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Design and Kinematics Analysis of Suspension System for ABAY Vehicle

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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 ABAY 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. 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.
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Keywords: Suspension modeling, Suspension analysis, kinematic simulation, Optimum Kinematics, double A-arm suspension system.

1. 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. All the components are rigid, and the springs have constant spring stiffness and perfect Kinematics & no compliances.[8][9].

Method for Determination of Suspension Hard Points

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.

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.

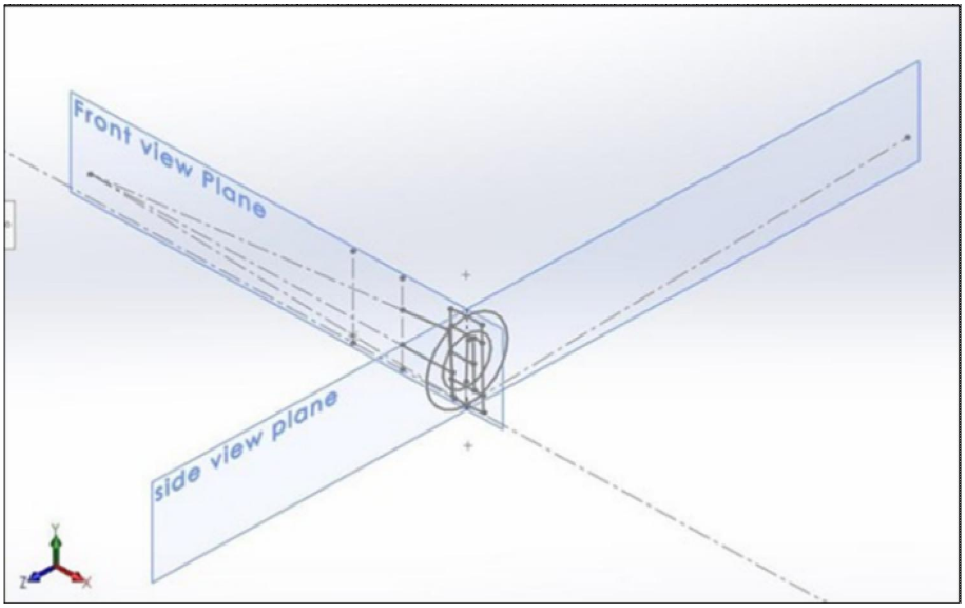

Fig. 1. 2D line repesentation

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 fount view swing arm (FVSA) length.

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

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.

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 1. So, an upper and lower ball joint x, y location input for Optimum kinematics software was obtained.

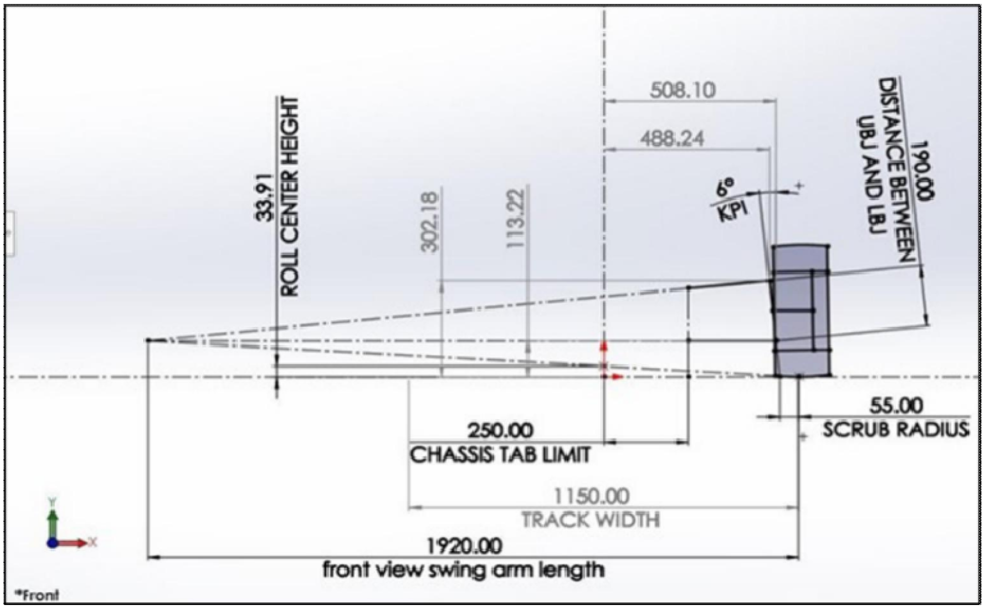

Fig. 2. *Front view swing arm line diagram*

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.

Selection of Chassis Tab Limit

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

Deciding of 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.

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.

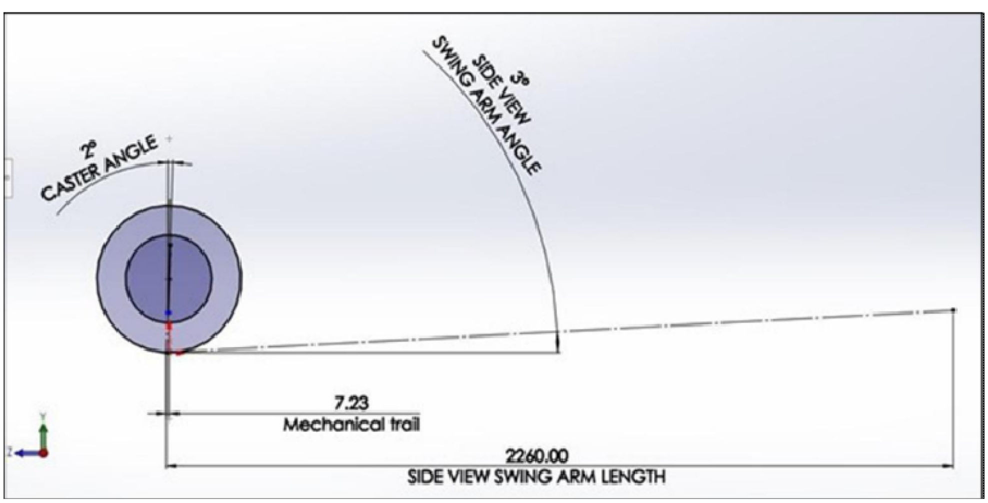

Fig. 3. *Side view swing arm line diagram*

$$\%Anti-dive = \frac{\tan\tan\,(front\ side\ view\ swing\ arm\ angle)}{\frac{CG\ height}{wheel\ base}} \quad [2]$$

$$\%Anti-squat = \frac{\tan\tan\,(back\ side\ view\ swing\ arm\ angle)}{\frac{CG\ height}{wheel\ base}} \quad [3]$$

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.

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.

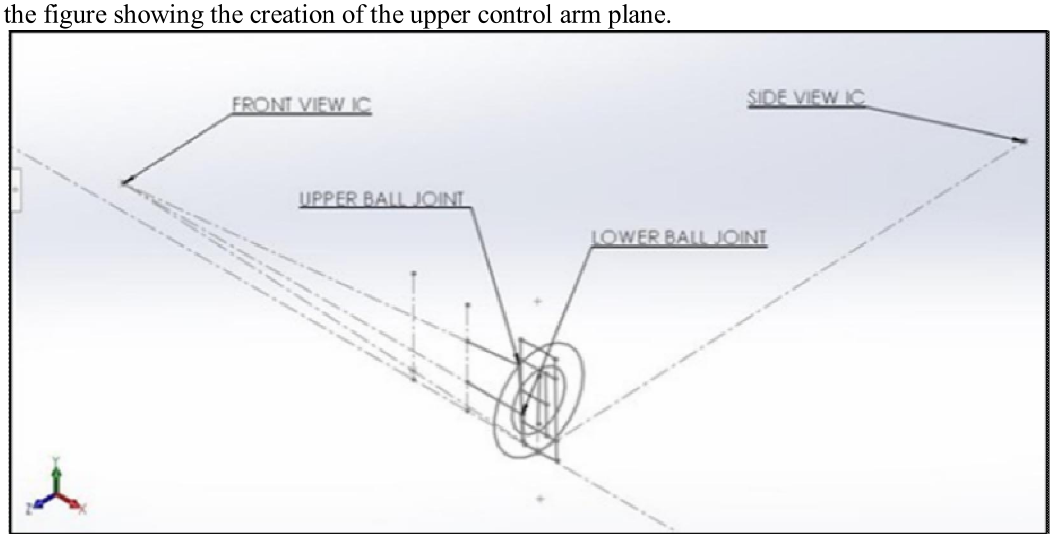

Fig. 4. 3D view of Instantaneous centres and ball joints

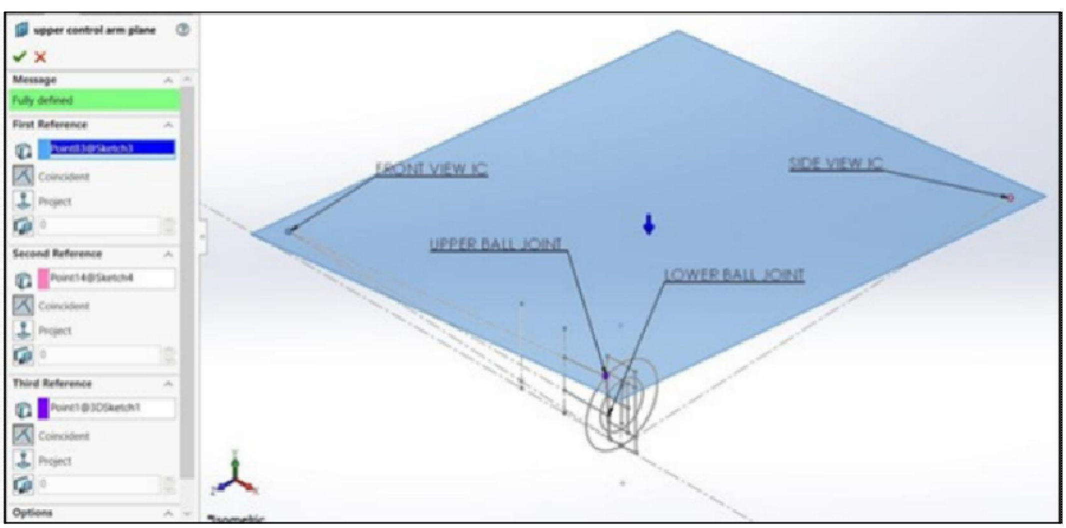

Fig. 5. *Plane creation from Ic's and ball joints*

Determination of Hard Points

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 6 cartesian coordinates of hardpoints are found, these hardpoints can be inputted into the kinematic simulation software to simulate the kinematic motion of the suspension.

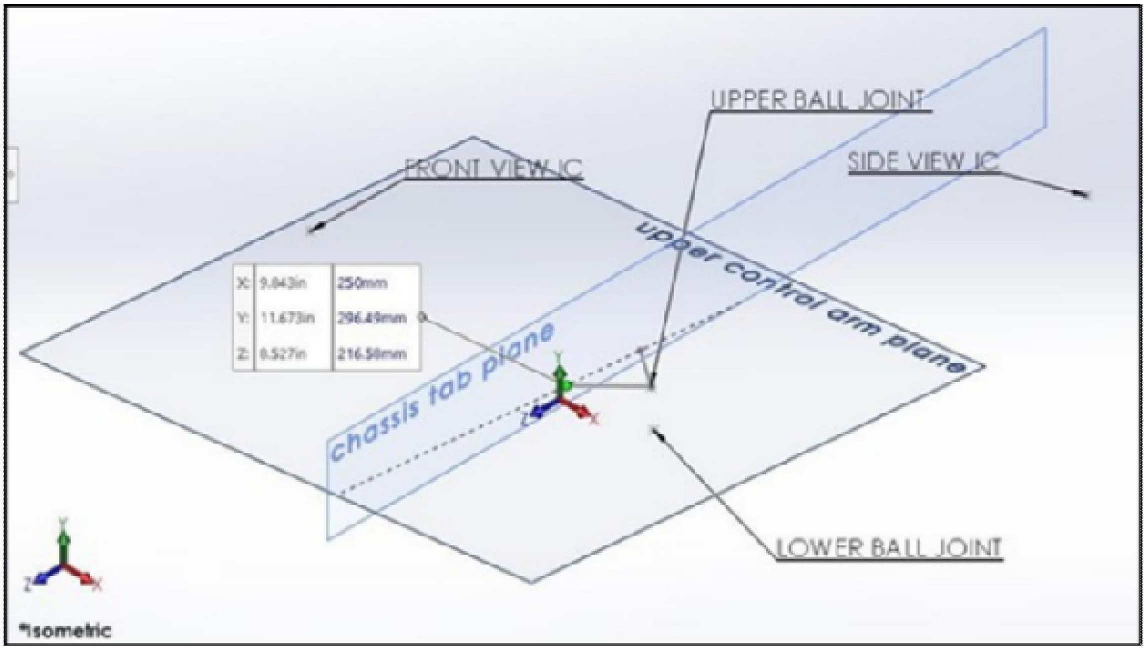

Fig. 6. *A-arm line diagrams*

Kinematic Simulation

Optimum Kinematics, developed by Optimum G, 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.

Advantages and Benefits

Optimum Kinematics is used by designers and analysts alike to layout the suspension hardpoint locations to achieve the required kinematic behavior. 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.

Procedure

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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.

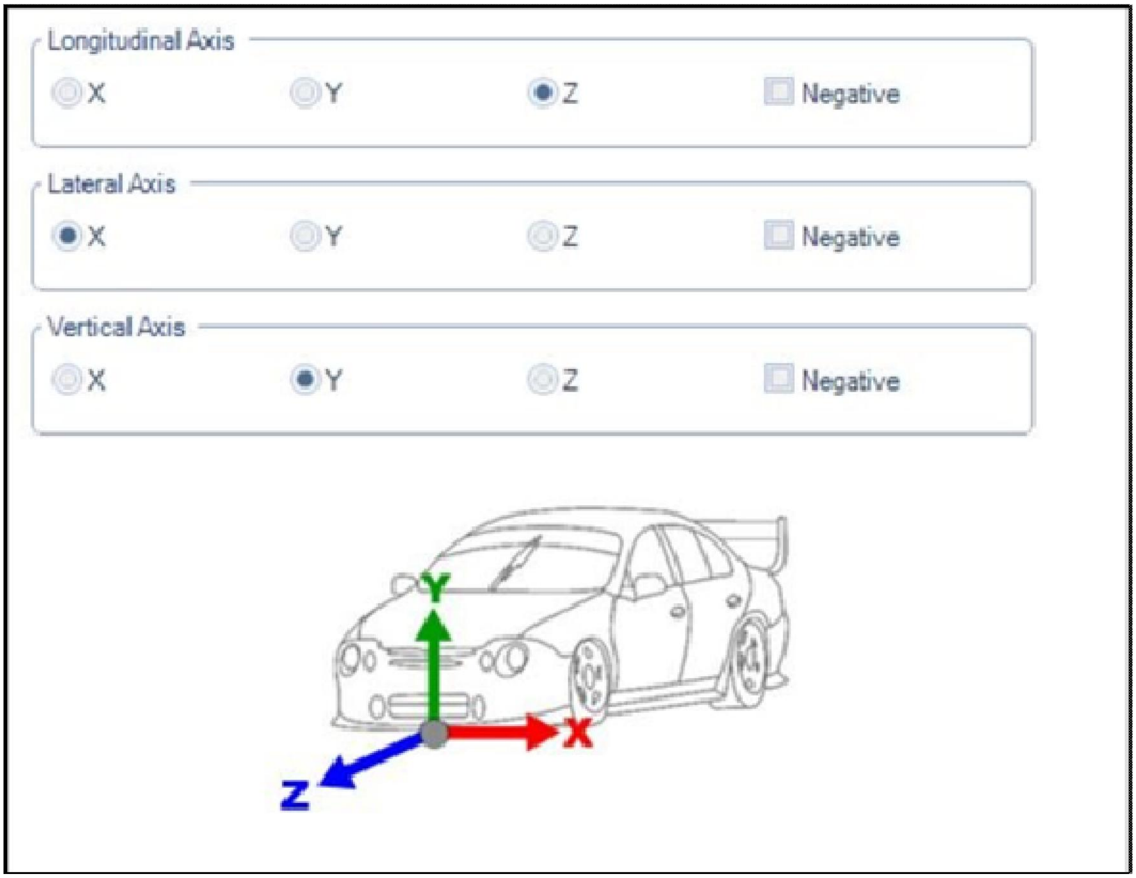

Fig. 7. *Vehicle axis system in Optimum Kinematics*

Sign Convention

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

Formulae, symbols and equations (*style: subsection*)

Formulae, symbols, Greek letter, and other character must be legible and carefully checked. Standard mathematical notation should be used. All symbols used in manuscript must be explained.

Single letter that denote mathematical constants, variables, and unknown quantities should be set in italic type, both in text and in equations. Numerals, operators, and punctuation should be set in Roman type (upright), as are commonly defined functions or abbreviations, e.g., cos, det, e (or exp), lim, log, max, min, sin, tan, d (for derivative) should also be upright. Vector and tensors should be set in bold italic; matrices should be set in bold.

Table 1. Vehicle Data

Property	Symbol	Value
Acceleration due to gravity	g	9.81 m/s2
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	μ	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

Result and Discussion

Combine

Formulae, symbols, Greek letter, and other character must be legible and carefully checked. Standard mathematical notation should be used. All symbols used in manuscript must be explained.

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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 front	47.75	mm
Kinematic roll axis inclination	0.5	deg

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

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

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.

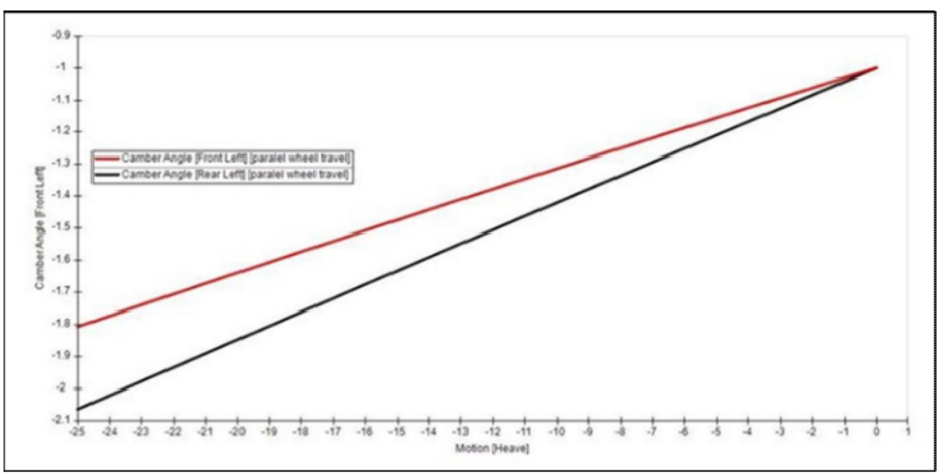

Fig. 8. *Camber angle (v.s) Heave Motion*

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

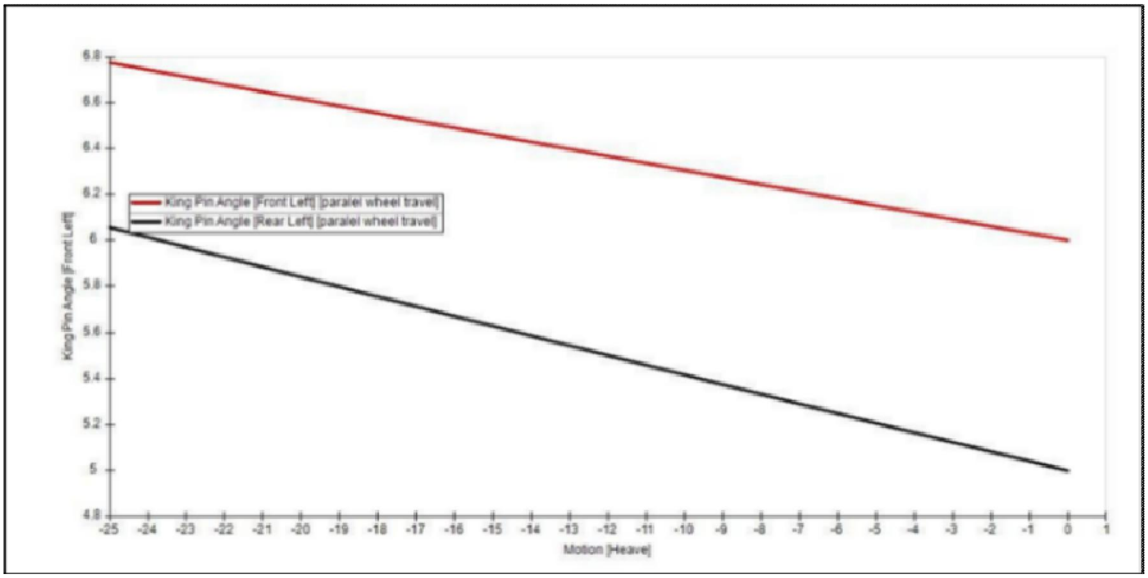

Fig. 9. KingpinAngle (v.s) Heave motion

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 20 to 2.650. 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.

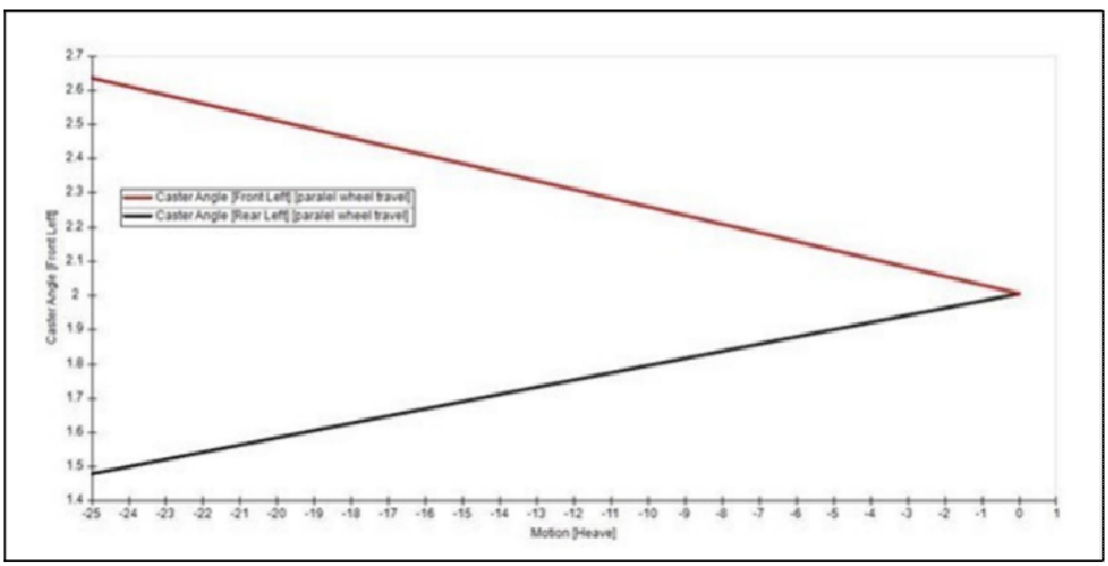

Fig. 10. Camber angle (v.s) Heave motion

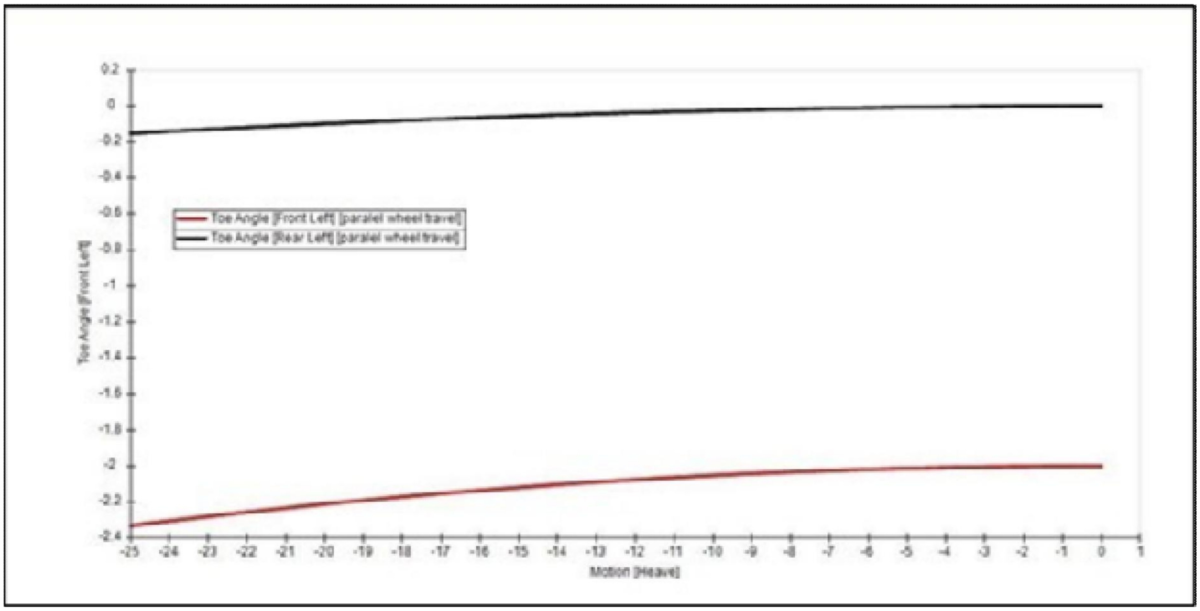

Fig. 11. Toe angle (v.s) Heave motion

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

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.

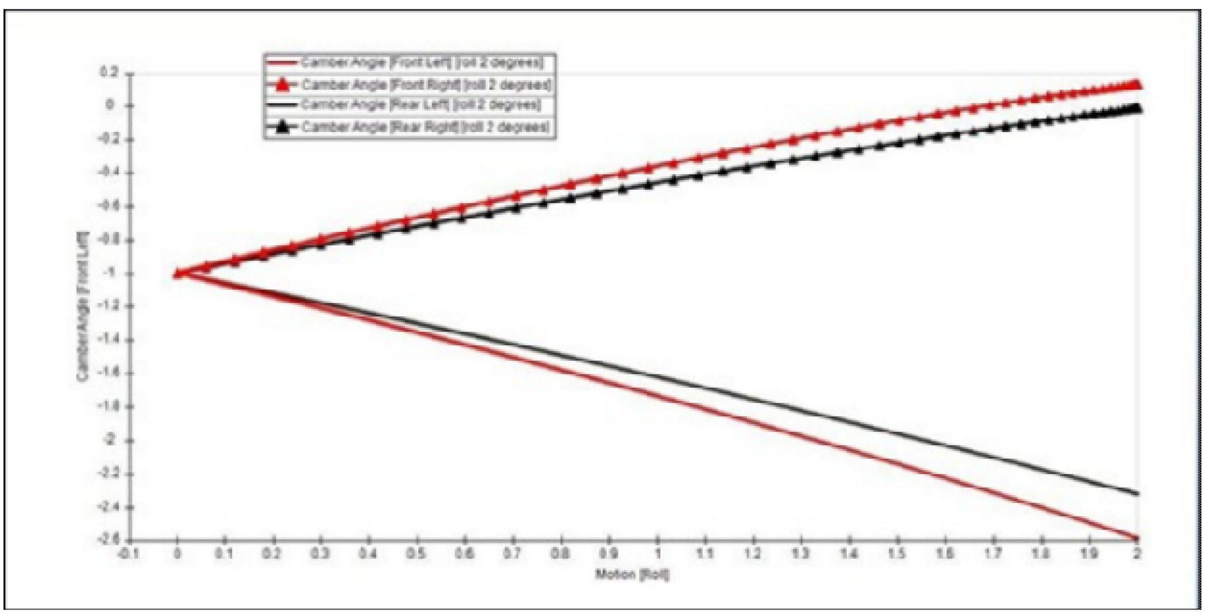

Fig. 12. Camber angle (v.s) Roll motion

The caster variation is within the max variation limits of under 0.50, so it wouldn't cause any issue while racing.

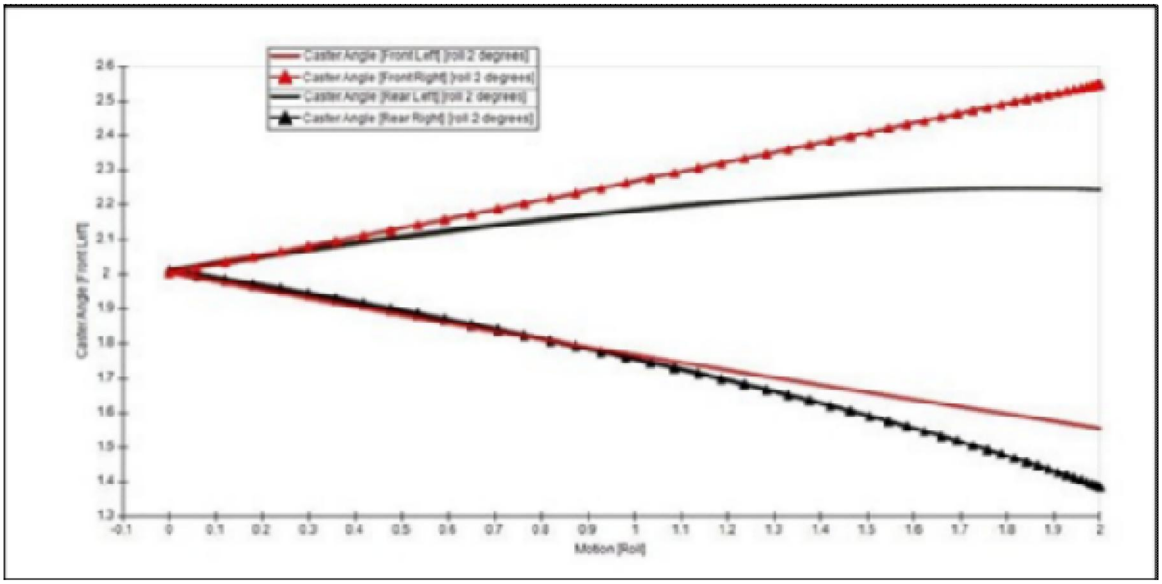

Fig. 13. Castor angle (v.s) Roll motion

The increase in KPI may result in increasing the steering effort, but the variation is under or equal to 10, so it wouldn't be a problem

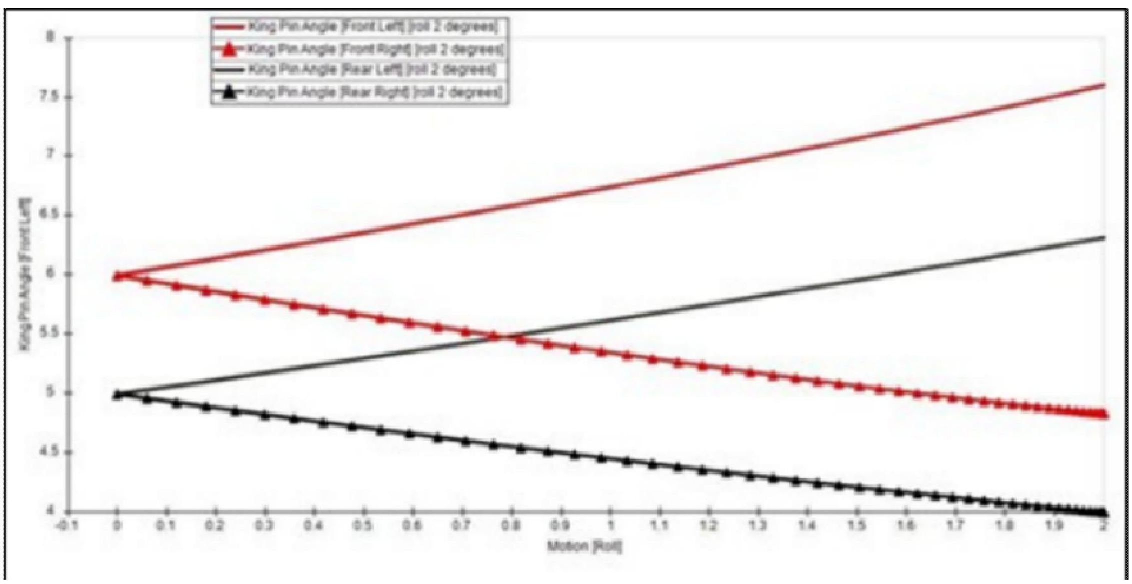

Fig. 14. Kingpin angle (v.s) Roll motion

The toe angle variation is almost negligible as the variation is under 0.5°. It wouldn't produce self-steering effects as the toe angle is so less.

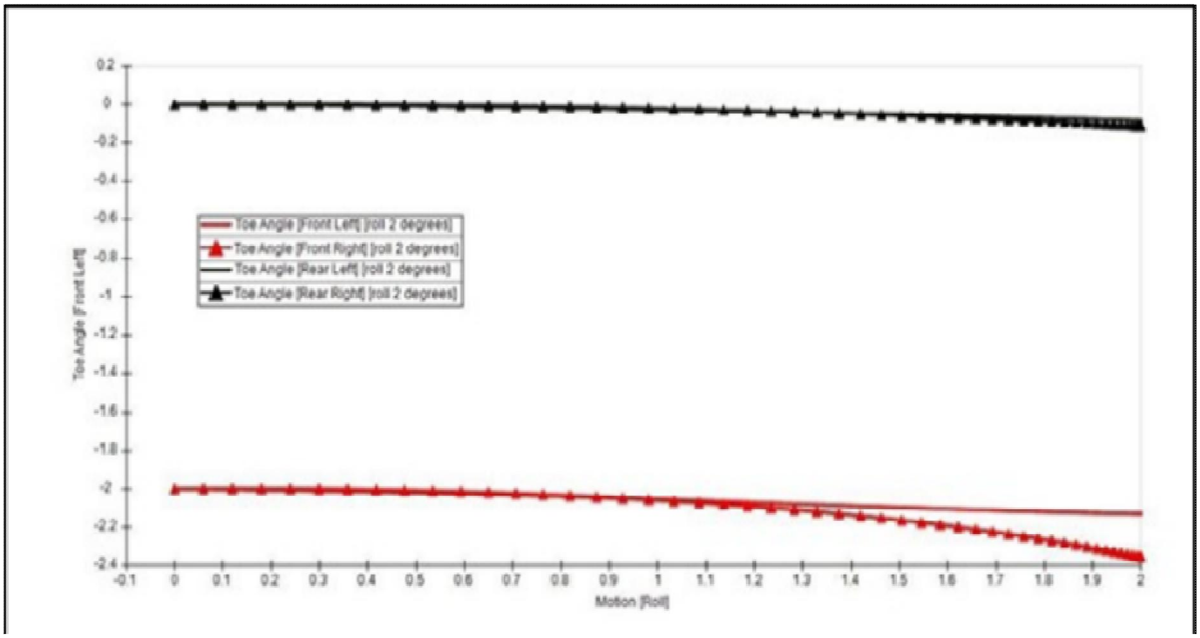

Fig. 15. Toe angle (v.s) Roll motion

Conclusion

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.

- The height of the Centre of gravity got reduced, providing stability for the vehicle during cornering.
- Optimum wear and tear were occurred due to minimum variation of suspension parameters.
- It helped in designing the roll cage, subsequently reducing its weight from 45kgs to 35kgs.
- It prevented road shocks from being transferred to the vehicle.
- It safeguarded the driver from road shocks.
- It maintained good traction during driving, cornering, and braking.

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