CZECH TECHNICAL UNIVERSITY IN PRAGUE

FACULTY OF MECHANICAL ENGINEERING

# MASTER'S THESIS ASSIGNMENT

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|---|---|
| **TITLE:** | Rear Suspension Sensitivity Study and Optimization |
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**GUIDELINES:**

Create a sensitivity study of rear independent suspension kinematics in terms of modifying its primary geometry with regards to toe, camber and caster angles. Research currently used suspension types on RWD & AWD cars. Build multibody model of rear suspension convenient CAE software based on provided hard points and create it kinematic characteristics. Optimize suspension kinematics based on sensitivity study i.e. by means of moving hard points or altering arm lengths – automate the optimization process using suitable programming interface / stand-alone system.

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MASTER OF AUTOMOTIVE ENGINEERING

Master's Thesis

# Rear Suspension Sensitivity Study and Optimization

Thesis project in association with Ricardo s.r.o., Prague, Czech Republic

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# Declaration

I, Nisant M Sethia declare that this thesis is completely written by me as a result of my own research and work. All the external sources used directly or indirectly are clearly marked and quoted. The sources are all acknowledged in the right manner.

This work has neither been submitted in part or full to any institute or authority for other qualification, nor it has been published elsewhere.

Prague, 11th January'19

Nisant M Sethia

# Abstract

The main objective of the thesis was to model a rear independent suspension using a CAE software and to understand the sensitivity behavior of the modeled suspension kinematics, by modifying its primary geometry with respects to camber, caster angels and to optimize the design.

During the course of study, a brief explanation of currently used suspension type for RWD and AWD are discussed. A detailed study on Multi-link suspension's modeling, kinematic characteristic along with study on multi-body modeling and simulation using ADAMS/Car is carried out in order to build a functional multi-body model of rear integral-link suspension.

The latter part of study is focused on sensitivity of integral-link suspension hardpoints. Highlighting which geometric point affect the kinematics. Using these hardpoints as input variable for the optimization process to obtain new geometry as per the performance targets defined. Then, compare its kinematic characteristic with the primary design.

Integral-link suspension modeling and simulation was carried out using ADAMS/Car. Toe, camber and caster variation with respect to vertical displacement of the wheel are obtained and the curve trends are compared with the existing model for validating the concept model. An automated optimization model using ADAMS/Insight is defined to achieve improved performance as closes to benchmarking target compared to the primary model by studying the sensitivity of integral-link suspension hardpoints.

**Keywords:** Rear Independent Suspension, Integral-link suspension, Multi-body modeling, Sensitivity study, Optimization, ADAMS/Car, ADAMS/Insight

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# Abbreviations

**AWD** – All Wheel Drive

**RWD** – Rear Wheel Drive

**MBS** – Multi Body Simulation

**SLA** – Short-long arm

**K&C** – Kinematic & Compliances

**ILS** – Integral-link Suspension

**UCA** – Upper Control Arm

**LCA** – Lower Control Arm

**CAE** – Computer-Aided Engineering

**DOF** – Degree of Freedom

**SPH** – Spherical Joint

**CYL** – Cylindrical Joint

**REV** – Revolute Joint

**K&C** – Kinematics and Compliances

**DOE** – Design of Experiments

**RSM** – Response Surface Method

# 1. Introduction

## 1.1 Motivation

In past decades, advancements in the world of automotive has changed drastically and now vehicles have become an essential part of day to day commute for humans. When it comes to buying or choosing an automobile the person undeniably looks for a comfortable ride along with good vehicle performance. Passenger opinion regarding what constitutes good ride quality obviously is extremely subjective. What one person considers the optimum ride may be completely undesirable to another. The person who prefers sports cars will be shocked by the handling of a large luxury vehicle, whereas the owner of the luxury vehicle will be quite dissatisfied with the ride of a sports car. This is where vehicle suspension comes into depiction and the crucial stage of the vehicle development is the design of the suspension geometry. Suspension's geometry affects vehicle behavior when running straight or through a curve, on bumps as well during braking and accelerating. The geometry directly affects force effects acting on the car and thus also its dynamic state, and indirectly it influences the way the driver and passenger feels the vehicle when driving. Therefore, it is important to evaluate many suspension kinematic characteristics and try to attain optimum values by modifying its geometry.

## 1.2 Modeling/Simulation

To study the general trends before investing profoundly on actual experimental testing industries these days rely heavily on computer modeling and simulation. Testing typically requires multiple vehicles, numerous sets of tires and expensive instrumentation to obtain indicative data on all the relevant parameters, and after all this there is still a possibility that the results show that the desired results cannot be obtained thus making the whole experiment a waste of resources, but virtual simulation doesn't. To lessen the expenses associated, a model is first created, and simulations are run on it, to make sure that the trend shows improvement if not, optimization on the virtual model to achieve the required trend is carried out and that one may indeed go ahead and invest in the experimentation.

## 1.3 Objective

The foremost purpose of this research work is to study the suspension sensitivity behavior, to optimize it with the aim of improving the kinematics and to automate the optimization process of the initial concept model. To achieve these goals, the following objectives are defined:

- General study on currently used suspensions types in RWD & AWD.
- To study the background of multi-link suspension types and its kinematics.
- To build a concept of a virtual multibody model of rear integral link suspension using MSC ADAMS/Car.
- To perform the MBS on the model for obtaining the kinematic characteristics.
- To do the sensitivity analysis of the model by altering the arm lengths(links) and moving hardpoints and determining how the kinematics vary according to it.
- To automate the optimization process using ADAMS/Insight for improving the performance by reaching as close to the benchmark targets.
- To discuss the findings of the research based on obtained results.

## 1.4 Outline

With the intention of achieving the objectives of this study, three steps were performed. In congruence with these steps, this thesis is organized in three succeeding chapters. These steps and the chapters are presented here in this section.

The first step was a background and literature review of the study in the field of vehicle's suspension systems with their kinematic characteristics and multibody modeling of multi-link type rear suspension along with the sensitivity behavior of wheel suspension and optimization methods. These topics are presented in the second chapter of this thesis. Firstly, a general definition and function of the suspension system presented. Then a general description on suspension system which are currently used in RWD and AWD are described. Finally, a detailed study on multi-link type suspension and suspension kinematic characteristics are labelled.

The second step was building a concept model for rear ILS system under development and creating its kinematic characteristics, which is one of the type of multi-link suspension. These subjects are presented in chapter three of this thesis. The design includes a multibody ADAMS/Car model for a RWD suspension followed by creating a suspension assembly to perform MBS and obtaining the kinematic characteristics.

The third step was to implement the study of suspension sensitivity for the concept ILS under development, followed by automating the optimization for the concept suspension model’s kinematics in terms of modifying its primary geometry with regards to toe, camber and caster angles and to improve the suspension performance to its primary one.

# 2. Literature Review

## 2.1 Background of Wheel Suspension

The wheel suspension defines the position of the wheels relative to the body. If the road with smooth surface were constructed, then there was no need of the suspension in the vehicle to provide a comfortable ride to the passengers as well as the driver but there are various types and irregularity of road surfaces in reality, and a suspension system is necessary in order to isolate the vehicle from the exciting vibrations of the road unevenness. A text book definition of a suspension system is, it is an interface between the vehicle body frame and the road surface and this statement assumes that the wheels and tires comprise part of the suspension system, which they indeed do [1]. The primary functions of suspension system and the requirements are listed below [3]:

- Carry the vehicle and its weight.
- Isolate passengers and cargo from vibration and shock.
- Improve mobility.
- Provide for vehicle control.
- Maintain the correct wheel alignment.
- Little space requirement.
- Easy to steer with the existing drive of the vehicle.
- Low weight.
- No mutual wheel influence for independent wheel suspension.

The last two characteristics are important for good road-holding, especially on bends with an uneven road surface. Besides these function, suspension systems should maintain a permanent contact between tires and road surface to provide good ride stability for vehicle. This ability is particularly important as vehicle is turning, accelerating, or braking. Briefly, a vehicle suspension provides ride comfort and driving stability for the vehicle. They are also necessary for protecting the road surface from excessive tire forces and pavement damage [2][4].

## 2.2 Types of Wheel Suspension

Generally, the suspension system is classified under the category of dependent and independent but in the following literature the suspension discussed are on the classification as per the driveline of the vehicle currently used in the automotive industry.

### 2.2.1 Double wishbone suspension

Double wishbone suspension is also known as SLA as the upper and lower control arm are of unequal length. The last two characteristics mentioned above can be easily achieved using double wishbone suspension. This system contains two transverse links/control arms either side of the vehicle, which are mounted to rotate on the frame, suspension subframe or body and in the case of the front axle these are connected on the outside to the steering knuckle with ball joints.

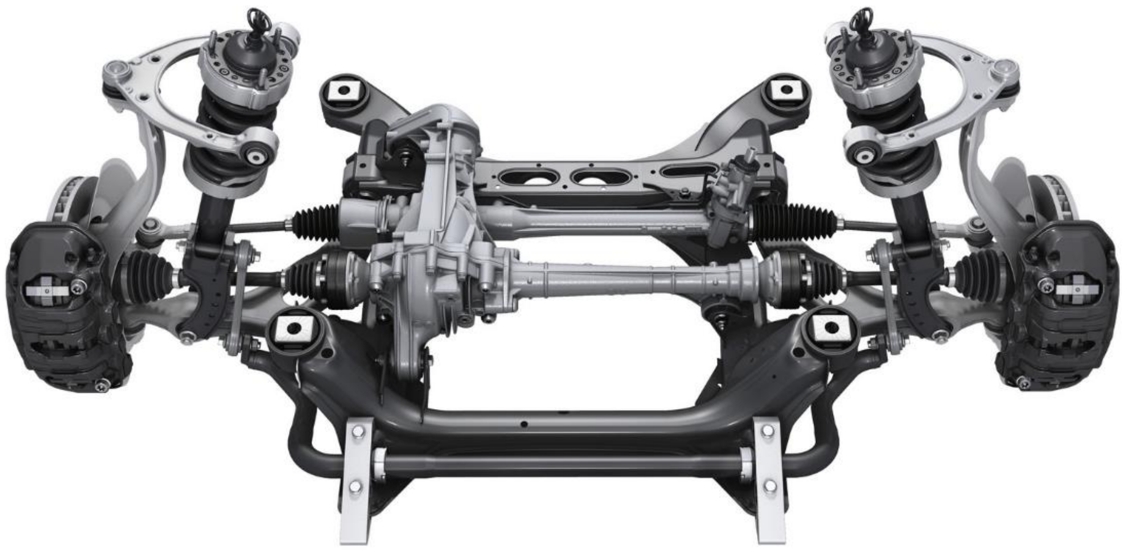


Figure 2.1:Double Wishbone Suspension Porsche Cayenne [source: Lecture Handouts by Porsche]

Originally double wishbone had equal-length upper and lower control arms to prevent camber changes during suspension deflection but under cornering condition the suspension deflection is caused by the body roll which causes camber. So, the modern double wishbone suspension has shorter upper control arm. This suspension is a perfect fit for rear wheel front engine car due to the package space for longitudinally oriented engine [2][4].

### 2.2.2 MacPherson Strut

MacPherson strut is a further development of SLA and the system was developed by Earle S. MacPherson, a Ford suspension engineer, in the 1940s. In this the upper control arm is replaced by a pivot point on the knuckle, which takes the end of the strut(damper) and the coil spring. The upper end of the damper is fixed to the chassis. In this system to avoid elastic caster and camber change the rod diameter in the piston should be increased to 18mm from 11mm and the diameter of the piston is in the range of 30mm to 32mm what works on the twin tube system which is either pressurized or non-pressurized. Due to the small amount of space required for this system, it is ideal for front-wheel-drive cars that use transversely mounted engines.

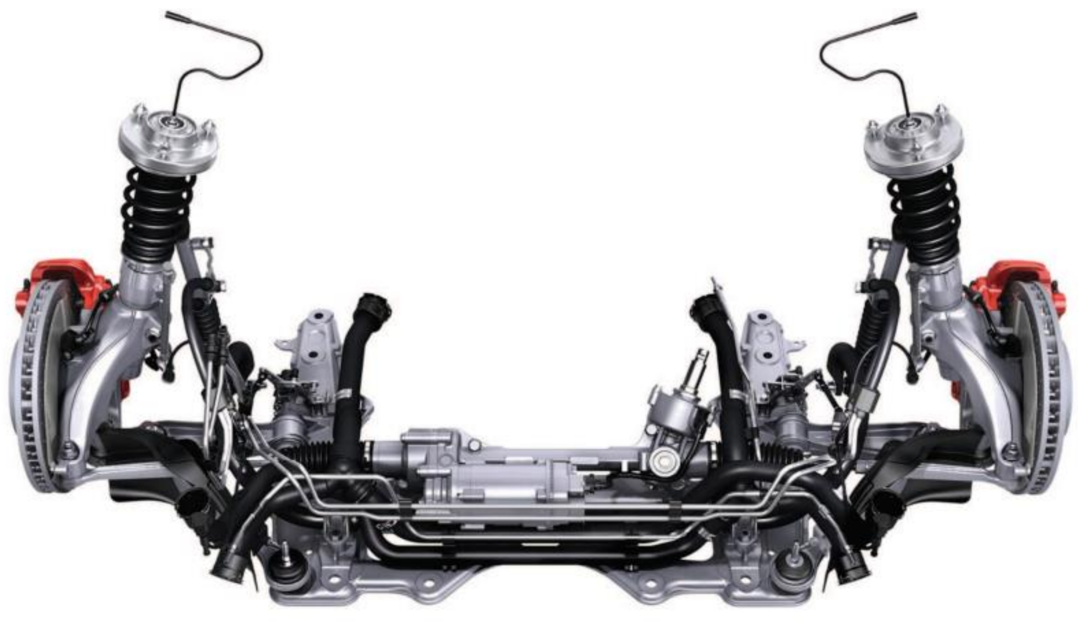


Figure 2.2: Front MacPherson Strut of Porsche 911 [source: Lecture Handouts by Porsche]

MacPherson struts are used almost exclusively on the front axle, but they are also fitted as the rear suspension with inverted leaf spring which replaces the coil spring. When fitted at the rear end of the car, the tail which is raised for aerodynamic reasons, allows a larger bearing span between the piston rod guide and piston. The main advantage of the MacPherson strut is that all the parts providing the suspension and wheel control can be combined into one assembly, but the system requires a larger amount of vertical space for installation and thus limits the designer's freedom to lower the hood height [2][4].

### 2.2.3 Trailing arm suspensions

This kind of suspension is used in the rear of the vehicle in AWD and it consists of a control arm lying longitudinally in the driving direction and mounted to rotate on a suspension subframe or on the body on both sides of the vehicle. Pure trailing arm suspension is used on high-performance car. It offers the advantage that the car body floor pan can be flat, and the fuel tank and/or spare wheel can be positioned between the suspension control arms. [2]

Figure 2.3: Trailing-arm rear suspension of the Mercedes-Benz A class (1997) [2]

Semi trailing-arm axle is a different kind of trailing arm suspension whose design is in between a swing axle and a pure trailing arm suspension, which is fitted mainly in RWD and AWD passenger cars. In this the length of control arm can positively affect the kinematics of the vehicle like toe, camber and height of roll center. The main advantage of semi-trailing arm is that it provides a wide luggage space with a level floor because its design hardly requires any overhead room.

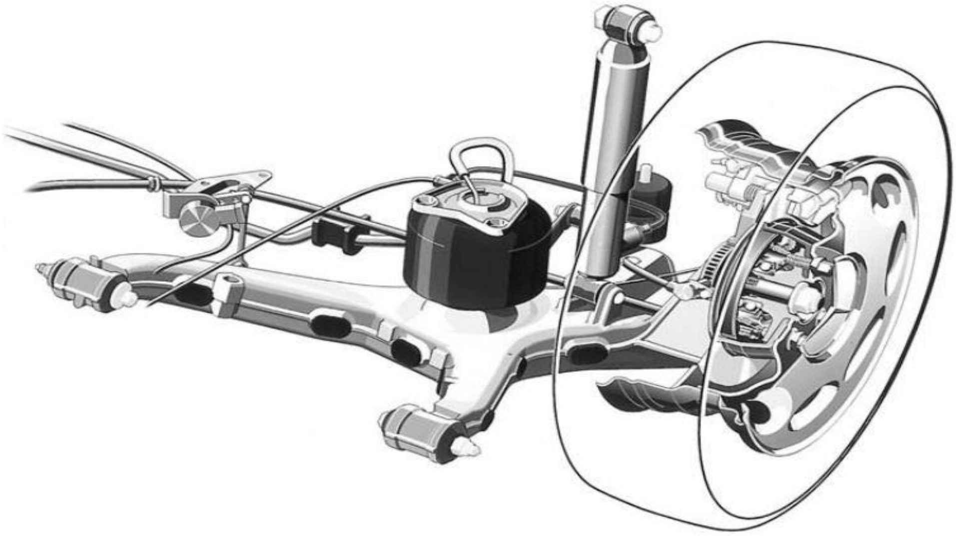


Figure 2.4: Semi-trailing Arm Mercedes-Benz V class [2]

### 2.2.4 Torsion Beam

The torsion bar is basically a length of metal rod anchored at one end to the car body and at the other end to the suspension lower link this can be also said that it consists of two trailing arms that are welded to a twistable cross-member and fixed to the body via trailing links. As the wheel passes over a bump the bar twists and returns to its original position when the bump is passed eventually restoring the car to its normal drive height. The resistance of the bar to twisting has the same effect as the spring used in more conventional suspension systems. [4]

Figure 2.5: Torsion Bar suspension [2]

### 2.2.5 Multi-Link Suspension

Mercedes-Benz was the first automotive firm to develop a kind of multi-link suspension in 1982 for the 190 series. Since then driven and non-driven multi-link front and rear suspensions are being used. A multi-link suspension can be characterized be ball joint connection at the end of linkages so there isn't any bending moment affect. As per designer four to five links are used to control wheel forces and torque depending on the geometry, kinematics, elastokinematics and force application of the axle. The arrangement of the links is according to the amount of packaging space available, there is extraordinarily wide scope for design when it comes to this king of independent wheel suspension. If the system consists of five links, then the additional link over constrains the wheel but take advantage on compliances in the bushings to provide more accurate kinematic properties. Because of the more open design, the wheel forces can be optimally controlled, i.e. without superposition, and introduced into the bodywork in an advantageous way with wide distances between the supports. [2][4]

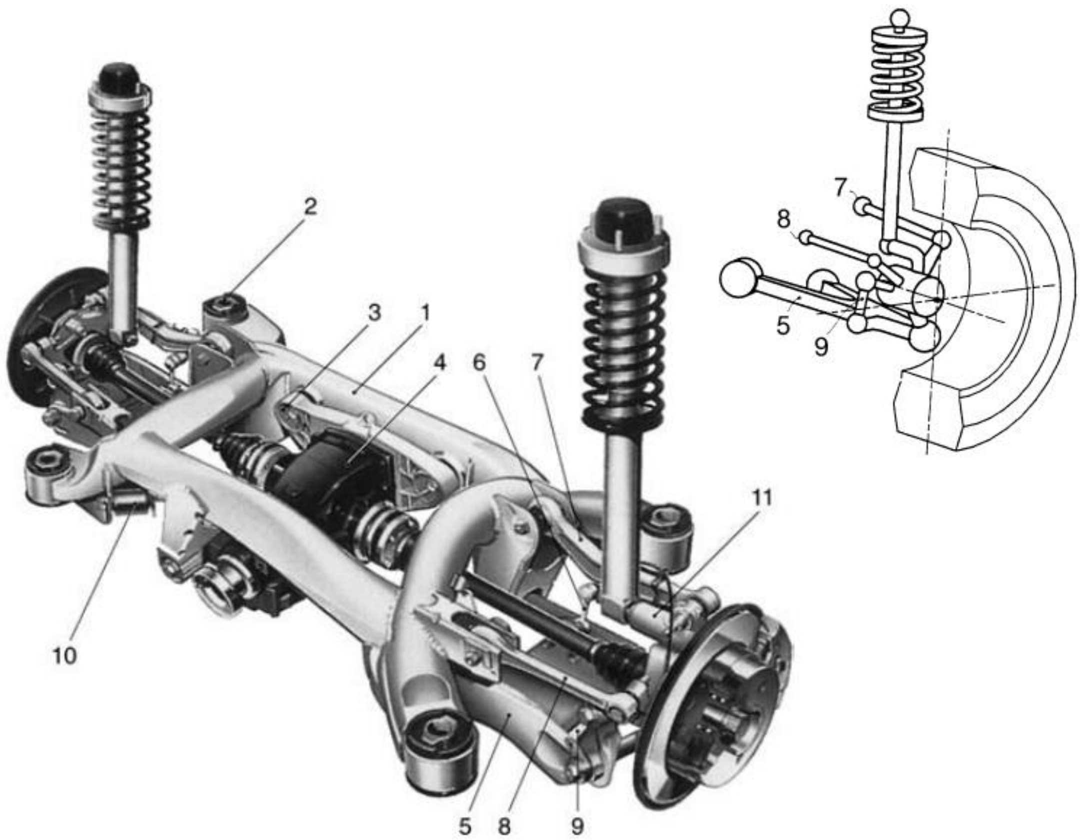



Figure 2.6: Integral-link rear suspension of the BMW 5 series (E39, 1996) [2]

In figure 2.6, The subframe (1), is attached to the bodywork by means of four large rubber mounts (2). These are soft in a longitudinal direction for the purposes of riding comfort and noise insulation and rigid in a transverse direction to achieve accurate wheel control. The differential gear also has compliant mounts (3). The wheel carrier is mounted on a U-shaped arm (5) at the bottom and on the transverse link (7) and inclined guide link (8) at the top. Because of this inclined position, an instantaneous center is produced between the transverse link and guide link outside the vehicle which leads to the desired brake understeer during cornering and the elastokinematic compensation of deformation of the rubber bearings and components. The driving and braking torque of the wheel carrier (11) is borne by the integral link (9) on the swinging arm (5), which is subject to additional torsional stress as a result. This design makes it possible to ensure longitudinally elastic control of the swinging arm on the guide bearing (10) for reasons of comfort. The stabilizer behind presses on the swinging arm (5) by means of the stabilizer link (6), whereas the twin-tube gas-pressure shock absorber and the suspension springs provide a favorably large spring base attached directly to the wheel carrier (11). [2]

Eun Ko *et al.,* (2006) [5] discusses about newly developed a multi-link suspension for Hyundai Elantra in their paper for better driving pleasure and confidence to driver with better K&C characteristics compared to its older car model. This rear suspension is completely modified from dual-link to multi-link suspension. The modeling of multi-link suspension consists of two upper control arms, a lower control arm, and a plane-shaped trailing arm. Front upper arm with cam-bolt plays a role in toe-control link. The hard point of main upper control arm is located as possible as highly with the constraint of straight upper arm shape. The lower control arm and two upper control arms support cornering force, and the trailing arm is located longitudinally for controlling the longitudinal braking and impact force. Two vibration isolators of spring and damper in rear suspension are separated for better ride performance.

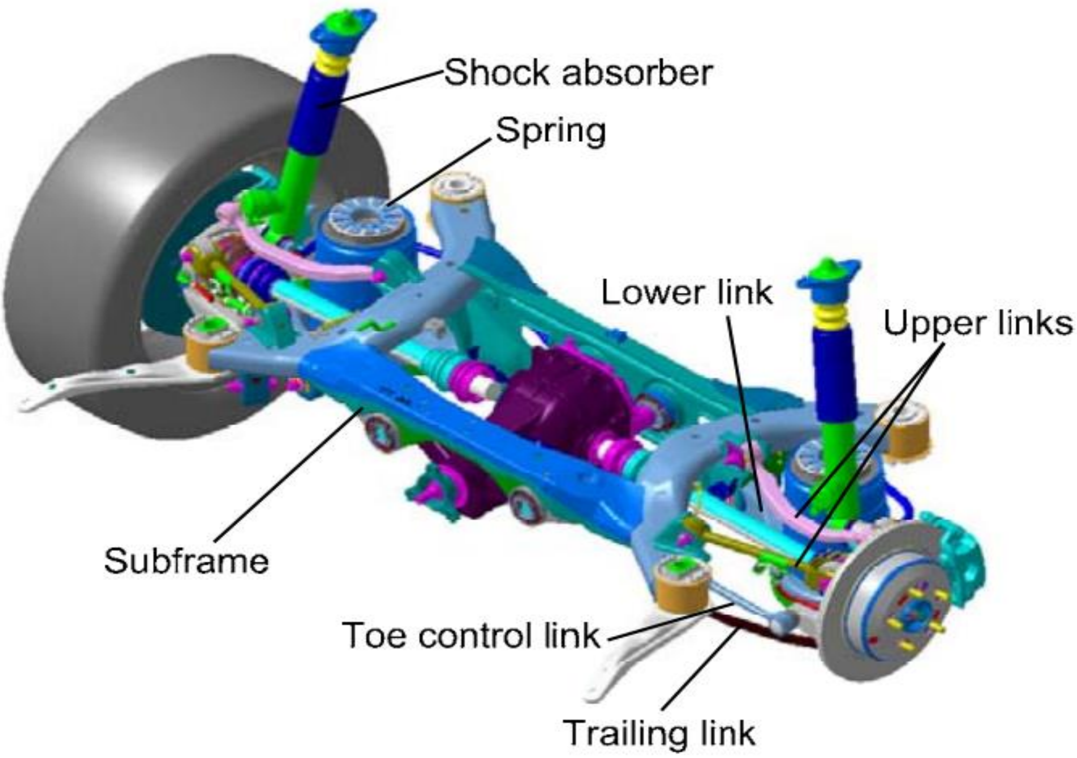



Figure 2.7: Hyundai Genesis real multi-link suspension [7]

Another paper by Hyundai Motor Company where Pyung Kim *et al.,* (2009) [6] discusses about the designed model of multi-link rear suspension for Hyundai Genesis and compares its toe behavior with respect to parallel wheel travel, brake & cornering forces with other competitive variants in the market then. It comprises of 5 links such as two upper links, lower link, trailing link, and toe control link. Spring and shock absorber are installed separately to lower control arm and axle carrier respectively. They are mounted to lower part of rear floor side member and wheel house, respectively. As it is a rear wheel drive the drive shaft is are installed at the rear suspension.

M.B.Gerrard., (2002) [7] discusses about the kinematic model and benefit of using such a scheme for the rear integral link suspension for Alfa Romeo 166. In figure 2.8, the rear ILS consists of a H-arm(LCA), two lateral links which are a camber link(UCA) and a toe link finally an integral link denoted as gear link which is connected between the knuckle/upright and wishbone. The benefits from this linkage configuration is that this model allows the individual static toe and camber adjustments.

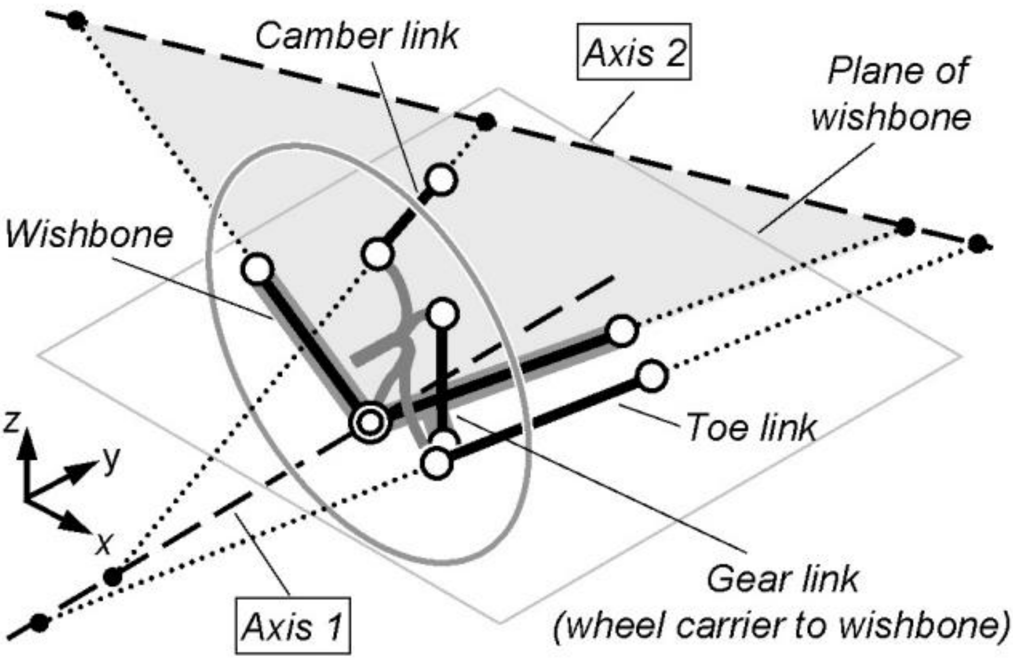



Figure 2.8 Alfa Romeo 166 rear ILS [7]

Fiat Chrysler Automobile press release [8] stated this ILS system improves performance to ensure optimum control of steering on corners and while braking, and great stability during overtaking and lane-change maneuver.

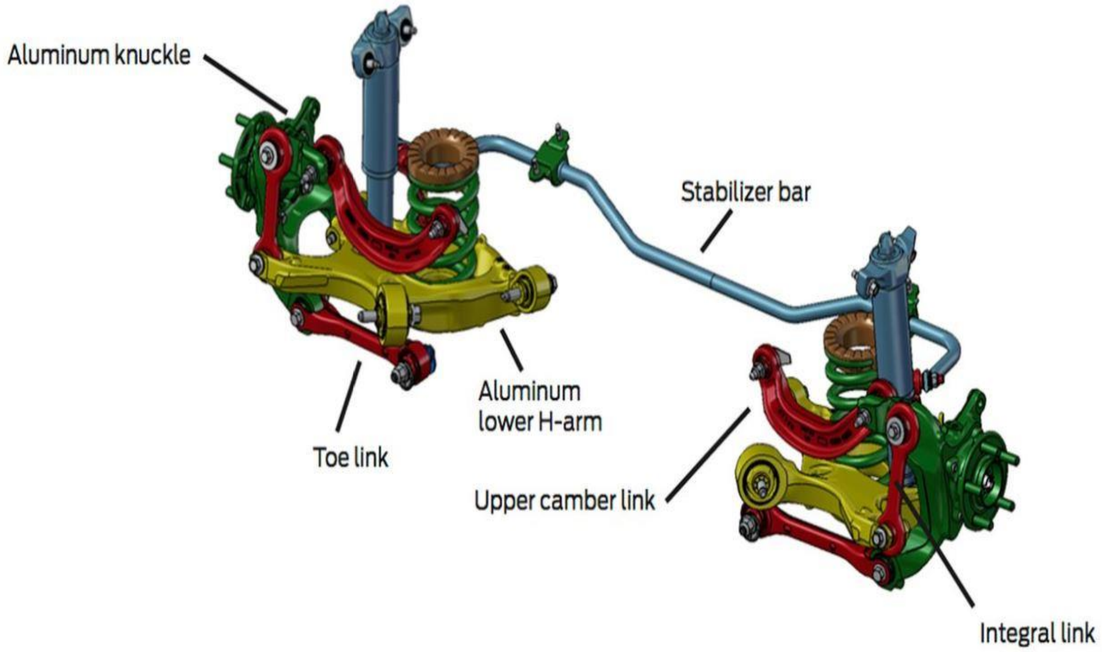



Figure 2.9 Ford Mustang 2015 Rear ILS [26]

In figure 2.9, the rear ILS developed by Ford for Mustang GT 2015 has similar design and configuration as Alfa Romeo 166 ILS except the lower wishbone it has an H arm but behaves as an A shaped arm, complete with two inner mounting points that define the pivot axis and a single outer bushing that locates the lower end of the knuckle. The camber link and the toe link are for tuning the cars camber and toe respectively. The integral link during acceleration and brake torque keeps the top end of the knuckle from moving front and back under even the most immense torque loads. The odd H-shape of the lower control arm is the result to give the integral link a solid base. In this the coil spring and the shock absorber have been mounted separately to reduce the intrusion of these suspension parts into the trunk area. This design made it possible to design a wider fuel tank along the cars lateral axis, allowing the length of the tank in the longitudinal direction to be reduced while achieving the same tank capacity and to increase the boot space by few centimeters [9].

## 2.3 Kinematics of Wheel Suspension

According to Mitchell *et al.,* (2008) [10] kinematics in a general sense is the study of motion without reference to mass or force. When it is applied to vehicle suspension systems, suspension kinematics describes the controlled orientation of road wheels by suspension links, based on assumptions of rigid parts and frictionless joints.

Porteš., (2011) [11] in his journal discussed the basic characteristics which includes dependence of the following parameters on wheel vertical displacement also known as parallel wheel travel during simulation.

- Steering angle on wheel vertical displacement (Toe angle)
- Wheel camber on wheel vertical displacement
- Wheel track on wheel vertical displacement
- Height of roll center on wheel vertical displacement
- Position of pitch center on wheel vertical displacement

In the case of the front suspension, the following dependences are added:

- Caster angle on wheel vertical displacement
- Kingpin inclination on wheel vertical displacement
- Caster offset on wheel vertical displacement
- Kingpin offset on wheel vertical displacement
- Wheel camber on steering angle
- Left steering angle in relation to right wheel (Ackermann geometry)

### 2.3.1 Camber

Camber is defined as the inward or outward tilt of the wheel at the top when viewed from the front of the vehicle, it is measured in degrees. A proper camber setting is essential on a vehicle as it improves road isolation and directional stability whereas incorrect camber will cause tire wear and handling problems.

Camber angle is measured in degrees, a wheel with zero-degree camber is vertical, it is positive if the wheel is inclined outwards and negative when inclined inwards shown in figure 2.10. Normally when a vehicle is cornering, the wheels on the outside of the turn go into positive camber relative to the ground reducing the lateral grip of the tire under load. To achieve this phenomenon, suspension designers tend to achieve positive camber in bump and negative camber in rebound motions, in their designs. Apart from this, a slight negative camber on a loaded vehicle would make the tires roll with maximum contact patch resulting in even wear and lower rolling resistance on a slightly curved road. The slip angle increases with increase in positive camber and the cornering force increases with increase in negative camber [2] [17].

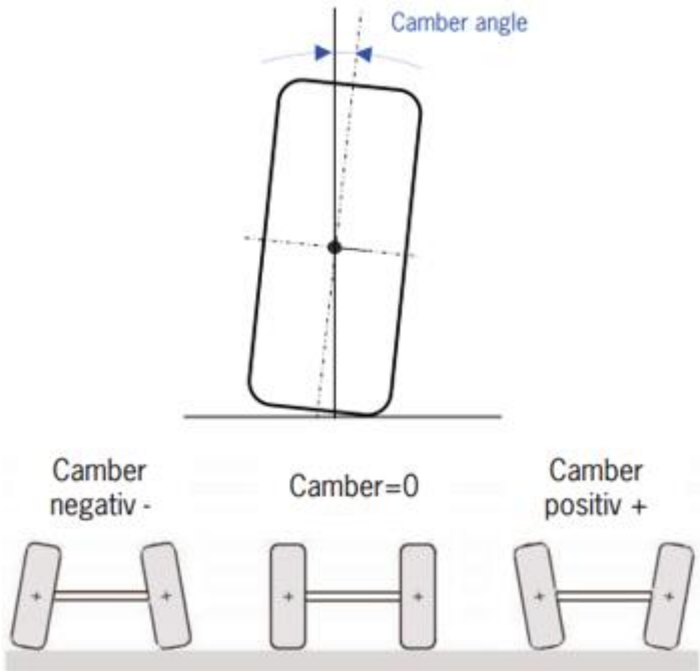
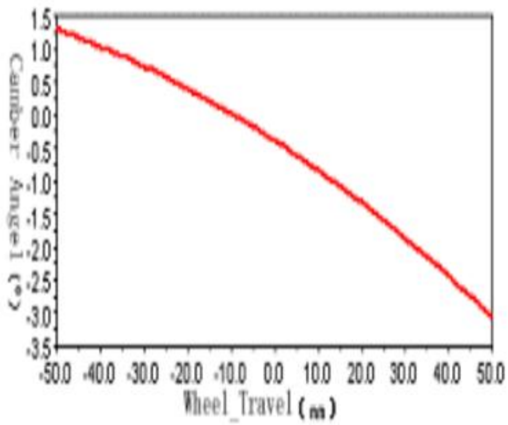


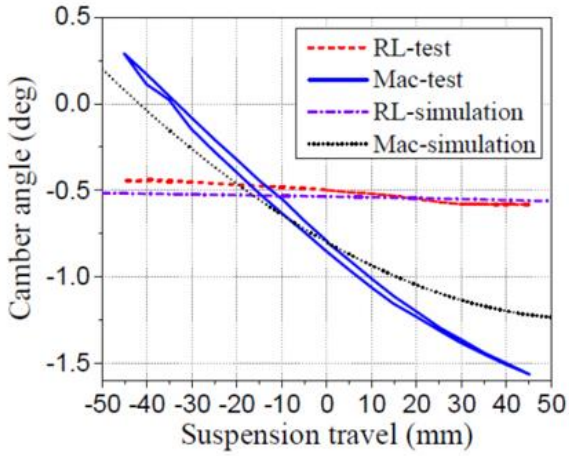

Figure 2.10: Camber Angle [source: Lecture Handouts by Porsche]

Lei Li *et al.*, (2007) [12] on analysis for kinematic characteristic of a front multi-link suspension exhibited the result shown in figure 2.11(a) and stated that the obtained characteristic i.e. a negative camber during jounce and positive camber during rebound can reduce the tire wear and improves handling quality.

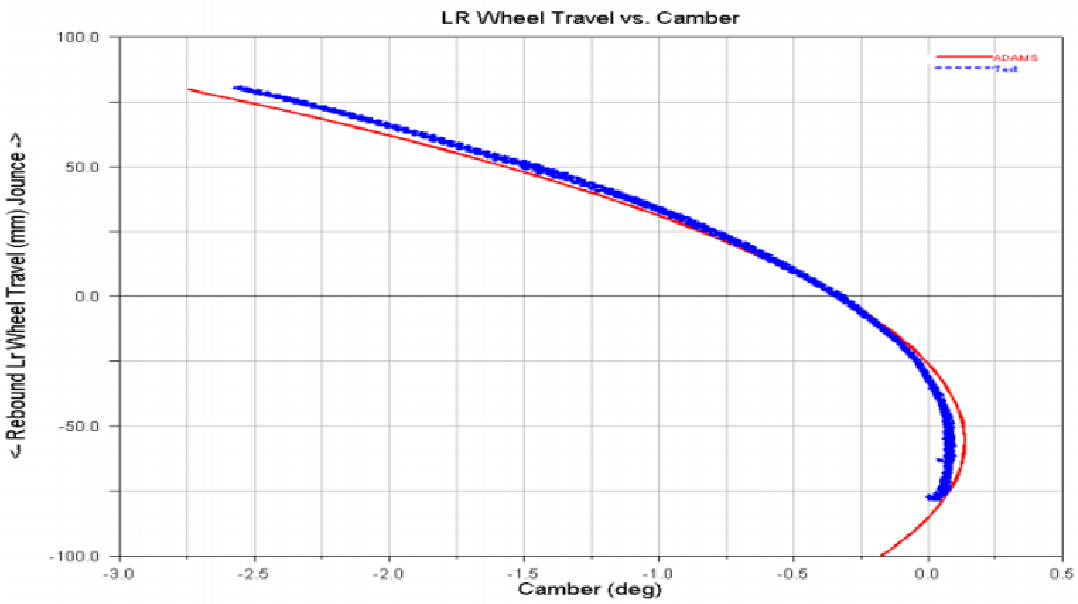
Liu *et al.*, (2015) [13] in their paper compared the kinematic characteristics of front suspensions Rectilinear (RL) with MacPherson Strut(Mac) based on ADAMS/Car simulation and test data shown in figure 2.11(b) and comes up with the conclusion that the variations of the alignment parameters of the rectilinear suspension are much smaller compared with the Macpherson suspension. So, it is easy to draw a decision that the rectilinear suspension is much better than Macpherson in kinematics.




(a)




(b)




(c)

Figure 2.11 Wheel Camber angel change with respect to wheel travel (a) for front Multi-link suspension[12] (b) for front Mac & RL suspension[13] (c) for rear double wishbone suspension[14]

Rao *et al.,* (2002) [14] developed a correlated full vehicle ADAMS model of the sedan using ADAMS/Pre and the test data and compares the virtual model suspension behavior with the real vehicles. Comparison of rear double wish bone suspension camber curves are shown if figure 2.11(c).

### 2.3.2 Toe Angle

Total toe is defined as the difference in the distance measured across the front of the tires compared to the distance measured across the rear of the tire, toe can also be defined as the measure of how far inward or outward the leading edge of the tire is facing, when viewed from the top. Excessive positive or negative toe will cause scuffing and wear on the shoulders of the tire. Toe is positive (toe-in) if the front of the tires is closer together than the rear of the tires. Similarly, it is negative (toe-out) when the rear of the tires is closer together than the front of the tires shown in figure 2.12.

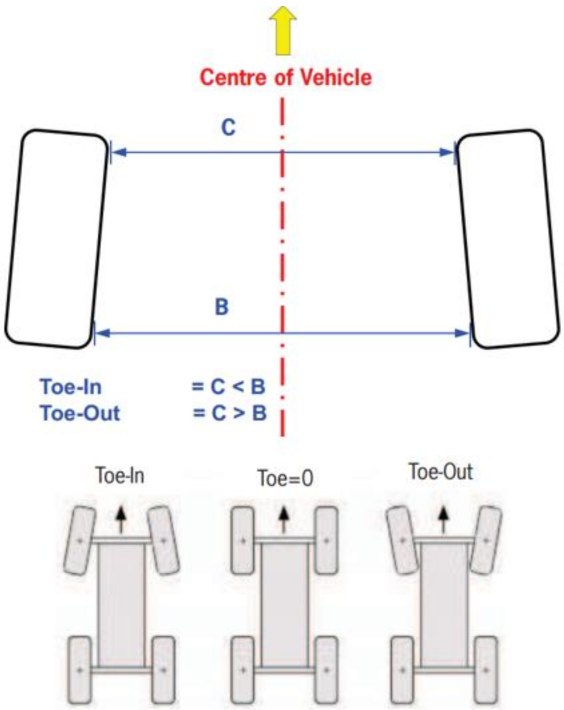
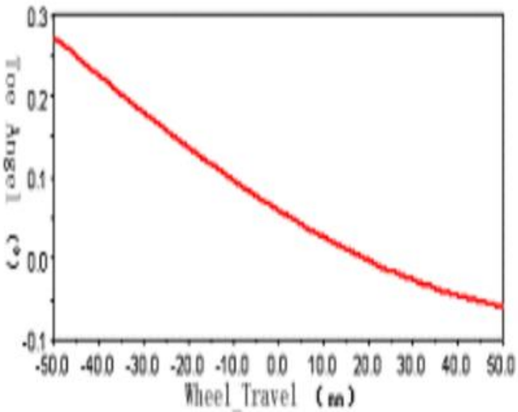


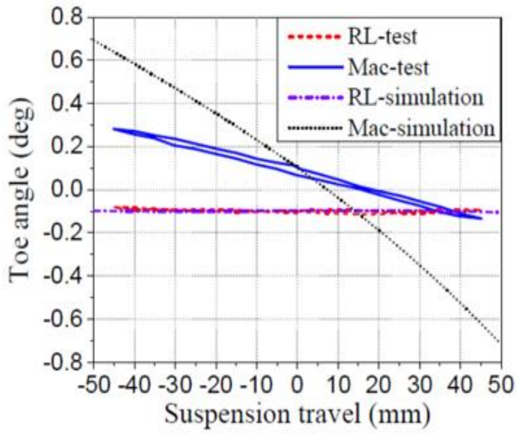

Figure 2.12: Toe angle [source: Lecture Handouts by Porsche]

Toe behavior with wheel travel for different suspensions are explained below from the papers referred in the section 2.2.2 respectively in figure 2.13 (a) (b) (c).

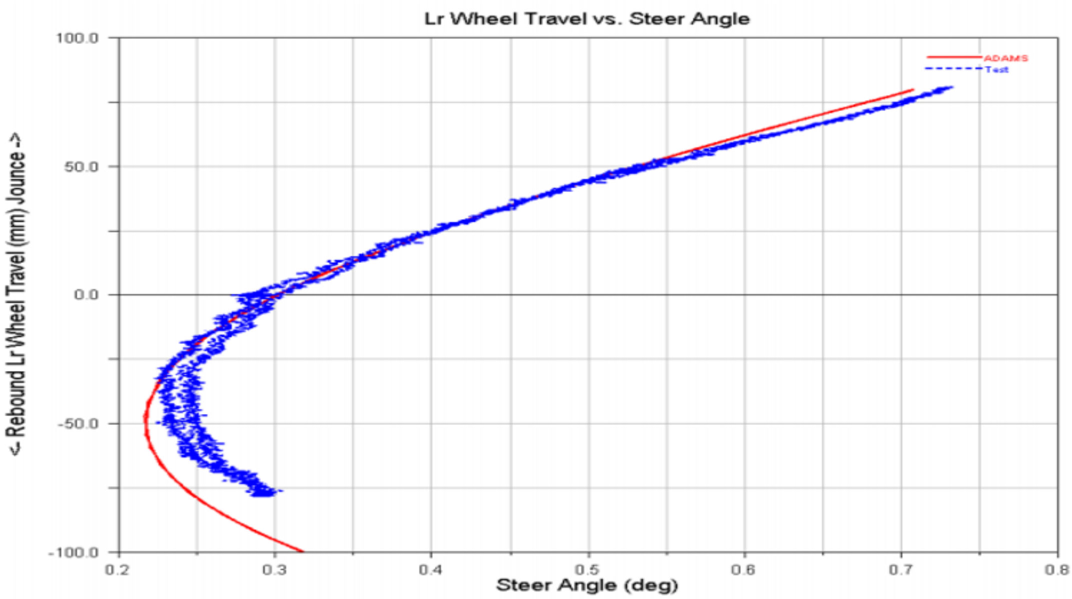
As the plots (a) (b) in figure 2.13 are of *front* Multi-link, Rectilinear and MacPherson suspension shows the same behavior irrespective of the suspension type i.e. a negative toe during jounce and positive toe during rebound but for rear suspension the toe behaves completely opposite which can be seen in the plot (c) in the same figure to reduce the tendency of oversteer said by Rao *et al.,* [14] in their paper.




(a)




(b)




(c)

Figure 2.13: Wheel Toe angel change with respect to wheel travel (a) for front Multi-link suspension [12] (b) for front Mac & RL suspension [13] (c) for rear double wishbone suspension[14]

### 2.3.3 Kingpin Inclination

The kingpin inclination is the angle σ which arises between the steering axis EG and a vertical to the road shown in figure 2.14, values on present passenger cars should be in between 11° to 15°30′. The precise position of the steer axis – also known as kingpin inclination axis – can only be determined if the center points E and G of the two ball joints are known [2].

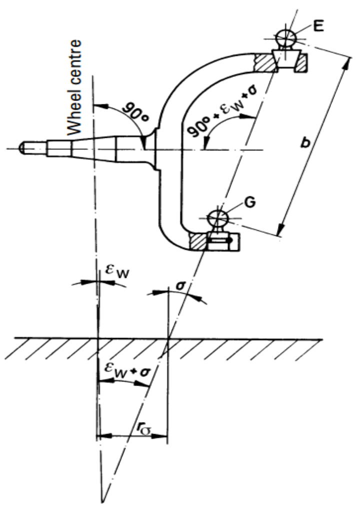



Figure 2.14: Kingpin Inclination [2]

### 2.3.4 Caster

Caster is the forward or rearward tilt of the steering axis compared to the vertical line viewed from side of the wheel, measured in degrees. It is positive when top of the steering axis is tilted backward and negative when it is tilted forward as shown in figure 2.14. The primary function of this characteristic is to improve directional stability and steering wheel returnability. Incorrect caster angle can cause excessive steering effort and even tire wear [2].

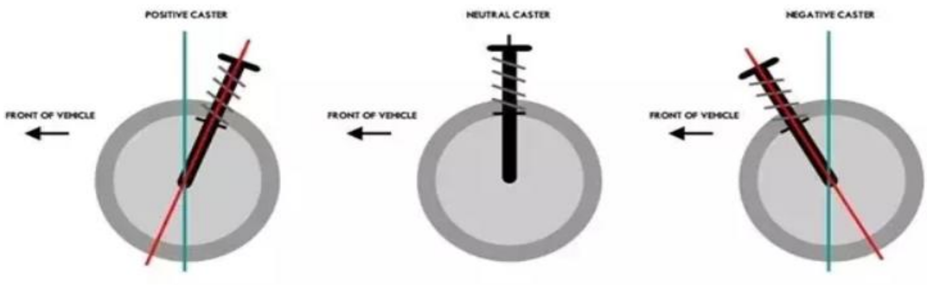



Figure 2.15: Castor Angle [source: Wikipedia]

## 2.4 Sensitivity of Wheel Suspension

Balike *et al.,* (2008) [18] discusses numerical method of sensitivity analysis for a multi-link suspension. The sensitivity analysis method reveals the influence of suspension joint locations in y and z direction i.e. lateral and vertical direction respectively on the wheel center trajectory and other kinematic parameters, to determine the influence of changes in the hard point location. It shows the upper front and lower rear control arms are found influencing the camber and roll center height the most, which helps in tuning of the suspension mechanism for the desired performance.

Rocca *et. al.,* (2016) [19] discusses the behavior of a rear multi-link suspension by varying the lengths of the linkages with respect to kinematic characteristics. In precise by varying only the joints coordinates on the chassis, to understand the influence of the length of individual link on a specific characteristic of the suspension, such as the steer angle and camber. After post-processing the sensitivity study it was highlighted the possibility of changing the various characteristic curves, and so the performances in terms of handling, by modifying the length of the various rods, eventually combining their effects.

Sub Yi *et al.,* (2014) [20] in their paper discusses two stage optimization process for better ride and handling performance for McPherson Strut in which the first stage is assigned for sensitivity analysis which is used for optimization for the independent suspension by Hyundai Motors. The sensitivity study using analysis of variance (ANOVA) tool is used, to reduce the variables for the optimization process then a multi-objective optimization is carried out for K&C characteristic using the optimization algorithm, Sequential Two-point Diagonal Quadratic Approximate Optimization (STDQAO) presented in the optimization software PIAnO is selected.

## 2.5 Study of Suspension Optimizations

Li *et al.,* (2007) [12] simulates multi-link suspension in ADAMS/Car. Studies the influence relations among multi-link suspension structure parameter and the wheel location parameter, wheel track, roll center. On the base of those parameters they aim for improving the ride safety and comfort by building multi-link suspension optimization model and by automating the optimization design by using ADAMS/Insight. The result showed that integrative use of ADAMS/Car and ADAMS/Insight in the kinematics analysis and optimized design of the suspension is quick and effective to improve the drive safety and comfort. The design method can be used in the design of multi-link suspension conveniently.

Chui *et al.,* (2009) [21] discusses about design problems and automated optimization of K&C characteristic and bushing stiffness of an independent wheel suspension for those problems. To execute an automatic analysis process, K&C analysis and handling simulation are sequentially integrated using by process integration and design optimization. This is done using an optimization software PIAnO which has in built function-based approximate optimization technique called Progressive Quadratic Response Surface Method (PQRSM) algorithm.

Qin *et al.,* [22] in their paper simulates the ADAMS model for McPherson strut to obtain the kinematic behavior and optimizes it using ADAMS/Insight. The optimization is carried out to reduce the tire wear, keeping side slip angle as the objective function of the linkage parameters and the side slip angle is minimized for better K&C characteristics. The coordinates of two inner points of the tie rod are changed through using optimized design parameters.

Hall et al., (2014) [23] automated the optimization process using MATLAB for K&C correlation of primary model, optimized model and experimental data. The paper explains how to link ADAMS/Car using command prompt, creating modified simulation code of ADAMS/Car for MATLAB. Running the experiment by integration of MATLAB and ADAMS/Car.

# 3. Suspension Modeling & Simulation

## 3.1 ADAMS/Car

The MSC ADAMS suite of software is extensively used in the automobile industry to perform up-front analysis of vehicle and suspension characteristics from the point of view of vehicle dynamics. It is also used for accident reconstruction and analysis, to understand the cause of the accident and improve the safety of the vehicle. The concept rear integral-link suspension model is created in MSC ADAMS suite, using the graphical user interface of ADAMS/Car to facilitate ease of modification later in the design. ADAMS/Car has two interfaces, standard interface is the first one, that can be used to quickly build computer models of a full-vehicle or a vehicle subsystem, run typical automotive events and postprocess the data for analysis and secondly template builder that is used to build templates for vehicle subsystems [14] [15].

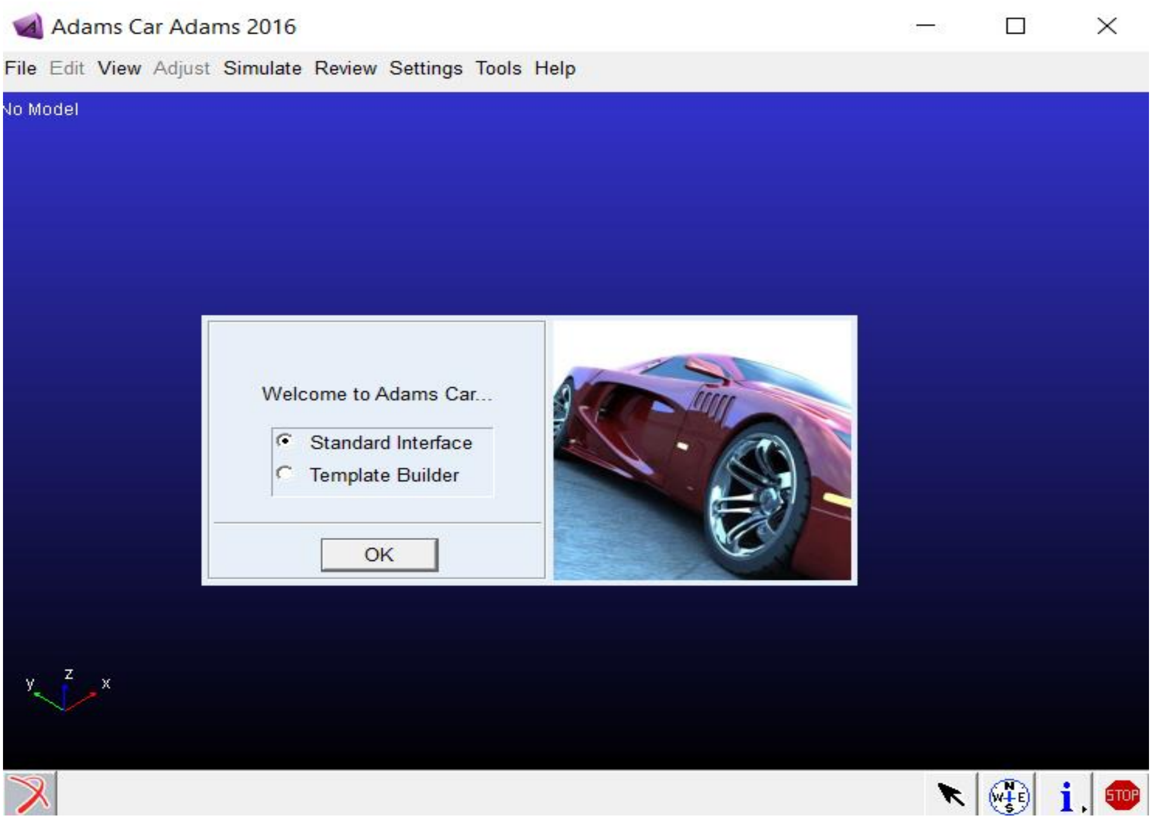



Figure 3.1: Adams/Car User Interface

### 3.1.1 Modeling/Simulation Overview

Starting with creating a template, hard points denoting the various key locations of the suspension system in the template. This is followed by creating links using those hard points, and finally adding joints and constraints between the links which are rigid bodies to complete the geometry are done in template builder. Followed by creating a subsystem and suspension assembly/full vehicle assembly in the standard interface for simulation.

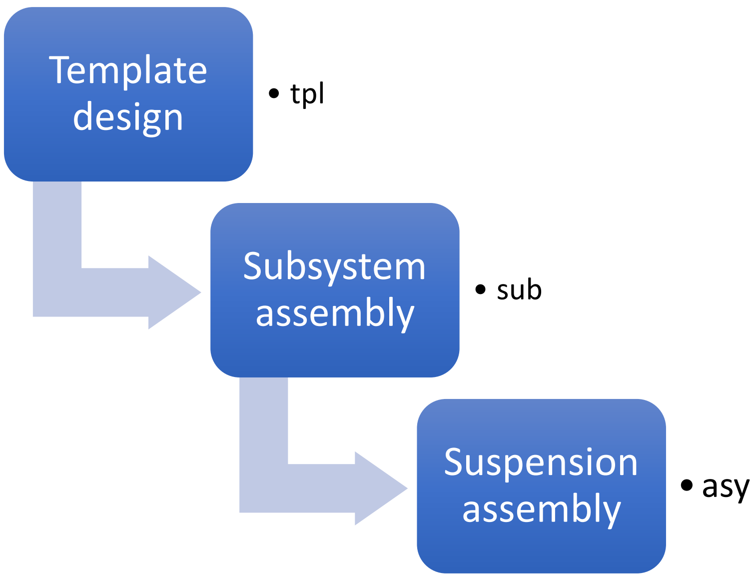



Figure 3.2: Modeling Flowchart

The above flowchart's each level is described below, which is needed to build the suspension model:

- *Template* defines vehicle sub-assemblies topology (that is, how the parts and joints fit together in the model, how information is transmitted, and so on). For example, a template could be a suspension type, which can be defined either as front and/or rear.
- *Subsystem* is a mechanical model that references a template and tailors it by supplying parameters that adjust the template (for example, locations that define part dimensions and spring/bushing stiffness). These models

are usually a major system of your vehicle, for example, front suspension, steering system, and body. The subsystem is a specific instance of the template in which the user has defined new hardpoint positions and property files.

- *Assembly* is a list of subsystem/subsystems and a single test rig combined in a vehicle or suspension assembly. A test rig is necessary to provide an actuation, in one's model, for analysis.

## 3.2 Modeling

A simplified kinematic scheme for rear ILS was drafted before the building the template in the template builder which would typically illustrate items such as the parts, joints, imparted motions and applied forces. The model includes other elements that include, for example, springs, dampers and beams etc. The drawing of a schematic is an important first step as it helps not only to plan the data that will need to be collated but more importantly to estimate the degrees of freedom in the system and develop an understanding of how the mechanism works eventually easing the template building process [15]. Figure 3.3 shows the structural sketch of rear ILS. It consists of a UCA (camber adjuster) H-I, Lower H Arm B-E-T-D, Tierod (toe adjuster) G-F, Integral Link U-T, Damper J-K, Wheel upright A-B-F-J-U.

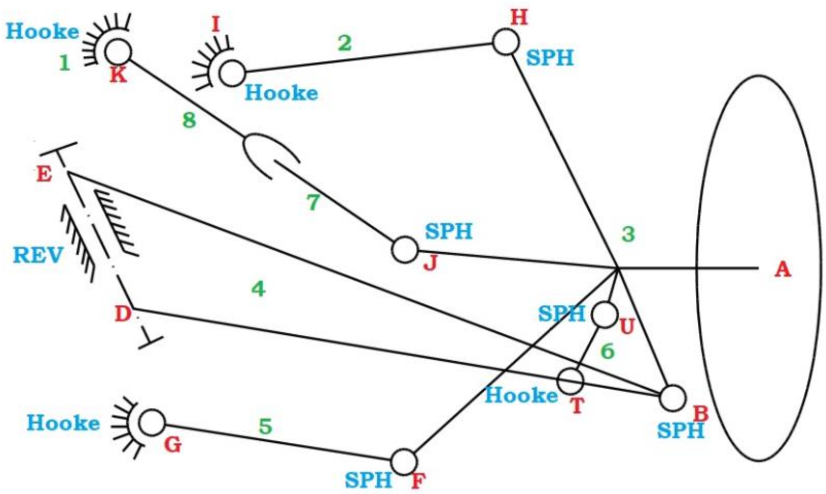



Figure 3.3: Kinematic Scheme

The scheme has 8 links(n), 5 spherical joints, 4 hooke joints, 1 revolute joint and 1 cylindrical joint and a parasitic link which is found in the damper. So, the DOF for the system is 1 calculated which was calculated using $Eq3.1$.

$$DOF = 6(n-1) - (SPH * 3) - (Hooke * 4) - (CYL * 4) - (REV * 5) - (parasitic\ link) \qquad (Eq3.1)$$

After the kinematic scheme was created, the following steps are carried out for modeling the suspension in the template builder.

- Defining hardpoint location
- Making construction frame
- Building general parts (rigid bodies)
- Building suspension geometry (links, arm etc.)
- Defining force elements (damper & spring)
- Making joints and constraints for suspension geometry
- Defining input and output communicators

### 3.2.1 Hardpoint Location

The ADAMS/Car pre-defined coordinate system was used while modeling the suspension, where the X-axis points back along the vehicle, the Y-axis points to the right of the vehicle and the Z-axis is up [15].

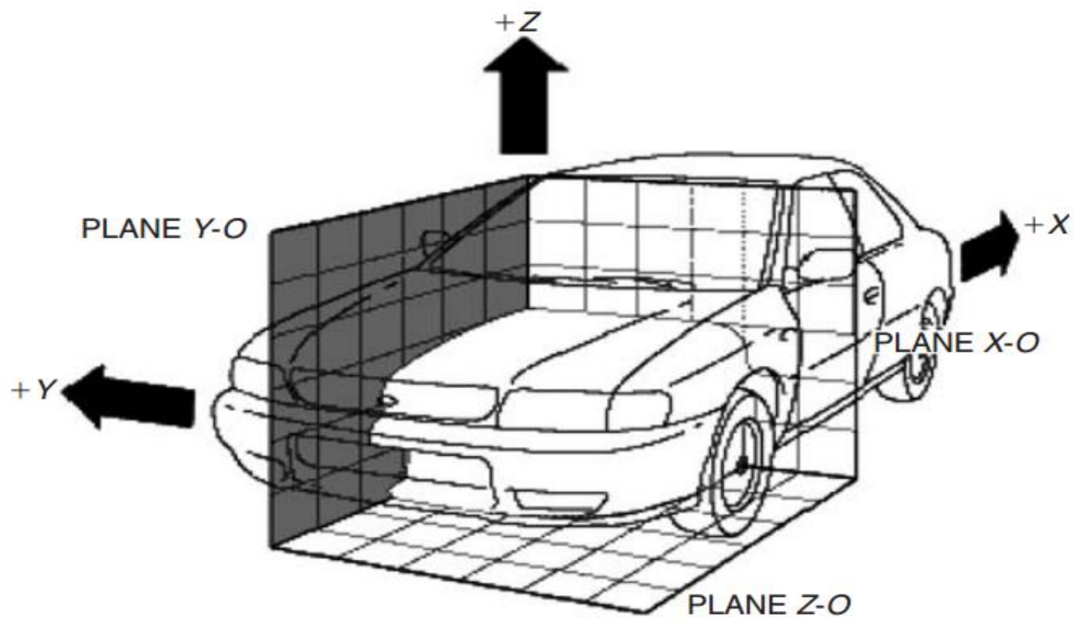



Figure 3.4: Co-ordinate reference used by ADAMS/Car [15]

The data for the hard point locations were generated by Ricardo s.r.o to build the concept model of the rear ILS. The generated co-ordinates also called as suspension geometry were comparable to as of Ford Edge SEL 2.0 Ecoboost for benchmarking the concept product under development, the suspension geometry of reference vehicle was generated from a CAD model by design department of Ricardo s.r.o. Table 3.1 lists the hard point coordinates of the left rear side suspensions hard points, these coordinates are defined with respect to the origin (0,0,0) which is at the center of the front axle. The right side hard points are coded as mirror images of left points automatically by the software. Hard-coding the right side as a mirror image of the left makes it simpler to make changes in the suspension geometry, as changing the coordinates of a point on the left side automatically maintains the symmetry of the right side [11]. By using these co-ordinates, suspension geometry and construction frames was generated in ADAMS/Car template builder for the modeling of the rear ILS. Hardpoints and construction frames are the ADAMS/Car elements that define all key locations and orientations in the model as they are the most elementary building blocks that is used to parameterize higher-level entities. Hardpoint locations define most parts and attachments. Hardpoints are only defined by their coordinate location. Coordinate frames are defined by their location and orientation.

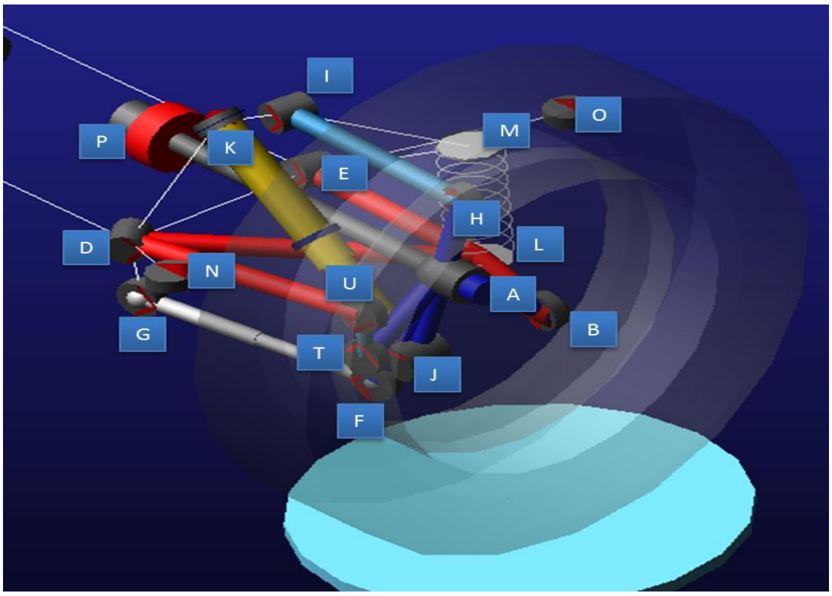



Figure 3.5: Reference image of CAE model rear ILS [source: Ricardo s.r.o]

| | | | | Coordinates (centre) | | |
|---|---|---|---|---|---|---|
| **Point** | **Description** | **Part** | **Part** | **X** | **y** | **z** |
| A | Wheel Centre | Wheel hub | Wheel | 2950.000 | -815.000 | 30.300 |
| B | LCA rear outer | LCA | Upright | 3052.900 | -765.700 | -64.900 |
| D | LCA front inner | LCA | Subframe | 2789.700 | -349.100 | -10.900 |
| E | LCA rear inner | LCA | Subframe | 3105.400 | -292.100 | -69.000 |
| F | Tierod outer | Tierod | Upright | 2863.300 | -684.200 | -140.100 |
| G | Tierod inner | Tierod | Subframe | 2786.839 | -366.970 | -104.400 |
| H | UCA outer | UCA | Upright | 2977.000 | -730.000 | 155.000 |
| I | UCA inner | UCA | Subframe | 2995.000 | -402.200 | 127.800 |
| J | Strut lower mount | Strut | Upright | 2901.000 | -712.542 | -97.000 |
| K | Strut upper mount | Strut | Subframe | 2883.358 | -413.215 | 191.708 |

| Point | Description | Part | Part | X | y | z |
|---|---|---|---|---|---|---|
| M | Spring upper mount | Spring | Subframe | 3113.351 | -579.612 | 113.838 |
| N | Subframe front | Subframe | Body | 2700.000 | -530.000 | 58.250 |
| O | Subframe rear | Subframe | Body | 3300.000 | -530.000 | 58.250 |
| P | Driveshaft inner | Driveshaft | Tripot | 2988.077 | -169.475 | 0.300 |
| T | Integral link lower | Integral link | LCA | 2840.000 | -697.600 | -75.000 |
| U | Integral link upper | Integral link | Upright | 2840.000 | -701.200 | 5.000 |

Table 3.1: Hardpoint Co-ordinate

### 3.2.2 Multi-Body Modeling of Integral-link Suspension

In ADAMS/Car templates are parameterize models which consists topology of vehicle components [11]. Suspension modeling was carried out in the template builder by creating a template for the concept suspension. The suspension parts are created using rigid bodies with hardpoints references. These are known as general parts which are movable parts which cannot deform and have mass, inertia properties. The hard points already created mark the end points of each of the suspension links, these points are used to create the suspension geometry. Each suspension element is modeled as a separate part, connected to other parts through joints. Figure 3.6 shows the model of concept suspension in the template builder of ADAMS/Car.

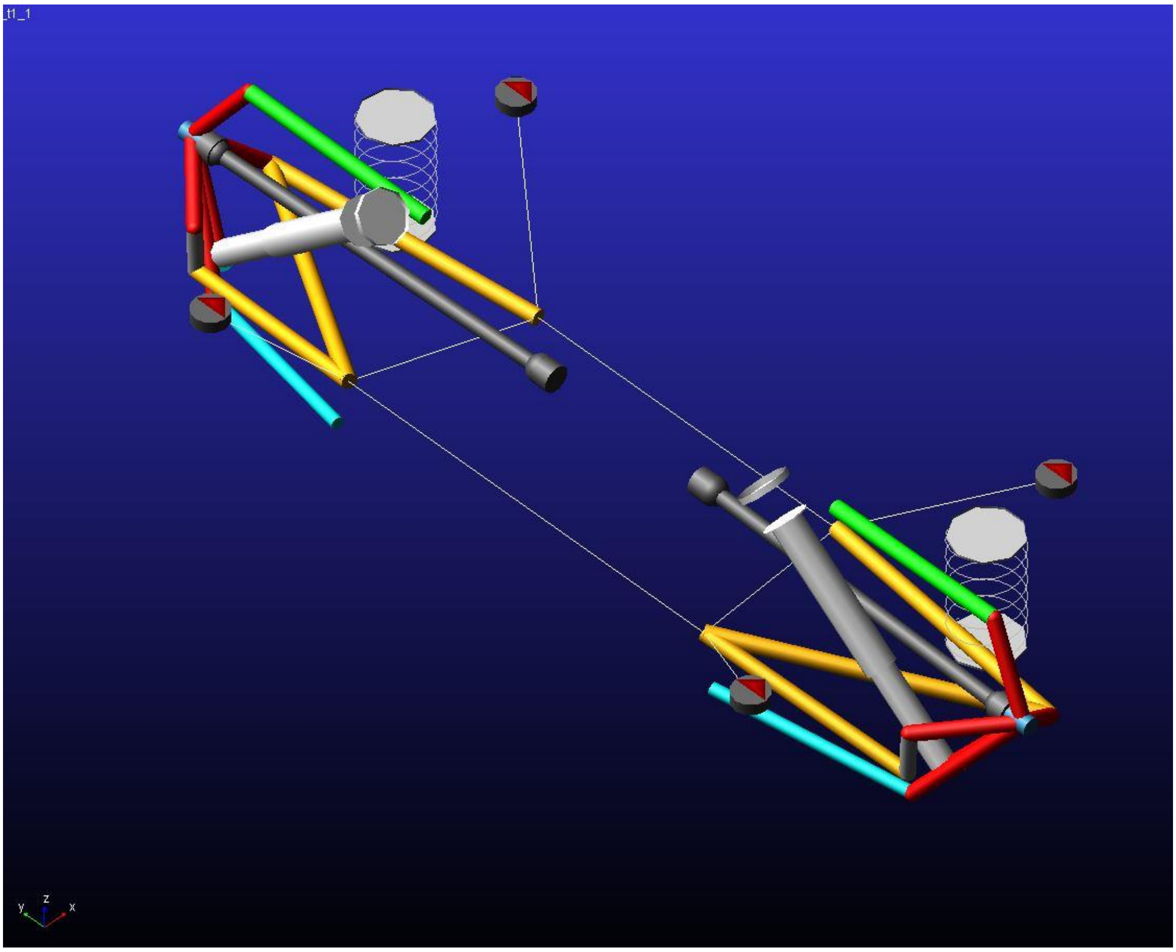



Figure 3.6: Front ISO view of multi-body model of concept rear ILS

The concept suspension is an integral link suspension for each side of the rear wheels consist of wheel hub/spindle, upright/knuckle, upper control arm, lower control arm, integral link, tierod, driveshaft and a common subframe as general part whose mass and inertia properties of the body are incorporated in the model. Once the parts are created, their mass and inertia properties are defined. The data for the mass and inertia properties were also measured for each component and provided by Ricardo s.r.o., the mass and inertia properties of all the parts of the left wheel suspension are shown in the Table 3.2. and the right wheel suspension has the same values.

| Number | Name | Mass (kg) | Ixx ($kg\ m^2$) | Iyy ($kg\ m^2$) | Izz ($kg\ m^2$) |
|---|---|---|---|---|---|
| 1 | Subframe | 34.2 | 655.005 | 845.012 | 249.00 |
| 2 | Wheel Hub | 3.35 | 0.09030 | 0.06888 | 0.04230 |
| 3 | Upright | 4.2 | 0.20124 | 0.08905 | 0.06895 |
| 4 | UCA | 3.89 | 0.20245 | 0.0925 | 0.07332 |
| 6 | LCA | 13.18 | 0.62095 | 0.34797 | 0.1954 |
| 7 | Integral Link | 1.4 | 0.0100 | 0.01350 | 0.00987 |
| 8 | TieRod | 2.0 | 0.01400 | 0.01800 | 0.01800 |
| 9 | Driveshaft | 5.26 | 0.26400 | 0.21200 | 0.15820 |

Table 3.2: Body Mass and Inertia data

Figure 3.3 shows the simplified scheme of the suspension geometry with the joints and the connection of each part to other and Table 3.3 shows the individual suspension component positions and topology modeling for each suspension and the connectivity of each component with other components via joint type as well as the number of degrees of freedom. The colour and no. for the component can be referred using Figures 3.3 & 3.6

| **Point** | **Component Name** | **Connected Part** | **Joint** | **DOF** |
|---|---|---|---|---|
| B | LCA outer | Upright | SPH | 3 Rotation |
| D | LCA front inner | Subframe | Rev | 1 Rotation |
| E | LCA rear inner | Subframe | Rev | 1 Rotation |
| F | Tierod outer | Upright | SPH | 3 Rotation |
| G | Tierod inner | Subframe | Hooke | 2 Rotation |
| H | UCA outer | Upright | SPH | 3 Rotation |
| I | UCA inner | Subframe | Hooke | 2 Rotation |
| J | Strut lower | Upright | SPH | 3 Rotation |
| K | Strut upper | Subframe | Hooke | 2 Rotation |
| T | Integral link lower | LCA front inner | Hooke | 2 Rotation |
| U | Integral link upper | Upright | SPH | 3 Rotation |

Table 3.3: Topology modeling for Integral-link suspension system

### 3.2.3 Spring and Shock Absorber

The coil springs are modeled using linear rates derived from a curve fit of spring load vs. displacement, between two points which are the mounting points of subframe and LCA by ADAMS/Car software, using available property file. The measured free lengths of the springs are defined between the hardpoint location of spring upper and lower mount. The damper also used property file defined by ADAMS/Car in the shared *acar folders*. The damper is aligned at an angle because it was decided earlier by Ricardo s.r.o, keeping in mind that such a design would give more boot space if attached to the subframe.

### 3.2.4 Communicators

The last part in template building is assigning communicators, which is very important as they are the key elements in the ADAMS/Car. Communicators enable subsystems that make up the assembly to exchange information with each other and with the test rig. An assembly requires two directions of data transfer between its subsystems. To provide for these two directions of data transfer, ADAMS/Car has two types of communicators first is the Input communicator which demand information from other subsystems or test rig, second is the Output communicator which provides information to other subsystems or test rigs.

### 3.3 Simulation

After the model of suspension is built in the template builder, the user interface for the ADAMS/Car is switched to standard interface from template builder. For initiating the simulation, a subsystem is generated using the template followed by the suspension assembly which completes the entire modeling process described in section 3.1.1.

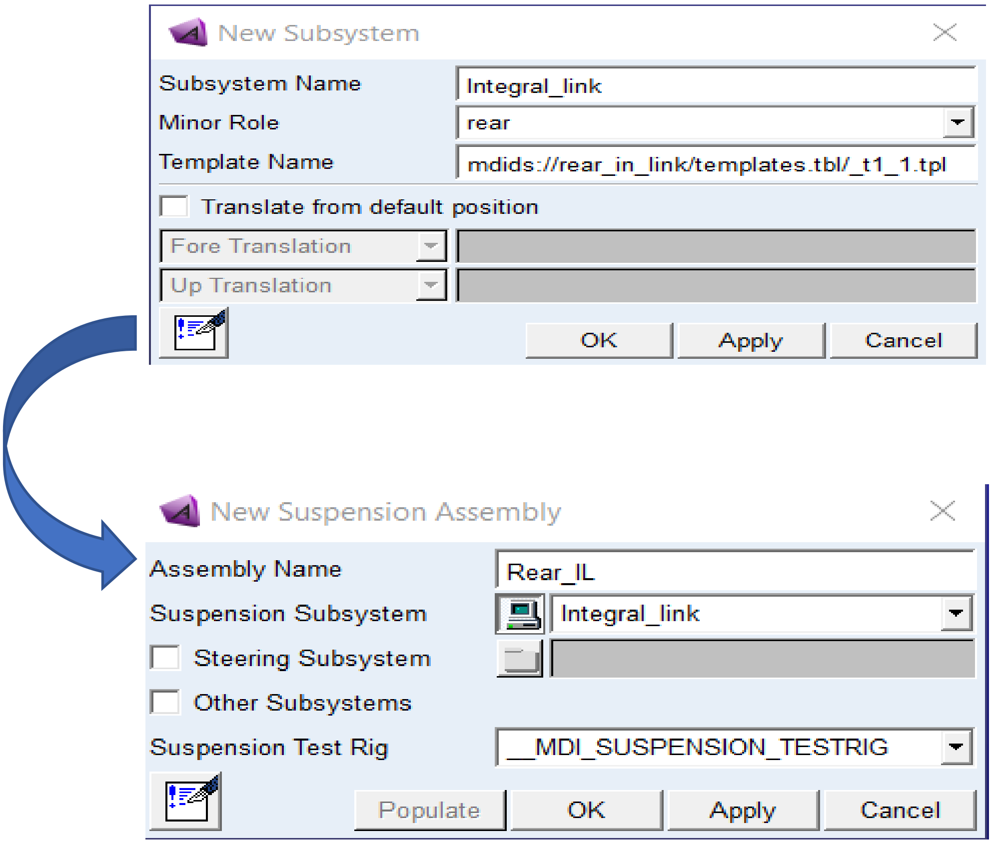



Figure 3.7: Defining Subsystem & Assembly

Quasi-static tests are used to validate the model. These tests validate the suspension parameters and the suspension K&C characteristic of the vehicle. The kinematic tests measure wheel position changes that occur due to vehicle position changes such as ride height and roll while longitudinal and lateral forces are zero. Compliance tests measure wheel position changes due to longitudinal and lateral force and aligning moment inputs [14].

### 3.3.1 Bounce Test

In the bounce tests also known as parallel wheel travel analysis or vertical wheel displacement test, a static load is applied to the chassis of the vehicle through a bar attached to the frame. The steer/toe angle, camber angle, caster angle and other wheel variations are measured against the suspension deflection for the suspension to be validated/tested [15].

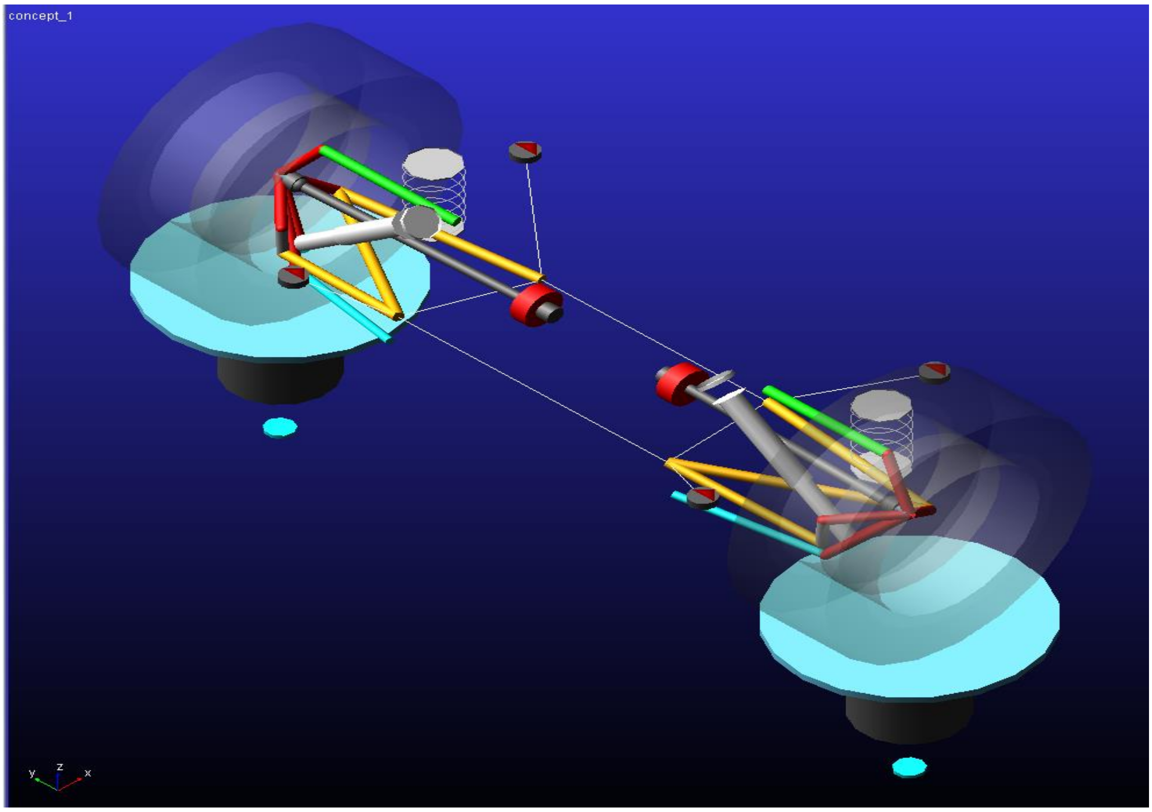



Figure 3.8: Rear ILS assembly on Test rig

Suspension parameters are setup before running the parallel wheel travel, which are assigned according to the specification of Vehicle X:

- Tire Model: ADAMS/Car has per defined property file for different range of tire and wheel sizes. Pac_2002_255/40R19 was used as it resembled the wheel and tire property of vehicle type for which the suspension is being developed.

- Vehicles sprung mass was calculated and set to 1056 kg as the mass of rear axle is 48% of curb weight i.e. 2200kg.
- Wheel base was set to 3004 mm.

The simulation was run in kinematic mode alone as the suspension is modeled as a pure kinematic suspension without compliances i.e. bushings as no data for compliances were available. The simulation parameters for bounce test are set as: no. of steps 50, bump travel 100mm and rebound travel = -100mm. Then the parallel wheel travel simulation is executed, and ADAMS/Solver runs the simulation only if the multi-body model doesn't have any design errors and gives the output in ASCII file which are result sets. ADAMS/Post-Processor is used to process the result sets and request files generated during simulation for plotting of curves.

## 3.4 Kinematic Characteristics

The simulation results of rear ILS are discussed in this section, which includes:

- Camber analysis
- Toe analysis
- Caster analysis

### 3.4.1 Camber Analysis vs Wheel Travel

Both left and right wheels show similar camber data. The variations of the camber angle due to the vertical wheel movement for left wheel suspension is shown in Figure 3.9 which follows the same variation trend compared to literatures studied in section 2.3.2. During maximum jounce and rebound, the camber value increases to -3.9° angle and decreases to 2.89° respectively, the camber angle increases continuously to negative side during jounce and towards positive when the suspension rebounds. During static height, the camber value is -0.0079° ≈ 0° and the gradient obtained is -34.4°/m which can be seen in ADAMS/Post-processor as it calculates and displays it, which is calculated by the equation of a line (Eq.2) by assuming that the curve is behaving linearly in the range of +20 mm jounce and rebound -20 mm.

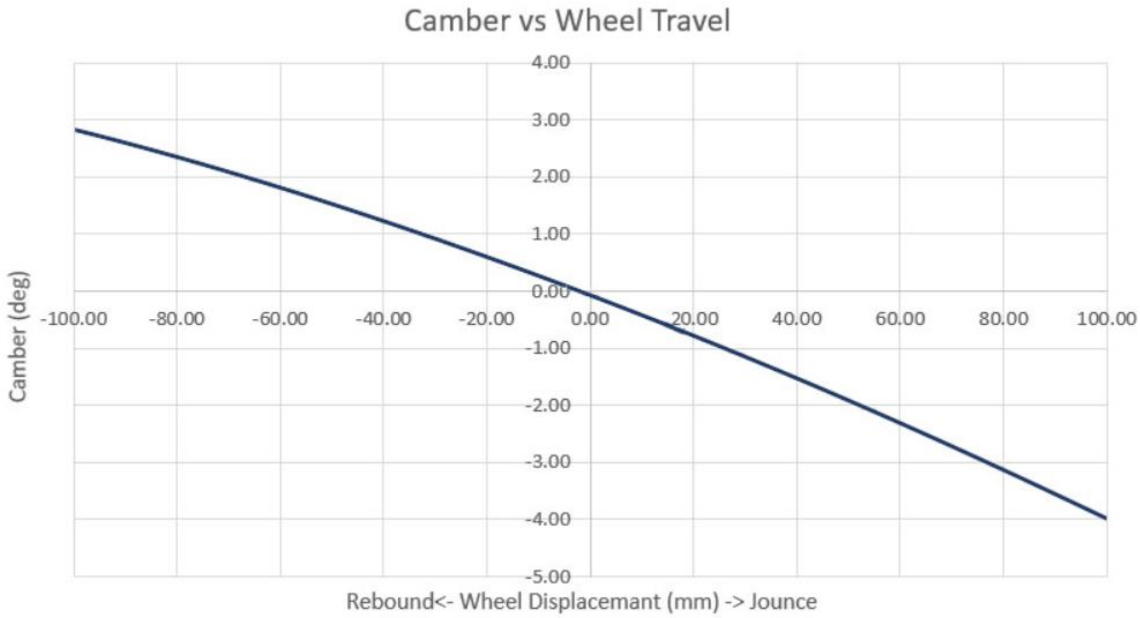



Figure 3.9: Left Wheel Camber vs vertical wheel displacement

$$slope = \left( {(y2 - y1)}/{(x2 - x1)} \right) * 1000 \qquad \text{.....................} \textit{ Eq.2}$$

### 3.4.2 Toe Analysis vs Wheel Travel

Both left and right wheels show similar toe data. The variations of the toe angle due to the vertical wheel movement for left wheel suspension is shown in Figure 3.10 and the curve obtained also shows similar variation trend compared to literature in section 2.3.2. The rear suspension toe variation should generate positive toe during jounce and negative toe during rebound [2]. During static height, the toe value is 0.00035°≈ 0° and the gradient is 11.65°/m which is calculated using Eq.2 mentioned above by the post-processor. During maximum jounce and rebound, the toe value increases to 1.97° angle and decreases to -0.49° respectively which satisfies the above conditions.

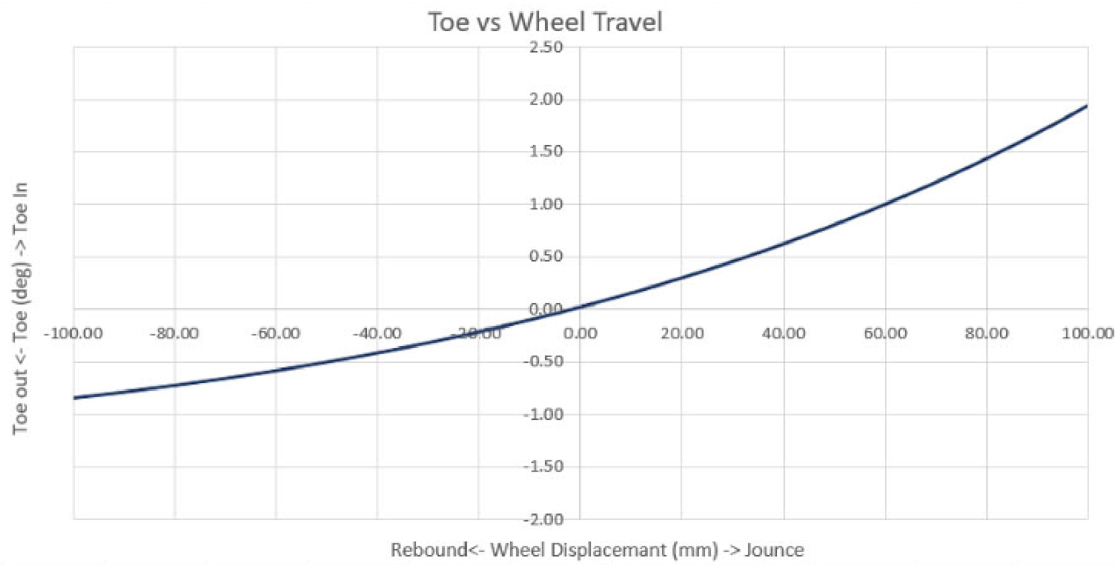



Figure 3.10: Left Wheel Toe vs vertical wheel displacement

### 3.4.3 Caster Analysis vs Wheel Travel

Similar camber and toe, caster data is same for right and left wheel suspension. The variations of the caster angle due to the vertical wheel movement for left wheel suspension is shown in Figure 3.10. The obtained plot shows a linear behavior in the caster change which is the function of steer axis and deigned model. During static height, the caster value is -13.11° and the gradient calculated is -55.03°/m using Eq. 2 by the post-processor.

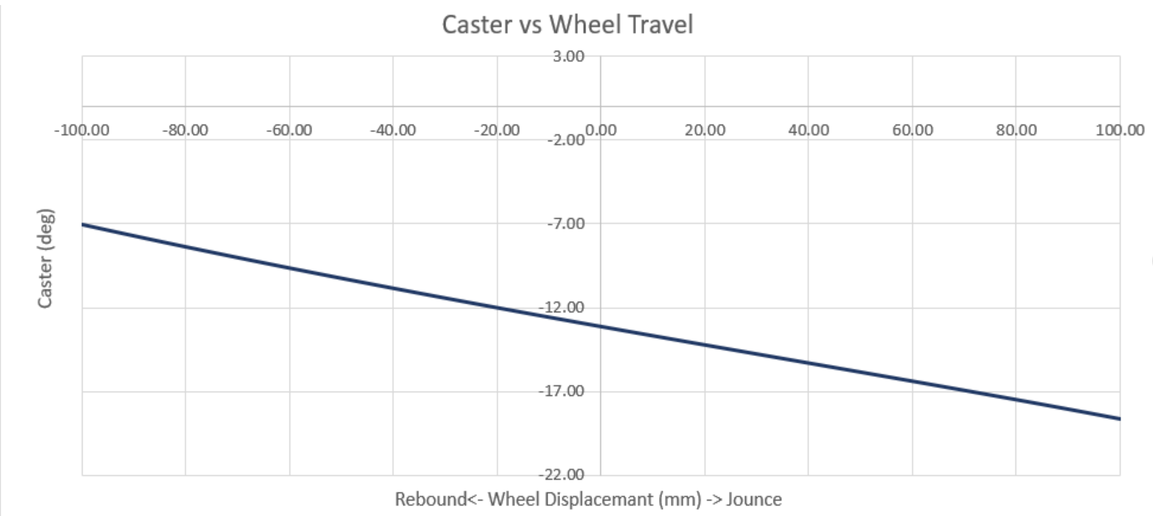



Figure 3.11: Left Wheel Caster vs vertical wheel displacement

The analysis of the computed result shows the following:

- The camber curve for rear ILS shows the variation range which is more than -3° to 3° with static angle almost near 0°. This which will cause tire wear during cornering and hinder the directional stability. Therefore, the variation should be minimized during optimization.
- The change in toe setting from the negative to the positive value for any multi-link type suspension system and the rear ILS follows the same trend but the variation should be reduced as much as possible as it will improve the stability which was discussed by Kim *et al.,* [6] when comparing toe variation of Hyundai Genesis with competitive models in the similar variant of vehicle type.
- The caster angle curve shows linear behavior, the caster angle depends on the position of UCA ball joint and LCA ball joint which are the function of design element and steer axis.

# 4. Sensitivity Study and Optimization

## 4.1 Sensitivity Study

The sensitivity of a system to changes in its parameter values is one of the basic process in the analysis of a system and knowledge of sensitivity is very valuable in optimizing the design. In the case of suspension kinematic studies, it would be appropriate to consider the suspension hard points location in the space as system parameters.

### 4.1.1 Rear Integral-Link Suspension Sensitivity Analysis

M. B. Gerrard [7], in his paper described the study of multi-link suspension linkages behavior said that the change in variation for camber angle is related to the length and position of upper and lower control arm in vertical and lateral direction. The toe angle variation is due to the length of tie rod and inclination of toe link from knuckle to chassis. The caster angle variation is difficult to define on any specific position as it depends kingpin inclination as well on the knuckle design and all the links shows some affect in every kinematic characteristic.

For studying the behavior of kinematic characteristics with reference to the study of Gerrard [7] [19], the hardpoints positions of links connected to upright which are UCA Outer, LCA outer and Tierod Outer were varied in lateral, vertical direction. Changes for hardpoint in longitudinal direction was only done for LCA outer because it results in the variation of kingpin inclination as the steer axis during modeling was defined using the hardpoint location of LCA. The hardpoint locations for the inner side are varied because of design constraint as the joints connection to the subframe should be fixed and not to be varied.

The sensitivity analysis for my modeled rear ILS showed the following results:

1. By changing the z co-ordinate of UCA ball joint to +5mm from initial position the gradient and angle variation for the camber curve was increased and decreased when it was lowered to -5mm at the extreme position during jounce and rebound. Decreasing the length i.e. by changing

the y co-ordinate showed negligible variation but made the curve a bit linear as shown in figure 4.1, where the x-axis is camber angle and y-axis is wheel travel. The red curve is the initial characteristic of the rear suspension and blue and green curve shows the variation respectively as discussed above. The pink dotted curve is obtained when the geometry of toe link was altered together with camber link.

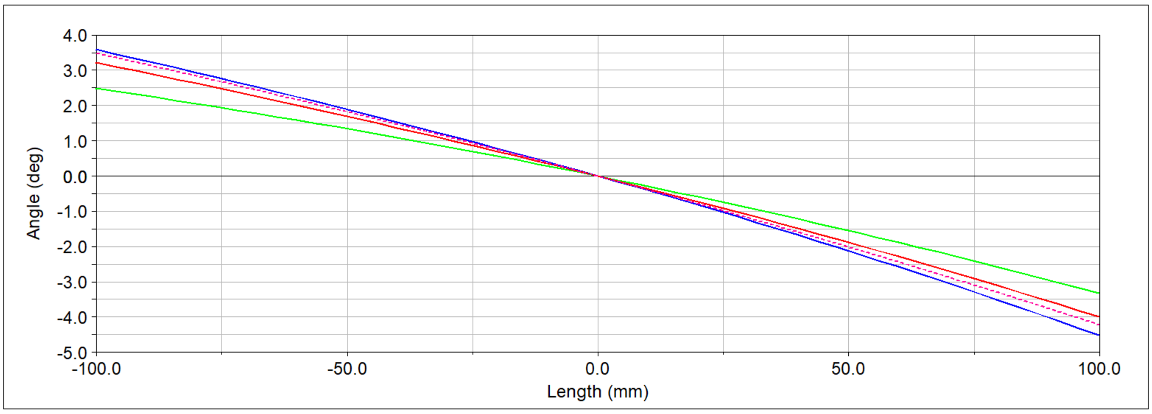



Figure 4.1: ADAMS/Postprocessor snapshot for Camber variation vs Wheel Travel

2. When z co-ordinate of Tierod outer was changed to +10mm from initial position, the gradient for the toe curve was increased and decreased when it was lowered to -10mm. Even the variation in position of UCA outer affected toe angle. The change in length of toe link by varying y co-ordinate of Tierod outer made the toe cure behave linear by reducing the end variation during jounce and rebound shown in figure 4.2 where the x-axis is toe angle and y-axis is wheel travel.

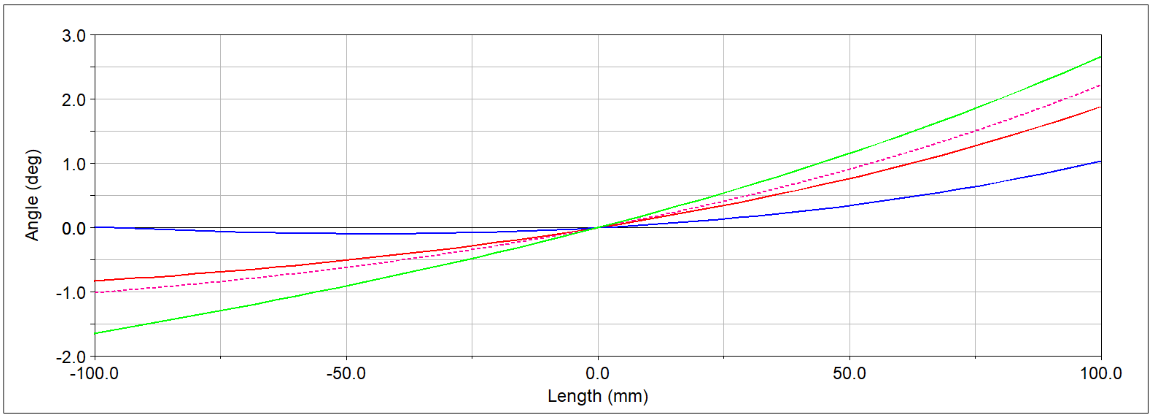



Figure 4.2: ADAMS/Postprocessor snapshot for Toe variation vs Wheel Travel

The red curve is the initial characteristic of the rear suspension and blue and green curve shows the variation respectively as discussed above. The pink dotted curve was obtained when the both the geometry toe link and camber link was altered together.

3. The change in x co-ordinate of LCA rear outer when shifted toward the side of wheel center by 15mm changed the caster curve drastically shown in figure 4.3 where the x-axis is caster angle and y-axis is wheel travel. The red curve is the initial characteristic of the rear suspension and blue and curve shows the variation when the geometry was altered. The pink dotted curve was obtained when the both the geometry toe link and camber link was altered together with LCA rear outers co-ordinate.

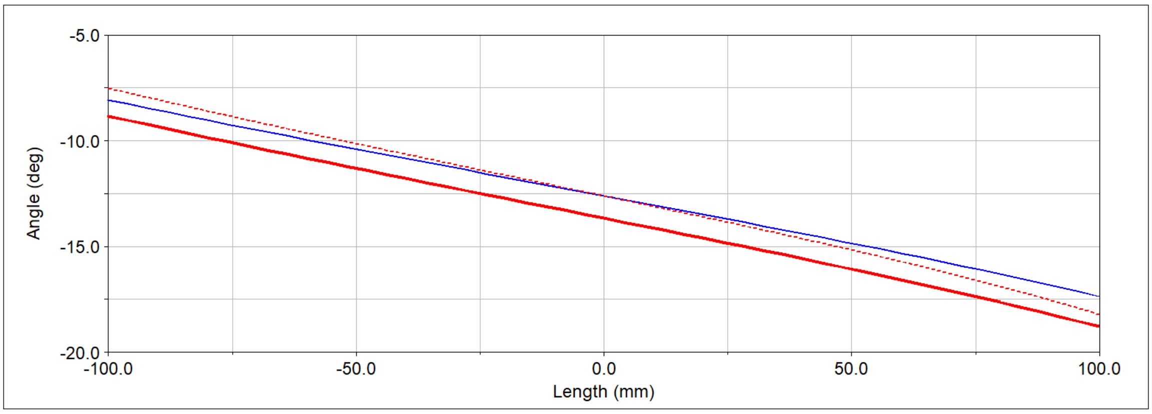



Figure 4.3: ADAMS/Postprocessor snapshot for Caster variation vs Wheel Travel

The study helped in choosing the specific input hardpoint variables which will affect the suspension geometry during optimization of the model. It was also observed that the what the literature [7] [19] explains is seen but the actual behaviors is different as the complete variation in the kinematics are combination of hardpoints of other linkages as well.

## 4.2 Optimization

Maximizing or minimizing some function relative to some set, often representing a range of choices available in a certain situation is known as optimization. The goal for optimization in this scenario is to solve and to get optimal design.

### 4.2.1 Optimization Using ADAMS/Insight

ADAMS/Insight is a stand-alone product/multi-objective optimization tool, part of MSC ADAMS suite of software that works with many other ADAMS products. Adams/Insight allows the user to design and analyze the results of sophisticated experiments for measuring the performance of the mechanical system. For simple design problems, ADAMS/Insight is used as an optimization tool to modify the behavior of the mechanical system to achieve the desired outcome by automating the simulation process. [24]

*Target Setting:*

The goal of the optimized suspension design is to achieve a better ride and handling performance compared to the initial model within a specific range of hardpoints and as close to the benchmarking model's performance target shown in table 4.1 below.

| Characteristic | Static Value (deg) | Gradient (deg/m) |
|---|---|---|
| Toe | 0.25 | 3.0 |
| Camber | -1.5 | -25.0 |
| Caster | -5.0 | -5.0 |

Table 4.1: Benchmarking Target

*Process Integration:*

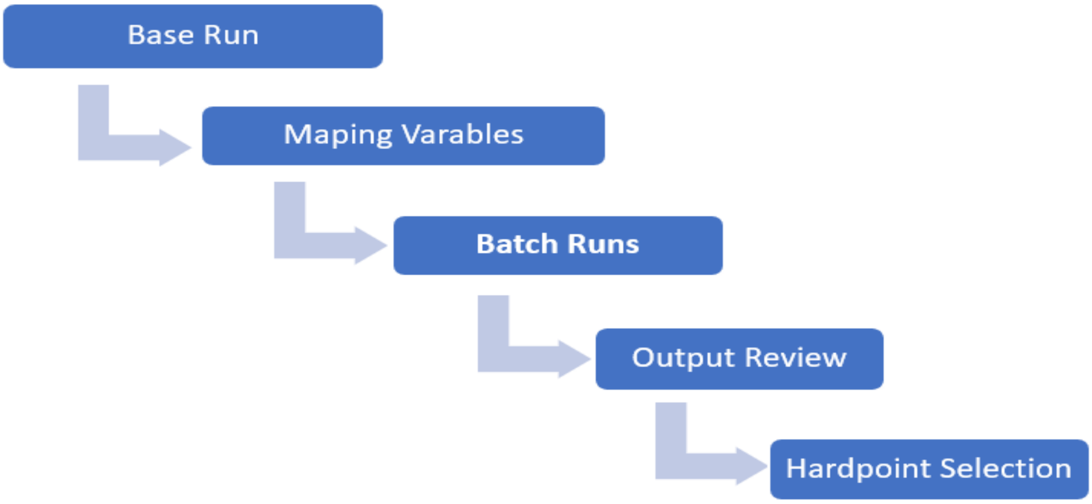



Figure 4.4: Process Integration

Optimization considers the design requirements of the suspension model, the design parameters which are to be manipulated must be determined and the simulations that quantify the design requirements should be executed in sequence. In addition, post-processing is necessary for simulation to transform the raw data into a form that specifies design requirements. To automate the suspension system analysis, this study conducted process integration in five steps: base run, mapping, batch runs, output review, and hardpoint selection.

*Base Run:*

For integrating ADAMS/Car with ADAMS/Insight the simulation for parallel wheel travel of the model is done again and save in a new directory. This is done because ADAMS/Car generates .ascii and .xml initial output file which contains the information of the complete model i.e. template, subsystem and the suspension assembly with the results set and the simulation script which is used to run the ADAMS/Car via command prompt, lastly the whole model with the result sets is exported to ADAMS/Insight A similar way to run the ADAMS/Car simulation by command prompt is discussed by Hall *et al.,* [23] where they automated the optimization process using MATLAB.

*Mapping Variable:*

The window for ADAMS/Insight opens 3-5 minutes later as the integration of both the software and exporting data consumes its. Mapping variable consists of following steps:

- *Assigning factors and constraint:* A factor is a design variable that vary in experiment. In this optimization model hardpoints location in x, y, z direction are factors and the range in which it the hardpoints will vary is the constraint. Because of sensitivity study the factors chosen are left sided (x, y, z) points of LCA rear outer, (y, z) points of UCA outer and (y, z) points of tierod outer. Only left side is chosen, because by varying the left points the right will vary automatically. The constraint for all design variables are assigned in a range which is important for wheel packaging and to get the best possible result could be achieved specifically in that set of range. The upper and lower limit i.e. the constraint was given by the design department according to the wheel design. The central value is the primary hardpoint coordinate, the upper and lower hardpoint constraint were different for every co-ordinate of the individual parts.

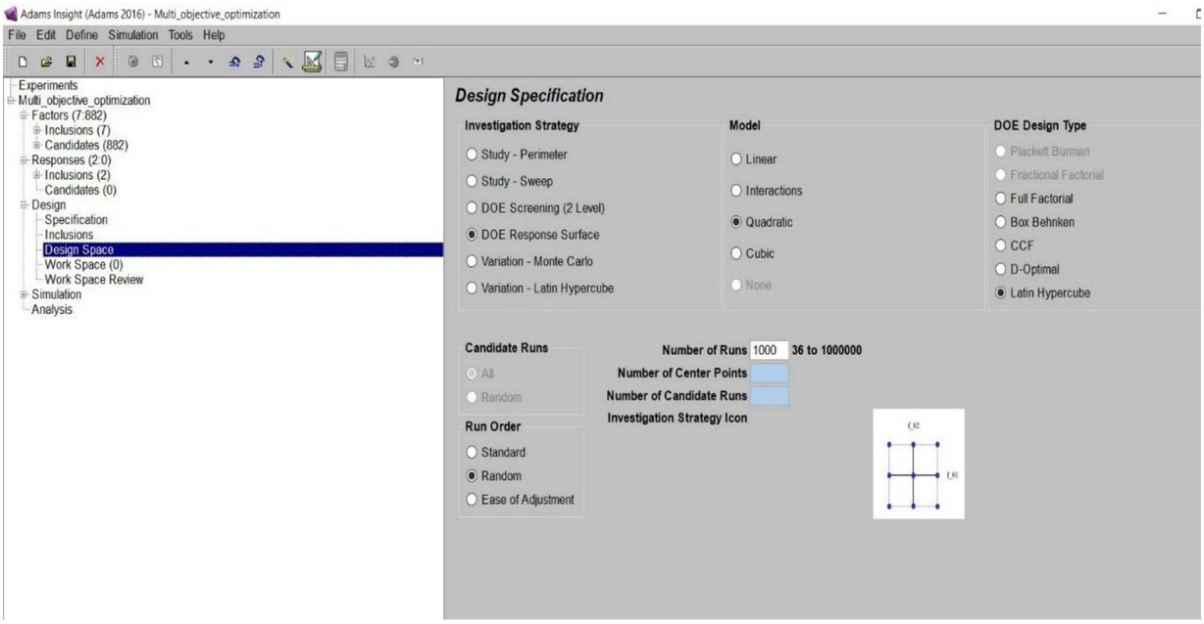



Figure 4.5: User Interface of ADAMS/Insight

- *Defining objective function:* A response can be considered the output or design objective/objective function. Camber and toe angle are defined as the response. Design objective was defined in the responses which was by minimizing the average variation for both camber and toe which was analyzed during sensitivity study at camber = -1.5° and toe = 0.25°.
- *Defining DOE type:* Latin Hypercube was defined because it uses as many values as possible for each factor. Each factor's values are randomly ordered so that each run has a random combination of factor values. [24] Continuous factors have a different value for each run. The values are equally spaced, running from the minimum value to maximum factor value. Discrete factors have a fixed number of values. If there are more runs than discrete values, there will be runs with duplicate factor values. If there are fewer runs than discrete values, then not all values will be used.
- *Choosing investigation type:* RSM models in ADAMS/Insight are statistical and numerical models that approximate the input/output behavior of the system under investigation [25]. DOE Quadratic Response Surface Method approximate optimization technique is used as it was understood from the literature [20] [21] will give desired results and was the best fit for the model. This optimization strategy works based on the behavior of independent variables (inputs), the RSM algorithm which is pre-coded by the MSC ADAMS software, the optimization tool guesses the value of the output function based on an assumption. The assumption is made by three level analysis in the specified range which is inserted in the response. The reason for building a response surface is that, although it is just an approximation, it can be used to estimate the set of input parameters yielding an optimal response. The result of an RSM-based optimization process is a predicted optimum within the input constraint, which can be seen as an estimate of the true optimum. The response surface is an analytical function, thus an optimization based on such model is very fast and does not require additional experiments or simulations to be performed.

*Batch Run:* Here, the simulations are executed sequentially after the mapping. All the simulations required for producing performance factors, considered as design requirements are included in this process. In this study, the batch run consists of simulations that provide kinematic characteristics. The analysis process was automated, and the optimization was carried out with complete MBS in the loop. As the no. of factors where reduced due to sensitivity study for each response was only 7 which made the size of the problem small and the total time was for the whole simulation was only 126 minutes.

*Output Review:* The batch run was carried out for 1000 times as it was defined in the optimizer and 1000 new designs were created with respect to fitting the criteria defined which had feasible, infeasible and error designs with individual camber and toe angel. While modifying factor value between maximum and minimum, the value of designed variable and objective function (optimization target) changes. When the trend of both the optimization targets appears to corelate then it can be chosen average value to take balance into consideration. This is done by scrolling the cursor in blue shown in Figure 4.5.

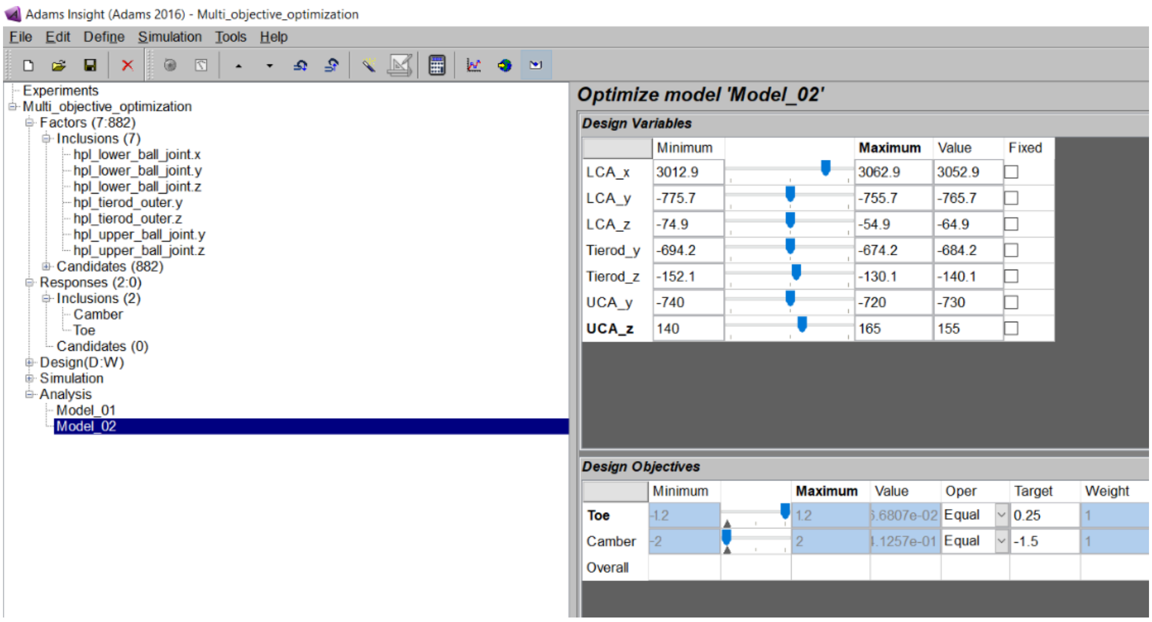



Figure 4.5: ADAMS/Insight Optimizer output toolbox

*Hardpoints:* The new set of hardpoints are selected by the above-mentioned process in the specific range of defined constraints. The new set of hardpoints are shown below in Table 4.2 and are compared with the hardpoints of initial design and the kinematic characteristic are obtained by replacing the old hardpoints with the new one in the ADAMS/Car. The parallel wheel travel analysis for new suspension design after optimization is again done by specifying the same analysis paraments discussed in section 3.3.1 and new plots for camber, toe and caster with wheel travel is generated.

| Point | Part & Location | Initial value | After Optimization |
|---|---|---|---|
| B | LCA outer_x | 3052.90 | 3021.846 |
| B | LCA outer_y | -765.70 | -758.628 |
| B | LCA outer_z | -64.90 | -75.223 |
| F | Ttirod outer_y | -684.20 | -672.883 |
| F | Tirod outer_z | -140.10 | -148.752 |
| H | UCA outer_y | -730.0 | -723.20 |
| H | UCA outer_z | 155.0 | 137.254 |

Table 4.2: New Hardpoint location

## 4.3 Comparing Results

The kinematic characteristic curves of the optimized models are shown and discussed below for each function respectively with the plots below the explanation. The blue curve represents the simulation before optimization, and the red curve represents the simulation after optimization.

*Variation in Camber Angle:*

It can be seen that, the total average variation is minimized, the new range variation is 1.88° during rebound and -2.6° during jounce. With the new hardpoints, a static camber of -0.2° is induced which increases lateral tire grip. And gradient of -23.478° is obtained, 93.94% of desired performance target is achieved for camber. When a vehicle is cornering, the wheels on the outside of the turn go into positive camber relative to the ground reducing the lateral grip of the tire under load. So, the increased lateral grip will compensate the condition. A slight negative camber maintains the contact patch of the tire with the road improving the directional stability along with reducing tire wear when moving in a straight line. This further results in higher lateral tire forces, which helps in cornering.

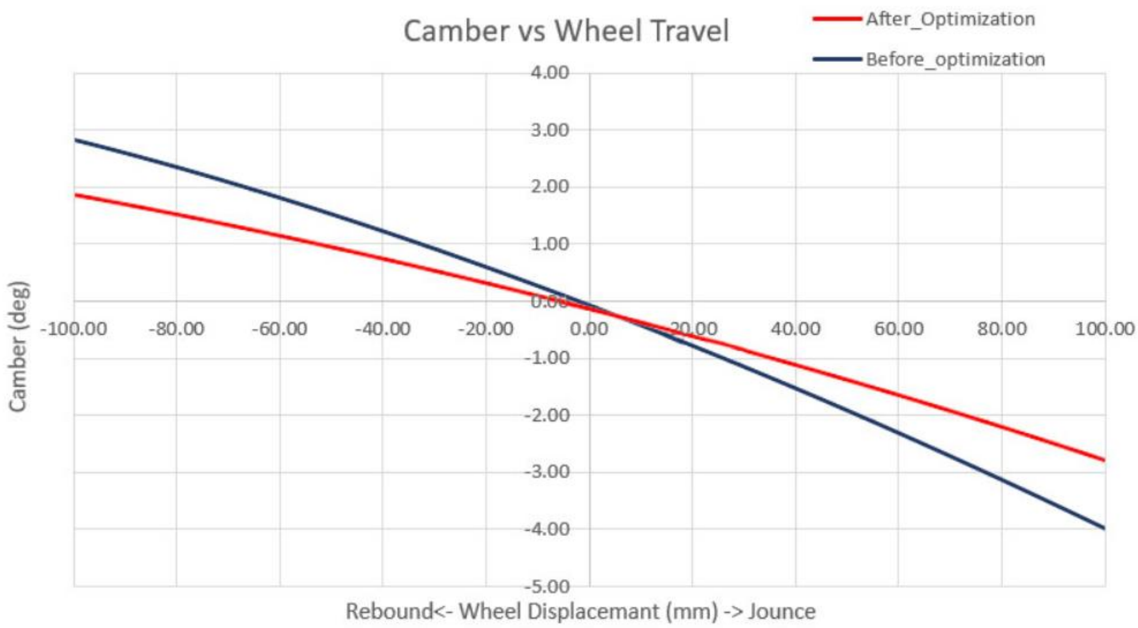



Figure 4.6: Comparison of Camber

*Variation in Toe angle:*

The total average variation is minimized, the new range variation is -0.49° during rebound and 1.487° during jounce. Static toe of 0.087° is attained, and the gradient is bought down to 4.821°/m from 11.65°/m. If there is negative static toe(toe-out) at the rear of the vehicle it will make driving condition difficult for the driver by causing lateral forces towards the outer side of the car causing oversteer while cornering which will cause the car to spin. So, a slight positive toe(toe-in) angle is needed during static condition to add more stability. Apart from that large variation in toe angle can causes tire wear to reduce the tire wear, variation of toe angle should be minimized, and it is achieved.

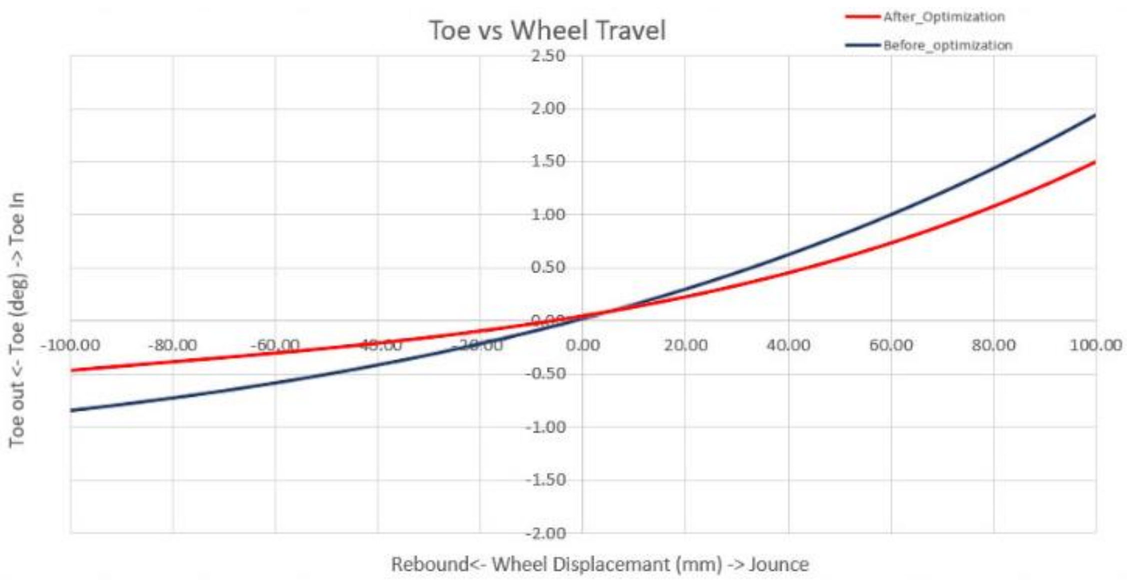



Figure 4.7: Comparison of Toe

*Variation in Caster:*

More emphasis was put on achieving the closest targets for camber and toe, as there was no objective function defined for caster during optimization. In multi-link suspension system type caster depend on various geometric points. As, it can be seen, by changes made in 7 parameters, caster shows a wide variation. This could be a result of the shift of the LCA x co-ordinate which was shifted by 28.264mm in combination of other hardpoints because the shift caused the change kingpin inclination angle. The gradient is reduced to -29.115°/m from -55.03°/m but the real time valve should be around -5°/m.

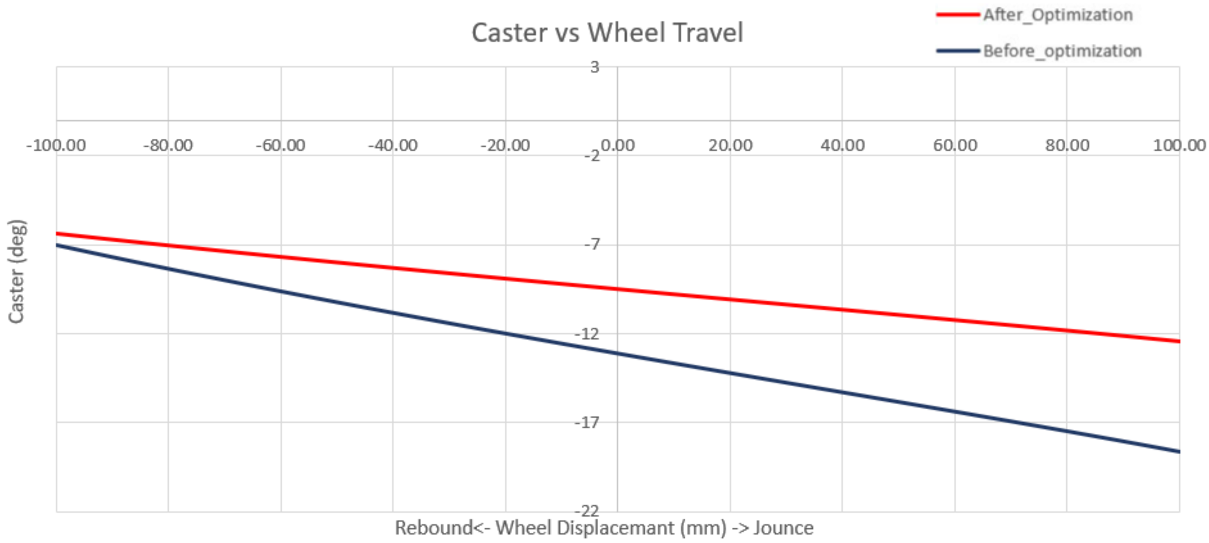



Figure 4.8: Comparison of Caster

When considering the producibility, the final design is modified and has better performance compared with the previous design under development process for camber, caster and toe angles. The target was to benchmark the concept ILS suspension which is a common practice in the industry, but the results obtained are accepted as the concept model fulfills the requirement of directional stability and reduction of tier wear mentioned at the end of section 3.4. It is observed that these are the best possible target which ILS can achieve in the certain limit of variation. To achieve the exact benchmarking target the complete structural design of ILS should be changed to of a four or five link multi-link suspension as its kinematic characteristic are better than that of ILS. This will increase the cost of the vehicle and reduce the boot space as ILS is designed to increase the boot space and larger fuel tank size, keeping the cost to a minimum.

## 4.4 Future Work

Much of future work towards defining an integral-link suspension whose caster angle behavior can be determined easily by a specific link for better kinematic performance. This will ease the optimization for rear steered cars as caster angle plays important role in returnability of wheel to its original position.

Elastokinematic study will give a better understanding of the wheel suspension as there are many bushing used in the suspension which influences suspension characteristics. Full vehicle dynamic simulation must be carried out using the same suspension on all the four wheels, this will give exact detailed performance report which can be used for advancement in the design process of the final concept product.

# 5. Conclusion

Rear integral-link suspension is the research object in this thesis and a kinematics simulation model of multi-link suspension is built based on the ADAMS/Car. By researching the influences suspensions structural geometric points and toe, camber angle a parameter optimization design of the modeled suspension is carried out.

The optimization is automated to reduce the trail-error method and to obtain a desired design, which has better performance matrix than the initial design in the specified hardpoint constraint, the automation is carried out by using ADAMS/Insight. The sensitivity study showed that a ILS system is very complicated mechanical system compared to other suspension types. The ILS has a specified links defined for camber and toe adjustments i.e. by varying their geometry the kinematic characteristics as per the named link will behave independently but this doesn’t happen completely. Variation in any links geometry affects the other characteristic is some way or the other. It was also found during the sensitivity study that the complete geometry has effects on caster angle. So, optimizing was difficult but with changes of other two parameter changed the caster behavior as well. ADAMS/Insight can be used for optimization by generating multiple DOE, but it is suitable with lesser amount of input variables and few design objectives. The results obtained had a considerable improvement over the primary design. The new design improves stability of the vehicle, increase the lateral tire grip and reduces tire wear.

The new design couldn’t achieve the exact performance target due its structural design. To get the exact performance target the structural design must be changed completely to a multi-link suspension.

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