SAEINDIA BAJA 2020 Report

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ABSTRACT

A team of Indian Institute of Technology Jammu students designed and manufactured an off-road vehicle. Efforts were made on designing cost-effective yet efficient systems including Suspension, Drivetrain, Steering, Roll Cage and Braking. Design criteria included originality, innovation, craftsmanship, manufacturing ergonomics, feasibility for mass production and serviceability. The final product meets these criteria and provides the customer with a functional product, and the producer with a product worth manufacturing.

INTRODUCTION

A team of 25 students from the Indian Institute of Technology Jammu designed and manufactured a prototype SAE Baja off-road vehicle. This report documents how and why the following components were designed: Roll Cage, Suspension, Drivetrain, Steering and Braking systems.
The design goals for the 2020 SAE Baja included:

- Minimize weight
- Maximize control with suspension design
- Optimize power efficiency
- Improve driving conditions for off-roading scenarios.
- Minimize cost

When producing an off-road vehicle, the largest concerns are cost, maintenance, safety, and driver comfort. Each of these aspects has been optimized to produce the best product.

ROLL CAGE

The goal was to design a lightweight, structurally sound and aesthetically pleasing frame. The purpose of the roll cage in an ATV is to keep the driver safe in case of collision and to support the components of the other subsystems and controls. The designing and analysis were done using Autodesk Inventor’19 and Ansys’19 respectively.

FRAME DESIGN

Triangulated space frame design was chosen, comprised of circular cross-section tubing to decrease the weight to strength ratio as much as possible. The standards along with the inputs from the other departments are considered in the designing process. The roll cage thus designed is a non-nose design because front bracing members extend to the front bumper. This choice was made to maximize internal space. Appropriate spacing on the sides of the vehicle is also provided for easier entry and exit during an emergency. Tubing varies throughout the roll cage depending on the structural needs of that area as well as the severity of possible failure modes. The final design complies with the rules mentioned in the BAJA SAE INDIA 2020 RULE BOOK.

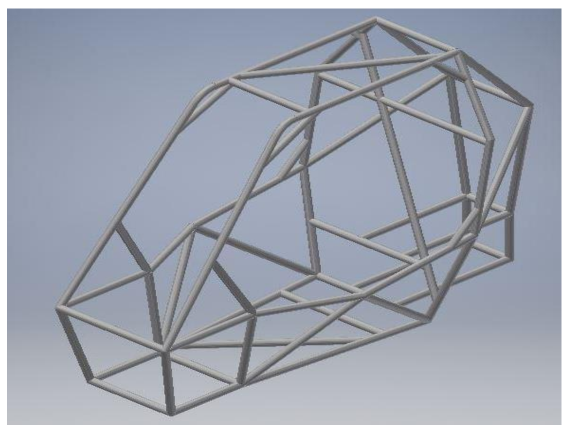


Figure 1: Isometric View of computer model using Autodesk Inventor’19

MATERIAL SELECTION

Material selection plays a crucial part in providing the desired strength, endurance, safety and reliability to the vehicle. As the vehicle is to be designed for racing, the weight also plays a crucial role. So, it is attempted to have a proper balance between the strength and weight requirements of the vehicle. Material is selected such that pipes have good bending stiffness, minimum weight, and maximum strength. We shortlisted AISI 1018 steel and AISI 4130 steel for further comparisons.

Table1. AISI 1018 vs AISI 4130

Property	AISI 1018	AISI 4130
Yield Strength	365 MPa	460 MPa
Ultimate Strength	440 MPa	560 MPa
Elongation on break	15%	21.5%
Weight/meter (same tube dimensions)	1.66 kg/m	1.63 kg/m

Cleary, AISI 4130 has a better strength to weight ratio but is expensive than AISI 1018. To have a lighter vehicle to have effective braking, acceleration, and maneuverability, AISI 4130 was finalized.

TUBE DIMENSIONS

We have to select dimensions for the primary member such that it satisfies at least one of the below conditions: -

1. AISI 1018 material, 25.4 mm outer diameter, 3 mm thickness
2. Bending stiffness greater than 2620 N-m^2 and bending strength greater than 373 N-m.

Considering the minimum requirements specified by SAE Baja and graphs between "bending strength vs. diameter" and "weight per meter vs. tube thickness" we select dimensions for the primary member.

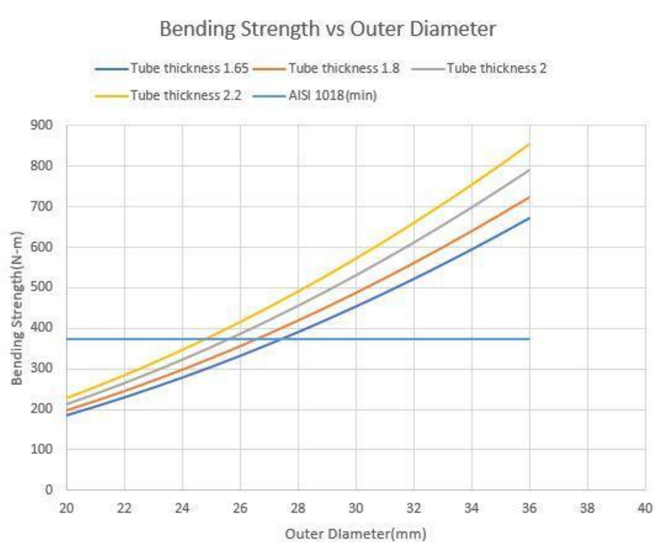


Figure 2: Bending Strength vs Outer Diameter

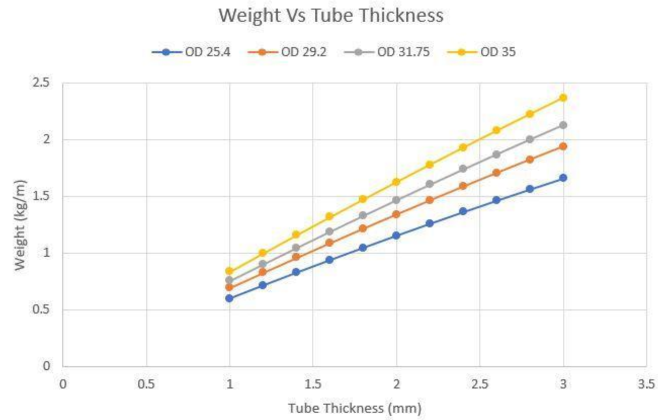


Figure 3: Weight vs Outer Diameter

Table2. Tube Dimensions

Tubes	Outer Diameter	Thickness
Primary	29.2 mm	2 mm
Secondary	25.4 mm	1 mm

For Selected material AISI 4130:

$$Modulus\ of\ Elasticity\ (E) = 205 * 10^3 \frac{N}{mm^2}$$

$$Yield\ Tensile\ Strength\ (S_x) = 460 \frac{N}{mm^2}$$

$$Outer\ Diameter\ (D_o) = 29.2\ mm$$

$$Inner\ Diameter\ (D_i) = 25.2\ mm$$

$$Distance\ from\ the\ neutral\ axis\ to\ extreme\ fiber\ (c) = 14.6\ mm$$

$$Second\ moment\ of\ area\ (I) = 3.14 * \frac{D_o^4 - D_i^4}{64} = 15890.52\ mm^4$$

$$Bending\ Stiffness = E * I = 3257.55\ N - m^2$$

$$Bending\ Strength = Sx * \frac{I}{c} = 500.66\ N - m$$

WELDING USED

Based on the weldability of the material and availability of the methods, Arc welding was the best option available to us.

FINITE ELEMENT ANALYSIS

An ATV should be capable of enduring harsh off-road conditions. To maximize the strength and durability of the roll cage and minimize the weight, ANSYS 19 was used to analyze the roll cage under predetermined loading conditions. The roll-cage was analyzed for various conditions like Front impact, Rear Impact, Side impact, Front roll over, Front Bump and Torsional Stiffness. According to the results, necessary changes were incorporated in design wherever necessary.

MESHING

2D meshing was carried out on roll cage with element size set to be 5 mm and element shape was Quad/Tria. Static analysis was done on the roll cage and the suspension mounting points were set as constraints.

IMPACT ANALYSIS

The impact forces were calculated using the following set of formulas.

$$Force = Mass \times Acceleration$$
$$Force = Rate\ of\ change\ in\ momentum$$
$$F \times T = M \times V$$

The projected mass of the vehicle including driver is 250 kg. Max. vehicle speed is 54 kmph or 15 m/s as given by the powertrain department.

FRONT IMPACT

Generally, in front impact vehicle hits a stationary object like a wall, tree, another vehicle, etc. Referring to various research papers impact timing is found to be around 0.2s.

$$F \times 0.2 = 250 \times 15$$
$$= 18750\ N$$

This is equivalent to **8G.**

Table3. Front Impact - Results

Force	Max Deformation (in mm)	Max Stress (MPa)	Factor of Safety
8G	2.99	398.19	1.16

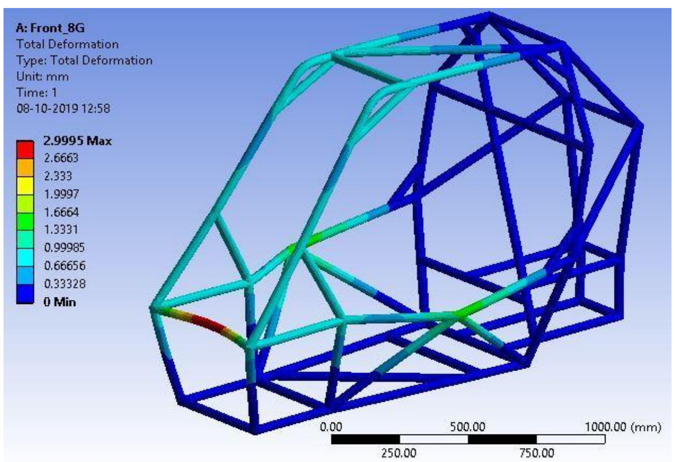


Figure 4.1: Front Impact – Total Deformation

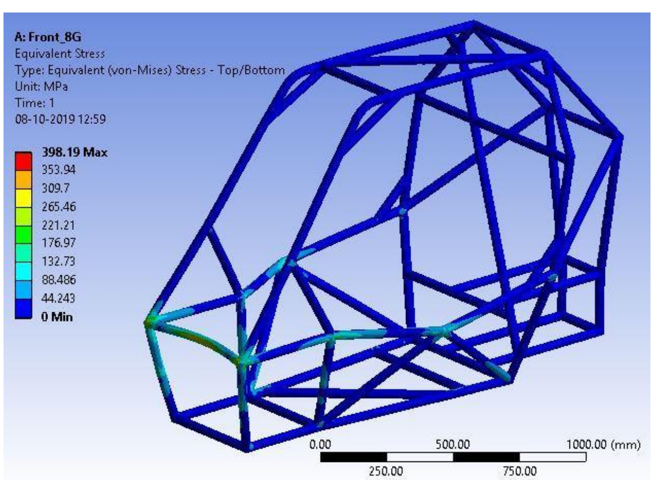


Figure 4.2: Front Impact – Equivalent Stress

REAR IMPACT

Generally, in rear impact vehicle hits by another vehicle while in motion. So relative velocity can be taken as 12m/s. Here both bodies are deformable so time of impact will be more than the front impact. Referring to various research paper it is found to be as around 0.3s.

$$F \times 0.3 = 250 \times 12$$
$$= 10000\ N$$

This is equivalent to **5G.**

Table4. Rear Impact - Results

Force	Max Deformation (in mm)	Max Stress (MPa)	Factor of Safety
5G	2.65	373.18	1.23

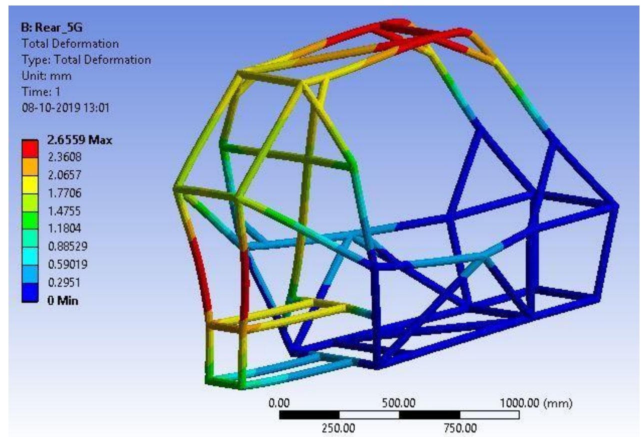


Figure 5.1: Rear Impact – Total Deformation

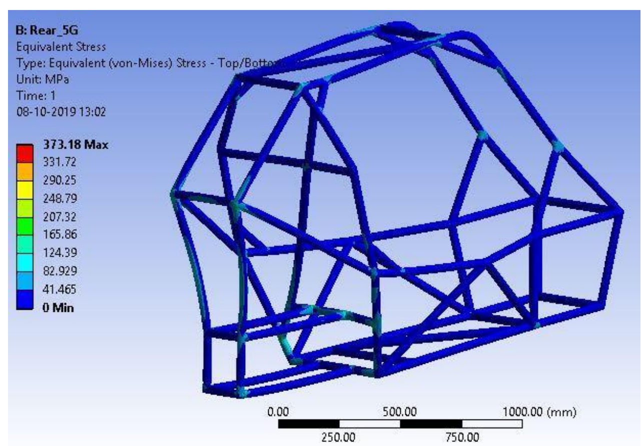


Figure 5.2: Rear Impact – Equivalent Stress

SIDE IMPACT

In side impact also both bodies are in deformable state so time of impact can be taken as 0.3s. Assuming vehicle to be at its maximum speed.

$$F \times 0.3 = 250 \times 15$$
$$= 12500\ N$$

This is equivalent to **5G.**

Table5. Side Impact - Results

Force	Max Deformation (in mm)	Max Stress (MPa)	Factor of Safety
5G	3.15	411.74	1.12

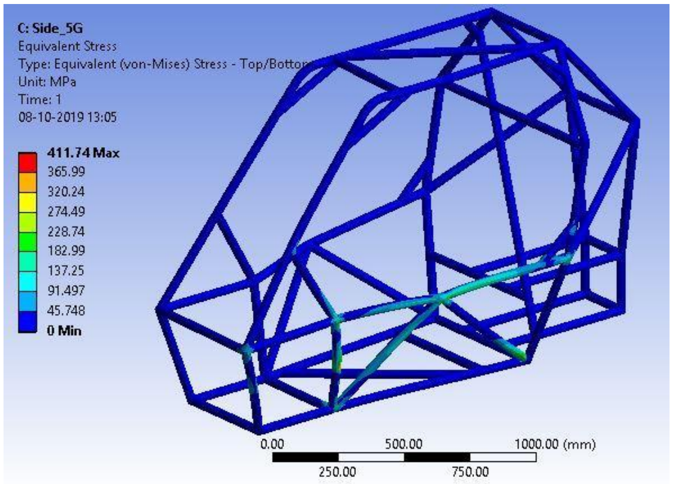


Figure 6.1: Side Impact – Total Deformation

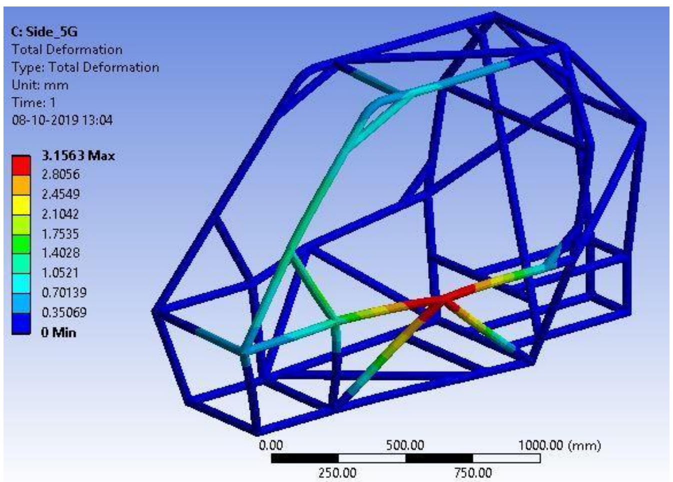


Figure 6.2: Side Impact – Equivalent Stress

ROLL OVER

Table6. Roll Over - Results

Force	Max Deformation (in mm)	Max Stress (MPa)	Factor of Safety
2G	1.67	280.66	1.64

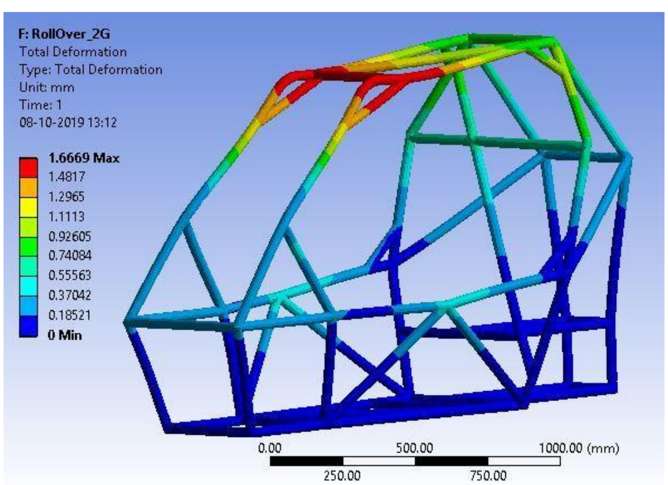


Figure 7.1: Roll Over – Total Deformation

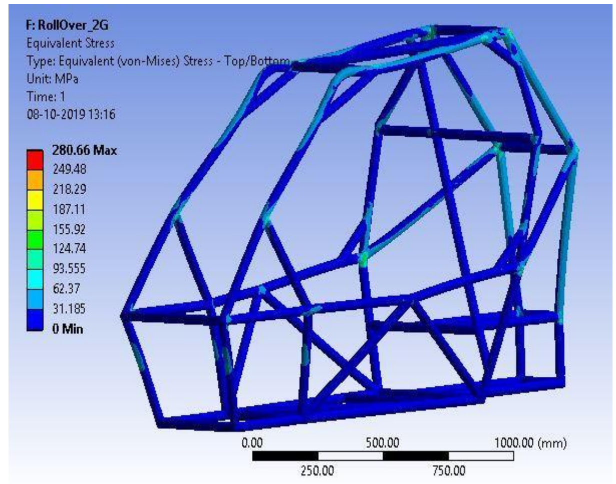


Figure 7.2: Roll Over – Equivalent Stress

FRONT BUMP

Table7. Front Bump - Results

Force	Max Deformation (in mm)	Max Stress (MPa)	Factor of Safety
4G	3.59	283.09	1.63

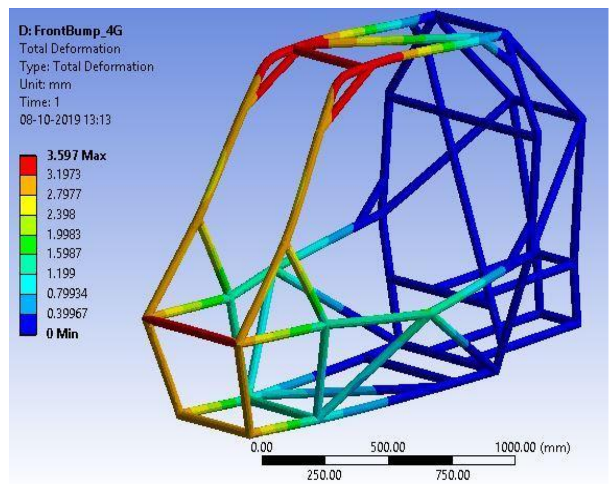


Figure 8.1: Front Bump – Total Deformation

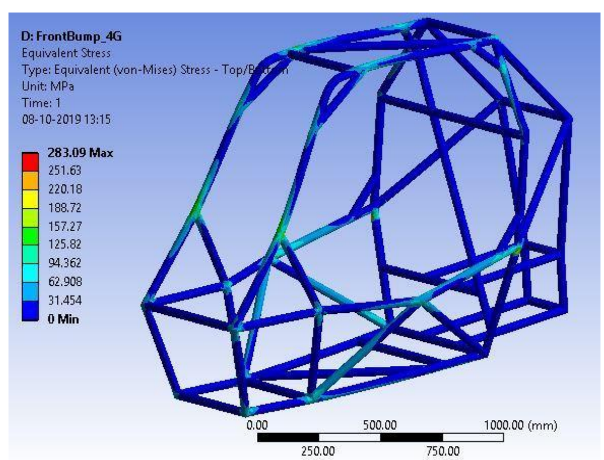


Figure 8.2: Front Bump – Equivalent Stress

TORSIONAL ANALYSIS

Table8. Torsional Analysis - Results

Force	Max Deformation (in mm)	Max Stress (MPa)	Factor of Safety
3G	2.37	389.74	1.18

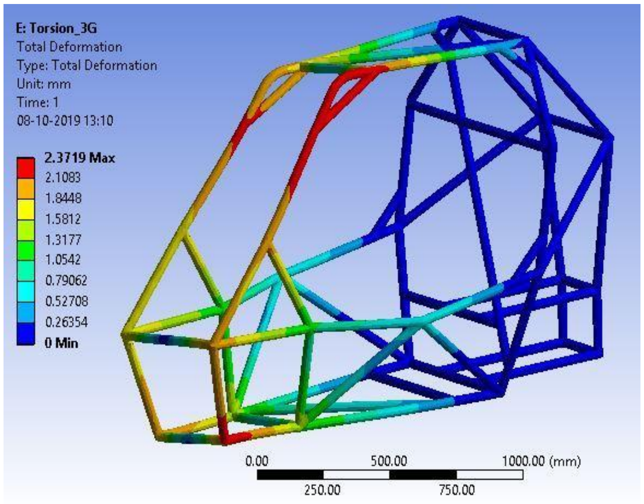


Figure 9.1: Torsional Analysis – Total Deformation

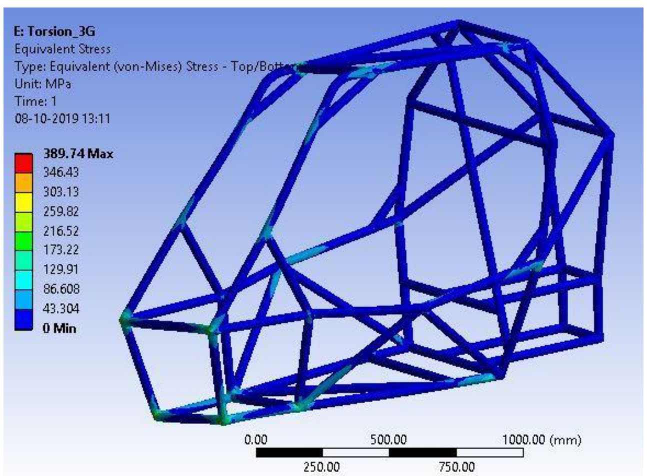


Figure 9.2: Torsional Analysis – Equivalent Stress

SUSPENSION

Suspension in a vehicle connects the wheel assembly or unsprung mass to the chassis/roll cage or sprung mass of the vehicle. The main function of the suspension system is to act as a cushion to protect the driver/passengers from shocks and bumps, i.e., maintaining good ride quality, while also providing proper handling to the vehicle. Following objectives were considered while designing the suspension system for SAE BAJA 2020:

1. To maintain contact with wheel and ground at all times.
2. To provide good ride quality and also handling.
3. To avoid toppling or roll-over.
4. To provide maximum wheel travel to absorb the shocks.
5. Suspension should be adjustable.

Suspension system design involved three phases:

1. Choice of suspension type and force calculations
2. Kinematic Analysis
3. Finite Element Analysis of all parts

A. CHOICE OF SUSPENSION

Independent vs. Dependent suspension

It was decided to use independent suspension as it allows each wheel to rise and fall on its own contrary to the dependent suspension in which both the wheel on right and left side of the vehicle are connected resulting in movement of one wheel with the movement in the other one. The dependent suspension system is undesirable in off-road applications.

FRONT SUSPENSION

Available suspension types:

1. Double wishbone
2. Macpherson strut

Double wishbone was chosen instead of Macpherson Strut for the following reasons:

1. Double wishbone provides a negative camber in bump. This is very useful, especially in the cornering of vehicle.
2. It is more adjustable compared to Macpherson strut. Static camber or caster angles can be adjusted easily.

REAR SUSPENSION

Available suspension types:

1. Double wishbone
2. Macpherson strut
3. Trailing arm/ Semi trailing arm
4. Trailing arm with camber links

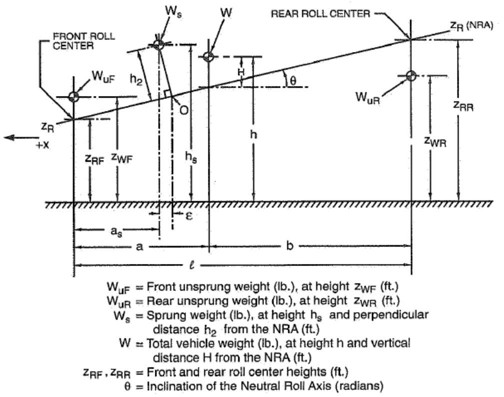
Trailing arms with camber links was chosen for the following reasons:

1. Macpherson strut and double wishbone were ruled out because of interference issues with the drive train. Trailing/semi-trailing arm provides for easy accessibility to the drive train system, which ensures easy maintenance of drive train.
2. Semi trailing arm involves design simplicity, but does not have enough camber gain compared to trailing arm with camber links.
3. The camber links provide an option to introduce negative camber I trailing arm suspension. Also, the camber links provide an option to adjust static camber and toe angle.

Force calculations

1. <u>Load Transfer – Lateral (Cornering)</u>

Lateral load transfer was calculated based on roll rate for front as well as rear.



The values are:

Mass:

$$M = 250\ kg$$

Spring Rate:

$$K_{SF} = 22.5N/mm$$
$$K_{SR} = 27.4N/mm$$

Motion Ratio:

$$MR_F = 0.51$$
$$MR_R = 0.70$$

Distances:

$$\mathrm{t_F} = 1340\ mm$$
$$\mathrm{t_R} = 1260mm$$
$$\mathrm{a = 928mm, b = 494mm, l = 1422mm}$$
$$\mathrm{H = 252.9mm}$$

The forces were calculated as:

1. Ride rate

$$\mathrm{K_{RF}} = K_{SF}(MR)^2$$
$$\mathrm{K_{RR}} = K_{SR}(MR)^2$$

2. Roll rate

$$\mathrm{K_{\phi F}} = \frac{K_{RF}t_f^2}{2}$$
$$\mathrm{K_{\phi R}} = \frac{K_{RR}t_R^2}{2}$$

3. Lateral load transfer

$$F_{LF} = \frac{M}{t_F}\left(\frac{HK_{\phi F}}{K_{\phi F} + K_{\phi R}} + \frac{bZ_{RF}}{l}\right)A_Y$$
$$F_{LR} = \frac{M}{t_R}\left(\frac{HK_{\phi R}}{K_{\phi F} + K_{\phi R}} + \frac{aZ_{RR}}{l}\right)A_Y$$

Substituting the values:

$$\mathrm{F_{LF}} = 29.83A_Y$$
$$\mathrm{F_{LR}} = 55.7A_Y$$

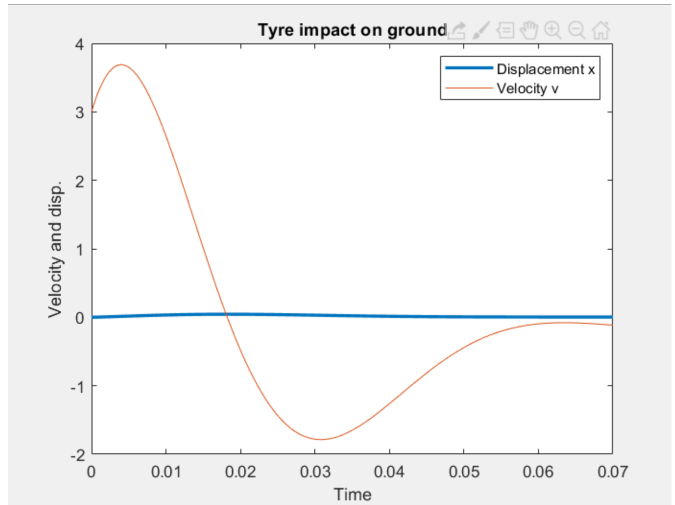
A_Y is the lateral acceleration:

$$\mathrm{A_Y} = \frac{v^2}{r}$$

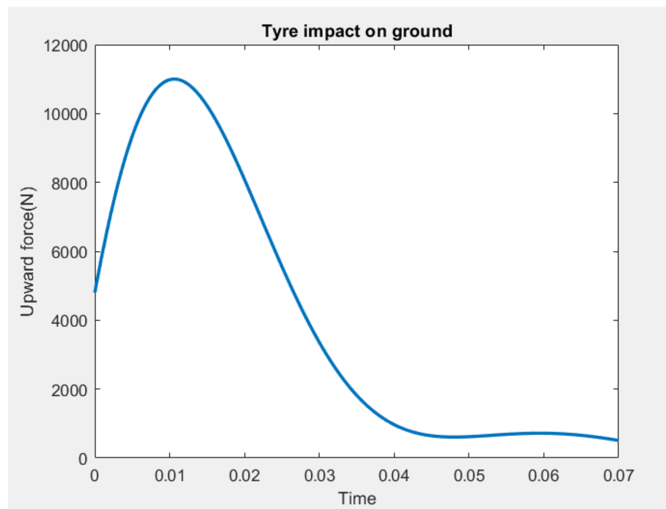
Hence we can get the lateral force as a function of lateral acceleration.

2. <u>Bump force:</u>

For calculation of bump force, a bump height of 0.5 m was considered and the vehicle falls from that height on a single tyre.
This situation was modelled using two spring-mass-damper systems in parallel- one for the actual dampers and one for the tyre. The entire vehicle mass of 250 kgs was considered. The solution was obtained numerically on MATLAB.
The impact duration is taken as the time when the tyre comes to a complete halt. The average upward force on the tyre during this duration was then taken to be the normal force.



Velocity and displacement vs time

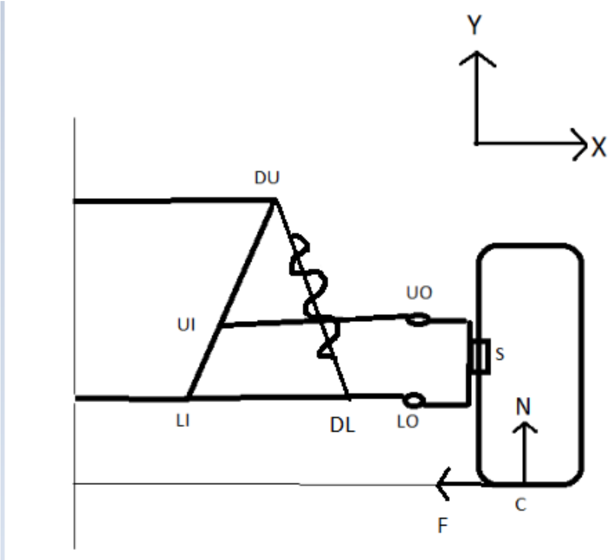


Upward force vs time

The impact duration is hence approximately 0.065 seconds.
The corresponding mean bump force is **4056 N.**

3. <u>Force on A-arms - front:</u>

We assume quarter car model for calculating the forces and the A-arms are parallel to each other as well as the ground.
The point are as follows, with the x and y direction.
(The forces at respective points are denoted by F with subscript point name,direction and distance by point1-point2 subscript direction)



We will obtain all forces corresponding to a particular value of Normal force acting on the tyre at contact patch(N) and lateral force acting at contact patch(F).
The forces were calculated as:

$$F_{UIX} = \frac{-Fc + Nl}{d - c}$$
$$F_{UIY} = 0$$
$$F_{UOX} = \frac{Fc - Nl}{d - c}$$
$$F_{UOY} = 0$$
$$\mathrm{F_{LOX}} = \frac{Nl - Fd}{d - c}$$
$$\mathrm{x} = \frac{\mathrm{N}}{(\mathrm{MR})\mathrm{k}\cos\theta}$$
$$\mathrm{F_{LOY}} = kx(MR)\cos(\theta)$$
$$\mathrm{F_{LIX}} = F_{LOX} + kx\sin(\theta)$$
$$\mathrm{F_{LIY}} = \left(1 - (MR)\right)kx\cos(\theta)$$

The distance values according to the design are:
$\mathrm{a} = 200\mathrm{mm}, \mathrm{b} = 442\mathrm{mm}, \mathrm{c} = 220\mathrm{mm}, \mathrm{d} = 395\mathrm{mm}, \mathrm{l} = 104\mathrm{mm}$
The resulting values of forces for various values of N and F are:

	F_{UIX}	F_{UOX}	F_{LOX}	F_{LOY}	F_{LIX}	F_{LIY}
Static(ride height) N = 430 N	254	254	254	430	676	520
Full bump- N = 4056N	2400	2400	2400	4056	6300	4900
Full cornering- F = 700 N, N = 905 N	346	346	1046	905	155	1095
Full Braking- N = 722 N	427	427	427	722	1137	873

Hence the suspension system needs to be designed for bump force

Same methodology was adopted for rear forces calculation.

B. Kinematic Analysis:

The objective of the kinematic analysis was to analyze the geometry of the suspension system and determine the optimum mounting points locations. These were decided based on the following factors:

1. A camber of -8 to -10 degrees for front and rear at maximum bump was desired. This was achieved by making the upper arm shorter than the lower arm.
2. The bump steer should be as low as possible. Also, the Ackerman percentage is needed to be close to 100%. This is achieved by choosing an appropriate position of the track rod inner and outer mounting points. However, a compromise has to be made between bump steer and Ackerman percentage as decreasing the bump steer decreases the Ackerman, and vice versa.
3. The scrub radius(front) has to be minimized. This can be done by choosing an appropriate Kingpin Inclination angle. This is achieved by setting appropriate positions of outer ball joints.
4. The roll centre height must be as close to the CG height as possible, to avoid excessive roll during cornering. Also, rear roll centre height should be greater than the front roll centre height.
5. The vehicle should avoid excessive dive and squat during braking and acceleration, respectively. A proper anti-dive and anti-squat percentage should be chosen for this purpose. This is accomplished by setting appropriate locations of the inner mounting points.
6. Ground clearance of 13 inches for the front as well as the rear is desired.

Both front and rear suspension geometry were modeled in Lotus Shark suspension design software.

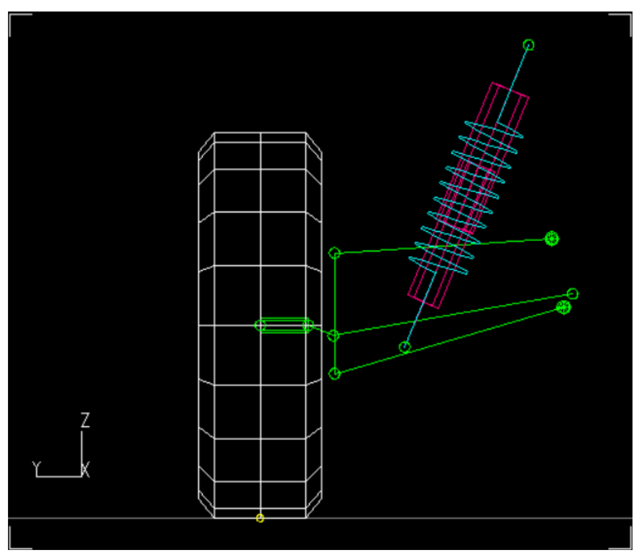


Figure 10.1: Lotus Shark modelling for Front Suspension

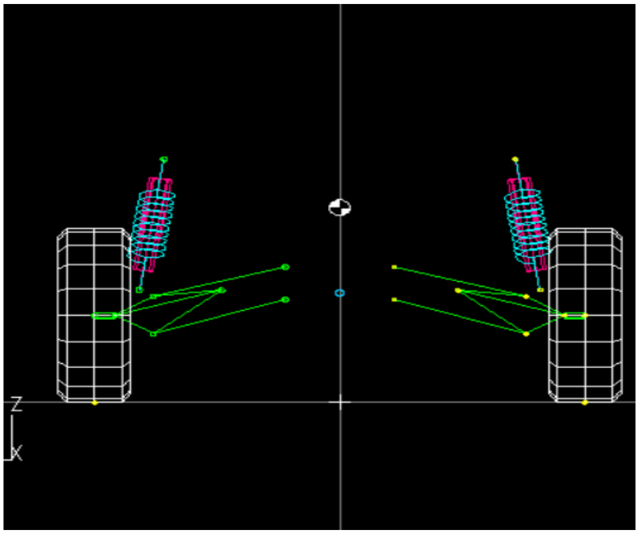


Figure 10.1: Lotus Shark modelling for Rear Suspension

Table9. Suspension Specifications

Parameters	Front	Rear
CG height	22 inches	
Ground clearance	12 inches	
Roll centre height	11.14 inch	13.81 inch
Motion ratio	0.51	0.70
Camber(deg.)	Static: 0 Bump: -10 Droop: 2.9	Static: 0 Bump: -9.8 Droop: 7.8
Castor(deg.)	0	0
KPI	0	0
Scrub radius	103 mm	--
Toe(deg.)	Static: 0 Bump: 0.1 Droop: 2.1	Static: 0 Bump: -1.8 Droop: -4.91
Wheel centre travel	Bump: 8 inches Droop: 4 inches	Bump: 7 inches Droop: 4 inches
Suspension(spring) travel	Bump: 4.5 inches Droop: 2 inches	Bump: 5 inches Droop: 3 inches

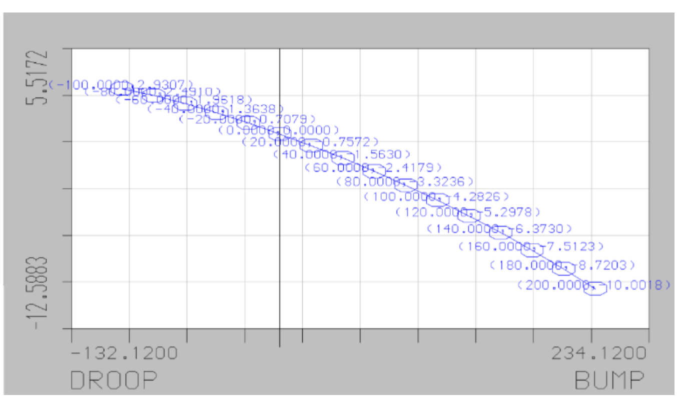


Figure 11.1: Camber vs Suspension travel for Front Suspension

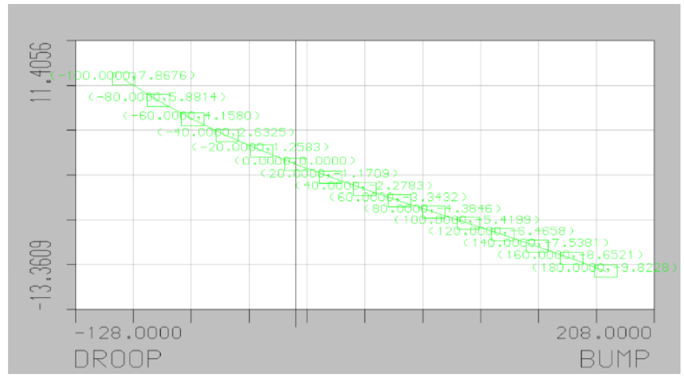


Figure 11.2: Camber vs Suspension travel for Rear Suspension

SUSPENSION ARMS

The suspension arms were modelled using Autodesk Inventor. The CAD file was prepared and further FE analysis was carried out in ANSYS Workbench.
The material chosen was AISI 4130 steel, based on yield strength, cost and availability factors.

DESIGN OF SHOCKS (SPRING & DAMPERS)

Custom made Coil-overs were chosen for both front and rear. The spring rate was calculated based on the quarter-car model, using the sprung mass of the vehicle and weight distribution to calculate the spring rate based on the desired ride frequency for an off-road vehicle.

Table10. Shocks Specification

Parameters	Front	Rear
Sprung mass	55	120
Unsprung mass	35	40
Ride frequency	2.1 Hz	2.3 Hz
Ride rate	6.09 N/mm	13.4 N/mm
Spring stiffness	22.5 N/mm	27.4 N/mm
Wire diameter	11 mm	13 mm
Outer diameter	80 mm	85 mm
Spring material	EN 47 Spring steel	
FOS	2.19	2.23
Eye to eye length	18 inches	20 inches

DESIGN OF UPRIGHTS

Al 7075 was chosen for both rear and front uprights. It was chosen based on factors like weight, yield strength and cost considerations.
The uprights were modelled in Autodesk Inventor and FEA was carried out in ANSYS Workbench. The following are details of FEA of uprights:

C. Finite Element Analysis

The stresses involved in some complex geometries cannot be accurately determined by analytical calculations. Also, several assumptions are made in analytical calculations, which can result in under-design of the component. Hence, performing a Finite Element Analysis (FEA) of such complex geometries is crucial.

A static analysis of the individual front and rear suspension arms was carried out in ANSYS Workbench.

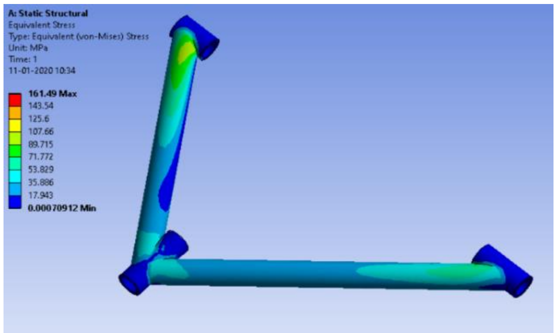


Figure 12.1: Equivalent Stress – Front Upper Arm

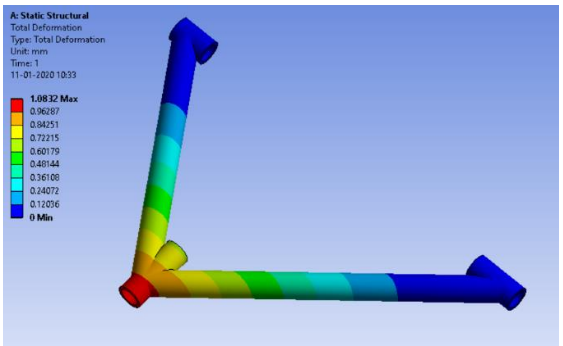


Figure 12.2: Total Deformation – Front Upper Arm

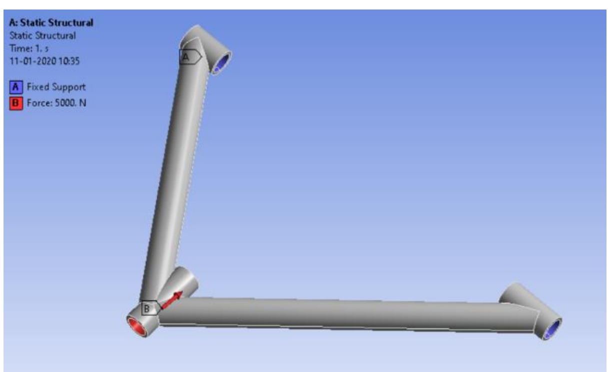


Figure 12.3: Boundary Conditions – Front Upper Arm

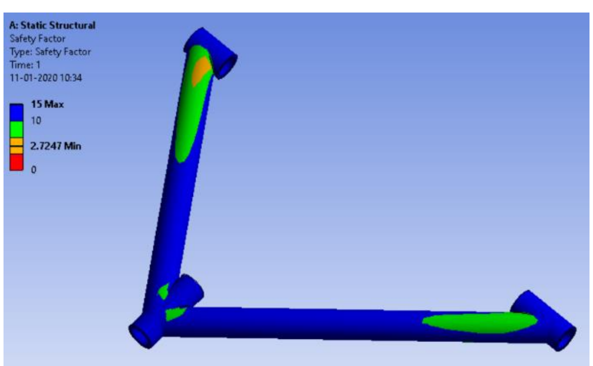


Figure 12.4: Factor of Safety– Front Upper Arm

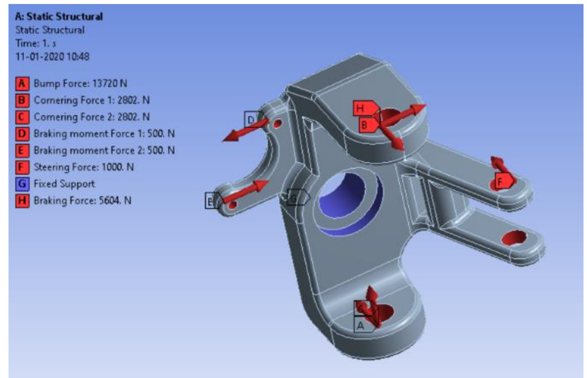


Figure 13.1: Boundary Conditions – Front Upright

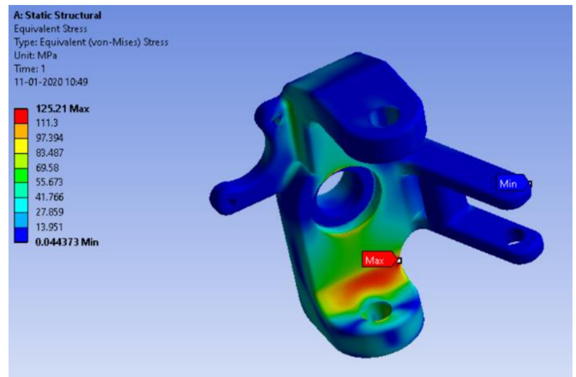


Figure 13.2: Equivalent Stress – Front Upright

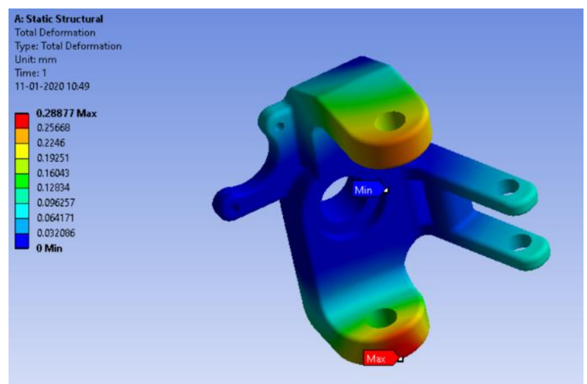


Figure 13.3: Total Deformation – Front Upright

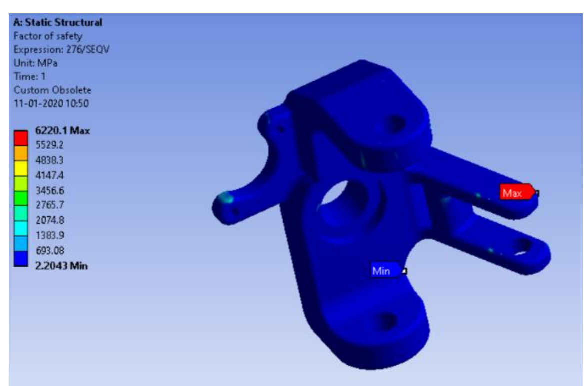


Figure 13.4: Factor of Safety – Front Upright

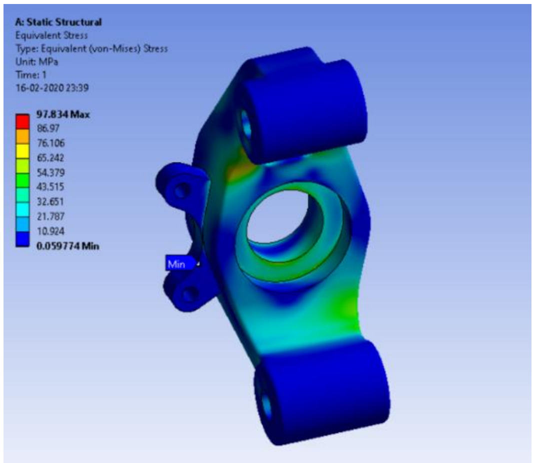


Figure 14.1: Equivalent Stress – Rear Upright

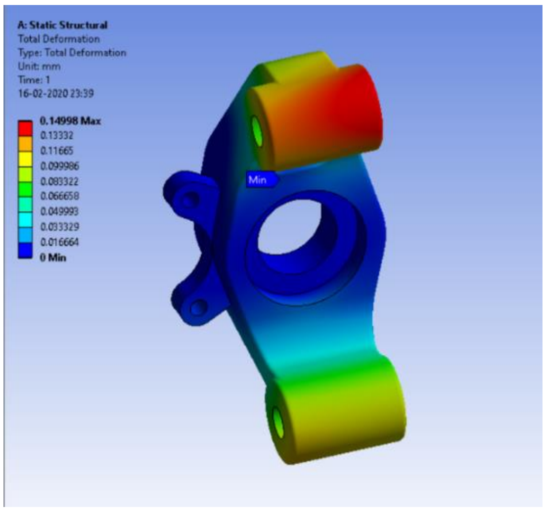


Figure 14.2: Total Deformation – Rear Upright

Table10. Summary of FEA Results

Sr. no.	Part	Material	Maximum Equivalent (Von-mises) stress (MPa)	Total Defor mation (mm)
1	Front upper control arm	AISI 4130	161.49	1.09
2	Front lower control arm	AISI 4130	259.9	0.94
3	Front upright	AL 7075	125.2	0.29
4	Rear upright	AL 7075	97.2	0.14

DRIVE TRAIN

The main objective of the drive-train is to convey power from the engine, through the transmission, to the wheels. The design team came across the challenge to amplify the engine's 10 horsepower and convey it to the wheels efficiently while not adding much weight to the car. The drive train includes Briggs and Stratton engine, CVTech CVT, custom made reduction gearbox and Half drive shafts.

ENGINE

The engine used is the provided Briggs and Stratton SAE BAJA 305cc 10HP engine. The engine top speed is observed to be 3800 rpm. From further researches we came to know that the optimal speed lied in the range of 2900 rpm to 3400 rpm for maximum efficiency. With limited engine power, the design team's important goal was to reduce the weight to the drive train as much as possible.

CVT (CONTINUOUSLY VARIABLE TRANSMISSION)

The CVT used is the CVTech GX9 BAJA Gaged CVT with overall gear ratio of 3.9:1 to 0.9:1. The only reason to used gaged CVT instead of normal CVTech CVT was to reduce weight. The gaged CVT weighted around 6.3kgs while the normal one weighted around 10kgs. The primary clutch consists of a centrifugal clutch that automatically shifts gears ratios under varying engine speeds and torque loads. The ability to tune the CVT comes from changing weights in the primary clutch that will change engagement RPMs and time to maximum ratio.

SPUR REDUCTION GEARS

The objective of a gear reducer is to lower the final gear ratio. Spur system is chosen over the more convenient chain system to make the reduction more effective in terms of drive efficiency, packing, and safety. This increased our cost of production but with mass production it can be lowered down. The custom-made reduction had a ratio of 8:1 and spline coupling output made with reference to the axles. This is done for efficient transmission of power.

DRIVE AXLES

Two drive axles are chosen with CV joints on hub end and on reduction end. CV joints were chosen as they give better performance at greater angles, which allows for more suspension travel. The axles delivered were longer than our requirement so we have to cut it down to our required size. UV joints came helpful in that as they are easy to reallocate. The axles and the reduction were fitted such that the weight distribution is uniform. The reduction is in the center with axles coming out of it in the opposite direction.

CALCULATIONS OF PERFORMANCE CHARACTERISTICS

The performance characteristics of BAJA vehicle strongly depends upon four main factors which are:
1. Gradeability
2. Top Speed
3. Maximum Acceleration
4. Reduction in lower gear ratio

For the calculations the mentioned factors following constant were used after referring to various research papers -

Table11. Constants for Calculations

Air density (ρ)	1.12gm/cc
Mass(m) (of the atv)	250kg
Acceleration due to gravity(g)	9.8m/s^2
Dynamic rolling friction(mr)	0.085
Air Drag coefficient (C_d)	1.08
Frontal area (A_f) (of the atv)	0.95m^2

GRADEABILITY

It is defined as the ability of a vehicle to climb uphill at a certain angle without slip occurring. It is an important factor that decides the maximum pulling force that can be applied by the engine on the wheels.

In general sense gradeability is defined in terms of percentage, let say X%. This means that vehicle will cover X m distance in vertical direction while moving 100 m in a horizontal direction.

ROLLING RESISTANCE

Rolling resistance is a result of energy loss in the tyre, which can be traced back to the deformation of the area of tyre contact and the damping properties of the rubber. These lead to the transformation of mechanical into thermal energy, contributing to warming of the tyre.

$$RR = m \times g \times \mu \times \cos\cos\theta$$
$$\theta = angle\ of\ elevation$$
$$\mu = friction\ coefficient$$

rolling resistance	
gradient%	rolling resistance
0	49
10	48.75682232
20	48.04845311
30	46.93348798
40	45.49535785
50	43.82693236
60	42.01715336
70	40.14236411
80	38.26257166

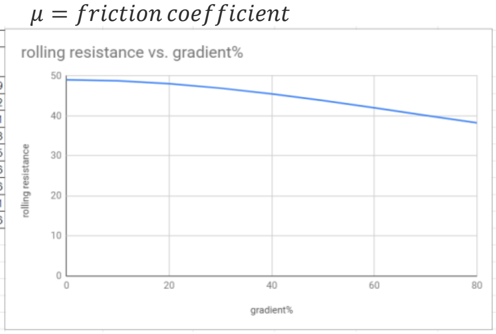


Figure 15.1: Rolling Resistance vs Grade (%)

AERODYNAMIC RESISTANCE

Aerodynamic drag of a vehicle is determined by the aerodynamic shape of the body, described by the drag coefficient (C_d), and by the projected frontal area of the vehicle (A_f). The aerodynamic force increases with the square of the vehicle's velocity(v).

$$AR = \frac{1}{2} \times \rho \times A_f \times V^2 \times C_d$$
$$V = Velocity\ of\ the\ car$$

Aero Res	
speed(km/hr)	aero res
0	0
5	1.05
10	4.2
15	9.45
20	16.8
25	26.25
30	37.8
35	51.45
40	67.2
45	85.05
50	105
55	127.05
60	151.2

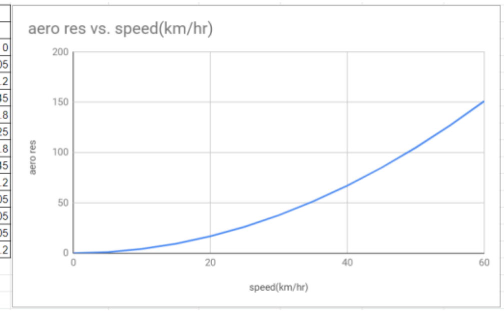


Figure 15.2: Aerodynamic Resistance vs Grade (%)

GRADE RESISTANCE

Grade resistance is the simplest form of resistance. It is the gravitational force acting on the vehicle. This force may not be exactly perpendicular to the roadway surface, especially in situations when an elevation is present.

$$GR = m \times g \times \sin\sin\theta$$

Gradient resistance	
gradient %	GR
0	0
10	243.7841116
20	480.4845311
30	704.0023196
40	909.9071571
50	1095.673309
60	1260.514601
70	1404.982744
80	1530.502867

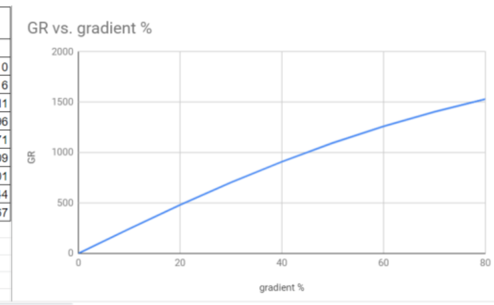


Figure 15.3: Grade Resistance vs Grade (%)

TOTAL TRACTIVE EFFORT

Total Tractive Effort is the net horizontal force applied by the drive wheels to the ground. If the design has two drive wheels, the force applied per drive wheel (for straight travel) is half of the calculated TTE. The total wheel torque does not change with the number of drive wheels. The sum of the individual drive motor torques must be greater than or equal to the computed Wheel Torque. The Maximum Tractive Torque represents the maximum amount of torque that can be applied before slipping occurs for each drive wheel. The total wheel torque must be less than the sum of the Maximum Tractive Torques for all drive wheels or slipping will occur.

$$TTE = RR + AR + GR$$

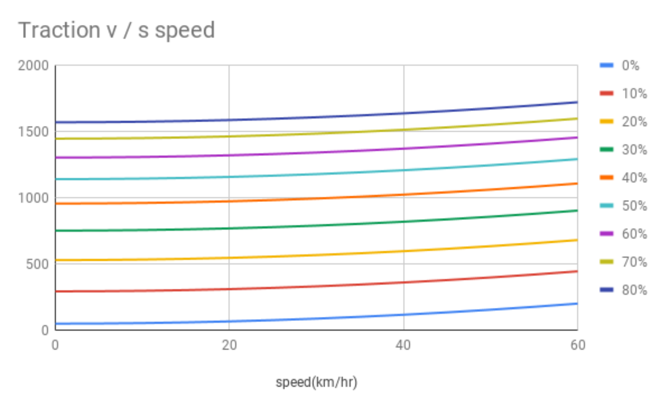


Figure 15.4: TTE vs Speed (kmph) for different Grade (%)

Maximum grade is achieved when we have our maximum acceleration, i.e. engine is working on its best performance. Using “Briggs & Stratton model 19 engine” report we get the maximum torque value to be: 18.57Nm.
For that particular torque, our vehicle achieves 3.362 m/s or 12.096 Km/hr.

gradient%	TEmax	12.096km/hr
0	1592.5	55.14515507
10	1584.596725	298.686089
20	1561.574726	534.6781393
30	1525.338359	757.0809627
40	1478.59913	961.54767
50	1424.375302	1145.645396
60	1365.557484	1308.676909
70	1304.626833	1451.270263
80	1243.533579	1574.910593

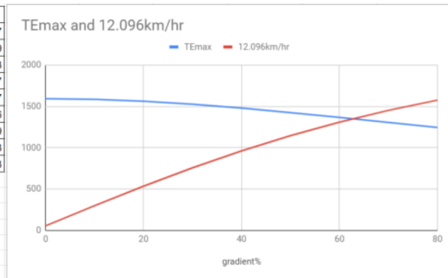


Figure 15.5: Graph to calculate gradeability

Using linear interpolation between gradient% of 60 and 70 we get the X(grade) and Y(N) as:
62.79 and 1348.54.

Gradeability = 62.79%.

TOP SPEED

Let the maximum velocity be V (m/s)
Maximum output from the engine (max_eng) = 10 hp = 7457 W
Maximum output at wheels (max_out) = max_eng x (transmission efficiency) = 7457 x 0.7 = 5219.9 W
For maximum velocity net force on a body must be zero.

Therefore,

$$RR = 220.5\ N$$
$$AR = 0.57456 \times V^2$$
$$Total\ effort \times\ V\ = 5219.9\ N$$
$$Net\ force\ =\ 0\ N$$

Power must be compensated by net resistive forces

$$RR \times\ V + AR \times\ V = max\_out$$
$$0.57456 \times (V^3) +\ 208.25 \times V\ - 5219.9 = 0$$

On solving,

$$V\ = \frac{15.26\ m}{s} = 54.93\ \frac{km}{hr}$$

MAXIMUM ACCELERATION

Maximum acceleration will be achieved when there will be least resistance to the vehicle.
Let ‘a’ be the acceleration that we need to calculate.
For a body in motion:

$$Force\ =\ m \times a$$

Torque delivered by engine,

$$Te\ = 18.66\ \ N - m$$

Overall reduction in low gear,

$$Ig_{low}\ =\ 30$$

Transmission efficiency,

$$n_t = 70\%$$

Torque at wheels,

$$Tw\ =\ Te \times Ig_{low} \times n_t$$
$$Tw =\ 419.85\ N - m$$

Pull at wheels,

$$Fw\ = Tw/dynamic\ radius\ of\ tire$$
$$Fw\ = \frac{419.85}{27.566} \times\ 100N$$
$$Fw\ =\ 1523.07\ N$$
$$RR\ =\ 49\ N \text{ (dry road)}$$
$$GR = m \times\ g \times sin(0) = 0$$
$$AR = 145.6870303\ N$$
$$TTE = 194.6870303\ N$$

$$a = \frac{Fw - TTE}{m}$$
$$a = 5.31 \frac{m}{s^2}$$

$$a = \frac{5.31}{9.8} g = 0541g$$

REDUCTION IN LOWER GEAR RATIO

$$Tw\ =\ Total\ Road\ Load \times dynamic\ rolling\ radius$$
$$Tw\ =\ (1387.473) \times (0.27566)$$
$$Tw =\ 387.65\ N - m$$
$$Te\ \ =\ 18.66\ N - m$$

The relation between Torque at wheels and Torque at engine is,

$$Tw\ =\ Te \times Ig_{low} \times n_t$$
$$Ig_{low} = 29.688$$

After the overall reduction is calculated, it is needed to decide between CVtech and Gaged CVT:
Reduction for CVtech = 29.688/3 = 9.89
Reduction for Gaged CVT = 29.688/3.9 = 7.61

Table12. Summary – Drive Train Calculations

Maximum velocity	**54.93 km/hr.**
Gradeability	**62.79**
Reduction to be used	• For CVTech = 9.81 = 10 approx. • For Gaged CVT=7.61= 8 approx.
Maximum acceleration	5.31 m/s^2

MOUNTINGS & PLACEMENT

AISI 4130 square pipes of outside dimension 1in and thickness 0,035 in is used – the minimum requirements for a rectangular section secondary roll cage pipe according to the BAJA SAE INDIA 2020 RULE BOOK.

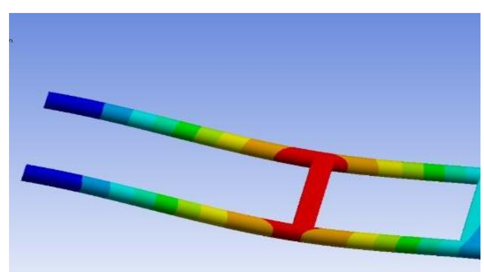


Figure 16.1: FEA of Engine & Reduction gearbox Mounting: Equivalent Stress = 91.625 MPa, Max. Deformation = 0.2769 mm, FOS = 4.74

The efficiency of vehicle gets reduced by the vibrations produced by the engine of the ATV. To increase the efficiency, it is needed to reduce NVH so rubber bushing, as per availability, are to be placed at engine and reduction mountings.

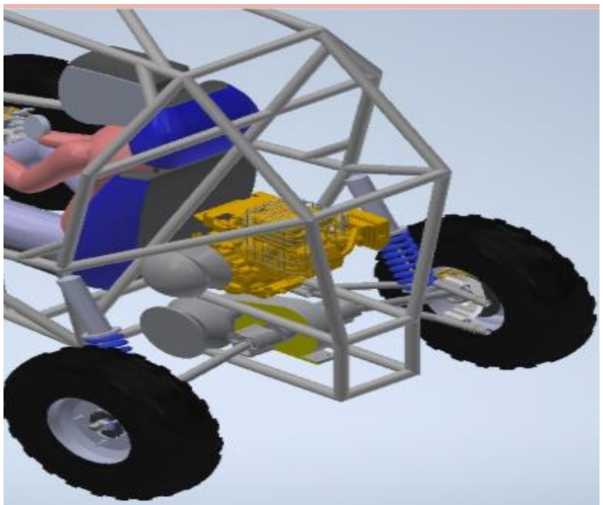


Figure 16.2: Placement of Engine and Reduction gearbox w.r.t Rear Frame

The engine and reduction are placed in such way to create a balance between wheelbase and the center of gravity of the vehicle with the considerations of rules mentioned in the BAJA SAE INDIA 2020 RULE BOOK.

CVT CASING

In order to prevent any debris from the track from hampering the smooth operation of our CVT and preventing the guard from getting injured due to rotating sheaves of the CVT, a casing to house the CVT using a 1.5mm thick mild steel sheet is designed with mesh like covering on the outer side for proper air circulation.

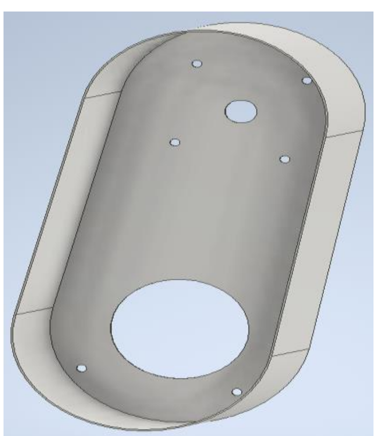


Figure 17: CVT Casing – without the mesh

STEERING

The main objective of the steering design that we have chosen is to minimise the turning radius as much as possible while maintaining easy manoeuvrability by the driver on any terrain, with a low steering ratio and less lock to lock rotation. The steering system contains the steering wheel, steering column, rack and pinion, tie rods, knuckles, and heim joints.

DESIGN

To obtain our desired objectives, we have used a centre driven steering rack, 17 inches eye to eye length, 1.5 lock to lock turns. According to our calculations, the turning radius of the buggy will be 8.3ft. One of the benefits of rack and pinion steering other than being light weight is, a lighter feel in the steering wheel due to a gear reduction. It gives a better road feel while driving and allows the driver to rely more on feel than sight. The chosen gear ratio provides very responsive steering, while not making it very difficult for the driver to steer. Two u-joints were used in the steering column to decrease the angle in each u-joint so as to reduce the stiffness while steering. Nature of steering is aimed to be oversteer to assist tight cornering using the phenomenon of drifting.

Figure 18: Steering Assembly

CALCULATIONS

Basic parameters:

Wheelbase = 56 in
Trackwidth = 50.3 in

After various iterations of the calculation of steering angle and corresponding turning radius, the following parameters were finalized:

Turning radius= 8.3 ft
Angle of inside lock= 38.50 degrees
Angle of outside lock= 24.89 degrees

Formulae Used

$$R = sqrt\left(turning\ rad^2 - \left(\frac{wheelbase}{2}\right)^2\right)$$

$$Angle\ of\ inside\ lock\ =\ tan-1\left(\frac{wheelbase}{R-\frac{trackwidth}{2}}\right)$$

$$Angle\ of\ outside\ lock\ =\ tan-1\left(\frac{wheelbase}{R+\frac{trackwidth}{2}}\right)$$

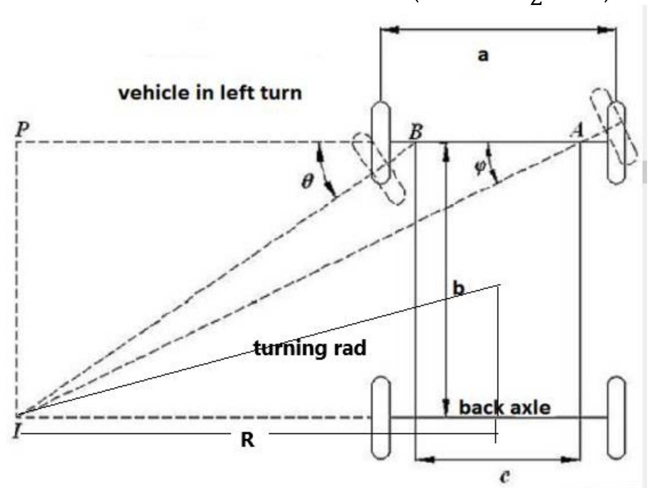


Figure 19.1: Calculation schematic

Lock to lock turns = 1.5
Steering Ratio = Degrees of steering wheel turn/ corresponding degrees of wheel turn= 7:1
Rack length (eye to eye) = 17 in
Rack travel = steer arm length*2*angle of inside lock(radians)= 5.42 in
C-factor = rack travel/lock to lock turns= 3.61

INCREMENTAL GEOMETRY VALUES

RACK TRAVEL (mm)	TOE ANGLE RHS (deg)	TOE ANGLE LHS (deg)	CAMBER ANGLE RHS (deg)	CAMBER ANGLE LHS (deg)	ACKERMANN (%)	TURNING CIRCLE RADIUS (mm)
-40.00	-30.44	26.00	0.00	0.00	37.31	2767.11
-35.00	-25.92	22.76	0.00	0.00	35.08	3275.02
-30.00	-21.73	19.54	0.00	0.00	33.40	3929.12
-25.00	-17.77	16.32	0.00	0.00	32.13	4820.79
-20.00	-13.99	13.09	0.00	0.00	31.18	6131.18
-15.00	-10.35	9.86	0.00	0.00	30.49	8281.71
-10.00	-6.82	6.61	0.00	0.00	30.02	12535.26
-5.00	-3.38	3.32	0.00	0.00	29.75	25204.46
0.00	0.00	0.00	0.00	0.00	29.66	0.00
5.00	3.32	-3.38	0.00	0.00	29.75	25204.46
10.00	6.61	-6.82	0.00	0.00	30.02	12535.26
15.00	9.86	-10.35	0.00	0.00	30.49	8281.71
20.00	13.09	-13.99	0.00	0.00	31.18	6131.18
25.00	16.32	-17.77	0.00	0.00	32.13	4820.79
30.00	19.54	-21.73	0.00	0.00	33.40	3929.12
35.00	22.76	-25.92	0.00	0.00	35.08	3275.02
40.00	26.00	-30.44	0.00	0.00	37.31	2767.11

Figure 19.2: Data for steering response – LOTUS simulation

Optimum suspension geometry was finalized by studying the different behavior parameters of the system during different dynamic conditions using LOTUS software. We found the maximum Ackerman percentage to be around **37%**.
After various iterations, the following parameter were finalized:
Steer Arm Length= 4.04 in
Steer Arm Angle= 24.2 degrees
Tie Rod length= 13.5 in

Commercially manufactured rods were used to get the required strength, durability and stress bearing capacity. Our main aim here was to prevent buckling of tie rods under stress.
The heim joints connecting the rack to the tie rods were commercially manufactured to ensure safety.

STEERING TORQUE

It is defined as the torque required to rotate the steering wheel by the driver.

Kingpin offset = 0.1073 m
Tie Rod length = 0.3429 m
Front axle weight = 97.44 kg
Width of Tyre = 0.1778 m
Friction coefficient = 0.65

$$kingpin\ torque = front\ axle\ weight \times g \times u \times sqrt\left(\frac{(width\ of\ tire)^2}{8} + kingpin\ offset^2\right) = 77.188\ N-m$$

$$steer\ torque = \frac{kingpin\ torque}{steering\ ratio} = 11.02686748\ N-m$$

NATURE OF STEERING (UNDERSTEER/ OVERSTEER)

$$Cornering\ force = M \times \frac{V^2}{r}$$

r = turning radius
FYF = Rear axle lateral force
FYR = front axle lateral force

$$FY\ =\ FYF\ +\ FYR$$

Calculation at 2.53m radius and 25 km/h speed i.e. 6.944 m/s.

$$FY\ = \frac{280 \times 6.944 \times 6.944}{2.53} = 5336.5\ N$$

Sprung mass of vehicle = 200 kg
Sprung mass at front axle and rear axle

$$Mf\ = \frac{sprung\ mass \times\ (distance\ of\ rear\ axle\ from\ CG)}{wheelbase}$$

$$Mr = \frac{sprung\ mass \times\ (distance\ of\ front\ axle\ from\ CG)}{wheelbase}$$

Using above equations,

$$Mf\ =\ 82.142\ kg$$
$$Mr\ =\ 117.854\ kg$$

At fixed radius of 2.53m and different velocities values of FYF and FYR are as follows

Table13. Lateral Forces dependence on speed

v (ms^{-1})	v (kmph)	FYF (N)	FYR (N)
0	0	0	0
1.39	5	62.73	90
2.78	10	250.91	360
4.16	15	561.864	806.14
5.55	20	1000.08	1434.86
6.99	25	1586.35	2276.03

From the book 'Fundamentals of Vehicle Dynamics: Thomas D. Gillespie', the values of slip angle are taken as:

$$\alpha f\ =\ 1.7\ deg$$
$$\alpha r\ =\ 2.5\ deg$$

Cornering stiffness (Cα)

$$C\alpha f = \frac{FYF}{af} = \frac{692.38}{1.7} = 407.28\frac{N}{deg}$$
$$C\alpha r = \frac{FYR}{ar} = \frac{993.4}{2.5} = 397.36\frac{N}{deg}$$

Weight on front axle,

$$(Wf) = MF \times 9.81 = 804.9916\,N$$

Weight on rear axle,

$$(Wr) = MR \times 9.81 = 1154.9692\,N$$

Oversteer condition

If $Wr/Car > Wf/Caf$, then vehicle will Oversteer otherwise it will have a tendency of Understeer.

$$\frac{Wr}{Car} = \frac{1154.9692}{397.36} = 2.9$$

$$\frac{Wf}{Caf} = \frac{804.9916}{407.28} = 1.977$$

$$Understeer\ coefficient = \frac{Wf}{Caf} - \frac{Wr}{Car} = -0.923$$

Since the value satisfies the condition of oversteer, therefore the vehicle has the tendency of oversteering at cornering.

BRAKING

The purpose of having a braking system in a car is to safely deaccelerate a car and bring it to a stop.

A good braking system is one in which the driver can actuate the brake pedal with ease and the vehicle retards and stops within a reasonable stopping distance with less pedal travel. The vehicle must decelerate in a controlled repeatable manner that helps maintain constant speed down-hill and must be able to hold vehicle stationary on a flat ground or on a gradient.

Actuating system of brakes can be mechanical, hydraulic or pneumatic. Hydraulic brakes use an enclosed fluid to transmit the pedal force to stop the vehicle. Force applied by the driver is multiplied in the braking system by a principle called Pascal's law. Friction between the rotating disc or drum and stationary pads are used as a tool to stop the vehicle within a considerable distance.

DESIGN

The objective was to produce light weight brake assembly, reliable throughout different terrains and effective to satisfy the wheel – lock condition. Weight and availability of space are two major constraints faced during the designing process. Initially, a balance bar system was planned so that adjustments could be made as per driver requirement. With the option to have dual brake pedals, the design was changed to have better cornering – depending on the diving skills of the driver.

CALCULATIONS

Table14. Input Parameters

Pedal Ratio	6:1
Master Cylinder Bore Diameter	5/8 in
Brake Disc Diameter	Front:5.5 in Rear: 5.5 in
Weight Distribution Ratio	34.8:65.2
Total Weight	250
Kerb Weight	180
Wheelbase	56 in

Table15. Calculated Parameters

Total braking force		1592.5 N
Dynamic weight transfer		63.84 kg
Maximum possible braking force	Front axle	960.85 N
	Rear axle	631.64 N
Braking torque required	Front wheel	164.23 N – m
	Rear wheel	88.24 N – m
Clamping force	Front wheel	1290.68 N
	Rear wheel	848.46 N
Pedal force		200*6=1200 N
Area of master cylinder		0.0002 m^2
System Pressure		600 KPa
Caliper piston diameter	front	1.24 in
	rear	1.04 in

PEDAL DESIGN

A 5mm mild sheet plate was selected as the pedal material – as per availability and FEA on ASYS'19 was performed to check the validity of the design.

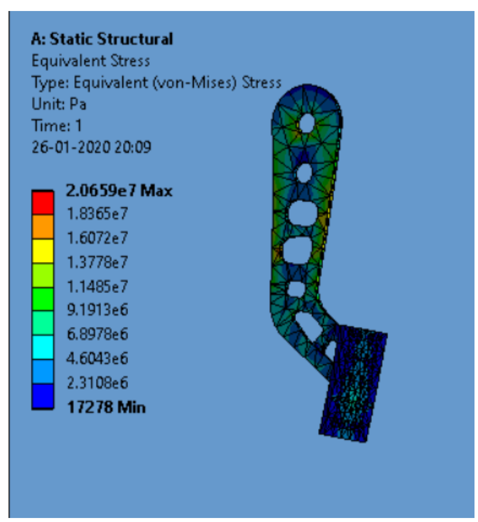


Figure 20.1: Equivalent Stress – Brake Pedal

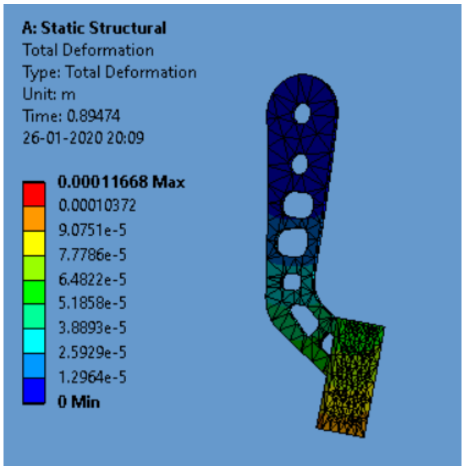


Figure 20.2: Total Deformation – Brake Pedal

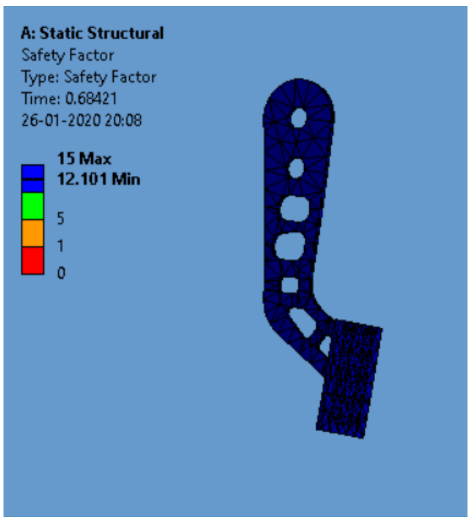


Figure 20.3: Factor of Safety – Brake Pedal

CONCLUSION

Design and construction of the BAJA SAE will result in the production of a reliable off-road vehicle which meets customer specifications. Design goals were met, resulting in a final product that will withstand the rigors of off-road travel while providing the driver with the necessary comforts. The vehicle is appealing to the customer in design, driver comfort and safety, and maintainability. The vehicle is appealing to the producer in manufacturability and reliability.

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APPENDIX