Vehicle No: 79

Design Report for BAJA SAEINDIA 2019 – Team CZAR

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ABSTRACT

BAJA SAEINDIA is a competition that fosters the growth of knowledge and skills by providing a platform for auto-enthusiasts to showcase engineering talent by building a safe and durable off-road vehicle in accordance with the rules laid out by BAJA SAEINDIA to compete in its various static and dynamic events.

This report provides a highlight of various departments objectives, their working methodology and the end results which was derived from their various considerations along with the justifications. Emphasis was laid to rectify past failures and improve performance while ensuring to provide proper corrective measures to prevent failure of any component.

INTRODUCTION

BAJA SAEINDIA aims at promoting sound engineering practices to students through the fabrication of an all-terrain vehicle, hence further providing a very technical platform for the participating students to showcase their innovative ideas and knowledge in the automobile field.

The teams raise the financial resources required and then fabricate the vehicle after designing the parameters for each component that is being used.

For the successful completion of each of the process involved, the team was divided into six primary departments (namely design, analysis, steering, suspension, powertrain and brakes), and five secondary departments (namely inventory, costing, documentation, logistics, and sponsorship). Each member of the team was allotted to one primary and one secondary department.

In addition to considering the previous team's mistakes, care was taken, not to make the vehicle over safe as it increases the cost involved and more importantly the bulk.

DESIGN

The design department is responsible for the task of realizing the kinematic models of various sub-departments into a robust, agile and compact ALL-TERRAIN VEHICLE, in accordance with the rulebook. This has to be done while ensuring that the vehicle as a whole maintains a harmonic balance between the various sub-systems, and that the assembly remains simple and sound, and the its components structurally rigid.

Design Methodology

The ATV is designed in accordance with the strict guidelines laid out in the rule book by BAJA SAEINDIA 2019, cumulative knowledge of past experiences and the goals set by the team this time. The primary objective of our ATV this year is create an agile and a compact ATV with high dynamic control.

The design was the ATV was made using NX 9 PLM software while the various designing and kinematic parameters for different subsystems was derived in ADAMS MSC software, LOTUS software, MATLAB and EES. The analysis for the designed components was tested for its structural rigidity and other parameters using NX 9 and ANSYS. Optimization analysis for weight and design was also carried out to further enhance the design all the while ensuring the components have a minimum life of a million cycles under fatigue.

Improvements

- The overall size of the ATV was reduced, with the wheel base being reduced to 57" and with a track width(F/R) of 49"/45".
- Triangulation in rear lower portion of rollcage to allow the engine to be shifted towards right which enabled in reduction of center to center distance between CVT pulleys making the transmission assembly more compact.
- The triangulation also resulted in addition of a member below CVT to help guard in case of bump.
- Single bent SIM member was used to increase knee clearance and reduce the strain of the driver's ankle, knee and thighs thus greatly improving the ergonomics.
- Reduction in overall weight of the ATV by removing redundant members.
- Incorporation of Data Acquisition System (DAQ) to gather essential data for further optimization of ATV.

Figure 1: Front view of the vehicle

Figure 2: Side view of the vehicle

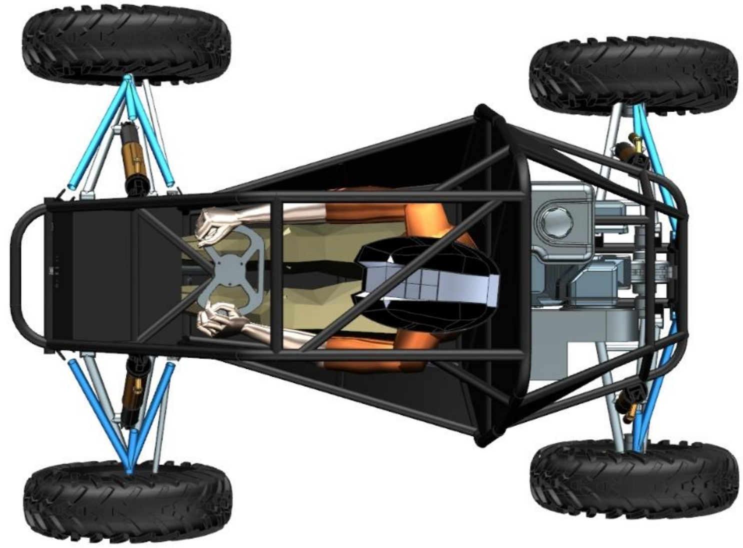


Figure 3: Top view of the vehicle

Component Design

- New design of rear upright with adjustments to change the wheels camber and toe angles.
- Custom designed and manufactured spacers to further increase misalignment capability of the heim joints.
- Further decrease in gear box dimensions for a smaller and a more compact design.
- Addition of Mechanically fastened fan to maintain the CVT temperature.

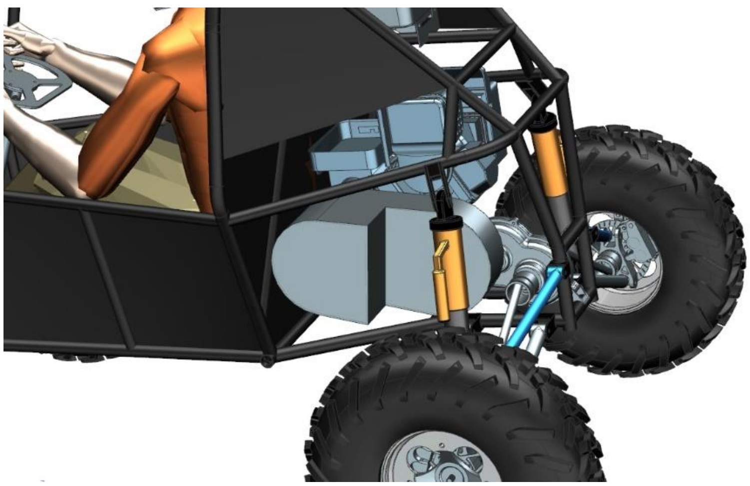


Figure 4: Triangulated member in rollcage

Ergonomic Features

- Single bend in SIM member to increase knee clearance for the driver.
- Increased leg room to reduce strain on driver's ankle, knee and thighs.

- Custom design of steering wheel for a compact and comfortable steering experience.
- Optimized angle of steering column to suit driver's comfort.
- RULA analysis was carried out while designing to optimize inner roll cage structure to improve driver comfort value (RULA Score) and using Portor 1998 analysis.
- Polyurethane foam is used in head, bottom and back seat for comfort during long driving sessions.
- Anti-slip pads added on pedal heads for more effective pedal action.

ANALYSIS

Analysis or Finite Element Analysis (FEA) is a crucial design quality and integrity control process whose primary tasks are to insure a design optimized for its structural rigidity and enhanced performance capacity. Computer aided engineering software was used to carry out this task of analyzing all the self-designed components by finite element method. These components comprise of rollcage, hub (front and rear), uprights (front and rear), A-Arms, Gear box, axels, roll bar, shafts, brake discs, tie rods, spacers and hinges. The various types of analysis that were carried out were static structural, dynamic, fatigue, thermal and modal analysis.

Parameters

Table 1: List of basic parameters considered during analysis

Parameter (Units)	Value
Mass (kg)	250
Gravitational acceleration (m/s^2)	9.81
Time of Impact(s)	0.1
Front Impact Speed(m/s)	16
Side Impact Speed(m/s)	12.5
Rear Impact Speed(m/s)	12.5
Braking torque (Front)(Nm)	183
Braking torque (Rear)(Nm)	133
Stopping time(s)	0.72
Torque (Nm)	400

Methodology

Each component is initially checked for its structural rigidity using static structural method which is followed by simulations of its actual combined loading conditions loading conditions.

The roll cage is initially analyzed in four primary cases of front, rear, side impact and roll over using static structural methods, it was then followed by dynamic and modal analysis to further check its structural rigidity and optimize its design in an interactive manner with the failure criterion of the roll cage being excessive deformation which compromise with the driver's clearances which was achieved by setting appropriate factor of safety.

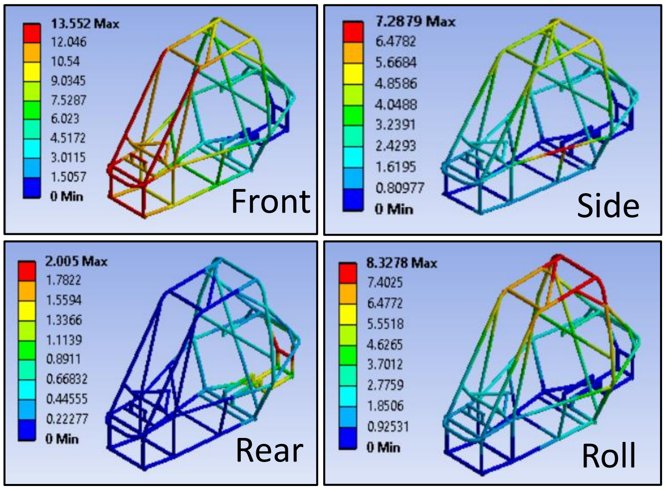


Figure 5: Rollcage analysis in different scenarios showcasing deformation.

The other components too were checked in various practical situations including impact, bump, cornering and other extreme conditions separately as well as collectively to ensure its consistent performance throughout its life. The failure criterion for these components was yielding. Each of the component was further analyzed to find out its fatigue life under soderberg criteria to ensure a minimum life of a million cycles.

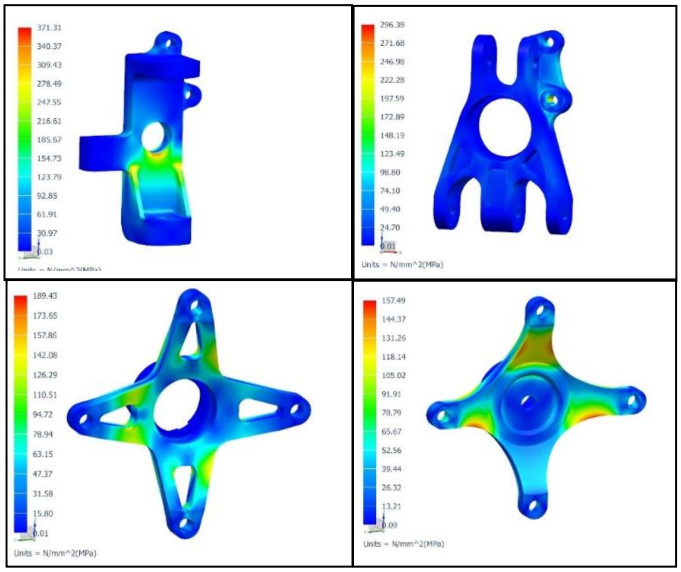


Figure 6: Upright and hub analysis (Front on left) in different scenarios showcasing von mises stress distribution.

The components were meshed using 3D hex and tetra elements of second order and 2D Quad and Tri elements of second order along with various other parameters to ensure a high mesh quality and avoid any points of singularity.

Materials Used

The material selection was given utmost importance since it is crucial for the overall strength, weight, desired endurance, safety and reliability of the vehicle.

Complying with the Baja SAEINDIA rules for 2019, AISI 4130 was used as the tubing material for the rollcage considering its higher yield strength. Hence least cross-sectional area of the tubes is optimized to attain the desired bending stiffness and strength.

Primary Member: 29.2 X 1.65 mm
Secondary Member: 25.4 X 1.00 mm

Physical Properties

Table 2: Material properties of AISI 4130

Physical Properties	
Density	7.87 g/cc
Mechanical Properties	
Tensile Strength	722 MPa
Yield Strength	761 MPa
Elongation at break	20.67%
Reduction of Area	58%
Modulus of Elasticity	205 GPa
Bulk Modulus	140 GPa
Poisson's ratio	0.29
Machinability	70%
Shear Modulus	80 GPa

Chemical Composition

Table 3: Chemical composition of AISI 4130

Chemical Composition	
Element	**Content (%)**
Carbon	0.29
Silicon	0.23
Manganese	0.495
Phosphorous	0.015
Sulphur	0.004
Chromium	0.910
Molybdenum	0.160

Other components such as uprights, hub, gear box casing and rack casing were manufactured from Aluminum 6061 T6, and Aluminum 7075. While hinges were manufactured using AISI 4340 and AISI 4140.

Data Acquisition system

This time, an offline data logging system has been incorporated to capture and store crucial data of the off-road vehicle during its testing phase and the actual event in order to further optimize its design using the more accurate results and actual scenario as captured by the data logger. The primary focus was to create a system which gathers information about the vehicles speed, acceleration, RPM of CVT pulleys and wheels and height, all calculated at the same position and same time along with a GPS tracking system which allows us to know the data of the car during its entire duration of running in the form of a map.

Also, several strain gauges will be mounted to capture the stress on the rollcage whose data will be used to minimize the number of members and optimize the pipe selection process which as a whole, results in rollcage optimization.

For the following purpose, the following are used:

- GPS module
- IR sensors
- Three axis accelerometers
- Arduino
- Data logger

POWERTRAIN

The agenda of the powertrain department is to calculate, choose and design the optimum transmission system that best fits with the team's requirement while ensuring its budget friendly and compact. The system is designed considering the teams past history, the racing track conditions, the ATVs parameters and various other dynamic and environmental parameters that might affect the torque and speed requirement of the ATV. Components such as the gearbox and axels are responsible for torque and speed transfer and hence it is essential that not only are they capable to withstand these fluctuating loads but are light in weight to reduce the overall weight of the ATV at the same time ensuring high acceleration, top speed as well as maximum output torque.

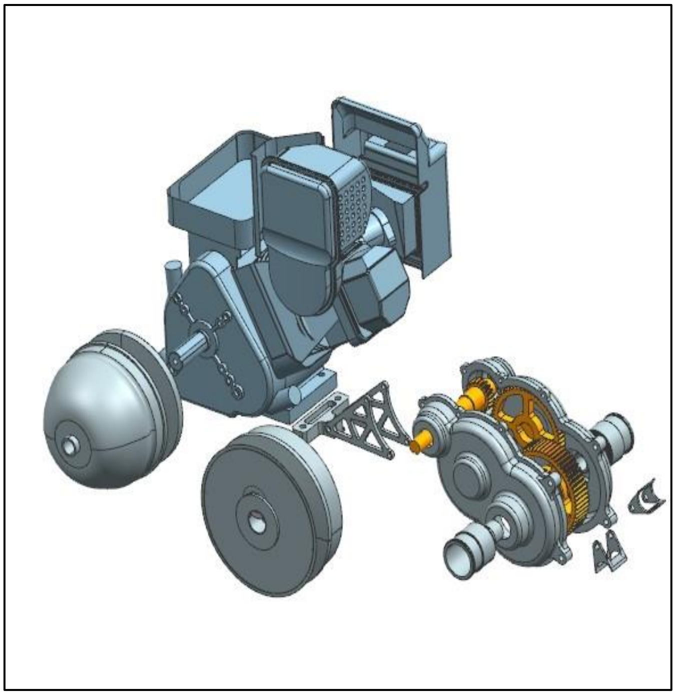


Figure 7: Expanded view of powertrain assembly

Parameters

Table 4: Specifications of powertrain assembly

Parameters	**2019**
Optimized Gear Ratio	9.33
Low	28
High	4
Gradeability	66.67
Max. Speed (Kmph)	56.1
Acceleration (M/s^2)	4.2
Centre to centre Distance (mm)	214
Ratio FAW to RAW	39.8:60.2

Sub Components

Engine - The air-cooled engine used is a 4 stroke, 305cc, 10hp Briggs & Stratton 19L100. The engine has a theoretical maximum torque of 19.65 N-m at 2800 rpm and maximum allowable speed of 3800 rpm.

Gearbox - A two-stage compound gear train type gearbox is used as a secondary transmitting unit with a gear ratio of 9.33 which ensures a top speed of 56.1 KMPH and a maximum torque output as 400 Nm is designed and implemented with further care to taken to ensure a light weight, rigid and compact design. These values were chosen after careful calculations of the AVT while factoring in the teams needs to achieve an optimum balance between the speed torque values. The gear box and the engine are mechanically fastened to each other to ensure that they vibrate in unison to ensure a constant center to center distance for the CVT pulleys attached to them.

CVT - The torque transmitting CVT used is made especially for m-BAJA application by CVTech, Canada with the low drive ratio of CVT being 3:1 and high drive ratio of 0.43:1 when the secondary pulley is fully engaged. The CVT is then further tuned to determine the appropriate shift speed and engagement speed for each of the dynamic and endurance event to obtain optimum value of speed and torque from amongst its 36 available iterations.

This time, we have also tried to create a MATLAB model for virtual tuning of our vehicle but are yet to produce any satisfactory results.

Justification - By observing the performance of last year car, the main problem faced was with the speed. As a result, this year the gear ratio was reduced to 9.33 from 10.458. So, the speed of car is increased from 47 Kmph to 56.1 Kmph.

Axle - Design of axel is based on considering it as a solid shaft under torsion and bending. Also, assuming constant torque of 400 N*m on the shaft. We have concluded to use 4340 steel solid shaft which is oil quenched with a diameter of 18mm. This material has higher strength as compared to what we use previous year, so chances for failure will be less.

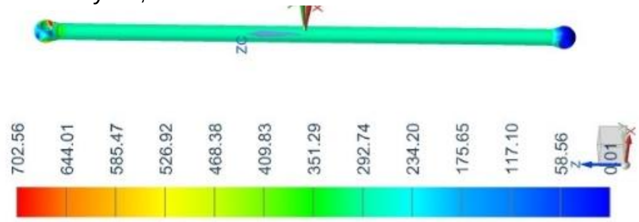
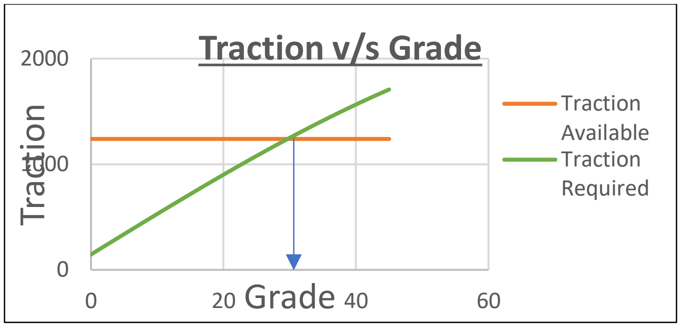


Figure 8: Structural analysis of axle showcasing von mises stress distribution.

Tires - The tire size has been configured according to ground clearance and gradeability. AT489 of Carlisle brand of size 23" X 7" X 10" has been selected which offers optimum performance in both wet and dry conditions while being lightweight, nimble in acceleration and durable throughout its demanded life.

The tire you chose for your particular vehicle will be decided majorly based on torque requirement, ground clearance and moment of inertia of the rotating wheel.



Graph 1: Traction versus Gradeability

BRAKES

The aim of this department is to provide a sufficient braking to lock all four wheels in a particular distance as per rules provided by BAJA SAEINDIA.
Some important rules required to follow are:

- Hydraulic braking system acting on all four wheels operated by a single pedal.
- At least two hydraulically operated braking circuits must be provided.

By following all the rules as stated in the rulebook provided by BAJA SAEINDIA, we have designed our brake system.

Parameters

Table 5: Parameters used by Brakes department

Pedal Force	400N
Pedal Ratio	6.5:1
Dynamic load transfer	658.58 N
Static tire roll radius	11.5"

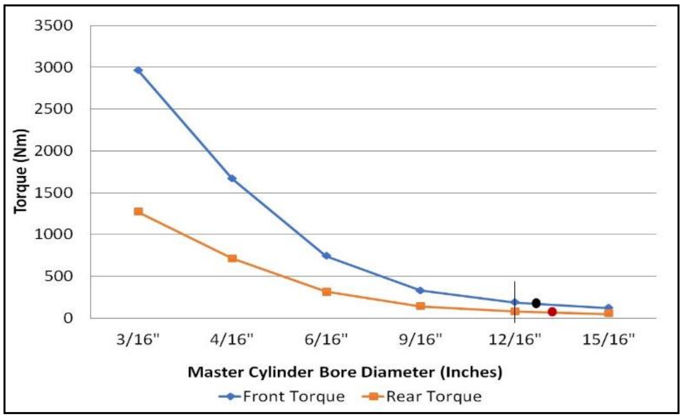
Assumptions

Table 6: Assumptions by Brakes department

Co-efficient of friction between tire and road	0.8
Co-efficient of friction between brake pads ad disk	0.42

Methodology

The force applied by the driver is transferred from pedal and is multiplied by pedal ratio. This force (Pedal Force*Pedal Ratio) is transferred to the bias bar which divides the force into proportions as per distances between the bar and the master cylinders (can be calculated by considering bias bar as beam). The force applied on master cylinder piston's area develops pressure which can be calculated by Pascal's law. Now this pressure is transferred by the incompressible fluid into calipers by flexible brake lines. The fluid then presses the caliper piston due to its increased pressure which further presses the brake pads to stop the disc. The force on brake pads is calculated by Pascal's law as we know piston area and pressure. We then multiply this force to the effective radius of the rotor i.e. from the center of the rotor to the center of the portion of the brake pad that touches the disc to get the torque provided to oppose motion of the vehicle. To get the force acting on wheels to promote motion, we draw free body diagram of a car when brakes are applied on all four wheels. From here we calculate reaction forces which when multiplied by friction forces provides us with net force acting on tire which supports motion. Now, multiplying this by radius of tire, we get required torque. Finally, we compare the required torque and the torque applied to choose the master cylinder.



Graph 2: Torque vs Master cylinder bore diameter

Results

Table 7: Specifications of Brakes department

Stopping Distance	7.87 m
Torque front available	183.88 Nm
Torque rear available	133.16 Nm
Maximum Temperature of Brake Disc	170 ˚C
Stopping Time	0.72 s

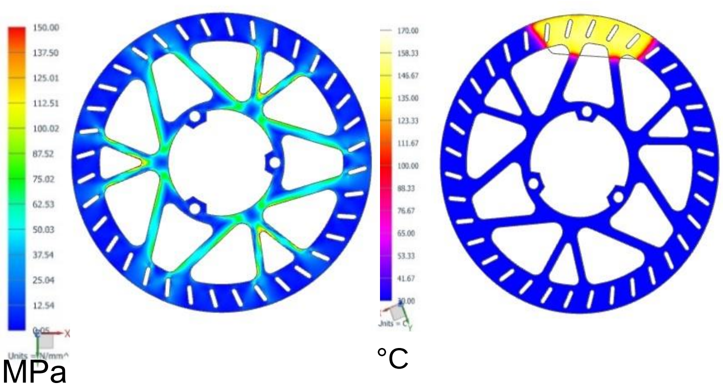


Figure 9: Structural analysis of brake disk showcasing von mises stress distribution on the left and thermal analysis of brake disk on the right showing temperature distribution.

STEERING

The steering subsystem is the most critical sub-system when it comes to the maneuverability event.

The aim was, thus, to build a steering system so as to turn the car in a minimum turning radius, with a minimum slip and minimum steering effort, while also being able to withstand the high impact stresses due to the rough terrain. Thus, the sub-system has to be made more robust while giving performance an equal importance.

The Ackermann geometry has been chosen for this year's ATV. This is because the caster effect dominates over the centrifugal force's effect in terms of lateral weight-shift. Also, since the Ackermann geometry gives a higher turning angle to the inner wheel, there would be an oversteer-effect, which is desirable along with the fact that the inner wheel shall get more traction because increased weight transferred on it.

It was also observed that in the previous Anti-Ackermann geometry, the outer-wheel was slipping a lot due to low traction on it at higher turning angle, at the speeds of around 30 kmph which again greatly favored incorporation of Ackermann geometry.

Parameters and Considerations

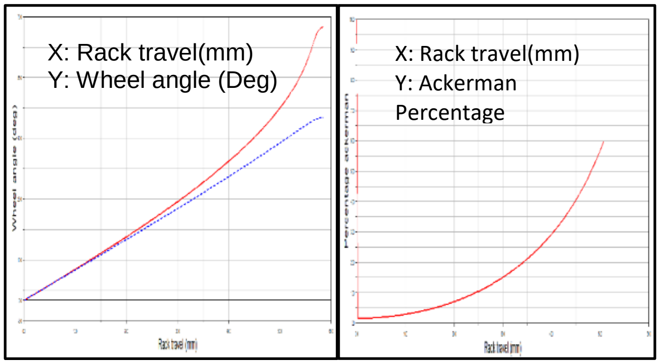
- The weight of the car is more on the rear side of the vehicle; hence traction is comparatively lower in the front.
- The centre of gravity of the ATV is higher as compared to the trackwidth length.
- Oversteer is desired for sharp cornering.
- Caster provides a weight shift to the side to which the vehicle is being turned. This is a static effect. The caster also provides a caster-trail which provides the self-aligning torque. The caster plays a role in steering effort, that is, the greater the positive caster, the lesser is the steering effort.
- A negative camber provides better traction while cornering.
- While turning, higher the speed, the more is the weight shift is towards the outer side. This is a dynamic condition effect. (It has been assumed that the speed while cornering would be low enough for the effect of the caster on the lateral weight shift is more than that due to the centrifugal force while turning).
- The kingpin inclination, which affects steering effort, and self-aligning torque.
- The scrub-radius also affects the self-aligning torque. The more the scrub-radius, the higher the self-aligning torque.

Methodology

- The kinematic analysis was done in MSC ADAMS. Also, motion simulation was done on NX 9.0 of the steering-suspension assembly.
- The rack and pinion were placed such that the instantaneous centre of the tie rod is coincident with that of the wishbones, thus minimizing bump-steer.
- The maximum rack-travel was restricted up to the locking of the four bar-bar chain.
- The rack and pinion gear-ratio was decided keeping in mind the sensitivity according to the steering-ratio required by the driver.
- The caster angle has been kept at 14 degrees. This was decided on the basis of the directional stability,

steering effort and decrease the impact forces on the knuckle during a bump travel.

- An important parameter is camber gain, which is to be increased for a better traction.



Graph 3: Rack travel versus wheel angle on the left and Rack travel vs Ackerman percentage on the right

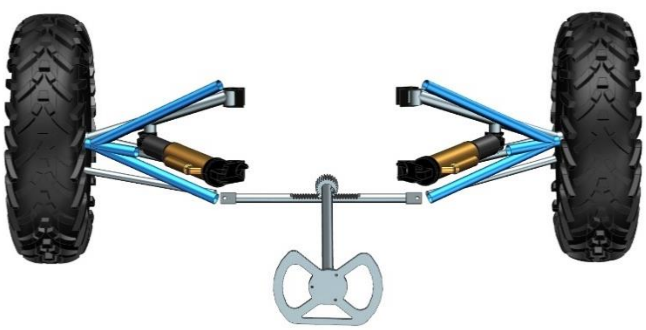


Figure 10: Assembly of steering system showcasing Ackermann geometry.

Results

Table 8: Specifications of steering department

Parameter	Value
Steering Geometry	Ackerman- 59.9 %
Inner/Outer Wheel Angle	53.4°/45.2°
Turning Radius	1.5 m-2.2m

SUSPENSION

Suspension sub system is a crucial subsystem especially for an off-road vehicle. The aim of the department was to build a vehicle to obtain the 3C's which are Control, Contact and Comfort.

In order to achieve the said objectives, double A-Arm type geometry was selected for the front system since it is the most suitable type geometry for an open wheel vehicle which also ensures that stability is obtained for the same while cornering. H-Arm with upper link was selected for the rear system to decrease the axle plunge.

Air dampers were selected since it gives the best value of damping ratio and the stiffness can be altered according to the comfort or terrain. Also jounce or bounce rates can be altered to further improve the performance.

Parameters

Table 9: Parameters used by suspension department

Parameter	Font	Rear
Mass Distribution	86.51 kg	117.25 kg
Sprung mass/Un-sprung mass	63.84 kg/22.64 kg	99.05 kg/18.2 kg
Ground clearance	15.5”	14.7”
Roll Centre height	13.33”	11.68”
COG position	531mm (20.9”)	
Suspension travel	Jounce=2”, Bounce =6”	

Methodology

In order to obtain the most suitable suspension geometry, The above-mentioned parameters were determined considering the design parameters and on the basis of ruggedness of the terrain and robustness of the vehicle. Software's such as MSC ADAMS, NX Dynamic Simulator.

The kinematic geometry for the same was prepared in Lotus using optimizer and modifications were made to obtain the best combination of the parameters stated above. Secondary parameters like ride rate, cornering stiffness, frontal pitch, yaw rates, thrust forces etc. were determined considering the designed weight, inertia and the nature of reaction forces.

Motion ratio was set so as to satisfy all the secondary parameters. Steering parameters and characteristics were also taken into consideration for the same.

Geometry was then designed in NX and its dynamic testing was performed to check the performance and robustness of the vehicle in dynamic environment. In the same dynamic environment, rugged terrain and extreme conditions was created for fine tuning of the designed suspension geometry to eliminate any problems.

Anti-roll bar was designed for the purpose of decreasing the roll angle of the rear system while cornering and hence avoiding the roll over.

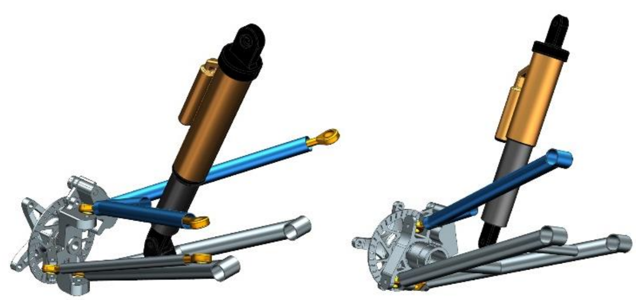
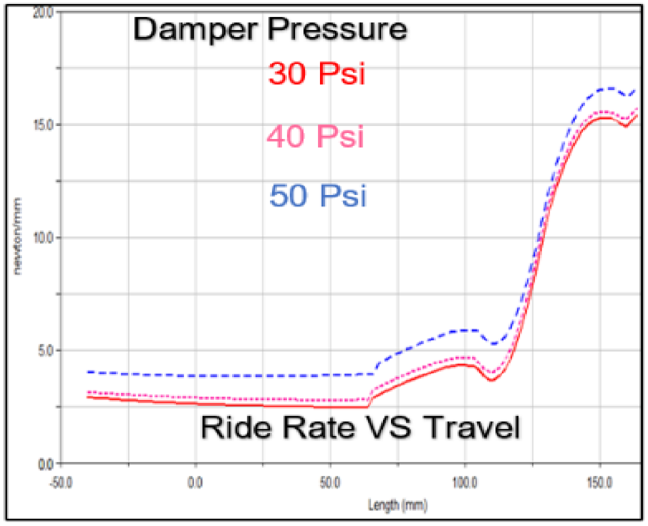


Figure 11: Suspension assembly for front on the left and rear on the right.

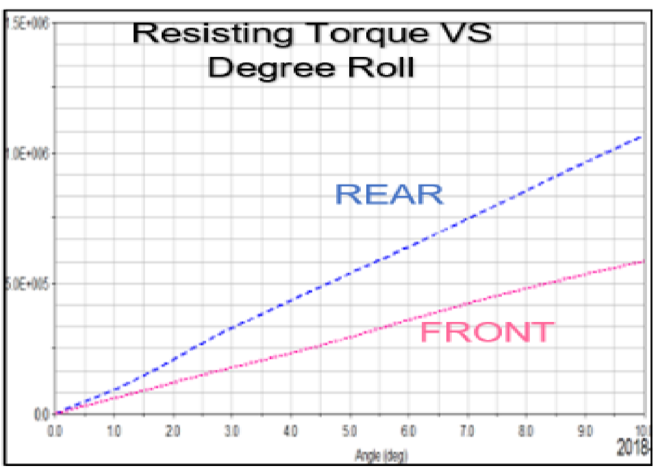
Results

- Car is inclined by nearly 5 degrees to rear with respect to front, this is done to prevent the frontal roll over due to pitching.

- Anti-dive geometry is introduced to reduce the effect of jacking forces.
- Ride rate is determined by keeping into consideration the cornering stiffness.
- Anti-roll bar is designed to reduce roll of 3 to 4 degrees for a roll of 7 degrees
- Damping ratios determined for different bump conditions and bounce jounce rates adjusted to decrease the force transmitted to the chassis.
- Vehicle dynamically simulated in NX taking into consideration all the dynamic parameters of the environment and the vehicle.
- Caster angle to increase the inner wheel load to nearly 25% while cornering, which is suitable for Ackermann steering geometry.



Graph 4: Ride rate versus wheel travel for different setting of suspension pressure



Graph 5: Resisting torque versus Degree roll for front and rear suspension geometries

Table 10: Specifications of suspension department

Parameters	Front	Rear
Camber Angle	-0.5^0	0^0
Toe angle	0^0	0^0
Motion Ratio	0.47	0.55
Ride Rate	5.13 – 845 N/mm	8.15 – 12.38 N/mm
Desired Natural Frequency	1.96 – 2.47 Hz	2.14 – 2.58 Hz
Caster Angle	14^0	
Kingpin Angle	6^0	
Scrub radius	0.49”	

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