

Research Article

Optimum Design and Material Selection of Baja Vehicle

Abhinav Sharma^{†*}, Jujhar Singh[†] and Ashwani Kumar[†]

[†]Mechanical Engineering, Rayat-Bahra Institute of Engineering and Bio-Technology, Mohali, India

Accepted 20 June 2015, Available online 27 June 2015, Vol.5, No.3 (June 2015)

Abstract

This paper provides a complete design and analysis of “Baja vehicle” or “All-Terrain Vehicle (ATV)”. While designing this Baja vehicle, all the design aspects were taken as per the rules of Society of Automotive Engineers (SAE)-2014. The main objective of this paper was to design and optimize the roll cage, front and rear suspension system, power train system. The finite element analysis (FEA) is also done on the roll cage for validating the design. Initially, a preliminary design of the roll cage was made as a 3-D model using CAD. The designed Baja vehicle is an off-road vehicle powered by 350 cc, four strokes, 10 BHP engine, driven by manual transmission. Material selection was based on the basis of factors like weight, cost, availability and performance. In present work, the overall design of vehicle has been categorized into subsystems: Roll cage, engine, Powertrain, suspension, steering, braking and ergonomics. This paper also provides an overview of the design for each of the subsystems of the vehicle. The designed and fabricated subsystems are equally strong and rugged.

Keywords: SAE Baja, All-Terrain Vehicle, Off-Road Vehicle, Baja Roll Cage, Baja Suspension.

1. Introduction

The term “Baja” or “All-Terrain vehicle” is used in a general sense to describe any of a number of small open motorized buggies and tricycles designed for off-road use. Baja or all-terrain vehicle as its name suggests is a single seated off-road vehicle that travels on low-pressure tires, with a seat that is straddled by the driver along with rack and pinion as steering mechanism for steering control. As its name implies, it is designed to handle a wider variety of terrain than most other vehicles. This vehicle is also known as quad, quad bike, three-wheeler, or four-wheeler. In three-wheeled all-terrain vehicle, the rider sits on and operates these vehicles like a motorcycle, but in case of four wheeled all-terrain vehicle the extra wheels give more stability at slower speeds. Although equipped with three or four wheels, six-wheel models exist for specialized applications. The All-Terrain vehicle cannot claim a history as deep or as long as the motorcycle but it certainly rouses the passion of owners and riders everywhere. This paper provides the complete design and analysis of Baja vehicle which is a four wheeled, powered by 350cc, four stroke, 10BHP (OHV-Intek) engine, driven by manual transmission and its body consisted of roll cage and frame of steel tubes is capable of taking abuse from rugged and off-road driving conditions in any weather conditions. Typical capabilities on basis of which Baja vehicle can be judged are hill climb, load pull, rock crawl, acceleration,

manoeuvrability, endurance on land as well as water and farming etc.

According to Baja SAE International rules 2014, engine could not be enhanced in any way to ensure uniform comparison of overall vehicle design. Thus, a large part of vehicle performance depends on the drive train and the manoeuvrability of the vehicle. By improving the drive-train efficiency, the vehicle was able to accelerate faster and achieve a higher top speed. The total weight of the vehicle including the driver weight has significant impact on performance. Overall, a light vehicle should perform better since the engine capacity is fixed. While designing the factors considered are safety and ergonomics, cost, market availability, weight and balance, endurance, standardization, serviceability and maneuverability.

Dessault Systems Catia V5 R21 software was used for designing and it was analyzed on ANSYS - 14. Figure 1.1 shows the isometric view of Baja vehicle and figure 1.2 to figure 1.5 shows the different views of the vehicle.

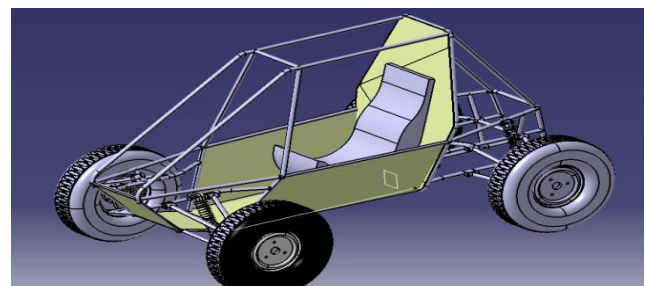


Fig.1.1: Isometric View of Baja Vehicle.

*Corresponding author: Abhinav Sharma

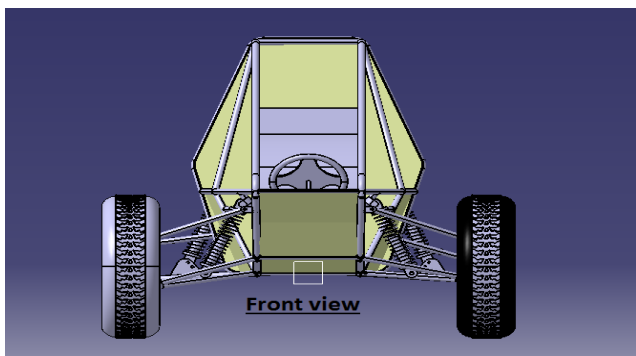


Fig.1.2: Front View of the Vehicle

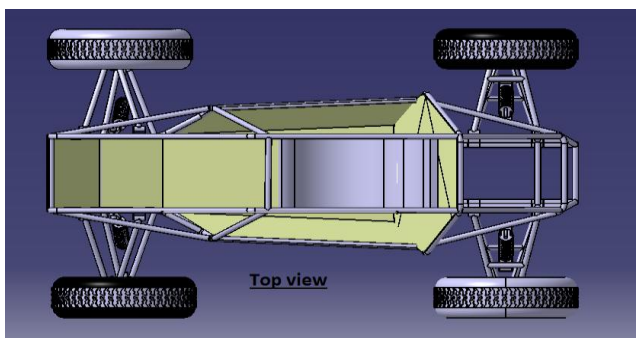


Fig.1.3: Top View of the Vehicle

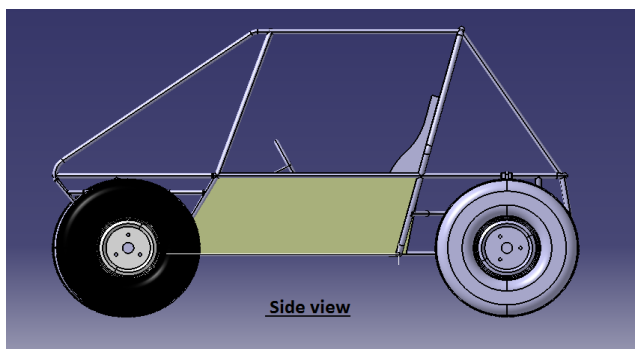


Fig.1.3: Side View of Vehicle

2. Roll Cage

The Roll Cage in a Baja vehicle is the most important basis for the entire design. The roll cage plays a crucial role in providing the desired strength, endurance, safety and reliability to the vehicle. It is the backbone of Baja vehicle as it interacts with every other system of the vehicle. The roll cage is designed in such a way that the driver seat, engine, transmission system, suspension system, brake system, fuel system and steering mechanism can be mounted on it. The objectives considered were that the roll cage must be designed with high yield and tensile strength steel tubes as a triangulated space frame, number of welded joints should be very less in favour of bent joints, strength and weight ratios should be maintained at all times when vehicle is in dynamic mode, must provide maximum spaces for the moving parts, must be designed in such a way that provides maximum driving

reliability and most importantly the driver’s safety, must have ease of serviceability by ensuring that the roll cage members do not interfere with other subsystems and the roll cage members should maintain their integrity in order to protect the driver in the event of a rollover or any impact. Figures 1.4 to 1.6 shows the various views of roll cage

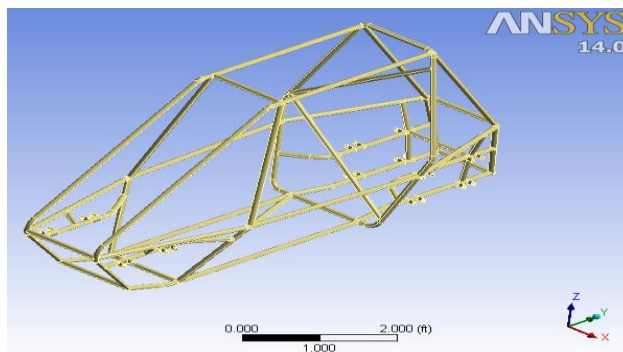


Fig. 1.4: Isometric View of Roll Cage

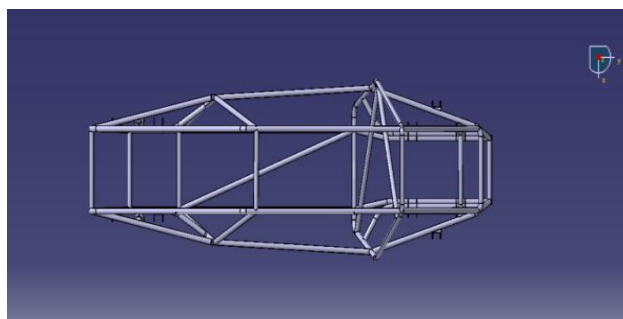


Fig.1.5: Top View of Roll Cage

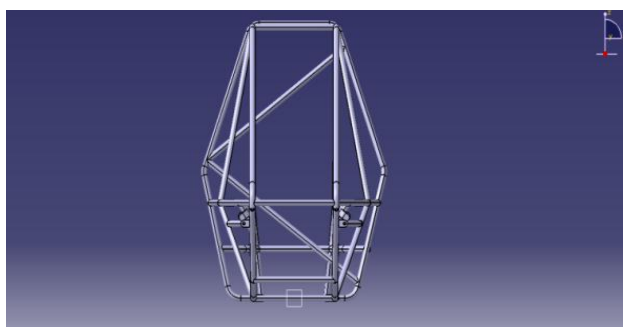


Fig.1.5: Rear View of Roll Cage

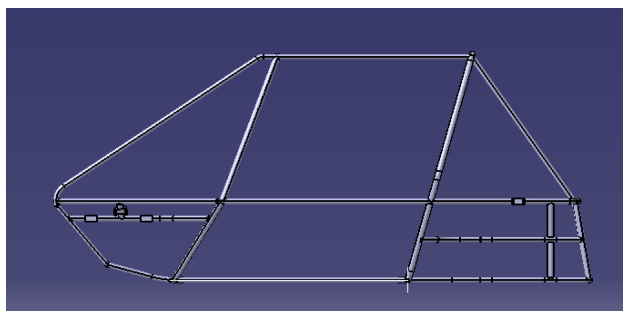


Fig.1.6: Side View of Roll Cage

2.1. Material Selection

The material used for the required roll cage was circular steel tubing with an outside diameter of 25 mm (1 inch), wall thickness of 3.05 mm (0.120 inch) and a carbon content of at least 0.18 (Baja SAE et al, 2014). The research was conducted to choose the best possible material. The choice of material was limited to steel as per SAE rules. The material was selected on the basis of cost, availability, performance and weight of material. After thorough research, two best materials were found for the designing of the roll cage i.e.: Steel AISI 4130 Chromoly alloy and Steel AISI 1018. The reasons for using round tubing (seamless) were it is lighter than square tube as smaller gauge sizes can be used to handle the same stress as a wider square tube and a round tube always out performs the square tube. Table 1.2 shows Mechanical properties of Steel AISI 1018 tube.

Table 1.2: Mechanical properties of Steel AISI 1018 Tube

Steel AISI 1018 Properties	
Density	0.284 lb/in ³
Ultimate Tensile Strength	63,800 psi
Yield Tensile Strength	53,700 psi
Modulus of Elasticity	29,700 ksi
Bulk Modulus	20,300 ksi
Shear Modulus	11,600 ksi
Poisson's Ratio	0.290
Elongation at Break (50 mm)	15 %
Hardness, Brinell	126

2.2. Design of Roll Cage

According to the constraint in the rulebook, the maximum speed of the vehicle is assumed to be 60 km/h or 16.66m/s. Calculations below were calculated in order to design the roll cage in best possible way.

Let W_{net} = Net work done, f = Force and d = Distance travelled

Now,

$$W_{net} = \frac{1}{2} mv^2_{final} - \frac{1}{2} mv^2_{initial} \tag{1}$$

$$W_{net} = - \frac{1}{2} mv^2_{initial} \tag{2}$$

$$\text{But, } W_{net} = \text{Impact force} \times d \tag{3}$$

It was considered that for static analysis, the vehicle comes at rest within 0.1 seconds after impact (Sania and Karan et al, 2013). Therefore, for a vehicle which moves at 16.66 m/s, the travel of the vehicle after impact is 1.66 m (Sania and Karan et al, 2013). From equations (1), (2) and (3)

$$\text{Impact force} = \frac{1}{2} mv^2_{initial} \times 1/d \tag{4}$$

$$\text{Impact force} = \frac{1}{2} \times 235 \times (16.66)^2 \times 1/1.66$$

$$\text{Impact force} = 19,632.852 \text{ N}$$

Therefore, Impact force by speed limit = 19,633 N

The Baja vehicle will have a maximum of 7.9 G's of force during impact, $G = \text{Mass of the vehicle} \times \text{Gravitational force acting on the vehicle}$ (Sania and Karan et al, 2013).

$$F = m \times a = 235 \times 7.9 \times 9.81 = 18,212.265 \text{ N}$$

$$\text{Impact force by acceleration limit} = 18,212 \text{ N}$$

The above calculated values are practically comparable.

To properly analyze the impact force, we needed to find the deceleration of the vehicle after impact. The forces which were impacted on the roll cage were decelerations and it was calculated as follows. According to research, a human body will pass out at forces much higher than 7.9 G's (Sania and Karan et al, 2013). Therefore, a value of 10 G's was considered for an extreme worst case collision (Sania and Karan et al, 2013). Therefore for static frontal impact analysis, the load on the vehicle is calculated.

$$F = m \times a \tag{5}$$

$$F = 235 \times 10 \times 9.81$$

$$F = 23,053.5 \text{ N}$$

Therefore, Impact force for worst case = 23,053 N

Thus, Steel used for roll cage of the vehicle has maximum yield point than the impact force for the impact force by speed limit, acceleration limit and worst case (where force value 10G's was considered). Initial design of the roll cage consisted of 4 main members: the roll hoop, horizontal hoop, and two perimeter hoops with the dimensions (Baja SAE et al, 2014). The Rear Roll Hoop (RRH) was first feature designed. It has been angled to the maximum tilt of 20 degrees from vertical to decrease the air resistance and maximize available space for the engine and transmission in the frame as low as possible (Baja SAE et al, 2014). Diagonal bracing members were added to RRH, no more than 5 inches from the end horizontal members. Working forward, the front end was designed according to suspension mounting points predetermined already. Members were drawn to accommodate the dual A-arms of the front suspension as well as a shock mounting point. Also in consideration, was leaving space for the brake reservoirs. The roll cage has been designed by simply connecting the roll cage to the highest point on the front end. Consideration was made for minimum head clearance for driver safety. The horizontal portion of the roll cage has been designed to maintain a large vertical clearance and forward clearance from the rear seat bottom. The vehicle's rear end has been designed with consideration for the engine size and orientation. Gearbox and rear suspension mounting points has been considered and decided. Frame has been analyzed to determine if this design was sufficient. Rollover and collision analysis has been performed. The design has been strengthened where necessary members were removed to save the weight. The firewall, front end,

and roll cage has been completed along with the design for the rear end. Every roll cage member has been designed with reducing weight in mind and overall chassis flex while cornering. Testing of the roll cage has been done in the event of rollover on front, side and top. The roll cage also needed a firm structure to aid it in protecting the driver in the event of a rollover or collision. To ensure that the frame could fulfill these specifications, the estimated forces that would be applied under several conditions (i.e. Front impact forces, side impact forces and roll over). A slow rollover would not be much more than weight of vehicle. The frame has to be designed to withstand these forces.

2.3. Finite Element Analysis (FEA)

As roll cage was designed by plotting key points, lines and splines, so every member of the roll cage was considered to be properly constrained at every joint. For boundary conditions for frontal impact test, the roll cage had to be fixed from the rear side and the front member will come across the applied load. In the similar way, for side impact test, one side of the roll cage elements was fixed while the other side would be applied with load. In order to determine the safety of our vehicle, deformation produced by applying different impact loads was checked (i.e. forces - 90.718 kgf on front, 45.359 kgf on side members and 90.718 kgf on top in case of roll over) with the help of analysis software. There are few important loading situations that should be analysed. These include frontal impact, side impact and roll over impact. Analysis from figures 1.7 to 1.9 shows the deformations produced by different impact loads. In frontal impact, there is a possibility of vehicle crashing into another vehicle head on during the race (Sania and Karan *et al*, 2013).

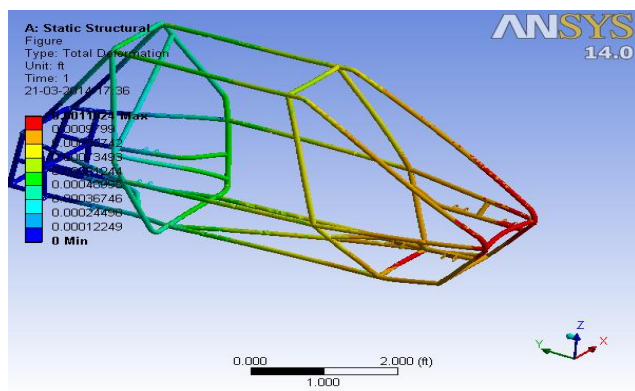


Fig. 1.7: Deformation due to Front Impact Load

The side impact analysis was carried out as there is a possibility of collision with another vehicle from either direction. Thus, the stresses acting on the side members of the roll cage are analyzed.

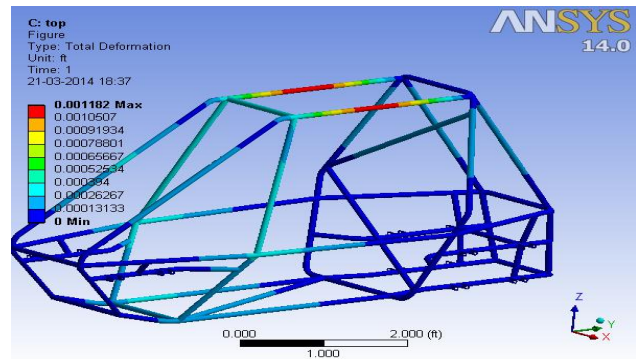


Fig. 1.8: Deformation due to Side Impact Load

The rollover impact analysis was carried out by considering the stresses induced on the members of the roll cage when the vehicle topples down from a slope with an angle of 45°. In this impact, the upper and rear members of the vehicle would bear the force.

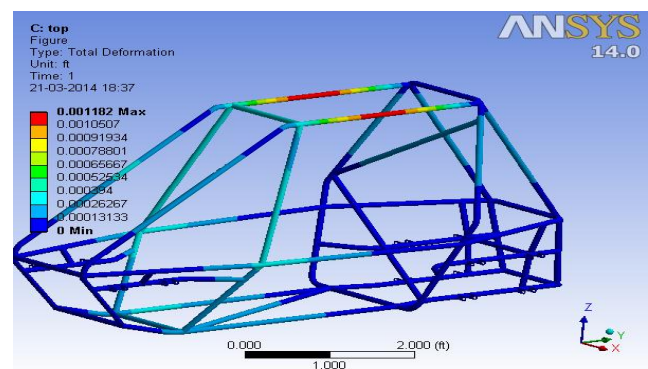


Fig. 1.9: Deformation due to Roll Over

3. Engine

Briggs & Stratton make (10 BHP and 19.659 Nm, 305 CC, single cylinder, four stroke, air cooled and OHV-intek) engine was used and it could not be modified in any way (Baja SAE *et al*, 2014). The engine had a governor that limits the power at high rpm to protect the engine. Governor was to be restricted at a 3,800 rpm or lower maximum speed (Baja SAE *et al*, 2014). The vehicle needed to be designed to maximize the torque available from this engine through the gearbox. Engine is mounted flatly in the lower rear part of the frame as it would help in reducing the overall centre of gravity in turn improve the handling of the vehicle, reduce the chances of rollover and the amount of mild steel plate needed for the gear frame. The plate on which engine is mounted is mild steel sheet with a length of 350.52 mm and width of 34.798 mm. The steel plate was welded to the middle frame rods and engine is then mounted on it with 4 bolts. The throttle pedal used to throttle the engine is of spring return with positive stroke type. The throttle pedal is connected to engine through throttle cable which is multi stand steel wire. Figure 1.10 and 1.11 shows the power curve and the torque curve for this engine.

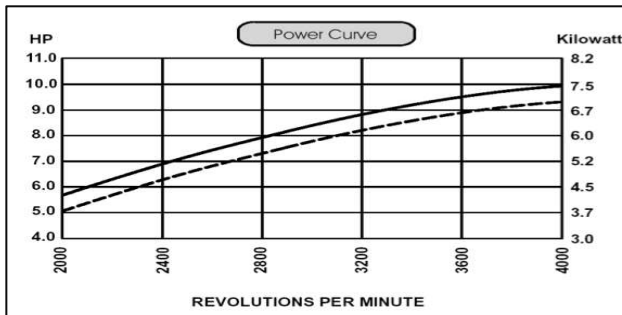


Fig.1.10: Power Curve of the Engine

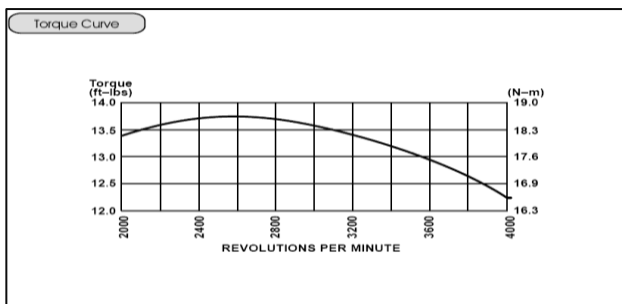


Fig.1.11: Torque Curve of the Engine

4. Powertrain System

A powertrain is an assembly of gears and associated parts by which power from the engine is transmitted to a driving axle, the tires, and finally the ground. It is essentially an arrangement that transmits power from the output shaft of the engine to the wheels. Many objectives were considered during the designing of the powertrain system. The primary objective was to maximize the torque delivered from the 10 BHP engine to the rear wheels for the vehicle speeds with a restraint of not exceeding engine governor speed of 3,800 rpm (Baja SAE et al, 2014). This objective was especially important when there was a restriction of any increment in 10 BHP horsepower, small and single cylinder engine. This restraint caused the design emphasis to be placed on the choice of transmission. The primary objectives considered were to reduce the weight of the powertrain system, durability, drivability and reliability. According to the constraint in the rulebook, modification of the engine was strictly prohibited but there was choice of selection of transmission. An “off-road enthusiast” would consider a manual transmission more “fun to drive” over any other transmission. Therefore, sequential manual transmission (4 speed with reverse gear mechanism, differential in-built) was considered because it has better acceleration from a standing start. This would give an off-road enthusiast more of a thrill of driving. This system would allow the driver to select the right gear from the available gears allowing more control over the vehicle. This gearbox is very easy to use for any driver as it has shifting pattern like 3 wheeler i.e. back and forth shifting movement instead of standard “H” pattern of an automobile.

4.1. Design of Powertrain

In order to keep a vehicle in motion, the engine has to develop sufficient power to overcome the opposing road-resistance, which is expressed as tractive resistance or tractive effort. In the process of selection of a suitable gearbox and according to the constraints of power requirement calculations was carried out to calculate traction at various gearing ratios i.e. different speeds of the engine. This traction was compared with the effort required to overcome various resistances offered to a vehicle in motion. The road resistance opposing the motion of the vehicle consists of three components (Chetan et al, 2013).

Roll Resistance (F_{roll}) is calculated as

$$F_{roll} = c \times W \times g \tag{6}$$

Where,

W = Weight of the vehicle = 235 kg, c = Rolling coefficient = 0.465, g = acceleration due to gravity = 9.81 m/s²

$$F_{roll} = 0.465 \times 235 \times 9.81, F_{roll} = 1071.987 \text{ N}$$

Air resistance (F_{air}) is calculated as

$$F_{air} = A/2 \times C_d \times D \times v^2 \tag{7}$$

Where,

A = Frontal area of the vehicle = $w \times h = 1.902 \text{ m}^2$

C_d = Coefficient of drag = 0.5

D = Density of air = 1.29 kg/m³

V = velocity of the vehicle (taken as 16.666 m/s)

$$F_{air} = 1.902/2 \times 0.5 \times 1.29 \times (16.66)^2$$

$$F_{air} = 170.251 \text{ N}$$

Gradient resistance ($F_{gradient}$) is calculated as

$$F_{gradient} = W \times \text{Sin } \theta \tag{8}$$

Where,

W = Weight of the vehicle

$\text{Sin } \theta$ = Inclination of road surface is taken as 45°

$$F_{gradient} = 235 \times \text{Sin } 45^\circ$$

$$F_{gradient} = 235 \times 0.707$$

$$F_{gradient} = 166.145 \text{ N}$$

Table 1.3 shows the forces at different gear ratios.

Table 1.3: Traction at different Gear Ratios

Gear Number	Gear Ratios	Velocity (km/h)	Roll Resistance (N)	Air Resistance (N)
1	01:55.1	13.07	64.7	8.01
2	01:32.7	23.04	66.71	14.13
3	01:20.0	37.79	71.72	23.18
4	01:13.4	55.5	80.94	34.04

A Sequential manual transmission (4 speed with reverse gear mechanism, differential in-built), (make Piaggio, India) is used for the vehicle. Gear box is

mated with rear drive axle through which the power transmitted to the wheels via the constant velocity joints (CV joints) (make Piaggio, India) which is Birfield joint type. Gear box is constant mesh and multi wet disc type. Shifting pattern is of 3 wheeler (i.e. back and forth movement). The main advantage of using this transmission was to get high torque and variation in speed. The clutch pedal used to operate the clutch in the vehicle is spring return type. Gear shifter is hand operated and is connected to gear lever through shifter cable which is dual wire push and pulls type. In this manual gearbox, the shift arm followed a linear cam so that as it could move fore and aft from center, it could also move inward (towards the driver). This inward movement would operate the clutch, disengaging the transmission from the engine, allowing for a shift to take place. When the driver releases the shifter, a spring holds it against the cam so that the clutch would re-engages the engine and transmission. To start moving from a stop, the driver would shift into first and ease the shifter forward (the shift pattern on this sequential manual gearbox is 1st is down, and, 2nd, 3rd, 4th are up) to slowly engage the clutch, while stepping on the gas. This design allowed the driver to feather the clutch if they have the need, by simply pushing the shifter towards them. The clutch linkage would be a cable, as it would need movement, the shift linkage would be a rod linkage as it would only move fore and aft. This manual gearbox helped in getting maximum torque from the engine at 3800 RPM (Maximum). The differential is inbuilt in gearbox and the type is Torque Limited Slip Differential (LSD). The gearbox is made out of steel and overall weights at roughly 12.156 kg +/- 0.453 kg (26.8 lbs +/- 1 lb). Gear ratios of gear box are given in table 1.3.

Table 1.3: Gearbox Ratios

Gear	Ratio
1st Gear	1:55.08
2st Gear	1:32.72
3st Gear	1:19.95
4st Gear	1:13.40
Reverse Gear	1:31.48

5. Suspensions

In order to resist the bumps and jerks that usually occur in an off-road track, an integrated approach of design is developed to obtain an optimized geometry of a system which contributes to the vehicle’s handling and braking for a better safety driving, to keep the driver as isolated as possible from bumps, vibrations and give the drivers a fun-to-drive experience. This geometrical system is termed as suspension system. The suspension system of an automobile is located between the rigid frame and the travel wheels. A Baja vehicle has its maximum running on off-road track which consists of all kinds of obstacles that could easily bind up the suspension of any other road vehicle.

Thus, to make Baja vehicle compatible to the off-road conditions, suspension system was designed in such a way that the vehicle can handle the roughest of bumps without affecting the vehicle’s stability. Objectives were considered during designing of the suspension system of the Baja vehicle. The primary objectives considered during designing were that the suspension system must provide maximum suspension travel as well as the comfort and greater control to the driver, to maximize the contact between the tire and the track surface, must be able to keep the vehicle stable and to reduce the vibrations of the vehicle over rough, unpredictable terrain and give the maximum ground clearance to the vehicle, must be light weighted which will increase the performance of the suspension system, must be neutral to slightly over steer due to the tight cornering in a maneuverability event and uniformly weight distribution among front and rear suspension system as it is used to locate the longitudinal center of gravity. The research was conducted to find out the best possible suspension system for the Baja vehicle. Different types of suspension systems were recognized, each with their own advantages and disadvantages. After thoroughly researching and ranking the different suspension systems characteristics, unequal length double wishbone coil-over front and rear suspension system was selected as double wishbone designs allows for the most control of ride behaviour, isolation of individual tire movement and provide excellent ride height and manoeuvrability for Baja vehicle. The best possible material used for suspension system components was steel AISI 1018 as it has all the four characteristics that were needed i.e. availability, cost of the material, material deformation with several impacts, and manufacturability. Table 1.4 shows the vehicle suspension specifications. Figure 1.12 shows the designed suspension system of the Baja vehicle.

Table 1.4: Vehicle Suspension Specifications

Vehicle Suspension Specifications	Values (at Ride Height)
Lateral Track Width (Front/Rear)	1524/1574.8 (mm)
Wheelbase	1625 mm
Ground Clearance	292.1 mm
Vehicle Weight	235 kg (518.086 lbs.)

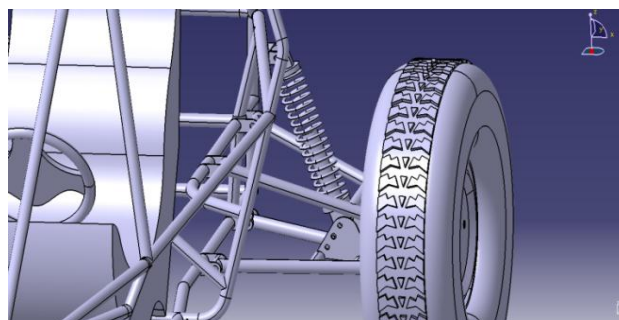


Fig.1.12: Suspension System

5.1. Design of Suspension

The suspension imparted to the vehicle was unequal length double wishbones coil-over front and rear suspension which provides maximum traction during cornering, stability in straight and to minimize the shock transferred to the roll cage and to provide enough ground clearance. Double-wishbone suspension consists of two wishbones shaped A-arms which have one attachment point on the knuckle and two on the frame. The suspension system was designed in such a way that the tires remain oriented properly in all modes of motion. This design took up a relatively large amount of space, but provides the most optimized wheel control, limiting tire scrub which could wear out tires quickly, and providing the maximum cornering grip. The purpose of such design was to reduce the cornering power of the front end relative to the rear during cornering over bumps. This was to ensure a safe and stable condition for the driver. Contradictory the suspension was also designed to over steer on smooth corners by employing a stiffer suspension at the rear, which will cause the rear to slide out during a turn. The front and rear suspension worked with a double A-arm system incorporating Nylon bushes. Both upper and lower A-arms of front and rear were not identical in length and thickness, so that the wheels vertical plane would maintained at all times during shock travel. Front and rear suspension systems were equipped with two fronts and two rear air shock absorber with 139.7 mm of travel. This setup gave 292.1 mm of total wheel travel, giving the car great ability to manage rocks, bumps and other obstacles while maintaining good traction. By using the steel AISI 1018 for suspension components, the weight of the suspension unit also called as un-sprung mass was reduced and stiffness of the mounting points on the sprung mass or the vehicle was increased which decreased the effects of inertia in the system allowing the system to react more rapidly to the bumps. This reduced un-sprung mass increased the suspension travel of the system, increased the control to the driver, increased the contact between the tire and track surface, keeps the vehicle more stable, reduced the vibrations of the vehicle over rough, unpredictable terrain and gives the maximum ground clearance to the vehicle. The suspensions has been designed in such a way that there are three choices for the selection between softer and harder suspensions which was quite during different races. Another important parameter was the camber and caster angle of the vehicle. The camber angle of Baja vehicle is negative as positive value is not stable and its value is 20 degrees as it improves grip during cornering and stability when landing rom a jump. Keeping the camber of the vehicle in the slightly negative range would ensure desirable handling characteristics in bump, and was examined more closely in roll to account for possible roll steer developed by this force. The roll camber coefficient graph given below is with a negative slope and it is close to one in bump travel while it increases with

droop travel. This helps in rolling motion as the weight of the vehicle is transferred to the outer wheel and the camber angle of the outer wheel remains close to the roll angle of the vehicle (Sania and Karan *et al*, 2013). This maximizes the contact patch the outer wheel supporting most of the weight of the vehicle while the angle of the inner wheel, which is unloaded rapidly, gains positive camber more than the roll angle thus further adding to the stability (Sania and Karan *et al*, 2013). Figure 1.13 shows the roll camber coefficient graph (Sania and Karan *et al*, 2013). Caster angle of the Baja vehicle is positive and its value is 10 degrees as it forces the front of the vehicle to be lifted slightly when steered, providing force feedback that centres the steering helping the driver to maintain a straight line. The most important parameter that affects camber was the design of the knuckle, specifically the kingpin angle.

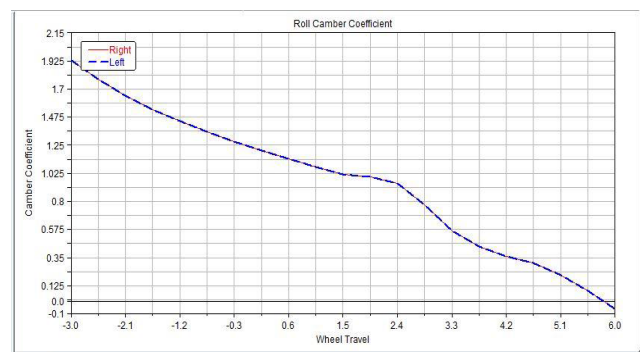


Fig.1.13: Roll Camber Coefficient Graph

As the kingpin angle increases, the effect of castor angle diminishes. Therefore, kingpin angle was angled to 2 degrees in order to maintain reasonable link and knuckle angles that would be able to handle the forces involved and be manufactured. An important parameter was the force center of gravity exerted on the vehicle. Center of gravity force is maximum at the rear end than the front end of the vehicle as weight distribution in Baja vehicle is 45/55 (%). Roll center of the rear end was kept high as compared to the front end so as to gain a balance between the lateral force at tires and the rolling moment. Thus, the roll center height of the rear end is 330.2 mm as compare to the 304.8 mm of the front end. This helps in inducing over steer and prevents heavy rolling which again helps in taking tighter turns by reducing the turning radius. The suspension geometry was setup in way such that the migration of the roll center was minimum at the rear thus inducing stability in the heavy rear part of the vehicle. Another important parameter is rake angle. Rake angle of the vehicle is 5 degrees as rotating the control arms counter clockwise in this plane allowed more of the horizontal component of the force from large bumps to be applied directly to the suspension travel. The entire hub assembly and square shaped knuckle joint were cast out of the Aluminium 6061-T6 blocks as they are very light in weight and drop some

weight in the suspension unit too. Hub is 3 cut flange type which is machined and welded to the driveshaft.

Figure 1.14 shows the designed wheel hub assembly of the suspension system. Scrub radius is an important parameter as positive scrub radius is very difficult to maintain specially in an all-terrain vehicle due to greater ground clearance and greater manoeuvrability of the vehicle. Positive scrub radius of the Baja vehicle is 127 mm as more the positive scrub radius, more the track width and stability of the vehicle. Track width is another important parameter as it affects a-arm length and stability. Maximum track width was limited to 1574.8 mm in order to be able to navigate certain obstacles (Baja SAE et al, 2014). The front upper a-arms are 419.1 mm each and front lower a-arms are 444.5 mm each. The rear upper a-arms are 444.5 mm each and rear lower a-arms are 469.9 mm each. The reasons for longer lower a-arms in both front and rear unit are increased suspension travel, vehicle stability and greater negative camber to get the optimum roll camber coefficient.

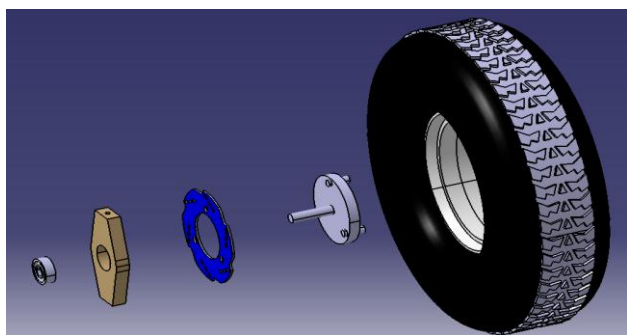


Fig. 1.14: Wheel Hub Assembly

Toe in with bump travel was incorporated in the geometry during sudden wheel movements as vehicle has a very high tendency to deviate from its direction just after sudden bumps (Sania and Karan et al, 2013). Figure 1.15 shows the result of the toe angle variation is plotted in graph, taking parameters with respect to wheel travel (Sania and Karan et al, 2013).

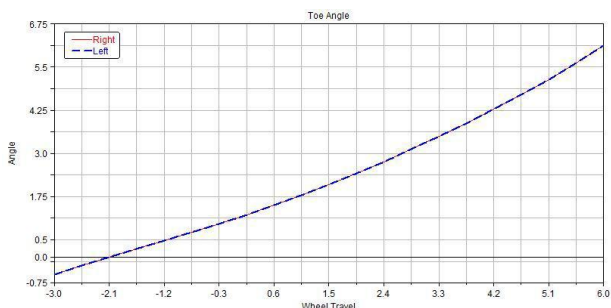


Fig.1.15: Toe Angle Variation with Wheel Travel

Figure 1.16 shows the roll camber coefficient variable with wheel travel (Sania and Karan et al, 2013).

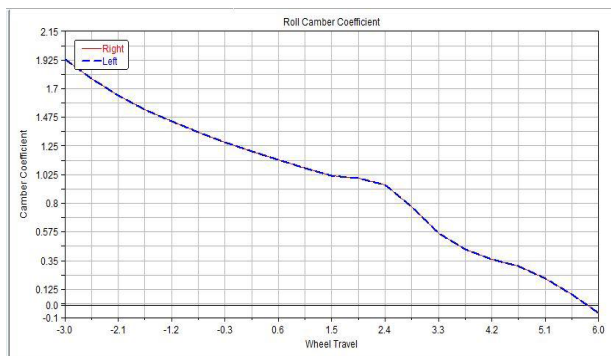


Fig.1.16: Roll Camber Coefficient Variable with Wheel Travel

Table 1.5 shows the suspension parameters of the Baja vehicle. Unlike a typical road car, Baja vehicle used all-terrain (ATV) tires which are rounder and has less contact area making camber less critical. Baja vehicle was designed to work with four tires having 25 inches outer diameter. This allowed the vehicle to reach a higher top speed by sacrificing some push force.

Table 1.5: Suspension Parameters of Baja Vehicle

Suspension Parameters	Values
Suspension Travel in Jounce	139.7 mm
Suspension Travel in Droop	63.5 mm
Front Roll Center Height	304.8 mm
Rear Roll Center Height	330.2 mm
Camber Angle	20°
Caster Angle	10°
Damper Travel	152.4 mm
Spring Rate	13.8
Spring Length (Front and Rear)	266.7 mm
Spring Wire Diameter (Front/Rear)	9 mm.
Number of Turns of Spring (Front/Rear)	18
Toe In (degrees)	0
Toe Change in Travel	Minimal
Weight Distribution Bias (Front/Rear)	45/55 %

The rims selected were made of steel grade, which gave enough strength to endure rough terrain. The width of rear rims was modified from original size (8 inch to 10 inch) with the help of steel sheet in accordance with the size of rear tires. Also, the thread pattern of the wheel gave the maximum traction on mud and loose terrain. Figures 1.17 and 1.18 are given in order to show the front and rear suspension systems in best possible way.

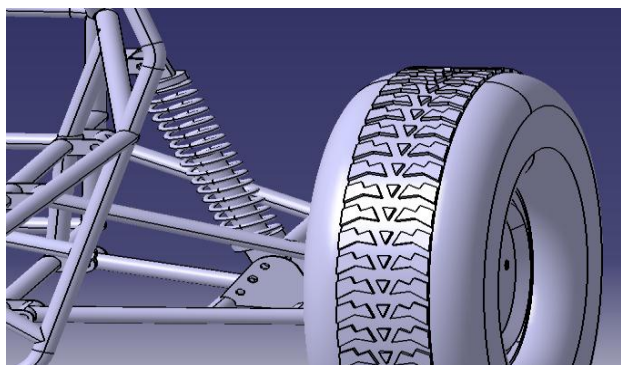


Fig. 1.17: Front Suspension System

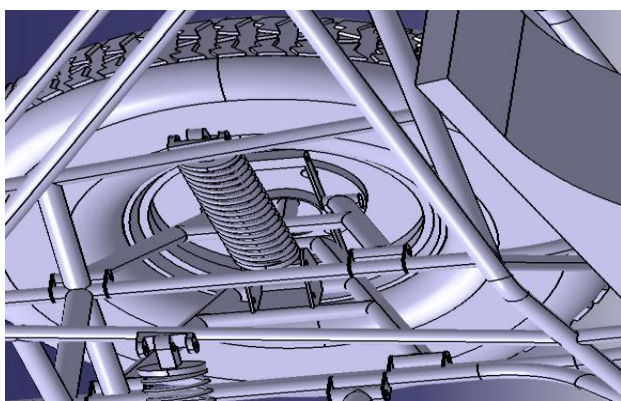


Fig. 1.18: Rear Suspension System

The tires selected were Maxxis (front) and Carlisle (rear). The size of rim and tires are given in table 1.3.

Table 1.3: Tire and Rim Size

Component	Front	Rear
Rims Size	12x8 in.	12x10 in.
Tires Size	25x8x12 in.	25x10x12 in.

6. Steering System

A linear and predictable steering system is vital to an all-terrain vehicle. A steering vehicle is responsible for the control of the lateral motion when vehicle is in longitudinal motion. A good steering system signs that the driver must be able to maintain control of the vehicle at all times. The principle that governed steering was fairly straightforward. The main factors on which steering system depends were - speed of response and driver’s input. There are many steering system each with their advantages and disadvantages. Mechanical steering mechanism with rack and pinion was chosen for Baja vehicle as this mechanism isn’t sloppy at the center point and gives the drive a large range of motion, provides a large degree of feedback and allows the driver to feel the ground, it places the pivot points of the steering system near the pivot points of the suspension system which greatly reducing bump steer and this unit is very compact and fits more easily into the front part of the frame. The steering

tube rods (or Tie rods) used for the controlling of the wheels were of steel AISI 1018 with the length of 13.5 in. each, inner and outer diameter of 11 mm and 15 mm respectively. Many objectives were considered during the designing of the steering mechanism.

The primary objective considered were that to control lateral motion while the vehicle is in longitudinal motion, to provide Ackerman geometry and to limit the steering wheel rotation from lock to lock, to obtain correct steering angles for each wheel and to achieve slight negative camber in the direction of the turn, to achieve a small turning radius and moreover, steering stability. Another important parameter considered was the distance between the kingpin axis and the mounting hole for the tie rod. This distance along with the ratio of the rack and pinion, dictated the overall steering ratio. It was required to have a harness holding the drivers wrists to the steering wheel (Baja SAE et al, 2014). This made it difficult for the driver to take his hands off the wheel to complete tight turning manoeuvres. To alleviate this problem, increase the steering ratio so that the car could be turned from lock to lock with 360 degrees or less of steering wheel rotation. While designing the steering system constraints that possessed were center alignment of steering system, track width, human effort at the steering wheel and the desired response of the steering system.

6.1. Design of Steering

Following calculations were assumed with the use of Tie rods (Singiresu S. et al, 2004).

$$\text{Tube rod moment of inertia, } I = \pi/64 (D^4 - d^4) \tag{9}$$

$$I = 3.141/64 (15^4 - 11^4)$$

$$I = 1766.027 \text{ mm}^4$$

$$\text{Tube rod cross-sectional area, } A = \pi/4 (D^2 - d^2) \tag{10}$$

$$A = 3.141/4 (15^2 - 11^2)$$

$$A = 81.66 \text{ mm}^2$$

$$\text{Solid rod moment, } I = \pi D^4/64 \tag{11}$$

$$I = 3.141 \times 15^4/64$$

$$I = 2484.580 \text{ mm}^4$$

$$\text{Solid rod cross-sectional area, } A = \pi D^2/4 \tag{12}$$

$$A = 3.141 \times 15^2/4$$

$$A = 39753.281 \text{ mm}^2$$

Maximum bending moment allowed by rod,

$$M = \sigma I/c \text{ ---- (13)}$$

$$M = 63,800 \times 2484.580/5.99$$

$$M = 26463.473 \text{ kg-mm}$$

Hence from equations (9–13) (Singiresu S. *et al*, 2004), maximum bending moment allowed by the rod is less as compared to yield tensile strength of rod.

The steering system designed was mechanical type which worked with rack and pinion (make Maruti-Suzuki, India). The rack and pinion rested behind the pedals in order to provide the maximum space needed in cockpit. The rack travelled two and a half turns from lock to lock which allowed effective control of the vehicle and better responding speed of the steering wheel. Rack is not constrained to translate in horizontal direction. It moves in slightly downward direction when the bush deflects. A Bush is made up of approximately 25 discrete elements as a flexible body as it helps to predict the rack motion and it has a cylindrical joint with rack. Pinion is mounted on bearings with a fixed axis of rotation. A Plunger has a prismatic joint with Yoke nut, but kept apart by spring preload of 29.999 kgf. This spring preload helps to maintain contact between the rack and pinion continuously and also provides damping.

The rack and pinion steering consists of a gear that is driven by the steering column and a gear rack that will mesh with the steering column gear. The rack is then connected to the tie rods that are connected to the hubs in a way where if they are pulled or pushed by the tie rods, the wheels will turn in the direction driven by the steering wheel. The types of rack and pinion steering are the spur gear type and the helical gear type. Rack is designed with teeth inclined at 15 degrees. The rack gain is 34.150 mm per revolution of the pinion. This has led to use of non-standard pinion in the gearbox with 6 numbers of teeth with 20 degrees pressure angle. Total depth of pinion is 3.213 mm which is divided into 2.938 mm addendum and 0.275 mm dedendum. The pinion do not have smooth flank surface due to series of Boolean cut operation. So the friction between the rack and bush needs to be incorporated in form of some function at the interface of two entities. Turning radius was one of the important parameters of an all-terrain vehicle. Lower the turning radius, lesser are the efforts for making tighter turns and in turn is more manoeuvrable in tight driving conditions. The turning radius of the Baja vehicle was reduced to 4.5 meters by maximized the angle that the wheels were able to turn left or right. Ackerman geometry was used for the proper steering mechanism. In Ackerman geometry, the centers of rotation of each of the front wheels must intersect the axis of the rear axle at the same point. An issue in Ackerman geometry was Ackerman angle in which the inner and outer wheels actually turn in a different radius. This variation in turning angle from one tire to another tire was accomplished by correct placement of the outer steering pivot points. The simplest way to account for Ackerman angle was to draw a construction line from the centre of the rear axle through the steering axis and extended it forward a couple inches. To have proper Ackerman steering angle, the outer tie rod pivots were laid on this line.

One important parameter was that the length of the tie rod should be in between length of the upper and the lower arm in order for the stable driving of the vehicle. Figure 1.19 shows Ackerman steering mechanism.

Another important parameter considers was the distance between the kingpin axis and the mounting hole for the tie rod. This distance along with the ratio of the rack and pinion dictated the overall steering ratio. The centres of rotation of each of the front wheels must intersect with the axis of the rear axle at the same point. To alleviate this, two steering arms intersects at the centre of the rear axle i.e. line drawn through the kingpin axis and the tie rod mounting hole intersects with the rear. Another issue that came in the designing of Baja vehicle was when large amounts of suspension travel involved was a change in the distance between the steering mechanism and the tie rod mount on the knuckle. During compression this distance actually becomes shorter thereby pushing the knuckle out and causing toe in. This could also cause wheel scrub and instability. To alienate this, the mounting point on the rack and pinion remained the centre of rotation of the knuckle during as much of the suspension travel as possible.

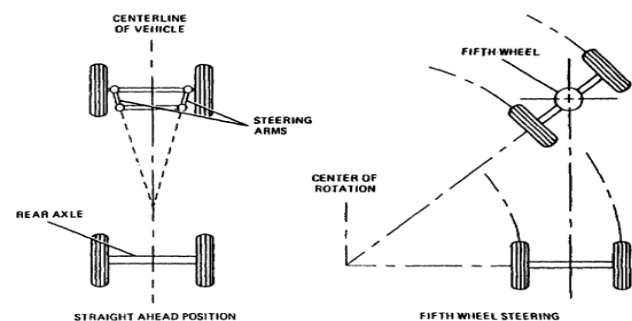


Fig.1.19: Ackerman Steering Mechanism

7. Braking System

The objective of the braking system is to provide a reliable and prompt deceleration for the vehicle or in simpler words, the purpose of the braking system is to slow down the speed of vehicle or to stop it. When a brake is applied to a vehicle, its kinetic energy is converted into heat energy and thus the motion of the vehicle stops. Important objectives were considered during the designing of braking system. The primary objectives were that it should be capable of stopping the Baja vehicle with a weight of 235 kg (518.086 lbs) running with a speed of 60 km/h or 16.66 m/s in 14.996m both in a static as well as dynamic conditions on a paved and unpaved surfaces (Baja SAE *et al*, 2014), pedal must directly actuate the master cylinder through a rigid link (Baja SAE *et al*, 2014), must be light in weight so that it should prevent air bubbles formation within the brake lines, brakes on the driven axle must operate through the final drive (Baja SAE *et al*, 2014), driver must have complete control of the vehicle while the brakes are activated, brakes should

be capable of locking up all four wheels while on the pavement (Baja SAE *et al*, 2014) and must have good ergonomics as every driver get suited to paddle assembly fitment. Braking system contained two independent hydraulic systems, each with their own fluid reserves (Baja SAE *et al*, 2014).

7.1. Design of Braking

The braking system was selected based on the braking torque requirements obtained from dynamic load on driveshaft. The changes in axle load during braking depend upon the static laden conditions and deceleration (R.S. Khurmi *et al*, 2011).

Front axle dynamic load is given as (W_f)

$$\begin{aligned} W_f &= W_1 + (\alpha/g) \times W \times (H/L) \\ W_f &= 105.75 + (6.37/9.81) \times 235 \times (11.5/64) \\ W_f &= 133.169 \text{ N} \end{aligned} \quad (14)$$

Rear axle dynamic load is given as (W_r)

$$\begin{aligned} W_r &= W_2 - (\alpha/g) \times W \times (H/L) \\ W_r &= 129.25 - (6.37/9.81) \times 235 \times (11.5/64) \\ W_r &= 156.669 \text{ N} \end{aligned} \quad (15)$$

Where,

W_1 = Static load on the front wheel (each) = 105.75 kg

W_2 = Static load on the rear wheel (each) = 129.25 kg

α = retardation of the vehicle

W = Weight of the vehicle = 235 kg

H = Distance of C.G from ground = 536.829 mm (21.13 in.)

L = Wheel Base = 1625.6 mm (64 in.)

Retardation of vehicle was obtained from the minimum stopping distance of vehicle at maximum speed travelling on a flat road. And the minimum stopping distance of the vehicle was calculated using the work energy principle. Retardation of the vehicle was found out to be $\alpha = 6.37 \text{ m/s}^2$. Thus, the value of dynamic load on front and rear axle was found to be 133.169 kg and 160.012 kg respectively.

The braking torque distribution available at the front and rear discs was obtained by using the following formulas.

Torque at each front disc is given as (T_f)

$$\begin{aligned} T_f &= W_f \times (f/g) \times R_f \\ T_f &= 136.169 \times (3.96/9.81) \times 12.5 \\ T_f &= 671.953 \text{ Nm} \end{aligned} \quad (16)$$

Torque at each rear disc is given as (T_r)

$$\begin{aligned} T_r &= W_r \times (f/g) \times R_r \\ T_r &= 156.669 \times (3.96/9.81) \times 12.5 \\ T_r &= 736.631 \text{ Nm} \end{aligned} \quad (17)$$

Where,

R_f = Radius of front tires = 12.5"

R_r = Radius of rear tires = 12.5"

f = Deceleration due to braking = 14.256 km/h (3.96 m/s²)

x = Distance of C.G from rear axle = 609.6 mm (24 in.)

Hydraulic disc brakes were used on all the four wheels strictly according to the Baja SAE International rules 2014. The hydraulic disc brakes [make TVS, India, Stainless steel with diameter 200 mm (7.87 in.)] were used as discs are directly in contact with cooling air, thus no separate cooling system for brakes, brake pads undergo uniform wear; this is beneficial since ATV requires frequent use of brakes, no loss of braking efficiency due to expansion/contraction of disc, total weight of disc-brake arrangement is less than other brake arrangements and applying brake is very easy as fluid exerts equal pressure everywhere; hence the required biasing of the brakes can be achieved at front and rear wheels easily. Also the system is simple in construction and due to absence of joints compared to mechanical brakes has less wear rate. The brake pedal used to apply the brake is of spring return foot lever type.

In the present braking system two separate circuits (Baja SAE *et al*, 2014), two rear master cylinders, a single pedal, four discs have been fitted. Master cylinders are very small with reservoirs mounted directly to the top which is advantageous in terms of weight and ergonomics. Two different brake calipers have been used in front wheels and rear wheels. Lines were run from both master cylinders to the front and rear calipers. For the front braking system, the brake line was split directly at the master cylinder and then ran to each side of the vehicle. The rear brake line run underneath the seat, up to the area of the final drive shaft, and then split at that location to both the left and right sides to minimize the length of cable needed. These calipers were connected to the master cylinder. The rear braking system was designed to be similar to the front setup, with two separate disc brakes, one for each wheel. The challenge was to design a rotor and caliper mount that would be installed with minimum modification to the rear knuckles, use the same rotor and caliper as the front wheels, not interfere with any other systems, be simple to design, and be as light as possible. There was enough room inside the rear knuckle to fit the rotor and caliper. A 3 cut flange hub was machined and welded to the driveshaft before the knuckle to allow for the rotors to be mounted. The rotor was mounted to the hub that was welded to the driveshaft and the caliper was mounted to a small plate. As there are two different braking circuits for front and rear which leads to the need of different pressures, a bias bar was installed between the brake pedal and the two cylinders used to adjust the amount of pressure being created in each circuit. The details of front and rear calipers and master cylinder are given as below.

7.2.1. Front Calipers

In front wheels, brakes calipers (make TVS, India) had been fitted. Calipers used were dual pistons 26.924 mm (1.06 in.) which were bigger and more efficient than rear brake calipers.

7.2.2. Rear Calipers

In rear wheels, brake calipers (make Royal Enfield, India) used were dual pistons 24.892 mm (0.98 in.).

7.2.3. Master Cylinder

A master cylinder is the part of the braking system that turns the mechanical energy from the brake pedal into hydraulic pressure will be sent to the brake calipers to stop the vehicle. Two rear 16mm (0.62 in.) Master Cylinders (make TVS, India) had been fitted in front and rear braking system. The master cylinders were mounted in parallel such that both the master cylinders were connected to a same linkage which connected them to a brake pedal and actuates braking in all the four wheels when foot pedal is pressed.

8. Ergonomics and Safety

Ergonomics is the science of design intended to maximize productivity and safety by reducing driver fatigue and discomfort. It was ensured that all of the components of the car will function properly when designed and assembled together. Ergonomics also includes the belly pan structure running over the entire length of the cockpit, foam padding of the roll cage, gear shifting indicators and such other things. Driver's safety was the most important concern for an all-terrain vehicle. For the comfort and safety of the driver in the rugged, up and down track the vehicle had been provided with 6 point harness seat belt system along with neck restraint and arm restraint. A pivoted bumper with spring support in the front of the vehicle was also installed on the front of the off road vehicle to absorb energy from collision. Fire extinguisher and kill switches were also used to meet the emergencies. SAE grade brake lights and reverse alarm were installed in the Baja vehicle with proper insulations. A transponder was also mounted on the vehicle in order to calculate the lap times. All electrical components were powered by a completely sealed 8 mAH DC dry cell battery that could not leak in the event of a roll over.

Conclusion

The work was divided into five subgroups in order to design all the main aspects of the vehicle. The subgroups were: roll cage, powertrain, suspension, braking and steering. The design work for each of the subsystems of the vehicle were made by the selection of components while keeping in mind the safety, performance, weight, reliability and last of all, cost. The primary goal was to build the lightest frame, best possible powertrain and best suspensions to maximize performance of the vehicle. The Dessault Systems Catia V5 R21 software was used for designing and it was analyzed on Ansys - 14. The performance, needs, serviceability, affordability and reliability were kept in mind while designing this Baja vehicle. The process of designing the Baja vehicle was not a simple task; as a matter of fact it took a lot of effort to make a best possible and final product.

References

- Baja SAE International Rules 2014, Society of Automotive Engineers (SAE) International.
- Oturkar Sania, Gujarathi Karan (2013), An Introduction to Computational Frontal Static Stress Analysis of a Baja Car, International Journal of Scientific and Research Publications, Vol. 3, pp. 1-5.
- Wadile Chetan, Dubal Rohal, Kohle Roshan, Rangaswamy Versha, Siddiqui Aqleem & Gaurav Nitin (2013), Selection, Modification and Analysis of Power Transmission and Braking System of an ATV, International Journal of Mechanical Engineering and Robotics, Vol. 1, pp. 97-102.
- Rao Singiresu S. (2004), Mechanical Vibration, Pearson Education Press.
- Khurmi R.S., Gupta J.K. (2011), Textbook of Theory of Machines, Eurasia Publishing House Pvt. Ltd.
- Pal Arindam, Sharma Sumit, Jain Abhinav, Naiju C.D. (2013), Optimized Suspension Design of an Off-Road Vehicle, The International Journal Of Engineering and Science (IJES), Vol. 2, pp. 57-62.
- Sharma Vikas, Purohit Divyanshu (2012), Simulation of an off-road vehicle roll cage a static analysis, International Journal of Engineering Research and Applications (IJERA), Vol. 2, pp. 126-128.