



**T.C.
BURSA TECHNICAL UNIVERSITY
GRADUATE EDUCATION INSTITUTE**

**TORQUE VECTORING AND REAR WHEEL STEERING CONTROL TO
IMPROVE HANDLING DYNAMICS OF AN ELECTRIC VEHICLE**

MASTER'S THESIS

Berkay ÇELİK

Department of Mechanical Engineering

JULY 2023

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JULY 2023

STATEMENT OF PLAGIARISM

I declare that all the information and results presented in visual, auditory, and written form in this thesis were obtained by me in accordance with academic and ethical rules, I have documented all the results and information in the thesis that are not specific to this study, by citing the source in the thesis, and I accept all kinds of legal consequences if the opposite arises.

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Signature:

FOREWORD

This thesis was written for my master's degree at Bursa Technical University, specialising in vehicle dynamics and chassis control. This is a fascinating research topic as it investigates combining two well-known chassis lateral dynamics control systems.

Firstly, I would like to express my most profound appreciation to my advisor, Dr Erdem Uzunsoy, for allowing me to work with him. I appreciate him for his valuable advice on vehicle dynamics, insightful comments, and everlasting help.

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Berkay Çelik
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ABBREVIATIONS

ABC	: Active Body Control
ACC	: Adaptive Cruise Control
AEB	: Automatic Emergency Braking
ARC	: Active Roll Control
ASS	: Active Suspension System
AWD	: All Wheel Drive
AWS	: All Wheel Steering
CoG	: Centre of Gravity
DOF	: Degree of Freedom
DoE	: Design of Experiment
ECU	: Electronic Control Units
EDU	: Electronic Drive Units
ESP	: Electronic Stability Program
EPAS	: Electric Power-Assisted Steering
EV	: Electric Vehicles
FCW	: Forward Collision Warning
FWD	: Front Wheel Drive
ICC	: Integrated Chassis Controller
ICLC	: Integrated Chassis Lateral Controller
KPI	: Key Performance Indicator
MBD	: Multi-Body Dynamics
MIL	: Model in the Loop
RAS	: Rear Axle Steering
RWD	: Rear Wheel Drive
RWS	: Rear Wheel Steering
TV	: Torque Vectoring
TVC	: Torque Vectoring Control
4WS	: Four Wheel Steering

SYMBOLS

v_x, v_y, v_z	: Speeds
a_x, a_y, a_z	: Accelerations
r_x, r_y, r_z	: Angular speeds
F_x, F_y, F_z	: Force Components
T_x, T_y, T_z	: Torque Components
M_x, M_y, M_z	: Moment Components
I_{xx}, I_{yy}, I_{zz}	: Inertia Components
t	: Time
g	: Gravity
μ	: Friction Coefficient
m	: Mass
C_f, C_r	: Tyre Cornering Stiffness
w	: Tyre Angular Velocities
α_r, α_r	: Tyre Slip Angles
κ_f, κ_r	: Tyre Slip Ratios
β	: Vehicle Side Slip Angle
a, b, l	: Wheelbase Distances
d	: Wheel Track Distance
h	: CoG Height
A_0, B_0, C_0	: Aero Resistance Coefficients
τ	: Ratios
η	: Factors
a_i, b_i, c_i, d_i	: Coefficients
K_p, K_i, K_d	: Gains

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ELEKTRİKLİ BİR ARACIN YANAL DİNAMİĞİNİ İYİLEŞTİRMEK İÇİN TORK DAĞITIMI VE ARKA AKS YÖNLENDİRME KONTROLÜ

ÖZET

Bu tez, arka aks yönlendirme (RWS) ve tork aktarımı kontrolü (TVC) sistemlerinin birlikte kullanılması yoluyla elektrikli araçların yol tutuşu ve stabilite iyileştirme potansiyelini araştırdı. Bu çalışmada birleşik lastik kayma etkilerine sahip doğrusal olmayan bir araç modeli kullanılmıştır. RWS ve TVC sistemleri, önerilen optimizasyon metodolojilerini kullanarak farklı sürüş modları için tasarlandı ve optimize edildi. RWS ve TVC sistemlerinin kontrolünü koordine etmek için bir entegre şasi yanal denetleyici (ICLC) sistemi geliştirildi. Önerilen sistemler, açık ve kapalı devre test senaryoları aracılığıyla araç dinamiği ve kararlılığı açısından değerlendirildi. Gelişmiş araç kontrol sistemlerinin çevrimdışı optimizasyonu için çıkarımlar yapıldı. Elektrikli araçlarda gelişmiş yol tutuş performansı ve geliştirilmiş sürüş deneyimi araştırıldı.

İlk olarak, RWS ve TVC sistemleri, özellikleri ve sınırlamaları dikkate alınarak farklı sürüş modları için tasarlandı ve optimize edildi. Elektrikli araçların yol tutuş performansını artırmayı amaçlayan bu sistemler için optimum parametrelerin belirlenmesi için önerilen optimizasyon metodolojileri uygulandı. Daha sonra, optimum RWS ve TVC sistemlerinin kontrolünü koordine etmek için bir ICLC sistemi geliştirildi ve bu sistemlerin daha iyi araç yanal dinamiği ve yol tutuş performansı sağlaması için sorunsuz entegrasyonu ve koordineli kontrolü sağlandı. Önerilen sistemler, yol tutuşu ve stabiliteyi artırmadaki etkinliklerini değerlendirmek için çeşitli test senaryoları aracılığıyla değerlendirildi. Sonuçlar, elektrikli bir araçta RWS ve TVC sistemlerini kullanmanın aracın yanal dinamiklerini önemli ölçüde iyileştirebileceğini, yol tutuş performansını ve stabilitesini artırabileceğini gösterdi. Gelişmiş araç kontrol sistemlerinin çevrimdışı optimizasyonu için pratik çıkarımlar yapıldı. Elektrikli araçların yol tutuş performansını artırmak için RWS, TVC ve entegre şasi sistemlerinin, nasıl etkili bir şekilde optimize edilebileceğine dair çıkarımlar yapıldı. Şasi sistemlerinin birlikte kontrolü için daha akıllı kontrol yöntemleri ve daha merkezi bir yaklaşımın gerekliliği önerildi. Hem RWS hem de TVC sistemleri için enerji verimli modları değerlendirmek ve kritik güvenlik hususlarını doğrulamak için gerçek araç testleri ile daha fazla araştırma yapılması önerildi. Ek olarak, gelecekteki araştırmalarda RWS ve TVC sistemlerinin merkezi olmayan kontrolünün potansiyel zorluklarının belirtilmesi gerektiği vurgulandı.

Sonuç olarak, bu tezin bulguları, elektrikli araçlarda yanal dinamiği iyileştirmek, elektrikli araç teknolojisinin ilerlemesine katkıda bulunmak ve araç dinamiği ve kontrolünde gelecekteki araştırmalar için bir temel sağlamak için RWS ve TVC sistemlerinin potansiyel birlikte kullanımına ilişkin fikir vermektedir.

Anahtar kelimeler: Araç dinamiği, Şasi kontrolü, Araç hareket kontrolü, Tork aktarımı, Arka aks yönlendirme, Optimizasyon.

TORQUE VECTORING AND REAR WHEEL STEERING CONTROL TO IMPROVE HANDLING DYNAMICS OF AN ELECTRIC VEHICLE

SUMMARY

This thesis investigated the potential for improving the handling and stability of electric vehicles through the combined use of rear-wheel steering (RWS) and torque vectoring control (TVC) systems. A non-linear vehicle model with combined tyre slip effects was used in this study. RWS and TVC systems were designed and optimized for different drive modes using proposed optimization methodologies. An Integrated Chassis Lateral Controller (ICLC) system was developed to coordinate the control of RWS and TVC systems. The proposed systems were evaluated regarding vehicle handling and stability through open and closed-loop test scenarios. Practical implications for offline optimization of advanced vehicle control systems were highlighted, paving the way for enhanced handling performance and improved driving experience in electric vehicles.

Firstly, considering their characteristics and limitations, RWS and TVC systems were designed and optimized for different drive modes. Proposed optimization methodologies were applied to determine the optimal parameters for these systems, aiming to enhance the handling performance of electric vehicles. An ICLC system was then developed to coordinate the control of optimum RWS and TVC systems, enabling seamless integration and coordinated control of these systems for improved lateral dynamics and handling performance. The proposed systems were evaluated through open and closed-loop test scenarios to assess their effectiveness in enhancing vehicle handling and stability. The results showed that using RWS and TVC systems in an electric vehicle can significantly improve its lateral dynamics, enhancing handling performance and stability. The findings of this thesis have practical implications for the offline optimization of advanced vehicle control systems. The optimized RWS and TVC systems, along with the integrated ICLC system, provide insights into how these systems can be effectively optimized to enhance the handling performance of electric vehicles. Further research is suggested to explore more intelligent control methods in a more centralized approach, evaluate energy-efficient modes for both RWS and TVC systems, and conduct actual vehicle tests to verify critical safety points. Additionally, the potential conflicts and challenges of decentralized control of RWS and TVC systems should be addressed in future research.

In conclusion, the findings of this thesis provide insights into the potential combined usage of RWS and TVC systems for improving lateral dynamics in electric vehicles, contributing to the advancement of electric vehicle technology, and providing a foundation for future research in vehicle dynamics and control.

Keywords: Vehicle dynamics, Chassis control, Vehicle motion control, Torque vectoring, Rear wheel steering, Optimization.

1. INTRODUCTION

Electric vehicles (EVs) are becoming more popular due to their environmental benefits, lower operating costs, improved performance, and driver feeling compared to traditional gasoline-powered vehicles. This has led to the need to re-design and develop systems such as power transmission, vehicle dynamics, energy management, advanced driver support systems, vehicle electronics architecture and safety to take advantage of all the features offered by electric motors and vehicle control units. Vehicle dynamics is a significant design fact in academia and the automotive industry from the perspective of electric vehicles. The systems used for vehicle dynamics improvements in the literature have been examined from the perspective that electric vehicle development brought innovations in longitudinal, vertical, and lateral dynamics. In general, different chassis control systems are used in electric vehicles to achieve better stability, handling, and comfort, especially in challenging driving conditions [2]. Some of the most-known chassis control systems are listed and explained below with the titles of longitudinal, vertical, and lateral dynamics.

1.1 Chassis Control Systems – Longitudinal Dynamics

Some of the chassis control systems related to longitudinal dynamics were listed and summarised below [3-5]:

- Adaptive Cruise Control (ACC): This system helps to maintain a safe following distance from the vehicle in front by automatically adjusting the vehicle's speed.
- Automatic Emergency Braking (AEB): This system helps to reduce the severity of collisions or avoid them altogether by automatically applying the brakes in emergency situations.
- Forward Collision Warning (FCW): This system alerts the driver when the vehicle approaches another vehicle too quickly but does not make any corrective inputs.

1.2 Chassis Control Systems – Vertical Dynamics

Below [6-8], a compilation and a brief overview of specific chassis control systems that pertain to vertical dynamics were expressed:

- Active Suspension System (ASS): This system helps to improve vehicle handling and ride comfort by automatically adjusting the suspension damping to suit driving conditions.
- Active Body Control (ABC): This system helps to improve vehicle stability and handling by automatically adjusting the suspension damping and ride height as needed.
- Active Roll Control (ARC): This system helps to improve vehicle stability and handling by automatically adjusting the suspension damping to reduce body roll during cornering.

1.3 Chassis Control Systems – Lateral Dynamics

Below [9], summary of select chassis control systems that are associated with lateral dynamics was expressed:

- Rear-Wheel Steering (RWS): This system helps to improve vehicle stability and manoeuvrability by automatically adjusting the angle of the rear wheels as needed.
- Torque Vectoring Control (TVC): This system helps to improve vehicle stability and cornering performance by automatically applying the torque or brake to individual wheels or adjusting the torque distribution to the wheels as needed.
- Electronic Stability Program (ESP): This system helps to improve vehicle stability and control by automatically applying the brakes to individual wheels or reducing engine power as needed.
- Electric Power-Assisted Steering (EPAS): This system is a type of steering system found in vehicles that uses an electric motor to help the driver in turning the wheels.

Both ESP and TVC are designed to improve vehicle stability and control. However, they work uniquely. ESC can help to prevent the vehicle from skidding or losing control in situations where the driver has lost traction or stability, such as when driving on slippery roads or during sudden manoeuvres. On the other hand, TVC is a continuously active system.

1.4 Integrated Chassis Control (ICC)

As seen above, many active chassis control systems serve different aspects of vehicle dynamics. As the demand for EVs increases, there is a corresponding need for advanced control systems to optimise the performance of these other chassis control systems. One such system is the integrated chassis control system, which plays a crucial role in an electric vehicle's overall control and stability. The integrated chassis control system combines various subsystems, such as suspension, steering, and braking. It uses sensors and control algorithms to monitor and adjust these systems in real-time continuously. This enables the vehicle to achieve better stability, handling, and comfort, especially in challenging driving conditions. Due to the increasing number of chassis actuators of modern human-driven and automated cars, Integrated Chassis Control (ICC) is one of the most appealing subjects for vehicle dynamics specialists and researchers. The ICC system optimises the performance of the available actuators by addressing their limitations and resolving conflicts and redundancies. To achieve optimal energy consumption, range, and overall performance, it is important for the ICC to be seamlessly integrated with the overall control architecture of the electric vehicle. This integration ensures that all subsystems are working together effectively and efficiently. In this study, the term ICLC (Integrated Chassis Lateral Controller) assumes the lateral dynamics part of the vehicle motion control system which includes RWS and TVC systems.

1.5 Research Aim

The primary purpose of this thesis is to investigate the potential for improving lateral dynamics in electric vehicles through the combined use of model-based RWS and TVC systems with feedforward and feedback that are used in industry. This study aimed to examine the effects of using these two systems together in an electric vehicle to enhance its handling performance. In addition to that, other purposes were stated below:

- To design a RWS system with feedforward and feedback in an electric vehicle with different drive modes.
- To optimise the RWS system for different drive modes and propose an optimization methodology for optimising RWS.
- To design a TVC system with feedforward and feedback in an electric vehicle with different drive modes.
- To optimise the TVC system for different drive modes and propose an optimisation methodology for optimising TVC.
- To design vehicle motion control with RWS and TVC systems and obtain an integrated chassis lateral controller (ICLC) system.

1.6 Organization

This study firstly introduced a literature review of vehicle dynamics models. Then, a detailed review of the RWS, TVC and ICLC systems related to lateral dynamics was explained. In the vehicle modelling chapter, non-linear vehicle and tyre models are considered and a co-simulation environment for chassis control systems was introduced. In the control system modelling chapter, RWS, TVC and ICLC were explained in detail. In the optimization of chassis control systems chapter, different drive modes with RWS and TVC are introduced with the addition of individually optimised RWS and TVC and optimization of combined ICLC methods. In the evaluation and results part, all proposed systems were evaluated in terms of vehicle handling and stability with open and close loop test scenarios. In the last section, the study was concluded.

2. LITERATURE REVIEW

This chapter is organized as follows; First, the vehicle dynamics models used in the literature were expressed, then the vehicle/control models used in this study were introduced. Later, the detailed review of RWS, TVC and combined systems were examined.

2.1 Vehicle Dynamics Models

Vehicle systems modelling is the process of creating mathematical models to represent the various systems that built up a vehicle, such as the suspension, steering, brakes, and powertrain. These models are used to predict the behaviour of the vehicle under different conditions, such as when driving on different road surfaces or different manoeuvres. In the literature, an important aspect of vehicle dynamics models is the number of degrees of freedom that they incorporate. The degrees of freedom of a model refers to the number of independent variables required to describe the motion of the vehicle. For example, a 2-degree-of-freedom (2DOF) model would require two variables to describe vehicle motion, such as lateral displacement and yaw angle. A 3DOF model would require three variables, and so on. In general, there are several vehicle dynamics models used in the literature, including the 2DOF, 3DOF, 7DOF, 14DOF and MBD (+30 DOF) models [10 -13]. These models differ in the number of degrees of freedom that they incorporate, and each has its purpose of usages. The general representation of different vehicle models was shown in Figure 2.1. The brief explanations of models from the literature are given below:

- 2DOF models are simple and easy to analyse, and they are often used to study the stability and handling characteristics of a vehicle. However, they can also be used as reference vehicle models as they remove much of the complexities and nonlinear behaviour of a complete vehicle system.
- 3DOF models are more complex than 2DOF models and capture much of the vehicle's roll dynamic behaviour. They are often used to study the interaction

between the tyres and the road surface, as well as the effect of various suspension parameters on the vehicle's behaviour.

- 7DOF models are more complex than 3DOF models and capture more of the vehicle's roll and ride dynamics behaviour. They are often used to study the ride dynamics of a vehicle.
- 14DOF vehicle models are used in some studies to represent the dynamic behaviour of a vehicle with a remarkably prominent level of detail. The 14 degrees of freedom in a 14DOF vehicle model include the 6 degrees of freedom that are present in a 6DOF model (3 translational DOF and 3 rotational DOF), as well as an additional 8 DOF that represent the motion of various components within the vehicle. These additional degrees of freedom may include the motion of the wheels and tyres, the motion of the suspension components, and the motion of the powertrain components, among others.
- Multi-Body Dynamics (MBD) systems can have a wide range of degrees of freedom, depending on the complexity of the system being modelled. However, in general, the software which has complex vehicle models has more than 30 degrees of freedom if all non-linear components and elements are considered. In the industry, MBD software's preferred to simulate vehicle dynamics behaviour [14] and +30 DOF models can generally be classified into two groups.
 - Multi-Body dynamics (MBD) models are mathematical models that represent the dynamics behaviour of a system using equations of motion. MBD models can capture the complex interactions between the various rigid and flexible components of the system and can be used to predict the system's behaviour under a wide range of conditions. Some MBD software widely used in the industry: MSC Adams, Altair Motion Solve, Simpack
 - Lookup table-based models, on the other hand, represent the system's behaviour using a table of pre-computed data. Lookup table-based models are typically simpler than MBD models and are easier to implement, but they are limited to the specific set of input conditions

for which data has been pre-computed. Some table-based software widely used in the industry: IPG CarMaker, AVL VSM, CarSim

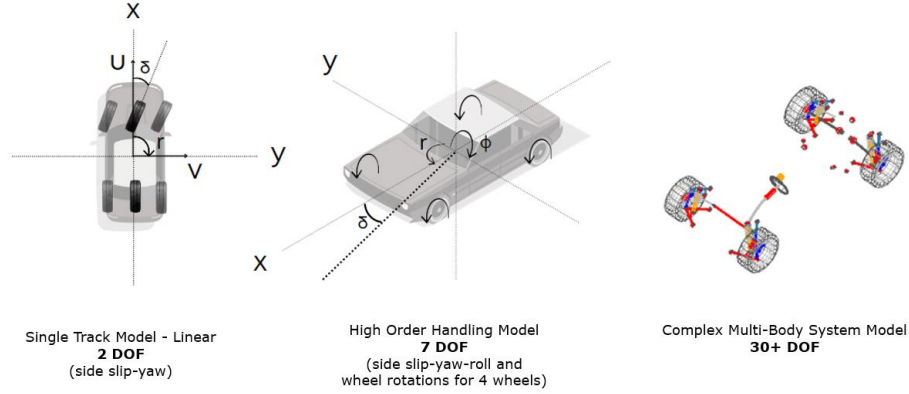


Figure 2.1: Vehicle models with different DOFs.

In this study, lookup table-based model (+30 DOF) was used as vehicle model since the implementation and co-simulation of chassis control system is easier than MBD models. Therefore, vehicle model that is used in this study allows taking advantage of the capabilities of nonlinear subsystem models (bushings, dampers, suspensions, complex tyre model, chassis etc.). Vehicle dynamics model was prepared in AVL VSM software [15]. 2DOF bicycle model was used for the RWS and TVC control systems since both systems directly influence the yaw stability of the vehicle. Table 1.1 shows the details and purposes of used models in this study.

Table 1.1: Vehicle and control system models in this study.

Model Description	DOF	Model/Controller Type	Software
Table-Based Non-Linear Vehicle Model	30+	Vehicle Dynamics Simulation Model	AVL VSM
RWS	2	RWS control with feedforward and feedback	MATLAB Simulink
TVC	2	TV control with feedforward and feedback	MATLAB Simulink
ICLC (RWS + TVC)	2	Combined control of RWS and TVC	MATLAB Simulink

Details of vehicle and control system modelling were examined in next chapters in detail, but also a general scheme of the vehicle and control system models is shown in Figure 2.2. A non-linear vehicle model provided by AVL VSM generates the required input data for the RWS and TVC chassis control systems. RWS and TVC outputs are

evaluated in ICLC, and adjusted signals are sent to vehicle model. A vehicle model with integrated chassis control can be simulated by this co-simulation environment.

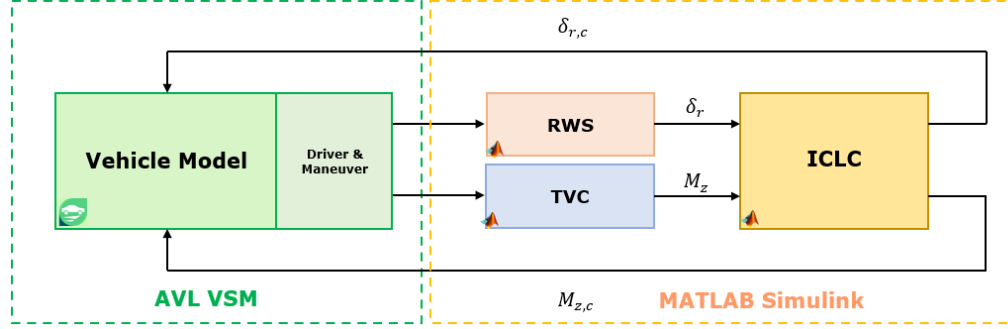


Figure 2.2: Vehicle and motion control system modelling scheme.

Before examining the systems that used for chassis control and lateral dynamics, the vehicle axis system should first be specified to make the terminology to be used more understandable. In this thesis, ISO vehicle coordinate system was used [16]. ISO coordinate system was shown in Figure 2.3. Inertial Reference Frame is fixed to earth with the vertical Z-axis and the X/Y horizontal plane. Vehicle Reference Frame is attached to the vehicle at fixed reference point (CoG) and +x towards to the front. In this axis system, when the vehicle turns left, the lateral displacement of the vehicle, the steering angle and the lateral yaw angle become positive.

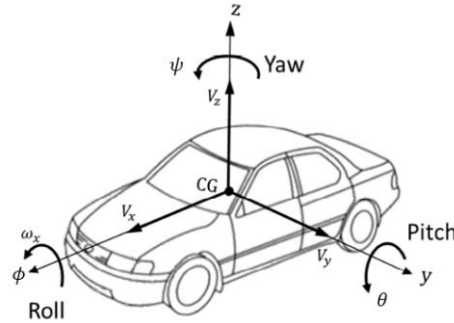


Figure 2.3: ISO axis system (ISO 8855-2011).

As mentioned above, two degrees of freedom (2DOF) vehicle models are widely used in the field of vehicle lateral dynamics due to their simplicity and effectiveness in providing accurate solutions for various problems. The term passive vehicle is used in the literature to express the basic vehicle model in which there is no control. There are already many different expressions to define the basic vehicle Newton-Euler equations [11]. Figure 2.4 shows the free body diagram of the vehicle represented by the well-known bicycle model in accordance with ISO axis system.

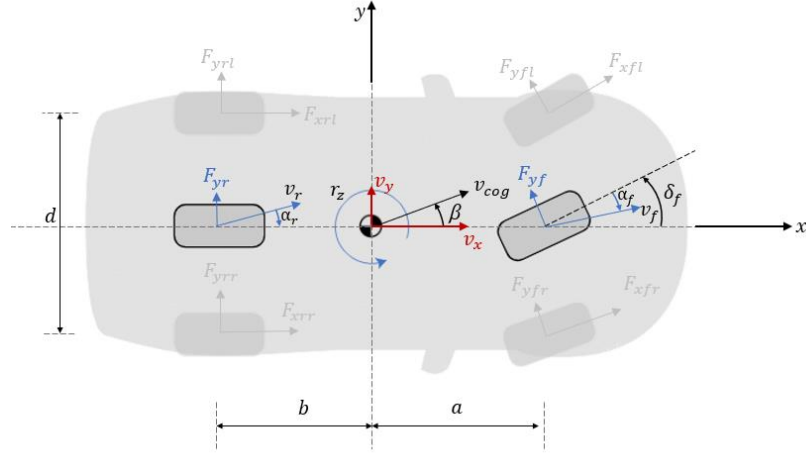


Figure 2.4: Passive vehicle (baseline) bicycle model.

The model includes lateral and yaw degrees of freedoms, as seen in equations (1) and (2), where v_y and v_x are the lateral and longitudinal velocities of the vehicle, m is the vehicle mass, F_{yf} and F_{yr} the lateral forces for the front and rear tyres, I_{zz} is yaw moment of inertia of vehicle, a and b are the front and rear axle distances to the CG, respectively:

$$m(\dot{v}_y + v_x r_z) = F_{yf} + F_{yr} \quad (1)$$

$$I_{zz}(\dot{r}_z) = aF_{yf} - bF_{yr} \quad (2)$$

The mean slip angles of the front (α_f) and rear (α_r) tyres can be derived as follows for the vehicle without rear wheel steering where δ_f is front tyres mean steering angle, r_z is the yaw rate of vehicle body and β is the side slip angle of vehicle body:

$$\alpha_f = \delta_f - \beta - \frac{r_z a}{v_x} \quad (3)$$

$$\alpha_r = -\beta + \frac{r_z b}{v_x} \quad (4)$$

Using the linear tyre model, expressions of the front and rear lateral tyre forces can be expressed as the functions of C_f and C_r , which are the cornering stiffness of the front and rear tyres, respectively:

$$F_{yf} = \alpha_f C_f \quad (5)$$

$$F_{yr} = \alpha_r C_r \quad (6)$$

By substituting equations (3) - (6) in model equations (1) and (2), the state-space representation of the vehicle dynamics model can be obtained:

$$\underbrace{\begin{bmatrix} \dot{v}_y \\ \dot{r}_z \end{bmatrix}}_{\dot{x}} = \underbrace{\begin{bmatrix} \frac{C_f + C_r}{mv_x} & \frac{aC_f - bC_r}{mv_x} - v_x \\ \frac{aC_f - bC_r}{I_{zz}v_x} & \frac{a^2C_f + b^2C_r}{I_{zz}v_x} \end{bmatrix}}_A \underbrace{\begin{bmatrix} v_y \\ r_z \end{bmatrix}}_x + \underbrace{\begin{bmatrix} -\frac{C_f}{m} \\ -\frac{aC_f}{I_{zz}} \end{bmatrix}}_B \underbrace{[\delta_f]}_u \quad (7)$$

$$\underbrace{\begin{bmatrix} v_y \\ r_z \end{bmatrix}}_y = \underbrace{\begin{bmatrix} 1 & 0 \\ 0 & 1 \end{bmatrix}}_C \underbrace{\begin{bmatrix} v_y \\ r_z \end{bmatrix}}_x + \underbrace{\begin{bmatrix} 0 \\ 0 \end{bmatrix}}_D \underbrace{[\delta_f]}_u \quad (8)$$

In the state-space representation of the 2 DOF bicycle model, the control input is only the front steering angle if the vehicle speed is assumed constant. The state variables are lateral velocity (v_y) and yaw rate (r_z). Here, some assumptions should be made to clarify the capability of the 2 DOF bicycle model [11].

- Suspension is assumed to be sufficiently rigid, so out of plane motions (roll and pitch) can be disregarded.
- The wheels of the vehicle are supposed to remain in a vertical position, i.e., zero camber.
- The external forces, cornering and traction or braking, are assumed to act from the ground to the four tyres in their contact centres, disregarding the self-aligning torques.
- The effects of tractive (or braking) force on tyre side force characteristics are neglected.
- Load transfer from inside wheels to outside wheels during cornering and the corresponding change in tyre side force characteristics are negligible.
- Steering control with equal steering angles of the left and right front wheels is assumed. Front tyres mean steer angle $\delta_f = (\delta_{fl} + \delta_{fr})/2$.

In chassis control development studies, the 2 DOF model is widely used [17,18]. In this study, RWS and TVC control system calculations were also done with the 2 DOF models. Although the control systems were modelled as 2 DOF since it is simpler to calculate the equations of motion, this study considered the full vehicle's non-linear behaviour.

2.2 Rear Wheel Steering (RWS)

There are many different systems in the literature to improve the lateral performance of vehicle dynamics with techniques such as the rear wheel steering (RWS) system. The system can also be called rear axle steering (RAS), all-wheel steering (AWS) and four-wheel steering (4WS) in some applications or papers [19]. The rear wheel steering system is a vehicle-integrated chassis control system in which the rear wheels can turn independently of the front wheels via hydraulic /electric actuators. The location of the RWS actuator can be seen in Figure 2.5. In general, rear-wheel steering actuators have their electronic control units (ECUs) in a vehicle's control system.



Figure 2.5: RWS actuator example.

Different RWS strategies are shown in Figure 2.6. Counter steering allows the vehicle to have a smaller turning radius when the rear wheels are steered opposite the front at low speeds. With parallel steering, vehicle handling and stability are improved when the rear axles are steered in phase at high speeds. In the automotive industry, companies are using RWS in their vehicles in the industry [20].

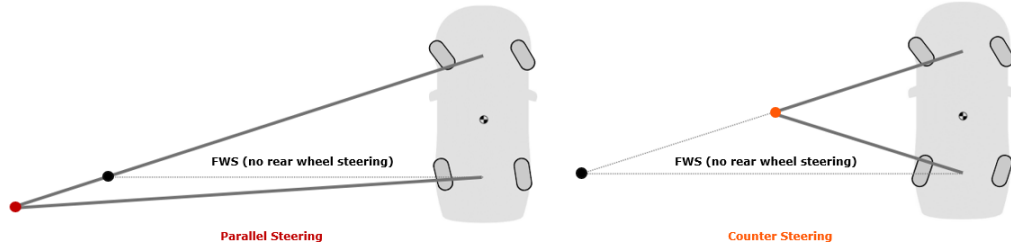


Figure 2.6: Steering strategies of RWS.

Melman et al. offered different drive modes for RWS function; sport and comfort modes [20]. In their study, the rear wheel steering angle was expressed as a function of the steering wheel angle and vehicle speed for sport and comfort modes. Accordingly, with the RWS, vehicle cornering response could be calibrated according to the desired driving mode. Figure 2.7 illustrates the cornering response of these drive modes with RWS and their cornering range with ramp steer manoeuvre. Thus, steering sensitivity, driver feeling, and safety-critical control strategies of RWS can be adjusted for RWS optimization and driver modes. The sport mode can make the vehicle more responsive and lead to more lateral acceleration with the same steering angle and vehicle speed as the comfort mode. Conversely, the comfort mode makes the vehicle less responsive.

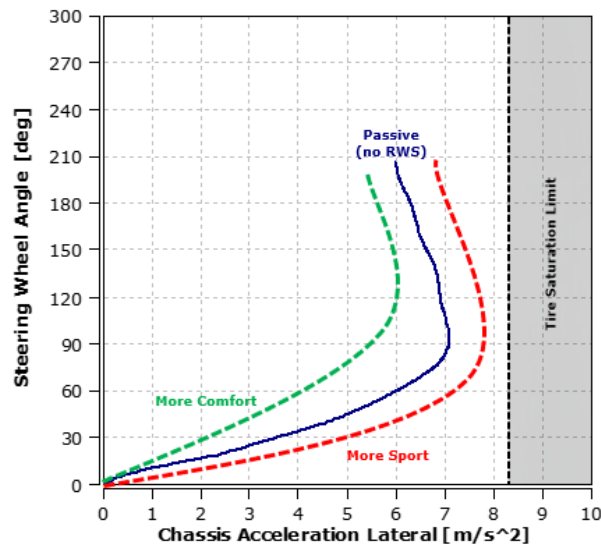


Figure 2.7: RWS with different drive modes

In addition, many papers in the literature propose different RWS control methods and schemes. Siahkalroudi and Naraghi divided the RWS control strategies into three categories: zero-side slip, zero yaw rate and combined model with reference tracking [21]. Additionally, Whitehead analysed the RWS with different approaches to minimise the vehicle side slip angle [22]. In this study, the system matrix is modified, and a vehicle's stability characteristic is changed to keep the vehicle side slip at zero during steady state and transient manoeuvres. The zero-side slip strategy aims to improve manoeuvrability by reducing the turning radius and enhancing vehicle stability to minimise side slip. Still, the system increases the yaw rate, which is unsuitable at higher speeds. Other than that, the zero-yaw rate strategy is more suitable for high speeds, but on the contrary, it needs to improve manoeuvrability. Sylvain proposed that a design approach with multiple strategies is better for overall control, recommending the use of tracking reference yaw rate and derivative of the side slip strategy [23]. This combined strategy has been used by many researchers, with the results proving an improvement in terms of vehicle handling and stability [24-26]. Although the RWS system proposed by the researchers enhances the vehicle handling performance, it has been examined that it is carried out with a limited vehicle model. Thus, this study offers an implementation of a calibratable RWS system on an electric vehicle with different drive modes and a model-based parameter optimisation methodology for suited rear-wheel steering system parameters integrated into a high-fidelity non-linear vehicle model. As mentioned, the rear-wheel steering angle is zero in a passive vehicle without a rear-wheel steering system. When the RWS is added to the system, equation (4) is modified, and the rear wheel slip angle becomes:

$$\alpha_r = \delta_r - \beta + \frac{r_z b}{v_x} \quad (9)$$

Thus, modifying the conventional bicycle model and force-moment equations is necessary. Firstly, the new free-body diagram with RWS is shown in Figure 2.8.

each wheel, which can help redistribute the vehicle's weight and improve its overall stability and handling. Torque vectoring systems can be implemented in several different ways, including electronic stability control systems, brake-based systems, or by adjusting the power output of individual wheels or motors in a hybrid or electric vehicle. Figure 2.9 represents the torque vectoring control system. Wheel torques are controlled in TVC and actuated to get ideal conditions for traction, cornering, stability, and efficiency. In the literature, the torque distribution strategies of TVC can be divided into two areas, longitudinal and lateral dynamics [27].

The purpose of longitudinal torque split in torque vectoring control is to improve traction and stability by adjusting the amount of torque transmitted to a vehicle's front and rear wheels. This can be done to help the vehicle accelerate more quickly or to improve its stability and handling during high-speed driving or severe cornering manoeuvres. In addition to improving traction, longitudinal split, and efforts to optimize longitudinal torque distribution are also associated with energy efficiency studies. Bovee et al. used experimental validation of the torque shaping algorithm, which includes a genetic algorithm, to optimize the energy management system of a vehicle equipped with two electric machines [28]. A torque allocation algorithm was proposed by Koehler et al. to provide the required propulsion force and the yaw torque with minimized power loss [29]. This proposed algorithm aims to increase energy efficiency based on torque distribution. An efficiency gain for numerous driving situations is calculated based on the two principles. The first principle is the calculation of a yaw torque supports the vehicle dynamics to reduce energy losses due to wheel slip. The second principle is the application of the torque demanded by the wheel of individual electric machines adaptively, taking their energy efficiency characteristics into account. Experimental results show that the proposed torque vectoring control system can reduce energy consumption by 10% for typical driving situations involving high and low lateral acceleration scenarios. The literature adds the energy-efficient mode to the TVC control strategies [30]. There is a methodology studied by Lenzo, who offers that choosing a front-to-total torque ratio can aim to minimize vehicle power losses. When the driver selects energy efficient mode, the TVC system only aims to reduce energy consumption.

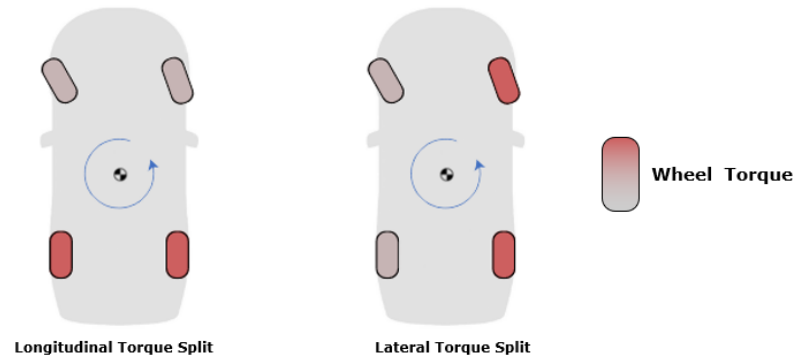


Figure 2.9: Torque vectoring control.

In addition, the purpose of lateral torque split in torque vectoring control is to improve cornering performance by adjusting the amount of torque transmitted to a vehicle's left and right wheels. There are some studies in the literature within the scope of optimization and control of the lateral controller. Mikulas et al. developed an optimization-based torque vectoring control strategy for an independent all-wheel drive electric race car to stabilize the vehicle at the handling limits and improve its track performance [31]. The proposed study has been evaluated in several simulation scenarios to assess the performance of the TVC system and without the TVC system, such as a 10 m radius U-turn scenario and steering step response. Based on the obtained results, the proposed torque vectoring systems were able to stabilize the vehicle close to the handling, prevent the vehicle spin and enable it to reach a smaller cornering radius than one possible with equal torque distribution at a given speed. Mangia et al. developed an integrated TVC framework allowing the driver to select drive modes that change vehicle cornering performance in separate ways or improve vehicle energy efficiency. The high-level controller in their study keeps the balance between the ideal yaw rate and ideal sideslip angle tracking, prioritizing the latter in dangerous situations and reducing excessive sideslip angles for increased vehicle safety. The study also expresses different drive modes of TVC: balanced (normal) mode, sport mode and energy efficient mode. Figure 2.10 illustrates the cornering response of these drive modes with the TVC in a ramp steer manoeuvre. It is clearly seen that TVC aims to decrease the understeer behaviour of a vehicle and can make the vehicle more responsive and leads to more lateral acceleration with the same steering angle.

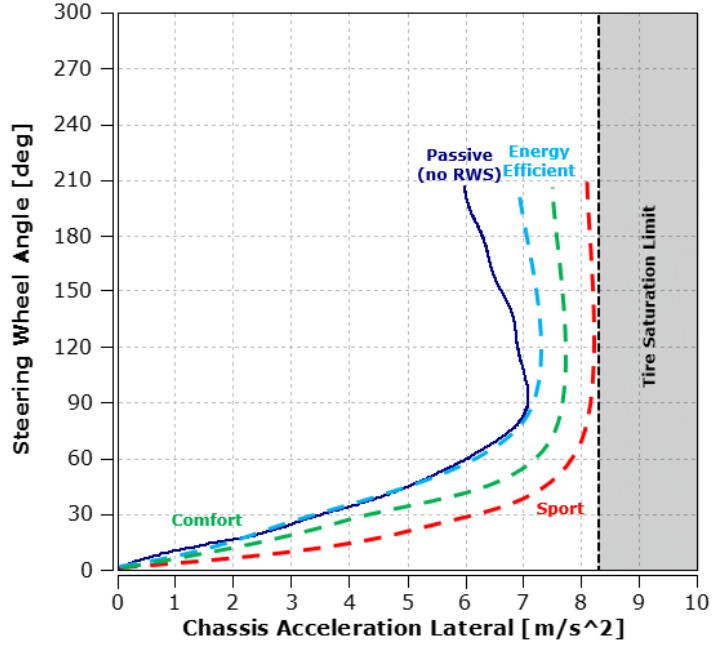


Figure 2.10: TVC with different drive modes.

Similarly, as in RWS, steering sensitivity, cornering response and safety-critical control strategies of TVC can be adjusted by calibration and driver modes.

From the literature, when the TVC is added to the system, the equation (2) is modified, and the moment equation becomes:

$$I_{zz}(\dot{r}_z) = aF_{yf} - bF_{yr} + M_z \quad (12)$$

Thus, the bicycle model and force-moment equations need to be changed. The free-body diagram with TVC is shown in Figure 2.11.

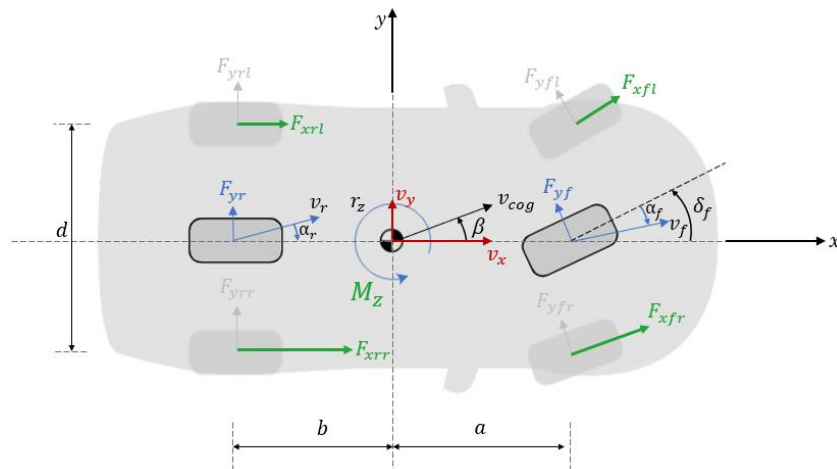


Figure 2.11: Bicycle model with TVC.

Since additional yaw moment is added to equation (7), the state-space representation of the vehicle dynamics model with TVC can be obtained:

$$\underbrace{\begin{bmatrix} \dot{v}_y \\ \dot{r}_z \end{bmatrix}}_{\dot{x}} = \underbrace{\begin{bmatrix} \frac{C_f + C_r}{mv_x} & \frac{aC_f - bC_r}{mv_x} - v_x \\ \frac{aC_f - bC_r}{I_{zz}v_x} & \frac{a^2C_f + b^2C_r}{I_{zz}v_x} \end{bmatrix}}_A \underbrace{\begin{bmatrix} v_y \\ r_z \end{bmatrix}}_x + \underbrace{\begin{bmatrix} -\frac{C_f}{m} & 0 \\ -\frac{aC_f}{I_{zz}} & \frac{1}{I_{zz}} \end{bmatrix}}_B \underbrace{\begin{bmatrix} \delta_f \\ M_z \end{bmatrix}}_u \quad (13)$$

$$\underbrace{\begin{bmatrix} v_y \\ r_z \end{bmatrix}}_y = \underbrace{\begin{bmatrix} 1 & 0 \\ 0 & 1 \end{bmatrix}}_C \underbrace{\begin{bmatrix} v_y \\ r_z \end{bmatrix}}_x + \underbrace{\begin{bmatrix} 0 & 0 \\ 0 & 0 \end{bmatrix}}_D \underbrace{\begin{bmatrix} \delta_f \\ M_z \end{bmatrix}}_u \quad (14)$$

In the state-space representation of the 2 DOF bicycle model with TVC, the control input is the front steering angle. An additional yaw moment control input is added to the system. The additional yaw moment is created by the difference in torque being applied to the wheels on either side of the vehicle. For example, suppose the torque at the right wheel is increased while the torque at the left wheel is decreased, as represented in Figure 11. In that case, this torque difference causes the vehicle to rotate around its vertical axis (yaw) in a counterclockwise direction. Thus, a yaw moment is applied via individual wheel torque distribution. The additional yaw moment M_z was controlled in the TVC system. The details of the torque vectoring controller were examined in detail in the control system modelling chapter.

2.4 Integrated Chassis Lateral Controller (ICLC)

In the industry, many chassis control systems are procured by automotive suppliers to enhance a particular vehicle's performance. Both original equipment manufacturers (OEMs) and chassis controller development companies are primarily interested in integrated chassis control systems. Different coordination architectures have been proposed to combine chassis systems into a vehicle [33,34]. Kissai et al. categorized ICC systems into two different coordination [34]: downstream and upstream coordination. The architecture of each ICC coordination strategy is represented in Figure 2.12 in the adapted form [33,34].

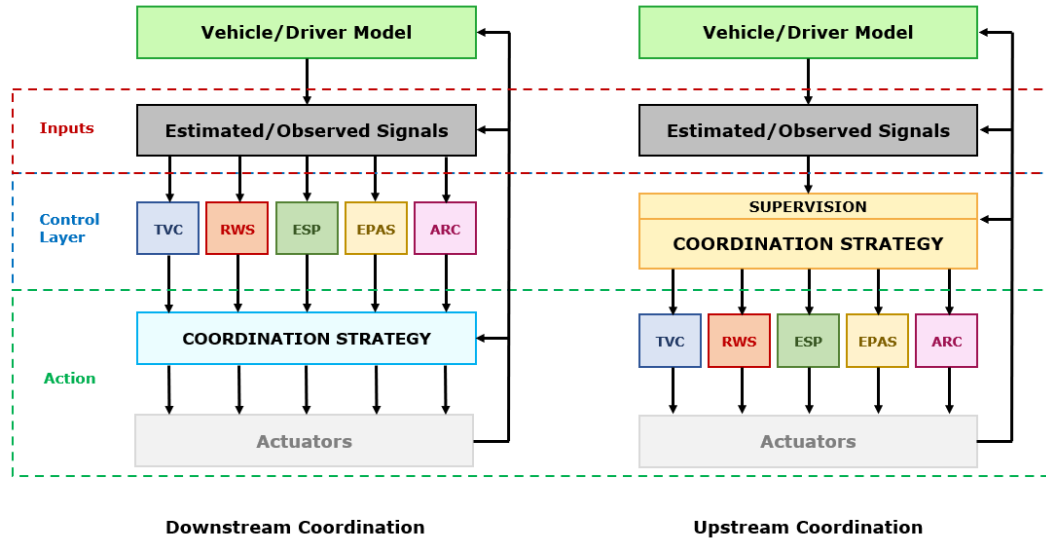


Figure 2.12: Example of downstream and upstream ICC architectures.

The main difference between the two different architectures is the position of the coordination strategy layer in the control flow, where TVC, RWS, ESP, EPAS and ARC are the main chassis control functions previously defined.

2.4.1 Downstream architecture

Downstream coordination architectures have a parallel structure that allows all chassis control systems to operate independently. The coordination of the different chassis control actions is defined in the lowest level of the control flow. Each chassis control function may have its own ECU, high-level controller, reference generator and control strategies. The coordination strategy block receives the outputs of each chassis control function and sends the corrected action signals to the actuators concerning some allocation rules or system. There are different coordination strategies for the downstream ICC architecture [34-36]:

- The most prominent module activation coordination strategy uses a straightforward approach. The highest input precedes the other chassis control inputs [35]. For example, a vehicle equipped with ESC and TVC needs a yaw torque demand in the cornering; the coordination strategy chooses the highest input from ESC and TVC outputs using switches. However, this switch transition between different actuators may lead to undesirable transients and information for the vehicle behaviour and driver feel. Thus, more intelligent

ways, such as artificial neural networks and fuzzy logic, provide more smooth transitions between actuations.

- The artificial neural network coordination strategy uses a voting-based non-linear interpolation function to optimize the weighting factors of coordination control of actuators [35].
- Fuzzy logic uses rule-based coordination to define the weighting factors of the actuators [36]. The fuzzy coordinator decides on ‘small’ and ‘large’ input signals for the smooth transition to switch the weighting factors of the actuators.

Different downstream ICC control architectures are represented in Figure 2.13, adapted from [35].

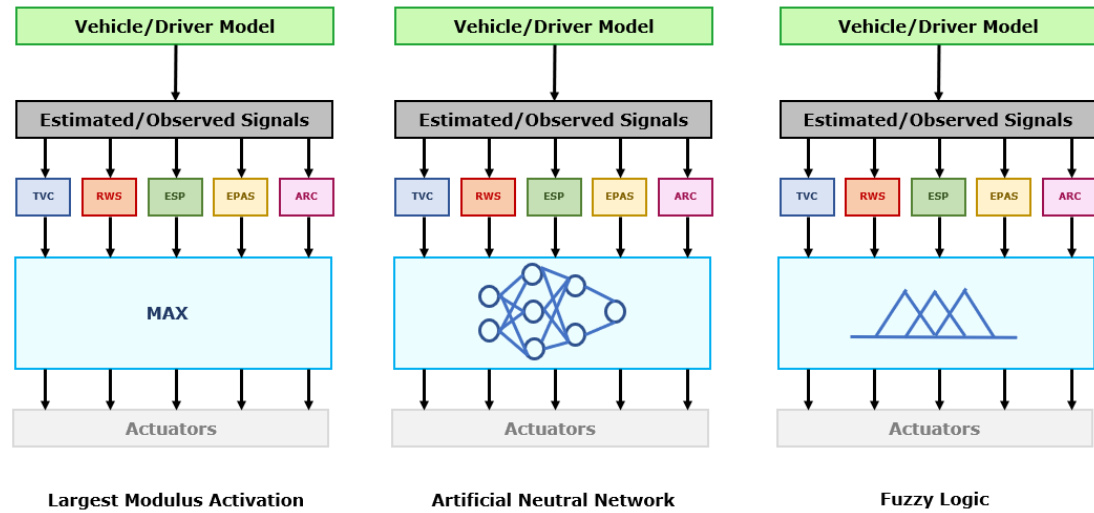


Figure 2.13: Scheme of different downstream ICC architectures.

The main advantage of using a downstream architecture for ICC is the ease of industrial implementation because the controllers/actuators of different chassis control systems can be purchased from other suppliers. As product owners of electric vehicles, many companies would prefer downstream architecture in the future for integration and cost reasons. In addition, downstream coordination requires less complexity. The methodology of the coordination strategy can be calculated using phase-portraits analyses [37] or lateral dynamics rules using fuzzy logic. Apart from that, the downstream architecture also has some disadvantages. Downstream ICC architecture could lead to safety issues in random situations because test cases for testing and optimization coordination strategy cannot consider all driving conditions.

Additionally, since each system has its own high-level control and reference values, it may affect the reliability of the coordination strategy. Therefore, using downstream architecture may not be appropriate in terms of adaptability and multi-objective coordination in an ICC.

2.4.2 Upstream architecture

Slightly different from the downstream method, the control inputs are decided and distributed before the chassis control systems in the upstream architecture to avoid conflicts. Other chassis control actions are coordinated in the mid-level of the control flow. ECUs, high-level controllers, and reference generators can be standard in different chassis control strategies. Thus, when considering the next generation of autonomous vehicles, managing other ICC systems may be easier with the upstream method. There are different coordination strategies for upstream ICC architecture [34]:

- The centralized coordination strategy uses a multi-variable controller to define the control actions of each chassis control system. A multi-variable control strategy can be designed according to the desired performance. For example, if the vehicle's stability is considered, the combined objectives are considered, and again fuzzy control or rule-based control methods can be applied for coordination. Since combined goals are considered, the conflicts between different chassis systems are decreased.
- In the supervisory coordination strategy, control actions are monitored, and driving states are defined. This allows the method to find the instabilities of driving behaviour and detects system failures. Thus, supervisory coordination takes advantage of functional safety topics.
- The Multi-Layer coordination strategy works similarly to supervisory coordination but is more modular. It allows the ICC to separate the functional safety requirements, monitoring the vehicle states and high-level controller design.

Different downstream ICC control architectures are represented in Figure 2.14, adapted from [34].

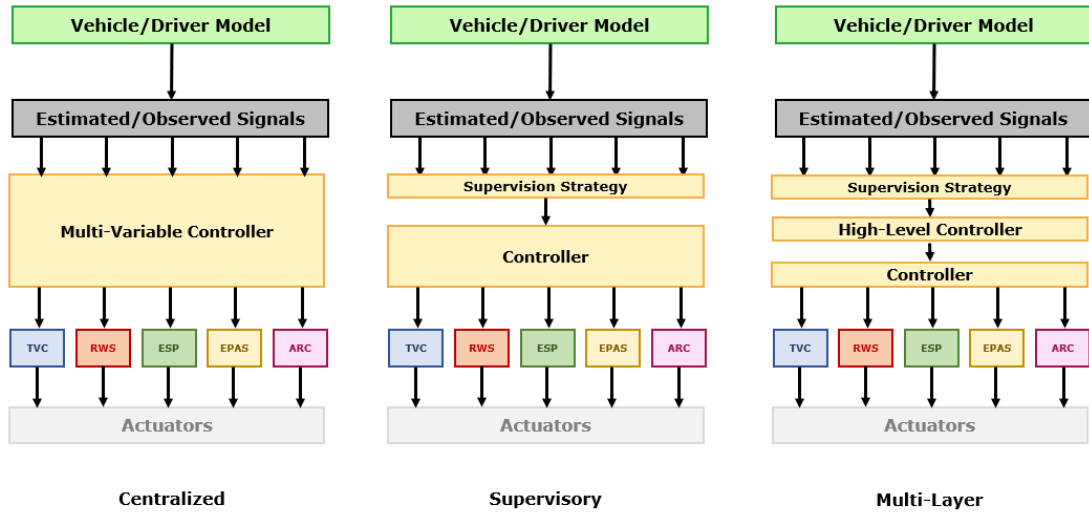


Figure 2.14: Scheme of different upstream ICC architectures.

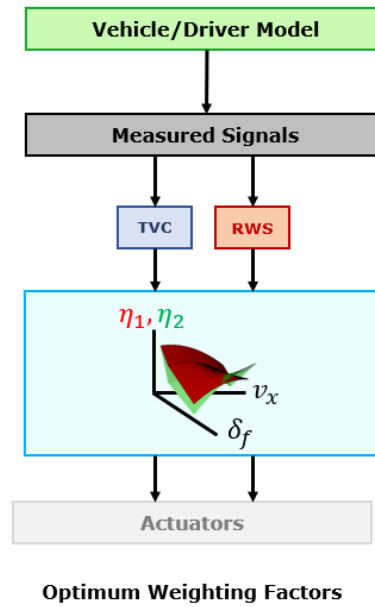
The main advantage of using the upstream architecture for ICC is its adaptability to face behavioural changes in the driver and environment. It is very compatible with the multi-objective coordination design. Also, designing a typical stability controller is easier with the upstream architecture. However, communicating and generating a common high-level controller may be challenging if an OEM takes a chassis control system from any supplier. Finally, stability, lateral and longitudinal targets and their cooperation are considered in creating a common high-level controller, or multi-variable controller design is more complex when compared to the downstream architecture.

Consequently, the literature has shown that integrated chassis control architecture is trendy and will become more popular when autonomous vehicles and x-by-wire technologies are considered [38]. Table 2.1 shows the comparison of these two ICC architectures. This thesis used the downstream architecture due to its simplicity and low computational cost. Also, the downstream architecture is more appropriate for OEM and supplier independence. With the downstream architecture, two different systems, RWS and TVC, can be purchased from other suppliers, and ICC coordination can be designed accordingly.

Table 2.1: Comparison of different ICC architectures [33].

Evaluation Criteria	Downstream	Upstream
Implementation cost	Appropriate	Neutral
Simplicity	Appropriate	Inappropriate
OEM - Supplier independency	Appropriate	Inappropriate
Computational cost	Appropriate	Neutral
Adaptability for connected control	Neutral	Appropriate
Supervision for the safety	Inappropriate	Appropriate
Multi-objective coordination	Inappropriate	Appropriate

As previously mentioned in Figure 2, RWS and TVC outputs are evaluated in ICLC, and the corrected signals are sent to the actuators as same as downstream coordination. A coordination strategy in this thesis is shown in Figure 2.15. It is the optimum distribution of weighting factors of RWS and TVC for different manoeuvres and conditions. In this study, optimization was made for both RWS and TVC systems separately. Additionally, the optimum weighting factor downstream coordination strategy and optimization methodology of the combined control for ICLC were proposed and introduced.

**Figure 2.15:** The downstream ICC architecture used in this study.

2.5 TVC-RWS Control

When the literature is examined, there are studies in which the two systems, RWS and TVC, are used together, and their effects are compared/shown [39-43]. All these papers suggest that it is possible to use torque vectoring control (TVC) and rear wheel steering (RWS) to control vehicle dynamics. All present different methods for designing such a combined ICC system. Vignati and Sabbioni proposed a phase portrait analysis to find weighting factors of RWS and TVC actuation [39]. A performance index was proposed to evaluate which weighing factor action capable of leading the vehicle to the nearest equilibrium point at the fastest rate. Two different systems used the same reference generator in the control scheme. In the second layer of the architecture, performance index maps were defined to evaluate the response of each control system for the yaw rate (r_z) and side slip angle (β) control. Performance index maps are pre-defined maps that are generated from phase-portrait (β - r_z) results with different speed and steering angles. The downstream architecture was used in their study. In another study, a feedforward control strategy was presented to follow desired target values using tyre information [40]. Additionally, some studies used a sliding mode controller with RWS and TVC studies to improve the driveability and stability of a car [41,42].

Similarly, it was found that the combined usage of RWS and TVC improves the capability of tracking desired values. Lastly, combining RWS and TVC were combined with the NMPC strategy for electric vehicles on low-friction surfaces [43]. Liu et al. used the upstream architecture in their study. A nonlinear model predictive controller (NMPC) was used as the coordination strategy. Performance and energy consumptions of RWS, TVC only and combined control strategies were evaluated on low friction coefficient conditions. According to their results, the proposed combined method had a better performance in performance improvement and less energy consumption than the traditional TVC-only control method. When the studies in the literature are examined with the scope of RWS and TVC, the recent publication shows the subject's popularity and the need for a more comprehensive investigation. Within the area of this thesis, the systems in which the TVC and RWS control systems of an electric vehicle are used separately and together were compared, and the effect of their use on the development of lateral dynamics was examined with a non-linear vehicle model. From the literature, when the RWS and TVC are added to the system, the

bicycle model and force-moment equations need to be changed. The new free-body diagram with RWS and TVC is shown in Figure 2.16.

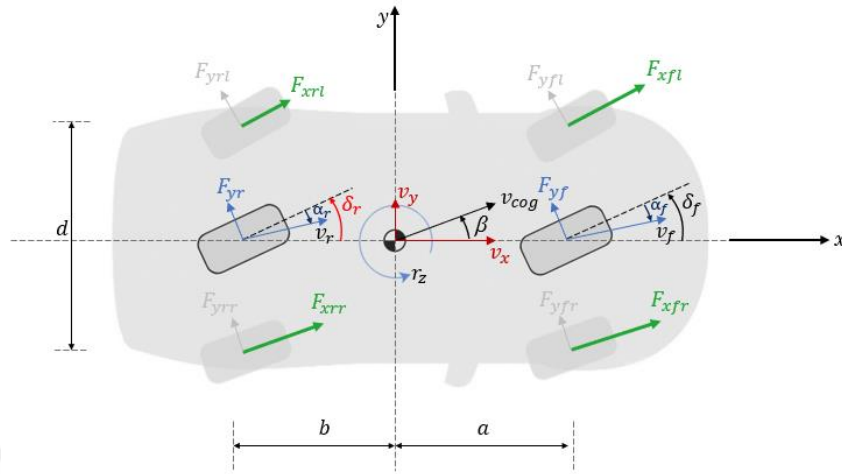


Figure 2.16: Bicycle model with RWS and TVC.

Since rear wheel steering input is added to equation (7), the state-space representation of the vehicle dynamics model with RWS can be obtained:

$$\begin{bmatrix} \dot{v}_y \\ \dot{r}_z \end{bmatrix} = \underbrace{\begin{bmatrix} \frac{C_f + C_r}{mv_x} & \frac{aC_f - bC_r}{mv_x} - v_x \\ \frac{aC_f - bC_r}{l_{zz}v_x} & \frac{a^2C_f + b^2C_r}{l_{zz}v_x} \end{bmatrix}}_A \underbrace{\begin{bmatrix} v_y \\ r_z \end{bmatrix}}_x + \underbrace{\begin{bmatrix} -\frac{C_f}{m} & -\frac{C_r}{m} & 0 \\ -\frac{aC_f}{l_{zz}} & \frac{bC_r}{l_{zz}} & \frac{1}{l_{zz}} \end{bmatrix}}_B \underbrace{\begin{bmatrix} \delta_f \\ \delta_r \\ M_z \end{bmatrix}}_u \quad (15)$$

$$\underbrace{\begin{bmatrix} v_y \\ r_z \end{bmatrix}}_y = \underbrace{\begin{bmatrix} 1 & 0 \\ 0 & 1 \end{bmatrix}}_C \underbrace{\begin{bmatrix} v_y \\ r_z \end{bmatrix}}_x + \underbrace{\begin{bmatrix} 0 & 0 & 0 \\ 0 & 0 & 0 \end{bmatrix}}_D \underbrace{\begin{bmatrix} \delta_f \\ \delta_r \\ M_z \end{bmatrix}}_u \quad (16)$$

In the state-space representation of the 2 DOF bicycle model with RWS and TVC, rear-wheel steering and direct (additional) yaw moment control inputs are added to the system together. A pseudo torque was directly applied to the centre of gravity, and the additional yaw moment M_z was controlled in the TVC system. Rear wheel steering δ_r was controlled in the RWS control system. As mentioned before, since this study uses the downstream ICC architecture, the control inputs were coordinated in the ICLC system with weighting factors, as in Figure 2.2.

Lastly, with the controllers of RWS and TVC, steering sensitivity, cornering response, and safety-critical control strategies can be adjusted by calibration and driver modes together. Figure 2.17 shows the effect of the RWS and TVC chassis control systems used on the lateral dynamic's behaviour. The blue line represents the passive vehicle

characteristic. TVC is more effective in shaping the understeer characteristic in all lateral acceleration ranges with respect to RWS. RWS can change a vehicle's understeer characteristic in low-medium lateral acceleration ranges effectively, but it is not effective as TVC in high lateral acceleration ranges due to safety reasons and steering limitations. Thus, using a combination of these systems can reduce the effort of each system and save the total energy usage since they can serve the same purpose in some applications or driving modes.

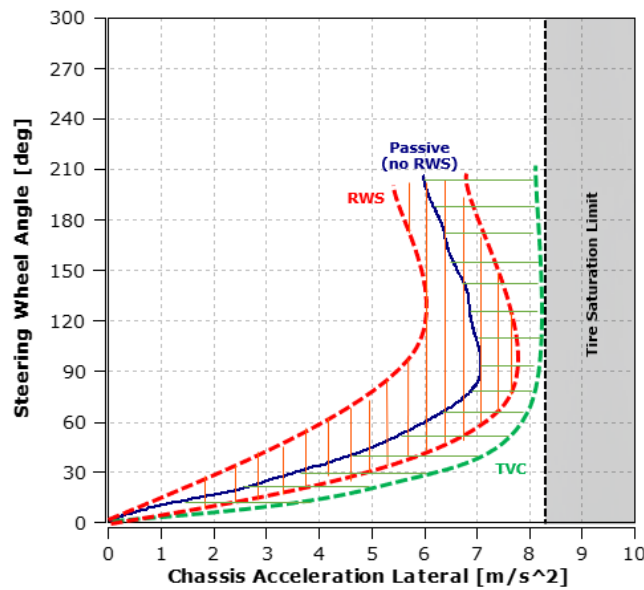


Figure 2.17: RWS and TVC approach on lateral dynamics.

2.6 Handling and Stability

In lateral vehicle dynamics, handling refers to a vehicle's ability to manoeuvre and respond to driver inputs [44]. Specifically, it relates to how well a car can turn corners, maintain a steady trajectory, and respond quickly and predictably to steering input. Handling is a performance evaluation criterion of a vehicle that can be determined by both the driver's skill and vehicle handling capabilities. Good handling in a vehicle refers to its ability to manoeuvre effectively and respond well to driver inputs [45]. A vehicle with good handling typically has a well-tuned suspension system that can effectively absorb bumps and other imperfections from the road. The tyres should also be in good condition and provide adequate grip on the road surface. A vehicle with good handling will also have precise and responsive steering, allowing the driver to easily steer the vehicle in the desired direction. When a vehicle with good handling is

driven, it feels stable and safe; it can change direction easily and make sharp turns with minimal body roll, which should increase the driver's confidence level in driving within the vehicle's limits.

On the other hand, lateral stability refers to a vehicle's ability to maintain a steady course or return to a regular course after being turned. A vehicle with good lateral stability will not drift on the road and will follow the driver's steering inputs. Yaw stability refers to a vehicle's ability to maintain or return to a steady heading after being rotated around the vertical axis. A vehicle with good yaw stability will not spin or skid when the driver turns the steering wheel.

Additionally, understeer, neutral-steer and oversteer are three handling characteristics that can occur in a vehicle. They refer to the behaviour of a car when the driver applies steering inputs. All the characteristic behaviours are shown in Figure 2.18, where a neutral steer is the desired behaviour. In general, an understeer is often considered less responsive and predictable than an oversteer.

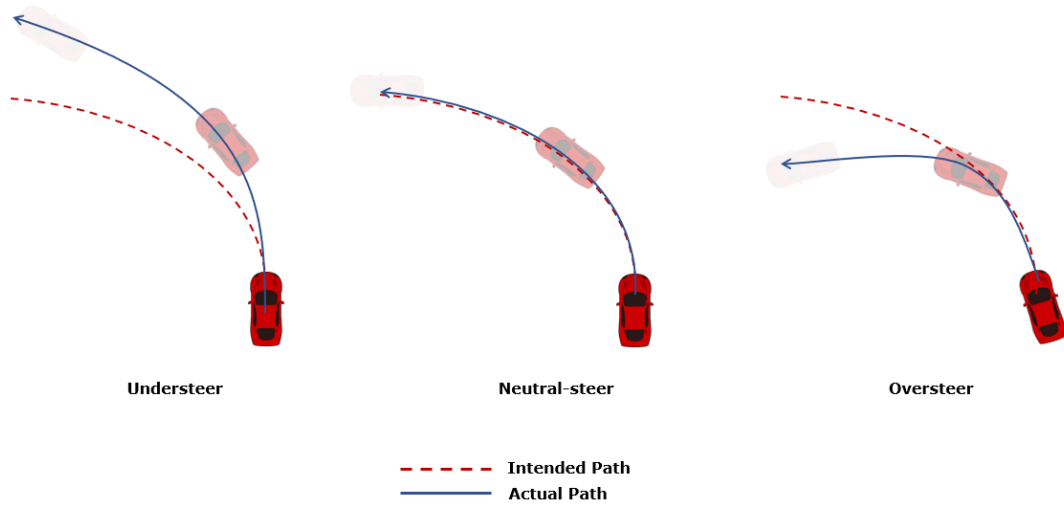


Figure 2.18: Understeer, neutral-steer and oversteer

- Understeer is a vehicle-handling characteristic that a vehicle having this characteristic cannot be turned as sharply as the driver intends. Using the equations developed for the passive vehicle bicycle model, for an understeer vehicle, the resulting absolute front wheels' mean slip angle is larger than the actual wheels' mean slip angle.

- $|\alpha_f| > |\alpha_r|$

- Neutral steer, also known as balanced or neutral handling, is when a vehicle's front and rear tires slip angles at a similar. In this case, the vehicle follows the intended curve smoothly. It is often considered the ideal handling characteristic as it allows for precise and predictable control for the driver.

- $|\alpha_f| \cong |\alpha_r|$

- Oversteer is a handling characteristic in which the vehicle turns more sharply than the amount commanded by the driver. For an oversteer vehicle, the rear wheels' mean slip angle is larger than the front wheel's mean slip angle.

- $|\alpha_f| < |\alpha_r|$

The evaluation of vehicle handling and stability can be done in both open-loop and closed-loop tests [44]. It is important to note that driving styles and skill levels can also affect how well a person perceives a vehicle to handle. For this reason, open-loop tests can be used to minimize or eliminate the driver's influence on vehicle handling. On the other hand, a close-loop test can be used to evaluate complex interactions between the vehicle system and the driver. In open-loop manoeuvres, the control inputs are predefined, as is seen in Figure 2.19.

- Open-loop tests: In the open-loop test, the vehicle follows a predefined test path or steering input, such as a ramp and step steer manoeuvres. The driver does not receive any feedback about the vehicle's motion or position on the test path and must rely solely on their skill and perception to control the vehicle. Open-loop tests are useful for evaluating a vehicle's handling characteristics without the driver's skills.
- Closed-loop tests: This type of test is characterized by monitoring the vehicle's position on the test path and comparing it to a desired trajectory, such as a double lane change [46]. The driver receives feedback on the vehicle's motion and can adjust their inputs accordingly. Closed-loop tests help evaluate the effectiveness of chassis control systems designed to improve a vehicle's handling, as well as driver-vehicle interactions.

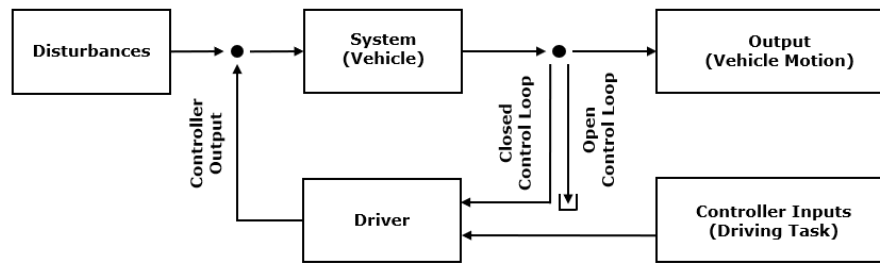


Figure 2.19: Closed and open loops.

3. VEHICLE MODELING

In this section, vehicle modelling and co-simulation environment details are given. As mentioned, the lookup table-based non-linear model (+30 DOF) was used as the vehicle model. In this study, a virtual vehicle including all the mechanical and moving parts, powertrain with an electric motor related components and a driver was modelled in AVL VSM [15]. RWS and TVC chassis control systems were modelled in MATLAB Simulink. Co-simulation environment, which was given in Figure 2.2, provides the capability for run-time coupling of AVL VSM vehicles and MATLAB Simulink chassis control system models. AVL VSM is a complete vehicle dynamics simulation software developed by AVL. The primary purpose of the AVL VSM software is to simulate the longitudinal and lateral dynamics behaviour of the vehicle. In AVL VSM, the user can model the following subsystems in detail: vehicle geometry and chassis, suspension kinematics of each axle, steering kinematics, tyres, spring, dampers and anti-roll bars, bump-stop and rebound-stop, drivetrain, engine/electric system, chassis controller co-simulation with Simulink.

Methodically, first, the sign conventions and coordinate system were given. Then, each subsystem was expressed in detail in separate sub-sections. The ISO vehicle coordinate system, used as default in AVL VSM is shown in Figure 3.1.

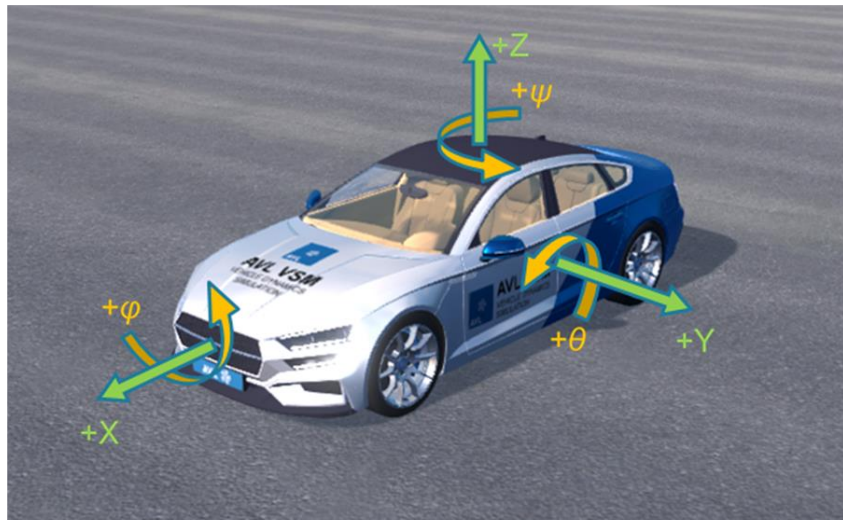


Figure 3.1: Default coordinate system used in AVL VSM.

3.1 Vehicle Geometry and Chassis

The vehicle geometry subsystem defines geometrically basic static dimensions, mass, weight distribution and inertia values. The vehicle parameters used in this study belong to a sedan car and are defined in Table 3.1. Detailed information of the electric system is given in the related subsystem section.

Table 3.1: Vehicle parameters.

Model Description	Symbol	Value	Units
Total vehicle mass	m	3000	kg
Roll moment of inertia	I_{xx}	1300	kg.m ²
Pitch moment of inertia	I_{yy}	4700	kg.m ²
Yaw moment of inertia	I_{zz}	5000	kg.m ²
Wheel radius	r_w	0.39	m
Wheelbase	l	3.06	m
Front axis to CoG distance	a	1.40	m
Rear axis to CoG distance	b	1.66	m
CoG Height above ground	h	0.6	m
Wheel track of the vehicle	d	1.72	m
Front EM maximum torque	$T_{max,F}$	420	N.m
Rear EM's maximum torque	$T_{max,R}$	420	N.m
Front gearbox ratio	τ_F	9.667	-
Rear gearbox ratios	τ_R	9.667	-
Front differential gear ratio	τ_d	1	-
1 st Aero resistance coefficient	A_0	300	N
2 nd Aero resistance coefficient	B_0	-5	N/kph
3 rd Aero resistance coefficient	C_0	0.1	N/kph ²

The aerodynamics parameters setup also can be parameterized in the chassis subsystem. The aerodynamic drag force is applied at the tyre contact patch in the AVL VSM, and a simple driving resistance equation is used. The drag force was calculated by Equation 17, and the related parameters are defined in Table 3.1. The coefficients of the drag force equation can be correlated via the coast-down test scenario.

$$F_d = A_0 + B_0 v_x + C_0 v_x^2 \quad (17)$$

3.2 Suspension and Steering Kinematics

Suspension kinematics & compliance and steering system parameters are highly effective on vehicle handling characteristics. The suspension adjustment is split into two setups, one for the front and one for the rear. Suspension kinematics, motion ratios

for springs and dampers and steering ratio/map were modelled for the front and rear static setups. As mentioned before, AVL VSM is table based non-linear vehicle model, and it uses look-up tables for each kinematic equation/relation in the suspension. Thus, different maps were entered describing the suspension kinematics. Every kinematic map is related to the vertical position of the wheel travel. A multibody dynamics software usually generates these necessary look-up tables for kinematics. As a multibody dynamics simulation tool, MSC Adams Car is also used in the industry and academy in vehicle dynamics modelling with high accuracy [13]. MSC Adams Car model was also prepared to create kinematic maps of both the front and rear suspension. AVL VSM has used the look-up tables of kinematics that are exported from MSC Adams Car. The following kinematic maps were exported for both the front and rear suspension:

- Steering Map (Road Wheel Angle vs. Hand Wheel Angle vs. Wheel Travel)
- Delta Camber vs. Wheel Travel vs. Steer Angle
- Delta Track vs. Wheel Travel
- Roll Centre Height vs. Wheel Travel
- Instantaneous Centre Height vs. Wheel Travel
- Anti-Lift vs. Wheel Travel
- Anti-Dive vs. Wheel Travel
- Caster vs. Wheel Travel
- Kingpin Inclination vs. Static Camber
- Scrub Radius vs. Static Camber

All maps were taken directly from MSC Adams Car suspension analyses and used in the AVL VSM suspension interface.

Similarly, all spring, damper and anti-roll bar details and characteristic curves were taken from the MSC Adams Car model. Since the purpose of this thesis is chassis controller development and evaluation, the author preferred to give something other than the details of spring/damper/ARB curves.

3.3 Tyre Modelling

The behaviour of the tyres has a significant effect on the overall behaviour of the vehicle, including its handling, stability, and braking performance, since the tyres are the only point of contact between the vehicle and the road. Thus, using a proper tyre model is essential in vehicle dynamics simulation. This study uses a semi-empirical Magic Formula (5.2) tyre model [47]. The Magic Formula can simulate the behaviour of a tyre's longitudinal and lateral forces and moments with combined slip conditions. The general form of the magic formula output holds for given values of vertical load and camber angle for pure slip condition:

$$y = D \sin[\text{Carctan}\{Bx - E[Bx - \arctan Bx]\}] \quad (18)$$

with,

$$Y(X) = y(x) + S_V \quad (19)$$

$$x = X + S_H \quad (20)$$

where, Y can be either output variable: longitudinal tyre force F_x , lateral tyre force F_y and, aligning moment M_z . X can be either input variable: lateral (side) slip α and longitudinal slip κ . B , C , D and E are stiffness factor, shape factor, peak value, and curvature factor, respectively. S_H and S_V are horizontal and vertical shifts. Figure 3.2 shows the output curve of the magic formula.

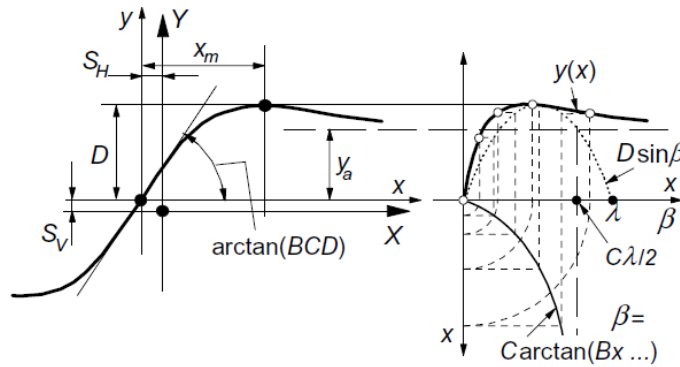


Figure 3.2: The Magic Formula curve parameters [47].

The pure longitudinal slip κ is the speed difference between the tyre's contact patch and the longitudinal component of the wheel centre [47]. The tyre's longitudinal force,

or its ability to accelerate or decelerate the vehicle, is highest when it operates at a pure longitudinal slip condition. Longitudinal force is generated during acceleration or braking. For example, in a wheel lock, κ becomes -1. The equation of longitudinal slip, where V_{sx} is the contact patch speed and V_x is the longitudinal wheel centre speed:

$$\kappa = \frac{V_{sx}}{V_x} \quad (21)$$

Pure lateral slip α is the speed difference between the contact patch and the road surface in the direction perpendicular to the vehicle's motion. The tyre's lateral force, or its ability to turn the vehicle, is the highest when the tyre is operating at a pure lateral slip condition. The equation of lateral slip, where V_{sy} is the lateral wheel speed, and V_x is the longitudinal wheel speed:

$$\tan \alpha = \frac{V_{sy}}{V_x} \quad (22)$$

In Figure 3.2, the left graph shows the lateral force-lateral slip behaviour, and the middle chart offers the longitudinal force-longitudinal slip behaviour. In the last diagram, the combined slip is represented. The magic formula covers the combined slip condition, which means it can simulate the tyre's behaviour when both longitudinal and lateral forces act on it simultaneously. Using the magic formula is the proper approach for this study, because the main purpose of the thesis is to investigate the effect of RWS and TVC systems, steering and traction of tyres, and combined effects are important.

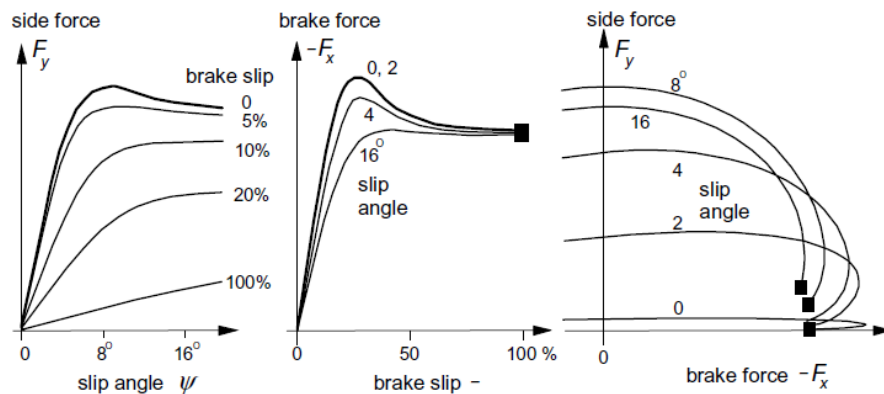


Figure 3.2: The Magic Formula pure and combined slip conditions [47].

As can be seen from Figure 3.1 and Figure 3.2, in the combined slip condition, the lateral force decreases as the longitudinal slip increases, and the longitudinal force

decreases as the lateral slip increases until the tyre reaches its maximum possible lateral or longitudinal force as represented in the last (third) graph in Figure 3.2. The reason for this, during the motion maximum resultant reaction force between the road and tyre contact point is limited by the friction ellipse. Also, Figure 3.3 has been added for clarity. The effect of longitudinal slip on lateral force and vice versa can be seen in both combined slip maps.

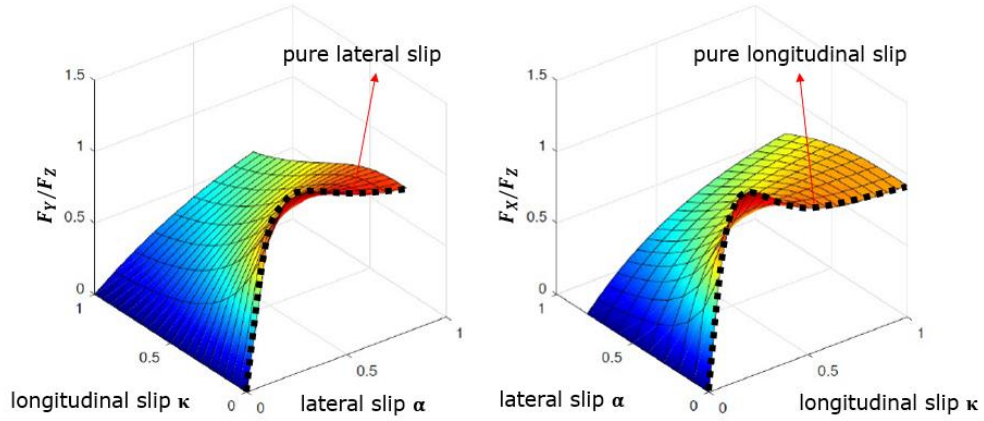


Figure 3.3: The Magic Formula combined slip maps [48].

Tyre saturation is another crucial point for the study. It refers to the point at which a tyre's ability to generate a longitudinal or lateral force reaches its maximum limit means the limits of friction ellipse shown in Figure 3.4. This occurs when the tyre operates at maximum slip angle or ratio. After the maximum tyre saturation point, the tyre's ability to generate force does not increase, regardless of any other slip angle or slip ratio increase. The friction ellipse represents the tyre's limits for generating longitudinal and lateral forces. For example, the red point in Figure 3.4 is where the tyre is fully saturated and has lost its ability to generate longitudinal and lateral forces since it is at the maximum limits of the friction circle. Thus, the tyre saturation is 100% at that point.

On the other hand, the green end with 20% tyre saturation shows that the tyre still can generate longitudinal or lateral force. If a tyre exceeds its saturation point on the friction ellipse, the tyre slips, which can lead to a loss of traction and stability. This can occur in high-speed driving, sharp turns, or sudden braking.

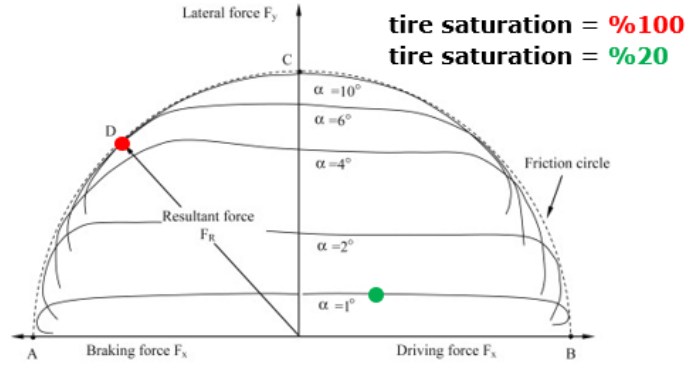


Figure 3.4: Friction ellipse and tyre saturation points [49].

3.4 Drivetrain and Electric System

An all-wheel-drive electric vehicle with three independent electric motors was used in this study. The drivetrain architecture is shown in Figure 3.5. The vehicle model has three electric motors, and the front electric motor is connected to the wheels via a differential. The front differential gear ratio was defined in Table 3.1.

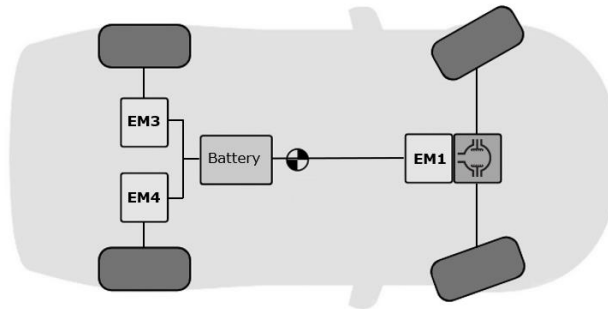


Figure 3.5: Drivetrain and electric system architecture.

All electric motors are identical in this vehicle. The torque characteristic over speed [rpm] can be entered for the motor and generator separately. The maximum torque characteristic of each engine is also shown in Figure 3.6. In addition, a pedal map is used to define the state of each electric motor at different motor speeds and accelerator pedal positions. The electric motor state describes how much torque the motor should deliver. For example, a state of 50 % means 50 % of the maximum available torque at this motor speed (defined in the electric machine setup), as can be seen in Figure 3.6.

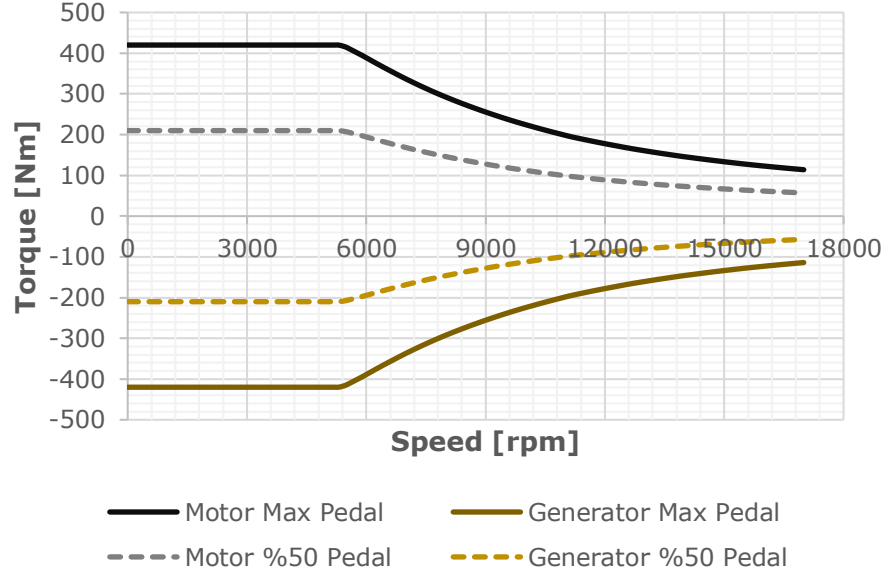


Figure 3.6: Electric motors torque-speed characteristics.

Similarly, all battery and energy storage details and characteristic curves were modelled in AVL VSM with look-up tables. Since the purpose of this thesis is chassis controller development and evaluation, the details of the battery and cell data of the electric system are not included.

3.5 Chassis Controller Integration and MIL Environment

This study used a co-simulation environment with vehicle/plant model, driver, and manoeuvre, RWS and TVC control models and optimization tools and scripts to simulate the vehicle and integrated chassis lateral controller. Co-simulation technique provides the capability for run-time coupling of AVL VSM and MATLAB Simulink models. The co-simulation and model in the loop (MIL) environment used can be seen in Figure 3.7.

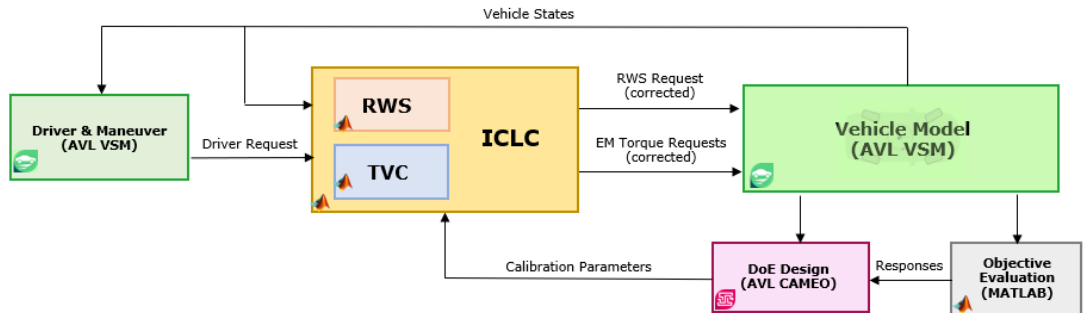


Figure 3.7: Co-Simulation environment.

Driver and manoeuvre are defined in AVL VSM. Driver demands (e.g., accelerator pedal position, brake pedal position, steering wheel angle) as well as manoeuvre information (e.g., track path and road friction coefficient) are inputs to the RWS and TVC controller logics that are generated by AVL VSM. Vehicle states (e.g., velocity, lateral acceleration, yaw rate) are provided by AVL VSM for RWS and TVC logic. Activation and mode selection parameters of RWS and TVC are also directly taken from AVL VSM. The integrated chassis lateral controller (RWS and TVC) requests corrected rear-wheel steering and EM torques to the vehicle model. The simulation software AVL VSM also offers multiple simulation features called to run in sequence. AVL VSM sequence runs were used to perform a variation on specific parameters or even whole parameter sets or setups. The calibration/optimization parameters of the RWS and TVC systems were varied in the design of the experiment (DoE) tool AVL CAMEO. These different optimization parameters were run with AVL VSM run in sequence capability. AVL VSM - AVL CAMEO - MATLAB Simulink toolchain provides both simulations of MIL systems and calibration of RWS, TVC and ICLC systems. The details of chassis controllers and optimization toolchain are expressed in the next section.

4. CONTROL SYSTEMS MODELING AND OPTIMIZATION

In this chapter, firstly, RWS controller modelling, parametrization and optimization were studied. Then, TVC controller modelling, parametrization and optimization were explained in detail. Thus, both RWS and TVC systems will be examined in detail, and all necessary studies will be expressed to ensure that the vehicle has optimum RWS and TVC chassis control systems. As a result, the vehicle with optimum RWS and TVC functions was obtained. Both RWS and TVC systems were modelled in MATLAB Simulink and connected to the vehicle model, as expressed in section 2. This study used both RWS and TVC systems with feedforward and feedback systems. Feedforward control is a control strategy to predict the effect of a disturbance on the control design and take action to counteract that disturbance before it affects the output. Feedback control, on the other hand, uses the outcome of a system to adjust the input to the system to maintain or improve the outcome. Since there were feedback mechanisms in both systems, the relationships between each other were also examined.

4.1 RWS Controller Design

AVL VSM and MATLAB Simulink co-simulation environment allows the user to implement the RWS control system. To implement RWS, three signals need to be changed: external rear wheel angle allowance, rear left wheel angle and rear right wheel angle. Figure 4.1 shows the implementation of RWS in Simulink.

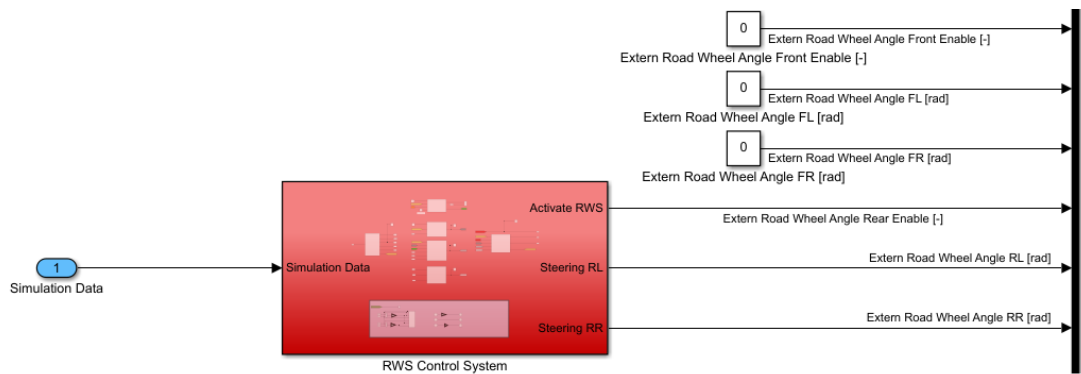


Figure 4.1: RWS system implementation in Simulink.

Park et al. proposed the RWS control algorithm, which includes feedforward and feedback parts [50]. With this algorithm compared to other studies, although this proposed algorithm does not use any tyre-road information, the RWS system can effectively improve vehicle manoeuvrability and stability. RWS system proposed by Park et al. was tested with a limited vehicle model (bicycle model). In this study, on the other hand, the proposed RWS system was used, tested, and optimized with a non-linear vehicle model. The general scheme of the proposed RWS is shown in Figure 4.2. The RWS system function has feedforward and feedback control parts with steady-state and transient control inputs. The rear wheels mean steering angle δ_r was controlled in the RWS control system. ICLC coordinator part takes the rear wheels mean steering angle δ_r and multiplies it with ICLC RWS bias factor η_{RWS} and sends the corrected rear wheels mean steering angle $\delta_{r,c}$ to the kinematic coordinator. In the last part of the RWS system, corrected rear wheels mean steering angle was distributed to the left and right wheels according to Ackermann kinematics.

So, the RWS system consist of six main parts:

- RWS Input Generator
- RWS Steady-State
- RWS Feedforward Transient
- RWS Feedback Transient
- ICLC Coordinator
- Ackermann Correction

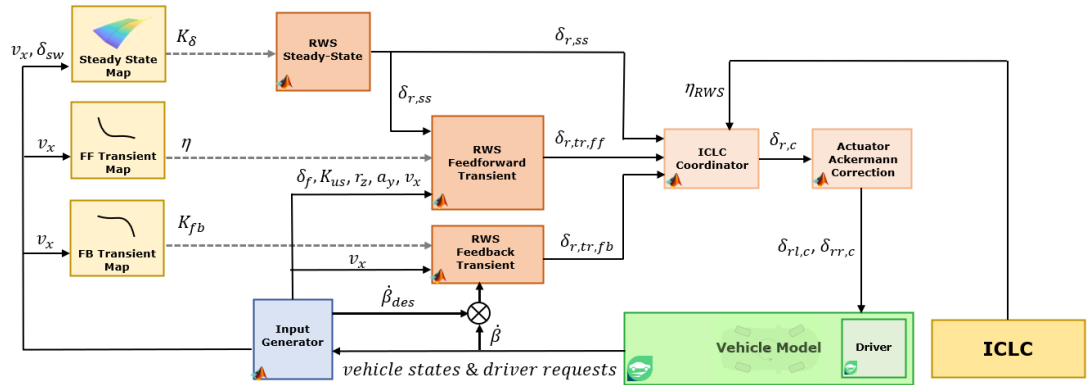


Figure 4.2: RWS system scheme.

4.1.1 RWS input generator

In the input generator block, the required signals were taken from the simulation data and the driver. Inputs of the RWS systems are:

- Vehicle side slip angle β [rad]
- Yaw rate r_z [rad/s]
- Vehicle speed v_x [m/s]
- Lateral acceleration a_y [m/s²]
- Steering wheel angle δ_{sw} [deg]
- Front tyres mean steering angle δ_r [rad]

As mentioned earlier in section 3.2, the mean steering angle, and the steering map (road wheel angle vs hand wheel angle vs wheel travel) were used to generate the front tyres' mean steering angle. The map converts driver steering wheel angles to left and right wheel angles using the steering ratio. The front tyres' mean steering angle is the average of the left and right tyres, as shown in Figure 4.3.

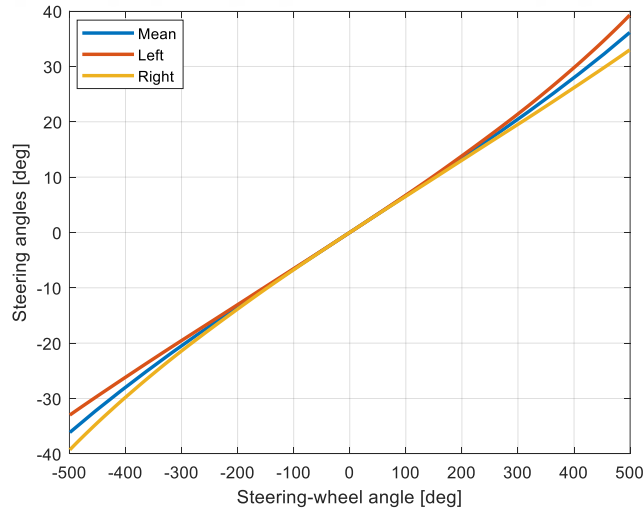


Figure 4.3: Steering-wheel angle vs front tyre steering angles.

Linear understeer gradient K_{us} can be defined as a function of the front steering angle and lateral acceleration according to different driver modes. Also, a more straightforward way of determining the understeer gradient is taking it as zero. If the desired behaviour of the vehicle is assumed to be neutral, then the linear understeer gradient is equal to 0. The formulation of K_{us} is expressed in the next section.

4.1.2 RWS steady-state

The equations of motion of the bicycle model with RWS were expressed in Chapter 2. From equations (3) and (9) with the condition of steady state ($\dot{v}_y=0$, $\dot{r}_z=0$), the vehicle cornering kinematics can be expressed as:

$$\delta_f - \delta_r = \frac{r_z(a+b)}{v_x} + \alpha_f - \alpha_r \quad (23)$$

$$\delta_f - \delta_r = \frac{l}{R} + \alpha_f - \alpha_r \quad (24)$$

Where l is wheelbase and sum of a and b terms, R is the radius of curvature ($R = v_x/r_z$). If the expressions of front and rear mean tyre slip angles α_f and α_r were added from equations (5) and (6), equation (24) becomes:

$$\delta_f - \delta_r = \frac{l}{R} + \frac{m(C_f a - C_r b)}{l C_f C_r} a_y \quad (25)$$

Where K_{us} is the understeer gradient:

$$K_{us} = \frac{m(C_f a - C_r b)}{l C_f C_r} \quad (26)$$

Vehicle cornering kinematics can be written as:

$$\delta_f - \delta_r = \frac{l}{R} + K_{us} a_y \quad (27)$$

Generally, the steering angle of a conventional vehicle ($\delta_r = 0$) consists of two parts, kinematic and dynamic [51]. Kinematic steering δ_{kin} , also known as Ackermann steering, is calculated by steering kinematics and comes from ideal steering geometry. On the other hand, the dynamic steering δ_{dyn} due to centrifugal force acting on the centre of gravity of the vehicle and comes from tyre lateral slip angles.

$$\delta = \delta_{kin} + \delta_{dyn} \quad (28)$$

$$\delta_{kin} = \frac{l}{R} \quad (29)$$

$$\delta_{dyn} = K_{us} a_y \quad (30)$$

Equation (28) can be written with steady-state and transient parts as:

$$\delta = \delta_{ss} + \delta_{tr} \quad (31)$$

Equation (10) can be resolved for steady-state condition ($\dot{v}_y=0$, $\dot{r}_z=0$) to find the steady-state part of the rear wheels mean steering angle δ_r :

$$\frac{1}{v_x} \begin{bmatrix} C_f + C_r & aC_f - bC_r - mv_x^2 \\ aC_f - bC_r & a^2C_f + b^2C_r \end{bmatrix} \begin{bmatrix} v_y \\ r_z \end{bmatrix} = \begin{bmatrix} C_f & C_r \\ aC_f & -bC_r \end{bmatrix} \begin{bmatrix} \delta_f \\ \delta_r \end{bmatrix} \quad (32)$$

The most straightforward approach for solving the state equations is to make the rear wheels' mean steering angle δ_r proportional to the front wheels mean angle δ_f [11].

Thus, the steady-state part of the rear wheels mean steering angle is as follows:

$$\delta_{r,ss} = K_\delta \delta_f \quad (33)$$

Where K_δ is the steady-state coefficient and can be a function of vehicle speed and front steering angle. The proportional gain (K_δ) of the steady state part is a tuneable parameter and is a 2D map in modelling. Negative K_δ means that the rear wheels are steered inversely according to the front wheels. A positive K_δ means that the rear wheels are steered in direct proportion to the front wheels and vice versa. The initial K_δ map is shown in Figure 4.4.

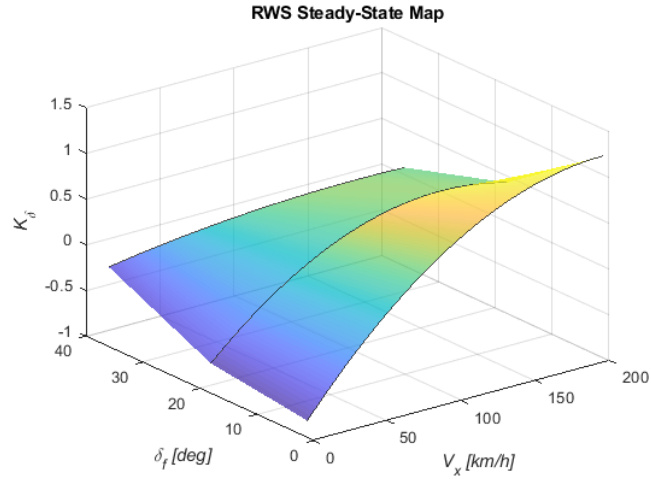


Figure 4.4: RWS steady-state coefficient 2D map.

Second-order polynomial equations were used to parameterize the RWS steady-state coefficient map. The equation is written as:

$$c_i x^2 + b_i x + a_i = 0 \quad (34)$$

Where a, b and c are coefficients of real numbers. Three different second-order polynomial equations were used to create the structure of the 2D map in MATLAB. The map generation codes of the RWS are shown in Appendix A. The main idea is to parametrize the map velocity-scheduled equations that placed a steering angle value and combine them to generate all maps. Defined initial second-order polynomial functions are shown in Figure 4.5.

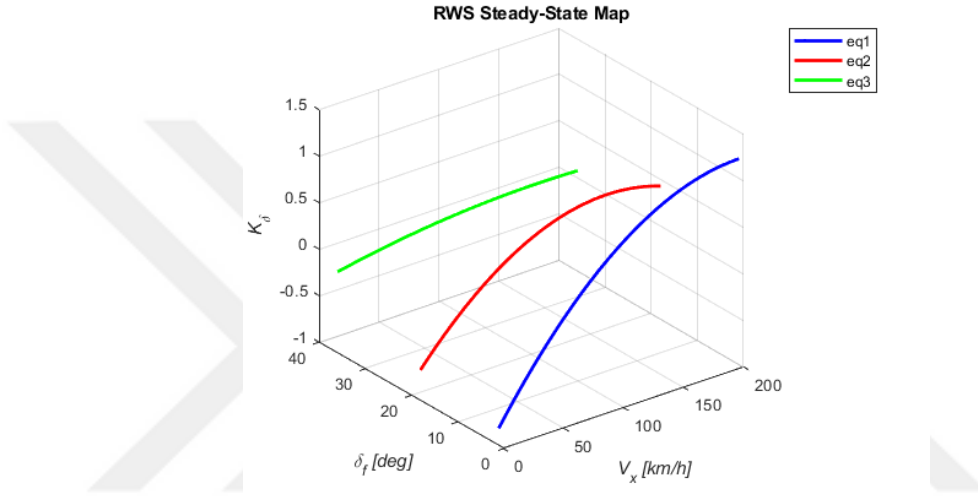


Figure 4.5: RWS steady-state coefficient 2D map equations.

Equations 32 and 33 were solved for reducing control inputs to one for generating the initial curve of the first equation of the RWS steady-state coefficient map:

$$\frac{1}{v_x} \begin{bmatrix} C_f + C_r & aC_f - bC_r - mv_x^2 \\ aC_f - bC_r & a^2C_f + b^2C_r \end{bmatrix} \begin{bmatrix} v_y \\ r_z \end{bmatrix} = \begin{bmatrix} C_f & K_\delta C_r \\ aC_f & -bK_\delta C_r \end{bmatrix} \delta_f \quad (35)$$

Equations 36 can be obtained if Equation 35 is solved for K_δ by eliminating r_z from the equation and setting v_y to zero. The main idea behind this approach is to achieve minimum side slip angle during cornering. This target can be achieved by setting the side slip angles to zero when solving the steady-state equation. This strategy can be called “zero-side slip” in the literature [11].

$$K_\delta = \frac{bL + m \frac{a}{C_r} v_x^2}{-aL + m \frac{b}{C_f} v_x^2} \quad (36)$$

If Equation 36 is solved according to vehicle speed it can be used as the first equation to generate the 2D map. The first equation coefficients are found by applying regression to the zero-sideslip approach curve, as shown in Figure 4.6. Later, in the optimization process, the first equation coefficients are found by numerical optimization techniques. The second and third equations are solved with random second-order polynomial equation coefficients. The second equation is placed with steering blending angle y_{ble} parameter to 2D map that is shown in the Figure 4.5. The steering blending angle y_{ble} is a tuneable parameter.

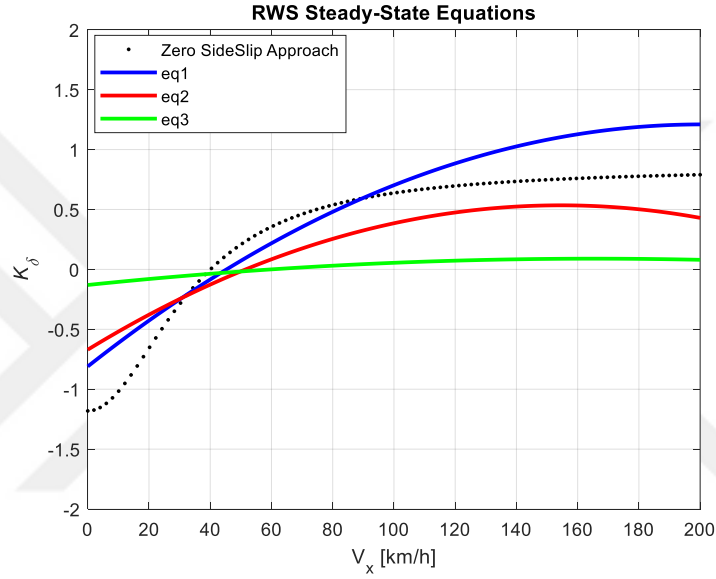


Figure 4.6: RWS steady-state coefficient map curves.

Initial second-order polynomial coefficients of different equations are shown in Equations 38,39,40. The values were shown in the last section of this chapter in Table 4.1.

$$eq_{RWS,ss,1} = c_1x^2 + b_1x + a_1 \quad (38)$$

$$eq_{RWS,ss,2} = c_2x^2 + b_2x + a_2 \quad (39)$$

$$eq_{RWS,ss,3} = c_3x^2 + b_3x + a_3 \quad (40)$$

MATLAB's "smoothdata" function was used in the last stage of map generation. After generating with the combination of first, second and third curves steady-state map, the "smoothdata" function was used to prevent the formation of sharp transitions. As

seen in Figure 4.7 finalized map with red was generated after smoothing the previous map was generated by second-order polynomial equations. RWS steady-state second and third equations coefficient parameters and steering blending angle y_{ble} were optimized in the next section.

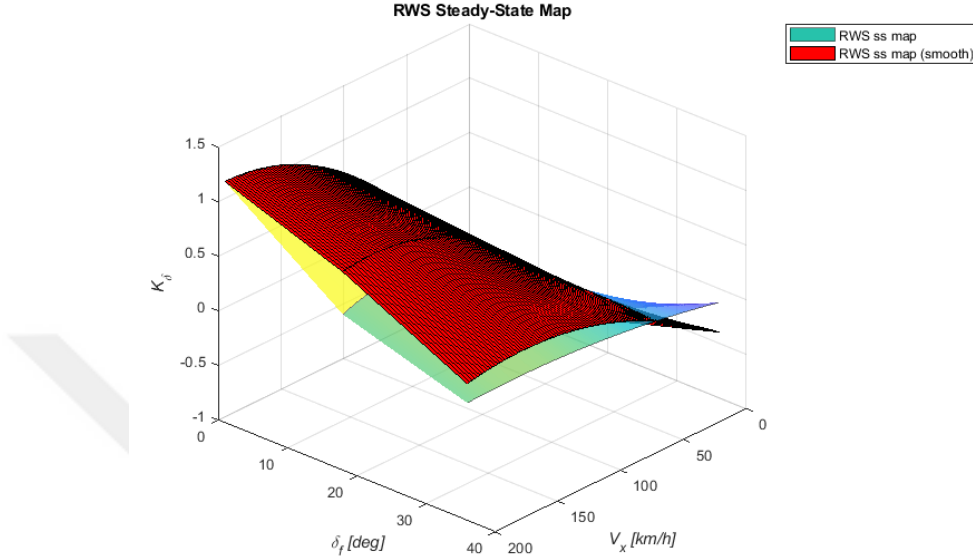


Figure 4.7: RWS steady-state coefficient 2D map, smooth.

4.1.3 RWS feedforward transient

The goal of the feedforward transient part is to improve the transient and dynamic responses. The first-order lag with a gain is added to the RWS transient part to enhance transient lateral performance in the Laplace domain with design parameter η as described in Equation 41. The parameter η is coefficient of the time constant of the lag. The transient feedforward part of the rear wheels' steering angle, as shown in Equation 42.

$$\delta_{r, tr, ff}(s) = \underbrace{\frac{1}{\eta\tau s + 1}}_{\text{delay}} \frac{v_x}{L} \left[1 - K_\delta - K_{us} \frac{a_y(s)}{\delta_f(s)} \right] \quad (41)$$

$$\delta_{r, tr, ff} = \left(\frac{1}{\eta} - 1 \right) \left\{ \delta_f (K_\delta - 1) + K_{us} a_y + \frac{L}{v_x} r_z \right\} \quad (42)$$

The transient response of the vehicle can be adjusted by changing the tuneable parameter η . When η is equal to 1, the system behaves the same as steady-state behaviour. If η is greater than 1, the yaw rate overshoot can be reduced, but the vehicle becomes less responsive to dynamic manoeuvres. In the opposite case, if η is lower

than 1, the system becomes more responsive, but overshoots can be increased, which is undesirable. Since the parameter η must be selected carefully, it is parametrized as a velocity-scheduled second-order polynomial equation as is done in the steady-state part. Second-order polynomial coefficients of the parameter η were shown in Equation 41 where x is the vehicle velocity.

$$\eta = c_4x^2 + b_4x + a_4 \quad (43)$$

The curve of parameter η is shown in Figure 4.8 with initial random parameters.

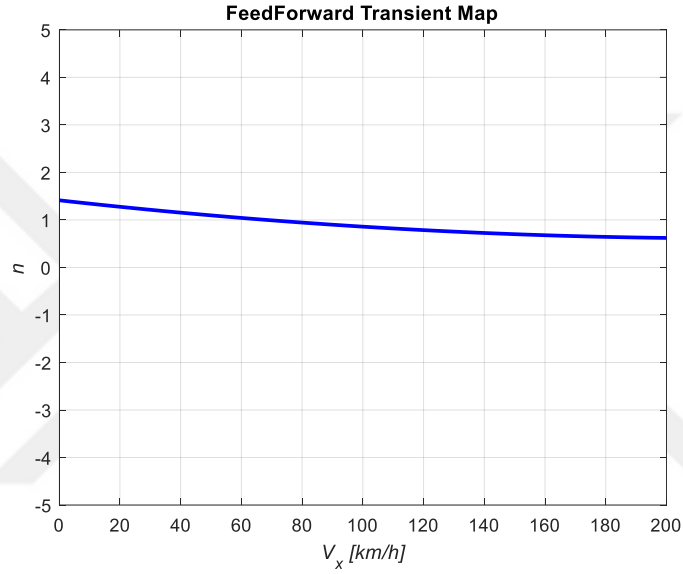


Figure 4.8: RWS feedforward transient coefficient 1D map.

4.1.4 RWS feedback transient

The transient feedback part aims to enhance transient behaviour by preventing high vehicle side-slip rates. Thus, the first-time derivative of the vehicle side slip as the feedback part of the RWS given in Equation 44 is added to the system:

$$\delta_{r, tr, fb} = -K_{fb}v_x(\dot{\beta} - \dot{\beta}_{des}) \quad (44)$$

Similarly, in the feedback part of the RWS, the feedback gain K_{fb} is a tunable parameter. The desired derivative of the vehicle side slip angle $\dot{\beta}_{des}$ was taken as zero in this study since the main aim of this part is to prevent excessive slip rates. K_{fb} was parametrized as velocity-scheduled second-order polynomial equation as in the other sections. The quadratic coefficients of the parameter K_{fb} are shown in Equation 45.

$$K_{fb} = (c_5 x^2 + b_5 x + a_5)/100 \quad (45)$$

The graph of feedback gain K_{fb} vs. vehicle speed is shown in Figure 4.9 with the initial parameters.

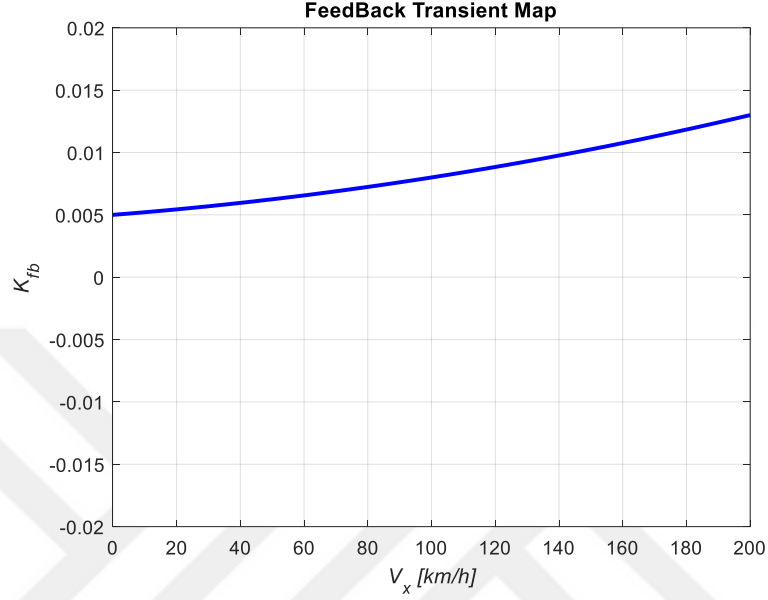


Figure 4.9: RWS feedback transient coefficient 1D map.

Finally, the general form of RWS can be obtained as follows:

$$\delta_r = \underbrace{K_\delta \delta_f}_{\text{Steady-State}} + \underbrace{\left(\frac{1}{\eta} - 1\right) \left\{ \delta_f (K_\delta - 1) + K_{us} a_y + \frac{L}{v_x} r_z \right\}}_{\text{Feedforward for Transient}} - \underbrace{K_{fb} v_x (\dot{\beta} - \dot{\beta}_{des})}_{\text{Feedback of Transient}} \quad (46)$$

The initial values of the design parameters of all RWS systems are shown in Table 4.1. Basically, the parameters $a_1, b_1, c_1, a_2, b_2, c_2, a_3, b_3, c_3$ and y_{ble} are used to generate velocity and steering angle scheduled K_δ map. Other parameters a_4, b_4 and c_4 are used to generate a velocity scheduled map of the feedforward tuneable parameter η . Similarly, parameters a_5, b_5 and c_5 are used to generate a velocity scheduled map of feedback gain K_{fb} . The limits of the design parameters were selected via open-loop tests by simulating the vehicle with ramp steer manoeuvres. Since the limits of the K_δ already were set by zero-side slip approach, no limit was specified for this parameter. The maximum and minimum limits of η were selected as 1.5 and 0.5 respectively

according to the open-loop test results. On the other hand, maximum and minimum limits of K_{fb} were selected as 0.02 and 0, respectively. The control engineer should check these parameters if the RWS is implemented later in any other vehicle model.

Table 4.1: RWS parametric variables – initial values.

RWS Parametric Variables	RWS Part	Initial Values
a_1	Steady-State	-0.81
b_1	Steady-State	4.03
c_1	Steady-State	-2.01
a_2	Steady-State	-0.67
b_2	Steady-State	3.12
c_2	Steady-State	-2.02
a_3	Steady-State	-0.13
b_3	Steady-State	0.53
c_3	Steady-State	-0.32
y_{ble}	Steady-State	18
a_4	FF Transient	1.413
b_4	FF Transient	-1.426
c_4	FF Transient	0.634
a_5	FB Transient	0.5
b_5	FB Transient	0.4
c_5	FB Transient	0.4

4.1.5 RWS ICLC coordinator

In the ICLC coordinator, steady-state, feedforward transients and feedback transients are collected and distributed according to the RWS mode selected by the user/driver. In the 1st mode of RWS, only the steady-state part was taken as an input. In the 2nd mode, both steady-state and feedforward transient parts were used. Lastly, in the 3rd mode, all subparts of the RWS system were used, as shown in equation 46. The RWS control modes are shown in Table 4.2.

Table 4.2: RWS control modes.

Mode Selection	RWS Parts
0	Off
1	Steady-State
2	Steady-State + FF Transient
3	Steady-State + FF Transient + Steady-State + FB Transient

After combining all parts of the RWS system with mode 3, the rear wheel mean angle can basically be expressed as follows:

$$\delta_r = \delta_{r,ss} + \delta_{r, tr,ff} + \delta_{r, tr,fb} \quad (47)$$

The vehicle motion control system corrects the rear wheel mean angle by adjusting its bias through the RWS weighting factor η_{RWS} . Parameterization and modelling details were expressed in the ICLC section. The main idea is to multiply the rear wheel mean angle with weighting factor η_{RWS} to get the corrected rear wheel mean angle $\delta_{r,c}$ that will be sent to the last part of the RWS controller.

$$\delta_{r,c} = \eta_{RWS} \delta_r \quad (48)$$

4.1.6 RWS actuator and kinematics correction

In the last part, the corrected rear wheel mean angle $\delta_{r,c}$ was distributed to the left and right wheels according to Ackermann's kinematics. In this study, the same kinematic correction parameters were used as the front ones that were shown in Figure 30 previously. The maximum selected rear wheels steering angle is 3.5 degrees. This is a very critical parameter for RWS design considering the boundary conditions of the system. Thus, in the RWS design, the left corrected wheel angle maximum $\delta_{rl,c,max}$ and right corrected wheel angle maximum $\delta_{rr,c,max}$ were fixed to ± 3.5 degrees, and finally, the left corrected wheel angle $\delta_{rl,c}$ and the right corrected wheel angle $\delta_{rr,c}$ eventually were sent to the vehicle model data as RWS outputs as shown in Figure 4.2. The final left and right wheel steering outputs were multiplied by a first-order filter with $\tau = 0.2$ sec to prevent excessive RWS control action for high-frequency inputs in the actuator block. The actuator plays an essential role in the application since the RWS output needs to work in the whole frequency range relevant to the vehicle lateral dynamics without significant phase lags. Figure 4.10 shows how this part is modelled in Simulink.

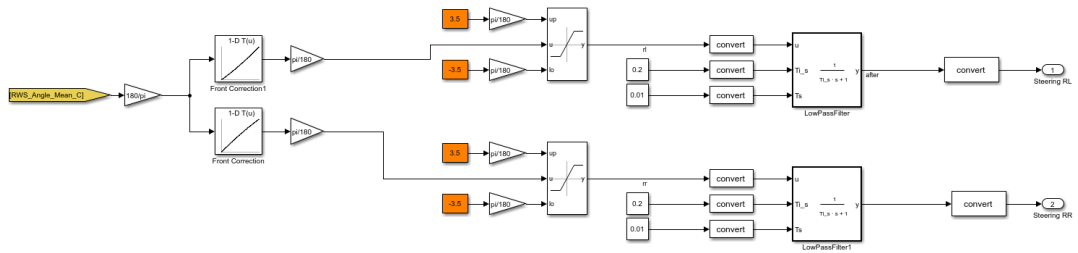


Figure 4.10: RWS Actuator and Ackermann Correction in Simulink.

4.2 RWS Optimization and MIL Evaluation

In this section, the proposed optimization methodology for RWS is expressed. A similar procedure was used to optimize TVC and ICLC parameters. Optimization of RWS was done using the DoE tool of AVL CAMEO, vehicle model of AVL VSM, objective evaluation tool of MATLAB and RWS control model in MATLAB/Simulink. Design of Experiments (DoE) is a statistical methodology used to determine the effect of certain factors or variables on a process or system and to optimize the process or system by identifying the settings of these factors that produce the best results. In the context of the RWS system, DoE can be used to optimize the performance of the system by identifying the parametric variables that affect the system's behaviour and determining the optimal settings for those variables. The optimized RWS system was tested with different test scenarios, and the results were shared. Different RWS modes were compared to show each mode's performance in terms of vehicle handling. The general order of the methodology is shown in Figure 4.11. Vehicle modelling, RWS control algorithm and parametrization of optimization parameters has already been expressed. To optimize the RWS system, the following steps need to be done in order:

- Defining test cases and their performance indicators.
- Scripting of objective evaluations in MATLAB to calculate the performance indicators and responses.
- Creating DoE design according to parametric variables in AVL CAMEO and defining general objective function space.
- Running all test cases in sequence with the parameters created in DoE design using AVL VSM – AVL CAMEO.
- Evaluation of results and detection of possible adjustments in the optimization according to the different drive modes.

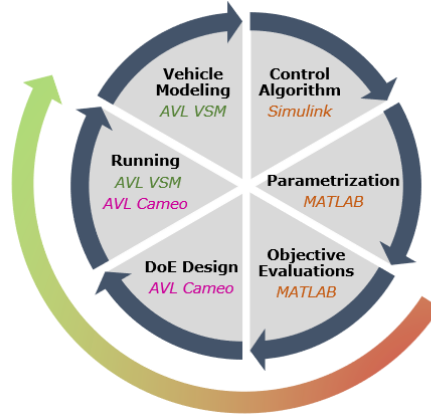


Figure 4.11: The optimization methodology.

4.2.1 Objective evaluations and test cases

Manoeuvres modelled in AVL VSM to optimize the RWS control system in terms of vehicle handling, are shown in Table 4.3. The objective functions of the test scenarios were generated via key performance indicators (KPI) of each manoeuvre, considering the main objectives of each manoeuvre. KPIs help achieve the objectives as they represent a type of performance measurement and objective functions of optimization. The methodology aimed to optimize RWS in all test cases with high- μ conditions. In general applications of chassis control software development, the control system observes or estimates different tyre-road friction coefficients and switches all calibration and control parameters according to the friction level [43]. However, in this study, only high friction level cases were considered since the same optimization methodology can be applied on medium and low μ conditions. All objective metrics were calculated after each AVL VSM simulation via a post-processing script written in MATLAB.

Table 4.3: RWS optimization aims of test cases.

Manoeuvre	Speed [kph]	Main Aim
Ramp steer	50	Decrease K_{us}
Ramp steer	100	Increase K_{us}
Ramp steer	150	Increase K_{us}
Step steer	100	Minimize a_y overshoot
Double-Lane Change	55	Minimize β
Turning Radius	5	Decrease R

4.2.1.1 Ramp steer manoeuvre

The ramp steer manoeuvre is an open-loop vehicle dynamics test case used to evaluate handling and stability when a vehicle is steered by a ramp input. During the test, the vehicle is driven towards a ramp at constant speed and then steered with a ramp input. There were three different ramp steer manoeuvres defined by vehicle speeds of 50, 100 and 150 kph. The purpose of the RWS control system in a ramp steer manoeuvre is to improve the stability and handling of the vehicle during the manoeuvre. The RWS works by allowing the rear wheels to turn in the opposite direction of the front wheels at low speeds, which provides better manoeuvrability, turning ability and responsiveness, by reducing the turning radius of the vehicle. At higher speeds, the rear wheels turn in the same direction as the front wheels to improve stability during high-speed cornering. In conclusion, the RWS system's main aim is to decrease the linear understeer gradient at 50 kph, increase the linear understeer gradient at 100 and 150 kph vice versa. The linear understeer gradient K_{us} is a metric which can be defined as the derivative of the steering angle with respect to the chassis lateral acceleration. The increased and decreased K_{us} vehicle dynamic lateral behaviour is represented in Figure 4.12. In general, linear understeer gradient is measured between 0.5 and 4 m/s² lateral accelerations.

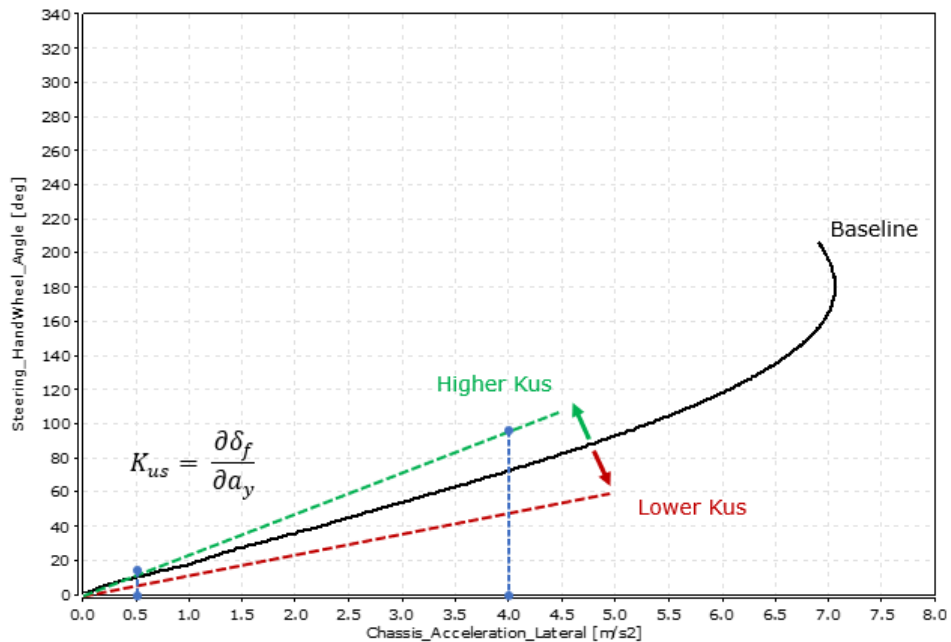


Figure 4.12: Ramp steer manoeuvre linear understeer gradient.

Oversteer behaviour is not considered to increase the efficiency of the estimated results of the optimization. Thus, the condition that the front tyre's side slip is smaller than the rear tyre's side slip angle was eliminated from the objective functions. The simplest form of the ramp steer manoeuvre objective functions for RWS optimizations is defined as follows:

$$\begin{aligned} f_{RWS,RS,50} &= \text{minimize } K_{us} \\ &\text{subject to } |\alpha_f| - |\alpha_r| < -0.1 \end{aligned} \quad (49)$$

$$\begin{aligned} f_{RWS,RS,100} &= \text{maximize } K_{us} \\ &\text{subject to } |\alpha_f| - |\alpha_r| < -0.1 \end{aligned} \quad (50)$$

$$\begin{aligned} f_{RWS,RS,150} &= \text{maximize } K_{us} \\ &\text{subject to } |\alpha_f| - |\alpha_r| < -0.1 \end{aligned} \quad (51)$$

4.2.1.2 Step steer manoeuvre

The step steer manoeuvre is an open-loop test case that aims to analyse the transient response of the vehicle. A fast-steering angle is given as input to the vehicle in short time with constant vehicle speed, and sufficient time is maintained to evaluate the vehicle's behaviour, as shown in Figure 4.13. The expectation from the RWS system is to decrease the overshoot and peak time of the step response.

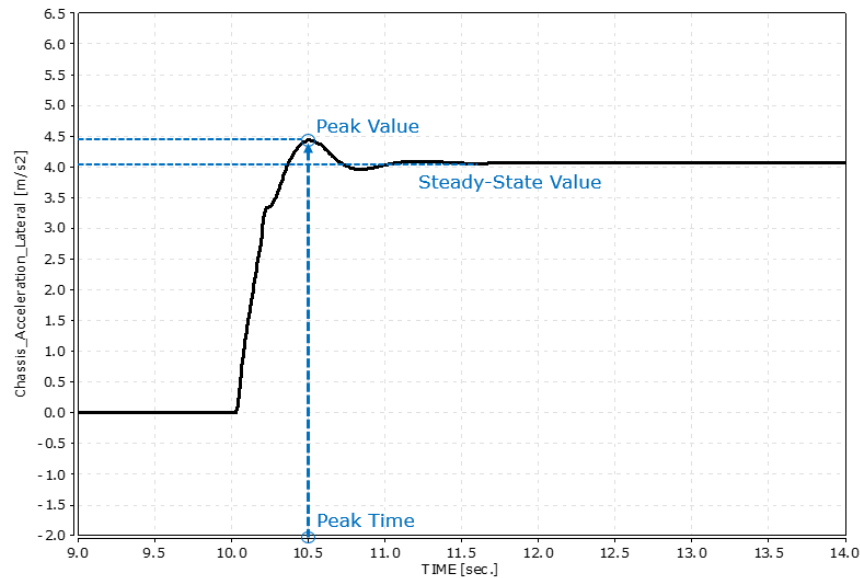


Figure 4.13: Step steer manoeuvre – lateral acceleration response.

In the step steer manoeuvre, the vehicle must also maintain a constant lateral acceleration of 4 m/s² during the final stage of the manoeuvre. Thus, the steering angles required to achieve 4 m/s² may depend on the use of different RWS modes. In the

evaluation section, although different steering inputs were used to analyse the step steer manoeuvre with different RWS modes, the difference between the peak and constant lateral acceleration value was defined as an objective function:

$$\begin{aligned} f_{RWS,SS} = & \text{minimize } a_{y,peak} - a_{y,steady-state} \\ & \text{subject to } 3 < a_{y,SS} < 5 \end{aligned} \quad (52)$$

4.2.1.3 Double lane change manoeuvre

The double lane change manoeuvre is a closed-loop test case defined by ISO standard 3888-2 and known as the “elk test” or “moose test” in the industry [53]. Maximum entry speed that guarantees to remain inside the track boundaries is checked in the test. The driver follows the defined path or track setup during the test, as represented in Figure 4.14. Thus, driver skill level can be highly crucial for this test. In AVL VSM, the saturation controller driver was chosen to use a more skilled driver for the DLC test case. The saturation-controlled driver takes tyre saturations into account while following the defined path. The entry speed was chosen as 55 kph by driving and trying the baseline vehicle on this manoeuvre. In general, an improvement in the entry velocity can be obtained as a performance metric. However, since every speed step is tried in the optimization procedure, the maximum side slip angle was selected as a performance indicator to simplify the optimization process. The maximum vehicle side slip angle is considered in double lane change evaluations with the same vehicle entry speeds for the baseline and RWS conditions [55].

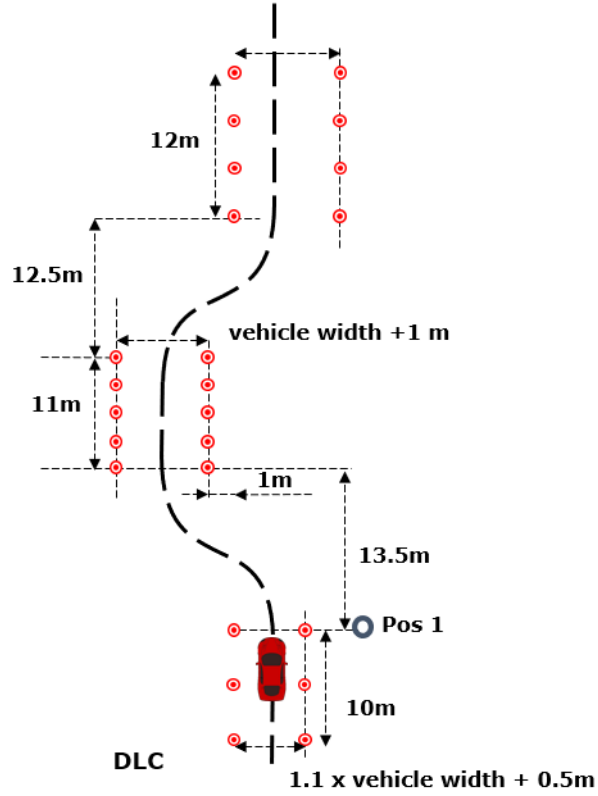


Figure 4.14: Double lane change - ISO 3888-2 test scheme.

The main purpose of the RWS system is to reduce the maximum side slip angle in this manoeuvre. Those with a maximum vehicle side slip angle greater than 45 degrees were excluded from the optimization evaluation.

$$\begin{aligned} f_{RWS,DLC} &= \text{minimize } \beta_{MAX} \\ \text{subject to } &-45 < \beta_{MAX} < 45 \end{aligned} \quad (53)$$

4.2.1.4 Turning radius manoeuvre

The turning radius is an open-loop test case that evaluates the vehicle's manoeuvrability. The vehicle is steered at maximum angle at very low speed, as represented in Figure 4.15.

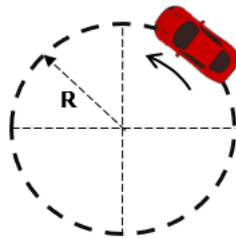


Figure 4.15: Turning radius manoeuvre.

The main purpose of RWS is to reduce the turning radius. The Rear wheels are expected to be steered at their maximum at low speeds and reduce the turning radius as much as possible. The objective metric was defined as the turning radius at full lock steering with a vehicle speed of 5 kph.

$$\begin{aligned} f_{RWS,TS} = & \text{minimize } R \\ & \text{subject to } 4 < R < 6 \end{aligned} \quad (54)$$

4.2.2 RWS mode evaluation

In this section, three different modes of RWS defined in Table 5 were evaluated with their initial parametric variables. Each mode of RWS was checked in the defined test cases. Then the RWS mode was decided, and the optimization was made with the selected mode.

4.2.2.1 RWS mode comparison - ramp steer

The three RWS modes were first evaluated in the ramp steer manoeuvre at 50 kph; since the ramp steer manoeuvre doesn't involve many transients, not much difference was expected between the different three modes. Figure 4.16 shows the lateral behaviour of the vehicle with different RWS modes and initial parametric variables.

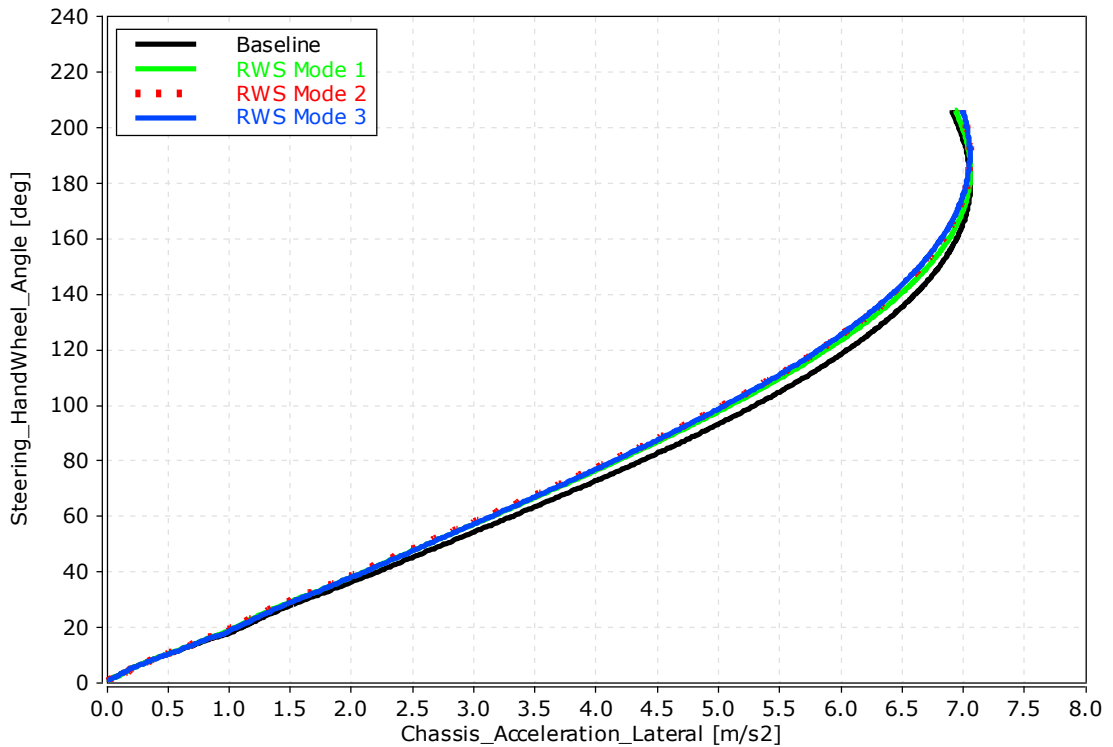


Figure 4.16: RWS modes - ramp steer - steering angle vs lateral acceleration.

From Figure 4.17, RWS mode 2 and RWS mode 3 gave similar results as there are not many transients in the manoeuvre. All RWS modes, as can be seen in Figure 4.6 forced the vehicle to understeer since the initially defined steady state RWS ratio was positive at that speed. Only at the end of the manoeuvre (after reaching $\sim 7 \text{ m/s}^2$) there is a slight difference between steady-state mode1 and transient included modes 2 and 3. The contribution of the steady-state and transient parts can be seen in Figure 4.18. In Figure 4.17 the transient feedforward part increased its effect, especially towards the end of the manoeuvre, but the transient feedback part compensated for it to prevent possible excessive side slip rates. A side slip phase plane was generated in Figure 4.18 to evaluate the stability behaviour clearer. As can be seen from Figure 4.18, the vehicle maintained its stability better with RWS mode 3 since a lower chassis slip angle and chassis slip rate were reached during the simulation. Thus, it can be said that RWS mode 3 performed best, although there was a slight difference.

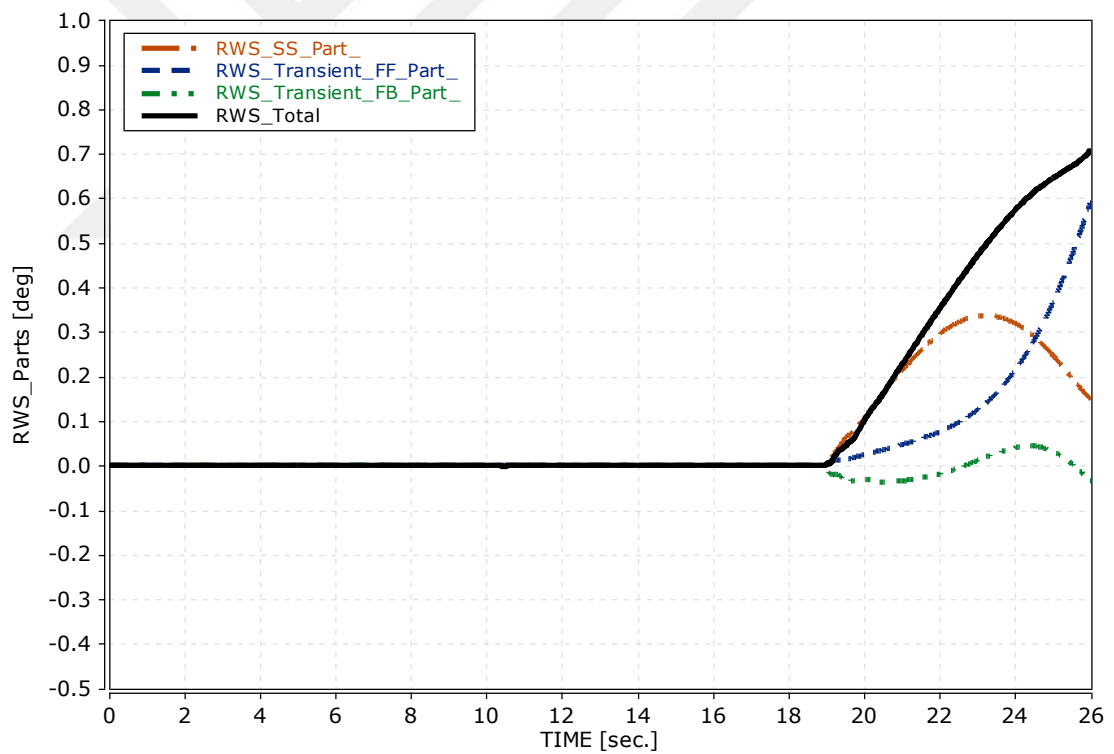


Figure 4.17: RWS modes - ramp steer - parts.

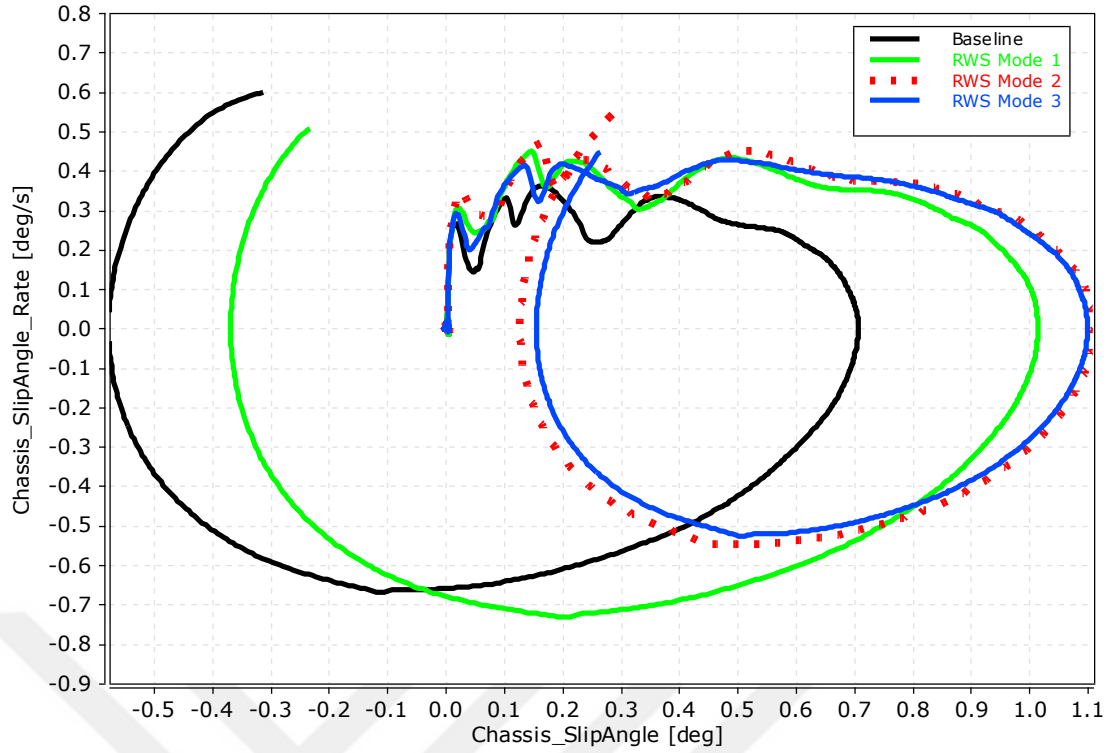


Figure 4.18: RWS modes - ramp steer - chassis slip angle phase plane.

4.2.2.2 RWS mode comparison - step steer

The second test consists of a simple step steering input of 72.4° at 50 kph vehicle velocity. All RWS control modes reached $\sim 3.8 \text{ m/s}^2$ steady-state lateral acceleration during the manoeuvre, which was lower than the baseline vehicle level, as seen in Figure 4.19. Although model 2 performed the best lateral acceleration overshoot, mode 3 performed the best stability against others, as seen from the chassis side-slip overshoots in Figure 4.19 and Figure 4.20. A similar interpretation can be made when Figure 4.21 is examined. The area of mode 3 in the chassis side-slip phase plane was less than the others. This means that the vehicle with RWS control mode 3 produces less side-slip and more slowly. Thus, RWS mode 3 again performed best in terms of vehicle handling and stability.

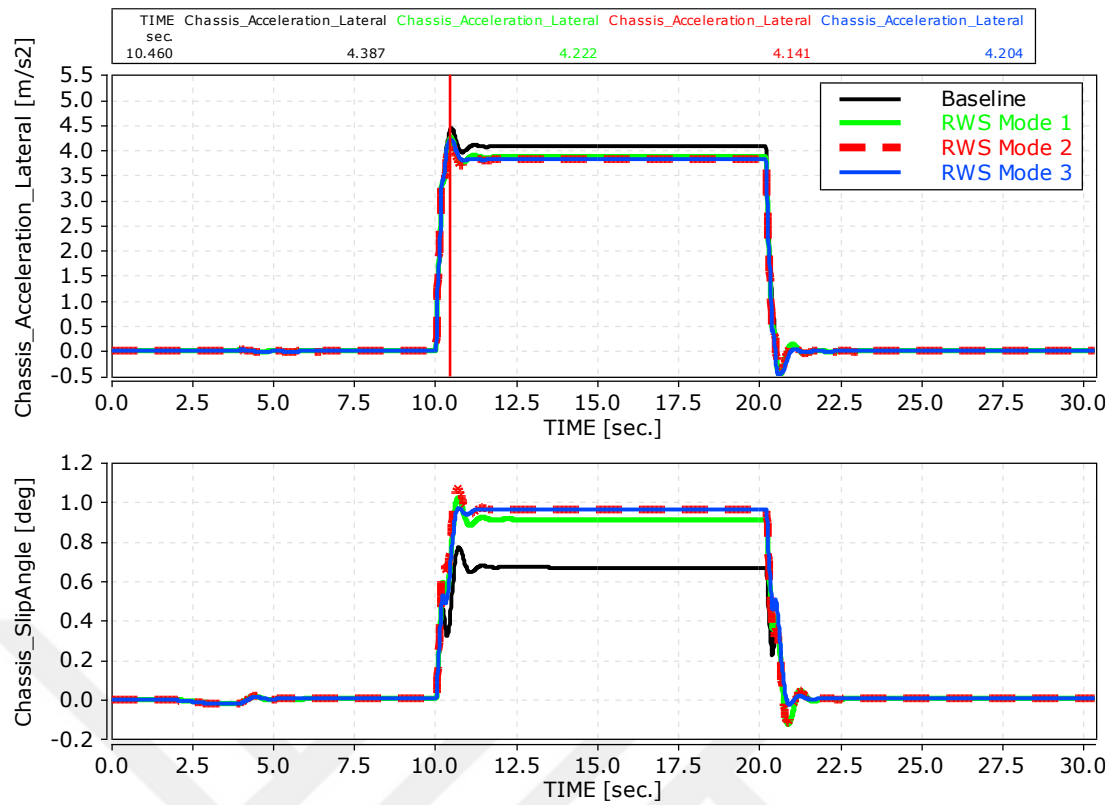


Figure 4.19: RWS modes - step steer - lateral acceleration and slip angle .

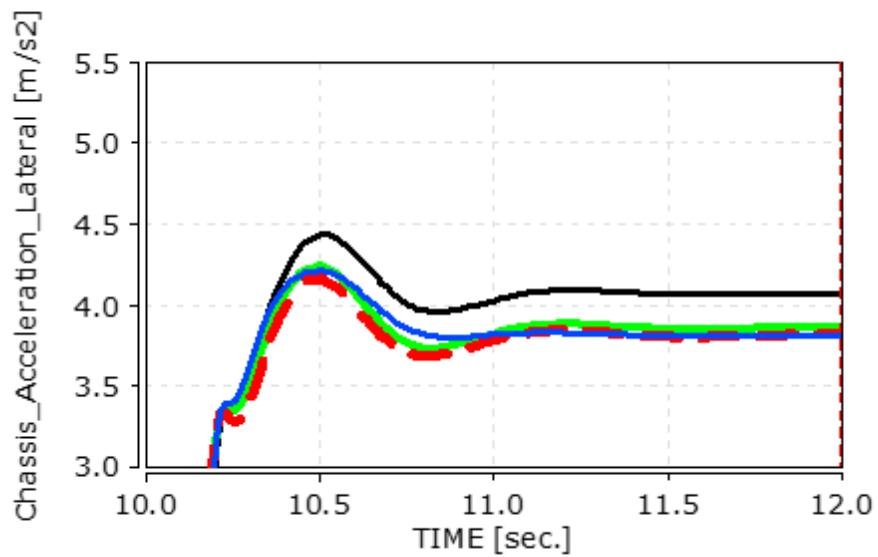


Figure 4.20: RWS modes - step steer - lateral acceleration detail.

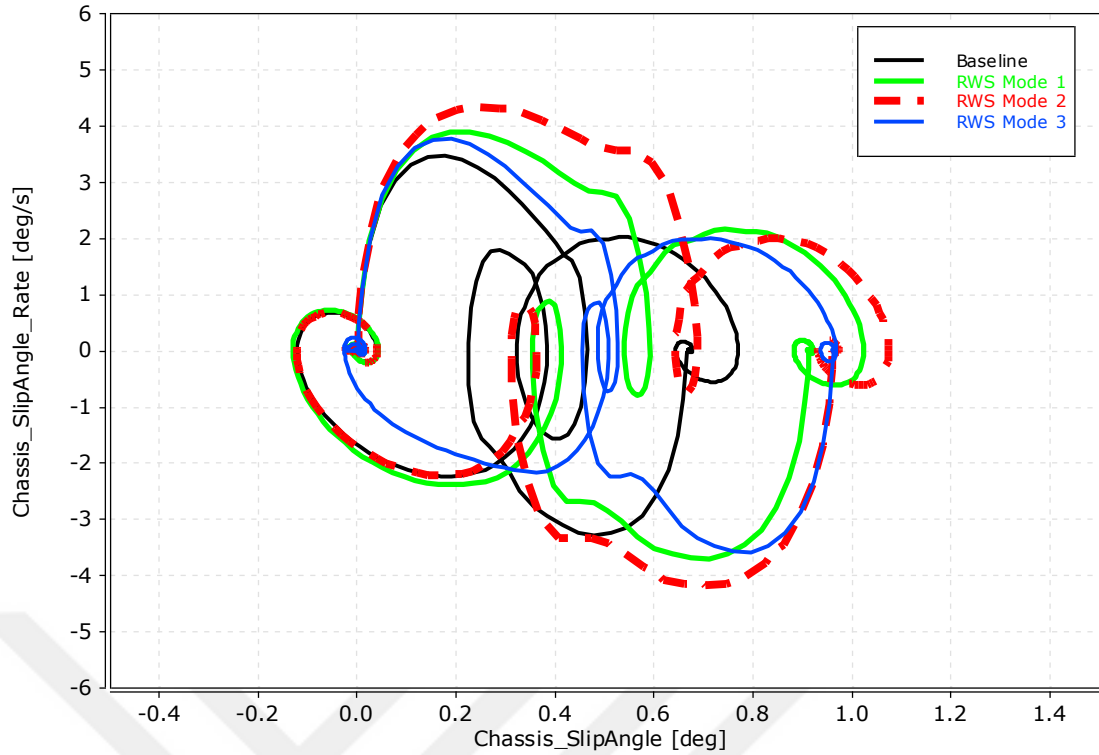


Figure 4.21: RWS modes - step steer - chassis slip angle phase plane.

4.2.2.3 RWS mode comparison - double lane change

The final comparison case was the double lane change. The track entry speed was 55 kph as previously defined. Results of the RWS modes and baseline vehicle showed that the baseline vehicle failed to complete the manoeuvre, as shown in Figure 4.22. The baseline vehicle exceeds the limits at the final stage of the track. The results of lateral acceleration, side slip angle, steering wheel angle and vehicle speed are given in Figure 4.23. It can be seen from Figure 4.23 that all RWS modes support the vehicle dynamics transient behaviour and support the vehicle to successfully complete the track. Similarly, since RWS modes 2 and 3 had similar results, a more detailed evaluation in the side-slip phase plane was needed.

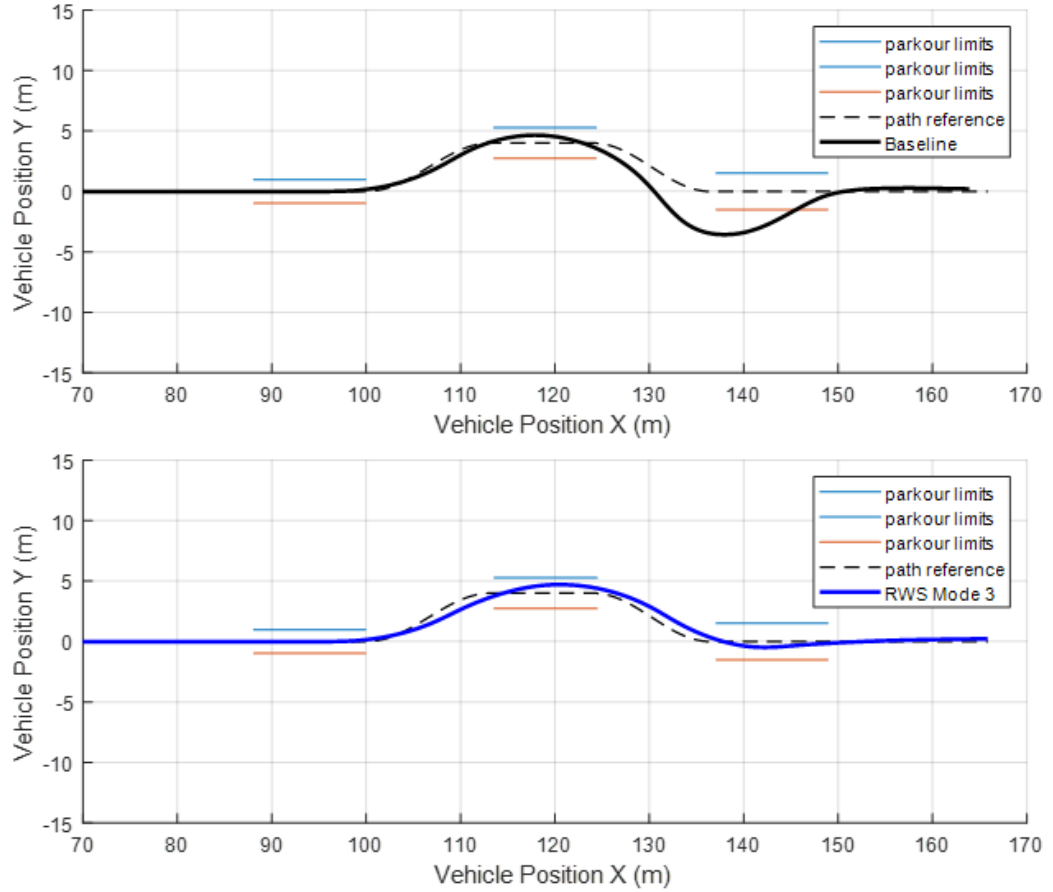


Figure 4.22: RWS modes - DLC - chassis trajectory.

The chassis sideslip phase plane in Figure 4.25 shows the valuable impact on vehicle stability in RWS mode 3. Again, the area of mode 3 in the chassis sideslip phase plane was less than the others. Although model 2 gave the best double lane change, providing the best exit speed of 43.76 kph, mode 3 performed best stability against the others, giving the minimum sideslip angle during manoeuvring. As shown in Figure 4.24, the RWS vehicle with mode 3 (transparent vehicle) kept its stability at the ninth second while the baseline vehicle (red) lost control during the simulation. In conclusion, these results showed the importance of the feedback part for vehicle stability in RWS control design. Thus, RWS mode 3 was selected for the RWS system. RWS control system DoE optimization was applied using mode 3 with all defined parametric variables.

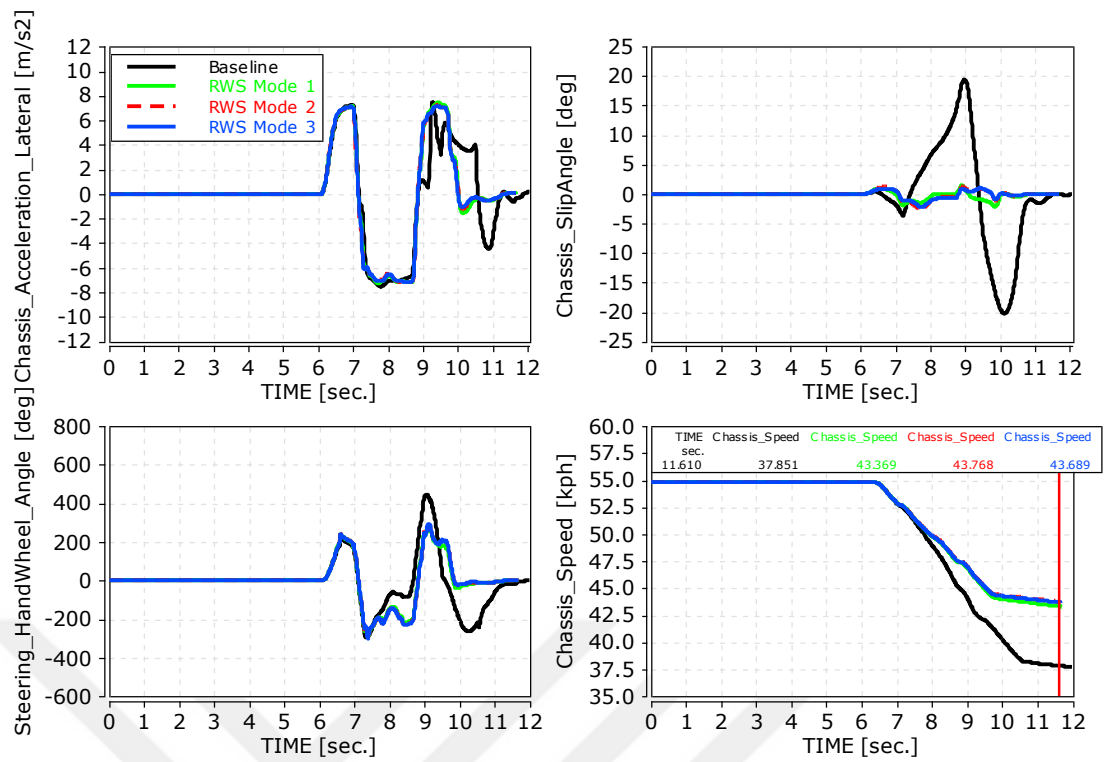


Figure 4.23: RWS modes - DLC - vehicle dynamics results.



Figure 4.24: AVL VSM Viewer - DLC - baseline vs RWS (mode 3).

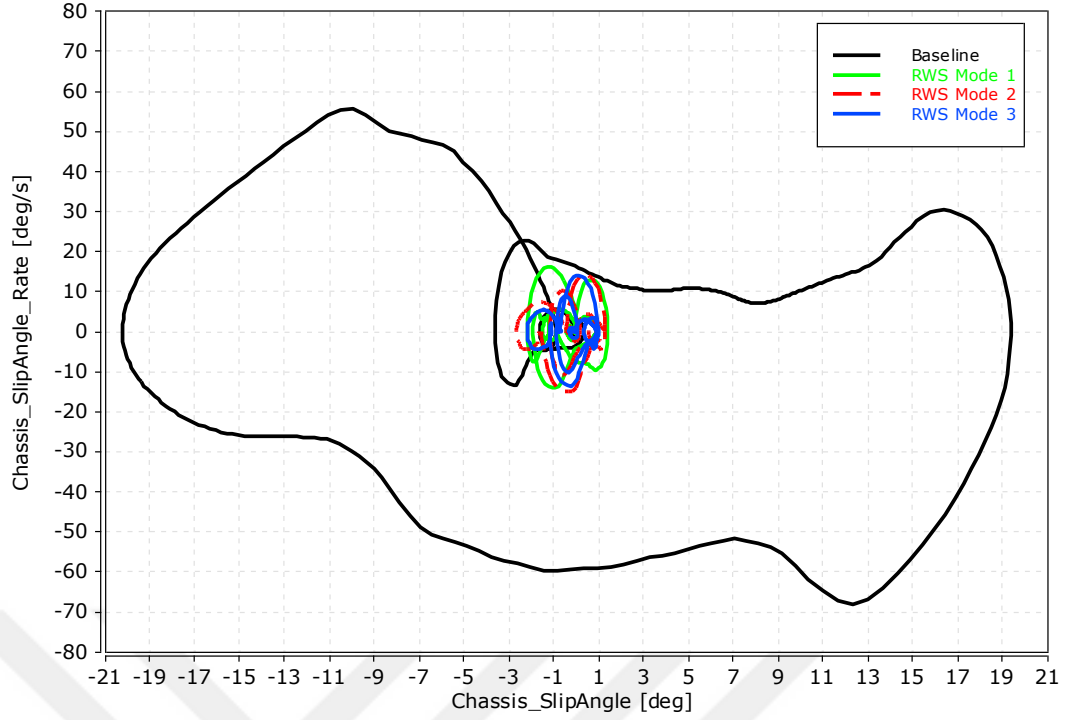


Figure 4.25: RWS modes - DLC - chassis slip angle phase plane.

4.2.3 DoE design and optimization toolchain

In the previous sections, the parametric variables to be optimized and objective functions of the RWS system were expressed, and the best RWS mode was selected. Accordingly, there were 16 parameters to be optimized for 4 test cases and 6 objective functions. The main objective of all RWS optimization procedures is to obtain optimum control parameters which work in all test conditions. The objective function space has multiple objectives defined in different test cases. The system has multi-input and procedures and variable constraints. In multi-objective optimization, there is no straightforward method to determine whether one solution is better than the others. Thus, as mentioned earlier in Section 2.2, biases are added to general optimization problems to adjust the desired target in many test cases. Desired targets are the driving modes that the user/driver can select. Thus, biases and constraints were selected according to the defined drive modes. The general multi-objective optimization problem with parameter variable constraints and biases was expressed as:

$$\begin{aligned}
opt J_{RWS} = & b_{RWS,1}f_{RWS,RS,50} + b_{RWS,2}f_{RWS,RS,100} + \\
& b_{RWS,3}f_{RWS,RS,150} + b_{RWS,4}f_{RWS,SS} + b_{RWS,5}f_{RWS,DLC} + \\
& b_{RWS,6}f_{RWS,TS}
\end{aligned}$$

$$\begin{aligned}
& \text{subject to} = \\
& -4 < a_1 < 4, -4 < b_1 < 4, -4 < c_1 < 4, \\
& -4 < a_2 < 4, -4 < b_2 < 4, -4 < c_2 < 4, \\
& -4 < a_3 < 4, -4 < b_3 < 4, -4 < c_3 < 4, \\
& -4 < a_4 < 4, -4 < b_4 < 4, -4 < c_4 < 4, \\
& -4 < a_5 < 4, -4 < b_5 < 4, -4 < c_5 < 4, \\
& 1 < y_{ble} < 36,
\end{aligned} \tag{55}$$

Where the biases of the multi-objective optimization equation were defined as follows; $b_{RWS,1}$, $b_{RWS,2}$, $b_{RWS,3}$, $b_{RWS,4}$, $b_{RWS,5}$, $b_{RWS,6}$. Therefore, these biases were chosen according to the drive modes. The limits of the parametric variables were selected via open-loop test cases and considering the system limits. The general RWS optimization scheme is shown in Figure 4.26. After each simulation case in the objective evaluation phase, all objective function metrics were calculated automatically via post-process MATLAB scripts. The calculated objective metrics/ responses were sent to the DoE tool AVL CAMEO. In AVL CAMEO, parametric variables were changed and selected according to the DoE design. The RWS control system MATLAB Simulink sent selected parametric variables to each sequencing run. Thus, all simulations were run with variable control parameters. Finally, the loop indicated in Figure 4.11 was completed for each scenario/sequence.

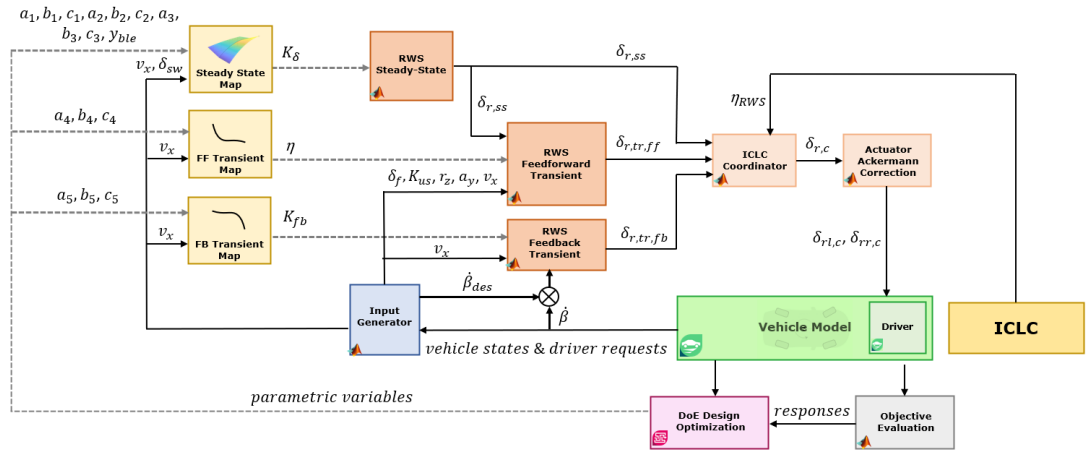


Figure 4.26: RWS system and optimization toolchain scheme.

In the design of experiment studies, there can be several DoE designs. The proper design should be selected for the application. The Sobol design was selected for the RWS optimization DoE design since the Sobol design was allowed to optimize up to

16 variations. This design uses a quasi-random algorithm and distributes the design point uniformly over the design hypercube, as represented in Figure 4.27 [55].

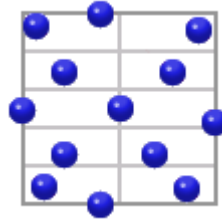


Figure 4.27: Sobol DoE design.

In the Sobol design, 1000 design points were selected for each objective function to increase the estimation of the regression performance. There were 6 objective functions, which means a total of 6000 simulations were run to optimize the RWS control system. In other words, the loop that was shown in Figure 4.26 ran 6000 times with different parametric variables for 6 different test scenarios defined. After the offline optimization procedure, all test results were evaluated in AVL CAMEO. In DoE, regression analysis was performed to estimate the effect of parameters on response. In general, higher R^2 values (> 0.8) can be accepted as successful estimations. Figure 4.28 shows the regression analysis of all runs and objective measurements. According to the results of the regression analysis, only the double lane change scenario could not give plausible results. The other 5 test case results showed accurate estimation to make an optimization. Therefore, regardless of the quality of the estimations, post-tests should be done to check and verify the optimization estimations.

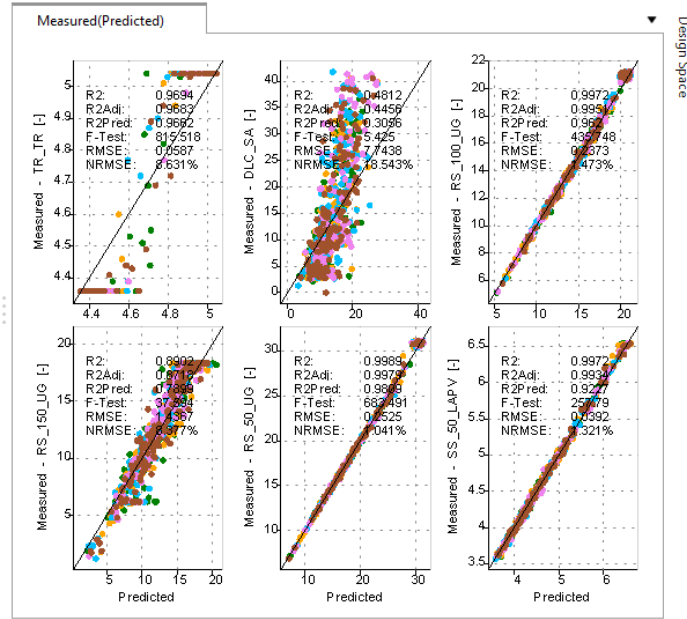


Figure 4.28: DoE Estimation Results of RWS optimization.

Two different driving modes are defined for optimising RWS: “comfort” and “sport” modes. In lateral vehicle dynamics, the terms “comfort” and “sport” refer to two different models of operation that can be selected by the driver according to their desire [20], as previously stated in section 2.2. These modes basically automatically adjust the handling characteristics of a vehicle after the selection. In the optimization process, the biases previously expressed in Equation 55 were selected according to the purpose of each driving mode. Since it is assumed that the comfort mode can be used by the driver in the daily driving, the ramp steer biases of the comfort mode ($b_{RWS,1}$, $b_{RWS,2}$, $b_{RWS,3}$) were selected higher than the other objective metric biases and second priority was given to the double lane change ($b_{RWS,5}$) scenario. The most critical cases were selected as step steer at 100 kph ($b_{RWS,4}$) and double lane change ($b_{RWS,5}$) for sport mode. The remaining biases are evenly distributed for sport mode optimization. Table 4.4 shows the biases used in RWS optimization.

Table 4.4: RWS optimization biases of different drive modes.

RWS Drive Mode	$b_{RWS,1}$	$b_{RWS,2}$	$b_{RWS,3}$	$b_{RWS,4}$	$b_{RWS,5}$	$b_{RWS,6}$
Comfort	0.3	0.3	0.3	0.1	0.2	0.1
Sport	0.1	0.1	0.1	0.3	0.3	0.1

Table 4.5 shows the optimized parametric variables found via offline DoE. In the next section, the RWS system was tested in defined test scenarios to validate the optimization and the entire RWS system.

Table 4.5: RWS different drive modes parametric variables.

RWS Parametric Variables	RWS Part	RWS Comfort Mode	RWS Sport Mode
a_1	Steady-State	-0.213	-0.116
b_1	Steady-State	1.760	0.752
c_1	Steady-State	-1.310	-0.688
a_2	Steady-State	-0.223	-0.760
b_2	Steady-State	1.860	2.910
c_2	Steady-State	-1.210	-2.190
a_3	Steady-State	-0.233	-0.640
b_3	Steady-State	1.960	2.690
c_3	Steady-State	-1.110	-2.090
y_{ble}	Steady-State	14.1	13.2
a_4	FF Transient	1.210	1.050
b_4	FF Transient	-0.510	-0.408
c_4	FF Transient	0.330	0.151
a_5	FB Transient	1.990	0.500
b_5	FB Transient	-0.190	0.407
c_5	FB Transient	-1.271	0.411

Figures 4.29, 4.30 and 4.31 show the graphical representations of the steady-state coefficient K_δ , tuneable parameter η and the feedback gain K_{fb} maps formed with the optimized parametric variables, respectively.

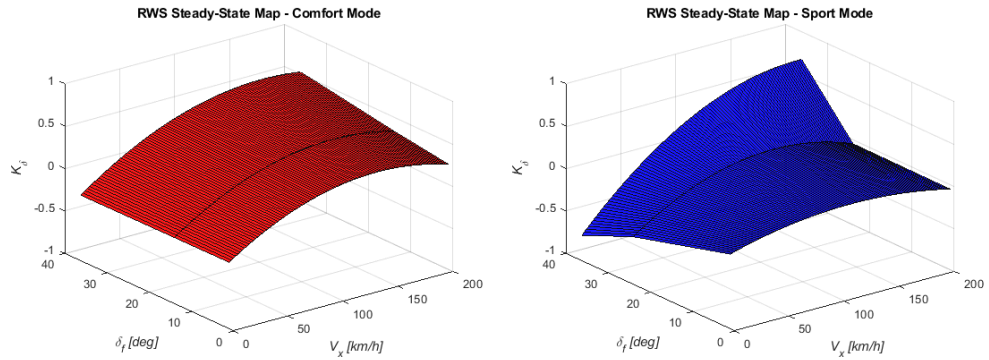


Figure 4.29: RWS modes steady-state coefficient maps.

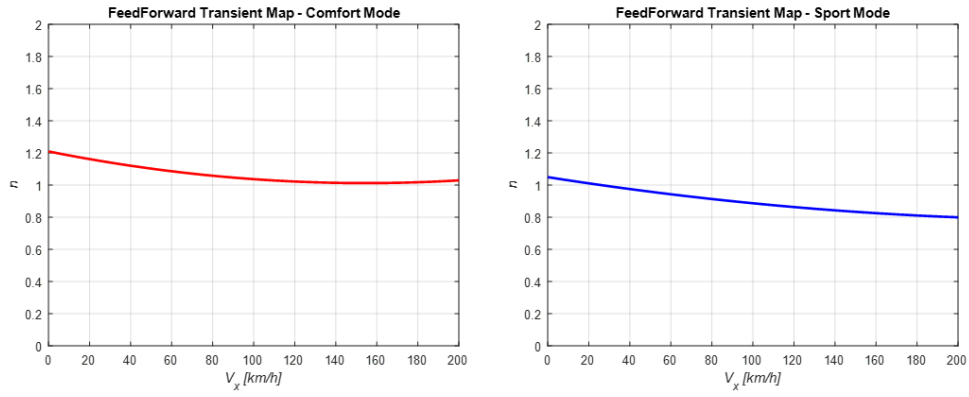


Figure 4.30: RWS modes feedforward tuneable parameter maps.

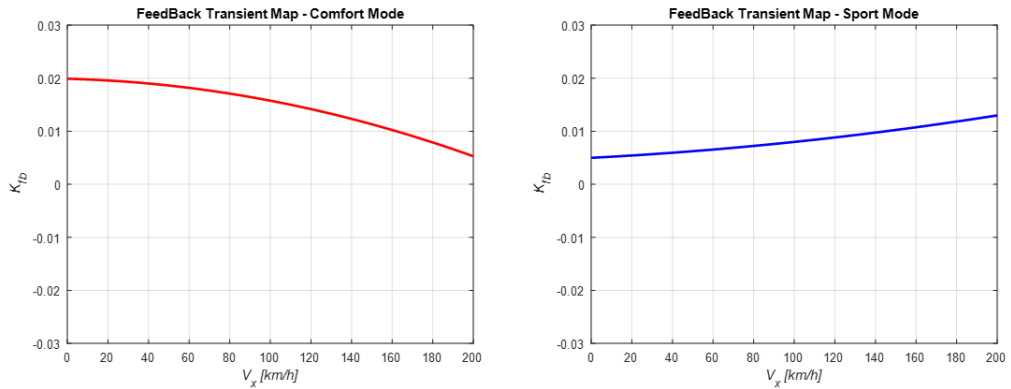


Figure 4.31: RWS modes feedback gain maps.

4.2.4 RWS evaluation and optimized results

In this section, the results of the comfort and sport RWS driving modes with AVL VSM simulations are presented. Before starting each simulation, the driving mode was selected in the AVL VSM user interface, as given in Figure 4.32. The RWS system and drive methods were evaluated with the same test cases we used in the optimization process. Test cases were ramp steer manoeuvre at 50, 100 and 150 kph, step steer at 100 kph, double lane change and turning radius. In all cases, RWS third mode was used as its benefits have already been discussed.

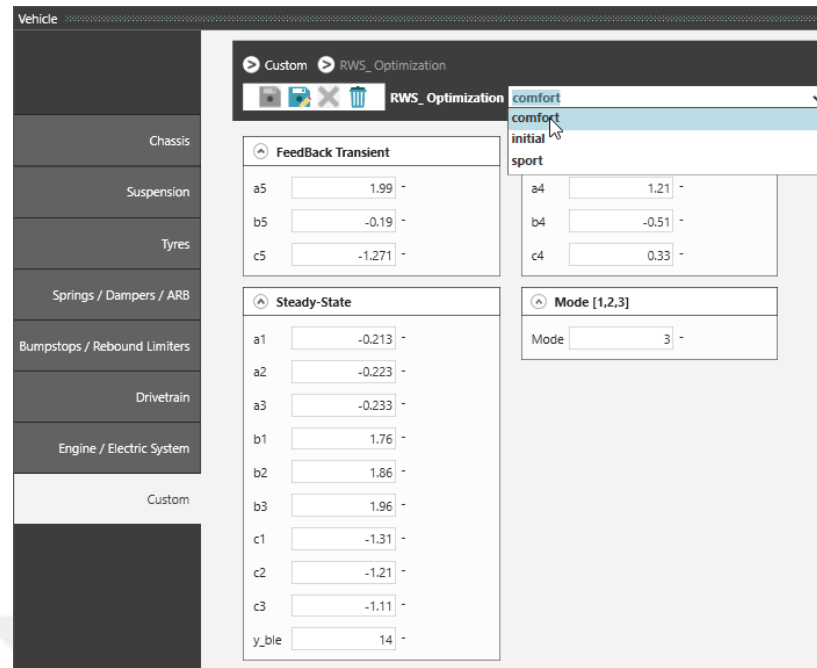


Figure 4.32: RWS driving mode selection in AVL VSM.

4.2.4.1 RWS results - ramp steer - 50 kph

Initially, two RWS drive modes were evaluated in the ramp steer manoeuvre at 50 kph. Figure 4.33 shows the lateral behaviour of the vehicle with the different RWS driving modes. As expected, the linear understeer gradient was increased with comfort mode. Using the sport mode for the RWS slightly decreased the understeer gradient. The steering angle input of sport mode was approximately 1.1 degrees lower than the baseline vehicle at the 5 m/s^2 lateral acceleration level. The optimization bias of sport mode for the ramp steer 50 kph $b_{RWS,1}$ case was lower than that of comfort mode. Thus, the expected result was achieved. Moreover, the sport mode made the vehicle more responsive at this speed. More responsive behaviour can also be seen in Figure 4.34. The rear wheels were steered opposite to the front ones in sport mode.

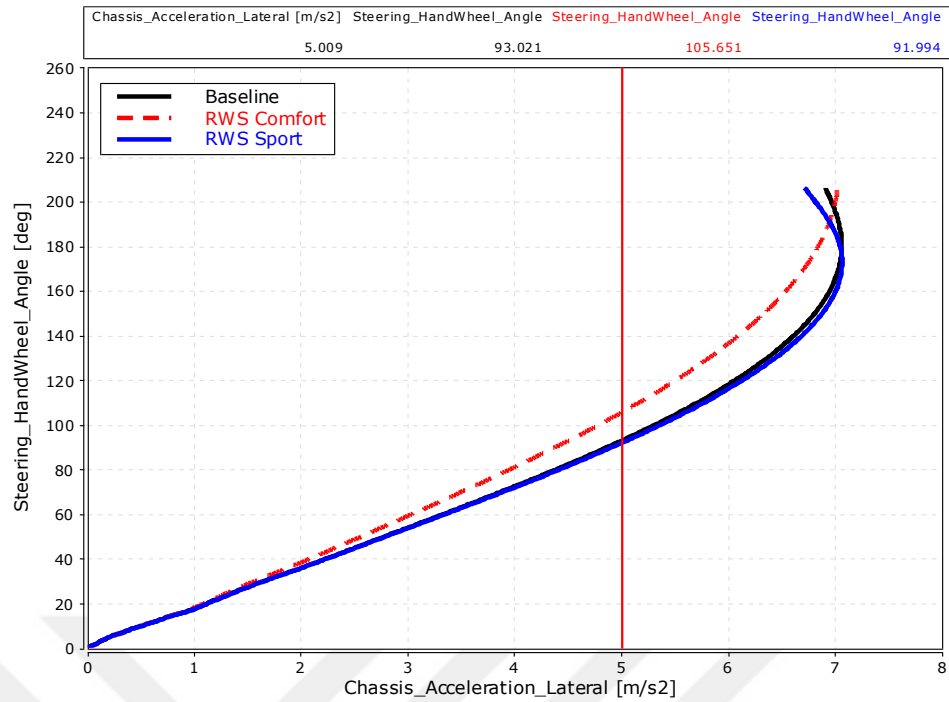


Figure 4.33: RWS results - ramp steer 50 kph - steering angle vs lateral acc.

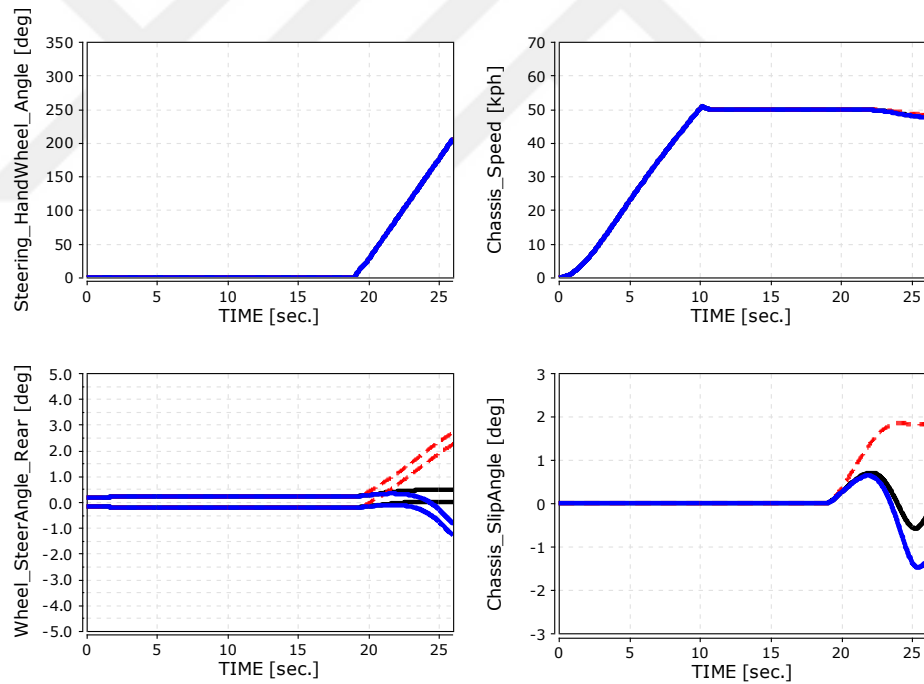


Figure 4.34: RWS results - ramp steer 50 kph - vehicle dynamics states

4.2.4.2 RWS Results - Ramp steer - 100 kph

In the ramp steer 100 kph results, both sport and comfort modes increased the linear understeer gradient, as shown in Figure 4.35. Again, the sport mode was the more responsive one. In both driving modes, the rear wheels steered in the same direction as the front ones. The comfort mode had minimal side slip angle since comfort mode

steered more appropriately according to the zero-side slip approach. It has been clearly seen that the maximum rear wheel angles locked at 3.5 degrees from the comfort mode results in Figure 4.36. The side slip result of the sport mode is also lower than the baseline vehicle for that speed.

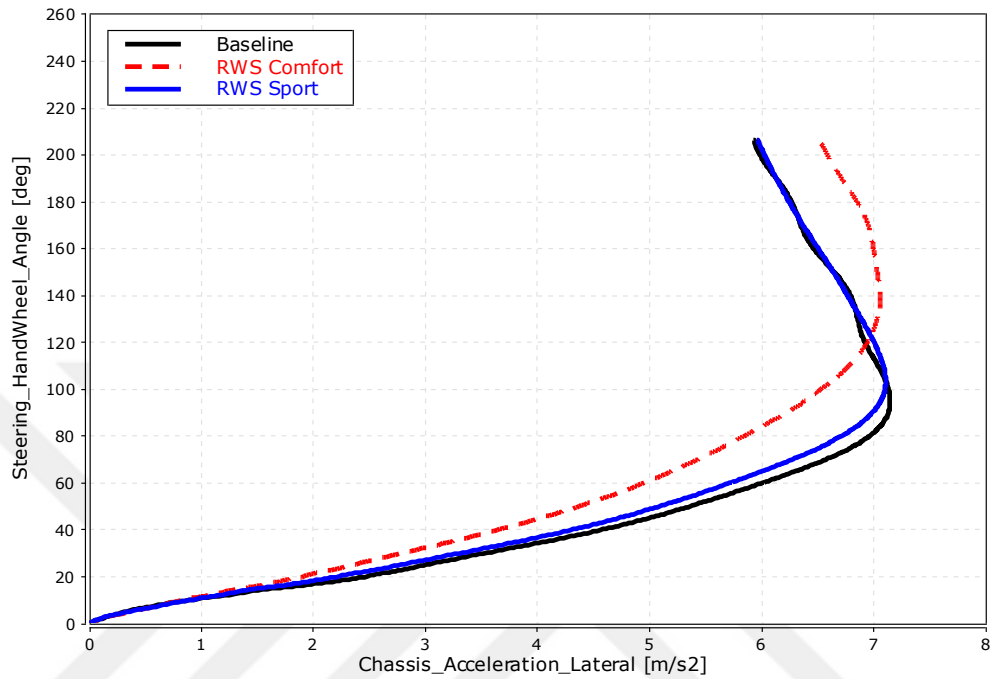


Figure 4.35: RWS results - ramp steer 100 kph - steering angle vs lateral acc.

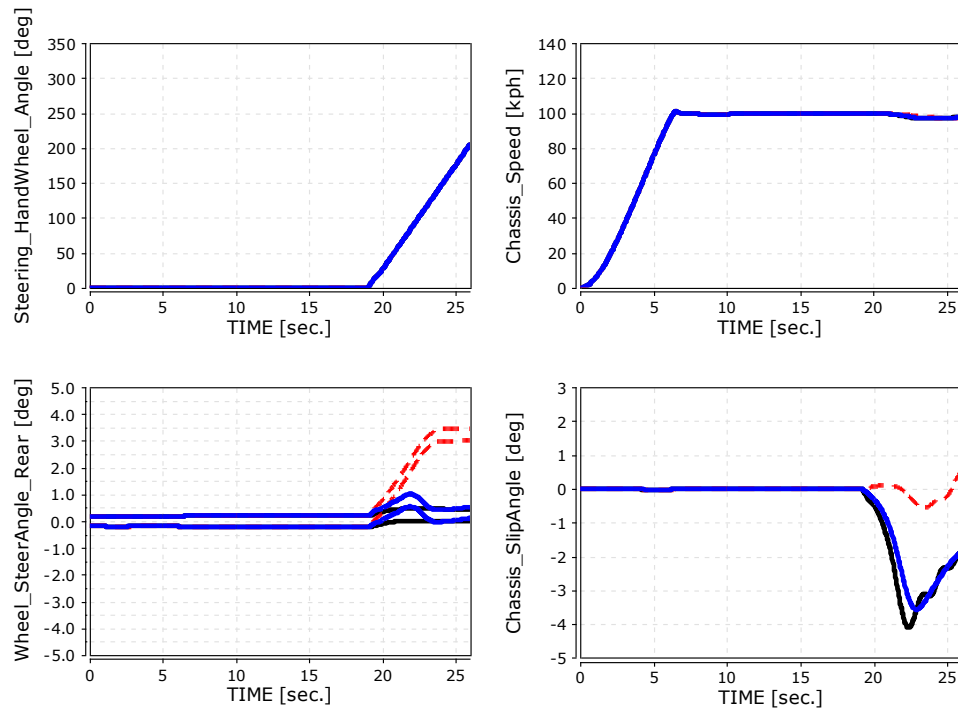


Figure 4.36: RWS results - ramp steer 100 kph - vehicle dynamics states.

4.2.4.3 RWS results - ramp steer - 150 kph

The final ramp steer manoeuvre speed was 150 kph. In this test case again, both sport and comfort modes increased the linear understeer gradient, as can be seen in Figure 4.37. The rear wheels were steered opposite to the front ones in sport mode in some moments. Thus, sport mode vehicle slip angle results were worst since it was far from the zero-side slip principle for this speed. The comfort mode had a minimum side slip angle again since the comfort mode steered more appropriately according to the zero-side slip approach. Rear wheel angles locked at 3.5 degrees from the comfort mode results in Figure 4.38. Since both modes increased the linear understeer gradient, targets were met in the ramp steer manoeuvre for the RWS.

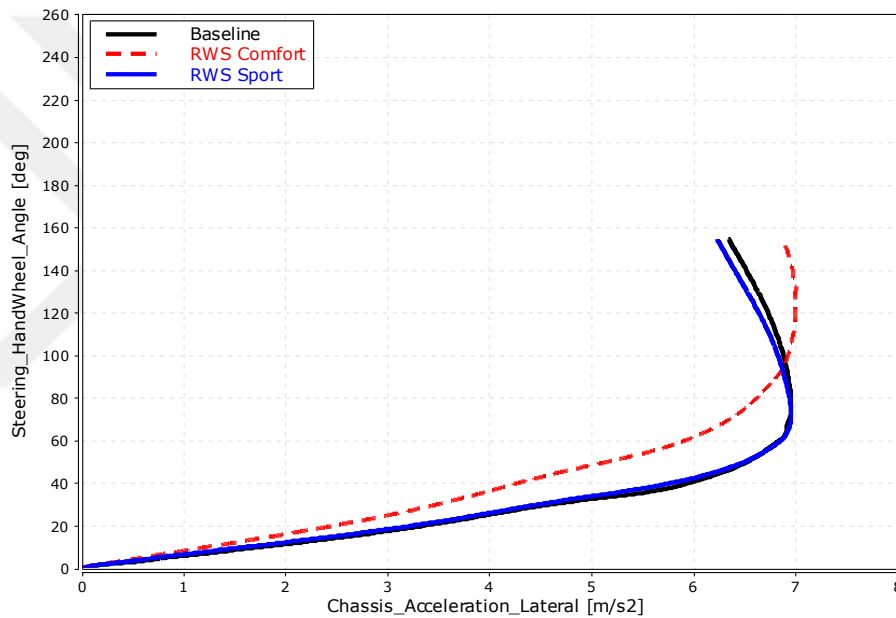


Figure 4.37: RWS results - ramp steer 150 kph - steering angle vs lateral acc.

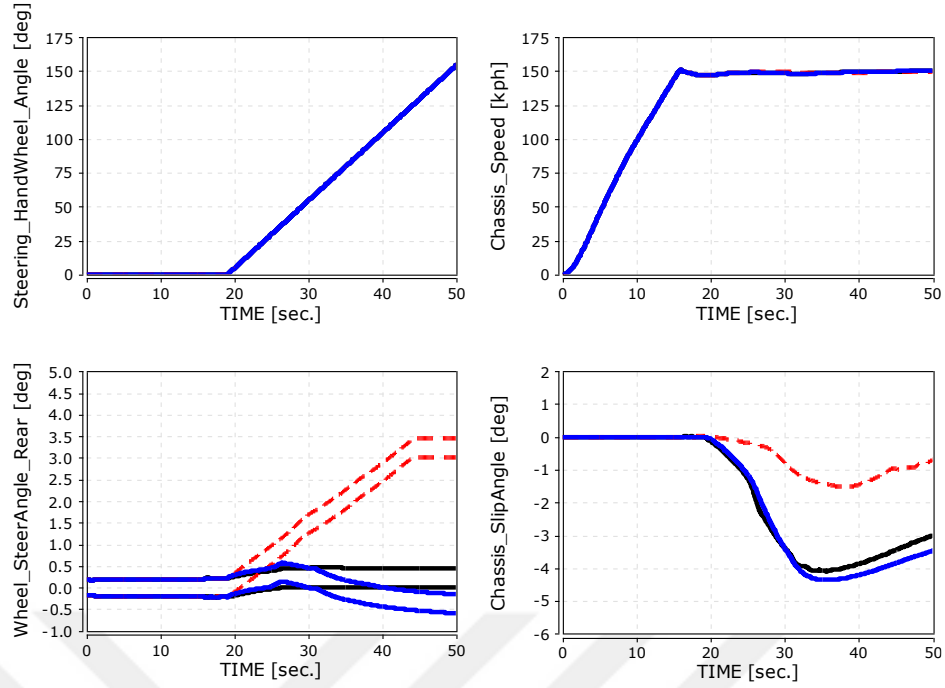


Figure 4.38: RWS results - ramp steer 150 kph - vehicle dynamics states.

4.2.4.4 RWS results - step steer

The step steer test consists of a simple step steering input of 30.9° at 100 kph vehicle velocity for the baseline vehicle to reach steady-state 4 m/s^2 lateral acceleration. The objective was to minimize the difference between peak and steady-state lateral acceleration values and to keep 4 m/s^2 steady-state lateral acceleration. From Figure 4.39, it was clearly seen that overshoots of lateral response were improved with both comfort and sports modes. RWS comfort mode increased the steering input to 44° to keep 4 m/s^2 steady-state lateral acceleration. This can create an understeer feeling in the driver. RWS sport mode increased the steering input to 32° to maintain the same lateral acceleration. The difference between the peak and steady-state lateral acceleration of the baseline vehicle was 0.350 m/s^2 . The difference value in RWS comfort mode was 0.117 m/s^2 . With RWS sport mode, the lateral acceleration did not exceed the steady state value and provided a smooth RWS action. The optimization bias of sport mode for the step steer 100 kph case was prominent, so sport mode met the expectations and gave the best result in overshoots. Figure 4.40 shows the chassis sideslip phase plane. Since the comfort mode area was the lowest, it gave the best results for vehicle stability.

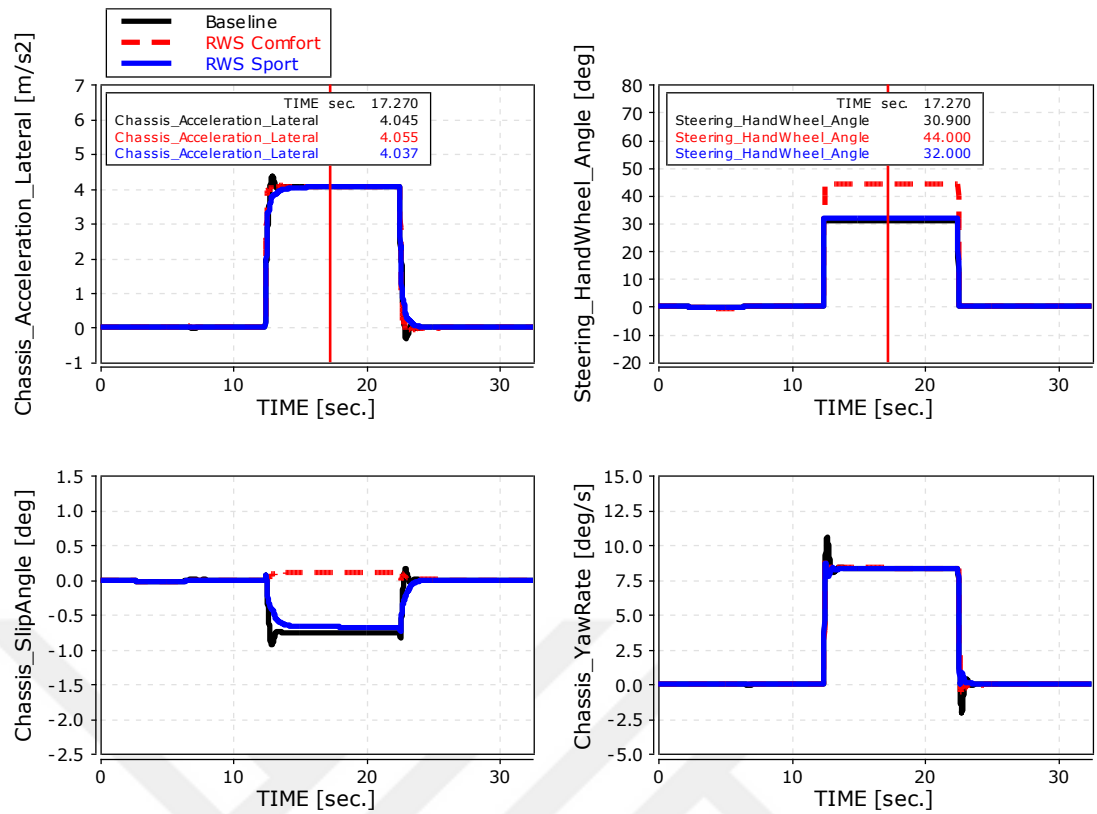


Figure 4.39: RWS results - step steer 100 kph – responses.

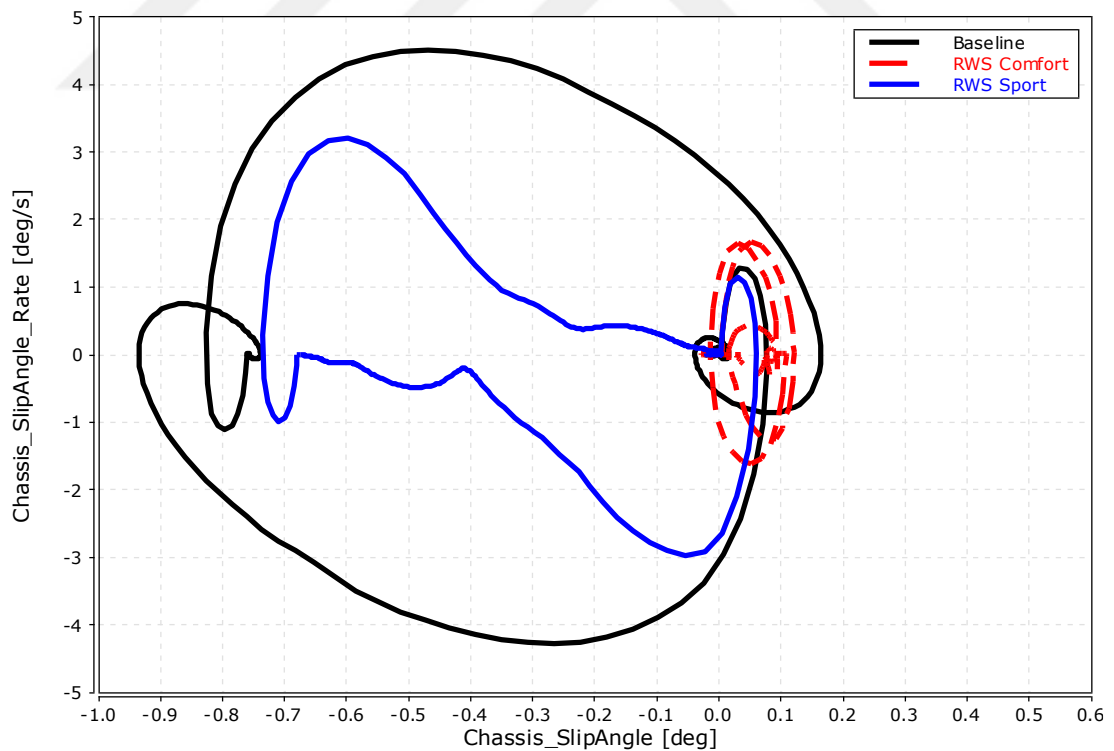


Figure 4.40: RWS results - step steer 100 kph - chassis slip angle phase plane.

4.2.4.5 RWS results - double lane change

The essential transient evaluation case was the double lane change. It was already known that the baseline could not complete the manoeuvre successfully. Both driving modes completed the manoeuvre without exceeding the path limits with a 55 kph entry speed, as seen in Figure 4.41. However, from the same figure, it was also seen that the sport mode came close almost to the limit of the track. Since the optimization bias of sport mode for the double lane change $b_{RWS,5}$ case was the highest one, sport mode was expected to give the lowest minimum side-slip angle result and best track-following results. However, the regression estimation analysis of the DLC results was of poor quality as the R^2 was less than 0.8, so the estimation of sport mode did not accomplish well. Nevertheless, the sport mode performed well in the DLC scenario. All vehicle dynamics results of RWS driving modes are given in Figure 4.42. It is seen from Figure 4.42 that the best exit speed belongs to the comfort mode. The maximum side slip angles during the manoeuvres for the comfort and sport mode were 2.4 and 4.2 degrees, respectively. Thus, the comfort mode gave the best results in terms of path tracking and vehicle stability in the DLC case. The chassis sideslip phase plane is given in Figure 4.43. As can be seen from this graph, the comfort mode provided the best results in terms of vehicle stability. Figure 4.44 shows the DLC scenario visually in the AVL VSM viewer. When the vehicle is at the 120th meter or approaching the track limits (from Figure 4.41), it is shown on the left in Figure 4.44. The difference in lateral displacement between the two different RWS driving modes was about 0.5 meters, which can be seen in both Figure 4.41 and Figure 4.44. Finally, Figure 4.45 shows that the feedforward part of the comfort mode worked more actively than the feedforward part of the sport mode. In both RWS modes, the feedback part was supported by the feedforward to enhance vehicle stability.

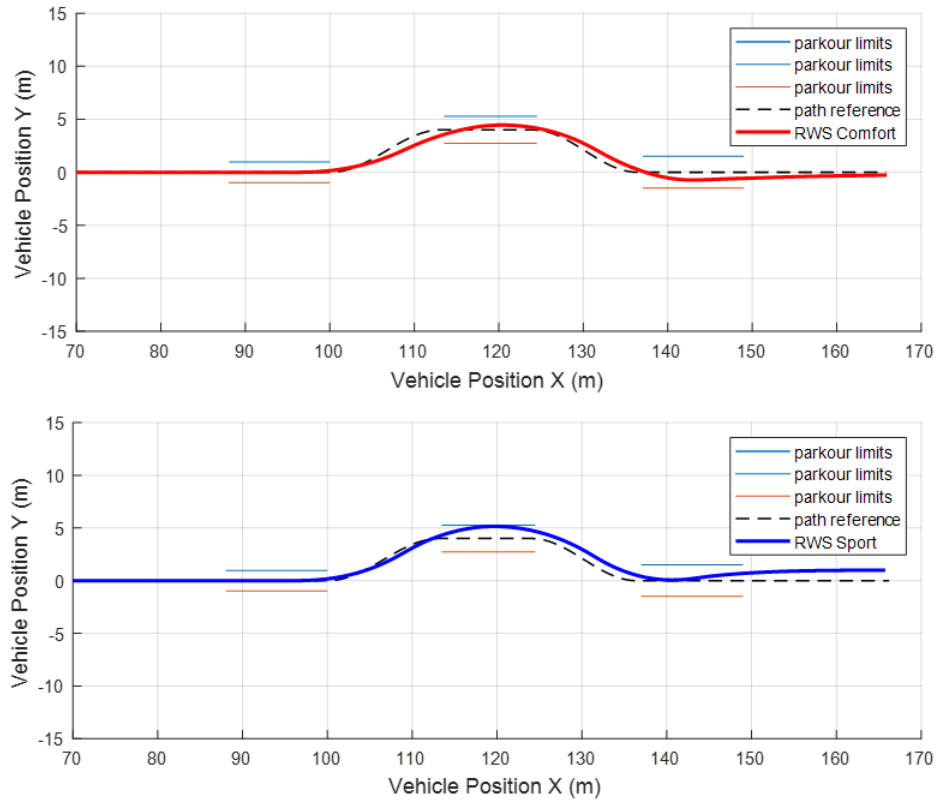


Figure 4.41: RWS driving modes - DLC - chassis trajectory.

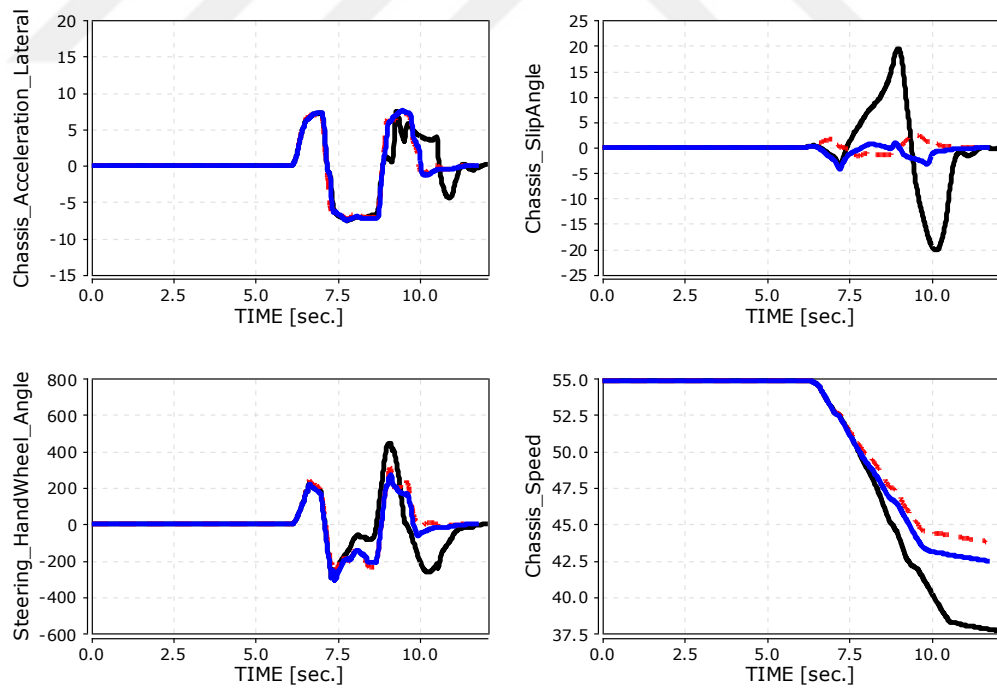


Figure 4.42: RWS results - DLC - vehicle dynamics results.

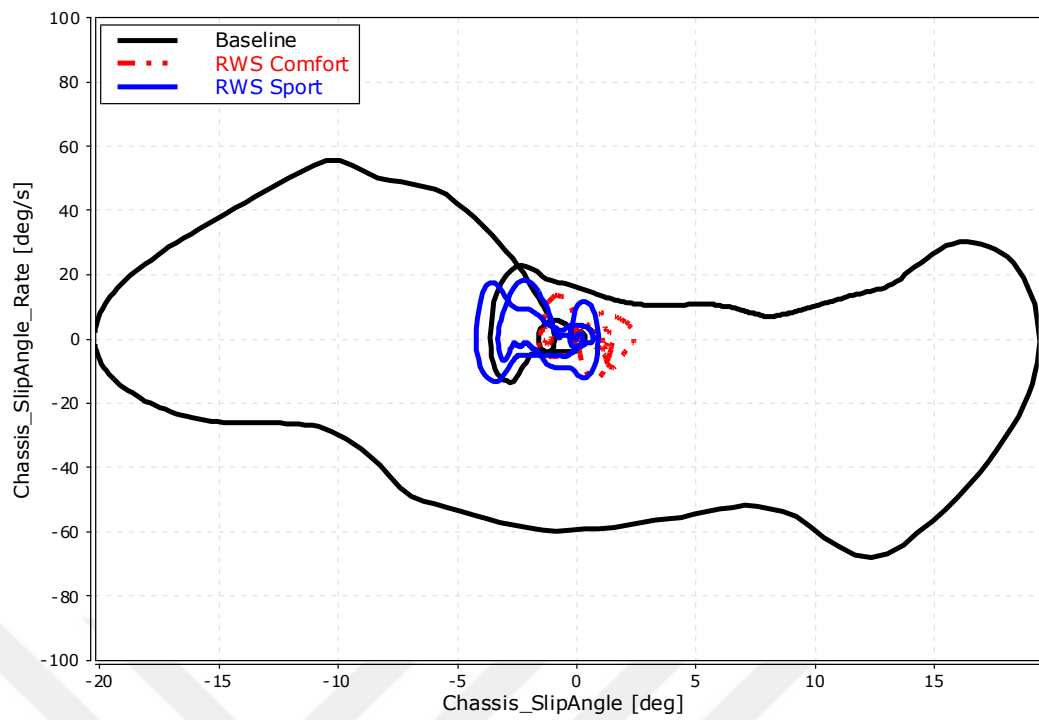


Figure 4.43: RWS results - DLC - chassis slip angle phase plane.

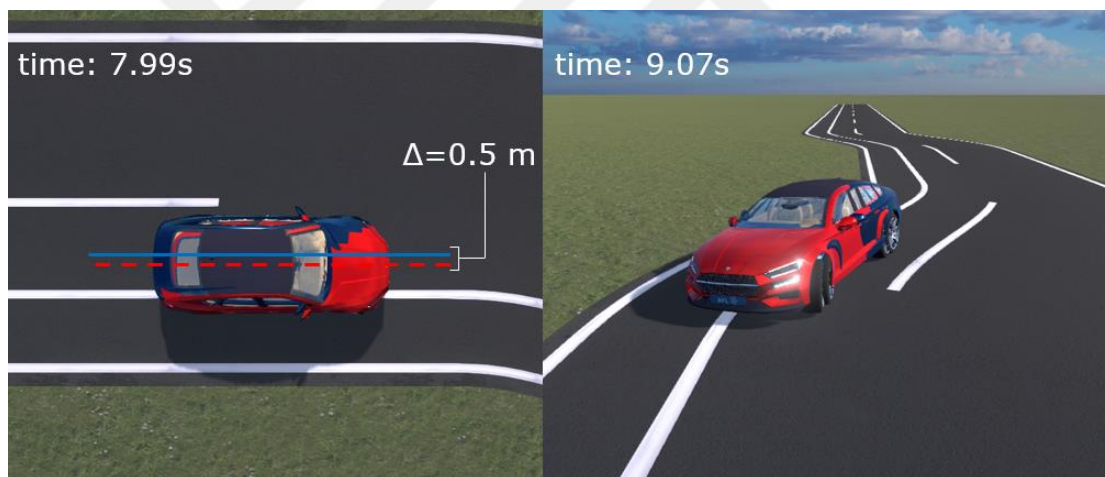


Figure 4.44: AVL VSM Viewer - DLC - RWS comfort vs RWS sport.

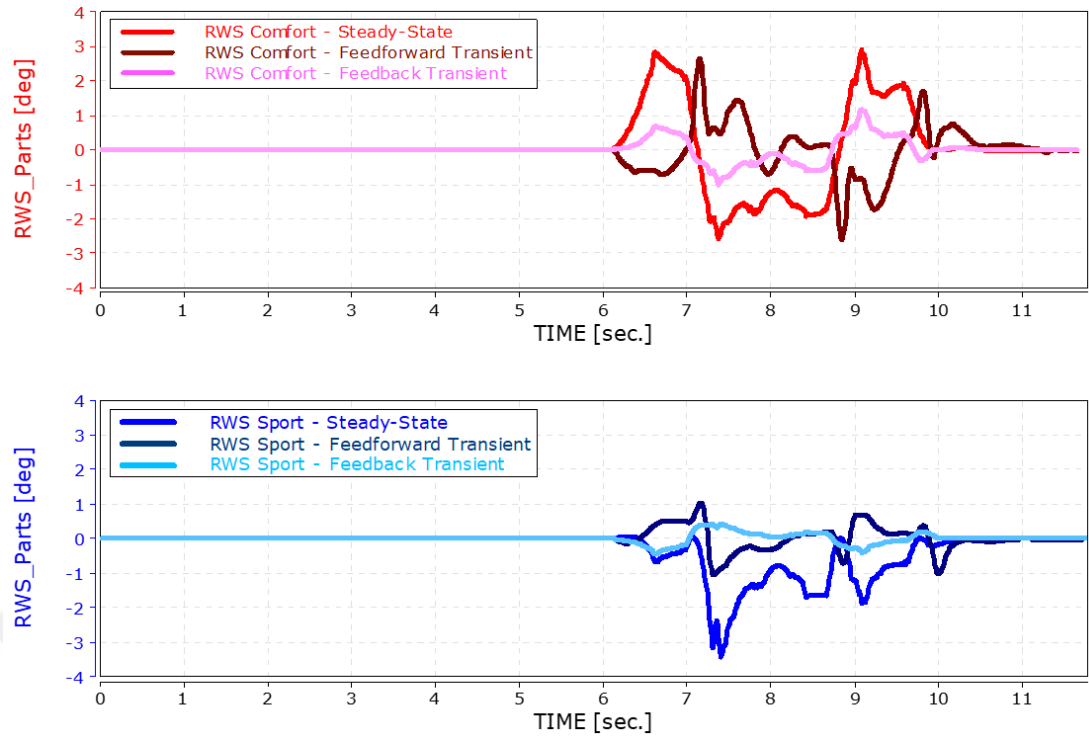


Figure 4.45: RWS results - DLC - RWS parts.

4.2.4.6 RWS results - turning radius

The last test case was the turning radius. Both comfort and sport RWS modes decreased the turning radius as expected. The turning radius was reduced from 4.67 to 4.36 meters with comfort and sport mode as shown in Figure 4.46.

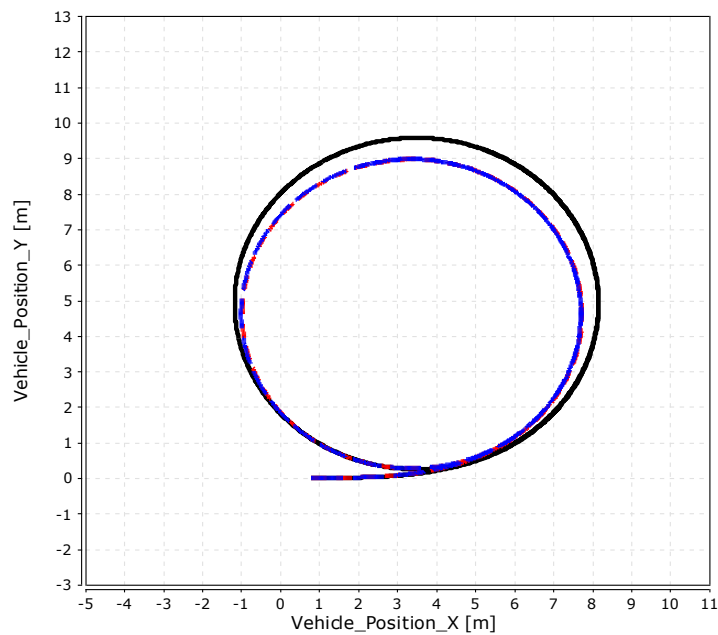


Figure 4.46: RWS driving modes – turning radius.

4.2.4.7 RWS evaluation

Consequently, all RWS drive modes and results were automatically evaluated and optimized with the calibration toolchain of the RWS system shown in Figure 4.26. According to the RWS driving mode results, the comfort mode successfully met the targets and increased the linear understeer gradient in all ramp steer with different speed conditions. In addition, the comfort mode decreased the lateral acceleration overshoot and enhanced the vehicle stability in the DLC case. In addition, the sport mode was provided with a more neutral driving experience for the driver below 50 kph. Also, the sport mode met the targets for increasing the understeer gradient for other speeds. As a best result, the sport mode was able to noticeably reduce the overshoot in the step steer manoeuvre. Both driving modes reduced the turning radius and increased the vehicle's manoeuvrability. All objective results are summarized in Table 4.6.

Table 4.6: RWS optimization and results of objective measurements.

Test Case	Speed [kph]	Objective Measurement	Base	RWS Comfort Mode	RWS Comfort Benefit	RWS Sport Mode	RWS Sport Benefit
Ramp Steer	50	K_{us} [deg s ² /m]	17.87	22.92	5.05	17.76	-0.11
Ramp Steer	100	K_{us} [deg s ² /m]	7.67	10.72	3.05	9.48	1.81
Ramp Steer	150	K_{us} [deg s ² /m]	6.35	9.15	2.8	6.41	0.06
Step Steer	100	a_y overshoot difference [m/s ²]	0.350	0.117	-0.233	0	-0.35
DLC	55	β [deg]	20.0	2.4	-17.6	4.2	-15.8
Turn Radius	5	R [m]	4.67	4.36	-0.31	4.36	-0.31

4.3 TVC Controller Design

In the TVC system, the torque of the all-electric drive units was controlled by the TVC system. With the established TVC system, both front-rear and right-left torque distribution were controlled. Therefore, all three electric motor signals need to be controlled in Simulink. Figure 4.47 shows the implementation of TVC in Simulink. By default, AVL VSM allows four EDU torques to be controlled. These four EDUs are front left, front right, rear left, and rear right, but a vehicle with three EDUs was

used in this study. Thus, only three electric motors were controlled. The front EDU was called “Electric Drive 1”. The left and right rear EDUs were named “Electric Drive 3” and “Electric Drive 4”, respectively.

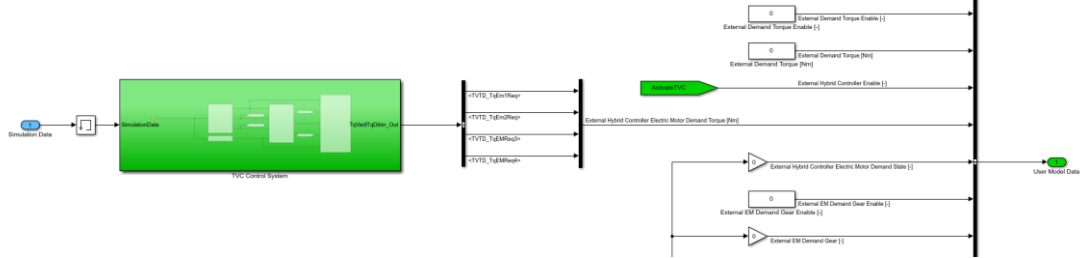


Figure 4.47: TVC system implementation in Simulink.

The design of the TVC model was based on the works of Monni and Lenzo [30] [56]. Basically, the general TVC algorithm is from Monni, and the feed-forward portion of TVC is from Lenzo's work. Thus, the overall TVC controller is a combination of two different TVC control systems, and the main aim and motivation were to obtain a detailed model based TVC system. The proposed TVC general scheme is shown in Figure 4.48. The TVC system, like the RWS, has feedback and a feedforward part for the lateral torque distribution. The details of each part are expressed in the following sections. The additional yaw moment M_z and longitudinal torque distribution were controlled by the TVC system. ICLC coordinator part takes the additional yaw moment M_z and multiplies it with ICLC TVC bias factor η_{TVC} and sends the corrected additional yaw moment $M_{z,c}$ to the torque distribution controller. In this way, the vehicle motion control system (ICLC) controls how much additional yaw torque should be applied to the vehicle.

The TVC system designed in this thesis consists of seven parts:

- TVC vehicle state and input generator
- TVC calculators
- TVC longitudinal controller
- TVC wheel-slip controller (WSC)
- TVC lateral controller
- TVC torque distribution controller

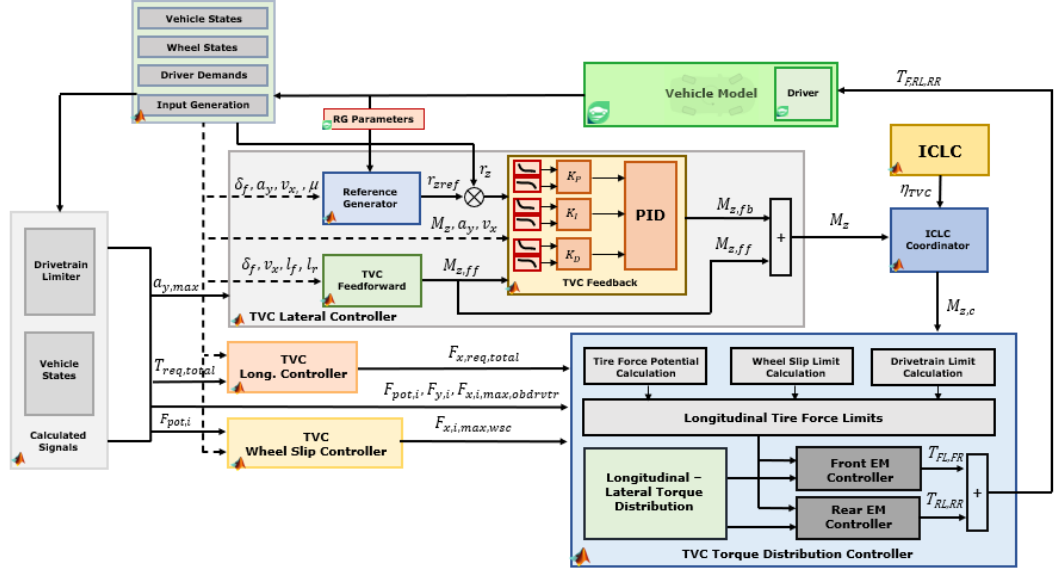


Figure 4.48: TVC system scheme.

4.3.1 TVC vehicle state and input generator

All the required signals were collected in the input generator, driver demands, wheel demands and vehicle demand block. Inputs of the TVC system are:

- Vehicle side slip angle β [rad]
- Yaw rate r_z [rad/s]
- Vehicle speed v_x [m/s]
- Lateral acceleration a_y [m/s²]
- Longitudinal acceleration a_x [m/s²]
- Steering wheel angle δ_{sw} [deg]
- Front tyres mean steering angle δ_r [rad]
- Wheel speeds [rpm]
- Wheel slip ratios [κ]
- Acceleration pedal position [%]
- Brake pedal position [%]
- Chassis, drivetrain, steering, wheel, and road parametrization

All the required simulation input signals were already expressed previously in this study before. Chassis geometric parameters are the primary vehicle design parameters such as mass, inertia, a centre of gravity location, wheelbase, and wheel track. The drivetrain parametrization covers all-electric drive unit information such as maximum torque-speed map in full-load condition, drivetrain efficiency and all gearbox ratios. The steering parametrization part covers the necessary information for the kinematic map and the front mean steering angle. The wheel parametrization was information about the wheel radius. Finally, in road parametrization, the system uses the information of tyre-road grip coefficient and gravity as environmental inputs. In general, the grip coefficients were taken as 90% since this study investigates the chassis control systems in high μ conditions. Consequently, this part collects all simulation data and performs unit conversions if required for the input data management.

4.3.2 TVC calculators

Observers, also known as estimators, are often used in chassis control software systems to improve the system's safety, accuracy, and reliability. Chassis control systems use sensors to measure required inputs such as vehicle speed, vehicle sideslip, road gradient angle, etc. However, sensor data are often subject to noise and errors. Thus, using only sensors can make the chassis control challenging to respond. Observers mostly use mathematical models and Kalman filters to estimate the vehicle's state or input parameters of the chassis control system with high accuracy [57]. From a safety point of view, using sensors and observers and monitoring critical vehicle states as vehicle side slip angle can make a vehicle system reliable. Observers can help compensate for possible sensor failures by improving the system's reliability.

On the other hand, the control system can more accurately determine the vehicle's behaviour by using observers and respond appropriately. Since designing an observer is not a scope of this study, essential vehicle dynamic parameters, such as vehicle side slip angle, wheel speeds, and gradient etc., were taken directly from the simulation model. In this study, drivetrain and vehicle state calculators were used to calculating the longitudinal and lateral tyre forces and power limitations.

4.3.2.1 Drivetrain limiter

The drivetrain limiter calculates the maximum, minimum, and driver-requested values of longitudinal forces based on the EDU limitations. First, the maximum and minimum torque demands of each EDU were calculated using the acceleration and braking pedal positions, wheel speed and look-up tables of each electric motor's torque-speed characteristics. In the first part of the limiter, longitudinal request and estimation of the longitudinal tyre forces were calculated using motor torque demands.

$$F_{x,i} = \frac{T_{x,i}}{r_w} \quad (55)$$

Where r_w is the wheel radius and $T_{x,i}$ is wheel level torques. Basically $T_{x,i}$ can be found by multiplying the EDU level torques $T_{EDU,i}$ of each electric motor by the gear ratio τ_i and the gear efficiency η_j .

$$T_{x,i} = T_{EDU,i} \tau_i \eta_j \quad (56)$$

The following equations were similarly used to estimate the maximum and minimum longitudinal tyre force limitations from the drivetrain.

$$F_{x,i,max,obdrvtr} = \frac{T_{x,i,max}}{r_w} \quad (57)$$

$$F_{x,i,min,obdrvtr} = \frac{T_{x,i,min}}{r_w} \quad (58)$$

Figure 4.49 shows the general scheme of the drivetrain limiter. Second, the wheel level longitudinal torque request was calculated using minimum (generator) and maximum (e-motor) EDU torques, acceleration pedal position p_a and brake pedal position p_b . The wheel torque request was calculated as:

$$T_{req,total} = \sum_1^4 T_{EDU,max,i} p_a + \sum_1^4 T_{EDU,min,i} p_b \quad (59)$$

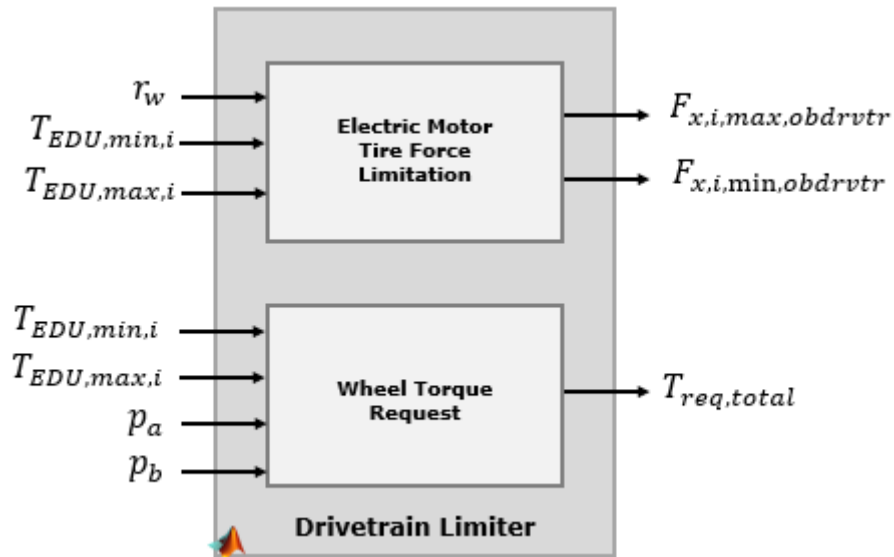


Figure 4.49: Drivetrain limiter scheme.

4.3.2.2 Vehicle states

Vehicle states calculate the maximum lateral acceleration, the potential of each tyre, and the lateral forces of each tyre. A basic scheme of the vehicle state calculator and its parts is shown in Figure 4.50. Vehicle state maximum lateral acceleration is crucial for TVC as it provides a limit on the vehicle's cornering performance. The controller needs to know the maximum lateral acceleration or capacity the vehicle can achieve without losing traction or stability.

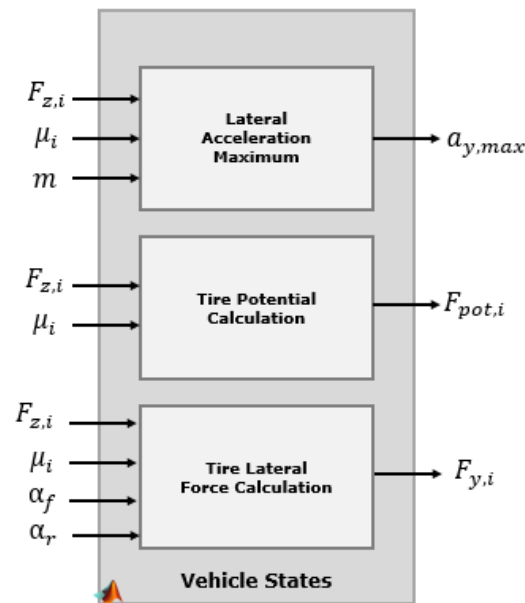


Figure 4.50: Vehicle states scheme.

First, the maximum lateral force of each tyre must be calculated to monitor the maximum lateral acceleration. The maximum lateral force generated by a tyre can be calculated using the formula:

$$F_{y,i,max} = \mu F_{z,i,sta} \quad (60)$$

where μ is the coefficient of friction between the tyre and the road surface. $F_{z,sta}$ is the stationary vertical force on the tyre. If the total vertical tyre force is considered as the total vertical force of the vehicle, then the weight of the vehicle is equal to its mass multiplied by the gravitational acceleration, as expressed in Equation 61.

$$F_{z,i,sta} = mg \quad (61)$$

Then, by adding up all the lateral tyre forces and dividing this by the total vehicle mass, the maximum lateral acceleration can be calculated:

$$a_{y,max} = \frac{\sum_i F_{y,i,max}}{m} \quad (62)$$

Then the maximum lateral acceleration can be represented by Equation 63:

$$a_{y,max} = \mu g \quad (63)$$

The second vehicle state estimator is calculation of the tyre potential. The tyre potential, known as the tyre saturation limit, refers to the maximum grip or traction a tyre can provide, as detailed previously in Section 3.3. Ideally, coefficient of friction, vertical tyre load, and tyre design/tyre modelling parameters influence tyre potential. The friction forces generated between the tyre and road surface in these contact patches are crucial for the vehicle acceleration, both longitudinally (braking and acceleration) and laterally (turning), as illustrated in Figure 4.51.

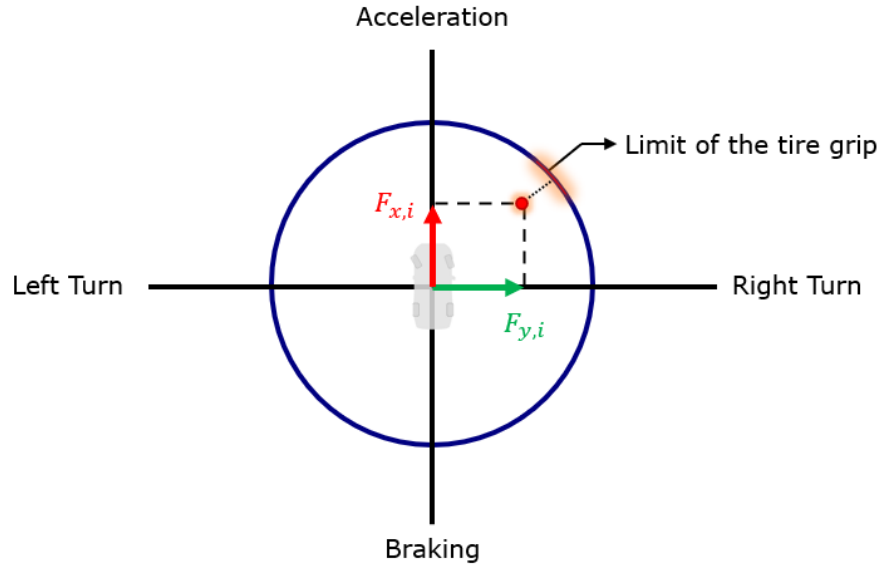


Figure 4.51: Tyre potential.

The tyre potential, or maximum grip, that a tyre can provide can be calculated using the coefficient of friction and the dynamic vertical wheel load as follows:

$$F_{pot,i} = \mu F_{z,dyn} \quad (64)$$

Ideally, each of the tyre's friction circle equations can be defined as follows:

$$\sqrt{F_x^2 - F_y^2} \leq \mu F_{z,dyn} \quad (65)$$

Figure 4.52 shows the tyre potential change during cornering for better understanding. In the left-hand turn cornering, as the vertical forces of the right tyres increase, the tyre grip capacity increases accordingly.

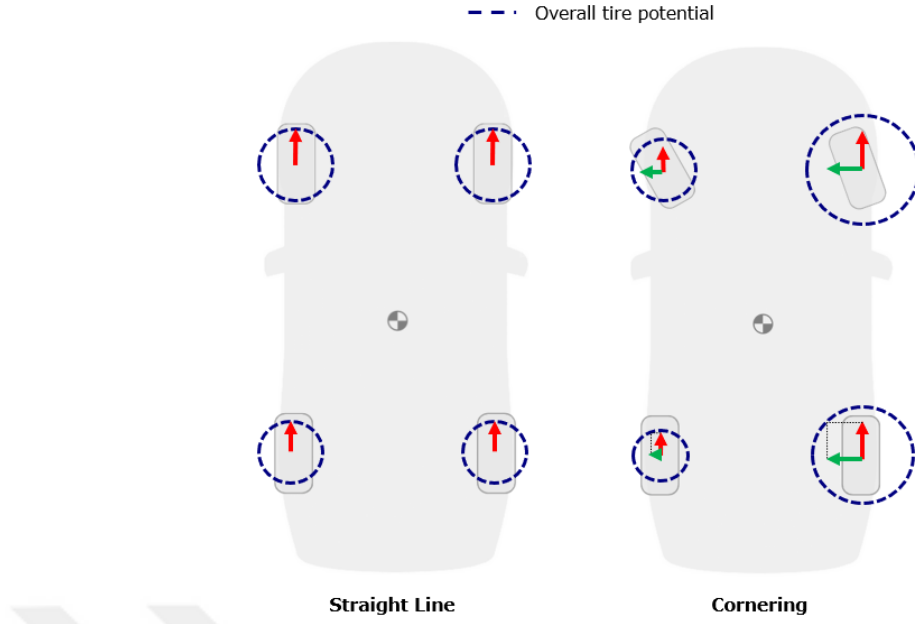


Figure 4.52: Tyre potential change during cornering.

The last vehicle state estimator is for the tyre lateral force. The lateral forces of the tyres can be calculated using vertical loads, tyre slip ratios and tyre slip angles, Pacejka tyre modelling parameters and coefficient of friction values.

$$F_{y,i} = F_{z,i} \eta_{F_y} \quad (66)$$

where η_{F_y} is the ratio of the tyre's lateral force to its vertical force and can be parametrized using the Pacejka formulations previously expressed in Section 3.3. The parameterized η_{F_y} map is shown in Figure 4.54.

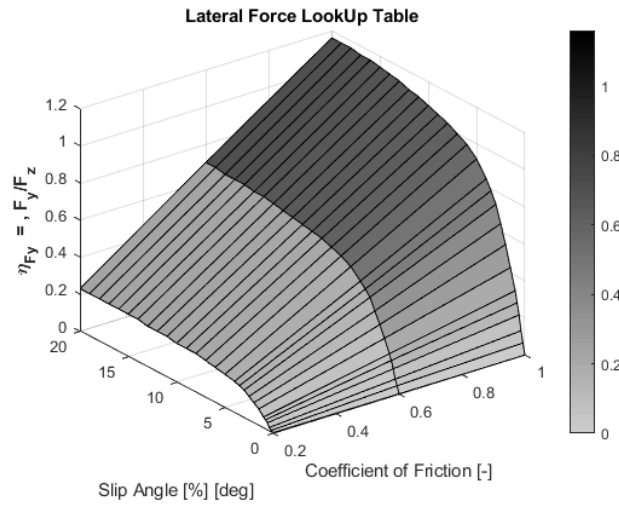


Figure 4.54: Lateral force / vertical force tyre modelling ratio map.

4.3.3 TVC longitudinal controller

Since the torque distribution controller operates based on tyre forces, the longitudinal controller must convert overall wheel torque requests into total longitudinal tyre force request. The following formula achieved the conversion of the torque request into the longitudinal tyre force request:

$$F_{x,req,total} = \frac{T_{req,total}}{r_w} \quad (67)$$

This means that the average tyre radius has an influence on the longitudinal tyre force request. The rolling wheel radius value was shown Table 3 before.

4.3.4 TVC wheel-slip controller

The goal of the wheel slip controller is to obtain a superior traction force from the optimal wheel slip (see Figure 3.2) to improve traction and driving stability during the acceleration phase and to prevent the one of the wheels from locking. This will be accomplished by a traction force limitation consisting of two subfunctions:

- WSC feedforward
- WSC feedback

The simplified interfaces and information flow of these two subfunctions are graphically displayed in Figure 4.55.

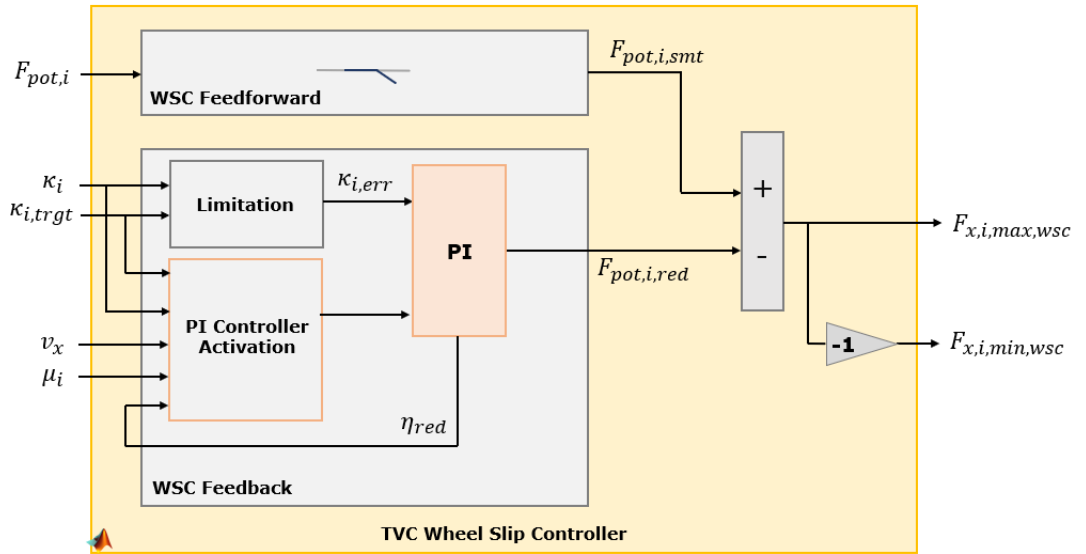


Figure 4.55: Wheel slip controller scheme.

4.3.4.1 WSC feedforward

The task of the feedforward part is to filter the tyre force potential signal with a low pass filter to get a smoothed tyre force potential signal $F_{pot,i,smt}$. It works so that the low pass filter allows the signal with frequencies below the cut-off frequency to pass attenuated, and the signals with frequencies higher than the cut-off frequency are damped. The first-order filter time constant was selected as 0.2.

4.3.4.2 WSC feedback

The task of the extended PI Controller is to calculate a reduction factor for tyre potential based on wheel slip and coefficient of friction. The first parameter is the friction coefficient that a map converts to a target wheel slip. This target wheel slip $\kappa_{i,trgt}$ and the current wheel slip κ_i were compared, and a difference, a target-current state deviation, was determined. The signal enters the PI controller via a limiter, which limits the signal upwards and downwards. The task of this control error $\kappa_{i,err}$ in the PI Controller is to represent the target-current state of wheel slip. The second parameter affects the PI Controller to indicate whether the PI Controller is enabled or not. The essential requirement is that the vehicle speed is greater than the vehicle speed threshold.

The controller switches on when the current wheel slip exceeds the target slip. It remains on until the output control is 0 and the recent wheel slip is lower than the target wheel slip. Only this allows the controller to switch off again. The braking force calculation behaves similarly only with negative wheel slips. A limiter between 0 and 100 now holds the outgoing signal of the PI controller. The signal is scaled by an operator to obtain a reduction factor between 0 and 1. When the PI-Controller is inactive, the reduction factor is 1 and does not affect the traction force limitation. The reduction factor is now multiplied by the smoothed tyre force potential, and after that, a difference between the just reduced and the smoothed tyre force potential was formed. For negative tyre slip, the signal is multiplied by -1 at the end before the signal leaves the controller. Finally, longitudinal tyre force limitations ($F_{x,i,max,wsc}, F_{x,i,min,wsc}$) are obtained from the wheel slip controller.

4.3.5 TVC lateral controller

The lateral controller aims to generate proper additional yaw moment requests considering the reference yaw rate target. This was realized in three sub-functions:

- TVC reference generator
- TVC feedforward
- TVC feedback

4.3.5.1 TVC reference generator

Generally, a control system needs a target value, such as a reference yaw rate for torque vectoring or vehicle chassis control systems, to serve as a goal or desired state for the system to achieve. The target value provides a reference against which actual system performance can be compared, and the control system can take appropriate actions to achieve the desired state. The control system can help improve vehicle handling, reduce understeer, or oversteer, and improve overall driving dynamics for better safety and performance by maintaining vehicle's dynamics close to the desired reference yaw rate. Steady-state bicycle model equations can be solved to obtain the yaw rate reference. To obtain the steering wheel angle in steady state condition ($\dot{\psi}_y=0, \dot{r}_z=0$) the equations of motion expressed in Section 2.1 can be rewritten as :

$$\delta_f = \frac{r_z l}{v_x} + K_{us} l v_x r_z \quad (68)$$

If Equation 68 is solved for yaw rate reference r_{zref} :

$$r_{zref} = \frac{v_x \delta_f}{l(1+K_{us} v_x^2)} \quad (69)$$

Here, choosing K_{us} was the critical factor for generating r_{zref} . In the literature [32], K_{us} can be defined as a function of front steering angle and lateral acceleration according to the different driver modes (see Figure 10). A similar approach was used in this part. The reference yaw rate r_{zref} was calculated as a function of front mean steering angle δ_f , vehicle speed v_x and reference understeer gradient $K_{us,linear}$. The function was designed to have neutral-steering characteristics at the lower value of lateral acceleration and increase the capacity or limit at the maximum lateral acceleration during cornering. The neutral-steering characteristic in vehicle dynamics

allows predictable and balanced handling, which can provide driving pleasure for the driver. First, the linear understeer gradient $K_{us,linear}$ is defined until a constant lateral acceleration value $a_{y,linear}$ is achieved to meet the neutral driving in the linear region of vehicle handling characteristics. After the linear region, the understeer gradient was defined as an exponential function until the maximum lateral acceleration value was $a_{y,max}$. The reference definition is graphically illustrated in Figure 4.56.

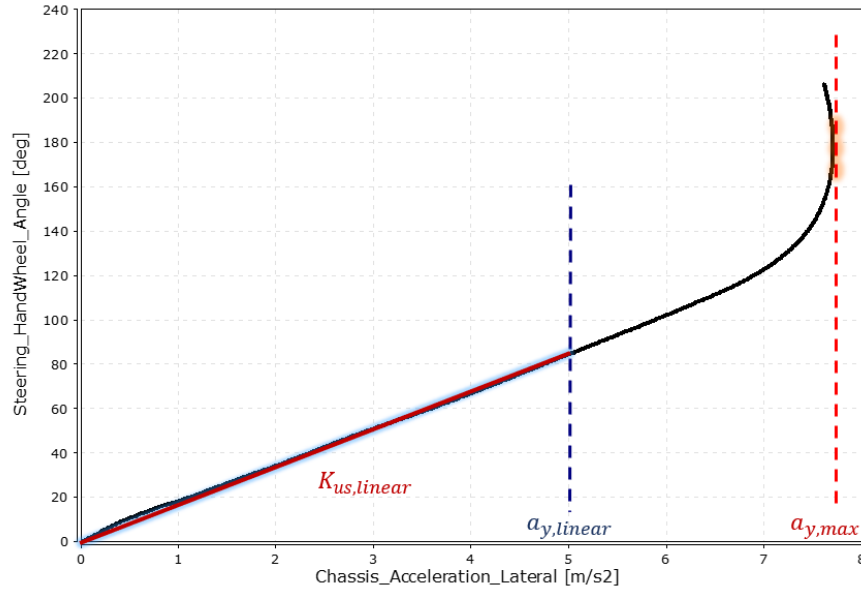


Figure 4.56: Example of reference understeer gradient curve.

The shape of the understeer gradient curve can be modified with the reference generation parameters “ n_1 ”, “ n_2 ”, “ e_1 ” and “ $K_{us,linear}$ ” [56]. The reference understeer gradient was calculated in the defined MATLAB function as follows:

$$\begin{aligned}
 & a_y^* = n_1 a_{y,max} \\
 & \text{if } a_y < a_y^* \\
 & \quad K_{us} = K_{us,linear}; \\
 & \text{else} \\
 & \quad K_{us} = K_{us,linear} + n_2 \left(\frac{a_y - a_y^*}{a_{y,max} - a_y^*} \right)^{e_1}; \\
 & \text{end}
 \end{aligned} \tag{70}$$

In the function, n_1 was used to define the lateral acceleration limit $a_{y,linear}$ value where the behaviour of the understeer gradient changes from linear to exponential. The parameters n_2 and e_1 were the shape parameters of the exponential part. $K_{us,linear}$ specified the value of desired linear understeer gradient. The reference yaw rate can be found by using the value of K_{us} value in equation 69. The understeer gradient K_{us}

must be positive to get understeer behaviour for safe driving. Therefore, saturation needs to be added to the reference yaw rate for safety reasons. The general approach [58] for the target yaw rate saturation is to use the friction coefficient, the gravitational acceleration, and the longitudinal velocity. The maximum achievable lateral acceleration is defined in Equation 63. Similarly, the maximum target yaw rate can be calculated as follows:

$$|r_{zref,max}| = \frac{\mu g}{v_x} \quad (71)$$

Lenzo [30] suggested adding a “safety factor” to limit the reference yaw rate at the safe margin. The reference generator safety factor η_{rg} was taken as 0.95 and added to the saturation of the reference yaw rate as follows:

$$|r_{zref,max}| = \eta_{rg} \frac{\mu g}{v_x} \quad (72)$$

A first-order filter with a time constant of 0.5 seconds was added to the yaw rate reference. The reference yaw rate block, its inputs and outputs are shown in Figure 4.57.

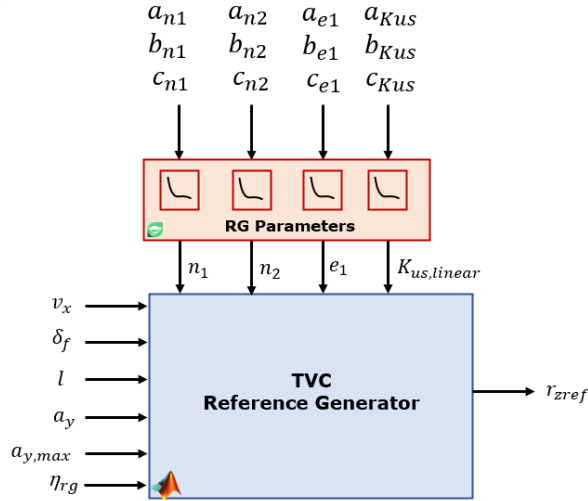


Figure 4.57: TVC reference generator block.

Here, reference generation parameters “ n_1 ”, “ n_2 ”, “ e_1 ” and “ $K_{us,linear}$ ” are tunable parameters of TVC reference generator. All tuneable parameters were parametrized as velocity-scheduled second-order polynomial equations in the reference generator part for the TVC optimization. The initial velocity-scheduled maps of the reference generator parameters are shown in Figure 4.58. The map generation codes of TVC are

shown in Appendix B. The initial values are shown in Table 10 at the end of the chapter.

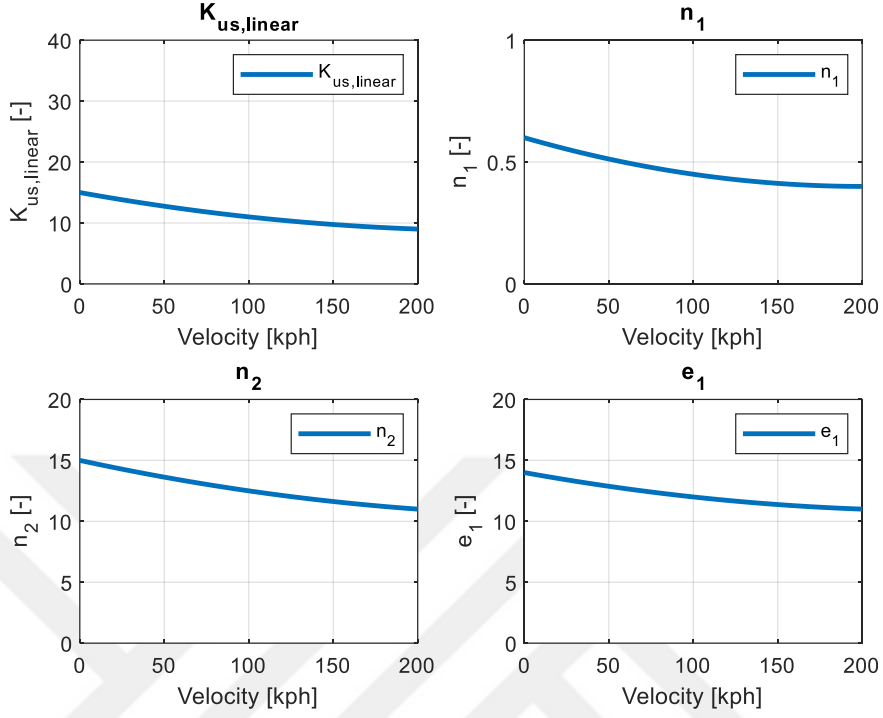


Figure 4.58: TVC reference generator parametric variables – initial values.

Second-order polynomial regression, as used in the RWS part, was used to parameterize these maps. Again, velocity scheduled maps were created by using a_{Kus} , b_{Kus} , c_{Kus} , a_{n1} , b_{n1} , c_{n1} , a_{n2} , b_{n2} , c_{n2} , a_{e1} , b_{e1} , c_{e1} parameters.

$$K_{us,linear} = c_{Kus}x^2 + b_{Kus}x + a_{Kus} \quad (73)$$

$$n_1 = c_{n1}x^2 + b_{n1}x + a_{n1} \quad (74)$$

$$n_2 = c_{n2}x^2 + b_{n2}x + a_{n2} \quad (75)$$

$$e_1 = c_{e1}x^2 + b_{e1}x + a_{e1} \quad (76)$$

Finally, an example graph was added to show the effect of ideal TVC implementation and generated yaw rate target. Figure 4.59 shows the baseline, and a vehicle with ideal TVC in a ramp steer manoeuvre at 50 kph. From the upper graph in the figure, TVC reduced the linear understeer gradient in the linear acceleration region and increased

the maximum lateral acceleration capacity. From the graph below, the ideal TVC achieved the reference yaw rate target.

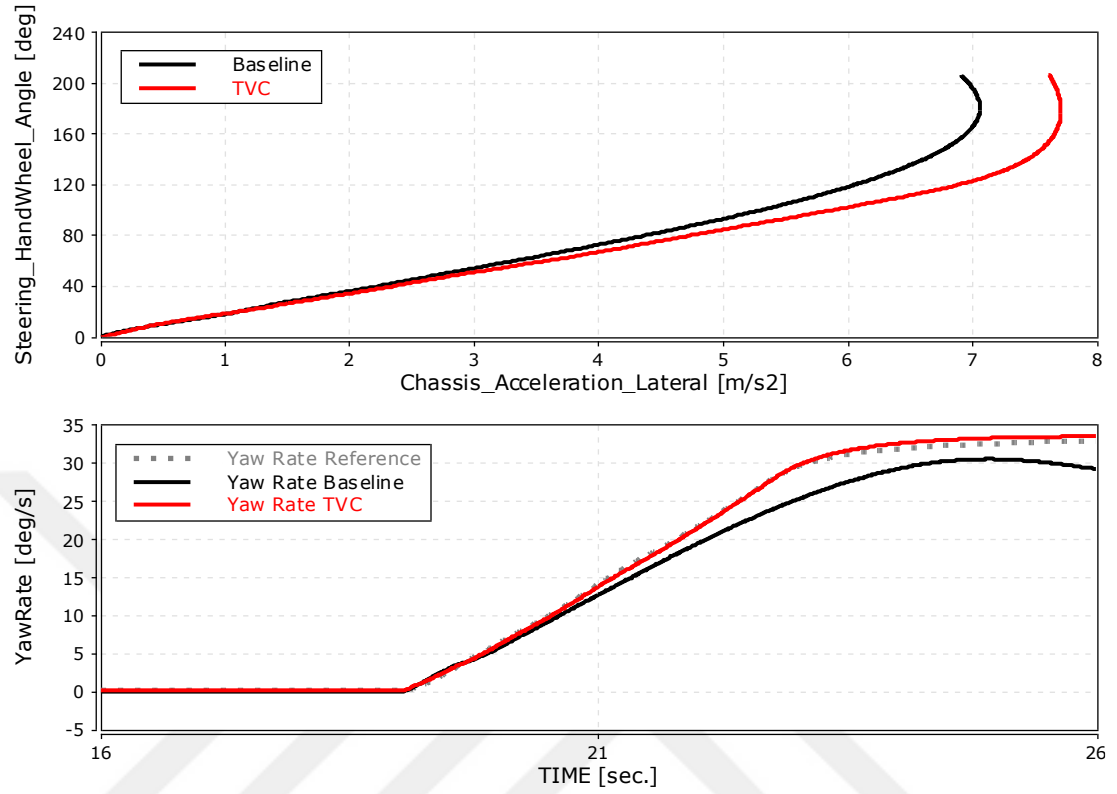


Figure 4.59: TVC reference yaw rate – baseline and TVC vehicles.

4.3.5.2 TVC feedforward

This section briefly explains the calculation of the additional yaw moment feedforward contribution. After getting the yaw rate target in a TVC, the appropriate yaw torque must be generated to meet the calculated yaw rate target. The feedforward part was determined using a mathematical model of the system to be controlled without considering the actual value of the yaw rate. The additional yaw moment that must be delivered from the feedforward can be calculated using Equations 1,12 and, setting $r_z = r_{zref}$ by using equation 69 as follows [30]:

$$M_{z,ff} = - \left(\frac{(-C_f a + C_r b) (C_f \delta_f v_x - r_{zref} (C_f a - C_r b + m v_x^2))}{v_x (m \dot{v}_x + C_f + C_r)} \right) + \frac{C_f (a^2 + b^2)}{v_x} r_{zref} - C_f a \delta_f \quad (77)$$

The feedforward yaw moment $M_{z,ff}$ was designed to minimize yaw rate error as soon as possible during steady state and transient manoeuvre without significant differences in handling characteristics. During the TVC design process, only the feedforward part was tried first; however, using only a feedforward controller does not guarantee an ideal yaw rate, as represented in Figure 4.60. Therefore, the feedforward part only needs to be improved. The vehicle can follow the yaw rate reference track more perfectly by using feedforward and feedback. Thus, it is important to add the feedback control contribution to the system together. Another solution for extreme transients would be to add a dynamic transient feedforward part in the Laplace domain [60]. This solution proposal is very similar to the feedforward transient part used in the RWS controller in this study. However, as can be seen in Figure 4.59, feedback controller contribution was chosen to prevent undesired yaw rate oscillations and overshoots because initial trials of feedforward plus feedback contribution were satisfactory in open-loop test cases.

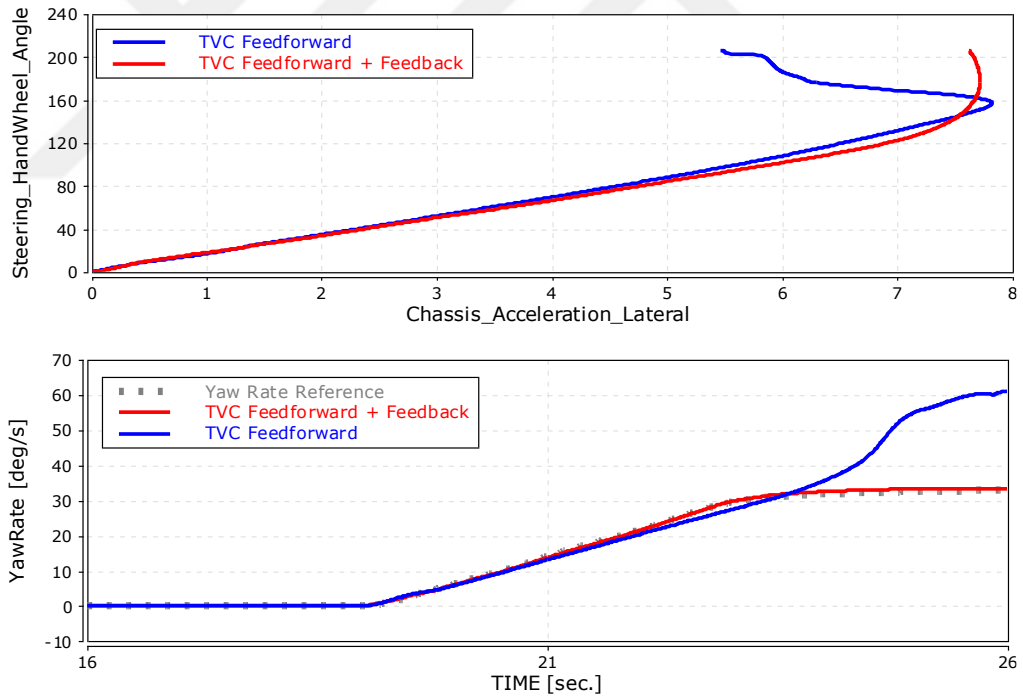


Figure 4.60: TVC only feedforward vs feedforward and feedback.

4.3.5.3 TVC feedback

A feedback control action was calculated based on the yaw rate error, that is, the difference between the yaw rate reference and the actual yaw rate, as follows:

$$r(t)_{z,e} = r(t)_{zref} - r(t)_z \quad (78)$$

For the feedback mechanism, the proportional integral derivative (PID) controller is a common approach in torque vectoring control system developments [48]. The feedback contribution of the additional yaw moment was modelled in the time domain as a PID controller as follows:

$$M(t)_{z,fb} = K_p r(t)_{z,e} + K_i \int_0^t r(t)_{z,e} dt + K_d \frac{dr(t)_{z,e}}{dt} \quad (79)$$

After adding the feedback part for the output of the TVC lateral controller, the total additional yaw moment can be calculated as follows:

$$M_z = M_{z,ff} + M_{z,fb} \quad (80)$$

In general, when an integral term is used, appropriate reset conditions must be included [30]. When the PID controller unable to control the process variable towards the setpoint due to physical constraints such as limits on actuator saturation, the integrator term of the controller can continue to integrate the error, leading to an overshoot or prolonged settling time once the boundaries are no longer in effect. This can cause poor system performance and instability. The anti-windup feature is designed to mitigate this issue by limiting or clamping the integrator term when the system is saturated, by preventing it from accumulating excessive error. There are several approaches to implementing anti-windup in a PID controller, such as back-calculation, clamping, and conditional integration, among others. Back-calculation, also known as “integrator reset” or “integrator tracking”, was used in this study. Thus, if the control effort exceeds the saturation limits, the integrator term is modified using the back-calculation approach. The modification was done by subtracting the saturation limit from the integral gain and then multiplying the result by the integrator output. The anti-windup factor K_{aw} was modeled as K_p/K_i in the back-calculation part. General scheme and anti-windup details were represented in Figure 4.61.

PID parameters can be optimised in many ways. First, the PID controller gains should be investigated in the frequency domain. Tuning PID gain parameters K_p , K_i , K_d in the frequency domain using open-loop cases involves analysing the system's frequency response without feedback control and adjusting the PID gains based on the open-loop

characteristics of the system. This method is also known as "gain and phase margin" tuning, where the goal is to achieve a certain amount of gain and phase margin at specific frequencies to ensure system stability and robustness. However, this way can only be a solution if a linearised vehicle plant model is used. When you have a non-linear vehicle/plant model (as in this study), there may be a different solution. Another way would be to use the well-known Ziegler-Nichols method. However, even with the Ziegler-Nichols method, it cannot be guaranteed that the system will behave properly in the non-linear region of the tyres. Therefore, the PID gains can be optimized with the proposed methodology in this study. To do that, again PID gain parameters K_p, K_i, K_d were parametrized as velocity-scheduled fourth order polynomial equation and acceleration-scheduled quadratic ansatz function. In parameterising these gains, 5 parameters were used to generate velocity scheduled contribution and 3 parameters were used to generate acceleration scheduled contribution. Then, velocity and acceleration scheduled contributions were multiplied, as shown in Figure 87.

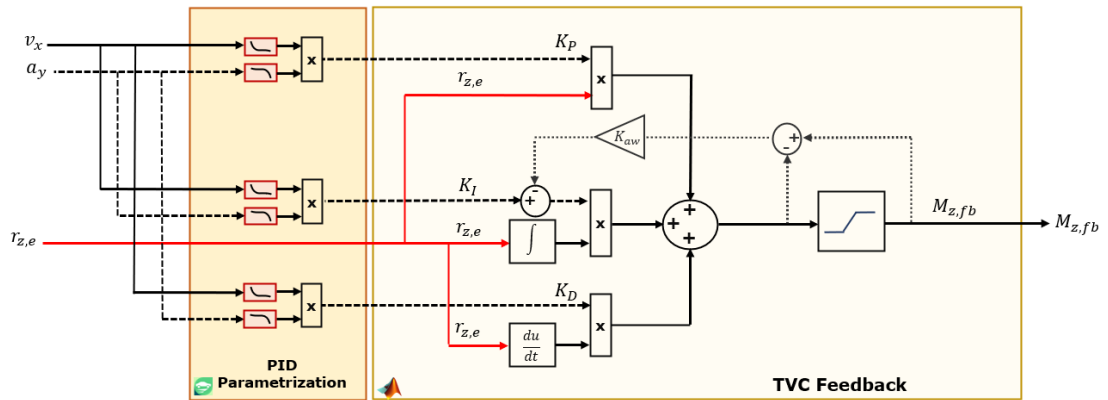


Figure 4.61: TVC feedback scheme.

Velocity-scheduled fourth-order polynomial equations were as follows for each gain:

$$K_{p,V} = k_{p,V,1}x^4 + k_{p,V,2}x^3 + k_{p,V,3}x^2 + k_{p,V,4}x + k_{p,V,5} \quad (81)$$

$$K_{i,V} = k_{i,V,1}x^4 + k_{i,V,2}x^3 + k_{i,V,3}x^2 + k_{i,V,4}x + k_{i,V,5} \quad (82)$$

$$K_{d,V} = k_{d,V,1}x^4 + k_{d,V,2}x^3 + k_{d,V,3}x^2 + k_{d,V,4}x + k_{d,V,5} \quad (83)$$

Acceleration-scheduled functions were defined as follows for each gain:

$$K_{P,A} = \begin{cases} (k_{P,A,1} + k_{P,A,2})^2 & a_y \leq k_{P,A,ble} \\ (k_{P,A,2})^2 & a_y > k_{P,A,ble} \end{cases} \quad (84)$$

$$K_{I,A} = \begin{cases} (k_{I,A,1} + k_{I,A,2})^2 & a_y \leq k_{I,A,ble} \\ (k_{I,A,2})^2 & a_y > k_{I,A,ble} \end{cases} \quad (85)$$

$$K_{D,A} = \begin{cases} (k_{D,A,1} + k_{D,A,2})^2 & a_y \leq k_{D,A,ble} \\ (k_{D,A,2})^2 & a_y > k_{D,A,ble} \end{cases} \quad (86)$$

Final feedback gains can be calculated using equations 80-85 as follows:

$$K_P = K_{P,V} K_{P,A} \quad (87)$$

$$K_I = K_{I,V} K_{I,A} \quad (88)$$

$$K_D = K_{D,V} K_{D,A} \quad (89)$$

All tuneable parameters of TVC feedback gains generation are listed in Table 4.7 with their initial values. Initial velocity-scheduled and acceleration-scheduled maps of the feedback gains are shown in Figure 4.62. The map generation codes of TVC are shown in Appendix B. The initial values are shown in Table 4.7 at the end of the chapter.

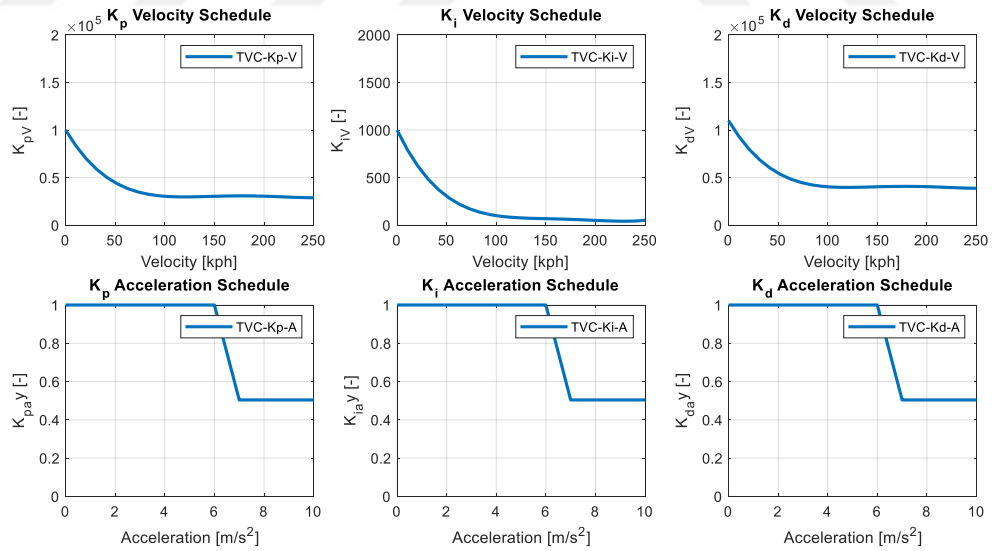


Figure 4.62: TVC feedback scheme.

The limits of the TVC feedback gains were selected through open-loop tests by simulating the vehicle with a ramp steer manoeuvre. According to the open-loop test results, the maximum and minimum limits of K_P were determined as 200000 and 0, respectively. The maximum and minimum limits of K_I were selected as 2000 and 0,

respectively. Finally, the maximum and minimum limits of K_D were selected as 200000 and 0 respectively. If TVC is implemented later another vehicle model, the control engineer should check these parameters. All TVC parametric variables in Table 4.7 were optimized using DoE for different drive modes as was done in the RWS controller.

Table 4.7: TVC parametric variables - initial values.

TVC Parametric Variables	TVC Part	Initial Values
a_{Kus}	Reference Generator	15
b_{Kus}	Reference Generator	-10
c_{Kus}	Reference Generator	4
a_{n1}	Reference Generator	0.6
b_{n1}	Reference Generator	-0.4
c_{n1}	Reference Generator	0.2
a_{n2}	Reference Generator	15
b_{n2}	Reference Generator	-6
c_{n2}	Reference Generator	2
a_{e1}	Reference Generator	14
b_{e1}	Reference Generator	-5
c_{e1}	Reference Generator	2
$k_{P,V,1}$	Feedback – Kp	8.2e-5
$k_{P,V,2}$	Feedback – Kp	-0.06
$k_{P,V,3}$	Feedback – Kp	15.75
$k_{P,V,4}$	Feedback – Kp	-1754.5
$k_{P,V,5}$	Feedback – Kp	100000
$k_{P,A,1}$	Feedback – Kp	0.29
$k_{P,A,2}$	Feedback – Kp	0.71
$k_{P,A,ble}$	Feedback – Kp	7
$k_{I,V,1}$	Feedback – Ki	1.079e-6
$k_{I,V,2}$	Feedback – Ki	-7.506e-4
$k_{I,V,3}$	Feedback – Ki	0.1921
$k_{I,V,4}$	Feedback – Ki	-21.783
$k_{I,V,5}$	Feedback – Ki	1000
$k_{I,A,1}$	Feedback – Ki	0.29
$k_{I,A,2}$	Feedback – Ki	0.71
$k_{I,A,ble}$	Feedback – Ki	7
$k_{D,V,1}$	Feedback – Kd	8.2e-5
$k_{D,V,2}$	Feedback – Kd	-0.06
$k_{D,V,3}$	Feedback – Kd	15.75
$k_{D,V,4}$	Feedback – Kd	-1753.5
$k_{D,V,5}$	Feedback – Kd	110000
$k_{D,A,1}$	Feedback – Kd	0.29
$k_{D,A,2}$	Feedback – Kd	0.71
$k_{D,A,ble}$	Feedback – Kd	7

4.3.6 ICLC coordinator

The vehicle motion control system corrects the additional yaw moment by adjusting its weighting factor via TVC weighting factor η_{TVC} . Parameterization and modelling details of η_{TVC} were expressed in the ICLC section. The corrected additional yaw moment $M_{z,c}$ to be sent to the last part of the TVC controller was calculated as follows:

$$M_{z,c} = \eta_{RWS} M_z \quad (90)$$

4.3.7 TVC torque distribution controller

A torque distribution controller, or low-level controller, calculates electric motors torques based on longitudinal and lateral requests. It considers the maximum force values of friction, wheel slip, and drivetrain power for controlling and actuating the electric motors. The system first collected the TVC longitudinal and TVC lateral controller outputs, total longitudinal force request $F_{x,req,total}$ and corrected additional yaw moment $M_{z,c}$, respectively. In this study, both longitudinal and lateral torque split functions were used. Lateral torque split is only applied to the rear electric motors. When an additional yaw moment was applied to the vehicle, a torque difference occurred at the rear left and right wheels. The longitudinal force difference on the rear left and right wheel can be defined as follows:

$$\Delta F_{x,LR} = \frac{M_{z,c}}{d} \quad (91)$$

where d is the rear wheel track. Then, if the total longitudinal force request is known, each wheel's longitudinal force request can be calculated as:

$$\begin{cases} F_{x,FL,req} = \left(\frac{F_{x,req,total}}{2} \right) \sigma \\ F_{x,FR,req} = \left(\frac{F_{x,req,total}}{2} \right) \sigma \\ F_{x,LR,req} = \left(\frac{F_{x,req,total}}{2} - \Delta F_{x,LR} \right) (1-\sigma) \\ F_{x,RR,req} = \left(\frac{F_{x,req,total}}{2} + \Delta F_{x,LR} \right) (1-\sigma) \end{cases} \quad (92)$$

Where σ is the longitudinal torque split factor. According to the literature, the general optimal longitudinal split solution is even distribution [61],[62]. Thus, in this study σ was taken as 0.5. While the torques generated from the front and rear axle electric motors shown in Figure 25 were uneven, in the low-level controller the torques were evenly distributed between the front and rear.

After calculating each wheel's longitudinal force request, the demand was controlled with 3 inputs. The demand comparison inputs are:

- Wheel slip controller limitations: $F_{x,i,max,wsc}, F_{x,i,min,wsc}$
- Drivetrain power limitations: $F_{x,i,max,obdrvtr}, F_{x,i,min,obdrvtr}$
- Tyre saturation limitations: $F_{x,i,max,tire}, F_{x,i,min,tire}$

Wheel slip controller and drivetrain limitations were previously calculated in the TVC calculator section. The friction circle calculation previously expressed in equation 65 was used to calculate the front tyre's longitudinal force capacity. From the vehicle state calculation, each tyre's potential and lateral forces were already known. Thus, the longitudinal force limit can be calculated as follows (see Figure 77):

$$F_{x,i,max,tire} = \begin{cases} \sqrt{F_{pot,i}^2 - F_{y,i}^2} & F_{y,i} < F_{pot,i} \\ 0 & F_{y,i} \geq F_{pot,i} \end{cases} \quad (93)$$

$$F_{x,i,min,tire} = \begin{cases} -\sqrt{F_{pot,i}^2 - F_{y,i}^2} & F_{y,i} < F_{pot,i} \\ 0 & F_{y,i} \geq F_{pot,i} \end{cases} \quad (94)$$

The final maximum limits of each tyre can then be calculated by taking minimum values of wheel slip controller, drivetrain power and tyre saturations limitations as follows:

$$F_{x,FL,max} = \min (F_{x,FL,max,wsc}, F_{x,FL,max,obdrvtr}, F_{x,FL,max,tire}) \quad (95)$$

$$F_{x,FR,max} = \min (F_{x,FR,max,wsc}, F_{x,FR,max,obdrvtr}, F_{x,FR,max,tire}) \quad (96)$$

$$F_{x,RL,max} = \min (F_{x,RL,max,wsc}, F_{x,RL,max,obdrvtr}, F_{x,RL,max,tire}) \quad (97)$$

$$F_{x,RR,max} = \min (F_{x,RR,max,wsc}, F_{x,RR,max,obdrvtr}, F_{x,RR,max,tire}) \quad (98)$$

In this way, excessive torque that cannot or should not be given to the wheels (due to wheel slip) was prevented. A similar approach can be used to find minimum limits. The final minimum limits of each tyre can be calculated by taking the maximum values of the wheel slip controller, drivetrain power and tyre saturations limitations as follows:

$$F_{x,FL,min} = \max (F_{x,FL,min,wsc}, F_{x,FL,min,obdrvtr}, F_{x,FL,min,tire}) \quad (99)$$

$$F_{x,FR,min} = \max (F_{x,FR,min,wsc}, F_{x,FR,min,obdrvtr}, F_{x,FR,min,tire}) \quad (100)$$

$$F_{x,RL,min} = \max (F_{x,RL,min,wsc}, F_{x,RL,min,obdrvtr}, F_{x,RL,min,tire}) \quad (101)$$

$$F_{x,RR,min} = \max (F_{x,RR,min,wsc}, F_{x,RR,min,obdrvtr}, F_{x,RR,min,tire}) \quad (102)$$

Finally, each tyres' the longitudinal force requests of each tyre $F_{x,FL,req}$, $F_{x,FR,req}$, $F_{x,RL,req}$ and $F_{x,RR,req}$ were saturated using the final maximum and minimum longitudinal force limits $F_{x,FL,max}$, $F_{x,FR,max}$, $F_{x,RL,max}$, $F_{x,RR,max}$, $F_{x,FL,min}$, $F_{x,FR,min}$, $F_{x,RL,min}$ and $F_{x,RR,min}$.

It is used directly, if the request is below these limits, otherwise, maximum, or minimum limits are used for unsatisfied demand for each tyre. The limited longitudinal forces of each tyre were named $F_{x,FL,lim}$, $F_{x,FR,lim}$, $F_{x,RL,lim}$ and $F_{x,RR,lim}$. The longitudinal wheel force demands were then converted to wheel torques using the following equation:

$$T_{x,i,limit} = F_{x,i,limit} r_w \quad (103)$$

EDU level torques were calculated using gear ratios and gear efficiencies as follows:

$$T_{EDU,i,limit} = \frac{T_{x,i,limit}}{\tau_i \eta_j} \quad (104)$$

Lastly, each electric motor torque was distributed in the last part. Figure 4.63 shows all processes performed on the torque distribution controller.

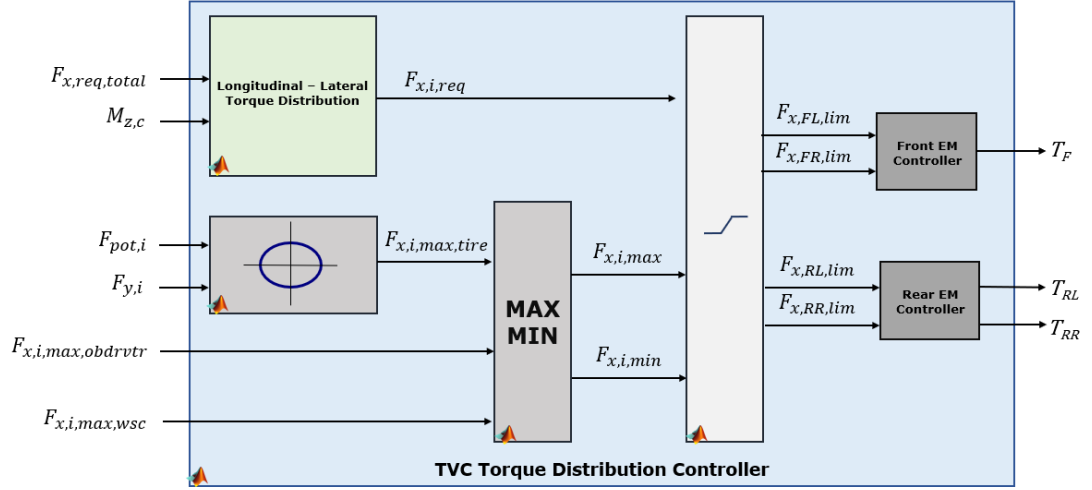


Figure 4.63: TVC torque distribution controller scheme.

4.4 TVC Optimization and MIL Evaluation

In this section, the same approach was used that was proposed in the RWS section earlier. Again, in MATLAB/Simulink, TVC optimization was performed using DoE tool AVL CAMEO, vehicle model AVL VSM, objective evaluation tool MATLAB and TVC control model.

4.4.1 Objective evaluations and test cases

Almost the same manoeuvres used in the RWS part were used for TVC optimization. Additionally, constant radius cornering manoeuvre has been added to the objective evaluation of torque vectoring. As the methodology was the same, the aim was to optimize TVC parametric variables listed in Table 4.7 in all high-mu test conditions. All TVC optimization aims are listed in Table 4.8.

Table 4.8: TVC optimization aims of test cases.

Manoeuvre	Speed [kph]	Main Aim
Ramp steer	50	Decrease K_{us} + Increase $a_{y,max}$
Ramp steer	100	Decrease K_{us} + Increase $a_{y,max}$
Ramp steer	150	Decrease K_{us} + Increase $a_{y,max}$
Step steer	100	Minimize a_y overshoot
Double-Lane Change	55	Minimize β
Constant Radius Cornering	100	Decrease K_{us} + Increase $a_{y,max}$

The main objectives of the ramp steer manoeuvre were different than RWS objectives, as torque vectoring was designed to make the vehicle more neutral in the linear acceleration range and increase the vehicle's cornering capacity. In other words, while vehicle stability was important in RWS controller design, driver desires and vehicle limits were considered more important in TVC design and optimization. The aim is therefore to decrease the linear understeer gradient and increase the maximum achievable lateral acceleration in ramp steer and constant radius cornering manoeuvre. Objective measurement results were obtained through post-processing in MATLAB after each AVL VSM simulation.

4.4.1.1 Ramp steer manoeuvre

Again, there were three different ramp steer manoeuvres defined by vehicle speeds of 50, 100 and 150 kph vehicle speeds. Since the details are already mentioned in the RWS section, only the objective functions were defined in this section for the ramp steer manoeuvre. The understeer gradient objective metrics of the TVC in the ramp steer case are as follows:

$$\begin{aligned} f_{TVC,RS,50,K_{us}} &= \text{minimize } K_{us} \\ &\text{subject to } |\alpha_f| - |\alpha_r| < -0.1 \end{aligned} \quad (103)$$

$$\begin{aligned} f_{TVC,RS,100,K_{us}} &= \text{maximize } K_{us} \\ &\text{subject to } |\alpha_f| - |\alpha_r| < -0.1 \end{aligned} \quad (104)$$

$$\begin{aligned} f_{TVC,RS,150,K_{us}} &= \text{maximize } K_{us} \\ &\text{subject to } |\alpha_f| - |\alpha_r| < -0.1 \end{aligned} \quad (105)$$

And the maximum lateral acceleration objective functions:

$$\begin{aligned} f_{TVC,RS,50,a_y} &= \text{maximize } a_y \\ &\text{subject to } |\alpha_f| - |\alpha_r| < -0.1 \end{aligned} \quad (106)$$

$$\begin{aligned} f_{TVC,RS,100,a_y} &= \text{maximize } a_y \\ &\text{subject to } |\alpha_f| - |\alpha_r| < -0.1 \end{aligned} \quad (107)$$

$$\begin{aligned} f_{TVC,RS,150,a_y} &= \text{maximize } a_y \\ &\text{subject to } |\alpha_f| - |\alpha_r| < -0.1 \end{aligned} \quad (108)$$

4.4.1.2 Step steer manoeuvre

A step steer steering input was applied when the vehicle was at 100 kph. The main aim was to decrease overshoots and peak value in this scenario again.

$$\begin{aligned}
f_{TVC,SS} = & \text{minimize } a_{y,peak} - a_{y,steady-state} \\
& \text{subject to } 3 < a_{y,ss} < 5
\end{aligned} \tag{109}$$

4.4.1.3 Double lane change manoeuvre

Similarly, the same conditions were run in the DLC test case that already ran in the RWS section. As discussed earlier, driver skills were crucial for vehicle behaviour in this case. The same driver skills were used to compare the chassis control systems and drive modes more precisely. The minimum side slip angle in the DLC scenario was measured as an objective with an entry speed of 55 kph:

$$\begin{aligned}
f_{TVC,DLC} = & \text{minimize } \beta_{MAX} \\
& \text{subject to } -45 < \beta_{MAX} < 45
\end{aligned} \tag{110}$$

4.4.1.4 Constant radius cornering manoeuvre

Constant radius cornering (CRC), also called constant radius turn (CRT), is a close loop test case in which the vehicle is driven on a constant radius ($R = 40 \text{ m}$) circular track and the vehicle speed is slowly increased ($a_x = 0.2 \text{ m/s}^2$) as represented in Figure 4.64. In this test, one of the known benefits of the torque vectoring system is that it allows the vehicle to increase the maximum lateral acceleration and extension of the region of linear cornering response for the baseline vehicle in a constant radius turn scenario. Thus, the simplest form of the linear understeer gradient and lateral acceleration objective functions were then expressed as follows:

$$\begin{aligned}
f_{TVC,CRC,Kus} = & \text{minimize } K_{us} \\
& \text{subject to } |\alpha_f| - |\alpha_r| < -0.1
\end{aligned} \tag{111}$$

$$\begin{aligned}
f_{TVC,CRC,a_y} = & \text{maximize } a_y \\
& \text{subject to } |\alpha_f| - |\alpha_r| < -0.1
\end{aligned} \tag{112}$$

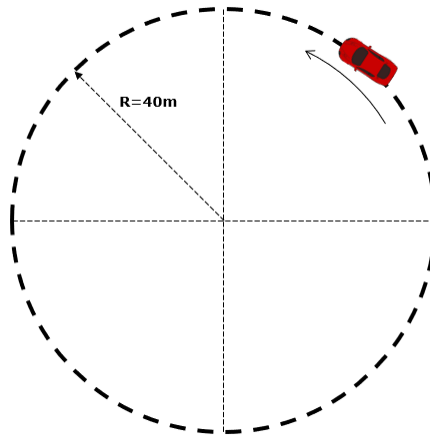


Figure 4.64: Constant radius cornering manoeuvre.

4.4.2 DoE design and optimization toolchain

In the previous section, parametric variables were to be optimized and the objective functions of the TVC system were determined. There are 36 parameters to be optimized for 10 objective functions with 4 test cases. The main purpose of the TVC optimization methodology is to obtain optimum control parameters which work under all test conditions. All biases in the multi-objective optimization design were taken the same in the TVC optimization procedure. Therefore, there was no need to define the bias in the optimization problem. The overall TVC optimization scheme is shown in Figure 4.65. MATLAB post-processing scripts were used to automatically calculate objective function metrics after each simulation. The calculated objective metrics/responses were sent to the DoE tool of AVL CAMEO. In AVL CAMEO, parametric variables were changed and selected according to the DoE design. In each sequence run, selected parametric variables were sent to the TVC control system in MATLAB Simulink. Thus, all simulations were run with variable control parameters. AVL CAMEO send variable feedback to parametric variables to the TVC feedback system and send variable reference generator parametric variables to the reference generator system in each sequence of run.

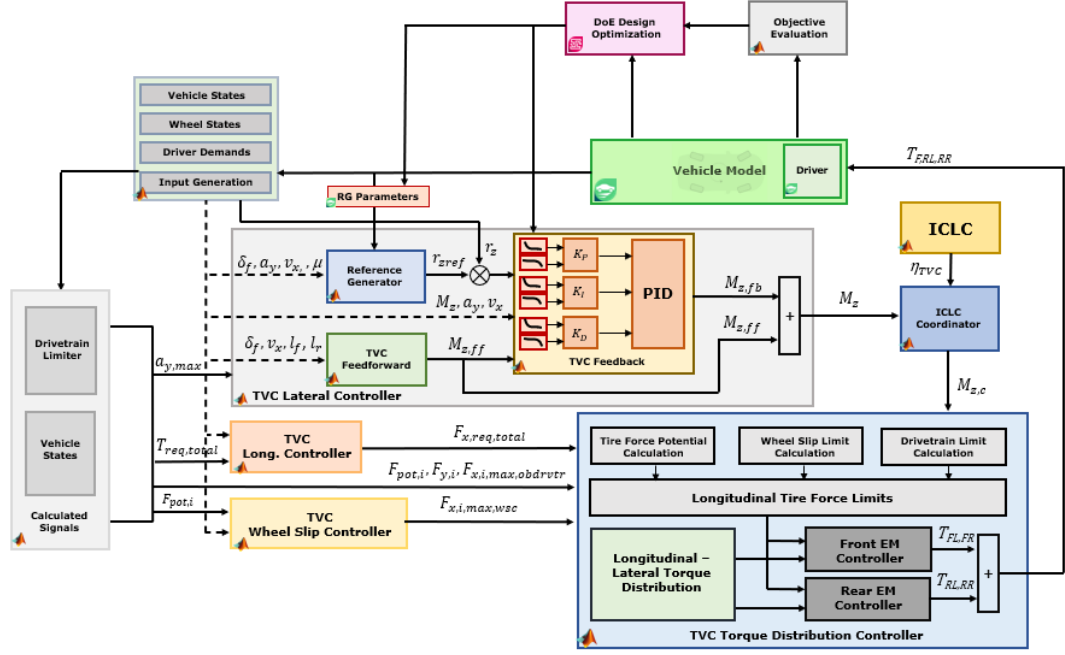


Figure 4.65: TVC scheme with optimization tools.

The Latin hypercube sampling design was selected for the TVC optimization DoE design because it allows to optimize up to 36 variations. This is a space-filling design especially suited to Neural Networks and responses with strong nonlinear behaviour, as shown in Figure 4.66. The user must choose a sufficiently high number of design points. Thus, 1000 design points were selected for each objective function.

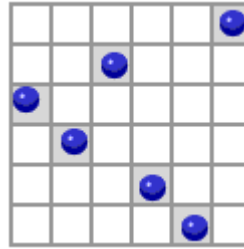


Figure 4.66: Latin Hypercube Sampling DoE design.

There were 10 objective functions; this means a total of 10000 simulations were run to optimize the TVC control system. In other words, the loop shown in Figure 4.65 ran 10000 times with different parametric variables for different test scenarios defined. Constraints of the variations and limits of the optimization points were selected according to the drive mode demands. TVC driving modes are shown in Figure 2.10. According to Figure 2.10, the maximum performance improvement that can be achieved in the TVC sport mode was aimed. In the comfort mode of the TVC, a more

“balanced” approach was chosen. In other words, it was aimed to reduce the linear understeer coefficient as much as possible in sport mode, while the minimum level was limited in comfort mode.

After the offline optimization procedure, all test results were evaluated in AVL CAMEO. According to the results of regression analysis in Figure 4.67, only the double lane change scenario could not give reasonable results. The DLC scenario R^2 was 0.7491, which was smaller than 0.8 and was considered an unsuccessful estimation. A very similar result was previously obtained in the RWS section. The DLC case was transient and depended on the driver’s performance. Objectively evaluating a highly subjective scenario, which depends on driver performance and feeling, can be an approach, but not conclusive. Nevertheless, regardless of the high or low quality of estimations, the final tests were done, and the final tuning was completed to verify and complete the TVC optimization. Table 4.9 shows the optimized parameters of TVC comfort and TVC sport modes.

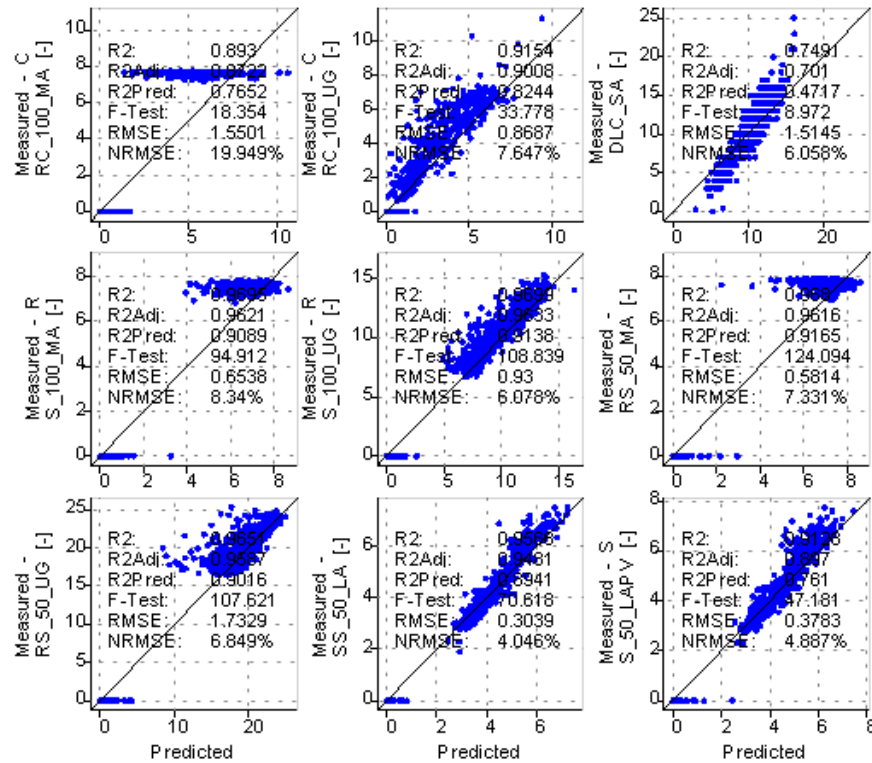


Figure 4.67: DoE estimation results of TVC optimization

Table 4.9: TVC different drive modes parametric variables.

TVC Parametric Variables	TVC Part	TVC Comfort Mode	TVC Sport Mode
a_{kus}	Reference Generator	25.3	24.4
b_{kus}	Reference Generator	-47.3	-53.8
c_{kus}	Reference Generator	25.2	31.1
a_{n1}	Reference Generator	0.80	0.61
b_{n1}	Reference Generator	-0.41	-0.40
c_{n1}	Reference Generator	0.21	0.20
a_{n2}	Reference Generator	12.1	14.2
b_{n2}	Reference Generator	-3.2	-3.6
c_{n2}	Reference Generator	1.6	1.0
a_{e1}	Reference Generator	7.0	10.3
b_{e1}	Reference Generator	-4.6	-4.0
c_{e1}	Reference Generator	2.0	2.1
$k_{p,v,1}$	Feedback – Kp	-0.00002	0.00007
$k_{p,v,2}$	Feedback – Kp	0.00468	-0.03333
$k_{p,v,3}$	Feedback – Kp	0.06056	3.83333
$k_{p,v,4}$	Feedback – Kp	-72.13420	-116.66667
$k_{p,v,5}$	Feedback – Kp	92100	101500
$k_{p,a,1}$	Feedback – Kp	0.31	0.34
$k_{p,a,2}$	Feedback – Kp	0.69	0.66
$k_{p,a,ble}$	Feedback – Kp	5.8	6.8
$k_{i,v,1}$	Feedback – Ki	0.000002	-0.000003
$k_{i,v,2}$	Feedback – Ki	-0.001154	0.001715
$k_{i,v,3}$	Feedback – Ki	0.147563	-0.313056
$k_{i,v,4}$	Feedback – Ki	-6.784593	11.693122
$k_{i,v,5}$	Feedback – Ki	1500	1980
$k_{i,a,1}$	Feedback – Ki	0.44	0.40
$k_{i,a,2}$	Feedback – Ki	0.56	0.60
$k_{i,a,ble}$	Feedback – Ki	5.0	5.4
$k_{d,v,1}$	Feedback – Kd	-0.000018	-0.000032
$k_{d,v,2}$	Feedback – Kd	0.003273	0.015407
$k_{d,v,3}$	Feedback – Kd	-0.172727	-3.577778
$k_{d,v,4}$	Feedback – Kd	2.727273	-3.439153
$k_{d,v,5}$	Feedback – Kd	91600	109565
$k_{d,a,1}$	Feedback – Kd	0.63	0.58
$k_{d,a,2}$	Feedback – Kd	0.37	0.42
$k_{d,a,ble}$	Feedback – Kd	4.1	6.2

In Figures 68, 69, 70 and 71 show graphical representations of the reference generator and feedback parameters of the comfort and sport driving modes of TVC, respectively.

TVC Comfort - Reference Generator Parameters

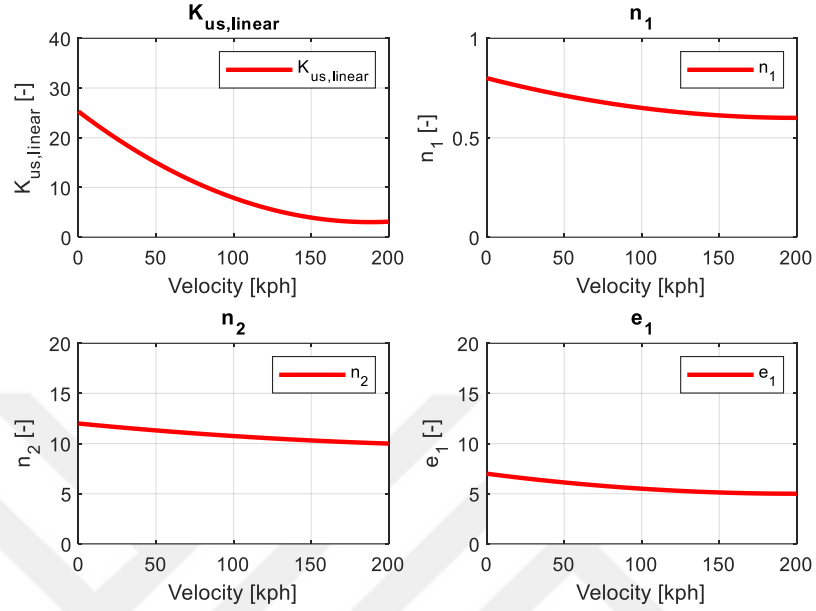


Figure 4.68: TVC results - reference generator parameters maps - comfort.

TVC Comfort - Feedback Parameters

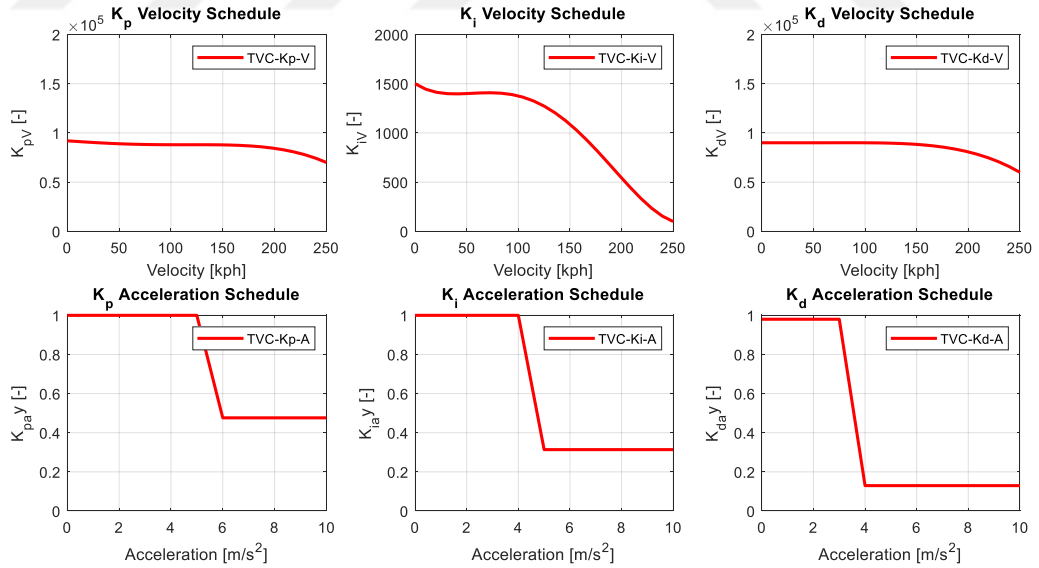


Figure 4.69: TVC results - feedback parameters maps - comfort.

TVC Sport - Reference Generator Parameters

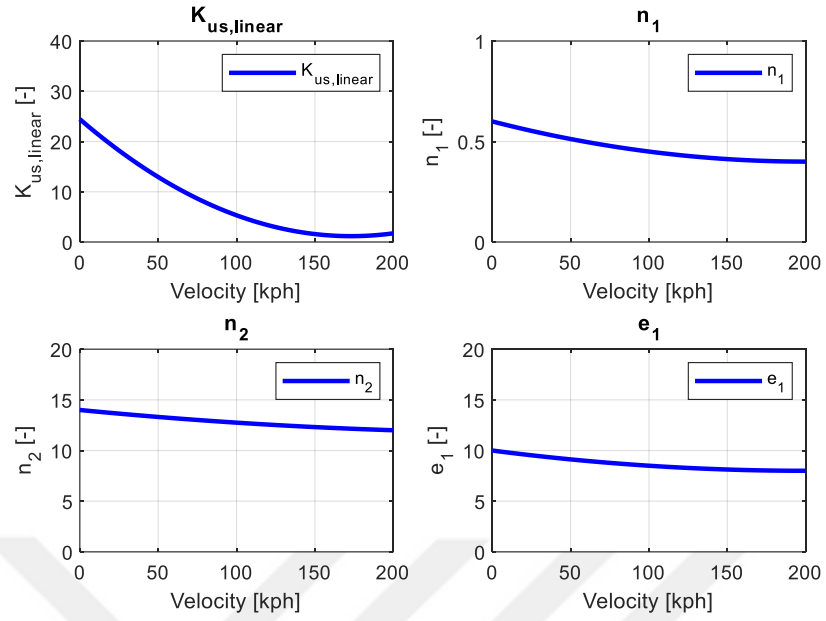


Figure 4.70: TVC results - reference generator parameters maps - comfort.

TVC Sport - Feedback Parameters

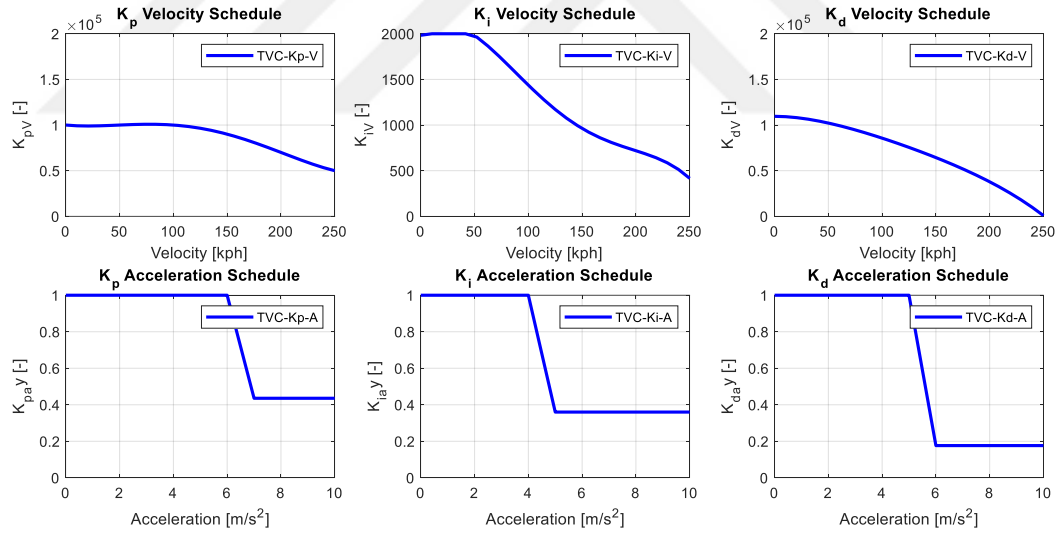


Figure 4.71: TVC results - feedback parameters maps - comfort.

4.4.3 TVC evaluation and optimized results

In this section, comfort, and sport TVC driving modes are tested with AVL VSM simulations. Before each simulation started, the driving mode was selected in the AVL VSM user interface.

4.4.3.1 TVC results - ramp steer - 50 kph

First, the two TVC drive modes were evaluated in the 50 kph ramp steer manoeuvre. Figure 4.72 shows the understeer characteristic of the baseline vehicle, the comfort and sport mode of the TVC. The linear understeer gradient was decreased, as expected, with comfort and sport modes. The linear understeer gradient was reduced to 16.09 degs²/m with TVC comfort mode and to 12.06 degs²/m with TVC sport mode. The TVC sport mode can meet the driver's expectations and give the driver a neutral feeling. In terms of cornering capacity, the vehicle reached a maximum lateral acceleration of 7.71 m/s² during the cornering with the TVC comfort mode. As a better result, the vehicle reached 7.80 m/s² maximum lateral acceleration with the TVC sport mode. All results satisfied expectations.

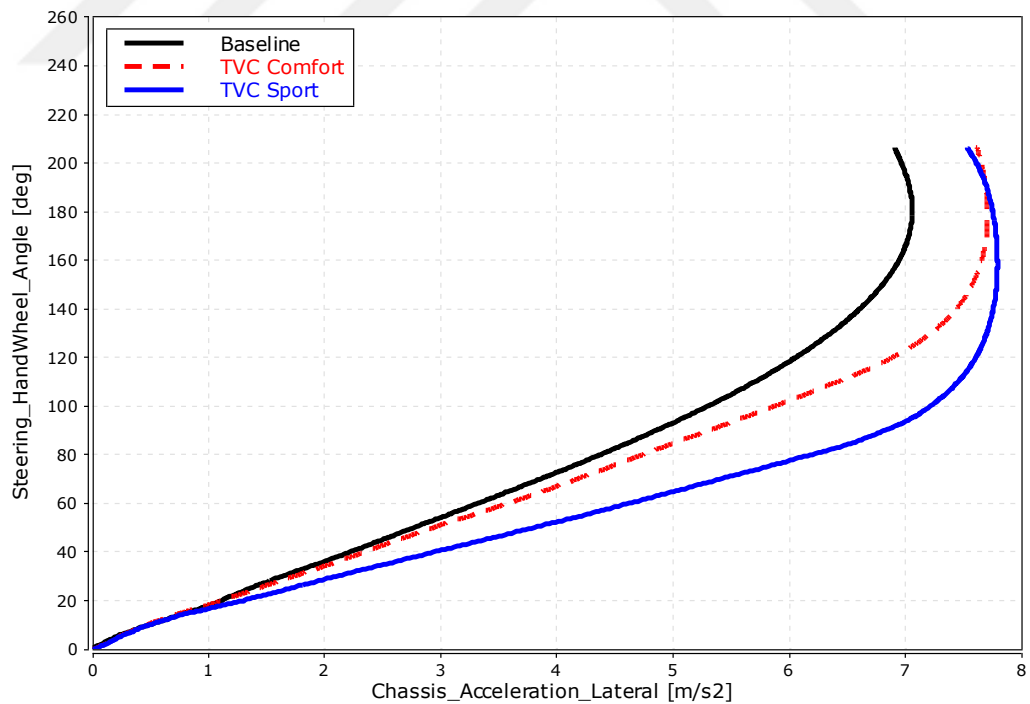


Figure 4.72: TVC results - ramp steer 50 kph - steering angle vs lateral acc.

Other details such as side slip angle, electric torques, and tyre potential utilization were also evaluated. Figure 4.73 shows the primary vehicle dynamics states. As can be seen

from the figure, generated reference yaw rates were different in the different TVC modes. The yaw rate tracking performances of each drive mode are shown in Figure 4.74. Yaw rate tracking performances satisfied the expectations as there were no oscillations and extreme overshoots.

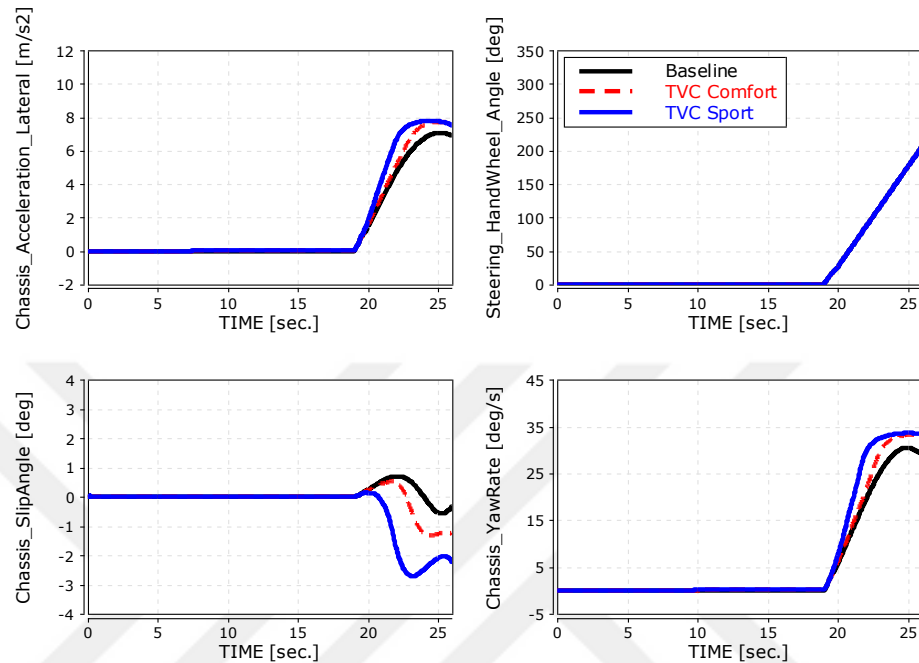


Figure 4.73: TVC results - ramp steer 50 kph - vehicle dynamics states.

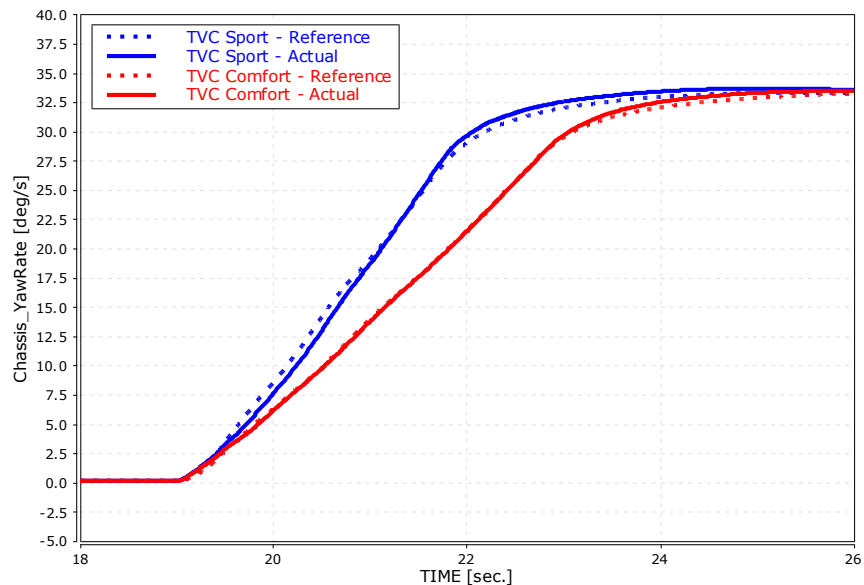


Figure 4.74: TVC results - ramp steer 50 kph - reference tracking.

In Figure 4.75, the front, rear left, and rear right electric motor torques were shown as EM1, EM3 and EM4, respectively. As can be seen from the Figure 4.75, the TVC sport mode worked more aggressively and generated more additional yaw moment during

the manoeuvre. Figure 4.76 shows the feedforward and feedback contributions of each drive mode. TVC sport mode added more additional yaw moment as the feedback part works harder.

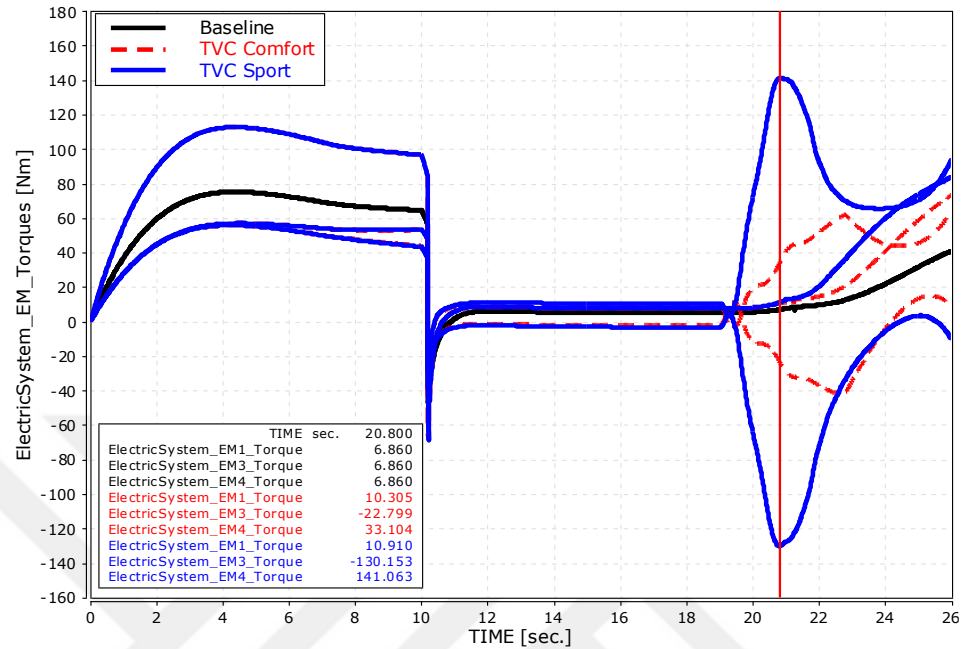


Figure 4.75: TVC results - ramp steer 50 kph - electric motor torques.

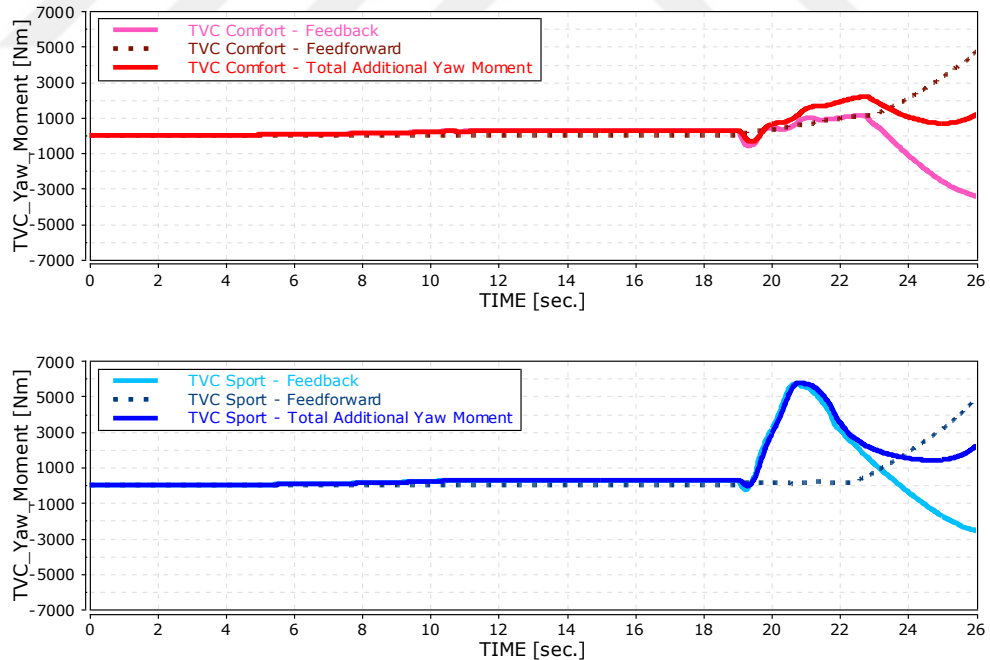


Figure 4.76: TVC results - ramp steer 50 kph - yaw moment contribution.

Also, in Figure 4.75, the longitudinal split function worked well. As mentioned earlier, the longitudinal split factor was chosen to be 0.5, which means that total front and rear torques must be evenly distributed. In the first 10 seconds, the EM1 torque and the

sum of EM3 and EM4 torques were equal. The final evaluation was made on the use of tyre saturation, which is a crucial point of the TVC controller. From Figure 4.77 it is seen that the sport mode uses the potential of the tyres more than the comfort mode. The aim was to use 100% tyre potential as much as the vehicle/tyres could. However, from the bottom graph, at some moment rear left tyre exceeds the tyre saturation limit for a short time. These results showed that the TVC sport mode was performed within safety margins in this manoeuvre. For a better explanation, Figures 4.78 and 4.79 were generated to express the situation.

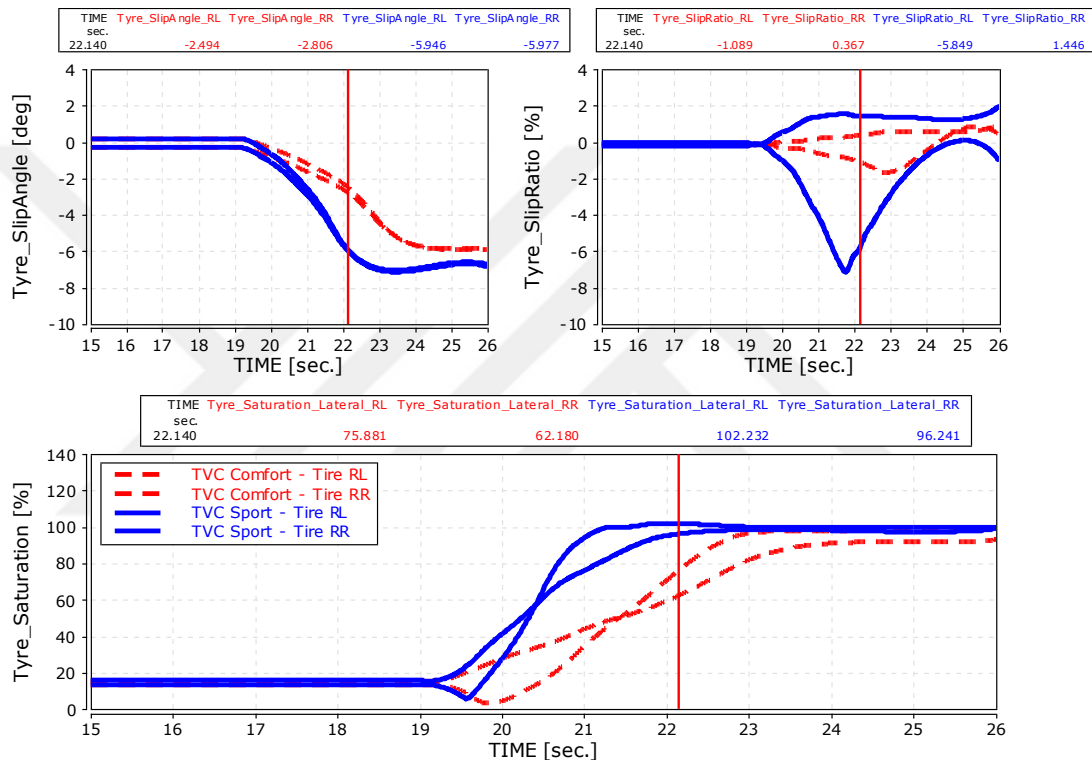


Figure 4.77: TVC results - ramp steer 50 kph - tyre saturations.

During the left-hand cornering, the left wheels' tyre potentials are decreased since the vertical force on those tyres is decreased. Torque vectoring system increases right wheel torque, decreases left wheel torque vice versa to add yaw moment to the system during left-hand turn. As can be seen from Figure 4.79, since the longitudinal force of the rear-left wheel decreases with TVC, staying at the same saturation level creates more lateral force. Figure 4.77 investigated the left-rear tyre saturation in detail during the ramp steer 50 kph manoeuvre. The middle graph of Figure 4.78 shows the tyre saturation usage history and limits of the rear-left tyre during the manoeuvre. The combined slip of the rear-left tyre can also be evaluated from Figure 4.77. Tyre saturation level can also indicate the usage of potential or safety margin overruns. The

saturation usage of the rear-left tyre can be seen from the bottom graph in Figure 4.77. Four points were indicated in this graph and the middle fiction circle graph for better understanding. Although the rear-left tyre exceeded the tyre potential limit at 22.14 seconds (red circle), it returned to potential limits (yellow circle) when it reached the maximum lateral acceleration level. This behaviour can create a safety problem. Thus, hands-on vehicle testing should have been performed to check for potential safety issues.

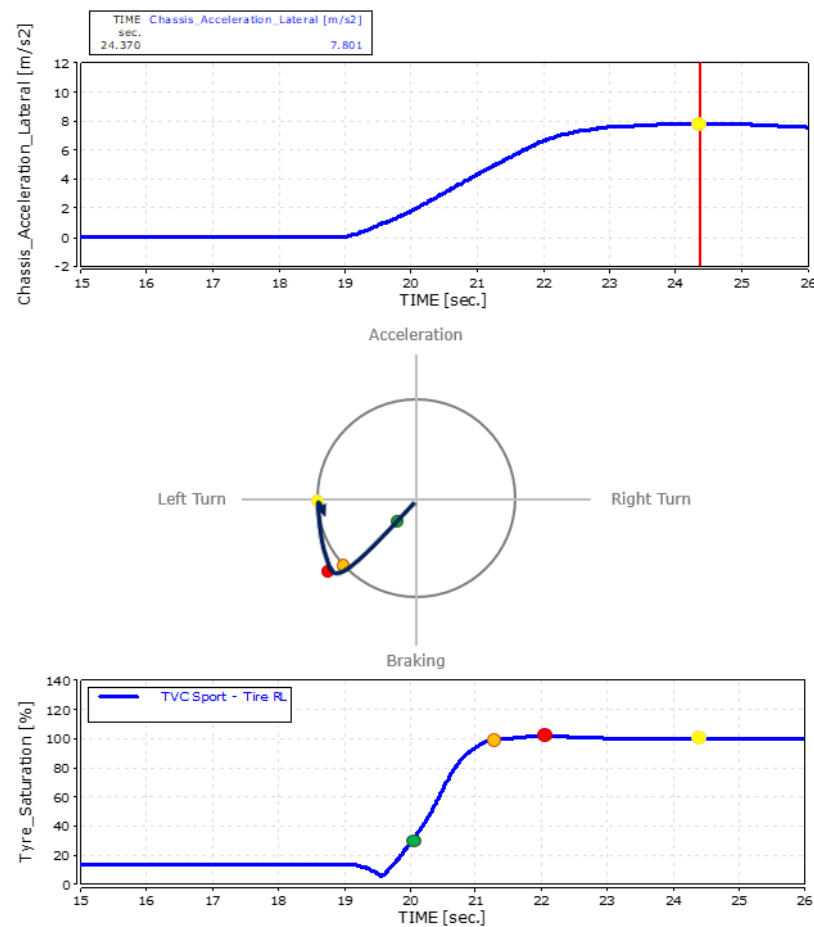


Figure 4.78: TVC results - ramp steer 50 kph - tyre saturation usage map.

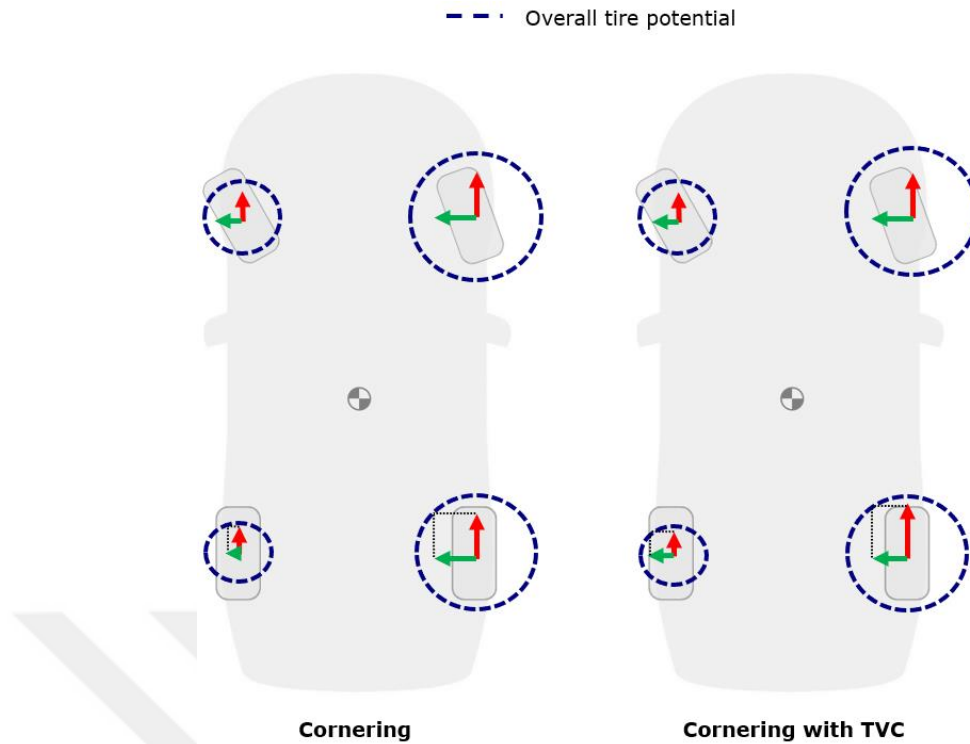


Figure 4.79: Cornering with TVC - tyre saturations.

4.4.3.2 TVC results - ramp steer - 100 kph

The second evaluation was made with the ramp steer 100 kph test scenario. Figure 4.80 shows the understeer characteristic of the baseline vehicle, comfort, and sport mode of the TVC with ramp steer input at 100 kph. The linear understeer gradient was reduced with comfort and sport modes as expected again. The linear understeer gradient was decreased to $6.76 \text{ degs}^2/\text{m}$ with TVC comfort mode and to $4.48 \text{ degs}^2/\text{m}$ with TVC sport mode. TVC sport mode provided a more neutral behaviour. In terms of cornering capacity, the vehicle reached 7.6 m/s^2 maximum lateral acceleration during cornering with the TVC comfort mode. As a better result, the vehicle achieved 7.76 m/s^2 maximum lateral acceleration with TVC sport mode. All results satisfied the expectations again, even at 100 kph. Figures 4.81 and 4.82 show basic vehicle dynamics states and electric motor torques, respectively. From Figure 4.83, it is seen that the sport mode uses the potential of the tyres more than the comfort mode, does not exceed the saturation limits and is kept below %100 level during the manoeuvre.

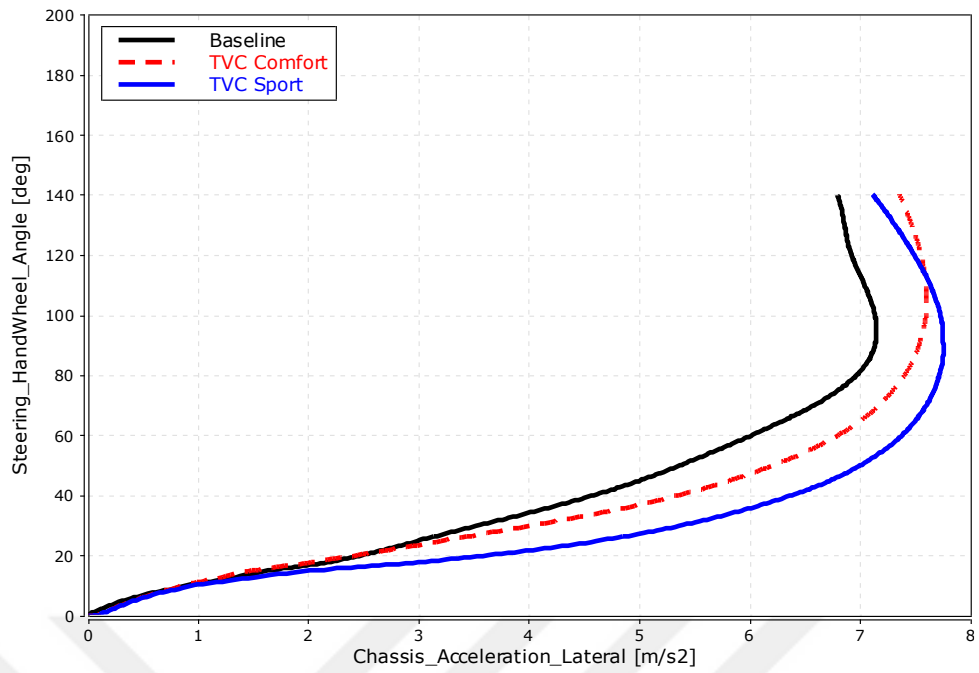


Figure 4.80: TVC results - ramp steer 100 kph - steering angle vs lateral acc.

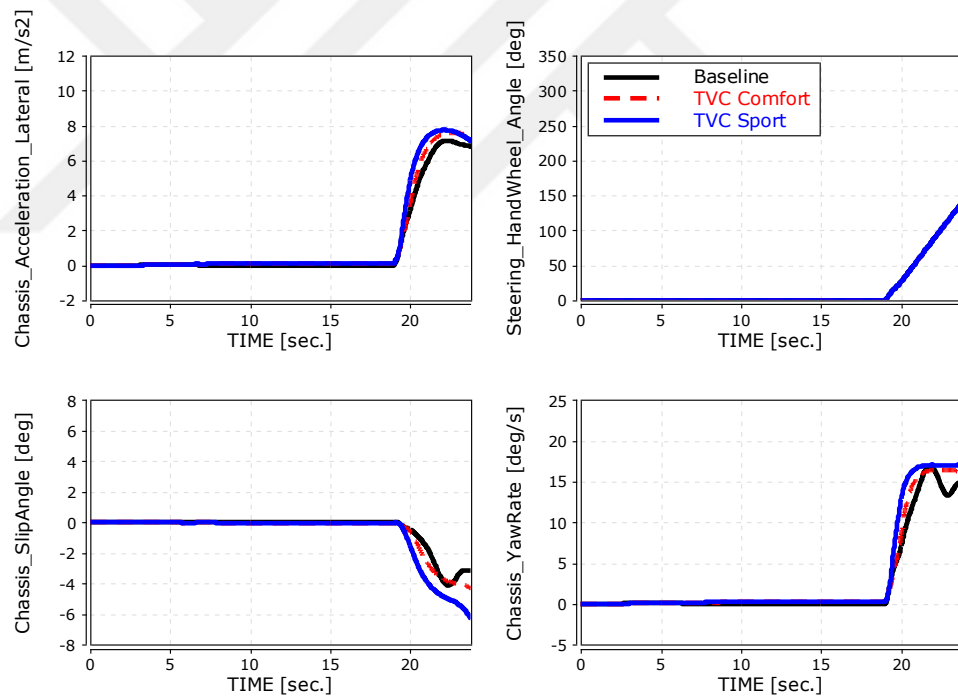


Figure 4.81: TVC results - ramp steer 100 kph - vehicle dynamics states.

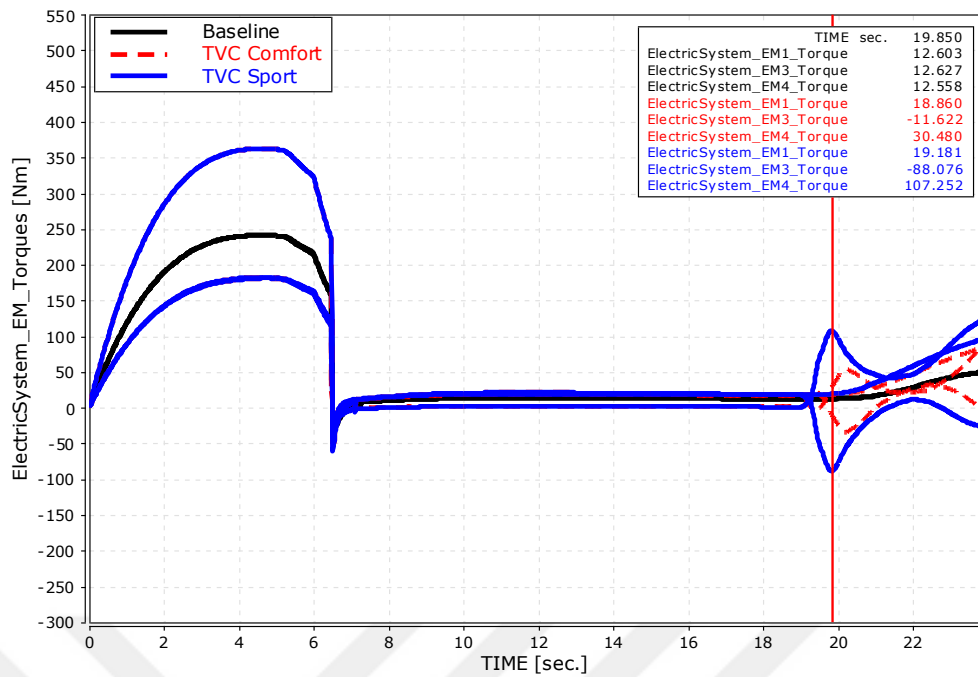


Figure 4.82: TVC results - ramp steer 100 kph - electric motor torques.

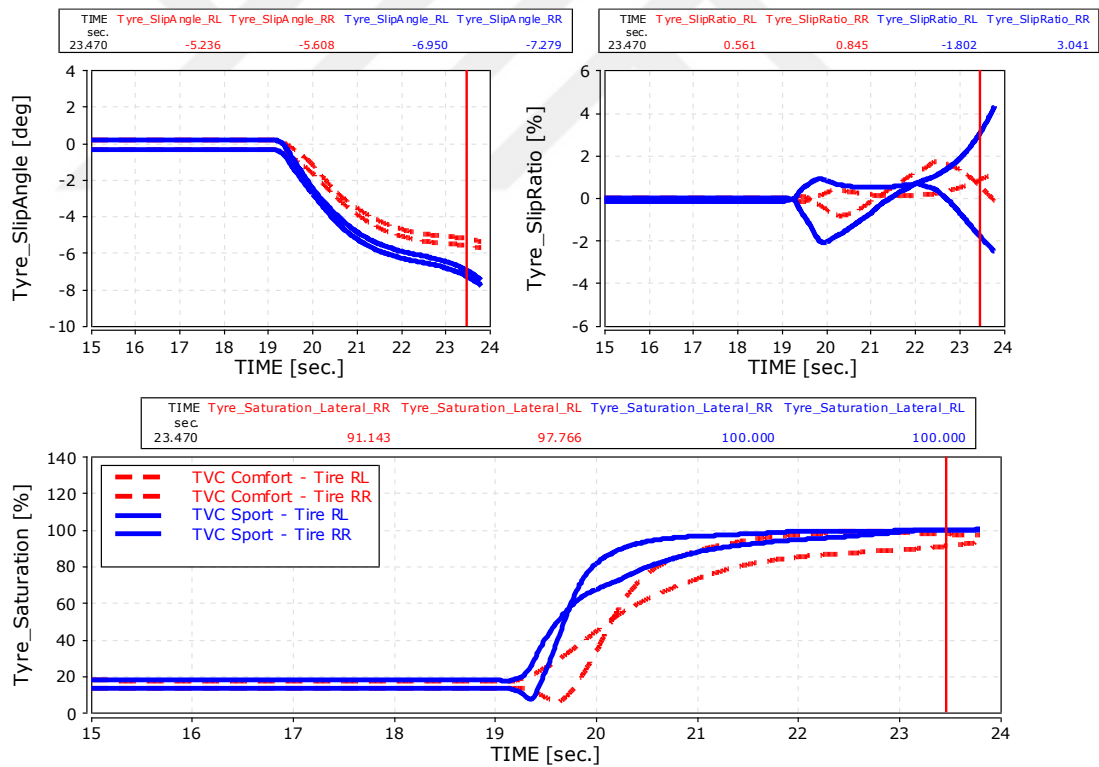


Figure 4.83: TVC results - ramp steer 100 kph - tyre saturations.

4.4.3.3 TVC results - ramp steer - 150 kph

Torque vectoring control system was also tested at 150 kph vehicle speed with ramp steer input. Figure 4.84 shows the understeer characteristic of the baseline vehicle, comfort, and sport mode of the TVC in ramp steer 150 kph case. Linear understeer gradient was decreased to 4.67 degs²/m with TVC comfort mode and to 3.23 degs²/m with TVC sport mode. The vehicle reached 7.31 m/s² and 7.40 m/s² maximum lateral accelerations during the cornering with the TVC comfort and sport modes, respectively. Figures 4.85 and 4.86 showed primary vehicle dynamics states and electric motor torques, respectively. In Figure 4.87, front tyre potentials have been added to the graph this time again since there was a crucial problem noticed. In the TVC optimization procedure, possible oversteer behaviour was already prevented. However, no action was taken for the understeer behaviour. The vehicle was driven in TVC sport mode and experienced understeer due to the increased front slip values in the last moments of the manoeuvre. Although potential of the rear tyres was below 100% limits, the front tyres exceeded the limits and slipped. Thus, this understeer behaviour was noticed, and it was planned to correct it with the contribution of RWS by vehicle motion control.

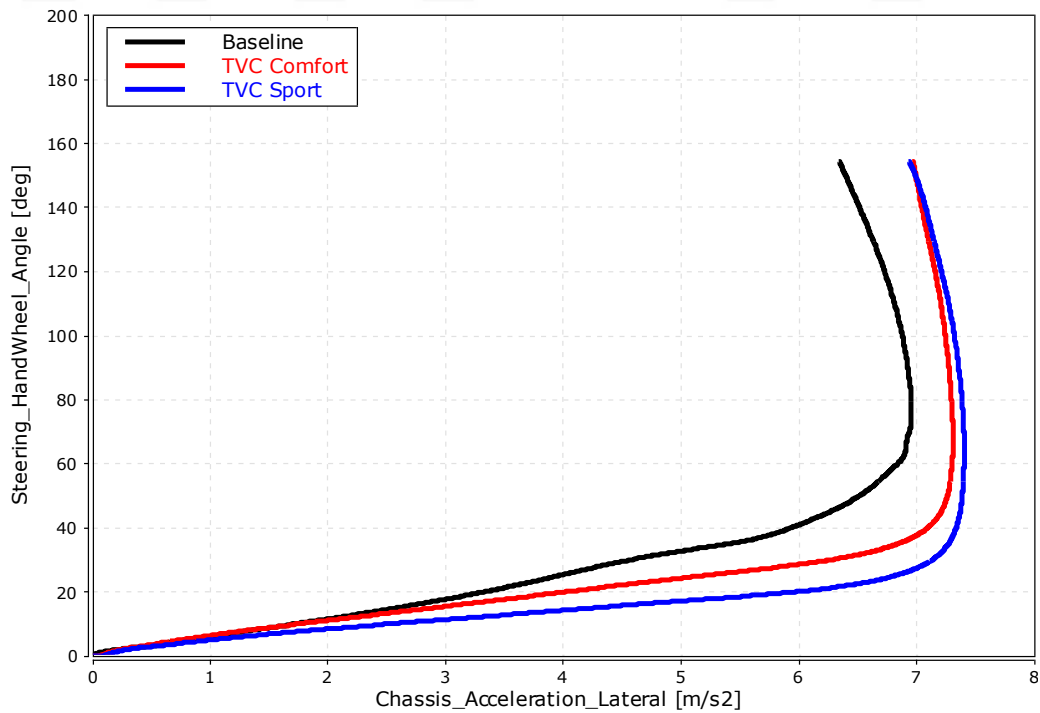


Figure 4.84: TVC results - ramp steer 150 kph - steering angle vs lateral acc.

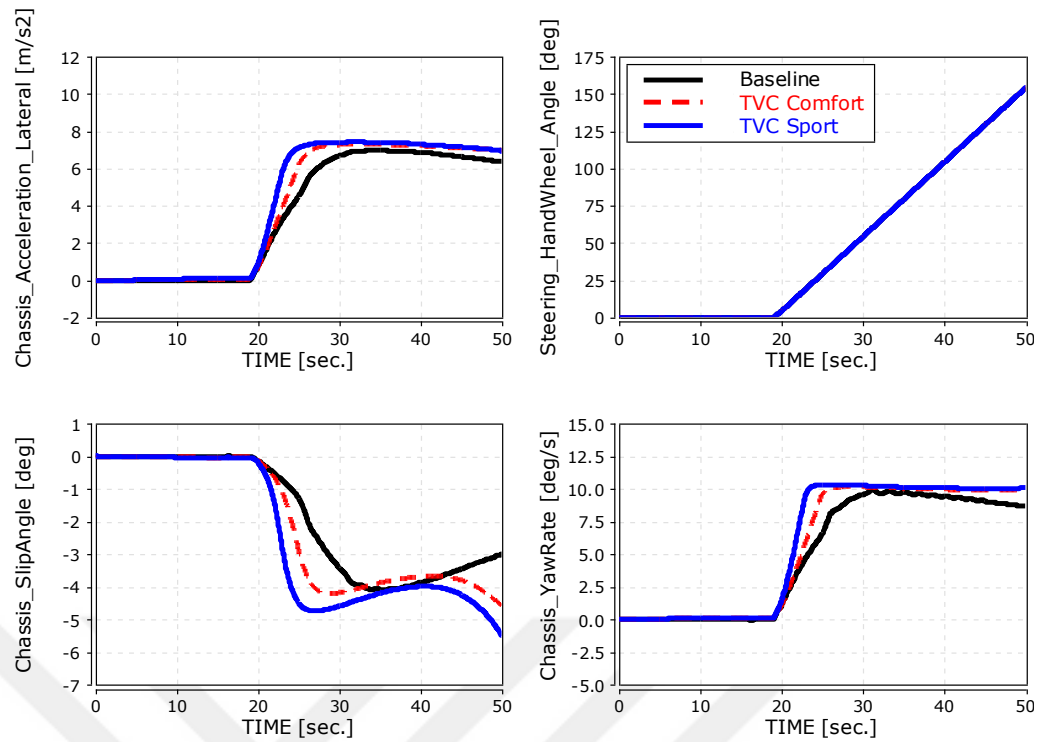


Figure 4.85: TVC results - ramp steer 150 kph - vehicle dynamics states.

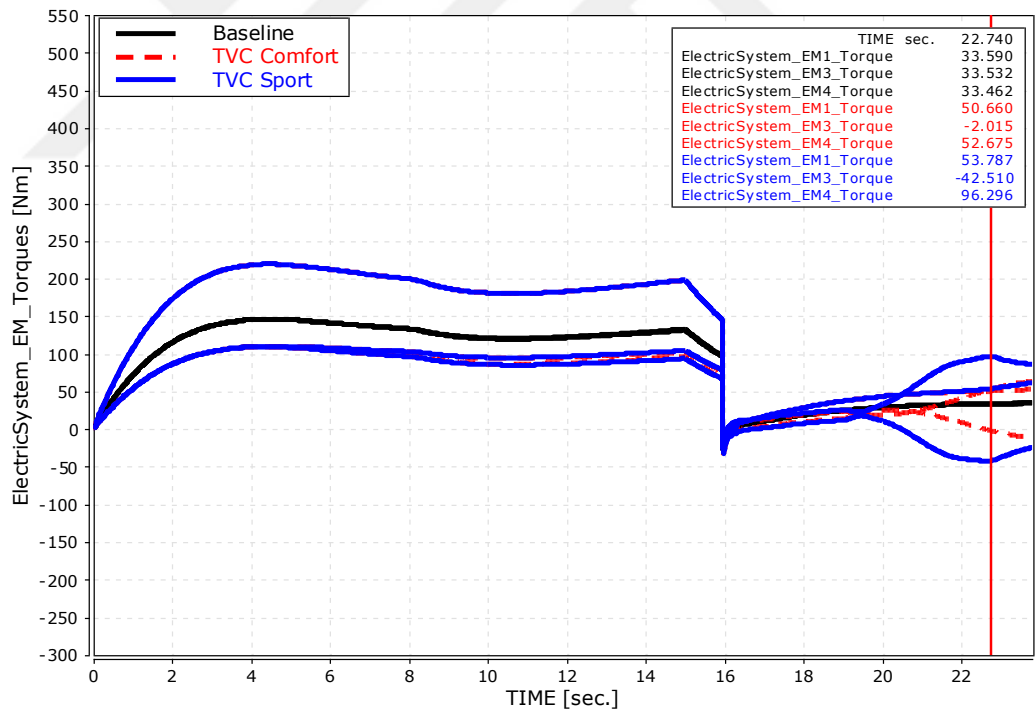


Figure 4.86: TVC results - ramp steer 150 kph - electric motor torques.

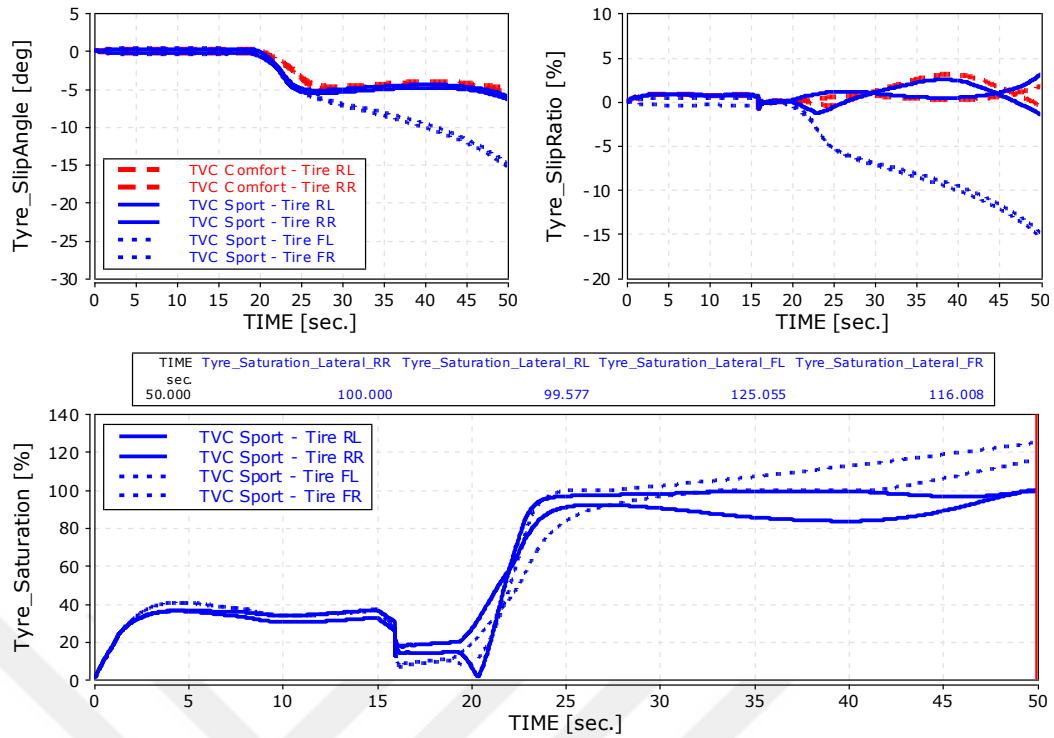


Figure 4.87: TVC results - ramp steer 150 kph - tyre saturations.

4.4.3.4 TVC results - step steer

A step steering input was applied to the vehicle at a vehicle speed of 100 kph to achieve a lateral acceleration of 4 m/s^2 at steady-state. The objective was to minimize the difference between peak and steady-state lateral acceleration value and to keep a steady-state lateral acceleration of 4 m/s^2 . From Figure 4.88, it is clearly seen that the overshoots of lateral response are improved with both TVC comfort and TVC sport modes. TVC comfort mode decreased the steering input to 14.3° to maintain lateral acceleration at 4 m/s^2 steady-state. This can create a more neutral driving feel, which can be preferable for the driver. The TVC sport mode steering input was 2.7° to achieve lateral acceleration at 4 m/s^2 steady-state. This can create an undesirable neutral driving feeling for the driver. This point was noticed during the evaluation. With the TVC comfort and sport mode, the lateral acceleration did not exceed the steady-state value. Figure 4.89 shows the yaw rate reference and the actual yaw rate data. Both modes successfully followed their yaw rate targets even in transient manoeuvre. Figure 4.90 shows the operating levels of the feedforward and feedback parts. As can be seen from the Figure 4.90, the feedback part in this manoeuvre worked very actively to meet the additional yaw moment demand.

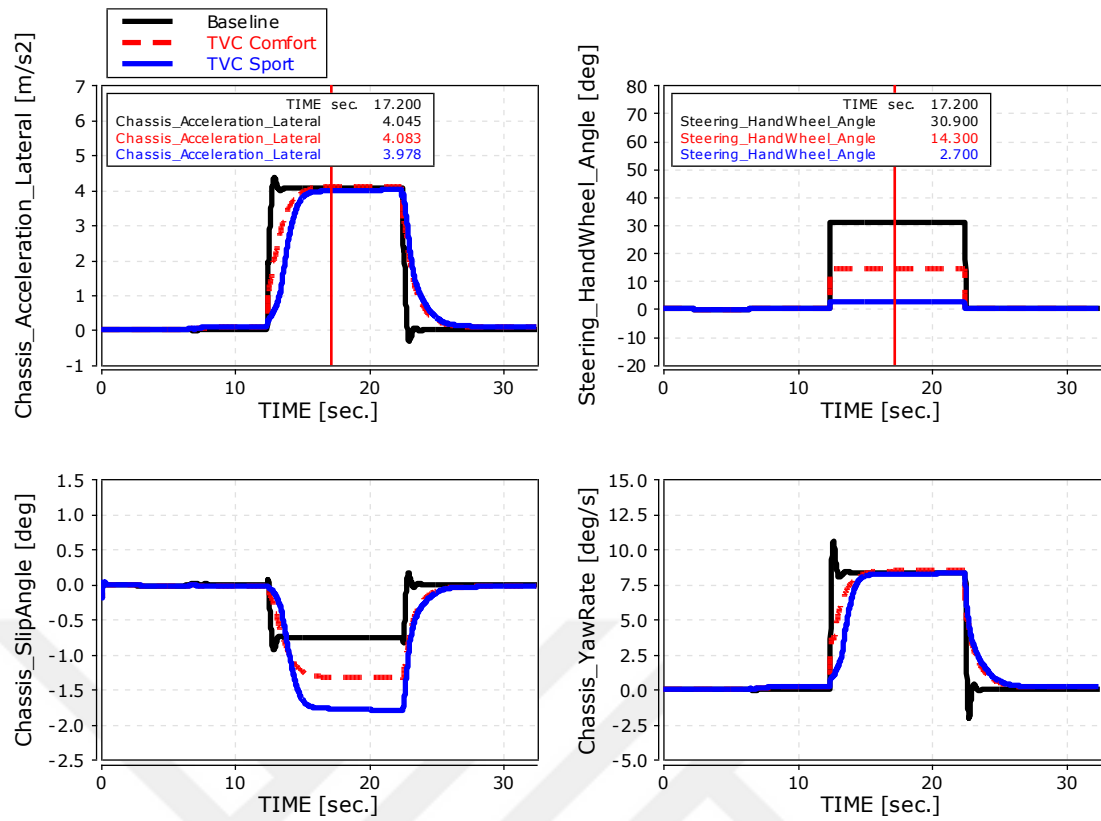


Figure 4.88: TVC results - step steer 100 kph - vehicle dynamics states.

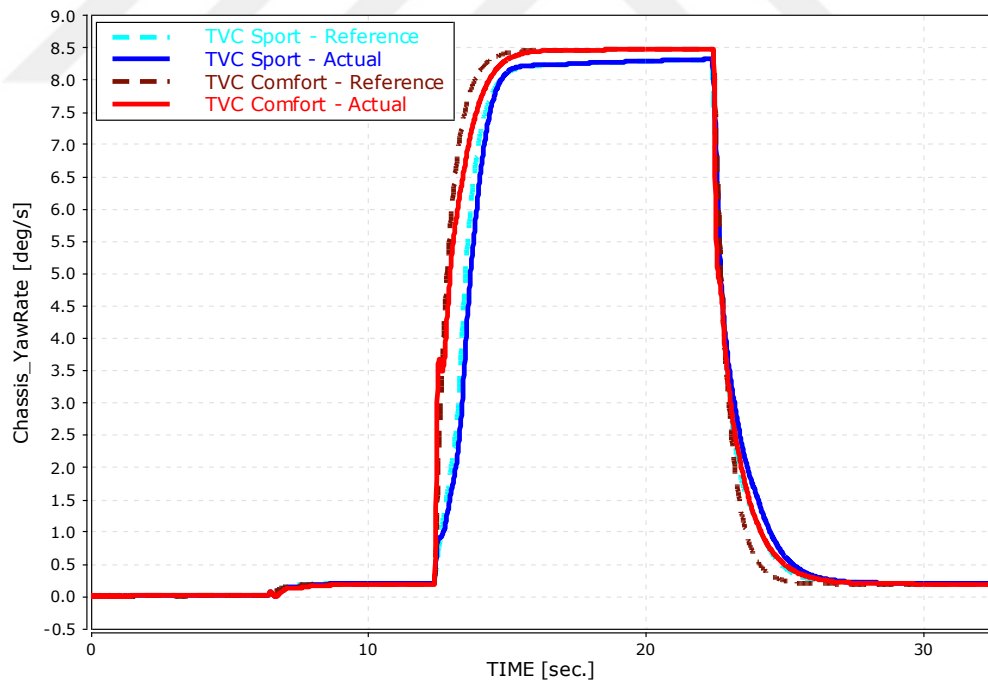


Figure 4.89: TVC results - step steer 100 kph - yaw rates.

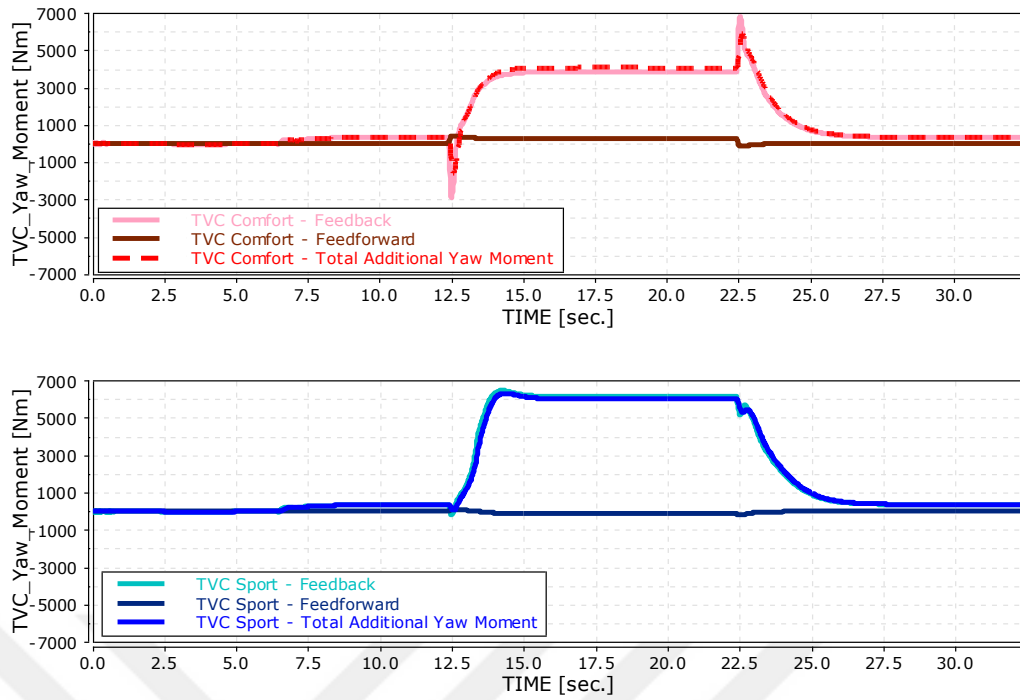


Figure 4.90: TVC results - step steer 100 kph - ff and fb parts.

4.4.3.5 TVC results - constant radius cornering

In constant radius cornering manoeuvre, both TVC driving modes optimally distributed the torques to the rear wheels to achieve more neutral steering behaviour and increase maximum lateral acceleration capacity. From Figure 4.91, the linear understeer gradient was decreased to 2.7 degs²/m with TVC comfort mode and to 1.23 degs²/m with TVC sport mode. The vehicle reached 7.49 m/s² and 7.6 m/s² maximum lateral accelerations during the cornering with the TVC comfort and sport modes, respectively. Figures 4.92 and 4.93 show the basic vehicle dynamics state and vehicle trajectory, respectively. From the trajectory graph, the vehicle lost the constant path-following ability and showed some understeer behaviour at the end of the manoeuvre. This understeer behaviour is also shown in Figure 4.94. Although the potentials of the rear tyres were below 100% limits, the front tyres slipped at the end of the manoeuvre. This result showed that an additional longitudinal torque split function should be considered instead of taking the longitudinal split factor as 0.5 directly during cornering. However, both TVC comfort and TVC sport mode understeer characteristics and yaw rate tracking performances satisfied expectations.

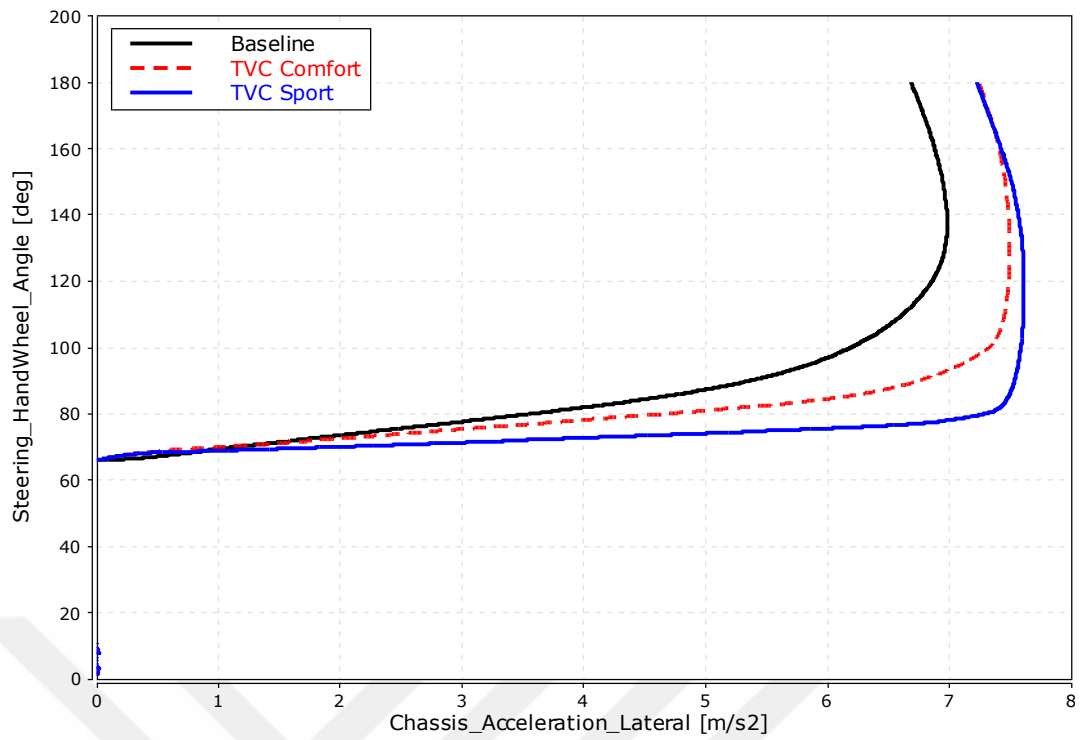


Figure 4.91: TVC results - CRC - steering angle vs lateral acc.

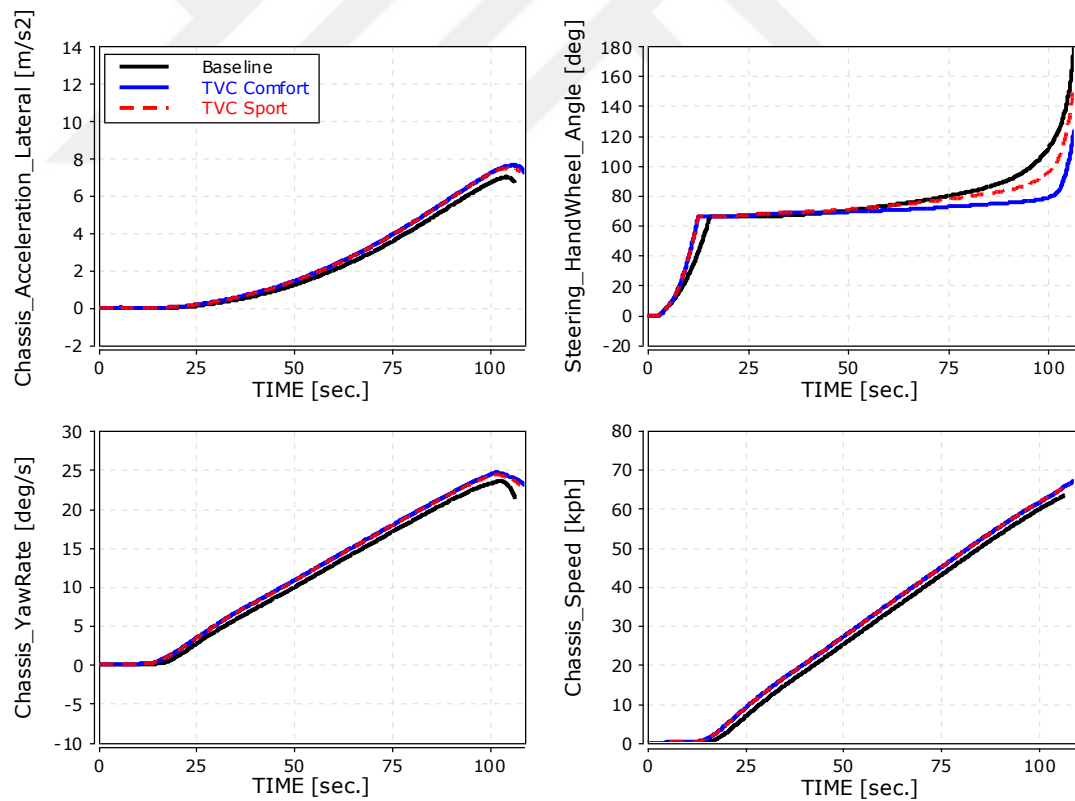


Figure 4.92: TVC results - CRC - vehicle dynamics states.

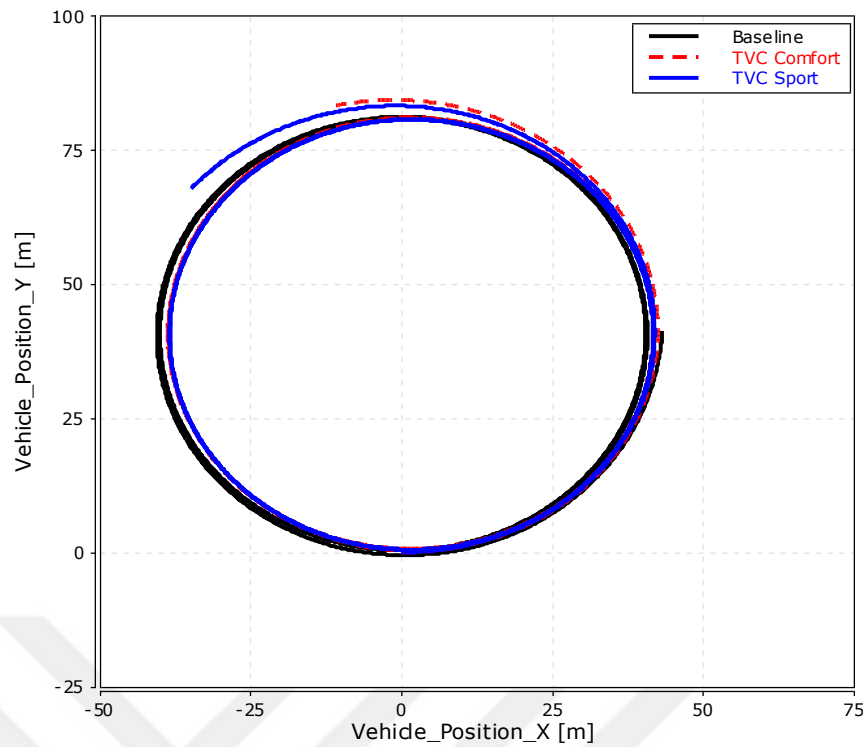


Figure 4.93: TVC results - CRC – trajectory.

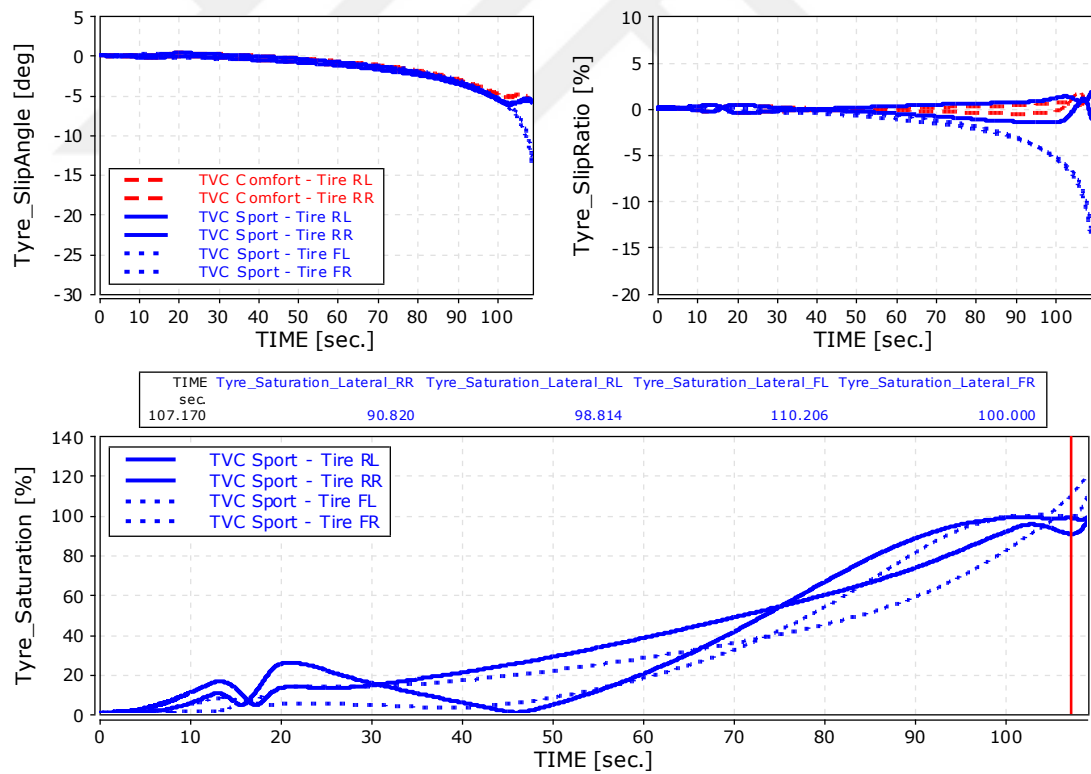


Figure 4.94: TVC results - CRC - tyre saturations.

4.4.3.6 TVC results - double lane change

Both TVC driving modes successfully completed the DLC manoeuvre without exceeding the path limits with an entry speed of 55 kph, as seen in Figure 4.95. Likewise, the TVC comfort mode almost came close to the limit of the track. TVC modes performed well in the DLC scenario, although the regression estimation analysis of the DLC results was of poor quality as R^2 was less than 0.8. Vehicle dynamics results of TVC driving modes are given in Figure 4.96. The best exit speed was belonged to the TVC sport mode. Maximum side slip angles during manoeuvres for TVC comfort and TVC sport mode were 3.5 and 2.0 degrees, respectively. Thus, TVC sport mode gave the best results in terms of path tracking and vehicle stability in the DLC case. The chassis sideslip phase plane is given in Figure 4.97. As can be seen from this graph, TVC sport mode gave the best results in terms of vehicle stability since its area was the lowest. And lastly, Figure 4.98 shows that the feedback part of the TVC sport mode worked more than the feedback part of the TVC comfort mode. Both TVC comfort and TVC sport mode path tracking performances satisfied expectations.

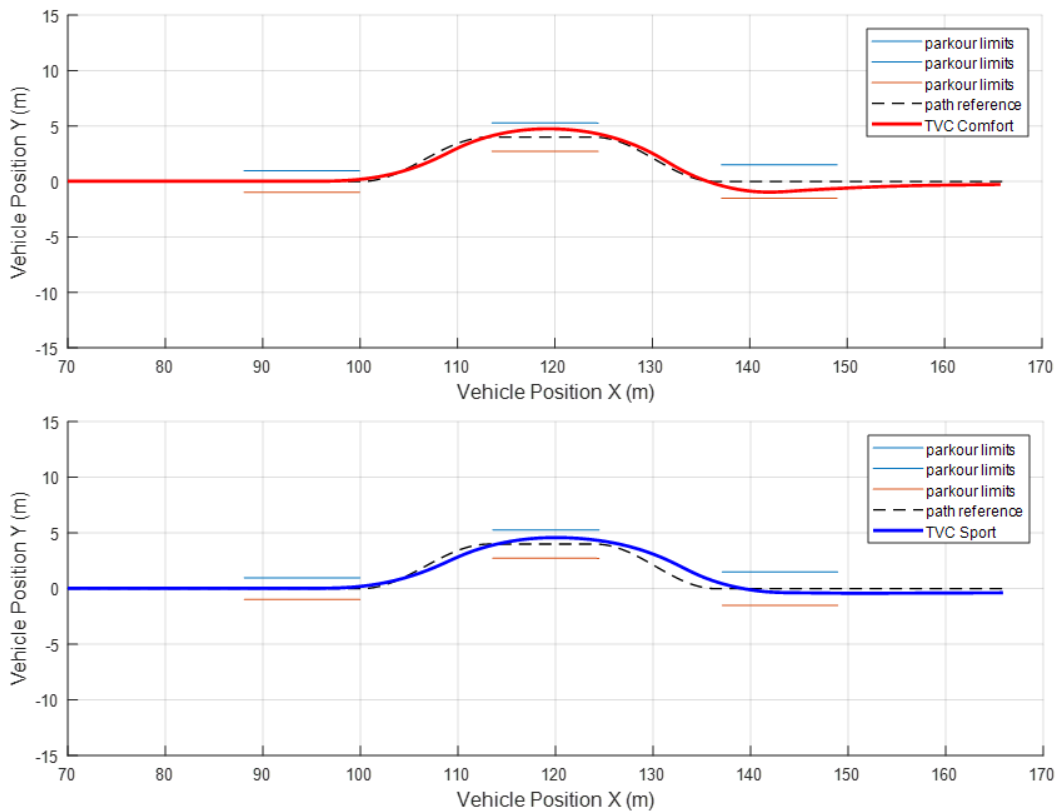


Figure 4.95: TVC results - DLC - chassis trajectory.

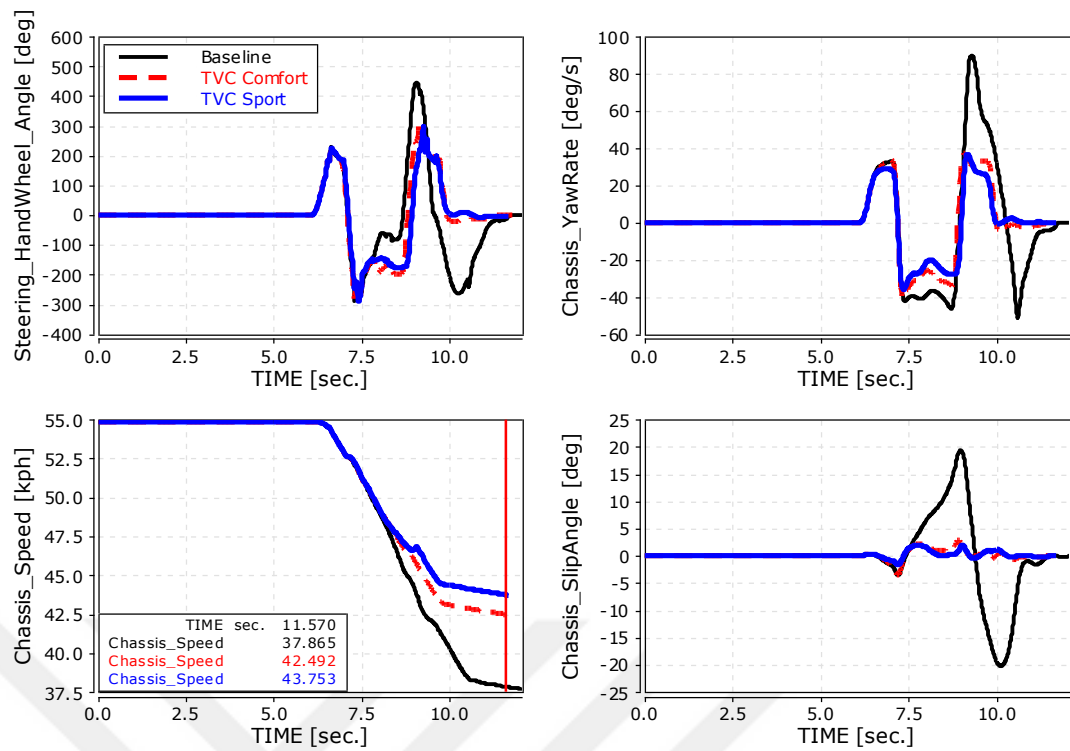


Figure 4.96: TVC results - DLC - vehicle dynamics states.

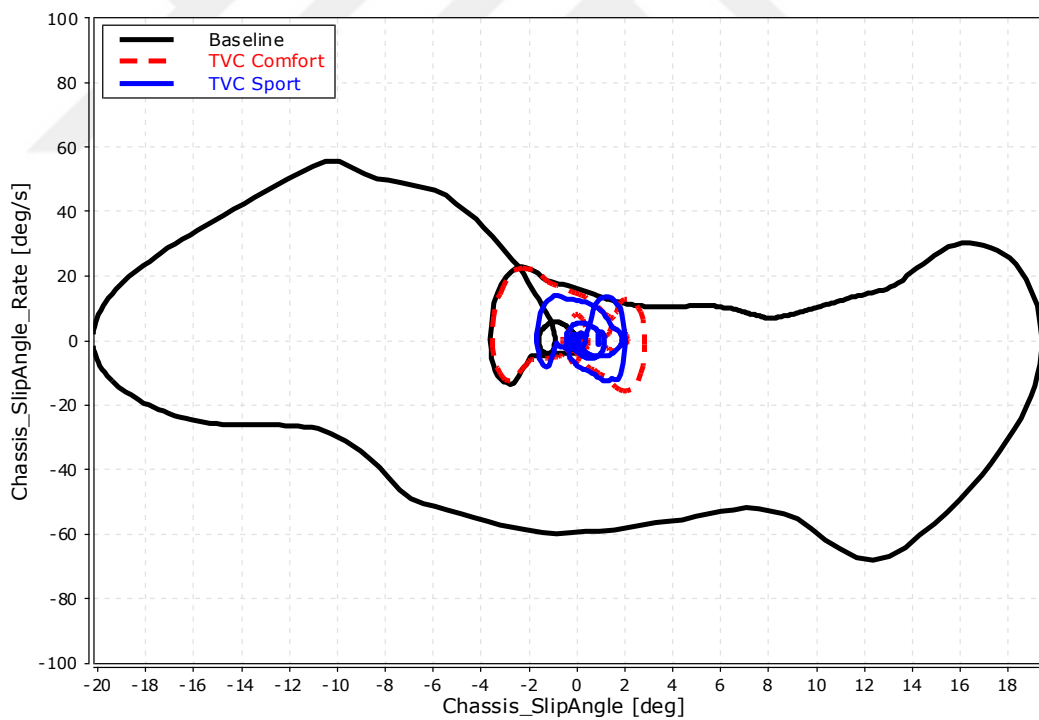


Figure 4.97: TVC results - DLC - sideslip phase plane.

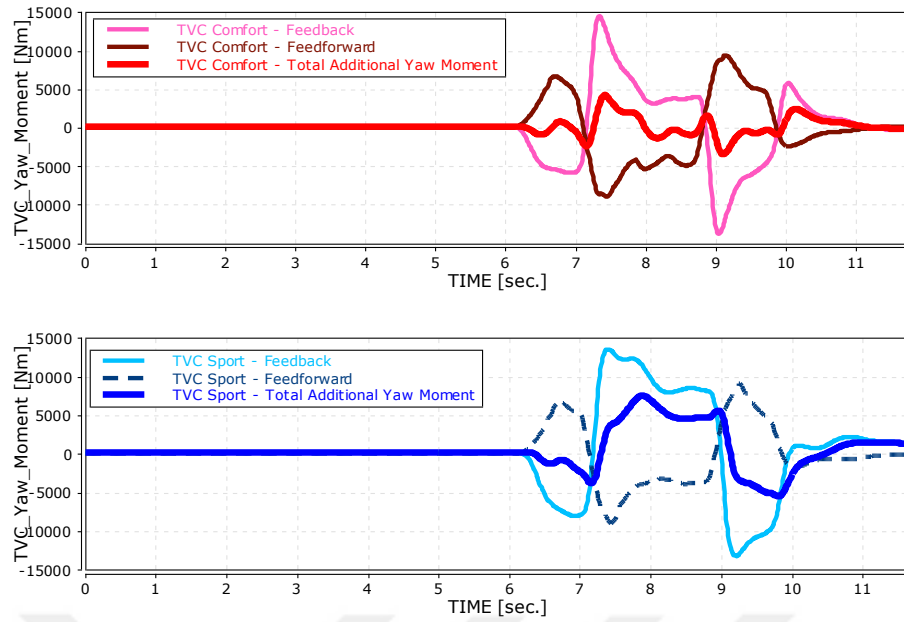


Figure 4.98: TVC results - DLC - ff and fb parts.

4.4.3.7 TVC evaluation

Consequently, all TVC drive modes and results were automatically evaluated and optimized with the optimization system previously used in the RWS. According to TVC driving mode results, TVC sport mode successfully met the targets, decreased the linear understeer gradient in all ramp steer with different speed conditions, and enhanced vehicle handling capability. TVC comfort mode reduced linear understeer gradient with less aggression. Sport mode provided a more neutral driving experience in all test conditions. As the best result, the sport mode was able to noticeably reduce the maximum side slip angle in the DLC manoeuvre noticeably. Both driving modes reduced overshoots in the step steer test case. All objective results are summarized in Table 4.10.

Table 4.10: TVC optimization and results of objective measurements.

Test Case	Speed [kph]	Objective Measurement	Base	TVC Comfort Mode	TVC Comfort Benefit	TVC Sport Mode	TVC Sport Benefit
Ramp Steer	50	K_{us} [deg s ² /m]	17.87	16.09	-1.78	12.06	-5.81
Ramp Steer	50	$a_{y,max}$ [m/s ²]	7.05	7.71	0.66	7.80	0.75
Ramp Steer	100	K_{us} [deg s ² /m]	7.67	6.76	-0.91	4.48	-3.19
Ramp Steer	100	$a_{y,max}$ [m/s ²]	7.15	7.60	0.45	7.76	0.61
Ramp Steer	150	K_{us} [deg s ² /m]	6.35	4.67	-1.68	3.23	-3.12
Ramp Steer	150	$a_{y,max}$ [m/s ²]	6.96	7.31	0.35	7.40	0.44
CRC	100	K_{us} [deg s ² /m]	4.23	2.70	-1.53	1.23	-3.00
CRC	100	a_y [m/s ²]	7.00	7.49	0.49	7.60	0.60
Step Steer	100	a_y overshoot difference [m/s ²]	0.35	0.00	-0.35	0.00	-0.35
DLC	55	β [deg]	20.0	3.5	-16.57	2.0	-18.07

5. VEHICLE MOTION CONTROL

Vehicle motion control in chassis control systems refers to the vehicle's ability to manage and manipulate the motion and behaviour of vehicles, actuators, and algorithms that are part of the vehicle's chassis control system. In general, the goal is to improve vehicle stability, handling, performance, and safety. In this study, vehicle motion control, also called integrated chassis "lateral" controller (ICLC), was designed to control, and manage two "lateral" dynamics controller actuators, RWS and TVC, to improve handling, stability, and performance. Until this part of the study, both chassis control systems RWS and TVC were thoroughly examined, designed, and optimized. RWS was designed to increase low-speed manoeuvrability and increase high-speed stability. TVC was designed to enhance vehicle responsiveness and cornering ability. Figure 5.1 shows the understeer gradient characteristics of comfort and sport modes of the RWS and TVC.

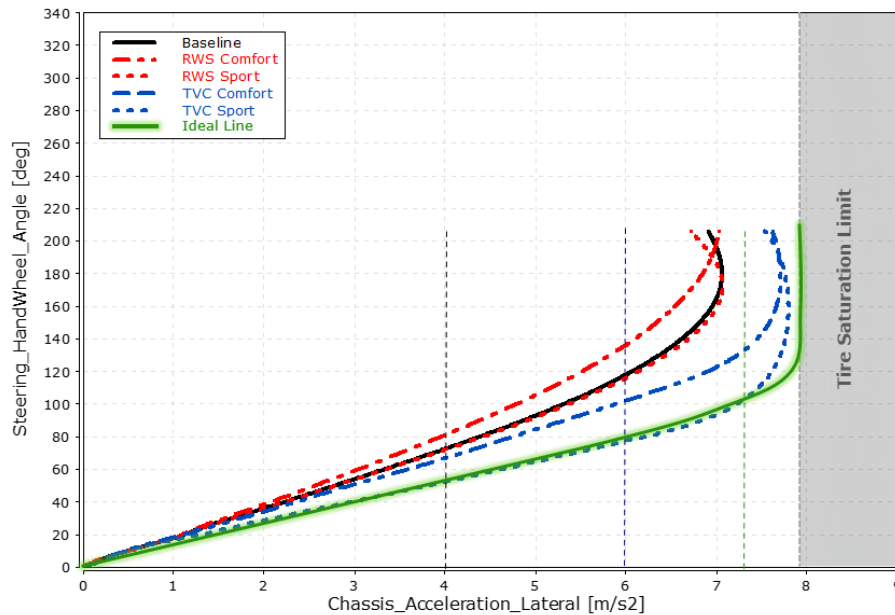


Figure 5.1: RWS and TVC driving mode results in ramp steer 50 kph.

Both chassis controller “comfort” mode provides less sensitivity than the “sport” mode while providing more stability. The same strategy was used when combining the two chassis systems in vehicle motion control. There are two different driving modes

defined again for the combined mode ICLC: comfort and sport. ICLC drive modes basically adjust the weighting factors of RWS and TVC according to the drive mode already selected by the driver. For better understanding, when a comfort mode is selected from the driver, ICLC uses RWS comfort and TVC comfort modes in an optimal manner. In the same way, when a driver selects sport mode, ICLC provides optimum weighting factors of RWS sport and TVC sport modes for the performance. In this way, when both chassis control systems are used, the decentralized vehicle motion control decides how they are used. All chassis control systems and modes are represented in Figure 5.2.

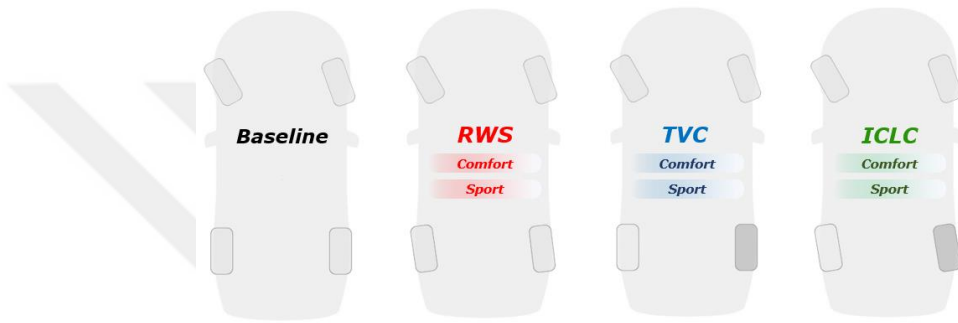


Figure 5.2: All chassis controllers and their drive modes.

As mentioned before, the vehicle motion control system coordinated all lateral dynamics actuators with a downstream architecture approach. The general scheme of this study and integrated chassis lateral controller is shown in Figure 5.1. In this section, firstly, the weighting factors of RWS and TVC actuators are parameterized. Then, the strategy to be applied to use optimum weighting factors for the performance is proposed. The weighting factors of RWS η_{RWS} and TVC η_{TVC} were parametrized as velocity and steering angle dependent. Thus, corrected additional yaw moment $M_{z,c}$ (see Equation 90) and corrected rear wheel steering mean angle $\delta_{r,c}$ (see Equation 49) were controlled according to the vehicle speed and steering wheel angle. As represented in Figure 5.3, RWS and TVC systems have no common high-level controller. Each system has its own controller inputs and outputs. Thus, each system's actuator can influence vehicle dynamics differently. Phase portrait analysis was used to understand the effects of each control system's actuator and drive modes. Phase plane analysis was used as a highly valuable method for assessing the influence of control actions on improving the performance and stability of vehicles [39][48][64][65].

figure, it was seen that phase portrait representation also can give information regarding the roots of the characteristic equation.

To generate phase portraits, state variables of the linear bicycle model differential equations were defined as side slip angle β and yaw rate r_z . State equations of linear bicycle model with RWS and TVC can also be written in terms of vehicle side slip angle with the acceptance of $\beta \cong v_y/v_x$ as follows using equations 1,2,15:

$$\dot{\beta} = \frac{C_f + C_r}{mv_x} \beta + \frac{aC_f - bC_r - mv_x^2}{mv_x^2} r_z - \frac{C_f \delta_f}{m} - \frac{C_r \delta_r}{m} \quad (113)$$

$$\dot{r}_z = \frac{aC_f - bC_r}{I_{zz}} \beta + \frac{a^2 C_f + b^2 C_r}{I_{zz} v_x} r_z - \frac{aC_f \delta_f}{I_{zz}} + \frac{bC_r \delta_r}{I_{zz}} + \frac{M_z}{I_{zz}} \quad (114)$$

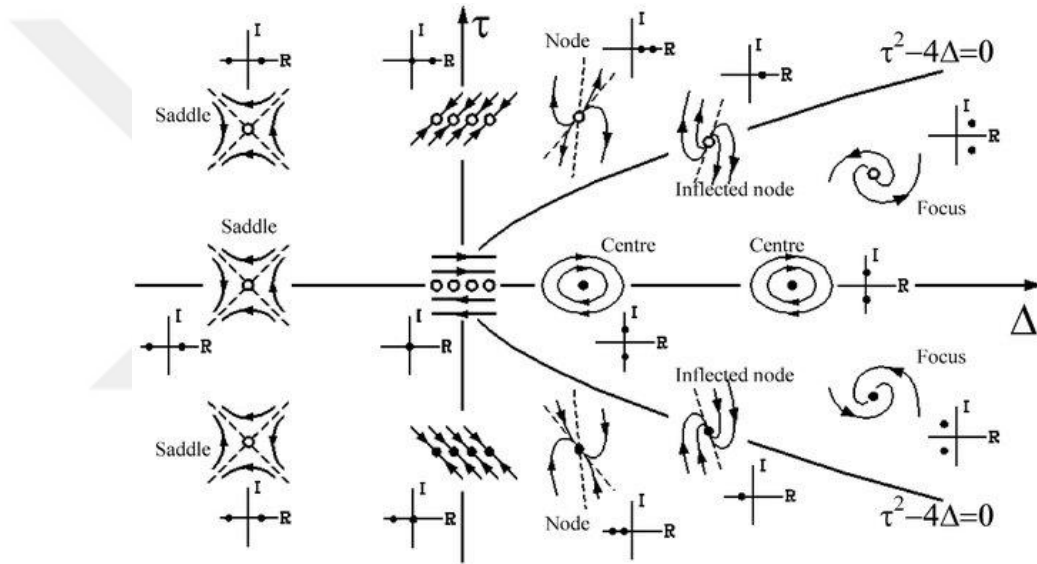


Figure 5.4: Types of phase portrait results [66].

The ordinary differential equation (ODE) solvers and “plotpp” function in MATLAB were used to generate phase portraits from a simulation result [67]. All parameters were taken from the simulation outputs. It means that initial conditions of state variables were taken directly from at a specific time of a simulation. Cornering stiffness values were also taken from the simulation data. In this way, phase plane analysis was adopted a non-linear vehicle and tyre model. Since phase portrait evaluation accounts all non-linearities and tyre combined slip effects, the effects of RWS and TVC actuators were shown more accurately. Vignati and Sabbioni state that “In fact, the phase portrait gives an instantaneous snapshot of the trajectories at each state combination for a pre-determined set of parameters and a fixed control input” and continue with ... “the trajectory of the vehicle states can only be viewed as it

would develop in time on single-phase portrait only if steering and speed remain constant” [39]. Thus, for the phase portrait analysis, an open loop manoeuvre with 50 kph vehicle speed and 2.5° front mean steering angle was defined. RWS and TVC comfort and sport modes were evaluated in phase plane to see their effectiveness on the vehicle dynamics behaviour

Figure 5.5 showed the baseline vehicle characteristic was generated at a constant speed of $v_x = 50$ kph and a constant steering angle of $\delta_f = 2.5$. The black dot in the middle of the graph indicates the equilibrium point. The effectiveness of the chassis control actuators is directly related with the convergence to this equilibrium point.

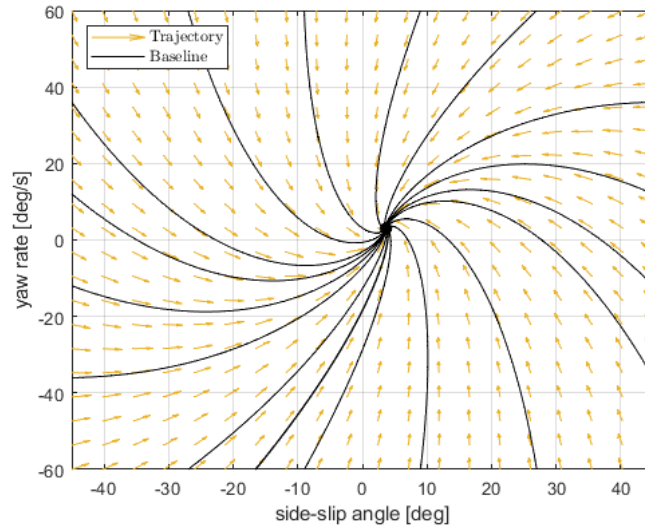


Figure 5.5: The phase portrait of baseline vehicle at $v_x = 50$ kph, $\delta_f = 2.5^\circ$.

The most effective actuator on vehicle system dynamics can be defined as the one that reaches the equilibrium point fastest. Thus, vehicle characteristics of RWS Comfort, RWS Sport, TVC Comfort and TVC Sport were generated at a constant speed of $v_x = 50$ kph and constant steering angle of $\delta_f = 2.5$ in a phase plane. Figures 5.6, 5.7, 5.8 and 5.9 showed the phase portraits of a vehicle with different chassis control driving modes variants. From figures, behaviours other than TVC sport mode can be called “stable spiral”, and they have complex eigenvalues with negative real parts. TVC sport mode can be called a “stable node”, and it has two real negative roots. So, it was obtained that the TVC sport mode had a big influence on vehicle dynamics behaviour and provided better stability than the others. The responsive characteristic of TVC sport mode was proven with that result. RWS sport was the worst in terms of actuator

efficiency on dynamic behaviour. Phase portrait results were also examined for other velocities (100 kph, 150 kph) and steering angles (5 deg, 10 deg).

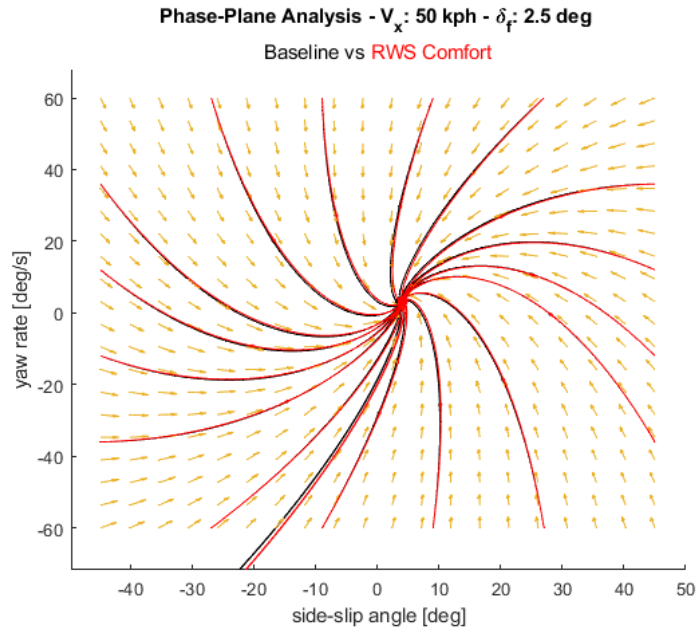


Figure 5.6: Phase portraits of RWS comfort mode at $v_x = 50$ kph, $\delta_f = 2.5^\circ$.

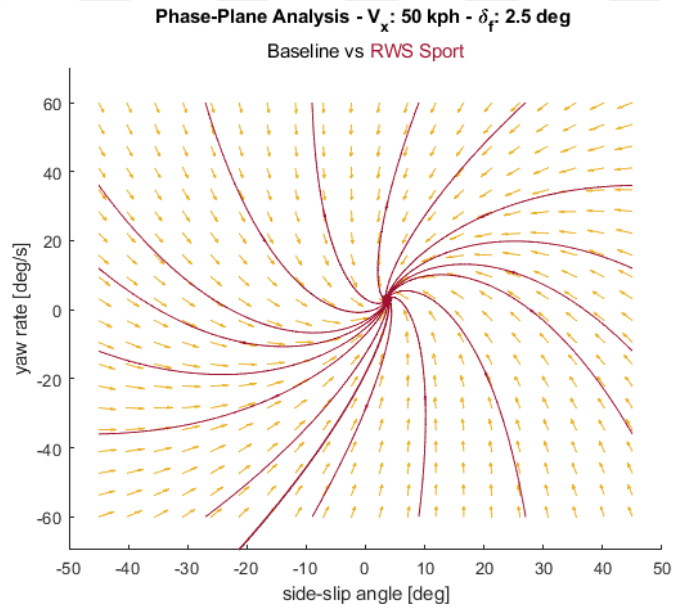


Figure 5.7: Phase portraits of RWS sport mode at $v_x = 50$ kph, $\delta_f = 2.5^\circ$.

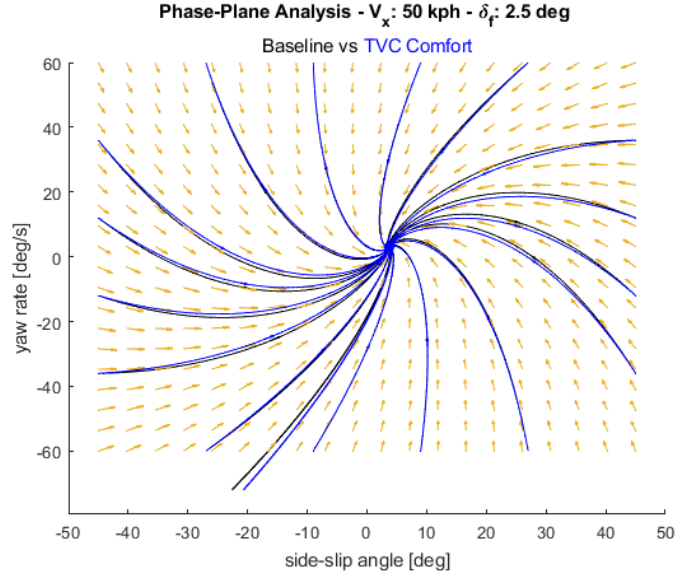


Figure 5.8: Phase portraits of TVC comfort mode at $v_x = 50$ kph, $\delta_f = 2.5^\circ$.

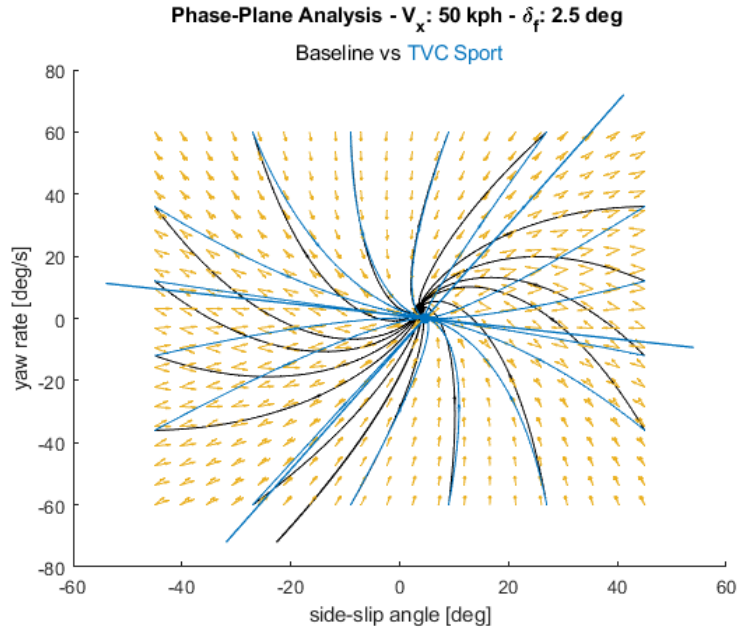


Figure 5.9: Phase portraits of TVC sport mode at $v_x = 50$ kph, $\delta_f = 2.5^\circ$.

After evaluating phase plane behaviours and efficiencies of the RWS and TVC actuators, it was concluded that RWS was not effective as TVC, and it was decided to use RWS as a supportive role in vehicle motion control to obtain ideal performance, as stated in Figure 5.9. The first plan was to optimise weighting factors for both RWS and TVC, but after analysing the effectiveness of each actuator, TVC weighting factors were kept as 1. Thus, only the RWS contribution was optimised. RWS weighting factor

was obtained through offline numerical optimisation via running ramp steer manoeuvre with speeds of 50,100,150,200, and 250 kph. Finalised weighting factors of RWS and TVC actuators according to different driving modes were represented in Figure 5.10.

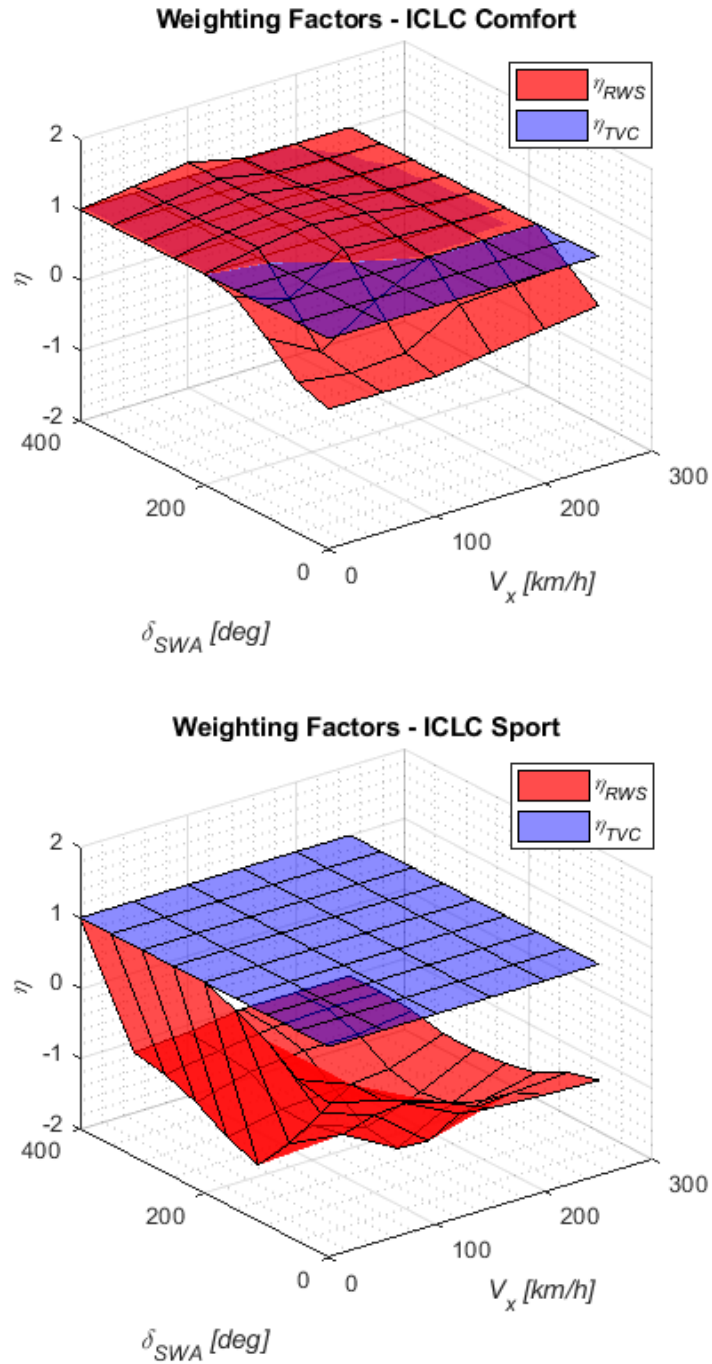


Figure 5.10: Optimum weighting factors of RWS and TVC actuators.

5.1 Robustness and conflicts

Since this study proposes a decentralized motion controller with downstream architecture, it was essential to investigate the possible robustness and redundancy issues due to overall communication. To achieve desired vehicle motion, integrated vehicle stability control systems typically rely on multiple actuators, such as torque vectoring and rear-wheel steering. However, actuators can fail for various reasons, such as mechanical or electrical faults. Fault-tolerant control using adaptive control allocation systems can detect and mitigate actuator failures in real time, allowing the system to maintain stability and control even in the presence of failures [68]. This increases the overall reliability and safety of the vehicle. Since this study already proposed two optimized and reliable chassis control systems, adaptive control allocation systems were not investigated. The proposed vehicle control strategy also allows the driver not to use a control system when the driver does not want to use it or in case of failure. For example, if the RWS system gives a failure and sport mode was selected already during the drive, the system automatically uses TVC sport mode. There can be an unintended characteristic change during switching from ICLC to TVC mode. However, since TVC has a yaw rate reference feedback, extreme changes are not expected. Nevertheless, similarly, if the TVC system gives a failure and sport mode was selected already during the drive, the system automatically uses RWS sport mode. Since RWS has side slip speed feedback, there will be no extreme slip changes. And, of course, the safety of the unintended yaw or slip characteristic was not a scope of this study.

Different actuators, such as torque vectoring and rear-wheel steering, may have overlapping control authority, leading to potential conflicts while reaching the defined targets. Vehicle motion control should prevent conflicts between actuators. As expressed in the previous section, only RWS contribution was optimized in this study, and RWS contribution should consider in an effective manner. For example, if the vehicle uses both RWS and TVC actuators %100 ($\eta_{RWS} = \eta_{TVC} = 1$), this scenario may cause some conflicts on both actuators and overall performance. There were two different ICLC modes defined and tested. From Figure 5.11, in the redundant ICLC mode (purple) RWS actuator was used %100 and added to ICLC. Since the yaw rate target was defined in the TVC system, the TVC actuator needed to generate more torque difference to meet the same yaw rate target. However, RWS actuators didn't

need to work much at those steering angles, so the optimum ICLC was added to the chart (orange) to understand the difference. The redundancy in the additional yaw moment can be seen from the left-bottom graph. The redundant ICLC produced 6700 Nm additional yaw moment to meet the same yaw rate target with unnecessary rear wheel steering input, which led to unneeded energy spent. Thus, the optimum ICLC strategy proposed that RWS should contribute after reaching tuneable steering angle values for each vehicle velocity level. The vehicle produced 3600 additional yaw moments with the optimum ICLC strategy, which was very close to only TVC mode. Thus, conflicts can be prevented by optimized weighting factors. The proposed idea was also tested in a double-lane change scenario. As shown in Figure 5.12, the optimum ICLC strategy showed the best performance in terms of exit speed with using less additional yaw moment. Thus, also DLC results showed that The RWS contribution directly affects the usage of TVC, and the RWS contribution should be chosen carefully to avoid redundancies.

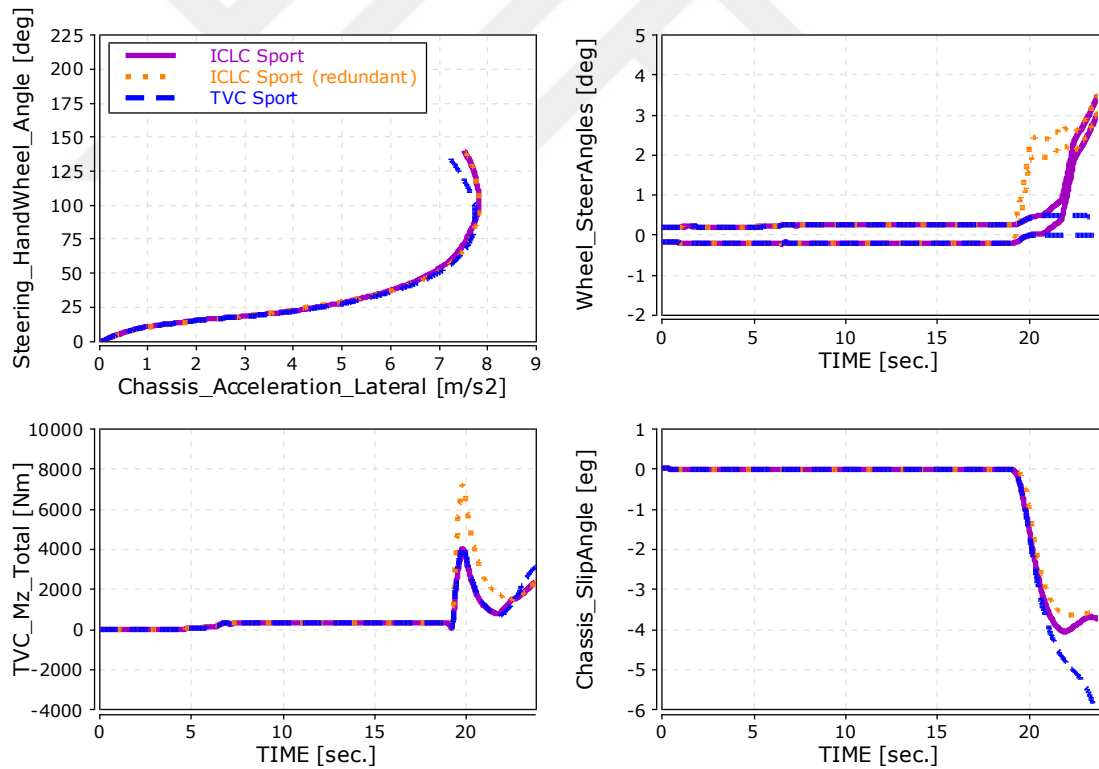


Figure 5.11: Redundancy example in ramp steer 100 kph.

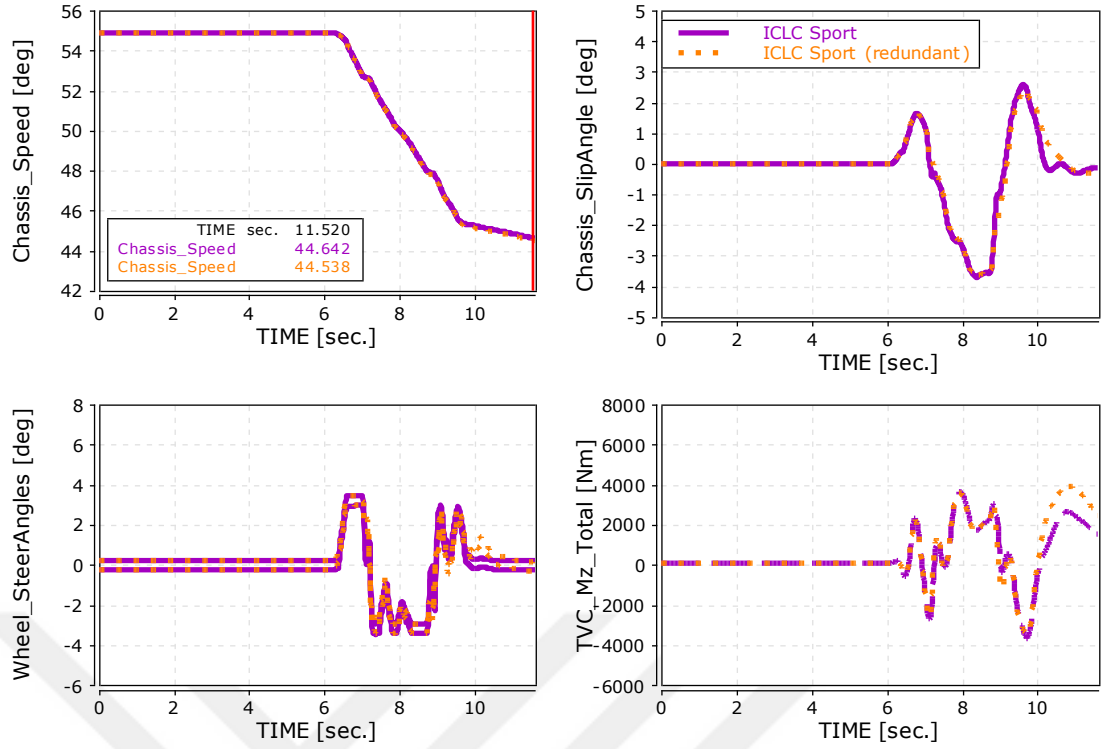


Figure 5.12: Redundancy example in DLC.

5.2 Integrated chassis lateral controller evaluation

The vehicle motion control was tested in ramp steer 50, 100 and 150 kph, step steer 100 kph, double-lane change 55 kph, and constant radius cornering 100 kph manoeuvres.

5.2.1 ICLC results - ramp steer - 50 kph

Firstly, two ICLC drive modes were evaluated in a 50 kph ramp steer manoeuvre. Figure 5.13 shows the understeer characteristic of baseline vehicle, comfort, and sport mode of the RWS, TVC and ICLC (combined). As expected, the linear understeer gradient was kept almost the same with ICLC. From the cornering capacity perspective, the vehicle reached 7.82 m/s^2 maximum lateral acceleration during the cornering with the ICLC comfort mode, it was 7.71 m/s^2 with only TVC. Thus, the vehicle motion control approach provided more lateral acceleration. A similar improvement was shown in the ICLC sport mode, the vehicle reached 7.83 m/s^2 with ICLC sport and this value is the maximum capacity that the vehicle can reach with the same tyres and maximum friction. Thus, the success of the integrated chassis lateral control on vehicle performance was shown. All results satisfied the expectations. The

detailed graph of the maximum acceleration level is shown in Figure 5.14. The vehicle dynamics states are shown in Figure 5.15. ICLC comfort and sport modes completed the manoeuvre with less side slip angle with respect to TVC comfort and sport modes. Figure 5.16 shows the usage of tyre saturations with ICLC and TVC comfort modes. From that figure, it was seen that ICLC was more successful than TVC, only to use rear tyre saturation more and at the end of the manoeuvre, ICLC front tyre saturation levels were lower. Therefore, ICLC was better suited to deal with the understeer characteristic.

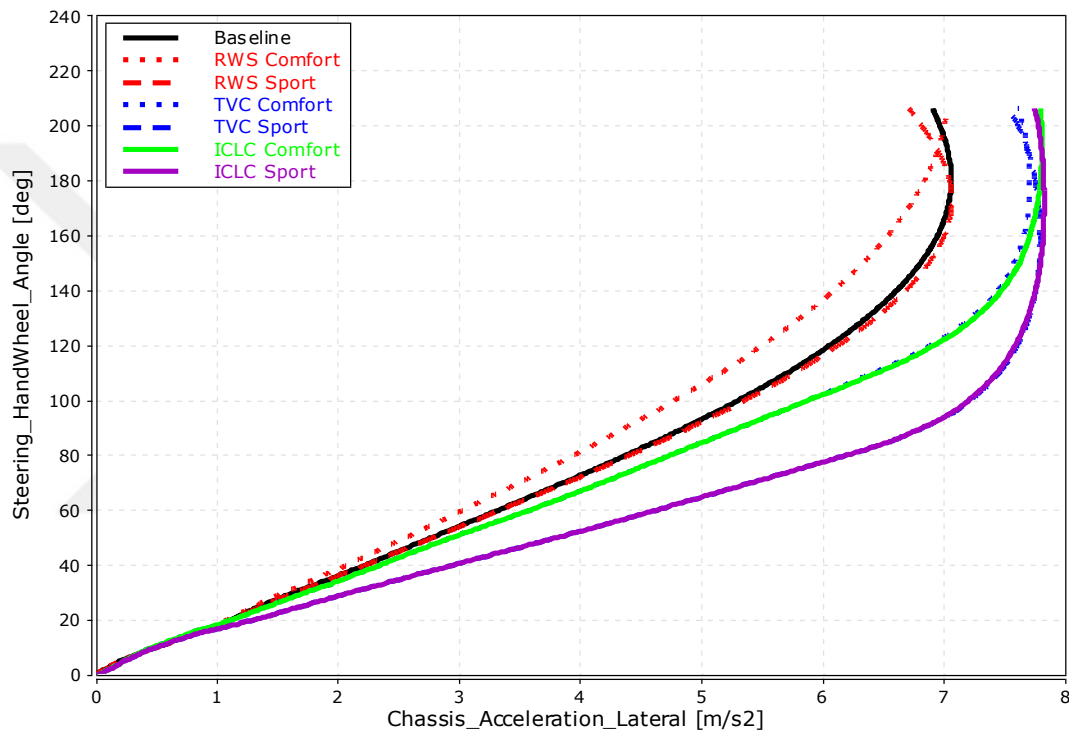


Figure 5.13: ICLC results - ramp steer 50 kph - steering ang. vs lateral acc.

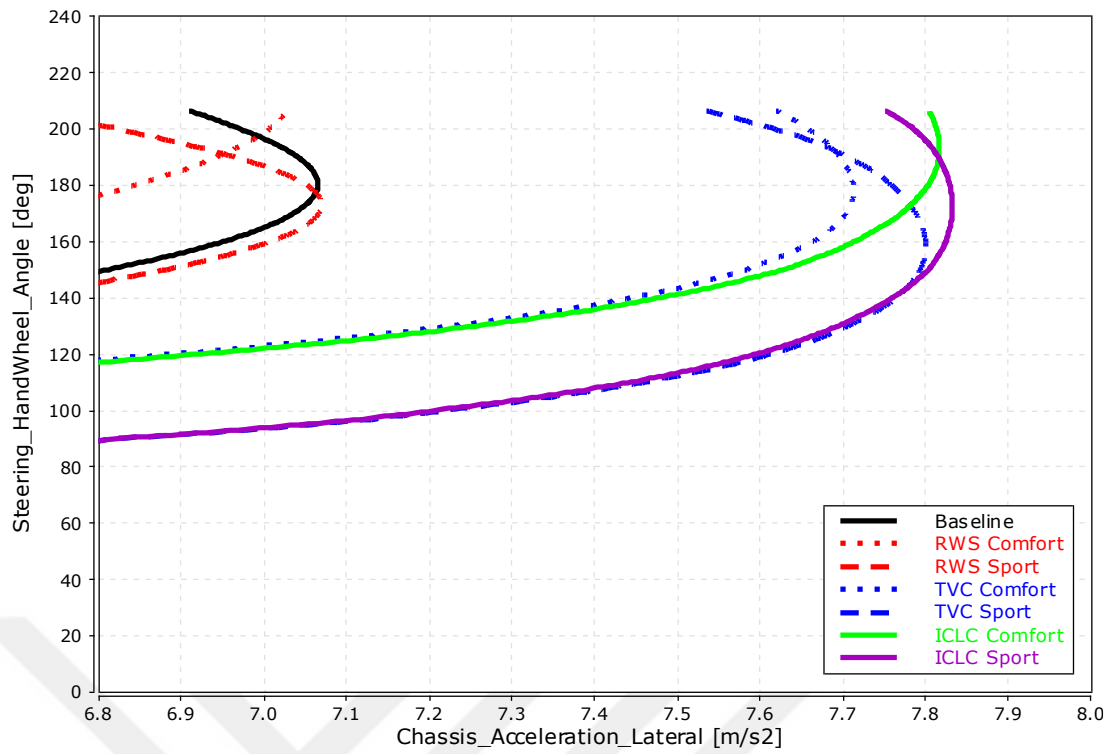


Figure 5.14: ICLC results - ramp steer 50 kph - max lateral acc. detail.

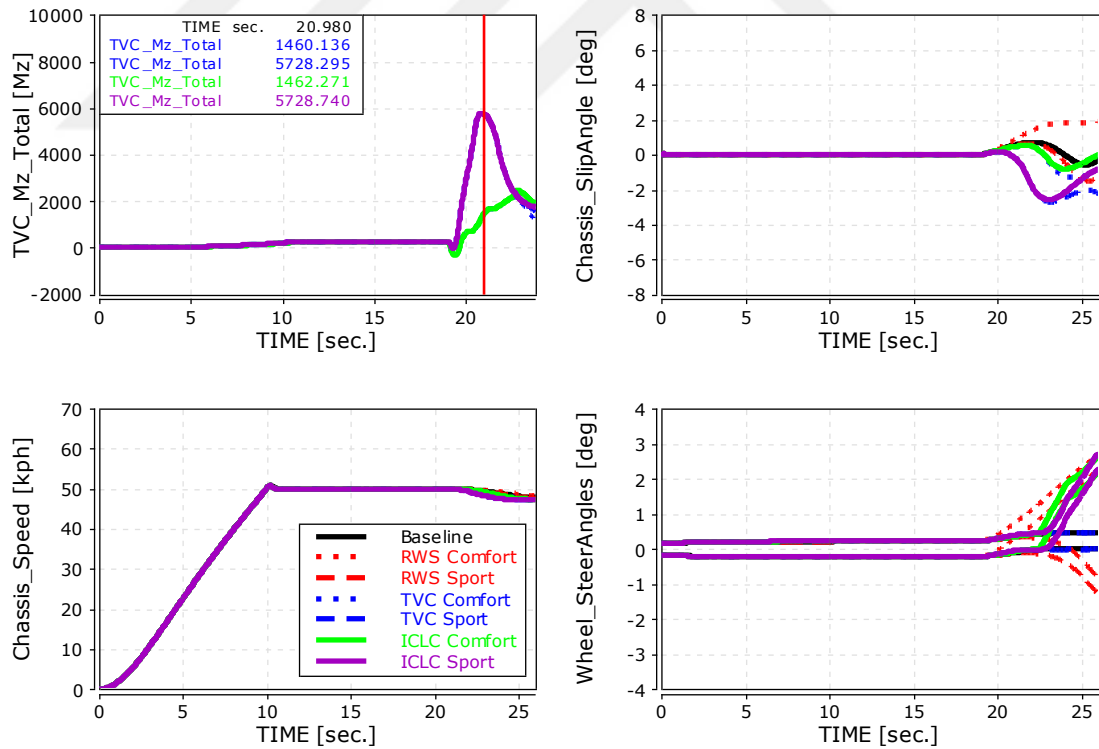


Figure 5.15: ICLC results - ramp steer 50 kph - vehicle dynamics states.

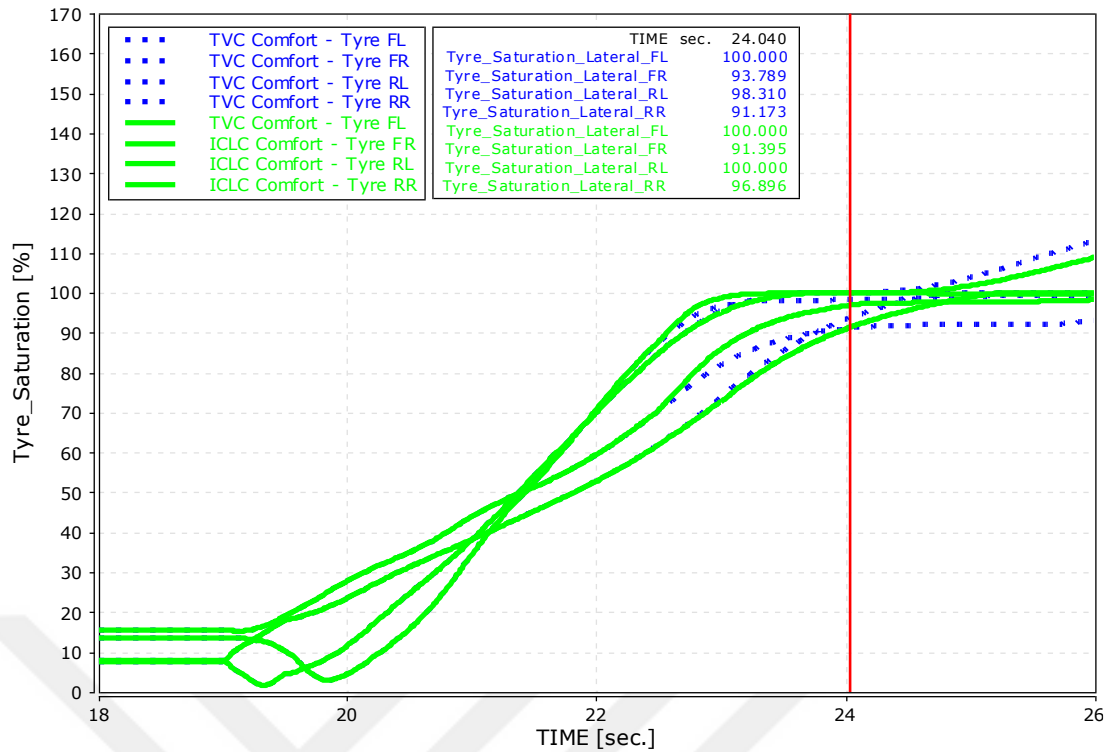


Figure 5.16: TVC vs ICLC comfort – ramp steer 50 kph - tyre saturation.

5.2.2 ICLC results - ramp steer - 100 kph

Figure 5.17 shows the understeer characteristic of baseline vehicle, comfort, and sport mode of the RWS, TVC and ICLC in ramp steer 100 kph. The vehicle reached 7.81 m/s² maximum lateral acceleration during the cornering with the ICLC comfort mode, it was 7.60 m/s² with only TVC. The similar improvement was shown in the sport mode, the vehicle reached 7.85 m/s² with ICLC sport and it was 7.76 m/s² with only TVC. Thus, integrated chassis lateral control enhanced vehicle handling at 100 kph. The vehicle dynamics states are shown in Figure 5.18. Both ICLC and TVC sport modes exceeded the side slip limits and behaved understeer characteristics at the end of the manoeuvre. This was because the safety factor η_{rg} in the TVC reference generator part was not fully optimized. Nevertheless, all results satisfied the expectations since all lateral acceleration capacities were increased.

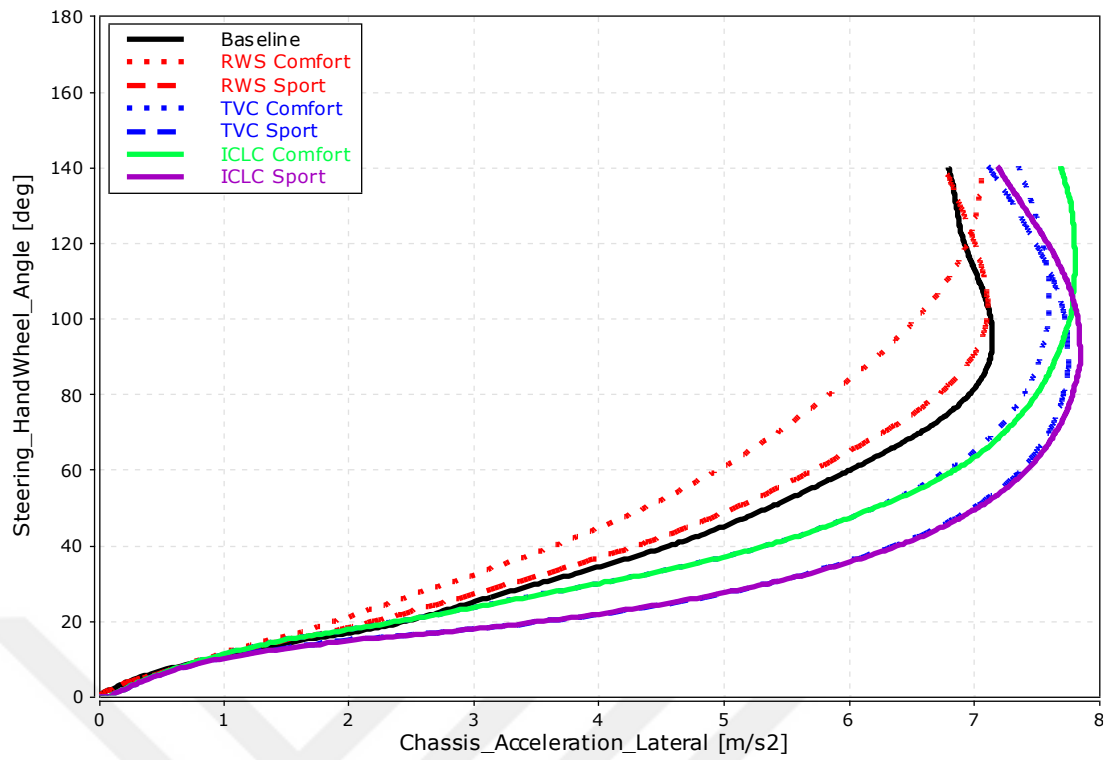


Figure 5.17: ICLC results - ramp steer 100 kph - steering ang. vs lateral acc.

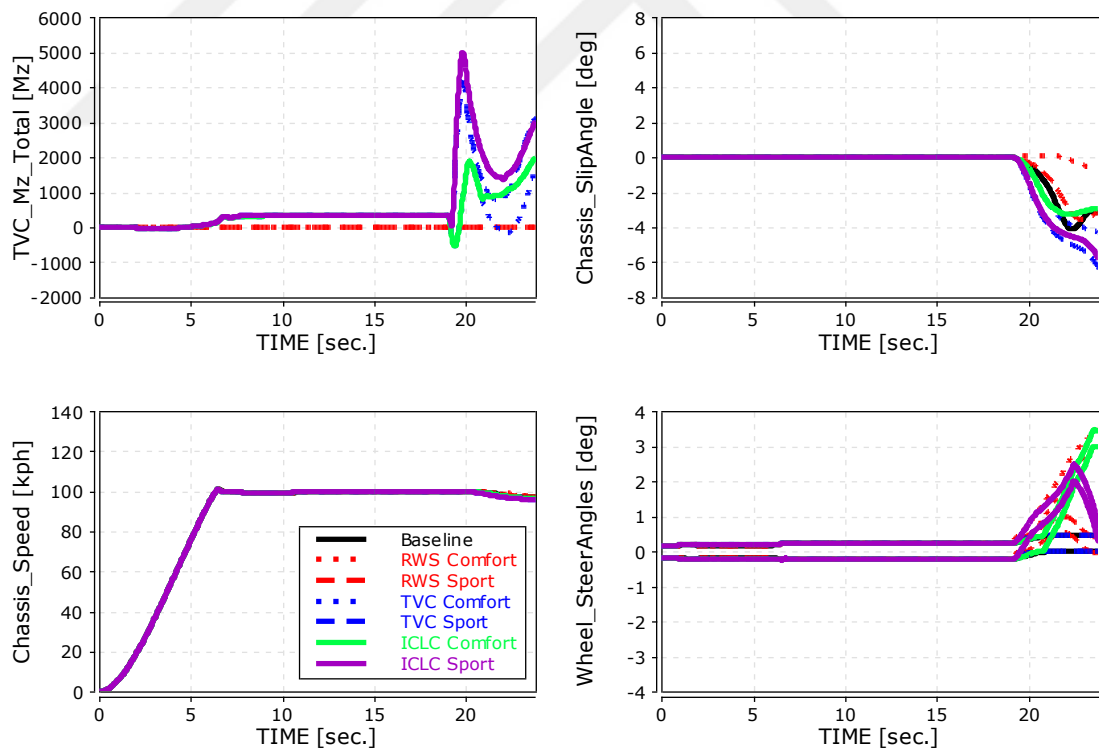


Figure 5.18: ICLC results - ramp steer 100 kph - vehicle dynamics states.

5.2.3 ICLC results - ramp steer - 150 kph

In the ramp steer 150 kph manoeuvre, both ICLC modes increased the lateral acceleration capacity. Figure 5.19 shows the understeer characteristic of baseline vehicle, comfort, and sport mode of the RWS, TVC and ICLC in ramp steer 150 kph. The vehicle reached 7.57 m/s^2 maximum lateral acceleration during the cornering with the ICLC comfort mode, and it was 7.31 m/s^2 with only TVC. A similar improvement was shown in the sport mode, the vehicle reached 7.58 m/s^2 with ICLC sport, and it was 7.40 m/s^2 with only TVC. Thus, integrated chassis lateral control enhanced vehicle handling at 150 kph. The vehicle dynamics states are shown in Figure 5.20. Since both ICLC comfort and sport mode side slip angles were relatively low, ICLC successfully prevented possible understeer characteristics at the end of the manoeuvre. All results satisfied the expectations since all lateral acceleration capacities were increased.

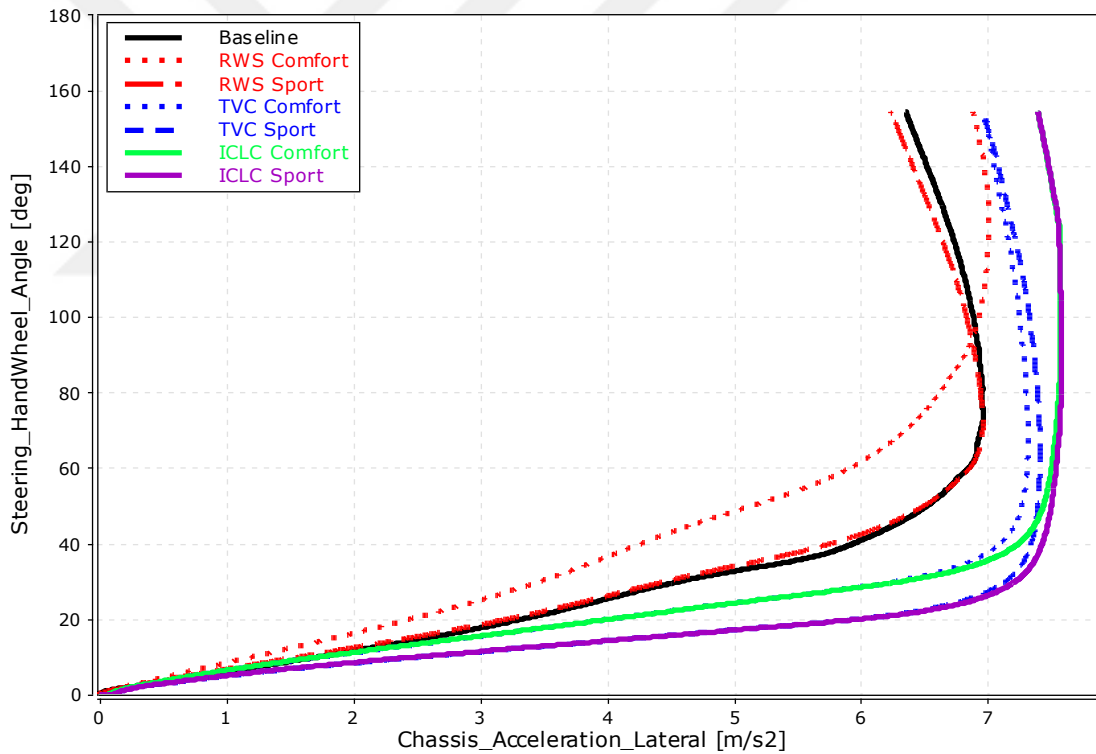


Figure 5.19: ICLC results - ramp steer 150 kph - steering ang. vs lateral acc.

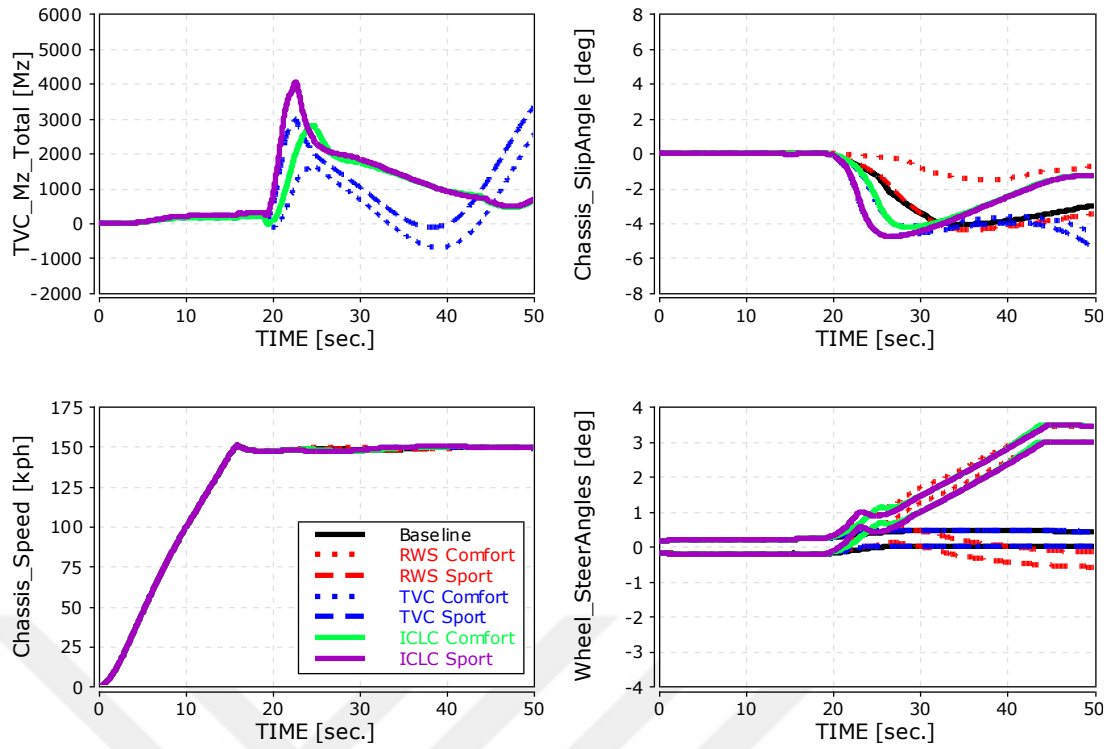


Figure 5.20: ICLC results - ramp steer 150 kph - vehicle dynamics states.

5.2.4 ICLC results - constant radius cornering

ICLC modes increased the lateral acceleration capacity in the constant radius cornering radius manoeuvre. Figure 5.21 shows the understeer characteristic of baseline vehicle, comfort, and sport mode of the RWS, TVC and ICLC in constant radius cornering manoeuvre. The vehicle reached 7.68 m/s^2 maximum lateral acceleration during the cornering with the ICLC comfort mode, and it was 7.49 m/s^2 with only TVC. The vehicle was reached 7.75 m/s^2 with ICLC sport and it was 7.60 m/s^2 with only TVC. ICLC worked well again and enhanced vehicle handling even in accelerated cornering scenarios. The vehicle dynamics states are shown in Figure 5.22. Interestingly, both ICLC modes need to generate higher additional yaw moments due to compensating RWS inputs between 10 and 30 seconds. The reason for this is that RWS was not optimized in the CRC scenario. Therefore, RWS worked undesirably for yaw rate tracking at the beginning of the scenario. Nevertheless, both ICLC modes could compensate for this undesirable behaviour. All results satisfied the expectations again since all lateral acceleration capacities were increased.

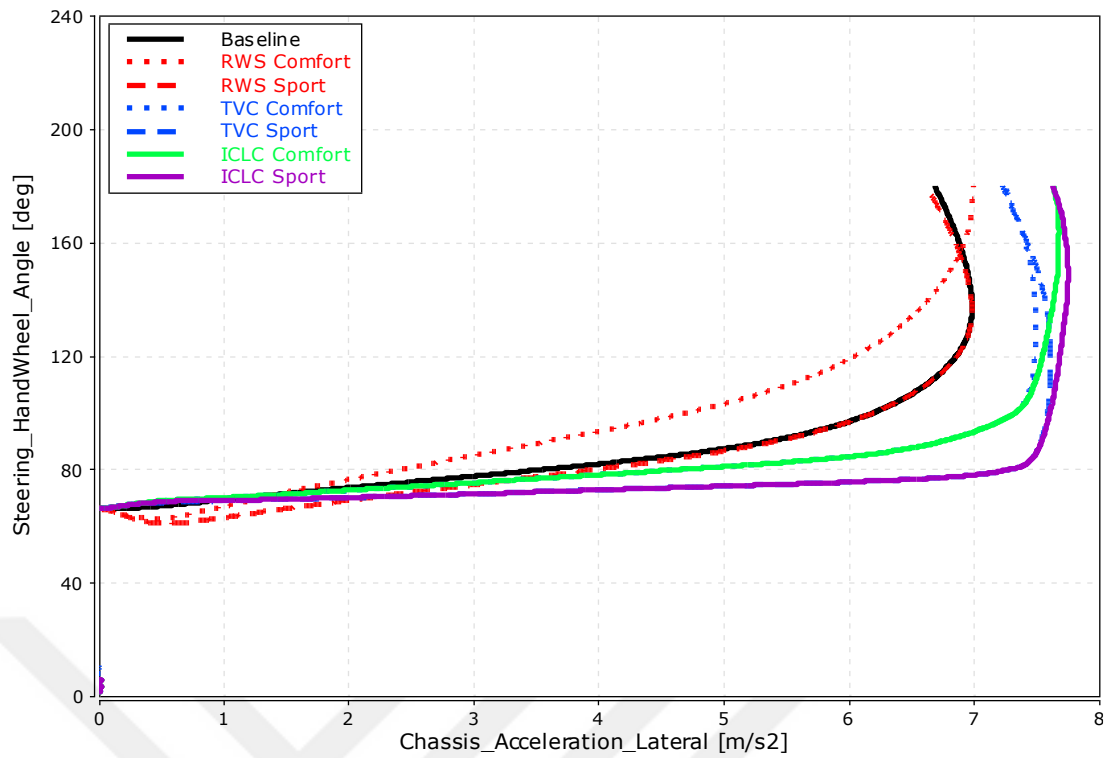


Figure 5.21: ICLC driving modes - CRC - steering ang. vs lateral acc.

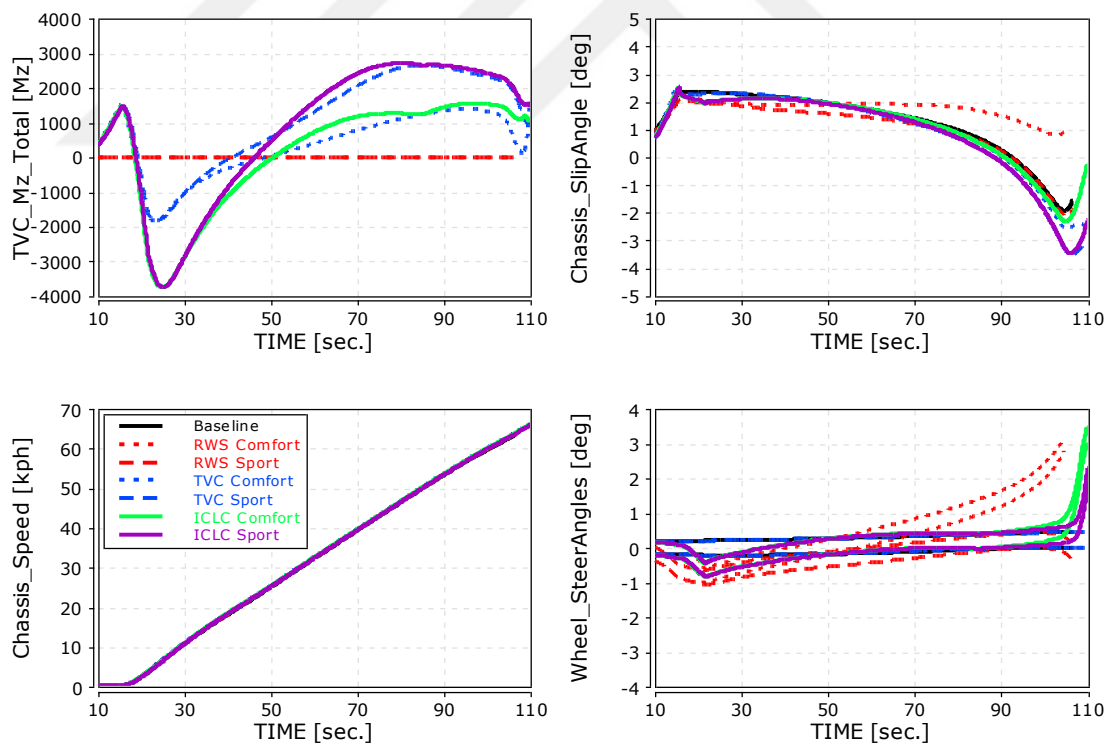


Figure 5.22: ICLC results - CRC - vehicle dynamics states.

5.2.5 ICLC results - double lane change

Both ICLC driving modes successfully completed the manoeuvre without exceeding the path limits with 55 kph entrance speed, as can be seen in Figure 5.23. Vehicle dynamics results of ICLC driving modes are given in Figure 5.24. The vehicle with the best exit speed (44.3 kph) was the ICLC sport. The maximum side slip angles during the manoeuvres for the ICLC comfort and sport modes were 2.8 and 3.0 degrees, respectively. Also, the ICLC comfort mode provides a higher exit speed with respect to other RWS and TVC-only modes. Since ICLC sport mode gave the best exit speed with a low side slip angle, ICLC sport mode showed the best performance in the DLC manoeuvre. ICLC sport mode provided less additional yaw moment with respect to TVC sport mode with the contribution of RWS actuator support. ICLC comfort and TVC comfort modes' additional yaw moment values were quite similar. Both ICLC comfort and sport modes reached all rear-wheel steering limits during the manoeuvre, but ICLC sport mode used RWS support less. In Figure 5.25, RWS Comfort, TVC Sport and ICLC sport modes, which were the scenarios that give the most successful results of all chassis control systems in the DLC scenario, were evaluated in terms of yaw rate tracking performance. From that figure, ICLC sport was closer to the target yaw rate during the manoeuvre. Here it can be said that RWS supported TVC for yaw rate tracking performance. Lastly, all chassis control system stability was checked in the side slip phase plane. Analysing the side slip phase plane can help identify potential stability issues and aid in the design and optimisation of chassis control systems [69]. It can be seen from Figure 5.26 that all chassis control functions and modes kept the vehicle in a stable region. And both ICLC comfort and sport mode gave the best results in terms of vehicle stability since their area was lower than other RWS and TVC modes.

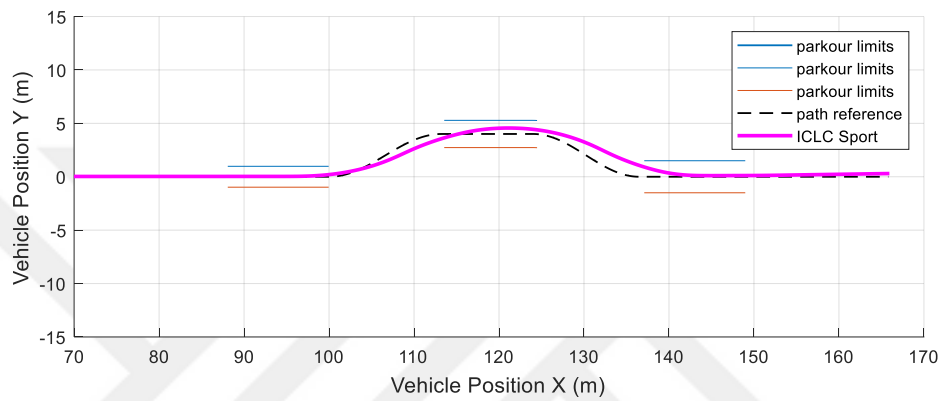
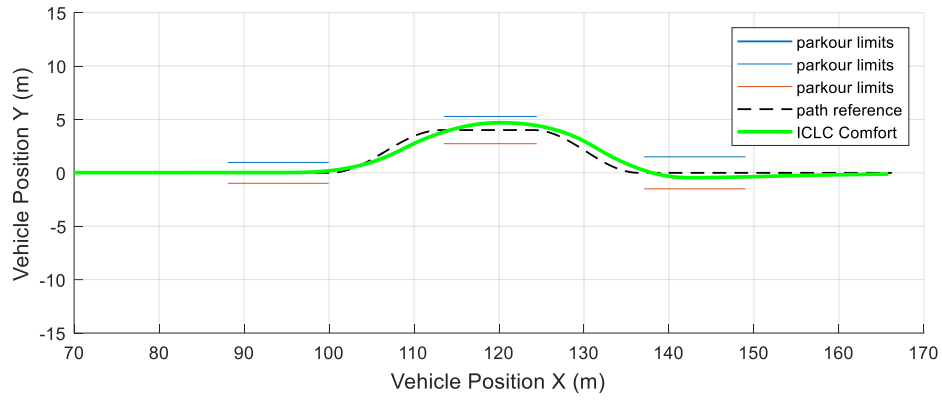


Figure 5.23: ICLC results - DLC - chassis trajectory.

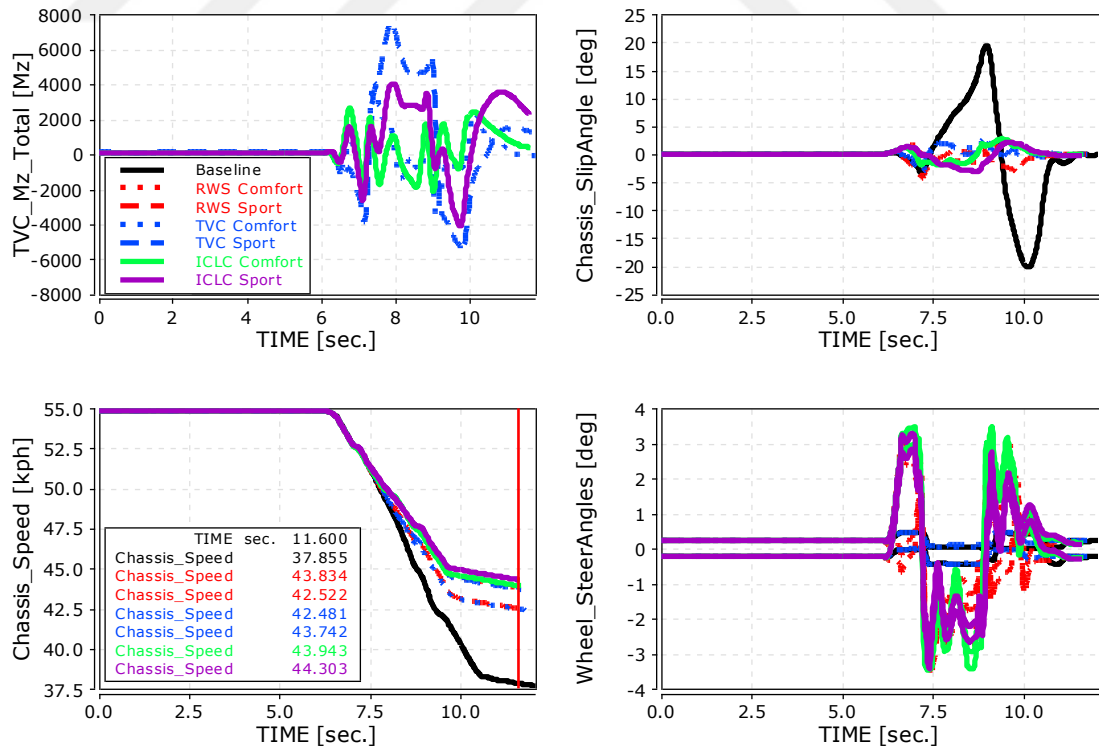


Figure 5.24: ICLC results - DLC - vehicle dynamics states.

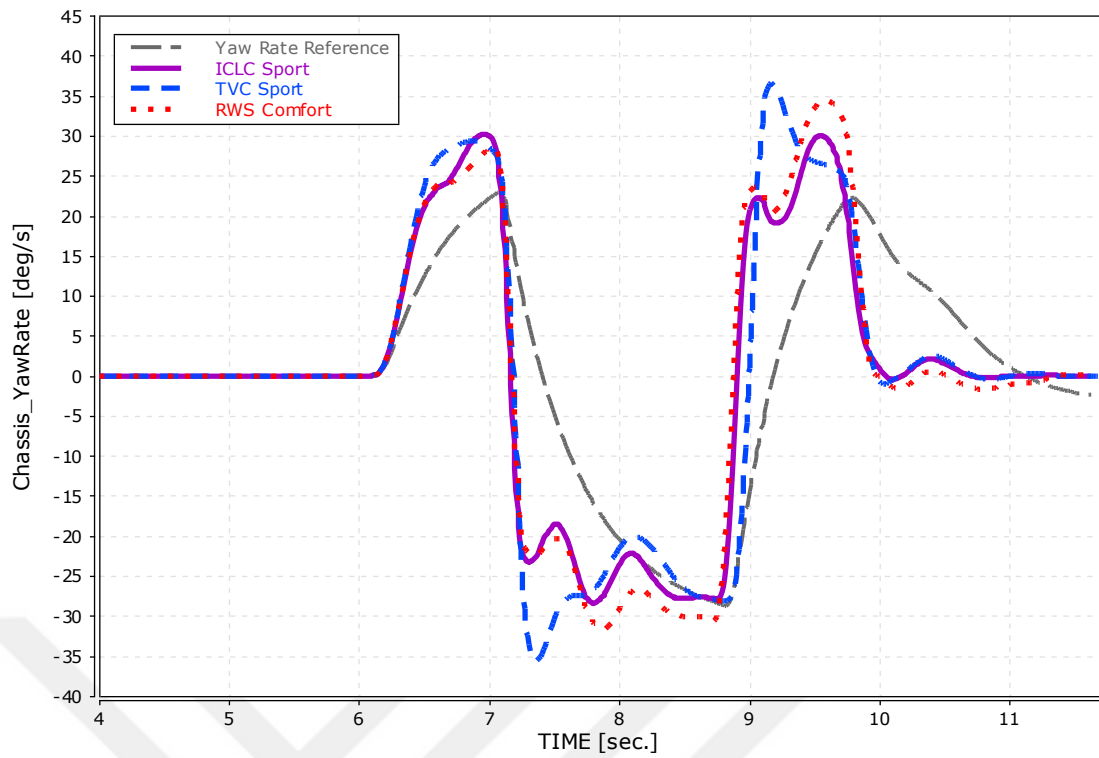


Figure 5.25: ICLC results - DLC - yaw rate tracking.

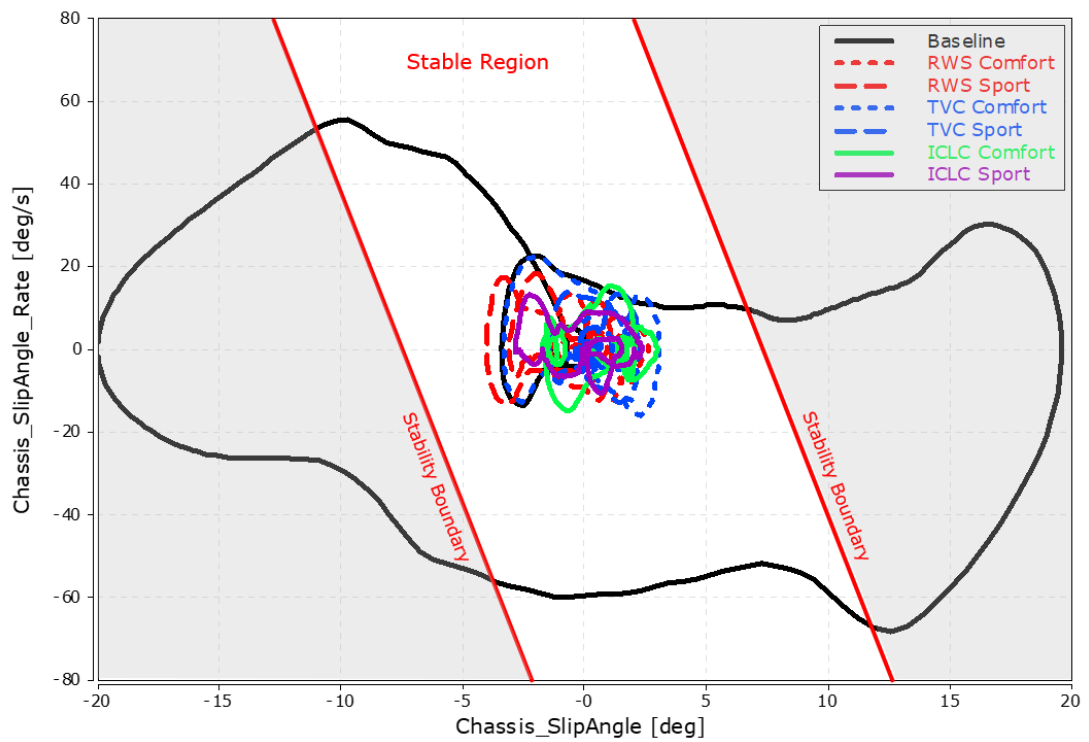


Figure 5.26: ICLC results – DLC - sideslip phase plane.

5.2.6 ICLC results - step steer

In the step steer manoeuvre, the results of ICLC driving modes were quite the same as the TVC modes, as expected. The reason for this, the usage of RWS actuators is already limited in low steering inputs to prevent redundancy. In addition, both TVC modes already performed well in the step steer manoeuvre with decreasing the overshoots. So already, there was no need for RWS support in the step steer. With both ICLC comfort and sport mode, the lateral acceleration didn't exceed the steady state value again. Figure 5.27 shows the vehicle dynamics states of each chassis controller in the step steer 100 kph manoeuvre.

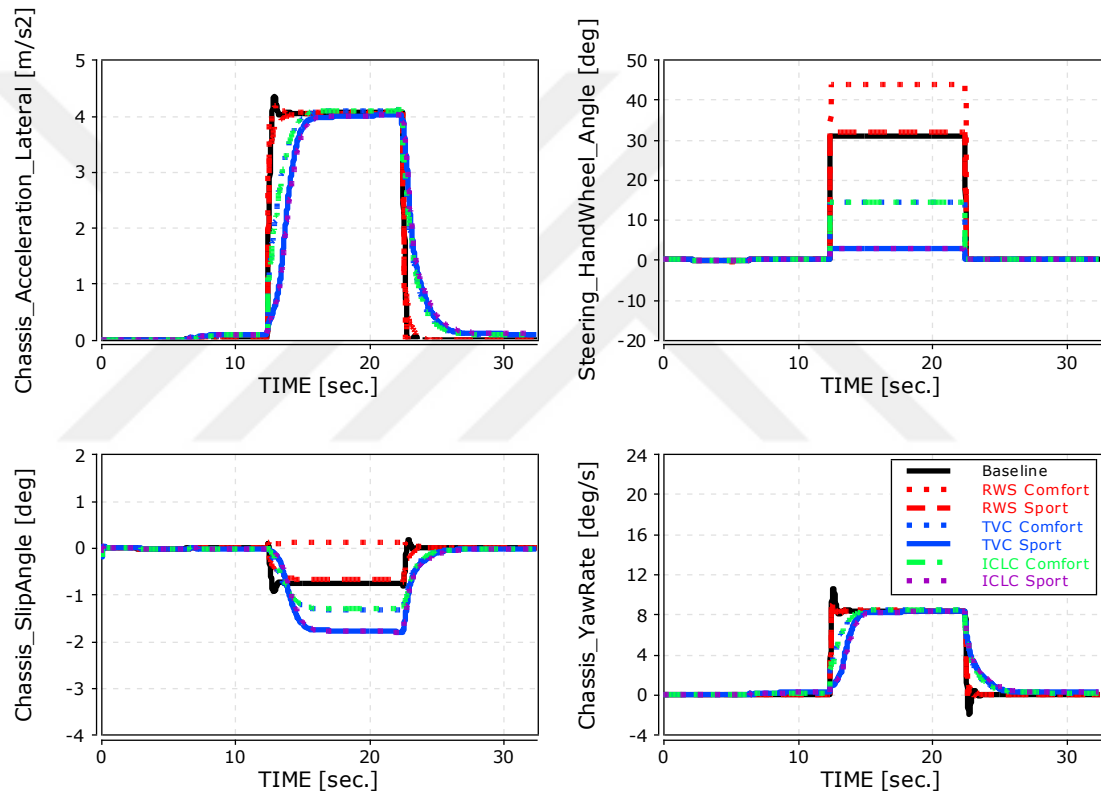


Figure 5.27: ICLC results - step steer 100 kph - vehicle dynamics states.

As a result, all ICLC drive modes and results were evaluated and optimized, considering possible redundancies, handling, and stability. According to the ICLC driving mode results, all ICLC modes enhanced the vehicle handling capability and improved only the TVC mode by using RWS support. As the best result, the ICLC sport mode gave the best performance in the DLC manoeuvre. All objective results were summarized in Table 5.1.

Table 5.1: ICLC improvements with respect to only TVC modes.

Test Case	Speed [kph]	Objective Measurement	Base	TVC Comfort Mode	ICLC Comfort Mode	TVC Sport Mode	ICLC Sport Mode
Ramp Steer	50	K_{us}	17.87	16.09	16.09	12.06	12.06
Ramp Steer	50	$a_{y,max}$	7.05	7.71	7.82	7.80	7.83
Ramp Steer	100	K_{us}	7.67	6.76	6.77	4.48	4.53
Ramp Steer	100	$a_{y,max}$	7.15	7.60	7.81	7.76	7.85
Ramp Steer	150	K_{us}	6.35	4.67	4.65	3.23	3.22
Ramp Steer	150	$a_{y,max}$	6.96	7.31	7.57	7.40	7.58
CRC	100	K_{us}	4.23	2.70	2.64	1.23	1.18
CRC	100	$a_{y,max}$	7.00	7.49	7.68	7.60	7.75
Step Steer	100	a_y overshoot difference [m/s ²]	0.350	0.00	0.00	0.00	0.00
DLC	55	β	20.0	3.5	2.8	2.0	3.0
DLC	55	$v_{x,exit}$	37.9	42.5	44.0	43.7	44.3

6. CONCLUSION

This research has been focused on to investigate the potential of improving lateral dynamics in electric vehicles through the combined use of rear-wheel steering (RWS) and torque vectoring control (TVC) systems with feedforward and feedback parts. The main motivation of the thesis is to examine the combination of two model-based control systems used in industry. The research objectives have been successfully achieved, and the findings contribute to vehicle dynamics and control. For the purpose, a non-linear vehicle and tyre model was used to design, implement, and optimize RWS, TVC and vehicle motion control systems to improve the handling dynamics of an electric vehicle. It is essential to have valid and compatible vehicle and tyre models for controller implantation. For this reason, before starting the control system modelling, firstly, the plausibility check of the vehicle/tyre model should be made. The passive yaw characteristics of the vehicle and the potentials of the tyres should be checked with open-loop manoeuvres. The need of a chassis control system for stability and handling improvement should be determined. Later, then it should be checked how much of this need can be met with chassis control systems.

Secondly, the design and optimization of both RWS and TVC system for different drive modes were accomplished. It was observed that RWS system is effective on the side-slip dynamics of the vehicle and RWS strategies based on the side slip angle can change the handling characteristics of the vehicle up to the RWS actuator limits. The RWS model with feedback part, which was tested on the bicycle model in the literature, was also tested on the non-linear vehicle model for the first time and its benefits were seen. The benefits of adding the feedback part to the RWS was seen in transient manoeuvres as it reduces the vehicle's slip speed. It was found that TVC system is effective on yaw rate dynamics via phase portrait analyses. TVC was chosen as the main contributor to the vehicle motion controller since the reference for handling and stability was the yaw rate. It was seen that generation of reference is very important part of active chassis control systems; thus, reference generation of yaw rate was modelled parametrically to cover all situations that vehicle can experienced. It was especially seen that the use of the feedback part in TVC, is very important in terms of

yaw rate reference tracking. Lastly, it was obtained that with TVC, the vehicle can be reached to ideal neutral driving behaviour with different TVC driving modes as desired.

This study was also aimed to propose an optimization methodology of RWS and TVC control improvement for vehicle handling performance in all defined test cases. The proposed optimization methodologies for RWS and TVC systems provide valuable insights into how these systems can be effectively optimized to enhance the handling performance of electric vehicles. However, since some manoeuvres were transient, optimization results couldn't give proper parameters for desired control. Therefore, fine-tuning will always be required on these chassis control systems.

Furthermore, integrating RWS and TVC systems into a vehicle motion control system, resulting in the developing of an Integrated Chassis Lateral Controller (ICLC) system, represents a significant contribution to the field. The ICLC system allows for coordinated control of the RWS and TVC systems, resulting in improved lateral dynamics and handling performance of the electric vehicle. Since RWS and TVC controllers have their own high-level controllers, the scenario of decentralized control was tested in this study. While combining RWS and TVC systems in the thesis, conflicts were avoided by optimization of weighting factors of those systems, but if the system had been combined in a more centralized approach, more intelligent control methods should be required in the future. In addition, the energy-efficient mode should also be evaluated for both system and all vehicle motion control. Safety critical points should be checked with the actual vehicle tests for further development.

Overall, the findings of this thesis provide valuable insight into the potential of RWS and TVC systems for improving lateral dynamics in electric vehicles. The research outcomes contribute to the advancement of electric vehicle technology and provide a foundation for future research in vehicle dynamics and control. The findings of this thesis have practical implications for offline optimization of advanced vehicle control systems, paving the way for enhanced handling performance and improved driving experience in electric vehicles.

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APPENDIX

APPENDIX A: RWS Tuneable Parameters Generation

```
%% Clean up
clc
clear all
close all

%% Model Parameters %%
% ISO axes are used in the model , left turn is positive side slip
(beta)
g = 9.80665;
m = 3000; %kg %Mass
J = 5000; %[kg*m^2] %Izz Inertia
Cf = -1800*180/pi; %[N/rad] %Front Tyre Linear Cornering Stiffness
Cr = -1800*180/pi; %[N/rad] %Rear Tyre Linear Cornering Stiffness
a = 1.40; %m %Front-Axle to COG distance
b = 1.66; %m %Rear-Axle to COG distance
W = m*g; %kg %Weight
L = a+b; %m %Lenght
d = 1.72 ; %wheeltrack

KUS = ((a/Cr)-(b/Cf))*W/L;

v_dec = 100; %decimation = 1/quantization of velocity
deltaf_dec = 40; %decimation = 1/quantization of front steering
angle
v_max = 200;
deltaf_max = 25;

x = linspace(0,1,v_dec);
d = linspace(0,1,deltaf_dec);

u = x.*v_max/3.6;
ratio_zss = zeros(1,v_dec);

%optimization parameters
a1 = -0.81;
b1 = 4.03;
c1 = -2.01;

a2 = -0.67;
b2 = 3.12;
c2 = -2.02;

a3 = -0.13;
b3 = 0.53;
c3 = -0.32;

y_ble = 18; %2nd blendind angle

%% parametrization of fist curve (ZSS - deltaf = 0)
```

```

RWS_first = c1*x.^2 + b1*x + a1;
if false
figure (1)
plot (x*v_max,RWS_first)
grid on
ylim([-2 2])
end
%% parametrization of 2nd curve

RWS_2nd = c2*x.^2 + b2*x + a2;
if true
figure (2)
plot (x*v_max,RWS_2nd, 'r','LineWidth',2)
grid on
ylim([-2 2])
xlabel('{V_x} [km/h]')
ylabel('\it{K_\delta} ')
end
%% parametrization of last curve

RWS_last = c3*x.^2 + b3*x + a3;
if true
figure (3)
plot (x*v_max,RWS_last, 'g','LineWidth',2)
grid on
ylim([-2 2])
xlabel('{V_x} [km/h]')
ylabel('\it{K_\delta} ')
end

for i=1:100
ratio_zss(i) = ((b*L)+m*(a/Cr)*(u(i).^2))/((-
(a*L)+m*(b/Cf)*(u(i).^2));
if true
figure(4)
plot(x*v_max,ratio_zss, 'k.', 'LineWidth',0.5)
grid on
xlabel('{V_x} [km/h]')
ylabel('\it{K_\delta} ')
ylim([-2 2])
end

end
hold on
plot(x*v_max,RWS_first, 'b', 'LineWidth',2)
title('RWS Steady-State Equations')
hold on
plot (x*v_max,RWS_2nd, 'r', 'LineWidth',2)
hold on
plot (x*v_max,RWS_last, 'g', 'LineWidth',2)
legend('Zero SideSlip Approach', 'eq1', 'eq2', 'eq3')

%% parametrization of feedforward transient curve
a4 = 1.413;
b4 = -1.426;
c4 = 0.634;

RWS_ff_n = c4*x.^2 + b4*x + a4;
if true

```

```

figure (6)
plot (x*v_max,RWS_ff_n,'b','LineWidth',2)
grid on
ylim([-5 5])
xlabel('\it{V_x} [km/h]')
ylabel('\it{n}')
title('FeedForward Transient Map')
end

%% parametrization of feedback curve

a5 = 0.01;
b5 = 0.4;
c5 = 0.4;

RWS_fb_K = (c5*x.^2 + b5*x + a5)/100;
if true
figure (7)
plot (x*v_max,RWS_fb_K,'b','LineWidth',2)
grid on
ylim([-0.02 0.02])
xlabel('\it{V_x} [km/h]')
ylabel('\it{K_f_b}')
title('FeedBack Transient Map')
end

%%
x1 = x*v_max;

y1 = repelem(1,100);
y2 = repelem(y_ble ,100);
y3 = repelem(36.2,100);
eq1 = RWS_first; % RWS_first %ratio_zss
eq2 = RWS_2nd;
eq3 = RWS_last;

xv = linspace(min(x), max(x), 25);
y1v = linspace(min(y1), max(y1), 25);
y2v = linspace(min(y2), max(y2), 25);
y3v = linspace(min(y3), max(y3), 25);
[Xm,Ym] = ndgrid([xv(:); xv(:); xv(:)], [y1v(:); y2v(:); y3v(:)]);
% Zm = griddata([x(:); x(:); x(:); x(:)], [y1v(:); y2v(:); y3v(:);
y4v(:)], [eq1(:); eq2(:); eq3(:); eq4(:)], Xm, Ym);

figure(8)
plot3(x*v_max, y1, eq1, '-b', 'LineWidth',2)
grid on
hold on
plot3(x*v_max, y2, eq2, '-r', 'LineWidth',2)
hold on
plot3(x*v_max, y3, eq3, '-g', 'LineWidth',2)
legend('eq1', 'eq2', 'eq3')
xlabel('\it{V_x} [km/h]')
ylabel('\it{\delta_f} [deg]')
zlabel('\it{K_\delta}')
title('RWS Steady-State Map')
% patch([y1 flip(y2)], [x flip(x)], [eq1 flip(eq2)], 'g',
'FaceAlpha',0.25, 'EdgeColor','none') % 'patch'

```

```

%%

if false
figure (9)
s1 =
surf(ChassisSpeed,RoadWheelAngleMeanFront,RoadWheelAngleMeanRear);
set(s1,'facealpha',1)
set(s1,'edgealpha',1)
grid on
xlabel('\it{V_x} [km/h]')
ylabel('\it{\delta_f} [deg]')
zlabel('\it{\delta_r} [deg]')
end

if true
figure (10)
% s2 =
surf(ChassisSpeed,RoadWheelAngleMeanFront,RWSRatio,'FaceColor','g');
% % shading interp;
% % colormap winter;
% set(s2,'facealpha',1)
% set(s2,'edgealpha',1)
% hold on
s3 = surf([x*v_max; x*v_max; x*v_max],[y1; y2; y3], [eq1; eq2; eq3],
'MeshStyle','row')
'surf'
grid on
set(s3,'facealpha',0.7)
set(s3,'edgealpha',0.7)
grid on
xlabel('\it{V_x} [km/h]')
ylabel('\it{\delta_f} [deg]')
zlabel('\it{K_\delta}')
title('RWS Steady-State Map')

end

x_RWS_cal = x*v_max;
y_RWS_cal= [y1(:,1) y2(:,1) y3(:,1)];
z_RWS_cal = [eq1; eq2; eq3];
z_RWS_cal_surf = smoothdata(z_RWS_cal);

if true
figure (11)
s4 = surf(x_RWS_cal,y_RWS_cal,z_RWS_cal,'MeshStyle','row');
% shading interp;
% colormap winter;
set(s4,'facealpha',0.7)
set(s4,'edgealpha',0.5)
xlabel('\it{V_x} [km/h]')
ylabel('\it{\delta_f} [deg]')
zlabel('\it{K_\delta}')
hold on
s5 = surf(x_RWS_cal,y_RWS_cal,z_RWS_cal_surf,'FaceColor','r');
title('RWS Steady-State Map')
legend('RWS ss map', 'RWS ss map (smooth)')
end

```

APPENDIX B: TVC Tuneable Parameters Generation

```
%% Clean up
clc
clear all
close all

%% TVC Optimization Parameters

%%
%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%% kP %%%%%%%%%
%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%

%% kP - velocity

TVC_Kp_V_1 = 8.2;
TVC_Kp_V_2 = -0.06;
TVC_Kp_V_3 = 15.75;
TVC_Kp_V_4 = -1753.5;
TVC_Kp_V_5 = 100000;

TVC_Kp_V_x = [0 10.4 20.8 31.25 41.6 52.08 62.5 72.91 83.33 93.75
104.16 114.58 125 135.41 145.83 156.25 166.66 177.08 187.5 197.91
208.33 218.75 229.16 239.58 250];
TVC_Kp_V_y = (TVC_Kp_V_1*1e-005).*TVC_Kp_V_x.^4 +
(TVC_Kp_V_2).*TVC_Kp_V_x.^3 + (TVC_Kp_V_3).*TVC_Kp_V_x.^2 +
(TVC_Kp_V_4).*TVC_Kp_V_x + (TVC_Kp_V_5);

for i = 1:length(TVC_Kp_V_x)
    if TVC_Kp_V_y(i) > 200000
        TVC_Kp_V_y(i) = 200000;
    end
    if TVC_Kp_V_y(i) < 0
        TVC_Kp_V_y(i) = 0;
    end
end

figure (1)
subplot(2,3,1)
plot(TVC_Kp_V_x,TVC_Kp_V_y,'LineWidth',2)
grid on
xlabel('{Velocity [kph]}')
ylabel('{K_p_V} [-]')
title('K_p Velocity Schedule')
legend ('TVC-Kp-V')
ylim([0 200000]);

%% kP - acceleration

TVC_Kp_A_1 = 0.29;
TVC_Kp_A_2 = 0.71;
TVC_Kp_A_ble = 7;

TVC_Kp_A_x = [0 1 2 3 4 5 6 7 8 9 10] ;

TVC_Kp_A_max = 10;
TVC_Kp_A_dec = length(TVC_Kp_A_x);

for i = 1:TVC_Kp_A_dec
```

```

a = (i/TVC_Kp_A_dec)*TVC_Kp_A_max;
if (a <= TVC_Kp_A_ble)
    gain = 1;
else
    gain = 0;
end
TVC_Kp_A_y(i,:) = (TVC_Kp_A_1*gain + TVC_Kp_A_2)^2;
end

for i = 1:length(TVC_Kp_A_x)
    if TVC_Kp_A_y(i) > 1
        TVC_Kp_A_y(i) = 1;
    end
    if TVC_Kp_A_y(i) < 0
        TVC_Kp_A_y(i) = 0;
    end
end

figure (1)
subplot(2,3,4)
plot(TVC_Kp_A_x,TVC_Kp_A_y, 'LineWidth',2)
grid on
xlabel('{Acceleration [m/s^2]}')
ylabel('{K_p_ay} [-]')
title('K_p Acceleration Schedule')
legend ('TVC-Kp-A')
ylim([0 1]);

%%
%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%% kI %%%%%%%%%
%% kI - velocity
TVC_Ki_V_1 = 1.0799;
TVC_Ki_V_2 = -7.506;
TVC_Ki_V_3 = 0.1921;
TVC_Ki_V_4 = -21.783;
TVC_Ki_V_5 = 1000;

TVC_Ki_V_x = [0 10.4 20.8 31.25 41.6 52.08 62.5 72.91 83.33 93.75
104.16 114.58 125 135.41 145.83 156.25 166.66 177.08 187.5 197.91
208.33 218.75 229.16 239.58 250];
TVC_Ki_V_y = (TVC_Ki_V_1*1e-006).*TVC_Ki_V_x.^4 + (TVC_Ki_V_2*1e-
004).*TVC_Ki_V_x.^3 + TVC_Ki_V_3.*TVC_Ki_V_x.^2 +
TVC_Ki_V_4.*TVC_Ki_V_x + (TVC_Ki_V_5);

for i = 1:length(TVC_Ki_V_x)
    if TVC_Ki_V_y(i) > 2000
        TVC_Ki_V_y(i) = 2000;
    end
    if TVC_Ki_V_y(i) < 0
        TVC_Ki_V_y(i) = 0;
    end
end

figure (1)
subplot(2,3,2)
plot(TVC_Ki_V_x,TVC_Ki_V_y, 'LineWidth',2)
grid on
xlabel('{Velocity [kph]}')
ylabel('{K_i_V} [-]')
title('K_i Velocity Schedule')

```

```

legend ('TVC-Ki-V')
ylim([0 2000]);

%% kI - acceleration
TVC_Ki_A_1 = 0.29;
TVC_Ki_A_2 = 0.71;
TVC_Ki_A_ble = 7;

TVC_Ki_A_x = [0 1 2 3 4 5 6 7 8 9 10] ;

TVC_Ki_A_max = 10;
TVC_Ki_A_dec = length(TVC_Ki_A_x);

for i = 1:TVC_Ki_A_dec
    a = (i/TVC_Ki_A_dec)*TVC_Ki_A_max;
    if (a <= TVC_Ki_A_ble)
        gain = 1;
    else
        gain = 0;
    end
    TVC_Ki_A_y(i,:) = (TVC_Ki_A_1*gain + TVC_Ki_A_2)^2;
end

for i = 1:length(TVC_Ki_A_x)
    if TVC_Ki_A_y(i) > 1
        TVC_Ki_A_y(i) = 1;
    end
    if TVC_Ki_A_y(i) < 0
        TVC_Ki_A_y(i) = 0;
    end
end

figure (1)
subplot(2,3,5)
plot(TVC_Ki_A_x,TVC_Ki_A_y, 'LineWidth',2)
grid on
xlabel('{Acceleration [m/s^2]}')
ylabel('{K_i_ay} [-]')
title('K_i Acceleration Schedule')
legend ('TVC-Ki-A')
ylim([0 1]);

%%
%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%% kD %%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%

%% kD - velocity

TVC_Kd_V_1 = 8.2;
TVC_Kd_V_2 = -0.06;
TVC_Kd_V_3 = 15.75;
TVC_Kd_V_4 = -1753.5;
TVC_Kd_V_5 = 110000;

TVC_Kd_V_x = [0 10.4 20.8 31.25 41.6 52.08 62.5 72.91 83.33 93.75
104.16 114.58 125 135.41 145.83 156.25 166.66 177.08 187.5 197.91
208.33 218.75 229.16 239.58 250];

```

```

TVC_Kd_V_y = (TVC_Kd_V_1*1e-005).*TVC_Kd_V_x.^4 +
(TVC_Kd_V_2).*TVC_Kd_V_x.^3 + TVC_Kd_V_3.*TVC_Kd_V_x.^2 +
TVC_Kd_V_4.*TVC_Kd_V_x + (TVC_Kd_V_5);

for i = 1:length(TVC_Kd_V_x)
    if TVC_Kd_V_y(i) > 200000
        TVC_Kd_V_y(i) = 200000;
    end
    if TVC_Kd_V_y(i) < 0
        TVC_Kd_V_y(i) = 0;
    end
end

figure (1)
subplot(2,3,3)
plot(TVC_Kd_V_x,TVC_Kd_V_y,'LineWidth',2)
grid on
xlabel('{Velocity [kph]}')
ylabel('{K_d_V} [-]')
legend('TVC-Kd-V')
title('K_d Velocity Schedule')
ylim([0 200000]);

%% kD - acceleration

TVC_Kd_A_1 = 0.29;
TVC_Kd_A_2 = 0.71;
TVC_Kd_A_ble = 7;

TVC_Kd_A_x = [0 1 2 3 4 5 6 7 8 9 10] ;

TVC_Kd_A_max = 10;
TVC_Kd_A_dec = length(TVC_Kd_A_x);

for i = 1:TVC_Kd_A_dec
    a = (i/TVC_Kd_A_dec)*TVC_Kd_A_max;
    if (a <= TVC_Kd_A_ble)
        gain = 1;
    else
        gain = 0;
    end
    TVC_Kd_A_y(i,:) = (TVC_Kd_A_1*gain + TVC_Kd_A_2)^2;
end

for i = 1:length(TVC_Kd_A_x)
    if TVC_Kd_A_y(i) > 1
        TVC_Kd_A_y(i) = 1;
    end
    if TVC_Kd_A_y(i) < 0
        TVC_Kd_A_y(i) = 0;
    end
end

figure (1)
subplot(2,3,6)
plot(TVC_Kd_A_x,TVC_Kd_A_y,'LineWidth',2)
grid on
xlabel('{Acceleration [m/s^2]}')
ylabel('{K_d_ay} [-]')
title('K_d Acceleration Schedule')

```

```

legend ('TVC-Kd-A')
ylim([0 1]);

%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%% ref generator %%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%
TVC_UG_Des = 1;
TVC_n1 = 0.55;
TVC_n2 = 12;
TVC_esp = 5;

v_dec = 100; %decimation = 1/quantization of velocity
v_max = 200;
x = linspace(0,1,v_dec);

%RG optimization parameters
a_ug_des = 15;
b_ug_des = -10;
c_ug_des = 4;

% a_ug_des = 26.3;
% b_ug_des = -48.3;
% c_ug_des = 25.1;

a_n1 = 0.6;
b_n1 = -0.4;
c_n1 = 0.2;

a_n2 = 15;
b_n2 = -6;
c_n2 = 2;

a_esp = 14;
b_esp = -5;
c_esp = 2;

%% parametrization of fist curve (ZSS - deltaf = 0)

TVC_RG_UG_des = c_ug_des*x.^2 + b_ug_des*x + a_ug_des;
TVC_RG_n1 = c_n1*x.^2 + b_n1*x + a_n1;
TVC_RG_n2 = c_n2*x.^2 + b_n2*x + a_n2;
TVC_RG_esp = c_esp*x.^2 + b_esp*x + a_esp;

if true
figure (2)
subplot(2,2,1)
plot (x*v_max,TVC_RG_UG_des,'LineWidth',2)
grid on
xlabel('{Velocity [kph]}')
ylabel('{K_u_s,_l_i_n_e_a_r} [-]')
title('K_u_s,_l_i_n_e_a_r')
legend ('K_u_s,_l_i_n_e_a_r')
ylim([0 40]);
end

if true
figure (2)
subplot(2,2,2)
plot (x*v_max,TVC_RG_n1,'LineWidth',2)
grid on

```

```

xlabel('{Velocity [kph]}')
ylabel('{n_1} [-]')
title('n_1')
legend ('n_1')
ylim([0 1]);
end

if true
figure (2)
subplot(2,2,3)
plot (x*v_max,TVC_RG_n2,'LineWidth',2)
grid on
xlabel('{Velocity [kph]}')
ylabel('{n_2} [-]')
title('n_2')
legend ('n_2')
ylim([0 20]);
end

if true
figure (2)
subplot(2,2,4)
plot (x*v_max,TVC_RG_esp,'LineWidth',2)
grid on
xlabel('{Velocity [kph]}')
ylabel('{e_1} [-]')
title('e_1')
legend ('e_1')
ylim([0 20]);
end

```

CURRICULUM VITAE

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- **B.Sc.** : 2018, Bilkent University, Mechanical Engineering
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PUBLICATIONS, PRESENTATIONS AND PATENTS ON THE THESIS:

- 10th International Congress on Engineering, Architecture and Design, Lateral Vehicle Dynamics Simulation Tool for Integrated Chassis Control Systems December 2022.

OTHER PUBLICATIONS, PRESENTATIONS AND PATENTS:

- International Journal of Vehicle Design, Vol 89 , Ride comfort parameter optimization of an 8x8 armoured military vehicle using Taguchi method, September 2021
- European Control Conference 2023, Stanley Controller Design for Enhancing Vehicle Lane Keeping and Departure Performance Using Active Rear Wheel Steering, March 2023