ENGINEERING DESIGN PRESENTATION

Vehicle Dynamics

GOALS:

- Maximize grip
  - Control camber change during cornering
  - Getting max output from aero devices in terms of downforce
- Stability
  - Minimize body roll
  - Optimum Ride frequency
  - Optimum Damping ratio.

Trackwidth and Wheelbase Selection

Tilt Test Criteria → Set track width to 1200 mm based on 300 mm C.G. height estimate for a 300 kg vehicle (with driver).

Wheelbase Setup →
-> Minimum Wheelbase: - 1525 mm
-> Wheelbase was selected after carefully fitting all the major components.

Parameter	Wheelbase	Front trackwidth	Rear Trackwidth
Value	1590 mm	1200 mm	1250 mm

Suspension System Overview

Front Suspension Package

- Double wishbone style control arms, unequal
- Pushrod direct actuation coil overs
- Anti-roll bar (ARB)
- Springs
- Dampers
- Rockers
- Steering tie rods

Rear Suspension Package

- Double wishbone style control arms, unequal
- Pushrod direct actuation coil overs
- Anti-roll bar (ARB)
- Springs
- Dampers
- Rockers

2D sketch of Front Suspension Hard Points

Front Wishbone 2d image

Parameter	Value
FVSA	1670.43
Roll Centre height	27.19
SVSA	970
Scrub Radius	30
Caster Angle	3.8
Kingpin Angle	7

Front Wheels

- The idea of a lower kingpin angle, stems from the fact that the higher the angle, the greater the jacking effect.
- SVSA selection was done such that the suspension provide optimum anti-dive percentage

Design Considerations

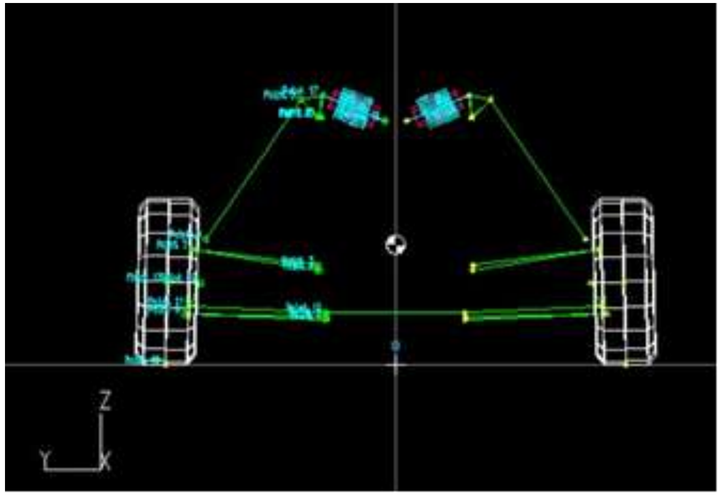
- Wheel offset is kept (+15mm) considering packaging constraints of the brake disc .
- Caster angle assumption was based on lower mechanical trail to reduce steering torque

Rear Wishbone 2D image

Parameter	Value
FVSA	1736.69
Roll Centre height	58.7
SVSA	1450
Scrub Radius	25
Caster Angle	-5.5
Kingpin Angle	7

Kinematic Simulation in Lotus

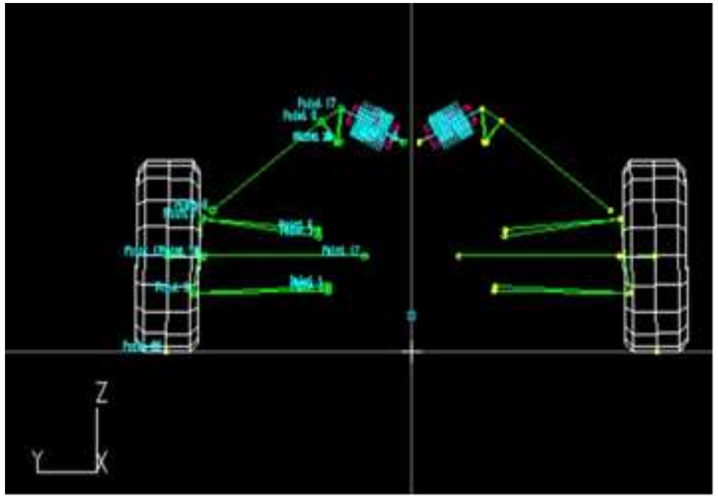
Iterative simulations in Lotus Shark provided optimal suspension hard points



Front Pushrod Suspension Lotus Setup

Point	x-coordinate	y-coordinate	z-coordinate
Tie rod inboard	80	90	142.75
LCAF inboard	140	230	125
LCAR inboard	-130	230.76	133.51
UCAF inboard	140	243	270
UCAR inboard	-130	240.6	243.24
Push rod inboard	-5.93	292.65	657.8
Tie rod outboard	70.65	565	161
LCA outboard	5.93	577.9	139.27
UCA outboard	-5.93	555.96	317.93
Pushrod outboard	-5.93	515.96	357.93

Final coordinates of Front Suspension System

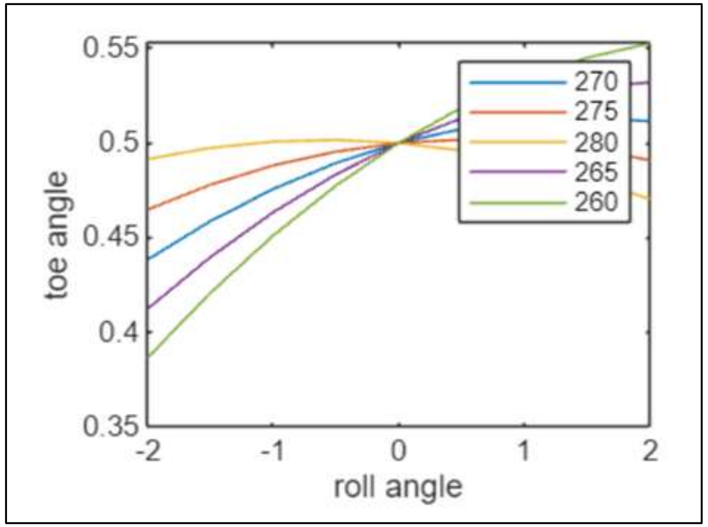


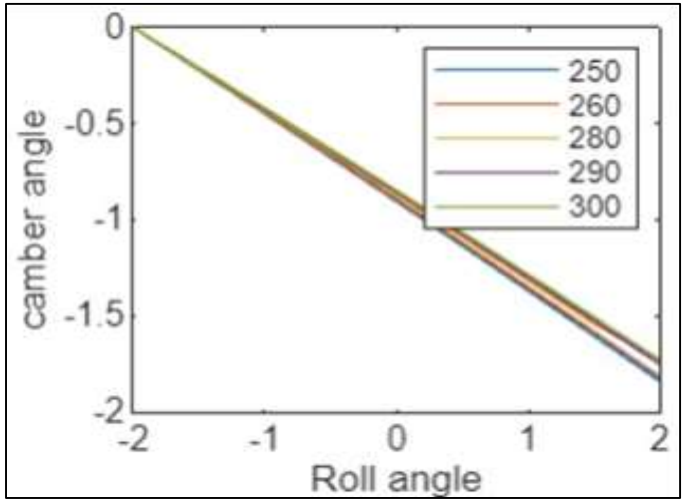
Rear Pushrod Suspension Lotus Setup

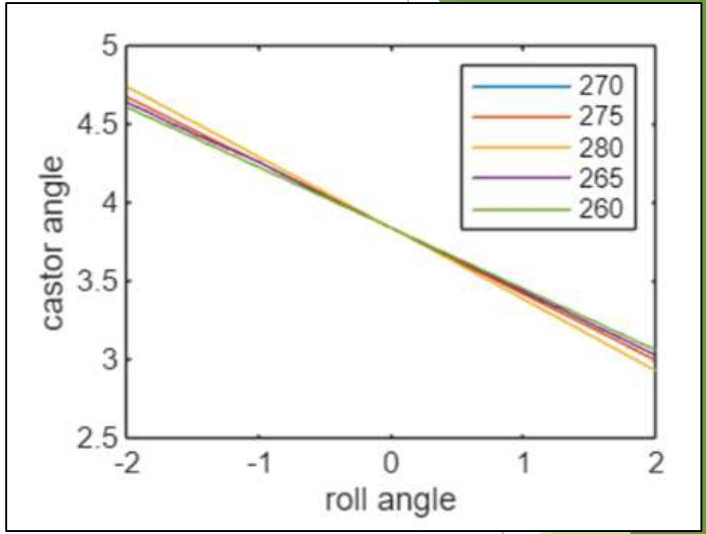
Point	x-coordinate	y-coordinate	z-coordinate
LCAF inboard	130	248.14	155.70
LCAR inboard	-140	246	145
UCAF inboard	130	273.05	280.23
UCAR inboard	-140	276	295
Push rod inboard	-5.93	331.53	529.24
LCA outboard	-8.6	557.9	139.27
UCA outboard	8.6	535.96	317.93
Pushrod outboard	-5.93	495.96	357.93

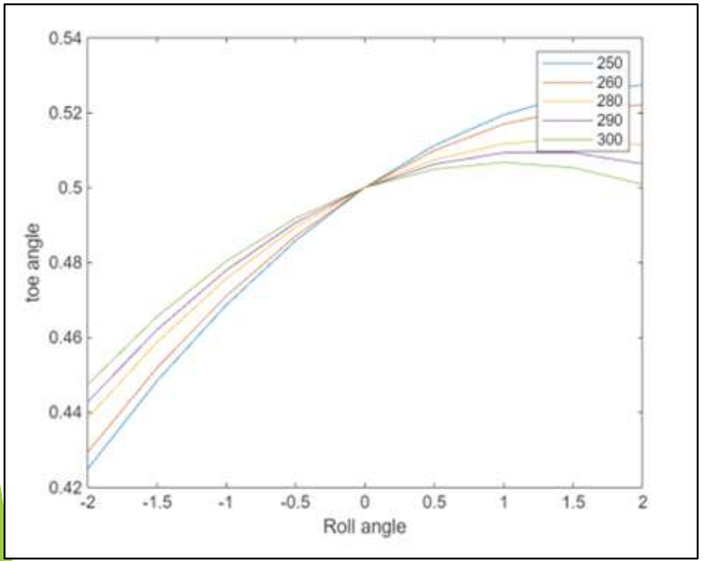
Final coordinates of Rear Suspension System

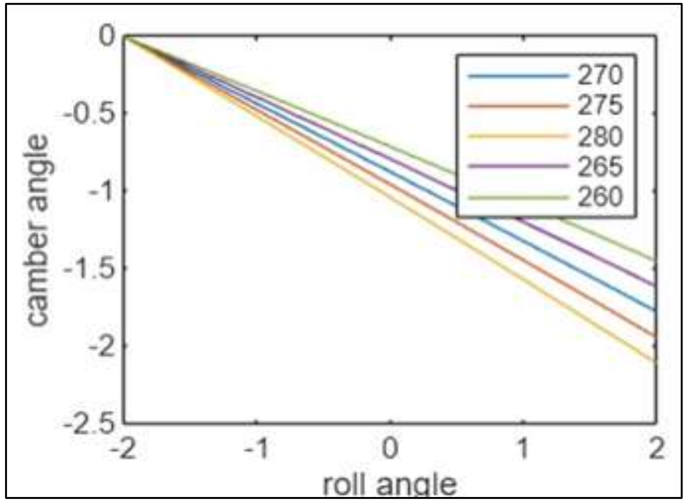
Change in Camber, Caster and Kpi angle for different Roll Angle

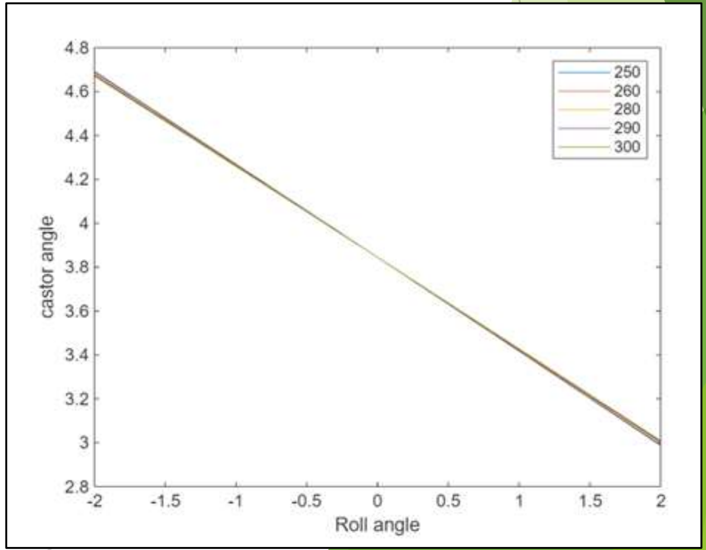




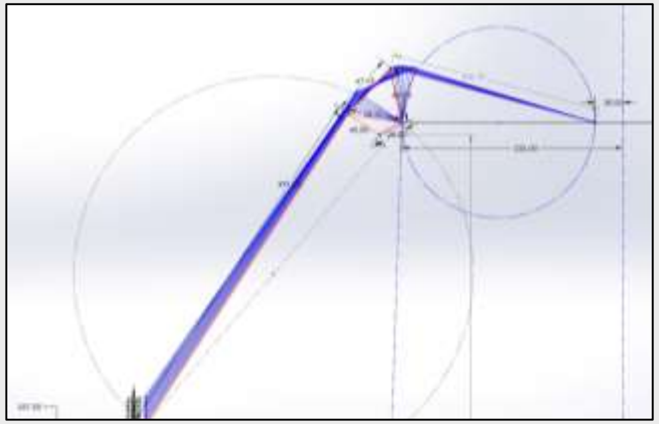


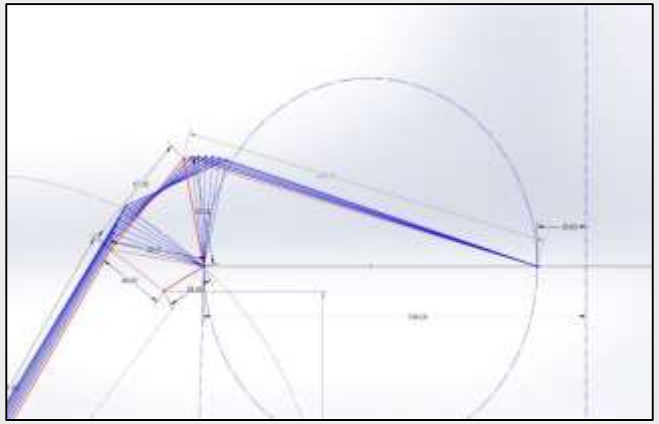


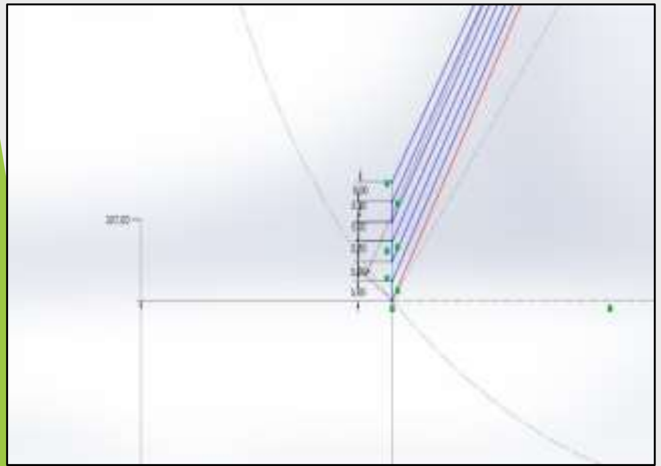




Rocker Motion Ratio







Sketch Considerations:

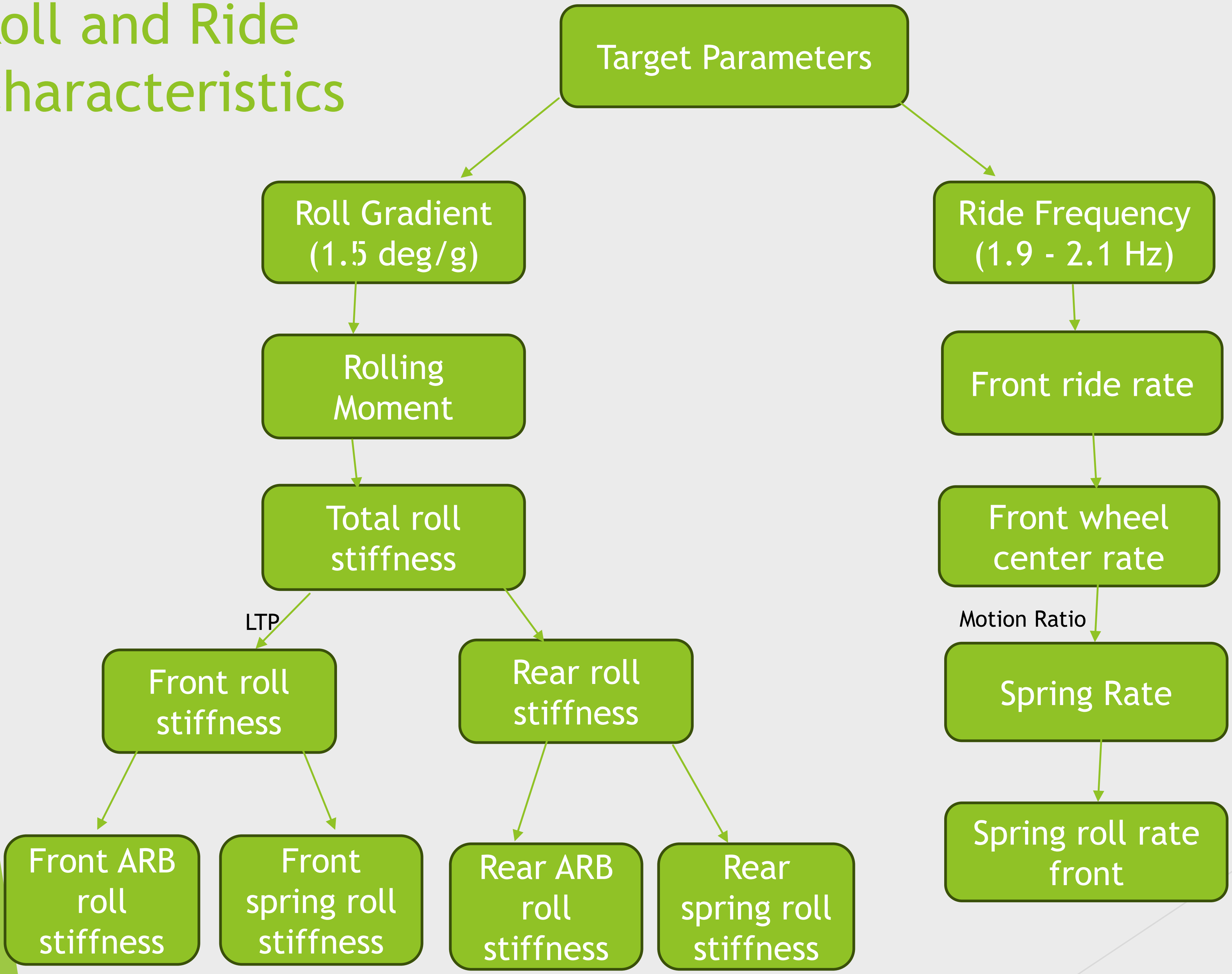
Damper Setup:

- 200mm at static position.
- 187mm at max bump.
- Damper has a stroke 70 mm.

Geometry Optimization:

- **90-degree angle** between push rod and rocker arm at max jounce (i.e 30 mm) for optimum force transmission)
- **90-degree angle** between damper and rocker arm at max jounce.
- Push rod upright point, push rod rocker end, rocker and damper are in same plane makes all the forces form tyre to travel to spring along pushrod. And reduces generation of unnecessary force in perpendicular directions.

Roll and Ride Characteristics



Parameters	Value	Unit
Total Roll Moment	779.8	Nm/g
Total Roll Stiffness	29786	Nm/rad
Front Frequency	2.173	Hz
Rear Frequency	1.954	Hz
Front Ride Rate	11061	N/m
Rear Ride Rate	11983	N/m
Front Wheel Rate	12083	N/m
Rear Wheel Rate	13192	N/m
Front Spring Rate	17570	N/m
Rear Spring Rate	28032	N/m
Front Spring Roll Stiffness	8641	Nm/rad
Rear Spring Roll Stiffness	8628	Nm/rad
Front ARB Stiffness	912	Nm/rad
Rear ARB Stiffness	1904	Nm/rad

ARB Design:

Calculation of front ARB diameter

- $k_{ARB} = \frac{k_{vehicle,f}}{(MR_{roll})^2} = 29786.34\ (Nm)/(rad)$
- $I = \frac{k_{ARB} * L}{G} = 604.89 * (10^{-9})\ (m^\wedge 4)$
- $d = \sqrt[4]{\frac{32 * I}{\pi}} = 20.84mm$

Shear stress calculation

- $\tau_{max} = \left(\frac{G*\theta}{l}\right) * r = 51.33\ MPa$
- $\tau_{mat=}\sigma_y/\sqrt{3}$ = 503/$\sqrt{3}$= 290.407 MPa
- FoS= $\frac{\tau_{mat}}{\tau_{max}}$= 5.66

$$Arb_{MR} = \frac{ARB\ Twist\ Angle}{Roll\ Angle}$$

ARB Motion Ratio	Arm Length	ARB Angle
2.49	141.48	4.78
3.526	100	6.7709
4.98	70.91	9.57

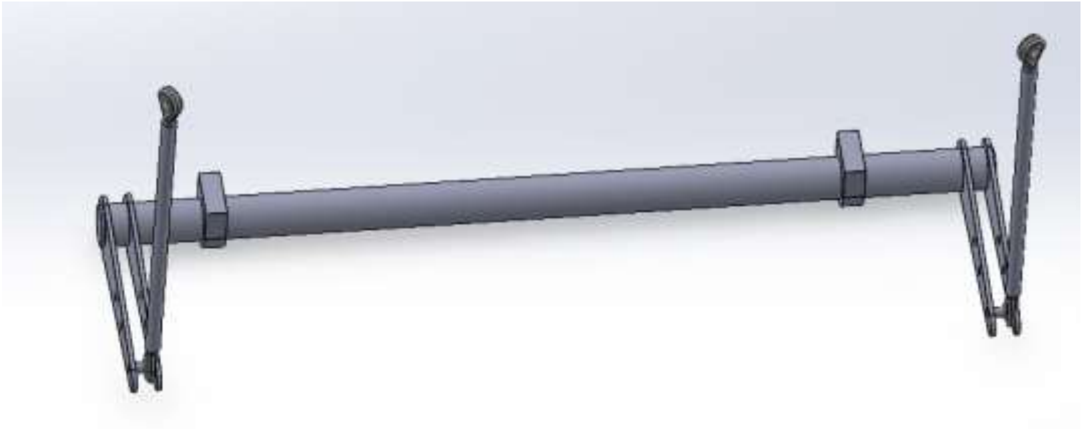
Calculation of rear ARB diameter

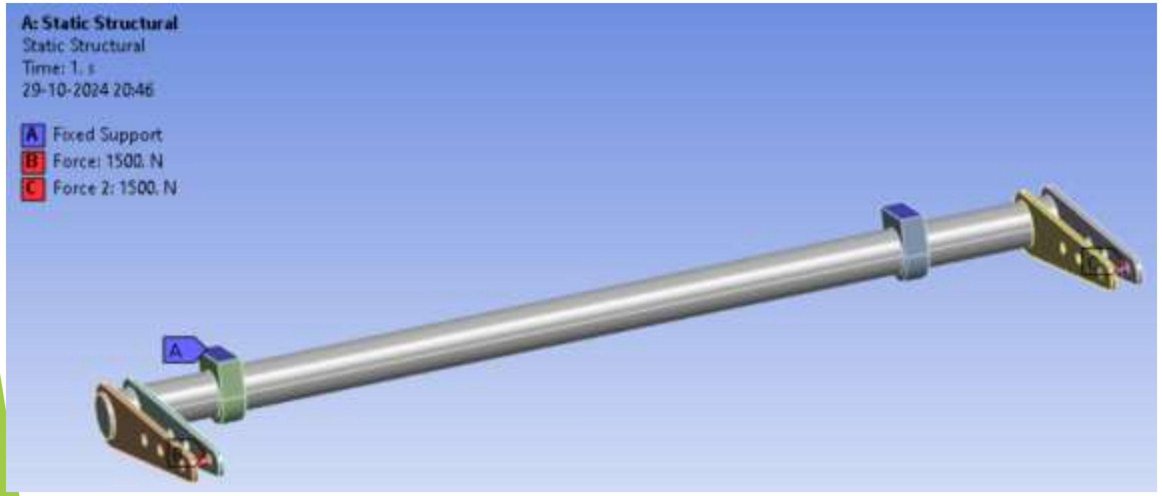
- $k_{ARB} = \frac{k_{vehicle,f}}{(MR_{roll})^2} = 29786.34\ (Nm)/(rad)$
- $I = \frac{k_{ARB} * L}{G} = 618.64 * (10^9)$
- $d = \sqrt[4]{\frac{32 * I}{\pi}} = 25.2mm$

ARB Motion Ratio	Arm Length	ARB Angle
3.79	60.34	7.29
2.684	85.28	5.153
1.898	120.52	3.644

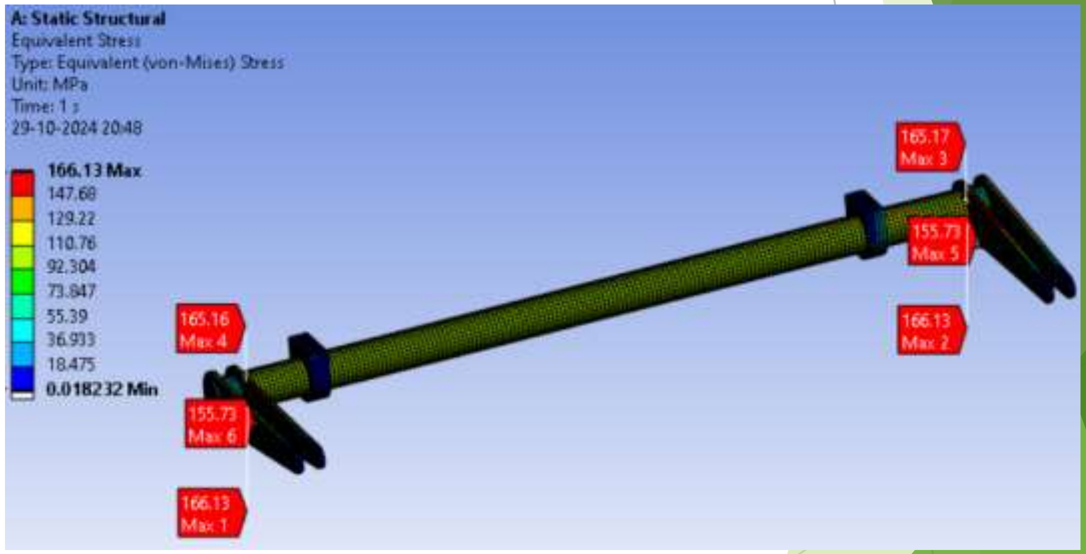
Shear stress calculation

- $\tau_{max} = \left(\frac{G*\theta}{l}\right) * r = 50.21\ MPa$
- $\tau_{mat=}\sigma_y/\sqrt{3}$ = 503/$\sqrt{3}$= 290.407 MPa
- FoS= $\frac{\tau_{mat}}{\tau_{max}}$= 5.784





Forces Applied



Stress generated

Force calculation on tire contact patch

Linear acceleration

Longitudinal force on one tire

$$F_{xf} = \frac{\frac{\mu * W}{l} * (l_2 + f_r * h)}{2 * \left(1 + \frac{\mu * h}{l}\right)}$$

$$F_{xr} = \frac{\frac{\mu * W}{l} * (l_1 - f_r * h)}{2 * \left(1 - \frac{\mu * h}{l}\right)}$$

Lateral Force = 0

Vertical force on one tire

$$F_{zf} = \left(\frac{W}{2}\right) * \frac{\frac{l_2}{l} - a_x * h}{l * g}$$

$$F_{zr} = \left(\frac{W}{2}\right) * \frac{\frac{l_1}{l} + a_x * h}{l * g}$$

Linear braking

Longitudinal force on one tire

$$F_{xf} = \frac{\mu}{2} * \left(W_{fs} + \frac{W * D_x * h}{l}\right)$$

$$F_{xr} = \frac{\mu}{2} * \left(W_{fs} - \frac{W * D_x * h}{l}\right)$$

Lateral Force = 0

Vertical force on one tire

$$F_{zf} = \frac{1}{2}(W_{fs} + \frac{W * D_x * h}{l})$$

$$F_{zr} = \frac{1}{2}(W_{fs} - \frac{W * D_x * h}{l})$$

Cornering

Longitudinal Force = 0

Lateral force on one tire

$$F_y = a_y * F_z$$

Vertical force on one tire

$$F_z = \pm \frac{M_\emptyset}{t_f} + \frac{W_{fs}}{2}$$

Braking + Cornering

Longitudinal force on one tire

$$F_{xf} = \frac{\mu}{2} * \left(W_{fs} + \frac{W * D_x * h}{l}\right)$$

$$F_{xr} = \frac{\mu}{2} * \left(W_{fs} - \frac{W * D_x * h}{l}\right)$$

Lateral force on one tire

$$F_y = a_y * F_z$$

Vertical force on one tire

$$F_z = (\frac{1}{2})(W_{fs} \pm \frac{W * D_x * h}{l} \pm \frac{M_\emptyset}{t_f})$$

Calculation of Suspension Forces

- To find the forces on the 6-suspension links we end up with 6 equations in 6 variables as follows:
- $\Sigma F_x = 0 = F_{TR}u_{TR_x} + F_{LCAF}u_{LCAF_x} + F_{LCAR}u_{LCAR_x} + F_{UCAF}u_{UCAF_x} + F_{UCAR}u_{UCAR_x} + F_{PR}u_{PR_x} + F_x$
- $\Sigma F_y = 0 = F_{TR}u_{TR_y} + F_{LCAF}u_{LCAF_y} + F_{LCAR}u_{LCAR_y} + F_{UCAF}u_{UCAF_y} + F_{UCAR}u_{UCAR_y} + F_{PR}u_{PR_y} + F_y$
- $\Sigma F_z = 0 = F_{TR}u_{TR_z} + F_{LCAF}u_{LCAF_z} + F_{LCAR}u_{LCAR_z} + F_{UCAF}u_{UCAF_z} + F_{UCAR}u_{UCAR_z} + F_{PR}u_{PR_z} + F_z$
- $\Sigma M_x = 0 = F_{TR}\left(n_z r_y - n_y r_z\right)_{TR} + F_{LCAF}\left(n_z r_y - n_y r_z\right)_{LCAF} + F_{LCAR}\left(n_z r_y - n_y r_z\right)_{LCAR} + F_{UCAF}\left(n_z r_y - n_y r_z\right)_{UCAF} + F_{UCAR}(n_z r_y -$

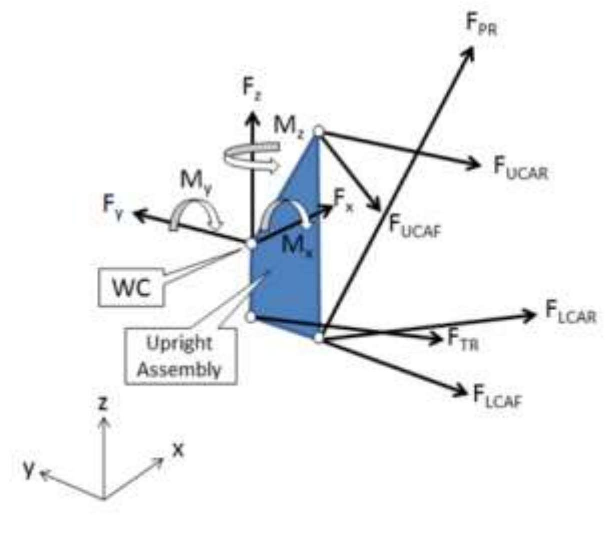


Figure 1.3 – FBD of the upright for the right-hand FR suspension.

To solve this, we use the matrix method.

The basic format would be A {x} = B.

$A =$

$$\begin{bmatrix} u_{TR_x} & u_{LCAF_x} & u_{LCAR_x} & u_{UCAF_x} & u_{UCAR_x} & u_{PR_x} \\ u_{TR_y} & u_{LCAF_y} & u_{LCAR_y} & u_{UCAF_y} & u_{UCAR_y} & u_{PR_y} \\ u_{TR_z} & u_{LCAF_z} & u_{LCAR_z} & u_{UCAF_z} & u_{UCAR_z} & u_{PR_z} \\ (n_z r_y - n_y r_z)_{TR} & (n_z r_y - n_y r_z)_{LCAF} & (n_z r_y - n_y r_z)_{LCAR} & (n_z r_y - n_y r_z)_{UCAF} & (n_z r_y - n_y r_z)_{UCAR} & (n_z r_y - n_y r_z)_{PR} \\ (n_z r_x - n_x r_z)_{TR} & (n_z r_x - n_x r_z)_{LCAF} & (n_z r_x - n_x r_z)_{LCAR} & (n_z r_x - n_x r_z)_{UCAF} & (n_z r_x - n_x r_z)_{UCAR} & (n_z r_x - n_x r_z)_{PR} \\ (n_y r_x - n_x r_y)_{TR} & (n_y r_x - n_x r_y)_{LCAF} & (n_y r_x - n_x r_y)_{LCAR} & (n_y r_x - n_x r_y)_{UCAF} & (n_y r_x - n_x r_y)_{UCAR} & (n_y r_x - n_x r_y)_{PR} \end{bmatrix}$$

$$\{x\} = \begin{bmatrix} F_{TR} \\ F_{LCAF} \\ F_{LCAR} \\ F_{UCAF} \\ F_{UCAR} \\ F_{PR} \end{bmatrix} \qquad B = \begin{bmatrix} F_x \\ F_y \\ F_z \\ M_x \\ M_y \\ M_z \end{bmatrix}$$

Results of Tire Forces at Contact Patch on each tire

	Fx	Fy	Fz
Linear Acceleration	Front - 718.884 N Rear - 1221.501 N	0	Front - 689.325 N Rear - 880.274 N
Braking	Front - 1180.494 N Rear - 923.427 N	0	Front - 880.965 N Rear - 689.124
Cornering	0	Front inner - 683.981 N Front outer - 800.859 N Rear inner - 906.673 N Rear outer - 1061.605 N	Front inner - 621.801 N Front outer - 728.054 N Rear inner - 824.248 N Rear outer - 965.095 N
Braking+Cornering	Front inner - 655.037 N Front outer - 2050.709 N Rear inner - 173.990 N Rear outer - 1676.085 N	Front inner - 537.717 N Front outer - 1683.418 N Rear inner - 142.827 N Rear outer - 1375.891 N	Front inner - 488.834 N Front outer - 1530.380 N Rear inner - 129.843 N Rear outer - 1250.810 N

Results of Forces on Suspension Links

Tie rod	859.907 N
Lower Control Arm Front	2054.891 N
Lower Control Arm rear	6889.856 N
Upper Control Arm front	1974.734 N
Upper Control Arm rear	2167.734 N
Pushrod	3252.966

Wishbone OD,ID, Thickness Calculation

By using the given formula for our maximum load, we are able to calculate the diameters of the wishbones. The material used is AISI4130 due to its high strength and good weldibility.

Maximum Buckling Load

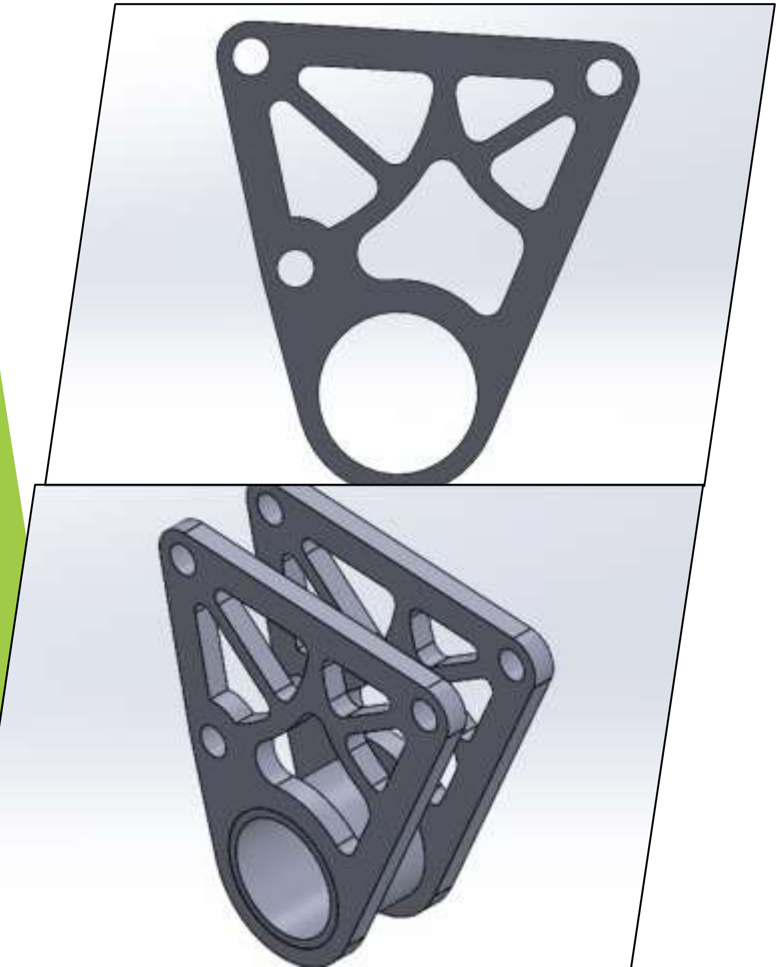
$$P_{cr} = \frac{\pi^2 * E * I}{(k * l)^2}$$

Maximum load from yield strength

$$F_{max} = \sigma_{yleld} * A$$

ID	OD	t
8mm	13mm	2.5mm

Rocker 3D Design



Key Design Features:

Push Rod and Rocker Mechanism:

- Push rod connected to the rocker; damper mounted between rocker and rear frame.
- Rocker rotates on a bearing holder on the bulkhead.
- Perpendicular force at max bump ensures damper efficiency and minimizes lateral forces.

Optimization:

- **90-degree angle** between push rod and rocker arm at max bump for optimum force transmission.
- Minimized bending moments in rocker arm, reducing stress and risk of deformation.

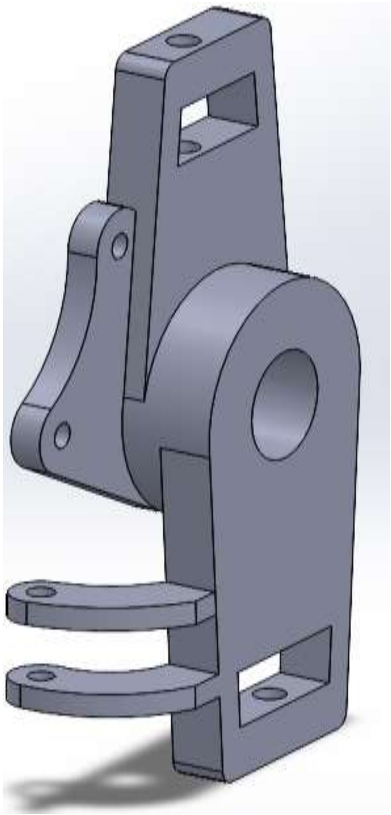
Damper Setup:

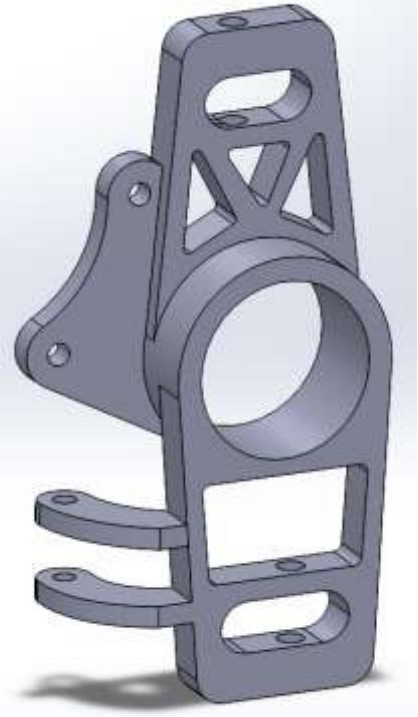
- 200mm at static position.
- 187mm at max bump.

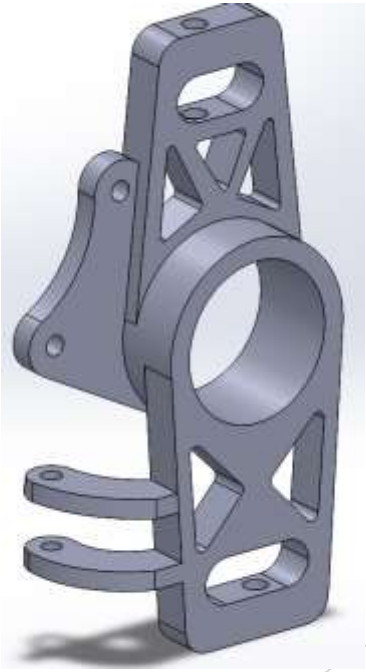
Upright Design

Constraints:

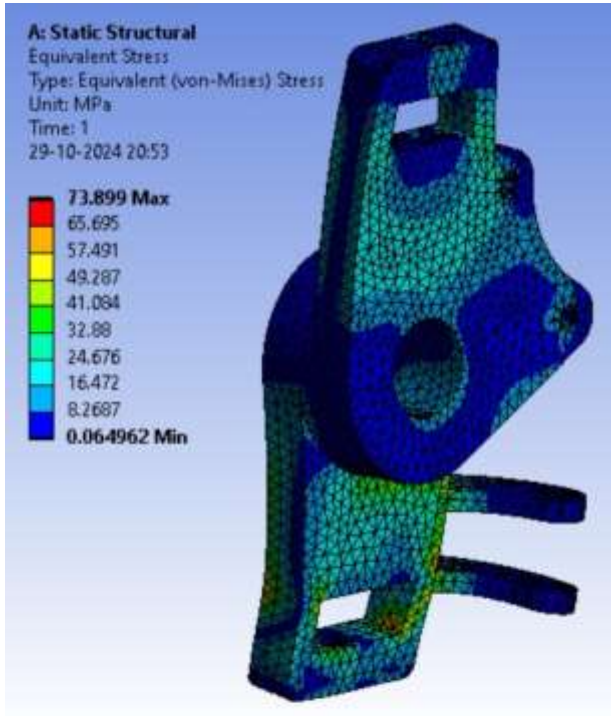
- Upper and Lower Ball Joint Points
- Tie Rod and Brake Caliper Points



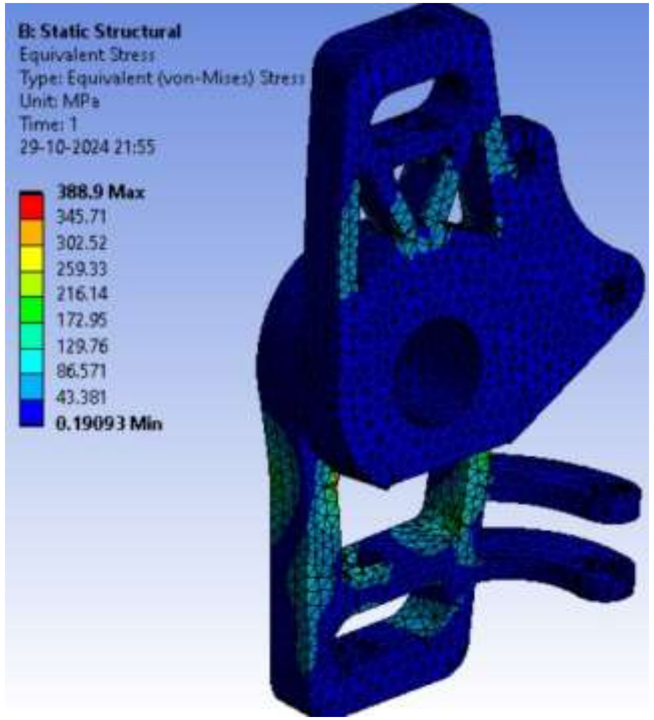




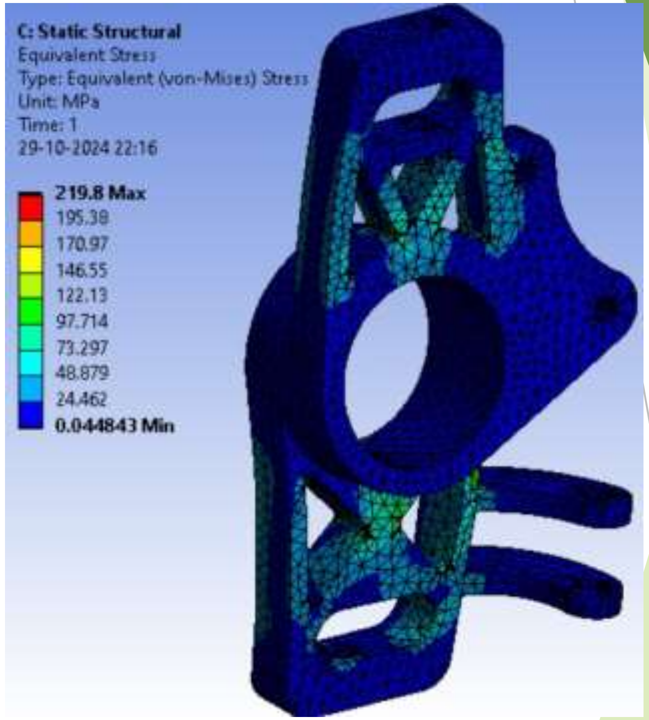
Upright Design Simulation



1st iteration



2nd iteration



3rd iteration

FOS = Yield strength/Allowable stress
= 270 MPa/219.8Mpa
=1.22

Groundforce aero:

- Goals

To increase the efficiency of the vehicle

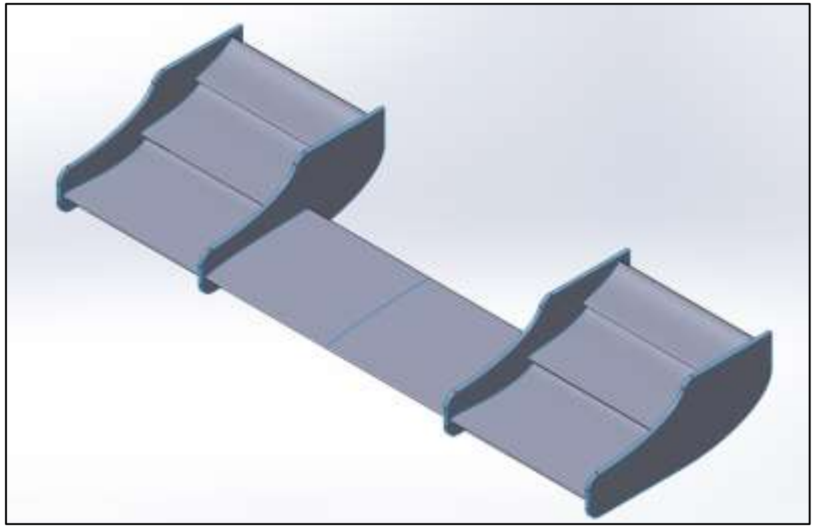
By decreasing the lap time

Increasing the downforce for better grip at corners

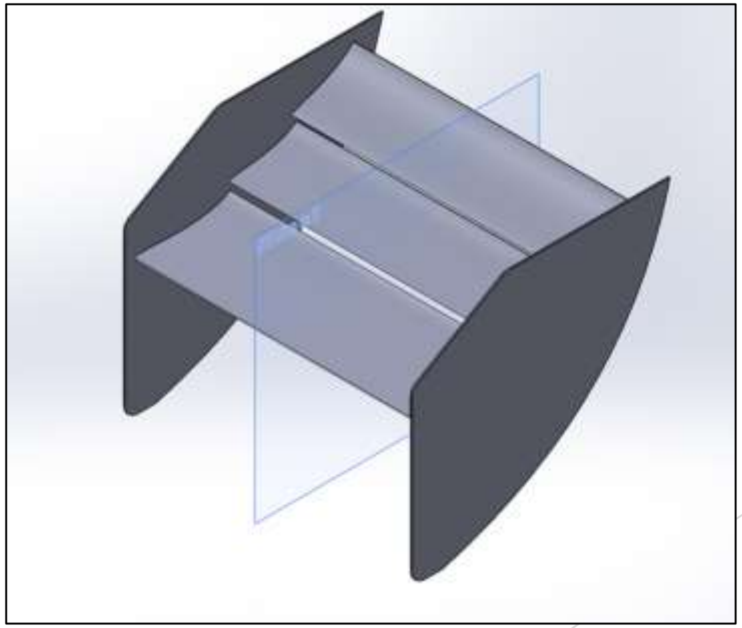
By decreasing the drag produced by the vehicle

Components:

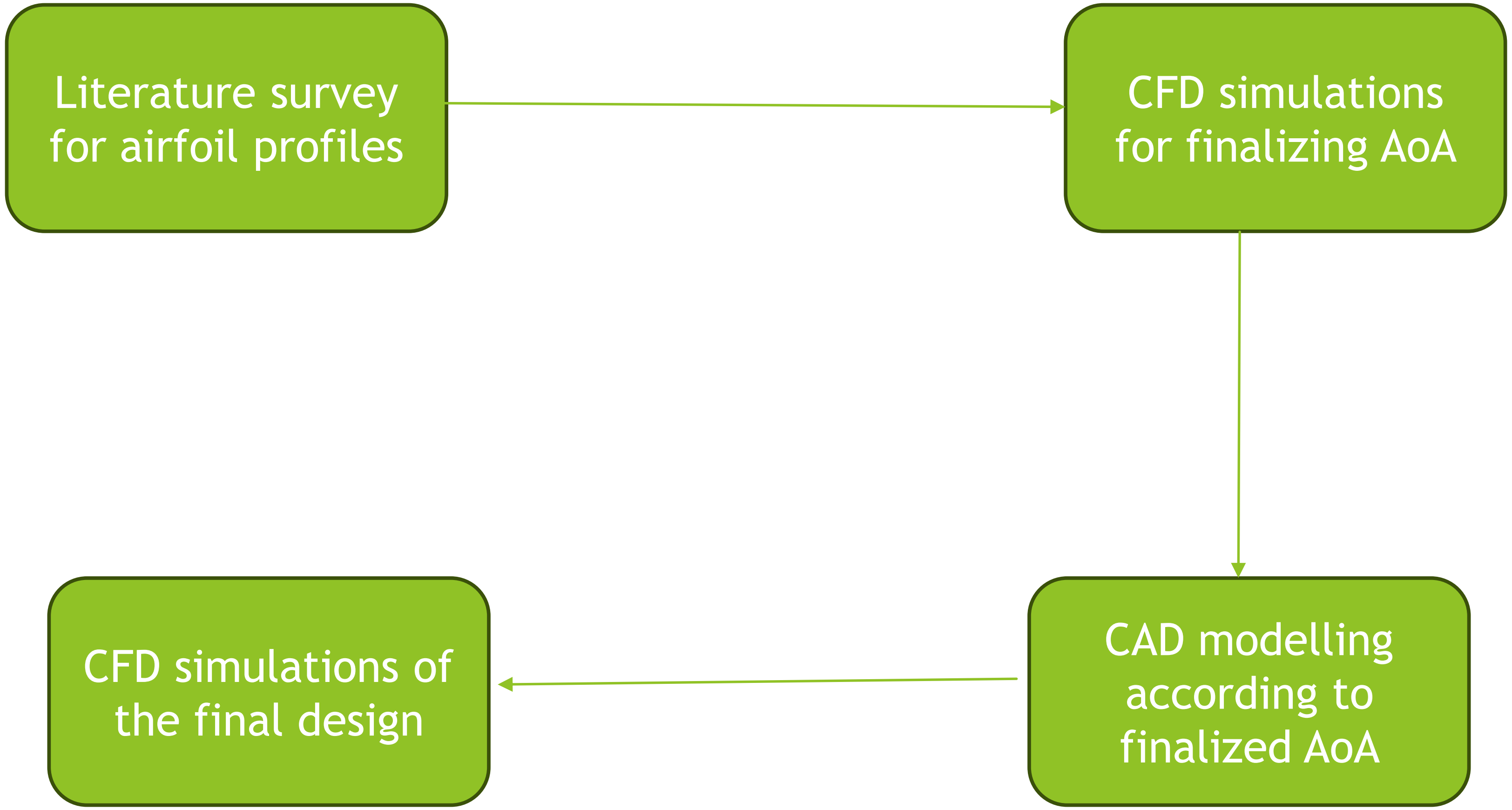
Front wing

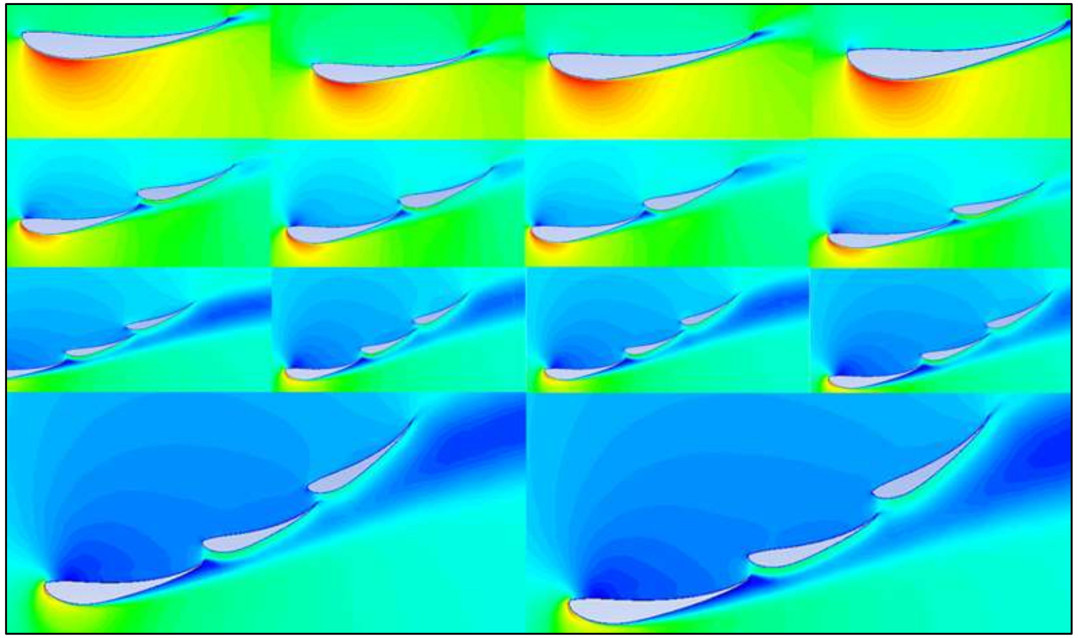


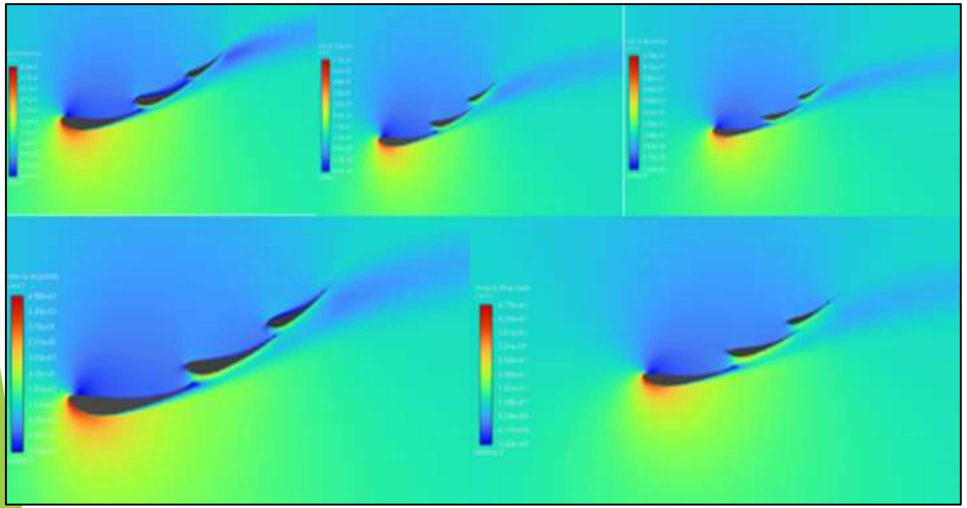
Rear wing

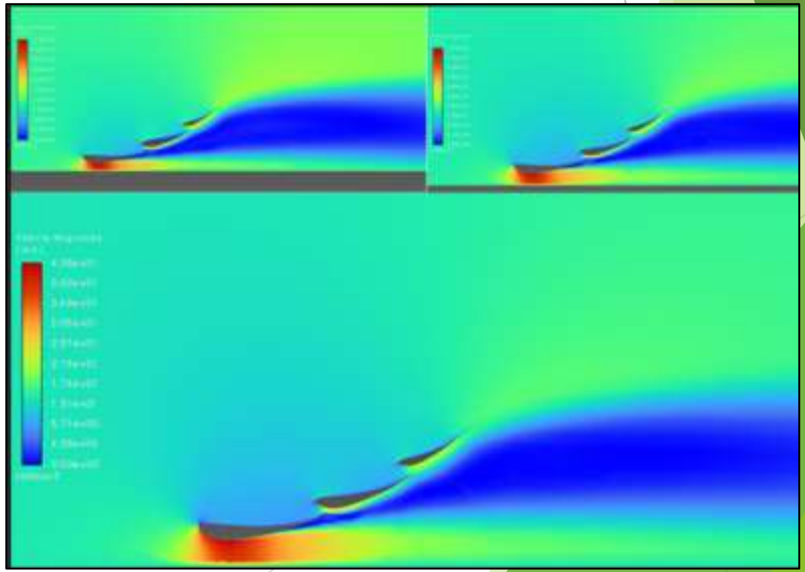


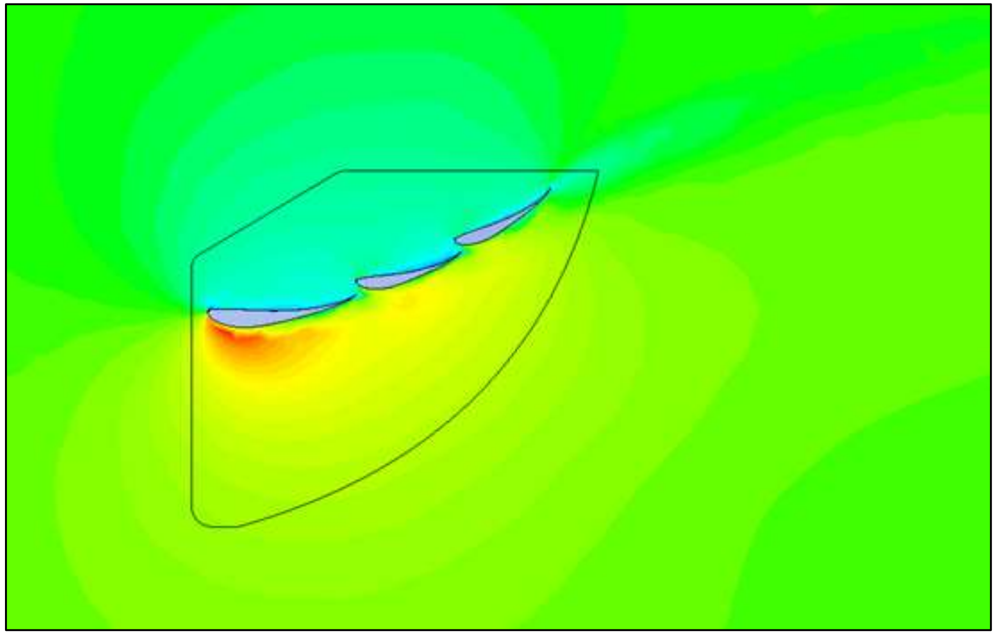
Tools:


Literature survey for airfoil profiles
CFD simulations for finalizing AoA
CAD modelling according to finalized AoA
CFD simulations of the final design


Velocity field of different AoA – Rear wing


Velocity field of s1223 tested for various gaps between the flaps – Front wing

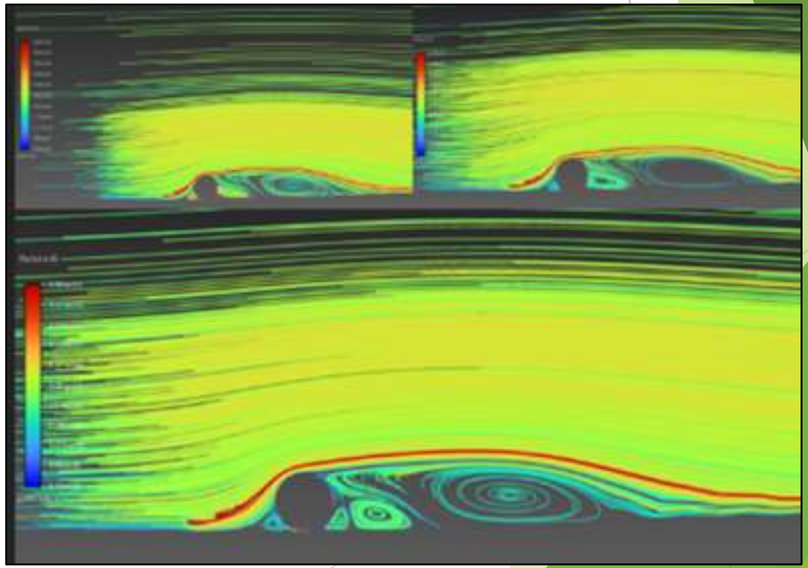

Velocity field of s1223 tested for various ground clearances



Velocity field past the rear wing – final model

Vehicle speed – 15m/s

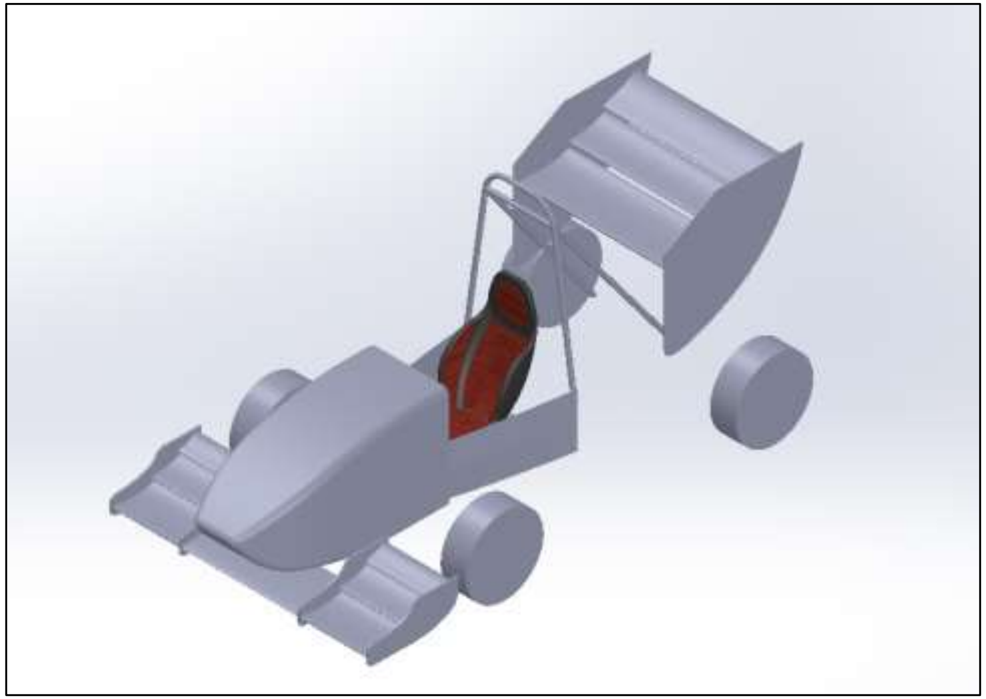
Front wing was tested for different ground clearances, varying distance from tyres, various gaps and overlaps between the flaps for efficient wing.



Pathlines of flow past front wing – final design

Endplates of both rear wing and front wing were simulated with different designs to get the optimal one, main concept is to give sufficient space for the low-pressure region to develop, otherwise there will be pressure loss from the sides of the plates and to reduce the weight of plates.

Rear wing	Downforce : 198.38N Drag : 46.934N	Cl : 1.88 Cd : 0.44 Efficiency : 4.27
Front wing	Downforce : 178N Drag : 30N	Cl : 1.63 Cd : 0.27 Efficiency : 6.037



Full aero package installed into the vehicle

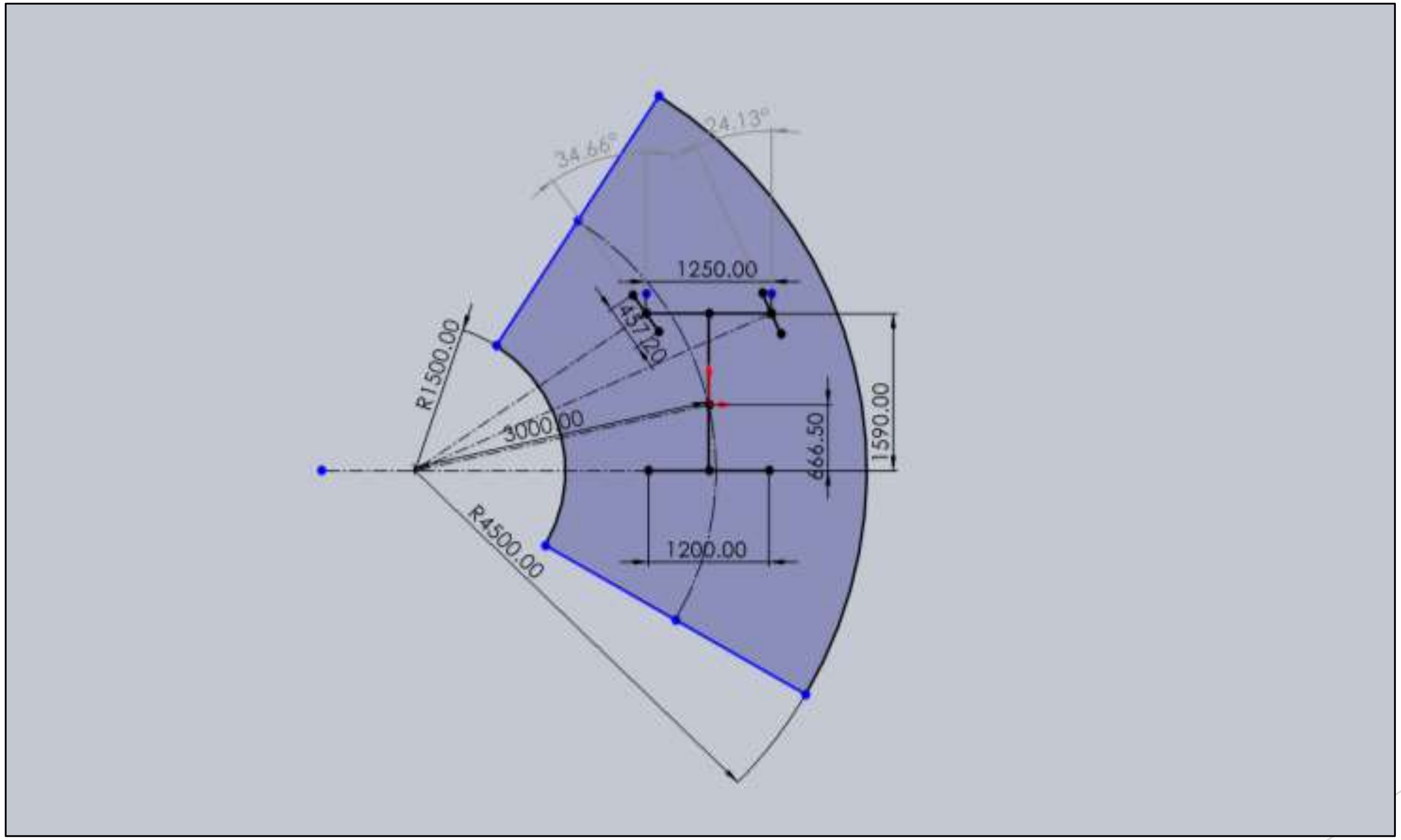
Steering System Overview

- Goals : 1) Steer Ratio - 3.062
  2) C-Factor - 59.064

Sl.No.	Parameters	Values
1	Wheelbase	1590 mm
2	Front Track Width	1250 mm
3	Rear Track Width	1200 mm
4	Tire Width	160 mm
5	Tire Radius	457.20 mm
6	Track Turning Radius	3000 mm

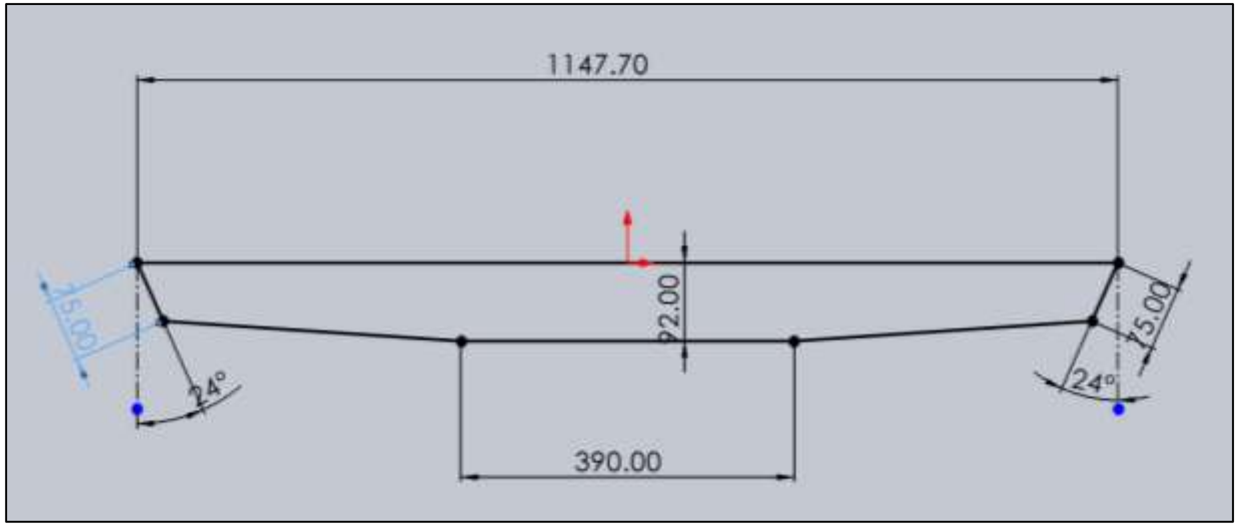
Steering System Design

- Design Considerations : Ackermann Geometry



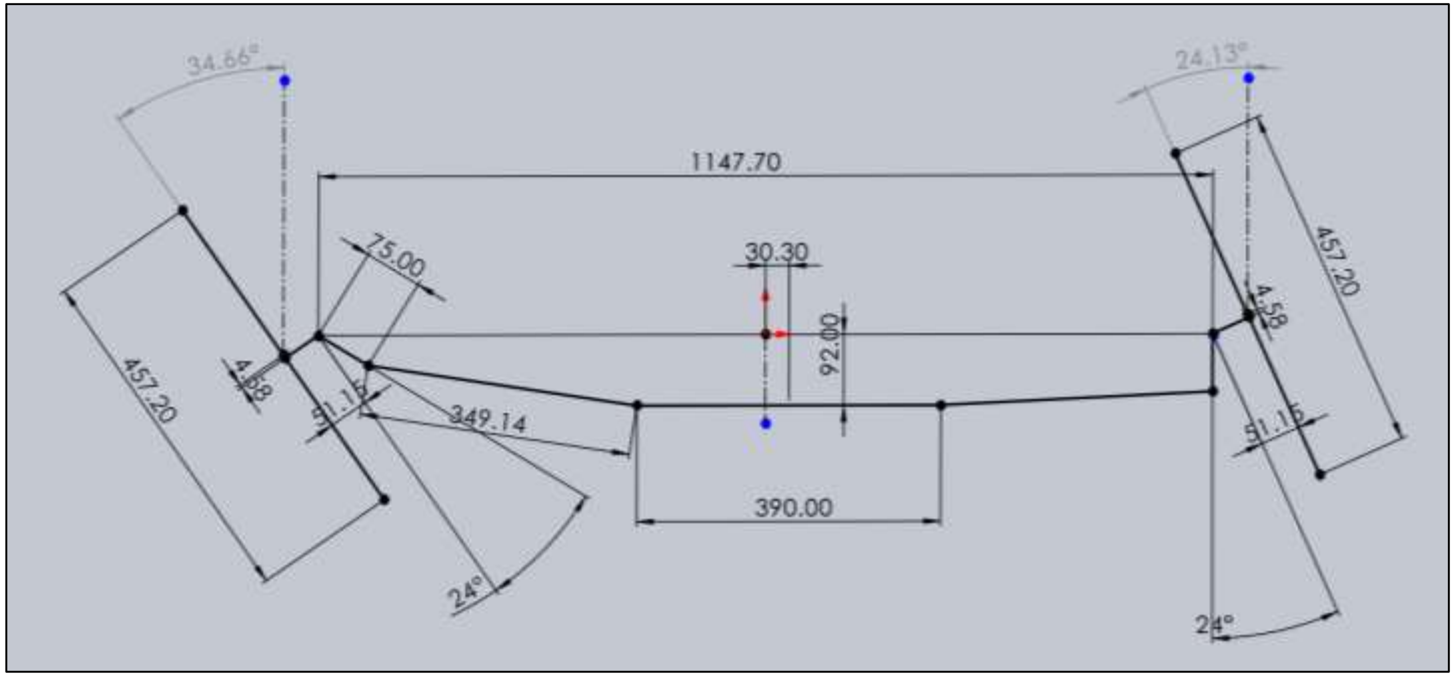
Tie Rod Calculations

- x= steering arm length=75mm
- y= tie-rod length
- z= rack ball joint center to ball joint length =390mm
- d= distance between front axis and rack center axis = 92mm
- B= distance between left and right kingpin centerline = 1147.70
- ß= Ackerman angle = 24 deg



From the above figure, we get
y = (((B − z)/2 − xsinß)^2 + (d − xcosß)^2)^(½)
The tie-rod length came out to be approximately 349.14 mm.

Rack Travel



Rack travel δ for maximum wheel turn angle was found in SolidWorks.
Rack Travel, δ = 30.30mm

Steering Ratio

The steering ratio is calculated as

Steering Ratio = Angle of steering wheel/(Angle of turning on the left + right wheels)

= 3.062

C-factor

C-factor is the linear distance travelled by the rack in one rotation of the pinion.

$C - \text{factor} = \text{Total Rack Travel(in mm)}/\text{Lock} - \text{to} - \text{lock angle(in rad)}$

Where, $\text{Lock} - \text{to} - \text{lock angle} = \theta i + \theta o = 1.026\text{rad}$

Therefore, C-factor = 59.064 mm

Pinion Calculations

It is assumed to get maximum rack travel at 180 steering wheel rotation. The maximum rack travel being 30.30 mm, the pitch diameter of the pinion can be found by the formula of the length of an arc.

The rotation angle of the pinion will not exactly match the steering wheel angle due to play in the universal joints and steering column. So, assuming an efficiency of 90%, we calculate

Arc length is, $s = r\omega$

$$30.30 = r \cdot (0.9) \cdot \left(180\pi/180 \right)$$

Therefore, $r = 10.71\text{mm}$

So, PCD of the pinion is $2r = 21.42 \approx 22\text{mm}$

$\text{Module}(M) = 2\ \text{PCD} = 22\text{mm}$

$\text{No.of teeth on pinion}(Tp) = \text{PCD}/M = 11$

$\text{Circular Pitch} = \pi \cdot M = 6.28\text{mm}$

Rack Calculations

Rack travel for both side = $2 \cdot \delta$ = 60.60mm

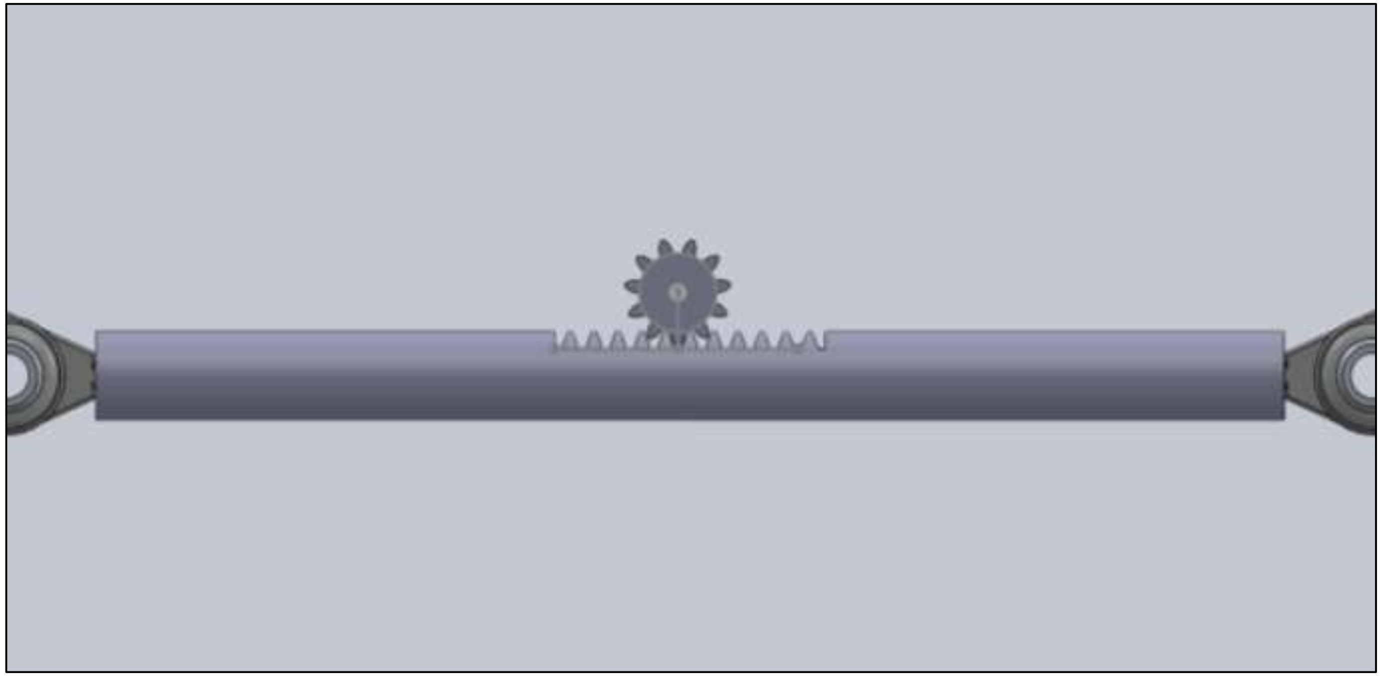
For perfect meshing of rack and pinion, the module should be the same for both.

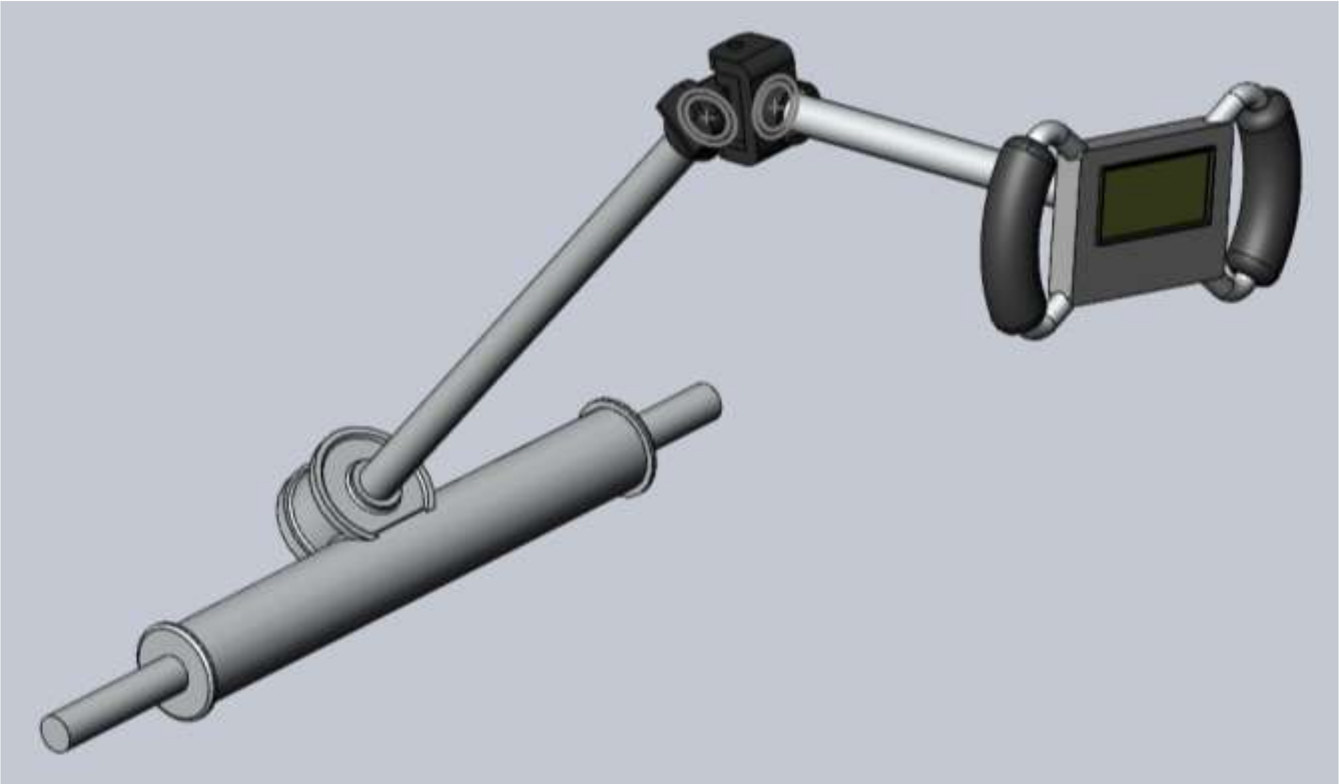
Therefore, Module(M) = 2

Axial pitch of rack = $\pi \cdot M$ = 6.28mm

No. of teeth on rack = $2 \cdot \delta / (\pi M) \approx 10$ (considering some amount of clearance)

Therefore, the actual length of rack (Lr) = $Tr \cdot \pi \cdot M$ = 62.83mm





Rack and Pinion Assembly

Brake System Overview

GOALS

- **Stopping Distance:** Achieve a stopping distance of less than **20 meters** under dry conditions, in line with high-performance standards for safety and control.

- **Brake Bias:** Target a front-to-rear brake bias of approximately **60:40** for optimal stability and performance, adjustable to match different track conditions.

- **Thermal Management:** Maintain optimum rotor temperatures during peak braking to prevent fade.

- **Pedal Force:** Design for optimum pedal force requirement to ensure driver comfort and precise control without excessive effort.

Component Selection

- Caliper and Brake Pads

We have currently selected the Wilwood GP200 brake caliper with the compatible brake pads for our braking system; however, this choice remains tentative and may be subject to change as further evaluations are conducted

Datasheet : Link

- Master Cylinder

For Master Cylinder we have decided to use Wilwood's Compact Remote Flange Mt Master Cyl-Banjo Outlet bore diameter 7/8 in were chosen which can withstand the pressure and transmit the force.

Datasheet : Link

Brake System Calculations

- Objectives :
   - Brake forces on front and rear wheels
   - Weight on front and rear wheels
   - Deceleration
   - Stopping Distance
   - Brake Line Pressure
   - Brake Effort

Brake-system Calculation

Brake force calculation

Front braking force, F_{bf}: 3357.07 N
Rear braking force, F_{br}: 1119.313 N

- $F_{bf} = U * \frac{\left(W*L_1 - \left(\frac{W}{g}\right)*Acc*H\right)}{L}$
- $F_b = \left(\frac{W}{g}\right) * Acc - R - Fd$

Weight transfer calculation

Weight on front wheel, W_f: 2156.741 N
Weight on rear wheel, W_r: 835.309 N

- $W_f = W * L_2 + H * \frac{(Fb+Fr*W)}{L}$
- $W_r = W * L_1 - \frac{H(F_b+Fr*W)}{L}$

Deceleration

Deceleration on front wheel :
D_{fw}=15.270 m/s^2
Deceleration on rear wheel :
D_{rw}=13.145 m/s^2

- $D_{fw} = \frac{F_{bf}*g}{W_f}$
- $D_{rw} = Fbr * \frac{g}{W_r}$

Brake Pressure Line Calculation

Front brake line pressure = 16.94 MPa
Rear brake line pressure = 5.64 MPa

- $\mathrm{P_{front}} = \frac{\mathrm{F_{c\,front}}}{\mathrm{U_c*2*Acp}}$
- $\mathrm{P_{rear}} = \frac{\mathrm{F_{c\,rear}}}{\mathrm{U_c*2*Acp}}$
- $\mathrm{F_{cfront}} = \mathrm{U_c} * \mathrm{Wf} * \frac{\mathrm{r}}{\mathrm{r_c}}$
- $\mathrm{F_{crear}} = \mathrm{Uc} * \mathrm{Wr} * \frac{\mathrm{r}}{\mathrm{r_c}}$

Brake Effort Calculation

Brake Effort = 1445.158 N

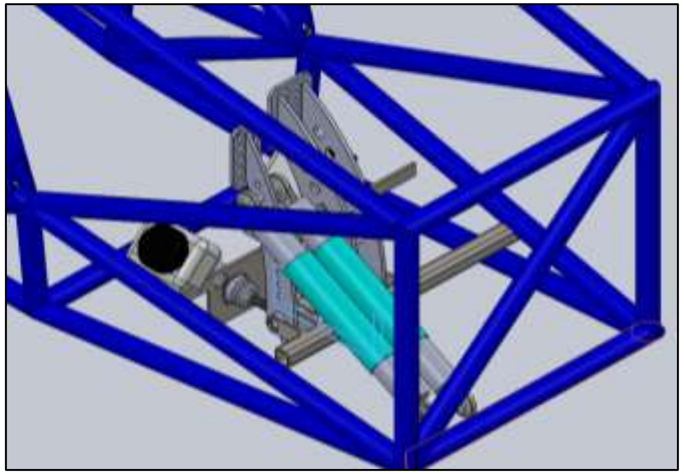
- $\mathrm{F} = (\mathrm{P_{front}} + \mathrm{P_{rear}}) * \mathrm{A_{mc}}$

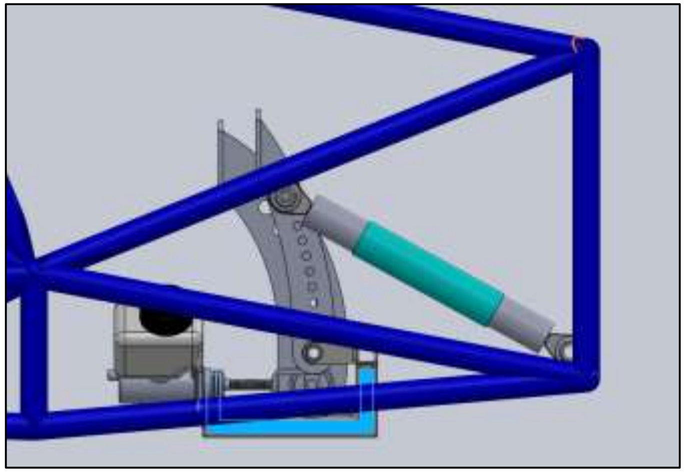
Stopping distance

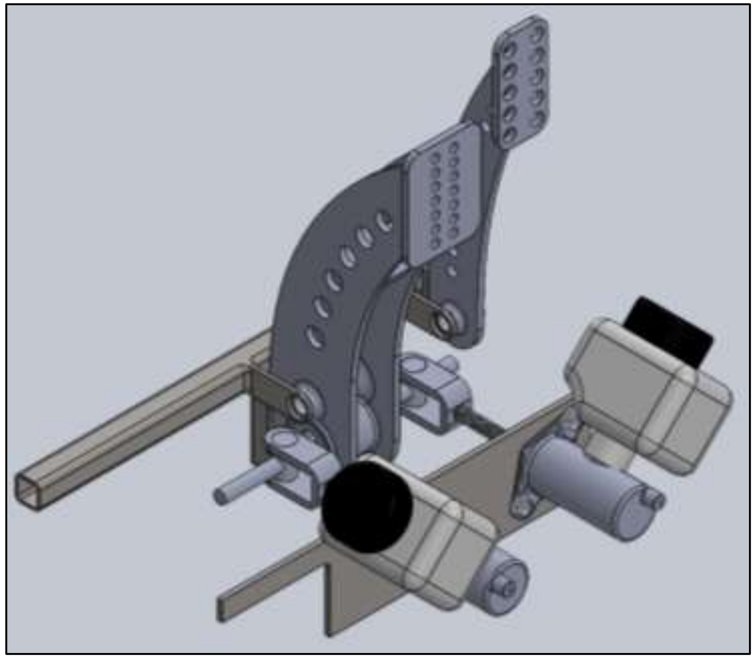
Stopping distance, s: 16.56 m
Wheel locking condition: Both wheels lock simultaneously

- $S = \left(Y_b * \frac{W}{2*g*Ca}\right) * ln\left(1 + Ca * \frac{V^2}{F_b+Rr}\right)$

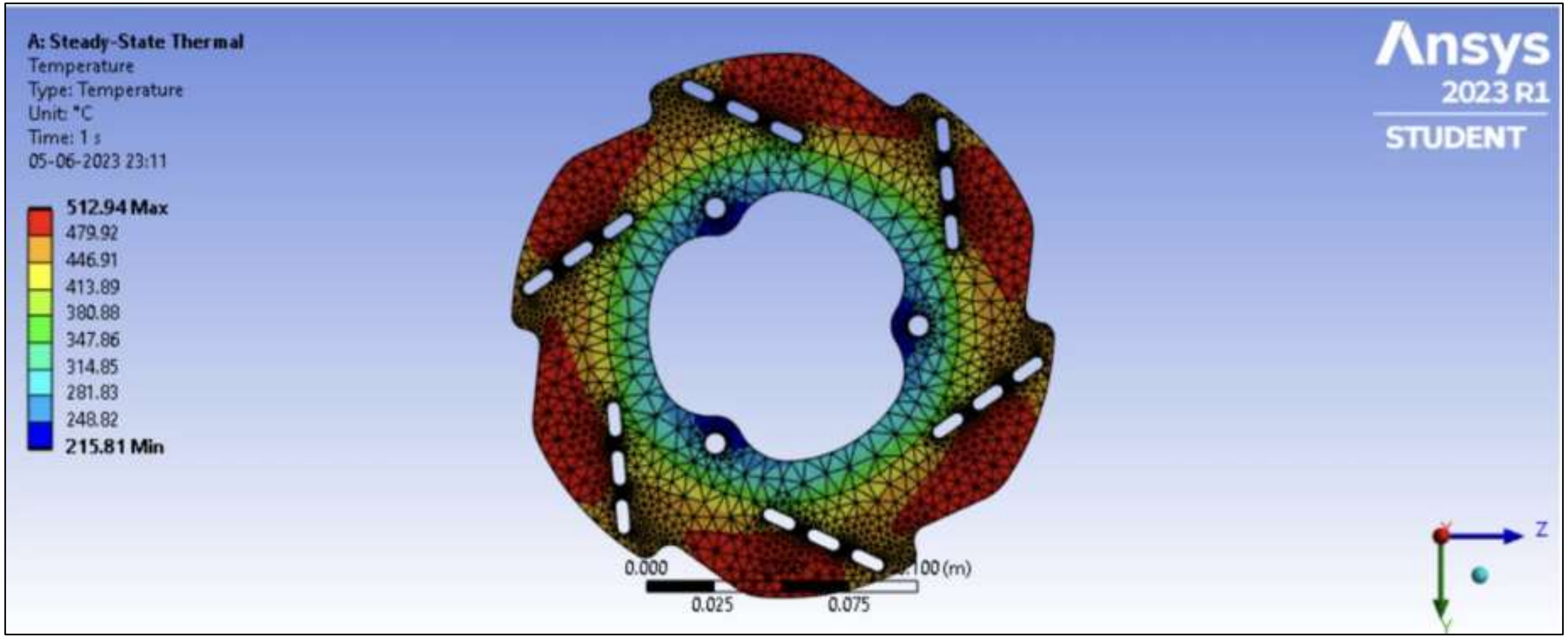
Pedal Box Design







Thermal Ansys of Rotor



Powertrain

- **Goals:** • Vehicle speed: 0 to 100 kmph in 3-5s
  • Maximum speed: 110-125 kmph

Force on car = $ma + \frac{1}{2} C_d \rho A v^2 + C_r W \cos\theta + W \sin\theta$

Vehicle Parameters	Values
Mass of the vehicle (m)	305kg
Drag Coefficient (C_d)	1.187
Density of air (ρ)	1.125
Road frictional resistance coefficient (C_r)	0.015
Slope Angle (θ)	0
Diameter of Wheel	45.72 cm (18'')

Powertrain

- Motor Selection

Make a list of considerable motors as permitted by Formula Bharat rules.

Tabulate maximum speed (rpm) without flux weakening.

Tabulate Final Gear Ratio for achieving design objectives of maximum vehicle speed.

Using the gear ratios and torque-speed characteristics, compare 0-100 kmph timings of each motor from vehicle dynamics calculation simulations.

Power density and cost considerations are taken into account along with timings to select appropriate motors.

Emrax 208

Bamocar d3 700

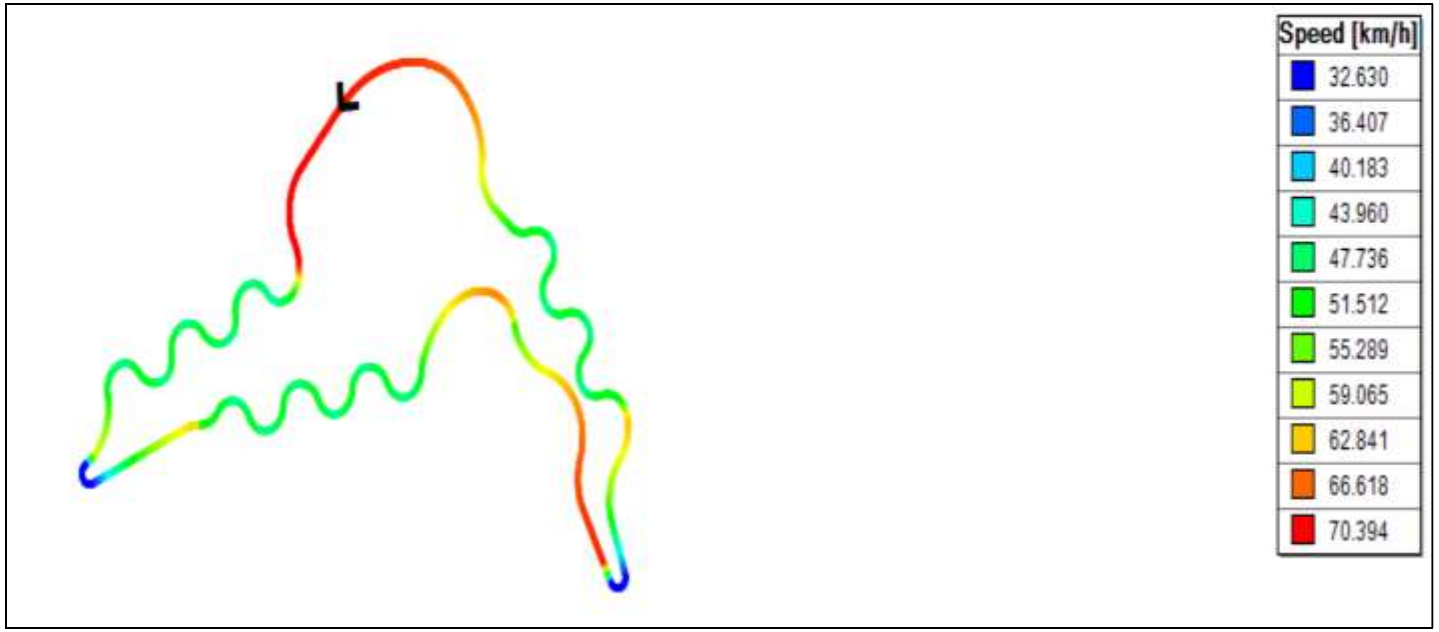
Powertrain

- Motor Selection

Motor	EMRAX 188	EMRAX 208	REX90
Maximum Power	52 kW	68 kW	70 kW
Weight	7kg	9kg	17.3 kg
Cost	3260 €	3580 €	9000 €
Maximum RPM	6500	6000	4000
Final Ratio for 150 kmph	3.73	3.45	2.30
Maximum Torque	90 Nm	140 Nm	200 Nm
0-100kmph Time	5.839 s	3.93 s	4.14 s

Powertrain

- Gear Ratio Selection
  - Track Details



- The track has been designed keeping in mind the guidelines of the endurance track mentioned in the Formula Bharat Rulebook.
- The track contains 3 straights, two of which are 50m in length and the third one 40m, 2 hairpins with an inner radius of 4m, and a couple of miscellaneous decreasing radius turns.

Powertrain

- Energy Requirements

Obtain velocity-acceleration data from lap time simulations and hence calculate the acceleration force (Fma=Mass of vehicle x longitudinal acceleration

Calculate Net Force using Fnet= Fma + downforce + Rolling Resistance

Hence calculate Power required at every instant using Power= Fnet x velocity and find average power

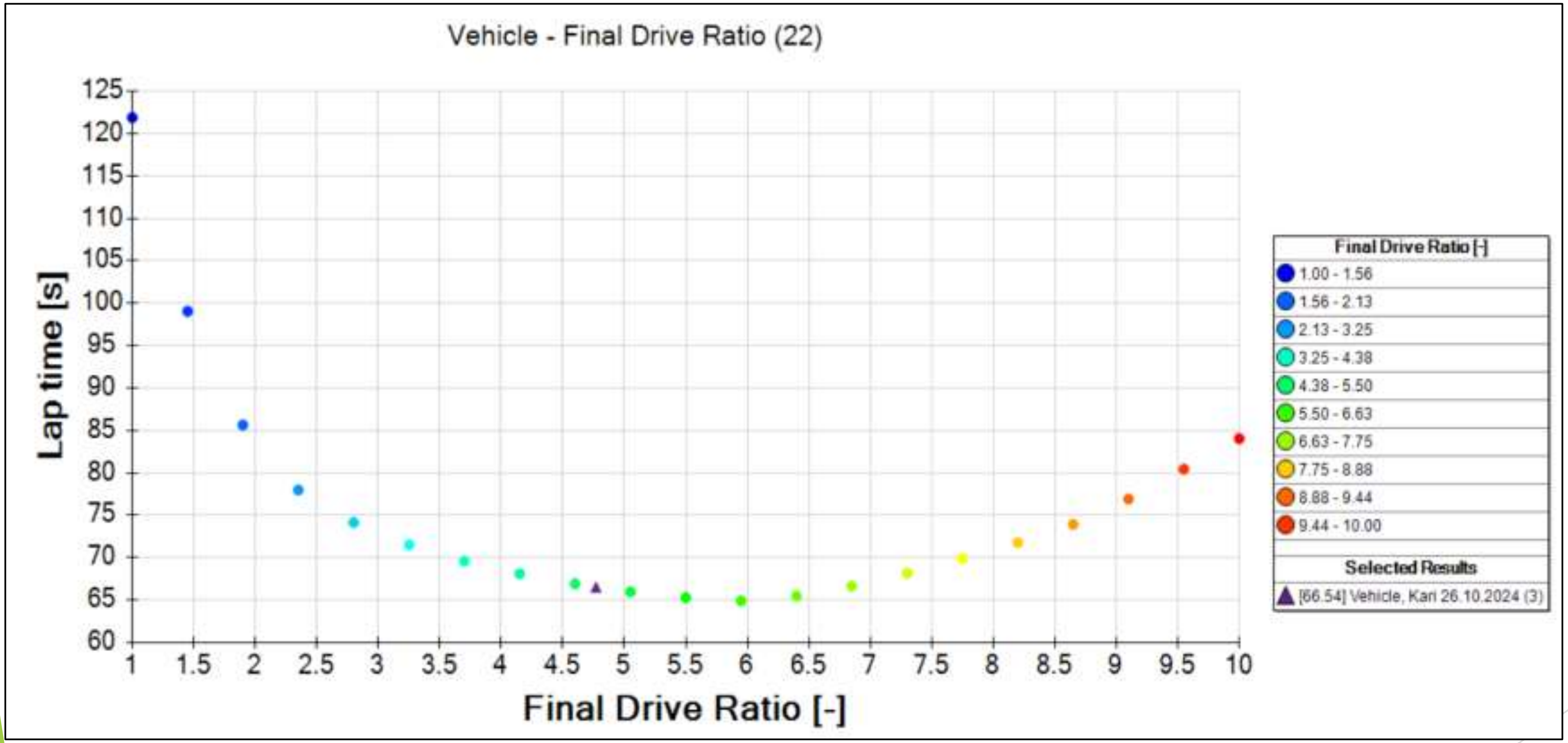
Multiply the average power by lap-time to calculate total energy required

Based on the calculations
Energy for one lap : 0.135024 kWh
Energy for 22 laps : 2.97 kWh

Powertrain

- Gear Ratio Selection (Lap Time Simulation)

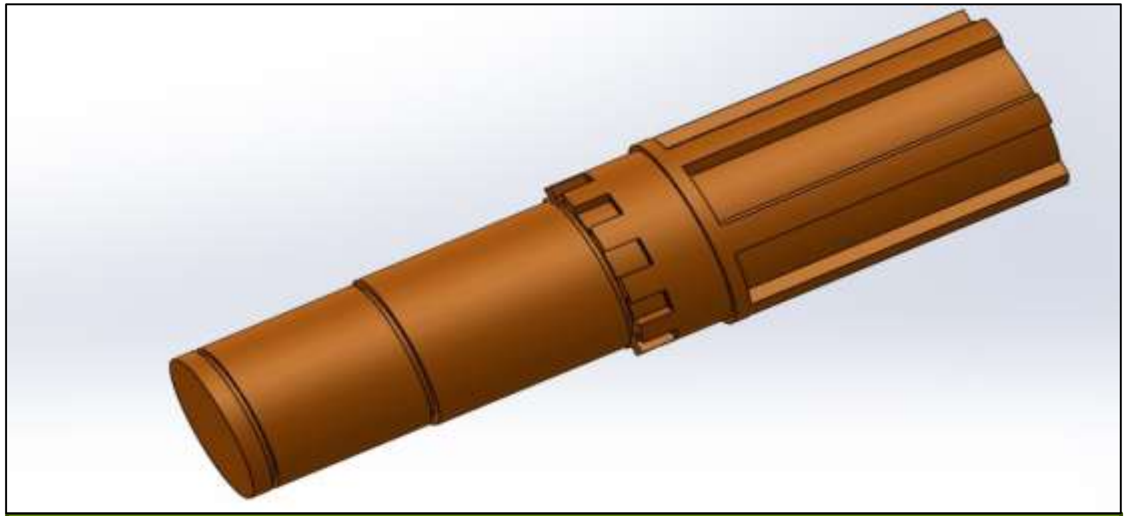
Optimum Lap software was used to carry out lap time simulation.

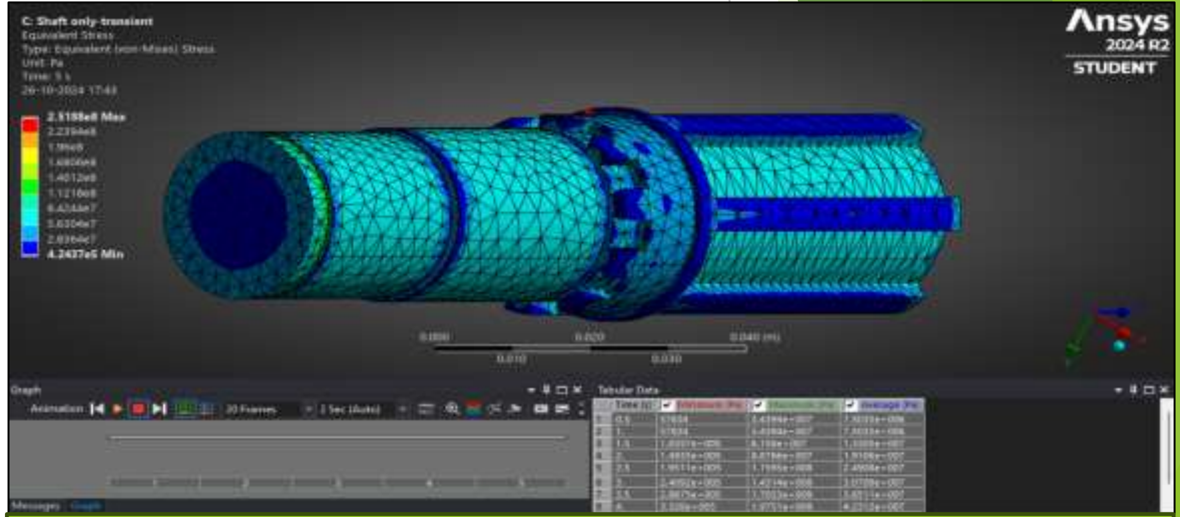


From the lap time v/s final drive ratio graph obtained, it was inferred that minimum lap time occurs at a gear ratio ranging from 4.5-6.5.
A larger gear ratio will result in lower maximum speeds on straights.
Thus, final drive ratio of 4.77 was chosen, which resulted in the desired max speed of around 70km/h and a lap time of 66.54s

Powertrain

- Components
  - Shaft





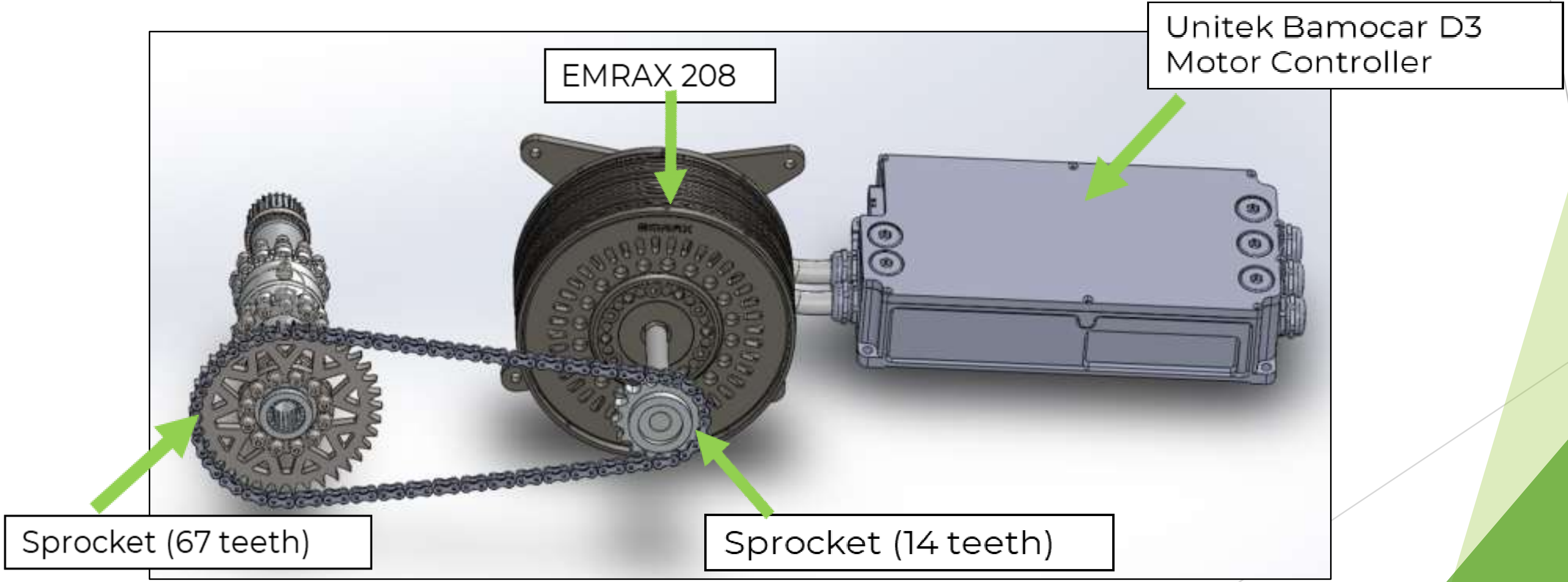
- Transmits power from motor to the drive sprocket.
- It has rectangular splines to facilitate coupling with sprocket.
- Material used: Annealed 4340 Steel, which has a yield strength of around 470 Mpa

FEA of shaft was carried out in Ansys Workbench. The end where shaft is supposed to be mounted to the chassis was considered fixed and dynamic torque was applied on the other end, which corresponded to the peak motor torque to account for the dynamic response of the system in the worst case scenario. Inertial loads and external resistance to the system were also taken into consideration.

Powertrain

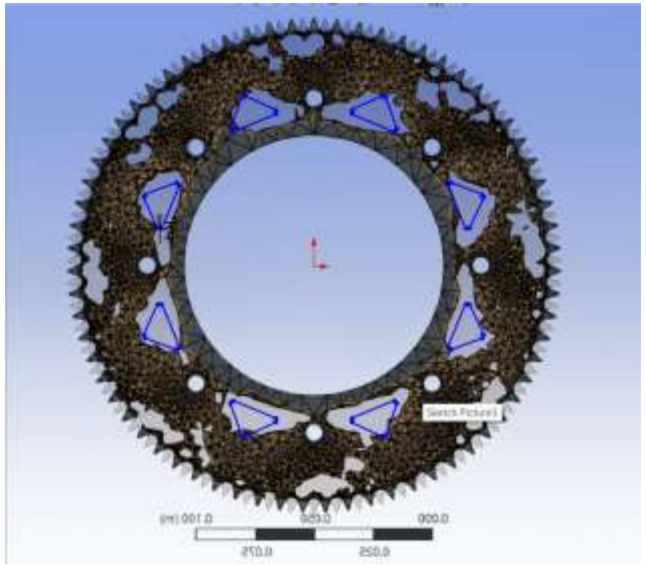
- Components
  - Chain drive system

As per the final gear ratio which is 4.77, we are taking the driver sprocket to have 14 teeth and the driven sprocket to have 67 teeth. The material needed should have high yield point to sustain enough load on the teeth. So we are using 201 annealed SS. We selected 06B1 chain for our vehicle



Powertrain

- Components
  - **Driven Sprocket Design Overview:**
    - The driven sprocket is manufactured using **7075-T6 aluminum**, a high-strength, lightweight alloy known for its excellent fatigue resistance. To achieve optimal performance, material removal follows the results of **topology optimization**, ensuring unnecessary mass is minimized without compromising structural integrity. This process reduces weight while maintaining durability, contributing to the overall efficiency of the system



	24 mm

Powertrain

- Components
  - Bearing Selection
    - Deep Groove Bearing – positioned between the Shaft and its mount

Torque from Motor	140 Nm
Gear Ratio	4.77
Torque applied on the differential(T)	140 × 4.77 = 667.8 Nm
Weight of Differential (Wd)	3.5 × 9.81 = 34.335 N
Diameter of big sprocket(mm) (ds)	203.111
Weight of Driven Sprocket (Ws)	0.32501 x 9.81 = 3.188 N

Powertrain

- Bearing Selection (Deep groove Bearing)

Torque by Motor (F)	140 N-m
Radius of small Sprocket (R)	24 mm
Mass of Shaft (m_s)	691.15 g
Mass of small sprocket (m_{SS})	22.44 g
Angle of wrap (α)	129.5^o

Tension force on sprocket (T) $= \frac{F}{R} = \frac{140}{24} = 5.83\ kN$

Weight of shaft (W_s) $= m_s * g = \left(\frac{691.15}{1000}\right) * 9.81 = 6.78\ N$

Weight of small sprocket (W_{ss}) = $m_{SS} * g$ = 0.22 N

$\beta = (\pi - \alpha)/2 = \frac{180-129.5}{2} = 25.25^o$

Tension force on sprocket from in radial direction

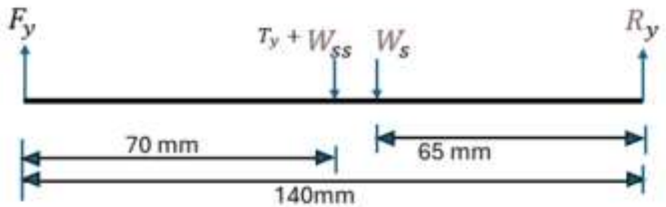
$(T_r) = 2 * T * \sin(\beta) = 2 * 5.83 * \cos(25.25^o) = 10.546\ kN$

Tension force along axis,

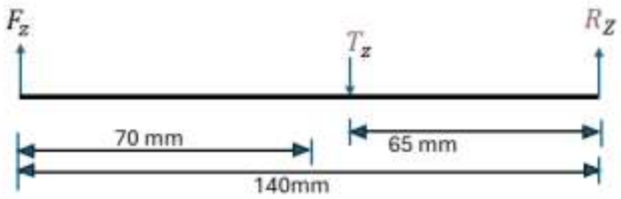
$T_z = 10.1221\ kN$ $T_y = 2.96\ kN$

Force Analysis of Bearing

Along Y-axis



Along Z-axis



On balancing force and net torque along Y-axis and Z-axis separately,

$F_z = 5.422\ kN$ $F_y = 2.755\ kN$

Radial Force $(F_r)^2 = ((F_z)^2 + (F_y)^2)$

$F_r = 6.0817\ kN$

Load factor =1.4

Equivalent Radial Load (P) = Load factor * $F_r = 8.5\ kN$

Bearing Life (L_{10}) =50 million rev.

Therefore,

Load Dynamic Capacity (C) = $P * (L_{10})^{\frac{1}{3}}$= 31.499 kN

Powertrain

- Bearing Selection

Calculation of Axial Forces due to differential

Torque by Motor	667.8 Nm
Module	4mm
Pressure Angle (α)	20°
Face Width	15.77 mm
Gear Ratio of Differential	2.144

Diameter of Pinion (D_p) = 80 mm

Mean Radius (r_m) = $\frac{D_p}{2} - \frac{b*\sin(\Upsilon)}{2}$ = 25.703 mm

Tangential Component of Force (P_t) = T/r_m = 25.981 kN

For axial force by differential,

$$P_a = P_t * \tan(\alpha) * \sin(\Upsilon)$$
$$= 8.57\ kN$$

For Selection of Bearing

Radial Force of A (F_{rA})	7.039 kN
Radial Force of B (F_{rB})	4.864 kN
Load Factor	1.3
Bearing Life	10.482 million revolutions

Here, $F_{rA} > F_{rB}$

Therefore, $F_{aA} = \frac{(0.5*F_{aB})}{Y}$

where Y is the thrust factor.

$F_{aB} = F_{aA} + (P_a)$ …… (4)

On solving (3) and (4), we get

$F_{aB} = \frac{(Y*P_a)}{Y+0.5}$

$F_{aA} = \frac{(P_a)}{Y+0.5}$

Equivalent Dynamic Load, P = $0.4 * F_r + Y * F_a$

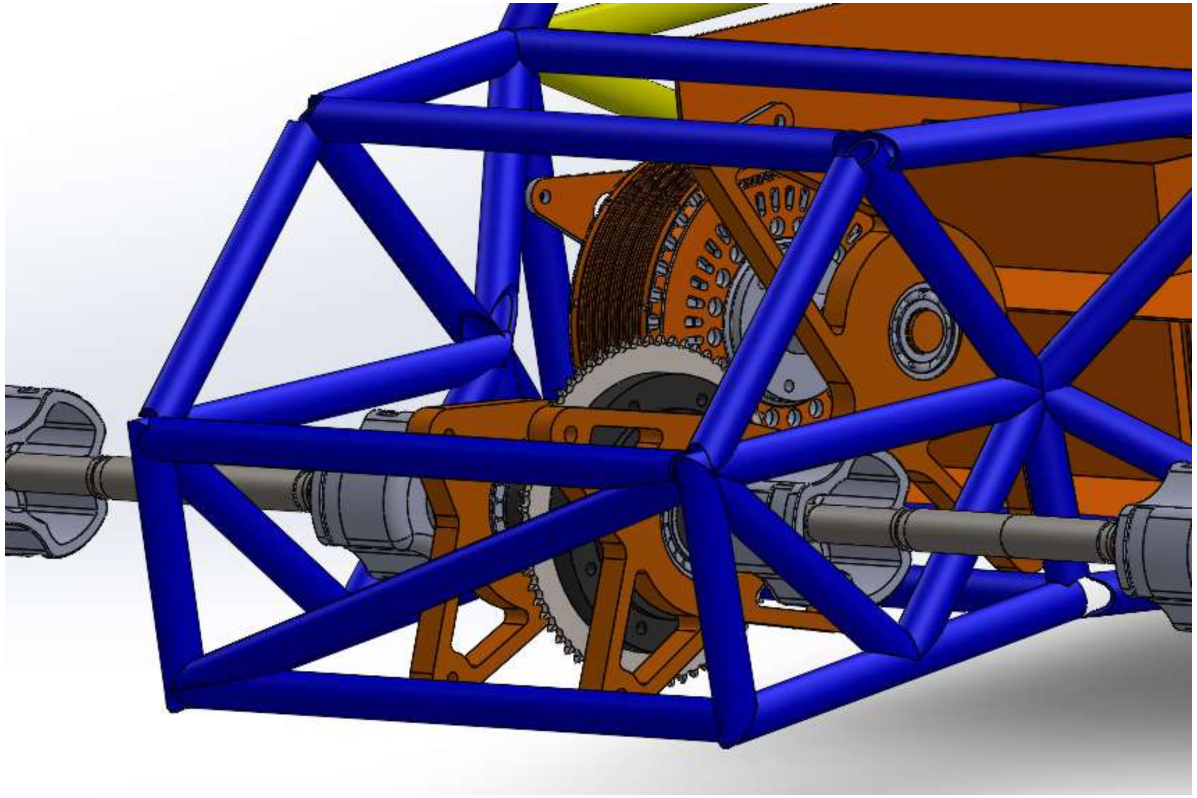
Equivalent Load Force, $C = P * (L_{10})^{\frac{1}{3}} * 1.3$

For Y = 1.5,
C_A = 26.2917 kN
C_B = 32.96 kN
Bore Diameter of Bearing = 50 mm

Powertrain

- Components
  - Mounts

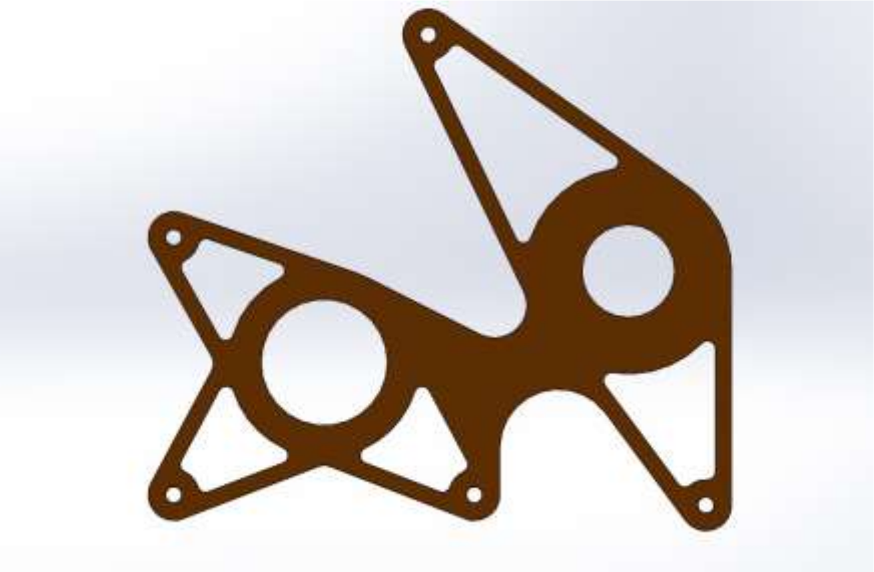
The mounts are designed to securely hold the differential and motor in the car. We selected **7075-T6 aluminum** for its **lightweight** nature and **high strength**, ensuring the mounts can withstand operational loads without exceeding the material's yield strength. This helps maintain the car’s performance and durability. The mount’s position within the car is shown in the photo shown.

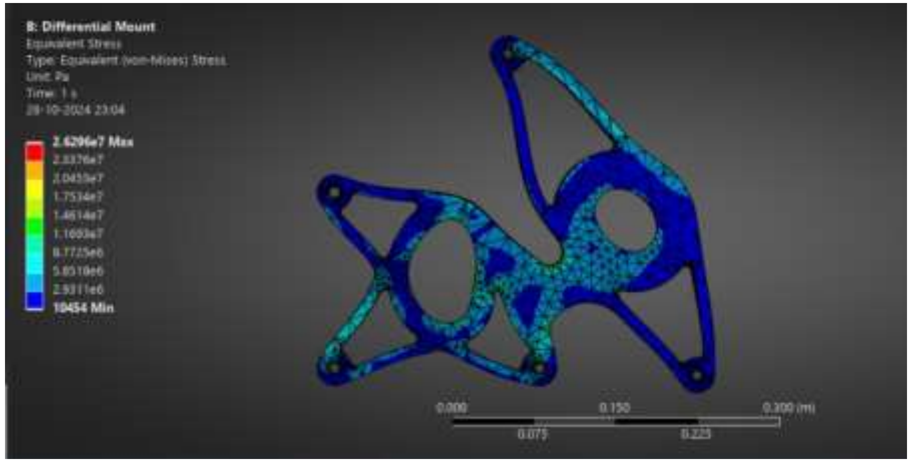


Powertrain

- Components
  - Differential cum Motor Mount

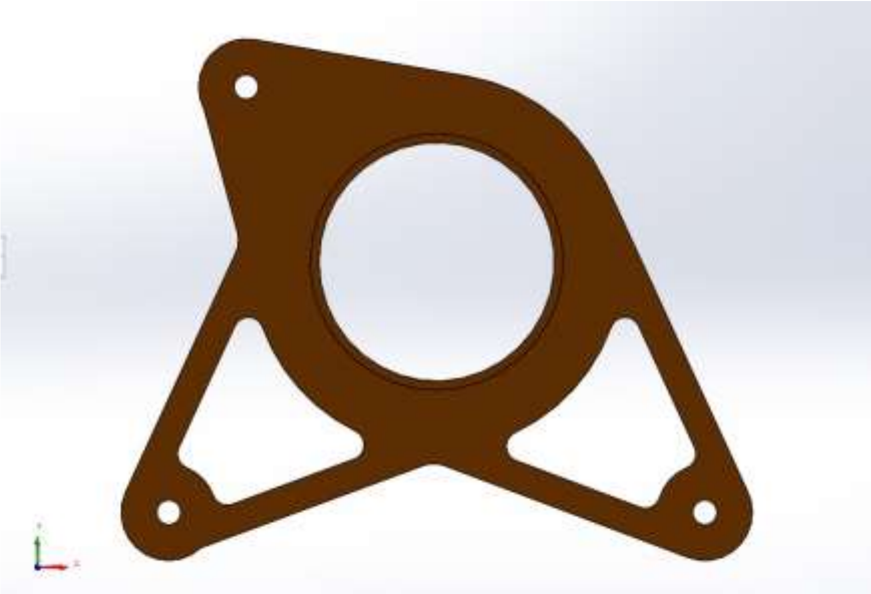
We are using aluminum 7075-T6 due to its excellent combination of lightweight properties and high strength, making it ideal for applications where minimizing weight is crucial without compromising structural integrity. Its superior strength-to-weight ratio ensures that it can sustain the applied forces while remaining within the material's yield strength limits, ensuring safety and durability

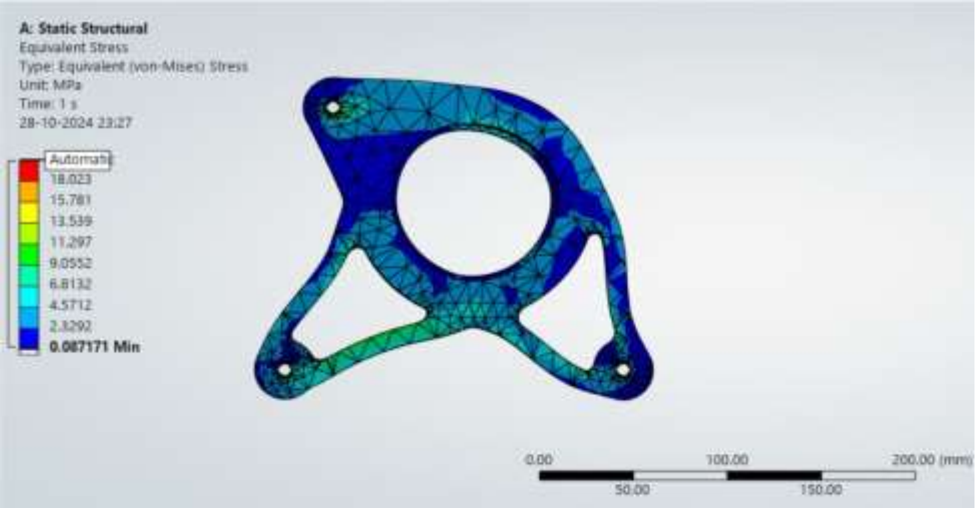




Powertrain

- Components
  - Differential Mount





Powertrain

Accumulator

- Goals

- Reliable and safe energy storage solution
  - High Performance
  - Satisfy energy requirements
  - Minimize Weight
    - Material Selection
    - Optimizing Space usage
    - Cells safety at 40g acceleration
  - Optimal Power output
    - Minimum energy losses
  - Maintain Safe operating temperature
    - Optimal thermal management

Powertrain
Accumulator

Energy Requirements

Tractive Energy required for 1 laps (1 km) = 139 Wh => for 25 laps (25 km) = 3.5 kWh
Motor efficiency = 94% ; Controller efficiency = 97% ; Transmission efficiency = 96%

Pack energy required = 3.99 kWh => Considering 15% Factor of Safety,
Final Pack energy = 4.59 kWh

Current Requirements

Max tractive power during the drive cycle = 40.1 kW => considering the efficiencies,
Max continuous power delivered by pack = 45.81 kW
Max peak power delivered by pack = 77.69 kW

Max continuous current required = 130.9 A ; Max peak current required = 221.9 A

Powertrain
Accumulator

Cell chemistry selection Parameter

1. Longer life cycle
2. High energy density
3. High Capacity & Voltage
4. Stable
5. Low self heating
6. Resistant to corrosion
7. Easily available and economical

For ref click here

Table -3: Voltage, Specific energy and volumetric energy for Li-ion batteries cell chemistry.

Material	Voltage (Average V/s Li/Li^{+})	Capacity (mAh/g)	Crystal Density (g/cm^3)	Tap Density (g/cm^3)	Specific Energy (Wh/kg)	Volumetric Energy (Wh/L)
LCO	3.8	150	5.10	2.9	570	2907
LMO	4.0	110	4.31	2.5	440	1896
NMC	3.7	170	4.75	2.5	629	2988
LFP	3.4	160	3.60	1.5	544	1958
NCA	3.7	185	4.85	2.5	685	3322

Cell Comparison

	BAK NMC N18650CNP	LPHDA88515 5	NMC (HHPOWER)	
Type	Cylindrical	Pouch	Pouch	
Voltage	3.6V	3.7V	3.7V	
Capacity(Ah)	2.5	16.8	32Ah	
Max. Discharge	12C	10C	8C	
Configuration	7P98S	1P95S	1P95S	
Pack Energy	6.17kWh	5.9kWh	11.25kWh	

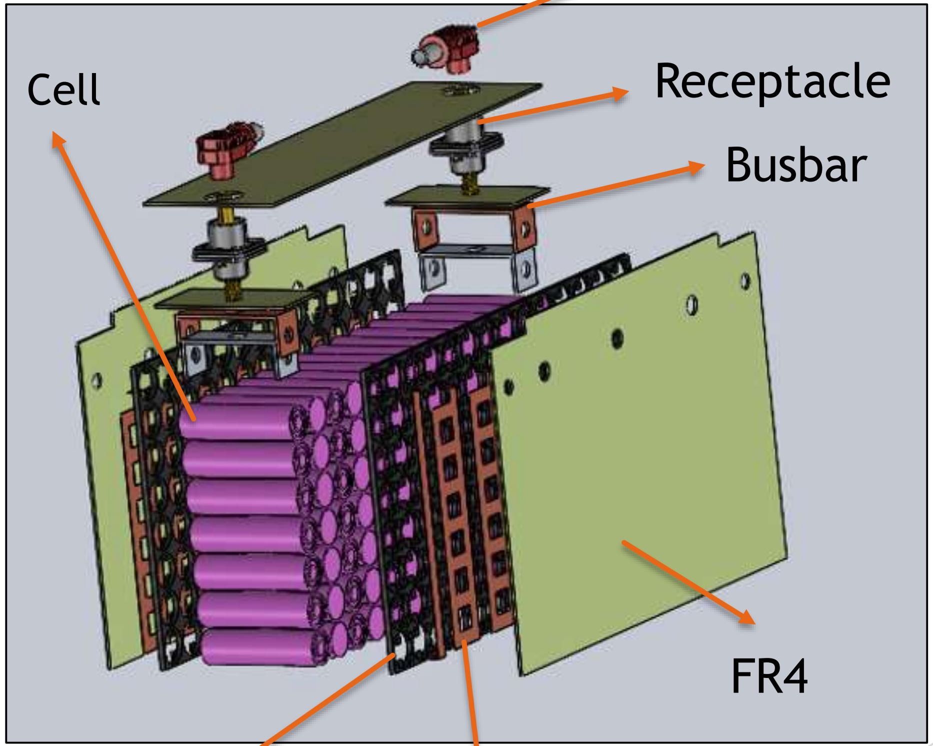
Cell Specifications

- Chemistry: NMC
- Capacity (mAh) : 2500
- Output Voltage: 3.6V
- Charge Rate: 0.5C
- Continuous Discharge Rate: 8C
- Peak discharge Rate: 35C for 1s
- Weight of single cell: 48gm

Powertrain
Accumulator

Module Configuration

- Total 7 modules
- Each module contains 98 cells
- 7P14S Configuration
- Each cell holds properly
- Light weight
- Rule compliant



Powertrain

Accumulator container

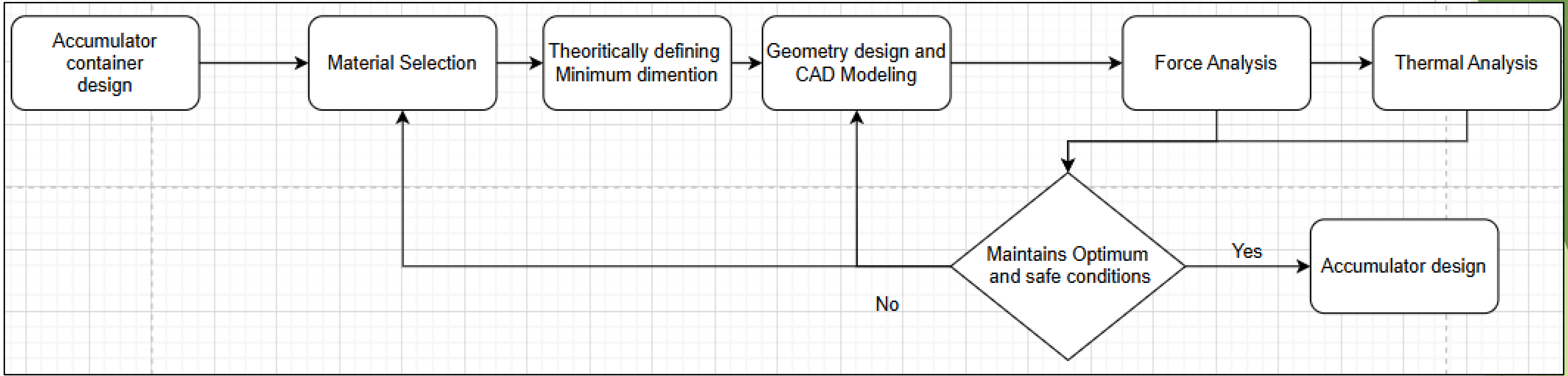
- Mechanical Design:
- Container material selection and container Design
- We have many options for material of accumulator container we have chosen Aluminium over other because of its weight reduction, corrosion resistance, Formability, thermal conductivity and recyclability.

	Al 6061 T6	A514 steel T1	Al 7075 T6
Yield Strength	270 MPa/39 x 10^3 psi	700 MPa	480 MPa/69 x 10^3 psi
Weldability	Very Good(25ksi)	Good	Not Weldable
Melting Point	580-660°C	1420-1460°C	477 - 635.0 °C
Thermal Conductivity	170 W/m-K	52 W/m-K	130 W/m-K
Corrosion Resistance	Excellent	Optimal	Poor
Machinability	Superior	Yes	Fair/Average

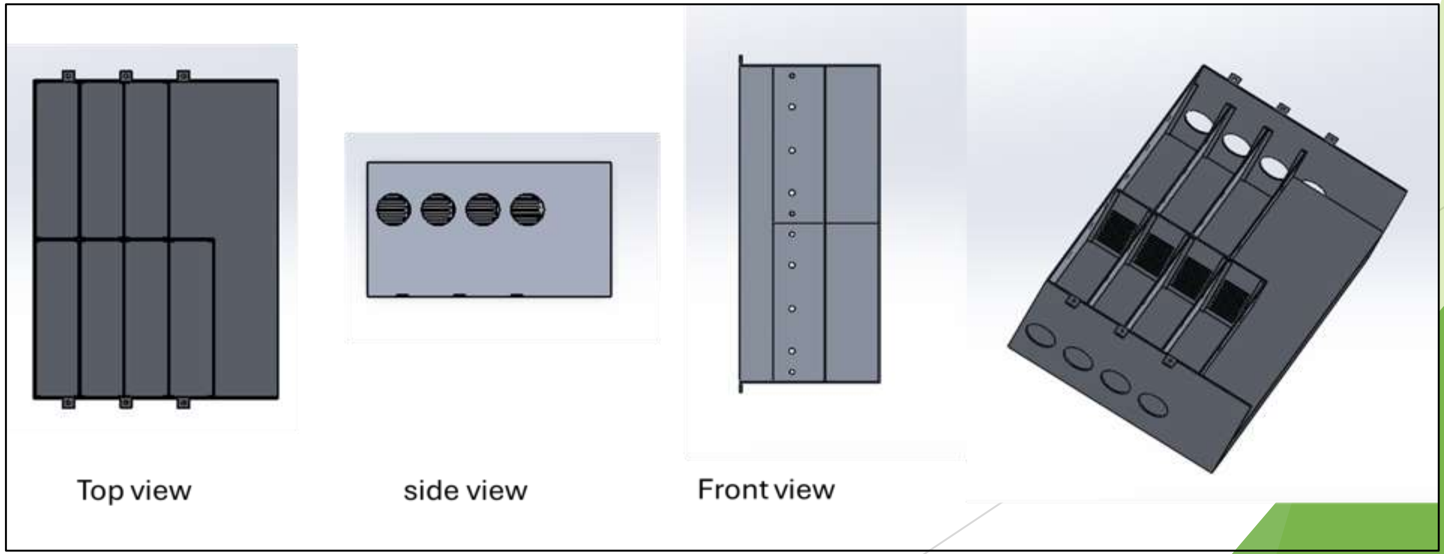
Powertrain

Accumulator Container

- Mechanical Design Flow Diagram



- Container Design



Powertrain

Accumulator Container Calculations

Internal Wall

Mass of Module		4.7	Kg
Modulus of elasticity (E)		68.9	GPa
Height of the mounting point a		0.185	m
Thickness of the inner wall(t)		0.0025	m
Width of the inner wall(b)		0.285	m
Modulus of elasticity (E)		68.9	Gpa
w=F/b	6471.157895 N/m		
Net force on inner wall	m*40g	1844.28	N
Max Bending moment (M) =	(w*b^2)/12	43.80165	Nm
Second moment of inertia(I)=	$(l*t^3)/12$	2.40885E-10	m^4
calculated stress(σ)	M*y/I (Max stress	227295048.6	Pa
Net deflection(δ)=	w*l^4/384 EI=	1.18935E-06	mm
FOS=	1.214307713		

Base Plate

Length of the plate(L)=		0.58	m
Width of the plate(w)=		0.42	m
Thickness of the plate(h)=		0.005	m
Modulus of elasticity(E)		68.9	Gpa
No of modules =		7	
Weight of one module =		4.7	Kg
The net force on the bottom plate =	7*4.7*20*9.81=	6454.98	N
Net force per unit length(f)=	F/l=	11129.27586	N/m
Maximum Bending Moment(M_{max})=	$f*l^2/12$	311.9907	Nm
I	wt^3/12	4.375E-09	
calculated stress(σ)=	M*y/I	178280400	Pa
Nett deflection(δ)=	w*l^4/384 EI=	10.8805382	mm
FOS		1.548126543	

External Plate

Modulus of elasticity (E)		68.9	GPa
Height of the mounting point a		243.5	mm
Thickness of the outer wall(t)		5	mm
Width of the outer wall(b)		423	mm
Modulus of elasticity (E)		68.9	Gpa
The net force at the mounting	(m*a*6)/6=	1962	N
Bending moment (M) =	(F*l)/2=	(1962*0.15)/8	
Second moment of inertia(I)=	$(b*h^3)/12$	4.40625E-09	m^4
calculated stress(σ)=	M*y/I	33.88276596	MPa
Nett deflection(δ)=	5*F*l^3/384*E*I	1.214916201	mm

Welding Calculation of internal and external walls

P1(Load internal)=	40*g*5.8	2275.92	N
P2 (Load external)=	40*g*5*4	7848	N
h(height of weld)=		1	mm
l1 (length of internal wall)		183.5	mm
l2 (length of external wall)		248.5	mm
T (Aluminium weld shear strength)=		120	Mpa
Max load on inernal wall weld can ha	h*l1*T/1.21=	18198.34711	N
Max load on extrnal wall weld can ha	h*l2*T/1.21=	29820	N
FOS1(factor of safety of internal wa	max.load/load ap	7.99603989	
FOS2(factor of safety of external wa	max.load/load ap	3.79969419	

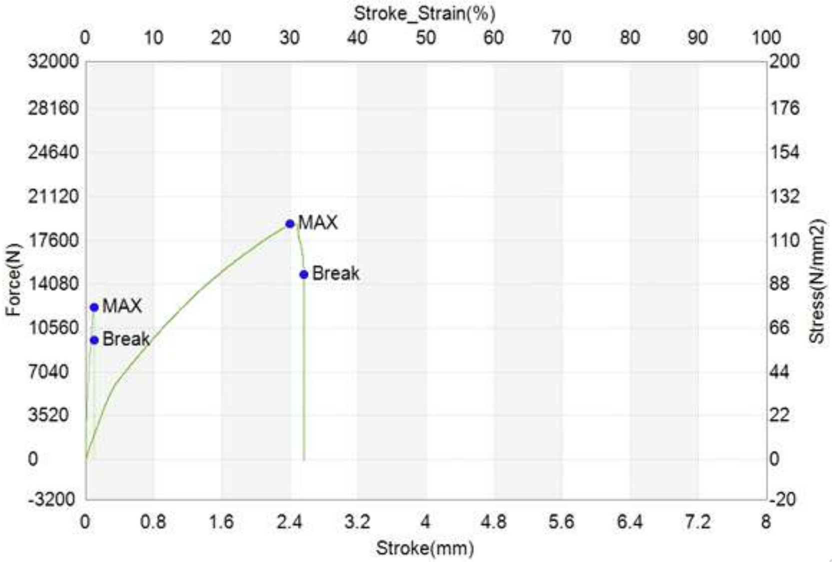
Shear strength calculation of bolts chassis to accumulator attachment

Grade		8.8		
Diameter		10	mm	
Allowable shear stress of grade 8.8 bolt=		369.28	MPa	0.577*yeild strength
mass of 7 module+ container=		50	Kg	
Total shear force on 5 bolts=		19620	N	{40*g*9.81}
Total shear on 1 bolt=		3924	N	Assuming equal distribution of force
Nominal Cross section area of bolts=		58	mm^2	
effective cross sectional are of bolt =	nominal*0.78	45.24	mm^2	
shear stress on each bolt=	F/A=	86.7374	MPa	
FoS(Shear)=	Allowed/used=	4.257448		

Powertrain

Strength Test for Aluminium

Universal Testing Machine



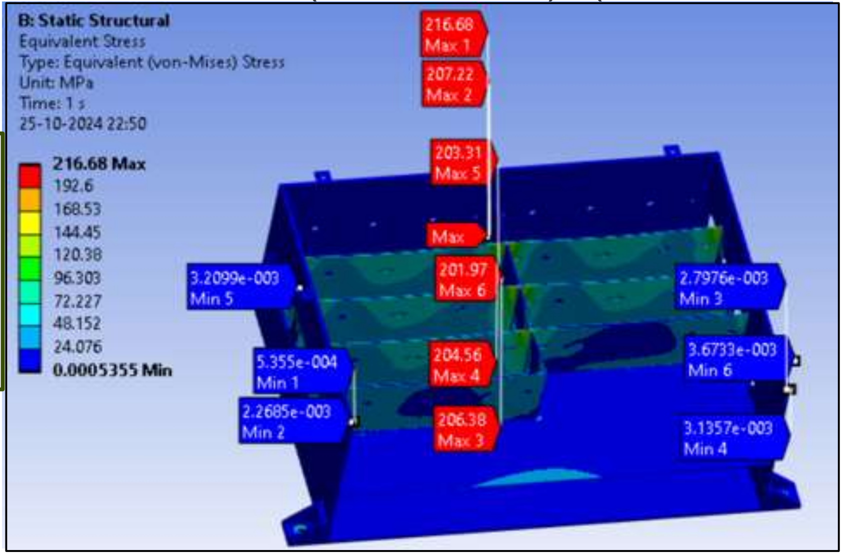
Failure of Lap Joint

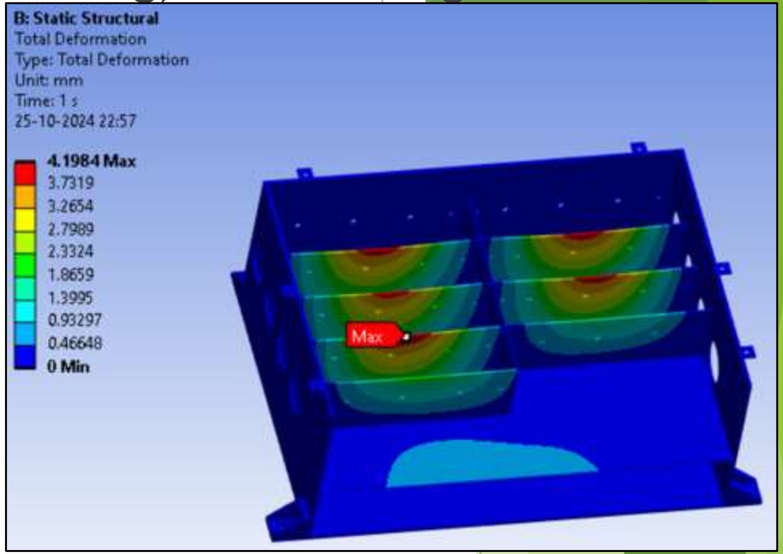
Powertrain

Accumulator Container

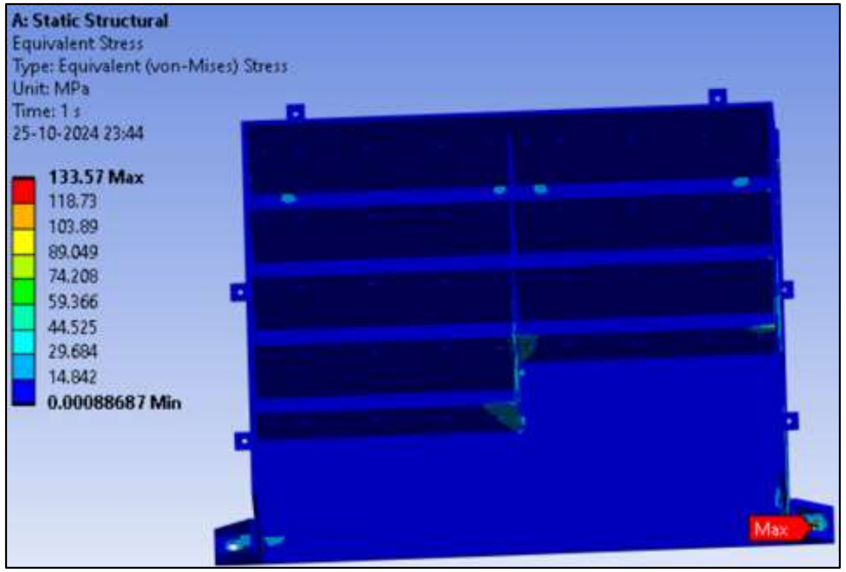
- Force analysis of Accumulator container(Al 6061 T6)=(40*mass of module*g)+Total mass*g

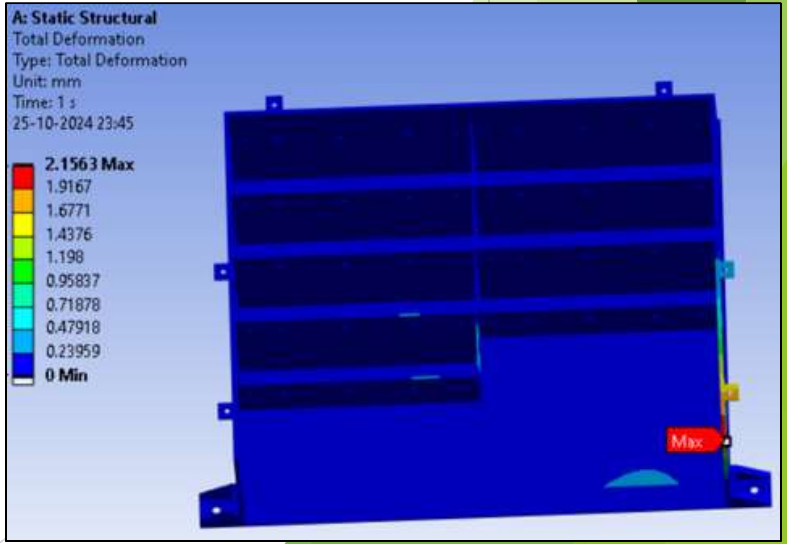
In x direction
Max. Stress=216.68MPa
Max. Deformation=4.19 mm





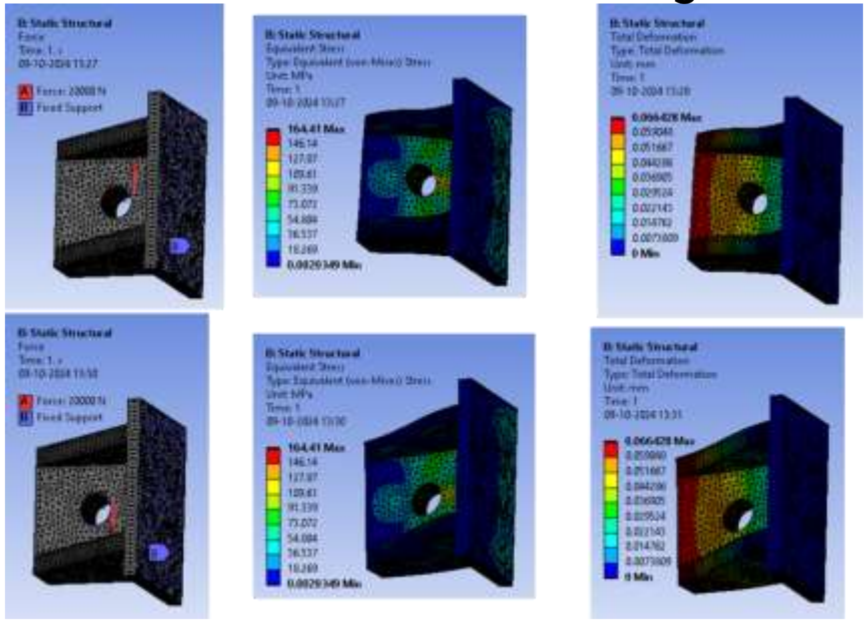
In y direction
Max. Stress=133.57Mpa
Max.
Deformation=2.1563mm





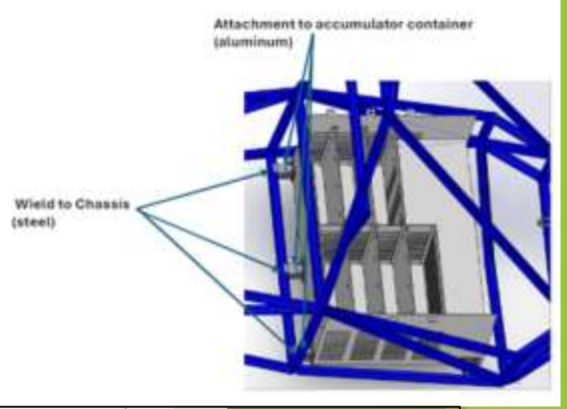
Powertrain
Mounting Design

Accumulator Side Mountings

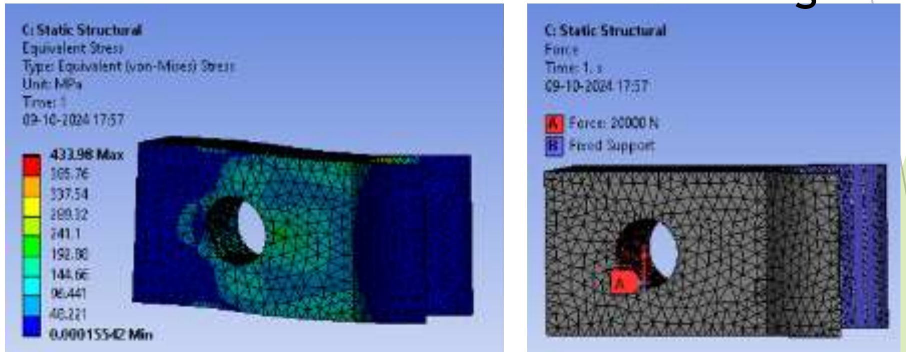


Shear strenth calculation of stack to accumulator container attachment

Grade		8.8	
Diameter		10 mm	
Allowable shear stress of grade 8.8 bolt		369.28 MPa	0.577*yeild strength
mass of 1 module		4.7 Kg	
Total shear force on 6 bolts		1844.28 N	(40*g*9.81)
Total shear on 1 bolt		307.38 N	Assuming equal distribution of force
Nominal Cross section area of bolts		58 mm^2	
effective cross sectional are of bolt	nominal*0.78	45.24 mm^2	
shear stress on each bolt	F/A	6.794429708 MPa	
FoS(Shear)	Allowed/used	54.35040406	



Chassis side accumulator mounting



Weld strenth calculation bracket pullout of acccumulator side bracket

The throat is the minimum cross-section of the weld
located at
45° to the leg dimension.

P1(Load inte	40*g*50/5	3924 N	
h(height of w		2 mm	
l1 (length of		40 mm	one of the minimum dimention is choosen to minimize the force allowed
T (Aluminiur		120 Mpa	
Max load on i	h*l1*T*0.	7933.884 N	
FOS1(factor	max.load/	2.021887	

Weld strenth calculation bracket pullout on chassis side

As a rule, the leg length h equals the plate thickness. The throat is the minimum cross-section of the weld
located at
45° to the leg dimension.

P1(Load internal): 40*g*50/5		3924 N	
h(height of weld)=		2 mm	
l1 (length of inter		35 mm	one of the minimum dimention is choosen to minimize the force allowed
T (Aluminium welc		120 Mpa	
Max load on inern	h*l1*T*0.707=	5938.8 N	
FOS1(factor of sa	max.load/load applied=	1.513455657	

Powertrain

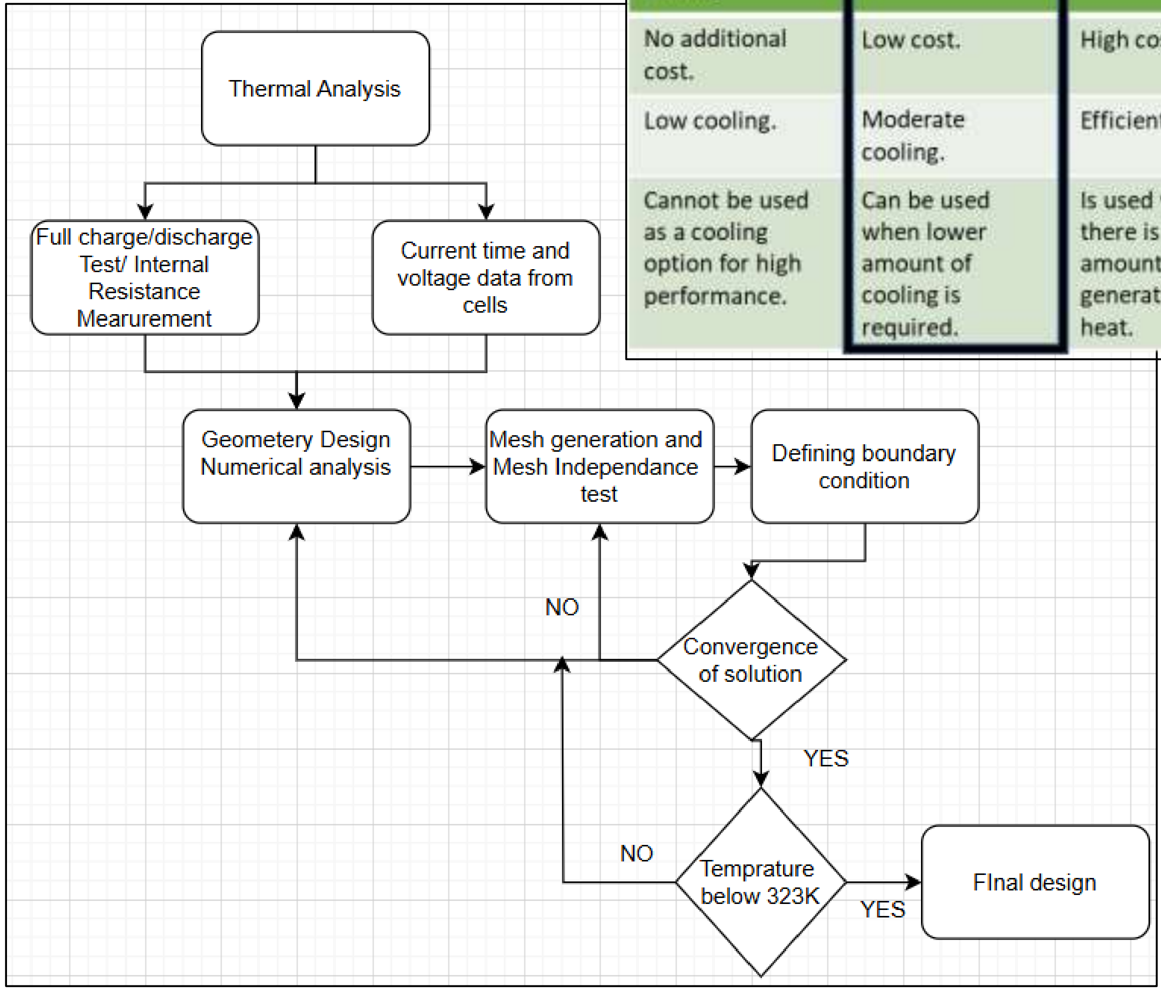
Accumulator Thermal Management

Reason for heat generation:

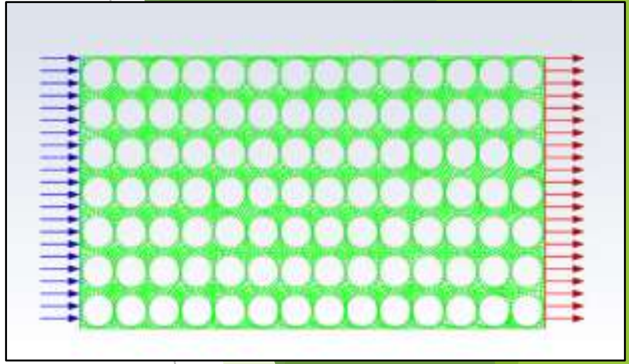
- Internal resistance due to flow of current.
- Heat generation due chemical reaction within cell.

Goals of cooling of battery pack:

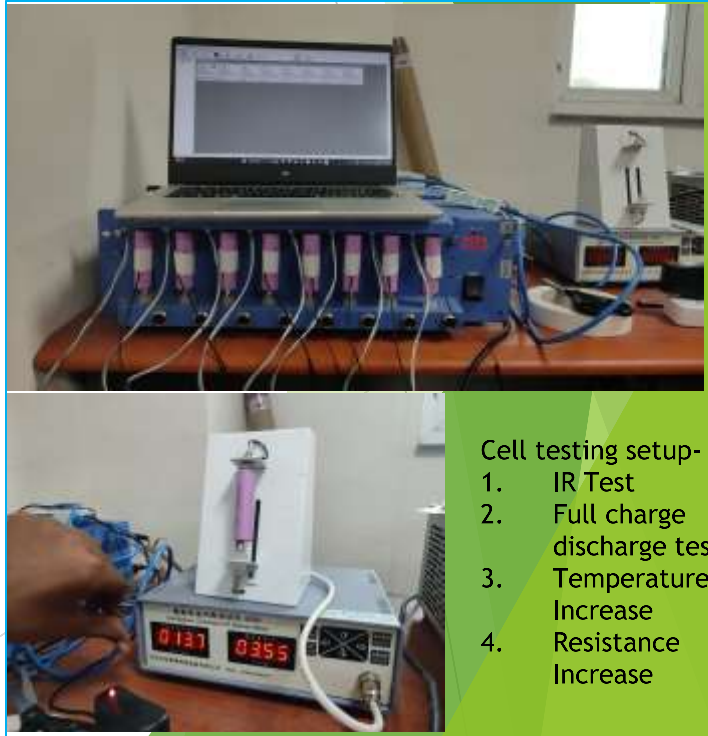
- Keeps the cells in operating temperature range under 60 degree celsius and keeps it inefficient range.
- Battery pack has been identified weakest component for thermal management and care should be taken during the design phase



Passive air cooling	Active air cooling	Liquid cooling
No additional cost.	Low cost.	High cost.
Low cooling.	Moderate cooling.	Efficient cooling.
Cannot be used as a cooling option for high performance.	Can be used when lower amount of cooling is required.	Is used when there is large amount of generation of heat.



Air domain



Cell testing setup-

1. IR Test
2. Full charge discharge test
3. Temperature Increase
4. Resistance Increase

In case of 40 degree Celsius temperature of air, the max temp obtained is 50 degree Celsius that is within specified limit

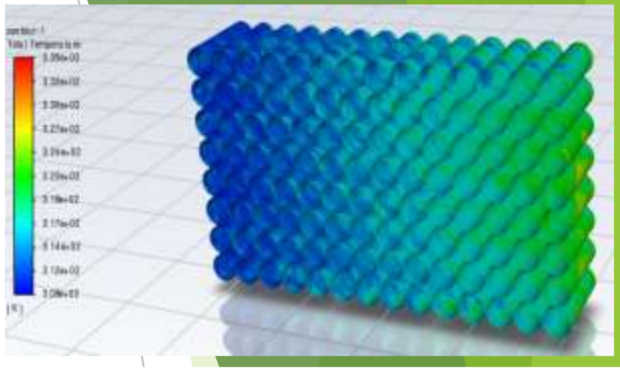
Powertrain

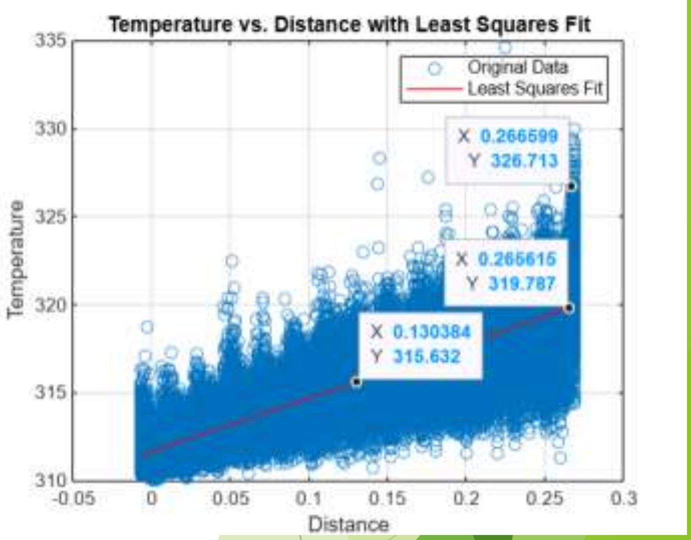
Accumulator Thermal Management

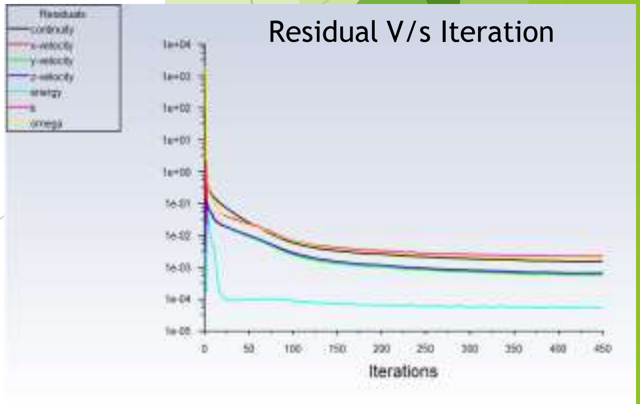
Cell Parameters	Average IR of each cell = 12mΩ Cell capacity=2.5Ah Discharge rate=8C
• *Determining the Current*	I=2.5Ah×8=20A
• *Heat Generation*	P= I^2 ×R=(20A)^2 ×0.012Ω=400×0.012 =4.8W
• *Surface Area of cell*	A=2πrh+2πr^2=2π×0.009 m×0.065 m+2π×0.009^2 ≈3.69× 10^(-3) m^2
• *Heat Flux*	q"= P/A = 4.8 W/(3.69×10^(-3)) m^2 ≈1300 W/m^2
• *Correction Factor*	f=[1+α(T-T2)]=0.05T-14.5 {Approx α=0.05, T2=310K}

These simulations are done with a heat generation linearly varying with temperature in Ansys fluent with initial temperature 37°C at 60 CFM.

Animated results of simulations on one module

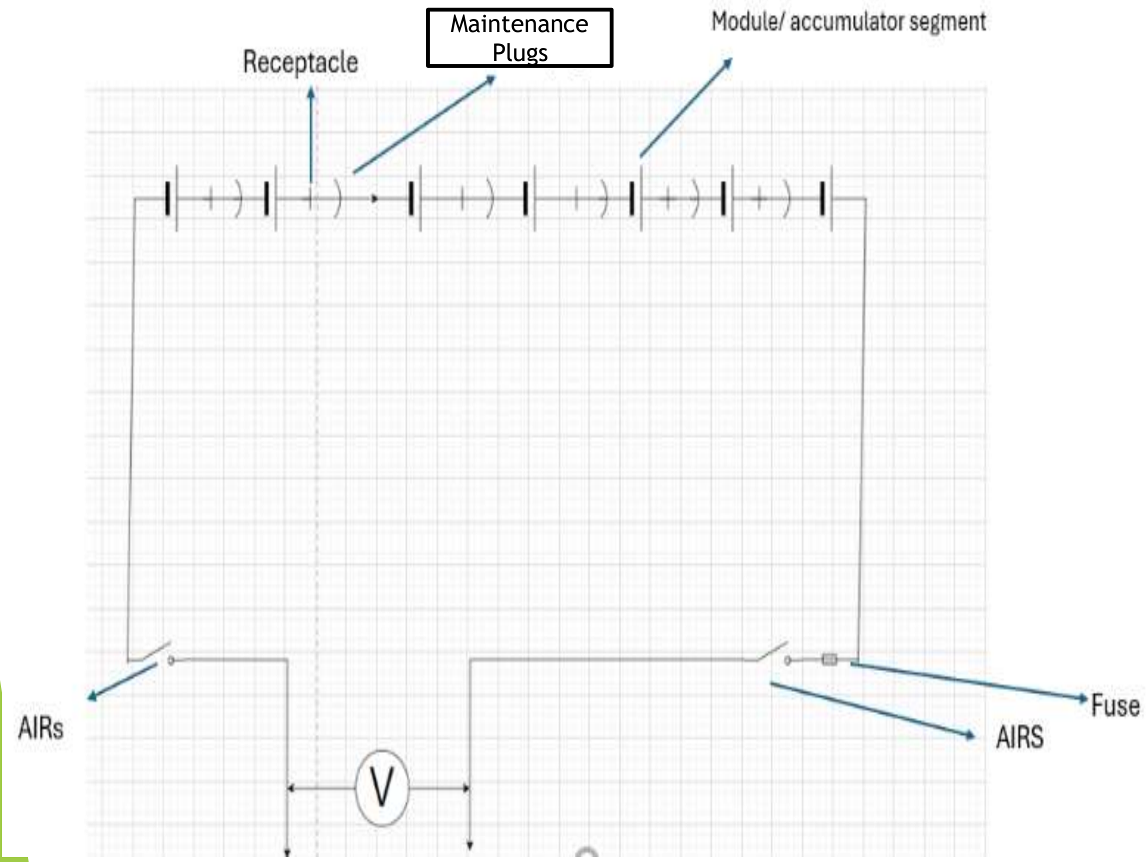




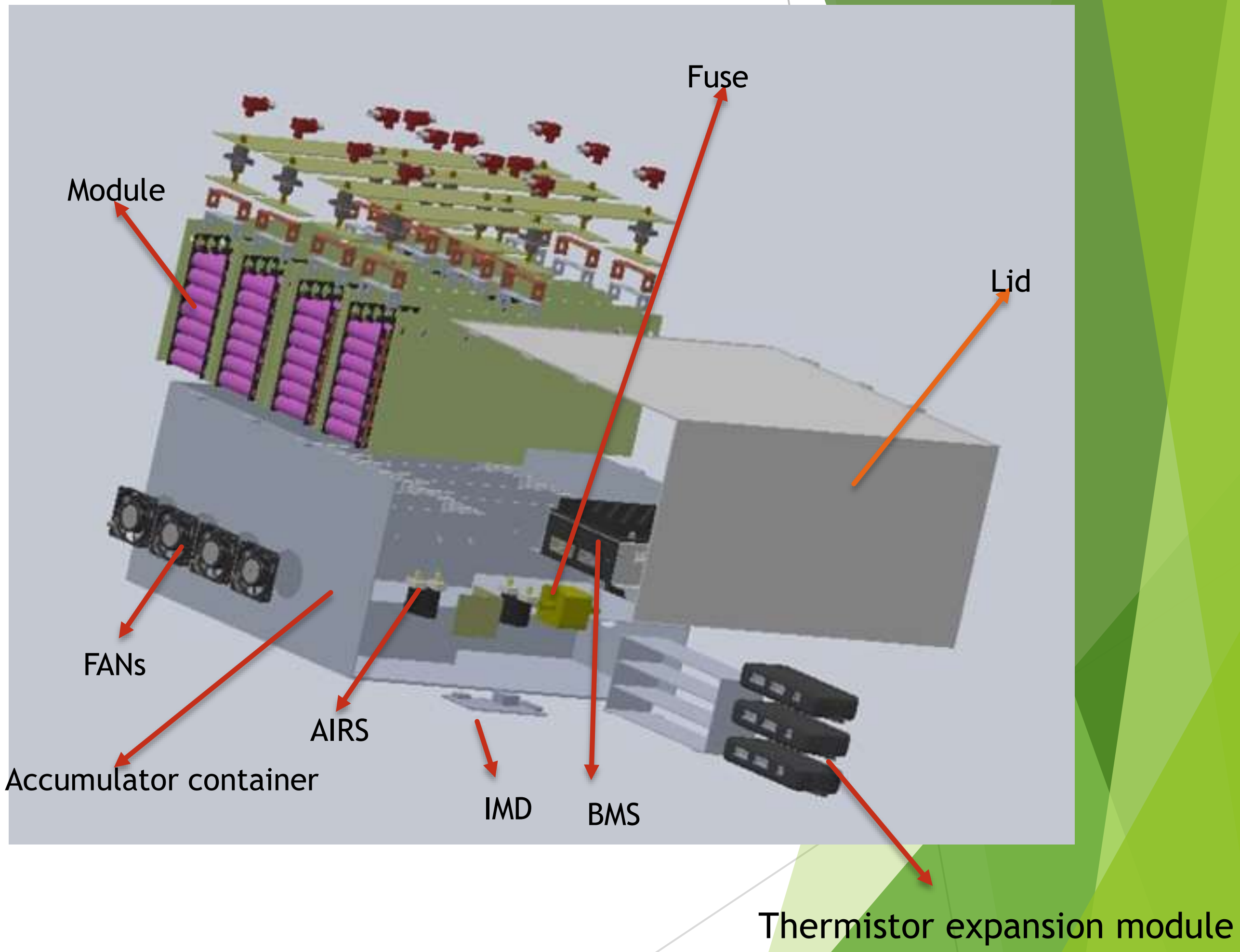


Powertrain

Accumulator



High Voltage path and connections of Accumulator



Exploded view of Accumulator assembly

Powertrain

BMS selection

Battery Protection

Against:
- Over-charge
- Over-discharge
- Overcurrent
- Isolation Faults
- Cell Short Circuits
- Extreme Temperatures

Battery Monitoring (charging and discharging)

Parameters like:
- SoC, SoH (Diagnostics)
- Cell voltages
- Battery Pack Current
- Module Temperature

Battery Optimization

Techniques like Cell Balancing improve the battery pack's:
- Life and Capacity
- Efficiency
- Reliability

Interface

- Regular Communication (reporting battery pack status)
- Logging of unusual errors or abuse events (for technician diagnosis)

BMS selection flow diagram

1. List BMS options sensor requirements by battery pack according to Formula Bharat's rule.
2. Tabulate cell balancing, temperature sensing SOC & SOH estimation method, noise reduction and accuracy.
3. Compare different methods, response time and ability to work accurately and safely above 323K temperature.
4. Accuracy and cost consideration along with availability are taken into account to select appropriate BMS

Powertrain

Orion BMS 2

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

Reliability

Operates through the highest class passenger vehicle load dump ISO 7637 Class IV (178V, 400mS, 0.5 ohm source.)

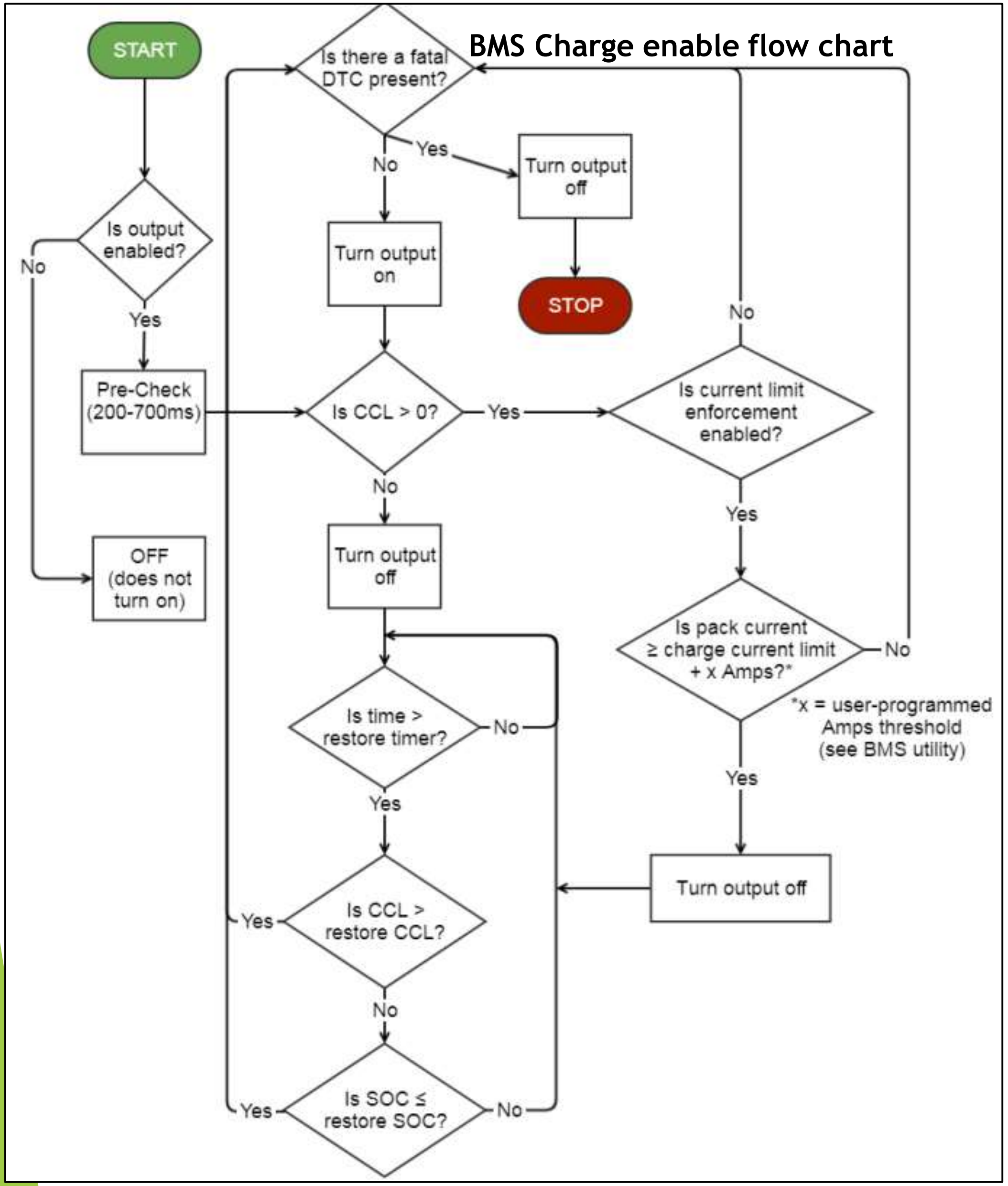
Operates through ISO 7637 “cold crank” brownouts down to 5v on input supply rail and can operate > 100mS with no power (with initial voltage of at least 12v)

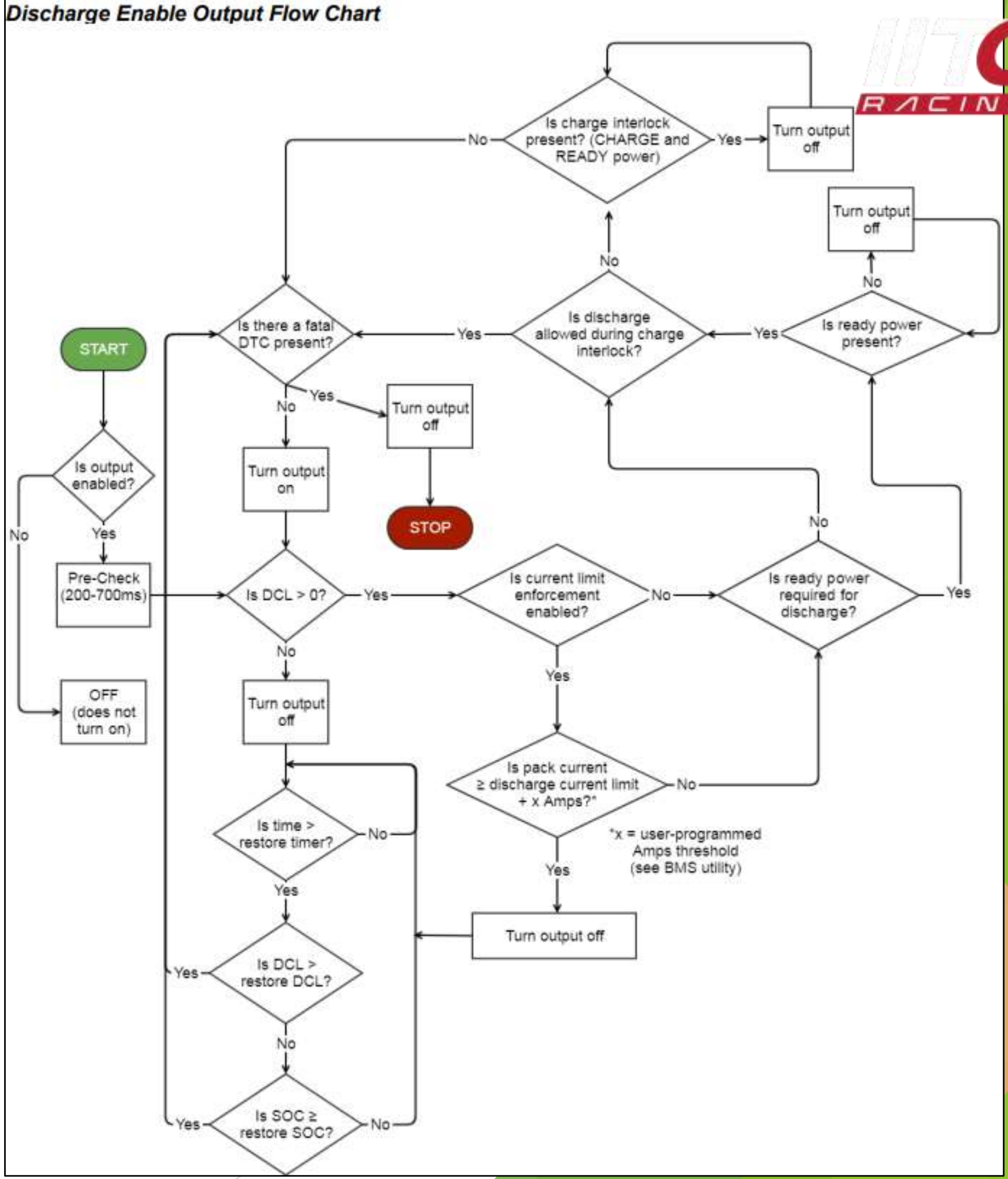
Meets EN 50498: 2010 EMC Aftermarket Vehicle Directive

Meets European UNECE Reg 10.05 (Replaced Road Vehicle Directive)

Powertrain

BMS Charge And discharge enable Flow chart





Powertrain

Charger

Charger Selection criterion

Charging Voltage = 400V
MAX. Charging Current = 20A

Charger comparison table

Charger specification	Madhura 6.6KW OBC	Dilong technology	Magenta power	Elcom
Output voltage	200 – 450V	250 - 420V	230 @50hz AC	321 - 417
Max charging currrent(A)	20	12	16	7
IP protection	IP67	IP67	IP54	IP46
Output power(kW)	6.6	3.3	3.3	3

Madhura 6.6KW OBC details

Output

Output voltage range	200V-450V (nominal 312V)
Max output current	20A
Output power	6600W@220VAC; 3300W@110VAC
Output way	CC/CV
Efficiency	>=94%
CV accuracy	±1%

Input

Input Voltage range	AC 90~265V
Frequency	47~63Hz
Input current	<=32A
Power Factor	>=0.98 @ >=1650W
Efficiency	>=93% full loading
Stand-by power consumption	<=5W
Starting inrush current	<=48A

Low voltage Output

Output way	CV
Output voltage	12V
Nominal current	5.5A
CV accuracy	±2%
Output power	<=66W
Ripple voltage coefficient	<=1%

Protection Functions

Input over-voltage protection	AC 270±5V
Input low-voltage protection	AC 85±5V
Output over-voltage protection	Stop output on exceeding highest voltage ±5V
Output low-voltage protection	Stop output when below lowest voltage ±5V
Over-temperature protection	Power start to decrease when internal temperature rise to 85°C, shut off when rise to 90°C
Output short circuit protection	Stop output
Output polarity reverse protection	yes
Grounding protection	<=100mΩ
CAN communication protection	Automatically stop output when CAN communication fails
Power-off protection	Yes

Powertrain

Other Components used in Accumulator

HVSL 1200

.Current Rating : 350A

.Voltage Rating : 700VCD

.Temperature Rating : -40 to 65 C

.Number of Contacts : 2

Eaton 200A Bolted Tag Fuse, MT, 500 V dc, 690V ac, 85mm

.Current rating : 250A

.Voltage Rating : 500VCD

.Diameter : 30mm

.Fuse Rating : Fast Blow

High Voltage wires

.Core material : Copper

. Current Rating : 600A

.Voltage Rating : 1.1KV

.Nominal Area : 50 sq mm

Hall effect current sensor

.No of Channel : 2

.Operating voltage: 5VDC

.Current measuring range
 . Channel 1 : -75 to 75 A
 . Channel 2 : -500 - 500 A

.Operating Temperature : -40 to 125 C

Powertrain

Other Components used in Accumulator

Thermistor Expansion Module

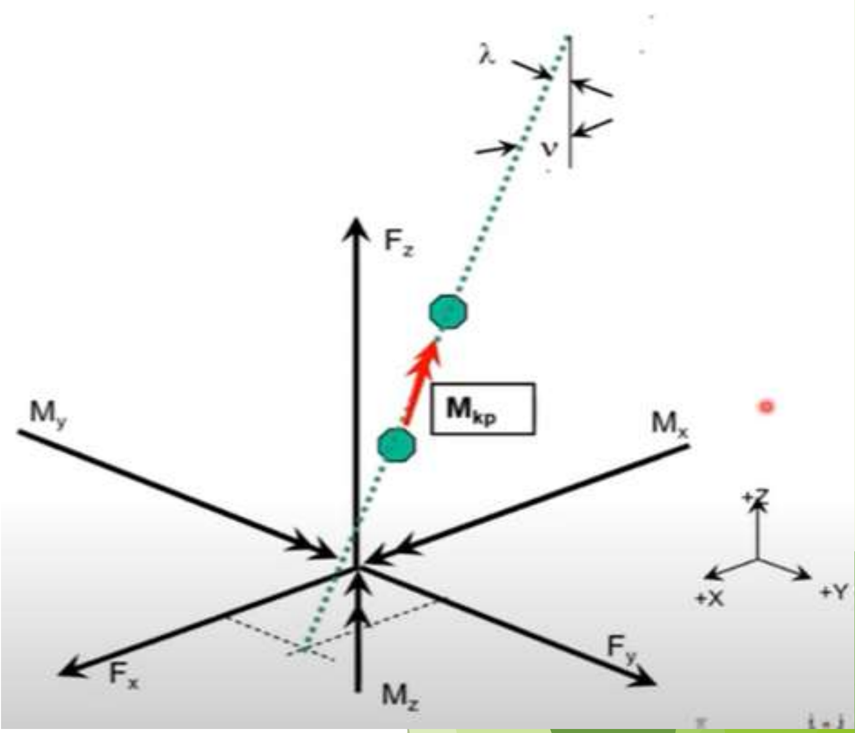
Data Logging Display Module

CANdapter

Insulation Material FR4

Dielectric Strength=20MV/m
Thickness=1.6mm
Temperature Resistance=140-150 degree Celsius
Tensile strength=450MPa
Flame retardant UL94V-0 rated

Kingpin Moment calculation



From lateral forces at contact patch

$$M_{kp,Fy} = -(F_{yr} + F_{yl}) * n_m * \cos \boldsymbol{\nu} * \cos \delta$$

From vertical force at contact patch

$$M_{kp,Fz} = -(F_{zr} + F_{zl}) * \sin \boldsymbol{\lambda} * [r_s * \sin \boldsymbol{\delta}] + (F_{zr} - F_{zl}) * \sin \boldsymbol{\nu} * [r_s * \cos \boldsymbol{\lambda}\}$$

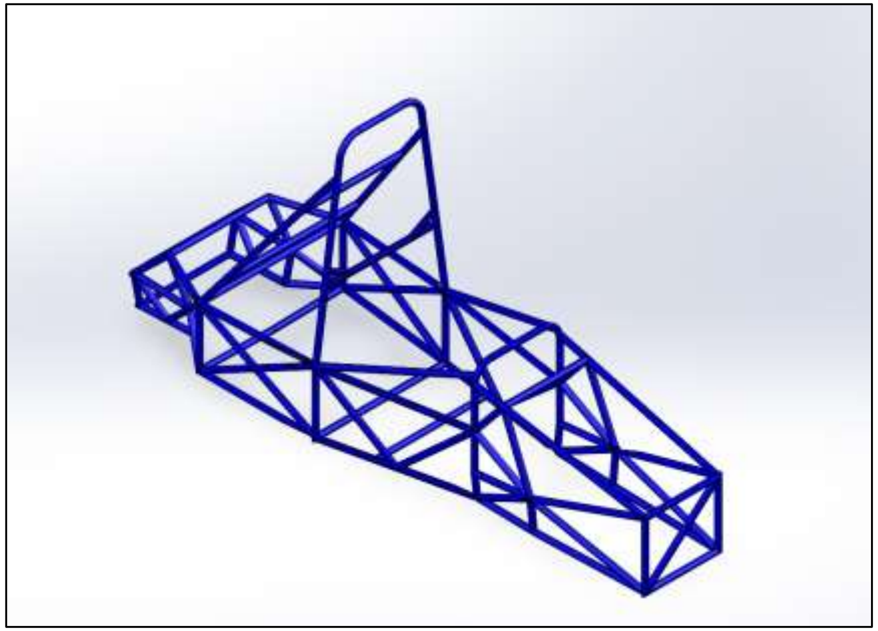
From aligning torque at contact patch

$$M_{kp,FMz} = -(M_{zr} + M_{zl}) * \cos(\boldsymbol{\lambda}^2 + \boldsymbol{\nu}^2)^{0.5}$$

From steering a 'locked tire'

$$M_{kp,sp} = -\mu * \left(F_{z,r} + F_{z,l}\right) * \{[r_s * \cos(\lambda)]^2 + [n_T * \cos(v)]^2\}^{0.5}$$

Structure



Structural Integrity & Safety

Ensuring driver safety by meeting FSAE crashworthiness standard

-Designing to withstand high-stress areas with a safety factor of 1.5.

Material Selection

Using AISI 4130 chromoly steel for its high strength and fatigue resistance.

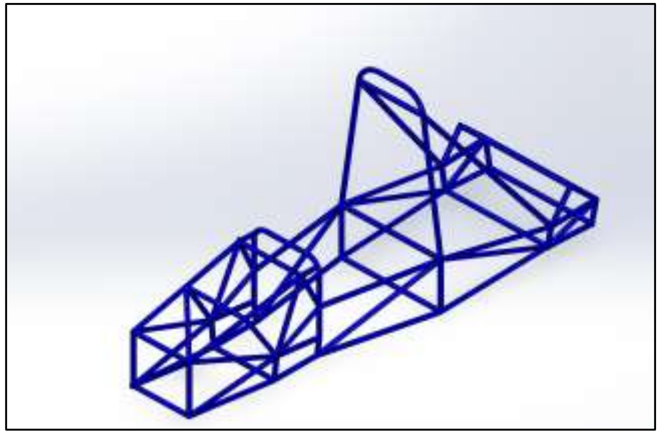
Design Simplicity

Prioritizing minimal complexity and ease of manufacturing.

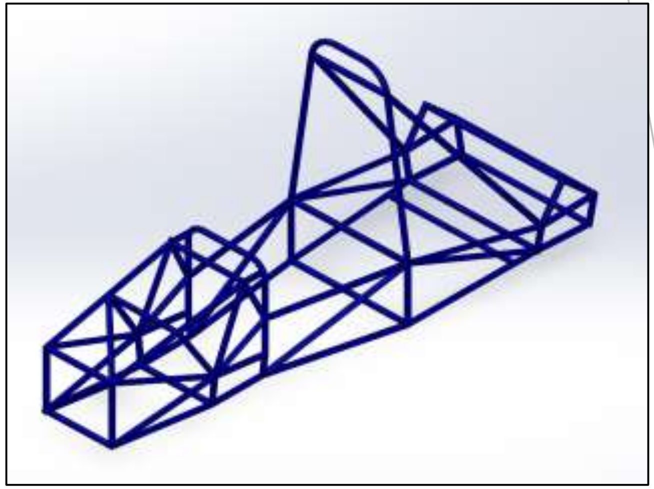
Performance Optimization

Ensuring optimal performance with simple, efficient design.

Structure



Iteration-1

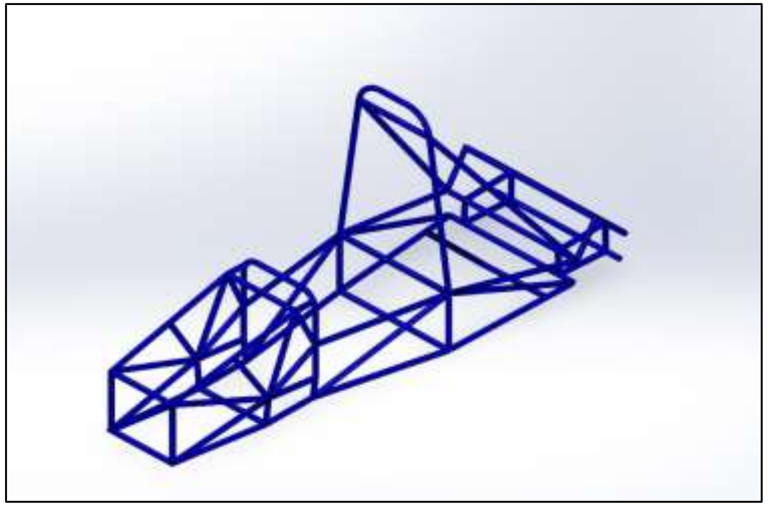


Iteration-2

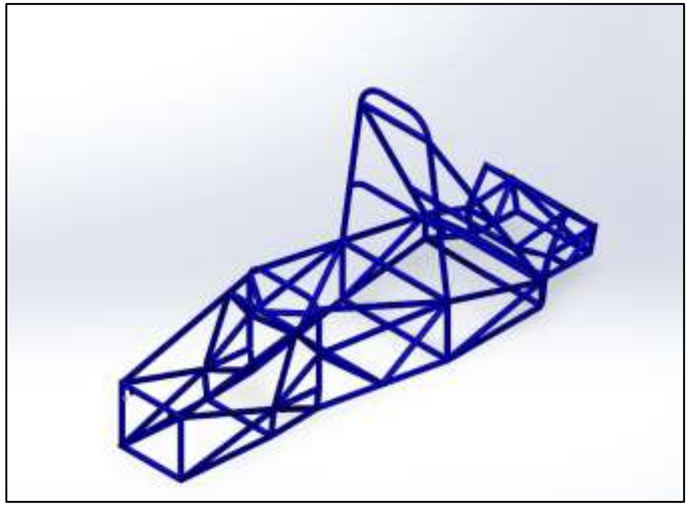
-Standard FSAE chassis
-Basic design to clear all chassis rule of rulebook

-The width of the accumulator container region has been increased to facilitate the easier removal and replacement of the battery pack.
-Along with the accumulator container rear suspension box width was also increased

Structure



Iteration-3

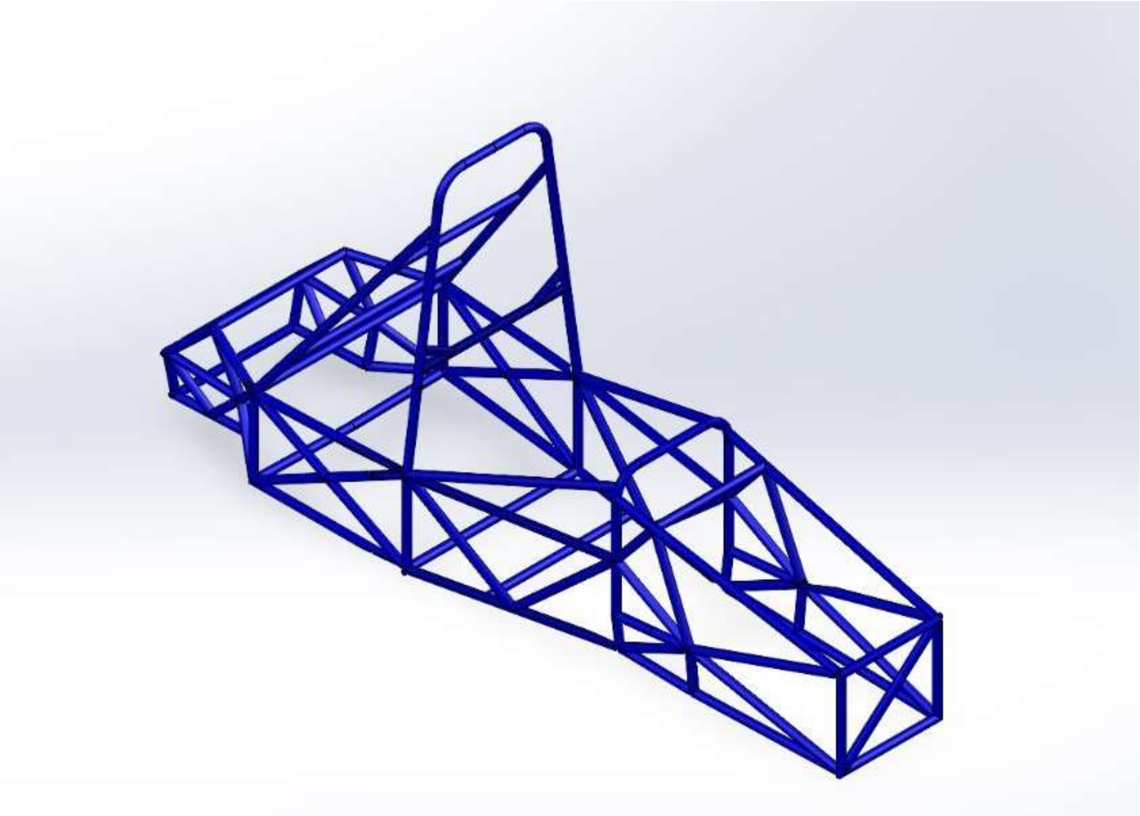


Iteration-4

*Rear suspension box width which was compromised in last design is changed to reduce the width

*Due to low rigidity of last design at rear suspension box new rod is added to reduce width without compromising torsional rigidity
*Increased height of SIS and Rear ACPSTS for better protection from impacts

Structure



Final-design

*New Shoulder harness design was included
*Increased diagonal rods for high torsional rigidity
*Final weight of design came out to be 63.3Kg

Structure:

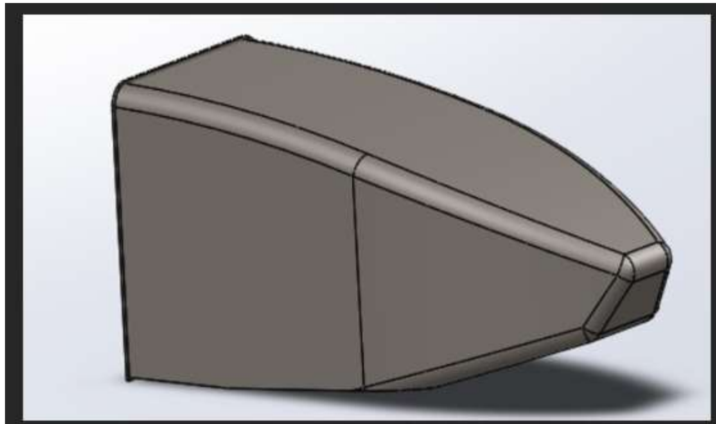
- Nose cone:

Literature survey for different shapes for nosecone

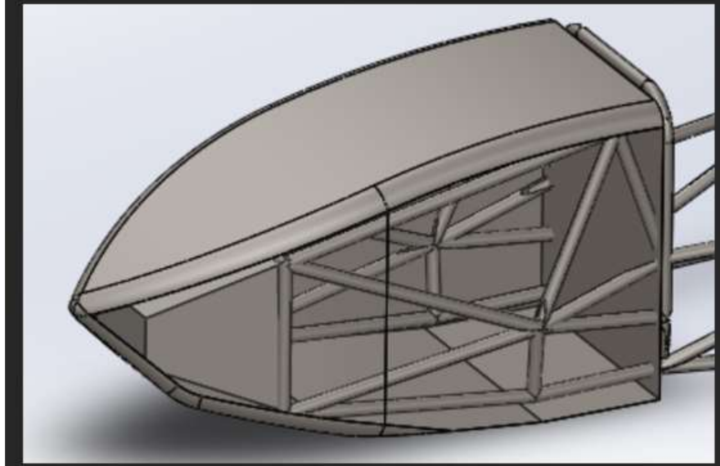
CAD modelled different nose cone shapes

Simulating all designs to finalize the efficient one

Structure:



Nosecone version 1



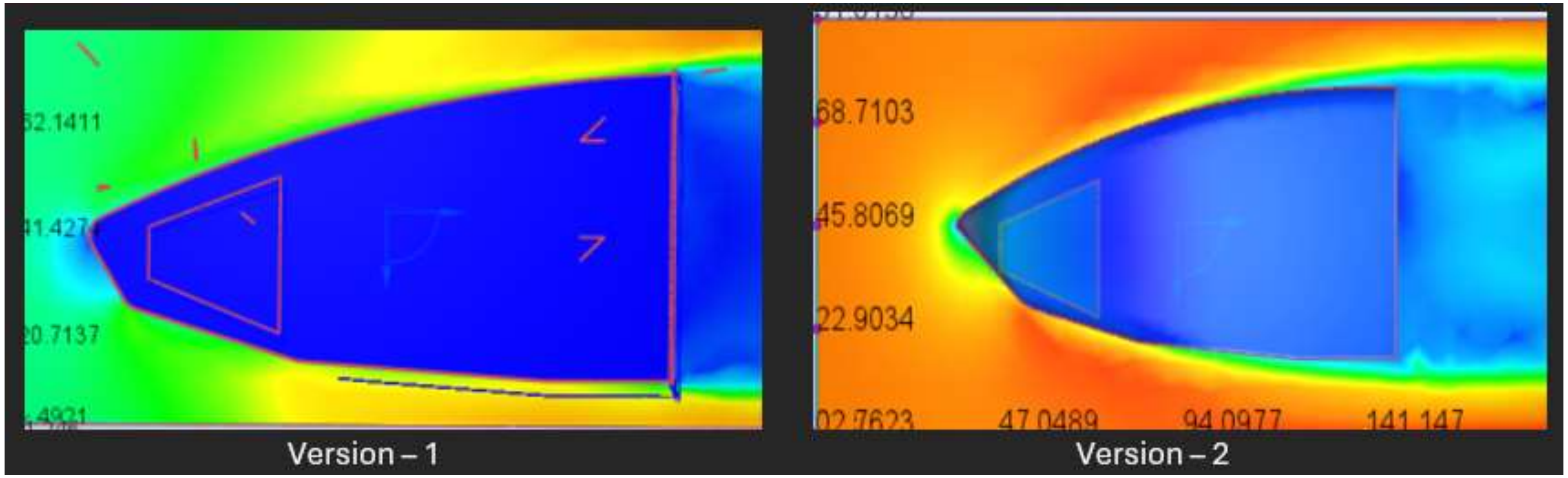
Nosecone version 2

*Two different shapes were simulated,
-Pointy end
-Stubbed end
*"Pointy end" version is aerodynamically efficient,
*But "stub end" version is easier to manufacture.

*CFD simulations were carried out in same conditions for both of the version.

*Simulations were completed at 400 iteration for both of the versions.

Structure



The area of stagnation points is much larger in version 1 compared to version 2, which is why version produces lot of drag and also there aren't flow separation in both of the versions of nosecone, which is very good because it's a huge factor in drag

Structure

Drag Force(Fy)	96.399N	Cd=2.37
Downforce(Fz)	14.53N	Cl=0.26
Lateral Force(Fx)	36.87N	-

Force data of version 1

Drag Force(Fy)	4.73832	Cd=0.116
Downforce(Fz)	-1.79644	Cl=0.033
Lateral Force(Fx)	-0.585479	-

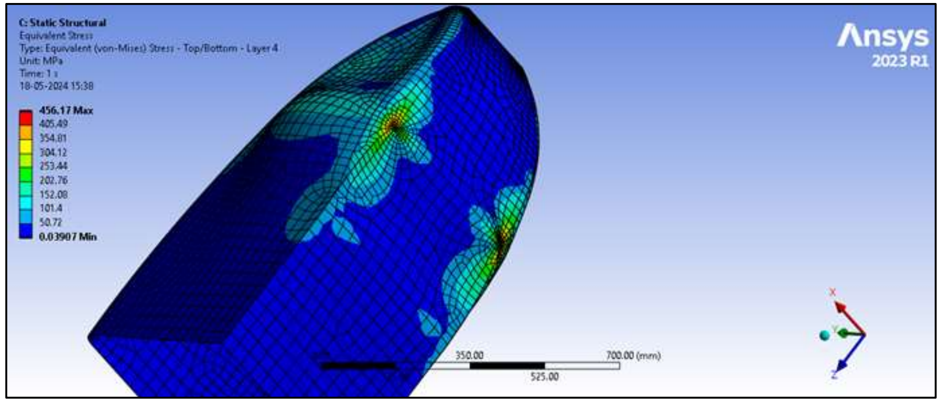
Force data of version 2

From the results, the "stub end" version proved to be more aerodynamically efficient for the velocity range of the vehicle.

Using these values and impact force values, composite analysis was done to determine the material.

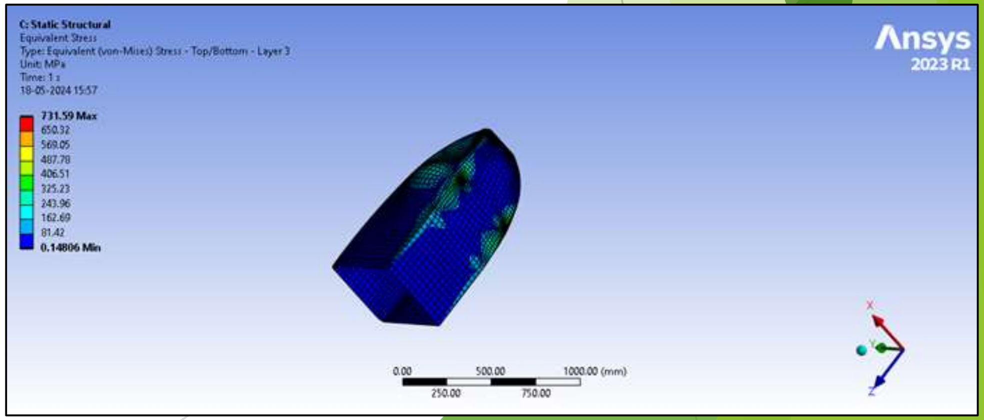
Structure

- After simulating the model with both carbon fiber and glass fiber, 3 layers of carbon fiber and 4 layers of glass fiber was rendered sufficient to withstand the forces.



4 layers of glass fiber/epoxy composite
biax, 300gsm

3 layers of carbonfiber/epoxy composite
biax, 300gsm

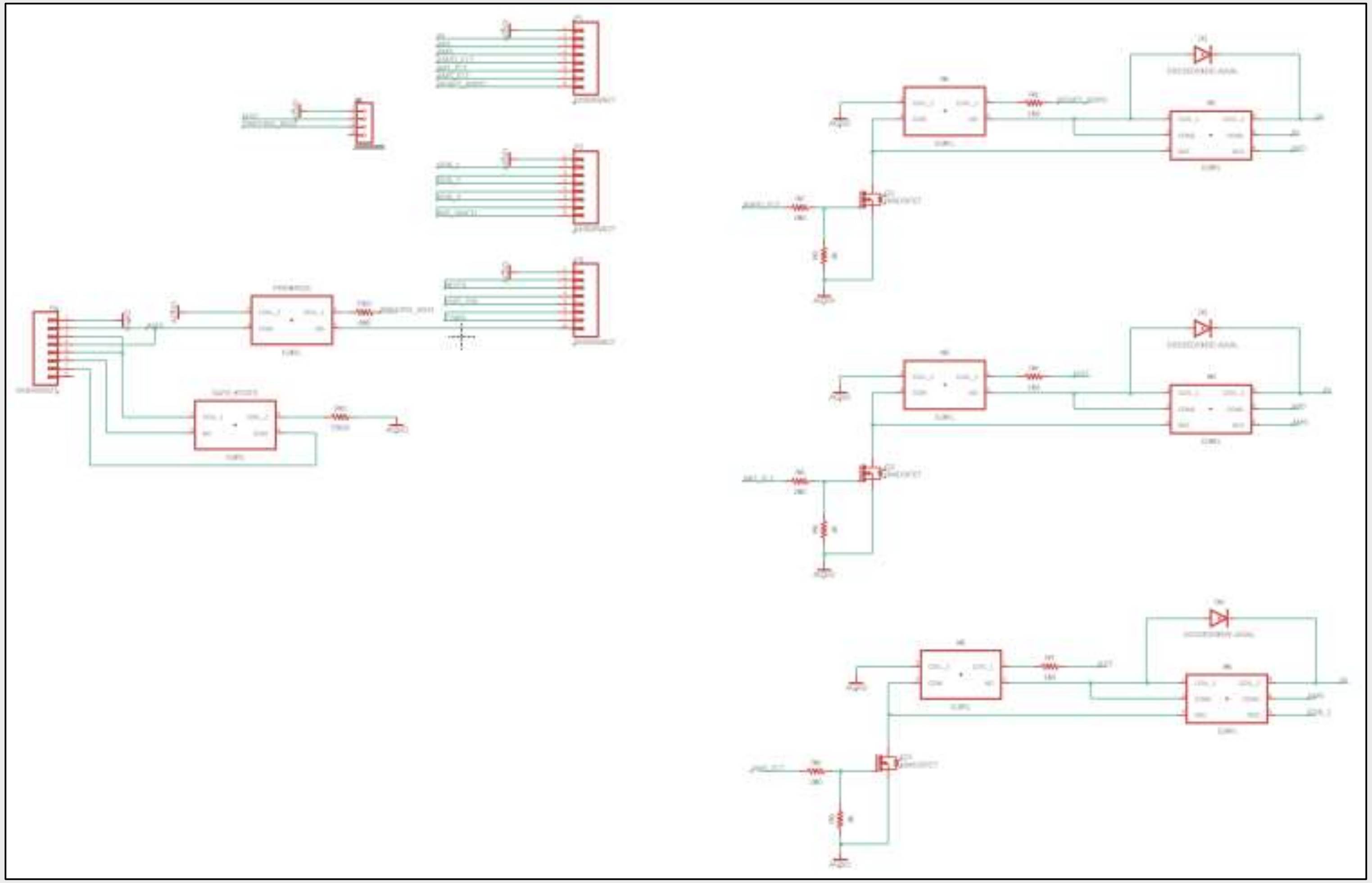


Low Voltage electronics

- The low voltage electronics consists of all the safety circuits and associated circuitry.

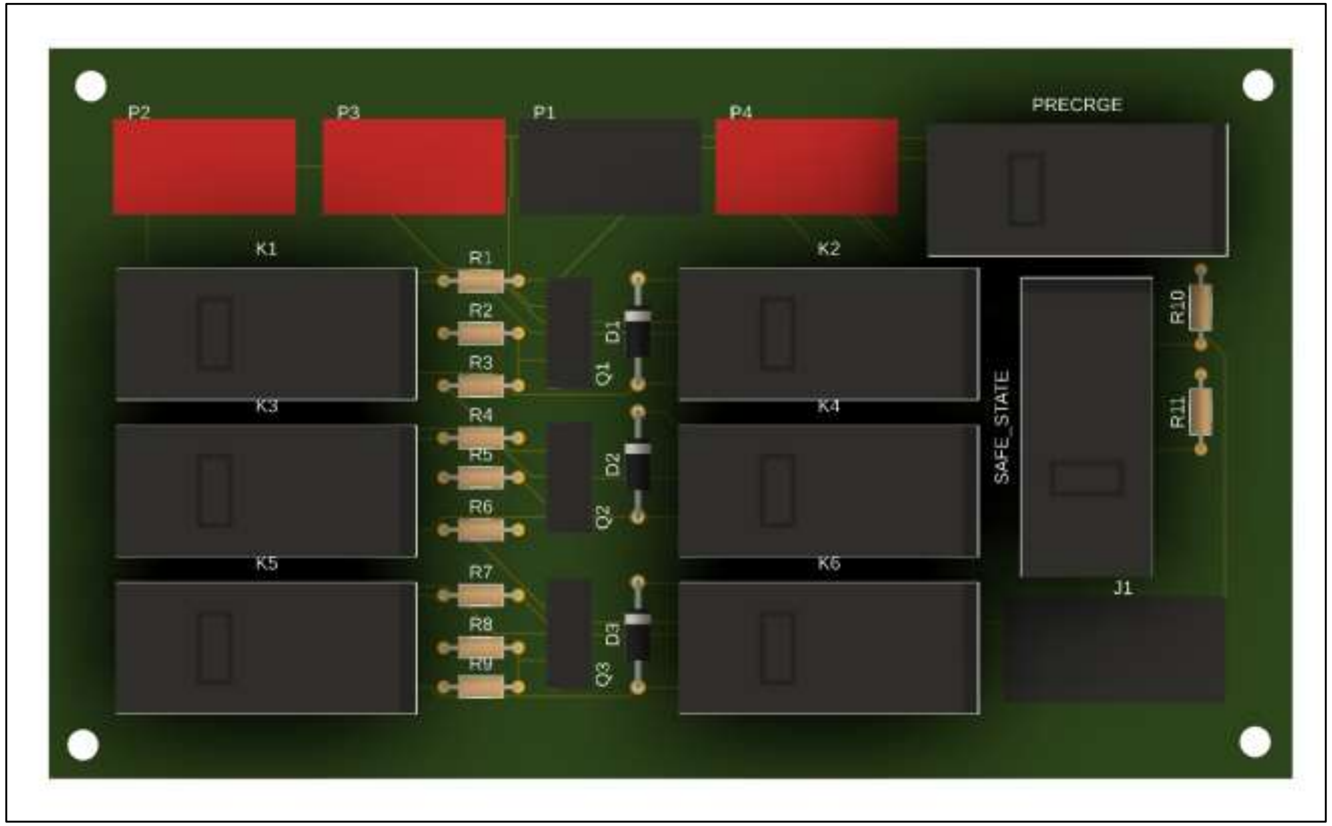
Shutdown Circuit	BSPD	RTDS
Precharge Circuit	TSAL	APPS

Shutdown Circuit



The shutdown circuit consists of the GLVMS, relays of BSPD, IMD, AMS fault detection, shutdown buttons, inertia switch, BOTS, HVD interlock, TSMS, Precharge circuitry and the AIR coils.

The final relay is used for the green light of TSAL which indicates the safe condition. Molex connectors (Mini fit Jr and Micro fit series) are used for the connections.



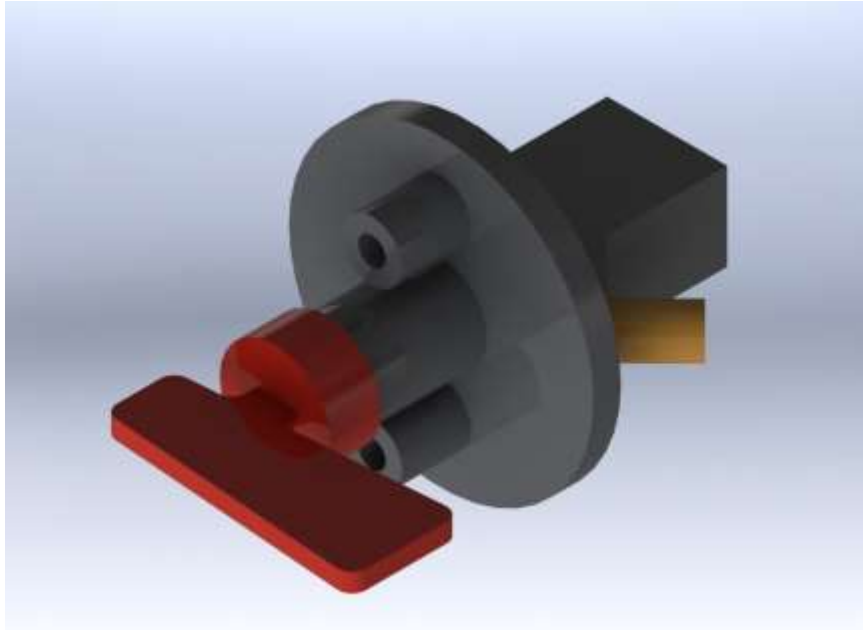
The 3D model of the shutdown circuit.

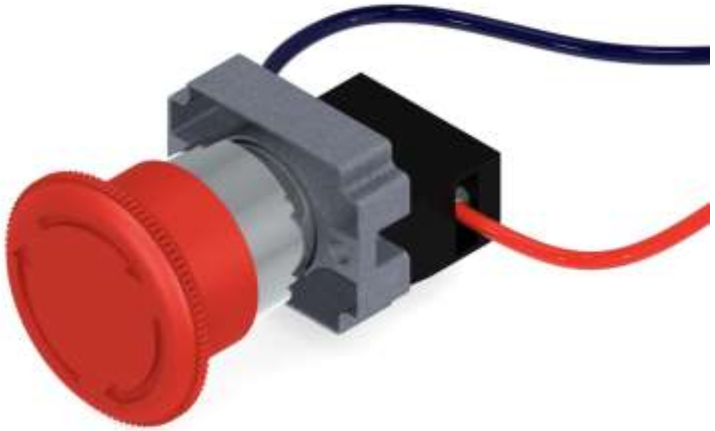
EV200 Series Contactor
(CZONKA Relay, Type

The AIRs used are KILOVAC EV200 Series Contactor (SPST-NO) with contacts rated 500 Amps, 12-900 VDC

We will be operating at 24V and the hold voltage and current are 7.5V and 0.07A respectively.

In addition to having auxiliary contacts, it is small, light weight and is fit for use here.

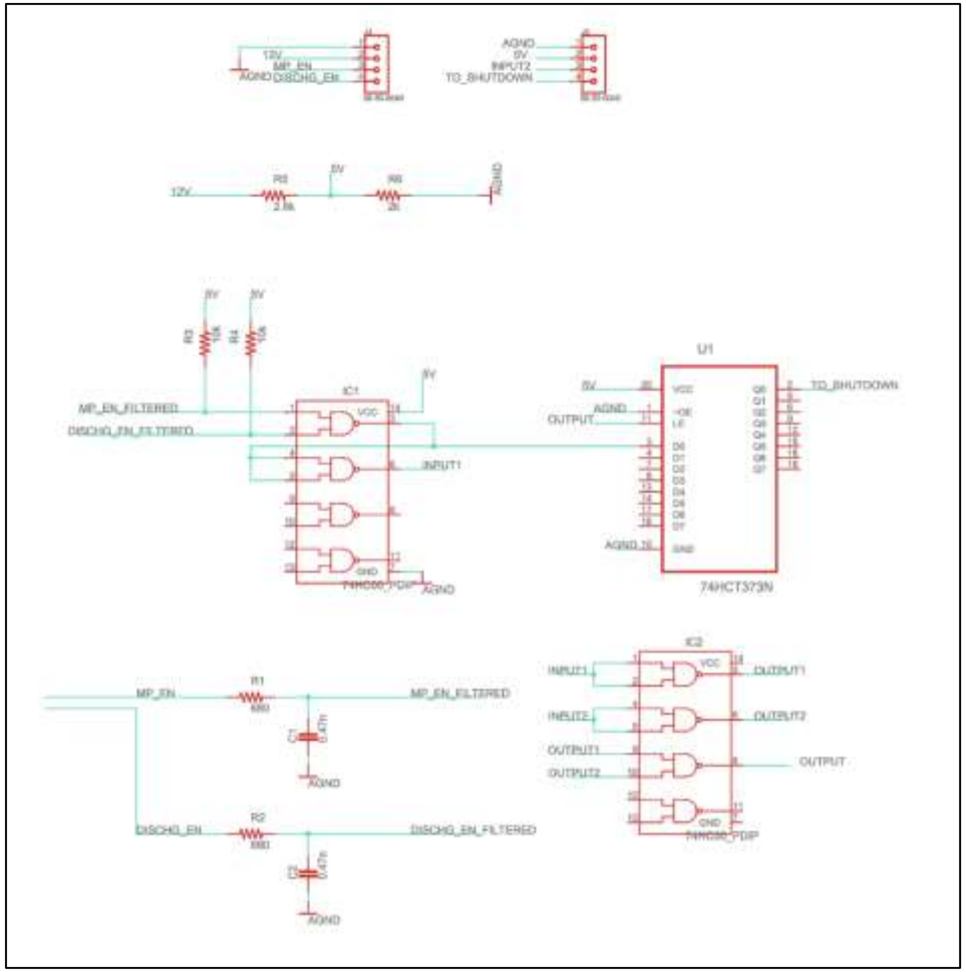


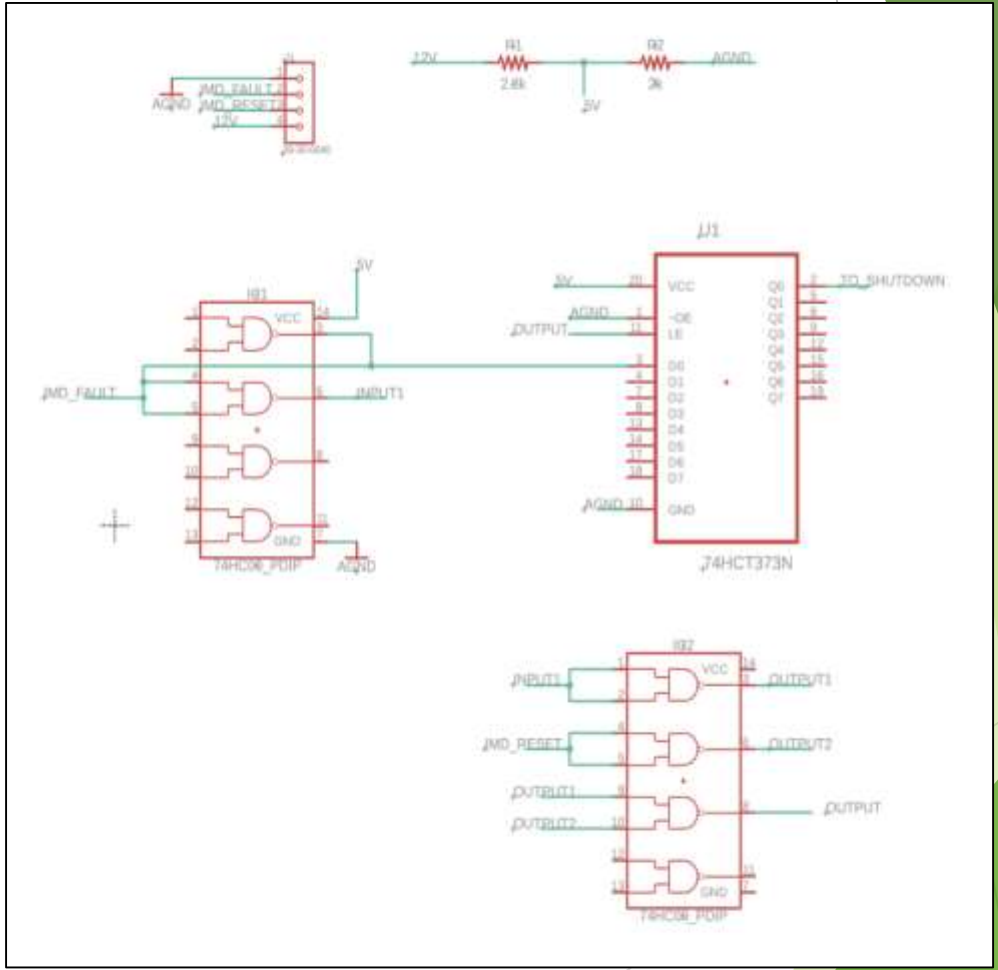


The master switches are of rotary type, with removable key. The switches are of good quality and made up of plastic.

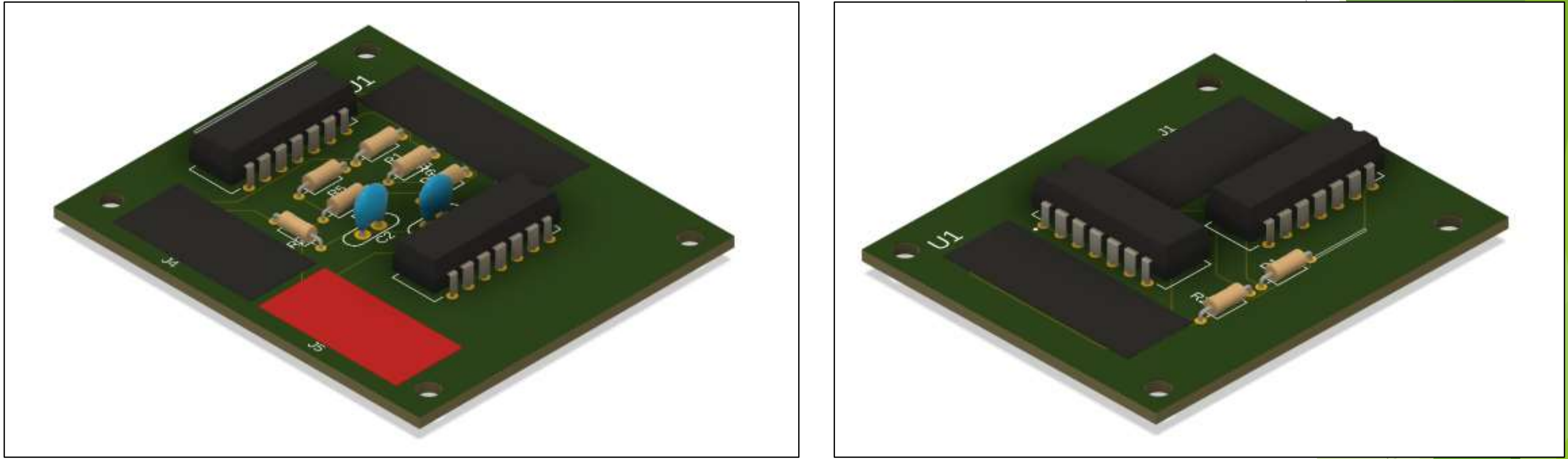
Shutdown buttons are of push-rotate type and are of good quality.

AMS and IMD fault latching circuits





The circuits above are simple latching circuits that detect and hold the error signal until reset and send them to the shutdown circuit.



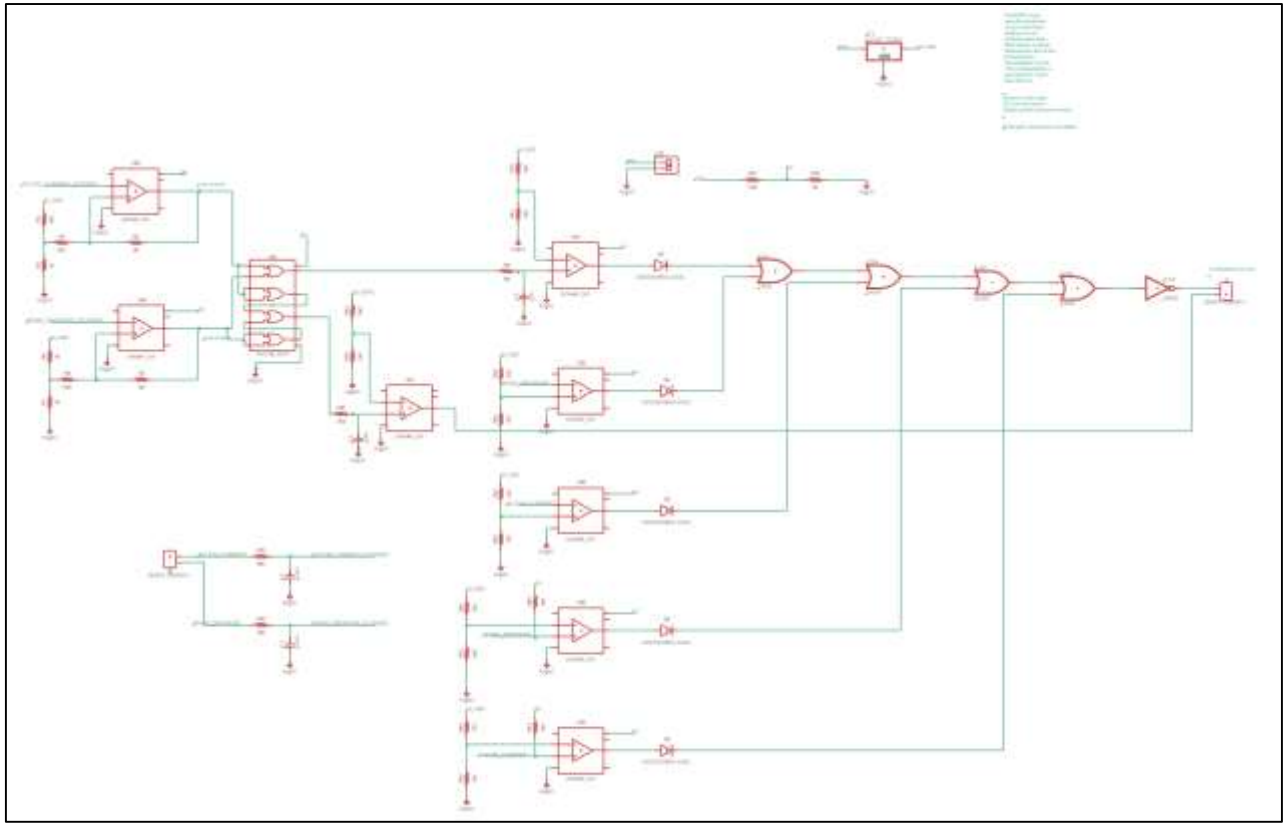
The 3D models of the respective circuits: AMS fault on the left, IMD fault on the right.

AMS/BMS and IMD

The BMS and IMD that we are using are the Orion BMS 2 and Isometer IR155.

They both are suitable for 12-24V systems. Orion BMS 2 has improved measurement for accuracy and resolution. Isometer IR155 monitors the condition of insulation on the DC side as well as on the AC motor side of the electrical drive system.

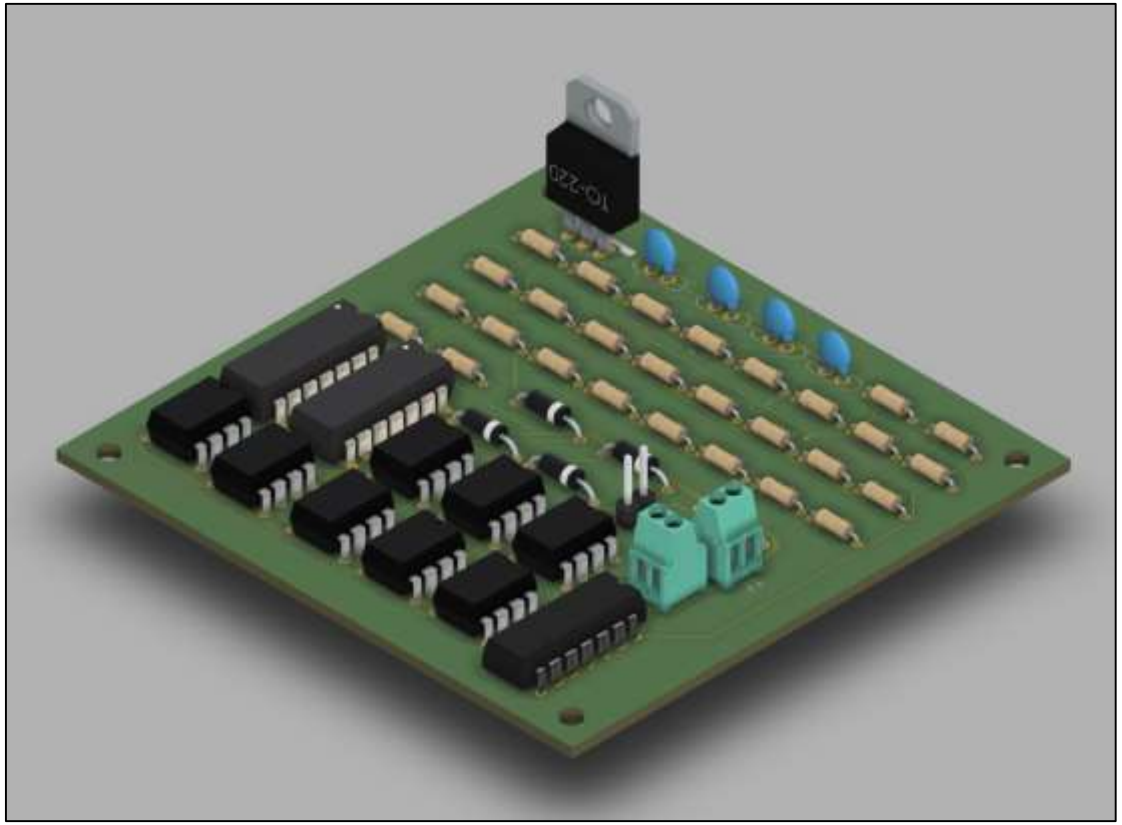
BSPD



BSPD must open the shutdown circuit when hard braking occurs whilst greater than 5kW power is being delivered to the motor for more than 500ms. A threshold of 30 bar is used for the condition of hard braking.

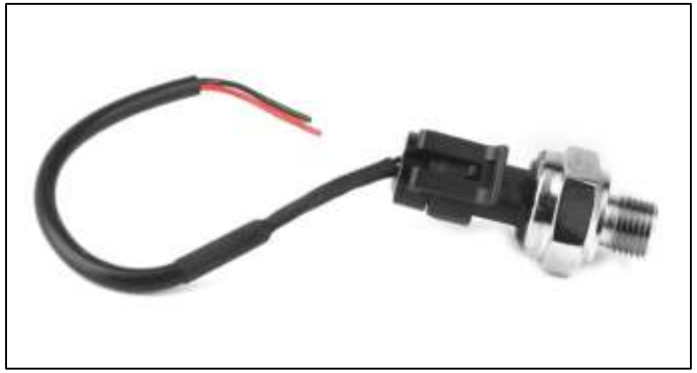
This is achieved by using a simple delay circuit made using a resistor and capacitor (of values 15kohm and 22uF).

A similar circuit is used for resetting the circuit if the implausibility no longer exists for more than 10 seconds.



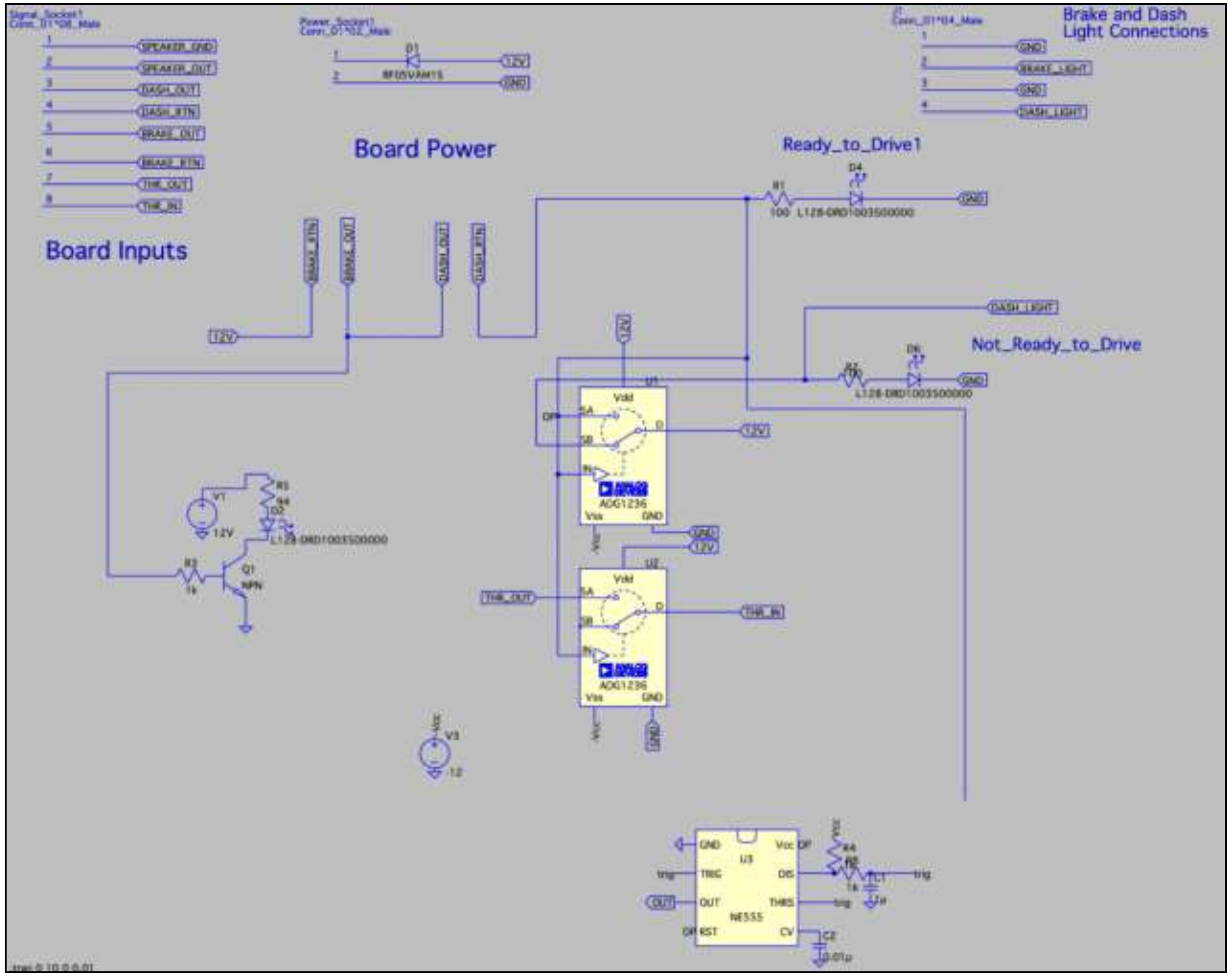
The 3D model of BSPD

The current sensor used is the Orion current sensor which is a hall effect sensor. Channel 2 is used for measuring currents here (limit: 500A). It also has low thermal sensitivity drift.



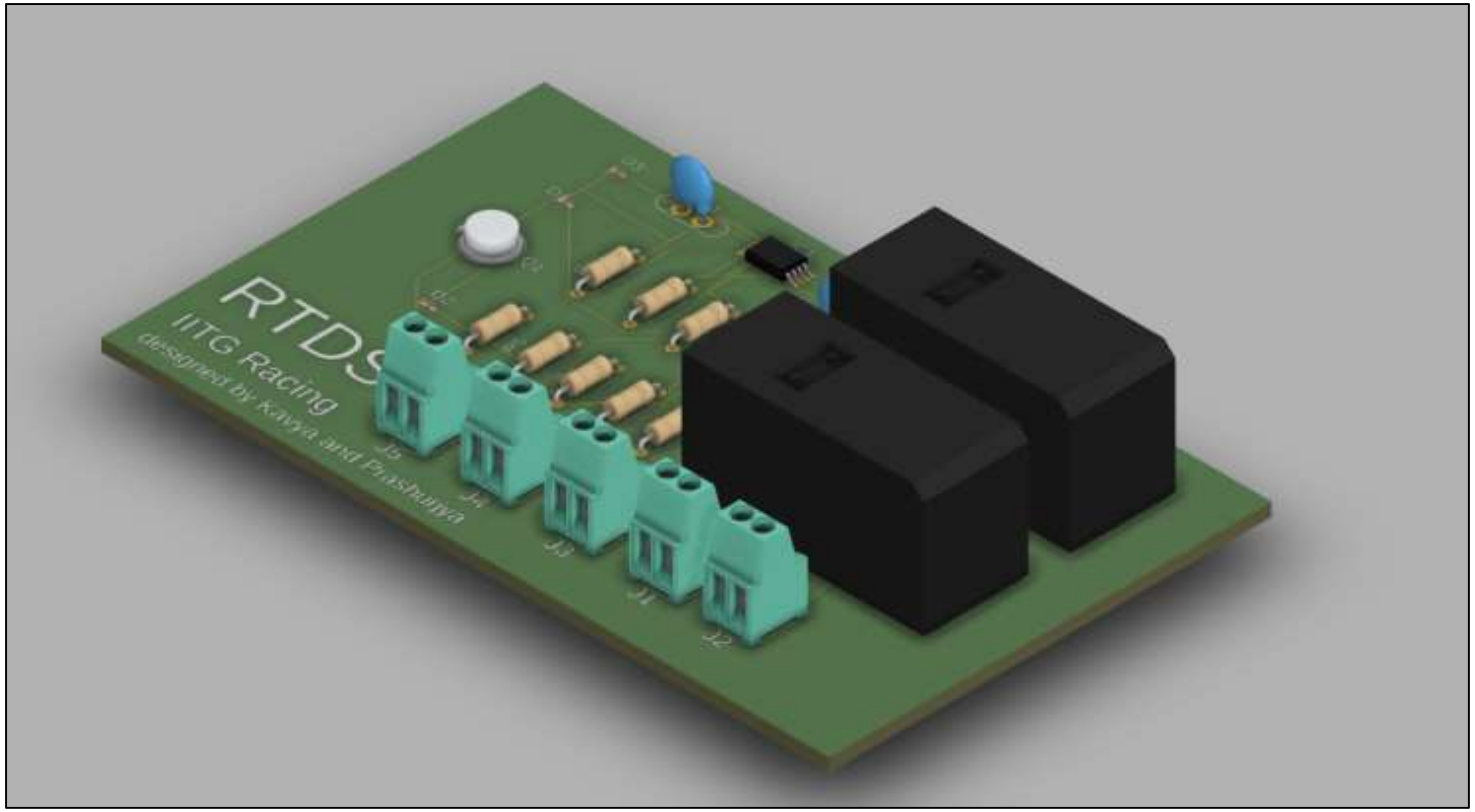
The brake pressure sensor used here is an analog pressure sensor. It is fit for measuring the brake pressure of 30 bar as it measures pressures up to 10Mpa.

RTDS



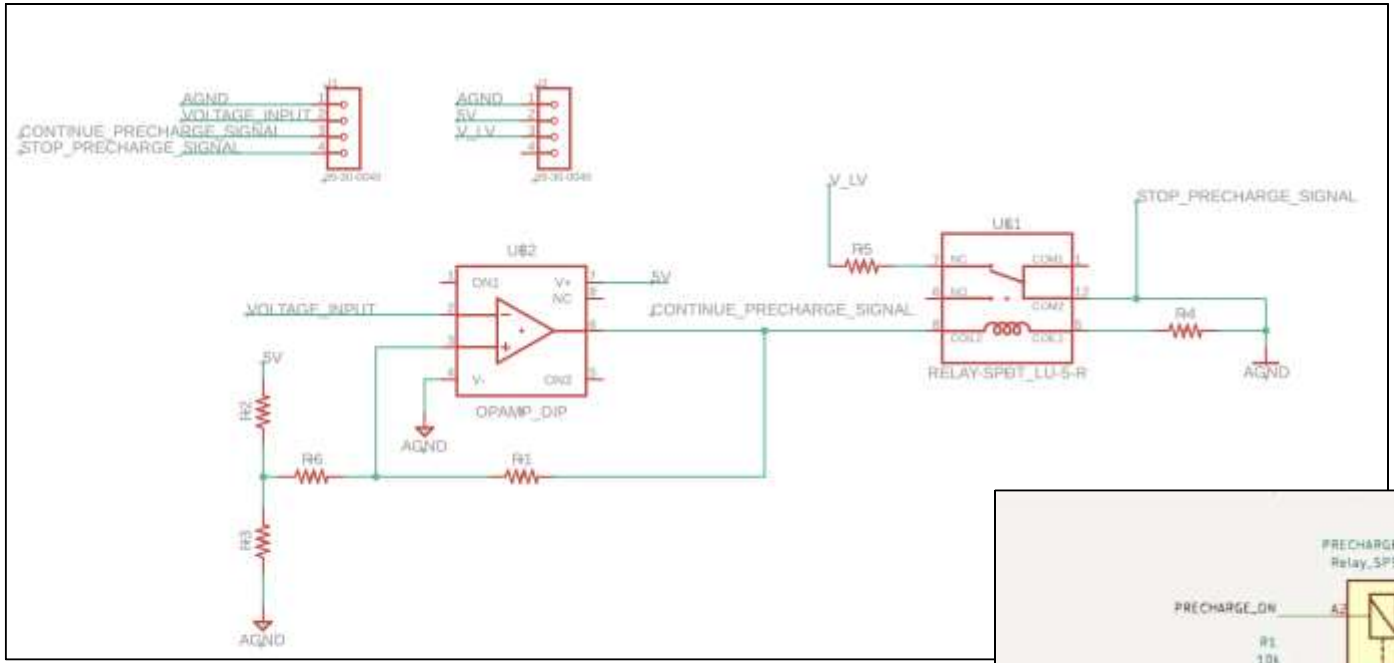
The ready to drive sound circuit is important to EVs as they don't make any sound during their start. It takes input from the brakes and the key (that is used to start the car).

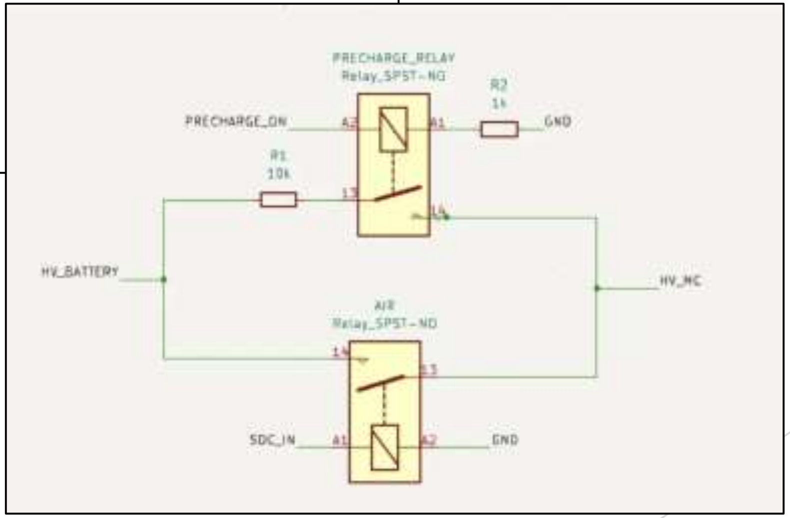
The speaker makes a characteristic sound that indicates that it is ready to drive.

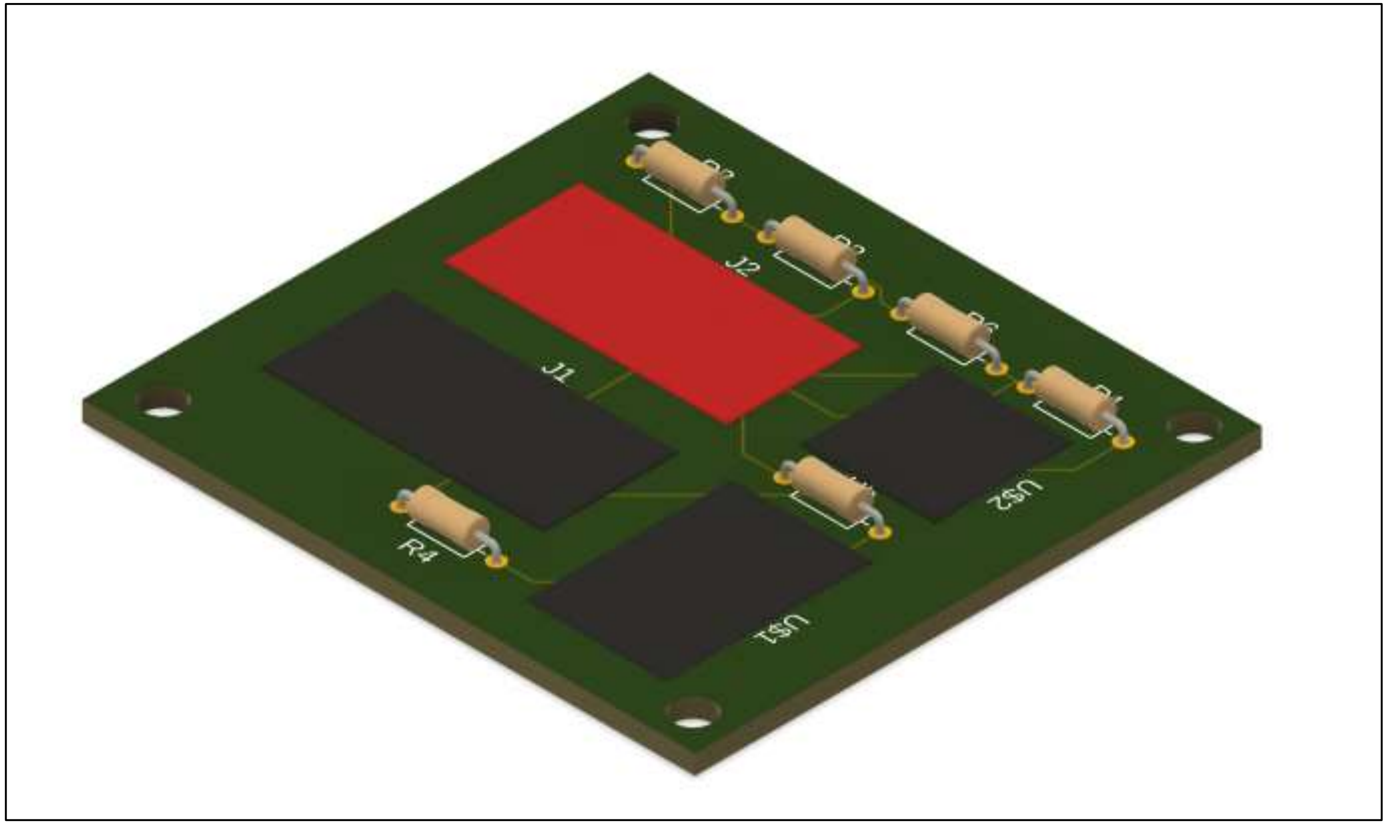


The 3D model of RTDS

Precharge Circuit







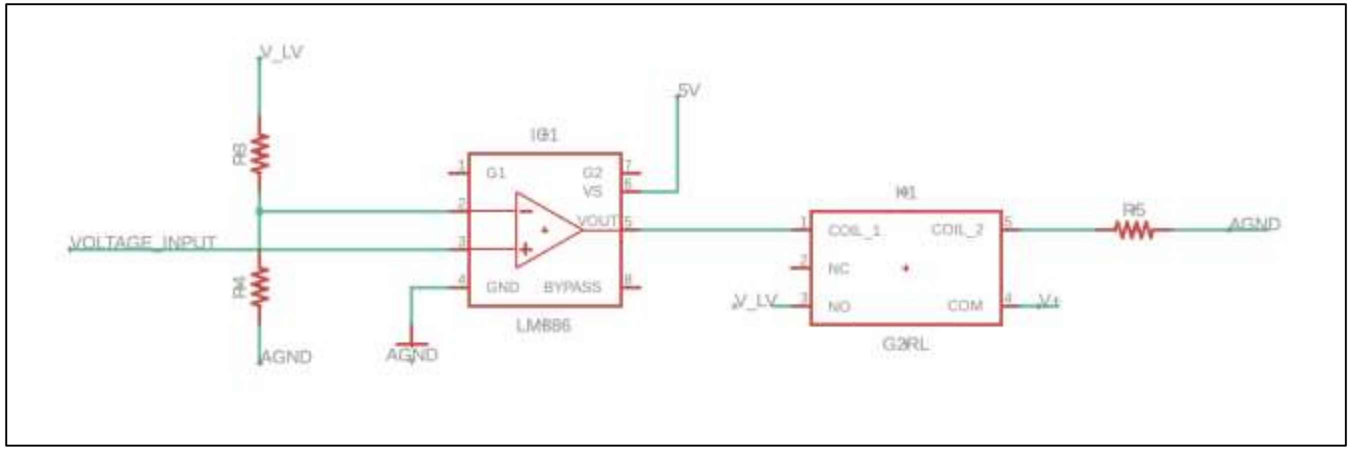
The 3D model of the precharge circuit.

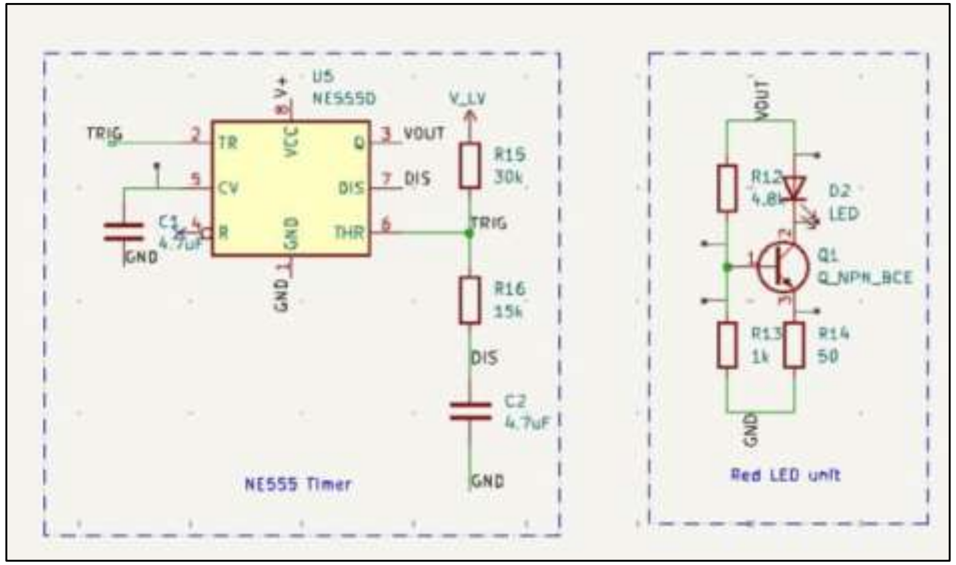
The Precharge circuit takes voltage input from the motor controller side of the HV path. As the motor controller is capacitive in nature, the capacitors need to be charged before full power can be supplied or else they would draw excessive current. Therefore, we add a resistor in path to limit the current, and after charging the capacitors to a certain level, the resistor is no longer used and is bypassed.

The precharge relay used is DCNLEV50 series high current high voltage DC contactor relay of SPST NO type.

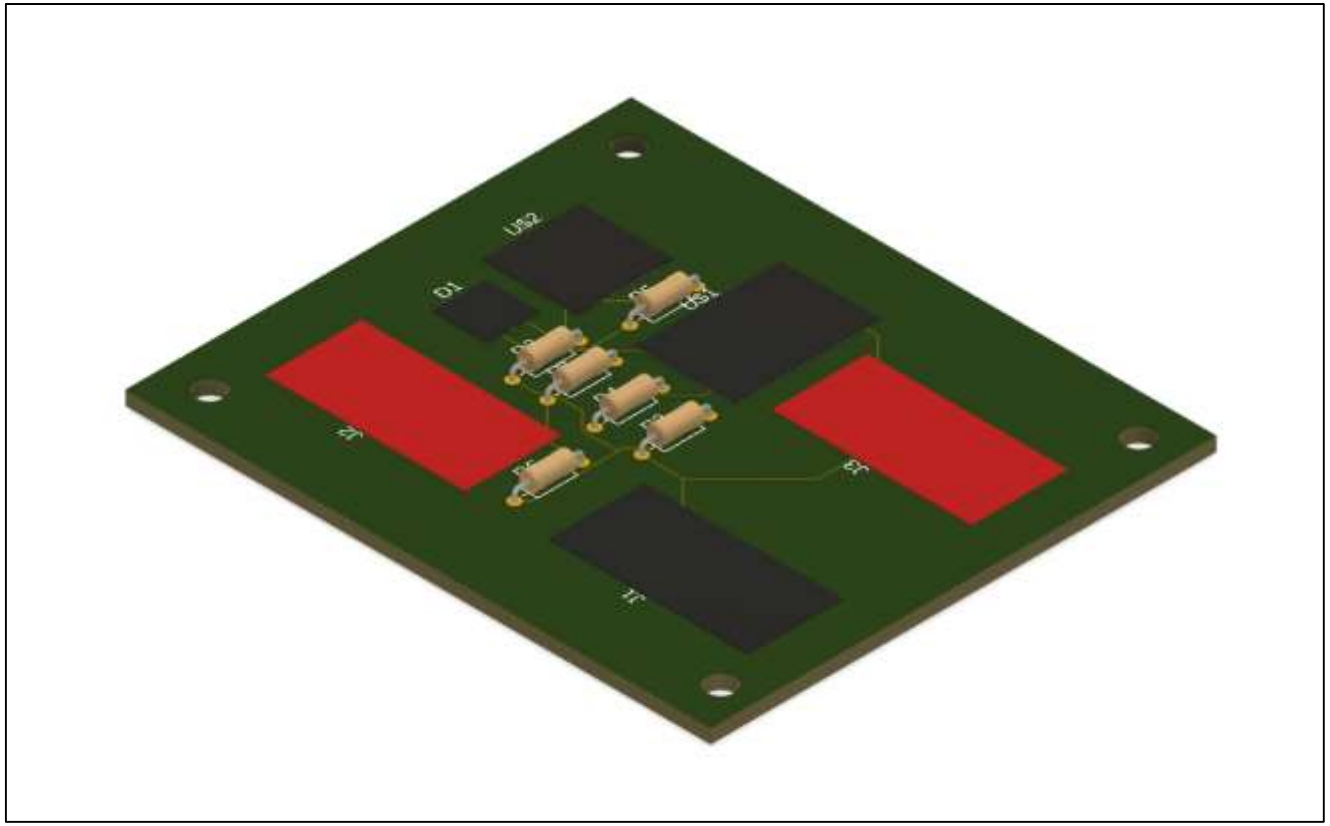
It is high voltage(900V) and high current(50A) contactor.

TSAL (Red Light)



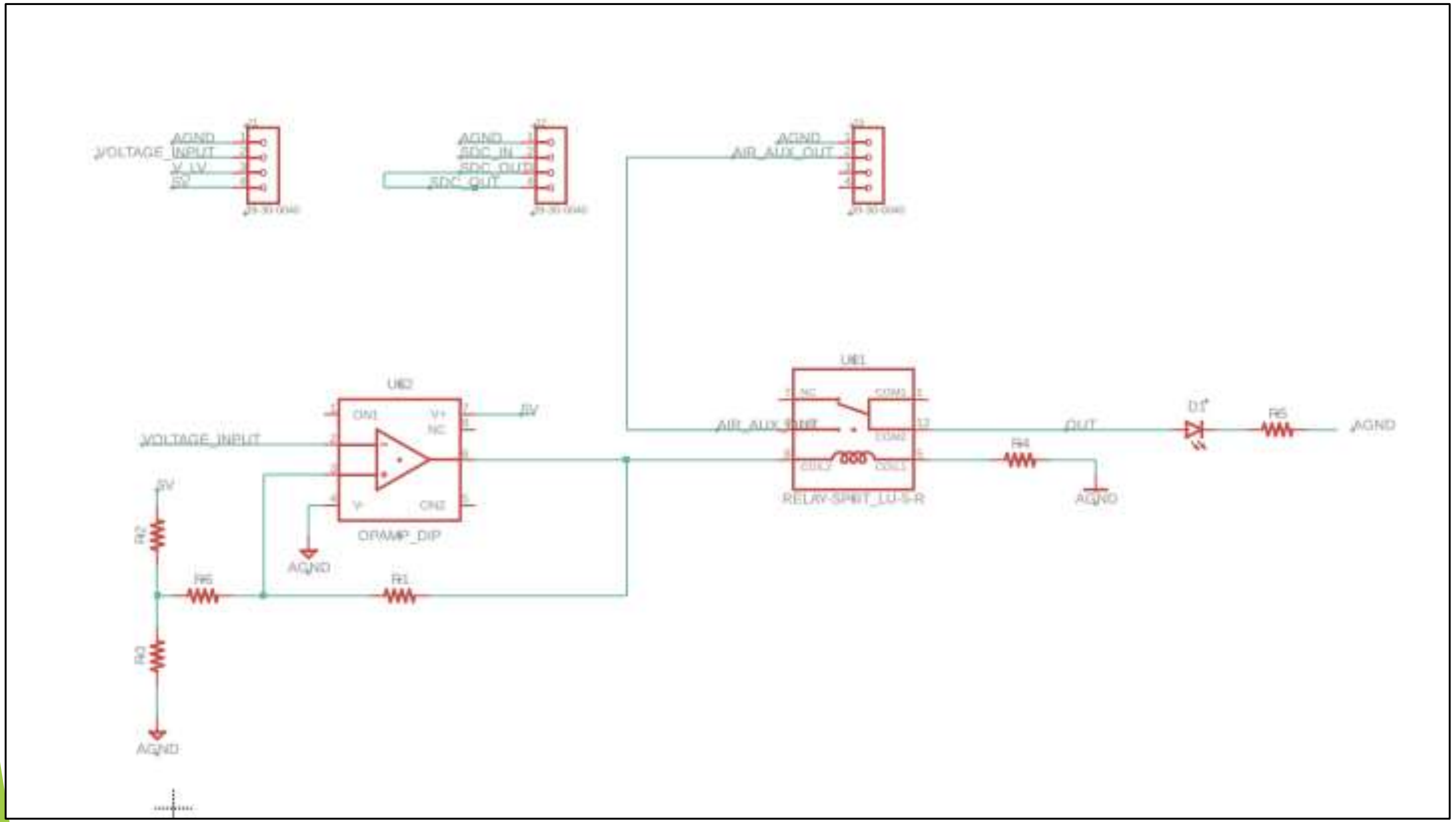


The red light should be flashing with a frequency of 5Hz if and only if LVS is active and voltage across DC link capacitors exceeds 60VDC.



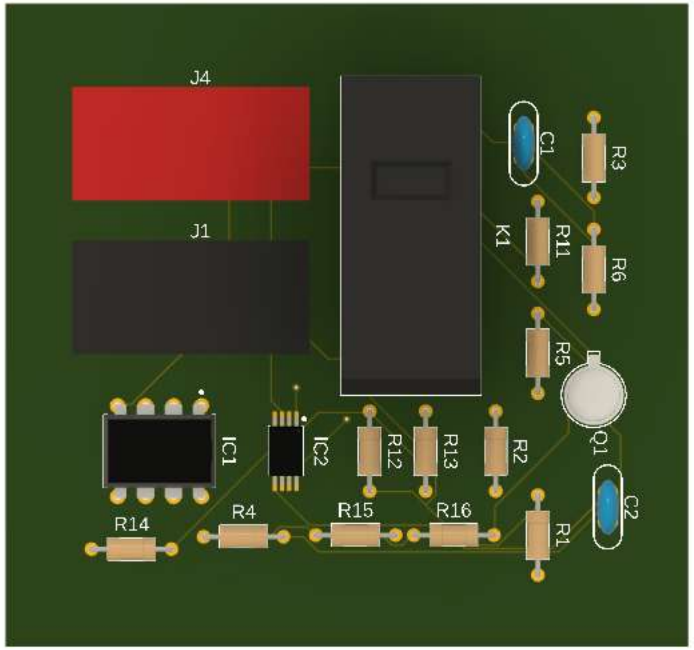
The 3D model of TSAL

TSAL (Green Light)



The green light must be on if and only if the LVS is active and the conditions below are true: All AIRs are opened, the precharge relay is opened, and the voltage across the vehicle side of AIRs inside TSAC does not exceed 60VDC.

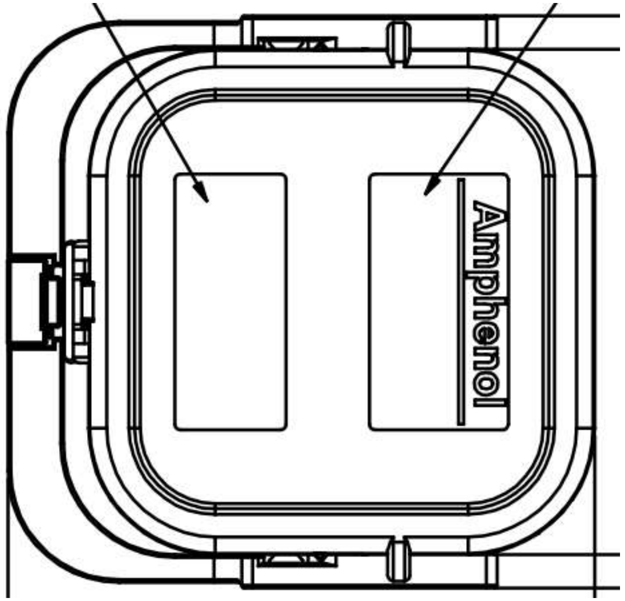
We are taking the input from shutdown circuit to check the conditions above.

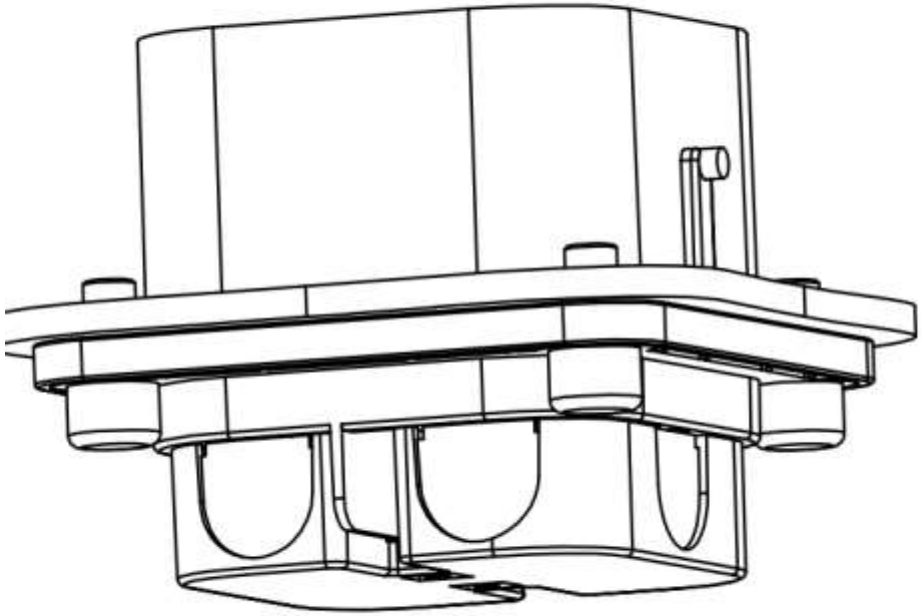


The 3D model of TSAL

HVD

- We are using Amphenol MSD HVD as it is easy to handle and operate. Its easy locking and unlocking mechanism makes it good for this purpose.





Dashboard

- The dashboard has various important components:

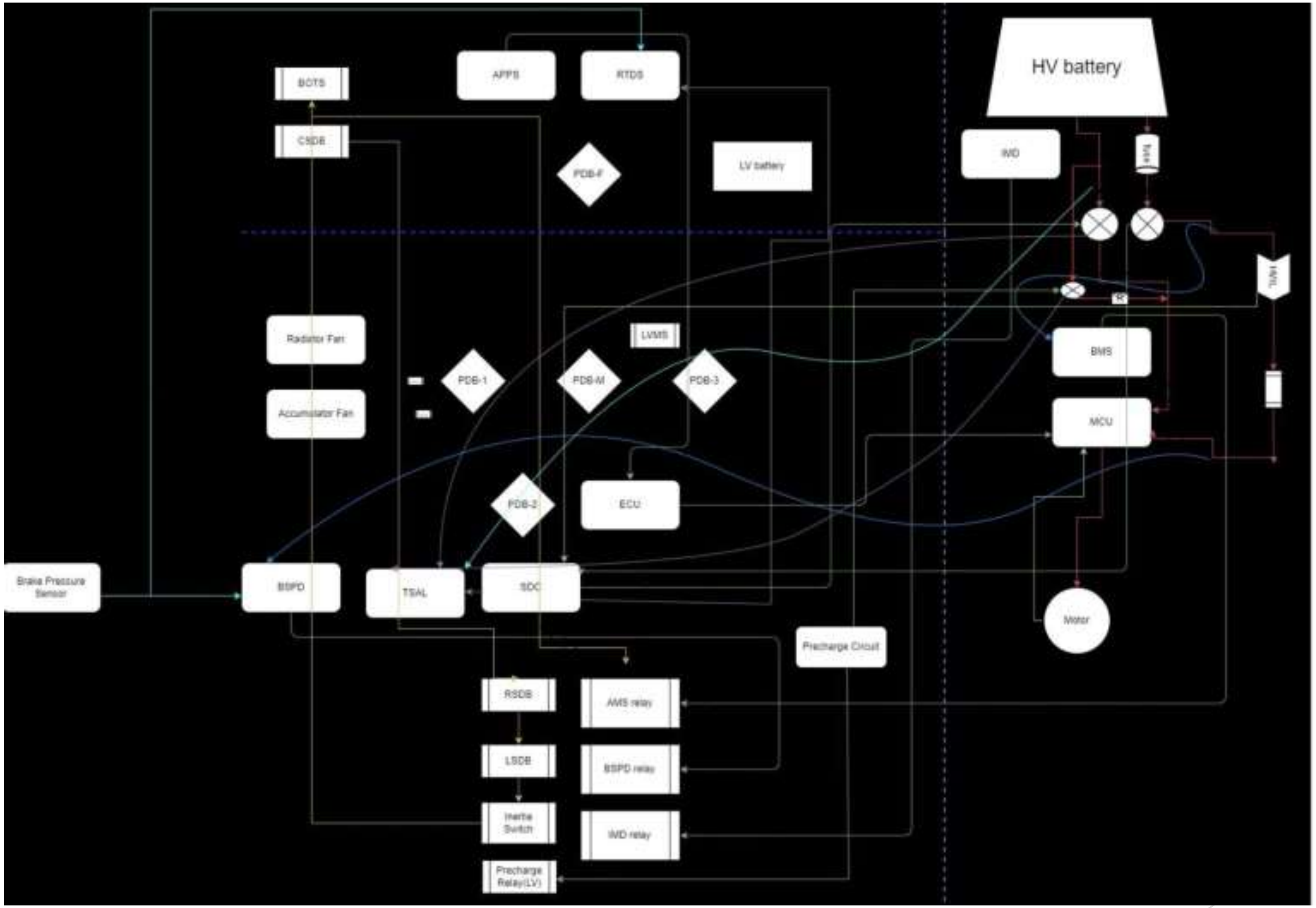
Start button/key

Fault indication lights and reset switches

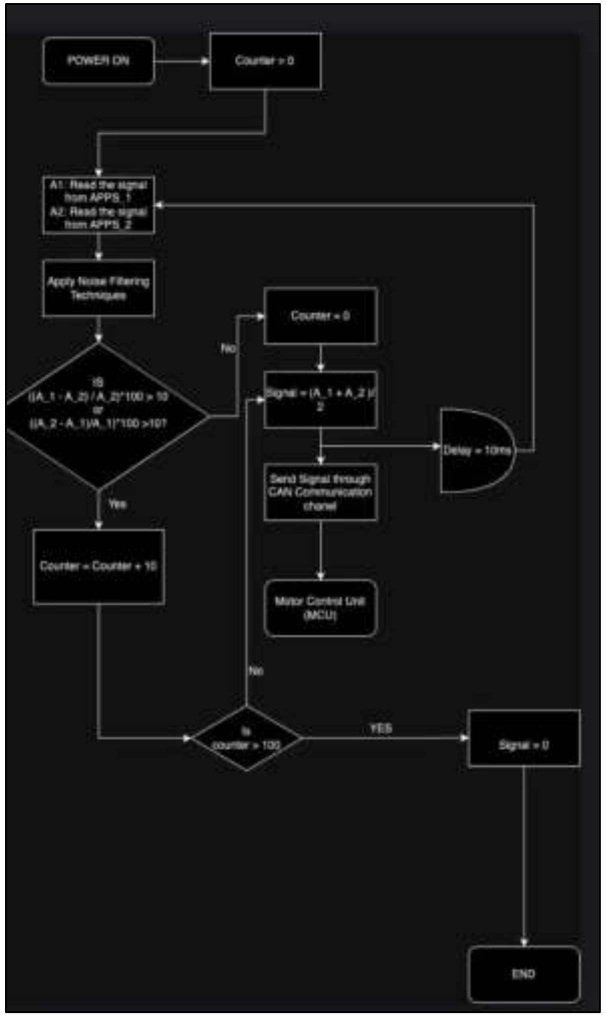
RTD indicator

Cockpit mounted shutdown button

High level diagram



APPS



Control Flow

Linear Potentiometer → AduinoUno and MCP2515 based CAN Protocol → Bamocar D3