ISTANBUL TECHNICAL UNIVERSITY ★ GRADUATE SCHOOL

MODEL REFERENCE ADAPTIVE CONTROLLER DESIGN WITH AUGMENTED ERROR METHOD FOR LANE TRACKING

M.Sc. THESIS

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İSTANBUL TEKNİK ÜNİVERSİTESİ ★ LİSANSÜSTÜ EĞİTİM ENSTİTÜSÜ

ŞERİT TAKİBİ KONTROLÜ İÇİN ARTIRILMIŞ HATA YÖNTEMİ İLE MODEL REFERANS UYARLANABİLİR KONTROLÖR TASARIMI

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KASIM 2023

Mehmet Nuri DİYİCİ, a M.Sc. student of ITU Graduate School student ID 518201038 successfully defended the thesis entitled “MODEL REFERENCE ADAPTIVE CONTROLLER DESIGN WITH AUGMENTED ERROR METHOD FOR LANE TRACKING”, which he/she prepared after fulfilling the requirements specified in the associated legislations, before the jury whose signatures are below.

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Date of Submission : 08 November 2023
Date of Defense : 20 November 2023

To my spouse and children,

FOREWORD

Autonomous driving could not be considered the main intention of designing the first automobile. Still, considerable improvements throughout the history of the automobile industry have made autonomous vehicles inevitable. Furthermore, automobiles have started to be called smart devices thanks to the rapid evolution of computerization in automobiles. In fact, ADAS technologies are the core of these smart devices. Considering that control is the basis of these technologies, this thesis will provide a piece of important knowledge about autonomous driving and advanced driver assistance systems. Specifically, an adaptive controller design, based on passivity, for lane tracking control of autonomous vehicles is explained in detail, and its performance is simulated. I hope this thesis inspires those who seek a solid and applicable solution for lane-tracking control of autonomous vehicles.

I would like to seize this chance to convey my deep appreciation. Foremost, I extend my sincere appreciation to my esteemed advisor, Doc. Dr. Yaprak Yalçın, whose invaluable guidance and wealth of knowledge have played an essential role in shaping the research presented in this thesis. I am also profoundly grateful to FEV Turkey Smart Vehicle Development Department, whose unwavering support has been instrumental during my tenure as a graduate student.

Lastly, I wish to convey my profound gratitude to my parents, whose steadfast support and unwavering presence have been a source of immense motivation throughout my educational journey. Their boundless encouragement has been a cornerstone of my academic achievements.

November 2023

Mehmet Nuri DİYİCİ
Senior Control & Software Engineer

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ABBREVIATIONS

ECU	: Electronic Control Unit
AI	: Artificial Intelligence
SAE	: Society of Automative Engineers
ADAS	: Advanced Driver Assistance Systems
ACC	: Adaptive Cruise Control
AEBS	: Advanced Emergency Braking System
LKA	: Lane Keeping Assistance
LDS	: Lane Detection Systems
BSD	: Blind Spot Detection
BSW	: Blind Spot Warning
LQR	: Linear Quadratic Requlator
NMPC	: Nonlinear Model Predictive Control
MPC	: Model Predictive Control
SMC	: Sliding Mode Control
GA	: Genetic Algorithm
LMI	: Linear Matrix Inequality
MRAC	: Model Reference Adaptive Control
STR	: Self Tuning Regulator
MFAC	: Model Free Adaptive Control
SPR	: Strictly Positive Real
ISP	: Input Strictly Passive
OSP	: Output Strictly Passive
DoF	: Degree of Freedom

SYMBOLS

δ	: Steering angle
$\delta_{\mathbf{f}}$	: Front steering angle
$\delta_{\mathbf{r}}$	: Rear steering angle
$\mathbf{L}$	: Vehicle wheel base
$\mathbf{R}$	: Road radius
$\mathbf{l_f}$	: Distance from front axle to vehicle center of gravity
$\mathbf{l_r}$	: Distance from rear axle to vehicle center of gravity
β	: Vehicle slip angle
$\mathbf{m}$	: Vehicle mass
$\mathbf{V}$	: Vehicle velocity
$\mathbf{V_x}$	: Vehicle longitudinal velocity
ψ	: Yaw angle
$\dot{\psi}$	: Yaw rate
$\mathbf{a_f}$, $\ddot{y}$	: Vehicle lateral acceleration
$\mathbf{I_z}$	: Vehicle moment of inertia
$\mathbf{F_{yf}}$, $\mathbf{F_{yr}}$	: Front and rear tire lateral tire force
$\mathbf{C_{\alpha f}}$, $\mathbf{C_{\alpha r}}$	: Front and rear tire cornering stiffness
$\alpha_{\mathbf{f}}$, $\alpha_{\mathbf{r}}$	: Front and rear tire slip angle
$\theta_{\mathbf{f}}$, $\theta_{\mathbf{r}}$	: Front and rear tire velocity angle
$\mathbf{E}$	: Lateral path error
ε	: Augmented error
$\mathbf{e}$	: Model error
η	: Error augmentation
γ	: Adaptation gain
$\mathbf{u_c}$	: Reference signal
$\mathbf{u}$	: Control signal
$\mathbf{y}$	: System output
$\mathbf{y_m}$	: Reference model output
$\mathbf{b_0}$	: System high frequency gain
$\mathbf{b_m}$	: Reference model high frequency gain
φ	: Filtered signal vector
θ	: Adaptable controller parameters vector

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MODEL REFERENCE ADAPTIVE CONTROLLER DESIGN WITH AUGMENTED ERROR METHOD FOR LANE TRACKING

SUMMARY

At the beginning of the automobile industry history, automobiles were simple mechanical systems. Starting from Ferdinand Verbiest's steam-powered vehicle in 1672, called a toy, automobiles have evolved into complex machines. Essential inventions, such as the design of the first internal combustion-powered vehicle by François Isaac de Rivaz in 1808 and the gasoline-powered automobile by Karl Friedrich Benz in 1886, automobiles became available for everyday use. Especially the introduction of the Ford Model T by the Ford Motor Company in 1908, which was the first mass-produced commercial automobile, automobiles became common on roads. Thus, the safety and riding comfort specifications became significant factors for automobile producers in the automobile industry.

In the early 20th century, the pace of evolution increased dramatically thanks to the computerization and electronics in automobiles, which led to the introduction of Electronic Control Units (ECUs) and onboard computers, allowing for more precise control over engine performance and emissions and vehicle stability. Moreover, these computers and electrical components were used to design driver assistance systems introduced for driving comfort and safety during the 20th century. The initial features, such as anti-lock braking system (ABS) and cruise control (CC), were worked effectively. Later, these features were improved, advanced, and varied for different purposes under autonomous driving. Recently, the automotive industry has undergone a distinctive transformation towards autonomy, which governments and leading companies like Tesla and Google support.

Advanced driver assistance systems (ADAS) play a crucial role in autonomous driving. ADAS includes features implemented in vehicles to enhance safety and riding comfort by improving user awareness and controlling vehicle movement. Driver support systems can be categorized from various perspectives, including active, passive, safety, and comfort. Active driver assistance systems assume control of certain vehicle functions, while passive control systems warn the driver. According to the vehicle control point of view, ADAS is covered under two main categories: longitudinal and lateral motion control. Features like ACC and AEBS, for example, are associated with controlling the longitudinal motion of a vehicle, primarily focusing on speed and distance management. On the other hand, LKA/LCA and BSD features are within the domain of lateral motion control, mainly concerned with maintaining proper alignment within a lane and detecting vehicles in adjacent blind spots.

Within the scope of this study, an adaptive controller is designed for lane tracking of autonomous vehicles. The controller algorithm aims to center the vehicle on the

lane by calculating the required front steering angle. The controller's performance is simulated and evaluated, and finally, further tasks are determined.

Lane tracking control design is handled either with a model-free or a model-based approach in the literature. Model-free methods provide an alternative option when creating models becomes inaccurate and challenging. These control strategies typically rely on data-driven techniques such as supervised learning, reinforcement learning, and fuzzy logic control. Model-based approaches, such as MPC, SMC, LQR, and H_∞, on the other hand, use the mathematical representation of the vehicle's lateral motion, which plays a significant role in controller design. Simulation of the vehicle system using this representation provides a clear perspective for the evaluation of designed controller performance and calibration.

Vehicle models for lane tracking controller design are categorized within various aspects. While the mathematical representation of a vehicle, whether it is linear or non-linear, is in one category, its configuration type is the second one. Three vehicle model configuration types are available in the literature: geometric, kinematic, and dynamic vehicle models. Each of these configuration types has advantages and disadvantages that must be considered while designing a controller.

The bicycle dynamic vehicle model is the popular representation used in this thesis. Lateral path error is derived as the function of vehicle lateral motion state variables (lateral, longitudinal velocity, and yaw angle of the vehicle) on the ego lane, which is the output of the control system according to the bicycle model. Then, this derived model is used to determine the adaptive control law to achieve the desired tracking performance.

The adaptive control method is one of the most promising methods to create reliable solutions to the difficulties faced by autonomous vehicles in lane tracking. Although different types of adaptive control design methods are available in the literature, model reference adaptive control (MRAC) is the most suitable in terms of clarity and low computational burden, as well as real-time application. In this thesis, it is clearly seen that the derived vehicle model is a perfect fit for the adaptation of feedforward gain with the output feedback based on the passivity. However, due to that the derived model's transfer function, based on the parameters of an autonomous large-size vehicle, is not SPR makes the model unsuitable for model reference adaptive controller design. As a solution, the augmented error method is used to enable the application of the passivity approach to determine the adjustment rules of controller parameters. Thus, the controller design, which ensures the input-output stability with MRAC, is derived based on the augmented error method.

As a result, it is seen that the model reference adaptive controller system with augmented error method showed a perfect tracking performance according to the simulations on Simulink. Considering similar studies, the control signal obtained in the simulation showed that the model is applicable for real-time application.

ŞERİT TAKİBİ KONTROLÜ İÇİN ARTIRILMIŞ HATA YÖNTEMİ İLE MODEL REFERANS UYARLANABİLİR KONTROLÖR TASARIMI

ÖZET

Otomobil endüstrisi tarihi incelendiğinde, başlarda otomobiller basit mekanik yapılardan oluşan sistemlerdi. 1672'de Ferdinand Verbiest tarafından tasarlanan ilk otomobil, buharla çalışan ve boyutlarından ötürü oyuncak olarak nitelendirilirken , büyük bir değişim göstererek günümüzde oldukça karmaşık akıllı makineler haline geldi. Bu değişim sürecinde, 1808'de François Isaac de Rivaz'ın ilk içten yanmalı motora sahip aracını tasarlaması ve Karl Friedrich Benz'in 1886'da benzinle çalışan otomobilini üretmesi gibi önemli buluşlarla otomobiller günlük kullanım için uygun hale geldi. Özellikle, Ford Motor Company tarafından 1908'de tanıtılan ilk seri üretim aracı Ford Model T'nin ardından, otomobiller günlük hayatta yolların önemli bir parası olmaya başladı. Bu durum, otomobil üreticileri için sürüş güvenliği ve konforu gibi özellikleri önemli gereksinimler haline getirdi.

20. yüzyılın başlarında, otomobillerdeki bilgisayarlaşma ve ve yarı iletken kullanımı sayesinde evrimin hızı büyük bir ivme ile arttı ve bu teknoloji, Elektronik Kontrol Üniteleri (ECU) ve yol bilgisayarları gibi, emisyonlar ve sürüş stabilitesi üzerinde daha hassas kontrol sağlayan otomobil bileşenlerinin tanıtılmasına yol açtı. Daha sonraları, bu bilgisayarlar ve sensörler, sürüş konforu ve güvenliği için tanıtılan ilk sürücü destek sistemlerini tasarlamak için kullanıldı. Başlangıçta, kilitleme önleyici fren sistemi ve hız sabitleme gibi sürücü destek sistemler etkili bir şekilde çalıştı. Daha sonra, bu sistemler otonom sürüş altında farklı amaçlar için ileri düzeyde geliştirildi ve çeşitlendirildi. Son zamanlarda otomotiv endüstrisi, hükümetler ve Tesla ve Google gibi önde gelen teknoloji şirketlerini önemli yatırımlarını alarak otonomiye doğru belirgin bir dönüşüm geçişine başladı.

Gelişmiş sürücü destek sistemleri (ADAS), otonom sürüşün önemli bir parçasıdır. ADAS, kullanıcı farkındalığını artırmak ve aracın hareketini kontrol etmek suretiyle sürüş güvenliği ve sürüş konforunu artırmak amacıyla araçlara uygulanan sistemlerin genel adıdır. Bu gelişmiş sürücü destek sistemleri, aktif, pasif, güvenlik ve konfor gibi çeşitli perspektiflerden kategorilere ayrılabilir. Aktif sürücü yardım sistemleri belirli araç fonksiyonlarını devralırken, pasif kontrol sistemleri sürücüyü uyarmak görevini üstlenir. Öte yandan, araç kontrolü bakış açısıyla ADAS, iki ana kategori altında ele alınır: boylamsal ve yanal hareket kontrolü. Örneğin, ACC ve AEBS gibi özellikler, bir aracın uzunlamasına hareketini kontrol etmekle ilişkilendirilir ve genellikle hız ve mesafe yönetimine odaklanır. Öte yandan, LKA/LCA ve BSD gibi özellikler, yanal hareket kontrolünün alanına girer ve genellikle bir şeritte uygun hizalamayı koruma ve yanıt alamayan kör noktalardaki araçları algılama konusunda önemli rol alırlar.

Başlıca gelişmiş sürücü destek sistemleri aşağıdaki gibi örneklendirilebilir.

Uyarlamalı hız sabitleyici (ACC): Aracın öndeki araçla arasındaki güvenli takip mesafesini korumak için aracın hızını uyarlarken aynı zamanda sürücü tarafından belirlenen hızı korur.

Gelişmiş acil durum frenleme sistemi (AEBS): Gelişmiş bir acil durum frenleme sistemi, diğer araçlarla, yayalarla veya engellerle arkadan çarpışmaları önlemek veya çarpışma etkisini azaltmak için acil durumlarda frenleri otomatik olarak uygulayarak araç güvenliğini artırmak üzere tasarlanmıştır.

Kör nokta asistanı (BSD): Kör nokta asistanı, sürücülerin görüş imkanının bulunmadığı aracın sağ ve sol arka çaprazında bulunan dinamik objelerle çarpışmasını önleme sistemi oluşturmak için ultrasonik sensörler kullanılır. Bu sistemler sadece sürücüyü uyarıp farkındalık oluştururlar, herhangi bir eyleyici aktif olarak konrtol etmedikleri için pasif sürücü destek sistemi olarak nitelendirilirler.

Şerit takip asistanı (LKA): Aracı takip edilen şerit içerisinde tutmak için gerekli direksiyon açısının hesaplayıp araca uygulayan ve ayrıca gerekli durumlarda sürücüyü uyaran bir gelişmiş sürücü destek sistemidir. Otonom sürüş fonksiyonlarında önemli bir rol oynar.

Bu çalışma kapsamında, otonom araçların şerit takibi için bir uyarlamalı kontrolör tasarlanmıştır. Kontrolör algoritması, otonom aracın takip edilen şerit üzerinde merkezlenmesi için gereken ön direksiyon açısını hesaplamak amacıyla tasarlanmıştır. Bu tasarım yapılırken belirsiz ve değiken sistem ve çevre dinamikleri göz önünde bulundurulmuştur. Arzu edilen sürüş konforunu sağlayan ideal bir referans model performansı belirlenerek kapalı çevrim uyarlamalı sistem yapısı bu modelin davranışını sergliyecek şekilde kontrolör parametrelerini güncelleyecek uyarlama kuralı belirlenmiştir. Kontrolör Simulink ortamında test edilmiş, performansı değerlendirilmiş ve son olarak gelecek çalışmalara değinilmiştir.

Literatürde, şerit takip kontrolör tasarımı, araç modelininin temsilinin kullanımına göre model tabanlı olmayan veya model tabanlı bir yaklaşım kullanılacak şekilde iki farklı bakış açısıyla ele alınır. Model tabanlı olmayan yaklaşımlar, modellemenin karmaşık veya doğru olmadığı durumlarda alternatif bir çözüm sunar. Bu kontrol stratejileri genellikle gözetimli öğrenme, pekiştirmeli öğrenme ve bulanık mantık kontrolü gibi veri tabanlı tekniklere dayanır. Öte yandan, MPC, SMC, LQR ve H_∞ gibi model tabanlı yaklaşımlarda, kontrolör tasarımı için araç yanal hareketinin matematiksel temsili kullanılır. Araç modeli matematiksel temsilinin simülasyonu, tasarlanmış kontrolör performansını değerlendirmek veya kalibrasyon etmek için net bir görünüm sağlar.

Şerit takip kontrolör tasarımı için kullanılan araç modelleri çeşitli açılardan sınıflandırılır. Araç modelin matematiksel ifadesinin doğrusal olup olamaması bir sınıf iken, yanal hareketliliğin ifade edilmesindeki konfigürasyon türü ise bir diğer sınıf olarak nitelendirlir. Literatürde üç farklı konfigürasyon türü bulunmaktadır: geometrik, kinematik ve dinamik araç modelleri. Her bir gösterim türünün kontrolör tasarımda kullanımı öncesinde dikkate alınması gereken avantajları ve dezavantajları vardır.

Bu tezde literatürde yaygın olarak kullanılan bisiklet araç dinamik modeli kullanılmıştır. Otonom aracın takip ettiği şerit merkezine göre yanal uzaklık hatası, araç yanal hareketliliğini açıklayan durum değişkenlerinin (yanal ve boylamsal hız ve yönelim açısı) bir fonksiyonu olarak türetilir ve bu değer, kontrol sisteminin çıkışı olarak kullanılmıştır. Ardından, türetilen model, istenen izleme performansını elde etmek için uyarlamalı kontrol yasasını belirlemek için kullanılmıştır.

Uyarlamalı kontrol yöntemi, otonom araçların şerit takibinde karşılaştığı zorluklara yönelik güvenilir çözümler oluşturmak için en umut verici yöntemlerden biridir. Literatürde farklı türde uyarlamalı kontrol tasarım yöntemleri bulunsa da, model referanslı uyarlamalı kontrol (MRAC), açıklık ve düşük hesaplama yükü, ayrıca gerçek zamanlı uygulama açısından en uygun olanıdır. Bu tezde, yukarıdan belirtilen araç modeli, çıkış geri beslemeline göre ileri yol kazancının pasifliğe dayalı karalı bir kapalı çevrimk sistemi elde edecek bir biçimde ayarlanması şerit takibi kontrolü çözüm yöntemi için mükemmel bir uyum sağladığı açıkça görülmektedir. Ancak, ağır vasıta otonom bir aracın parametreleri kullanılarak elde edilen transfer fonksiyonu incelendiğinde, bu teoremin kullanılabilmesi için gerekli koşul olan pozitif reellik sağlanmadığı açıktır. Bu nedenle, çıkış geri beslemelisine göre kontrol parametrelerinin uyarlama kuralını elde etmek için artırılmış hata yöntemi kullanılmıştır. Bu şekilde giriş çıkış anlamında pasifliği grantileyecek MRAC kontrolör tasarımı artırılmış hata yöntemine dayalı olarak türetilmiştir.

Sonuç olarak, artılımış hata yöntemine göre tasarlanmış kontrol sisteminin performansı Simulink ortamında test edilmiş ve arzu edilen şerit pozisyonuna göre mükemmel bir izleme performansı gösterdiği görülmüştür. Benzer çalışmaları dikkate alındığında, simülasyonda elde edilen kontrol sinyali, modelin gerçek zamanlı uygulanma için uygun olduğunu göstermiştir.

1. INTRODUCTION

1.1 Introduction to Automotive

The automotive industry has experienced significant change and constant innovations throughout history. Automobiles have evolved from simple mechanical systems powered by steams to complex machines equipped with driver assistance systems using several energy sources, such as electric, hydrogen, etc., over the past century. More importantly, autonomous driving capabilities have recently placed the center of his sophisticated evolution.

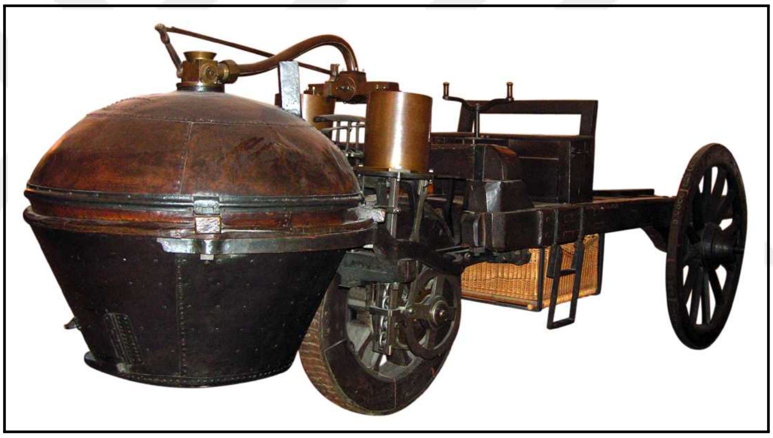

Figure 1.1 : Cugnot's Steam Wagon [1].

The development of automobiles can be traced back to 1769 when Nicolas-Joseph Cugnot constructed the first steam-powered automobile capable of carrying humans. His work was inspired by the invention of the first steam-powered vehicle designed by Ferdinand Verbiest in 1672. On the other hand, François Isaac de Rivaz designed and constructed the first internal combustion-powered vehicle in 1808, while Karl Friedrich Benz patented the marketable gasoline-powered automobile designed for everyday use in 1886. In 1908, the Ford Motor Company marked a notable breakthrough in the automotive manufacturing industry with the launch of the Ford Model T. This vehicle became the first automobile to be mass-produced using a moving assembly line, revolutionizing the automotive industry [1]. During the mid-20th century, safety and comfort features became prior automobile specifications, and the automotive industry experienced notable progress on these features.

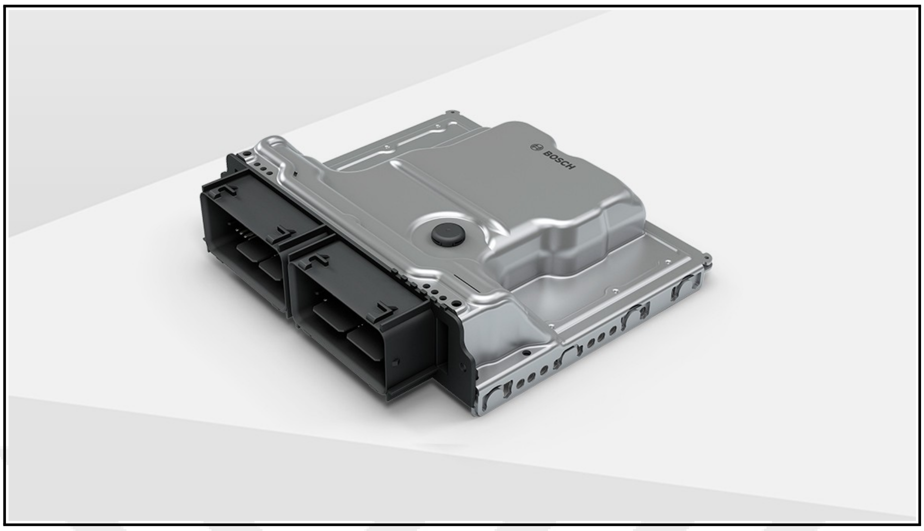

Figure 1.2 : A Car Electronic Control - Unit Bosch Mobility Solutions.

The late 20th century witnessed a shift towards computerization and electronics in automobiles. The introduction of Electronic Control Units (ECUs) and onboard computers allowed for more precise control over engine performance and emissions. Moreover, electronic systems began to play a role in vehicle stability and traction control. Furthermore, computers and electronic components such as sensor suits led the designers to introduce driver assistance systems, thanks to the rapid technological improvements in semiconductors. Initially, features like the anti-lock braking system developed by Bosch and Speedostat (Cruise Control) used by General Motors became available in premium vehicles. These features marked the beginning of a paradigm shift towards autonomy.

Today, the automotive industry shows a noticeable transformation with the development of autonomous vehicles. Governments support this eco-friendly evolution with their start-up, such as TOGG in Turkey and Vinfast in Vietnam. Companies like Tesla, Google's Waymo, and traditional automakers have invested heavily in research and development to make self-driving cars real use on roads. These vehicles are equipped with sensors, cameras, advanced artificial intelligence (AI) algorithms, and highly defined maps that enable them to navigate roads autonomously.

1.1.1 Autonomous Driving and Driver Assistance Systems

The development of self-driving vehicles, also called autonomous driving, is commonly seen as incorporating an additional layer of intelligence onto traditional vehicle platforms [2]. The concept of autonomy level for automobiles was classified by SAE International (Society of Automotive Engineers) in 2014, and according to this classification, there are six levels of driving automation that indicate a vehicle's capability to operate without human intervention. This classification, established by SAE, typically serves as the industry's reference point for assessing vehicle automation [3]. The explanation of these six levels is as follows.

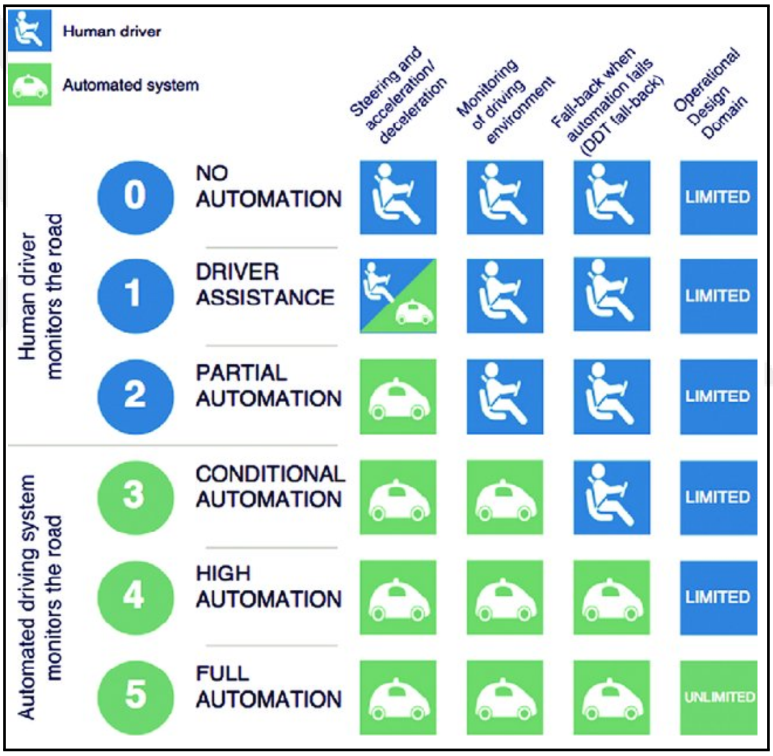

Figure 1.3 : Automation levels according to SAE International [2].

Level 0: (No Automation) At this stage, the human driver has complete control of the vehicle. There may be basic driver assistance features like warnings or brief interventions, but they don't take over any aspect of driving.

Level 1: (Driver Assistance) "Hands-on" stage where vehicles come equipped with systems that can aid the driver with acceleration/deceleration. An example is adaptive cruise control.

Level 2: (Partial Automation) "Hands-off" level where the vehicle can manage both steering and acceleration/deceleration concurrently, but the driver must stay actively involved and monitor the surroundings. Examples include advanced driver-assistance systems (ADAS) like Tesla's Autopilot [4].

Level 3: (Conditional Automation) "Eyes-off" level, where vehicles are capable of handling most driving tasks under specific conditions, such as highway driving. However, they still require a human driver as a backup. The driver can disengage from active control but should be ready to take over if prompted by the system.

Level 4: (High Automation) "Minds-off" level where vehicles can operate autonomously in particular conditions and environments without human intervention. Nevertheless, they may have limitations, and the driver might need to step in during certain situations.

Level 5: (Full Automation) "Steering-wheel optional" level where vehicles achieve full autonomy, capable of handling all driving tasks under any conditions without any need for human intervention. There is no requirement for a steering wheel or pedals at this vehicle level.

ADAS plays a crucial role in autonomous driving. It is a set of safety and convenience features in vehicles that use advanced technology to assist drivers in various aspects of driving, and they are designed to enhance vehicle safety, improve driver awareness, and provide additional convenience. These systems have the capability to offer crucial information about the environment, and they can assess the driver's level of fatigue and distraction, issuing precautionary alerts when necessary, and provide feedback on driving performance along with suggestions for improvement. Moreover, they can assume control in assessing potential threats, carry out straightforward tasks such as cruise control, and execute more complex maneuvers like overtaking and parking [5]. Studies have proven that ADAS's systems decrease road fatalities by minimizing human errors [6].

According to the vehicle control point of view, ADAS can be divided into two categories, namely longitudinal and lateral motion control. The popular ADAS features are adaptive cruise control (ACC), advanced emergency braking system

(AEBS), lane-keeping assistance (LKA), and blind spot detection (BSD). While adaptive cruise control and advanced emergency braking systems are related to longitudinal motion control, lane-keeping assistance and blind spot detection are in the scope of lateral motion control of a vehicle.

ACC maintains a set speed while also adjusting the vehicle's speed to maintain a safe following distance from the vehicle ahead. It enables the vehicle to adapt its velocity according to the detected preceding vehicle based on the pre-selected gap by the driver.

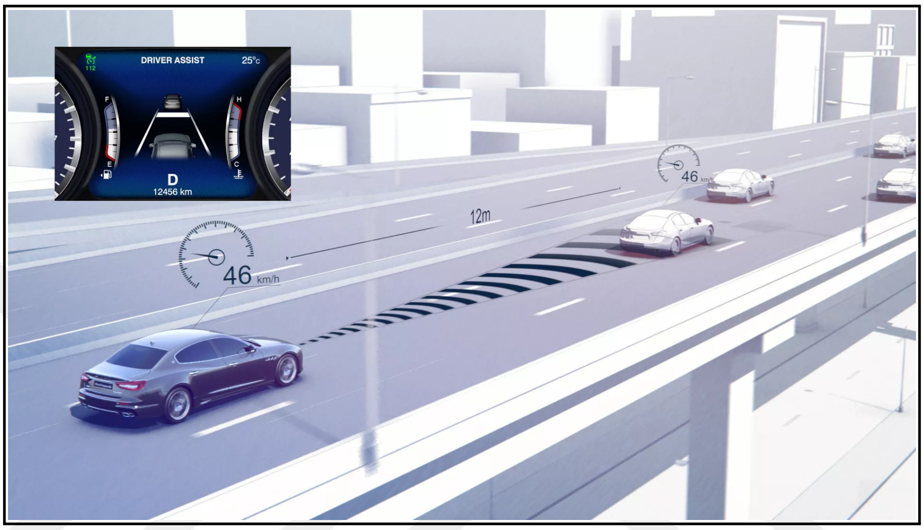

Figure 1.4 : Adaptive cruise control working scenario [7].

AEBS is designed to enhance vehicle safety by automatically applying the brakes in emergency situations to prevent or mitigate rear-end collisions with other vehicles, pedestrians, or obstacles. This is accomplished by issuing collision warnings to the driver, prompting them to initiate braking, considering the time to collision (TTC) value. If the driver does not respond to the warning or does not take appropriate action to avoid the collision, AEBS can initiate automatic emergency braking. It can apply the brakes to reduce the vehicle's speed or, in some cases, bring it to a complete stop to prevent or mitigate the impact.

LKA automatically makes steering adjustments when it determines that an adjustment is necessary to align the vehicle with the reference points within the lane. Lane centering, on the other hand, is an advanced driver assistance system where the steering angle is calculated and applied to the system to make the vehicle centered in the lane. It is mainly designed for vehicles with a higher level of autonomy, level 3 and above.

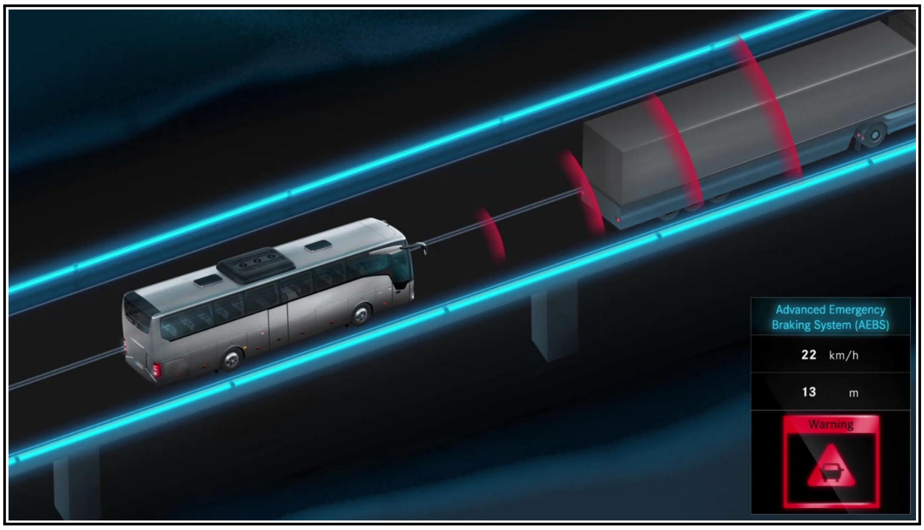

Figure 1.5 : Advanced emergency braking system working scenario [8].

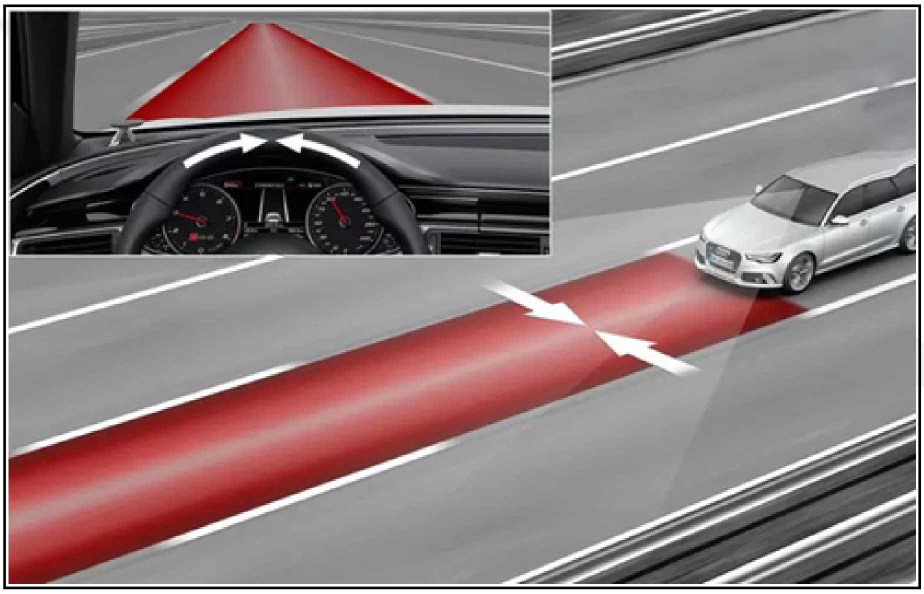

Figure 1.6 : Lane centering assistance system working scenario [9].

In BSD systems, ultrasonic sensors are used to create a prevention system for rear collisions. In BSW (Blind Spot Warning) systems that rely on visual information, algorithms such as optical-flow detection efficiently identify approaching vehicles. Still, their computational demands may restrict their capabilities and usage. When a vehicle or obstacle is detected in the blind spots of the ego vehicle, the system generates haptic, acoustic, or visual warnings that are perceptible to the driver.

Since the significant subjects of ADAS are mostly related to controlling the vehicle or warning the driver to keep the vehicle under control, the main topic of this thesis is a controller design for vehicle lane tracking in which the purpose is to make the vehicle centered on the tracked lane.

1.2 Purpose

In the literature, there exist many types of lateral control systems for lane tracking, and research in this field remains an ongoing and actively explored area of study. Different approaches have been used to provide the desired tracking performance. Nevertheless, ensuring stability for uncertain dynamics and varying parameters by considering ride comfort and safety is a challenging task to achieve. In fact, a sustainable lane tracking controller design to implement in real-time application requires a clear and effective design procedure. In this thesis, the aim is to design an adaptive lane tracking controller for an autonomous vehicle and test the controller performance on a large-size autonomous bus.

1.3 Literature Review

The future of the automotive industry is being shaped by state-of-the-art technologies such as connected services, automated driving, shared mobility, and electrified powertrains. These are the optimal solutions for the future of mobility.

Autonomous driving, especially, has the potential to revolutionize mobility and transportation by significantly reducing road accidents, alleviating traffic congestion, and mitigating air pollution [10] - [11]. Autonomous vehicles are complex systems that are comprised of multiple modules designed to execute functions like planning, perception, decision-making, and control. Considering these modules are the functions of advanced driving assistance systems, it can be said that autonomous driving capability is achieved through the collaboration of some of ADAS functions.

Control plays a fundamental role in realizing autonomous driving as the center of ADAS, and it is primarily divided into two components: longitudinal control, responsible for maintaining speed and tracking, and lateral control, which guarantees precise steering [12]. Lateral control is particularly critical in path-tracking, similarly called lane tracking or road following, applications by providing a precise steering angle.

Lane tracking control design is also divided into two approaches based on the necessity of a vehicle model: model-based and model-free. Lateral tracking control systems in model-based approaches require a representation of the vehicle's lateral motion. These representations are typically geometric, kinematic, or dynamic models chosen based on the specific controller type employed [12]. The strengths and weaknesses are the other important factors for the decision of the model type for controller design [13]. Controllers with geometric vehicle models are designed using the geometric connection between the path and the vehicle [14]. The most known design methods are Pure-Pursuit, Stanley, and vector pursuit, while the combination of these methods was also proposed as a solution for path tracking control by Cibaloglu et al. [14]. Controllers designed based on kinematic vehicle model, on the other hand, use the vehicle's position, speed, and acceleration to describe the vehicle's lateral motion to design lane tracking controller, which Hoffmann et al. provided their type of improved Stanley algorithm based on these variables [15]. Alaca et al. proposed a lateral controller for autonomous vehicles using a linear parameter variable kinematic vehicle model based on Linear Quadratic Regulator (LQR) control method [16]. They tested the controller performance with real scenario test cases.

The nonlinearity of a vehicle's lateral motion representation via dynamic model, however, is a fact unless vehicle speed is constant and the tire model is assumed as linear [17]. Although nonlinear approaches have been studied in the literature, such as the Nonlinear Model Predictive Controller (NMPC) by Alcala et al. [18], the computational load is considered a problem for real-time applications. Therefore, the linearized vehicle dynamic model is an option for low computational load, but the rapid changes in vehicle speed must be considered. Yao et al. designed an MPC with longitudinal speed compensation to address the variability in speed changes, thus addressing this assumption [19]. Sliding Mode Controller (SMC) is another method that effectively provides a solution for lane tracking. Wu et al., for example, proposed an SMC for a linearized vehicle model lane tracking problem, and the model in loop simulations showed good performance with overcoming uncertainties and rejecting disturbances [20].

Model uncertainties and disturbances, on the other hand, are rigorously handled with robust control design methods, such as H_∞ is one of them, to provide robustness. Hu et al., for example, designed a H_∞ robust controller that the vehicle lateral velocity is not required. They tested this controller for a path following control and also used Genetic Algorithm (GA) and Linear Matrix Inequality (LMI) for optimization of feedback gain [21].

Adaptation according to the changing dynamics and parameter uncertainties is an expected behavior from a controller for lane tracking. Model Reference Adaptive Control (MRAC) is one effective model-based controller method that provides promising results. Moreover, the transparency of MRAC is sufficient for a complex lateral control problem. Bryne and Abdallah designed an MRAC for vehicle road following, and simulations showed perfect tracking results. However, the oscillatory steering angle signal within a large range was unsuitable for real-time application [22].

To deal with the model uncertainties and varying dynamics, model-free approaches, such as machine learning and fuzzy logic controllers, have been studied recently as another option [23] - [24]. Nevertheless, the lack of transparency and the requirement for expert knowledge of these model-free methods are considered a handicap for real-time applications. Classical control techniques are limited due to the high-order problem formulation of lane tracking control with real-time application; even these methods are improved for lane tracking with optimization methods in the literature, such as adaptive PID with supervised neural networks [25].

1.4 Hypothesis

The lateral path error is the distance between the nearest point to the lane center and the point of the vehicle's center of gravity. The lateral path error can be defined as a function of the vehicle's lateral and longitudinal speed and the orientation angle [22]. Thus, the tracking error can be considered as the output of the vehicle system, whereas the input is the steering angle. Highly precise systems have been developed to detect the lateral path error in the scope of lane detection systems in the automotive industry, which makes this error to be defined as a perfect system output to feed into the control system. In the concept of MRAC with output feedback that ensures passivity of output error and the closed loop system (input-output stability) [26] requires the system transfer function to be strictly positive real. Nevertheless, the transfer function of a vehicle describes a fourth-order system with relative degree two because of which the strictly positive realness cannot be met to use the theorem directly. The error augmentation method can meet the condition of strictly positive realness, and the stable closed loop systems in terms of input-output stability is obtained. In this thesis, a model reference adaptive controller for lane tracking of a vehicle is designed with augmented error method for the first time in the literature.

The thesis work is divided into five parts. Chapter 2 provides general information about the vehicle modeling for lateral control in the literature and the model used in the thesis where the transfer function has the output as lateral path error and the input steering angle. Chapter 3 presents the fundamental information about adaptive control and the model reference adaptive controller design with augmented error for lane tracking of an autonomous vehicle. In Chapter 4, the simulation results on Simulink are represented. Finally, the recommendations and future works are discussed in Chapter 5.

2. VEHICLE MODEL FOR LANE TRACKING

2.1 Introduction to Vehicle Model for Lane Tracking

Vehicle model holds significant importance when it comes to designing a lane tracking controller, influencing it in two essential aspects. One is the simulation of the vehicle system during which the controller properties are examined. If the controller parameters are not adaptable, calibration of controller parameters is performed during the simulation stage to yield the desired performance. The second aspect is that the control laws for lane tracking control are mostly derived from the mathematical depiction of the vehicle system [27]. On the other hand, while the nonlinear vehicle model is used in some studies [28], the linear approach is seen as a popular approach when the literature is observed, so this is considered another categorization. Thus, several types of vehicle models can be seen in the literature according to the aim of the study.

The typical approach in vehicle modeling involves treating the vehicle body as a solid, rigid structure with a centralized mass situated at the center of gravity. When it comes to controlling the vehicle's path, this is similar to handling control, which focuses on how the vehicle behaves in the longitudinal plane (forward and backward motion). This approach is often referred to as the handling model, and it primarily considers the vehicle's lateral (side-to-side) and yaw (rotation around the vertical axis) movements. In this handling model, vertical movements, which pertain to the up-and-down motion of the vehicle, are usually assumed to be constant due to the assumption of a flat road surface. This means that factors like ride comfort and vertical forces from the suspension are considered negligible [27]. Figure 2.1 illustrates the conventional depiction of a vehicle model commonly employed in studies related to vehicle handling. This representation is called the full-vehicle model.

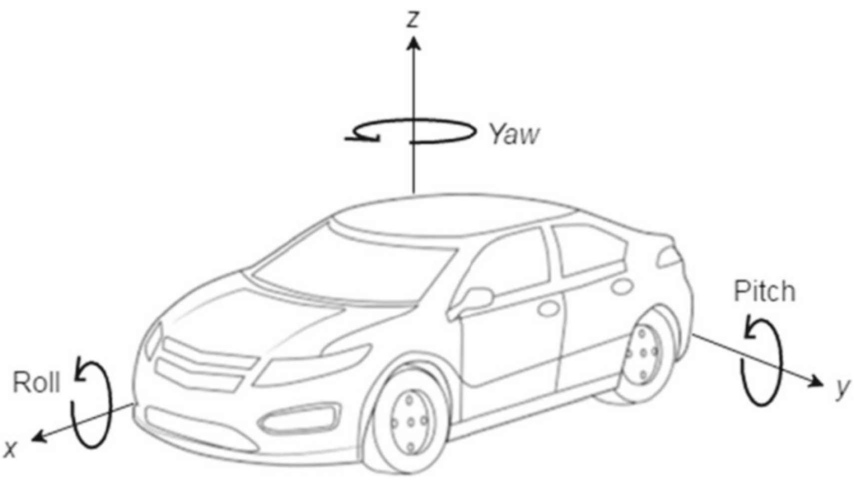

Figure 2.1 : Perspective of full vehicle used in vehicle modelling terminology [27].

The bicycle vehicle model, on the other hand, is the most popular approach based on the single-track concept of a vehicle, where the vehicle is simplified to a two-tire configuration at both the rear and front. This simplification assumes that the behavior of the left and right tires is the same. Consequently, the left and right tires are combined and represented as a single tire at the front and another at the rear.

In summary, vehicle models for lane tracking controller design are categorized within various aspects. While the linearity of the mathematical representation is one aspect, the configuration type is the second one. Generally, there are three configuration types: geometric, kinematic, and dynamic models, and the simplification of a full vehicle as the single-track bicycle is a popular simplification for all these configurations. This chapter gives information about these vehicle models used in lane tracking control in the literature by explaining each of their advantages and disadvantages, and the model used in this thesis is described in detail.

2.2 Geometric Vehicle Model

The vehicle geometric model provides a fundamental framework for understanding and predicting the vehicle's trajectory within its lane, contributing to the development of autonomous driving systems. The geometric model exclusively encompasses the vehicle's geometric dimensions without taking into account the kinematic aspects, such as acceleration and its derivatives, as well as dynamic factors (including forces, inertia,

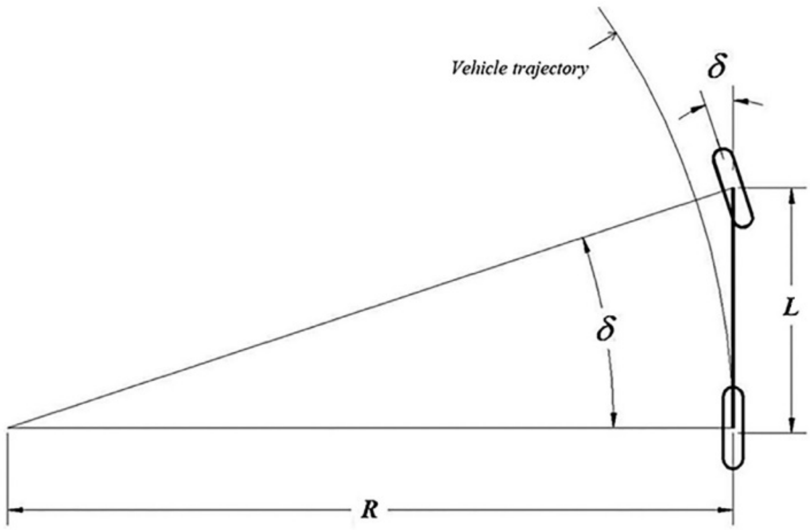

Figure 2.2 : Geometric bicycle model.

and energy properties) [27]. Geometric model methodologies rely on utilizing the geometric bicycle model, an abstraction derived from Ackerman-steered vehicles [29]. In this model, the fundamental principle is that the line perpendicular to each vehicle wheel must intersect at vehicle's central point when the vehicle is cornering. Figure 2.2 represents the Ackerman-steered geometric bicycle model, where δ represents the angle of the front wheel in radians, R and L are the curvature radius of the trajectory and the wheelbase of the vehicle in meters, respectively.

Utilizing this model for understanding the vehicle's motion, the underlying concept of a fundamental geometric lateral control approach is to determine the angle δ to be applied to the vehicle's front wheels at a specific moment. This determination relies on the vehicle's geometric attributes, notably its wheelbase L, the intended vehicle trajectory, and the present state of the vehicle, which typically encompasses parameters such as position, orientation, velocity, acceleration, and, for pure rolling stability, the slipping forces. The analysis focuses on the rear wheel movement, as the positions of the other vehicle components can be deduced from the rear wheel's position. This relation is represented in Equation 2.1 below.

$$\delta = \tan^{-1}\left(\frac{L}{R}\right) \tag{2.1}$$

Popular controller design methods, such as Pure-Pursuit [30] and Stanley [15], are based on geometric vehicle models in the literature. Conversely, Vector-Pursuit is another geometric control approach that relies on the principles of screw theory [31]. While these approaches transform the path-tracking issue into a straightforward geometric problem, they are problematic at high speeds.

2.3 Kinematic Vehicle Model

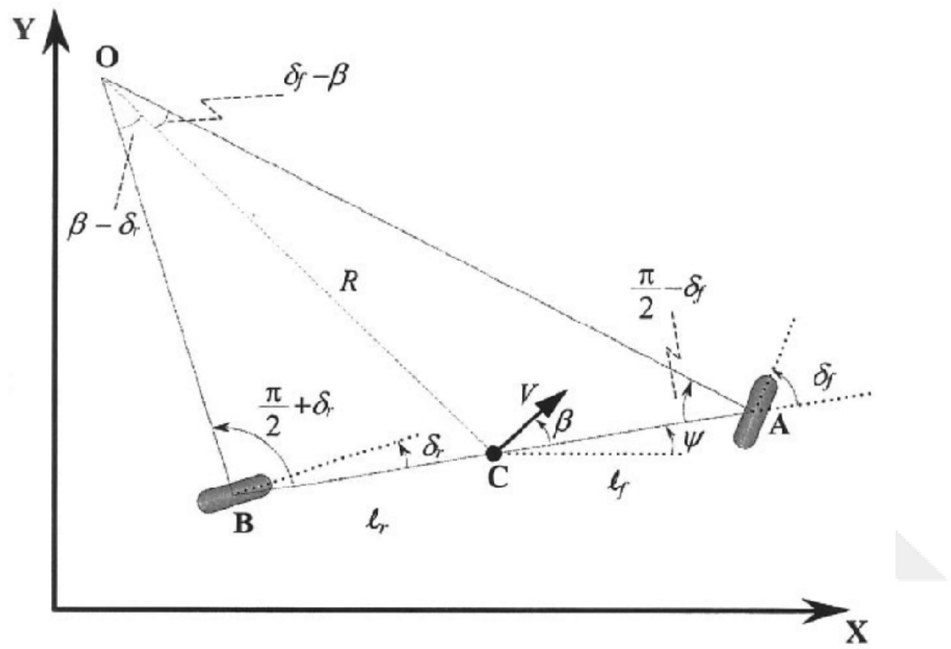

Figure 2.3 : Single track kinematic bicycle model [17].

In the realm of mechanics, kinematics is defined as the examination of a body's motion apart from its internal forces, inertia, and energy. As a result, the kinematic vehicle model describes a vehicle's motion based on its position and derivatives, omitting the internal dynamics of the vehicle. This approach solely accounts for the vehicle's geometry and kinematic properties.

Numerous approaches have been employed previously to construct and elucidate vehicle kinematics [13]. Figure 2.3 represents a basic single-track kinematic bicycle model, where both left and right front wheels are illustrated by one single wheel at point A. Similarly, both rear wheels are combined as a single point at B. δ_f and δ_r are the front and rear steering angles of the front and rear wheels, respectively. It is assumed that this model describes that both rear and front wheels can steer the vehicle. If the vehicle configuration is based only on the front wheel, then the rear steering angle is considered zero [17]. While C represents the vehicle's center of gravity ($c.g.$), l_f and l_r describe the distances from A and B points to $c.g.$ of the vehicle, respectively. L is the wheelbase of the vehicle, which is derived by summation of l_f and l_r. The point O signifies the vehicle instantaneous rolling center, which is the junction of lines AO and BO, both of which are extended perpendicularly from the orientations of the two rolling wheels. The radius of the vehicle's trajectory, denoted as R, is established by the extent of line OC. Consequently, the orientation of the velocity at the center of gravity, relative to the vehicle's longitudinal axis, is termed the vehicle's slip angle, denoted as β.

The equations of the vehicle's motion are derived under some assumptions. The central assumption is that slip angles at both rear and front wheels are zero, which is logical when the vehicle operates at low speeds. In such scenarios, the lateral forces produced by the tires are small, where the overall lateral force generated by both tires is $\frac{mV^2}{R}$ [17]. The second assumption is that the vehicle has planar motion, and to describe the vehicle's motion adequately, three coordinates are essential: X, Y, and ψ. In this context, (X, Y) represent the inertial coordinates corresponding to the center of gravity's position, while ψ signifies the description of the vehicle's orientation, commonly known as yaw angle. Lastly, assuming that alterations in the vehicle's path radius occur gradually due to its low speed, the rate of change of the vehicle's orientation ($\dot{\psi}$) must equivalently match the vehicle's angular velocity.

Considering these assumptions, the overall equations of motion based on the kinematic vehicle model are derived as in Equations 2.2, 2.3, and 2.4 [17].

$$\dot{X} = V\cos(\psi + \beta) \tag{2.2}$$

$$\dot{Y} = V\sin(\psi + \beta) \tag{2.3}$$

$$\dot{\psi} = \frac{V\cos(\beta)}{l_f + l_r}\Big(\tan(\delta_f) - \tan(\delta_r)\Big) \tag{2.4}$$

Although the kinematic model permits the design of a controller that is globally stable under the assumption that the vehicle has negligible inertia, this assumption is effective for low-speed driving as in geometric vehicle model-based controllers [15].

2.4 Dynamic Vehicle Model

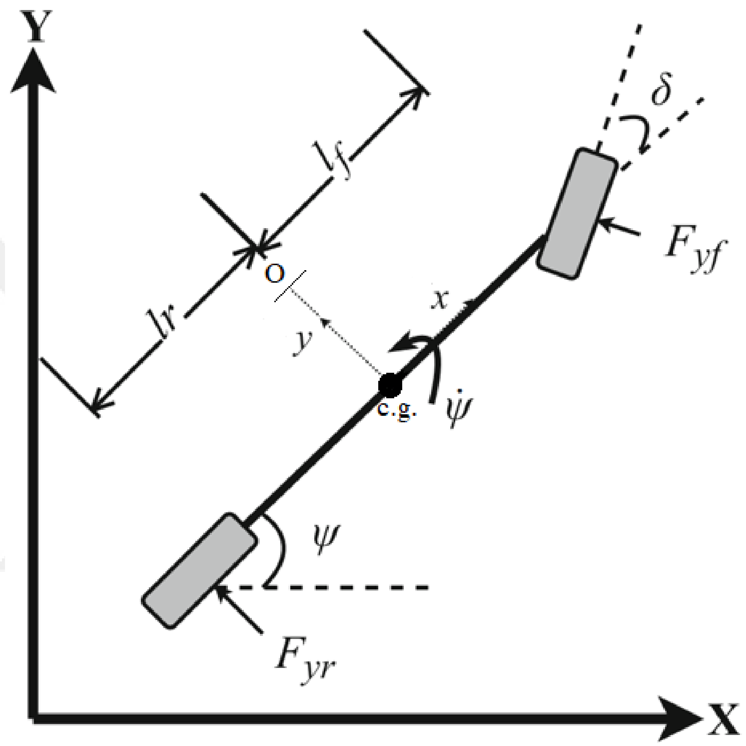

Figure 2.4 : Bicycle vehicle dynamic model.

Dynamic vehicle model for lateral control is a fundamental framework used to comprehend and regulate its lateral motion, particularly during maneuvers like lane changes and turns. Differing from the kinematic model, the dynamic vehicle model characterizes the motion of the vehicle by considering their acceleration and its integrals (velocity and position). This approach incorporates internal forces, energy, and momentum within the system to provide a comprehensive understanding [27].

A bicycle dynamic model of the vehicle, characterized by two degrees of freedom, is taken into account, as illustrated in Figure 2.4, to derive the equation of the vehicle's motion. These two degrees of freedom encompass the vehicle's lateral position y and its yaw angle ψ. The lateral position is gauged along the vehicle's lateral axis, reaching point O, which serves as the vehicle's rotation center. While the longitudinal velocity of the vehicle at its center of gravity is indicated by V_y, the yaw angle ψ is measured relative to the global X axis [17].

A prevalent technique employed to derive the mathematical model for vehicle handling dynamics involves utilizing the principles of Newtonian equations of motion. Two main equations are derived for describing each movement of the degree of freedom. The first one is the lateral force balance equation due to the motion along the y axis, which is derived as in Equation 2.5 under the assumption that the road bank angle is zero.

$$ma_y = F_{yf} + F_{yr} \tag{2.5}$$

where the lateral inertial acceleration of the vehicle at the inertial center of gravity (c.g.) is directed along the y-axis and is represented by a_y, additionally, F_{yf} and F_{yr} denote the lateral tire forces exerted by the front and rear wheels, respectively, while m is the mass of the vehicle. The acceleration $\ddot{y}$ along the y axis and centripetal acceleration $V_x\dot{\psi}$ are the two terms contributing to a_y. Therefore, the lateral acceleration a_y is the summation of these two terms. Substituting these terms into Equation 2.5, the equation describing the lateral translational motion of the vehicle is derived as follows.

$$m(\ddot{y} + V_x\dot{\psi}) = F_{yf} + F_{yr} \tag{2.6}$$

Applying a moment balance around the z-axis results in the equation governing the yaw dynamics, which can be expressed as in Equation 2.7.

$$I_z\ddot{\psi} = l_f F_{yf} - l_r F_{yr} \tag{2.7}$$

Here the I_z denotes the moment of inertia of the vehicle about the Z axis. The tire forces, F_{yf} and F_{yr},on the other hand, are remaining part of the governing equations to be clarified. Empirical findings indicate that the lateral tire force of a tire is linearly related to the slip angle, particularly for small slip angles [17]. The angle formed between the tire's orientation and the orientation of the velocity vector of the corresponding wheel is characterized as the slip angle of a tire, which is illustrated in Figure 2.5 for the front wheel.

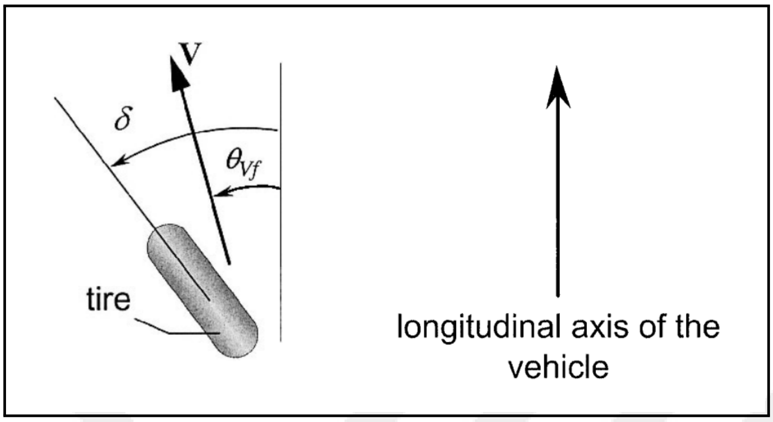

Figure 2.5 : Slip-angle of a tire [17].

The slip angle for the front and rear wheels is determined in Equation 2.8 and Equation 2.9, respectively.

$$\alpha_f = \delta - \theta_{Vf} \tag{2.8}$$

$$\alpha_r = -\theta_{Vr} \tag{2.9}$$

Consequently, the lateral tire force acting on the front and rear wheels of the vehicle can be formulated as follows:

$$F_{yf} = 2C_{\alpha f}(\delta - \theta_{Vf}) \tag{2.10}$$

$$F_{yr} = 2C_{\alpha r}(-\theta_{Vr}) \tag{2.11}$$

The $C_{\alpha f}$ and $C_{\alpha r}$ are the cornering stiffness of each front and rear tire, which are the proportionality constant, respectively. While δ stands for the front wheel steering angle, the factor of 2 is incorporated to accommodate the presence of two front wheels. θ_{Vf} and θ_{Vr} signify the velocity angle of the front and rear tire, respectively, and the following equations can be employed to compute these angles.

$$\tan(\theta_{Vf}) = \frac{\dot{y} + l_f\dot{\psi}}{V_x} \tag{2.12}$$

$$\tan(\theta_{Vr}) = \frac{\dot{y} + l_r \dot{\psi}}{V_x} \tag{2.13}$$

Utilizing approximations for small angles, the following calculations can be employed for velocity angles.

$$\theta_{Vf} = \frac{\dot{y} + l_f \dot{\psi}}{V_x} \tag{2.14}$$

$$\theta_{Vr} = \frac{\dot{y} + l_r \dot{\psi}}{V_x} \tag{2.15}$$

Finally, substituting the values from Equations (2.8), (2.9), (2.14), and (2.15) into Equations (2.6) and (2.7), the state-space representation can be expressed as:

$$\frac{d}{dt}\begin{bmatrix} y \\ \dot{y} \\ \psi \\ \dot{\psi} \end{bmatrix} = \begin{bmatrix} 0 & 1 & 0 & 0 \\ 0 & -\frac{2(C_{\alpha f}+C_{\alpha r})}{mV_x} & 0 & -V_x - \frac{2(C_{\alpha f}l_f - C_{\alpha r}l_r)}{mV_x} \\ 0 & 0 & 0 & 1 \\ 0 & -\frac{2(C_{\alpha f}l_f - C_{\alpha r}l_r)}{I_zV_x} & 0 & -\frac{2(C_{\alpha f}l_f^2 + C_{\alpha r}l_r^2)}{I_zV_x} \end{bmatrix} + \begin{bmatrix} 0 \\ \frac{2C_{\alpha f}}{m} \\ 0 \\ \frac{2C_{\alpha f}l_f}{I_z} \end{bmatrix} \delta_f \tag{2.16}$$

Observing the equations above, it becomes evident that the primary influential external factor within this model is the tire forces. In fact, the vehicle's lateral motion can be controlled under the assumption that these tire forces are proportional to the slip angles of each wheel being small. Under these circumstances, the lateral tire force becomes contingent upon the slip angle, the normal tire load F_z, the tire-road friction coefficient μ, and additionally the magnitude of the simultaneously generated longitudinal tire force [17]. These tire forces serve as both the main source of external disturbances and the key determinants of the vehicle's motion, encompassing both disturbances and traction aspects [27]. In this case, although the incorporation of non-linear tire dynamics provides a more accurate simulation of the vehicle's response, particularly in scenarios involving high speeds and significant steering angles, it makes the problem much more complicated.

2.5 Vehicle Dynamic Model based on Lateral Path Error

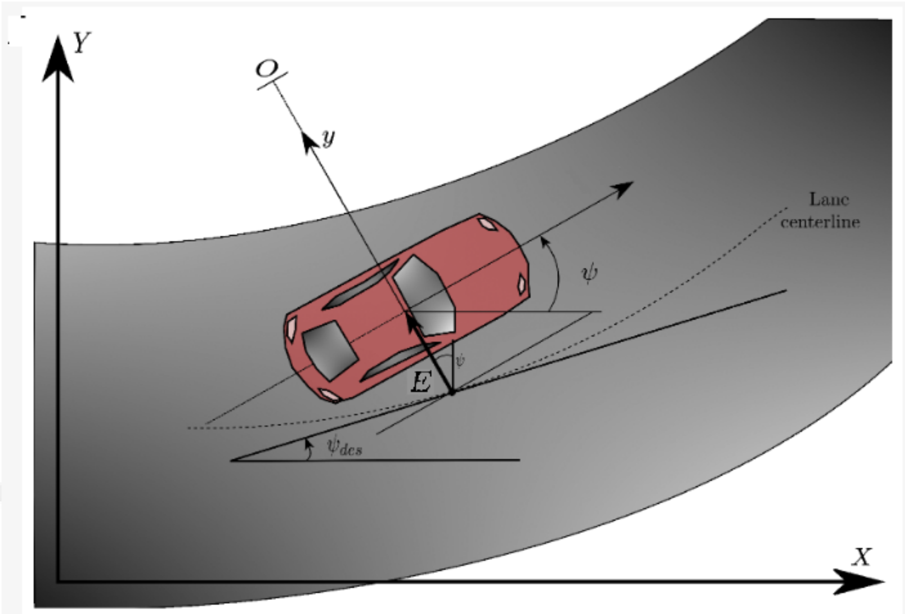

Figure 2.6 : Vehicle lateral path error based on the lane center [32].

In the realm of automotive engineering, the meticulous study and refinement of vehicle modeling play a pivotal role in enhancing both lateral stability, ensuring steadfast path-tracking, and overall driving performance. Therefore, the previously explained dynamic vehicle model approach can be customized based on the aim of the control point. In this case, the vehicle state-space model is customized for lane-tracking control purposes. In order to make a vehicle follow a lane center the vehicle's lateral position and orientation should be controlled. The lateral path error of a vehicle is the distance between the point of the vehicle's center of gravity and the nearest point at the lane center along the rolling center of the vehicle as illustrated in Figure 2.6. Furthermore, the dynamic of lateral path error E is represented in Equation 2.16, where $\dot{E}$ is related to vehicle lateral velocity, longitudinal velocity, and the yaw angle.

$$\dot{E} = \dot{y} + V_x \psi \tag{2.17}$$

Using this relation and restructuring Equation 2.16, the state-space model of the vehicle is derived as follows.

$$\frac{d}{dt}\begin{bmatrix}\dot{y}\\ \dot{\psi}\\ \psi\\ \dot{E}\end{bmatrix} = \begin{bmatrix} -\frac{2(C_{\alpha f}+C_{\alpha r})}{mV_x} & -V_x-\frac{2(C_{\alpha f}l_f-C_{\alpha r}l_r)}{mV_x} & 0 & 0\\ -\frac{2(C_{\alpha f}l_f-C_{\alpha r}l_r)}{I_zV_x} & -\frac{2(C_{\alpha f}l_f^2+C_{\alpha r}l_r^2)}{I_zV_x} & 0 & 0\\ 0 & 1 & 0 & 0\\ 1 & 0 & V_x & 0\end{bmatrix}\begin{bmatrix}\dot{y}\\ \dot{\psi}\\ \psi\\ \dot{E}\end{bmatrix} + \begin{bmatrix}\frac{2C_{\alpha f}}{m}\\ \frac{2C_{\alpha f}l_f}{I_z}\\ 0\\ 0\end{bmatrix}\delta_f \tag{2.18}$$

It should be noted that lateral path error is a measurable state via several methods, as explained in [33]. Thus, matrix C, representing the system output, is shown as follows.

$$C = \begin{bmatrix}0 & 0 & 0 & 1\end{bmatrix} \tag{2.19}$$

Before deriving the transfer function of the system, it must be clarified that the vehicle speed remains stable in order to follow a linear vehicle model approach [26]. In this thesis, it is assumed that the vehicle is an autonomous bus, and its speed is controlled by an ACC (Adaptive Cruise Controller) to make the speed remain. Thus, the system transfer function, where the output to be controlled is lateral path error E, and the input is the steering angle δ, is derived via Equation 2.20.

$$\frac{E(s)}{\delta(s)} = G(s) = C(sI-A)^{-1}B+D \tag{2.20}$$

Consequently, the system transfer function is derived as follows, using the parameters represented in Table 2.1 in Equation 2.18 and applying Equation 2.20.

$$\frac{E(s)}{\delta(s)} = \frac{31.83(s^2+2.52s+24.87)}{s^2(s^2+7.4s+3.75)} \tag{2.21}$$

When the transfer function above is examined, it is seen that the relative degree of the system is two, and zeros are on the left side of the real axis; therefore, the numerator polynomial is Hurwitz. In addition, the high-frequency gain is known as a positive value. According to these system features, it is expected that a controller that provides good results for lane tracking purposes during the parameter uncertainties and varying conditions should be designed.

Table 2.1 : Vehicle parameters.

Parameter	Value	Description
m	16500 kg	vehicle mass
l_f	4.07 m	distance from front axle to center of gravity
l_r	2.03 m	distance from front axle to center of gravity
$C_{\alpha f}$	262570 KN/rad	cornering stiffness of front tires
$C_{\alpha r}$	262570 KN/rad	cornering stiffness of rear tires
I_z	128800 kgm^2	moment of inertia about z-axis
ψ	-	vehicle yaw angle
$\dot{\psi}$	-	vehicle yaw rate
V_x		vehicle longitudinal velocity
$\dot{y} = V_y$	-	vehicle lateral velocity
δ	-	steering angle of the vehicle

For this purpose, an MRAC with augmented error is designed by considering that the system's transfer function structure is known, but its parameters are unknown or changing over time in Chapter 3.

3. MODEL REFERENCE ADAPTIVE CONTROLLER DESIGN FOR LANE TRACKING

3.1 Introduction to Adaptive Control Systems

In everyday terminology, "adapt" typically refers to altering one's actions to align with new circumstances. Conventional controllers are not suitable for application in systems with dynamic parameter variations. This leads to the need for adaptive controllers. An adaptive control is a control system methodology used in engineering and automation to adjust the control parameters or strategies of a system in real time based on its changing characteristics or operating conditions. The primary goal of adaptive control is to maintain desired system performance even when the system's parameters, dynamics, or external disturbances are uncertain or vary over time.

Adaptive control systems can be categorized differently [34]. Generally, it's essential to differentiate between:

1. Feedforward adaptive control
2. Feedback adaptive control

Additionally, there's a distinction between:

1. Direct methods
2. Indirect methods

Direct methods involve using estimated parameters directly in the adaptive controller, while indirect methods utilize estimated parameters to calculate the necessary controller parameters. There are four main types of adaptive systems [26]:

1. Gain Scheduling
2. Self-tuning Regulators
3. Dual Control
4. Model-Reference Adaptive Control

3.1.1 Gain scheduling

Gain scheduling is a method used in adaptive control systems to improve the performance and stability of a control system when the system's dynamics change or vary over time. It is a form of control strategy where the controller's parameters, often referred to as gains, are adjusted or scheduled based on certain operating conditions or the state of the system. The primary idea behind gain scheduling is to tailor the controller's behavior to match the changing dynamics of the system, ensuring that the control system remains effective and stable under different operating conditions.

The gain scheduling block diagram for an adaptive system is illustrated in Figure 3.1, where it is seen that the system has two loops. In the inner loop, there is a controller generating the controller signal to the process while the controller parameters are updated based on the operating conditions in the outer loop.

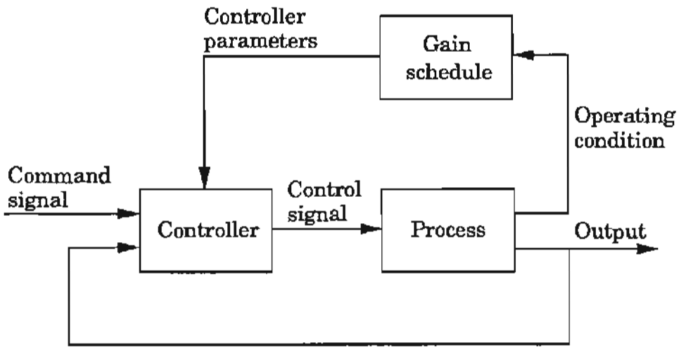

Figure 3.1 : Gain scheduling adaptive control system block diagram [26].

Gain scheduling is commonly used in various applications, such as aerospace control systems (where aircraft operate at different altitudes and speeds) and industrial processes (where environmental conditions, load, or process dynamics change). It helps achieve better control system performance and robustness in the face of changing operating conditions.

3.1.2 Self-tuning regulators

A Self-Tuning Regulator (STR) is a type of adaptive control system that continuously adjusts its controller parameters based on the real-time behavior of the controlled process. It is designed to improve the performance and robustness of control systems in the presence of changing system dynamics, disturbances, or uncertainties [26]. Consisting of two loops, an STR system is represented in Figure 3.2, where the inner loop is like an ordinary closed-loop system, while the controller parameters are adjusted according to the estimation of process parameters at each sampling period in the outer loop.

Direct and indirect methods are available contents in STR. In the indirect method, as illustrated in Figure 3.2, the controller parameters undergo indirect updates through design calculations. However, in certain cases, it becomes feasible to transform the process into a representation directly correlating with the controller parameters. This simplifies the algorithm by eliminating the need for design calculations, "Controller design" in direct method.

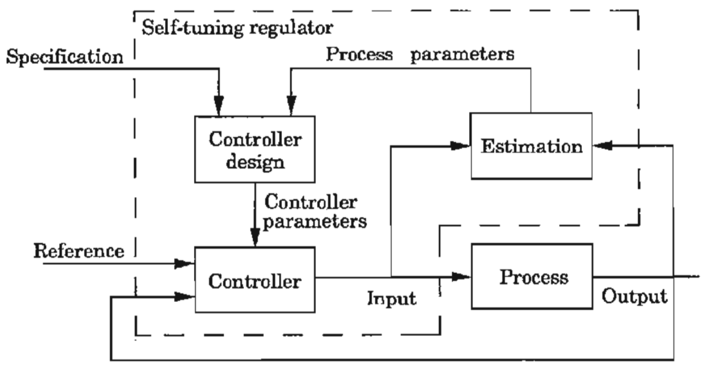

Figure 3.2 : Block diagram of an adaptive system with self-tuning regulator [26].

Self-tuning regulators are applied across different industries, encompassing manufacturing, process control, automotive control systems, and robotics. They prove especially valuable in scenarios where the controlled process experiences substantial variations over time or when creating an accurate model of the process is difficult due to complex or uncertain dynamics.

3.1.3 Dual control

Dual control theory is a specialized field within control theory that focuses on managing systems with initially unknown characteristics [35]. In dual control, an abstract problem formulation and optimization theory are employed to offer a solution. As shown in Figure 3.3, the controller structure has two parts. A nonlinear estimator is one part that produces the state's conditional probability distribution based on the measurements of the control signal u and system output y; also, this distribution is referred to as hyperstate calculation. The second part is the nonlinear feedback controller, which transforms the hyperstate into the controller variables' domain [26]. Applications of dual control can be found in various fields, including manufacturing, robotics, and process control. However, it should be noted that the dual control approach is complicated.

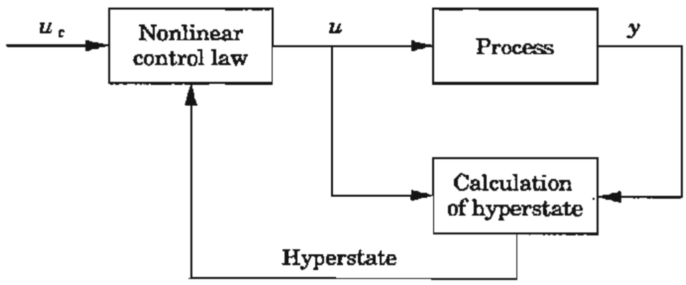

Figure 3.3 : Block diagram of a dual controller system [26].

3.1.4 Model-reference adaptive control

Model reference adaptive control (MRAC) is one of the most promising methods in the adaptive control approach that revolves around addressing a problem where performance specifications are defined in relation to a reference model. The goal is to make the system's behavior match the desired behavior specified by the reference model, even when the system's dynamics are uncertain or changing. A block diagram of a model reference adaptive control system is illustrated in Figure 3.4. MRAC consists of three primary components:

1. Reference Model: It gives the expected response to the input command signal so that the controller parameters are adapted to make the plant response match its behavior.
2. Controller: The structure of the controller depends on the problem to be solved. Generally, it comprises feedback and feed-forward terms, which are the polynomials, and the parameters of these polynomials are adapted.
3. Adjustment Mechanism: This critical component is used to modify the system's parameters or settings to align it with the reference model. The purpose of such mechanisms is to ensure that the controlled system behaves in a way that

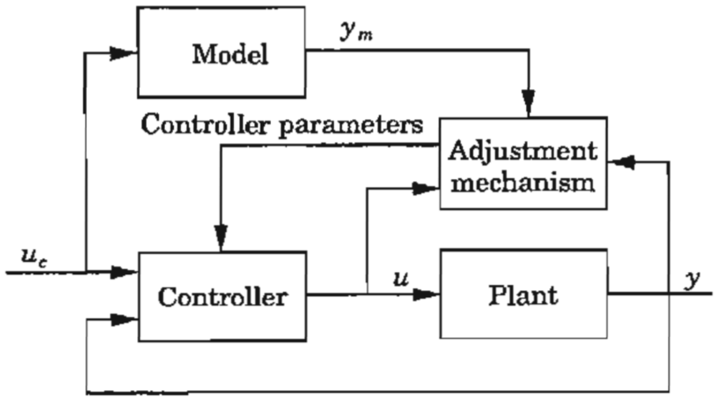

Figure 3.4 : Block diagram of a model reference adaptive control system [26].

meets specified performance criteria and achieves the desired outcomes. Different mathematical approaches, such as Lyapunov theory, the MIT rule, and the theory of augmented error, serve as tools to create adjustment mechanisms [36].

Choosing a reference model according to the expected result is not the only advantage of MRAC, but it also offers benefits ordered as follows:

- Robustness
- Stability
- Tuning simplicity (fine-tuning
- Adaptation to changing dynamics
- Versatility
- Real-time application implementation

However, it's important to note that the choice of control technique depends on the specific characteristics of the system and the control objectives. MRAC may not be the best choice for all applications. Other adaptive control techniques, such as the Self-Tuning Regulator (STR) or Model-Free Adaptive Control (MFAC), may be more suitable in certain scenarios. The effectiveness of a particular adaptive control technique depends on the system's dynamics, the level of uncertainty, and the control performance requirements. In this thesis, an MRAC with augmented error method is chosen to solve the lane tracking problem for an autonomous large-size vehicle whose transfer function was derived in the previous chapter.

In this section, the general features of adaptive systems are described briefly. The following section explains the background of augmented error theory and the design of an MRAC with augmented error method for lane tracking control in detail, respectively.

3.2 Model Reference Adaptive Control with Augmented Error

Model reference adaptive controller with augmented error is an adaptive controller design method that ensures stability based on the passivity of input-output of the closed-loop system together with estimation error dynamics of controller parameters [26]. The method provides promising results, where the system to be controlled is linear and has one input and output. The essential assumption is that the term augmented error must be linearly correlated to the adaptable controller parameters [26].

To better understand the background of the augmented error method, the adaptation of feedforward gain based on Lyapunov theory, which describes the stability solution of the output feedback adaptive system based on the state-space view, is explained first. Then, the relation between stability analysis according to state-space view and input-output (passivity) view will be described.

3.2.1 Lyapunov theory (state-space) based model reference adaptive control

3.2.1.1 Adaptation of feedfoward gain

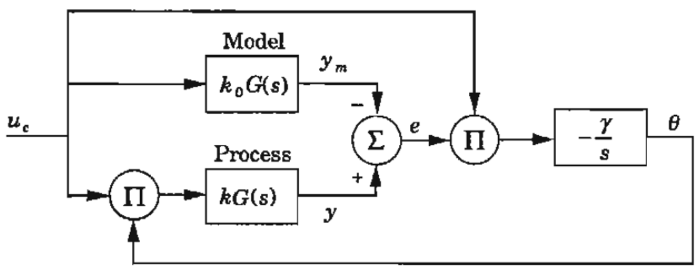

Figure 3.5 : Block diagram of MRAC system with the adaptation of feedforward gain based on Lyapunov Rule [26].

Considering Figure 3.5, the system transfer function is $kG(s)$, where $G(s)$ is known, but the value of k remains unknown, and the reference model defined by the transfer function $k_0G(s)$ provides the desired response. The model error e is expressed by

$$e = kG(s)(\theta - \theta^0)u_c \tag{3.1}$$

where $\theta^0 = k_0/k$ and represents the initial value of the controller parameter while u_c is the command signal. Using this equation and casting $G(s)$ into a state space form, the relation between error e and adaptable controller parameter θ is written as follows.

$$\begin{aligned} \frac{dx}{dt} &= Ax + B(\theta - \theta^0)u_c \\ e &= Cx \end{aligned} \tag{3.2}$$

If the homogeneous system $\dot{x} = Ax$ is asymptotically stable, then there is positive definite matrices P and Q that the following Lyapunov equation is satisfied [26]:

$$A^TP + PA = -Q \tag{3.3}$$

Thus, by choosing a candidate Lyapunov function as in Equation 3.4 and using the parameter updating rule described in Equation 3.5, the state vector x and the error $e = Cx$ go to zero, which means the stability of the process is guaranteed [26].

$$V = \frac{1}{2}(\gamma x^TPx + (\theta - \theta^0)^2) \tag{3.4}$$

$$\frac{d\theta}{dt} = -\gamma u_c B^TPx \tag{3.5}$$

Nonetheless, all the state variables must be known to obtain such a stable closed-loop system, which is quite limiting. In the case where there exists a positive definite matrix P that satisfies Equation 3.3 and Equation 3.6, output feedback can be used instead of state feedback as follows.

$$B^TP = C \tag{3.6}$$

$$\frac{d\theta}{dt} = -\gamma u_c e \tag{3.7}$$

Here, it is necessary to state Kalman-Yakubovic Lemma that relates this result to the strictly positive realness of a system transfer function [26].

Lemma **1** *Kalman-Yakubovic lemma*

Let the time-invariant linear, completely controllable, and observable system be as follows.

$$\begin{aligned} \frac{dx}{dt} &= Ax + Bu \\ y &= Cx \end{aligned} \tag{3.8}$$

The transfer function

$$G(s) = C(sI - A)^{-1}B \tag{3.9}$$

is strictly positive real if there exist positive definite P and Q such that

$$A^T P + PA = -Q \tag{3.10}$$

and

$$B^T P = C \tag{3.11}$$

Based on the Kalman-Yakubovic lemma, the following theorem can be given [26].

Theorem **1** *MRAC using the Lyapunov rule*

Considering the problem of adapting a feedforward gain, the output error in Equation 3.2 goes to zero with the parameter adjustment rule derived below, assuming that the transfer function G is strictly positive real.

$$\frac{d\theta}{dt} = -\gamma u_c e \tag{3.12}$$

Here, the γ is the positive constant known as adaptation gain.

At this point, the definition of strictly positive realness should be stated below [37].

Definition **1** *A rational function $G(s)$ is strictly positive real if and only if*

$$\begin{aligned} &G(s) \text{ is analytic in } Re[s] \geq 0, \\ &Re[G(jw)] > 0 \; \forall w \in (-\infty, \infty), \\ &lim_{w^2 \to \infty} w^2 Re[G(jw)] > 0 \text{ when } n = 1, and \\ &lim_{|w| \to \infty} (G(jw))/jw > 0 \text{ when } n = -1, \end{aligned} \tag{3.13}$$

where n is the relative degree of $G(s)$.

The parameters adjustment rule, which is given in Theorem 1, ensures the stability of the system G if it is SPR. It must be noted that the system description is determined with the help of Lemma 1, and it is the state-space view [19]. Finally, the stability based on this approach is proved by Definition 1.

3.2.2 Input-output (passivity theory) based model reference adaptive control

In this subsection, the concept that allows the solution of adaptive control problems with output feedback via considering input-output stability is formulated based on passivity theory. Passivity is a fundamental method closely associated with the concept of positive realness. In order to explain this relation, dissipativity and properness must be explained first.

Definition **2** *The dissipativity of a system is explained as follows.*

For the systems;

$$\begin{aligned} \dot{x} &= f(x,u), u \in U \\ y &= h(x,u), y \in Y \end{aligned} \tag{3.14}$$

X is the state value at time t_1 that is reached from the initial condition X with $u(0)$ input signal,
$\forall x_0 \in X$, $\forall t_1 \geq t_0$, and for all input functions, if there exists a storage function $V(t)$ where

$$V(0) = 0 \tag{3.15}$$

and

$$V(x(t_1)) \leq V(x(t_0)) + \int_{t_0}^{t_1} S(u(t), y(t))\, dt \tag{3.16}$$

then the system is dissipative with respect to S.

Some special cases of dissipativity are defined in Definition 3 stated below [26].

Definition **3** *If a system is dissipative according to*

$$\begin{aligned} &S = u^T y \text{ then it is passive} \\ &S = u^T y - \delta ||u||^2 \text{where } \delta > 0 \text{ then it is input strictly passive (ISP)} \\ &S = u^T y - \varepsilon ||y||^2 \text{where } \varepsilon > 0 \text{ then it is output strictly passive (OSP)} \end{aligned} \tag{3.17}$$

On the other hand, the properness property of a system is defined as below.

Definition **4** *Proper system :*

A system $G(s)$ is proper if the relative degree between the denominator and numerator polynomials is larger than or equal to zero.

Finally, declaring a SPR system is also defined as OSP, and proper, proof of the input-output stability based on passivity is defined with the help of Kalman-Yakubovic-Popov lemma as follows, which is referred as a cornerstone of the control system applications in absolute stability, dissipativity, passivity, optimal control, adaptive control, stochastic control and filtering [38].

Lemma **2** *Kalman-Yakubovic-Popov lemma*

Let the completely controllable and observable system be as follows.

$$\begin{aligned} \frac{dx}{dt} &= Ax + Bu \\ y &= Cx \end{aligned} \tag{3.18}$$

The (pxp) transfer function

$$G(s) = C(sI - A)^{-1}B + D \tag{3.19}$$

is strictly positive real and passive if and only if

$$\exists P = P^T > 0, P \in R^{nxn}, w \in R^{pxp} \text{ and } L \in R^{pxn} \text{ and } \varepsilon > 0 \tag{3.20}$$

such that

$$
\begin{aligned}
A^T P + PA &= -L^T L - \varepsilon P \\
PB &= C^T - L^T w \\
w^T w &= D + D^T
\end{aligned}
\tag{3.21}
$$

Then system $G(s)$ has all the poles, A is Hurwitzian, left-hand plane, and is stable [39].

In summary, that a system G is SPR represents the same meaning it is OSP and proper. As as a result, a parameter adjustment rule similar to Theorem 1 can be derived to ensure the input-output stability based on passivity. In next section this similar theorem is explained.

3.2.3 The augmented error

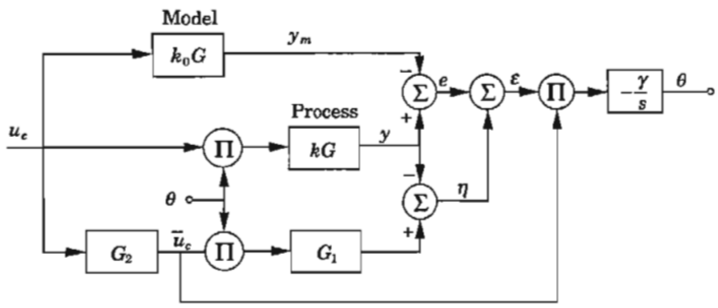

Figure 3.6 : Block diagram of a model reference adaptive control system based on augmented error [26].

The adaptation of feedforward gain for a stable closed loop system requires an SPR system according to Theorem 1, as described before. However, Definition 1 defines that systems in which G has a relative degree larger than one are not SPR. This is solved with augmented error, an insight by passivity theory [26]. The system transfer function G is factorized as

$$G = G_1 G_2 \tag{3.22}$$

such that the transfer function G_1 is SPR. Thus, the error e in Figure 3.5 is rearranged as

$$
\begin{aligned}
e &= G(\theta-\theta^0)u_c = G_1G_2(\theta-\theta^0)u_c \\
&= G_1(G_2(\theta-\theta^0)u_c + (\theta-\theta^0)G_2u_c - (\theta-\theta^0)G_2u_c) \\
&= G_1((\theta-\theta^0)G_2u_c) - G_1((\theta-\theta^0)G_2u_c - G_2(\theta-\theta^0)u_c)
\end{aligned} \tag{3.23}
$$

and introducing the augmented error ε defined as follows

$$\varepsilon = e + \eta \tag{3.24}$$

where η is the error augmentation determined by

$$
\begin{aligned}
\eta &= G_1(\theta-\theta^0)G_2u_c - G(\theta-\theta^0)u_c \\
&= G_1(\theta G_2u_c) - G\theta u_c
\end{aligned} \tag{3.25}
$$

Here, the term θ represents the adaptable unknown controller parameters, while θ^0 is the true value for known conditions. The error augmentation term η vanishes during the adaptation if the controller parameter θ to be adapted is constant. Thus, the augmented error is rewritten as follows.

$$\varepsilon = G_1(\theta-\theta^0)G_2u_c) = G_1(\theta-\theta^0)u_{cf} \tag{3.26}$$

where the u_{cf} is the reference signal filtered through the factor G_2. Now, the following theorem can be stated by applying the results in Theorem 1 to the augmented error in Equation 3.26 as given in [26].

Theorem **2** *Stability using augmented error*

Consider a model-reference system for adaptation of feedforward gain for a system with the transfer function G. Let G_1G_2 be a factorization of G such that G_1 is SPR. The parameter adjustment law

$$\frac{d\theta}{dt} = -\gamma\varepsilon(G_2u_c) \tag{3.27}$$

where

$$\varepsilon = e + G_1(\theta G_2u_c) - G(\theta u_c) \tag{3.28}$$

gives a closed-loop system in which the error goes to zero as time goes to infinity.

3.2.4 Model reference adaptive controller design for lane tracking

In this section, the design steps of an adaptive control system with model reference using the augmented error method are explained. The system model, which is already derived in Chapter 2, represents an electric bus of an international heavy vehicle provider aiming to achieve autonomous capabilities.

Before explaining the design steps in detail, the reason why the model reference adaptive controller with augmented error is chosen in this thesis can be summarized as:

1. The advantages of MRAC mentioned previously.

2. ADAS professional tools, such as smart cameras, are employed for measuring lateral path error in the scope of the lane detection system (LDS). This concept is crucial when designing an adaptive controller with output feedback, ensuring observability.

3. The system transfer function in Equation 2.21 is not SPR. Therefore, error augmentation, which is the main step of augmented error method mentioned before, eliminates this problem (controllability).

Model reference adaptive controller design with augmented error method is handled in three primary stages. First, a suitable controller structure that can follow the desired output is determined. Afterward, to have a stable closed-loop system, on the basis of input-output, the augmented error expression is derived. Finally, the controller parameter adaptation rules are derived based on this error expression to guarantee the passivity of the closed-loop system. It should be noted that the vehicle system used in the thesis, in Equation 2.21, is considered for the following calculations.

3.2.4.1 Reference model and controller structure determination

In model reference adaptive control, determining a controller structure that can follow the reference model output is a crucial step. In this thesis, the controller structure is determined based on the pole-placement design method [19], and this method is summarised as follows.

In pole placement, the system represented in Equation 3.29 has the A and B polynomials, which are relatively prime, and A is monic.

$$Ay(t) = Bu(t) \tag{3.29}$$

A general two-degree-of-freedom linear controller structure can be defined as follows.

$$Ru(t) = Tu_c(t) - Sy(t) \tag{3.30}$$

Here, R, S, and T are polynomials of the controller, and $y(t)$ is the system output while $u(t)$ and $u_c(t)$ are the control and command signals, respectively. The control structure includes a feedback term $-S/R$ and a feedforward term T/R. Thus, the closed-loop control system is derived as in Equation 3.31.

$$y(t) = \frac{BT}{AR + BS} u_c(t) \tag{3.31}$$

The denominator term, thus, is the closed-loop characteristic equation

$$AR + BS = A_c \tag{3.32}$$

In the literature, this equation, referred to as the Diophantine equation, plays a significant role in determining the desired system solution through the determination of polynomials R and S [26]. Furthermore, as long as there is no common factor of A and B polynomials, at least one solution exists for this equation. Besides, there is always a solution if the S polynomial's degree is lower than that of the A polynomial [26]. This solution of Diophantine introduces R and S polynomials with unknown parameters. To determine the T controller polynomial with its unknown parameters, the causality and compatibility conditions are used. To do this, the desired model structure is defined as follows.

$$A_m y_m(t) = B_m u_c(t) \tag{3.33}$$

Then, Equation 3.31 must satisfy the condition below to achieve the perfect model following.

$$\frac{BT}{AR+BS} = \frac{BT}{A_c} = \frac{B_m}{A_m} \tag{3.34}$$

In the model following design, unstable zeros cannot be canceled, and only stable zeros can be canceled. Therefore BT and A_c should be factorized accordingly. The B polynomial, which includes the zeros of the system, is factorized based on the stable and unstable zeros as

$$B = B^+B^- \tag{3.35}$$

where B^+ defines the factor including stable zeros that the controller can cancel, while B^- includes the unstable zeros that cannot be canceled.

It is, thus, evident that the reference model zeros polynomial B_m must include the factor B^-, and it is rearranged like below.

$$B_m = B'_mB^- \tag{3.36}$$

Besides, the A_c must include zeros to be canceled B^+ and A_m.

$$A_c = A_0A_mB^+ \tag{3.37}$$

Considering Equations 3.32 and 3.37, the R polynomial is factorized like below.

$$R = R_1B^+ \tag{3.38}$$

Thus, after the cancellation of the B^+ factor, the Diophantine equation is rearranged as

$$AR_1 + B^-S = A_0A_m = A'_c \tag{3.39}$$

Finally, substituting Equations 3.35, 3.36, and 3.37 into 3.34, the controller polynomial T is derived as follows.

$$T = A_0 B_m' \tag{3.40}$$

At this stage, to complete the controller structure design based on the pole placement method for the model following, the causality and compatibility conditions must be defined.

Definition **5** *Causality Conditions*

A causal controller, in both discrete time and continuous time, must satisfy the below conditions.

$$\begin{aligned} deg(S) &\leq deg(R) \\ deg(T) &\leq deg(R) \end{aligned} \tag{3.41}$$

Definition **6** *Compatibility Conditions*

$$\begin{aligned} deg(A_m) &= deg(A) \\ deg(B_m) &= deg(B) \\ deg(A_0) &= deg(A) - deg(B) - 1 \end{aligned} \tag{3.42}$$

Finally, considering these conditions and the explained design steps, there is always a solution where the degree of S controller polynomial is at most deg $A - 1$, which is called the minimum-degree solution to the Diophantine Equation [26].

Considering the minimum degree solution of the Diophantine equation according to the above statement, the MRAC design for lane tracking with augmented error method for the autonomous bus is explained as follows.

In this thesis, the controller structure is selected considering the pole placement control design. The reference model determination and details of MRAC design for lane tracking of considered vehicles are given in the following.

Consider the following model that represents linear time-invariant systems with single input and single output.

$$Ay(t) = b_0 B u(t) \tag{3.43}$$

A and B are the polynomials representing the system dynamics, and it is assumed that they do not have a common factor A and B, and they are monic. On the other hand, b_0 stands for the high-frequency gain. Therefore, when the vehicle system, in Equation 2.21, is considered, the dynamic polynomials and high frequency are obtained as below.

$$\begin{aligned} A &= s^4 + 7.4s^3 + 3.75s^2 \\ B &= s^2 + 2.52s + 24.87 \\ b_0 &= 31.83 \end{aligned} \tag{3.44}$$

Substituting Equation 3.30 (2 DoF controller) into 3.43 and making the necessary simplifications, the closed-loop system is obtained as follows.

$$y(t) = \frac{b_0 BT}{AR + b_0 BS} u_c(t) \tag{3.45}$$

Thus, the closed-loop system characteristic polynomial A_c is then as follows.

$$AR + b_0 BS = A_c \tag{3.46}$$

In this context, with the minimum degree solution principle of the Diophantine equation, and considering the degree of polynomial A is 4, the degree of the S polynomial is determined as below.

$$degree(S) = degree(A) - 1 = 3 \tag{3.47}$$

Thus, an ordinary polynomial S with a degree of 3 can be written as follows.

$$S = s_0 s^3 + s_1 s^2 + s_2 s + s_3 \tag{3.48}$$

In order to determine the degree of polynomials R and T, the causality conditions which is described in Definition 5 must be considered [26]. Thus, the degrees of R and T polynomials are determined the same as the degree of S.

$$degree(S) = degree(R) = degree(T) = 3 \tag{3.49}$$

It is clear that the form of polynomials R and T will be the same as S, as in Equation 3.50 below.

$$\begin{aligned} R &= s^3 + r_0 s^2 + r_1 s + r_2 \\ T &= t_0 s^3 + t_1 s^2 + t_2 s + t_3 \end{aligned} \tag{3.50}$$

In order to determine an algorithm for the parameters of these controller polynomials, the reference model structure must be determined at this stage. It should be noted that, for the determination of the reference model structure based on the minimum degree pole placement method, the compatibility conditions (Definition 6) must be taken into consideration [26]. Thus, the reference model used in this thesis is determined as follows.

$$G_m(s) = \frac{b_{m0} B_m}{A_m} = \frac{19.15(s+2.1)(s+0.56)}{(s+2.04)(s+2.93)(s^2+7.4s+3.75)} \tag{3.51}$$

Since the aim of the controller design is lane tracking for a vehicle, the desired performance characteristics should be evaluated with a focus on driving comfort during the determination of the reference model. For this reason, overshooting is not a desired behavior in lane tracking control. Therefore, the two stable poles of the vehicle system (Equation 2.21) are retained while determining the reference model transfer function. Nevertheless, one of the stable poles of the system, which is located very close to the stability boundary at -0.54, is a possible problem for stability. In order to eliminate its influence, a zero at -0.56 is placed. The two integrator poles of the vehicle system at the origin are shifted to the left side of the real axis, and these poles are placed at -2.04 and -2.93 points to ensure that the system control effort does not exceed the required level. Finally, the high-frequency gain b_{m0} is calculated to match the unity continuous-time gain of the reference model (dc gain=1) [22].

In this study, since the zeros of the system, denoted as B, are stable, these stable zeros can be eliminated based on the model following approach with zero cancellation. To achieve this, the control polynomial R, whose degree has been previously determined, must include these stable system zeros. In other words, the polynomial B must be a factor of R. Thus, the polynomial B can be expressed as one of the terms of the polynomial R, as follows.

$$R = R_1 B \tag{3.52}$$

Thus, considering A_c includes B, A_m and A_0 in order to eliminate the stable zeros and to achieve the desired characteristic of reference model The Diophantine equation (Equation 3.32) can be rearranged with this information.

$$AR_1 B + b_0 BS = A_c = A_0 A_m B \tag{3.53}$$

Here, to determine the R_1 component of the polynomial R, the degree of the A_0 polynomial, which must be monic and stable, can be determined via Definition 6. Therefore, A_0 is selected as a one-degree monic polynomial as follows.

$$A_0 = s + a_0 \tag{3.54}$$

Since the highest-degree term in Equation 3.53 is AR_1B, and it must be equal to the degree of A_c, it is evident that the degree of the R_1 term must be 1.

$$R_1 = s + r_0 \tag{3.55}$$

As a result, when Equation 3.55 is substituted into 3.52 , the controller polynomial R is obtained as follows.

$$R = (s + r_0)(s^2 + b_1 s + b_2) \tag{3.56}$$

The polynomial T is determined to match the closed-loop system zeros to the reference model zeros. In this context, when the closed-loop system transfer function given in Equation 3.34 is equated to the reference model transfer function as follows, it is seen that the T polynomial should be selected in the form of $T = A_0 b_{m0} B_m / b_0$.

$$\frac{b_0BT}{AR+b_0BS}=\frac{b_0BT}{A_0A_mB}=\frac{b_{0m}B_m}{A_m} \tag{3.57}$$

Considering the reference model zeros with the polynomial B_m, the T polynomial is derived as in Equation 3.58 .

$$T=\frac{A_0b_{m0}B_m}{b_0}=\frac{(s+a_0)b_{m0}(s^2+b_{m1}s+b_{m2})}{b_0} \tag{3.58}$$

In this context, to verify that the control structure is appropriately selected, the Diophantine equation (3.32) is solved with known (nominal) values in Equations 3.44 and 3.51 for the coefficients of R and S, and the control structure is tested in a simulation environment. In the specified simulation, observer polynomial parameter a_0 was chosen as 10; the result was presented in Figure 3.6. Besides, the controller polynomials were derived as follows.

$$\begin{aligned} R &= s^3+17.5s^2+62.65s+372.30 \\ S &= 1.75s^3+14.82s^2+20.45s+7.04 \\ T &= 0.59s^3+7.58s^2+16.66s+7.04 \end{aligned} \tag{3.59}$$

For the case where the parameters are unknown, parametric expressions for controller polynomials given in Equations 3.48 and 3.50 , are considered. For the case, where it is necessary to create update rules (adaptation rules) in order to update the controller parameters appropriately while maintaining the closed-system stability. These update rules are designed for the parameters in the following error augmentation section.

3.2.4.2 Error augmentation

Model reference adaptive control with output feedback requires transfer function of the system to be strictly positive real (Theorem 2) to ensure stability based on passivity. Since, the system's transfer function (Equation 2.21) is not SPR, the augmented error method will be employed to enable the derivation of updating rules that guarantee stability based on passivity.

In the previous section, it was mentioned that if there is no common factor between the A and B polynomials, there always exist R and S polynomials that satisfy Diophantine

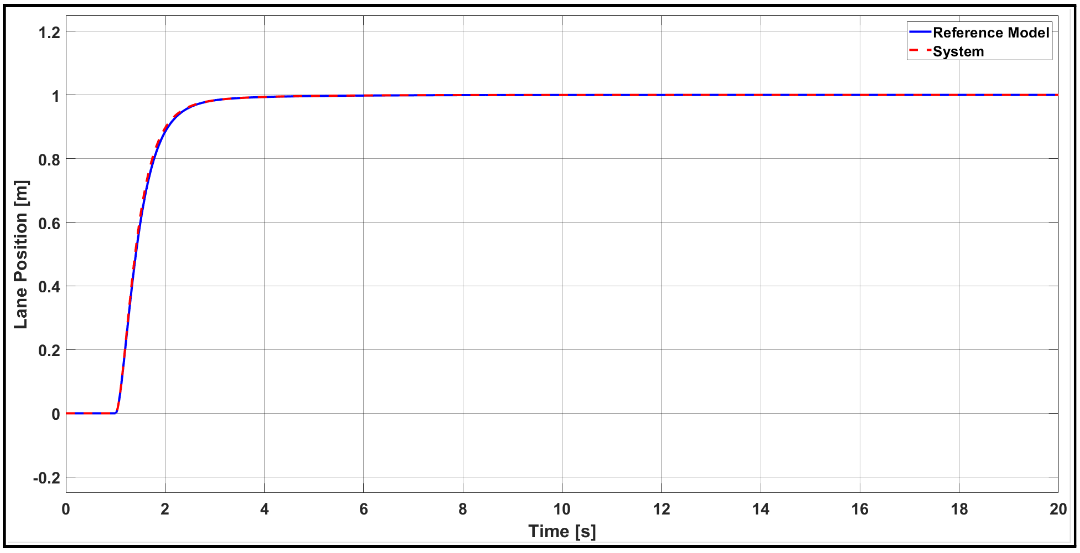

Figure 3.7 : Step response of reference model and closed-loop control system.

equation. Since it is known that the system zeros are stable, when the R polynomial is chosen as in Equation 3.52, the Diophantine equation (3.32) can be expressed as follows.

$$(AR_1 + b_0 S) = A_0 A_m \tag{3.60}$$

Furthermore, the structure in which stable zeros are eliminated for Equation 3.45 can be rearranged as follows.

$$(AR_1 + b_0 S)y = b_0 T u_c \tag{3.61}$$

Considering the closed-loop system given in Equation 3.61, when $T = t_0 A_0$ is chosen, it is seen that a selected reference model, as chosen for any A_m in Equation 3.62, can be tracked with the chosen control structure (model matching can be achieved) [26].

$$A_m y_m(t) = b_0 t_0 u_c(t) \tag{3.62}$$

Here, when the control polynomial obtained in Equation 3.58 is observed, it is seen the term t_0 is determined as follows.

$$t_0 = \frac{b_{m0}B_m}{b_0} \tag{3.63}$$

The error e is defined as the difference between the system output and reference model output.

$$e = y - y_m \tag{3.64}$$

Equations 3.60 and 3.61 can be expressed as follows when multiplied by y and A_0, respectively.

$$\begin{aligned} A_0A_my &= (AR_1 + b_0S)y \\ A_0A_my_m(t) &= A_0b_0t_0u_c(t) \end{aligned} \tag{3.65}$$

Thus, the error expression can be rearranged as below when the expressions in Equation 3.65 are substituted into Equation 3.64.

$$e = \frac{b_0}{A_0A_m}(Ru + Sy - Tu_c) \tag{3.66}$$

Since the transfer function, b_0/A_0A_m, in Equation 3.66 is not strictly positive real, determining adjustment rules that guarantee stability for a stable closed-loop system is not possible yet. Therefore, the tracking error is filtered with Q/P to obtain an appropriate error expression. Thus, the filtering process generates the filtered error e_f.

$$e_f = \frac{b_0Q}{A_0A_m}(\frac{R}{P}u + \frac{S}{P}y - \frac{T}{P}u_c) \tag{3.67}$$

The terms Q and P are polynomials, and the choice of Q, such that its degree does not greater than the degree of A_0A_m, will ensure that the condition for strictly positive realness. Additionally, the P polynomial, which filters the control u, system response y, and reference u_c signals, is generally expressed as follows.

$$P = P_1P_2 \tag{3.68}$$

It is crucial at this stage that the P_2 term is monic and has the same degree as R; thus, the filtered error expression can be rewritten as shown below [26].

$$e_f = \frac{b_0 Q}{A_0 A_m}\left(\frac{1}{P_1}u + \frac{R - P_2}{P}u + \frac{S}{P}y - \frac{T}{P}u_c\right) \tag{3.69}$$

The unknown parameters and the filtered measurable variables in Equation 3.69 can be expressed as in Equations 3.70 and 3.71 below.

$$\theta^0 = (r_1^{'} ... r_2^{'} s_0 ... s_3 t_0 ... t_3)^T \tag{3.70}$$

$$\varphi^T = \left(\frac{s^{k-1}}{P(s)}u ... \frac{1}{P(s)}u + \frac{s^l}{P(s)}y ... \frac{1}{P(s)}y - \frac{s^m}{P(s)}u_c ... - \frac{1}{P(s)}u_c\right) \tag{3.71}$$

Here, k, l and m represents the degrees of $R - P_2$, S and T, respectively. Thus, the filtered error expression is simplified with this vector representation as follows [26].

$$e_f = \frac{b_0 Q}{A_0 A_m}\left(\frac{1}{P_1}u + \varphi^T \theta^0\right) \tag{3.72}$$

It can be seen that the parameters θ^0 in the above expression are the nominal parameters, and this error value could be driven to zero if these parameters were known by choosing the control signal as $u = -\varphi^T \theta^0$. Since these parameters are unknown, the error expression is rearranged to have the adaptable parameters by adding and subtracting the term $\varphi^T \theta$ [26].

$$e_f = \frac{b_0 Q}{A_0 A_m}\left(\frac{1}{P_1}u + \varphi^T \theta + \varphi^T(\theta^0 - \theta)\right) \tag{3.73}$$

In the expression above, the first two terms are related to error augmentation, which is represented as η. In this context, with the following definition,

$$\eta = -\left(\frac{1}{P_1}u + \varphi^T \theta\right) \tag{3.74}$$

the augmented error ε is derived as in Equation 3.75.

$$\varepsilon = e_f + \frac{b_0 Q}{A_0 A_m}\eta = \frac{Q}{P}(y - y_m) + \frac{b_0 Q}{A_0 A_m}\eta \tag{3.75}$$

Thus, the augmented error, with $b_0 Q / A_0 A_m$ being strictly positive real, is obtained in the following standard form [26].

$$\varepsilon = \frac{b_0 Q}{A_0 A_m}\varphi^T(\theta^0 - \theta) \tag{3.76}$$

3.2.4.3 Parameter adjustment rules determination

Considering Theorem 2, it is seen that the input-output stability guarantee of the closed-loop system is achieved for the system with augmented error by choosing the parameter adjustment rule as in the equation below

$$\frac{d\theta}{dt} = \gamma\varphi\varepsilon \tag{3.77}$$

where, γ is a positive number representing the adaptation gain.

In order for the output tracking error (filtered error in Equation 3.72) to converge to zero, the error augmentation variable must also converge to zero in continuous-time, which means $\frac{1}{P_1}u + \varphi^T\theta$ must be zero. In this case, the control law for u may be chosen as follows [26].

$$u = -P_1\varphi^T\theta \tag{3.78}$$

However, $-P_1\varphi^T\theta$ is a function that contains derivatives. Since u in Equation 3.78 is a scalar, its transpose will be the same; therefore, a suitable control law for practical application can be chosen as in Equation 3.79, instead of an expression involving derivatives of θ [26].

$$u = -\theta^T(P_1\varphi) \tag{3.79}$$

In this case, η is rearranged as follows.

$$\eta = \frac{1}{P_1}\theta^T - \varphi^T \theta \tag{3.80}$$

Consequently, the needed equations for the realization of the controller design implementation can be summarized as follows [26].

$$\begin{aligned} y_m &= \frac{b_{m0}B_m}{A_m}u_c \\ e_f &= \frac{Q}{P}e = \frac{Q}{P}(y - y_m) \\ \eta &= -(\frac{1}{P_1}u + \varphi^T\theta) \\ \varepsilon &= e_f + \frac{b_0 Q}{A_0 A_m}\eta \\ \frac{d\theta}{dt} &= \gamma\varphi\varepsilon \\ u &= -\theta^T(P_1\varphi) \end{aligned} \tag{3.81}$$

Also, the block diagram of the model reference adaptive controller with augmented error is represented in Figure 3.8.

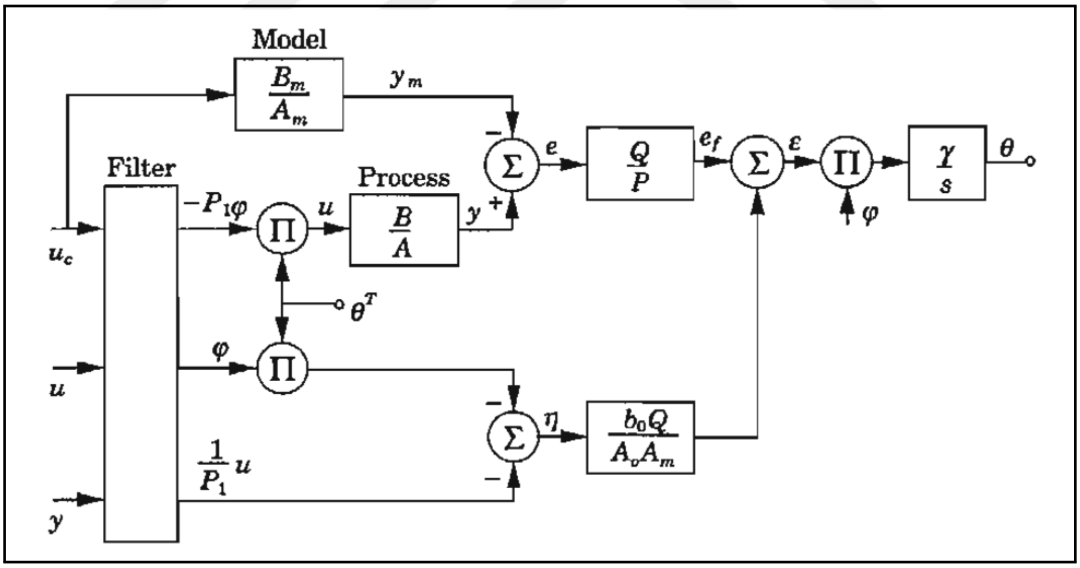

Figure 3.8 : Implementation block diagram of MRAC based on the augmented error [26].

It should be noted that the choice of the Q and P filter expressions used in this designed controller, as part of the augmented error method, significantly impacts the controller's performance. In this context, a common approach is to select the Q polynomial as

A_0A_m. At the same time, the P_1 and P_2 polynomial is chosen A_m and A_0, respectively [26]. However, the suggested choice for P is unsuitable for the condition SPR. Thus, in this implementation, the P_1P_2 expression is determined as follows, with the condition that the degree of P_2 is 3, such that the gain of Q/P is 1.

$$\begin{aligned} P_1 &= (s^2+7.4s+3.75) \\ P_2 &= (s+a_0)(s+2.04)(s+2.93) \\ Q = A_0A_m &= (s+a_0)(s+2.04)(s+2.93)(s^2+7.4s+3.75) \end{aligned} \tag{3.82}$$

where, a_0 is determined as 10 during the simulations, and the simulation results are represented in the following chapter.

4. SIMULATION

In this section, the model reference adaptive controller designed with augmented error method Simulink results are presented. The vehicle model designed in Chapter 2 is controlled via the controller designed in Chapter 3.

4.1 Simulink Results

The reference input $u_c(t)$ below is the desired lane position the vehicle must track, which is a sine wave with an amplitude of 1 meter. It is mentioned earlier that the vehicle speed, controlled by an adaptive cruise controller, is assumed to be constant at 20 m/s.

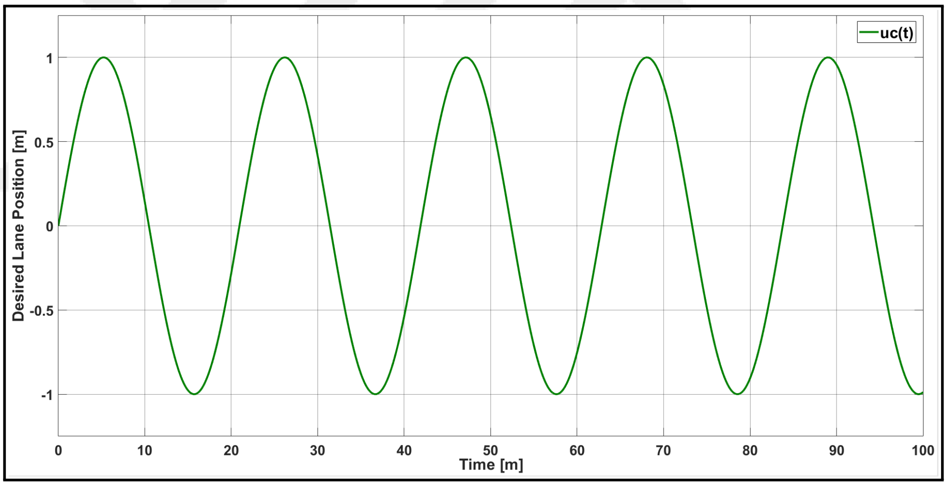

Figure 4.1 : Desired lane position.

Thus, the reference model and the system response according to the desired lane position are represented in Figure 4.2.

In this case, the adaptation of controller parameters of R, S, and T during the simulation is shown in Figure 4.3, Figure 4.4, and Figure 4.5, respectively. It should be noted that the adaptation adaptation gain γ is chosen 90.

In Figure Figure 4.6, it can be observed that the closed-loop system-reference model error value e reaches a steady-state response at approximately 40 seconds, with the

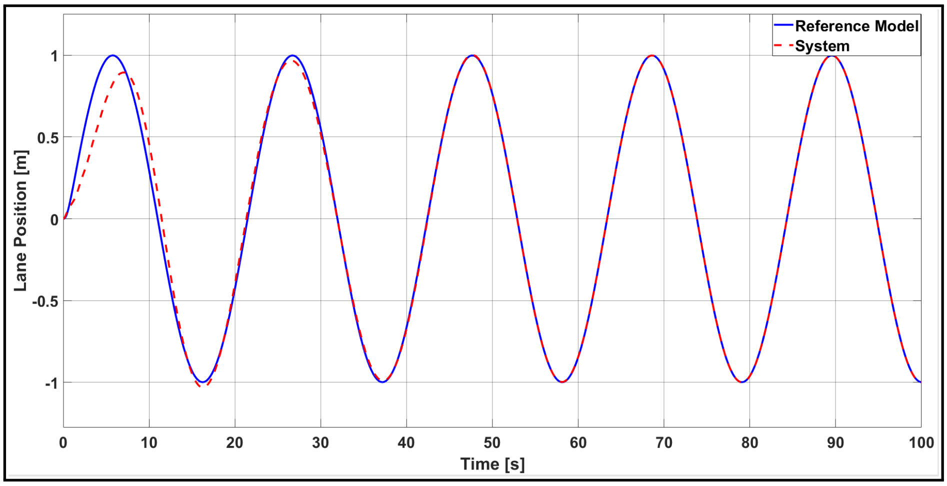

Figure 4.2 : Reference model and System lane position response.

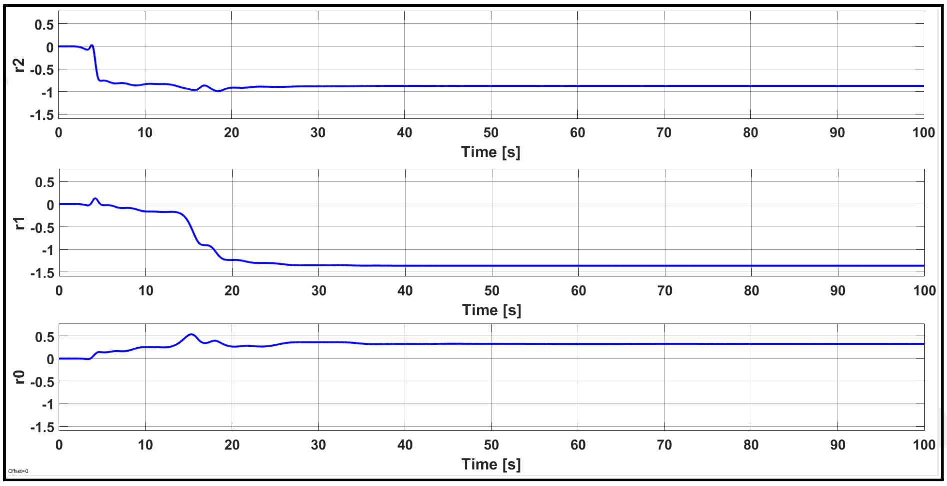

Figure 4.3 : Adaptation of R polynomial parameters.

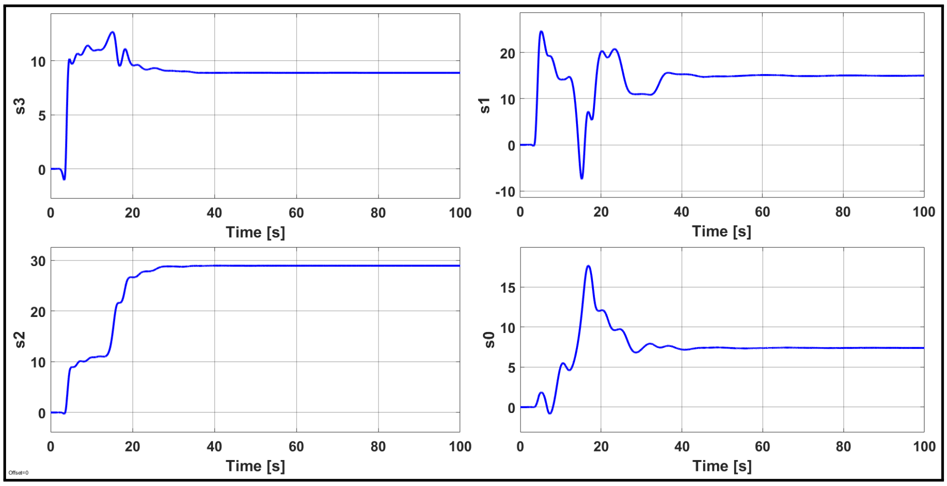

Figure 4.4 : Adaptation of S polynomial parameters.

maximum value in the transient state being approximately 0.27 meters. In contrast, the steering angle δ applied to the system, which is the control signal $u(t)$, is obtained as

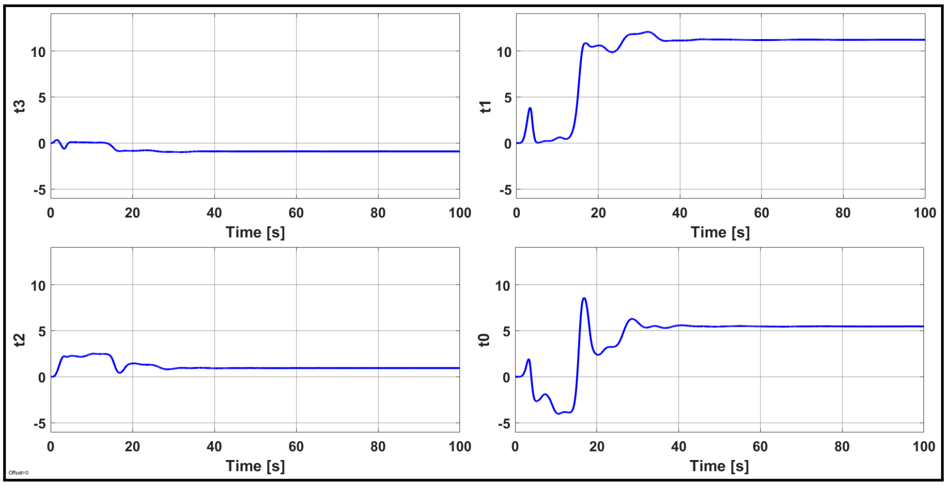

Figure 4.5 : Adaptation of T polynomial parameters.

in Figure 4.7. It must be noted that the fluctuation of $u(t)$ during the transient is within the ±0.03 rad range, which is a good response in terms of the desired driving comfort.

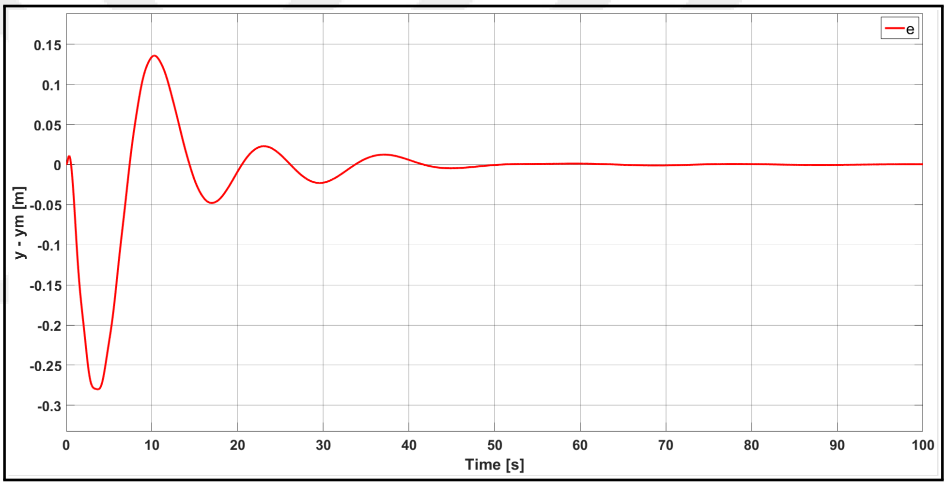

Figure 4.6 : Model following error.

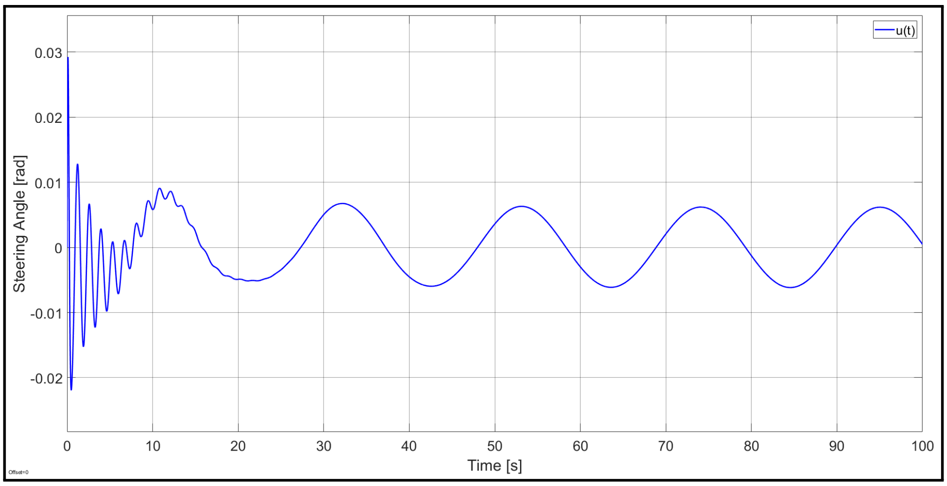

Figure 4.7 : Steering angle (controller signal).

5. CONCLUSION AND RECOMMENDATIONS

5.1 Conclusion

In this thesis, the milestones of the automotive industry and the importance of advanced driving assistance systems for autonomous driving are explained. More importantly, the vehicle models used for lane tracking control are described, and a vehicle model transfer function was derived for a large-size autonomous vehicle. Then, by explaining the adaptive control methods briefly at first, a model reference adaptive controller design with augmented error method are described, and controller design is performed for considered large-size autonomous vehicle.

During the controller design, the initial step involved selecting a reference model, and the controller structure responsible for tracking the output of this model was designed based on the minimum-degree pole placement method [26]. Subsequently, considering the scenario where the system parameters are unknown, a stability-guaranteed adaptive control system has been developed based on passivity using the augmented error method. The design of controller parameter update rules was carried out to enable this adaptation while ensuring stability.

In this case, when examining the simulation results, it is observed that the adaptation process takes approximately 10 *sn* (seconds). The steering angle exhibits oscillations within a narrower range compared to similar studies [22]. Consequently, it has been obtained in a manner that is applicable in real-time.

5.2 Recommendations and Future Works

The filter polynomials are considered to might have an influence on the transient performance. Therefore, it will be a primary step to investigate the optimal polynomial decision rules for future work. On the other hand, an additional method will be investigated to implement the control system in order to provide robustness. Since the real-time application is one of the main claims based on the result, the model will be used on a testing environment such as a model in the loop (MIL) platform.

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