

# International Journal of Innovative Research in Science, Engineering and Technology

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## Design, Analysis and Mockup of Semi-Trailing Rear Suspension for an All-Terrain Vehicle (ATV)

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**ABSTRACT:** The main idea behind the paper is to design, analyze as well as simulate the semi trailing suspension system geometry installed in the rear of an All-Terrain Vehicle(ATV). An ATV as the name suggests is designed to handle a vast variety of terrain than the other vehicles. The Suspension system which is the most important constraint for handling has undergone a substantial amount of development in the past. Thus, the topic is focused on designing the aforementioned suspension system incorporating dynamics of the vehicle with providing optimum performance of the vehicle while maximizing driver comfort. Parameters like camber, wheel travel, roll center were evaluated to attain the desired performance. As the ATV's generally undergo a tremendous amount of shocks in rough terrains, the stability of the vehicle was also given importance. The springs were designed from scratch using calculations in CATIA, followed by arms in Solidworks. The components were analyzed in Ansys software and the complete geometry was simulated in Lotus Suspension Analysis software.

**KEYWORDS:** ATV, Suspension, Catia, Ansys, Lotus Suspension Analysis

### I. INTRODUCTION

All-Terrain Vehicle is one of the category in vehicles which is capable of maneuvering all type of terrains. This is achieved by low wheel pressures and accurate ratio of sprung and un-sprung mass for minimising the momentum to be controlled by the suspension springs. An ATV can comprise of single seat or dual seat as per the manufacturer. ATV's can be equipped with three, four or six wheels according to the specialized use. Because of their tantalising aesthetics, versatile usage- ATV's are used in variety of industries which include border patrol, surveying, adventure activities, etc. Sports variant of the same are also manufactured keeping performance way up ahead than utility. Safety always has been a major issue as it offers negligible protection in case of rollover. Thus, suspension forms the major part of starter pack for a sound ATV.

As with most other aspects of car design, suspension is very much a compromise. The main requirement is to keep all four wheels, and particularly the two driving wheels, in contact with the road at all times. Suspension has to be designed and tuned in such a way that it imparts the ATV with suitable handling characteristics, to insulate the car from the road shocks and to provide road-holding throughout the full range of power transmission. It is in the compromise between road-holding and the ride that most difficulties arise.

### II. RELATED WORK

The suspension systems of the ATV have been briefly discussed before with focussing mainly on the front suspension system, mainly the double wishbone system which is the most common setup. For the rear, the double wishbone has shown better results as the setup provides multiple kinematic parameters for tuning to the accurate performance. Kinematic analysis has been performed previously on the same verifying its advantages. Also, McPherson setup has been worked upon having non-complex geometry and lower un-sprung mass. Because of this, the MacPherson setup is being used in most of the passenger vehicles available today. But MacPherson setup has its own shortcomings due to which trailing and semi-trailing comes into the spotlight. Previous work showcases the importance of trailing setup with camber links. Also, instead of employing conventional springs and dampers, work has been done on using air

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shocks, semi-automatic suspension systems. Software like MATLAB has been used to tune the same. To improve the performance of springs, composite materials have been used proving the better efficiency with lowering the spring weight substantially. Efforts have been made to reduce the unsprung mass as much as possible by manufacturing geometries from Carbon Fibre. Some work has been done related to active suspensions providing better performance than the passive type.

### III. SUSPENSION AND IT'S TYPES

The suspension system is discussed not only just considering the system of springs and dampers but also includes the linkages that connect vehicle and wheels which control the relative motion between the wheels and the vehicle body. This is done to isolate the vehicle chassis from the road shocks which may be in the form of bounce, pitch, roll or sway. These propensities lead to uncomfortable ride and cause additional stress in the automobile frame. Also, maintaining the maximum amount of wheel contact with the road is considered designer's chief goal. This means keeping the wheels, and particularly the outside wheels upright during cornering. Under cornering loads, a very considerable proportion of a car's weight is transferred on to its outside wheels, and thus keeping these wheels upright is far more important than keeping the inside front wheel on the ground. Besides the dynamic parameters, other factors considered in the design process are cost, weight, package space, manufacturability, assembly, etc.

Contingent upon the type and usage, different categories of vehicles are installed with varying suspension geometries. Generally, suspension is classified as Dependent suspension system and independent suspension system.

**A. Dependent Suspension System:** Such suspension system performs as a rigid beam such that behavior of one wheel affects the movement of the other wheel. Basically, force affecting one wheel alters adjacent wheel. Such systems are usually installed in the rear of heavy vehicles like trucks and buses. Some examples are:

- Panhard rod
- Leaf spring Suspension
- Watt's Linkage
- Satchell link
- WOblink

**B. Independent Suspension System:** As the name suggests, with this type of Suspension system the front or rear sets of wheels act independently while acting upon bumps or droops. These are not connected with a rigid beam altogether, thus the nature of forces of one wheel does not transfers to the adjacent wheel. Some examples are:

- Double Wishbone Suspension
- MacPherson Strut
- Swing axle
- Multi-link Suspension
- Trailing/Semi-Trailing Arm

### IV. SUSPENSION CONSIDERATION

While deciding the rear geometry for an ATV, there are several criteria which needs to be satisfied. Criteria's like space constraint, dynamic nature of wheel, up-sprung weight, reliability or even type of usage comes into the picture. There were several geometries which were considered for installation in the rear. The first option being the double wishbone setup, which has the geometry of two arms in parallel or non-parallel setup. Double Wishbone setup could be beneficial as it allows the designer to evaluate the kinematic parameters easily and control the wheel characteristics in the event of wheel movement vertically. But due to its excessive un-sprung weight and extreme requirement for space, the double wishbone suspension was not considered. Next for the consideration was the Trailing arm geometry. Trailing arm suspension employs two trailing arms which are pivoted to the car body at the arm's front edge. Now, this would allow the wheels to move up and down in the case of bump. In the event of rolling the trailing arm also tends to roll with same degree as with the chassis thus leading to the camber gain in the rear wheels with respect to the road surface. Because of this even this geometry was dropped.

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Lastly, moving on to semi-trailing setup, it has the trailing arm pivoted at inclined angle of about 60 to 70 degrees. As the name suggests “semi”, this geometry is half trailing and half transverse. This results in neutral steering response and controls both lateral and vertical motions. Thus, semi-trailing geometry was chosen for the rear. Regarding the shock absorbers, conventional system of springs and dampers were chosen.

## V. DESIGN

### A. Introduction

The design process for the rear suspension began with the evaluation of overall track width and wheelbase of the vehicle. The track width was kept minimum as possible to aid in manoeuvrability. The wheelbase was also kept as minimum as possible to reduce the turning radius and prevent from rolling in the event of harsh turns. To maximize obstacle avoidance, a ground clearance of 13.5 inches from the ground to the lowermost member on the chassis was chosen.

The following components were designed as per the requirements:

- Semi-Trailing geometry
- Rear upright
- Spring

The ATV dimensions are as follows

**Table 1: Vehicle Dimensions**

| Vehicle Dimensions at Static Height | Value in inches |
|-------------------------------------|-----------------|
| Track Width                         | 52              |
| Wheelbase                           | 61              |
| Ride Height                         | 13.5            |

### B. Design for Semi-Trailing Geometry

The design of the semi trailing geometry becomes the initial step for the suspension system. The material was chosen based on the requirements. The roll center was determined considering the center of gravity and influence of jacking forces. Finally, the geometry was modeled in CATIA.

#### I. Material Selection

The strength of the material should be well enough to withstand all the loads acting on it in dynamic conditions. The material selection also depends on a number of factors such as carbon content, material properties, availability and the most important parameter is the cost.

At first, three materials were shortlisted based on their availability in the market which are AISI 1018, AISI 1040 and AISI 4130.

For the arm geometry of the semi-trailing system, AISI 4130 was finally chosen based on Pugh’s concept of optimization.

#### II. Pugh’s Concept

Pugh’s method or decision-matrix method is a qualitative technique used to rank the multi-dimensional options of a set. It basically consists of establishing a set of criteria options which are scored and summed to gain a total score which can be then ranked accordingly. The final symbols or ratings get converted into scores and are combined in the matrix to yield scores for each option.

#### III. Using Pugh’s Concept

The major distinguishing properties of distinguishing materials are as follows

**Table 2: Characteristics of Metals**

| Properties             | AISI 4130 | AISI 1040 | AISI 1018 |
|------------------------|-----------|-----------|-----------|
| Carbon Content (%)     | 0.30      | 0.40      | 0.18      |
| Tensile Strength (MPa) | 560       | 620       | 440       |
| Yield Strength (MPa)   | 460       | 415       | 370       |
| Hardness (BHN)         | 217       | 201       | 126       |
| Cost (Rs. /metre)      | 700       | 425       | 300       |

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**Table 3: Selection using Decision-matrix**

| Description             | AISI 4130 | AISI 1040 | AISI 1018 |
|-------------------------|-----------|-----------|-----------|
| <b>Total Weight</b>     | +2        | 0         | -1        |
| <b>Tensile Strength</b> | 0         | +2        | -1        |
| <b>Yield Strength</b>   | +1        | 0         | -1        |
| <b>Cost</b>             | -1        | 0         | +1        |
| <b>Net elongation</b>   | +2        | 0         | -2        |
| <b>Total Score</b>      | <b>+4</b> | <b>+2</b> | <b>-4</b> |

As the net score for AISI 4130 is highest, it is selected for the suspension material.

#### IV. Stress Calculation

The allowable stress for AISI 4130 is evaluated from the following relation-

$$\sigma = \frac{S_{yt}}{fos}$$

Assuming the factor of safety 2 as AISI 4130 is a ductile material

$$\sigma = \frac{460}{1.5}$$

$$\sigma = 306.6 \text{ MPa}$$

This is the value for allowable stress in the suspension arms. The suspension will remain safe until and unless the induced stress is less than the allowable stress.

#### V. Evaluation of Roll Center

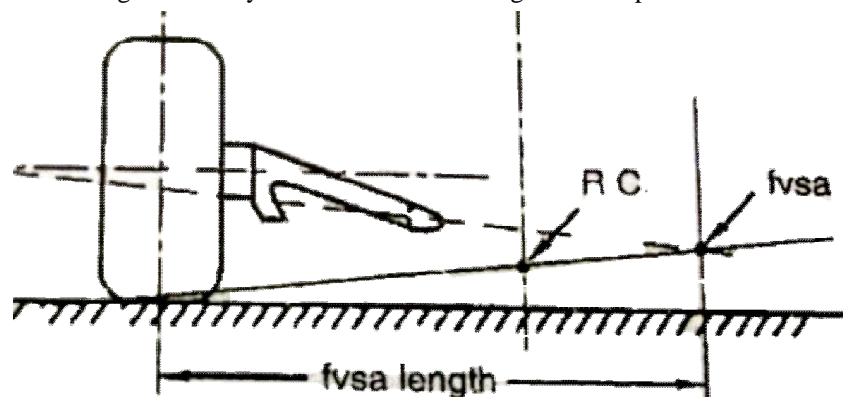
Roll center is that virtual point in the transverse vertical plane through any pair of wheel centers and equidistant from them, at which lateral forces may be applied to the sprung mass without producing an angular (roll) displacement of the sprung mass.

Basically, roll center is a geometrical point at which the cornering forces are reacted on the vehicle body. The point varies for front and rear suspension accordingly and also is contingent upon the suspension geometry.

Roll center establishes a force coupling point between sprung and un-sprung mass. Higher the roll center, lateral forces will act higher off the ground leading to small moment across the center of gravity causing jacking forces. Lower the roll center, lateral forces act at lower height thus eventually large roll moment meaning suspension travel is used by the roll.

For the semi-trailing geometry, the evaluation of the roll center is shown in Fig. 1. The arms are extended till the point of convergence and are joined with the tire center point at the contact patch.

Thus, evaluation of the roll center height is usually a tradeoff between a high and low point.



**Fig. 1- Location of RC in Semi Trailing Geometry**

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## VI. Modelling of Semi-Trailing Arm

The suspension was modelled in Solidworks as shown in Fig. 2. Clamps are also shown in the arms to provide provision for attaching hub and the wheel.

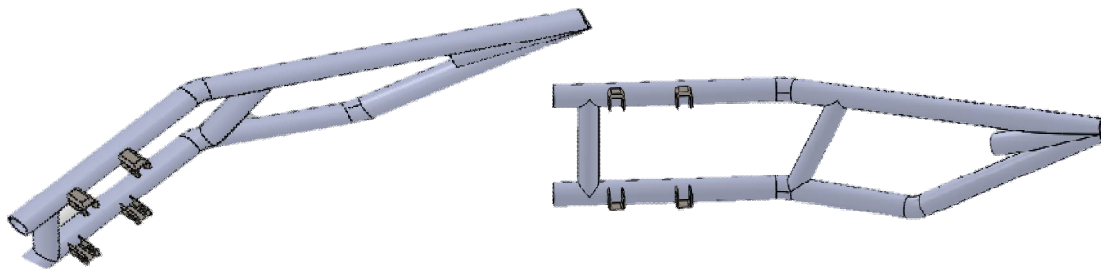


Fig. 2- Semi-Trailing Arm Model

## C. Design of Rear Spring

The coil springs are used mainly for the application in independent type suspensions. The spring basically stores the elastic energy and is compressed due the force exerted on it. The force it exerts is proportional to the change in its length. The springs are accompanied by dampers or shock absorbers. Shock absorbers slow down and reduce the magnitude of vibratory motions by turning the kinetic energy of suspension movement into heat energy that can be dissipated through hydraulic fluid. While the design of spring has been considered, the design and calculations of dampers are out of the scope of this article.

As the ATV undergoes braking, acceleration or body roll (while turning) load transfer takes place. Due to braking load will be transferred to the front wheels, while accelerating load is transferred to the rear wheels. In the event of turning the maximum load is transferred to the outer wheels due to lateral forces acting on the body. With the harsh use of ATV, there are some extreme cases when the complete load is transferred onto a single wheel. That wheel has to undergo a tremendous amount of compression and if the suspension system isn't designed for the same, it could result in a major fatal accident. Thus, as per the following calculations and parameters the suspension spring model was designed as shown in the Fig. 3.

Mass of the ATV considered with driver = 250kg

Hence, the net load acting on a single wheel =  $250 \times 9.81 = 2453 \text{ N}$

According to the design the centre to centre distance of suspension arm is 28.27 inches. The spring and damper system would be mounted 14 inches from the front end.

Thus, considering the forces acting on the arm in the event of bump, the spring comes out as follows-

From the moment equation, we get

$$2453 \times 28.28 = SF' \times 20$$

$$SF' = 3468.54 \text{ N}$$

This SF' force is the vertical component of the spring force acting in upward direction.

Taking the cos component, net force on the spring comes out as-

$$SF = \frac{SF'}{\cos(20^\circ)}$$

$$SF = \frac{3468.54}{\cos(15^\circ)} = 3590.89 \text{ N}$$

By considering the appropriate suspension travel as 5.5 inches to accommodate extreme forces, we get

$$\text{Spring Stiffness, } k = \frac{3590.89}{4.5 \times 25.4}$$

$$k = 31.41 \text{ N/mm}$$

With the spring stiffness calculated, selection of spring material comes next. There are several factors which are to be kept in mind for the spring material. Some are as follows-

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- Net load acting on the spring
- The range of stresses through which spring operates
- The expected fatigue life
- Environmental conditions like temperature, humidity

The cold-drawn steel wires were considered for our use as they are made of high carbon steel containing 0.85-0.95% of carbon. They have high tensile strength, high elastic limit and ability to withstand high stresses under repeated loadings. Further calculations are as follows-

Permissible shear stress,  $\tau = 0.50 \times 650 = 325 \text{ N/mm}$ , Spring Index (C) assumed = 7

$$\text{Wahl factor, } K = \frac{4C-1}{4C-4} + \frac{0.615}{C} = 1.212$$

$$\text{Thus, wire diameter, } d = \sqrt{\frac{8FCCK}{T\pi}} = 15.45 \text{ mm}$$

$$\text{Mean Coil Diameter, } D = Cd = 108.15 \text{ mm}$$

$$\text{Number of Active Coils, } N = \frac{Gd^4}{8D^3K} = 14.7 \approx 15, \text{ where } G = 81370 \text{ N/mm}^2$$

For this configuration, square and ground ends type of springs are used

$$\text{Total gap} = (N-1) \times \text{gap between two adjacent coils} = 28 \text{ mm}$$

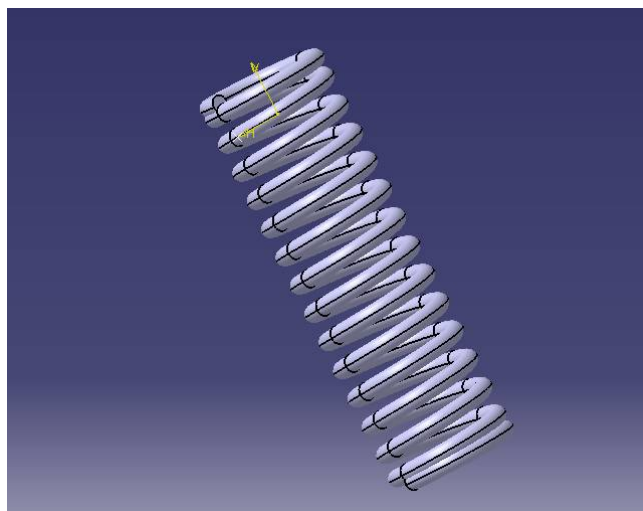
$$\text{Solid Length} = N \times d = 15 \times 15.45 = 231.75 \text{ mm}$$

$$\text{Free Length} = \text{Solid length} + \text{Total Gap} + \delta = 231.75 + 28 + (25.4 \times 4.5) = 374.05 \text{ mm}$$

$$\text{Pitch of the spring, } p = \frac{\text{Free Length}}{N-1} = 26.71 \text{ mm}$$

**Table 4: Summary of Spring parameters**

| Sl. No. | Parameter                       | Value      |
|---------|---------------------------------|------------|
| 1.      | Mean diameter of wire (D)       | 108.15 mm  |
| 2.      | Wire diameter (d)               | 15.45 mm   |
| 3.      | No. of turns (N)                | 15         |
| 4.      | Solid length                    | 231.75 mm  |
| 5.      | Free length                     | 374.05 mm  |
| 6.      | Pitch of spring (p)             | 26.71 mm   |
| 7.      | Spring Rate (k)                 | 31.41 N/mm |
| 8.      | Maximum deflection ( $\delta$ ) | 114.3 mm   |



**Fig. 3- Spring Model**

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## VI. ANALYSIS

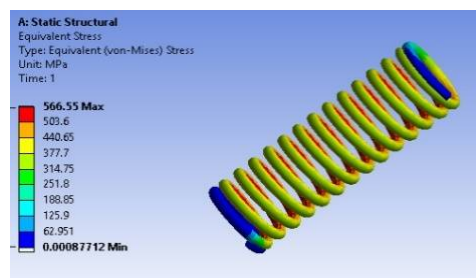
While designing any component/assembly it becomes mandatory to verify whether the system will sustain in extreme conditions. It becomes necessary to verify the defined geometries and analyze the same in real like conditions. Thus, the analysis was performed in Ansys software. For the same, respective ultimate stresses were considered as maximum value. Various forces were considered and applied to the respective geometries parallel with real life off-roading scenarios. The results plotted are shown below.

### A. Spring Analysis

The analysis of spring was done in Ansys software as shown in Fig. 4 to determine the deflection in the spring equivalent to the maximum spring force. In the analysis, one end was kept fixed and a vertical load was applied on the opposite end.

**Table 5: Summary of Spring parameters**

| Parameters            | Value      |
|-----------------------|------------|
| Maximum Force Applied | 3600 N     |
| Maximum Deflection    | 91.5 mm    |
| Stress Induced        | 566.55 MPa |

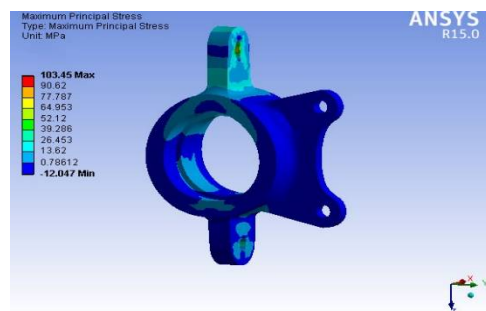


**Fig. 4- Equivalent stresses in Spring**

### B. Upright Analysis

An upright assembly, commonly referred as wheel upright assembly is an automotive component used in the majority of vehicles to hold certain mechanical components like suspension arms, brake drums or discs. The upright is mounted on the chassis using a set of bolts. The wheel is further mounted on the upright assembly through bolts. Wheel upright forms the critical part of the suspension system as it not only holds the wheel in place but also mounts the suspension arm. Because of this, the upright undergoes various loads like radial load, axial load, tangential load, fatigue load during several running conditions.

Because of these huge loads acting on the rear wheel upright, the Finite element analysis (FEA) is used after designing process for checking factor of safety and what would be possible changes that can provide the adequate design of it.



**Fig. 5- Equivalent stresses in Upright**

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As mentioned earlier, the upright undergoes huge loads in running conditions. Still, the greatest loading that it undergoes is during landing on a single wheel with upright tolerating the complete vehicle load. For the same, analysis as shown in Fig. 5 was done by keeping the mounting points for brake caliper fixed. This resulted in Maximum Principal stress distribution as shown in the figure. The maximum stress was found to be approximately 103 MPa which quite below the recommended yield strength of 276 MPa for 6000 series Aluminum alloys proving the upright is safe for use in given conditions.

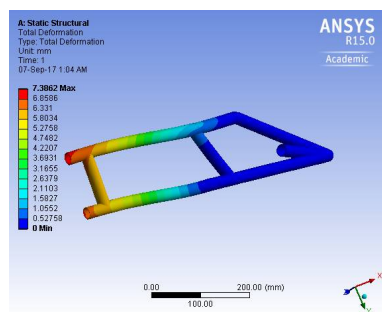


Fig. 6- Max. Deformation in Suspension arm

### C. Suspension Arm Analysis

The analysis of the suspension arm was done. The maximum deformation is shown in Fig. 6 which is 7.3 mm, well under limits. The maximum stress is 105 MPa which is also under limits as the material used was AISI 4130 having allowable stress of 306.6 MPa with factor of safety 1.5.

## VII. MOCKUP

To simulate the final geometry as per the actual scenario, the rear suspension geometry was plotted in Lotus Suspension Analysis. The geometry was tuned as per the need. For fixing the suspension geometry to the chassis hard points are required, which are obtained from Lotus suspension analysis. This software provides a broad range of parameters for tuning the same. Tire configuration is 23x7-10. The results are shown below.

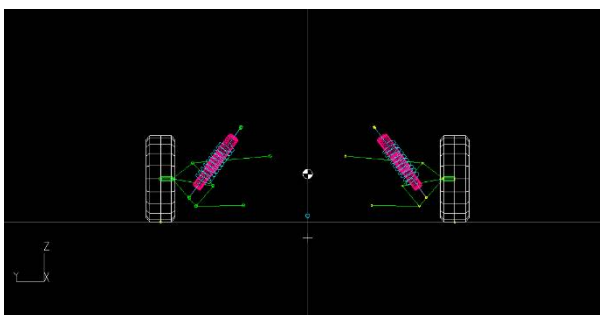


Fig. 7- Suspension- Static Display

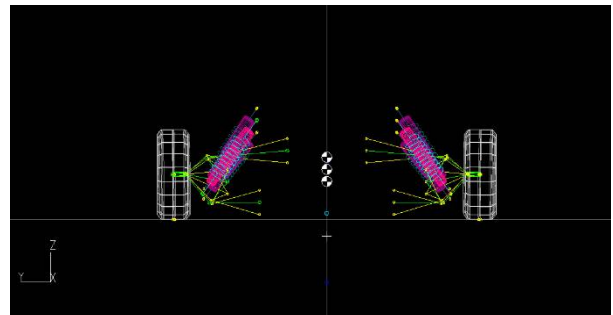


Fig. 8- Suspension- 3D Bump Simulation

The rear suspension was simulated in Lotus software as per the initial 3D parameters as shown in Table 6. Camber angle was set neutral as there isn't a requirement for camber gain in cornering for the rear wheels as such. Bump and rebound travel were set accordingly considering the spring designed. The tire dimension is 23x7-10. Fig. 7 shows the suspension geometry in the static condition. Fig.8 illustrates the system in dynamic condition undergoing a bump followed by roll condition in fig. 10.



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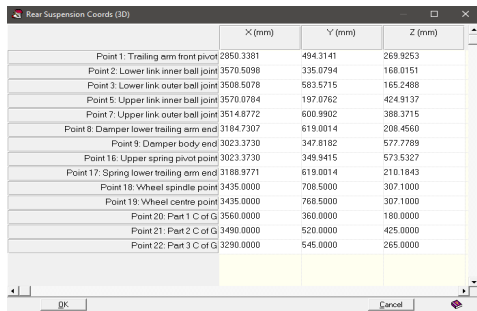
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Fig. 11 illustrates the change in camber angle undergoing a bump and droop. The maximum camber angle gained is  $+1.6^\circ$  in the incident of the bump. But while the ATV undergoes a droop it gains a slight negative camber angle helping in the generation of more mechanical grip thus more traction at the wheels. This geometry as helps in gaining negative camber angle in the event of rolling thus ensuring optimum amount of traction while cornering.

As all the conditions are satisfied the final hard points were noted for manufacturing as shown in Fig. 9.

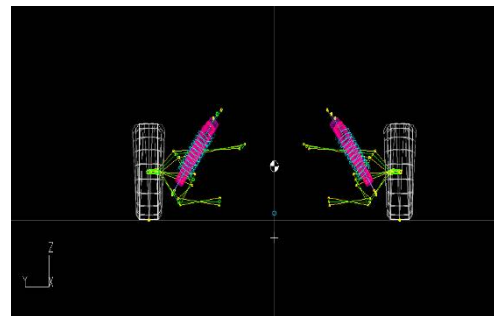
**Table 6: Initial 3D parameters**

| Parameter             | Value          |
|-----------------------|----------------|
| <b>Bump Travel</b>    | <b>76.2 mm</b> |
| <b>Rebound Travel</b> | <b>50.8 mm</b> |
| <b>Wheelbase</b>      | <b>1550 mm</b> |
| <b>Camber Angle</b>   | <b>0°</b>      |

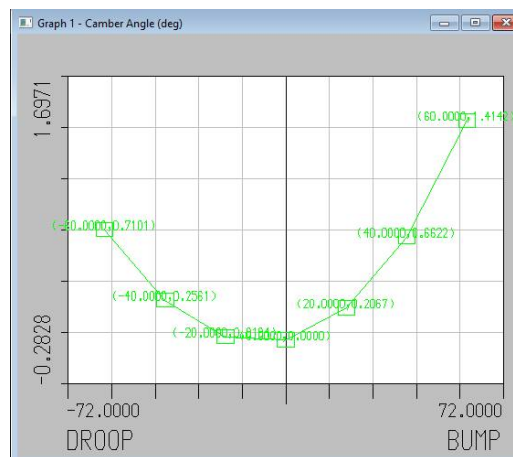


|   | X (mm)    | Y (mm)   | Z (mm)   |
|---|-----------|----------|----------|
| Point 1: Trailing arm front pivot       | 2850.3381 | 494.3141 | 269.9253 |
| Point 2: Lower link inner ball joint    | 3570.5098 | 335.6794 | 168.0151 |
| Point 3: Lower link outer ball joint    | 3508.5078 | 583.5715 | 165.2488 |
| Point 5: Upper link inner ball joint    | 3570.0784 | 197.0762 | 424.9137 |
| Point 7: Upper link outer ball joint    | 3514.8772 | 600.9902 | 388.3715 |
| Point 8: Damper lower trailing arm end  | 3184.7307 | 619.0014 | 208.4560 |
| Point 9: Damper body end                | 3023.3730 | 347.8182 | 577.7789 |
| Point 16: Upper spring pivot point      | 3023.3730 | 349.9415 | 573.5327 |
| Point 17: Spring lower trailing arm end | 3188.9771 | 619.0014 | 210.1843 |
| Point 18: Wheel spindle point           | 3435.0000 | 708.5000 | 307.1000 |
| Point 19: Wheel centre point            | 3435.0000 | 708.5000 | 307.1000 |
| Point 20: Part 1 C of G                 | 3560.0000 | 360.0000 | 160.0000 |
| Point 21: Part 2 C of G                 | 3490.0000 | 520.0000 | 425.0000 |
| Point 22: Part 3 C of G                 | 3280.0000 | 545.0000 | 265.0000 |

**Fig. 9- Suspension- Hard Points for Manufacturing**



**Fig. 10- Suspension- 3D Roll Simulation**



**Fig. 11- Camber Variation with bump and droop**

## VIII. CONCLUSION

The design, analysis, and mock-up were carried out in this article. Initially the suspension components like spring, arms and upright were designed in the CAD software. The calculations for the appropriate spring has been carried out. The components have been analysed in Ansys for any failures and evaluating the nature of the components under various forces. Lotus Suspension Analysis gave suspension simulation results under by plugging in the exact values of vehicle geometry. The required objectives of optimum suspension travel, driver comfort, and better handling were achieved maximising the performance of the suspension system.

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The suspension further can be improved by decreasing the un-sprung mass like improving the geometry of the upright and installing an antiroll bar for controlling the roll centre height and jacking forces. Instead of a conventional system of springs and dampers, air shocks could be installed for better tuning and efficiency although air shocks come with many caveats.

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