Vehicle Number: 134

Design Report for BAJA SAEINDIA 2020 – Team CZAR

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DESIGN

The design department is at the helm of incorporating the kinematic models of various sub-departments into a light-weight and vigorous All-Terrain Vehicle consistent with the rulebook. While doing so, the simplicity and strength of each component need to be maintained and all systems must be congruous together.

Design methodology

The ATV was designed in NX 12 PLM software and all the subsystems' kinematic parameters were extracted using MSC Adams, LOTUS and MATLAB. Analysis of the components was done using NX's Nastran solver and Ansys. Geometric Optimization was also done for enhancing the strength-weight ratio furthermore a minimum life of a million cycles under fatigue was also ensured. Chassis was also checked with RULA Analysis to avoid strain on the driver.

Improvements

Overall improvements

- The overall size of the ATV was reduced to improve vehicle dynamics, the wheelbase was reduced by 4 inches and brought to 53' and track width (F/R) is 48'/45'
- The engine was moved to the right to obtain the COG of the ATV closer to the center.
- Double bent SIM was used to give more space to the driver.
- Reduction in the overall weight of the chassis by improving the brake assembly mounts and getting rid of FAB by reducing the angle between FBMUP and vertical and increasing the size of RRH&RHO gusset to get equivalent strength as last year.
- Increased the angle of attack for better off-road ability.
- Incorporation of even more Data Acquisition features for collection necessary data for further improvement of the ATV and live data feed also added for better planning.

Component improvements

- Drastically increased the FOS/Weight ratio by using Al 7075 over Al 6061.
- Gear box's dimension were reduced to make it more compact.
- Nylon bushes used in hinges for reducing weight.
- Changed the bearing in front wheel assembly to reduce the hub offset and decrease the weight of the wheel assembly.
- Wider rack and pinion casing designed for better protection of the rack during impact.

Ergonomic Features

- Chassis, Brake pedal, accelerator pedal and steering wheel designed and positioned with REBA analysis in consideration for better ergonomics.
- Double bent added to the SIM to provide more space to the driver around his knees.
- Steering wheel's design was changed for a more comfortable operation.

Figure 1: Front view of the vehicle

Figure 2: Side view of the vehicle

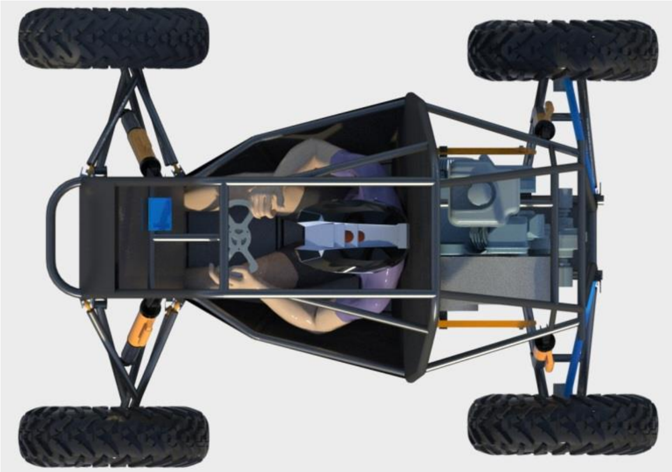

Figure 3: Top view of the vehicle

Figure 4: Triangulated member in roll-cage

ANALYSIS

To achieve the objectives of the FEA activities performed this year, it was necessary to have a shift in the philosophy. For this, emphasis was placed on accurately replicating the loading scenarios and modelling them using software tools adequately to achieve the best results possible, thereby improving the reliability of the results obtained. The simulations were modelled so as to accurately mimic actual scenarios thus facilitating the design process.
The goals of the department were to add new simulation techniques to the design process and to accurately replicate the physical scenarios.

Improvements

To achieve the objectives of the FEA activities, the new simulation techniques added to the design process were-

- Assembly FEM to accurately predict the behavior of a set of inter-connected components to eliminate the assumption of other bodies as rigid bodies.
- Fatigue analysis for an estimate of the life of components in the vehicle.
- Computational Fluid Dynamics to predict the temperatures of fluid inside the CVT cover region and at the pulleys.

Assembly FEM

The assumption taken in most simulations is that all other parts connected to the component being analyzed are rigid. This assumption sets a limit on the optimization possible as this results in higher stress than actual scenarios in the component being analyzed. To accurately predict the behavior of these components it is essential to remove this assumption and for that sub-assemblies were analyzed as a whole. This led to more accurate results as the deformation of other components meant the stress induced in the component of interest were lower, thereby providing grounds for further optimization.

Table 1: Conditions used for setting up the simulation of a front wheel sub-assembly.

Mesh Element Type	3D Tetrahedral (all)
Mesh Element Size	2mm
Constraint type	Fixed (suspension points)
Forces	3G bump + 2G corner + braking + steering
Simulation tools	Surface to surface glue

Using the parameters mentioned in table 1, the assembly FEM of the front and rear wheel sub-assemblies was performed. The results obtained were then compared with static structural analysis of the part under study. It was found that the stress generated in the components was lower in assembly FEM due to the ability of other connected components to share the load.

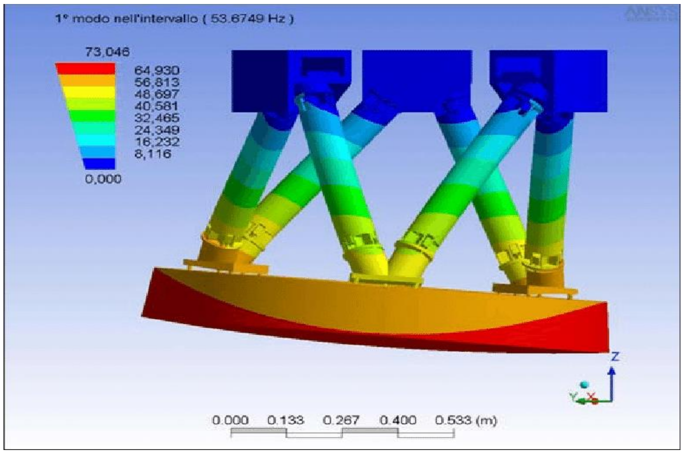

Figure 1: Von-Mises stress results from the assembly FEM with the specified conditions in Table 1.

By the use of this technique we could reduce the un-sprung mass of the vehicle by 800g, which showed a major improvement in the vehicle's dynamics.

Computational Fluid Dynamics

The objective of performing a Computational Fluid Dynamics analysis was to get understanding of the temperatures at which the CVT pulleys will exist, so that the design process of the CVT cover can be made more efficient and effective, so as to prevent excess heating in the CVT and thus maintain the performance of the transmission system

Table 3: Conditions for setup of CFD analysis of CVT cover

Solid object Mesh element type	3D tetrahedral
Fluid Mesh Element Type	3D tetrahedral
Solid object mesh element size	2 mm
Fluid mesh element size	3 mm
Thermal Loads	800 W (primary) 720 W (secondary)
Constraints	Convection to environment
Simulation Tools	Solid Blockage (pulley) Inlet velocity (10 m/s)

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The result of the CFD analysis was obtained at steady state conditions, and proved that the design of CVT cover under investigation provided sufficient cooling without the addition of an external fan, which simplifies the design all while keeping it effective. The results of the analysis showed satisfactory temperature of the CVT pulleys.

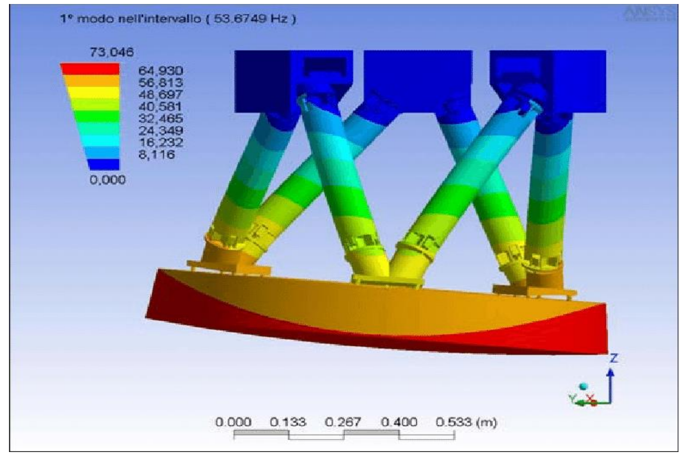

Figure 2: Results obtained under steady state conditions from the specified simulation setup

The Model was checked for maximum fluid temperature and maximum pulley temperature.

Thus, we have achieved our target of keeping the temperature in the CVT cover below a certain threshold value to improve the efficiency of the transmission system.

Fatigue Analysis

The wheel sub-assemblies are subjected to not just static loads, but dynamic loads too. It is often these loads that are the reason for the failure of major load bearing components in the wheel sub-assemblies, and thus it becomes essential to perform durability checks on all these components to ensure their longevity.
The acceptability criteria of our fatigue analysis were taken as 10^5 loading cycles. This value, as well as the value of the load was found by analyzing data obtained from the previous year's endurance event, to accurately model the loading conditions. The fatigue criteria were taken as the Soderberg criteria.

Table 5: Simulation setup of the fatigue analysis of front hub

Mesh Element Type	3D-tetrahedral
Mesh Element Size	2 mm
Loads	4000N (bump) + 2500N (cornering) + braking
Constraints	Fixed (bearing holes)
Loading cycle	Fully-reversed cycle
Fatigue criteria	Soderberg

The results obtained from the fatigue analysis were deemed as satisfactory and the design for the given component was finalized.

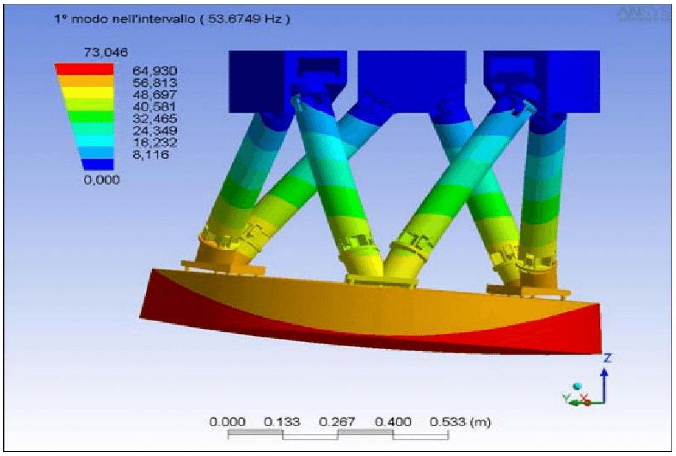

Figure 3: No. of cycles endured by the component

As the life of the component cleared the acceptability criteria, the component was deemed to be acceptable, thus having ensured the longevity of the component.

DAQ

Employing the already existing systems, the primary aim was to rectify the failures while incorporating additional sensors for further improvisation. Offline data logging systems have been incorporated for capturing and storing data for the entire duration of testing phase and the event. The system gathers information and data about vehicle's Speed, acceleration, RPM of CVT pulleys and wheels, CVT temperature and forces on the wishbones, simultaneously along with the data from GPS system which provides mapping of the data to the location on the track, thus enabling better analysis with respect to the obstacles. In addition to the above systems, ZigBee modules were used for transmitting live location to the pit, for optimized utilization of resources. A GSM module was used for driver-to-pit communication, and additionally a RPM display was placed in the cockpit.
The forces and data hence obtained was used for analyzing the components for weight reduction and the validation of real-time forces acting on the components, namely hub, knuckle and wishbones. For the following purpose, the following were used:

- GPS Module - Location
- Hall Sensors - RPM
- Three axis Accelerometer & Gyroscope - Forces
- Temperature Sensor
- ZigBee Pro module - RF Data Transmission
- GSM Module - Driver Communication
- Data Logger - Offline Data Storage
- Arduino - Micro controller

POWERTRAIN

The agenda of the powertrain department is to calculate, choose and design the optimum transmission system that suites the team's requirement while ensuring its cost-efficiency, effectiveness and compactness. The system is designed considering the criteria including the team's past performance, tractive effort, acceleration, gradeability, required speed and torque of the vehicle and environmental parameters that might affect the desired performance of the ATV. Components such as the gearbox and axles are responsible for torque and speed transfer, for which it is not only essential to give maximum output acceleration, speed and torque, but the ability to withstand fluctuating loads and light-weightiness to reduce the overall weight of the vehicle are also crucial.

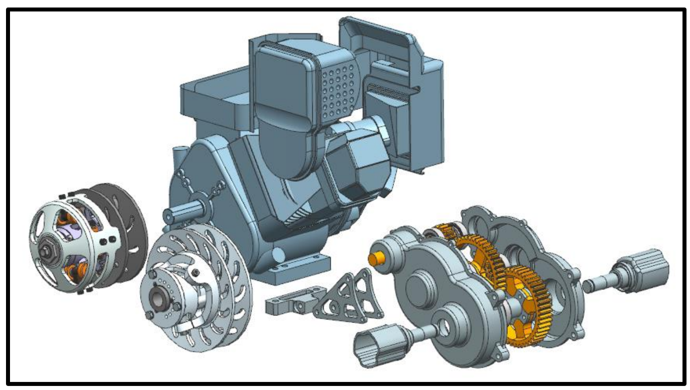

Figure 3: Powertrain layout

Parameters

Table Specifications of powertrain assembly

Parameters	2020
Optimized gear ratio	7.64
Low	6.8
High	29.8
Gradeability (%)	71
Maximum speed (kmph)	53
Acceleration (m/s^2)	5.9
Centre to centre distance (mm)	214
Ration FAW to RAW	42:58

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Sub-components

Engine- The air-cooled engine used is a 4-stroke, 305cc, 10hp Briggs and Stratton 19L2320054. 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 53 kmph and a maximum torque output of 425 N-m. It is designed and implemented with further care taken to ensure a light weight, rigid and compact design. These values were chosen after careful calculations of the ATV 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 and that a constant center to center distance for the CVT pulleys attached to them is maintained.

CVT – The CVT used is Gaged GX-9, with the low drive ratio of the CVT being 3.9:1, and the high drive ratio of 0.9:1, when the secondary pulley is fully engaged. The CVT has been chosen keeping in mind, it's weight, the ratios provided and the incorporated roller fly-weights, which improves the time response, compared to the CVTech CVT that was used in the previous season. 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.
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 initial torque and the inability of the CVT to fully shift and thus, not utilizing it to the fullest extent. So, accordingly changes has been made in the CVT choice this year which provided us a better initial torque and better time response compared to CVtech CVT.

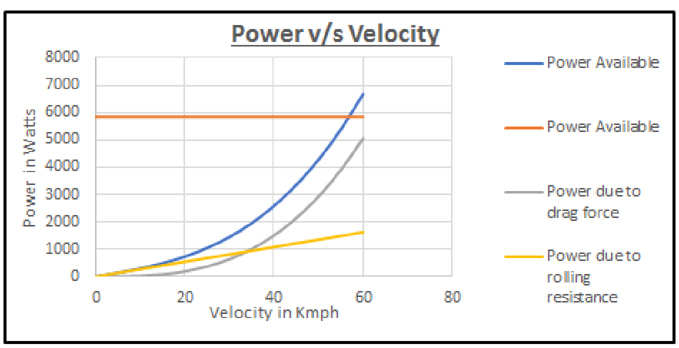

Figure 3: Power versus velocity

Axle - Design of axel is based on considering it as a solid shaft under torsion and bending. Also, assuming constant torque of 425 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.

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.

NVH Considerations

The engine and the gearbox 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. A sheet of polyurethane is used to damp the engine vibrations. The vibrational damping of the polyurethane sheet has been validated by the use of accelerometers, which on comparison with rubber sheets gives better results.

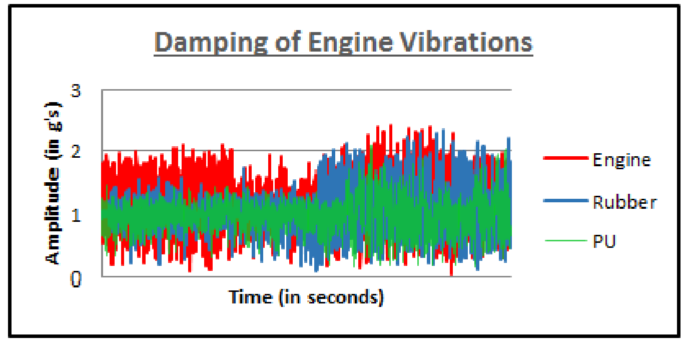

Figure 3: Plot for amplitude of the vibrations

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 (N)	350
Pedal Ratio	6.5:1
Dynamic load transfer (N)	745.92
Static tire roll radius (inches)	11.5

Assumptions

Table 6: Assumptions by Brakes department

Co-efficient of friction between tire and road	0.9
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 pivot and the plungers of the master cylinders (which 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 through 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 the 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 the vehicle when brakes are applied on all four wheels. From here we calculate reaction forces which when multiplied by friction forces provides us with the net force acting on tires which supports motion. Now, multiplying this by radius of tires, we get the required torque. Finally, we compare the required torque and the torque applied to choose the brake-calipers.

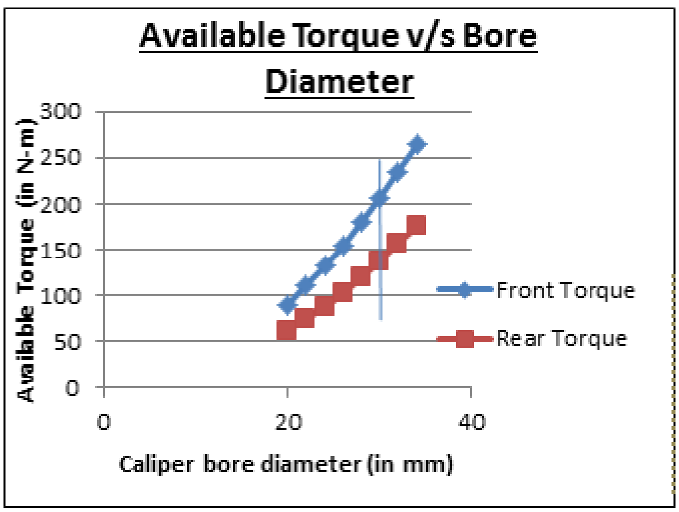

Figure 3: Caliper bore dia. Vs available torque

Results

Table 7: Specifications of Brakes department

Stopping Distance (m)	6.99 m
Torque front available (N-m)	206.19
Torque rear available (N-m)	137.08
Maximum Temperature of Brake Disc (°C)	170
Stopping Time (seconds)	0.65

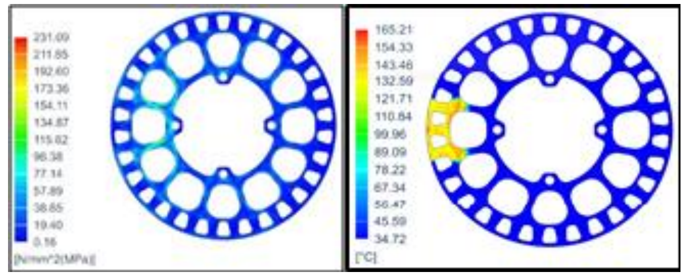

Figure 9: Structural analysis of brake disk showcasing von mises stress distribution on the left and thermal analysis of brake disk on the right showing temperature distribution.

STEERING

When it comes to the maneuverability event, the steering sub-system is the most crucial sub-system. It is also one of the most important components, when it comes to handling, and ride characteristics. The ultimate aim of the steering-subsystem is to minimize the turning radius. For this cornering stability, minimum slip, and robustness to endure the rough and strength-testing terrain are some requisites.

We have implemented the Ackermann geometry for this year's ATV. This is because caster's effect dominates over the centrifugal force's effect in terms of lateral weight-shift, at speeds at which the ATV would generally operate, and most importantly on the maneuverability event track. 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. Considering the close dependency of the steering and suspension sub-systems on each other, we are running high caster, which leads to its effect on lateral load-shift dominating. Considering the close dependency of the steering and suspension sub-systems on each other, we are running high caster, which leads to its effect on lateral load-shift dominating.

We were able to reduce the slipping issue in last year's ATV, as compared to its predecessor. But there was still observable slipping, while cornering at steady-state speeds. To counter this, we have changed the fore-aft weight distribution. The FAW has been increased to 42% from 40%. This would result in an increased lateral traction in the front.

Parameters and Considerations

- The weight distribution of the vehicle is more towards the rear; hence lateral traction that we can exploit is limited.
- The wheel-base of the car is to be kept to a minimum, to reduce the turning radius.
- The center of gravity of the ATV is higher as compared to the trackwidth length.
- The maneuverability event has sharp corners. This cannot be tackled without an oversteer characteristic.
- Caster provides a weight shift to the side to which the vehicle is being turned in the Ackermann geometry. This is a static effect, that is, it can be observed even when the car is not moving. The caster also provides a caster-trail which provides the self-aligning torque. Caster angle plays a crucial role in 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, during the maneuverability event).
- The kingpin-inclination, which affects steering effort, and self-aligning torque.
- The scrub-radius also affects the steering effort. More the scrub-radius, higher the steering effort.

Methodology

- The kinematic analysis was done in MSC ADAMS.
- The rack was placed such that the instantaneous centre of the tie-rod is coincident with that of the wishbones, thus minimizing bump-steer.
- For this year, the main motive of the design methodology was to reduce the iterations being done while using some boundary conditions.
- Zero bump steer was used as the boundary condition.
- For this, the equations of the planes of the suspension front control-arms were calculated. Then the line of intersection of these planes was taken. Then the locus of points lying in the plane of the inboard points of the control arms, where the projection of the tie-rod would pass through the line earlier found. This would be the ideal condition for bump-steer to be zero.
- The rack length was hence, first constrained in the range of the lateral position of the inboard side coordinates of the control-arms.
- Thus, a restricted range of heights (Z-coordinates) with respect to a given longitudinal coordinate (X-coordinate) was obtained. Hence, the iterations were done only for this restricted range. This saved much of the fine-tuning time. This also improved the results, because more iterations could be carried out with the relevant values of the various parameters.
- From this, the results for the optimum rack length and longitudinal rack position were obtained.
- The maximum rack-travel was restricted up to the locking of the equivalent four bar-bar chain.
- The rack and pinion's gear-ratio was decided keeping in mind the sensitivity according to the steering-ratio required by the driver.
- The pinion pitch diameter was calculated using the lateral travel required for the maximum travel and the angle it has to be rotated to achieve it. According to the driver the best angle was 135°.
- The caster has been changed from 14° to 12°. It was changed to reduce the steering effort.
- For calculations related to dynamics, we used NX's motion simulation. From these simulations, various forces were obtained, which were used for structural analysis.
- Also, graphs related to various parameters like steering-angles, camber and caster change in bump travel, etc. were obtained.

Component improvements

Rack and pinion-

- A custom rack and pinion was designed for this year.
- The main problems in last year's design were:
    1. The bending stress was high. This resulted in the rack getting bent during the testing phase.
    2. The losses in the casing were very high. This resulted in a high steering effort. The main reasons for these losses were the dirt entering the bearing (because it was open from one side) and sliding friction between the rack and the casing.
- To reduce the bending stress, two changes were made:
    1. The diameter of the rack was increased from 20mm to 22mm
    2. The length of the casing was increased from 120mm to 170mm
- To tackle the steering effort problem, two changes were made:
    1. Brass bushes were introduced. This reduces the coefficient of sliding friction between the rack and the casing, because of contact of dissimilar metals.
    2. The clearance between the input shaft of the pinion and the hole for the input shaft was minimized. This minimized the chances of dirt entering the bearing.

Steering column assembly-

- Another issue contributing to the steering effort was the sliding friction that was present because of the normal force that the column tube was exerting on the mounting tube. This was because no bearing was used.
- So, to prevent this, for this year, we have used a steering bearing, which is externally spherical
- This has the advantage that, when the assembly is being done, the column can be first adjusted according to the position of the rack, and then tightened. This ensures that the pinion input shaft, and the column mounting are exactly coaxial.

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 adjust and 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.
Air Dampers are 50% lighter in weight compared to coil-over shocks and can give the same optimum results by tuning its valving system to get highest possible Rebound damping without getting the shocks in bottoming-out situation.

Parameters

Parameter	Font	Rear
Mass Distribution	91.51 kg	126.4 kg

Sprung mass/Un-sprung mass	68.5 kg/25 kg	102.7 kg/26.24 kg
Ground clearance	15.5"	14.5"
Roll Centre height	12.85"	11.02"
COG position	529mm (20.8")	
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 Lotus Shark, MSC ADAMS and NX Dynamic Simulator were used.

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.

The Front and Rear geometry are designed with proper 'Anti' characteristics to balance the longitudinal load transfer of vehicle. These properties give reduced load stresses in wheel assembly components.

The suspension system is differentiated in three parts i.e. Outboard suspension, Inboard suspension and the central suspension system.
The outboard suspension includes the wheel alignment and operating angles as stated below:

Table 7: wheel-alignment and operating angles

Parameters	Front	Rear	Reasons
Camber angle	-0.5º	0.0º	A small value negative camber is desirable for higher addition in kerb weight to maintain negative camber change.
Toe angle	0º	0º	Toe angle can be adjusted performance wise but is kept zero to get good acceleration performance.
Caster angle	12º		Optimized to get better lateral load shift for steering performance and minimize driver effort.
Kingpin angle	6.92º		
Scrub radius	1.01"		Increased Aligning Torque and reduce scrub to reduce the wheel moment effort.

The inboard suspension incudes an anti-roll bar in the rear suspension system which is adjustable and was designed for the purpose of decreasing the roll angle of the rear system while sharp and high-speed cornering and hence avoiding the roll over as per track conditions.
The Central system includes the wishbone assemblies and Air damper with its geometrical mountings designed as per Anti-properties and Motion Ratio whose dynamic results are as stated below:

Table 7: Specifications of Brakes department

Parameters	Front	Rear
Motion Ratio	0.49	0.57
Ride Rates	6.4 – 10.5 N/mm	9.6 – 13.4 N/mm
Desired Natural Frequency	1.71 – 2.3 Hz	2.12 – 2.5 Hz

Results

- Car is inclined by nearly 10 to rear with respect to front, this is done to prevent the frontal roll over due to pitching.
- Bump caster is decreased by 35% to reduce steering effort by retaining anti-Dive properties.
- Ride rate is determined by keeping into consideration the cornering stiffness.
- The Anti-Squat geometry incorporated in Rear suspension increases the tractive force.
- Anti-roll bar is designed to reduce roll from 8 degrees to 5 degrees and 3 degrees at respective tuning by changes in arm length.
- 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.
- Decrease in unsprung mass leads to better dynamic performance by isolating the response gain through wheel hop frequency by approximately 26%.