



# Design and Kinematics Analysis of Suspension System for a Formula Society of Automotive Engineers (FSAE) Car

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## Authors' contributions

*This work was carried out in collaboration among all authors. All authors read and approved the final manuscript.*

## Article Information

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## ABSTRACT

The main objective of the Suspension of a vehicle is to maximize the contact between the vehicle tires and the road surface, provide steering stability and provide safe vehicle control in all conditions, evenly support the weight of the vehicle, transfer the loads to springs, and guaranteeing the comfort of the driver by absorbing and dampening shock. This paper discusses the kinematic design of a double a-arm Suspension system for an FSAE Vehicle. The hardpoint's location can be determined using this procedure to simulate motion in any kinematic simulation software. Here, Optimum Kinematics is used as kinematic simulation software, and the results are verified using Msc Adams simulation. The method illustrated deals with the basics of Kinematics which helps to predict the characteristics of the Suspension even before simulating it in the kinematic simulation software.

**Keywords:** *Suspension modeling; suspension analysis; kinematic simulation; optimum kinematics; double a-arm suspension system.*

## 1. INTRODUCTION

The kinematic performance of the vehicle depends on the location and orientation of the A-arms. [1] Therefore, it helps to determine the variation of suspension parameters like camber, toe, caster, kingpin inclination, to name a few. [2,3]. The range of the parameters mentioned above is dependent on various factors such as tire used, loads on the tire, slip angle of the tire. [4] For this paper, Hoosier 43070 tire with DWT 10" x 6.0" rims were considered. However, for any tire and rim combination, the procedure illustrated can be used.

A designer can significantly simplify the suspension design by considering the plane formed by the A-arm points rather than the points themselves [5]. Considering planes also has an added advantage of being able to vary a-arm angles, chassis hardpoints freely without significantly changing the kinematic of the Suspension [6,7].

The assumption made by optimum kinematics software is that all joints are spherical joints without any friction in-between. A kinematic-based optimization is performed in Adams, which includes Parallel and Opposite wheel tests of a half vehicle. All the components are rigid, and the

springs have constant spring stiffness and perfect Kinematics & no compliances [8,9].

## 2. METHOD FOR DETERMINATION OF SUSPENSION HARDPOINTS

3 points are needed to create any plane to decide the three crucial points for the A-arm plane. A body in 2D motion has an instantaneous center; similarly, a body in 3D motion body has an instantaneous axis. So, to make an axis, two points are needed; the two points selected are the front view instantaneous center and the side view of the instantaneous center. The line joining these two points is the instant axis. As for the third point, the upper ball joint of the Suspension upright to get the upper control arm plane and the lower ball joint for, the lower control arm plane can be selected. The step-by-step procedure for hardpoint determination is illustrated below [10,11].

### 2.1 The Decision of Planes

The front view plane is selected such that it is perpendicular to the ground with the normal along the car travel direction and coincides with the axel of the wheel (it may be front or back). The side view plane is selected such that the plane is perpendicular to the ground and front view plane and coincides with the wheel center.

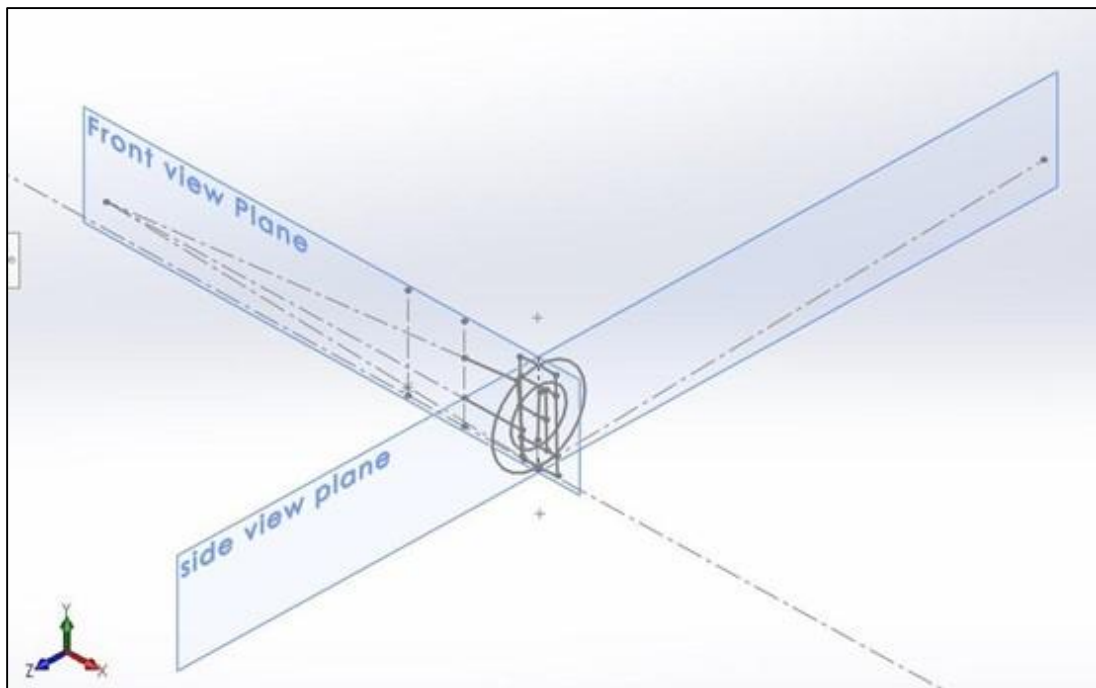


Fig. 1. 2D line representation

## 2.2 Deciding the Front view Instant Center

The location of the front view swing arm center decides the camber change rate, roll center, and lateral tire scrub based on its location w.r.t to ground and tire. The designer chooses the location front view IC as per their requirements.

The camber change rate in heave motion is proportional to the arctan of the inverse of the front view swing arm (FVSA) length.

$$\text{camber change rate} \left( \frac{\text{deg}}{\text{mm}} \right) = \tan^{-1} \left( \frac{1}{\text{front view swing arm length}} \right) \quad [6]$$

Suppose the car is rolling longer the FVSA length, the larger the camber gain. For example, if the FVSA length is infinite and the car's roll angle is 3°, then the camber gain is also 3°, so a decision must be taken based on the designer's requirements. For the design, 1920 mm is selected.

The intersection of lines joining the FVSA instant center, the tire's contact patch center, and the symmetric line is the Roll center. It is the lateral force coupling point for sprung and unsprung mass. The force acting on the center of gravity

can be transferred to the roll center by a matching pair of force and moment. The roll moment is inversely proportional to the roll-center height. So, the lower the roll center, the higher the rolling moment of the roll center. Roll center height of 33.91mm was chosen for the design to prevent the vehicle's jacking during tight cornering.

The roll height to be above the ground was selected. This results in the FVSA instant center being above the ground resulting in scrub out during wheel travel, i.e., increases the track width resulting in better stability. And so that the wheel path does not vary excessively on extreme turns.

## 2.3 Deciding Lower Ball Joint and Upper Ball Joint Location in Front View

The KPI and scrub radius were decided based on the steering effort and so that the ball joints are outside of the rim, so more space is obtained while designing uprights. The final values that have been decided on are 7° KPI and 55 mm scrub radius. The distance between ball joints is assumed as 190 mm to prevent interference with the wheel's rim based on the construction in Fig. 2. So, an upper and lower ball joint x, y location input for Optimum kinematics software was obtained.

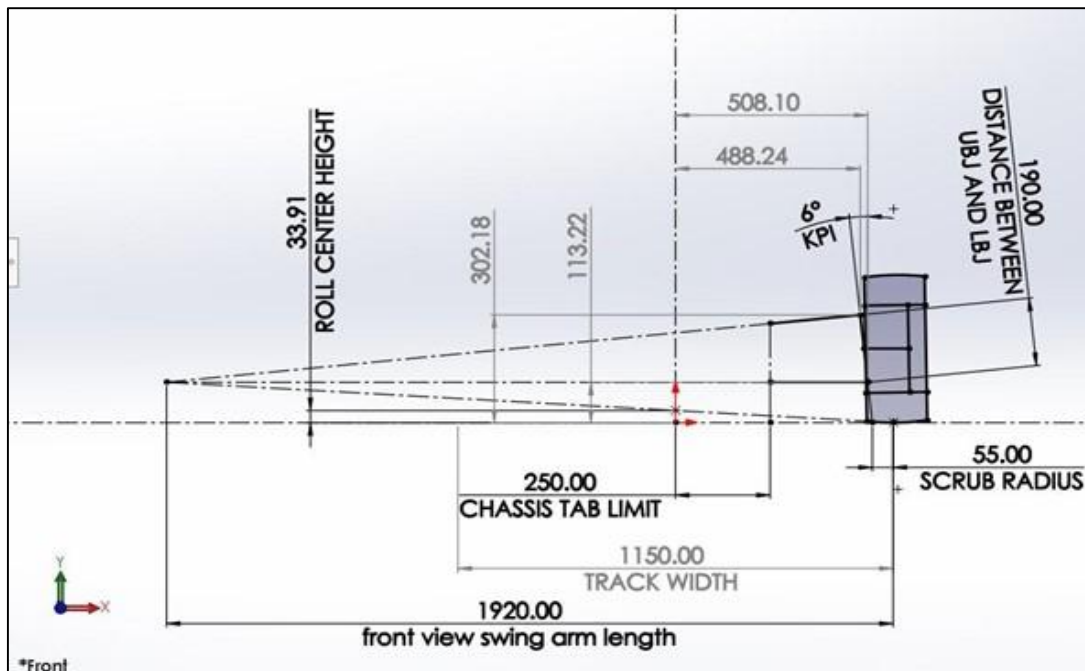


Fig. 2. Front view swing arm line diagram

### 2.4 Selection of Track Width

The track width of 1150mm was selected based on the steering and chosen to achieve the highest possible wishbone lengths, which directly affect the vertical travel of the Suspension. In addition, having a large amount of suspension travel was preferred as this would give flexibility with the actuation setup.

### 2.5 Selection of Chassis Tab Limit

The chassis tab limit was selected purely based on the chassis dimension [12].

### 2.6 Deciding Lower Ball Joint and Upper Ball Joint Location in Side View

UBJ, LBJ must be at the same y height to maintain the KPI in the front view. Finally, the line joining UBJ and LBJ is given a caster angle and mechanical trail to get the final z location of the ball joints.

### 2.7 Deciding the Side View Instant Center

The side view swing arm (SVSA) controls the anti-lift, anti-dive, anti-squat, caster change rate, and wheel path.

$$\%Anti - dive = \frac{\tan(\text{front side view swing arm angle})}{\frac{CG \text{ height}}{\text{wheel base}}} \quad [6]$$

$$\%Anti - squat = \frac{\tan(\text{back side view swing arm angle})}{\frac{CG \text{ height}}{\text{wheel base}}} \quad [6]$$

The percentage of anti-determines the amount of load transferred to the A-arms. for example, if the anti is 100%, the A-arm resists the entire longitudinal load transfer. Therefore, the springs do not take any transferred force. Thus, no suspension deflection occurs.

The selected length of SVSA is to prevent the change of caster during the acceleration and deceleration in cornering as any change increase in caster during this phase results in additional effort from the driver to maintain the car on the optimal path.

### 2.8 Creating Control Planes

The upper and lower control arm planes can be created with ball joints, front and side view instant centers. Below is the figure showing the creation of the upper control arm plane.

### 2.9 Determination of Hardpoints

As the plane is created, lines can be drawn representing each arm of the A-arm, with both lines originating from the ball joint. The below figure contains an example of this. As the cartesian coordinates of hardpoints are found, these hardpoints can be inputted into the kinematic simulation software to simulate the kinematic motion of the Suspension [12].

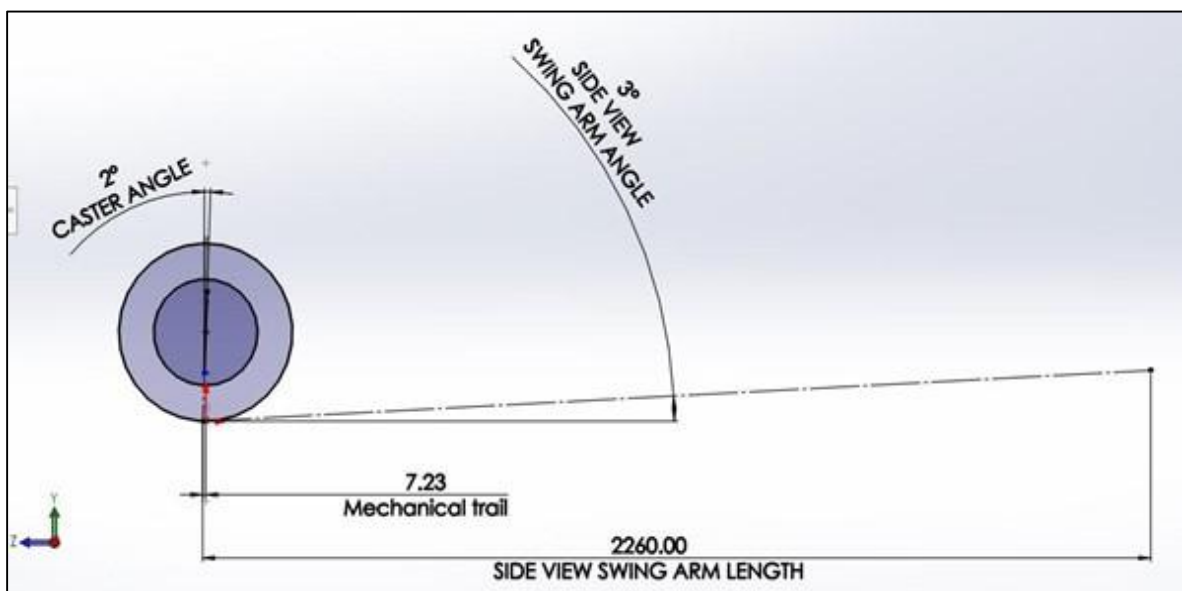


Fig. 3. Side view swing arm line diagram

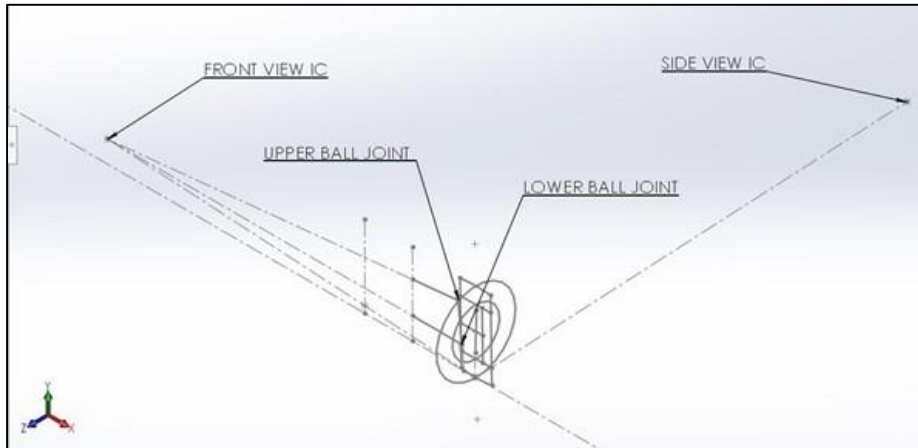


Fig. 4. 3D view of Instantaneous centres and ball joints

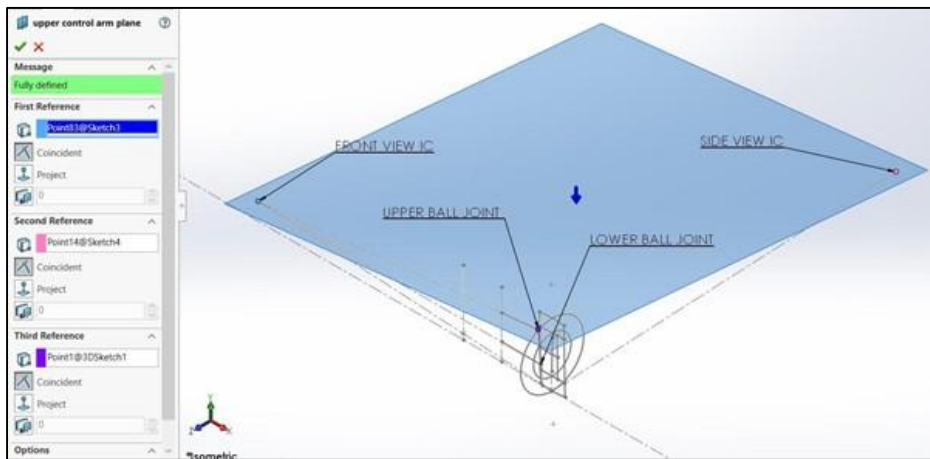


Fig. 5. Plane creation from IC's and ball joints

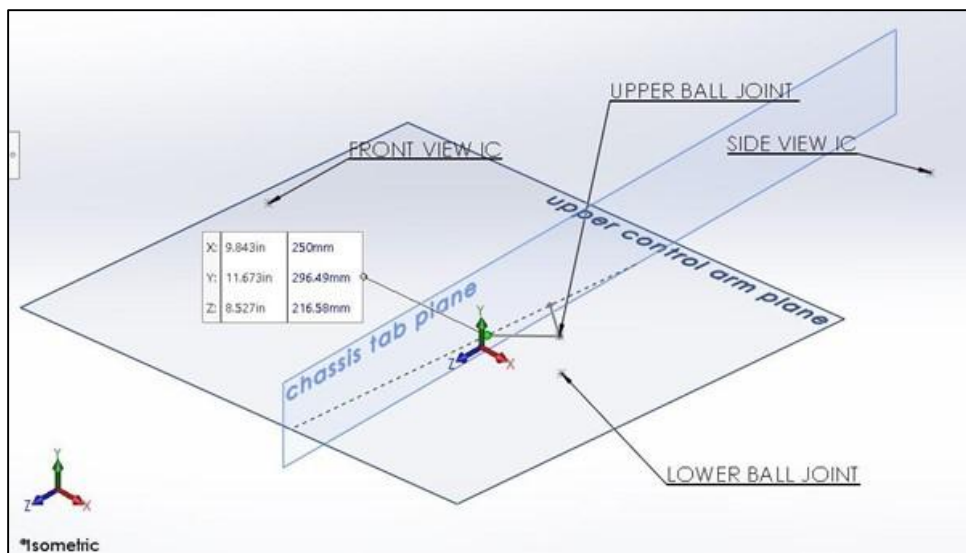


Fig. 6. A-arm line diagrams

### 3. KINEMATIC SIMULATION

Optimum Kinematics, developed by OptimumG, is a suspension simulation software. It is specifically designed with a user-friendly interface to make the process of suspension design, analysis much faster and more convenient. Optimum Kinematics application. It is a unique approach toward 2D/3D designing, and with its numerous features and variety of tools, it is easy for users to organize and boost their workflow.

#### 3.1 Advantages and Benefits

Optimum Kinematics is used by designers and analysts alike to layout the suspension hardpoint locations to achieve the required kinematic behaviour. Several results can be displayed graphically, like Camber and Toe angles, against various motions like a bump, roll, pitch, and steering motion, to name a few. These results are updated in 'real time' as the suspension hardpoints are updated.

#### 3.2 Procedure

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#### 3.2.1 Vehicle axis system

The coordinate system is a right-handed system, the origin of which must be in the car's front axle and coincide with the vehicle's longitudinal centerline and ground plane.

- The X-axis is along the lateral direction of the vehicle and positive toward the left of the car.
- Y-axis is along the vertical direction and positive upwards.
- Z-axis is along with the vehicle and positive in the forward direction.

#### 3.2.2 Sign convention

Standard SAE sign convention is used for all other parameters like camber, caster, Kingpin angle, to name a few.

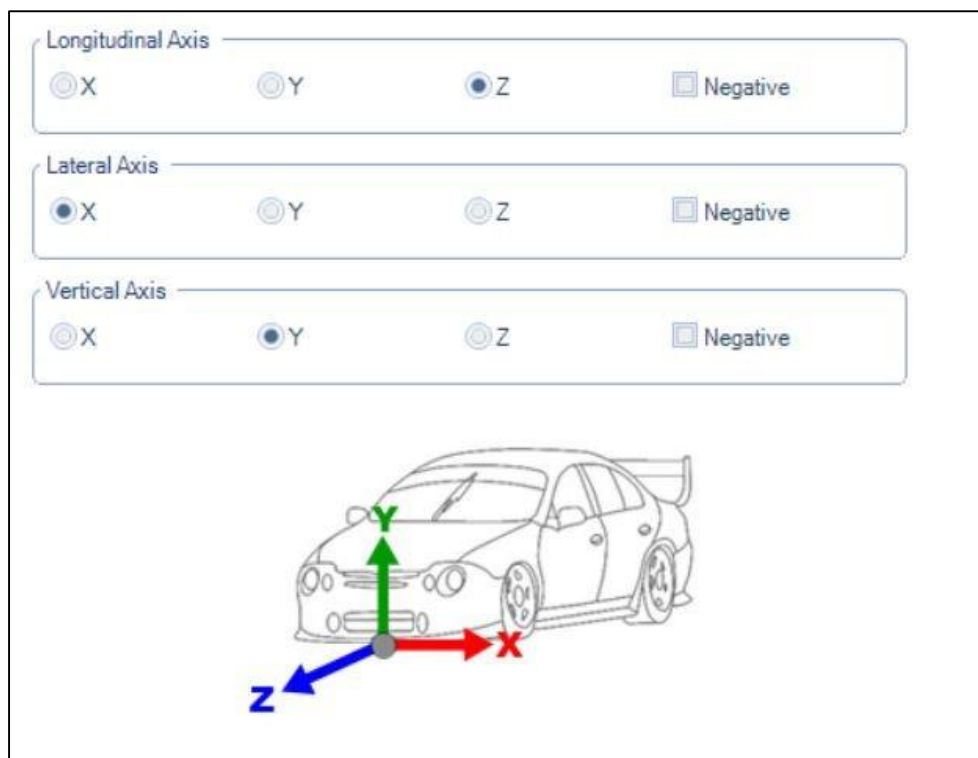


Fig. 7. Vehicle axis system in Optimum Kinematics

### 3.2.3 Vehicle data

**Table 1. Vehicle data**

Property	Symbol	Value
Acceleration due to gravity	g	9.81 m/s <sup>2</sup>
Sprung mass of Car	Ms	110 kg
Unsprung mass of the vehicle	Mu	40 kg
Mass of driver	Md	70 kg
Wheelbase	l	1540 mm
Track width(F/R)	t	1150 mm
CG x location	Z"	-4.11 mm (neglected)
CG y location	h	245.44 mm
CG z location	a	808.31 mm
Turn radius	R	5 m
Turn speed	V	45 kmph or 12.5 m/s
acceleration	Aa	0.8g
Deceleration	Ad	1.5g
Spring travel	ST	35 mm(F) and 30 mm(R)
Spring constant	K	78.88 N/mm
Coefficient of friction	$\mu$	0.5
Braking torque front	Tf	118.3 N-m
Braking torque rear	Tr	72.8 N-m
Radius of wheel	R	207.1 mm

## 4. RESULTS AND DISCUSSION

### 4.1 Combine

**Table 2. Combine vehicular data**

Parameter	Value	Unit
Wheelbase	1540	mm
Track width	1150	mm
Kinematic pitch center X	697.99	mm
Kinematic pitch center Y	575	mm
Kinematic pitch center Z	35.51	mm
Roll center height front	34.43	mm
Roll center height rear	47.75	mm
Kinematic roll axis inclination	0.5	deg

### 4.2 Front

**Table 3. Front Wheel Data**

Parameter	Value	Unit
Camber Angle [Left]	-1	deg
Caster Angle [Left]	2.01	deg
King Pin Angle [Left]	6	deg
Wheel Center Z [Left]	206.98	mm
Wheel Center Y [Left]	-571.39	mm
Wheel Center X [Left]	0	mm
Front View Swing Arm Angle [Left]	3.43	deg
Side View Swing Arm Length [Front Right]	2,261.72	mm
Side View Swing Arm Angle [Front Right]	2.91	deg
Front View Swing Arm Length [Left]	1,917.10	mm
King Pin Angle [Right]	6	deg
Toe Angle [Left]	-2	deg
Scrub Radius [Front Left]	54.99	mm

### 4.3 Rear

**Table 4. Rear Wheel Data**

Parameter	Value	Unit
Camber Angle [Left]	-1	deg
Caster Angle [Left]	2.01	deg
King Pin Angle [Left]	6	deg
Wheel Center Z [Left]	206.98	mm
Wheel Center Y [Left]	-571.39	mm
Wheel Center X [Left]	0	mm
Front View Swing Arm Angle [Left]	4.75	deg
Side View Swing Arm Length [Front Right]	2719.63	mm
Side View Swing Arm Angle [Front Right]	2.41	deg
Front View Swing Arm Length [Left]	1384.58	mm
King Pin Angle [Right]	5	deg
Toe Angle [Left]	0	deg
Scrub Radius [Front Left]	70	mm

### 4.4 Parallel wheel Travel v/s Geometry Changes

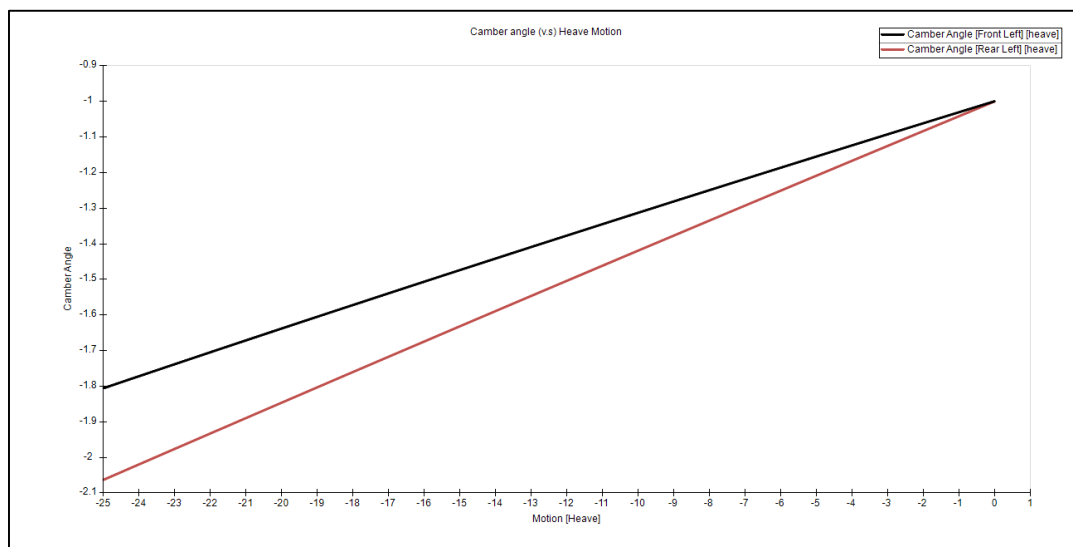
Due to the Suspension's symmetric nature, graphs are provided only for the left wheel to show the geometry variation [13].

The camber angle varies towards the negative values when there is heave motion, which helps during the vehicle's cornering. The camber angle is always below zero. If camber is positive, it results in wobbly movement of the vehicle

As the wheels move upwards, the KPI value is reducing this helps reduce the vehicle jacking during bump and cornering. The KPI has a maximum variation of 0.75° in front and 1.3° in the rear.

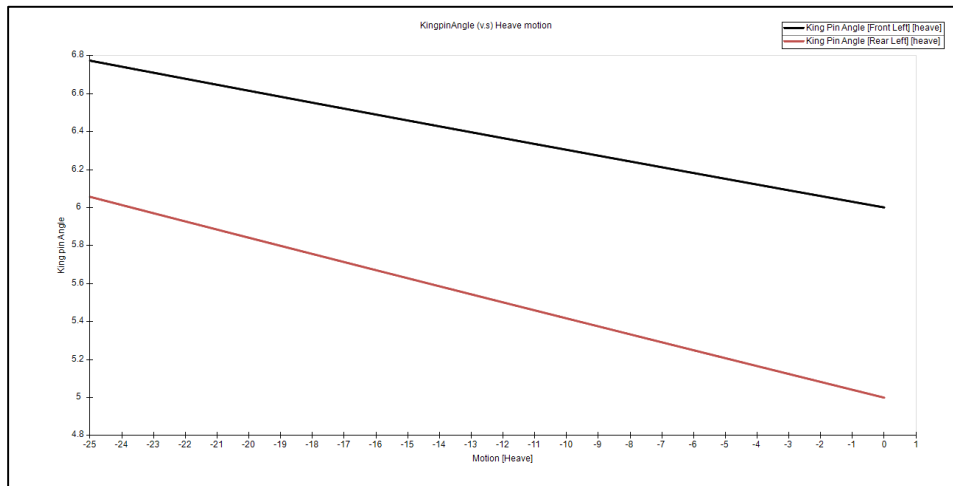
It is observed that the caster angle in the front rises from 2° to 2.65°. due to this, the self-aligning force acting wheel increases, resulting in better stability. The rear caster varies from 2° to 1.45° degrees as this helps in reducing the opposing forces on the rear wheel during acceleration of the vehicle and also induce stability.

The toe angles during parallel wheel travel are consistent with little to no change and a max variation of 0.12° in the front and 0.10° in the rear. As variation is minimal, it prevents self-steering of the vehicle when it experiences a bump or heavy acceleration and deceleration.

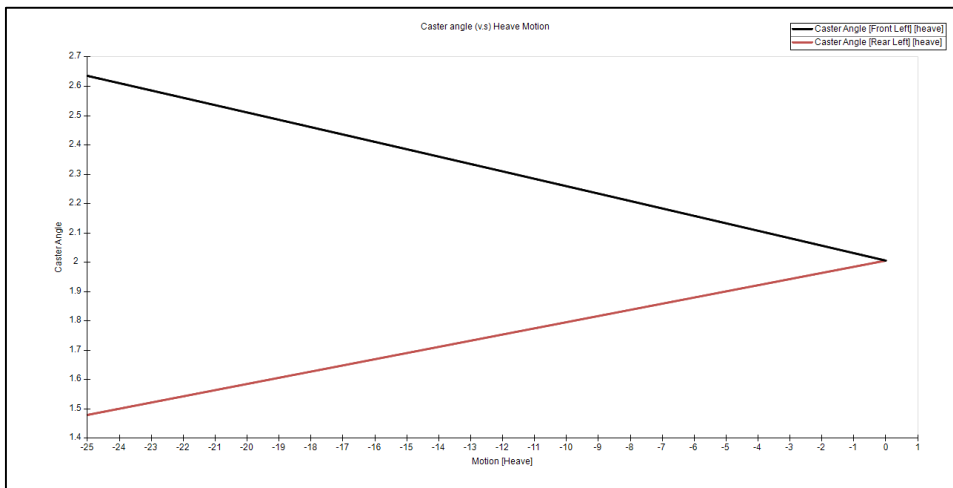


**Fig. 8. Camber angle vs. Heave Motion**

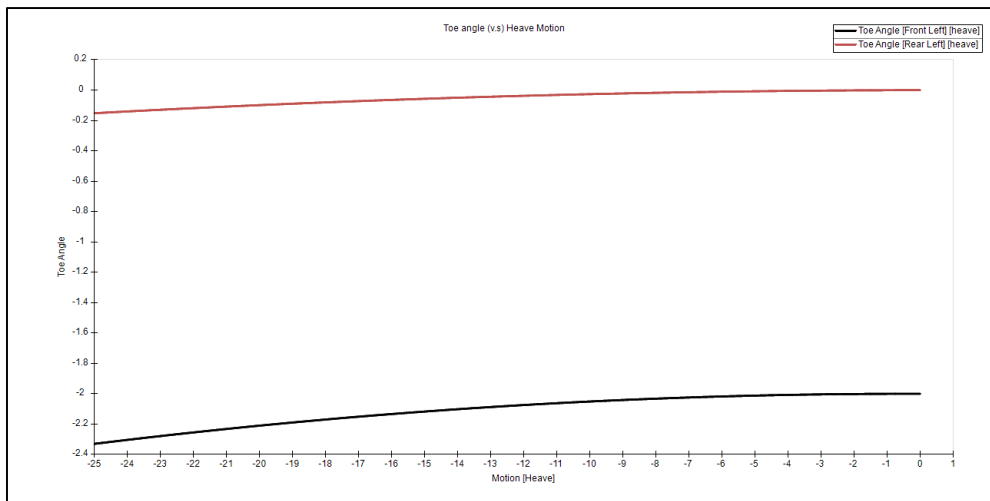




**Fig. 9. Kingpin Angle vs. Heave motion**



**Fig. 10. Caster angle vs. Heave motion**



**Fig. 11. Toe angle vs. Heave motion**

### 4.5 Roll v/s Geometry Changes

The inner and outer wheel would not have equal grip during the rolling condition. Therefore, the inner wheels have a lower grip force compared to the outer wheels. So the parameters must be maintained only for outer wheels.

The caster variation is within the max variation limits of under  $0.5^\circ$ , so it wouldn't cause any issue while racing.

The increase in KPI may result in increasing the steering effort, but the variation is under or equal to  $1^\circ$ , so it wouldn't be a problem.

The toe angle variation is almost negligible as the variation is under  $0.5^\circ$ . It wouldn't

produce self-steering effects as the toe angle is so less.

### 4.6 Pitch v/s Geometry Changes

Pitching motion is when the front of the car dips and the rear of the car raises. Due to the symmetric nature of the Suspension, graphs are provided only for the left wheel to show the geometry variation.

The camber angle variation of the front wheel is  $1^\circ$ . Therefore, even though the rear wheel camber tends toward positive camber, it wouldn't be an issue as it is close to zero, and the positive camber is on the rear wheel, which would have lower loading due to forward pitching.

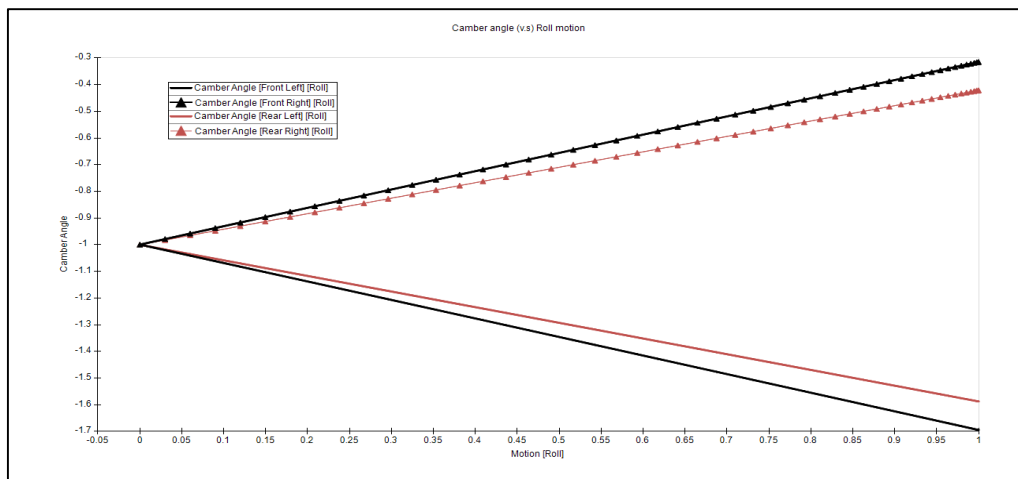


Fig. 12. Camber angle vs. Roll motion

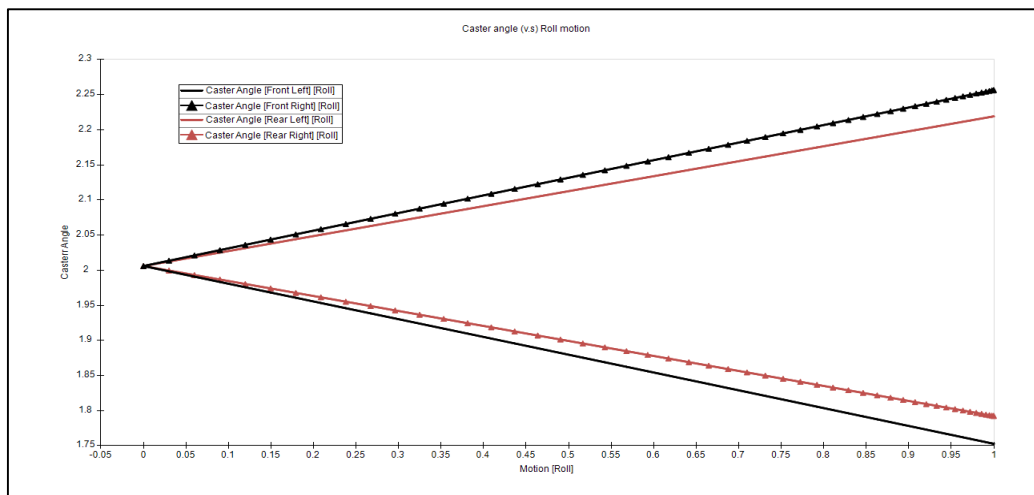
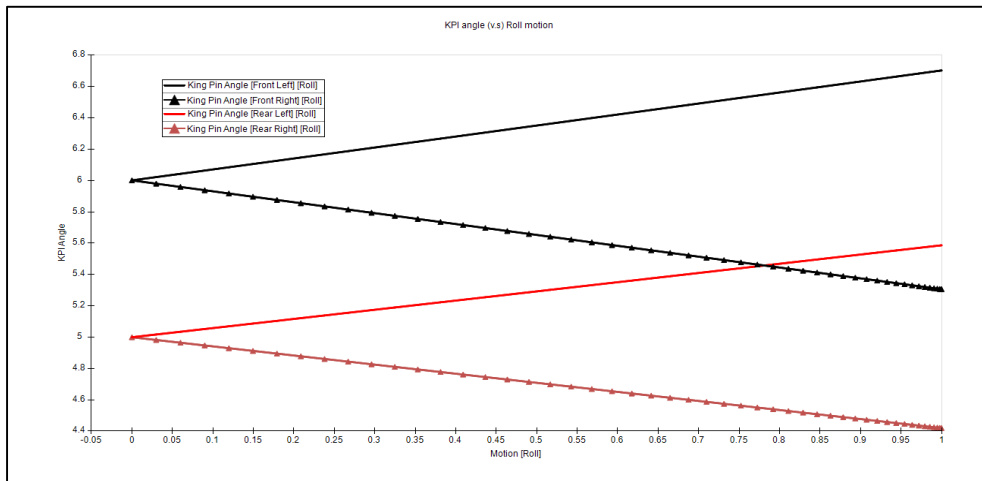
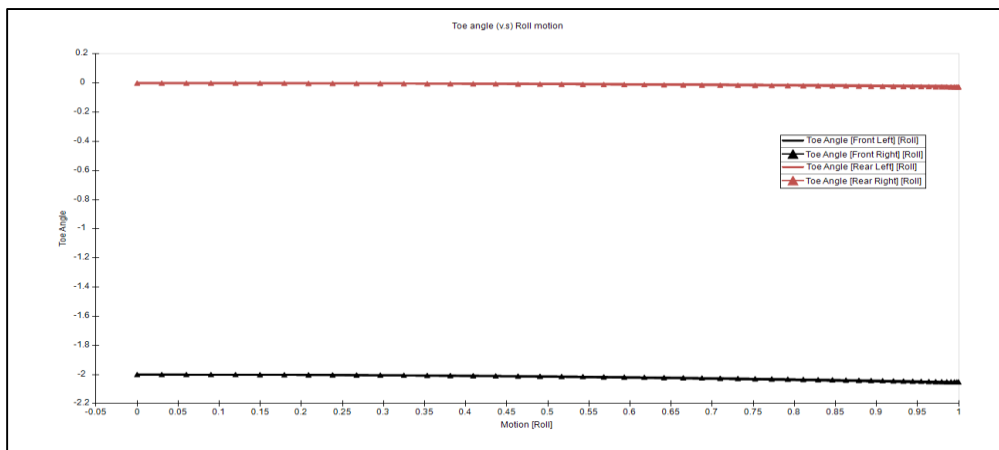


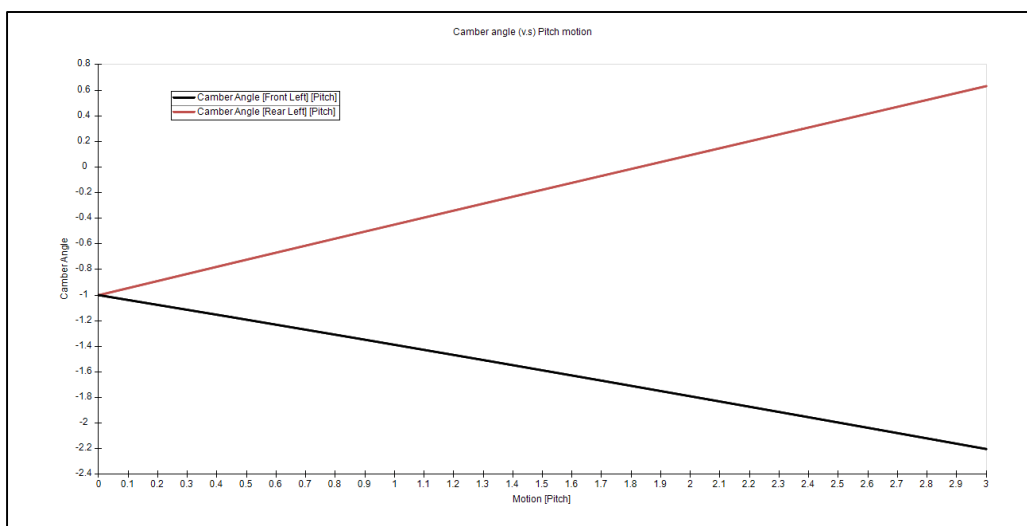
Fig. 13. Caster angle vs. Roll motion



**Fig. 14. Kingpin angle vs. Roll motion**



**Fig. 15. Toe angle vs. Roll motion**



**Fig. 16. Camber angle vs. Pitch motion**

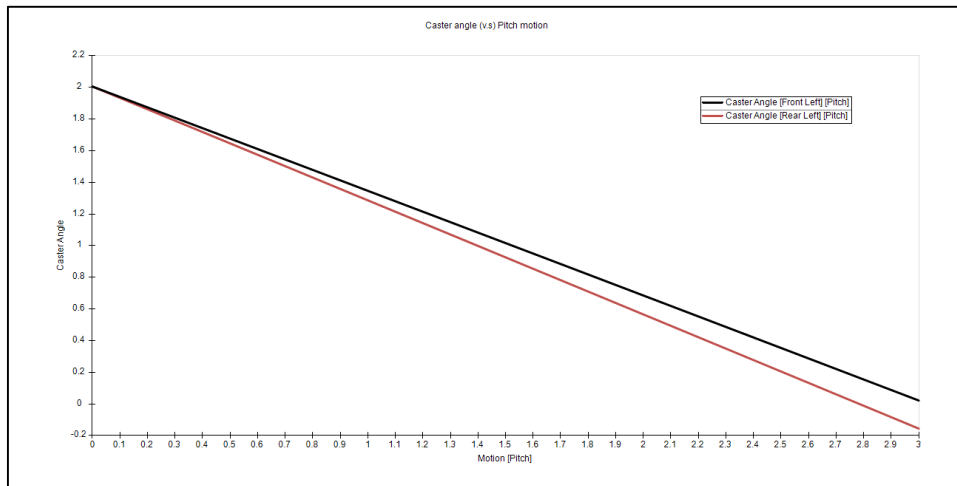


Fig. 17. Caster angle vs. Pitch motion

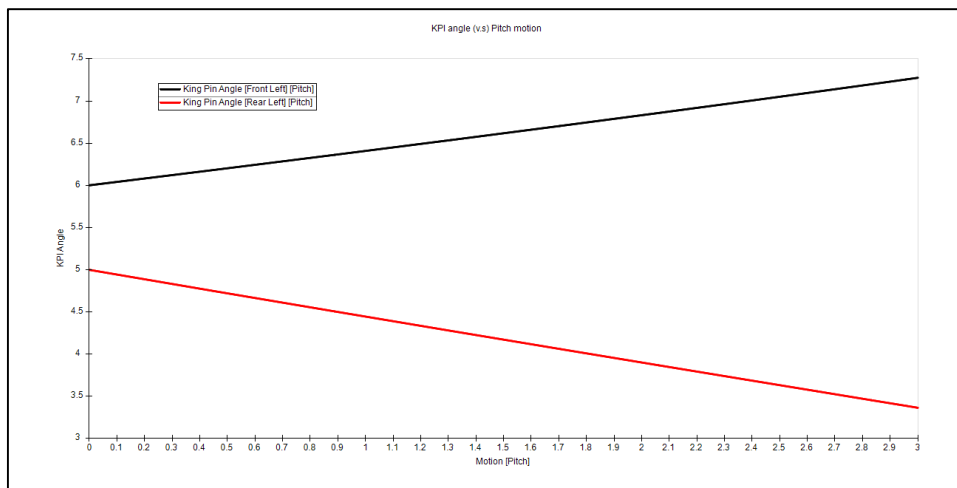


Fig. 18. KPI angle vs. Pitch motion

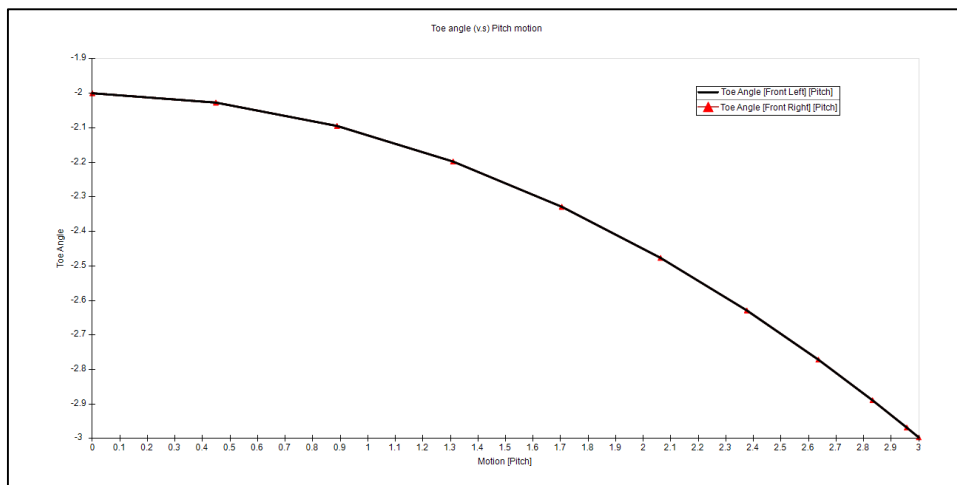


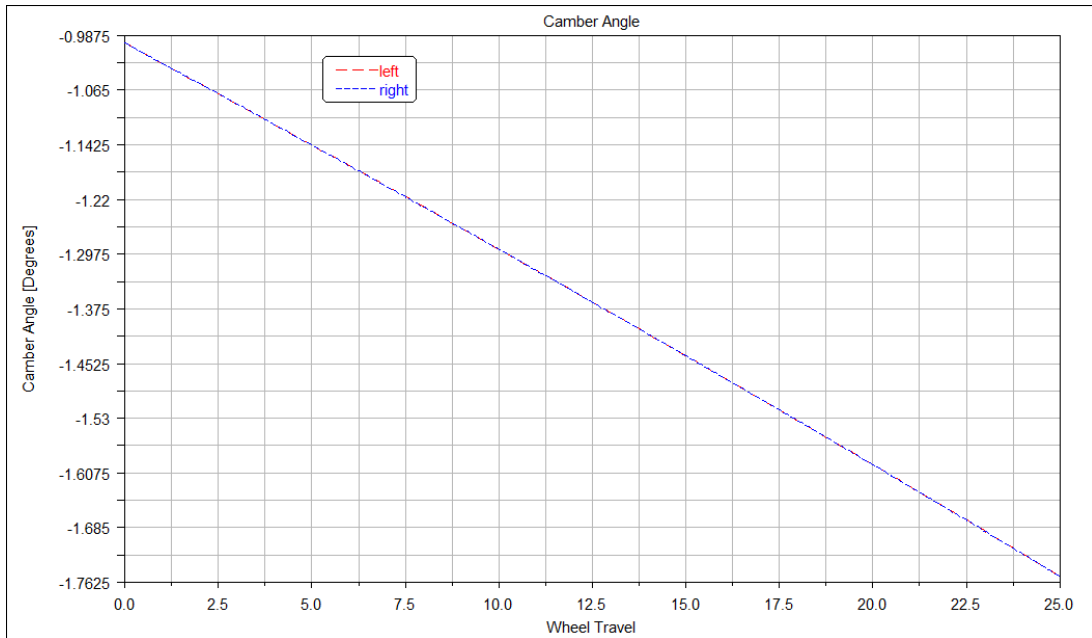
Fig. 19. Toe angle vs. Pitch motion

The caster angle on the front tends to zero; as a result, it reduces the self-centering effect. The car may move in wobbly motion at extreme pitching in the rear due to the positive caster.

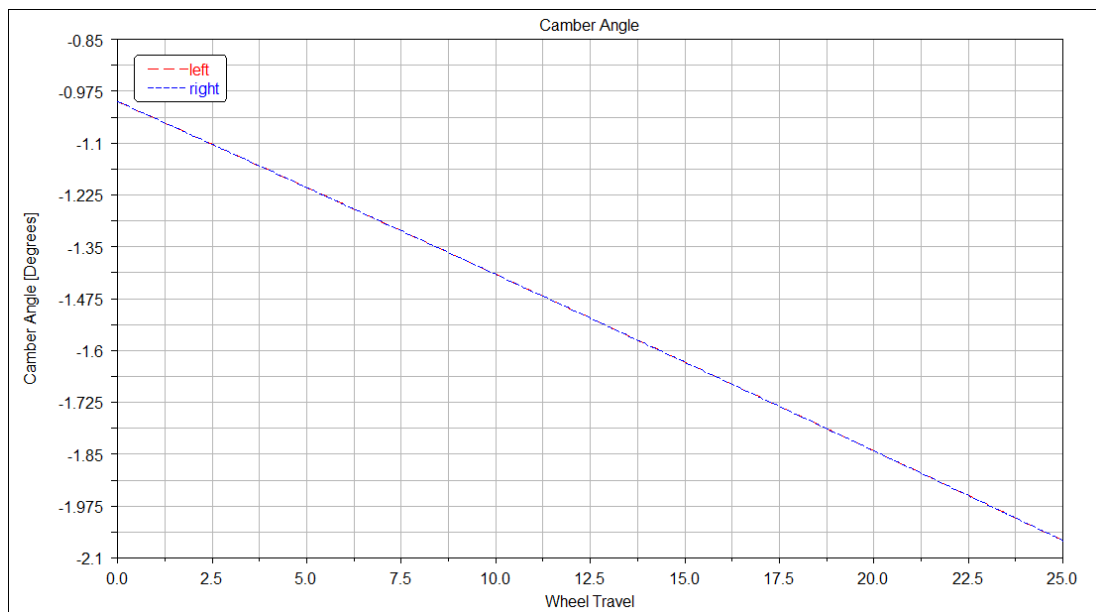
the scrubbing would decrease, resulting in lower steering effort. The KPI has a maximum variation of 2° in front and 1.5° in the rear.

As the KPI value reduces in the back, it helps reduce the vehicle jacking during a bump. But in the front, as there is an increase in the KPI value,

The front and rear wheel toe angle variations coincide as the wheels are symmetrically moving in opposite directions. As a result, both wheels have a variation of 1° inwards, resulting in higher vehicle stability during braking.



**Fig. 20. Wheel Travel vs. Camber Angle (Front) (In Adams)**



**Fig. 21. Wheel Travel vs. Camber Angle (Rear) (In Adams)**

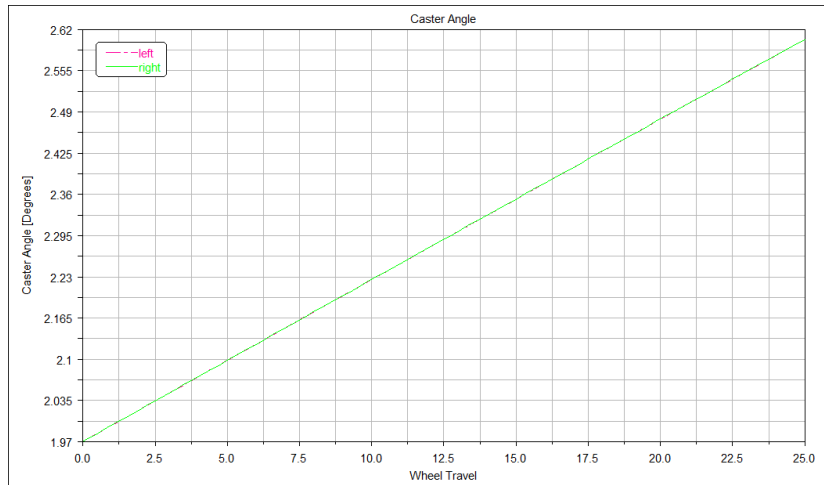


Fig. 22. Wheel Travel vs. Castor Angle (Front) (In Adams)

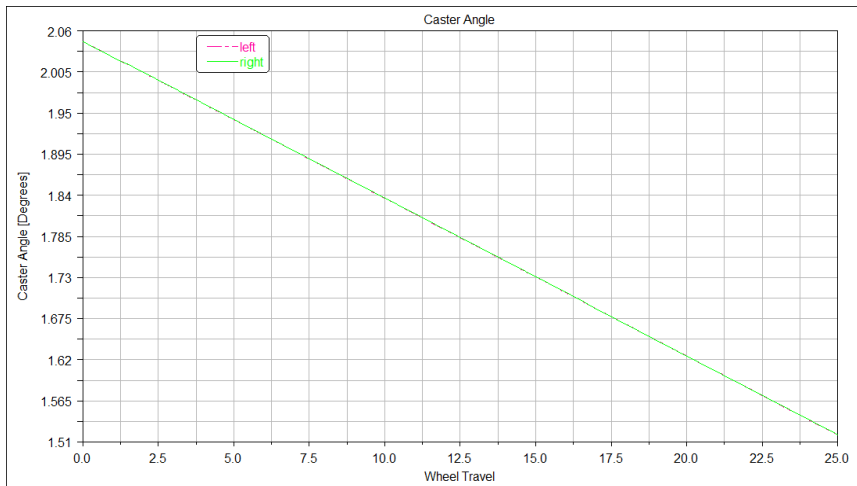


Fig. 23. Wheel Travel vs. Castor Angle (Rear) (In Adams)

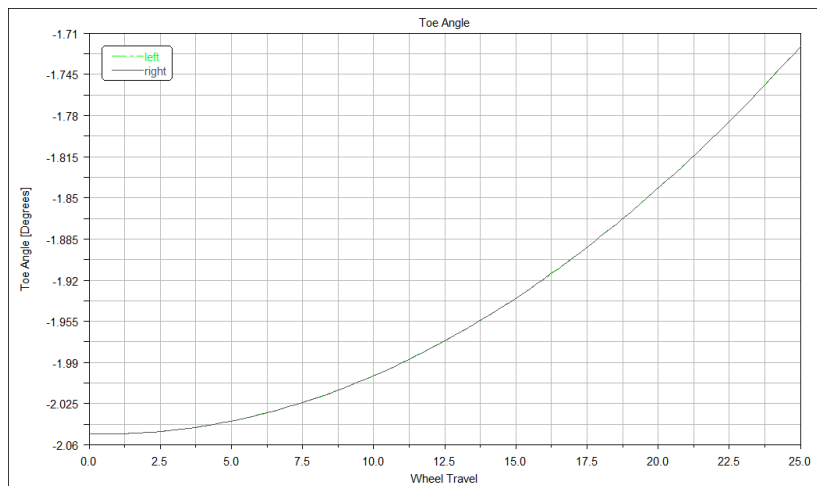
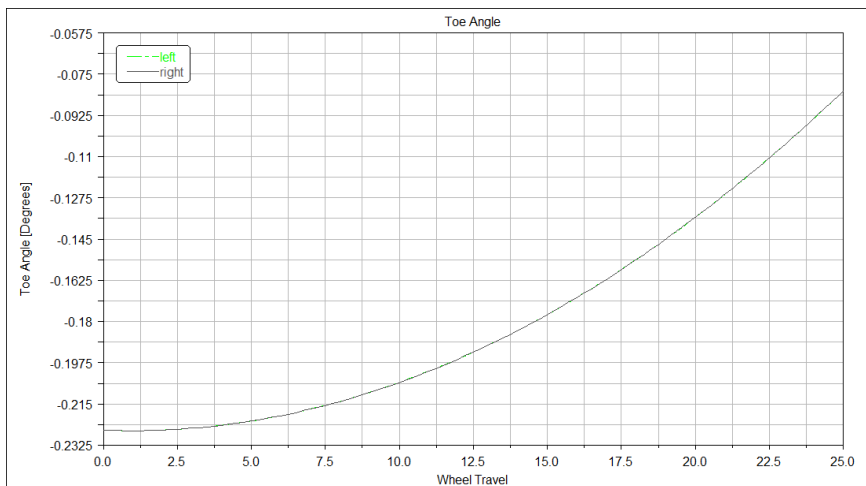


Fig. 24. Wheel Travel vs. Toe Angle (Front) (In Adams)



**Fig. 25. Wheel Travel vs. Toe Angle (Rear) (In Adams)**

## 5. VERIFICATIONS

We have modeled the same Suspension in Adams [14], performed a similar analysis, and obtained plots for the verification. We have used Camber, Caster, and Toe plots for verification as they significantly impact the vehicle performance.

### 5.1 Parallel Wheel Test

As we can see, the camber angle variation is similar to that of what we obtained in the Optimum kinematics software, with variation in front being  $0.1^\circ$  and in the rear being  $0.3^\circ$

The caster angle variation is similar to what we obtained in the Optimum kinematics software, with variation in front being  $0.08^\circ$  and in the rear being  $1^\circ$ . Thus, the higher caster angle at the rear would increase the required steering effort application. But this is not an issue as the driver does not control the rear wheels.

The toe angle variation in the front is  $0.55^\circ$  and in the rear is  $0.2^\circ$ . The values are almost similar to those we have obtained in Optimum Kinematics.

After running the similar Suspension in Adams, we can observe that our values are almost similar. So Similar results would be obtained for roll and pitching motion analysis. Thus, our Suspension should behave similarly even in the real world.

## 6. CONCLUSIONS

Hence, the project of analyzing the double-wishbone suspension system has been

systematically executed. During the literature survey, the type of suspension system and the actuation have been thoughtfully chosen. Furthermore, parametric modeling of suspension geometry is done, which is helpful while designing the chassis and associated components. Verification of the designed Suspension is done using Adams

- The variation on camber angle in all motion conditions is maintained under  $1.2^\circ$
- The variation on caster angle in all motion conditions is maintained under  $2^\circ$
- The variation on toe angle in all motion conditions is maintained under  $1^\circ$

Although the successful modeling suspension system is done, there are a few limitations of the work, as described below:

- An anti-roll bar can be incorporated to have better control over the rolling characteristics of the vehicle.
- A model can be made considering the stiffness of components to obtain a complete understanding of the vehicle.
- Dynamic analysis can test the vehicle characteristics in various tests such as skid pad, acceleration, braking, to name a few.

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any litigation but for the advancement of knowledge. Also, the research was not funded by the producing company rather it was funded by personal efforts of the authors.

### COMPETING INTERESTS

Authors have declared that no competing interests exist.

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