROPERTIES:

**Table 6.3: Material Properties**

| PROPERTIES | METRIC |
|---|---|
| TENSILE STRENGTH | 510 MPa |
| YIELD STRENGTH | 410 MPa |
| MODULUS OF ELASTICITY | 70 GPa |
| SHEAR MODULUS | 26 GPa |
| POISSON'S RATIO | 0.32 |

## DESIGN ANALYSIS

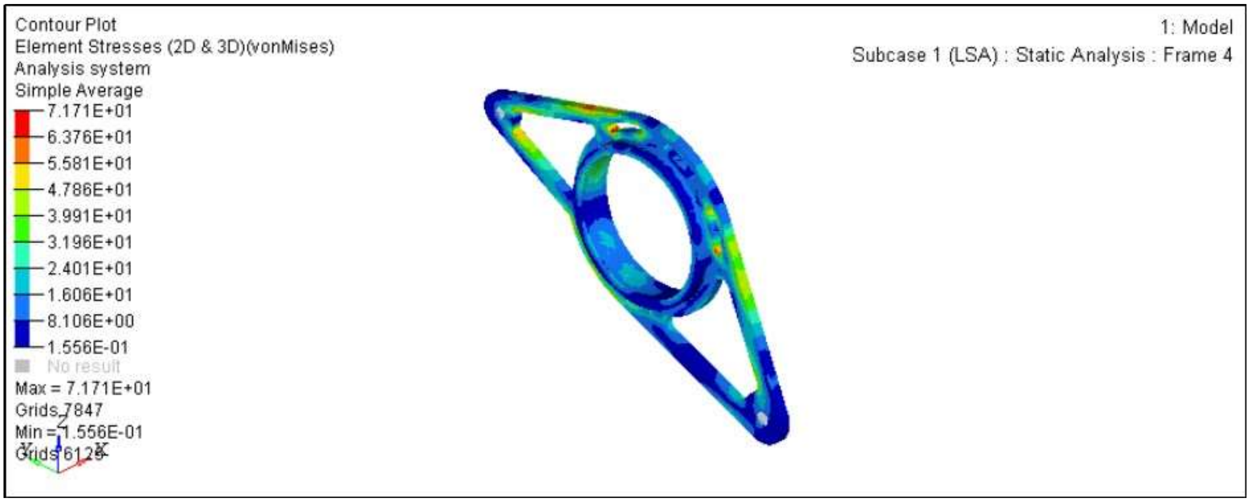



**Fig 6.12: Stress Analysis**

As per the above analysis the maximum stress that is induced in the differential mount is 71.71MPa.

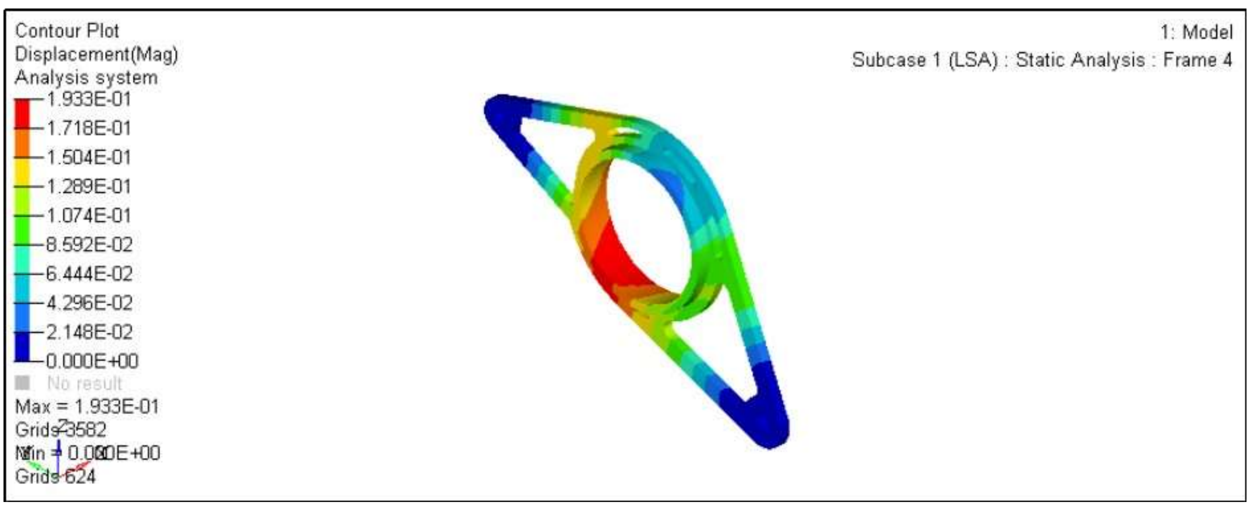



**Fig 6.13: Under dynamic condition**

As per the above analysis the maximum displacement is 0.1933 mm.

## 6.13 TURNBUCKLE

### DESIGN & PERFORMANCE REQUIREMENT

Turn buckles are the chain tensioning system for the car. We consider the turn buckles for our car as it not heavy or clunky and is reliable.

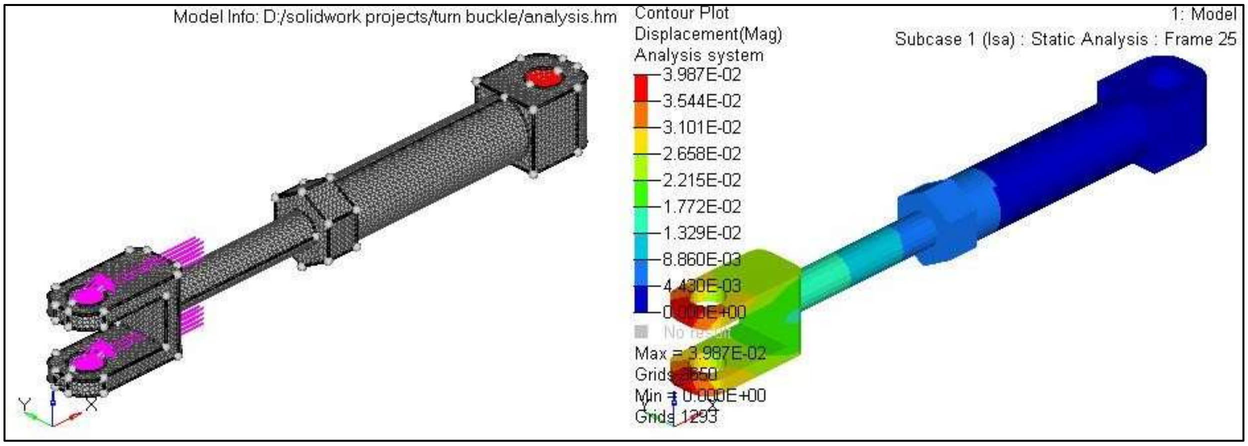



**Fig 6.14: Static Structure Analysis**

## 6.14 HALF SHAFTS & CV JOINTS

### DESIGN & PERFORMANCE REQUIREMENT

Half shafts are the axle connecting the transmission to the driven wheels. We are using the OEM shafts joined by the couplers for our vehicle. Tripod bearing are also the OEM whereas the we have the inboard as well as outboard tripod housing made out of aluminum by CNC turning.

### CONSTRAINTS & CONSIDERATIONS

- CV joints on the either end, allowing the driven wheels to maintain constant velocity
- Splines to transmit power between the differential, CV joints, shaft and wheel hub
- Suspension travel

## 6.15 ONE DIMESIONAL ENGINE SIMULATION AND MODELLING

### 6.15.1 DESIGN & PERFORMANCE REQUIREMENTS

With the selection of the 2014 KTM RC390 engine, performance targets can be set and Evaluated for the chosen platform. In general, the goal of any high performance FSAE powertrain should be to maximize the average power of the engine over the entire operating range with a 20 mm restrictor attached to its intake manifold. This will provide the vehicle with highest possible acceleration over the entire lap, thus will reduce the lap time. There are other goals such as extracting maximum power output and torque from engine by iterating the intake and exhaust manifold design. The operating range of the vehicle was determined 2000 RPM to 10000 RPM. This range was chosen on the basis of engine platform and driver feedback. With working range determined the performance needed can be obtained. Using data collected from previous year events and testing, it’s reasonable to expect a peak power output of 40hp from a naturally aspirated 373cc single cylinder engine. The goal was set to achieve similar power output from the engine with a 20 mm restrictor. This would provide a decent a power to weight ratio to the vehicle. To achieve this power goal Ricardo Wave simulation software was employed to predict the performance of the power unit, this also helped to determine the failure points and inefficiencies of the unit.

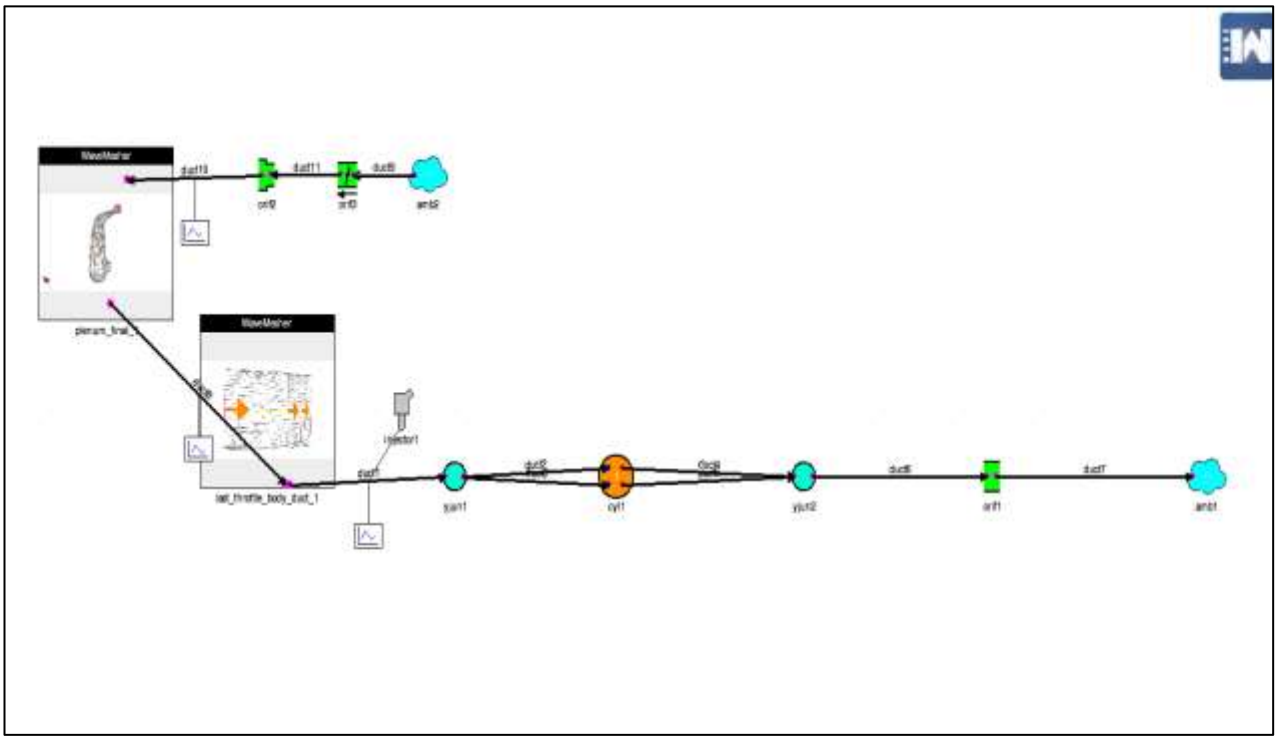


**Fig 6.15: Ricardo Wave 1D Model**

Using this 1-D simulation model the performance of the engine package can be predicted. This allows for quick and easy modifications to the engine design. Many different configurations of the engine can be tested to determine if they are worth pursuing before investing in the design. This will ultimately save time and money, both of which are scares in FSAE. All aspects of the engine design will we evaluated to reach the performance target. These include the intake manifold, fuel delivery system, exhaust system, and power transmission system.

## Modelling Parameters

- ***Engine Parameters***

| | |
|---|---|
| **No. of Cylinders** | **1** |
| **Bore** | **89mm** |
| **Stroke** | **60mm** |
| **Connecting Rod Length** | **105mm** |
| **Clearance Height** | **3.6167mm** |
| **Head Surf Multiplier** | **1.2** |
| **Piston Surface Multiplier** | **1.0** |
| **Compression Ratio** | **12.88** |

- ***Valves***

| | |
|---|---|
| **Reference Dia.** | **36mm (intake)/ 29mm (exhaust)** |
| **Valve Lift Profile** | **(from FMSCI report)** |

- ***Combustion Model***

| | |
|---|---|
| **Type** | **SI WIEBE** |
| **Location of 50% Burn Point** | **10º** |
| **Combustion Duration** | **1º/1000 RPM** |
| **Exponent in Wiebe Function** | **8** |
| **Profile Control** | **1.0** |

- ***Heat Transfer Model***

| | |
|---|---|
| **Type** | **WOSCHNI 1** |
| **Model Type** | **Original** |
| **Heat Transfer Multiplier (IVO)** | **1.0** |
| **Heat Transfer Multiplier (IVC)** | **1.0** |

- ***Y-Junction***

| | |
|---|---|
| **Type** | **Simple** |
| **Diameter** | **44** |
| **Wall Friction Multiplier** | **1.0** |
| **Heat Transfer Multiplier** | **1.0** |
| **Initial Pressure** | **1.0 bar** |
| **Initial Fluid Temperature** | **300.0 K** |
| **Wall Temperature** | **300.0 K** |

- ***Ducts***

| | | |
|---|---|---|
| **Shape** | **Circular** | **Circular** |
| **Left Dia.** | **20.0** | **35.1** |
| **Right Dia.** | **44.4** | **40.0** |
| **Discretization Length** | **30.0** | **30.0** |

| | | |
|---|---|---|
| **Overall Length** | **42.0** | **800.0** |
| **Bend Angle** | **0.0** | **0.0** |
| **Minimum Taper Angle** | **15.9454** | **0.175468** |

- ***<u>Injector</u>***

| | |
|---|---|
| **AFR** | **14.7** |
| **Distance from left end of the duct** | **6.0** |

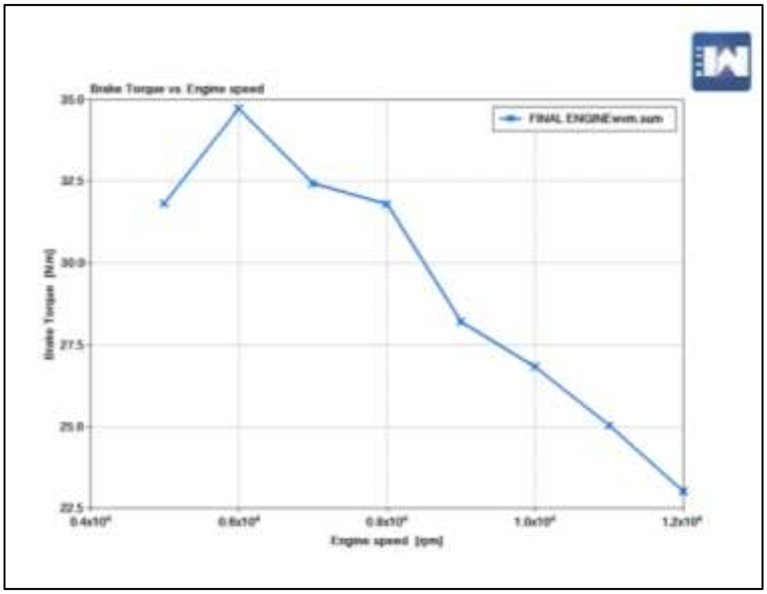



**Fig 6.16: Brake Torque Vs Engine Speed**

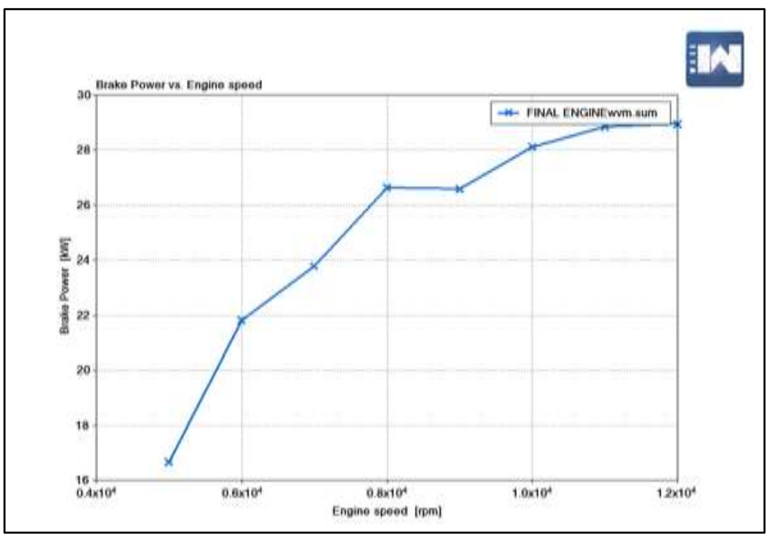



**Fig 6.17: Brake Power Vs Engine Speed**

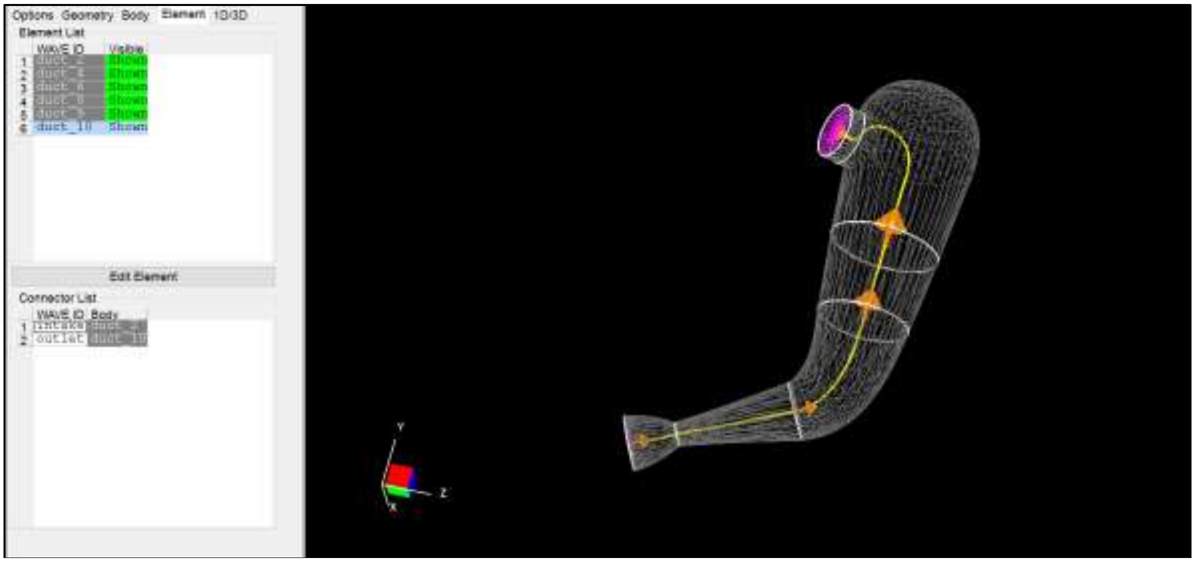



Fig 6.18: Simulation of Plenum

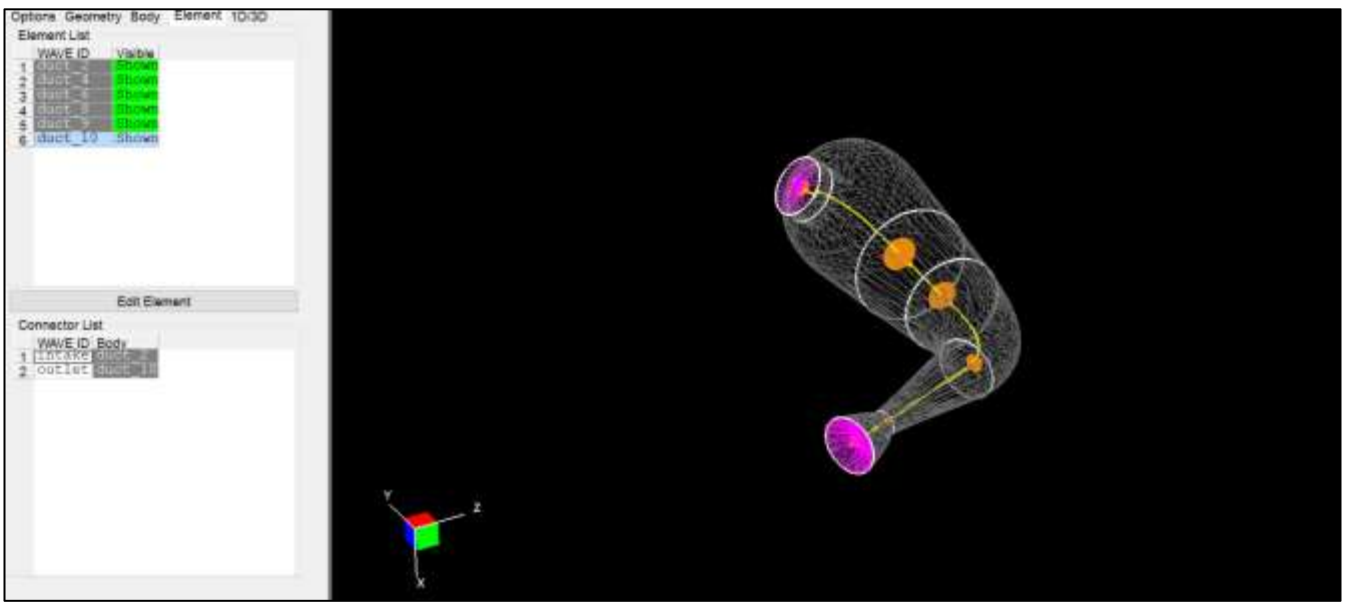
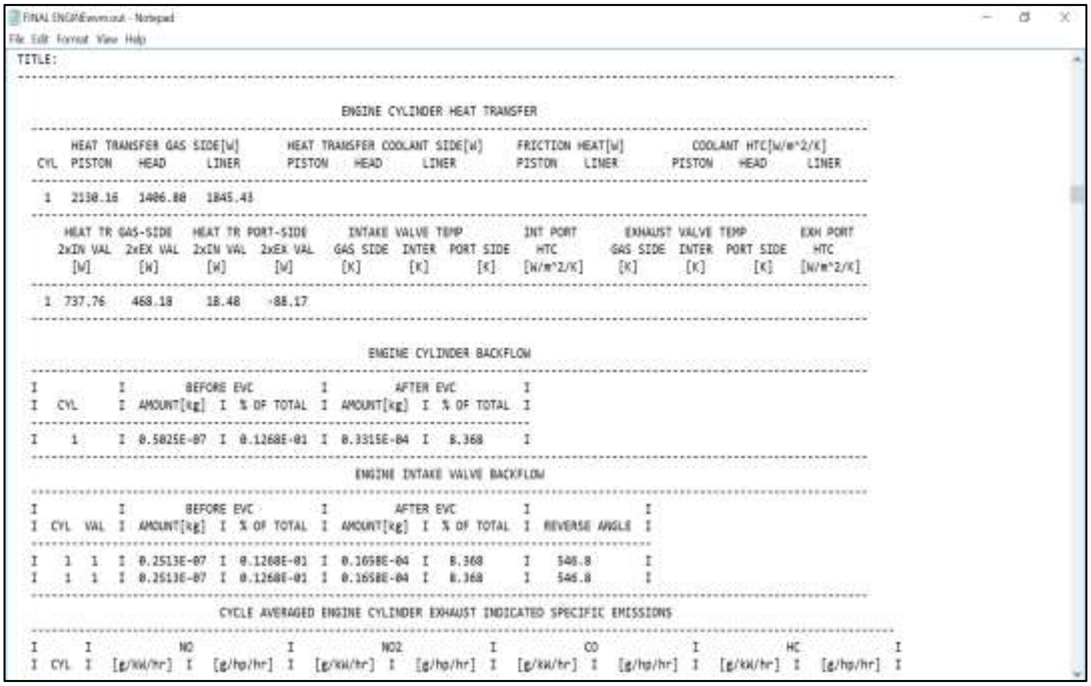
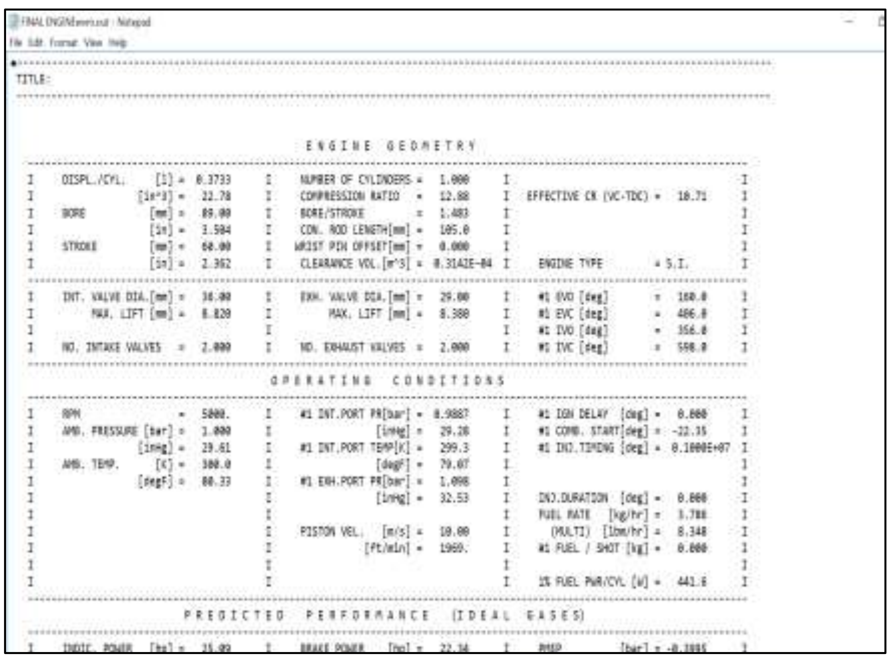



Fig 6.19: Simulation of Plenum



```
TITLE:
                         ENGINE CYLINDER HEAT TRANSFER
   HEAT TRANSFER GAS SIDE[W]     HEAT TRANSFER COOLANT SIDE[W]   FRICTION HEAT[W]      COOLANT HTC[W/m^2/K]
CYL  PISTON    HEAD    LINER     PISTON    HEAD    LINER         PISTON    LINER       PISTON    HEAD    LINER
 1  2130.16   1406.80  1845.43
   HEAT TR GAS-SIDE   HEAT TR PORT-SIDE    INTAKE VALVE TEMP        INT PORT    EXHAUST VALVE TEMP        EXH PORT
  2xIN VAL  2xEX VAL  2xIN VAL  2xEX VAL  GAS SIDE  INTER  PORT SIDE  HTC       GAS SIDE  INTER  PORT SIDE  HTC
    [W]       [W]       [W]       [W]       [K]      [K]      [K]   [W/m^2/K]    [K]      [K]      [K]   [W/m^2/K]
 1 737.76    468.18    18.48    -88.17
                          ENGINE CYLINDER BACKFLOW
I        I        BEFORE EVC          I        AFTER EVC          I
I  CYL   I  AMOUNT[kg]  I  % OF TOTAL  I  AMOUNT[kg]  I  % OF TOTAL  I
I   1    I  0.5025E-07  I  0.1268E-01  I  0.3315E-04  I  8.368       I
                          ENGINE INTAKE VALVE BACKFLOW
I        I        BEFORE EVC          I        AFTER EVC          I                 I
I CYL VAL I  AMOUNT[kg]  I  % OF TOTAL  I  AMOUNT[kg]  I  % OF TOTAL  I  REVERSE ANGLE  I
I  1   1  I  0.2513E-07  I  0.1268E-01  I  0.1658E-04  I  8.368   I   546.8   I
I  1   1  I  0.2513E-07  I  0.1268E-01  I  0.1658E-04  I  8.368   I   546.8   I
        CYCLE AVERAGED ENGINE CYLINDER EXHAUST INDICATED SPECIFIC EMISSIONS
I     I          NO            I          NO2           I          CO            I          HC            I
I CYL I [g/kW/hr] I [g/hp/hr] I [g/kW/hr] I [g/hp/hr] I [g/kW/hr] I [g/hp/hr] I [g/kW/hr] I [g/hp/hr] I
```

Fig 6.20: Heat Transfer Values



```
TITLE:
                              E N G I N E   G E O M E T R Y
I  DISPL./CYL.     [l] =  0.3733   I  NUMBER OF CYLINDERS =  1.000   I                               I
I               [in^3] =  22.78    I  COMPRESSION RATIO   =  12.88   I  EFFECTIVE CR (VC-TDC) =  10.71 I
I  BORE           [mm] =  89.00    I  BORE/STROKE         =  1.483   I                               I
I                 [in] =  3.504    I  CON. ROD LENGTH[mm] =  105.0   I                               I
I  STROKE         [mm] =  60.00    I  WRIST PIN OFFSET[mm] = 0.000   I                               I
I                 [in] =  2.362    I  CLEARANCE VOL.[m^3] = 0.3142E-04 I  ENGINE TYPE     = S.I.      I
I  INT. VALVE DIA.[mm] =  36.00    I  EXH. VALVE DIA.[mm] =  29.00   I  #1 EVO [deg]    =  180.0    I
I       MAX. LIFT [mm] =  8.820    I       MAX. LIFT [mm] =  8.380   I  #1 EVC [deg]    =  486.0    I
I                                  I                                 I  #1 IVO [deg]    =  356.0    I
I  NO. INTAKE VALVES   =  2.000    I  NO. EXHAUST VALVES  =  2.000   I  #1 IVC [deg]    =  598.0    I
                        O P E R A T I N G   C O N D I T I O N S
I  RPM                 =  5000.    I  #1 INT.PORT PR[bar] =  0.9887  I  #1 IGN DELAY  [deg] =  0.000  I
I  AMB. PRESSURE [bar] =  1.000    I               [inHg] =  29.28   I  #1 COMB. START[deg] = -22.35  I
I               [inHg] =  29.61    I  #1 INT.PORT TEMP[K] =  299.3   I  #1 INJ.TIMING [deg] = 0.1000E+07 I
I  AMB. TEMP.      [K] =  300.0    I               [degF] =  79.07   I                                I
I               [degF] =  80.33    I  #1 EXH.PORT PR[bar] =  1.098   I                                I
I                                  I               [inHg] =  32.53   I  INJ.DURATION [deg] =  0.000   I
I                                  I                                 I  FUEL RATE   [kg/hr] =  3.700   I
I                                  I  PISTON VEL.   [m/s] =  10.00   I  (MULTI)  [lbm/hr] =  8.348    I
I                                  I              [ft/min] = 1969.   I  #1 FUEL / SHOT [kg] =  0.000  I
I                                  I                                 I                                I
I                                  I                                 I  1% FUEL PWR/CYL [W] =  441.6   I
                   P R E D I C T E D   P E R F O R M A N C E   ( I D E A L   G A S E S )
I  INDIC. POWER   [kW] =  25.09    I  BRAKE POWER    [hp] =  22.34   I  BMEP          [bar] = -0.3895  I
```

Fig 6.21: Values of Heat

# CHAPTER 7: BRAKES AND SAFETY

## 7.1 DESIGN AND PERFORMANCE REQUIREMENTS

With a strategy to improve the whole braking efficiency of the vehicle during the dynamic conditions the following design and technical requirements were taken into account:

- Ability of the brake pedal to withstand a force of 2000N without any failure of the brake system or the pedal box.
- Determination of a suitable pedal ratio for the proper travel of push rod leading to sufficient amount of hydraulic pressure.
- Appropriate design of rotors to withstand the maximum possible force exerted by the calipers and to dissipate the maximum amount of heat at the time of barking.
- Brake biasing in order to enhance suitable load transfer during braking.

## 7.2 LITERATURE SURVEY

Brakes are the mechanical components which converts kinetic energy of the vehicle to heat energy. The whole braking system of the vehicle depends upon the proper pressure calculation, the forces exerted on the rotors by the caliper and also the reaction forces which is being imparted from the caliper to the upright during the braking condition. Correct pad material and proper alignment of the rotor and the pads is necessary in order to prevent steps and wear and tear of the pads, which ensures more pad and rotor life. Less and smooth pedal travel leading to required pressure exertion is one of the most important elements in relevance to the comfort of the driver.

### 7.3 CONSTRAINTS AND CONSIDERATIONS

Talking about the constraints and considerations, the above topic was subdivided into several general problems and the solutions that the team faced starting from the phase of conceptualization to the testing phase.

- **DESIGN CONSTRAINT**:

  For the brake pedal the first constraint was the 2000N force rule. In addition to that as the two master cylinders, used were positioned towards the driver, a number of iterations (line diagrams) were made to setup a proper angle for the master cylinder for the push rod travel.

- **MACHINABILITY AND COST CONSTRAINT:**

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The whole pedal box was manufactured in such a way in order to minimize the total cost of the water jet machining and MIG Welding involved. Similarly, use of suitable material for the rotors and also the machining process was quite necessary in order to enhance the total braking efficiency of the vehicle.

Looking at the above constraints the following considerations were taken into account to minimize complications and other major problems.

- **DESIGN CONSIDERATION:**

  The master cylinder was mounted with an angle of 4.5 degrees to the horizontal leading to a pedal ratio of 3.8:1 and a total push rod travel of 22mm without bleeding. Similarly, the rotors were drilled and slotted in unidirectional pattern to provide enough friction between the pads and suitable heat dissipation.

- **MACHINABILITY AND COST CONSIDERATION:**

  A suitable market research was done in order to select the calipers which can provide the highest amount of brake pressure at maximum acceleration and turnout to be most efficient. This concept led to the selection of Wilwood PS-1 fixed piston calipers. Maximum fillets and round edges were used in the pedals leading to proper stress flow while force exertion, leading to minimization of the total cycle time for the machine and ultimately the cost.

- **PACKAGING AND MOUNTING:**

  Proper space between the 3 pedals were given for proper driver comfort and pedal play during any dynamic event. Moreover, L clamps were welded on the front part of the ladder in order to mount the pedal box using fasteners to enhance ease of assembly/disassembly of the whole component.

- **MATERIAL SELECTION**

Material selection was the second procedure after the conceptualization phase. In order to select the appropriate material for pedals, base plate and the rotors the following parameters were taken into account:

1. **STRENGTH**

   Able to withstand high impact forces, pressure and also considerable amount of fatigue.

2. **MACHINABILITY**

One of the major factors as machining of hard materials like steel using CNC or other conventional manufacturing process may lead to increase in total cycle time and ultimately the cost.

3. **WELDING**

Supporting components like pedal clamps and master cylinder clamps were welded to the base plate which require high weldability of the material to provide proper strength during panic braking conditions.

4. **COST**

Obviously with a limitation in machining processes the cost of material has to be within the team's budget for a minimum final cost analysis of the whole component.

5. **WEIGHT**

The pedal box and its components are considered to be the sprung mass of the vehicle. With a reduction of sprung mass there can be an overall decrease in the overall mass of the vehicle for an efficient power to weight ratio. Similarly, rotors are considered to be the un-sprung mass of the vehicle and the similar concept is being applied to the un-sprung mass.

6. **CORROSION RESISTANT**

As the rotors are subjected to moisture every time the team cannot afford to use material which corrode within a specified testing time and may lead to damage of the pads. After looking at the above parameters **Al-T6-6061** was selected for the whole pedal box and its components and **SS304** for the rotors.

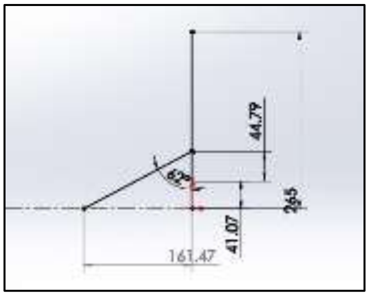
## 7.4 CALCULATIONS

### Pedal Ratio

The main parameter to be determined was the pedal ratio of brake pedal. Using line diagrams, the major factor kept in mind was the mounting of the master cylinder and the piston travel in the master cylinder. A pedal ratio of 3.8:1 was finally calculated with the master cylinder mounted towards the driver at an angle of 5 degrees to the horizontal. A number of iterations using line diagrams in SolidWorks were performed to meet the required pedal ratio and also the piston travel without bleeding condition.

## ITERATION 1

**Fig 7.1: Master Cylinder towards front bulkhead**




The above line diagram was the first iteration involved with the master cylinder positioned towards the front bulkhead at an angle of 28 degrees with the horizontal. Positioning the master cylinder in the above-mentioned position was able to provide the sufficient piston travel, but a loophole was involved. This position would make the front part of the chassis much longer and also violating the rule T2.20.2.

## ITERATION 2

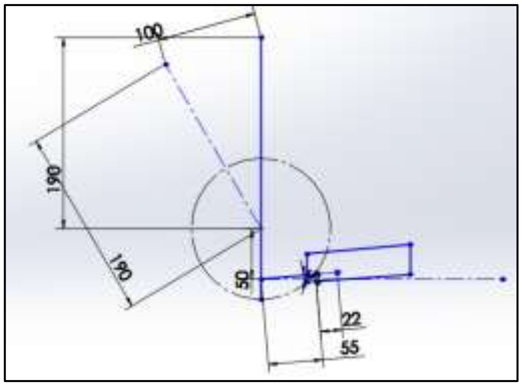



**Fig 7.2: Master Cylinder towards driver**

The above line diagram was the 2nd iteration and the final iteration involved. At first the master cylinder was positioned parallel to the pedal box which resulted only 15mm of push rod travel without bleeding which was unacceptable. The angle was then increased to 5 degrees which made a full piston travel and also made the pedal travel for the driver smoother and more efficient. The mounting point for the piston was made 50mm below the pivot point of the pedal.

After knowing the above values, the final pedal ratio was calculated given by:

Pedal Ratio = Distance between the face of pedal to the pedal pivot Distance between pedal pivot to the piston mounting= 19050= 3.8

The final pedal ratio for the brake pedal with the master cylinder mounted was **3.8:1.**

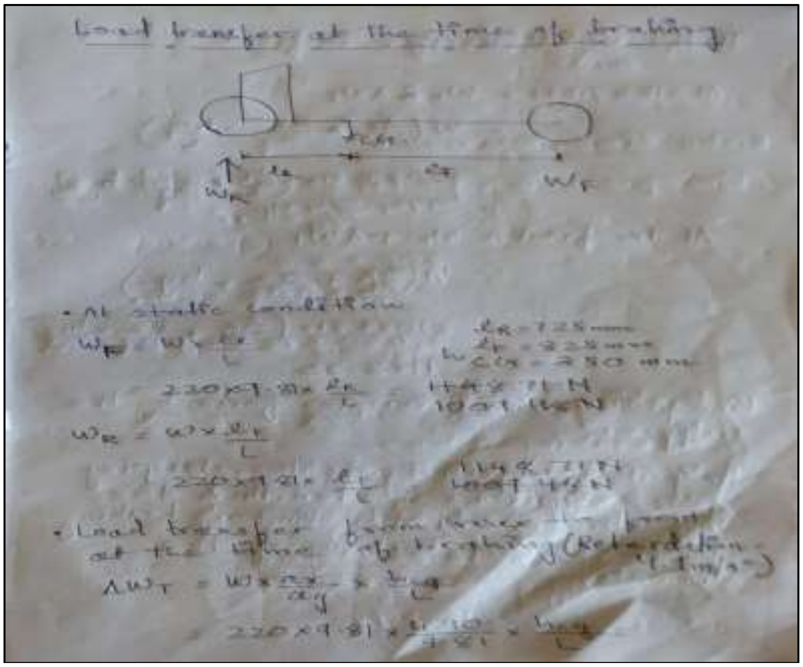
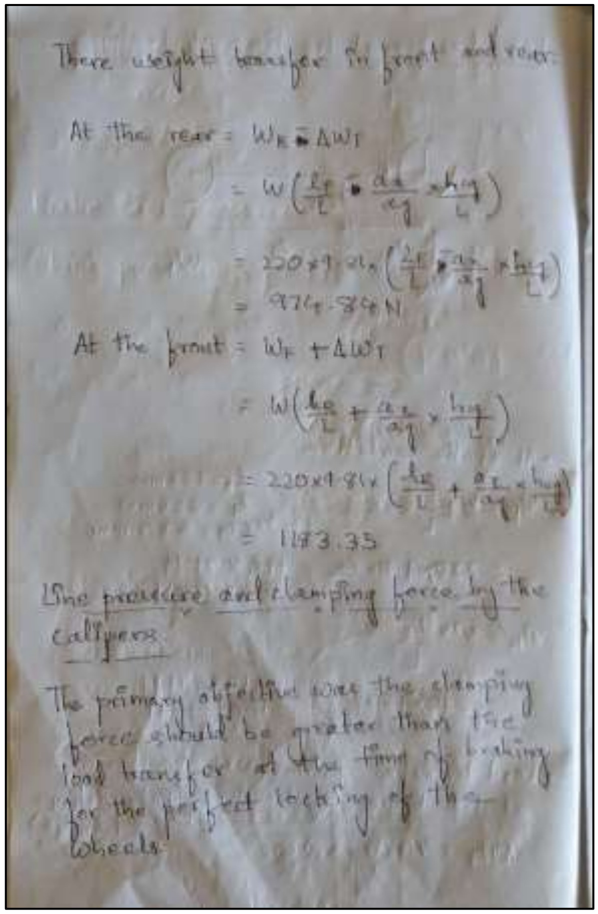



**Fig 7.3: Calculations for Pedal Ratio**

**ISSN: 2278-0181**


There weight transfer to front and rear:

At the rear = $W_R - \Delta W_T$

$= W\left(\frac{l_f}{L} - \frac{a_x}{g} \times \frac{h_g}{L}\right)$

$= 220 \times 9.81 \times \left(\frac{l_f}{L} - \frac{a_x}{g} \times \frac{h_g}{L}\right)$

$= 974.854$ N

At the front = $W_F + \Delta W_T$

$= W\left(\frac{l_r}{L} + \frac{a_x}{g} \times \frac{h_g}{L}\right)$

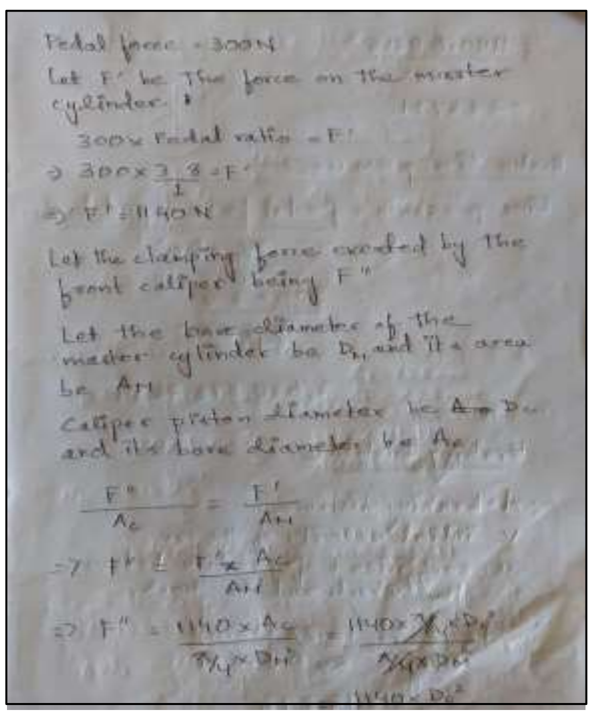
$= 220 \times 9.81 \times \left(\frac{l_r}{L} + \frac{a_x}{g} \times \frac{h_g}{L}\right)$

$= 1173.35$

The pressure and clamping force by the calipers

The primary objective was the clamping force should be greater than the load transfer at the time of braking for the perfect locking of the wheels

**Fig 7.4: Calculations for Pedal Ratio**


Pedal force = 300 N

Let F' be the force on the master cylinder

300 × Pedal ratio = F'

$\Rightarrow 300 \times \frac{3.8}{1} = F'$

$\Rightarrow F' = 1140$ N

Let the clamping force exerted by the front caliper being F''

Let the bore diameter of the master cylinder be $D_1$, and its area be $A_M$

Caliper piston diameter be $D_c$ and its bore diameter be $A_c$

$\frac{F''}{A_c} = \frac{F'}{A_M}$

$\Rightarrow F'' = \frac{F' \times A_c}{A_M}$

$\Rightarrow F'' = \frac{1140 \times A_c}{\pi/4 \times D_M^2} = \frac{1140 \times \pi/4 \times D_c^2}{\pi/4 \times D_M^2}$

$1140 \times D_c^2$

**Fig 7.5: Calculations for Pedal Ratio**

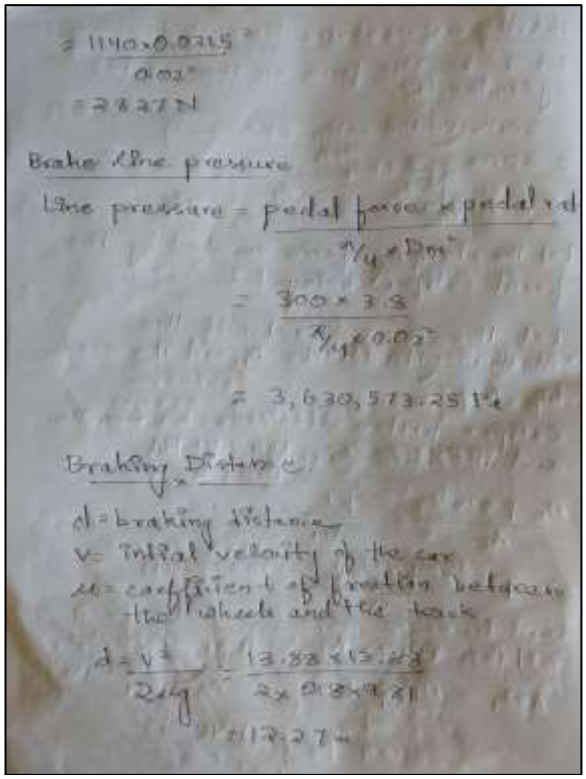


Fig 7.6: Calculations for Braking Distance

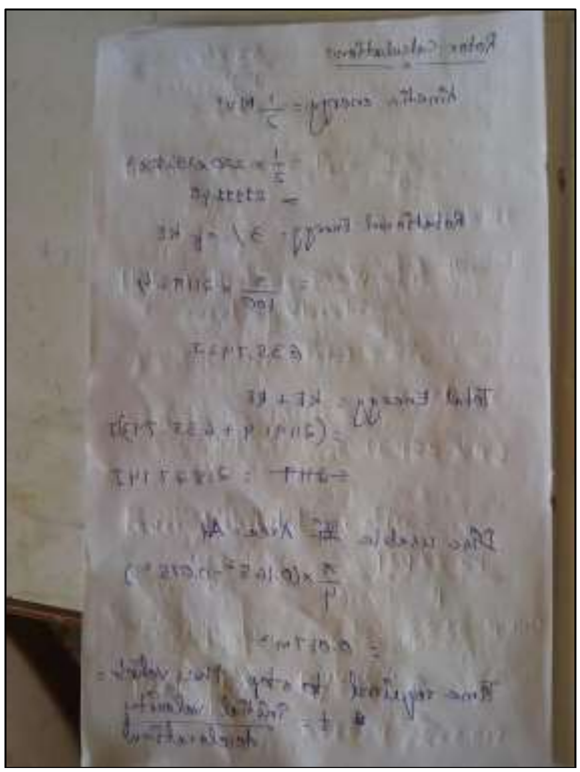


Fig 7.7: Calculations for Braking Distance

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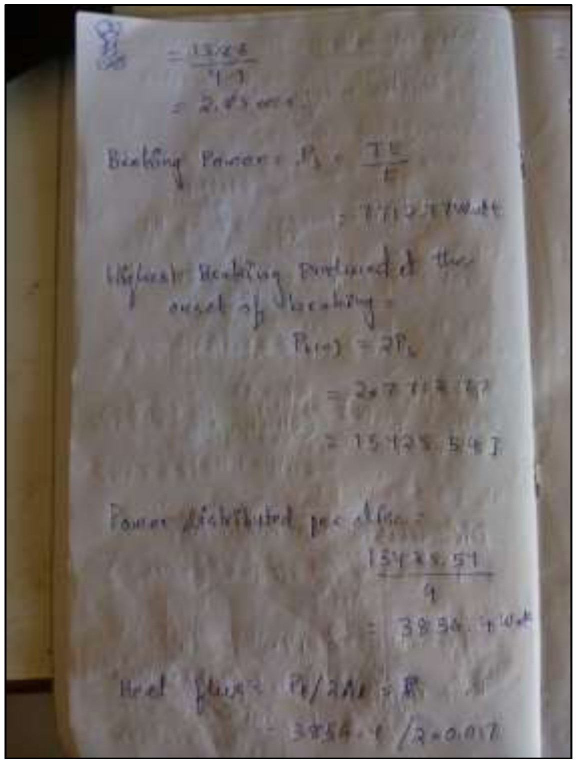


Fig 7.8: Calculations for Braking Power

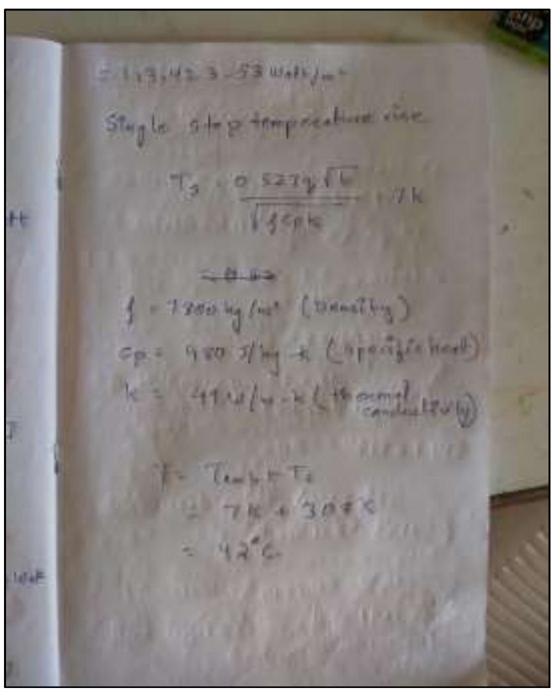


Fig 7.9: Calculations for Braking Power

## 7.6 FINAL DESIGN

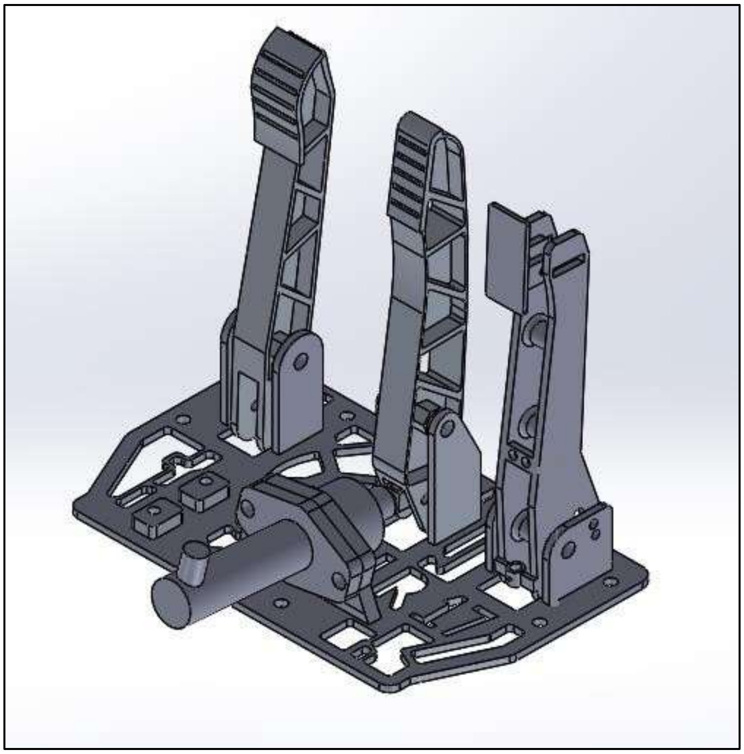


**Fig 7.10: Final Pedal Box Assembly**

## 7.7 SAFETY

**Fig 7.11: 5-point harness**

**Fig 7.12: 6-point harness**

- 6-point harness is better distributing the shock compare to 5 point.
- Justify rule of 30-degree angle.

ISSN: 2278-0181

# CHAPTER 8: BODYWORKS

## 8.1 DESIGN AND PERFORMANCE PARAMETERS

To develop a body taking into account several factors to develop an optimum body model as a final result. The factors are:

- Lightweight
- Less drag
- Easy to dismantle
- Aesthetics

## 8.2 LITERATURE SURVEY

The radiator is housed inside the side-pod and it should be designed to act as air-duct. The entrance has a rounded edge to prevent flow separation.

## 8.3 CONSTRAINTS AND CONSIDERATION

### DESIGN CONSTRAINT:

- The nose cone is designed such that it should accommodate the IA assembly, easy to dismantle, should not block the driver's field of view and at the same time it should be visually appealing and comply with the rules- minimum 38mm radius at the front, front face angle should be more than 45 degrees.
- The side pods are designed to accommodate the radiator and exhaust and also to aid in cooling. It does not extend further than vertical plane touching the outboard face of the front and rear wheel/tire.
- Dashboard is designed to accommodate all the switches and displays with an ease to operate. It should not hinder the driver at the time of ingress and egress.

### MATERIAL SELECTION

There are two main types of material used in common practice, which are glass fibre and carbon fibre composite. We opted for carbon fibre because it has more strength to weight ratio, easier to work on, has a better surface finish and glass fibre was used on last year vehicle but did not presented the expected characteristics.
So, the material selected was CARBON FIBRE of 2x2 twill.

**Table 8.1: Material Properties of CFRP**

| Density(g/cm3) | Filament Diameter(µm) | Tensile Strength(MPa) | Tensile Modulus(GPa) | Elongation (%) | Sizing |
|---|---|---|---|---|---|
| 1.8 | 7 | 3450 | 230 | 1.5 | Epoxy Compatible |

Epoxy type resin is used to get better bond strength.

**Table 8.2: Material Properties of Resin**

| Product Code | Viscosity At 25° C - (Unit: MPas) | Density At 25°C - (Unit: G/Cc) | Gel Time At 25° C - (Unit: Minutes) | Full Cure Time At 25 C - (Unit: Hours) |
|---|---|---|---|---|
| Hinpoxy C – Resin | 9000 - 12000 | 1.15 - 1.20 | 120 | 24 |
| Hinpoxy C - Hardener | ? 50 | 0.94 - 0.95 | 120 | 24 |

## 8.4 DESIGN ALTERNATIVES

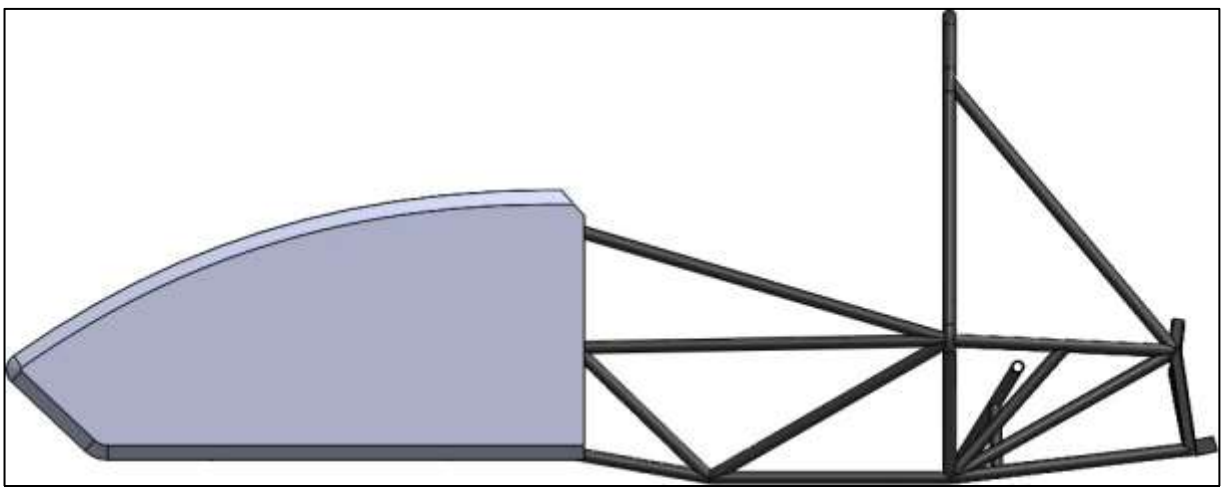


Fig 8.1: CAD of Nose

This design was rejected because it was bulky, weighs more and it was hard to install and dismantle. Also, the air resistance was slightly higher.

**Table 8.3: Physical Properties of CFRP**

| Goal Name | Unit | Value | Averaged Value | Minimum Value | Maximum Value | Progress [%] |
|---|---|---|---|---|---|---|
| SG Normal Force (X) 1 | [N] | -28.92961924 | -28.86004136 | -29.46174743 | -28.51089259 | 100 |
| SG Normal Force (Y) 1 | [N] | -8.522741878 | -9.586898247 | -10.33456294 | -8.168403045 | 100 |

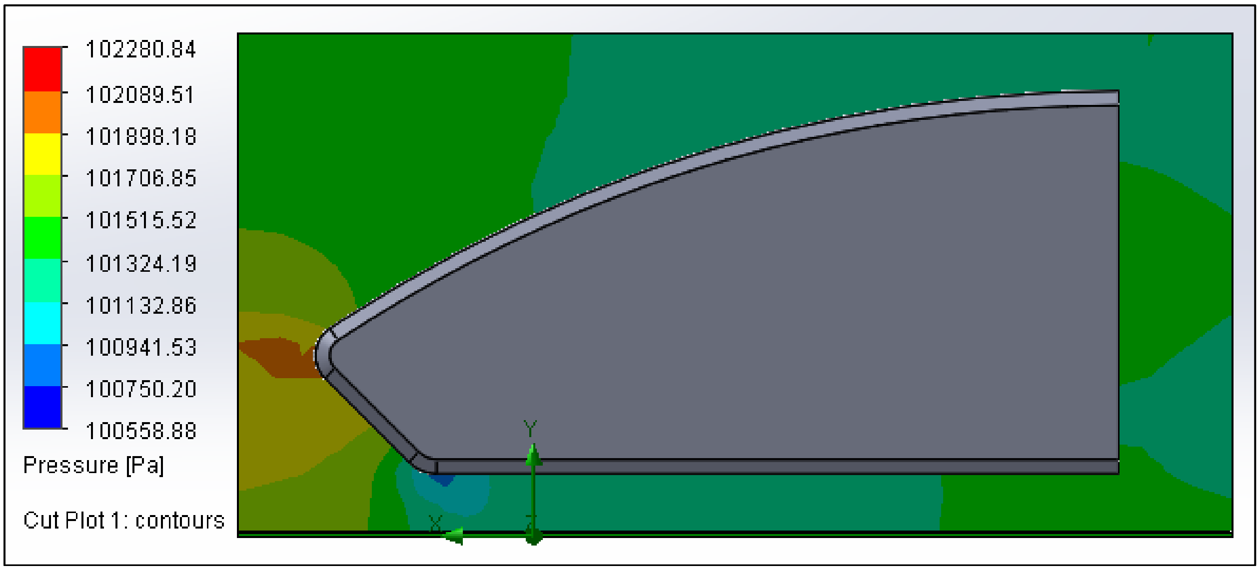



**Fig 8.2: CAE of Nose**

## FINAL DESIGN

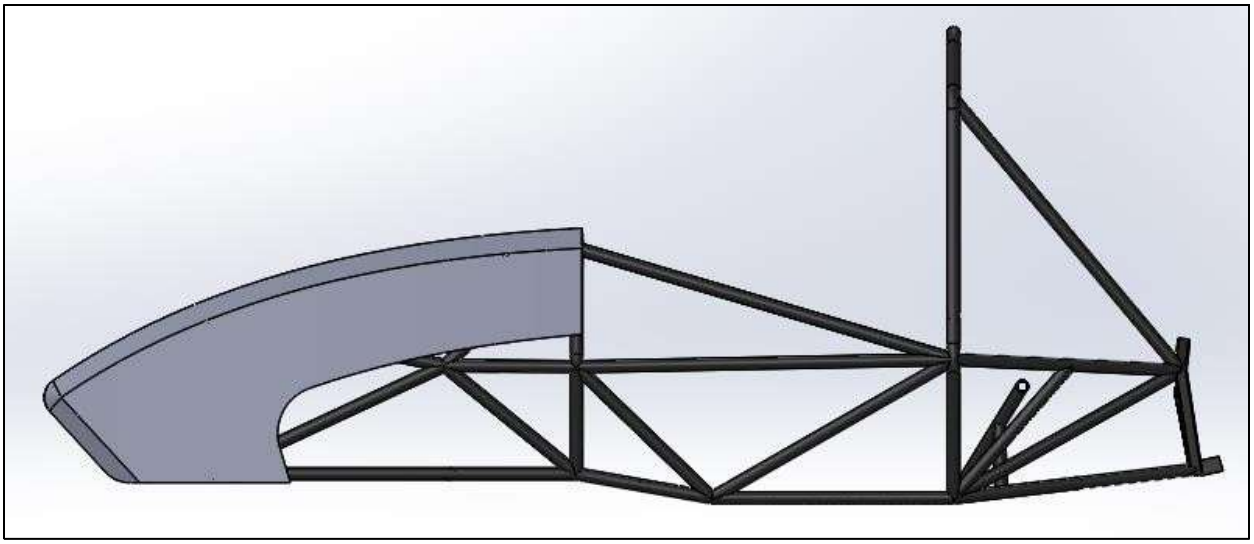


**Fig 8.3: Final Nose Design CAD**

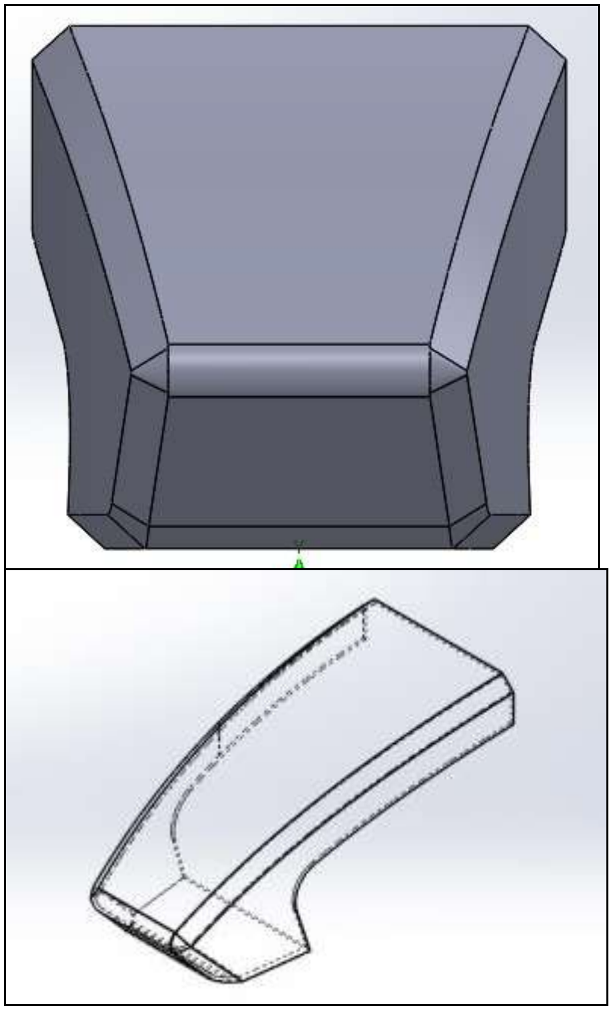
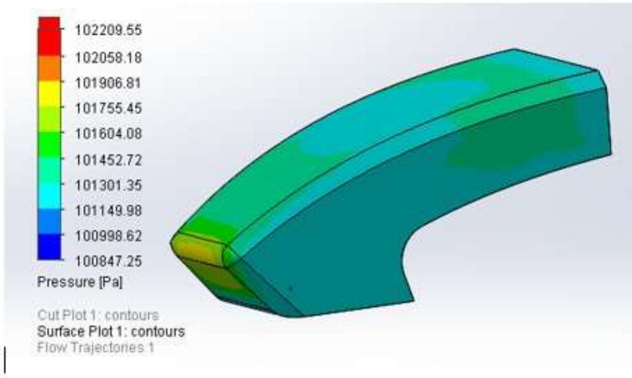


**Fig 8.4: Front View of Nose**




| Goal Name | Unit | Value | Averaged Value | Minimum Value | Maximum Value | Progress [%] |
|---|---|---|---|---|---|---|
| SG Normal Force (X) 1 | [N] | -14.9733372 | -15.03515187 | -15.15577777 | -14.97223664 | 100 |
| SG Normal Force (Y) 1 | [N] | -14.7074766 | -14.51744384 | -14.70747657 | -14.26445948 | 100 |

**Fig 8.5: Nose Analysis**

As the air drag resistance was slightly lower for this design, it is easier to install, looks clean, it was selected as final design.

## 8.5 DASHBOARD:

## 8.6 SIDE PODS:

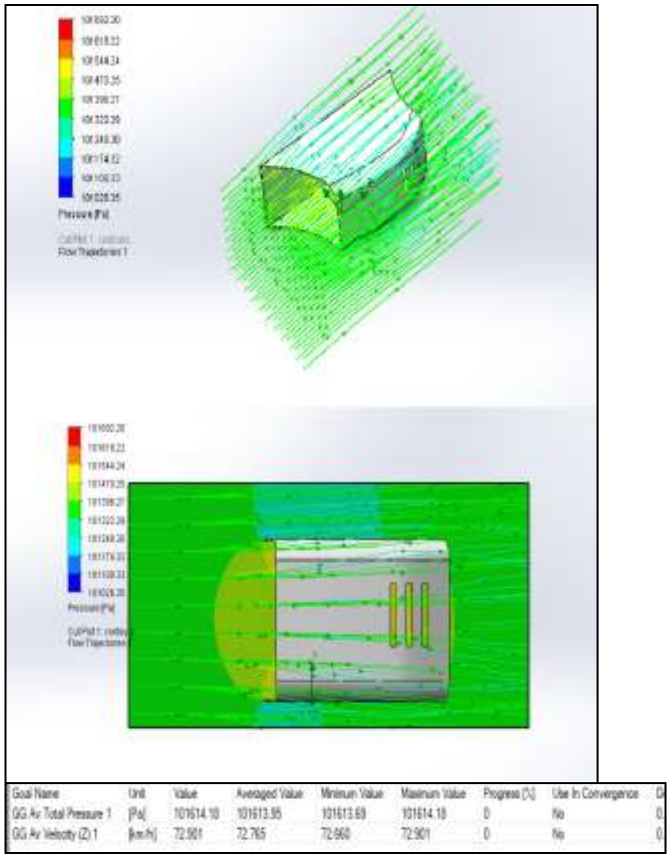



**Fig 8.7: Side Pod Analysis**

# CHAPTER 9: ELECTRICAL & ELECTRONICS

## 9.1 DESIGN AND PERFORMANCE REQUIREMENT

- The main motive of the designing of the electrical and electronics of our car is improved performance, reliability, ease of troubleshooting, better packaging, and giving proper feedback and control of the vehicle to the driver.
- We have placed proper switches for fuel pump, radiator fan and Launch Control has also been placed on the Car’s dash, giving Major controls to the Driver.
- An Engine Temperature Indicator has also been placed on the Dash which gives the driver an idea of how much the engine is heated, and also when to turn on the radiator Fan.
- The custom Harness for the Engine provides better packaging, because it only has the connection to the Critical Components of the engine. The not so important parts of the harness have been removed such as rollover sensors, side stand sensor, lights and etc.


**Fig 9.1: RD 1401**

- We have used Race Dynamics Standalone ECU RD 1401 which gives us access to the fuel maps and the ignition timing of the engine and we could change it according to the driver's wish.

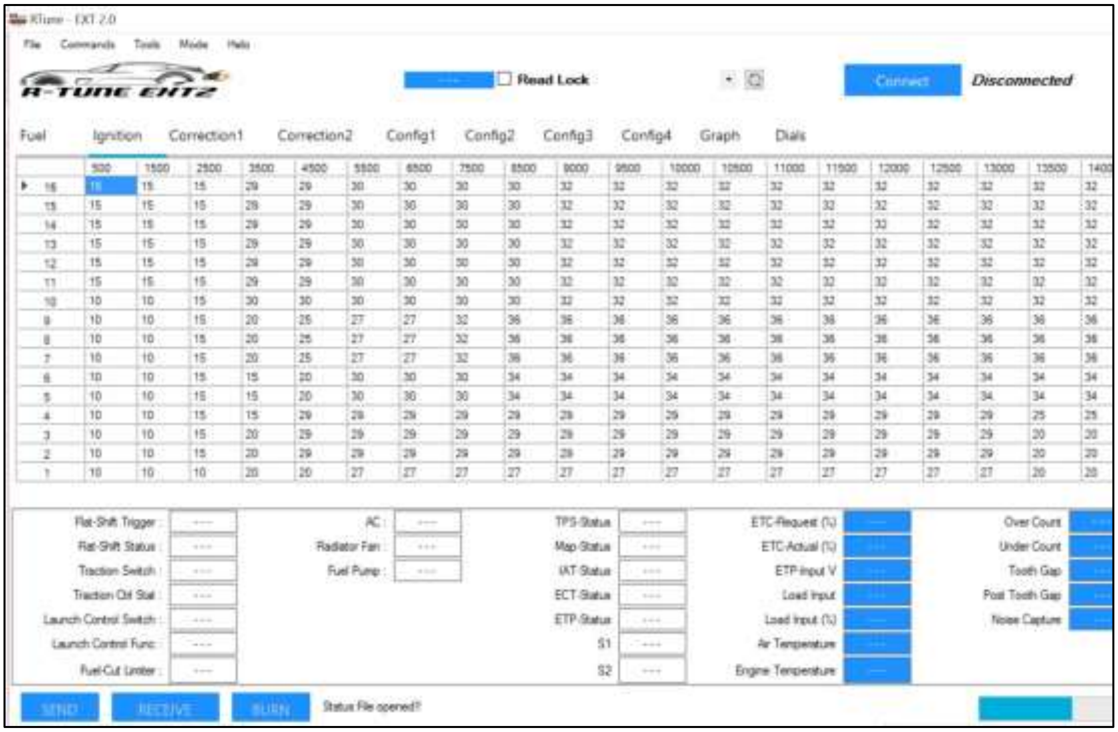



Fig 9.2: MAP Values

- We are able to manipulate these data with the help of R-Tune ETX2. It is an application software which acts as an interface between us as the RD_1401. It helps us to read and write data and instruction to the ECU.

## 9.2 ENGINE HARNESS

- **The engine harness consists of these following.**
- **Igniter**
- **CDI unit and spark plug**
- **CKP sensor**
- **fuel injector sensor**
- **Throttle position sensor**

## 9.3 MATERIAL SELECTION

- The ECU has been selected by keeping performance, reliability, and affordability in mind.

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- The Custom Engine harness provides us better packaging.
- Nearly all of the electrical components were sourced from the market and these were combined to form a working electrical circuit.

The wires used in the circuit are of havells which offer insulation with (HR) Heat resistant along with Flame Retardant properties which are suitable to bear a temperature up to 85°C whereas ordinary PVC is suitable only up to 70°C.

# CHAPTER 10: COST ANALYSIS

## 9.1 SMART MANUFACTURING TECHNIQUES AND COST REDUCTION

SMART MANUFACTURING TECHNIQUES USED THIS SEASON AND COST REDUCTIONS DUE TO THAT

| SPROCKET | | | | | | | |
|---|---|---|---|---|---|---|---|
| 2016-17 | | | | 2017-18 | | | |
| **Material Name-** Mild Steel | **Material Cost(per kg= ₹27** | ₹ | 614.43 | **Material Name-** Aluminium T7 | **Material Cost(per kg= ₹273)** | ₹ | 489.35 |
| **Machining Proccess-** Laser cutting, Drilling, Grinding | **Machining Cost** | ₹ | 962.00 | **Machining Proccess-** Waterjet Cut | **Machining Cost** | ₹ | 488.00 |
| | **Proccess Cost-** | | | | **Proccess Cost-** | | |
| | Assemble | ₹ | 42.60 | | Assemble | ₹ | 42.60 |
| | Drilled Holes | ₹ | 49.42 | | Drilled Holes | ₹ | - |
| | **Fasteners Cost** | | | | **Fasteners Cost** | | |
| | Fastners(*12) | ₹ | 427.20 | | Fastners(*8) | ₹ | 149.76 |
| | Tighten | ₹ | 130.00 | | Tighten | ₹ | 130.00 |
| | **TOTAL** | ₹ | **2,225.65** | | **TOTAL** | ₹ | **1,299.71** |

| BRAKE ROTOR | | | | | | | |
|---|---|---|---|---|---|---|---|
| 2016-17 | | | | 2017-18 | | | |
| **OEM PART COST= ₹1865/pc** | | | | | | | |
| **Material Name-** Mild Steel | **Material Cost** | ₹ | - | **Material Name- Stainless Steel** | **Material Cost(per kg=₹146.2** | ₹ | 109.69 |
| **Machining Proccess-** | **Machining Cost** | | | **Machining Proccess-** Waterjet Cut | **Machining Cost** | ₹ | 1,352.65 |
| | **Proccess Cost-** | | | | **Proccess Cost-** | | |
| | Assemble | ₹ | 338.00 | | Assemble | ₹ | 338.00 |
| | Drilled Holes | ₹ | 24.92 | | Drilled Holes | ₹ | - |
| | **Fasteners Cost** | | | | **Fasteners Cost** | | |
| | Fastners(*4) | ₹ | 71.52 | | Fastners(*4) | ₹ | 71.52 |
| | Tighten | ₹ | 85.00 | | Tighten | ₹ | 85.00 |
| | **TOTAL** | ₹ | **2,384.40** | | **TOTAL** | ₹ | **1,956.86** |

**Fig 10.1: Cost Comparision of Smart Manufacturing Techniques used**

| ROCKER | | | | | | | |
|---|---|---|---|---|---|---|---|
| 2016-17 | | | | 2017-18 | | | |
| **Material Name- Aluminum Premium** | **Material Cost(per Kg= ₹273)** | ₹ | 757.39 | **Material Name- Aluminum T7** | **Material Cost(per kg= ₹273)** | ₹ | 346.43 |
| **Machining Proccess-** CNC Milling | **Machining Cost** | ₹ | 987.67 | **Machining Proccess-** CNC Milling | **Machining Cost** | ₹ | 514.13 |
| | **Proccess Cost-** | | | | **Proccess Cost-** | | |
| | Assemble | ₹ | 85.00 | | Assemble | ₹ | 85.00 |
| | | | | | | | |
| | **Fasteners Cost** | | | | **Fasteners Cost** | | |
| | Fastners(*3) | ₹ | 58.64 | | Fastners(*3) | ₹ | 58.64 |
| | Tighten | ₹ | 48.75 | | Tighten | ₹ | 48.75 |
| | **TOTAL** | ₹ | **1,937.45** | | **TOTAL** | ₹ | **1,052.95** |

**Fig 10.2: Cost Comparision of Smart Manufacturing Techniques used**

| AIR INTAKE ASSEMBLY | | | | | | | |
|---|---|---|---|---|---|---|---|
| 2016-17 | | | | 2017-18 | | | |
| **Material Name-** Plastic ABS with CFRP | **Material Cost (₹214.50+ ₹13000)** | ₹ | 14,475.00 | **Material Name-** Additively manufactured carbon fibre composite | **Material Cost(per kg= ₹1300(** | ₹ | 12,753.00 |
| **Machining Proccess-** Rapid Prototyping | **Machining Cost** | ₹ | 3,753.30 | **Machining Proccess-** Rapid Prototyping | **Machining Cost** | ₹ | 2,222.48 |
| | **Proccess Cost-** | | | | **Proccess Cost-** | | |
| | Assemble | ₹ | 42.60 | | Assemble | ₹ | 42.60 |
| | | | | | | | |
| | **Fasteners Cost** | | | | **Fasteners Cost** | | |
| | Fastners(*4) | ₹ | 142.40 | | Fastners(*4) | ₹ | 142.40 |
| | Tighten | ₹ | 130.00 | | Tighten | ₹ | 130.00 |
| | **TOTAL** | ₹ | **18,543.30** | | **TOTAL** | ₹ | **15,163.22** |

| UPRIGHT | | | | | | | |
|---|---|---|---|---|---|---|---|
| **2016-17** | | | | **2017-18** | | | |
| **Material Name- Aluminum Premium** | **Material Cost(per Kg= ₹273)** | ₹ | 1,784.45 | **Material Name- Aluminum T7** | **Material Cost(per kg=₹273)** | ₹ | 1,104.00 |
| **Machining Proccess-** CNC Milling | **Machining Cost** | ₹ | 3,896.10 | **Machining Proccess-** CNC Milling | **Machining Cost** | ₹ | 2,566.45 |
| | **Proccess Cost-** | | | | **Proccess Cost-** | | |
| | Assemble | ₹ | 42.50 | | Assemble | ₹ | 42.50 |
| | Drilled Holes(x6) | ₹ | 176.00 | | | | |
| | **Fasteners Cost** | | | | **Fasteners Cost** | | |
| | Fastners(*5) | ₹ | 78.91 | | Fastners(*9) | ₹ | 132.00 |
| | Tighten | ₹ | 561.22 | | Tighten | ₹ | 561.22 |
| | **TOTAL** | ₹ | **6,539.18** | | **TOTAL** | ₹ | **4,406.17** |

**Fig 10.3: Cost Comparision of Smart Manufacturing Techniques used**

## 9.2 SEASON BUDGET

| BUDGET 2019-20 | | | | | |
|---|---|---|---|---|---|
| TOTAL EXPECTED BUDGET | ₹ | 8,40,000.00 | TOTAL INCOMING | ₹ | 7,56,000.00 |

**Fig 10.4: Budget 2019-2020**

| BODYWORKS | | | CHASSIS | | | MANUFACTURING | | | MISCELLANEOUS | | |
|---|---|---|---|---|---|---|---|---|---|---|---|
| **Carbon Fibre and Resi** | ₹ | 26,000.00 | **Tubes** | ₹ | 33,000.00 | **Safety** | ₹ | 1,000.00 | **FB** | ₹ | 80,000.00 |
| **Moulds** | ₹ | 1,000.00 | **Jigs** | ₹ | 10,000.00 | **Spool** | ₹ | 1,200.00 | **Supra** | ₹ | 70,000.00 |
| **TOTAL** | ₹ | 27,000.00 | **Tube Bending** | ₹ | 3,000.00 | **nuts/bolts** | ₹ | 1,500.00 | **Transportation** | ₹ | 1,50,000.00 |
| | | | **Ms Clamps** | ₹ | 1,100.00 | **misc.** | ₹ | 5,000.00 | **IA** | ₹ | 30,000.00 |
| | | | **TOTAL** | ₹ | 47,100.00 | **Powder Coat** | ₹ | 2,000.00 | **Safety Gloves** | ₹ | 5,000.00 |
| | | | | | | **TOTAL** | ₹ | **10,700.00** | **Suit** | ₹ | 25,000.00 |
| | | | | | | | | | | | |
| | | | | | | | | | **TOTAL** | ₹ | **3,60,000.00** |

**Fig 10.5: Departmental Budget-1**

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| POWERTRAIN | | | VD | | TYRES | | | BRAKES | | | BODYWORKS | | |
|---|---|---|---|---|---|---|---|---|---|---|---|---|---|
| AT Power | ₹ | 50,000.00 | Upright | 58000 | Rims | ₹ | 48,000.00 | Callipers | ₹ | 52,000.00 | Carbon Fibre and Resi | ₹ | 26,000.00 |
| Throttle Body 390 | ₹ | 11,000.00 | Hubs | 60000 | Slicks(if sponsored) | ₹ | 12,000.00 | Rotors | ₹ | 8,000.00 | Moulds | ₹ | 1,000.00 |
| Driveshaft and Housing | ₹ | 12,000.00 | Bearings | 8000 | TOTAL | ₹ | 60,000.00 | Brake Lines | ₹ | 10,000.00 | TOTAL | ₹ | 27,000.00 |
| Sprocket | ₹ | 5,000.00 | Rod Ends | 7000 | | | | Master Cylinders | ₹ | 14,000.00 | | | |
| Muffler | ₹ | 4,000.00 | Inserts | 1200 | | | | TOTAL | ₹ | 84,000.00 | | | |
| Exhaust | ₹ | 5,000.00 | Rocker | 4000 | | | | | | | | | |
| Harness | ₹ | 11,000.00 | TOTAL | 138200 | | | | | | | | | |
| Sensors | ₹ | 5,000.00 | | | | | | | | | | | |
| Miscellaneous | ₹ | 10,000.00 | | | | | | | | | | | |
| TOTAL | ₹ | 1,13,000.00 | | | | | | | | | | | |

**Fig 10.6: Departmental Budget-2**

# CHAPTER 11: RESOURCE PLANNING

## 11.1 JOB SCHEDULING AND WORKFORCE PLANNING

| | At Risk | Task Name | Start Date | End Date | Assigned To | Durat... | % Complete |
|---|---|---|---|---|---|---|---|
| 1 | | Watch how-to video on this template (4:16) | | | | | |
| 2 | | | | | | | |
| 3 | | Team Setup | 02/10/17 | 02/13/17 | Zeshan Ahr | 2d | 100% |
| 4 | | project planning | 02/13/17 | 05/19/17 | | 70d | 95% |
| 5 | | budget planning | 02/13/17 | 02/16/17 | Rohan Rai | 4d | 100% |
| 6 | | technical planning | 02/16/17 | 02/20/17 | Abhilash Mi | 3d | 100% |
| 7 | | resource planning | 02/21/17 | 03/01/17 | Saheb Sing | 7d | 100% |
| 8 | | designing phase | 03/02/17 | 04/19/17 | Aditya Abhi | 35d | 100% |
| 9 | | material procurement / Orders | 04/19/17 | 05/19/17 | Jutika Dowa | 23d | 85% |
| 10 | | Manufacturing | 04/05/17 | 01/05/18 | | 198d | |
| 11 | | chassis prototype | 04/05/17 | 04/11/17 | Diksha | 5d | |
| 12 | | chassis fixtures | 04/12/17 | 04/19/17 | Chandan | 6d | |
| 13 | | chassis | 04/20/17 | 05/03/17 | Sidhant | 10d | |
| 14 | | mounts/ chassis clamps | 05/04/17 | 05/09/17 | Sharat Anar | 4d | |
| 15 | | control arms | 05/05/17 | 05/09/17 | G Sashank | 3d | |
| 16 | | machining | 04/06/17 | 05/17/17 | Saheb Sing | 30d | |
| 17 | | first assembly | 05/19/17 | 05/29/17 | Narendra S | 7d | |
| 18 | | primer coating | 06/02/17 | 06/06/17 | Devashish | 3d | |
| 19 | | testing | 09/01/17 | 01/05/18 | | 91d | |

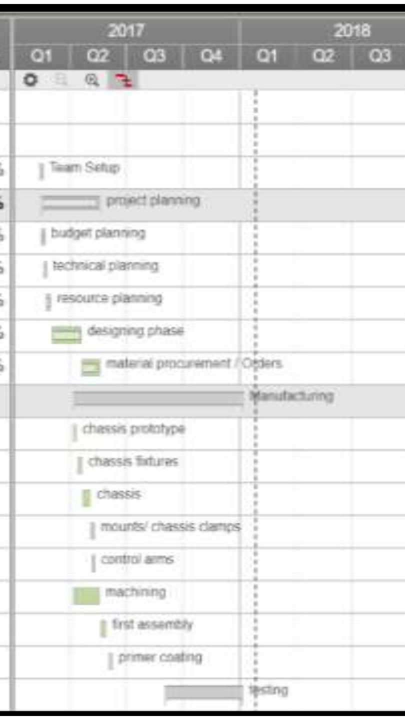



**Fig 11.1: Gant Chart workforce planning**

| At Risk | Task Name | Start Date | End Date | Assigned To | Durat... | % Complete |
|---|---|---|---|---|---|---|
| | testing | 09/01/17 | 01/05/18 | | 91d | |
| | first testing | 09/01/17 | 09/01/17 | | 1d | |
| | maintenance/ inspection | 09/02/17 | 09/11/17 | | 7d | |
| | testing | 09/11/17 | 09/13/17 | | 3d | |
| | maintenance/ inspection | 09/13/17 | 09/21/17 | | 7d | |
| | testing | 09/23/17 | 09/28/17 | | 5d | |
| | driver selection | 10/01/17 | 10/04/17 | | 4d | |
| | maintenance/ inspection | 10/05/17 | 10/13/17 | | 7d | |
| | driver practice | 10/15/17 | 12/18/17 | | 47d | |
| | final maintenance/ inspection | 12/20/17 | 12/28/17 | | 7d | |
| | check test | 01/02/18 | 01/03/18 | | 2d | |
| | final testing | 01/03/18 | 01/05/18 | | 3d | |

Fig 11.2: Gant Chart for job distribution

# CHAPTER 12: CAUSE AND EFFECT

## 12.1 ISHIKAWA DIAGRAM FOR EFFECTS

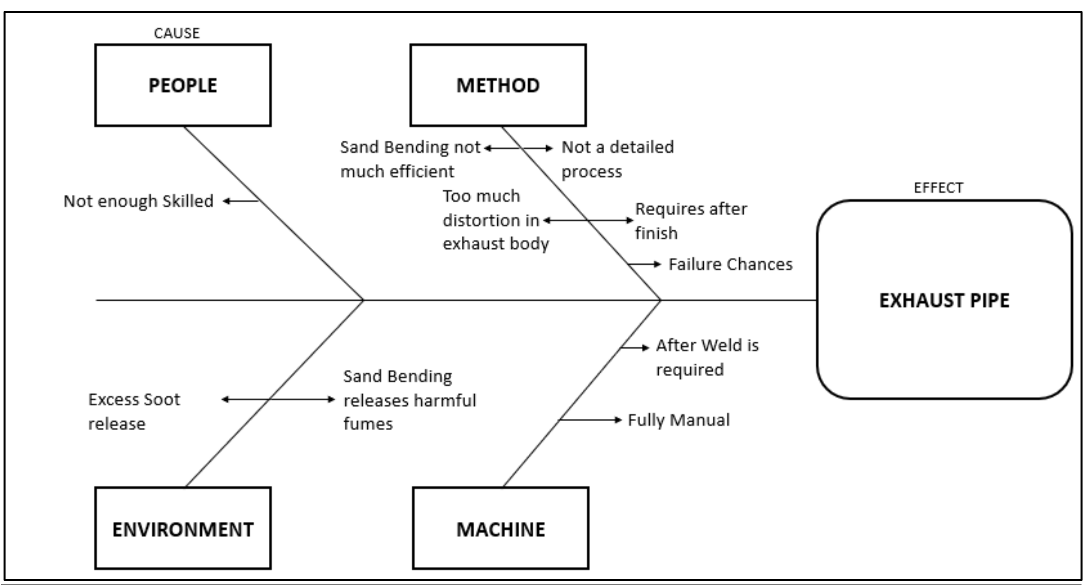



Fig 12.1: Fishbone Diagram for Exhaust Pipe

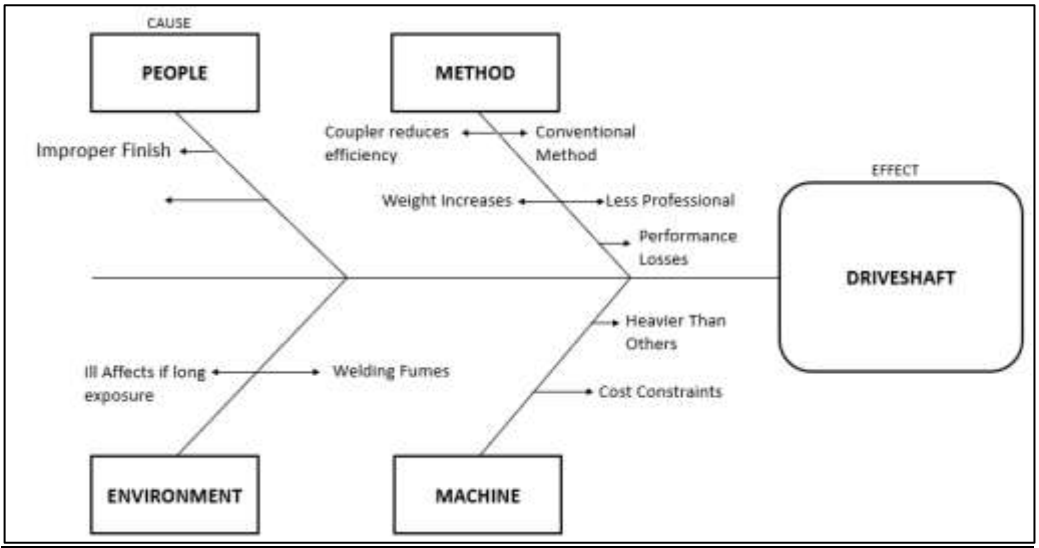



**Fig 12.2: Fishbone Diagram for Drive Shaft**

# CHAPTER 13: ENVIRONMENTAL INFLUENCE

## - WELDING FUMES WHILE STAINLESS STEEL WELDING-

Preferred Gas welding over Arc welding of stainless steel because it reduces the emission of harmful Chromium (VI) forming due to Potassium/Sodium Silicates.

## - MIG WELDING GAS-

Mixed Argon Gas with Carbon dioxide to reduce Ozone production.

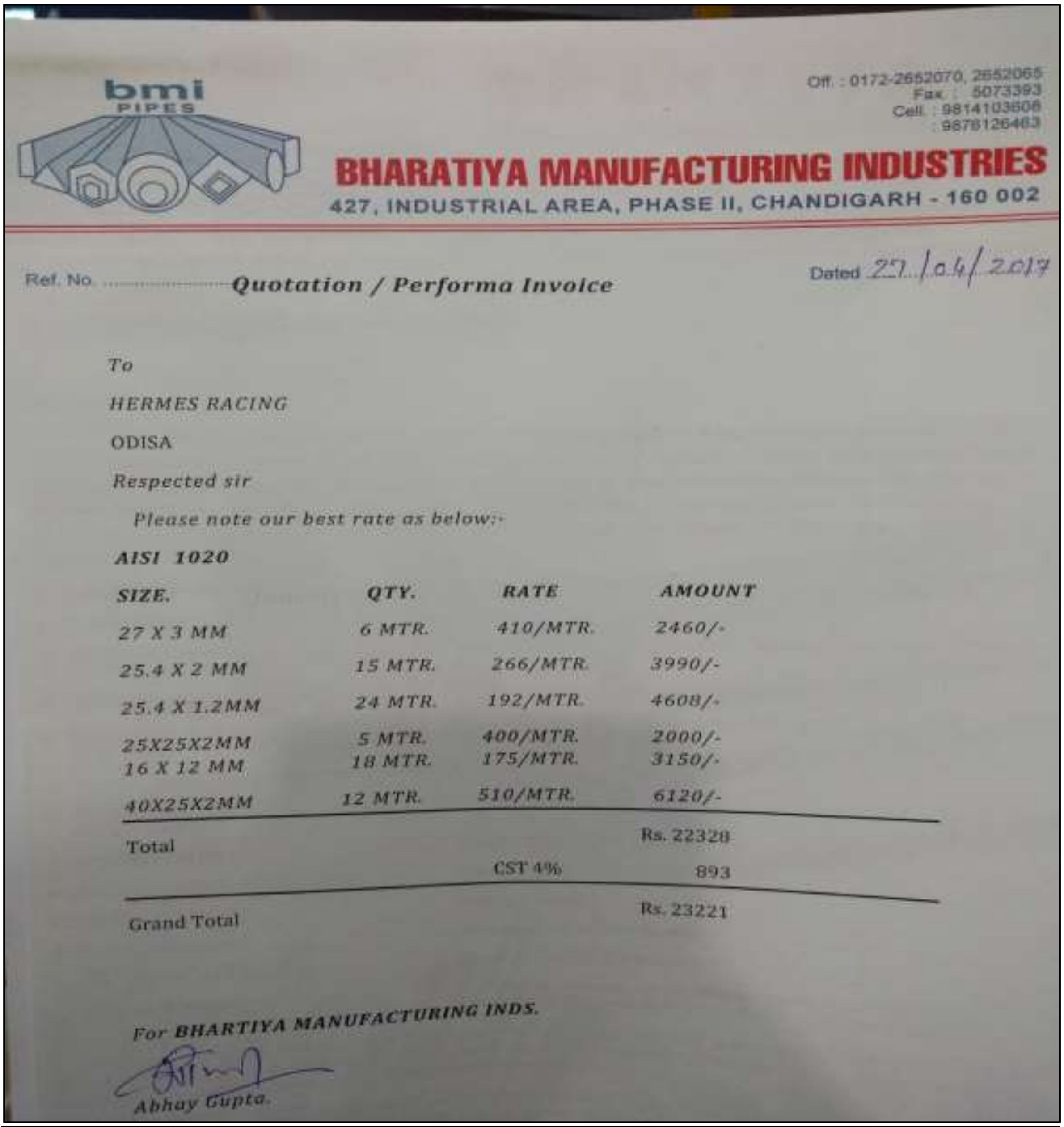
## -ENGINE CUBIC CAPACITY (cc)-

Less cc engine used which releases lesser emissions, thus lowering carbon footprint.

## -SAND BENDING-

Should be condemned because while heating releases Toxic Gases such as CARBON MONOXIDE. Thus, alternative methods should be preferred.

# CHAPTER 13: MATERIAL VALIDATION



bmi PIPES

Off. : 0172-2652070, 2652065
Fax : 5073393
Cell : 9814103606
: 9878126463

**BHARATIYA MANUFACTURING INDUSTRIES**
427, INDUSTRIAL AREA, PHASE II, CHANDIGARH - 160 002

Ref. No. ............ *Quotation / Performa Invoice* Dated 27/04/2017

*To*
*HERMES RACING*
ODISA
*Respected sir*
*Please note our best rate as below:-*

***AISI 1020***

| *SIZE.* | *QTY.* | *RATE* | *AMOUNT* |
|---|---|---|---|
| 27 X 3 MM | 6 MTR. | 410/MTR. | 2460/- |
| 25.4 X 2 MM | 15 MTR. | 266/MTR. | 3990/- |
| 25.4 X 1.2MM | 24 MTR. | 192/MTR. | 4608/- |
| 25X25X2MM | 5 MTR. | 400/MTR. | 2000/- |
| 16 X 12 MM | 18 MTR. | 175/MTR. | 3150/- |
| 40X25X2MM | 12 MTR. | 510/MTR. | 6120/- |
| Total | | | Rs. 22328 |
| | | CST 4% | 893 |
| Grand Total | | | Rs. 23221 |

*For BHARTIYA MANUFACTURING INDS.*

*Abhay Gupta.*

**Fig 13.1: Material and Dimension of Tubes**

# CHAPTER 14: IMPACT ATTENUATOR DATA

## 14.1 IMAPACT ATTENUATOR SPECIFICATIONS

**Table 14.1: Data-Sheet**

| | |
|---|---|
| **Material(s) Used** | IMPAXX™700 |
| **Description of form/shape** | IMPAXX™700 |
| **IA to Anti-Intrusion Plate mounting method** | BOLTED USING GI-SHEET |
| **Anti-Intrusion Plate to Front Bulkhead mounting method** | WELDED |
| **Peak deceleration (<= 40 g's)** | 5.7g |
| **Average deceleration (<= 20 g's)** | 5.2g |

**Table 14.2: Dimensions of Attenuator**

| | | | | | |
|---|---|---|---|---|---|
| **Length (fore/aft direction) (mm)** | 304.8mm | **Width (lateral direction) (mm)** | 355.6mm | **Height (vertical direction) (mm)** | 254mm |

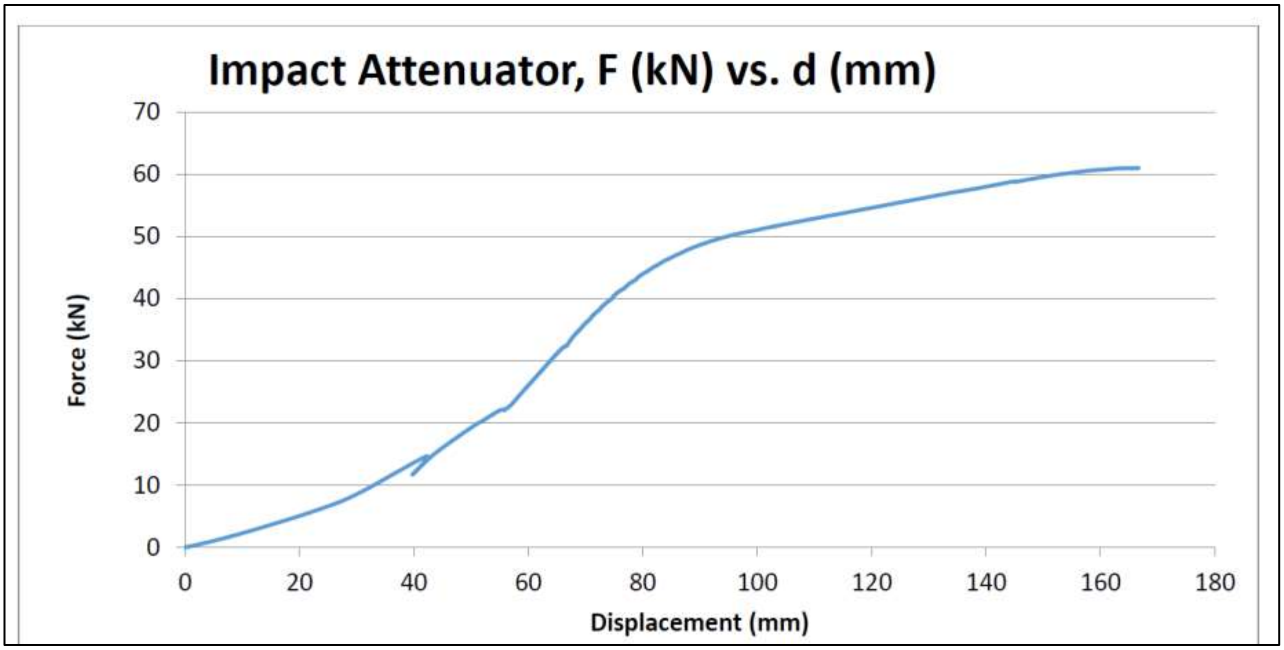



**Figure 14.1: Force-Displacement Curve (dynamic tests must show displacement during collision and after the point v=0 and until force becomes = 0)**

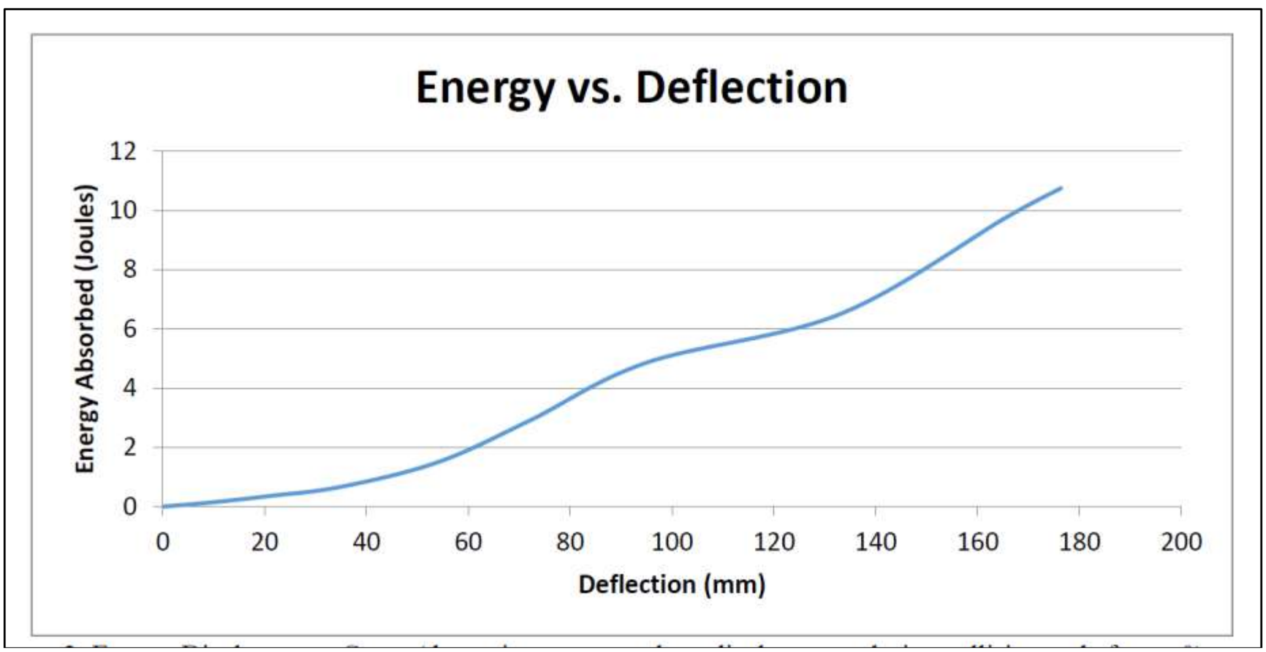



**Figure 14.2: Energy-Displacement Curve (dynamic tests must show displacement during collision and after v=0)**

International Journal of Engineering Research & Technology (IJERT)
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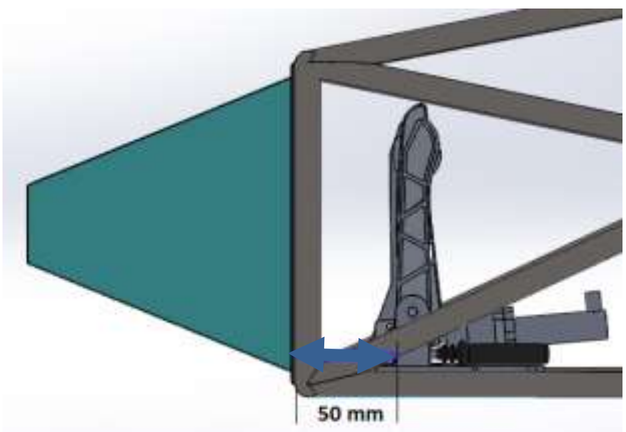


Figure 14.3: Attenuator as Constructed

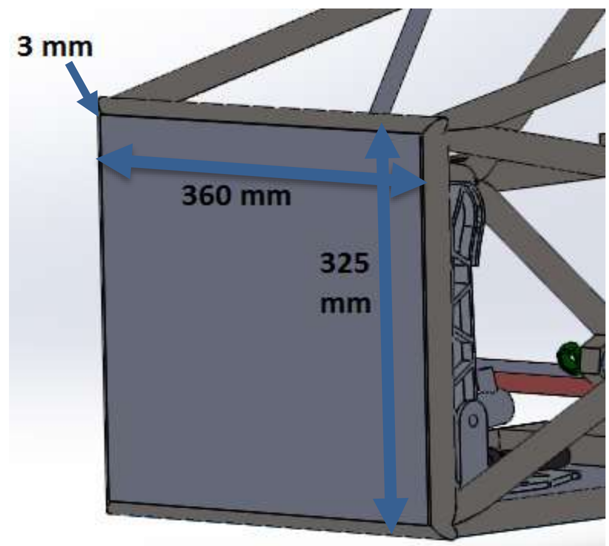


Figure 14.4: Attenuator as Constructed

Table 14.3: Parameters of Test

| | | | |
|---|---|---|---|
| Energy Absorbed (J): Must be >= 7350 J | | Vehicle includes front wing in front of front bulkhead? | NO |
| IA Max. Crushed Displacement (mm): | | Wing structure included in test? | NO |

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| | | | |
|---|---|---|---|
| **IA Post Crush Displacement - demonstrating any return (mm):** | | **Test Type: (e.g. barrier test, drop test, quasi-static crush)** | |
| **Anti-Intrusion Plate Deformation (mm)** | | **Test Site: (must be from approved test site list on website for dynamic tests)** | |

## 14.2 TECHNICAL DRAWINGS

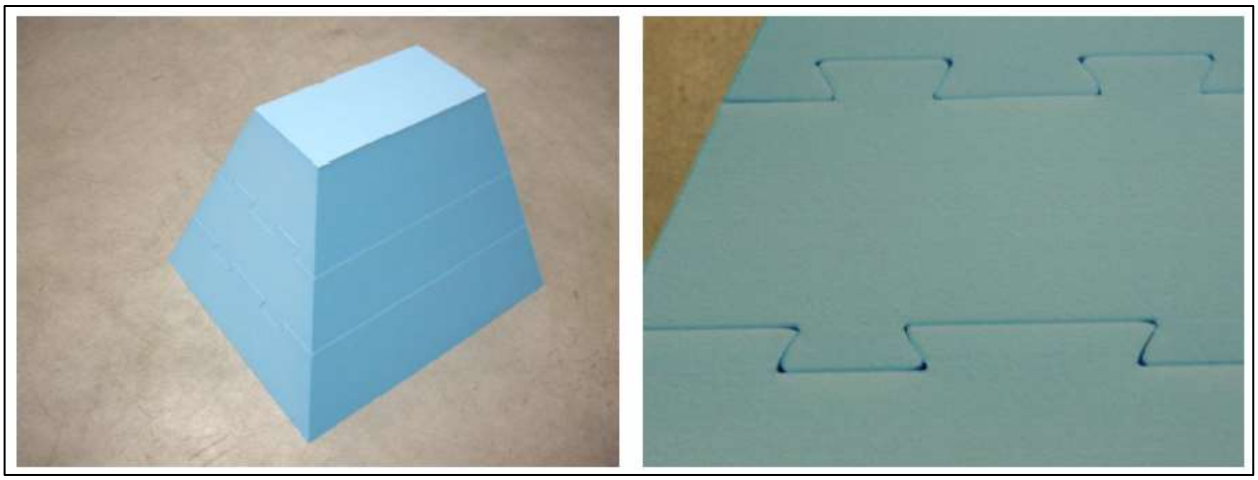

**Fig 14.5: Impact Attenuator Surface**

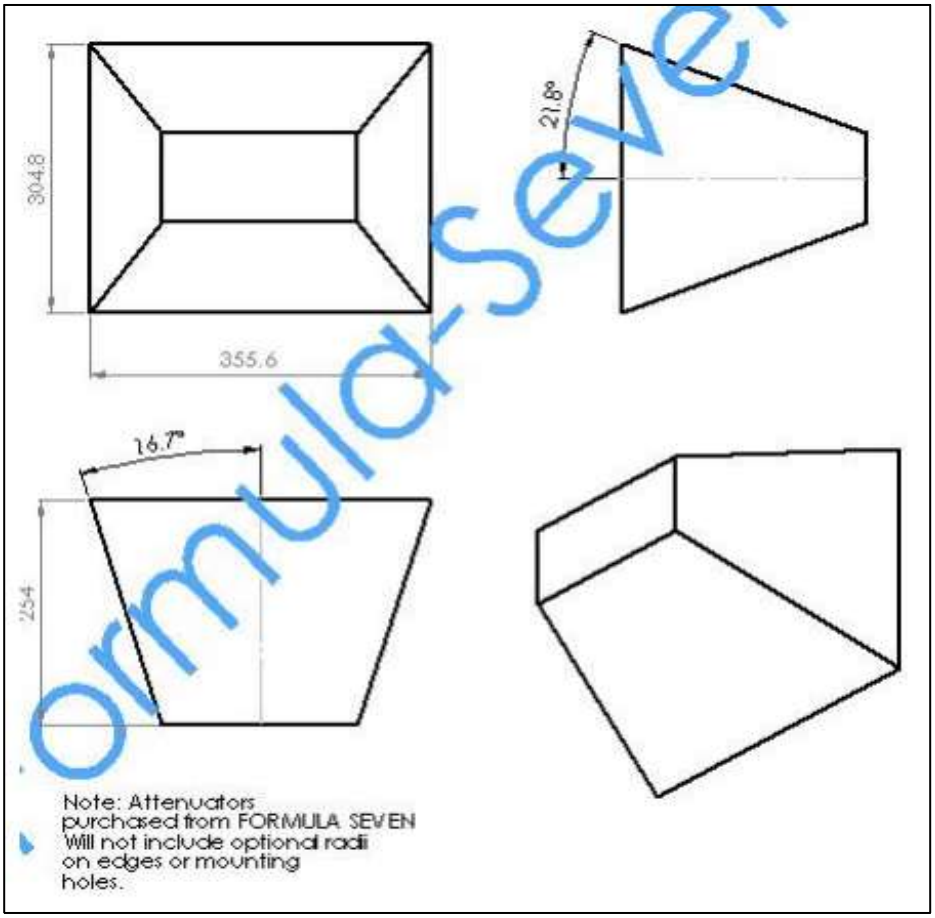


**Fig 14.6: Views OF IA's Engineering Drawing**

## 14.3 REASONS FOR SELECTION

1. Specific energy absorption was maximised and impact force minimised for this design.

2.The frustum square tube absorbed the highest amount of energy,.

3. IMPAXX™700 energy absorbing foam offers good strength, vibration

and shock absorbency and water resistance characteristics.

4. IMPAXX™700 foam is easily available.

5.IMPAXX™700 foam is less weight foam

## 14.4 RECEIPTS/PACKING SLIP

**TAX-INVOICE**

Mobile : 9883256155
E-mail : contact@rajputautocorp.com
Website : www.rajputautocorp.com

**₹AJPUT AUTOCORP** ORIGINAL-BUYER'S COPY

AG-8, Santa Abasan, Hela Battala,
Hatiara, Baguiati,
Kolkata- 700 157

No. 082/OR082/15-16 Date: 26-06-15

**INVOICE TO**
**KIIT University**
Bhubaneshwar- 751024
Odisha

Your Order No.- via email Challan No. 082/15-16 Date: 26-06-15
Date: 04-06-15

| Item | DESCRIPTION | Quantity | Rate (₹) | Per | Amount (₹) | |
|---|---|---|---|---|---|---|
| 1 | Part No.: CHN-DIFF-52TTH -530- LPC-TYPE-2<br>SPARE PARTS: DIFFERENTIAL WITH HOUSING | 1 No. | 16250/- | Each | 16250 | 00 |
| 2 | Part No.: STRNG-WHL-DIA 245 MM-DHU-MS-AD<br>SPARE PARTS: STEERING WHEEL | 1 No. | 5000/- | Each | 5000 | 00 |
| 3 | Part No.: BE-CR-CUST-AA6063-SKF-IN<br>SPARE PARTS: BELL CRANKS | 4 Nos. | 3800/- | Each | 15200 | 00 |
| 4 | Part No.: RCNG-SHCK -PSB-M8-200-MM<br>SPARE PARTS: SHOCK ABSORBER | 4 Nos. | 5600/- | Each | 22400 | 00 |
| 5 | Part No.: POS-10 –IKO-PSB-INSRTS<br>SPARE PARTS: ROD END BEARINGS WITH INSERTS | 30 Nos. | 470/- | Each | 14100 | 00 |
| 6 | Part No.: UPRT-AA-6063-T6<br>SPARE PARTS: UPRIGHT/KNUCKLE ASSEMBLY | 4 Nos. | 15000/- | Each | 60000 | 00 |
| 7 | Part No.: FRP-ST-RT-WF-WA-BL<br>SPARE PARTS: FRP SEAT WITHOUT COVER | 1 No. | 4500/- | Each | 4500 | 00 |
| 8 | Part No.: IMPCT-ATTNTR-DOW-IMPX<br>SPARE PARTS: FSAE IMPACT ATTENUATOR MANUFACTURED FROM DOW IMPAXX™ 700 FOAM | 1 No. | 23500/- | Each | 23500 | 00 |
| 9 | Part No.: PDL-ASML-ABC-BB<br>SPARE PARTS: PEDAL BOX | 1 No. | 13500/- | Each | 13500 | 00 |
| 10 | Part No.: STRNG-GBOX-CTR-MNTD-TYPE-2<br>SPARE PARTS: STEERING GEARBOX | 1 No. | 17500/- | Each | 17500 | 00 |
| | | | | | 191950 | 00 |
| | Packing & Forwarding Charge | | | | 6954 | 76 |
| | | | | | 198904 | 76 |
| | ADD: CST @ 5% | | | | 9945 | 24 |
| | TOTAL | | | | 208850 | 00 |

**Rupees** Two Lac Eight Thousand Eight Hundred Fifty Only 208850.00

VAT No.: 19671206472
CST No.: 19671206472

For Rajput AutoCorp
V. Singh
( VIVEK SINGH )

**Fig 14.7: Receipt**

ISSN: 2278-0181

Product Information

# Tech Data Sheet
# IMPAXX™ 700 Energy Absorbing Foam

IMPAXX™ 700 Energy Absorbing Foam is a higher-performing, and lower-cost alternative to traditional energy absorbing solutions such as polyurethane and expanded polypropylene bead foams.

IMPAXX foam is a highly engineered, extruded, thermoplastic foam that utilizes Dow proprietary process technology to maximize efficiency and minimize *weight*. It is a strong, *low-density*, closed-cell foam.

IMPAXX foam is highly suited for applications in a variety of industries requiring enhanced safety features through energy absorbing countermeasures.

Color: Blue
Sizes available: sheets, basic blocks and custom-fabricated parts.

| Physical Properties[1] | Test Method | Direction | Value U.S. / Metric |
|---|---|---|---|
| Density | ASTM D 3575, Suffix W, Method B<br>DIN 53420 | N/A | 2.8 pcf<br>45 Kg/m³ |
| Compression Strength | ASTM D1621, 23°C | | Psi / kPa |
| @ 10% | | Vertical | 101 / 700 |
| @ 25% | | Vertical | 104 / 718 |
| @ 50% | | Vertical | 121 / 835 |
| Compression Strength | ASTM D1621, -15°C | | Psi / kPa |
| @ 25% | | Vertical | 114 / 788 |
| @ 50% | | Vertical | 138 / 954 |
| Compression Strength | ASTM D1621, 60°C | | Psi / kPa |
| @ 25% | | Vertical | 73 / 504 |
| @ 50% | | Vertical | 85 / 586 |
| Thermal Stability (linear change @ 80°C) | ASTM D 3575, Suffix S or DIN 53431 | N/A | < 2% |
| Flammability** | FMVSS 302 | Extrusion | Pass |
| Fogging | SAE J1756 | | 100 Fog Number |
| Water Absorption | ASTM D3575, Suffix L | N/A | 6% |

(1) The data presented for this product is for unfabricated foam. While values shown are typical of the product, they should not be construed as specification limits.
** Results from this test do not represent performance under actual fire conditions.

- *See reverse side for additional properties and product information.*

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**Fig 14.8: Receipt Proof**

ISSN: 2278-0181

# CONCLUSION

The vehicle HR-19 was designed by the designers of team Hermes Racing, using industry level software such as the DS Solidworks and Altair Hyper-works. Further all the designs were analyzed and validated using genuine and authentic software such as Ansys and Altair Hyper-mesh and Hyper-works.

The Suspension Geometry was determined using Lotus Software and the physical parameters of the vehicle's suspension was fixed using simulations and results from LS Dynatune. Further the parts were machined and manufactured using advanced manufacturing techniques such as 5-axis CNC Machining, Waterjet cutting, Laser Cutting and Raid Prototyping.

The aerodynamics of HR19 was made after fixing the values from a perfect computational flow dynamics and material used was 1-ply Woven Carbon Fiber Fabric.

Under the Powertrain division the vehicle was perfectly tuned using a Chassis Dynamometer and the most efficient MAP values were selected, as per the vehicle's overall dimension and weight of 210 kg. The 1-D engine simulation and tuning was also done by Ricardo software and an accurate and detailed study of the vehicle was done.

Lastly, the vehicle was checked all-throughout in terms of Safety Measures and Guidelines as per the Rulebook of Formula Student Germany and declared safe for taking out on the Track and proceed forward for the Dynamic Testing of the vehicle for a minimum distance of 350 km within a time span of 5 months.

Thus, the vehicle project was assembled under the prescribed deadline and within the allotted budget, as per the required work-force planning.

ISSN: 2278-0181

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## E. REFERENCES

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[3] Documental Repositories, Team Hermes Racing, KIIT University, http://hermesracing.in/index.html the abbreviation

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