

MINIMIZATION OF THE EFFECT OF MACPHERSON STRUT SUSPENSION SYSTEM ON WHEEL ALIGNMENT: A KINEMATIC ANALYSIS



Amanuel Nasir Yimam

A Thesis submitted to the Department of Mechanical Engineering

School of Mechanical, Chemical and Materials Engineering.

Presented in Partial Fulfillment of the Requirements for the Degree of Master's
in Automotive Engineering

Office of Graduate Studies

Adama Science and Technology University

June 2024

Adama, Ethiopia

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DECLARATION

I hereby declare that this Master Thesis entitled “*Minimization of the effect of MacPherson strut suspension system on wheel alignment: A kinematic analysis*” is my original work. That is, it has not been submitted for the award of any academic degree, diploma, or certificate in any other university. All sources of materials that are used for this thesis have been duly acknowledged through citation.

Amanuel Nasir Yimam

June 2024

Name of the student

Signature

Date

APPROVAL

I/we, the advisors of the thesis entitled “*Minimization of the effect of MacPherson strut suspension system on wheel alignment: A kinematic analysis*” and developed by **Amanuel Nasir Yimam** hereby certify that the recommendation and suggestions made by the board of examiners are appropriately incorporated into the final version of the thesis.

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APPROVAL FROM BOARD OF EXAMINERS

We, the undersigned, members of the Board of Examiners of the thesis by **Amanuel Nasir Yimam** have read and evaluated the thesis entitled “*Minimization of the effect of MacPherson strut suspension system on wheel alignment: A kinematic analysis*” and examined the candidate during open defense. This is, therefore, to certify that the thesis is accepted for partial fulfillment of the requirement of the degree of Master of Science in Automotive Engineering.

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ACKNOWLEDGEMENT

I would like to express my gratitude to the almighty God for giving me time, strength and health during the preparation of this thesis. My deepest gratitude goes to my Advisor, Ramesh Babu Nallamothe (PhD), for his help, advice and guidance that gave me direction and made me put intelligent effort on this thesis work. My special thanks go to my family and my friends, for their unwavering support and encouragement throughout my thesis. Then, I would like to express my sincere gratitude and appreciation to my classmates.

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ACRONYMS AND ABBREVIATIONS

<i>3D</i>	<i>Three dimensional</i>
<i>A</i>	<i>The point of the front joint of the lower control arm</i>
<i>a₁₁, a₁₂, a₂₁, a₂₂</i>	<i>Coefficients</i>
<i>B</i>	<i>The point of the rear joint of the lower control arm</i>
<i>C</i>	<i>The point of connection of knuckle and lower control arm</i>
<i>CAD</i>	<i>Computer Aided Design</i>
<i>CVT</i>	<i>Continuously variable transmission</i>
<i>D</i>	<i>The point of the joint of the top of suspension and car body</i>
<i>DOF</i>	<i>Degree of Freedom</i>
<i>E</i>	<i>The point of the lower installed position of the damper</i>
<i>G</i>	<i>The point of the wheel center</i>
<i>H</i>	<i>The point of the center of the outer surface of wheel</i>
<i>IBJ</i>	<i>Inner ball joint</i>
<i>K</i>	<i>The ground point of the wheel</i>
<i>KPI</i>	<i>Kingpin Inclination</i>
<i>L_A</i>	<i>Length of the control arm</i>
<i>LBJ</i>	<i>Lower ball joint</i>
<i>LCA</i>	<i>Lower control arm</i>
<i>M</i>	<i>The point of the outer joint of the tie rod</i>
<i>MPSS</i>	<i>MacPherson Strut Suspension</i>
<i>MPSSS</i>	<i>MacPherson Strut Suspension System</i>
<i>N</i>	<i>The point of the inner joint of the tie rod</i>
<i>NVH</i>	<i>Noise, Vibration, and Harshness</i>
<i>O</i>	<i>Origin of the coordinate system</i>
<i>OBJ</i>	<i>Outer ball joint</i>
<i>TCD</i>	<i>Turning circle diameter</i>
<i>x_C</i>	<i>x coordinate of point C</i>

x_D	<i>x coordinate of point D</i>
x_G	<i>x coordinate of point G</i>
x_H	<i>x coordinate of point H</i>
y_A	<i>y coordinate of point A</i>
y_{A1}	<i>y coordinate of static equilibrium position of point A</i>
y_B	<i>y coordinate of point B</i>
y_C	<i>y coordinate of point C</i>
y_{C1}	<i>y coordinate of static equilibrium position of point C</i>
y_D	<i>y coordinate of point D</i>
y_G	<i>y coordinates of point G</i>
y_H	<i>y coordinates of point H</i>
y_J	<i>y coordinate of point J</i>
y_{J1}	<i>y coordinate of static equilibrium position of point J</i>
y_P	<i>y coordinate of point P</i>
y_{P1}	<i>y coordinate of static equilibrium position of point P</i>
z_A	<i>z coordinate of point A</i>
z_{A1}	<i>z coordinate of static equilibrium position of point A</i>
z_B	<i>z coordinate of point B</i>
z_C	<i>z coordinate of point C</i>
z_{C1}	<i>z coordinate of static equilibrium position of point C</i>
z_D	<i>z coordinate of point D</i>
z_G	<i>z coordinates of point G</i>
z_H	<i>z coordinates of point H</i>
z_J	<i>z coordinate of point J</i>
z_{J1}	<i>z coordinate of static equilibrium position of point J</i>
z_P	<i>z coordinate of point P</i>
z_{J1}	<i>z coordinate of static equilibrium position of point P</i>
z_r	<i>Excitation of road disturbance</i>
α	<i>Slope of the strut with respect to the vertical axis</i>
γ_c	<i>Camber angle</i>

γ_{cas}	<i>Caster angle</i>
γ_{kp}	<i>Kingpin inclination angle</i>
γ_t	<i>Toe angle</i>
θ	<i>Angle of the control arm</i>
θ_1	<i>Initial angle of the control arm resulting from static deflection</i>
φ	<i>Rotation angle of the wheel about x-axis</i>

ABSTRACT

The MacPherson strut suspension system, despite its widespread use in the automotive industry, has been shown to have issues with wheel alignment. Lower control arm of this suspension is prone to damage affecting camber, caster, and toe. This can lead to uneven tire wear, reduced handling, and increased fuel consumption. To address this, a kinematic analysis of the MPSSS was conducted using Adams Car software to identify the wheel alignment parameters, followed by optimization of these parameters to reduce the effect of wheel alignment using DESIGN EXPERT software. The analysis focused on a Toyota Corolla 2022 model and identified camber, toe, kingpin inclination, and caster angles as key factors affecting wheel alignment. The optimization aimed to improve these parameters through adjustments to the front, lower, and rear control arms. Accordingly, for the current model, minimum and maximum camber angles are -1.09° and 2.07° , minimum and maximum caster angles are -2.4° and 1.6° , minimum and maximum kingpin inclination angles are 3.09° and 5.0° , and minimum and maximum toe angles are -3.09° and 2.25° were obtained through kinematic analysis. Then, by changing the z positions of front control arm, rear control arm, and outer control arm from -50, -100, and -75 to -50.9, -89.69, and -77.36, respectively, the optimized model was able to reduce 24.17%, 23.75%, 7.44%, 4.21%, 2.4%, 1.33%, and 1% of the minimum caster, maximum caster, minimum toe, minimum kingpin, maximum camber, maximum toe, maximum kingpin angles, respectively. Hence, for the optimized model, minimum and maximum optimized camber angles are -1.27° and 2.02° , minimum and maximum optimized caster angles are -1.82° and 1.22° , minimum and maximum optimized kingpin inclination angles are 2.96° and 4.95° , and minimum and maximum optimized toe angles are -2.86° and 2.22° were obtained. Therefore, by changing the z positions of front control arm, rear control arm, and outer control arm wheel alignment can be reduced.

Keywords: *ANN, Adams Car, Design Expert, Macpherson Strut Suspension, Wheel Alignment.*

CHAPTER ONE

INTRODUCTION

1.1. Background of the Study

Any vehicle, from light-duty trucks to passenger automobiles, must have a suspension system (Deo, 2007). It is essential to maintaining the safety, comfort, and stability of the car. Its purpose is to cushion the vehicle's wheels from road shocks and bumps so they stay in touch with the surface (Liang and Xin, 2012). Additionally, it helps maintain the car's balance and levelness even when navigating difficult terrain, which enhances ride quality and lessens driver and passenger fatigue (Reddy et al., 2016).

There are various kinds of suspension systems, as shown in *Figure 1.1*; the MacPherson Strut Suspension (MPSS) is one of the more popular kinds because of its affordability, simplicity, and ease of maintenance (Abdelrahman and Nabawy, 2020). Additional suspension system options include the leaf spring suspension, which is ideal for towing large loads, and the double wishbone suspension, which provides excellent handling and performance (Berghian et al., 2018).

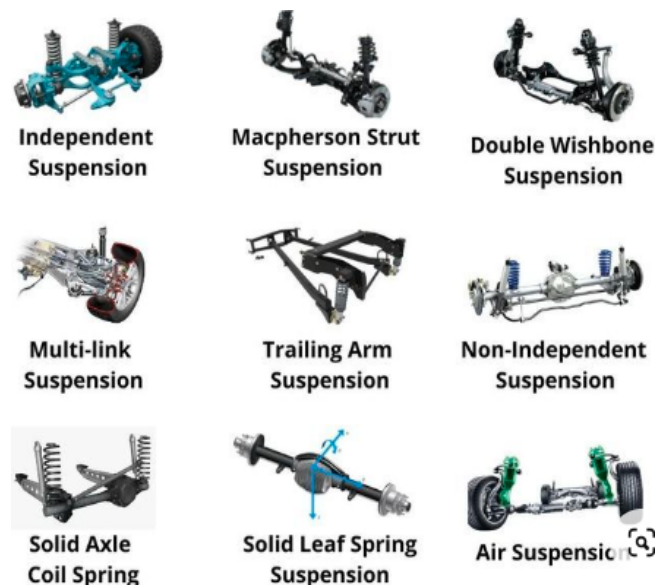


Figure 1.1: Various types of Suspension systems (The Engineers Post, 2020)

The MacPherson strut suspension system (MPSSS) is a popular choice for an independent suspension system, especially for the front wheels, in modern vehicles (Fallah et al., 2014). It integrates the capabilities of a suspension spring and a shock absorber into one compact and efficient device, making it a popular option among automakers due to its simplicity, cost-effectiveness, and compactness (Shinde et al., 2018). As shown in *Figure 1.2*, it is made up of the wheel hub, steering knuckle, control arm, and strut, among other parts (Reddy et al., 2016).

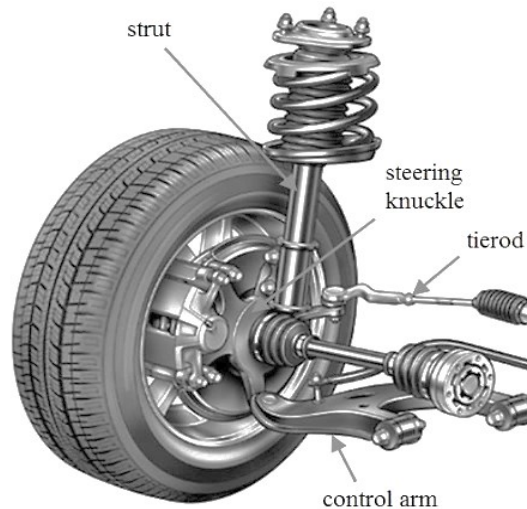


Figure 1.2: Basic components of MacPherson suspension system (Saviotto, 2020)

A prismatic joint is used for attaching one end of the strut to the knuckle and a spherical joint is used to fix the other end to the chassis. In addition, the lower control arm is connected to chassis by a revolute joint on one end. Moreover, the steering knuckle is attached to the other end by a spherical joint. Furthermore, the steering link is attached to the tie-rod by spherical joints and the tie-rod is connected to the rack with a universal joint (Shi et al., 2019).

The MPSSS has many advantages, including simplicity and cost-effectiveness, compactness, and good handling and performance, especially in front-wheel drive vehicles (Cronin, 1981). However, it also has some disadvantages, such as a harsher ride quality than other types of suspension systems, changes in wheel camber and alignment that can affect handling and tire

wear, and limited adjustability and torque-steer in certain more potent front-wheel-drive applications (Fallah et al., 2014).

Maintaining good wheel alignment is one of a suspension system's essential duties (Chen et al., 2012). The angle at which the wheels are positioned in relation to the vehicle's chassis is known as wheel alignment (Deo, 2007). This is important because it reduces uneven tire wear, increases fuel efficiency, and improves vehicle handling (Anumukonda et al., 2021). A misaligned suspension system can cause a number of problems, including uneven tire wear, poor steering response, and poor performance all around. By preventing the wheels from bouncing about on the road and reducing the force of driving over bumps and dips, the suspension system is essential to preserving wheel alignment (Saviotto, 2020).

Wheel alignment refers to the angles and orientation of the wheels relative to the vehicle and the road surface (Liang and Xin, 2012). There are three main angles that are measured and adjusted during a wheel alignment: camber, toe, and caster, as shown in *Figure 1.3*. Camber refers to the angle of the tires when viewed from the front of the vehicle, toe refers to the angle of the tires when viewed from above the vehicle, and caster refers to the angle of the steering axis when viewed from the side of the vehicle (Shojaeefard et al., 2017). Proper wheel alignment is essential for several reasons, primarily because it directly impacts tire wear, vehicle handling, and overall performance (Nasiri et al., 2015). Misaligned wheels can cause premature and uneven tire wear, affect the vehicle's handling, and reduce the vehicle's overall performance (Patil, 2016).

Several factors can affect wheel alignment, including road conditions, impact damage, suspension wear and tear, and the type of suspension system used in the vehicle (Reddy et al., 2016). The MPSSS, which is a popular type of suspension system used on many front-wheel drive vehicles, is more susceptible to wheel alignment issues than other suspension systems. This is because it is a single strut that serves as both the suspension component and the steering component, and it is more likely to be knocked out of alignment by road conditions and impact damage (Saviotto, 2020). Any wear or damage to the MacPherson strut can lead to

misalignment issues, causing the wheels to toe in or out, resulting in uneven tire wear, as shown in *Figure 1.3*, and compromised vehicle handling (Wang et al., 2016).

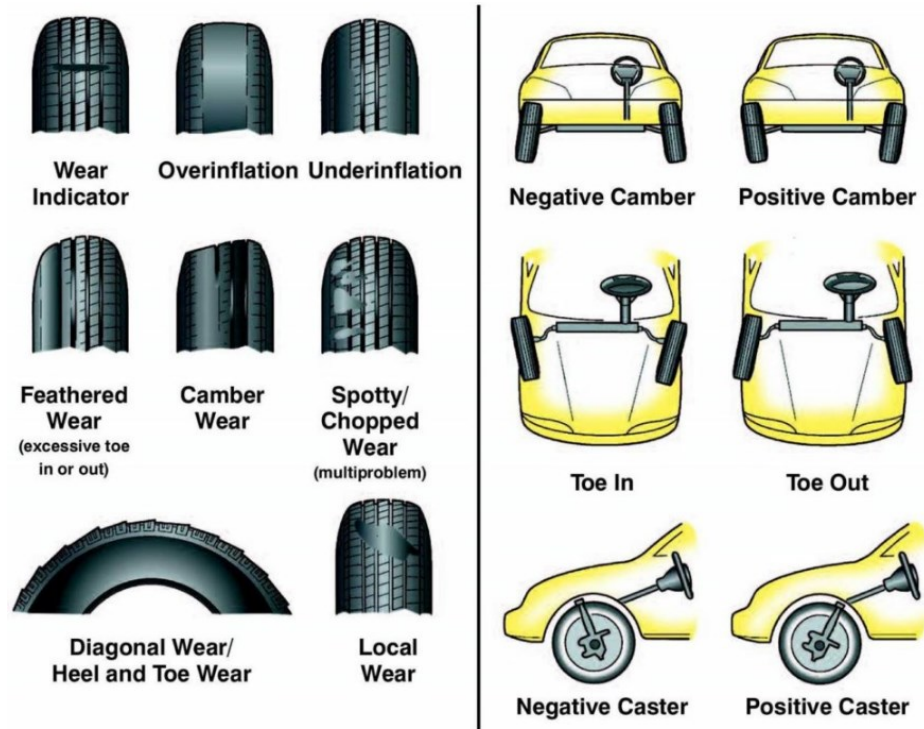


Figure 1.3: Tire wear and wheel alignment (Quick Trick, 2023)

MPSSS pose significant issues, especially when it comes to preserving constant camber and toe angles during suspension movement. In contrast to suspension systems like as double-wishbone or multi-link, which offer a more stable and controlled wheel alignment across the whole range of motion, the MPSS's single control arm design frequently causes noticeable camber changes when the wheel moves up and down. This may result in uneven tire wear and poor handling, particularly when the vehicle is loaded or cornering sharply and the wheels may tilt too much, impairing the vehicle's stability and grip (Reddy et al., 2016).

There are specific challenges with MPSSS that can lead to adverse effects on wheel alignment (Abdelrahman and Nabawy, 2020). Some of these challenges include vertical and lateral forces, strut deflection, and the impact of road irregularities. Vertical forces, such as bumps

and potholes on the road, can cause the strut to compress and extend, leading to changes in wheel alignment and uneven tire wear (Anumukonda et al., 2021). Lateral forces, such as those experienced during cornering, can cause the strut to deflect, inducing positive camber and leading to reduced traction and uneven tire wear (Berghian et al., 2018). Strut deflection occurs when the strut bends due to the weight of the vehicle or the impact of road irregularities leading to uneven tire wear and poor handling (Chen et al., 2012).

Despite its wide use, kinematics of the MPSSS is very complex and it has not been sufficiently studied yet compared to the other suspension systems (Shojaeefard et al., 2017). The equation representing the kinematic solutions for MPSSS is 64 degree polynomial, and this shows the complexity observed in the kinematics of the mechanism (Patil, 2016). Thus, it has been the focus of many researchers in the past decades and several analysis methods have been proposed for its kinematic study. However, most of them are limited to linear analysis which does not account for the non-linear load aspects that the system faces, and this has been the main reason behind the early-stage failures of the system. Because of lack of extensive studies in this regard, vehicles equipped with the MPSSS usually experience early wheel misbalancing issues subsequent effects on vehicle handling and riding quality (Shinde et al., 2018). Thus, in order to solve these problems, a kinematic analysis and optimization of the MPSSS is performed to extensively study the subsequent effects on the performance of the mechanism.

1.2. Statements of the Problem

The MacPherson strut suspension system (MPSSS) is widely used in the automotive industry due to its simplicity and cost-effectiveness. However, it is known to have issues with wheel alignment, which can lead to uneven tire wear, reduced handling, and increased fuel consumption (Shinde et al., 2018). Lower control arm of this suspension is prone to damage affecting camber, caster, and toe (Fallah et al., 2014). Though, parts of MPSS can be optimized to reduce the effects of suspension travel on wheel alignment. However, MPSSS undergoes complex dynamics, and its characteristics are difficult to determine in real operating conditions. As such, usually softwares such as ADAMS Car are used to perform the kinematic

analysis. Majority of the existing studies represented the conditions by single effects ignoring the combination effects in their analysis. Such assumptions have led to unrealistic generalization that cannot be used to predict their properties in real conditions. To address this, a kinematic analysis of the MPSSS was conducted using Adams Car software to identify the wheel alignment parameters, followed by optimization of these parameters to reduce the effect of wheel alignment using DESIGN EXPERT software.

1.3. Research Question

The following research questions are used as a guide for this thesis:

- ✎ What are the key kinematic parameters that affect wheel alignment in MPSSS?
- ✎ How can the kinematic analysis of MPSS be performed to determine wheel alignment?
- ✎ What modifications can be made to the MPSSS to reduce the effects of suspension travel on wheel alignment?

1.4. Objectives

1.4.1. General Objective

The general objective of this thesis is to perform minimization of the effect of MacPherson strut suspension system on wheel alignment through a kinematic analysis.

1.4.2. Specific Objectives

The specific objectives of this thesis are:

- ✎ To investigate the key kinematic parameters that affect wheel alignment in MPSS
- ✎ To develop a kinematic model of a MPSS for determining wheel alignment
- ✎ To make suitable design optimizations to the MPSS that can reduce the effects of wheel alignment

1.5. Scope of the study

In this study, kinematic analysis and optimization of the MPSSS is performed using empirical and simulation to determine the factors that contribute to its poor wheel alignment performance. Suitable analytical model is established and utilized to gain good insight into the understanding phenomena when the system is subjected to various operating conditions. The analysis focused on response behavior of the MPSSS components such as displacement of the components. Based on the observed results, the factors contributing to the early-stage wheel alignment distortion of the system is identified and optimizations is made on the parameters.

1.6. Significance of the Study

The significance of this study lies in its potential to provide detailed kinematic analysis of the MPSSS, involving both analytical and finite element analysis methods. By conducting a kinematic analysis and optimization of MPSS, the study aims to reduce the negative effects on wheel alignment. This is important because proper wheel alignment is crucial for vehicle stability, handling, and tire wear. By reducing the negative effects on wheel alignment, the modified suspension system could potentially improve vehicle safety and performance, leading to a better driving experience for the user. Additionally, the study could contribute to the development of more efficient and effective suspension systems in the automotive industry.

1.7. Limitations of the Study

While the kinematic analysis conducted in this thesis provides valuable insights into the alignment issues of the MPSSS, it is important to note that the findings are based on simulations using Adams Car software. The accuracy and reliability of the simulation results heavily depend on the input parameters and model assumptions. However, the study's limitations include potential oversimplifications in the simulation model, assumptions about the representativeness of the Toyota Corolla 2022 model, and the exclusion of dynamic factors and real-world variations that may affect wheel alignment.

1.8. Organization of the thesis

This thesis work is consisting of six chapters. The first chapter is concerned with the introduction of the thesis, providing background information on MPSSS and its effect on wheel alignment. Whereas, the second chapter is mainly concerned with the reviews of previous research work on the analysis of MPSSS. In addition, research gap is identified and included in in this chapter of the thesis. The materials, methodologies, and methods used for the research are explained and discussed in the third chapter of this thesis. Moreover, chapter four is concerned with kinematic analysis, optimization, and validation of MPSSS to reduce the effects on wheel alignment. Furthermore, in the fifth chapter of the thesis the results of chapter five are discussed in detail. Finally, based on the insights gained from chapter five, conclusions and recommendations are explained in chapter six.

CHAPTER TWO

LITERATURE REVIEW

2.1. Overview of suspension systems

A suspension system is a network of components that link a vehicle's wheels to its body, serving to absorb the jolts and impacts encountered on the road (Shojaeefard et al., 2017). Additionally, suspension systems are instrumental in ensuring vehicle stability and control, particularly during maneuvers like braking or cornering (Abdelrahman and Nabawy, 2020). This system encompasses various parts, including tires, springs, shock absorbers, air springs, steering, sway bars, and axles (Anumukonda et al., 2021). In essence, a well-functioning suspension system contributes to a smooth ride, improved handling, and enhanced braking and steering control (Shojaeefard et al., 2017).

Although there are many other kinds of suspension systems used in cars, independent and non-independent suspension are the most widely utilized types (Deo, 2007). Better handling and a smoother ride are made possible by independent suspension, which permits each tire to move independently of the others. Control arms, shock absorbers, springs, and connecting links are frequently used in conjunction with one another to provide independent suspension. Because of its improved stability, comfort, and handling, it is common in contemporary cars (Berghian et al., 2018). Conversely, non-independent suspension is less expensive to produce than independent suspension since it links the two wheels on the same axle. It does not, however, offer as good handling or as smooth a ride as independent suspension (Chen et al., 2012).

The MPSSS is independent suspension that is commonly used in passenger cars due to its simplicity and cost-effectiveness (Cronin, 1981). It include a strut, coil spring, control arm, and ball joint (Fallah et al., 2014). The strut is structural component that supports the vehicle's weight and acts as a shock absorber, while the coil spring is located around the strut and helps to absorb the bumps and shocks of the road surface. The control arm is a link that connects the strut to the vehicle's chassis and helps to control the movement of the wheel and keep it aligned,

while the ball joint is a joint that connects the control arm to the wheel hub and allows the wheel to move up and down without changing its alignment (Liang and Xin, 2012).

MPSSS are easy-to-use, have excellent handling and ride comfort, and comparatively have low maintenance needs (Nasiri et al., 2015). In addition to being more compact and lightweight than other suspension systems, it is a well-liked option for contemporary cars (Patil, 2016). Nevertheless, there are certain drawbacks to the MPSSS, such as its restricted adjustability and propensity to transfer road noise and vibrations to the car's body (Abdelrahman and Nabawy, 2020). Heavy-duty vehicles and off-road driving may not be as suitable for it, and its handling may not be as good as that of other suspension systems, like double wishbone suspension (Reddy et al., 2016).

For passenger cars, the MPSSS is a popular option due of its somewhat easy-to-use, low-cost design, which offers excellent handling and ride comfort (Wang et al., 2016). But off-roading and heavy-duty vehicles are not as good a fit for it (Su et al., 2018). Although they are more complicated and costly, alternative suspension systems, like double wishbone suspension, can offer greater off-road performance and handling (The Engineers Post, 2020). The MPSSS has a more compact and lightweight construction than other suspension systems like double wishbone or multi-link suspensions (Shinde et al., 2018). It works well with a variety of cars because it has outstanding handling and stability qualities (Saviotto, 2020). It might not, however, provide as much performance or adjustability as more intricate suspension systems (Shi et al., 2019).

Double wishbone suspension systems offer a greater range of adjustability and control compared to MPSSSs (The Engineers Post, 2020). They provide better handling and stability during aggressive driving maneuvers but are generally more complex and expensive to manufacture (Reddy et al., 2016). Multi-link suspension systems offer improved wheel control and handling performance. They provide better isolation of road imperfections and allow for more precise tuning of suspension characteristics. However, they can be more challenging to design and maintain (The Engineers Post, 2020).

2.2. Kinematic analysis of MacPherson strut suspension system

Kinematic analysis of the MPSSS is important in understanding its behavior and performance (Chen et al., 2012). Kinematic analysis involves studying the motion and movement of the suspension system without considering the forces involved (Shinde et al., 2018). This analysis helps engineers optimize the design and improve the handling of the vehicle (Patil, 2016). A variety of kinematic analysis techniques can be used to study suspension systems, including geometric analysis, kinematics of mechanisms, and multibody system dynamics (Fallah et al., 2014).

The key kinematic parameters of a MPSSS include camber angle, caster angle, toe angle, kingpin inclination (KPI), track width, and wheelbase (Patil, 2016). The camber angle refers to the angle between the wheel plane and the vertical axis, affecting tire contact patch and vehicle handling. The caster angle is the angle between the steering axis and the vertical axis, influencing stability and steering. Toe angle represents the angle between the longitudinal axis of the vehicle and the wheel plane, impacting tire wear and handling. Kingpin inclination is the angle between the steering axis and the wheel plane, affecting steering and handling (Nasiri et al., 2015). These parameters play a significant role in the vehicle's ride, handling, and performance (Reddy et al., 2016).

A kinematic analysis of a MPSSS can be performed using geometric analysis, kinematics of mechanisms, or multibody system dynamics, depending on the specific goals of the analysis and the resources available (Anumukonda et al., 2021). To perform a kinematic analysis of a MPSSS, engineers typically use computer-aided drafting or design (CAD) software and specialized simulation tools (Chen et al., 2012). These tools allow them to model the suspension system, input design parameters, and simulate the system's kinematic behavior (Cronin, 1981).

A MPSSS is influenced by various design parameters that affect its kinematics. These include the strut angle, control arm angle, knuckle height, and tie rod length (Deo, 2007). Adjusting these parameters allows for customization of the suspension system's kinematic characteristics.

For instance, increasing the strut angle enhances camber change over bumps, improving vehicle handling. However, this adjustment may compromise ride comfort (Fallah et al., 2014).

Additionally, there are other design parameters such as spring rate, damper rate, suspension arm length, strut length, mounting position, wheel offset, tire size, vehicle weight, road conditions, and driving style that also impact the kinematics of the MPSSS (Liang and Xin, 2012). Each parameter influences different aspects of the suspension system, including ride height, handling, comfort, camber angle, caster angle, kingpin inclination, scrub radius, tire wear, cornering grip, steering response, stability, steering feel, high-speed performance, toe angle, roll center height, body roll, suspension geometry, anti-dive characteristics, brake dive, weight transfer, and braking performance (Saraf and Satpute, 2017).

There are several techniques used in studying the MPSSS. These techniques include graphical methods, analytical methods, and numerical methods. Graphical methods involve the use of diagrams and graphs to represent the motion of the system. Analytical methods involve the use of mathematical equations to describe the motion of the system. Numerical methods involve the use of computer simulations to analyze the motion of the system (Berghian et al., 2018; Nasiri et al., 2015).

Over the past decade, several attempts have been made to study the kinematics of MPSSS (Reddy et al., 2016) used Microsoft Excel to study the kinematic characteristics of MPSSS. They found that the final equation representing all the kinematic solutions for a given pair of steering and road profile inputs is of 64 degrees and they could not find the exact solutions.

(Saviotto, 2020) analyzed side forces on the upper mount of a MPSSS as it is assembled on a vehicle using ADAMS CAR software by applying linear loads. Inclination of the spring axis with respect to the shock absorber axis was changed several to see the physical limits imposed by the geometry of the suspension. Based on the results, he introduced the use of a spherical joint on the top mount of the strut to reduce the side forces.

(Liang and Xin, 2012) optimized control art topology of MPSSS for SUVs through finite element analysis when the system is exposed to different loads. The analysis started with the

simulation of existing model under linear loads, and based on the results, optimized new model was developed for the same purpose.

(Saraf and Satpute, 2017) optimized tie-rod length for light-duty motor vehicles to reduce their tendency of side pulling and improve maneuverability during steering. They compared and validated the theoretical results with the experimental test results obtained by the natural frequencies on FFT analyzer.

(Su et al., 2018) studied the causes of the upper strut premature failure. To estimate the lifetime of this layout a lot of experimental test and numerical analyses has been conducted. The results showed that impulsive loads mainly lead to damage of the upper strut, and other loads cannot be compared with the mentioned load. However, they did not provide any optimal solutions for the problem.

(Abdelrahman and Nabawy, 2020) investigated sensitivity of Macpherson suspension mechanisms is to reduce distortion of the wheel alignment angles from the pre-set values using ADAMS CAR software. The coordinates of two inner points and the tie rod inner point were changed and the subsequent effects were analyzed. They concluded that the camber angle, toe angle and track width are highly sensitive to changes of these parameters. However, they did not give optimal values of the parameters.

A mathematical model to represent operating conditions of MPSSS has been developed by (Berghian et al., 2018). From the results, they concluded that the behavior of the system at high fidelity has been reproduced by mathematical models. The results contain the major effects that determine the contact between the tire and the road and how the forces resulting from the interaction of the tire and road transferred through the chassis suspension. Though, they did not provide details about connections of the transmission or structure compliance.

2.3. Effects of MacPherson strut suspension system on wheel alignment

Wheel alignment is the process of adjusting the suspension parameters such as angles of the wheels so that they are perpendicular to the ground and parallel to each other (Chen et al., 2012). Proper wheel alignment is essential for safe driving, even tire wear, and optimal vehicle handling. Wheel alignment integrates all the factors of steering and suspension geometry to provide safe handling, good ride quality, and maximum tire life (Deo, 2007). The primary angles that are adjusted during wheel alignment are camber, caster, and toe. Camber refers to the angle of the wheel when viewed from the front or rear of the vehicle, caster refers to the angle of the steering pivot when viewed from the side of the vehicle, and toe refers to the extent to which the tires turn inward or outward when viewed from above. Proper wheel alignment helps to maintain stability, improve fuel efficiency, and reduce tire wear (Abdelrahman and Nabawy, 2020).

There are two main types of wheel alignment: front-end alignment and four-wheel alignment (Quick Trick, 2023). Front-end alignment adjusts the camber, toe, and caster angles of the front wheels, while four-wheel alignment adjusts the camber, toe, and caster angles of all four wheels (Saraf and Satpute, 2017). Technicians use these angles to properly align the wheels, and proper alignment can improve handling, extend the lifespan of tires, and ensure a safe and comfortable ride (Saviotto, 2020).

The alignment of the wheels can be influenced by the MPSSS in several ways (Shi et al., 2019). If the strut is worn or damaged, it can cause changes in the camber and toe angles. Similarly, a bent or damaged upper control arm can also lead to alterations in the camber and toe angles (Su et al., 2018). Moreover, the elimination of the upper control arm in this suspension system affects the camber, caster, and toe adjustments as the suspension compresses (Wang et al., 2016). Despite these effects, the MacPherson strut minimizes camber gain during compression and can cause negative camber due to the inward tilt of the strut (Liang and Xin, 2012).

The impact of MPSSSs on wheel alignment have been the subject of numerous investigations. According to (Chen et al., 2012), the MPSSS technology significantly impacted the wheels'

camber and caster angles. (Anumukonda et al., 2021) discovered that compared to other suspension systems, the MPSSS had a bigger impact on the alignment angles. Potential alignment problems may arise from variations in the camber and toe angles caused by the design of the strut assembly. Wheel alignment may be impacted by modifications in suspension geometry brought about by the strut's placement and orientation (Reddy et al., 2016).

The wear and tear on the MPSSS can also have a significant impact on wheel alignment (Saraf and Satpute, 2017). When replacing struts, it is important to perform an alignment to ensure that the vehicle's alignment is within factory specifications. Failure to do so can lead to problems like premature tire wear and other customer comebacks (Shojaeefard et al., 2017). Camber, toe, and caster settings can be affected by installing new struts, and if the alignment isn't adjusted back to manufacturer specifications, the customer could end up with uneven tire wear (Su et al., 2018).

Misaligned wheels can cause a number of problems, including reduced fuel economy, increased tire wear, decreased handling and stability, and premature wear and tear on the suspension components (Wang et al., 2016). In a MPSSS, misaligned wheels can also cause changes in the camber and caster angles, which can affect steering geometry and handling performance (Su et al., 2018). Therefore, it is important to regularly check and adjust the wheel alignment of a vehicle with a MPSSS (Deo, 2007).

To reduce the effects of wheel alignment in MPSSSs, several modification techniques can be employed. These include adjusting the ride height, installing camber plates, upgrading control arms, and utilizing adjustable suspension components (Saraf and Satpute, 2017; Shi et al., 2019). Geometric modifications, such as adjusting the ride height and installing camber plates, can optimize the alignment settings for specific driving conditions and ensure even tire contact with the road surface (Nasiri et al., 2015). Upgrading control arms with stiffer and more adjustable components can enhance suspension control, reducing the likelihood of misalignment (Fallah et al., 2014). Polyurethane bushings and stronger control arms can also help to reduce the effects of wheel alignment issues by making the suspension system more

rigid (Chen et al., 2012). Using adjustable camber plates, which allow the camber angle to be changed, improves handling and reduces tire wear (Anumukonda et al., 2021).

MPSSSSs can be modified to reduce the effects on wheel alignment (Abdelrahman and Nabawy, 2020). Cambering the strut by 0.5 degrees can reduce camber change by up to 50%, while adjusting the offset of the wheel by 5mm can reduce toe change by up to 25% (Berghian et al., 2018). Each modification targets specific alignment parameters, allowing for more precise adjustment. Reducing the impact on wheel alignment by altering a MPSSSS can improve handling, increase tire wear, and boost stability, among other benefits (Chen et al., 2012). However, certain changes might also come with trade-offs, such as less comfortable ride, more noise, or a stiffer ride. To make sure that the modifications are suitable for the vehicle and driving conditions, it is crucial to carefully weigh the benefits and drawbacks of each alteration and to speak with an expert (Fallah et al., 2014).

The MPSSSS has been extensively researched, with studies focusing on various aspects of the system (Liang and Xin, 2012). These include ride comfort, handling and stability, structural integrity, noise, vibration, and harshness (NVH), weight reduction, impact on vehicle dynamics, manufacturing and assembly, performance enhancement, and the influence of suspension geometry (Nasiri et al., 2015). Studies have investigated the durability and fatigue life of the strut and its components, ensuring that the system can withstand the demanding conditions of everyday driving (Patil, 2016). The influence of suspension geometry on the performance of the MPSSSS has also been extensively studied. Researchers have examined the effects of parameters such as camber angle, caster angle, and toe angle on tire wear, steering response, and vehicle stability (Reddy et al., 2016).

2.4. Research gap

The kinematics of MPSSSSs are highly complex, and existing literature lacks adequate information to accurately predict their behavior under real operating conditions. While static behavior, such as camber and toe angles, has been extensively studied, there is a lack of research on the optimization of the MPSSSS components. Thus, this thesis aims to address these

limitations by conducting a kinematic analysis of the MPSSS using ADAMS CAR software. Moreover, the optimization of the locations of MPSSS components are performed here. By simulating the mechanism under bump and rebound conditions and analyzing the results, this research aims to identify ways to improve weak points and enhance the performance of the MPSSS.

CHAPTER THREE

MATERIALS AND METHODOLOGY

In order to make a kinematic analysis of the MPSSS, behavior of MPSSS under different loads including the effects of the non-linear loads is investigated. Through the development of a kinematic model, this study aims to unravel the intricate relationships between various suspension components and their effects on wheel alignment. The methods and methodology used for design and modeling is broken down into several steps. The key procedures in this study are represented in *Figure 3.1*.

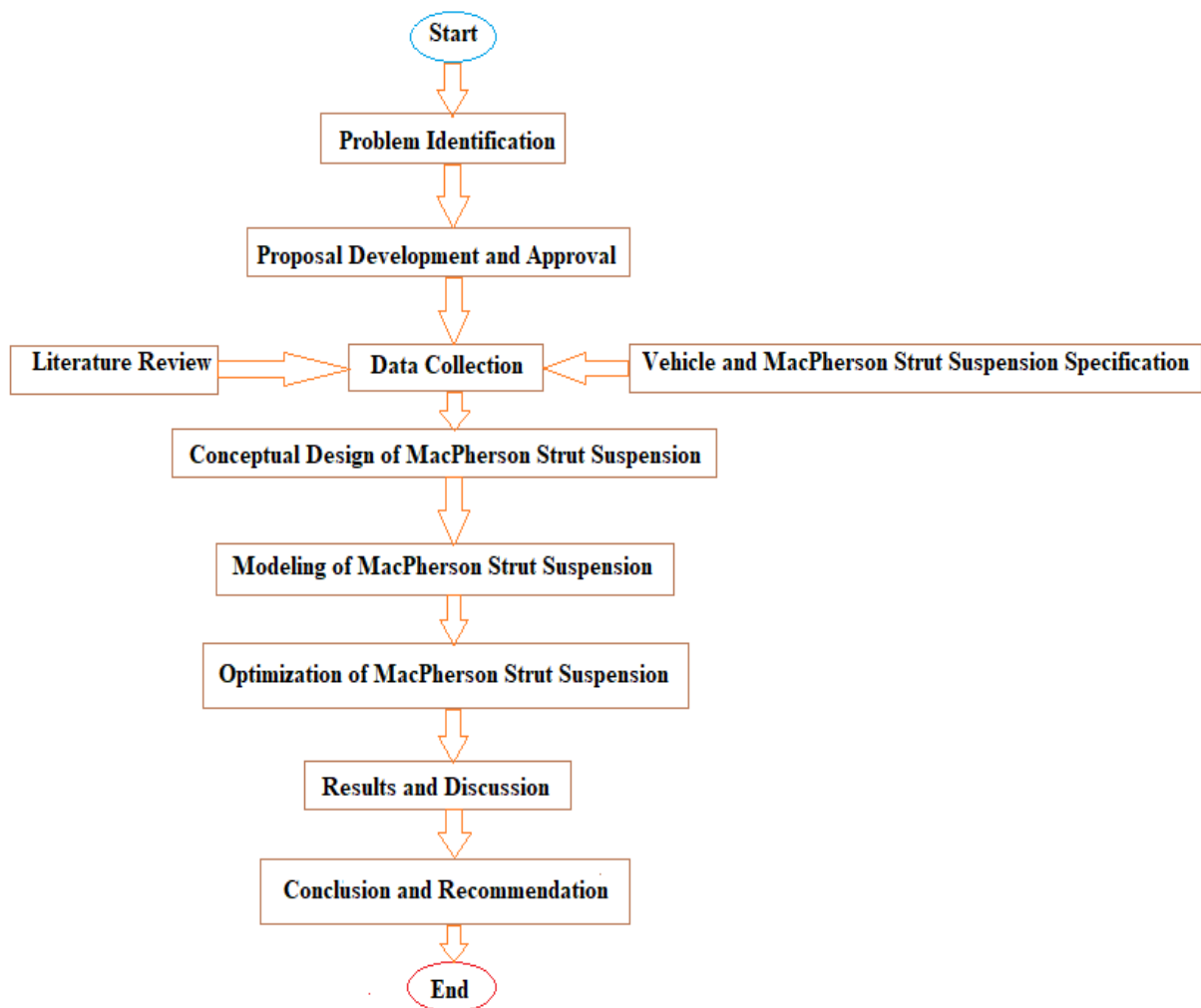


Figure 3.1: A Methodology flowchart

3.1. Research Design

This thesis focusses on a kinematic analysis of MPSSS and its modification to reduce the effects on wheel alignment. The study used a mixed-methods approach, combining quantitative and qualitative data collection methods. The quantitative data survey focusses to collect data on the general specification of the vehicle, and the dimensions of the MPSSS. Therefore, data is collected from the manufacturer of the selected vehicle. The qualitative data focusing on the methods, approaches and procedures of the MPSSS modeling is collected through document review. Based on the collected data, the thesis will develop a kinematic model of MPSSS. Moreover, by investigating the critical factors contributing to misalignment, the thesis endeavors suitable design optimizations to the MPSSS that can effectively mitigate the adverse effects on wheel alignment.

3.2. Data Collection

In this thesis, for data collection, a mixed-method approach is employed. It consists of two main approaches: document review, and vehicle and MPSSS specification. A review of documents related to design and modeling of suspension systems, kinematic analysis of MPSSS, and wheel alignment are conducted. This encompasses studying literature on various types of suspension systems, design, modeling and kinematic analysis of MPSSS, the effects of MPSSS design on wheel alignment, and design modification of MPSSS for better wheel alignment. For modeling and simulation of a MPSSS for lightweight vehicles to reduce the vehicle weight and improve its fuel economy, the specification of the selected vehicle and its leaf spring are required. Thus, the specifications of Toyota Corolla 2022 model like that of overall dimensions, weight of the vehicle, fuel economy and the dimensions of the leaf spring are collected from the website of the manufacturer. For this particular thesis, Toyota Corolla 2022 model, shown in *Figure 3.2*, is considered for the modeling and FEA of a MPSSS. Hence, the specification shown in *Table 3.1* are collected accordingly.

Table 3.1: Toyota Corolla 2022 Model Specification (Edmunds, 2022)

S.No.	Parameter	Value
1	Length (m)	4.639
2	Width (m)	1.780
3	Height (m)	1.455
4	Wheelbase (m)	2.700
5	Ground Clearance (m)	0.135
6	Number of Cylinder	4
7	Displacement (Liter)	1.8
8	Turning Circle (m)	0.90424
9	Power (kW)	103.65 @ 6,100 rpm
10	Torque (Nm)	170.83 @ 3,900 rpm
11	Transmission	Continuously variable transmission (CVT)
12	Drivetrain	Front wheel drive
13	Fuel Tank Capacity (Liter)	49.96744
14	Kerb Weight (kg)	1290
15	Payload (kg)	460
16	Suspension type	Four-wheel independent suspension



Figure 3.2: Toyota Corolla 2022 Model (Edmunds, 2022)

The Toyota Corolla 2022 model has an independent MPSSS with stabilizer bar, coil springs, and hydraulic shock absorbers. The rear suspension of the Toyota Corolla 2022 model is a torsion beam suspension with coil springs. The MPSSS is a type of independent suspension system used in the Toyota Corolla 2022 model.

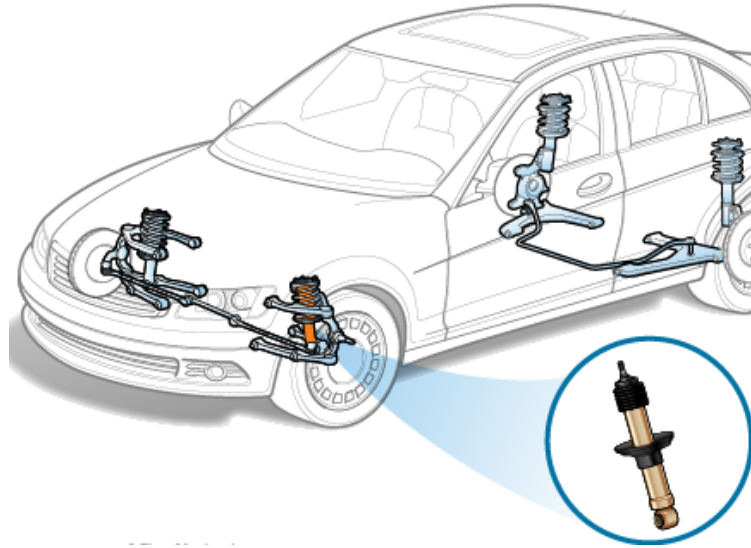


Figure 3.3: MacPherson strut front suspension in Toyota car (Edmunds, 2022)

Thus, various specifications of MPSSS, shown in *Table 3.2* is collected. This includes the angle of the strut assembly, the length of the suspension spring, the length and angle of the toe links, camber angle, and the angle of the lower control arm. These parameters are used to develop appropriate analytical model that is used for the kinematic analysis MPSSS.

Table 3.2: Toyota Corolla 2022 Model Specification (Edmunds, 2022)

S.No.	Parameter	Front	Rear
1	Strut assembly angle ($^{\circ}$)	1.5	-1.5
2	Length of the suspension spring (m)	0.36	0.37
3	Length of Toe links (m)	0.5	0.51
4	Angle of Toe Links ($^{\circ}$)	1.5	-1.5
5	Camber angle ($^{\circ}$)	-0.5	-1.0
6	Angle of the lower control arm ($^{\circ}$)	10.0	12.0

3.3. Conceptual Design of Macpherson Strut Suspension System

After collecting the technical specification of the Toyota Corolla 2022 model, and its MPSSS's dimensions, the conceptual design of the MPSSS is performed. This is done by considering the kinematic analysis of MPSSS.

Kinematics, in the context of vehicle suspension systems, focuses on the controlled positioning of road wheels through suspension links, assuming rigid parts and frictionless joints. It delves into suspension setup, geometry, and kinematic behavior, emphasizing the necessity of setting angles precisely to achieve peak performance and avoid unexpected behaviors (Nasiri et al., 2015).

In the MPSSS, its kinematics dictate the precise alignment of components like the lower control arm, spindle, and strut to ensure optimal wheel movement and stability (Patil, 2016). The geometry and arrangement of these components affect how the wheel moves and reacts to road irregularities, ultimately influencing the vehicle's handling and ride comfort. Adjusting these elements within specified parameters is critical for enhancing both performance and safety on the road (Shojaeefard et al., 2017).

The kinematic analysis of a MPSSS is concerned with the relationship between the motion of the wheel and the motion of the chassis without considering the forces that cause the motion. It involves:

- ✓ The kinematic modeling of the suspension system,
- ✓ Derivation of constraint equations for kinematic links of the suspension system, and
- ✓ Analysis of the displacement, velocity, and acceleration of the suspension system.

3.3.1. Kinematic parameters of Macpherson strut suspension system

The kinematic parameters of a MPSSS, shown in *Figure 3.4*, encompass a set of critical variables that define the dynamic behavior of a vehicle's wheel alignment (Wang et al., 2016). These parameters include elements such as camber, caster, toe, and kingpin inclination (KPI), each playing a vital role in determining the vehicle's stability, handling, and overall

performance on the road (Shi et al., 2019). They represent specific angles and orientations of the wheels and suspension components, influencing how the vehicle interacts with the road surface and responds to steering inputs (Reddy et al., 2016). Understanding and fine-tuning these parameters is fundamental to optimizing the vehicle's behavior, ensuring precise handling, and enhancing both safety and comfort for the driver and passengers (Fallah et al., 2014).

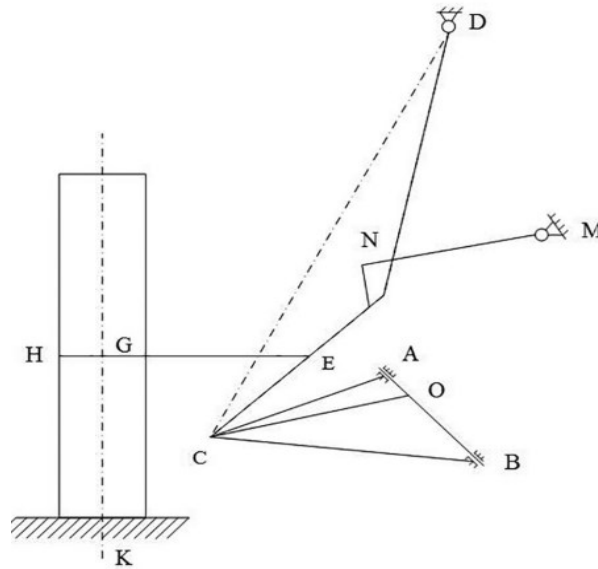


Figure 3.4: Kinematic mechanism of the MacPherson suspension system (Shi et al., 2019)

The points on the *Figure 3.4*, can be discussed as below:

- ✓ The Point A is the front joint of the lower control arm,
- ✓ The Point B is the rear joint of the lower control arm,
- ✓ The Joint C is the connection between the knuckle and lower control arm,
- ✓ The Point D is the joint of the top of suspension and vehicle body,
- ✓ The Point E is the damper's lower installed position,
- ✓ The Point G is the point of the wheel center,
- ✓ The Point H is the center of the outer surface of wheel,
- ✓ The Point K is the ground point of the wheel,

- ✓ The point N is the inner joint of the tie rod, and
- ✓ The point M is the outer joint of the tie rod.

a. Camber Angle

Camber angle, as shown in *Figure 3.5*, refers to the tilt of a vehicle's wheel when viewed from the front, either inward or outward at the top (Reddy et al., 2016). A positive camber occurs when the tire tilts outward, away from the vehicle, while a negative camber happens when the tire points inward, toward the vehicle (Saviotto, 2020). Neutral camber, with a zero angle, indicates wheels that are parallel to the vertical axis and perpendicular to the ground, offering a balanced, non-angled orientation for optimal tire contact with the road surface (Saraf and Satpute, 2017).

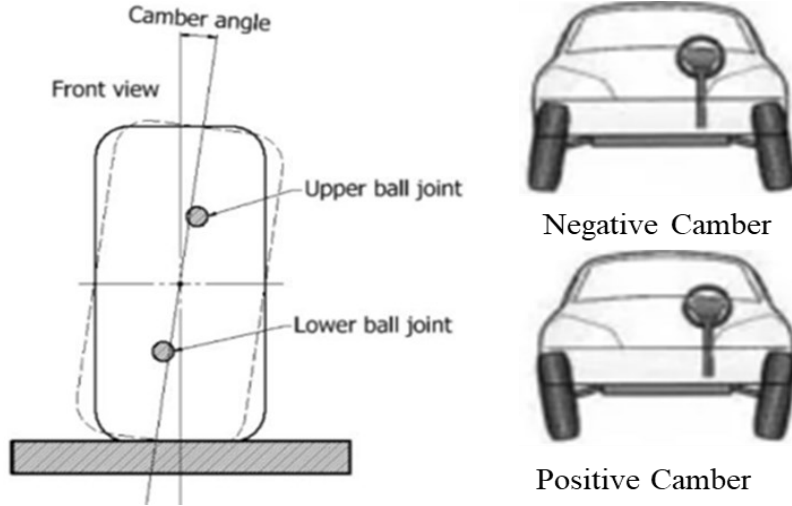


Figure 3.5: Camber angle terminologies (Patil, 2016; Saviotto, 2020)

Based on *Figure 3.4*, camber angle can be determined as:

$$\gamma_c = \tan^{-1} \left[\frac{(z_G - z_H)}{(y_G - y_H)} \right] \quad \text{Equation 3.1}$$

Where,

- ✓ γ_c is the camber angle,
- ✓ y_G and z_G are point G's the y and z coordinates, and

✓ y_H and z_H are point H's the y and z coordinates.

During cornering, a vehicle's outside wheels experience positive camber, reducing lateral grip due to the tire's altered angle relative to the ground. Suspension designers strategically incorporate positive camber in bump and negative camber in rebound motions to address this issue, optimizing tire contact and wear (Saviotto, 2020). Slight negative camber, particularly on a loaded vehicle, maximizes tire contact for even wear and decreased rolling resistance on curved roads. The relationship between camber and slip angle dictates cornering force, with positive camber increasing slip angle and negative camber enhancing cornering force (Reddy et al., 2016).

Proper camber settings significantly improve road isolation and directional stability, preventing tire wear and handling issues. In race cars, negative camber is predominantly used to bolster grip and high-speed stability, while positive camber examples often resulted in suboptimal race outcomes, emphasizing the critical role of camber in tire grip and overall vehicle performance (Shojaeefard et al., 2017).

b. Caster Angle

The caster angle, as shown in *Figure 3.6*, determined by the tilt of the steering axis in relation to the vertical line from the wheel's side view, impacts a vehicle's directional stability and steering wheel returnability (Saviotto, 2020). Positive caster occurs when the top of the steering axis tilts backward, while negative caster happens when it tilts forward. This angle's primary role is to enhance steering stability and facilitate the steering wheel's ability to return to its central position after turns (Liang and Xin, 2012). Incorrect caster angles can lead to increased steering effort and potential tire wear, emphasizing the significance of proper caster settings for optimal steering performance and tire longevity (Saraf and Satpute, 2017).

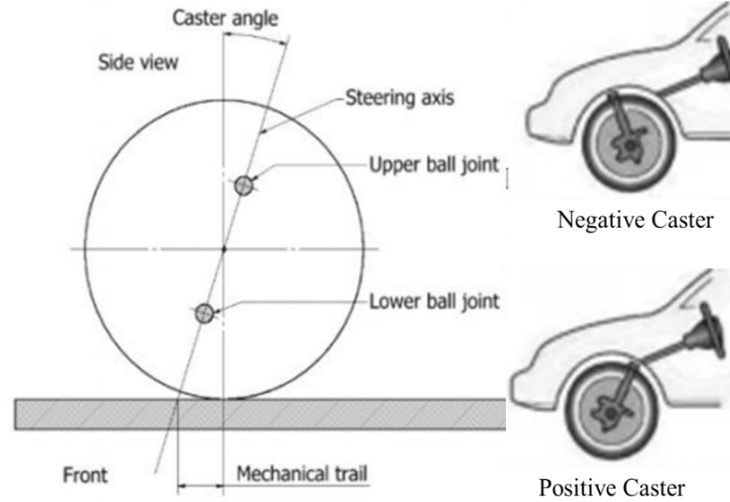


Figure 3.6: Caster angle terminologies (Patil, 2016; Saviotto, 2020)

Based on *Figure 3.4*, caster angle can be determined as:

$$\gamma_{cas} = \tan^{-1} \left[\frac{(x_C - x_D)}{(z_C - z_D)} \right] \quad \text{Equation 3.2}$$

Where,

- ✓ γ_{cas} is the caster angle,
- ✓ x_C and z_C are point C's the x and z coordinates, and
- ✓ x_D and z_D are point D's the x and z coordinates.

In race cars, positive caster is the preferred setup for its stability and handling advantages during cornering, while negative caster is seldom used due to reduced stability and poorer handling (Shojaeefard et al., 2017). Neutral caster, found in setups avoiding negative caster, offers some steering assistance without compromising stability, yet lacks the benefits of positive caster's handling enhancements. Positive caster, widely adopted in the vehicle industry, provides better cornering, increased stability at higher speeds, and improved straight-line tracking due to the self-aligning torque it generates (Abdelrahman and Nabawy, 2020). However, extreme dynamic camber resulting from positive caster can lead to mid-corner understeering, and an increase in steering effort proportional to the positive caster might be

noticeable, particularly in vehicles lacking power steering. Despite these drawbacks, positive caster remains popular due to its overall performance benefits (Shi et al., 2019).

c. Kingpin Inclination

The kingpin angle, as shown in *Figure 3.7*, determined by the steering axis projection and the vertical line to the road, influences steering effort in conjunction with wheel offset and scrub radius (Saviotto, 2020). Positive KPI values are primarily studied due to the absence of negative or neutral KPI in most suspension types. KPI, along with the scrub radius, impacts the lifting force and self-centering effect, increasing steering effort but enhancing straight-line stability, particularly at lower speeds (Berghian et al., 2018).

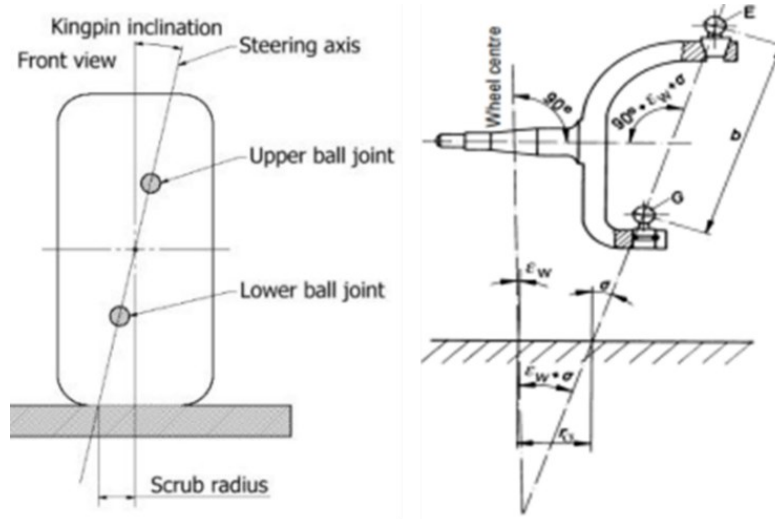


Figure 3.7: Kingpin Inclination (Berghian et al., 2018; Deo, 2007)

Based on *Figure 3.4*, kingpin inclination angle can be determined as:

$$\gamma_{kp} = \tan^{-1} \left[\frac{(y_C - y_D)}{(z_C - z_D)} \right] \quad \text{Equation 3.3}$$

Where,

- ✓ γ_{kp} is the kingpin inclination angle,
- ✓ y_C and z_C are point C's the y and z coordinates, and

✓ y_D and z_D are point D's the y and z coordinates.

During cornering, the kingpin angle contributes to lifting the car and facilitating steering return as weight transfer realigns, although achieving maximum stability by aligning the kingpin axis with the tire's vertical axis is practically challenging (Reddy et al., 2016). This inclination also affects the wheel's camber angle, leading to positive camber, which in turn aids in overall vehicle stability. The intricate relationship between these angles and forces within the suspension system significantly influences steering behavior and the vehicle's stability under various driving conditions (Liang and Xin, 2012).

d. Toe angle

Total toe, as shown in *Figure 3.8*, the difference between the front and rear distances across the tires or the measure of the leading edge's inward or outward orientation when viewed from the top, defines toe alignment (Su et al., 2018). Excessive positive or negative toe, whether the front or rear tires are closer together, leads to wear and scuffing on the tire shoulders. Positive toe (toe-in) occurs when the front tires are closer together than the rear, while negative toe (toe-out) happens when the rear tires are closer together than the front (Anumukonda et al., 2021).

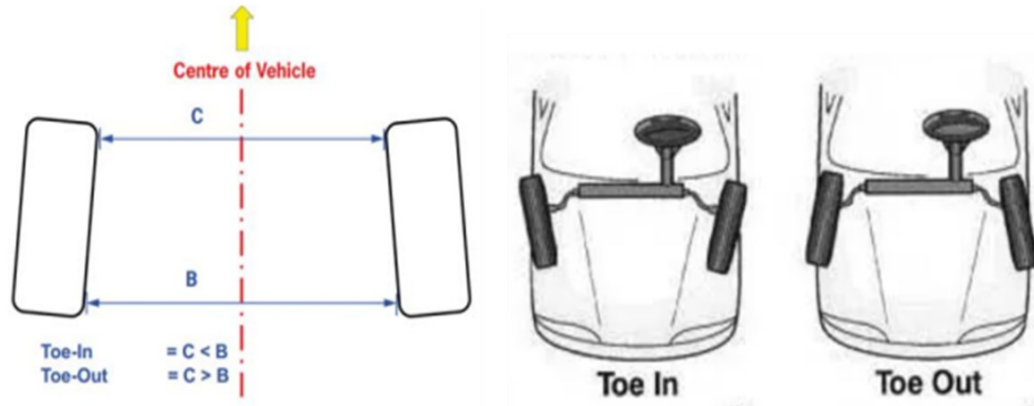


Figure 3.8: Toe angle terminologies (Berghian et al., 2018; Deo, 2007)

Based on *Figure 3.4*, toe angle can be determined as:

$$\gamma_t = \tan^{-1} \left[\frac{(x_G - x_H)}{(y_G - y_H)} \right] \quad \text{Equation 3.4}$$

Where,

- ✓ γ_t is the toe angle,
- ✓ x_G and y_G are point G's the x and y coordinates, and
- ✓ x_H and y_H are point H's the x and y coordinates.

Negative toe angle (toe out) enhances turning and cornering stability but can induce yaw motion, leading to a wobbly feel, especially at higher speeds. In the rear, negative toe improves grip for acceleration but might cause oversteer in rear-wheel drive vehicles (Chen et al., 2012). A neutral toe angle minimizes tire wear and power loss, offering a balanced approach in both front and rear suspension systems, prioritizing stability in straight-line driving while compromising stability during cornering (Shi et al., 2019). Positive toe angle (toe in) bolsters straight-line stability and grip, enabling higher acceleration, but accelerates tire temperature increase, potentially reducing tire life and energy efficiency. Uneven tire wear due to poor wheel adjustment and slower steering response are also drawbacks of a positive toe angle setup (Abdelrahman and Nabawy, 2020).

Proper toe adjustment is crucial in maintaining even tire wear and optimizing vehicle handling and performance. The choice of toe angle setup involves trade-offs between straight-line stability, cornering performance, and tire longevity, each configuration offering distinct advantages and drawbacks for vehicle handling and tire wear (Deo, 2007).

3.3.2. Kinematic model of Macpherson strut suspension system

The kinematic modeling of the suspension system involves the identification of the kinematic links of the suspension system and the determination of the degrees of freedom of the suspension system (Saviotto, 2020). It is a framework that elucidates the intricate dynamics governing the movement and behavior of this suspension design (Nasiri et al., 2015). This model intricately analyzes the interplay of components such as the lower control arm, strut,

spindle, and various joints within the system, shedding light on how these elements collectively impact the wheel's motion, alignment, and response to external forces (Wang et al., 2016).

As shown in *Figure 3.9*, the kinematic model MPSSS comprises a quarter-car body, a spindle and a tire, a helical spring, control arm, load disturbance, f_d , and an actuation force, f_a . The MPSSS is excited by road disturbance, z_r . When the mass of the strut is ignored and the bushing at point D is assumed to be a pin joint, the structure consists of two degrees of freedom (DOF) including the vertical displacement of the sprung mass and the rotational motion of the control arm (Fallah et al., 2014).

Figure 3.9: Simplified model of Macpherson suspension of right wheel (Fallah et al., 2014)

- ✎ The sprung mass has only vertical displacement and the motion of the body in other directions is zero.
- ✎ The unsprung mass (spindle and tire) is connected to the car body through the damper and spring and through the control arm.
- ✎ The values of z_s and θ are measured from the static equilibrium position and are considered as generalized coordinates.
- ✎ The camber angle is assumed to be zero when the suspension system is at static equilibrium.
- ✎ Compared to other links, the mass of the strut is negligible.
- ✎ The coil spring deflection, the tire deflection and the damping force are assumed to be linear.

As shown in *Figure 3.9*, the origin of the coordinate system, O, is at point B. Moreover, (y_A, z_A) , (y_B, z_B) , (y_C, z_C) , (y_D, z_D) , (y_J, z_J) and (y_P, z_P) , denotes the coordinates of the points A, B, C, D, J and P, respectively. However, under road disturbances, z_r , the position of the key points on the suspension changes. Let points A, B, C, D, J and P be at (y_{A1}, z_{A1}) , (y_{B1}, z_{B1}) , (y_{C1}, z_{C1}) , (y_{D1}, z_{D1}) , (y_{J1}, z_{J1}) and (y_{P1}, z_{P1}) , respectively, under static equilibrium conditions with the origin of the coordinate system. For instance, under road disturbances, the positions of the key points on the chassis are changing as:

$$y_B = 0 \quad \text{Equation 3.5}$$

Where,

- ✓ y_B is y coordinates of point B

$$z_B = z_s \quad \text{Equation 3.6}$$

Where,

- ✓ z_B is the point B's z coordinate, and
- ✓ z_s is the static equilibrium position.

$$y_D = y_{D1}$$

Equation 3.7

Where,

- ✓ y_D is the point D's y coordinate, and
- ✓ y_{D1} is the static equilibrium position of point D.

$$z_D = z_{D1} + z_s$$

Equation 3.8

Where,

- ✓ z_D is the point D's z coordinate,
- ✓ z_{D1} is the static equilibrium position of point D, and
- ✓ z_s is the static equilibrium position.

In addition, let the sprung mass be translated by z_s upwards and the control arm be rotated by θ in the counterclockwise direction.

The displacement components of the main points on the spindle are:

$$\begin{bmatrix} y_C & y_J & y_P \\ z_C & z_J & z_P \\ 1 & 1 & 1 \end{bmatrix} = \begin{bmatrix} a_{11} & a_{12} & y_A - (a_{11}y_{A1} + a_{12}z_{A1}) \\ a_{21} & a_{22} & z_A - (a_{21}y_{A1} + a_{22}z_{A1}) \\ 0 & 0 & 1 \end{bmatrix} \begin{bmatrix} y_{C1} & y_{J1} & y_{P1} \\ z_{C1} & z_{J1} & z_{P1} \\ 1 & 1 & 1 \end{bmatrix}$$

Equation 3.9

Where,

- ✓ y_C and z_C are point C's y and z coordinates,
- ✓ y_J and z_J are point J's y and z coordinates,
- ✓ y_P and z_P are point P's y and z coordinates,
- ✓ a_{11} , a_{12} , a_{21} and a_{22} are the coefficients,
- ✓ y_A and z_A are point A's y and z coordinates,
- ✓ y_{A1} and z_{A1} are the y and z coordinates of static equilibrium position of point A,
- ✓ y_{C1} and z_{C1} are the y and z coordinates of static equilibrium position of point C,
- ✓ y_{J1} and z_{J1} are the y and z coordinates of static equilibrium position of point J, and

- ✓ y_{P1} and z_{P1} are the y and z coordinates of static equilibrium position of point P.

Moreover,

$$\begin{aligned} a_{11} &= a_{22} = \cos\varphi \\ a_{12} &= -a_{21} = \sin\varphi \end{aligned} \quad \text{Equation 3.10}$$

Where,

- ✓ φ is the rotation angle of the wheel about x-axis.

Equation 3.9 consists of six set of equations. However, there are nine unknown parameters, which are (y_A, z_A) , (y_C, z_C) , (y_J, z_J) , (y_P, z_P) , and φ .

3.3.3. Constraint equations of MacPherson strut suspension system

To solve set of equations in *Equation 3.9*, it is necessary to use constraint equations of the mechanism. Since a link including cylindrical joint and spherical joint establishes the strut of the MPSSS, the constraint of this link at points C and J can be described as follows (Saviotto, 2020):

$$z_C = \alpha(y_C - y_D) + z_D \quad \text{Equation 3.11}$$

Where,

- ✓ y_C and z_C are point C's y and z coordinates,
- ✓ y_D and z_D are point D's y and z coordinates, and
- ✓ α is the slope of the strut with respect to the vertical axis

$$z_J = \alpha(y_J - y_D) + z_D \quad \text{Equation 3.12}$$

Where,

- ✓ y_D and z_D are point D's y and z coordinates,
- ✓ y_J and z_J are point J's y and z coordinates, and
- ✓ α is the slope of the strut with respect to the vertical axis

The control arm is a link between a revolute joint and a spherical joint. It rotates around a fixed axis and has a constant length. Thus, the constraint equations of control arm are (Fallah et al., 2014):

$$y_A = L_A \cos(\theta + \theta_1) + y_B \quad \text{Equation 3.13}$$

Where,

- ✓ y_A is point A's y coordinate,
- ✓ L_A is the length of the control arm,
- ✓ θ is the angle of the control arm,
- ✓ θ_1 is the initial angle of the control arm resulting from static deflection,
- ✓ y_B is point B's y coordinate.

$$z_A = L_A \sin(\theta + \theta_1) + z_B \quad \text{Equation 3.14}$$

Where,

- ✓ z_A is point A's z coordinate,
- ✓ L_A is the length of the control arm,
- ✓ θ is the angle of the control arm,
- ✓ θ_1 is the initial angle of the control arm resulting from static deflection,
- ✓ z_B is point B's z coordinate.

$$L_A^2 = (y_A - y_B)^2 + (z_A - z_B)^2 \quad \text{Equation 3.15}$$

Where,

- ✓ L_A is the length of the control arm,
- ✓ y_A and z_A are point A's y and z coordinates, and
- ✓ y_B and z_B are point B y and z coordinates.

With the addition of *Equation 3.11* to *Equation 3.15*, the number of unknowns increase to ten. Thus,

$$y_c = a_{11}(y_{c1} - y_{A1}) + a_{21}(z_{A1} - z_{c1}) + y_A \quad \text{Equation 3.16}$$

$$z_c = a_{21}(y_{c1} - y_{A1}) + a_{11}(z_{c1} - z_{A1}) + z_A \quad \text{Equation 3.17}$$

$$y_J = a_{11}(y_{J1} - y_{A1}) + a_{21}(z_{A1} - z_{J1}) + y_A \quad \text{Equation 3.18}$$

$$z_J = a_{21}(y_{J1} - y_{A1}) + a_{11}(z_{J1} - z_{A1}) + z_A \quad \text{Equation 3.19}$$

$$y_P = a_{11}(y_{P1} - y_{A1}) + a_{21}(z_{A1} - z_{P1}) + y_A \quad \text{Equation 3.20}$$

$$z_P = a_{21}(y_{P1} - y_{A1}) + a_{11}(z_{P1} - z_{A1}) + z_A \quad \text{Equation 3.21}$$

$$z_c = \alpha(y_c - y_{D1}) + (z_{D1} - z_s) \quad \text{Equation 3.22}$$

$$z_J = \alpha(y_J - y_{D1}) + (z_{D1} - z_s) \quad \text{Equation 3.23}$$

$$y_A = L_A \cos(\theta + \theta_1) \quad \text{Equation 3.24}$$

$$z_A = L_A \sin(\theta + \theta_1) + z_s \quad \text{Equation 3.25}$$

Where,

- ✓ y_c and z_c are point C's y and z coordinates,
- ✓ y_J and z_J are point J's y and z coordinates,
- ✓ y_P and z_P are point P's y and z coordinates,
- ✓ a_{11} , a_{12} , a_{21} and a_{22} are the coefficients,
- ✓ y_A and z_A are point A's y and z coordinates,
- ✓ y_{A1} and z_{A1} are the y and z coordinates of static equilibrium position of point A,
- ✓ y_{c1} and z_{c1} are the y and z coordinates of static equilibrium position of point C,
- ✓ y_{J1} and z_{J1} are the y and z coordinates of static equilibrium position of point J,
- ✓ y_{P1} and z_{P1} are the y and z coordinates of static equilibrium position of point P.
- ✓ L_A is the length of the control arm,

- ✓ y_A and z_A are point A's y and z coordinates,
- ✓ y_B and z_B are point B's y and z coordinates,
- ✓ θ is the angle of the control arm, and
- ✓ θ_1 is the initial angle of the control arm resulting from static deflection.

Moreover, when solving the aforementioned system of equations, φ would be determined as a function of the generalized coordinates θ and z_s . The value of φ as a function of θ and z_s is:

$$\varphi = -\frac{[S_1 + S_2 \cos(\theta + \theta_1) + S_3 \sin(\theta + \theta_1)]}{2} + \frac{\sqrt{[S_1 + S_2 \cos(\theta + \theta_1) + S_3 \sin(\theta + \theta_1)]^2 - 4[S_4 + S_2 \cos(\theta + \theta_1) + S_4 \sin(\theta + \theta_1)]}}{2} \quad \text{Equation 3.26}$$

Where,

- ✓ φ is the rotation angle of the wheel about x-axis.
- ✓ θ is the angle of the control arm,
- ✓ θ_1 is the initial angle of the control arm resulting from static deflection, and

$$S_1 = \frac{(y_{C1} - y_{J1})y_{D1} + (z_{J1} - z_{C1})z_{D1}}{(y_{C1} - y_{A1})(z_{J1} - z_{A1}) + (z_{A1} - z_{C1})(y_{J1} - y_{A1})} \quad \text{Equation 3.27}$$

$$S_2 = \frac{(y_{J1} - y_{C1})L_A}{(y_{C1} - y_{A1})(z_{J1} - z_{A1}) + (z_{A1} - z_{C1})(y_{J1} - y_{A1})} \quad \text{Equation 3.28}$$

$$S_3 = \frac{(z_{C1} - z_{J1})L_A}{(y_{C1} - y_{A1})(z_{J1} - z_{A1}) + (z_{A1} - z_{C1})(y_{J1} - y_{A1})} \quad \text{Equation 3.29}$$

$$S_4 = 1 + \frac{(z_{C1} - z_{J1})y_{D1} + (y_{C1} - y_{J1})z_{D1}}{(y_{C1} - y_{A1})(z_{J1} - z_{A1}) + (z_{A1} - z_{C1})(y_{J1} - y_{A1})} \quad \text{Equation 3.30}$$

And, subsequently the other unknown parameters, including (y_A, z_A) , (y_J, z_J) , (y_P, z_P) , (y_C, z_C) and α can be specified. So far, the displacements of all key points have been determined as functions of the independent variables θ and z_s .

The next step is to find the velocities of key points. The system of velocity components of the main points are obtained from the above equations as follows (Fallah et al., 2014):

$$\dot{y}_C = \dot{\phi}(z_A - z_C) + \dot{y}_A \quad \text{Equation 3.31}$$

$$\dot{z}_C = \dot{\phi}(y_C - y_A) + \dot{z}_A \quad \text{Equation 3.32}$$

$$\dot{y}_J = \dot{\phi}(z_A - z_J) + \dot{y}_A \quad \text{Equation 3.33}$$

$$\dot{z}_J = \dot{\phi}(y_J - y_A) + \dot{z}_A \quad \text{Equation 3.34}$$

$$\dot{y}_P = \dot{\phi}(z_A - z_P) + \dot{y}_A \quad \text{Equation 3.35}$$

$$\dot{z}_P = \dot{\phi}(y_P - y_A) + \dot{z}_A \quad \text{Equation 3.36}$$

$$\dot{z}_C = \alpha \dot{y}_C + \dot{\alpha}(y_C - y_D) + \dot{z}_S \quad \text{Equation 3.37}$$

$$\dot{z}_J = \alpha \dot{y}_J + \dot{\alpha}(y_J - y_D) + \dot{z}_S \quad \text{Equation 3.38}$$

$$\dot{y}_A = -L_A \sin(\theta + \theta_1) \dot{\theta} \quad \text{Equation 3.39}$$

$$\dot{z}_A = L_A \cos(\theta + \theta_1) \dot{\theta} + \dot{z}_S \quad \text{Equation 3.40}$$

Where,

- ✓ $\dot{y}_C$ and $\dot{z}_C$ are the y and z components of point C velocity,
- ✓ y_C and z_C are point C's y and z coordinates,
- ✓ $\dot{y}_J$ and $\dot{z}_J$ are the y and z components of point J velocity,
- ✓ y_J and z_J are point J's y and z coordinates,
- ✓ $\dot{y}_P$ and $\dot{z}_P$ are the y and z components of point P velocity,
- ✓ y_P and z_P are point P's y and z coordinates,
- ✓ a_{11} , a_{12} , a_{21} and a_{22} are the coefficients,
- ✓ $\dot{y}_A$ and $\dot{z}_A$ are the y and z components of point A velocity,
- ✓ y_A and z_A are point A's y and z coordinates,
- ✓ y_{A1} and z_{A1} are the y and z coordinates of static equilibrium position of point A,
- ✓ y_{C1} and z_{C1} are the y and z coordinates of static equilibrium position of point C,

- ✓ y_{J1} and z_{J1} are the y and z coordinates of static equilibrium position of point J,
- ✓ y_{P1} and z_{P1} are the y and z coordinates of static equilibrium position of point P.
- ✓ L_A is the length of the control arm,
- ✓ y_B and z_B are point B's y and z coordinates,
- ✓ $\dot{\phi}$ is the velocity of rotation angle of the wheel about x-axis,
- ✓ α is the slope of the strut with respect to the vertical axis,
- ✓ $\dot{\alpha}$ is the velocity of the slope of the strut with respect to the vertical axis,
- ✓ $\dot{\theta}$ is the velocity of angle of the control arm,
- ✓ θ is the angle of the control arm, and
- ✓ θ_1 is the initial angle of the control arm resulting from static deflection.

In solving the aforementioned system of equations, the value of $\dot{\phi}$ is determined as follows:

$$\dot{\phi} = \frac{(\dot{z}_A - \alpha \dot{y}_A - \dot{z}_S)(y_C - y_J)}{h} \quad \text{Equation 3.41}$$

Where,

- ✓ $\dot{\phi}$ is the velocity of rotation angle of the wheel about x-axis,
- ✓ α is the slope of the strut with respect to the vertical axis,

$$h = (y_C - y_A + \alpha z_C - \alpha z_A)(y_J - y_D) - (y_J - y_A + \alpha z_J - \alpha z_A)(y_C - y_D) \quad \text{Equation 3.42}$$

3.3.4. Equations of motion

The equations of motion of the new model can be derived using Lagrange's method. The kinetic energy is given by:

$$E_k = \frac{1}{2}(m_s + m_{ca})(\dot{z}_S)^2 + \frac{1}{2}m_u(\dot{y}_P^2 + \dot{z}_P^2) + \frac{1}{2}I_u\dot{\phi}^2 + \frac{1}{2}I_{ca}^B\dot{\theta}^2 \quad \text{Equation 3.43}$$

Where,

- ✓ E_k is the kinetic energy,

- ✓ m_s is the mass of the body,
- ✓ m_u is the mass of the wheel,
- ✓ m_{ca} is the mass of the control arm,
- ✓ I_u is the moment of inertial of the wheel, and
- ✓ I_{ca}^B is the moment of inertial of the control arm at point B.

Whereas, the potential energy is:

$$E_p = \frac{1}{2}K_s(\Delta L)^2 + \frac{1}{2}K_t(\Delta z)^2 \quad \text{Equation 3.44}$$

Where,

- ✓ E_p is the potential energy,
- ✓ K_s is the stiffness coefficients of the sprung mass,
- ✓ K_t is the stiffness coefficients of the unsprung mass
- ✓ ΔL is the deflection of the spring, and
- ✓ Δz is the deflection of the tire.

The deflection of the spring can be determined from:

$$\Delta L = \sqrt{[(y_C - y_D)^2 + (z_C - z_D)^2]} - \sqrt{[(y_{C1} - y_{D1})^2 + (z_{C1} - z_{D1})^2]} \quad \text{Equation 3.45}$$

Where,

- ✓ ΔL is the deflection of the spring,
- ✓ y_C and z_C are point C's y and z coordinates,
- ✓ y_D and z_D are point D's y and z coordinates,
- ✓ y_{C1} and z_{C1} are the y and z coordinates of static equilibrium position of point C,
- ✓ y_{D1} and z_{D1} are the y and z coordinates of static equilibrium position of point D.

The deflection of the tire can be determined from:

$$\Delta z = (y_{A1} - y_{P1})\varphi + (z_{P1} - z_{A1}) + L_A \sin(\theta + \theta_1) + z_s - z_r \quad \text{Equation 3.46}$$

Where,

- ✓ Δz is the deflection of the tire,
- ✓ y_{P1} and z_{P1} are the y and z coordinates of static equilibrium position of point P,
- ✓ y_{A1} and z_{A1} are the y and z coordinates of static equilibrium position of point A,
- ✓ L_A is the length of the control arm,
- ✓ φ is the rotation angle of the wheel about x-axis,
- ✓ θ is the angle of the control arm, and
- ✓ θ_1 is the initial angle of the control arm resulting from static deflection.

The damping function, D, is given by:

$$D = \frac{1}{2} C_p (\Delta \dot{L})^2 \quad \text{Equation 3.47}$$

Where,

- ✓ D is the damping function,
- ✓ C_p is the damping coefficient, and
- ✓ $\Delta \dot{L}$ is the relative velocity of the damper.

The relative velocity of the damper can be determined from:

$$\Delta \dot{L} = [\dot{y}_C(y_C - y_D) + (\dot{z}_C - \dot{z}_D)(z_C - z_D)] \sqrt{[(y_C - y_D)^2 + (z_C - z_D)^2]} \quad \text{Equation 3.48}$$

Where,

- ✓ $\Delta \dot{L}$ is the relative velocity of the damper,
- ✓ $\dot{y}_C$ and $\dot{z}_C$ are the y and z coordinates velocity of point C,
- ✓ $\dot{y}_D$ and $\dot{z}_D$ are the y and z coordinates velocity of point D,
- ✓ y_C and z_C are point C's y and z coordinates, and
- ✓ y_D and z_D are point D's y and z coordinates.

$$\begin{aligned}
f_1 \ddot{z}_s + f_2 \ddot{\theta} = & (m_s + m_u + m_{ca}) \ddot{z}_s \\
& + m_u L_A \left[\cos(\theta + \theta_1) \right. \\
& + \left. \left[\frac{(y_C - y_J)(y_P - y_A)}{h} \right] \cos(\theta + \theta_1) \right. \\
& \left. + \alpha \sin(\theta + \theta_1) \right] \ddot{\theta}
\end{aligned}$$

Where,

- ✓ $\ddot{z}_s$ is the acceleration of static equilibrium position,
- ✓ $\ddot{\theta}$ is the acceleration of the angle of the control arm,
- ✓ m_s is the mass of the body,
- ✓ m_u is the mass of the wheel,
- ✓ m_{ca} is the mass of the control arm,
- ✓ L_A is the length of the control arm,
- ✓ θ is the angle of the control arm
- ✓ θ_1 is the initial angle of the control arm resulting from static deflection.
- ✓ y_C is point C's y coordinates,
- ✓ y_J is point J's y coordinates,
- ✓ y_P is point P's y coordinates, and
- ✓ y_A is point A's y coordinates.

Using Lagrange's equations along the generalized coordinates z_s and θ , and by substituting the values of $\dot{y}_P$ and $\dot{z}_P$

Equation 3.50

$$\begin{aligned}
f_3 \ddot{z}_s + f_4 \ddot{\theta} = m_u & \left[L_A \cos(\theta + \theta_1) + (y_P - z_A) \frac{\partial \dot{\phi}}{\partial \dot{\theta}} \right] \ddot{z}_s \\
& + \left[m_u L_A \left[\{ \cos(\theta + \theta_1) \right. \right. \\
& + \alpha \sin(\theta + \theta_1) \} \left\{ \frac{(y_C - y_J)(z_A - z_P)}{h} \right\} \\
& \left. \left. - \sin(\theta + \theta_1) \right] \left[-L_A \sin(\theta + \theta_1) + (z_A - z_P) \frac{\partial \dot{\phi}}{\partial \dot{\theta}} \right] \right. \\
& + m_u L_A \left[L_A \cos(\theta + \theta_1) \right. \\
& + (y_P - z_A) \frac{\partial \dot{\phi}}{\partial \dot{\theta}} \left. \right] \left[\{ \cos(\theta + \theta_1) \right. \\
& + \alpha \sin(\theta + \theta_1) \} \left\{ \frac{(y_C - y_J)(y_P - y_A)}{h} \right\} \\
& \left. \left. + \cos(\theta + \theta_1) \right] \right. \\
& \left. + \left[I_u L_A \cos(\theta + \theta_1) + \alpha \sin(\theta + \theta_1) \frac{\partial \dot{\phi}}{\partial \dot{\theta}} + I_{ca} \right] \ddot{\theta} \right]
\end{aligned}$$

Since the above equations are highly nonlinear, the higher-order nonlinearities are ignored in.

In addition, define:

Equation 3.51

$$\begin{aligned}
H_1 &= -\frac{\partial E_k}{\partial z_s} \\
H_2 &= \frac{\partial E_p}{\partial z_s} \\
H_3 &= -\frac{\partial D}{\partial \dot{z}_s} \\
H_4 &= -\frac{\partial E_k}{\partial \theta} \\
H_5 &= \frac{\partial E_p}{\partial \theta} \\
H_6 &= -\frac{\partial D}{\partial \dot{\theta}}
\end{aligned}$$

This results in:

$$f_1\ddot{z}_s + f_2\ddot{\theta} + H_1 + H_2 = \frac{\partial \vec{r}_C}{\partial \dot{z}_s} \cdot f_a + f_d + H_3 \quad \text{Equation 3.52}$$

$$f_1\ddot{z}_s + f_2\ddot{\theta} + H_1 + H_2 = \frac{\partial \vec{r}_C}{\partial \dot{z}_s} \cdot f_a + f_d + H_3 - H_1 - H_2 = F_1$$

And,

$$f_3\ddot{z}_s + f_4\ddot{\theta} + H_4 + H_5 = \frac{\partial \vec{r}_C}{\partial \dot{z}_s} \cdot f_a + H_6 \quad \text{Equation 3.53}$$

$$f_3\ddot{z}_s + f_4\ddot{\theta} + H_4 + H_5 = \frac{\partial \vec{r}_C}{\partial \dot{z}_s} \cdot f_a + H_6 - H_4 - H_5 = F_2$$

Therefore, the equations of motion are obtained as:

$$f_1(z_s, \dot{z}_s, \theta, \dot{\theta})\ddot{z}_s + f_2(z_s, \dot{z}_s, \theta, \dot{\theta})\ddot{\theta} = F_1(z_s, \dot{z}_s, \theta, \dot{\theta}) \quad \text{Equation 3.54}$$

$$f_3(z_s, \dot{z}_s, \theta, \dot{\theta})\ddot{z}_s + f_4(z_s, \dot{z}_s, \theta, \dot{\theta})\ddot{\theta} = F_2(z_s, \dot{z}_s, \theta, \dot{\theta})$$

When solving the aforementioned system of equations, the accelerations of the generalized coordinates are obtained as follows:

$$\ddot{z}_s = \frac{f_4 F_1 - f_2 F_2}{f_4 f_1 - f_2 f_3} = g_1(z_s, \dot{z}_s, \theta, \dot{\theta}) \quad \text{Equation 3.55}$$

$$\ddot{\theta} = \frac{f_1 F_2 - f_3 F_1}{f_4 f_1 - f_2 f_3} = g_2(z_s, \dot{z}_s, \theta, \dot{\theta})$$

At this point, introducing the state variables as $[x_1, x_2, x_3, x_4]^T = [z_s, \dot{z}_s, \theta, \dot{\theta}]^T$ yields:

$$\dot{x}_1 = x_2 \quad \text{Equation 3.56}$$

$$\dot{x}_2 = g_1(z_s, \dot{z}_s, \theta, \dot{\theta}, f_a, f_d, z_r)$$

$$\dot{x}_3 = x_4$$

$$\dot{x}_4 = g_2(z_s, \dot{z}_s, \theta, \dot{\theta}, f_a, f_d, z_r)$$

3.3.5. Linearization of equations

The highly nonlinear equations of motion are at the equilibrium state $(x_{1e}, x_{2e}, x_{3e}, x_{4e}) = (0,0,0,0)$. The resulting linearized equations are:

$$\begin{aligned}\dot{x} &= Ax(t) + B_1 f_a(t) + B_2 z_r(t) + B_3 f_d(t) \\ x(0) &= x_e\end{aligned}\tag{Equation 3.57}$$

Where,

$$\begin{aligned}\checkmark \quad A &= \begin{bmatrix} 0 & 1 & 0 & 0 \\ \frac{\partial g_1}{\partial x_1} & 0 & \frac{\partial g_1}{\partial x_3} & \frac{\partial g_1}{\partial x_4} \\ 0 & 0 & 0 & 1 \\ \frac{\partial g_2}{\partial x_1} & 0 & \frac{\partial g_2}{\partial x_3} & \frac{\partial g_2}{\partial x_4} \end{bmatrix}_{x_e} \\ \checkmark \quad B_1 &= \begin{bmatrix} 0 & \frac{\partial g_1}{\partial f_a} & 0 & \frac{\partial g_2}{\partial f_a} \end{bmatrix}_{f_a=0} \\ \checkmark \quad B_2 &= \begin{bmatrix} 0 & \frac{\partial g_1}{\partial z_r} & 0 & \frac{\partial g_2}{\partial z_r} \end{bmatrix}_{z_r=0} \\ \checkmark \quad B_3 &= \begin{bmatrix} 0 & \frac{\partial g_1}{\partial f_d} & 0 & \frac{\partial g_2}{\partial f_d} \end{bmatrix}_{f_d=0}\end{aligned}$$

Let the output variables $y(t)$ be:

$$y(t) = [z_s \theta]^T$$

Then, the output equation is obtained as:

$$y(t) = Cx(t) + D_1 f_a(t) + D_2 z_r(t) + D_3 f_d(t)$$

Where,

$$\begin{aligned}\checkmark \quad C &= \begin{bmatrix} 1 & 0 & 0 & 0 \\ 0 & 0 & 1 & 0 \end{bmatrix} \\ \checkmark \quad [D_1^T \quad D_2^T \quad D_3^T] &= 0\end{aligned}$$

3.4. Modeling, Optimization and Validation Softwares

After the required specifications of the Toyota Corolla 2020 model MPSSS are obtained through data collection, the software modeling of the MPSSS is performed using various softwares. Moreover, ADAMS CAR is used to perform the simulation of the MPSSS. Furthermore, DESIGN EXPERT 13 is used for optimization of selected parameters of the MPSS. The required softwares are listed in *Table 3.3*.

Table 3.3: Required softwares for CAD modeling and Analysis

S.No.	Software	Purpose
1	ADAMS CAR 2020	For MPSS Modeling and Simulation
2	MICROSOFT EXCEL 2021	For determining the relationship between parameters
3	DESIGN EXPERT 13	For Optimization
4	MATLAB R2023a	For Validation

ADAMS CAR is a specialized software program which is widely used to perform FEA in automotive industries (Liang and Xin, 2012). It provides a suite of options for accurately predicting the behavior of vehicle structures, including their response to load, vibration, impact, and fatigue. Furthermore, its scalability and flexibility enables users to customize the simulation environment to match their specific requirements (Saviotto, 2020).

MICROSOFT EXCEL, developed by Microsoft Corporation, is a versatile spreadsheet software widely used for data analysis, calculation, and visualization. It has several applications ranging from simple data entry to sophisticated statistical analysis and financial modeling (Microsoft, 2024). Determining the correlations between factors is one of its main advantages. With the aid of these features, users can compute correlation coefficients, carry out single and multiple regressions, and optimize solutions by establishing constraints and variables inside the program (Segal, 2024).

DESIGN EXPERT, developed by Stat-Ease Inc., is a software program specifically designed to assist with experimental design and statistical analysis (Stat-Ease Inc, 2024). It offers a wide range of applications, from screening factors to optimizing product formulations. It is widely

acclaimed for its intuitive interface and robust functionality (Ritme, 2024). With its array of advanced features including factorial, response surface, and mixture designs, DESIGN EXPERT streamlines the process of optimizing product formulations, processes, and designs (Stat-Ease Inc, 2024). This software boasts advantages such as user-friendly design wizards, analysis tools, and effective graphical outputs for communicating results (Science Plus Group, 2024).

MATLAB, developed by MathWorks, is a high-level programming language and environment widely used for mathematical modeling, data analysis, and visualization. It provides a range of applications such as machine learning, data analysis, and optimization (Mathworks, 2024). It offers the Neural Network Toolbox for creating and utilizing Artificial Neural Networks (ANNs) for a range of uses, including pattern recognition, regression analysis, and optimization. ANNs in MATLAB allow users to optimize solutions by varying the network's weights and biases, enabling the modeling of complex relationships between inputs and outputs (Mathworks, 2018).

CHAPTER FOUR

MODELING AND OPTIMIZATION OF MACPHERSON STRUT SUSPENSION SYSTEM

For the purpose of reducing the effects on wheel alignment, a kinematic analysis of the Macpherson strut suspension system using ADAMS CAR is performed. The model is designed based on the design specifications acquired from Toyota Corolla 2022 model. This is done by considering various suspension parameters such as camber angle, toe angle, kingpin inclination angle, and caster angle. In addition, DESIGN EXPERT is used for optimization of Macpherson strut suspension system. Moreover, for creating the CAD model, SOLIDWORKS software is used. The detail process used and model of MPSSS are discussed in the following sections.

4.1. Modeling of Macpherson Strut Suspension System

As shown in *Figure 4.1*, Macpherson strut suspension system consists of various parts, links, arms and connections. Thus, in order to analyze the kinematics of MPSS in ADAMS CAR software, there are six general processes that may be followed. These are:

- Creating hardpoints,
- Creating Parts,
- Defining joints and attachments,
- Creating communicators,
- Creating suspension sub assembly,
- Creating suspension assembly,
- Configuring and performing kinematic suspension analysis, and
- Plotting configuration files.

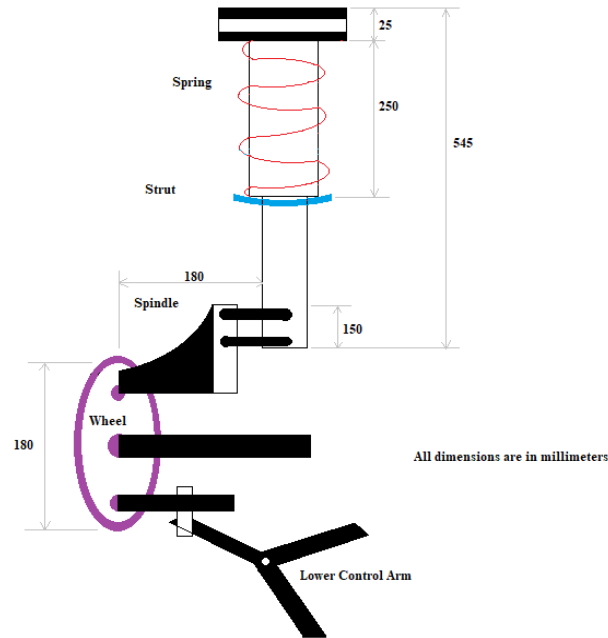


Figure 4.1: Simplified Model of MPSS

4.1.1. Hardpoints of Macpherson Strut Suspension System

Hardpoints, in ADAMS CAR, are the spatial points that are used to define key locations on the part of the suspension system. They are used to create parts, links, arms, and joints.

Table 4.1: The hardpoints of the current MPSS Model

S.No.	Name of the MPSS Parts	Hard_Points		
		x	y	z
1	Wheel_Center	0	-890	0
2	Arm_Outer	0	-710	-75
3	Arm_Front	-100	-360	-50
4	Arm_Rear	140	-360	-100
5	Strut_Lower	0	-710	240
6	Strut_Upper	0	-660	515
7	Spring_Lower	0	-710	250
8	Tie_Rod_Outer	150	-710	240
9	Tie_Rod_Inner	200	-360	240

4.1.2. Control Arm of Macpherson Strut Suspension System

The control arm of the MPSS is a triangle shaped object. Thus, as shown in *Table 4.1*, three hardpoints are needed to define the key locations of the control arm. These are, front arm, rear arm, and outer arm. Hence, by defining the hardpoints of these three positions, the geometry of control arm can be created by combining these positions, as shown in *Appendix 1*. Moreover, the thickness is added to create the real geometry of the control arm.

4.1.3. Wheel Carrier of Macpherson Strut Suspension System

Wheel carrier of MPSS is a triangle shaped object that consists of two general geometries as a form of arm and link. The carrier arm is based on wheel_center, strut_lower and arm_outer hardpoints. Whereas, carrier ink is based on strut_lower and tierod_outer hardpoints. Thus, based on hardpoints shown on *Table 4.1*, the geometry of wheel carrier is created as shown in *Appendix 2*. Then, the thickness is added to create the real geometry of the wheel carrier.

4.1.4. Tie rod of Macpherson Strut Suspension System

Tie rod of the MPSS is a cylinder-shaped object. Thus, as shown in *Table 4.1*, two hardpoints are needed to define the key locations of the tie rod. These are, inner tie rod, and outer tie rod. Hence, by defining the hardpoints of these two positions, the geometry of tie rod can be created by combining these positions. Moreover, the geometry of tie rod is created by adding the radius of the cylinder, as shown in *Appendix 3*.

4.1.5. Damper of Macpherson Strut Suspension System

Damper of the suspension is a cylinder-shaped object. Thus, two hardpoints are needed to define the key locations of the damper, as shown in. After hardpoints are created for damper by combining two defining locations of upper strut, and lower strut, the geometry of damper is created by adding the diameter of the cylinder, as shown in *Appendix 4*.

4.1.6. Spring of Macpherson Strut Suspension System

Spring of suspension system can be created by combining two defining locations of upper strut, and lower spring. Then, the geometry of spring is created by adding the diameter of the spring and number of coils, as shown in *Appendix 5*.

4.1.7. Strut of Macpherson Strut Suspension System

Strut of the suspension system can be created by entering the location, orientation, and mass properties for the strut part. However, since the strut geometry would not be visible from inside the damper, there is no need to give the strut any geometry. As such, the strut geometry can be created as shown in *Appendix 6*.

4.1.8. Suspension Parameters of Macpherson Strut Suspension System

Furthermore, suspension parameters and variables defining toe and camber angles needed to be created for the suspension analysis. These are the values of toe and camber angles at equilibrium state. Thus, toe and camber angles suspension parameters are defined as shown in *Appendix 7*.

4.1.9. Hub of Macpherson Strut Suspension System

Hub of a suspension parameter, which is a cylinder as shown in *Appendix 10*, can be created by combining hub bearing construction frame, shown in *Appendix 8* **Error! Reference source not found.**, and hub general part, shown in *Appendix 9*. Hub bearing construction frame is based on suspension parameters.

4.1.10. Joints and Attachments of Macpherson Strut Suspension System

After all the parts of the suspension system are created, it is required to create the joints and attachments that holds these parts. Moreover, it is required to create the degree of freedom for each part of the suspension system. The joints of the suspension system, as shown in *Table 4.2* include translational joint, revolute joint, bushing attachment, cylindrical joint and spherical joint.

Table 4.2: Joint and attachments of MPSS

S.No.	Joint Type	Part 1	Part 2
1	Revolute	Control Arm	Body subframe
2	Spherical	Control Arm	Wheel_Carrier
3	Bushing	Front Control Arm	Body subframe
4	Bushing	Front Control Arm	Body subframe
5	Translation	Wheel_Carrier	Strut
6	Bushing	Strut	Body subframe
7	Spherical	Strut	Body
8	Spherical	Tierod	Wheel_Carrier
9	Hooke	Tierod	Steering

4.1.11. Communicators of Macpherson Strut Suspension System

Communicators are the tools in Adams Car that define how the suspension is connected the suspension test rig. As shown in *Appendix 11*, this is done by defining the communicators that attach the hub to the suspension test rig at the wheel center. It includes the description of which part it needs to be connected to where the connection takes place, and which is the upright part.

4.1.2. Sub-assembly of Macpherson Strut Suspension System

After all the parts of the suspension system, and their joint and attachments are created using Adams Car template builder, the sub-assembly of the suspension system is created. This is done as shown in *Figure 4.2Appendix 12*. The sub-assembly consists of both left and right components of the suspension system.

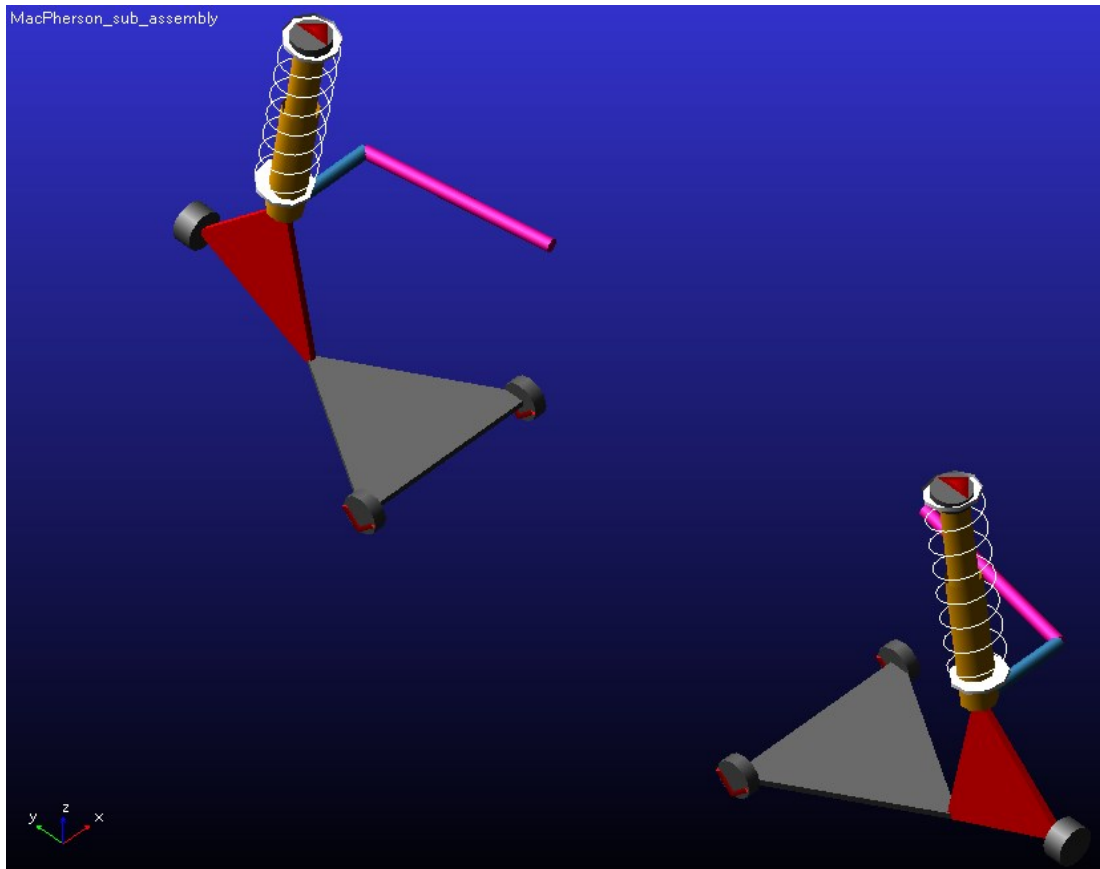


Figure 4.2: Creating a sub-assembly for suspension system in Adams Car

4.1.3. Assembly of Macpherson Strut Suspension System

After the sub-assembly of the suspension system is created using Adams Car standard model builder, the assembly of the suspension system is created. This is done as shown *Appendix 12*. In addition to the components of the suspension system, the assembly consists of wheels and road bumping model.

4.1.4. Kinematic Analysis of Macpherson Strut Suspension System

After the assembly of the suspension system is successfully created, using Adams Car standard model builder, the layout for the kinematic analysis of the suspension system is created. This is done as shown in *Appendix 13*. It is done in three step process. That is changing the kinematic

toggle to the kinematic mode, setting the suspension parameters and selecting a parallel wheel travel to set that both wheels are going to move upside down in a parallel movement.

After the parameters for kinematic analysis of the suspension system is created, using Adams Car standard model builder, the layout for the result display is created. The result of the kinematic analysis is in two forms, including plots and animations. The plot may include the variations of camber angle, caster angle, kingpin inclination angle versus wheel travel.

4.3. Optimization of Macpherson strut suspension system

As shown in *Table 4.1*, MPSS is modeled based on nine parts. Each of these parts consists of three hard points. That is, there are a total of twenty-seven hard points that may affect the performance of the MPSS. Thus, it is necessary to investigate the effect of these points on the performance of the MPSS. However, from those points, few of the points are based on the connection with the main vehicle parts. As such, there are constraints on these points. These points are, Wheel_Center, Strut_Upper, and Spring_Lower. Hence, by considering these constraints, eighteen points are selected for further analysis.

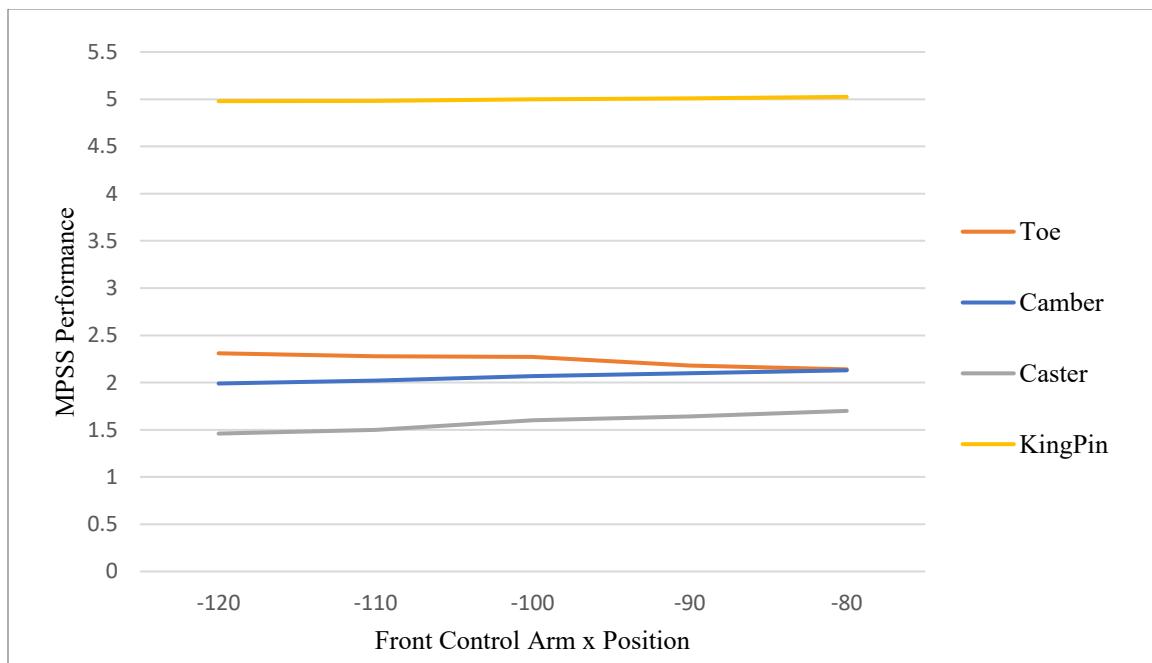


Figure 4.3: Effect of front control arm x position on maximum angle of MPSS

Figure 4.3 shows the effect of x position of front control arm on maximum angle of MPSS. In addition, Table 4.3 illustrates the effects of changes of x position of front control arm on maximum angle of MPSS. As such, 10 % change in front control arm along x direction results in 2.11%, 4.23%, 10 %, 2.21% change in camber, caster, KPI, and toe maximum angles, respectively.

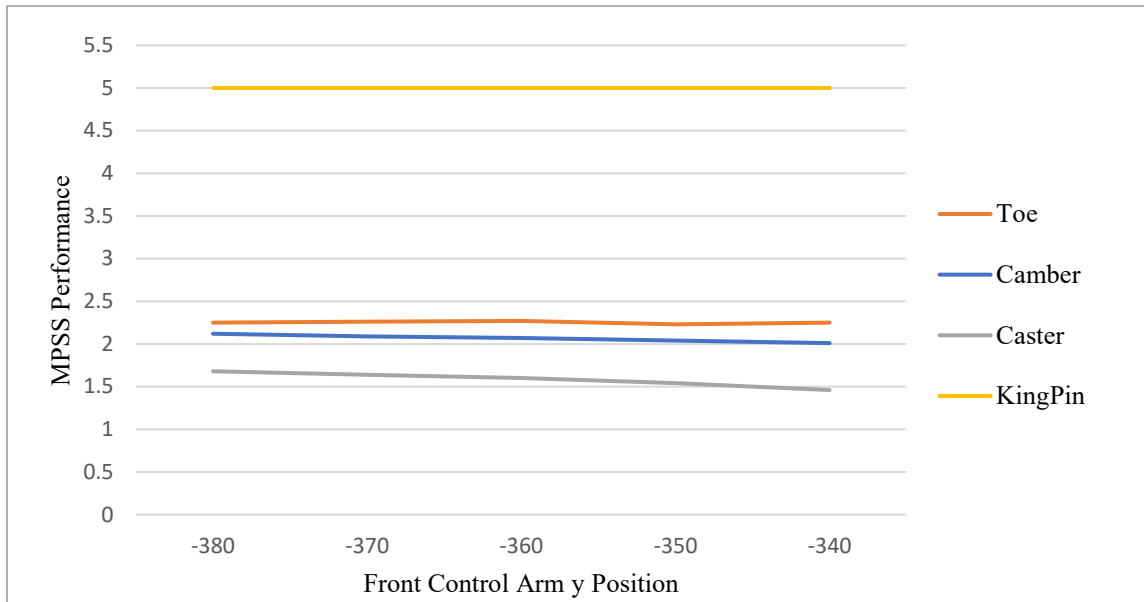


Figure 4.4: Effect of front control arm y position on maximum angle of MPSS

As shown in Figure 4.4 and Table 4.3, 10 % change in front control arm along y direction results in 4.93%, 14.3%, 0%, 10 % change in camber, caster, KPI and toe maximum angles, respectively. Similarly, as shown in Figure 4.5 and Table 4.3, 10 % change in front control arm along z direction results in 12.8%, 10%, 1.04%, and 5.66% change in camber, caster, KPI, and toe maximum angles, respectively.

Thus, front control arm changes along z direction have more effect than front control arm change along x and y directions, for camber, KPI, and toe maximum angles. However, front control arm changes along y direction as more effect than front control arm change along x and z directions for maximum caster angles.

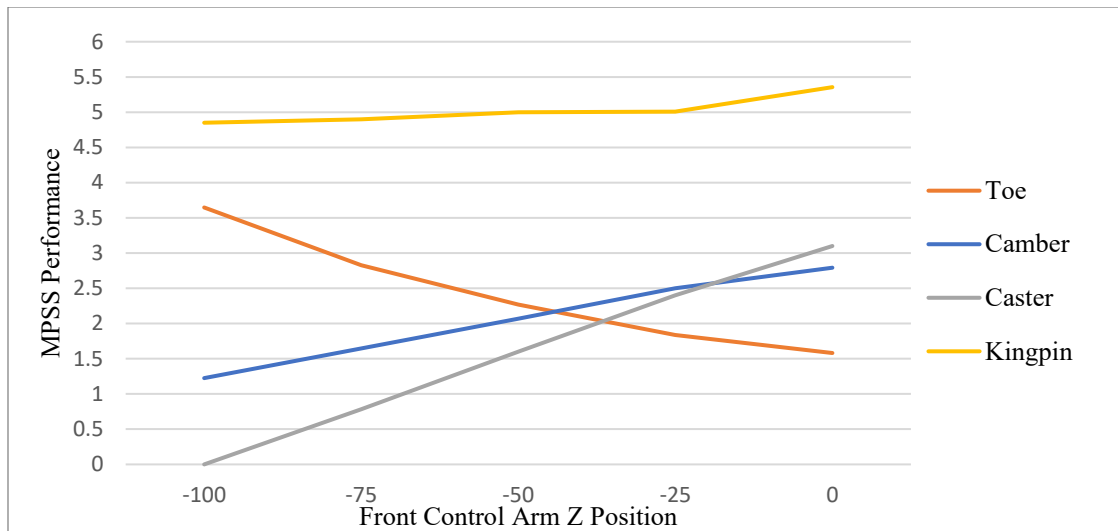


Figure 4.5: Effect of front control arm z position on maximum angle of MPSS

Figure 4.6 shows the effect of x position of front control arm on minimum angle of MPSS. In addition, Table 4.3 illustrates the effects of changes of x position of front control arm on minimum angle of MPSS. As such, 10 % change in front control arm along x direction results in 2.26%, 4.61%, 1.43%, and 3.06% change in camber, caster, KPI, and toe minimum angles, respectively.

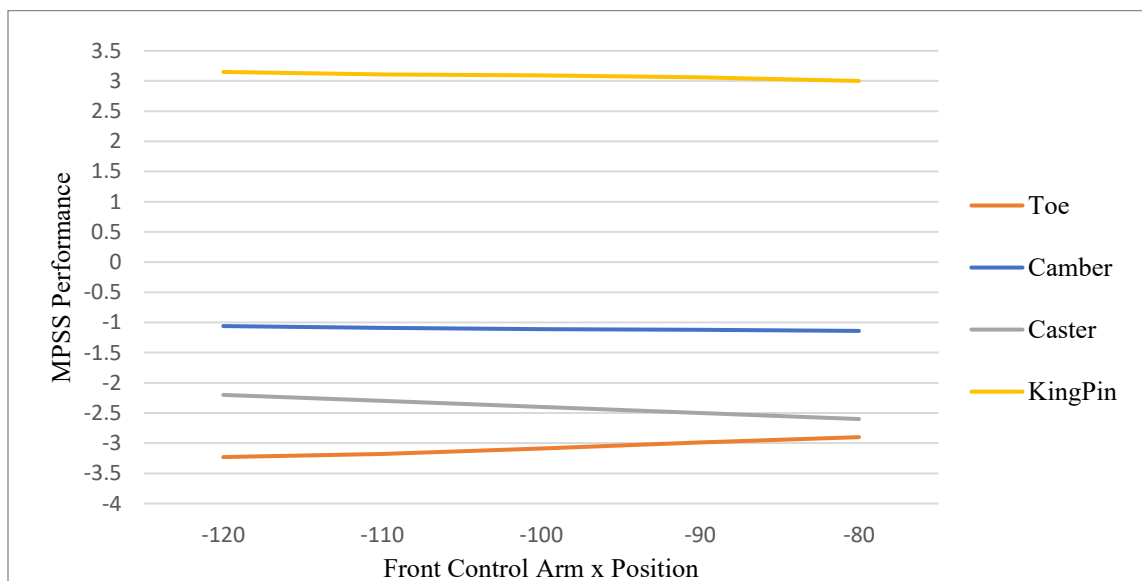


Figure 4.6: Effect of front control arm x position on minimum angle of MPSS

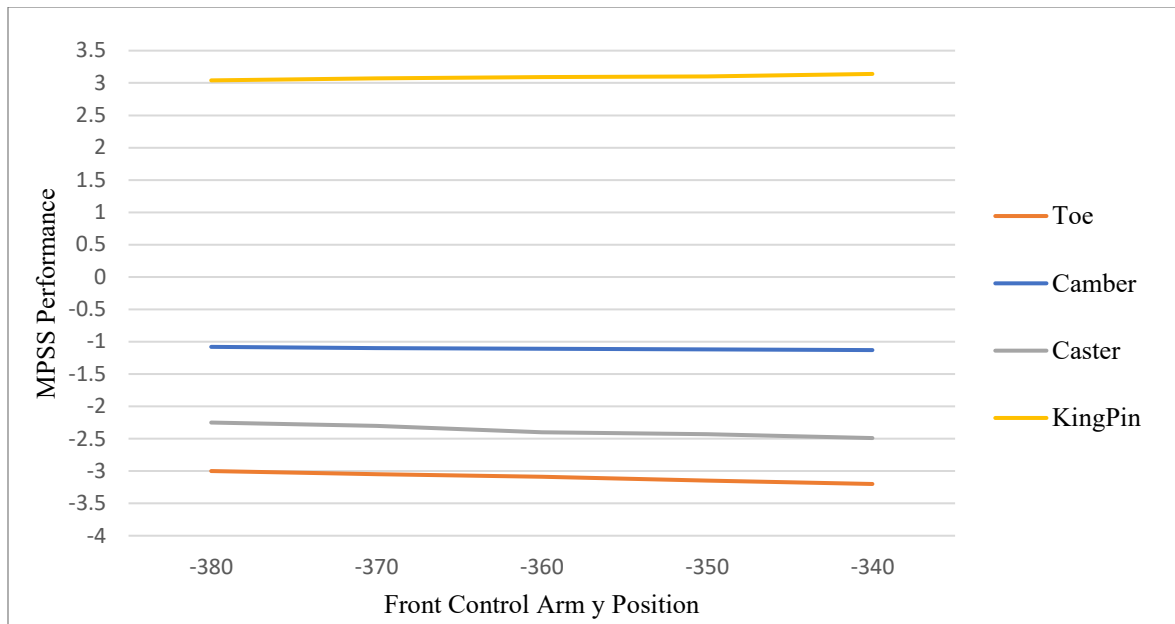


Figure 4.7: Effect of front control arm y position on minimum angle of MPSS

As shown in *Figure 4.7* and *Table 4.3*, 10 % change in front control arm along y direction results in 4.39%, 9.16%, 3.12%, and 6.33% change in camber, caster, KPI, and toe minimum angles, respectively.

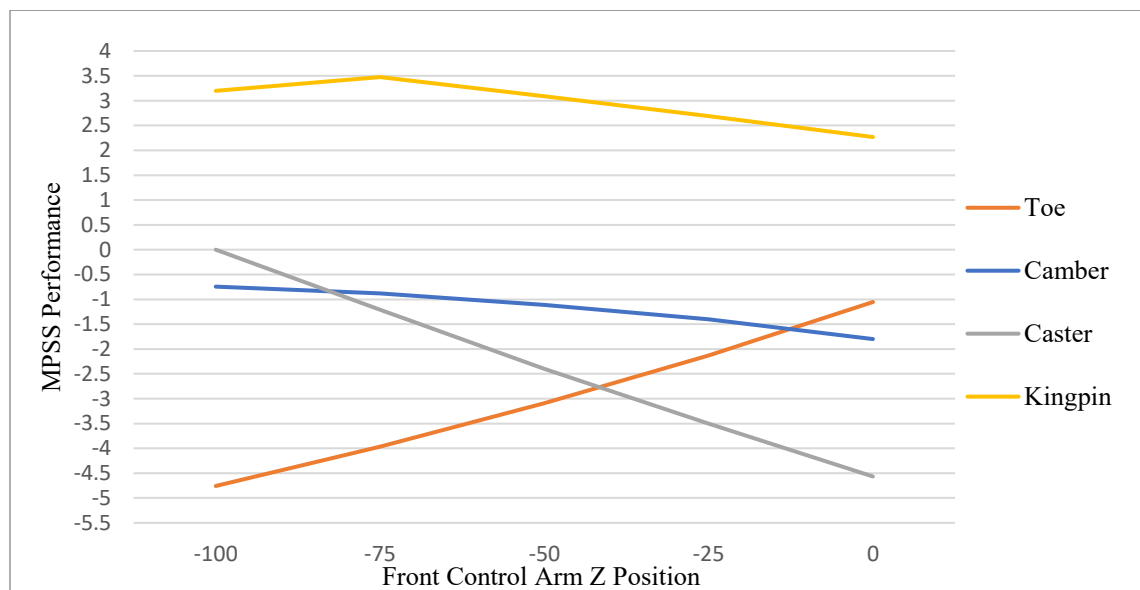


Figure 4.8: Effect of front control arm z position on minimum angle of MPSS

As shown in *Figure 4.8* and *Table 4.3*, 10 % change in front control arm along z direction results in 14.2%, 10%, 2.9%, and 7.78% change in minimum camber, caster, KPI, and toe minimum angles, respectively.

Thus, front control arm changes along z direction as more effect than front control arm change along x and y directions, for camber, caster, and toe minimum angles. Whereas, front control arm change along y direction as more effect than front control arm change along x and z directions for minimum Kingpin angles. Hence, the change in front control arm along z direction has more effect than change in front control arm along x and y direction, for the six of eight performance parameters. Therefore, as shown in *Table 4.4*, change in front control arm along z position is selected as an input parameter for optimization.

Regarding the rear control arm, as shown in *Figure 4.9* and *Table 4.3*, 10 % change in rear control arm along x direction results in 0.74%, 4.51%, 0.24%, and 0.53% change in camber, caster, KPI, and toe maximum angles, respectively.

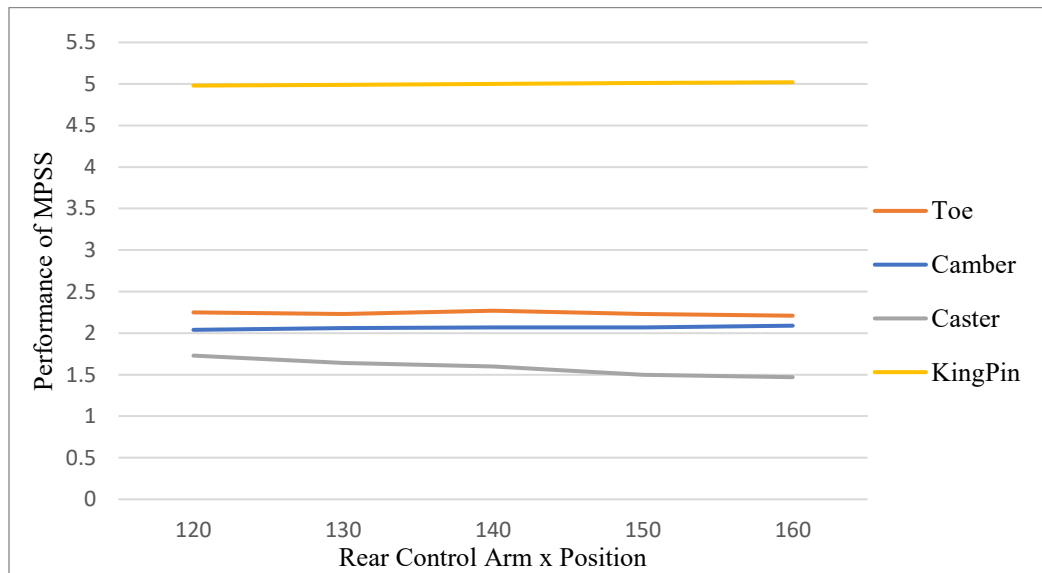


Figure 4.9: Effect of Rear Control Arm x position on Maximum Angle of MPSS

As shown in *Figure 4.10* and *Table 4.3*, 10 % change in rear control arm along y direction results in 3.15%, 13.2%, 0%, and 0% change in camber, caster, KPI, and toe maximum angles, respectively.

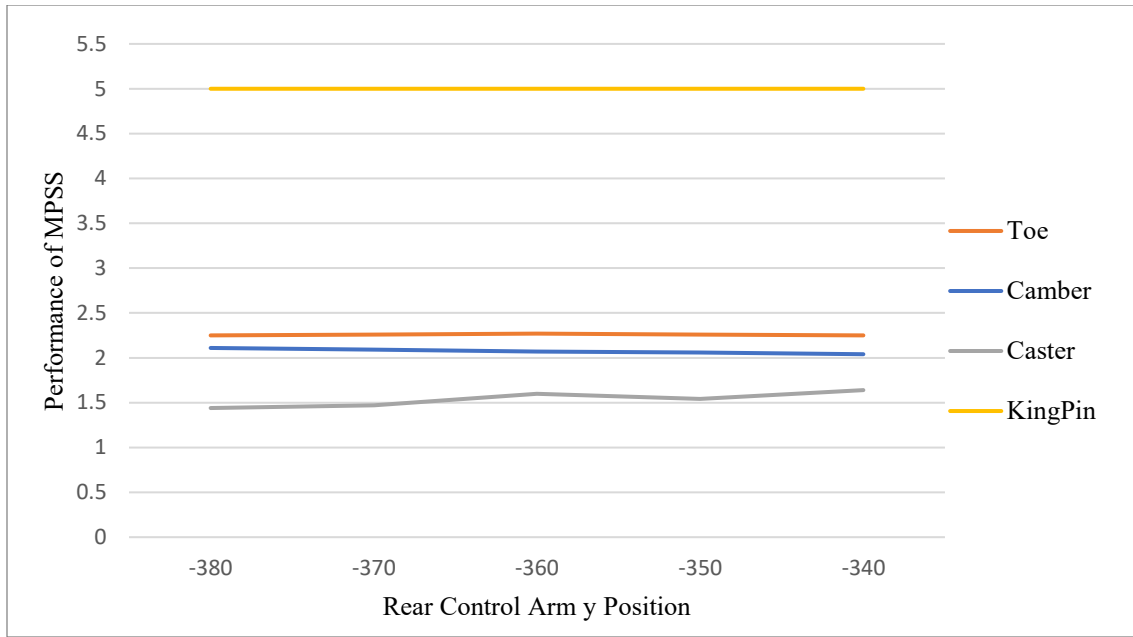


Figure 4.10: Effect of Rear Control Arm y position on Maximum Angle of MPSS

Regarding the rear control arm, as shown in *Figure 4.11* and *Table 4.3*, 10 % change in rear control arm along z direction results in 5.01%, 15%, 1.41%, and 1.62% change in camber, caster, KPI, and toe maximum angles, respectively.

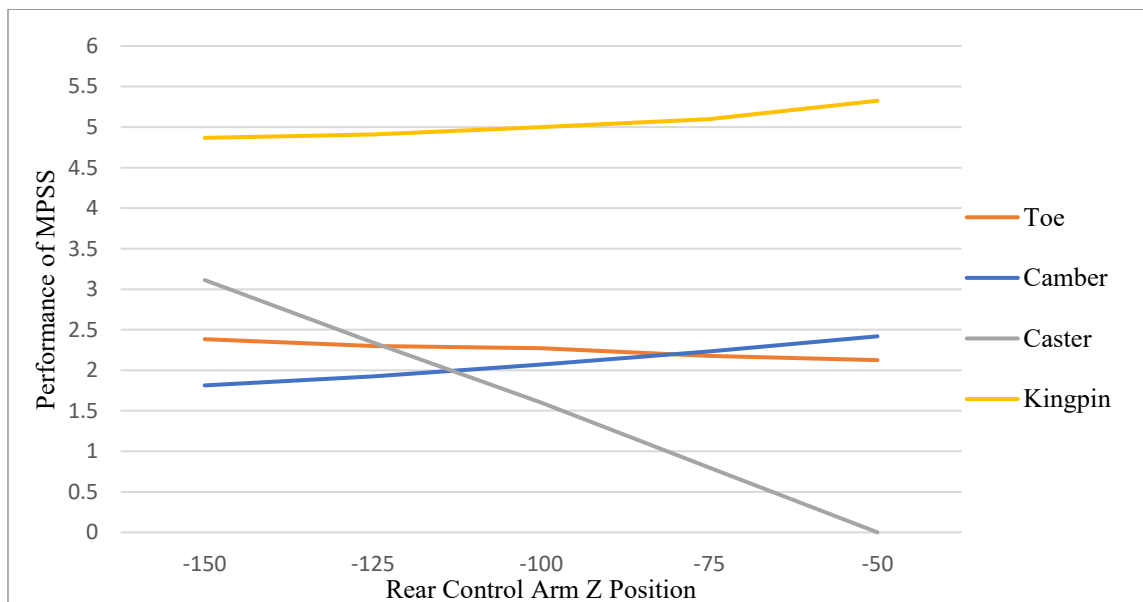


Figure 4.11: Effect of Rear Control Arm z position on Maximum Angle of MPSS

Thus, rear control arm changes along z direction as more effect than rear control arm change along x and y directions, for the camber, caster, KPI, and toe maximum angles.

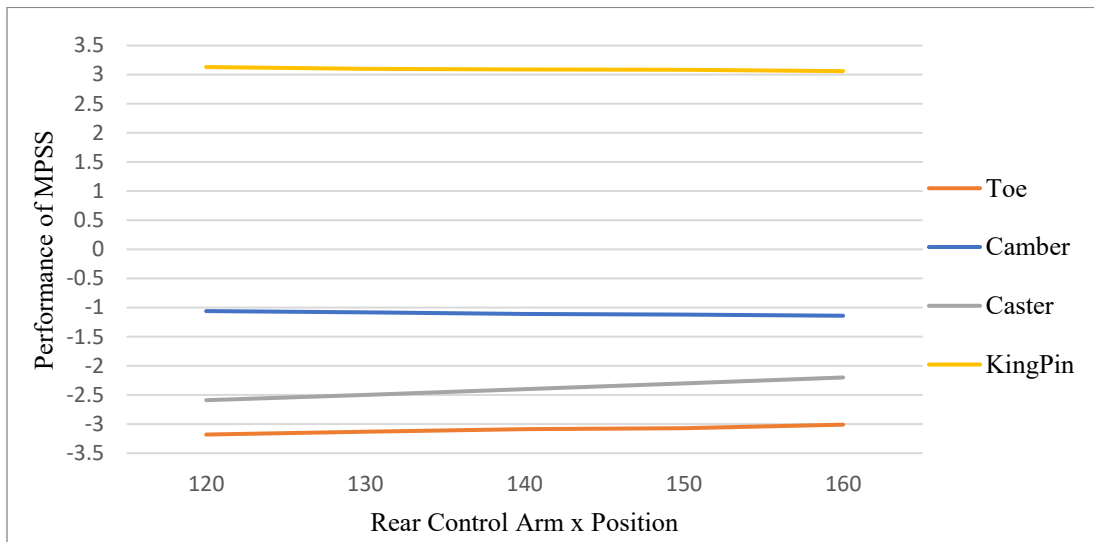


Figure 4.12: Effect of Rear Control Arm x position on Minimum Angle of MPSS

Similarly, as shown in *Figure 4.12* and *Table 4.3*, 10 % change in rear control arm along x direction results in 2.26%, 4.5, 0.67%, and 1.6% change in camber, caster, KPI, and toe minimum angles, respectively.

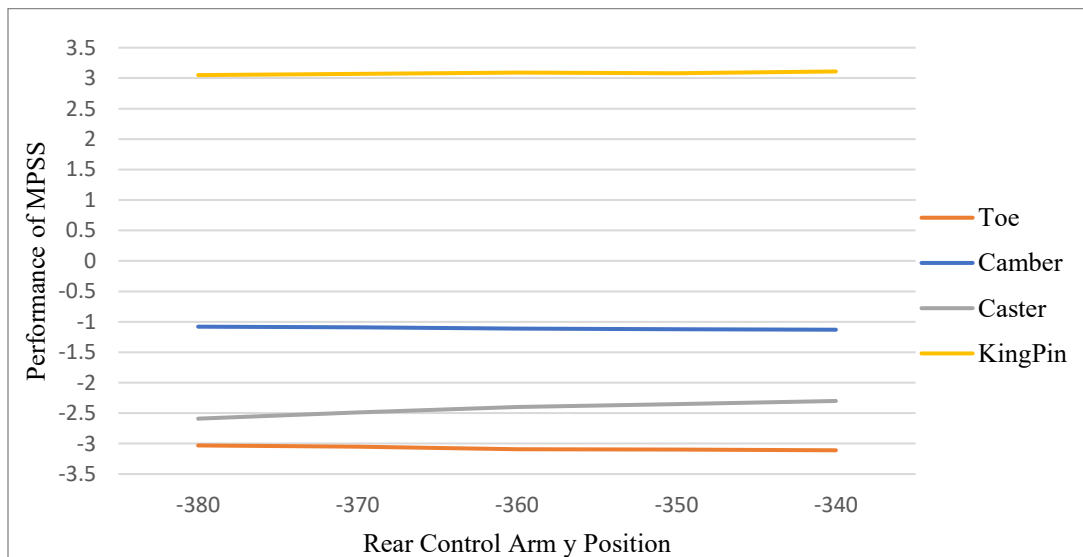


Figure 4.13: Effect of Rear Control Arm y position on Minimum Angle of MPSS

As shown in *Figure 4.13* and *Table 4.3*, 10 % change in rear control arm along x, y, and z direction results in 4.39%, 22.9% change in camber, caster, KPI, and toe minimum angles, respectively.

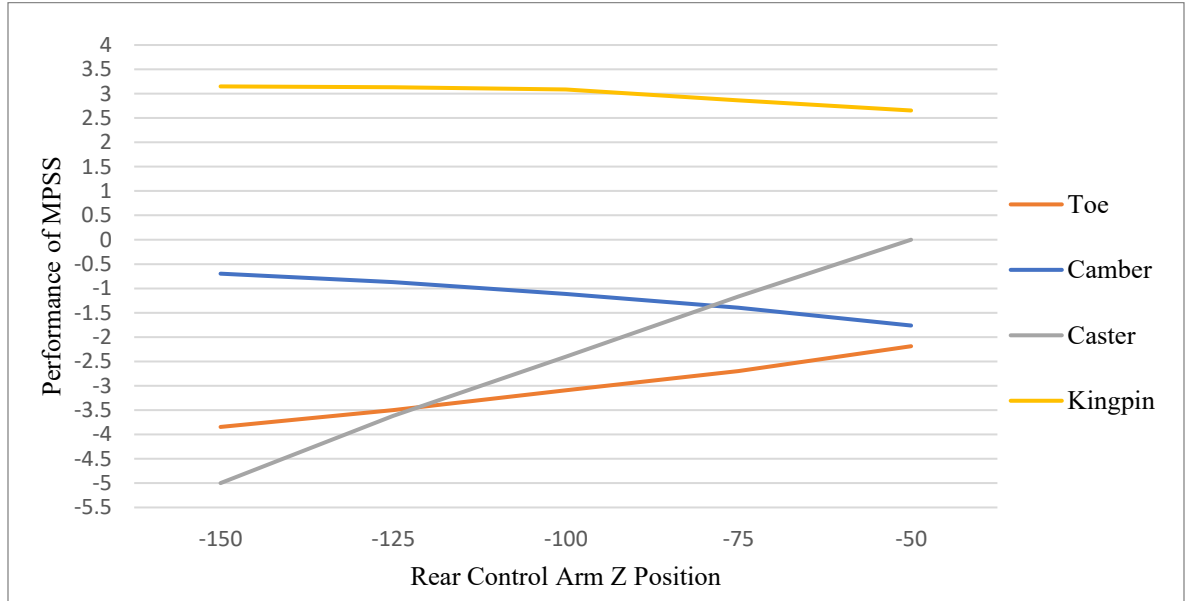


Figure 4.14: Effect of Rear Control Arm z position on Minimum Angle of MPSS

As shown in *Figure 4.14* and *Table 4.3*, 10 % change in rear control arm along z direction results in 22.9%, 15%, 2.36%, and 6.46% change in camber, caster, KPI, and toe minimum angles, respectively. Thus, rear control arm changes along z direction as more effect than rear control arm change along x and y directions, for camber, caster, KPI, and toe minimum angles.

Hence, the change in rear control arm along z direction has more effect than change in rear control arm along x and y direction, for all of the eight performance parameters. Therefore, as shown in *Table 4.4*, change in rear control arm along z position is selected as an input parameter for optimization.

Regarding the outer control arm, as shown in *Figure 4.15* and *Table 4.3*, 10 % change in outer control arm along x direction results in 0.95%, 5%, 0.09%, and 0.69% change in camber, caster, KPI, and toe maximum angles, respectively.

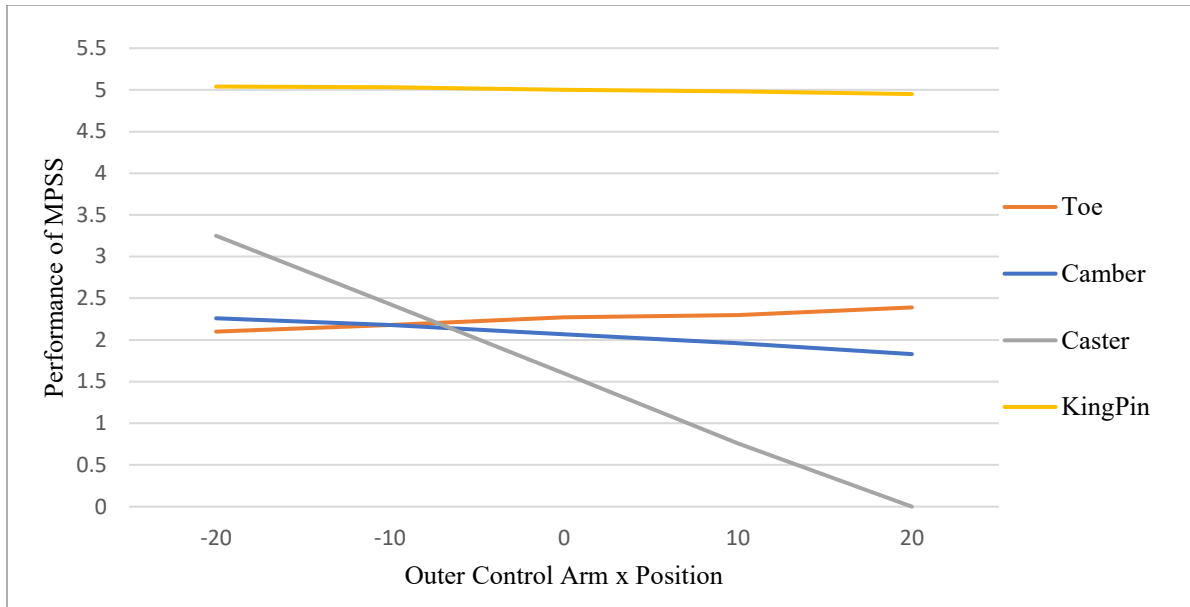


Figure 4.15: Effect of Outer Control Arm x position on Maximum Angle of MPSS

In addition, as shown in *Figure 4.16* and *Table 4.3*, 10 % change in outer control arm along y direction results in 10.9%, 5.62%, 25.4%, and 234% change in camber, caster, KPI, and toe maximum angles, respectively.

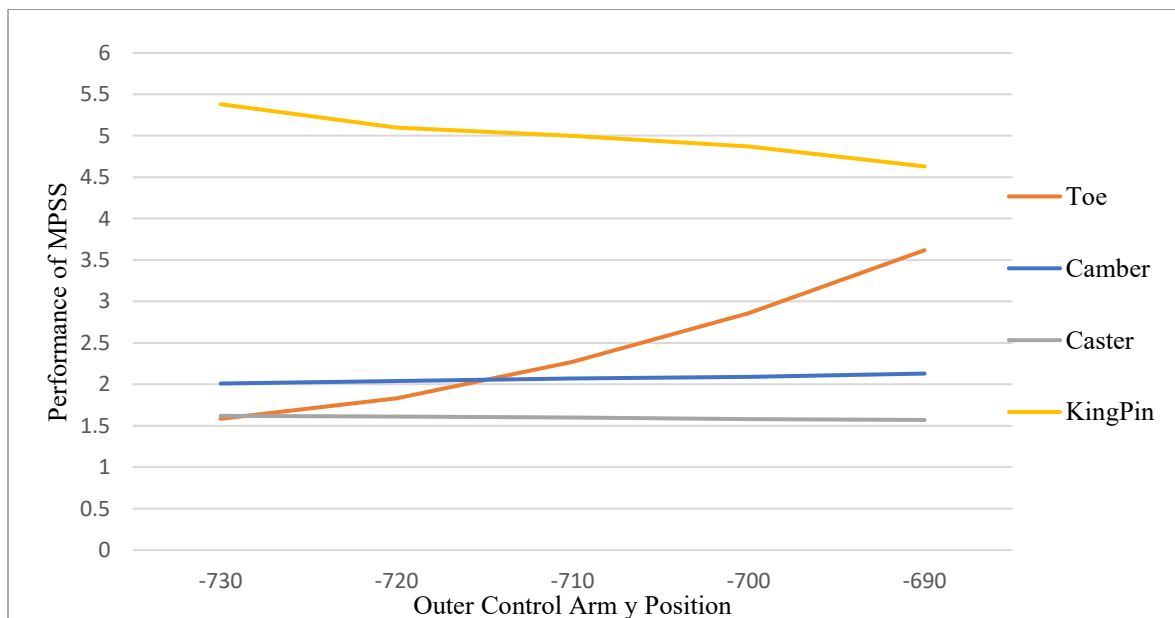


Figure 4.16: Effect of Outer Control Arm y position on Maximum Angle of MPSS

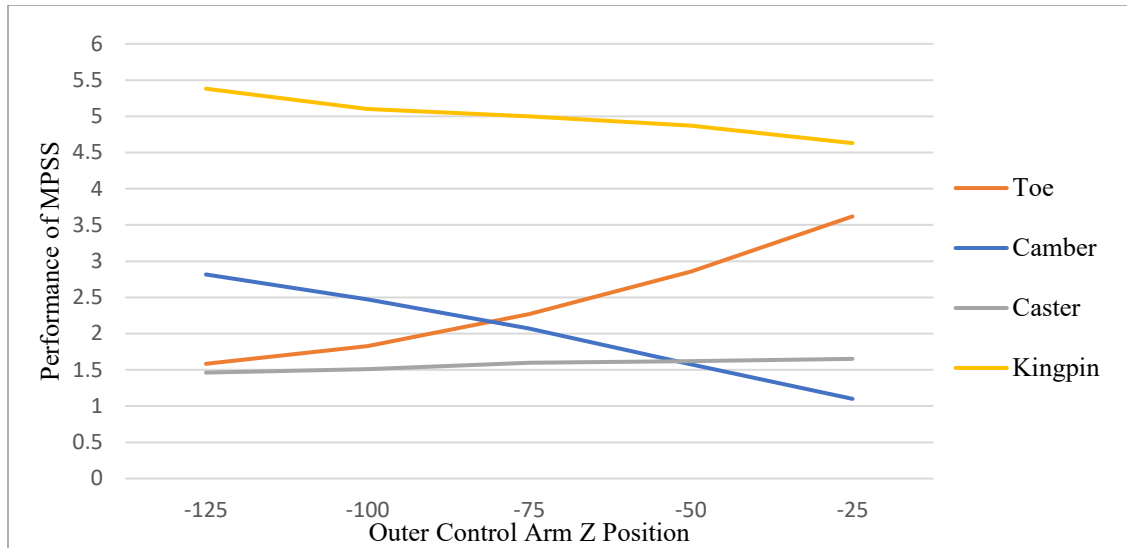


Figure 4.17: Effect of Outer Control Arm z position on Maximum Angle of MPSS

Moreover, as shown in *Figure 4.17* and *Table 4.3*, 10 % change in outer control arm along x, y, and z direction results in 7.62%, 1.62%, 1.74%, and 16.1% change in camber, caster, KPI, and toe maximum angles, respectively. Thus, outer control arm changes along y direction as more effect than outer control arm change along x and z directions for camber, caster, KPI, and toe maximum angles.

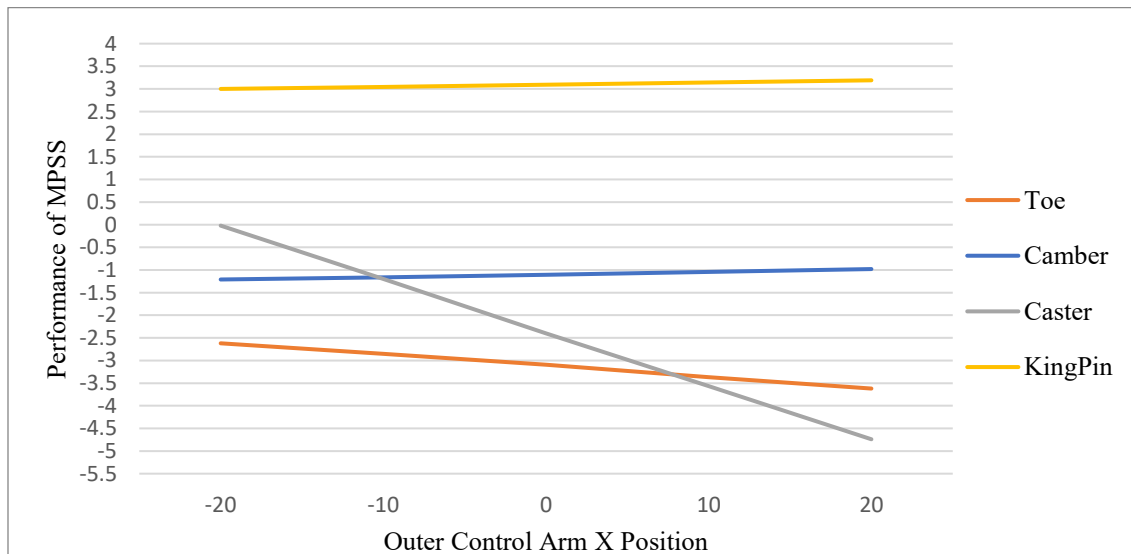


Figure 4.18: Effect of Outer Control Arm x position on Minimum Angle of MPSS

Similarly, as shown in *Figure 4.18* and *Table 4.3*, 10 % change in outer control arm along x, direction results in 0.95%, 1180%, 0.32%, and 1.9% change in camber, caster, KPI, and toe minimum angles, respectively.

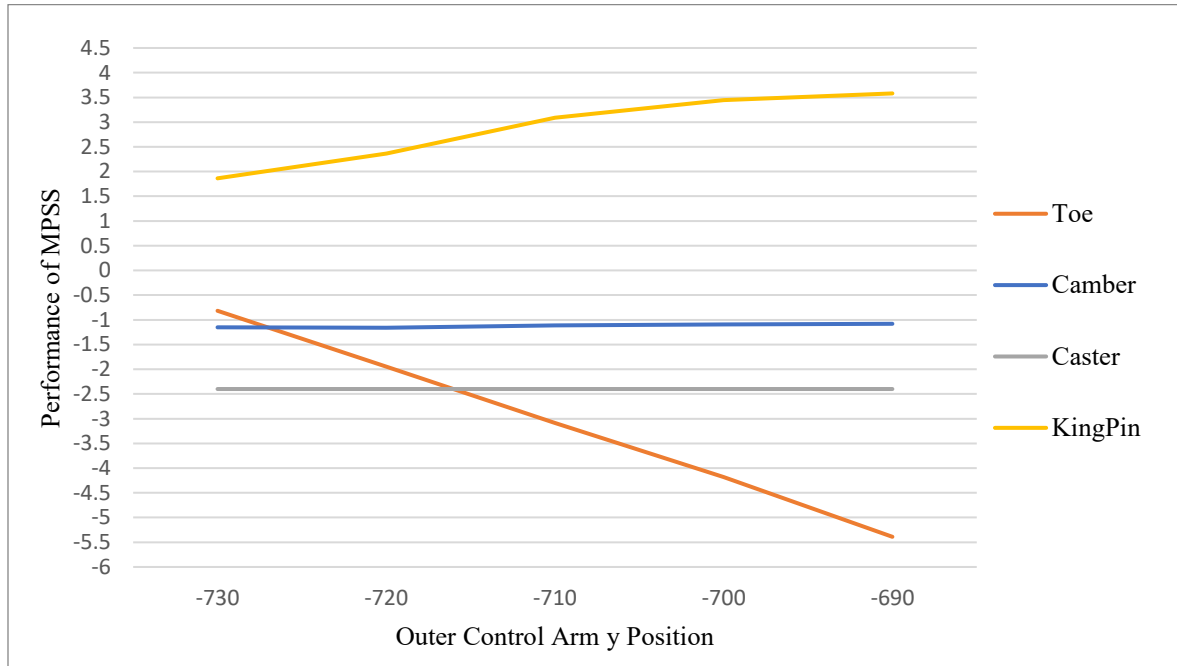


Figure 4.19: Effect of Outer Control Arm y position on Minimum Angle of MPSS

In addition, as shown in *Figure 4.19* and *Table 4.3*, 10 % change in outer control arm along y direction results in 11.1%, 0%, 168%, 1021% and change in camber, caster, KPI, and toe minimum angles, respectively.

Moreover, as shown in *Figure 4.20* and *Table 4.3*, 10 % change in outer control arm along z direction results in 9.72%, 5.29%, 11.5%, and 69.9% change in camber, caster, KPI, and toe minimum angles, respectively. Thus, outer control arm changes along y direction as more effect than outer control arm change along x and z directions for camber, caster, KPI, and toe minimum angles.

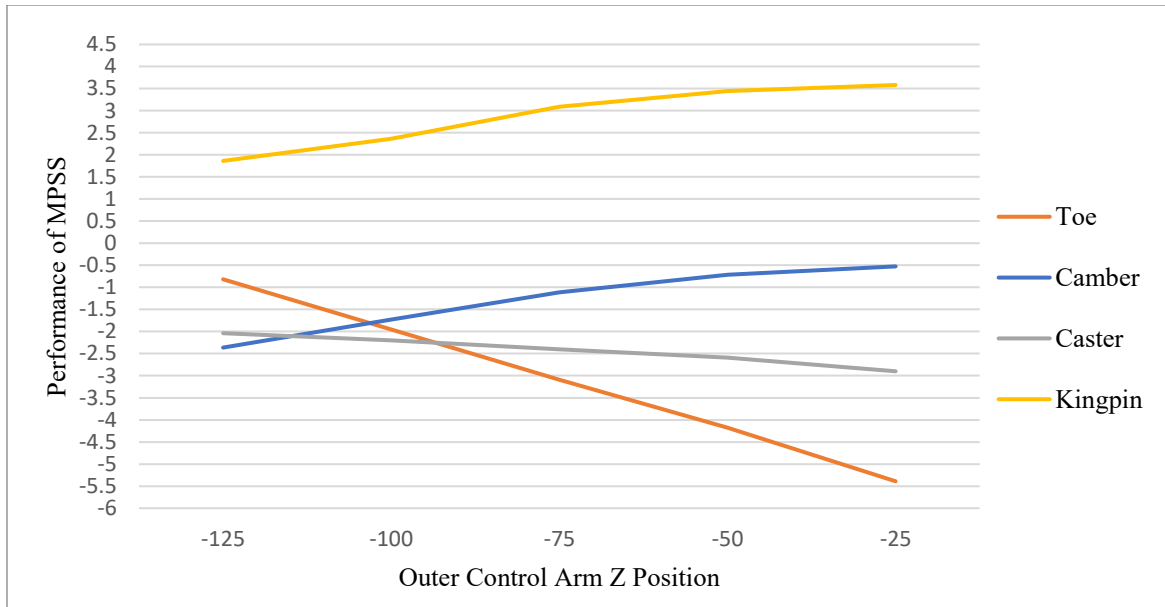


Figure 4.20: Effect of Outer Control Arm z position on Minimum Angle of MPSS

However, the relationship of outer control arm with performance parameters are complex. For instance, the 10% change in the outer control arm along y direction results in 1021% change in maximum toe angle changes. Whereas, the 10% change in the outer control arm along x direction results in 1180% change in minimum caster angle changes. And, these may result in out-of-range values. Moreover, for such difference in parameter changes, optimization is difficult. Hence, even though the change in outer control arm along z direction has average effect than change in outer control arm along x and y direction, as shown in *Table 4.3*, change in outer control arm along z position is selected as an input parameter for optimization.

Regarding the inner tierod, as shown in *Figure 4.21* and *Table 4.3*, 10 % change in inner tierod along x direction results in 0.29%, 0%, 0%, and 1.67% change in camber, caster, KPI, and toe maximum angles, respectively.

In addition, as shown in *Figure 4.22* and *Table 4.3*, 10 % change in inner tierod along y direction results in 1.81%, 0%, 0%, and 6.54% change in camber, caster, KPI, and toe maximum angles, respectively.

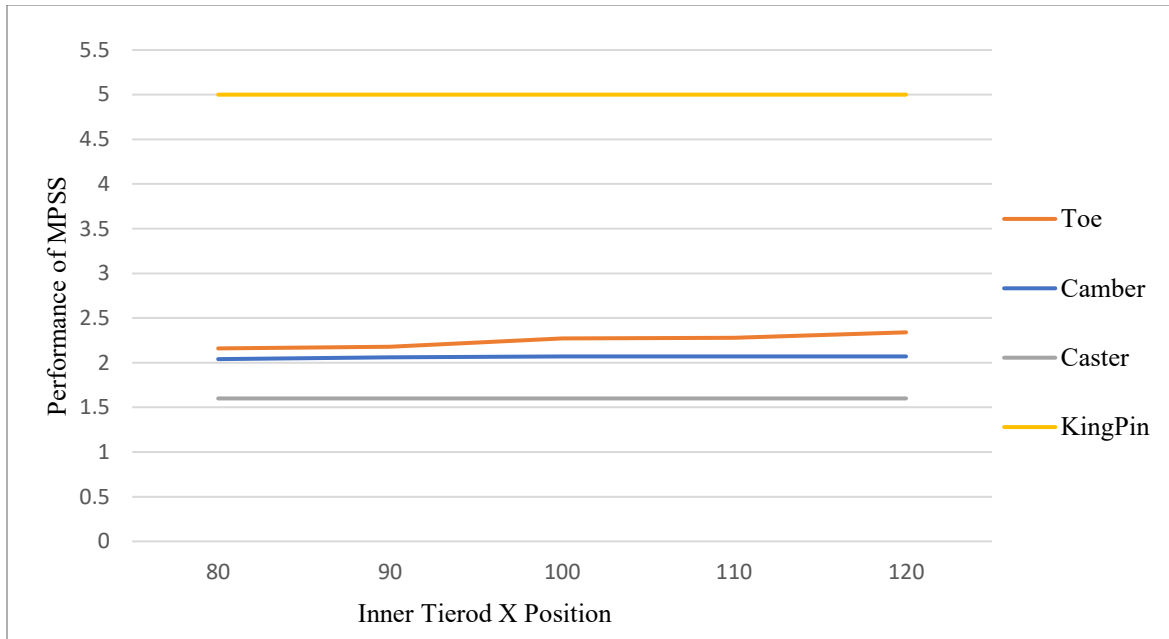


Figure 4.21: Effect of Inner Tierod x position on Maximum Angle of MPSS

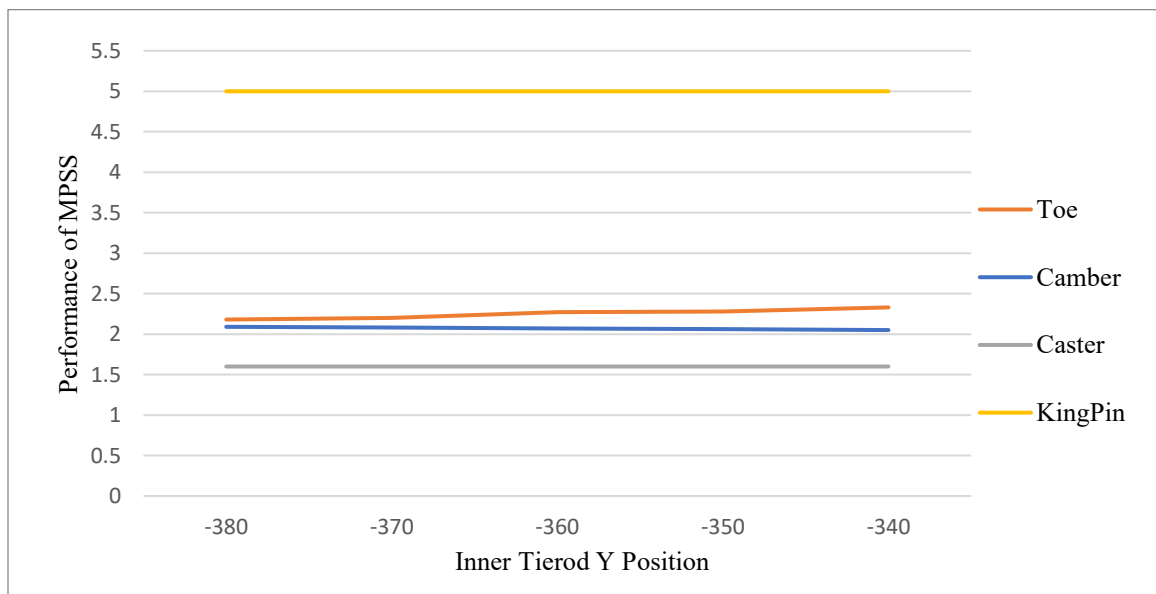


Figure 4.22: Effect of Inner Tierod y position on Maximum Angle of MPSS

Similarly, as shown in *Figure 4.23* and *Table 4.3*, 10 % change in inner tierod along z direction results in 4.24%, 0%, 0%, and 81.7% change in camber, caster, KPI, and toe maximum angles, respectively.

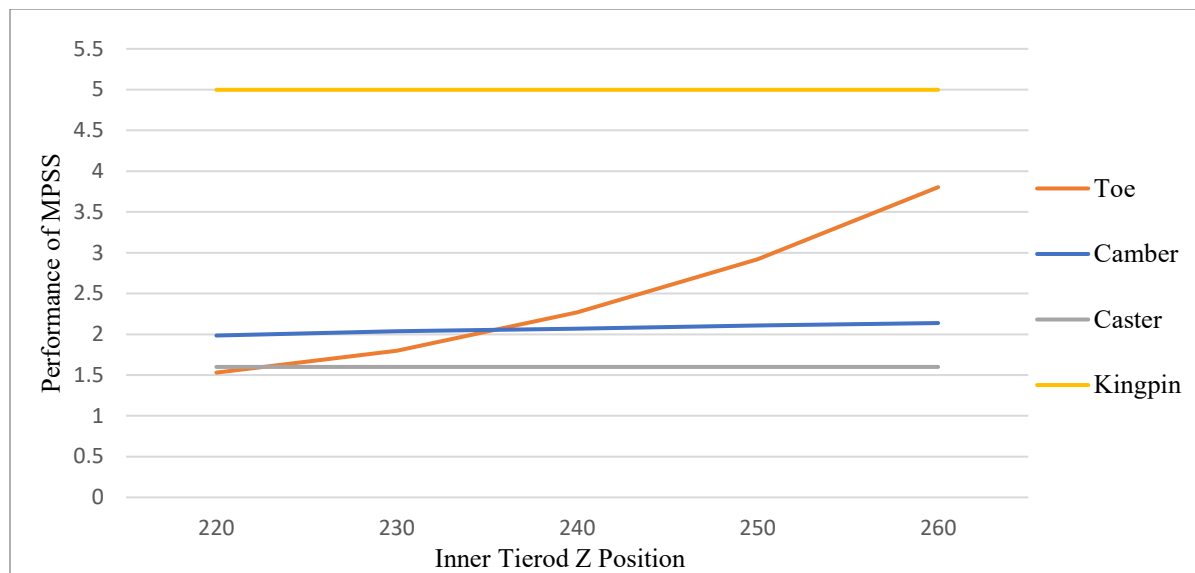


Figure 4.23: Effect of Inner Tierod z position on Maximum Angle of MPSS

Thus, inner tierod change along z direction as more effect than inner tierod change along x and y directions, for both the camber and toe maximum angles. However, inner tierod change has no effect on caster and KPI maximum angles.

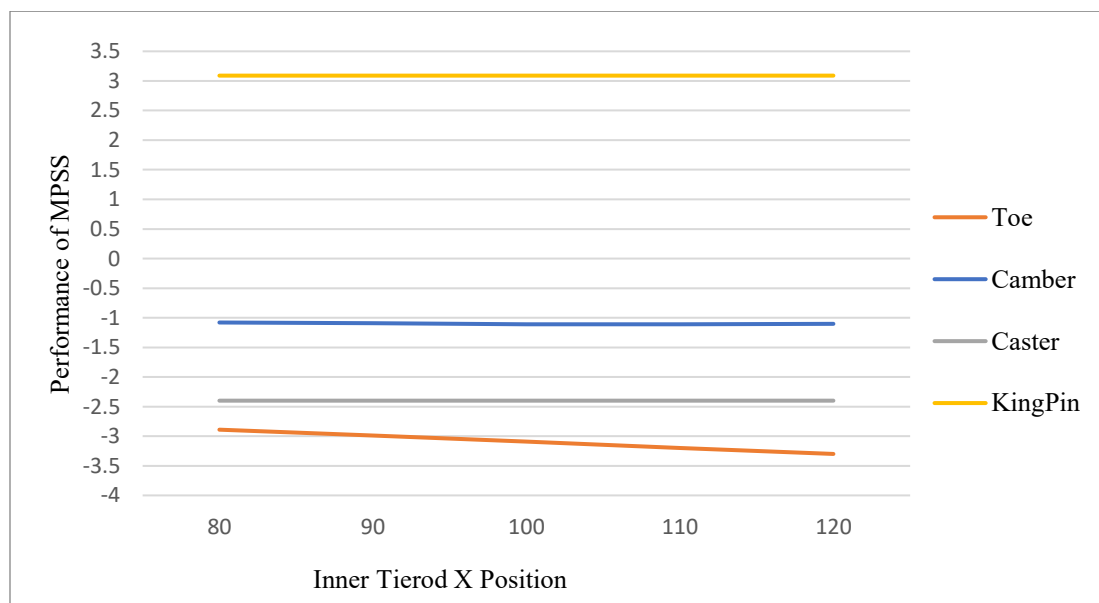


Figure 4.24: Effect of Inner Tierod x position on Minimum Angle of MPSS

Whereas, as shown in Figure 4.24 and Table 4.3, 10 % change in inner tierod along x direction results in 0.37%, 0%, 0%, and 2.84% change in camber, caster, KPI, and toe minimum angles, respectively.

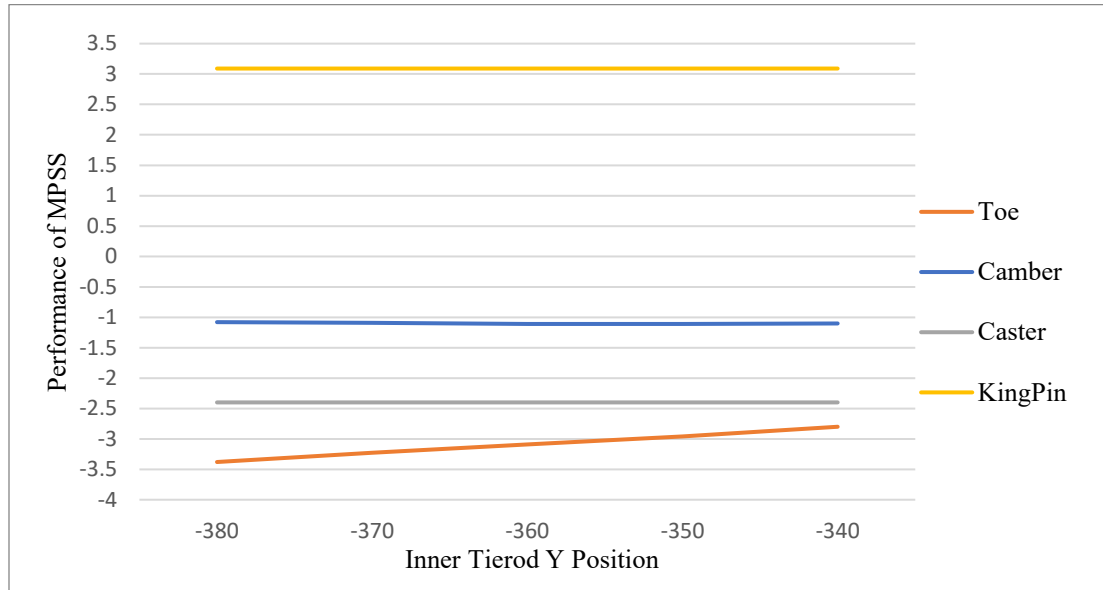


Figure 4.25: Effect of Inner Tierod y position on Minimum Angle of MPSS

Similarly, as shown in Figure 4.25 and Table 4.3, 10 % change in inner tierod along y direction results in 1.76%, 0%, 0%, and 16.3% change in camber, caster, KPI, and toe minimum angles, respectively.

In addition, as shown in Figure 4.26 and Table 4.3, 10 % change in inner tierod along z direction results in 5.88%, 0%, 0%, and 202% change in camber, caster, KPI, and toe minimum angles, respectively. Thus, inner tierod change along z direction as more effect than inner tierod change along x and y directions, for both the camber and toe minimum angles. However, inner tierod change has no effect on caster and KPI minimum angles.

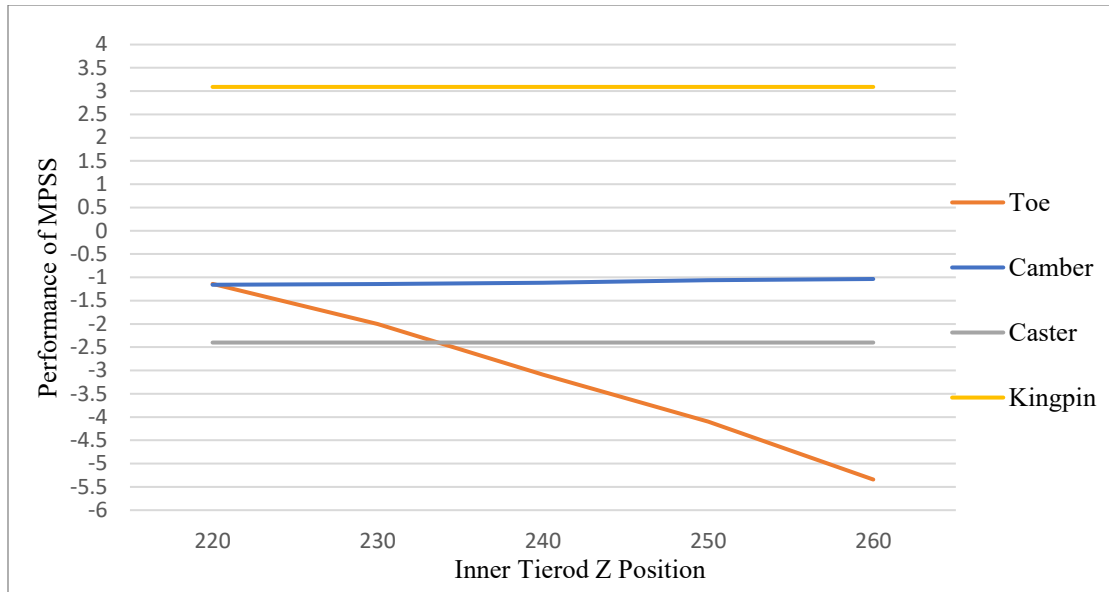


Figure 4.26: Effect of Inner Tierod z position on Minimum Angle of MPSS

Hence, the change in inner tierod has no effect, for the four of eight performance parameters. Moreover, the change in inner tierod has little effect on maximum and minimum camber angles. Therefore, as shown in *Table 4.4*, change in inner tierod is not selected as an input parameter for optimization.

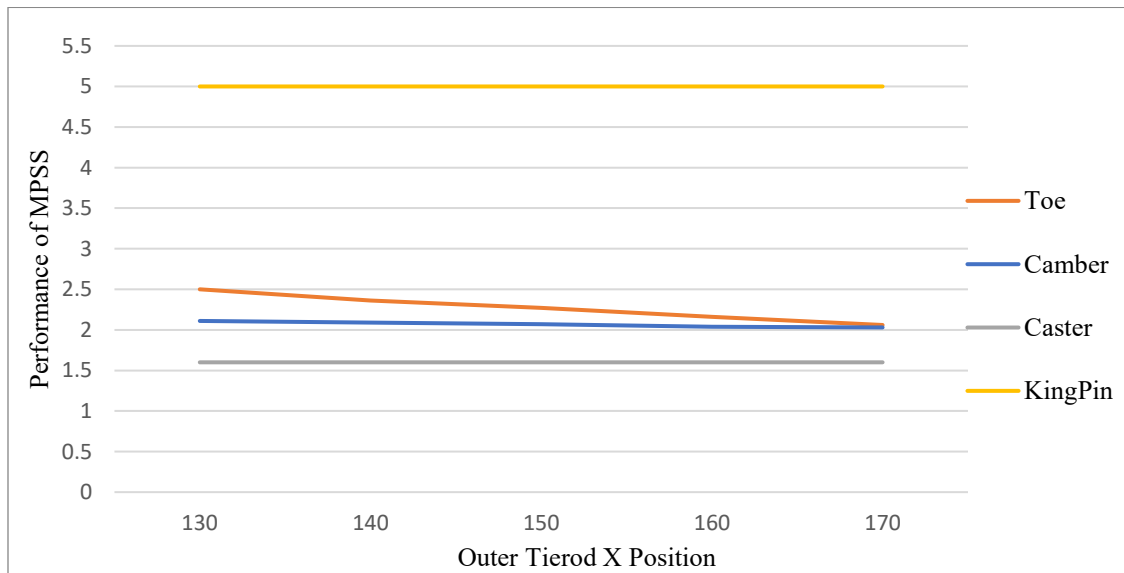


Figure 4.27: Effect of Outer Tierod x position on Maximum Angle of MPSS

Regarding the outer tierod, as shown in *Figure 4.27* and *Table 4.3*, 10 % change in outer tierod along x direction results in 1.23%, 0%, 0%, and 5.72 change in camber, caster, KPI, and toe maximum angles, respectively.

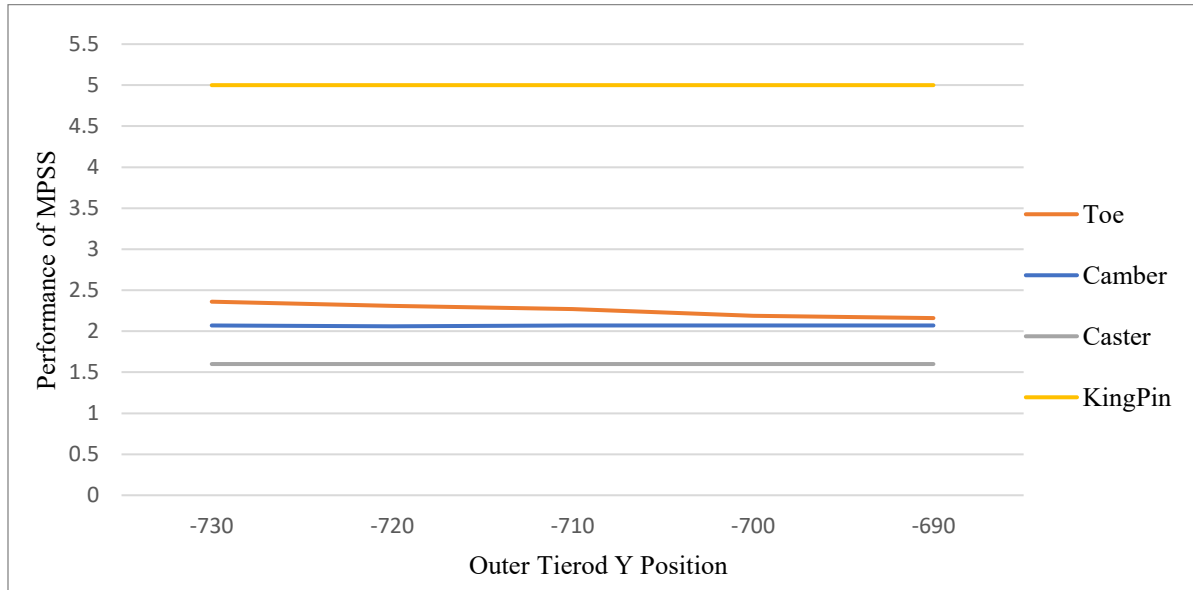


Figure 4.28: Effect of Outer Tierod y position on Maximum Angle of MPSS

As shown in *Figure 4.28* and *Table 4.3*, 10 % change in outer tierod along y direction results in 0%, 0%, 0%, and 15.4% change in camber, caster, KPI, and toe maximum angles, respectively.

In addition, as shown in *Figure 4.29* and *Table 4.3*, 10 % change in outer tierod along z direction results in 3.34%, 0%, 0%, and 31.6% change in camber, caster, KPI, and toe maximum angles, respectively.

Thus, outer tierod change along z direction as more effect than outer tierod change along x and y directions, for camber and toe maximum angles. However, outer tierod change has no effect on caster and KPI maximum angles.

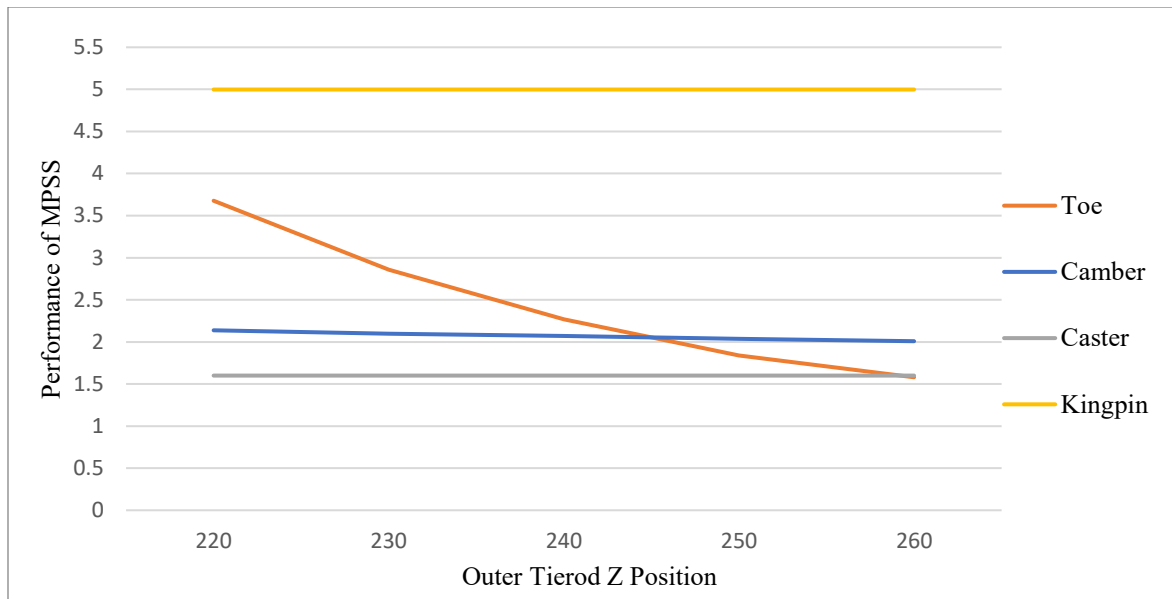


Figure 4.29: Effect of Outer Tierod z position on Maximum Angle of MPSS

Whereas, as shown in *Figure 4.30* and *Table 4.3*, 10 % change in outer tierod along x direction results in 2.43%, 0%, 0%, and 12.8% change in camber, caster, KPI, and toe minimum angles, respectively.

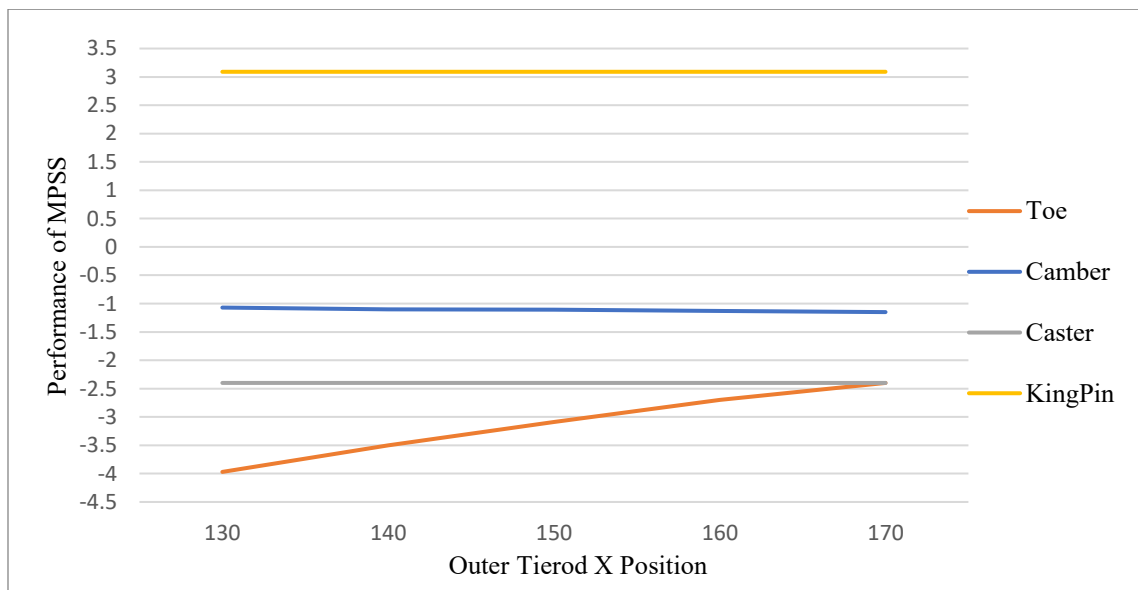


Figure 4.30: Effect of Outer Tierod x position on Minimum Angle of MPSS

Similarly, as shown in *Figure 4.31* and *Table 4.3*, 10 % change in outer tierod along y direction results in 13.6%, 0%, 0%, and 15.9% change in camber, caster, KPI, and toe minimum angles, respectively.

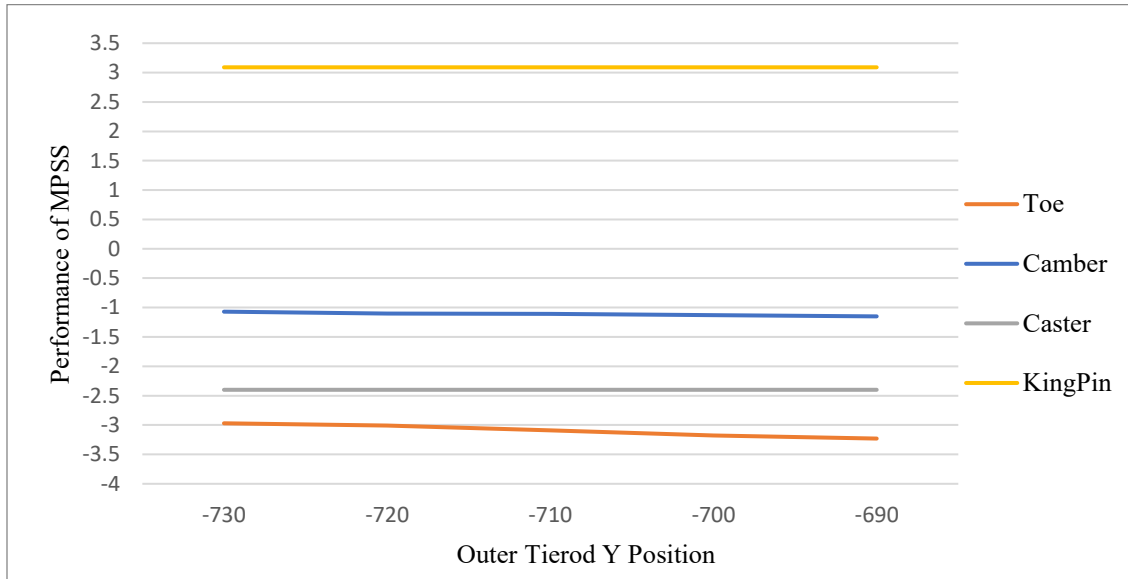


Figure 4.31: Effect of Outer Tierod y position on Minimum Angle of MPSS

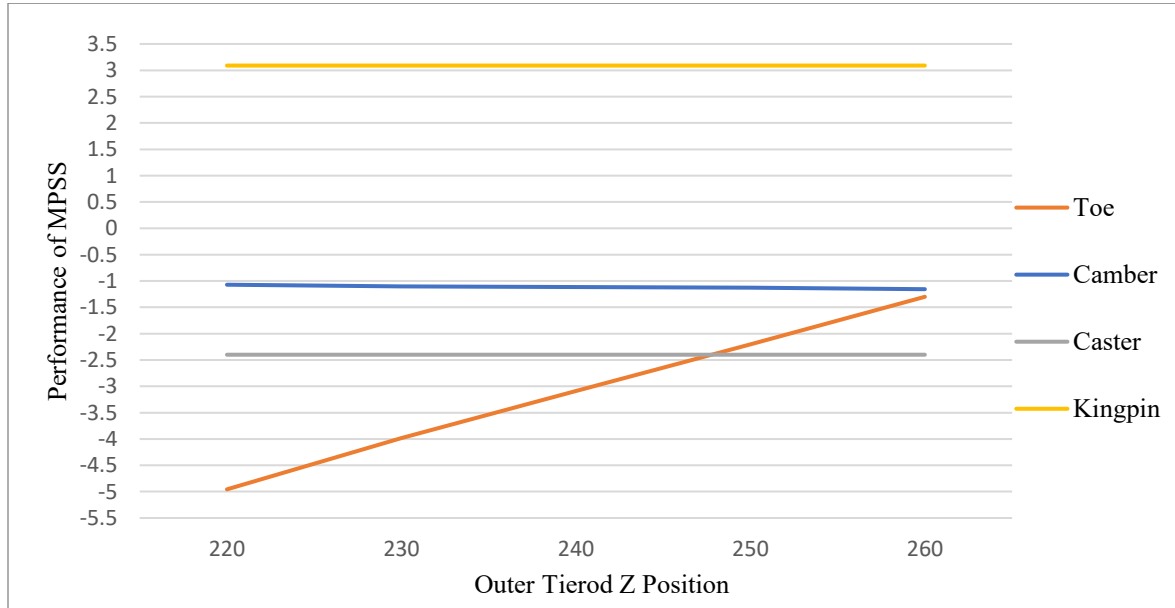


Figure 4.32: Effect of Outer Tierod z position on Minimum Angle of MPSS

Moreover, as shown in *Figure 4.32* and *Table 4.3*, 10 % change in outer tierod along z direction results in 4.36%, 0%, 0%, and 40.5% change in camber, caster, KPI, and toe minimum angles, respectively.

Thus, outer tierod change along z direction as more effect than outer tierod change along x and y directions, for camber and toe minimum angles. However, outer tierod change has no effect on caster and KPI minimum angles.

Hence, the change in outer tierod has no effect, for the four of eight performance parameters. Therefore, as shown in *Table 4.4*, change in outer tierod is not selected as an input parameter for optimization.

Regarding the lower strut, as shown in *Figure 4.33* and *Table 4.3*, 10 % change in lower strut along x direction results in 0.61%, 653%, 0%, and 0.3%, change in camber, caster, KPI, and toe maximum angles, respectively.

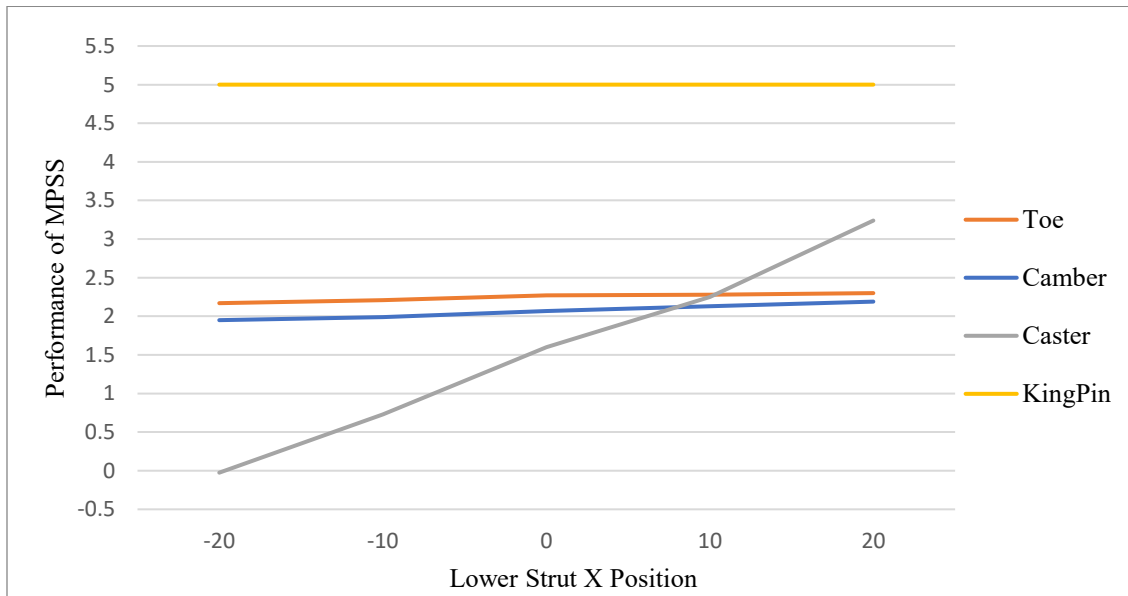


Figure 4.33: Effect of Strut Lower x position on Maximum Angle of MPSS

In addition, as shown in *Figure 4.34* and *Table 4.3*, 10 % change in lower strut along y direction results in 79.6%, 2.31%, 0%, and 88.4% change in camber, caster, KPI, and toe maximum angles, respectively.

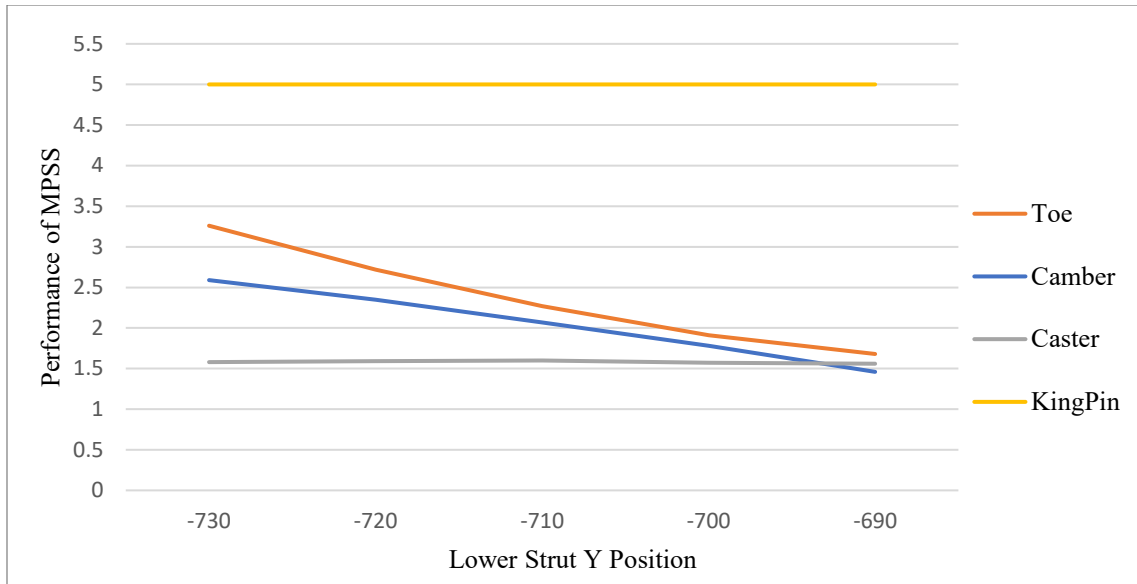


Figure 4.34: Effect of Strut Lower y position on Maximum Angle of MPSS

Moreover, as shown in *Figure 4.35* and *Table 4.3*, 10 % change in lower strut along x, y, and z direction results in 5.28%, 0%, 0%, and 7.23% change in camber, caster, KPI, and toe maximum angles, respectively.

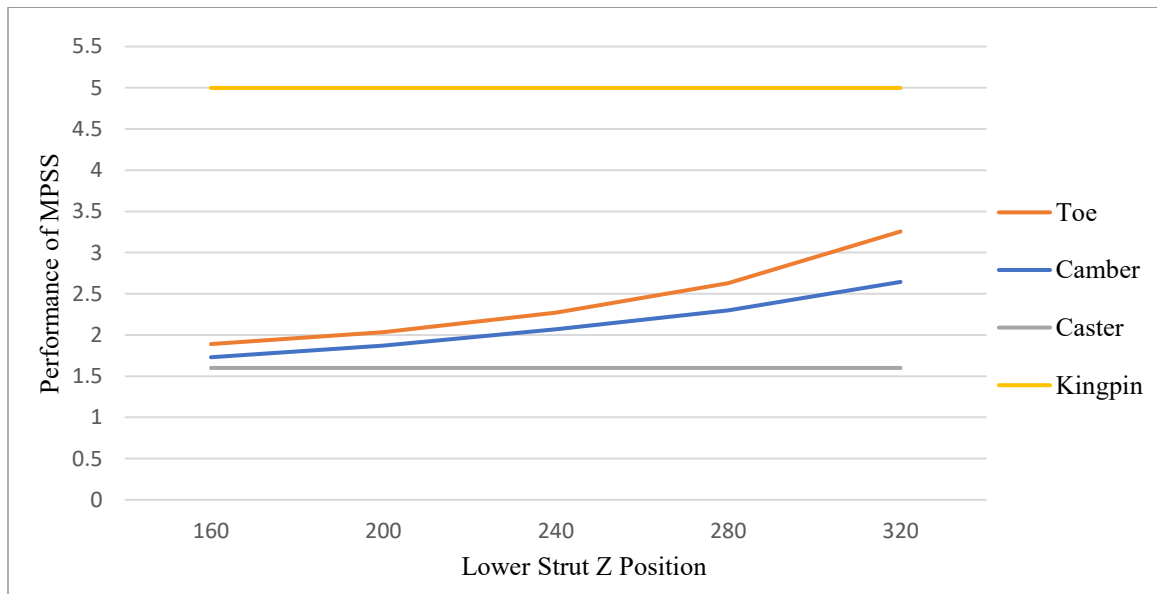


Figure 4.35: Effect of Strut Lower z position on Maximum Angle of MPSS

Thus, lower strut changes along y direction have more effect than lower strut change along x and z directions, for camber and toe maximum angles. Whereas, lower strut changes along x direction have more effect than lower strut change along y and z directions, for maximum caster angles. However, lower strut change has no effect on maximum Kingpin angles.

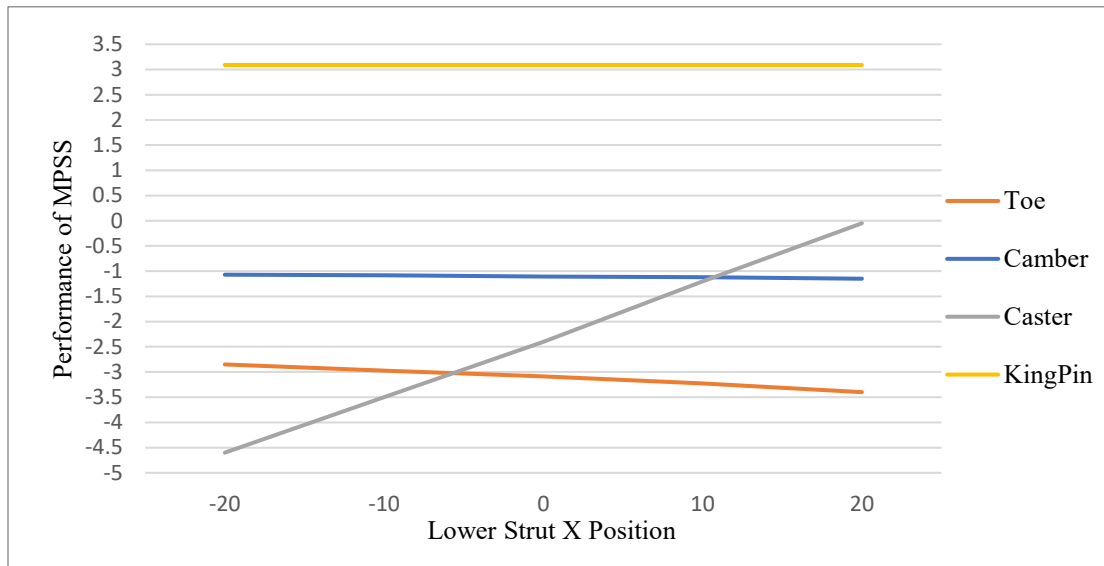


Figure 4.36: Effect of Strut Lower x position on Minimum Angle of MPSS

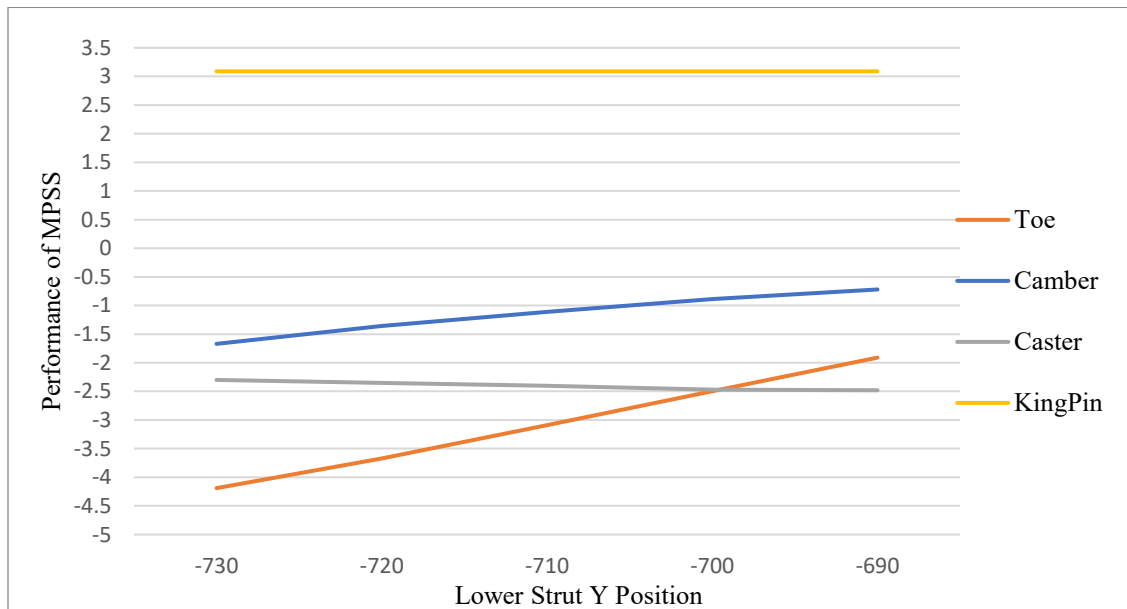


Figure 4.37: Effect of Strut Lower y position on Minimum Angle of MPSS

Similarly, as shown in *Figure 4.36* and *Table 4.3*, 10 % change in lower strut along x direction results in 0.37%, 4.94%, 0%, and 0.96% change in camber, caster, KPI, and toe minimum angles, respectively.

In addition, as shown in *Figure 4.37* and *Table 4.3*, 10 % change in lower strut along y direction results in 103%, 14.3%, 0%, and 99.3% change in camber, caster, KPI, and toe minimum angles, respectively.

Moreover, as shown in *Figure 4.38* and *Table 4.3*, 10 % change in lower strut along z direction results in 9.79%, 0%, 0%, and 7.19% change in camber, caster, KPI, and toe minimum angles, respectively.

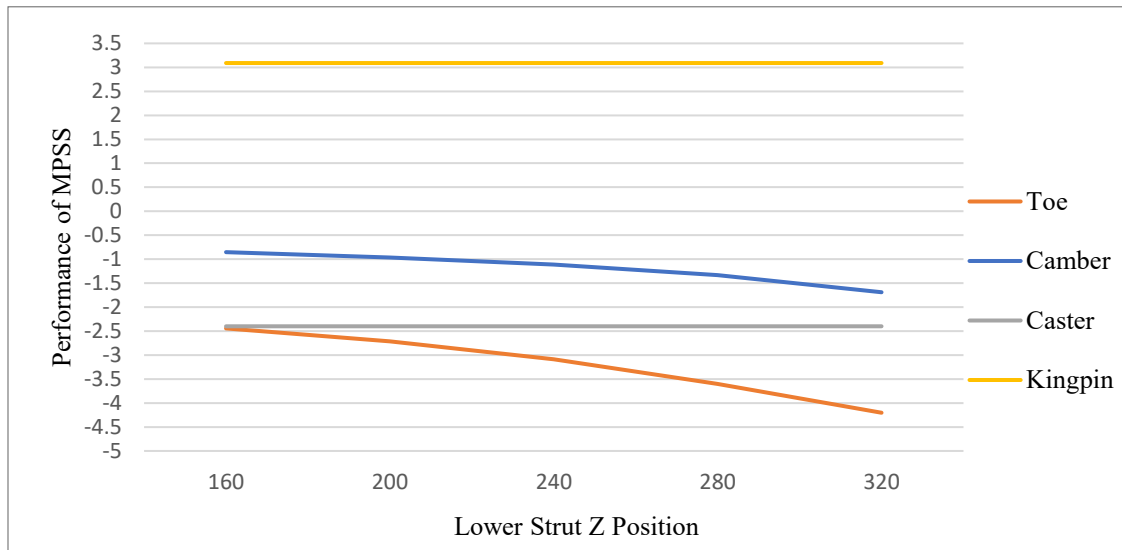


Figure 4.38: Effect of Strut Lower z position on Minimum Angle of MPSS

Thus, lower strut changes along y direction have more effect than lower strut change along x and z directions, for camber, caster and toe minimum angles. However, lower strut change has no effect on maximum Kingpin angles.

Hence, the change in lower strut has no effect, for the two of eight performance parameters. Moreover, the 10% change in the lower strut along x direction results in 653% change in maximum caster angle. And, these may result in out-of-range values. Therefore, as shown in *Table 4.4*, change in lower strut is not selected as an input parameter for optimization.

Table 4.3: Summary of the effect of change in input parameters on the performance of MPSS

Output Parameters	10 % Change in Front Control Arm			10 % Change in Rear Control Arm			10 % Change in Outer Control Arm			10 % Change in Tierod Inner			10 % Change in Tierod Outer			10 % Change in Strut Lower		
	X	Y	Z	X	Y	Z	X	Y	Z	X	Y	Z	X	Y	Z	X	Y	Z
Change in Maximum Output Parameters																		
Camber (%)	2.11	4.93	12.8	0.74	3.15	5.01	0.95	10.9	7.62	0.29	1.81	4.24	1.23	0	3.34	0.61	79.6	5.28
Caster (%)	4.23	14.3	10	4.51	13.2	15	5	5.62	1.62	0	0	0	0	0	0	653	2.31	0
Kingpin (%)	0.27	0	1.04	0.24	0	1.41	0.09	25.4	1.74	0	0	0	0	0	0	0	0	0
Toe (%)	2.21	0	5.66	0.53	0	1.62	0.69	234	16.1	1.67	6.54	81.7	5.72	15.4	31.6	0.3	88.4	7.23
Change in Minimum Output Parameters																		
Camber (%)	2.26	4.39	14.2	2.26	4.39	22.9	0.95	11.1	9.72	0.37	1.76	5.88	2.43	13.6	4.36	0.37	103	9.79
Caster (%)	4.61	9.16	10	4.5	10.6	15	1180	0	5.29	0	0	0	0	0	0	4.94	14.3	0
Kingpin (%)	1.43	3.12	2.9	0.67	1.87	2.36	0.32	168	11.5	0	0	0	0	0	0	0	0	0
Toe (%)	3.06	6.33	7.78	1.6	2.51	6.46	1.9	1021	69.9	2.84	16.3	202	12.8	15.9	40.5	0.96	99.3	7.19

Table 4.4: Selection of input parameters on the performance of MPSS

Output Parameters	10 % Change in Front Control Arm along Z	10 % Change in Rear Control Arm along Z	10 % Change in Outer Control Arm along Z
Change in Maximum Output Parameters			
Camber (%)	12.8	5.01	7.62
Caster (%)	10	15	1.62
Kingpin (%)	1.04	1.41	1.74
Toe (%)	5.66	1.62	16.1
Change in Minimum Output Parameters			
Camber (%)	14.2	22.9	9.72
Caster (%)	10	15	5.29
Kingpin (%)	2.9	2.36	11.5
Toe (%)	7.78	6.46	69.9

Thus, with these considerations, front control arm, rear control arm, and outer control arm along z directions are selected as input parameter for optimization. In addition, for each of these parameters five variations are considered. These variations are the current parameter, two below the current parameter, and two above the current parameter. As such, total of one hundred twenty-five various models are modeled on ADAMS CAR software and the results are recorded accordingly, as shown in *Appendix 14*. Then, the optimization is performed using DESIGN EXPERT software as discussed below.

The optimization is performed through the following general steps. These are:

- ✎ Importing and setting the result data from ADAMS CAR,
- ✎ Selecting and configuring the analysis,
- ✎ Analyzing the data,
- ✎ Selecting the criteria for optimization, and
- ✎ Previewing the results.

In order to analyze the data in DESIGN EXPERT the input parameters should be categorized as “Factor”, and the output parameters should be categorized as “Response”, as shown in *Figure 4.39*.

Factor	Name	Units	Type	SubType	Minimum	Maximum	Coded Low	Coded High	Mean	Std. Dev.
A	FCA		Numeric	Continuous	-100.00	0.0000	-1 ↔ -100.00	+1 ↔ 0.00	-50.00	35.50
B	RCA		Numeric	Continuous	-150.00	-50.00	-1 ↔ -150.00	+1 ↔ -50.00	-100.00	35.50
C	OCA		Numeric	Continuous	-125.00	-25.00	-1 ↔ -125.00	+1 ↔ -25.00	-75.00	35.50

Response	Name	Units	Observations	Minimum	Maximum	Mean	Std. Dev.	Ratio
R1	Max_Camber		125.00	0.88	4.53	2.26	0.7112	5.15
R2	Max_Caster		125.00	0	4.87	1.87	1.24	N/A
R3	Max_KingPin		125.00	4.58	6.68	5.25	0.3783	1.46
R4	Max_Toe		125.00	1.5	6.4	2.63	1.17	4.27
R5	Min_Camber		125.00	-3.7	-0.4	-1.35	0.8281	9.25
R6	Min_Caster		125.00	-8.62	0	-2.75	2.07	N/A
R7	Min_KingPin		125.00	0	3.85	2.40	0.8134	N/A
R8	Min_Toe		125.00	-7.63	0.04	-3.20	1.83	N/A

Figure 4.39: Factor and Response parameters of Optimization in DESIGN EXPERT

For the analysis of the variance two types of models are used, as recommended by the software. As such, quartic model is used for all other remaining angles. Hence, *Equation 4.1* to *Equation 4.8* show the final equation in terms of coded factors showing the association between the input factors under investigation and the output parameters based on the quartic model.

$$\begin{aligned}
Cam_{max} = & 1.98 + 0.8508A + 0.333B - 1.04C + 0.0937AB + 0.0466AC - 0.2142BC \\
& + 0.1037A^2 + 0.1284B^2 + 0.065C^2 + 0.5374ABC - 0.0787A^2B \\
& + 0.3098A^2C - 0.1046AB^2 - 0.6419AC^2 + 0.2089B^2C - 0.5047BC^2 \\
& - 0.0413A^3 - 0.0371B^3 + 0.4053C^3 - 0.0671A^2B^2 - 0.0399A^2BC \\
& + 0.2545A^2C^2 - 0.0954AB^2C + 0.4296ABC^2 + 0.16B^2C^2 - 0.0501A^3B \\
& - 0.0744A^3C - 0.0061AB^3 - 0.4131AC^3 - 0.0435B^3C - 0.2661BC^3 \\
& - 0.0408A^4 + 0.0045B^4 + 0.1365C^4
\end{aligned}$$

Equation 4.1

$$\begin{aligned}
Cas_{max} = & 1.42 + 1.72A - 1.69B + 0.0731C - 0.5131AB + 0.1552AC - 0.1058BC \\
& + 0.5169A^2 + 0.5302B^2 - 0.0272C^2 - 0.1123ABC + 0.9329A^2B \\
& + 0.0899A^2C - 0.9233AB^2 - 0.0268AC^2 + 0.0461B^2C + 0.0274BC^2 \\
& - 0.2976A^3 + 0.2635B^3 + 0.0083C^3 + 0.6476A^2B^2 + 0.1364A^2BC \\
& + 0.0066A^2C^2 - 0.1426AB^2C + 0.0074ABC^2 + 0.0065B^2C^2 - 0.2674A^3B \\
& - 0.0454A^3C - 0.2542AB^3 + 0.0106AC^3 + 0.0345B^3C + 0.001BC^3 \\
& - 0.256A^4 - 0.2801B^4 + 0.012C^4
\end{aligned}$$

Equation 4.2

$$\begin{aligned}
KPI_{max} = & 4.99 + 0.2798A + 0.2413B - 0.0618C + 0.2349AB - 0.3883AC - 0.367BC \\
& + 0.1403A^2 + 0.0776B^2 + 0.3646C^2 - 0.0266ABC + 0.0021A^2B \\
& + 0.0437A^2C + 0.0114AB^2 - 0.0453AC^2 + 0.0021B^2C + 0.0005BC^2 \\
& - 0.0099A^3 - 0.0133B^3 + 0.0309C^3 - 0.0099A^2B^2 + 0.0157A^2BC \\
& - 0.0393A^2C^2 + 0.023AB^2C - 0.0362ABC^2 - 0.0328B^2C^2 - 0.0123A^3B \\
& + 0.0077A^3C - 0.0096AB^3 + 0.0344AC^3 + 0.0336B^3C + 0.0296BC^3 \\
& - 0.023A^4 + 0.013B^4 - 0.0137C^4
\end{aligned}$$

Equation 4.3

$$\begin{aligned}
Toe_{max} = & 2.23 - 0.9843A - 0.164B + 0.9871C - 0.0279AB - 0.6412AC - 0.0014BC \\
& + 0.3974A^2 + 0.1384B^2 + 0.2752C^2 - 0.1738ABC + 0.1137A^2B \\
& + 0.0954A^2C - 0.0179AB^2 - 0.0737AC^2 + 0.0305B^2C + 0.0845BC^2 \\
& - 0.0459A^3 + 0.0261B^3 + 0.0125C^3 - 0.0119A^2B^2 - 0.0257A^2BC \\
& + 0.1291A^2C^2 - 0.0369AB^2C + 0.0234ABC^2 + 0.0413B^2C^2 + 0.0061A^3B \\
& - 0.104A^3C - 0.0312AB^3 - 0.0653AC^3 + 0.0323B^3C - 0.0064BC^3 \\
& - 0.0256A^4 - 0.1003B^4 + 0.0397C^4
\end{aligned}$$

Equation 4.4

$$\begin{aligned}
Cam_{min} = & -1.13 - 0.5111A - 0.5283B + 1.01C - 0.2091AB + 0.4426AC + 0.4094BC \\
& - 0.095A^2 - 0.0622B^2 - 0.3347C^2 - 0.0705ABC - 0.0026A^2B \\
& - 0.0582A^2C + 0.0079AB^2 + 0.1158AC^2 - 0.0934B^2C + 0.1466BC^2 \\
& - 0.0173A^3 - 0.0035B^3 - 0.0576C^3 + 0.0032A^2B^2 - 0.0506A^2BC \\
& + 0.0565A^2C^2 - 0.0641AB^2C + 0.1555ABC^2 + 0.0558B^2C^2 - 0.0352A^3B \\
& - 0.0437A^3C - 0.0192AB^3 - 0.0744AC^3 - 0.0592B^3C - 0.0565BC^3 \\
& - 0.0061A^4 - 0.0114B^4 + 0.0219C^4
\end{aligned}$$

Equation 4.5

$$\begin{aligned}
Cas_{min} = & -2.23 - 2.43A + 2.59B - 0.3562C + 0.5253AB - 0.3905AC + 0.4363BC \\
& - 0.4368A^2 - 0.6596B^2 - 0.0177C^2 + 0.1196ABC - 0.9217A^2B \\
& - 0.008A^2C + 0.8964AB^2 - 0.0639AC^2 - 0.1339B^2C + 0.0933BC^2 \\
& + 0.2795A^3 - 0.2349B^3 - 0.0448C^3 - 0.6329A^2B^2 - 0.148A^2BC \\
& + 0.0106A^2C^2 + 0.0874AB^2C + 0.0085ABC^2 - 0.0342B^2C^2 + 0.241A^3B \\
& + 0.0837A^3C + 0.253AB^3 - 0.0336AC^3 + 0.0547BC^3 + 0.2497A^4 \\
& + 0.283B^4 - 0.0566C^4
\end{aligned}$$

Equation 4.6

$$\begin{aligned}
KPI_{min} = & 3.19 - 0.2481A + 0.0397B + 0.6639C - 0.6416AB + 1.36AC + 1.18BC \\
& - 0.549A^2 - 0.3226B^2 - 1.16C^2 - 0.211ABC - 0.0952A^2B - 0.2181A^2C \\
& - 0.0794AB^2 + 0.523AC^2 - 0.144B^2C + 0.4169BC^2 - 0.0211A^3 \\
& - 0.0168B^3 - 0.3171C^3 + 0.1073A^2B^2 - 0.3717A^2BC + 0.1992A^2C^2 \\
& - 0.3051AB^2C + 0.4889ABC^2 + 0.1456B^2C^2 + 0.0949A^3B - 0.1051A^3C \\
& + 0.1056AB^3 - 0.2811AC^3 - 0.0947B^3C - 0.1787BC^3 + 0.1272A^4 \\
& + 0.0072B^4 + 0.1352C^4
\end{aligned}$$

Equation 4.7

$$\begin{aligned}
Toe_{min} = & -3.03 + 2.03A + 0.8824B - 2.4C - 0.0565AB + 0.0075AC + 0.0522BC \\
& + 0.0161A^2 + 0.0396B^2 - 0.2543C^2 + 0.2157ABC - 0.1201A^2B \\
& + 0.7493A^2C - 0.0175AB^2 - 0.6784AC^2 + 0.0192B^2C - 0.1129BC^2 \\
& - 0.2816A^3 - 0.0043B^3 + 0.1837C^3 - 0.0309A^2B^2 + 0.1458A^2BC \\
& - 0.5584A^2C^2 + 0.0366AB^2C - 0.1616ABC^2 + 0.0131B^2C^2 - 0.0048A^3B \\
& + 0.3525A^3C - 0.0248AB^3 + 0.2405AC^3 - 0.0125B^3C + 0.0061BC^3 \\
& + 0.0293A^4 + 0.0053B^4 + 0.124C^4
\end{aligned}$$

Equation 4.8

After the setting of the parameters are completed, the next step is to select and configure the analysis. Then, DESIGN EXPERT uses analysis of variance (ANOVA) for analyzing the data.

Table 4.5: Criteria for optimization of MPSS in DESIGN EXPERT

S.No.	Parameters	Goal	Lower Limits	Upper Limits	Importance
1	Front Control Arm	In range	-100	0	3/5
2	Rear Control Arm	In range	-150	-50	3/5
3	Outer Control Arm	In range	-125	-25	3/5
4	Maximum Camber	Minimize	-0.5	2.5	3/5
5	Maximum Caster	Minimize	0	12	3/5
6	Maximum Kingpin	Minimize	5	10	3/5
7	Maximum Toe	Minimize	1.5	2.5	3/5
8	Minimum Camber	Minimize	-2	-0.5	3/5
9	Minimum Caster	Maximize	-1	0	5/5
10	Minimum Kingpin	Minimize	0	5	3/5
11	Minimum Toe	Minimize	-3	1.5	3/5

Finally, after the analysis is completed, the criteria for optimization can be set to optimize the results. This is done by selecting the goal of the optimization. Here, the goal, limits, and importance of the optimization is set, as shown in *Table 4.5*. The lower and upper limits are

fixed based on the results from the analysis, the equilibrium results, and the standards required. Whereas, the importance is set to ensure all the parameters are considered in the optimization. Then, the DESIGN EXPERT gives the list of optimum solution in the form of graphs, charts and tables. Finally, the user can select the optimum solution based on the higher desirability factor and required range of parameters.

4.4. Validation of the Optimization of Macpherson strut suspension system

After the optimization process is completed using the DESIGN EXPERT, the next step is to validate the results. Here, this is performed using Artificial Neural Network (ANN) under the MATLAB software using Neural Network fitting tool. ANNs generate predictions and decisions by representing complex correlations in data. ANN neurons have synaptic weight connections, which enable the neurons to store information. Front control arm, rear control arm, and outer control arm along z direction were employed as independent input variables in a combined data set that included 125 variations. Whereas, the maximum, minimum, bump and rebound point of camber, caster, kingpin, and toe angles were used as dependent target variables.

Thus, inorder to validate the result, as shown in *Figure 4.40* and *Figure 4.41*, an ANN based on Levenberg-Marquardt model consisting of three input layers, ten hidden layers, and one output layer was examined for each required output. Thus, performance metrics such as mean square error (MSE), and coefficient of correlation (R^2) were utilized to validate the accuracy of the RSM and ANN models.

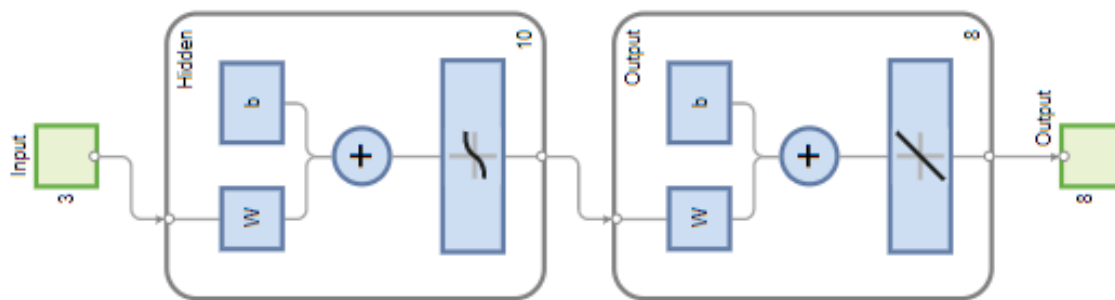


Figure 4.40: ANN Architecture

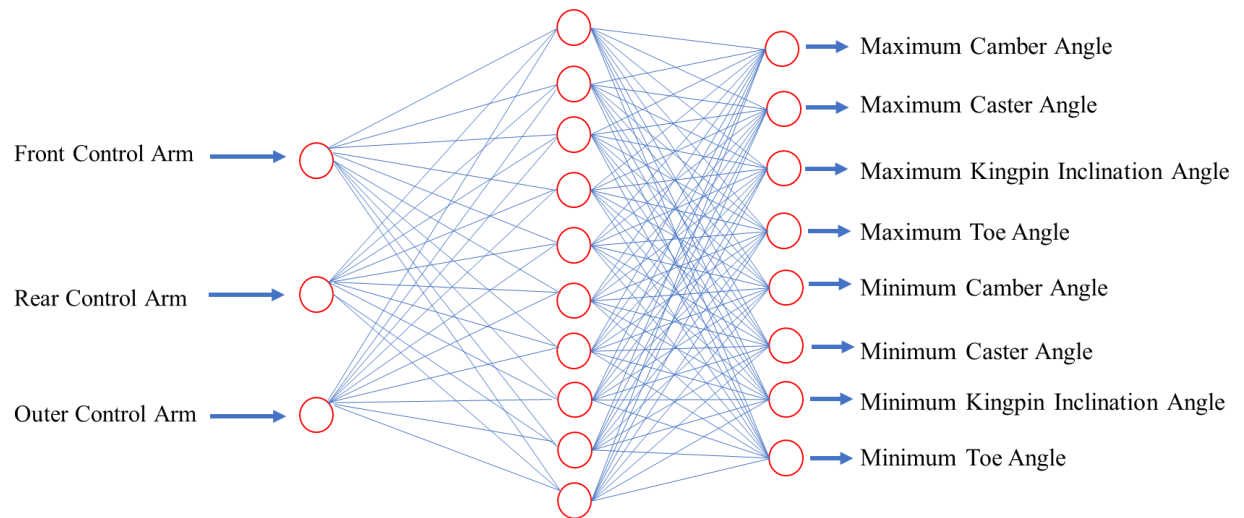


Figure 4.41: Layers of the Artificial Neural Network model

CHAPTER FIVE

RESULT AND DISCUSSION

For the purpose of reducing the effects on wheel alignment, a kinematic analysis of the Macpherson strut suspension system using Adams Car is performed, as shown in *Figure 5.1*. The model is designed based on the design specifications acquired from Toyota Corolla 2022 model. This is done by considering various suspension parameters such as camber angle, toe angle, kingpin inclination angle, and caster angle. Hence, the results obtained both from the simulation of the kinematic analysis are discussed below.

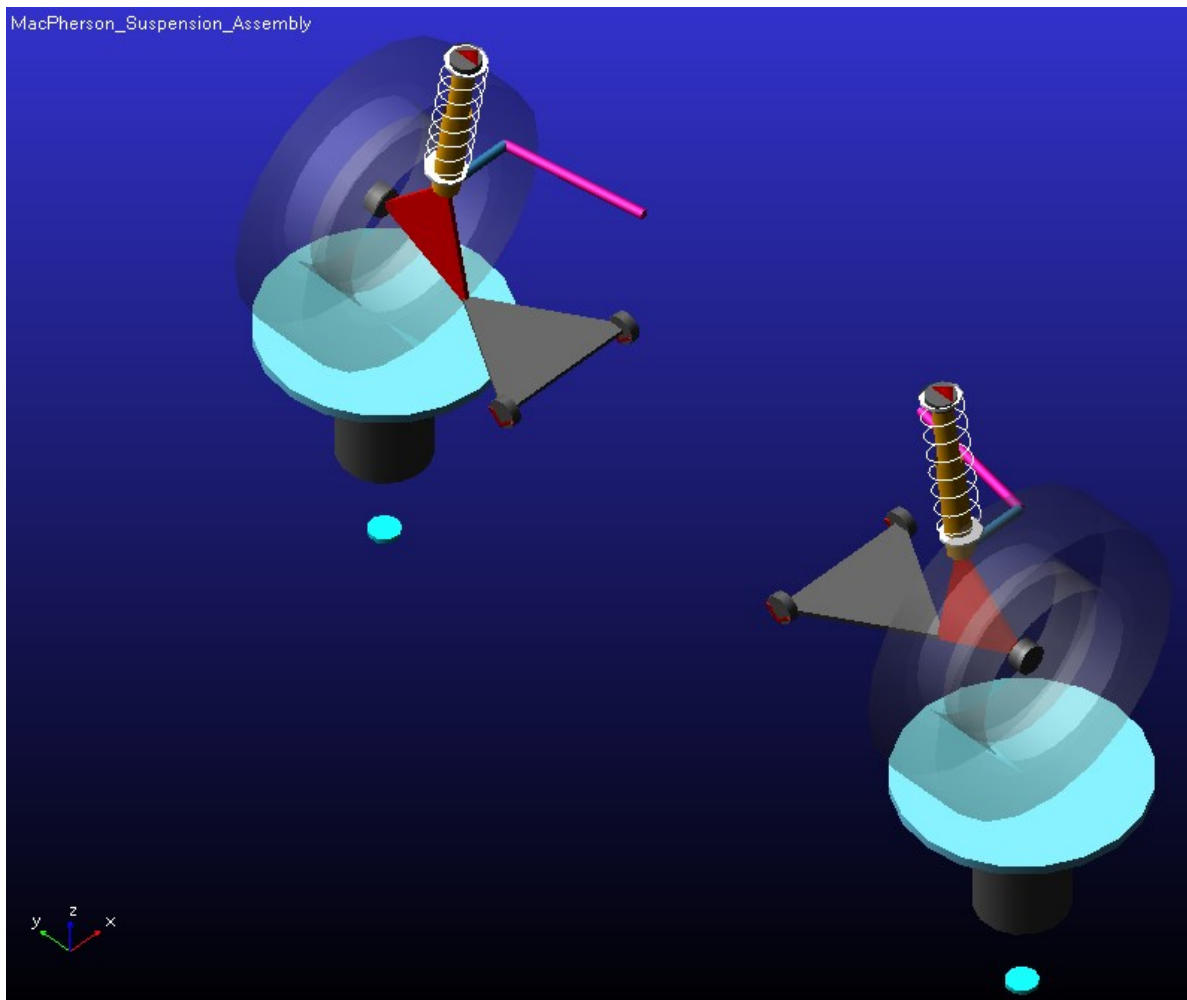


Figure 5.1: Assembly of MacPherson suspension system in Adams Car

5.1. Current Model of MPSS

For the current model of MPSS, four principal performance parameters are obtained graphically. These are camber angle, caster angle, kingpin inclination angle, and toe angle. In addition, for each of these angles, maximum and minimum values are recorded, as shown in *Table 5.1*. As such, minimum and maximum camber angles are -1.09° and 2.07° , respectively. Whereas, at static balance, the camber angle is -0.5° . In addition, minimum and maximum caster angles are -2.4° and 1.6° , respectively. Whereas, at static balance, the caster angle is 0° . Moreover, minimum and maximum kingpin inclination angles are 3.09° and 5.0° , respectively. Whereas, at static balance, the kingpin inclination angle is 4.85° . Furthermore, minimum and maximum toe angles are -3.09° and 2.25° , respectively. Whereas, at static balance, the toe angle is 1.5° .

Table 5.1: The performance results of the current MPSS Model

	Maximum	Equilibrium	Minimum	Rebound	Bump
Camber	2.07	-0.5	-1.09	2.07	-0.94
Caster	1.6	0	-2.4	1.6	-2.4
Kingpin	5.0	4.85	3.09	3.09	4.25
Toe	2.25	1.5	-3.09	-3.09	1.2

5.2. Optimized Model of MPSS

After optimizing the parameters, the DESIGN EXPERT provided four alternative solutions, as shown in *Table 5.2*. It includes the optimized parameters for all the 8 output parameters. As such, the optimized model recommends to use -50.9, -89.69, and -77.36 for front control arm, rear control arm, and lower control arms Z positions, respectively. As such, minimum and maximum optimized camber angles are -1.176° and 2.017° , respectively. In addition, minimum and maximum optimized caster angles are -2.158° and 1.376° , respectively. Moreover, minimum and maximum optimized kingpin inclination angles are 3.142° to 4.988° ,

respectively. Furthermore, minimum and maximum optimized toe angles are -2.948° and 2.196° , respectively.

Table 5.2: Alternative Solutions of Optimization in DESIGN EXPERT

No.	FCA	RCA	OCA	Max_Camb	Max_Cast	Max_KPI	Max_Toe	
1	-50.9	-89.69	-77.36	2.017	1.376	4.988	2.196	Selected
2	-92.36	-150	-125	2.034	1.74	4.622	2.223	
3	-91.69	-150	-125	2.045	1.761	4.626	2.212	
4	-92.05	-149.24	-125	2.043	1.73	4.626	2.219	
				Max_Camb	Max_Cast	Max_KPI	Max_Toe	
1	-50.9	-89.69	-77.36	-1.176	-2.158	3.142	-2.948	Selected
2	-92.36	-150	-125	-1.122	-2.555	2.778	-3.244	
3	-91.69	-150	-125	-1.128	-2.584	2.778	-3.244	
4	-92.05	-149.24	-125	-1.132	-2.537	2.786	-3.245	

The response surfaces obtained from the DESIGN EXPERT software from the optimized model are shown in *Figure 5.2* to *Figure 5.5*. As such, the response surface of *Figure 5.2* shows that, for the given outer control arm and rear control arm, as front control arm position increases, maximum camber angle increases and minimum camber angle decreases. Similarly, for the given outer control arm and front control arm, as rear control arm position increases, maximum camber angle increases and minimum camber angle decreases. Whereas, for the given rear control arm and front control arm, as outer control arm position increases, maximum camber angle decreases and minimum camber angle increases.

In addition, the response surface of *Figure 5.3* shows that, for the given outer control arm and rear control arm, as front control arm position increases, maximum caster angle increases and minimum caster angle decreases. Similarly, for the given rear control arm and front control arm, as outer control arm position increases, maximum caster angle increases and minimum caster angle decreases. Whereas, for the given outer control arm and front control arm, as rear

control arm position increases, maximum caster angle decreases and minimum caster angle increases.

Moreover, the response surface of *Figure 5.4* shows that, for the given outer control arm and rear control arm, as front control arm position increases, maximum Kingpin inclination angle increases and minimum Kingpin inclination angle decreases. Similarly, for the given outer control arm and front control arm, as rear control arm position increases, maximum Kingpin inclination angle increases and minimum Kingpin inclination angle decreases. Whereas, for the given rear control arm and front control arm, as outer control arm position increases, maximum Kingpin inclination angle decreases and minimum Kingpin inclination angle increases.

Moreover, the response surface of *Figure 5.5* shows that, for the given outer control arm and rear control arm, as front control arm position increases, maximum toe angle decreases and minimum toe angle increases. Similarly, for the given outer control arm and front control arm, as rear control arm position increases, maximum toe angle decreases and minimum toe angle increases. Whereas, for the given rear control arm and front control arm, as outer control arm position increases, maximum toe angle increases and minimum toe angle decreases.

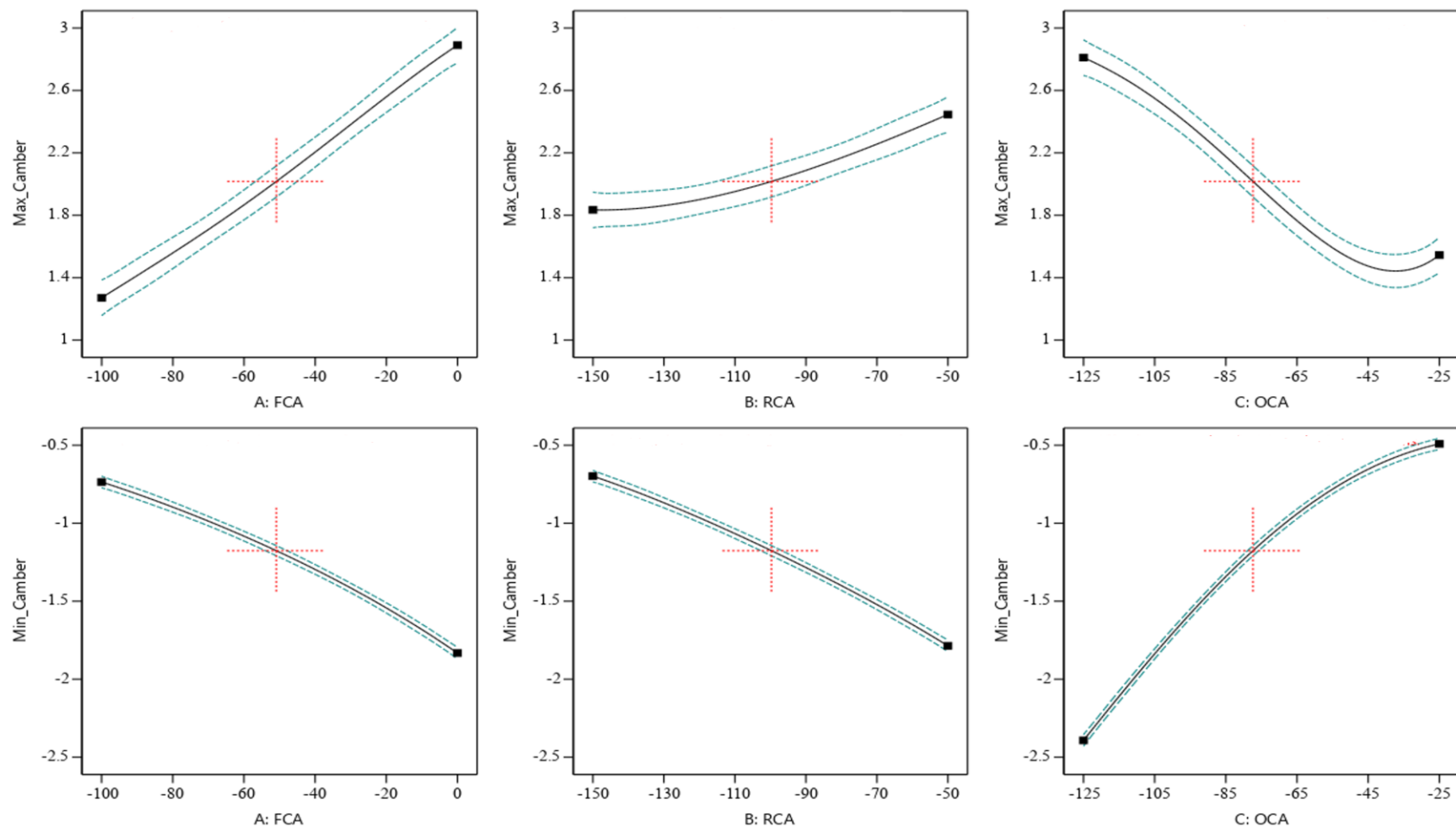


Figure 5.2: Surface response of Camber angles variation of MPSS for Optimized Model in DESIGN EXPERT

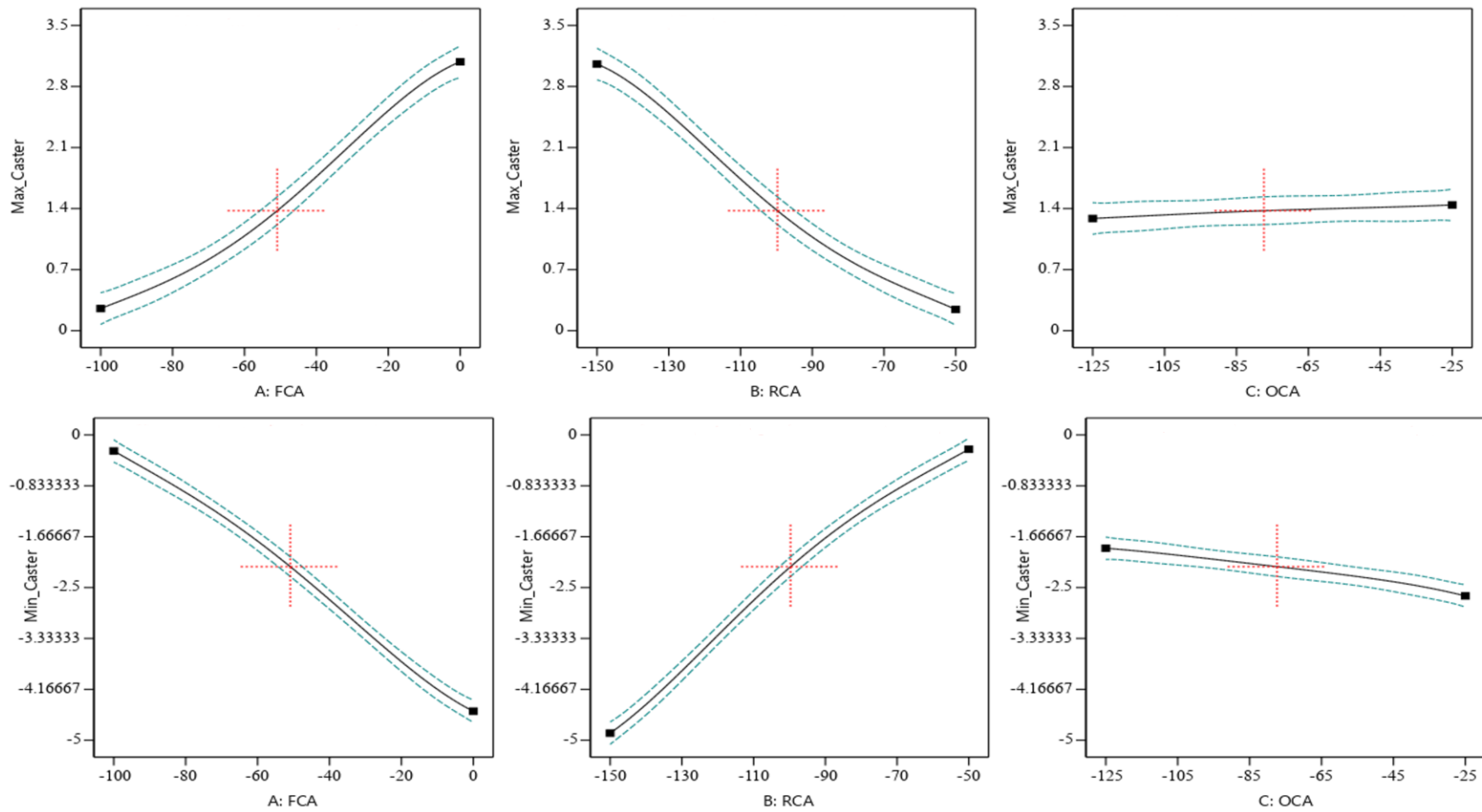


Figure 5.3: Surface response of Caster angles variation of MPSS for Optimized Model DESIGN EXPERT

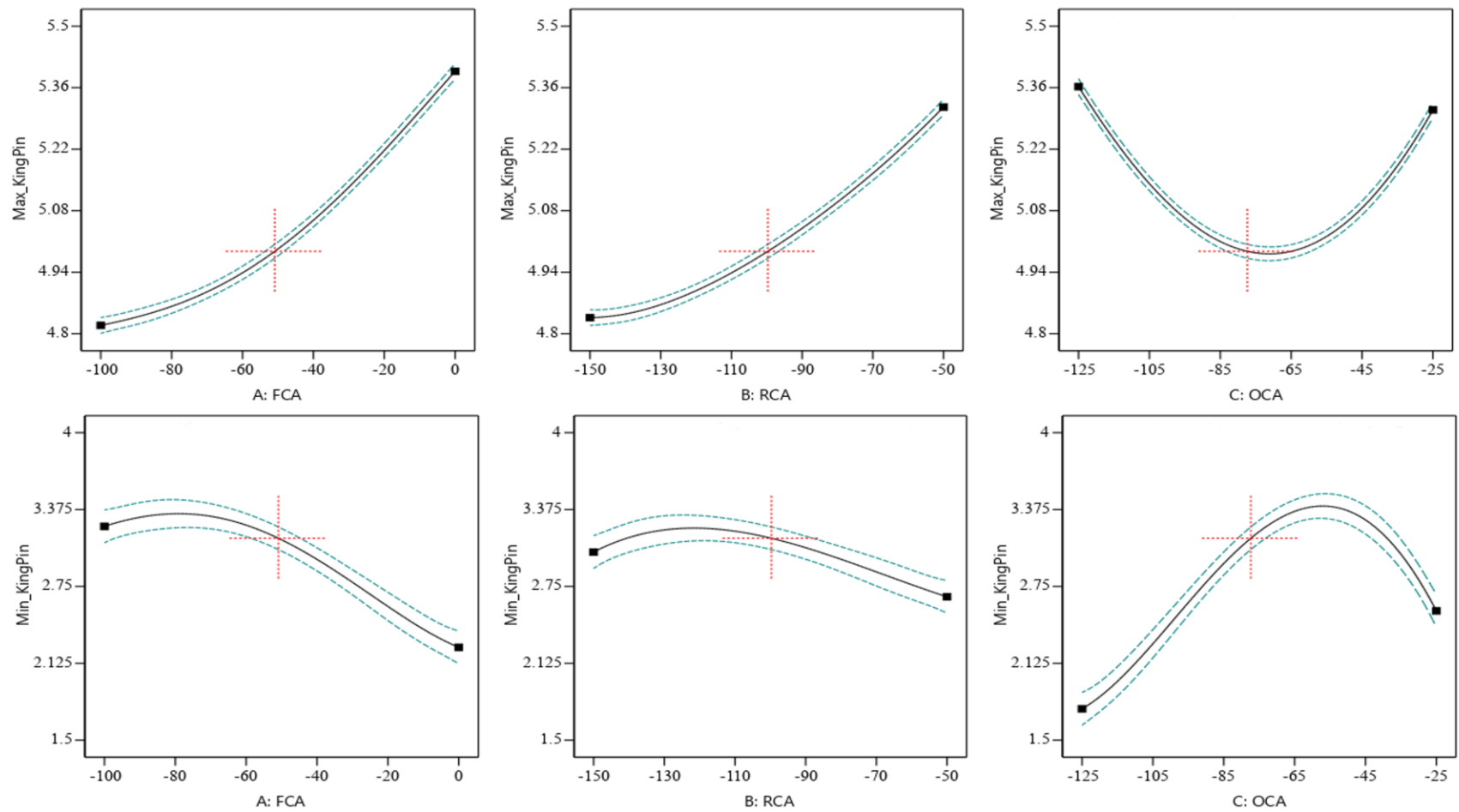


Figure 5.4: Surface response of Kingpin inclination angles variation of MPSS for Optimized Model DESIGN EXPERT

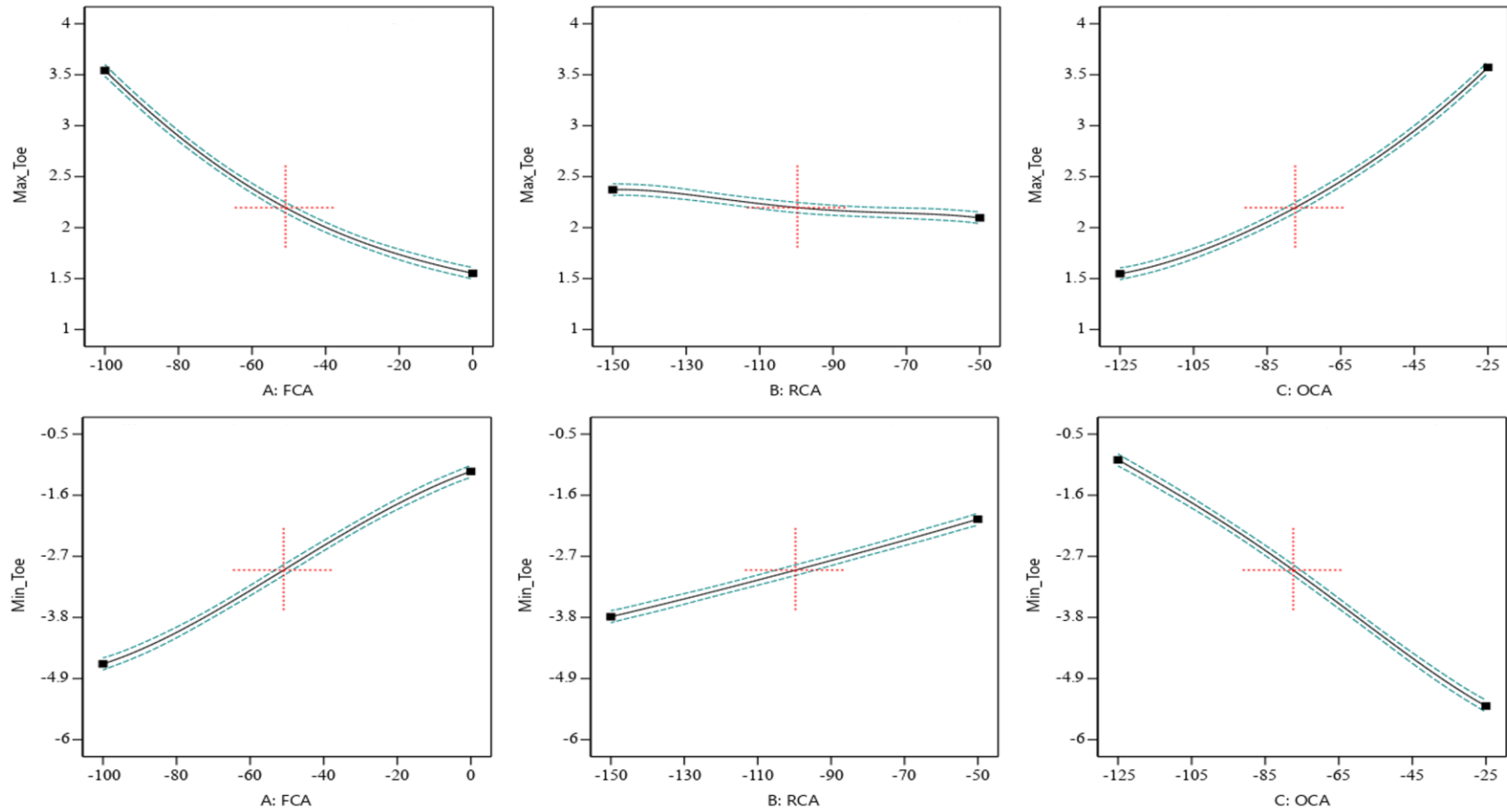


Figure 5.5: Surface response of Toe angles variation of MPSS for Optimized Model DESIGN EXPERT

However, the final optimized result should be obtained from ADAMS CAR software. Thus, the new optimized model is developed based on the recommended values of input parameters. Thus, minimum and maximum optimized camber angles are -1.27° and 2.02° , respectively. Whereas, at static balance, the camber angle is -0.5° . In addition, minimum and maximum optimized caster angles are -1.82° and 1.22° , respectively. Whereas, at static balance, the caster angle is 0° . Moreover, minimum and maximum optimized kingpin inclination angles are 2.96° and 4.95° , respectively. Whereas, at static balance, the kingpin inclination angle is 4.5° . Furthermore, minimum and maximum optimized toe angles are -2.86° and 2.22° , respectively. Whereas, at static balance, the toe angle is 1.5° .

5.3. Validation Using ANN

In this validation, the ANN model was established with a structure of three input layers, ten hidden layers, and one output neuron layer, for each of the outputs. As such, Mean squared error performance plot, shown in *Figure 5.6* to *Figure 5.13*, and regression analysis plot, shown in *Figure 5.14* to *Figure 5.17* are obtained. Moreover, *Table 5.3* compares the results correlation of correlations of response surface model with ANN model.

As such, as shown in *Figure 5.6* and *Figure 5.10*, the best validation points are obtained at 0.02432, and 0.00174 MSE for the maximum and minimum camber angles, respectively. In addition, as shown in *Figure 5.7* and *Figure 5.11*, the best validation points are obtained at 0.00044 and 0.00203 MSE for the maximum and minimum caster angles, respectively. Moreover, as shown in *Figure 5.8* and *Figure 5.12*, the best validation point is obtained at 0.00051 and 0.02768 MSE for the maximum and minimum Kingpin inclination angles, respectively. Furthermore, as shown in *Figure 5.9* and *Figure 5.13*, the best validation point is obtained at 0.00431 and 0.10085 MSE for the maximum and minimum toe angles, respectively. Hence, since MSE of all the models are less than 0.2, all of the developed models are accurate.

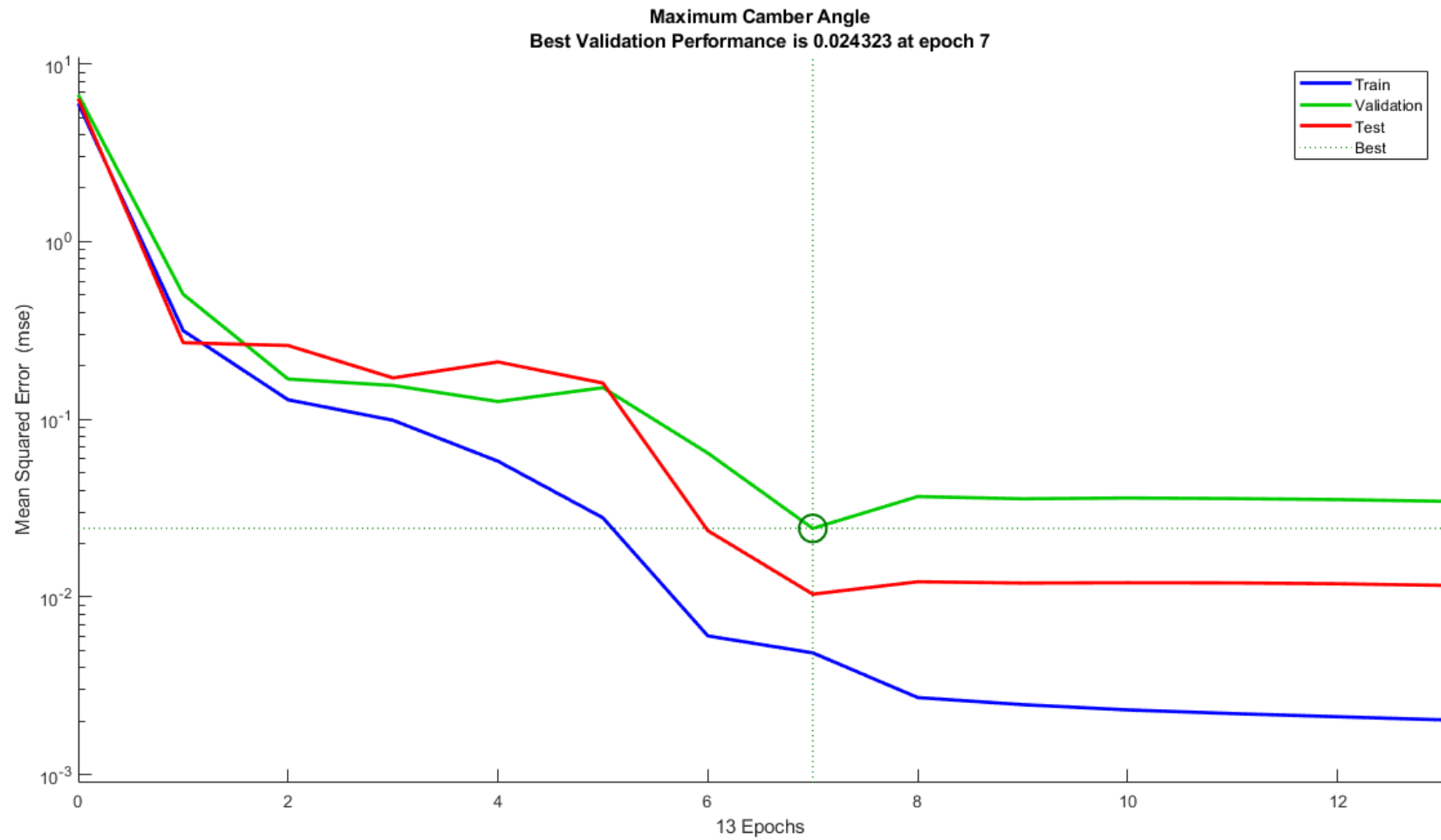


Figure 5.6: ANN model performance plot for Maximum Camber angle

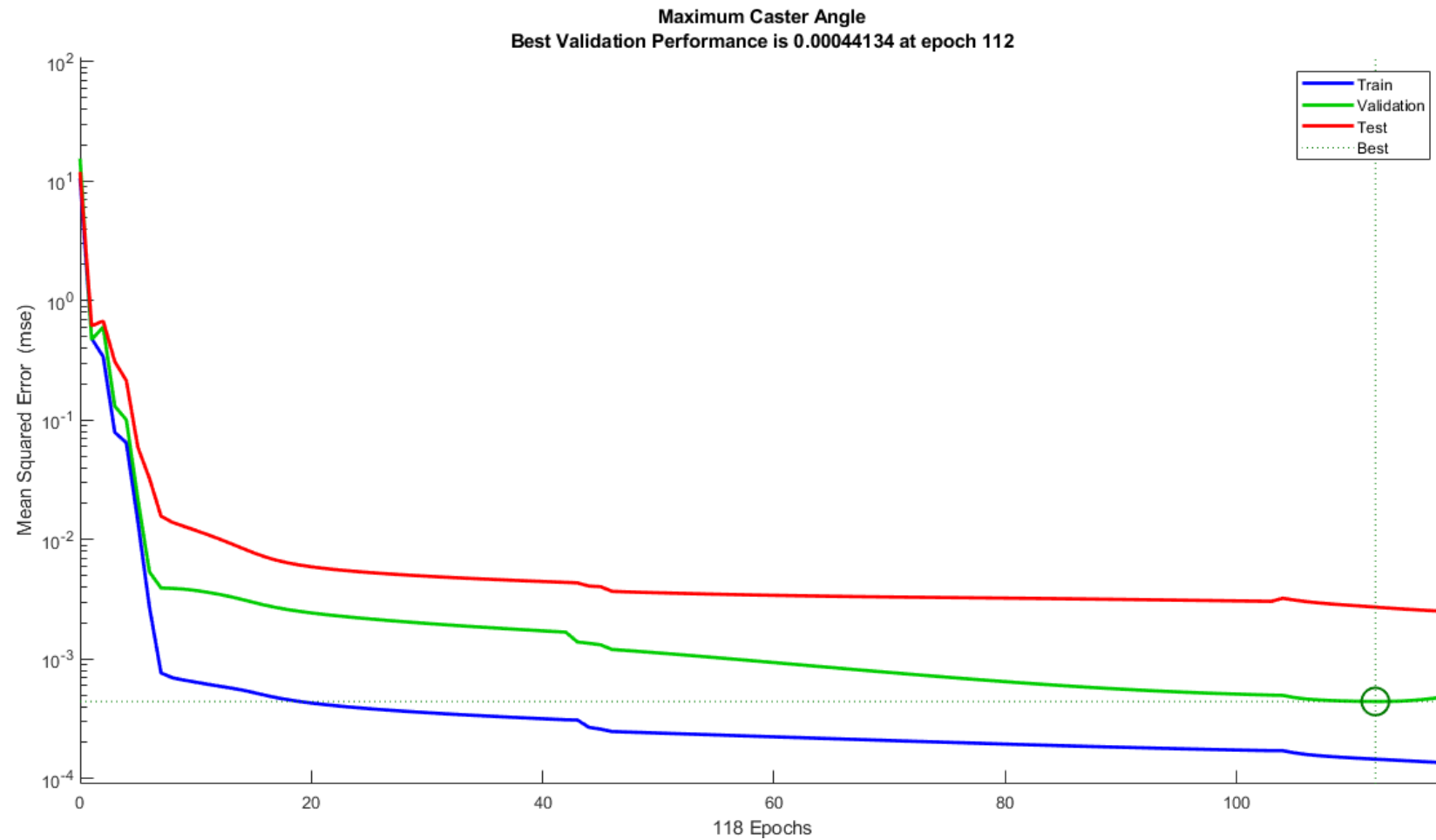


Figure 5.7: ANN model performance plot for Maximum Caster angle

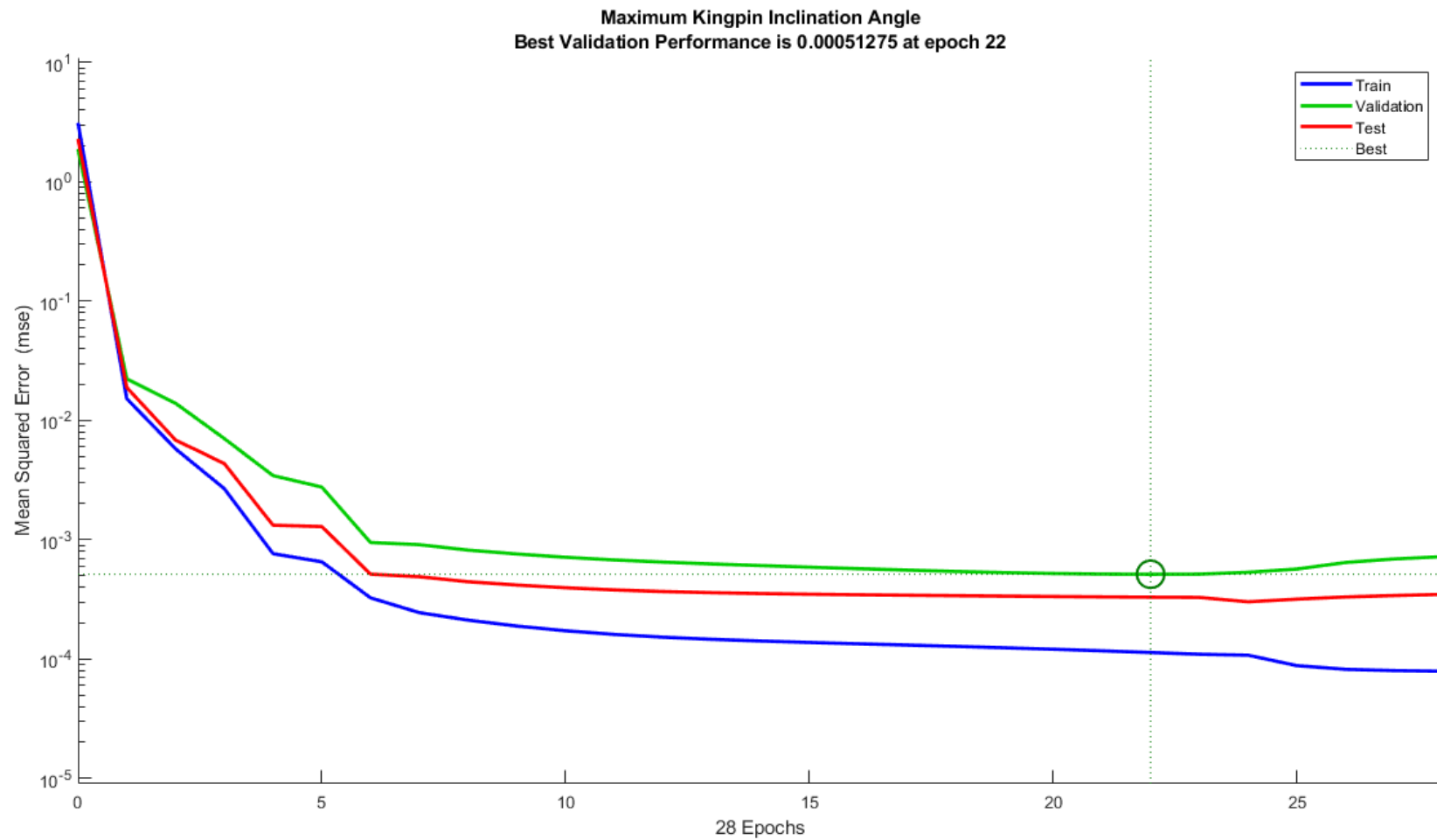


Figure 5.8: ANN model performance plot for Maximum Kingpin Inclination angle

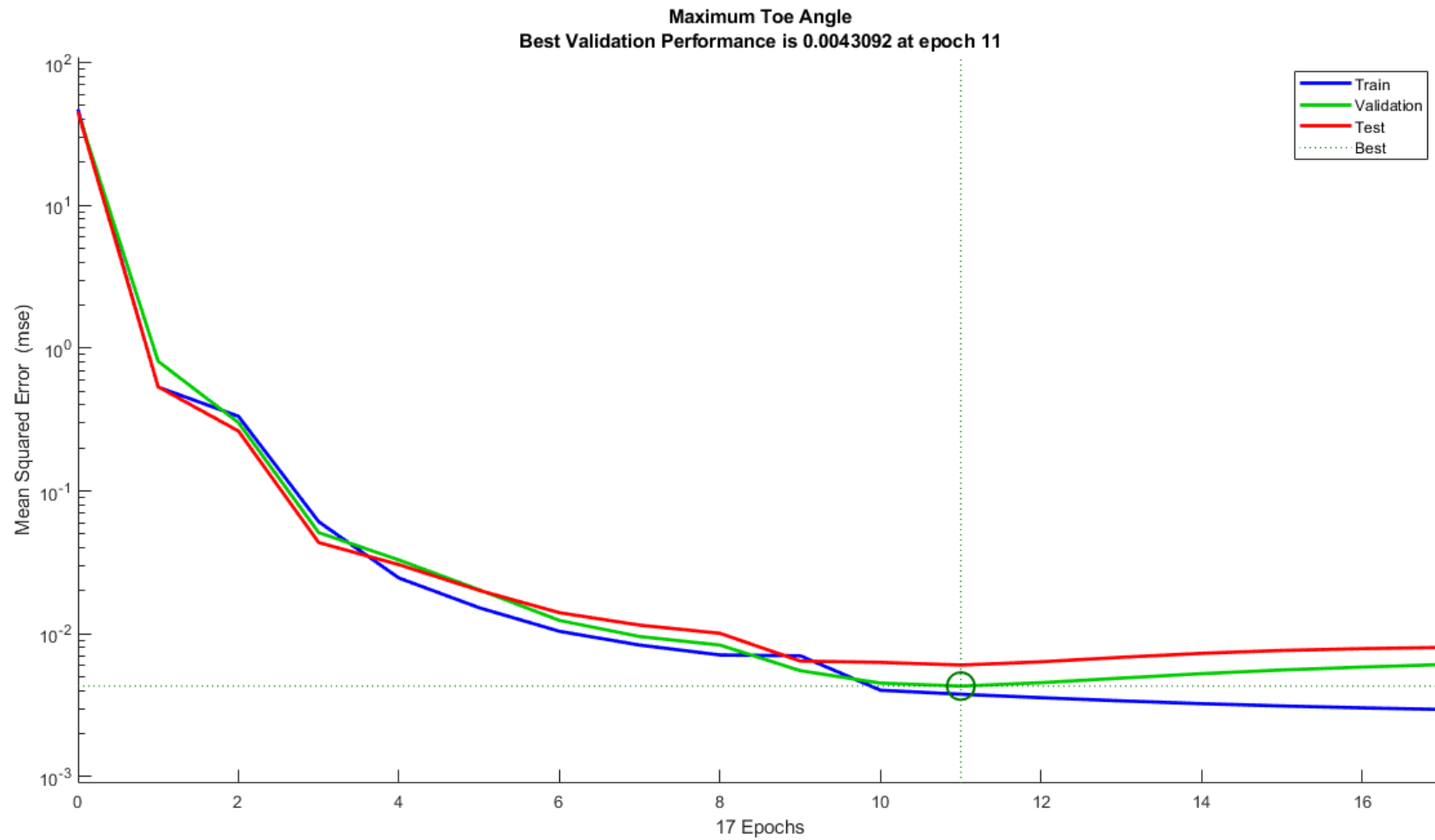


Figure 5.9: ANN model performance plot for Maximum Toe angle

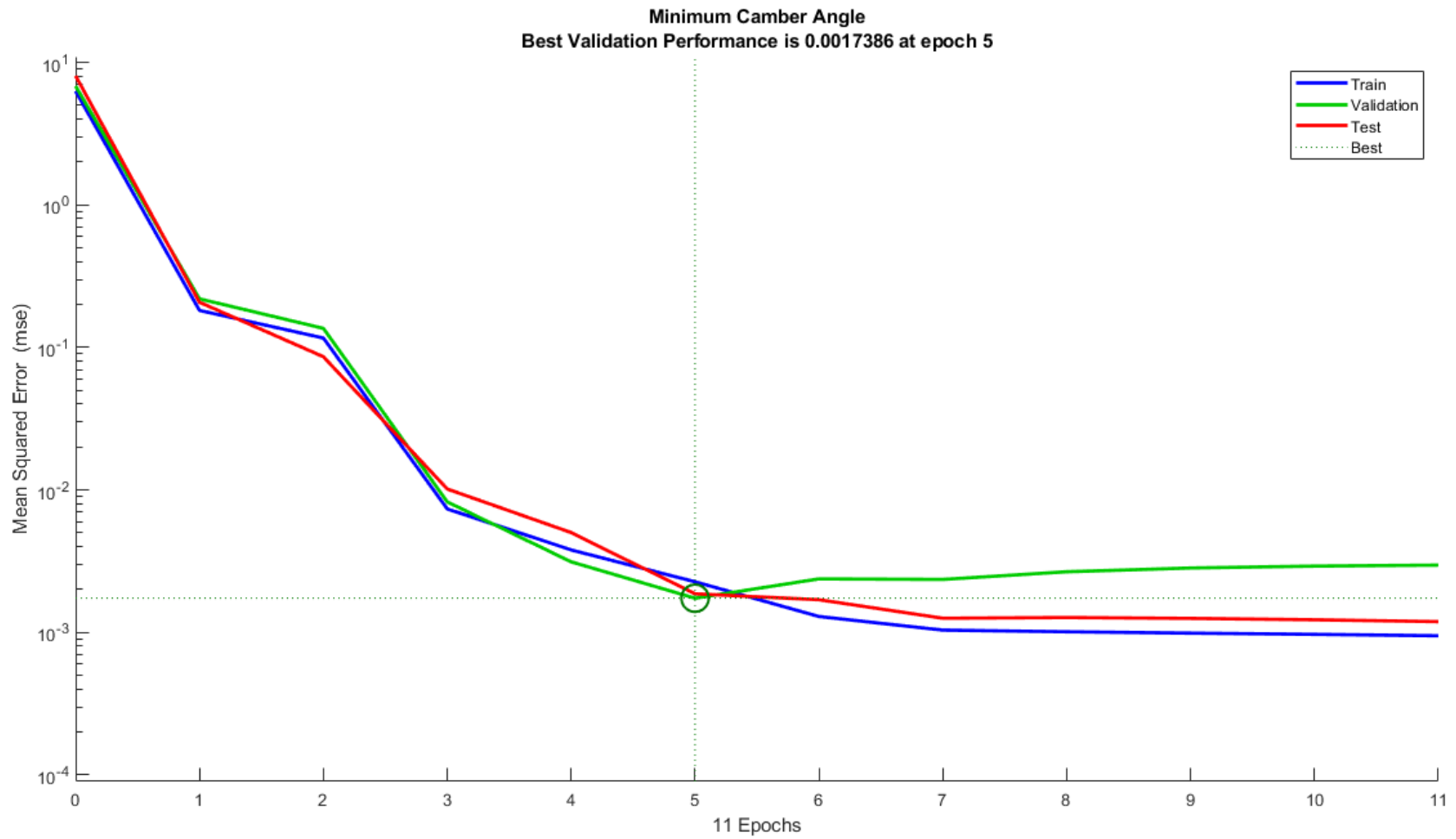


Figure 5.10: ANN model performance plot for Minimum Camber angle

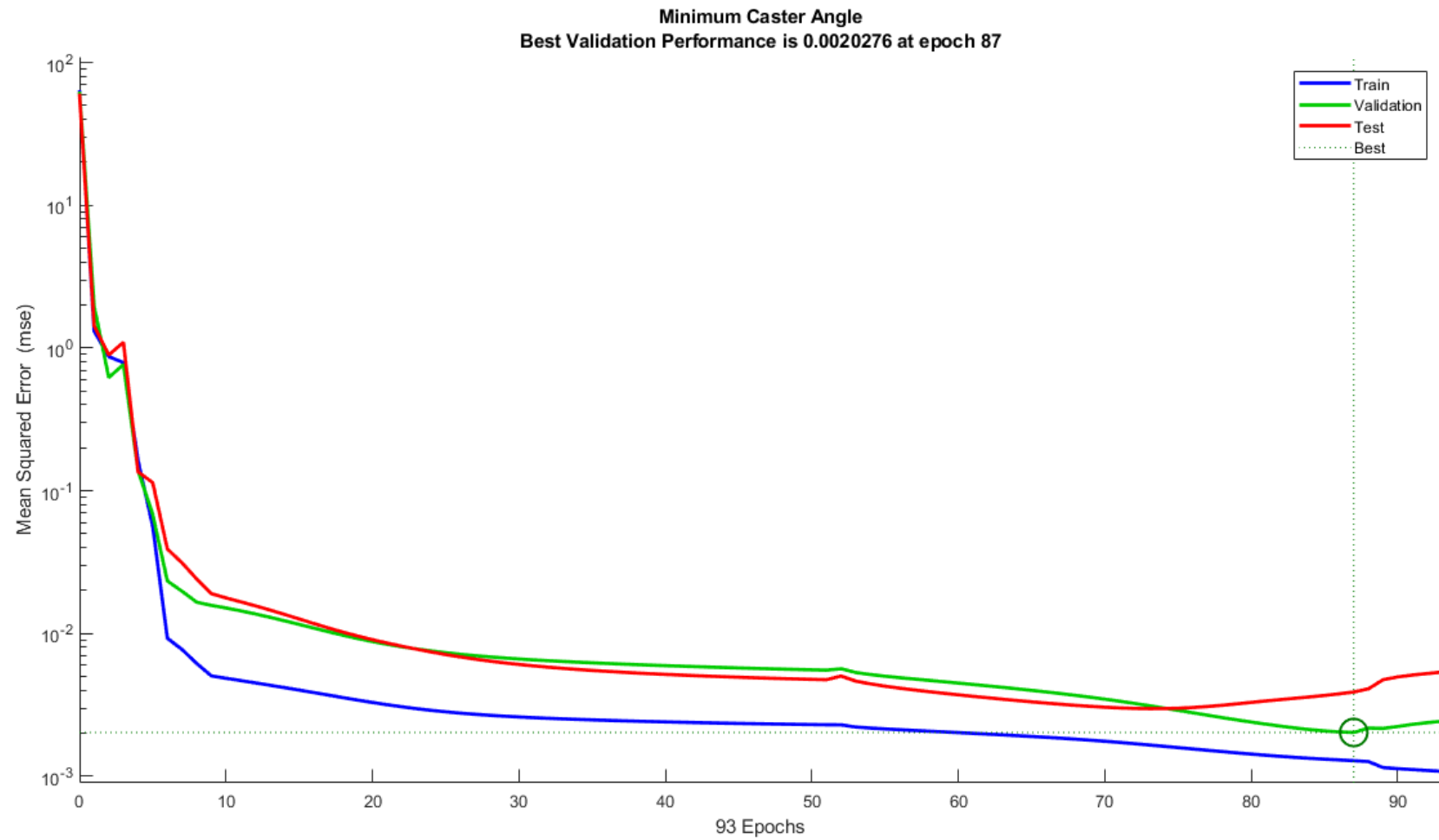


Figure 5.11: ANN model performance plot for Minimum Caster angle

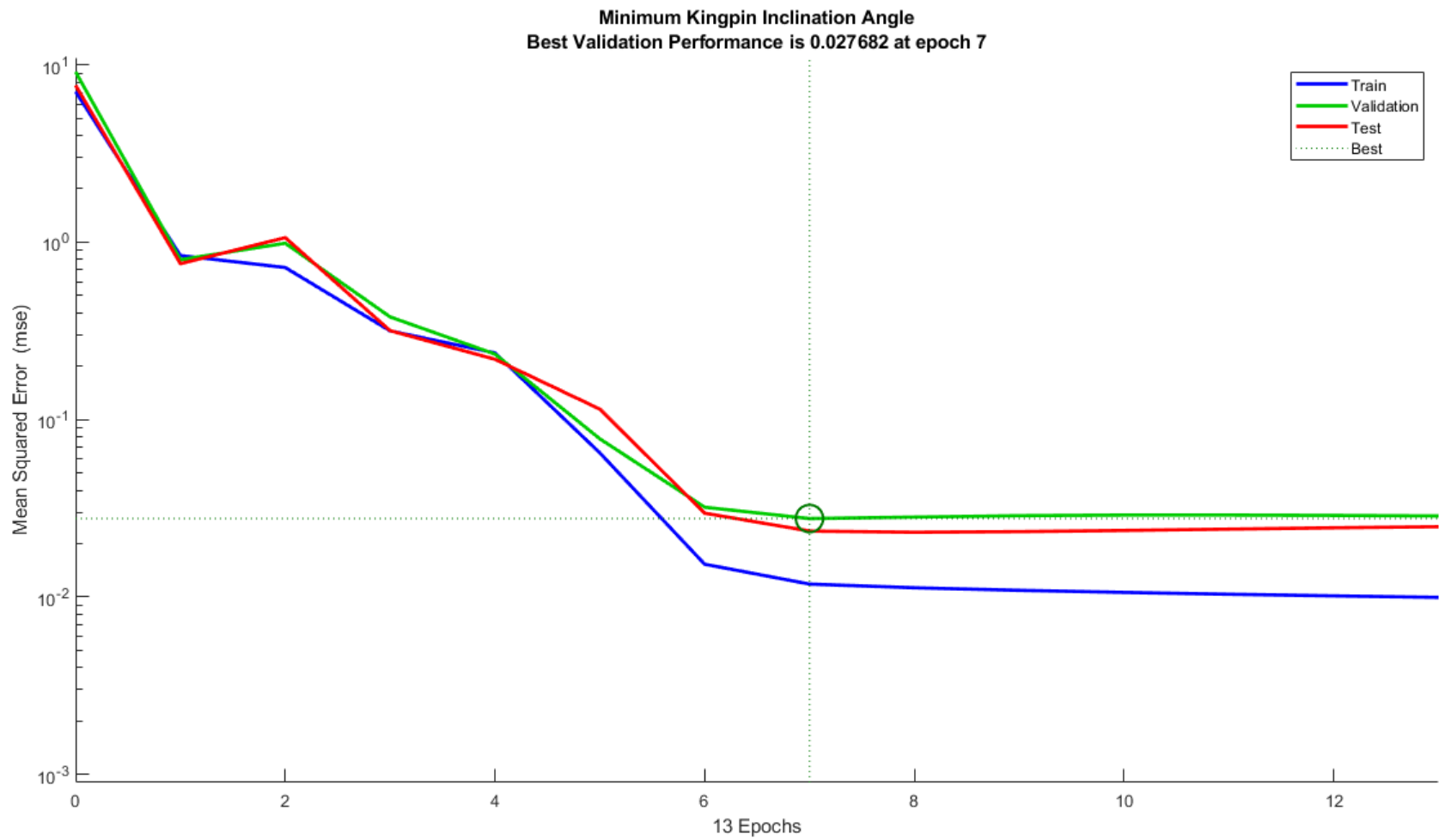


Figure 5.12: ANN model performance plot for Minimum Kingpin Inclination angle

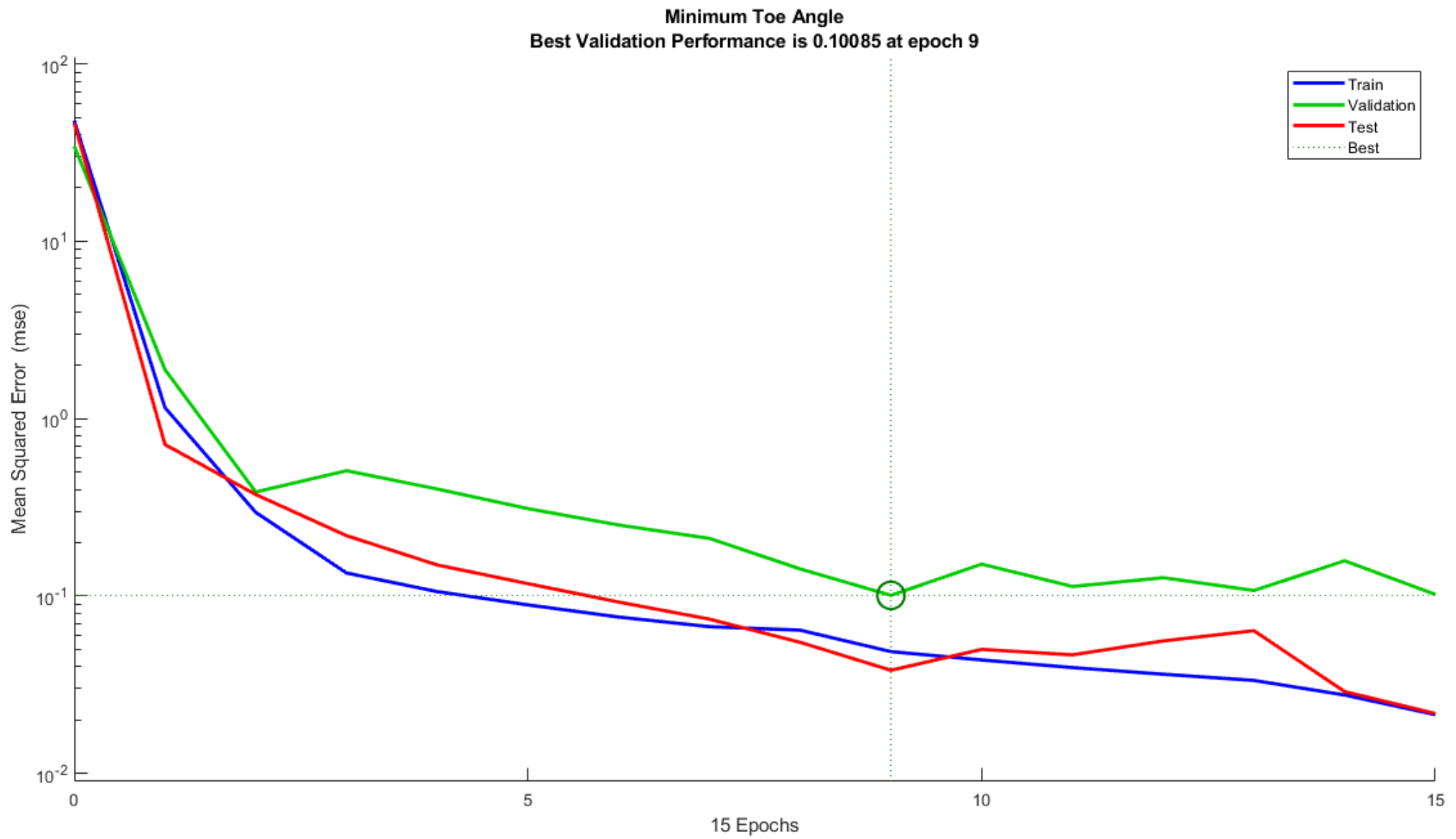


Figure 5.13: ANN model performance plot for Minimum Toe angle

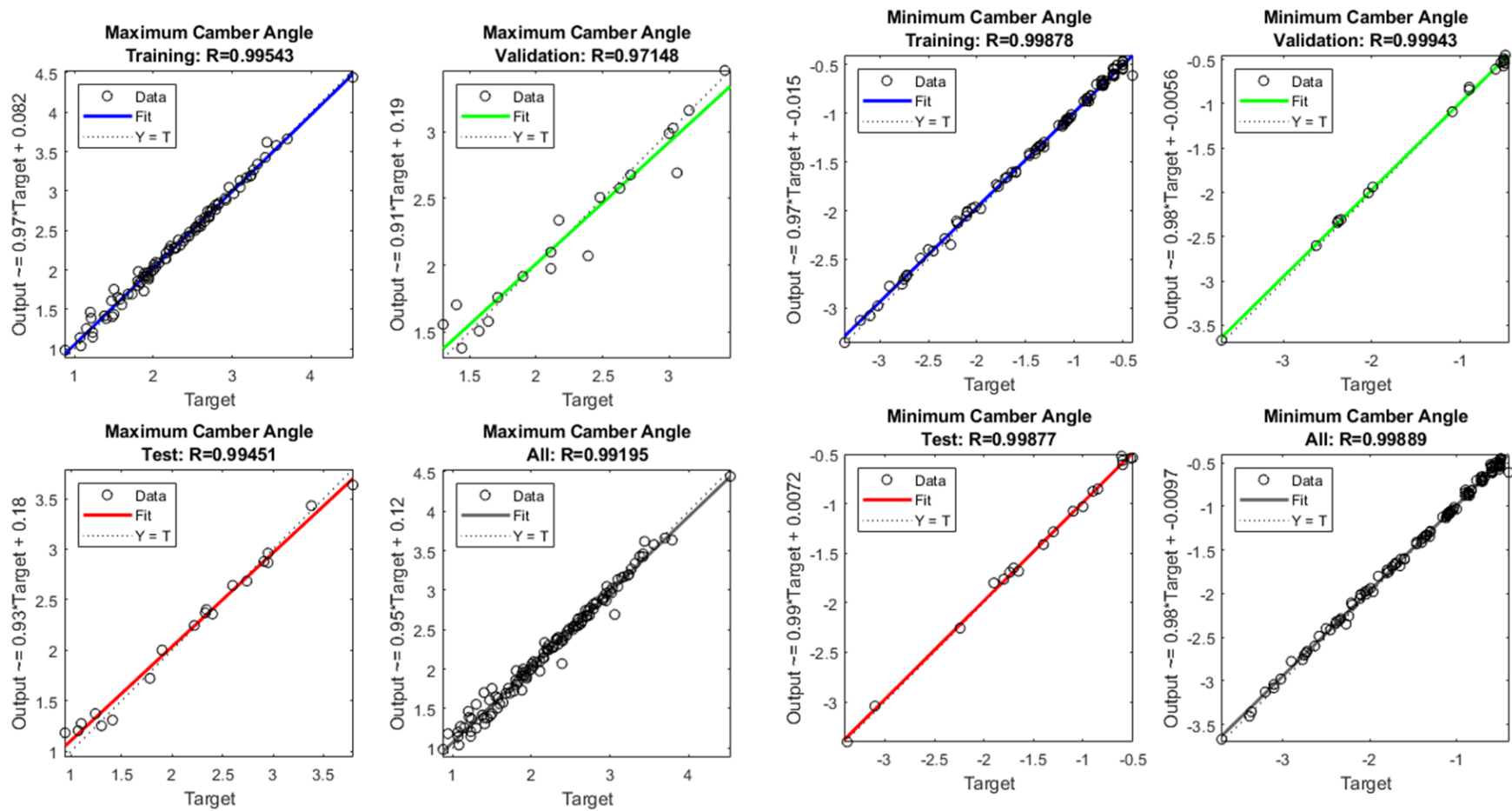


Figure 5.14: ANN model regression analysis plot Maximum and Minimum Camber angles

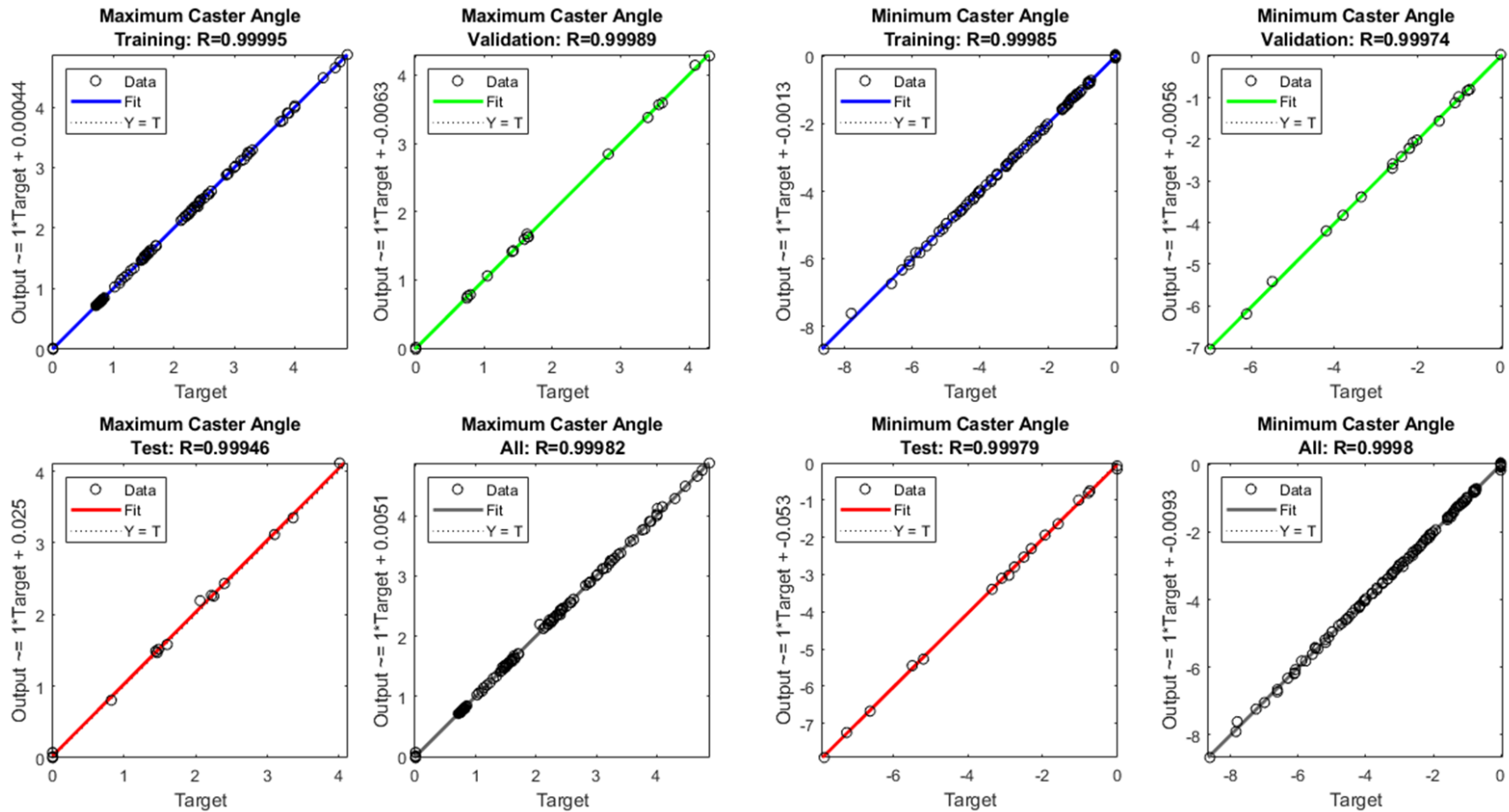


Figure 5.15: ANN model regression analysis plot Maximum and Minimum Caster angles

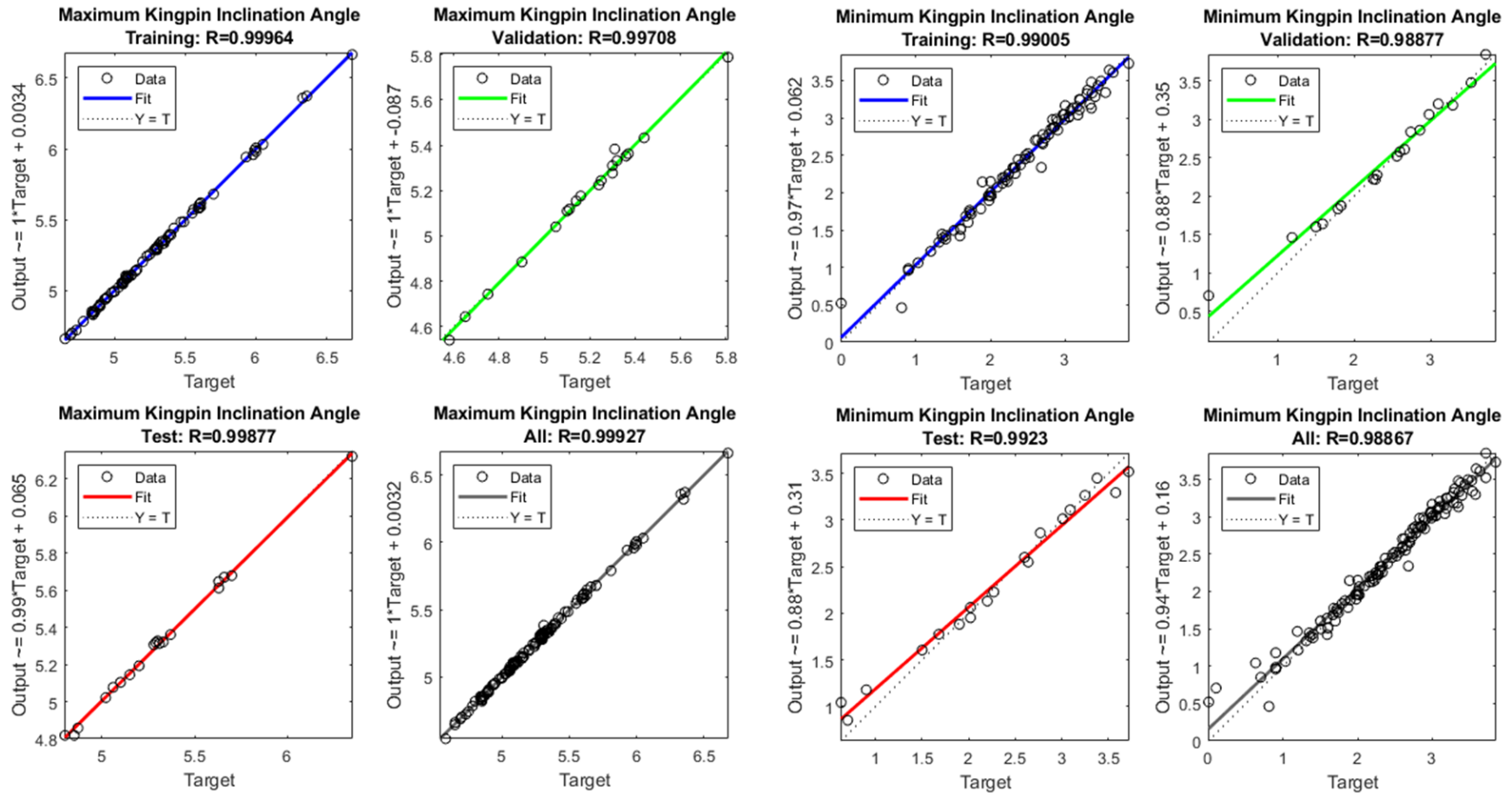


Figure 5.16: ANN model regression analysis plot Maximum and Minimum Kingpin Inclination angles

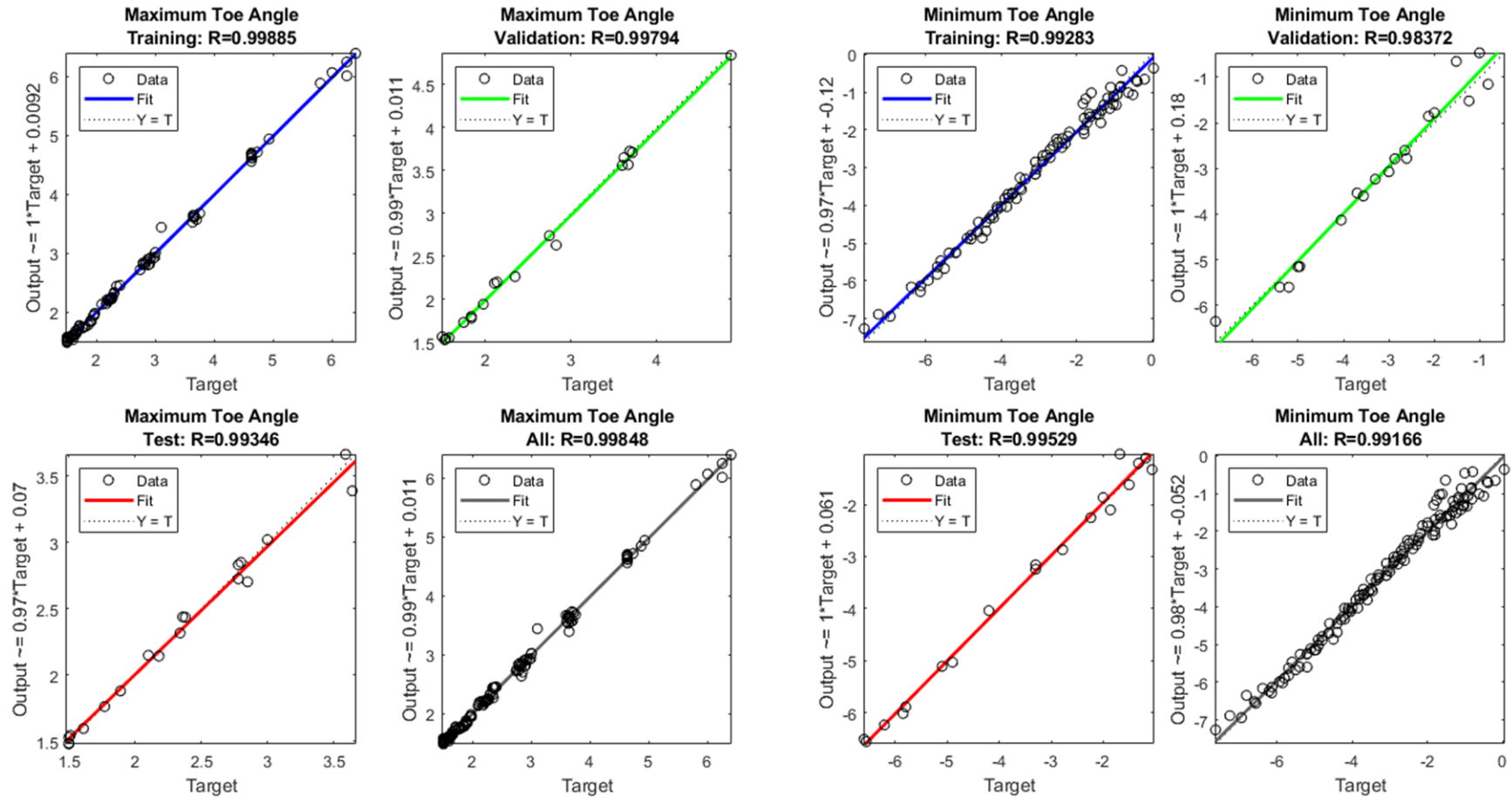


Figure 5.17: ANN model regression analysis plot Maximum and Minimum Toe angles

Similarly, as shown in *Figure 5.14*, 0.9919 and 0.9988 R values are obtained for the maximum and minimum camber angles, respectively. In addition, as shown in *Figure 5.15*, 0.9998 and 0.9998 R values are obtained for the maximum and minimum caster angles, respectively. Moreover, as shown in *Figure 5.16*, 0.9993 and 0.9887 R values are obtained for the maximum and minimum Kingpin inclination angles, respectively. Furthermore, as shown in *Figure 5.17*, 0.9985 and 0.9917 R values are obtained for the maximum and minimum toe angles, respectively. Hence, since all of the models obtained more than 0.98 R values, the developed models are accurate.

Table 5.3: Comparison of ANN Model and DESIGN EXPERT Response Surface

MPSS Angles	Correlation of Correlations (R^2)		
	Response Surface	ANN Model	Deviation (%)
Maximum Camber	0.9762	0.9838	0.77
Maximum Caster	0.9802	0.9996	1.98
Maximum Kingpin	0.9979	0.9986	0.07
Maximum Toe	0.9977	0.9970	0.07
Minimum Camber	0.9982	0.9976	0.06
Minimum Caster	0.9927	0.9996	-0.69
Minimum Kingpin	0.9879	0.9775	1.05
Minimum Toe	0.9967	0.9835	1.32

As shown in *Table 5.3*, 0.9762, 0.9802, 0.9979, and 0.9977 R^2 values are obtained for the camber, caster, Kingpin, and toe maximum angles, respectively, in response surface method. Whereas, 0.9838, 0.9996, 0.9986, and 0.9970 R^2 values are obtained for the camber, caster, Kingpin, and toe maximum angles, respectively, in ANN. In addition, 0.9982, 0.9927, 0.9879, and 0.9967 R^2 values are obtained for the camber, caster, Kingpin, and toe minimum angles, respectively, in response surface method. Whereas, 0.9976, 0.9996, 0.9775, and 0.9835 R^2 values are obtained for the camber, caster, Kingpin, and toe minimum angles, respectively, in ANN.

Hence, the response surface model has the minimum R^2 value of 0.9802 at maximum caster angle. Whereas, the ANN model has the minimum R^2 value of 0.9775 at minimum kingpin angle. Moreover, despite the maximum deviation to be 1.98% at maximum caster angle, the deviation between ANN and response surface method are below 1% for the most of the angles. Thus, based on the validation, the model performs well.

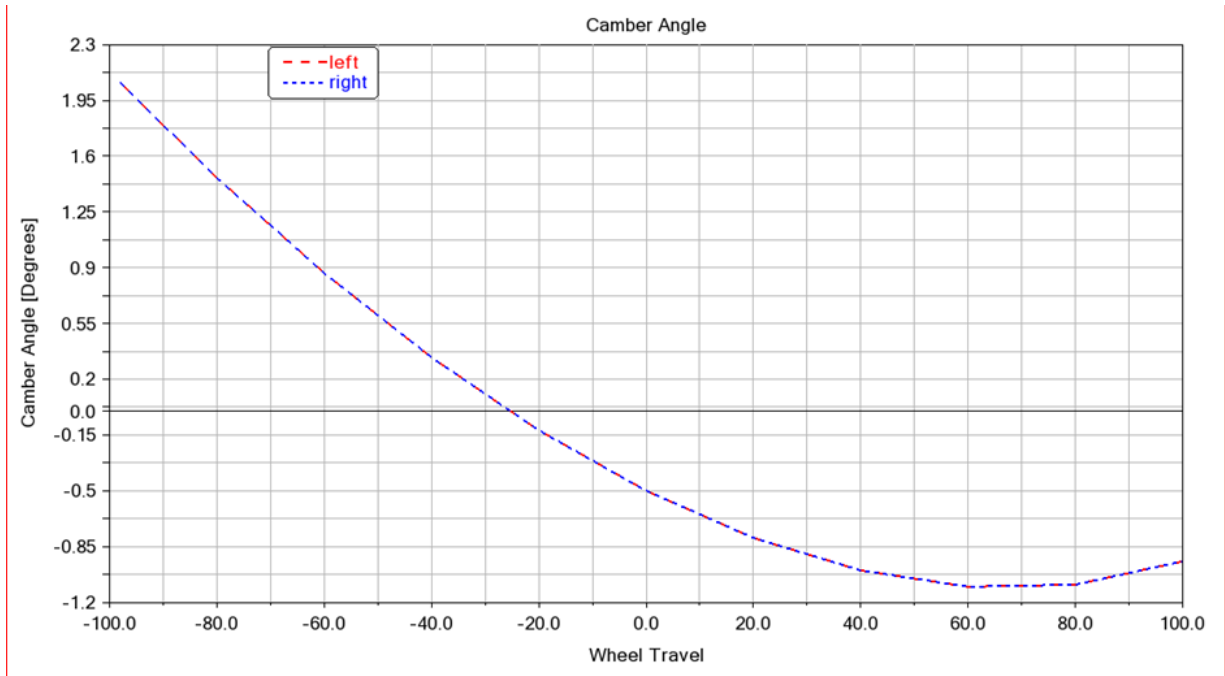
5.4. Comparison of Current Model with Optimized Model of MPSS

As shown in *Figure 5.18* and *Table 5.4*, for the current model, minimum and maximum camber angles are -1.09° and 2.07° , respectively. Whereas, for the optimized model, minimum and maximum camber angles are -1.27° to 2.02° , respectively. That is, the optimized model was able to reduce the maximum camber angle. However, the minimum camber angle increased.

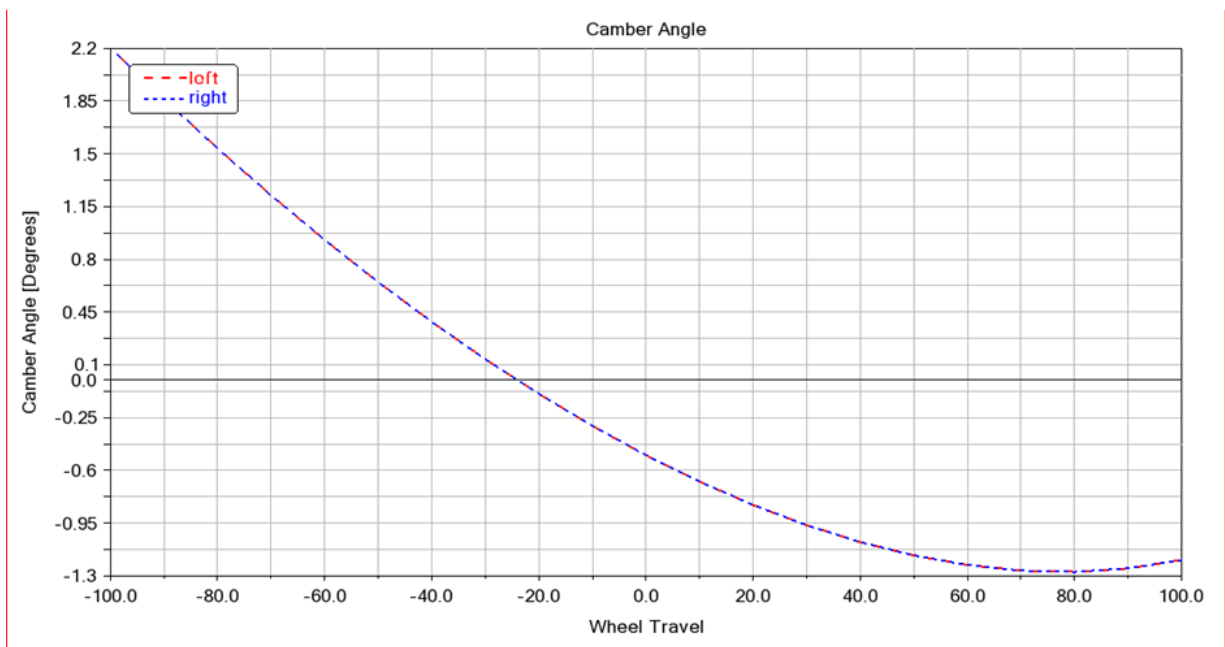
In addition, as shown in *Figure 5.19* and *Table 5.4*, for the current model, minimum and maximum caster angles are -2.4° and 1.6° , respectively. Whereas, for the optimized model, minimum and maximum caster angles are -1.82° and 1.22° , respectively. That is, the optimized model was able to reduce both the maximum and minimum caster angles.

Moreover, as shown in *Figure 5.20* and *Table 5.4*, for the current model, minimum and maximum kingpin inclination angles are 3.09° and 5.0° , respectively. Whereas, for the optimized model, minimum and maximum kingpin inclination angles are 2.96° to 4.95° , respectively. That is, the optimized model was able to reduce both the maximum and minimum kingpin inclination angles.

Furthermore, as shown in *Figure 5.21* and *Table 5.4*, for the current model, minimum and maximum toe angles are -3.09° and 2.25° , respectively. Whereas, for the optimized model, minimum and maximum toe angles are -2.86° and 2.22° , respectively. That is, the optimized model was able to reduce both the maximum and minimum toe angles.

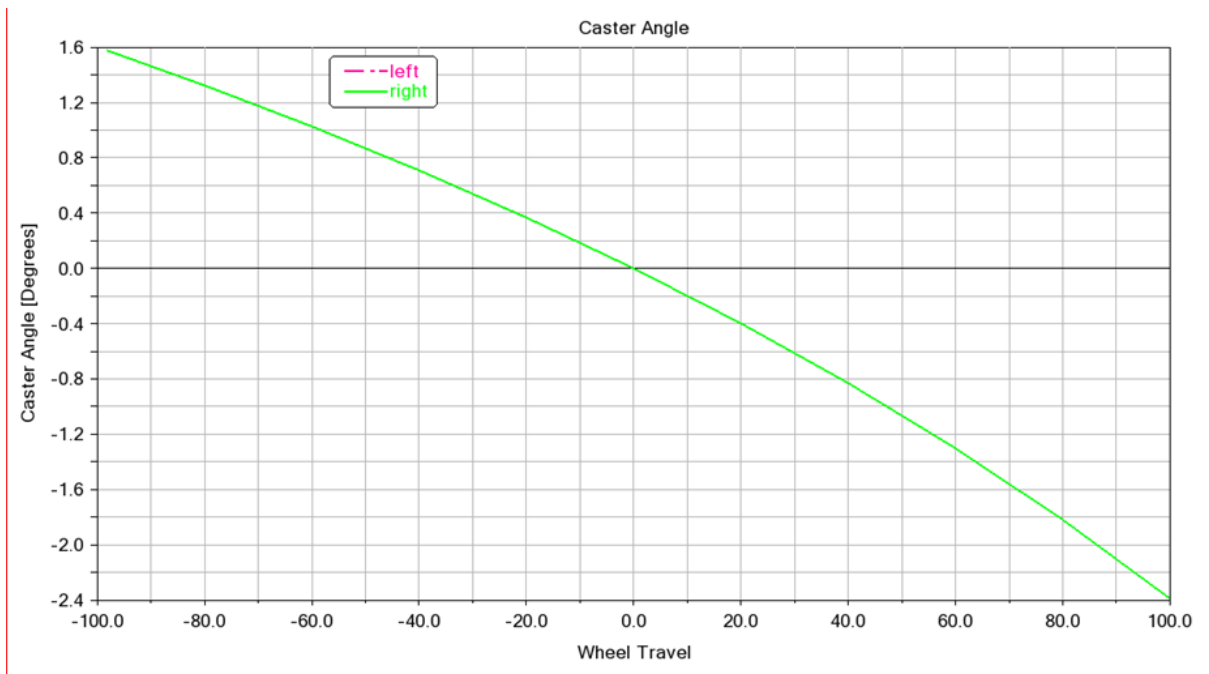


Current Model

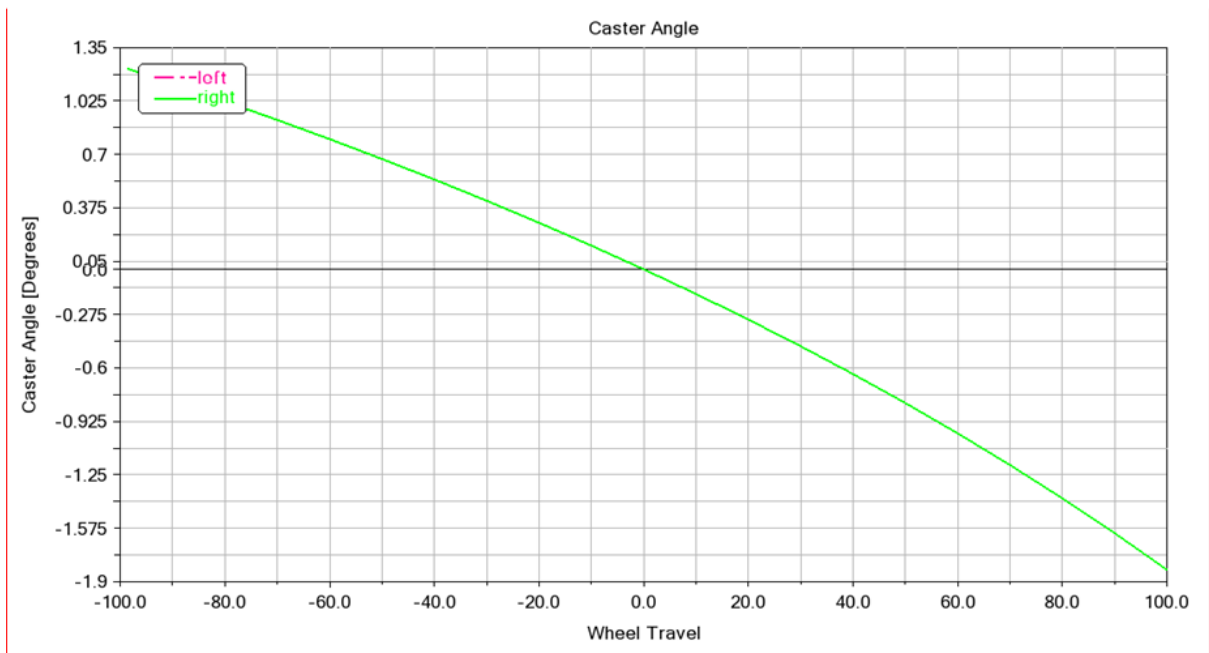


Optimized Model

Figure 5.18: Variation of Camber angle of MPSS

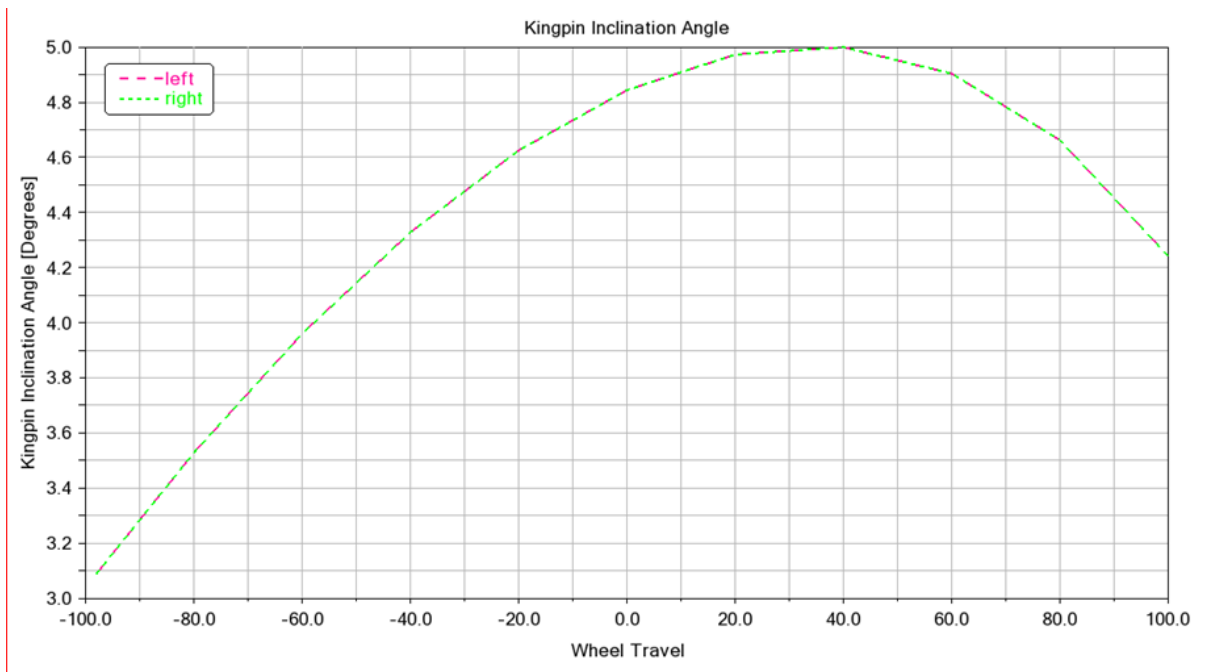


Current Model

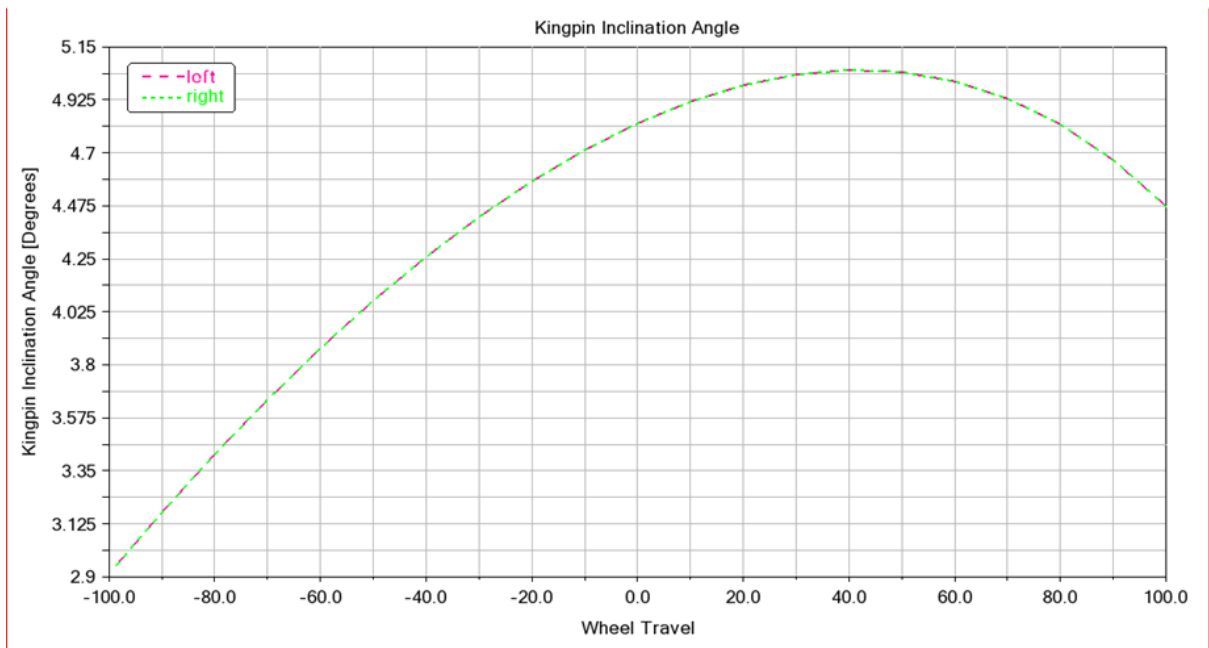


Optimized Model

Figure 5.19: Variation of Caster angle of MPSS

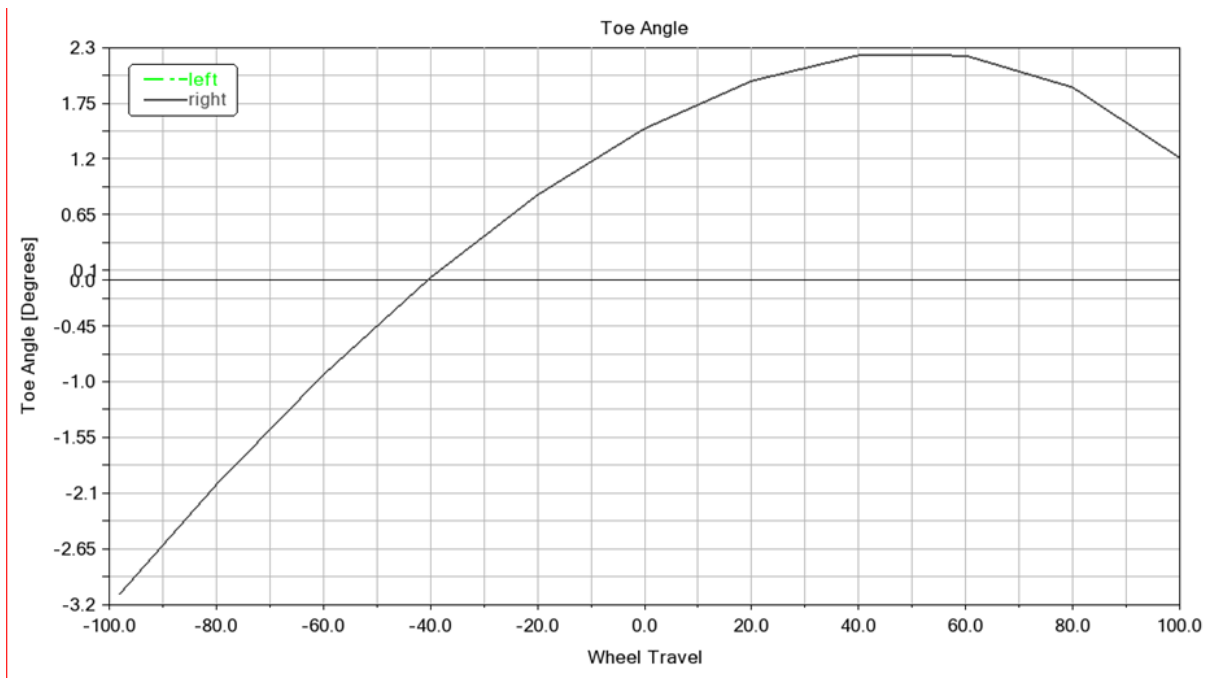


Current Model

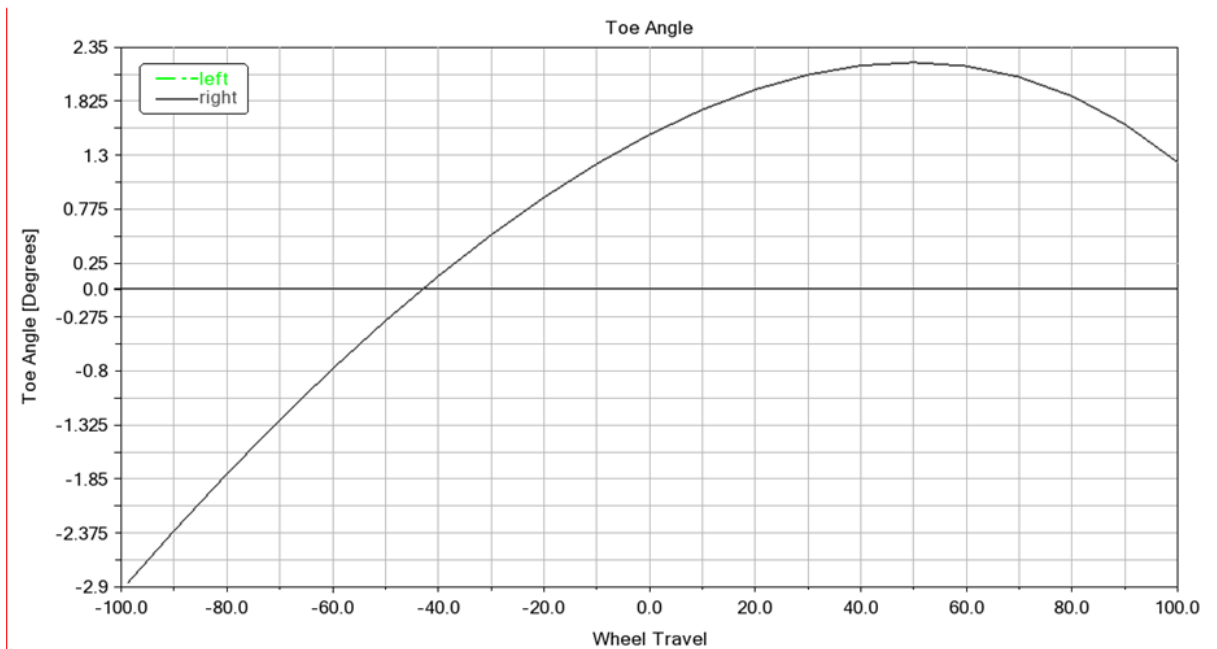


Optimized Model

Figure 5.20: Variation of Kingpin inclination angle of MPSS



Current Model



Optimized Model

Figure 5.21: Variation of Toe angle of MPSS

Table 5.4: Comparison of Current Model with Optimized Model of MPSS

	Current Model	Optimized Model	Reduced Wheel alignment (%)
Maximum Camber	2.07	2.02	2.4
Maximum Caster	1.6	1.22	23.75
Maximum Kingpin	5.0	4.95	1.0
Maximum Toe	2.25	2.22	1.33
Minimum Camber	-1.09	-1.27	-16.5
Minimum Caster	-2.4	-1.82	24.17
Minimum Kingpin	3.09	2.96	4.21
Minimum Toe	-3.09	-2.86	7.44

The optimized model was able to reduce the effect of the wheel alignment of seven parameters. 24.17%, 23.75%, 7.44%, 4.21%, 2.4%, 1.33%, and 1% of the minimum caster, maximum caster, minimum toe, minimum kingpin, maximum camber, maximum toe, maximum kingpin angles, respectively, of the wheel alignment are reduced. Though, the intended target of reducing wheel alignment is not achieved for minimum camber angle.

(Su et al., 2018) used unified weight combination method to reduce toe angle, caster angle, camber angle, and kingpin inclination angle by 55.03%, 12.97%, 6.9%, and 2.8% respectively. Here, in this thesis 7.44%, 24.17%, 2.4%, and 4.21% % of the toe, caster, camber, kingpin angles, respectively.

5.4.1. Negative Effects of the Optimization of MPSS

While performing the optimization for the MPSS, unfortunately, the optimized model was able to increase the effect of wheel alignment for minimum camber angle. However, the effect of the optimization on the minimum camber angle is within the standard recommended ranges of -2° to 2.5° (Su et al., 2018).

5.5. Summary of the Results

The kinematic analysis of the current model of MPSS is performed, and four principal performance parameters are obtained graphically. As such, minimum and maximum camber angles are -1.09° and 2.07° , respectively. In addition, minimum and maximum caster angles are -2.4° and 1.6° , respectively. Moreover, minimum and maximum kingpin inclination angles are 3.09° and 5.0° , respectively. Furthermore, minimum and maximum toe angles are -3.09° and 2.25° , respectively.

Then, three input parameters are selected to as optimization parameters. As such, considering front, rear, and outer control arm as optimization input parameters, the DESIGN EXPERT recommends to use -50.9, -89.69, and -77.36 for front control arm, rear control arm, and lower control arms Z positions, respectively. Hence, minimum and maximum optimized camber angles are -1.176° and 2.017° , respectively. In addition, minimum and maximum optimized caster angles are -2.158° and 1.376° , respectively. Moreover, minimum and maximum optimized kingpin inclination angles are 3.142° to 4.988° , respectively. Furthermore, minimum and maximum optimized toe angles are -2.948° and 2.196° , respectively.

Thus, after the results are crosschecked with ADAMS CAR software, minimum and maximum optimized camber angles are -1.27° and 2.02° , respectively. In addition, minimum and maximum optimized caster angles are -1.82° and 1.22° , respectively. Moreover, minimum and maximum optimized kingpin inclination angles are 2.96° and 4.95° , respectively. Furthermore, minimum and maximum optimized toe angles are -2.86° and 2.22° , respectively.

In addition, ANN model was established to check the model accuracy of DESIGN EXPERT. As such, 0.9762, 0.9802, 0.9979, and 0.9977 R^2 values are obtained for the camber, caster, Kingpin, and toe maximum angles, respectively, in response surface method. Whereas, 0.9838, 0.9996, 0.9986, and 0.9970 R^2 values are obtained for the camber, caster, Kingpin, and toe maximum angles, respectively, in ANN. In addition, 0.9982, 0.9927, 0.9879, and 0.9967 R^2 values are obtained for the camber, caster, Kingpin, and toe minimum angles, respectively, in response surface method. Whereas, 0.9976, 0.9996, 0.9775, and 0.9835 R^2 values are obtained

for the camber, caster, Kingpin, and toe minimum angles, respectively, in ANN. Hence, despite the maximum deviation to be 1.98% at maximum caster angle, the deviation between ANN and response surface method are below 1% for the most of the angles. Thus, based on the validation, the model performs well.

Finally, the comparison is made between the current model and the optimized model. As such, for the current model, minimum and maximum camber angles are -1.09° and 2.07° , respectively. Whereas, for the optimized model, minimum and maximum camber angles are -1.27° to 2.02° , respectively. In addition, for the current model, minimum and maximum caster angles are -2.4° and 1.6° , respectively. Whereas, for the optimized model, minimum and maximum caster angles are -1.82° and 1.22° , respectively. Moreover, for the current model, minimum and maximum kingpin inclination angles are 3.09° and 5.0° , respectively. Whereas, for the optimized model, minimum and maximum kingpin inclination angles are 2.96° to 4.95° , respectively. Furthermore, for the current model, minimum and maximum toe angles are -3.09° and 2.25° , respectively. Whereas, for the optimized model, minimum and maximum toe angles are -2.86° and 2.22° , respectively.

Thus, the optimized model was able to reduce the effect of the wheel alignment of seven parameters. 24.17%, 23.75%, 7.44%, 4.21%, 2.4%, 1.33%, and 1% of the minimum caster, maximum caster, minimum toe, minimum kingpin, maximum camber, maximum toe, maximum kingpin angles, respectively, of the wheel alignment are reduced. Though, the intended target of reducing wheel alignment is not achieved for minimum camber angle. However, the effect of the optimization on the minimum camber angle is within the standard recommended ranges of -2° to 2.5° (Su et al., 2018).

Therefore, by changing the z positions of front control arm, rear control arm, and outer control arm from -50, -100, and -75 to -50.9, -89.69, and -77.36, respectively, 24.17%, 23.75%, 7.44%, 4.21%, 2.4%, 1.33%, and 1% of the minimum caster, maximum caster, minimum toe, minimum kingpin, maximum camber, maximum toe, maximum kingpin angles, respectively, of the wheel alignment can be reduced.

CHAPTER SIX

CONCLUSION AND RECOMMENDATION

6.1. Summary

Even though MacPherson strut suspension system is widely used in the automotive industry due to its simplicity and cost-effectiveness, it is known to have issues with wheel alignment. And this can lead to uneven tire wear, reduced handling, and increased fuel consumption. Hence, this thesis made a kinematic analysis of the MPSSS through Adams Car software and perform the optimization using DESIGN EXPERT software.

The model is designed based on the design specifications acquired from Toyota Corolla 2022 model. This is done through the development of a kinematic model with the aim to unravel the intricate relationships between various suspension components and their effects on wheel alignment. As such, camber angle, toe angle, kingpin inclination angle and caster angle are identified as suspension parameters that contribute to misalignment. In addition, the optimization is performed by considering three input parameters of front control arm, lower control arm, and rear control arm of the MPSSS.

6.2. Conclusion

For the purpose of reducing the effects on wheel alignment, a kinematic analysis of the MPSS using Adams Car is performed, based on the design specifications acquired from Toyota Corolla 2022 model. Accordingly, for the current model, minimum and maximum camber angles are -1.09° and 2.07° , minimum and maximum caster angles are -2.4° and 1.6° , minimum and maximum kingpin inclination angles are 3.09° and 5.0° , and minimum and maximum toe angles are -3.09° and 2.25° were obtained through kinematic analysis.

In addition, by changing the z positions of front control arm, rear control arm, and outer control arm from -50, -100, and -75 to -50.9, -89.69, and -77.36, respectively, the optimized model was able to reduce the effect of the wheel alignment of seven parameters. As such, 24.17%, 23.75%, 7.44%, 4.21%, 2.4%, 1.33%, and 1% of the minimum caster, maximum caster,

minimum toe, minimum kingpin, maximum camber, maximum toe, maximum kingpin angles, respectively, of the wheel alignment are reduced. In doing so, even though the parameter is within the standard recommended ranges, unfortunately, the optimized model was able to increase the effect of wheel alignment for minimum camber angle. Hence, for the optimized model, minimum and maximum optimized camber angles are -1.27° and 2.02° , minimum and maximum optimized caster angles are -1.82° and 1.22° , minimum and maximum optimized kingpin inclination angles are 2.96° and 4.95° , and minimum and maximum optimized toe angles are -2.86° and 2.22° were obtained.

Therefore, by changing the z positions of front control arm, rear control arm, and outer control arm from -50, -100, and -75 to -50.9, -89.69, and -77.36, respectively, 24.17%, 23.75%, 7.44%, 4.21%, 2.4%, 1.33%, and 1% of the minimum caster, maximum caster, minimum toe, minimum kingpin, maximum camber, maximum toe, maximum kingpin angles, respectively, of the wheel alignment can be reduced.

6.3. Recommendation

A kinematic analysis and optimization of MPSSS is performed with the aim of reducing the effects on wheel alignment. This is done through Adams Car software and DESIGN Expert, by considering the front wheel part of the vehicle. However, the rear part of the vehicle may have the effect on the front wheel and its suspension. Thus, it is recommended that:

- The full vehicle analysis may be considered for further analysis and better analysis.

6.4. Future Works

Here, in this thesis, optimization of MPSSS is performed with the aim of reducing the effects on wheel alignment based on three control arm hardpoint using Adams Car software and DESIGN Expert. However, the effect of these optimization is on fuel economy is not explicitly studied here. Thus, the optimization of MPSSS could be done in the future with the aim of reducing fuel economy.

REFERENCES

- Abdelrahman, A. A., & Nabawy, A. E. (2020). Dynamic Finite Element Analysis of Flexible Double Wishbone Suspension Systems with Different Damping Mechanisms. *European Journal of Computational Mechanics*, 28(6), 573–604. <https://doi.org/10.13052/ejcm2642-2085.2862>
- Anumukonda, V. P., Singh, L., & Singh, A. (2021). Literature review on kinematic and dynamic analysis of McPherson Suspension. *International Research Journal of Engineering and Technology*, 8(5), 2225–2227.
- Berghian, A. B., Baneasa, C. B., & Topor, M. (2018). Mathematical Shaping of the Suspension System of Macpherson Type. *Annals of Faculty Engineering Hunedoara - International Journal of Engineering*, 16(2), 107–110.
- Chen, Y. C., Tsai, P. Y., & Lai, I. A. (2012). Kinematic Analysis of Roll Motion for a Strut / SLA Suspension System. *International Journal of Aerospace and Mechanical Engineering*, 6(5), 1009–1013.
- Cronin, D. L. (1981). MacPherson Strut Kinematics. *Mechanism and Machine Theory*, 16(6), 631–644.
- Deo, H. V. (2007). *Axiomatic Design of Customizable Automotive Suspension Systems*. Massachusetts Institute of Technology.
- Edmunds. (2022). *2022 Toyota Corolla - Specs & Features*.
- Fallah, M. S., Bhat, R., Xie, W. F., & Bhat, R. (2014). New model and simulation of Macpherson suspension system for ride control applications. *Vehicle System Dynamics: International Journal of Vehicle Mechanics and Mobility*, 47(2), 37–41. <https://doi.org/10.1080/00423110801956232>
- Liang, J., & Xin, L. (2012). Simulation analysis and optimization design of front suspension based on ADAMS. *Mechanika*, 18(3), 337–340.

- Mathworks. (2018). *Optimizing and Accelerating your MATLAB Code*. Videos and Webinars. <https://www.mathworks.com/videos/optimizing-and-accelerating-your-matlab-code-107711.html>
- Mathworks. (2024). *MATLAB Performance Optimization*. Help Center. <https://www.mathworks.com/help/comm/rtlsdrradio/ug/matlab-performance-improvements.html>
- Microsoft. (2024). *Define and solve a problem by using Solver*. Microsoft Support. <https://support.microsoft.com/en-us/office/define-and-solve-a-problem-by-using-solver-5d1a388f-079d-43ac-a7eb-f63e45925040>
- Nasiri, S., Sina, N., & Eslami, A. (2015). Multi-objective Optimisation of McPherson Strut Suspension Mechanism Kinematics using Random Search Method. *Indian Journal of Science and Technology*, 8(16). <https://doi.org/10.17485/ijst/2015/v8i16/62548>
- Patil, A. A. (2016). Mathematical Model for Kinematic Analysis of McPherson Strut Suspension. *SAE Technical Papers*, 28. <https://doi.org/10.4271/2016-28-0184>. Copyright
- Quick Trick. (2023). *What is wheel allignment*.
- Reddy, K. V., Kodati, M., Chatra, K., & Bandyopadhyay, S. (2016). A comprehensive kinematic analysis of the double wishbone and MacPherson strut suspension systems. *Mechanism and Machine Theory*, 105, 441–470. <https://doi.org/10.1016/j.mechmachtheory.2016.06.001>
- Ritme. (2024). *Design-Expert*. <https://ritme.com/en/software/design-expert/>
- Saraf, S., & Satpute, N. V. (2017). Optimization & FEA of Inner Tie Rod for Light Motor Vehicles (LMV). *REST Journal on Emerging Trends in Modelling and Manufacturing*, 3(3), 63–68.
- Saviotto, G. (2020). *Simulation of MacPherson strut side force reduction and influence in the dynamics of vehicle in Adams*. Politecnico Di Torino.
- Science Plus Group. (2024). *DESIGN EXPERT*.

- Segal, T. (2024). *Linear Regression Excel: Step-by-Step Instructions*. Investopedia. <https://www.investopedia.com/ask/answers/062215/how-can-i-run-linear-and-multiple-regressions-excel.asp>
- Shi, Q., Peng, C., Chen, Y., He, J., Li, P., & Chen, J. (2019). Robust kinematics design of MacPherson suspension based on a double-loop multi-objective particle swarm optimization algorithm. *Journal of Automobile Engineering*. <https://doi.org/10.1177/0954407018821556>
- Shinde, S. D., Maheshwari, S., & Kumar, S. (2018). Literature review on analysis of various Components of McPherson suspension. *Materials Today: Proceedings*, 5(9), 19102–19108. <https://doi.org/10.1016/j.matpr.2018.06.263>
- Shojaeefard, M. H., Khalkhali, A., & Yarmohammadisatri, S. (2017). An efficient sensitivity analysis method for modified geometry of Macpherson suspension based on Pearson correlation coefficient. *Vehicle System Dynamics - International Journal of Vehicle Mechanics and Mobility*, 1–26. <https://doi.org/10.1080/00423114.2017.1283046>
- Stat-Ease Inc. (2024). *DESIGN EXPERT*. <https://www.statease.com/software/design-expert/>
- Su, Z., Xu, F., Hua, L., Chen, H., Wu, K., & Zhang, S. (2018). Design optimization of minivan MacPherson-strut suspension system based on weighting combination method and neighborhood cultivation genetic algorithm. *Journal of Automobile Engineering*. <https://doi.org/10.1177/0954407018789303>
- The Engineers Post. (2020). *Different Types of Suspension Systems*.
- Wang, J., Qu, X., Han, J., & Xing, S. (2016). The K & C Analysis and Optimization of Macpherson Suspension. *International Journal of Applied Science and Mathematics*, 3(6), 205–210.

APPENDIX

Appendix I – ADAMS Model of Procedures

Create Arm Geometry

Arm Name	Control_Arm
General Part	._New_MacPherson_Suspension.gel_Control_arm
Coordinate Reference #1	._New_MacPherson_Suspension.ground.hpl_arm_outer
Coordinate Reference #2	._New_MacPherson_Suspension.ground.hpl_arm_front
Coordinate Reference #3	._New_MacPherson_Suspension.ground.hpl_arm_rear
Thickness	10
Color	red

☐ Calculate Mass Properties of General Part

Density	☒ Material ☐ User Entered
Material Type	steel

Appendix 1: Creating a geometry for control arm in Adams Car

General Part Wizard

General Part Name	Wheel_Carrier
Type	☒ left ☐ right ☐ single
Geometry Type	Arm

Coordinate Reference #1	._New_MacPherson_Suspension.ground.hpl_Wheel_Center
Coordinate Reference #2	._New_MacPherson_Suspension.ground.hpl_arm_outer
Coordinate Reference #3	._New_MacPherson_Suspension.ground.hpl_Strut_Lower
Thickness	10
Color	dark gray

Density ☒ Material ☐ User Entered

Material Type materials.aluminum

Link Name	Carrier_Link
General Part	._New_MacPherson_Suspension.gel_Wheel_Carrier
Coordinate Reference #1	._New_MacPherson_Suspension.ground.hpl_Strut_Lower
Coordinate Reference #2	._New_MacPherson_Suspension.ground.hpl_Tie_rod_Outer
Radius	10
Color	dark gray

☐ Calculate Mass Properties of General Part

Density	☒ Material ☐ User Entered
Material Type	steel

Appendix 2: Creating a geometry for wheel carrier in Adams Car

General Part Name	Tierod
Type	☒ left ☐ right ☐ single
Geometry Type	Link
Coordinate Reference #1	._New_MacPherson_Suspension.ground.hpl_Tie_rod_Outer
Coordinate Reference #2	._New_MacPherson_Suspension.ground.hpl_Tie_rod_inner
Radius	10

Appendix 3: Creating a geometry for tie rod in Adams Car

Ad Create Damper		×
Damper Name	Damper	
I Part	._New_MacPherson_Suspension.gel_Wheel_Carrier	
J Part	._New_MacPherson_Suspension.gel_Strut	
I Coordinate Reference	._New_MacPherson_Suspension.ground.hpl_Strut_Lower	
J Coordinate Reference	._New_MacPherson_Suspension.ground.hpl_Strut_Upper	
Property File	mdids://acar_shared/dampers.tbl/mdi_0001.dpr	
Color	maize	
Damper Diameter	15	

Appendix 4: Creating a geometry for damper in Adams Car

Ad Create Spring		×
Spring Name	Main_Spring	
I Part	._New_MacPherson_Suspension.gel_Wheel_Carrier	
J Part	._New_MacPherson_Suspension.gel_Strut	
I Coordinate Reference	._New_MacPherson_Suspension.ground.hpl_Spring_Lower	
J Coordinate Reference	._New_MacPherson_Suspension.ground.hpl_Strut_Upper	
Installed Length	304.1381265149 DM(iCoord,jCoord)	
Property File	mdids://acar_shared/springs.tbl/mdi_0001.spr	
Color	white	
Spring Diameter	30	
Number of Coils	5	

Appendix 5: Creating a geometry for spring in Adams Car

Create General Part

General Part:

Type: ☒ left ☐ right ☐ single

Location Dependency:

Location values:

Orientation Dependency:

Orient using: ☒ Euler Angles ☐ Direction Vectors

Euler Angles:

X Vector:

Z Vector:

Mass: Sprung Mass %:

Ixx: ☐ Off-Diagonal Terms

Iyy: Izz:

CM Location Relative to Part:

Density: ☒ Material ☐ User Entered

Material Type: 

Appendix 6: Creating a geometry for strut in Adams Car

Set Toe & Camber Values

	Left Side	Right Side
Toe Angles	1.5	1.5
Camber Angles	-0.5	-0.5

Appendix 7: Creating suspension parameters in Adams Car

Ad

Create Construction Frame

×

Construction Frame

Hub_Bearing

Type

☒ left
☐ right
☐ single

Location Dependency

Delta location from coordinate

Coordinate Reference

._New_MacPherson_Suspension.ground.hpl_Wheel_Center

Location

0,0,0

Location in

☒ local
☐ global

Orientation Dependency

Toe/Camber

Variable Type

Parameter Variables

Toe Parameter Variable

._New_MacPherson_Suspension.pvl_toe_angle

Camber Parameter Variable

._New_MacPherson_Suspension.pvl_camber_angle

Appendix 8: Hub bearing construction frame for creating hub geometry

Ad

Create General Part

×

General Part

Hub

Type

☒ left
☐ right
☐ single

Location Dependency

Delta location from coordinate

Coordinate Reference

._New_MacPherson_Suspension.ground.cfl_Hub_Bearing

Location

0,0,0

Location in

☒ local
☐ global

Orientation Dependency

Delta orientation from coordinate

Construction Frame

._New_MacPherson_Suspension.ground.cfl_Hub_Bearing

Orientation

0,0,0

Appendix 9: Hub general part for creating hub geometry

Create Cylinder Geometry

Cylinder Name	Hub
General Part	._New_MacPherson_Suspension.gel_Hub
Construction Frame	._New_MacPherson_Suspension.ground.cfl_Hub_Bearing
Radius	30
Length In Positive Z	30
Length In Negative Z	0

Appendix 10: Creating hub geometry in Adams Car software

Communicators Info

Model Names: .__MDI_SUSPENSION_TESTRIG

Type: ☐ Input ☐ Output ☒ All

Entity: ☒ Many ☐ All

☒ array	☐ motion
☐ bushing	☒ mount
☐ contact geometry	☐ orientation
☐ differential equation	☐ parameter integer
☐ force	☐ parameter real
☐ joint for motion	☐ part
☐ joint	☐ solver variable
☒ location	☐ spline
☐ marker	

Create Output Communicator

Output Communicator Name: Suspension_Upright

Matching Name(s):

Type: ☒ left ☐ right ☐ single

Entity: mount

To Minor Role: inherit

To Subsystem Index: inherit

Part Name: ._New_MacPherson_Suspension.gel_Wheel_Carrier

Create Output Communicator

Output Communicator Name: Wheel_Center

Matching Name(s):

Type: ☒ left ☐ right ☐ single

Entity: location

To Minor Role: inherit

To Subsystem Index: inherit

Coordinate Reference Name: ._New_MacPherson_Suspension.ground.hpl_Wheel_Center

Creating Communicator information

Create Output Communicator

Output Communicator Name: Suspension_Mount

Matching Name(s):

Type: ☒ left ☐ right ☐ single

Entity: mount

To Minor Role: inherit

To Subsystem Index: inherit

Part Name: ._New_MacPherson_Suspension.gel_Hub

Creating Suspension mount

Creating Wheel center location

Test Communicators

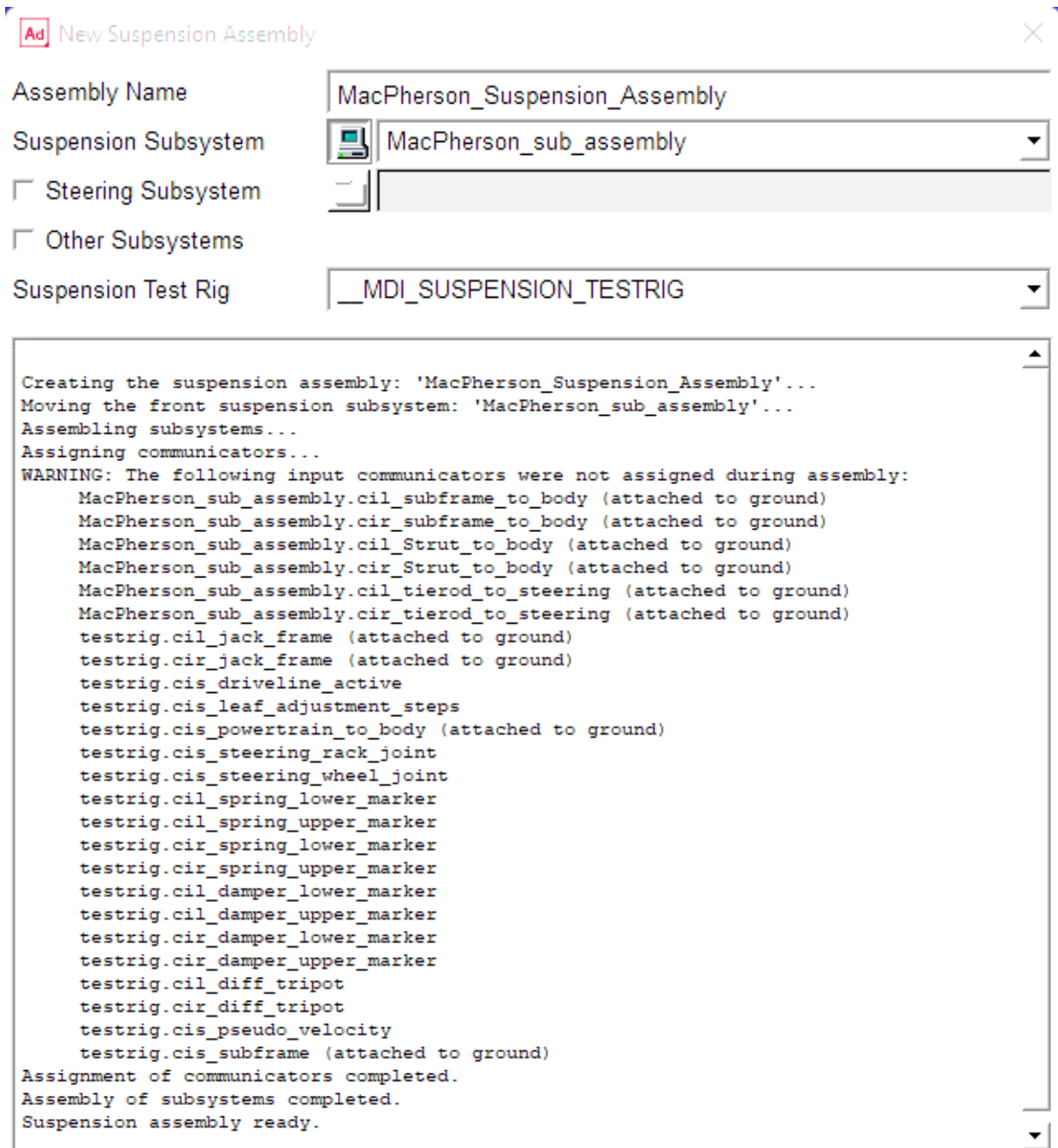
Model Names: .__MDI_SUSPENSION_TESTRIG, ._New_MacPherson_Suspension

Minor Roles: any, front

☒ Clear Information Window

Creating Communicator test

Appendix 11: Creating a communicator for suspension system in Adams Car



Appendix 12: Creating assembly for suspension system in Adams Car

Ad

Toggle Kinematic Mode

×

Subsystem

MacPherson_Suspension_Asse

▼

Current Mode

☒ Kinematic
 ☐ Compliant

Ad

Suspension Analysis: Parallel Travel

×

Suspension Assembly

MacPherson_Suspension_Assembly

▼

Assembly Variant

default

▼

↔

Output Prefix

NMPS_Am_01

Number of Steps

20

Mode of Simulation

interactive

▼

Vertical Setup Mode

Wheel Center

▼

Travel Relative To

Wheel Center

▼

Control Mode

☒ Absolute
 ☐ Relative

Fixed Steer Position

Steering Input

☒ Angle
 ☐ Length

Coordinate System

Vehicle

▼



Bump Travel

100

Rebound Travel

-100

☒ Create Event Log File

Ad

Suspension Analysis: Setup Parameters

×

Suspension Assembly

MacPherson_Suspension_Assembly

▼

Suspension Settings

Actuation Type

☒ Contact Patch
 ☐ Wheel Center

Tire Model

User Defined

▼

Tire Unloaded Radius

260.0

Tire Loaded Radius

Tire Stiffness

235.0

Tire Property File

RIGID_WHEEL

Wheel Mass

1.0

Cm Offset

0.0

Dual Wheels

☐ Active
 ☒ Inactive

Dual Wheel Offset

300.0

Dual Cm Offset

0.0

Vehicle Parameters

Sprung Mass

300.0

CG Height

306.0

Wheelbase

1560.0

Drive Ratio (% Front)

◀

▶

0

Brake Ratio (% Front)

◀

▶

75

Appendix 13: Creating kinematic analysis for suspension system in Adams Car

Appendix II – ADAMS CAR Result for Optimizations

FCA	RCA	OCA	Maximum				Minimum			
			Camber	Caster	KPI	Toe	Camber	Caster	KPI	Toe
-100	-150	-125	1.9	1.46	4.58	2.35	-1.04	-2.2	2.9	-3.5
-100	-150	-100	1.44	1.5	4.65	2.96	-0.7	-2.35	3.1	-4.6
-100	-150	-75	1.1	1.54	4.88	3.66	-0.51	-2.6	2.2	-5.6
-100	-150	-50	2.57	1.58	5.25	4.64	-0.49	-2.87	0.9	-6.6
-100	-150	-25	4.53	1.59	5.81	5.8	-0.61	-3.24	0.81	-7.63
-100	-125	-125	2.01	0.74	4.65	2.26	-1.31	-1.06	2.69	-3.09
-100	-125	-100	1.57	0.75	4.69	2.87	-0.85	-1.15	3.36	-4.11
-100	-125	-75	1.07	0.76	4.84	3.62	-0.59	-1.25	2.74	-5.2
-100	-125	-50	1.78	0.79	5.16	4.63	-0.5	-1.4	1.6	-6.21
-100	-125	-25	3.44	0.791	5.66	6.25	-0.56	-1.57	0.1	-7.25
-100	-100	-125	2.2	0	4.85	2.25	-1.65	0	2.46	-2.7
-100	-100	-100	1.73	0	4.75	2.84	-1.08	0	3.07	-3.7
-100	-100	-75	1.23	0	4.85	3.66	-0.71	0	3.2	-4.8
-100	-100	-50	1.07	0	5.1	4.63	-0.5	0	2.2	-5.69
-100	-100	-25	2.59	0	5.55	6	-0.51	0	0.63	-6.81
-100	-75	-125	2.41	1.05	5	2.21	-1.96	-0.73	2.24	-2.24
-100	-75	-100	1.93	1.13	4.86	2.79	-1.34	-0.75	2.9	-3.3
-100	-75	-75	1.41	1.23	4.87	3.64	-0.86	-0.77	3.58	-4.3
-100	-75	-50	0.88	1.34	5.05	4.73	-0.59	-0.8	2.73	-5.38
-100	-75	-25	1.88	1.48	5.44	6.25	-0.5	-0.81	1.51	-6.38
-100	-50	-125	2.65	2.06	5.2	2.2	-2.24	-1.46	1.99	-1.72
-100	-50	-100	2.15	2.22	4.98	2.8	-1.6	-1.5	2.64	-2.8
-100	-50	-75	1.6	2.41	4.93	3.72	-1.06	-1.55	3.33	-3.85
-100	-50	-50	1.08	2.62	5.06	4.93	-0.7	-1.58	3.1	-4.9
-100	-50	-25	1.23	2.89	5.34	6.4	-0.5	-1.58	2	-5.93
-75	-150	-125	2.28	2.2	4.73	1.96	-1.31	-3.19	2.5	-2.62
-75	-150	-100	1.83	2.24	4.7	2.34	-0.83	-3.5	3.19	-3.7
-75	-150	-75	1.4	2.29	4.84	3	-0.6	-3.8	2.7	-4.8
-75	-150	-50	1.94	2.38	5.15	3.69	-0.5	-4.17	1.59	-5.85
-75	-150	-25	3.79	2.42	5.61	4.63	-0.5	-4.71	0	-6.94
-75	-125	-125	2.4	1.48	4.89	1.91	-1.7	-2.11	2.33	-2.23
-75	-125	-100	1.97	1.5	4.78	2.29	-1.08	-2.3	2.99	-3.3
-75	-125	-75	1.5	1.54	4.84	2.91	-0.7	-2.5	3.29	-4.4
-75	-125	-50	1.2	1.59	5.08	3.59	-0.52	-2.71	2.28	-5.5
-75	-125	-25	2.74	1.65	5.47	4.63	-0.5	-3.04	0.9	-6.56

-75	-100	-125	2.53	0.73	5.1	1.86	-2.04	-1.03	2	-1.8
-75	-100	-100	2.16	0.76	4.9	2.23	-1.38	-1.12	2.78	-2.89
-75	-100	-75	1.64	0.78	4.9	2.83	-0.87	-1.21	3.48	-4
-75	-100	-50	1.15	0.79	5.05	3.6	-0.6	-1.33	2.88	-5.1
-75	-100	-25	1.94	0.81	5.4	4.63	-0.5	-1.48	1.7	-6.13
-75	-75	-125	2.72	0	5.31	1.77	-2.35	0	1.97	-1.32
-75	-75	-100	2.34	0	5.06	2.2	-1.69	0	2.56	-2.5
-75	-75	-75	1.81	0	4.95	2.79	-1.09	0	3.25	-3.56
-75	-75	-50	1.3	0	5.05	3.63	-0.77	0	3.38	-4.62
-75	-75	-25	1.24	0	5.33	4.63	-0.51	0	2.3	-5.69
-75	-50	-125	2.91	1.02	5.6	1.71	-2.63	-0.73	1.72	-0.83
-75	-50	-100	2.53	1.1	5.23	2.14	-2.02	-0.75	2.3	-2
-75	-50	-75	2.03	1.18	5.08	2.78	-1.4	-0.8	2.99	-3.1
-75	-50	-50	1.48	1.29	5.09	3.64	-0.9	-0.8	3.72	-4.2
-75	-50	-25	0.94	1.44	5.3	4.87	-0.6	-0.82	2.83	-5.2
-50	-150	-125	2.6	2.9	4.9	1.65	-1.64	-4.2	2.16	-1.64
-50	-150	-100	2.2	3.02	4.8	1.95	-1.03	-4.6	2.78	-2.78
-50	-150	-75	1.8	3.11	4.85	2.36	-0.7	-5	3.17	-3.9
-50	-150	-50	1.4	3.21	5.06	3	-0.51	-5.5	2.16	-5
-50	-150	-25	3.06	3.3	5.42	3.75	-0.5	-6.1	0.7	-6.14
-50	-125	-125	2.71	2.19	5.09	1.63	-2.05	-3.09	2.02	-1.24
-50	-125	-100	2.38	2.29	4.94	1.89	-1.36	-3.36	2.6	-2.39
-50	-125	-75	1.93	2.34	4.9	2.3	-0.85	-3.67	3.29	-3.5
-50	-125	-50	1.47	2.4	5.05	2.92	-0.6	-3.99	2.86	-4.63
-50	-125	-25	2.11	2.5	5.37	3.7	-0.5	-4.5	1.6	-5.8
-50	-100	-125	2.83	1.46	5.31	1.58	-2.39	-2.02	1.83	-0.82
-50	-100	-100	2.48	1.51	5.1	1.81	-1.74	-2.2	2.4	-2
-50	-100	-75	2.07	1.6	4.998	2.27	-1.113	-2.4	3.09	-3.09
-50	-100	-50	1.58	1.6	5.06	2.88	-0.72	-2.61	3.44	-4.24
-50	-100	-25	1.3	1.63	5.31	3.1	-0.51	-2.9	2.68	-5.4
-50	-75	-125	3	0.72	5.7	1.53	-2.72	-1.01	1.67	-0.39
-50	-75	-100	2.65	0.76	5.3	1.77	-2.1	-1.1	2.23	-1.5
-50	-75	-75	2.22	0.8	5.11	2.18	-1.4	-1.18	2.86	-2.7
-50	-75	-50	1.71	0.81	5.12	2.78	-0.9	-1.28	3.59	-3.8
-50	-75	-25	1.21	0.83	5.28	3.64	-0.6	-1.4	3	-4.96
-50	-50	-125	3.18	0	5.99	1.5	-3.02	0	1.5	0.04
-50	-50	-100	2.82	0	5.56	1.71	-2.27	0	2.02	-1.02
-50	-50	-75	2.45	0	5.29	2.11	-1.78	0	2.66	-2.19
-50	-50	-50	1.9	0	5.2	2.75	-1.12	0	3.35	-3.36

-50	-50	-25	1.38	0	5.3	3.67	-0.72	0	3.85	-4.5
-25	-150	-125	2.91	3.61	5.07	1.5	-1.99	-5.19	1.79	-1.12
-25	-150	-100	2.63	3.8	4.94	1.67	-1.3	-5.58	2.37	-1.8
-25	-150	-75	2.17	3.88	4.93	1.98	-0.83	-6.09	3.05	-2.9
-25	-150	-50	1.87	4	5.06	2.4	-0.6	-6.62	2.63	-4.09
-25	-150	-25	2.39	4.01	5.34	2.99	-0.5	-7.81	1.35	-5.2
-25	-125	-125	2.96	2.87	5.32	1.51	-2.38	-4.06	1.68	-1.03
-25	-125	-100	2.71	3.02	5.12	1.61	-1.71	-4.4	2.25	-1.36
-25	-125	-75	2.34	3.16	5.02	1.9	-1.07	-4.8	2.84	-2.53
-25	-125	-50	1.9	3.23	5.07	2.3	-0.7	-5.21	3.35	-3.7
-25	-125	-25	1.5	3.36	5.29	2.9	-0.54	-5.78	2.27	-4.9
-25	-100	-125	3.15	2.15	5.7	1.5	-2.75	-3.01	1.5	-0.93
-25	-100	-100	2.79	2.25	5.33	1.6	-2.11	-3.23	2.09	-0.95
-25	-100	-75	2.46	2.4	5.14	1.84	-1.4	-3.5	2.7	-2.13
-25	-100	-50	2.03	2.44	5.15	2.26	-0.87	-3.82	3.4	-3.3
-25	-100	-25	1.55	2.58	5.28	2.85	-0.59	-4.2	3.01	-4.4
-25	-75	-125	3.27	1.43	6.05	1.51	-3.1	-2.01	1.36	-0.85
-25	-75	-100	2.95	1.5	5.63	1.53	-2.5	-2.12	1.9	-0.51
-25	-75	-75	2.6	1.58	5.35	1.77	-1.8	-2.3	2.52	-1.67
-25	-75	-50	2.16	1.64	5.24	2.18	-1.1	-2.49	3.19	-2.87
-25	-75	-25	1.68	1.7	5.29	2.83	-0.4	-2.75	3.64	-4.05
-25	-50	-125	3.42	0.71	6.33	1.56	-3.36	-1	1.19	-0.8
-25	-50	-100	3.1	0.75	5.98	1.5	-2.77	-1.05	1.74	-0.2
-25	-50	-75	2.75	0.78	5.59	1.7	-2.2	-1.13	2.33	-1.18
-25	-50	-50	2.33	0.82	5.37	2.09	-1.46	-1.2	2.97	-2.38
-25	-50	-25	1.83	0.85	5.36	2.74	-0.9	-1.34	3.72	-3.6
0	-150	-125	3.23	4.3	5.3	1.51	-2.33	-6.12	1.4	-1.83
0	-150	-100	2.95	4.47	5.09	1.5	-1.6	-6.62	1.98	-1.4
0	-150	-75	2.6	4.67	5.03	1.66	-1	-7.25	2.6	-1.86
0	-150	-50	2.22	4.75	5.08	1.97	-0.68	-7.85	3.05	-3
0	-150	-25	1.81	4.87	5.3	2.38	-0.5	-8.62	1.89	-4.2
0	-125	-125	3.32	3.56	5.61	1.57	-2.73	-5.1	1.31	-1.76
0	-125	-100	3.02	3.76	5.3	1.5	-2.1	-5.42	1.87	-1.28
0	-125	-75	2.7	3.9	5.16	1.62	-1.35	-5.9	2.47	-1.48
0	-125	-50	2.32	4	5.15	1.89	-0.86	-6.31	3.16	-2.65
0	-125	-25	1.89	4.09	5.3	2.34	-0.59	-7	2.77	-3.82
0	-100	-125	3.42	2.82	6	1.62	-3.1	-4	1.2	-1.68
0	-100	-100	3.1	3	5.63	1.5	-2.45	-4.3	1.72	-1.18
0	-100	-75	2.79	3.1	5.36	1.58	-1.8	-4.56	2.31	-1.06

0	-100	-50	2.45	3.26	5.25	1.84	-1.1	-4.99	2.98	-2.3
0	-100	-25	2	3.4	5.3	2.25	-0.71	-5.49	3.54	-3.46
0	-75	-125	3.56	2.12	6.36	1.76	-3.38	-2.94	1.03	-1.61
0	-75	-100	3.24	2.25	6	1.5	-2.9	-3.16	1.59	-1.1
0	-75	-75	2.92	2.35	5.61	1.54	-2.21	-3.35	2.19	-0.63
0	-75	-50	2.54	2.44	5.39	1.75	-1.46	-3.67	2.81	-1.81
0	-75	-25	2.11	2.56	5.38	2.17	-0.9	-4.03	3.53	-3.06
0	-50	-125	3.7	1.41	6.68	1.96	-3.7	-1.93	0.9	-1.52
0	-50	-100	3.38	1.46	6.35	1.58	-3.2	-2.07	1.4	-1.01
0	-50	-75	3.03	1.54	5.93	1.51	-2.58	-2.227	2.01	-0.42
0	-50	-50	2.69	1.64	5.6	1.69	-1.9	-2.39	2.6	-1.34
0	-50	-25	2.27	1.71	5.49	2.1	-1.16	-2.62	3.3	-2.61

Appendix 14: ADAMS CAR result of the new models