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# DESIGN AND OPTIMIZATION OF SUSPENSION GEOMETRY FOR AN ELECTRIC ALL-TERRAIN VEHICLE

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**Abstract -** *A suspension system is one of the most important systems of an automobile that deals with the dynamics of the vehicle. It is the intermediate flexible system that connects the wheels with the main frame of the vehicle; suspension system is a combination of various components like knuckle or upright (that have important angles like king-pin, caster), Arms or linkages and shock absorber that comes together and enables the relative motion between the tyre and the mainframe. Suspension systems must support both road holding/handling and ride quality, which are at odds with each other. The tuning of suspensions involves finding the right compromise. The optimization of suspension geometries is done using LOTUS software. The obtained values are expressed as graphs to visually understand the relationship between suspension parameters and vehicle performance. The process of iterative optimization is followed to improve the design to meet the loading conditions. The results are noted down, the designs are optimized and the most promising values are concluded*

***Key Words***: **Suspension, double wishbone, A-arm, H-arm, ATV, shock absorber.**

## 1. INTRODUCTION

The suspension system is made up of several components, including the chassis, which holds the cab of the car. The springs support the vehicle weight and absorb and reduce excess energy from road shocks, along with the shock absorbers and struts. Specifically, the suspension system maximizes the friction between the tires and the road to provide steering stability and good handling. Suspension geometry varies for automobile based on off road and on road conditions. An off-road vehicle is considered to be any type of vehicle which is capable of driving on and off paved or gravel surface. It is generally characterized by having large tires with deep, open treads, a flexible suspension. All-Terrain Vehicles (ATV) are small, open, and single�seated motor vehicles with three or four tires. These tires have deep-treaded structures that allow the vehicles to be driven on rocky, muddy, and root cover terrains. With a host of applications, ATV's are ideal for wetlands and sand dune topologies [1].

### 1.1 DIFFERENT TYPES OF SUSPENSION SYSTEM

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.

**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. [2].

## 2. 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 For the rear wheels, the main factors applicable are weight, cost and functionality. The H-arm with camber link suspension system is used to provide good balance between ground clearance, travel and adjustability.

## 3. KINEMATIC ANALYSIS

Kinematic analysis is done to validate the wheel travels for the corresponding damper position during sag, jounce, rebound. These values are incorporated in dynamic analysis to determine the travel limitations of the geometry. They are also vital in damper selection. In kinematic analysis the individual components of a RC diagram are blocked together and suitable constraints are given to simulate the kinematic model of the geometry.

### 3.1 ROLL CENTRE DIAGRAM

Kinematic analysis of suspension geometry is performed using Solidworks. Roll centre diagram is first generated in order to determine the position of the suspension components. We include the necessary suspension parameters like ground clearance, KPI, wheel end thickness and tire dimensions. This helps us to generate the 2D diagram of the suspension geometries.

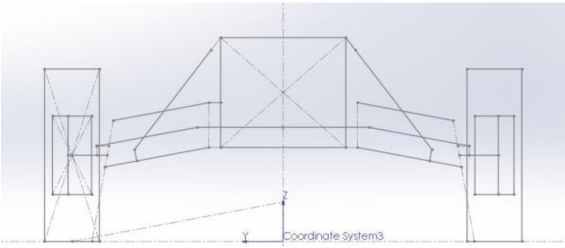


**Fig -1**: Roll centre diagram of double wishbone

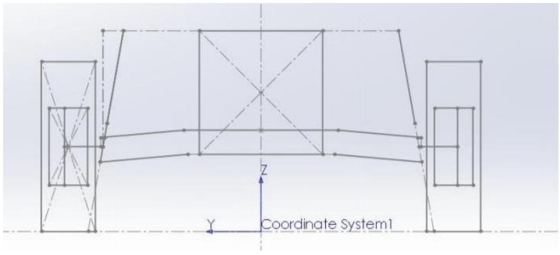


**Fig -2**: Roll centre diagram of H-arm

### 3.2 SAG

Suspension sag is the amount of suspension travel used when you are sitting on your automobile in a natural riding position

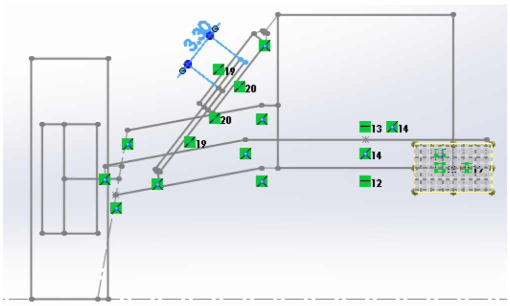


**Fig -3**: Sag position (Double wishbone)

For the above mentioned geometry the sag position is obtained at a suspension travel of 2.2 inches. For the selected damper the entire travel is about 5.5 inches. When a sprung weight of 200 kg acts on a damper it is compressed to 2.2 inches to obtain the natural riding position. This can be validated by measuring the ground clearance of 12 inches which was set in the roll centre diagram.

### 3.3 JOUNCE

Jounce refers to the bounce or vertical movement of the vehicle suspension upward when it contacts a bump in the road.

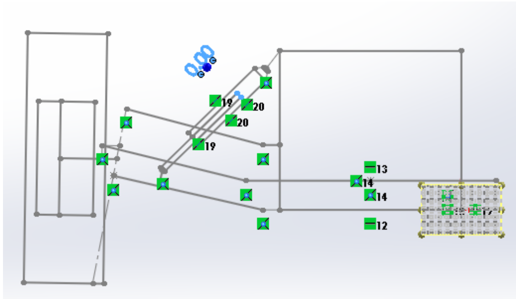


**Fig -4**: Jounce position (Double wishbone)

For the above mentioned geometry the jounce condition is obtained for the maximum damper travel of 5.5 inches. When the vehicle encounters a bump the maximum damper travel is obtained. From the kinematic analysis it is calculated that this enables a vertical wheel travel around 6 inches (150mm).

### 3.4 REBOUND

Rebound refers to the movement of the vehicle suspension in the opposite direction of jounce. For this geometry the rebound condition is obtained when the damper retracts a length of 2.2 inches from its sag position. When the vehicle enters a pothole the damper expands to enable better contact to the road. From the kinematic analysis it is calculated that the rebound wheel travel around 4 inches (100mm).

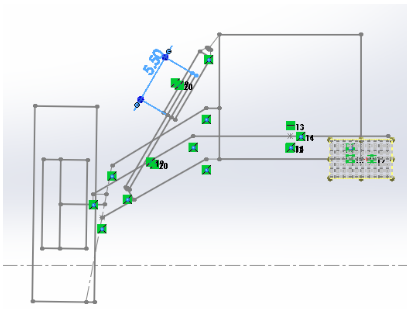


**Fig -5**: Rebound position (Double wishbone)

From the kinematic analysis we conclude that for a damper travel of 5.5 inch we obtain an entire wheel travel of 250 mm for the above constructed geometry. These values are noted down and are incorporated in dynamic analysis of LOTUS SHARK.

## 4. DYNAMIC ANALYSIS

The Lotus Suspension Analysis SHARK module is a suspension geometric and kinematic modelling tool, with a user- friendly interface which makes it easy to apply changes to proposed geometry and instantaneously assess their impact through graphical results. The main purpose of performing dynamic analysis is to validate the changes in suspension parameters such as camber, castor, toe angle etc.

during the dynamic conditions of the vehicle. This is done by constructing a 3D model in LOTUS SHARK using the co-ordinates obtained from the roll centre diagram

### 4.1 METHODOLGY

The required type of suspension is first selected and the 3D co-ordinates are entered. From the kinematic analysis the wheel travel values are integrated with the LOTUS SHARK and dynamic analysis is carried out. The important parameters like tire data, CG value and wheelbase are added which are vital for the calculation [3]. Then the geometry is animated for the conditions of bump, roll and steer. This enables us to visualize the suspension motion for those conditions. The dynamic change of the parameters with respect to wheel travel is obtained in results.

```
                 STATIC VALUES
         Camber Angle  (deg):      0.00
     Toe Angle {Plane} (deg):      0.00
       Toe Angle {SAE} (deg):      0.00
         Castor Angle  (deg):      5.00
    Castor Trail (hub)  (mm):      0.00
  Castor Offset (grnd)  (mm):     24.44
        Kingpin Angle  (deg):     10.00
  Kingpin Offset (w/c)  (mm):    114.30
 Kingpin Offset (grnd)  (mm):     65.04
Mechanical Trail (grnd) (mm):     24.35
    ROLL CENTRE HEIGHT  (mm):    125.22

             GENERAL DATA VALUES
   TYRE ROLLING RADIUS  (mm):    279.40
             WHEELBASE  (mm):   1473.20
        C OF G HEIGHT   (mm):    508.00
 BREAKING ON FRONT AXLE  (%):     60.00
    DRIVE ON FRONT AXLE  (%):      0.00
   WEIGHT ON FRONT AXLE  (%):     40.00
       OUTBOARD FRONT BRAKES:
  INDEPENDENT FRONT SUSPENSION:
RACK TYPE STEERING ARTICULATION:
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**Fig -6**: Lotus static values

INCREMENTAL GEOMETRY VALUES

| Bump Travel (mm) | Camber Angle (deg) | Toe Angle (SAE) (deg) | Castor Angle (deg) | Kingpin Angle (deg) | Damper1 Ratio (-) | Spring1 Ratio (-) |
|---|---|---|---|---|---|---|
| 75.00 | -1.3959 | 1.2491 | 6.0016 | 11.5452 | 0.989 | 0.989 |
| 70.00 | -1.3619 | 1.2598 | 5.9208 | 11.5092 | 0.996 | 0.996 |
| 65.00 | -1.3176 | 1.2546 | 5.8424 | 11.4613 | 1.003 | 1.003 |
| 60.00 | -1.2637 | 1.2343 | 5.7663 | 11.4023 | 1.010 | 1.010 |
| 55.00 | -1.2005 | 1.1995 | 5.6925 | 11.3327 | 1.017 | 1.017 |
| 50.00 | -1.1290 | 1.1513 | 5.6207 | 11.2532 | 1.037 | 1.037 |
| 45.00 | -1.0488 | 1.0893 | 5.5510 | 11.1642 | 1.044 | 1.044 |
| 40.00 | -0.9605 | 1.0144 | 5.4831 | 11.0662 | 1.050 | 1.050 |
| 35.00 | -0.8646 | 0.9272 | 5.4171 | 10.9595 | 1.056 | 1.056 |
| 30.00 | -0.7614 | 0.8280 | 5.3529 | 10.8447 | 1.062 | 1.062 |
| 25.00 | -0.6510 | 0.7172 | 5.2903 | 10.7220 | 1.067 | 1.067 |
| 20.00 | -0.5338 | 0.5951 | 5.2292 | 10.5916 | 1.073 | 1.073 |
| 15.00 | -0.4099 | 0.4621 | 5.1698 | 10.4540 | 1.078 | 1.078 |
| 10.00 | -0.2796 | 0.3185 | 5.1118 | 10.3094 | 1.083 | 1.083 |
| 5.00 | -0.1430 | 0.1646 | 5.0552 | 10.1580 | 1.088 | 1.088 |
| 0.00 | -0.0004 | 0.0005 | 4.9999 | 10.0000 | 1.092 | 1.092 |
| -5.00 | 0.1483 | -0.1735 | 4.9460 | 9.8356 | 1.097 | 1.097 |
| -10.00 | 0.3027 | -0.3572 | 4.8933 | 9.6649 | 1.101 | 1.101 |
| -15.00 | 0.4628 | -0.5505 | 4.8418 | 9.4882 | 1.105 | 1.105 |
| -20.00 | 0.6286 | -0.7532 | 4.7914 | 9.3056 | 1.109 | 1.109 |
| -25.00 | 0.7999 | -0.9651 | 4.7422 | 9.1172 | 1.113 | 1.113 |
| -30.00 | 0.9766 | -1.1861 | 4.6941 | 8.9230 | 1.116 | 1.116 |
| -35.00 | 1.1586 | -1.4163 | 4.6470 | 8.7233 | 1.120 | 1.120 |
| -40.00 | 1.3460 | -1.6554 | 4.6009 | 8.5180 | 1.123 | 1.123 |
| -45.00 | 1.5386 | -1.9034 | 4.5557 | 8.3072 | 1.126 | 1.126 |
| -50.00 | 1.7365 | -2.1603 | 4.5116 | 8.0911 | 1.128 | 1.128 |

**Fig -7**: Lotus dynamic values

As the result of the dynamic analysis the changes in suspension parameters such as camber, toe, castor, KPI, etc. are obtained with respect to the change in wheel travel. These parameters help us define the vehicles performance in terms of straight line stability, cornering and steering. By modifying the 3D coordinates, these values can be adjusted and suitable performance of the vehicle can be obtained.

## 5. OPTIMIZATION OF DOUBLE WISHBONE SUSPENSION

In double wishbone suspension geometry we have assumed both the upper and lower control arms to be parallel and of unequal lengths [4]. The tie rod length is calculated to be in relation with the lengths of the control arms, and is mounted parallel to them to enable better steering.

**Table -1:** Double wishbone static values

| S.No | Suspension parameters | Values |
|---|---|---|
| 1 | Castor | +5 deg |
| 2 | camber | 0 deg |
| 3 | Toe | 0 deg |
| 4 | KPI | +10 deg |
| 5 | Scrub radius | 65.034mm |
| 6 | Track width | 1371.6mm |
| 7 | Wheel base | 1473.2mm |
| 8 | Bump | 150mm |
| 9 | Rebound | 100mm |

**Table -2:** Double wishbone co-ordinates

| S.No | POINT NAME | X(mm) | Y(mm) | Z(mm) |
|---|---|---|---|---|
| 1 | Lower wishbone front pivot | -114.3 | 241.3 | 304.800 |
| 2 | Lower wishbone rear pivot | 114.3 | 241.3 | 304.800 |
| 3 | Lower wishbone outer ball joint | -3.333 | 578.116 | 241.879 |
| 4 | Upper wishbone front pivot | -152.4 | 241.3 | 449.941 |
| 5 | Upper wishbone rear pivot | 114.3 | 241.3 | 449.941 |
| 6 | Upper wishbone outer ball joint | 9.999 | 507.264 | 319.964 |
| 7 | Damper wishbone end | 0 | 482.575 | 298.486 |
| 8 | Damper body end | 0 | 234.242 | 620.187 |
| 9 | Outer track rod ball joint | -50.8 | 604.310 | 309.400 |
| 10 | Inner track rod ball joint | -76.2 | 279.4 | 370.097 |
| 11 | Wheel spindle point | 0 | 571.500 | 279.400 |
| 12 | Wheel centre point | 0 | 698.500 | 279.400 |

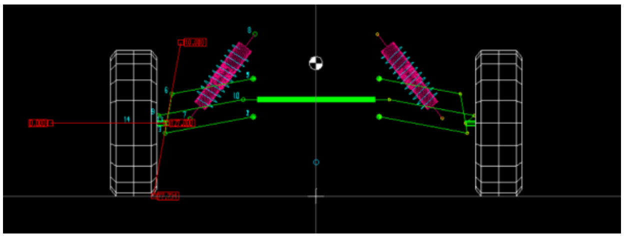

**Fig -8**: Optimized double wishbone

From the above values we constructed the base geometry for double wishbone suspension system and performed dynamic analysis. The optimization process was carried out by modifying these co-ordinates
and recording the results.

### 5.1 MODIFICATION OF KPI

By modifying the KPI values in terms of 6, 8 , 10, 12 & 14 degrees in the roll centre diagram and the corresponding changes in the co-ordinates are noted and the dynamic analysis is performed.

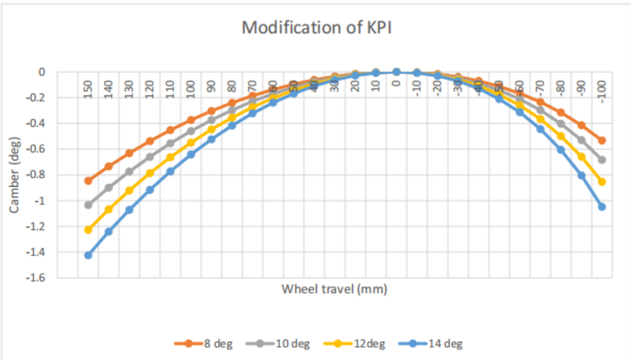


**Chart -1**: Wheel travel vs camber for modification of KPI

Thus increasing the KPI values in terms of 6, 8, 10, 12 & 14 degrees increases the camber gain. From the graph we can also conclude that this geometry produces negative camber during bump and positive camber during rebound. Negative camber during bump increase cornering effect and positive camber during rebound gives a comfortable driving experience.

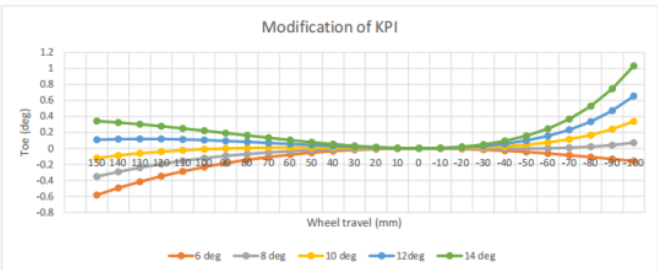


**Chart -2**: Wheel travel vs Toe for modification of KPI in Double wishbone

From the above graph we conclude that increasing KPI increases positive toe rather than increasing the overall toe gain. So in double wishbone geometry higher KPI increases straight line stability which provides better control to the driver.

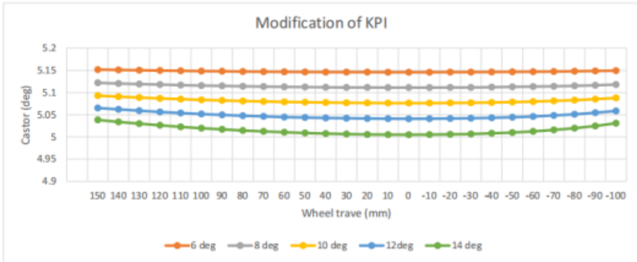


**Chart -3**: Wheel travel vs castor for modification of KPI in Double wishbone

Dynamic castor changes are very less in double wishbone geometry. Increase in KPI slightly reduces the Dynamic castor change. Its overall effect on the vehicles performance is relatively less.

### 5.2 MODIFICATION OF UCA LENGTH

The modification of the upper control arm (UCA) length from 315.721mm by either increasing or decreasing its current length, certain changes in the suspension parameters could be observed. These changes and their effects are shown below in the following charts.

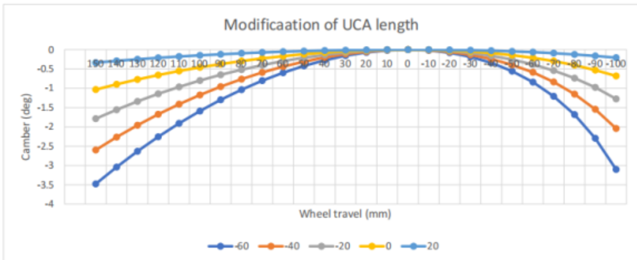


**Chart -4**: Wheel travel vs camber for modification of UCA length in Double wishbone

The above graph depicts the modification of UCA in terms of -60mm, -40mm, -20mm and 20 mm from its original length of 315.721mm and its effect on suspension parameters. It is observed that by reducing the UCA length, camber gain increases. These in turn increases negative camber during bump and rebound conditions. Negative camber provides better cornering effect.

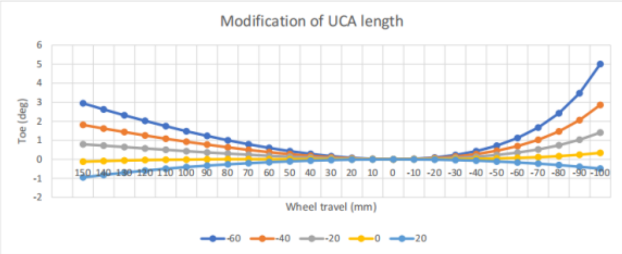


**Chart -5**: Wheel travel vs Toe for modification of UCA length in Double wishbone

By decreasing the UCA length, positive toe increases. Positive toe or toe in provides straight line stability during bump. By increasing the UCA length negative toe or toe out is increased. Negative toe results in under steering of the vehicle. The UCA length cannot exceed the length of the LCA.

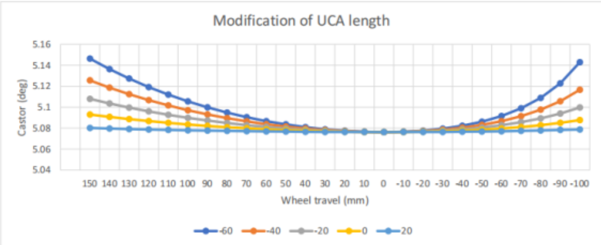


**Chart -6**: Wheel travel vs castor for modification of UCA length in Double wishbone

Here decreasing the length of the UCA increases caster gain. Positive castor creates a lot of align torque which improves straight line stability of the car. Due to the geometry of positive caster it also will increase negative camber gain when turning.

## 6. OPTIMIZATION OF H-ARM SUSPENSION

The H-arm geometry considered is assumed to have control arms of equal lengths and parallel mountings. The analysis is performed by adjusting the length of the camber link. In this geometry the camber link length is always less than or equal to that of the H-arm.

**Table -3:** H-arm static values

| S.No | Suspension parameters | Values |
|---|---|---|
| 1 | Toe | 0 deg |
| 2 | camber | 0 deg |
| 3 | Track width | 1320.8mm |
| 4 | Wheel base | 1473.2mm |
| 5 | Bump | 150mm |
| 6 | Rebound | 100mm |

**Table -4:** H-arm co-ordinates

| S.No | POINT NAME | X(mm) | Y(mm) | Z(mm) |
|---|---|---|---|---|
| 1 | Lower wishbone front pivot | -152.4 | 165.10 | 304.8 |
| 2 | Lower wishbone rear pivot | 101.6 | 165.10 | 304.8 |
| 3 | Lower wishbone outer front pivot | -50.8 | 632.6 | 209.55 |
| 4 | Lower wishbone outer rear pivot | 50.8 | 632.6 | 209.55 |
| 5 | Upper link inner ball joint | 152.4 | 165.10 | 444.5 |
| 6 | Upper link outer ball joint | 0 | 632.6 | 349.25 |
| 7 | Damper wishbone end | -50.8 | 495.369 | 276.893 |
| 8 | Damper body end | -50.8 | 362.331 | 660.40 |
| 9 | Wheel spindle point | 0 | 632.6 | 279.4 |
| 10 | Wheel centre point | 0 | 660.4 | 279.4 |

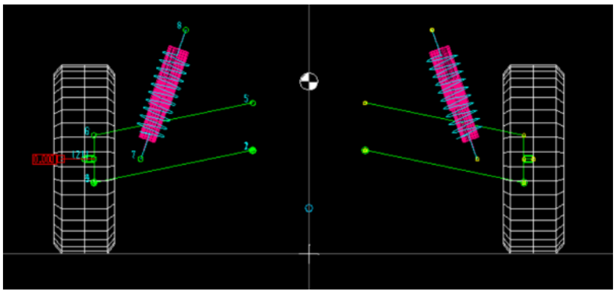

**Fig -9**: Optimized H-arm suspension

From the above values we constructed the base geometry for H-arm suspension system and performed dynamic analysis. The optimization process was carried out by modifying these co-ordinates and recording the results.

### 6.1 MODIFICATION OF CAMBER LINK LENGTH

The modification of the camber link length from 477.105mm by decreasing its current length, certain changes in the suspension parameters could be observed. These changes are depicted in the graphs below.

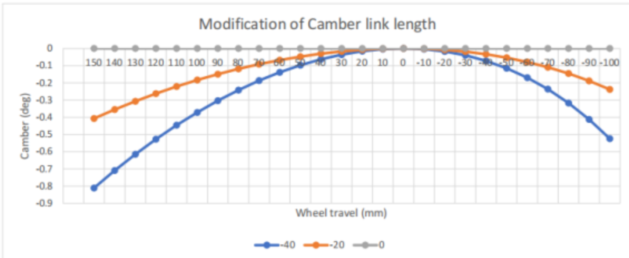


**Chart -7**: Wheel travel vs camber for modification of camber link length in H-arm

The camber link is decreased by 40 mm and 20 mm from its original length. It is observed that by decreasing the camber link length the camber gain is increased. This enables greater negative camber during both jounce and rebound. Negative camber in the rear wheel greatly increases the cornering ability of the vehicle.

## 6.2 TYPE OF LENGTH REDUCTION

The length of the camber link can be modified in three ways

- By reducing the length at the wheel end.
- By reducing the length at the frame mounts.
- By reducing length equally at both ends.

By these three methods the analysis was performed and the obtained values are depicted below in the form of the graph [5]. Despite the length of the camber link, the method of modification plays an important role in the dynamic analysis.

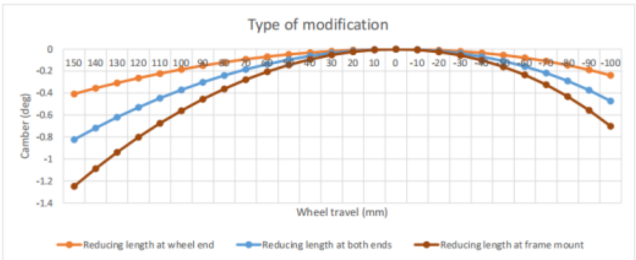


**Chart -8**: Wheel travel vs camber for various types of modification in H-arm

Thus it is observed that by modifying the camber link length at the wheel end, the corresponding camber gain is relatively low. Whereas when the length is reduced at the frame mount the camber gain values increases. It is also noted that when the adjustment is done equally at both ends the camber gain values are maintained at a nominal level.

## 7. CONCLUSION

In Double wishbone suspension, it is observed that by increasing KPI, the Camber and Toe increases and slightly decreases castor. Which shows that KPI is directly proportional to camber and Toe, but inversely proportional to castor. It is also observed that, by decreasing the UCA length, Camber gain, Toe and Castor increases. This shows that UCA length is inversely proportional to Camber gain, Toe and Castor In H-arm suspension system, it is observed that by decreasing the Camber link length, Camber gain increases. This shows that Camber link length is inversely proportional to Camber gain. It is also observed that Maximum Camber gain is obtained when Camber link length is reduced at the mount. It can also be concluded that Length of the Camber link is always less than or equal to the Length of Arm. Therefore it can be considered that the optimized set of values will render a very comfortable ride as far as normal roads are considered and optimum performance in the case of off-road conditions.

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