INSTITUTE OF SCIENCES AND ENGINEERING

MECHATRONIC ENGINEERING PROGRAM

USING OF FUZZY PID CONTROLLER TO IMPROVE VEHICLE STABILITY FOR PLANAR AND FULL VEHICLE MODELS

By

ABDUSSALAM ALI AHMED OMAR

B.SC& M.SC MECH. ENG.

THESIS

SUBMITTED IN PARTIAL FULFILLMENT
OF THE REQUIREMENTS
FOR DEGREE OF
DOCTORATE OF PHILOSOFY

October 2017

Mechatronic Engineering

USING OF FUZZY PID CONTROLLER TO IMPROVE

VEHICLE STABILITY FOR PLANAR AND FULL VEHICLE MODELS

THESIS

by

ABDUSSALAM ALI AHMED OMAR

submitted to the Department of Mechatronic Engineering of

OKAN UNIVERSITY

IN PARTIAL FULFILLMENT OF THE REQUIREMENTS FOR DEGREE OF

DOCTORATE OF PHILOSOFY

Approved by:

Asst. Prof. Başar Özkan
Supervisor

Prof. Ramazan Nejat Tuncay
Member

Asst. Prof. Mümin Tolga Emirler
Member

Assoc.Prof. Özgür ÜSTÜN
Member

Member

October 2017

Program: Mechatronic Engineering

ABSTRACT

USING OF FUZZY PID CONTROLLER TO IMPROVE

VEHICLE STABILITY FOR PLANAR AND FULL VEHICLE MODELS

Stability control system plays a significant role in vehicle dynamics to improve the vehicle handling and achieve better stability performance**.** In this thesis, vehicle stability control system is constructed and studied by using two vehicle models which are a planar vehicle model and full vehicle model, design of a complete vehicle dynamic control system is required to achieve high performance of vehicle stability and handling, this control system structure in this thesis includes three parts which are a reference model (2 DOF vehicle model) used for controller design and yaw stability control analysis, actual vehicle model, and the controller. In this thesis, a fuzzy PID controller (FPIDC) was used and designed to improve the stability of the vehicle. The simulation results of this work show that, the structures of control systems for vehicle models used in this thesis were successful to achieve better vehicle handling and stability. To make sure this controller works well it has been tested at two cases of input steering angle which are a step signal and a lane change maneuver.

Keywords: Vehicle model, Simulation, vehicle stability, Fuzzy PID controller (FPIDC), sideslip angle, yaw rate.

ÖZET

BULANIK PID KONTROLÖR KULLANARAK DÜZLEMSEL ARAÇ MODELİ VE TAM ARAÇ MODELİ İÇİN ARAÇ KARARLILIĞININ GELİŞTİRİLMESİ

Kararlılık kontrol sistemleri, araç dinamiğinde, araç yol tutuşu ve kararlılık performansının iyileştirilmesi için önemli bir rol oynar. Bu tezde, araç kararlılık kontrol sistemleri, düzlemsel araç modeli ve tam araç modeli kullanarak araştırılmıştır. Yüksek performanslı araç kararlılığı ve yol tutuşu elde etmek için tam bir araç dinamik kontrol sisteminin tasarlanması gerekmektedir. Bu tezde bu kontrol sisteminin yapısı, savrulma kontrol analizi ve kontrolör tasarımı için kullanılan referans modeli (2 serbestlik derecesi olan araç modeli), gerçek araç modeli ve kontrolör olarak üç parçadan oluşmaktadır. Bu çalışmada bir bulanık mantık PID kontrolör araç kararlılığını iyileştirmek için kullanılmış ve tasarlanmıştır. Çok sayıda simülasyon göstermektedir ki bu tezde kullanılan kontrol sistem yapıları başarılıdır ve araç yol tutuşu ve kararlılığını iyileştirmektedir. Kontrol sisteminin iyi çalıştığından emin olmak için dümenleme girişi olarak basamak ve şerit değişimi manevraları test edilmiştir.

Anahtar Kelimeler: Araç simülasyonu, Bulanık PID kontrolör, taşıt kararlılığı, savrulma oranı, yana kayma açısı

Dedicated to my Mother, my father, my wife Aisha , my Sisters, my brothers and all my friends .

ACKNOWLEDGMENT

In the name of God, I would like to express my heartfelt to the almighty, who is the light of my life for giving me the will, strength, patience and the perseverance to pursue and the joy that I needed to work hard and complete my PhD degree.

Enthusiastically, I would like to express my gratitude and appreciation to my dear advisor **Dr. Başar Özkan** for them invaluable guidance, dedicated efforts, supervision and continuous support throughout the study. This research could not have been completed without his help. Sincere thanks and appreciation also go to my sponsor "Ministry of Higher Education in Libya" for financing my study that has enabled me to complete this thesis.

TABLE OF CONTENTS

LIST OF TABLES

LIST OF FIGURES

SYMBOLS

a, b	The displacement of the COG of the vehicle to both of front and rear axle.
c_i	The suspension damper coefficient at each wheel.
C_{af}	The longitudinal stiffness of the front wheel.
C_{ar}	The lateral stiffness of the rear wheel.
d	Half of the wheel track.
f_1, f_2	Front active suspension control force.
f_3, f_4	Rear active suspension control force.
f_r	rolling resistance coefficient.
F_x^{fL}, F_y^{fL}, F_x^{fR}, F_y^{fR}	The force components for the front tires.
F_x^{rL}, F_y^{rL}, F_x^{rR}, F_y^{rR}	The force components for the rear tires.
F_{z1}, F_{z2}	The total suspension force which acting on the front sprung mass.
F_{z3}, F_{z4}	The total suspension force which acting on the rear sprung mass.
g	acceleration due to gravity.
K_P	Proportional gain.
K_I	Integral gain.
K_D	Derivative gain.
h	The vertical distance between the roll center and the COG of the sprung mass.
I_w	The tire moment of inertia about the spin axis.
I_x	The roll moment of inertia (sprung mass).
I_y	The pitch moment of inertia (sprung mass).

I_z	The yaw moment of inertia of the sprung mass.
I_{xz}	The product of inertia of the vehicle body mass about the yaw and roll axes.
k_{af}	The anti-roll bars stiffness for the front suspension.
k_{ar}	The anti-roll bars stiffness for the rear suspension.
k_{si}	The suspension stiffness at tire *i*.
k_{ti}	The tire stiffness at tire *i*.
m	The total vehicle mass.
m_s	The vehicle body or sprung mass.
m_{ui}	The unsprung mass at each wheel.
R_w	The tire effective radius.
PB	Proportional band.
T_i	Applied torque at wheel *i*.
T_I	Integral time constant.
v	vehicle speed.
v_x	The vehicle longitudinal velocity.
v_y	The lateral velocity of the vehicle.
Z_{gi}	The excitation of the road.
Z_{si}	The sprung mass displacement due to vertical direction.
Z_{ui}	The unsprung mass displacement due to vertical direction.
β	The vehicle sideslip angle at the COG.
β_d	The desired vehicle sideslip angle.

δ_f	The front steering angle.
ϕ	The roll angle.
θ	The pitch angle.
w_i	The angular velocity at each tire.
w_z	The vehicle yaw rate.
w_{zd}	The desired vehicle yaw rate.
α_{ij}	The slip angle each wheel.
μ	The coefficient of road-tire friction.
σ_x	The tire longitudinal slip ratio.

ABBREVIATIONS

4WS	FOUR WHEEL STEERING
ASS	ACTIVE SUSPENSION SYSTEM
2DOF	TWO DEGREE OF FREEDOM
FIS	FUZZY INFERENCE SYSTEM
FLC	FUZZY LOGIC CONTROL
FPIDC	FUZZY PID CONTROLLER
LQ	LINEAR QUADRATIC
LQR	LINEAR QUADRATIC REGULATOR
LQG	LINEAR QUADRATIC GAUSSIAN
EV	ELECTRIC VEHICLE
COG	CENTER OF GRAVITY
VDC	VEHICLE DYNAMIC CONTROL

I. INTRODUCTION

1.1. Introduction

In vehicle dynamic control, controlling the lateral dynamic control motion is very important where it will determine the control and stabilization of the vehicle. One of the prominent approaches that are reported in the literature for the lateral dynamics control is a system of the stability of yaw control. To make a successful control system design, it is essential to determine an appropriate design elements of the system of yaw stabilization and control. In this thesis, parts of yaw stabilization control system, that are vehicle dynamic models, control objectives, active chassis control, and its control strategies as depicted in the figure below.

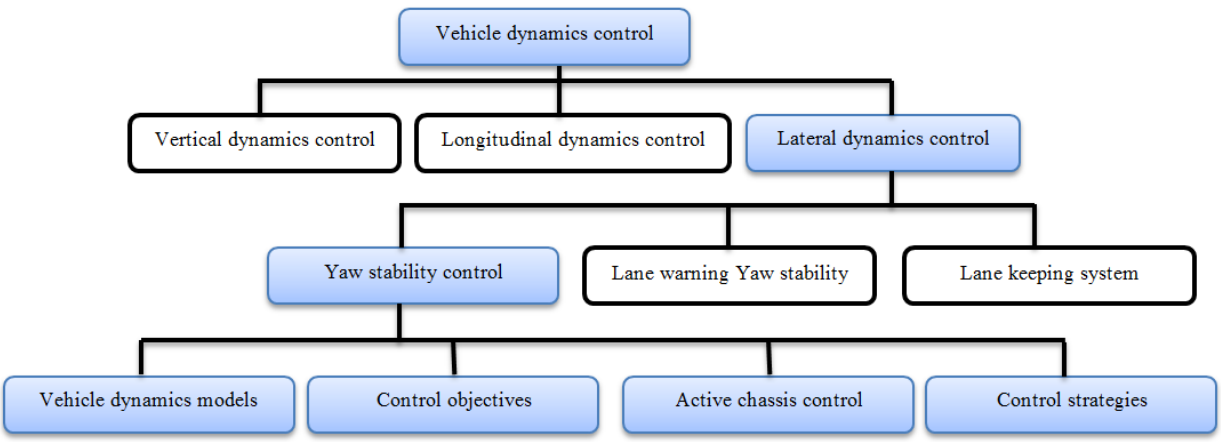


Figure 1.1. Yaw stabilization control system for lateral vehicle dynamic.[1]

In order to design, analyze, and examine design the control system for yaw stabilization control, some of vehicle dynamics models are essential where the mathematical model of vehicle dynamic motion is completed based on Newton's 2nd law that describes the moments and forces which acting on the vehicle body and wheels. In general, there are two kinds of models for vehicle dynamic, that are, model of non-linear vehicle (might be 3DOF, 7 DOF,8 DOF, and 14 DOF) and linearized vehicle model (Two degrees of freedom vehicle model) as obtained in figure 1.2.

In this thesis, two vehicle dynamic models will be used which are planar vehicle model and full vehicle model with four in-wheel motors will be obtained to study vehicle handling, stabilization, and comfort.

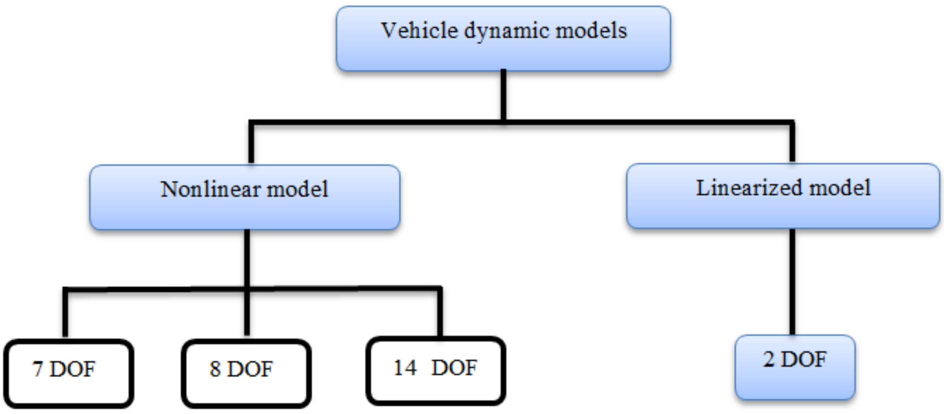


Figure 1.2: Vehicle dynamic model. [1]

1.2. Aim of the Thesis

A perfect vehicle dynamic model is desired to represent the performance of the car in order to implement and design accurate vehicle control system such as yaw rate control, and handling high comfort for a passenger car etc. The main goal of this work is construct complete vehicle model by using Matlab/ Simulink contains 2-DOF vehicle model as desired or reference model and non-linear vehicle models (Two nonlinear vehicle models will be used which are planar vehicle model and full vehicle model) , and controller . In this work Fuzzy PID controller (FPIDC) will be used.

1.3. Structure of the Thesis

This introduction concludes with a brief outline of the thesis chapters. The outline of this thesis shown in figure 1.3.

Chapter one : includes the introduction and the contributions of this work.

Chapter two: Literature survey and the Background of the subject. Also it includes discussing published literature on the vehicle models and control strategies.

Chapter three: Vehicle models.

This chapter presents fundamentals of vehicle dynamics by introducing vehicle models and tire model which have been widely adopted for vehicle motion control. This helps to get basic of what parameters and states of a vehicle are important in vehicle motion control. This chapter divided to four sections: (1) vehicle planar model (2) full vehicle model (3) linear single track model to design the controller (4) tire model and dynamic.

Chapter four : Principles of PID and Fuzzy PID controllers (FPIDC).

This chapter presents some principles of PID and Fuzzy PID controllers. In the thesis design Fuzzy PID controller (FPIDC) is important to get better vehicle stabilization. This controller can tune and adjust the three equivalent parts of the PID controller (proportional P, integral I and the derivative D control elements). The design steps of FPIDC will be summarized in this chapter.

Chapter five: Control strategy and simulation results.

This chapter aims to create two control systems for both of planar vehicle model and full vehicle model. Every control system consists of three models which are: the non-linear model (actual vehicle), the reference model (Bicycle model), and the controller model (FPIDC). A structures of vehicle models with the controller are explained in this chapter.

Chapter six: Conclusion and future work.

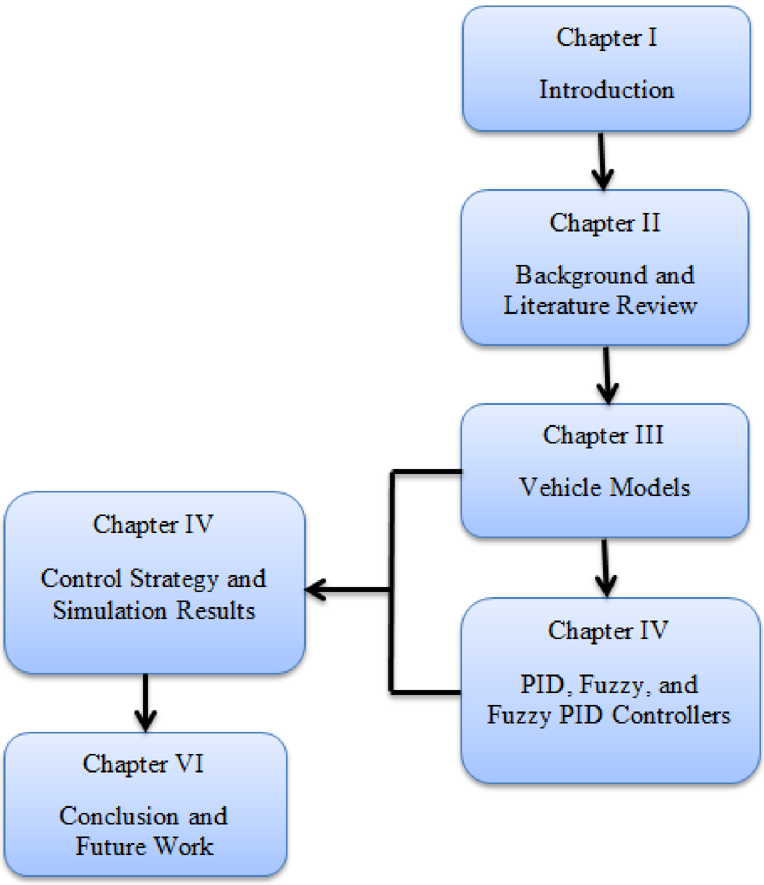


Figure 1.3. Thesis Outline.

1.4. Contributions Overview

Due to all requirements needed that PhD work enhance the state-of-the-art of a knowledge in the thesis field, it is important and necessary to clearly enumerate the predictable the significant contributions of the thesis which can be listed and summarized as follow:

1. According to our literature survey of trends of research in the topic, a lot of researchers had studied the vehicles dynamic control VDC, stabilization, comfort of passengers, and handling. Every work gave a different results and practical design steps for estimation parameters such as vehicle lateral acceleration, yaw rate, wheel sideslip angle and, but most of researchers used just vehicle planar models in their studies with many control methods that means they didn't take into consideration pitch, heave, and roll movements of sprung mass. So, this thesis gives simulation results for both of full and planar car models.

2. The main contribution of this work is using the fuzzy PID controller (FPIDC) with full car model to get better stability and performance at two cases of input steering angle which are a step signal and a lane change maneuver. This controller was not used previously, it was used only in the case of planar car models not in the case of full vehicle models.

The goal of this thesis will be the evolution and improve the stabilization of vehicles with in-wheel motors. These contributions are presented to be unique work which improves significantly to state-of-the-art in vehicle dynamic field and control.

II. BACKGROUND AND LITERATURE SURVEY

2.1. Introduction

During several years in the past and present, researchers and designers in vehicle and automotive engineering are working for the developments of a state-of-the-art car with a view to providing better ride comfort, handling and stabilization characteristics and reliable operation. The focus of their work seems to be on trying to find good solutions to improve the above characteristics using different vehicle models (Quarter, half and full vehicle models) and different control strategies.

Before developing the methodology to reach the aims of the thesis, it is important and necessary to evaluate and research the past works in this topic. The literature review of this work can be divided into four categories: suspension system which must be examined and discussed when studying the complete vehicle model, vehicle dynamic models, tire models and some control methods used to improve the vehicles' stabilization and road handling. From the literature. All of the categories above and their relevance to this topic will be summarized and analyzed.

Finally, because the requirements which PhD work enhance the state-of-the-art of realization in the field, it is important to discuss clearly about the predictable the main contributions of the thesis.

2.2. Review of Suspension System

It is known that, the suspension system is a major part in old and modern vehicles and plays a key and important role in the performance and handling of vehicles performance, like vehicle stabilization improvement ,handling and ride comfort. The four main tasks of the suspension systems are mentioned [2] : one is to separate a vehicle body to get better driving and ride quality, and the second one is to keep good road holding, and the third one is to provide good handling, and the last one is to support the vehicle static weight.

Generally, three known types of suspension system namely as ; active suspension, passive suspension, and semi- active suspension that have been widely investigated by many researchers with different techniques and algorithms [3]. In all types of vehicle conventional suspension systems have two basic elements. These elements are dampers and springs. In any vehicle suspension system the function of the spring is supporting the static weight. The damper dissipates the energy of vibration and controls any road input transmitted to the car.

A passive suspension system has fixed characteristics, where the features of the parts (dampers & springs) are fixed. The features are specified by the suspension system designers depending on the goals of the intended applications and the design. In the active suspension, a force actuator replaces either the passive damper or both the spring and passive damper. This force actuator has the ability to dissipate or add action from the system, whereas a damper can just dissipate the energy. In contrast, the force actuator in an active suspension system could apply the independent force of the relative speed or displacement through suspension.

Main differences between passive and active suspension systems are discussed by [4, 5]. They mentioned that the passive system is a control system with open loop. It just designed to enhance on specific conditions. Passive Suspension systems are fixed, and no ability to be changed or adjusted by any part of the system. The passive suspension problem is if it designed as too damped a suspension it will transmit a lot of input of the road or throw the vehicle in relation to the unevenness of the road. So, if it lightly damped and soft suspension it will give reduction in the stability of the vehicle when turns or when changing path or it will swing the vehicle. So , the road profile is the main key determinant of the passive suspension system. Alternatively, a better performance can be achieved from an active suspension system using a force actuator, in a closed loop control system. The actuator force is added inside the system, a mechanical part that is controlled by the controller.

2.3. Review of Vehicle Model and Control Strategies

When studying ride safety, stabilization and the handling capabilities of a vehicle, three types of vehicle suspension models can be used which are divided into quarter , half and full vehicle models. The quarter and half vehicle models are the ones most commonly used for passenger cars. These have 2 DOF and 4 DOF respectively. Since the reduced number of DOFs certain information about the vehicle performance such as pith and roll information is unobtainable from these two models. The half vehicle model loses roll information and the quarter vehicle model loses both of pitch and roll information [6].

For more complex study and more information about ride safety, passengers comfort, stabilization and the handling behaviors of the vehicle, a full model must be used. In this case the basic modeling is still the same as quarter and half vehicle models, but there is additional consideration regarding the sprung mass roll, pitch or heave, yaw of the car, lateral and longitudinal motion, rotation movement and vertical movement of the four wheels, all of which need to be quantified.

According to our research trends survey in the topic, a lot of researchers had studied the control of the car dynamics, stabilization, handling, and comfort of passengers. Every work gave a good functional results for exact estimation parameters such as lateral acceleration, tire sideslip angle, and car yaw rate but most of researchers used just vehicle planar models in their studies with many control methods that means they didn't take into consideration pitch, heave, and roll motions of the car body or the sprung mass. So in the next part of this literature survey , we will note that studying ride safety , stabilization and the handling capabilities of a vehicle can be done by using a planar vehicle model or by additional use of a full vehicle model including suspension parts with different control strategies by many researchers results will obtained.

Starting with planar vehicle models, there are many published literature surveys concerning vehicle stabilization and handling using planar model with different control methods such as sliding mode control, PID control, Sky-hook, Fuzzy Logic Control, H_∞ control, and LQ/LQG control, etc.

[7, 8] discuss the vehicle control system stabilization depending on yaw torque and body dimensions to study side-slip angle in electric vehicles. They used a planar model of vehicle which is displayed in figure 2.1 and sliding mode control technique which is displayed in figure 2.2 to describe vehicle stabilization performance. Researchers prove that the design of controller using sliding mode technique for planar vehicle model made the car has response better than without the controller. The yaw rate and the body sideslip angle of the vehicle can track the desirable model, and improve the car stabilization.

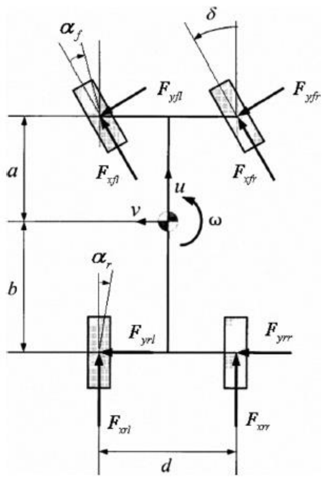


Figure 2.1. Vehicle planar motion model. [7]

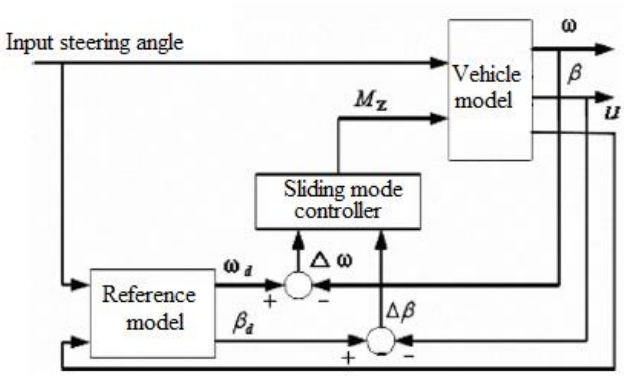


Figure 2.2. Control system diagram of vehicle stability including sliding mode controller. [7]

Improving and preserving of four wheel electric cars handling and stability using a robust controller is presented by [9]. A yaw plane model presented to describe the movements of four wheel independently-actuated car motions. Simulation results show that the vehicle model evidence demonstrates that it is possible to achieve vehicle stabilization and the desired maneuverability by using the proposed robust control system.

[10] represents a technique for controlling the lateral stabilization of hybrid and electric cars using several in-motors that are connected to the wheels of the car. The control of independent motor torque built on PID and Linear Quadratic Control is utilized to give negative brake and positive drive torques around the yaw axis to imposing an aligning moment which delivers the functionality of the lateral stabilization.

[11] Discusses an active suspension due to the control in planar vehicle model. In this work the application of Active suspension force is done by producing a negligible impact on the chassis roll, pitch, and heave dynamics. Two of control techniques allocation are suggested for yaw rate tracking to improve the vehicle performances. Firstly a control allocation, optimal, albeit impractical problems are expressed to give a benchmark for the performances of subsequent Simplified controller. Assumptions are simplified to make and generate functional different of the controller. Secondly a robust analyses are led to demonstrate the suboptimal system stabilization as a consequence of the simplified assumption , it presented the suboptimal control system which can robustly track the yaw rate of the car.

Another control method is map-based control for car stabilization enhancement is discussed by [12]. Performance of a car using this method is compared to the method of traditional model referenced control. In control of model referenced the sliding mode technique is applied to determine the compensated yaw rate map; on the other hand, the suggested map based control applied the compensated yaw rate map gained by the analysis of car stabilization. A Pacejka' s tire model and 2-DOF car model and are used in the paper to estimate the suggested Map-based control technique. Results from simulations demonstrate that the suggested Map-based control technique demonstrates and improved the stability of vehicle and performance.

[13,14] discuss another control method, which is fuzzy logic control technique (FLC) for developing the stabilization and controlling behavior of a vehicle during control of motor torques and yaw rate for Electric car with four-wheel-drive (4WD). The car model discussed here is a 4WD electric car, just taking into consideration the motion the planar movement : lateral, longitudinal and yaw. The car is designed as three-degrees-of-freedom of rigid body. The roll and pitch motions are neglected. Results of this research have shown that controller used not just enhances car stabilization but also modified the car maneuverability.

Because the importance of improve the in-wheel electric cars handling and stabilization, [15] studied the yaw rate control based FPIDC (fuzzy PID controller). This paper indicated that using a fuzzy PID controller can ensure good yaw stabilization for in-wheel-motored electric vehicles. Also, in this paper a planar model with 8-DOF is used to enhance and improve the performance of vehicle.

[16] studied the car stability control system and simulation with the Fuzzy PI control technique. The model of car used in this work is the same as in figure 2.1. The stabilization control system is studied using a linear 2DOF car model. Car stabilization control strategy is built based on the yaw moment control. Compared to PI control technique , Fuzzy PI control method is able to adjust the parameters of integral (I) and proportional (P) and increase the system response. In this study, the design of Fuzzy PI Controller led to increase and improve the electric vehicle yaw stabilization. The controller used could be decomposed to the equivalent integral and proportional control parameters. The authors of this paper designed a full control

system that contains three parts: the controller, planar vehicle model (Actual vehicle), and the reference model (Bicycle model). The structure of the Simulink model with the controller used is displayed in figure 2.3.

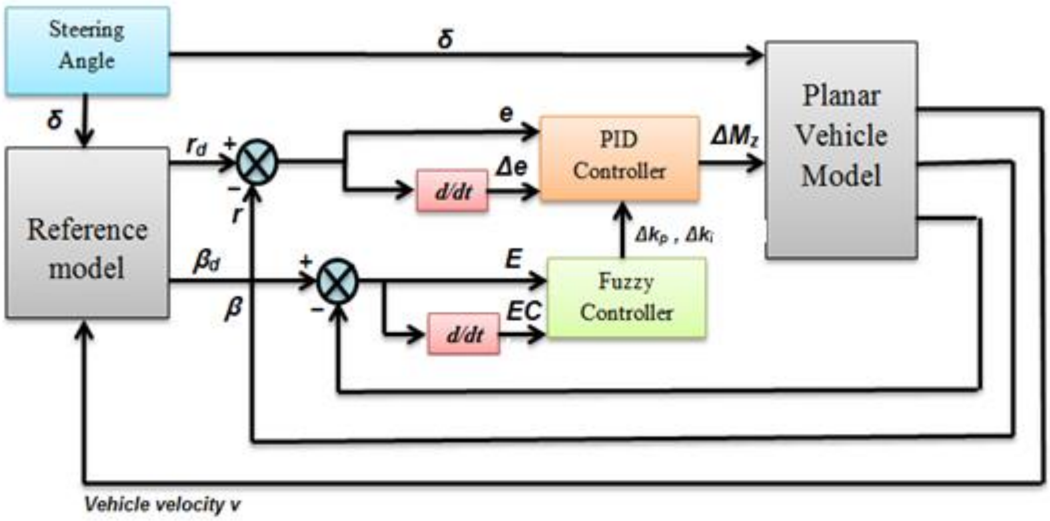


Figure 2.3. The structure of yaw moment controller. [16]

The Simulink model used in this thesis similar to the above structure but using of Fuzzy PID control method not Fuzzy PI control method and full vehicle model instead of a planar vehicle model. So it's necessary to clarify how the Fuzzy PI controller works.

A method for determining the ideal values of vehicle yaw rate and sideslip angle can be done using reference model and based on the δ which can be derived through the action of the driver. The PI controller is commonly used control algorithm in various industries because of its simplicity, ready availability and faster processing time. The critical regulation of the PI parameters is a complex design task, while it is a very simple decision-making process in case of using fuzzy controller. In this work, the comparison between the vehicle sideslip angle β and the vehicle desired value β_d is made and the desired yaw rate r_d is also compared with actual yaw rate r. Then the calculation of the sideslip angle error E and its change rate is done. The implementation of the fuzzy control is made as fellows: the first step is fuzzification of the input parameters such as the error of sideslip angle and its rate of change EC, followed by the fuzzification of the direct yaw moment M_z. Fuzzification of the input parameters transforms the actual input variables into fuzzy terms according to the selected member functions.

In this study [16], the efficiency of the vehicle stability control is achieved by developing a nonlinear planar vehicle model through the fuzzy PI control technique. The Matlab simulation results and performance of the vehicle is optimized markedly using Fuzzy PI controller. Also, the computer simulations show that the side-slip angle and vehicle yaw rate are considerably suppressed. Hence, the control system of a vehicle using a fuzzy PI control method can greatly verifies and improves the cornering capabilities and performance of the vehicle accelerated even on the low frication roads.

Other researchers and investigators have studied the stability of vehicles and handling using full vehicle models with many control techniques such as those mentioned above and they took into account additional considerations such as the rolling, pitching and bouncing motions.

[17] presented a cooperative control system for an EV body based on hierarchical, architecture using a full vehicle dynamic model which is obtained in the next figure.

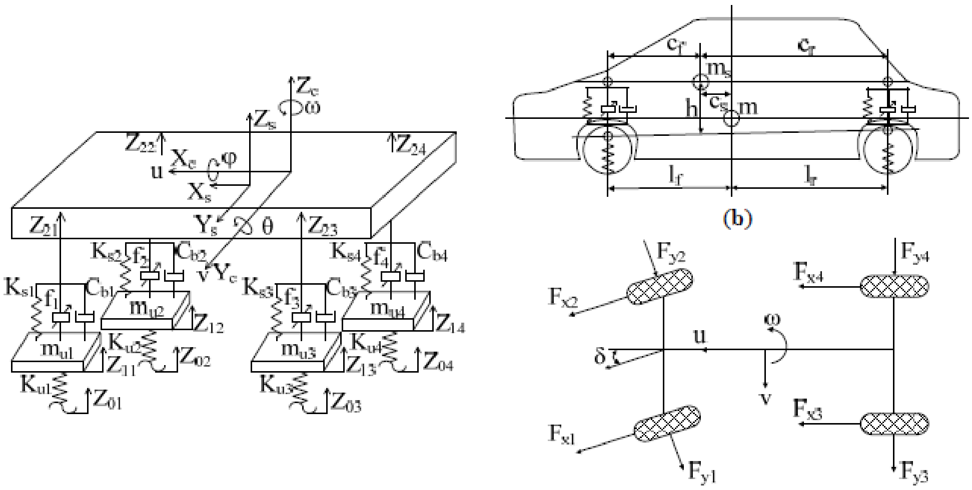


Figure 2.4. Full car model. [17]

The vehicle dynamic model includes vertical, longitudinal and lateral rotational movements; thus, the car lateral, longitudinal, and vertical dynamics should be take into account to perform and construct a full control car model. The model which used in this paper consists of fourteen DOF as illustrated in figure 2.4. with the assumption of a small vehicle angular movements.

To provide improved handling, stability and passengers comfort, the centralized controller designed for the steering system and suspension was used. The simulation results in this study show that an optimized vehicle stability, braking safety, and handling is achieved by the proposed collaborative control system.

Another control method used to analyze the full vehicle model is linear quadratic control LQR by [18]. This work reports the control performances analysis of car with active suspension by CAN (Controller Area Network) based on a model of full car. The model which be used is developed based upon four sets of suspension which constitutes 14 state variables communicates through a six CAN nodes.

The Linear Quadratic Regulator (LQR) method reduces pitch, heave and roll variation to deliver a required performance of the active suspension. The computer simulations are performed and analyzed using Matlab/Simulink with True Time toolbox.

[19] discussed full dynamic control system of a vehicle based on linear parameters varying LPV/ H infinity and flatness approaches. This article discusses an combination of two advanced vehicle controllers. The first controller is designed for control of lateral and longitudinal vehicle movements. The design takes advantages of algebraic identification methods for de-noising, numerical differentiations, and the differential flatness of non-linear systems. The second is a LPV/ H infinity controller for a suspension system which designed to adapt the vertical dynamics of a vehicle to the lateral vehicle dynamics to enhance performance objective. The computer simulation results confirmed the success of the collaborative strategy used for enhance the lateral, longitudinal and vertical dynamic and have obtained the efficiency of the required approach. The article authors proved that using the LPV technique allows them to simplify the implementation procedure.

Another paper describes another integrated dynamic control system of vehicle dynamics through coordinating of electronic stability plan and active suspension ASS by [20]. This paper investigates complete dynamics control of the vehicle using an electronic program of stability and coordinating of active suspension to improve the overall vehicle performance including comfort, handling and stability. Also, the full model of a vehicle is used in this paper to achieve the desired goal. The findings of this paper demonstrate that the suggested control system improves the multiple performance indices of the vehicle including both the ride comfort and lateral stability when compared to the equivalent nonintegrated control system.

[21, 22] described full vehicle control systems and suspension systems. The goal of all of the researchers above is to enhance vehicle stability handling and passenger ride comfort. They used some fuzzy logic techniques, which are Neuro Fuzzy control and Mamdani fuzzy logic technique.

After briefly discussing the published literature on vehicle stability, handling and comfort, we can say that the main different between this thesis and all other works mentioned above will be in the use of the fuzzy PID control method with a full vehicle model, and this difference will be a new contribution in the field of vehicle dynamics.

2.4. Review of Tire Model

The tire is the main part which interacting directly with the road. The general performance and behavior of a car is most affected by the key features of its all tires. Tires influence on the fuel consumption of a vehicle, traction, ride comfort and handling. To comprehend the importance of this, it is sufficient to remember that it is only through lateral, longitudinal, and vertical force system generated through the tires that a vehicle can maneuver [24].

When studying vehicle dynamic for handling, stabilization, and ride comfort it is necessary to calculate the acting of forces on tire which is: The longitudinal force of the tire which is generated during acceleration and deceleration and the lateral tire force which is generated when cornering.

Over past decades, many tire models have been used in the field of vehicle dynamic, like Magic formula, Brush and Dugoff' s tire models.

Magic Formula is a widely used empirical tire model that could be used to categorize long tire forces and experimental tire data to define the relationship between slip angle, lateral tire force and slip angle and self-aligning moment. There is a certain expression for this model in many references which deals with vehicle dynamic field, one of these references is [24].

Dugoff’s tire model is commonly used tire model, which is a substitute to the model by Pacejka and Sharpe to produce combined lateral and longitudinal forces generation, and to elastic analytical model developed by the tire model Fiala in1954 for the generation of lateral force.

Dugoff's model allows to calculate the tire forces under combined longitudinal and lateral tire forces generation. In this model a uniform distribution of pressure is assumed effect on the tire contact . This is simpler version than the more real parabolic distribution of pressure assumed in Sharp and Pacejka. Moreover, the model is offering one important advantage, it permits for independent amounts of the tire stiffness in the longitudinal and lateral orientations. This is a very important advantage, since the latitudinal stiffness in a tire might be completely different from longitudinal stiffness. Compared to magic tire model, Dugoff’s model is more accurate, being analytically derived from balance calculations of force. in addition, the longitudinal and lateral and forces are immediately related to the road friction coefficient in more clear equations [2].

From all notes above about the advantage of Dugoff’s tire model compared to Magic Formula Tire Model, we selected this model for tire force calculations used in this thesis and the all equations of the model will be described.

III. VEHICLE MODEL

3.1. Introduction

This chapter presents fundamentals of vehicle dynamics by introducing vehicle models and tire model which have been widely adopted for vehicle motion control. This helps to get a basic idea of what parameters and states of a vehicle are important in vehicle motion control. This chapter is separated into four sections: (1) vehicle planar model (2) full vehicle model (3) 2-degrees of freedom vehicle model (bicycle model) to design the controller and (4) Model of wheel dynamic.

3.2. **Coordinate system**The system of coordinates that is used to describes the vehicle motion as shown below in the figure.

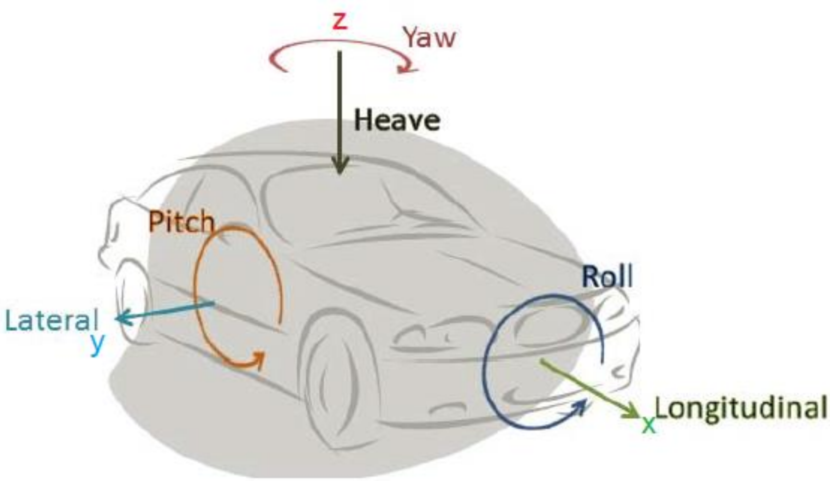


Figure 3.1. A system of coordinates of a vehicle fixed to COG. [50]

It is in according to the ISO standards, as described in ISO 8855. Using this coordinate system, the vehicle forward motion is depicted in the positive x axis and the lateral motion is depicted by the y axis, being positive when oriented to the driver's left side position, and the z axis represents the vertical motion. The rotations of the vehicle cabin are also included in this system of coordinates. The pitch rotation is defined about the y axis and the roll rotation about the x axis, while the yaw motion about the z axis.

A local coordinate system will be used independently for each tire in addition to this the system of coordinates, also according to (ISO 8855). The coordinate system for a single wheel can be obtained in figure 3.2.

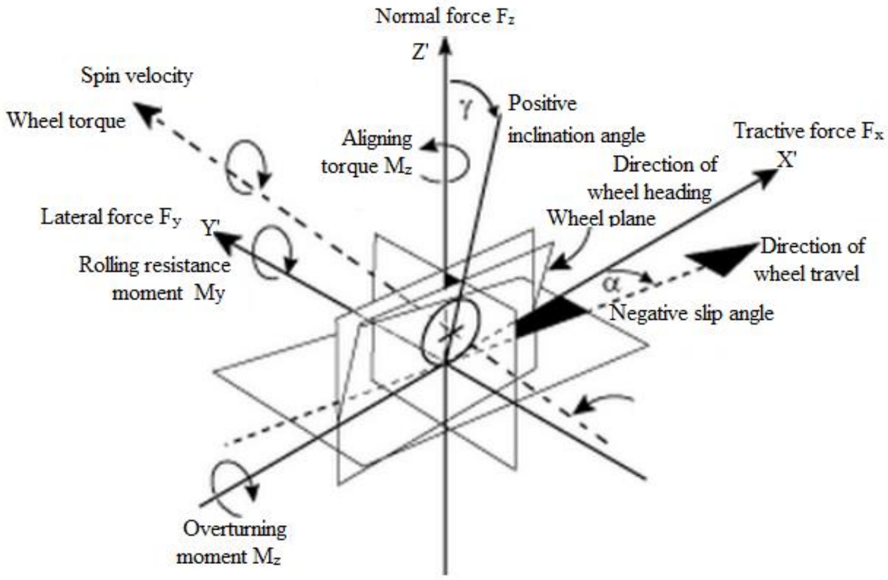


Figure 3.2. Wheel local coordinates system.[51]

3.3. Vehicle Planar Model

The vehicle planar model is formulated from the following three equations of motion of a four wheel vehicle with front steering. Figure 3.3 describes the sketch of the vehicle model and the parameters concerned. The positive x axis starts at the COG and points in the front direction of the vehicle, this direction is also indicated to as a longitudinal direction, while the y axis is corresponded to as the lateral direction and starts from the model center line. It is assumed that the front wheels have the same steering angle and the roll, pitch and bob motions are neglected.

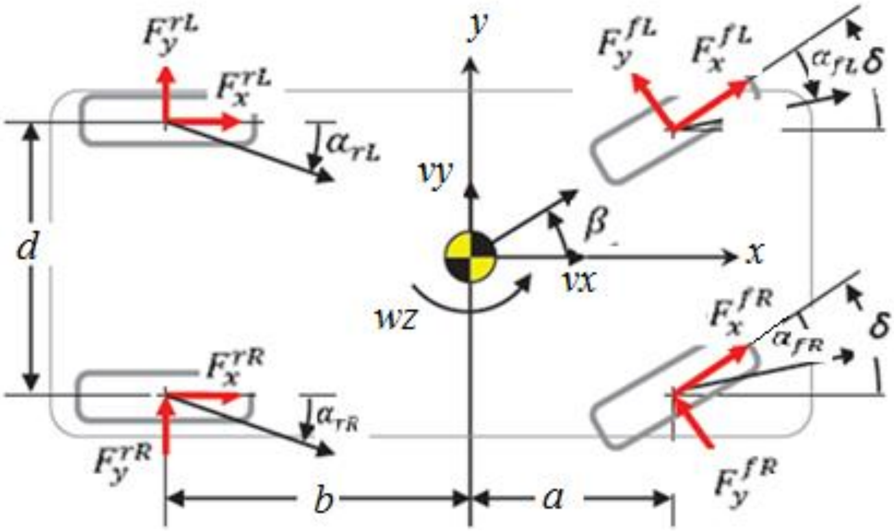


Figure 3.3. Vehicle planar motion model**.**

The mathematical equations of vehicle motions can be expressed as follows:

For yaw movement:

$$I\dot{w}_z = [a(F_x^{fR} + F_x^{fL})\sin(\delta) + a(F_y^{fR} + F_y^{fL})\cos(\delta) - b(F_y^{rL} + F_y^{rR}) + \frac{d}{2}(F_x^{fR} - F_x^{fL})\cos(\delta) + \frac{d}{2}(F_x^{rR} - F_x^{rL}) + \frac{d}{2}(F_y^{fL} - F_y^{fR})\sin(\delta)] \tag{3.1}$$

For longitudinal movement:

$$\dot{v}_x - v_y w_z = \frac{1}{m}[(F_x^{fR} + F_x^{fL})\cos(\delta) - (F_y^{fR} + F_y^{fL})\sin(\delta) + F_x^{rL} + F_x^{rR}] \tag{3.2}$$

For lateral movement:

$$\dot{v}_y + v_x w_z = \frac{1}{m}[(F_y^{fL} + F_y^{fR})\cos(\delta) + (F_x^{fL} + F_x^{fR})\sin(\delta) + F_y^{rL} + F_y^{rR}] \tag{3.3}$$

Whrer $F_x^{fL}, F_y^{fL}, F_x^{fR}, F_y^{fR}, F_x^{rL}, F_y^{rL}, F_x^{rR}$, F_y^{rR} are the components of forces for the front left tire, front right tire, rear left tire, and the rear right tire along *x* axis and *y* axis coordinates; *a* ,*b* are the displacement of the COG of the vehicle to both of front and rear axle; L_W is the displacement between left and right tires ; v_x, v_y are the car longitudinal and the car lateral velocitiy, w_z is the vehicle yaw rate, δ is the front wheel steering angle, *m* is the vehicle total mass, *I* is the vehicle moment inertia about its yaw.

The slip angle at each wheel α_{ij} is expressed and derived using the geometry of the vehicle and the vectors of wheel speed. If the velocity at each wheel road contact point is known then, it can easily derive the tire slip angle at each tire geometrically and can be expressed as follows:

$$\alpha_{fR} = \arctan\left(\frac{v_y + a.w_z}{v_x + \frac{d}{2}.w_z}\right) - \delta \tag{3.4}$$

$$\alpha_{fL} = \arctan\left(\frac{v_y + a.w_z}{v_x - \frac{d}{2}.w_z}\right) - \delta \tag{3.5}$$

$$\alpha_{rL} = \arctan\left(\frac{v_y - b.w_z}{v_x + \frac{d}{2}.w_z}\right) \tag{3.6}$$

$$\alpha_{rL} = \arctan\left(\frac{v_y - b.w_z}{v_x - \frac{d}{2}.w_z}\right) \tag{3.7}$$

The above vehicle model is analyzed and simulated using Matlab Simulink. It is assumed that the vehicle used in this case moves at a constant speed *(v_x)* = 20 m/s, the road friction coefficient is 0.4, and the vehicle receives a input steering from the tire. Firstly, the input steering will set as a step signal which have an amplitude of two degrees (0.035 radians) as illustrated in the figure below. Also, the input steering will set as a lane change maneuver with amplitude of front steering angle of 0.035 radians as obtained figure 3.5.

The performance of vehicle will be obtained and compared in this thesis using the two cases of input steering angle (a step signal and a lane change maneuver).

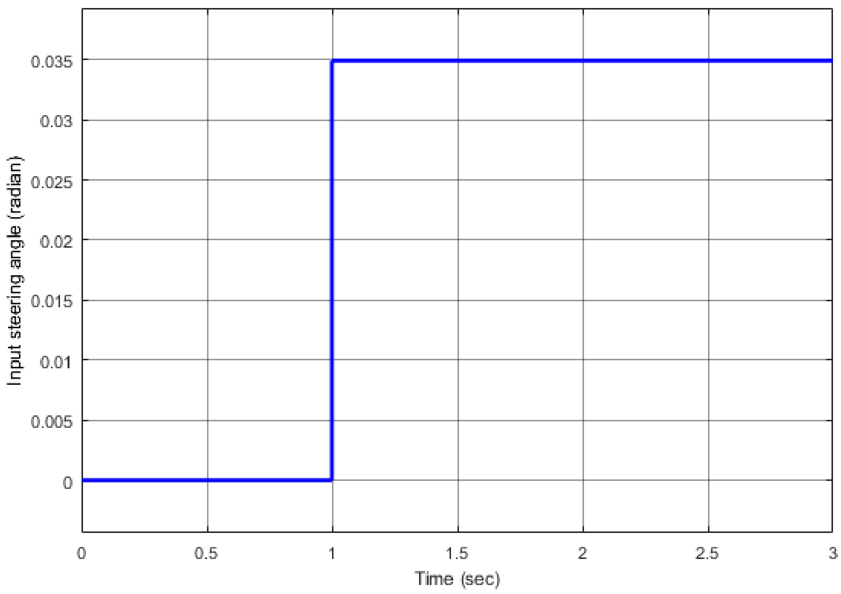


Figure 3.4. the steering input of vehicle maneuver**.**

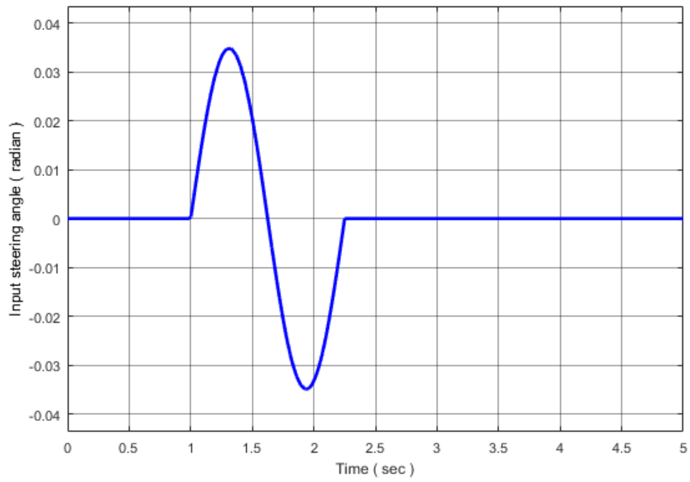


Figure 3.5. the steering input of vehicle with a lane change maneuver**.**

The following figures represent the performance of the car in the case of planar model which performed at a step signal of steering single and the lane change maneuver of the front wheels . The vehicle longitudinal velocity is obtained in figures 3.6 and 3.7 . Figures 3.8 and 3.9 display the vehicle lateral acceleration. As it can be shown clearly, the lateral acceleration reaches its maximum rapidly during start of the second two.

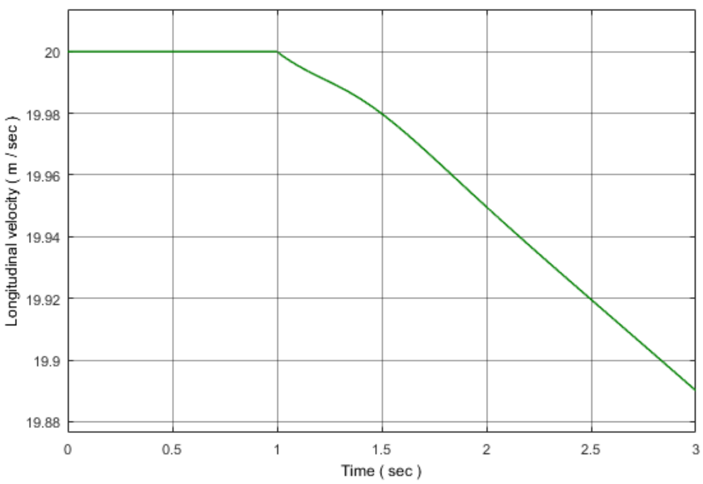


Figure 3.6. the vehicle longitudinal velocity at a step signal of steering angle

(Planar model).

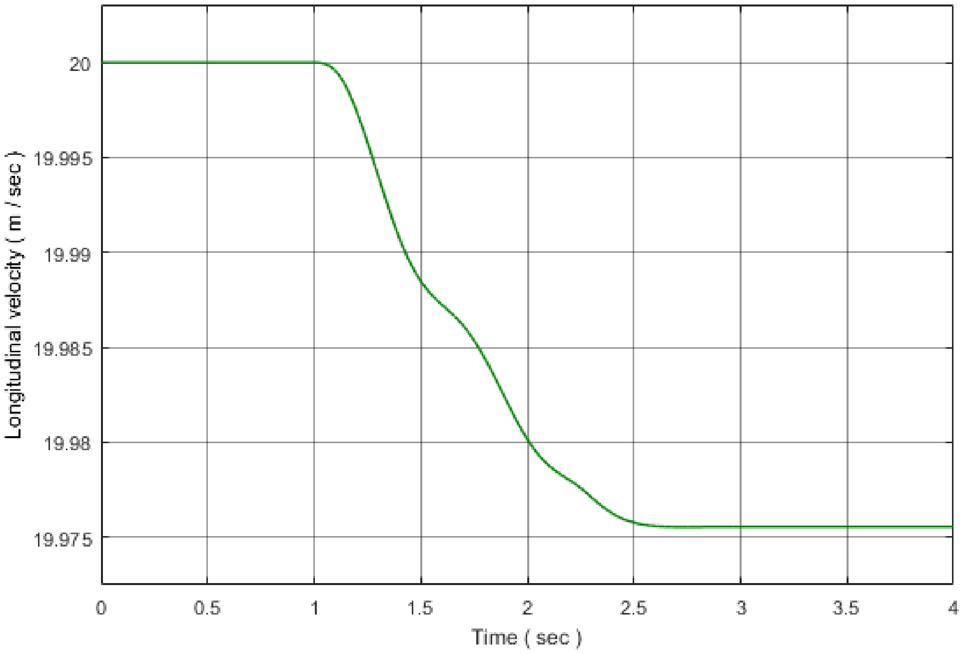


Figure 3.7. the vehicle longitudinal velocity on a lane change maneuver (Planar model).

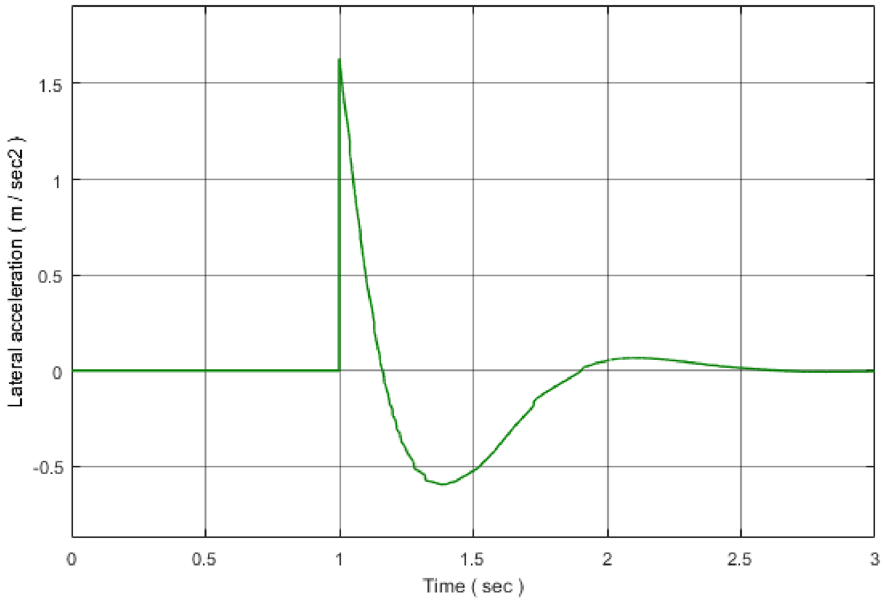


Figure 3.8. the vehicle lateral acceleration at a step signal of steering angle (Planar model).

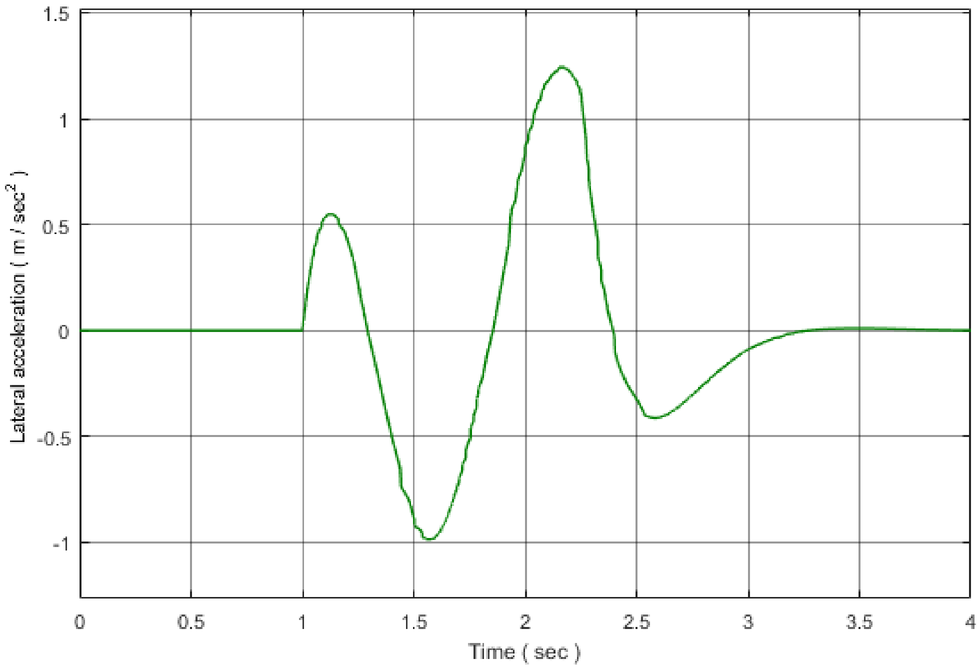


Figure 3.9. the vehicle lateral acceleration on a lane change maneuver (Planar model).

Figures 3.10 and 3.11 show the vehicle sideslip angle change with respect to time under specified conditions and figures 3.12 and 3.13 depict the yaw rate of the vehicle versus time. It is obvious that from figures, the vehicle yaw rate at a step signal of steering angle initially increases as it is expected from the steer input. The peak value of the yaw rate at a step signal of steering angle is approximately 0.135 rad/s while the peak value of the yaw rate on a lane change maneuver is about +0.12 rad/s and – 0.12 rad/s.

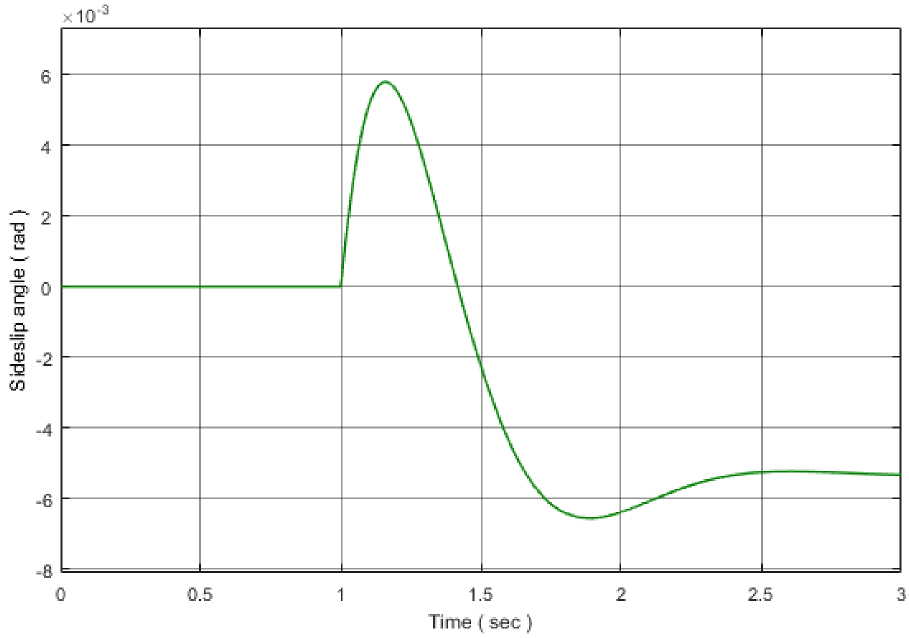


Figure 3.10. the vehicle sideslip angle at a step signal of steering angle (Planar model).

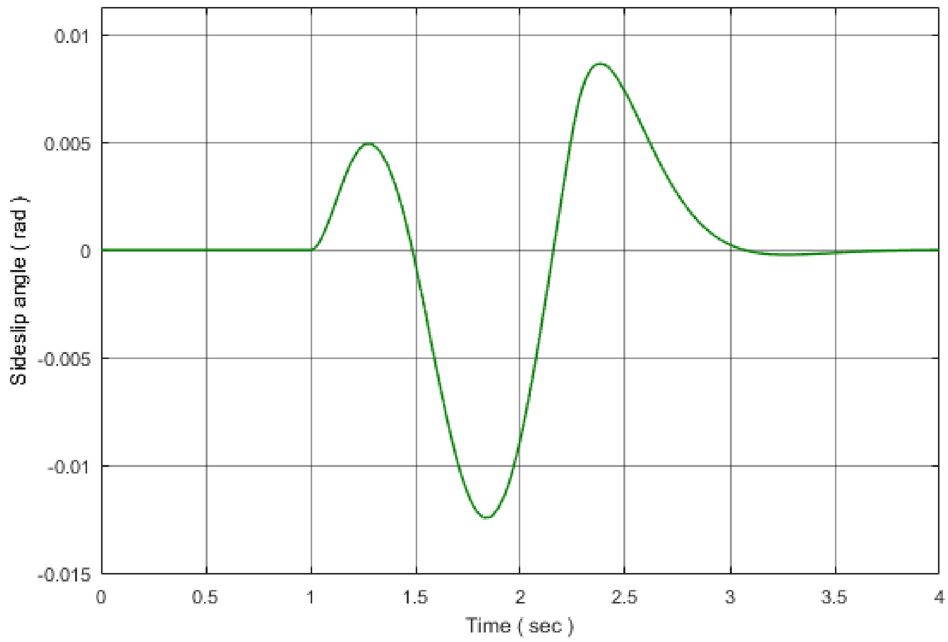


Figure 3.11. the vehicle sideslip angle on a lane change maneuver (Planar model).

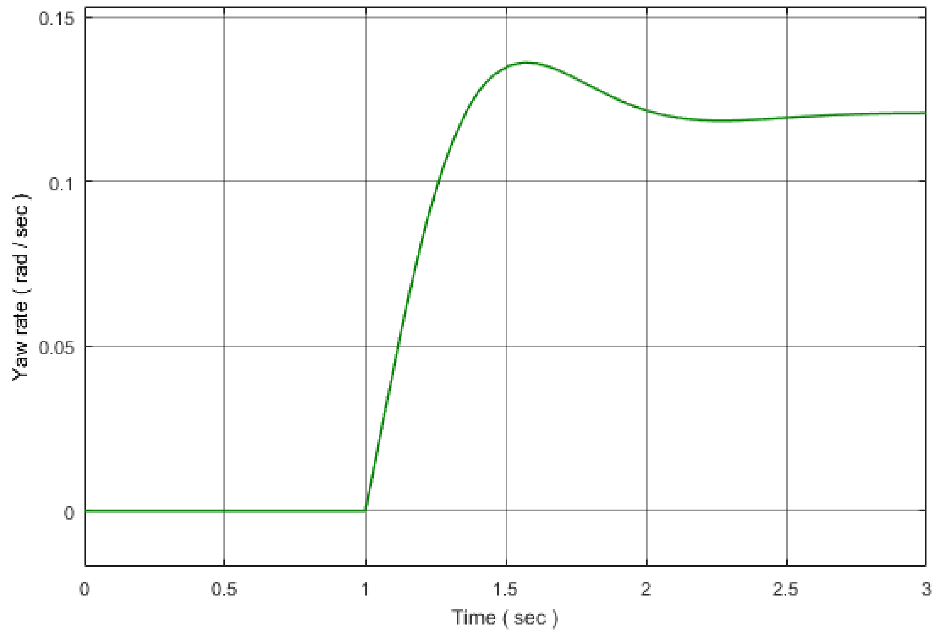


Figure 3.12. the vehicle yaw rate at a step signal of steering angle (Planar model).

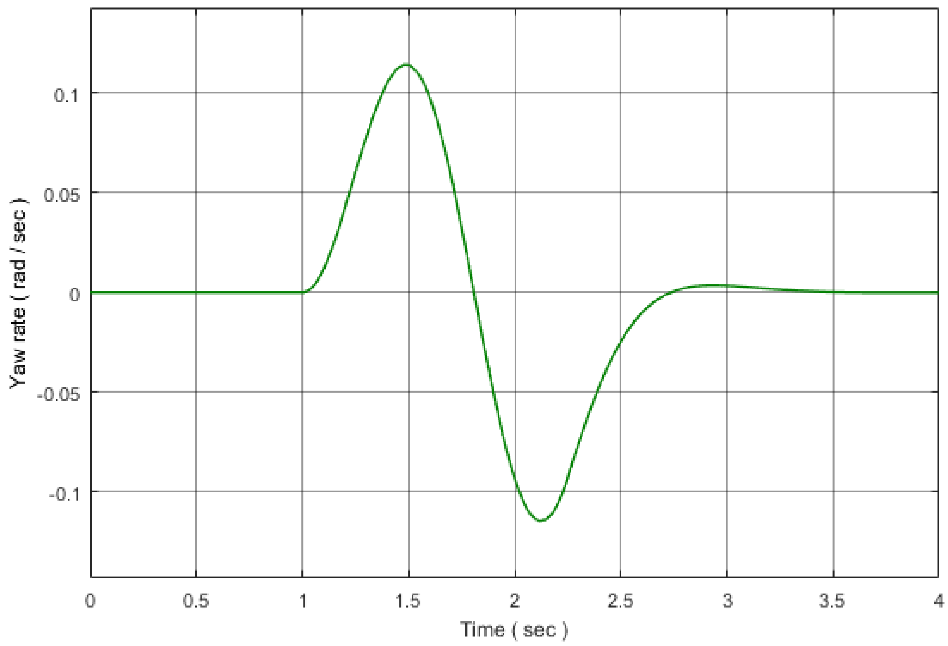
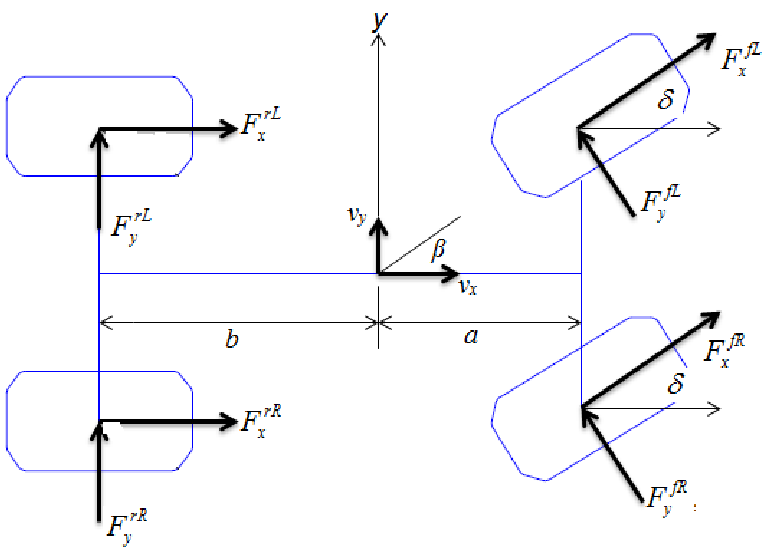


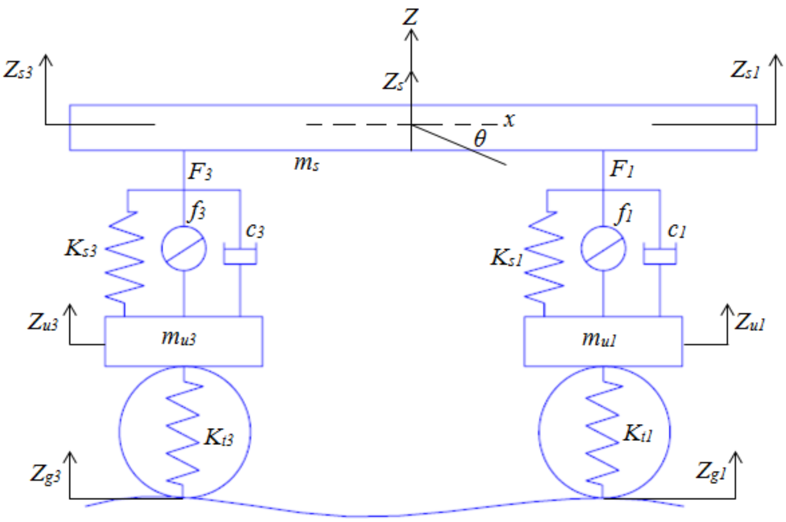
Figure 3.13. the vehicle yaw rate on a lane change maneuver (Planar model).

3.4. Full Vehicle Model

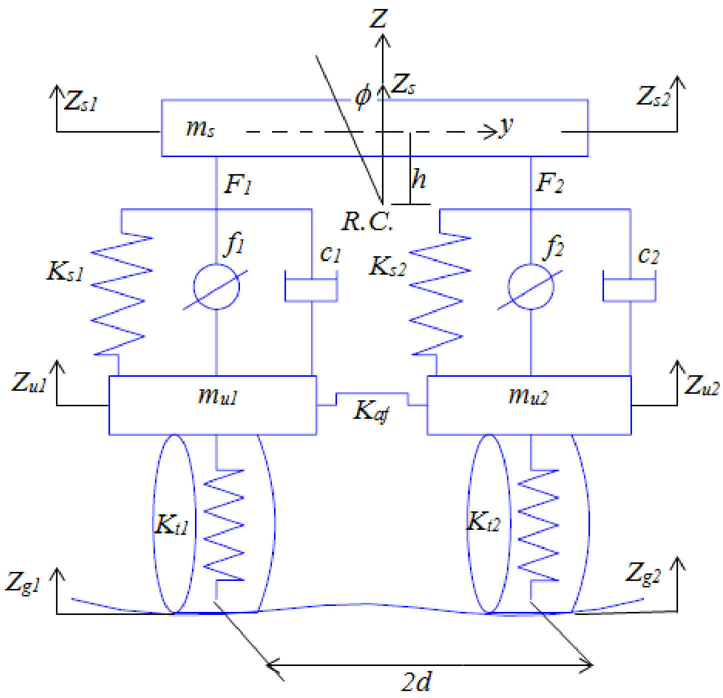
In this vehicle model, the sprung mass or body is free to roll, pitch and heave, which means that the full vehicle model dynamics involves the rotation and movement on the vertical, longitudinal and lateral directions; therefore, the vehicle, vertical, longitudinal and lateral dynamics have to take into considerations for a complete car model. The suspension pats connects the body or the sprung mass to four unsprung masses which are rear-left, rear-right, front-left, and front-right tires. They are free to bouncing vertically relative to the vehicle body. The vehicle model adopted here consists of fourteen degrees of freedom (14 DOF) as obtained in the following figure.



(a)



(b)



(c)

Figure 3.14. Full vehicle model includes: the yaw (a), pitch (b), and roll(c) motions. [49]

The mathematical equations of the vehicle motion can be written as:

For the yaw movement of the sprung mass in Figure 3.4.a

$$\dot{w}_z I_z - \ddot{\phi} I_{xz} = a(F_x^{fR}\sin(\delta) + F_y^{fR}\cos(\delta) + F_x^{fL}\sin(\delta) + F_y^{fL}\cos(\delta)) - b(F_y^{rL} + F_y^{rR}) \qquad (3.8)$$

Where w_z is the vehicle yaw rate, I_z is the sprung mass yaw moment of inertia ,a and b is the distance between the COG. of the vehicle and the rear, front axle, and ϕ is the roll angle of sprung mass.

And the differential equations of motion due to lateral and longitudinal directions can be derived as:

$$m(\dot{v}_x + v_y w_z) - m_s h \dot{w}_z \phi = [F_x^{fR}\cos(\delta) - F_y^{fR}\sin(\delta) + F_x^{fL}\cos(\delta) - F_y^{fL}\sin(\delta) + F_x^{rL} + F_x^{rR}] - f_r mg \tag{3.9}$$

Where m is total mass of the vehicle, v_y is the vehicle speed in the lateral direction, v_x is the speed of the vehicle in the x axis direction (longitudinal direction), f_r is the rolling resistance coefficient, and h is the vertical displacement between the roll center and the COG of the sprung mass.

$$m(\dot{v}_y + v_x w_z) + m_s h \ddot{\phi} = [(F_x^{fR}\sin(\delta) + F_y^{fR}\cos(\delta) + F_x^{fL}\sin(\delta) + F_y^{fL}\cos(\delta) + F_y^{rL} + F_y^{rR}] \tag{3.10}$$

The following equation describes the pitch motion of the body or sprung mass:

$$I_y \ddot{\theta} = b(F_{z3} + F_{z4}) - a(F_{z1} + F_{z2}) \tag{3.11}$$

Where $F_{z1}, F_{z2}, F_{z3}, F_{z4}$ are the total suspension forces which acting on the front and rear sprung masses, I_y is the pitch moment of inertia of sprung mass, and θ is the sprung mass pitch angle.

And for roll motion of the sprung mass:

$$I_x \ddot{\phi} + m_s(\dot{v}_y + \dot{v}_x w_z)h - I_{xz}\dot{w}_z = m_s g h \phi + d(F_{z2} + F_{z3} - F_{z1} - F_{z4}) \tag{3.12}$$

Where I_x is the sprung mass roll moment of inertia and I_{xz} is the product of inertia of the vehicle body mass about the yaw and roll axes.

The equations which describes the vertical motions of the unsprung masses sprung mass are as follows:

$$m_s\ddot{Z}_s = F_{z1} + F_{z2} + F_{z3} + F_{z4} \tag{3.13}$$

Where Z_s is the vertical displacement of the vehicle body.

$$m_{u1}\ddot{Z}_{u1} = k_{t1}(Z_{g1} - Z_{u1}) - F_{z1} \tag{3.14}$$

$$m_{u2}\ddot{Z}_{u2} = k_{t2}(Z_{g2} - Z_{u2}) - F_{z2} \tag{3.15}$$

$$m_{u3}\ddot{Z}_{u3} = k_{t3}(Z_{g3} - Z_{u3}) - F_{z3} \tag{3.16}$$

$$m_{u4}\ddot{Z}_{u4} = k_{t4}(Z_{g4} - Z_{u4}) - F_{z4} \tag{3.17}$$

m_{ui} is the unsprung mass at tire *i*, Z_{ui} is the vertical distance of unsprung masses, k_{ti} is the tire stiffness at tire *i*, and $Z_{g1}$is the road excitation.

The total suspension forces which acting on the front and rear sprung masses can be expressed as:

$$F_{z1} = k_{s1}(Z_{u1} - Z_{s1}) + c_1(\dot{Z}_{u1} - \dot{Z}_{s1}) - \frac{k_{af}}{2d}\left[\phi - \frac{(Z_{u2} - Z_{u1})}{2d}\right] + f_1 \tag{3.18}$$

$$F_{z2} = k_{s2}(Z_{u2} - Z_{s2}) + c_2(\dot{Z}_{u2} - \dot{Z}_{s2}) - \frac{k_{af}}{2d}\left[\phi - \frac{(Z_{u2} - Z_{u1})}{2d}\right] + f_2 \tag{3.19}$$

$$F_{z3} = k_{s3}(Z_{u3} - Z_{s3}) + c_3(\dot{Z}_{u3} - \dot{Z}_{s3}) - \frac{k_{ar}}{2d}\left[\phi - \frac{(Z_{u3} - Z_{u4})}{2d}\right] + f_3 \tag{3.20}$$

$$F_{z4} = k_{s4}(Z_{u4} - Z_{s4}) + c_4(\dot{Z}_{u4} - \dot{Z}_{s4}) - \frac{k_{ar}}{2d}\left[\phi - \frac{(Z_{u3} - Z_{u4})}{2d}\right] + f_4 \tag{3.21}$$

Where k_{si} is the suspension stiffness at each wheel, c_i is the damper coefficient of the suspension at each wheel, k_{af} is the stiffness of anti-roll bars for the front suspension, k_{ar} is the stiffness of the anti-roll bars for the rear suspension , and f_i is the control force of front and rear active suspension controller.

It is assumed that both of the sprung mass pitch angle θ and the sprung mass roll angle ϕ have a small values, the next approximations can be achieved.

$$Z_{s1} = Z_s - a\theta - d\phi \tag{3.22}$$

$$Z_{s2} = Z_s - a\theta + d\phi \tag{3.23}$$

$$Z_{s3} = Z_s - b\theta + d\phi \tag{3.24}$$

$$Z_{s4} = Z_s + b\theta - d\phi \tag{3.25}$$

Figures 3.15 to 3.22 represent the performance of the vehicle in the case of full model at a step signal of steering single and the lane change maneuver of the front wheels. The vehicle longitudinal velocity is obtained in figures 3.15 and 3.16, while figures 3.17 and 3.18 describe the lateral acceleration behavior of the vehicle.

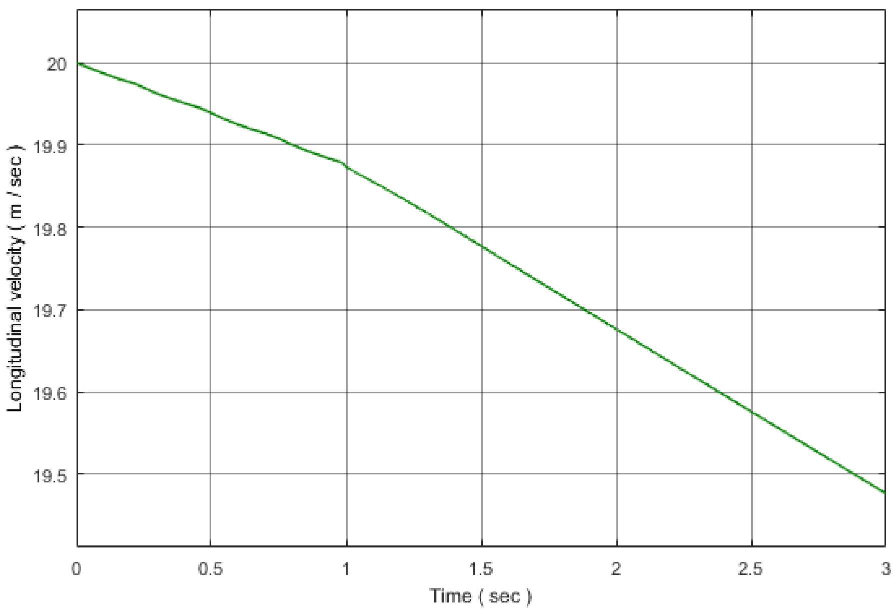


Figure 3.15. the vehicle longitudinal velocity at a step signal of steering angle (Full model).

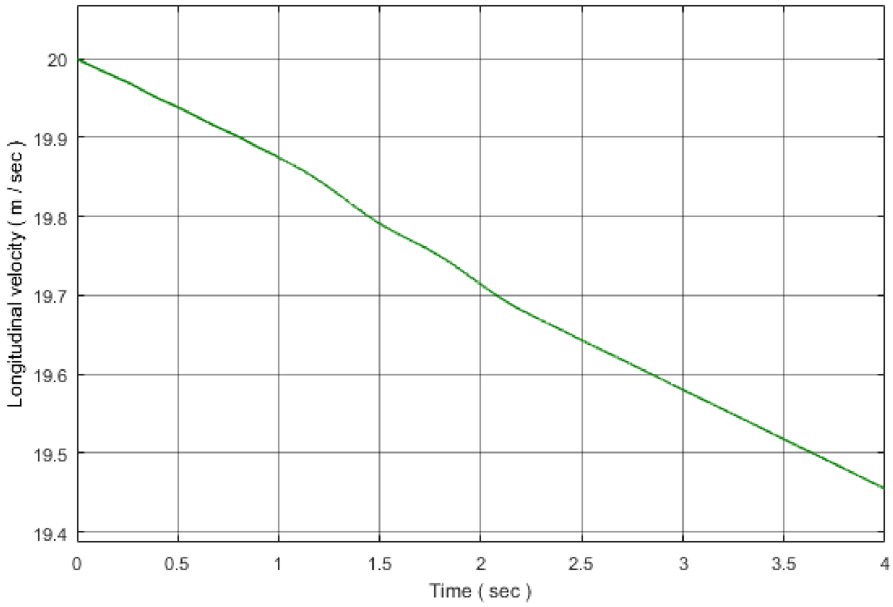


Figure 3.16. the vehicle longitudinal velocity on a lane change maneuver (Full model).

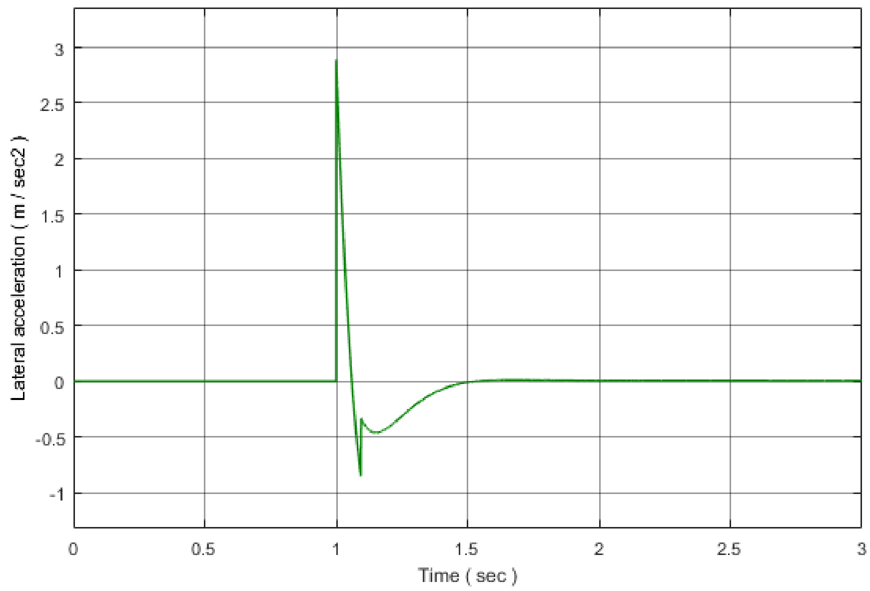


Figure 3.17. the vehicle lateral acceleration at a step signal of steering angle (Full model).

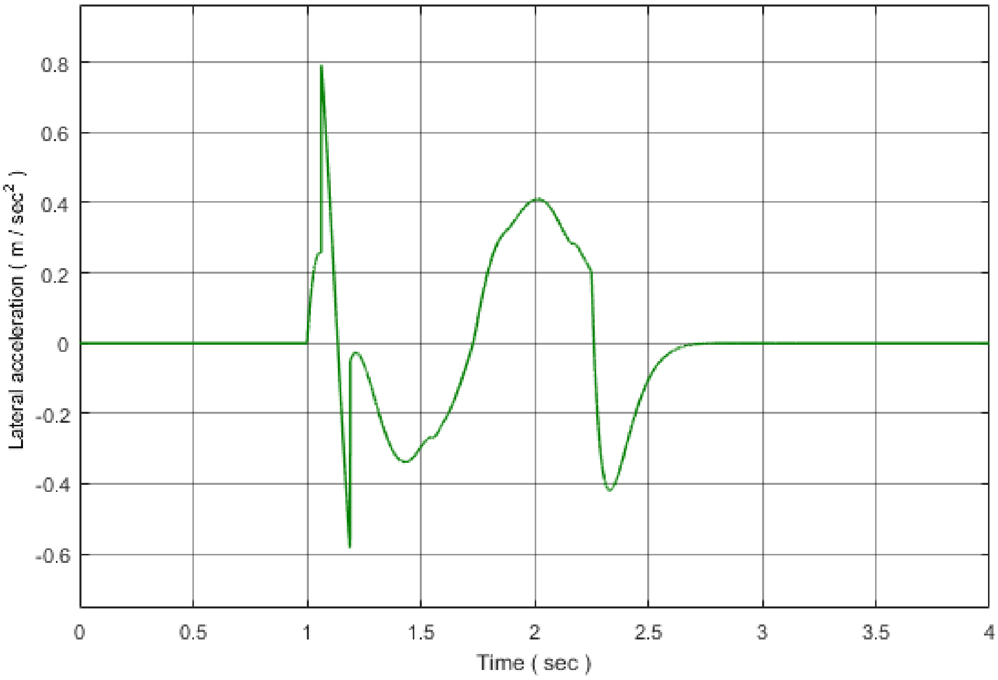


Figure 3.18. the vehicle lateral acceleration on a lane change maneuver (Full model).

Figures 3.19 and 3.20 show the vehicle sideslip angle change with respect to time under specified conditions and figures 3.21 and 3.22 depict the vehicle yaw rate versus time. As it can be shown from the figures, the yaw rate initially increases as it is expected from the steer input. The peak value of the yaw rate at a step signal of steering angle is approximately 0.22 rad/s while the peak value of the yaw rate on a lane change maneuver is about +0.2 rad/s and – 0.2 rad/s.

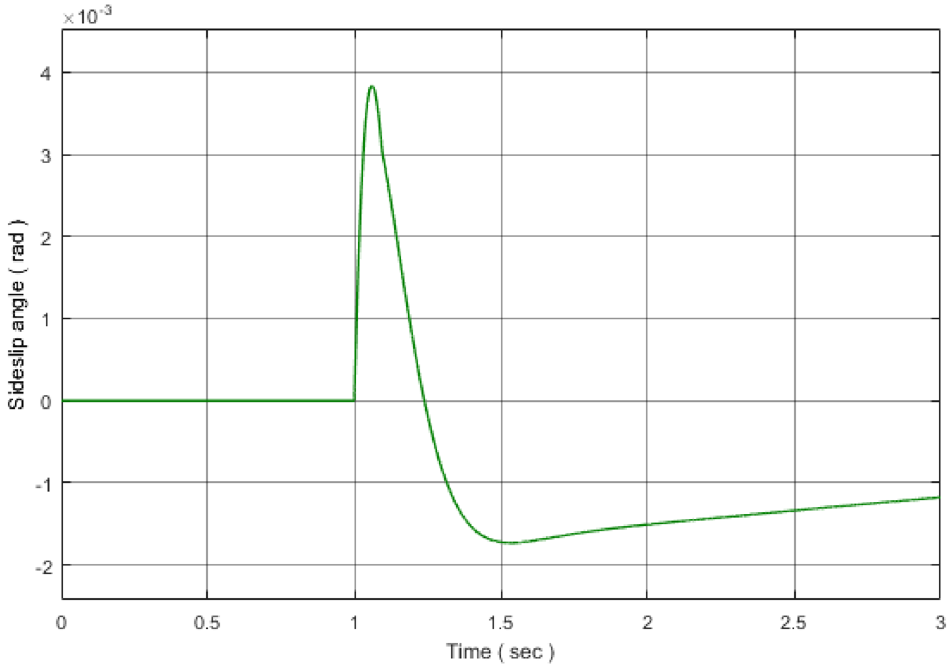


Figure 3.19. the vehicle sideslip angle at a step signal of steering angle (Full model).

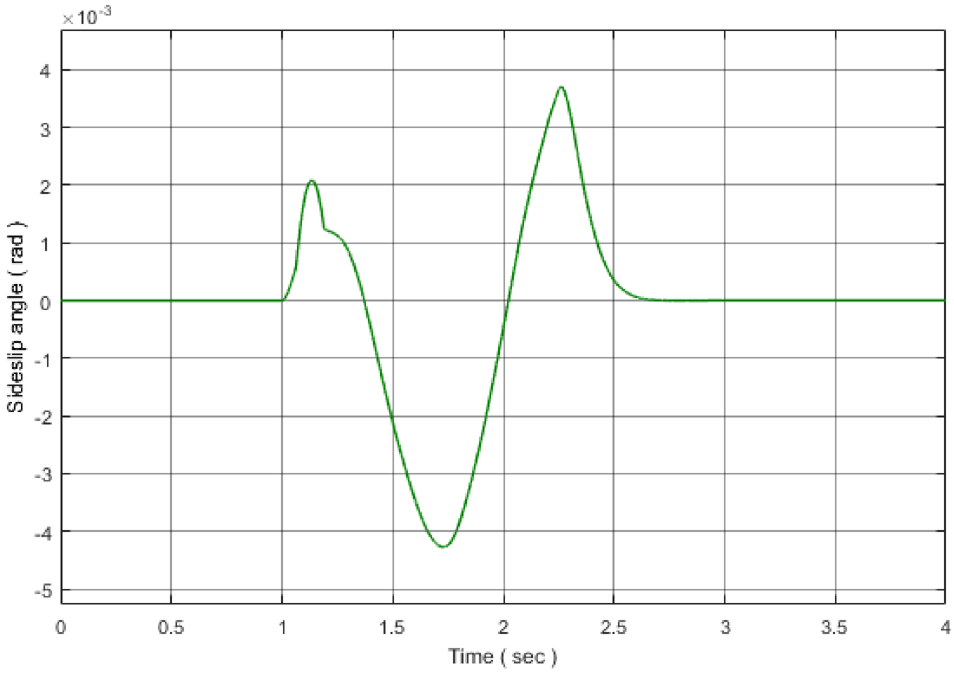


Figure 3.20. the vehicle sideslip angle on a lane change maneuver (Full model).

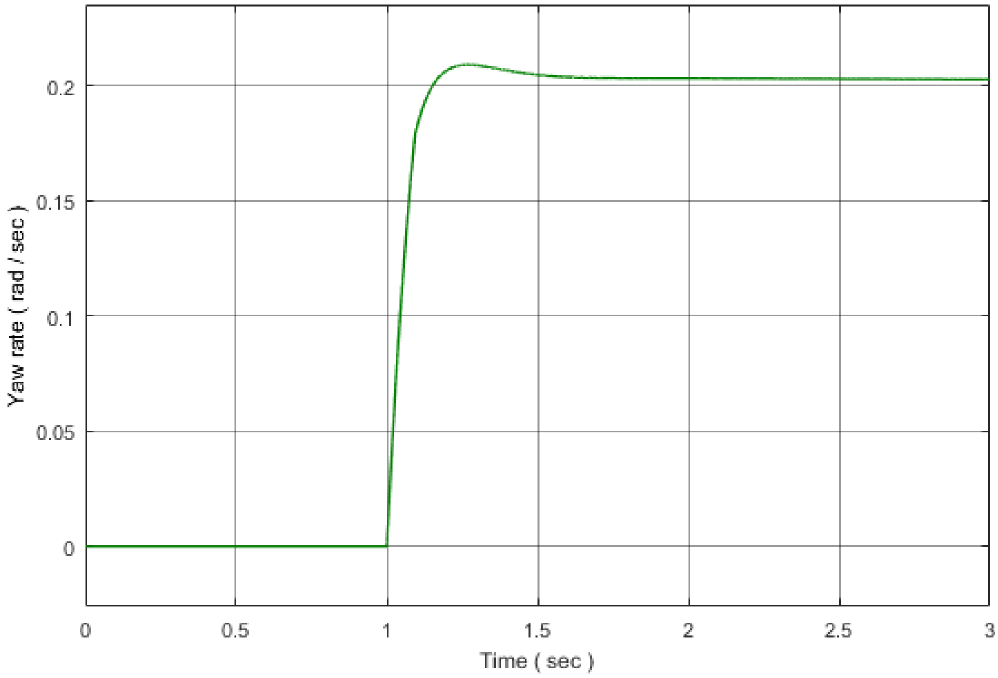


Figure 3.21. the vehicle yaw rate at a step signal of steering angle (Full model).

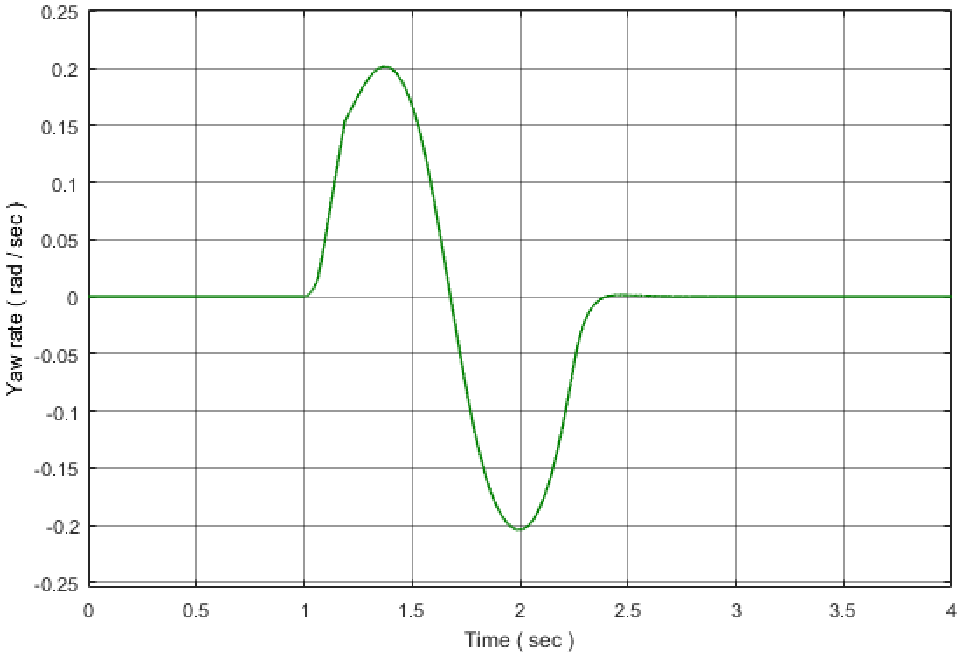


Figure 3.22. the vehicle yaw rate on a lane change maneuver (Full model).

3.5. Vehicle Reference Model

The bicycle model (reference vehicle model) as illustrated in figure 3.23 is an important part in the automotive engineering studies and vehicle dynamic field because it uses for controllers design and the analysis of yaw stability control prominently. According to some assumptions, it is possible to linearize the actual model of the vehicle (a nonlinear model), these assumptions are : the vehicle moves on plane surface or flat road (planar motion), the tires forces operate in a linear region, and the left and the right tires at the rear and front axle are placed in an unattached tire at the center line of a vehicle.

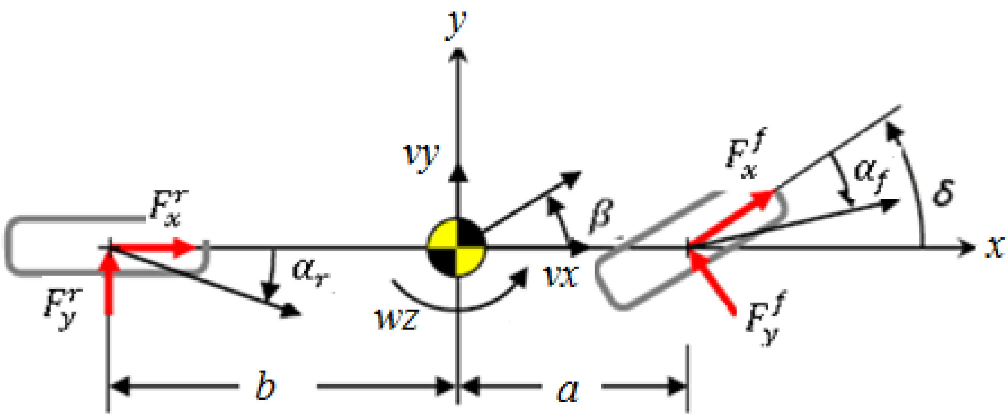


Figure 3.23. Vehicle refernce model(bicycle model).

The driver tries to control the vehicle's stability during normal and moderate cornering from the steer-ability point of view. The bicycle model gives a relationship between the stability factors and driver performance of the vehicle. Hence, the model is designed to generate a desired values of the vehicle yaw rate w_{zd} and the sideslip angle β_d at each instance according to the driver's steering angle input and the vehicle velocity while considering of a constant forward velocity.

The following equations represent the differential equations of lateral and yaw motions of the reference model [1]:

$$mv(\dot{\beta}_d - w_{zd}) = (F_y^f + F_y^r) - w_{zd} \tag{3.26}$$

$$I_z w_{zd} = a.F_y^f - b.F_y^r \tag{3.27}$$

$$F_y^f = C_{af}.\alpha_f \tag{3.28}$$

$$F_y^r = C_{ar}.\alpha_r \tag{3.29}$$

$$\alpha_f = \delta - \beta_d - \frac{a.w_{zd}}{v} \tag{3.30}$$

$$\alpha_r = -\beta_d + \frac{b.w_{zd}}{v} \tag{3.31}$$

Where:

w_{zd}, β_d are the desired vehicle yaw rate and desired vehicle sideslip angle.

C_{af}, C_{ar} are the longitudinal and the lateral stiffness of front wheel and rear wheel.

a, *b* are the displacement of the COG of the vehicle to both of front and rear axle.

The simulation result for the vehicle reference model which represents the desired vehicle yaw rate is obtained in figures 3.24 and 3.25.

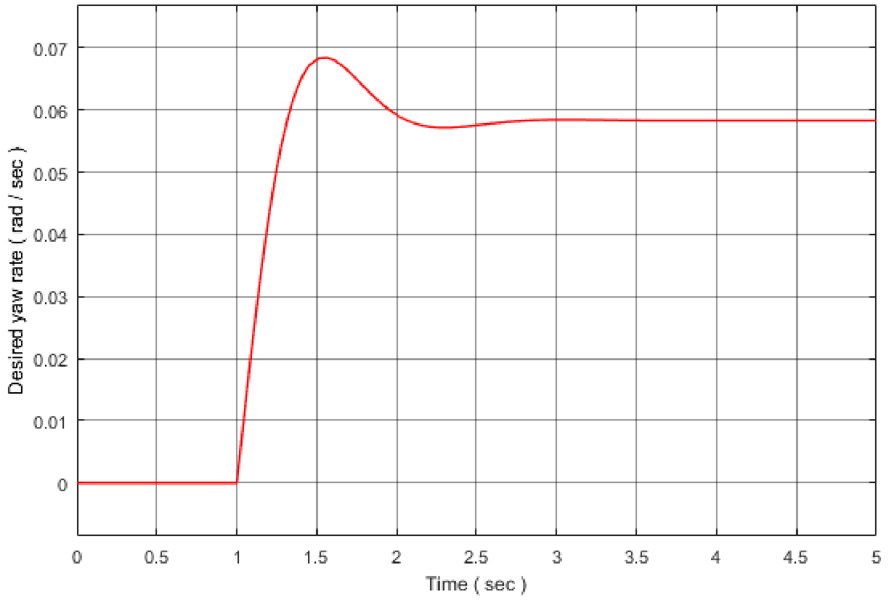


Figure 3.24. The desired vehicle yaw rate at a step signal of steering angle

.

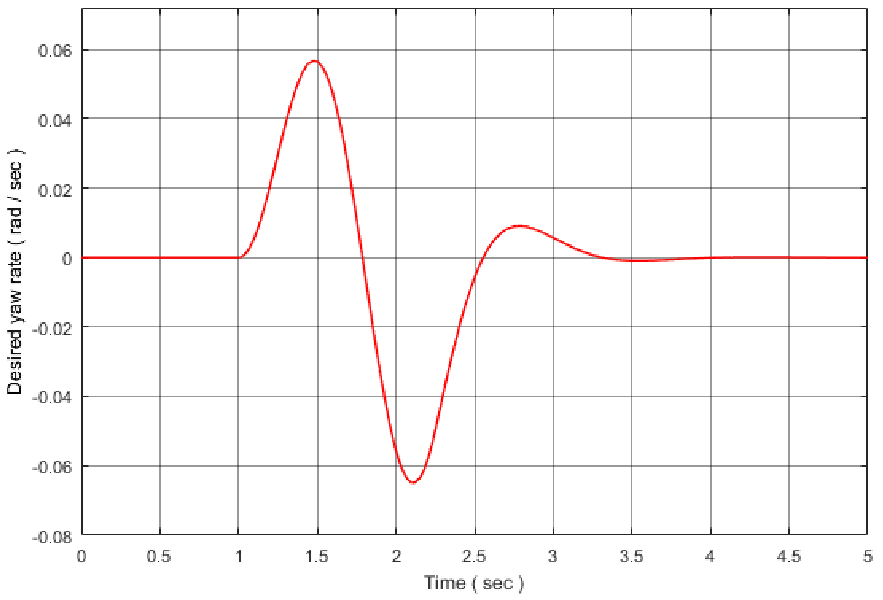


Figure 3.25. The desired vehicle yaw rate on a lane change maneuver.

3.6. Model of Tire Dynamic

A sketch diagram of a modeled tire is illustrated in figure 3.26. The moment of inertia of the wheel is I_w and the effective radius is R_w. The torque applied on each tire T and longitudinal force of the tire F_x is produced at the tire bottom. The tire is rotating with angular speed ω and travels at a longitudinal speed v_x. the total moments about the rotation axis of the tire generates the dynamic equation derived in the following equations:

$$T_1 - F_{x1}.R_w = I_w \dot{\omega}_1 \tag{3.32}$$

$$T_2 - F_{x2}.R_w = I_w \dot{\omega}_2 \tag{3.33}$$

$$T_3 - F_{x3}.R_w = I_w \dot{\omega}_3 \tag{3.34}$$

$$T_4 - F_{x4}.R_w = I_w \dot{\omega}_4 \tag{3.35}$$

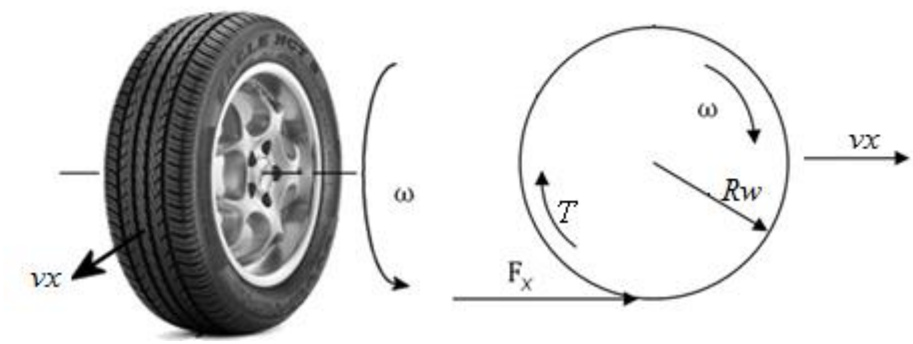


Figure 3.26. Wheel schematic diagram.

3.7. Review of Existing Tire Models

The widely used experiential tire model is called the Magic tire formula which can be used to fit empirical tire data for describing the relationship between the longitudinal force of the tire and slip ratio, the lateral tire force and the slip angle, and the relationship between self-aligning moment and the slip angle. It is derived as follows:

$$y(x) D \sin\{C \arctan[Bx - E(Bx - \arctan Bx)]\} \quad (3.36)$$

$$Y(X) = y(x) + S_v \quad (3.37)$$

$$x = X + S_h \quad (3.38)$$

where Y (X) indicates the output variables: the longitudinal and lateral tire forces, or self- align moment, and X indicates slip angle α or slip ratio σ. The coefficient B is the stiffness factor, C is called shape factor, D is peak factor, and E the factor of curvature. S_v and S_h and are called the vertical shift and horizontal shift, respectively.

The curve which crosses through the origin point, $x = y = 0$ is produced from equation (3.36), and reaches the maximum where $x = x_m$, as illustrated in figure 3.27 for the given values coefficients, the curve presents an antisymmetric style with respect to the origin point, $x = y = 0$. Figure 3.27 shows the physical meaning of the coefficients mentioned above in equation (3.36). For example, if Figure 3.27 represents the relationship between the slip angle and lateral tire force, then coefficient D indicates the higher value with respect to *x and y* coordinates and the factor BCD represents to the curve slop in the origin point, representing the tire cornering stiffness.

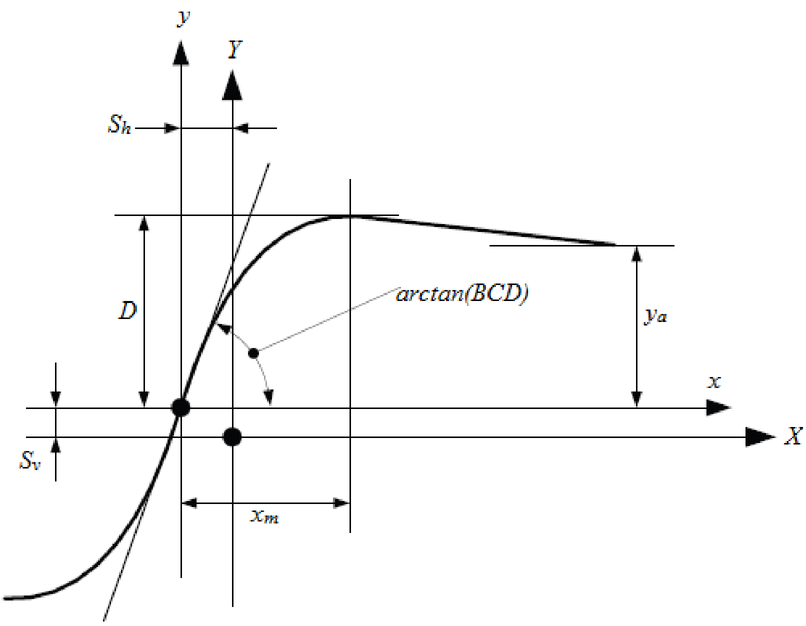


Figure 3.27. Characteristics of the Magic Formula. [2]

It is noted that some of the coefficients in (3.39) and (3.40) are a functions of the vertical tire force F_z and/or camber angle of tire. Coefficients B, C, D, and E can be derived as a functions of the vertical tire force F_z and the road friction coefficient μ as follows:

$$D = a_1(F^z)^2 + a_2 F^z \tag{3.39}$$

Where a_1 and a_2 are experiential coefficients.

For stiffness of tire cornering (i.e., the slope of the lateral tire force and slip angle curve):

$$(BCD) = a_3 \sin[a_4 \arctan(a_5 F^z)] \tag{3.40}$$

And for the align stiffness (the slope of the self-aligning moment and the slip angle curve) or the longitudinal stiffness (the slope of the longitudinal tire force and the slip ratio):

$$BCD = \frac{a_3 (F^z)^2 + a_4 F^z}{e^{a_5 F^z}} \quad (3.41)$$

The stiffness factor B and the curvature factor E are given by:

$$B = \frac{BCD}{CD} \quad (3.42)$$

$$E = a_6 (F^z)^2 + a_7 F^z + a_8 \quad (3.43)$$

where $a_1, a_2, \cdots, a_8$ are empirical coefficients.

Another widely used of tire models is the Dugoff's tire formula. The Dugoff's model represents an interesting alternate to the elastic foundation of analytical tire model modified by Fiala in 1954 for the generation of lateral tire force and by Sharp and Pacejka for combined lateral and longitudinal and forces generation. The Dugoff model permits independent tire stiffness values in both of the lateral and longitudinal directions. Compared to Magic Formula, the Dugoff tire model has the merit of being an analytical derived formula developed from the calculations of force balance. Furthermore, the lateral and longitudinal forces of the tire have a direct relation to the road-tire friction coefficient in more clear equations [2].

The longitudinal F_x and lateral F_y tire forces are defined as follows:

$$F_x = C_1 \frac{\sigma_x}{1+\sigma_x} f(\lambda) \quad (3.44)$$

$$F_y = C_2 \frac{\tan\alpha}{1+\sigma_x} f(\lambda) \quad (3.45)$$

Where C_1 and C_2 are the longitudinal and cornering tire stiffness, σ_x is the tire longitudinal slip ratio and can be defined in equations (3.44) and (3.45), and α_i is the tire slip angle at each tire.

$$\sigma_x = 1 - \frac{r_{eff}\,\omega_i}{v_x} \qquad \text{in the case of deceleration} \tag{3.46}$$

$$\sigma_x = -1 + \frac{v_x}{r_{eff}\,\omega_i} \qquad \text{in the case of acceleration} \tag{3.47}$$

The slip angle for each tire is derived and calculated depend on the geometry dimensions of the vehicle and the wheel speed vectors. If it is possible to determine the velocity at each wheel road contact point then, the it can derive the tire slip angle easily at each tire geometrically and can be expressed as follows:

$$\alpha_{fR} = \arctan\left(\frac{u + L_f .r}{v + L_w .r}\right) - \delta \tag{3.48}$$

$$\alpha_{fL} = \arctan\left(\frac{u + L_f .r}{v - L_w .r}\right) - \delta \tag{3.49}$$

$$\alpha_{rL} = \arctan\left(\frac{u - L_r .r}{v + L_w .r}\right) \tag{3.50}$$

$$\alpha_{rL} = \arctan\left(\frac{u - L_r .r}{v - L_w .r}\right) \tag{3.51}$$

The variable λ and the function $f(\lambda)$ are given by

$$\lambda = \frac{\mu F_z (1 + \sigma_x)}{2\sqrt{\{(C_1\sigma_x)^2 + (C_2 \tan(\alpha))^2\}}} \tag{3.52}$$

And

$$\begin{aligned} f(\lambda) &= (2-\lambda)\lambda \qquad if\ \lambda < 1 \\ f(\lambda) &= 1 \qquad if\ \lambda \geq 1 \end{aligned} \tag{3.53}$$

Where (μ) is the coefficient of road-tire friction and F_i^z represents the vertical tire force for each wheel.

IV. PRINCIPLES OF PID AND FUZZY PID CONTROLLERS

The purpose of this chapter is to give some principles and explanation about two of control methods which are conventional proportional, integral, and derivative (PID) controller and the controller used in this thesis which is Fuzzy PID controller (FPIDC). The most commonly structure, design, and meaning of Fuzzy PID controller will be obtained in this chapter in details.

4.1. Principles Of PID Controller

PID Controller Feedback system is a mechanism of control which uses an information from the measurements. The output is sensed in this control system. Tow principle kinds of the feedback control systems are known which are called positive feedback and negative feedback. The first one increases the input size, but in the negative feedback system, the feedback decreases the input size. It can be said that, the negative system is generally stable. A PID controller has been mostly used as feedback control system of many industrial procedures in the markets since decades, and has remained until these days. Thus, this controller is clearly understood which takes into account the past, present, and future of the errors. Once digital application and implementation has been introduced and a change in the structure of control system was adopted and has been proposed in several applications. however although, that the change does not affect the most primary part when analysis and design of the PID controllers. A PID controller (proportional P, integral I, and derivative D controller) is a technique of the control feedback system. This technique is consisted of a three controllers [52]:

a. PC: Proportional control

b. IC: Integral control

c. DC: Derivative control

4.1.1. The proportional controller role (PC)

It's possible to say that, the role of this controller depends upon the error of present, I on the accumulation of the past error while D on the prediction of future error. These three actions and its weighted sum are used to adjust the proportional control, and it is a simple and a widely used technique of control for many types of systems. Generally, in the proportional controller (PC) steady state error inversely depends on proportional gain (if this gain is made larger then, error goes down). In this controller the proportional response can be checked and adjusted by multiply the error (*Error(t)*) by a constant named (K_p) and called the proportional gain, where the proportional term is derived as follow [52]**:**

$$(P) = K_p . Error(t) \tag{4.1}$$

If this proportional gain *P* is very high, then, the system has the potential to be unstable. In contrast, a small proportional gain K_p results in a small output response to a large error of input. If this gain is very small, the control action might insufficient when responding to the system disturbances. In consequence, the effect of a proportional controller (K_p) will be decreasing the rise time, but never prevent or eliminate the steady-state error. practically, the proportional band is derived as a percentage as follows:

$$(PB)\% = \frac{100}{K_p} \tag{4.2}$$

4.1.2. The integral controller role (IC)

The integral part of a controller provides a signal depending on how long the error is persisted. It works to stop and prevent this persistence of the error by increase the control signal with time. This action will reduce the steady state error some cases, depending upon the type of the system and the type of the reference signal, eliminates it. Integral control *I* is not used on its own usually, however it is an effective than Proportional control *P* for eliminate the step response steady-state error of the first order plant. And for the second order system, using of *I* control leads to a third order system that, depending upon the parameters of system, can result in unstable oscillation [52].

The integral of the error *e(t)* proportional to the control signal of the integral controller output $u_i(t)$ is:

$$u_i(t) = K_i \int_0^t e(t).d(t) \tag{4.3}$$

K_i is called the integral gain of the controller.

4.1.3. The derivative controller role (DC)

The derivative component of this process error can be calculated by multiply change rate by the derivative gain K_d and by confirming the rate of slope of error. In this case the derivative gain slows the change rate in the controller output. The influence of a derivative control (K_d) component will be to increase the stability of the system, reducing the overshoot, and improving the transient response. The derivative gain is derived as [52]:

$$(D) = K_d \frac{dError(t)}{d(t)} \tag{4.5}$$

Table (4.1) shows the Effect of each component K_p, K_d, and K_i on a the closed-loop system .

4.2. structure and design of the PID controller

Figure 4.1 demonstrates an ideal structure of a control system including PID controller and figure 4.2 illustrates a structure of a control system with PID controller. The error e(t) of the signal is used to produce the three components of the controller (P, I, and D actions), as a resulting of signals sum weighted to form the signal u(t) which applied to the process model.

.

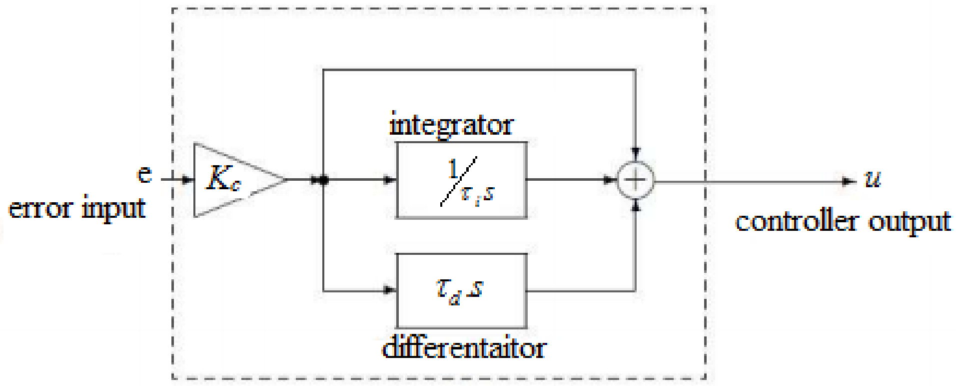


Figure 4.1. A PID controller structure. [52]

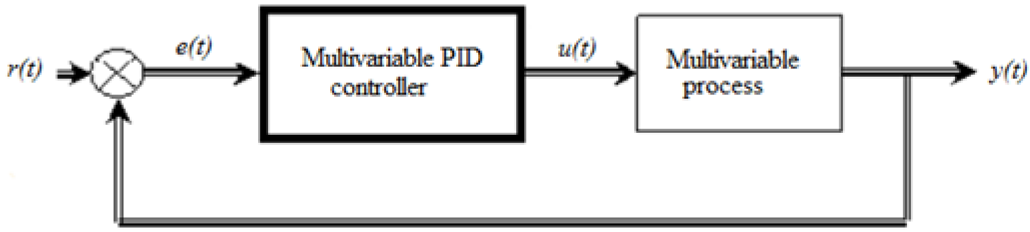


Figure 4.2. A block diagram of control system including PID controller.

Table 4.1. The effect of a PID controller parameter in a closed loop system.

Parameter	Rise time	Overshoot	Settling time	Steady-state error
K_p	Decrease	Increase	Small change	Decrease
K_i	Decrease	Increase	Increase	Decrease significantly
K_d	Minor decrease	Minor decrease	Minor decrease	No effect in theory

The multi-variable processes has an input signal *u(t)*, the error signal *e(t)* can be derived as $e(t) = r(t) - y(t)$, where *r(t)* is a reference signal and represents the input variable. The standard structure of the PID controller is known also as a three-term controller. This standard mode of action can be demonstrated by the parallel connection of the proportional, integral and derivative elements which illustrated in figure 4.3.

$$G(s) = K_p(1 + \frac{1 + T_I.T_D.S^2}{T_I.S}) = K_p(1 + \frac{1}{T_i s} + T_D s) \qquad (4.6)$$

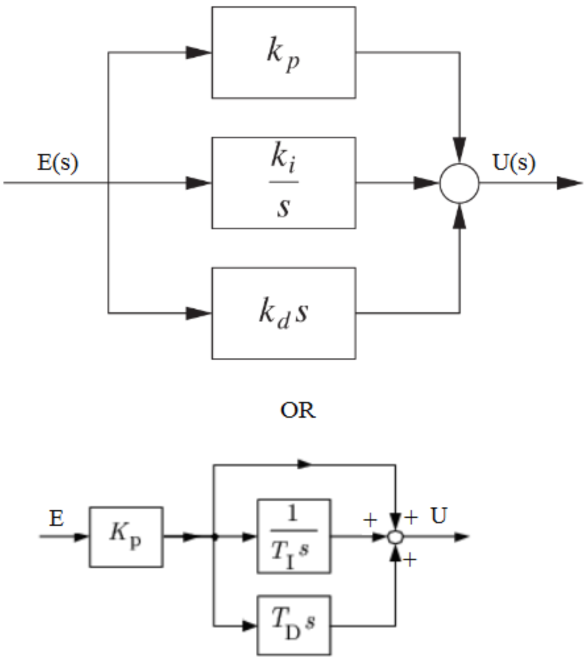


Figure 4.3. The PID Compensator Parallel Form. [52]

Where T_I is a constant and represents an integral time constant, K_P, T_D are the proportional gain and the derivative time constant respectively, K_I and K_D can be calculated as ($K_I = K_P / T_I$) and ($K_D = K_P . T_D$). The functions of the three components of the PID controller listed below:

- The proportional term P: can provide a comprehensive control action proportional to the error signal through all pass gain factor.
- The integral term I: can reduce the steady state error due to low frequency compensation using an integrator.
- The derivative term D: can improve the transient response due to a high frequency compensation using a differentiator.

The three variables were mentioned above T_I , K_P and T_D are generally tuned due to given ranges. Thus, they are often named the tuning parameters of the controller.

During a suitable choice of these tuning parameters a PID controller can be adapted for a certain plant to make a perfect behavior of the system which controlled.

The controller output time response is:

$$u(t) = (K_p)(e(t) + \frac{\int_0^t e(t)dt}{(T_i)} + (T_d)\frac{de(t)}{dt}) \tag{4.7}$$

Using above relationship due to a step input of error *e (t),* (i.e. *e(t) = δ(t))* , the step response of the PID controller *r(t)* can be easily calculated. The result is illustrated in the figure below. One has to observing that the length of the arrow *($K_T P_D$*) of the *D* action is a measure of the impulse weight *δ*.

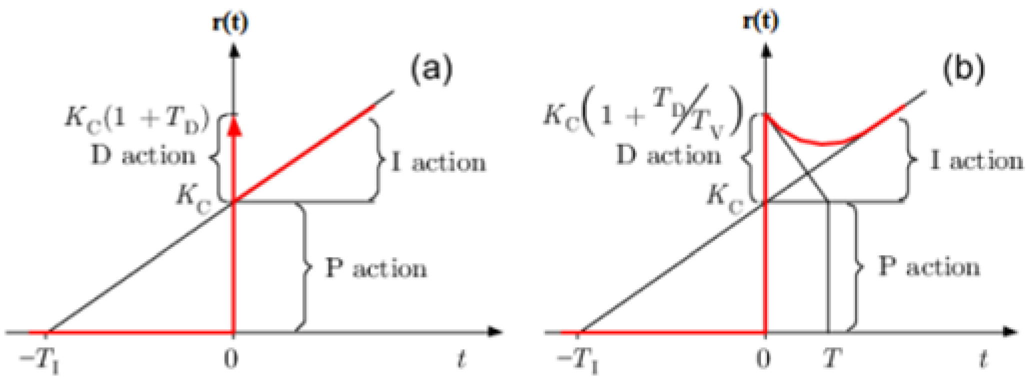


Figure. 4.4. a) Step response of PID ideal form b) Step response of PID real form. [52]

4.3. Fuzzy PID Controller

4.3.1. The Meaning Of Fuzzy PID Controller

Self-tuning FLPIDC means that the PID controller parameters which are Kp, Ki and Kd are tuned by using the fuzzy logic tuner automatically. There are available two different kinds of fuzzy logic controllers: the Sugeno and Mamdani types. There are mainly two differences in their fuzzy rule consequent: a Sugeno fuzzy controller can employ the linear functions of the input variables while, a Mamdani fuzzy controller employs a fuzzy sets as the consequent. Although fuzzy logic controller can be designed implemented using a simple modification of some IF-THEN rules of fuzzy logic into the control system of conventional controllers, this approach in general complicates the overall design and prevent to produce new FPIDCs which properly reflect an essential nature and characteristics of the conventional PID controllers. In addition, they usually have no analytical formulas for stability analysis control and specification. The FPIDC to be introduced in this chapter is natural extensions of its conventional version, that preserve the PID controllers linear structure, with conventional and simple analytical forms as a final results of the required design. So, they can replace the conventional controllers (PID controllers) in any operating plants or processes in control systems directly. The conventional PID controller design was modified and a new fuzzy PID controller was implemented and designed. Instead of the summation influence a Mamdani-based fuzzy inference system is constructed and the inputs to the Mamdani based FIS "fuzzy inference system" are “change in error” and error itself”. [54]

The conventional PID controller coefficients are not predominant properly tuned and adjusted for the nonlinear systems, with unpredictable variations of parameter. Hence, it is important to tune the three parameters of the PID controller automatically by using a suitable fuzzy rule base.

4.3.2. Fuzzy PID Controller Structure

The performance of system can be improved and modified by proper adjusting and scheduling of the three parameters of PID controller (K_p, K_i and K_d). The most known structure of the Fuzzy PID controller (FLPIDC) is obtained in the following figure.

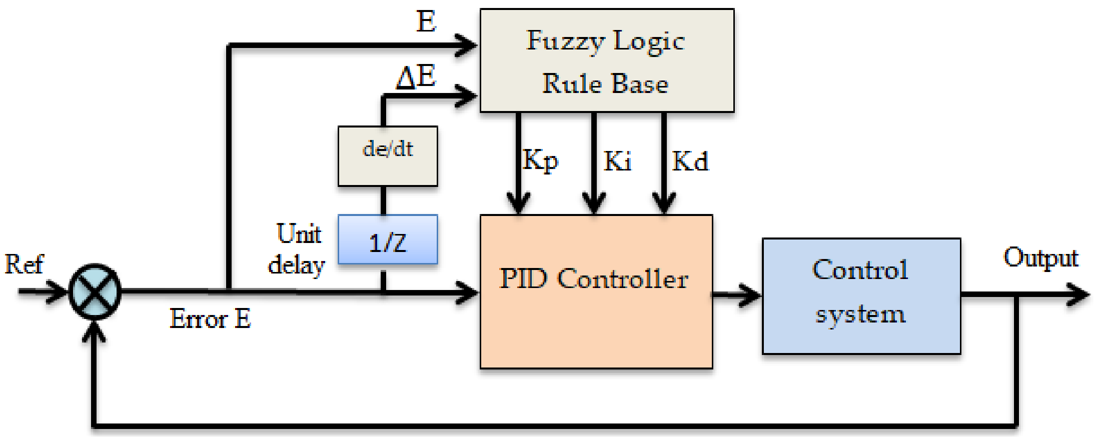


Figure 4.5. The structure of FPIDC. [54]

In Fuzzy self-adjusting PID controller, it takes the error E (the deviation between the desired set point and the output of the system) and ΔE (change in error or error rate with unit delay) as a two inputs. The parameters of PID controller (K_p, K_i, K_d) are tuned and adjusted by using fuzzy inference automatically to meet and verify the requirements of E and ΔE for PID parameter self-setting in different time. [54]

4.3.3. Build Of Mamdani Systems in Matlab

This section demonstrates how to build design a Fuzzy Inference System (FIS) using Matlab Software. Figure 4.8 shows the graphical tools required to design and build, view, and fuzzy inference systems.

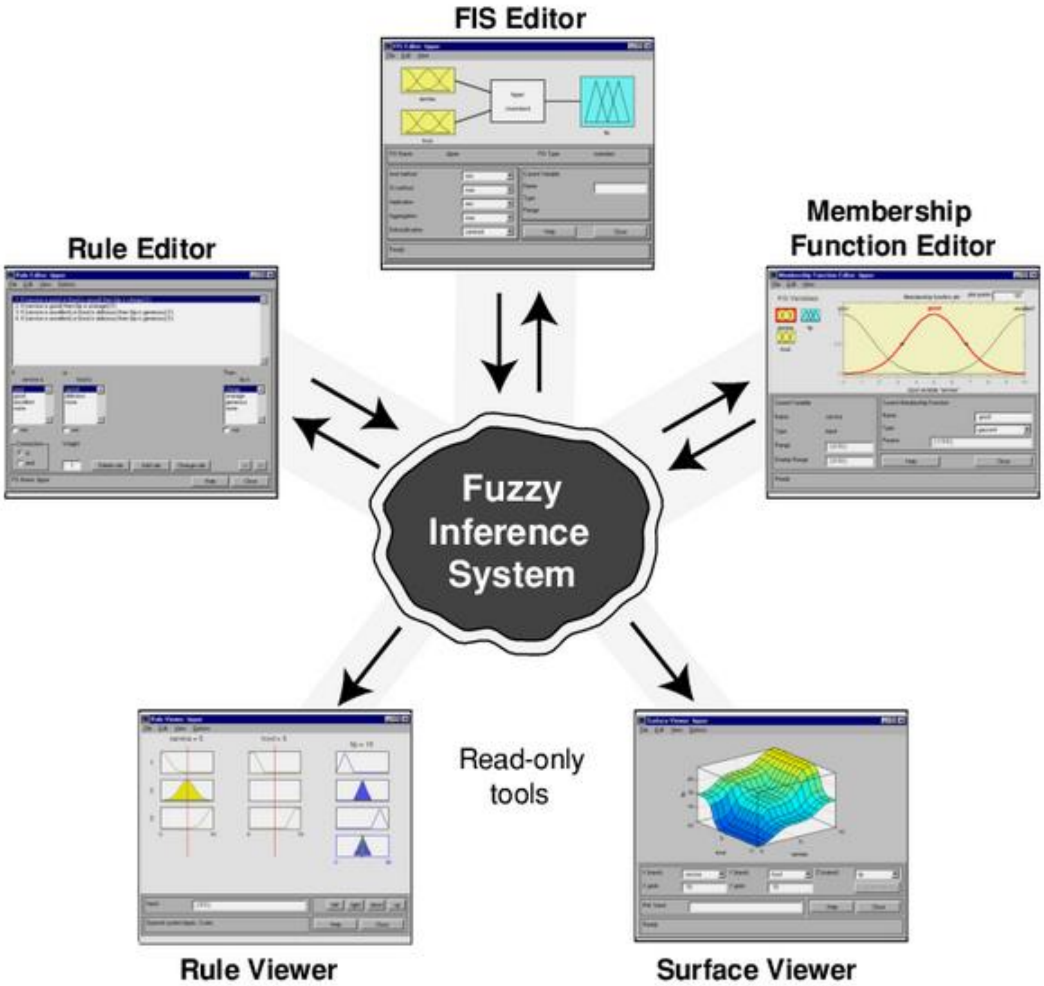


Figure 4.6. The graphical tools required to build, view, and edit fuzzy inference system.

- Fuzzy Logic Designer to get and handle a high level issues for the control system Its necessary to ask how many output and input variables of the fuzzy controller? What is the name of each one?

The steps of design are started in sequence by create a Fuzzy Inference System editor (FIS) shown in figure 4.7.

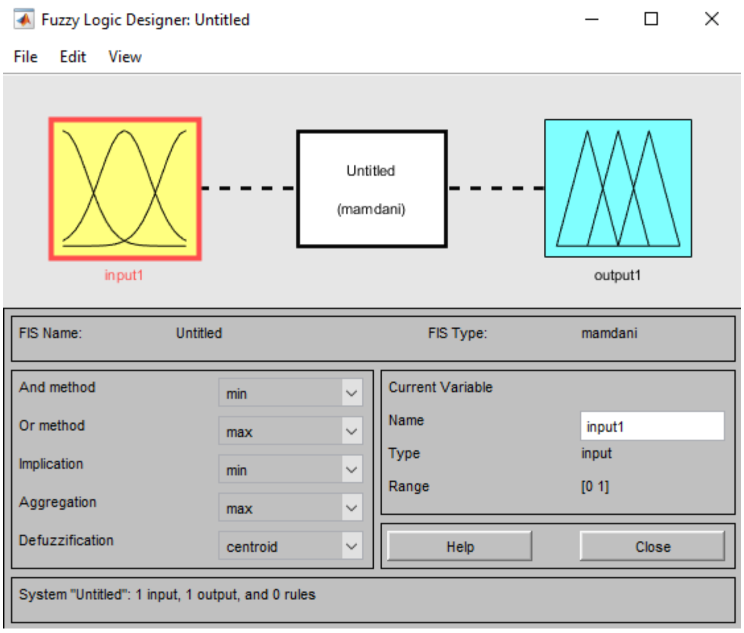


Figure 4.7. FIS editor in Matlab.

Taking into account the controller needed in this thesis, two input variables and other three output variables are added and shown in figure 4.8. The Fuzzy Inference System (FIS) editor offers the relevant information of the Adaptive gain scheduling FLPIDCs.

Fuzzy Logic in Matlab does not itself limit the number of inputs variables. However, the computer memory available may impose a certain constraint. If the inputs number is too great, or the membership functions number too many, consequently, difficulty will be found when analysis of the FIS using other tools.

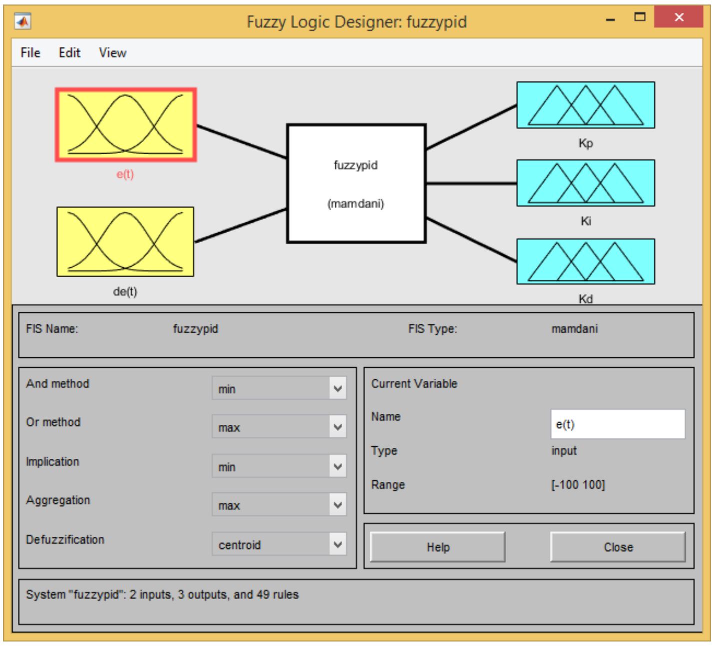


Figure 4.8. FIS editor of proposed FLPIDC.

- <u>The editor of membership functions</u> to define the membership functions shapes of associated with each input variable and output variable.

The membership functions of the input variables (E and ΔE) and the output variables (K_p, K_i, and K_d) are displayed in figures 4.9 and 4.13.

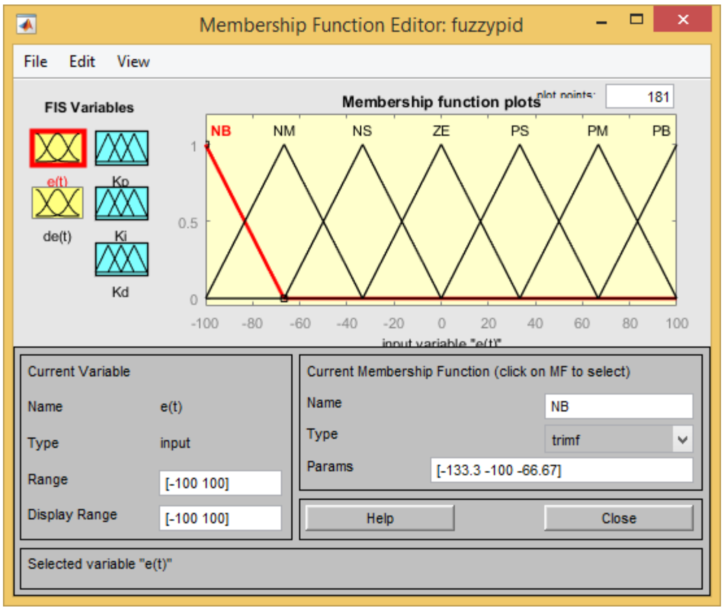


Figure 4.9. The membership function of the error E.

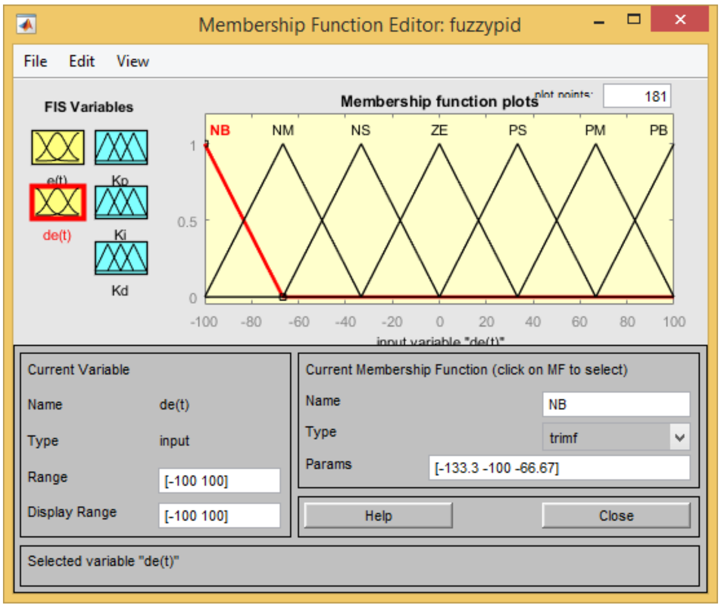


Figure 4.10. The membership function of the change of error ΔE.

The ranges of output variables for e(t) and de(t) are between 0 up to 1.

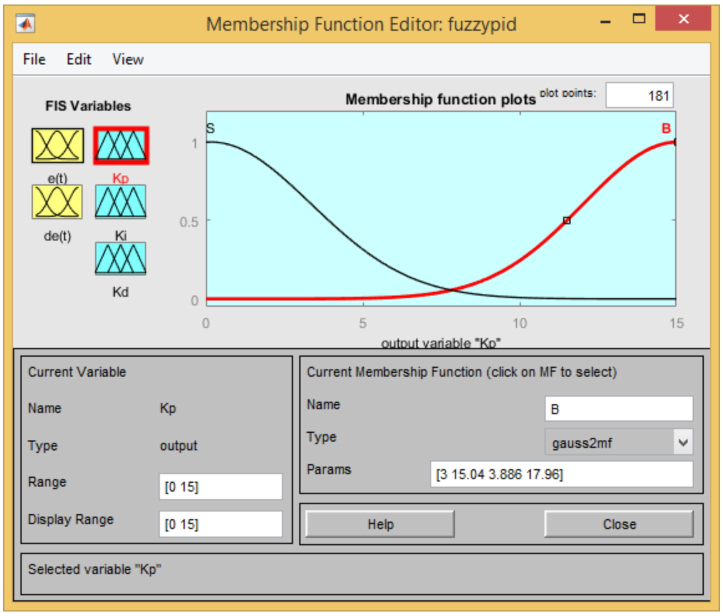


Figure 4.11. The membership function of K_p.

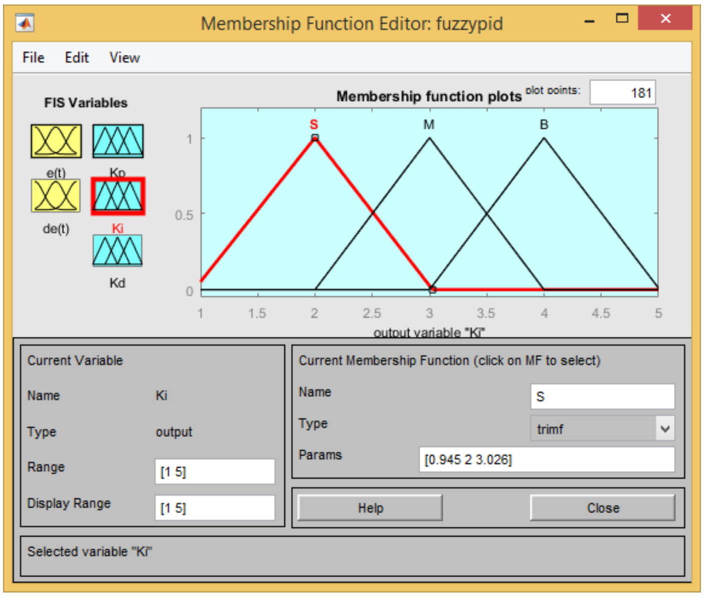


Figure 4.12. The membership function of K_i

.

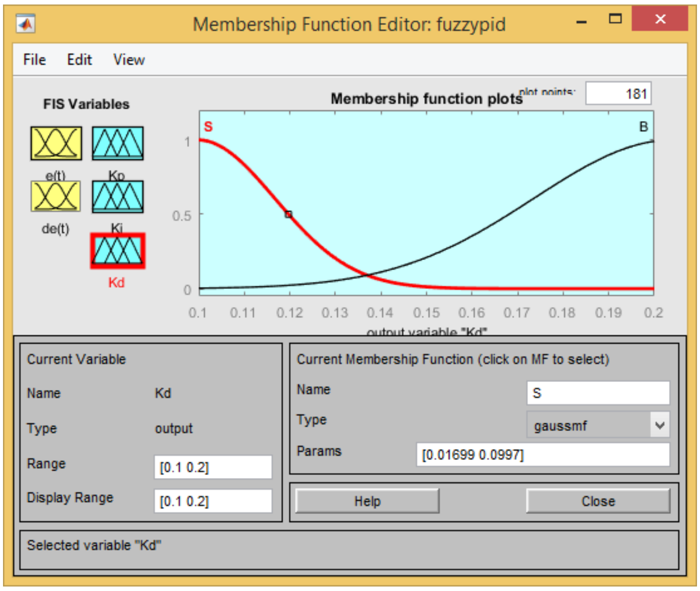


Figure 4.13. The membership function of K_d.

- Rule Editor: used to edit the list of rules which defines the performance and behavior of the system.

The fuzzy PID rules for Kp, Ki and Kd parameters are illustrated in the following three tables, where each cell in the tables shows one of the output membership function of the control rule with a two input membership functions. The control rules are built based upon the IF statement: IF input1 and input2 THEN output1, output2, and output3.

For example to read the tables, consider the fifth column and third row in table 4.3. IF the error e(t) is NS and the change in error de(t) is PS, THEN K_p is B and K_i is M and K_d is B.

Table 4.2. Fuzzy PID rule for K_p parameter.

de(t) / e(t)	NB	NM	NS	ZE	PS	PM	PB
NB	B	B	B	B	B	B	B
NM	S	B	B	B	B	B	S
NS	B	S	B	B	B	S	S
ZE	S	S	S	B	S	S	S
PS	S	S	B	B	B	S	S
PM	S	B	B	B	B	B	S
PB	B	B	B	B	B	B	B

Table 4.3. Fuzzy PID rule for K_i parameter.

de(t) / e(t)	NB	NM	NS	ZE	PS	PM	PB
NB	S	S	S	S	S	S	S
NM	M	M	S	S	S	M	M
NS	B	M	M	S	M	M	B
ZE	B	B	M	M	M	B	B
PS	B	M	M	S	M	M	B
PM	M	M	S	S	S	M	M
PB	S	S	S	S	S	S	S

Table 4.4. Fuzzy PID rule for K_d parameter.

de(t) / e(t)	NB	NM	NS	ZE	PS	PM	PB
NB	S	S	S	S	S	S	S
NM	B	B	S	S	S	B	B
NS	B	B	B	S	B	B	B
ZE	B	B	B	B	B	B	B
PS	B	B	B	S	B	B	B
PM	B	B	S	S	S	B	B
PB	S	S	S	S	S	S	S

.

- Rule Viewer: to view the diagram of the fuzzy inference diagram. this viewer is used as a diagnostic to see the active rules, and how each one of the membership function shapes affect the results.
- Surface Viewer: to show the dependency of one output on the inputs which is, can generate and draw an output surface shape for the system. Figure 4.14 shows an example to know the surface viewer in Matlab.

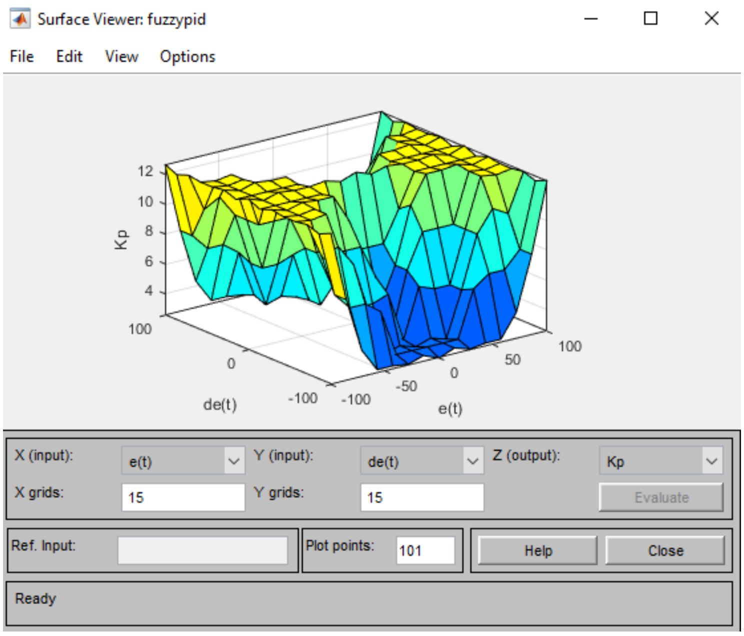


Figure 4.14. Surface viewer shape in Matlab.

V. CONTROL STRATEGY AND SIMULATION RESULTS

In this chapter the simulation results for planar vehicle model and full vehicle model will be presented. In these two models the FPIDC is used in this thesis to enhance the desired improvement of the vehicle dynamic stability, the desired vehicle model (Actual vehicle) as obtained in figure 3.23 is used prominently for vehicle stabilization control analyses and control system design.

5.1 Vehicle system control strategy

The proposed control system of the planar vehicle model is implemented based on equations 3.1, 3.2, and 3.3 which are obtained in chapter three. The structure of the implemented model with the controller for planar model is illustrated in the following figure.

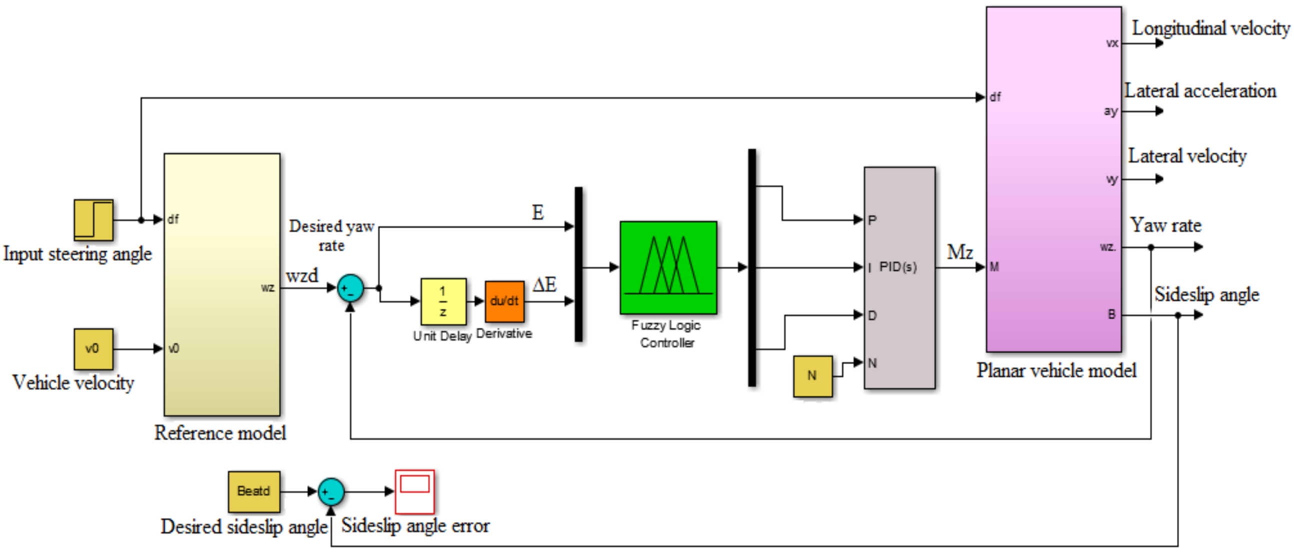


Figure 5.1. The vehicle model structure with controller for planar model.

In this thesis, designing of Fuzzy PID Controller (FPIDC) is implemented to improve the vehicle yaw stability in the case of vehicle planar model and full vehicle model. The fuzzy PID controller can be tune the equivalent components of the PID controller which are proportional control PC, integral control IC, and the derivative control DC components. The control system which be designed includes three parts: the 2 DOF model which represents the reference vehicle model, non-linear vehicle model (Actual model), and the controller FPIDC. The structures of the planar and full vehicle models including the controller are illustrated in figure 5.1 and figure 5.2.

A method for determining the ideal values of vehicle yaw rate and sideslip angle can be done using reference model and based on the steering angle (δ) which can be derived through the driver action. The PID controller is commonly used control algorithm in various industries because of its simplicity, ready availability and faster processing time. The critical regulation of the PID parameters is a complex design task, while it is a very simple decision-making process in case of using fuzzy controller FPIDC. In this study, the comparison between the sideslip angle β and the ideal sideslip β_d is made and the ideal yaw rate w_{zd} is also compared with the actual yaw rate w_z. Then the calculation of the yaw rate error E and its change rate is done. The implementation of the fuzzy control is made as fellows: the first step is fuzzification of the input parameters such as yaw rate error E and its rate of change ΔE, followed by the fuzzification of the direct yaw moment M_z. Fuzzification of the input parameters transforms the actual input variables into fuzzy terms according to the selected member functions. [53]

The Simulink model of full vehicle model without control is built based on differential equations from (3.8) to (3.25). The structure of vehicle model with controller for the full vehicle model shown in figure 5.2 same to the structure model in the case of planar vehicle model.

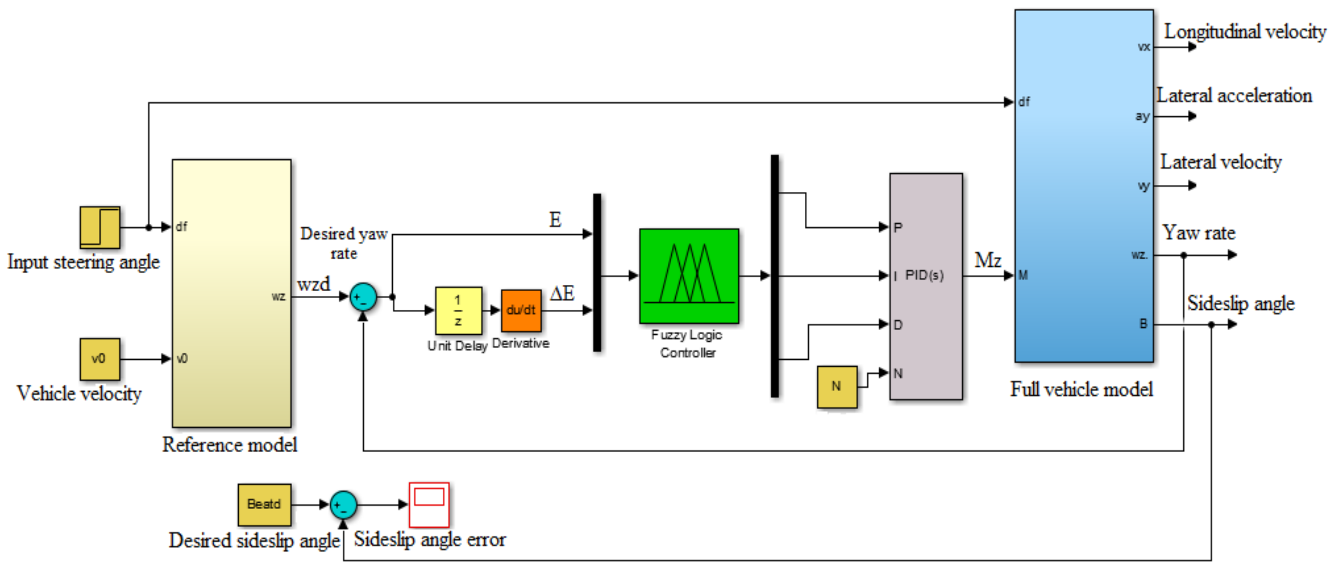


Figure 5.2. The structure of vehicle model with controller for full model.

5.2 Simulation Results and Discussion

In order to analyze and evaluate the control system performance, some simulation investigations are performed. The dynamic behaviors and performance of the developed control system using FPIDC in the case of planar and full vehicle models are performed using Matlab Simulink. It is assumed that the car travels at a constant speed *(v)* = 20 m/sec and the road friction coefficient μ equal 0.4 . One driving condition is performed which is the input of steering. The input steering will set as a step signal which have an amplitude of two degrees (0.035 radians) and will set also as a lane change maneuver with amplitude of front steering angle of

0.035 radians as obtained in chapter three (Figures 3.4 and 3.5). Comparison between the performance of the vehicle (Planar and full models) in both cases of the steering angle will be discussed.

The vehicle parameters values which are used in the thesis simulations are listed below in table 5.1.

In chapter three, the mathematical models of planar vehicle dynamic model and the full model of vehicle model have been presented and the simulation results of both models are presented in this chapter.

Table 5.1. The physical parameters of the vehicle.[49]

Parameter	**Unit**	**Value**	**Parameter**	**Unit**	**Value**
m_s	*Kg*	810	c_1	*(KN.s/m)*	1570
m	*Kg*	1030	c_2	*(KN.s/m)*	1570
a	*m*	0.968	c_3	*(KN.s/m)*	1760
b	*m*	1.392	c_4	*(KN.s/m)*	1760
d	*m*	0.64	ks_1	*(KN/m)*	20.6
R_w	*m*	0.303	ks_2	*(KN/m)*	20.6
I_w	$Kg.m^2$	4.07	ks_3	*(KN/m)*	15.2
μ	---	0.4	ks_4	*(KN/m)*	15.2
C_1	*KN/rad*	52.526	k_{af}	*(N.m/ rad)*	6695
C_2	*KN/rad*	29000	k_{ar}	*(N.m/ rad)*	6695
C_{af}	*KN/rad*	95.117	I_x	$(Kg.m^2)$	300
C_{ar}	*KN/rad*	97.556	I_y	$(Kg.m^2)$	1058.4
g	m/s^2	9.81	I_z	$(Kg.m^2)$	1087.8
m_{u1}	*Kg*	26.5	v_0	*m/s*	20
m_{u2}	*Kg*	26.5	f_r	---	0.015
m_{u3}	*Kg*	24.4	h	*m*	0.505
m_{u4}	*Kg*	24.4			

The simulation results of planar vehicle model include comparison of the vehicle performance and behavior in case of without control and in the other case of using Fuzzy PID controller.

Figures 5.3-5.8 indicate the results which selected of the planar model of the vehicle in case the controller is not used and with controller, where in the first four figures (5.3 to 5.6) show the vehicle longitudinal velocity and the vehicle lateral acceleration at a step signal of steering angle and on lane change maneuver respectively, while the vehicle yaw rate is illustrated in figures 5.7 and 5.8.

Figures 5.9 and 5.10 present a comparison between the behavior of the car sideslip angle with the controller and without it. From these figures we can say, the control system used with of FPIDC clearly improved the stability of vehicle and performance.

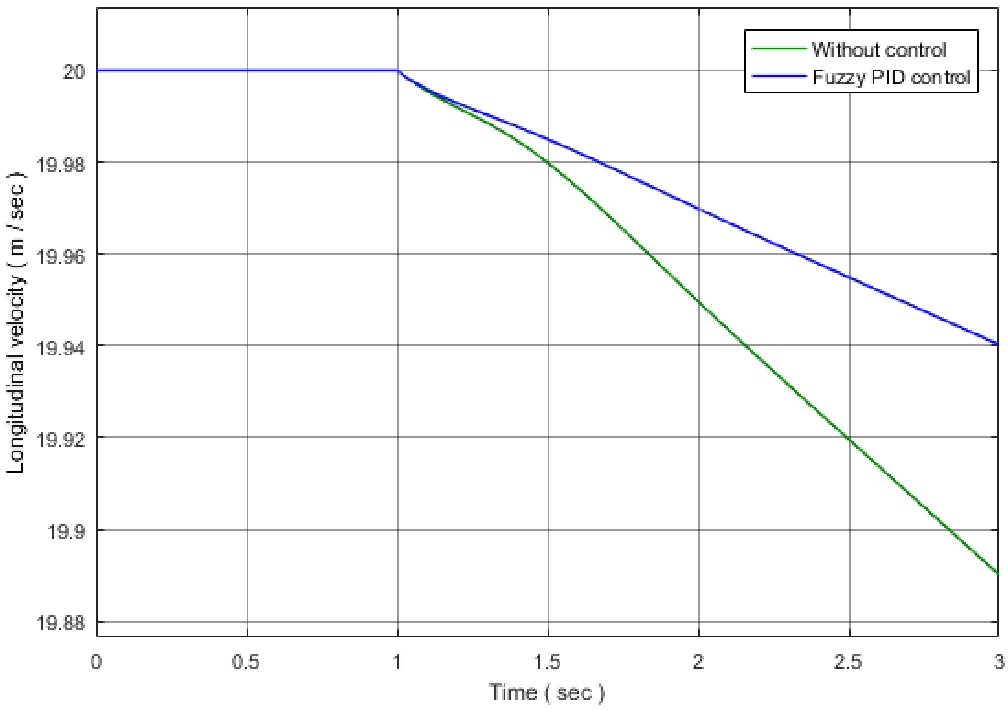


Figure 5.3. Comparison of the vehicle longitudinal velocity with FPIDC and without FPIDC at a step signal of steering angle (Planar model).

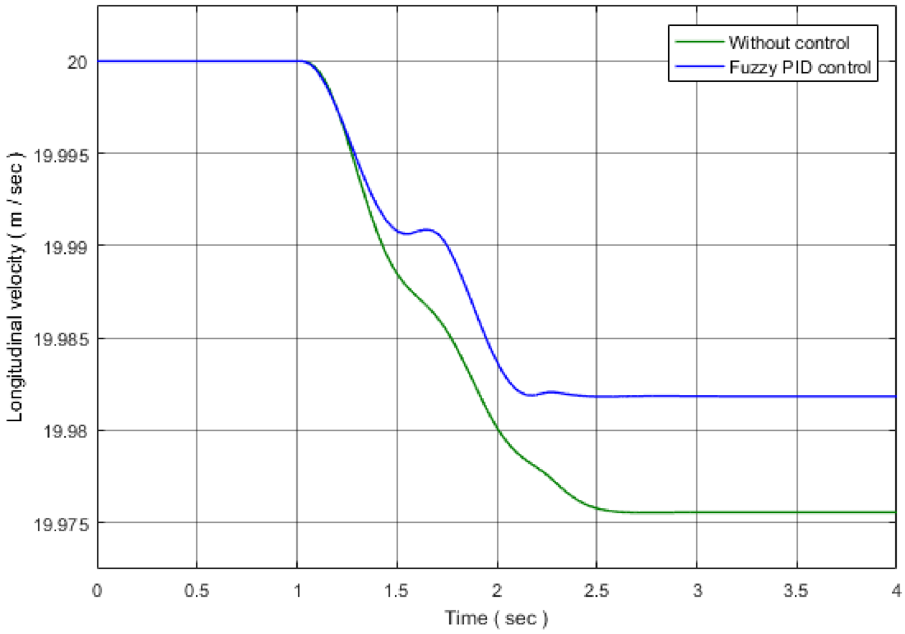


Figure 5.4. Comparison of the vehicle longitudinal velocity with FPIDC and without FPIDC on a lane change maneuver (Planar model).

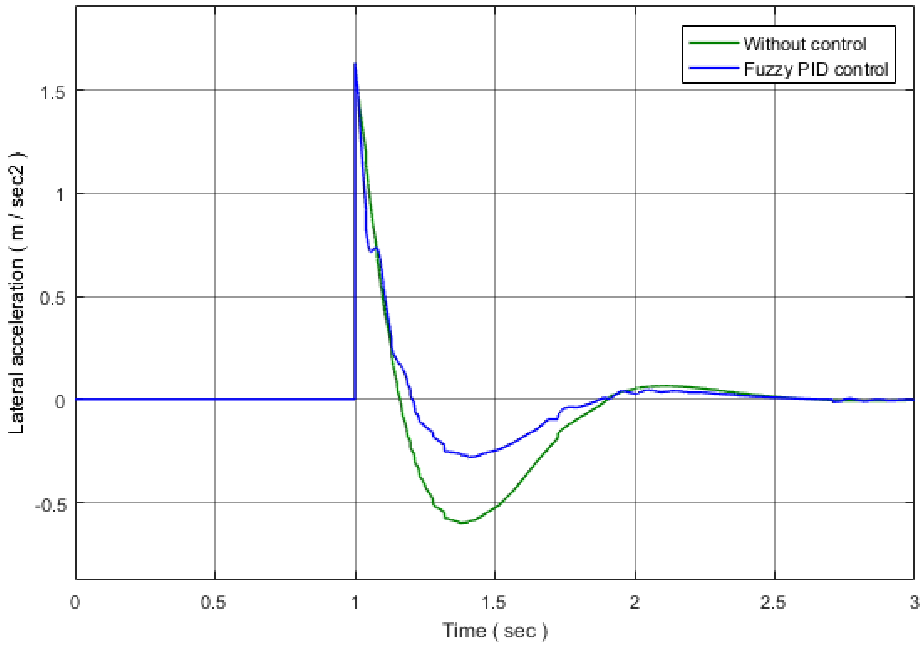


Figure 5.5. The vehicle lateral acceleration behavior with FPIDC and without FPIDC at a step signal of steering angle (Planar model).

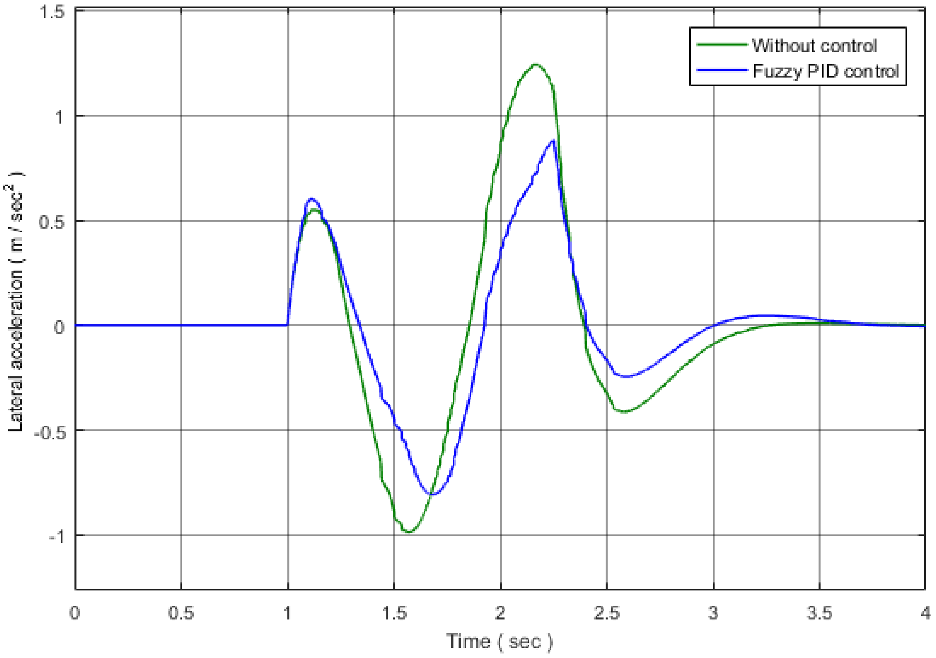


Figure 5.6. The vehicle lateral acceleration behavior with FPIDC and without FPIDC on a lane change maneuver (Planar model).

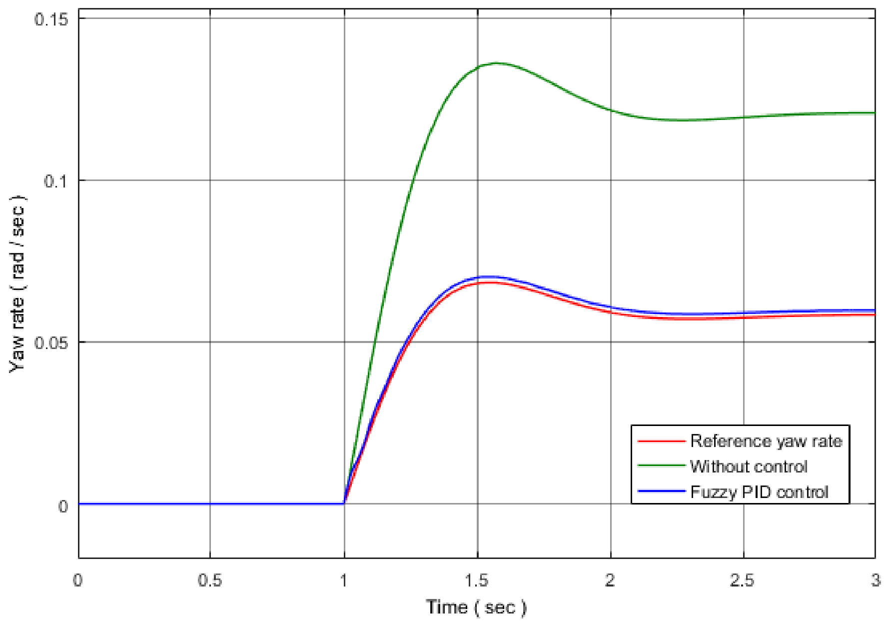


Figure 5.7. The vehicle yaw rate behavior with FPIDC and without FPIDC at a step signal of steering angle (Planar model).

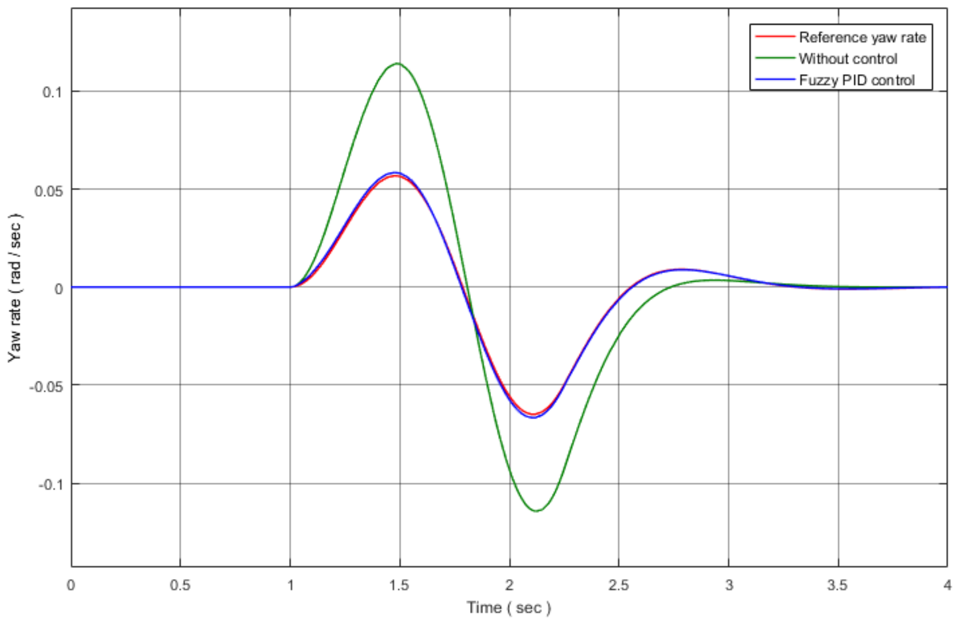


Figure 5.8. The vehicle yaw rate behavior with FPIDC and without FPIDC on a lane change maneuver (Planar model).

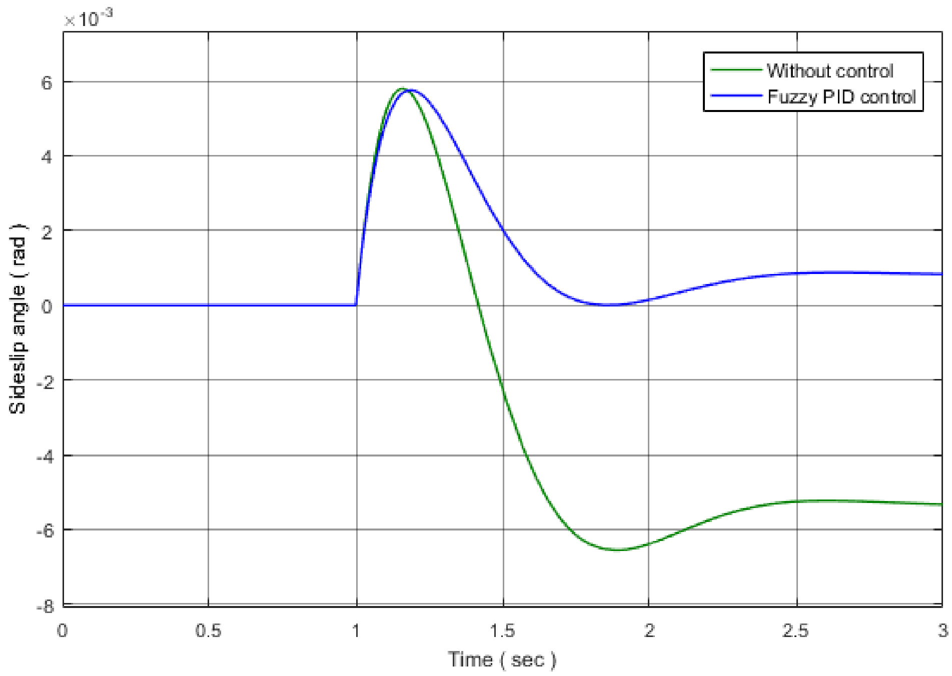


Figure 5.9. The vehicle Sideslip angle behavior with FPIDC and without FPIDC at a step signal of steering angle (Planar model).

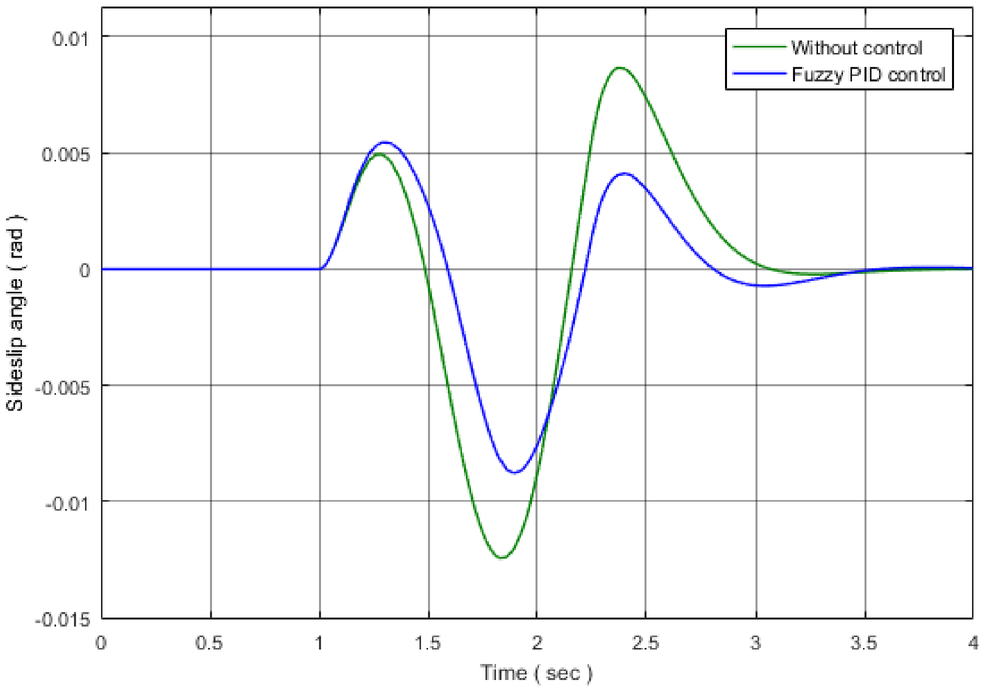


Figure 5.10. The vehicle Sideslip angle behavior with FPIDC and without FPIDC on a lane change maneuver (Planar model).

Fuzzy rule viewer shown in figure 5.11 gives the influence of control signal for rules viewer changing. It is used as a specific diagnostic test to create and find which rules is active and how every single member function forms are influencing the final results. After that rules are estimated and crisp values are created by defuzzification of the membership functions corresponding.

From the rules viewer of fuzzy, screen shot as shown in figure 5.7, it is illustrated that the tuned parameters of Fuzzy logic PID controller at this particular instance is $K_p = 2.69$ and $K_i = 4$, $K_d = 0.176$, for e(t) = -6.12, de(t) = -100. Figure 5.12 presents both the output and input relations in the shape of Cartesian rule surfaces.

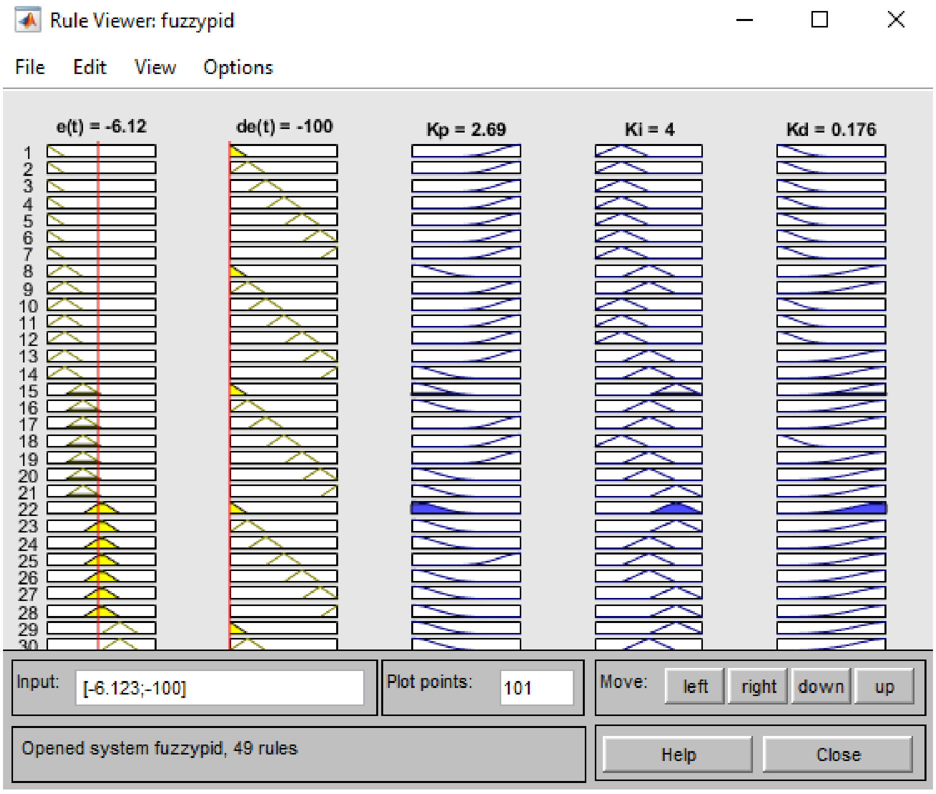
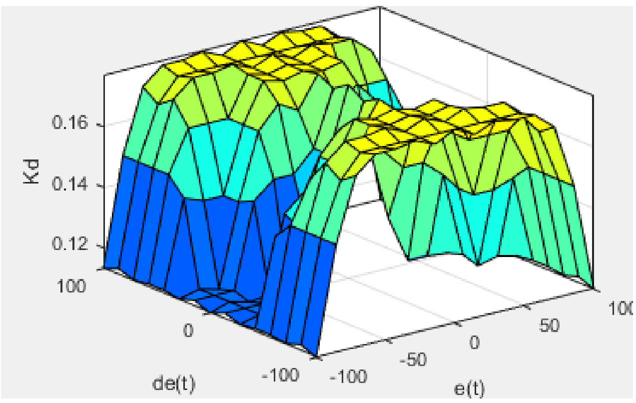
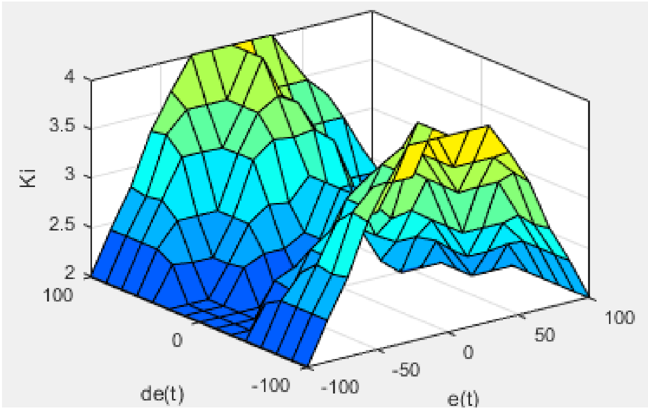


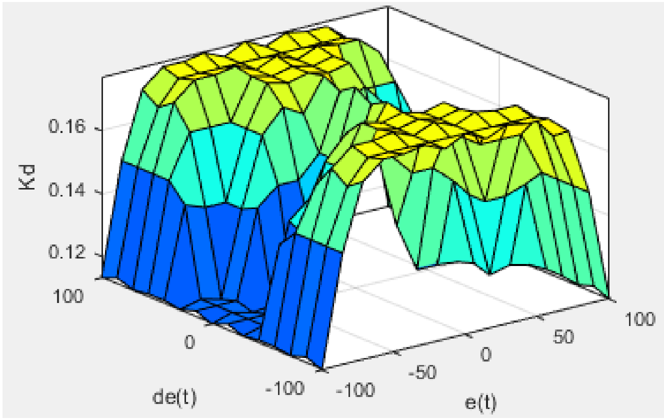
Figure 5.11. Rule viewer of fuzzy PID controller in the case of planar vehicle model.



(a)



(b)



(c)

Figure 5.12. The relations of inputs and outputs in the shapes of Cartesian rules surfaces of P, I, and D control variables, (a) P variable; (b) I variable, and (c) D variable.

Figures 5.13 to 5.20 present the selected results of full vehicle model for all of vehicle longitudinal velocity, the lateral acceleration , yaw rate ,and the vehicle sideslip angle when using the controller (FPIDC) and if not used at a step signal of steering angle and on lane change maneuver respectively. Also, in this model the vehicle stability behavior with fuzzy PID controller is improved especially due to vehicle yaw rate.

From the rules viewer of fuzzy, screen shot as shown in figure 5.21, it is obvious that the tuned parameters of Fuzzy logic PID controller at this particular instance are K_p = 12.7 and K_i = 1.99, K_d = 0.133, for e(t) = -67.3, de(t) = 0. Figure 5.22 demonstrates the output and input relations in the shape of Cartesian rules surfaces.

It can be seen that from figures 5.3 to 5.10 and figures 5.13-5-20 which represent how the Fuzzy PID controller used in this work affected on the vehicle performance in both planar and full vehicle models and how to the controller improved the vehicle stability and handling.

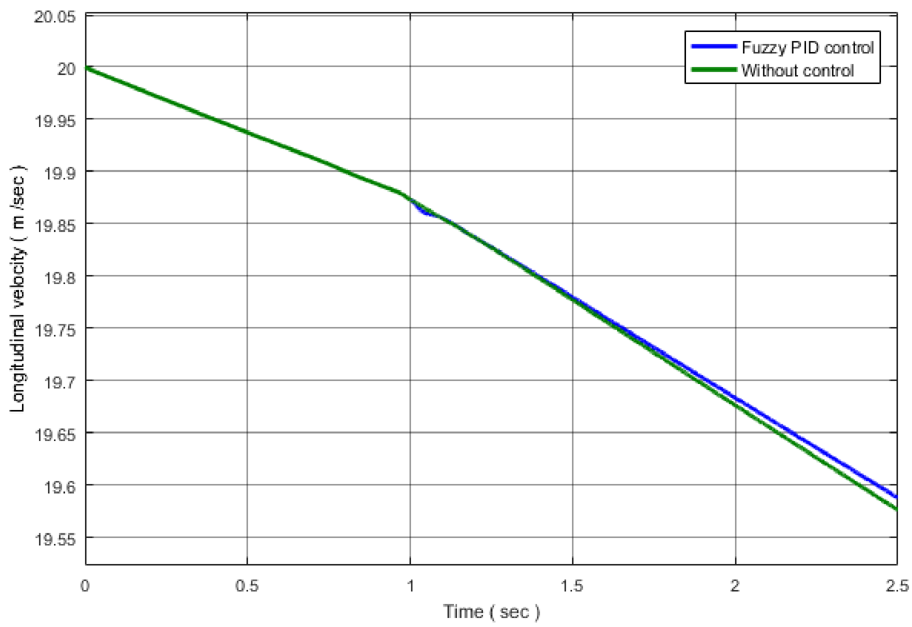


Figure 5.13. The vehicle longitudinal velocity performance with FPIDC and without FPIDC at a step signal of steering angle (Full model).

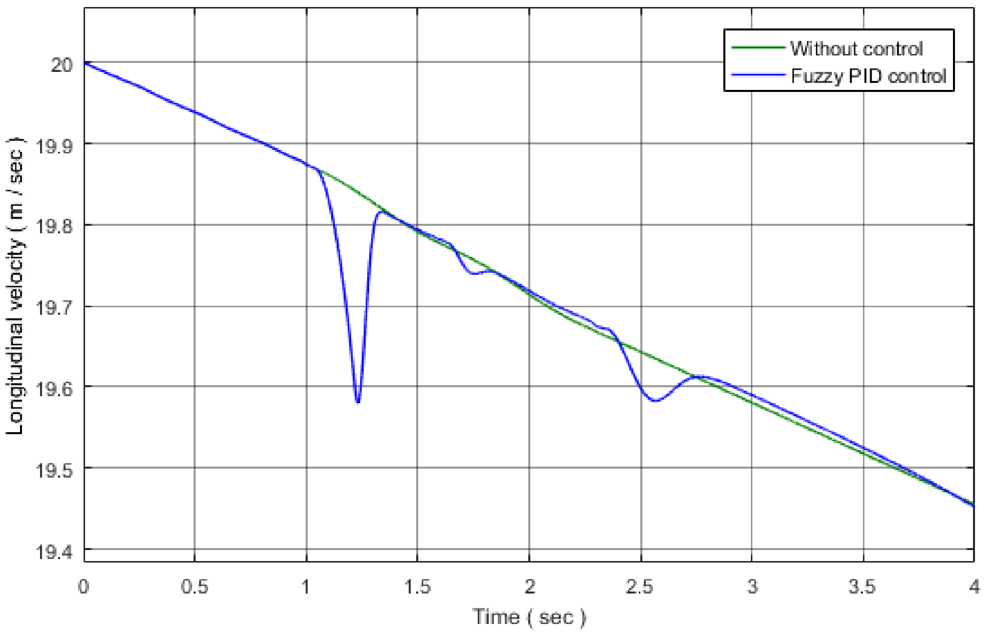


Figure 5.14. The vehicle longitudinal velocity performance with FPIDC and without FPIDC on a lane change maneuver (Full model).

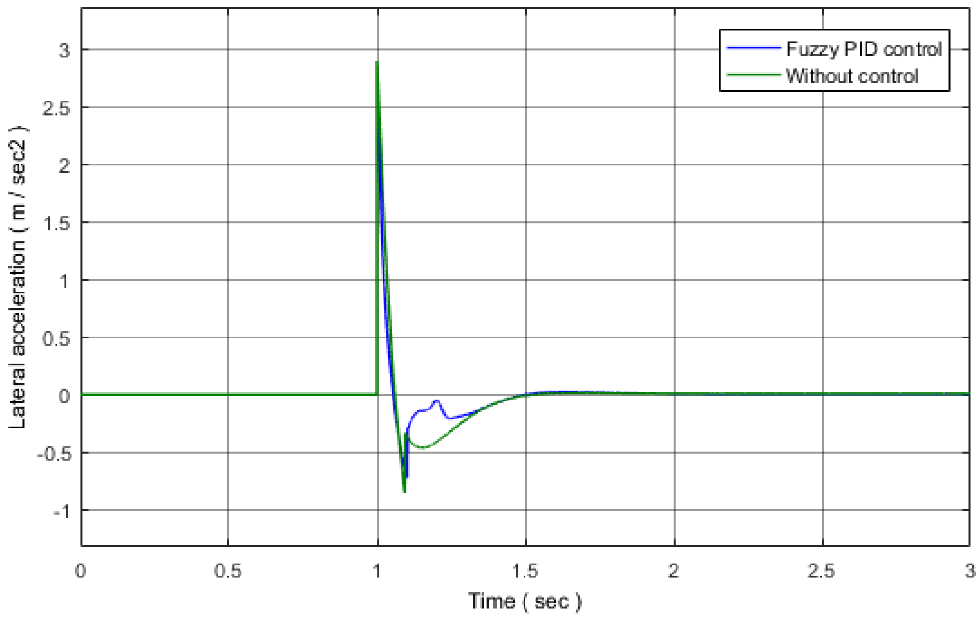


Figure 5.15. Comparison of the lateral acceleration with FPIDC and without FPIDC at a step signal of steering angle (Full model).

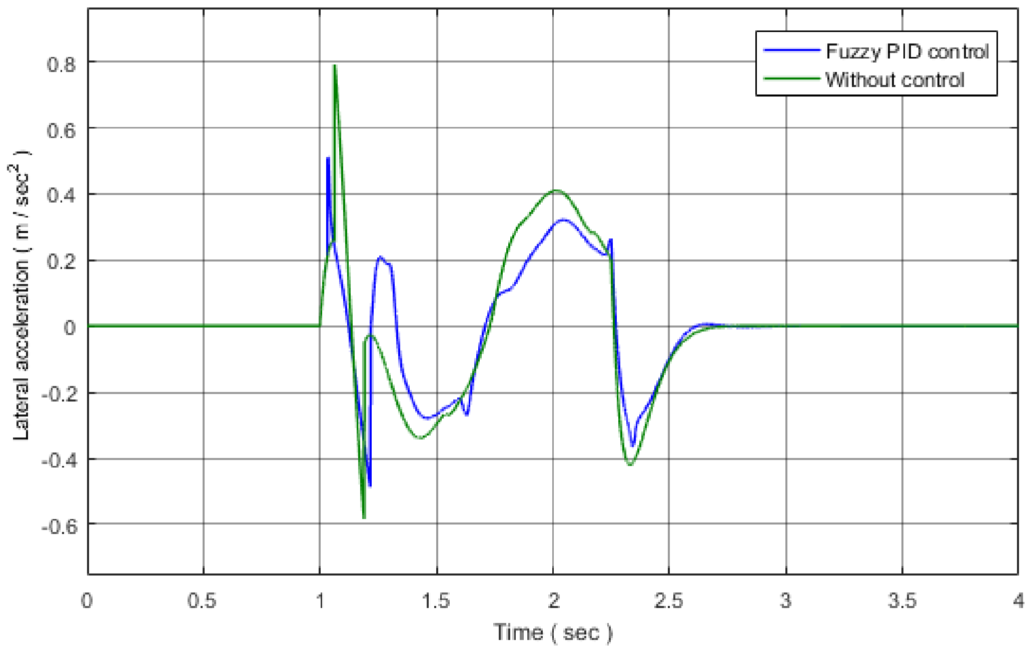


Figure 5.16. Comparison of the lateral acceleration with FPIDC and without FPIDC on a lane change maneuver (Full model).

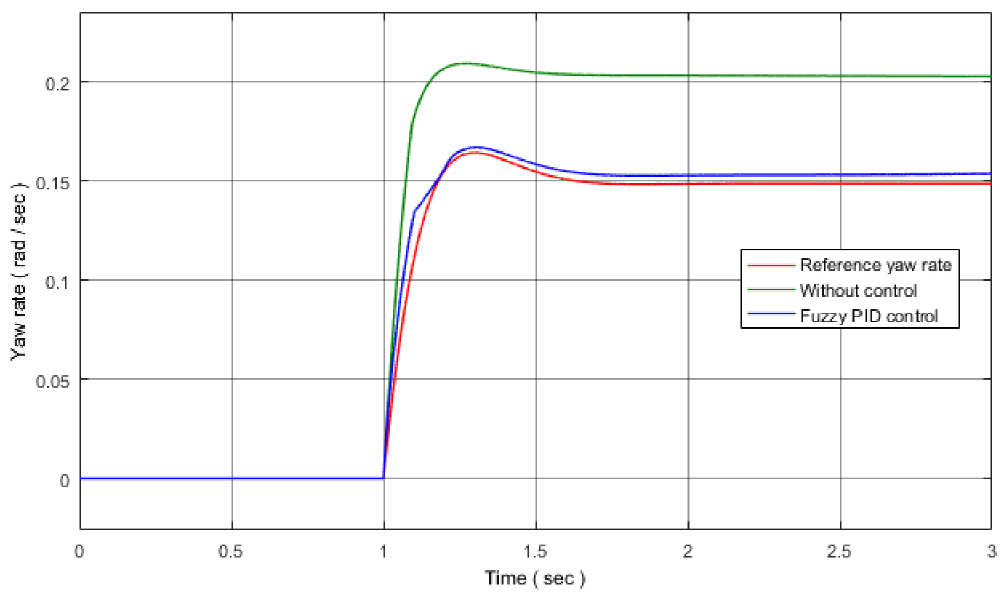


Figure 5.17. The car yaw rate behavior with FPIDC and without FPIDC at a step signal of steering angle (Full model).

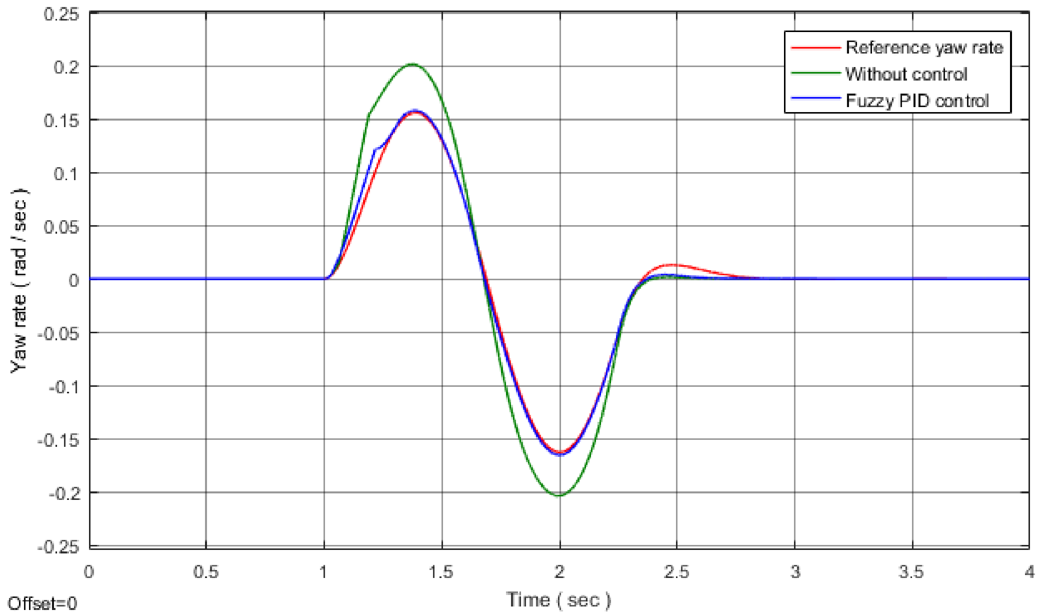


Figure 5.18. The car yaw rate behavior with FPIDC and without FPIDC on a lane change maneuver (Full model).

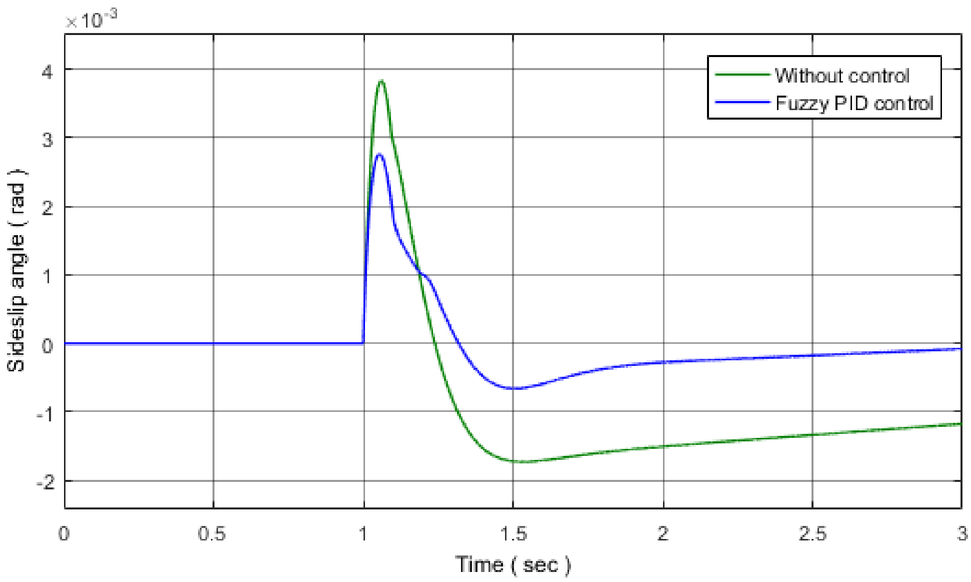


Figure 5.19. The car sideslip angle behavior with FPIDC and without FPIDC at a step signal of steering angle (Full model).

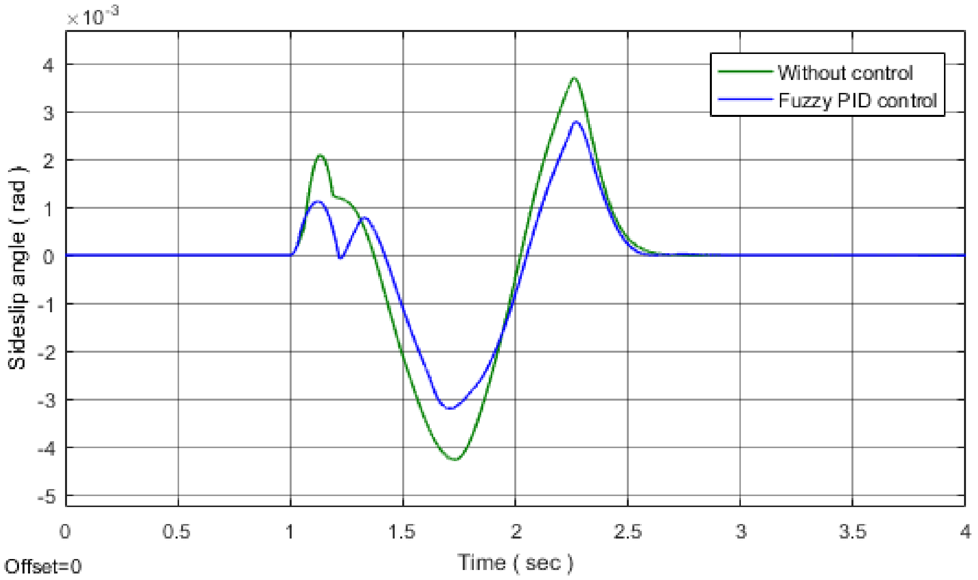


Figure 5.20. The car sideslip angle behavior with FPIDC and without FPIDC on a lane change maneuver (Full model).

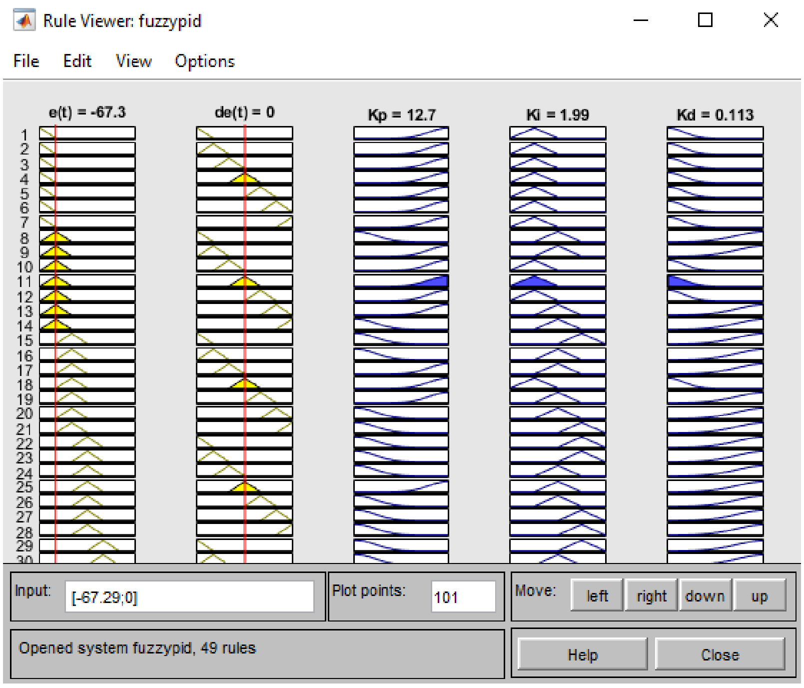
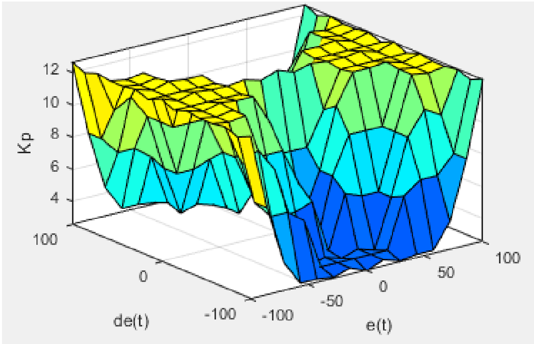
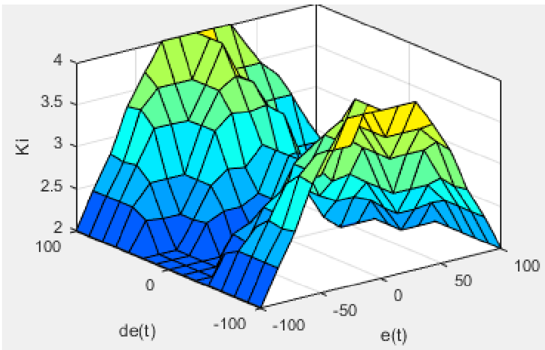


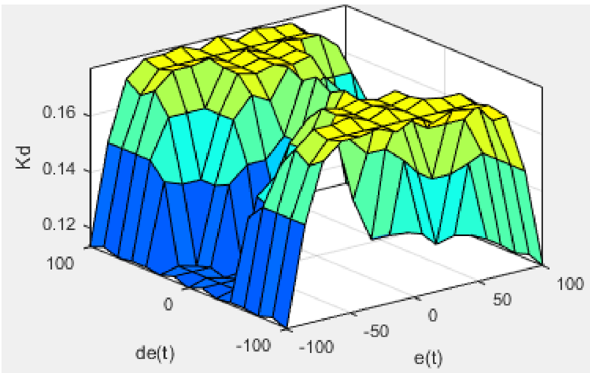
Figure 5.21. Rule viewer of fuzzy PID controller in the case of full vehicle model.



(a)



(b)



(c)

Figure 5.22. The relations of inputs and outputs in the shapes of Cartesian rules surfaces of P, I, and D control variables, (a) P variable; (b) I variable, and (c) D variable.

As mentioned before in the beginning of this thesis, a Fuzzy PID controller was used and designed to improve the performance and stability of the vehicle in the case of planar model and full model. All the simulation results of the two models in figures 5.3 to 5.21 show that, the structures of control systems for vehicle models used in this thesis were able to give a remarkable improvement of the vehicles handling and stability.

After studying and designing the controller (FPIDC) used in this thesis and its influence on the vehicles stability, it is also possible to compare the vehicle performance with this controller and other conventional controller such as PID controller.

Figures 5.23 to 5.30 illustrate the effect of using of both the FPIDC and the conventional PID controller on the stability of the vehicle and its performance in case of planar model at two states of input steering angle which are a step signal and a lane change maneuver. It is obvious that using of Fuzzy PID controller(FPIDC) gave better performance of the vehicle compared to using of conventional PID controller.

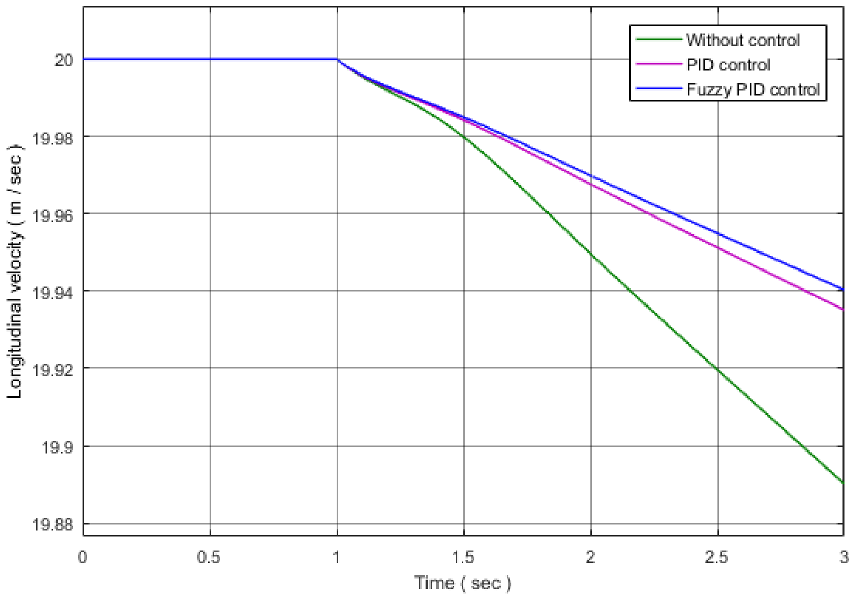


Figure 5.23. The longitudinal velocity behavior using Fuzzy PID and PID controllers at a step signal of steering angle (Planar model).

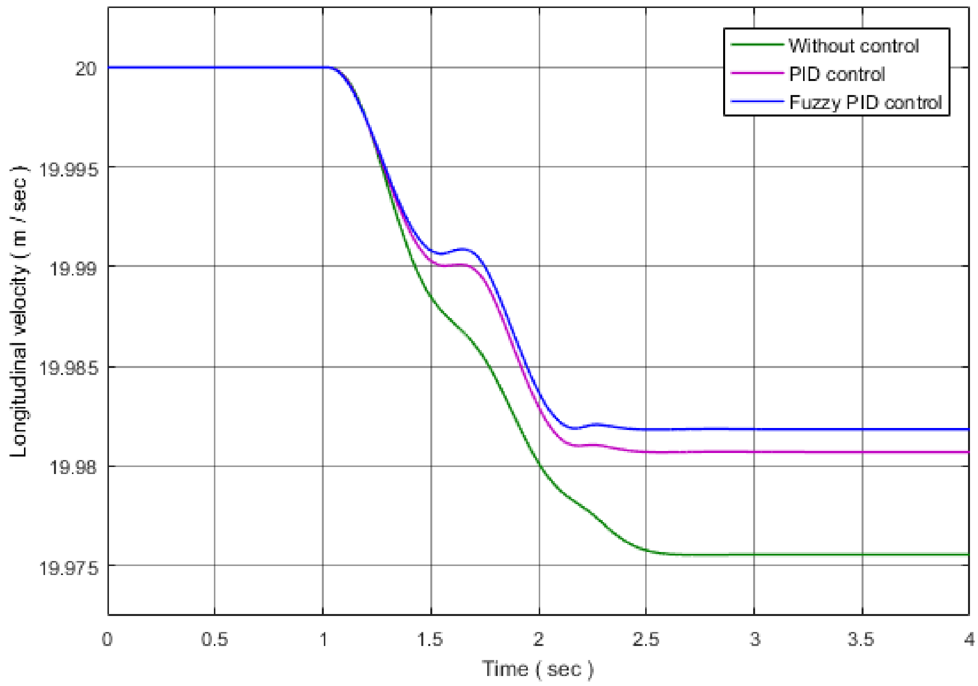


Figure 5.24. The longitudinal velocity behavior using Fuzzy PID and PID controllers on a lane change maneuver (Planar model).

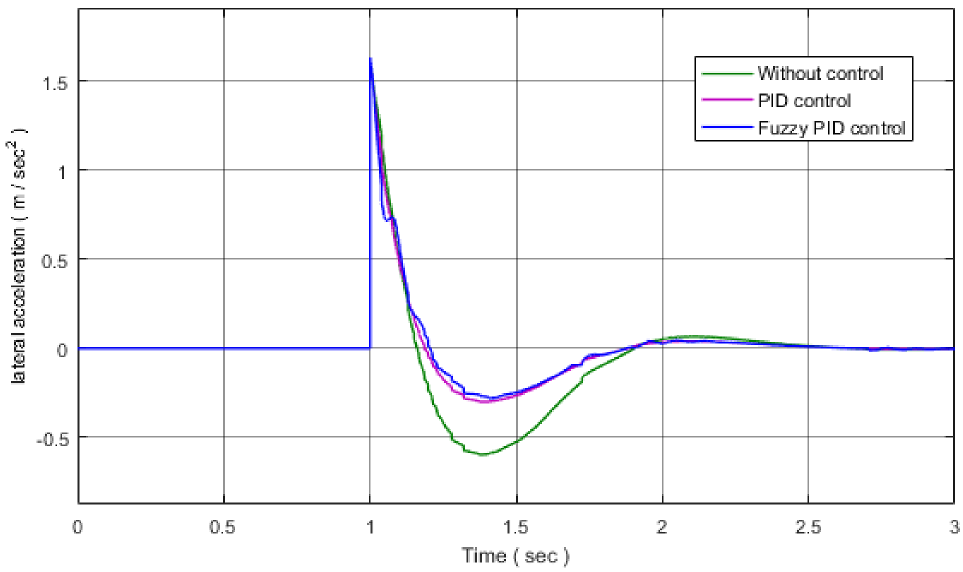


Figure 5.25. The lateral acceleration simulation result using Fuzzy PID and PID controllers at a step signal of steering angle (Planar model).

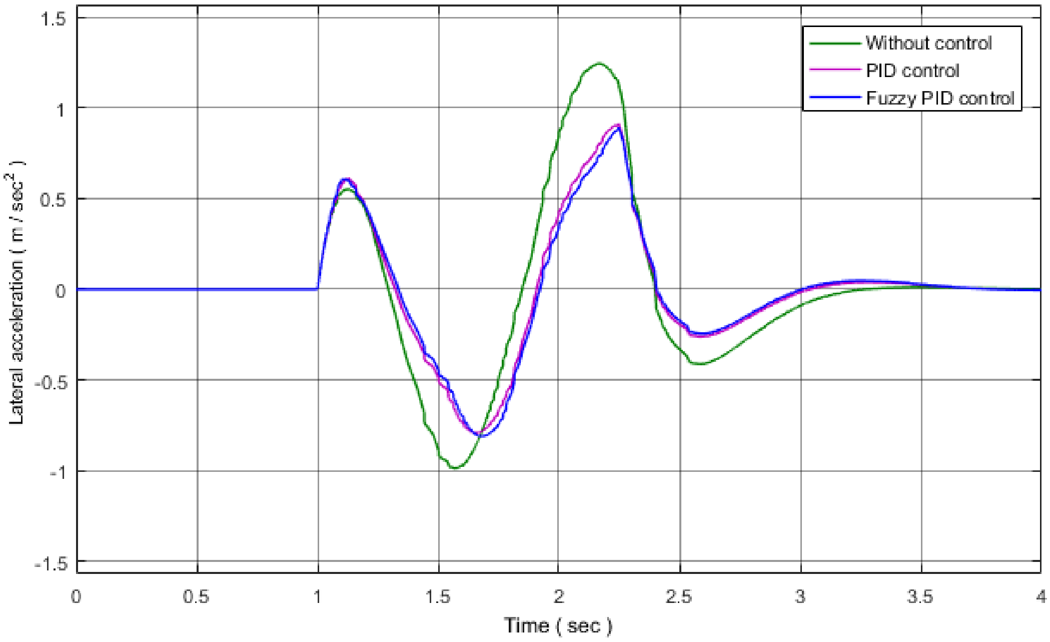


Figure 5.26. The lateral acceleration simulation result using Fuzzy PID and PID controllers on a lane change maneuver (Planar model).

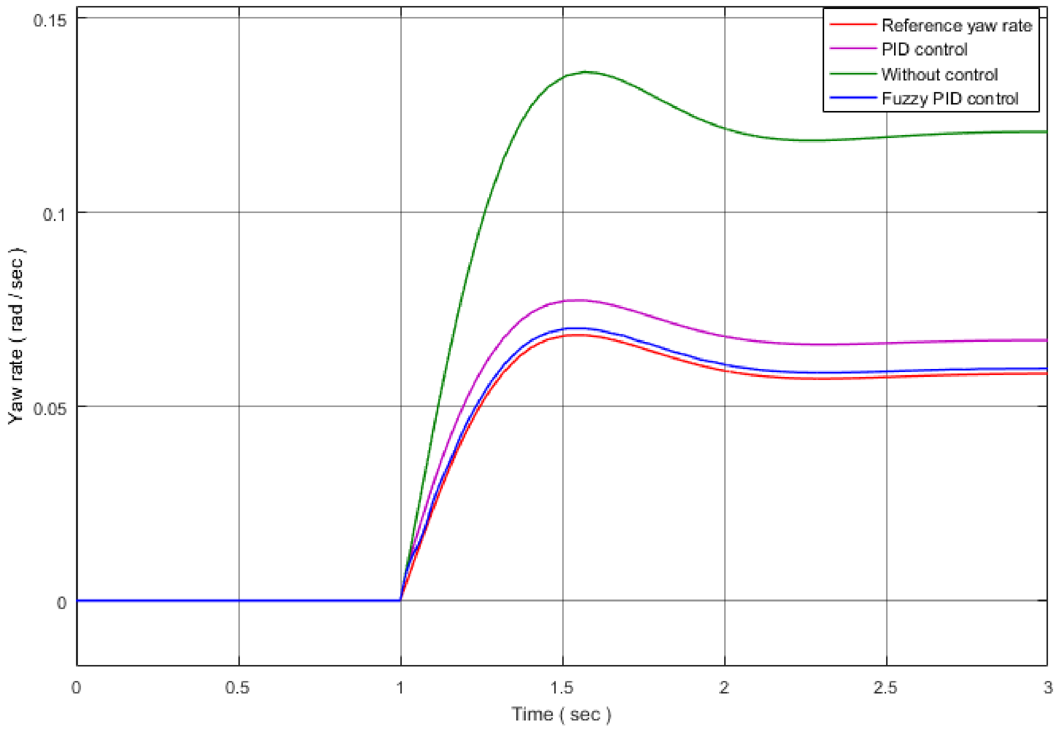


Figure 5.27. The yaw rate Simulation result using Fuzzy PID and PID controllers at a step signal of steering angle (Planar model).

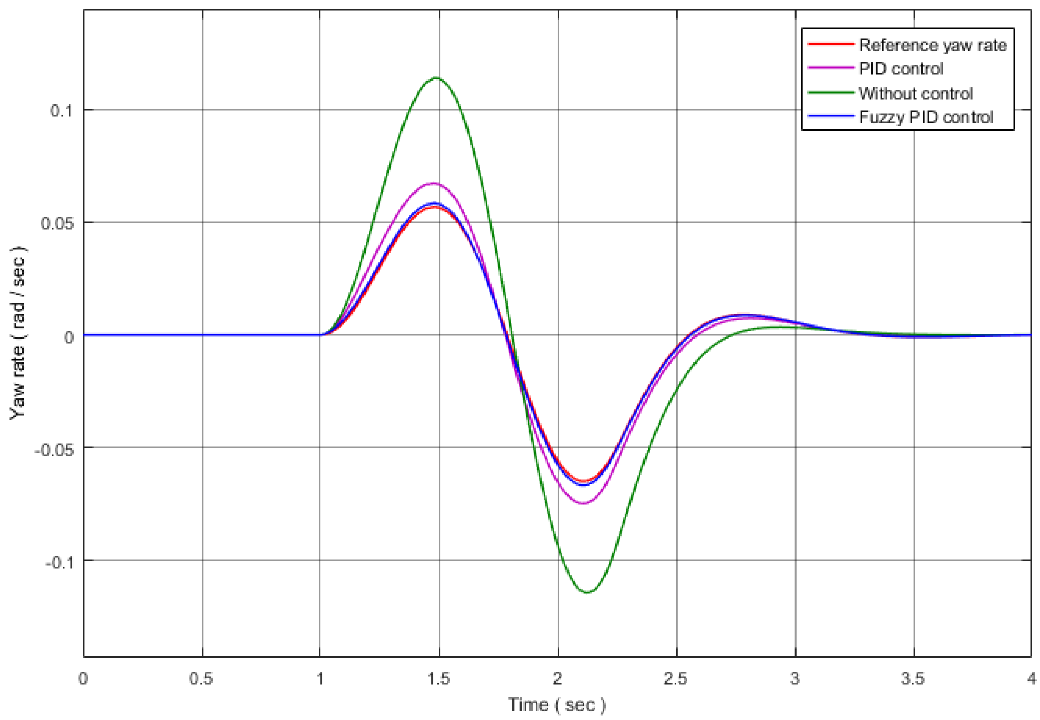


Figure 5.28. The yaw rate Simulation result using Fuzzy PID and PID controllers on a lane change maneuver (Planar model).

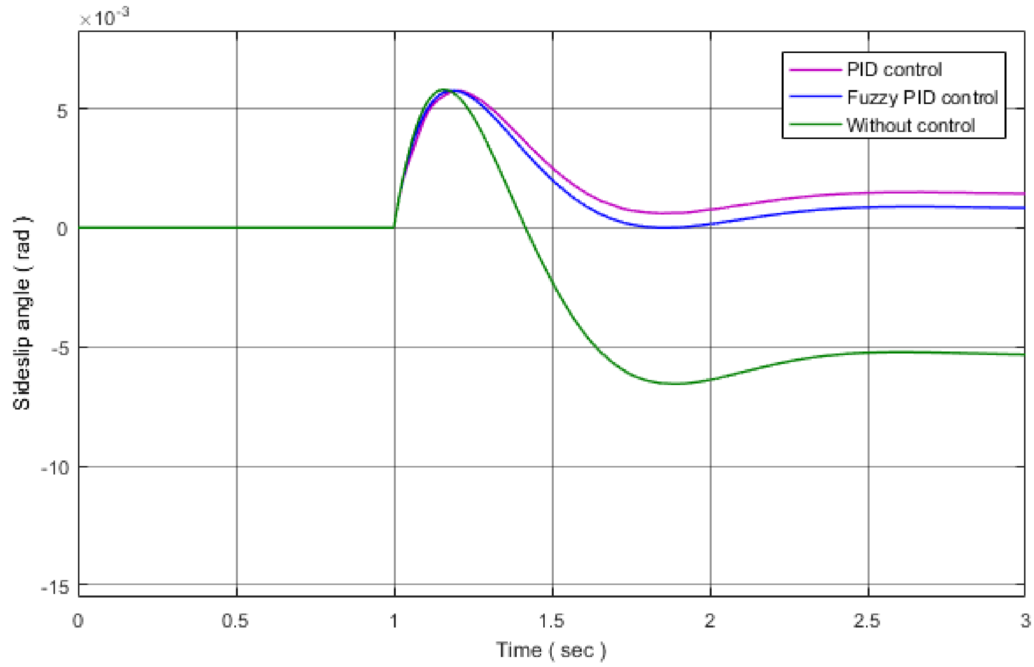


Figure 5.29. The sideslip angle behavior using Fuzzy PID and PID controllers at a step signal of steering angle (Planar model).

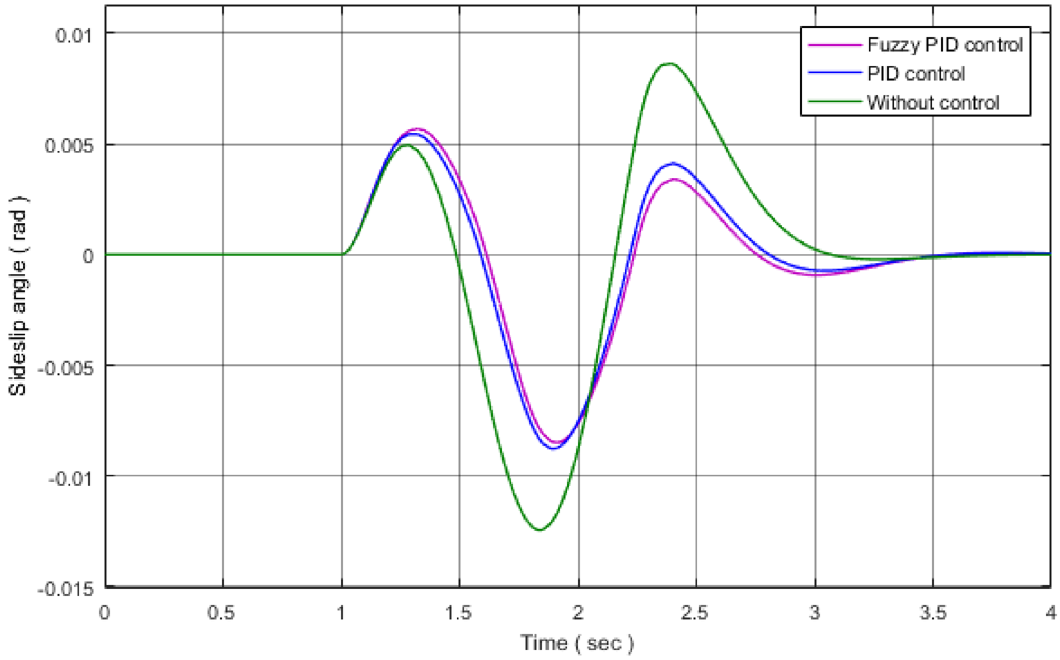


Figure 5.30. The sideslip angle behavior using Fuzzy PID and PID controllers on a lane change maneuver angle (Planar model).

Figures 5.31 to 5.38 show the effect of using of both the PID controller and FPIDC on the stability of the vehicle and its performance for full model at a step signal of steering angle and at a lane change maneuver. It's very clear that using of FPID controller gave better performance of the vehicle compared to using of the conventional PID controller and the effect is clear to understand due to the yaw rate behavior which is shown in figure 5.35 and figure 5.36.

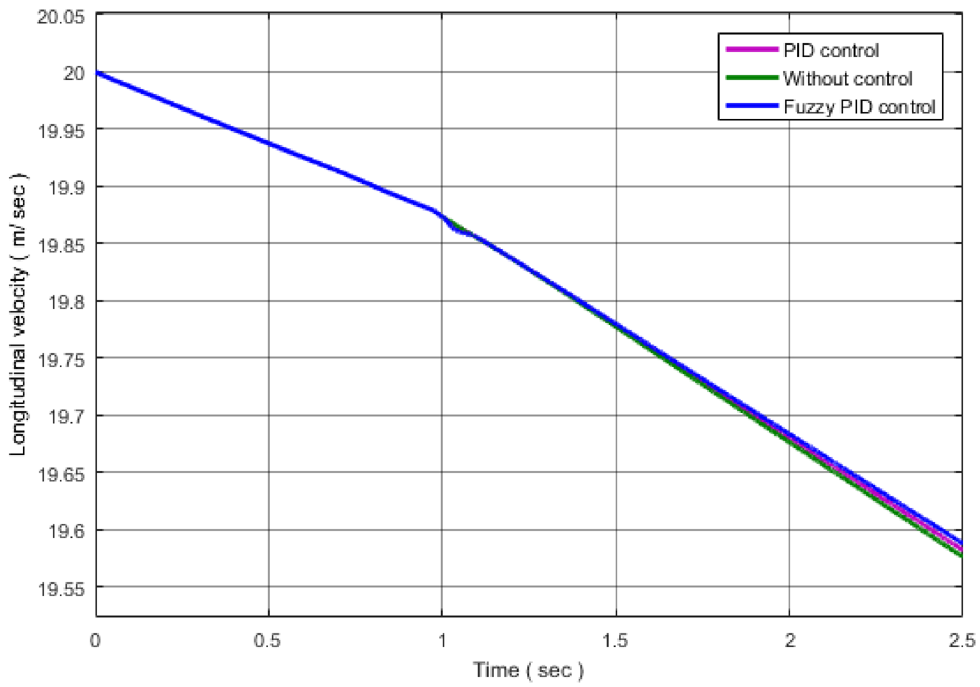


Figure 5.31. The longitudinal velocity behavior using Fuzzy PID and PID controllers at a step signal of steering (Full model).

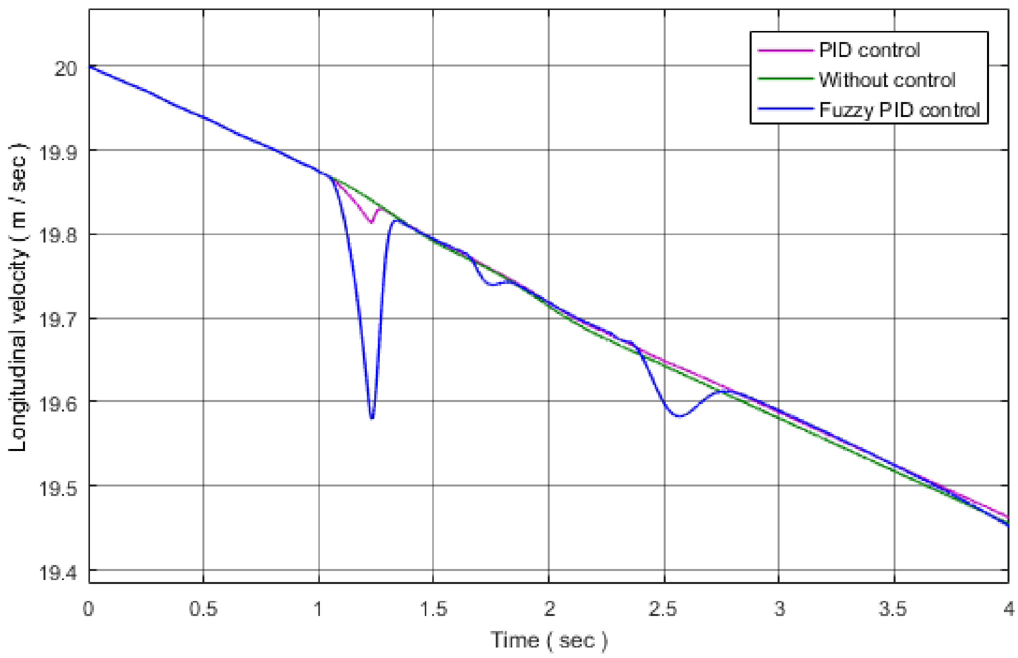


Figure 5.32. The longitudinal velocity behavior using Fuzzy PID and PID controllers on a lane change maneuver (Full model).

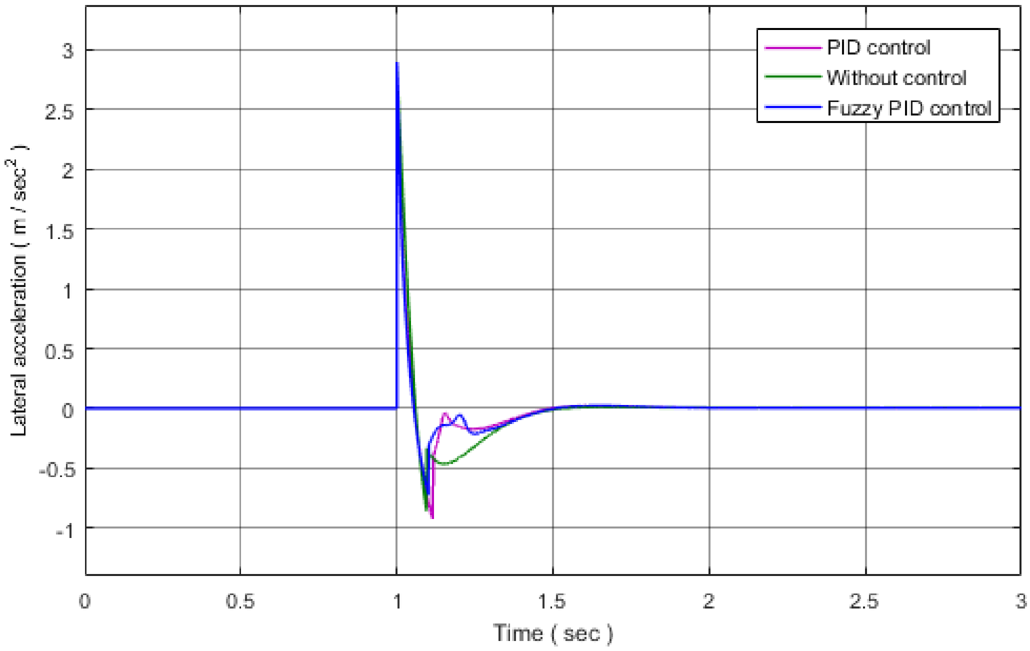


Figure 5.33. Simulation results of lateral acceleration using Fuzzy PID and PID controllers at a step signal of steering (Full model).

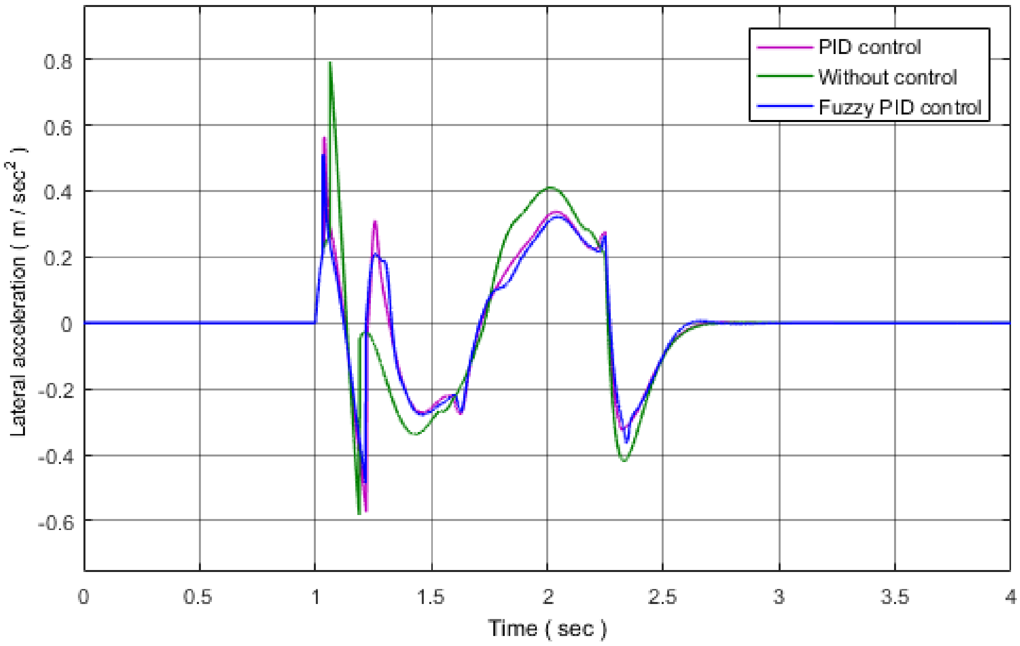


Figure 5.34. Simulation results of lateral acceleration using Fuzzy PID and PID controllers on a lane change maneuver (Full model).

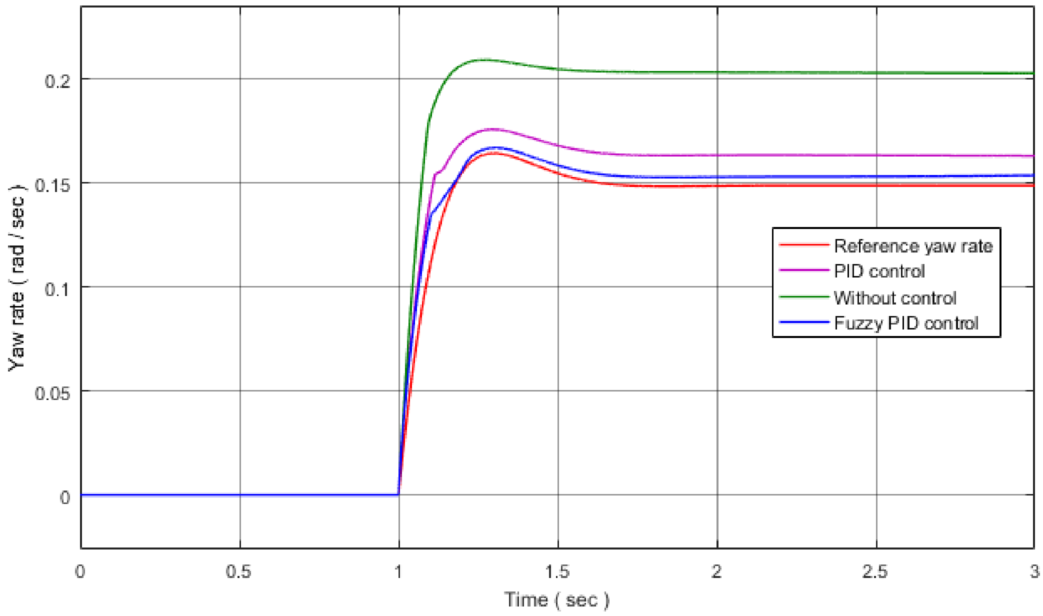


Figure 5.35. Simulation results of yaw rate using Fuzzy PID and PID controllers at a step signal of steering (Full model).

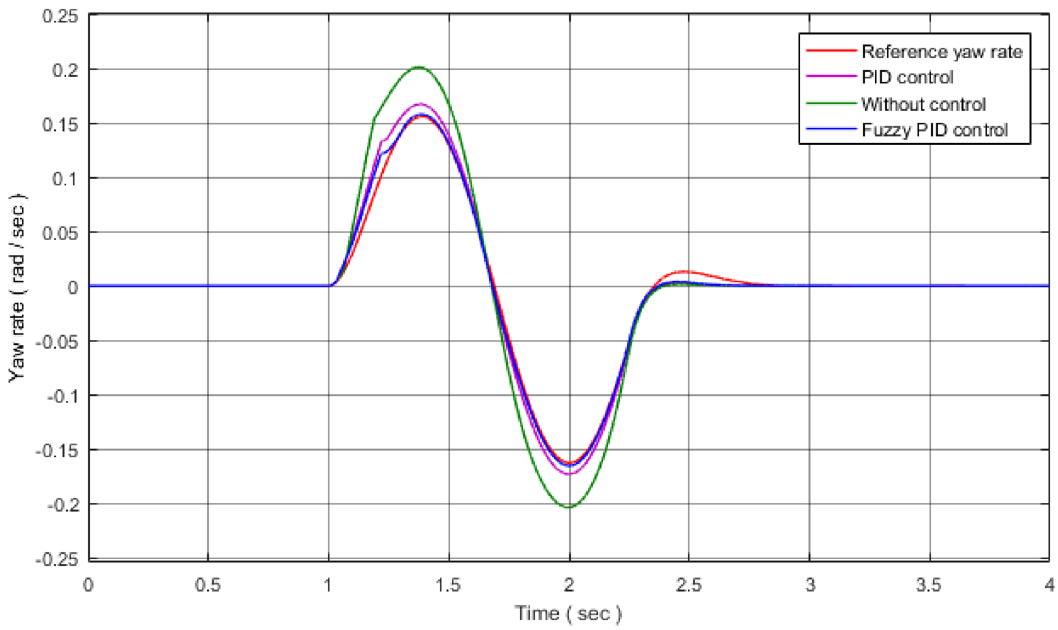


Figure 5.36. Simulation results of yaw rate using Fuzzy PID and PID controllers on a lane change maneuver (Full model).

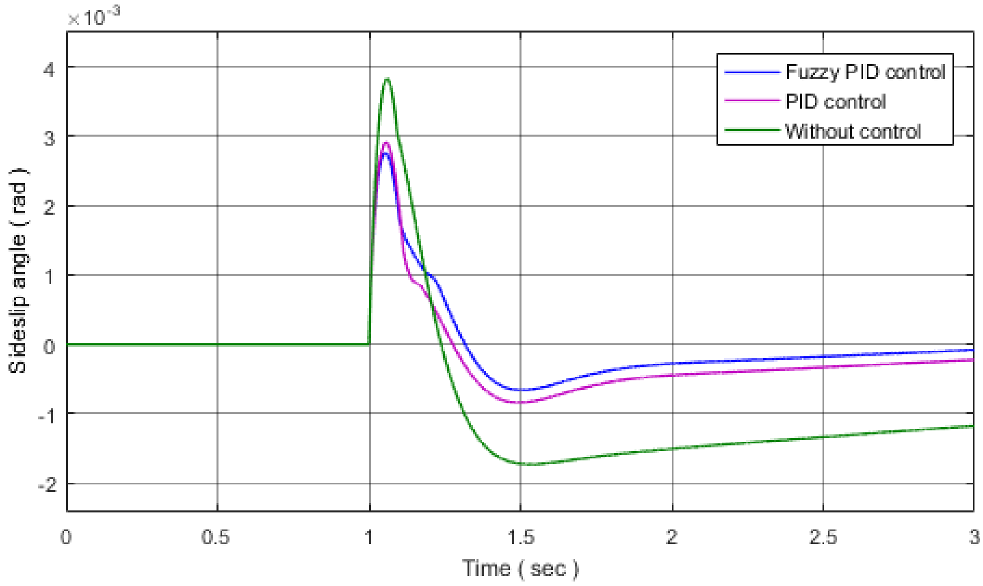


Figure 5.37. Simulation results of sideslip angle using Fuzzy PID and PID controllers at a step signal of steering (Full model).

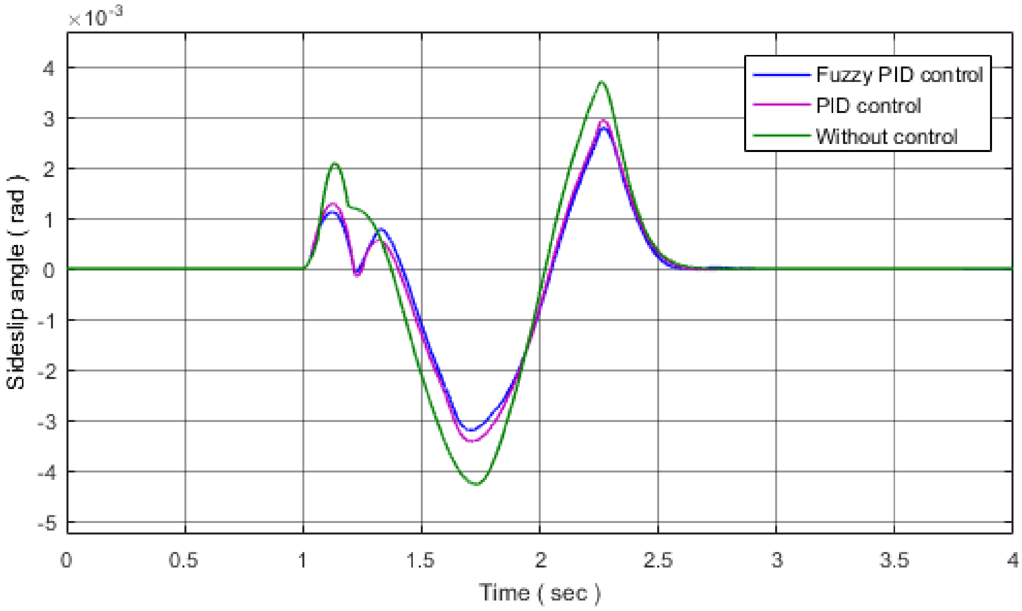


Figure 5.38. Simulation results of sideslip angle using Fuzzy PID and PID controllers at a step signal of steering (Full model).

VI. THESIS CONCLUSIONS AND FUTURE WORK

The main objective of the thesis was to achieve a vehicle dynamic stability improvement. Two vehicle models used in this work: planar vehicle model and full vehicle model. The first model, only the lateral, yaw, and longitudinal motions of the vehicle were considered while in the full vehicle model the heave, roll and pitch movements of the vehicle body are included by addition to yaw, longitudinal, and lateral motions. A fuzzy PID controller FPIDC was used and designed to get better stability of the vehicles, This controller was tested at two states of the front steering angle which are a step signal and a lane change maneuver. Also 2DOF bicycle model (the reference model) is prominently used for design the controller and to analyze the yaw stability control.

At the end of this study, using of a conventional PID controller was compared to using of Fuzzy PID controller.

In next section the main contributions of this thesis are illustrated, also the future work which can be done is presented in final section.

Summary

- In chapter three a two vehicle dynamic models were modeled which are planar and full vehicle models and this is what distinguishes this work compared to other researches related in this field.
- Another contribution of this thesis is using of fuzzy PID controller FPIDC with full vehicle model to improve the stability of vehicles. Some works

are mentioned in chapter one used same controller but in quarter car model or planar vehicle model not in full vehicle models.

- Using of FPIDC in the case of planar vehicle model and full vehicle model gave a significant improvement in the vehicle performance.
- The controller used in this thesis has been tested at a step signal of the front steering angle and on a lane change maneuver.
- Simulation results illustrate the effect of using of both the Fuzzy PID controller and the conventional PID controller on the stability of the vehicle and its performance in case of planar model and full vehicle model. It is obvious that the two controllers gave a good performance of the vehicle but the Fuzzy PID controller was able to achieve the desired stability of the vehicle more than the conventional controller.

6.2. Future work

- This research only focused on the using of FPIDC to improve the performance of the stability of vehicles. Also, using of this controller has also been compared to a conventional PID controller.
- Various of control methods and strategies may be used for vehicle dynamic control and stability such as H_∞ Control, Sliding Mode Control, and Linear Quadratic Regulation (LQR), etc. Then it can possible to compare the performance of vehicles using these control techniques.

REFERENCES

[1] M. K. Aripin, Yahaya Md Sam, Kumeresan A. Danapalasingam, Kemao Peng, N. Hamzah, andM. F. Ismail," A Review of Active Yaw Control System for Vehicle Handling and Stability Enhancement", International Journal of Vehicular Technology, Volume 2014, Article ID 437515, 15 pages.

[2] R. Rajamani, *Vehicle dynamics and control*: Springer Science & Business Media, 2011.

[3] N. M. Ghazaly and A. O. Moaaz, "The future development and analysis of vehicle active suspension system," *IOSR Journal of Mechanical and Civil Engineering*, vol. 11, pp. 19-25, 2014.

[4] R. B. DARUS, "Modeling and control of active suspension for a full car model," University Teknologi Malaysia, 2008.

[5] V. R. Patil, G. B. Pawar, and S. A. Patil, "A Comparative Study between the Vehicles' Passive and Active Suspensions–A Review," 2002.

[6] J. Happian-Smith, An introduction to modern vehicle design: Elsevier, 2001.

[7] H. Zhang and J. Zhang, "Yaw torque control of electric vehicle stability," in *2012 IEEE 6th International Conference on Information and Automation for Sustainability*, 2012, pp. 318-322.

[8] S. Zhang, S. Zhou, and J. Sun, "Vehicle dynamics control based on sliding mode control technology," in *2009 Chinese Control and Decision Conference*, 2009, pp. 2435-2439.

[9] G. Yin, R. Wang, and J. Wang, "Robust control for four wheel independently-actuated electric ground vehicles by external yaw-moment generation," *International Journal of Automotive Technology,* vol. 16, pp. 839-847, 2015.

[10] I. Yoğurtçu, S. Solmaz, and S. Ç. Başlamışlı, "Lateral Stability Control Based on Active Motor Torque Control for Electric and Hybrid Vehicles," in *IEEE European Modelling Symposium*, 2015.

[11] S. Varnhagen, O. M. Anubi, Z. Sabato, and D. Margolis, "Active suspension for the control of planar vehicle dynamics," in *2014 IEEE International Conference on Systems, Man, and Cybernetics (SMC)*, 2014, pp. 4085-4091.

[12] M.-Y. Yoon, S.-H. Baek, K.-S. Boo, and H.-S. Kim, "Map-based control method for vehicle stability enhancement," *Journal of Central South University,* vol. 22, pp. 114-120, 2015.

[13] L. Feiqiang, W. Jun, and L. Zhaodu, "On the vehicle stability control for electric vehicle based on control allocation," in *2008 IEEE Vehicle Power and Propulsion Conference*, 2008, pp. 1-6.

[14] S. Krishna, S. Narayanan, and S. D. Ashok, "Fuzzy logic based yaw stability control for active front steering of a vehicle," *Journal of Mechanical Science and Technology,* vol. 28, pp. 5169-5174, 2014.

[15] H. Zhou, H. Chen, B. Ren, and H. Zhao, "Yaw stability control for in-wheel-motored electric vehicle with a fuzzy PID method," in *The 27th Chinese Control and Decision Conference (2015 CCDC)*, 2015, pp. 1876-1881.

[16] Y. Dei Li, W. Liu, J. Li, Z. M. Ma, and J. C. Zhang, "Simulation of vehicle stability control system using fuzzy PI control method," in *IEEE International Conference on Vehicular Electronics and Safety, 2005.*, 2005, pp. 165-170.

[17] J. Peng, H. He, and N. Feng, "Simulation research on an electric vehicle chassis system based on a collaborative control system," *Energies,* vol. 6, pp. 312-328, 2013.

[18] M. B. N. Shah, A. R. Husain, and A. S. A. Dahalan, "An Analysis of CAN Performance in Active Suspension Control System for Vehicle," *a− a,* vol. 4, p. 2, 2012.

[19] S. Fergani, L. Menhour, O. Sename, L. Dugard, and B. A. Novel, "Full vehicle dynamics control based on LPV/$\mathcal{H}\infty$ and flatness approaches," in *Control Conference (ECC), 2014 European*, 2014, pp. 2346-2351.

[20] H. Xiao, W. Chen, H. Zhou, and J. W. Zu, "Integrated vehicle dynamics control through coordinating electronic stability program and active suspension system," in *2009 International Conference on Mechatronics and Automation*, 2009, pp. 1150-1155.

[21] P. He, Y. Wang, Y. Zhang, and Y. Xu, "Integrated control of semi-active suspension and vehicle dynamics control system," in *2010 International Conference on Computer Application and System Modeling (ICCASM 2010)*, 2010, pp. V5-63-V5-68.

[22] C. Huang, L. Chen, H. Jiang, C. Yuan, and T. Xia, "Fuzzy chaos control for vehicle lateral dynamics based on active suspension system," *Chinese Journal of Mechanical Engineering,* vol. 27, pp. 793-801, 2014.

[23] R. N. Jazar, Vehicle dynamics: theory and application*: Springer Science & Business Media*, 2013.

[24] J. C. Dixon, "Tires, suspension and handling," *Training,* vol. 2014, pp. 12-15, 1996.

[25] A. Visioli, "Tuning of PID controllers with fuzzy logic," *IEE Proceedings-Control Theory and Applications,* vol. 148, pp. 1-8, 2001.

[26] K. H. Ang, G. Chong, and Y. Li, "PID control system analysis, design, and technology," *IEEE transactions on control systems technology,* vol. 13, pp. 559-576, 2005.

[27] C.-T. Chen, *Analog and digital control system design: transfer-function, state-space, and algebraic methods*: Oxford University Press, Inc., 1995.

[28] Q.-G. Wang, B. Zou, T.-H. Lee, and Q. Bi, "Auto-tuning of multivariable PID controllers from decentralized relay feedback," *Automatica,* vol. 33, pp. 319-330, 1997.

[29] K. J. Åström and T. Hägglund, *Advanced PID control*: ISA-The Instrumentation, Systems and Automation Society, 2006.

[30] J. Ziegler and N. Nichols, "Optimum settings for automatic controllers," *Journal of dynamic systems, measurement, and control,* vol. 115, pp. 220-222, 1993.

[31] V. Kumar and A. Mittal, "Parallel fuzzy P+ fuzzy I+ fuzzy D controller: Design and performance evaluation," *International Journal of Automation and Computing,* vol. 7, pp. 463-471, 2010.

[32] M. Dotoli, B. Maione, and B. Turchiano, "Fuzzy-Supervised PID Control: Experimental Results," in *the 1st European Symposium on Intelligent Technologies*, 2001, pp. 31-35.

[33] J. Surdhar and A. S. White, "A parallel fuzzy-controlled flexible manipulator using optical tip feedback," *Robotics and Computer-Integrated Manufacturing,* vol. 19, pp. 273-282, 2003.

[34] G. Thomas, P. Lozovyy, and D. Simon, "Fuzzy robot controller tuning with biogeography-based optimization," in *International Conference on Industrial, Engineering and Other Applications of Applied Intelligent Systems*, 2011, pp. 319-327.

[35] C.-C. Lee, "Fuzzy logic in control systems: fuzzy logic controller. II," *IEEE Transactions on systems, man, and cybernetics,* vol. 20, pp. 419-435, 1990.

[36] T. Terano, K. Asai, and M. Sugeno, *Fuzzy systems theory and its applications*: Academic Press Professional, Inc., 1992.

[37] G. Feng, "A survey on analysis and design of model-based fuzzy control systems," *IEEE Transactions on Fuzzy systems,* vol. 14, pp. 676-697, 2006.

[38] Yahaya Md Sam, Hazlina Selamat, M.K Aripin,and Muhamad Fahezal Ismail," Vehicle Stability Enhancement Based on Second Order Sliding Mode Control",*IEEE International Conference on Control System, Computing and Engineering*, 23 - 25 Nov. 2012, Penang, Malaysia.

[39] DIRK E. SMITH and JOHN M. STARKEY," Effects of Model Complexity on the Performance of Automated Vehicle Steering Controllers: Model Development, Validation and Comparison",*Vehicle System Dynamics*, 24 (1995), pp. 163-181.

[40] C. I. Chatzikomis · K. N. Spentzas," Comparison of a vehicle equipped with Electronic Stability Control (ESC) to a vehicle with Four Wheel Steering 4WS", *Forsch Ingenieurwes,* (2014) 78:13–25.

[41] Shingo Tamaru, Jinxin Zhuo, Masahiro Oyam and Qiang Wang," A method to improve the stability of adaptive steerin driver–vehicle systems",*Artif Life Robotics* (2011) 16:343–347.

[42] YIN GuoDong, JIN XianJian, QING ZhiYong & BIAN ChenTong," Lateral stability region conservativeness estimation and torque distribution for FWIA electric vehicle steering",*SCIENCE CHINA, Technological Sciences,* Vol.58 No.4: 669–676, April 2015.

[43] Wongun Kim, Kyongsu Yi, and Jongseok Lee," Drive control algorithm for an independent 8 in-wheel motor drive vehicle", *Journal of Mechanical Science and Technology*, Vol. 25 (6) 1573-1581, (2011).

[44] S. Krishna, S. Narayanan and S. Denis Ashok," Fuzzy logic based yaw stability control for active front steering of a vehicle", *Journal of Mechanical Science and Technology*, Vol. 28 (12) 5169-5174,(2014).

[45] Y. E. KO and C. K. SONG ," VEHICLE MODELING WITH NONLINEAR TIRES FOR VEHICLE STABILITY ANALYSIS",*International Journal of Automotive Technology,* Vol. 11, No. 3, pp. 339−344, (2010).

[46] Laiq Khan, Shahid Qamar, and M. Umair Khan," Comparative Analysis of Adaptive NeuroFuzzy Control Techniques for Full Car Active Suspension System",*Arab J Sci Eng*, 39:2045–2069, (2014).

[47] Junmin Wang, Javier M. Solis, and Raul G. Longoria," On the Control Allocation for Coordinated Ground Vehicle Dynamics Control Systems",*Proceedings of the 2007 American Control Conference,* July 11-13, 2007, New York City, USA.

[48] Kangwon Lee, Youngwoo Kim and Jinhee Jang," A study of integrated chassis control Algorithm with Brake Control and Suspension Control Systems for vehicle stability and handling performance",*ICROS-SICE International Joint Conference*, August 18-21, 2009, Japan.

[49] Wuwei Chen, Hansong Xiao, Liqiang Liu, Jean W. Zu and HuiHui Zhou," Integrated Control of Vehicle System Dynamics: Theory and Experiment, Advances in Mechatronics",InTech,2011.

[50] Rana Raouf Hasan Farag"Active Neuro-Fuzzy Integrated Vehicle Dynamics Controller to improve the vehicle handling and stability at complicated maneuvers" PhD thesis , July 2013.

[51] JORGE GÓMEZ FERNÁNDEZ "A Vehicle Dynamics Model for Driving Simulators" Master thesis, Chalmers university, 2012.

[52] Kambiz Arab Tehrani and Augustin Mpanda," PID Control Theory, Introduction to PID Controllers - Theory, Tuning and Application to Frontier Areas", ",InTech, February, 2012.

[53] Abdussalam Ali Ahmed and Başar Özkan, “Stability And Dynamic Control Of Four In-Wheel Motored Vehicle Using Fuzzy PID Controller”, International Journal of Mechanical & Mechatronics Engineering IJMME-IJENS, Vol:16 No:03. June 2016.

[54] Dhanalakshmi R, “Regulation of frequency and generated power on isolated renewable hybrid power system using intelligent techniques ”, PhD thesis, Anna University. August 2013.

Apendix A

CURRICULUM VITAE

PERSONAL INFORMATION
Surname, Name: Abdussalam Ali Ahmed Omar
Date and Place of Birth: 29.07.1979
Marital Status: Married
Phone: +90 5464771711
+218922841583
Email: abd112890@yahoo.com
abd112890@gmail.com

EDUCATION

Degree	Institution	Year of Graduation
Ph.D.	**…………**	
M.Sc.	**Mechanical Engineering**	**Fall 2005**
B.Sc.	**Mechanical Engineering**	**Spring 2002**
High School	**Mosab Bin Omair**	**1996-1997**

- **FOREIGN LANGUAGES: English and Beginner Turkish.**
- **Supervised many graduate projects.**

Publications:

[1] **Abdussalam Ali Ahmed** and Başar Özkan, "Evaluation Of Effect Of In-Wheel Electric Motors Mass On The Active Suspension System Performance Using Linear Quadratic Regulator Control Method", International Journal of Engineering Research & Technology (IJERT), Vol. 4 Issue 01,January-2015.

[2] **Abdussalam Ali Ahmed** and Başar Özkan, “Analysis of effect of in-wheel electric motors mass on passive and active suspension systems”, ARPN Journal of Engineering and Applied Sciences, Vol. 10, No. 14, August 2015.

[3] **Abdussalam Ali Ahmed** and Başar Özkan, “Stability And Dynamic Control Of Four In-Wheel Motored Vehicle Using Fuzzy PID Controller”, International Journal of Mechanical & Mechatronics Engineering IJMME-IJENS, Vol:16 No:03. June 2016.

[4] **Abdussalam Ali Ahmed** and Başar Özkan, “ Using of Fuzzy PID Controller to Improve Vehicle Stability for Planar Model and Full Vehicle Models”, International Journal of Applied Engineering Research ISSN 0973-4562 Volume 12, Number 5 (2017) pp. 671-680.

Apendix B

MATLAB code for planar vehicle model

```
clc
clear all
close all
controlfuzzypid = readfis('controlfuzzypid.fis');
%Full vehicle parameters:
a=0.968;
b=1.392;
L=a+b;
d=2*0.64;
m=1030;
g=9.81;
reff=0.303;
J=4.07;
mu=0.9;
mu11=0.4;
Fzf=(m*g*Lr)/(Lf+Lr)/2;
Fzr=(m*g*Lf)/(Lf+Lr)/2;
C1=52526;
C2=29000;
I=2765;
v=20;
%sreerong angle values
df=2*(pi/180);
%wheel torque
Twf=0;
Twr=0;
Caf = 49100;
Car = 64500;
k=(m*(b*Car-a*Caf))/((a+b)*(Caf*Car);
```

MATLAB code for full vehicle model

```
clc
clear all
close all
controlfuzzypid = readfis('controlfuzzypid.fis');
```

```
%Vehicle physical parameters
a=0.968;
b=1.392;
L=a+b;
h=0.505;
d=2*0.64;
Rw=0.303;
Iz=1087.8;
Ixz=100;
Iy=1058.4;
Ix=300;
Iw=4.07;
m=1030;
ms=810;
g=9.81;
Tr =0;
Tf =0;
fr =0.015; v0 = 20;
%Input steering angle
df=2*(pi/180);
%Suspension system parameters
mu1=26.5;
mu2=26.5;
mu3=24.4;
mu4=24.4;
Kt1=13800;Kt2=13800;Kt3=13800;Kt4=13800;
Ks1=20600;Ks2=20600;Ks3=15200;Ks4=15200;
c1=1570;c2=1570;c3=1760;c4=1760;
Kar = 6695;
Kaf = 6695;
%Dugoff's tire model
C1=52526;
C2=29000;
mu=0.9;
mu11=0.4;

%Vehicle bicycle model
Caf=49100;
Car=64500;
k=(m*(b*Cr-a*Cf))/((a+b)*(Cf*Cr))
```

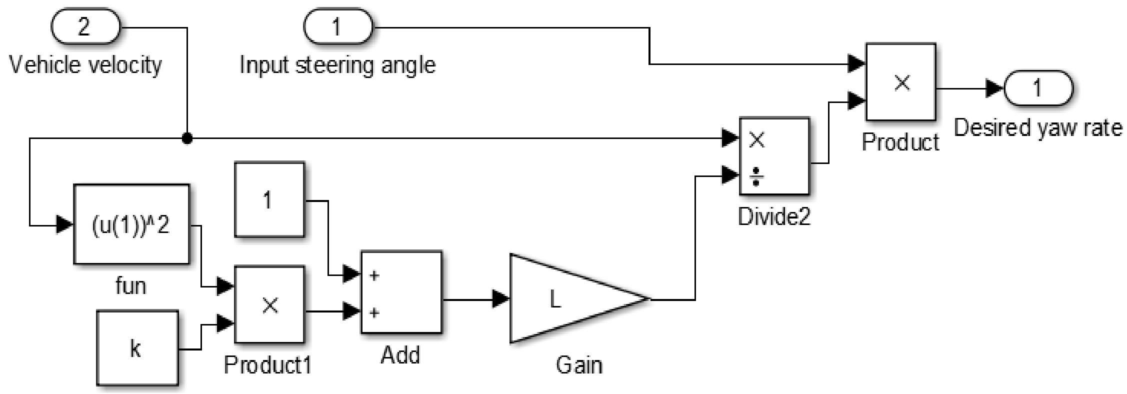
Apendix C

- **Vehicle reference model:**

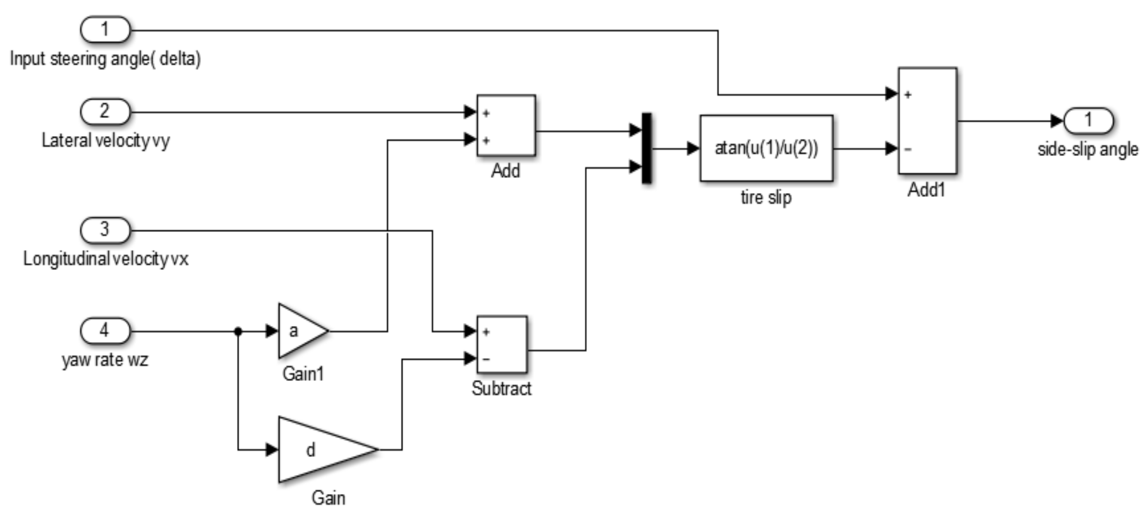
$$w_{zd} = \frac{v_x}{(L)\left(1 + k\ v_x^2\right)}\delta$$

$$k = m(L_r C_{ar} - L_f C_{af}) / LC_{af} C_{ar}$$

Where w_{zd} is the desired yaw rate, k is the understeer parameter, L is the wheel base, v_x is the vehicle velocity and C_{af}, C_{ar} are longitudinal and lateral stiffness of front and rear tire.



- **Side slip angle in Matlab Simulink:**



- Dugoff tire model simulink model: