

Development of Suspension System Model for Designing Anti-Roll Bar

by

Heong Joo Yoo

A thesis

presented to the University of Waterloo

in fulfillment of the

thesis requirement for the degree of

Master of Applied Science

in

Mechanical and Mechatronics Engineering

Waterloo, Ontario, Canada, 2024

© Heong Joo Yoo 2024

Author's Declaration

I hereby declare that I am the sole author of this thesis. This is a true copy of the thesis, including any required final revisions, as accepted by my examiners.

I understand that my thesis may be made electronically available to the public.

Abstract

This thesis presents the development and validation of a comprehensive full-vehicle Adams Car model for the University of Waterloo Alternative Fuels Team LYRIQ. The primary objective of this research is to optimize the suspension system, particularly the anti-roll bar, to accommodate the increased weight and altered weight distribution resulting from the integration of an all-wheel-drive electric powertrain. The full-vehicle model facilitates a detailed analysis of the vehicle's dynamic behavior under various conditions, enabling rapid prototyping and evaluation of suspension parameters.

Through rigorous simulation tests, including fishhook and double lane change maneuvers, the study identified key areas where the UWAFS LYRIQ exhibited higher body roll and reduced yaw rate responsiveness compared to the stock LYRIQ model. These findings underscored the necessity of optimizing the ARB stiffness to enhance the vehicle's dynamic performance. Adjustments to the ARB stiffness, achieved by shortening the moment arm, successfully reduced body roll, bringing it closer to the levels observed in the stock model. However, improvements in lateral acceleration and yaw rate were less pronounced, highlighting the significant role of overall vehicle weight in influencing agility and handling characteristics.

The research provides valuable insights into the impact of increased vehicle weight and altered weight distribution on handling performance, emphasizing the importance of fine-tuning suspension components in electrified vehicles. Future work may focus on further optimization of suspension parameters and the exploration of advanced materials and technologies to enhance vehicle performance while maintaining safety and comfort. This study contributes significantly to the field of vehicle dynamics, particularly in the context of electric vehicle development, and lays the groundwork for ongoing advancements in this rapidly evolving domain.

Acknowledgements

I would like to extend my heartfelt thanks to my supervisor, Dr. Roydon Fraser, for his unwavering passion and dedication in supporting both myself and the University of Waterloo Alternative Fuels Team. His commitment to providing us with unforgettable experiences and invaluable learnings has significantly contributed to my personal and professional growth.

I am also deeply grateful to my teammates on the University of Waterloo Alternative Fuels Team. Without your camaraderie, those long nights and stressful days would not have been as memorable or rewarding.

Additionally, I would like to express my sincere gratitude to the organizers and mentors, particularly Jordie McAuley, for their relentless support in fostering this exceptional program. Your guidance has been instrumental in helping students like myself develop into the next generation of industry leaders.

Dedication

To my family.

Table of Contents

Author's Declaration	ii
Abstract	iii
Acknowledgements	iv
Dedication	v
List of Figures	ix
List of Tables.....	xiii
List of Abbreviations.....	xiv
Chapter 1 Introduction.....	1
1.1 Research Motivation.....	1
1.1.1 Electrification of Powertrain System.....	1
1.2 Research Objectives	3
Chapter 2 Literature Review	4
2.1 Ride Comfort.....	4
2.2 Handling and Stability.....	5
2.3 Vehicle Suspension Development.....	8
2.4 Definition of Suspension	9
2.4.1 Anti-Roll Bar.....	10
2.5 Vehicle Parameters.....	12
2.5.1 Roll Center and Roll Axis	13
2.5.2 Sprung and Unsprung Mass.....	14
2.5.3 Camber Angle.....	14
2.5.4 Toe Angle	15
2.6 Point Mass Model.....	16

2.7 Bicycle Model	17
2.8 Roll Dynamic Model	19
2.9 9 Degree of Freedom Model.....	22
2.10 Adams Car Model	24
Chapter 3 Model Development	26
3.1 Vehicle Features	26
3.2 Motor	28
3.3 Bushing.....	30
3.4 Spring and Damper.....	33
3.5 Bump stop and Rebound stop.....	35
3.6 Steering System.....	36
3.7 Front and Rear Anti-Roll Bar	37
3.8 Front Suspension	39
3.9 Rear Suspension	40
3.10 Chassis.....	42
3.11 Tire	43
3.12 Full Vehicle	45
Chapter 4 Full Vehicle Handling Simulation	48
4.1 Handling Maneuvers	48
4.1.1 Fishhook	48
4.1.2 Double Lane Change	49
4.2 Simulation Results.....	50
4.2.1 Fishhook Test	51
4.2.2 Double Lane Change Test	54

4.3 Modified UWAFB ARB.....	58
4.3.1 Fishhook Test	59
4.3.2 Double Lane Change Test	62
Chapter 5 Conclusion	67
5.1 Future Work and Recommendations	67
References	69
Appendix A	74
Appendix B.....	75

List of Figures

FIGURE 1.1: DIAGRAM SUMMARIZING THE VEHICLE ELECTRIFICATION LEVELS [7]	2
FIGURE 2.1: OSCILLATION FREQUENCY RANGES ASSOCIATED WITH THE TERMS NOISE, VIBRATION, AND HARSHNESS [9].....	4
FIGURE 2.2: ILLUSTRATION OF THE VEHICLE VELOCITY, YAW RATE, VEHICLE SIDESLIP ANGLE, TIRE- SLIP ANGLE, AND ROAD-TIRE FRICTION FORCES [13]	6
FIGURE 2.3: VEHICLE RESPONSE SHOWING THE EFFECTS OF CG LOCATION IN HANDLING PERFORMANCE [14]	8
FIGURE 2.4: TOYOTA MOTOR CORPORATION'S RESEARCH AND DEVELOPMENT EXPENSES FROM YEAR 2007 TO 2024 [15].....	9
FIGURE 2.5: EXAMPLE OF AN ARB IN A SUSPENSION SYSTEM [21]	10
FIGURE 2.6: ANTI-ROLL BAR GEOMETRY USED IN SAE SPRING DESIGN MANUAL [22]	11
FIGURE 2.7: ILLUSTRATION OF ISO 8855-2011 VEHICLE COORDINATE SYSTEM [23]	12
FIGURE 2.8: ROLL CENTER FOR A DOUBLE WISHBONE SUSPENSION [20].	14
FIGURE 2.9: ILLUSTRATION OF POSITIVE AND NEGATIVE CAMBER ANGLES [26].....	15
FIGURE 2.10: ILLUSTRATION OF TOE ANGLE IN FRONT WHEEL [27].....	16
FIGURE 2.11: VEHICLE POINT MASS MODEL [29].....	17
FIGURE 2.12: 3 DOF DYNAMIC MODEL [29]	18
FIGURE 2.13: ILLUSTRATION OF A HALF CAR MODEL WITH A ROLL CENTER [32]	19
FIGURE 2.14: DIAGRAM OF THE ARB FORCES AND VARIABLES [32]	21
FIGURE 2.15: DIAGRAM OF 9 DOF VEHICLE MODEL [34].....	22
FIGURE 2.16: MSSP FULL VEHICLE MODEL BUILT FOR THE UWAFI CAMERO SHOWING HARDPOINTS IN GREEN AND MODELED SUSPENSION COMPONENTS.....	25
FIGURE 3.1: 2023 CADILLAC LYRIQ	27
FIGURE 3.2: TRANSLATION OF CAD MODEL TO MSSP MODEL SUSPENSION HARDPOINTS SHOWN IN GREEN TEXTS	28
FIGURE 3.3: EXAMPLE OF AN ELECTRIC MOTOR EFFICIENCY CURVE FOR ELECTRIC DRIVE UNIT [38].	29
FIGURE 3.4: EXAMPLE OF A MOTOR TORQUE AND POWER VS. SPEED GRAPH OF AN AUTOMOTIVE TRACTION MOTOR [39]	30
FIGURE 3.5: SUSPENSION BUSHING SIGN CONVENTION. PICTURE OF THE BUSHING [41]	31
FIGURE 3.6: TRANSLATIONAL BUSHING STIFFNESS FOR INNER FRONT UPPER CONTROL ARM	31

FIGURE 3.7: TORSIONAL BUSHING STIFFNESS FOR INNER FRONT UPPER CONTROL ARM.....	32
FIGURE 3.8: FORCE VS. DISPLACEMENT GRAPH FOR FRONT AND REAR SPRING.....	34
FIGURE 3.9: FORCE VS. VELOCITY GRAPH FOR FRONT AND REAR DAMPER	34
FIGURE 3.10: FORCE VS. DEFORMATION GRAPH OF BUMP AND REBOUND STOPS	36
FIGURE 3.11: MSSP STEERING SYSTEM MODEL SHOWING THE STEERING RACK IN RED AND COLUMN IN GREY	37
FIGURE 3.12: MSSP MODEL OF THE STOCK FRONT ARB IN GREY AND DROP LINK IN BLUE.....	38
FIGURE 3.13: MSSP MODEL OF THE STOCK REAR ARB IN GREY AND DROP LINK IN BLUE.....	38
FIGURE 3.14: MSSP FRONT SUSPENSION MODEL ASSEMBLY WITH THE TEST RIG SETUP	39
FIGURE 3.15: FRONT SUSPENSION PARALLEL WHEEL TRAVEL SIMULATION WHERE RIGHT PICTURE SHOWS THE VERTICAL WHEEL MOVEMENT AND FORCES GENERATED BY THE SUSPENSION COMPONENTS	40
FIGURE 3.16: FRONT SUSPENSION OPPOSITE WHEEL TRAVEL SIMULATION SHOWING THE SEPARATE MOVEMENT OF THE LEFT AND RIGHT WHEEL ASSEMBLY AND FORCES GENERATED BY THE COMPONENTS	40
FIGURE 3.17: MSSP REAR SUSPENSION MODEL ASSEMBLY WITH THE TEST RIG SETUP	41
FIGURE 3.18: REAR SUSPENSION PARALLEL WHEEL TRAVEL SIMULATION SHOWING THE VERTICAL AND CAMBER FORCES GENERATED BY THE SUSPENSION COMPONENTS	42
FIGURE 3.19: REAR SUSPENSION OPPOSITE WHEEL TRAVEL SIMULATION SHOWING SEPARATE MOVEMENT OF THE LEFT AND RIGHT WHEEL ASSEMBLY WITH FORCES GENERATED BY SUSPENSION COMPONENTS	42
FIGURE 3.20: MSSP CHASSIS MODEL SHOWING THE LOCATION OF CG IN YELLOW SPHERE	43
FIGURE 3.21: MSSP TIRE MODEL SHOWING ONE SET OF TIRES FOR AXLE IMPLEMENTED WITH PAC2002 MODEL	45
FIGURE 3.22: MSSP FULL VEHICLE MODEL ON SPMM TEST RIG.....	46
FIGURE 4.1: STEERING INPUT FOR FISHHOOK MANEUVER [47]	49
FIGURE 4.2: DOUBLE LANE CHANGE MANEUVER COURSE DIMENSIONS [48] [49].....	50
FIGURE 4.3: BODY ROLL ANGLE RESPONSE OF STOCK AND UWAF T MODEL FROM FISHHOOK MANEUVER	51
FIGURE 4.4: BODY ROLL ACCELERATION RESPONSE OF STOCK AND UWAF T MODEL FROM FISHHOOK MANEUVER	51

FIGURE 4.5: LATERAL ACCELERATION RESPONSE OF STOCK AND UWAFI MODEL FROM FISHHOOK MANEUVER	52
FIGURE 4.6: YAW RATE RESPONSE OF STOCK AND UWAFI MODEL FROM FISHHOOK MANEUVER	52
FIGURE 4.7: SLIP ANGLE RESPONSE OF THE STOCK AND UWAFI MODEL FROM FISHHOOK MANEUVER	53
FIGURE 4.8: ROLL ANGLE RESPONSE OF STOCK AND UWAFI MODEL FROM DOUBLE LANE CHANGE MANEUVER	54
FIGURE 4.9: ROLL ACCELERATION RESPONSE OF STOCK AND UWAFI MODEL FROM DOUBLE LANE CHANGE MANEUVER	55
FIGURE 4.10: LATERAL ACCELERATION RESPONSE OF STOCK AND UWAFI MODEL FROM DOUBLE LANE CHANGE MANEUVER.....	55
FIGURE 4.11: YAW RATE RESPONSE OF STOCK AND UWAFI MODEL FROM DOUBLE LANE CHANGE MANEUVER	56
FIGURE 4.12: SLIP ANGLE RESPONSE OF THE STOCK AND UWAFI MODEL FROM DOUBLE LANE CHANGE MANEUVER	56
FIGURE 4.13: ROLL ANGLE RESPONSE OF STOCK AND MODIFIED UWAFI MODEL FROM FISHHOOK MANEUVER	59
FIGURE 4.14: LATERAL ACCELERATION RESPONSE OF STOCK AND MODIFIED UWAFI MODEL FROM FISHHOOK MANEUVER	60
FIGURE 4.15: YAW RATE RESPONSE OF STOCK AND MODIFIED UWAFI MODEL FROM FISHHOOK MANEUVER	60
FIGURE 4.16: ROLL ACCELERATION RESPONSE OF UWAFI MODELS FROM FISHHOOK MANEUVER ...	61
FIGURE 4.17: SLIP ANGLE RESPONSE OF STOCK, UNMODIFIED, AND MODIFIED UWAFI MODEL FROM THE FISHHOOK MANEUVER	62
FIGURE 4.18: ROLL ANGLE RESPONSE OF STOCK AND MODIFIED UWAFI MODEL FROM DOUBLE LANE CHANGE MANEUVER	63
FIGURE 4.19: LATERAL ACCELERATION RESPONSE OF STOCK AND MODIFIED UWAFI MODEL FROM DOUBLE LANE CHANGE MANEUVER.....	63
FIGURE 4.20: YAW RATE RESPONSE OF STOCK AND MODIFIED UWAFI MODEL FROM DOUBLE LANE CHANGE MANEUVER	64
FIGURE 4.21: ROLL ACCELERATION OF UWAFI MODELS FROM DOUBLE LANE CHANGE MANEUVER	64

FIGURE 4.22: SLIP ANGLE RESPONSE OF STOCK, UNMODIFIED, AND MODIFIED UWAFT MODEL FROM THE FISHHOOK MANEUVER	65
---	----

List of Tables

TABLE 3.1: SUMMARY OF VEHICLE PARAMETER COMPARISON OF STOCK TO UWAFI VEHICLE	27
TABLE 3.2: SUMMARY OF BUSHING USED FOR FRONT AND REAR SUSPENSION	32
TABLE 3.3: VEHICLE PARAMETER DATA GATHERED FROM SPMI ANALYSIS.....	47
TABLE 4.1: SUMMARY OF FISHHOOK MANEUVER SIMULATION RESULTS	54
TABLE 4.2: SUMMARY OF DOUBLE LANE CHANGE MANEUVER SIMULATION RESULTS	58
TABLE 4.3: VEHICLE PARAMETER COMPARISON BETWEEN MODIFIED AND UNMODIFIED MODELS	59
TABLE 4.4: SUMMARY OF FISHHOOK SIMULATION RESULTS COMPARING UWAFI MODELS	62
TABLE 4.5: SUMMARY OF DOUBLE LANE CHANGE RESULTS COMPARING UWAFI MODELS.....	66

List of Abbreviations

ARB	Anti-Roll Bar
AWD	All-Wheel Drive
CAD	Computer Aided Design
CG	Center of Gravity
DOF	Degree of Freedom
NHTSA	National Highway Traffic Safety Administration
NVH	Noise, Vibration, and Harshness
OEM	Original Equipment Manufacturer
RMS	Root Mean Square
RWD	Rear-Wheel Drive
SPMM	Suspension Parameter Measurement Machine
UWAF	University of Waterloo Alternative Fuels Team

Chapter 1

Introduction

1.1 Research Motivation

The automotive industry has increasingly adopted virtual engineering design methodologies, driven by advancements in computing power and sophisticated simulation software. This approach facilitates the efficient design, testing, and optimization of complex systems, significantly reducing both development time and costs compared to traditional methods [1].

One of the key innovations in modern engineering is the use of modeling and simulation. With the availability of powerful computational devices, the industry now relies heavily on simulations to evaluate design and manufacturing changes with high confidence. This reliance minimizes the need for costly physical testing. Additionally, the reduced time required to obtain similar results allows engineers to implement changes directly during the design process or on the manufacturing floor, enhancing the overall efficiency of the development cycle.

Automotive engineers use sophisticated software packages to predict mechanical failures, fluid flow characteristics, thermal dynamics, and multibody dynamics and kinematics. For vehicle dynamics, a subset of multibody dynamics and kinematics, engineers have various software tools to design and test vehicle suspension concepts. Vehicle models are based on mathematical and geometrical equations and can be created using multibody dynamics software like MSC Adams and MapleSim or automotive-specific software like CarSim [2] [3] [4]. Full vehicle test rigs and vehicle component test machines provide relevant data to input into these models, and onboard data acquisition systems gather data to tune and validate the vehicle model.

1.1.1 Electrification of Powertrain System

In recent years, electrified vehicles have rapidly increased their global market share due to several factors. These include shifting consumer preferences toward sustainability and efficiency, stricter regulatory demands from government bodies, and greater commitment and investment from the automotive industry. A meta-analysis of automotive market trends conducted by Aachen University suggests that 54.3% of all new vehicles will be electrified by 2030 [5]. The anticipated dominance of electrified vehicles in the automotive market has spurred a surge in research efforts focused on their design, thereby accelerating the development and maturity of electrified vehicle technologies.

The term "electrified vehicles" refers to vehicles that use an electric propulsion system in their powertrain [6]. These systems typically include a battery pack and an electric motor, with their utilization varying based on the vehicle's level of electrification. Electrified vehicles encompass mild hybrid electric vehicles (MHEVs), full hybrid electric vehicles (FHEVs), and full electric vehicles (EVs). The primary difference between these types of propulsion systems lies in their dependence on an internal combustion engine (ICE). MHEVs primarily use an ICE for propulsion, FHEVs can operate solely on electric power for a limited range before switching to an ICE, and EVs rely entirely on electric power with no use of an ICE. Figure 1.1 summarizes the different level of vehicle electrification technology in today's market.

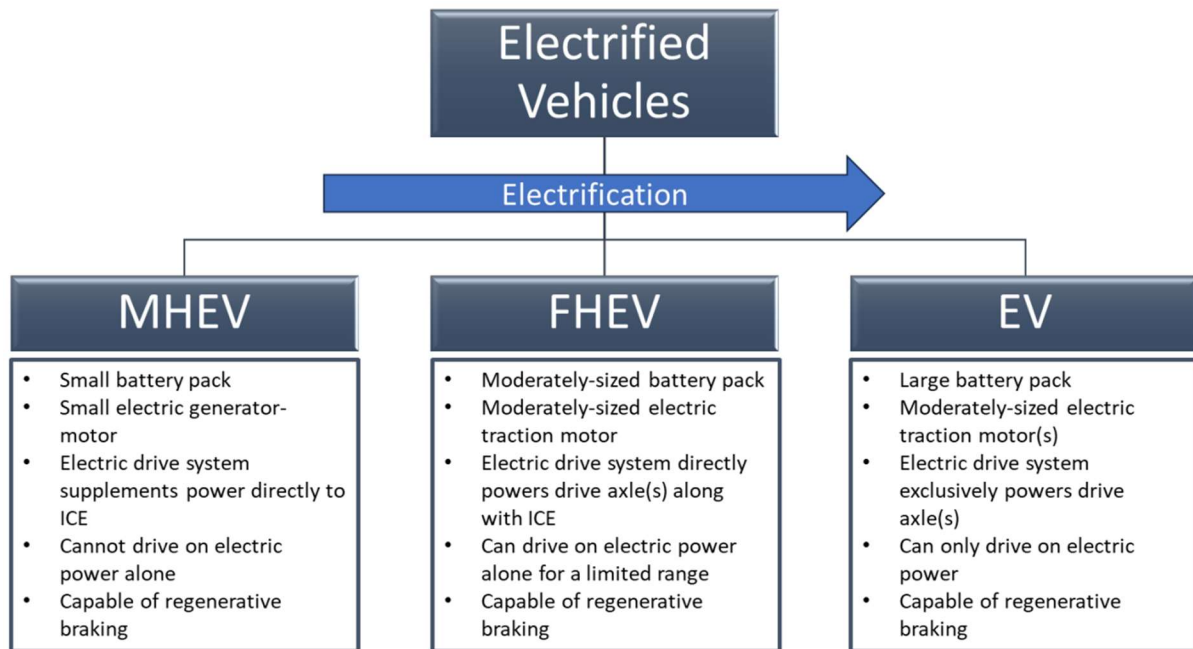


Figure 1.1: Diagram summarizing the vehicle electrification levels [7]

With electrification of the vehicle in today's market, it came with draw backs. The vehicle's overall weight has increased due to the addition of heavy battery pack and electric motor. The heavy battery pack and electric motor increased the sprung and center of gravity (CG) height, generally considered undesirable in the view of vehicle dynamics. In addition, a heavier vehicle body combined with a higher CG tends to yield larger dynamic lateral load transfers during steering maneuvers, which have negative effects in view of vehicle handling and directional stability controls [8].

1.2 Research Objectives

This thesis is set within the context of the University of Waterloo EcoCAR Team (UWAF) and their participation in the EcoCAR EV Challenge. The competition's primary objective is to design and build an electric vehicle with SAE Level 2 autonomous capabilities. For this research, the platform used is based on the 2023 Cadillac LYRIQ, a vehicle generously donated by OEM sponsor, General Motors, to UWAF for the challenge. The main goal of this research is to support the development of the suspension systems for the UWAF LYRIQ.

The research aims to create a multi-body simulation environment to understand the vehicle dynamics response of the electric vehicle's suspension system and to use this environment to redesign components for customization of the vehicle platform. This simulation environment facilitates rapid prototyping of suspension parameters to achieve the desired vehicle dynamics performance for the 2023 Cadillac LYRIQ. The objectives of this research are divided into three parts:

1. Detailing the process of creating full-vehicle models using MSC Adams Car software.
2. Conducting a parametric study of the ARB properties to identify a suitable design for UWAF's LYRIQ.
3. Proposing a custom ARB design for UWAF's LYRIQ based on insights gained from the simulations.

This research primarily focuses on vehicle dynamics modeling and component design, utilizing the results from vehicle dynamics simulations. It is specifically tailored for student design teams engaged in vehicle construction, providing a robust framework for understanding and enhancing vehicle performance through advanced suspension system design.

Chapter 2

Literature Review

In this section of the thesis, the research into the process of designing ARBs within the context of both industry practices and student design teams is identified and examined. Subsequently, the theoretical background of vehicle suspension components and the relevant dynamics equations are reviewed and discussed.

2.1 Ride Comfort

Ride comfort is a multifaceted concept that defies precise definition. In the automotive industry, ride comfort is typically segmented into three primary components: noise, vibration, and harshness (NVH). This classification is based on distinct frequency ranges, as illustrated in Figure 2.1 [9]. Frequencies ranging from 0.1 to 20 Hz are referred to as tactile oscillations or vibrations, frequencies of 100 Hz and above are categorized as audible oscillations or noise, and the intermediate range is known as harshness [9]. This division helps in systematically analyzing and addressing the various elements that contribute to overall ride comfort.

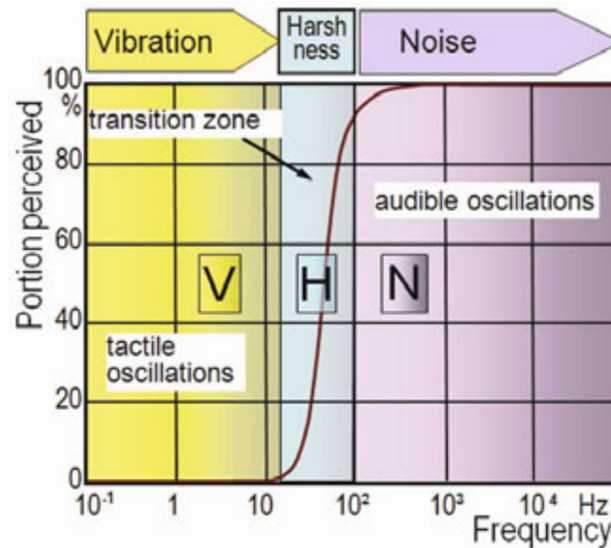


Figure 2.1: Oscillation frequency ranges associated with the terms noise, vibration, and harshness [9]

Ibicek et al. [10] demonstrated that vehicle roll significantly influences the perception of discomfort. By placing a vehicle on a four-post rig, human subjects were exposed to roll movements quantified using root mean square (RMS) values of roll acceleration. Participants were then asked to rate their level of discomfort. The study found that discomfort levels increased proportionally with the RMS values of roll acceleration within the studied range of 0 to 0.6 m/s². Additionally, Koumura et al. [11] identified that, for a specific vehicle, the resonance frequency for roll accelerations occurs around 2 Hz, with a secondary peak around 5 Hz. These frequencies closely align with the natural frequencies of the human body, indicating that roll is a critical factor in perceived discomfort. Koumura et al. further established that the roll acceleration gain is primarily dependent on roll stiffness and the roll-damping coefficient. Around the resonance frequency, an increase in roll stiffness was shown to amplify the roll response, whereas an increase in roll-damping reduced the roll response. This suggests that optimizing roll stiffness and damping is essential for mitigating discomfort caused by vehicle roll.

2.2 Handling and Stability

Handling and stability are two critical properties of a vehicle, significant for multiple reasons. Primarily, the vehicle's behavior must ensure safety, meaning that driver inputs should not lead to excessive responses that render the vehicle uncontrollable. While a vehicle inherently possesses certain physical limits, such as tire friction thresholds, surpassing these limits can result in complete tire slippage [12].

Moreover, car manufacturers strive to ensure that their vehicles exhibit smooth handling behavior, as judged by both test drivers and customers. The vehicle should provide the driver with a sense of control, responsiveness to inputs, quick damping of sudden deviations, and limited body roll. This level of performance should be maintained over a wide range of both longitudinal and lateral accelerations, which may sometimes conflict with the need to detect and correct critical behavior in a timely manner. Balancing these requirements is essential for delivering a vehicle that is both safe and enjoyable to drive.

One critical parameter that assists engineers in comprehending vehicle handling behavior is the vehicle slip angle data. The vehicle slip angle is an instantaneous, calculated variable that quantifies the lateral slippage of a vehicle during cornering maneuvers. This angle is defined as the difference between the vehicle's actual trajectory, often referred to as the heading or course over ground, and the orientation of the vehicle's body, known as the true heading [13].

It is essential to distinguish the vehicle slip angle from the tire slip angle, as the former pertains to the vehicle coordinate system, which is discussed in a later section. Vehicle slip angle is a pivotal metric in determining whether a vehicle is understeering or oversteering during cornering. A vehicle is understeering if the axle steering angle must be increased with an increase in vehicle forward speed to follow the desired path of the corner. Conversely, a vehicle is oversteering if the steering angle must be decreased with an increase in vehicle forward speed to maintain the desired path [12]. Therefore, the vehicle slip angle is higher in an understeering vehicle compared to an oversteering vehicle. Figure 2.2 illustrates the vehicle slip angle on a vehicle coordinate system.

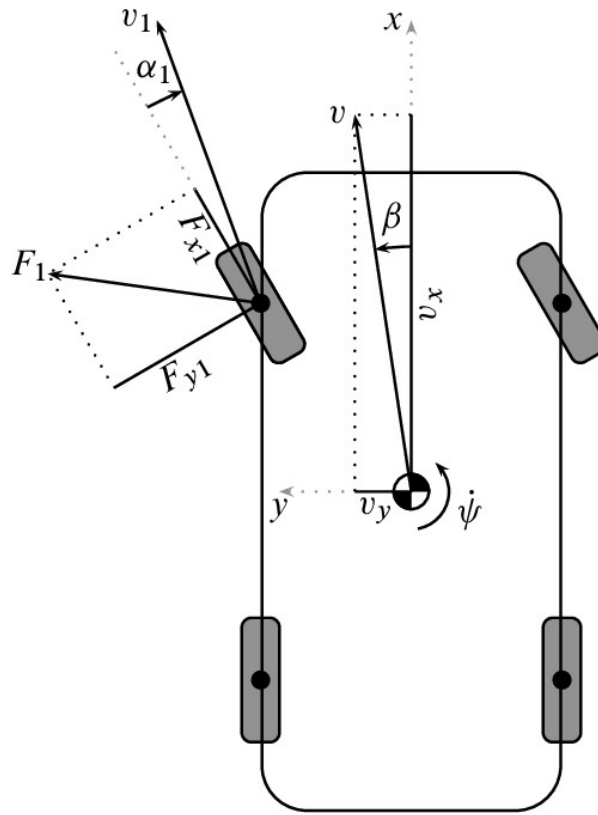


Figure 2.2: Illustration of the vehicle velocity, yaw rate, vehicle sideslip angle, tire-slip angle, and road-tire friction forces [13]

The vehicle's velocity vector v at the CG can be broken down into two components: the longitudinal velocity v_x , which is positive in the forward direction, and the lateral velocity v_y , which is positive in the leftward direction. The angle β between the longitudinal x -axis and the velocity vector v is known as the vehicle sideslip angle and is defined as positive in the counterclockwise direction. For the front

left wheel, the velocity vector v_1 at the wheel center is depicted. The angle α_1 between v_1 and the tire's longitudinal axis is referred to as the tire slip angle and is considered positive in the clockwise direction. A positive tire slip angle, as shown in the figure, generates a positive lateral tire force F_{y1} . The longitudinal tire force F_{x1} arises from nonzero longitudinal tire slip. Together, F_{y1} and F_{x1} the tire force vector F_1 . Similar road-tire friction forces are generated for the other three tires as well.

Nordeen [14] explored the impact of the center of gravity (CG) location on vehicle handling performance. It reveals that a forward CG decreased the handling performance, requiring more steering input to achieve the same turning radius, and thus making the vehicle less responsive to quick maneuvers. Conversely, a rearward CG improves agility but reduces stability, making the vehicle more prone to oversteer. The CG location also affects roll behavior, with a lower CG reducing body roll and enhancing stability and comfort, while a higher CG increases body roll, leading to greater instability and discomfort. The paper further discusses how CG location influences transient response, noting that a forward CG can result in a slower and more oscillatory response to sudden steering inputs, particularly at higher speeds. Overall, optimizing CG location in conjunction with other vehicle parameters, such as tire properties, suspension geometry, and damping characteristics, is essential to achieving a balance between stability and responsiveness, thereby enhancing both safety and performance. Figure 2.2 shows the results of the vehicle response with different CG locations, where a and b are the distance between the CG location to the front and rear axle, respectively.

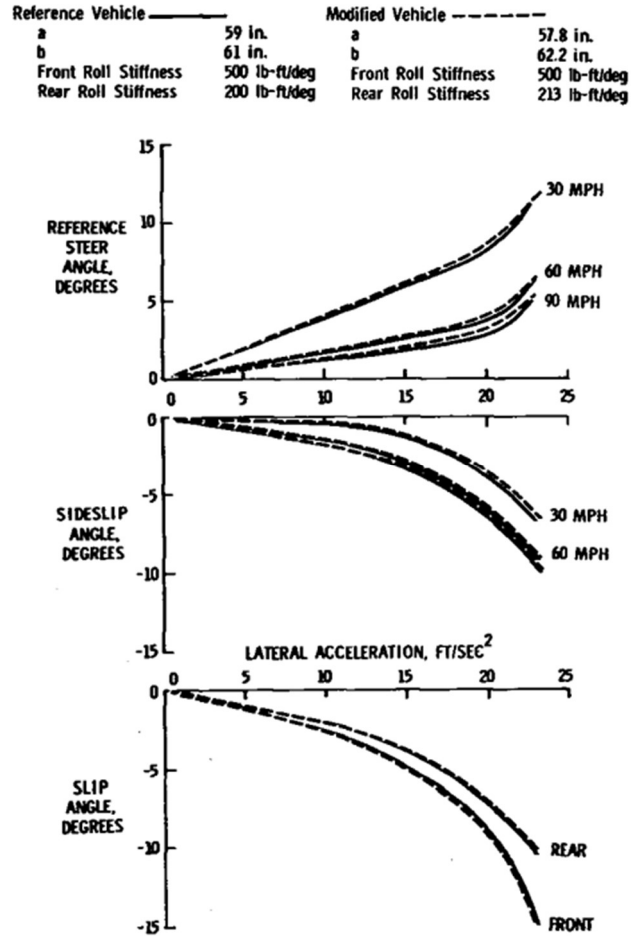


Figure 2.3: Vehicle response showing the effects of CG location in handling performance [14]

2.3 Vehicle Suspension Development

With the advent of more complex mechanisms and designs for vehicle applications, the process of building prototypes to validate these designs has become increasingly costly and time-consuming for Original Equipment Manufacturers (OEMs), see Figure 2.3 [15]. To mitigate the resource strain associated with validating vehicle performance and safety, the automotive industry is more aggressively adopting virtual environments for vehicle development and testing [16]. The virtual environment facilitates rapid iteration and result generation throughout the vehicle development process. However, a comprehensive virtual design integral to the vehicle development process and capable of establishing an industrial standard has not yet been fully realized [17]. Badiru et. al. [18] notes that the current state

of this process often relies on the basic ride comfort design of predecessor vehicles. The ability to virtually tune vehicle behavior or use objective methods to achieve a near-acceptable state for ride comfort would substantially enhance development efficiency.

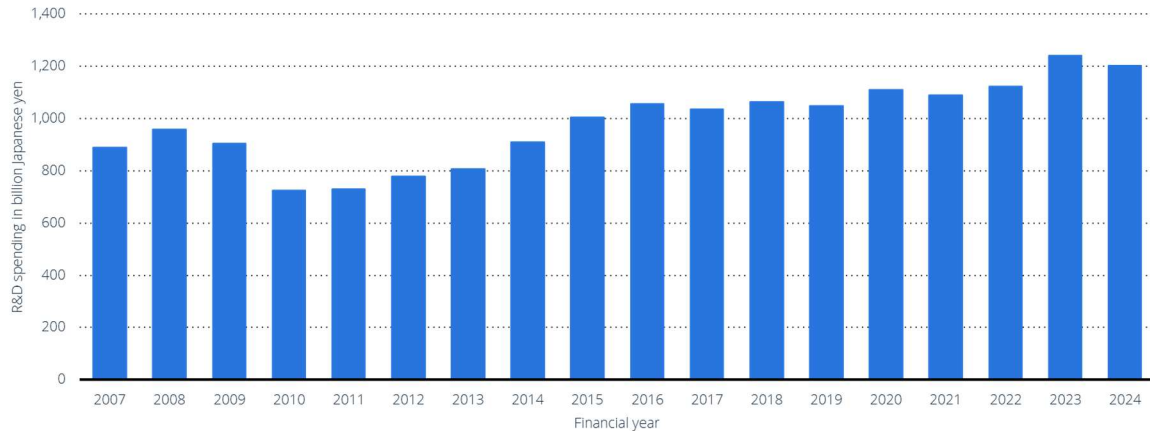


Figure 2.4: Toyota Motor Corporation's research and development expenses from year 2007 to 2024 [15]

As observed in the industry, this scenario mirrors the approach taken by student design teams developing vehicle suspension systems. Typically, these teams utilize designs from previous years as a foundational understanding of vehicle dynamics, iterating upon existing designs. While this method ensures the reliability of the vehicle system, it often slows innovation and hinders the enhancement of vehicle performance. Implementing a virtual vehicle suspension development environment would benefit student design teams similarly to its impact on the industry, offering a faster and more cost-effective means of achieving the desired vehicle performance.

2.4 Definition of Suspension

Vehicle suspension refers to the intricate system of components designed to connect a vehicle to its wheels, allowing for relative motion between them [19]. This system plays a crucial role in ensuring the vehicle's performance, comfort, and safety. The primary functions of a vehicle suspension system include supporting the vehicle's weight, absorbing and dampening shocks from road irregularities, and maintaining tire contact with the road surface [20]. These functions are essential for good handling, stability, and braking efficiency.

The key components of a vehicle suspension system include springs, dampers, linkages, ARB, and bushings. Springs, which can be coil springs, leaf springs, torsion bars, or air springs, store energy from road bumps and help maintain the vehicle's height. Dampers control the movement of the springs and suspension, preventing excessive oscillation and ensuring consistent tire contact with the road. Linkages are mechanical connections that allow for controlled movement of the wheels relative to the vehicle body. Bushings are used to prevent direct contact between two metal parts, helping to reduce noise and minimize vibration. Made from soft materials like rubber, bushings serve as vibration isolators, connecting various moving components to the vehicle body or suspension frame. There are many types of bushings, classified based on the number of degrees of freedom (DOF) they allow between the two connected parts.

2.4.1 Anti-Roll Bar

An ARB, also known as a stabilizer or sway bar, is an essential part of a vehicle's suspension system. The ends of the ARB are attached to the suspension links near the wheels, while the center is connected to the vehicle's frame, allowing it to rotate freely. When both ends of the bar move equally, it rotates in its bushings without providing torsional resistance. The bar's torsional stiffness is key to its ability to reduce body roll. Although most vehicles come with a front ARB, installing ARBs on both the front and rear can further decrease body roll. Figure 2.4 shows an example of a ARB integrated to the suspension system.

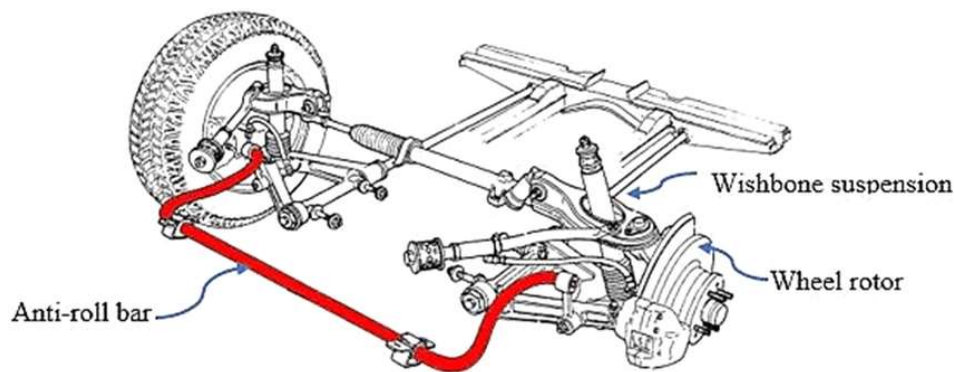


Figure 2.5: Example of an ARB in a suspension system [21]

ARB enhances vehicle handling and driver confidence by minimizing body roll. It serves two main purposes: reducing body roll and redistributing cornering loads between the front and rear wheels. This redistribution allows for modifications in the vehicle's handling characteristics. By adjusting the roll

stiffness of the ARBs at the front and rear suspensions, handling can be tailored. A stiffer front ARB increases lateral load transfer to the front tires, reducing deflection and resulting in more understeer. Conversely, a stiffer rear ARB increases roll stiffness at the rear, leading to oversteer. Therefore, ARBs are also utilized to improve directional control and stability.

Designing an ARB involves achieving the necessary torsional stiffness to enhance vehicle stability and handling without surpassing the material's mechanical limits. Analyzing anti-roll bars is a straightforward process, which is why published studies on this specific topic are scarce. Manufacturers typically conduct standard design analyses, but these results are not publicly available. Instead, available studies often focus on the characteristics of bushings and the fatigue life of anti-roll bars, with some research dedicated to design automation. Anti-roll bars are considered a subset of torsion bars. Useful formulas for calculating roll stiffness and deflection under a given load are provided in manuals. However, these formulations apply only to bars with standard shapes, such as simple, torsion bar-shaped anti-roll bars. Figure 2.5 shows the ARB design geometry used in SAE Spring Design Manual.

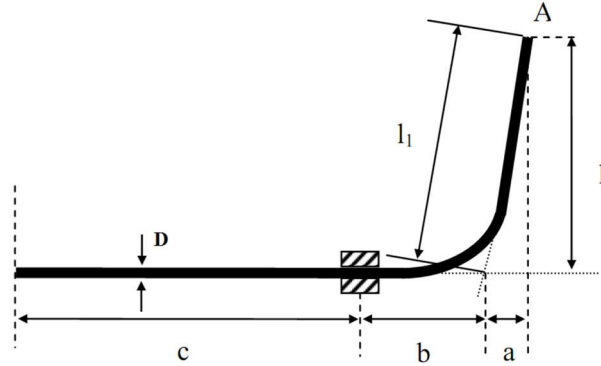


Figure 2.6: Anti-roll bar geometry used in SAE Spring Design Manual [22]

To calculate the torsional stiffness of the ARB with point load F at point A into the plane of the page, following equations are used:

$$k_{ARB} = \frac{FL^2}{2\delta_a} \quad \frac{Nmm}{rad} \quad (2.1)$$

where,

L = Half-length of ARB:

$$L = a + b + c \quad (2.2)$$

δ_a = Deflection of point A:

$$\delta_a = \frac{F}{3EI} \left[l_1^3 - a^3 + \frac{L}{2}(a + b)^2 + 4l_2^2(b + c) \right] \quad (2.3)$$

I = Moment of inertia of ARB:

$$I = \frac{\pi D^4}{64}$$

with, D = outer diameter of the ARB and E = elastic modulus of the ARB material.

2.5 Vehicle Parameters

Vehicle dynamics is a critical field in automotive engineering that focuses on understanding and optimizing the movement of vehicles. It encompasses the study of forces and moments that influence vehicle motion, providing insights essential for the design and development of vehicles to ensure safety, comfort, and performance. Key aspects of vehicle dynamics are longitudinal, lateral, vertical, yaw, roll, and pitch dynamics. Figure 2.6 shows the International Organization of Standardization (ISO) vehicle coordinate system that is used for this research.

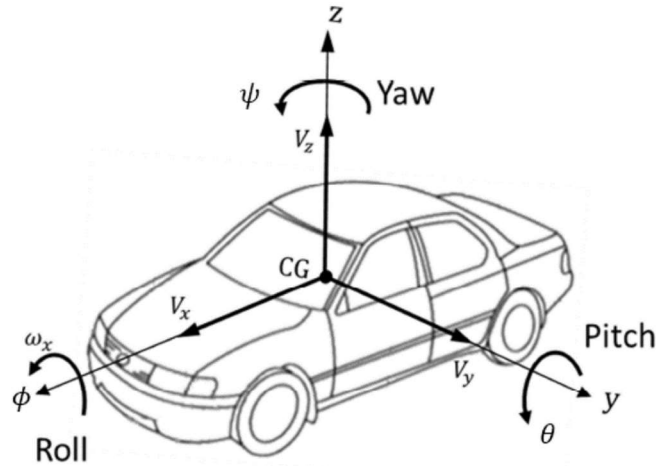


Figure 2.7: Illustration of ISO 8855-2011 vehicle coordinate system [23]

Longitudinal dynamics involve forces that act along the vehicle's direction of travel, such as acceleration, braking, and aerodynamic drag. The main goal of studying longitudinal dynamics is to optimize the vehicle's acceleration and braking performance while maintaining stability and control.

These dynamics are governed by Newton's second law, which states that the force acting on a vehicle equals the mass of the vehicle multiplied by its acceleration [24]. This principle is essential in designing powertrains and braking systems that effectively manage these forces.

Lateral dynamics pertain to forces acting perpendicular to the vehicle's direction of travel, which greatly influence handling and stability during cornering. These forces arise mainly from the interaction between the tires and the road surface. Key factors in lateral dynamics include tire slip angles, lateral tire forces, and vehicle sideslip angles [25]. Understanding these elements is crucial for developing suspension components, such as ARB, that enhance handling and prevent skidding.

Vertical dynamics focus on the forces acting vertically on the vehicle, which are vital for ride comfort and road holding. These forces, caused by road irregularities, are managed by the vehicle's suspension system. The study of vertical dynamics involves analyzing the spring and damper characteristics of the suspension system to balance comfort and handling performance. Yaw dynamics involve the rotation of the vehicle around its vertical axis and are essential for understanding stability during maneuvers like lane changes and skids. Factors influencing the yaw moment include vehicle speed, tire characteristics, and the location of the CG.

Roll dynamics refer to the tilting of the vehicle around its longitudinal axis during cornering, where the vehicle body tilts outward due to centrifugal forces. Pitch dynamics occur during acceleration and braking, causing the vehicle's nose to lift or dip. These dynamics are particularly critical for high-performance vehicles and SUVs, where the risk of rollover is higher.

2.5.1 Roll Center and Roll Axis

The roll center is a specific point on the Y-Z and X-Z planes where lateral forces can act on the vehicle body without causing it to roll. The roll axis is an imaginary line in the X-Z plane that connects the front and rear roll centers, and it represents the axis around which the vehicle body rolls. The positioning of the roll centers and the roll axis is crucial for a vehicle's handling and ride quality. Figure 2.7 illustrates roll center of a double wishbone suspension.

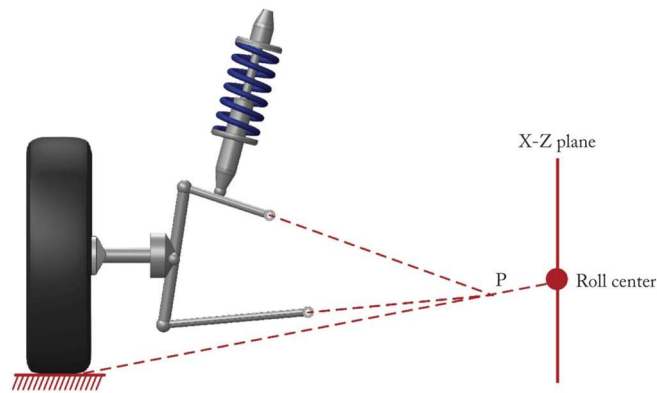


Figure 2.8: Roll center for a double wishbone suspension [20].

2.5.2 Sprung and Unsprung Mass

In vehicle dynamics, masses are categorized into two types: sprung mass and unsprung mass. Sprung mass includes all the components supported by the suspension system, such as the vehicle's main body, chassis, drivetrain, passengers, and cargo. This mass benefits from the suspension's damping effects, contributing to ride comfort and stability. Unsprung mass, conversely, encompasses the components not supported by the suspension, including the wheels, tires, brake assemblies, and parts of the suspension system itself, like the axles and control arms. Together, the sprung and unsprung masses constitute the total vehicle mass, with each type playing a distinct role in influencing the vehicle's performance and handling.

2.5.3 Camber Angle

The camber angle is the angle between the X-Z plane of the wheel and a line perpendicular to the plane of road on the Y-Z plane of the vehicle. This angle is negative if the wheel is inclined inward and positive when inclined outward, as shown in Figure 2.8. A positive camber would be useful for a reduction in tire wear due to the road profile. On the other hand, a negative camber provides better tire grip and improves handling [20].

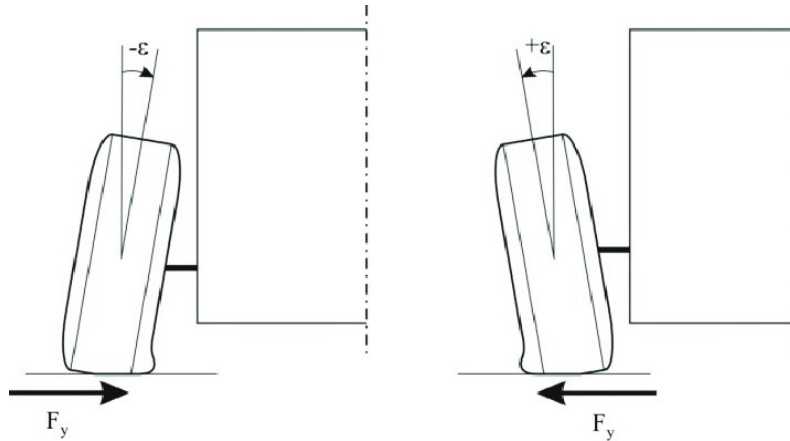


Figure 2.9: Illustration of positive and negative camber angles [26]

2.5.4 Toe Angle

The toe angle in vehicle dynamics is the angle between the wheel's X-Z plane and the vehicle's X-Z plane, see Figure 2.9. “Toe-in” occurs when the front of the wheel points inward, while “toe-out” happens when the wheel points outward. When rolling resistance and braking forces push the wheel backward, it tends to create a toe-out condition. Conversely, when traction pulls the wheel forward, it results in a toe-in situation. For minimal tire wear and rolling resistance, the ideal toe angle is zero. Manufacturers typically set the static toe angle so that it becomes zero under operating conditions. To counteract the toe-out effect caused by braking forces on the front axle, manufacturers often set a toe-in angle on the front wheels. A rear drive axle is usually set with a toe-out angle due to traction forces, while a rear non-drive axle is adjusted to toe-in to offset rolling resistance.

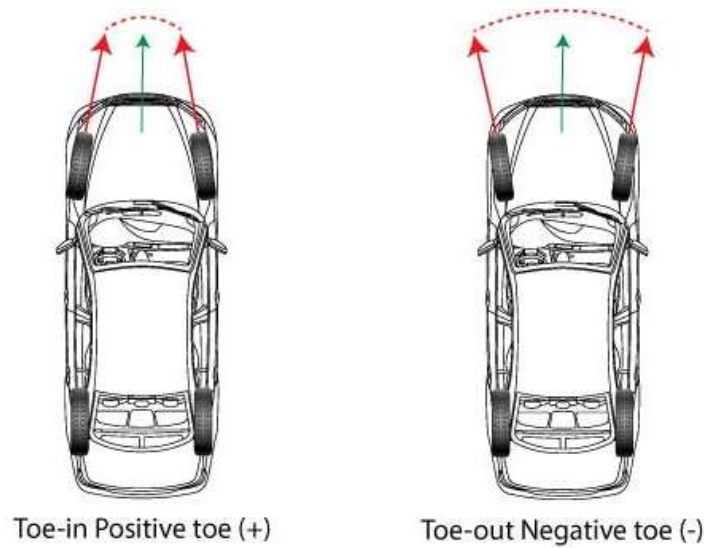


Figure 2.10: Illustration of toe angle in front wheel [27]

2.6 Point Mass Model

A point mass model is a simplified representation used in vehicle dynamics to analyze the motion and behavior of a vehicle. In this model, the entire mass of the vehicle is assumed to be concentrated at a single point, typically located at the CG. This simplification helps in studying the fundamental aspects of vehicle motion without the complexities involved in a full vehicle structure [28]. By concentrating the mass at a single point, the model eliminates the need to consider the distribution of mass throughout the vehicle.

The point mass model is primarily used to study linear motion, such as acceleration, deceleration, and straight-line stability. It simplifies the analysis of longitudinal dynamics, where the primary forces are along the direction of travel. Additionally, it is useful in basic studies of lateral dynamics, including the response to lateral forces during cornering, though it lacks the detail needed for in-depth analysis of handling and stability.

In terms of mechanics, the model uses basic Newtonian equations, where force is equal to mass times acceleration ($F = ma$). This applies to both longitudinal and lateral forces. However, while the point mass model is useful for basic analysis, it does not account for rotational dynamics (pitch, roll, and yaw), detailed interactions between different vehicle components, or the effects of suspension geometry and tire characteristics [29]. Thus, it is less effective for high-fidelity simulations required in advanced

vehicle design and performance optimization. Figure 2.10 shows the variables in the vehicle point mass model.

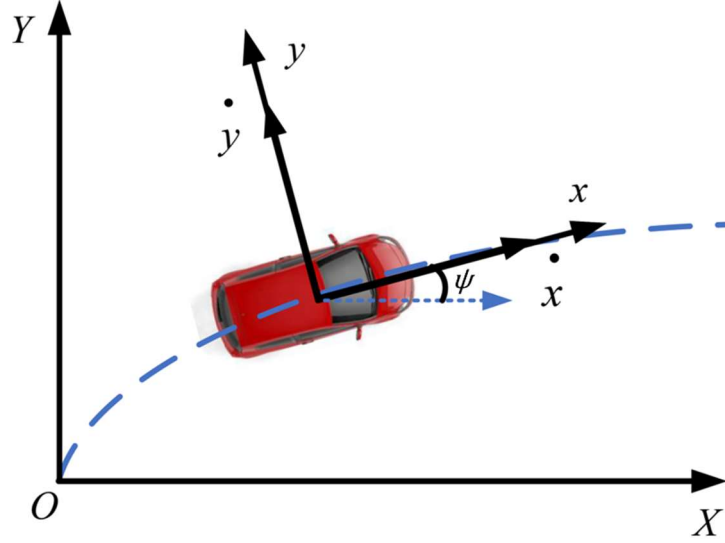


Figure 2.11: Vehicle point mass model [29]

The vehicle point mass model can be expressed as [30]:

$$\dot{X} = \dot{x} \cos \psi - \dot{y} \sin \psi \quad (2.4)$$

$$\dot{Y} = \dot{x} \sin \psi + \dot{y} \cos \psi \quad (2.5)$$

where $\dot{X}$ and $\dot{Y}$ are the longitudinal and lateral velocities in the fixed coordinate system, respectively. Meanwhile, $\dot{x}$ and $\dot{y}$ are the longitudinal and lateral velocities in the vehicle coordinate system. Additionally, ψ signifies the vehicle's course angle.

2.7 Bicycle Model

A bicycle model, also known as a single-track model, represents a vehicle as a rigid body with two wheels: one at the front and one at the rear. The front wheel is steerable and incorporates a tire model to capture cornering dynamics. This model has 3 DOF: x-translation, y-translation, and yaw-rotation. It serves as a reduced-order model, developed as an advancement from the point mass model, primarily used as a benchmark for validating more complex models. Figure 2.11 shows the illustration of 3 DOF dynamics model.

There are several variations of the bicycle model, each incorporating different assumptions and levels of complexity. Vehicle motion in a bicycle model can be described using two primary approaches: kinematic and dynamic. The kinematic approach focuses on the geometry of the system to describe vehicle motion, while the dynamic approach takes into account the forces generated by the tires, applying the corresponding moments and forces to the vehicle body to generate motion.

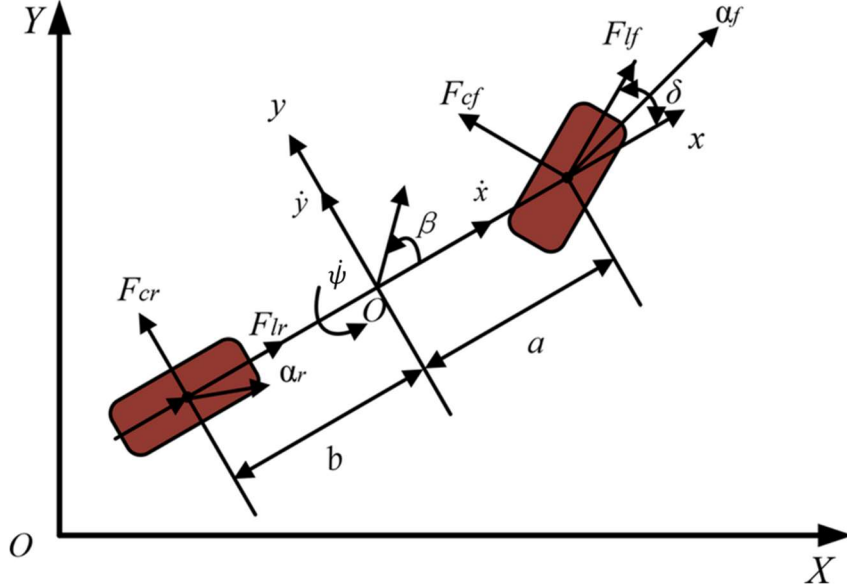


Figure 2.12: 3 DOF dynamic model [29]

The following equations describe the handling dynamics of a rigid vehicle, starting with the general equation of a body in fixed reference frame [31]:

$$m\ddot{X} = \sum F_x \quad (2.6)$$

$$m\ddot{Y} = \sum F_y \quad (2.7)$$

$$I_z\ddot{\psi} = \sum M_z \quad (2.8)$$

where m is the total vehicle mass, I_z is the vehicle yaw moment of inertia, $\ddot{X}$ is the vehicle longitudinal acceleration, $\ddot{Y}$ is the vehicle lateral acceleration, and $\ddot{\psi}$ is the yaw angle acceleration.

Assuming that the vehicle travels on a horizontal road and ignores the effect of air resistance, the vehicle dynamic equation in vehicle reference frame is as follows:

$$m\ddot{x} = m\dot{y}\dot{\psi} + 2(F_{lf} + F_{cf}\delta + F_{lr}) \quad (2.9)$$

$$m\ddot{y} = -m\dot{x}\dot{\psi} + 2(F_{cf} + F_{cr}) \quad (2.10)$$

$$I_z\ddot{\psi} = 2(aF_{cf} - bF_{cr}) \quad (2.11)$$

where F_{lf} and F_{lr} are the longitudinal forces of the front and rear tires, respectively. F_{cf} and F_{cr} are the lateral forces of the front and rear tires, respectively.

2.8 Roll Dynamic Model

A roll dynamic model with a roll center can be used to describe vehicles roll dynamics, see Figure 2.12.

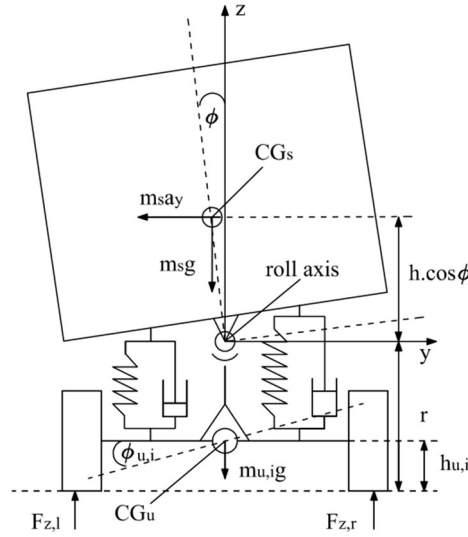


Figure 2.13: Illustration of a half car model with a roll center [32]

Assuming that the body roll angle ϕ is small, the moment equilibrium around the roll center for the car body results in equation [33]:

$$m_s a_y h + m_s g h \phi + \frac{w}{2} (F_{z,l} - F_{z,r}) = I_{xx} \ddot{\phi} \quad (2.12)$$

where m_s is the vehicle sprung mass, a_y is the lateral acceleration of the sprung mass, h is the distance of center of gravity from the roll center, w is track width of the front and rear suspension, $F_{z,l}$ and $F_{z,r}$ are forces on the wheels at each corner, I_{xx} is the moment of inertia of a sprung mass, and $\ddot{\phi}$ is the roll acceleration.

The vehicle can move both longitudinally and laterally and can yaw. The sprung mass can rotate around the roll axis, while the unsprung masses can also rotate in roll, allowing the model to include the effect of the tires' vertical compliance on roll performance. Suspension springs, dampers, and anti-roll bars generate moments between the sprung and unsprung masses in response to roll motions. The effects of aerodynamic inputs (wind disturbances) and road inputs (cross-gradients, dips, and bumps) are neglected. The roll stiffness and damping of the vehicle suspension systems are assumed to remain constant for the range of roll motions considered. The system's equation of motion can be derived using Newton's law [32]:

$$(I_{xx} + m_s h^2) \ddot{\phi} - I_{xz} \ddot{\psi} = m_s g h \phi + m_s v h (\dot{\beta} + \dot{\psi}) - k_f (\phi - \phi_{uf}) - b_f (\dot{\phi} - \dot{\phi}_{uf}) + M_{ARf} - k_r (\phi - \phi_{ur}) - b_r (\dot{\phi} - \dot{\phi}_{ur}) + M_{ARr} \quad (2.13)$$

$$-r F_{yf} = m_{uf} v (r - h_{uf}) (\dot{\beta} + \dot{\psi}) + m_{uf} g h_{uf} \phi_{uf} - k_{tf} \phi_{uf} + k_f (\phi - \phi_{uf}) + b_f (\dot{\phi} + \dot{\phi}_{uf}) + M_{ARf} \quad (2.14)$$

$$-r F_{yr} = m_{ur} v (r - h_{ur}) (\dot{\beta} + \dot{\psi}) + m_{ur} g h_{ur} \phi_{ur} - k_{tr} \phi_{ur} + k_r (\phi - \phi_{ur}) + b_r (\dot{\phi} + \dot{\phi}_{ur}) + M_{ARr} \quad (2.15)$$

where I_{xz} is the yaw-roll inertial of the sprung mass, $\ddot{\phi}$ is the sprung mass roll angular acceleration, $\ddot{\psi}$ is the yaw acceleration, g is the gravitational acceleration, v is the vehicle's longitudinal velocity, $\dot{\beta}$ is the vehicle's side slip angular velocity, $\dot{\psi}$ is the yaw angular velocity, $k_{f,r}$ and $b_{f,r}$ are suspension roll stiffness and damping on the front and rear axle respectively. ϕ_{uf} and $\dot{\phi}_{uf}$ is the roll angle and roll angular velocity of the unsprung mass, respectively. The lateral tire forces, F_{yf} and F_{yr} in the direction of the velocity at the wheel ground contact points are modelled by using linear stiffness coefficients as [32]:

$$F_{yf} = \mu C_f \alpha_f \quad (2.16)$$

$$F_{yr} = \mu C_r \alpha_r \quad (2.17)$$

with the tire side slip angles:

$$\alpha_f = -\beta + \delta_f - \frac{l_f \dot{\psi}}{v} \quad (2.18)$$

$$\alpha_r = -\beta - \frac{l_r \dot{\psi}}{v} \quad (2.19)$$

where μ is the road adhesion coefficient, $C_{f,r}$ are tire cornering stiffness on the front and rear axles, respectively. $l_{f,r}$ are the length of the front and rear axle from the CG, and δ_f is the steering angle.

Equations for the moments M_{ARf} and M_{ARr} in equation system equations above is given as [32]:

$$M_{ARf} = 4k_{AOf} \frac{t_a t_b}{c^2} \phi - 4k_{AOf} \frac{t_a^2}{c^2} \phi_{uf} \quad (2.20)$$

$$M_{ARr} = 4k_{AO r} \frac{t_a t_b}{c^2} \phi - 4k_{AO r} \frac{t_a^2}{c^2} \phi_{ur} \quad (2.21)$$

where $k_{AOf,r}$ are the torsional stiffness of the front and rear ARB, t_a is the half the distance between the two suspensions, t_b is the half the distance between the chassis, c is the length of the ARB. Figure 2.13 shows the diagram of the ARB with the acting forces and variables.

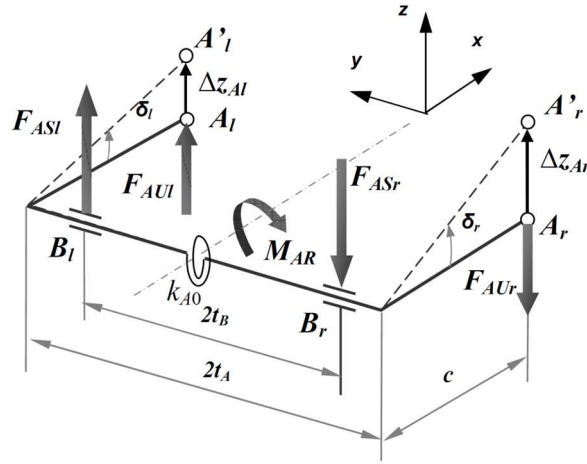


Figure 2.14: Diagram of the ARB forces and variables [32]

Typically, numerical methods are required to effectively solve these equations using tools like MATLAB/Simulink due to the nonlinearity, the dependency of tire forces on slip angles, and the significant coupling between terms. It is important to note that the results are only valid for the specific vehicle whose parameters were used in the model.

2.9 9 Degree of Freedom Model

A 9 DOF model is a vehicle model that is another step above a bicycle model. It uses the same equations and includes the same degrees of freedom but incorporates 6 additional degrees of freedom: pitch, yaw, and 4 rotational degrees of freedom to account for each of the 4 wheels. The reason why the z translational degree of freedom is neglected is because the road is assumed to be flat with road elevation and road imperfections neglected. Figure 2.14 shows a diagram denoting variable in the 9 DOF vehicle model.

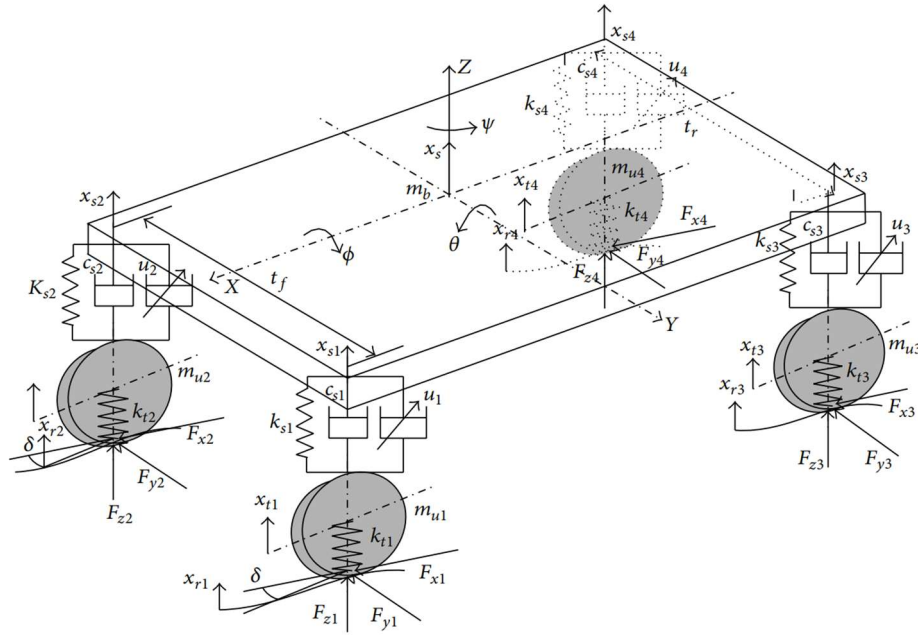


Figure 2.15: Diagram of 9 DOF vehicle model [34]

Assuming that the tires maintain real-time contact with the road, and ignoring the effects of air resistance, road slope, and inclination, the parameters of tire location and suspension deformation on handling stability are not considered. The 9 DOF vehicle dynamics equations using the Newton's second law is shown as:

$$m(\dot{u} - v\dot{\psi}) = (F_{x1} + F_{x2})\cos\delta - (F_{y1} + F_{y2})\sin\delta + F_{x3} + F_{x4} \quad (2.22)$$

$$m(\dot{u} - v\dot{\psi}) = (F_{y1} + F_{y2})\cos\delta - (F_{x1} + F_{x2})\sin\delta + F_{y3} + F_{y4} \quad (2.23)$$

$$I_{b,x}\ddot{\psi} = h_r[(F_{y1} + F_{y2})\cos\delta - (F_{x1} + F_{x2})\sin\delta + F_{y3} + F_{y4}] + m_b g h_0 \sin\phi + (F_{s1} - F_{s2})\frac{t_f}{2} + (F_{s3} - F_{s4})\frac{t_r}{2} \quad (2.24)$$

$$I_{b,y}\ddot{\theta} = h_p[(F_{x1} + F_{x2})\cos\delta + (F_{y1} + F_{y2})\sin\delta + F_{x3} + F_{x4}] + m_b g h_0 \sin\theta - a(F_{s1} + F_{s2}) + b(F_{s3} + F_{s4}) \quad (2.25)$$

$$m_b \cdot \ddot{x}_s = \sum_{i=1}^4 F_{si} \quad (2.26)$$

$$m_{ui}\ddot{x}_{ti} = k_{ti}(x_{ri} - x_{ti}) + c_{ti}(\dot{x}_{ri} - \dot{x}_{ti}) - F_{si} \quad (i = 1,2,3,4) \quad (2.27)$$

$$I_{wi}\dot{\omega}_{wi} = F_{xi}R_i - T_{bi} \quad (i = 1,2,3,4) \quad (2.28)$$

where F_{xi} and F_{yi} ($i = 1,2,3,4$) are the tire longitudinal and lateral force, respectively, which are calculated by magic formula of nonlinear tire model [35]. F_{si} is vertical loads of the suspension and equations are shown as:

$$F_{s1} = k_{s1}(x_{t1} - x_{s1}) + c_{s1}(\dot{x}_{t1} - \dot{x}_{s1}) - \frac{k_{\phi f}}{2t_f}\left(\phi - \frac{x_{t1} - x_{s1}}{2t_f}\right) \quad (2.29)$$

$$F_{s2} = k_{s2}(x_{t2} - x_{s2}) + c_{s2}(\dot{x}_{t2} - \dot{x}_{s2}) + \frac{k_{\phi f}}{2t_f}\left(\phi - \frac{x_{t2} - x_{s2}}{2t_f}\right) \quad (2.30)$$

$$F_{s3} = k_{s3}(x_{t3} - x_{s3}) + c_{s3}(\dot{x}_{t3} - \dot{x}_{s3}) + \frac{k_{\phi r}}{2t_r}\left(\phi - \frac{x_{t3} - x_{s3}}{2t_r}\right) \quad (2.31)$$

$$F_{s4} = k_{s4}(x_{t4} - x_{s4}) + c_{s4}(\dot{x}_{t4} - \dot{x}_{s4}) - \frac{k_{\phi r}}{2t_r}\left(\phi - \frac{x_{t4} - x_{s4}}{2t_r}\right) \quad (2.32)$$

where $u, v, \psi, \theta, \phi, \delta, \omega_{wi}$ are vehicle longitudinal velocity, lateral velocity, yaw angle, pitch angle, roll angle, steering wheel angle, and the wheel angle speed separately. m is the total vehicle mass, m_b is sprung mass, m_{ui} is unsprung mass, $I_{b,x}$ and $I_{b,y}$ are rotational inertia about the x and y axis separately. I_{wi} is the rotational inertia of the wheels, $k_{s,i}$ and $c_{s,i}$ are suspension stiffness and damping coefficient, k_{ti} is the tire radial stiffness, c_{ti} is tire damping, x_s is the vertical displacement of vehicle body, $x_{s,i}$ is the vertical displacement of suspension, $x_{t,i}$ is the vertical displacement of tires. $k_{\phi f}$ and $k_{\phi r}$ are the front and rear suspension roll stiffness, R_i is the wheel's effective rolling radius, a and b are distances from CG to the front and rear axle, t_f and t_r are the distances between the front wheels and rear wheels,

h_0 is the height of CG, h_r is the roll center height, h_p is pitch center height, F_{xi} and F_{yi} are the tire longitudinal and lateral force separately. Lastly, T_{bi} is the wheel braking torque.

2.10 Adams Car Model

Multibody simulation software package (MSSP), like Adams Car, is a specialized software tool used for simulating and analyzing the dynamics of vehicles. MSSP allows engineers to build and test virtual prototypes of vehicle subsystems and complete vehicles within a comprehensive simulation environment. By leveraging multibody dynamics, the software simulates the movement and interaction of vehicle components under various conditions, such as driving maneuvers, road surfaces, and different loading scenarios. This enables detailed analysis of the vehicle's performance and behavior.

One of the standout features of MSSP is its ability to handle models with many degrees of freedom, allowing for complex and realistic simulations of vehicle dynamics without requiring users to develop the underlying mathematical models themselves. This user-friendly interface allows engineers to define key parameters like suspension hard points, mass, inertia, and spring and damping coefficients. The software then automatically manages the dynamic behavior, making it accessible even to those who may not be experts in dynamics. The applications of MSSP are vast, ranging from vehicle handling and stability analysis to ride comfort studies.

The key development effort for this research is invested into creating the full vehicle model. This full vehicle model is used to replicate the computer aided design (CAD) model of the 2023 Cadillac LYRIQ that was donated to UWAF. In depth model development process is covered in the next chapter. Figure 2.15 shows the example of a full vehicle model built for previous UWAF's Camero.

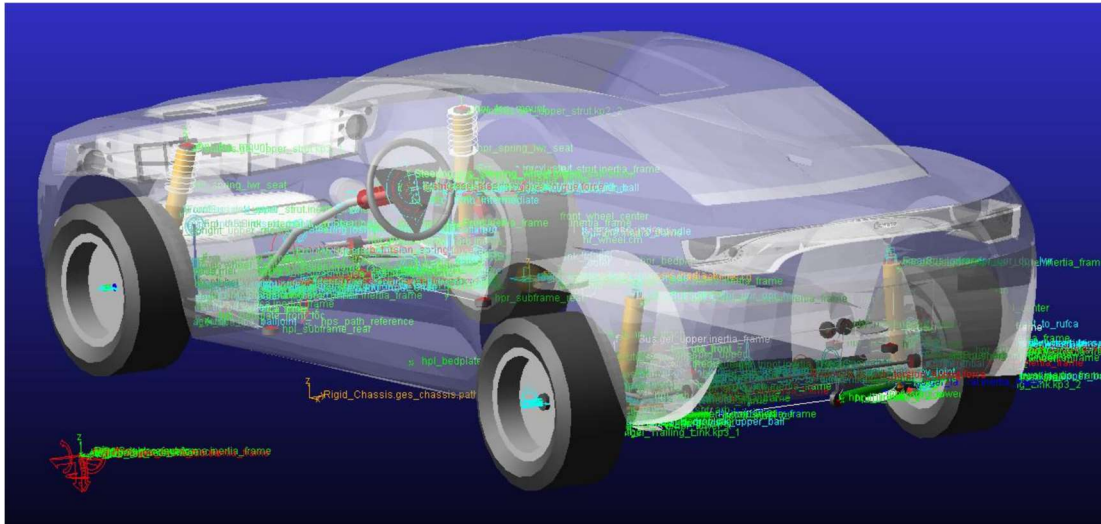


Figure 2.16: MSSP full vehicle model built for the UWFT Camero showing hardpoints in green and modeled suspension components

Chapter 3

Model Development

The following section of the thesis will outline the in depth process of developing UWAFY LYRIQ's full vehicle MSSP model. Firstly, the known vehicle geometric data and suspension force parameters will be presented. Secondly, the creation of the front and rear suspensions with the known parameters will be shown with each of it validated separate to each other. Lastly, the assembly of the full vehicle model will be presented, and vehicle dynamics simulation test methodologies will be discussed.

3.1 Vehicle Features

The donated 2023 Cadillac LYRIQ is a high-end electric sports utility vehicle (SUV) designed to offer exceptional handling and ride comfort for both drivers and passengers [36]. This luxury SUV is equipped with advanced five-link independent front and rear suspension systems. These systems are engineered to provide precise control over the vehicle's dynamic motion, ensuring stability and comfort during various driving conditions, including sharp turns and uneven road surfaces.

The specific model donated is the rear-wheel-drive (RWD) variant, featuring an electric motor that powers the rear wheels. This motor is capable of generating impressive peak power and torque, with 340 horsepower and 325 lb.-ft., respectively [36]. This combination of power and advanced suspension technology ensures that the vehicle delivers a smooth and responsive driving experience, characteristic of Cadillac's commitment to luxury and performance. Figure 3.1 shows a photo of the 2023 Cadillac LYRIQ.



Figure 3.1: 2023 Cadillac LYRIQ

In alignment with the EcoCAR EV Challenge competition, UWAFI has developed and implemented an in-house electric powertrain to transform the vehicle from RWD to all-wheel drive (AWD). This transformation was achieved by integrating two foreign electric drive units into the front and rear axles of the vehicle, creating a robust AWD system. This custom electric powertrain has significantly boosted the total power output of the vehicle.

The addition of an electric drive unit to the front axle has notably altered the vehicle's weight distribution compared to the stock RWD model. This shift in weight distribution can affect various aspects of vehicle dynamics, including handling, stability, and performance during different driving conditions. Table 3.1 shows the summary of the real-life vehicle measurement data comparing the stock and UWAFI vehicle.

Table 3.1: Summary of vehicle parameter comparison of stock to UWAFI vehicle

Parameter	Stock	UWAFI
Total vehicle curb weight [kg]	2368	2497
Weight distribution [% front loaded]	50.75	54.31
Drivetrain configuration	RWD	AWD

To thoroughly understand the implications of this weight distribution change, UWAFI has employed the MSSP model. This advanced simulation tool allows the team to analyze and predict the vehicle's

dynamic behavior under various scenarios. By simulating the modified vehicle in MSSP, UWAF can assess how the added weight and altered distribution impact the handling and stability of the vehicle. The model enables detailed examination of the vehicle's responses to different inputs, helping the team fine-tune the suspension to accommodate the new powertrain configuration. This process ensures that the modified vehicle meets the competition's performance and safety standards while maintaining the high level of innovation and engineering excellence expected in the EcoCAR EV Challenge.

OEM sponsor has provided detailed CAD model for the team to utilize for custom powertrain design and integration work. This CAD model is used to gather the hardpoints of the suspension elements and geometry of the parts. Due to the confidentiality of the data, they are not presented in this thesis in detail. Figure 3.2 shows the result of transformation of the hardpoints from the CAD model to MSSP model for the front suspension [37]. OEM sponsor's model is not shown due to the confidentiality of the CAD model.

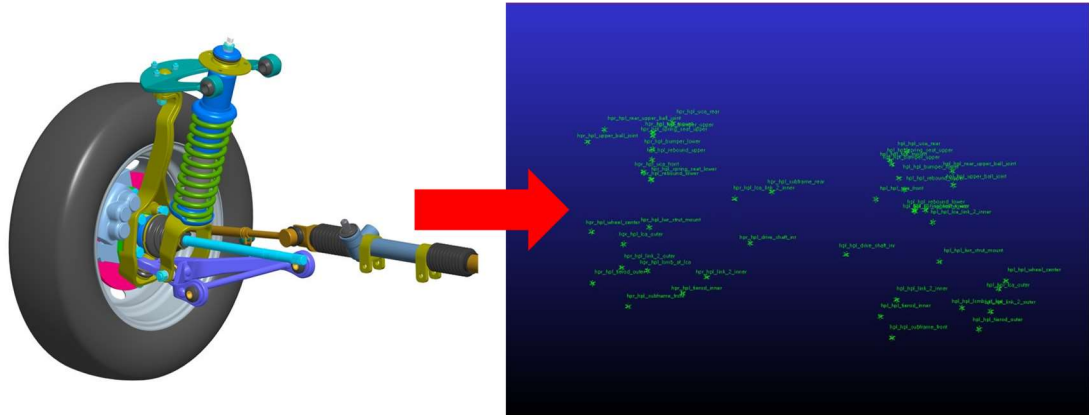


Figure 3.2: Translation of CAD model to MSSP model suspension hardpoints shown in green texts

3.2 Motor

The full vehicle model is developed incorporating fully electric drive units on both the front and rear axles, designed to deliver identical performance characteristics with equal efficiency and power ratings. The motor model for the vehicle was created using detailed data provided by the motor manufacturer, encompassing key performance metrics such as speed, torque, power, and efficiency curves. This model was then adapted to integrate seamlessly with the MSSP, enabling precise simulations of the vehicle's dynamic behavior. A critical component of this model is the motor efficiency curve, which illustrates

the motor's efficiency across various operating conditions; however, to maintain confidentiality, specific axis values are omitted. Figure 3.3 shows an example of an efficiency curve of an automotive traction motor [38]. The actual efficiency curve implemented into the MSSP model is omitted due to the confidentiality reason with the motor provider.

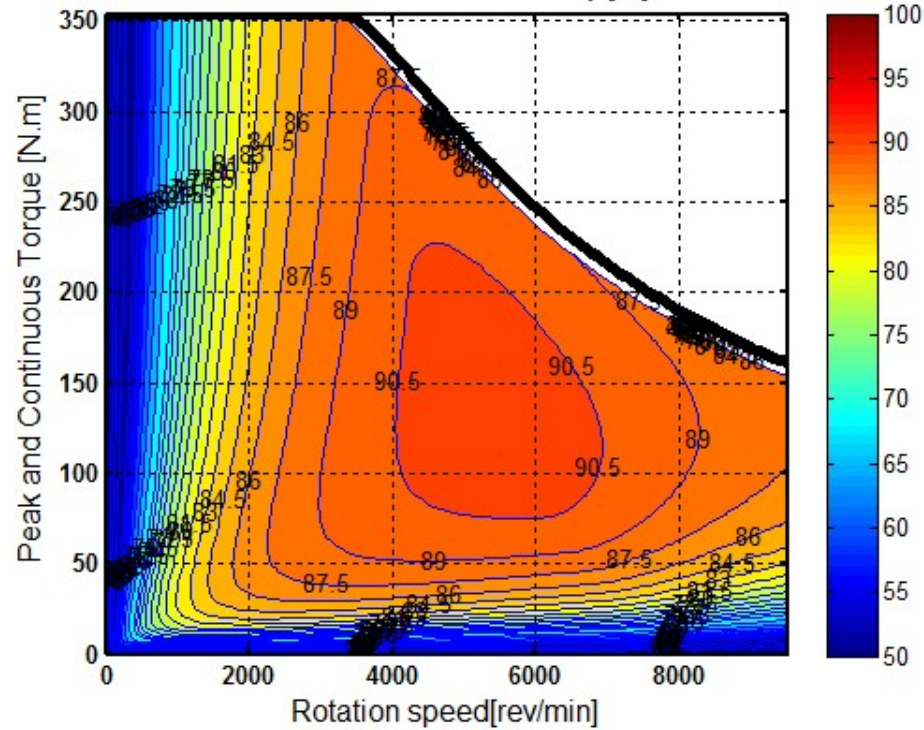


Figure 3.3: Example of an electric motor efficiency curve for electric drive unit [38]

Figure 3.4 shows the Exagon Furtive e-GT's electric drive motor's torque and power vs. speed graphs [39]. The motor data UWAF is using is omitted for the confidentiality. The detail values are omitted again for the confidentiality.

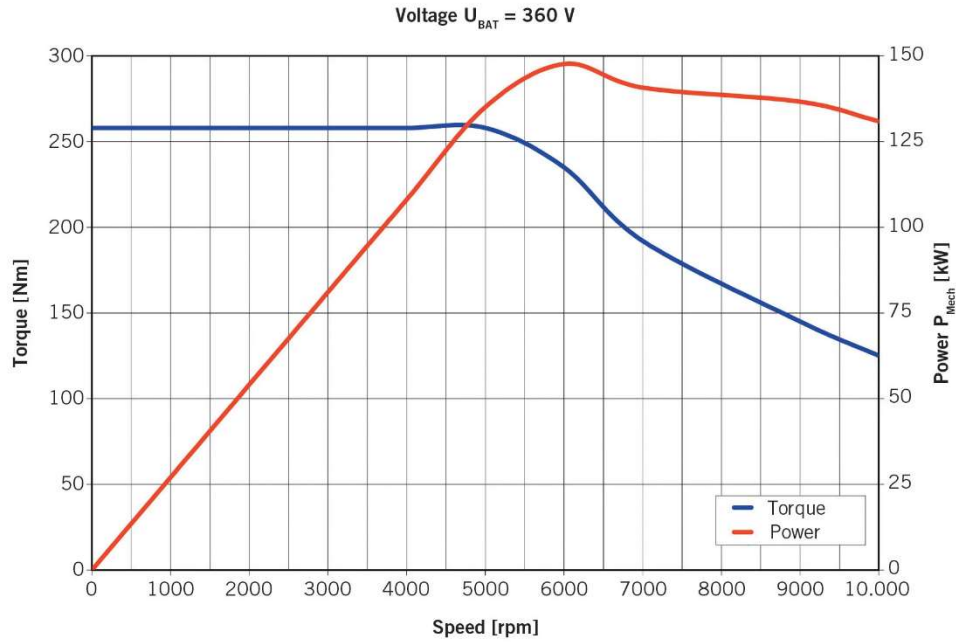


Figure 3.4: Example of a motor torque and power vs. speed graph of an automotive traction motor [39]

3.3 Bushing

Suspension bushing is a critical component in the suspension assembly, especially in five-link suspension design. Bushing is a component that allows the relative movement of the suspension arms and knuckles and prevents small amplitudes vibrations from the road irregularity to transferred to the vehicle chassis [40]. It is a critical component in tuning the overall vehicle suspension characteristic for handling and ride comfort performances. Data for all suspension bushing was not available from OEM sponsor for implementation to the model, therefore original templates bushing templates from MSC Software is utilized to build the suspension model. Figure 3.5 shows the sign convention of the suspension bushing and its force and torque directions. Figure 3.6 and 3.7 shows an example of a bushing property integrated to the suspension model and its compliance parameters.

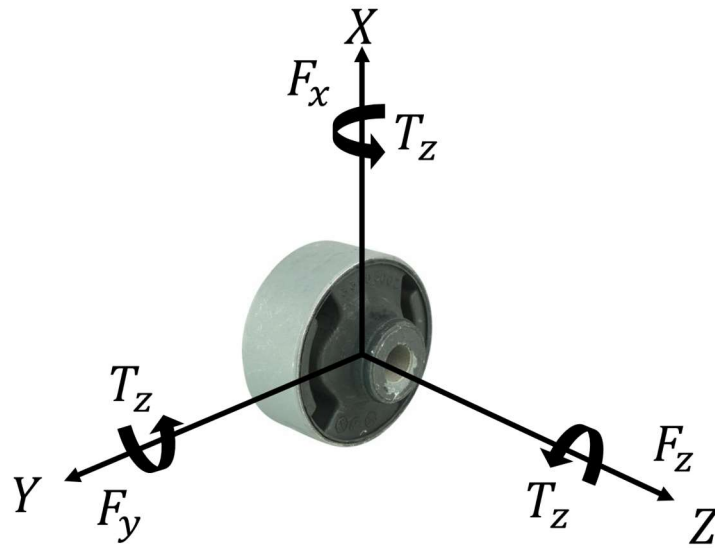


Figure 3.5: Suspension bushing sign convention. Picture of the bushing [41]

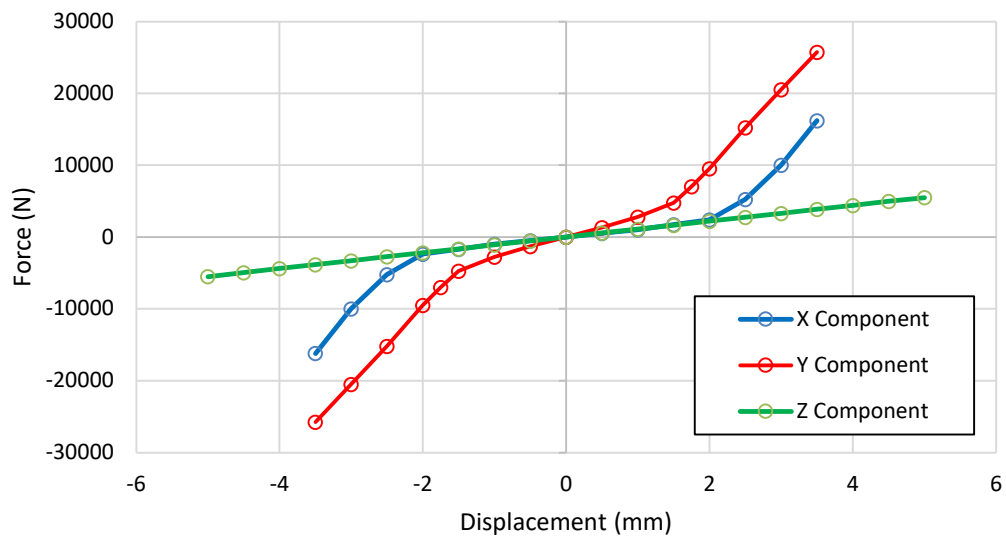


Figure 3.6: Translational bushing stiffness for inner front upper control arm

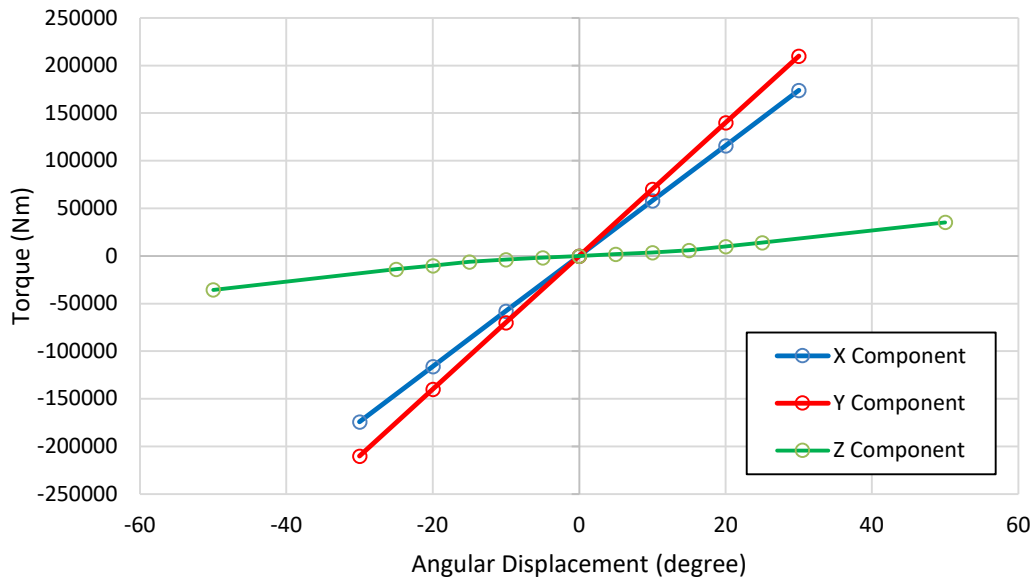


Figure 3.7: Torsional bushing stiffness for inner front upper control arm

The front suspension alone has nine unique bushings with each having its own translational and torsional characteristics. A summary of the bushings used for the front and rear suspension is outlined in Table 3.2.

Table 3.2: Summary of bushing used for front and rear suspension

Bushing Name	Parts connecting	
	i Part	j Part
uca_link_2_inner	Upper rear control arm	Chassis
uca_inner	Upper front control arm	chassis
top_mount	Spring and damper upper mount	Chassis
tierod_inner	Tierod inner	Steering rack output
Subframe_rear	Suspension subframe rear mount point	Chassis
Subframe_front	Suspension subframe front mount point	Chassis
Lwr_strut	Damper lower mount point	Front lower control arm
Link_2_inner	Lower front control arm	Subframe
Lca_link_2_inner	Lower rear control arm	Subframe

3.4 Spring and Damper

Springs and dampers are the primary force-generating elements in a vehicle's suspension assembly. These components play a crucial role in supporting the vehicle's sprung mass and ensuring that the wheels maintain optimal contact with the road surface, which is vital for both handling and ride comfort. Springs are designed to vertically support the vehicle's weight. They act as the primary load-bearing component, absorbing the vertical forces generated by the vehicle's mass and any additional loads. By compressing and decompressing in response to road conditions, springs help to maintain the vehicle's ride height and prevent excessive body roll during cornering. Dampers, often referred to as shock absorbers, work alongside the springs to control the suspension's motion. They dissipate the kinetic energy generated by the vertical movement of the suspension through hydraulic resistance. Inside the damper, fluid flows through orifices or valves, creating resistance that converts kinetic energy into heat, which is then dissipated. This process helps to dampen the oscillations of the springs, ensuring that the vehicle returns to a stable position quickly after encountering a bump or dip in the road.

The interplay between springs and dampers is critical for vehicle dynamics. Springs provide the necessary support to keep the vehicle stable, while dampers control the rate of suspension movement, preventing excessive bouncing and improving ride quality. This combined functionality is essential for maintaining strong tire contact with the road, which is crucial for handling, braking, and overall vehicle stability. In Chapter 2, it was presented how these components impact the system forces within the suspension through various vehicle dynamics models.

The spring and damper characteristics are represented using a lookup table and subsequently inputted into the model. The spring is modeled with a linear characteristic, taking into account the integration of bump stops and rebound stops to manage maximum compression and rebound scenarios. To simplify the analysis of the impact of weight distribution changes, it is assumed that the front and rear springs have identical characteristics for this research. These parameters were derived from the spring and damper models provided by the manufacturers. Figures 3.8 and 3.9 illustrate the spring and damping forces as functions of displacement and velocity, respectively, for the suspension system.

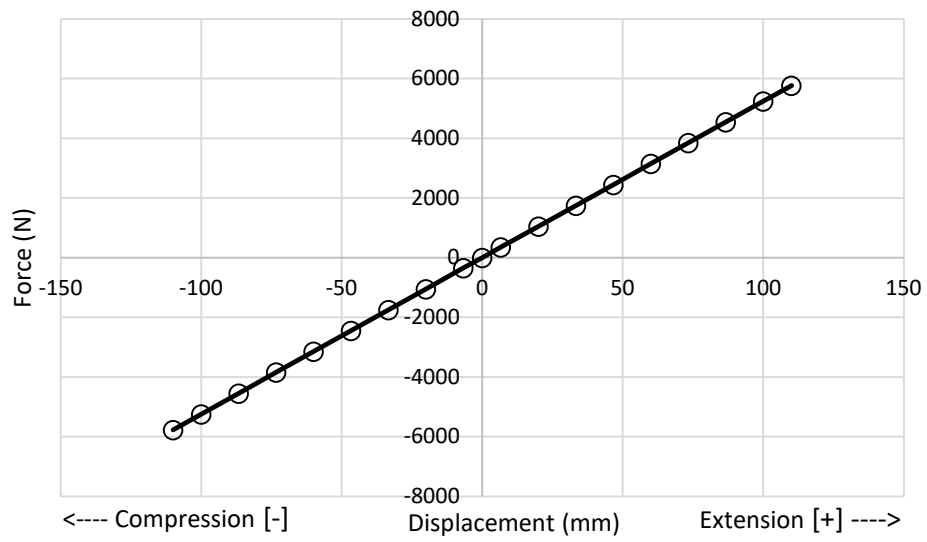


Figure 3.8: Force vs. Displacement graph for front and rear spring

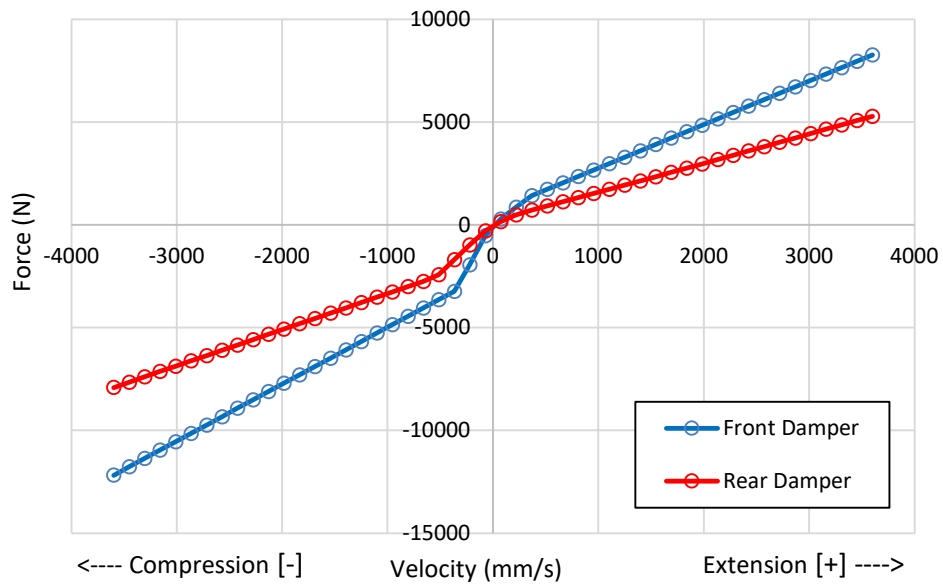


Figure 3.9: Force vs. Velocity graph for front and rear damper

3.5 Bump stop and Rebound stop

Bump stops are critical components in vehicle suspension systems designed to limit the maximum compression of the suspension. They play a crucial role in preventing the suspension from bottoming out, which occurs when the suspension components are compressed to their limit. This can potentially cause damage to the vehicle or discomfort to the passengers. Typically made from materials such as rubber or urethane, bump stops are mounted on various suspension components, such as the chassis or control arms. In 2023 Cadillac LYRIQ, the bump stops are integrated to the damper assembly carried by the piston rod. When the suspension compresses beyond a certain point, the bump stop absorbs the excess energy, thereby reducing the impact force transmitted to the vehicle structure. This not only protects the suspension components and vehicle chassis from damage but also helps maintain ride comfort and enhances handling performance.

Rebound stops serve to limit the maximum extension or rebound of the suspension. These components are essential in preventing the suspension from overextending, which can occur when the vehicle's wheels leave the ground or when there is a rapid change in road surface. Like bump stops, rebound stops are typically made from durable materials such as rubber or urethane and are strategically placed to limit the downward travel of suspension components. By doing so, they protect the suspension components and vehicle frame from damage due to excessive extension. Moreover, rebound stops contribute significantly to vehicle stability by ensuring that the wheels maintain contact with the road, thus improving handling. They also enhance ride quality by preventing harsh suspension rebound, which can cause a jolting sensation for the passengers.

The bump and rebound stops exhibit a quadratic relationship between force and deformation. For the purposes of integration into the model, it was assumed that the bump and rebound stops possess identical force versus deformation characteristics due to the unavailability of precise real-world data at the time of model creation. This assumption simplifies the modeling process while providing a reasonable approximation of the stops' behavior. Figure 3.10 illustrates the force versus deformation graph for the bump and rebound stops, demonstrating their quadratic nature.

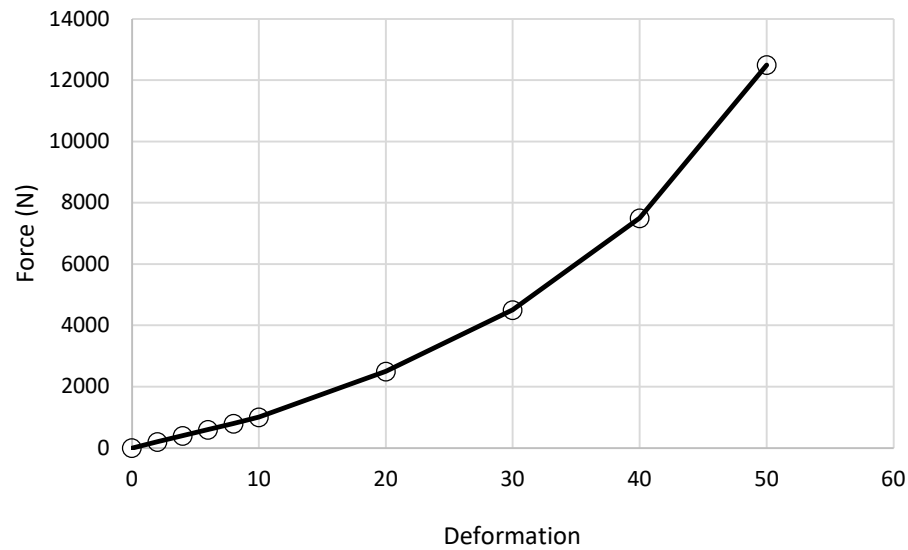
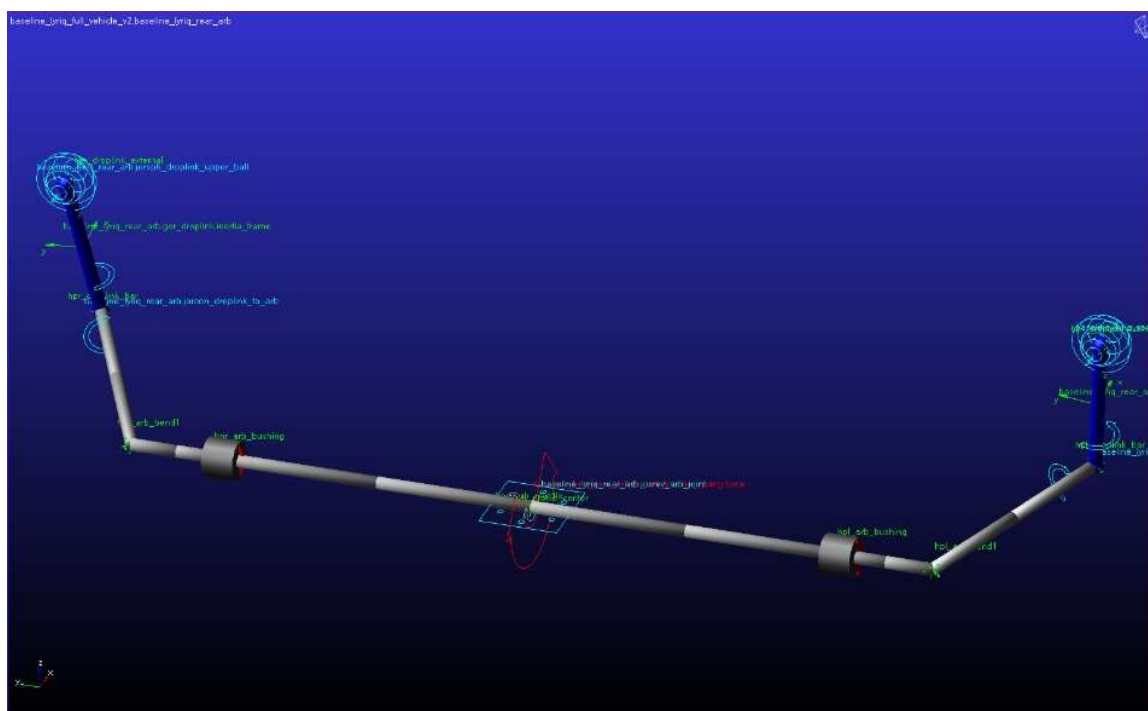
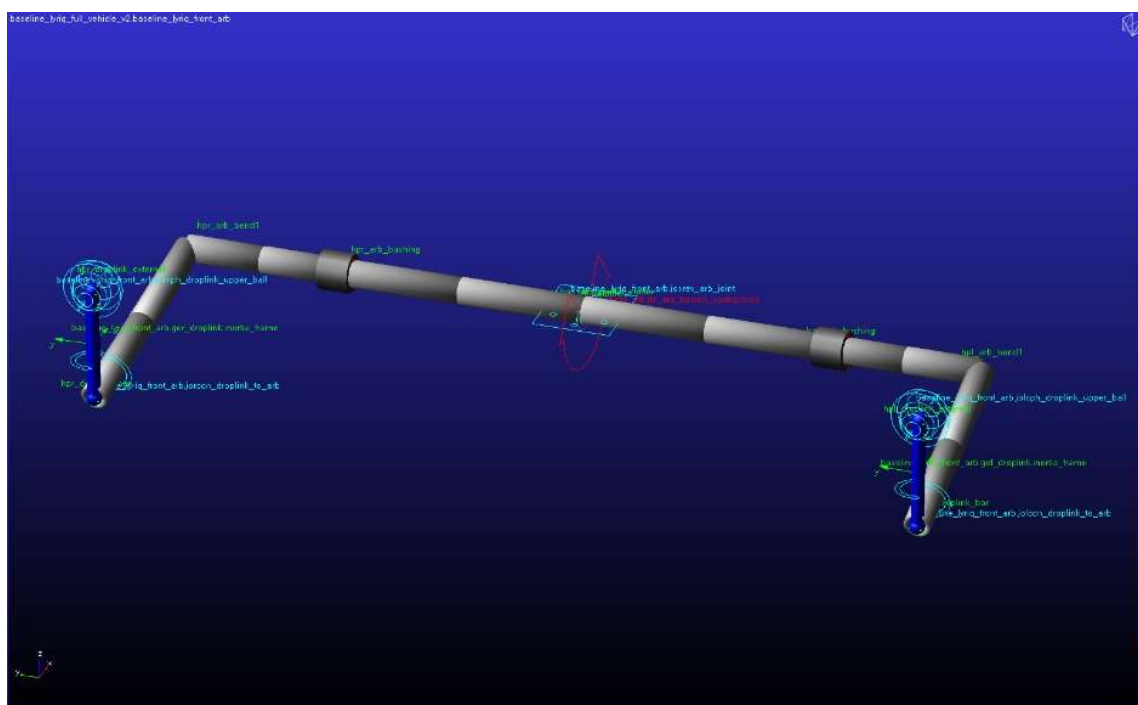


Figure 3.10: Force vs. Deformation graph of bump and rebound stops

3.6 Steering System

The steering system in this model is based on the standard rack and pinion steering system template provided in MSSP. While the stock vehicle is equipped with an electric power steering assist system, this feature has been omitted in the simulation for simplicity. The steering wheel is connected to the upper column through a cylindrical joint. This upper column is then connected to the intermediate shaft via a Hooke joint, which in turn is connected to the steering shaft through another Hooke joint. The steering shaft drives the pinion, which ultimately actuates the steering rack. This configuration provides an accurate and responsive steering mechanism, effectively replicating the mechanical linkages and dynamics inherent in a conventional rack and pinion steering system. Figure 3.11 shows the steering model representing vehicle's steering system.



3.8 Front Suspension

By integrating the detailed characteristics of the bushings, springs, dampers, bump stops, and rebound stops into the model, a comprehensive front suspension model has been developed. This front suspension model was subsequently subjected to parallel and opposite wheel travel tests to validate the dynamic movements of the interconnected components, ensuring their proper function before full vehicle assembly. Figure 3.14 depicts the fully assembled front suspension model with its dynamic motion successfully validated, demonstrating the effectiveness of the integration and testing process.

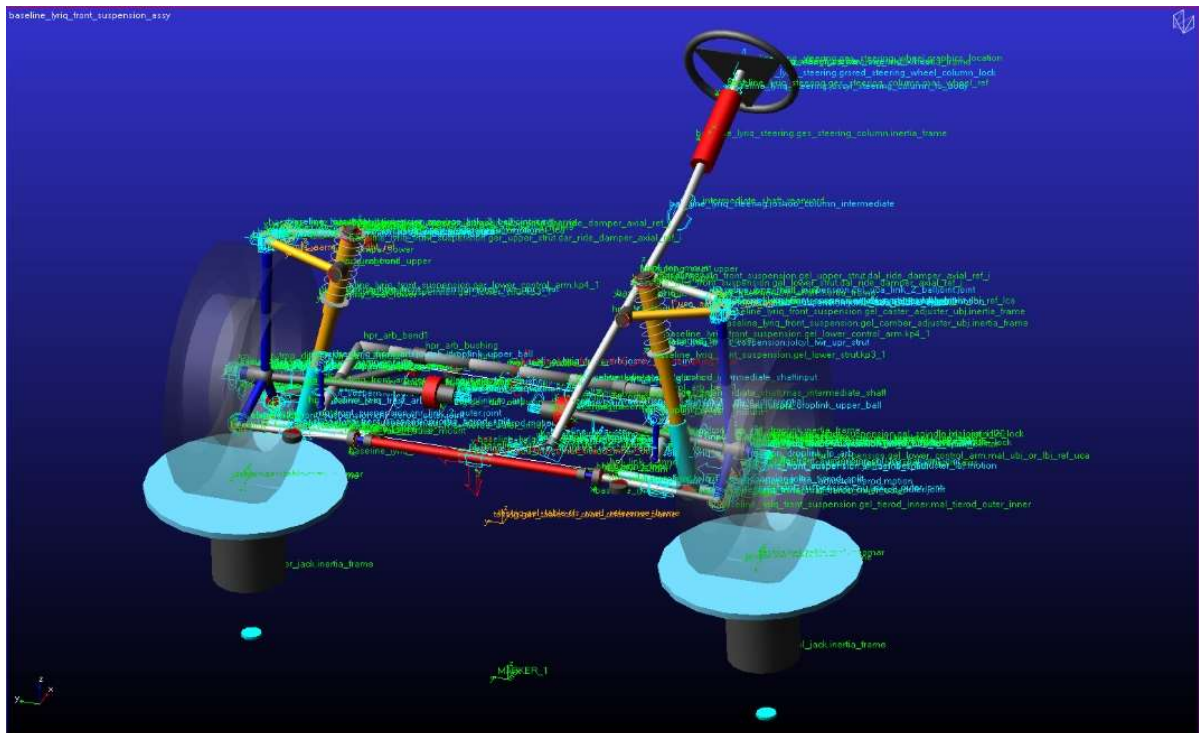


Figure 3.14: MSSP front suspension model assembly with the test rig setup

As depicted in Figure 3.14, the front suspension assembly is configured with test rigs in the model. These test rigs facilitate parallel and opposite wheel movement tests, enabling the observation of component movements within the assembly and providing various output measurements to validate the assembly. To ensure accurate validation, parallel and opposite wheel travel tests were conducted with displacements of +45 mm and -45 mm for bump and rebound relative to the wheel center. Figures 3.15 and 3.16 demonstrate that the model operates as intended, with all components moving correctly in relation to one another.

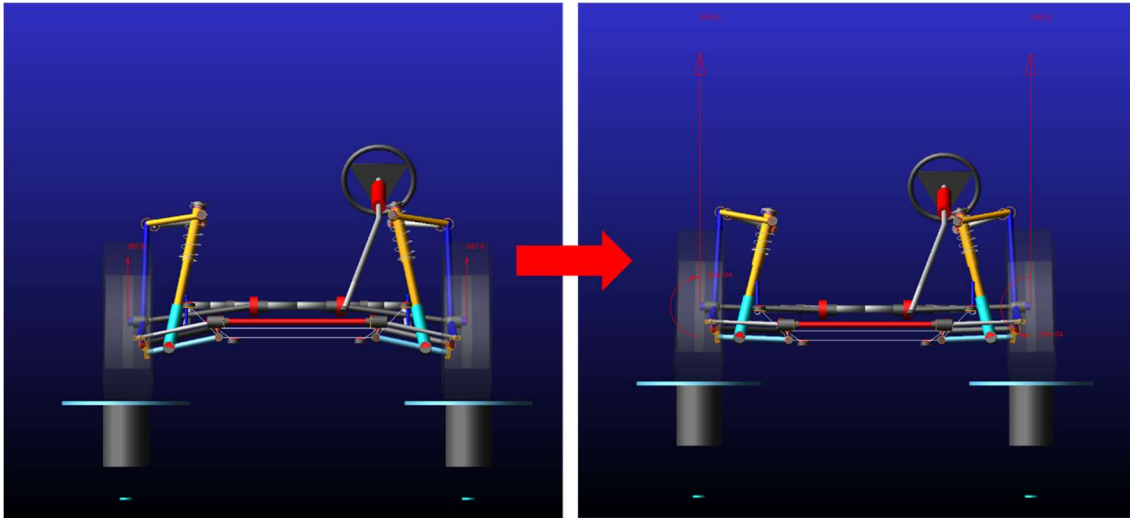


Figure 3.15: Front suspension parallel wheel travel simulation where right picture shows the vertical wheel movement and forces generated by the suspension components

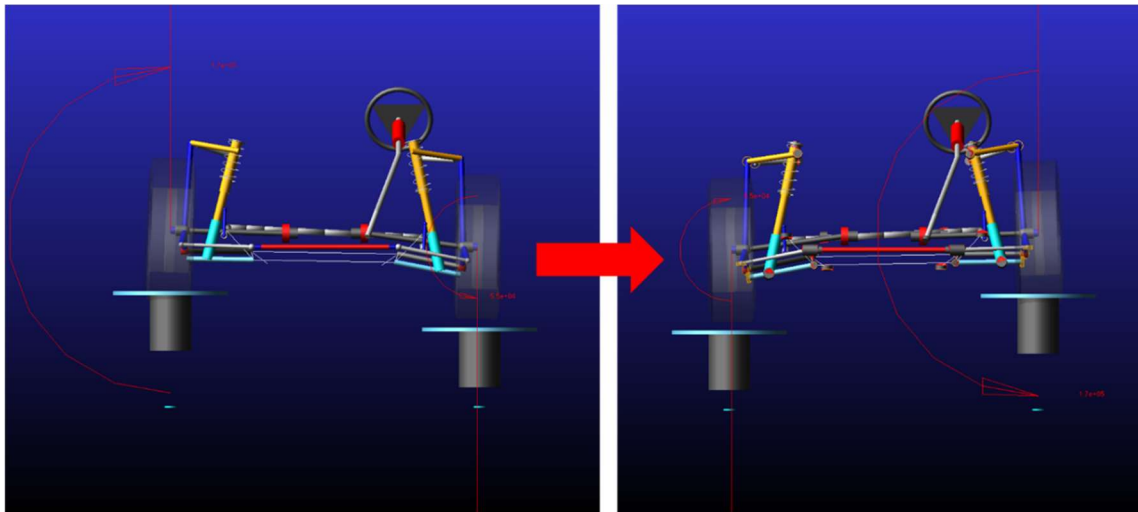


Figure 3.16: Front suspension opposite wheel travel simulation showing the separate movement of the left and right wheel assembly and forces generated by the components

3.9 Rear Suspension

The rear suspension model is constructed in a manner similar to that of the front suspension. However, because the rear suspension lacks steering functionality, the steering system is not integrated into this model. Instead, the rear suspension includes half shafts, as it serves as the driven axle in the stock vehicle powertrain configuration. Additionally, in this setup, the spring is positioned separately from

the damper, being connected to both the chassis and the rear lower control arm. Figure 3.17 illustrates the fully modeled rear suspension assembly, highlighting the configuration and integration of these components.

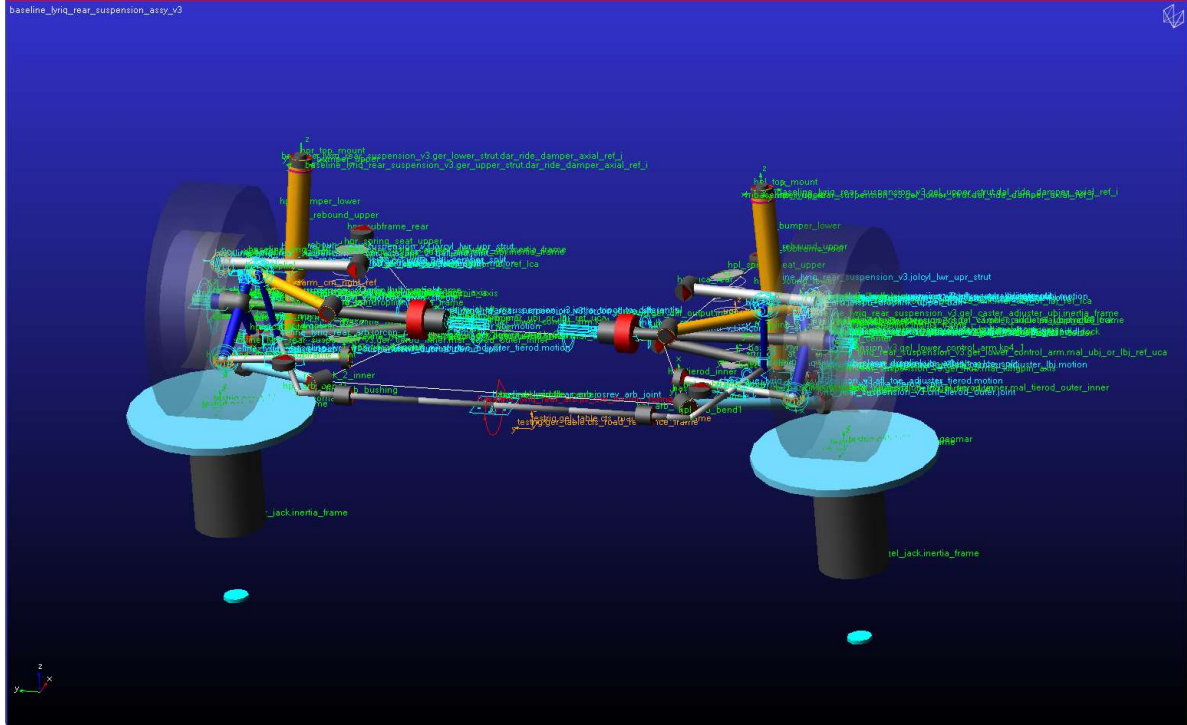


Figure 3.17: MSSP rear suspension model assembly with the test rig setup

Following the methodology employed for the front suspension model, the rear suspension model was subjected to analogous parallel and opposite wheel travel simulations to validate the model's outputs and dynamic responses. Figures 3.18 and 3.19 present the results of these simulations, demonstrating the accuracy and reliability of the rear suspension model in replicating the expected behaviors under varying conditions.

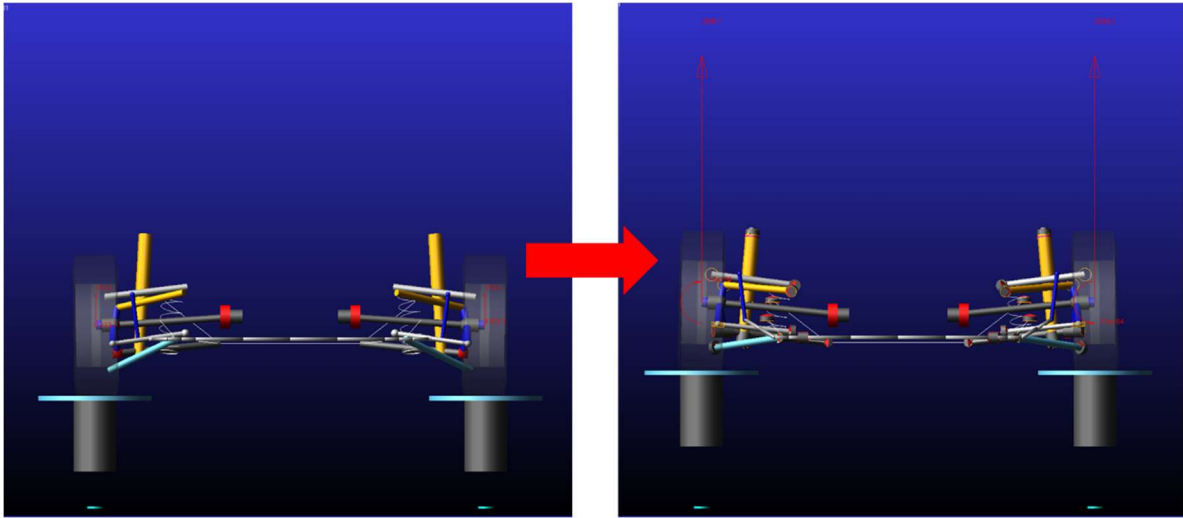


Figure 3.18: Rear suspension parallel wheel travel simulation showing the vertical and camber forces generated by the suspension components

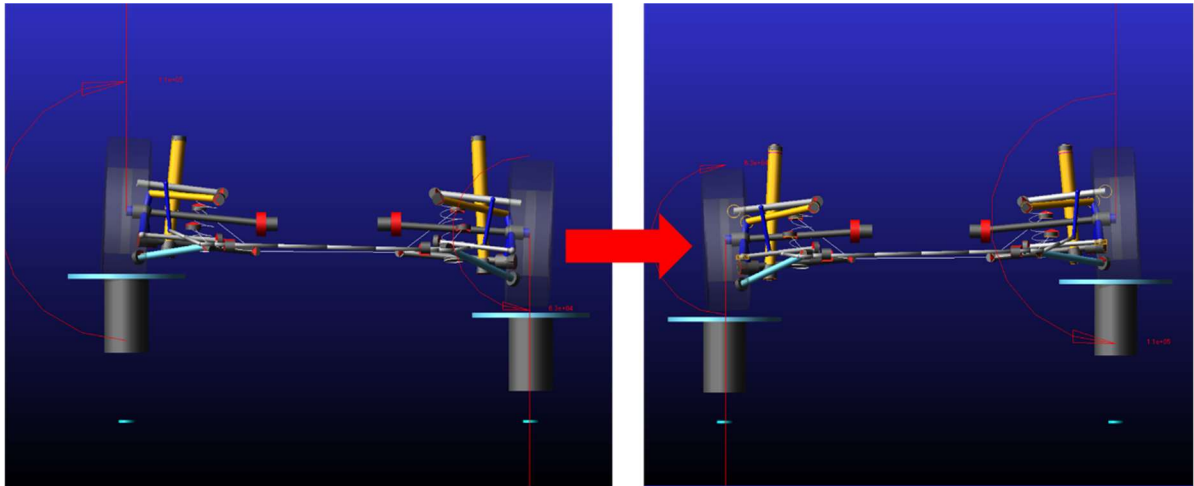


Figure 3.19: Rear suspension opposite wheel travel simulation showing separate movement of the left and right wheel assembly with forces generated by suspension components

3.10 Chassis

The chassis system of the vehicle is modeled as a single rigid body, which defines the location of all the sprung mass components within the vehicle's coordinate system. The CG of the chassis system will be fine-tuned during the creation of the full vehicle model. This adjustment is crucial for accurately modeling the vehicle's corner balance, ensuring that the distribution of weight among the vehicle's

corners is properly represented and optimized for performance and handling characteristics. Figure 3.20 shows the vehicle's chassis model in MSSP.



Figure 3.20: MSSP chassis model showing the location of CG in yellow sphere

3.11 Tire

Tires are intricately engineered products, and engineers have developed various sophisticated tire models for vehicle dynamics simulations. These tire models function as best-fit curves that approximate the tire behavior characteristics derived from physical test data. This is because tires behave as nonlinear elastic structures with small hysteresis due to rate-dependent energy losses. Although it is feasible to model the interaction between the tire and the surface using finite element analysis, such simulations require substantial computational power due to their complexity. Consequently, simplified models are often employed to balance the need for accuracy with computational efficiency in vehicle dynamics simulations.

The tire model employed for the MSSP model is the PAC2002 tire model. Developed by Adams and based on the Magic Formula Tire model [43], this model is renowned for its precision in describing tire behavior on smooth roads and is widely utilized within the automotive industry [44]. The PAC2002

tire model accounts for various complex tire dynamics, including camber thrust, non-zero forces and moments at zero slip angle, and varying lateral and longitudinal tire stiffness with respect to normal load. Additionally, it can represent combined lateral and longitudinal slip, providing a comprehensive and accurate simulation of tire performance under diverse conditions.

The formulation of the PAC2002 tire model is grounded in a trigonometric curve fitting approach utilizing six coefficients: B, C, D, E, S_h , and S_v . These coefficients are integral to the Magic Formula, which describes the tire's force and moment characteristics with high accuracy. The fundamental equations of the PAC2002 tire model are delineated as follows [44]:

$$y(x) = D \sin[C \arctan\{Bx - E(Bx - \arctan(Bx))\}] \quad (3.1)$$

$$y(x) = D \cos[C \arctan\{Bx - E(Bx - \arctan(Bx))\}] \quad (3.2)$$

$$x = X + S_h \quad (3.3)$$

$$Y(X) = y(x) + S_v \quad (3.4)$$

where, $Y(X)$ represents the forces or moments to be determined as a function of X , which can be either longitudinal slip or slip angle. The variables y and x are utilized as intermediate variables to account for the vertical offset S_v and horizontal offset S_h of the curves. B represents the stiffness factor, C represents the shape factor, D represents the peak factor, and E represents the curvature factor. The coefficients B, C, D, E, S_h , and S_v are derived using equations proposed by Pacejka, which incorporate a comprehensive set of parameters related to the tire's properties. Due to their complexity and length, these equations and parameters are not included in this thesis. Figure 3.21 shows the PAC2002 tire model implemented to the MSSP model.

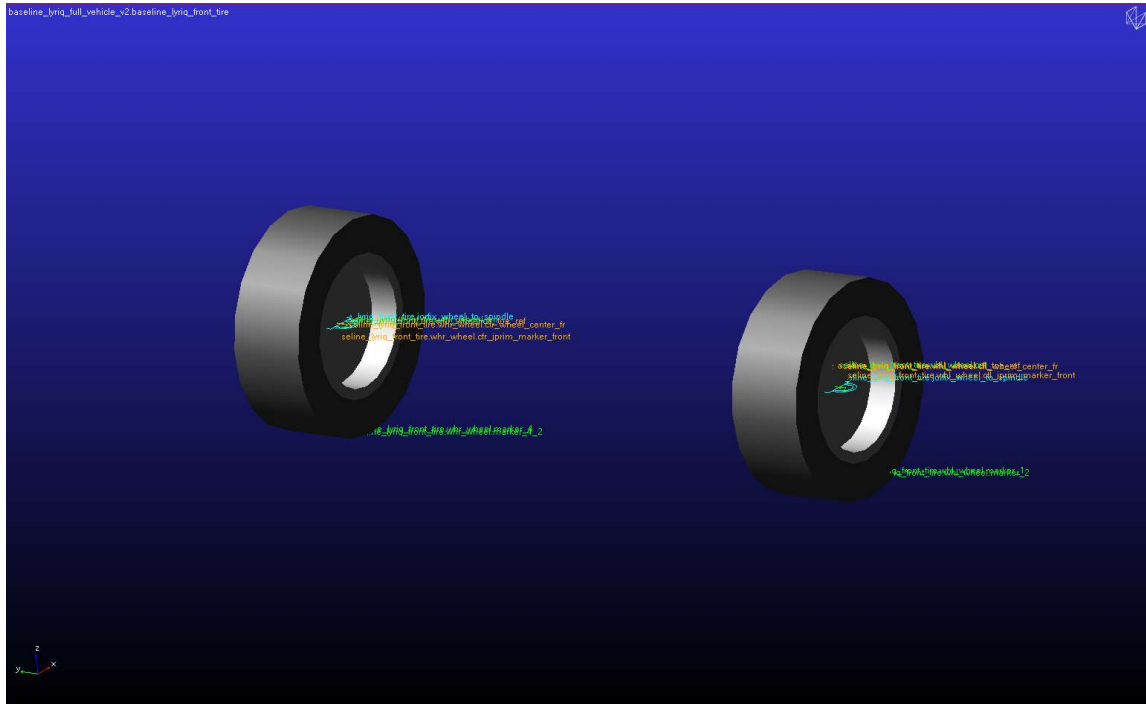


Figure 3.21: MSSP tire model showing one set of tires for axle implemented with PAC2002 model

3.12 Full Vehicle

With all subsystems, including the front and rear suspensions, modeled and validated, the full vehicle model is assembled in MSSP. The integration of these subsystems is achieved by setting up communicator variables that align with the corresponding components requiring interconnection across different subsystems. It is essential to accurately match these communicator variables to ensure the model successfully passes the equilibrium calculation. This meticulous integration process ensures that the assembled vehicle model accurately reflects the dynamic interactions between its various subsystems. Outline of the components required to assemble the full vehicle model is outlined in Appendix A

Upon assembling the model, initial simulation validations are conducted. These validations encompass the equilibrium test and static vehicle measurement test, both of which are essential for verifying the model's accuracy in terms of overall vehicle geometry and parameters. The static equilibrium analysis is particularly crucial as it checks for any components within the model that do not achieve zero acceleration at time $t = 0$. The MSSP solver facilitates this by perturbing the components with small

displacements or rotations until the entire model achieves zero acceleration within an acceptable error tolerance.

Following the confirmation that the model reaches static equilibrium without errors, a Suspension Parameter Measurement Machine (SPMM) analysis is performed. This analysis measures the kinematic and compliance characteristics of the vehicle, which result from the geometries and compliances of the suspension and steering systems. These compliances include those from suspension springs, anti-roll bars, elastomeric bushings, and component deformations. The SPMM applies known displacements, forces, and torque inputs to measure the static suspension characteristics, which are crucial for understanding ride and handling performance. Figure 3.22 shows the SPMM analysis setup of the full vehicle MSSP model of the stock vehicle.

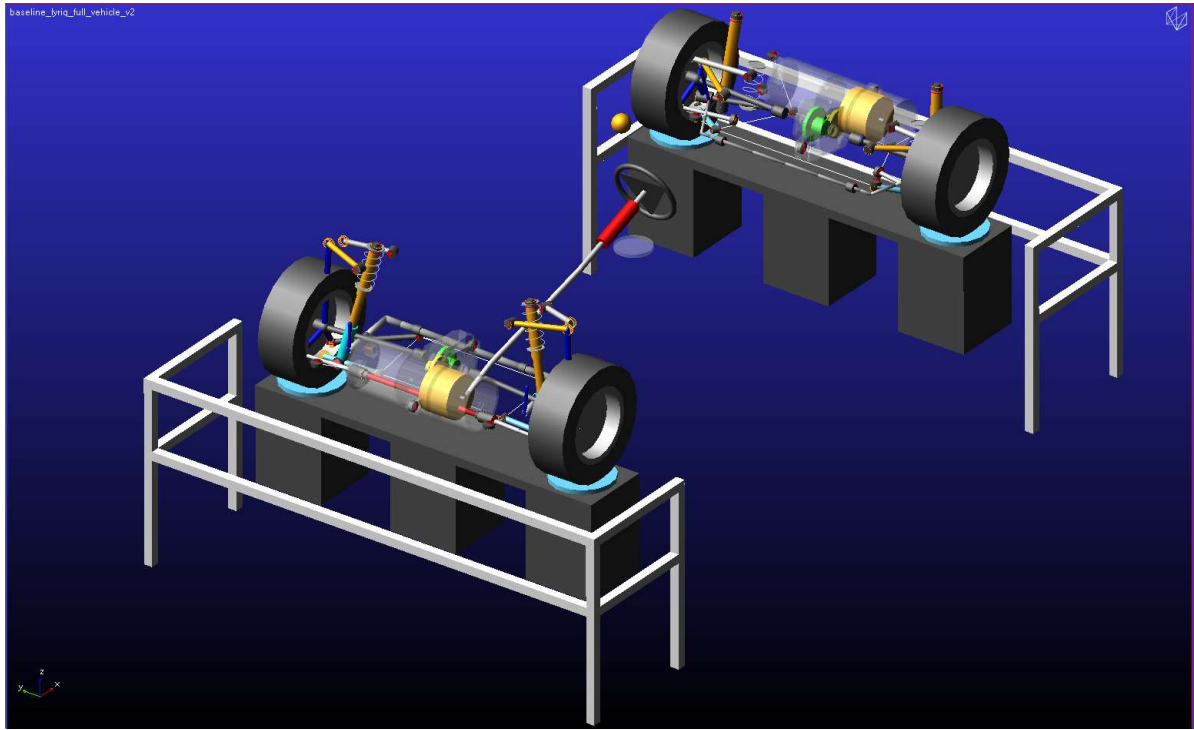


Figure 3.22: MSSP full vehicle model on SPMM test rig

The data from the SPMM analysis was utilized to fine tune the location of the chassis point mass in order to achieve the vehicle's weight distribution data presented in Table 3.2. Table 3.3 details the vehicle parameters output data gathered from SPMM analysis. Full result of the kinematic and compliance test conducted for stock MSSP model with the SPMM test rig is attached in Appendix B.

Table 3.3: Vehicle parameter data gathered from SPMM analysis

Parameter	Value
CG height [mm]	744
Total roll inertia $\left[\frac{kg}{m^2}\right]$	474
Total pitch inertia $\left[\frac{kg}{m^2}\right]$	238
Total yaw inertia $\left[\frac{kg}{m^2}\right]$	245

Chapter 4

Full Vehicle Handling Simulation

This section elucidates the comprehensive vehicle handling simulation based on industry-standard vehicle testing methodologies. To analyze the vehicle handling performance, the road model is configured as a flat surface with a constant friction coefficient. Fishhook and double lane change maneuvers are simulated using the full vehicle models for both the stock and UWAF variants to assess the impact of changes in vehicle weight distribution. The results of these simulations are discussed in this section, providing insights into how the alterations in weight distribution affect the vehicle's handling dynamics.

4.1 Handling Maneuvers

To comprehensively understand vehicle handling performance, the automotive industry has established standard handling maneuvers for vehicle testing. Among these, the most commonly utilized maneuvers are the fishhook and double lane change tests. These maneuvers are highly regarded for their objective repeatability, ease of execution, and their capacity to evaluate a vehicle's maximum handling performance. Furthermore, they provide an accurate representation of real-world driving scenarios, thereby offering valuable insights into the vehicle's dynamic behavior under critical conditions [45].

4.1.1 Fishhook

The Fishhook test standard, developed by the National Highway Traffic Safety Administration (NHTSA), is designed to conduct dynamic rollover tests for motor vehicles as part of a consumer information program [46]. In this research, the test is executed at a target vehicle entrance speed of 80 km/h. The steering input angle is standardized, with both the initial and countersteer steering magnitudes set symmetrically. This standardization is achieved by multiplying the steering wheel angle, which generates an average lateral acceleration of 0.3 g in the Slowly Increasing Steer maneuver, by a scalar of 6.5.

The Slowly Increasing Steer maneuver is a characterization test where the vehicle is driven in a straight line at 80 km/h while maintaining the speed. During this maneuver, the steering wheel is turned linearly to a 270-degree position at a rate of 13.5 degrees per second. The results from this test are utilized to calculate the steering wheel input for the Fishhook maneuver.

The fishhook maneuver begins with the vehicle driven in a straight line at a speed slightly exceeding the target entrance speed. The initial calculated steering input is then applied, followed by a 250-millisecond interval before the countersteer is initiated. The steering wheel rates for both the initial steer and countersteer ramps are set at 720 degrees per second. Figure 4.1 illustrates the steering wheel input profile for the fishhook maneuver.

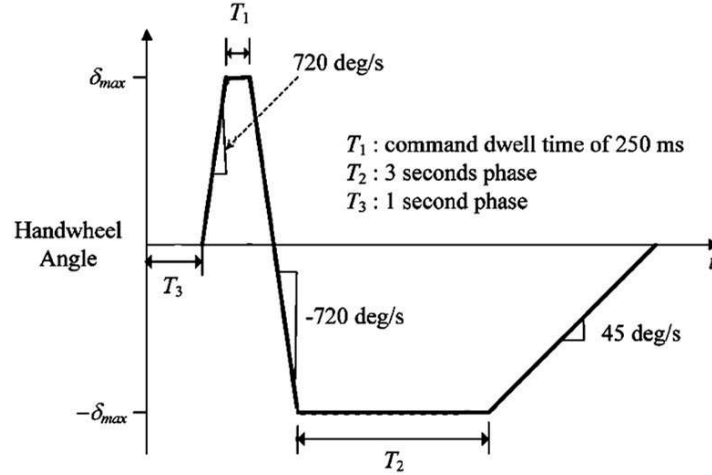


Figure 4.1: Steering input for fishhook maneuver [47]

4.1.2 Double Lane Change

The double lane change maneuver is a standardized vehicle handling test commonly used in the automotive industry to evaluate a vehicle's dynamic stability and handling characteristics. This maneuver simulates an emergency situation where a driver must quickly change lanes to avoid an obstacle and then return to the original lane. The test is designed to assess the vehicle's agility, steering responsiveness, and overall stability during sudden directional changes.

The double lane change maneuver is conducted on a flat track. The vehicle is driven at a constant speed towards the start of the test section. Upon reaching the entry point, the driver executes a rapid lane change to the adjacent lane, followed immediately by a return to the original lane, forming a "S" shaped path. The vehicle's performance during this maneuver is measured and analyzed based on several parameters, including lateral acceleration, yaw rate, and the stability of the vehicle's trajectory.

The vehicle's speed at the entry point is crucial and is typically standardized for consistency. Common entry speeds range from 60 to 80 km/h, depending on the specific requirements of the test. For this research, constant vehicle speed of 80 km/h is used for simulations. The steering wheel angle and rate of input are controlled to ensure reproducibility. The steering input is applied at a constant rate to achieve a precise lane change.

The double lane change maneuver is defined by several international standards, including the ISO 3888-2, which specifies the test setup, procedure, and evaluation criteria. The ISO 3888-2 standard outlines the dimensions of the test course, the required entry speed, and the criteria for assessing the vehicle's performance. Figure 4.2 shows the test course dimensions for ISO 3888-2 double lane change maneuver. The stock vehicle has a width of 1976 mm, so $A = 1.1 \times 1976 + 250 = 2,423.6 \text{ mm}$ and $B = 2976 \text{ mm}$.

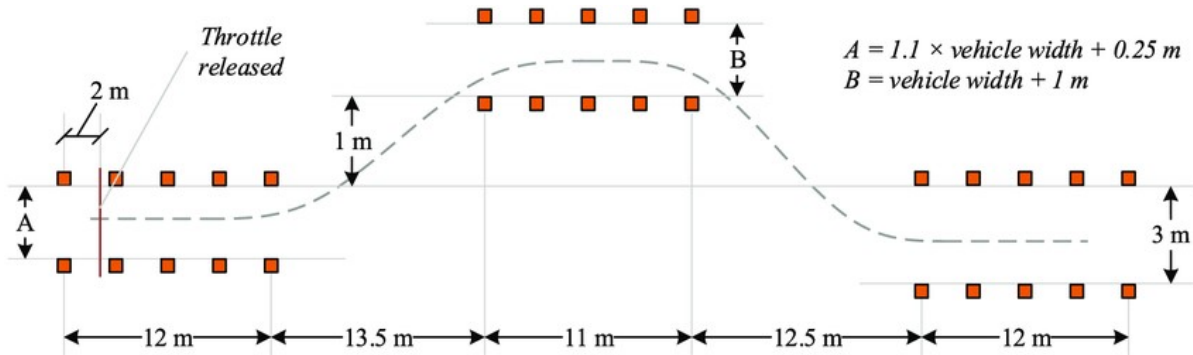


Figure 4.2: Double lane change maneuver course dimensions [48] [49]

4.2 Simulation Results

With the full-vehicle model constructed and the fishhook and double lane change maneuvers defined, a series of simulations were conducted to assess the impact of changes in vehicle weight distribution on handling performance. The handling performance results of the stock and UWAF models were compared to provide insights for fine-tuning the ARB stiffness values for the UWAF. This comparative analysis aims to enhance the vehicle's handling characteristics by optimizing the ARB stiffness in response to the observed effects of weight distribution changes.

4.2.1 Fishhook Test

Fishhook maneuver simulations were conducted separately for both the stock and the UWAFI models. The results were compared by plotting lateral acceleration and total body roll against time. Figures 4.3, 4.4, 4.5, and 4.6 present the outcomes of these tests, illustrating the dynamic responses of each vehicle model.

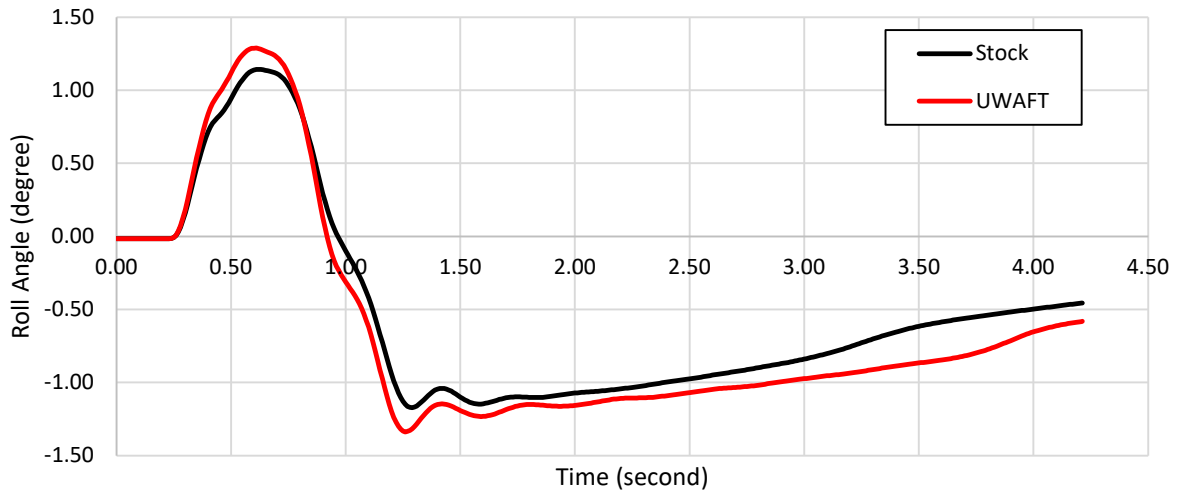


Figure 4.3: Body roll angle response of stock and UWAFI model from fishhook maneuver

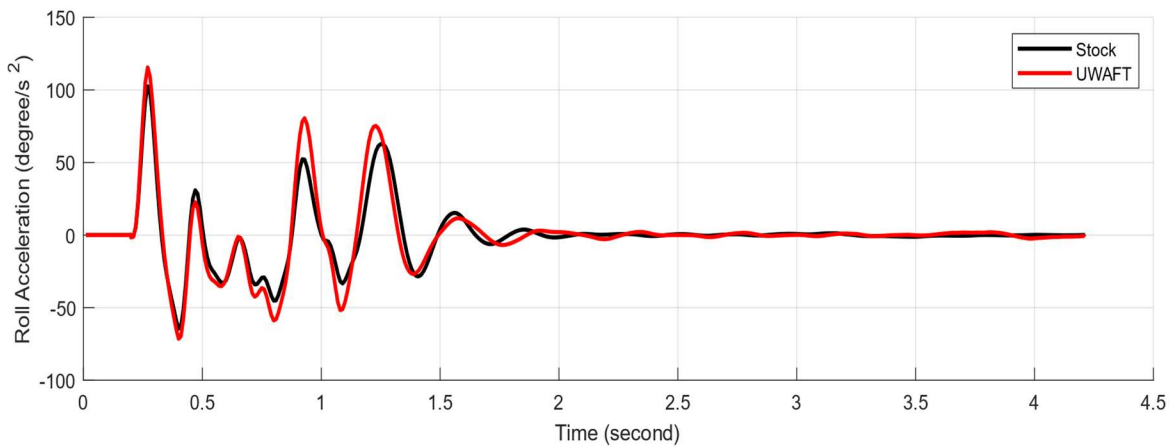


Figure 4.4: Body roll acceleration response of stock and UWAFI model from fishhook maneuver

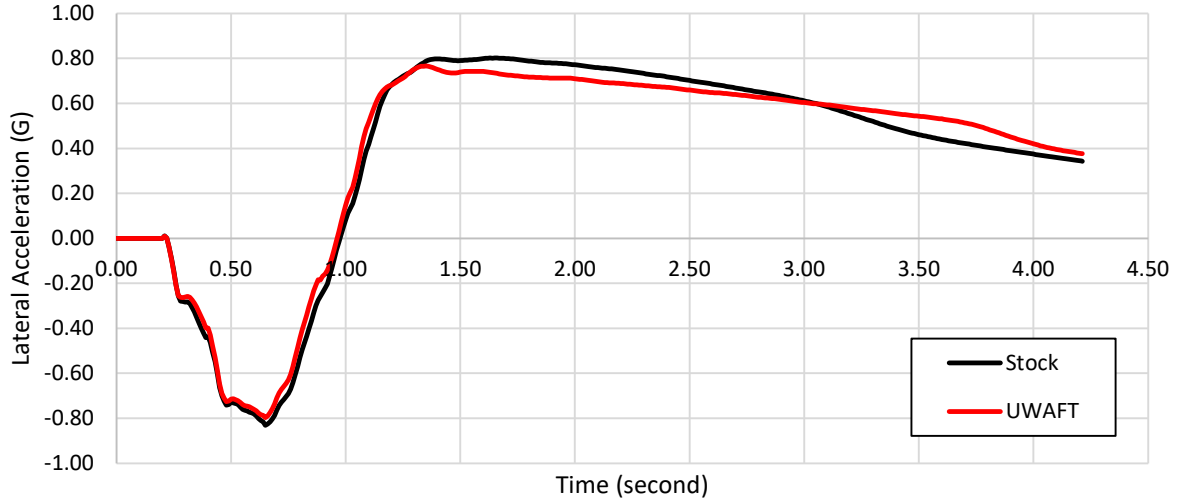


Figure 4.5: Lateral acceleration response of stock and UWAF model from fishhook maneuver

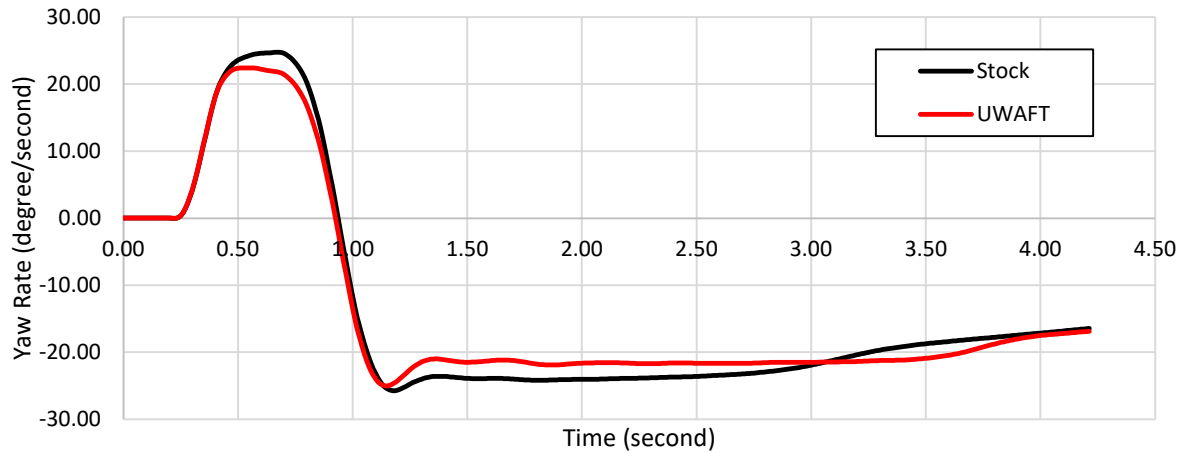


Figure 4.6: Yaw rate response of stock and UWAF model from fishhook maneuver

As depicted in Figure 4.3, the UWAF model demonstrates a markedly greater degree of body roll compared to the stock model. This pronounced increase in body roll indicates a reduction in the UWAF model's stability and precision in handling, which could potentially compromise both the comfort and safety of the vehicle. Furthermore, this reduction in performance is also realised by the roll acceleration response of the body, as shown in Figure 4.4, where the UWAF model exhibits consistently higher peak roll accelerations throughout the maneuver.

When comparing maximum lateral acceleration, as shown in Figure 4.5, the UWAF model demonstrated a performance similar to that of the stock, although it exhibited a slightly lower peak

lateral acceleration during the transition of the steering angle. The yaw rate response, depicted in Figure 4.6, which indicates the vehicle's handling responsiveness, showed that the UWAF model underperformed relative to the stock model. Specifically, the UWAF displayed a lower peak yaw rate, indicating a slower turning response and translation of the vehicle body. These results underscore the impact of increased vehicle weight and altered weight distribution, as evidenced by the outcomes of the fishhook maneuver simulations. Table 4.1 summarizes the handling performance differences between the stock and UWAF models.

Analyzing the slip angle response of the stock and UWAF models indicates that the UWAF model exhibits a more pronounced oversteer-biased response compared to the stock model, evidenced by a lower peak slip angle during the fishhook maneuver test. This outcome is consistent with theoretical expectations, wherein increased force applied to the front axle generates higher tire lateral force, thereby producing greater lateral force and enabling the vehicle to turn with reduced slip. Figure 4.7 presents a comparative analysis of the slip angles for the stock and UWAF models during the fishhook maneuver.

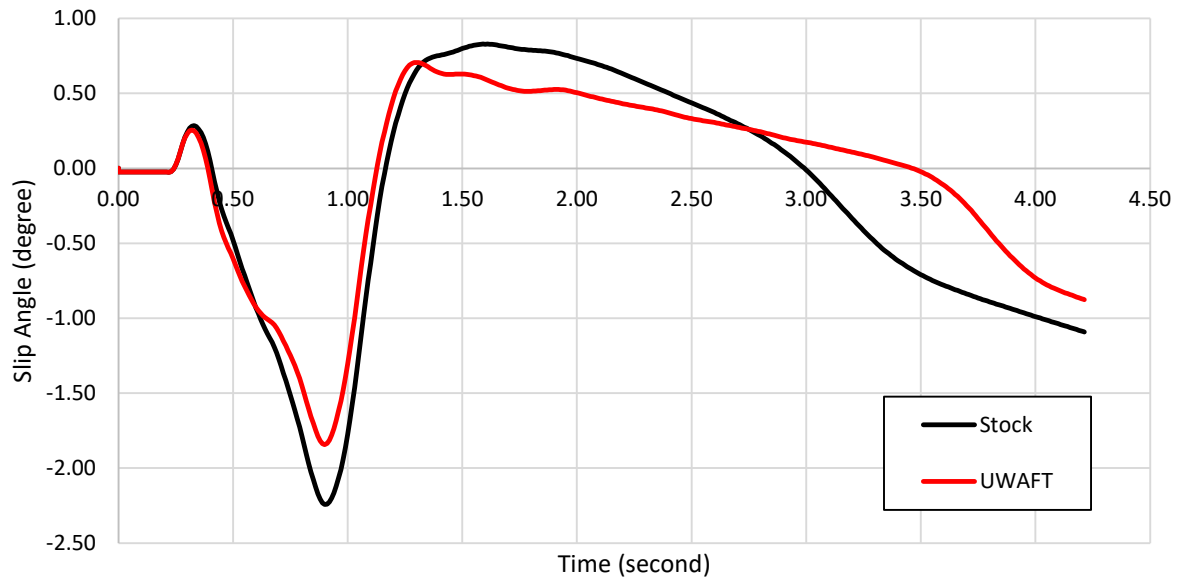


Figure 4.7: Slip angle response of the stock and UWAF model from fishhook maneuver

Table 4.1: Summary of fishhook maneuver simulation results

Variable	Stock	UWAF	% Difference
Absolute maximum body roll [degree]	1.17	1.35	+ 14.29
Absolute maximum lateral acceleration [G]	0.83	0.80	- 3.68
Absolute maximum yaw rate $\left[\frac{\text{degree}}{\text{s}}\right]$	24.75	22.42	- 9.88
Absolute maximum slip angle [degree]	2.24	1.84	- 19.61

4.2.2 Double Lane Change Test

As outlined in Section 4.1.2, the double lane change maneuver was configured in the MSSP environment using both the stock and UWAF full vehicle models. The simulations were conducted under identical conditions with consistent steering inputs. Figures 4.8, 4.9, 4.10, 4.11, and 4.12 present the vehicle responses of the stock and UWAF full vehicle models, respectively.

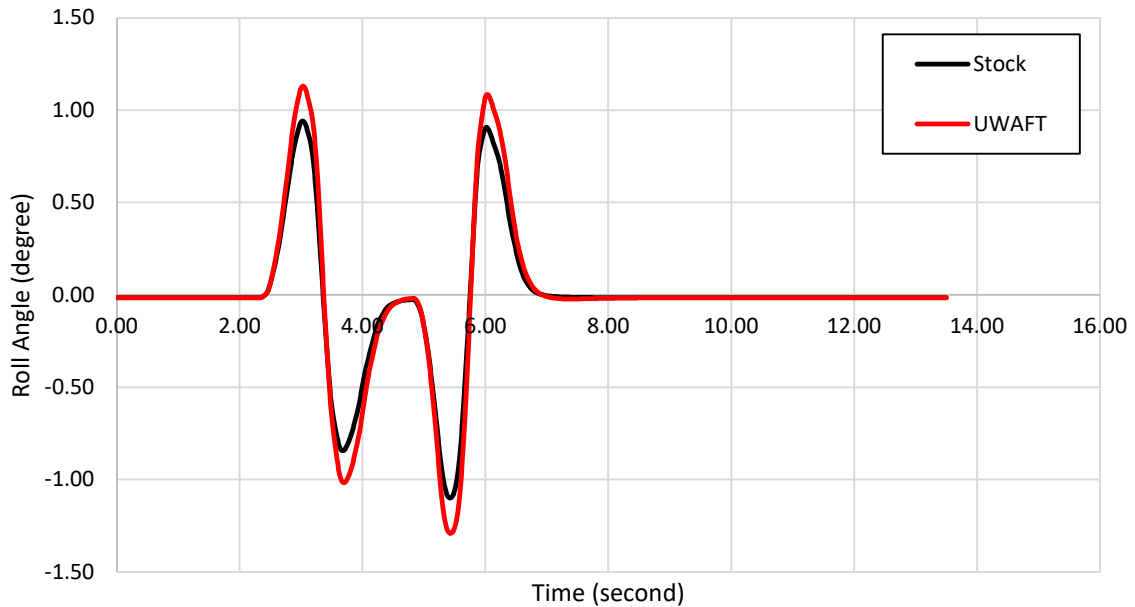


Figure 4.8: Roll angle response of stock and UWAF model from double lane change maneuver

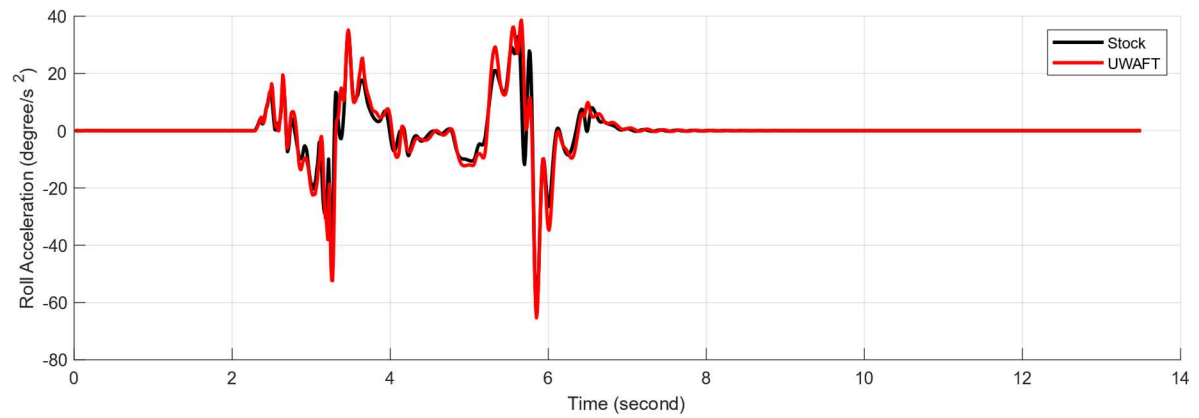


Figure 4.9: Roll acceleration response of stock and UWAF model from double lane change maneuver

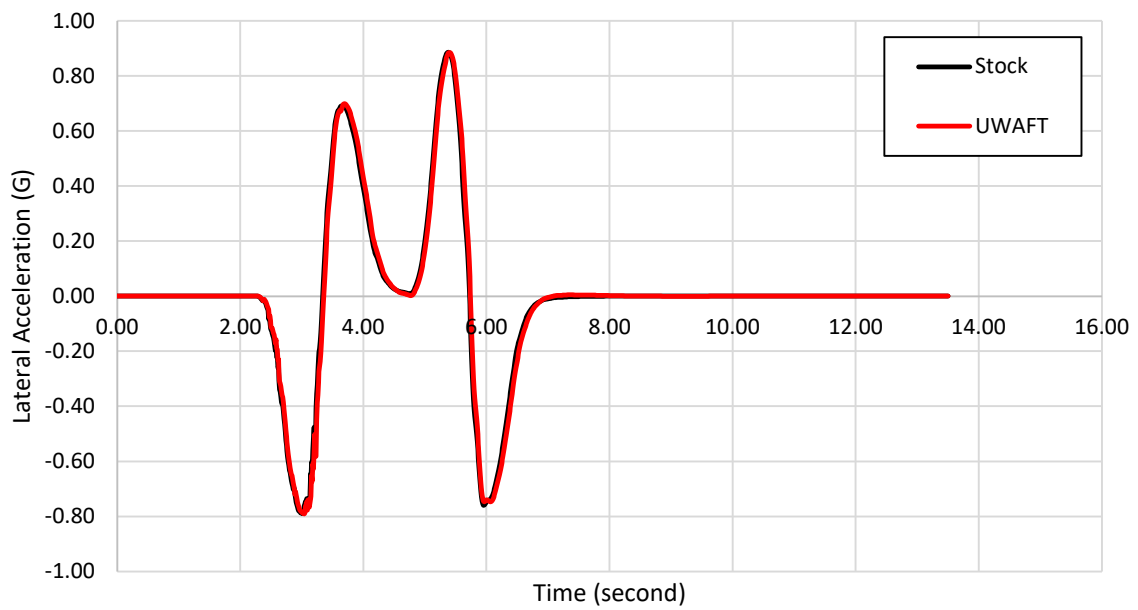


Figure 4.10: Lateral acceleration response of stock and UWAF model from double lane change maneuver

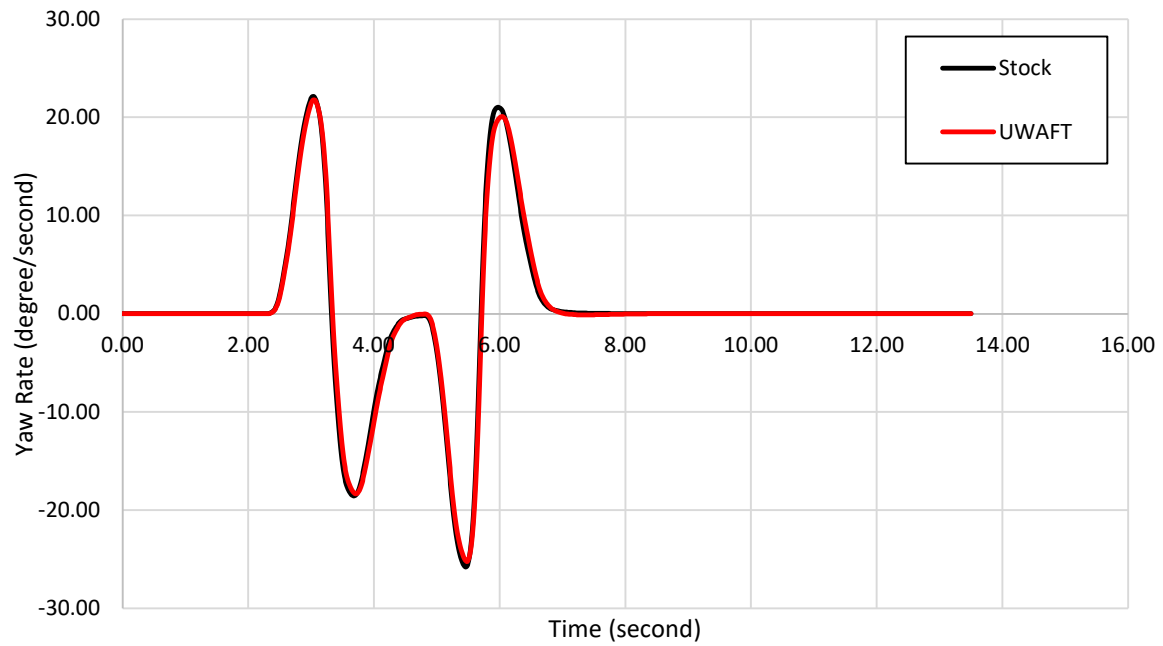


Figure 4.11: Yaw rate response of stock and UWAF model from double lane change maneuver

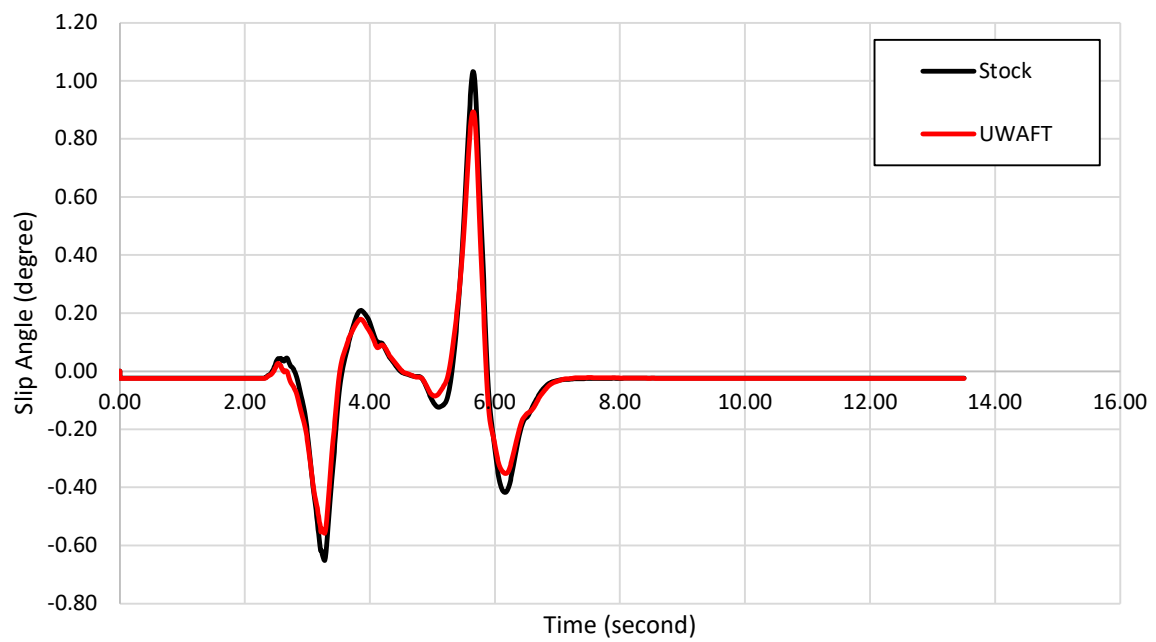


Figure 4.12: Slip angle response of the stock and UWAF model from double lane change maneuver

Consistent with the findings from the fishhook maneuver simulations, the UWAF model exhibited inferior performance compared to the stock model. Specifically, the UWAF model demonstrated a higher roll angle than the stock model throughout the simulations. This increased body roll indicates reduced stability and less precise handling performance in the UWAF model. Furthermore, this reduction in performance is also realised by the roll acceleration response of the body, where the UWAF model exhibits consistently higher peak roll accelerations throughout the maneuver. The heightened body roll could adversely impact vehicle comfort and safety by causing more pronounced lateral movement within the cabin, potentially unsettling passengers.

The analysis of lateral acceleration indicated that both models exhibited similar trends. Although the UWAF model maintains lateral grip comparable to the stock model, the increased body roll and altered weight distribution likely contribute to its diminished handling performance. The analysis of yaw rate revealed that the stock model exhibited a more responsive yaw rate compared to the UWAF model. Specifically, the UWAF model demonstrated a lower peak yaw rate, indicative of a slower turning response and reduced agility during maneuvering. This diminished responsiveness could adversely affect the vehicle's overall handling performance, particularly under dynamic driving conditions.

The slip angle response observed from the double lane change maneuver is consistent with the results of the fishhook maneuver, demonstrating that the UWAF model exhibits a more pronounced oversteer-biased response compared to the stock model. While a higher tendency towards oversteer can potentially lead to the driver losing control of the vehicle and unpredictable vehicle behavior, the consistently high slip angles exhibited by both the stock and UWAF models throughout the maneuver indicate that both models maintain safe handling characteristics.

These findings underscore the significant impact of increased vehicle weight and altered weight distribution on the handling characteristics of the UWAF model. The higher body roll and lower yaw rate responsiveness highlight areas where further optimization, such as adjusting the anti-roll bar stiffness, may be necessary to enhance the vehicle's dynamic performance. Table 4.2 summarizes the results of double lane change simulations with the percentage difference between the stock and UWAF model.

Table 4.2: Summary of double lane change maneuver simulation results

Variable	Stock	UWAFt	% Difference
Absolute maximum body roll [degree]	1.10	1.29	+ 15.90
Absolute maximum lateral acceleration [G]	0.89	0.89	0
Absolute maximum yaw rate $\left[\frac{\text{degree}}{s}\right]$	25.78	25.23	- 2.16
Absolute maximum slip angle [degree]	1.03	0.89	- 14.58

4.3 Modified UWAFt ARB

The fishhook and double lane change maneuver simulations clearly indicate that the UWAFt model necessitates an increase in roll stiffness to mitigate overall body roll and enhance yaw rate, thereby aligning its performance with that of the stock model. The method selected to augment the torsional stiffness of the ARB involves shortening its moment arm. As described in Chapter 2, Equation 2.3, reducing the length of the ARB arms proportionally increases their stiffness, thus enhancing the vehicle's overall torsional rigidity.

The full vehicle and subsystem models are constructed to be easily modifiable, enabling the simulation of varying ARB torsional stiffness effects on vehicle dynamics. In the simulation, ARB arm lengths are incrementally shortened by 5%, and the modified roll stiffness of each model is compared to that of the stock full vehicle model to ascertain the degree of ARB shortening required to achieve the target roll stiffness. This iterative simulation process is both efficient and rapid, as the model is designed to function as a fast prototyping environment and results generator.

By the conclusion of the iterative design cycle, the front and rear ARB arm lengths were each optimized and shortened by 35%. With these adjusted arm lengths, the UWAFt full vehicle model was reassembled and subjected to the same fishhook and double lane change maneuvers for comparative analysis against the stock vehicle. Table 4.3 provides a summary of the vehicle parameters, comparing the performance of the UWAFt model before and after the ARB modifications.

Table 4.3: Vehicle parameter comparison between modified and unmodified models

Parameter	Before Modification	After Modification	% difference
Front roll stiffness $\left[\frac{Nm}{degree}\right]$	4726	5287	+11.20
Rear roll stiffness $\left[\frac{Nm}{degree}\right]$	3085	3849	+22.03
Combined roll stiffness $\left[\frac{Nm}{degree}\right]$	7811	9136	+15.63

4.3.1 Fishhook Test

The results of the fishhook maneuver simulation test are shown in figure 4.13, 4.14, and 4.15.

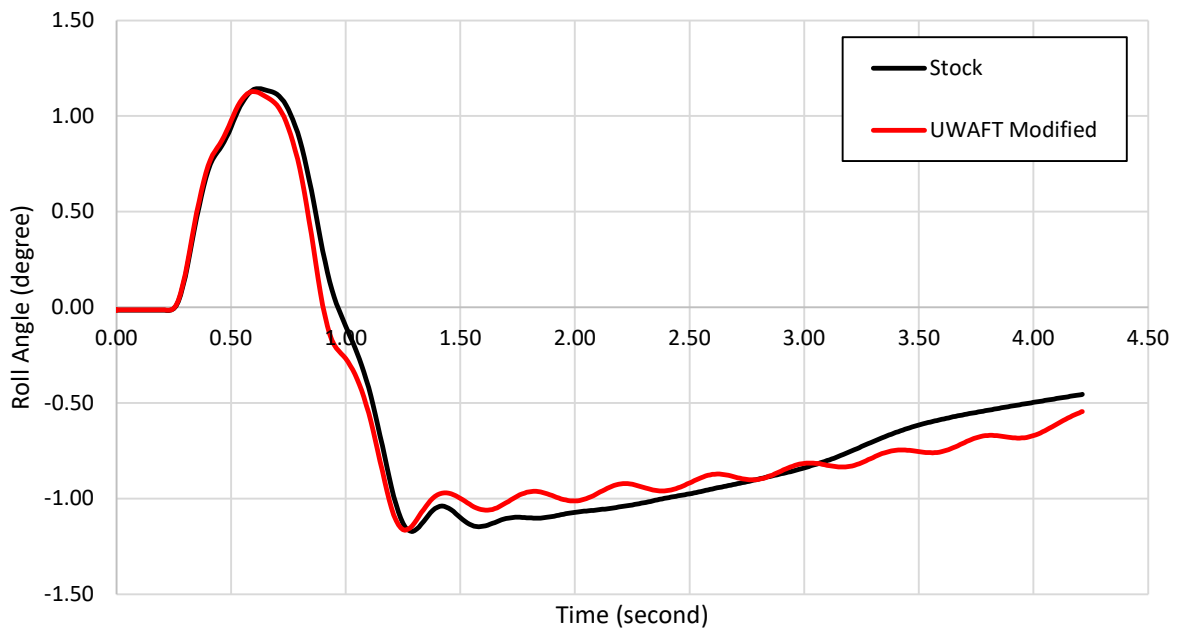


Figure 4.13: Roll angle response of stock and modified UWAFT model from fishhook maneuver

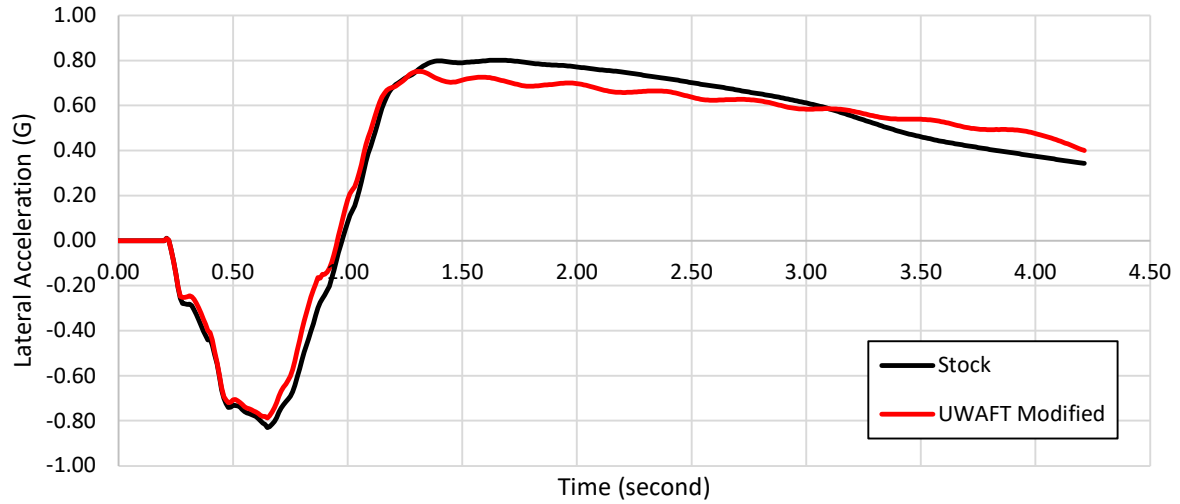


Figure 4.14: Lateral acceleration response of stock and modified UWAF model from fishhook maneuver

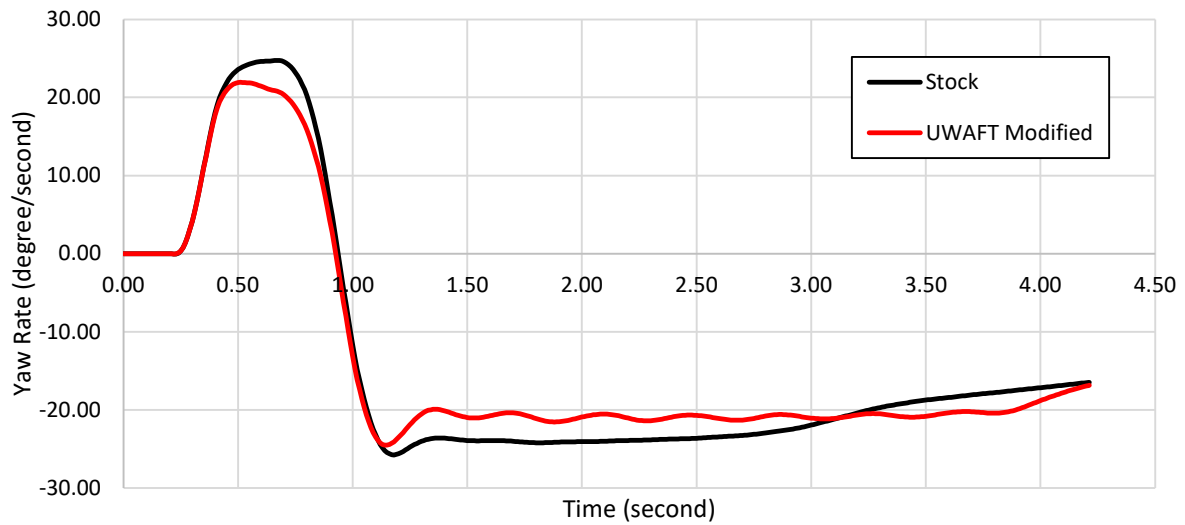


Figure 4.15: Yaw rate response of stock and modified UWAF model from fishhook maneuver

The most significant improvement resulting from the adjustment of the ARB stiffness was observed in the body roll angle response of the model. The maximum roll angle of the modified UWAF model closely approximates that of the stock model, indicating that increasing the ARB stiffness enhances overall vehicle comfort by reducing body roll. However, the lateral acceleration and yaw rate responses did not show comparable improvement relative to the unmodified UWAF model. This phenomenon

can be attributed to the substantial impact of overall vehicle weight on agility and dynamic performance.

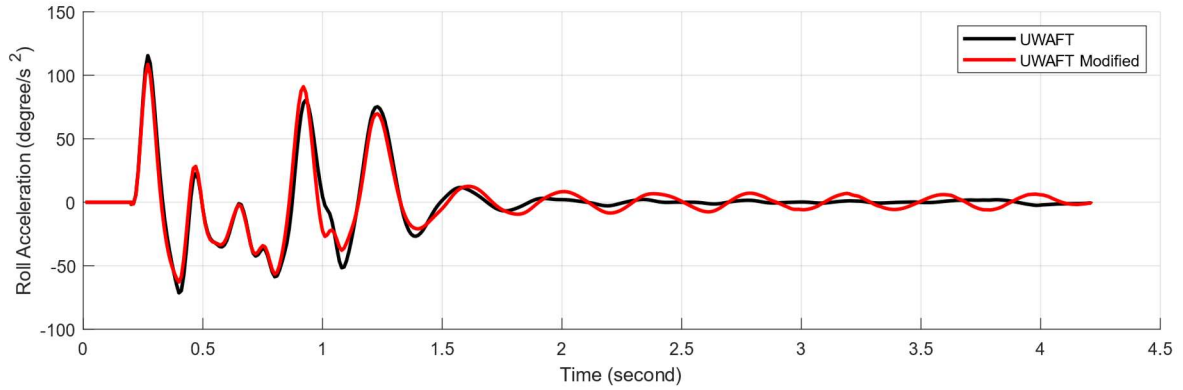


Figure 4.16: Roll acceleration response of UWAF T models from fishhook maneuver

Comparing the roll acceleration of the UWAF T models, illustrated in Figure 4.16, the modified ARB model has slightly lowered the overall magnitudes of the roll acceleration throughout the maneuver, except at around time 0.9 where the peak has increased slightly. This decrease in roll acceleration results in more comfortable ride throughout the maneuver, proving the increase in roll stiffness has improved the ride comfort of the vehicle. The error in the model, high peak at around time 0.9 seconds and oscillation after 1.5 seconds, requires further investigation in the future for better accuracy of the model.

The increase in roll stiffness in modified UWAF T model has decreased the slip angle, resulting more tendency to oversteer during cornering. This results in vehicle with tighter handling feel, where driver can feel the car be more responsive to the same wheel angle input during cornering. Figure 4.14 shows the slip angle response comparison of the three models from the fishhook maneuver. Table 4.4 summarizes the fishhook test data of the modified UWAF T model in comparison to the unmodified UWAF T model.

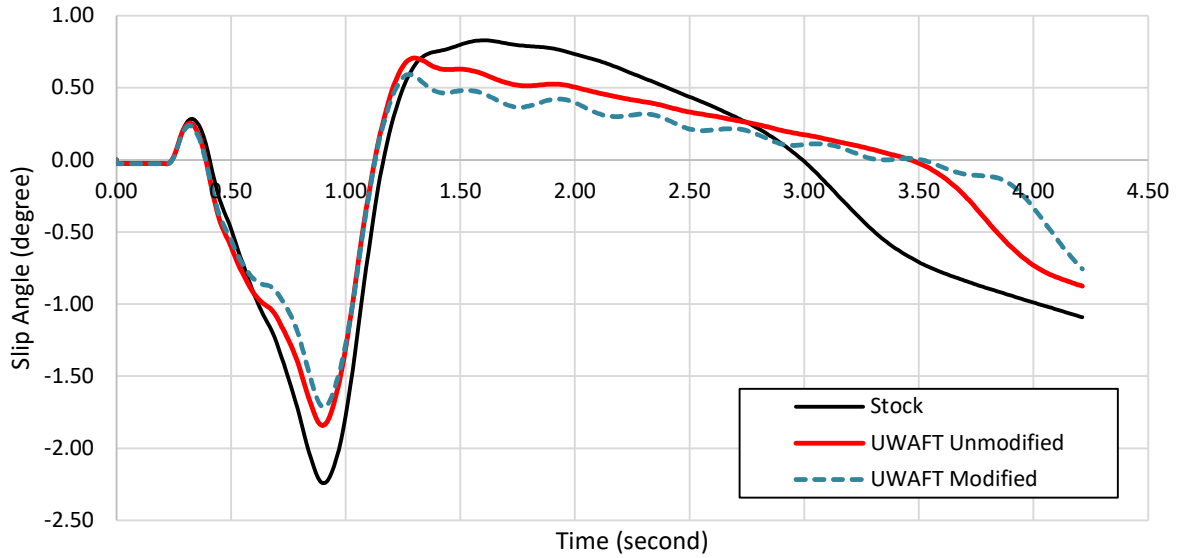


Figure 4.17: Slip angle response of stock, unmodified, and modified UWAF model from the fishhook maneuver

Table 4.4: Summary of fishhook simulation results comparing UWAF models

Variable	Unmodified	Modified	% Difference
Absolute maximum body roll [degree]	1.35	1.16	- 15.14
Absolute maximum lateral acceleration [G]	0.80	0.79	- 1.26
Absolute maximum yaw rate $\left[\frac{\text{degree}}{\text{s}}\right]$	22.42	21.93	- 2.21
Absolute maximum slip angle [degree]	1.84	1.71	- 7.32

4.3.2 Double Lane Change Test

A double lane change test was conducted with the modified UWAF model to assess the impact of increased ARB stiffness during an emergency maneuver. Similar to the results of the fishhook maneuver tests, the most significant improvement was observed in the body roll angle during the double lane change test. Although the modified UWAF model still marginally underperforms compared to the stock model, the results are deemed acceptable from a vehicle dynamics standpoint, considering the significant increase in weight and the forward shift in weight balance. Figures 4.18, 4.19, and 4.20 present the results from the double lane change tests.

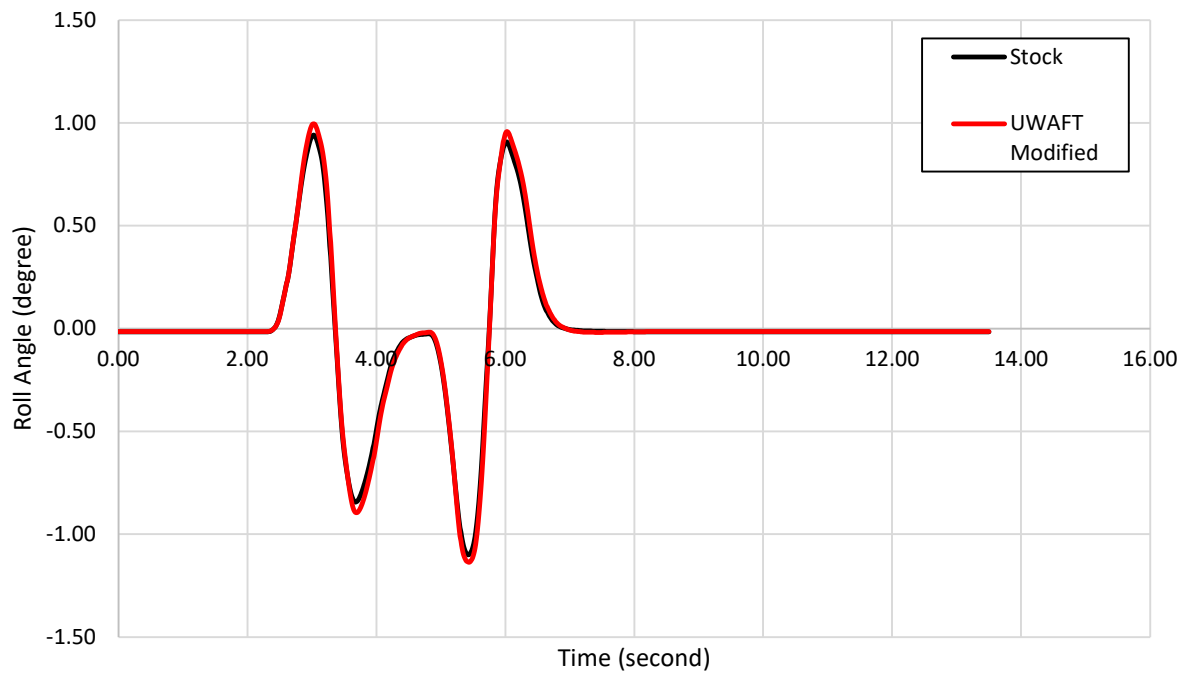


Figure 4.18: Roll angle response of stock and modified UWAF model from double lane change maneuver

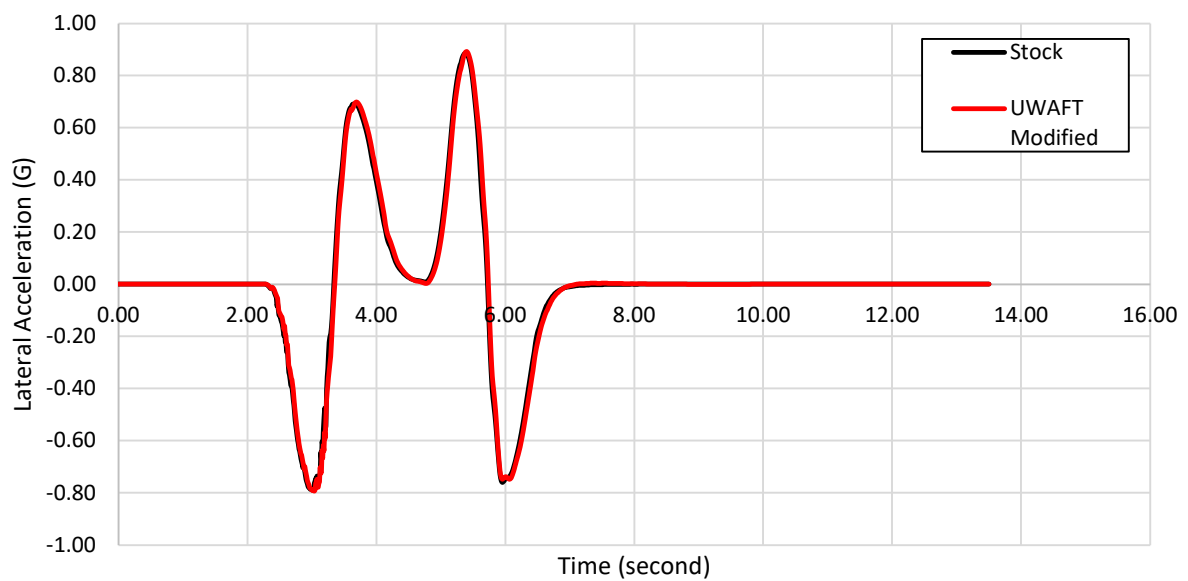


Figure 4.19: Lateral acceleration response of stock and modified UWAF model from double lane change maneuver

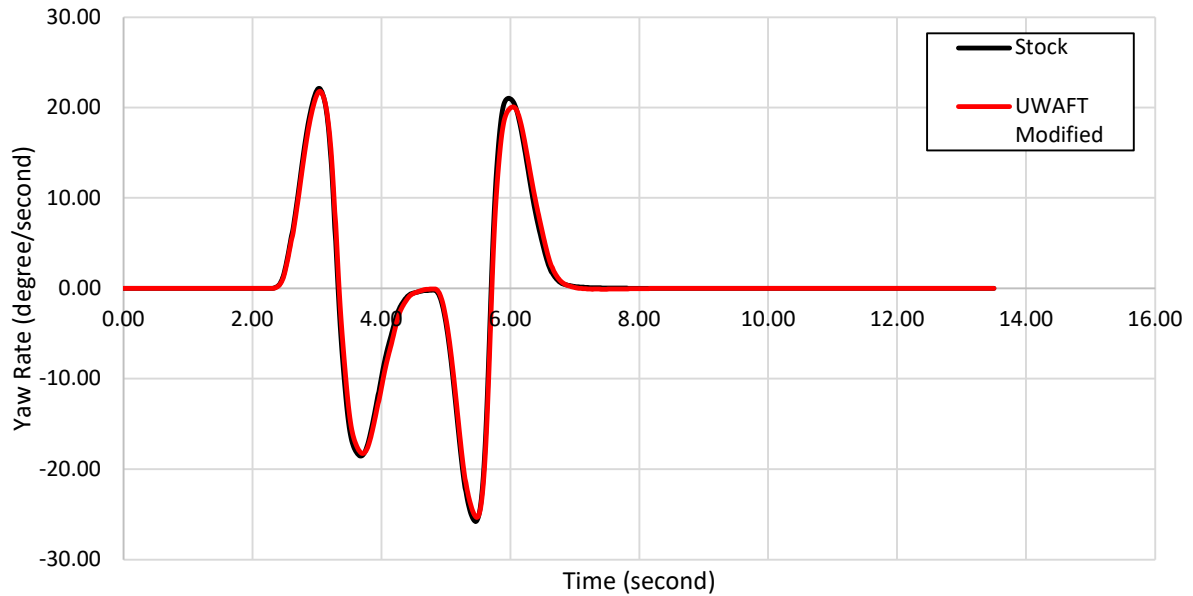


Figure 4.20: Yaw rate response of stock and modified UWAF model from double lane change maneuver

The overall increase in performance was observed with the modified UWAF model compared to the unmodified UWAF model. The stiffer ARB increased the roll resistance performance over the unmodified model. This increased in performance have direct impact on providing more comfortable vehicle operation experience and safety of the vehicle at an emergency action.

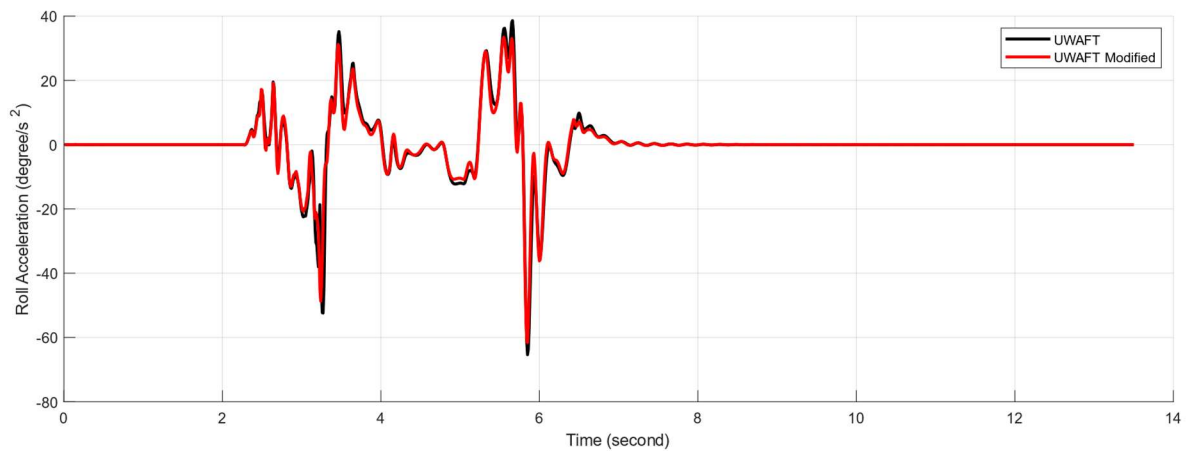


Figure 4.21: Roll acceleration of UWAF models from double lane change maneuver

As illustrated in Figure 4.21, the roll acceleration of the modified UWAF model has diminished in comparison to the unmodified version. This reduction in roll acceleration indicates an enhancement in ride comfort, achieved through an increase in roll stiffness.

The increase in roll stiffness in modified UWAF model has decreased the slip angle as similar from the fishhook maneuver, although significantly less pronounce in double lane change maneuver. This resulted in modified UWAF model to have more tendency to oversteer during cornering. Figure 4.22 shows the slip angle response comparison of the three models from the double lane change maneuver. Table 4.5 summarizes the double lane change test data of the modified UWAF model in comparison to the unmodified UWAF model.

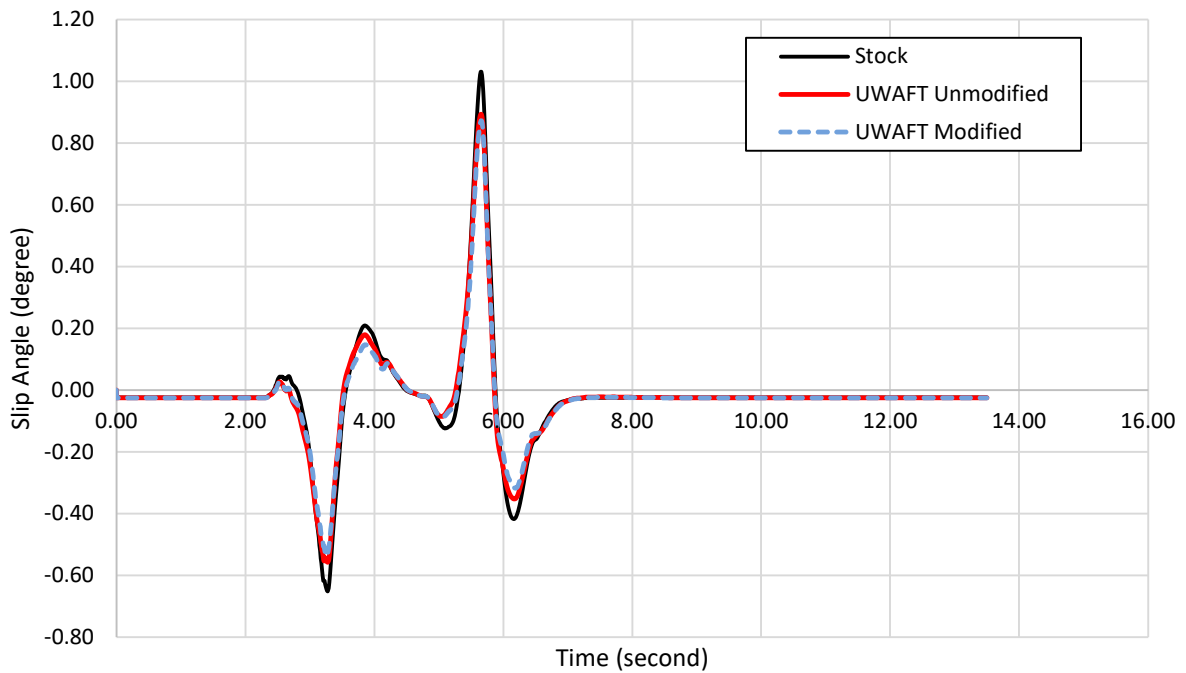


Figure 4.22: Slip angle response of stock, unmodified, and modified UWAF model from the fishhook maneuver

Table 4.5: Summary of double lane change results comparing UWAF T models

Variable	Unmodified	Modified	% Difference
Absolute maximum body roll [degree]	1.29	1.14	- 12.35
Absolute maximum lateral acceleration [G]	0.89	0.89	0
Absolute maximum yaw rate $\left[\frac{degree}{s}\right]$	25.23	25.35	+ 0.47
Absolute maximum slip angle [degree]	0.89	0.87	- 2.27

Chapter 5

Conclusion

Designing an effective suspension system in a passenger vehicle is a safety-critical task. Understanding and testing the dynamic characteristics of a vehicle is particularly challenging and time-consuming when relying solely on real-life test data. This research has developed and validated a comprehensive full-vehicle MSSP model for the UWAFI team, facilitating the redesign of the ARB to accommodate the custom powertrain configuration, increased vehicle weight, and altered weight distribution.

The development and validation of the full-vehicle MSSP model for the UWAFI have provided significant insights into the impact of increased vehicle weight and altered weight distribution on handling performance. Through rigorous simulation tests, including fishhook and double lane change maneuvers, it was determined that the UWAFI model exhibited higher body roll and reduced yaw rate responsiveness compared to the stock model. These findings highlighted the necessity of optimizing the vehicle's ARB stiffness to enhance its dynamic performance.

Adjustments to the ARB stiffness, achieved by shortening the moment arm, successfully reduced body roll, bringing it closer to the levels observed in the stock model. However, improvements in lateral acceleration and yaw rate were less pronounced, underscoring the substantial role of overall vehicle weight in influencing agility and handling characteristics.

The comprehensive analysis conducted in this research underscores the importance of fine-tuning suspension components to mitigate the adverse effects of increased weight and altered distribution in electrified vehicles. Future work may focus on further optimization of suspension parameters and exploring advanced materials and technologies to enhance vehicle performance while maintaining safety and comfort. This study contributes valuable knowledge to the field of vehicle dynamics, particularly in the context of electric vehicle development, and provides a foundation for ongoing advancements in this rapidly evolving domain.

5.1 Future Work and Recommendations

The successful development and validation of the full-vehicle MSSP model has established a robust foundation for future research endeavors aimed at enhancing the model's accuracy and fidelity for an in-depth understanding of vehicle dynamics. Building upon the insights gained from this study, several

promising areas for future research include the development of active anti-roll bars (ARBs) for dynamic control of roll stiffness based on the vehicle's state. Such advancements could significantly improve the adaptability and performance of suspension systems, thereby enhancing overall vehicle stability and handling.

To facilitate the smooth implementation of the learnings from the MSSP model, a high-fidelity ARB model can be developed to iterate the design and study packaging constraints within the assembly. This can be pursued within CAD environments. Additionally, creating a finite element model of the ARB to validate its stiffness and assist in designing the geometry is recommended.

Other areas for future improvement to increase the vehicle dynamics model's accuracy and ease of implementation include:

1. Developing more accurate models of suspension components, especially bushings, to replicate real-life vehicle behavior in the simulation.
2. Incorporating accurate weight and moment of inertia data for suspension components to enhance the model's precision. It is critical to input the correct weight of components, as this impacts the accuracy of the sprung and unsprung mass calculations.
3. Acquiring an accurate tire model is essential for accurately representing the vehicle. Efforts should be made to obtain the OEM's tire data.
4. Validating the simulation data with real-life test data gathered from track testing of the LYRIQ to ensure the model's reliability.
5. Conducting kinematic and compliance testing of the real-life vehicle to correlate simulation data with real-world performance. This testing can be conducted by a third-party vehicle shop equipped with the necessary test facilities.

These recommendations aim to refine the vehicle dynamics model, ensuring its precision and applicability in real-world scenarios.

References

- [1] K. Kamberov, G. Todorov and T. Ivanov, "Automotive Product Validation through Virtual Prototyping at Early Design Stage," *AIP Conference Proceedings*, vol. 3064, no. 1, 2024.
- [2] Hexagon , *Adams Car*, Hexagon, 2024.
- [3] MapleSoft, *MapleSim*, Waterloo: MapleSoft, 2024.
- [4] Applied Intuition, *CarSim*, Applied Intuition, 2024.
- [5] A. Kampker, C. Offermanns, H. Heimes and P. Bi, "Meta-analysis on the Market Development of Electrified Vehicles," *ATZ worldwide*, vol. 123, pp. 58-63, 2021.
- [6] "Electrified Vehicles," Carnegie Mellon University Vehicle Electrification Group, [Online]. Available: <https://www.cmu.edu/cit/veg/electrified%20vehicles/index.html>. [Accessed 8 July 2024].
- [7] J. F. Marrone, "Adapting Regenerative Braking Strength to Driver Preference," University of Waterloo, 2023.
- [8] D. A. Crolla and D. Cao, "The impact of hybrid and electric powertrains on vehicle dynamics, control systems and energy regeneration," *International Journal of Vehicle Mechanics and Mobility*, vol. 50, no. Supplement 1, pp. 98-109, 2012.
- [9] B. Heißing and M. Ersoy, *Chassis Handbook*, Springer Fachmedien Wiesbaden GmbH, 2011.
- [10] T. Ibicek and A. N. Thite, "Quantification of Human Discomfort in a Vehicle Using a Four-Post Rig Excitation," *Journal of Low Frequency Noise, Vibration and Active Control*, vol. 31, no. 1, pp. 29-42, 2012.
- [11] S. Koumura and T. Ohkita, "Ride Comfort Evaluation through Analysis of Roll and Lateral Vehicle Behaviors Due to Road Input," *SAE International Journal of Passenger Cars - Mechanical Systems*, vol. 1, no. 1, pp. 472-479, 2009.
- [12] J. P. Pauwelussen, *Essentials of Vehicle Dynamics*, Elsevier, 2015.

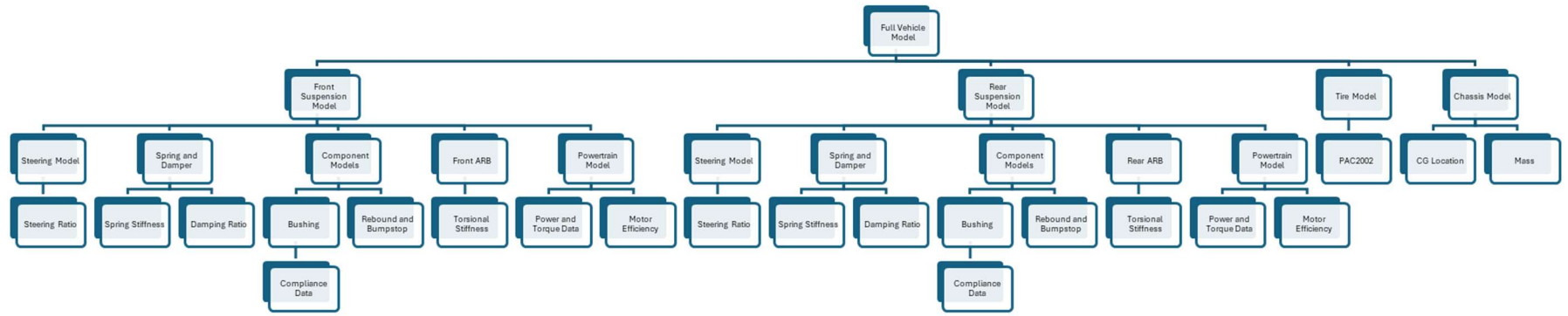
- [13] H. F. Grip, L. Imsland, T. A. Johansen, J. C. Kalkkuhl and A. Suissa, "Vehicle Sideslip Estimation: Design, implementation, and experimental validation," *IEEE control systems*, pp. 36-52, October 2009.
- [14] D. L. Nordeen, "Vehicle Handling: Its Dependence Upon Vehicle Parameters," in *SAE World Congress & Exhibition*, 1963.
- [15] Statista, "Toyota Motor Corporation's research and development (R&D) expenses from FY 2007 to FY 2024," Statista, 2024.
- [16] P. T. Faithfull, R. J. Ball and R. P. Jones, "An investigation into the use of hardware-in-the-loop simulation with a scaled physical prototype as an aid to design," *Journal of Engineering Design*, vol. 12, no. 3, pp. 231-243, 2001.
- [17] C. Deubel, S. Ernst and G. Prokop, "Objective evaluation methods of vehicle ride comfort—A literature review," *Journal of Sound and Vibration*, vol. 548, 2023.
- [18] I. Badiru and W. Cwycyshyn, "Customer Focus in Ride Development," in *SAE World Congress & Exhibition*, 2013.
- [19] R. N. Jazar, *Vehicle Dynamics: Theory and Application*, Springer International Publishing, 2017.
- [20] A. Goodarzi and A. Khajepour, *Vehicle suspension system technology and design*, Springer, 2022.
- [21] S. Sunil Kelkar, P. Gautam, P. Sanjay Agrawal and R. Manoharan, "A detailed study on design, fabrication, analysis, and testing of the anti-roll bar system for formula student cars," *SN Applied Sciences*, 10 February 2021.
- [22] SAE, *Spring Design Manual*, Warrendale: Society of Automotive Engineers, 1996.
- [23] M. Kissai, B. Monsuez, X. Mouton, D. Martinez and A. Tapus, "Adaptive Robust Vehicle Motion Control for Future Over-Actuated Vehicles," *Machines*, 19 April 2019.
- [24] J. Y. Wong, *Theory of ground vehicles*, John Wiley & Sons, 2001.
- [25] R. Rajamani, *Vehicle Dynamics and Control*, Springer, 2012.

- [26] S. Nasiri, N. Sina and A. Eslami, "Multi-objective Optimisation of McPherson Strut Suspension Mechanism Kinematics Using Random Search Method," *Indian Journal of Science and Technology*, vol. 8, no. 16, 2015.
- [27] Automotive Service Excellence, "ASE Certification Practice Test," FreeASEStudyGuides, 2008.
- [28] L. Shengqin and F. Xinyuan, "Study of structural optimization design on a certain vehicle body-in-white based on static performance and modal analysis," *Mechanical Systems and Signal Processing*, 1 January 2020.
- [29] B. Miao and C. Han, "Intelligent vehicle obstacle avoidance path-tracking," *Mechanical Sciences*, pp. 247-258, 15 June 2023.
- [30] M. Guiggiani, *The Science of Vehicle Dynamics Handling, Braking, and Ride of Road and Race Cars*, Switzerland: Springer, 2023.
- [31] D. Tan, "Nonlinear Optimal Control of Automated Vehicles in a Connected Environment," University of Waterloo, Waterloo, 2019.
- [32] v. T. Vu, "Enhancing the roll stability of heavy vehicles by using an active anti-roll bar system," HAL, 2017.
- [33] H. Agrawal and J. Gustafsson, "Investigation of active anti-roll bars and development of control algorithm," KTH Royal Institute of Technology, Stockholm, 2017.
- [34] S.-e. Zhao, Y. Li and Z. Qu, "Vehicle Chassis Integrated Control Based on Multimodel and Multilevel Hierarchical Control," *Mathematical Problems in Engineering*, 28 April 2014.
- [35] MathWorks, "Tire-Road Interaction (Magic Formula)," [Online]. Available: <https://www.mathworks.com/help/sdl/ref/tireroadinteractionmagicformula.html>. [Accessed 23 July 2024].
- [36] General Motors, "LYRIQ FEATURES," General Motors, [Online]. Available: <https://www.cadillac.com/electric/lyriq/features#performance>. [Accessed 23 July 2024].
- [37] V. Derevianchuk, "off-road vehicle suspension and steering," GrabCAD, 2014.

- [38] F. Badin, F. Le Berr, G. Castel and J. C. Dabadie, "Energy Efficiency Evaluation of a Plug-in Hybrid Vehicle under European Procedure, Worldwide Harmonized Procedure and Actual Use," *World Electric Vehicle Journal*, vol. 7, no. 3, pp. 475-488, 2015.
- [39] T. Moser, F. X. Wagner and D. Van Riesen, "Electric motor technology in a high-performance application," *MTZ Worldwide*, pp. 44-48, 03 September 2013.
- [40] P. Verissimo and J. A. Ambrosio, "Improved Bushing Models for Vehicle Dynamics," in *European Conference on Computational Mechanics*, 2008.
- [41] Kcivsou, "6pcs Control Arm Suspension Bushing Kit Front Lower Inner Outer Side 2331513287 51393SDAA02 51391-SDB-A03," Amazon.
- [42] M. Doody, "Design and development of a composite automotive anti-roll bar," University of Windsor, Windsor, 2013.
- [43] E. Bakker, L. Nyborg and H. B. Pacejka, "Tyre Modelling for Use In Vehicle Dynamics Studies," *SAE Technical Paper*, 1987.
- [44] E. Kuiper and J. J. M. Van Oosten, "The PAC2002 advanced handling tire model," *Vehicle System Dynamics*, vol. 45, no. 1, pp. 153-167, 2007.
- [45] G. J. Forkenbrock, W. R. Garrot, M. Heitz and B. C. O'Harra, "An Experimental Examination of J-Turn and Fishhook Maneuvers That May Induce On-Road, Untripped, Light Vehicle Rollover," *Journal of passenger car: Mechanical systems journal*, vol. 112, pp. 1112-1127, 2003.
- [46] National Highway Traffic Safety Administration, "Laboratory Test Procedure for Dynamic Rollover: The Fishhook Maneuver Test Procedure," U.S. Department of Transportation, Washington, 2013.
- [47] J. Marzbanrad, G. Soleimani, M. Mahmoodi-k and A. H. Rabiee, "Development of fuzzy anti-roll bar controller for improving vehicle stability," *Journal of Vibroengineering*, vol. 17, no. 7, pp. 3856-3864, 2015.
- [48] A. Feher, S. Aradi and T. Becsi, "Hierarchical Evasive Path Planning Using Reinforcement Learning and Model Predictive Control," *IEEE Access*, pp. 187470-187482, 14 October 2020.

- [49] International Organization for Standardization, "Passenger Cars—Test Track for a Severe Lane-Change Manoeuvre—Part2: Obstacle Avoidance," International Organization for Standardization, Geneva, 2011.
- [50] P. Eisenstein, "A New Breed of American Electric Luxury: 2023 Cadillac Lyriq Review," Gearjunkie, 2022.

Appendix A



Appendix B

SPMM TEST REPORT

<lyriq>/assemblies.tbl/baseline_lyriq_full_vehicle_v2.asy

(PARAMETER) (UNITS) (TOTAL) (LEFT) (RIGHT)

Total weight N 26361.3528 13230.6266 13130.7262

% 50.4821 49.5179

Front ground reaction N [%] 18229.2476 [50.4688] 9144.0632 9085.1844

Rear ground reaction N [%] 18132.1052 [49.5312] 9086.5634 9045.5418

Total roll inertia kg-mm**2 480244301.5930

Total pitch inertia kg-mm**2 2388646280.4374

Total yaw inertia kg-mm**2 2447888383.5030

Total product Ixy kg-mm**2 2068649.4846

Total product Ixz kg-mm**2 -9738114.7517

Total product Iyz kg-mm**2 2701803.6760

Global C.G location (X/Y/Z) mm 3056.2335 -8.0997 1282.9473

Ground plane elevation mm 539.2089

C.G. height mm 743.7384

Body yaw angle deg 0.000

Body pitch angle deg -0.2276

Body roll angle deg 0.0159

Wheelbase mm 3098.2517 3098.266 3098.2373

Average track width mm 1659.3653

Static stability factor - 1.1156

FRONT AXLE:

<lyriq>/subsystems.tbl/baseline_lyriq_front_suspension.sub

<acar_shared>/tires.tbl/mdi_tire01.tir

<acar_shared>/tires.tbl/mdi_tire01.tir

(PARAMETER) (UNITS) (AVERAGE) (LEFT) (RIGHT)

STATIC:

Static toe deg 0.0709 0.0574 0.0844

Static camber deg -0.7703 -0.7569 -0.7836

Static caster deg 5.0233 5.0229 5.0238

Kingpin angle deg 0.0026 0.0037 0.0014

Caster trail mm 24.9087 24.9107 24.9066

Scrub radius mm 1.4962 1.4884 1.5040

Track width mm 1660.0126

Roll center height mm 160.4668

Roll center lateral shift mm 0.6410

Wheel center rise mm 15.1228 15.2574 14.9882

Tire loaded radius mm 301.465 301.37 301.5599

BOUNCE TEST:

Wheel rate N/mm 29.9095 29.8841 29.9349

Spring rate N/mm 52.4510 52.4510 52.451

Spring ratio mm/mm 0.6456 0.6457 0.6456
Damper ratio mm/mm 0.6456 0.6457 0.6456
Spring rate at wheel N/mm 21.8643 21.8674 21.8612

Bump steer deg/mm -0.0053 -0.0053 -0.0053
Bump caster deg/mm 0.0212 0.0212 0.0212
Bump camber deg/mm -0.0159 -0.0159 -0.0159

Longitudinal wheel center recession mm/mm -0.0112 -0.0112 -0.0112
Lateral wheel center migration mm/mm -0.0042 -0.0041 -0.0044

ROLL TEST:

Wheel rate in roll N/mm 200.6414 200.4006 200.8821
Roll stiffness Nmm/deg 4824899.5308

Roll steer (per degree) deg/deg -0.0945 -0.0948 -0.0941
Roll steer deg/mm 0.0 0.0092 -0.0092
Roll camber w.r.t. chassis (per degree) deg/deg -0.2486 -0.2466 -0.2506
Roll camber (per degree) deg/deg 0.751 0.753 0.7490
Roll camber deg/mm 0.0 -0.0278 0.0278

Steering wheel torque vs roll angle Nmm/deg -957.0745

STEER TEST:

Rack travel @ steering input mm N/A
Max steer angle deg 0.0 -0.0 0.0
@steering input (left turn)
Max steer angle deg 0.0 -0.0 0.0
@steering input (right turn)

Full lock angle deg N/A N/A N/A

@pvs_max_rack_disp (left turn)

Full lock angle deg N/A N/A N/A

@pvs_max_rack_disp (right turn)

LONGITUDINAL COMPLIANCE TEST (Traction):

Traction Steer deg/kN -0.0748 -0.076 -0.0735

Wheel center longitudinal stiffness N/mm 972.3933 972.1484 972.6381

Caster compliance - traction deg/kN 0.104 0.1043 0.1037

LONGITUDINAL COMPLIANCE TEST (Braking):

Brake Steer deg/kN 0.1332 0.1342 0.1321

Contact patch braking stiffness N/mm 772.0347 771.8777 772.1918

Caster compliance - braking deg/kN 0.3324 0.3322 0.3326

LATERAL COMPLIANCE TEST - PARALLEL:

Lateral compliance steer deg/kN -0.3697 -0.3688 -0.3706

Lateral compliance camber deg/kN 0.1318 0.1313 0.1322

Wheel center lateral stiffness N/mm 1535.9299 1535.6164 1536.2434

LATERAL COMPLIANCE TEST - PARALLEL: With Trail 0.0

Lateral compliance steer deg/kN -0.3697 -0.3688 -0.3706

Lateral compliance camber deg/kN 0.1318 0.1313 0.1322

Wheel center lateral stiffness N/mm 1535.9299 1535.6164 1536.2434

LATERAL COMPLIANCE TEST - OPPOSITE:

Lateral compliance steer deg/kN -0.078 -0.078 -0.0779

Lateral compliance camber deg/kN 0.047 0.047 0.0469

Wheel center lateral stiffness N/mm 6037.927 6035.2793 6040.5747

ALIGNING TORQUE TEST - PARALLEL:

Aligning torque - steer deg/kNmm -0.0057 -0.0057 -0.0057

Aligning torque - camber deg/kNmm 0.0008 0.0008 0.0008

ALIGNING TORQUE TEST - OPPOSITE:

Aligning torque - steer deg/kNmm -0.0004 -0.0004 -0.0004

Aligning torque - camber deg/kNmm 0.0001 0.0001 0.0001

REAR AXLE:

<lyriq>/subsystems.tbl/baseline_lyriq_rear_suspension_v3.sub

<acar_shared>/tires.tbl/mdi_tire01.tir

<acar_shared>/tires.tbl/mdi_tire01.tir

(PARAMETER) (UNITS) (AVERAGE) (LEFT) (RIGHT)

STATIC:

Static toe deg -0.5219 -0.5222 -0.5216

Static camber deg -0.846 -0.8389 -0.8531

Static caster deg -10.2555 -10.2685 -10.2425

Kingpin angle deg -0.0944 -0.0946 -0.0943

Track width mm 1658.718

Roll center height mm 217.5401

Roll center lateral shift mm 0.9615

Wheel center rise mm 27.5615 27.7245 27.3985

Tire loaded radius mm 301.6215 301.5554 301.6877

BOUNCE TEST:

Wheel rate N/mm 27.4065 27.3973 27.4158

Spring rate N/mm 52.4510 52.4510 52.451

Spring ratio mm/mm 0.5119 0.5119 0.5119

Damper ratio mm/mm 0.8075 0.8076 0.8075

Spring rate at wheel N/mm 13.7438 13.745 13.7426

Bump steer deg/mm -0.0079 -0.008 -0.0079

Bump caster deg/mm -0.0322 -0.0321 -0.0323

Bump camber deg/mm -0.0219 -0.0219 -0.0219

Longitudinal wheel center recession mm/mm -0.1155 -0.1154 -0.1157

Lateral wheel center migration mm/mm 0.0474 0.0476 0.0473

ROLL TEST:

Wheel rate in roll N/mm 129.0657 129.2161 128.9152

Roll stiffness Nmm/deg 3098875.8725

Roll steer (per degree) deg/deg 0.0158 0.0124 0.0193

Roll steer deg/mm 0.0 -0.0004 0.0004

Roll camber w.r.t. chassis (per degree) deg/deg -0.5681 -0.571 -0.5652

Roll camber (per degree) deg/deg 0.4319 0.429 0.4349

Roll camber deg/mm 0.0 -0.055 0.0550

LONGITUDINAL COMPLIANCE TEST (Traction):

Traction Steer deg/kN 0.0 -0.0 -0.0

Wheel center longitudinal stiffness N/mm 350.5759 351.0757 350.0760

Caster compliance - traction deg/kN -0.0259 -0.0288 -0.023

LONGITUDINAL COMPLIANCE TEST (Braking):

Brake Steer deg/kN -0.0991 -0.0982 -0.1

Contact patch braking stiffness N/mm 1198.084 1199.2152 1196.9528

Caster compliance - braking deg/kN 1.0574 1.0577 1.0571

LATERAL COMPLIANCE TEST - PARALLEL:

Lateral compliance steer deg/kN 0.4284 0.418 0.4387

Lateral compliance camber deg/kN 0.5005 0.4803 0.5208

Wheel center lateral stiffness N/mm 2251.2354 2246.3669 2256.1038

LATERAL COMPLIANCE TEST - PARALLEL: With Trail 0.0

Lateral compliance steer deg/kN 0.4284 0.418 0.4387

Lateral compliance camber deg/kN 0.5005 0.4803 0.5208

Wheel center lateral stiffness N/mm 2251.2354 2246.3669 2256.1038

LATERAL COMPLIANCE TEST - OPPOSITE:

Lateral compliance steer deg/kN 0.4096 0.4096 0.4096

Lateral compliance camber deg/kN 0.3542 0.3533 0.355

Wheel center lateral stiffness N/mm 12826.0919 12642.0924 13010.0913

ALIGNING TORQUE TEST - PARALLEL:

Aligning torque - steer deg/kNmm -0.0062 -0.0062 -0.0062

Aligning torque - camber deg/kNmm -0.0012 -0.0012 -0.0011

ALIGNING TORQUE TEST - OPPOSITE:

Aligning torque - steer deg/kNmm -0.0061 -0.0061 -0.0061

Aligning torque - camber deg/kNmm -0.0011 -0.0011 -0.0011

SPMM REPORT SIGN CONVENTIONS

PARAMETER CONVENTION FOR POSITIVE VALUE

Toe change Wheel toe in when moved into jounce

Caster change Caster increases when wheel moves into jounce

Camber change Top of tire moves outboard when wheel moves into jounce

Wheel center rise Wheel travelling into jounce

Lat. force deflection Wheel moves to the right when pushed inboard

Lat. force steer Wheel toe in when pushed inboard

Lat. force camber Top of tire moves out when pushed inboard

Brake steer Toe in due to a rearward force at the tire contact patch

Recession steer Toe in due to a rearward force at the wheel center

Tractive force steer Toe in due to a forward force at the wheel center

Aligning torque steer Toe in due to an aligning torque at wheel center

Aligning torque camber Top of tire moves outboard due to an aligning torque at wheel center