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DESIGN OF A STUDENT FORMULA RACING CAR WITH COMPUTATIONS AND ANALYSIS

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Abstract - *Engineers cannot be nurtured only in classroom; it requires practical exposure for overall development of theoretical knowledge of what they study in classroom. FFS INDIA is great platform for the students to learn and gain practical knowledge. FMAE organizes engineering design event every year, since 2017 to provide practical exposure to students who are automobile enthusiast. The competition invites team from different colleges, institutes and universities to design, fabricate and develop the formula style prototype car under the rules mentioned in the rulebook released by FMAE and compete with various other teams in static and dynamic event. This report will discuss about designing and fabrication aspects of different sub parts of the car, namely chassis, suspension, transmission, steering, braking, aerodynamic package etc.*

***Key Words*: *FMAE, Suspension, steering,* transmission, braking.**

1.INTRODUCTION

In this report, the criteria and methods used to develop the design of various subsystems of the car is discussed. While developing the car, students acquire diverse practical knowledge, which is not only limited to machinery and electronics but also increases the performance, reduce cost and improve the vehicle marketability. Leadership and teamwork among members are fostered with a strong sense of camaraderie. The competition thus enhances the ability of a student to identify and resolve problems on their own. The competition intends to aim at nurturing engineers who are rich in originality through an environment of object creation, in which they can learn the essence of object creation and the processes this entails, as well as experiencing team activities, and the difficulty, interest, and enjoyment of object creation. The marks awarded for various static and dynamic events is listed below.

Table 1. Points Table

Events	Static/Dynamic	Max. Points
Technical Inspection	Static	Nil (Qualifying)
Design Presentation	Static	150
Sales Presentation	Static	75
Cost Presentation	Static	100
Brake	Dynamic	Nil (Qualifying)
Acceleration Test	Dynamic	75
Skid Pad	Dynamic	75
Auto cross	Dynamic	100
Fuel Economy	Dynamic	100
Endurance	Dynamic	325
TOTAL		1000

In the process of developing the car, drivability, reliability, weight reduction and manufacturability were chosen as the main area of focus. Design goal for the car is mentioned below:

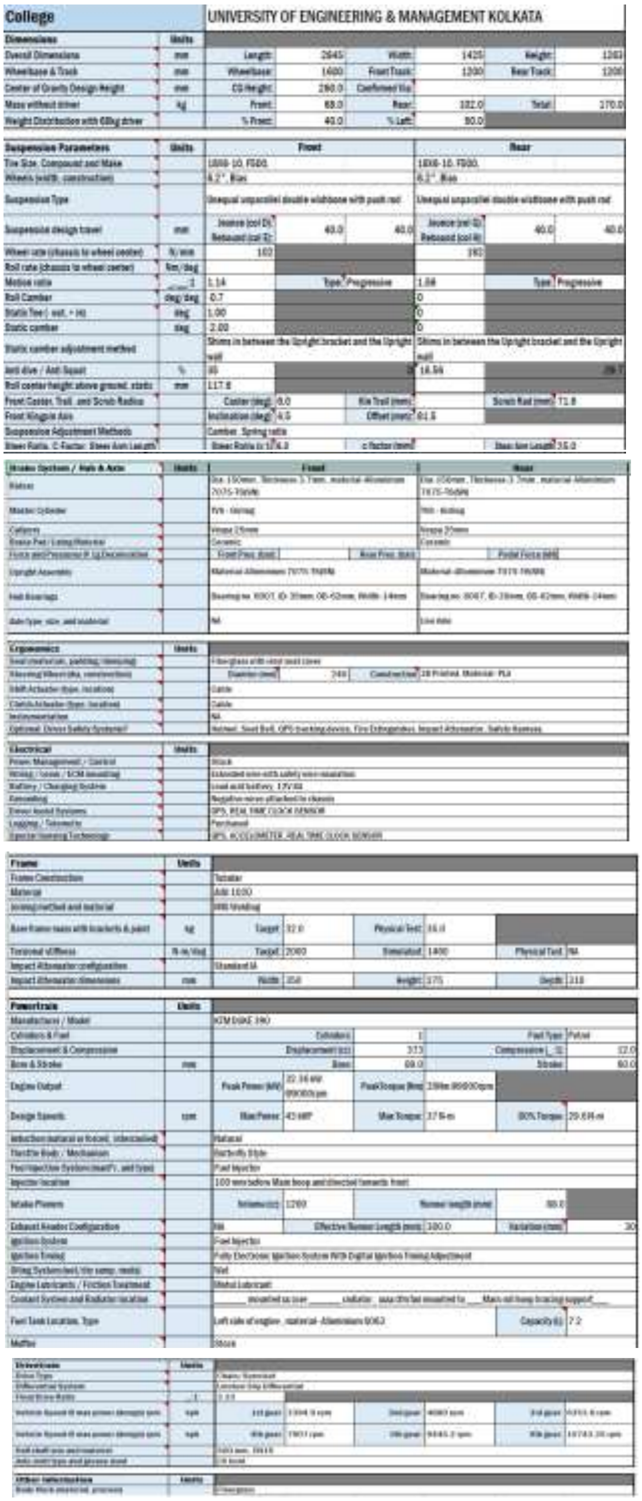
Drivability: Engine must provide steady torque from 7000 to 10500rpm.

Reliability: The vehicle must complete all dynamic events.

Weight Reduction: The vehicle target weight was set at 150kg, to increase its speed.

Manufacturability: The vehicle must be designed in such a way that it should be simple in manufacturing and cost effective.

2. TECHNICAL SPECIFICATION OF THE VEHICLE



College	UNIVERSITY OF ENGINEERING & MANAGEMENT KOLKATA						
Dimensions	Units						
Overall Dimensions	mm	Length	2645	Width	1425	Height	1283
Wheelbase & Track	mm	Wheelbase	1600	Front Track	1200	Rear Track	1200
Center of Gravity Design Height	mm	CG Height	280.0	Confirmed Via			
Mass without driver	kg	Front	68.0	Rear	102.0	Total	170.0
Weight Distribution with 68kg driver		% Front	40.0	% Left	50.0		

3. CHASSIS FRAME CONDIGURATION, DESIGN & MATERIAL SELECTION

Frame or roll cage acts as skeleton for any vehicle which provides sufficient support and encloses all the sub systems of the vehicle in it. Material of the frame should be strong enough to withstand heavy load and stresses & light enough to reduce the overall weight of the vehicle and should also be cost effective. Market survey is conducted to collect the information of the material available in the market, which is tabulated as follows:

Table 2. Material selection for chassis frame.

Sl. No.	Properties	Material		
		AISI 1018	AISI 1020	AISI 4130
1	Carbon %	0.15 - 0.20	0.18 - 0.23	0.28 - 0.33
2	Density (kg/m3)	7.87 x 103	7.7 x 103	7.85 x 103
3	Tensile yield Strength (MPa)	370	350	435
4	Tensile ultimate Strength (Mpa)	440	400	560
5	Modulus of Elasticity (GPa)	205	200	210
6	Poisson's Ratio	0.29	0.29	0.29

Above is the material specification of the available seamless pipes in the market and for the manufacturing of chassis, the below decision matrix is used to arrive at the final decision in which 1= poor, 2= Ok, 3=good, 4= best.

Table 3. Decision Matrix Rating

SL. NO.	PARAMETERS	AISI 1018	AISI 1018	AISI 1018
1.	Availability	3	3	3
2.	Cost	2	3	2
3.	Weld Ability	2	3	1
4.	Total	7	9	6

With reference to above matrix, AISI 1020 material is selected because of its suitability and the chassis is made with the pipes of diameter (1) OD = 25.4mm, ID

= 20.4mm for main hoop, (2) OD = 25.4mm, ID = 20.4mm for ladder, front hoop and shoulder harness bar & (3) OD = 25.4mm, ID = 22.1 mm for front bulk head, bulk head support, front hoop bracing, side impact, firewall members, main roll hoop bracing support and for all other triangulated member. At first a dummy chassis was made in Solidworks 3d modeling software with the rules as stated in the FMAE FFS rulebook with the driver seated inside the chassis to check the compatibility of the real chassis with the CAD model. so that the design can be finalized for the final manufacturing of chassis with AISI 1020 material. Targets set by the team to manufacture the chassis are tabulated as follows:

Table 3. Roll cage design considerations

Sl. No.	Considerations	Priority	Reason
1	Light weight	Essential	Light car is fast car
2	Durable	Essential	Must not deform in any circumstances
3	Requirements	Essential	must meet requirements to compete
4	Simple Frame	High	Fabrication to be done in house
5	Attractive Design	desired	Attractive looks of the car helps in good product sale
6	cost	Low	It has to meet the budget of the project

After completion of the chassis designing process, finite element analysis (FEA) was performed using Solidworks and Ansys software to ensure that expected load does not exceed material strength and the final design of the chassis design is displayed below with the wall thickness shown in different color.

RED ⟶ 25.4mm X 2.5mm
GREEN ⟶ 25.4mmX 1.65mm
YELLOW ⟶ 25.4mmX1.25mm

Fig- 1: Chassis configuration

4. IMPACT ANALYSIS AND CALCULATION FOR THE CHASSIS

Now, after the selection of chassis material, types of analysis to be performed to ensure its stability under various conditions in mentioned below:

(a) Rear impact analysis
(b) Front impact analysis
(c) Side impact analysis
(d) Front torsional analysis
(e) Rear torsional analysis
(f)Modal or frequency analysis
(g) Static vertical bending analysis

Most of the formula one driver usually experience 5G force while braking, 2G force while accelerating and 4G to 6G force while cornering. Several softwares are available in the market for analysis purpose. Here SOLIDWORKS software has been used to perform the analysis and the steps are mentioned below:

(a) Open the required design or import the chassis file if it is designed in any other software.
(b) Apply the material properties to the design.
(c) Fix the required points of the chassis using fixtures (usually the suspension points) shown the green arrow head in Fig-2.

(d) Create the mesh of the chassis.
(e) Once, meshing is done. Run the analysis.
(f) Using analysis results, i.e. Von Mises stress, equivalent strain, displacement and factor of Safety, the chassis will be judged for its stability.

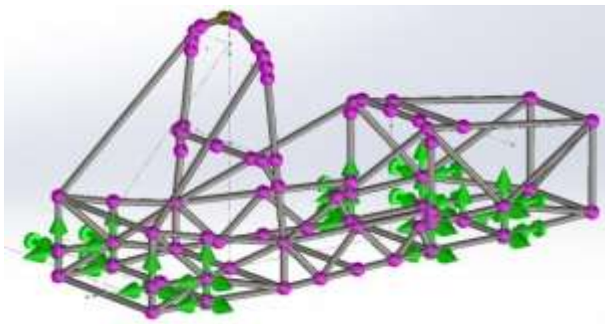


Fig – 2: Fixture and Load applied on Chassis

Now the sample calculations and results of the above discussed analysis types is discussed below.

A. *Rear Impact analysis*

In this type of analysis also, the front and rear suspension points are fixed as shown with green arrow and load is applied on the rear 4 nodes as shown with pink arrow in Fig – 3.

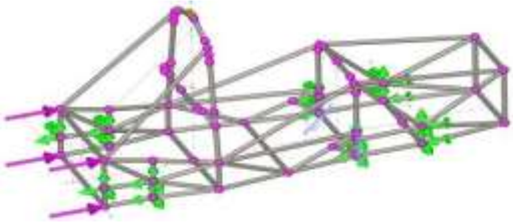
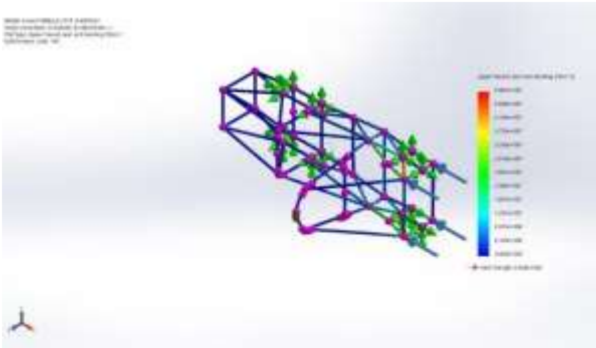


Fig-3: Boundary condition during Rear Impact Analysis

Weight of the entire vehicle is 240 Kg and 2G-Force (240 x 2 x 9.8 = 4704/4=1176N) is applied on each nodes of the rear section of the chassis for structural analysis as shown in the fig 4 above to check its stability. Maximum stress of 5.0024e+007 N/m^2, maximum displacement of 0.300207 mm and minimum Factor of Safety as 9.2 is recorded in this analysis.



Fig–4: Stress generated during Rear Impact

B. *Front Impact Analysis*

In this analysis type, front and rear suspension points are fixed as shown with green arrow and load is applied on the front 4 nodes as shown with pink arrow in fig – 5.

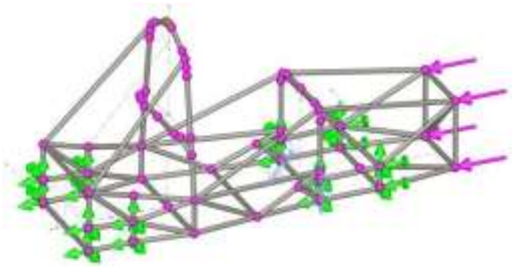
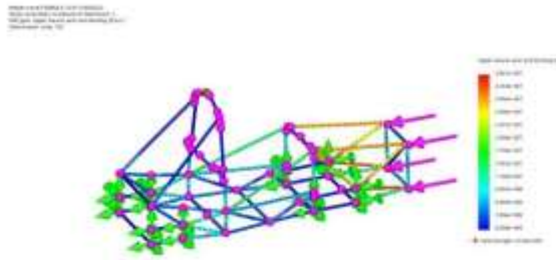


Fig-5: Boundary condition during front impact analysis

Weight of the entire vehicle is 240 Kg and 2 G-Force (240 x 2 x 9.8 = 4704/4=1176N) is applied on each node of the front bulk head of the chassis for structural analysis as shown in the fig 5.



Maximum stress of 3.58095e+007 N/m^2, maximum displacement of 0.279657 mm and minimum Factor of Safety as 9.8 is recorded in this analysis.

C. *Side Impact Analysis*

In this type, the side members which will first face the impact in case of collision is tested with 2 G – Force as shown in the Fig – 7. In this analysis also, front and rear suspension points are constrained as shown in Fig– 7.

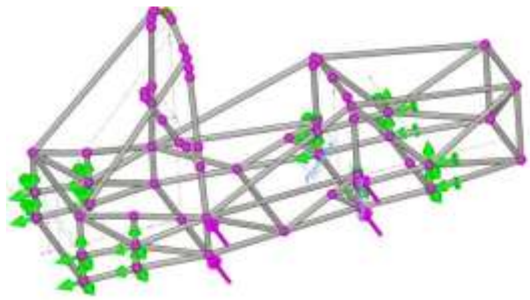


Fig - 7: Boundary condition during side Impact Analysis

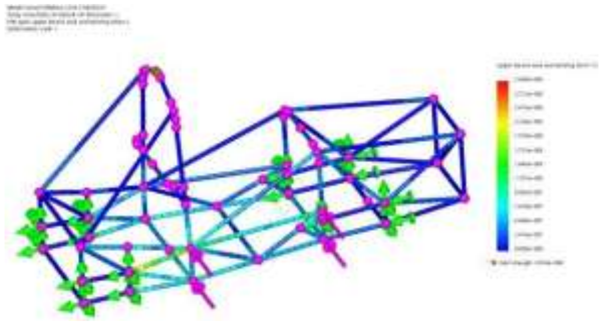


Fig – 8: Stress generated during Side Impact

Maximum stress of 2.96789e+008 N/m^2, maximum displacement of 2.70527 mm and minimum Factor of Safety as 1.5 is recorded in this analysis.

D. Torsional Analysis

Torsional analysis is considered to be one of the most important analysis on which the structural stability of the chassis depends upon. In this test, the chassis is set to act like a cantilever with one end fixed and other end is subjected to torque. The analysis is to be performed for both the front and rear section. Chassis should be designed for high torsional stiffness with low weight of the vehicle. If the chassis vibrates due to significant twisting, it affects the vehicle handling performance. The torsional rigidity can be determined using the formula stated below with the given figure.

$K = R/\theta$

$K = (F \times L)/ \tan^{-1}[(\Delta y1 + \Delta y2)/2L]$

Where, K = Torsional stiffness T = Torque;
Θ = Angular Deflection

F = Shear force

y1, y2 = Translation displacement

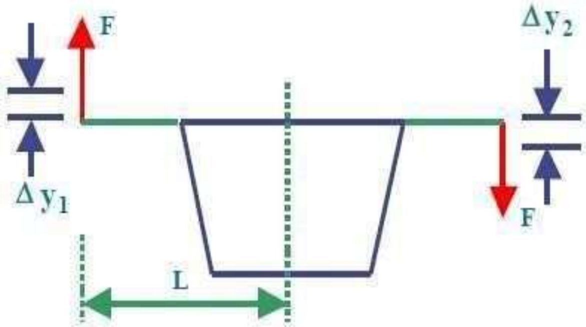


Fig – 9: Suspension testing loads

- ***Front Torsional Analysis***

2 G - Force is considered for this analysis. Weight distribution of the car is taken as 40-60%. So, the weight on front suspension points is 96kg (40/100 x 240= 96kg). 48kg (96/2 = 48kg) of weight should be exerted on each side but to make the chassis safe, the analysis will be carried out considering 96kg of load on each side. Upward load (96 x 2 x 9.8 = 1881.6N) to be applied on 4 nodes of one side of the front suspension and downward force 1881.6N to be applied on the 4 nodes of the other side. Upwards force is shown with blue arrow and downward force is shown with pink arrow in fig – 10 below.

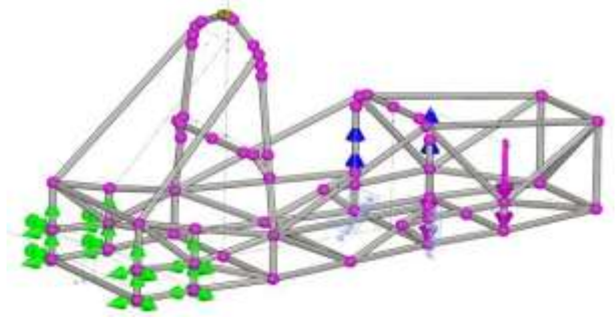


Fig – 10: Boundary condition for Front Torsional Analysis

Rear suspension points are fixed and the load is applied on the front suspension points as shown in the figure above. Maximum stress 1.79999e+008 N/m^2, maximum displacement of 8.19914 mm and minimum Factor of Safety as 2 is recorded in this analysis. Front torsional rigidity can now be calculated with the above obtained values.

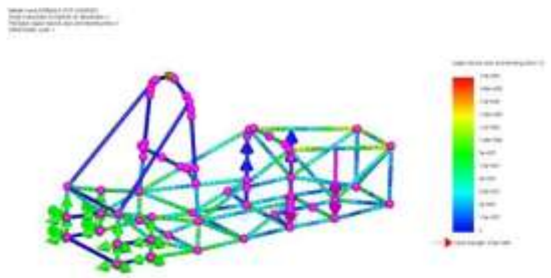


Fig – 11: Stress generated for Front Torsional Analysis

$\mathbf{K} = 1881.6 \times 0.600/ \tan^{-1}[(0.0081 + 0.0081)/2 \times 0.600]$

$= 1459.64$ Nm/deg

- ***Rear Torsional Analysis***

2G – Force is again considered for this analysis. Rear section has 60% of the total load. So, the load on rear suspension points is 144kg. 72kg of load should be applied on each side of suspension but to ensure better stability of the chassis, 144kg of load is considered on each side for worst case scenario. Here, front suspension points are constrained and loads are applied on rear suspension points. Upward load (144 x 2 x 9.8 = 2882.4N) to be applied on 4 nodes of one side of the front suspension and downward force of 2882.4N to be applied on the 4 nodes of the other side. Upwards force is shown with pink arrow and downward force is shown with blue arrow in fig – 12 below.

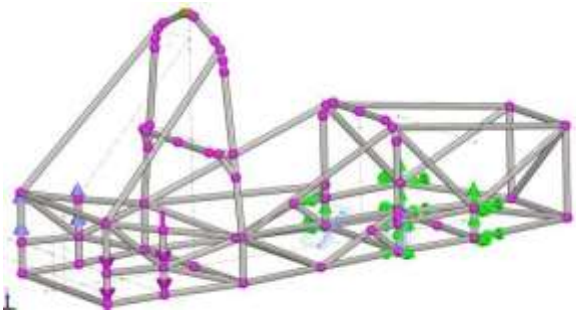


Fig – 12: Boundary condition for Rear Torsional Analysis

Maximum stress 1.4416e+008 N/m^2, maximum displacement of 3.99721 mm and minimum Factor of Safety as 2.4 is recorded in this analysis. Rear torsional rigidity can now be calculated with the above obtained values.

$$\mathbf{K} = 2882.4 \times 0.600 / \tan^{-1}[(0.0039 + 0.0039)/2 \times 0.600]$$

$$= 4643.82 \text{ Nm/deg}$$

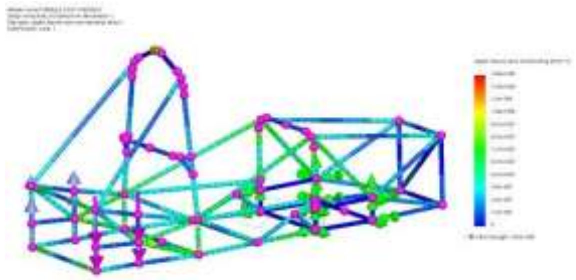


Fig – 13: Stress generated for Rear Torsional Analysis

E. Modal or Frequency Analysis

Every structure has the tendency to vibrate at certain frequencies which is commonly known as Natural Frequency. The lowest frequency of vibration is called Fundamental frequency and higher frequencies are known as Harmonics. Every natural frequency is linked with certain shape of a structure called Mode Shape. If dynamic load coincides with any of the natural frequency, they can undergo large displacements which is known as Resonance. Resonance causes infinite motions. Natural frequencies depend on:

(a) Geometry of the structure

(b) Mass and material properties

(c) Fixtures and support conditions

(d) In plane loads.

Resonance can be avoided by altering these characteristics. Computation of natural frequencies in mode shapes is known as: (a) Modal analysis (b) Frequency Analysis and (c) Normal Mode analysis. Frequency is given by Hertz or Hz and it can be given by RPM = Hz * 60 [3]. Modal analysis is important to test the structural behavior of the chassis during certain range of frequencies. In this analysis, the front and rear suspension points are constrained and only the structural mass of the chassis is considered with no load applied on it. SOLIDWORKS software has shown the results from **15 Hertz to 42 Hertz** automatically. At high speed, the engine frequency is around 100Hz [2]. The results obtained during the analysis is tabulated below.

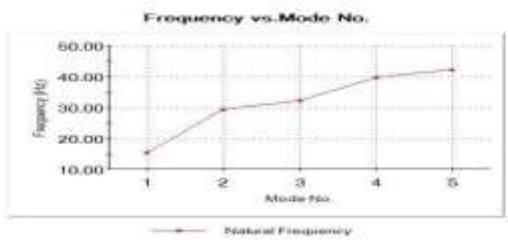
Table 4. Frequency analysis results

Frequency Number	Rad/sec	Hertz	Seconds
1	96.941	15.429	0.064814
2	184.66	29.39	0.034025
3	203.51	32.39	0.030873
4	250.06	39.798	0.025127
5	266.08	42.348	0.023614

Table 5. Mass participation

Mode Number	Frequency(Hertz)	X direction	Y direction	Z direction
1	15.429	2.3999e-006	0.66569	1.7282e-005
2	29.39	0.01212	0.0036293	0.45184
3	32.39	0.00037094	0.060214	0.02062
4	39.798	3.6071e-005	0.00028574	0.0014959
5	42.348	0.00056974	0.00036289	0.039602
		Sum X = 0.013099	Sum Y = 0.73019	Sum Z = 0.51358

It is noted that, from the above data, none of the frequency matches with the natural frequency of the four-stroke single cylinder petrol engine which is 100Hz. So, the chassis is said to be safe during vibration. Frequency Vs Mode number graph is given below for better understanding of the results.



Graph 1: Frequency Vs Mode No. graph

F. Static vertical bending analysis

The sub systems of the car which is installed in the chassis also exerts a Vertical downward force to the chassis especially on the ladder of the frame as shown in figure – 14 below with blue arrow. A force of 240 x 9.8 = 2352N is applied downwards along the driver and engine compartment.

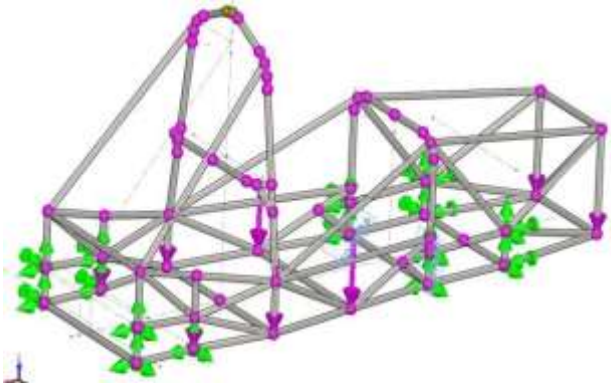


Fig – 14: Boundary condition for Vertical bending Analysis

Front and Rear suspension points are constrained in this analysis type. Maximum stress of 6.31907e+007 N/m^2, maximum displacement of 0.522724 mm and minimum Factor of Safety as 5.6 is recorded in this analysis. Stress generated during vertical bending is shown below.

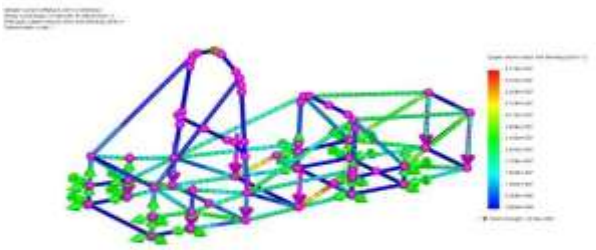


Fig – 15: Stress generated for Vertical bending

4. SUSPENSION

Suspension plays a vital role in the stability of a vehicle. It also enhances vehicle handling while cornering at high speeds and isolates the unsprung mass from road shocks. High amount of weight is transferred in lateral and longitudinal direction and to eradicate this, a functional suspension system should be installed in the vehicle. After an extensive research unequal, unparallel double wishbone system is selected for the vehicle suspension system.

Front Suspension: Push rod with unequal unparallel double wishbone is selected for the front suspension. This wishbone system minimizes tread change to avoid excessive tire wear and also gains negative camber in case of rolling. Push rod suspension packaging is also easy andeffective.

Rear Suspension: Rear suspension system is equipped with double wishbone with push rod geometry to provide enough stiffness to wheel travel to avoid problems of transmitting power from gearbox to wheel. Push rod system is used because of its simple geometry but still effective. Rear suspension consists of low camber and castor angle to transmit the power to wheels effectively. Suspension parameters are tabulated asfollows:

Table 7. Parameters of Front and Rear shocks.

Sl. No.	Parameter	Front	Rear
1	Length of spring	130mm	130mm
2	Total Length (Spring + Damper)	210mm	210mm
3	Wire Diameter	9mm	9mm
4	Mean Diameter	50mm	50mm
5	No. of active turns	6	6
6	Total no. of turns	8	8

Here is the CAD view of the Suspension system of the formula student type vehicle generated in SOLIDWORKS:

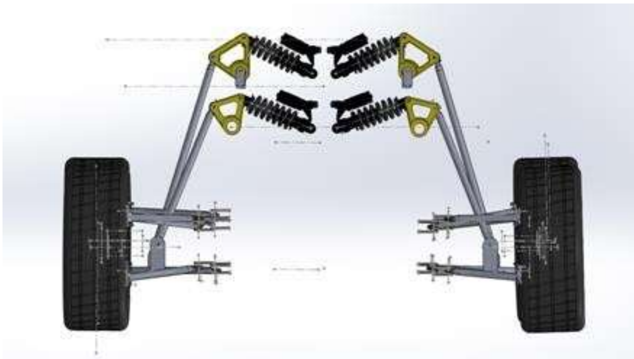


Fig – 16: Front view of the suspension system of student formula vehicle.

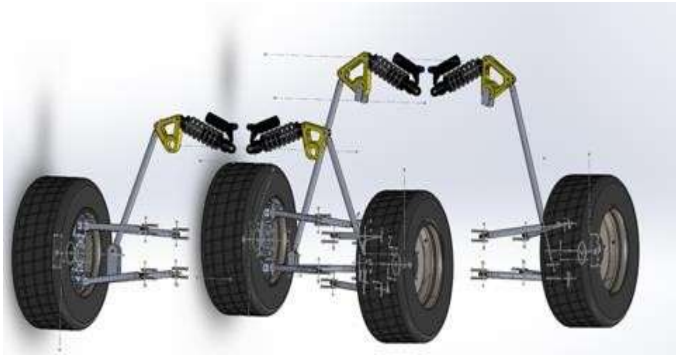


Fig – 17: Isometric view of suspension system of student formula vehicle.

It is very important to choose the correct suspension springs because it is responsible for stability of the vehicle. Below is the parameters mentioned for the spring and damper selected.

Table 8. Specification of suspension systems

Sl. No.	Parameter	Front	Rear
1	Type	Unequal unparallel double wishbone	Unequal unparallel double wishbone
2	Roll center height	117.55mm	108.27
3	Motion Ratio	1.14	1.56
4	Suspension Travel	30mm	30mm
5	Spring and Damper	DNM RCP 2S	DNM RCP 2S
6	Spring rate	78.8 N/mm	78.8 N/mm

- **Components of Suspension System**

Suspension system comprises of the following components: (A) Upright, (B) wishbone, (C) push or pull rod, (D) Bell crank (E) wheel Hub and (F) spring.

A. Front Upright

Upright is a component which connects the vehicle body with tyre with the help of a wishbone. Upright transfers the load experienced by it to the contact point of the ground and tyre through wheel hub. The design of front and rear upright is made different with different dimensions. Below is the analysis result of the front upright:

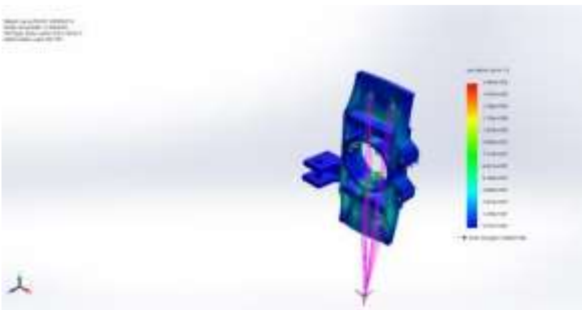


Fig - 18: Front Upright Static analysis

Maximum stress of 1.54351e+008 N/m^2, maximum displacement of 0.222971mm and minimum Factor of safety as 3.3 is recorded in the analysis. Weight of the vehicle is assumed as 280kg to be on the safe side. Also, torsion is applied on the brake mounting points. The Remote 3G load is applied on the upright to carry out the analysis. The bearing hole is constrained and 1500 N as longitudinal force, 2000 N as lateral and 2500 N as vertical or bump load is applied on the upright wishbone mounting point and a separate load of 500N is applied on the tie rod mounting point for the analysis.

Fatigue is the weakening of a material caused by cyclic loading that results in progressive and localized structural damage and the growth of cracks.

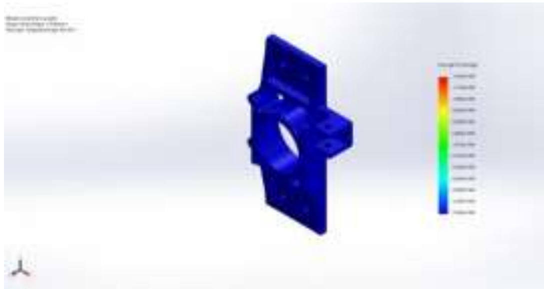


Fig- 19: Front upright Fatigue analysis (Damage)

Front upright has analyzed with **10000000000 cycles**. It is observed that Minimum Damage at 25000 cycle and Maximum Damage at 120405 cycle.

Fig- 20: Front upright Fatigue analysis (Life)

Front upright has analyzed with **10000000000 cycles**. It is observed that Minimum Life is 8.30532e+006 cycle and Maximum Life is 4e+007 cycle.

B. Rear Upright

The rear upright is also constrained at the bearing hole and the remote 3G load is applied on the wishbone mounting points and a separate load of 500N is applied on the follower bar mounting on the upright, same as the front upright to carry out the analysis. Also, torsion is applied on the brake mounting points. Maximum stress of 9.71582e+007 N/m^2, maximum displacement of 0.208041 mm and minimum Factor of safety as 5.2 is recorded in the analysis. In this case also, the weight of the car is assumed as 280kg. Static analysis of the rear upright is shown below.

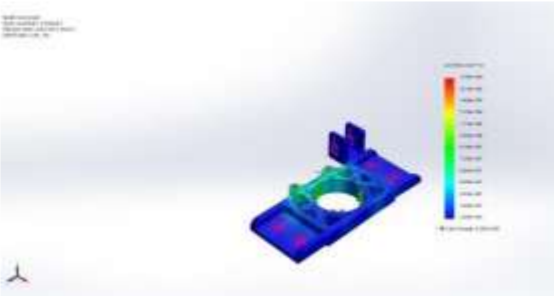


Fig - 21: Rear Upright Static analysis

Rear upright has analyzed with **10000000000 cycles**. It is observed that Maximum Damage at 25000 cycle.

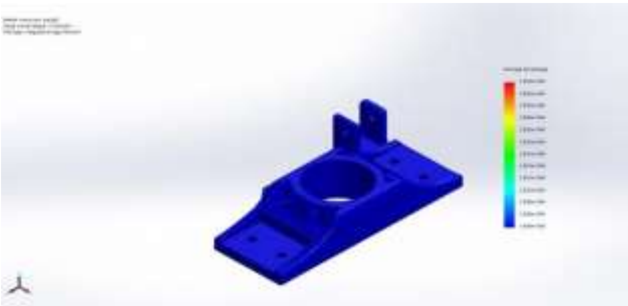


Fig – 22: Rear upright Fatigue analysis (Damage)

Rear upright has analyzed with **10000000000 cycles**. It is observed that Maximum Damage at 4e+007 cycle.

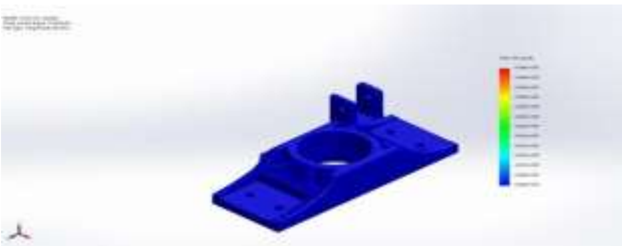


Fig – 23: Rear upright Fatigue analysis (Life)

C. Wheel Hub

Wheel hub dimension is same for both front and rear upright. Rear hub has spline cut in it to accommodate axle inside it. Remote 3G load is applied on the hub to carry out the analysis. Image below shows the static analysis being conducted to hub to test its performance during the application of load. Weight of the vehicle is assumed as 280kg. Torsion is applied on the brake disc mounting point, which in this case is assumed as 300N-m. The hub is constrained at the center, where the bearing fits on it and a remote load of 1500 N as longitudinal force, 2000 N as lateral and 2500 N as vertical or bump force is applied on it. Maximum stress of 1.84496e+008 N/m^2, maximum displacement of 0.54321 mm and minimum Factor of safety as 2.7 is recorded in the analysis. Figure 20 shows the static analysis performed on it.

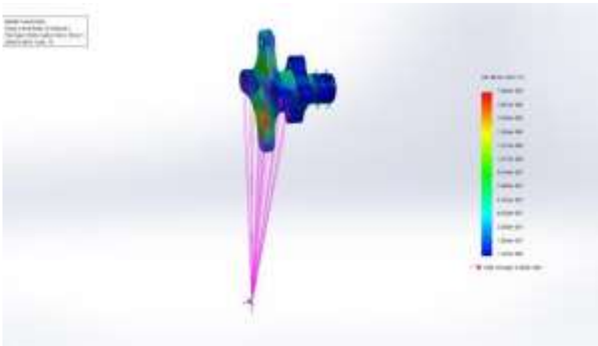


Fig - 24: Wheel hub Static analysis

Wheel hub has analyzed with **1000000000 cycles**. It is observed that Minimum Life is 2500 cycle and Maximum Life is 1.42857e+007 cycle.

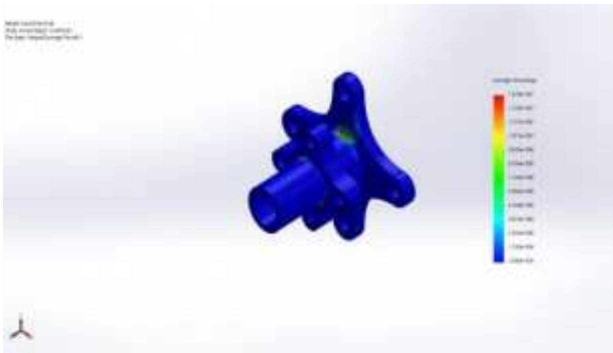


Fig – 25: Wheel hub Fatigue analysis (Damage)

Wheel hub has analyzed with **1000000000 cycles**. It is observed that Minimum Life is 7000 cycle and Maximum Life is 4e+007 cycle.

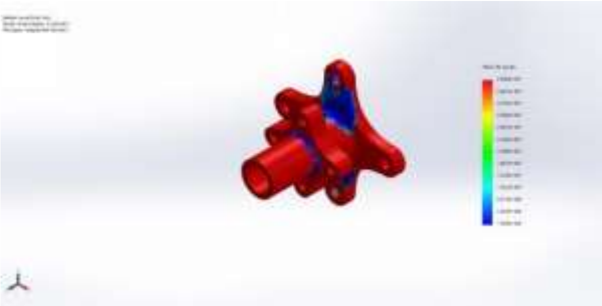


Fig – 26: Wheel hub Fatigue analysis (Life)

D. Upright Bracket

Brackets is separately mounted on the upright to avail the camber adjustment option and so it is also analyzed with the load conditions. In these two conditions is assumed for loading. The first option is the rear portion of the bracket is fixed and the load is applied on the wishbone mounting points. In the second case, the hole which is created on the bracket to mount it on the upright is made fixed and then, the load is applied on the wishbone mounting points. Below is the figure which shows two of the cases.

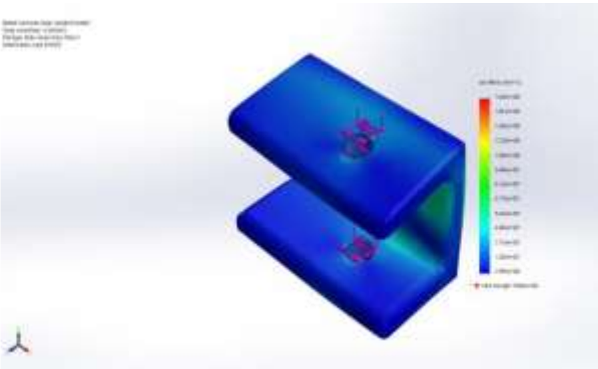


Fig -27: Rear portion is fixed of the bracket

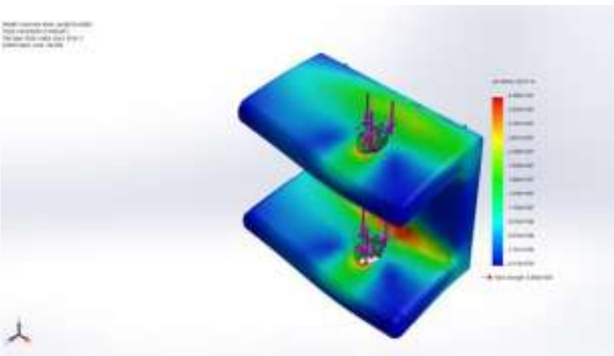


Fig -28: Bracket mounting point is fixed of the bracket

E. Front Rocker

Bell Crank transfers the force to the spring which it receives from the push or pull rod. So, it is necessary to analyze all these components to check its behavior under various loading conditions.

Maximum stress of 4.26015e+007 N/m^2, maximum displacement of 0.0292501 mm and minimum Factor of safety as 11.854 is recorded in the analysis.

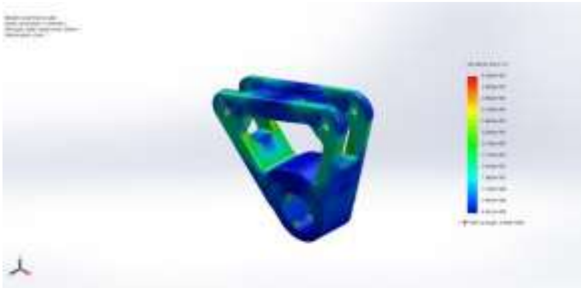


Fig – 29: Front Rocker Static analysis

Fatigue analysis has also been performed on the front rocker with the following results:

Front rocker has analyzed with **1000000000 cycles**. It is observed that Minimum damage is 2500 cycle.

Fig – 30: Front rocker Fatigue Analysis (Damage)

Front rocker has analyzed with **1000000000 cycles**. It is observed that Minimum life is 4e+007 cycle.

Fig – 31: Front rocker Fatigue Analysis (Life)

F. Rear Rocker

Maximum stress of 5.64733e+007 N/m^2, maximum displacement of 0.0227842 mm and minimum Factor of safety as 8.9 is recorded in the analysis.

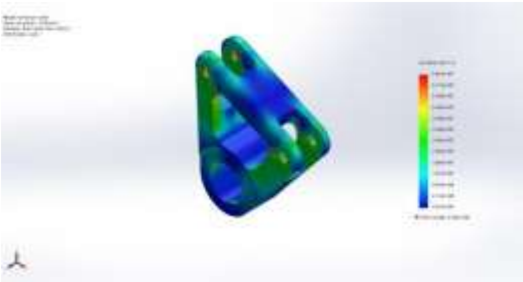


Fig- 32: Rear Rocker Static analysis

Fatigue analysis has also been performed on the rear rocker with the following results:

Rear rocker has analyzed with **100000000 cycles**. It is observed that Minimum damage is 250 cycle.

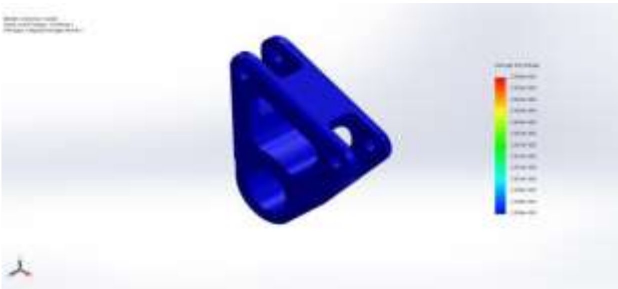


Fig – 33: Rear rocker Fatigue Analysis (Damage)

Rear rocker has analyzed with **100000000 cycles**. It is observed that Minimum life is 4e+007 cycle.

Fig – 34: Rear rocker Fatigue Analysis (Life)

G. Steering System & Wheel Geometry

Steering system is used to control the direction of the vehicle when the vehicle is in motion. There are different types of steering systems used in automotive industry, amongst them for formula student vehicle; rack pinion type steering is widely used.

While designing, the major factor is the type of geometry to be used for the steering system. The three possible geometries that can be used are Ackermann, anti- Ackermann steer geometry.

As the event consists of more low speed corners it was decided to use Ackermann steering geometry as in this geometry the inner tire turns more as compared to the outer tire thus giving an added advantage for tracks with low speed turns.

Now that the geometry has been decided the percent Ackermann has to be decided. 100% Ackermann is considered to be the best solution for low speed maneuvers but due to design constraints, an Ackermann percent of around 60 to 80 percent was considered to be the best solution.

Let us assume either the inner angle of turning radius or the turning radius itself to proceed for the steering calculation. In this case, the inner angle θ_i as 45^o. Now, that we have the inner turning angle, we will find out the outer steering angle and the turning radius. Below is the figure given to illustrate the steering geometry.

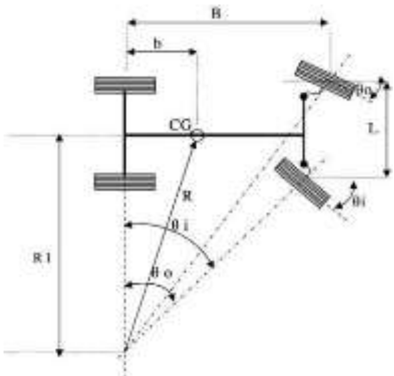


Fig – 35: Steer angle for Ackerman principle

Where,

θ_0 = turn angle of the wheel on the outside of the turn

θ_i = turn angle of the wheel on the inside of the turn

B= track width
L = wheel base

b = distance from rear axle to center of mass
Now, b = 633mm; L = 1200mm; B = 1600mm

We know, $R = \sqrt{(R1)^2+b^2}$ -------(1)

$R1 = B/\tan(\theta i) + L/2$ mm

$= 1600/\tan(45^0) + 1200/2$ mm

$= 1600 + 600$

$= 2200$ mm

Putting the value of R1 in equation

(1), $R = \sqrt{(2200)^2+(633)^2}$ mm = 2.9 m

We know,

$\text{Cot}(\theta 0) - \text{Cot}(\theta i) = L/B$

$\Rightarrow \text{Cot}(\theta 0) - \text{Cot}(\theta i) = 1200/1600$

$\Rightarrow \text{Cot}(\theta 0) = 1.75$

$\Rightarrow \theta 0 = \text{Cot}^{-1}(1.75)$

$\Rightarrow \theta 0 = 29.7^0$

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.

Assuming 45^0 to be maximum turn and steering wheel movement to be 270^0, the steering ratio can be calculated as:

$SR = 270^0 / 45 = 6:1$

As the steering ratio has been calculated, then Rack Travel needs to be calculated.

Radius of steering wheel = 120mm

So, Steering wheel travel to 1 complete rotation

$= 3\pi R = 360 * 3.14 = 1130.4\text{mm} = 1.1304\text{m}$

Now, we can calculate racktravel.

Rack Travel = (Steering wheel travel) / Steering Ratio = 1.1304/6 = 0.188m = 188mm

So, the rack travel is 188mm.

Now, Diameter of steering wheel = 240mm Weight on one wheel = 50kg = 500N Radius of pinion = 21mm

Maximum coefficient of friction μ = 1 Torque on pinion = 500*21 N = 10500N

Force on steering wheel= Torque/Radius of steering wheel = 10500 / 120 = 87.5N

The torque on steering wheel is 87.5N which is under a pretty 100N decent value.

Parameter of the steering Rack and Pinion gear is as follows:

Sl. No.	Parameter	Value
1	Steering wheel lock to lock angle	540
2	Maximum rack Travel (mm)	188
3	Steering Ratio	6:01
4	Turning Radius(m)	2.9
5	Innner wheel turning angle	45
6	Outer wheel turning angle.	29.71

Table 9: Steering parameters

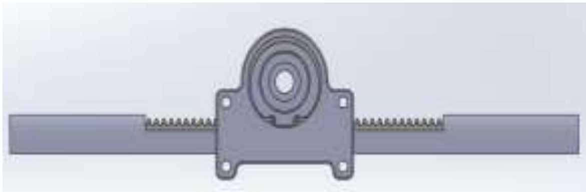


Fig 36: Front view of Rack Pinion steering system

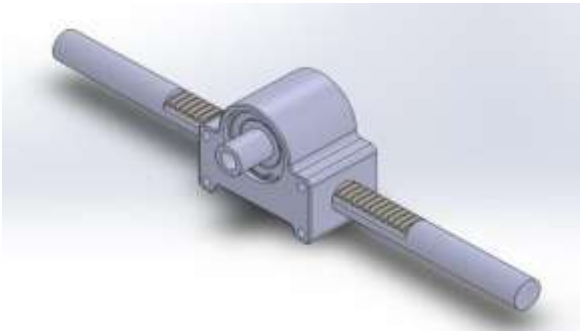
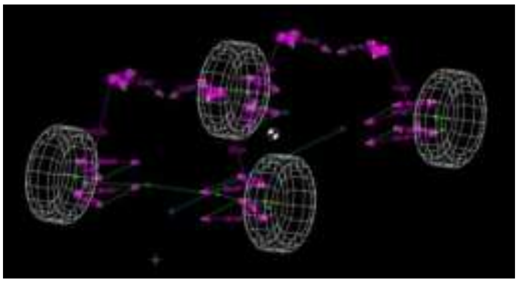


Fig 37: Isometric view of Rack Pinion Steering system

Wheel geometry defines how the vehicle will behaveat the time of cornering at high speeds. So, it is very important to fix the wheel geometry judiciously to enhance the vehicle stability at high speed. Below are the parameters fixed for the vehicle.

Table 10. Static Wheel parameters

Sl. No.	Wheel Geometry	Front Wheel	Rear Wheel
1	Camber Angle	-2	0
2	Castor Angle	6.22	2.12
3	Scrub Radius	71.56	70.75
4	King Pin Angle	5	2



Fig–38: LOTUS SHARK analysis of the wheel geometry

With the above-mentioned data, the steering wheel is designed using 3D printing technology. This technology is used to produce the exact product as designed. Below is the design of the 3D printed steering wheel.

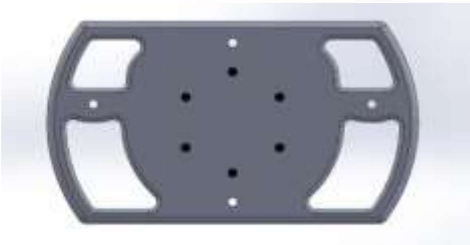


Fig – 39: 3D Printed Steering Wheel

H. Braking System

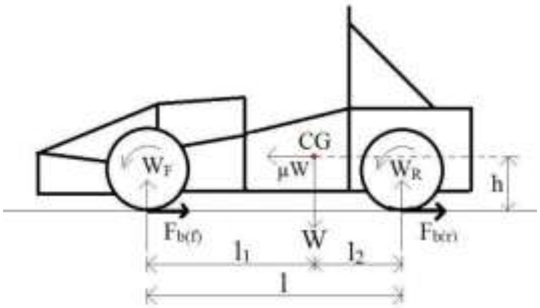


Fig – 40: Braking Geometry

Calculations are performed considering initial velocity at 60 kmph and final velocity of 0 kmph. During braking there is a weight transfer due to which load on front axle increases.

Load on front tyres,

$F(f) = W_C - F_r + W_C*\mu*Y_{CG}/l$

$= 300 - 180 + 300*0.7*0.27/1.6$

$= 155.4$ kg $= 1525$N

Where, W_C = Weight of the car;

F_r = Load on rear wheels;

Y_{CG} = Height of Center of Gravity from ground.

Load on rear tyres,

$F(r) = (300*0.96/1.6) - (300*0.7*0.27)$

$= 144.5$ kg $= 1418.13$N

Weight due to load transfer on front tyres

$= W_C*\mu*Y_{CG}/l = 300*0.7*0.27/1.6$

$= 35.4375$ kg $= 348.2$N

Total weight on front tyres due to braking

$= 1525+348.2$ N $= 1873.2$ N

Torque on wheels,

$T_F = (F_f/2) * R_{wheel} = (1875.2/2) * 0.229$N

$= 214.48$N

Where, R_{wheel} = Radius of wheel.

$T(r) = F_r * R_{wheel} / 2 = 1418*0.229/2$ Nm

$= 162.3$ Nm

Considering a pedal force of 150N Pedal ratio = 4:1

Force on master cylinder push rod = 4*150 N = 600N Diameter of master cylinder piston = 18mm Hydraulic pressure at master cylinder,

P_{mc} = (Master cylinder force / Area of piston)

$= 600/254.5$ N/mm^2

$= 2.357$ N/mm^2 or MPa

Brake Caliper Calculation

Diameter of piston = 34mm Force at caliper,

$F_{cal} = P_{mc}$*Area of piston*No. of piston

$= 2.35*3.14*17*17*2$ N

$= 2.35*908*2$ N

$= 4267.5$ N

Frictional force $= \mu*2*F_{cal}$ (For 2 piston)

$= 0.7*2*4267.5$ N

= 2560.5 N

Torque = Frictional force *
reffective reffective = (r1 + r2) /
2 = 0.095m

Braking torque at the disc= 2560.5*0.095

= 243.2475Nm

- ***Selection of Brake Disc and Analysis***

There are various Brake Disc materials available in the market but we have chosen SS420. Grade 420 stainless steel is a high-carbon steel with a minimum chromium content of 12%. Like any other stainless steel, grade 420 can also be hardened through heat treatment. It offers good ductility in its annealed state and excellent corrosion resistance properties when the metal is polished, surface grounded or hardened. This grade has the highest hardness among all the stainless-steel grades with 12% chromium.

Martensitic stainless steels are ones with high hardness and high carbon content. These steels are generally fabricated using methods that require hardening and tempering treatments. The operating conditions of martensitic steels are affected by loss of material's strength at high temperatures, and decrease in ductility at negative temperatures.

Table 11: Mechanical properties of grade 420 stainless steels

Tempering Temperature (°C)	Tensile Strength (MPa)	Yield Strength (MPa)	Elongation (% in 50mm)	Hardness Brinell (HB)
Annealed	655	345	25	241 Max

Reasons for selecting SS420 material:

Under hardened conditions, grade 420 steels are resistant to fresh water, alkalis, air, foods and mild acids. The steel grades with a smooth surface finish has excellent performance. The corrosion resistance properties of grade 420 will tend to fall under annealed conditions.

Grade 420 stainless steels have a scaling resistance at temperatures of up to 650°C. However, temperatures above standard tempering temperature are not suitable for this grade.

Grade 420 steels can be easily machined in their annealed state, but they are difficult to machine having a hardness greater than 30HRC.

Analysis of Brake Disc

Brake disc experiences torque as well as thermal stresses. Therefore, it becomes mandatory to analyze the brake disc for static stress and thermal simulation to achieve better performance.

Fig – 41: Brake disc static analysis

Maximum stress of 7.59301e+007 N/m^2, maximum displacement of 0.0174545 mm and minimum Factor of safety as 4.6 is recorded in the analysis.

Fig – 42: Brake Disc thermal analysis

Minimum Temperature of brake disc is 52.5212 Celsius and Maximum Temperature is 125.26 Celsius.

This car can be driven by a person whose weight is 100kg. So, we are assuming total weight of the car with driver is 270kg.

Kinetic energy K.E = $mv^2/2$ = 0.5*270*30.5*30.5

= 125583.75 J

[v = 110kmph = 30.5mps]

Heat power (Total) = K.E / dt = 125583.75/3

= 41861.25 W

Heat Power one brake = 0.6*41861.25/2 = 12558.375 W During Computational Fluid Dynamics (CFD), it is observed that the wheel assembly section has convection coefficient is 230W/m^2K.

Stopping Distance

Stopping distance = V^2 / 2μg

= $(120*1000/3600)^2$ / (2*0.6*9.81)

= 94.3m

Now, consider the speed to be 60kmph

= $(60*1000*3600)^2$ / (2*0.6*9.81)

= 23.59 m

I. Transmission System

KTM 390 Gear Ratio:

Primary reduction = 30/80 = 2.66

1st Gear = 2.66

2nd Gear = 1.85

3rd Gear = 1.421

4th Gear = 1.142

5th Gear = 0.956

6th Gear = 0.84

Final reduction = 3.33

Final sprocket ratio = 3.33 (Secondary reduction)

Radius of tyre = 9 inch = 0.2286m

Effective tyre radius = 96 % of 0.2286 m

= 0.219 m

- **Torque Calculation**

T = Taverage * Primary reduction*secondary reduction* Gear reduction

Torque at 1st Gear = 37*2.66*3.33*2.66

= 871 Nm

Torque at 2nd Gear = 37*2.66*3.33*1.85

= 606.31 Nm

Torque at 3rd Gear = 37*2.66*3.33*1.421

= 465.71 Nm

Torque at 4th Gear = 37*2.66*3.33*1.142

= 374.27 Nm

Torque at 5th Gear = 37*2.66*3.33*0.956

= 313.31 Nm

Torque at 6th Gear = 37*2.66*3.33*0.84

= 275.30 Nm

- **Velocity Calculation**

KTM 390 Specifications:

Max Power – 43 bHP @ 9000 rpm

Max Torque – 37 Nm @ 7000 rpm

Velocity V = (2πRN * 60)/(1000*Primary reduction*secondary reduction*selected gear ratio*μ)

Velocity at 1st Gear = (2*3.14*0.2286*9000*60) / (1000*2.66*3.33*2.66*0.85)

= 38.7 Kmph

Velocity at 2nd Gear = (2*3.14*0.2286*9000*60) / (1000*2.66*3.33*1.85*0.85)

= 55.65 Kmph

Velocity at 3rd Gear = (2*3.14*0.2286*9000*60) / (1000*2.66*3.33*1.421*0.85)

= 72.45 Kmph

Velocity at 4th Gear = (2*3.14*0.2286*9000*60) / (1000*2.66*3.33*1.142*0.85)

= 90.161 Kmph

Velocity at 5th Gear = (2*3.14*0.2286*9000*60) / (1000*2.66*3.33*0.956*0.85)

= 107.7 Kmph

Velocity at 6th Gear = (2*3.14*0.2286*9000*60) / (1000*2.66*3.33*0.84*0.85)

= 122.5 Kmph

- ***Engine RPM Calculation***

Diameter of Tyre = 18 inch =0.45m

Radius of Tyre = 9 inch

Circumference = 22/7 * 0.45m = 1.413m

At 80kmph tyre will rotate at = (80000/60)/1.413

= 943.618 rpm

For 1 kmph = 943.618/80 = 11.79rpm.

Now engine rpm for 100 kmph

= 11.79*2.66*3.33*0.84*100

= 8780 rpm

At 122 kmph = 10711.4 rpm

- **RPM calculations for Gears**

RPM for 1st Gear = 11.79*2.66*3.33*0.84*38.7

= 3374.9 rpm

RPM for 2nd Gear = 87.7*55.65 = 4880 rpm

RPM for 3rd Gear = 87.7*72.45 = 6353.8 rpm

RPM for 4th Gear = 87.7*90.161 = 7907 rpm

RPM for 5th Gear = 87.7*107.7 = 9445.2 rpm

RPM for 6th Gear = 87.7*122.5 = 10743.25 rpm

Sprocket

A **sprocket** or **sprocket-wheel** is a profiled wheel with teeth, or cogs, that mesh with a chain, track or other perforated or indented material. The name 'sprocket' applies generally to any wheel upon which radial projections engage a chain passing over it. It is distinguished from a gear in that sprockets are never meshed together directly, and differs from a pulley in that sprockets have teeth and pulleys are smooth.

Fig – 43: Sprocket

- ***Calculation of Sprocket pitch diameter***

The sprocket pitch diameter is an imaginary circle through which the chain pin centers move around the sprocket. The pitch diameter is the fundamental design geometry that determines the size shape and form of the sprocket teeth dimensions. Below is the basic diagram given to understand the terminology related to sprocket.

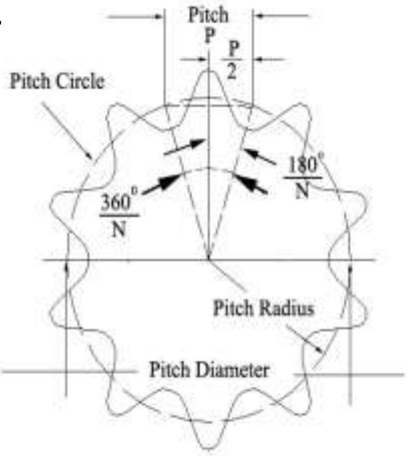


Fig – 44: Sprocket pitch diameter calculation

PD = P / sin(180^0/N)

Where PD = Pitch diameter

P = Chain pitch in inches

N = No. of teeth in the sprocket

Chain used in KTM DUKE 390 has a pitch of 5/8 inch. So, the driven sprocket needs to be designed according to the pitch. Number of teeth is assumed is 50. The calculation for driven sprocket pitch diameter is given below.

PD = 0.625/sin (180/50) = 9.953 inch = 252.82 mm

Outer sprocket diameter = P(0.6+cot(180^0/N))

= 0.625(0.6+cot (180/50))

= 10.3 inch

= 261.8mm or 262mm

Sprocket Analysis

It is very important to analyze sprocket to check whether it can sustain the maximum torque produced by the Engine. Below are the steps followed to analyze the sprocket:

1. Calculate the amount of torque on the shaft over which the sprocket is being mounted.
2. Divide the torque by radius of the sprocket to get the force.
3. Divide the force value by number of teeth engaged with the chain. That will be the value of force on a single teeth. Apply it tangentially on each teeth fixing the center face. For this, consider only the teeth engaged. Don't consider total number of teeth.

Fig – 45: Static analysis of sprocket

Maximum stress of 1.3597e+008 N/m^2, maximum displacement of 0.107399 mm and minimum Factor of safety as 3.7 is recorded in the analysis.

Differential hanger is used to hold the differential of the vehicle in one particular position. It is also analyzed and the procedure of analysis is given below:

1. Fixed the points which will be mounted on chassis.
2. Apply 880 Nm torque to the hollow section.
3. Apply 500 N force downwards due to the gravity.
4. Click on "Mesh"
5. Select "Apply Material" and Click on "Aluminum 6063-T6"

Run the process and determine FOS.

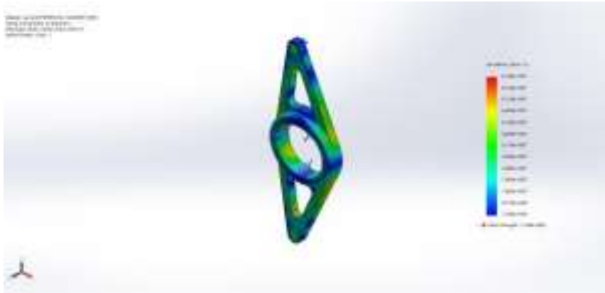


Fig – 46: Differential hanger static analysis

Maximum stress of 6.25022e+007 N/m^2, maximum displacement of 0.102322 mm and minimum Factor of safety as 3.4 is recorded in the analysis.

J. INTAKE SYSTEM RESTRICTOR

An **air restrictor** is a system installed at the intake of an engine to limit its power. This kind of system is used in automobile racing, to limit top speed to provide equal level of competition, and to lower costs.

The FFSINDIA 2019 competitions rules dictates that:

1. If more than one engine is used, the air for all engines must pass through a single air intake restrictor.

2. In order to limit the power capability of the engine, a single circular restrictor must be placed in the intake system and all engine airflow must pass through the restrictor. The only allowed sequence of components are the following:
(a) For naturally aspirated engines, the sequence must be: throttle body, restrictor, and engine.
(b) For turbocharged or supercharged engines, the sequence must be: restrictor, compressor, throttle body, engine.

3. The maximum restrictor diameters which must be respected at all times during the competition are:
(a) Gasoline-fueled vehicles - 20mm
(b) E-85 fueled vehicles - 19mm

4. The restrictor must be located to facilitate measurement during the inspection process.

5. The circular restricting cross-section may not be movable or flexible in any way, e.g. the restrictor must not be part of the movable portion of a barrel throttle body.

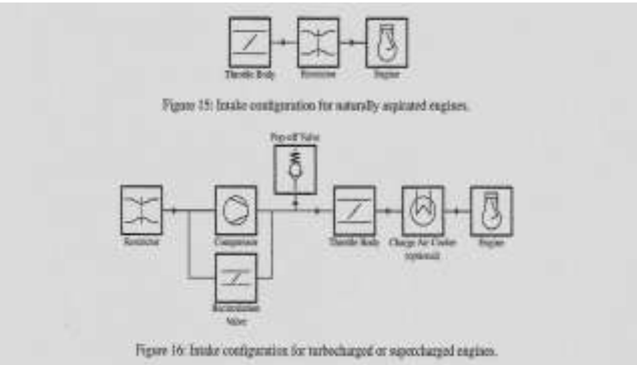


Fig – 47: Intake Configuration

TURBOCHARGERS AND SUPERCHARGERS

1. The intake air may be cooled with an intercooler. Only ambient air may be used to remove heat from the intercooler system. Air-to-air and water-to-air intercoolers are permitted. The coolant of a water-to-air intercooler system must be pure water without any additives.

2. If pop-off valves, recirculation valves, or heat exchangers (intercoolers) are used, they may only be positioned in the intake system as shown in Figure 47.

3. Plenums anywhere upstream of the throttle body are prohibited. A "plenum" is any tank or volume that is a significant enlargement of the normal intake runner system.

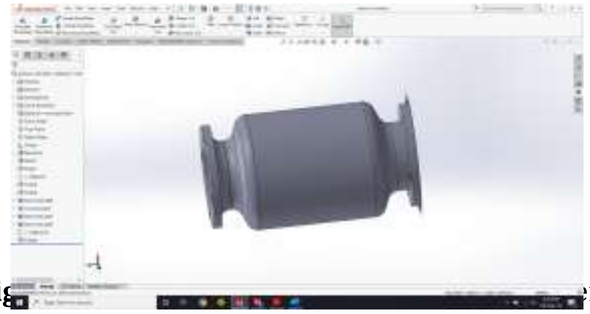
4. The maximum allowable internal diameter of the intake runner system between the restrictor and throttle body is 60mm diameter, or the equivalent area of 2827mm2 if non-circular.

CRANKCASE/ENGINE LUBRICATION VENTING

1. Any crankcase or engine lubrication vent lines routed to the intake system must be connected upstream of the intake system restrictor.

2. Crankcase breathers that pass through the oil catch tank(s) to exhaust systems, or vacuum devices that connect directly to the exhaust system, are prohibited.

Keeping all these things in mind, Planum Chamber, Restrictor and Throttle body is designed in such a way that it can follow all the rules.



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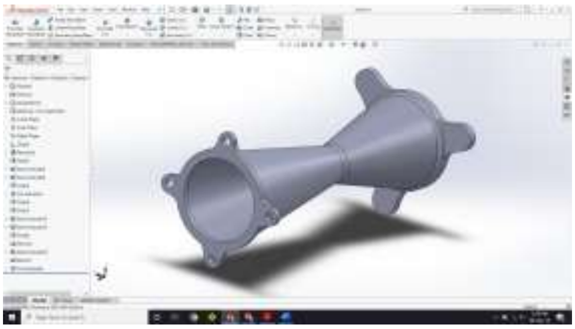

Fig – 49: Design Configuration of restrictor

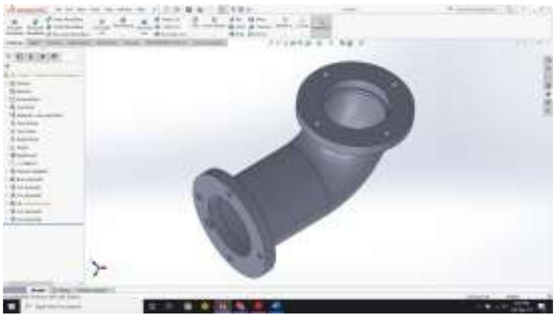

Fig – 50: Design Configuration of runner

The assembly view of intake system is given below:

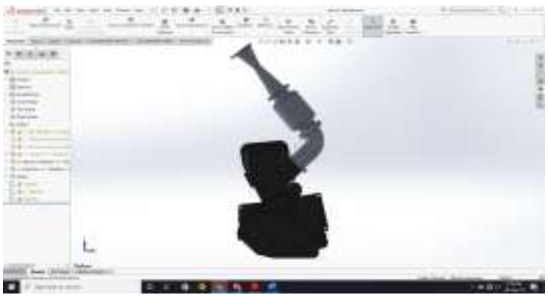

Fig – 51: Assembly view of intake system

Simulation of intake system is the most important thing. There are some steps need to follow for simulating the intake system are given below:

1. Create Lids and select opening side of restrictor and engine runner in Flow Simulation.
2. Select Wizard and adjust computational domain.

3. Insert Boundary condition for Inlet velocity and Environmental pressure.
4. Define Velocity flow rate in surface goals.
5. Insert equation.
6. Run the analysis.
7. Insert cut plots and select the opening mouth of the restrictor and select required plane.

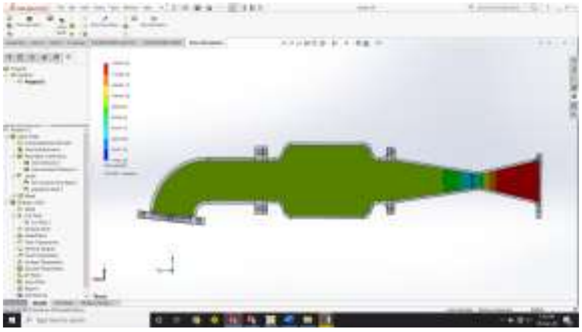

Fig – 52: Simulation of Intake System

K. FUEL TANK

A fuel tank or petrol tank is a safe container for flammable fluids. There are few rules of designing the fuel tank. They are given below:

CV2 FUEL AND FUEL SYSTEM
FUEL

1. The available fuel types will be unleaded gasoline 98RON and E85.

2. The vehicles must be operated with the fuels provided at the competition.

3. No agents other than fuel (gasoline or E85), and air may be induced into the combustion chamber.

4. The temperature of fuel introduced into the fuel system may not be changed with the intent to improve calculated efficiency.

FUEL TANKS

1. The fuel tank is defined as that part of the fuel containment device that is in contact with the fuel. It may be made of a rigid material or a flexible material.

2. Fuel tanks made of a rigid material cannot be used to carry structural loads and must be securely attached to the vehicle structure with mountings that allow some flexibility such that chassis flex cannot unintentionally load the fuel tank.

3. Any fuel tank that is made from a flexible material, for example, a bladder fuel cell or a bag tank, must be enclosed within a rigid fuel tank container which is securely attached to the vehicle structure. Fuel tank containers (containing a bladder fuel cell or bag tank) may be load carrying.

4. The fuel system must have a provision for emptying the fuel tank if required.

5. The fuel tank, by design, must not have a variable capacity.

FUEL LINES FOR LOW-PRESSURE SYSTEMS

1. Fuel lines between the fuel tank and fuel rail and return lines must have:
• Reinforced rubber fuel lines or hoses with an abrasive protection with a fuel hose clamp which has a full 360° wrap, a nut and bolt system for tightening and rolled edges to prevent the clamp cutting into the hose, or
• Metal braided hoses with crimped-on or reusable, threaded fittings.

2. Fuel lines must be securely attached to the vehicle and/or engine.

3. All fuel lines must be shielded from possible rotating equipment failure or collision damage.

FUEL INJECTION SYSTEM REQUIREMENTS

Low-Pressure Injection (LPI) fuel systems are those functioning at a pressure below 10bar and High-Pressure Injection (HPI) fuel systems are those functioning at 10bar pressure or above. Direct Injection (DI) fuel systems are those where the injection occurs directly into the combustion chamber.

1. The following requirements apply to LPI fuel systems:
• The fuel rail must be securely attached to the engine cylinder block, cylinder head, or intake manifold with mechanical fasteners. The threaded fasteners used to secure the fuel rail are considered critical fasteners and must comply with T9.
• The use of fuel rails made from plastic, carbon fiber or rapid prototyping flammable materials is prohibited. However, the use of unmodified Original Equipment

Manufacturer (OEM) Fuel Rails manufactured from these materials is acceptable.

2. The following requirements apply to HPI and DI fuel systems:

• All high-pressure fuel lines must be stainless steel rigid line or Aeroquip FC807 smooth bore PTFE hose with stainless steel reinforcement and visible Nomex tracer yarn. Use of elastomeric seals is prohibited. Lines must be rigidly connected every 100mm by mechanical fasteners to structural engine components.

• The fuel rail must be securely attached to the engine cylinder head with mechanical fasteners. The fastening method must be sufficient to hold the fuel rail in place with the maximum regulated pressure acting on the injector internals and neglecting any assistance from in-cylinder pressure acting on the injector tip. The threaded fasteners used to secure the fuel rail are considered critical fasteners and must comply with T9.

• The fuel pump must be rigidly mounted to structural engine components.

• A fuel pressure regulator must be fitted between the high and low-pressure sides of the fuel system in parallel with the DI boost pump. The external regulator must be used even if the DI boost pump comes equipped with an internal regulator.

• Prior to the tilt test specified in IN7, engines fitted with mechanically actuated fuel pumps must be run to fill and pressure the system downstream of the high-pressure pump.

FUEL SYSTEM LOCATION REQUIREMENTS

1. All parts of the fuel storage and supply system must lie within the surface defined by the top of the roll bar and the outside edge of the four tires. In side view, no portion of the fuel system can project below the lower surface of the frame.

2. All fuel tanks must be shielded from the side or rear impact collisions. Any fuel tank which is located outside the side impact structure required by T2.16 must be shielded by a structure built to T2.16. Any portion of the fuel system that is less than 350mm above the ground must be within the primary structure.

3. All parts of the fuel storage and supply system must be adequately protected against any heat sources and located at least 70mm from any exhaust system component.

FUEL TANK FILLER NECK AND SIGHT TUBE

1. All fuel tanks must have a filler neck which is:

• At least 35mm diameter at any point between the fuel tank and the top of the fuel filler cap.

• At least 125mm vertical height above the top level of the tank.

• Angled at no more than thirty degrees (30°) from the vertical an

• accompanied by a clear fuel resistant sight tube with a length of at least 125mm vertical height for reading the fuel level.

2. A clear filler neck tube may be used as a sight tube.

3. A permanent, non-moveable, clear and easy visible fuel level line must be located between 12mm and 25mm below the top of the visible portion of the sight tube. This line will be used as the fill line for the tilt test and before and after the endurance test to measure the amount of fuel used during the endurance event.

4. The filler neck opening must be directly accessible without removing any parts of the vehicle except for the fuel filler cap.

5. The filler neck must have a fuel filler cap that can withstand severe vibrations or high pressures such as could occur during a vehicle rollover event.

TANK FILLING REQUIREMENT

1. The fuel tank must be capable of being filled to capacity without manipulating the tank or the vehicle in any manner. The fuel system must be designed in a way that during refueling of the vehicle on a level surface, the formation of air cavities or other effects that cause the fuel level observed at the sight tube to drop after movement or operation of the vehicle (other than due to consumption) is prevented.

2. The fuel system must be designed such that the spillage during refueling cannot contact the driver position, exhaust system, hot engine parts, or the ignition system.

3.Belly pans must be vented to prevent accumulation of fuel. At least two holes, each of a minimum diameter of 25mm, must be provided in the lowest part of the structure in such a way as to prevent accumulation of volatile liquids.

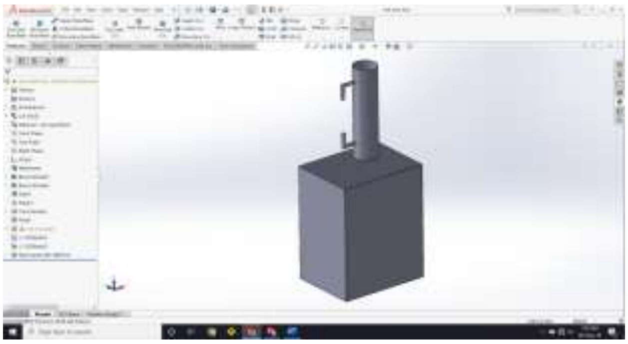

Fig – 53: Design of fuel tank

L. AERODYNAMICS DEVICES

Aerodynamics is study of motion of air. The word "Aerodynamics" is taken from the Greek word "Aero" which means "Air" and the word "*Dynamikos*" which means "Dynamics".

To get maximum output out of the car, a high lift and maximum amount of drag is required. So proper wings profile should be designed for better output. FX74 CL5 140 wings profile is used for base wing and E423 is used for the flap wings. The cross-section of the wings profile is given below:

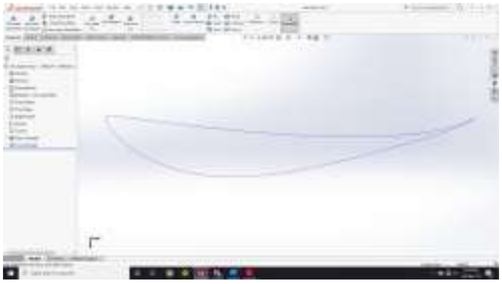

Fig – 54: Wings Profile for FX74 CL5 140

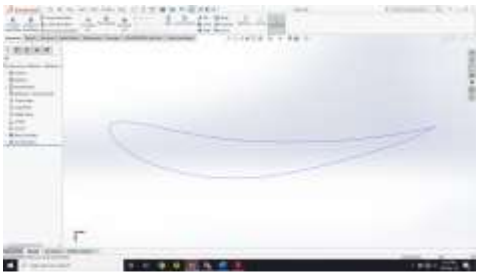

Fig – 55: Wings Profile for E423

I. 2D AERO FOIL CONFIGURATION

The FFSINDIA 2019 competitions rules dictates that:
1. Vehicles with aerodynamic devices and/or environment perception sensors in front of the IA must not exceed the peak deceleration of the combination of their IA assembly and the non-crushable object(s). Any of the following three methods may be used to prove the design does not exceed 120kN: (a) Physical testing of the IA assembly including any attached non-crushable object(s) in front of the AIP. (b) Combining the peak force from physical testing of the IA assembly with the failure load for the mounting of the non-crushable object(s), calculated from fastener shear and/or link buckling. (c) Combining the "standard" IA peak load of 95kN with the failure load for the mounting of the non-crushable object(s), calculated from fastener shear and/or link buckling.

DEFINITION AERODYNAMIC DEVICE:

1. A specifically designed structure mounted on the vehicle to guide the airflow around the vehicle, increasing the downforce on the vehicle and/or lowering its drag. The mounting of this structure is not regarded as an aerodynamic device unless it is intentionally designed to be one.

Ground Effect Devices
1. Power ground effects are prohibited. No power device may be used to move or remove air from under the vehicle except fans designed exclusively for cooling.

RESTRICTIONS FOR AERODYNAMIC DEVICES:

1. HEIGHT RESTRICTIONS:

- All aerodynamic devices forward of a vertical plane through the rearmost portion of the front face of the driver head restraint support, excluding any padding, set to its most rearward position, must be lower than 500mm from the ground.
- All aerodynamic devices in front of the front axle and extending further outboard than the most inboard point of the front tire/wheel must be lower than 250mm from the ground.
- All aerodynamic devices rearward of a vertical plane through the rearmost portion of the front face of the driver head restraint support, excluding any padding, set to its most rearward position must be lower than 1.2m from the ground.

2. WIDTH RESTRICTIONS:

• All aerodynamic devices lower than 500mm from the ground and further rearward than the front axle, must not be wider than a vertical plane touching the outboard face of the front and rear wheel/tire.
• All aerodynamic devices higher than 500mm from the ground, must not extend outboard of the most inboard point of the rear wheel/tire.

3. LENGTH RESTRICTIONS:

All aerodynamic devices must not extend further rearward than 250mm from the rearmost part of the rear tires.
4. All restrictions must be fulfilled with the wheels pointing straight and with any suspension setup with or without driver seated in the vehicle.

MINIMUM EDGE RADII OF AERODYNAMIC DEVICES:

1. All forward facing edges of aerodynamic devices that could contact a pedestrian must have a minimum radius of 5mm for all horizontal edges and 3mm for vertical edges.

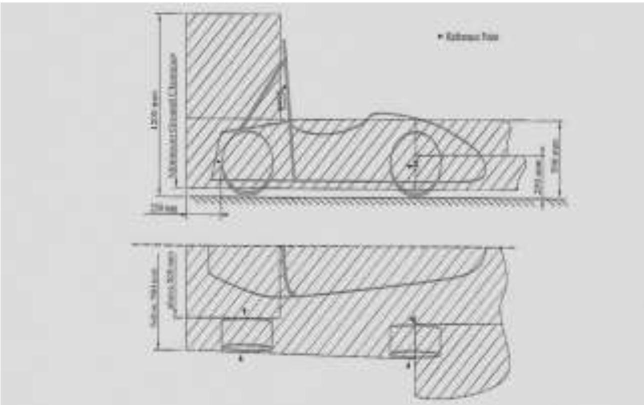

Fig – 56: Overview of aerodynamic devices

AERODYNAMIC DEVICES STABILITY AND STRENGTH:

1. Any aerodynamic device must be able to withstand a force of 200N distributed over a minimum surface of 225cm2 and not deflect more than 10mm in the load carrying direction.
2. Any aerodynamic device must be able to withstand a force of 50N applied in any direction at a point and not deflect more than 25mm.

2. SIMULATION

Simulation is the most important thing for aerodynamics. In Solidworks, Computational Fluid Dynamics (CFD) simulation for both rear and front wings. The steps of CFD simulation is given below:

- Open Flow Simulation in Solidworks. Click on Wizard.
- Select Next and define unit system.
- Select External and define reference axis.
- Add Air from Gases and provide velocity 25m/s
- Edit Computational Domain and define Global Goals for X, Y &Z axis.
- Run the analysis. Apply Global Goals.
- Define Spheres from Flow Trajectories.
- Play the Analysis.

After analyzing the wings, following results are found:

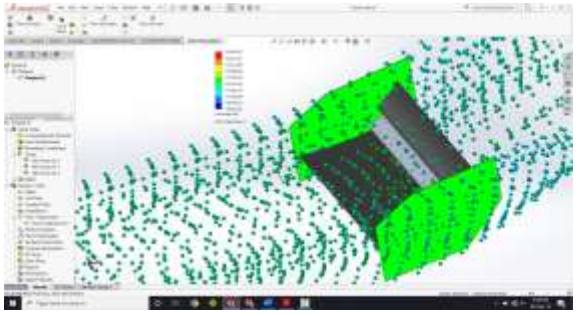

Fig – 57: Rear Wings flow simulation

It is observed that through this design, the value of drag is 94.1597 N and the value of downforce is 31.898 N.

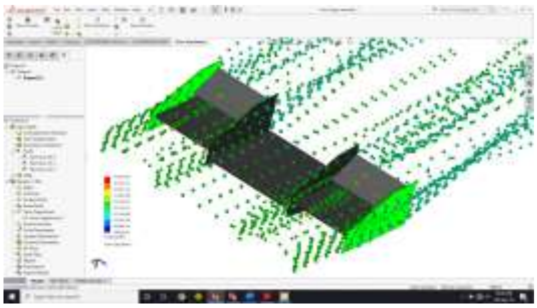

Fig – 58: Front Wings flow simulation

It is observed that through this design, the value of drag is 31.589 N and the value of downforce is 108.765 N.

The complete CAD overview of the vehicle is given below:

Fig – 59: Overview of the car

CONCLUSION

The first part of the paper discusses about the method of selection of various components of the Chassis material selection, analysis of all the components used in the racing car. The second part is all about the aerodynamics simulation. These methods were then verified using SOLIDWORKS CFD analysis.

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