

Design, Analysis, Simulation and Validation of Suspension System for an Electric All-Terrain Vehicle (ATV)

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ABSTRACT: Suspension is the most vital sub-system in an automobile. Its main functions are load transfer to the wheels and protection of the driver from road shocks. The purpose of this paper is to select suitable suspension system for the front and the rear of an All-Terrain Vehicle (ATV) with rear electric drive and to thereafter design, analyze, simulate and test the suspension systems for optimum performance of the vehicle, driver safety and maximum driver comfort. The camber and caster angles, toe and the Ackermann variations were given due consideration. The stability of the vehicle was given importance and the system was designed to be durable enough to withstand shocks from the harsh terrain where ATVs are generally used. The springs were designed by calculations and the components were designed using CATIA. The components were analyzed using NASTRAN/PATRAN commercial FEA software and the front and rear systems were simulated using Lotus software. The system was then fabricated and its performance was duly tested.

KEYWORDS: Suspension; ATV; Springs; Simulation; Catia; Nastran

I. INTRODUCTION

Suspension is of extreme importance in any road vehicle. It consists mainly of the spring and damper that perform the function of shock absorption and the linkages that connect the vehicle and the shock absorbers to the wheels. When the wheel of a vehicle goes over an obstruction, the movement is termed as a 'bump' and the suspension system, through the motion of the linkages and the spring-damper arrangement absorbs the forces created by this motion. The force depends on the unsprung mass at each wheel; greater sprung to unsprung mass ratio implies the occupants will be less affected by road imperfections. [1] The main role of suspension system is to support weight of the vehicle and to provide comfort to the passenger.

The methodology of this paper is as below:

- Selection of suitable suspension system for all terrain purpose
- Finalizing the arrangement/geometry of the suspension based on the steering requirements, stability etc.
- Locating the chassis mounting points
- Designing the spring based on load considerations of the vehicle
- Designing and modelling the other components based on geometry previously selected
- Analysing these components and reiterating if required
- Simulating the suspension systems for 3-D compatibility
- Manufacturing the components, assembling the vehicle
- Making assembly modifications based on simulation results
- Testing of the vehicle performance

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II. RELATED WORK

The Double Wishbone system is one of the most commonly used front suspension systems in ATVs, thanks to its flexibility and availability of components. Previous work done on Double Wishbone systems have concentrated on design and analysis of spring and wishbone, with simulation of camber variation with roll angle; the camber gain was shown to be achieved by unequal wishbones. [2] Further work on optimizing the double wishbone system with unequal arms demonstrated the camber as well as toe angle variation for the front wishbone suspension. At the rear, the MacPherson strut was used with a toe link for optimum performance and decreased unsprung mass. [3]

Although the MacPherson strut has its advantages, the Trailing Arm system is one of the most preferred rear suspension systems due to its functional design and sturdiness. It undergoes high bending and torsional stresses. Its advantage lies in the fact that there are no camber or toe-in changes, along with a constant track width. [4] The semi-trailing arm has also been considered for the rear suspension system. They have provisions for the adjustment of toe and camber angle for the rear wheels [5], and can be used to induce oversteer. A variation on this, in the form of a trailing arm with camber links has also been designed and analysis of the lateral, vertical and longitudinal forces acting on the components have shown this system to be a viable option. A degree of adjustment of the camber has been shown to be possible. [6] Some vehicles have also been run with wishbones at the rear. This gives good ride quality and toeing of the wheels has been eliminated; this has been confirmed by kinematic analysis. [7] An alternate approach to suspension kinematics is through control systems; second order equations have been used to present the suspension in control system form. The motion can then be observed by applying 'disturbances' to the equation in passive or active systems. [8]

III. DIFFERENT TYPES OF SUSPENSION SYSTEMS

Different types of vehicle applications require different types of suspensions. For eg., early stage coaches of very high weight required a primitive form of suspension, with layers of wrought iron beams providing the core of the system, and the two wheels at either the rear or the front dependent on each other [1]. However, modern commuter vehicles require independent motion of each wheel, as comfort is the most important prerequisite in passenger cars. Hence, suspension systems can be broadly classified into:

A. Dependent Suspension Systems

This type of suspension acts as a rigid structure in which any movement of one wheel is transmitted to the other wheel as well, in the rear or the front. The force is also transmitted from one wheel to the other. It is mainly used in heavy vehicles. Examples are Leaf Spring Suspension, Watt's Linkage etc. [9]

B. Independent Suspension Systems

This type of suspension allows each wheel to move vertically without affecting the other wheel, in the rear or the front. This type of suspension is used mainly in passenger cars and light trucks. They have better resistance to steering vibrations and also provide greater space for the engine. Examples are MacPherson Strut, Trailing Arm and Double Wishbone etc. [9]

IV. SELECTION OF SUSPENSION SYSTEM

For the front wheels, Double Wishbone suspension was chosen. This contains two lateral control arms or 'A-arms' which are of unequal length. This type of suspension is well suited to large travel requirements of ATVs. It can also be configured to provide a camber geometry which can be varied as required. Design of the geometry of Double Wishbone suspension system along with design of the spring and the steering system plays an important role in maintaining the stability of the vehicle. [1] This type of suspension system provides increasing negative camber gain all the way to full bump travel unlike the MacPherson strut. They also enable easy adjustment of wheel parameters such as camber. Double Wishbone suspension system has superior dynamic characteristics as well as load handling capabilities. [9]

For the rear wheels, the main factors applicable are weight, cost and functionality. [5] The Trailing Arm suspension system, with attached hub and control arms was chosen based on these factors. This configuration provides the vehicle

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with a good balance between ground clearance, travel and adjustability. It has a simple and economical design. The wheels will remain parallel to the body with no camber or body roll. The control arms (trailing arms) absorb the longitudinal forces and braking moments, and control squat and lift. [9]

For the spring selection, the various springs in the market were explored including Air Shocks. However, due to a multitude of factors such as affordability, simplicity and availability, helical springs were chosen and the spring design has been based on them.

V. DESIGN

The design procedure for the chosen suspension system is as follows:

- Design and development of the components of the suspension
- Considering dynamic factor and hence modifying the design parameters
- Static testing
- Dynamic testing
- Modifying the component design based on Dynamic Testing results

The following components were designed:

- Knuckle
- A-arms or wishbones
- Spring

The basic consideration for the design of the various suspension components is the overall geometry of the suspension, based on parameters given by SAE [10].

A. DESIGN OF THE SUSPENSION SPRING

The various design considerations made for the vehicle suspension are as follows:

Sprung Mass= 400 kg

Mass distribution (Front : Rear)= 40:60

Mass per Front Wheel= 80 kg

Mass per Rear Wheel= 120 kg

Static to Dynamic amplification Factor = 2.5

i. Design of the Front Spring

Fig.1 is a representation of the front suspension geometry and Fig.2 is a schematic of the forces acting on the wishbone with their points of action.

Length of the wishbone = 12" (30.48 cm) (from suspension geometry)

Angle at which the spring-damper arrangement is inclined (to the horizontal) = 60°

Mounting point of the spring on the lower wishbone = 10" (25.4 cm) from the chassis (from suspension geometry)

The reaction force from the ground when wheel goes over a bump

= (Mass per wheel * 9.81) N

= (80 kg * 9.81) N = 784.8 N

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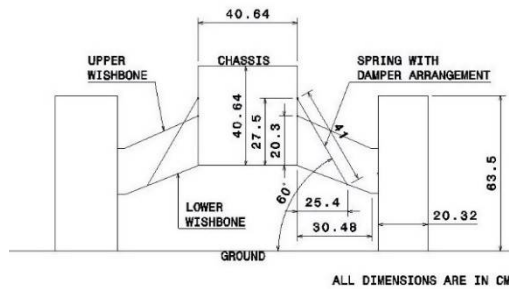


Fig.1: Front suspension geometry

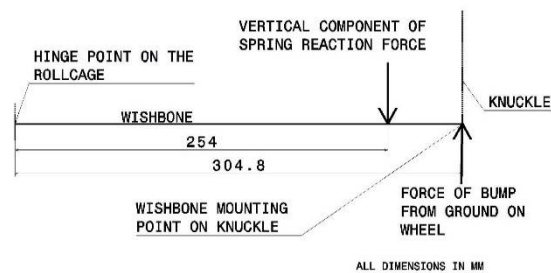


Fig.2: Forces on front wishbone

A force slightly greater than this will cause the wheel to move upwards. For the purpose of further calculation, this is the value of force that is considered. Also, as the angle of inclination of the spring-damper arrangement to the horizontal is close to 90°, the spring force can be approximated to this particular condition. Taking moment about the wishbone hinge point on the chassis,

$$784.8 * 12 = \text{Spring Force} * 10$$

Therefore, Spring Force = 941.76 N

Applying the dynamic amp. Factor,

Dynamic force acting on the spring = 2354.4 N

An all-terrain vehicle must have a significant amount of wheel travel for optimum performance and driver comfort. [4]

A minimum of 4" of spring deflection is an ideal condition for ATV front wheel travel. Thus, the required Spring Stiffness =

$$\frac{\text{Dynamic Spring Force}}{\text{Spring travel}} = \frac{2354.4}{101.6} = 23.1732 \text{ N/mm}$$

Deriving the number of coils in the spring from the force, using the formula:

$$k = \frac{Gd^4}{8nD^3}$$

and making the following considerations based on availability and ease of manufacture:

n= Number of active coils in the spring

k= Stiffness of the spring (Calculated separately for front and rear)

G= Modulus of Rigidity of the spring material= $79.3 \times 10^3 \text{ MPa}$

d= Diameter of the spring wire=10.5 mm

D= Mean diameter of the spring coil= 69.5 mm

D'= Outer spring coil diameter= 80mm

“Active coils” are the helically wound wires that contain pitch, initial tension to exert energy when a spring is deflected or extended.

$$\text{Substituting, } 23.1732 = \frac{79.3 \times 10^3 \times 10.5^4}{8n \times 69.5^3}$$

$$n = 15.488 = \mathbf{16 \text{ coils}}$$

ii. Design of the Rear Spring

Fig.3 shown is a schematic diagram of the forces acting on the trailing arm with their points of action.

The data for the trailing arm is as below:

Length of the trailing arm = 20" (50.8 cm) (from suspension geometry)

Angle at which the spring-damper arrangement is inclined (to the horizontal) = 70°

Mounting point of the spring on the trailing arm = 14" (35.56 cm) from the end of the chassis (by length limitation of spring-damper arrangement)

The reaction force from the ground when wheel goes over a bump

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$$= (\text{Mass per wheel} * 9.81) \text{ N}$$

$$= (120 \text{ kg} * 9.81) \text{ N} = 1177.2 \text{ N}$$

As with the front wheel, a force slightly greater than this will cause the wheel to move upwards. For the purpose of further calculation, this is the value of force that is considered. Also, as the angle of inclination of the spring-damper arrangement to the horizontal is close to 90°, the spring force can be approximated to this particular condition.

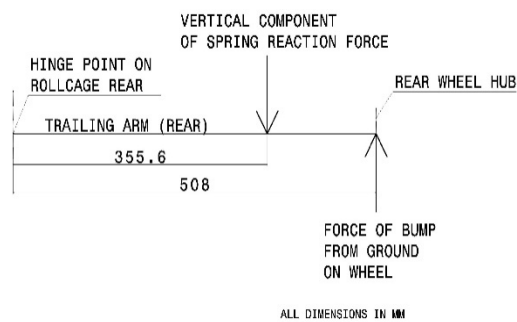


Fig.3: Forces on rear trailing arm

Taking moment about the trailing arm hinge point on the rear end of the chassis,

$$1177.2 * 20 = \text{Spring Force} * 14$$

Therefore, Spring Force = 1681.714 N

Applying the dynamic amp. Factor,

$$\text{Dynamic force acting on the spring} = 4204.285 \text{ N}$$

Similar to the front wheel, travel requirements for ATVs are taken into consideration and a 4” spring deflection is used as the basis for calculating the spring stiffness.

$$\frac{\text{Dynamic Spring Force}}{\text{Spring Deflection}} = \frac{4204.285}{101.6} = 42.35 \text{ N/mm}$$

Deriving the number of coils in the spring from the force using the formula:

$$k = \frac{Gd^4}{8nD^3}$$

$$\text{Making the same considerations as previously made for the front spring and substituting, } 42.35 = \frac{79.3 \times 10^3 \times 10.5^4}{8n \times 69.5^3}$$

$$n = 8.4748 = \mathbf{9 \text{ coils}}$$

Thus, according to the calculations, the number of active coils in the front and rear springs are 16 and 9 respectively.

The complete specifications and dimensions of the spring are as below:

Sl. No.	Parameter	Front Spring	Rear Spring
1	Diameter of wire	10.5 mm	10.5 mm
2	Outer coil diameter	80 mm	80 mm
3	No. of turns	16	9
4	Free length of spring	410 mm	260 mm
5	Pitch of spring	20 mm	25 mm
6	Eye-to-eye length of spring-damper (unloaded)	503 mm	350 mm
7	Stiffness of spring	23.17 N/mm	42.35 N/mm
8	Maximum travel	101.12 mm	100.88 m

Table 1: Spring Parameters

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B. WISHBONE DESIGN (FRONT)

The design parameters that govern the dimensions of the wishbone are given below. These are based on the chassis dimensions as well as track width restrictions as given by SAE BAJA [10]. Based on the chassis dimensions, the available length for wishbones was 12 inches and available width was 10 inches. The front wishbones were designed to be A-arms with a rounded end for the ease of manufacture as well as to prevent fouling with the shocks as shown in Fig.4. The dimensions were finalized based on the regulations given by the SAE. The upper and lower wishbones were designed to be of unequal length, with the upper being longer.

The advantage of having different lengths is that when the car takes a turn a negative camber is induced which increases the stability. [7] The spring-damper arrangement is mounted on the lower wishbone and the wishbone is attached to the knuckle through a ball joint. The positive static camber provided by the wishbone arrangements provides a reduction of the scrub radius of the wheel-steering knuckle combination. This reduces the steering effort required and the rate of tyre wear [9]. All of these increase the comfort of the driver which is one of the most important functions of a suspension system.

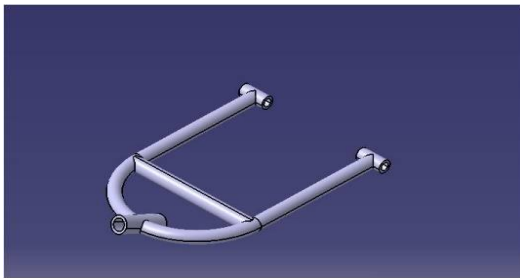


Fig.4: Model of Front Wishbone (Lower)

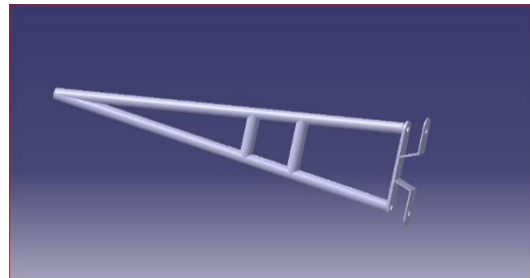


Fig.5: Model of Rear Trailing arm

C. TRAILING ARM DESIGN (REAR)

When the vehicle takes a turn, due to the horizontal forces acting on the attachments of the wishbones and the knuckle and due to lack of steering on the rear wheels, there may be toe-in or toe-out of the wheels. This may lead to improper steering and may result in unbalance. Excess toe-in or toe-out may cause oversteer or understeer which leads to loss of control. To avoid any toeing, the trailing arms were selected for the rear. This prevents formation of an axis and hence eliminates the possibility of toeing of the wheels, ensuring proper alignment. [6] Fig.5 shows the model of the trailing arm. Fig.6 gives the full assembly of the rear suspension with the motor and gearbox arrangement. The type of trailing arm finalized was a version with two control arms that increase the torsional strength of the arrangement.

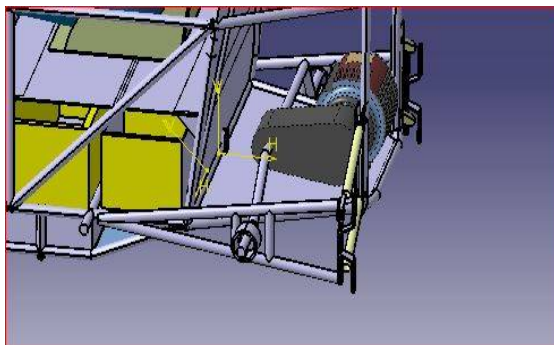


Fig.6: Rear Trailing Arm Assembly

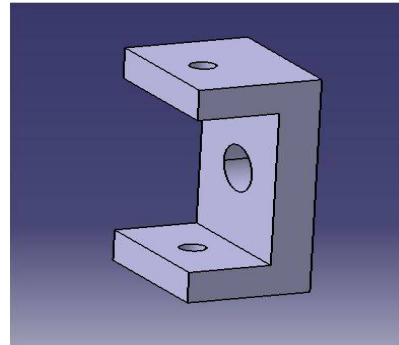


Fig.7: Model of Front Knuckle

D. STEERING KNUCKLE DESIGN

In automotive suspension, a steering knuckle is that part which contains the wheel hub or spindle, and attaches to the suspension components. The wheel and tire assembly attach to the hub or spindle of the knuckle where the tire/wheel

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rotates while being held in a stable plane of motion by the knuckle/suspension assembly. Since the drive was given to the rear wheels, the front knuckle was designed suitable to a non-drive suspension, where the wheels rotate independently.

The design process was started by evaluating the market options and their designs. The suspension system of the various vehicles available commercially is of the MacPherson type. These are different from the Double Wishbone type of suspension system in their mounting and their working. Hence, the design of the knuckle was based on these and modifications were made to the design so as to accommodate the requirements of the Double Wishbone system. A provision was provided at the side for attachment of the steering link attachment. The aim was to keep the design simple so as to facilitate easy manufacturing. The designed knuckle is shown in Fig.7.

VI. ANALYSIS RESULTS

The principles of Finite Element Analysis were applied to the major components of the front suspension in order to verify geometries and material thicknesses. Simulation was used to perform the various studies laid out in this section using NASTRAN/PATRAN software. Before any stress analysis could be performed, some basic forces were calculated using free body diagrams and information from previous Baja projects. Based on the weight of the original vehicle and the average weight of the driver, the total weight of the system was estimated to be around 400 kg. Based on previous projects, it was determined that the maximum acceleration a Baja vehicle is likely to endure during competition is 3g [6]. With this knowledge, 1200kg forces were distributed on each component in various directions in order to simulate the stresses encountered during landing from a jump, lateral acceleration due to turning, and frontal impact. The resulting stress plots are presented in the following sections.

A. KNUCKLE ANALYSIS

The knuckle undergoes many stresses during braking, bump, droop, side impact and front impact. However, the greatest loading that it undergoes is during landing, where the entire load of the vehicle will fall on the knuckle. It was thus tested in order to identify any concentrated stresses in the design. Fixed geometry constraints were added to the ball joint holes at the top and bottom of the knuckle. The knuckle was first loaded with a simple vertical force through the spindle similar to the vehicle landing from a jump. The resulting Von-Mises stress plot for the knuckle is shown in Fig.8. The maximum stress was found to be approximately 159 N/mm², which is well below the yield strength of 270N/mm² for TN8 steel. This shows that the design of the knuckle is safe, and will withstand heavy forces that are acting on it at all times.

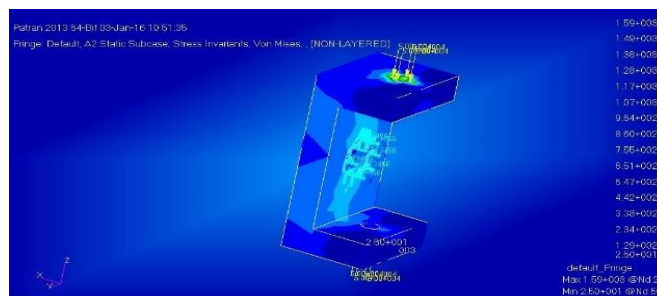


Fig.8: Knuckle Von Mises Stress Analysis during Landing

B. LOWER A-ARM ANALYSIS

The lower arm was also tested for strength during a frontal impact. Being an off-road vehicle, the car must be durable enough to withstand minor collisions and bumps without failure. A 1200 kg force applied at the ball joint housing towards the rear of the vehicle produces a maximum stress of about 161N/mm², as shown in Fig.9, which is even less than the yield for AISI 1018 steel which is 370 N/mm². The maximum displacement is found at the wheel end of the wishbone as shown in Fig.10.

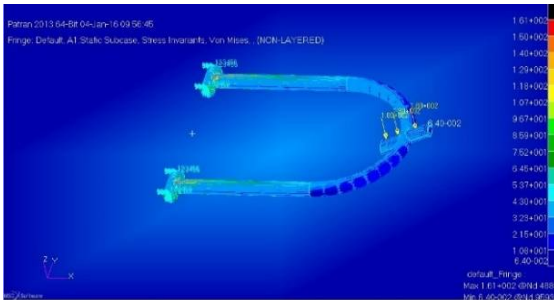


Fig.9: Lower A-Arm Stress Analysis

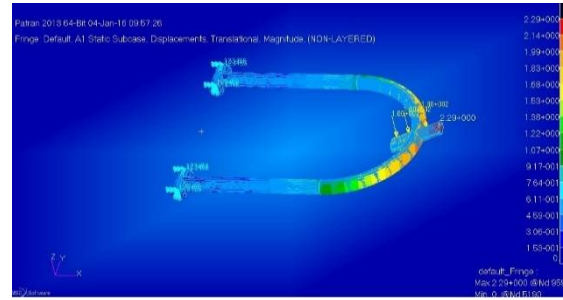


Fig.10: Lower A-Arm Max. Displ. Values

C. UPPER A-ARM ANALYSIS

The lack of shock mount on the upper arm means that the member is always either in pure tension, or pure compression; both cases produce nearly identical maximum stresses. Fig.11 and Fig.12 show the dynamics of the loading including the force applied at the ball joint housing and the fixed geometry constraints applied to the rod end housings. The maximum stress occurs at the corners where the main tube segments meet the rod end housings. At these locations the stress reaches about 280N/mm², as shown in Fig.11; Fig.12 shows a maximum displacement of 6.28mm. These are well below the values for yield strength of AISI 1018 steel (370 N/mm²) and maximum displacement.

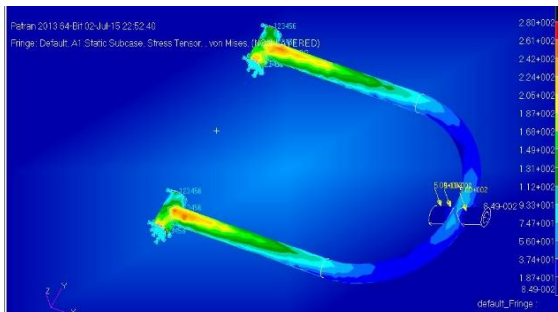


Fig.11: Upper A-Arm Stress Analysis (Frontal Impact)

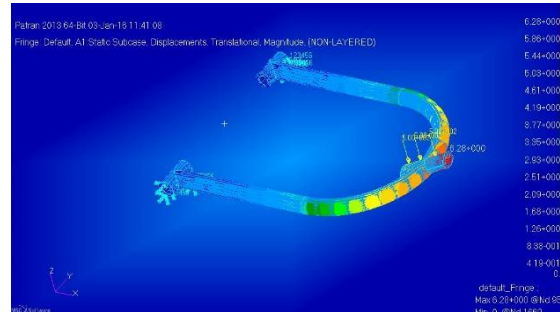


Fig.12: Upper A-Arm Max. Displ. Values

D. TRAILING ARM ANALYSIS

Trailing arm is a part of rear suspension system. The load applied is 800kgs of force and the maximum stress is found to be 176 N/mm², as shown in Fig.13. The material used was ASTM a106b with yield strength of 240 N/mm²; hence the design was found to be safe. Fig.14 shows maximum displacement to be 6.07 mm, which is also a safe value for the given material.

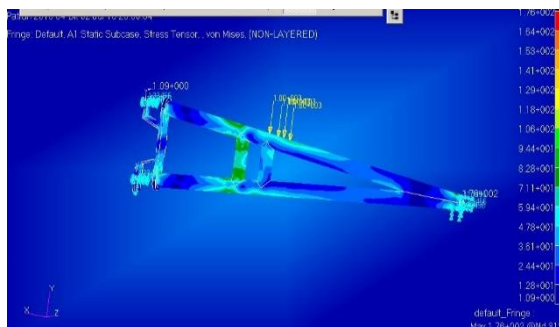


Fig.13: Maximum Stress result of trailing arm

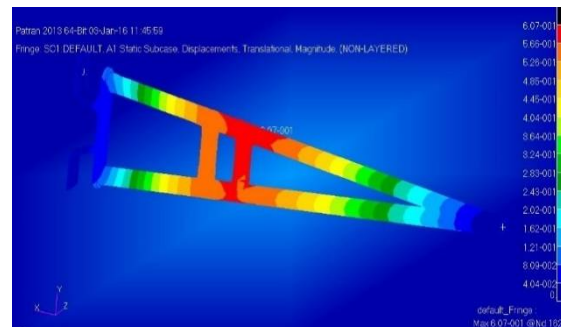


Fig.14: Max. Displ. Values of trailing arm

VII. SIMULATION AND RESULTS

Based on the geometry shown and the design requirements, the suspension systems were independently simulated on Lotus engineering software and the results are given below.

A. FRONT SUSPENSION SIMULATION

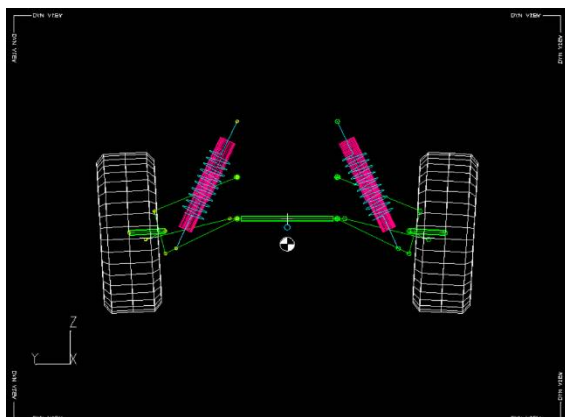


Fig.15: Front Suspension-Static

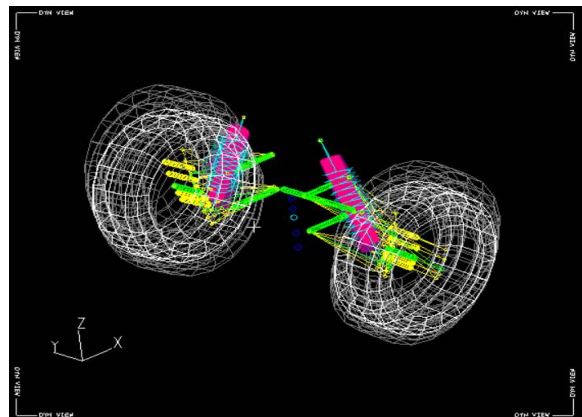


Fig16: Front Suspension- Dynamic Simulation

Based on the front suspension geometry, wishbone design and limitations given by SAE, the hard points on the chassis were calculated. The ideal travel for an all-terrain vehicle with the sprung mass considerations as given before was evaluated to be 4” in bump and 2” in droop. The static camber of the wheels was taken as +3°, to ensure stability of the vehicle in dynamic conditions. These values were fed into the software. Fig.15 is the static condition of the system and Fig.16 shows the complete simulated motion of the wheels in bump and droop.

The results are shown in Fig.17 and Fig.18. Fig.17 gives the camber angle variation of the front wheels as they travels upwards(bump) and then downwards(droop). The camber is positive in droop with a maximum of +4° and

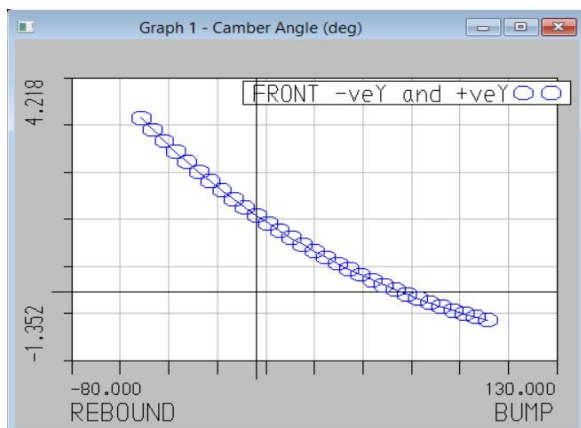


Fig.17: Front Susp.- Camber var. (with Bump and Drp.)

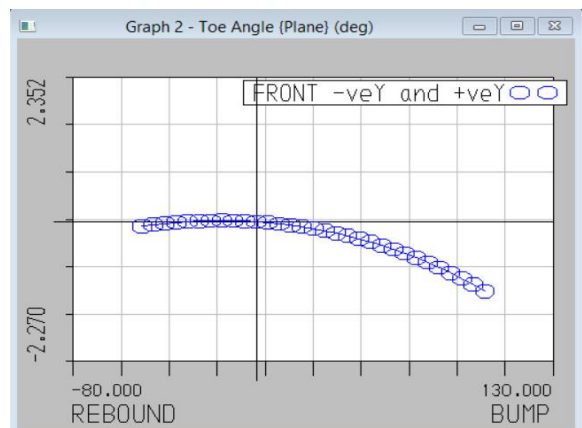


Fig.18: Front Susp.- Toe var. (with Bump and Drp.)

becomes negative in the bump condition with a maximum of -0.7°; this will increase traction of the wheel. Fig.18 shows the variation of toe angle with the motion of the wheel. In an ideal condition, there should be zero toe variation, but due to steering constraints the toe angle varies up to -1.1°. This was rectified by providing suitable positive toe angle during the vehicle assembly. The design requirements of camber gain and near zero toeing in motion were hence met.

B. REAR SUSPENSION SIMULATION

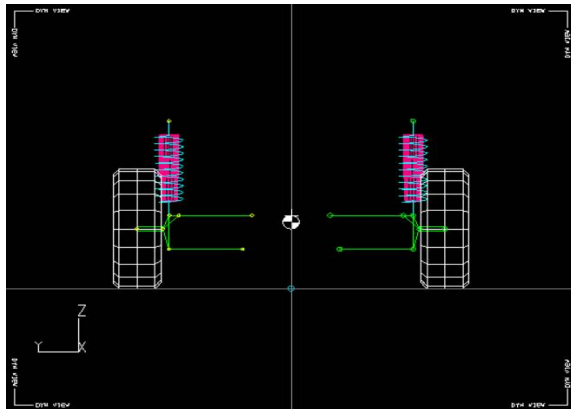


Fig.19: Rear Suspension- Static

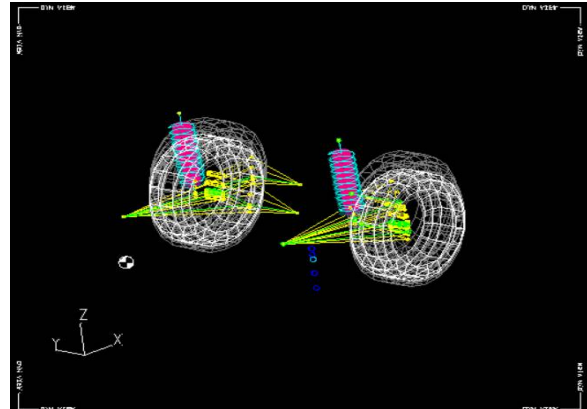


Fig.20: Rear Suspension- Dynamic Simulation

The rear suspension hard points were calculated considering the chassis dimensions and the length limitations of the shocks. The travel was limited to 3” in bump and 1” in droop due to presence of the rear split-axle. No initial camber was given. The simulation was carried out using these values. Fig.19 shows the static condition of the system and Fig.20 shows the simulated positions of the wheels in bump and in droop.

The results are given in Fig.21 and Fig.22. Fig.21 represents the variation of camber angle with bump and droop motion of the wheels. The maximum camber angle achieved in motion is $+0.1^\circ$, which is negligible and will not affect the system greatly. However, the toe angle which must ideally be zero at all times, reaches a maximum of $+1.68^\circ$ as shown in Fig.22. When the motor supplies torque to the wheels, they will tend to move outwards, achieving negative toeing and hence this flaw will be balanced out during vehicle operation. Thus, the design requirements of zero camber and zero toe angle were achieved in the trailing arm system.

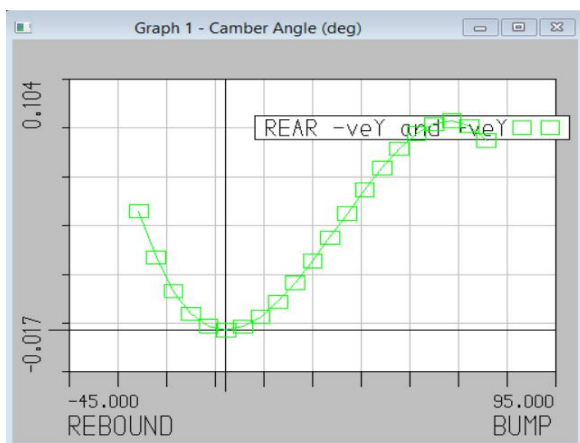


Fig.21: Rear Susp.- Camber var. (with Bump and Drp.)

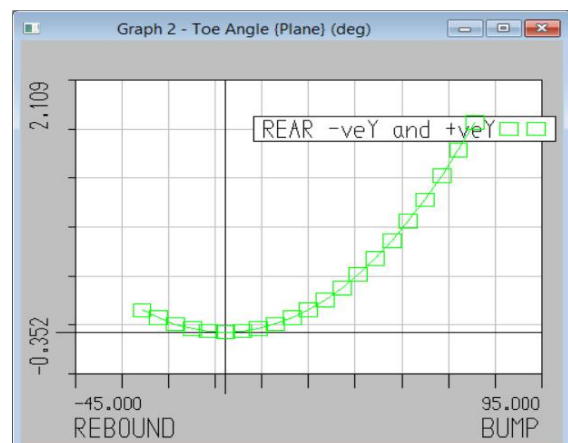


Fig.22: Rear Susp.- Toe var. (with Bump and Drp.)

VIII. VALIDATION AND CONCLUSION

The components were fabricated according to the design and assembled. The vehicle performance was then validated by performing various tests. Measurement of the static parameters such as track width, height, length was carried out and was found to be within the limits specified by the SAE for the E-Baja vehicles [10]. The dynamic testing of the

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vehicle was carried out multiple times and was found to be satisfactory in the aspects of driver safety, driver comfort, ease of steering and manoeuvrability. In conclusion, the implementation of the design in the vehicle achieved the goals of optimum performance by flexibility of camber adjustment, minimum toeing, driver comfort and maximum travel; these were achieved at a reasonable cost.



Fig.23: Vehicle at SAE E-Baja Event

The vehicle is shown in Fig.23. It performed well without any issues at the SAEINDIA E-Baja competition held at Pithampur, Indore in February 2015 and secured 4th place overall.

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