Formula Bharat 2024 Supporting Document

EV37-IITK Motorsports

December 11, 2023

Contents

Chapter 1

Chassis and Aerodynamics

1.1 Frame Impact Tests :

Each impact test is a worst case scenario that could potentially occur at the competition. There are four tests: a drop test, front collision test, rear impact test, and side impact test. The three collision tests simulate different 16 m/s impacts with stationary objects or other vehicles.

The front collision test simulates the vehicle hitting a solid, immovable object at a speed of 16 m/s. This is the maximum top speed the vehicle is expected to reach. The rear impact test simulates the vehicle being rear-ended by another 227 kg Formula vehicle, again at a speed of 16 m/s. To make this test as difficult as possible, the front of the vehicle is resting against a solid wall. The side impact test is identical to the rear impact, but the vehicle is oriented sideways relative to the motion of the incoming 227 kg vehicle. In reality the wheels and suspension of the vehicle would absorb some of the energy in the side impact test, but these were removed from the simulation to make it an absolute worst-case scenario.

For the impact tests, the following Eq1.1 is used to calculate the force on the vehicle. An impulse time of 0.3 seconds was used.

$$F = \frac{m.v}{t} \tag{1.1}$$

Where:

F : Force
m : Mass
v : Initial Velocity
t : Impulse Time

1.1.1 Calculations

Front and Rear Impact– m = 250 kg; v = 45 km/h = 12.6 m/s; t = 0.3 s

$$F = m \cdot \frac{v}{t} \tag{1.2}$$
$$= 250 \cdot (42)\ N \tag{1.3}$$
$$= 10,500\ N \tag{1.4}$$

1.1.2 Front Impact Analysis-

- Load application-

$$F_x = 0N \quad F_y = 0N \quad F_z = 50.01kN$$

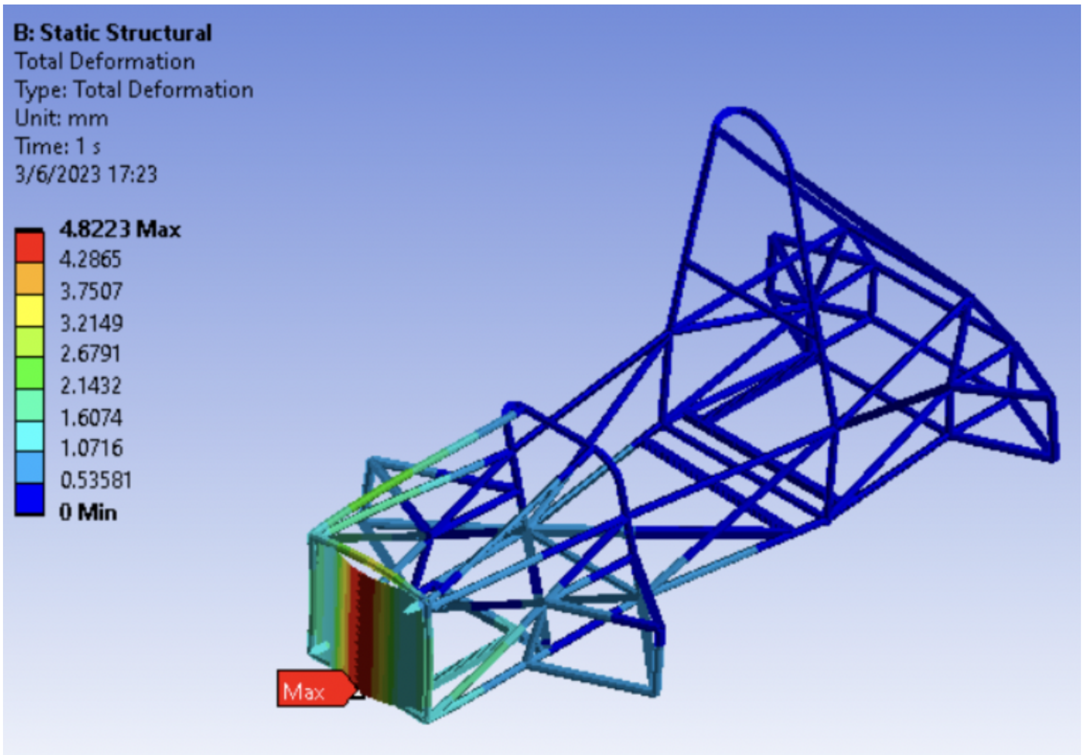


Figure 1.1: Front Impact Analysis

- Application Points - Nodes of the Front Bulkhead.
- Boundary Condition - Fixed displacement (x,y,z) but not rotation about the nodes of the Rear Bulkhead.
- Maximum Allowable deflection = 25 mm
- Maximum Deformation produced = 4.832 mm

1.1.3 Side Impact Analysis-

- Load Application -

$$F_x = 0 \quad F_y = 29.16kN \quad F_z = 0$$

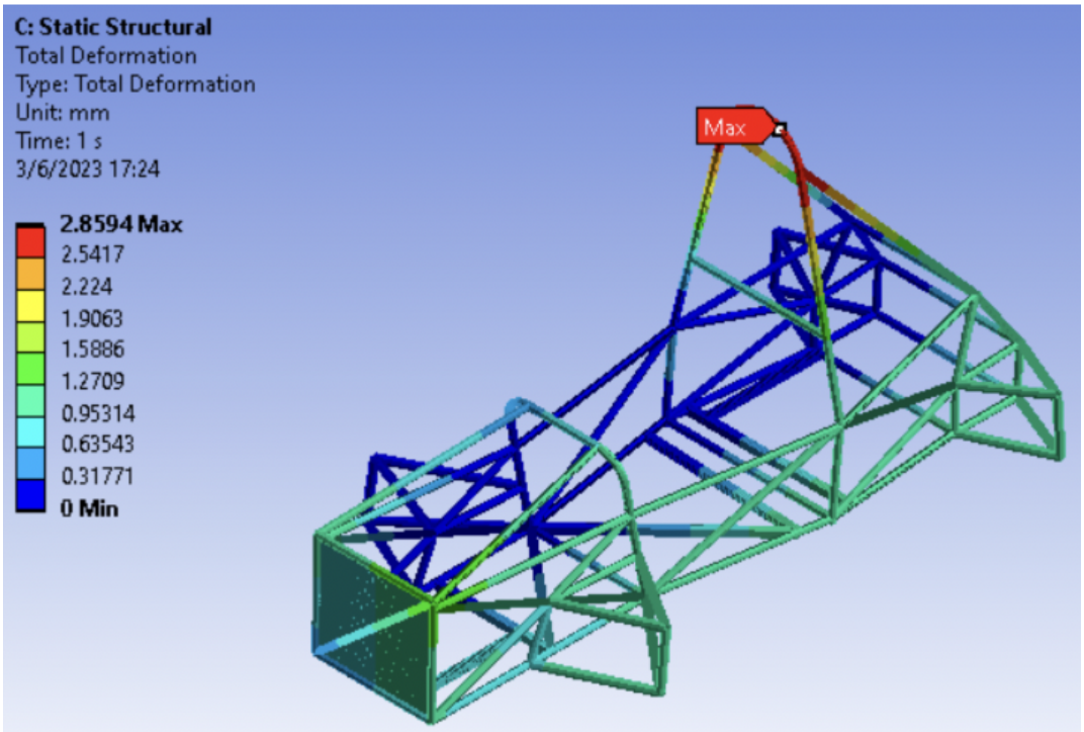


Figure 1.2: Side Impact Analysis

- Application point: Left members of FBH, front roll hoop, main roll hoop, rear bulkhead .
- Boundary Condition: Simple supports provided at suspension inboard points.
- Max Allowable Deflection: 25 mm
- Max Deformation: 2.85 mm

1.1.4 Rear Impact Analysis

- Load application-

$$F_x = 0N \quad F_y = 0N \quad F_z = 50kN$$

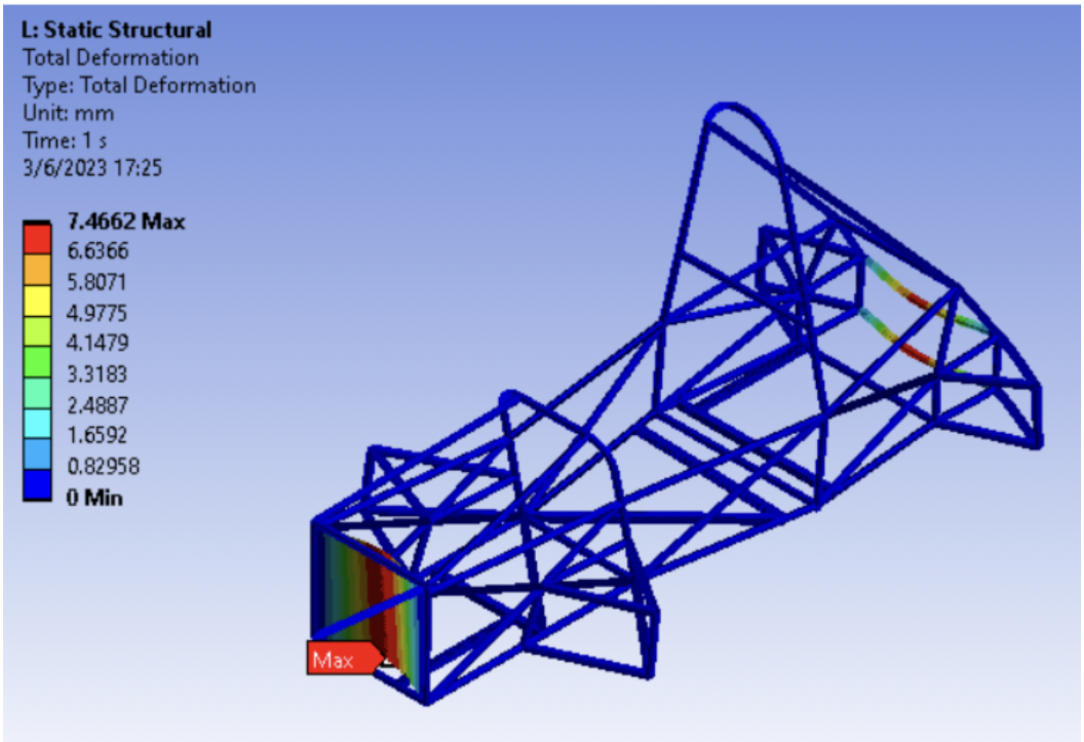


Figure 1.3: Rear Impact Analysis

- Application Points - Nodes at triangulated points of Rear Bulkhead
- Boundary Condition - Fixed displacement (x,y,z) but not rotation about the nodes of the Rear Bulkhead.
- Maximum Allowable deflection = 25 mm
- Maximum Deformation produced = 7.4662 mm

1.1.5 Main Roll Hoop Impact (Roll-Over)

- Load Application -

$$F_x = 5kN \quad F_y = -9kN \quad F_z = -6kN$$

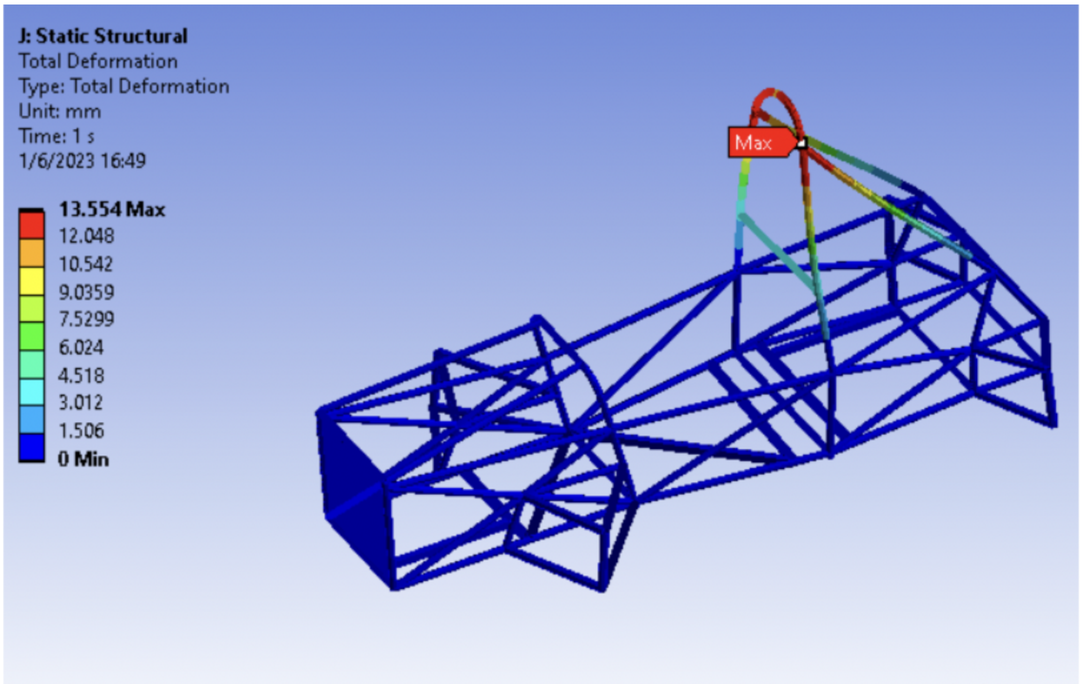


Figure 1.4: Main Roll Hoop Impact

- Application Points - Top of main Roll Hoop and Front Hoop.
- Boundary Condition - Fixed displacement (x, y, z) but not rotation of the bottom nodes of front and main roll hoop.
- Maximum Allowable deflection = 25 mm
- Maximum Deformation produced = 13.554 mm

Front Roll Hoop Impact (Roll-Over)

- Load Application

$$F_x = 5kN \quad F_y = -9kN \quad F_z = -6kN$$

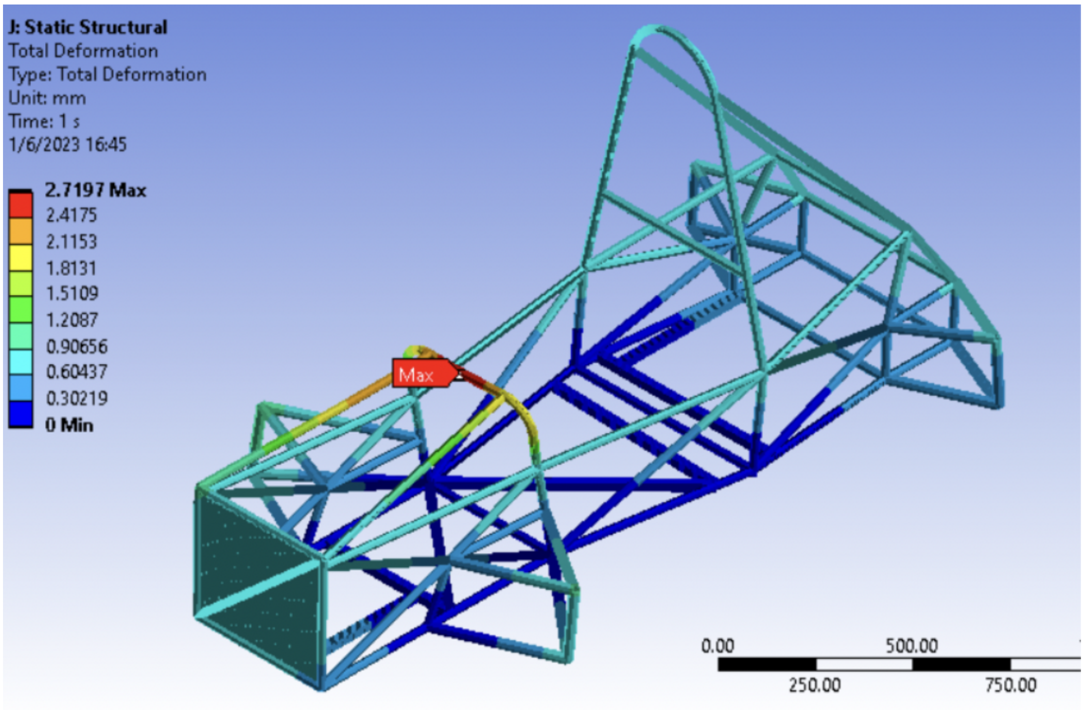


Figure 1.5: Front Roll Hoop Impact

- Application Points - Top of the Front Hoop.
- Boundary Condition -Fixed displacement (x, y, z) but not rotation of the bottom nodes of Front and main roll hoops.
- Maximum Allowable deflection = 25 mm
- Maximum Deformation produced = 2.7197 mm

Torsional Test (Front)

- Load Application :

$$1.F_x = 0N \quad F_y = 3kN \quad F_z = 0N(\text{at front left suspension})$$

$$2.F_x = 0N \quad F_y = 3kN \quad F_z = 0N(\text{at front right suspension})$$

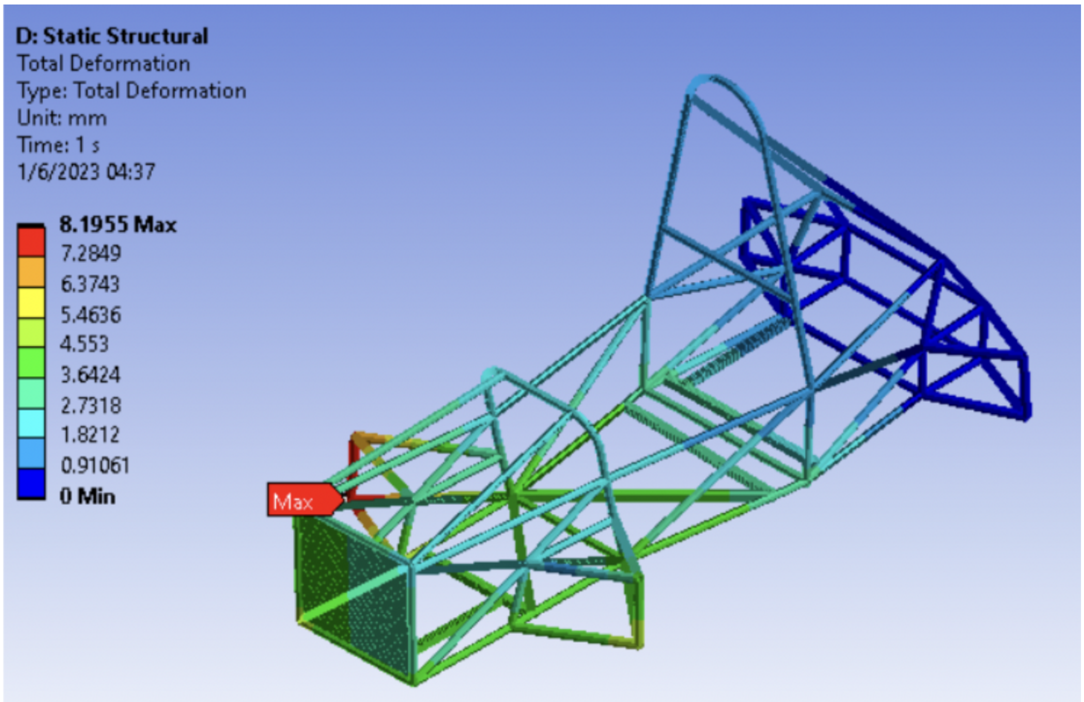


Figure 1.6: Torsional Impact

- Application Points - Nodal force at front left and right suspension points, upward and downward respectively.
- Boundary Condition - Fixed displacement (x, y, z) and rotation of the chassis members attached to rear bulkhead points.
- Torsional Constant J

$$\text{Total Weight} = 300\ Kg \tag{1.5}$$
$$\text{Force} = 10 \cdot 300 \tag{1.6}$$
$$= 3.0\ KN \tag{1.7}$$
$$\text{Torque} = F \cdot 0.5 \cdot (\text{Track Width}) \tag{1.8}$$
$$= 3000 \times 0.605 \tag{1.9}$$
$$= 1815\ N \cdot m \tag{1.10}$$
$$\theta = tan^{-1} \frac{\text{Vertical Displacement}}{0.5 \cdot \text{Track Width}} \tag{1.11}$$
$$= tan^{-1} \frac{8.2}{0.5 \cdot 1210} \tag{1.12}$$
$$= 0.78 \text{ rad} \tag{1.13}$$
$$\text{Torsional Stiffness}(k) = \frac{\text{Torque}}{\text{Angle of Deflection}} \tag{1.14}$$
$$= \frac{1815}{0.78} \tag{1.15}$$
$$= 2327\ Nm/degree \tag{1.16}$$

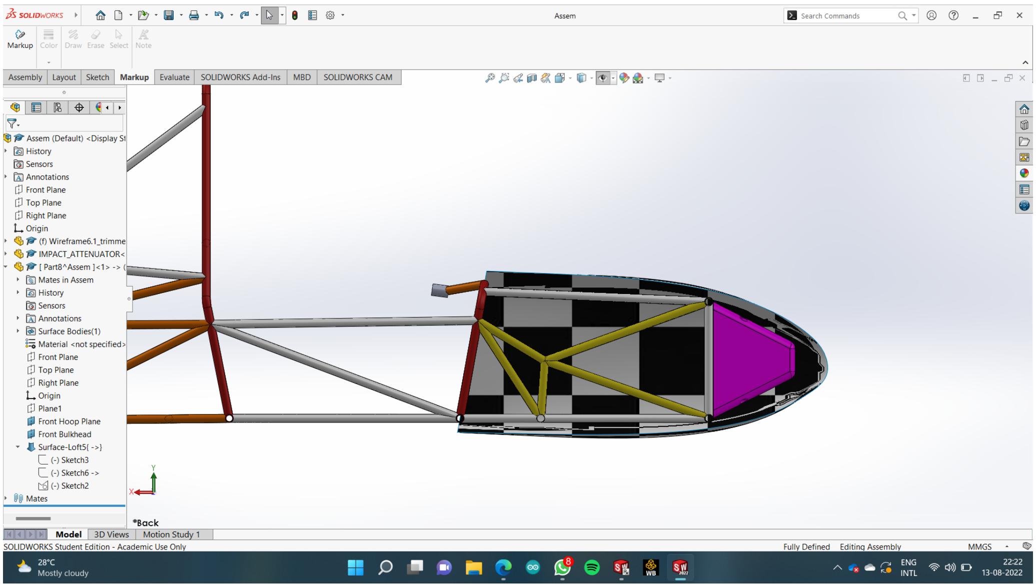
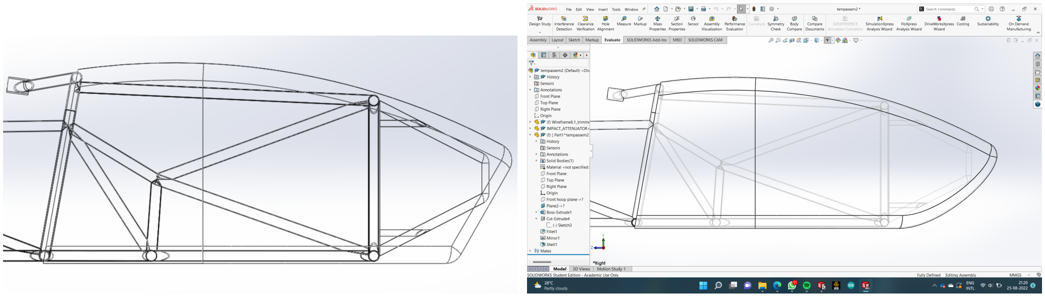


Figure 1.7: Iteration-1



(a) Iteration-2

(b) Iteration-3

1.2 Bodyworks

1.2.1 Designing the Nose Cone using Surface Modelling

Started designing the nose cone of the car using surface modelling. Initially started with a pretty basic design. Further continued and modified this design to account for the margins, aerodynamic shape, and to remove the bulkiness. All of this was done keeping manufacturing in mind.

1.2.2 Analysis on Nose Cone

After the designing of the nose cone was complete, analysis was done on Ansys(Fluent) using CFD to realize the airflow around the nose cone. Initially the analysis was done on the Ahmed Body to understand and learn how different parameters work and use it to learn the various contours that can be plotted such as Velocity Contours, streamlines, Pressure contours etc. After that the analysis was done on the iterations of the nose cone designed. The results were used to analyze the airflow, and subsequently redesign it to get better results. All the iterations and the relevant analysis can be seen in Fig 1.7 to 1.14.

Some changes were made in the final iterations keeping manufacturing in mind.

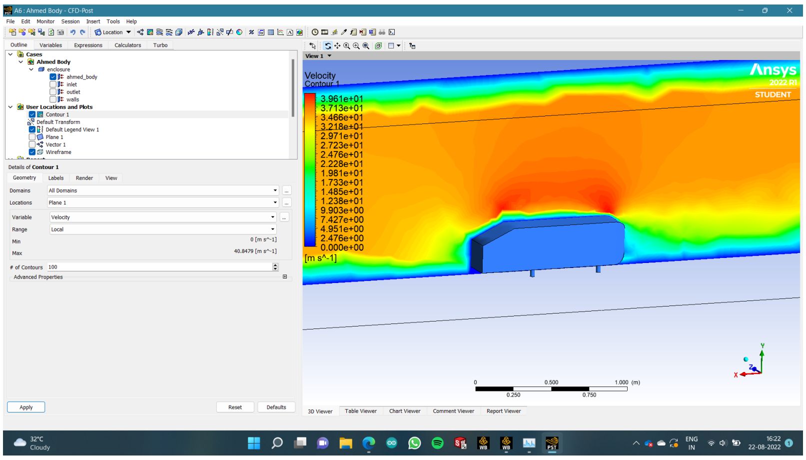


Figure 1.9: Ahmed Body

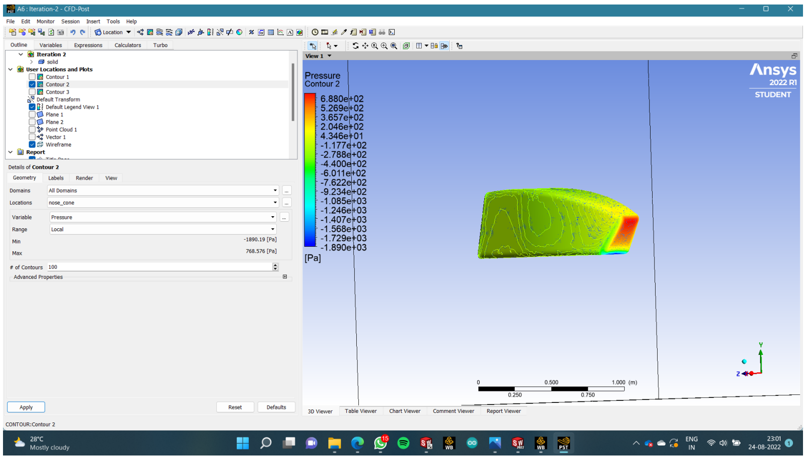


Figure 1.10: Pressure contour on Iteration-2

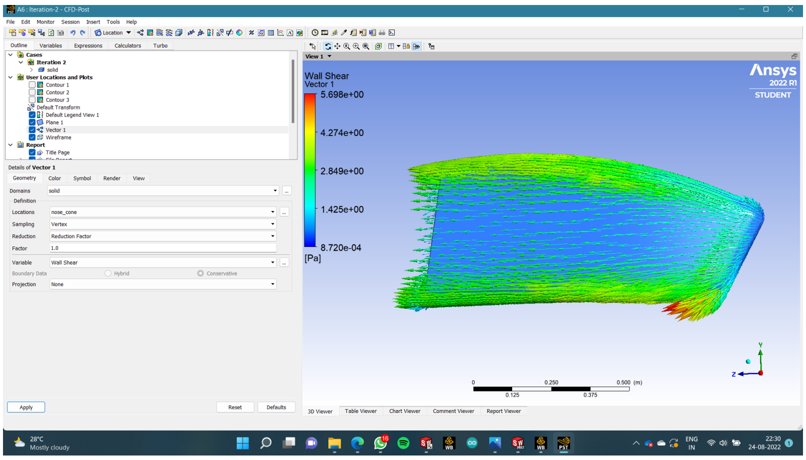


Figure 1.11: Wall Shear on Iteration-2

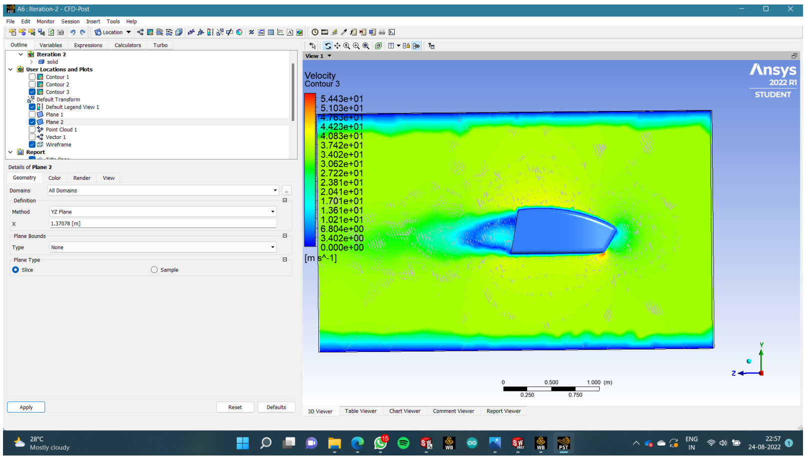


Figure 1.12: Velocity Contour of Iteration-2

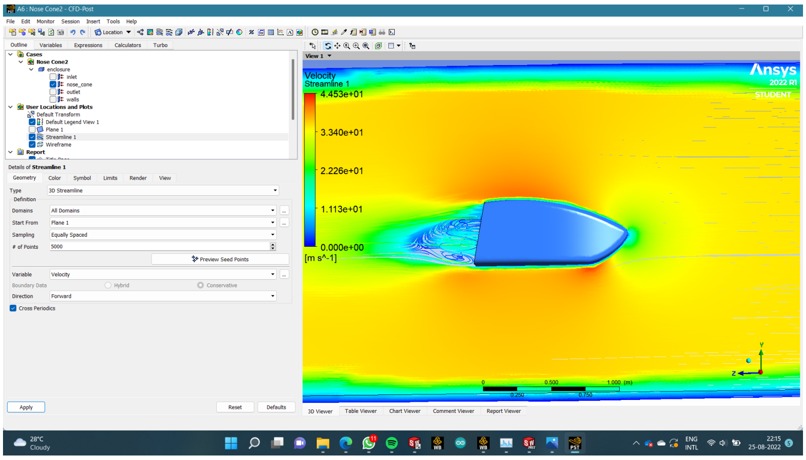


Figure 1.13: Velocity Contour Iteration-3

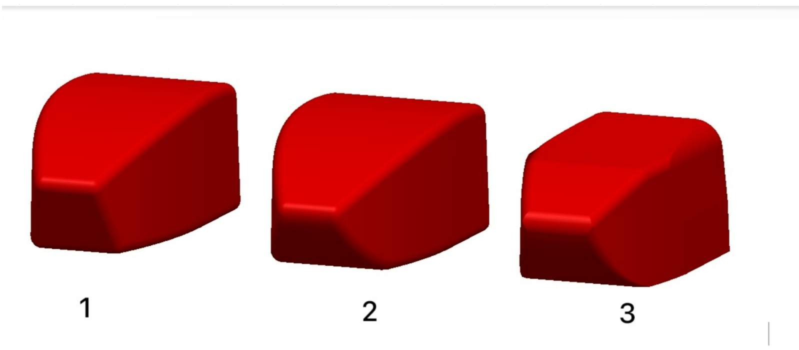


Figure 1.14: Comparison between different Iterations

1.3 Manufacturing - Spaceframe

1.3.1 Jigs

Keeping in mind the importance of precision in chassis manufacturing for other subsystems and also making manufacturing cost-effective, we designed the jigs such that they are easy to handle and assemble along with providing us one fixed geometry of the tubes to be placed into.

Previous seasons' jigs were analyzed to find out major faults and features.

Pros:

- Cost-effective
- Easy to understand and assemble
- Easy availability of material.

Cons:

- Holes were not made keeping in mind the angle of tubes with the face
- Less pickup points
- Jigs were made for whole chassis at once making them difficult to handle

For this season, we designed our jigs inheriting the features of last year and keeping the following things in mind additionally:

1. Precise jigs adjusted to both intersection on either face on tube
2. Easy to handle, assemble and manufacture
3. No part of the jig large enough to flex
4. Elimination of flexing of plywood
5. Better access to welder

Figure 1.15: Assembled Jigs

Figure 1.16: Node after Jigs assembly

The designed jigs was a sum total of three individual, independent parts which didn't require any member of the chassis to assemble. Plywood of 9 mm was used to make faces in Z-Y plane and plywood of 12 mm was used to make other faces. Slots were given in each jig, thus making the assembly as easy as sliding plywood pieces into each other.

Also pins were made on the jig to denote the direction of positive axes of the car which made the assembling even more convenient. The three different parts were welded independently and then welded to each other using a height gauge and a height adjustable trolley.

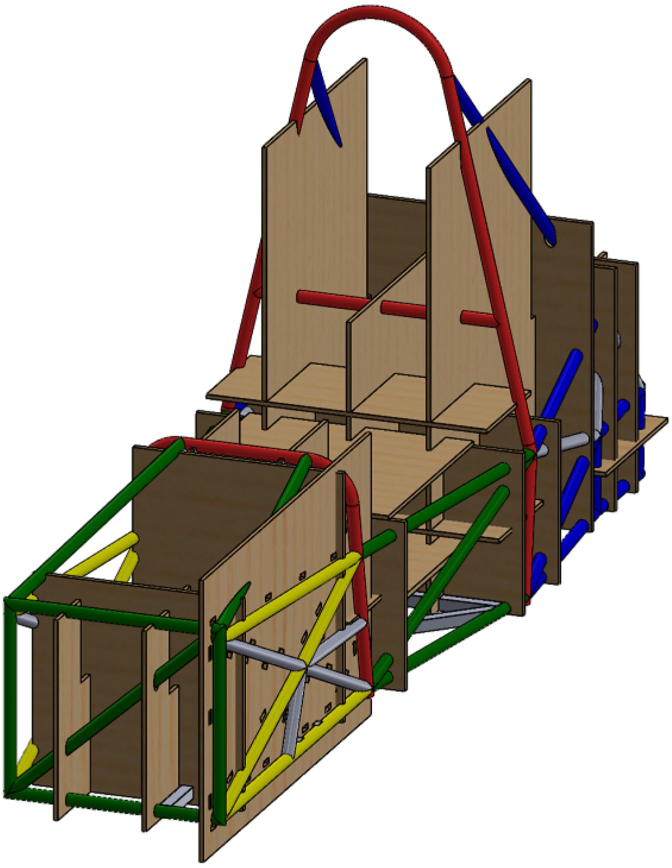


Figure 1.17: Front jigs' assembly

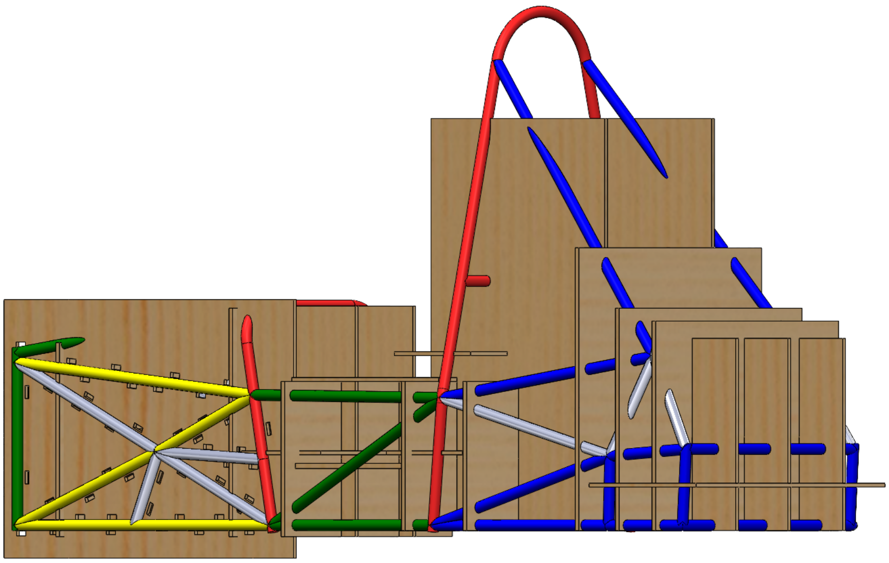


Figure 1.18: Rear jigs' assembly

1.3.2 Tube Notching

- Physical cutting

Figure 1.19: Notched tubes with grinder

Precision machining using hand grinder cutting for tube notching is a vital process in metal fabrication and construction industries, ensuring accurate and clean cuts for seamless tube connections. The method involves marking the notch location on the tube, securing it to prevent movement, and using a specialized grinder with a cutting wheel or abrasive disc. We carefully guide the grinder along marked lines, removing material with precision. This technique offers high customization and control, making it ideal for complex structures. Despite demanding labour, physical grinder cutting for tube notching remains a reliable and efficient solution for industries requiring utmost precision in tube fabrication..

- Bending

Figure 1.20: Bended Main Hoop

Bending was required for making Front Roll Hoop, Main Roll Hoop and Rear Bulkhead. Though we could have designed a planar Main Roll Hoop but the advantages of torsional stiffness provided us an incentive to go for the higher cost 3-D bending. Besides that, the 3-D main roll hoop also helps to support driver sitting angle and provides for aerodynamics of the vehicle.

- Use of square tubes

Square tubes were used as a part of the floor tubes which provides us with better accuracy in welding of taps which are of prime importance such as differential mount tabs, front damper tabs and rack mount.

Figure 1.21: Square tube

1.4 Welding Process

For welding the tubes, we had three options based on cost and general availability:

1. **SMAW (Shielded Metal Arc Welding)**
2. **GMAW (Gas Metal Arc Welding)**
3. **GTAW (Gas Tungsten Arc Welding)**

Among the above three, SMAW was eliminated because it forms a very large Heat Affected Zone, hence altering properties of the tube. Besides, it delivers a very untidy and aesthetically bad job. GTAW and GMAW welding, GTAW welding was preferred as the weld is more controllable, the apparatus is more portable, ensures a strong yet light weld, and a much cleaner job than a GMAW weld.

1.4.1 Welding Material

The **ER70 series** rods are recommended for low carbon steel such as AISI 1018 itself. Narrowing our choices based on availability, we shortlisted:

1. **ER70S-2**
2. **ER70S-6**

While ER70S-6 has more silicon which makes the weld puddle more fluid, ER70S-2 has various elements such as Zirconium, Aluminium, Titanium, etc. This property makes ER70S-2 much more desirable; pre-weld surface preparation required is less, and that makes up for the welding conditions with less flowing weld puddle allowing easy welding at difficult and unusual positions. ER80S-D2 was considered for its more rigid welds, but the 70,000 psi offered by the 70 series was considered to be more than enough. Besides, the D2 mixture makes relatively brittle joints and requires more preparation as compared to ER70S-2.

Welding was done using ER70S-2 filler rod with a DC TIG 80-100A.

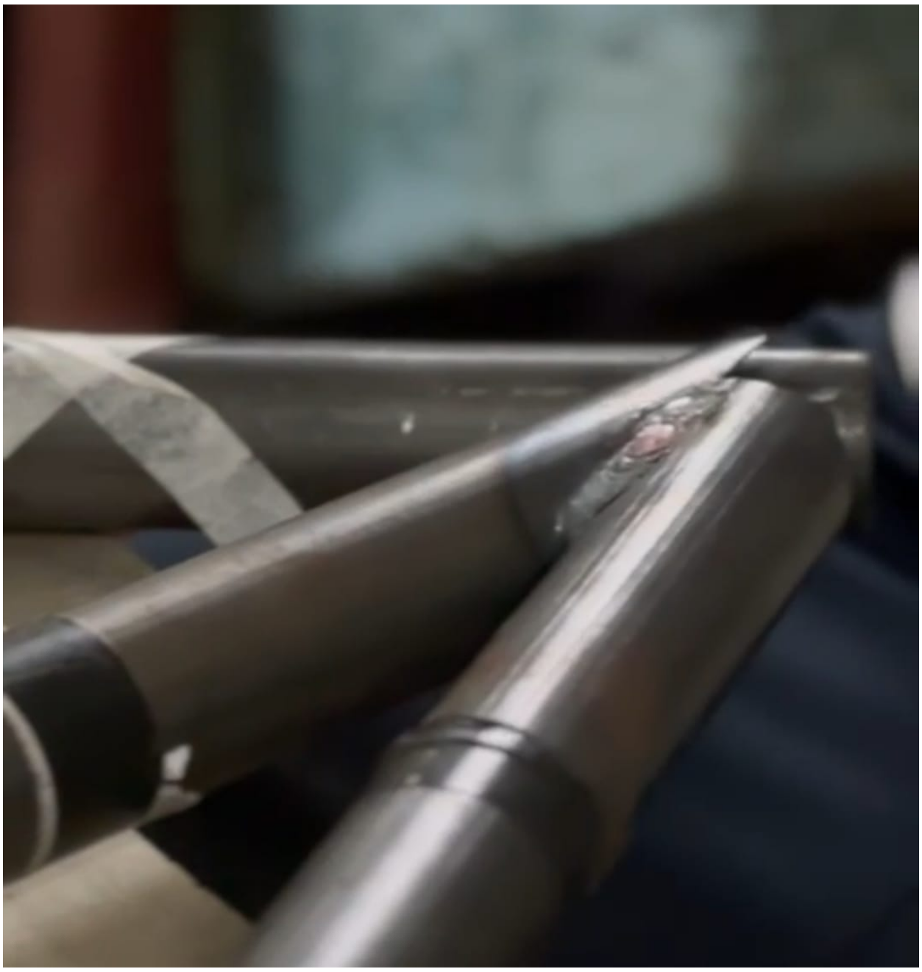


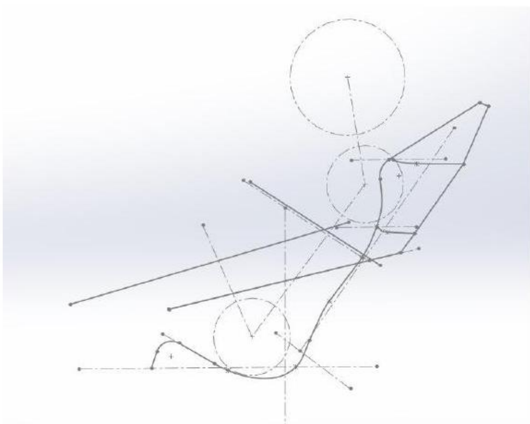
Figure 1.22: Weld at front bulkhead

Figure 1.23: Welded chassis

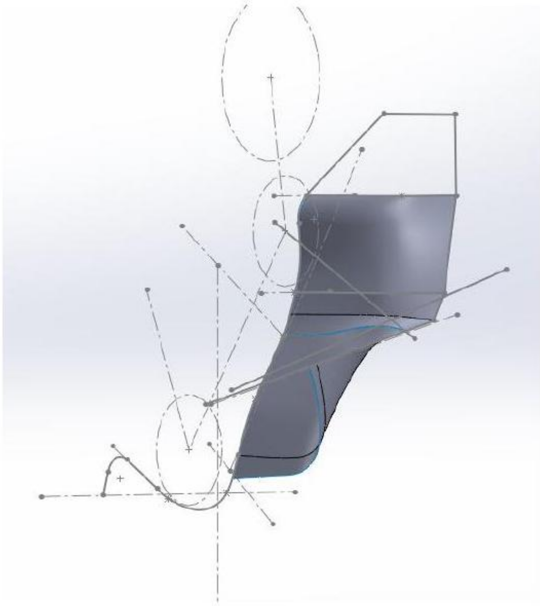
1.5 Designing of Seat

Seat design being the most important part of driver ergonomics was thought of carefully. A good seat design allows the driver to comfortably drive for longer period of time, supports all the body parts in contact especially the driver's back, butt and shoulders, provides a sturdy support during acceleration, doesn't restrict driver's motion in the cockpit and provides for proper securing of driver using driver harness. Following were the steps followed in designing of seat:

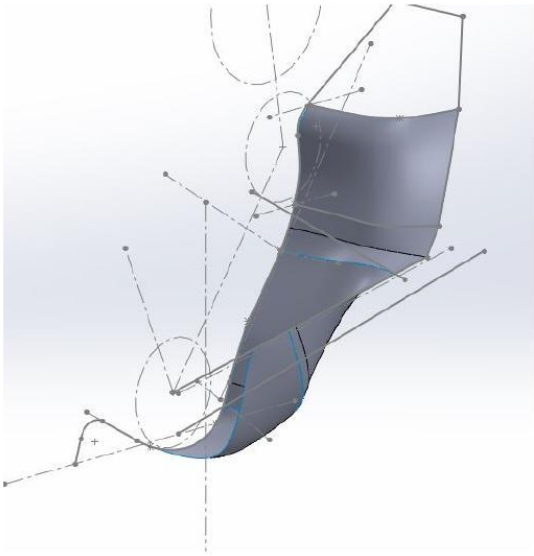
- First, a sketch of driver model with required parameters along with required members of the chassis was made.



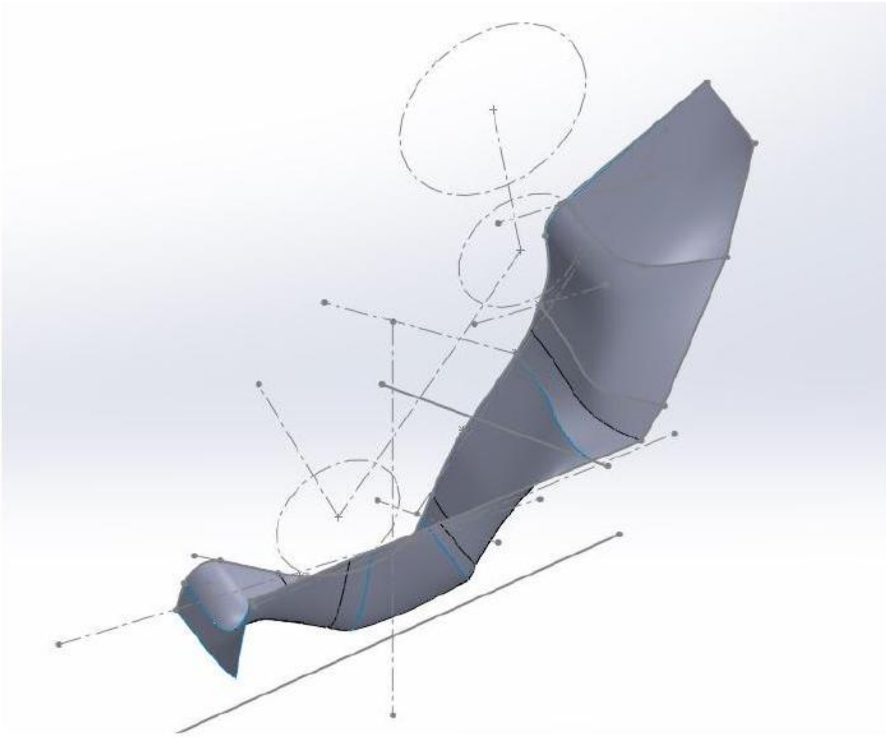
- Then the back support was made according to the driver sitting angle and the shoulder support was made according to the shape and size of the shoulder circle.



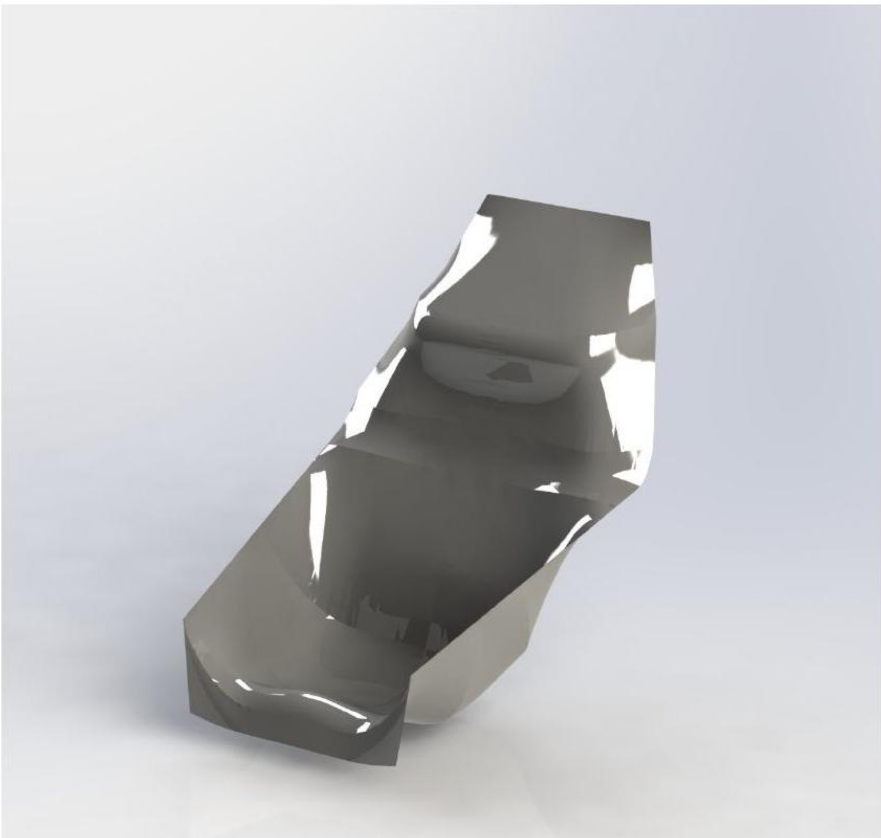
- Then the butt support was added. The curve coming from the back support was smoothly curved around the butt representative circle.



- Thereafter, the top part was added to meet the shoulder harness mounting bar to add aesthetics to the seat design apart from protecting driver's neck from direct contact with the firewall. The bottom part was completed as well continuing the lower curve.



- After mirroring, and final touch ups, the following design was obtained.



Chapter 2

Powertrain

2.1 Motor Selection

2.1.1 Power Calculation

Our two main assumptions for the power calculations of the motor are:-

$V_{\text{max}} = 120\text{Kmph}$.
We took V_{max} as 120Kmph is the technical achievable speed of our car.
$M = 300\text{Kg}$

There are two types of opposing force.

1. Rolling friction $= \mu \cdot M \cdot g$
   here,
   - μ = Coefficient of rolling friction
   - M = Mass of vehicle
   - g = acceleration due to gravity
2. Air resistance $= 0.5 \cdot \rho \cdot A \cdot C_d \cdot V_{\text{max}}^2$
   - ρ = density of air
   - A = area of front of the car
   - C_d= Drag coefficient
   - V_{max} = Maximum velocity of car

so $F_{\text{total}} = \mu \cdot M \cdot g + 0.5 \cdot \rho \cdot A \cdot C_d \cdot V_{\text{max}}^2$
power $= F \cdot V_{\text{max}}$

$$P = \frac{(\mu \cdot M \cdot g \cdot V_{\text{max}} + 0.5 \cdot \rho \cdot A \cdot C_d \cdot V_{\text{max}}^3)}{(\eta \cdot K_p)}$$

η = Efficiency of motor
k_p = Compensation coefficient of capacity

$\mu = 0.025$
c_d =0.69
ρ =1.2 kgm^{-3}
g =9.8 ms^{-2}
η =0.9
K_p =0.95
A = 1.56 m^2

$P_{\text{peak}} = 30$ kW
Also, Power = $torque \cdot \omega$
$\omega = \frac{v}{r}$
r = Radius of tyre
r = 25 cm
ω = 133.33 rad/s
Torque = 144.67 Nm

2.1.2 Motor Specifications

After doing all the calculations we searched in the market for the motors to meet all our requirements and we found that **TSUYO THWM-P-J-96** is meeting all requirements.

Constant power/peak power	15/30 (kW)
Rated voltage	96 (VDC)
Rated current	135 (A)
Rated torque/peak torque	35/140 (N.m)
Rated speed/peak speed	4093/8500 (rpm)
Resolver pole pair number	4 Poles
weight	43Kg
Operating temperature	-40℃ to +60℃

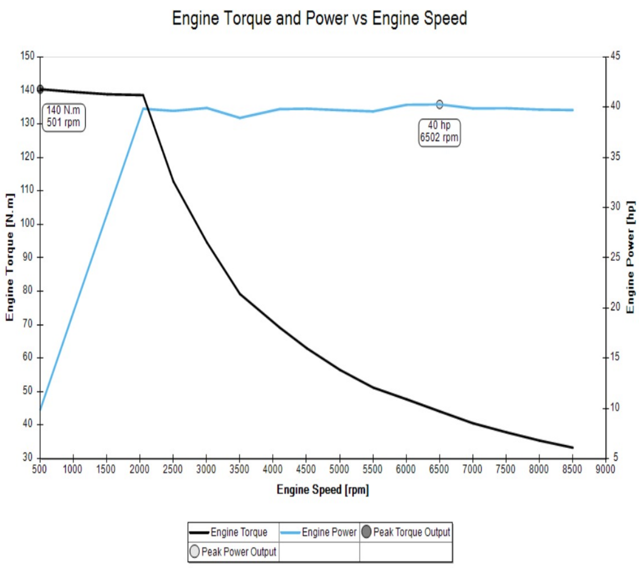


Figure 2.1: Peak characteristics curve

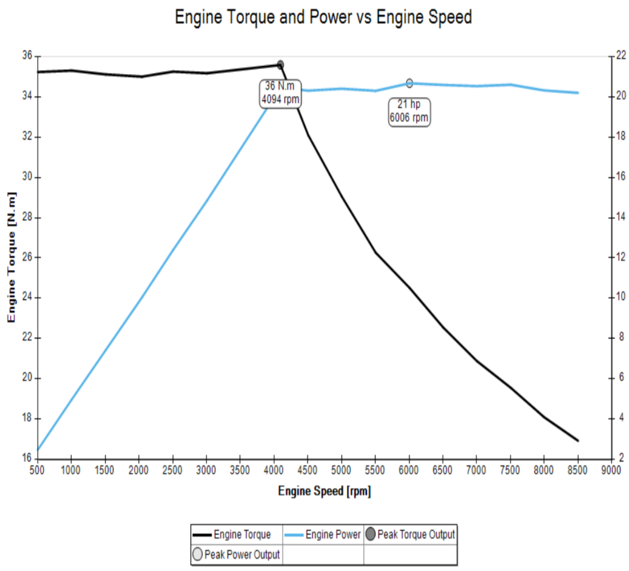


Figure 2.2: Rated characteristics curve

2.2 Reduction ratio optimisation

2.2.1 Lap-time Simulation

The simulations were done on **OptimumLap software** for the **Kari Motor speedway track**. Data assumed by us are as follows:

- Mass of vehicle: 300 kg
- Drag Coefficient: -0.69
- Downforce Coefficient: - 0.29
- Frontal Area: 1.8 m^2
- Lateral friction: 1.40
- Longitudinal friction: 1.40
- Rolling resistance: 0.03
- Air density: 1.2 kg/m^3
  The air density of Coimbatore is considered as 1.2 kg/m^3 according to the weather conditions prevailing there.
- Tire Radius: 0.25 m
- Engine thermal efficiency: 90%
- Drivetrain efficiency: 80%

Data at different reduction ratios was compiled for collective analysis. Accordingly the reduction ratio of 4 was found to be suitable as it aligned with all the design goals.
The link to the simulation results is given below
Link

2.3 Chain and Sprocket

2.3.1 Chain-sprocket VS Single Gear VS Multi Gear VS Belt Drive VS CVT

	Chain sprocket	**Single Gear**	**Multi Gear**	**Belt Drive**	**CVT**
Design	Chain drive system consists of chain wrapped along one sprocket on motor shaft and another one on the wheel axle	Single gear system consists of a single gear and connects the electric motor directly to the wheels without any gear changing mechanism	Multi gear system consists of multiple gears and can vary the gear ratio between the electric motors and the wheel	Belt drive consists of belt, shaft and pulleys	CVT consists of a v shaped belt and two cone shaped halves
Efficiency	High	Moderately-high	Very high	High	High
Performance	good performance but with a single reduction ratio	good performance but have limited top speed and top acceleration	Offers high top speed and top acceleration due to availability of more gear ratios	Generally quieter and produces less vibration than chain drives and suitable for medium torque requirement	Offers infinite gear ratio and gear transition is so smooth
Simplicity	Simple and very lightweight	Simple	Complex and heavy	Simple and few components	complex and lightweight
Maintenance	Requires periodic lubrication and maintenance	Requires more maintenance than chain and sprocket	Requires high maintenance and frequent lubrication.	Does not require heavy maintenance	Repair is more expensive and requires regular service
Life expectancy	Moderate	High	High	Low-moderate	High

From all these power transmitting systems we have chosen the **chain and sprocket system**. This is mainly because of the simplicity and high efficiency of the mechanism.

2.3.2 Length of Chain and Centre Distance

Let

T_1 = Number of teeth on the smaller sprocket,

T_2 = Number of teeth on the larger sprocket,

p = Pitch of the chain,

x = Centre distance

The length of the chain (L) must be equal to the product of number of chain links (K) and the pitch of the chain (p). Mathematically,

$$L = K.p$$

The number of chain links may be obtained from the following expressions, i.e.

$$K = \frac{T_1 + T_2}{2} + \frac{2x}{p} + \left[\frac{T_2 - T_1}{2\pi}\right]^2 \frac{p}{x}$$

The value of K as obtained from the above expression must be approximated to the nearest even number.

The centre distance is given by

$$x = \frac{p}{4}\left[K - \frac{T_1 - T_2}{2} + \sqrt{\left(K - \frac{T_1 + T_2}{2}\right)^2 - 8\left(\frac{T_2 - T_1}{2\pi}\right)^2}\right]$$

In order to accommodate initial slack in the chain, the value of the centre distance obtained from the above equation should be decreased by 2 to 5 mm.

2.3.3 Pitch and Number of Teeth

By using Optimum Lap, it was found that a gear ratio of 4 is required. A gear ratio of 2 is provided by the reducer. So a gear ratio of 2 is required for the chain and sprocket system. So a table was constructed for different number of teeth on smaller sprocket with a gear ratio of 2.

1. The chassis subsystem will give the approx distance between the engine output shaft and the wheels (x).
2. Then set the no.of teeth in the smaller sprocket (T1) and multiply the gear ratio to T1 to get T2 (no. of teeth in the larger sprocket).
3. Decide a pitch of available chains (p).
4. Calculate the No. of links (K) from (1) formula in the above figure.

5. Now, the K will be some rational number convert it to an integer (Read the odd even section below to how to choose the integer)
6. Then put the integer K in the 2 formula in the above figure to get the exact (x). And give that information to the chassis subsystem.

The centre distance as provided by Chassis subsystem is 230 mm. Based on this data and considering sufficient factor of safety, the final pitch is taken as 15.875 mm (5/8 in).

Pitch (mm)	**Gear Ratio**	**Centre Distance (mm)**	T_1	T_2	**Number of Links**
15.875	2	230	15	30	50.6278
15.875	2	230	16	32	52.1845
15.875	2	230	17	34	53.7448
15.875	2	230	18	36	55.3087
15.875	2	230	19	38	56.8764
15.875	2	230	20	40	58.4477
15.875	2	230	21	42	60.0226

Final number of teeth on smaller sprocket is taken as 21 and that on the larger sprocket is 42. This gives number of links to be 61.24. Converting this to integer, number of links are 62.

After incorporating the required slack we get the centre distance 233.8 mm.

2.3.4 Forces

The equation for force on each teeth of a sprocket is given by the equation-

$$T_i = T_0 \left[\frac{sin(\phi)}{sin(\phi + 2\beta)} \right]^{(i-1)}$$

T_1 = Tension on i^{th} teeth

T_0 = Tension on the first teeth

$2\beta = \frac{360}{N}$ in Degrees [N=number of teeth on the sprocket]

$\phi = 17 - \frac{64}{N}$ in Degrees

Forces we calculated for the 21 teeth sprocket

N	**Pitch (mm)**	T_0 **(N)**	T_1 **(N)**	T_2 **(N)**	T_3 **(N)**	T_4 **(N)**	T_5 **(N)**	T_6 **(N)**	T_7 **(N)**
21	15.875	3667	1350.97	497.69	183.35	67.54	24.88	9.16	3.37

Forces we calculated for the 42 teeth sprocket

N	Pitch (mm)	T_0 (N)	T_1 (N)	T_2 (N)	T_3 (N)	T_4 (N)	T_5 (N)	T_6 (N)	T_7 (N)
42	15.875	3667	2598.52	1841.31	1304.75	924.55	655.13	464.22	328.95

N	Pitch (mm)	T_8 (N)	T_9 (N)	T_{10} (N)	T_{11} (N)	T_{12} (N)	T_{13} (N)	T_{14} (N)	T_{15} (N)	T_{16} (N)
42	15.875	233.09	165.17	117.04	82.93	58.76	41.64	29.50	20.90	14.81

2.3.5 Analysis on Ansys

Mild Carbon Steel

- Density - 7850 kg m^3
- Coefficient of thermal expansion- $1.1 * 10^{-5} C^{-1}$
- Young's Modulus - 200 GPa
- Posisson's Ratio - 0.28
- Tensile Yield Strength - 250 MPa
- Tensile Ultimate Strength - 440 MPa
- Compressive Ultimate Strength - 350 MPa

Analysis on Ansys

	Minimum	**Maximum**
Stress (MPa)	$2.9971 * 10^{-3}$	190.24
Deformation (mm)	0	$2.2435 * 10^{-2}$
Safety factor	1.2494	15

AISI 4140

- Density - $7.850 gcm^{-}3$
- Coefficient of thermal expansion- $1.2 * 10^{-5} C^{-}1$
- Bulk Modulus - 140 GPA
- Shear Modulus - 80 GPa
- Tensile Yield Strength - 655 MPa
- Tensile Ultimate Strength - 1020 MPa
- Compressive Ultimate Strength - 1580 MPa

	Minimum	**Maximum**
Stress (MPa)	$3.8953 * 10^{-4}$	237.22
Deformation (mm)	0	$2.3688 * 10^{-2}$
Safety factor	2.7612	15

AISI 4340

- Density - $7.850gcm^{-3}$
- Coefficient of thermal expansion- $1.2 * 10^{-5}C^{-1}$
- Bulk Modulus - 140 GPA
- Shear Modulus - 80 GPa
- Tensile Yield Strength - 862 MPa
- Tensile Ultimate Strength - 1282 MPa
- Compressive Ultimate Strength - 1700 MPa

2.3.6 Miscellaneous

Slack in Chain

In the chain and sprocket system, the chain is usually slacked, making the chain slighly longer than that required. The reasons for this slack are as followed-

- **Compensation for Misalignment** Misalignment between driving and driven sprockets can occur due to manufacturing tolerances, installation errors, or operational conditions. The slack in the chain accommodates this misalignment and prevents excessive stress on the chain, sprockets, and other components.
- **Tension Variation** Chain tension can vary during operation due to load, speed, or environmental conditions fluctuations. By providing slack, the chain can absorb these tension variations without becoming overly tight or loose. It helps maintain a more consistent, optimal tension level for efficient power transmission.
- **Heat Expansion** During operation, chains can experience heat buildup due to friction. This heat can cause the chain components to expand. By allowing slack, the chain has room to expand without becoming excessively tight, which could lead to increased wear, binding, or even chain failure.
- **Shock and Vibration Damping** Slack in the chain acts as a buffer to absorb shock and vibration during operation. It helps reduce the transmission of these forces to the chain, sprockets, and other connected machinery. This damping effect improves the overall smoothness of the system, reduces noise, and helps prolong the chain's life and related components' life.

It is worth noting that while some slack is necessary, excessive slack can also be detrimental. Too much slack can lead to chain jumping off the sprockets, reduced power transmission efficiency, and increased wear. Therefore, it is essential to strike a balance and ensure that the chain has an appropriate amount of slack based on the specific application and manufacturer's recommendations.

Odd-even numbers of teeth

Research has revealed that achieving enhanced durability in a sprocket-chain system can be accomplished by adjusting the number of chain links and the number of teeth on both sprockets. Specifically, it has been observed that when the number of chain links is even, it is beneficial to have an odd number of teeth on each gear, and conversely, when the number of chain links is odd, an even number of teeth on each gear is preferred. This arrangement significantly improves durability by ensuring that the same tooth does not engage with the same link during every cycle, thereby minimizing the occurrence of vibration and wear within that specific gear-chain pairing.

Polygon Effect

Polygon effect is a phenomenon which is prominent in chain drives with small links or when operating at high speeds. When a chain drive is engaged, power is transmitted from the driving sprocket to the driven sprocket via the chain. The chain consists of a series of links, typically in the form of rollers connected by pins. As the chain engages with the sprocket teeth, it experiences a periodic acceleration and deceleration due to the discrete nature of the chain links. This acceleration and deceleration result in speed fluctuations during each engagement between the chain and sprocket. These variations in speed can cause fluctuations in the torque transmitted from the driving sprocket to the driven sprocket. The torque fluctuations can lead to vibration, noise and increased wear on the chain and sprockets.

2.3.7 Failure modes of chain drive

- **Wear and tear of chain** Wear and tear of chain Due to friction and vibration chain wears and elongates over time. As a result it becomes loose and may skip one or more teeth of sprockets
- **Overload** Increasing the load beyond the design limit of the chain may result in exceeding stress on chain or even breakdown
- **Corrosion** With time chain starts rusting due to reaction with chemicals and moisture, also due to insufficient lubrication. This weakens the chain and reduces its ultimate strength
- **Misalignment** Incorrect alignment between chain and sprocket can lead to more wear, increased friction and slipping of chain.

2.4 GEARBOX

Torque : 140 N-m

Reduction Ratio aimed by the gear: 2

- the smallest number of teeth on a spur pinion and gear which can exist without interference

$$N_p = \frac{2k}{(1+2m_G)(\sin\phi)^2}\{m_G + \sqrt{{m_G}^2 + (1+2m_G)\sin\phi^2}\}; m_G = \frac{N_G}{N_P}$$

where k is 1 for full depth and 0.8 for stub depth
N_p = Number of Teeth on pinion

N_G = Number of Teeth on gear
ϕ = Pressure Angle

- the largest gear for a specified pinion to avoid interference

$$N_G = \frac{N_p^2 \sin\phi^2 - 4k^2}{4k - 2N_p \sin\phi^2}$$

where k is 1 for full depth and 0.8 for stub depth
N_p = Number of Teeth on pinion
N_G = Number of Teeth on gear
ϕ = Pressure Angle

- Contact Stress

$$\sigma_c = Z_E \sqrt{W_t K_O K_V K_S \frac{K_H}{d_P F} \frac{Z_R}{Z_I}}$$

F is Face Width, d_p is PCD of pinion
Z_I is Geometry Factor for Pitting Resistance
Z_R is Surface Condition Factor
$Z_E = \sqrt{\frac{1}{\pi[\frac{1-v_P^2}{E_P} + \frac{1-v_G^2}{E_G}]}}$ is Elastic Coefficient
W_t is Tangential Force, K_O is Overload Factor
K_V is Dynamic Factor, K_S is Size Factor

- Factor of Safety

$$S_H = \frac{S_c}{\sigma_c} \frac{Z_N Z_W}{Y_\theta Y_Z}$$

S_c is Contact Strength(N/mm^2),
Z_N is Stress Cycle Factor for Contact Strength
Z_W is Hardness Ratio, σ_c is Contact Stress
Y_θ is Temperature Factor, Y_Z is Reliability Factor

Matlab Code For reference:-

```
%Pinion
torque=140;
table=[0,0,0,0,0,0];
ratio=2;
k=1;
s=sin(deg2rad(20))*sin(deg2rad(20));
Nmin=2*k*(ratio+(ratio^2 + (1+ 2*ratio)*s)^0.5)/((1+ 2*ratio)*s);
N=14;
Nmax= ((N*N*s)-4*k*k)/(4*k-2*N*s);
Ng=85*1.5;
T= [18 0.299 0.306 0.312 0.317 0.324;
19 0.306 0.314 0.320 0.324 0.330;
20 0.312 0.320 0.325 0.331 0.338;
21 0.318 0.325 0.331 0.337 0.344;
22 0.324 0.330 0.337 0.343 0.350;
23 0.330 0.339 0.344 0.349 0.356;
24 0.333 0.343 0.351 0.355 0.363;
25 0.337 0.344 0.353 0.360 0.367;
26 0.343 0.350 0.357 0.365 0.372;
```

```
27 0.349 0.353 0.363 0.369 0.378;
28 0.350 0.357 0.365 0.374 0.381;
29 0.353 0.361 0.371 0.377 0.386;
30 0.357 0.367 0.375 0.383 0.389;
35 0.371 0.379 0.389 0.398 0.405;
40 0.381 0.389 0.400 0.411 0.418;
45 0.387 0.399 0.409 0.419 0.426;
50 0.393 0.405 0.416 0.425 0.434;
60 0.406 0.417 0.429 0.441 0.450;
80 0.419 0.429 0.442 0.454 0.466;
125 0.431 0.445 0.456 0.472 0.485;
275 0.447 0.461 0.473 0.487 0.500;];

N2=[17    25    35    50    85]';
N1=T(:,1);
Yj_T= T(:,2:6);
Yj1=interp1(N1,Yj_T,(1:275)','pchip');
Yj2=interp1(N2,Yj1',(1:275)','pchip');
Yj=Yj2';

for module = 1:0.5:4
for Np = N:1:fix(Ng/ratio)
for times = 3:0.1:5
%Sigma(Bending) Allowed
St=((0.749*363)+110)*(10^6);
Ythe=1;
Yz=1.25;
cycle=10^7;
Yn=1.6831*(cycle^(-0.0323));

Sall= St*Yn/Ythe/Yz;

%Sigma(Bending)
x=Yj(N,85);


Rp=Np*module/2;
Wt=(torque/Rp)*1000;
Kb=1;
Ks=1;
Ko=2.25;
Kh=1.3;
V=(Rp*68)/torque; %68kWh is Power of motor
Qv=7;
B=0.25*((12-Qv)^(2/3));
A=50+56*(1-B);
Kv=(1+((200*V)^(1/2))/A)^B;
F=times*pi*module;

S=Wt*Ko*Kv*Ks/F/module*Kh*Kb/x*1000*1000;      %1000 for module and
    1000 for F

SF=Sall/S;
if(SF>1.5)
table=[table;Np,module,times,SF,module*Np,F];
end
end
end
```

```
end
```

The table listed no values for safety factor greater than 1.5. We took this value because we wanted to account both translational and vibrational effects for which FOS greater than 1 or precisely 1.5 is expected. Hence, we concluded that single/multi sage gears cannot be used for such torque.
Planetary gear system analysis:

- Highly complicated and required very high precision, thus can't be easily manufactured.
- In case of planetary gears, the driving and driven members must be concentric,which was limiting our designs as it did not allow us to incorporate the differential.

Since gears were incompatible, the pros and cons of belt drive system and chain drive system were compared so that the most efficient and suitable transmission drive can be selected.

2.5 Driveshaft

Driveshaft calculations were done keeping in mind the following points:

- Minimise rotational inertia so that angular acceleration of the car is not affected much by the rotational inertia of the driveshaft itself.
- Minimise mass of the shaft as this was one of the primary design goals of the car.

The equation relating the inner and outer diameters of the driveshaft to the acceleration of the car:

$$\dot{u} = \frac{16\cdot r\cdot T_{Wheel}}{(\rho\cdot L\cdot\pi\cdot[OD^4-ID^4+8\cdot r^2\cdot(OD^2-ID^2)]+16\cdot r^2\cdot m_{\text{Shaftless}}}$$

To ensure that the shaft does not fail due to fatigue under the applied loads, the ASME Elliptic failure criterion was used. The equation relating the inner and outer diameter of the shaft:

$$\text{ID} = (OD^4 - \frac{16\sqrt{3}}{\pi}\frac{\sqrt{OD^2{n_f}^2{S_E}^2{S_Y}^2({S_Y}^2{T_{\text{Amp}}}^2+S_2{T_{\text{Mean}}}^2)(K_{\text{FS}})^2{}_{\text{Torsion}}}}{S_E^2S_Y^2})^{\frac{1}{4}}$$

This relation was substituted in the earlier equation to get acceleration of the car in terms of the outer diameter. The shaft material was chosen as AISI 4340 steel as it is the most common material used for shafts by other formula student teams and our team has used it in the past as well.
The maxima of the acceleration function came out to be at a surprisingly unrealistic value of outer diameter = 500 mm. The following observations were made:

- Minima for mass does not exist. Mass of the shaft decreases as outer diameter increases, with the constraint that the inner and outer diameter values satisfy the ASME Elliptic failure criterion equation. Also, there is not much variation in mass with outer diameter.

- The acceleration function does not vary much with outer diameter. Hence, it is not necessary to choose outer diameter for which acceleration function is maximum.

As it is difficult from the manufacturing side to drill a hole in a 50 cm long shaft so we took inner diameter = 0 and value of outer diameter according to it is 21mm
Matlab code for calculations:

```
nf = 1.5;                              % FOS against yield
Kfs = 1;                               % fatigue SCF
Se = 333*(10.^6);                      % endurance limit
Sy = 710*(10.^6);                      % yield strength
r = 0.25;                              % wheel radius
T_max = 336;                           % max torque
T_brake = -700;                        % braking torque
l = 0.6;                               % length
T_amp = (T_max - T_brake)/2;
T_mean = (T_max + T_brake)/2;
ef = 16*sqrt(3)*nf*Kfs*sqrt((Sy.^2)*(T_amp.^2)+(Se.^2)*(T_mean.^2))/(pi
    *Se*Sy);
c = 8*r*r;
od_min = ef.^(1/3)
od = 0.03
id = ((od.^4)-ef*od).^0.25
```

Listing 2.1: Matlab code for calculations:

Splines:
We decided to use involute spline profile as it has a better torque-bearing capacity. Calculations were done for the factor of safety against various stresses. Module and number of teeth were fixed such that a healthy factor of safety was maintained.
Matlab code for calculation of factor of safety:

```
gr = 3.5;
T = 140000*gr*2.3/3.3;
rpm =6000/gr;
m = 1.75;                                       % module
N = 20;                                      % no. of teeth
phi = 30*pi/180;                                % Pressure Angle
D = N*m;                               % Pitch Diameter, in mm
Dre = (N-1.35)*m;                  % External Minor Diameter, in mm
Do = (N+1)*m;                      % External Major Diameter, in mm
Dri = (N+1.35)*m;                   % Internal Major Diameter, in mm
Di = (N-1)*m;                      % Internal Minor Diameter, in mm
Le = 0.741*D;                          % Effective Length of Spline
Dsi = 20.7;                                     % Shaft ID
Ka = 1.5;    % Spline ApplicationFactor, Uniform Power-Inttermittent
    shock
                        load
Km = 1;          % Load DistributionFactor for Misalignment of Splines,
Km=1 for fixed spline
Kf = 1;                      % Fatigue-Life Factor for 10000 torque
    cycles
Kw = 1.4;                 % Wear Life Factor for 1E7 revolutions of
    spline
```

Listing 2.2: Matlab code for calculation of factor of safety:

Shear Stress Under Roots of External Teeth

```
Ssr = 16*T*Dre*Ka/(pi*(Dre^4-Dsi^4)*Kf);            % MPa
```

Shear Stress at Pitch Diameter of Teeth

```
t = m*pi/2;                                  % Tooth thickness, in mm
Ssp = 4*T*Ka*Km/(D*N*Le*t*Kf);                   % MPa
```

Compressive Stress on Side of Spline Teeth

```
h = 0.9*m;                                   % depth of engagement, in mm
Sc = 2*T*Km*Ka/(9*D*N*Le*h*Kf);                      % MPa
```

Burst Stresses on Splines

```
tw = 5;      % wall thickness = outer diameter - internal spline major
    diameter
S1 = T*tan(phi)/(D*tw*Le);        % Radial Load Tensile Stress, MPa
S2 = 17.697*rpm^2*(Di^2+0.212*Dri^2)/1e12;   % Centrifugal Tensile
    Stress, MPa
Y = 1.5;             % Lewis Form Factor, Y=1.5 for 30 deg spline
S3 = 4*T/(D^2*Le*Y);         % Beam Bending Tensile Stress, MPa
St = (Ka*Km*(S1+S3)+S2)/Kf;         % Total Tensile Stress, MPa
```

Strength of Material:

```
Sy = 710;                             % Yeild Strength, in MPa
Sts = Sy*0.577;          % Maximum Allowable Shear Stress, in MPa
Stc = Sts/10;        % Maximum Allowable Compressive Stress, in MPa
Stt = Sts*1.1;          % Maximum Allowable Tensile Stress, in MPa
d_min = 3*sqrt(16*(T*0.738/1000)*Ka*1.5/(pi*Sts*(10.^6)*Kf))
```

Factor of Safety

```
Fsr = Sts/Ssr
Fsp = Sts/Ssp
Fc = Stc/Sc
Fb = Stt/St
```

For module = 1.75 and number of teeth = 20, Fsr = 4.5757; Fsp = 9.9773; Fc = 10.2899; Fb = 4.1635. **FEA Analysis:**

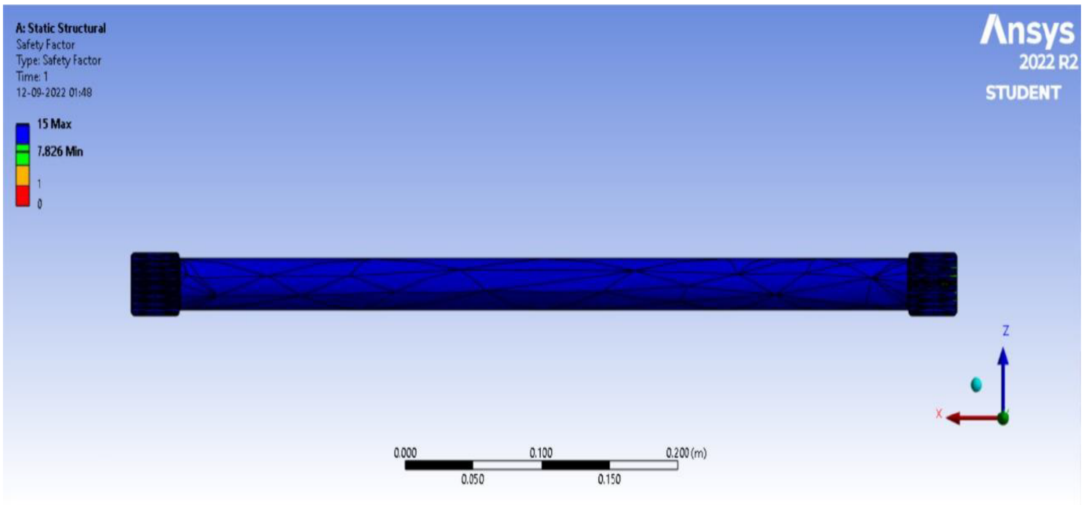


Figure 2.3: Factor of Safety = 1.826

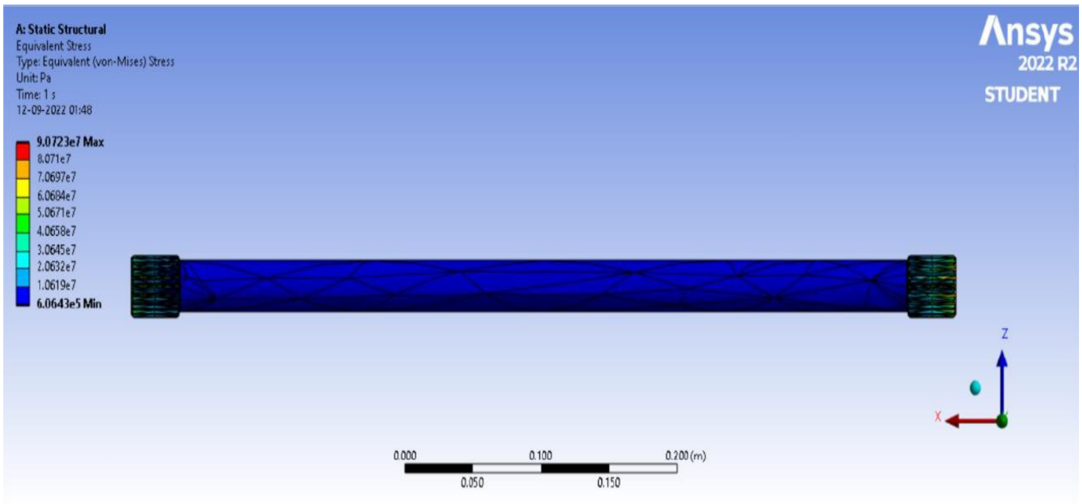


Figure 2.4: Equivalent Stress

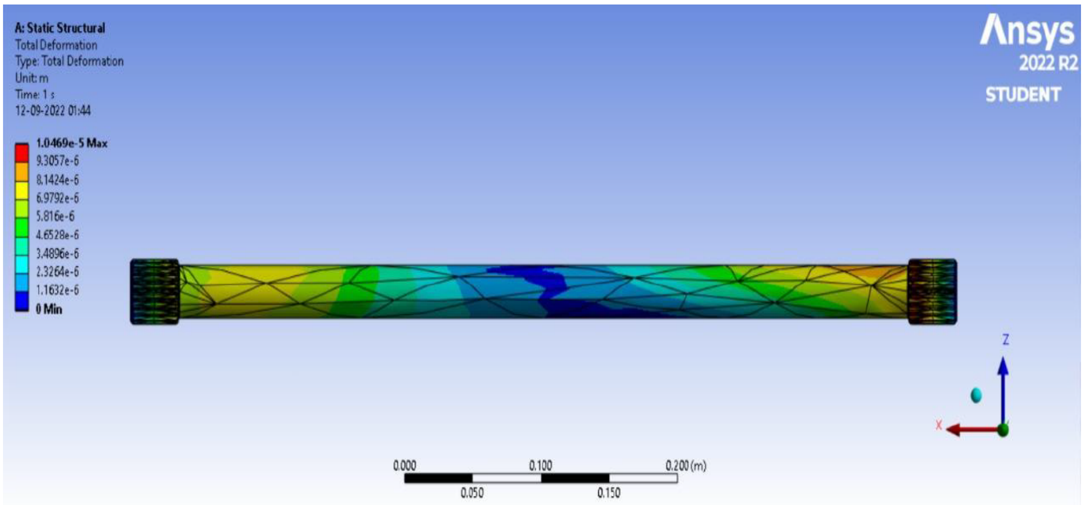


Figure 2.5: Total Deformation

2.6 Lap Time Simulation

Lap time for our vehicle in the track has been calculated with a simplified point mass model. This helps us to know how good our vehicle is in track and will help us to check how any optimisation in the vehicle affects its performance in the track. The code for the lap time simulation:.

2.6.1 optimum time lap

This part of the code has all the global variables declared and its starts the simulation by calling the otl function. This outputs the total time taken by the vehicle to complete the track once.

```
function [time] = optimum_time_lap()
%importing Track Data
%column1 right left straight  column 2 Arc length column 3 Radius of
%curvature
global SvT;
global trackdata;
global i;
global V;
global T;
global k;
k=0;
V=table2array(SvT(:,1));
T=table2array(SvT(:,2));

%defining frictional cofficiant, gravitational acc, and max
    dcceleration
%possible in car
u=1.4; g=9.8; dcc=1*g;

track_no=height(trackdata);
```

```
vi=0; time=0;

for i=1:(track_no-1)
    z=otl(vi,i);
    time=time+z(1,1);
    time=double(time);

    vi=z(1,2);
    vi=double(vi);
end
if(trackdata(track_no,1)=='S')
 time=double(time+str2double((trackdata(track_no,2)))/vi)

elseif(trackdata(track_no,1)=='L' || trackdata(track_no,1)=='R')
    Radius=str2double(trackdata(i,3));
    Arclen=str2double(trackdata(i,2));
    v_max= sqrt(u*g*Radius);

    if (vi==v_max)
        time=Arclen/vi;
        v_exit=vi;
    end

    if (vi<v_max)
        fun1 = @(v) 1 ./ (sqrt((u*g).^2-(v.^4 ./Radius.^2)));
        fun2 = @(v) v ./ (sqrt((u*g).^2-(v.^4 ./Radius.^2)));

        s1 = integral(fun2,vi,v_max);
        s2=Arclen-s1;

        if(s2>=0)
            time = integral(fun1,vi,v_max)+s2/v_max;
            v_exit = v_max;
        end
        if(s2<0)
            integralFunc = @(v) integral(fun2, vi, v);
            v = fminbnd (@(v) abs(integralFunc(v) - Arclen), vi, v_max)
    ;

            time = integral(fun1,vi,v);
            v_exit = v;

        end
    end

     if(vi>v_max)
        'error in'
        i
        time = Arclen/vi;
        v_exit = v_max;
     end

end
end
```

2.6.2 Simulation Code

This part of the code takes into account of all the possibilities the car may encounter and calculates the time taken for all the cases. This code uses the track data to find out the max velocities possible in terms of a turn and makes sure the velocity does not exceed that max velocity in case of a turn.

```
function [z] = otl(vi,i)
%importing Track Data
%column1 right left straight  column 2 Arc length column 3 Radius of
%curvature
global trackdata;
%defining frictional cofficiant, gravitational acc, and max
   dcceleration
%possible in car
u=1.4; g=9.8; dcc=2*g;



%encountring straight track with curve at the end
if (trackdata(i,1)=='S' && (trackdata(i+1,1)== 'R' || trackdata(i+1,1)
   == 'L'))
    Radius=str2double(trackdata(i+1,3));%...........Radius of curve
    Arclen=str2double(trackdata(i+1,2));%...........Arclength of curve
    distance=str2double(trackdata(i,2));%...........straight track
   distance.
    vmax= sqrt(u*g*Radius)%......................  speed targated to
    be achived at the end of straight line.

    v_final_possible= sqrt(vi^2 + 2 * u*g * distance);
    aaa=1
    %Maximum possible speed at the end acclerating through full
   potential


    if (vmax >= v_final_possible)
         bbb=2
         time=(v_final_possible-vi)/(u*g);
         v_exit=v_final_possible;
         VDAmain(V,T,vi,vf,distance);
    end

    if (vmax < v_final_possible)
         ccc=3
         time=straightlinetime(vi,vmax,distance);
         v_exit=vmax;
         VDAmain(V,T,vi,vmax,distance);
    end

end




if ((trackdata(i,1)== 'R' || trackdata(i,1)== 'L') && (trackdata(i+1,1)
   == 'S'))
    %ddd=4
```

```
    Radius=str2double(trackdata(i,3));
    Arclen=str2double(trackdata(i,2));
    v_max= sqrt(u*g*Radius);


    if (vi==v_max)
        eee=5
        time=Arclen/vi;
        v_exit=vi;
        VDAmain(V,T,vi,vf,Arclen)
    end

    if (vi<v_max)
        fff=6
        fun1 = @(v) 1 ./ (sqrt((u*g).^2-(v.^4 ./Radius.^2)));
        fun2 = @(v) v ./ (sqrt((u*g).^2-(v.^4 ./Radius.^2)));

        s1 = integral(fun2,vi,v_max);
        s2=Arclen-s1;

        if(s2>=0)
            iii=7
            time = integral(fun1,vi,v_max)+s2/v_max;
            v_exit = v_max;
            VDAmain(V,T,vi,v_max,s1);
            VDAmain(V,T,v_max,v_max,s2);
        end
        if(s2<0)
            jjj=8
            integralFunc = @(v) integral(fun2, vi, v);
            v = fminbnd (@(v) abs(integralFunc(v) - Arclen), vi, v_max)
    ;

            time = integral(fun1,vi,v);
            v_exit = v;
            VDAmain(V,T,vi,v,Arclen);

        end
    end

    if(vi>v_max)
        %kkk=9
        'error in'
        i
        time = Arclen/vi;
        v_exit = v_max;
    end

end


if ((trackdata(i+1,1)== 'R' || trackdata(i+1,1)== 'L') && (trackdata(i
    ,1)== 'R' || trackdata(i,1)== 'L'))
    Radius_1=str2double(trackdata(i,3));
    Arclen_1=str2double(trackdata(i,2));
    Radius_2=str2double(trackdata(i+1,3));
    Arclen_2=str2double(trackdata(i,2));

```

```
    v_max_1= sqrt(u*g*Radius_1);
    v_max_2= sqrt(u*g*Radius_2);
    lll=10


    if (v_max_1<=v_max_2)
        mmm=11

        if (vi==v_max_1)
            nnn=12
            time=Arclen_1/vi;
            v_exit=vi;
            VDAmain(V,T,vi,vi,Arclen_1);
        end


        if (vi<v_max_1)
            ooo=13
            fun1 = @(v) 1 ./ (sqrt((u*g).^2-(v.^4 ./Radius_1.^2)));
            fun2 = @(v) v ./ (sqrt((u*g).^2-(v.^4 ./Radius_1.^2)));
            s1 = integral(fun2,vi,v_max_1);
            s2=Arclen_1-s1;


            if(s2>=0)
                ppp=14
                time = integral(fun1,vi,v_max_1)+s2/v_max_1;
                v_exit = v_max_1;
                VDAmain(V,T,vi,v_max_1,s1);
                VDAmain(V,T,v_max_1,v_max_1,s2);
            end

            if(s2<0)
                qqq=15
                integralFunc = @(v) integral(fun2, vi, v);
                v = fminbnd(@(xo) abs(integralFunc(v) - Arclen_1), vi,
   v_max_1);
                time = integral(fun1,vi,v);
                v_exit = v;
                VDAmain(V,T,vi,v,Arclen_1);
            end
        end

        if(vi>=v_max_1)
            rrr=16
           % 'error in'
            %i
            time = Arclen_1/vi;
            v_exit = vi;
        end
    end

    if(v_max_1>v_max_2)
        sss =17
        fun_dis_acc = @(v) v./(sqrt((u*g).^2-(v.^4./Radius_1.^2)));
        fun_dis_dcc = @(v) -v./(sqrt((u*g).^2-(v.^4./Radius_1.^2)));

        fun_time_acc = @(v) 1./(sqrt((u*g).^2-(v.^4./Radius_1.^2)));
```

```

        integralFunc = @(vo) integral(fun_dis_acc, vi, vo) + integral(fun_dis_acc,v_max_2,vo);
        vo = fminbnd(@(vo) abs(integral(fun_dis_acc, vi, vo) + integral(fun_dis_acc,v_max_2,vo)-Arclen_1), v_max_2, 4*v_max_1);


        if(vo<=v_max_1)
            ttt=18
            time = integral(fun_time_acc, vi, vo) + integral(fun_time_acc, vo, v_max_2);
            v_exit = v_max_2;
            VDAmain(V,T,vi,vo,Arclen_1);
            VDAmain(V,T,vo,v_max_2,Arclen_1);

        end

        if(vo>v_max_1)
            if(vi==v_max_1)
                 s_acc=0;
                 s_dcc = integral(fun_dis_acc, v_max_2, v_max_1);
                 time=integral(fun_time_acc, vi, v_max_2) + (Arclen_1-s_acc-s_dcc)/v_max_1;
                 v_exit = v_max_2;
                 VDAmain(V,T,vi,vi,Arclen_1-s_dcc);
                 VDAmain(V,T,vi,v_exit,s_dcc);
            else
            uuu=19
            s_acc = integral(fun_dis_acc, vi, v_max_1);
            s_dcc = integral(fun_dis_acc, v_max_2, v_max_1);
            time  = integral(fun_time_acc, vi, v_max_1) + integral(fun_time_acc, vi, v_max_2) + (Arclen_1-s_acc-s_dcc)/v_max_1;
            v_exit = v_max_2;
            VDAmain(V,T,vi,v_max_1,s_acc);
            VDAmain(V,T,vi,v_max_2,s_dcc);
            VDAmain(V,T,v_max_2,v_max_2,arclen_1-s_acc-s_dcc);
            end
        end
    end
end
z=[time,v_exit];
```

2.6.3 straightline time computation

This part of the code is used to calculate the time in case of a straight line. In this we assume the acceleration of the car to be constant, this assures the applicability of Newton's laws of motion. Solving three of the available equations helps us finding the time taken to cover the straight line distance.

```
function [time] = straightlinetime(vi,vf,d)

%global trackdata;
u=1.4; g=9.8; dcc=1*g;

syms vo s1  s2
eq1=vi^2+2*u*g*s1-vo^2==0;
eq2=vf^2+2*dcc*s2-vo^2==0;
```

```
eq3=s1+s2==d;


result=vpasolve([eq1,eq2,eq3],[vo,s1,s2]);

vo=result.vo
vo = vo(result.vo > 0);


time = (vo-vi)/(u*g)+(vo-vf)/dcc;

end
```

2.6.4 VDAmain

This part of the code is called in multiple places to get and store the values of V,D,a, which is plotted at the last to get some informations. +

```
function [table] = VDAmain(V,T,vi,vf,d)
%for getting the velocity vs displacement vs acceleration graph
v1=vi;
n=1000;
h=(vf-vi)/n;
s=0;
if(vi==vf)
 for k=k:(k+n)           %using k to update values consecutively in the
    table
  v2=vi+k*h;
  s1=integral(@(v1) v1./((interp1(V,T,v1,'linear'))/(250*0.2)),v1,v2);
     %finding distance travelled for a small change in velocity
  v1=v2;
  s=s+s1;
  TAB(k+1,1)=v2;
  TAB(k+1,2)=s;
  TAB(k+1,3)=((interp1(V,T,v1,'linear'))/(250*0.2));
 end
else
    for k=k:(k+n)
    s1=d/1000;
    s=s+s1
    v=vi
  TAB(k+1,1)=v;
  TAB(k+1,2)=s;
  TAB(k+1,3)=0
    end
end
 x=TAB(:,1);          %velocity
 y=TAB(:,2);          %displacement
 z=TAB(:,3);          %acceleration
 table=TAB
end
```

2.7 Differential

Differential is chosen based on required minimum torque capacity = 140 N-m
According to the requirement, there are few options available in market

- Quaife LSD differential (QDF7ZR)
  Torque capacity = 15000 N-m
  TBR = Adjustable
  Cost = 37000 INR
- NC MX-5 TORSEN LSD (SMX LEGAL)
  Torque capacity = 5500 N-m
  TBR = 2.7:1
  Cost = 121786.7047 INR

We had used Quaife LSD differential (QDF7ZR) in our previous FSAE car.Despite it is having a very large torque capacity than what we required, we chose this as we are going to reuse that differential.

2.8 Differential Mount

Points kept in mind while choosing the dimensions for differential mount: -

- Angle between the arms of differential mount is fixed so that there should be only compressible/Tensile forces on the tabs connecting differential mount and chassis.
- There is Chain tensioner between upper differential mount arm and chassis, so according to length of chain tensioner, The length of upper arm is chosen.
- Thickness of differential mount is decided based of FEA analysis and bearing thickness.
- Diameter of mount is chosen as outer diameter of bearing.

2.8.1 Design:

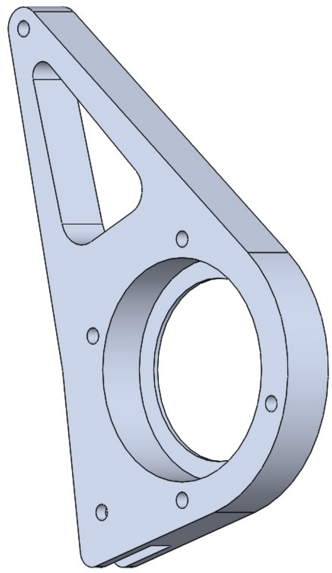
2.8.2 Force calculation of differential mount:

Parameters:

- Maximum torque of the motor = 140 N-M
- Maximum torque output after reduction through a reducer gearbox of gear ratio 2 = 280 N-M
- Radius of front sprocket = 53.25 mm
- Radius of the rear sprocket = 106.21 mm
- Distance between rear and front sprocket = 233.8 mm

Tension in the chain = $\frac{\text{Torque}}{\text{Radius of front sprocket}} = \frac{280}{53.25X10^{-3}} = 5258.2N$

Angle of force with horizontal = $sin^{-1}\frac{R-r}{l} = sin^{-1}\frac{106.51-53.25}{233.8} = 13.16°$



2.8.3 Material selection:

Aluminium 7075-T6 was chosen because it is light in weight compare to other materials yet it has great durability and can be easily machined.
FEA Analysis:

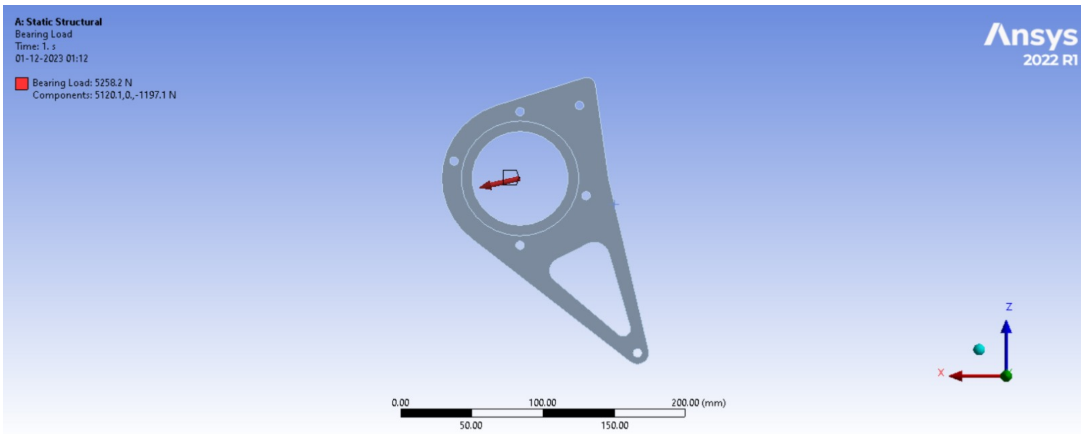


Figure 2.6: Bearing Load

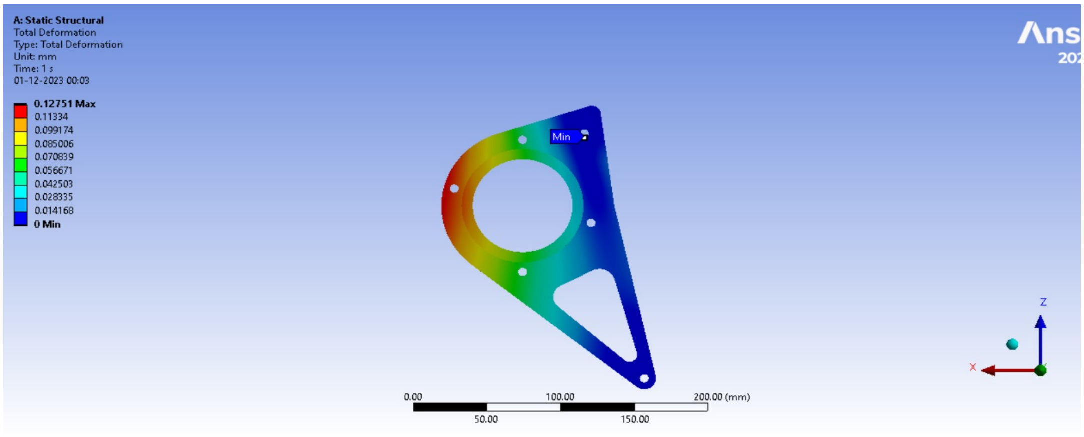


Figure 2.7: Bearing Load Deformation

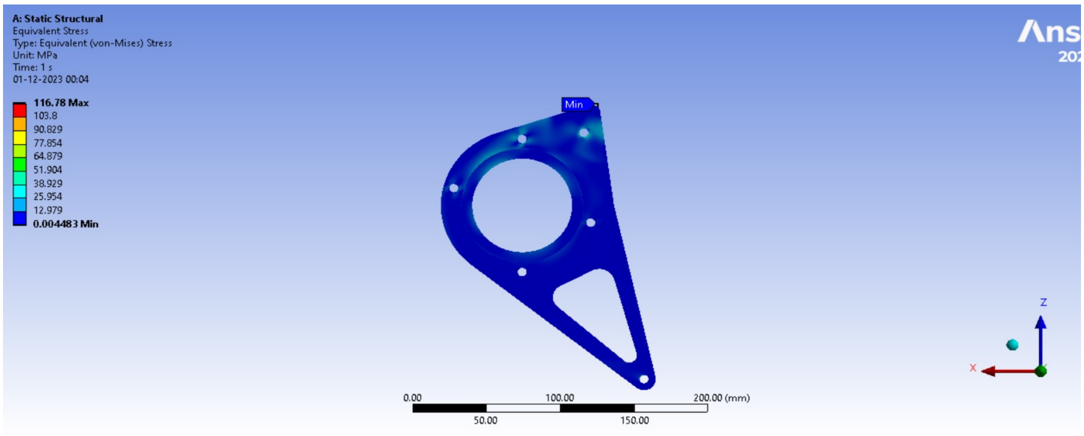


Figure 2.8: Equivalent Stress

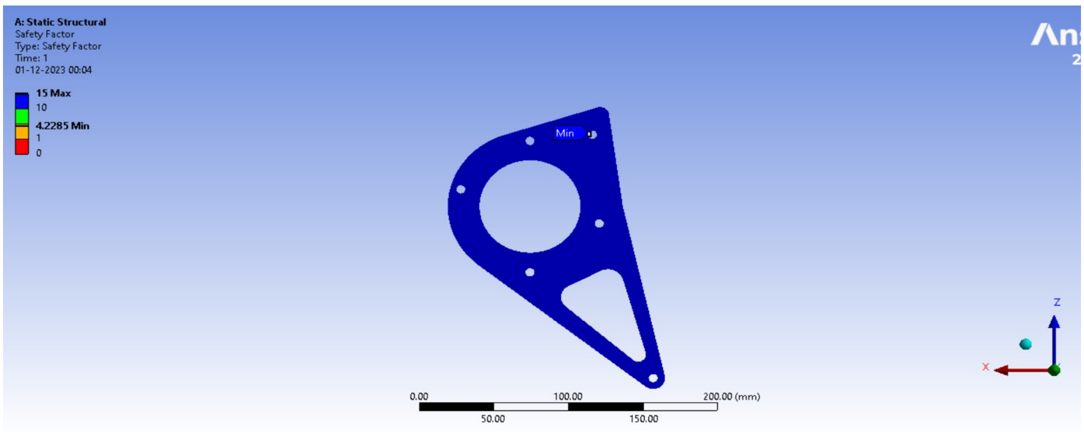


Figure 2.9: Safety Factor

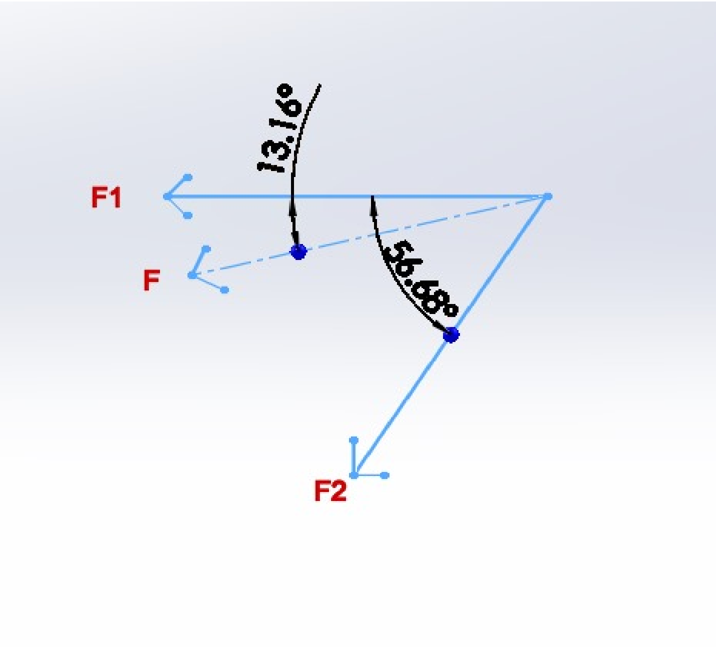
- Minimum FOS = 2.1
- Maximum Equivalent stress =$2.33 \times 10^{7}\ Pa$
- Maximum deformation = $2.5 \times 10^{-4}\ m$

2.9 Turnbuckle:

Subparts: - 2 rod ends + sleeve

- Rod ends was selected based on load bearing capacity.
- Dimensions of Sleeve is taken according to the selected rod ends.
- Sleeve's material was chosen as mild steel because of high tensile and impact strength.

Calculations:



- F = 5258.2 N
- $F_1cos(\theta_1) + F_2cos(\theta_2) = Fcos(\theta)$
- $F_1sin(\theta_1) - F_2sin(\theta_2) + Fsin(\theta) = 0$
  On solving,
  $F_1 = 4333.14$ N
  $F_2 = 1432.64$ N

- So, load on turnbuckle = 4333.14 N
- Load bearing capacity of the rods should be greater than 3903.40 N

Based on the above data, M6 rod ends are chosen SKF (SAKB 6 F).

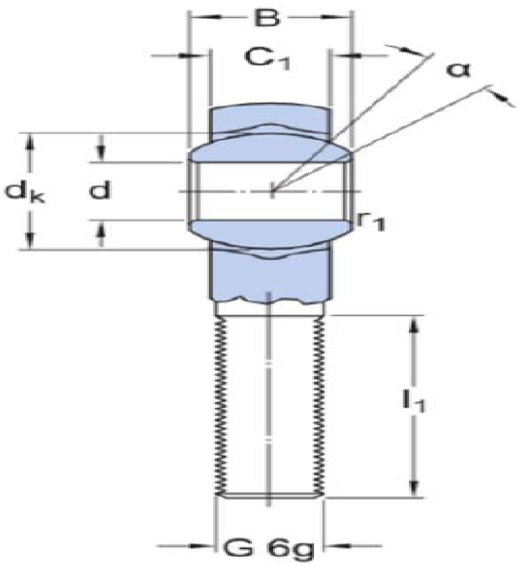


Figure 2.10: Rodend

l

d	Bore diameter	6 mm
	Basic static load capacity	4.25 KN
	Basic dynamic load capacity	6.8 KN
d2	Diameter head	max 21 mm
B	Width inner ring	9 mm
G	Thread	M 6
C1	Width head	max 7.5 mm
h	Height shank end face -center rod end eye	36 m
α	angle of tilt	17 degrees
dk	raceway diameter inner ring	12.7 mm
l1	length thread	min 21 mm
l2	length housing	max 48 mm
r1	chamfer dimension bore	min 0.3 mm

2.10 Bearings for differential assembly:

Two ball bearing are needed in differential assembly which will be connected between differential shaft and differential mount.
Points kept in mind while selecting the bearings:

- Radial and axial loads on bearing. Skf selection tool is used to choose bearings. All the calculations are done using this tool. Link for that has been given here
- Radial load = 5258.2 N
- Axial load = 0 N

2.11 Bearing Selected:

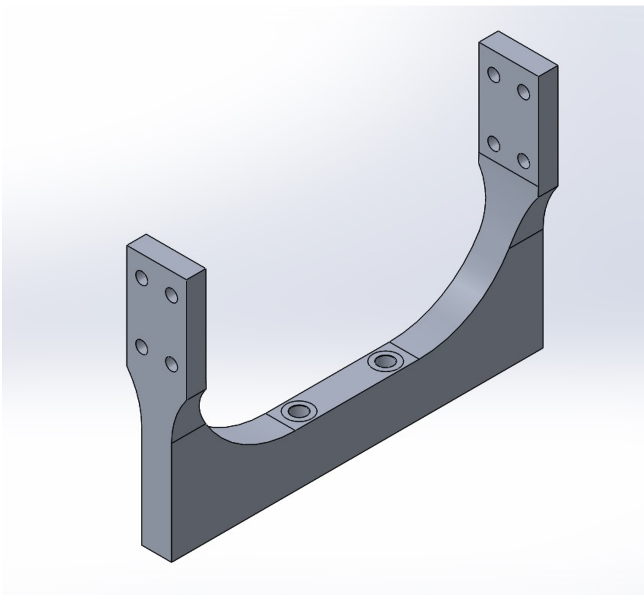
Type	**Deep groove ball bearing**
Model Name	SKF RLS 12
Bore diameter	38.1 mm
Outside diameter	82.55 mm
Width	19.05 mm
Basic dynamic load rating	30.7 kN
Basic static load rating	19 kN
Mass bearing	0.45 kg
Static safety factor	4.75

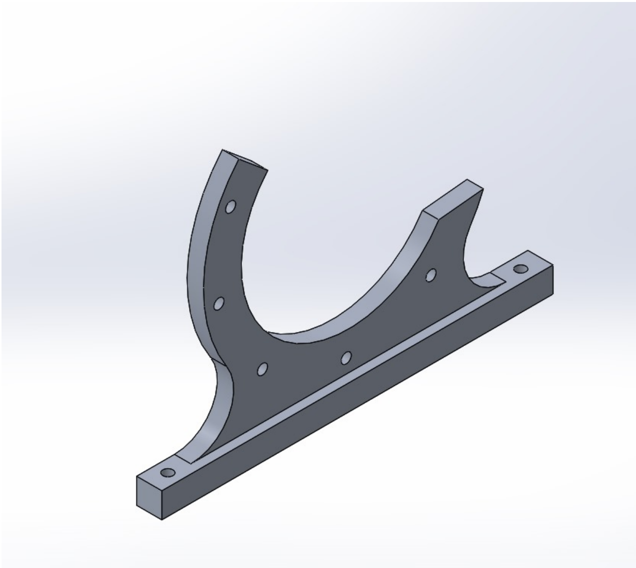
2.12 Motor Mounts

2.12.1 Methodology:

- Motor mounts have been designed with constraints to motor size and mounting positioning.
- Al7075 T6 is chosen because it is lightweight and easily manufacturable.
- Force calculation has been done, followed by FEA analysis.

2.12.2 Design:





2.12.3 Force Calculation:

Considering the weight of the motor = 35kg
Forces in X direction:

- Force due to tension in the chain = -F*cos(13.16°)= = -5120.1 N
- Force due to acceleration = -M*Max de-acceleration=35*14.89=521.15

TOTAL = -5641.25 N
Forces in Y direction:

- Force due to weight = -1000N
- Force due to tension in chain = -F*Sin(13.16°) = -1197.14N

TOTAL = -2197.14N
Forces in Z direction:

- Force due to lateral acceleration = M*A-lateral = 35 * 13.61 = 476.35 N

2.12.4 FEA Results:

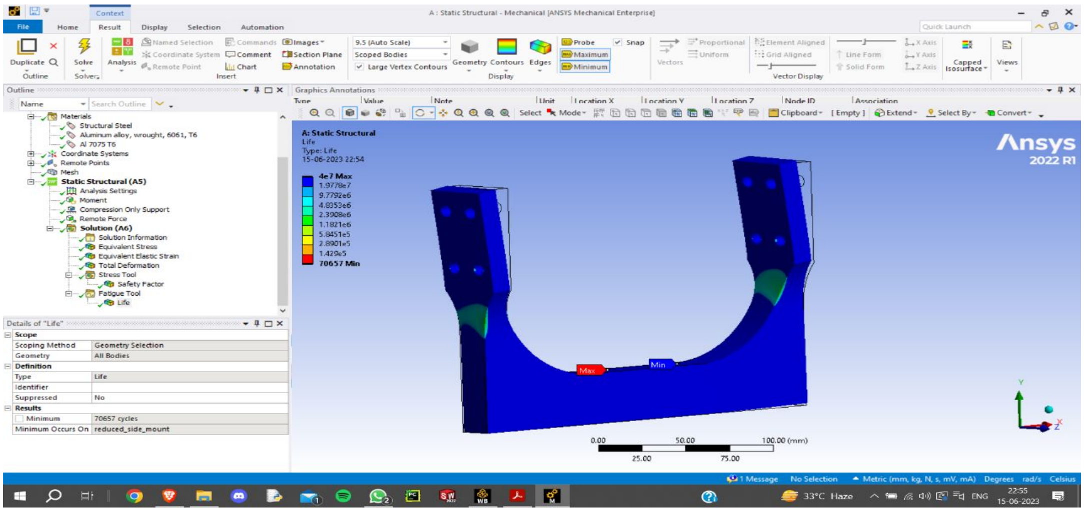


Figure 2.11: Bearing Load

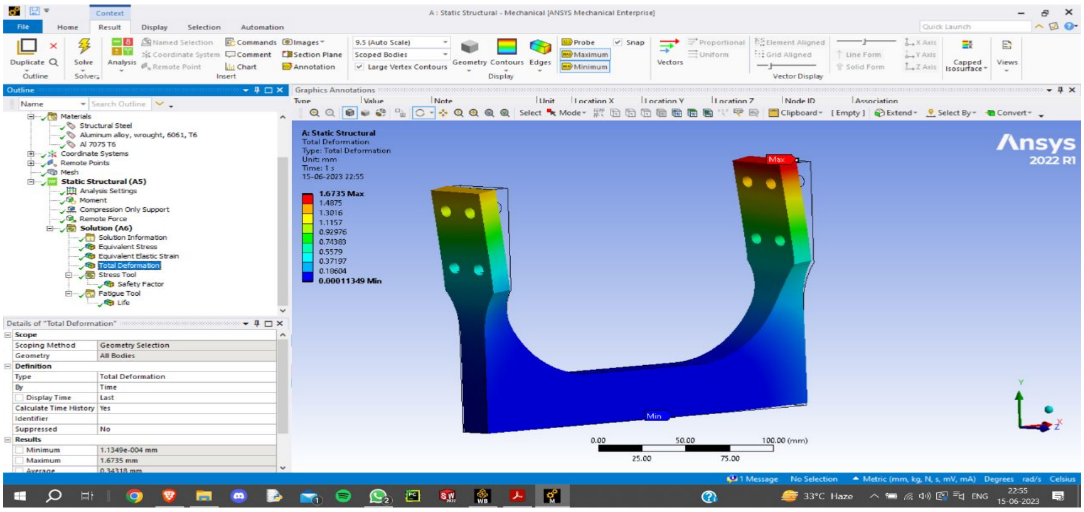


Figure 2.12: Total Deformation

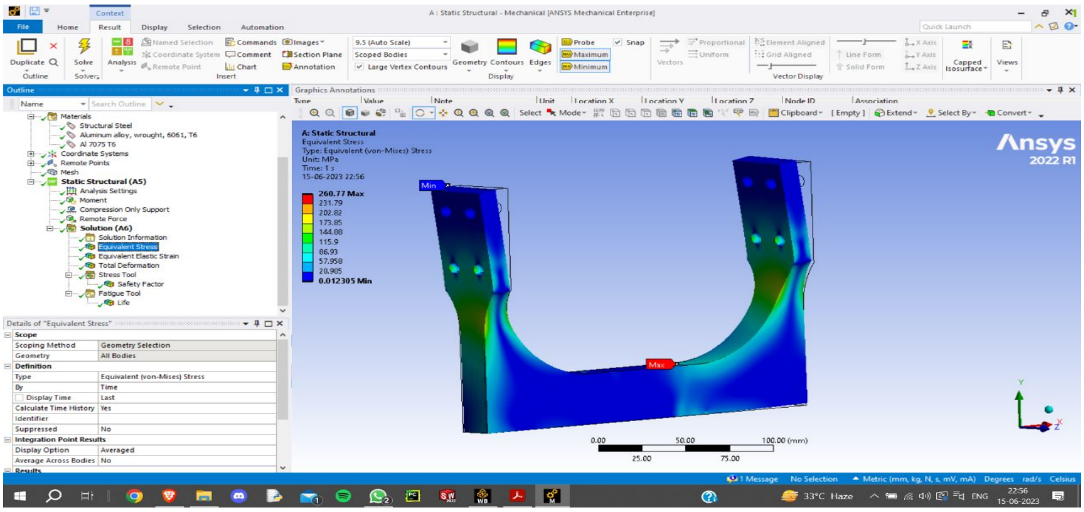


Figure 2.13: Equivalent Stress

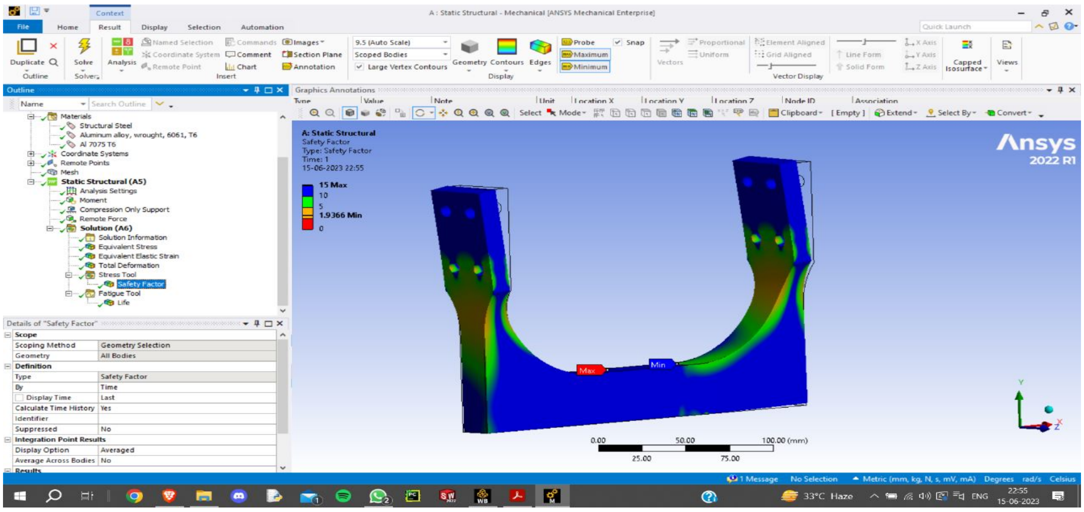


Figure 2.14: Safety Factor

2.13 Planetary Gears

- The 1st code systematically explores combinations of sun, planet, and ring sizes for a planetary gear system, applying various conditions and constraints to filter out valid configurations. The resulting valid combinations are stored in the matrix N for further analysis or use.

```
N = zeros(1,6); \\          % a,b,s,p,r,R,m1,m2,d,Rp,Rs,l(distance btw shafts)
 for module_pln = 1.5:0.5:2\\
     for  ring = 60:120\\
         for  planet = 20:ring/2\\
             sun = ring - 2 * planet;\\
             Reduction_pln = 1+ring/sun;\\
             if (floor(sun*Reduction_pln/3)==sun*Reduction_pln/3)
```

```
                & (sun > 14)\\
                if planet+2 < (sun+planet)*sin(pi/3)\\
                if Pin_Min(sun,planet) < sun\\
                if (gcd(sun,planet)==1 & (gcd(planet,ring)==1))\\
                    ring_od = ring*module_pln+25;\\
                if ring_od < 200\\\
                    N = [N; sun,planet,ring,module_pln,
    Reduction_pln,ring_od];\\

```

- The 2nd code consists of MATLAB function, FOS-Seq which computes factors of safety (fos-b and fos-c) for different load cases in a gear system. It uses input parameters related to gear sizes, forces, torques, and empirical factors to calculate stresses, strengths, and subsequently, factors of safety. The function iterates through load cases (10, 19, 20, 30), recursively calling itself, and stores the resulting factors of safety in arrays (FOS-10, FOS-20, FOS-30) for further analysis or storage.

```
function[fos_b, fos_c] = FOS_Seq(n1,n2,m,F,T,rpm,N)

 Tangential Load
D = m*n1/1000;
Wt = 2*T/D;

% Overload Factor
Ko = 1.25;

%Dynamic Factor
Q = 8;
v = pi*D*rpm/60;
B = 0.25*power((12-Q),2/3);
A = 50 + 56*(1-B);
Kv = power((A+sqrt(200*v))/A, B);

%Size Factor
load 'Y.mat';
Y = Yn(n1-14);
Ks = 0.8433*power((m*F*sqrt(Y)),0.0535);

%Load Distribution Factor
Cmc = 1;            % Uncrowned Gears
if F<25.4                               % Facewidth Misallignment
    Magnification
    Cpf = F/(10000*D)-0.025;
else
    Cpf = F/(10000*D) - 0.0375 + 0.0125*F/25.4;
end
Cpm = 1;            % Mounting Position
A = 0.0675;
B = 0.0125;
C = -0.926E-4;
Fi = F/25.4;
Cma = A + B*Fi + C*Fi*Fi;       % Manufacturing Accuracy
Ce = 1;                 % Extra Adjustment;
Kh = 1 + Cmc*(Cpf*Cpm + Cma*Ce);

% Rim Thickness Factor
Kb = 1;                 % For all practical purposes, Tr/Ht>1.2
```

```
]
%Geometric Factor
load 'J.mat';
Yj = J(n2-14,n1-14);

% Bending Stress
B_Stress = Wt*Ko*Kv*Ks*(1/(F*m))*(Kh*Kb/Yj);
\begin{lstlisting}[language = Matlab]
%Elasticity Coefficient
Ze = 191;
]
%Surface Condition Factor
Zr = 1;                          % should be taken >1 for
    detrimental surface finish effect

%Surface Strength Geometry Factor
phi = 20;
mg = n2/n1;
mn = 1;                          % for spur gears
Zi = (cosd(phi)*sind(phi)/(2*mn))*(mg/(mg+1));       % mg-1 for
    internal gear

% Contact Stress
C_Stress = Ze*sqrt((Wt*Ko*Kv*Ks*Kh*Zr/(1000*D*F*Zi)));

% Strength
Hbc = 363;                        % Nitrided AISI 4140 core Brinell
    Hardness
St = 0.749*Hbc + 110;             % Grade 2 Nitrided Through
    Hardened
Yn = 1.3558*power(N,-0.0178);    % Ctress Cycle Factor;
Kt = 1;
Kr = 1.25;                       % Reliability = 0.999
B_Strength = St*Yn/(Kt*Kr);

Hbs = 700;               % Nitralloy Case brinell hardness
Sc = 2.41*Hbs + 237;
Zn = 1.4488*power(N,-0.023);
Zw = 1;
Kt = 1;
Kr = 1.25;                       % Reliability = 0.999
C_Strength = Sc*Zn*Zw/(Kt*Kr);

fos_b = B_Strength/B_Stress;
fos_c = C_Strength/C_Stress;

rpm_carier = rpm_sun./N(:,5);
rpm_planet = (rpm_sun-rpm_carier).*(N(:,2)./N(:,1));
torque_carier = torque_inp*N(:,5);
torque_planet = torque_inp*(N(:,2)./N(:,1))/3;
% fos_sun, fos_planet
FOS_10=[0,0,0,0,0];
FOS_20=[0,0,0,0,0,0];
FOS_30=[0,0,0,0,0];

for i = 2:169
    [sunb,sunc] = FOS_Seq(N(i,1),N(i,2),N(i,4),10,torque_inp/3,
    rpm_sun-rpm_carier(i),1e7);
```

```
    [planetb,planetc] = FOS_Seq(N(i,2),N(i,1),N(i,4),10,torque_planet(i),rpm_planet(i),2e7);
    FOS_10 = [FOS_10; sunb sunc planetb planetc N(i,5)];

    [sunb,sunc] = FOS_Seq(N(i,1),N(i,2),N(i,4),19,torque_inp/3,rpm_sun-rpm_carier(i),1e7);
    [planetb,planetc] = FOS_Seq(N(i,2),N(i,1),N(i,4),20,torque_planet(i),rpm_planet(i),2e7);
    FOS_20 = [FOS_20; sunb sunc planetb planetc N(i,5) N(i,4)];

    [sunb,sunc] = FOS_Seq(N(i,1),N(i,2),N(i,4),30,torque_inp/3,rpm_sun-rpm_carier(i),1e7);
    [planetb,planetc] = FOS_Seq(N(i,2),N(i,1),N(i,4),30,torque_planet(i),rpm_planet(i),2e7);
    FOS_30 = [FOS_30; sunb sunc planetb planetc N(i,5)];
End
```

- The 3rd code computes the forces acting on a planetary gear system's sun and planet gears under varying load conditions.The loop iterates through load factors to analyze gear interactions and behavior for different load scenarios in the planetary gear assembly, storing the results in a matrix for further examination.

```
N = [0 0]
E = 2E+11
T = 40                                      %Torque input in N-m
rpm = 16000                                 % rpminput in revolution per minute
reduction_plntry = 4.636                    %reduc=gear redution
rs = 0.022                                  %radius of sun gear in m
rp = 0.029                                  %radius of planet gear in m
phi = 20*(pi/180)                  %phi = pressure angle
rpmC = rpm/reduction_plntry                 %carrier rpm
torque_carrier   = T*reduction_plntry       %Torque carrier in N-m(torque coming on 2nd plate of carrier)==inputT for pinion
L = rs + rp                                 %L=length of arm of carrier
Ft_sun_planet = T/(3*rs)                        %Ftinput = F tangential input
Fn_sun_planet = Ft_sun_planet*tan(pi/9)
F1 = torque_carrier/(3*L)                   %Ft = tangential force in N
dist_bw_carrier_plates = 32                 %measured from outer surface(plate thickness - 5mm)

for Nr = 1:0.01:1.5
    %SUN-PLANET MESH WITH Nr TIMES LOAD
    Fn_sun_planet1 = Nr*Fn_sun_planet
    Ft_ring_planet1 = Nr*(F1 - Ft_sun_planet)
    Fn_ring_planet1 = Ft_ring_planet1*tan(pi/9)
    Fn_net1 =  Fn_sun_planet1 - Fn_ring_planet1

    %REMAINING 2 PLANETS
    Fn_sun_planet2 = ((3-Nr)/2)*Ft_sun_planet * tan(pi/9)
    Ft_ring_planet2 = ((3-Nr)/2)*(F1 - Ft_sun_planet)
    Fn_ring_planet2 = Ft_ring_planet2*tan(pi/9)
    Fn_net2 = Fn_sun_planet2 - Fn_ring_planet2
```

```

    Fn_net = Fn_net1 - Fn_net2

    N = [N;Nr Fn_net]
    %Fn_net should be 0...

end
```

- The 4th code computes various force-related parameters, distribution, and resultant forces on a pinion gear based on the applied torque, gear geometry, and radii considerations.

```
torque_pinion = T * reduction_plntry
F_tangential_pinion = torque_pinion*1000/(3*41/2)
F_radial_pinion = F_tangential_pinion * tan(phi)
F = sqrt(F_radial_pinion^2 + F_tangential_pinion^2)
r1 = 60              %in mm
r2 = 24              %in mm
R1_t = F_tangential_pinion * r2/(r1+r2)
R1_r = F_radial_pinion * r2/(r1+r2)
R2_t = F_tangential_pinion * r1/(r1+r2)
R2_r = F_radial_pinion * r1/(r1+r2)

R1 = sqrt(R1_t^2 + R1_r^2)
R2 = sqrt(R2_t^2 + R2_r^2)
len = 84
SF = R1/3
BM = R1*0.0265
BM_at_pinion = R1*0.060

d_pin = 10

I = (pi/32)*0.030/(power(0.030,4) - power(0.025,4))
```

- The 5th code computes bending moments, fatigue-related parameters, critical diameter, stresses, fatigue life, and deflection. By analyzing these factors, it assesses the planet axle's ability to withstand loads and endure repeated stress cycles, aiding in designing a robust and reliable mechanical system.

```
M1 = 1.1*F1 * dist_bw_carrier_plates*0.001/2
M2 = 1.1*Fn_net1 * dist_bw_carrier_plates*0.001/2
M = sqrt(power(M1,2) + power(M2,2))
torque_planet = 0
n = 1.75
Din = 0

q = 0.82                  %Notch Sensitivity(from graph)
qshear = 0.85

Kt1 = 1.7;
Kf1 = 1.7;
Kts1 = 1.5;
Kfs1 = 1.5;


Kb1 = 0.9;
Kts = 1.42;                  %Stress concentration for shaft under
    torsion(from grap(A-15-8)).
```

```
Kt = 1.65;                    %Stress concentration for shaft under
    bending(from graph(A-15-9)).
Kf = 1 + q*(Kt-1);
Kfs = 1 + qshear*(Kts-1);


% ENDURANCE LIMIT
Sy=470 * power(10,6);
Sut=745 * power(10,6);
Se1 = 0.5*Sut;
Ka = 4.51*(Sut*power(10,-6))^(-0.265);
Kc = 1;                               % 1 For Bending    0.59 for Torsion
Kd = 1.020;                           %For temp. approx. 100deg C
Ke = 0.702;                           %For 99.99% reliability
kf = 1;                               % Misc.Factor
Se_dash = Ka*0.9*Kc*Kd*Ke*kf*Se1;

a = (2*Kf1*M/Se_dash) + (sqrt(3)*Kfs1*torque_planet/Sut);
D_ref = power((16*n/pi)*a,1/3)
Kb = 1.24 * power(D_ref*1000,-0.107);   %Kb = 1.24*power(Dout
   *1000,-0.107);
Se = Ka*Kb*Kc*Kd*Ke*kf*Se1;

sigma_a = 32*Kf*M/(pi*D_ref^3);
sigma_m = sqrt(3)*16*Kfs*torque_planet/(pi*D_ref^3);


nf = 1/((sigma_a/Se) + (sigma_m/Sut));

%DEFLECTION
I1 = (pi/32)*power(D_ref,4)
def01 = F1 * power(dist_bw_carrier_plates*0.001,3)*5/(12*E*I1)
```

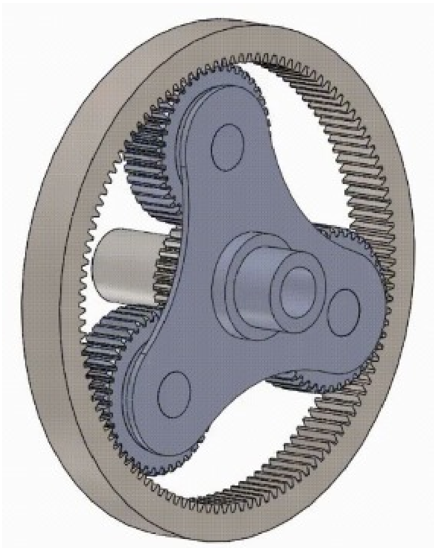


Figure 2.15: Planetary Gears

.

Chapter 3

HV Accumulator and AMS

3.1 Position of HV Components in Chassis

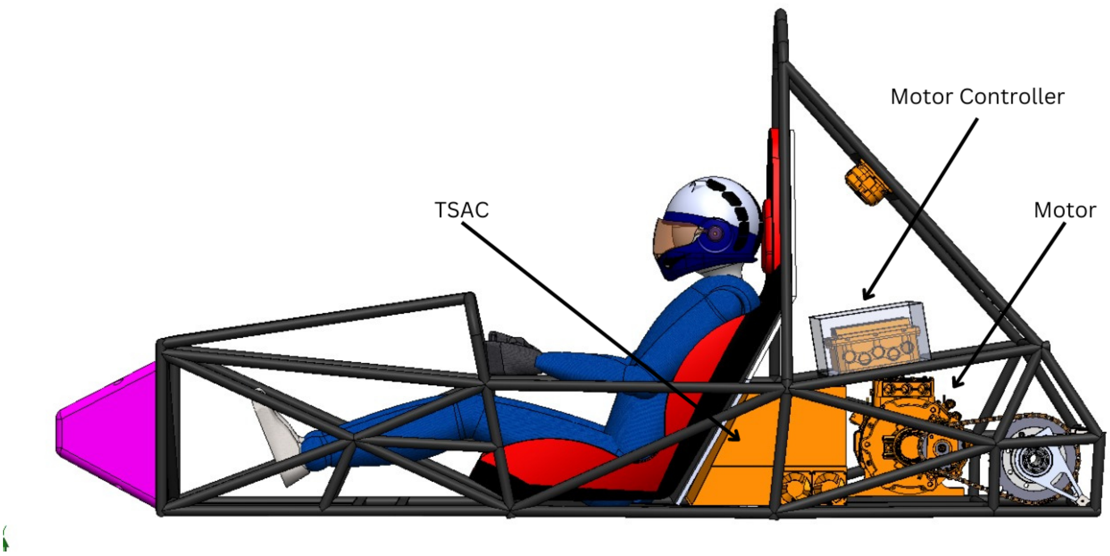


Figure 3.1: Accumulator, Motor and Motor Controller position in Chassis

The accumulator is placed midway below the main hoop and its highest point lies below the upper SIS. It is separated from the cockpit by the firewall and seat. HV wires beginning from the accumulator go to the motor controller which is kept above the supporting tubes and below the main hoop braces. 3 phase AC wires then carry current from the motor controller to the motor placed behind the TSAC.

3.2 Battery Pack Parameters

- Peak Motor Power(P_{max}): 30 kW
- Continuous Motor Power(P_{cont}): 15 kW
- Maximum Motor Current(I_{max}): 520 A

- Continuous Motor Current(I_{cont}): 135 A
- Maximum battery voltage = 96 V DC
- Motor Controller operating voltage = 96 V
- Max cell voltage = 4.2 V
- Nominal cell voltage = 3.7 V

3.3 Number of Cells Calculation

$$96/4.2 < N_s < 96/3.7 \implies 22.85 < N_s < 25.95$$
$$\mathbf{N_s = 24}$$

$$\text{Number of Cells in Parallel} = N_p \tag{3.1}$$

$$N_p = \frac{\text{Energy required in } Wh}{N_s \cdot \text{Cell Capacity} \cdot \text{Cell Nominal Voltage}} \tag{3.2}$$

$$\tag{3.3}$$

$$\text{Energy Required from simulation} = 2.8 \text{ kWh} \tag{3.4}$$
$$\text{Energy used for Design (Saftey factor)} = 3.7\text{kWh} \tag{3.5}$$
$$\tag{3.6}$$

$$= \frac{3.7 \cdot 1000}{24 \cdot 21 \cdot 3.7} \qquad = 1.98 \tag{3.7}$$
$$\tag{3.8}$$

$$\mathbf{N_p} \approx 2 \tag{3.9}$$

$$\begin{aligned}
\textbf{Total Capacity of Battery} &= N_p \cdot N_s \cdot \text{Cell Nominal Voltage} \cdot CellCapacity && (3.10)\\
&= 7 \cdot 27 \cdot 3.7 \cdot 9.7 && (3.11)\\
&= \mathbf{3.7\ KWh} && (3.12)\\
\textbf{Total Number of Cells} &= 2 \times 24 && (3.13)\\
&= \mathbf{48} && (3.14)
\end{aligned}$$

3.4 FR4 Datasheet

Datasheet for the FR4 sheet used is provided here

3.5 Melasta Lithium-Polymer Cell Datasheet

Datasheet for the $LiCoO_2$ cells used is provided here

3.6 Cell Connections

Parallel assemblies within a segment are connected to each other using copper busbars. Copper busbars are proven to have low terminal resistance and better heat dissipation. Bolting of tabs leads to lower area for the current to flow, resulting in higher terminal resisitance.
The negative terminal of one PA is separated from the postive terminal of the next using FR4 sheets.

- Area of Copper busbar (A) = $30mm \cdot 3mm$
- Resistivity of copper (ρ)= $1.724 \cdot 10^{-8}\rho$
- Length between 2 tabs(L) $\approx 45mm$

Terminal resistance due to one copper busbar between 2 PAs =

$$\frac{\rho \cdot L}{A} \tag{3.15}$$

$$\implies \mathbf{8.62 \cdot 10^{-6}\,\Omega}$$

3.7 High Voltage Wiring Calculations(inside Accumulator)

- Max current in HV circuit(I) = 600 Amp
- The conductivity of the material used (copper) = $58.14x10^{\,S/m}$.
- Approximate length of the wire (L) = 1 m
- % of Maximum allowable voltage drop (U) = 1% of rated HV voltage
- Area of the cross-section of the wire A $\geq \frac{\times L}{G \times U} = 20.64\ mm^2$

3.8 AMS

The goal behind selecting the AMS is as follows:

- Overcharge and over discharge protection
- Over current protection
- Under temperature and Over temperature protection
- Short circuit and Reverse polarity protection

Orion AMS 2 with the following specification is perfectly compatible with the rest of vehicle and the above objectives. Its centralized nature allows to optimize space and structural requirements.

Specification Item	Min	Typ	Max	Units
Input Supply Voltage	8		30	Vdc
Supply Current—Active (at 25 degrees Celsius)		< 2		Watts
Supply Current—Sleep (at 25 degrees Celsius, 12vDC)		450		µA
Operating Temperature	-40		80	C
Sampling Rate for Current Sensor		8		mS
Sampling Rate for Cell Voltages		25	40	mS
Isolation Between Cell Tap #1 and Chassis / Input Supply	1.5			kVrms
Isolation Between Cell Taps #2+ and Chassis / Input Supply	2.5			kVrms
Isolation Between Cell Tap Connectors	2.5			kVrms
Digital Output Switching Voltage (Open Drain)			30	V
Digital Output Sink Continuous Current (**Some outputs can pulse up to 4A for contactors**—see wiring manual for details)			175	mA
Cell Voltage Measurement Range	0.5		5	V
Cell Voltage Measurement Error (over 1-5v range)			0.25	%
Cell Balancing Current			200	mA
Cell Current (Operating)		0.5		mA
Cell Current (Low Power Sleep)		50		µA
Thermistor Accuracy		1		C
Cell Voltage Reporting Resolution		0.1		mV

Figure 3.2: Orion AMS 2 Specification

3.9 HV Schematics

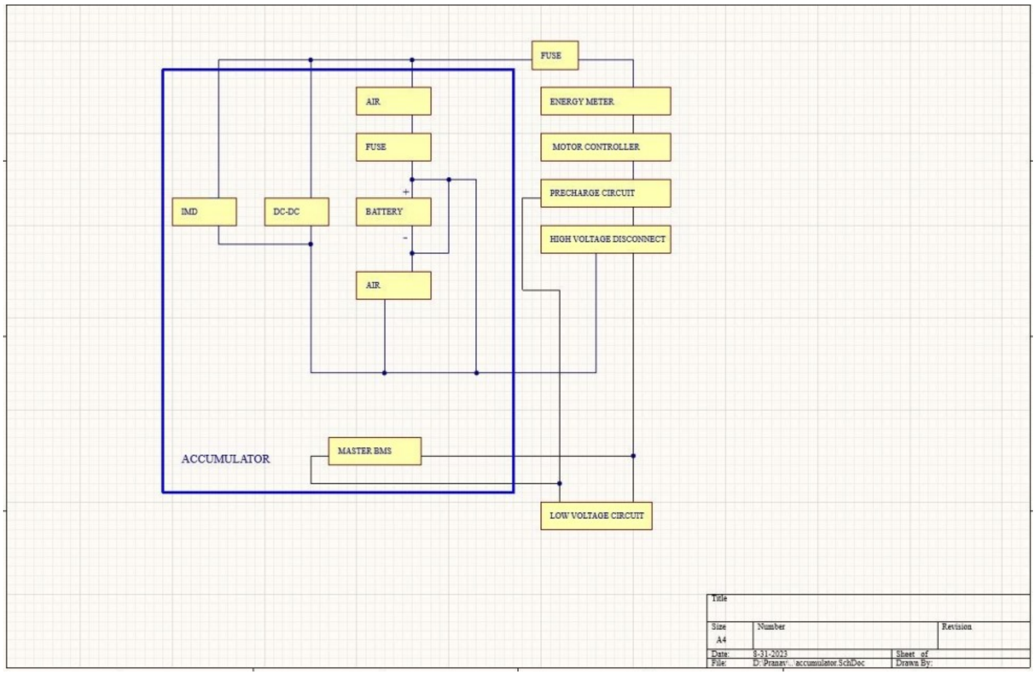


Figure 3.3: Electric Layout of Accumulator

Chapter 4

Electronics

4.1 LV Wiring Calculations:

- Current in the LV circuit (I) = **1.25 Amp**
- The resistivity of the material used (copper) = $1.72 * 10^{-8}\ \Omega \cdot m$.
- Approximate length of the wire (L) = 8 m
- % of Maximum allowable voltage drop (U) = 0.5% of 12V = **0.006 V**
- Area of the cross-section of the wire A $\geq \frac{2\times I\times L}{G\times U}$ $=$ **0.57336 mm²**
- Suitable AWG rating of the wire = 18AWG

4.2 HV Wiring Calculations

4.2.1 Wiring Calculations for Motor Controller

- Max current in HV circuit(I) = **200 Amp**
- The resistivity of the material used (copper) = $1.72 * 10^{-8}\ \Omega - m$.
- Approximate length of the wire (L) = 2 m
- % of Maximum allowable voltage drop (U) = 1% of rated HV voltage
- Area of the cross-section of the wire A $\geq \frac{2\times I\times L}{G\times U}$ $=$ **11.008 mm²**

Hence 50 mm^2 HV shielded cable is selected.

4.2.2 Wiring Calculations for Motor

- Max current in HV circuit(I) = **200 Amp**
- The resistivity of the material used (copper) = $1.72 * 10^{-8}\ \Omega \cdot m$.
- Approximate length of the wire (L) = 1.5 m
- % of Maximum allowable voltage drop (U) = 1% of rated HV voltage

- Area of the cross-section of the wire A $\geq \frac{2\times I\times L}{G\times U} = \mathbf{10.7569\ mm^2}$

Hence 50 mm^2 HV shielded cable is selected.

4.3 Insulation Monitoring Device(IMD)

The IMD device used is the **Bender ISOMETER IR155-3204**. It is an insulation monitoring device that continuously measures the resistance between the HV-conductors of an electrical system and reference earth (chassis ground). The IMD continuously monitors whether the insulation resistance is above the response resistance set at $100K\Omega$ and the response value of IMD at $500\Omega/V$. The IMD is placed inside the accumulator container. If the resistance measurement drops below this threshold of $100K\Omega$, an under-voltage is detected, and **the IMD will pull its status output to Low** and open the shutdown circuit. A similar response occurs in the case of earth faults and interruption of the earth connection. The IMD has an automatic self-test ability.

4.3.1 Wiring

The IMD is connected to the HV bus after the AIR using Molex Minifit connectors and ring terminals. LV connections enter the accumulator container through Deutsch DTM series connectors and are connected to the IMD using Samtec connectors. The output from the IMD exits the accumulator container through Deutsch DTM series connectors and enters the motor controller enclosure, and connects to the latching circuit using fixed terminal connectors. The IMD indicator light signal from the latching circuit is connected to the indicator located on the dashboard. HV connections to the IMD are made using 18 AWG wires and low voltage connections with 18 AWG wires.

4.4 Reset/Latching for IMD and AMS

The Latching circuitry of IMD and AMS are similar, except that IMD status is an active LOW signal while AMS status is not. The latching circuit is designed such that if the AMS or IMD throws a fault, the shutdown circuit latches off until the reset button is pressed. In both, the circuits relay K1 and K2 are solid state relays, normally open relays.

The status of both IMD, i.e., when no fault is detected, IMD gives a HIGH signal and a LOW signal if any fault is detected. The latching circuit is designed such that to close the Shutdown circuit; one needs to press the SET-RESET button for a moment initially. If IMD status is OK, then Relays K1 and K2 will close as soon as the SET-RESET button is pressed, and then the DPST relay (shown in the Diagram) will close and thereby closing the SDC. After the switch is released, the relay K1 will open, but the DPST relay will remain closed. If the IMD throws a fault, relay K1 opens, and the shutdown circuit latches off until the reset button is pressed. An additional NOT gate is added in case of AMS latching to account for the fact that the AMS signal is not active LOW.

- Maximum Accumulator Voltage Output = **96V**
- Response Value = $500 * 96 = \mathbf{48000\Omega}$

4.4.1 Wiring

The output from the IMD will exit the accumulator container through Deutsch DTM series connectors. It will enter the motor controller enclosure through the same and be connected to the power stage by fixed terminal blocks mounted on PCB with **18** AWG-rated wires. Wires inside the accumulator indicator will be rated for max TS voltage and surrounding temperature.

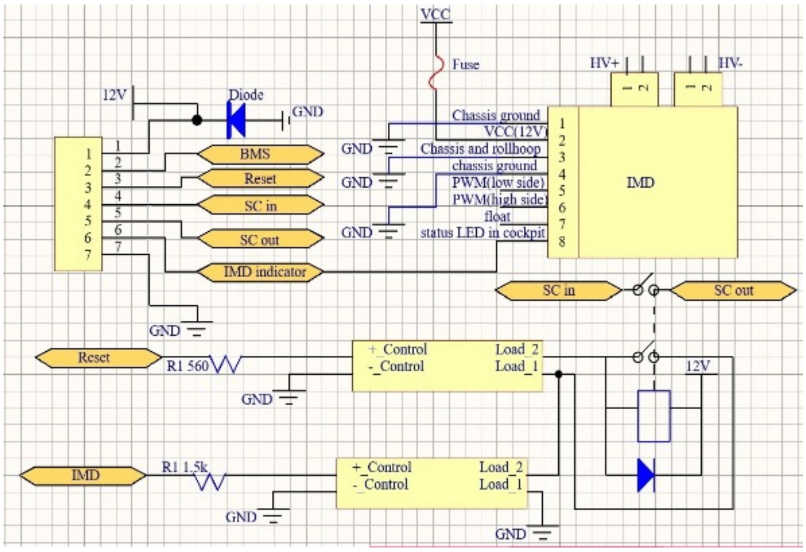


Figure 4.1: TSALReset/Latching for IMD and AMS

4.5 Brake System Plausibility Device (BSPD)

It takes inputs from brakes sensors and current sensor. Then by comparators they are checked if they lie between given range of voltages(plausible) then output will be 0, else(implausible) the output will be 5V. The outputs are sent to AND gate and a NOR gate to check if both are high or both are low. Output of AND gate is passed to an RC circuit followed by a comparator, which works as a time delay of 500ms, Similarly that of NOR gate is sent to a time delay of 10s. If output of 500ms delay is HIGH, it means that for previous 500ms both the inputs are implausible, so Shutdown circuit must be switched off. If output of 10s delay is HIGH, it means that for previous 10 seconds both the inputs are in plausible ranges so the shutdown circuit should be closed. To do this function, we gave the outputs of the delays to a SR latch, and its output is sent to a relay which controls the shutdown circuit.

4.5.1 Wiring

The current sensor signal, brake pressure signal, power supply, and output signals enter the motor controller enclosure through a DTM series connector and connect to the fixed terminal connector mounted on PCB. Inline connections are made using 18 AWG wires.

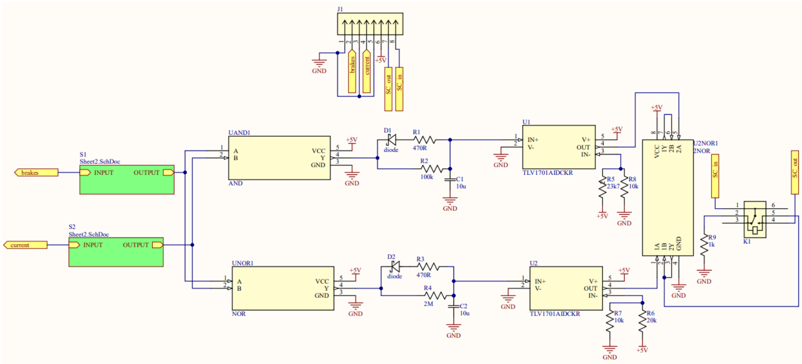


Figure 4.2: Brake System Plausibility Device (BSPD)

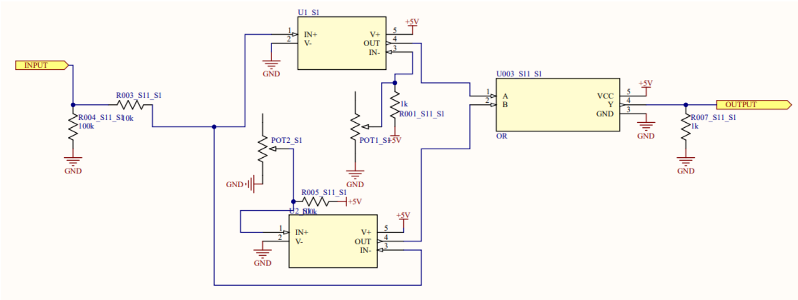


Figure 4.3: Filter Circuit for BSPD (Click here for enlarged photo of the schematic)

4.6 Tractive System Active Light (TSAL)

The Tractive system active light is used to indicate the state of the tractive system, which glows **Green**; if the tractive system is off and the mechanical state of AIRs and precharge relay are opened, and less than 60 V is detected by the DC-DC converter, which is placed inside the Accumulator container. It glows **Red** if any of the AIRs or precharge relay is closed and greater than 60V is detected by the DC-DC converter placed inside the Accumulator container. TSAL **does not glow** when GLV is powered off, or there is implausibility between the two DC-DC converters used in our system. One DC-DC converter is placed in the accumulator container across the AIR, and the other in the motor controller enclosure is wired parallel to the discharge relay. Inputs are taken from HV+ and HV-. HV+ are fed into the DC-DC converter in the accumulator container. This checks if the voltage outside the accumulator is greater than 60V though it is placed in the accumulator container. If the input voltage to the DC-DC converter is greater than 60V, then it will output a voltage of 12V, and if the input voltage is less than 60V, the output will be 0V. The outputs from the 2 DC-DC converters are given to an XNOR gate whose output will act as a power supply for all the ICs in the TSAL circuit. The state of the AIR and Precharge relays are inferred by the auxiliary contacts, where the output from the auxiliary contacts will be high if the Relays are closed. The outputs from the DC-DC converter and auxiliary contacts of the AIRs and precharge relay are given to a four-input OR gate, and the output from the OR gate is fed such that it produces red or green light according to the given rules. If the tractive system is on and the AIRs and precharge relay are closed, output from the AND gate will power the oscillator through an nMOSFET (T2). The output from the oscillator is given to the Red LED through an nMOSFET(T3) which illuminates that red bulb with a 3.6Hz frequency. If conditions are not satisfied, a green bulb will illuminate continuously through a pMOSFET(T4). Both the red and green LEDs are placed in a single housing.

4.6.1 Wiring

The output from the DC-DC converter will exit the accumulator container through a Deutsch DTM series connector by 18 AWG wires rated for max TS voltage and surrounding temperature. Connections to the TSAL PCB will be made through the fixed terminal blocks mounted on PCB by 18 AWG wires. These connections will enter the Motor controller enclosure through the Deutsch DTM series connector.

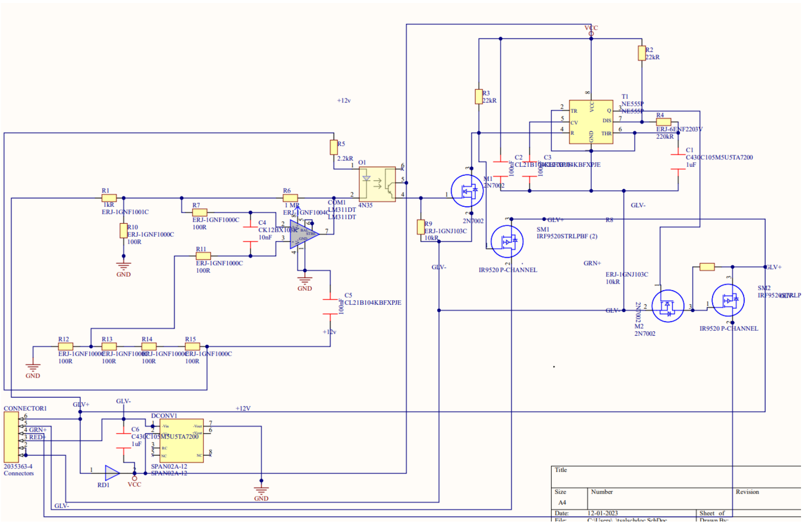


Figure 4.4: TSAL (Click here for enlarged photo of the schematic)

4.7 Ready-To-Drive-Sound (RTDS)

RTDS is used to put the car into Ready-to-drive mode. The functionality of RTDS is through programmable logic. The RTDS will not be able to function when the shutdown circuit has been activated. Debouncing the RTDS switch is taken care of in the code.

The car is ready-to-drive mode; a blue LED on the dashboard will illuminate, and the speakers will play a sound 3 seconds long. The RTDS board also takes input from the shutdown circuit line (after TSMS) and converts this signal to a 5V signal, which will be given to the ECU.

4.7.1 Wiring

The connection to the ECU from switches and sensors is made using 22 AWG wires. The power supply and input connections enter the steering dashboard and are connected to the RTDS PCB. The output from the RTDS unit is connected to the speakers using 22 AWG connectors.

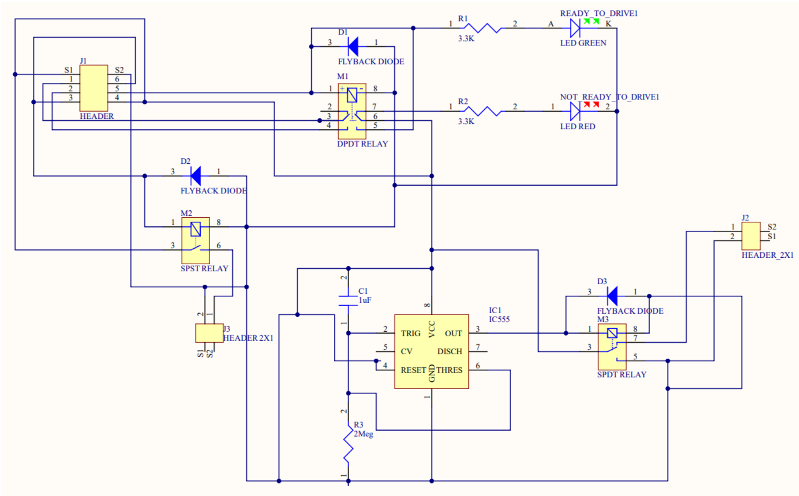


Figure 4.5: Ready-To-Drive-Sound (RTDS) (Click here for enlarged photo of the schematic)

4.7.2 RTDS Arduino Code

```
int rtds = 2;
int speaker = 8;
int breakpedalsensor = A1;
int apps = A0;
int speakerState = LOW;
int buttonState = LOW;
int lastbuttonState = LOW;
int scline = 13;
unsigned long lastDebouncetime = 0;
    unsigned long debounceDelay = 50;

void setup() {
  Serial.begin(9600);
  pinMode (rtds,INPUT);
  pinMode (breakpedalsensor,INPUT);
  pinMode (speaker,OUTPUT);
  pinMode (apps,INPUT);
  pinMode (scline,INPUT);
}

void loop() {
  delay(1000);
  int sclinestatus = digitalRead(scline);
  int breakpedalVal = analogRead(breakpedalsensor);
  int breakpedalPress = LOW;
  if(breakpedalVal>102.3 && breakpedalVal<920.7){
     breakpedalPress = HIGH;
  }

  // To take care of switch debouncing
  int reading = digitalRead(rtds);
  if (reading != lastbuttonState){
    lastDebouncetime = millis();
  }

```

```
if ((millis()-lastDebouncetime)>debounceDelay){
    if(reading!=buttonState){
      buttonState = reading;
      if(buttonState==HIGH && breakpedalPress==HIGH && sclinestatus==
   HIGH){
        speakerState = !speakerState;
      }
    }
  }
  lastbuttonState = reading;
  digitalWrite(speaker,!speakerState);
  int appsVal = analogRead(apps);
}
```

Listing 4.1: RTDS Arduino Code

4.7.3 Position in the car

RTDS unit, along with an ECU, is placed in an enclosure named steering electronics unit, which is placed near the steering such that it is easily accessible to the sensor inputs from pedal sensors and sensor sensors.

4.7.4 Wiring and Cables

All LV wires used are 18 AWG cables.

4.8 APPS - Acceleration Pedal Position Sensor

The accelerator pedal is directly connected to the APPS. APPS is used to convert pedal travel to throttle input provided to the motor controller. The APPS consists to two separate APPS sensors (primary and secondary). The two sensors have non intersecting transfer curves. The output of both the sensors are sent to the ECU. The ECU checks for plausibility i.e.. if the output of the two sensors differ by more than 10 percent for more than 100ms. If plausibility exists both LV and tractive system are shutdown, else primary APPS sensor output signal is sent to motor controller.

4.9 Pre-Charge/Discharge circuit

Pre-charging internal capacitors in motor controller input circuits to 96% of battery voltage is important to avoid inrush current and pitting of contactors. Discharge of internal capacitors in motor controller input circuits is important for avoiding electric shock after deactivation of the tractive system. In our car, those two systems are combined and will use the same resistor for both pre-charging and discharging of the motor controller's internal capacitors.

4.9.1 Wiring, Cables, Connectors

Connections to the precharge and time delay relay are made using a female quick disconnect. The Precharge resistor is connected using ring lugs. Control terminals of the relays are connected using 18 AWG wires and enter into the accumulator container using DTM series connectors. Load terminals of the pre-charge relay are connected using shielded wires rated for 1000VDC.

4.9.2 Voltage and Current Calculations

- Intermediate capacitance of one motor controller 10.797 mF

Pre-Charge Circuit

- Pre-charge Resistor = 100Ω
- $V_{max} = 96\ V$
- $I_{max} = \frac{V_{max}}{R} = \mathbf{960\ mA}$
- Energy Precharge Event = $\frac{1}{2}CV^2 = \mathbf{49.75\ J}$
- Precharge Time = 3RC = **3.2391 s**
- Power = $\frac{\text{Energy Precharge Event}}{\text{Precharge Time}} = \mathbf{15.359\ \ W}$
- Peak Power = $\frac{V^2}{R} = \mathbf{92.16\ W}$
    - Voltage V/s Time: $V = V_{max}(1 - e^{\frac{-t}{R \times C}}))$
    - Current V/s Time: $I = I_{max}(e^{\frac{-t}{R \times C}}))$

Discharge Circuit

- Discharge Resistor = 100 Ω
- $V_{max} = 96\ V$
- $I_{max} = \frac{V_{max}}{R} = \mathbf{960\ mA}$
- Time to Discharge = $R \times C \times \ln \frac{V}{60} = \mathbf{0.507\ s}$
    - Voltage V/s Time: $V = V_{max}(e^{\frac{-t}{R \times C}}))$
    - Current V/s Time: $I = I_{max}(e^{\frac{-t}{R \times C}}))$

4.9.3 Driver Interface

Driver interface allows the driver to keep track of functionality of various systems existing in the car. A new system, called Tire Pressure Monitoring System (TPMS), which monitors tire pressure of each tire, has been incorporated in our car this year. The interface is located in the cockpit, right in front of the driver. It consists of various buttons such as RTD button and shutdown button. The status of TS, IMD, AMS, BSPD, battery state of charge, battery temperature and tire pressure can also be seen on the same.

4.9.4 Wiring

The connection to the Driver Interface from switches and sensors is made using 18 AWG wires. The power supply and input connections enter the Interface and are connected to its display.

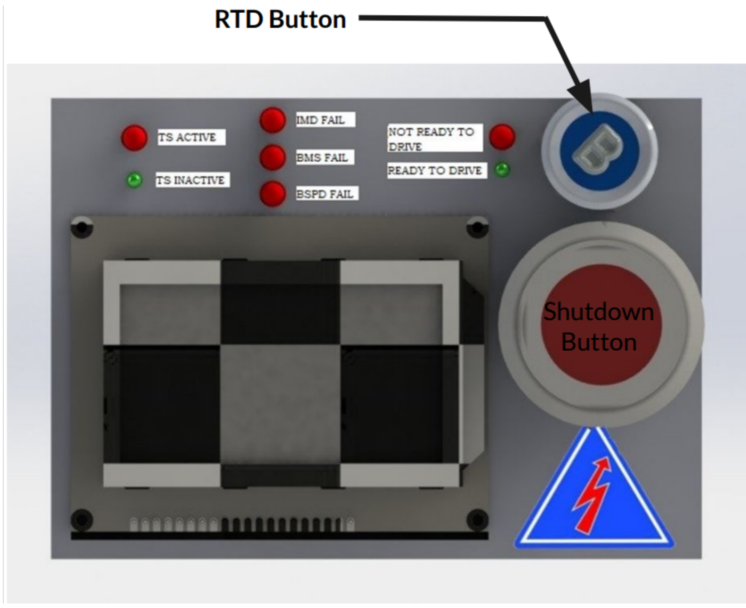


Figure 4.6: Driver Interface

Chapter 5

Thermal Management

5.1 Thermal Monitoring

5.1.1 Accumulator

The cell temperatures are monitored using NTC type thermistors. Each module having 12 cells is monitored using two thermistors which are lowered down into the respective busbars connected to the negative terminals of the cells, such that all the terminals which the particular thermistor is monitoring is within the limits of 10 mm. In total, 8 such thermistors are monitoring all the cells. Also, a thermistor is mounted over the slave board to keep in check the temperature of the board during the cell balancing. The cut-off temperature for the AMS to open the shutdown circuit is kept at 60°C, in accordance with cell's specification limits.

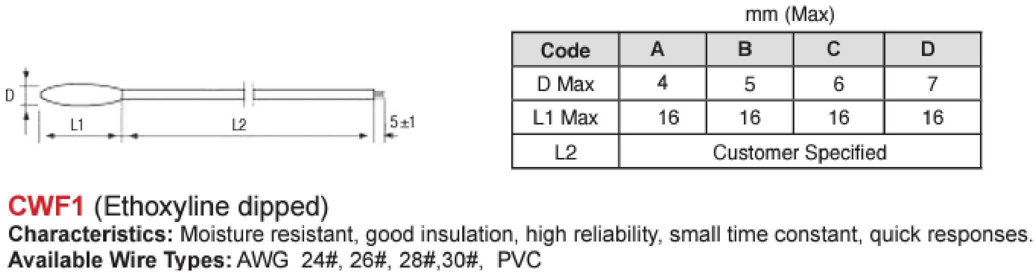


Figure 5.1: Dimension diagram of Thermistor

We will be using 10K Ohms NTC thermistor with the following characteristics:-

Thermistor Manufacturer	Cantherm
Resistance at 25C	10,000 Ohms
Operating Temperature	-55C to 125C
Tolerance	$\pm 1\%$
B-value	33380 $\pm 1\%$
Dissipation factor	2.2mW/K
Dimension of Head	Diameter (5mm) * Length(16mm)
Thermal Time Constant	$\leq 70s$

The centralized AMS is connected to the thermistors which measure the temperatures of the segment. In case of a cell reaches above the cut-off temperature, the thermistors signal the AMS to open the shut-down circuit, which ultimately switches off the vehicle.

5.1.2 Cooling Circuit and Mechanism

Structural features of TSAC

The input from the thermistors to AMS is used to control the functioning of fans fitted into the sides of the TSAC. The cooling fans are switched on during the entire running of the car. The air inputted to cooling vent by fans has a speed of 4m/s and air output from the vent has a speed of 8m/s.

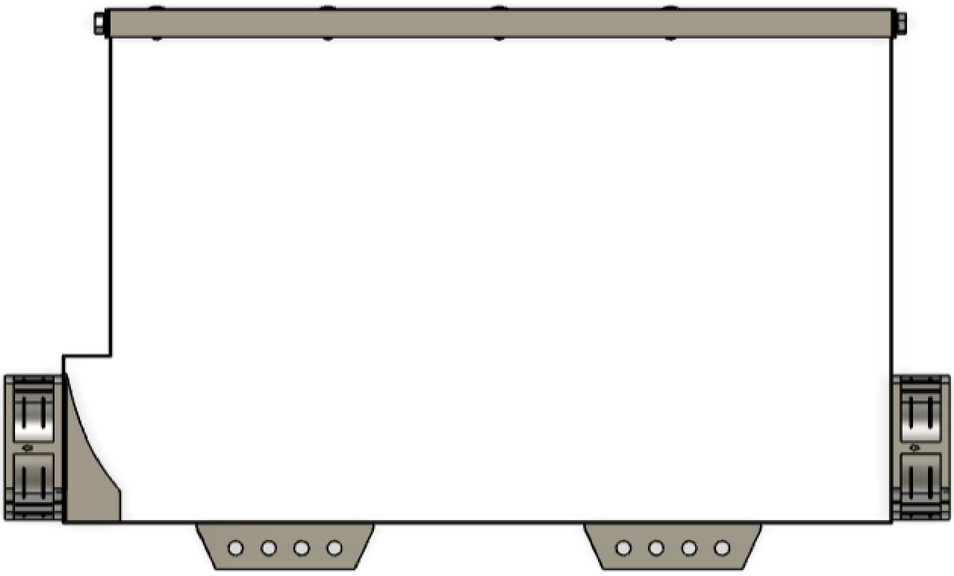


Figure 5.2: Side view of Fans and cooling Vent

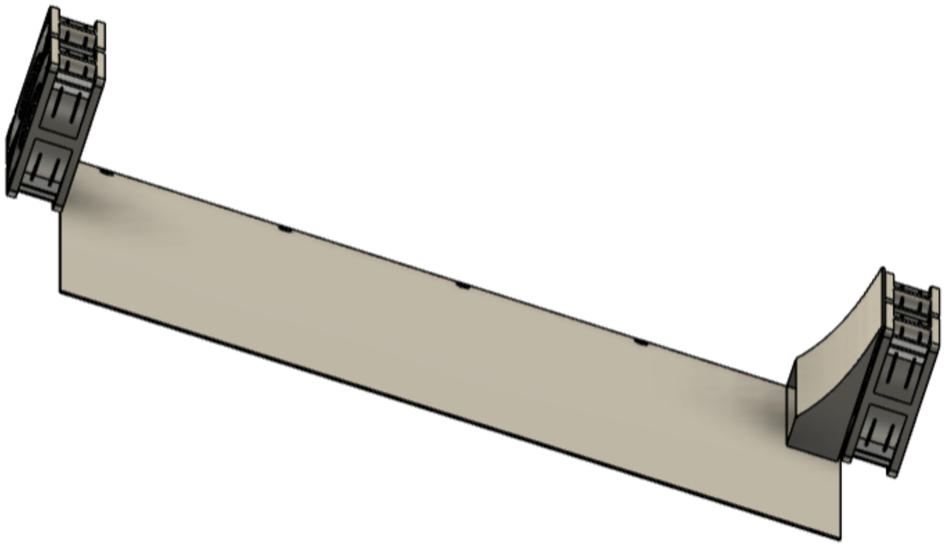


Figure 5.3: Diagonal view of Fans and cooling Vent

Structural features of the TS Segments

All the cells in a segment are separated by cooling fins which are used the transfer the heat created in the cells to wards the bottom of the TSAC.The parallel to flow of air and have their lower end cooled by it.

Heat Calculations

$$Area\,of\,fin\,exposed\,to\,air(A) : Length \cdot Breadth \tag{5.1}$$

- Length = 75.5 mm
- Breadth = 14 mm
  $\implies A = 1057\text{mm}^2$
- Internal Resistance of each cell(r) = $0.8m\Omega$
- $N_p = 2$

$$Internal\ Resistance\ of\ 1\ Parallel\ assembly = \frac{r}{N_p} \tag{5.2}$$

- Heating loss from 1 Cell (testing) = 2W

- h_{cW} = heat transfer coefficient of fins ($W/m^{2\circ}C$)

$$h_{cW} = 12.12 - 1.16v + 11.6v^{1/2} \tag{5.3}$$

- v = speed of air (8 m/s)

$$\implies h_{cW}\ for\ fins \approx \mathbf{35.64 (W/m^{2\circ}C)}$$

Considering these factors, the cooling system was designed accordingly. Following is the result of thermal analysis on fins.

The thermal analysis is performed in steady state with current of 100 Amp, the temper-

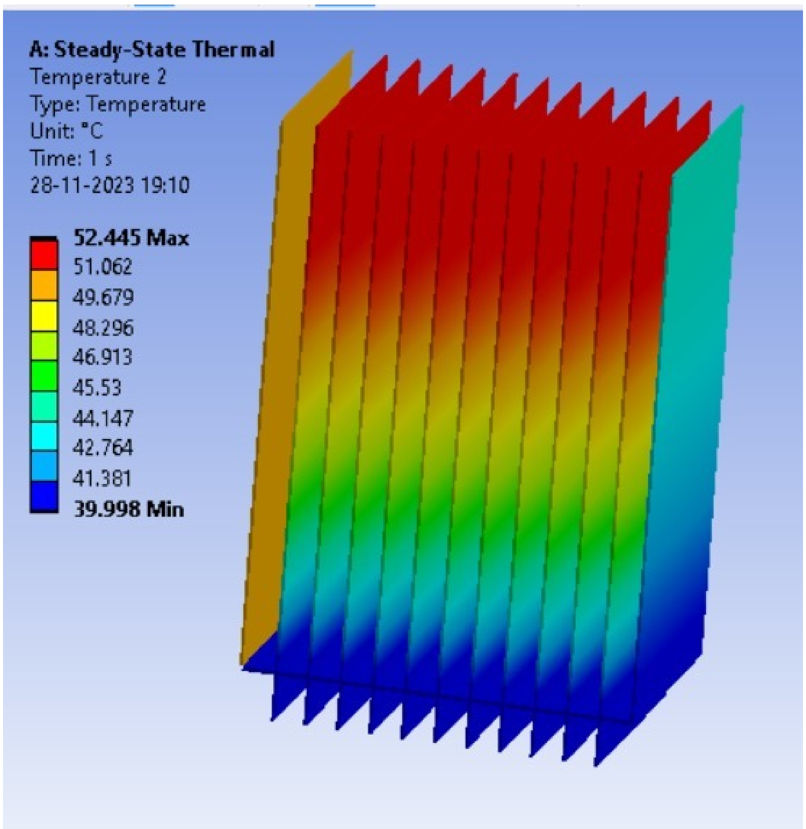


Figure 5.4: Thermal analysis of Fins of a Segment

ature is within the specified limitations of the cell specifications and the rule-book. The highest temperature of 52.445C occurs at the top of the segment and cooling occurs as we more towards the bottom which is closer to the path of coolant(air).

Chapter 6

Brakes and Wheel Assembly

6.1 Wheel Loads

This MATLAB code takes lateral and longitudinal acceleration values as input and provides the net wheel load transfer on the four vehicle wheels as output.

```
function z = WheelLoadTransfer (Ax , Ay)

l = 1.561 ; % Wheel base
a = 0.887 ; % Dist between front axle and COM in longitudinal direction
tF = 1.21 ; % Front Track Width
tR = 1.177 ; % Rear Track Width
h = 0.223 ; % Height of total COM of car
zrf = 0.05 ; % Height of Front Roll Center
zrr = 0.06 ; % Height of Rear Roll Center
zwf = 0.254 ; % Height of Front Unsprung Mass
zwr = 0.254 ; % Height of Rear Unsprung Mass
W = 282.6 ; % Weight of the whole car
Wuf = 25.366 ; % Weight of front unsprung mass
Wur = 26.27662 ; % Weight of rear unsprung mass
Ws = (282.6 - Wuf - Wur) ; % Weight of sprung mass
Kf = 9779.4 ; % N.m/rad _ front wheel stiffness
Kr = 21800 ; % N.m/rad _ rear wheel stiffness

Nr = (W * a) / l ;
Nf = W - Nr ;
b = l - a ;
bs = ((W * b) - (Wuf * l)) / Ws ;
as = l - bs ;
h2 = - (as*(zrr-zrf)-(h*l)+(zrf*l))/sqrt((l*l)+((zrr-zrf)*(zrr-zrf))) ;
Kf1 = Kf - ((l - as) * Ws * h2 / l ) ;
Kr1 = Kr - (as * Ws * h2 / l ) ;

delWf = ((Ay * Ws) * (((h2 * Kf1) / (Kf + Kr - (Ws * h2))) +
(l - as) * (zrf / l)) / tF ) + ((Ay * Wuf) * (zwf / tF)) ;

delWr = ((Ay * Ws) * (((h2 * Kr1) / (Kf + Kr - (Ws * h2))) +
(as * (zrr / l))) / tR ) + ((Ay * Wur) * (zwr / tR)) ;

delWx = (h * W * Ax) / l ;
```

```
Nfl = (Nf / 2) - delWf - (delWx / 2) ;
Nfr = (Nf / 2) + delWf - (delWx / 2) ;
Nrr = (Nr / 2) + delWr + (delWx / 2) ;
Nrl = (Nr / 2) - delWr + (delWx / 2) ;

z = [Nfl , Nfr , Nrr , Nrl] ;
disp (Nfl + Nfr + Nrl + Nrr) ;

end
```

6.2 Brakes

6.2.1 Abstract

The system is one based on hydraulic braking.The objective of the system is to convert the kinetic energy of the vehicle into thermal energy, allowing the vehicle to decelerate optimally and safely. The design includes three major categories: calculation and evaluation of the hydraulic system in order to select calipers and master cylinders, the design of the pedal box, and the design of the rotors.

6.2.2 Calculations

Generic parameters of the vehicle that were used in the system design were a total vehicle weight, including the driver, of 300kg and a maximum velocity of 130 km/h.

The three main categories/ components designed and analyzed for this system are the following:

1. **Calculation and evaluation of the hydraulic system in order to select calipers and master cylinders**: For the first component, the overall functionality and free variables of the braking system will be analyzed for optimum caliper, master cylinder and rotor size selection.

2. **The design of the pedal box**: This design includes the gas pedal, brake pedal and master cylinder orientation as well as the throttle senor and emergency stop placement and orientation.

3. **The design of the rotors**: The main focus on the design of the rotors will be material selection as well as the geometry of the rotors.

   The objective of these major component designs will be to minimize the weight for the lightest design possible while at the same time designing to all of the mechanical and thermal conditions that the system would be subjected to.

6.2.3 Design and Calculations

The main objective of the braking system is to convert the kinetic energy of the vehicle into thermal energy, thus allowing the vehicle to decelerate. The braking system was designed as a hydraulic system with two master cylinders, one for the braking of the front two tires and one for braking of the rear two tires. Attached to each master cylinder are two floating calipers, one located at each of the tires for a total of four calipers for the system, as well as four rotors or brake disks.

The flow of the braking system is as follows: the driver exerts a force on the brake pedal, the brake pedal channels that force to the master cylinders, thus displacing the braking fluid in the master cylinders. The displaced fluid then exerts a pressure on each of the calipers allowing the caliper pistons to exert a clamping force on the rotors. Therefore, the input of the system is the driver's applied foot force and the output is the clamping force of the calipers exerted on the rotors. Based on the competition rules and conditions the designed braking system must be able to **lock all four tires** of the vehicle completely during an emergency stop braking scenario. What this translates to physically is that the moment generated from the caliper force placed on the rotor must be equal to or greater than the moment the tire exerts on the surface of the road.

Formulae

During braking, under the condition the vehicle is moving in the direction of its front tires, the greatest deceleration rate will translate as a weight transfer from the rear tires to the front tires. Therefore, it is necessary to calculate this weight transfer based on the average coefficient of friction between the tires and the road , the weight of the vehicle , the height of the center of mass above the road , and the wheelbase . This weight transfer will be denoted as and calculated-

$$F_T = \frac{\mu_B W H}{L} \tag{6.1}$$

From this weight transfer calculation the normal force exerted on the front two tires (F_{NP}) and the rear two tires (F_{NR}) are calculated:

$$F_{NF} = \frac{W L_R}{L} + F_T \tag{6.2}$$

$$F_{NR} = \frac{W L_F}{L} - F_T \tag{6.3}$$

where L_F and L_R are the lengths of the front and rear axles to the center of mass of the vehicle.

Neglecting the weight on the tires, wheels and rotors and taking a summation of moments about the center of rotation of one of the tires, it is observed for the front and rear systems:

$$\sum M_O = 0 \tag{6.4}$$

$$\text{Front:} F_{FCP} \cdot R_{FCP} - \mu_B F_{NF} R_{FT} = 0 \tag{6.5}$$

$$\text{Rear:} F_{RCP} \cdot R_{RCP} - \mu_B F_{NR} R_{RT} = 0 \tag{6.6}$$

where, F_{FCP} and F_{RCP} are the sum of the friction forces generated from the front two calipers and rear two calipers, R_{FCP} and R_{RCP} are the mean radii of these friction forces, and R_{FT} and R_{RT} are the radii of the front and rear tires.

The total friction force of the calipers for the front and rear can be written in terms of the normal force or "clamping force" that the calipers exert on the rotor multiplied by the brake pad coefficient of friction.

$$F_{FCP} = \mu_{CF} \cdot F_{NFCP} \tag{6.7}$$
$$F_{RCP} = \mu_{CF} \cdot F_{NRCP} \tag{6.8}$$

The normal forces of each set of calipers, both front and rear, can be written in terms of the pressure in each brake line and the area of each caliper piston:

$$F_{NFCP} = (\#\text{ of front caliper pistons}) \cdot P_F \cdot A_{FCP} \tag{6.9}$$
$$F_{NRCP} = (\#\text{ of rear caliper pistons}) \cdot P_R \cdot A_{RCP} \tag{6.10}$$

Thus, the pressure in both the front and rear braking lines can be written:

$$P_F = \frac{F_{NFCP}}{\#\text{ of front caliper pistons}) \cdot A_{FCP}} \tag{6.11}$$
$$P_R = \frac{F_{NRCP}}{\#\text{ of rear caliper pistons}) \cdot A_{RCP}} \tag{6.12}$$

The forces on both master cylinders can then be expressed from the pressure in the front and rear lines multiplied by the bore area of each master cylinder:

$$F_{FMC} = P_F \cdot A_{FMC} \tag{6.13}$$
$$R_{RMC} = P_R \cdot A_{RMC} \tag{6.14}$$

Taking a moment about the braking pedal, the foot force applied to the pedal can be written in terms of the forces from the master cylinders:

$$a \cdot F_{foot} = b \cdot (F_{FMC} + F_{RMC}) \tag{6.15}$$

where a is the distance from the foot force to the pivot point and b is the distance from where the master cylinder mounts to the brake pedal to the pivot point.

6.2.4 Results from Calculations and Selection of Components

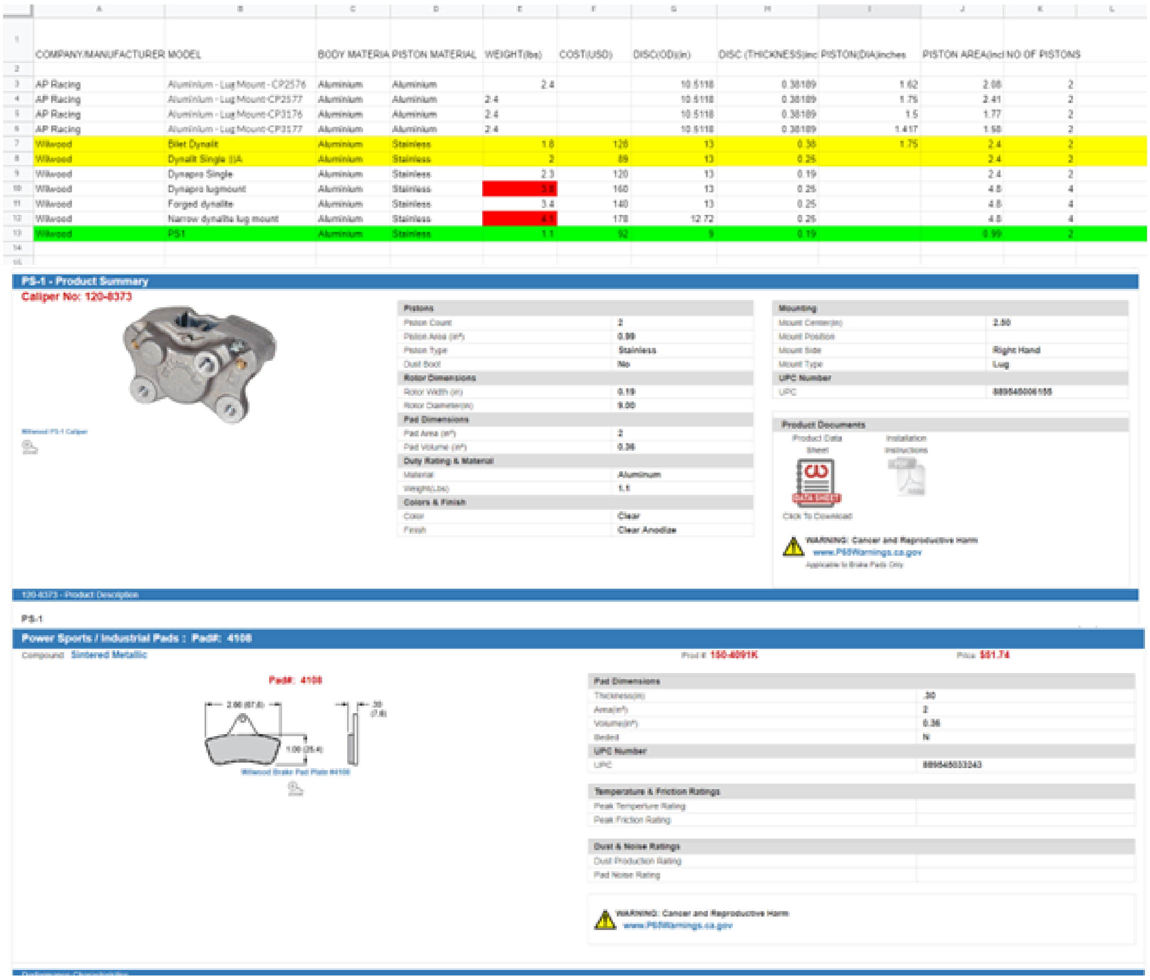
From previous years:

- Selected Wheels: 7"/13", 22mm neg. Offset, Forged Al
- Selected Tires: 7.2"/20"-13"

Tyre size	Rim choice	Rim used	Measured at		Diameter		Section		Tread		Revolutions		Tread
			psi	Bar	ins	mm	ins	mm	ins	mm	miles	km	patterns
6.2/20.0-13	5.5 - 6.5	6	20.0	1.4	19.90	505	7.20	183	6.10	155	1013	630	View
7.2/20.0-13	6.0 - 8.0	8	20.0	1.4	20.20	513	9.02	229	7.09	180	998	620	View
8.2/20.0-13	7.0 - 9.0	9	20.0	1.4	19.95	507	9.76	248	8.50	216	1011	628	View
9.0/20.0-13	8.0 - 10.0	10	20.0	1.4	20.00	508	10.67	271	9.21	234	1008	627	View
10.0/20.0-13	9.0 - 11.0	10	20.0	1.4	19.80	503	11.10	282	10.20	259	1019	633	View
10.0/21.5-14	8.5 - 11.0	11	20.0	1.4	22.10	561	11.97	304	9.84	250	913	567	View

Figure 6.1: Tire Data

(As we are using same tires and wheels from the previous years) the radii of the tires for both the front and rear were given as 10.10 inches . Given the hollow rim design, where the rotor sits inside of the rim with the caliper mounted, the rotor's diameter was restricted. The inner diameter of the rim was approximated to 13 inches and the caliper clearance was approximated to be 1.5 inches , (given a final selection has not been made yet). After modeling the rotor, caliper and hub in SolidWorks the largest rotor diameter that could fit the assembly was 8.5 inches (approximated) . This was diameter used, as the largest rotor diameter would produce the greatest torque from the caliper, for a lower line pressure, during braking. Thus the friction forces for the front and rear calipers could be solved for:



COMPANY/MANUFACTURER	MODEL	BODY MATERIA	PISTON MATERIAL	WEIGHT(lbs)	COST(USD)	DISC(OD)(in)	DISC (THICKNESS)inc	PISTON(DIA)inches	PISTON AREA(inc	NO OF PISTONS
AP Racing	Aluminium - Lug Mount - CP2576	Aluminium	Aluminium	2.4		10.5118	0.38189	1.62	2.06	2
AP Racing	Aluminium - Lug Mount-CP2577	Aluminium	Aluminium	2.4		10.5118	0.38189	1.75	2.41	2
AP Racing	Aluminium - Lug Mount-CP3176	Aluminium	Aluminium	2.4		10.5118	0.38189	1.5	1.77	2
AP Racing	Aluminium - Lug Mount-CP3177	Aluminium	Aluminium	2.4		10.5118	0.38189	1.417	1.58	2
Wilwood	Billet Dynalit	Aluminium	Stainless	1.8	128	13	0.38	1.75	2.4	2
Wilwood	Dynalit Single IIIA	Aluminium	Stainless	2	89	13	0.25		2.4	2
Wilwood	Dynapro Single	Aluminium	Stainless	2.3	120	13	0.19		2.4	2
Wilwood	Dynapro lugmount	Aluminium	Stainless	3.8	160	13	0.25		4.8	4
Wilwood	Forged dynalite	Aluminium	Stainless	3.4	140	13	0.25		4.8	4
Wilwood	Narrow dynalite lug mount	Aluminium	Stainless	4.1	178	12.72	0.25		4.8	4
Wilwood	PS1	Aluminium	Stainless	1.1	92	9	0.19		0.99	2

Figure 6.2: Product Selection

After selecting a Wilwood caliper, a Wilwood brake pad was chosen that would fit the style and size of the caliper. The Polymatrix Compund A Wilwood brake pad was chosen for its overall high coefficient of friction. Based off of the company's performance chart (see below) between temperatures of $100°F$ and $700°F$ the average coefficient of friction was approximated as 0.6215.

Tilton 75 series 7/10" bore size master cylinder selected.

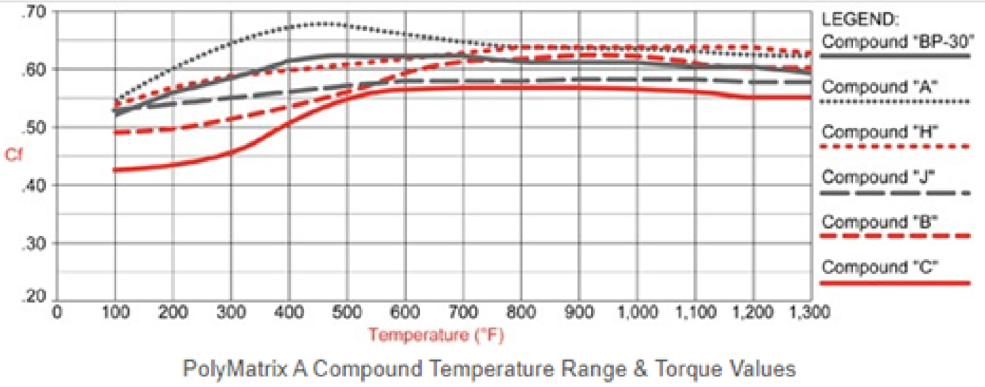


Figure 6.3: PolyMatrix A Compound Temperature Range and Torque Values

6.2.5 Analysis of Brake pedal CAD model

For designing the CAD model of brake pedal, we used SolidWorks. We started off with a simple design of pedal which had no cuts. After analysing the model on Ansys, we introduced some triangular cuts in the design as our objective is to make it as light as possible while maintaining a good Factor of Safety(FOS). The following images depict the static structural analysis of the final design on Ansys.

Static Structural Analysis:

- Mesh Element Size : 2 mm.
- Cylindrical Support : Pivot (Hole with smaller radius)
- Compressive Support : Hole which connects to the master cylinder
- Normal Force on pedal : 2000.00 N

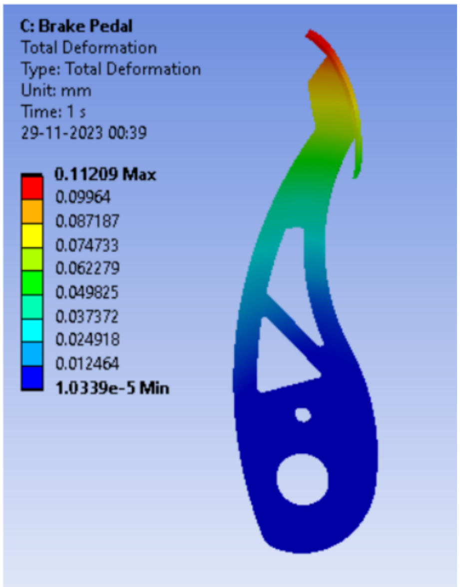


Figure 6.4: Total deformation

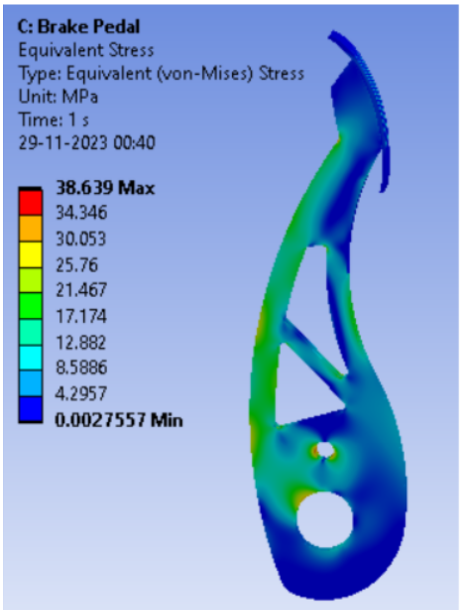


Figure 6.5: Equivalent stress

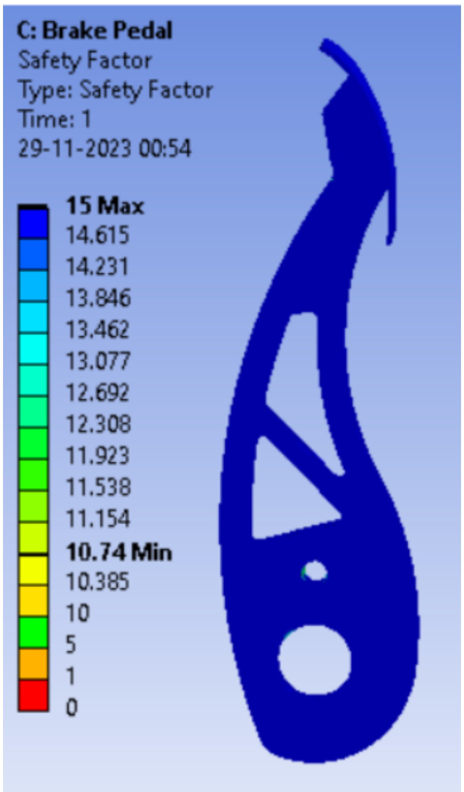


Figure 6.6: Factor of Safety

6.2.6 Brake bias code

Brake bias refers to how braking force is distributed between a vehicle's front and rear wheels, determined by the load transfer on the front and rear tires. This Matlab code takes the acceleration in x and y direction as input and calculates the brake bias percentage in the respective tyres.

```
function z = brakebias2 (Ax , Ay)

l = 1.561 ; % Wheel base
a = 0.887 ; % Distance between front axle and COM in longitudinal
    direction
tF = 1.21 ; % Front Track Width
tR = 1.177 ; % Rear Track Width
h = 0.223 ; % Height of total COM of car
zrf = 0.05 ; % Height of Front Roll Center
zrr = 0.06 ; % Height of Rear Roll Center
zwf = 0.254 ; % Height of Front Unsprung Mass
zwr = 0.254 ; % Height of Rear Unsprung Mass
W = 282.6 ; % Weight of the whole car
Wuf = 25.366 ; % Weight of front unsprung mass
Wur = 26.27662 ; % Weight of rear unsprung mass
Ws = (282.6 - Wuf - Wur) ; % Weight of sprung mass
Kf = 9779.4 ; % N.m/rad _ front wheel stiffness
Kr = 21800 ; % N.m/rad _ rear wheel stiffness

Nr = (W * a) / l ;
Nf = W - Nr ;
b = l - a ;
bs = ((W * b) - (Wuf * l)) / Ws ;
as = l - bs ;
h2 = - (as * (zrr - zrf) - (h * l) + (zrf * l)) / sqrt ( (l * l) + ((
    zrr - zrf) * (zrr - zrf)) ) ;
Kf1 = Kf - ((l - as) * Ws * h2 / l ) ;
Kr1 = Kr - (as * Ws * h2 / l ) ;

delWf = ((Ay * Ws) * (((h2 * Kf1) / (Kf + Kr - (Ws * h2))) + (l - as) *
     (zrf / l)) / tF ) + ((Ay * Wuf) * (zwf / tF)) ;
delWr = ((Ay * Ws) * (((h2 * Kr1) / (Kf + Kr - (Ws * h2))) + (as * (zrr
     / l))) / tR ) + ((Ay * Wur) * (zwr / tR)) ;
delWx = (h * W * Ax) / l ;

Nfl = (Nf / 2) - delWf - (delWx / 2) ;
Nfr = (Nf / 2) + delWf - (delWx / 2) ;
Nrr = (Nr / 2) + delWr + (delWx / 2) ;
Nrl = (Nr / 2) - delWr + (delWx / 2) ;

biasfront = (Nfl+Nfr)/2;
biasrear = (Nrl+Nrr)/2;
biasfrontpercent = biasfront*100/(biasfront+biasrear);
biasrearpercent = biasrear*100/(biasfront+biasrear);

z = [biasfrontpercent biasrearpercent];

end
```

Listing 6.1: Brake Bias

6.3 Brake Rotor

6.3.1 Material Selection

AISI 4340 stands out as the optimal choice, aligning with both the strength and thermal considerations of our project.

Properties of Steel 4340 :

1. Materials Compressive strength (MPa) = 745
2. Friction coefficient (μ) = 0.5
3. Specific heat capacity (J/g/$\circ$C) = 0.475
4. Specific gravity (g/cc) = 7.85

6.3.2 Selecting the type of Rotor

Rotor types can be categorized into two:

(i) Fixed Rotor
(ii) Floating Rotor

Our team has opted for floating rotors in our project, considering the following advantages:

1. When brakes are applied, the kinetic energy of the vehicle is primarily absorbed by the rotors, causing them to heat up and expand. Floating rotors outperform fixed rotors in this scenario, allowing for easy expansion without distorting their shape.

2. The floating rotor naturally aligns itself with the brake pad surface during application, ensuring maximum contact and enhancing braking efficiency while minimizing wear.

3. From a manufacturing perspective, floating rotors offer the added benefit of accommodating nominal bends in the material sheet.

6.3.3 Rotor Design

The rotor in our design is mounted onto the hub. Mounting points and rotor dimensions are considered accordingly. Drills and slots have been strategically added to the rotor. The addition of drills and slots provides the following advantages:

1. These cuts play a crucial role in preventing brake fading during intense braking.

2. During heavy braking, the brake pad temperature can rise, leading to the formation of a thin gas layer between the pad and the rotor. This gas layer reduces the pad's friction, resulting in less effective braking. The holes or slots in the rotor serve to create a venting effect, helping clear out the gas layer and maintaining optimal brake performance.

3. These design features contribute to lighting the rotor and enhance its appearance, adding aesthetic value.

4. Increased surface area exposure makes it easier for the rotor to release heat efficiently.

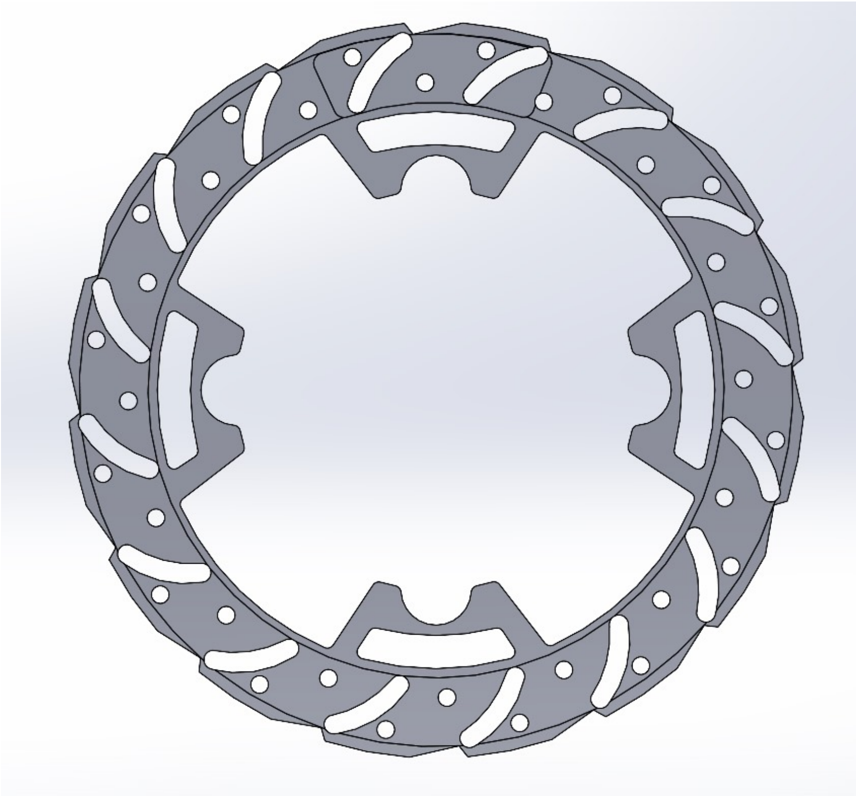


Figure 6.7: Brake Rotor CAD

Outer Radius : 105 mm.
Inner Radius : 80 mm.

6.3.4 Braking Torque and Pressure Calculations

This MATLAB code takes the maximum deceleration of the vehicle (a) as input and provides the required braking torque, net applied braking torque, and pressure on the brake rotor as output.

```
function z = BrakeRotor(a)

M = 282.6 ; % Total weight of the car (in kg)
g = 9.81 ;
fWR = 0.254 ; % Front wheel radius
rWR = 0.254 ; % Rear wheel radius
Wb = 1.561 ; % Wheel base (in m)
lf = 0.8872 ; % Front axle to CG (in m)
lr = 0.6738 ; % Rear axle to CG (in m)
h = 0.2233 ; % Height of CG (in m)
utg = 1.4 ; % Coefficient of friction between tyres and ground
ubr = 0.4 ; % Coefficient of friction between brake pad and rotor
A_piston = 0.785 ; % Inches area caliper piston , diameter = 1 inch
A_MC = 0.3066 ; % Inches area master cylinder, diameter = 5/8 inch
```

```

% Static Weight Transfer
Wf = M * (lr / Wb); % Front
Wr = M * (lf / Wb); % Rear

% Dynamic Weight Transfer
dW = M * a * h / Wb ; % Longitudinal Weight Transfer
dWF = Wf + dW ; % Dynamic weight Front
dWR = Wr - dW ; % Dynamic weight Rear

% Required Braking Torque
Torque_wf = utg * dWF/2 * g * fWR ;  % Torque on Front Wheel
Torque_wr = utg * dWR/2 * g * rWR ; % Torque on Rear Wheel

Fp = 500 ; % Pedal force (in N)
P_ratio = 5 ; % Pedal ratio
Fb = Fp * P_ratio ; % Force at Balance bar

Bf = 0.65;
Br = 0.35;

% Normal Clamping Force
F_clamp_front = 2 * (Fb * Bf * A_piston / A_MC) ;
% Clamp Force in Front (in N)
F_clamp_rear = 2 * (Fb * Br * A_piston / A_MC) ;
% Clamp Force in Rear (in N)

% Friction force applied by Brake Pad
Friction_Front = F_clamp_front * ubr ;
Friction_Rear = F_clamp_rear * ubr ;

% Dimensions of the Brake Rotor
R_Outer = 105 ; % Center to Outer Brake pad (in mm.)
R_Inner = 80 ; % Center to Inner Brake pad (in mm.)

Reff = (2/3) * (R_Outer^3 - R_Inner^3) / (R_Outer^2 - R_Inner^2) ;
% Effective radius of Rotor

% Net torque on the rotors from both pads
Net_Torque_Front = Friction_Front * Reff * 0.001 ;
Net_Torque_Rear = Friction_Rear * Reff * 0.001 ;

% Pressure
Pressure = (500 * 5 * 0.65) / A_MC ; % in N/in ^ 2

z = [Torque_wf,Torque_wr,Net_Torque_Front,Net_Torque_Rear,Pressure] ;

end
```

6.3.5 Thermal Calculations on Brake Rotor

This MATLAB code calculates the Heat flux, Heat power, and maximum Temperature of the rotor under a given maximum deceleration (1.53g). Wilwood Brake Pads and its corresponding dimensions have been used.

```
v_max = 33.33 ; % 120 kmph in m/s
```

```
M = 315 ; % in kgs
KEtotal_car = 0.5 * M * v_max ^ 2 ; % total max KE of the car
T_amb = 298 ; % in kelvin

% Wilwood Brake Pads
od = 0.106 ; % Outer diameter
id = 0.0835 ; % Inner diameter
A_max = pi * (od) ^ 2 ;
A_min = pi * (id) ^ 2 ;
A_contact = A_max - A_min ;

% Rotor AISI 4340
density = 7850 ; % in kg/m3
k = 44.5 ; % Thermal conductivity in W/mK
c = 475 ; % Specific heat capacity in J/kgK

% Stopping time
stop_t = v_max /(1.53 * 9.81) ;

% Stopping Distance (Method - 1)
stop_dist1 = KEtotal_car / (2 * (2239 + 595)) ;
% KE lost by car will be used by friction (i.e. friction force on all 4
    rotors), here deceleration = 2 * (2239 + 595) / 315 = 17.99

% Stopping Distance (Method - 2) - seems more accurate
stop_dist2 = v_max * stop_t - 0.5 * (1.53 * 9.81) * (stop_t^2) ;
% s = ut + 1/2at2 , here deceleration = 1.53g , hence more accurate.

% Thermal calculations for one side of brake pad and rotor
E_rotors = KEtotal_car * 0.9 ;
% 90 percent dissipated through rotors while braking

E_frotors = E_rotors * 0.82 ;
% Proportional to dynamic weight, P.S. and not brake bias

E_rrotors = E_rotors * 0.18 ;
% Proportional to dynamic weight, P.S. and not brake bias

E_fr = E_frotors / 2 ; % per front rotor
E_rr = E_rrotors / 2 ; % per rear rotor

HeatPower_fr = E_fr / stop_t ;
HeatPower_rr = E_rr / stop_t ;

Heatflux_fr = HeatPower_fr / A_contact ;
Heatflux_rr = HeatPower_rr / A_contact ;


T_max_fr = T_amb + (0.527 * Heatflux_fr * (stop_t) ^ 0.5) / (density *
    c * k) ^ 0.5 ;
T_max_rr = T_amb + (0.527 * Heatflux_rr * (stop_t) ^ 0.5) / (density *
    c * k) ^ 0.5 ;
```

6.3.6 Analysis of Rotor CAD Models

For our project, we used SolidWorks to design the CAD model of our rotor. We began with a basic rotor, devoid of any cuts or slots. Subsequently, we applied fixed geometry to the points connecting to the hub and the slider fixture where the pad contacts the rotor surface. Next, we applied calculated forces to simulate the normal and frictional forces on both sides where the pad contacts the rotor. After applying the material and employing curvature-based meshing, we conducted the simulation.

Static Structural Analysis:

- Mesh Element Size : 2 mm.
- Fixed Support : Points of the Brake Rotor connecting to the Hub.
- Rotational Velocity
- Normal Force on one side of the Rotor : 2629.3 N
- Frictional Force on the Rotor by the Brake Pads : 1314.7 N

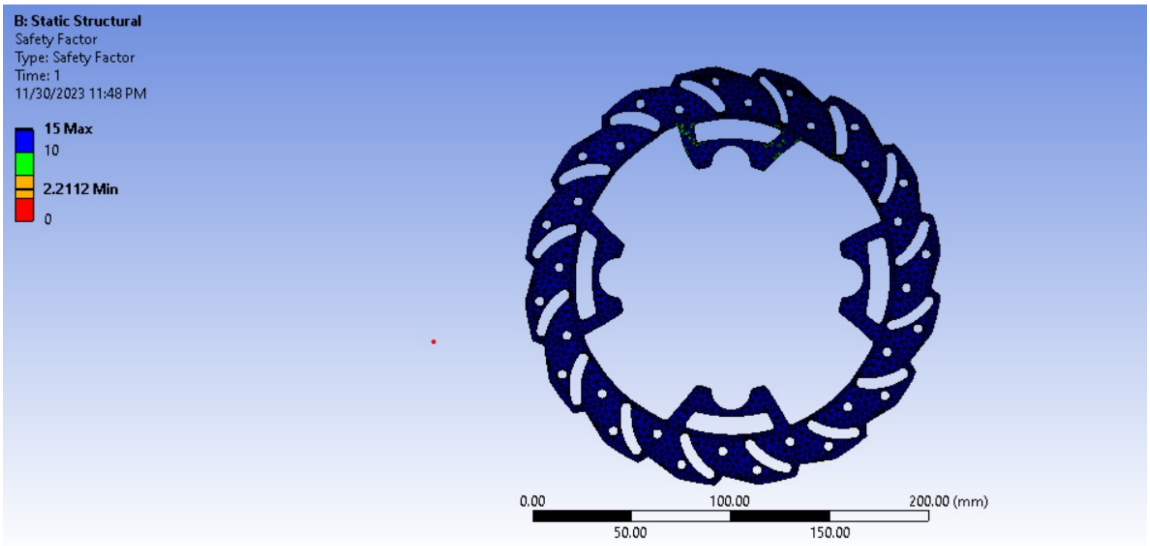


Figure 6.8: Factor of Safety

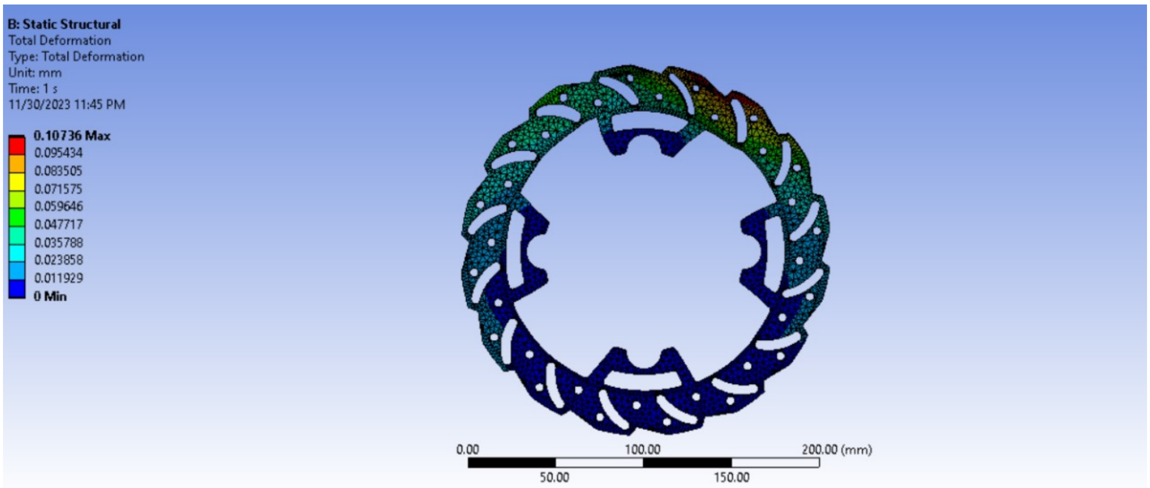


Figure 6.9: Total Deformation

Steady - State Thermal Analysis :

- Mesh Element Size : 2 mm.
- Initial Temperature : 25 ∘C
- Convection film co-efficient : 2.3 X $10^{-5} W/mm^2$.∘C
- Convection ambient temperature : 20 ∘C
- Radiation ambient temperature : 20 ∘C
- Heat Flux magnitude : 2.5 X $10^{-2} W/mm^2$

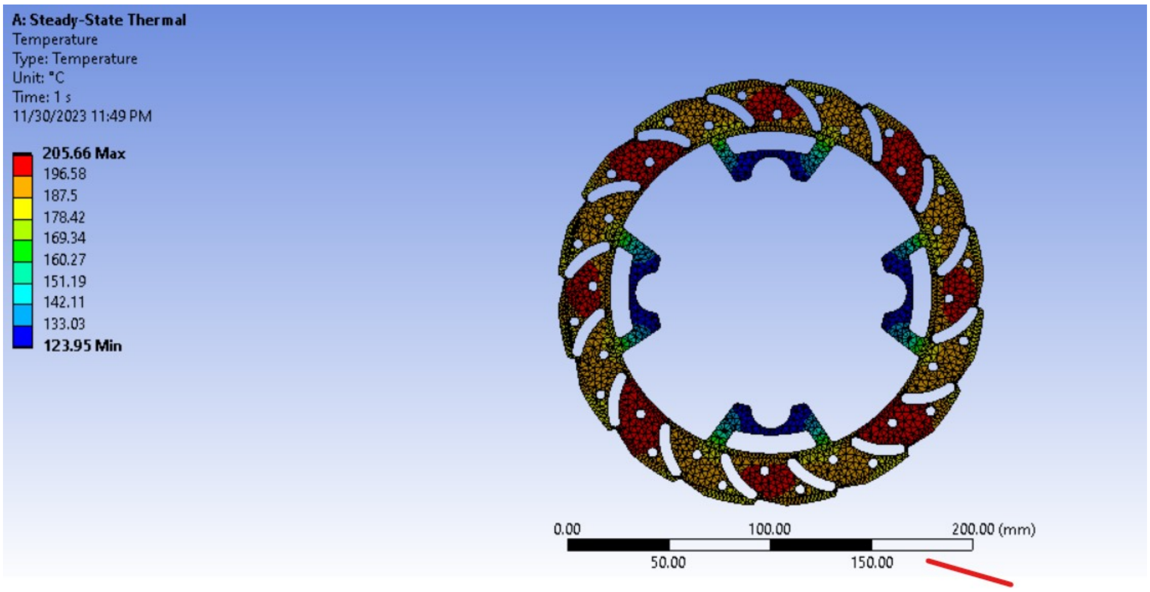


Figure 6.10: Temperature Distribution

The results obtained after conducting both analysis were satisfactory, and thus, design of the rotor was finalized.

6.4 Force Calculation for Knuckle

6.4.1 Approach for Calculation

Forces applied on the knuckle will derive from the 3 ball joints, which are the lower and upper ball joints and the tie rod joint. Forces acting on the suspension arms were then applied to respective ball joints to calculate forces acting on knuckle. We balance force and torque in x, y, and z directions and equate them to the forces due to acceleration/braking and cornering and solve the equations using a Matlab code.

Torque applied on brake mounts were calculated as follows:-

$$\text{Maximum Braking Deceleration} = 2 \cdot g \tag{6.16}$$

$$\text{Wheel Load on Front Wheel while Braking (Wlb)} = 2122.2\ N \tag{6.17}$$

$$\text{Frictional Force}(F_{rc}) = \text{Wheel Load * Coefficient of Friction} \tag{6.18}$$

$$= W_{lb} \cdot \mu/\sqrt{2} \tag{6.19}$$

$$= 2100.87\ N \tag{6.20}$$

$$\text{Braking Torque}(B_t) = \text{Frictional Force} \times \text{Radius of Tire} \tag{6.21}$$

$$= F_{rc} \cdot R \tag{6.22}$$

$$= 172490\ N \cdot mm \tag{6.23}$$

$$\text{Force}_{\text{ Exerted on Caliper Mounting}(F_c)} = \frac{\text{Braking Torque}}{\text{Distance of caliper mount}_{\text{ from center of spindle}}} \tag{6.24}$$

$$F_c = \frac{B_t}{r} \tag{6.25}$$

$$= 1889.3\ N \tag{6.26}$$

6.4.2 Bump Force

$$\text{Vertical Force Exerted on Push Rod}(F_p) = 1955.6\ N \tag{6.27}$$

$$\text{Perpendicular Distance from Lower A arm Hinge point}(L_1) = 301.47\ mm \tag{6.28}$$

$$\text{Torque of Push Rod about lower A arm hinge point}(T_p) = F_p * L_1 \tag{6.29}$$

$$= 589550\ N - mm \tag{6.30}$$

$$\text{Bump Force on Upright}(F_b) = \frac{T_p}{\perp \text{ dist. b/w lower ball joint and lower a-arm hinge point}} \tag{6.31}$$

$$= 1786.5\ N \tag{6.32}$$

6.5 Knuckle Forces Code

```
wt=315; ax=-14.46;   ay=13.61;  wb=1.561;   h=0.329;    ax_a=6.36;
g=9.81; a=0.79;   b=wb-a;   tr=1.177;   tf=1.210;

%uaf=[85.21 254.2 329.7]; %upper A-arm front
%uar=[-221.49 254.2 313.24]; %upper a arm rear
%direction_vector_u=uaf-uar;
%laf=[157.17 254.2 156.27]; %lower a arm front
%lar=[-249.58 254.2 156.27]; %lower a arm rear
%direction_vector_l=laf-lar;

%position vector of 9 points
LAFI=[157.17 254.2 156.27]; %lower a arm front
LARI=[-249.58 254.2 156.27]; %lower a arm rear
```

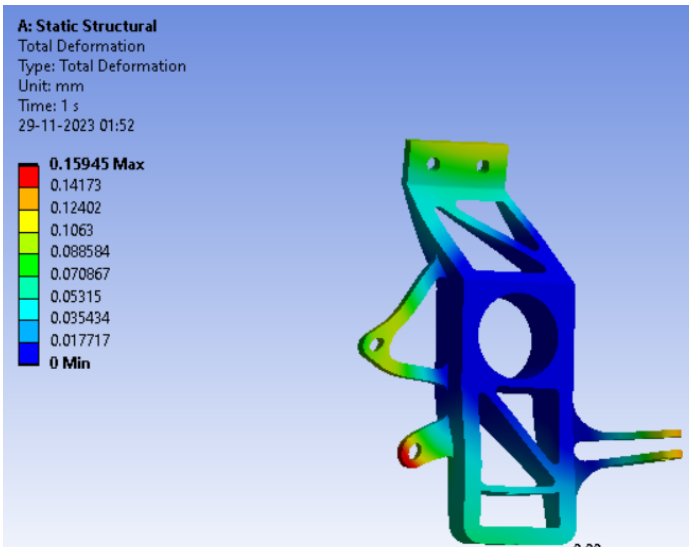
```
UAFI=[85.21 254.2 329.7];%upper A-arm front
UARI=[-221.49 254.2 313.24]; %upper a arm rear
UBJ=[-14.05 565.6 353.86];%upper ball joint
LBJ=[14.05 572.58 153.99];%lower ball joint
PRI=[-14.05 334.39 126.7];%pull rod inboard point
PRO=[-14.05 468.40 319.54];%pull rod outboard point
TCP=[0 605  0];%tire contact point
TRI=[91.43 243.87 183.97]; %tie rod chassis point
TRO=[99.1400  539.7100  187.3900]; %tie rod knuckle point

Spindle = TCP + [0 -40.87 253];
BD = PRI + [0 -37.95 98.99];
Dbody = BD + [0 -176.31 -17.66]
BellAxis = [-14.05 284.2 156.27];

APri = norm(PRI - BellAxis);
adam = norm(BD - BellAxis);
damper = norm(Dbody-BD);


%vector in the direction of forces
vec1=LAFI-LBJ; %lower a arm fore
vec2=LARI-LBJ; %lower a arm aft
vec3=UAFI-UBJ %upper a arm fore
vec4=UARI-UBJ %upper a arm aft
vec5=PRI-PRO; %pull rod
vec6 =TRI -TRO; %tie rod
vec7 = UBJ-TCP;
vec8 = LBJ-TCP;
vec9 = PRO-TCP;
vec10 = TRO-TCP;
vec11 = UBJ-LBJ

%magnitude of vectors defined above
mag1=norm(vec1); %lower a arm fore
mag2=norm(vec2); %lower a arm aft
mag3=norm(vec3); %upper a arm fore
mag4=norm(vec4); %upper a arm aft
mag5=norm(vec5); %pull rod
mag6=norm(vec6) %tie rod



vecu1=vec1/mag1; %lower a arm fore
vecu2=vec2/mag2; %lower a arm aft
vecu3=vec3/mag3; %upper a arm fore
vecu4=vec4/mag4; %upper a arm aft
vecu5=vec5/mag5; %pull rod
vecu6 =vec6/mag6; %tierod

A =[vecu1(1) vecu2(1) vecu3(1) vecu4(1) vecu5(1) vecu6(1);
    vecu1(2) vecu2(2) vecu3(2) vecu4(2) vecu5(2) vecu6(2);
    vecu1(3) vecu2(3) vecu3(3) vecu4(3) vecu5(3) vecu6(3);
     (vecu1(3)*vec8(2)- vecu1(2)*vec8(3))/1000  (vecu2(3)*vec8(2)-
   vecu2(2)*vec8(3))/1000   (vecu3(3)*vec7(2)- vecu3(2)*vec7(3))/1000
    (vecu4(3)*vec7(2)- vecu4(2)*vec7(3))/1000   (vecu5(3)*vec9(2)-
   vecu5(2)*vec9(3))/1000  (vecu6(3)*vec10(2)- vecu6(2)*vec10(3))/1000;
     -(vecu1(3)*vec8(1)- vecu1(1)*vec8(3))/1000 -(vecu2(3)*vec8(1)-
```

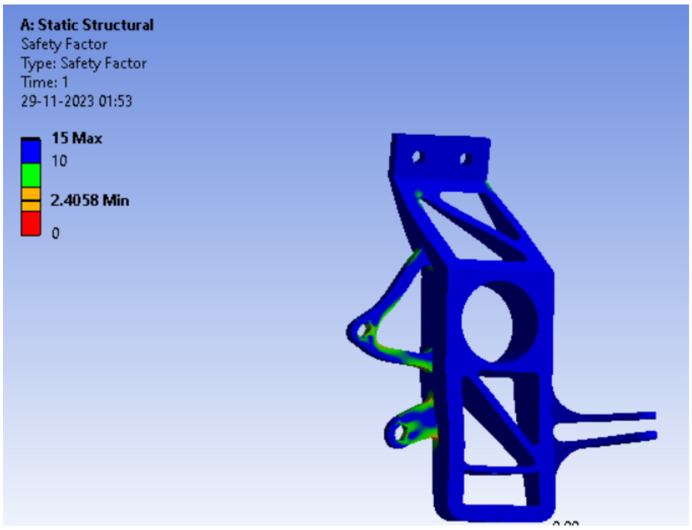
```
    vecu2(1)*vec8(3))/1000  -(vecu3(3)*vec7(1)- vecu3(1)*vec7(3))/1000
    -(vecu4(3)*vec7(1)- vecu4(1)*vec7(3))/1000  -(vecu5(3)*vec9(1)-
    vecu5(1)*vec9(3))/1000 -(vecu6(3)*vec10(1)- vecu6(1)*vec10(3))/1000;
      (vecu1(2)*vec8(1)- vecu1(1)*vec8(2))/1000  (vecu2(2)*vec8(1)-
    vecu2(1)*vec8(2))/1000   (vecu3(2)*vec7(1)- vecu3(1)*vec7(2))/1000
     (vecu4(2)*vec7(1)- vecu4(1)*vec7(2))/1000   (vecu5(2)*vec9(1)-
    vecu5(1)*vec9(2))/1000  (vecu6(2)*vec10(1)- vecu6(1)*vec10(2))
    /1000];

N2=(wt*(b*g-ax*h)/(wb))*((1/2)+h*ay/(tf*g));
N4_a=(wt*(a*g+ax_a*h)/(wb))*((1/2)+h*ay/(tr*g));
%Enter Value of Forces and Torques in X,Y,Z,Tx,Ty respectively in B
Fn = 1391.8;
Fb = 1.4*Fn;
Fcw = 1.4*Fn
B = -[0; 0; Fn; 0; 0; 0];

my_ans = A\B

C1 = [vecu1(1) vecu2(1) vecu3(1) vecu4(1) vecu5(1) vecu6(1);
     vecu1(2) vecu2(2) vecu3(2) vecu4(2) vecu5(2) vecu6(2);
     vecu1(3) vecu2(3) vecu3(3) vecu4(3) vecu5(3) vecu6(3)];


Ct = (transpose(C1));

my_ans = transpose(my_ans);
LAF = (my_ans(1))*Ct(1,:)
LAR = (my_ans(2))*Ct(2,:)
UAF = (my_ans(3))*Ct(3,:)
UAR = (my_ans(4))*Ct(4,:)
PR = (my_ans(5))*Ct(5,:)
TR = (my_ans(6))*Ct(6,:)

F1mag =sqrt(sum(LAF.*LAF));
F2mag =sqrt(sum(LAR.*LAR));
F3mag =sqrt(sum(UAF.*UAF));
F4mag =sqrt(sum(UAR.*UAR));
F5mag =sqrt(sum(PR.*PR));
F6mag =sqrt(sum(TR.*TR));



Fx = LAF(1)+LAR(1)+UAF(1)+UAR(1)+PR(1)+TR(1)
Fy = LAF(2)+LAR(2)+UAF(2)+UAR(2)+PR(2)+TR(2)
Fz = LAF(3)+LAR(3)+UAF(3)+UAR(3)+PR(3)+TR(3)

Fubj = UAF+UAR+PR
Flbj = LAF+LAR
Ftie = TR

z=[F1mag;F2mag;F3mag;F4mag;F5mag;F6mag];
z=norm(z)
```

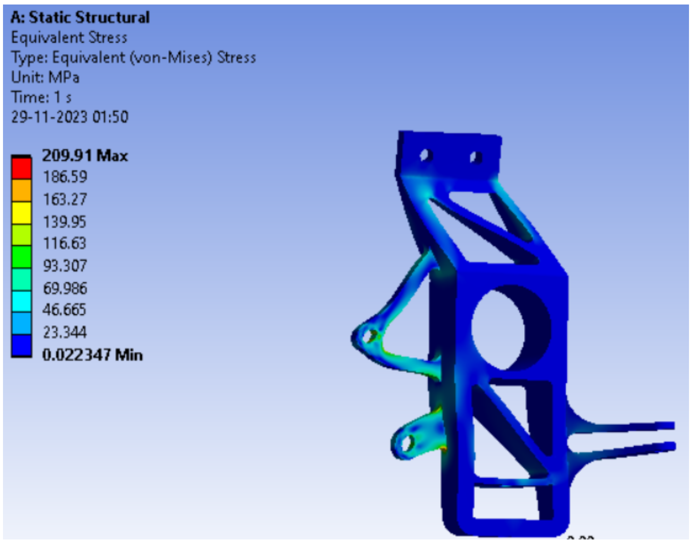
Listing 6.2: Knuckle Forces Loads

6.6 Knuckle Images

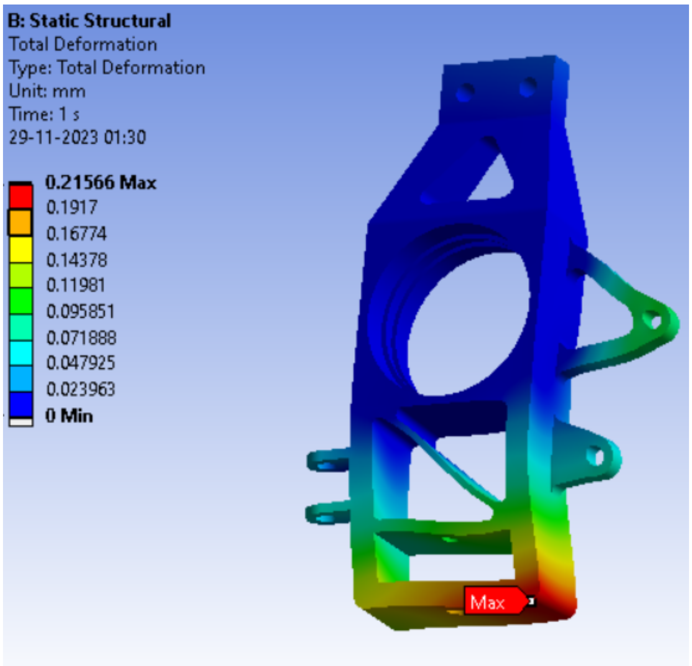
6.6.1 Front Knuckle

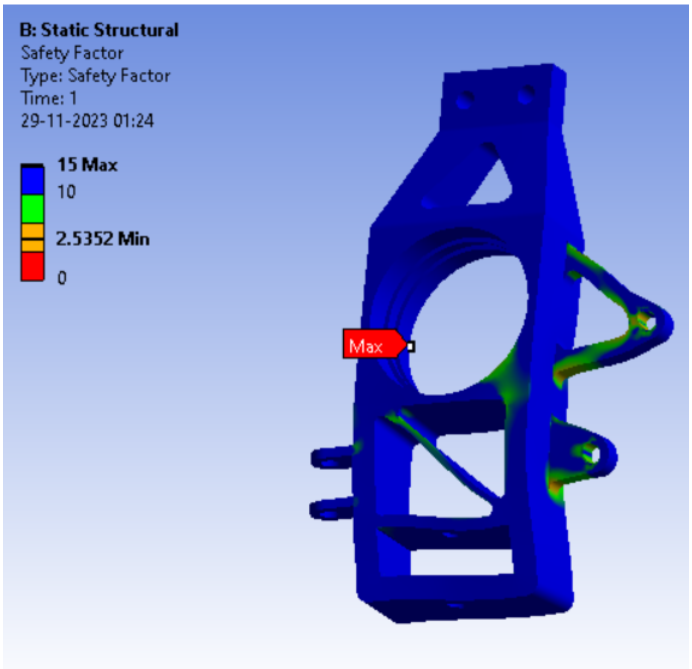


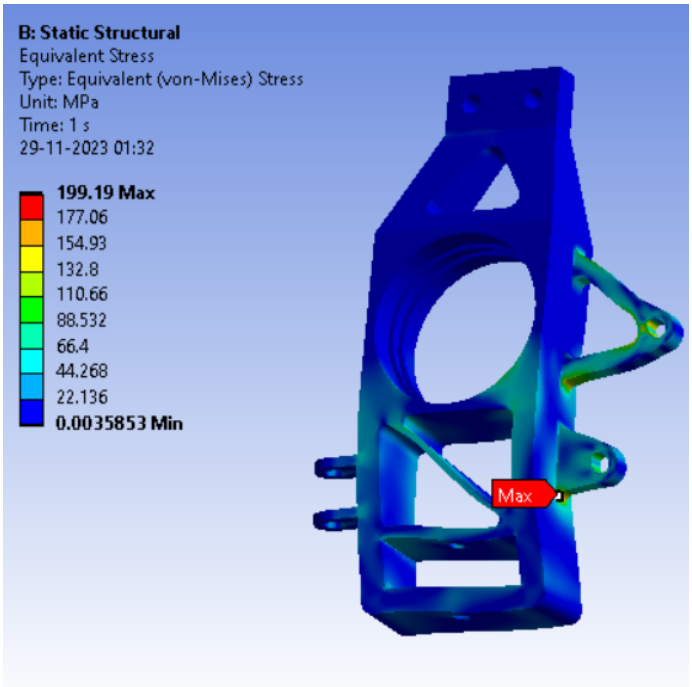




6.6.2 Rear Knuckle







6.7 Hub Analysis

6.7.1 Material selection

Despite the material's unfavorable fatigue strength, aluminum 7075-T6 was chosen to make the knuckle and hub assembly because it has a higher strength to weight ratio than steel with strong fatigue strength.

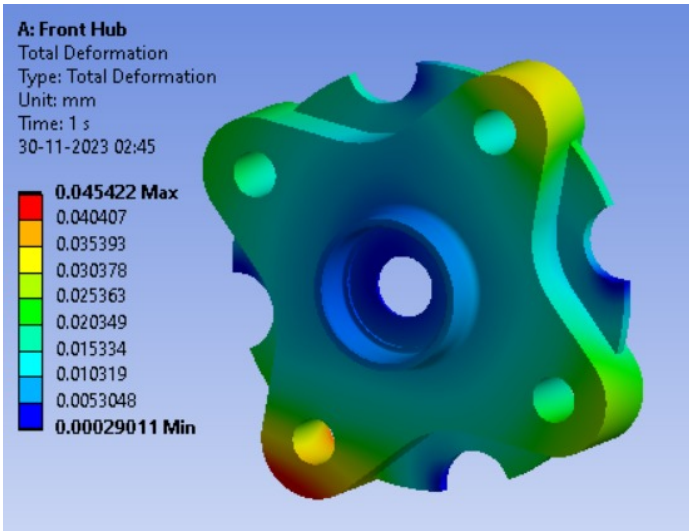
6.7.2 Approach for design

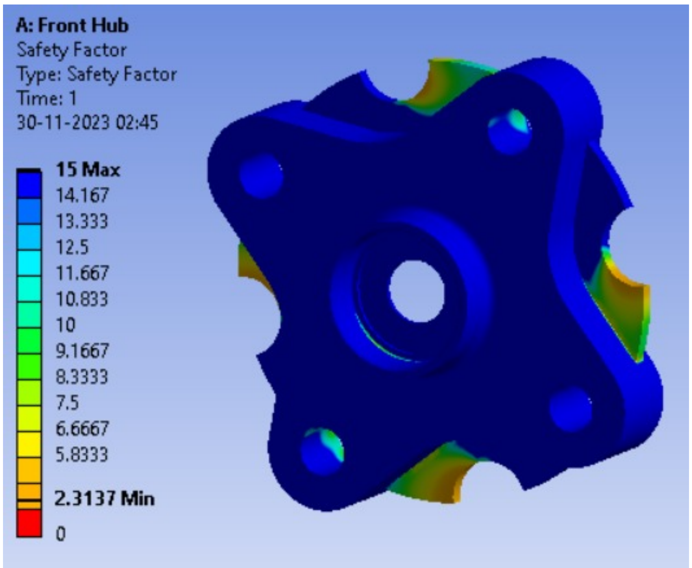
To start, we use the vehicle's selected rim to calculate the PCD diameter and hub stud location for the design. Next, we analyze the data using Ansys to find the optimal design that takes manufacturing and budget constraints into consideration. We simply iterate by removing undesirable regions while keeping the factory of safety in mind to arrive at the best possible design.

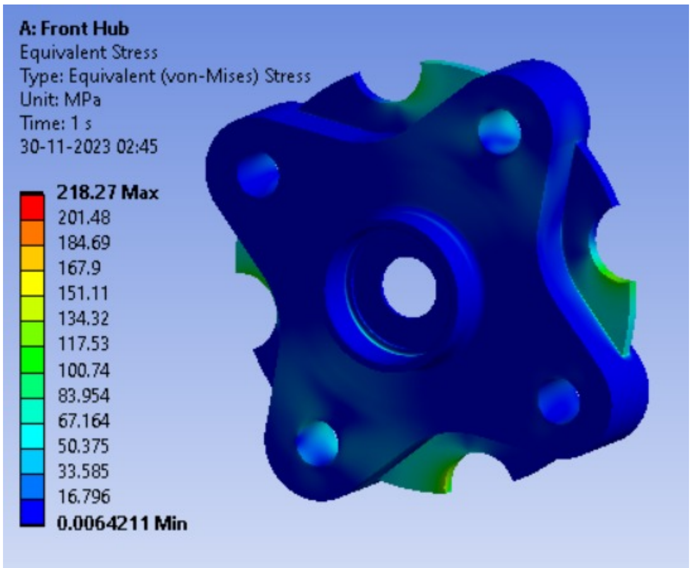
6.7.3 Design Analysis

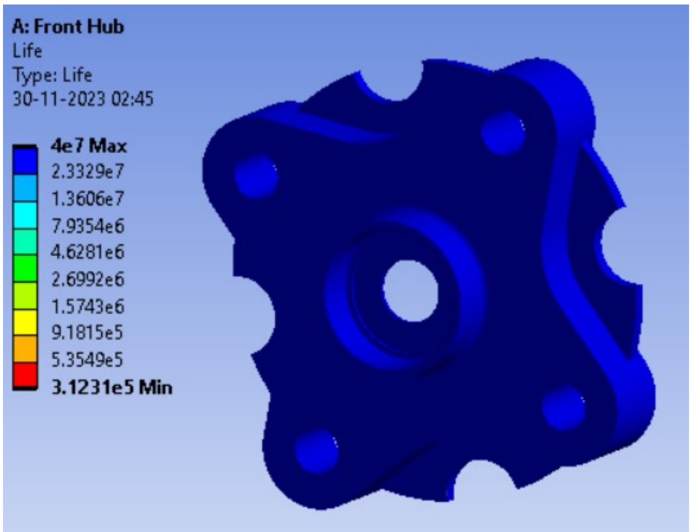
For our design analysis in ansys we kept the square extrusion that was in contact with the spindle to be fixed for the rear spindle.
Now, to get the optimized solution we considered the case in which the car was breaking and turning at the same time, We got the respective force from the wheel load codes that was previously explained and the moment applied is by the motor that is equivalent to 140Nm.
For our rear hub analysis we also used remote force equivalent to (-2942.4 N, -2942.4 N, -1451.1 N) in x, y and z direction; The remote force was applied from the tyre contact patch for analysis, The values for the forces are found while taking in consideration the worst case possible this case eliminates all the other cases thus best for our analysis.
We used remote force analysis to simulate real life scenarios as the force on the hub is due to the contact patch of the vehicle.
Similarly, we did the analysis for the front hub we just changed the wheel load force that was obtained for the front tyres.
We defined the mesh for our analysis to be 0.5mm to make our analysis as accurate as possible, an ideal analysis consist of an object containing 10e6 elements.

Front Hub

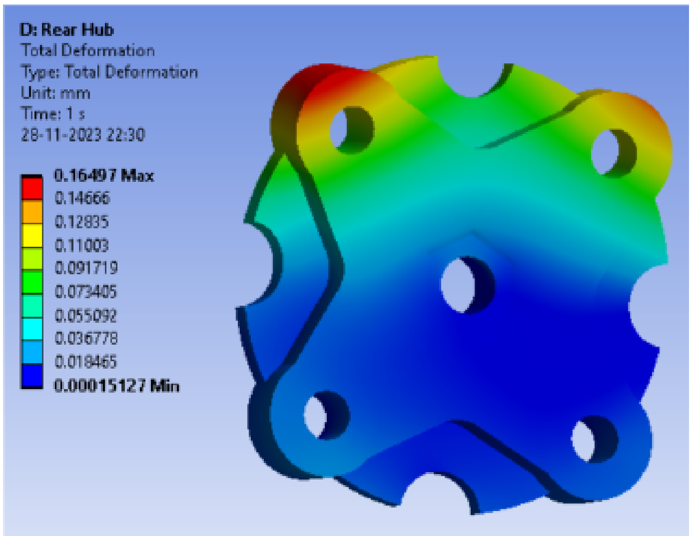


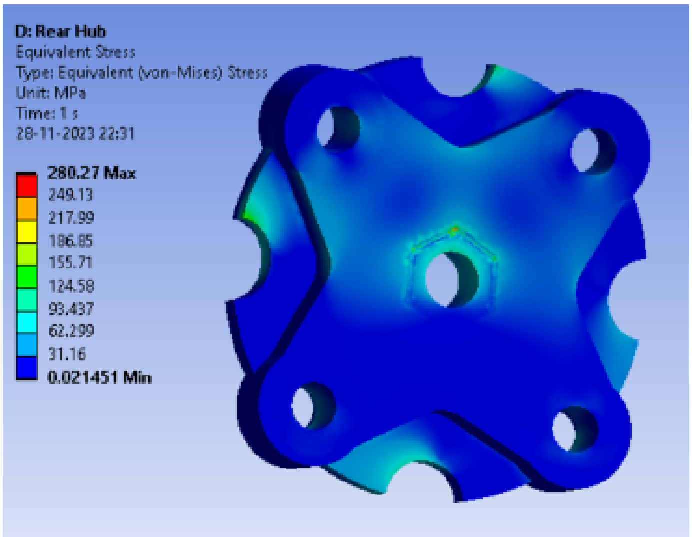

A: Front Hub
Safety Factor
Type: Safety Factor
Time: 1
30-11-2023 02:45
15 Max
14.167
13.333
12.5
11.667
10.833
10
9.1667
8.3333
7.5
6.6667
5.8333
2.3137 Min
0

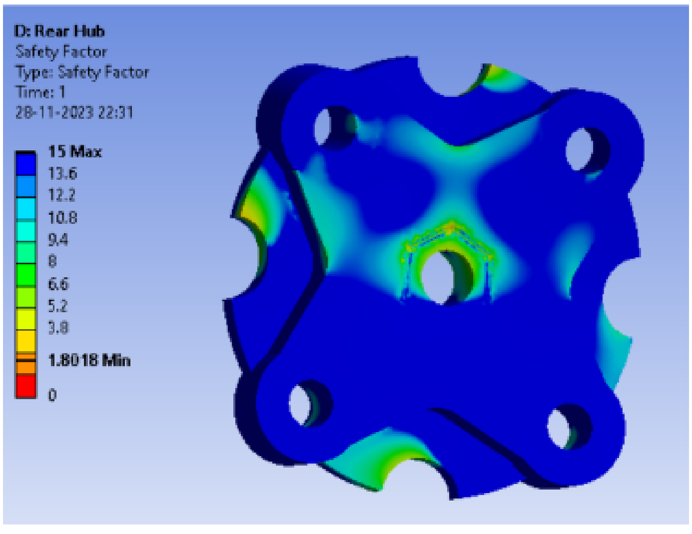

A: Front Hub
Equivalent Stress
Type: Equivalent (von-Mises) Stress
Unit: MPa
Time: 1 s
30-11-2023 02:45
218.27 Max
201.48
184.69
167.9
151.11
134.32
117.53
100.74
83.954
67.164
50.375
33.585
16.796
0.0064211 Min


A: Front Hub
Life
Type: Life
30-11-2023 02:45
4e7 Max
2.3329e7
1.3606e7
7.9354e6
4.6281e6
2.6992e6
1.5743e6
9.1815e5
5.3549e5
3.1231e5 Min

Rear Hub


D: Rear Hub
Total Deformation
Type: Total Deformation
Unit: mm
Time: 1 s
28-11-2023 22:30
0.16497 Max
0.14666
0.12835
0.11003
0.091719
0.073405
0.055092
0.036778
0.018465
0.00015127 Min


D: Rear Hub
Equivalent Stress
Type: Equivalent (von-Mises) Stress
Unit: MPa
Time: 1 s
28-11-2023 22:31
280.27 Max
249.13
217.99
186.85
155.71
124.58
93.437
62.299
31.16
0.021451 Min


D: Rear Hub
Safety Factor
Type: Safety Factor
Time: 1
28-11-2023 22:31
15 Max
13.6
12.2
10.8
9.4
8
6.6
5.2
3.8
1.8018 Min
0

6.8 Spindle Analysis

6.8.1 Material Selection

Priority was given to weight reduction over steel with high fatigue strength, so AISI 4340 alloy steel was used in our spindle manufacturing even though Al 7075 T6 was chosen for the hub and knuckle because it has a high strength to weight ratio when compared to other materials like Al 6061 and AISI 1020, but the fatigue strength is low.

6.8.2 Approach for design

For spindle we firstly took a cylinder which can withstand torque exerted by the drive shaft and at the same time can rotate the wheel assembly by maintaining its structural integrity. Then we optimise the design such that we reduce the weight and maintain the ease of assembly.

Rzeppa joints were used to transmit the torque from driveshaft to the wheel assembly.

We decided to use dead spindle in the front, hub and knuckle were designed accordingly.

We decided to use two bearings to ensure smooth rotation of the hub, and to restrict the axial motion of the individual components by incorporating respective abutments for the bearings.Bearing used are SKF6306 and SKF62022z. Finally, we used a castle nut and cotter pin to secure the wheel assembly from the outer end.

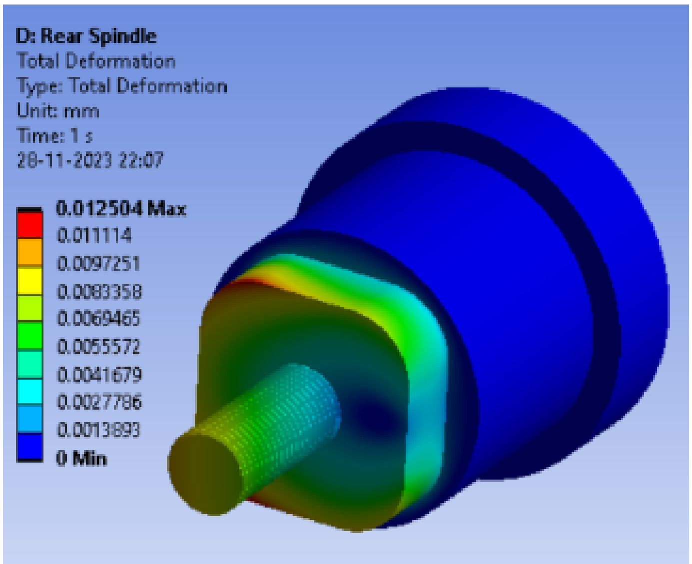
For rear spindle we used live spindle as per our requirement of our vehicle. The only difference except this is that we use the same two bearings between knuckle and spindle them being SKF 61910.

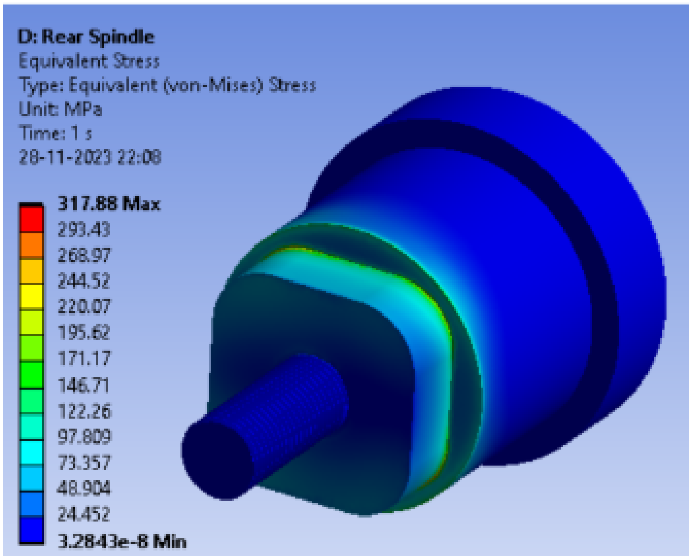
6.8.3 Design Analysis

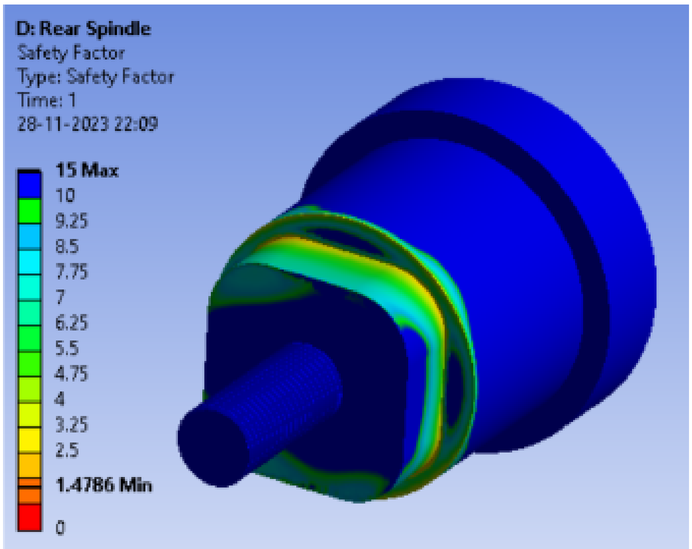
In order to facilitate the transmission of torque from the drive shaft to the wheel assembly, we employed remote force on the square extrusion that we developed throughout the design process, together with cylindrical support on the bearings and compression support in the spline area for the rear spindle analysis.

We employed bearing load where the bearings are located and fixed support in the knuckle area for the front spindle study. Additionally, we imparted distant force to the abutment, which is essentially lateral force brought on by bearing cornering.

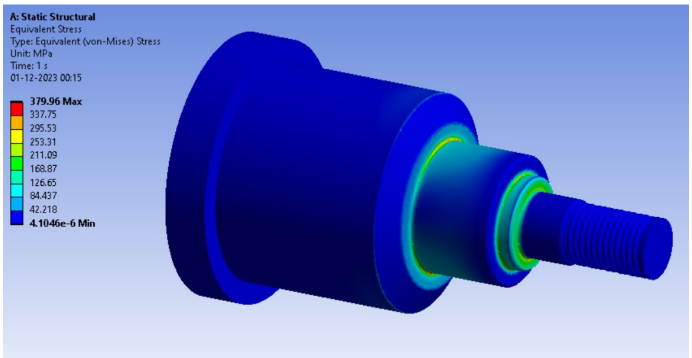
Rear Spindle

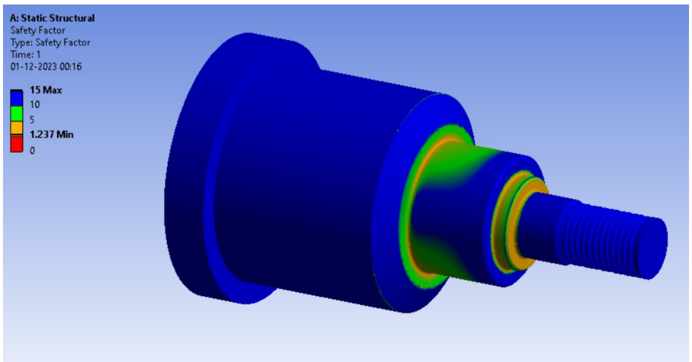






Front Spindle





6.9 Optimum time lap

Lap time for our vehicle on the track has been calculated with a simplified point mass model. This helps us determine how well our vehicle performs on the track and allows us to assess how any optimizations to the vehicle affect its performance on the track. The code is divided into four parts.

```
function [time] = optimum_time_lap()
%importing Track Data
%column1 right left straight  column 2 Arc length column 3 Radius of
%curvature
global SvT;
global trackdata;
global i;
global V;
global T;
global k;
k=0;
V=table2array(SvT(:,1));
T=table2array(SvT(:,2));

%defining frictional cofficiant, gravitational acc, and max
    dcceleration
%possible in car
u=1.4; g=9.8; dcc=1*g;

track_no=height(trackdata);

vi=0; time=0;

for i=1:(track_no-1)
    z=otl(vi,i);
    time=time+z(1,1);
    time=double(time);

    vi=z(1,2);
    vi=double(vi);
end
if(trackdata(track_no,1)=='S')
 time=double(time+str2double((trackdata(track_no,2)))/vi)

elseif(trackdata(track_no,1)=='L' || trackdata(track_no,1)=='R')
    Radius=str2double(trackdata(i,3));
    Arclen=str2double(trackdata(i,2));
    v_max= sqrt(u*g*Radius);

    if (vi==v_max)
        time=Arclen/vi;
        v_exit=vi;
    end

    if (vi<v_max)
        fun1 = @(v) 1 ./ (sqrt((u*g).^2-(v.^4 ./Radius.^2)));
        fun2 = @(v) v ./ (sqrt((u*g).^2-(v.^4 ./Radius.^2)));

        s1 = integral(fun2,vi,v_max);
        s2=Arclen-s1;

        if(s2>=0)
            time = integral(fun1,vi,v_max)+s2/v_max;
            v_exit = v_max;
        end
        if(s2<0)
            integralFunc = @(v) integral(fun2, vi, v);
```

```
            v = fminbnd (@(v) abs(integralFunc(v) - Arclen), vi, v_max)
   ;

            time = integral(fun1,vi,v);
            v_exit = v;

        end
    end

     if(vi>v_max)
        'error in'
        i
        time = Arclen/vi;
        v_exit = v_max;
     end

end
end
```

Listing 6.3: Matlab code for finding lap time of our vehicle in the track

```
function [z] = otl(vi,i)

%importing Track Data
%column1 right left straight  column 2 Arc length column 3 Radius of
%curvature
global trackdata;
%defining frictional cofficiant, gravitational acc, and max
   dcceleration
%possible in car
u=1.4; g=9.8; dcc=2*g;



%encountring straight track with curve at the end
if (trackdata(i,1)=='S' && (trackdata(i+1,1)== 'R' || trackdata(i+1,1)
   == 'L'))
    Radius=str2double(trackdata(i+1,3));%............Radius of curve
    Arclen=str2double(trackdata(i+1,2));%............Arclength of curve
    distance=str2double(trackdata(i,2));%............straight track
   distance.
    vmax= sqrt(u*g*Radius)%........................  speed targated to
    be achived at the end of straight line.

    v_final_possible= sqrt(vi^2 + 2 * u*g * distance);
    aaa=1
    %Maximum possible speed at the end acclerating through full
   potential


    if (vmax >= v_final_possible)
        bbb=2
        time=(v_final_possible-vi)/(u*g);
        v_exit=v_final_possible;
        VDAmain(V,T,vi,vf,distance);
    end

    if (vmax < v_final_possible)
        ccc=3
```

```
        time=straightlinetime(vi,vmax,distance);
        v_exit=vmax;
        VDAmain(V,T,vi,vmax,distance);
    end

end




if ((trackdata(i,1)== 'R' || trackdata(i,1)== 'L') && (trackdata(i+1,1) == 'S'))
    %ddd=4
    Radius=str2double(trackdata(i,3));
    Arclen=str2double(trackdata(i,2));
    v_max= sqrt(u*g*Radius);


    if (vi==v_max)
        eee=5
        time=Arclen/vi;
        v_exit=vi;
        VDAmain(V,T,vi,vf,Arclen)
    end

    if (vi<v_max)
        fff=6
        fun1 = @(v) 1 ./ (sqrt((u*g).^2-(v.^4 ./Radius.^2)));
        fun2 = @(v) v ./ (sqrt((u*g).^2-(v.^4 ./Radius.^2)));

        s1 = integral(fun2,vi,v_max);
        s2=Arclen-s1;

        if(s2>=0)
            iii=7
            time = integral(fun1,vi,v_max)+s2/v_max;
            v_exit = v_max;
            VDAmain(V,T,vi,v_max,s1);
            VDAmain(V,T,v_max,v_max,s2);
        end
        if(s2<0)
            jjj=8
            integralFunc = @(v) integral(fun2, vi, v);
            v = fminbnd (@(v) abs(integralFunc(v) - Arclen), vi, v_max);

            time = integral(fun1,vi,v);
            v_exit = v;
            VDAmain(V,T,vi,v,Arclen);

        end
    end

    if(vi>v_max)
        %kkk=9
        'error in'
        i
        time = Arclen/vi;
```

```
        v_exit = v_max;
    end

end


if ((trackdata(i+1,1)== 'R' || trackdata(i+1,1)== 'L') && (trackdata(i
    ,1)== 'R' || trackdata(i,1)== 'L'))
    Radius_1=str2double(trackdata(i,3));
    Arclen_1=str2double(trackdata(i,2));
    Radius_2=str2double(trackdata(i+1,3));
    Arclen_2=str2double(trackdata(i,2));

    v_max_1= sqrt(u*g*Radius_1);
    v_max_2= sqrt(u*g*Radius_2);
    lll=10


    if (v_max_1<=v_max_2)
        mmm=11

        if (vi==v_max_1)
            nnn=12
            time=Arclen_1/vi;
            v_exit=vi;
            VDAmain(V,T,vi,vi,Arclen_1);
        end


        if (vi<v_max_1)
            ooo=13
            fun1 = @(v) 1 ./ (sqrt((u*g).^2-(v.^4 ./Radius_1.^2)));
            fun2 = @(v) v ./ (sqrt((u*g).^2-(v.^4 ./Radius_1.^2)));
            s1 = integral(fun2,vi,v_max_1);
            s2=Arclen_1-s1;


            if(s2>=0)
                ppp=14
                time = integral(fun1,vi,v_max_1)+s2/v_max_1;
                v_exit = v_max_1;
                VDAmain(V,T,vi,v_max_1,s1);
                VDAmain(V,T,v_max_1,v_max_1,s2);
            end

            if(s2<0)
                qqq=15
                integralFunc = @(v) integral(fun2, vi, v);
                v = fminbnd(@(xo) abs(integralFunc(v) - Arclen_1), vi,
   v_max_1);
                time = integral(fun1,vi,v);
                v_exit = v;
                VDAmain(V,T,vi,v,Arclen_1);
            end
        end

        if(vi>=v_max_1)
            rrr=16
```

```
            % 'error in'
             %i
             time = Arclen_1/vi;
             v_exit = vi;
         end
     end

     if(v_max_1>v_max_2)
         sss =17
         fun_dis_acc = @(v) v./(sqrt((u*g).^2-(v.^4./Radius_1.^2)));
         fun_dis_dcc = @(v) -v./(sqrt((u*g).^2-(v.^4./Radius_1.^2)));

         fun_time_acc = @(v) 1./(sqrt((u*g).^2-(v.^4./Radius_1.^2)));

         integralFunc = @(vo) integral(fun_dis_acc, vi, vo) + integral(
    fun_dis_acc,v_max_2,vo);
         vo = fminbnd(@(vo) abs(integral(fun_dis_acc, vi, vo) + integral
    (fun_dis_acc,v_max_2,vo)-Arclen_1), v_max_2, 4*v_max_1);


         if(vo<=v_max_1)
             ttt=18
             time = integral(fun_time_acc, vi, vo) + integral(
    fun_time_acc, vo, v_max_2);
             v_exit = v_max_2;
             VDAmain(V,T,vi,vo,Arclen_1);
             VDAmain(V,T,vo,v_max_2,Arclen_1);

         end

         if(vo>v_max_1)
             if(vi==v_max_1)
                  s_acc=0;
                  s_dcc = integral(fun_dis_acc, v_max_2, v_max_1);
                  time=integral(fun_time_acc, vi, v_max_2) + (Arclen_1-
    s_acc-s_dcc)/v_max_1;
                  v_exit = v_max_2;
                  VDAmain(V,T,vi,vi,Arclen_1-s_dcc);
                  VDAmain(V,T,vi,v_exit,s_dcc);
             else
             uuu=19
             s_acc = integral(fun_dis_acc, vi, v_max_1);
             s_dcc = integral(fun_dis_acc, v_max_2, v_max_1);
             time  = integral(fun_time_acc, vi, v_max_1) + integral(
    fun_time_acc, vi, v_max_2) + (Arclen_1-s_acc-s_dcc)/v_max_1;
             v_exit = v_max_2;
             VDAmain(V,T,vi,v_max_1,s_acc);
             VDAmain(V,T,vi,v_max_2,s_dcc);
             VDAmain(V,T,v_max_2,v_max_2,arclen_1-s_acc-s_dcc);
             end
         end
     end
end
```

```
z=[time,v_exit];

function [time] = straightlinetime(vi,vf,d)

```

```
%global trackdata;
u=1.4; g=9.8; dcc=1*g;

syms vo s1  s2
eq1=vi^2+2*u*g*s1-vo^2==0;
eq2=vf^2+2*dcc*s2-vo^2==0;
eq3=s1+s2==d;


result=vpasolve([eq1,eq2,eq3],[vo,s1,s2]);

vo=result.vo
vo = vo(result.vo > 0);


time = (vo-vi)/(u*g)+(vo-vf)/dcc;

end
```

```
function [table] = VDAmain(V,T,vi,vf,d)
%for getting the velocity vs displacement vs acceleration graph
v1=vi;
n=1000;
h=(vf-vi)/n;
s=0;
if(vi==vf)
 for k=k:(k+n)           %using k to update values consecutively in the table
  v2=vi+k*h;
  s1=integral(@(v1) v1./((interp1(V,T,v1,'linear'))/(250*0.2)),v1,v2); %finding distance travelled for a small change in velocity
  v1=v2;
  s=s+s1;
  TAB(k+1,1)=v2;
  TAB(k+1,2)=s;
  TAB(k+1,3)=((interp1(V,T,v1,'linear'))/(250*0.2));
 end
else
    for k=k:(k+n)
    s1=d/1000;
    s=s+s1
    v=vi
  TAB(k+1,1)=v;
  TAB(k+1,2)=s;
  TAB(k+1,3)=0
    end
end
 x=TAB(:,1);          %velocity
 y=TAB(:,2);          %displacement
 z=TAB(:,3);          %acceleration
 table=TAB
end
```

Chapter 7

Vehicle Dynamics

7.1 Calculating control rod mounting points:

Initial Steps :

- Brake rotors : The wheel offset is worked out in conjunction with fItting the brake caliper to clear the inside surface of the wheel.
- Lower Ball joint : With the rotor location comes the absolute farthest outboard location for the lower ball joint; this sets the lower ball joint lateral position .The height of the lower ball joint such that :-

  A deflated tire ground clearance might be in order.
  If it is totally inside the wheel all it has to do is clear the wheel and the brake rotor under all travel and load conditions. From all these considerations, we used a knuckle height of 202 mm in the front and 200 mm in the rear to calculate the LBJ coordinates.
- Upper Ball joint : The decision about the kingpin angle in the front view (2 deg) and in the rear view(4 deg) will ultimately decide the coordinates of UBJ.

  From now on the geometry is made using 2 perpendicular projection planes. One Along the wheels and the other perpendicular to it containing the kingpin axis. Set instant axis of the independent wheel, thereby giving the two instant centres
  Next step is to choose the desired control arm length and constructing a plane by selecting the inner pivot point 1.
  Followed by taking projection onto the tire plane as point 3 and then transferring it onto the side view by extending line connecting point 3 and the side view IC. mark it as point 4 , helps decide A-arm plane.
  By projecting these lines again on the front plane we get the planes containing the A-arms , all the upper A-arm point 1 through 4 must line on the same plane.
  Project a line from point 4 to point 2 (extend this till point 1, assuming inner pivot points to be parallel to the center-line of the vehicle)
  Draw vertical line through point 1 ; establishing the UCA (upper control arm) axis

Now project point 5 on the side view and the control arm pivot point must lie on this line connecting 1-5.
Select any two desired points based on calculations of (iterative process)

- Forces on the suspension links
- Desired plane of the links
- Limiting unsprung weight
- Vehicle integration (with other sub-system like chassis)

The wheel is constrained to rotate around the steering axis defined by upper and lower ball joint so the outer tie rod point in constraint to move along a defined path. This path can be approximated by setting the tie rod to pass through the instant centre of the suspension assembly. Now, as the suspension moves, the tie rod can be considered rotating about the IC.

Push/Pull rod Geometry Design Process and calculation

The main insights of constructing this geometry were :

- Optimizing Motion Ratio for the vehicle during wheel travel
- Maintaining a single plane of motion of the components
- Packaging of the assembly
- Integration with node points of chassis
- Force on components
- Easy Serviceability for chassis tuning during testing

Steps for geometry construction :

1.Deciding lower pick-up point of the pullrod based on these factors

- Force transmitted in the components - Angle
- Motion ratio
- Weight

2.Deciding the upper pick-up point based

- Packaging of spring-dampers
- Integration with node points of chassis

3.Designing bell-crank with iterative process considering

- Motion ratio

- Force - Weight balance

The following multivariable nonlinear optimisation was performed

Factors put into consideration while preparing optimisation function :

- Motion Ratio of Bellcrank
- Unsprung and lower pickup point tab (weight)
- Bending force on Suspension link

The motion ratio was studied on Mathworks Matlab software and the following data for different upper pick-up location was studied for a range of -35mm to 35mm bump.
* the bump was considered vertical instead of giving considerations to instant axis as the results were similar.

The force in the pullrod was iterated through the Matlab code for a different set of values along the lateral directions of the upper pick up point.

The pullrod force was analyzed for 3-point bending in the A-arms. The deflection was calculated as a function of upper pick up point location in the y-axis.

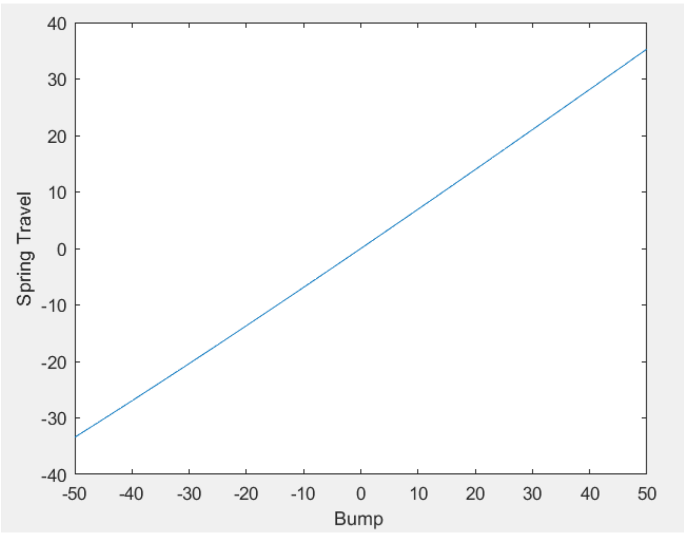


Figure 7.1: Motion ratio

The bending deflection was calculated for the 3 point bending and the deflections were calculated using the superposition method.

We have also developed a kinematic suspension model using Matlab. It uses the principle of axis angle rotation for rotation and Newton Raphson method for the approximation of suspension geometry. The model gave almost identical results to Lotus Shark Software with an error of less than 3%.

Roll Angle (deg)	Camber Angle (deg)	Toe Angle {SAE} (deg)	Castor Angle (deg)	Kingpin Angle (deg)
3.00	0.6605	0.1172	8.4745	-0.0788
2.90	0.5926	0.1108	8.4586	-0.0098
2.80	0.5245	0.1045	8.4428	0.0591
2.70	0.4565	0.0985	8.4269	0.1281
2.60	0.3883	0.0926	8.4111	0.1971
2.50	0.3201	0.0868	8.3953	0.2662
2.40	0.2519	0.0813	8.3794	0.3352
2.30	0.1836	0.0759	8.3636	0.4043
2.20	0.1152	0.0707	8.3478	0.4735
2.10	0.0468	0.0656	8.3321	0.5426
2.00	-0.0217	0.0608	8.3163	0.6118

(a) Lotus Shark Model

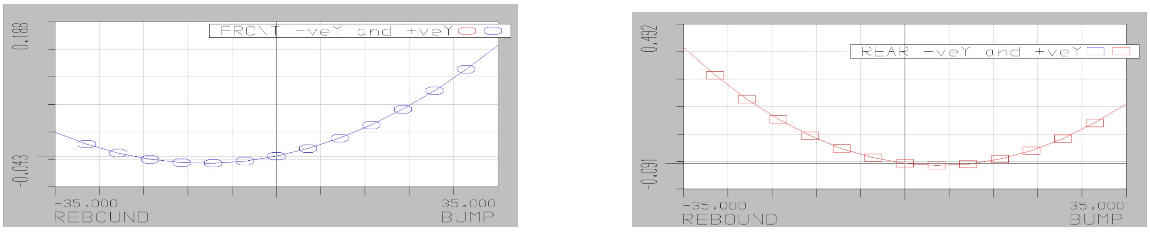
Roll(deg)	Camber(deg)	Toe(deg)	Castor(deg)	KPI(deg)
3.0000	0.6688	0.1158	8.4711	-0.0852
2.9000	0.6006	0.1094	8.4554	-0.0161
2.8000	0.5324	0.1032	8.4397	0.0531
2.7000	0.4641	0.0972	8.4239	0.1223
2.6000	0.3957	0.0914	8.4082	0.1915
2.5000	0.3273	0.0858	8.3925	0.2608
2.4000	0.2588	0.0803	8.3768	0.3301
2.3000	0.1903	0.0750	8.3611	0.3994
2.2000	0.1217	0.0698	8.3454	0.4687
2.1000	0.0530	0.0648	8.3297	0.5381
2.0000	-0.0157	0.0600	8.3140	0.6075

(b) Our Model

Figure 7.2: Comparision between our model and Shark model

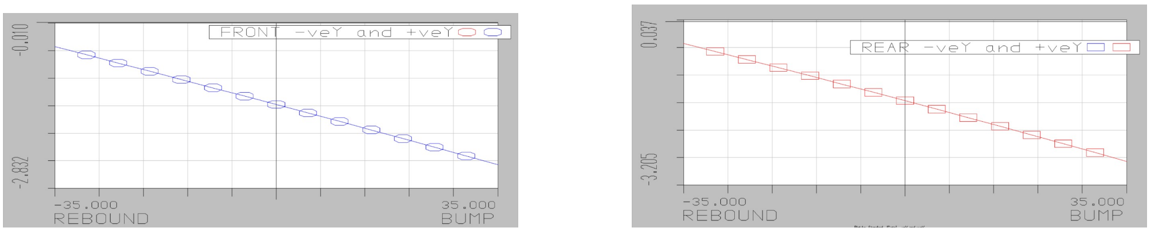
The above geometry was analyzed on Lotus Shark software and the following study was obtained :-

Graphs of camber, caster, toe angles vs bump/droop and roll :-



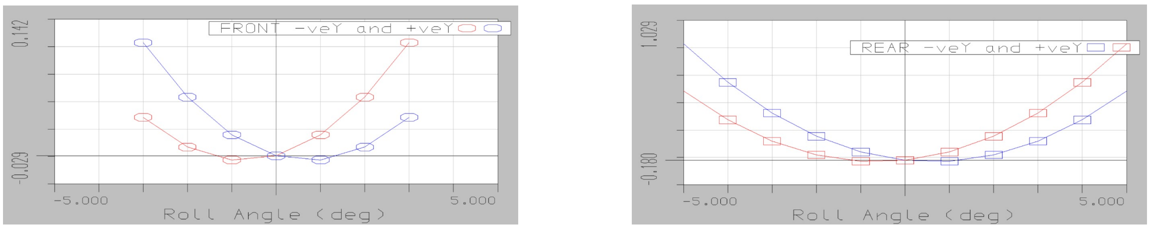
(a) Front (b) Rear

Figure 7.3: Bump vs Toe



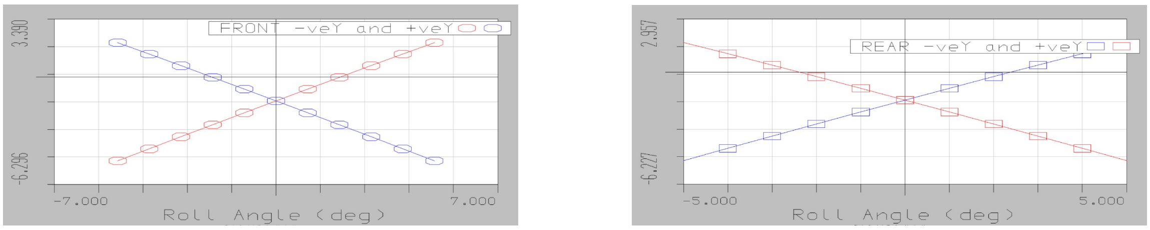
(a) Front (b) Rear

Figure 7.4: Bump vs Camber



(a) Front (b) Rear

Figure 7.5: Roll vs Toe



(a) Front (b) Rear

Figure 7.6: Roll vs Camber

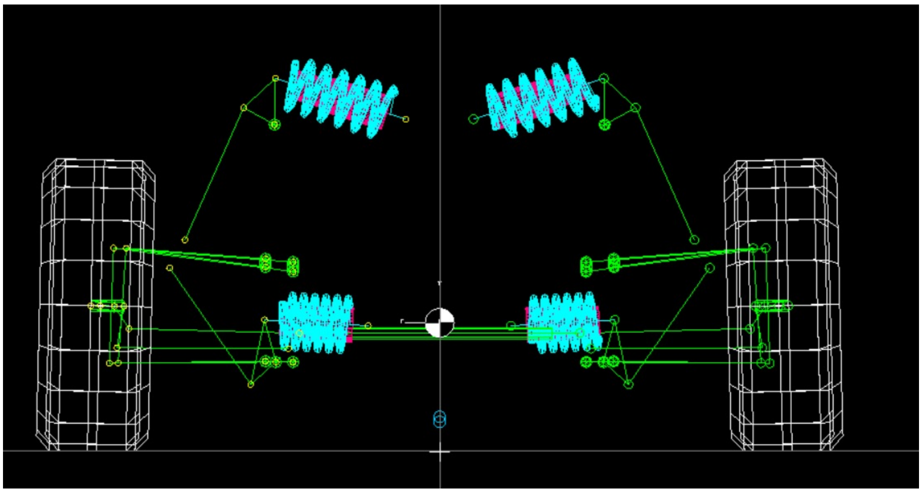


Figure 7.7: Suspension Geometry

7.2 Spring Dampers

7.2.1 Spring

Spring has mainly two deciding characteristics:

(a) Stiffness (Spring Rate)

The spring rate is proportional to the ride stiffness, for a fixed motion ratio. Stiffer the spring would be, stiffer would be the ride. We want appropriate ride stiffness because large stiffness will give a very rough ride and very small stiffness will give a bouncy ride. The variable which defines ride stiffness is ride frequency which we discussed above. Also there is a relation between spring rate and wheel rate which is related by motion ratio. So, by changing motion ratio we change the ride stiffness, without changing the spring rate. We nearly fixed our motion ratio and ride frequency so that we could easily get spring rate.

(b) Length

Now, the length of spring is important because of the travel we want. If the spring would be smaller in length then it would not be able to provide the appropriate compression needed. So, we will have to choose the spring length according to our need for suspension travel.

Coil Over Springs have a fixed spring rate, it can't be changed. Only Preloaded can be adjusted. We select the spring by first deciding the length of spring and then finding a spring with spring rate value close to our calculated value.
Spring rate is fixed so we need to carefully choose which spring rate we are going to use.

7.2.2 Dampers

- At first we select the length and travel.
- Then we see that if we have to use a damper with fixed velocity vs. force characteristic or adjustable ones. Adjustable are better because we can always tune the according to our need.
- Dual speed adjustment is much better but adds a lot of cost, and also the tuning become a bit more difficult.

7.2.3 Selection

The damper lengths were decided on the basis of assembly requirements in integration with chassis and steering subsystem. The following specifications were chosen for the damper

7.2.4 Specifications:

- Overall length = 200 or 267 mm (center to center of spherical bearings, fully extended)
- Stroke = 57 or 90mm
- Weight = 57mm stroke = 394g without spring, 90mm stroke = 446 g without spring
- Spherical Bearing dimensions:
- ID = 8 mm

- Ball Width = 8 mm
- OD = 15 mm

The springs were selected out of the given list of springs and which was closest to our calculated value from ride and roll rate calculations

7.3 Design and analysis of suspension parts

Codes for Dynamic force analysis We have selected AISI4130 for making the arms as It has high machinability, high strength, high ductility and good weldability
Iterations for dimension analysis have been shown below :

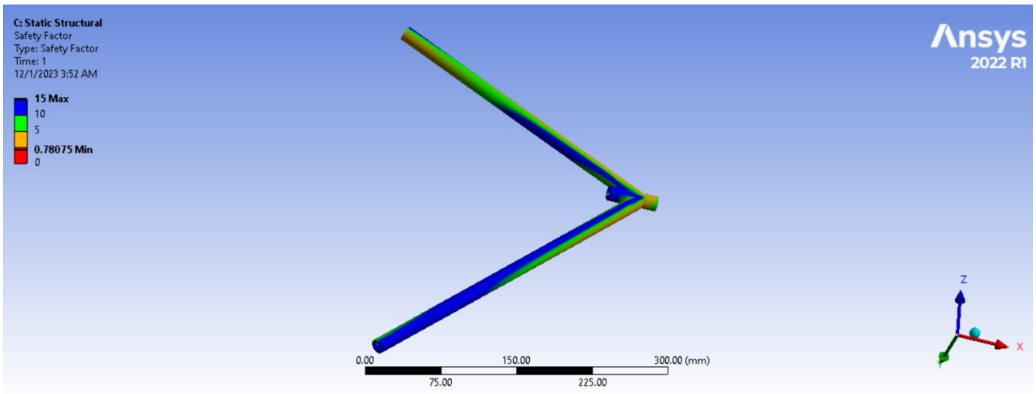


Figure 7.8: Front Lower Arm

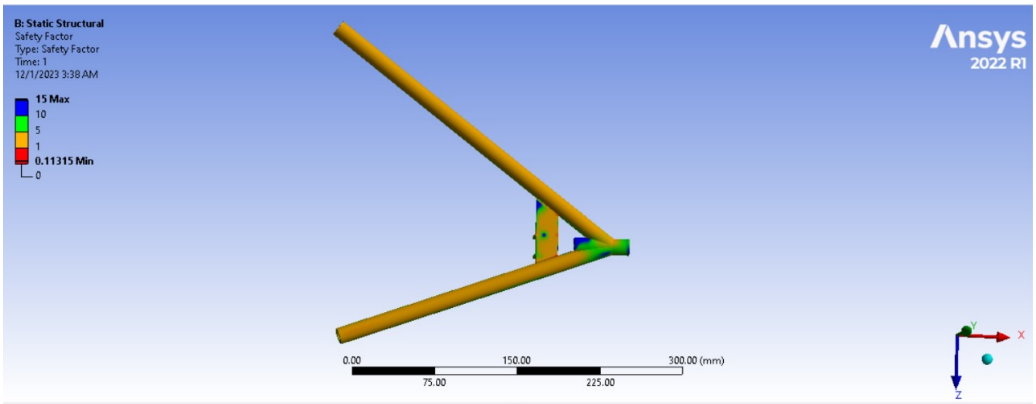


Figure 7.9: Front Upper Arm

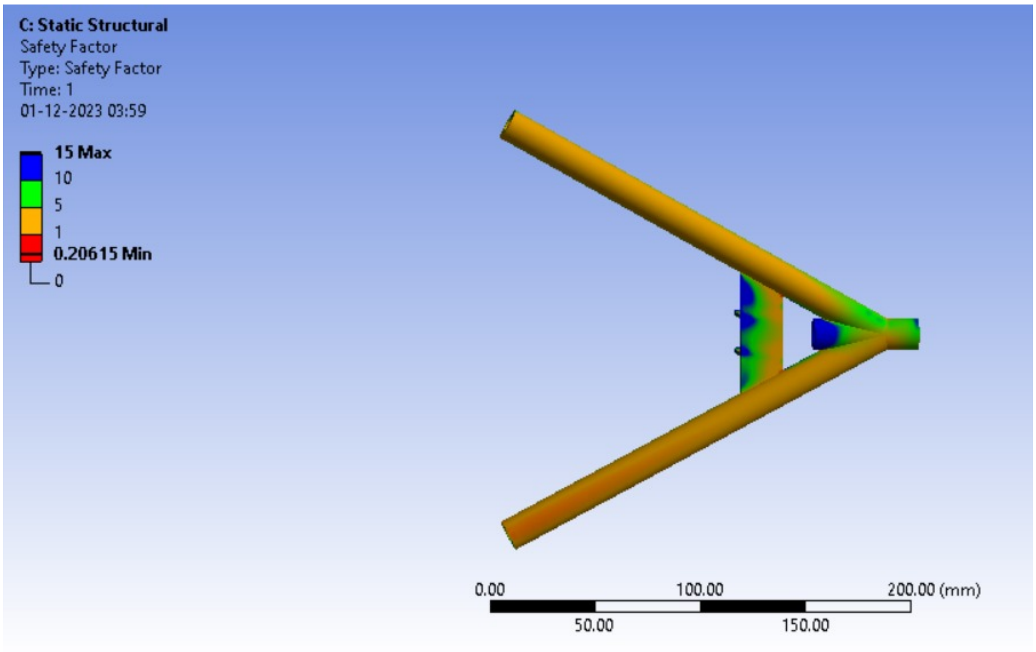


Figure 7.10: Rear Upper Arm

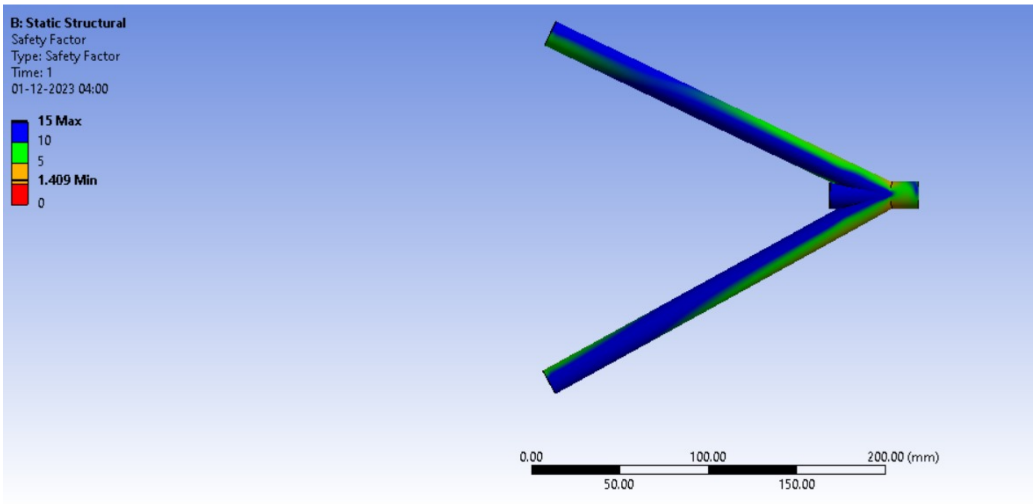


Figure 7.11: Rear Lower Arm

Similar analysis was done for the front and rear bellcranks used.

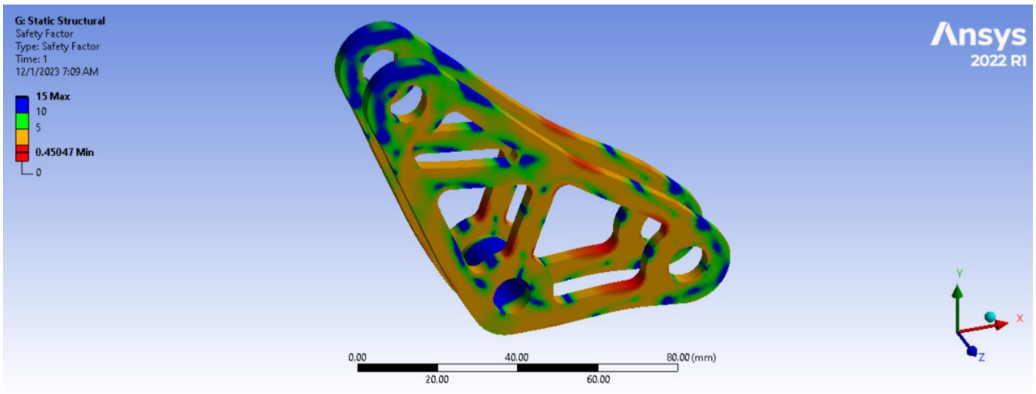


Figure 7.12: Front Bellcrank

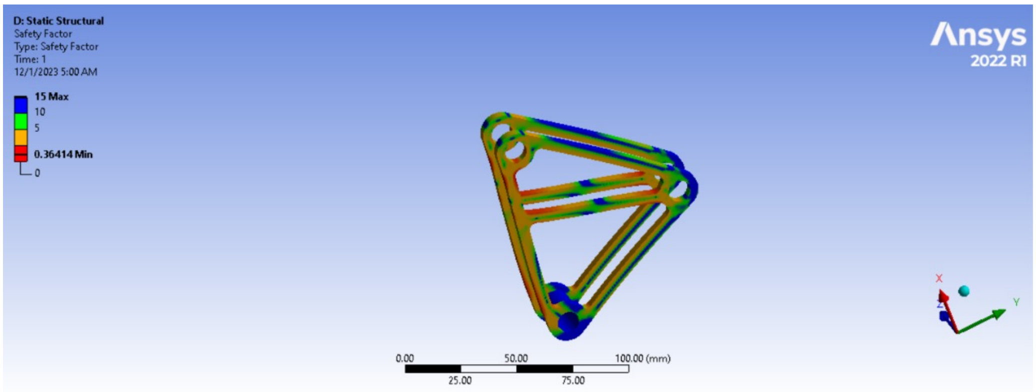


Figure 7.13: Rear Bellcrank

7.4 Steering

Relevent codes for calculation of steering geometry.

7.4.1 Steeering Torque

Steering torque was determined by assessing the force transmitted to the rack through the knuckle and tie rod, while also considering the self-balancing torque. The calculated steering torque was rigorously managed to remain below 10 N-m, especially during maximum cornering conditions. Compromises between Steering Ratio and C-factor values were made to reduce steering torque along with ensuring sufficient driver feedback would be received. Codes for steering torque calculation

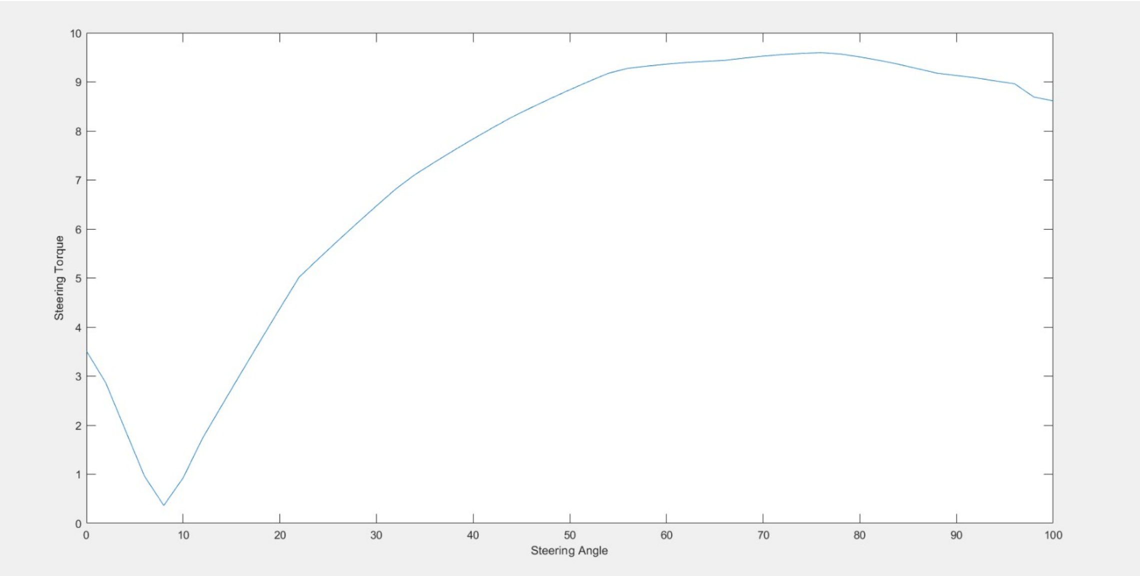


Figure 7.14: Steering torque vs angle

7.4.2 Assembly Components

Steering Wheel: The steering wheel is used to produce the turning effect. A custom steering wheel is used to reduce weight. Material used is Carbon fibre.

Quick Release: The steering wheel is affixed to the steering column via a quick release. The quick release is used for ease of removing steering wheel whenever required.

Steering Columns:

- The tubes used in the steering column are made of EL-2.
- All three steering columns has outer diameter of 21 mm and Inner diameter of 16 mm.
- The tubes sufficiently can propagate a torque of 100 N-m through. (The maximum torque required is 9.1017 N-m)

Universal Joint: Keeping in mind the comfort of the driver and the position of the rack with respect to the chassis 2 U-joint are used.

Needle bearing: To ensure that the steering column assembly stays in place needle bearings are used.They are used due to their following advantages:

- Higher load capacity than single-row ball or roller bearings of comparable OD. The ability to handle a larger, more rigid shaft in a given application.
- Excellent rolling characteristics within a small cross section.

Clamp Housing: Clamps made of mild steel were used to hold the bush and steering column. These were used in place of common cylindrical housing to provide better serviceability or easy removal of components.

- The housing is used to align the steering column with the rack; so that the rotation motion of the steering wheel is transferred perfectly.

- It is installed with a Delrin bush to avoid metal-metal contact.

Bush: A bush made of Delrin is fitted between the clamps and the lower steering column to minimize friction during column rotation.

Rack and Pinion: A rack and pinion system is employed as it is easily available, cheap and simple in its make, has a long life and does not yield compliance.

- Rack is a toothed bar contained in a metal housing. It is similar to the parallelogram center link in that its sideways movement in the housing is what pulls or pushes the tie-rods to change wheel directions
- Pinion is a toothed or worm gear mounted at the base of the steering column assembly where it is moved by steering wheel. The pinion gear meshes with the teeth in the rack so that the rack is propelled sideways in response to the turning of the pinion.
- Module is 2
  Diameter is 30 mm
  Number of teeth is 15

Steering Arm: The steering arm is a part of the knuckle and thus is almost compliance free. The two major parameters optimized above are defined by the steering arm.

7.4.3 Why not one universal joint?

Considering the need for the optimum angle between the upper steering column and the steering pinion axis, if one universal joint is preferred then the height difference between the universal joint and the center of steering wheel is coming out to be 19.25 mm which is very less and the torque transfer will not be proper in the universal joint. So, we have preferred using 2 universal joints. The scenario with using one universal joint is indicated in Fig. **??**

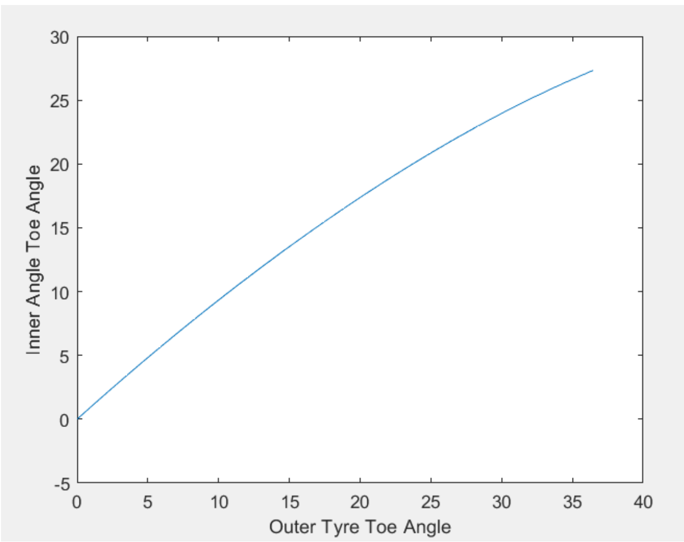


Figure 7.15: Inner Wheel Angle V/s Outer Wheel Angle

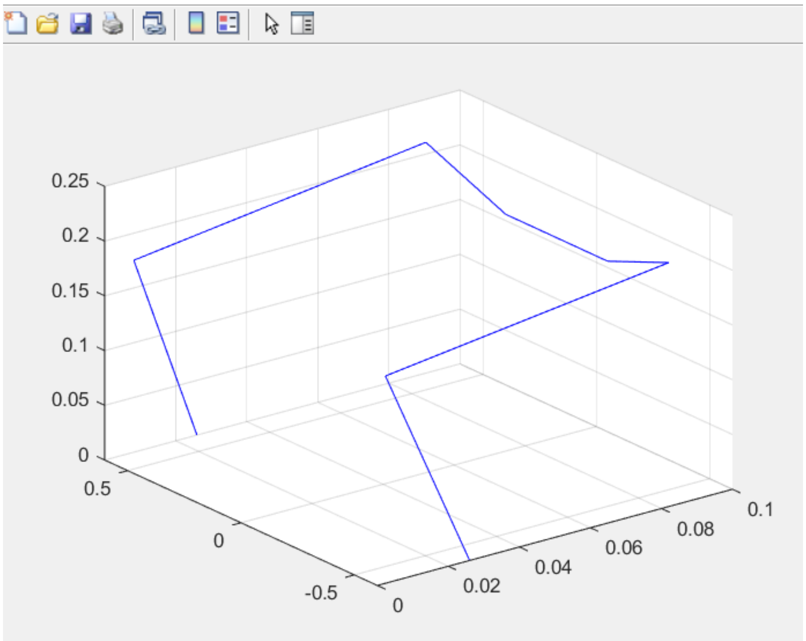


Figure 7.16: Steering Geometry modeled on Matlab