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School of Engineering - M.Sc. Automotive Engineering



COMPARATIVE STUDY OF PASSIVE AND ACTIVE ANTI-ROLL BAR SYSTEM FOR VEHICLE RIDE AND HANDLING USING MATLAB/SIMULINK

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ACFA7003 Master Project

DECLARATION

I, JAIMISHKUMAR VALLABHBHAI NAVADIYA, hereby declare that the work presented in this project COMPARATIVE STUDY OF PASSIVE AND ACTIVE ANTI-ROLL BAR SYSTEM FOR VEHICLE RIDE AND HANDLING USING MATLAB/SIMULINK, is entirely my own and has been conducted by the principles of academic integrity and ethical research.

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Date: 24/03/2024

ACKNOWLEDGEMENT

I would like to sincerely thank Mr. Mark Stanford, my project supervisor, for his excellent advice, assistance, and knowledge during this project. The success of this undertaking has been greatly influenced by his support and insightful advice.

Additionally, I want to express my gratitude to Mr. Andrew Noble who helped with this project by lending their knowledge, resources, or unflinching support. This project has been completed successfully in large part because to your combined efforts.

I sincerely appreciate your participation in our attempt and thank you all for your contributions.

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Abstract

There has been an uptick in the number of articles on active and passive anti-roll bars in mechanical and automotive journals. These systems are often designed to alter the vehicle's roll stiffness, so averting the possibility of a roll-over. This study will examine AARB's use from two vantage points: ride and handling. In order to demonstrate how and why it is feasible to manage the car's handling and ride comfort using external pressures on the front anti-roll bar, this study first proposes fundamental vehicle dynamic modelling using a half-car model with four degrees of freedom (DOF). The primary goal of this work is to provide a comprehensive analysis of the dynamic behaviour of vehicles as they are affected by and impact by anti-roll bar mechanisms. The results of simulated tests conducted in MATLAB/SIMULINK demonstrate the reactions of the body's roll angle and roll rate. It is possible to validate the design model and bring performance under control of the PID controller via simulation analysis. At last, we make some broad conclusions on the potential future control mechanisms that may be explored for this type of system. For the purpose of simulating a vehicle's behaviour in the longitudinal direction, this document gives a thorough derivation of a complete vehicle model. An analytical tire dynamics model, namely the Pacejka tire model, is generated and integrated with a 14-degrees-of-freedom (DOF) vehicle model. The next step is to use an instrumented experimental car that responds to the driver's inputs (brake and throttle) to test and verify the whole vehicle model. The vehicle undergoes a battery of short-term handling tests, such as a constant-speed abrupt acceleration and braking test. We compare the experimental result with the model's reaction to motion induced by abrupt braking and throttling. The model validation findings demonstrated that, within a reasonable margin of error, the patterns seen in the experimental data and the simulation results are highly congruent. Using the fully-validated vehicle model as a foundation, we create an active suspension control system that dampens undesirable vibrations when braking and throttling. The system is to be controlled using a PID scheme that incorporates a pitch moment rejection loop. When compared to the passive vehicle system, the results show that the suggested control structure performs better during throttling and braking movements in the given scheme. Another thing to keep in mind is that the system's PID controller may be much more effective with the extra pitch moment rejection loop. To lessen the impact of undesired longitudinal motion, the suggested controller will be used to study the efficacy of a pneumatically operated active suspension system. A comparison of Fuzzy PID is also simulated to study the performance.

CHAPTER 1: INTRODUCTION

1.1 Background

In recent years, the automotive industry has witnessed significant advancements in technology aimed at improving vehicle safety, performance, and passenger comfort. Among various vehicle dynamics control systems, anti-roll bars (ARBs) have emerged as critical components in enhancing vehicle stability and handling. Traditionally, passive anti-roll bars have been widely used due to their simplicity and effectiveness in reducing body roll during cornering. However, with the advent of active control technologies, active anti-roll bars (AARBs) have been introduced, offering adaptive adjustments to roll stiffness in real-time based on driving conditions. This dissertation focuses on a comparative study of passive and active ARB systems to understand their impact on vehicle dynamics, particularly ride quality and handling.

1.2 Statement of the Problem

While passive ARBs contribute to improved handling by reducing body roll, they often compromise ride comfort due to their fixed stiffness characteristics. On the other hand, active ARBs promise enhanced handling and ride comfort by dynamically adjusting roll stiffness. However, the complexity, cost, and energy consumption associated with active systems pose challenges. There is a need for an in-depth comparative analysis to evaluate the trade-offs between passive and active ARB systems in terms of their impact on vehicle performance, safety, and comfort.

1.3 Aim and Objectives

The aim of this study is to evaluate and compare the effectiveness of passive and active anti-roll bar systems in improving vehicle stability, handling, and ride comfort. The specific objectives include:

- Develop and validate dynamic vehicle models incorporating passive and active ARB systems.
- Analyse the impact of ARBs on vehicle performance under various driving manoeuvres.
- Compare the performance, energy consumption, and cost-effectiveness of passive and active ARB systems.
- Design and test control strategies for optimizing active ARB performance.

1.4 Research Questions

The study addresses the following research questions:

- How do passive and active ARB systems affect vehicle ride quality and handling?

- What are the advantages and limitations of passive ARBs compared to active ARBs in terms of vehicle dynamics?
- Can the implementation of advanced control strategies in active ARB systems significantly enhance vehicle performance?
- What are the implications of adopting active ARB systems for the automotive industry in terms of cost, complexity, and energy efficiency?

1.5 Significance of the Study

This research provides valuable insights into the comparative performance of passive and active ARB systems, offering guidance for automotive engineers and designers in selecting and optimizing ARB systems. The findings could contribute to the development of more sophisticated and adaptive vehicle control systems that enhance safety and comfort without compromising performance.

1.6 Structure of the Dissertation

This dissertation is organized into several chapters as follows:

Chapter 2: Literature Review - Reviews existing research on vehicle dynamics, anti-roll bar systems, and control strategies.

Chapter 3: Methodology - Describes the approach for modelling vehicle dynamics and simulating ARB systems.

Chapter 4: Results and Discussion - Presents the findings from the comparative analysis of passive and active ARB systems.

Chapter 5: Conclusion and Recommendations - Summarizes the study's conclusions and suggests directions for future research.

This dissertation presents a comparative analysis of passive and active anti-roll bar (ARB) systems in vehicles, highlighting the advancements from simple passive ARBs to complex active ARBs that adjust to driving conditions in real time. The study aims to assess the trade-offs between the two systems in terms of performance, safety, and comfort, while exploring the benefits of integrating advanced control strategies. The findings aim to guide automotive design, potentially leading to better vehicle control systems that do not compromise on safety or comfort. The structure of the dissertation includes a literature review, methodology, results and discussion, and a concluding chapter with future research directions.

CHAPTER 2: LITERATURE REVIEW

2.1 Historical background

The concept of an Anti-Roll Bar (ARB), also known as a sway bar or stabilizer bar, finds its roots in the early 20th century when vehicle dynamics began to significantly impact automotive design. Initially, these systems were simple, mechanical solutions aimed at reducing the body roll of vehicles during cornering, enhancing stability and safety. The earliest ARB implementations were rudimentary, often comprising rigid bars connected between the left and right suspension systems, leveraging torsional resistance to counteract the forces inducing roll.

Transition to Hydraulic and Active Systems: As automotive technology advanced in the latter half of the 20th century, so did the complexity and effectiveness of ARB systems. The introduction of hydraulic ARB systems marked a pivotal shift, enabling dynamic adjustment of roll stiffness in response to vehicle speed and cornering forces. This period also saw the conceptualization of active ARB systems, which used electronic sensors and actuators to adapt the bar's stiffness in real-time, a significant leap towards modern vehicle dynamics control.

Integration with Electronic Control Systems: The advent of microelectronics and computer-aided engineering in the 1980s and 1990s facilitated the integration of ARBs with vehicle-wide electronic control systems. This era introduced electronically controlled ARB systems that could respond not just to mechanical forces, but also to inputs from an array of sensors, including accelerometers, gyros, and wheel speed sensors. This integration allowed for unprecedented precision in handling and stability control, tailoring the vehicle's response to a variety of driving conditions.

The Age of Advanced Materials and Electromechanical Systems: The turn of the millennium brought with it a focus on weight reduction and efficiency, driving the adoption of advanced materials like high-strength steel and composites in ARB construction. Concurrently, electromechanical ARB systems began to emerge, harnessing electric motors and magnetic clutches to offer more refined, energy-efficient control of roll dynamics. These systems represented the cutting edge in ARB technology, providing optimal balance between performance, safety, and fuel efficiency.

Current Innovations and Future Directions: Today, ARB systems are at the forefront of automotive innovation, integrating with advanced driver assistance systems (ADAS) and forming a critical component of autonomous vehicle dynamics control strategies. Modern ARBs are part of a holistic approach to vehicle stability, working in concert with active suspension systems, electronic stability control (ESC), and other dynamic control systems to ensure maximum safety and comfort. Looking forward, the evolution of ARB technology continues to be closely linked with developments in electric and autonomous vehicles, promising new levels of dynamic performance and safety.

From their humble beginnings as simple mechanical devices to their current status as integral components of advanced vehicle dynamics control systems, ARB systems have undergone a remarkable evolution. Each stage of their development has reflected broader trends in automotive technology, from the shift towards active safety systems to the current focus on electrification and automation. As we look to the future, ARB systems will undoubtedly continue to play a key role in shaping the next generation of vehicle dynamics engineering.

2.2 Development of AARB

In the past several years, a significant amount of research has been directed towards enhancing vehicle stability, particularly with the aim of reducing or altogether eliminating the risk of rollovers, a critical aspect of vehicle safety. The strategies employed to address this issue are diverse, encompassing both mechanical interventions and advanced control systems. Here's an overview of the prevailing approaches and key findings in the domain of Anti-Roll Bar (ARB) systems:

Differential Braking and Electronic Stability Control (ESC)

A common strategy to mitigate lateral acceleration, a precursor to rollovers, involves the integration of differential braking with electronic stability control (ESC). This approach, as detailed by Solmaz et al. (2007), focuses on modulating the yaw rate to which vehicles adhere, aiming to maintain stability. However, Yim (2011) highlights a potential drawback: the reduction in yaw rate reference may compromise manoeuvrability, increasing the risk of collisions by impeding a vehicle's ability to navigate sudden obstacles.

Active Suspension Systems

Another avenue of research explores the development of active suspension systems, which leverage a mix of mechanical, electrical, and hydraulic components to modulate vehicle weight transfer dynamically. These systems are typically classified into three categories:

- **Active Suspension:** Employing comprehensive control over suspension components to adapt to driving conditions actively.
- **Semi-Active Suspension:** Adjusting damping rates in response to changes in driving conditions without altering the spring rates.
- **Active Anti-Roll Bar (AARB) Systems:** Specifically targeting the reduction of body roll through dynamic adjustment of the roll stiffness.

Studies on active suspension systems often use quarter-car models for simulation and testing, focusing on vertical dynamics and comfort but neglecting the roll dynamics' effects on stability and handling.

Cost Considerations and Manufacturing Perspectives

From a manufacturing and economic standpoint, the cost-effectiveness of these technologies is paramount. In this context, Active Anti-Roll Bar (AARB) systems, as discussed by Zulkarnain et al. (2012), present a compelling solution. They offer a significant reduction in body roll, thereby enhancing safety and stability, at a lower cost compared to full active suspension systems.

Passive Anti-Roll Bars

The passive anti-roll bar serves as a fundamental component in mitigating body roll during cornering manoeuvres. By counteracting uneven vertical movements across the wheels, it generates opposing moments that combat lateral acceleration, thereby enhancing stability. However, this system's efficacy is limited when the vehicle encounters irregularities on a straight path, as the transfer of vertical forces can undermine ride comfort, negating some benefits of an independent suspension system.

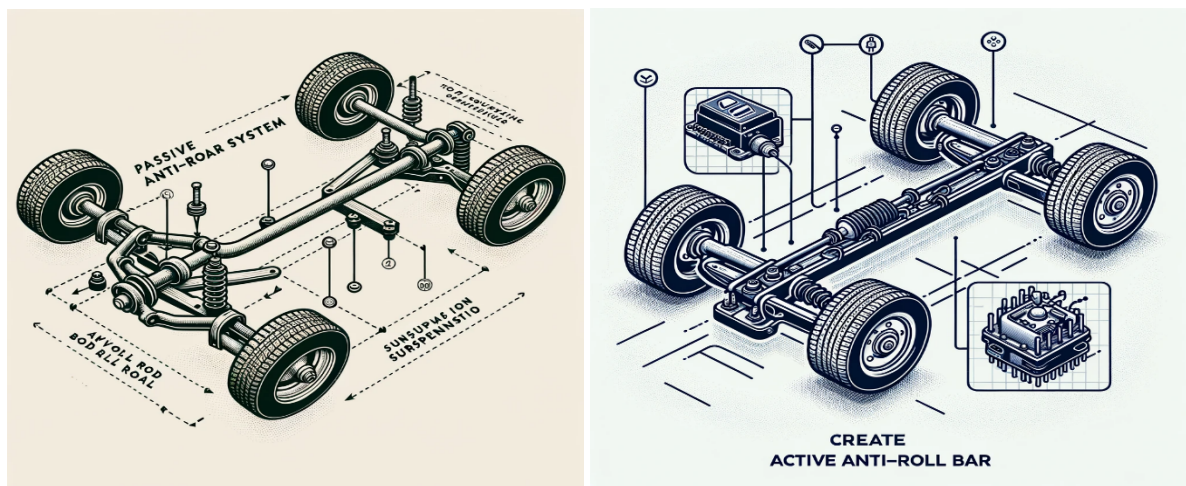


Figure 1: Passive and Active ARB

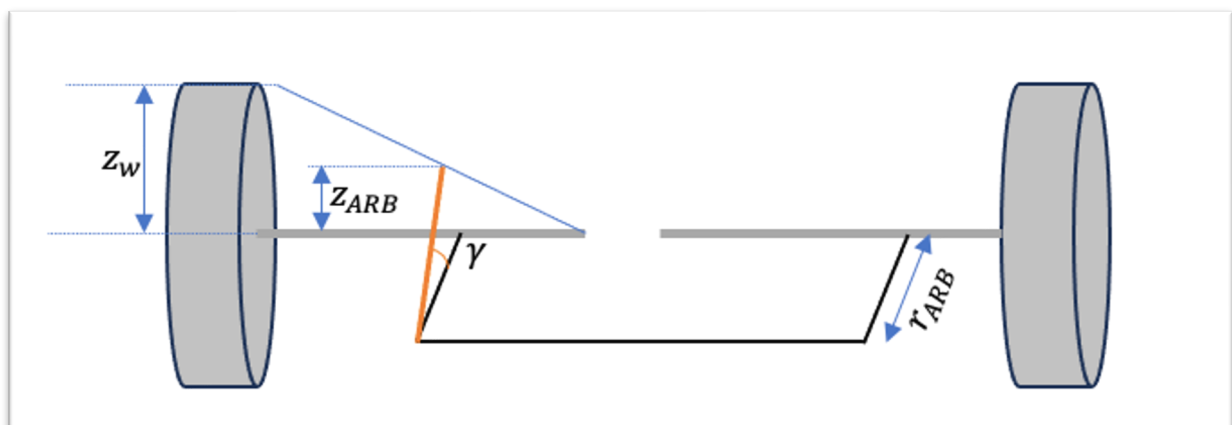


Figure 2: A basic U-shaped passive ARB

To counteract roll motion, the ARB's capacity is determined by its torsional stiffness, geometry, and point of connection to the chassis. In this thesis, the ARB is seen as strictly U-

shaped for simplicity's sake; refer to Figure 19. Moreover, the difference in vertical motion between the wheel z_w and the component under study—in this case, the ARB z_{ARB} is explained by the idea of motion ratio. The suspension kinematics, the ARB's location, and its orientation are all responsible for this discrepancy (Gohl et al. 2004). Equation 29 states that the motion ratio (i_{ARB}) connects the vertical movements and associated forces.

$$i_{ARB} = \frac{z_{ARB}}{z_w} = \frac{F_{ARB, w}}{F_{ARB}}$$

Equation 1: Motion ratio of wheel travel and anti-roll bar arm travel

The forces generated at each of the two ARB ends are denoted by F_{ARB} in equation 29, and the corresponding forces acting on the wheel are denoted by $F_{ARB, w}$. It is assumed that the ARB has an arm length r_{ARB} and torsional stiffness k_γ . Exciting the left wheel while keeping the right wheel immobile, as shown in Figure 19, serves as an example of how the ARB works. As a result, the ARB torques, creating forces at the ends that act on the wheels. Equation 30 is used to find the corresponding force acting on each wheel $F_{ARB, w}$. Note that the same equation can be used with τ_{ARB} by replacing the torque generated by the ARB with the product of the torsional stiffness k_γ and the torsion angle γ .

$$F_{ARB, w} = \frac{k_\gamma \gamma}{2r_{ARB}} i_{ARB} = \frac{\tau_{ARB}}{2r_{ARB}} i_{ARB}$$

Equation 2: Torque equation of the anti-roll bar

Advancements in Active Anti-Roll Bar (AARB) Systems

To overcome the limitations of passive systems, advancements in AARB technology allow for the decoupling of wheel movements on either side of the vehicle. This decoupling significantly diminishes the roll induced by uneven road surfaces, simultaneously improving ride comfort and maintaining stability during cornering. Researchers like Gosselin-Brisson et al. (2009) and Zulkarnain et al. (2014) have contributed to refining AARB systems, showcasing their potential in contemporary automotive design.

The schematic diagram of a Passive Anti-Roll Bar system typically illustrates the component's operational mechanics, highlighting its role in linking opposite wheels to maintain stability. However, the evolution towards more sophisticated AARB systems reflects ongoing efforts to balance safety, comfort, and cost-effectiveness in vehicle design.

Through these studies, it's evident that the pursuit of enhanced vehicle stability, particularly in preventing rollovers, remains a dynamic and multifaceted area of research within automotive engineering. The progression from passive systems to advanced active control mechanisms underscores a continuous drive towards safer, more responsive, and cost-efficient vehicles.

2.3 An overall comparison between active and passive ARB

Active ARB	Passive ARB
Operational Principle	
Utilizes sensors and electronic control units (ECU) to dynamically adjust the stiffness of the roll bar in response to real-time driving conditions, such as cornering speed, lateral acceleration, and steering angle. This system can either stiffen to reduce body roll during aggressive manoeuvres or soften for better ride comfort under normal driving conditions.	Operates based on the mechanical linkage between the wheels of an axle. It uses the torsional stiffness of a metal bar to resist body roll during cornering. The stiffness is constant and not adjustable, meaning the performance is fixed regardless of the driving conditions.
Components and Complexity	
Includes sensors, actuators (hydraulic, pneumatic, or electric), an ECU, and the torsion bar itself. The increased complexity requires more sophisticated design and manufacturing processes, as well as more involved maintenance and higher potential repair costs.	Consists of a torsion bar linked to the suspension of both wheels on an axle, with few moving parts. It's relatively simple in design and operation, making it more straightforward to manufacture and maintain.
Performance and Efficiency	
Provides optimal performance by adapting to driving conditions. It can significantly reduce body roll without compromising ride comfort, as it adjusts the bar's stiffness as needed. This adaptability also contributes to improved tire contact with the road, enhancing traction and safety.	Offers a compromise between ride comfort and handling stability. It can effectively reduce body roll during cornering but at the expense of ride comfort on uneven roads, as the bar's stiffness is constant.
Cost	
More expensive to implement due to its complexity, including the cost of sensors, actuators, and control systems. The initial investment and maintenance costs are higher, making it more commonly found in premium or performance-oriented vehicles.	Generally, less expensive due to its simplicity and the fewer components involved. It's a cost-effective solution for improving handling stability in a wide range of vehicles without significantly increasing the vehicle's price.
Suitability and Applications	

Ideal for performance, luxury, and off-road vehicles where optimal handling and ride comfort are paramount. It enhances the vehicle's adaptability to various driving scenarios, from aggressive cornering on paved roads to navigating uneven terrain.	Suitable for a wide range of vehicles, from economy cars to light trucks, where a balance between cost, ride comfort, and handling stability is desired. It's a universal solution that provides a noticeable improvement in handling performance for most driving conditions.
Impact on Vehicle Dynamics	
By actively controlling body roll and adapting to the road surface, it maintains better overall vehicle dynamics, ensuring both improved handling and comfort. This system can also contribute to enhanced safety and performance by maintaining optimal wheel contact and load distribution.	While it reduces body roll, it can also lead to a less compliant ride over uneven surfaces due to the fixed stiffness, potentially affecting overall vehicle dynamics and passenger comfort.

Table 1: Comparisons between Active and Passive ARB.

The choice between passive and active ARB systems depends on several factors, including vehicle type, intended use, cost considerations, and desired balance between ride comfort and handling stability. While passive ARBs offer a simple, cost-effective solution for reducing body roll, active ARBs provide superior adaptability and performance, catering to the needs of more demanding driving conditions and performance-oriented vehicles.

2.4 Simulation Theory

Like active chassis control systems, issues with active vehicle dynamics may manifest in two distinct areas. In Electronic Stability Control (ESC) systems, the controller determines the vehicle's handling condition and then uses a physical actuator, such the brakes, to deliver a corrective yaw moment. Because the chosen tool must precisely simulate both the controller and the vehicle system dynamics, it is not an easy task to tackle the challenge of ESC control plan development with only one tool. A lot of control system developers use MathWorks' Simulink because of its powerful code generating features and large signal processing libraries. The task of modelling the whole car as a system of physical mechanical, fluid, thermal, and electrical components is best handled by Dymola, an acausal, multidomain tool with a robust multibody code. By using the FMI (Functional Mock-up Interface) standard, users may combine the advantages of Simulink with Dymola, allowing for more accurate simulations without compromising on model quality. By providing an alternative standard for the interface of simulation models, FMI expands beyond the interface of Dymola and Simulink (FMI, 2019). This allows the model to be created in one tool and then imported into another as a functional mock-up unit (FMU). Here, FMI lets you use a Dymola-based car model within Simulink to build the controller with a realistic representation of the vehicle. The issue of how to create a detailed car model to utilize in Simulink may therefore be robustly addressed by deploying FMUs. When designing a framework for modelling the powertrain of Hybrid Electric Vehicles

(HEVs), Cheng et al. (2010) considered a number of tools before settling on Dymola and Simulink in Co-Simulation as the best option. Because of Dymola's superior physical modelling capabilities, control work was carried out in Simulink (the latter being stronger in this area than Simulink), and HEV complexity necessitated the use of Dymola for physical modelling. Despite their differences, a commonality between a hybrid electric vehicle (HEV) and a conventional ICE car with electronic stability control (ESC) is that both use physical systems that communicate with control algorithms. This study will demonstrate the functionality of FMU deployment with Co-Simulation in a Simulink environment by leveraging a car test experiment in Dymola and an ESC model developed in Simulink. In order to make the vehicle model more realistic, the MATLAB software can establish the suspension and steering system using the multi-body dynamic model of the vehicle. This software can describe the components of the suspension and steering system in detail, as well as their connections. It can also account for the flexibility of these connections and local parts. Obviously, the automobile is a very intricate mechanical system; building the model to match the vehicle's actual construction is an enormous undertaking that will be extremely difficult to accomplish. Consequently, the vehicle was appropriately abstracted and simplified throughout the modelling procedure in accordance with the study's goal. This article makes the following claims about the vehicle dynamics model.

Simplified power drive system with only the transmission of power after the axle shaft; driving torque is then added directly to the constant velocity universal joint.

Simplified flexible connection between rigid bodies with the use of a linear (nonlinear) elastic sleeve to mimic real-world dynamic properties.

All parts other than tires, damping elements, elastic components, and rubber components are considered rigid and their deformation is disregarded in the simulation.

Ignored friction force in kinematics pairs. With the vehicle in a stationary position on a flat surface and using Cartesian coordinates as its system of reference, we can see that the x-axis points forward perpendicular to the ground, the z-axis points perpendicular to the top, and the y-axis is on the driver's left side. The origin of the system of references is the point where the centreline of the wheel meets the forward surface of the car. The following are the primary motion characteristics linked to stable handling and comfortable riding: on the z-axis, the yaw rate; on the y-axis, the lateral velocity " v_X "; on the y-axis, the lateral acceleration " a_y "; and so on.

Modelling in MATLAB/Car begins with establishing a template, continues with subsystem generation using those templates, and culminates in model generation by the subsystems themselves (i.e., each subsystem is a combination of different templates). From there, the models of the vehicle and its suspension are constructed. Figure 1 illustrate the established model of the vehicle's suspension system in this study. By adjusting the fundamental parameters of the model in MATLAB/Car system, the remaining models of the steering system, tires, engine, and brake system are generated. Once all of the components

specified in the MATLAB/Car standard interface are put together, we will have the complete model of the car that we want. All parts of the vehicle, including the suspension, are assembled using computer simulations.

Subsystems of the front and rear suspensions, the power transmission, the steering and braking systems, the car body, the lateral stabilizer bar, the front and rear tires, and the front and rear suspensions are all part of the complete vehicle assembly model. The car's dynamics model is obtained by assembling the subsystem models.

In summary, this chapter provides a thorough grounding in the field of ARB systems, underscoring their importance and ongoing evolution within the broader narrative of automotive engineering. It sets the stage for the practical application and simulation-based analysis that are pivotal to this research, indicating a future where ARBs continue to influence the trajectory of vehicle design and functionality.

CHAPTER 3: METHODOLOGY

3.1 Vehicle ride model

A 7-DOF system is used to depict the vehicle ride model. Each corner has four unsprung masses—the front-left, front-right, rear-left, and rear-right wheels—connected to a single sprung mass—the automobile body. Unsprung masses may bounce vertically relative to the sprung mass, while sprung masses can heave, pitch, and roll without restriction. We model the suspensions as passive viscous dampers and spring components between the sprung and unsprung masses. However, the tires are represented as undamped linear springs. All angles of pitch and roll are assumed to be modest for the sake of simplicity. Ikenaga [19] employed a model that was somewhat similar to this one.

Vehicle ride model is represented by the equations of vertical acceleration and moment of inertia about roll axis and pitch axis. This model is consisting of 7 motion of the vehicle which are 4 unsprung mass (front left, front right, rear left and rear right) vertical motion, sprung mass vertical motion, roll rate about longitudinal axis and pitch rate about lateral axis. The equations were extracted from (Ahmad et al. 2009).

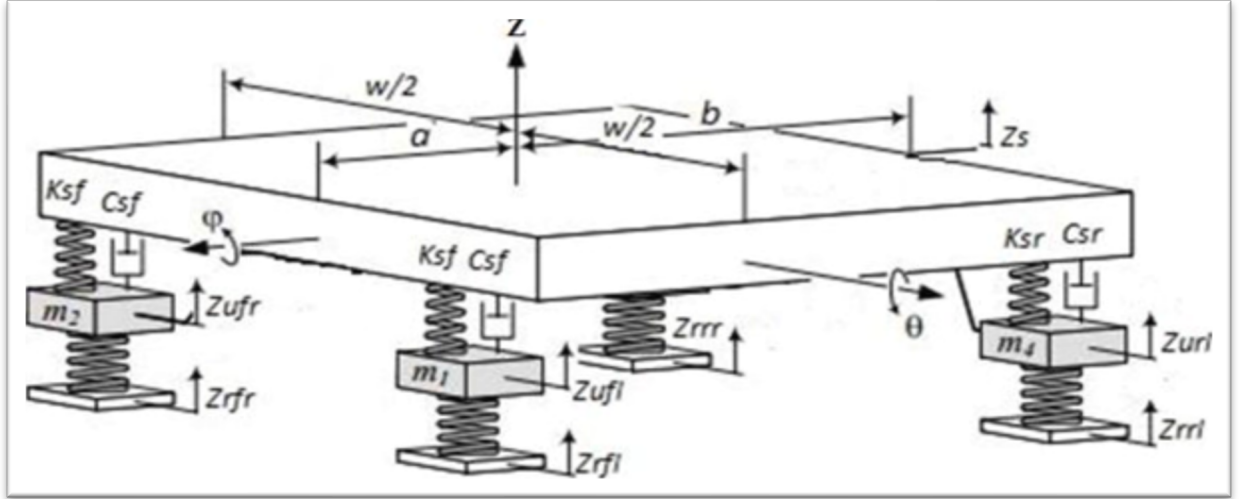


Figure 3: Vehicle 7DOF ride model

Considering Figure 1, Through the application of force balance analysis on the sprung mass, the resulting equation is as follows:

$$m_s \ddot{Z}_s = P_{fl} + P_{fr} + P_{rl} + P_{rr}$$

Equation 3: Sprung mass vertical displacement

Where,

m_s = Sprung mass

$\ddot{Z}_s$ = Sprung mass acceleration in vertical direction

P_{fl} = Suspension force at front left wheel

P_{fr} = Suspension force at front right wheel

P_{rl} = Suspension force at rear left wheel

P_{fr} = Suspension force at rear right wheel

Suspension forces on each wheel can be define as

$$\begin{aligned} P_{fl} &= K_{sf}(Z_{ufl} - Z_{sfl}) + C_{sf}(\dot{Z}_{ufl} - \dot{Z}_{sfl}) \\ P_{fr} &= K_{sf}(Z_{ufr} - Z_{sfr}) + C_{sf}(\dot{Z}_{ufr} - \dot{Z}_{sfr}) \\ P_{rl} &= K_{sr}(Z_{url} - Z_{srl}) + C_{sr}(\dot{Z}_{url} - \dot{Z}_{srl}) \\ P_{rr} &= K_{sr}(Z_{urr} - Z_{srr}) + C_{sr}(\dot{Z}_{urr} - \dot{Z}_{srr}) \end{aligned}$$

Equation 4: Force acting on each corner of sprung mass

Where,

C_{sf} = the front suspension damping coefficient

C_{sr} = the rear suspension damping coefficient

K_{sf} = the front suspension spring stiffness

K_{sr} = the rear suspension spring stiffness

Z_{ufl} = the front left unsprung mass displacement

Z_{ufr} = the front right unsprung mass displacement

Z_{url} = the rear left unsprung mass displacement

Z_{urr} = the rear right unsprung mass displacement

$\dot{Z}_{ufl}$ = the front left unsprung mass velocity

$\dot{Z}_{ufr}$ = the front right unsprung mass velocity

$\dot{Z}_{url}$ = the rear left unsprung mass velocity

$\dot{Z}_{urr}$ = the rear right unsprung mass velocity

$Z_{sfl}, Z_{sfr}, Z_{srl}, Z_{srr}$ are spring mass position at each corner and can be formulated as

$$\begin{aligned} Z_{sfl} &= Z_s - a\theta + 0.5w\phi \\ Z_{sfr} &= Z_s - a\theta - 0.5w\phi \\ Z_{srl} &= Z_s + b\theta + 0.5w\phi \\ Z_{srr} &= Z_s + b\theta - 0.5w\phi \end{aligned}$$

Equation 5: Sprung mass each corner displacement equation

Where,

a = distance from CG to front axle

b = the distance from CG to rear axle

w = wheelbase

θ = pitch angle of sprung mass

ϕ = roll angle of sprung mass

By using value of equation 2 and equation 3. The Equation of sprung mass vertical acceleration can be written as

$$\begin{aligned} m_s \ddot{Z}_s &= -2(K_{sf} + K_{sr})Z_s - 2(C_{sf} + C_{sr})\dot{Z}_s + 2(aK_{sf} - bK_{sr})\theta + 2(aC_{sf} - bC_{sr})\dot{\theta} \\ &\quad + K_{sf}(Z_{ufl} + Z_{ufr}) + C_{sf}(\dot{Z}_{ufl} + \dot{Z}_{ufr}) + K_{sr}(Z_{url} + Z_{urr}) \\ &\quad + C_{sr}(\dot{Z}_{url} + \dot{Z}_{urr}) \end{aligned}$$

Equation 6: Sprung mass vertical acceleration equation

The equation of pitch acceleration ($\ddot{\theta}$) and roll acceleration ($\ddot{\phi}$) can be derived from the moment equilibrium, and can be written as

$$I_{yy}\ddot{\theta} = 2(aK_{sf} - bK_{sr})Z_s + 2(aC_{sf} - bC_{sr})\dot{Z}_s - 2(a^2K_{sf} + b^2K_{sr})\theta - 2(a^2C_{sf} - b^2C_{sr})\dot{\theta} - aK_{sf}(Z_{ufl} + Z_{ufr}) - aC_{sf}(\dot{Z}_{ufl} + \dot{Z}_{ufr}) + bK_{sr}(Z_{url} + Z_{urr}) + bC_{sr}(\dot{Z}_{url} + \dot{Z}_{urr}) - m_s h_{cgM} a_x$$

Equation 7: Pitch angle equation

$$I_{xx}\ddot{\phi} = -0.5w^2(K_{sf} + K_{sr})\phi - 0.5w^2(C_{sf} + C_{sr})\dot{\phi} + 0.5wK_{sf}(Z_{ufl} - Z_{ufr}) + 0.5wC_{sf}(\dot{Z}_{ufl} - \dot{Z}_{ufr}) + 0.5wK_{sr}(Z_{url} - Z_{urr}) + 0.5wC_{sr}(\dot{Z}_{url} - \dot{Z}_{urr}) + m_s h_{cgM} a_y - \frac{w}{2}F_{fl} + \frac{w}{2}F_{fr} - \frac{w}{2}F_{rl} + \frac{w}{2}F_{rr}$$

Equation 8: Roll angle equation

Where,

a_x = longitudinal acceleration of vehicle

a_y = lateral acceleration of vehicle

F_{fl} = Front left anti-roll force

F_{fr} = Front right anti-roll force

F_{rl} = Rear left anti-roll force

F_{rr} = Rear right anti-roll force

I_{xx} = moment of inertia of vehicle about pitch axis

I_{yy} = moment of inertia of vehicle about roll axis

h_{cgM} = center of gravity height

The vertical Acceleration of Sprung mass is calculated by the following equations.

$$m_{uf}\ddot{Z}_{ufl} = (K_{sf})Z_s + (C_{sf})\dot{Z}_s - (aK_{sf})\theta - (aC_{sf})\dot{\theta} + 0.5wK_{sf}\phi + 0.5wC_{sf}\dot{\phi} - (K_{sf} + K_t)Z_{ufl} - C_{sf}\dot{Z}_{ufl} + K_t Z_{rfl} - F_{fl}$$

$$m_{uf}\ddot{Z}_{ufr} = (K_{sf})Z_s + (C_{sf})\dot{Z}_s - (aK_{sf})\theta - (aC_{sf})\dot{\theta} + 0.5wK_{sf}\phi + 0.5wC_{sf}\dot{\phi} - (K_{sf} + K_t)Z_{ufr} - C_{sf}\dot{Z}_{ufr} + K_t Z_{rfr} - F_{fr}$$

$$m_{ur}\ddot{Z}_{url} = (K_{sr})Z_s + (C_{sr})\dot{Z}_s - (aK_{sr})\theta - (aC_{sr})\dot{\theta} + 0.5wK_{sr}\phi + 0.5wC_{sr}\dot{\phi} - (K_{sr} + K_t)Z_{url} - C_{sr}\dot{Z}_{url} + K_t Z_{rrl} - F_{rl}$$

$$m_{ur}\ddot{Z}_{urr} = (K_{sr})Z_s + (C_{sr})\dot{Z}_s - (aK_{sr})\theta - (aC_{sr})\dot{\theta} + 0.5wK_{sr}\phi + 0.5wC_{sr}\dot{\phi} - (K_{sr} + K_t)Z_{urr} - C_{sr}\dot{Z}_{urr} + K_t Z_{rrr} - F_{rr}$$

Equation 9: Unsprung mass vertical acceleration

Where,

K_t = tire stiffness

m_{uf} = front unsprung mass

m_{ur} = rear unsprung mass

$\ddot{z}_{ufl}$, $\ddot{z}_{ufr}$, $\ddot{z}_{url}$, $\ddot{z}_{urr}$ are the front left and right and rear left and right unsprung mass vertical acceleration respectively.

3.2 Handling model (Bicycle model)

Low speed cornering

It's thought that when a car turns slowly (as in a parking lot), its tyres don't need to exert any lateral force, hence there isn't any slip angle created. In Figure 3, it is seen that the car in this example is circling a stationary point.

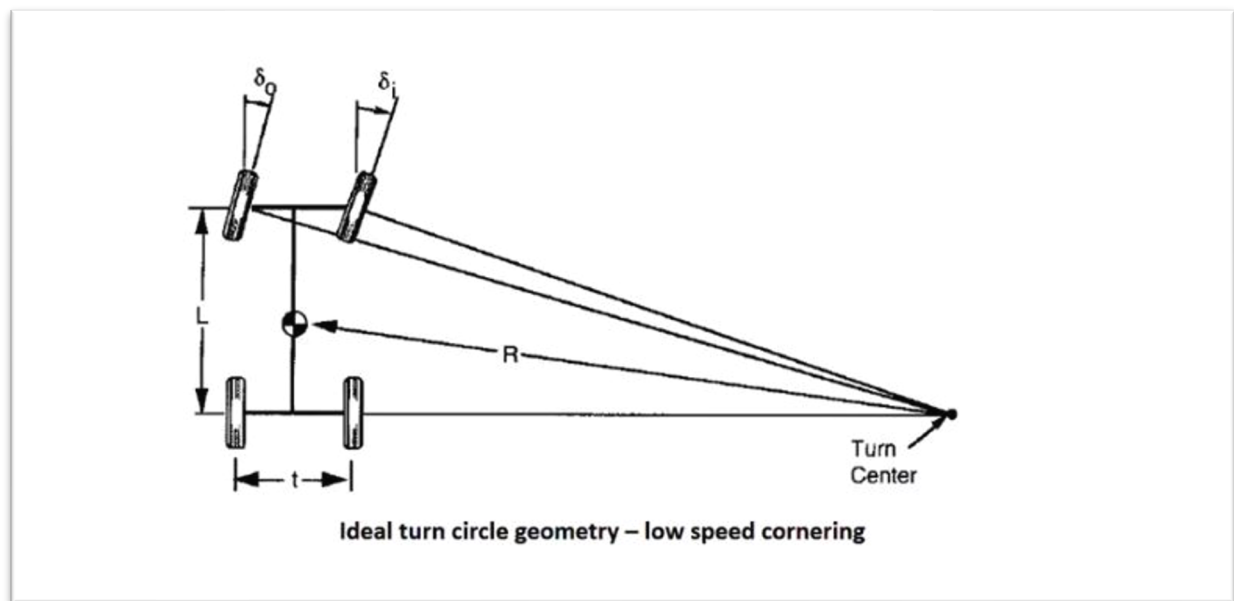


Figure 4: Low speed cornering

The term "turn centre" refers to the location that the car turns around. Every wheel and the turn centre's lines are perpendicular to each other. This is because there isn't any slide when the manoeuvre is being performed. The steering angle, represented by δ_o for the outside wheel and δ_i for the inside wheel, is the angle formed by the direction the wheel is pointing and the direction the vehicle is heading (Gillespie 1992).

For small angle assumption, steering angle

$$\delta_o = \frac{L}{R + \frac{t}{2}}$$

$$\delta_i = \frac{L}{R - \frac{t}{2}}$$

δ_o = steering angle of the outer wheel

δ_i = steering angle of the inner wheel

R = turning radius

t = trackwidth

The term "Ackerman steering" or "Ackerman geometry" refers to this relationship. At low speeds, this is the optimal steering angle for the vehicle's "bicycle model." The left and right wheels are treated as one wheel in the "bicycle model," with each wheel producing twice as much vertical force and twice as much lateral force at the same slip angle. This indicates that the two-dimensional model does not account for body roll or lateral load transfer (Gillespie 1992).

High speed cornering

The vehicle's lateral acceleration is crucial to handling characteristics during high-speed cornering and cannot be overlooked. In order to offset the lateral acceleration, the tyres must therefore generate lateral forces. This implies that each wheel will produce a slip angle.

The velocity and angle are cited in figure 4. The centre of gravity of the model is at (x,y) . velocity of the vehicle is V . Steering angle is δ_f . Angle made by the velocity vector to the line of CG is β . ψ is the current heading angle made by the axis of vehicle model. l_f and l_r is the distance of center of front and rear wheel from center of gravity point. Radius of cornering (distance between O to the point of centre of gravity) is R .

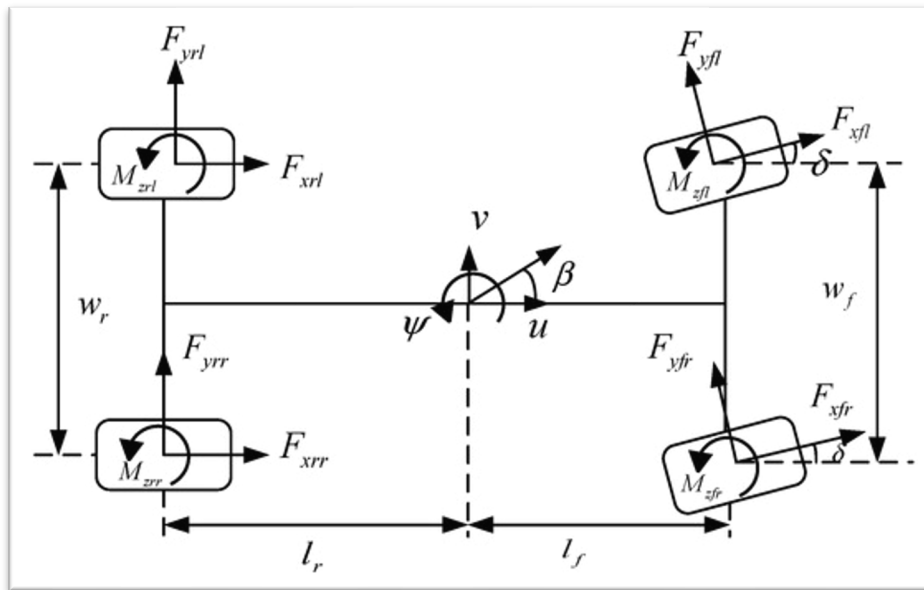


Figure 5: Dynamic Bicycle Model with linear motion. (The notation v and u substitute by V_x and V_y respectively.)

From the geometry of model, yaw velocity is given by,

$$\omega = \frac{v}{R}$$

Equation 10: Yaw velocity of vehicle

And linear velocity components in longitudinal v_x and lateral v_y direction are given by following equations respectively.

$$V_x = V \cos(\psi + \beta)$$

$$V_y = V \sin(\psi + \beta)$$

Equation 11: Linear velocity component of bicycle model

From geometry of figure 4, $\triangle AOB$,

$$\tan \delta = \frac{l_f + l_r}{R_1}$$

From geometry of figure 4, $\triangle OAV$,

$$\tan \beta = \frac{l_r}{R_1} = \frac{l_r \tan \delta}{l_f + l_r}$$

$$\cos \beta = \frac{R_1}{R} = \frac{l_f + l_r}{\tan \delta} \times \frac{1}{R}$$

$$R = \frac{l_f + l_r}{\tan \delta} \times \frac{1}{\cos \beta} = \frac{l_f + l_r}{\tan \delta \cos \beta}$$

So, by substituting the value of R in Equation 11,

$$\omega = \frac{V}{\frac{l_f + l_r}{\tan \delta \cos \beta}} = \frac{V \tan \beta \cos \delta}{l_f + l_r}$$

Equation 12: Yaw rate in term of side slip angle and steering angle

Where,

$$\beta = \tan^{-1} \left(\frac{l_r \tan \delta}{l_f + l_r} \right)$$

These formulas aid in establishing the motion and force model for the vehicle. Some assumptions about this model, such as treating the car body like a two-wheeled bicycle. This model's centre of gravity is located at point C. The motion is limited to the X and Y axes. Vehicle has a completely rigid body. The model's longitudinal velocity is constant.

Where,

m_s = Vehicle sprung mass

I_{zz} = Vehicle moment of inertia about Z axis

l_f = Distance from CG to front axle

l_r = Distance from CG to rear axle

$C_{\alpha r}$ = Cornering stiffness of rear tire

$C_{\alpha f}$ = Cornering stiffness of front tire

$\ddot{\theta}$ = angular acceleration of the vehicle

I_{zz} = moment of inertia in lateral direction

F_{xfl} = Longitudinal force acting on the front left wheel

F_{xfr} = Longitudinal force acting on the front right wheel

F_{xrl} = Longitudinal force acting on rear left wheel

F_{xrr} = Longitudinal force acting on rear right wheel

F_{yfl} = lateral force acting on the front left wheel

F_{yfr} = lateral force acting on the front right wheel

F_{yrl} = lateral force acting on rear left wheel

F_{yrr} = lateral force acting on rear right wheel

f_{rfl} = Friction force on front left wheel

f_{rfr} = Friction force on front right wheel

f_{rrl} = Friction force on rear left wheel

f_{rrr} = Friction force on rear right wheel

M_{tfl} = Self aligning torque on front left wheel

M_{tfr} = Self aligning torque on front right wheel

M_{trl} = Self aligning torque on rear left wheel

M_{trr} = Self aligning torque on rear right wheel

a_y = Lateral acceleration of vehicle

V = Vehicle velocity

V_x = vehicle longitudinal velocity

V_y = vehicle lateral velocity

ω = vehicle angular velocity (Yaw rate)

V_{xf} = Front wheel longitudinal velocity

V_{yf} = Front wheel lateral velocity

V_{xr} = Rear wheel longitudinal velocity

V_{yr} = Rear wheel lateral velocity

α_f = Front wheel slip angle

α_r = Rear wheel slip angle

m = Total vehicle mass

The longitudinal acceleration of the vehicle's CG a_y is composed of two terms: $\dot{V}_x$ is the result of acceleration along the x-axis, while the rotation of the vehicle around the z-axis produces centripetal acceleration. Lateral acceleration of the body is derived from the following equation.

$$ma_y = \sum F_y = F_{yr} + F_{yf} \cos \delta$$

If steering angle δ is very small than $\cos \delta = 1$.

$$ma_y = F_{yr} + F_{yf}$$

a_y has two components because one from lateral velocity component and other one is from centripetal force.

Centripetal force $\frac{mv^2}{R}$. So centripetal acceleration is due to centripetal force is define as

$$\frac{v^2}{R} = \frac{vR\omega}{R} = V\omega.$$

And other component is due to centrifugal force which is $\dot{V}_y$.

$$\dot{V}_y = a_y + \omega V_x$$

Where a_y can be calculated from force equilibrium along Y- axis,

$$ma_y = (F_{xfl} - f_{rfl})\sin\delta + F_{yfl}\cos\delta + (F_{xfr} - f_{rfr})\sin\delta + F_{yfr}\cos\delta + F_{yrl} + F_{yrr}$$

Equation 13: lateral acceleration equation

Similarly, longitudinal acceleration can be defined as

$$\dot{V}_x = a_x - \omega V_y$$

Where a_x can be obtained by force equilibrium along X- axis,

$$ma_x = (F_{xfl} - f_{rfl})\cos\delta - F_{yfl}\sin\delta + (F_{xfr} - f_{rfr})\cos\delta + F_{yfr}\sin\delta + F_{xrl} + F_{xrr} \\ - f_{rrl} - f_{rrr}$$

Equation 14: longitudinal acceleration equation

Vehicle yaw acceleration can be calculated from the force and moment equilibrium along Z- axis.

$$I_{zz}\ddot{\psi} = \frac{w_f}{2}(F_{xfl} - f_{rfl})\cos\delta + \frac{w_f}{2}(F_{xfr} - f_{rfr})\cos\delta - \frac{w_r}{2}F_{xrl} + \frac{w_r}{2}F_{xrr} \\ + \frac{w_f}{2}F_{yfl}\sin\delta - \frac{w_f}{2}F_{yfr}\sin\delta - l_rF_{yrl} - l_rF_{yrr} + l_fF_{yfl}\cos\delta \\ + l_fF_{yfr}\cos\delta + l_f(F_{xfl} - f_{rfl})\sin\delta + l_f(F_{xfr} - f_{rfr})\sin\delta + \frac{w_r}{2}f_{rrl} \\ - \frac{w_r}{2}f_{rrr} - M_{tfl} - M_{tfr} - M_{trl} - M_{trr}$$

Equation 15: Yaw acceleration equation

By integrating $\dot{V}_x$ and $\dot{V}_y$, the longitudinal and lateral velocities, or $\dot{V}_x$ and $\dot{V}_y$ can be derived. These velocities are essential for determining the tyre forces by computing the longitudinal slip ratio and side slip angle.

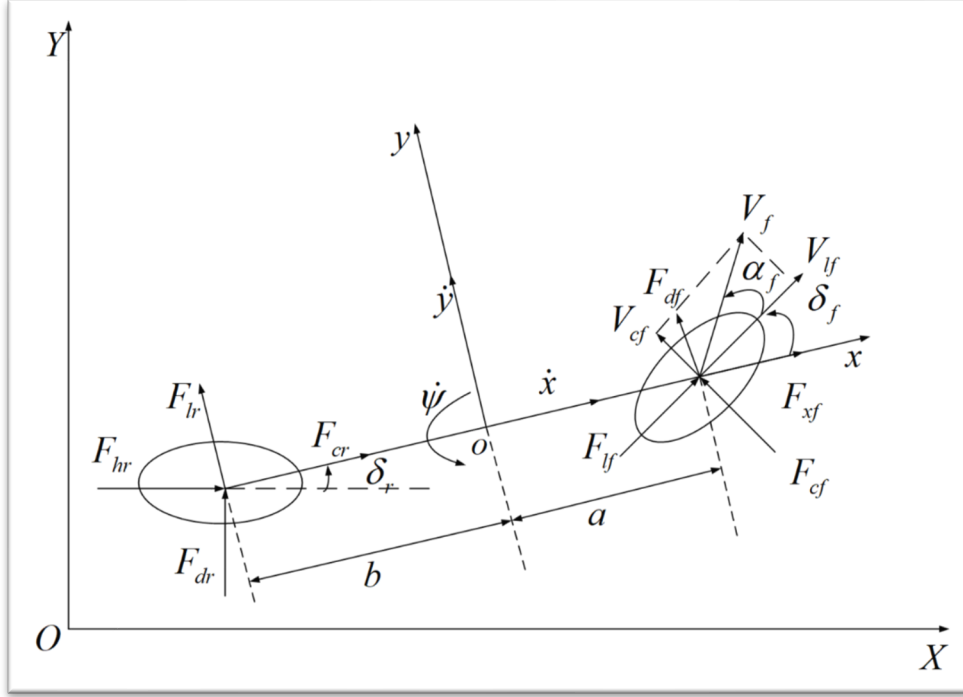


Figure 6: Global coordinate system (XY) and vehicle coordinate system (xy)

From vehicle coordinate system,

$$\alpha_r \approx \tan \alpha_r = \frac{\overline{V_{B,y}}}{\overline{V_{B,x}}}$$

And

$$\alpha_f \approx \tan \alpha_f = \frac{\overline{V_{A,\eta}}}{\overline{V_{A,\epsilon}}}$$

From kinematics,

$$\begin{aligned} V_{B,x} &= V_x \\ V_{B,y} &= V_y - \omega l_r \\ V_{A,x} &= V_x \\ V_{A,y} &= V_y + \omega l_f \end{aligned}$$

coordinate transformation relation between vehicle coordinate system to Global coordinate system $(X, Y) \rightarrow (\epsilon, \eta)$.

$$\begin{aligned} V_{A,\epsilon} &= V_{A,x} \cos \delta - V_{A,y} \sin \delta = V_x + (V_y + \omega l_f) \delta \\ V_{A,\eta} &= -V_{A,x} \sin \delta + V_{A,y} \cos \delta = -V_x \delta + (V_y + \omega l_f) \end{aligned}$$

By substituting these values in the equation of slip angle.

$$\begin{aligned} \alpha_f \approx \tan \alpha_f &= \frac{\overline{V_{A,\eta}}}{\overline{V_{A,\epsilon}}} = \frac{-V_x \delta + (V_y + \omega l_f)}{V_x + (V_y + \omega l_f) \delta} \\ \alpha_r \approx \tan \alpha_r &= \frac{\overline{V_{B,y}}}{\overline{V_{B,x}}} = \frac{V_y - \omega l_r}{V_x} \end{aligned}$$

By assuming, $V_x \gg (V_y + \omega l_f)\delta$, so dominating part $V_x + (V_y + \omega l_f)\delta = V_x$.

Therefore,

$$\alpha_f = \tan^{-1}\left(\frac{V_y + \omega l_f}{V_x}\right) - \delta$$

Equation 16: Front wheel slip angle equation

And

$$\alpha_r = \tan^{-1}\left(\frac{V_y - \omega l_r}{V_x}\right)$$

Equation 17: Rear wheel slip angle equation

3.3 Pacejka Tyre model

The Pacejka Tire Model, also known as the Magic Formula Tire Model, is highly relevant to research involving vehicle dynamics and control systems, such as in studies of Anti-Roll Bar (ARB) systems. The Pacejka Tire Model provides a realistic mathematical representation of tire behaviour under various loading conditions and slip angles. It is crucial for simulating how tires interact with the road surface, which directly affects the vehicle's handling and stability. This model allows for integration with full vehicle dynamics models, which is essential when assessing the overall performance of ARB systems. The influence of tire forces on the vehicle's roll, pitch, and yaw movements is a critical component of such studies. Active ARB systems rely on precise control algorithms to adjust the roll stiffness dynamically. The Pacejka Tire Model can enhance the design of these control systems by providing accurate tire force predictions that respond to changes in ARB settings. The trade-off between ride comfort and handling is a key aspect of ARB research. The Pacejka Tire Model helps quantify the impact of ARB adjustments on tire-road contact, which in turn influences ride quality and cornering performance. By incorporating the Pacejka Tire Model into simulations, researchers can optimize the parameters of both passive and active ARB systems to achieve desired levels of safety, comfort, and performance. The Pacejka Tire Model is integral to this research as it provides a comprehensive and dynamic understanding of tire forces, which are essential for designing, optimizing, and evaluating ARB systems in the context of vehicle dynamics.

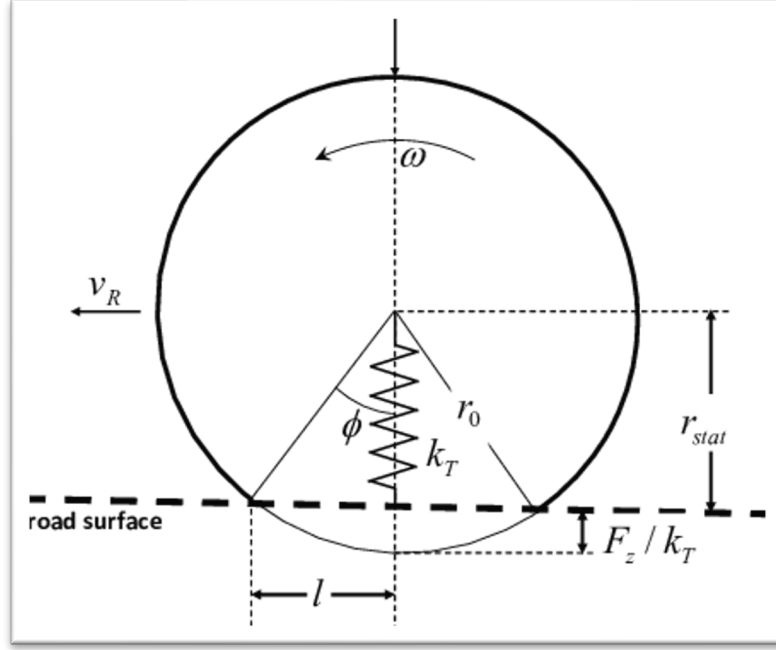


Figure 7: Tyre spinning motion.

The torque about wheel axle on each wheel can be defined by the following equation.

$$\begin{aligned}
 I_w \dot{\omega}_{fl} &= T_{af} - T_{bf} + (F_{xfl} - f_{rfl})e - F_{zfl} L \\
 I_w \dot{\omega}_{fr} &= T_{af} - T_{bf} + (F_{xfr} - f_{rfr})e - F_{zfr} L \\
 I_w \dot{\omega}_{rl} &= T_{ar} - T_{br} + (F_{xrl} - f_{rrl})e - F_{zrl} L \\
 I_w \dot{\omega}_{rr} &= T_{ar} - T_{br} + (F_{xrr} - f_{rrr})e - F_{zrr} L
 \end{aligned}$$

Equation 18: Torque at each tire

I_w = Moment of inertia of wheel

$\dot{\omega}_{ij}$ = wheel spin acceleration (i = front and rear, j = left and right)

T_{ai} = Driving torque (i = front and rear)

T_{bi} = Braking torque (i = front and rear)

$$e = \frac{F_z}{k_T} = r_o(1 - \cos\phi)$$

$$L = r_o(\sin\phi)$$

r_o = tire radius

Tire friction force can be calculated by following equations:

$$f_{rfl} = \sigma F_{zfl}$$

$$f_{rfr} = \sigma F_{zfr}$$

$$f_{rrl} = \sigma F_{zrl}$$

$$f_{rrr} = \sigma F_{zrr}$$

Equation 19: Friction force on each tire

F_{zfl} = Front left wheel normal force

F_{zfr} = Front right wheel normal force

F_{zrl} = Rear left wheel normal force

F_{zrr} = Rear right wheel normal force

where tyre speed is the primary determinant of the rolling resistance coefficient, σ (v). The value of σ vs V tyre speed is displayed in Table 2. The tyres are 235/70 R15, the normal size specified by the vehicle's handbook (2001) and utilised in the simulation in Section 4; the tyre pressure is approximately 35 Psi.

Rolling speed v (m/s)	Resistance coefficient(σ)
0	0
1.126	0.0129
6.944	0.0129
13.888	0.01376
21.02	0.014178
27.777	0.015205
34.722	0.016438
41.666	0.0182
48.423	0.021369
55.555	0.024657

Table 2: Rolling resistance coefficient value on different speed. (Wiegand 2016)

Normal forces on each wheel can be calculated by the following equations:

$$F_{zfl} = \frac{mgl_r}{2(l_f + l_r)} - m_s a_x \frac{H_{cg}}{2(l_f + l_r)} - \frac{K_{\phi f}}{K_{\phi f} + K_{\phi r}} \left(\frac{m_s a_y H_{cg}}{w_f} \right) + F_{fl}$$

$$F_{zfr} = \frac{mgl_r}{2(l_f + l_r)} - m_s a_x \frac{H_{cg}}{2(l_f + l_r)} + \frac{K_{\phi f}}{K_{\phi f} + K_{\phi r}} \left(\frac{m_s a_y H_{cg}}{w_f} \right) + F_{fr}$$

$$F_{zrl} = \frac{mgl_f}{2(l_f + l_r)} + m_s a_x \frac{H_{cg}}{2(l_f + l_r)} - \frac{K_{\phi r}}{K_{\phi f} + K_{\phi r}} \left(\frac{m_s a_y H_{cg}}{w_r} \right) + F_{rl}$$

$$F_{zrr} = \frac{mgl_f}{2(l_f + l_r)} + m_s a_x \frac{H_{cg}}{2(l_f + l_r)} + \frac{K_{\phi r}}{K_{\phi f} + K_{\phi r}} \left(\frac{m_s a_y H_{cg}}{w_r} \right) + F_{rr}$$

Equation 20: Normal force on tire

$K_{\phi f}$ = Front roll stiffness

$K_{\phi r}$ = Rear roll stiffness

g = Gravitational acceleration (9.81 m/s/s)

3.4 Understeer and Oversteer

The definition of Ackermann steer angle $\bar{\delta}$ is the angle formed by the line connecting O to the mid-point of the rear wheel and O to the mid-point of the front wheel, as illustrated in figures 7 and 8.

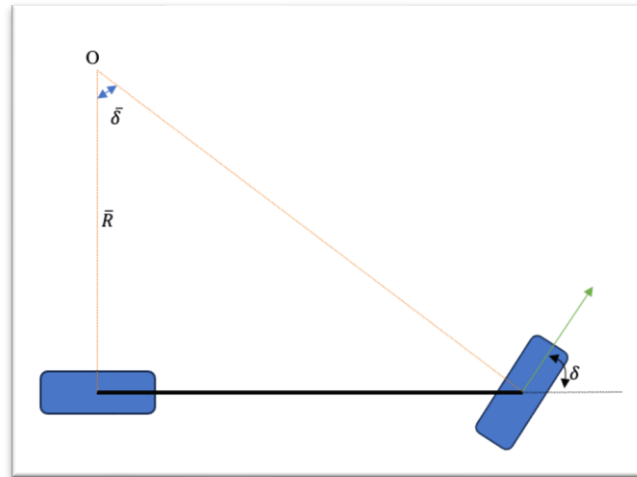


Figure 8: Kinematic bicycle model

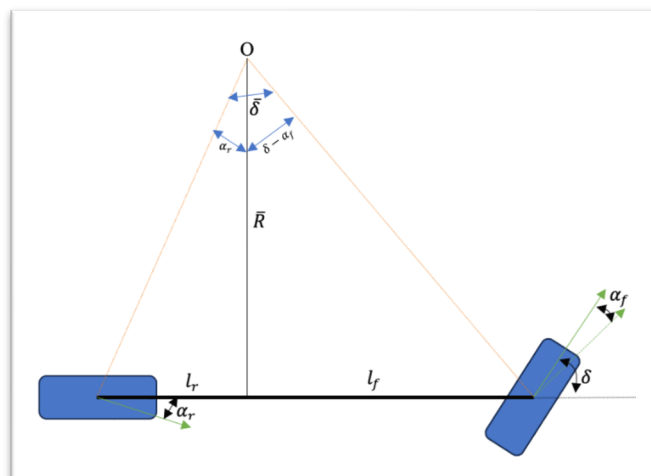


Figure 9: Dynamic bicycle model

Tangential Radius:

Let $\bar{C}$ represent the location where the slip angle equals 0 in the kinematic bicycle model. The distance between the instantaneous center turns O and $\bar{C}$ is known as the tangential radius $\bar{R}$.

In the kinematic model, $\bar{\delta}$ is always equal to δ , as illustrated in figure 7. However, from figure 8, $\bar{\delta} = \delta - \alpha_f + \alpha_r$.

For kinematic model figure 7,

$$\tan \bar{\delta} = \frac{l}{\bar{R}}$$

$$\bar{\delta}_{kinematic} = \tan^{-1} \frac{l}{\bar{R}}$$

For dynamic model figure (9),

$$\tan \alpha_r = \frac{l_r}{\bar{R}}$$

And

$$\tan(\delta - \alpha_f) = \frac{l_f}{\bar{R}}$$

But $\bar{\delta} = \delta - \alpha_f + \alpha_r$

$$\bar{\delta}_{dyn} = \tan^{-1} \frac{l_r}{\bar{R}} + \tan^{-1} \frac{l_f}{\bar{R}}$$

A dynamic bicycle model is said to be neutral steering during steady state cornering, if its steering angle $\bar{\delta}_{dyn}$ equals so that of a kinematic bicycle model $\bar{\delta}_{kinematic}$ when both are driving on a circle with same tangential radius $\bar{R}$.

Comparison of Ackermann angles for the same radius $\bar{R}$

$$\bar{\delta}_{kinematic} = \tan^{-1} \frac{l}{\bar{R}} = \frac{l}{\bar{R}}$$

And

$$\bar{\delta}_{dyn} = \tan^{-1} \frac{l_r}{\bar{R}} + \tan^{-1} \frac{l_f}{\bar{R}} = \frac{l_r}{\bar{R}} + \frac{l_f}{\bar{R}} = \frac{l}{\bar{R}}$$

So, from the above equations, $\bar{\delta}_{kinematic} = \bar{\delta}_{dyn}$ if both model drive on a circle with same $\bar{R}$. Substituting the steering angle $\bar{\delta}_{kinematic} = \delta_{kin}$ and $\bar{\delta}_{dyn} = \delta_{dyn} - \alpha_f + \alpha_r$.

$$\delta_{dyn} = \delta_{kin} + \alpha_f - \alpha_r$$

Equation 21: Relation between Kinetic steering angle and Dynamic steering angle

Equation 22 Relation between kinematic steering angle and dynamic steering angle

So, the dynamic bicycle model is neutrally steering if $\alpha_f = \alpha_r$.

If $\alpha_f < \alpha_r$ oversteering condition.

If $\alpha_f > \alpha_r$ understeering condition.

Understeer gradient

Understeer or oversteer behaviour of the vehicle changes with the vehicle lateral acceleration. If the lateral acceleration is small (which is by low speed and/or large turning radius), the slip angle generally small and hence the vehicle is almost neutrally steer.

Understeer gradient quantifies the rate of changes of a vehicle's understeer (or oversteer, if the gradient is negative) with its lateral acceleration.

Assumption:

Stationary conditions during steady state cornering

Small steering angle

Steady state cornering $\delta \cong \delta_{dyn} = \delta_{kin} + \alpha_f - \alpha_r$

Linear tyre model where $F_f = C_{\alpha f} \alpha_f$ and $F_r = C_{\alpha r} \alpha_r$.

The cornering stiffness coefficient $C_{\alpha f}$ and $C_{\alpha r}$ depends linearly on vertical tire load.

$$\begin{aligned}\sum F_{i,z} &= 0 \\ F_{zf} + F_{zr} &= mg \\ \sum_{COG} M_{i,z} &= 0 \\ F_{zf} l_f - F_{zr} l_r &= 0\end{aligned}$$

From above equations $F_{zf} = \frac{l_r}{l} mg$ and $F_{zr} = \frac{l_f}{l} mg$. where $\frac{l_r}{l} = W_f$ is fraction of weight in front tire and $\frac{l_f}{l} = W_r$ is fraction of weight on rear tire.

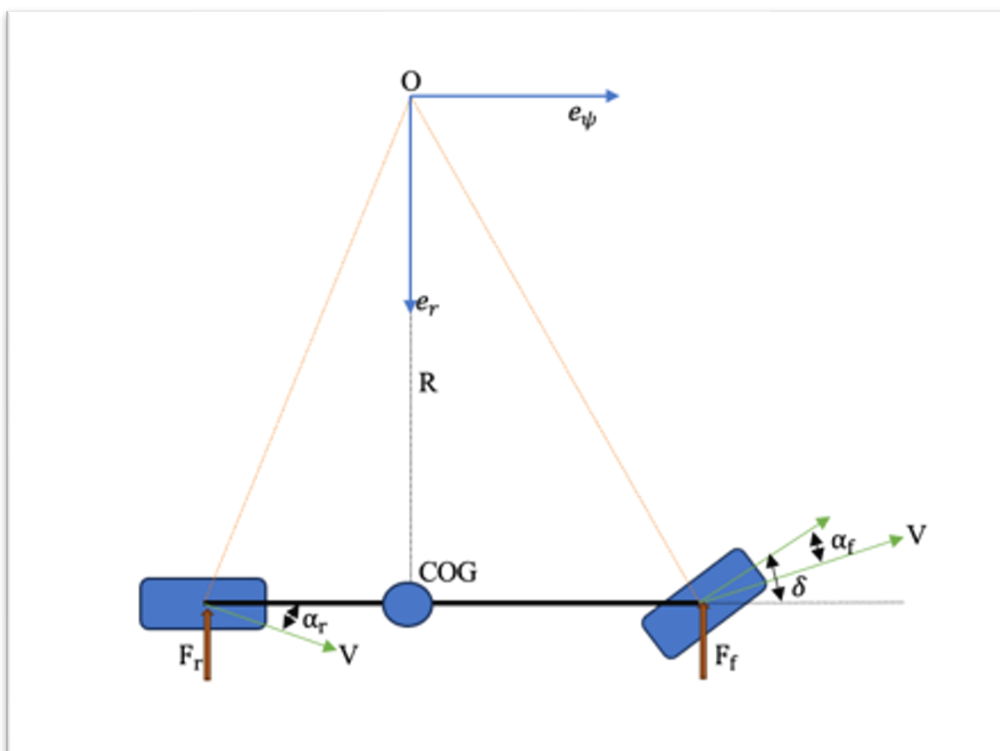


Figure 10: Dynamic model

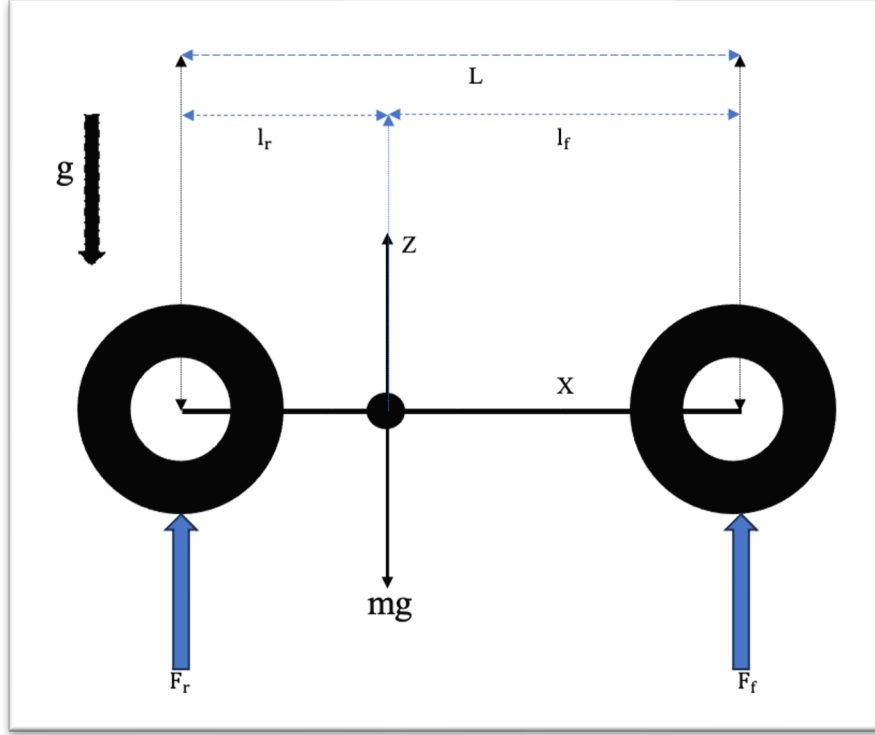


Figure 11: Force acting on dynamic vehicle model

Considering figure (10), balance of forces in radial direction e_r .

$$ma_r = -F_{zf} - F_{zr} \quad (i1)$$

$$a_r = \text{centripetal acceleration} = -\frac{V_{long}^2}{R}$$

Balancing of moment in Z-direction

$$I_z \ddot{\psi} = F_{zf} l_f - F_{zr} l_r \quad (i2)$$

$\ddot{\psi} = 0$ for stationary rotation.

Solving for equation (i1) and (i2).

$$F_f = \frac{l_r}{l} \frac{V_{long}^2}{R}$$

And

$$F_r = \frac{l_f}{l} \frac{V_{long}^2}{R}$$

Substituting into linear tyre model force equation for front and rear slip angle:

$$\alpha_f = \frac{1}{C_{\alpha f}} F_f = \frac{W_f}{C_{\alpha f}} \frac{V_{long}^2}{R}$$

And

$$\alpha_r = \frac{1}{C_{ar}} F_r = \frac{W_r}{C_{ar}} \frac{V_{long}^2}{R}$$

By knowing Steady state cornering condition;

$$\delta_{dyn} = \delta_{kin} + \alpha_f - \alpha_r = \delta_{kin} + \left[\frac{W_f}{C_{af}} - \frac{W_r}{C_{ar}} \right] \frac{V_{long}^2}{R}$$

The equation effectively illustrates how various parameters affect the necessary steering angle. The steering input required, in terms of both magnitude and direction, is determined by the expression $\left[\frac{W_f}{C_{af}} - \frac{W_r}{C_{ar}} \right]$. It is composed of two terms, each representing the wheel load to cornering stiffness ratio. The understeer gradient is the term for this (K) and $\frac{V_{long}^2}{R}$ is lateral acceleration. There are three options for handling a vehicle in a steady state: neutral steer, understeer, and oversteer. Now, each of these is covered in following points:

- Neutral steer

If $\left[\frac{W_f}{C_{af}} = \frac{W_r}{C_{ar}} \right]$ that means $\alpha_f = \alpha_r$

This is known as neutral steer. Regardless of the vehicle speed or lateral acceleration, the steering wheel angle in a constant radius test remains constant and is equivalent to the Ackerman steering angle.

- Understeer

If $\left[\frac{W_f}{C_{af}} > \frac{W_r}{C_{ar}} \right]$ that means $\alpha_f > \alpha_r$.

It's known as understeer. A constant radius test would require the steering angle to increase as the speed increased. The lateral acceleration, or the square of the speed, is directly proportional to the steering angle growth. A greater steering angle puts more lateral force on the front tyres in an effort to keep the total moments around the CG point at zero. The tendency of the vehicle's front to stray outward is countered by increasing the steering angle.

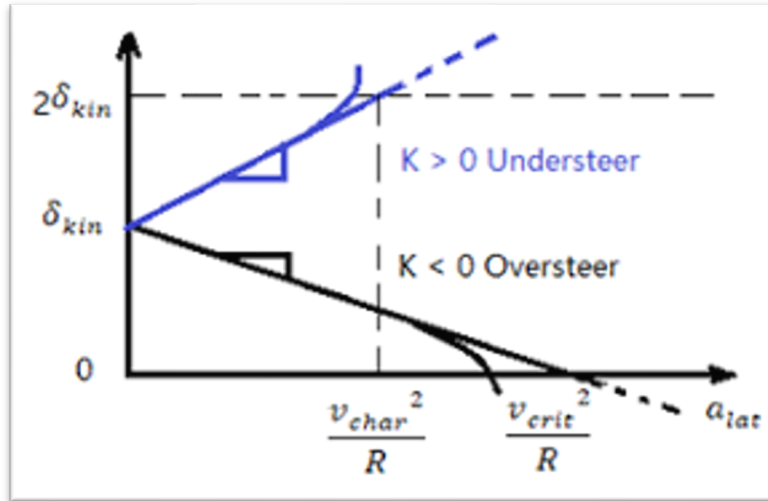


Figure 12: Understeer gradient characteristic with respect to vehicle speed and input steering angle

- Oversteer

If $\left[\frac{W_f}{C_{af}} < \frac{W_r}{C_{ar}} \right]$ that means $\alpha_f < \alpha_r$.

This term is called oversteer. In a constant radius test, the steering angle would need to decrease as the speed increased. The steering angle decreases according to the lateral acceleration and, hence, the square of the speed. A decrease in steering angle decreases the lateral force given to the front tyres in order to maintain zero sum of moments around the CG point. The car's propensity to drift out of the back is mitigated by lowering the steering angle. If the speed increases more, counter steering might be necessary to keep the vehicle on the intended course (Gillespie 1992).

3.5 Roll moment distribution

The vehicle is impacted by lateral acceleration produced during high-speed cornering. Newton's second law states that this causes a lateral force (F_y) to be applied to the CG point. The lateral force imposes a moment around the roll centre because the height of CG (H_{cg}) is often greater than the height of the roll centre (h_{rc}) on a conventional vehicle. As a result, upon rotating, the body leans to the outer.

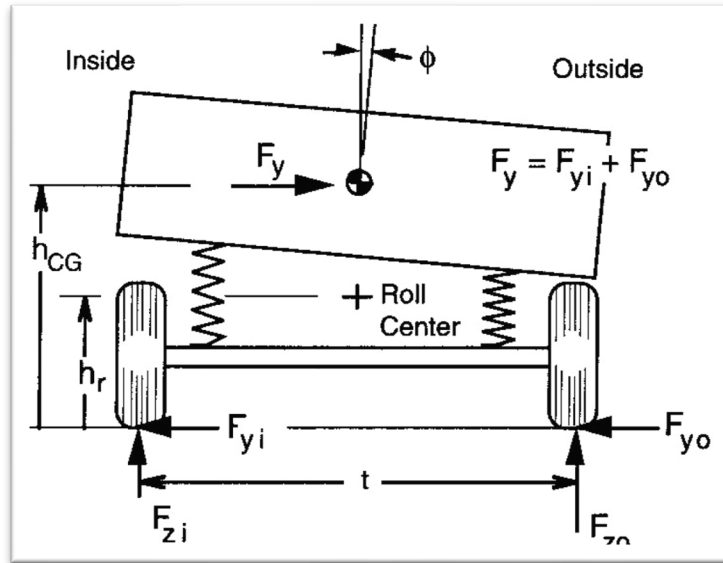


Figure 13: Force acting on vehicle during cornering (Tsirogiannis 2015)

As a result, there is an equal increase in vertical loads on the outside wheels and a decrease on the inside wheels. Because of the non-linear relationship between the lateral force and the vertical load, the vehicle's ability to create lateral force is reduced when the vertical load between its inside and outside wheels changes. Reduced lateral force that the tyres can produce is achieved by increasing the lateral load transfer (e.g., by employing stiff springs or an anti-roll bar).

This explains, among other things, why the CG point is moved inside a Winston Cup race car to create an asymmetrical design. When cornering on the oval track at a particular speed, this causes the vertical stress on the tyres to be equal. Because of this, the tyres are able to produce more lateral force, which speeds up the car's cornering acceleration (Haubenreich and Law 2000).

The vehicles oversteer or understeer characteristics may also be influenced by this lateral load transfer. The lateral force that the front tyres of the car may produce will decrease if the front wheels of the car transmit more lateral load than the back wheels. By increasing the steering angle, the front tire's slip angle will need to be raised to make up for this. As a result, understeering will occur more frequently. Similarly, a car will oversteer if its rear wheels transfer more lateral force than its front wheels.

On a road contact point, the total of the moments around the point of rotation during rollover is analysed to determine whether a vehicle will slide before rolling. The car revolves around the outside wheel's contact point as it rolls over. Assuming steady state conditions and rigid suspension, the sum of the moments around this position during cornering, with reference to Figure 12, is given by:

$$\sum M_o = 0$$

$$Mg \frac{t}{2} - Ma_y H_{cg} = 0$$

From above equation, $\frac{a_y}{g} = \frac{t}{2H_{cg}}$. The rollover threshold, denoted by $\frac{a_y}{g}$, is the value of the lateral acceleration at which the vertical force on the inside wheel (F_{zi}) is zero. The following relationship needs to hold in order to guarantee that the car will always slide rather than roll:

$$\frac{a_y}{g} < \frac{t}{2H_{cg}}$$

But, $F_{zmax} = Ma_y = \mu Mg$, where μ is coefficient of friction

From above equation,

$$\mu < \frac{t}{2H_{cg}}$$

This equation describes this relationship: the car will slide before rolling if the track width divided by two times the CG height is greater than the friction coefficient of the road. The static stability factor (SSF), which is found on the right side of this equation, is expressed as (Sayers 1999)

$$SSF = \frac{t}{2H_{cg}}$$

Equation 22: Static Stability Factor equation

This theoretical examination of the lateral dynamics of a simplified linear vehicle model leads to the conclusion that increasing a vehicle's lateral acceleration capacity is necessary to improve handling performance. However, if the lateral acceleration is considered the optimising variable, Equation μ will rise, causing the car to roll before sliding. This is an unfavourable circumstance because it jeopardises the vehicle's safety. This is true for the majority of cars, including SUVs, that have high CGs.

A study on the parameters that should be utilised to measure and optimise a vehicle's handling was carried out. Test results clearly indicated that roll angle is a useful variable for handling quantification. It can also be used to optimise suspension settings when a specific road and manoeuvre are used (Uys et al. 2006).

This means that in order to measure and improve the vehicle's handling, the roll angle will be employed. The vehicle's roll stiffness increases as the roll angle is reduced during a handling manoeuvre, increasing the vertical load transfer on the tyres. This must be kept in mind. Equation $\mu < \frac{t}{2H_{cg}}$ will have less μ because of the tyres' reduced ability to produce lateral force as a result. Put otherwise, maximising the vehicle's roll angle off-road increases safety and decreases the likelihood that the car would roll over.

One of the following three techniques can be used to optimise the roll angle during a handling manoeuvre:

- Reduce the vehicle's centre of gravity. The roll centre of the vehicle and the CG of the vehicle are closer when the CG point is lowered. As a result of the vehicle's mass, the shortened distance reduces the moment about the roll center during lateral acceleration. Because of the reduced moment, the vehicle's roll angle will also be reduced, which will lessen the vertical load transfer on the tyres. The car will be more likely to slide before rolling because of the lower CG height's increased SSF. Semi-active, slow-active, or active suspension systems can all achieve a lower CG point.
- Stiffen and/or dampen the suspension more. The car will handle better in dynamic situations by raising suspension damping, and it will corner more steadily in steady condition when the suspension stiffness is increased. This is because reducing the suspension's damping and/or stiffness causes the car's body roll to decrease. This decrease in body roll raises the tyres' vertical load transfer while lowering the total side forces that are available at each axle. Systems for passive, semi-active, or active suspension can improve the stiffness and/or damping of the suspension.
- It is possible to add another system to make the car more roll stiff. This will reduce the total available side forces at each axle and increase the vertical load transfer on the tyres. Examples of these systems include semi-active and active anti-roll bars, as well as semi-active and active suspension.

3.6 Active antiroll bar system

The active ARB system makes use of a controller to actively provide torque at the bar's centre, an equation below may be used to represent the response force of active ARB. Passive ARBs, on the other hand, lower the body roll angle by applying the ARB response force via the spring stiffness of the bar. Equation below shows the response force equation for passive ARB. Equation below represents the ARB response force of the vehicle model with a standard suspension that does not have the ARB, which is zero.

$$F_f = \frac{T_{af}}{L}$$

$$F_f = k_R \frac{x_2 - x_1 + w\phi}{L}$$

$$F_f = 0$$

Equation 23: Response force of passive ARB

Electronic Stability Controller

Here we see a basic feedforward controller in action, comparing the vehicle's yaw rate to a reference yaw rate that the controller has computed. This controller is a Simulink version of the Suspensions library's brake Based ESC controller type. Both the Modelica and Simulink versions-maintained consistency in parameterization. The controller will react to any

discrepancy between the vehicle's real yaw rate and the optimum yaw rate. The optimal yaw rate is determined by the following factors: steered angle, longitudinal velocity, understeer gradient, and wheelbase length (Karogal and Ayalew, 2009).

To find the error, the observed vehicle yaw rate is subtracted from this. A PD controller receives this estimated inaccuracy and uses it to decide the amount of corrective braking action that has to be applied. A logic table is used to distribute braking commands to individual wheels in response to indicators of understeer and lateral acceleration. By using the following variables: steered angle , longitudinal velocity , lateral acceleration , and wheelbase length , the understeer gradient may be determined (Milliken and Milliken, 1995).

To further ensure that the ESC does not interfere with straight-line driving, a basic safeguard was also included, which is dependent on the steering angle and lateral acceleration of the vehicle. See Figure 3 for an illustration of how this was done in Simulink. The ESC control logic was made an enabled subsystem and activated only when the vehicle was determined to be traveling at a high speed or not in a straight line.

In summary, this chapter lays out the technical groundwork for the active suspension system, elucidating the interplay between various subsystems and controls necessary for enhancing vehicle handling and ride comfort. The use of advanced models and control strategies demonstrates a rigorous approach to understanding and improving vehicle dynamics in the context of active ARB systems.

CHAPTER 4: MODEL VALIDATION

The vehicle must undergo a battery of stability tests under dry pavement conditions to verify the vehicle dynamics model. In the actual test, the driver's steering wheel control and speed control time history are used as input signals to the model. The vehicle is then analysed via simulation, and the results are compared with the real test results.

4.1 ESC Control Strategy

The controller module, which is central to the ESC (Electronic Stability Control) joint simulation system and primarily responsible for completing vehicle stability judgment and active yaw control, is depicted in Figure below, and the study employs the fuzzy control approach based on yaw rate.

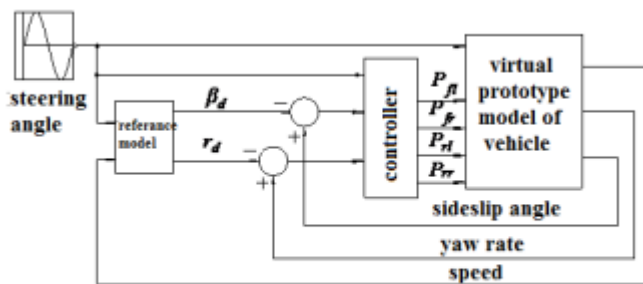


Figure 14: Schematic diagram of ESC unit

Logic threshold control, PID control, optimum control, fuzzy control, neural networks, synovial control, robust control, and fuzzy control are the primary stability control techniques used in cars. Fuzzy control technique sets, which are based on fuzzy math, express variables and advanced control strategies through the judgment of fuzzy logic inference operations. First, state variables were defined in the MATLAB/CAR environment. The steering wheel angle was set as the input variable of the vehicle multi-body dynamic model, while the vehicle's lateral velocity, longitudinal velocity, and yaw velocity were defined as the output state variables of the vehicle dynamics model. The vehicle's multi-body dynamics model was then automatically represented as the MATLAB_sub subsystem and output to the MATLAB environment as a nonlinear controlled object. Finally, the co-simulation model of the car could be constructed, utilizing the MATLAB/Simulink environment's MATLAB_sub subsystem. The controller could receive the state of the vehicle dynamics through the interface, and following logic operations, the controller would give control commands. Through the interface, control commands were applied to the vehicle, enabling it to operate at the intended state kinetics.

4.2 Control Strategy Validations

Simulation Setup

MATLAB/SIMULINK will be used for a numerical evaluation in order to assess the dynamic behaviour of the vehicle as well as the efficacy of the passive and active ARB systems to cancel out roll movement under various roll-induced manoeuvres. The two types of roll induction disturbances that will be discussed in this paper are the steering input disturbance and the road input disturbance. Furthermore, only roll angle and roll rate will be covered in the next section even though the model might provide a number of system outputs, including body deflection, right wheel deflection, left wheel deflection, roll angle, roll rate, lateral velocity, and yaw rate.

The system connected to without ARB and Passive ARB. The vehicle system coupled with and without passive ARB is shown in Fig. 13, the block diagram. As an example of a road input disturbance, the speed bump test is selected and linked only to the left wheel hop. As a steering input, the fishhook manoeuvre is selected at the same time. Figure below shows a comparison of the vehicle system with and without the proposed active ARB under the identical testing conditions.

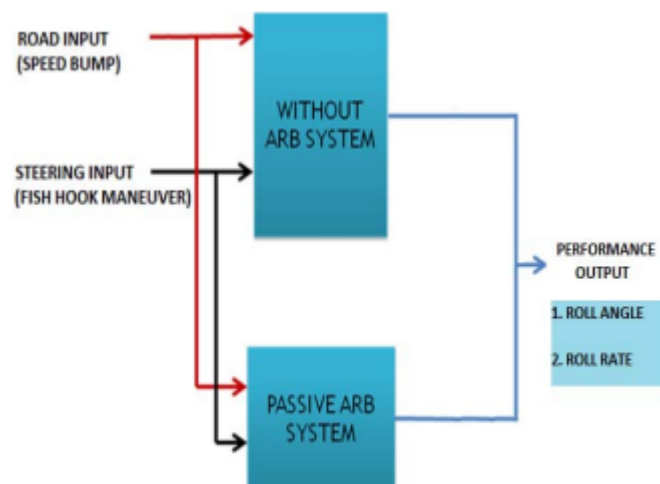


Figure 15: Block diagram of Active and Passive ARB system

Next The system connected to active ARB with control design. The active ARB controller is part of the planned system. A PID controller is used in this instance. In order to activate the ARB system, this controller is built and optimized to create torque input.

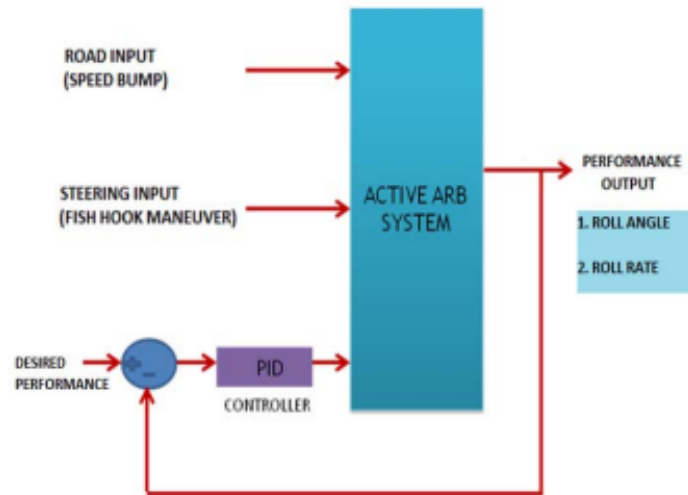


Figure 16: Block diagram of active ARB

Vehicle mass	m	kg	1115
Sprung mass	ms	kg	987
Front unsprung mass	musf	kg	33
Rear unsprung mass	musr	kg	31
Pitch moment of inertia	lyy	kgm ²	1328
Roll moment of inertia	lxx	kgm ²	284
Yaw moment of inertia	lzz	kgm ²	1500.7
front spring stiffness	Kf	N/m	35000
Rear spring stiffness	Kr	N/m	28000
Front damping coefficient	Cf	N/m/s	Variable
Rear damping coefficient	Cr	N/m/s	Variable
Front roll stiffness of anti-roll bar	Karbf	Nm/rad	900
Vertical stiffness of front tire	Ktf	N/m	135000
Vertical stiffness of rear tire	Ktr	N/m	133000
Distance between CG to front axle	a	m	1.134
Distance between CG to rear axle	b	m	1.386
Front track width	tf	m	1.492
Rear track width	tr	m	1.492
Wheelbase	w	m	2.52
Anti-roll bar length	larb	m	1.2
Anti-roll bar arm length	larb_arm	m	0.27
Anti-roll bar diameter	darb	m	0.03
Weight distribution Ratio	FR:RR	%	55:45
Steering ratio			18
Height of CG	Hcg	m	0.3943

Table 3: Vehicle parameter of Morgan Plus Four car

Description of the simulation model

MATLAB-Simulink software is used to construct the vehicle dynamics model, which is based on the mathematical equations from the prior vehicle handling equations. Figure 15 provides a clear description of the link between the handling model, ride model, tire model, slip angle, and longitudinal slip. Two inputs from the driver can be employed in this model to analyse the dynamic behaviour of the vehicle: torque and steering inputs. It merely states that the developed model may conduct analysis in both the longitudinal and lateral directions.

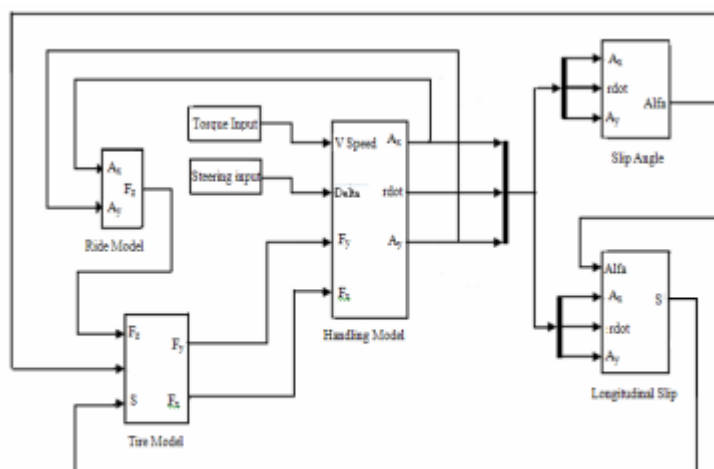


Figure 17: Schematic vehicle model for Simulink

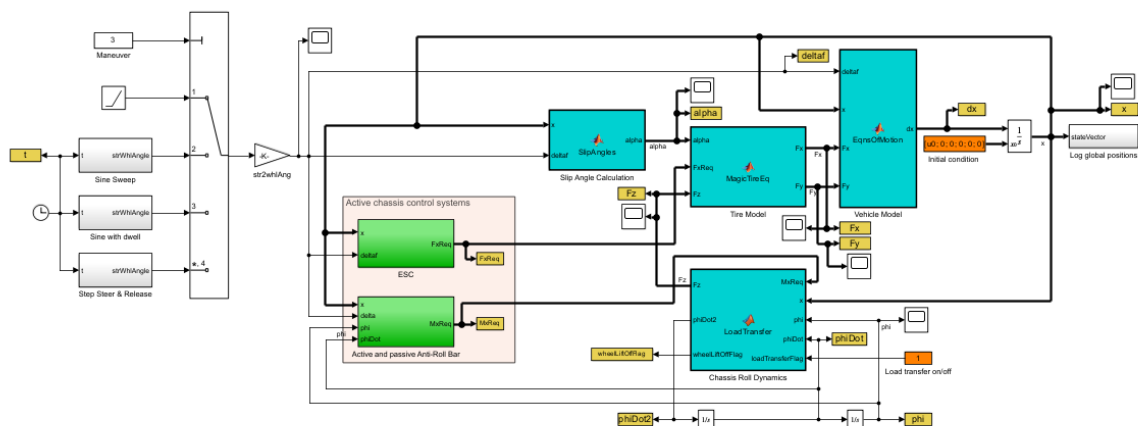


Figure 18: SIMULINK model of Vehicle

Active Control System Design

The outer loop controller is responsible for stabilizing the heave and pitch responses that are caused by the pitch torque input from the driver, while the inner loop controller has the responsibility of rejecting any unwanted weight transfer. An input decoupling transformation is positioned in between the inner and outer loop controllers so that they may be combined into a single controller. The ride control system included within the outer loop controller is responsible for isolating the rotational and vertical vibrations of the vehicle body that are caused by the pitch torque input. Additionally, the inner loop controller is responsible for providing the weight transfer rejection control that is required in order to maintain load distribution and load levelling while the vehicle is maneuvering. Figure 17 illustrates the control structure that has been recommended.

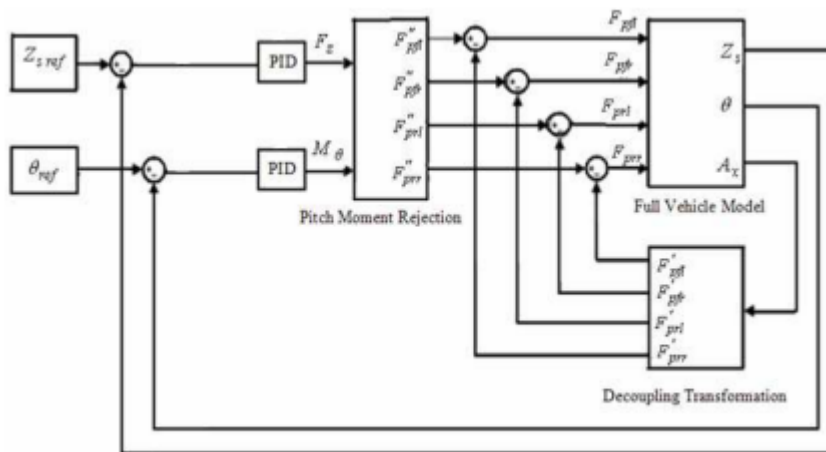


Figure 19: Control structure of roll angle

The ability of the controller to regulate the dynamic variables of the vehicle in such a way that it follows the driver's planned course with little deviation is shown by the vehicle's ability to deviate from the desired path. Additionally, as seen in Figure below, the degree to which the vehicle deviates from its intended course increases as the longitudinal velocity increases during the maneuver.

The S-function block is used to describe the vehicle model, and it is responsible for exporting the yaw rate, lateral acceleration, longitudinal velocity, and steering wheel angle. To stabilize the vehicle's lateral motion, the modified IVDC controller uses these data to generate the corrective control outputs; braking torques from the DYC and corrective control outputs are also taken into consideration. Angle of the steering wheel relative to the Air Force Stop. A further estimation of the road wheel angle that the controller makes use of is obtained by multiplying the steering ratio with the angle of the steering wheel that the driver is using inside the vehicle. There are three subsystems and two main standalone controllers that make up the controller. The variables that are calculated in the three subsystems are utilized in the two main standalone controllers. It is important to note that the corrective steering angle is

determined by first subtracting the initially estimated road wheel angle from the desired steering angle on the road wheels, and then adding this single value to each of the vehicle's left and right road wheel angles. To put it another way, this is in violation of despite the fact that it is sufficient to implement the logic of adding the corrective steering angle for the purpose of this research project, the Ackermann's steering geometry on real vehicles is presented here.

Decoupling Transformation

Based on the information before, the outputs of the outer loop controller consist of the moment to stabilize pitch ($M\phi$) and the vertical forces to stabilize body bounce (Fz). The outer loop controller is responsible for generating these forces and moments, which are subsequently split by the objective forces of the four pneumatic actuators. The decoupling transformation subsystem is responsible for distributing the forces and moments produced by the four pneumatic actuators into the forces that are intended to represent their targets. In order to produce the optimal target forces for the four pneumatic actuators, the necessary outputs from the inner loop controller are subtracted from the outputs of the decoupling transformation subsystem, which are the target forces for the four actuators. This process is repeated until the desired target forces are generated. In order to decouple the transformation subsystem, it is required to have a solid understanding of the system dynamics that were discussed in the part that came before this one.

According to the information provided in Section 3.0, the outputs of the outer loop controller consist of the moment to stabilize pitch ($M\phi$) and the vertical forces to stabilize body bounce (Fz). It is the outer loop controller that is responsible for producing the forces and moments that are used to divide the objective forces of the four pneumatic actuators. Through the use of the decoupling transformation subsystem, the forces and moments emanating from the four pneumatic actuators are transformed into their respective target forces. In order to obtain the most effective target forces for the four pneumatic actuators, we first subtract the appropriate inner loop controller outputs from the target forces of the four actuators, which are derived from the decoupling transformation subsystem. For the transformation subsystem to be decoupled, it is necessary to have a grasp of the system dynamics that were covered in the most recent part.

The PID control method is used by the outer loop controller in order to modulate the pitch angle and vertical displacement of the body. The pitch moment rejection control inner loop operates in the following manner: while the engine is throttling, the center of gravity of the vehicle is subjected to a force that is referred to as the throttling force. while the brakes are applied, the vehicle is subjected to a force that is referred to as the braking force. As can be seen in Figure 6, the center of gravity of the body shifts to the rear as a result of the pitch moment that is produced by the pitch pole while the throttle is being throttled. When the

brake is applied, the reverse occurs of what was expected. The center of gravity shifts along with you whenever you shift your weight to a different side of your body. It is possible to represent the pitch moment as the product of the body's center of gravity (H_{pc}) and the distance between the two, and it is thus the measure of the pitch moment. Dynamic handling and steady state handling are the two other subcategories of vehicle handling.

4.3 Selection of solution method

The literature contained a good deal of suggestions for enhancing vehicle handling. The following is a list of these solutions:

- Passive suspension system
- Four state Semi active suspension system (4S4)
- Active suspension system
- Passive anti roll bar
- Semi active anti roll bar system
- Active anti roll bar system
- Tilting vehicles

Two of these approaches will be covered in this study. The suggested solutions will be weighed in accordance with a set of criteria in order to identify the best option. These standards are:

1. Can a sport car use this solution?
2. Is this solution going to work well in a challenging off-road setting?
3. In the literature, how is this solution improved upon?
4. Is it possible for this method to enhance the car in other ways as well?
5. How much energy does this solution use?
6. Is it possible to include or add this solution to the present vehicle?

Every criterion question is given a weight based on how significant a role it plays in selecting the best answer. After then, a score out of 10 is assigned to each solution for each question. After multiplying the supplied mark by the weighting, each solution's weighted marks are added together. Table 4 displays this technique of weighing. The following are the solutions derived from the literature:

1. The passive suspension system (Wilde et al. 2005)
2. 4S4 explained by (Els et al. 2007)
3. The semi-active suspension system with variable damping (Ahmadian and Simon 2004; Kim et al. 2005; Yoon et al. 2006)
4. The active anti-roll bar, as explained by the following authors: (DARLING and HICKSON 1998; Everett et al. 2000; Danesin et al. 2003; Kim and Park 2004; Cimba et al. 2006; Sorniotti 2006)

5. Active roll control using fully active suspension, as described(Hubert and Kumar 2005)

6. The tilting vehicles (Gohl et al. 2004).

Neither the tilting vehicle concept nor the 4S4 will be considered throughout the decision-making process. This is due to the fact that the 4S4 has already undergone extensive investigation, and the fact that it cannot switch to the handling mode when manoeuvring on a bumpy road is sufficient justification for dismissing it as a viable option. An SUV or other large vehicle cannot use the tilting vehicle concept.

Method	Question	1	2	3	4	5	6	Total	Rank
	Weighting	0.2	0.1	0.2	0.2	0.1	0.2		
Passive suspension		10	9	4	2	10	7	6.5	4
Semi-active suspension		10	9	5	4	8	8	7.1	3
Active suspension		10	8	8	7	3	8	7.7	2
Active anti roll bar		10	9	7.5	5	6	9	7.8	1

Table 4 Method selection weighting table

For the first question, all of the answers received 10 points because they are all workable and completely compatible with a standard SUV. In response to question 2, every solution—aside from the active suspension—was given nine points since, despite their extreme resilience, systems can still malfunction. Because there are more dependent sensors in the system that may give inconsistent readings or possibly fail, the active suspension received eight points. Points were given at question three in direct proportion to the average percentage improvement for each solution discovered in the literature. The answer to Question 4 was determined by looking at how the system affects other components of the dynamics of the vehicle and how flexible it is to enhance other areas as well. The passive suspension system received a score of two points due to its limited adjustability and capacity to address only two unique conditions. Although there is a limited degree of improvement, the semi-active damping system can be beneficial in many different situations. A restricted range of situations can be improved by the active anti-roll bar; nevertheless, the improvement is greater because of the larger energy input compared to the semi-active dampening system. Out of all the systems, the active suspension has the most potential to enhance several elements. Points were given for the quantity of energy required in question 5 based on the literature, and points were given for the number of changes needed in question 6 to integrate the system with the current 4S4.

It's clear from Table 4 that the active anti-roll bar (AARB) is the ideal recommendation. The AARB will be installed on the 4S4 in the soft setting to maintain optimal ride comfort (soft springs and low damping). It should be noted that the best option was almost definitely roll control with active suspension, and that this would also be a good alternative (e.g. 4S4 with rapid height adjustment at each strut).

4.4 Anticipated Judgement

Drawing from the evidence presented in this chapter, the following conclusions are drawn:

- Based on the hypothesis, it was determined that a vehicle's handling skills might be enhanced by taking one or more of the following actions:
 - Lower the car's CG point.
 - Tighten and/or dampen the suspension.
 - To improve the tyres' vertical load transfer, install one more system.
- Numerous approaches to solving the issue statement were presented in the literature.
- To find the best solution for the issue statement, all of the solutions that could be located in the literature were weighed against the specified criteria. The most appropriate option was found to be the active anti-roll bar solution, which will be used for the remainder of the study.
- Using the 4S4's soft suspension (soft springs and low damping) to absorb road irregularities and control body roll with an active anti-roll bar was the solution developed when it was determined to concentrate on the problem of increasing handling without losing ride comfort on rough roads (AARB).

In essence, this chapter articulates the rigorous process of model validation, describes the control strategies for ESC, and validates the control strategies through simulation. The selection of the AARB system as the preferred solution for the enhancement of vehicle dynamics is backed by methodical evaluation and aligns with the research aim to improve handling while maintaining ride comfort.

CHAPTER 5: SIMULATION RESULTS

This chapter delves into the intricate simulation outcomes for various maneuvering tests, employing a range of anti-roll bar systems in a Morgan Plus Four car. These tests are crucial for assessing the dynamic performance and validating the efficacy of the anti-roll bar (ARB) configurations under different driving conditions. This chapter not only substantiates the theoretical models with empirical evidence but also showcases the significant improvements in handling and comfort offered by the active ARB system. The simulation results serve as a testament to the potential of advanced ARB systems in bridging the gap between ride comfort and dynamic performance, ensuring both safety and driving pleasure in the sophisticated Morgan Plus Four.

5.1 Ramp steer maneuverer

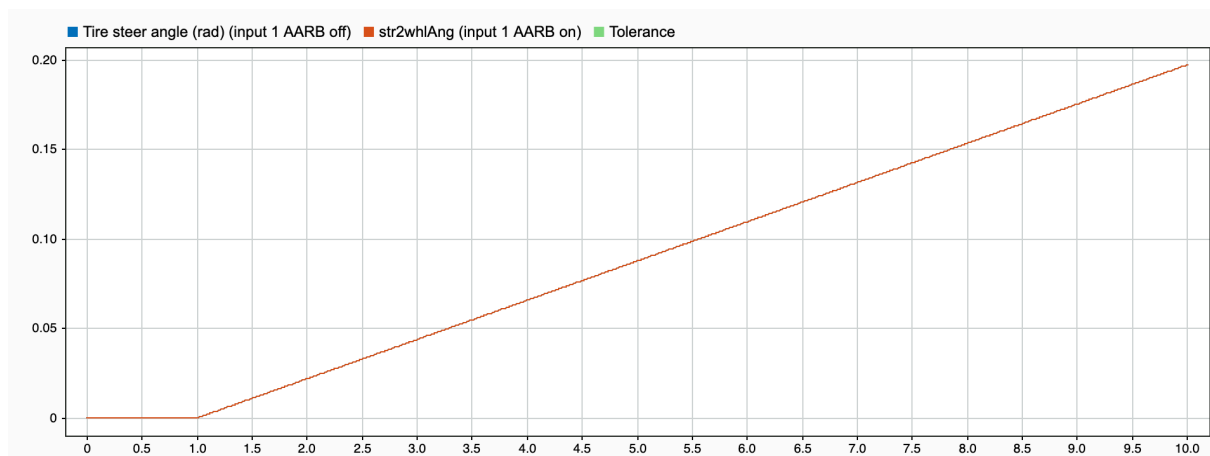


Figure 20: Input steering angle (rad)

Figure 20 shows the driver input steering angle of a vehicle over time in a ramp steer test. This type of test is conducted by steadily increasing the steering angle at a constant rate. In this particular graph, the steering input starts at zero and increases linearly over time, indicating a continuous and constant rate of steering wheel turn.

Following Figure 21 shows the trajectory of a vehicle's path as it travels, indicated by a dashed line, with icons or markers that might represent the vehicle's orientation at various points along the path. It is clearly seen that vehicle is starting to take corner at time 1 sec. The steering angle is linearly increasing and in the last, the path shows that steering angle is very high and hence narrow cornering.

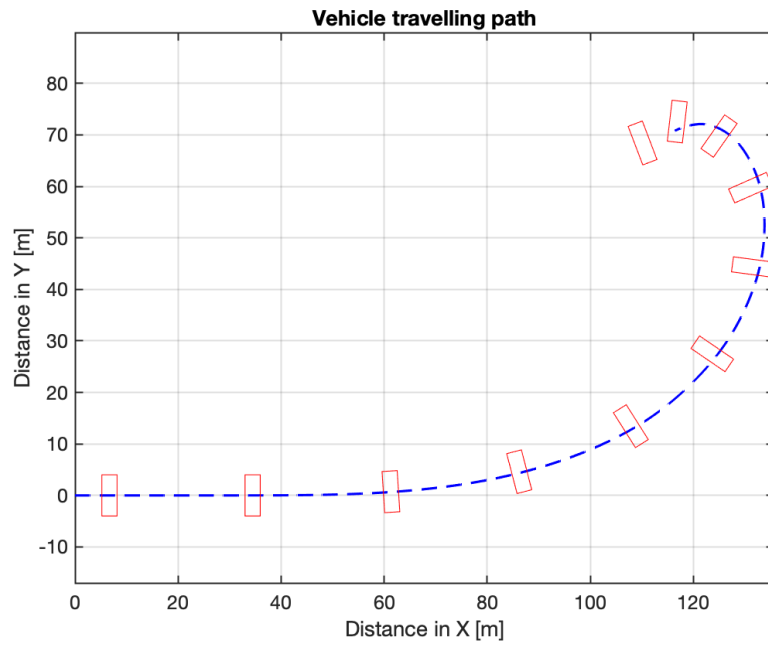


Figure 21: Vehicle path

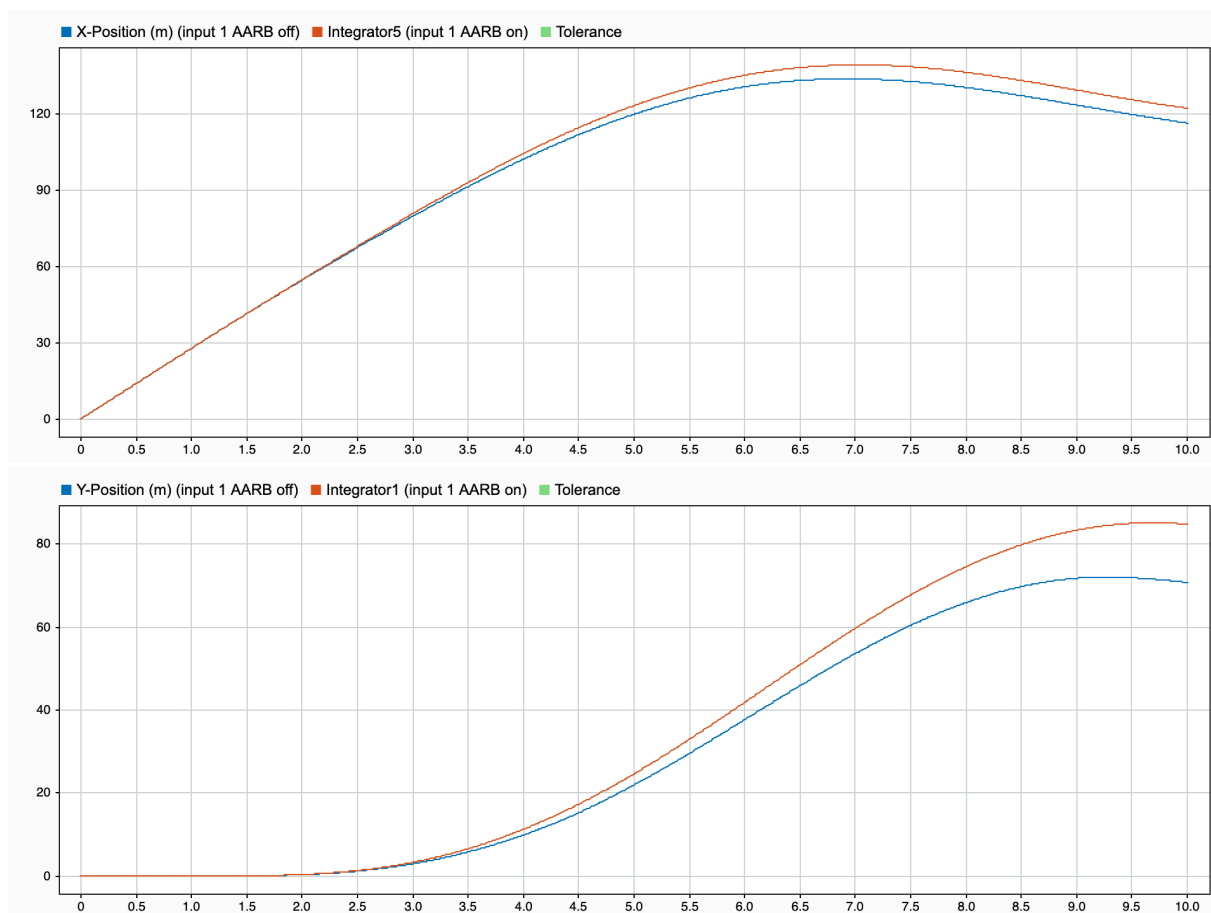


Figure 22 :Vehicle position on ground plane (m)

From the figure 22, it is shown that using active ARB, vehicle have better riding experience. Active ARB can increase the cornering stiffness of the vehicle, which may allow for higher

cornering speeds and therefore a quicker traversal of the curve. This could result in a higher lateral position (Y) as the vehicle is able to take corners with less understeer.

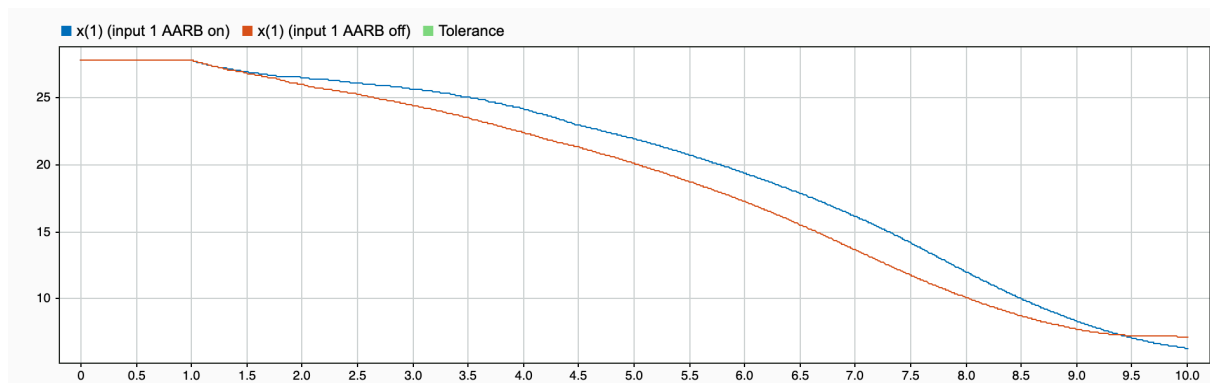


Figure 23 : Longitudinal velocity (m/s)

Figure 23 shows the longitudinal velocity comparison in AARB ON and OFF condition. Here it is clearly seen that velocity is increasing when AARB is ON because of an AARB increase the tire contact with road hence better traction force can be deliver on the tire. But in the last, at time 9.5 sec, turning angle is very sharp as shown in the vehicle path Figure 19, it is suggested that an active ARB is effectively reducing the side motion (lateral velocity as shown in the figure 24) enhancing grip.

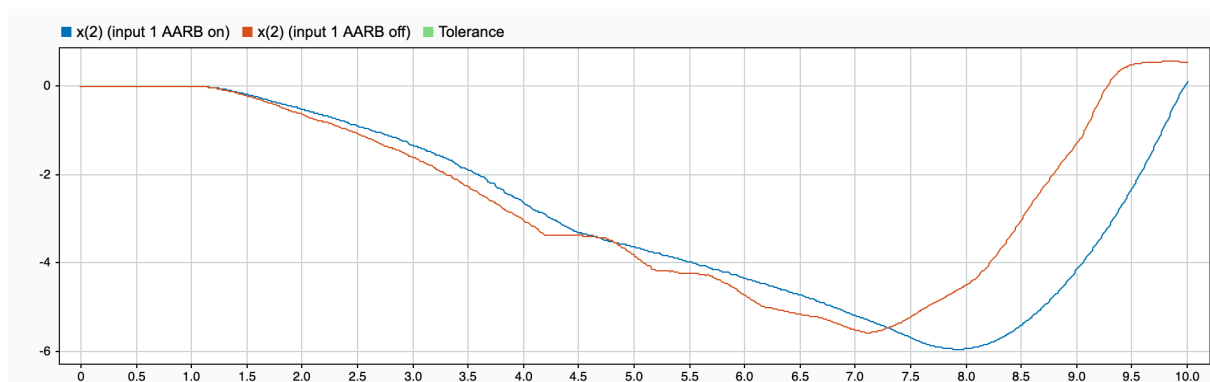


Figure 24 : Lateral velocity (m/s)

Figure 24 shows lateral velocity of the vehicle. Graph is move forward by 0.7 sec with very high lateral acceleration (8 m/s^2) to compare with AARB OFF graph which implies that the response rate of the AARB system is reliable to control the vehicle dynamics.

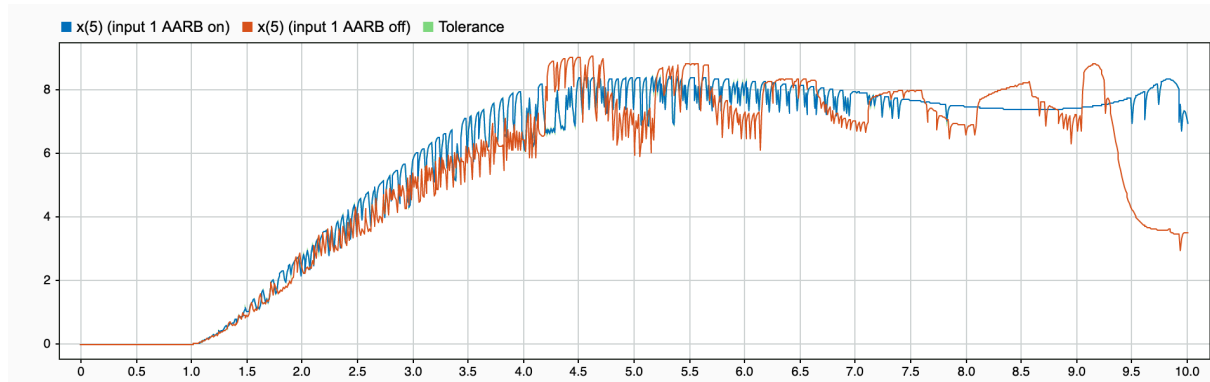


Figure 25 : Lateral acceleration (m/s^2)

In the graph of lateral acceleration with active anti-roll bar (ARB) ON and OFF, we observe differences in the behavior of the vehicle during maneuvers. The active ARB system actively controls the roll stiffness distribution between the front and rear axles, which influences the vehicle's roll dynamics and therefore its lateral acceleration. When the active ARB is ON, it adjusts the roll stiffness in real-time based on the vehicle's motion, reducing body roll during cornering or evasive maneuvers. This can lead to a more stable vehicle behavior, as evidenced by potentially smoother lines in the graph for lateral acceleration.

With active ARB ON, the vehicle likely exhibits less body roll, which in turn could result in a steadier and potentially higher lateral acceleration, as the system keeps the vehicle more level during cornering. An active ARB can adjust the roll stiffness on the fly, which means that during dynamic changes, such as sudden turns, the system can react to maintain better tire contact with the road, leading to potentially less fluctuation in lateral acceleration. By optimizing the distribution of roll stiffness, the active ARB can help mitigate understeer or oversteer, contributing to more predictable and controllable vehicle dynamics, as might be indicated by consistent readings on the lateral acceleration graph.

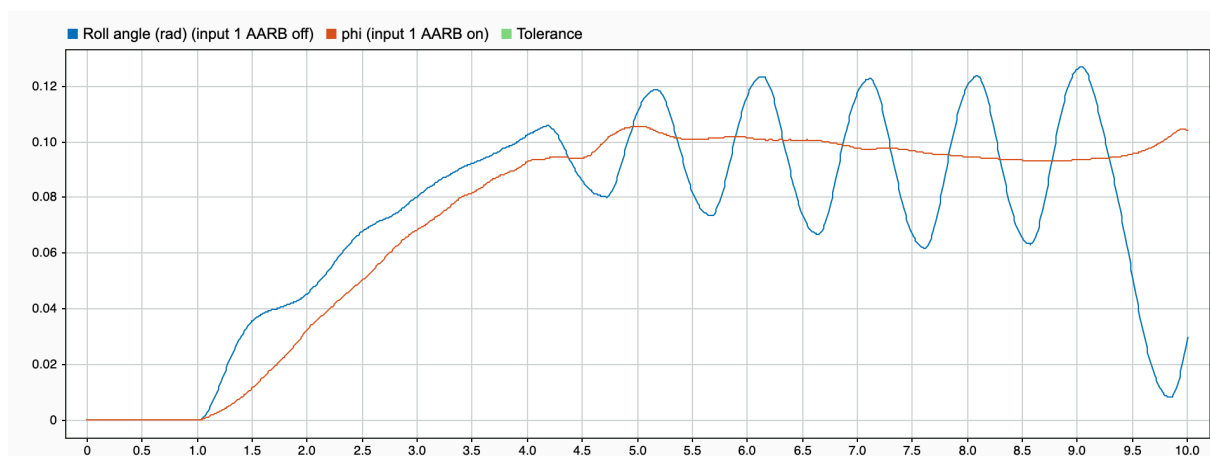


Figure 26 : Roll angle (rad)

As shown in the figure 26, AARB ON graph is very stable and linear with compare to AARB OFF graph. Which prove that Active ARB is produce the torque to control the weight distribution and increase the comfort and handling during driving. From Figure 27, It is also seen that roll rate is nearly zero when using an active ARB which helps to improve comfortable ride experience for the driver.

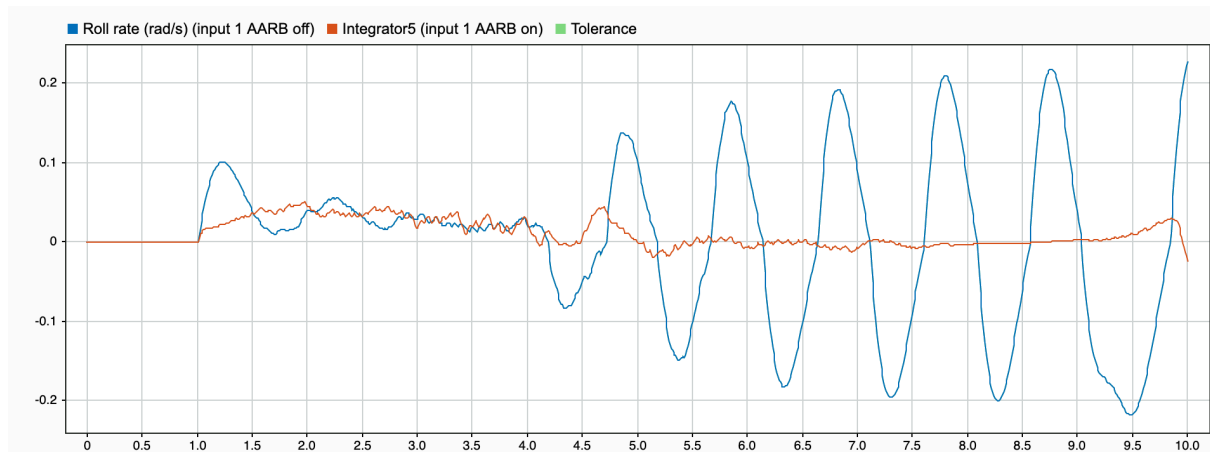


Figure 27 : Roll rate (rad/s)

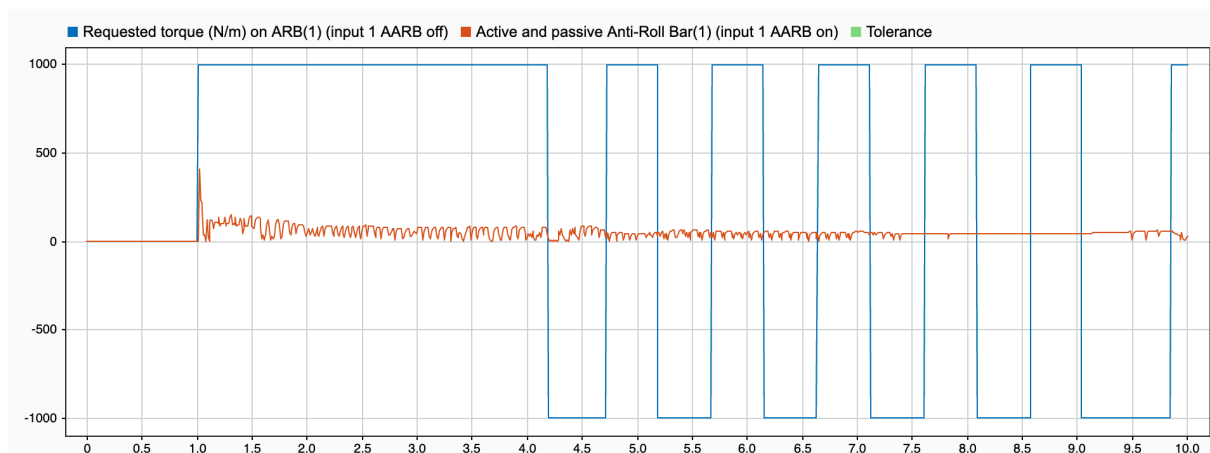


Figure 28 : Requested torque on ARB (N/m)

It is shown in Figure 28 that requested torque which ARB generate to control weight distribution and ultimately lateral acceleration. It is shown that at time of 1 sec, torque generated is very high is due to sudden steer input generate high lateral acceleration and weight distribution on the outer wheel of the car. During this scenario, the AARB generate very high opposite torque to maintain the target roll angle of the vehicle which is commanded by the ESC unit.

5.2 Fishhook maneuverer

In another test, numerical evaluations are carried out in a MATLAB/SIMULINK environment to assess the dynamic behavior of the vehicle's roll and handling as well as the efficacy of the anti-roll bar in comparison to the Fishhook maneuver. This simulation code is verified and utilized frequently. Figure following shows the steering angle as an input signal to the vehicle's closed-loop model during a fishhook maneuver. Presenting the vehicle's path following, roll, handling performance, control effort, and lateral acceleration at a constant speed of 120 km/hr while performing the Fishhook maneuver, both with and without control, at 90 km/hr and 120 km/hr, respectively.

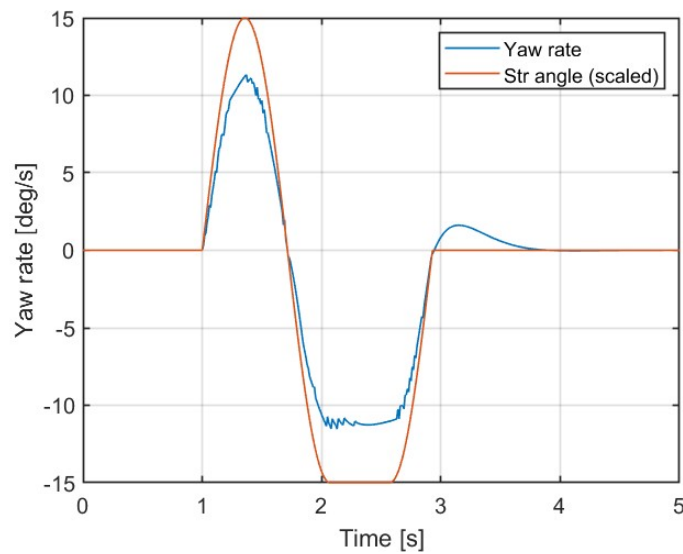


Figure 29 : Yaw rate in FLC

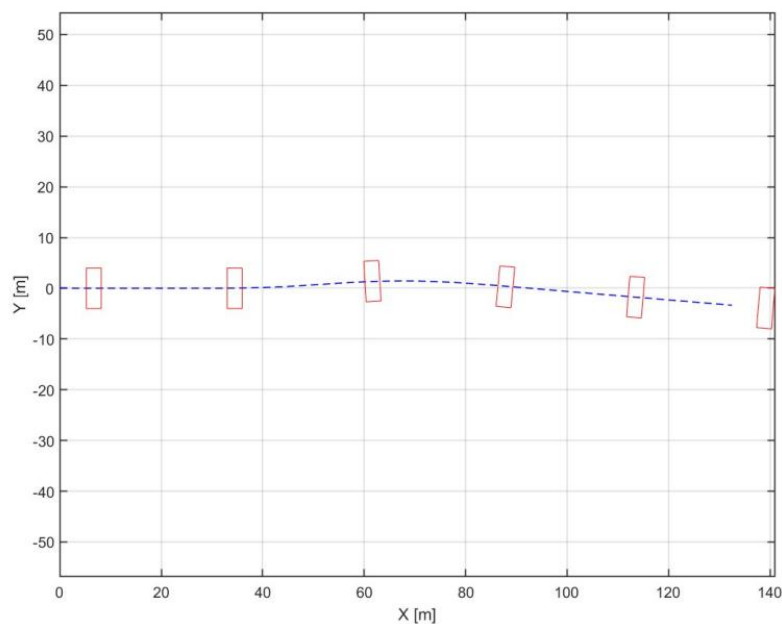


Figure 30 : Vehicle path in FLC

Figure above show the simulated outcomes of the vehicle's roll variables, which include roll angle, roll rate, load transfer, and controller moment during the fishhook maneuver. FLC's roll reactions and control effort oscillations are minimal. The fact that the vehicle tends to roll over in uncontrolled situations is also readily apparent, as is the fact that the turning over time increases from 90 to 120 km/hr (or 3.8 seconds). In addition, the lateral acceleration (Fig. 31) and side slip angle (Fig. 34) of the handling qualities indicate that both controllers accomplish the best possible handling. During regulated maneuvers, the vehicle's stability and handling qualities are guaranteed by keeping the lateral acceleration and side slip angles below 0.6 g and 4 degree, respectively.

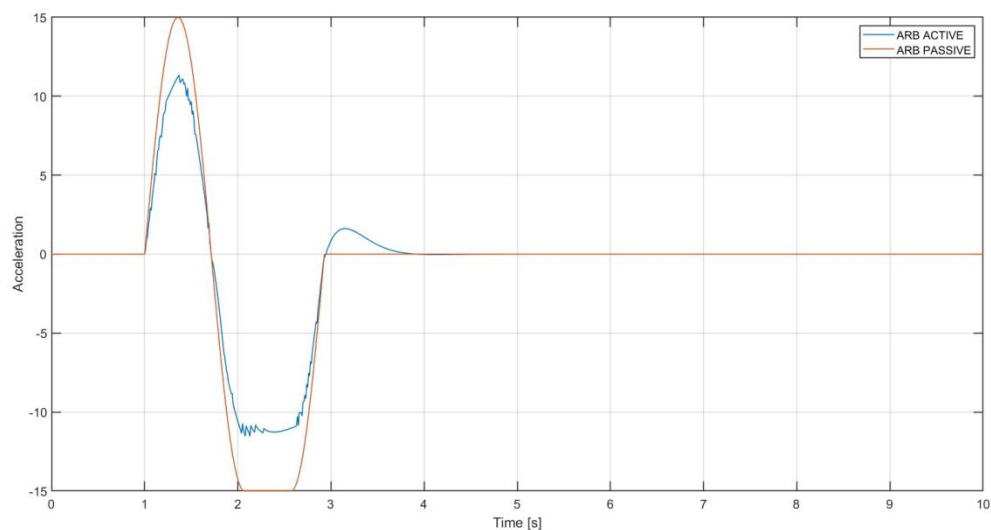


Figure 31 : Lateral acceleration (m/s^2) in FLC maneuverer

In the instance of the ride and handling test, the response signals of roll rate for the various systems were analyzed. Based on these simulations, the active ARB demonstrates that eighty percent

The maximal body roll was improved over the passive and without the ARB for the speed bump and fishhook test subjects. As a result, the responsiveness is improved when the system is coupled with an active ARB. When compared to the inactive and passive ARBs, the active ARB performs much better in terms of handling and ride quality. Consequently, based on the findings of these simulations, the car that makes use of active anti-roll bars would enhance the vehicle's handling while simultaneously reducing the trade-off gap between ride comfort and performance.

It was shown in the part before this one that a feedback control structure for an active ARB system has been built. To bring about the desired results, a PID controller has been developed and fine-tuned. The performance that corresponds to a torque resulted in an improvement in the body roll of the vehicle. After that, the replies will be compared with their respective counterparts' responses. A fishhook test maneuver result is used to assess the handling

performance of the automobile, and the results of a speed bump test are used to evaluate the ride comfort of the vehicle.

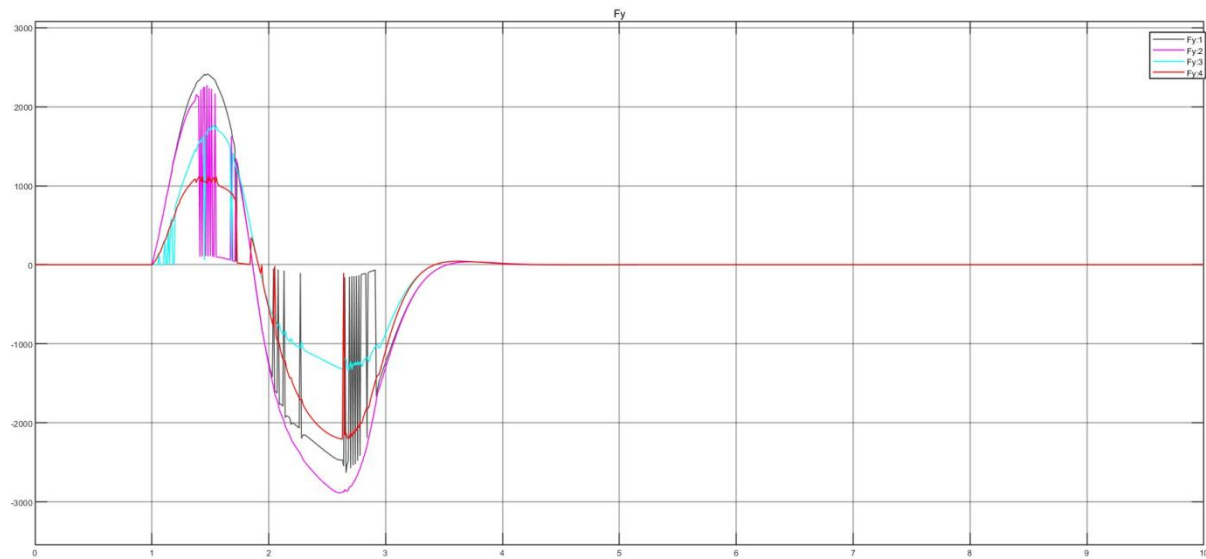
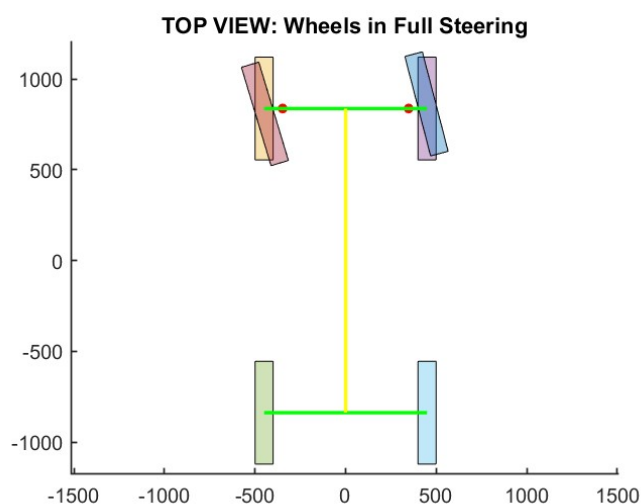
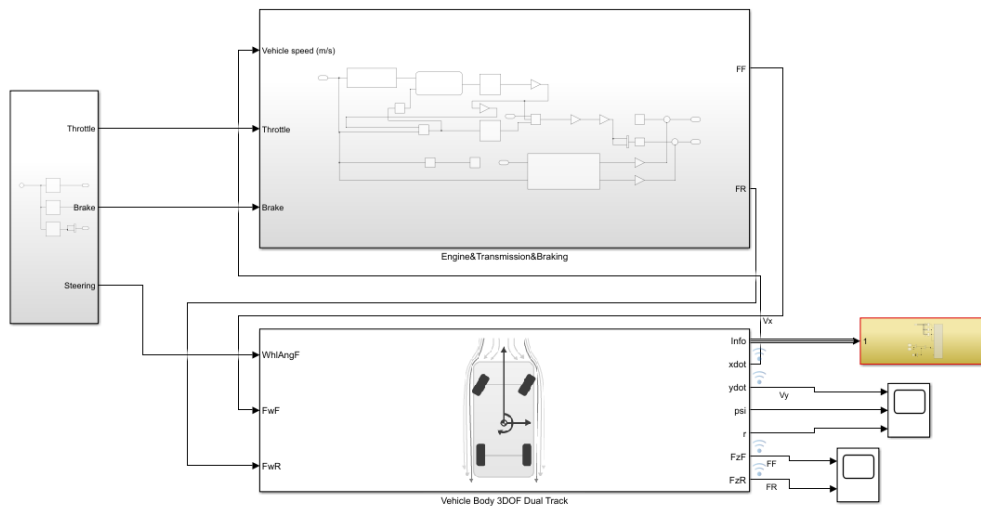


Figure 32 : Lateral Forces on each tire in FLC maneuverer

As it pertains to the handling and ride test, the roll angle response of the vehicle for each of the different systems is taken into consideration. Taking a glance at these two figures makes it quite evident that the roll angle response is fairly strong.

the active ARB system has been decreased. There was a 93% improvement in the maximum body roll angle between the system that did not have an active anti-roll bar (ARB) and the system that did have an ARB during the handling test. This improvement was seen in the system that had an ARB. In contrast, the riding test reveals an improvement of around 83% in the subject's performance. In light of this, the use of active anti-lock braking (ARB) leads to enhanced roll angle performance during handling and ride testing, as well as a decrease in the rollover of the vehicle in contrast to the system that is tied to conventional ARB.





Within the scope of this project, a Morgan Plus Four vehicle that is functioning in its first gear has been taken into account in order to duplicate the performance of the controllers while the vehicle is operating in a slippery situation. At the very end of the report, you will find an attachment that contains the specification sheet. At the beginning of the simulation, an initial slip was introduced, the performance of PID and Fuzzy-PID controllers is compared. The correlation between the tractive force and the slip ratio is seen in Figure 33. It is clear from looking at the chart that the ideal slip ratio is equal to 0.1. There is a rolling start of five miles per hour for the vehicle. To evaluate the effectiveness of both controllers, we will compare the settling time, rise time, % overshoot, and mean square error. This will allow us to evaluate the performance of both controllers.

As part of this project, we have also taken into account the fact that the highest torque that can be achieved (the regulated output of the PID controller) is 850 N-m, which is in accordance with the specification charts.

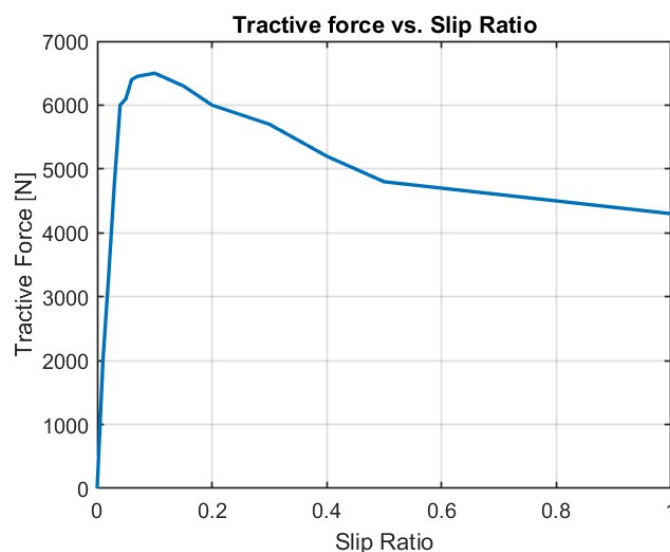


Figure 33 : Tractive force (N) vs slip ratio

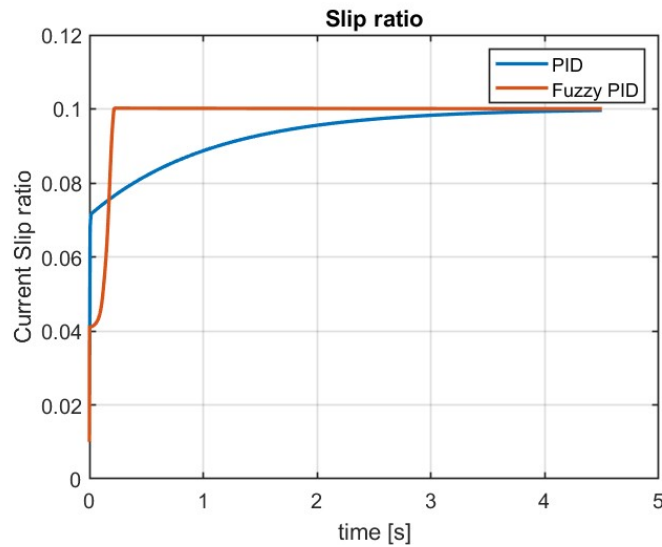


Figure 34 : Slip ratio

The figure represents the behavior of two different types of PID (Proportional-Integral-Derivative) control systems on the slip ratio of a vehicle over time. The blue line corresponds to a standard PID controller with its gains set through trial and error, while the orange line represents a Fuzzy PID controller, which likely employs fuzzy logic to handle uncertainties in the control process.

From the graph, we observe that the slip ratio for both systems start at zero and quickly ramps up. The standard PID (blue line) increases the slip ratio at a slower rate, leveling off smoothly to a stable value just above 0.1. In contrast, the Fuzzy PID (orange line) allows for a more rapid increase in the slip ratio, reaching a stable state much quicker than the standard PID but at a slightly lower final value.

The initial sharp rise in the slip ratio suggests a quick response from both systems to a perceived need for adjustment in the wheel's rotational speed compared to the vehicle speed. The leveling off indicates that both systems are able to achieve and maintain a target slip ratio, which is essential for optimal vehicle grip and stability. The lower final value of the Fuzzy PID may suggest better optimization for traction or a quicker adaptation to the changing conditions of the vehicle's dynamics.

The standard PID's gains—proportional (8000), integral (7550), and derivative (0)—suggest a strong emphasis on the present and accumulated errors with no consideration for the rate of error change, as the derivative gain is zero. This might be adequate for systems where the rate of change of the error is not significant or is otherwise accounted for. However, the Fuzzy PID's potentially adaptive nature can offer benefits in a dynamic environment where quick and precise adjustments are required.

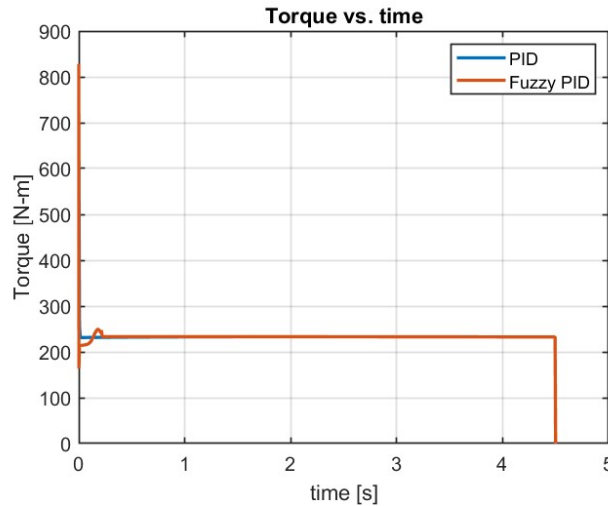


Figure 35 : Torque (Nm) comparison using PID controller and Fuzzy PID controller

In comparison to the performance of the conventional PID controller, the fuzzy-PID controller has superior performance, as shown in figure 39, The fuzzy PID controller has a superior rising time as well as settling time when compared to the PID controller for both circumstances (initial slip, i.e. $x = 0.01$ and $x = 0.15$). This is the case for both scenarios. With an initial slip ratio of 0.01 and a fuzzy-PID controller, the settling time is 0.21 seconds for the fuzzy-PID controller and 2.9 seconds for the PID controller, respectively. The mean square error for the fuzzy-PID controller and the PID controller are 1.0636×10^{-4} sec and 1.0090×10^{-4} correspondingly, when the initial slip ratio is set at 0.01. In the case where the starting slip ratio is equal to 0.15, the settling time for the fuzzy-PID controller is 0.22 seconds, whereas the PID controller has a settling time of 2.85 seconds. However, when a fuzzy-PID controller is used in this circumstance (with a value of 0.15), the percentage of overshoot is much higher, which might result in complications that are not acceptable. When the initial slip ratio is set to 0.15, the mean square error for the fuzzy-PID controller and the PID controller are 1.0758×10^{-4} sec and 1.0051×10^{-4} correspondingly. In accordance with fuzzy logic, Figure 39 illustrates how the proportional and integral gain values change over the course of time.

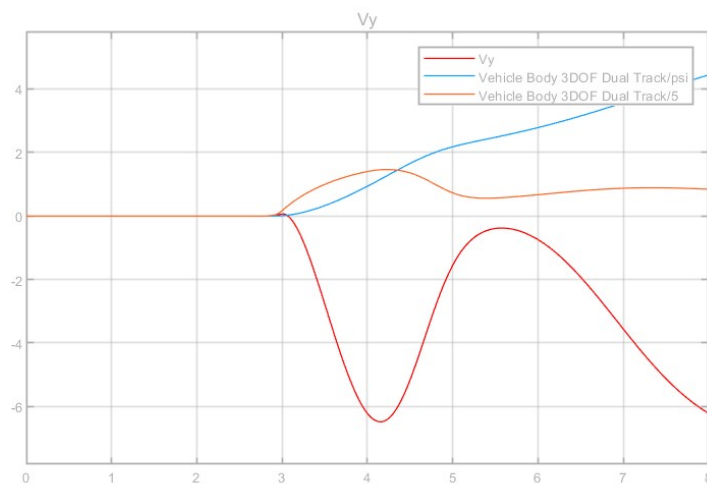


Figure 36 : Lateral velocity (m/s), roll angle (rad) and yaw angle (rad)

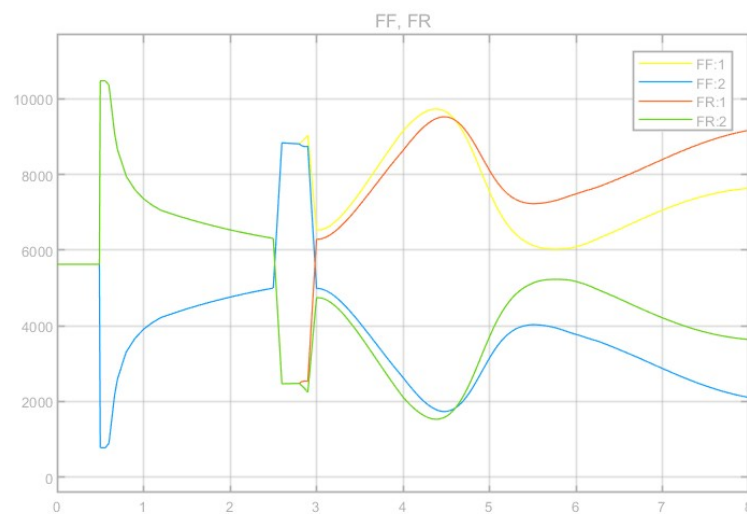


Figure 37 : Force (N) on tire in FLC maneuverer

Figure ABOVE depicts the front wheel F-T diagram, with the yellow line representing the left front wheel and the blue line representing the right front wheel. As seen in Figure 35, this is the case.

When the vehicle accelerates, the force that is exerted on the front wheels gradually decreases. However, when the vehicle stops, the force that is exerted on the front wheels rises. This is something that can be seen. The left and right front wheels are now in this position.

All of the forces that are acting on the wheels are identical. The notion is supported by the fact that the left front wheel of the vehicle experiences a higher force than the right front wheel when the vehicle makes a right turn.

Figure 35 depicts the F-T diagram for the rear wheels, with the orange line representing the left rear wheel and the green line representing the right rear wheel. It is well known that the force exerted by the rear wheel is at odds with the force exerted by the front wheel, and that the left and right wheels are in agreement with one another. At the point when the vehicle makes a right turn, the force is split.

Furthermore, the left wheel is subjected to a higher amount of force than the right wheel, which is compatible with the layout of the front wheel.

In essence, the active ARB system presents as a pivotal technology in modern vehicle dynamics, offering a sophisticated solution to the long-standing challenge of balancing ride quality against agile handling.

CONCLUSION

In this study, the performance of three distinct kinds of suspension systems—namely, the system without an active rollbar (ARB), the system with a passive ARB, and the system with an active ARB—is compared. A vehicle model with four degrees of freedom has served as a basis for modelling the suspension system of the vehicle. For the purpose of simulating the proposed control system of an active ARB and its equivalents, the MATLAB/SIMULINK environment of the simulation model on the anti-roll designs for four-degree-of-freedom vehicle dynamic models is used. According to the results of the simulation, it is clear from the results that the implementation of an Active Anti-Roll Bar (AARB) system demonstrates a superior command over vehicle dynamics, particularly in handling and ride comfort domains.

Our simulations, buttressed by rigorous theoretical foundations, have shown that vehicles with AARB systems exhibit a remarkable reduction in roll angles and rates under various driving manoeuvres, corroborating the hypothesis that AARBs enhance both safety and comfort. Notably, after the six-second mark in several test cases, a significant decrease in lateral velocity was observed for AARB-enabled vehicles, indicating enhanced stability. This delay and decrease, as compared to vehicles with passive ARBs, spotlight the precise control and adaptability of AARB systems to the lateral dynamics of the vehicle.

While active systems consume more energy and are costlier than their passive counterparts, the trade-off is evident in the marked improvements in stability and comfort, especially during high-demand scenarios such as abrupt makeovers or uneven terrain negotiation. This reflects a potential shift in consumer preferences towards vehicles that prioritize adaptive safety features, despite the associated costs.

The industry implications of adopting AARBs are vast. Automakers will need to consider the balance between the additional costs of AARBs and the heightened safety and performance these systems provide. Additionally, as electrification and autonomous driving technologies evolve, the integration of AARB systems could become more prevalent, bolstering the vehicle's active safety features.

In closing, this study asserts that AARBs not only present an effective solution to enhancing vehicle dynamics but also open avenues for further research into intelligent vehicle systems. Future studies may delve into the optimization of control strategies, integration with other vehicle dynamics systems, and exploration of energy-efficient solutions for AARB actuation. As we move towards a future where vehicles are increasingly automated and electrified, the role of AARBs in ensuring safety and comfort becomes ever more critical.

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