Integration of Active Chassis Control Systems for Improved Vehicle Handling Performance

by

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Abstract

This thesis investigates the principle of integration of vehicle dynamics control systems by proposing a novel control architecture to integrate the brake-based electronic stability control (ESC), active front steering (AFS), normal suspension force control (NFC) and variable torque distribution (VTD).

A nonlinear 14 degree of freedom passive vehicle dynamics model was developed in Matlab/Simulink and validated against commercially available vehicle dynamics software CarSim. Dynamics of the four active vehicle control systems were developed. Fuzzy logic and PID control strategies were employed considering their robustness and effectiveness in controlling nonlinear systems. Effectiveness of active systems in extending the vehicle operating range against the passive ones was investigated.

From the research, it was observed that AFS is effective in improving the stability at lower lateral acceleration (latac) region with less interference to the longitudinal vehicle dynamics. But its ability diminishes at higher latac regions due to tyre lateral force saturation. Both ESC and VTD are found to be effective in stabilising the vehicle over the entire operating region. But the intrusive nature of ESC promotes VTD as a preferred stability control mechanism at the medium latac range. But ESC stands out in improving stability at limits where safety is of paramount importance. NFC is observed to improve the ability to generate the tyre forces across the entire operating range.

Based on this analysis, a novel rule based integrated chassis control (ICC) strategy is proposed. It uses a latac based stability criterion to assign the authority to control the stability and ensures the smooth transition of the control authority amongst the three systems, AFS, VTD and ESC respectively. The ICC also optimises the utilisation of NFC to improve the vehicle handling performance further, across the entire operating regions. The results of the simulation are found to prove that the integrated control strategy improves vehicle stability across the entire vehicle operating region.

Publications

1. **Rengaraj, C.**, and Crolla, D.A., (2011). ‘Integrated Chassis Control to Improve Vehicle Handling Dynamics Performance’, SAE 2011-01-0958.

2. **Rengaraj, C.**, Crolla, D.A., Wheatley. A. and Hilton, G., (2009), ‘Integration of Active Driveline, Active Steering, Active Suspension and Active Brake for an Improved Vehicle Dynamics Performance’ , 21st International Symposium on Dynamics of Vehicles and on Roads and Tracks, International Association of Vehicle System Dynamics

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4. **Rengaraj, C.**, Crolla, D.A., Wheatley. A. and Adgar, A, (2007), ‘Integration of Brake Based Vehicle Stability Control and Active Suspension for Improved Vehicle Handling’, Automotive Congress, European Automobile Engineers Corporation.

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Notations

M_x	Moment about the roll axis at CoG of vehicle
M_y	Moment about the pitch axis at CoG of vehicle
M_z	Moment about the yaw axis at CoG of vehicle
I_x	Sprung mass roll moment of inertia at CoG of vehicle
I_x	Sprung mass pitch moment of inertia at CoG of vehicle
I_z	Sprung mass yaw moment of inertia at CoG of vehicle
V_x	Sprung mass longitudinal velocity at CoG
V_y	Sprung mass lateral velocity at CoG
V_z	Sprung mass vertical velocity at CoG
m_s	Vehicle sprung mass
m_{us_fl}	Vehicle un-sprung mass at front left corner
m_{us_fr}	Vehicle un-sprung mass at front right corner
m_{us_rl}	Vehicle un-sprung mass at rear left corner
m_{us_rr}	Vehicle un-sprung mass at rear right corner
m_v	Total vehicle mass
F_{xi}	Longitudinal tyre force on i^{th} tyre
F_{yi}	Lateral tyre force on i^{th} tyre
F_{zi}	Vertical tyre force on i^{th} tyre
i =	{front left, front right, rear left, rear right}
M_{zc}	Corrective yaw moment
K_t	Tyre vertical stiffness
K_s	Suspension spring stiffness

C_s	Suspension damper stiffness
Z_s	Sprung mass vertical displacement
$\dot{Z}_s$	Sprung mass vertical velocity
Z_{us}	Unsprung mass vertical displacement
$\dot{Z}_{us}$	Unsprung mass vertical velocity
F_s	Suspension vertical force
F_a	Active suspension force
F_t	Tyre vertical force
λ	Tyre longitudinal slip ratio
α	Tyre lateral slip angle
ω_w	Wheel angular velocity
T_b	Braking Torque
T_d	Driving Torque
R_w	Dynamic wheel radius
a_x	Longitudinal acceleration at CoG
a_y	Lateral acceleration at CoG
a_z	Vertical acceleration at CoG
g	Gravitational acceleration
θ	Vehicle pitch angle at CoG
ϕ	Vehicle roll angle at CoG
ψ	Vehicle yaw angle at CoG
$\dot{\theta}$	Vehicle pitch rate at CoG
$\dot{\phi}$	Vehicle roll rate at CoG
$\dot{\psi}$	Vehicle yaw rate at CoG
t_f	Vehicle front track width
t_r	Vehicle rear track width
l_f	Distance of vehicle CoG from front axle
l_r	Distance of vehicle CoG from rear axle

C_f	Front tyre cornering stiffness
C_r	Rear tyre cornering stiffness
F_n	Tyre normal force
δ_f	Front steering angle at wheels
δ_d	Front steering angle at wheels by driver
δ_c	Corrective steering angle by active suspension

Abbreviations

ABS	Anti-lock Braking System
AFS	Active Front Steering
4WS	Active Four Wheel Steering
ARS	Active Rear Steering
AYC	Active Yaw Control
CoG	Centre of Gravity
DoF	Degree of Freedom
DSC	Dynamic Stability Control
FWD	Front Wheel Drive
FLC	Fuzzy Logic Control
LSD	Limited Slip Differential
NFC	Normal Force Control
NLVM	Nonlinear Vehicle Model
PID	Proportional Integral Derivative
RMD	Roll Moment Distribution
RWD	Rear Wheel Drive
SFC	Suspension Force Control
SMC	Sliding Mode Control
TCS	Traction Control System
ICC	Integrated Chassis Control
GCC	Global Chassis Control
UCC	Universal Chassis Control
DoF	Degree of Freedom

Chapter 1

Introduction

1.1 Vehicle Dynamics

The dawn of the motor vehicle was set-in when Nicholas Joseph Cugnot built a three wheeled steam-driven vehicle in 1769 (Richard Fine, 1969). But the credit of inventing the first practical automobiles powered by gasoline engines in 1886 should go to Karl Benz and Gottlieb Daimler. Over the decades automobiles were developed by many other pioneers. In 1908 Henry Ford manufactured the first 'Model T' at the General Motors Corporation. Automotive engineers in the early 20th century were mainly focussing on the invention of new designs to improve the vehicle performance, comfort and reliability at higher speeds.

Following the achievement of an automobile capable of operating at higher speeds, soon the research was focussed on the high speed dynamic behaviour of those vehicles, particularly during turning and braking. Many engineers such as Fredrick William Lancaster, Segel, Olley have contributed to the early development of automotive dynamics (Gillespie, 1992). And, finally, this gave birth to the field of vehicle dynamics. During the second half of the 20th century, dynamics played an important role in vehicle design and development.

Research in the field of vehicle dynamics mainly focuses on the three primary forces generated at each of the four tyre-road contact patches, in the case of a four wheeled vehicle. The three forces acting at the contact tyre patch are oriented at three different directions, longitudinal, lateral and vertical. The longitudinal forces are generated due to the application of braking and steering torques at the wheel hub. The vertical forces are created due to the vehicle suspension systems. Apart from these three primary forces there are three moments acting on the tyre-road contact patch. Gaining an understanding of these forces and moments is essential in the attempt to address vehicle dynamics problems. As these forces and moments play a major role in defining the dynamic performance of a vehicle, it is essential to understand the mechanics of force generation by the tyre. As these forces and moments are generated at the tyre-road contact patch, it is not only the tyre, but the road and the environment that plays a crucial role in defining the dynamic performance of a vehicle. For example, a dynamically well designed vehicle's performance may change depending on the road surface conditions such as a dry, wet, icy or a gravel road.

When the vehicle responds to the driver's input such as steering, how well it can cope up with the input and respond is called the vehicle's 'directional response' or' handling'. Generally a vehicle's handling response can be characterised by dynamic parameters such as lateral acceleration, yaw rate, side-slip angle etc.

Every vehicle or vehicle design has its own comfort zone (in the driver's point of view – how safe/confident a driver feels) or performance zone that can be defined /characterised in terms of these vehicle dynamic parameters. Pushing the vehicle out of its performance zone will make the vehicle behave unpredictably, especially to the driver's input. A vehicle can be pushed out of its performance window during various situations, such as a driver's input to the vehicle which is not suitable for the road conditions. When a vehicle is pushed out of its performance window, the values of these vehicle dynamic parameters will grow and spiral out and the vehicle will move from its confident zone into the critical zone and will finally end up in a dangerous situation, such as a collision with other external objects (other vehicles, tree, buildings etc). This behaviour of the vehicle is basically called by the vehicle dynamics community as passive in nature. This problem was addressed by the research community in vehicle dynamics which led to the invention of the field of active vehicle dynamics.

1.2 Vehicle dynamics and Control

The rapid development of electronics, sensor and actuator technologies had helped the researchers to address the problem of passive vehicle dynamics (Junje, Crolla et al, 2006). The development in the field of digital electronics has been applied to various automobile subsystems for nearly five decades (Fodor et al, 1998). Initially digital controls were used to improve vehicle fuel economy, but later, applied to improve the dynamic performance of vehicles.

A generalised concept of active vehicle dynamic control can be defined as shown in figure 1.1. With reference to this concept, the driver's inputs are applied to a vehicle model and to a reference model (normally a linear model whose performance is predictable to driver's input). Then the response of the vehicle model is compared with that of the reference model. The output is then used by a controller and its actuator to force the vehicle response towards the linear response of the reference model. Application of this concept to major vehicle subsystems alters the overall vehicle dynamics performance.

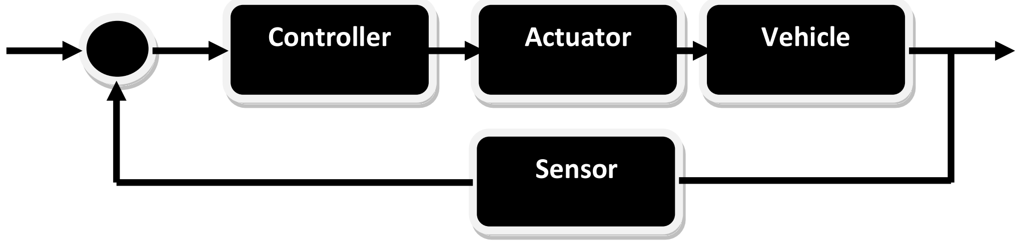

Figure 1.1 - Generalised Control Concepts of Active Vehicle Dynamics

Vehicle dynamic control systems can be categorised either based on the direction of the vehicle dynamics they affect or based on the function of the vehicle subsystems they control. In terms of the direction of control on vehicle dynamics, they can be categorised into three areas: Longitudinal control, Lateral Control and Vertical Control (Junje, 2006). Based on the function of the vehicle subsystems, they can be categorised as follows, active suspension, active braking, active driving, active steering, active power train etc (Rengaraj et al, 2011). For the purpose of this thesis the functional approach towards the active vehicle dynamics categorisation is followed and the focus will be limited to those active systems that influence the vehicle handling dynamics.

The active chassis control systems that influence the handling of a vehicle are generally called vehicle stability control systems. As mentioned earlier in this chapter the vehicle stability can be controlled by controlling the longitudinal, lateral and vertical tyre forces. These forces in turn can be controlled by changing the characteristic behaviour of the respective functional systems. A vast research literature is available under the subject of active vehicle dynamic control systems. It is a good idea to first outline the major strategies used by researchers to influence the vehicle handling dynamics before embarking on the analysis of what has been done in this field.

The abundant research literature available in this field highlights the fact that there are four major strategies used to influence the vehicle handling dynamics. They are basically to control the three forces acting at the tyre-road interface as follows:

- Controlling the Longitudinal Braking Forces
- Controlling the Longitudinal Driving Forces
- Controlling the Lateral Steering Forces

- Controlling the Vertical Suspension Forces

1.3 Thesis Outline

This section provides an overview of the thesis and what the reader can expect in the following chapters:

Chapter 2 This chapter reviews the literature available in the field of active vehicle dynamics and the integrated chassis control. It begins with a description of the four major vehicle dynamic control strategies practiced by the industry and the academia around the world. Then a brief introduction to the six active control systems from the four major vehicle functions is provided. A detailed review of the research literature available about these systems is conducted. This is followed by a detailed study on the state of the art of integrated chassis control systems. A critical review on the literature available in the field of integrated chassis control is provided. Based on the critical review of the literature, a research question and hypotheses is formed. In order to answer the research question and to test the hypotheses, a set of aims and objectives for the thesis are set.

Chapter 3 This chapter discusses the development of passive vehicle dynamics models. It begins by explaining the fundamental theories and terms in vehicle dynamics followed by a description about various vehicle dynamics models developed in the literature. The development of a full vehicle model to be used in this thesis is discussed. Then a brief study about the theory of tyre modelling is discussed followed by the classification and types of tyre models for simulation purposes. A description about the automotive toolbox developed for this thesis in Matlab / Simulink is presented. Finally some of the standard test manoeuvres used internationally to evaluate the vehicle handling dynamics are described followed by validation of the passive vehicle dynamics model developed against a well known commercial software vehicle model.

Chapter 4 This chapter discusses the development of the models and controllers of active vehicle dynamic systems. It begins by discussing the history of active vehicle dynamic systems in the research literatures. Then a detailed discussion on the development of mathematical models of antilock brake system, electronic stability control, active front steering, suspension normal force control and variable torque distribution systems are provided. This is followed by the discussion on the controller development for each of the active system. The remaining sections of this chapter provide a comparative analysis on the performances of these active systems against their passive counterparts respectively.

Chapter 5 This chapter discusses the integration of four active chassis control systems developed in the previous chapter to improve the current vehicle handling dynamics performance. The chapter starts with the analysis of individual active chassis control systems and establishes their control authorities on vehicle handling dynamics. Then it discusses the development of an integrated chassis controller by starting the integration of electronic stability control and active front steering. After the successful integration of these two systems, the variable torque distribution system was integrated to further augment the handling performance. Finally the normal suspension force control system is added to produce this research goal of a fully integrated chassis controller.

Chapter 6 This chapter highlights the key conclusions of the thesis and recommendations for further research.

Chapter 2

Literature Review

2.1 Introduction

This chapter presents a detailed investigation of literature available in the field of integrated chassis control. It starts with a description of the four major strategies used to actively control the vehicle handling dynamics. This includes the literature on the six fundamental building blocks / active vehicle dynamics systems used for this research. Then the need for integration of active chassis control systems is investigated followed by a detailed review about the state of the art in integrated vehicle dynamics and control. A critical review of the literature is presented and the justification for the research methodology followed is also presented. Based on the literature review the key research question and research hypotheses are formed. In order to answer the research question, the necessary aims and objectives for this thesis are derived based on the presented literature and the critical review. The chapter concludes with a summary.

2.2 Major Strategies for Vehicle Dynamics and Control

As discussed in chapter 1, there are four major strategies used for vehicle dynamics and control to improve vehicle handling. They are,

- Active brake based systems control
- Active drive-torque based systems control
- Active steering based systems control and
- Active suspension based systems control

The following sections will describe these strategies in detail and review the research literature available in those fields.

2.3 Active Brake based Chassis Handling Systems

The first brake based vehicle handling system was invented by Bosch in 1995. Bosch named it as the electronic stability programme (ESP). ESP is an active safety technology that assists the driver to keep the vehicle on the intended path and thereby helps to prevent accidents. When the ESP detects an unstable situation, through various sensors such as yaw rate, lateral acceleration, steering angle, the controller actuates the vehicle braking system to apply a calculated braking torque on a particular wheel to correct the under steering or over steering behaviour of the vehicle. When the braking torque is applied at one of the wheels, it generates a braking force. This braking force acts at the Centre of Gravity of the vehicle to produce a corrective yaw moment that maintains or brings the vehicle back to the stable zone.

2.3.1 Introduction to Anti-lock Braking System (ABS)

A brake based electronic stability control is built upon the fundamentals of an anti-lock braking system. So any attempt to study an ESC system should start with the review of an ABS system.

ABS is an electronically controlled brake based active chassis system that prevents wheels from locking when a brake torque is applied during a sudden / panic braking on dry road conditions or during an excessive brake application on slippery conditions such as wet, icy or snowy roads. The primary objective of an ABS control systems is to prevent the wheels from locking during sudden braking. The prevention of locking of the wheels, especially the steered wheels, also provides steerability to the vehicle during emergency avoiding manoeuvres. As a consequence of this an ABS control system provides the vehicle with directional stability during emergency braking. As an ABS maintains the wheel slip ratio at an optimal value, it reduces the vehicle stopping distance by generating the optimum braking force from all tyres. Locking of the rear wheels before the front wheels, accompanied by any lateral input, makes the vehicle unstable as the locked rear wheels lose the ability to generate the lateral forces. Since an ABS prevents the locking of wheels during braking, it aids in enhancing the stability of a vehicle.

2.3.2 Literature on Anti-lock Brake System (ABS)

Since its conceptual introduction in the 1950s into the automotive industry, a vast amount of research has been done in developing and improving the ABS controller. Various control strategies have been developed and implemented. One of the most widely used industrial control strategies is a Proportional, Integral and Derivative (PID) controller. A vast amount of work has been done on PID based ABS controllers. Jun. C (1998) studied the control of an ABS system with PID control along with various other control strategies to evaluate its performance. This control strategy is very simple, widely used and proven. But it does come with some drawbacks. Tuning of a PID controller is an important issue to be tackled. Optimising the P, I, and D control gains for the desired controller performance is called the tuning.

There are various tuning methods available in literature starting from a simple method such as Ziegler and Nicolas Technique to highly complex mathematical optimizing algorithms. Jiang. F et al (2001), proposed a non-linear PID controller that facilitated robust performance and ease of tuning. Mauer F.G.(1995) examined an ABS braking system with a fuzzy logic controller, which was robust and good at controlling nonlinear systems such as an automobile. Yu. F et al (2002), Zhang J. et al (2008) used a fuzzy logic based online optimal slip ratio method and a ratio of derivative of friction to that of the slip and their derivatives respectively, in order to get an improved ABS performance. Alleyne A. (1998) developed a sliding mode controller and demonstrated its robustness in improving the ABS performance against key vehicle parameters and actuator dynamics. An adaptive PID controller was proposed by Chen C. et al (2004) where a fuzzy logic strategy was used to tune the PID gain parameters to make it robust and nonlinear for various vehicle and surface friction conditions.

2.3.3 Introduction to Electronic Stability Control (ESC)

Electronic stability control, as the name implies, is an active vehicle control system that improves the stability of a vehicle. Having said this, the need to define and explain the word 'stability' of a vehicle is required. Vehicle dynamics specialists around the world generally measure or define the stability of a ground vehicle whether it is a commercial vehicle or a passenger vehicle by a few important vehicle state parameters. They are a vehicle's yaw rate, side slip angle and lateral acceleration, popularly known as 'latac'.

Yaw rate is the rotational velocity of a vehicle about its inertial vertical 'Z' axis and is generally measured in radian/sec (S.I. unit) or degree/sec. The side-slip angle (SSA), also known as the body slip angle (BSA) is the angle between the vehicle's longitudinal 'x' axis (in the body coordinate system) and the direction of the vehicle's velocity vector. In other words, it is the angle between the direction in which the vehicle is facing/ pointing and the direction in which the vehicle is actually moving. The side-slip angle is normally measured in radians (S.I. unit) or degrees. The 'latac', is the vehicle body acceleration in the lateral direction, in other words, in the 'Y' axis and is measured in m/s^2 (S.I. unit), or as a function of the gravitational constant, 'g'.

In this thesis yaw rate and side slip angle are the two vehicle state parameters used to define whether a vehicle is stable or not. The magnitudes and the trends of these two parameters are highly complex and nonlinear processes which are generally controlled by many vehicle parameters, such as a vehicle's inertial properties, its ability to generate the lateral forces and the rate at which it can generate the lateral forces, the amount of steering input, the speed at which a vehicle operates etc. In a typical passenger car operating under defined road conditions (dry and wet roads) the yaw rate can be observed in the range 10, 20, 30 and even up to 40 deg/s as a nonlinear function of all the above mentioned parameters. In day to day driving a normal driver can experience a side-slip angle that is no more than ± 2°. For a sporty driving style the side-slip angle can increase further in magnitude but any increase beyond the limit slip angle value for a given road and speed conditions will move the car towards instability.

In day to day driving we use our cars on various types of roads, such as city roads in built-up areas where we travel up to a speed of 30mph (13..3m/s), to the dual-carriage ways and motorways , where we travel up to a legal speed of 70mph (31.11m/s). From our earlier discussion the two key factors that affect the stability of a vehicle are the speed and the road conditions. The higher the speed the worse the stability and vice-versa. The road condition, also known as the surface condition, can broadly be classified into three regions, dry, wet and icy. These three road conditions can be characterised numerically by a variable called surface coefficient of friction, μ. The μ for dry, wet and icy road conditions are 0.85, 0.5 and 0.2 respectively. A decrease in the surface coefficient of friction increases the instability of a vehicle.

Currently we are living in a busy world where the demands on the time available for people to efficiently complete their daily tasks both at the office and at the home are increasing. So both the people and the government are constantly looking for ways to save time to increase the efficiency and in turn the economy. This has led to increase in the legal speeds at which we are allowed to travel supported by the constantly developing road infrastructure systems. Again a growing world and population has led to an increase in the number vehicles on the existing road infrastructure. This increase in traffic makes vehicles more prone to accidents as the probability of collision is increasing. This prompts drivers travelling at high speeds to take sudden evasive actions such as emergency avoidance manoeuvres, panic braking etc. When these emergency actions are executed at high speeds and/or when the road conditions are poor, the consequences can be serious.

Instability of any ground vehicle can be classified into two broad scenarios, under-steer (US) and the over-steer (OS). These two scenarios are defined based on how good a vehicle tracks the driver's steering input or steering intention. In case of an under-steer condition the actual path followed by a vehicle deviates away from the driver's intended path. The vehicle that is operating under this condition is normally termed as 'pushing' in layman's terms. An under-steer vehicle can be characterised by less yaw rate and smaller side-slip angle. For an over-steer vehicle, the actual path followed by the vehicle moves in towards the centre of the curvature of turn with respect to the driver's intended / desired path. In other words the vehicle is said to be 'spinning'. An over-steer vehicle can be characterised by higher yaw rate and larger side-slip angle. Even though both of these conditions are undesirable, over-steer is considered more dangerous than under steer. There is another scenario of vehicle stability or vehicle dynamic characteristics, the neutral steer (NS), where the vehicle exactly follows the driver's input or intention and the vehicle is considered as stable. However as a neutral steer vehicle has certain undesirable characteristics, such as its proximity to the more dangerous over-steer condition, production vehicles are generally designed with a bias towards under-steer.

At low vehicle speeds the driver needs to input more steering angle to follow a desired path, whereas at higher speeds the reverse is true. This means that if the driver inputs more steering angle than required, then the vehicle might over-steer and ‘spin’. In that situation a corrective yaw moment needs to be applied in the opposite direction to the driver generated yaw moment, to stabilise the vehicle. Similarly, in another scenario where the vehicle travels on a patch of low friction surface and the driver applies a steering input to change the path of the vehicle, to avoid some obstacles or to do a lane change for example. In this case, due to the lower surface friction, the lateral tyre force generated (which is a nonlinear function of variables such as the normal tyre load, the tyre lateral slip angle, the slip ratio and the surface coefficient of friction) is much less than in a normal driving condition. This means the vehicle response to the driver input diminishes and the vehicle deviates away from the driver’s desired path, or under-steers. This situation demands a corrective torque to be applied in the yaw direction supporting the yaw torque generated by the driver’s steering input. The strategy of controlling vehicle stability or influencing a vehicle’s dynamic behaviour by generating either a supporting or an opposing yaw torque as explained above is called active yaw control.

As mentioned at the beginning of the thesis, there are three fundamental ways by which this active yaw torque can be generated in a vehicle:

1) By developing a longitudinal force through the application of different brake torques between the left and the right wheels.
2) By developing lateral forces through steering either the front and/or the rear wheels.
3) By developing longitudinal forces through the application of differential drive torques between the left and the right wheels.

In today's modern ESC system both the brake and drive torque can be applied for this purpose. For the purpose of this thesis, an ESC system that is based only on brake torque is considered.

2.3.4 Literature on Electronic Stability Control (ESC)

Abe, M., et al (2001) used a side-slip control based ESC system to stabilise the lateral dynamics of a vehicle. Their investigation proved that the side-slip control based ESC has a higher ability to stabilize the vehicle motion compared with 4WS because the vehicle losses its stability due to deterioration of rear tire characteristics. The side-slip control is also proved to be superior to the yaw rate control to compensate for loss of stability due to nonlinear tire characteristics. The study used a planar vehicle model to arrive at the conclusions.

Investigation of the use of a nonlinear control allocation scheme for yaw stabilization of the vehicle was conducted by Tondel and Johansen (2005). The control allocation allows a modularization of the control task, such that a higher level control system specifies a desired moment to work on the vehicle, while the control allocation distributes this moment among the individual wheels by commanding appropriate wheel slips. Simulations show that the controller stabilizes the vehicle in an extreme manoeuvre where the vehicle yaw dynamics otherwise becomes unstable.

The feed forward and state feedback controller strategy along with an estimator for sideslip angle was used by Park et al (2001) to implement the stability control theory. The research used a 14 DoF vehicle model along with a brush tyre model and a simple 2DoF reference control model with a simple linear tyre model. The simulations indicated that the designed electronic stability control system could successfully improve lateral vehicle dynamic properties.

A robust sliding mode control strategy based on a quarter car model was used by Bang, H.S. et al, (2001) to enhance a non-linear vehicle model's yaw dynamics performance. This controller showed good longitudinal performance in tracking reference slip ratio regardless of modelling errors and disturbances. However, when cornering was combined with braking or there was a yaw moment disturbance due to a mu-split road, it was difficult to achieve the desired performance using this controller

A fuzzy logic based yaw rate control strategy by assigning the desired wheel slip to each corner of the vehicle by applying a calculated brake torque was demonstrated by Buckholtz, K.R., (2002) of Delphi automotive systems. The fuzzy logic controller used in this research combined two controller inputs into a single input and the rule table elements are adjusted based on expertise. Each wheel is classified in relation to the turn of the vehicle and these classifiers are then assigned to the appropriate wheel online. Even though the yaw control improves the stability, the results show that pure yaw rate control, without addressing vehicle sideslip angle may not be an acceptable method.

A further research by Buckholtz, K. R., (2002), used a yawrate and sideslip angle based fuzzy logic controller to improve the stability of the vehicle. The research investigated the effect of limiting the vehicle side slip angle as a part of the fuzzy supervisory control design. The control was shown to display improved results over the yaw rate only fuzzy electronic stability control system.

Khajavi et al (2009) designed a fuzzy logic controller to enhance the directional stability of vehicle under difficult maneuvers. Their strategy was based on applying braking forces on inner or outer tyres with reference to the direction of vehicle deviation from the desired path. They used a feed forward fuzzy controller with stering angle and vehicle lateral velocity as input and the correcting yaw moment as the output. The membership functions were tuned by trail and error method. The results showed the stability of the controlled vehicle is enhanced compare to the uncontrolled vehicle.

Boada et al., (2005) developed a fuzzy logic controller that generates a suitable yaw moment which is obtained from the difference of the brake forces between the front wheels so that the 8DoF vehicle model follows the target values of the yaw rate and the sideslip angle. The simulation results showed the effectiveness of the proposed control method when the vehicle is subjected to different cornering steering manoeuvres.

2.4 Active Driveline based Chassis Handling System

The first driveline based vehicle handling system was introduced by Honda in 1995 on their Honda Prelude vehicle. Torque from the engine is transferred equally to the left and right drive wheels and thet longitudinal force components are used as only driving forces. ATTS successfully makes yaw moment during cornering by using driving forces. The limit for under steering when accelerating during cornering is extended and vehicle manoeuvrability is dramatically improved.

2.4.1 Introduction to Traction Control Systems (TCS)

Variable torque distribution is driveline based vehicle stability control system. This system is built upon the fundamentals of a traction control system, popularly called as TCS in the automotive community. So any attempt to model an VTD system should start with the modelling of an TCS system.

TCS is an electronically controlled driveline based active chassis system that prevents wheels from spinning when a demand for drive torque is applied during a sudden / panic acceleration (such as starting from a give way junction) on dry road conditions or during an excessive throttle application on slippery conditions such as wet, icy or snowy roads.

The primary objective of an TCS control systems is to prevent the wheels from spinning during sudden acceleration. The prevention of spinning of the wheels especially the steered wheels also provides steer-ability to the vehicle during emergency avoiding manoeuvres such as trying to steer away from a nearby vehicle on an icy road conditions. As a consequence of this a TCS control system provides the vehicle with directional stability during emergency acceleration. As an TCS maintains the wheel slip ratio at an optimal value, it reduces the drop in the vehicle longitudinal acceleration by generating the optimum driving force from all tyres for the given road condition. Spinning of the rear wheels before the front wheels accompanied by any lateral vehicle input, makes the vehicle unstable as the spun rear wheels loses the ability to generate the lateral forces. Since an TCS prevents the spinning of wheels during acceleration, it aids in enhancing the stability of a vehicle.

2.4.2 Literature on Traction Control System (TCS)

Traction control system is also popularly known by another name called anti-slip regulator (ASR). As the TCS shares the hardware with the ABS system it does not as a standalone and always comes with an ABS system. This makes more sense in including the driveline based control system in the integration research with a brake based system. However the TCS required change in the slip control system logic and the necessary hardware to implement the traction control strategy, such as engine spark retard/cut, or driveline disconnection to a particular wheel corner.

A wide variety of TCS are available to improve the traction performance of the vehicle and the number and type of the components vary widely along with their control strategy employed (Jeonghoon Song and Kwangsuck Boo, 2004). It is possible to construct a TCS using a braking system, like an ABS, which provides a faster reduction in drive torque at the spinning wheel, but an engine torque control improves the traction on poor road & tyre surfaces, increases the life of the tyre, brake pad and disc and reduces fuel consumption (Borning, Bete, 1992).

The design of a traction control system is complicated by several factors. The system is highly non-linear, vehicle parameters and road conditions may change significantly with time, and the tire-road interaction is difficult to measure and estimate (Tan 1989, Leiber 1983, Layne 1993, Kachroo 1994). Traction controllers based on conventional control approaches have been successfully designed and implemented by many researchers. Gain scheduling traction controllers (Lieber 1983, Schurr 1984) and robust control algorithms based on sliding mode theory has been developed (Tan 1988, Tan 1989, Kachroo 1994). The uncertainty and non-linearity associated with traction control makes a fuzzy-logic control approach appealing (Lee 1990, Wang 1992, Layne 1993, Bauer 1995).

Chun and Sunwoo, (2004), proposed robust wheel slip control using the moving sliding surface technique which improves the robustness and chattering. Kabganian and Kazemi, (2001) developed a TCS based on the dynamic surface control method. They used a sliding mode control strategy to engine torque by controlling the throttle valve. So, the traction control system is well developed and implemented chassis system.

2.4.3 Introduction to Variable Torque Distribution System (VTD)

Variable Torque Distribution, as the name implies is an active vehicle control system that improves the stability of a vehicle by distributing the driveline torque to the four corners of the vehicle in the required ratio. As for as vehicle stability control principle is concerned, VTD is similar to an ESC system, but the control forces at the tyre-road interaction are applied in the opposite direction with much slower driving dynamics.

In the recent decades, the automotive industry has seen a growth in the use of four wheel drive systems on passenger vehicles. Following which the use of four wheel systems for yaw control using torque management systems has been investigated by many researchers. Systems that have been proposed in this area include the use of front-back torque control couplers by Nissan V-TCS, Haldex LSC, BMW xDrive, and Bosch CCC, the use of limited slip differentials together with on-demand couplings by GKN TMD, the use of left-right torque control by Honda SH-AWD and Mitsubishi AYC to enhance cornering performance and stability.

In case of four wheel drive vehicles (4WD), the VTD is implemented by a centre coupler along with a front / rear limited slip differential. Since the vehicle model used in this thesis is that of a small ‘class A’ front wheel driven (FWD) category, the need for centre coupler is eliminated and the VTD system is developed only with a front limited slip differential mechanism.

2.4.4 Literature on Variable Torque Distribution System (VTD)

In a study conducted by Ghelardoni, (2004), the feasibility of engine torque distribution between the axles is investigated using a simple, planar Simulink based vehicle model. It was concluded that the redistribution of torque system is not able to prevent both oversteer and understeer on the same vehicle. On the contrary the ESP can correct both of these undesired motions.

Pinnel, A. et al. (2004) investigated a simple yaw torque controller by means of variable drive torque distribution. The research used a PI control algorithm implemented in Matlab/Simulink integrated with a vehicle simulation model on VeDYNA. In demanding driving situations with traction forces acting, the control system provided substantial support for the driver in stabilising the vehicle despite the control algorithms simplicity. The research was limited itself to the drive torque distribution between the right and left wheels of a given axle and did not investigated the effect between the front and rear axles.

A novel approach to the control of modern torque vectoring differentials based on the approach of side-slip angle minimisation was conducted by Croft-White, M. and Harrison, M. (2006). They used a side-slip angle based PI control strategy to implement the VTD principle and improved the vehicle lateral stability. The effect of front to rear and left to right torque vectoring was analysed and compared.

Cheli,F. et al., (2009) used a feed forward and feedback control strategy to develop a torque vectoring algorithm for a high-performance 4WD vehicle. They used a multi-layer control logic to control the clutch and the differential. The results demonstrated the improvement in the lap performance of the active vehicle.

The research by Russell Osborne and Tayhyun Shim, (2006) has demonstrated that the AWD technology has reached a sophisticated level by means of controlling the torque on independent wheel. A vehicle model of a typical sports sedan was developed in Simulink for this research, with fully independent control of torque distribution. Box–Behnken experimental design was employed to determine which torque distribution parameters have the greatest impact on the vehicle course and acceleration. A proportional-integral control strategy was implemented, applying yaw rate feedback to vary the front–rear torque distribution and lateral acceleration feedback to adjust the left–right distribution. The resulting system shows a significant improvement over conventional driveline configurations under aggressive cornering acceleration on a high-μ surface.

2.5 Active Steering based Chassis Handling Systems

Active steering based vehicle handling systems controls front/rear wheel steering angle within the linear range of tire characteristics to improve the automobile handling and stability. Its purpose is to control the steering angle at the front/rear wheel to eliminate the error between actual and the reference value, making the yaw rate response follow the steady-state reference model.

2.5.1 Introduction to Active Front Steering (AFS)

As mentioned in section 2.3.3, one of the three fundamental ways by which an active corrective yaw moment can be generated is by developing lateral tyre forces through steering of front and/or the rear wheels. The control strategy to actively develop additional corrective steer angle to support the driver's steering input is called active steering. This strategy, if applied at the front wheels, is known as active front steering (AFS) and if at applied at the rear, as active rear steering (ARS).

There are two ways how AFS is currently implemented on vehicles. In the first method, the steering ratio is actively varied a function of the speed of the vehicle. At low speeds, such as during parking manoeuvres, the steering system is operated at higher ratios, providing direct steering to easily manoeuvre the vehicle. At higher speeds, such as highway driving, where maintaining the vehicle directional stability is of more concern, the steering system is operated at higher ratios providing more indirect steering to help the driver to maintain the directional stability. In the second method, the steering ratio between the steering wheel and the road wheel is actively varied by adding a planetary gear box and an electric motor in the steering system. In this method the steering input from the driver is fed through a planetary gear box that drives the rack and pinion steering system. An additional steering angle input is either added or subtracted by an electric motor through the planetary gearbox, thereby changing the overall steering ratio and in turn the steer angle given to the road wheel.

The power assistance in a vehicle's steering system is provided by two means, hydraulic or electric. The hydraulic power steering system, where a hydraulic actuator is used to provide the assisting force, is widely implemented in today's vehicles. I the case of an electric power steering system, which has started to appear in modern vehicles, an electric motor replaces the hydraulic actuator. For the purpose of this thesis the method of AFS with a hydraulic power steering system is used.

2.5.2 Literature on Active Front Steering (AFS)

Handling improvement using active steering has been extensively studied in the two past two decades by many researchers around the world. The earliest study on such a concept was carried out by Kasselmann and Karanen (1969). Their work on adaptive steering control used a proportional feedback of yaw rate from a gyro to generate an additive steering angle input for the front wheels. In 1982, Ackermann (1982), who principally contributed to the research of active steering, developed a robust steering controller for varying operating conditions such as mass, velocity and tyre contact. A concept to use yaw rate feedback in active front and rear steering was proposed by Ackermann (1990). Ackermann and Bunte

(1996) conducted a detailed analysis of the contribution of active steering in handling improvement of passenger cars based on decoupled steering dynamics. They presented an analytical method to robustly decouple the yaw rate from the lateral dynamics. Using active steering as a tool to influence vehicle yaw and roll dynamics was demonstrated by Ackermann et al, (1999), where they summarised two concepts to improve yaw attenuation and to reduce rollover risk respectively.

Said Mammar and Damien Koenig, (2002), analysed the improvement of vehicle handling by active steering by implementing a driver steering angle feed forward controller coupled with a yaw rate feedback controller. The results show that this strategy increases the robustness of the AFS controller against model parameter variations and disturbances. A robust sliding mode controller (SMC) based active steering system is used to improve the vehicle handling behaviour in split-mu braking condition by Roderick et al (2004). In 2004, Willy Klier et al discussed the modular system concepts of active steering systems and their respective advantages. Their research underlines the need for developing dynamic models of steering systems before developing an active steering system controller.

2.6 Active Suspension based Chassis Handling Systems

Although active suspension control has been studied and used for many decades, most of the research focussed on vehicle comfort. Recently the capability of active suspension system to influence vehicle handling has been explored by researchers (Wang, Crolla et al, 2005). An active roll control concept is used by employing hydraulic actuators to stiffen up the suspension system by TRW to enhance the vehicle handling (Seewald, 2000). Kou et al, (2004) employed a continuously varying damping control (CDC) to vary the suspension damping forces to improve the vehicle stability during cornering. Effect of wheel load intervention on yaw moment generation is investigated by Saeger et al (2003). An intelligent system to influence the vehicle roll dynamics is investigated by Ansgar Trachtler (2004).

2.6.1 Introduction to Normal Force Control (NFC)

As mentioned in section 2.3.3 of this chapter, the industry wide practice is that, vehicle yaw and side slip dynamics are improved by one of the three well known chassis control strategies, active brake intervention, active steering intervention and active drive force intervention. The active suspension systems are mainly employed as comfort improvement systems in terms of roll, pitch and vertical dynamics. Taking into account the tyre force generation mechanism and how the tyre normal load can influence the lateral and longitudinal force generation, active suspension system has the potential of influencing vehicle handling if the correct control strategy is used. But obtaining that objective should not affect the main control objective of the active suspension, such as roll control etc.

Before analysing any active suspension system it is worth to start by discussing the fundamentals of the passive system. The two main purposes of an automotive suspension system are passenger isolation from road roughness and road holding. Road isolation is the ability of a suspension system to isolate the sprung mass (passenger compartment, passengers and payloads) from the road vibration inputs. Whereas, road holding defines the ability of a suspension system to maintain the tyre and road contact. Road holding plays a key role in vehicle handling since it controls the generation of longitudinal and lateral tyre forces. The three main elements of a suspension system through which these two objectives can be achieved are springs, dampers and anti-roll bars. Spring is an energy storage element which stores the energy transferred from the road. The characteristics of a spring can be described by its force vs deflection characteristics. Since a spring can only store energy temporarily and cannot dissipate it oscillation occurs in the sprung mass. This discomforting behaviour points to the need for an element that dissipates this stored energy at a faster rate which is done by the dampers.

There are various types of dampers used in suspension system such as frictional dampers, hydraulic dampers etc. But most of the today's modern automobiles use velocity based hydraulic dampers. These dampers can be characterised by their velocity vs force generation characteristics. In addition to these two, the third passive fundamental element used in a suspension system is an anti-roll bar (ARB), which is basically a spring that resists the vehicle roll motion and it is characterised by the resistance it provides for a unit roll angle. These passive elements together aid the suspension to achieve the two earlier mentioned aims, road isolation and road holding. But, increasing the road isolation ability of a suspension system will decrease the road holding and vice-versa. So suspension system design should strike a compromise between these two totally opposite aims.

Designers of these passive elements always fine tune the stiffness and damping coefficient of a suspension to strike a balance. A softer suspension system increases the passenger comfort by isolating the sprung mass better whereas a stiffer suspension system increases the road holding ability and increases the vehicle handling. But an increase in road holding deteriorates the passenger comfort and vice-versa. So, it is always a trade-off as the characteristics of these three passive suspension elements are fixed by design. Dynamically varying these characteristics as driving conditions are changed will help to achieve the individual performance objectives for both the comfort and the road holding from a same suspension system. This adaptability to improve both the suspension objectives led to the development of active suspension systems.

Active suspension system can generally be divided into two classes, active and semi-active. A semi-active suspension system is where a damper's force generation characteristic can be varied as a function of different driving conditions. Here, the suspension deflection is sensed actively and the damping coefficient of the damper is varied. Hence a semi-active damper provides different damping forces for a given suspension velocity which affect the total suspension force generated from a suspension system and finally the rate at which the stored energy is dissipated. Whereas an active suspension system, nor like a semi-active suspension where the energy can only be dissipated, contains power sources such as hydraulic or electric actuators that can dynamically change a suspension's force generating characteristics by adding energy to the overall vehicle system.

As we discussed earlier, suspension affects the road isolation and the road holding property of a vehicle. But as this thesis is mainly focused on the vehicle stability, only road holding objective of the active suspension is taken into account. There are two general principles of active suspension widely used in vehicle control systems, suspension normal force control (NFC) and roll moment control (RMC). In case of suspension normal force control individual actuators are used at each of the four wheels to apply positive controlled normal force that changes the ratio of normal force distribution between the front and the rear axles. This change in the ratio of normal force distribution between the axles affects the tyre force generation characteristics at the respective axles. Generally longitudinal and lateral tyre force generation is a function of many parameters including the tyre normal loads. Increasing the proportion of the front axle normal loads by individual wheel normal force actuators will increase the lateral force generated at the front.

More lateral force at the front than the rear makes the vehicle to overseer and vice versa. Hence the vehicle handling characteristics can be changed by changing the distribution of normal wheel forces between the front and rear axles.

Another active suspension principle used to influence the vehicle stability is by actively controlling the roll moment distributed between the front and rear axles. This principle similar to the suspension normal force control changes the normal force distribution between the front and rear axles by reducing the dynamic tyre load variation between the wheels. This can be achieved through an active anti-roll bar whose roll stiffness can be dynamically changed as a function of the lateral acceleration of a vehicle, for example.

2.6.2 Literature on active suspension systems

An extensive amount of research has been done on active suspension systems in the last four decades even though the first research publication on active suspension dates back to 1950s (Gugliemino et al, 2008) the first commercially available electronically controlled damper systems were introduced in the 1980s (Ueki et al, 2004). Crolla, D. A. has done an extensive contribution to the field of passive and active suspension research. Sharp and Crolla (1987) and Crolla and Nour (1988) produced a comparative reviews of advantages and disadvantages of various types of suspensions. In 1995 Crolla presented a historical review where he detailed some of the key design criteria for a suspension system.

Williams, (1997) has conducted a detailed analysis on the basic principles of active suspension systems and its practical consideration. In the first part of his work he reviews the compromises of a passive suspension system and how these compromises can be changed by the addition of active components. He also studied the benefits of various active suspension technologies and their strengths and weaknesses.

Two fundamental suspension control principles discussed in the literature are skyhook and ground hook damping controls. Skyhook damper technique is a control technique used in active suspension systems, which is based on the absolute measurement of the body velocity of the car proposed in the 1970s by Karnoop. The skyhook control technique was designed to produce a superior road isolation performance. A contrary control strategy called ground hook control logic, which was investigated by Valasek et al (1998) where the absolute measurement of the un-sprung mass velocity is used to reduce the dynamic tyre forces. During the last few years there has been a tremendous amount of applications of intelligent control techniques such as fuzzy logic and neural network in controlling the active suspension systems for automotive application. An optimal fuzzy controller was proposed by Tadeo Armenta and Miguel Stefazza, (2007), to improve the ride comfort performance of a bus suspension system. Salem and Aly (2009) used a quarter car model to compare the performance of fuzzy and PID control logics in improving the both the road isolation and road holding.

Another important component of any active suspension research is the type of actuator used in producing the extra energy to be put into the system. It's a general practice to use a first order model of a displacement actuator in analysing the performance of active suspension system. Foda (2000) used a first order actuator model with time constant and a simple fuzzy logic controller to improve the vehicle ride performance under various road conditions. But more detailed and accurate study requires the use of a nonlinear model of the actuator dynamics. Chantranuwathana and Peng, (2004) used a mathematical model of nonlinear hydraulic actuator in analysing the vehicle active suspension performance through robust force control technique. In their study on the development of adaptive observers for active suspension systems, Rajamani and Hedrick, (1995) used a similar mathematical model of a hydraulic suspension actuator and used the skyhook damper technique to calculate the desired actuator force. A linearised version of the nonlinear actuator model is used by Shen and Peng (2003) to make use of the classical control techniques in studying the force control and displacement control problems of an active suspension system.

2.7 Need for the Integration of Chassis Control Systems

The above mentioned active chassis systems were developed at different points of time during the history of automobiles. For example the early research on active suspension systems dates back to the 1950s. The first active steering systems were developed in the 1960s. Bosch and Honda invented the brake and driveline based stability control system in the mid 1990s respectively. These active chassis systems were designed for various vehicle dynamic purposes and have different control objectives from each other. When activated these systems control the vehicle motions indirectly by influencing the generation of tyre forces and moments through different actuating mechanism available on the vehicle. But the special motion of the vehicle in the three translational and three rotational degrees of freedom are interconnected and changing the force in one direction will have its effect on the other degrees of freedom of the vehicle (Junje, H. Et al., 2006).

It is evident that by nature of their development these systems were developed as standalone systems. A standalone control system can be defined as a system that has its own sensors, actuators and ECUs (Electronic Control Unit). They act on their own to achieve their own control goals without any regard to the other chassis control systems that exist in the same vehicle. Research shows that today's modern vehicles have more than 40 active control systems to control different functions (Rengaraj, C et al. 2009). When combined in a vehicle environment, some of the systems might coexist and achieve their control objective without affecting other control systems. Some might conflict with others in the process of obtaining their own control goals and might deteriorate the performance of other systems. For example, an active suspension system, in the process of achieving its control goal of providing passenger comfort might make the suspension softer and take the load off the front wheels. This might conflict with the control goal of an active brake system to have the maximum possible load on front wheels to achieve optimum brake efficiency.

With the rapid development of chassis control systems in today's modern vehicles, the information and resources can be shared between individual control systems to reduce the cost and improve the overall vehicle performance and efficiency. The above mentioned analysis highlights the fact that these stand alone chassis systems when operated in a combined manner are good candidates for a vehicle dynamics researcher to look into the possibilities to integrate them. There are two different integration possibilities, as mentioned earlier: functional integration (suspension, steering etc) and hardware integration (sharing of sensors, actuators etc). Being in an academic environment this thesis will focus only on the functional integration which can be modelled, analysed within a reasonable time and cost frame. A bottom up approach is followed in this thesis where two or more existing stand alone control systems will be used to develop the integrated control system. In this thesis one chassis control system from each of the four main vehicle functions will be examined for the final integration.

2.8 State of the Art of Integrated Chassis Control

The concept of integrated chassis control appeared in the automotive industry in a primitive form a few decades ago. An early prediction about the future direction of automotive control systems was made by Toyota. Toyota predicted the possibility and benefits of sharing information based on chassis control forces between various automotive systems in their future models to achieve improved performance, safety and cost reduction (Kizu et al, 1988). This was merely a proposal and how the chassis control technology would move in the future. A proposal of harmony between various control concepts and mechanical systems was made by Honda in the middle of the last decade. Shibahata, Y. (2004) concluded in his research that the automotive control logic existed then was still undeveloped and it would be necessary to establish new control concepts which were unique to automotive chassis control systems in future Honda vehicles. One of the key ideas for integration proposed here was between the driver and the vehicle itself. Major industrial research on integrated vehicle control was done by TRW Automotive Chassis in early 2000, where a road map for their future chassis control concepts was presented. A concept to take vehicle dynamics control systems to incorporate occupant safety, collision avoidance, navigation and intelligent transportation was proposed (Seewald, A. (2000).

One of the key academic research publications that discussed the concept of integration of various chassis systems was published by Selby. M. et al, (PhD thesis, 2003). The research presents a two level control strategy to calculate generic actuation forces, such as individual wheel torques, suspension forces and steer rates to achieve the vehicle dynamic motion and to co-ordinate the individual chassis systems to produce the generic actuation forces. Selby used simulation and a sliding mode control method to present his findings. His research provides a detailed discussion of the implications of a coordinated control approach showing it to be a powerful tool. The limitations of the approach are discussed. The most significant limitations are a) the difficulty in proving the optimality of a heuristic control structure, b) the difficulty in assessing the controller behaviour and its interaction with a real driver and c) the likely complexity of the rule base for coordinating more than 2 or 3 systems or describing more complex interactions than were observed here.

In their review about yaw rate and side-slip angle controllers for passenger cars, Crolla, D. and Manning, W. (2007) have highlighted the fact that the integrated approach of different automotive systems would offer the best solution in different areas of the vehicle handling regime. So it is evident that integration of chassis control systems has a potential to improve various vehicle functions such as performance, safety, navigation etc.

As this thesis is about the functional integration of active chassis control systems it is a good starting point to review the key research publications published about various vehicle functions. Integration can take place between a number of systems in an automobile, including the driver. However as mentioned earlier the four key functional areas considered for the purpose of this thesis are braking, steering, driving and suspension. Therefore literature that discussed the integration of these in a system will only be taken into consideration for the purpose of this literature review.

Integration of several vehicle control systems is being studied by several researchers around the world. One of the very early studies on the integration of suspension and braking systems was done by Alleyne (1997). This research showed that a performance enhancement of 5-9% can be seen in longitudinal deceleration with an integrated anti-lock braking system and active suspension system. Smakman (2000) has compared the effect of an integrated braking and wheel load control system on the lateral motion of a vehicle. The research highlights the significance of braking on the lateral dynamics and the little effect of wheel load control on longitudinal dynamics. However, neither work analysed the interaction between braking and suspension systems in detail.

Wang, J. and Shen, S. (2008) has demonstrated design vehicle ride and roll control functions using an active suspension. Their research demonstrates that the integrated suspension control can act against the roll moment induced due to steering manoeuvres and paves way for future integrated chassis control. However their research results show an increase in roll angular acceleration. This violates the one of the fundamental aims of integrated chassis control, not to affect the current vehicle performance after integration.

A similar approach of integrating two vehicle variables was carried out by Ghoneim et al (2000). Their research integrates vehicle yaw-rate and sideslip angle to improve vehicle stability using state equation and transfer function approaches. Both these works are based on the integration of two different vehicle variables using a same chassis function, suspension and brakes respectively. The integration between various chassis functions was not considered here.

Integration of two chassis functional systems is investigated by many researchers. Work by Saeger, M and Andreas, G. (2003), establishes a process for the quantification of yaw moments generated by interventions of a roll moment distribution system and an active brake system. Results show the potential for generating stabilising yaw moment by these systems in standalone and combined manner. A strategy for improved performance by preventing interference is analysed. The benefit of integrating active steering and a dynamic yaw control was proposed by Selby et al. (2001). The concept of pro and contra cornering moment was defined and how these moments can be demanded from active steering and DYC was shown. The rules of integration between these two systems across the vehicle operating range were investigated. The final results show that coordination of these two controllers provides additional benefits in overall vehicle handling behaviour.

A technique to integrate the active front steering (AFS) and active roll moment control (ARMC) was proposed by Elbeheiry et al. (2001). They used a sliding mode controller to influence the steering input of the driver by adding a correction steering angle to maintain the vehicle yaw rate under control. ARMC is used to differentiate the front and rear suspension forces to alter the vehicle yaw rate and to eliminate the vehicle roll motion. SMC technique is used to realise the control objectives. The research demonstrates the coexistence behaviour of AFC and ARMC systems and their ability to reduce vehicle yaw rate.

Research on the integration of braking torque and active suspension forces was carried out by Chou, H. and D’Andrea-Novel, B. (2005) in collaboration with Peugeot-Citroen. They used a simple horizontal dynamics model to demonstrate the effect of differential braking torques and a vertical model to demonstrate the effect of suspension forces on vehicle dynamics. Finally, they merged the two planar dynamics to form a global vehicle control problem.

Daofei, L and Fan, Y. (2007), investigated the ability of direct yaw moment control and active steering to coordinate improved vehicle handling performance. They followed sliding mode control technique to calculate the desired stabilising forces combined with a quadratic programming based control allocation approach to optimally distribute the tyre forces. A Carsim model was used to demonstrate the performance improvement obtained by the integrated controller.

A fuzzy logic based control strategy was developed to integrate suspension and front steering systems by March, C. And Shim, T. (2007). A reasonably detailed vehicle model was used to simulate the integrated strategy. The results proved that both active suspension and active steering have a potential to influence the lateral dynamics and integrating them enhances the vehicle performance. The above research literature have contributed significantly to the field of integrated vehicle dynamics and control, but all of them have been limited in their contribution to integrating any two functions of active chassis systems.

Research on the integration of a vehicle dynamic controller (VDC), a four wheel steering (4WS) controller and an active suspension controller was first carried out by Kitajima, K. and Peng, H. (2000). They used a feed forward control algorithm and a H_∞ control algorithm for coordination. But this research was mainly used to prove the effectiveness of the H_∞ control algorithm.

The method of effecting vehicle yaw dynamics using controllable brakes, suspension and steering are discussed by Hac, A. and Bodie, M. (2002). The research demonstrates how small change in the balance of tyre forces between front and rear axles may affect the yaw moment and stability. The ability of each of the systems considered to generate a corrective yaw moment is evaluated and used for the integration. The results demonstrate the benefits of integration in terms of handling response, stability and reduced driver steering effort. But the research used a simple yaw plane vehicle model, which did not take into account the interactions between the other degrees of freedom, and a poor representation of a real vehicle.

Kou et al., (2004) from Mando Corporation in South Korea, were the first to demonstrate the integration concept between continuously variable damping control (CDC), rear active toe control (AGCS) and an electronic stability program

(ESP). They used multi-body dynamics based ADAMS software to simulate their vehicle and control models. In the integrated controller, the CDC was used to keep the body flat by a hard damping force, AGCS was used to change the rear toe angle and the ESP was used to stabilise the vehicle through braking forces.

Daofei et al., showed that integration of 4WS and DYC can significantly improve the handling performance. When these two systems are integrated with an ARC, it greatly reduces the roll angle and also contributes to yaw control.

The research literature reviewed above clearly highlights the potential benefits to the vehicle dynamics community of integrating various active chassis systems. But much of this research is limited to 2 or 3 vehicle systems, whereas there are four key vehicle functions, braking, steering, power-train and suspension that affect the vehicle dynamics performance. From the research it is clear that each of these systems has the potential to play a role in altering the present dynamics of a vehicle.

The review of research literature on integrated chassis control will be incomplete without mentioning the following three key phrases, ‘ Stand-alone Control Systems’, ‘Combined Control Systems’ and ‘Integrated Control Systems’. They are repeatedly used and emerged as the main concepts from the integrated chassis control literature review. These words basically define the way in which each system interacts with other systems in the vehicle. Junje et al. (2005) define these as follows:

2.8.1 Stand-alone control Systems:

> “A stand-alone control system is defined as the system which is designed to achieve a specific control objective with its own control algorithm and

corresponding hardware and without any knowledge of other control systems”.

Any research to integrate two or more systems first needs to model and establish the individual control systems in a standalone manner. Then their regions of effectiveness, also called control authority need, to be established in terms of key vehicle dynamics parameters.

2.8.2 Combined control systems

“A combined control system is defined as being one with multiple stand-alone control systems operating in parallel and without any communication between each other”. These systems will provide a baseline structure for further integration analysis.

2.8.3 Integrated control systems

“An integrated control system is referred to as being one in which various stand-alone control systems are functionally rather than simply physically superimposed using different design approaches, ranging from local to global integration”.

These systems aim to improve overall vehicle performance by identifying coexisting systems to increase the interactions between them, and by identifying conflicting systems to avoid the possible interactions in order to reduce the potential of negative vehicle performance when they are combined.

So it is evident from the research literature that the actively controlled chassis systems existing on today’s modern vehicles were originally developed as standalone systems. Employing two or more systems on vehicles will create a situation defined as combined chassis systems. Every active chassis system will have their own behaviour and region where they are dominant and a region where they are not. Identification of the strengths and weaknesses of each of these systems will be interesting and will provide the fundamental information required to devise novel intelligent strategies to integrate them, to improve present vehicle dynamic performance and to extend the boundary of vehicle operation.

2.9 Critical Review of the Literature

With reference to the integrated chassis control literature reviewed in section 2.8, many successful research outcomes and limitations are highlighted. The research by Alleyne (1997) and Smackman (2000) did not analyse the interaction between braking and suspension in detail. The research on integrated suspension control by Wang. J. and Shen, S. (2008) showed an increase in the roll angular acceleration which is a violation of one of the fundamental principles of integrated chassis control. Ghoneim (2000) limited his research within a particular vehicle function such as brakes or suspension. His research did not consider integration across the four major vehicle functions.

Research by Saeger (2003), Selby (2001), Elbeheiry (2001) are limited with the integration of two chassis functions. The research by Daofei, L. and Fan, Yu. (2007) and Kou (2004) used commercially available softwares such as CarSim and Adams/Car. No attempt was made to develop a detailed vehicle model with all its subsystem dynamics. Kitajima and Keng's (2000) research on the integration of four wheel steering and active suspension focussed more to prove the effectiveness of H_∞ control and did not focus on the integrated chassis control principles. The research by Hac, A and Bodie, M. (2002) used a simple yaw plane model which did not take into account the interaction between the other DoF.

It is clear that, most of the research work reviewed above, either used simple vehicle models, such as the yaw plane model, individual horizontal and vertical dynamics models, non-linear models with limited degrees of freedom. In some cases, the researcher used standard commercially available bought out softwares such as CarSim and MSc.ADAMS for their base vehicle models. Those approaches either provide little or no interaction between various key degrees of freedom of a vehicle and their effects upon each other. Using commercial software has limitations on details of the model information, such as the dynamics of each standalone system and their actuator, in particular. This thesis attempts to address those problems by developing a detailed non-linear vehicle model with the necessary actuator dynamics wherever possible. Matlab / Simulink software is used as the main working environment to develop vehicle and control models. A modular approach in the model development is used to facilitate the portability and easy reuse of vehicle systems and subsystems in future by other researchers. In order to achieve this aim an automotive toolbox is developed in Matlab/Simulink.

Also, most of the integrated chassis control studies reviewed, considered only two stand alone chassis systems that represent any of the four key vehicle functional domains for the purpose of integration. Some of the researchers have attempted the integration of three standalone systems. But nowadays, with the rapid development of digital electronics, sensor and actuator technologies, the high end modern vehicles boast many active chassis systems, at least one for each of the four key functions. This will become a norm for most of the vehicles in future. This will lead to a situation where more combined systems at least one from each vehicle function need to interact with other. So, this thesis considers four standalone chassis control systems, one each from braking, steering, power-train and suspension. Figure 2.1 details the various possible routes to achieve the goal of a fully integrated chassis controller (Crolla,D.A., 2005) . The highlighted path describes one of the ICC methodologies starting from an individual controller to a fully integrated system of four chassis controllers. The clockwise path from,

braking → steering → driveline → suspension is used in this thesis.

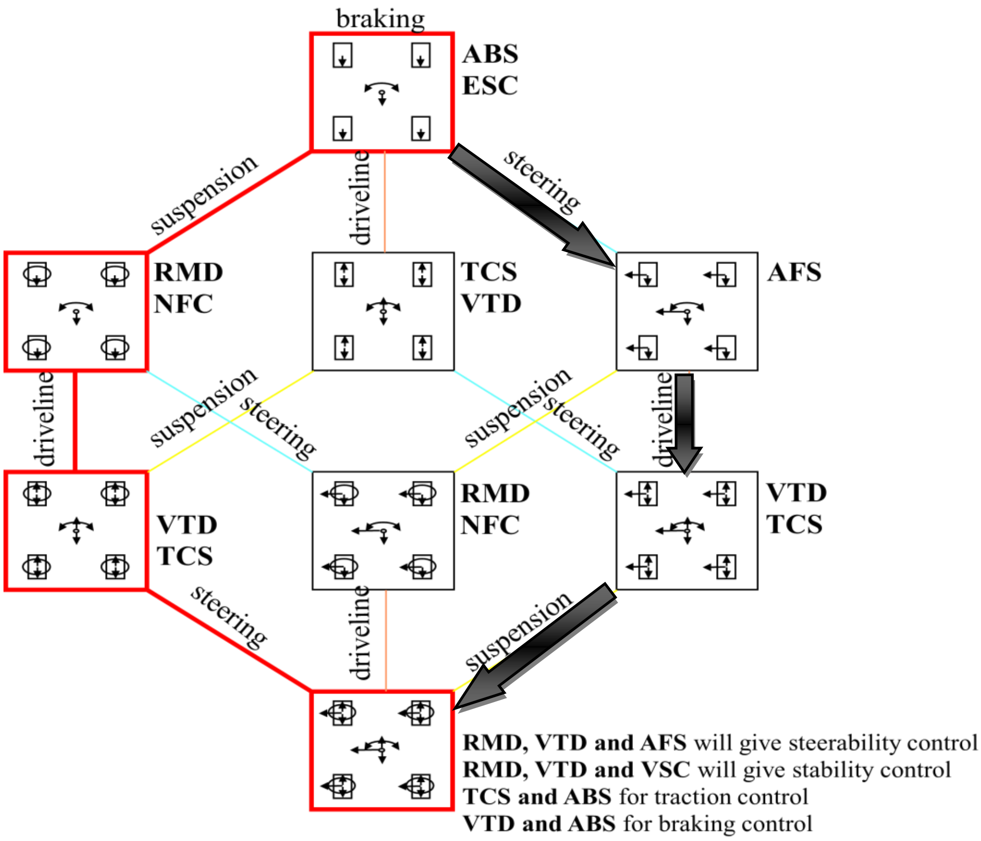

Figure 2.1 – Various Integrated Chassis Control Strategies (Crolla,D.A.,2005)

To start with the study on integrated vehicle dynamics control, two stand-alone vehicle dynamics control systems, namely brake based electronic stability control (ESC) and active front steering (AFS), are considered in this thesis to build the integration strategy. For simplicity the focus will be mainly on vehicle dynamics theory and fundamentals than on using sophisticated complex control system techniques. In order to help to achieve this aim rule based fuzzy logic control and PID control algorithms will be used where necessary. Detailed models of actuators will be implemented during the development of most of the standalone control systems. To facilitate a faster simulation and reduce the model complexity simple first order actuator models are also considered in this research. The following aims and objectives will define the nature of the work undertaken in this thesis and follow directly from the above review and discussion.

2.10 Research Aims and Objectives

From the above literature review a key research question can be raised. Having said that the active vehicle dynamic systems are developed at different points in time with their own control objectives in mind,

- Will it be possible to integrate various (more than three) active chassis control systems to improve vehicle handling dynamics performance?

To answer the research question the following hypotheses are formed:

- Can the various active control systems on a vehicle co-exist without affecting the vehicle dynamic performance?
- Is it possible to make them work together to improve the vehicle dynamics performance?

In order to answer the research question and to test the hypotheses the following aims are made for this thesis:

- To identify the region of effectiveness of different active chassis subsystems.
- To analyse the co-existence behaviour and potential conflicts among them.
- To propose an integrated control strategy to improve vehicle dynamic performance.

In order to achieve the above aims, the objectives were:

- Develop a suitable non-linear vehicle model with all its necessary functional systems.
- Conduct a study on the principles of various tyre models used in the industry and finalise a suitable mathematical tyre model that is capable of handling the complex tyre force generation mechanisms.
- Incorporate these models of vehicle and its subsystems in the form of a Matlab/Simulink toolbox.
- Develop detailed models of standalone vehicle control systems with actuator dynamics.
- Simulate the passive vehicle model and validate the results.
- Conduct a detailed literature review about the four active chassis control systems chosen for the purpose of this research and to develop the fundamental framework and actuator models, controller strategy for them.
- Simulate and compare the passive and the active systems.
- Study and analyse two standalone active chassis systems (electronic stability control and active front steering), in a combined manner and understand the conditions of coexistence and conflicts among them.
- Devise a novel integrated strategy to make these two standalone systems to exist functionally integrated on a vehicle.
- Incorporate a third active system onto the earlier integrated system and study the combined effect due to the new system. Enhance the integrated control strategy to accommodate the new system.
- Finally do a similar investigation to integrate the fourth active system.

2.11 Summary

A detailed investigation of literature available in the field of integrated chassis control is presented with a description of the four major strategies used to actively control the vehicle handling dynamics. Then the literature on the six fundamental building blocks / active vehicle dynamics systems used for this research is reviewed followed by an investigation on the need for integration of active chassis control systems. Then a detailed review about the state of the art in integrated vehicle dynamics and control is presented. A critical review of the literature provided the justification for the research methodology followed. Finally the key aims and objectives for this thesis are derived.

Chapter 3

Modelling of Passive Vehicle Dynamics

3.1 Introduction

This chapter discusses the development of passive vehicle dynamics models. It begins by explaining the fundamental theories and terms in vehicle dynamics followed by a description about various vehicle dynamics models developed in literature. Then the development of a full vehicle model to be used in this thesis is discussed. Then a brief study about the theory of tyre modelling is discussed followed by the classification and types of tyre models for simulation purposes. A description about the automotive toolbox developed for this thesis in Matlab/Simulink is presented. Finally some of the standard test manoeuvres used internationally to evaluate the vehicle handling dynamics are described followed by validation of the passive vehicle dynamics model developed against a well known commercial software vehicle model.

3.2 Theory of Vehicle Dynamics

Dynamics of a rigid vehicle may be considered as the motion of a rigid body with respect to a global coordinate frame. The Newton and Euler equations of motion that describe the translational and rotational motion of a rigid body are the basis for deriving the equations of motion for a vehicle.

3.2.1 Co-ordinate systems

In the domain of rigid body dynamics, co-ordinate systems are used to define the position, orientation and motion of a rigid body with respect to the origin of the co-ordinate system. For the same reason, it is used in the modelling of vehicle dynamics to calculate the vehicle’s position, orientations, velocities and accelerations.

There are two types of co-ordinate systems worthy to mention here, an inertial axis system (also known as global / earth-fixed coordinate system) and a vehicle axis system (also known as body-fixed coordinate system). The inertial axis system is fixed to the earth and is a non-moving system. It is primarily used to calculate the position of a vehicle. The vehicle axis system is assumed to be fixed to the centre of gravity of a vehicle and it is primarily used to calculate the velocities and accelerations of a vehicle. Initially these two systems are aligned with each other at the origin. As the vehicle moves the position and the orientation of the vehicle is calculated as a difference between these two systems.

In the field of vehicle dynamics the two standard coordinate reference frames followed widely are SAE and ISO frames. Both are based on the right hand rule principle. A pictorial representation of the right hand rule is shown in Figure3.1.

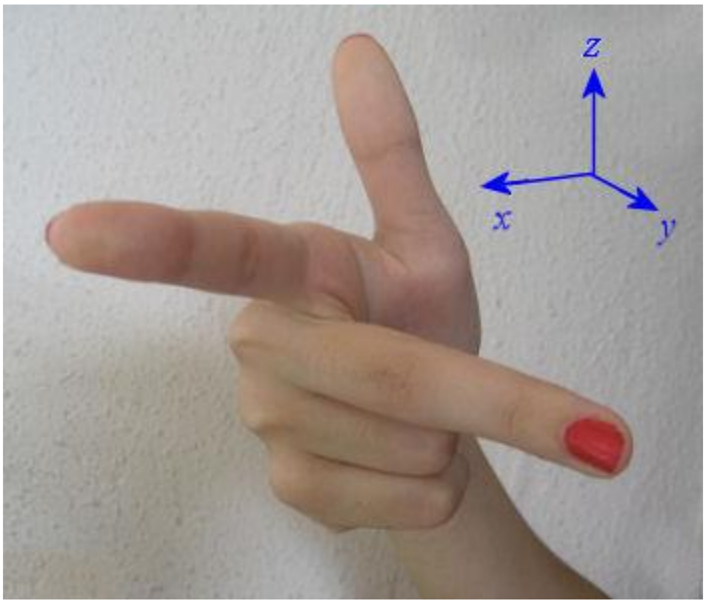

Fig. 3.1: A pictorial representation of right hand rule

The SAE system has its positive X axis towards the front of the vehicle, positive Y axis towards the right side of the vehicle and the positive Z axis downwards (into the earth). An SAE vehicle reference frame is shown in Figure 3.2. The ISO system has it positive X axis defined towards the front of the vehicle, the positive Y axis towards the left hand side of the vehicle and the positive Z axis upwards. An ISO vehicle reference frame is shown in Figure 3.3. Both are widely used in the field vehicle dynamics modelling. The modelling work in this thesis is based on the ISO co-ordinate system because it is more popular than the SAE system and also due to fact that the vehicle dynamics community is considering standardising the use of the ISO reference frame in the future.

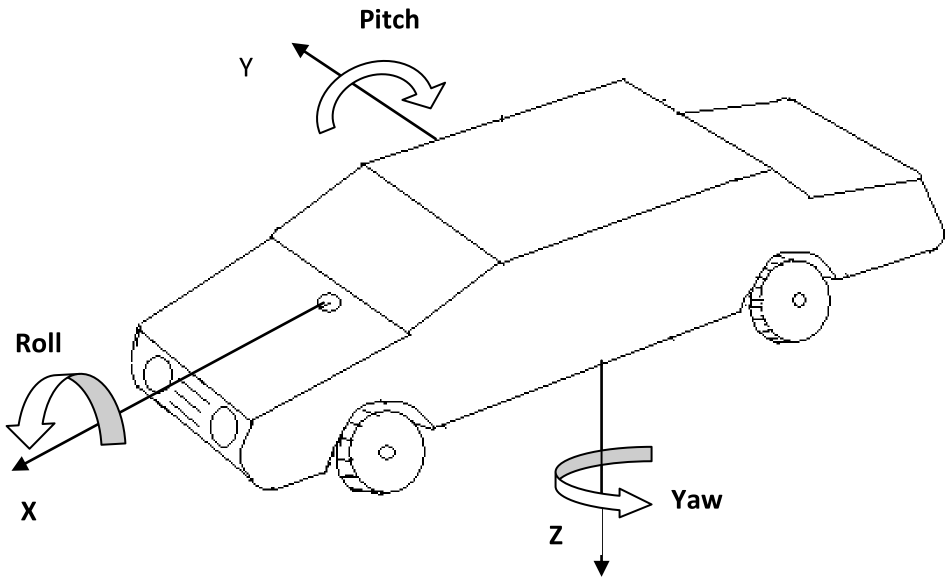

Fig. 3.2: SAE Vehicle Axis System

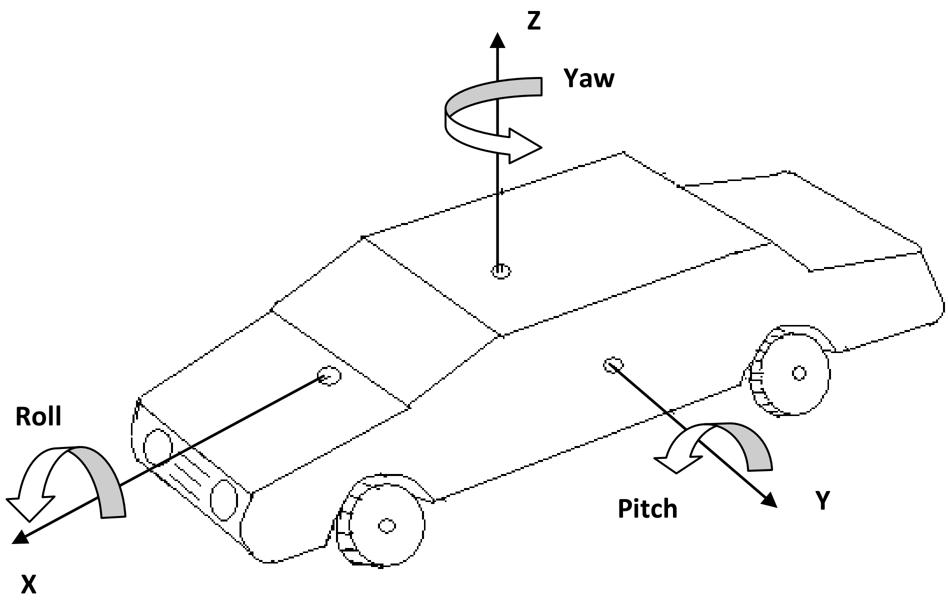

Fig. 3.3: ISO Vehicle Axis System

Co-ordinate systems are the basis to derive the equations of motion for a rigid body such as a vehicle. According to Newton's law of motion the translational dynamics of a rigid body can be expressed in the body-fixed coordinate frame as

$$ {}^{G}F = m\ {}^{G}a_{B} \tag{3.1}$$

where,

- **F** is the vector of resultant external forces acting on the rigid body of mass m
- **a** is the vector of resultant accelerations of the body mass center in global frame.

Equation 2.1 can be expressed in vehicle fixed coordinate frame as

$$ {}^{B}F = m\ {}^{B}_{G}a_{B} + m\ {}^{B}_{G}\omega_{B} \times {}^{B}V_{B} \tag{3.2}$$

$$\begin{bmatrix} F_x \\ F_y \\ F_z \end{bmatrix} = \begin{bmatrix} m\,a_x + m\left(\omega_y v_z - \omega_z v_y\right) \\ m\,a_y + m\left(\omega_x v_z - \omega_z v_x\right) \\ m\,a_z + m\left(\omega_x v_y - \omega_y v_x\right) \end{bmatrix} \tag{3.3}$$

The rotational dynamics of a rigid body can be defined using the following Euler equation.

$$^{B}M = {}^{B}I\ {}_{G}^{B}\dot{\omega}_{B} + {}_{G}^{B}\omega_{B} \times \left({}^{B}I\ {}_{G}^{B}\omega_{B}\right) \tag{3.4}$$

The expanded form Euler equation can be reduced to the following simple form in a special coordinate frame called the principal coordinate frame.

$$\begin{bmatrix} M_x \\ M_y \\ M_z \end{bmatrix} = \begin{bmatrix} I_x\ \dot{\omega}_x - \left(I_y - I_z\right)\omega_y\omega_z \\ I_y\ \dot{\omega}_y - \left(I_z - I_x\right)\omega_z\omega_x \\ I_z\ \dot{\omega}_z - \left(I_x - I_y\right)\omega_x\omega_y \end{bmatrix} \tag{3.5}$$

3.2.2 Vehicle Dynamics

A vehicle has three translational and three rotational motions. Based the ISO coordinate system, the dynamics of vehicle motion is defined in the following six directions as follows:

Longitudinal dynamics: This is the dynamics of vehicle in the ± X axis. Application of acceleration and braking are the primary actions that affect the longitudinal dynamics of a vehicle. The motion in the forward direction is defined as positive and vice versa. The load transfer takes place from front to rear during acceleration and vice versa.

Lateral dynamics: This is the dynamics of the vehicle in the ± Y axis. Motion in this direction is primarily as a result of the application of steering input. The movement of the vehicle towards the left hand side is defined as a positive lateral displacement and vice versa. In general, a pure lateral motion with constant acceleration leads to a load transfer from inner to outer wheels.

Vertical dynamics: This is the dynamics of the vehicle in the ± Z axis. This type of motion is called the bounce of a vehicle. Vertical dynamics mainly discusses about the ride comfort of passengers and the vertical forces applied from the road to the vehicle body through the suspension systems. Input from the road, such as speed bumps or pot holes, are the primary factors that affect the dynamics in this direction. The movement of the vehicle mass in the upward direction is defined as positive and the movement towards the earth is negative.

Yaw dynamics: This is the rotational dynamics of a vehicle about its vertical Z axis. Again the primary input that induces a yaw motion in a vehicle is the application of steering angle. Sometimes the unbalanced longitudinal forces generated between left and right hand sides of a vehicle will also generate a yaw moment. Rotation of the vehicle mass in the anti-clockwise direction (towards the left hand side of the vehicle) is defined as a positive yaw motion and vice versa.

Roll dynamics: This is the rotational dynamics of a vehicle about its X axis. Roll is primarily caused by steering inputs and uneven road inputs between left and right wheels. During a roll motion, load transfer takes place from inner wheels to outer wheels.

Pitch Dynamics: This is the rotational dynamics of a vehicle about its Y axis. Pitch is caused by braking, acceleration and uneven road inputs between front and rear wheels. Load transfer between the front and rear wheels are a typical phenomenon during a pitch motion.

3.3 Various Models of Vehicle Dynamics

Dynamics of a vehicle can be studied and analysed either by means of real-time vehicle experiments or by computer simulations. Experimental analysis involves an instrumented test vehicle and conducting various standard testing manoeuvres to acquire and analyse the vehicle dynamics characteristics. The advantages of analysing the vehicle dynamics experimentally are that it is in real vehicle and results are more accurate and reliable. But vehicle experiments are more time consuming, hard to iterate and expensive. Analysis using computer simulation involves developing models of vehicles and their subsystems on computer, simulating them for given test conditions and analysing the results. Computer simulation of vehicle dynamics is faster, easier to iterate and cheaper. But the results are only as accurate as the models and the input data used. The limitation on the processor speed restricts the complexity of the models used and as simulation is not done in real time the inference of the results needs to be approached with caution.

Nowadays however, simulation technology has improved so much that more complex models can be simulated with many of the experimental analysis advantages. Nevertheless an experimental analysis can never be replaced as all the simulated results need to be validated experimentally before implementation.

Many techniques have been developed to model dynamic systems over the years. Some of the common techniques are mathematical modelling, physical modelling, empirical modelling and multi-body modelling. Considering the simplicity to use, the ability to iterate and the capacity to create highly complex models, mathematical modelling has been the first choice of many researchers in the field of vehicle dynamics. Hence mathematical modelling is used to develop the vehicle and subsystem models in this thesis.

From the abundant literature available in this field one can find various dynamic models have been used to study the dynamics of road vehicles. Wagner and Keane (1997) categorised these vehicle dynamics models into three main groups: (i) Low order, (ii) Medium order and (iii) Higher order models. According to Rodic (2002) these models can also be classified as planar and spatial models based on the types of vehicle motion analysed.

3.3.1 Low-order Models

One DOF model: A one degree of freedom model is sufficient in cases where a lumped mass approach is acceptable to generate a vehicle’s speed. The equation of motion in the longitudinal direction is

$$m\dot{V}_x = \Sigma F_x \tag{3.6}$$

where,

m = mass of the vehicle

V_x = longitudinal velocity of the mass

F_x = Longitudinal tyre force

This description has been successfully used in power train simulations where only an approximate speed is required to emulate the vehicle’s speed sensor for engine algorithm testing.

Quarter car model: The quarter car model represents a comparatively simple model of vehicle dynamics. This model constitutes a quarter of the mass of the vehicle body, called a sprung mass, and a quarter of the mass of the axles and under carriage, called an unsprung mass. These two masses are connected by a quarter of the suspension system (i.e. a spring and a damper) and one wheel.

When suspension modelling and control are considered, the quarter car model is often used. This model allows studying the vertical behaviour of a vehicle according to the suspension characteristic, whether passive or active. The pictorial representation of a quarter car model is given in Figure 3.4.

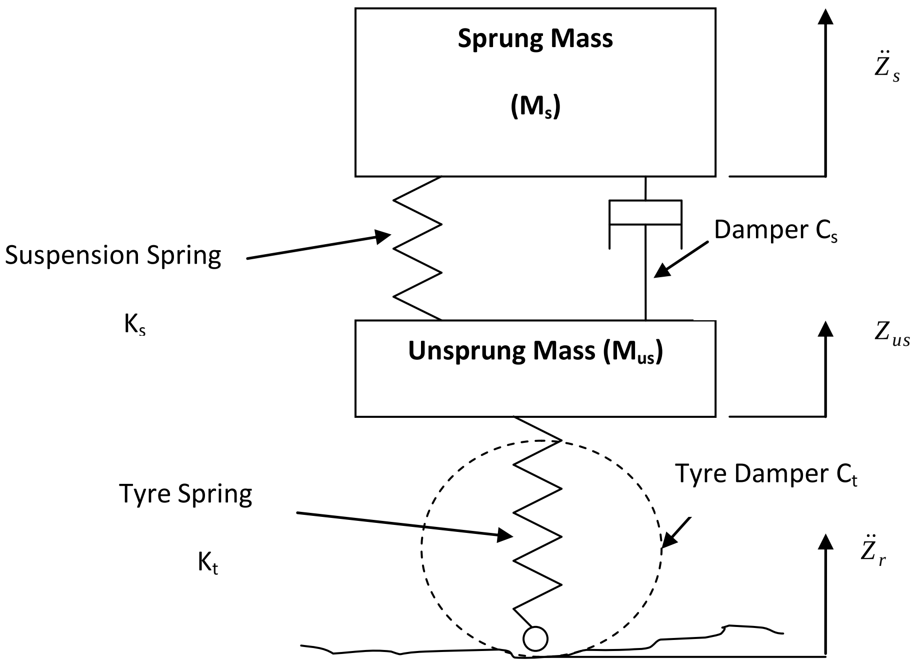

Fig. 3.4: Quarter Car Model

The vertical force generated by the suspension and tyre can be described by the following equations,

$$F_{tz} = K_t (Z_{us} - Z_r) + C_t (\dot{Z}_{us} - \dot{Z}_r) \quad (3.7)$$

$$F_{sz} = K_s (Z_s - Z_{us}) + C_s (\dot{Z}_s - \dot{Z}_{us}) \quad (3.8)$$

The suspension and tyre stiffness (K_S and K_t respectively) and damping factors (C_S and C_t respectively) are non-linear in real vehicle applications. But it is a wide practice among vehicle dynamics researchers to assume them as linear elements to simplify the models. The tyre damping coefficient $C_{t,}$ is generally ignored in vehicle dynamics modelling due to its negligible effect against the high tyre stiffness value. Using the Newton's law of motion the dynamics of a quarter car can be modelled as follows,

$$\begin{aligned} M_s \ddot{Z}_s &= F_{sz} \\ M_{us} \ddot{Z}_{us} &= -F_{sz} + F_{tz} \end{aligned} \tag{3.9}$$

Extended quarter car model: The classical quarter car model allows modelling only vehicle bounce of the chassis and the wheel. A natural extension consists in adding the longitudinal dynamics, i.e., the wheel dynamics as shown in figure 3.5,

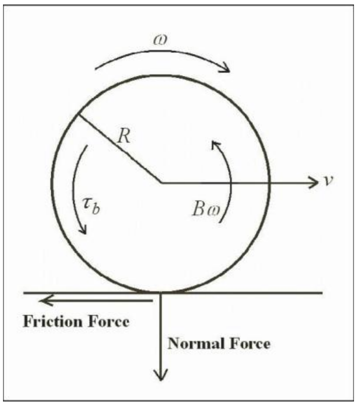

Fig. 3.5: Extended Quarter Car Model

This extended quarter car model with longitudinal dynamics is usually used when braking control and ABS are studied and this model only involves the longitudinal slip (λ), wheel angular velocity (ω) and the vehicle longitudinal velocity (V_x). In addition to equation 2.9, the dynamics of the extended model can be completely described further by the following equations,

$$\lambda(\%) = \frac{V_x - R\omega}{V_x} \times 100 \tag{3.10}$$

$$\dot{\omega} = \frac{RF_{tx}(\mu, \lambda, F_n) - T_b}{I_\omega} \tag{3.11}$$

$$V_x = \frac{-F_{tx}(\mu, \lambda, F_n)}{M_v} \tag{3.12}$$

where,

M_v = the total mass of quarter car

R = the dynamic radius of tyre,

I_ω = the inertia of the wheel

$F_{tx}(\lambda, \mu, F_n)$ = the longitudinal tyre/road friction force and

T_b = the braking torque applied at the center of the wheel.

The coupling phenomenon between (Z_s, Z_{us}) and (λ, ω, V_x) is the normal load F_n which is function of the suspension force and defined as:

$$F_n = M_v g + F_{tz} \tag{3.13}$$

where g is the gravitational constant. This model is being studied more and more as this can connect the work of both suspension and brake control communities.

Bicycle model: The bicycle model is widely found and one of the most extensively used models in the vehicle dynamics literature. This model was developed by Reickert and Schunck (Ackermann, J. and Sienel, W, 1990). Lateral vehicle and yaw dynamics of motion are mostly studied using this bicycle model. It is a single track, two DoF model where the front and rear tyres are collapsed into single front and rear wheels. The roll and weight transfer effects are neglected. This model permits the lateral direction response of a vehicle to be examined for small angle steering manoeuvres at constant longitudinal speed. Both inertial and vehicle fixed co-ordinate systems are used to describe the dynamics. This model is widely used in steering controller studies. The equations of motion for forces along the y-axis and moments about the z-axis are

$$m(V_y + V_x\dot{\psi}) = \Sigma F_y$$
$$I_{zz}\ddot{\psi} = \Sigma M_z \tag{3.14}$$

The pictorial representation of a bicycle model is given in Figure 3.6. All the three models discussed so far are planar models.

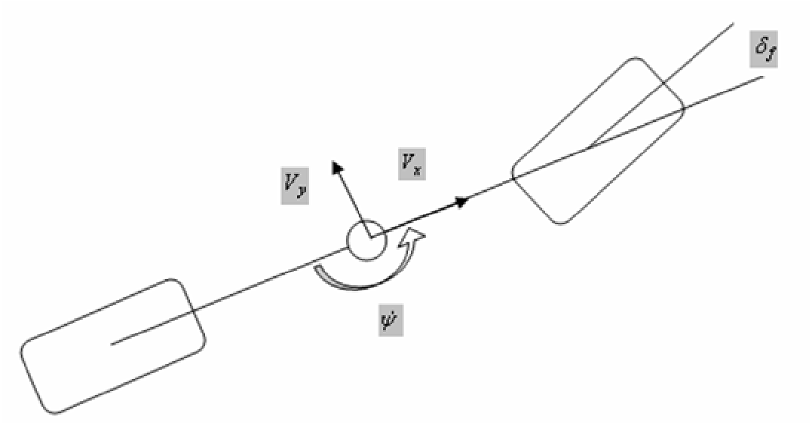

Fig. 3.6: Bicycle Model

3.3.2 Medium-order Modelst2

Model of longitudinal and lateral vehicle dynamics: Another important model that comes under the group of planar models is the model of longitudinal and lateral vehicle dynamics. This three DOF model describes the vehicle dynamics behaviour in the longitudinal and lateral directions as well as in the yaw direction. This model is suitable for preliminary ABS and TCS studies. In addition to equation (2.14) which defines the lateral and yaw motion of the vehicle, the equation of motion in the longitudinal direction can be defined as,

$$m(V_x - V_y\dot{\psi}) = \Sigma F_x \tag{3.15}$$

Pitch/ Roll Model: A four DOF model to describe the longitudinal, lateral, yaw and pitch/roll motions is an important model used in vehicle dynamics studies. These models provide a general purpose description of vehicle dynamics which can serve both powertrain and chassis applications. These two models belong to the group of spatial models. Junje et al, (2005) used an 8 DoF nonlinear vehicle model to study the integrated chassis control systems. The nonlinear model involving roll dynamics is described by equation 2.16 as follows.

$$\begin{aligned}
\ddot{Z}_s &= \frac{(F_{szl} + F_{szr})}{M_s} \\
\ddot{Z}_{usl} &= \frac{(-F_{szl} + F_{tzl})}{M_{usl}} \\
\ddot{Z}_{usr} &= \frac{(-F_{szr} + F_{tzr})}{M_{usr}} \\
\ddot{\phi} &= \frac{F_{szl}t_f - F_{szr}t_f}{I_{xx}}
\end{aligned} \tag{3.16}$$

where,

the index {l,r} = {left, right}

F_{szij} = the suspension forces

F_{tzij} = the tyre forces

I_{xx} = the roll Inertia

t_f = the half front track and

Z_s and Φ = the chassis bounce and roll at the centre of gravity respectively

Z_{usf} and Z_{usr} = the vertical displacements of front and rear unsprung masses respectively

The nonlinear vehicle model involving pitch dynamics is given by the following equations:

$$\ddot{Z}_s = \frac{(F_{szf} + F_{szr})}{M_s}$$
$$\ddot{Z}_{usf} = \frac{(-F_{szf} + F_{tzf})}{M_{usf}}$$
$$\ddot{Z}_{usr} = \frac{(-F_{szr} + F_{tzr})}{M_{usr}} \qquad (3.17)$$
$$\ddot{\theta} = \frac{-F_{szf} l_f + F_{szr} l_f}{I_{yy}}$$

3.3.3 Higher-order Models

A common characteristic of all the above described models is that none of them describes the overall dynamics of the vehicle, but only its partial dynamics. If a more sophisticated vehicle description is needed to study dynamic interactions, such as integration of chassis controllers, then a more complete higher order model needs to be used. The simulation by Garret and Scott (1980) considers a three mass system with sprung mass of six DoF, front unsprung and rear unsprung masses. Each has two DoF. Overall this vehicle model provides a comprehensive description of the systems dynamics. Work by Allen et al. (1998) to analyse the vehicle lateral and directional stability used a chassis with a sprung mass of four DoF and un-sprung masses with two DoF each. March and Shim (2007) used a 14 DoF vehicle model to study the integration of active front steering and active suspension systems. From the literature it is evident that a higher order nonlinear vehicle model is required to effectively analyse vehicle dynamics and control systems, especially for the integrated chassis control application.

3.3.4 Full vehicle Model

The body of the vehicle model used in this thesis is assumed to be rigid and has six degrees of freedom (three translational and three rotational). The vehicle axis co-ordinate system used is assumed to be fixed at the centre of gravity (CoG) of the vehicle body. The vehicle equations of motion are derived with reference to both the vehicle and inertial co-ordinate systems. It is assumed that a suspension unit is attached at each corner of the vehicle with linear spring and damper elements. The dynamics of the un-sprung mass and tyre at each corner are also included in this model. As full vehicle modelling is not a simple task and it involves many subsystems and coupled nonlinear system dynamics, certain modelling assumptions are made and are explained here.

- The self aligning moments of the tyre are neglected, as they do not disturb the vehicle dynamics by bringing back the steering wheel to the initial position.
- The kinematic effects due to suspension geometry are neglected. So the suspensions only provide vertical force to the chassis.
- The gyroscopic effects of the sprung mass are neglected. The only external forces acting on the vehicle are assumed to the longitudinal, lateral and vertical forces generated by the tyres.
- The tyre cambering is considered in tyre modelling.
- The vehicle chassis plane is considered parallel to the ground.
- The aerodynamic and wheel friction effects are neglected as in this work study of those effects is not of great interest.

- The effects due to the toe-in and toe-out of the tyres are neglected.

A schematic view of the nonlinear vehicle model used is shown in Figure 3.7.

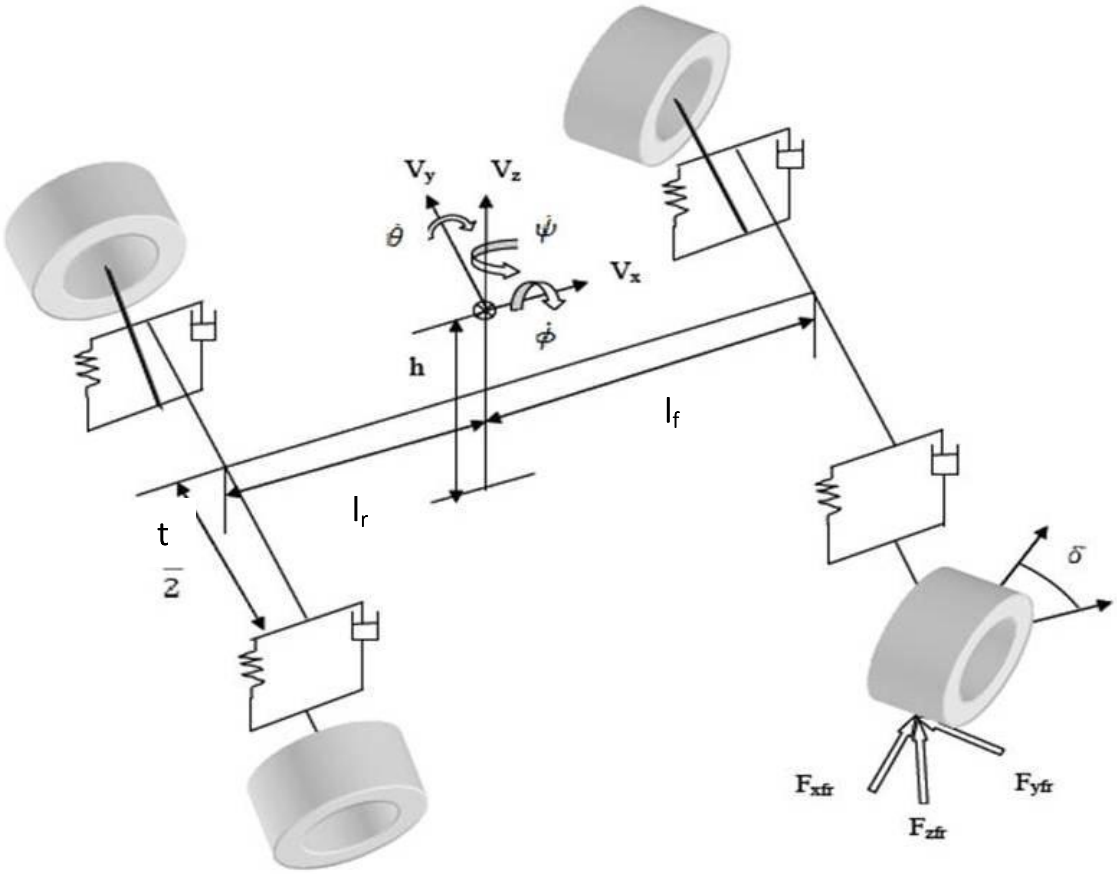

Fig. 3.7: Schematic of nonlinear vehicle model

The kinematic equations are mainly due to the vehicle geometry. Each corner of the vehicle is identified with a {i, j} index, where i = {f, r} stands for front/rear and j = {l, r} stands for left/right. The displacements of the sprung mass on chassis corners are described by,

$$Z_{sfj} = Z_s - l_f \sin(\theta) \pm t_f \sin(\varphi)$$
$$Z_{srj} = Z_s + l_r \sin(\theta) \pm t_r \sin(\varphi) \qquad (3.18)$$

where Z_S is the CG of the sprung mass, Φ and θ are the roll and pitch angle of the chassis respectively, l_f, l_r, t_f, t_r stands for the vehicle geometry, The full vehicle model is defined by the following non-linear dynamical equations.

$$\dot{V}_x = \left\{\frac{\Sigma F_{xij}}{M_v}\right\} - V_y\dot{\psi} + V_z\dot{\theta} \tag{3.19}$$

$$\because \quad \Sigma \mathrm{F_{xij}} = (\mathrm{F_{xij}} + \mathrm{F_{xij}})\cos(\delta) + \mathrm{F_{xij}} + \mathrm{F_{xij}} - \left(\mathrm{F_{yij}} + \mathrm{F_{yij}}\right)\sin(\delta)$$

$$\dot{V}_y = \left\{\frac{\Sigma F_{yij}}{M_v}\right\} + V_x\dot{\psi} - V_z\dot{\phi} \tag{3.20}$$

$$\because \quad \Sigma F_{yij} = (F_{yij} + F_{yij})\cos(\delta) + F_{yij} + F_{yij} + (F_{xij} + F_{xij})\sin(\delta)$$

$$\dot{Z}_s = \frac{(\Sigma F_{sij})}{M_s} + V_x\dot{\theta} - V_y\dot{\phi} \tag{3.21}$$

$$\because, \quad \Sigma F_{sij} = F_{sfl} + F_{sfr} + F_{srl} + F_{srr}$$

$$\ddot{Z}_{usij} = \frac{(-F_{szij} + F_{tzij})}{M_{usij}} \tag{3.22}$$

$$\ddot{\theta} = \frac{((F_{srl} + F_{srr})l_r - (F_{sfl} + F_{sfr})l_f - M_s h_{cg} a_x + (I_{zz} - I_{xx})\dot{\psi}\dot{\phi})}{I_{yy}} \tag{3.23}$$

$$\ddot{\varphi} = \frac{\left(\left(F_{sfl} + F_{srl}\right)t_r - \left(F_{sfr} + F_{srr}\right)t_f - M_s h_{cg} a_y + \left(I_{yy} - I_{zz}\right)\dot{\psi}\dot{\theta}\right)}{I_{xx}} \tag{3.24}$$

$$\ddot{\psi} = \frac{\left(\Sigma F_{mz} + \left(I_{xx} - I_{yy}\right)\dot{\phi}\dot{\theta}\right)}{I_{zz}} \tag{3.25}$$

and, $\Sigma F_{mz} = \Sigma F_{yfj} \cos(\delta) l_f - \Sigma F_{yrj} l_r + \Sigma F_{xfj} \cos(\delta) t_f - \Sigma F_{xrj} t_r$

The forces are given by the following equations:

Tyres:
$$\begin{cases} F_{txij} = F_{tx}\left(\mu_{ij}, \lambda_{ij}, F_{nij}\right) \\ F_{tyij} = F_{ty}\left(\mu_{ij}, \alpha_{ij}, F_{nij}\right) \\ F_{tzij} = F_{tz}\left(Z_{us} - Z_r\right) \end{cases} \tag{3.26}$$

Suspensions: $F_{szij} = F_{sz}\left(Z_{sij} - Z_{usij}\right)$

(3.27)

The normal force on each tyre is calculated based on the following equation,

$$F_{nzij} = F_{ns\_ij} + F_{tzij} \tag{3.28}$$

where F_{ns_ij} is the static load acting on the ij^{th} tyre.

A synopsis of data flow between various vehicle subsystems in this model is given in Figure 3.8

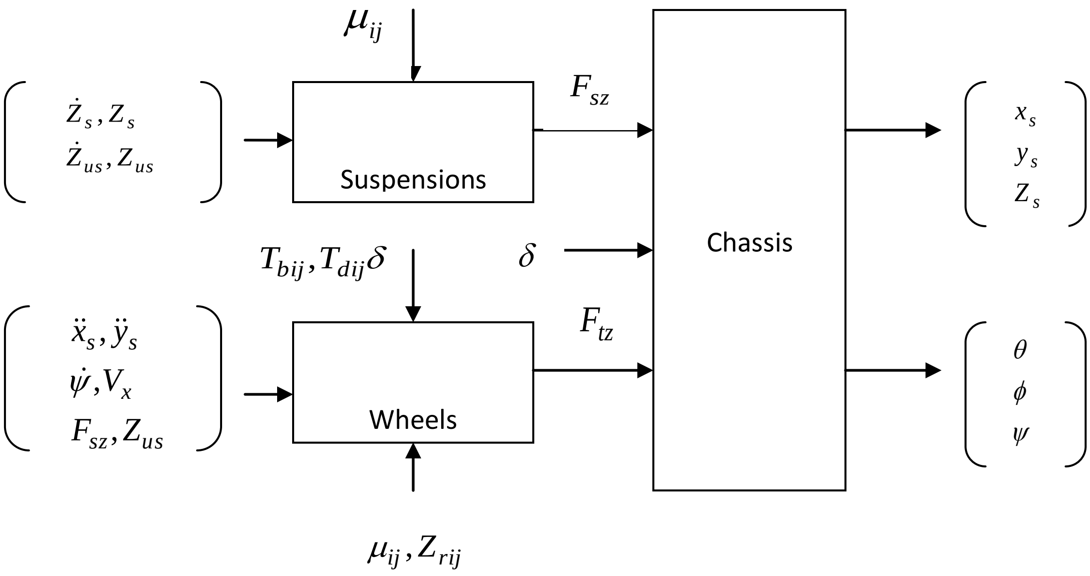

Fig. 3.8: Full vehicle model synopsis

3.4 Justification for the inclusion of 3 rotational DoF

A vehicle's sprung mass has 6 DoF such as an object that moves in space. To develop the equations of motion of such a vehicle, one needs to define the kinematic characteristics first. With reference to equation 3.3, the Newton equations of motion of the vehicle are:

$$\begin{bmatrix} F_x \\ F_y \\ F_z \end{bmatrix} = m \begin{bmatrix} \dot{V}_x - \left(\dot{\theta} V_z + \dot{\psi} V_y\right) \\ \dot{V}_y - \left(\dot{\phi} V_z + \dot{\psi} V_x\right) \\ \dot{V}_z + \left(\dot{\phi} V_y - \dot{\theta} V_x\right) \end{bmatrix} \tag{3.28a}$$

From the above equation it can be observed that the 3 rotational DoF, roll, pitch and yaw influence the translational accelerations. Moreover the 3 rotational DoF plays a role in the calculation of vertical suspension and tyre forces which in turn affect the vehicle translational and rotational dynamics. Hence including them in the equation of motion will improve the accuracy of the vehicle model.

3.5 Modelling of Tyres

Tyres are perhaps the most important component of a vehicle. And they are the only means of contact between a vehicle and the road. The dynamic behaviour of a road vehicle is controlled by the application of steering, braking and acceleration inputs. This ultimately results in forces generated between the tyre and road interface. This explains the necessity to accurately model such an important component of a vehicle. But modelling the tyre behaviour accurately is probably the most difficult and important problem to tackle while building a vehicle dynamics model. Various tyre models have been developed in the past to try and solve this problem.

Any modelling work on tyres cannot be started without discussing the two very important tyre parameters that define and control the force generation in tyres. They are the tyre longitudinal slip (λ), generally called slip, and the tyre lateral slip angle (α), generally called, slip angle.

Slip*:* Tyre slip can be defined as the ratio of the difference in rotational speed and the longitudinal speed of the tyre, to either the rotational speed or the longitudinal speed, depending upon whether the vehicle is under acceleration or braking, respectively. Slip is described either as a percentage or as a number between '-1' to '+ ∞'. A slip of '-1' represents a locked wheel, '+∞' represents a wheel that is spinning and '0' means a free rolling wheel (neither accelerating nor braking). The longitudinal tyre slip is defined as

$$\lambda_{ij} = \frac{R\omega_{ij} - V_x}{\max\left(R\omega_{ij}, V_x\right)} \tag{3.29}$$

Slip angle: Tyre slip angle can be defined as the arctangent of the angle between the direction of the tyre centre plane and the direction of the tyre velocity. This is normally expressed in radians for vehicle dynamics calculations. A positive steer angle always produces a negative slip angle and vice versa.

$$\alpha_{fj} = \tan^{-1}\left\{\frac{V_y + l_f\dot{\psi}}{V_x \pm t_f\dot{\psi}}\right\} - \delta_f$$
$$\alpha_{rj} = \tan^{-1}\left\{\frac{V_y - l_r\dot{\psi}}{V_x \pm t_r\dot{\psi}}\right\} \tag{3.30}$$

3.5.1 Classification of tyre models:

The tyre models widely used for vehicle dynamic simulations can be classified into two major categories, Linear and Nonlinear tyre models.

Linear tyre model: A linear tyre model is assumed to have constant tyre stiffness co-efficient in both the longitudinal and lateral directions. Hence, the longitudinal and lateral tyre forces produced are proportional to the longitudinal slip and tyre slip angle respectively. The linear tyre models are easy to implement and can be used in the design of linear controllers for active vehicle systems. A linear tyre model can be defined typically by means of the following equations:

$$F_x = C_\lambda \lambda$$
$$F_y = -C_\alpha \alpha \tag{3.31}$$

Where, C_λ and C_α are the longitudinal and lateral tyre stiffness respectively. Since the actual tyre behaviour is highly non-linear from medium to higher tyre slip and slip angles, the applicability of this model is limited to small tyre longitudinal slips and tyre slip angles only.

Nonlinear tyre model: The tyre force in a nonlinear tyre model saturates as the slip or slip angles are increased. In order to capture this behaviour a simple tyre model called a piecewise tyre model was developed. These models have a linear region at smaller slip / slip angle and a saturated region at higher slip / slip angle. This model tried to capture the non-linear behaviour to some extent but did not represent true nonlinear tyre behaviour as an actual nonlinear tyre has three distinctive regions. From the experimental results of tyre force generation characteristics, a tyre has a linear region at smaller slip angles where the tyre force is generated linearly proportional to the slip angles, at higher slip angles the generation tyre force saturates, leads to a saturation region, irrespective of the increase in slip angle (for some tyres it even decrease) and the third region is called a transition region where the tyre force is in transition from linear to saturation behaviour. Figure 3.9 gives a comparison of linear, piecewise and nonlinear tyre characteristics. From the figure 3.9 it is clear that the linear tyre model is only valid within a small slip angle region which is generally 5° to 8° depending on the tyre design.

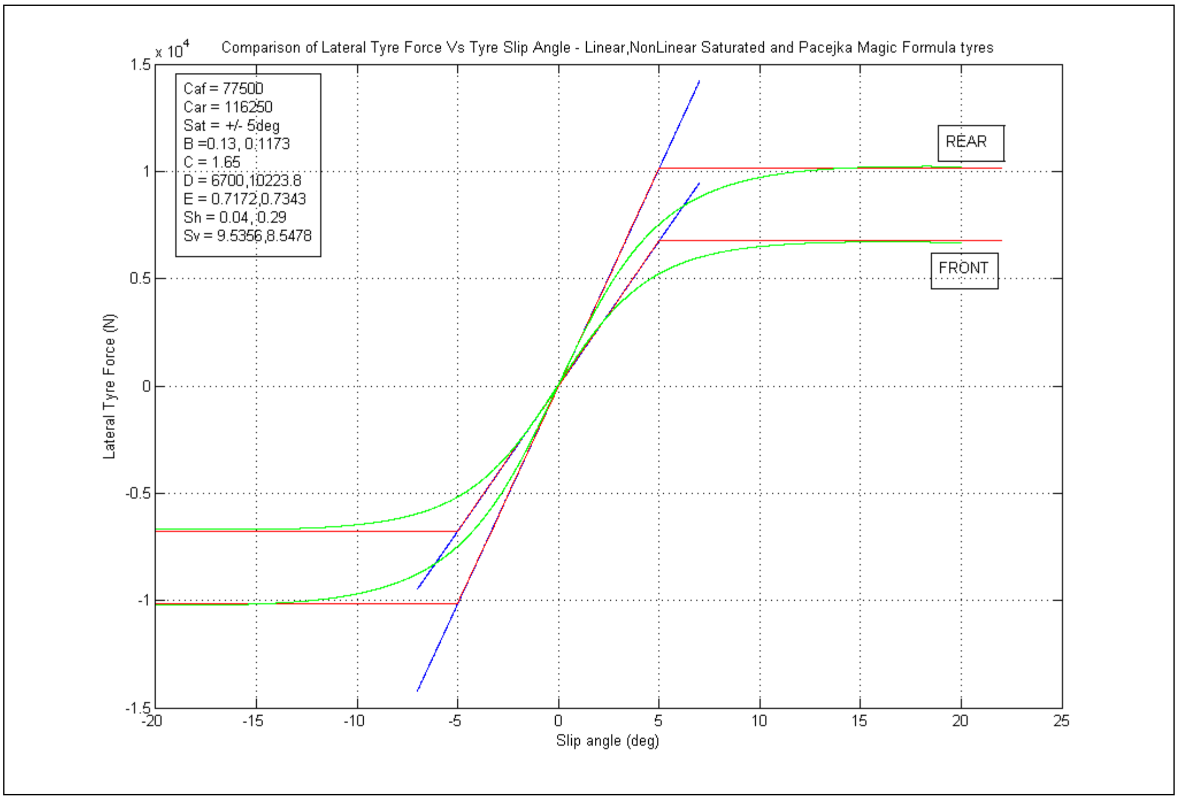

Fig. 3.9: Comparison of linear, piecewise and nonlinear tyre characteristics

3.5.2 Types of nonlinear tyre models:

There are a number of nonlinear tyre models developed for vehicle dynamics simulation by various researchers around the world in the last century. To name a few, Sakai tyre model, Buckardt tyre model, Brush model, Dugoff model, Pacejka tyre model, Delft tyre model. Discussing them would go beyond the scope of this thesis; hence it was decided to compare three of these models based on their popularity among the vehicle dynamics community and choose one model to use as a tyre model in this thesis. The models that will be discussed in the following paragraphs are the Brush model, the Dugoff model and the well known Pacejka tyre model.

Brush model: The brush model consists of a row of elastic bristles that touches the road plane and can deflect in a direction parallel to the road surface. These bristles are called tread elements. Their compliance represents the elasticity of the combination of carcass, belt and actual tread elements of the real tyre. As the tyre rolls, the first element that enters the contact zone is assumed to stand perpendicularly with respect to the road surface. When the tyre rolls freely (that is without the action of a driving or braking torque) and without side slip, camber or turning, the wheel moves along a straight line parallel to the road and in the direction of the wheel plane. In that situation, the tread elements remain vertical and move from the leading edge to the trailing edge without developing a horizontal deflection and consequently without generating a fore and aft or side force. The force and moment generation using a brush model is detailed in Figure 3.10. The longitudinal and lateral forces determined according to the Brush model consist of two components – adhesion and sliding. Longitudinal and lateral forces are determined according to equations (A2.1 to A2.5) in Appendix A. A flow chart for tyre force calculation is given on figure 3.11

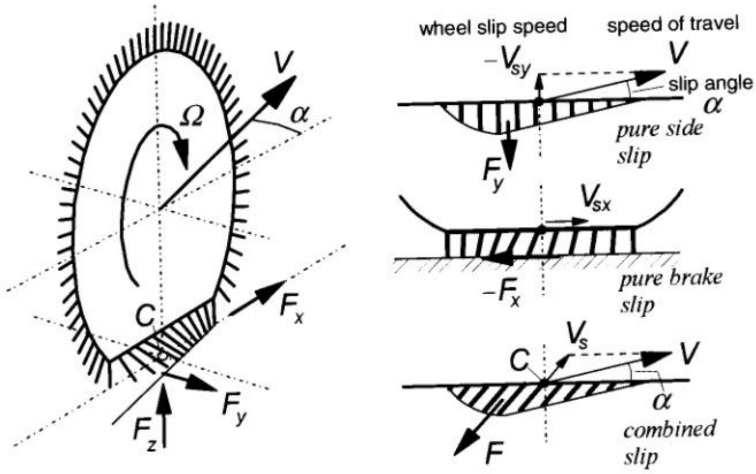

Fig. 3.10: The brush tyre model

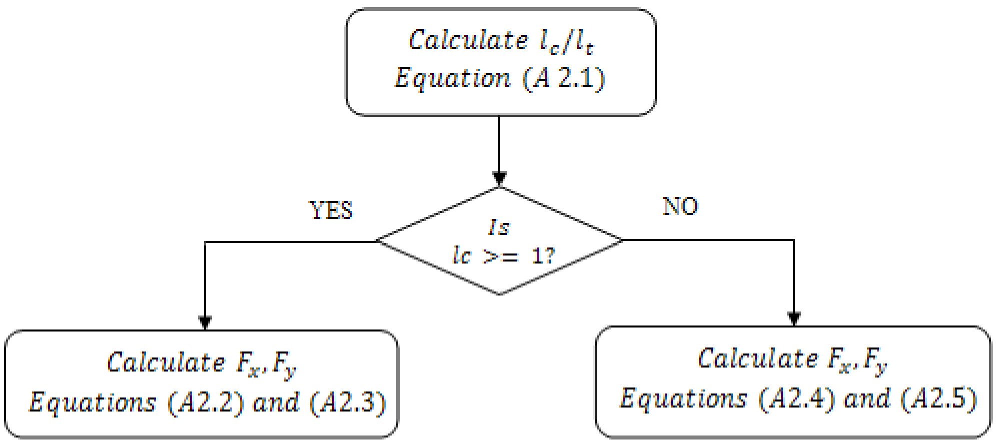

Fig. 3.11: Flowchart for Brush model tyre force calculations

Dugoff tyre model: Dugoff tyre model (Dugoff et al, 1969) assumes a uniform vertical pressure distribution to calculate the longitudinal and lateral tyre forces. Dugoff's model has the advantage of being an analytically derived model developed from force balance calculations. Further, the lateral and longitudinal forces are directly related to the tyre road friction coefficient in more transparent equations. Guntur et al. (2003) presented a simplified method for calculating the longitudinal and lateral tyre forces according to the Dugoff model. The Dugoff tyre model has been developed in MATLAB as recommended by (Guntur et al, 2003). In Dugoff tyre model, the longitudinal and lateral forces of the tyre are given by equations (A2.6) to (A2.10). Simplified equations used for developing the tyre model are given in equations (A2.11) to (A2.15). Figure 3.12 shows the algorithm used for developing the Dugoff tyre model.

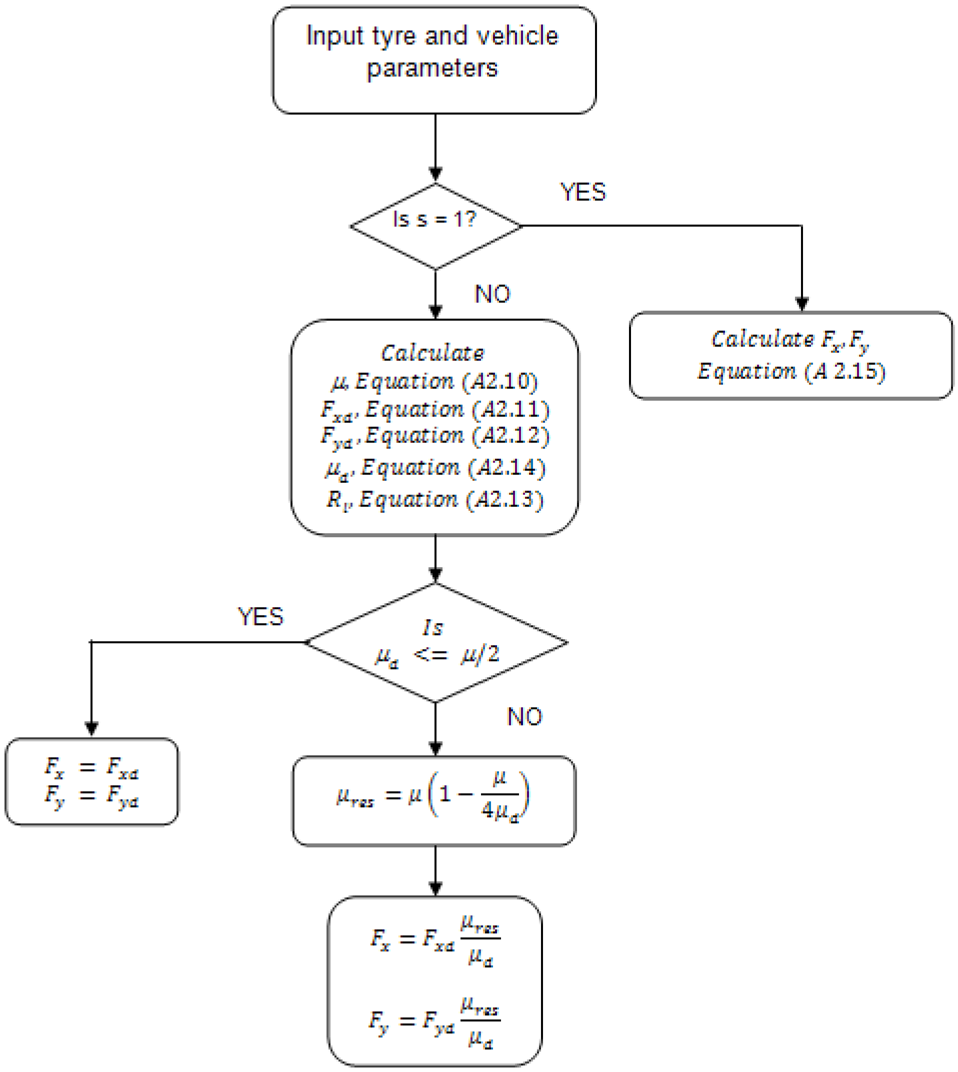

Fig 3.12: Flowchart for Dugoff model tyre force calculations

Pacejka 'magic formula' tyre model: Pacejka, Bakker and Nyborg proposed a new method (Pacejka et al, 1987) for representing tyre data obtained from measurements. They have developed a series of tyre models over the last 20 years. These models were named the 'magic formula' because there is no particular physical basis for the structure of the equations chosen, but they fit a wide variety of tyre constructions and operating conditions. Each tyre is characterized by 10-20 coefficients for each important force that it can produce typically lateral and longitudinal forces, and self-aligning torques, as a best fit between experimental data and the model. These coefficients are then used to generate equations showing how much force is generated for a given vertical load on the tyre, camber angle and slip angle. The Pacejka tyre models are widely used in professional vehicle dynamics simulations, and racing car games, as they are reasonably accurate, easy to program, and solve quickly.

3.5.3 Pure Cornering and Braking: The formula, popularly known as the Magic Formula is given by equations (2.32) and (2.33).

$$Y = D\sin\left(C\arctan\left(B\phi\right)\right) + S_v \tag{3.32}$$

with

$$\phi = \left(1 - E\right)\left(X + S_h\right) + \left(E / B\right)\arctan\left(B\left(X + S_h\right)\right) \tag{3.33}$$

where Y – lateral force, longitudinal force or aligning moment

X – slip angle α or longitudinal slip s

Coefficients used in the formula are explained with the help of Figure 3.13. D is the peak value and the product BCD equals the slip stiffness at zero slip. The coefficient E makes it possible to accomplish a local extra stretch or compression of the curve in such a way that the stiffness and peak value remain unaffected. Coefficient C defines the extent of the sine function which will be used and therefore determine the shape of the curve. Due to ply steer, conicity, rolling resistance and camber, the characteristics will be shifted in horizontal and / or vertical directions. These shifts are represented by S_h and S_v respectively. The coefficients are named as follows:

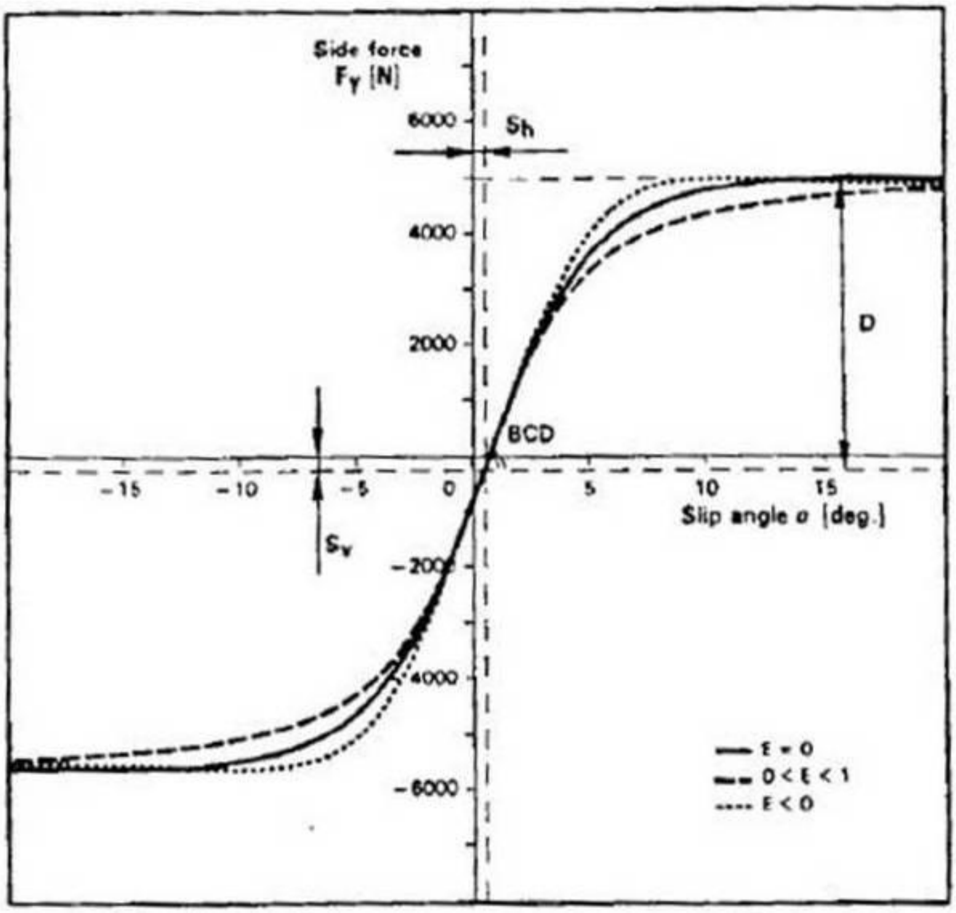

Fig 3.13: Coefficients in Magic Formula (Pacejka,1997)

B – stiffness factor

C – shape factor

D – peak factor

E – curvature factor

S_h – horizontal shift

S_v – vertical shift

The coefficients as function of normal load F_z are given in equations (3.34) to (3.39).

$$D = a_1 F_z^{\,2} + a_2 F_z \tag{3.34}$$

For lateral force,

$$BCD = a_3 \sin\left(a_4 \arctan\left(a_5 F_z\right)\right) \tag{3.35}$$

For longitudinal force and self aligning torque,

$$BCD = \frac{a_3 F_z^{\,2} + a_4 F_z}{e^{a_5 F_z}} \tag{3.36}$$

$$B = \frac{BCD}{CD} \tag{3.37}$$

For lateral force, C = 1.3

Longitudinal force, C = 1.65

Self aligning torque, C = 2.4 (3.38)

$$E = a_6 F_z^{\,2} + a_7 F_z + a_8 \tag{3.39}$$

Values of a_1 through a_8 for lateral force, longitudinal force and self-aligning torque are given in Appendix B. Note that F_z is in kN, α is in degrees, while outputs F_y, F_x and M_z are in N, N and N-m respectively.

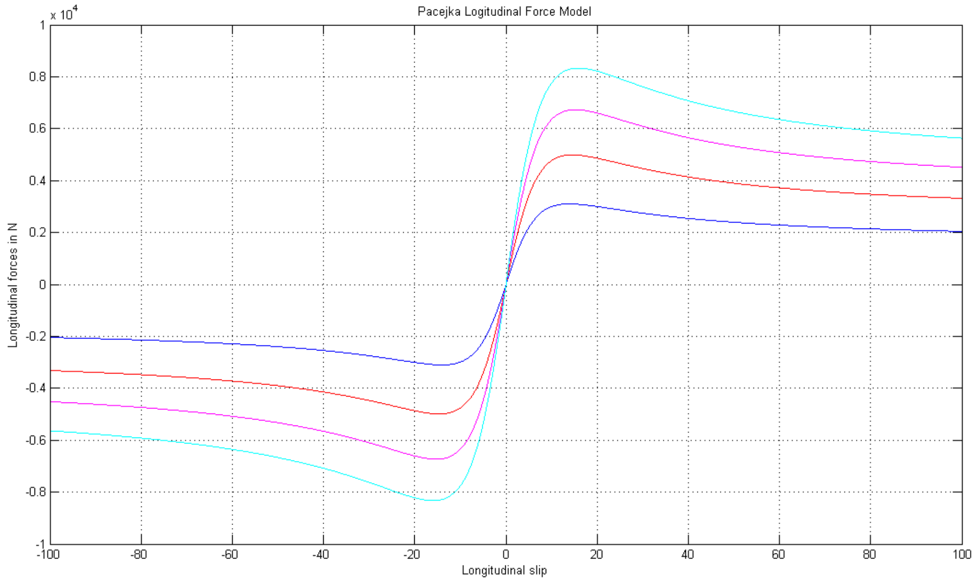

Fig 3.14: Pacejka Longitudinal tyre force – Pure Braking/Driving

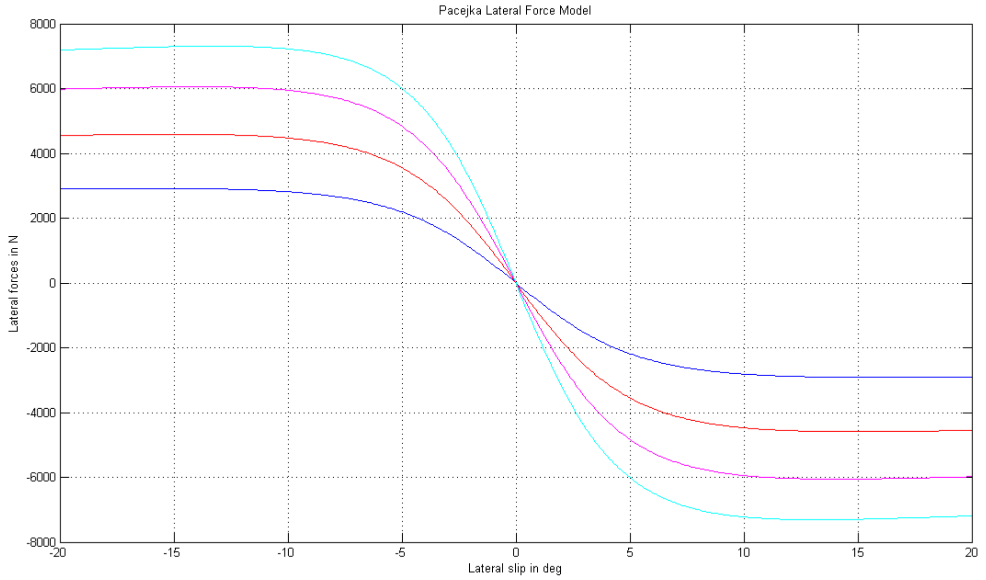

Fig 3.15: Pacejka Lateral tyre force – Pure Cornering

A comparison of these three nonlinear tyre models is shown in figure 3.16 below.

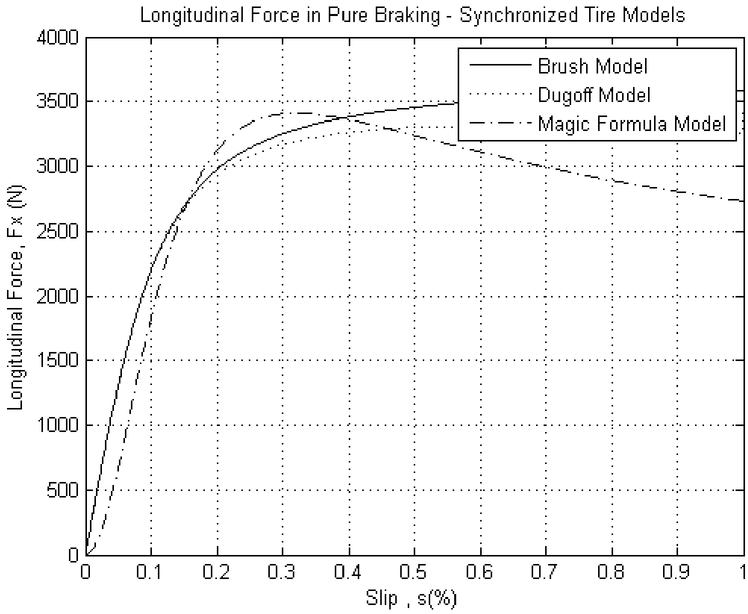

Fig 3.16: Comparison of Brush, Dugoff and Pacejka Tyre models

All three models simulate the actual tyre behaviour better than a linear model. But the Pacejka's magic formula model simulates all the three regions of the tyre characteristic, the linear, the transition and the sliding regions, more accurately than the others. Moreover the literature available on tyre and vehicle modelling shows that this is the most popular and widely used model by the research community both in the academia and in industry. So it was decided to use the Pacejka's magic formula model as the tyre model in thesis.

3.5.4 Combined Slip Conditions: Pacejka's above mentioned mathematical representation is limited to steady state conditions during either pure cornering or pure braking. But in reality, a tyre comes across situations where it experiences combined braking/acceleration and steering. In these situations, for a given tyre normal load and camber angle, the lateral force produced is a function the tyre slip angle and the longitudinal slip. The same holds good for the longitudinal force generation. The parameters used for calculating lateral and longitudinal forces during combined slip conditions are given in equations (3.40) to (3.45).

$$\sigma_x = \frac{s}{(1+s)} \tag{3.40}$$

$$\sigma_y = \frac{\tan\alpha}{(1+s)} \tag{3.41}$$

$$\sigma = \sqrt{\sigma_x^{\ 2} + \sigma_y^{\ 2}} \tag{3.42}$$

$$\sigma_x^{\ *} = \frac{\sigma_x}{\sigma_{xm}} \tag{3.43}$$

$$\sigma_y^{\ *} = \frac{\sigma_y}{\sigma_{ym}} \tag{3.44}$$

$$\sigma^* = \sqrt{\sigma_x^{\ *2} + \sigma_y^{\ *2}} \tag{3.45}$$

σ_{xm}, σ_{ym} are the slip values which occur at the peak of the respective characteristic (braking and cornering). Forces F_{xo} and F_{yo} are obtained by calculating F_x and F_y in pure conditions (using Equation (3.32)) and writing them as a function of normalized slip σ^*. The algorithm for developing the Magic Formula tyre model in combined slip conditions is shown in Figure 3.17.

Pacejka and Bakker (1993) modelled the tyre's response to combined slip by using physically based formulae. A newer more efficient method way is purely empirical. This method was developed by Michelin and published by Bayle et al (1993). It describes the effect of combined slip on lateral force and longitudinal force characteristics by introducing a weighing function G, when multiplied with the original pure slip functions produce the interaction effects of λ on F_y and α on F_x. The weighing functions have a hill shape. The cosine version of the magic formula is used to represent the hill shaped function:

$$G = D\cos\left[C\arctan\left(Bx\right)\right] \qquad (3.46)$$

where x is either λ or α.

The combined lateral force is calculated using the following formulae:

$$F_y = G_{yk} \cdot F_{yo} + S_{vk} \qquad (3.47)$$

The effect due to ply-steer (S_{vk}) is assumed to be zero to reduce the complexity of the tyre model. The function G_{yk} is used as described in equation 3.18.

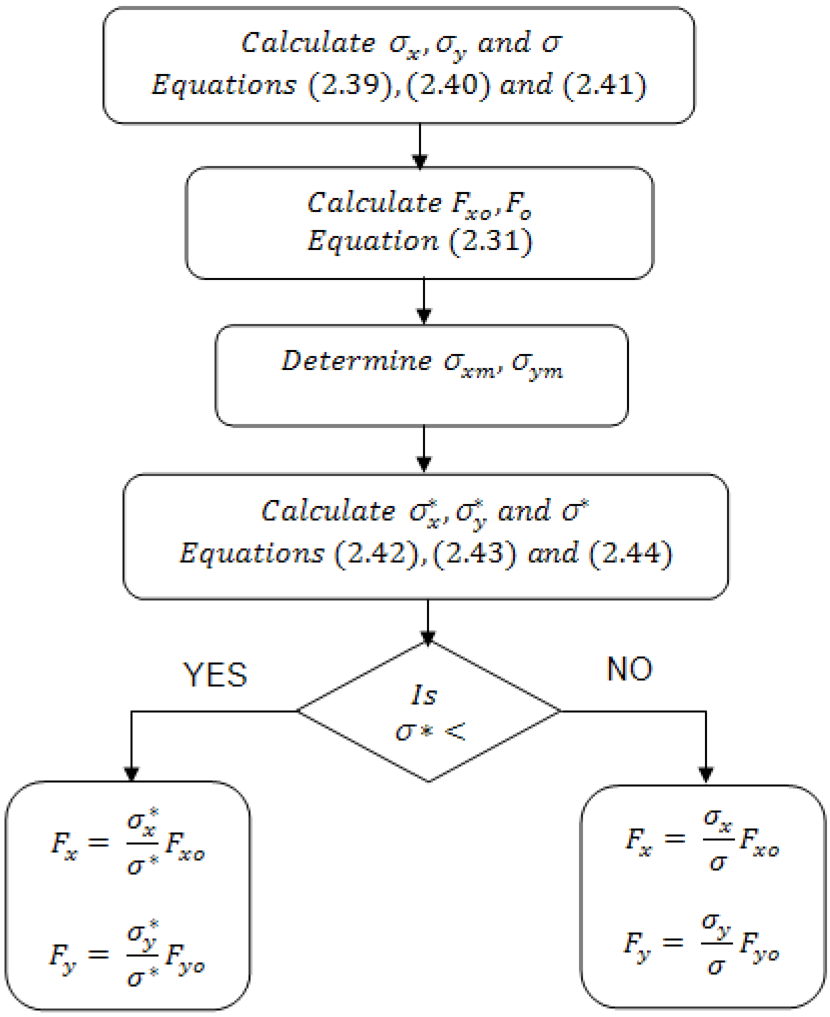

Fig 3.17: Flowchart for Magic Formula tyre force calculations

$$G_{yk} = \frac{\cos\left[C_{yk} \arctan\left(B_{yk} k_S\right)\right]}{\cos\left[C_{yk} \arctan\left(B_{yk} S_{Hyk}\right)\right]} \tag{3.48}$$

And, the combined longitudinal force is described by the following formulae:

$$F_x = G_{x\alpha} \cdot F_{xo} \tag{3.49}$$

where $G_{x\alpha}$ is described as follows,

$$G_{x\alpha} = \frac{\cos\left[C_{x\alpha} \arctan\left(B_{x\alpha}\left(\alpha + S_{Hx\alpha}\right)\right)\right]}{\cos\left[C_{x\alpha} \arctan\left(B_{x\alpha} S_{Hx\alpha}\right)\right]} \tag{3.50}$$

The plots of the combined longitudinal and lateral force modelled are given in figure 3.18 and 3.19.

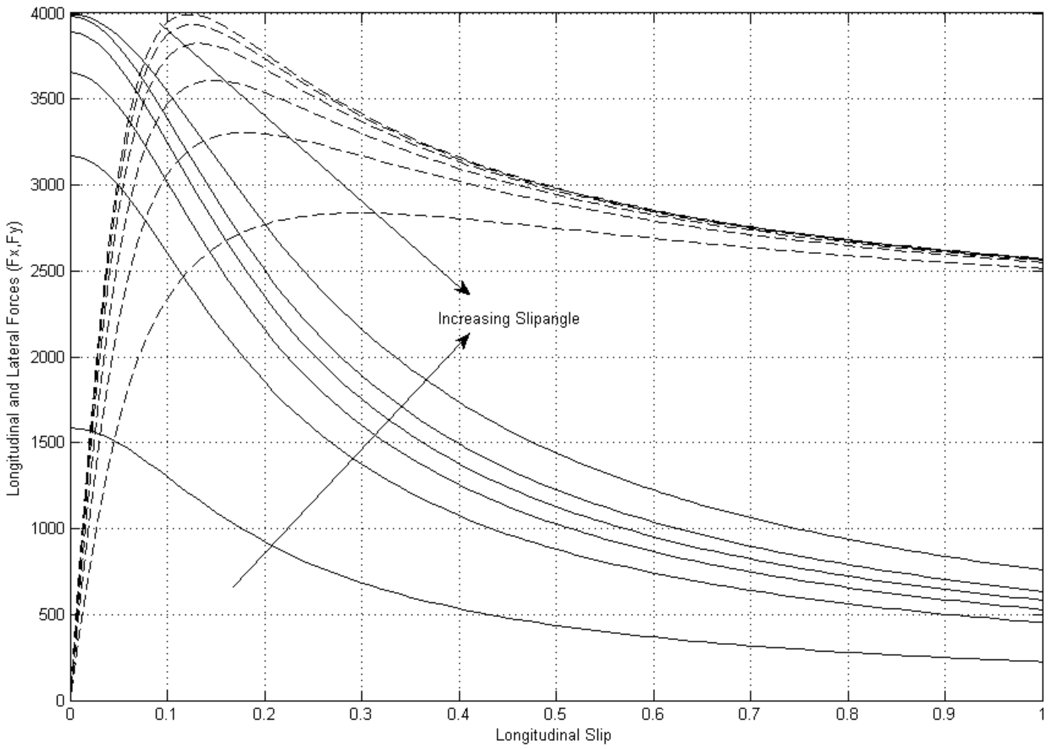

Fig 3.18: Combined Longitudinal and Lateral tyre force Vs slip ratio

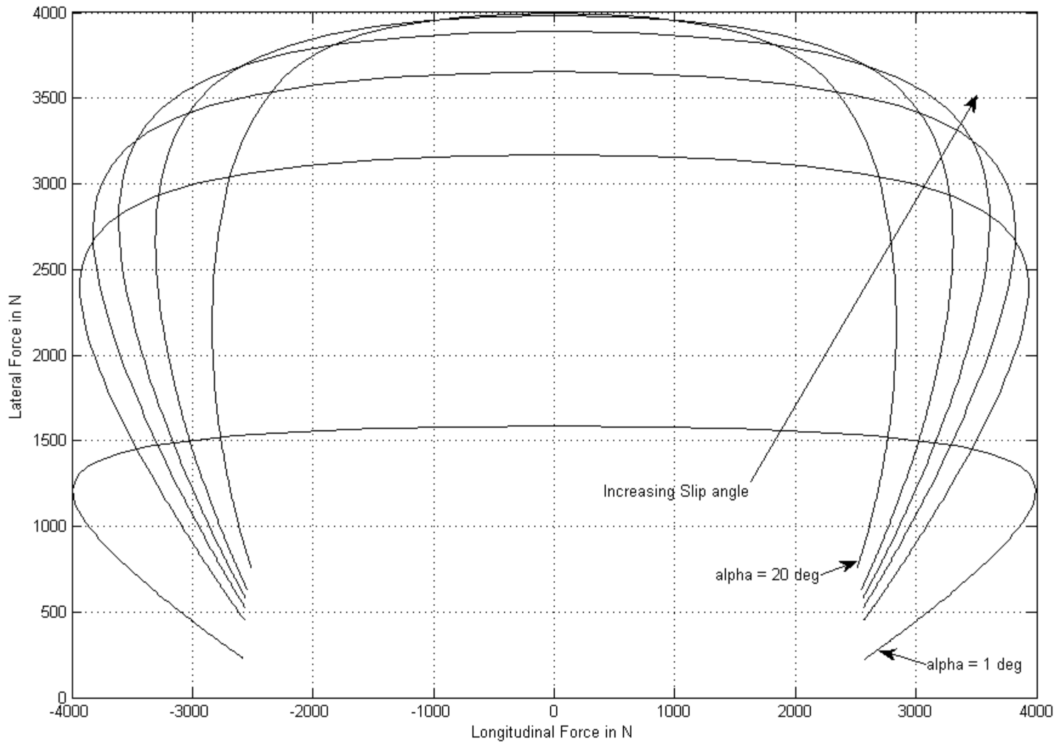

Fig 3.19: Tyre forces during combined braking and cornering

3.5.5 Transient tyre behaviour: The above mentioned magic formula equations are only valid for steady state operating conditions. Under realistic vehicle driving conditions however the influence of the input velocities cannot be neglected. Moreover the tyre carcass has compliance with respect to the rim in both the longitudinal and lateral directions. This causes a lag in the response to the lateral and longitudinal slip. This low frequency behaviour is called transient tyre behaviour or tyre dynamics. Pacejka (2002) explains two methods to account for tyre transient behaviour: the stretched-string model and the contact mass model. Stretch-string model is used in this thesis to account for the tyre dynamics. The deflection of the leading point in contact with the road v_1 can be calculated with the following differential equation:

$$\sigma_{\alpha} = \frac{dv_1}{dt} + |V_x| v_1 = \sigma_{\alpha} V_{sy} \qquad (3.51)$$

The string deflection in the longitudinal direction is given in a similar way:

$$\sigma_{\kappa} = \frac{du_1}{dt} + |V_x| u_1 = -\sigma_{\kappa} V_{sx} \qquad (3.52)$$

In this thesis the tyre transient behaviour is modelled as a first order dynamic system as a function of the tyre relaxation length and vehicle longitudinal velocity.

3.6 Development of Automotive Toolbox in Matlab/Simulink

Simulation of dynamic systems such as vehicles is a complex and time consuming task. Most of the time the modelling tasks need to be repeated in order to perform system analysis such as "if-what" scenarios. Developing a toolbox will modularise the whole modelling process and reduce the model development and analysis time. A. Rodic, (2003), developed a specialised piece of commercial software for modelling, control design and simulation of road vehicles. Poussot-Vassal, (2007), developed a unique toolbox during the course of his doctoral research to analyse active suspension and active brake systems. The author greatly acknowledges the advice provided by both A.Rodic and C.Poussot in developing this toolbox. The automotive toolbox developed in this thesis provides Simulink models and Matlab tools for vehicle dynamic simulation, analysis and development of vehicle dynamic control systems. It has modular Simulink models for quarter car, extended quarter car, half car for roll and pitch, vertical vehicle model, full vehicle model, linear, nonlinear tyre models and other vehicle subsystem models.

This toolbox provides a flexible environment for vehicle dynamic research. It contains libraries with Simulink graphical blocks and Matlab functions, which can be connected to build vehicle models. Using this toolbox, it is also possible to subdivide the whole vehicle model into a number of smaller vehicle subsystems, which can be arranged in a neat way and validated separately. The use of block-diagrams greatly facilitates computer representation of vehicle dynamic systems.

A screen shot of the top layer of the automotive toolbox is given in figure 3.20.

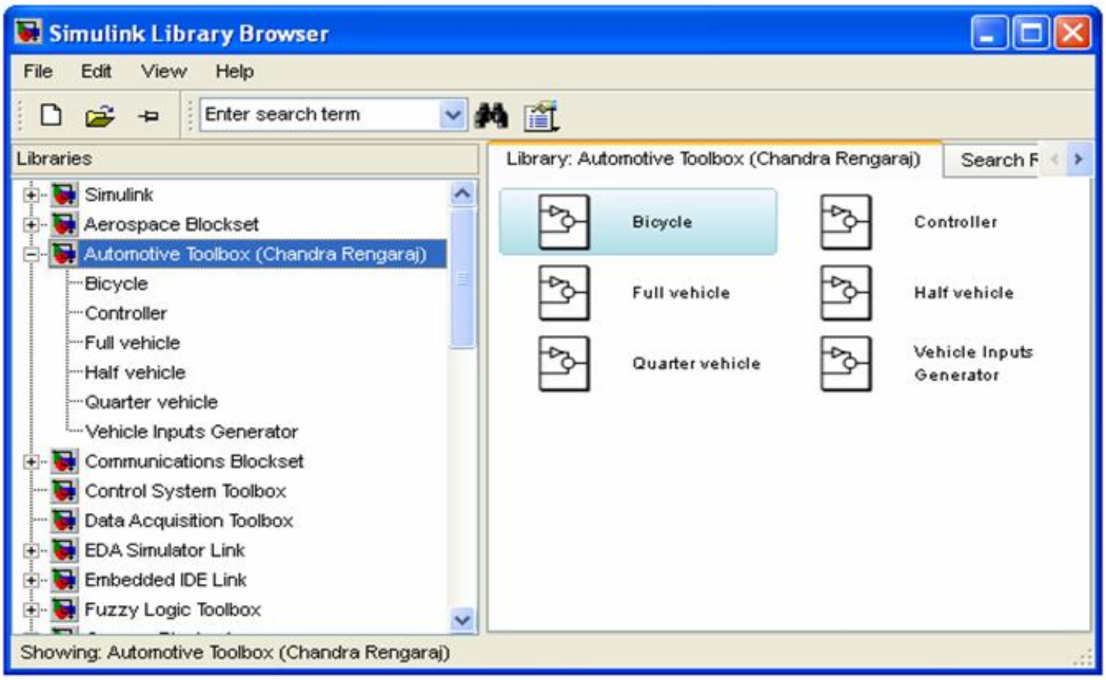

Fig 3.20: Screen shot of the automotive toolbox developed for this thesis

3.7 Description of Matlab/Simulink Vehicle Model Developed

This section describes the detailed vehicle model developed as a part of this research which became part of the above automotive toolbox as mentioned in the earlier section. The full passive vehicle model is developed based on the fundamental vehicle dynamic equations mentioned earlier in this chapter. Figure 3.21 shows the top layer of the vehicle model. The model is set to receive inputs from the active systems and the driver. It provides the simulated vehicle dynamic parameters as outputs to post processing.

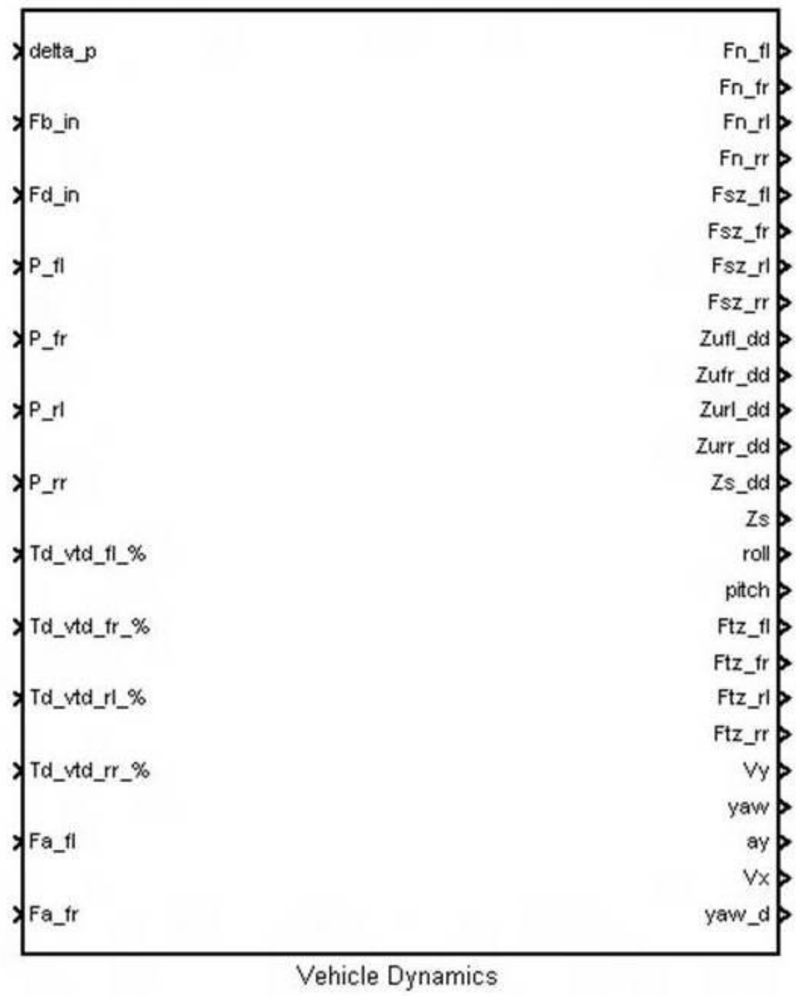

Fig 3.21: Screen shot of the Vehicle Model Developed - Top Layer

The next lower layer has blocks to calculate vehicle sprung mass positions, normal tyre loads, suspension and tyre forces, vehicle sprung and un-sprung mass accelerations, steering, brake and drive dynamics, vehicle longitudinal, lateral, yaw, roll and pitch dynamics. The tyre forces block includes wheel dynamics and tyre lateral slip angle calculations. Matlab embedded function approach is used for the computations of most of the parameters for ease of use and effective data handling.

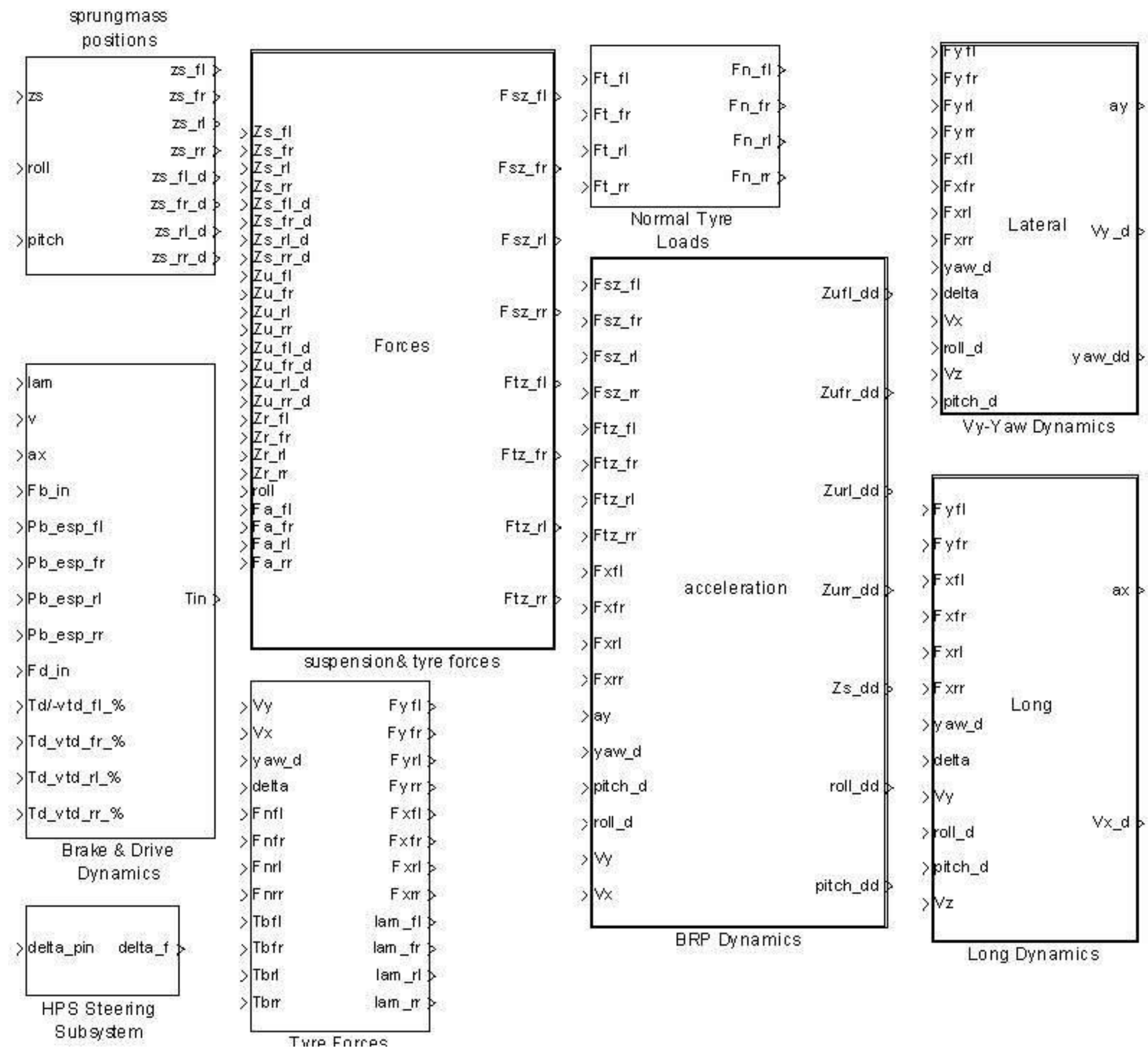

Fig 3.22: Screen shot of the Vehicle Model Developed – Layer 2

An overall view of the full passive vehicle model is given in figure C1 on Appendix C which is used for the simulations of vehicle dynamics test manoeuvres described in the next section.

3.8 Description of Test Manoeuvres

There are a number of test manoeuvres that can be simulated to determine the effectiveness of vehicle dynamics. As the focus of this thesis is mainly on the improvements of vehicle handling in a stability point of view, test manoeuvres that can analyse a vehicle's stability are only considered. All the test manoeuvres defined are based on the corresponding ISO standards. Three vehicle parameters widely used to define a vehicle's stability are the yaw rate, the vehicle side slip angle and the lateral acceleration of the vehicle (popularly known as latac) These are the design parameters in this thesis and an improvement in vehicle stability is considered as a reduction in the yaw rate and sideslip angle at a given latac.

3.8.1 Straight-line braking

Objective: According to ISO21994, this test is used to evaluate actual braking deceleration and vehicle stability. But the overall objective of this test is to demonstrate a design of the braking system which is suitable for the particular vehicle by combining good levels of comfort (responsiveness, operating force, etc.) with the shortest possible stopping distances. According to statutory requirements a brake system must assure a vehicle deceleration of up to 0.8 g and above. And the front wheels always lock before the rear wheels, because locking rear wheels result in the vehicle's instability.

Test Procedure: The test is carried at an initial speed of 100km/h with maximum brake pressure until the vehicle comes to a complete halt. During the test, the following data must be logged as per the ISO 21994:

- Vehicle speed
- Time when braking begins
- Braking distance over the defined measurement duration

- Brake pedal force (or brake pressure in brake master cylinder)

Characteristic parameters for the **deceleration ability** of a vehicle include, for example:

- Braking distance as a function of initial speed or
- Average deceleration as a function of brake pressure.

To evaluate vehicle stability and directional stability, the following characteristics, for example, are measured as per the literature:

- Lateral deviation across the braking distance
- Yaw speed across the duration of braking (average deceleration)

This test is also used to evaluate the performance of ABS systems in straight line braking situations.

3.8.2 Step steer input

Objective of the Driving Manoeuvre: According to ISO 7401 this test serves the main objective of describing the transient dynamic behaviour of a vehicle. It defines characteristic values and functions required for both the time domain and the frequency domain. Key criteria in the time domain include, among others:

- Time shift between steering wheel angle, lateral acceleration and yaw speed
- Gain factor of yaw speed

Key criteria in the frequency range include, among others:

- Lateral acceleration related to steering wheel angle
- Yaw speed related to steering speed

Test Procedure: From straight-line driving at a constant speed of approx. **80 km/h** the steering wheel is moved as fast as possible to the angle position that will result in a lateral acceleration of 4 m/s². The test can be repeated with various steady state accelerations such as 6, 8 m/s² by either varying the steer angle keeping the vehicle speed constant or vice versa. . According to ISO 7401 the following data must be logged during the test:

- Steering wheel angle
- Lateral acceleration
- Yaw speed
- Steady-state float angle
- Longitudinal speed
- Lateral speed or unsteady float angle
- Roll angle
- Steering wheel torque
- Forces and moments acting on the wheels
- Slip angle on the wheels

The vehicle's response to sudden step steering input enables statements to be made about the speed of response, vehicle stability under the existing conditions as well as the precision of the steering system. In case of a major phase delay between steering wheel input and yaw speed the vehicle can be perceived as inert and possessing poor cornering ability.

If, during the change from the unsteady to the steady-state phase of the step steering input, yaw speed and lateral acceleration exhibit large amplitudes and long transient periods, then vehicle stability may be jeopardized.

The gain factor, the quotient of yaw speed and the steering wheel angle, is a measure of how much steering angle the driver needs in order to generate a certain yaw response. A precise steering system is characterized by a large gain factor.

3.8.3 Double lane change (ISO 3888 – Part 2)

Objective of the Driving Manoeuvre: Originally, this manoeuvre was named "moose test" or "elk test" and designed to provide a criterion to prove the tilt stability of a vehicle. In Scandinavian and American countries, these animals sometime cross the road. When it occurs, the driver has to perform a quick avoidance manoeuvre that may destabilise the vehicle. An actual scenario is shown in figure 3.21. Today, this test is named the double lane change manoeuvre and is widely used in the automotive industry as a means to evaluate the stability of a vehicle, and in the development of active stability systems such as ESP. The international standard for this test manoeuvre is described in ISO 3888 – Part 2.

Test Procedure: The test procedure consists of driving a vehicle through a set track, which simulates a double lane change manoeuvre. The vehicle is driven at 80km/h from the initial lane to the parallel lane and back to the original lane. During the test, significant movement parameters such as vehicle longitudinal and lateral speeds, lateral acceleration and steering wheel angle are measured.

Fig 3.23: Moose crossing a road, Alaska, USA

The drive-in speed is increased step by step up to the maximum vehicle speed or 110 ± 3km/h (whichever is maximum), and none of the cones may be touched during the lane change test.

Summary: This test is suitable for demonstrating how precisely, fast and spontaneously the vehicle responds to the driver's steering angle inputs.

3.8.4 Braking on split-mu

Objective of the Driving Manoeuvre: This test manoeuvre examines a vehicle's ability to maintain straight ahead directional stability during braking on a split mu surface.

Test Procedure: In this test the vehicle is driven straight ahead at a speed of 100km/h on a split mu surface where the wheels on the left side of the vehicle are on an icy surface ($\mu = 0.2$) and the wheels on the right side on a dry surface ($\mu = 1.0$). A step input in brake torque, which produces a longitudinal deceleration of –0.4g, is then applied. The difference in brake force generated between these two frictional surfaces causes a yaw moment about the center of gravity of the vehicle and destabilises the vehicle about the vertical axis.

Summary: This test evaluates a vehicle's stability in split-mu braking situations and is suitable for the development of active chassis control systems, such as ABS, ESP.

3.9 Vehicle Model Validation

This section describes the validation of the full vehicle model developed in the previous sections. The handling dynamics are evaluated and the simulation results compared against industry standard software.

Any software vehicle model developed needs to be validated either against experimental results or against other proven simulation software results. The vehicle model developed in this thesis is validated against the well-known commercial software called CarSim. CarSim is a vehicle dynamics simulation software developed by Mechanical Simulation Corporation in Ann Arbor, USA. It is a parametric modelling software widely used both in academia and industry to simulate, predict and analyse vehicle dynamic behaviour.

The validation methodology consists of three phases:

- describing the validation test condition and procedures,
- simulation of full vehicle model and
- comparison of simulation prediction with the CarSim vehicle model simulation data.

In order to be used in this research the model developed must be capable of evaluating vehicle dynamics both in normal and limit driving situations. Two standard test manoeuvres are used to evaluate the vehicle model. First a step steer input at a constant speed was provided so that it generates a lateral acceleration of 0.3g, 0.6g and 0.8g respectively. This evaluates the model across all the lateral acceleration range from low to the limit handling. The results of the step steer input are shown in figures 3.24, 3.25, 3.26 and 3.27.

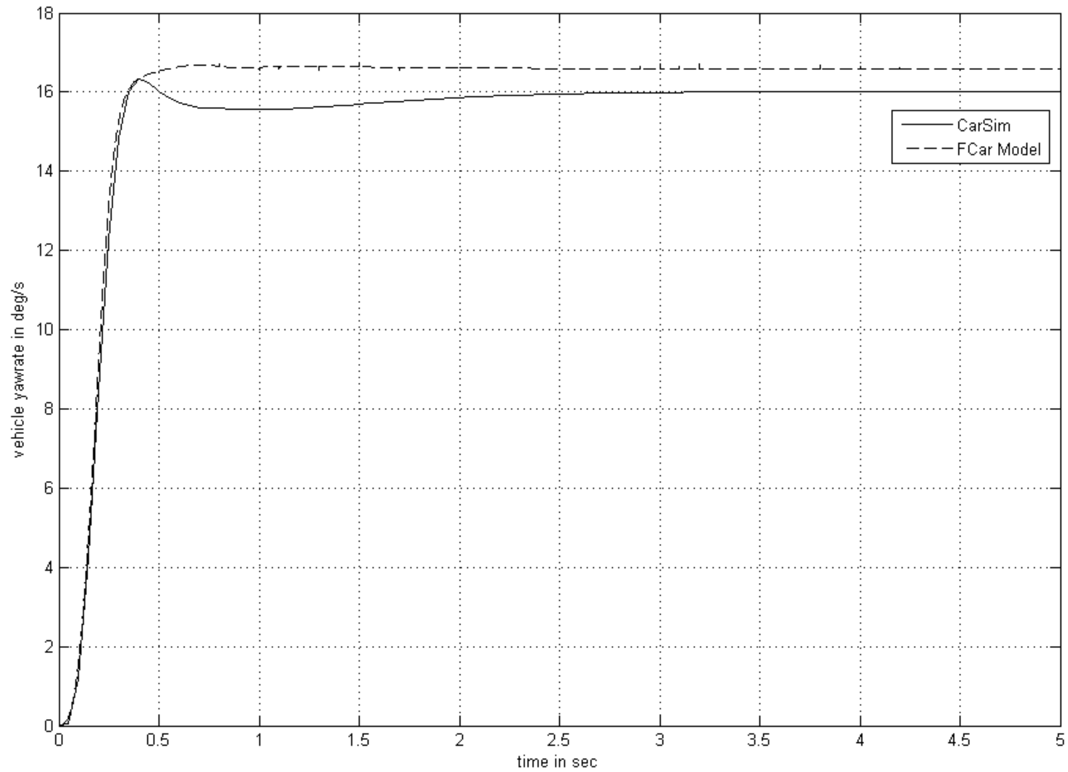

Fig 3.24: Comparison of yaw rate at 0.3g latac

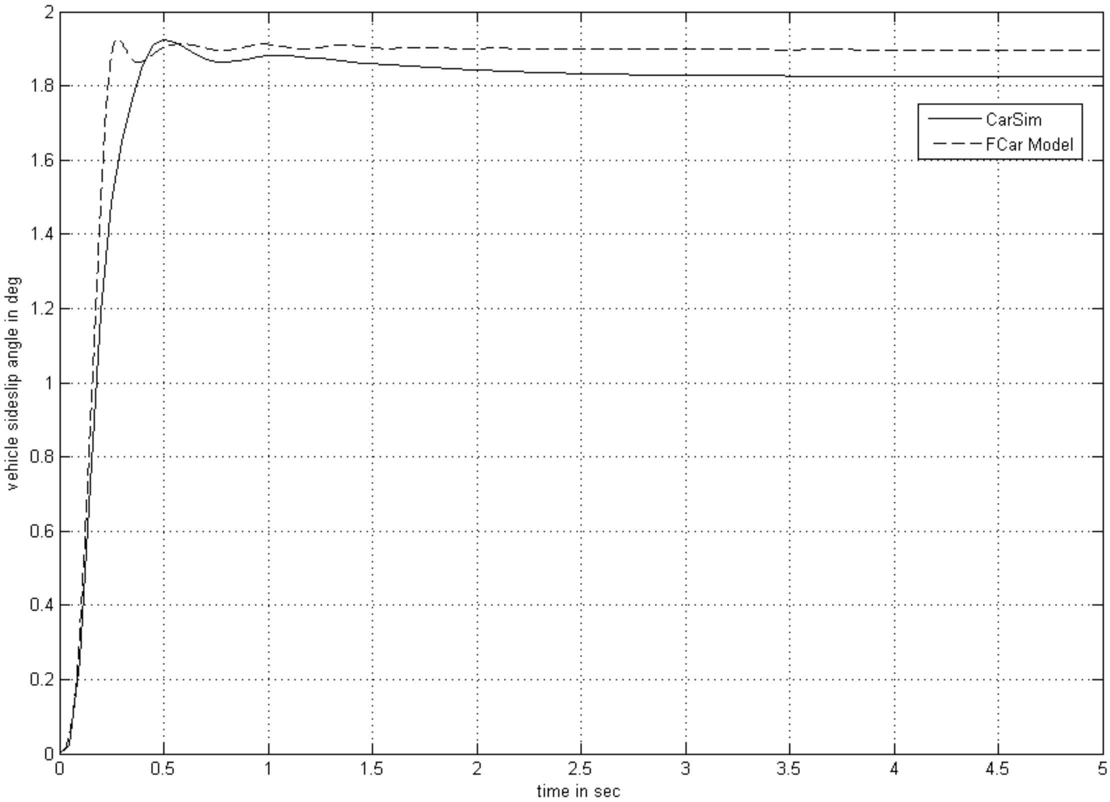

Fig 3.25 Comparison of vehicle side slip angle at 0.3g latac

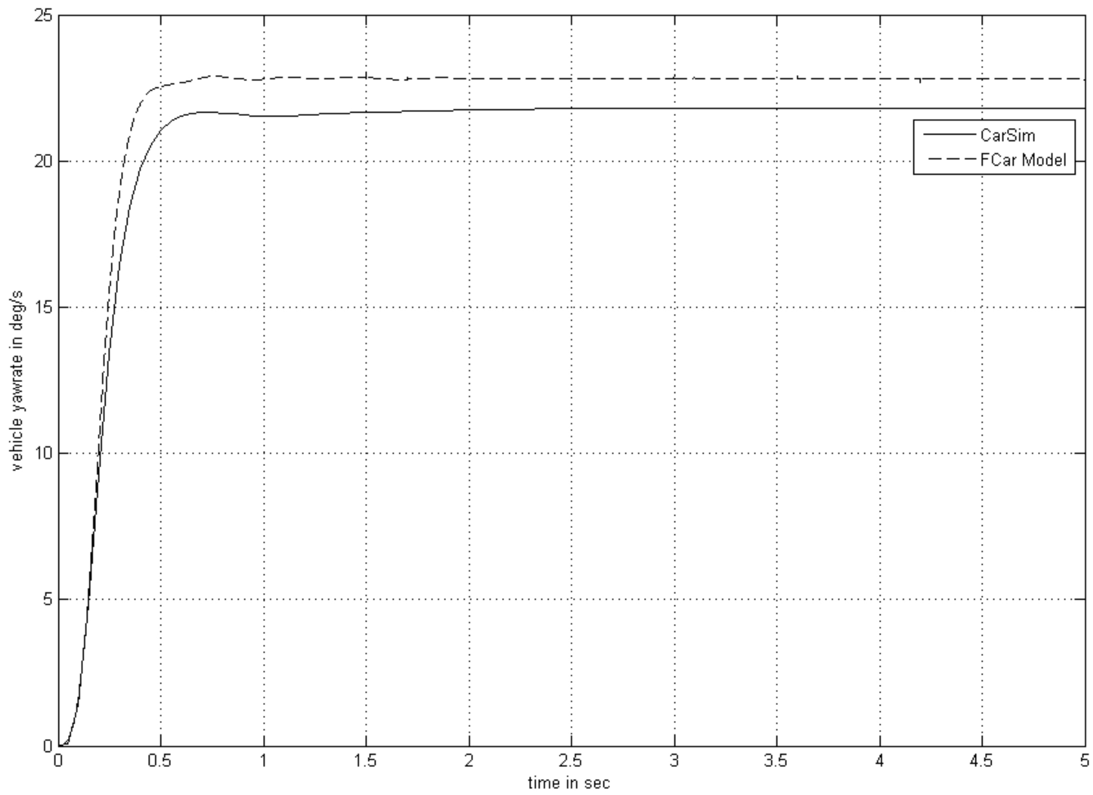

Fig 3.26 Comparison of yaw rate at 0.6g latac

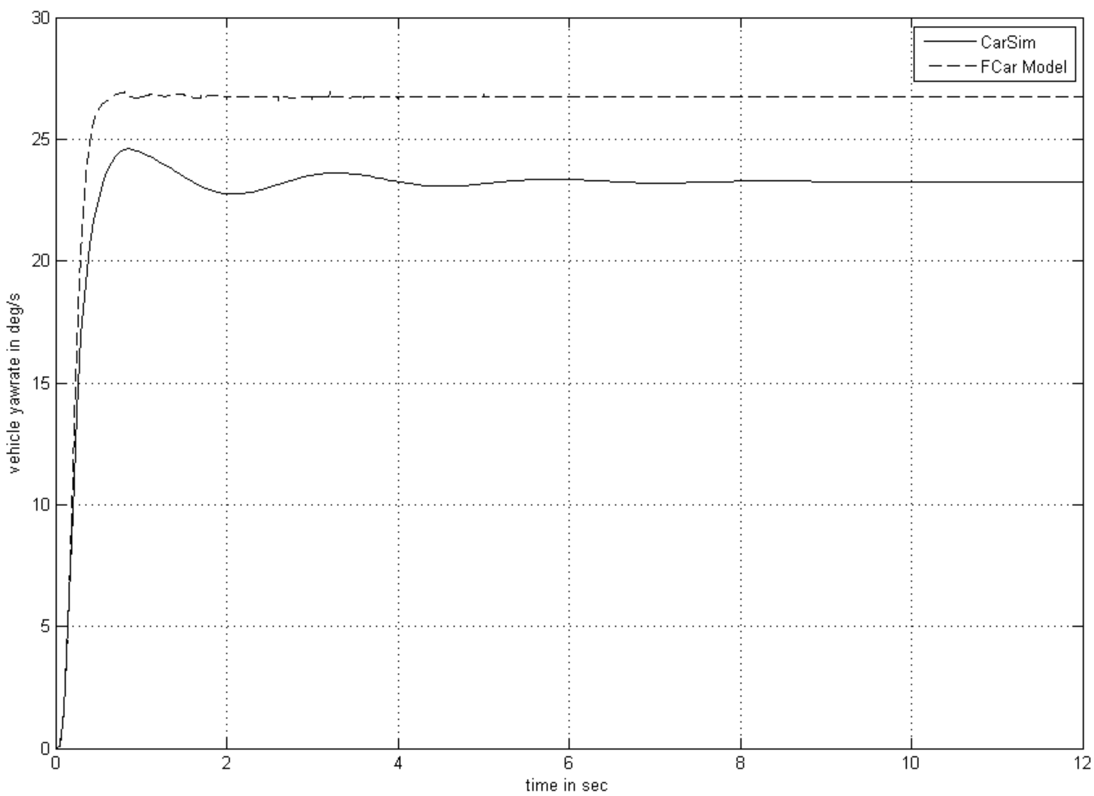

Fig 3.27 Comparison of yaw rate at 0.8g latac

Then a double land change manoeuvre was also performed to validate the vehicle model. The test was performed at a speed of 80km/h on a flat dry surface whose coefficient of friction was 1(μ = 1). First the test was carried out using CarSim software. The vehicle parameters for a D Class Sedan were used. The results were imported to Matlab/Simulink workspace. Then the Full vehicle model was characterised with the CarSim vehicle parameters. The same steering data used to simulate the CarSim model was used as steering input to the full vehicle model. The simulation results of the full vehicle model are plotted along with the CarSim results for comparison in figures 3.28, 3.29and 3.30.

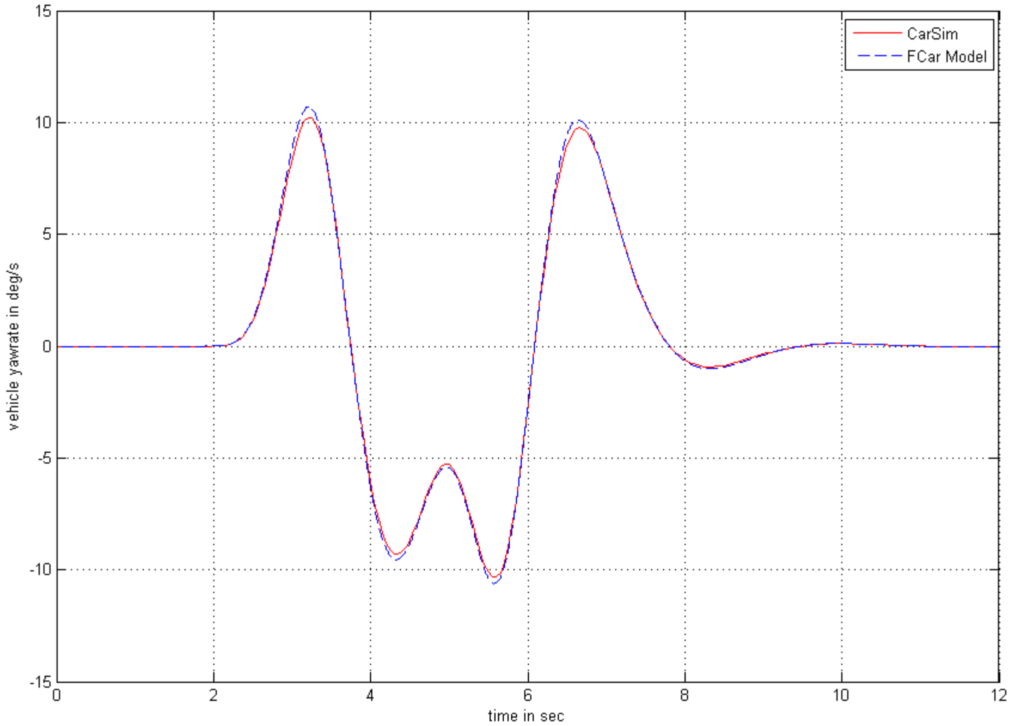

Fig 3.28: Comparison of vehicle yaw rate between CarSim and full vehicle model during an 80km/h double lane change manoeuvre

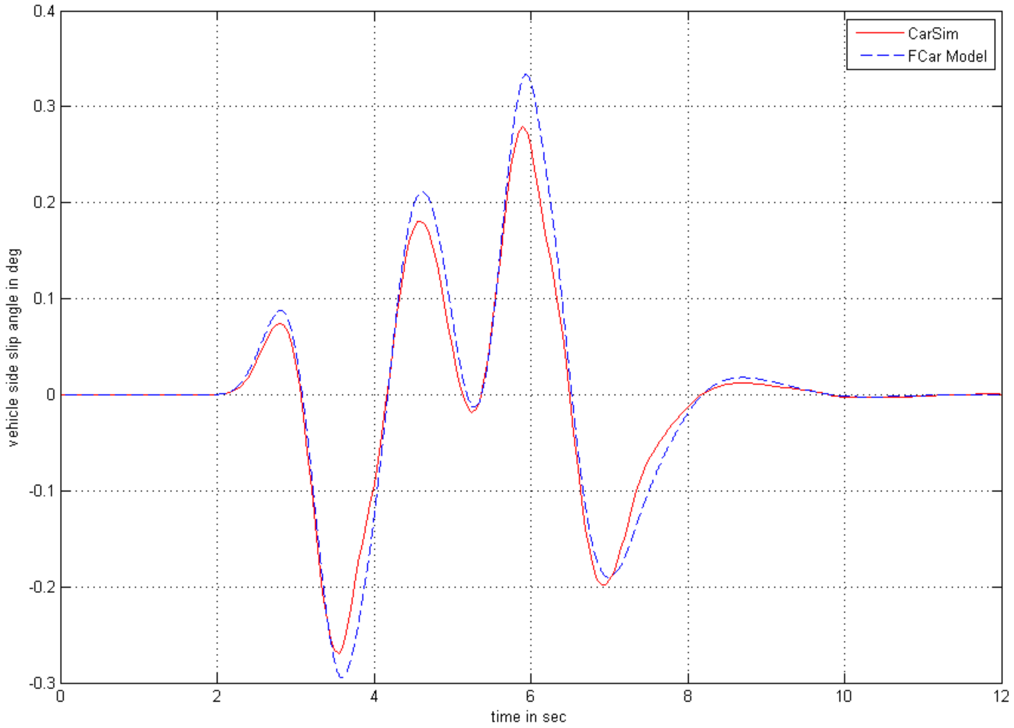

Fig 3.29: Comparison of vehicle sideslip angle between CarSim and full vehicle model during an 80km/h double lane change manoeuvre

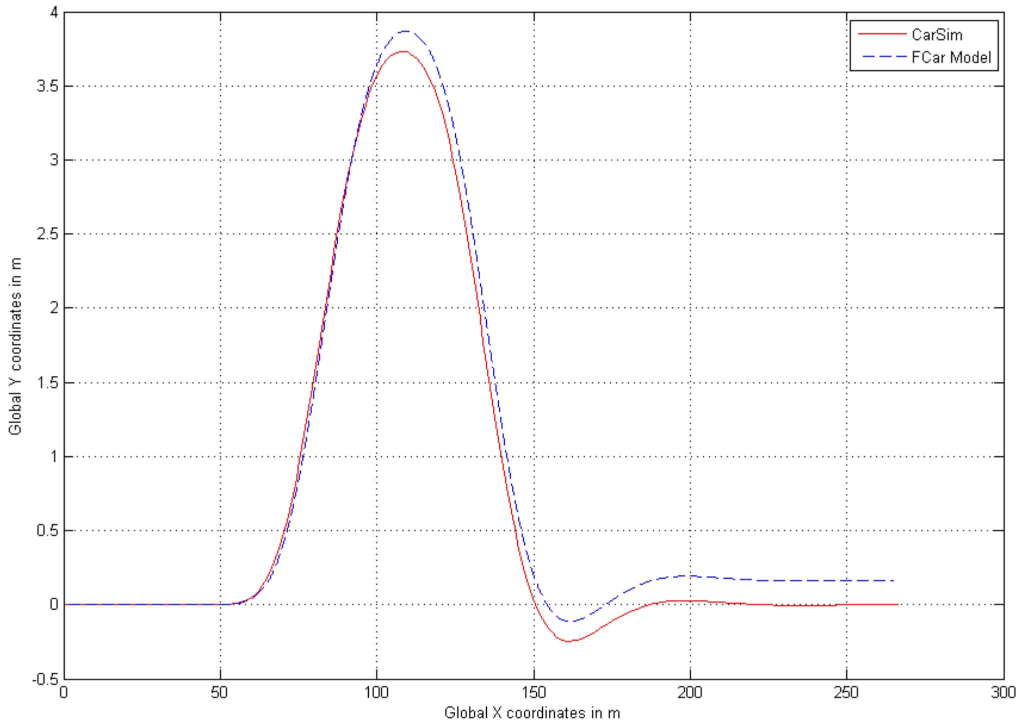

Fig 3.30: Comparison of vehicle path between CarSim and full vehicle modelduring an 80km/h double lane change manoeuvre

From the above simulation results it can be concluded that the responses of the full vehicle model developed follow closely the responses of the CarSim vehicle model across various lateral acceleration ranges. The minor deviations observed in the medium and high latac range are largely due to the differences in the suspension kinematics between the models, the nonlinearity in the suspension elements in CarSim and limitations in transferring all the CarSim vehicle parameters into the full vehicle model. Moreover, as the CarSim model is validated against real-time experiments, a conclusion can be derived that the full vehicle model is also validated indirectly against experimental results. So it can be concluded that the full vehicle model developed is on a par with the widely used commercial software vehicle model and is suitable to use in the vehicle dynamics studies such as integrated chassis control systems.

3.10. Summary

This chapter discussed the development of passive vehicle dynamics models. It began by explaining the basic theory of vehicle dynamics followed by various vehicle dynamics models developed with increasing complexity. A detailed discussion about the development of tyre models for simulation purposes is then carried out. Then the modelling of four major vehicle subsystems along with the wheel dynamics was discussed followed by a discussion on vehicle handling dynamics. A description about the automotive toolbox developed for this thesis in Matlab/Simulink was presented. Finally some of the important standard test manoeuvres used internally to evaluate the vehicle handling dynamics was described followed by the validation of the passive vehicle dynamics model developed against a commercially available software model.

Chapter 4

Modelling of Active Vehicle Dynamics

4.1 Introduction

Modern day automobiles have originally been developed as a combination of passive systems. These vehicles perform well under typical operating conditions, such as dry and smooth roads at moderate speeds. Under these conditions, they produce predictable dynamic behaviours to driver inputs, such as steering, accelerating and braking. This part of the operating region of a vehicle is called the linear operating / driving region. As long as the vehicle is within this linear region the driver enjoys the driving and feels safe and confident. However, in adverse operating conditions such as slippery (caused by rain or snow), uneven roads and / or at higher vehicle speeds, the dynamic behaviour of the vehicle to the driver's inputs is no longer linear and becomes unpredictable. This part of the region of driving is called the non-linear operating / driving region. Operating a vehicle in this region increases the driving stress and reduces the safety of the vehicle, its occupants and of course pedestrians. This drawback of a passive vehicle is tackled to some extent by active chassis systems. Thus, the primary motivation of active vehicle dynamic control systems is to increase the range of conditions under which the vehicle behaves predictably (Fodor M et al, 1998, Hardy D et al, 2004). Additionally, active chassis control systems can be used to enhance the vehicle comfort and response under typical operating conditions.

4.2 History of Active Vehicle Dynamics

Active controls have been applied to various automobile subsystems for nearly four decades. Some of the technologies were originally developed for rail and aircraft industries and found their way into the automotive industry. One such system is the anti-lock brake system popularly known as ABS. ABS started its journey in automobiles in the early 1970s. It is a system designed to avoid locking of the wheels of a vehicle during a sudden excessive braking process and at the same time it maintains the driver’s ability to steer the vehicle. ABS primarily influences the longitudinal dynamics or motion of a vehicle.

Tractional Control System also known as TCS is another system that influences the longitudinal dynamics of a vehicle but in the opposite direction and it is much similar to ABS in operation. TCS avoids spinning or slipping of the wheels when sudden excessive torque is applied while starting or cruising on slippery roads or slopes (Jung.H. et al., 2000). The two most popular principles used to prevent a wheel from spinning are, either applying brake torque or cutting of the drive torque momentarily (e.g. through spark or injection cut) on the particular wheel that is spinning. When awareness about pollution due to vehicle exhaust emissions became more of a concern, electronically controlled engine management systems (EMS) were invented. An EMS system optimises the engine air-fuel ratio for the specified engine performance and reduces the pollution from an engine. Parts of the world such as USA and Europe, where long-straight stretches of highway are the norm in road design for inter city traffic, felt the need for an active system that maintains the speed without the driver pressing down the accelerator pedal continuously. This lead to the invention of adaptive cruise control (ACC) which maintains the set vehicle speed as long as there is no intervention from the driver either through the brake / accelerator pedals.

Electronic Stability Control (ESC) also known by many names such as electronic stability programming (ESP), Active Yaw Control (AYC), Vehicle Stability Control (VSC) etc., is a brake and/or driveline based vehicle dynamic control system designed to improve the stability of a vehicle. It was initially proposed by Shibahata et al, (1992), Matsumoto et al, (1992) and Inagaki et al, (1994). This technology was later commercialised by Robert Bosch Gmbh under the name of

Vehicle Dynamic Control (VDC). ESC influences the Lateral and yaw dynamics of a vehicle. Active steering is another domain where active control systems played a key role in improving the vehicle lateral dynamics. There are many active steering systems developed such as active front steering (AFS), active rear steering (ARS), four wheel steering (4WS) etc. Active vehicle control system technology has also been applied to the vehicle drive train and lead to the invention of various chassis control systems such as Variable Torque Distribution (VTD), All Wheel Drive (AWD) etc. These systems are similar to ESC but use the just the drive torque as the control input.

A continuous drive to strike a balance between occupant ride comfort and vehicle handling has lead to the invention of active suspension systems. In general, electronic control of vehicle suspensions can be divided into semi-active (SAS) and fully active systems (ASS). In a SAS, the damper coefficient is controlled to alter the applied suspension force on the vehicle sprung and unsprung masses. Systems such as continuous damping control (CDC), Magneto Rheological (MR) Damper belong to this category. An ASS, a hydraulic actuator is used to provide the suspension force required to affect the dynamics of the sprung and unsprung masses to improve the vehicle ride and handling dynamics. Another suspension based active control system is a roll moment distribution (RMD) system that uses an active anti-roll bar to distribute different roll moments between the front and rear axles and amongst the wheels, and improves the vehicle dynamics behaviour. Normal force controller (NFC) is a type of active suspension system used to optimise the tyre normal forces on a particular wheel to enhance the lateral stability of a vehicle. In general these active suspension control systems influence the vehicle dynamics in vertical and lateral directions.

Many such active systems have been developed. According to Crolla. D.A. (2005), more than 40 such systems exist for the use of today's modern automobile. A detailed analysis of these systems reveals that all can be clubbed together into four broad categories: braking, steering, suspension and driving, the four basic functions of any automobile. For example systems such as ABS, ESP etc. belong to active brake systems. AFS, ARS, 4WS belong to active steering systems. Active suspension systems include CDC, RMD, ASS etc and TCS, AWD, VTD belong to active driveline systems.

As the focus of this thesis is to develop an integrated control system amongst these four vehicle functional domains, one active chassis control system from each of these domains is chosen for the purpose of active vehicle dynamics modelling and further analysis.

The four systems chosen are a brake based electronic stability control (ESC), active front steering (AFS), normal suspension force controller (NFC) and a driveline based variable torque distribution (VTD). These systems were finalised considering their wide availability in today's modern vehicles and the influential role they are going to play as a part of global chassis control in the future. The following sections of this chapter will discuss the development and analysis of these four active chassis systems.

4.3 Modelling of Anti-lock Brake system (ABS)

4.3.1 Mathematical model of the dynamics of brake system

Modelling the dynamic behaviour of a brake system plays an important role in designing the control system. It is a widely used practice to model the dynamics of brake system as a combination of time delay and first order dynamics (Allyene.A, 1997, Pilutti. T, et al,1998, Eldemerdash.S.M et al., 2006). But detailed models accounting for the brake hydraulics give more realistic brake behaviour. The hydraulic brake considered in this thesis is based on the model used by Fletcher. I. et al, 2004 and consists of the following components: a mechanical brake pedal, a servo brake booster, a master cylinder, proportioning valve, hydraulic brake callipers and the friction pads for each wheel. A schematic of the hydraulic brake structure used is given in figure 4.1.

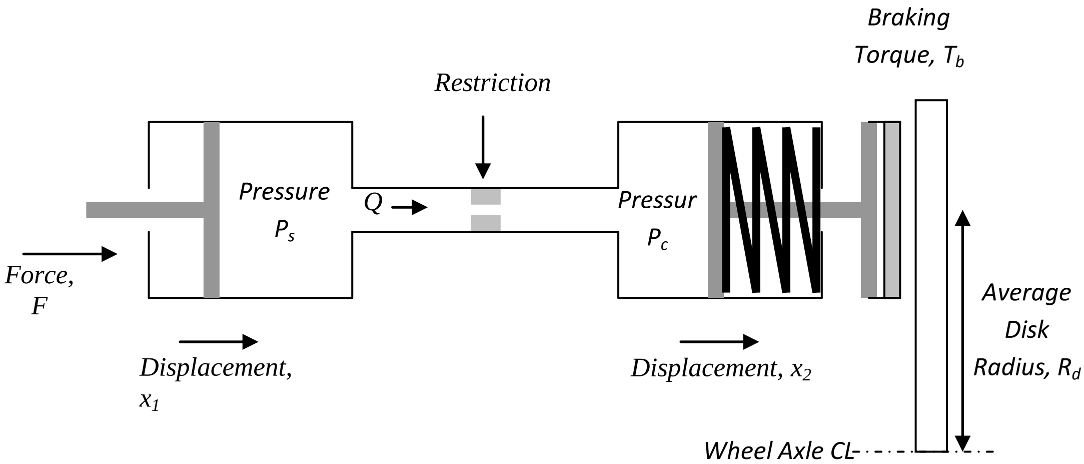

Fig. 4.1 Schematic of the brake hydraulics

The brake mechanics considered in this study are explained below. The mechanical brake input is amplified by the servo booster. This is further amplified and converted to a hydraulic pressure called line pressure/supply pressure, which is fed through the brake lines. The hydraulic equation governing the non-linear laminar, incompressible flow of brake fluid through brake pipe lines is as follows:

$$Q = C_d A \sqrt{\frac{2}{\rho}(P_s - P_c)} \tag{4.1}$$

Where, Q = the flow rate of the brake fluid in m^3/s.

A = the area of the brake pipe in m^2.

P_s and P_c = the supply and calliper pressures in N/m^2

ρ = the density of the brake fluid in kg/m^3

C_d = the coefficient of discharge

The development of brake torque at the wheels is a very complex nonlinear process which is a function of many factors such as the brake disc temperature that develops during a braking process. Neglecting those factors for the sake of simplicity, brake torque can be expressed accurately to a great extent as a linear function of the caliper pressure as per the following equation:

$$T_b = \mu_d A_{wc} r_{bd} P_c \tag{4.2}$$

Where, μ_d = the friction coefficient of brake disc,

A_{wc} = the area of the wheel calliper.

r_{bd} = the radius of the brake disc.

μ_d, A_{wc}, r_{bd} , can be clubbed together as K_b , the brake constant.

During braking, the dynamic load transfer takes place from rear to the front wheels, which leads to an increase in the tyre normal force at the front wheels and a decrease at the rear wheels. This dynamic load transfer stresses the need for a proportioning of brake pressure between the front and rear wheels or else the excess pressure at rear wheels will lock them during braking. The brake pressure between front and rear wheels can be proportioned based on many parameters such as the vertical load on front and rear axles, the deceleration of the vehicle etc. A vertical load based proportioning is basically done for preset factory conditions, e.g., laden and un-laden vehicles require two completely different pressure settings. Similarly the setting for a dry road will not work for icy or wet roads as the dynamic load transfer will be different in these situations. Another well-known proportioning method is based on the vehicle deceleration, which accounts for the change in surface coefficient of friction. As the research work in this thesis is focused only on simulations of a laden vehicle moving at high speeds on different road surface conditions, a deceleration based brake pressure proportioning method as per the following formula is used.

$$P_{brj} = \left(\frac{l_f * g + a_x * h_{cg}}{l_r * g - a_x * h_{cg}} \right) * P_{bfj} \tag{4.3}$$

The theory of brake performance triangles (Gillespie, 1992) can be used to explain and to calculate the necessary proportioning of brake pressures for a given vehicle-brake system combination.

4.3.2 Development of ABS controller

Having modelled the necessary dynamics of a hydraulic brake system, the control strategy to be used on the ABS controller needs to be finalised next. Considering the key features of a control system such as ease of design, simplicity to implement, the ability to control nonlinear systems and maintain robustness against parameter variations, a fuzzy logic based ABS controller is developed for use in this thesis.

Fuzzy Logic Control

Fuzzy logic control is conceptually a powerful control strategy based on linguistic variables. According to Anthony and Stanislaw (2000), it provides a means of converting linguistic variables into automatic control variables. Fuzzy control theory, on which fuzzy controllers are based, allows imprecise and qualitative inputs to be processed for decision making. Since fuzzy controllers deal with inaccuracies in a better manner, they are effective at handling uncertainties and nonlinearities associated with complex systems such as vehicle sub systems (Beyer et al, 1993).

Justification for using Fuzzy Control Strategy

Introduction to section 4.3.2 highlights some of the reasons why a fuzzy logic control strategy is chosen. But following are the key factors that influenced the choice of using this control strategy over a conventional control for this research work.

- First of all fuzzy logic control is not an alternative for the conventional control strategies. It is one of the options available and the author has decided to use it for this research work.
- Researchers around the world have successfully used fuzzy logic in designing control systems for automobile applications.
- Fuzzy logic is easy to understand as a concept.
- Since fuzzy logic is flexible, it is easy to change instead of starting the control design process from the beginning.
- Fuzzy logic’s ability to control non-linear systems and to maintain robustness against parameter variations is one of the key factors.
- Matlab / Simulink has a well developed fuzzy logic toolbox which helps the design and development of a fuzzy logic control system for the Matlab / Simulink based vehicle model developed for this thesis.
- Last but not the least, the author is interested in using fuzzy logic as a control strategy in his research work as fuzzy logic control can be built upon the experience of experts. The author’s many years of industry and academic experience on vehicle dynamics and automobiles is an asset in choosing the fuzzy input-outputs, linguistic variables and values, building

the rule base, defining the range for input and output scaling factors if necessary, and tuning of the output gains, membership functions, universe of discourse etc to match with the vehicle dynamics.

Process undertaken in Designing Fuzzy Control Systems

The fuzzy controller is to be designed in such a way that how a human expert who is successful at this task would control the system. Even though the design process of fuzzy logic controller varies to a certain extent depending upon the application, control needs and the expertise of the control engineer, the overall process is quite straight forward. Craig K., (2001) discusses one such design process for fuzzy logic controller development which is followed in this research.

- **Choosing the inputs and outputs**

It is important to make sure that the fuzzy controller has the proper information to make good decisions and has proper control inputs to be able to drive the system in the direction needed to be able to achieve high performance. In practical situations such this research, we have choice in choosing the inputs and outputs of the fuzzy control system. The choice of the controller inputs and outputs is a fundamentally important part of the controller design process.

- **Choosing the linguistic variables and values**

The linguistic variables and values provide a language for the experts to express his/her ideas about the control decision making process in the context of framework established by the choice of fuzzy controller inputs and outputs as made earlier. Then linguistic quantification is used to specify a set of linguistic rules that captures the experts knowledge about how to control the plant. The linguistic rules are formed from linguistic variables and values. The number of linguistic rules is a function of the number of linguistic variable and values. A fuzzy rule table is formed at the end of this process.

- **Choosing the input and output membership functions**

Membership functions are used to quantify the meaning of linguistic values. The definition of membership function is subjective hence it is a choice of the control engineer. The shape of the membership functions can be defined as a function that suits the control task from the point of view of simplicity, convenience, speed and efficiency. Membership functions are specified for each linguistic variable for each fuzzy input and output. For example if a fuzzy logic system has two inputs and one output and each of the input and output has five linguistic variables each, then a total of fifteen membership functions are specified. The Fuzzification and Defuzzification are explained in the respective controller development sections for each active system.

- **Choosing the input/output scaling**

The input and /or output scaling gains can be applied if necessary. The change in the scaling gains at the input and the output of fuzzy controller can have significant impact on the performance of the resulting fuzzy controller. These scaling gains can be used to normalise a fuzzy controller.

- **Tuning of fuzzy controllers / Shaping the non-linearity**

In general the following three parameters can be used as good candidates for tuning a fuzzy logic controller.

- Scaling Gains – Since the change in the scaling gains at the input and the output of fuzzy controller can have significant impact on the performance of the resulting fuzzy controller.
- Universe of Discourse
- Output Membership functions – Membership functions are a good candidate for tuning fuzzy controllers. But the problem is that there are too many parameters, such as, membership function shapes, positioning, number and type of rules etc...

But the ultimate goal of tuning is to shape the non-linearity that is implemented by the fuzzy controller. This non-linearity sometime called the control surface is affected by all the main fuzzy control parameters. As with conventional control design, a process of trial and error is generally needed. The above mentioned general designed process is followed for all the four active system in this thesis.

The fuzzy logic controller used for ABS in this thesis is a slip controller, where the error between the desired longitudinal slip and the actual slip is driven to zero during braking to avoid locking of the wheels. With respect to the tyre force and slip characteristics in figure it can be understood that the desired slip is the slip where the maximum tyre force is produced. The tyre force-slip characteristics also show that any tyre has three distinctive regions, either during the application of brakes or during acceleration. The linear region is where the tyre generates forces proportional to the tyre longitudinal slip. This happens at lower slip values, generally up to a slip ratio of 5 – 8 %. A further increase in brake application increases the slip, but the tyre force generation becomes nonlinear, i.e. the increase in tyre force is not proportional to the tyre slip and eventually reaches a maximum friction point, thus producing the maximum tyre force. Any further application of braking or throttling increases the slip faster and the tyre force generation suddenly starts to decrease and/or to saturate in magnitude. The tyre then accelerates towards the maximum slip (100%) and locks. A locked wheel stops rolling and starts sliding. The friction at this operating point is called the sliding friction, which is much less than the peak friction and hence produces less longitudinal force than the wheel that is operating at the peak friction.

So the aim of a wheel slip controller is to maintain the tyre near the maximum friction point during braking, thereby producing the maximum tyre force that helps to increase the braking performance, at the same time avoiding the locking of the wheels by not letting them slip towards the maximum slip (100%) and thus maintaining the ability to steer the vehicle during braking.

The schematic of the typical ABS control system used in this thesis is shown in figure 4.2.

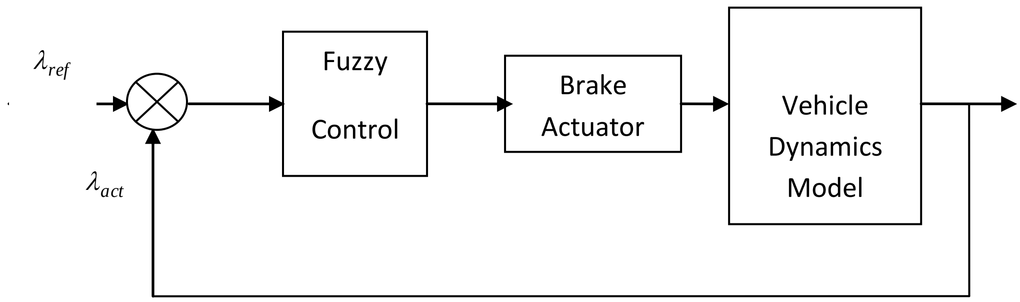

Fig. 4.2 Block diagram representation of anti-lock brake systems

The fuzzy ABS controller used in thesis has two inputs and one output. The error between the desired and actual slips and its derivative are the input variables and the change in the control signal that actuates the brake actuator is the output variable.

$$\lambda_{error} = \lambda_{ref} - \lambda_{actual} \tag{4.4}$$

$$\frac{d\lambda_{error}}{dt} \approx \frac{\Delta\lambda}{\Delta t} \tag{4.5}$$

The two input variables λ_{error} and $\frac{d\lambda_{error}}{dt}$ have three triangular and two trapezoidal membership functions. The membership function of the output variable has similar format of membership functions as the input variables. The control rules are formulated using these input and output variables. The following table describes the control rules used.

Change in Control Signal		$\boldsymbol{\lambda_{error}}$				
$\boldsymbol{\frac{d\lambda_{error}}{dt}}$		PB	PS	ZO	NS	NB
	PB	NB	NB	NB	NS	ZO
	PS	NB	NB	NS	PS	PS
	ZO	NB	NS	ZO	PS	PB
	NS	NS	NS	ZO	PB	PB
	NB	NS	NS	PS	PB	PB

Table 4.1: Fuzzy rules table for the ABS controller

4.3.3 Simulations:

The performance of an ABS system can be evaluated by the following three test manoeuvres - straight line braking (SLB), split –mu braking and a combination of braking (until locking) and steering also known as an emergency avoidance manoeuvre.

Straight line braking manoeuvre reproduces the typical vehicle braking behaviour encountered by drivers on a daily basis. This test normally evaluates the performance of a braking system in terms of the stopping distance in a straight ahead condition. Split-mu braking test is where a vehicle is braked in a straight ahead condition when the left and right side wheels are on surfaces with different coefficients of friction, e.g. left wheels on ice (μ = 0.2 to 0.3) and right wheels on dry road (μ = 0.85 to 0.9). Braking under this condition creates uneven braking forces between the left and right wheels and that causes a destabilising yaw moment whose direction depends on whether the low frictional surface is under the left wheels or the right wheels. Normally this test is used to evaluate the directional stability of a vehicle while braking. The third and final test is used to evaluate the steerability of a vehicle in an emergency braking condition. In order to verify the two main objectives of an ABS system, stopping at a shorter distance without locking the wheels and the ability to steer the vehicle during braking, only the first and third tests are simulated here.

Initially, a vehicle with a passive braking system (without ABS) is simulated on a dry road. The vehicle is assumed to be travelling at an initial speed of 27.8 m/s (100km/h). The road surface condition is assumed to be dry (the road surface coefficient of friction, $\mu = 1$) and a gradual braking input is applied. The performance of the vehicle can be observed by monitoring the following output variables, vehicle velocity, four wheel angular velocities, stopping distance and stopping time.

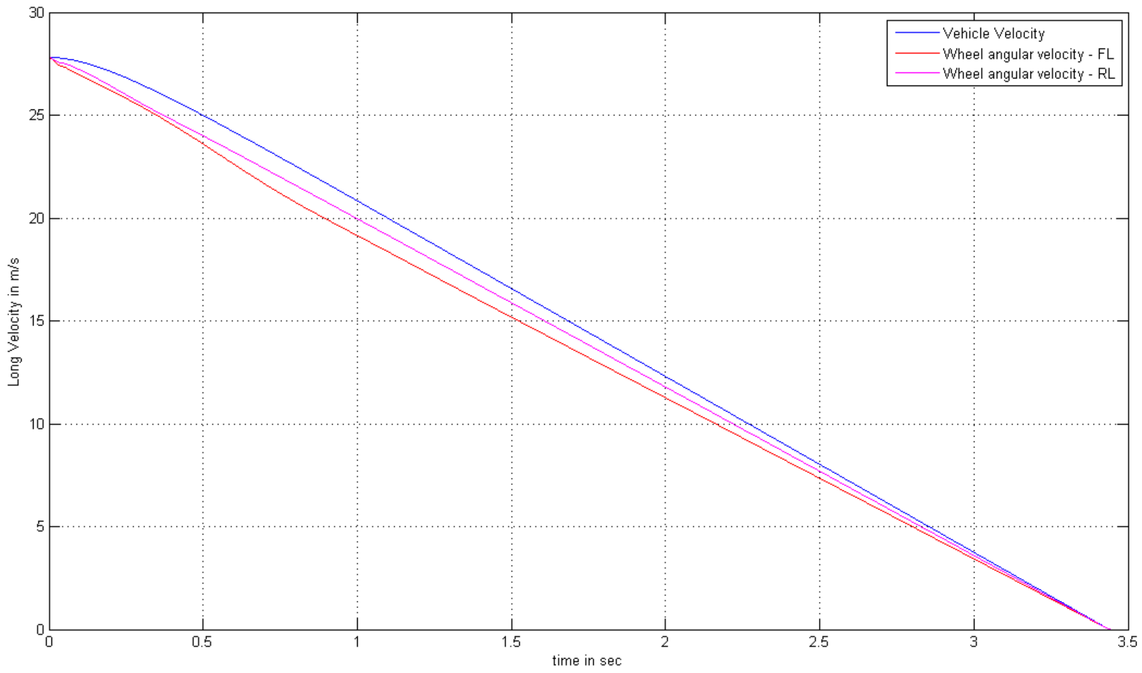

Fig. 4.3 Vehicle and wheel velocities during gradual braking w/o ABS

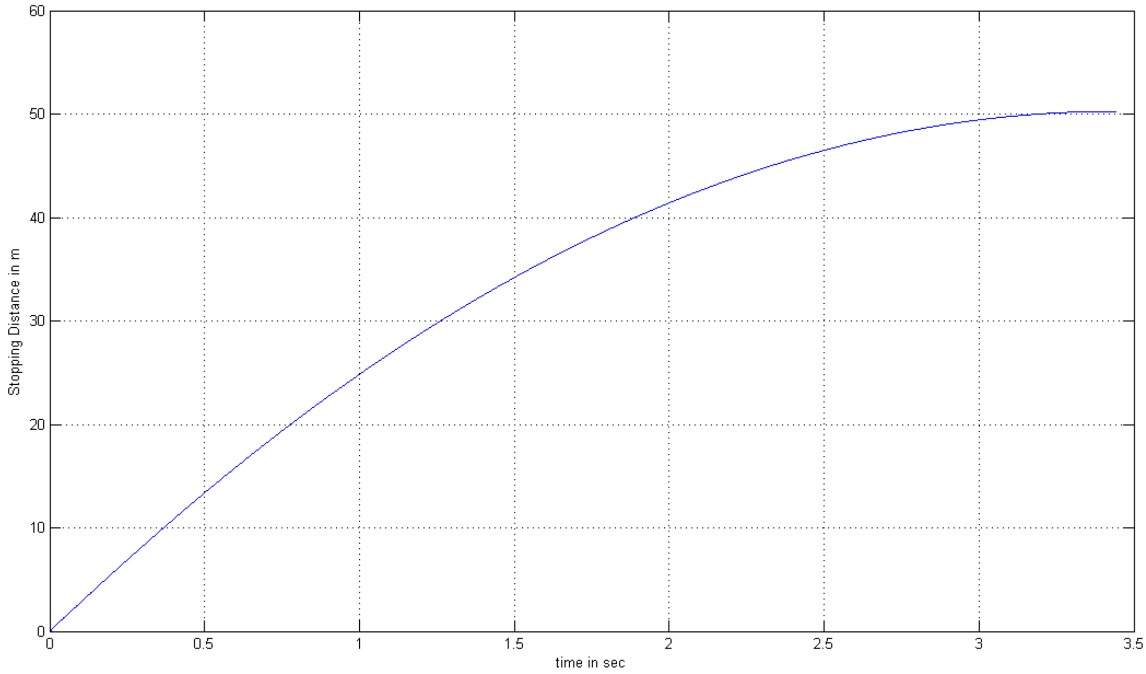

Figure 4.4 Vehicle stopping distance during gradual braking w/o ABS

From figure 4.3 it is clear that the vehicle decelerates gradually during the braking process from a speed of 27.8m/s to a standstill in 3.4 seconds. During this process, the wheel speeds are also observed to follow the vehicle speed closely and come to zero at almost at the same time as the vehicle, which is an indication that the wheels are not locked during braking. Figure 4.4 shows that the vehicle took nearly 50m to come to a complete stop during the gradual application of the brakes. One point to be noted here is that this braking simulation is not the optimised passive braking, so the stopping distance obtained is not an optimised (shortest possible) one either.

Having demonstrated the ability of the model to simulate a physical braking process, a sudden braking input has been given to the vehicle model keeping all other conditions same, and the results are shown in figure 4.5. From the results it can be seen that after a panic/sudden braking input, both the front and rear wheel speeds have dropped sharply to zero (which means the wheels have been locked). This makes the tyre slip ratios to reach 100% in approximately 0.25 and 0.5 seconds for the front and rear wheels respectively. As the wheels are continuing to operate in this condition, they produce less braking force and in turn less longitudinal acceleration. The vehicle is stopped at 49.15m in 3.53 seconds from the application of the brake.

Next, the panic straight-line braking test was repeated with ABS switched ON, which is an active control case. From the results given in figure 4.6, it can be seen that the wheel velocities are closely following the vehicle speed and both the vehicle and the wheels have come to a stop at approximately the same time, with the wheels a little bit in advance. This property is an inherent design implemented in ABS systems by switching OFF the ABS when the vehicle speed reaches below a threshold value to avoid excessive actuation of the ABS system.

It can also be seen that the tyre slips have been maintained at the desired optimum value throughout the braking process. This produces the maximum possible braking force and deceleration without locking the wheel. The vehicle stopping distance is 40.7m in 2.9 seconds.

A typical emergency braking and steering manoeuvre (avoidance manoeuvre) situation is shown in figure 4.7.

The situation is simulated as follows: The vehicle was driven at 100km/h (27.8m/s) in a straight ahead condition, keeping all the parameters the same as above. Then a sudden (panic) braking was applied and held, which locked the wheels. An avoidance steering input manoeuvre was initiated after the wheels were locked. From the results in figure 4.7, it can be seen that the passive vehicle without ABS system has lost the ability to generate the lateral tyre forces effectively and therefore was not able to respond to the driver’s steering input. The lateral displacement response produced by the vehicle is only 0.5m from its straight

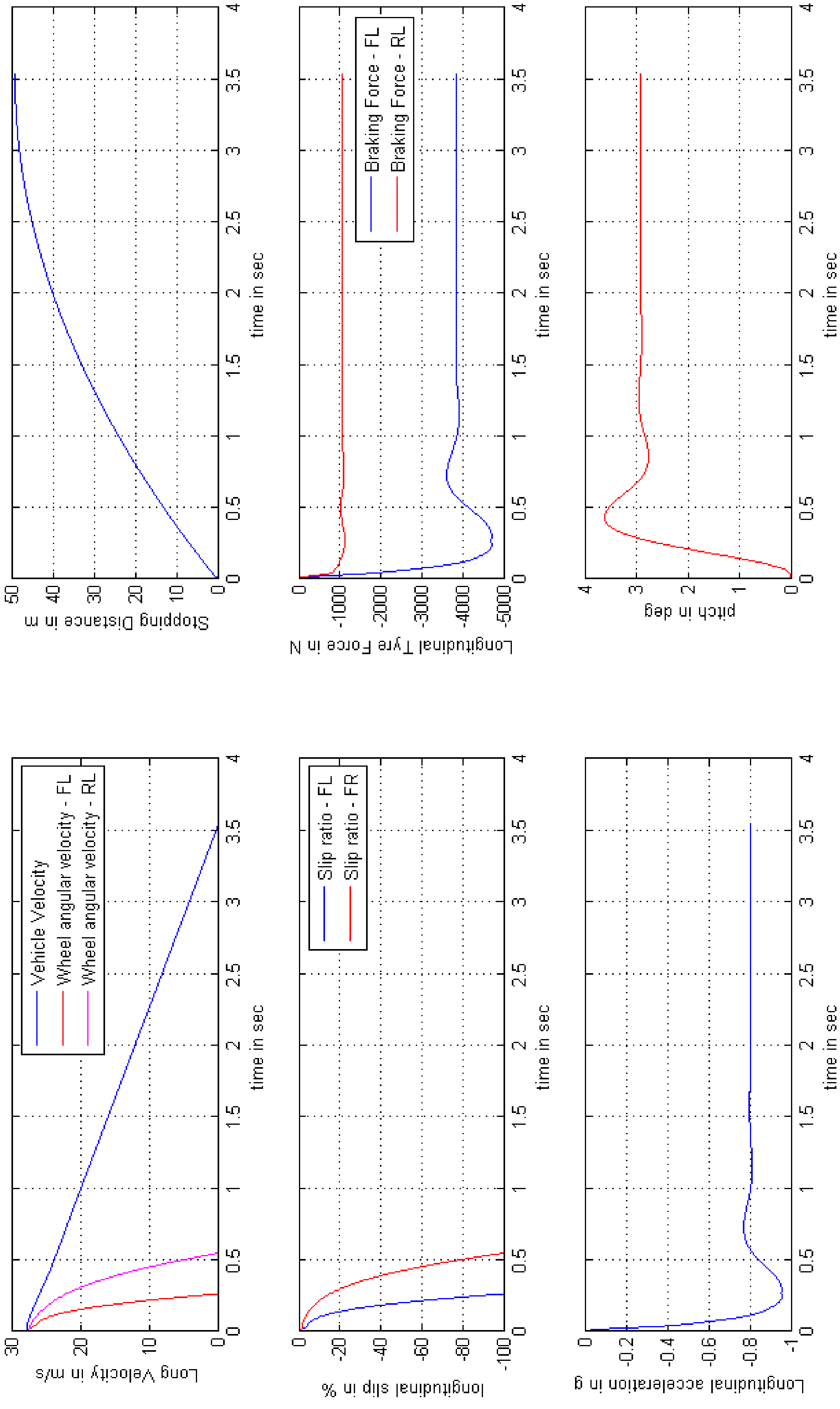

Fig. 4.5 Vehicle braking during panic braking on dry road without ABS

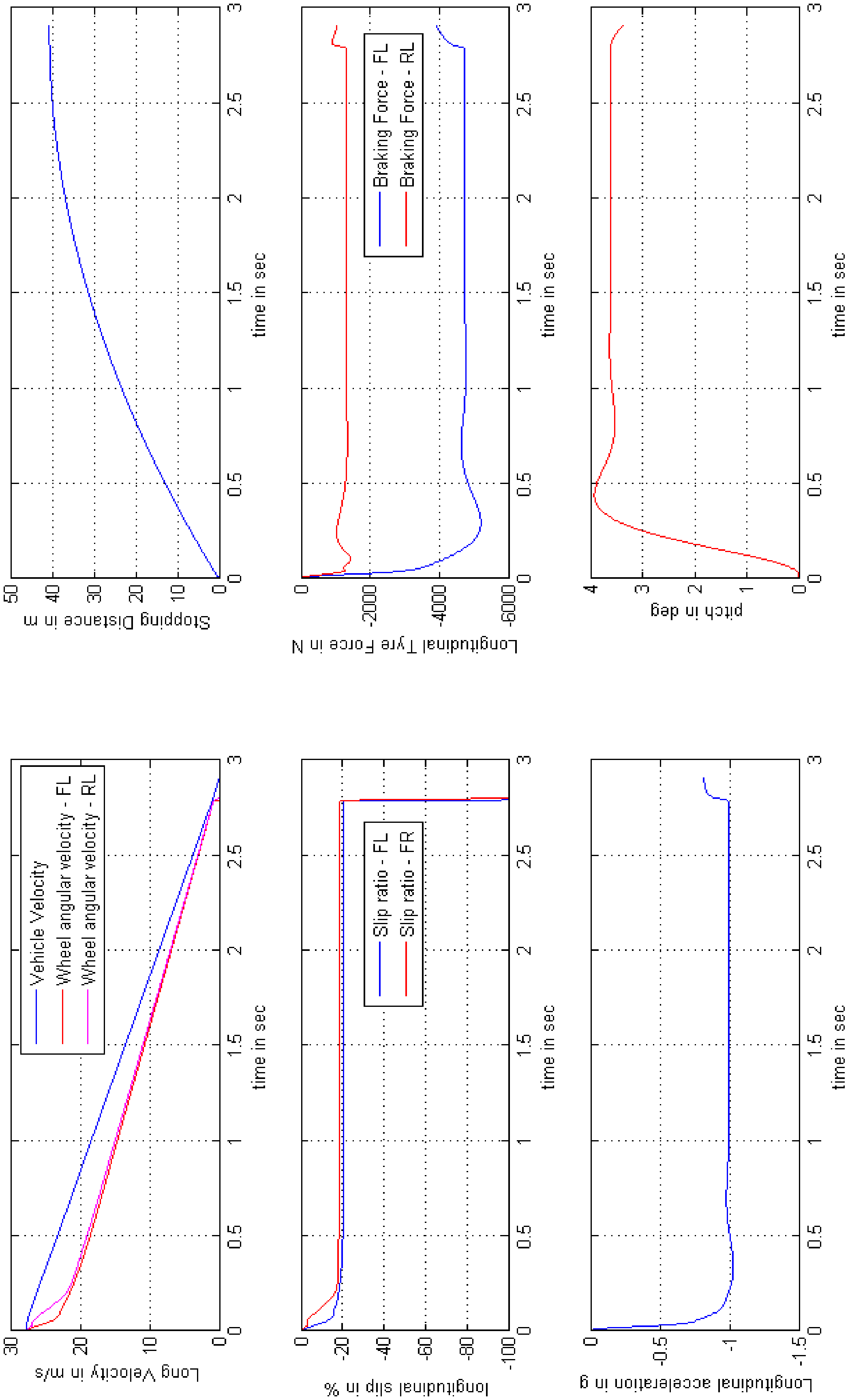

Fig. 4.6 Vehicle braking during panic braking on dry road with ABS

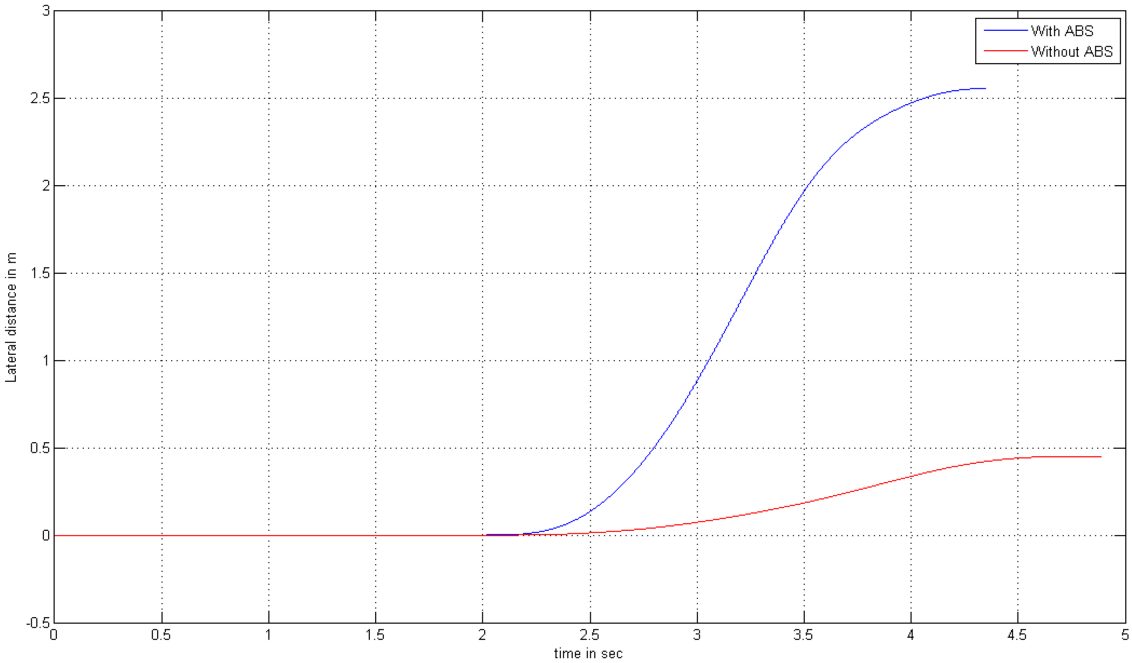

Fig. 4.7 Vehicle steer-ability during a panic braking and avoidance steering manoeuvre with and without ABS

ahead position. That means the passive vehicle has lost its steerability in this critical driving situation, which could easily occur during normal driving.

The same test was repeated with ABS ON, keeping all other parameters the same. It can be seen that the vehicle has responded to the driver's steering input and avoided the obstacle by laterally moving approximately 2.5m.

From the above simulations it can be proved that the active braking system, in this case the ABS, extends the vehicle performance and the operating region by reducing the stopping distance, the stopping time, and by increasing the ability to steer the vehicle in emergency situations such as the one explained above.

4.4 Modelling of Electronic stability Control (ESC)

4.4.1 Modelling of an ESC system

The dynamic model of the brake based Electronic Stability Control (ESC) system is built upon the ABS model developed in the previous section of this chapter. As an ESC system needs to compare the driver's intention with the vehicle's actual behaviour, needs to decide the appropriate action to be taken and implement the decision taken, it requires the necessary sensors, a decision making module, also known as electronic control unit (ECU) and the necessary actuators to implement the decision.

In order to check the driver's intention an ESC system uses a steering angle sensor, which measures the amount of steering input given by the driver. A yaw rate sensor is used to measure the yaw velocity of the vehicle. Another state variable to be measured is the side-slip angle. Owing to the complexity in measuring the side-slip angle, this parameter is generally estimated while designing the vehicle dynamic control systems. Based on the input signals from these sensors the desired vehicle behaviour is obtained and compared against the actual behaviour. The difference between the two, known as the error, is sent to the ECU, where an appropriate controlling decision is made. Based on the decision made at the ECU a signal to actuate the appropriate brake is sent to the brake actuator. In the case of an excessive or sudden command of this brake torque, the wheels would be locked. To prevent that, the wheel speeds need to be monitored continuously and the applied brake pressures need to be modulated. This is done by the existing wheel speed sensors and the ECU of the ABS system.

4.4.2 Development of ESC Controller

The ESC controller used in this thesis is developed based on the model reference control technique where the desired vehicle states are generated from a linear 2 DoF reference vehicle model. As a function of the vehicle parameters, vehicle longitudinal speed and the steering input, the reference model generates the desired vehicle state trajectories to be tracked by the actual vehicle. The desired yaw rate can be expressed as shown in the following equation:

$$\dot{\psi}_{ref} = \frac{V_x}{L\left(1+\frac{V_x^2}{V_{ch}^2}\right)} \times \delta_f \tag{4.6}$$

Where,

$\dot{\psi}_{ref}$ = the reference yaw rate in rad/s

V_{ch} = the characteristic speed m/s

L = wheel base in m

δ_f = the front wheel steering angle in radian

V_x = the longitudinal speed in m/s

The characteristic speed V_{ch} in the previous equation can be calculated as follows:

$$V_{ch} = \sqrt{\frac{gL}{K_{us}}} \tag{4.7}$$

Where,K_{us} is called the under steering gradient which is a function of the vehicle parameters.

$$K_{us} = \frac{mg}{L}\left(\frac{l_f}{C_r} - \frac{l_r}{C_f}\right) \tag{4.8}$$

Where C_f and C_r are the tyre cornering stiffness of the linear tyre model.

The calculated desired yaw rate from the above equation is valid only on dry roads with high surface coefficient of friction. The maximum desired yaw rate developed is limited by the surface coefficient of friction. As the surface coefficient decreases the desired yaw rate also decreases.

The lateral acceleration is a function yaw rate and the vehicle longitudinal velocity,

$$a_y = \dot{\psi} \times V_x \tag{4.9}$$

Since the maximum lateral acceleration developed by a vehicle cannot exceed the surface coefficient of friction,

$$|a_y| \leq \mu g \tag{4.10}$$

Taking this into consideration extends the validity of the desired yaw rate calculation. So the maximum desired yaw rate is limited by the following condition:

$$\dot{\psi}_{limit} \leq \mp \frac{\mu g}{V_x} \tag{4.11}$$

The same logic is implemented in the desired yaw rate calculation block as follows:

$$\dot{\psi}_{des} = \begin{cases} \dot{\psi}_{des} \ , & if \ |\dot{\psi}_{des}| < \frac{\mu g}{V_x} \\ \frac{\mu g}{V_x} \ sign(\dot{\psi}_{des}) \ , & if \ |\dot{\psi}_{des}| \geq \frac{\mu g}{V_x} \end{cases} \tag{4.12}$$

The calculation of desired side-slip (β_{des}) angle is made simpler by assuming it to be zero, i.e. $\beta_{des} = 0$, as driving the vehicle side-slip angle to the minimum increases the vehicle stability.

Next the calculation of the actual vehicle states is carried out. In order to do that, the same steering input (δ_f) used to calculate the desired state values is also given to the nonlinear vehicle model which generates the actual vehicle states. Then the desired and the actual values of yaw rate and side-slip angle are compared and the errors are used to generate the desired corrective yaw moment.

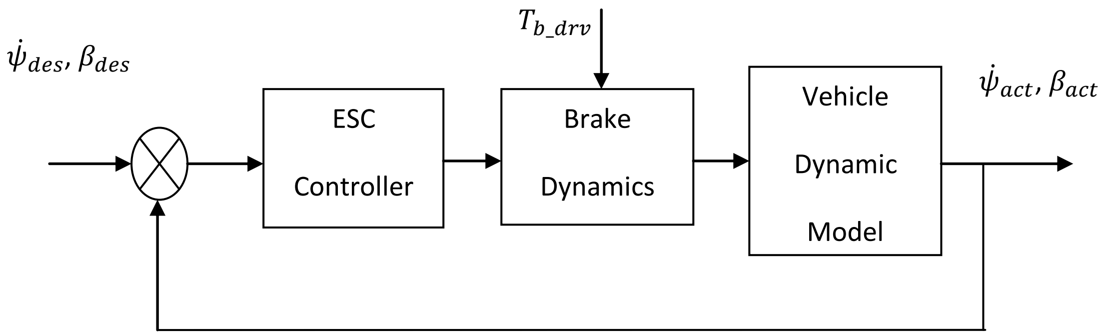

Fig. 4.8 The schematic of the ESC controller

The ESC controller used in this thesis has a three layer hierarchical architecture. The upper layer determines the desired corrective yaw moment, the middle layer calculates the required brake pressure to develop the corrective yaw moment and finally, the lower layer allocates the desired yaw moment to the appropriate wheel to improve the stability of the vehicle.

For the same reasons explained during the development of the ABS controller, such as the simplicity, its ability to robustly control the non-linear systems fuzzy logic control strategy is used to calculate the desired corrective yaw moment from the yaw rate and side-slip angle errors.

The fuzzy ESC controller has two inputs, the yaw rate error and side slip angle error and one output the normalised desired corrective yaw moment. This fuzzy controller has an output scaling block which converts ESC controller output to the desired corrective yaw moment.

Then the longitudinal brake force required to develop the desired corrective yaw moment is calculated from the kinematics of the brake-tyre force transmission system.

$$F_{x\_ij} = \frac{2\,M_{zc}}{t_i} \tag{4.13}$$

where, t_i is the track width of the vehicle in m.

M_{zc} , is the desired corrective yaw moment in Nm.

i and j stand for {front , rear} and {left, right} respectively.

Then the brake pressure required to generate this brake force is calculated as a function of the brake system parameter.

$$P_{b\_ij} = \frac{F_{x_{ij}} \times r_w}{K_b} \tag{4.14}$$

where, r_w is the radius of the wheel.

K_b is the brake gain pf the brake system in Nm/MPa

P_{b_ij} is the brake pressure in Mpa

Finally, the allocation of this desired brake pressure on a particular wheel is determined at the lower layer of the ESC controller. This control pressure allocation strategy is based on the direction of steering input (left or right) and the sign of the yaw rate error (under-steer or over-steer). This is explained in table 4.2.

The research by Ken Koibuchi et al, (1996), Kilong Park and Seung-Jin Heo, (2003), H.T. Smakman, (2000), on brake pressure allocation show the effectiveness of individual wheels generating the corrective yaw moment for the ESC system. The effectiveness of the front outer wheel and rear inner wheel in developing the pro and contra cornering moment is analysed. In order to simplify the concept the one wheel control strategy is followed in this thesis. The front outer wheel and front inner wheel are used for brake intervention.

$e_{\dot{\psi}}$ > 0	$\delta_f > 0$	OS	FLW
	$\delta_f < 0$	US	FRW
$e_{\dot{\psi}}$ < 0	$\delta_f > 0$	OS	FRW
	$\delta_f > 0$	US	FLW
$e_{\dot{\psi}}$ = 0	*for all* δ_f	-	HOLD

Table 4.2 Control allocation of braking force on individual wheels using ESC

If the pressure demanded by the ESC system produces a brake torque that drives the wheels to lock (wheel speeds to zero), then the ABS system intervenes and releases the excess pressure from individual wheels.

The sum of the pressures demanded by the ESC system, by the ABS system (if activated) and any pressure demand from the driver (if the brake pedal is pressed) is supplied at the wheels to produce the differential brake forces which generates the desired corrective yaw moment.

The schematic of the summation of three pressures is given in figure 4.9.

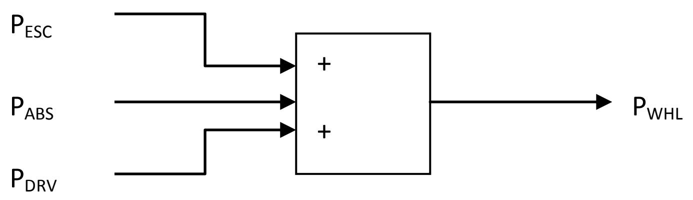

Fig. 4.9 Schematic of the summation of brake wheel cylinder pressure

As the main objective of the ESC system is to minimise the yaw rate and side-slip angle errors, to obtain the desired vehicle response, the fuzzy logic controller requires two input values:

$$\text{INPUT 1: } e_{\dot{\psi}} = \psi_{des} - \psi_{act} \tag{4.15}$$

$$\text{INPUT 2: } e_{\beta} = \beta_{des} - \beta_{act} \tag{4.16}$$

As the purpose of this layer of the controller is to calculate the desired corrective yaw moment, the same has been designed as the output.

$$\text{OUTPUT 1: } M_{zc} \tag{4.17}$$

The architecture of the fuzzy logic controller has four steps as described below:

Fuzzification: makes the controller inputs compatible with the linguistic variables shown in table 4.3

	Linguistic variables
NB	Negative Big
NM	Negative Medium
NS	Negative Small
ZE	Zero
PS	Positive Small
PM	Positive Medium
PB	Positive Big

Table 4.3 Linguistic variables used in ESC Fuzzy logic controller

Five fuzzy sets are used for both the inputs and seven fuzzy sets are used for the output. : $e_{\dot{\psi}}$ and e_{β} have a set of values between NB and PB which is defined as follows:

$$\{ e_{\dot{\psi}} , e_{\beta} \} = \{NB, NS, ZE, PS, PB\}$$

And $\widehat{M}_Z$ has a set of values between NB and PB which is defined as follows

$$\{\widehat{M}_Z \} = \{NB, NM, NS, ZE, PS, PM, PB\}$$

Fuzzy decision Process: processes a list of rules from the knowledge base using fuzzy input from the previous step to produce the fuzzy output. Table 4.4 show the fuzzy rules used in the controller.

$\hat{M}_{zc}$		e_{β}				
$e_{\dot{\psi}}$		NB	NS	ZE	PS	PB
	PB	NB	NB	NM	NB	NB
	PS	NB	NM	NS	NM	NS
	ZE	NS	NS	ZE	PS	PS
	NS	PB	PM	PS	PM	PS
	NB	PB	PB	PM	PB	PB

Table 4.4 Fuzzy rules table for the ESC controller

The fuzzy controller uses the Mamdani Fuzzy Inference System (FIS), which is characterised by the following rule:

IF e_{β} is A and $e_{\dot{\psi}}$ is B THEN $\hat{M}_{zc}$ is C

Defuzzification: Scales and maps the fuzzy output from fuzzy decision process to produce an output value which is the input value to the system being controlled, in our case, the corrective yaw moment. The defuzzification method used here is the centre of area. The universe of discourse of the inputs is selected considering the range of yaw rate and side-slip angle without controller. The universe of discourse of the output is normalised [-1, 1].

Output scaling: The controller output $\hat{M}_{zc}$ is scaled to map the yaw moment from the normalised interval.

$$M_{zc} = K_{fuz\_esc} \times \hat{M}_{zc} \quad (4.18)$$

K_{fuz_esc} = output scaling factor for ESC fuzzy controller

The scaling factor is tuned through multiple simulations.

4.4.3 Simulations

The performance of an ESC system can generally be evaluated by the following three standard test manoeuvres – ISO 3888 Double Lane Change (DLC), Single Lane Change and Federal Motor Vehicle Safety Standard (FMVSS) 126.

Due to its growing popularity and its mandatory nature to validate ESC systems, FMVSS 126 test method is used to prove that the active vehicle with ESC ON extends the range of vehicle handling limit compared to a passive vehicle.

As per the FMVSS 126, the steering angle (amplitude is equal to A_{swd}) required to produce a lateral acceleration of 0.3g is determined first. The test speed is set as 80 km/h (22.22m/s) and the road surface is assumed to be dry (μ = 0.85). Starting from the steer angle with amplitude of 1.5*A_{swd} and a frequency of 0.7 Hz, a Sine With Dwell (SWD) steer input is given to the vehicle. A typical SWD input is shown in figure 4.10. The vehicle stability indicators such as yaw rate and side-slip angle are then measured. The simulation is repeated for various steer angles increased in a step of 0.5*A_{swd} until the passive vehicle becomes unstable.

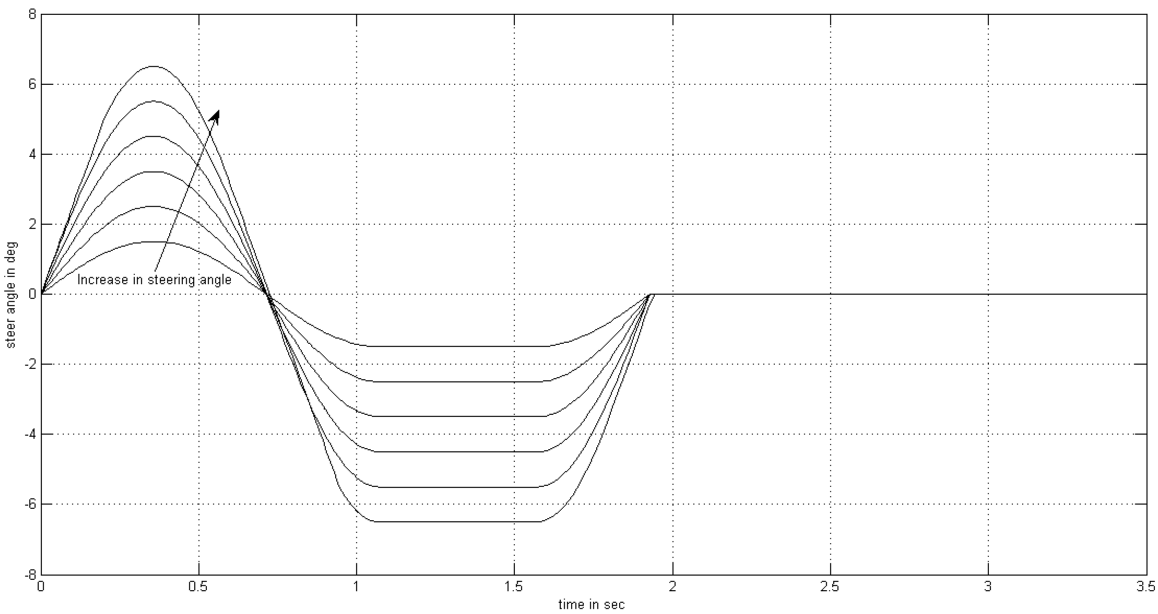

Fig. 4.10: Sine with Dwell steer angle input for FMVSS 126 test

In figure 4.11 and 4.12, the yaw-rate and the side-slip angle of the passive vehicle can be seen spiralling out of bound and indicate the unstable condition of the vehicle as the vehicle is pushed to the limit lateral acceleration (figure 4.13) by increasing steer angle.

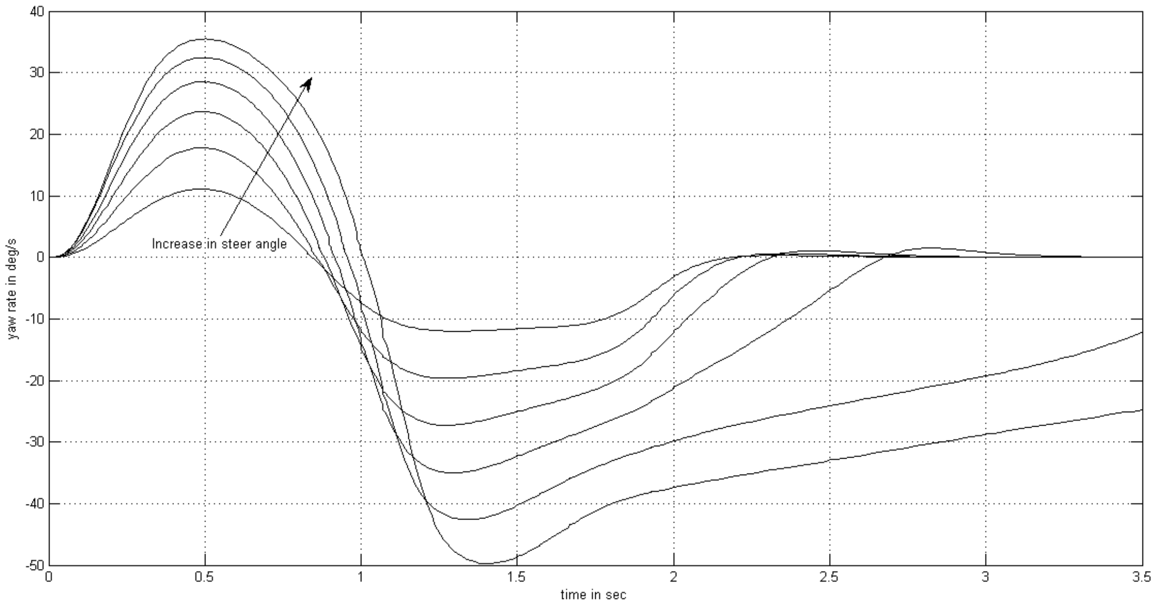

Fig. 4.11: Yaw rate response of the passive vehicle in the FMVSS 126 test

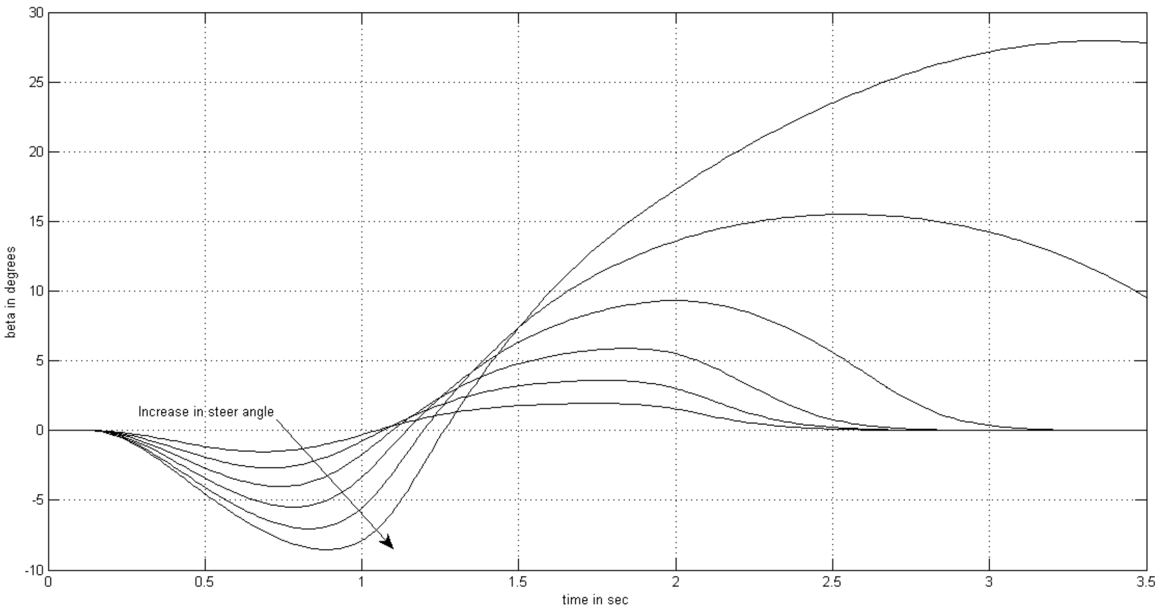

Fig. 4.12: Side-slip angle response of the passive vehicle in the FMVSS 126

It can also be observed that the yaw rate is not following the lateral acceleration and lateral acceleration of the vehicle is saturated towards the limits due to the saturation of tyre force.

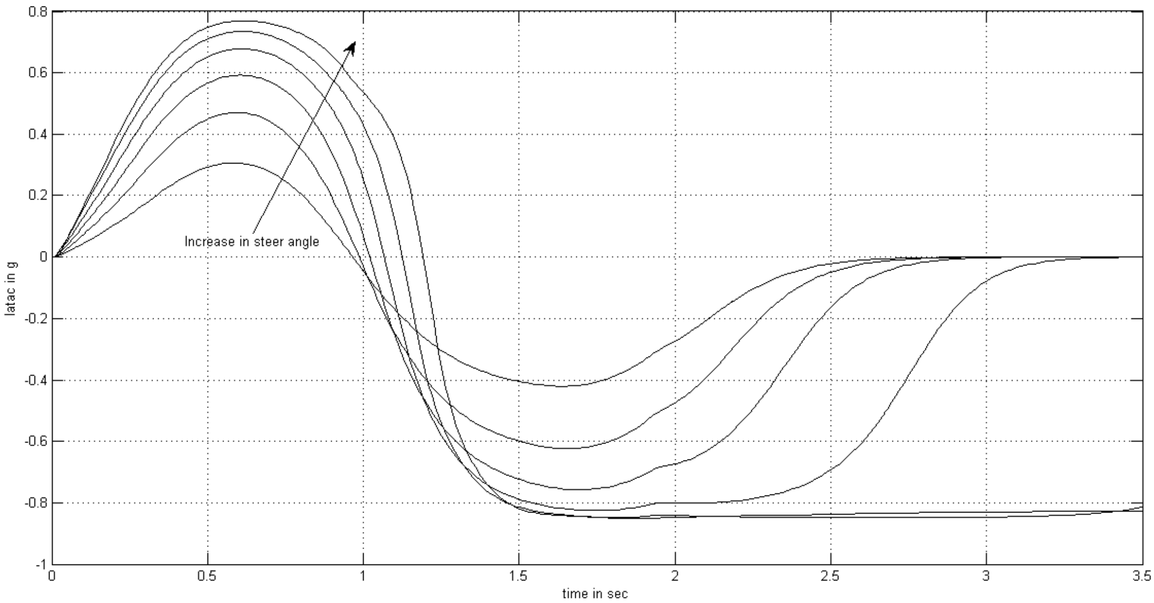

Fig. 4.13: 'Latac' response of the passive vehicle in the FMVSS 126 test

Then the ESC system is switched ON and the test is repeated keeping all other parameters the same. The figure 4.14 and figure 4.15 show that both the yaw rate and the sideslip angle of the vehicle with ESC ON is less than the passive vehicle. This improvement in limit handling is obtained through the differential braking forces applied by the ESC system.

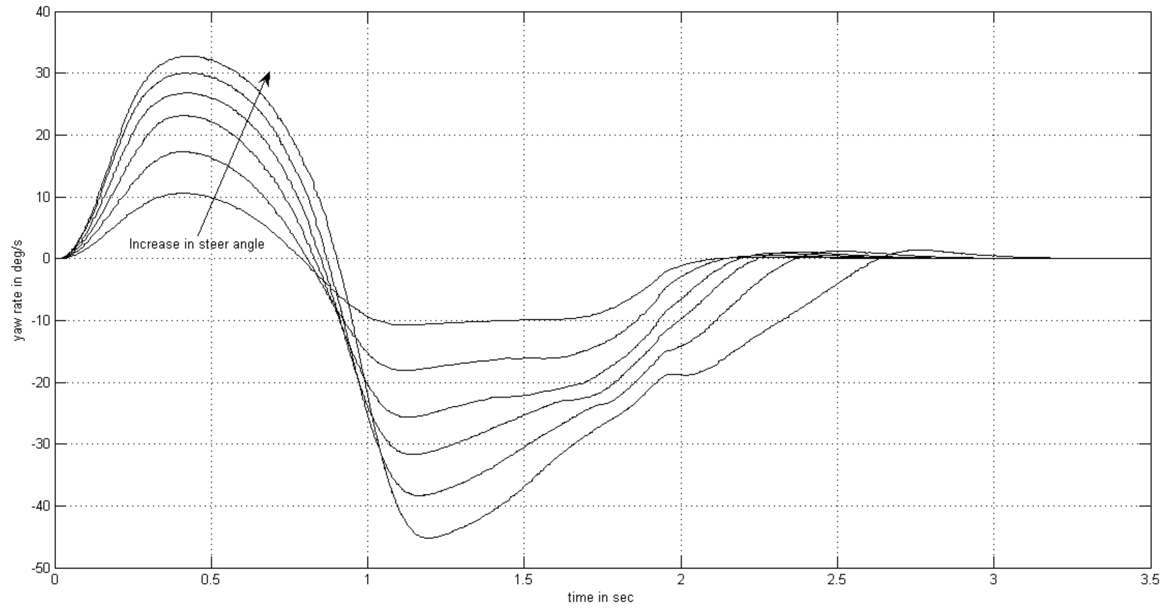

Fig. 4.14: Yaw rate response of the vehicle with ESC in the FMVSS 126 test

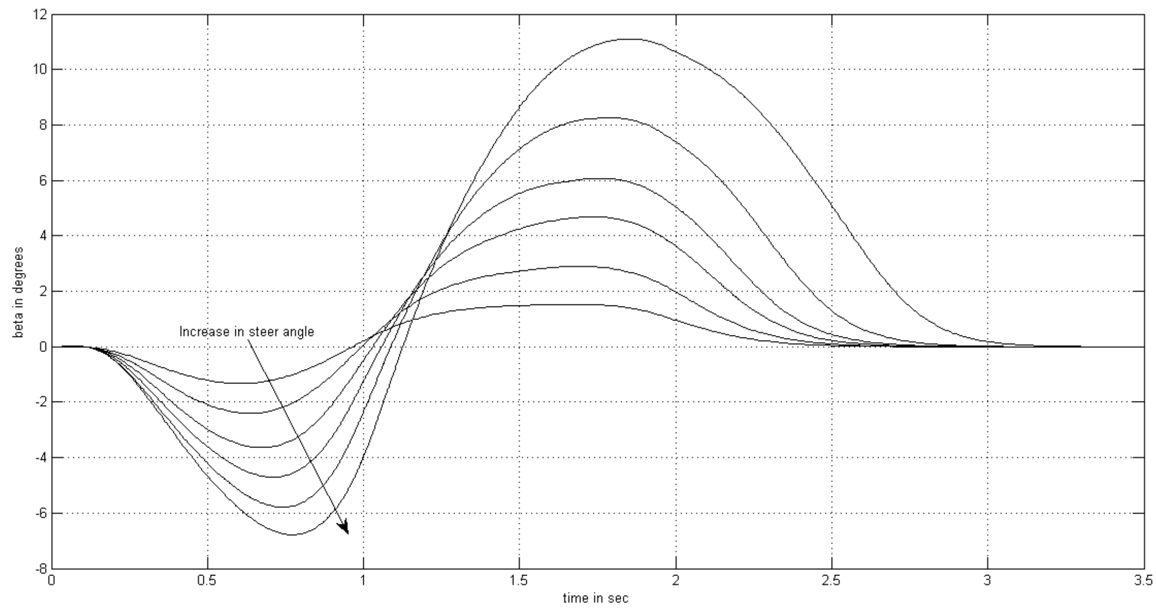

Fig. 4.15: Side-slip angle response of the vehicle with ESC in the FMVSS 126

4.5 Modelling of Active Front Steering (AFS)

4.5.1 Mathematical Modelling of steering dynamics

In this section, the dynamics of a hydraulic power steering system is developed to provide the necessary steering assistance to the driver of modern cars.

The simple steering system modelled in this research is a hydraulic power steering based on a model developed by C.Messener, (2006). The main components of the systems modelled are the steering column, the torsion valve, the hydraulic cylinder, the rack and pinion gearbox. The steering column is modelled as two parts, upper and lower columns, which are connected by the torsion valve. The steering column is assumed to be rigid and the torsional valve is a modelled as a torsional spring with constant spring stiffness. At the end of the lower steering column is a steering pinion gear that is engaged with a steering rack. The input to the steering system model is the angle of the steering wheel, also known as the hand wheel, while the output is the position of the steering rack, which determines the angle of the front wheels. The rack is mechanically connected with a steering pinion gear, which converts the rotational motion of the steering column to translational motion of the rack to turn the wheels. The rack is subjected to three forces, pinion-rack contact force, internal frictional force and the hydraulic force. The rack linear velocity can be obtained for a given steering wheel input by solving these three forces:

$$\dot{x}_{rack} = \frac{1}{m_{rack}} \int F_{contact} - F_{friction} + F_{hydraulic} \qquad (4.18)$$

Where

$\dot{x}_{rack}$ = the steering rack velocity in m/s

m_{rack} = the mass of the steering rack

The rack-pinion contact force can be calculated as:

$$F_{contact} = \frac{k_{stg}(\delta_{sw} - \delta_{pinion})}{r_{sc}} \tag{4.19}$$

Where,

k_{stg} = the torsional valve stiffness in Nm

δ_{sw} = the steering wheel angle in radians

δ_{pinion} = the pinion rotational angle in radians

r_{sc} = the radius of steering column in m

$$F_{friction} = b_{stg}\dot{x}_{rack} \tag{4.20}$$

Where

b_{stg} = the steering system friction

$$F_{hydraulic} = P\left(\delta_{sw} - \delta_{pinion}\right)A_{stg\_hc} \tag{4.21}$$

Where,

P = the hydraulic pressure as a function of the pinion-steering column angular difference.

A_{stg_hc} = The area of the steering hydraulic cylinder.

The power assistance is provided by a hydraulic piston attached to the rack. The torsion valve determines the direction of flow of the pressurised hydraulic fluid. The difference between the angular position of the steering wheel and the angular position of the pinion determines the fractional opening of the torsion valve. The power assistance continues until the difference between the steering wheel position and pinion position is approximately zero.

4.5.2 Development of AFS controller

The AFS controller used in this thesis is a yaw rate error and side slip angle error based fuzzy logic steering controller. A fuzzy logic control strategy is used for the same reasons as mentioned earlier in this chapter. The aim of the AFS controller is to minimise the yaw rate and side slip error by modulating the front wheel steer angle, using model reference control technique. The AFS controller receives two inputs the yaw rate and side slip angle errors and provides one output the normalised corrective steering angle. Then an output scaling operation is carried out to convert the normalised steering angle ($\hat{\delta}_C$) to the required corrective steering angle (δ_c).

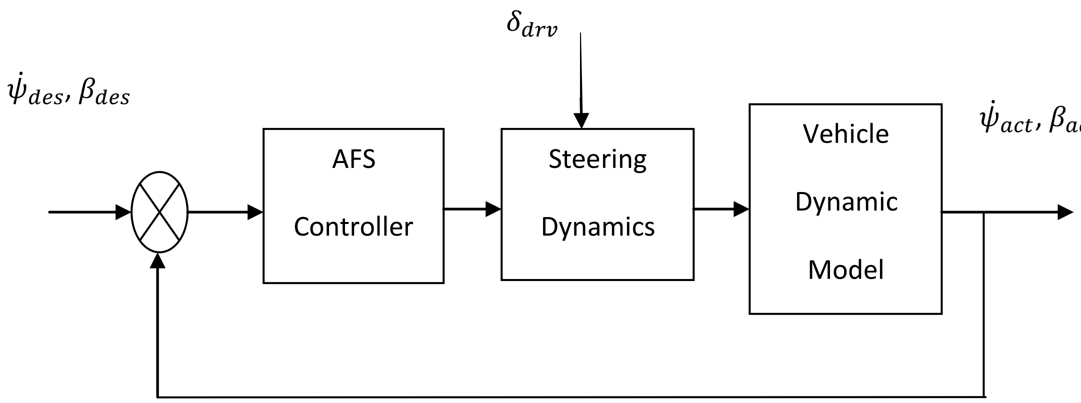

Fig. 4.16 Schematic of the Active Front steering (AFS)

Fuzzy Input / Output Selection: As the main objective of the AFS system is to minimise the yaw rate and side-slip angle errors, to obtain the desired vehicle response, the fuzzy logic controller requires two input values:

$$\text{INPUT 1: } e_{\dot{\psi}} = \psi_{des} - \psi_{act} \tag{4.22}$$

$$\text{INPUT 2: } e_\beta = \beta_{des} - \beta_{act} \tag{4.23}$$

As the purpose of this layer of the controller is to calculate the corrective steer angle, the same has been designed as the output.

$$\text{OUTPUT 1: } \delta_c$$

The architecture of the fuzzy logic controller has four steps shown in figure 4.16

Fuzzification: makes the controller inputs compatible with the linguistic variables shown in table 4.5

	Linguistic variables
NB	Negative Big
NM	Negative Medium
NS	Negative Small
ZE	Zero
PS	Positive Small
PM	Positive Medium
PB	Positive Big

Table 4.5: Table of linguistic variables for the fuzzy AFS controller

Five fuzzy sets are used for both the inputs and seven fuzzy sets are used for the output. : $e_{\dot{\psi}}$ and e_β have a set of values between NB and PB which is defined as follows:

$$\{ e_{\dot{\psi}} , e_\beta \} = \{\text{NB, NS, ZE, PS, PB}\}$$

And $\hat{\delta}_C$ has a set of values between NB and PB which is defined as follows

$$\{\hat{\delta}_C\} = \{\text{NB, NM, NS, ZE, PS, PM, PB}\}$$

Fuzzy decision Process: processes a list of rules from the knowledge base using fuzzy input from the previous step to produce the fuzzy output. Table 4.6 show the fuzzy rules used in the controller.

$\hat{\boldsymbol{\delta}}_{\boldsymbol{C}}$		$\boldsymbol{e_\beta}$				
$\boldsymbol{e_{\dot{\psi}}}$		NB	NS	ZE	PS	PB
	PB	NB	NB	NM	NB	NB
	PS	NB	NM	NS	NM	NS
	ZE	NS	NS	ZE	PS	PS
	NS	PB	PM	PS	PM	PS
	NB	PB	PB	PM	PB	PB

Table 4.6 Fuzzy Rule for the AFS Controller

The fuzzy controller uses the Mamdani Fuzzy Inference System (FIS), which is characterised by the following rule:

IF e_β is A and $e_{\dot{\psi}}$ is B THEN $\hat{\delta}_C$ is C

Defuzzification: Scales and maps the fuzzy output from fuzzy decision process to produce an output value which is the input value to the system being controlled, in our case, the corrective yaw moment. The defuzzification method used here is the centre of area. The universe of discourse of the inputs is selected considering the range of yaw rate and side-slip angle errors without controller. The universe of discourse of the output is normalised to [-1, 1].

Output scaling: The controller output $\hat{\delta}_C$ is scaled to map the corrective steer angle from the normalised interval.

$$\delta_c = K_{fuz\_afs} \cdot \hat{\delta}_C \tag{4.24}$$

K_{fuz_afs} = output scaling factor for AFS fuzzy controller

From the research literature (Junje et al, 2005) the upper and lower bound of the corrective steering angle is limited to ±3°. This active steering actuator saturation is implemented through the output scaling stage of the controller development.

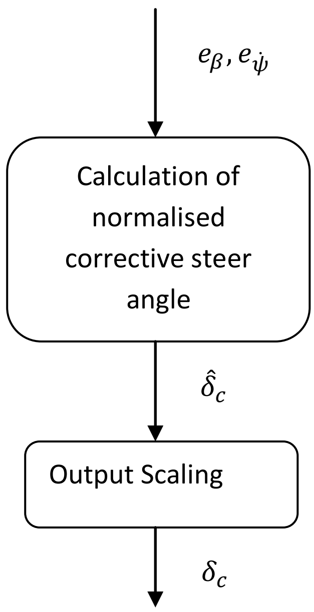

Fig 4.17 AFS Control Architecture

4.5.3 Simulations

To compare the passive and active vehicle (AFS) performance a single lane change manoeuvre is performed. The vehicle is driven at a speed of 80kmph (22.22 m/s) on a dry road with $\mu = 0.85$. Then a single sine steering input of 57° with a frequency of 0.5 Hz is given. The steering ratio obtained through the steering system is 19:1. This produces a sine steer angle equivalent of 3° amplitude at the wheels. The sample single lane change sine steer input is shown in figure 4.18.

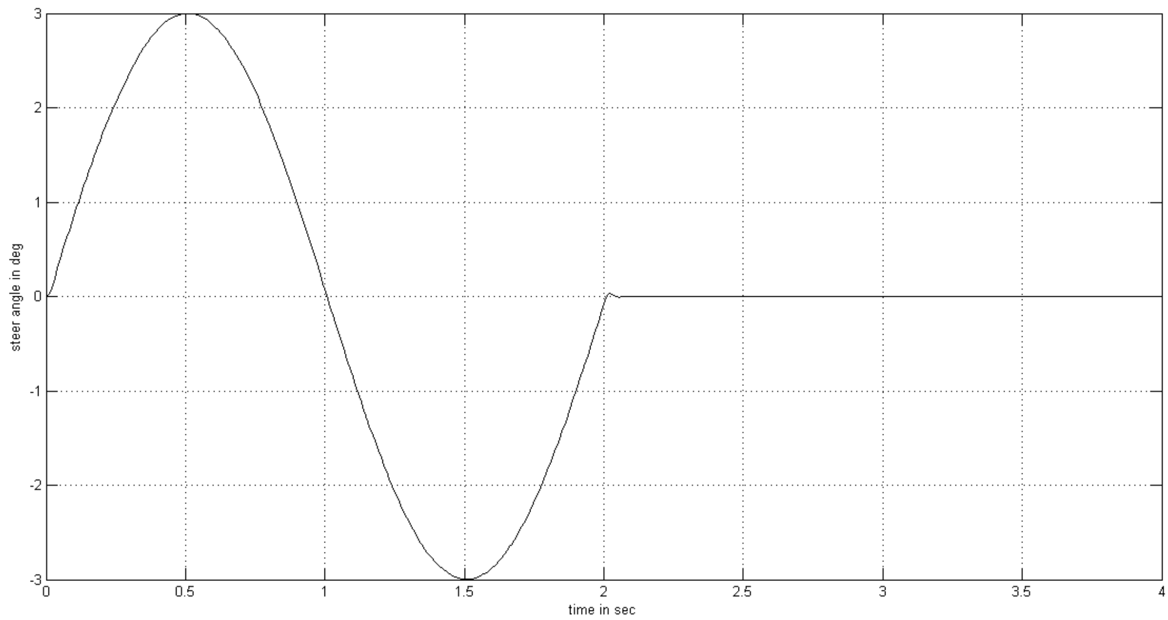

Fig 4.18 Steer angle input for the Single Lane Change (SLC) Manoeuvre

The yaw rate and side slip angle output of the nonlinear vehicle model is recorded.

Then the AFS controller is switched ON and the test is repeated for the same test conditions. From figure 4.19 the lane change path followed by both the controlled and uncontrolled vehicle can be seen.

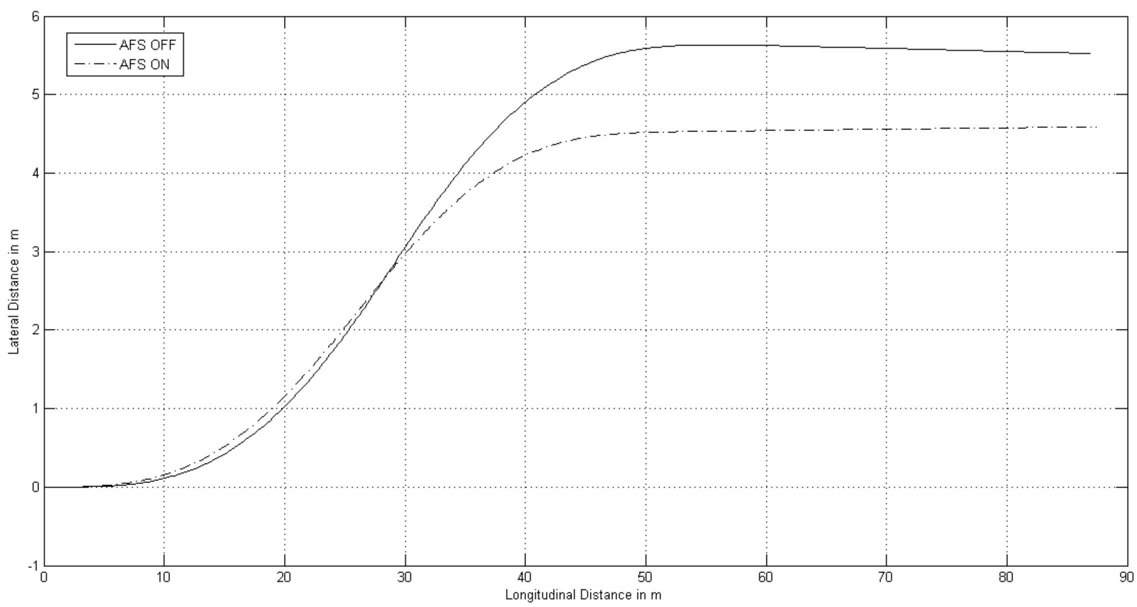

Fig 4.19 Lateral Path Deviation in the SLC with and without AFS on high μ

The following figures 4.20 and 4.21 show that the yaw rate and the side slip angle of the controlled vehicle is less than the uncontrolled vehicle during the single lance change manoeuvre.

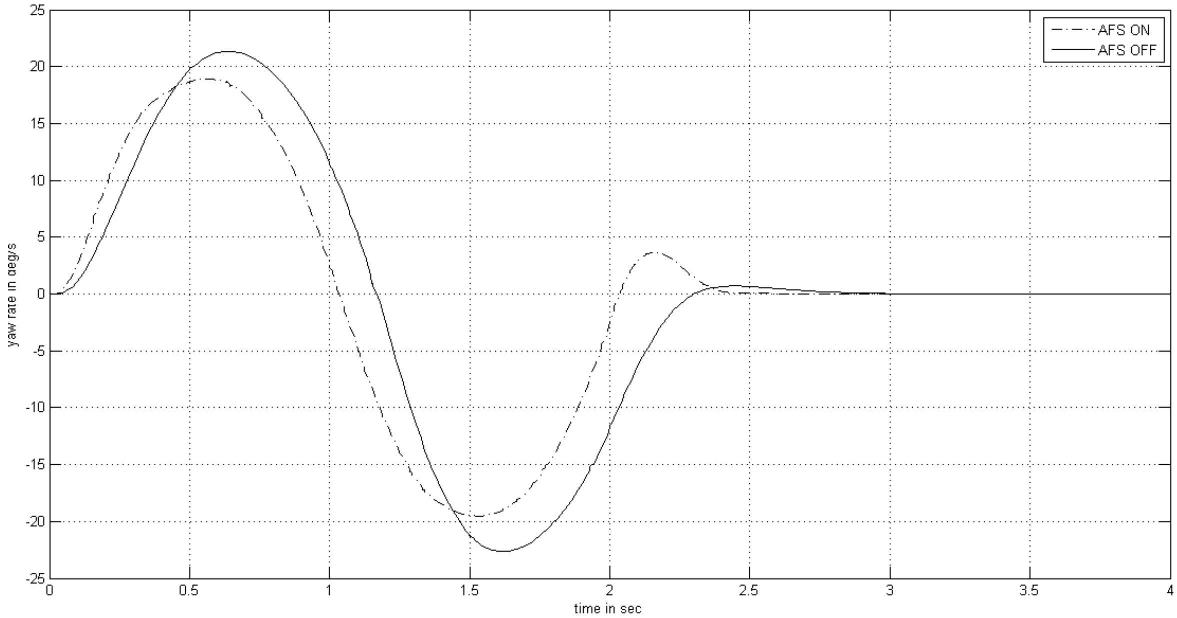

Fig 4.20: Yaw rate response during the SLC with and without AFS on high μ

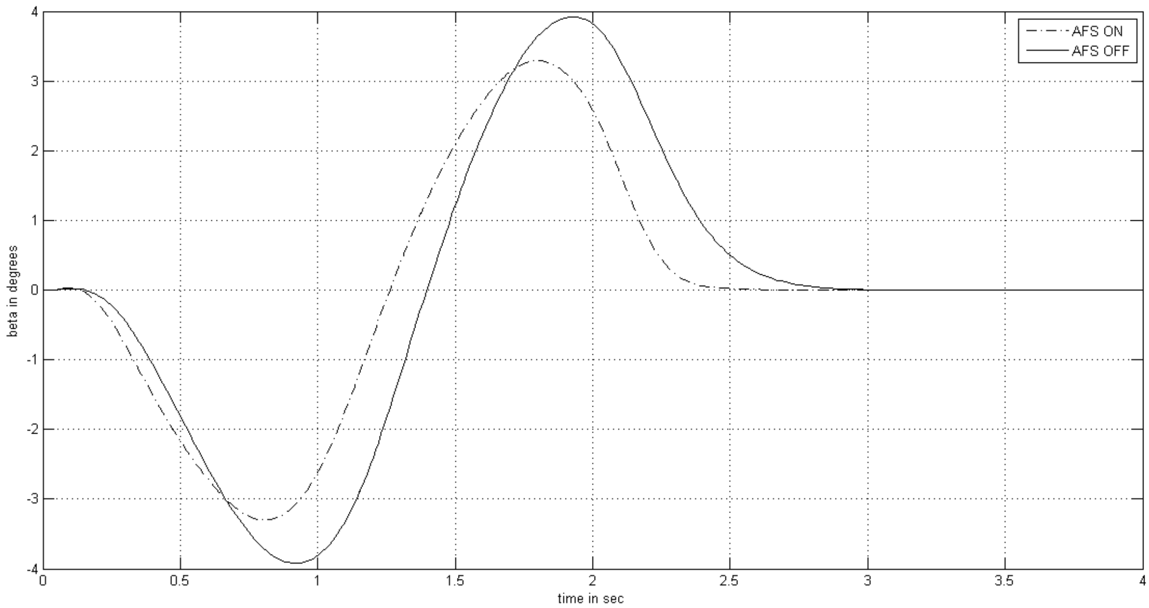

Fig 4.21: Side-slip angle during the SLC with and without AFS on high μ

To check the robustness of the developed fuzzy controller the test is repeated on a low frictional icy surface with a surface coefficient of friction $\mu = 0.3$. Here it can be seen that the passive vehicle is unable to perform the lane change and breaks away from the intended path whereas the active vehicle is maintain its trajectory in a controlled manner.

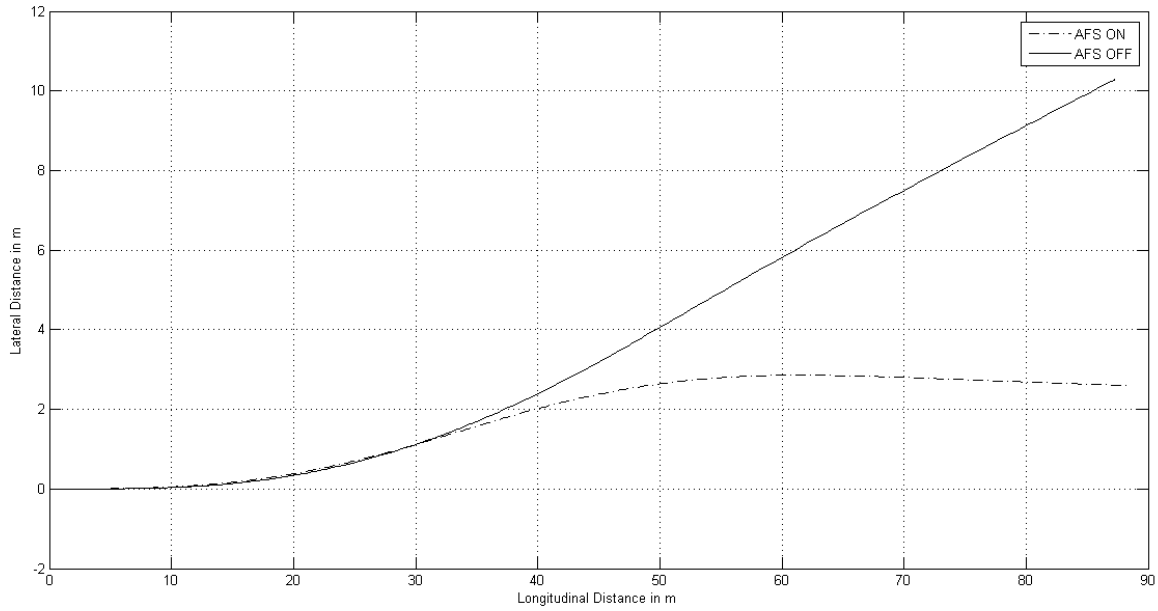

Fig 4.22: Lateral Path Deviation in the SLC with and without AFS on low μ

Moreover from the following figures 4.22 and 4.23 it can be observed that the yaw rate and the side slip angle are less than the passive vehicle and follows the driver steering input, indicating the vehicle with active front steering increases the stability and extending the operating range compare to its passive counterpart.

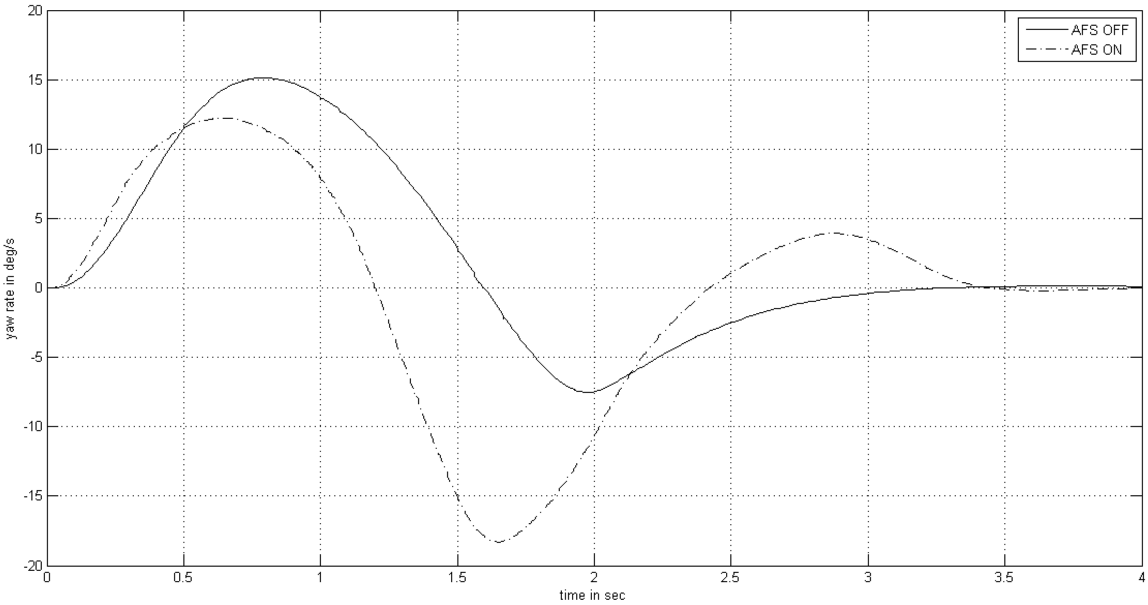

Fig 4.23: Yaw rate response during the SLC with and without AFS on low μ

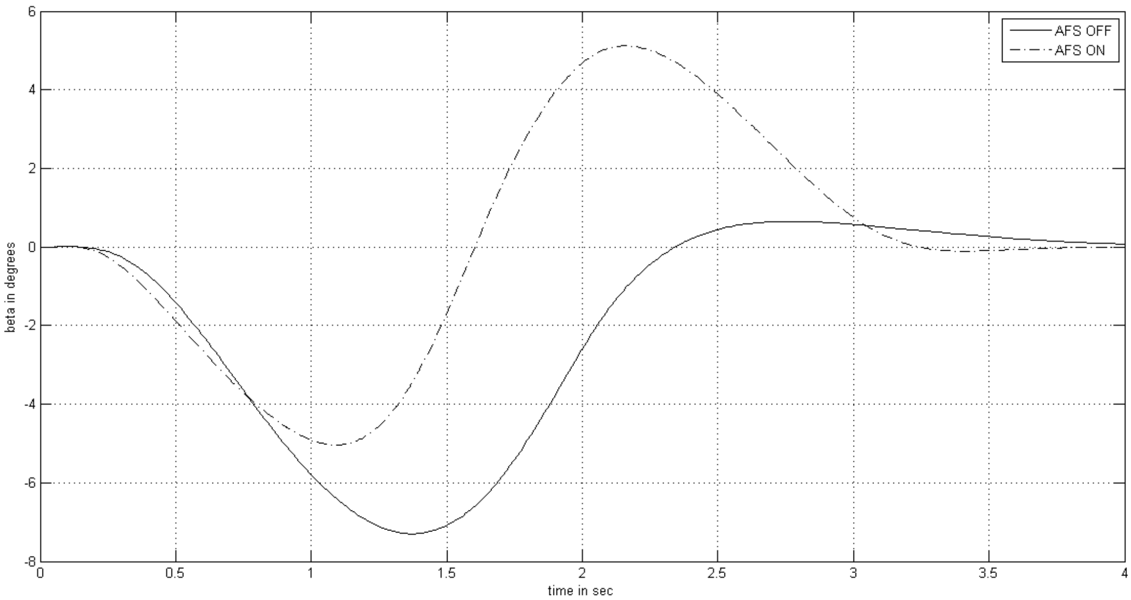

Fig 4.24: Side-slip angle during the SLC with and without AFS on low μ

4.6 Modelling of Normal Force Control (NFC)

4.6.1 Model of active suspension dynamics

The electro hydraulic actuator model used in this these has a spool servo valve model and a hydraulic cylinder model.

A first order dynamic model is used to characterise the behaviour of the spool valve and the mathematical form of its dynamics is given below:

$$\dot{x}_v = \frac{-1}{\tau} x_v + \frac{K_a}{\tau} i_v \tag{4.25}$$

Where,

x_v = the spool valve displacement in m

τ = the valve time constant

K_a = the spool valve gain in m/ A

i_v = the spool valve control current in mA

The displacement of this spoolvalve and the pressure difference across both sides of the hydraulic cylinder causes a load flow, Q_L.

And using Bernoulli's equation this load flow can be calculated as follows:

$$Q_L = C_d w x_v \sqrt{\frac{1}{\rho}(P_S - P_L)} \tag{4.26}$$

Where,

C_d = the discharge coefficient

w = area gradient of servo valve

ρ = the suspension hydraulic oil density

P_S = the supply pressure = 110 bar

P_L = the load pressure

Now using the continuity equation the load pressure P_L can be calculated as a function of the load flow and the suspension deflection velocity.

$$\dot{P}_L = \frac{4\beta_e}{V_t}\left[Q_L - C_t P_L - A\left(\dot{Z}_s - \dot{Z}_{uij}\right)\right] \tag{4.27}$$

Where

β_e = the bulk modulus of the suspension oil

V_t = the effective volume of the hydraulic cylinder

C_t = the leaking coefficient of the hydraulic cylinder

A = the effective cross section area of the hydraulic cylinder

$\dot{Z}_s - \dot{Z}_{uij}$ = is the suspension deflection where the ij stands for individual location of the actuators on the front/rear and left / right wheels.

Finally the output force from the actuators,F_a, can be calculated as product of the load pressure and the cross sectional area of the hydraulic cylinder.

$$F_a = P_L \cdot A \tag{4.29}$$

4.6.2 Development of NFC controller

For the purpose of this thesis, the suspension control strategy used has the following objectives:

- To add the required amount of active suspension forces at the individual wheel corners to reduce the vehicle yaw rate and side-slip angle.
- To reduce or maintain roll angle compared to a passive vehicle.

Two different NFC controller strategies are investigated in this thesis. The first one is a yaw rate error and side slip angle error based fuzzy logic normal force controller. A fuzzy logic control strategy is used for the same reasons as mentioned earlier in this chapter. The main aim of the NFC controller is to minimise the yaw rate and side slip angle error by modulating the front tyre normal forces, using fuzzy feedback control strategy. The NFC controller receives two inputs the yaw rate and side slip angle errors and provides two outputs the normalised active suspension control force. Then an output scaling operation is carried out to convert the normalised active suspension control forces ($\hat{F}_{a\_fl}, \hat{F}_{a\_fr}$) to the required corrective suspension normal forces (F_{a_fl}, F_{a_fr}).

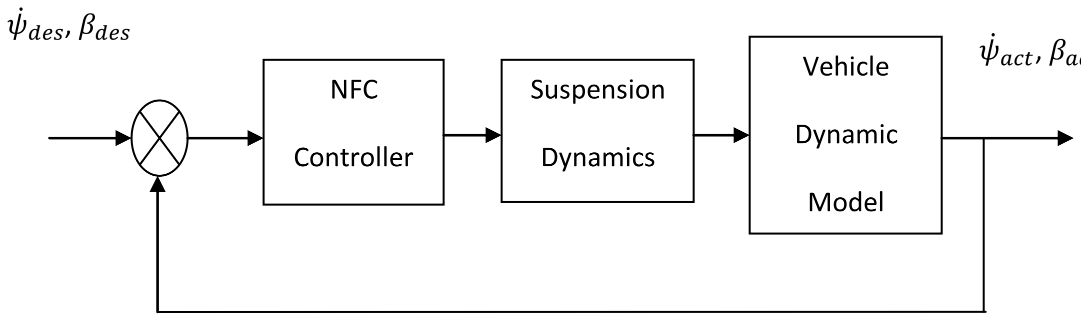

Fig. 4.25: Schematic of the Normal Force Controller (NFC)

Fuzzy Input / Output Selection: As the main objective of the NFC system is to minimise the yaw rate and side-slip angle errors, to obtain the desired vehicle response, the fuzzy logic controller requires two input values:

$$\text{INPUT 1: } e_{\dot{\psi}} = \psi_{des} - \psi_{act} \tag{4.30}$$

$$\text{INPUT 2: } e_\beta = \beta_{des} - \beta_{act} \tag{4.31}$$

As the purpose of this layer of the controller is to calculate the normalised active suspension forces, the same has been designed as the output.

$$\text{OUTPUT 1: } \hat{F}_{a\_fl}, \hat{F}_{a\_fr}$$

The architecture of the fuzzy logic controller has four steps shown in figure 3.25

Fuzzification: makes the controller inputs compatible with the linguistic variables shown in table 4.7

	Linguistic variables
OK	Okay
P	Positive
N	Negative
NC	No Change
PS	Positive Small
PM	Positive Medium
PB	Positive Big

Table 4.7: Table of Linguistic variables for the fuzzy AFS controller

Four and three fuzzy sets are used for both the inputs respectively and four fuzzy sets are used for the output(s). : $e_{\dot{\psi}}$ has a set of values between OK and PB which is defined as follows:

$$\{ e_{\dot{\psi}} \} = \{\text{OK, PS, PM, PB}\}$$

and e_β has a set of values between N and P which is defined as follows:

$$\{e_\beta \} = \{\text{N, OK, P}\}$$

And $\hat{F}_{a\_fl}, \hat{F}_{a\_fr}$ each has a set of values between NC and PB which is defined as follows

$$\{\hat{F}_{a\_fl}, \hat{F}_{a\_fr} = \{NC, PS, PM, PB\}$$

Fuzzy decision Process: processes a list of rules from the knowledge base using fuzzy input from the previous step to produce the fuzzy output. Table 4.8 show the fuzzy rules used in the controller.

$\hat{F}_{afl}, \hat{F}_{afr}$		$e_{\dot{\psi}}$			
		OK	PS	PM	PB
e_{β}	N	NC	PS	PM	PB
	OK	NC	PS	PM	PB
	P	PM	PM	PB	PB

Table 4.8 Fuzzy Rule for the NFC Controller

The fuzzy controller uses the Mamdani Fuzzy Inference System (FIS), which is characterised by the following rule:

IF e_{β} is A and $e_{\dot{\psi}}$ is B THEN $\hat{F}_{afl}, \hat{F}_{afr}$ is C

Defuzzification: Scales and maps the fuzzy output from fuzzy decision process to produce an output value which is the input value to the system being controlled, in our case, the corrective yaw moment. The defuzzification method used here is the centre of area. The universe of discourse of the inputs is selected considering the range of yaw rate and side-slip angle errors without controller. The universe of discourse of the output is normalised to [0, 1].

Output scaling: The controller output $\hat{F}_{afl}, \hat{F}_{afr}$ is scaled to map the corrective active suspension force from the normalised interval.

$$F_{a\_fl} = K_{fuz\_nfc} \cdot \hat{F}_{afl} \tag{4.32}$$

$$F_{a\_fr} = K_{fuz\_nfc} \cdot \hat{F}_{afr} \tag{4.33}$$

K_{fuz_afs} = output scaling factor for NFC fuzzy controller. The controller output is the desired suspension normal force which is then demanded and generated by the individual wheel suspension actuators using simple PID controller.

From the research literature the upper and lower bound of the active suspension force is limited in the ranges of 2500N to 4500N. This active steering actuator saturation is implemented through the output scaling stage of the controller development.

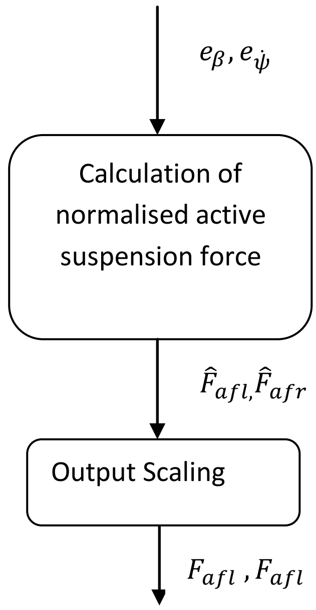

Fig 4.26: NFC Control Architecture – Strategy 1

4.6.3 A novel suspension force control (SFC) strategy

In the course of this research, a novel suspension force control strategy is developed and called as Suspension Force Control (SFC). Instead of using yaw rate error and side slip angle error as in the previous case, the SFC uses the vehicle roll angle as the control input. The control output, desired active suspension force, is produced as a function of the absolute value of the roll angle.

A control allocation strategy is used to allocate the active suspension force at the inner or outer wheels as a signum function of the roll angle. The desired force is then produced by the suspension actuator. This control strategy is different from the active roll moment control strategy where, the roll moment distribution between the front and rear axles is controlled as a function of the vehicle roll angle and or lateral acceleration.

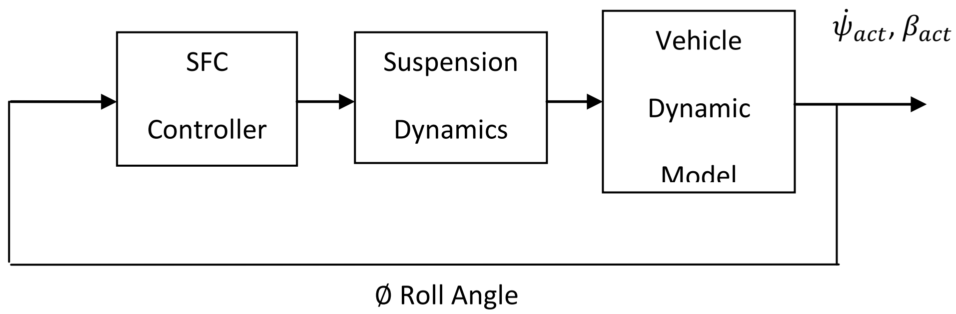

Fig 4.27: NFC Control Schematic – Strategy 2

4.6.4 Simulations

To compare the passive and active vehicle (NFC) performance a single lane change manoeuvre is performed similar to the one used in the AFS controller evaluation. The vehicle is driven at a speed of 96kmph (27.77 m/s) on a dry road with μ = 0.85. Then a single sine steering input of 38° with a frequency of 0.5 Hz is given. The steering ratio obtained through the steering system is 19:1. This produces a sine steer angle equivalent of 2° amplitude at the wheels. The yaw rate and side slip angle output of the nonlinear vehicle model is recorded.

Then the VTD controller is switched ON and the test is repeated for the same test conditions. From figure 4.28 the lane change path followed by both the controlled and uncontrolled vehicle can be seen. It can be seen that for the same steering angle input the NFC increases the generation of front lateral forces by increasing the normal load on front wheels. This increase in front lateral tyre force aids and improves the vehicle's lane changing behaviour.

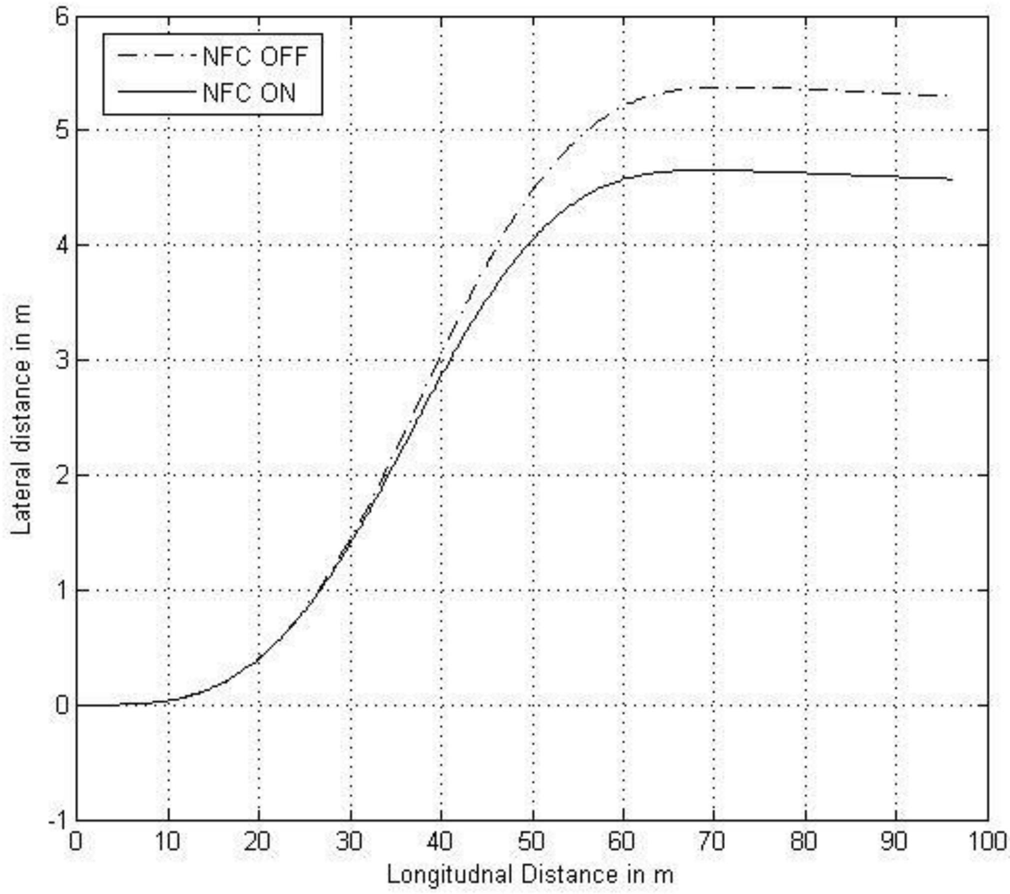

Fig 4.28: Lateral Path Deviation in the SLC with and without NFC on high μ

The figure 4.29 shows that the yaw rate and the side slip angle of the controlled vehicle is less than the uncontrolled vehicle during the single lane change manoeuvre.

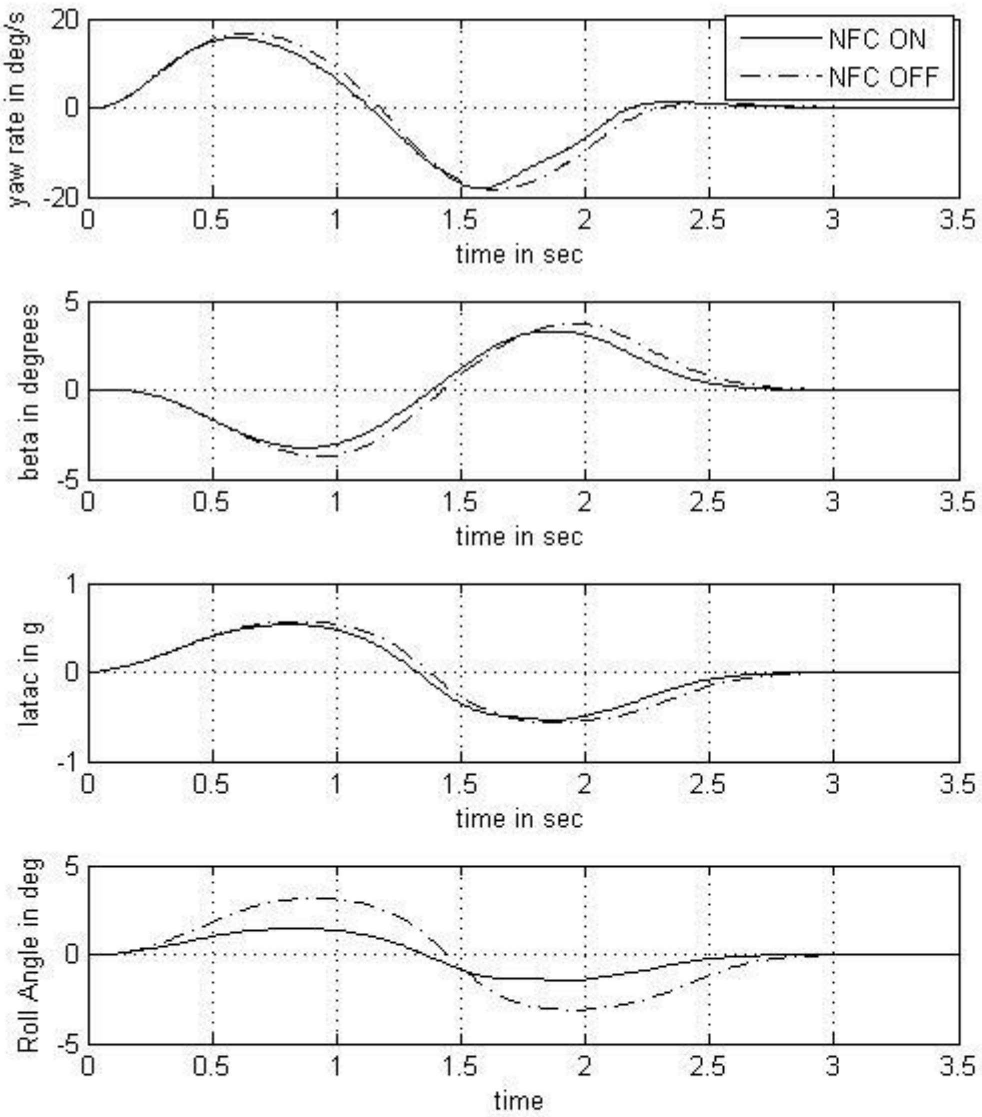

Fig 4.29: Vehicle stability during the SLC with and without NFC on high μ

From the simulation results it can concluded that the controlling of suspension normal forces on individual wheels do have an effect on the vehicle yaw and side-slip dynamics without affecting the roll dynamics. This indicates that the vehicle with active suspension force control increases the stability and extending the operating range compare to its passive counterpart Hence this active chassis control strategy can be considered as one of the key systems for integration in the next chapter.

4.7 Modelling of Variable Torque Distribution (VTD)

4.7.1 Dynamics of TCS:

For the purpose of this research, as TCS is used only as a fundamental building block for the VTD system, a simple first order drive line dynamics is implemented in generating the drive torque. This system is capable of delivering the driving torque at the individual wheels as required. To make the control system design simple and to focus on the research aim, engine and other unnecessary driveline dynamics are neglected. It is assumed that the vehicle is driven at a constant velocity initially and a sudden driveline torque is given to make the wheels spin.

4.7.2 Development of TCS Controller:

Having described the simple driveline dynamics used in this modelling, the control strategy to be used on the TCS controller needs to be finalised next. Considering the key features of a control system such as easy to design, simple to implement, ability to control nonlinear systems and robustness against parameter variation, a fuzzy logic based TCS controller similar to the one developed for the ABS is used.

The TCS fuzzy logic controller used in this thesis is a slip controller, where the error between the desired longitudinal slip and the actual slip is driven to zero during starting from stop and sudden acceleration while moving to avoid spinning of the wheels.

The aim of a TCS wheel slip controller is to maintain the tyre to operate near the maximum friction point during sudden acceleration, thereby producing maximum tyre force that aids to increase the acceleration performance, at the same time, avoiding the spinning of the wheels by not letting them to slip towards the maximum slip (100%) and thus maintaining the ability to steer the vehicle during acceleration.

The schematic of the typical TCS control system used in this thesis is shown in figure 4.30.

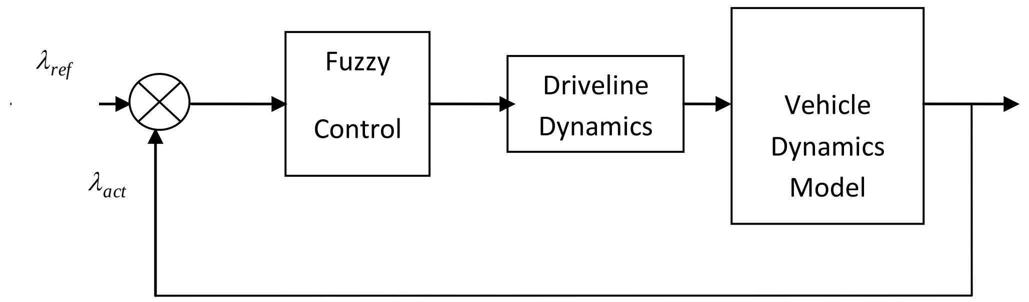

Fig 4.30: TCS Control Architecture

The fuzzy control rules formulated using the input and output variables are described in the following table:

Change in Control Signal		$\boldsymbol{\lambda_{error}}$				
$\boldsymbol{\frac{d\lambda_{error}}{dt}}$		PB	PS	ZO	NS	NB
	PB	NB	NB	NB	NS	ZO
	PS	NB	NB	NS	PS	PS
	ZO	NB	NS	ZO	PS	PB
	NS	NS	NS	ZO	PB	PB
	NB	NS	NS	PS	PB	PB

Table 4.9: Fuzzy rules table for the TCS controller

To start with, a typical open differential model is built and converted to a limited slip differential. A limited slip has the same components as an open differential except for a clutch that provides an additional oath for torque transfer. In fugure # Td is the drive torque transmitted to the front differential, Tdiff is the torque transmitted through the differential gears, and Tct is the torque transmitted through the clutch. Assuming the following:

1) Efficiency of the torque transmission is 100%
2) Differential gear ratio from the prop shaft to differential is 1

Then we have,

$$T_f = T_{ct} + T_{diff} \tag{4.34}$$

Since T_{diff} is equally distributed to the left and right front wheels, then the net torque to the front left and rear wheels are given by:

$$T_{fl} = T_{ct} + \frac{T_{diff}}{2} = \frac{T_f + T_{ct}}{2} \tag{4.35}$$

$$T_{fr} = \frac{T_{diff}}{2} = \frac{T_f - T_{ct}}{2} \tag{4.36}$$

To reduce the complexity of the modelling and save the simulation time the LSD is modelled as a system along with the equations to represent the necessary dynamics.

4.7.3 Development of VTD Controller:

A driveline based yaw control strategy proposed by Rajamani, (2007) is followed with significant modifications to the driveline dynamics and control strategy to suit the purpose of this thesis. The control architecture of the VTD system is hierarchical as used in the ESC controller development in this thesis. The upper controller has the objective of ensuring yaw stability control and assumes that it can command any desired value of yaw moment within the capability of the driveline system. The measurement from the wheel speed sensors, yaw rate sensor, an estimation of the vehicle side-slip angle and a steering angle sensor are used. A fuzzy logic control strategy uses these measurements and computes the desired value of the corrective yaw moment. The lower controller ensures that the desired value of yaw torque commanded by the upper controller is indeed obtained from the torque management system. The lower controller uses the driveline dynamics and controls the biasing of the drive torque management system to provide the desired yaw torque for the vehicle. The following figure describes a schematic of the VTD system used in this thesis.

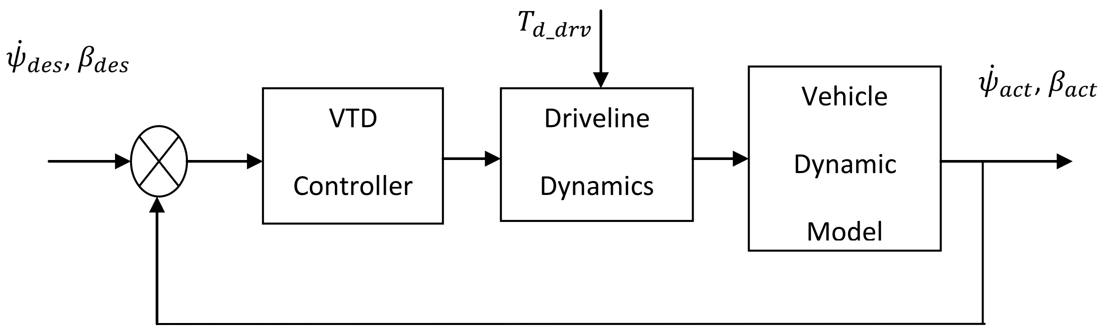

Fig 4.31: VTD Control Architecture

The required active yaw torque is calculated by the fuzzy controller in the upper layer as described in the following table.

$\widehat{M}_{zc}$		e_{β}				
$e_{\dot{\psi}}$		NB	NS	ZE	PS	PB
	PB	NB	NB	NM	NB	NB
	PS	NB	NM	NS	NM	NS
	ZE	NS	NS	ZE	PS	PS
	NS	PB	PM	PS	PM	PS
	NB	PB	PB	PM	PB	PB

Table 4.10: Fuzzy rules table for the VTD controller

Having calculated the required corrective yaw torque the controller allocates through the LSD system to either the front left wheel or right wheel as defined by the following algorithm.

$e_{\dot{\psi}}$ > 0	$\delta_f > 0$	OS	FRW
	$\delta_f < 0$	US	FLW
$e_{\dot{\psi}}$ < 0	$\delta_f > 0$	OS	FLW
	$\delta_f > 0$	US	FRW
$e_{\dot{\psi}}$ = 0	*for all* δ_f	-	HOLD

Table 4.11: Allocation of braking force on individual wheels using VTD

4.7.4 Simulations:

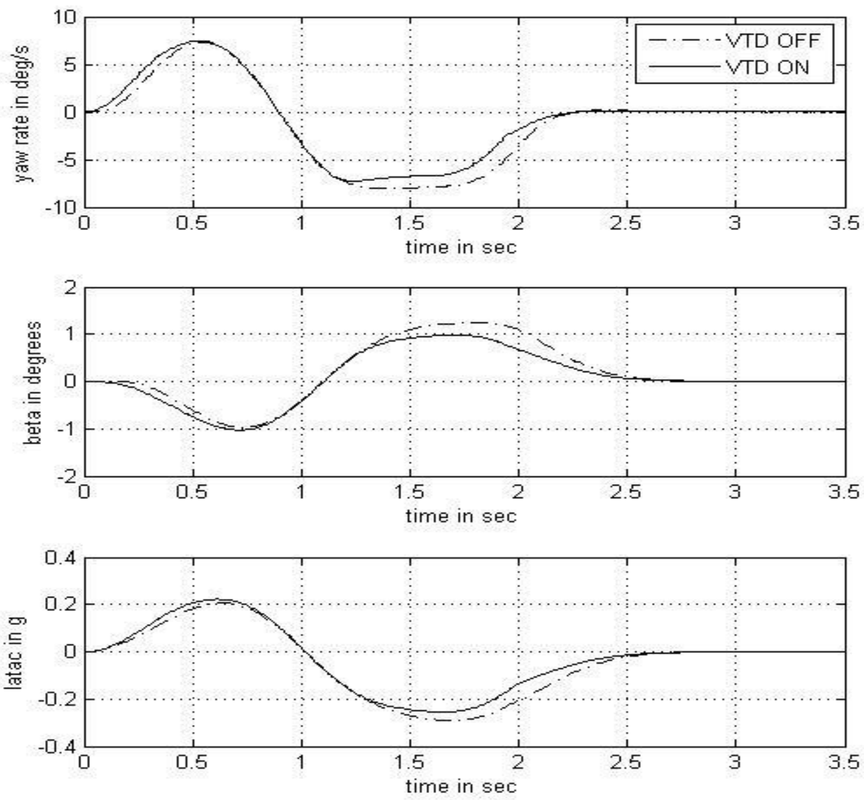

Fig 4.32: Stability during the SLC with and without VTD

4.8 Summary

This chapter discussed the development of active vehicle dynamics models. It began by discussing the history of active vehicle dynamic systems in the research literatures. Then a detailed discussion about the development of Simulink models for electronic stability control, active front steering, suspension normal force control and variable torque distribution systems is carried out. The two fundamental building blocks of electronic stability control and variable torque distribution, the anti-lock braking system and traction control system are developed respectively. The controllers for the respective active systems are developed using fuzzy and PID strategies. Handling simulations are carried out to compare the each active system against their passive counterparts. The simulation results prove that the active systems are better in reducing the yaw rate and sideslip angle compared to the vehicle with passive systems.

Chapter 5

Integration of Active Chassis Control

5.1 Introduction

As discussed earlier in this thesis, a vehicle can have its vehicle dynamics characteristic as an under-steer, a neutral steer or an over-steer depending upon the vehicle (statics & dynamics) and the environmental parameters. For a passive vehicle this characteristic is defined at the design stage but can't be controlled or altered during the dynamic operation. For an active vehicle this characteristic is designed together with an ability to vary or adapt within a predefined control range based on the dynamic operating conditions.

The control range of these dynamic operating conditions are defined by the ability of each active system in generating the desired control effort. That means within this control range each of these systems is capable of generating the required corrective control output to enhance the system performance and to achieve its individual control objectives. Beyond that range these systems might become ineffective and might deteriorate in their ability to provide improved performance. This range of control can be called the 'control authority' of active systems.

The overall characteristic of such a control authority for each active system is based on the overall dynamics of the system. However the details, such as the amplitude of the corrective control forces, are a function of many vehicle parameters such as geometry, actuator capacity and type. So the control authority of an active suspension system will be the same irrespective of whether it's a vehicle with a smaller wheel base, or a longer wheel base or with a higher CoG / lower CoG or a vehicle with a smaller active suspension actuator or bigger one. However the amplitude of the control forces generated will vary as a function of all such parameters.

Every active chassis control system has such a range and the aim of the early sections of this chapter is to identify those ranges, or control authorities, of each of the four active systems under consideration by analysing their control characteristics. This chapter is concerned with using them to develop a novel integrated chassis controller strategy that enhances the vehicle dynamic performance.

For the ease of reading the results/plots a consistent line style approach is followed throughout this chapter. For a comparative study between the passive and the active systems (i.e. ESC OFF and ESC ON) a dotted/dashed line style is used for passive systems and a solid line style for active systems. This is also clearly highlighted in the plot legend. If the results/plots are about a comparative study between active systems and active systems in standalone manner, then the active systems are referred with dashed/discontinuous lines and the standalone systems are referred with solid lines. Again this is clearly highlighted in the legend section of the plots. If the results/plots are about the performance comparison between standalone and integrated systems, the standalone systems are referred with dashed lines and the integrated systems are with solid lines.

5.2 Analysis of Standalone systems

The active chassis control systems can be classified as standalone systems if each system has its own sensor(s), controller and actuator(s) modules. The standalone systems do not interact with each other in terms of resources and information sharing. They also individually try to achieve their own control objective(s) without taking into account whether it affects the control objective(s) of other active systems or not.

The section to analyse the standalone control systems is divided into four subsections, one for each active chassis system in consideration. The overall aim of all these systems is to improve the vehicle stability by reducing the yaw rate and side slip angle, but they achieve it through different methods, such as controlling the distribution of braking, driving, steering and suspension forces. Studying and analysing the ability of each of these four systems in developing their control outputs will highlight their individual control authority in improving vehicle handling.

To improve the stability of any vehicle that deviates from its desired trajectory (under steer or over steer), a corrective control action is generated and applied. As seen from the development of individual chassis control systems in the previous chapter, this control action is called the application of a corrective yaw moment or corrective yaw torque at the CofG of the vehicle. The amplitude of this corrective yaw moment generated by each chassis control system depends on their individual system characteristics, the operating conditions of the vehicle, the capacity and the dynamics of the actuators themselves. So in the next few sections the ability of each of these four chassis control systems in generating the corrective yaw moment across the vehicle operating regions will be simulated and analysed.

The range of operation of a vehicle can be defined as a function of the dynamic environment within which the vehicle can be driven either in a passive or active manner. The boundary of this range of operation can be defined using many of the vehicle dynamic parameters, such as lateral acceleration (popularly known as ‘latac’), side-slip angle, yaw angle, vehicle speeds and their respective rates etc. One of the most popular methods of defining this range by the vehicle dynamics community is using the lateral acceleration of a vehicle. As lateral acceleration is a function of two key vehicle parameters, the velocity and the radius of turn, the whole vehicle operating range can be defined by means of this unique parameter. So, in terms of the lateral acceleration, a vehicle operating range can be divided into three distinct operating ranges as follows:

Low Lateral Acceleration Range = up to 0.3g

Medium Lateral Acceleration Range = 0.3g to 0.6g

High Lateral Acceleration Range = above 0.6g

Within each of these ranges how effective are the chosen four active systems in improving the vehicle handling show the control authority of these systems. For example if active chassis system 'A' dominates in reducing the yaw rate and sideslip angle in the low latac range but its influence diminishes in the medium latac range and if it ceases to play any role in improving the vehicle handling dynamics performance at all in the high latac range, then it can be said that the active chassis system 'A' is the most efficient system to use and has a strong control authority until the vehicle lateral acceleration reaches 0.3 g. And if both systems 'A' and 'B' are dominant in medium latac and high latac ranges in improving vehicle handling, but system 'A' negatively influences other vehicle performance parameter(s) compared to 'B', then the strategy should be to hand over the control authority to system 'B' at the high latac range to protect the current vehicle dynamics performance.

5.2.1 Control authority of Electronic Stability Control System

The control authority of electronic stability control system has been analysed by running the vehicle model on dry, wet and icy road conditions at 0.2g and 0.3g for the low latac, 0.4g, 0.5g and 0.6g for the medium latac and at 0.7g and 0.8g for the high latac operating ranges respectively. The control authority of ESC at the handling limits is also investigated. First the corresponding steering angles to produce these lateral accelerations are calculated through simulations using step steer inputs. Then the resultant yaw rate, sideslip angle, lateral acceleration (for verification) and longitudinal vehicle speed are obtained through the full vehicle simulations for a “Sine with Dwell” steering input with and without ESC activated.

From the simulation results it can be seen that ESC improves the vehicle handling by reducing the peak yaw rate by 12% at 0.2g and by 9% at 0.3g latac on a dry road. Similarly a 23% reduction in the peak slip angle is obtained at 0.2g and a 20% reduction at 0.3g. One important observation concerning the activation of the ESC controller is that it reduced the longitudinal vehicle speed by 1.4% at 0.2g and by 2.0% at 0.3g latac. This highlights the intrusive nature of this control system in the longitudinal dynamics of the vehicle. This is generally not a preferable characteristic for a vehicle from a driver’s point of view, especially in the low lateral acceleration range, which is not a safety critical operating range.

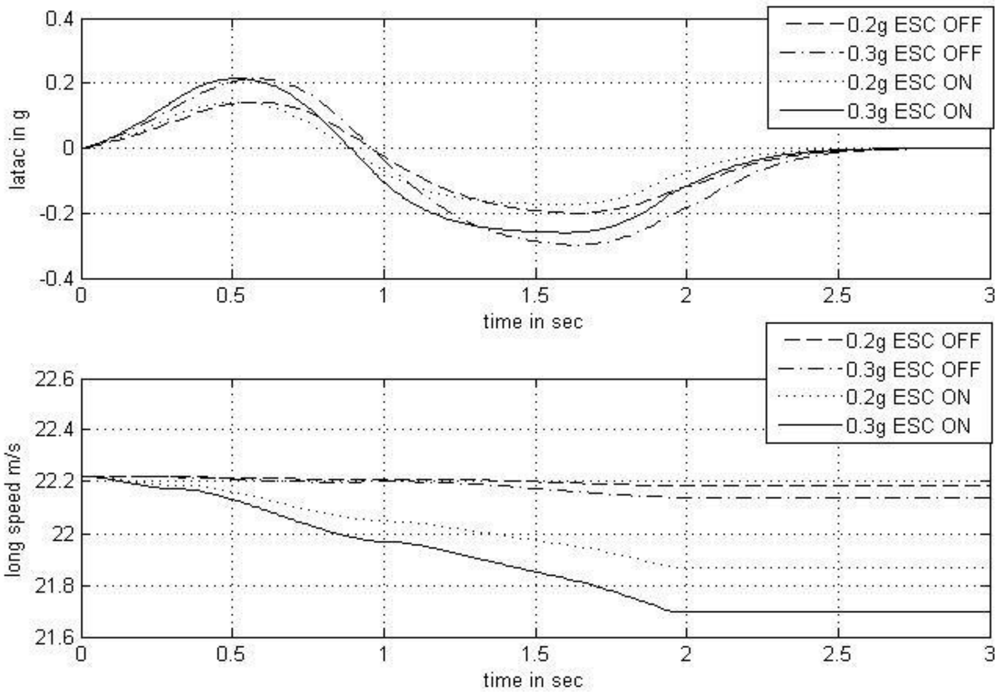

Fig. 5.1: Intrusive nature of ESC on longitudinal dynamics in low latac

In the medium latac range, the ESC improves the vehicle handling by reducing the peak yaw rate by 12% at 0.4g, by 10% at 0.5g and by 6% at 0.6g latac on a dry road. Similarly a 22% reduction in the peak slip angle is obtained at 0.4g, 21% at 0.5g and a 20% reduction at 0.6g. Also it is observed that the activation of ESC controller reduces the longitudinal vehicle speed by 2.5% at 0.4g, 2.95% at 0.5g and by 3.2% at 0.6g latac. This is again not a preferable characteristic for a vehicle from a driver’s point of view in the medium lateral acceleration range which is not a safety critical operating range.

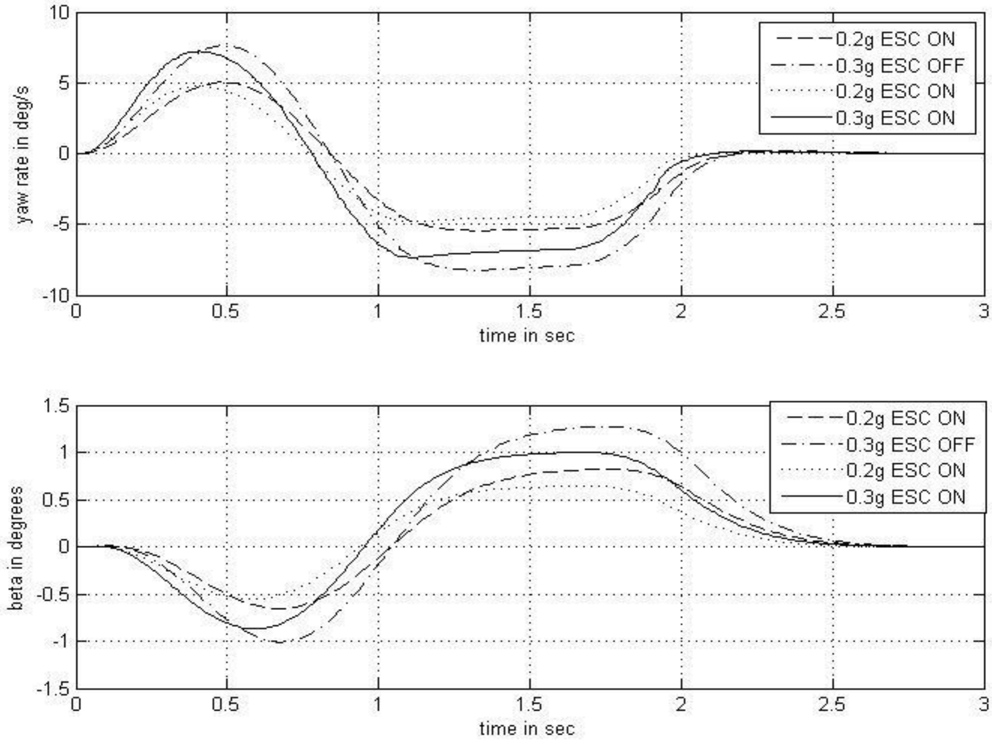

Fig. 5.2: Control authority of ESC during low latac

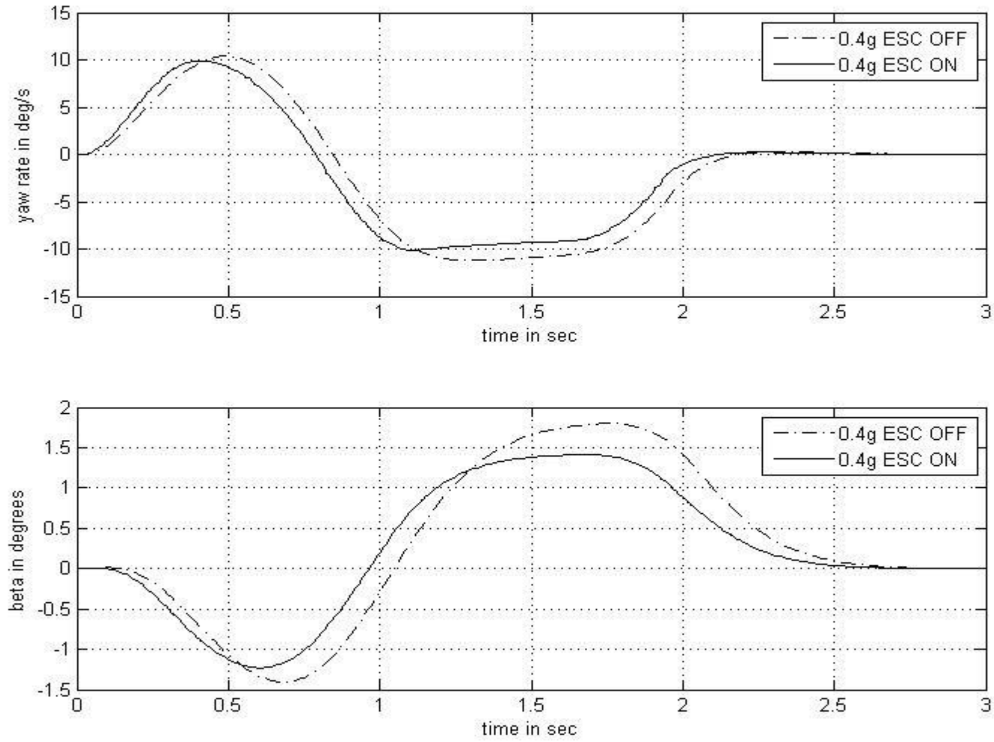

Fig. 5.3: Control authority of ESC at 0.4g

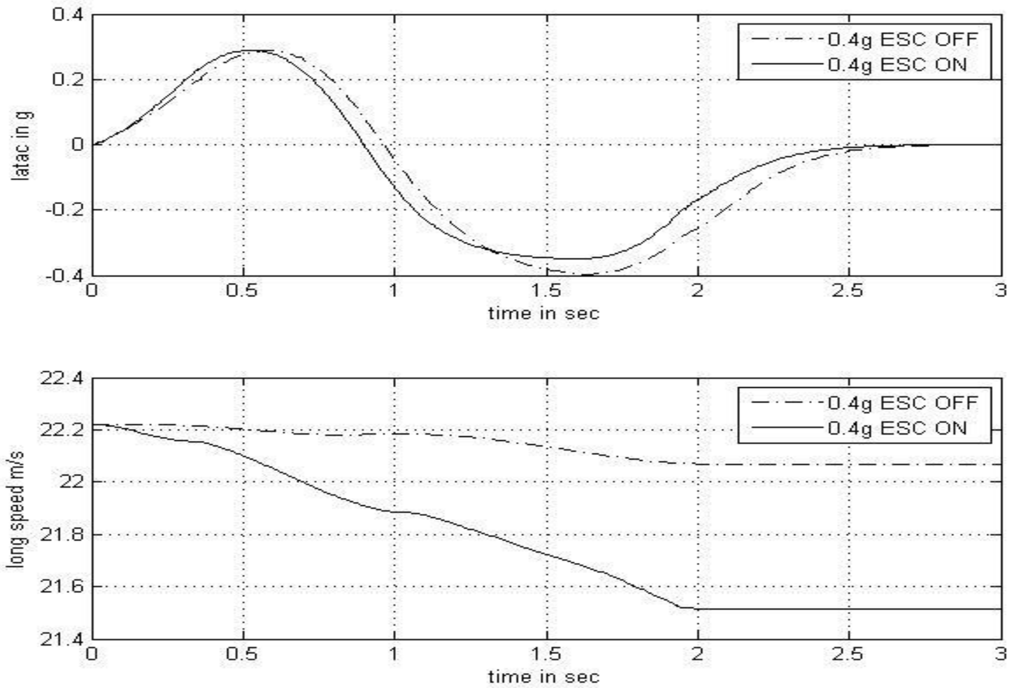

Fig 5.4: Intrusive nature of ESC on longitudinal dynamics at 0.4g latac

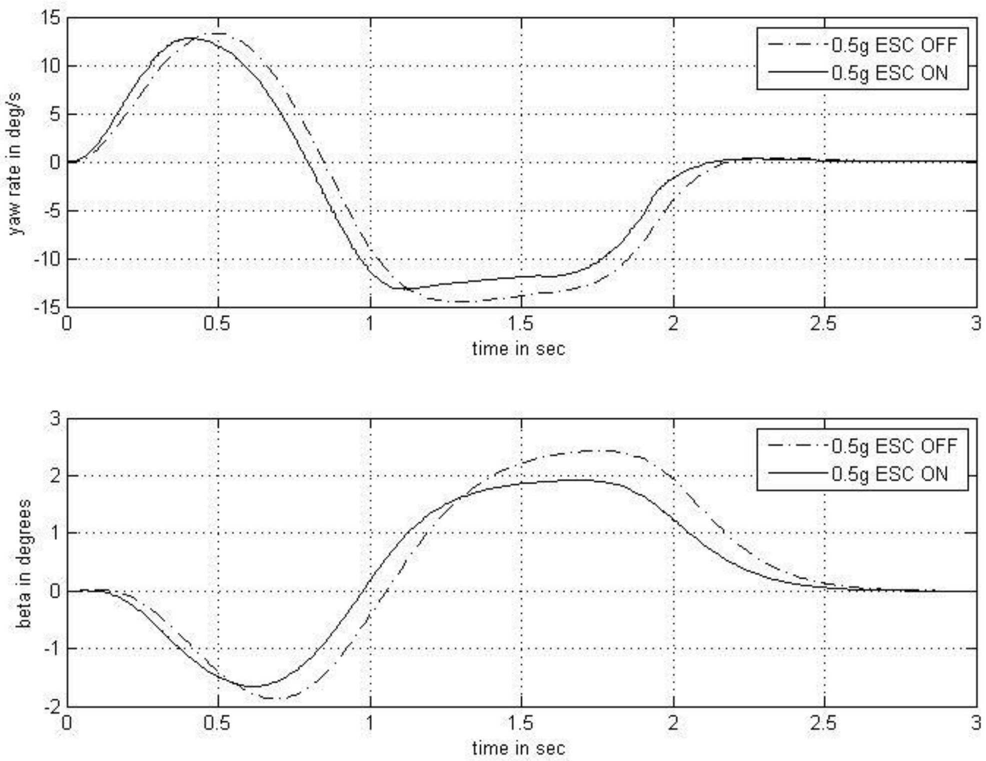

Fig 5.5: Control authority of ESC at 0.5g latac

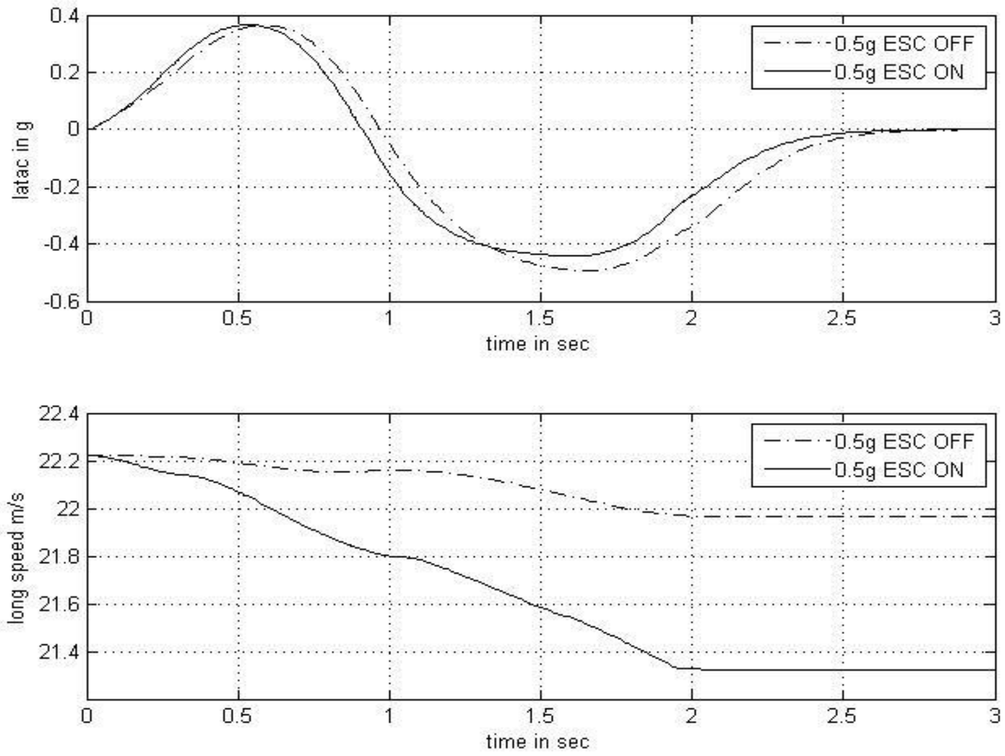

Fig 5.6: Intrusive nature of ESC on longitudinal dynamics at 0.5g latac

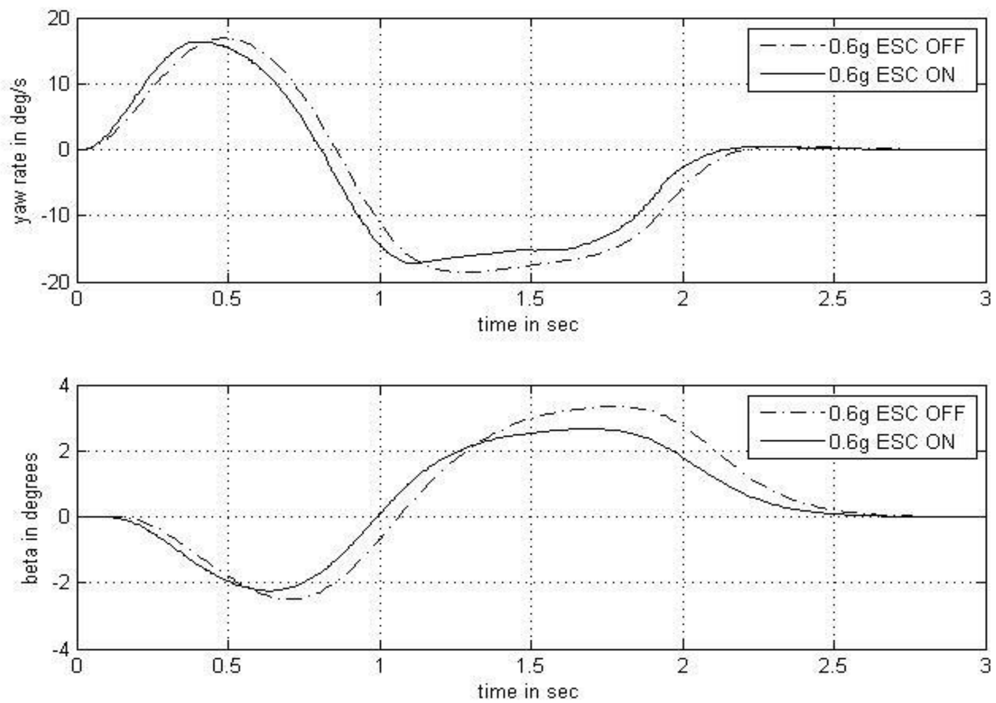

Fig 5.7: Control authority of ESC at 0.6g latac

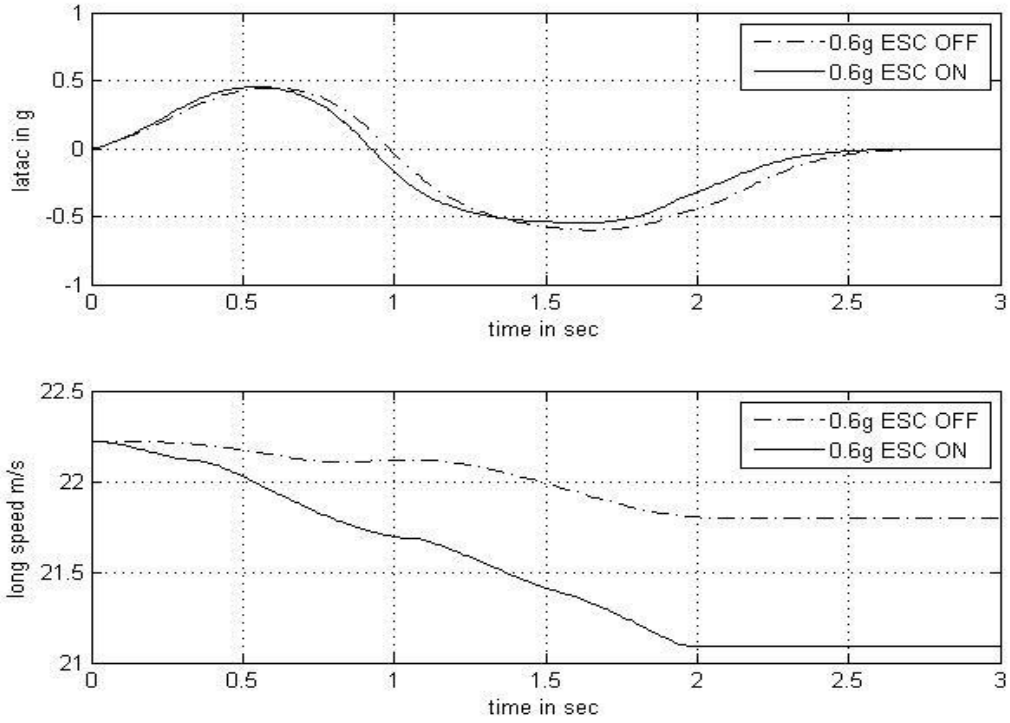

Fig 5.8: Intrusive nature of ESC on longitudinal dynamics at 0.6g latac

In the high latac range ESC improves the vehicle handling by reducing the peak yaw rate by 12% at 0.7g and by 9.6% at 0.8g latac on a dry road. Similarly a 23% reduction in the peak slip angle is obtained at 0.7g and a 27% reduction at 0.8g. Another important observation concerning the activation of ESC controller is that it reduced the longitudinal vehicle speed by 1.4% at 0.7g and by 2.9% at 0.8g latac.

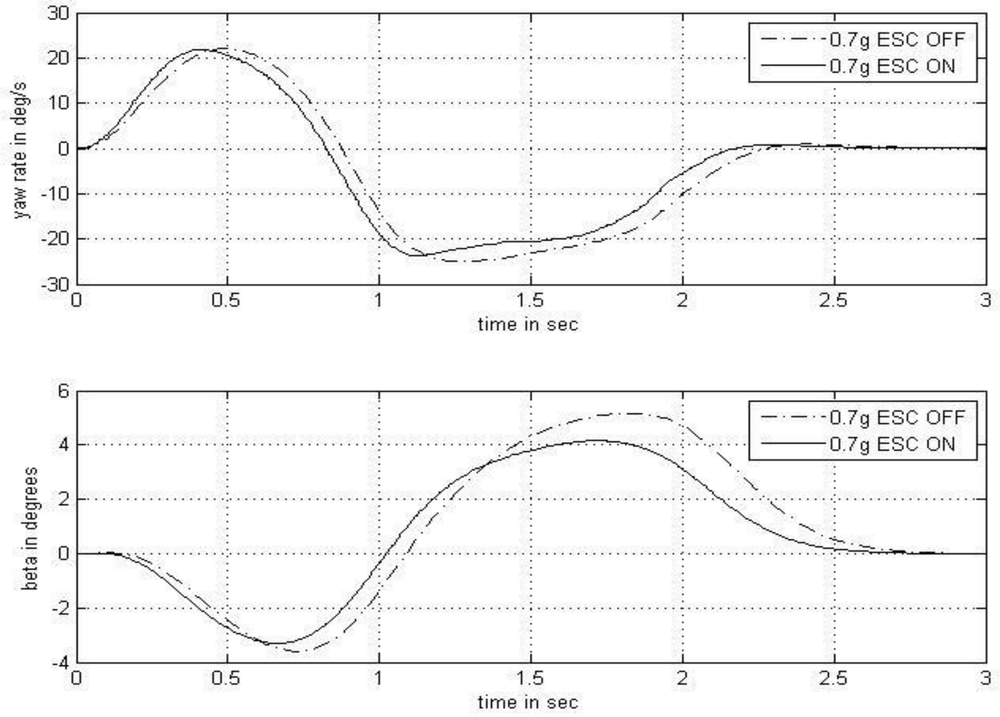

Fig 5.9: Control authority of ESC at 0.7g latac

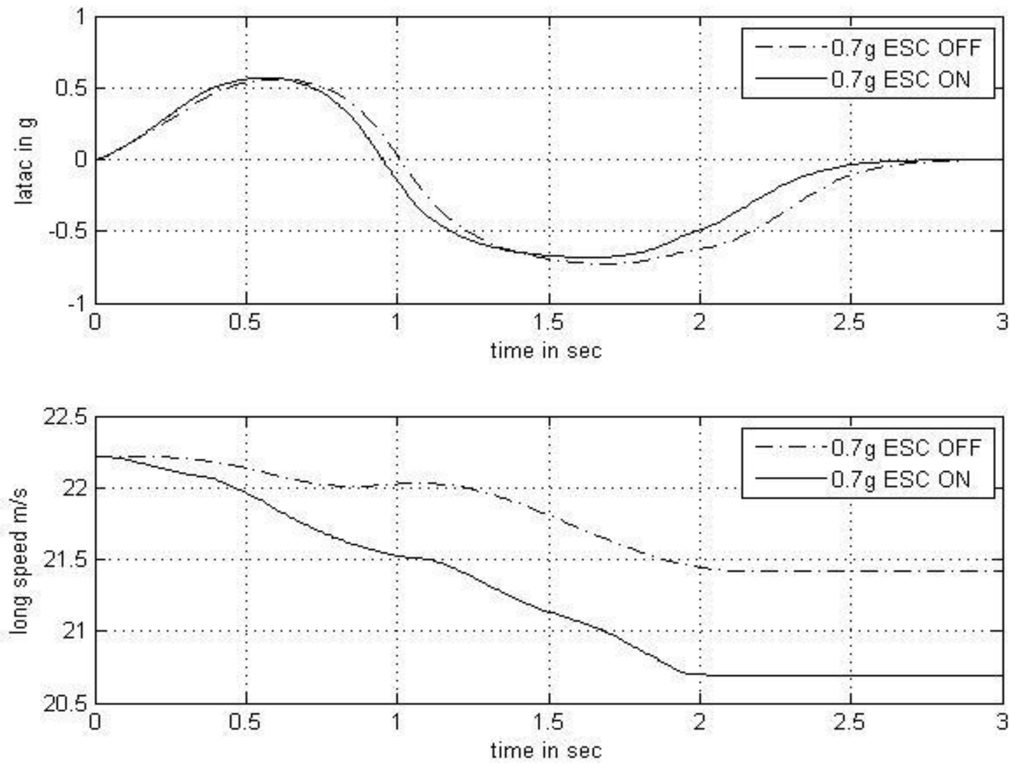

Fig 5.10: Intrusive nature of ESC on longitudinal dynamics at 0.7g

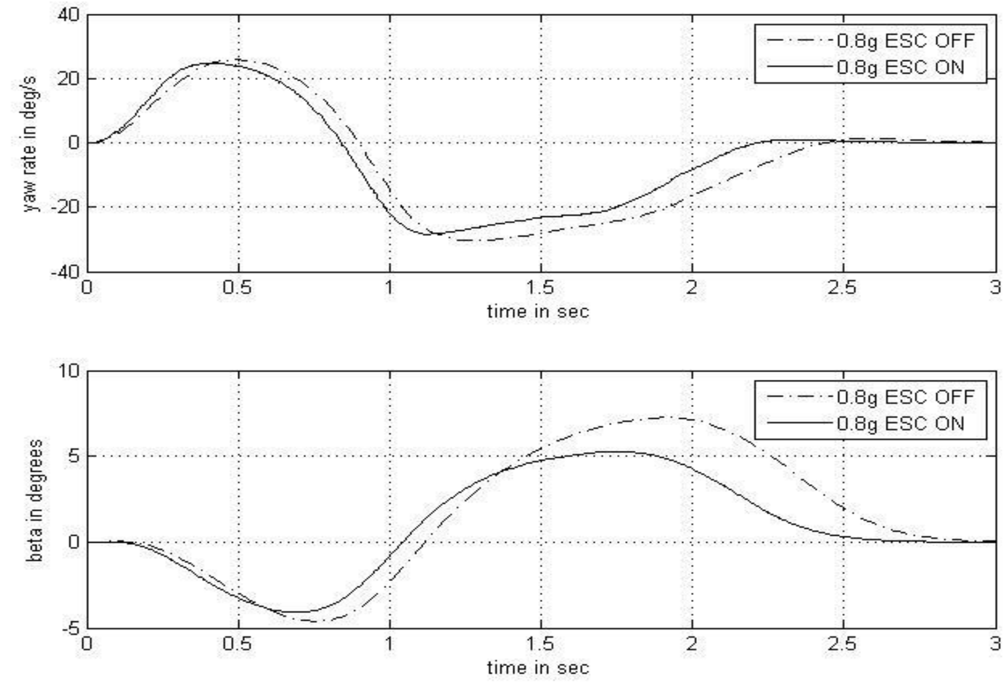

Fig 5.11: Control authority of ESC at 0.8g latac

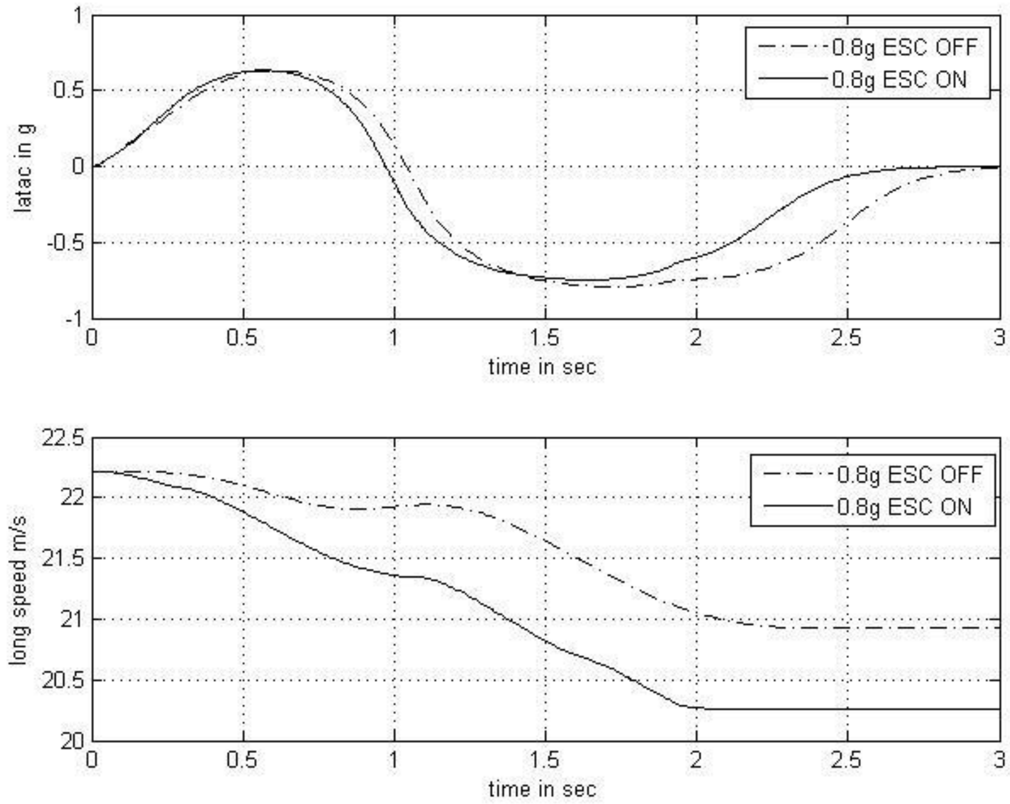

Fig 5.12: Intrusive nature of ESC on longitudinal dynamics at 0.8g

When the vehicle is being driven at the maximum possible lateral acceleration, then it is said to be at its handling limits. Two of the key parameters that define the vehicle lateral acceleration are the speed at which the vehicle is driven and radius of curvature of the turn, which is a function of the steering angle input to the vehicle. But the maximum possible lateral acceleration or limit handling lateral acceleration is limited by the coefficient of friction between the tyre and the road. If pushed beyond the limits the frictional contact between tyres and road breaks and the vehicle loses its ability to generate forces to react at the tyre-road contact patch in response to the lateral acceleration that is acting through the CoG of the vehicle.

From the simulations it is observed that when the vehicle is being operated at it limits with the ESC not activated then the vehicle yaw rate does not follow and respond to the steering input change and saturates to a maximum possible value. Also the vehicle side slip angle spins out of control and keep on increasing. When the simulation is repeated with ESC activated, it can be observed that the ESC still can influence the vehicle handling by making the vehicle yaw rate and sideslip angle to respond to the steering input. As limit handling operation is a safety critical situation the loss of longitudinal speed and the intrusion of ESC on the longitudinal dynamics of the vehicle is of less importance. This proves that the ESC has an ability to influence the vehicle handling by reducing yaw rate and sideslip angle, even at the handling limits.

From the above analysis, it is evident that the ESC has a strong control authority in improving the vehicle handling by reducing the yaw rate and vehicle sideslip angle at the low, medium and high latac ranges. It also has a reasonable control authority to make the vehicle yaw rate and side-slip angle to respond and follow the driver's steering input even at the vehicle handling limits.

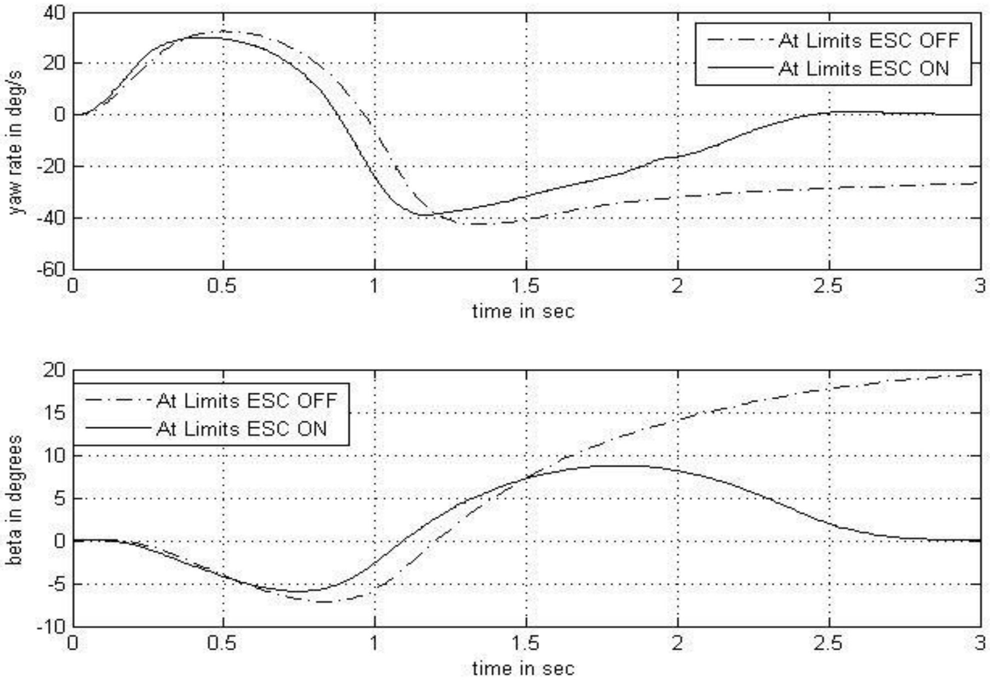

Fig 5.13: Control authority of ESC at the limits

When simulated over a wet road with a coefficient of friction 0.5 in the medium latac range at 0.2g, the electronic stability control improves the rate of response of yaw rate with a marginal or no improvement in the peak yaw rate value and let the vehicle to track the steering input better. It is also observed that the vehicle completes the manoeuvre earlier than the passive vehicle due to the brake assistance from the ESC system. But there is a significant improvement in the vehicle stability in terms of vehicle peak side-slip angle reduction with ESC. And a similar characteristic is observed at 0.3g and at 0.4g in the low and medium latac ranges respectively.

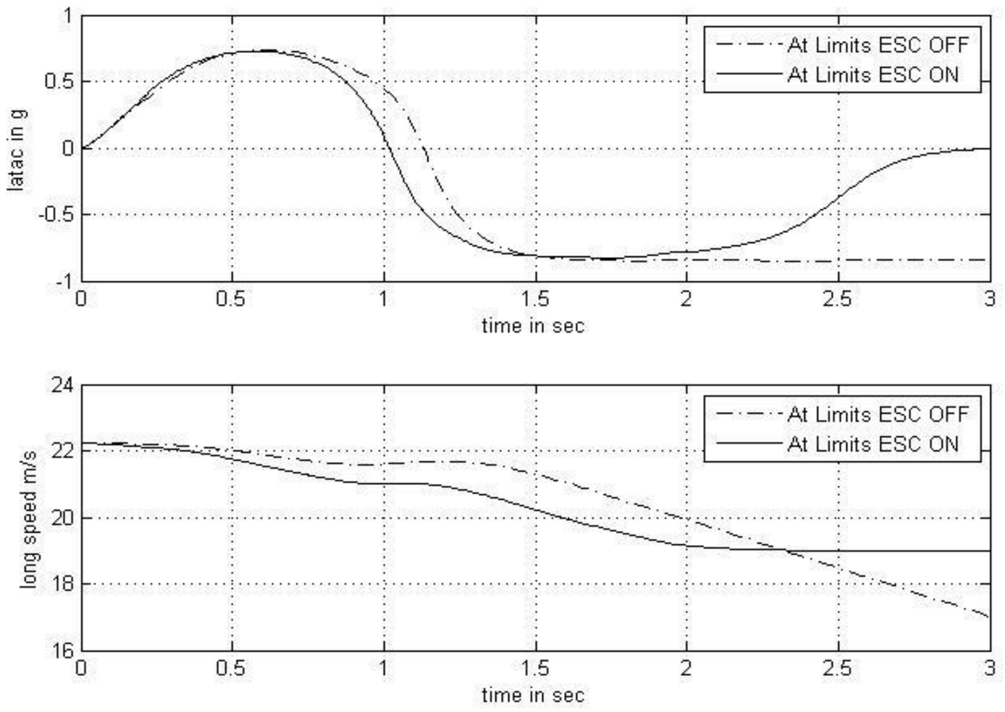

Fig 5.14: Influence of ESC on longitudinal dynamics at the limits

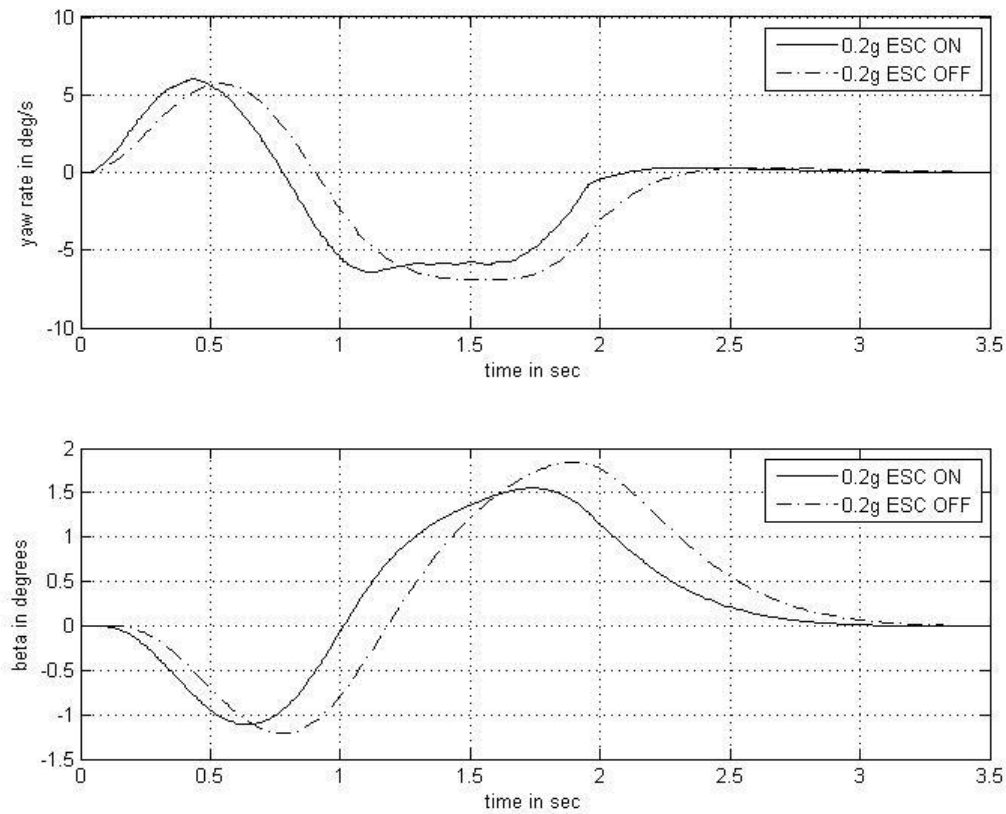

Fig 5.15: Control authority of ESC at 0.2g on wet road conditions

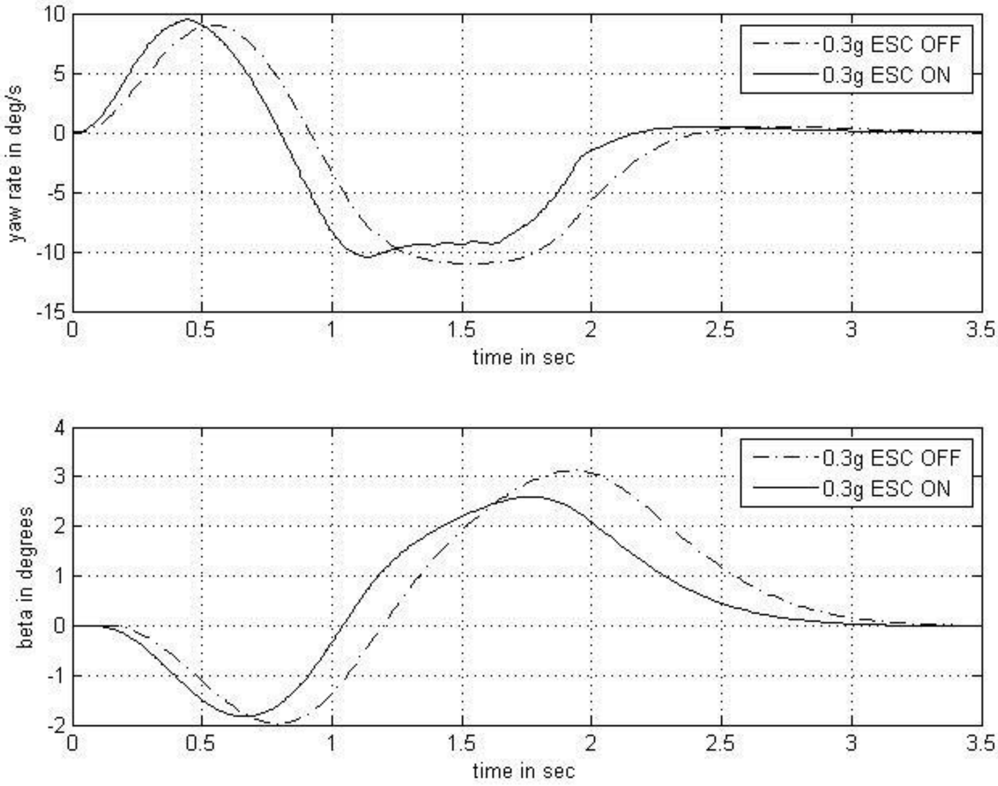

Fig 5.16: Control authority of ESC at 0.3g on wet road conditions

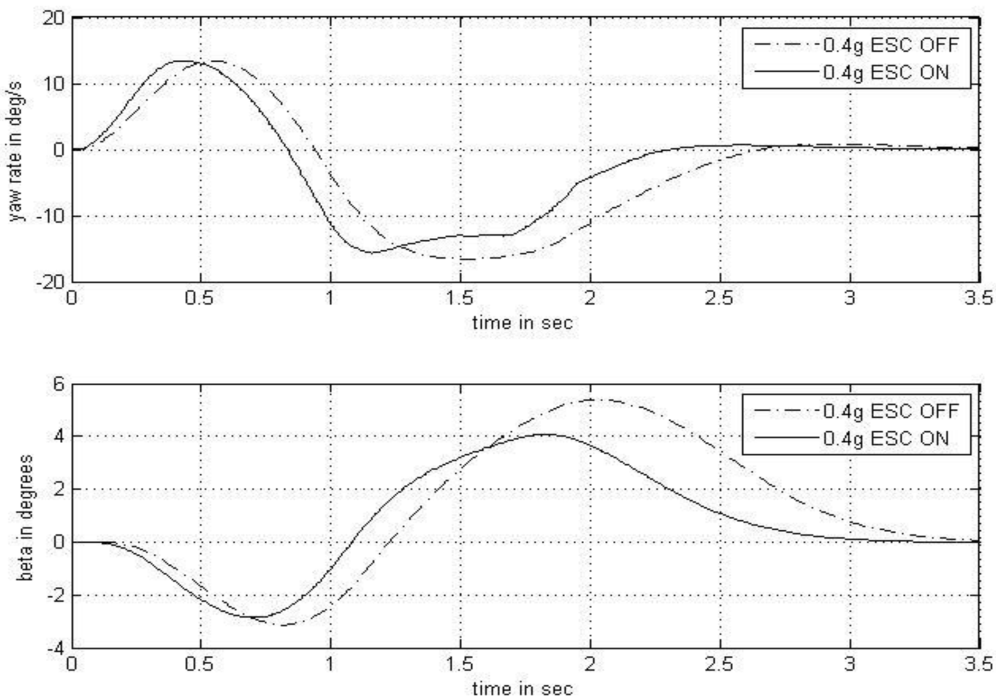

Fig 5.17: Control authority of ESC at 0.4g on wet road conditions

Activation of ESC at the limits on wet and icy roads also show that electronic stability control system has the control authority to provide a desired influence in improving vehicle handling.

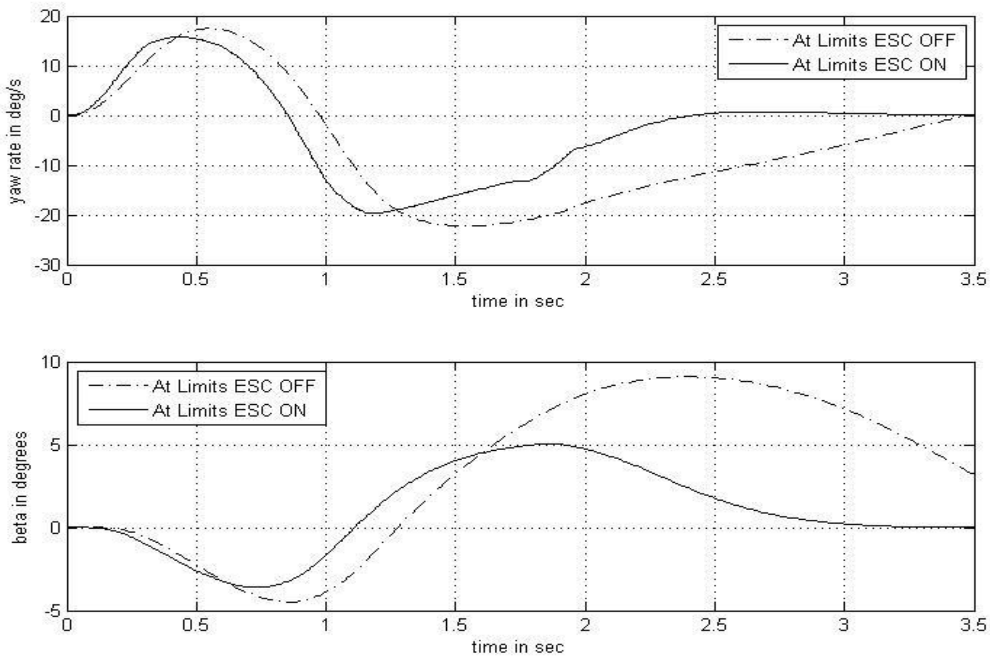

Fig 5.18: Control authority of ESC at the limits on wet road conditions

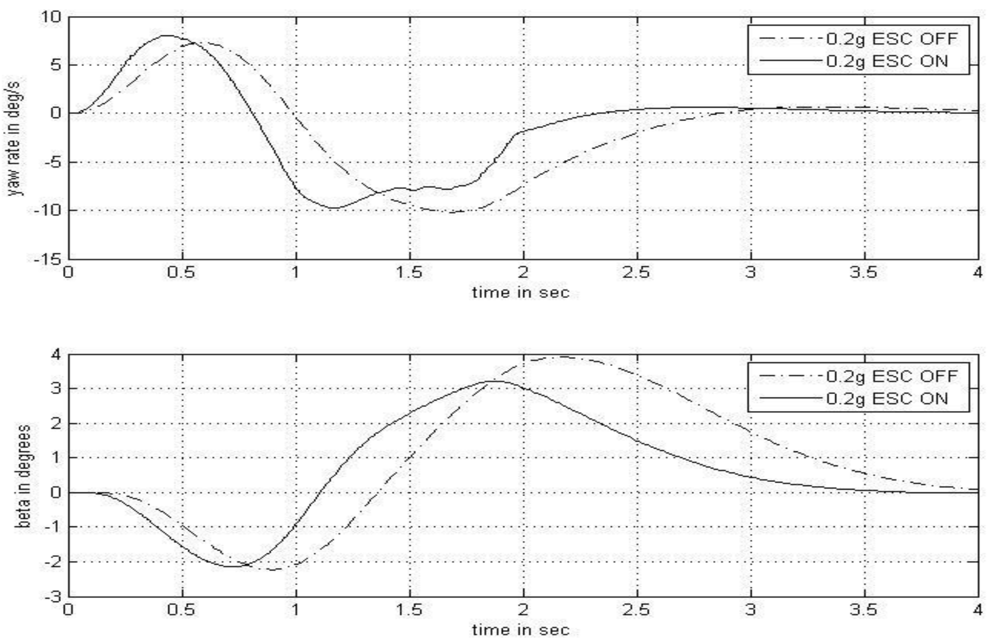

Fig 5.19: Control authority of ESC at 0.2g on Icy road conditions

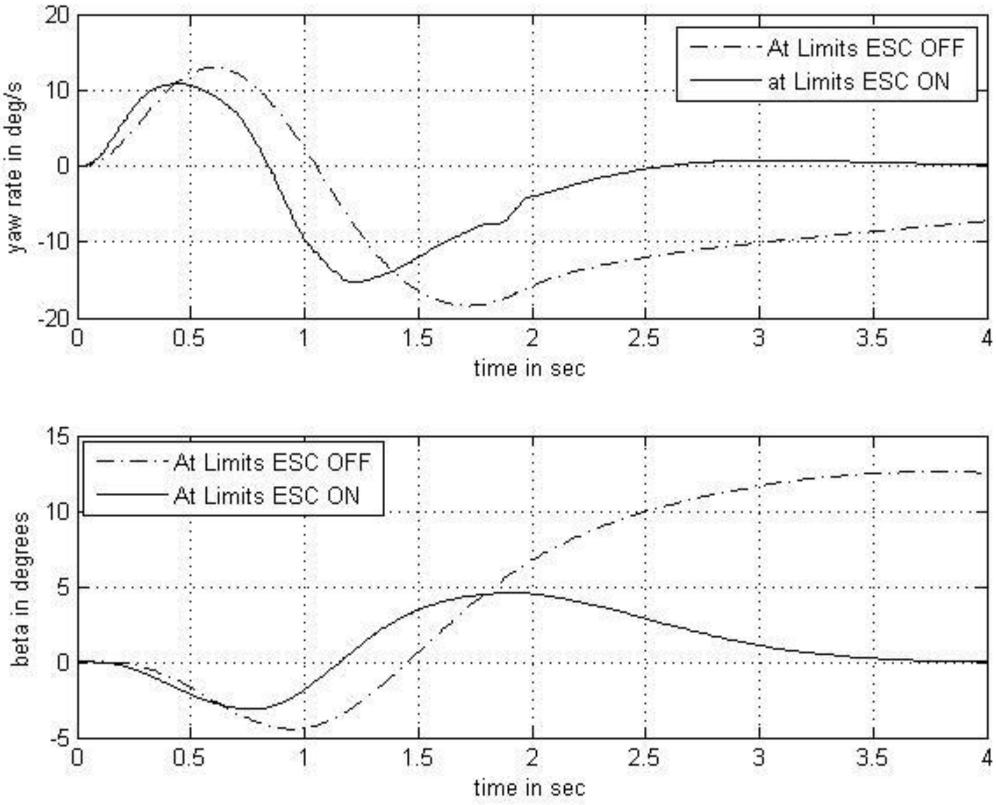

Fig 5.20: Control authority of ESC at the limits on Icy road conditions

The following table 5.1 describes a rating scale used in this section to classify the intrusive nature of the active chassis systems in longitudinal dynamics. It rates the systems from 1 to 4, 1 being the least intrusive and 4 being the most intrusive.

Rating	Description
1	Best
2	Better
3	Good
4	worst

Table 5.1: Rating based on the intrusion on longitudinal dynamics

	Low Latac up to 0.3g			Medium Latac 0.3g to 0.6g			High Latac above 0.6g			At the Limits
	$\dot{\psi}$	β	$\mathbb{V}_x$	$\dot{\psi}$	β	$\mathbb{V}_x$	$\dot{\psi}$	β	$\mathbb{V}_x$	ψ β
Dry	✓	✓	4	✓	✓	4	✓	✓	4	✓
Wet	✓	✓	4	✓	✓	4	✓	✓	4	✓
Icy	✓	✓	4	✓	✓	4	✓	✓	4	✓

Table 5.2: Summary of control authority of ESC over the vehicle latacs

In summary, the ESC system has the ability to improve the vehicle handling in low, medium and high latac vehicle operation ranges on all the three possible road frictional conditions, such as dry, wet and icy. It even has the ability to influence the vehicle handling at the limits of vehicle operation. But due to its inherent nature of using the braking forces to generate the corrective yaw moment, it intrudes in the longitudinal dynamics of the vehicle and reduces the exit speed at the end of the manoeuvre and the overall driving feel. This is not an issue at the safety critical handling limits, but will be considered as an intrusion at the low and medium latac ranges by the driver, especially in the dry road conditions.

5.2.2 Control authority of Active Front steering

The control authority of active front steering system has been analysed by running the vehicle model on dry, wet and icy roads at 0.2g and 0.3g for the low latac, 0.4g, 0.5g and 0.6g for the medium latac and at 0.7g and 0.8g for the high latac operating ranges respectively. The control authority of AFS at the handling limits is also investigated. The steering angle inputs from the driver to produce these lateral accelerations calculated through simulations using step steer inputs in the earlier section are used. Then the resultant yaw rate, sideslip angle, latac (for verification) and the longitudinal vehicle speed are obtained through the full vehicle simulations for a “Sine with Dwell” steering input with and without AFS.

From the simulation results it can be seen that AFS improves the vehicle handling by reducing the peak yaw rate by 14% at 0.2g and by 8% at 0.3g latac on a dry road. Similarly a 17% reduction in the peak slip angle is obtained at 0.2g and a 15% reduction at 0.3g. One important observation concerning the activation of AFS controller is that due to improved tracking of yaw rate and reduction of side slip angle the exit speed is better compared to the passive vehicle. Also as the AFS system is less intrusive on the longitudinal dynamics of the vehicle, unlike the brake based ESC system. So the exit speed at the end of the manoeuvre is better at both 0.2g and 0.3g latac compared to the ESC activated condition. This is a much more preferable characteristic from a driver’s point of view, especially in the low lateral acceleration range which is not a safety critical operating range.

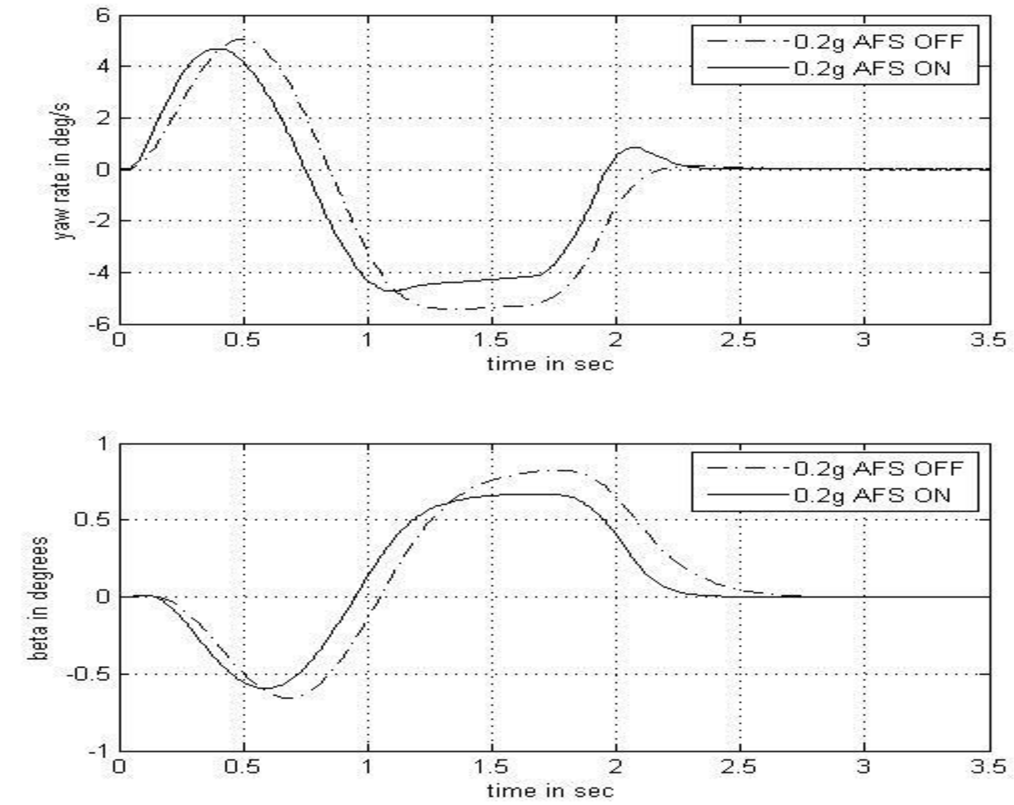

Fig 5.21: Control authority of AFS at 0.2g on dry road conditions

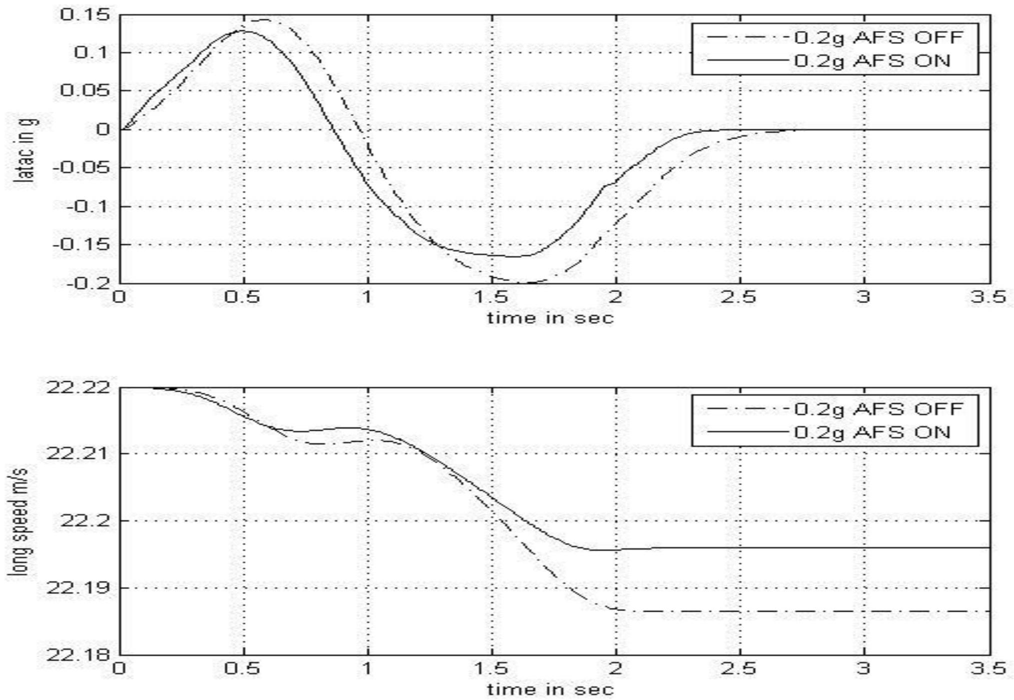

Fig 5.22: Influence of AFS on longitudinal dynamics at 0.2g

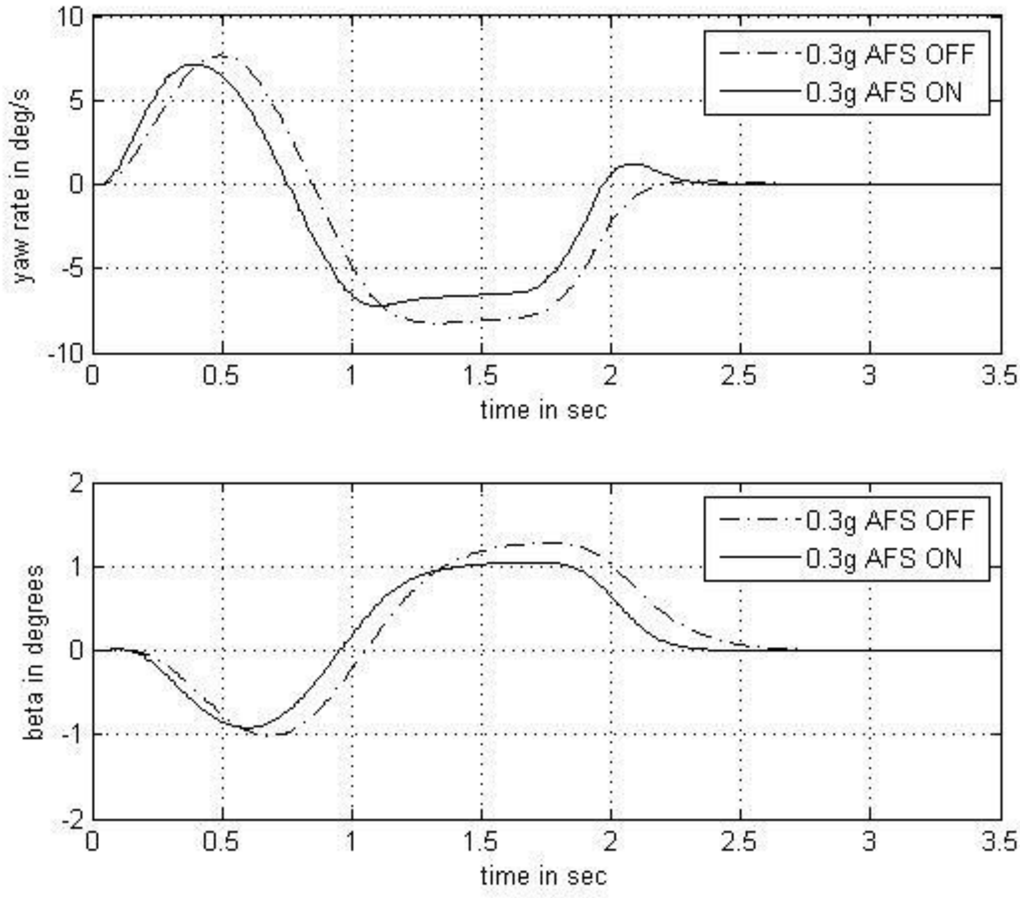

Fig 5.23: Control authority of AFS at 0.3g on dry road conditions

In the medium latac range, the AFS improves the vehicle handling by reducing the peak yaw rate by 17% at 0.4g, by 10% at 0.5g and by 7% at 0.6g latac on a dry road. Similarly a 17% reduction in the peak slip angle is obtained at 0.4g, 20% at 0.5g and a 20% reduction at 0.6g. Again the AFS does not affect the longitudinal vehicle speed at the end of the manoeuvre and the longitudinal vehicle speed is on a par with the passive vehicle at 0.4g and better by 0.5% at 0.5g. This highlights the non-intrusive nature of this control system in the longitudinal dynamics of the vehicle in the medium latac range as well.

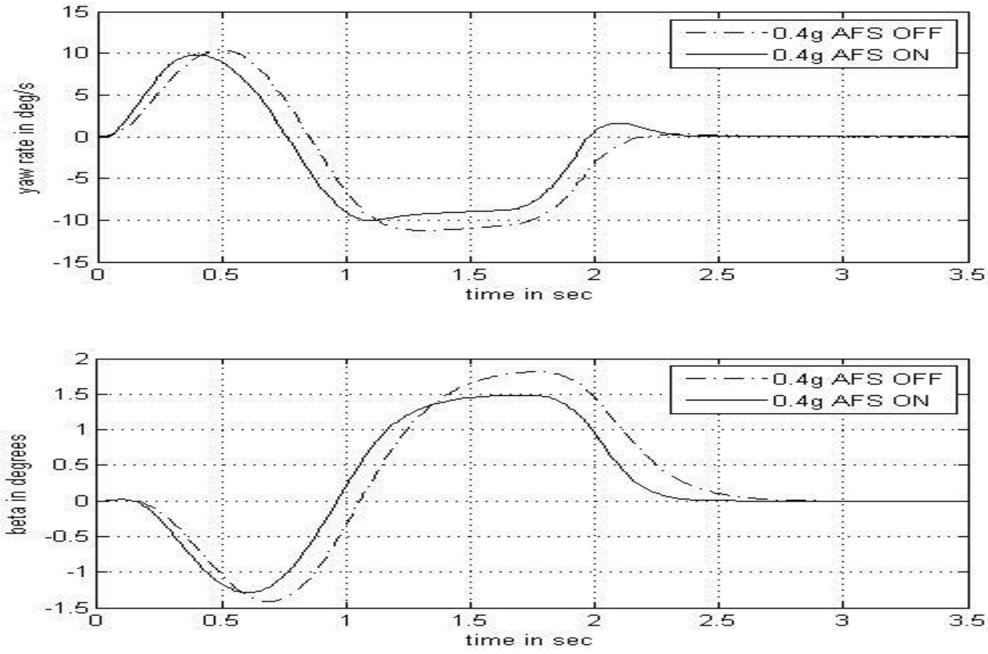

Fig 5.24: Control authority of AFS at 0.4g on dry road conditions

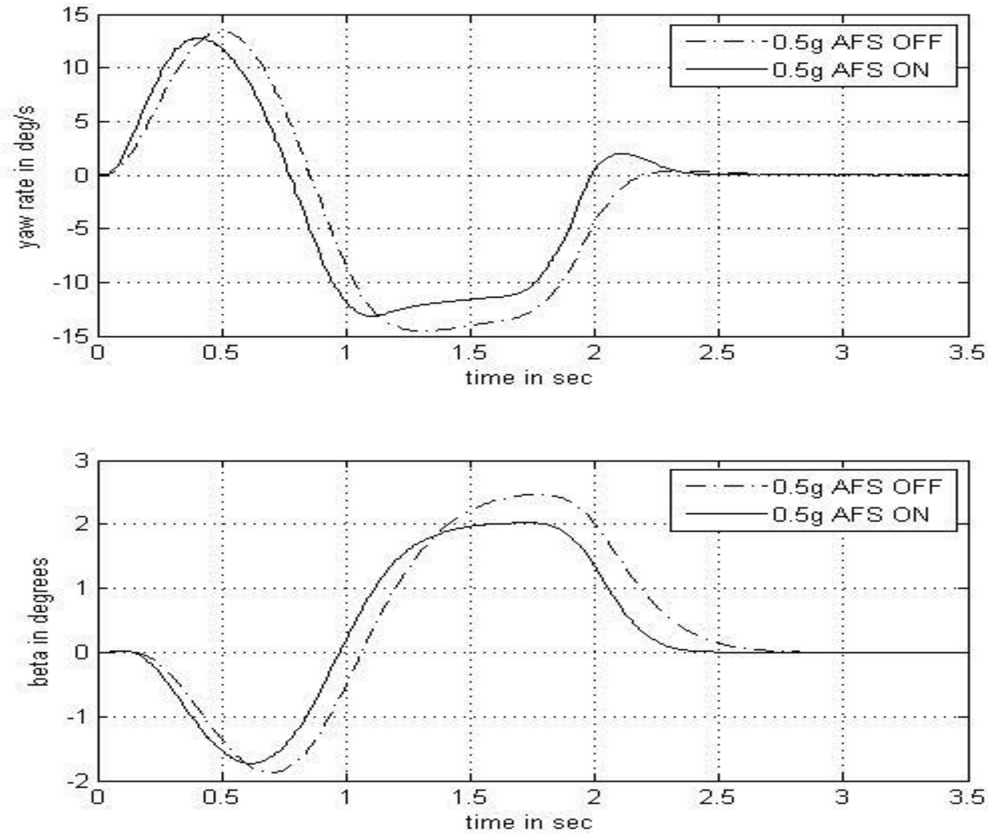

Fig 5.25: Control authority of AFS at 0.5g on dry road conditions

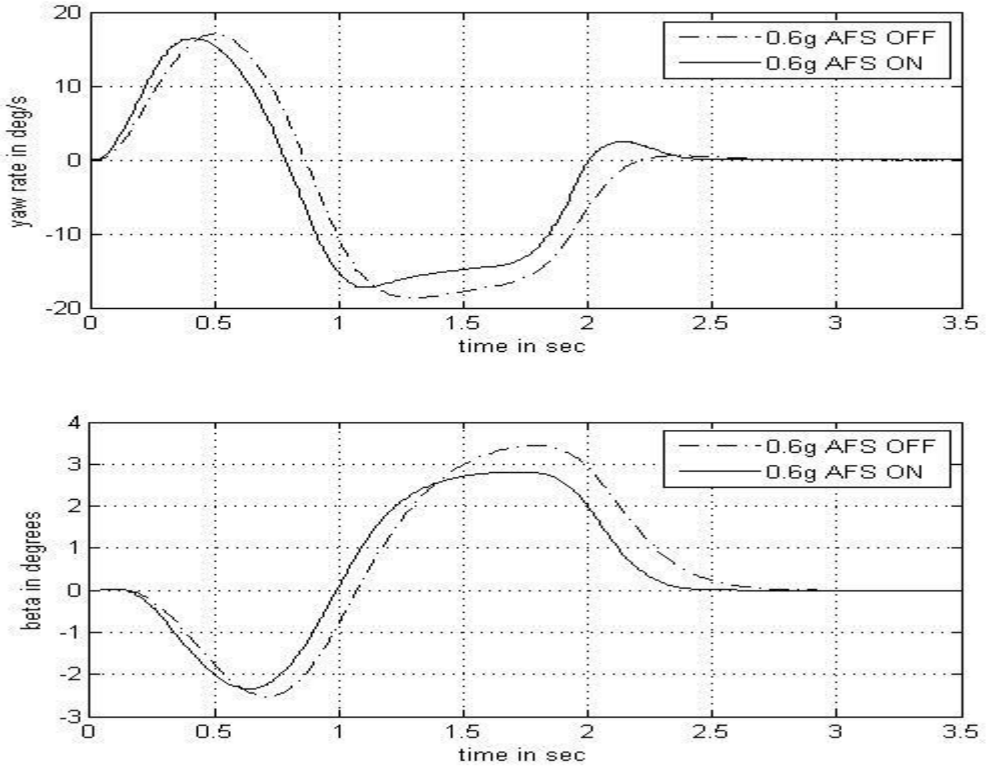

Fig 5.26: Control authority of AFS at 0.6 on dry road conditions

In the high latac range AFS improves the vehicle handling by reducing the peak yaw rate by 4% at 0.7g and by 4% at 0.8g latac on a dry road. Similarly a 18% reduction in the peak slip angle is obtained at 0.7g and a 25% reduction at 0.8g. Again the AFS does not affect the longitudinal vehicle speed at the end of the manoeuvre and the longitudinal vehicle speed is better by 0.8% at 0.7g and by 1.8% at 0.8g. This highlights the non-intrusive nature of this control system in the longitudinal dynamics of the vehicle in the high latac range .

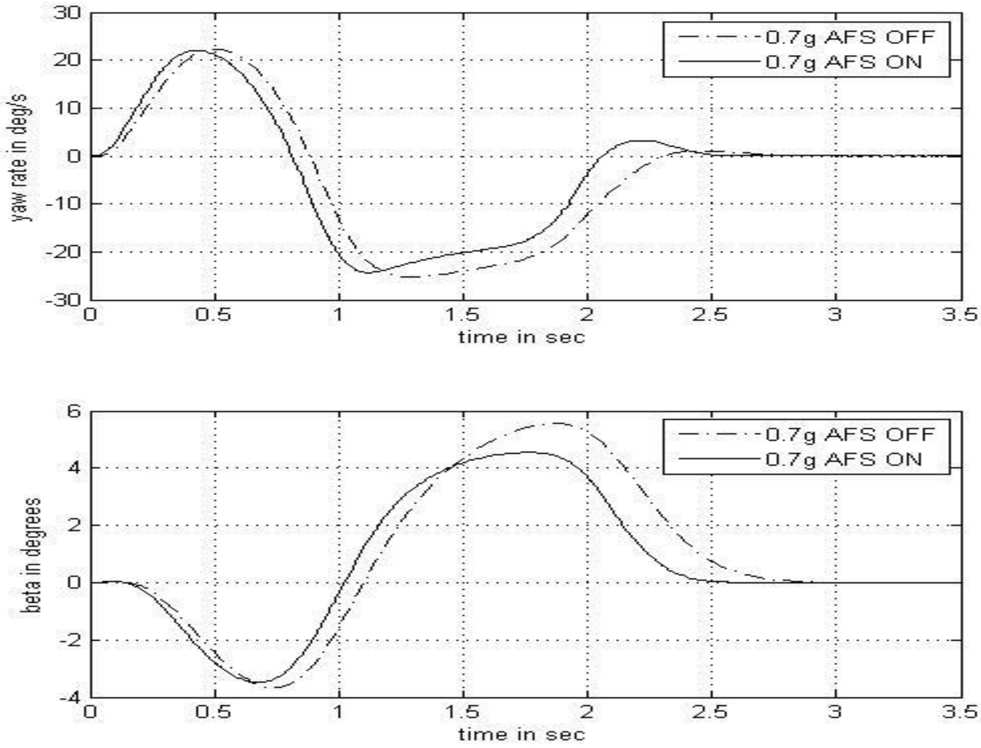

Fig 5.27: Control authority of AFS at 0.7 on dry road conditions

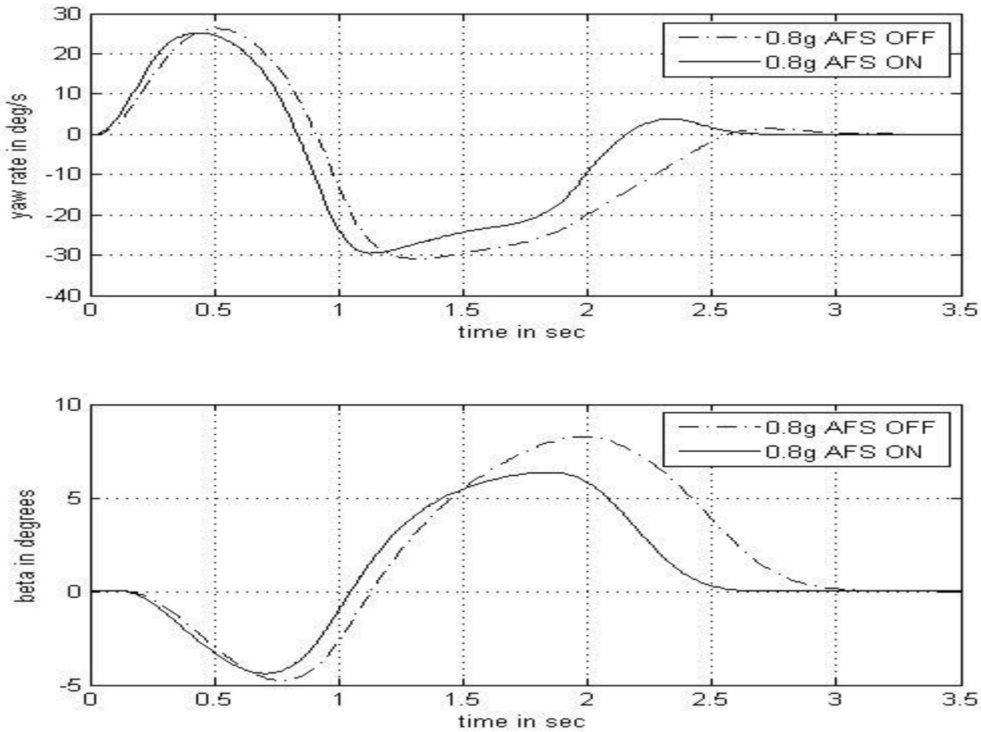

Fig 5.28: Control authority of AFS at 0.8g on dry road conditions

From the simulations it is observed that when the vehicle is being operated at its limits with the AFS deactivated then the vehicle yaw rate does not follow and respond to the steering input change and saturates to a maximum possible value. Also the vehicle side slip angle grows out of control and is unbounded. When the simulation is repeated with AFS activated, it can be observed that the AFS loses its control authority in improving vehicle handling. This proves that the AFS does not have an ability to influence the vehicle handling at the handling limits.

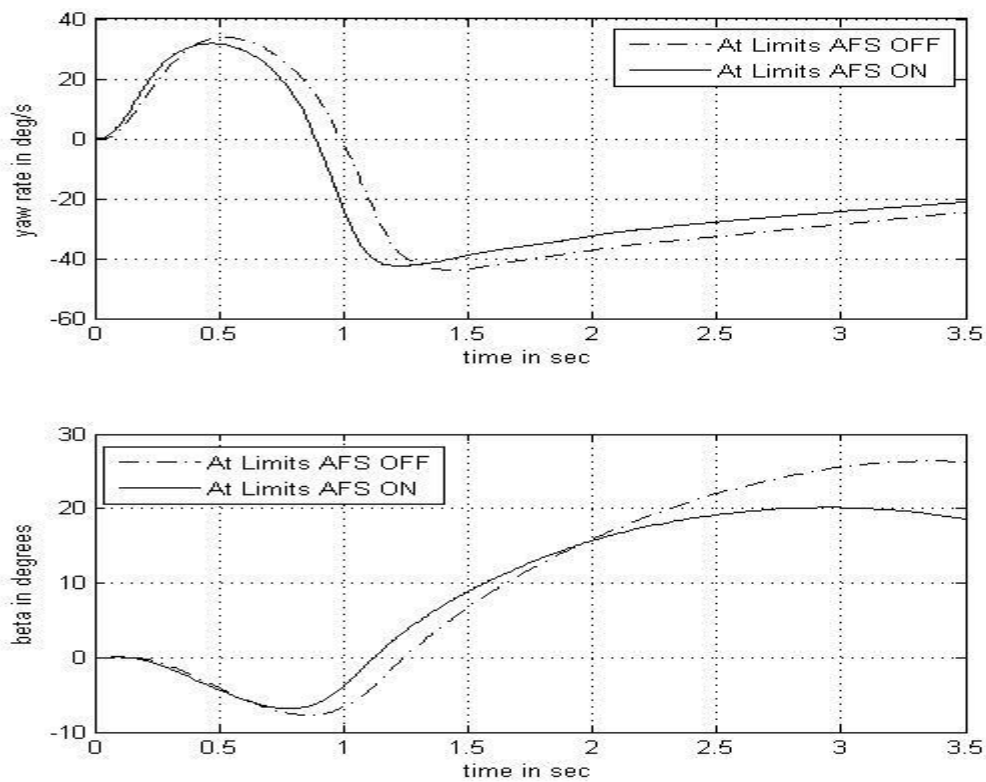

Fig 5.29: Control authority of AFS at the limits

From the above analysis, it is evident that the AFS has a good control authority in improving the vehicle handling by reducing the yaw rate and vehicle sideslip angle at the low and medium latac ranges. However its control authority starts to diminish in the high latac range and it does not have any control authority to make the vehicle yaw rate and side-slip angle to respond and follow the driver's steering input at the handling limits. Even though effective compared to the ESC system, the AFS does have a less control authority in reducing yaw rate and sideslip angle across the vehicle handling regions. However its less intrusive nature on the vehicle longitudinal dynamics makes it a preferable candidate over the low latac for better driving feel.

	Low Latac up to 0.3g			**Medium Latac 0.3g to 0.6g**			**High Latac above 0.6g**			**At the Limits**
	$\dot{\boldsymbol{\psi}}$	$\boldsymbol{\beta}$	$\mathbb{V}_{\boldsymbol{x}}$	$\dot{\boldsymbol{\psi}}$	$\boldsymbol{\beta}$	$\mathbb{V}_{\boldsymbol{x}}$	$\dot{\boldsymbol{\psi}}$	$\boldsymbol{\beta}$	$\mathbb{V}_{\boldsymbol{x}}$	$\boldsymbol{\psi}$ $\boldsymbol{\beta}$
Dry	✓	✓	1	✓	✓	1	×	×	1	×
Wet	✓	✓	1	✓	✓	1	×	×	1	×
Icy	✓	✓	1	✓	✓	1	×	×	1	×

Table 5.3: Summary of control authority of AFS over the vehicle latacs

5.2.3 Control authority of Variable Torque Distribution

The control authority of variable the torque distribution system has been analysed by running the vehicle model on dry, wet and icy roads at the low, medium and the high latac operating ranges respectively. The control authority of VTD at the handling limits is also investigated. The steering angle inputs from the driver to produce these lateral accelerations calculated through simulations using step steer inputs in the earlier sections are used. Then the resultant yaw rate, sideslip angle, latac (for verification) and the longitudinal vehicle speed are obtained through the full vehicle simulations for a "Sine with Dwell" steering input with and without VTD. From the simulation results it can be seen that VTD improves the vehicle handling by reducing the peak yaw rate by 12% and the peak slip angle by 28% in the low latac region on a dry road. In the medium latac range, the VTD improves the vehicle stability by reducing the peak yaw rate by 11% and the peak slip angle by 20%. A 6% peak yaw rate improvement and 37% peak side slip angle improvement is obtained with VTD against a passive vehicle. One important observation concerning the activation of the VTD controller is that due to addition of driving torque at the wheels to improve yaw rate tracking and stability the reduction in the exit speed is less compared to the ESC and AFS vehicles. Also the reduction in the exit speed at high latac region is much more pronounced than at the low and medium latac region, but still much better than the passive vehicle. Unlike the brake based ESC system, VTD does not intrude with the vehicle's longitudinal dynamics. This is also a much more preferable characteristic from a driver's point of view, especially in the low lateral acceleration range which is not a safety critical operating range.

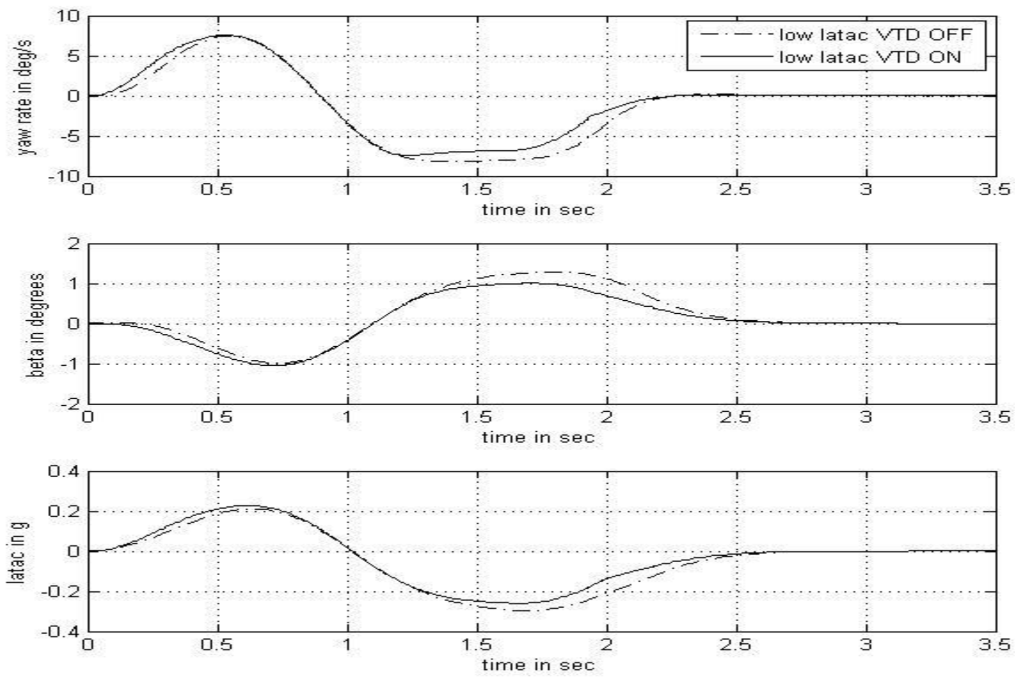

Fig 5.30: Control authority of VTD at low latac

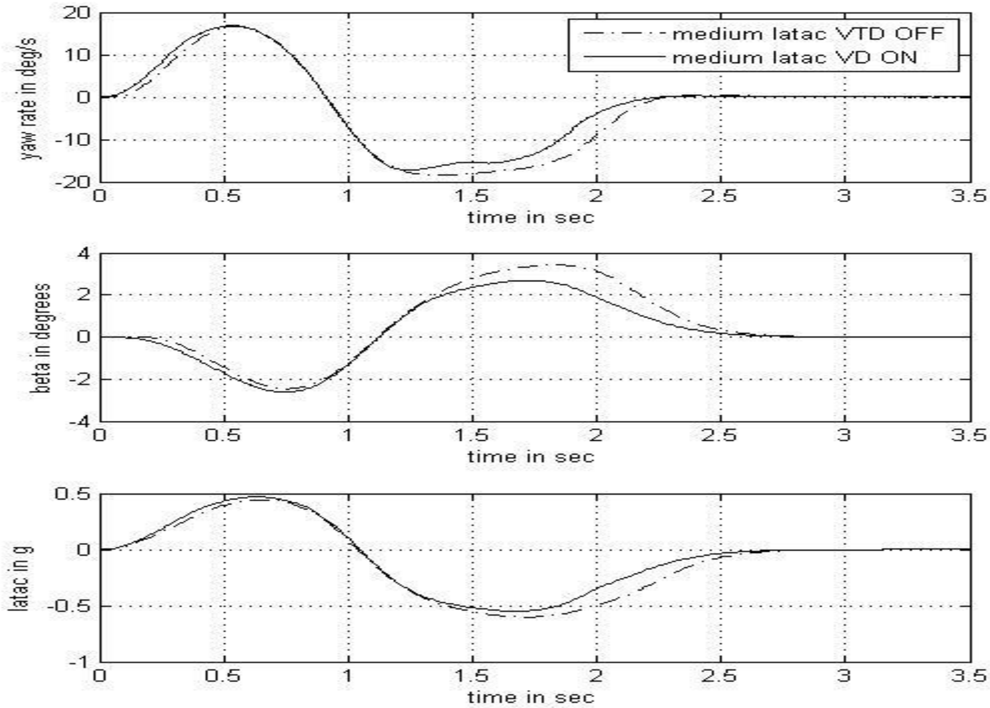

Fig 5.31: Control authority of VTD at medium latac

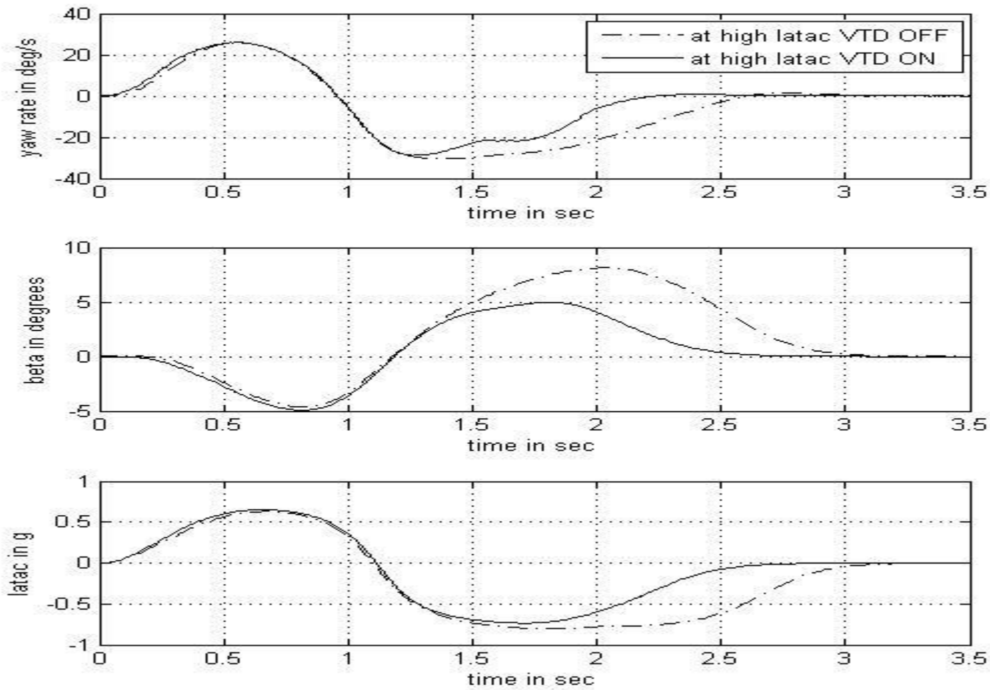

Fig 5.32: Control authority of VTD at high latac

From the simulations it is observed that when the vehicle is being operated at its limits with VTD ON, it can be observed that the VTD does lose the control and fails to track the vehicle steering angle input. Also the resultant yaw rate and the side slip angle are uncontrolled and much higher than the brake based electronic stability control system. So it is clear that the VTD's control authority diminishes as the vehicle moves towards its limits. This is mainly due to the addition of the drive torque to the vehicle, which makes the vehicle to operate at a higher latac or limit latac than a brake based electronic stability control, thereby increasing the sideslip angle and yaw rate of the vehicle.

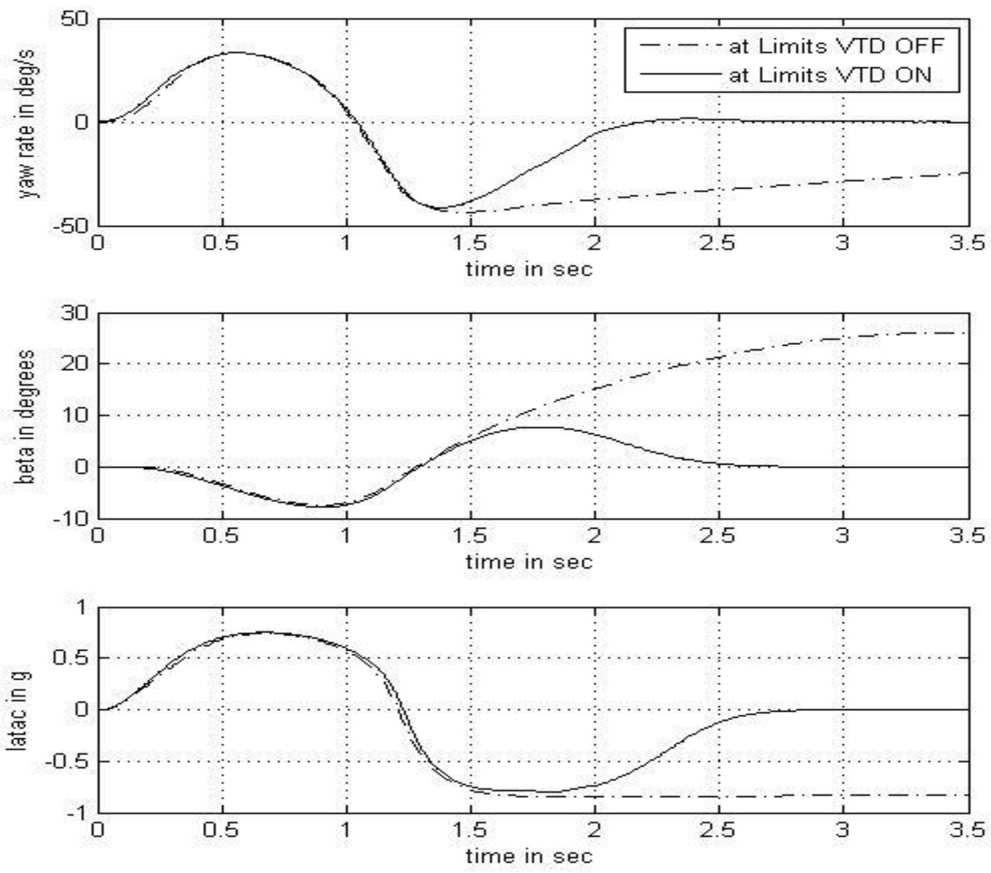

Fig 5.33: Control authority of VTD at the limits

5.2.4 Control authority of Normal force Control

The control authority of the suspension normal force control system on vehicle handling has been analysed by running the vehicle model on dry, wet and icy roads at the low, medium and the high latac operating ranges respectively. The control authority of NFC at the handling limits is also investigated. The steering angle inputs from the driver to produce these lateral accelerations calculated through simulations using step steer inputs in the earlier sections are used. Then the resultant yaw rate, sideslip angle, latac (for verification) and the longitudinal vehicle speed are obtained through the full vehicle simulations for a “Sine with Dwell” steering input with and without NFC.

From the simulation results it can be seen that when NFC is activated in the low latac region it does improve the vehicle handling by reducing the peak yaw rate and the peak slip angle, but the improvement is negligible. This is because the lateral load transfer between the outer and inner wheels is not very large during low latac. Also, the control strategy optimises the addition of suspension normal force as a function of the vehicle roll angle, which is reduced by the controller. However we can observe an improvement in this trend with more reduction in the peak yaw rate and the peak side slip angle as the vehicle moves into the medium latac zone. The superiority of the active system continues in the high latac range as well but with a deminishing effect on the control authority. At the limits we can see that the control authority of NFC vanishes and the vehicle behaves in a way much similar to the passive vehicle.

In all the three latac regions a good roll control is obtained, except at the limits. The main reason for this behaviour of the NFC system is that, at low latac, the tyre is operating at its linear region and hence producing lateral force as a function of the slip angle and the normal wheel load. Being operated at the same slip angle, between passive and active vehicles with little effect on lateral load transfer reduction by NFC system, the output lateral tyre force produced by the active system provides a negligible improvement in the reduction of yaw rate and slip angle. But at the medium latac zone, supported with a greater reduction in lateral load transfer, the NFC system produces better handling compared to a passive vehicle. Again, at the high latac, the trend continues, but a reduced efficiency due to the addition of more active suspension normal force results in a tyre normal load instability that affects the effective generation of lateral and longitudinal forces. This limits the extent / capacity of the normal suspension force actuator. So it is evident that the normal force control does have the capability to improve the vehicle stability at the medium latac but its control authority is limited and diminished at low and high latacs respectively. At the limits, the NFC ceases to display any ability to improve the vehicle handling compared to the passive vehicle.

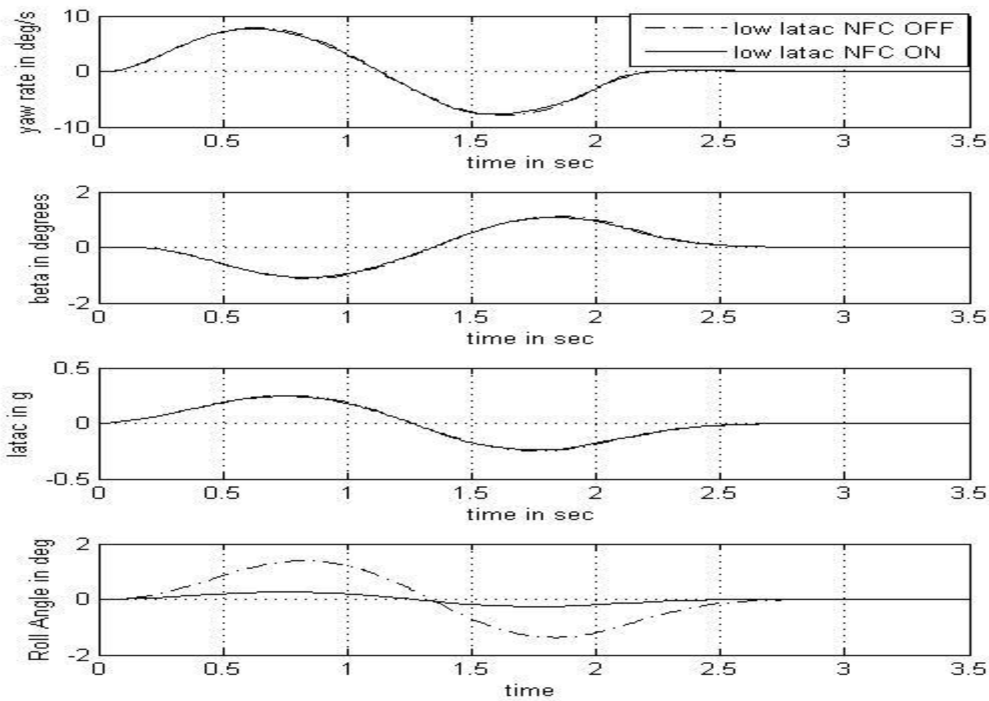

Fig 5.34: Control authority of NFC at low latac

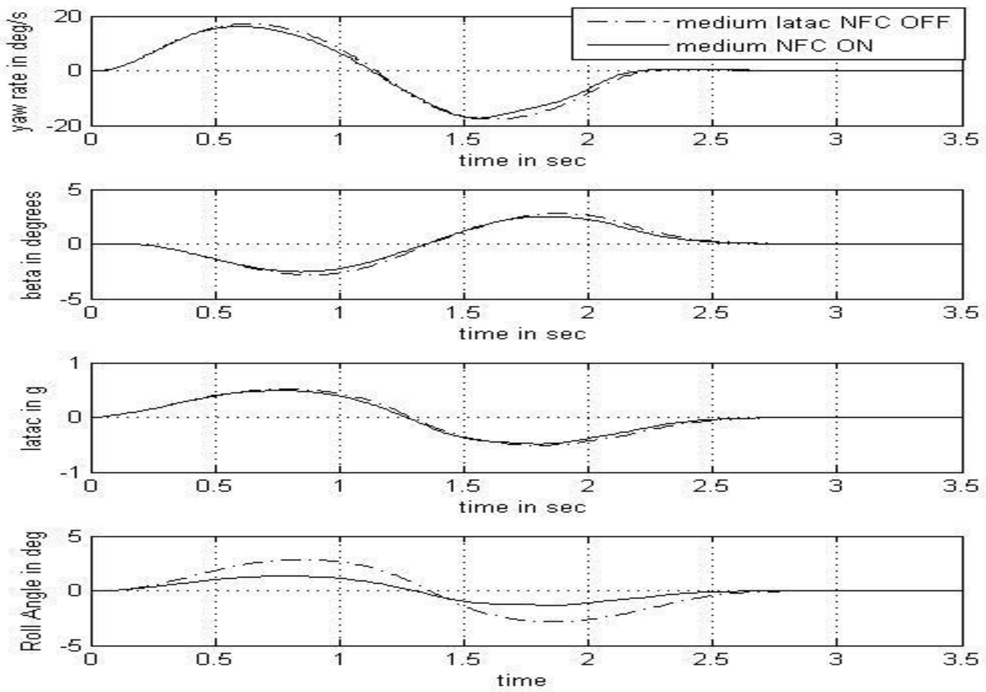

Fig 5.35: Control authority of NFC at medium latac

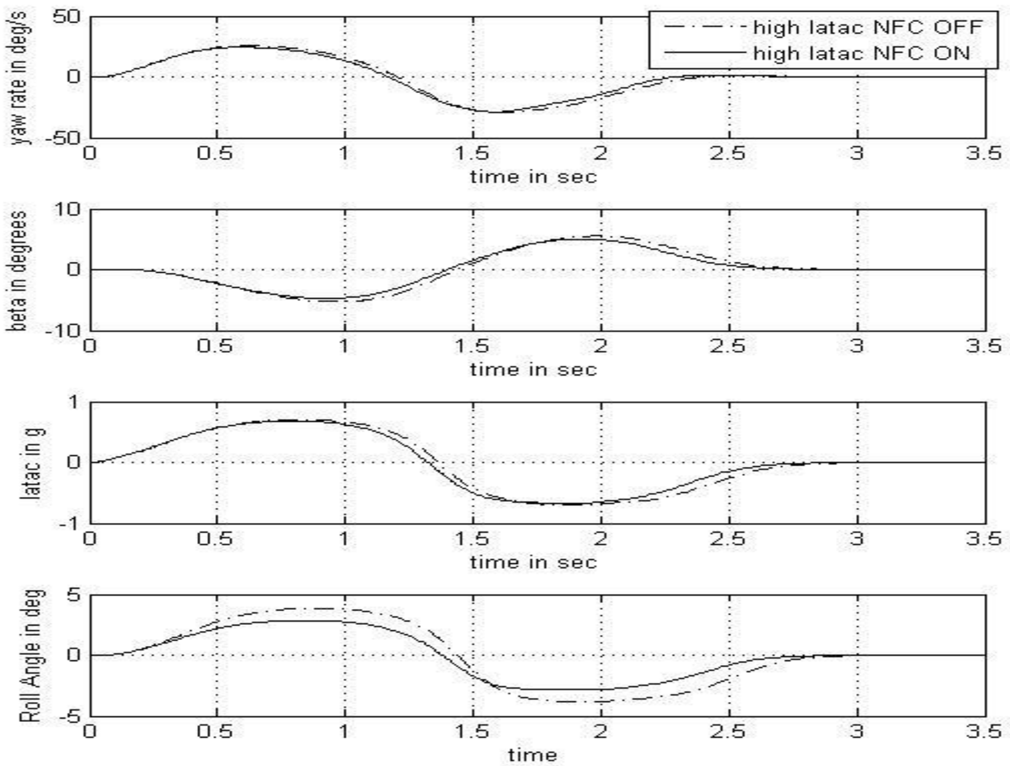

Fig 5.36: Control authority of NFC at high latac

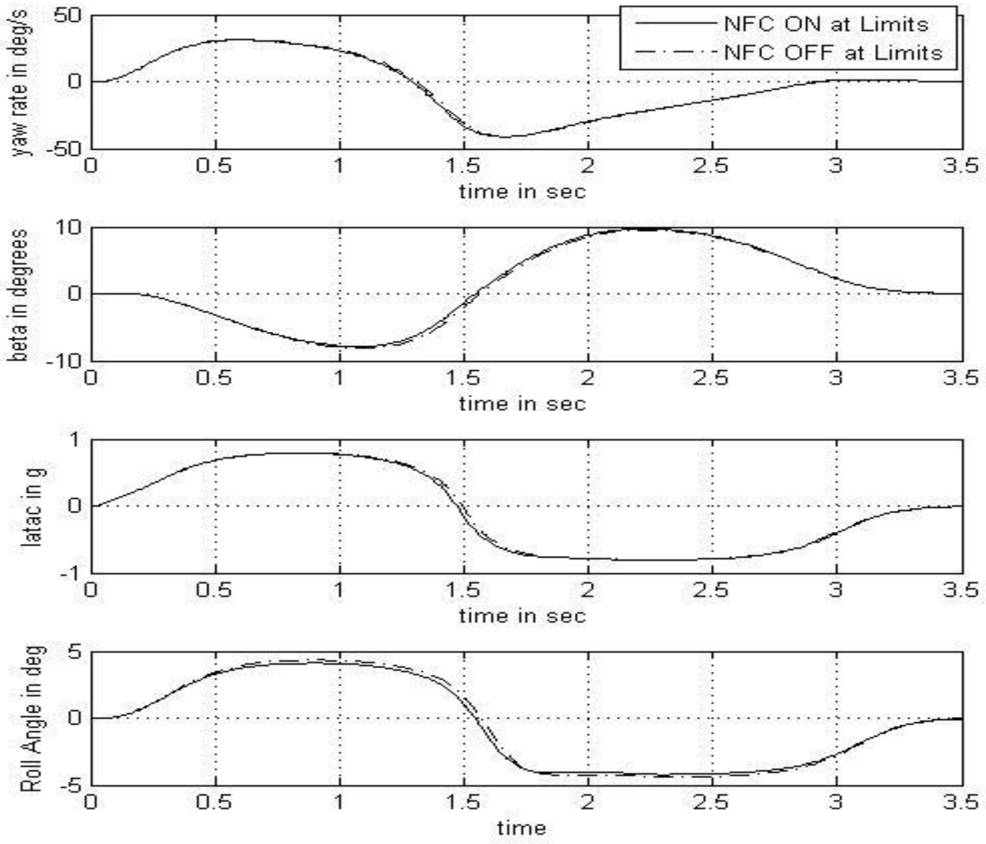

Fig 5.37: Control authority of NFC at the limits

5.3 Integration of ESC and AFS

Having investigated the individual behaviour and the control authorities of each of the four chassis control systems, the development of an integrated control strategy is carried out as follows. First the electronic stability control and the active front steering systems are activated individually and the vehicle yaw rate, sideslip angle, lateral acceleration and the longitudinal vehicle speed are recorded. Then both of these control systems are activated in standalone mode and the results were compared against that of the individual controllers.

From figure 5.38, when AFS and ESC are activated in a standalone manner, they reduce the yaw rate and the sideslip angle better than when they are activated individually. This shows that both the AFS and the ESC controllers complement each other in improving the vehicle handling performance.

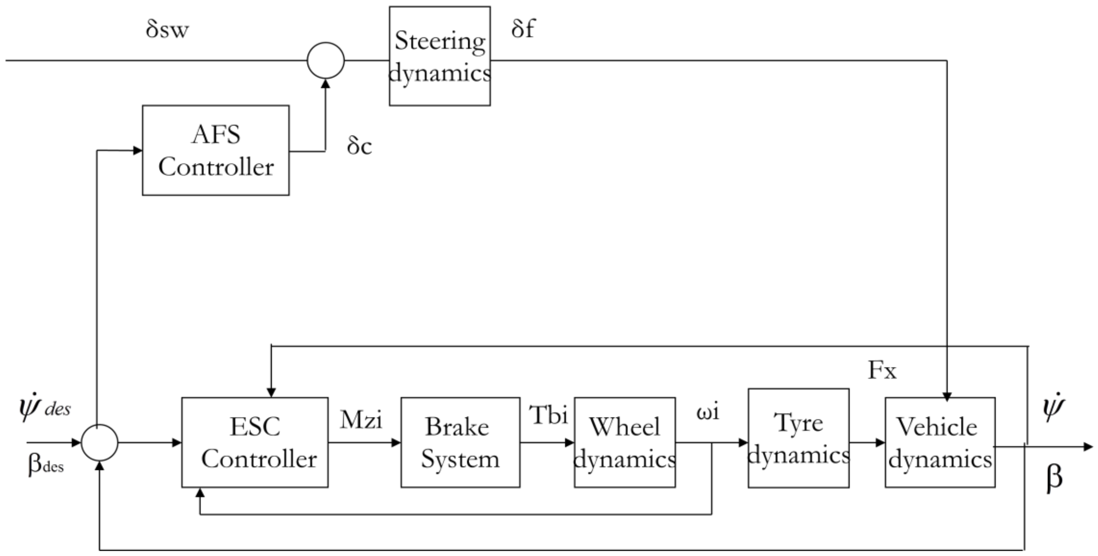

Fig 5.38: Schematic of AFS+ESC Standalone Controller

Compared to the ESC only activated scenario, the AFS and ESC standalone controller performs less intrusively in reducing the longitudinal vehicle speed and aiding a better driving feel. But AFS still dominates in providing the less safety critical low latac region of vehicle operation.

Again, both in the medium and high latac regions the AFS+ESC standalone controller performed better than the individual ones. The results are shown in figures 5.39 and 5.40 respectively.

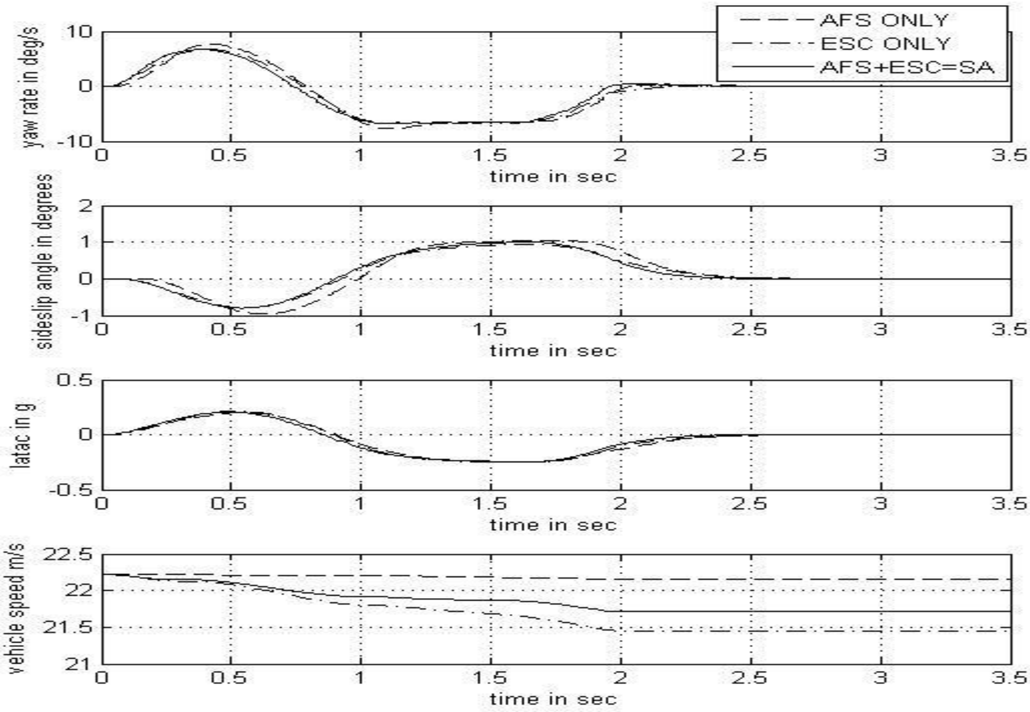

Fig 5.39: Low latac performance of AFS+ESC in Standalone Mode

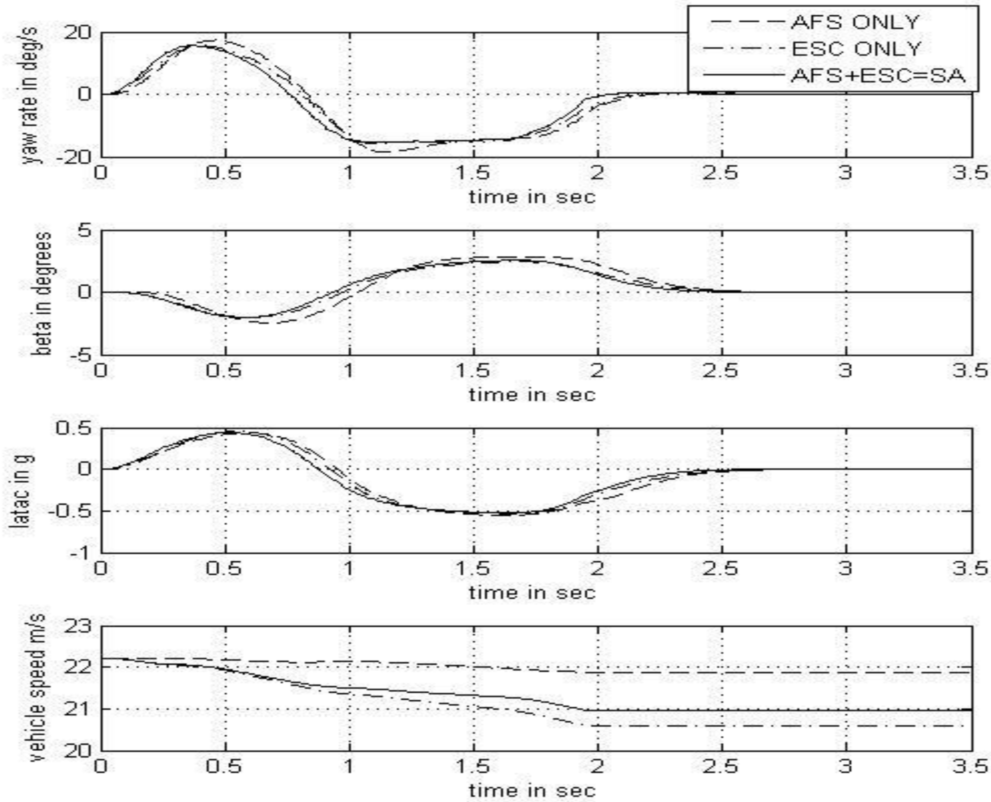

Fig 5.40: Medium latac performance of AFS+ESC in Standalone Mode

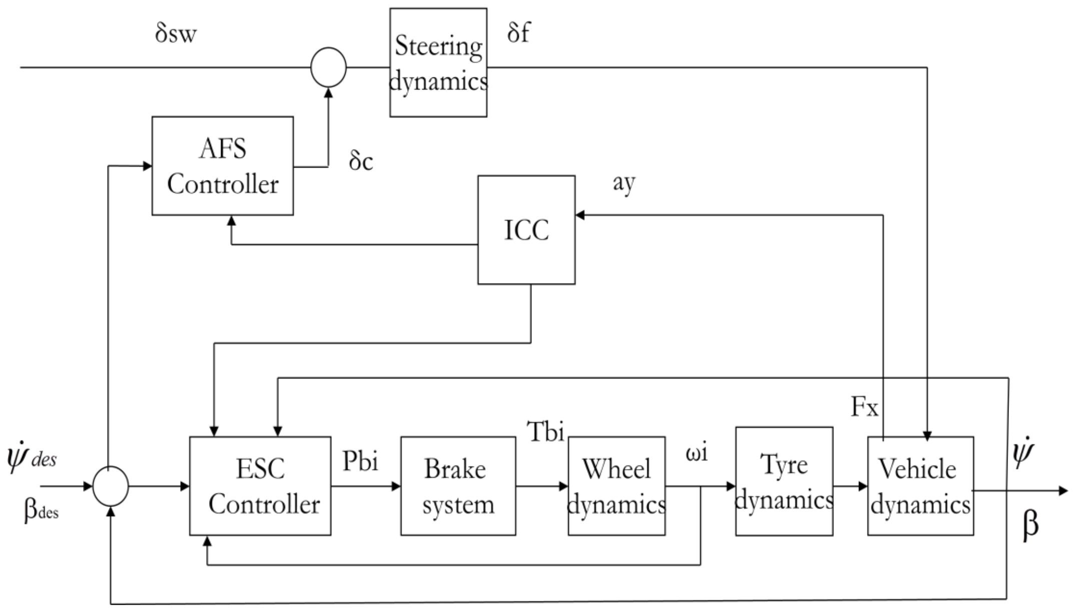

Fig 5.41: Schematic of AFS+ESC Integrated Controller (ICC)

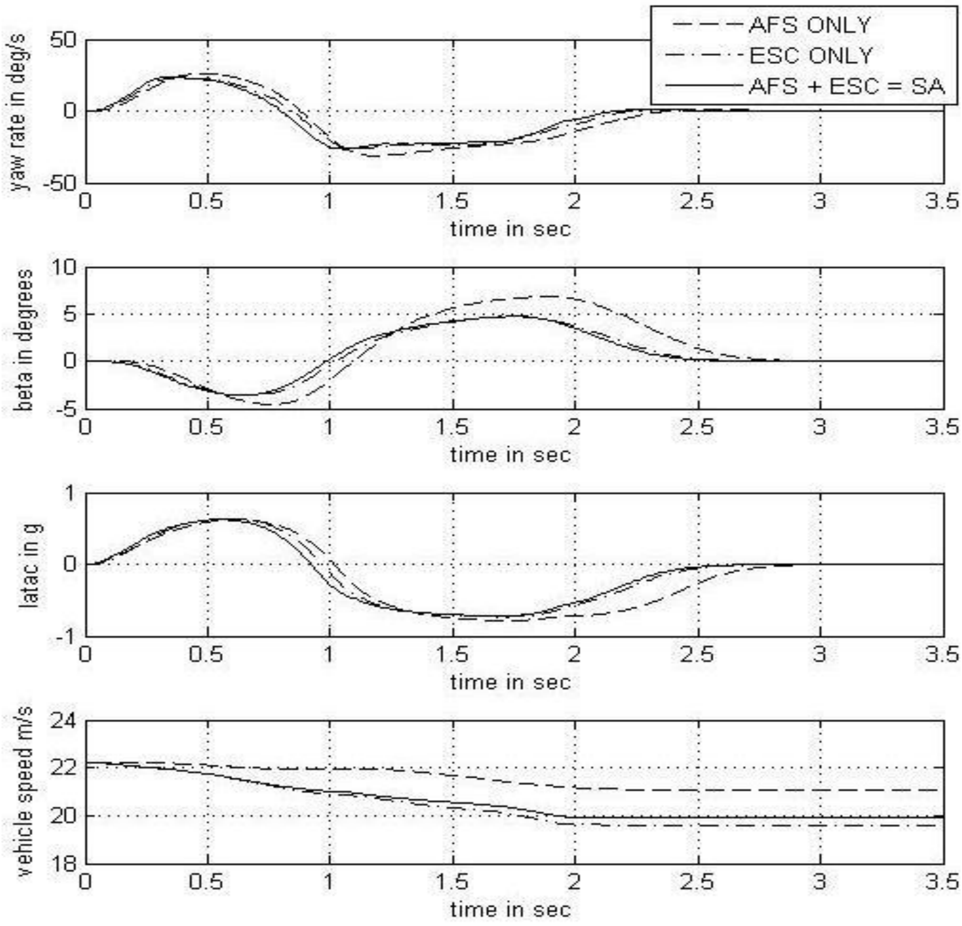

Fig 5.42: High latac performance of AFS+ESC in Standalone Mode

Following the above analysis of the AFS and ESC in standalone manner on low, medium and high latac regions, a rule based integrated chassis control (ICC) strategy is developed.

5.3.1 Rule based Integrated Control Strategy

Hence, in order to avoid undesirable interactions between the active front steering and electronic stability control subsystems and reduce performance trade-offs in vehicle handling, a novel rule based integration scheme is proposed to coordinate the control actions of the two stand-alone controllers. In light of the previous analysis of stand-alone active subsystems, the proposed integrated control system will be designed to achieve the following objectives:

- To improve vehicle steerability at low to mid-range lateral accelerations;
- To maintain vehicle stability close to and at the limit of handling;
- To minimize the influence of brake intervention on the longitudinal vehicle dynamics

This strategy needs to determine the activation sequences and active regions of the two stand-alone controllers in terms of the current vehicle operating point to avoid conflicts and to enhance the coexistence. It is therefore necessary to measure the vehicle operating point. The operating point of the vehicle ranges from normal driving to limit handling. A quantitative measure of this is the lateral acceleration of the vehicle. The relationship between the operating point and the lateral acceleration is a function of the road surface coefficient of friction. It is assumed that the road surface coefficient of friction can be measured or estimated. Hence lateral acceleration can be used as a measure of the operating vehicle point in the integration strategy.

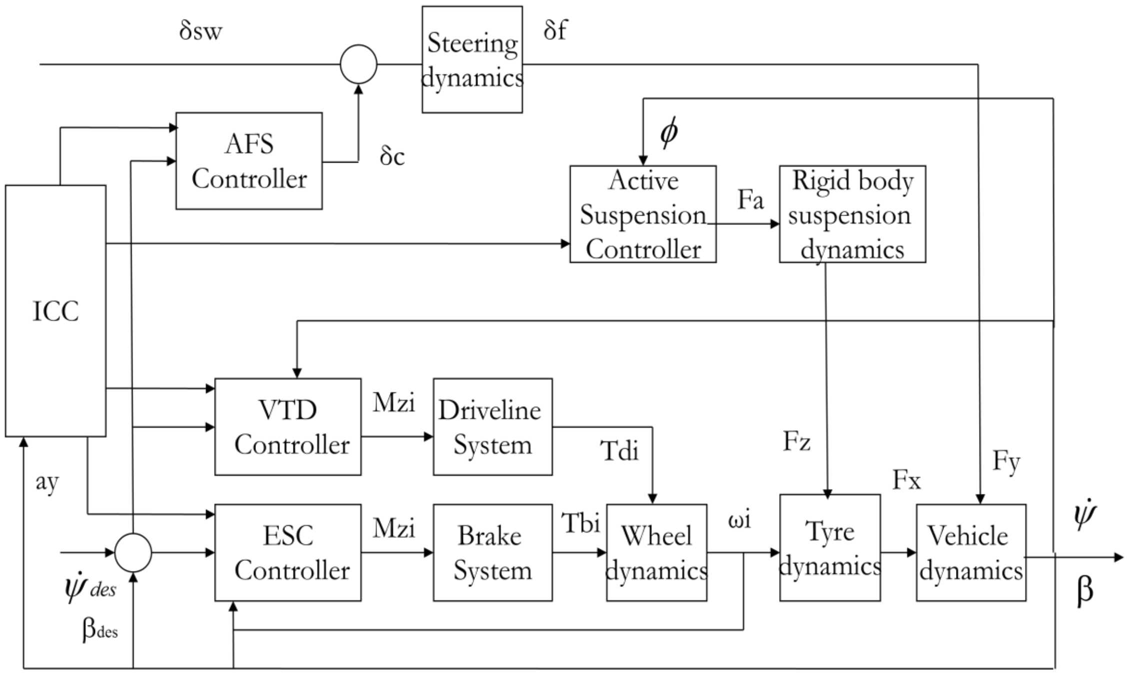

Fig 5.43: Schematic of the integrated Control Strategy

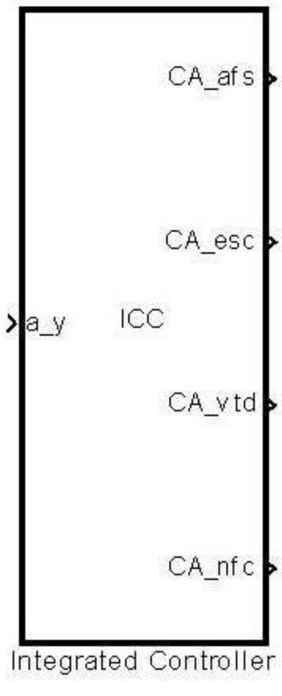

Fig 5.44: Block diagram of the rule based integrated controller

The developed integrated controller for AFS and ESC has one input and two outputs. The vehicle lateral acceleration is fedback to the integrated controller as the input and is used to determine the vehicle operating region. Having determined the vehicle operating region, the integrated controller allocates the vehicle dynamics control authority between the AFS and the ESC.

The rule based integrated controller activates the AFS in the low latac range until 0.3g and then handover the control authority to ESC. As the low latac range is within 0.3g, the ICC utilises the ability of the AFS to reduce the vehicle yaw rate and sideslip angle. At the same time, since the ESC is not activated, the ICC does not intrude in the vehicle longitudinal dynamics and aids a better driving feel.

From the figures 5.45 and 5.46, when the vehicle is operated in the medium and high latac regions, the integrated controller performs better than the standalone controller in improving the vehicle handling. Due to the deactivation of AFS and the intervention of ESC beyond the 0.3g latac, the exit speed of the manoeuvre is less than the standalone controller, but better than the ESC only system.

In summary, the integrated controller (AFS+ESC) performs on a par with the standalone system in the low latac and performs better than the standalone controller by reducing the vehicle yaw rate and sideslip angle at the medium and the high latac regions. The exit speed of the manoeuvre with ICC is less than the standalone controller, but better than the ESC only system.

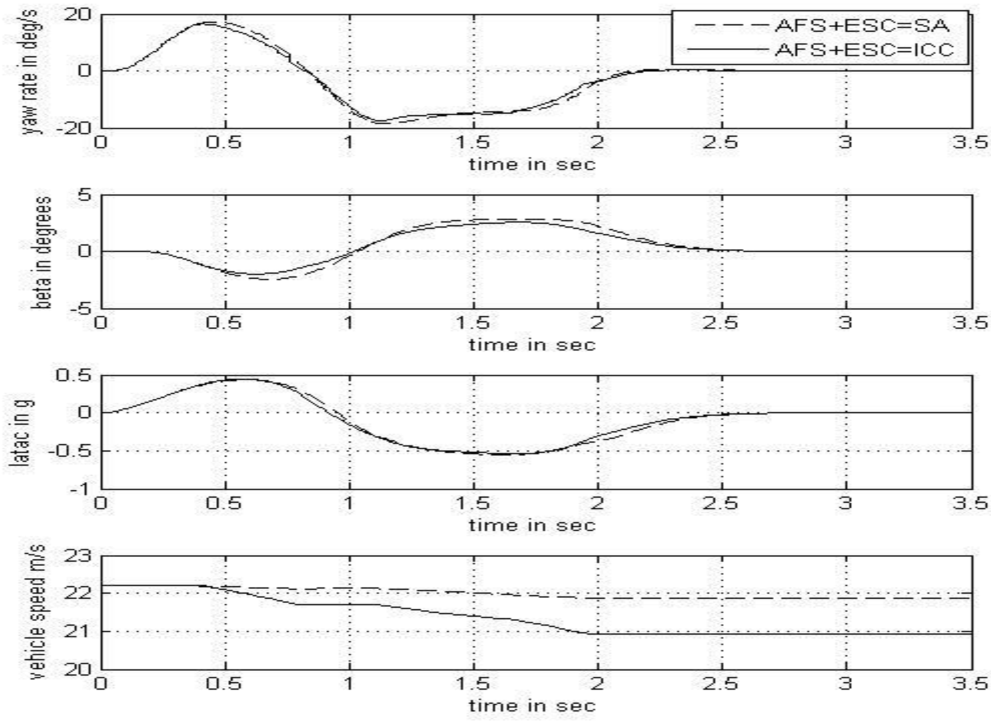

Fig 5.45: Performance of ICC (AFS+ESC) at medium latac

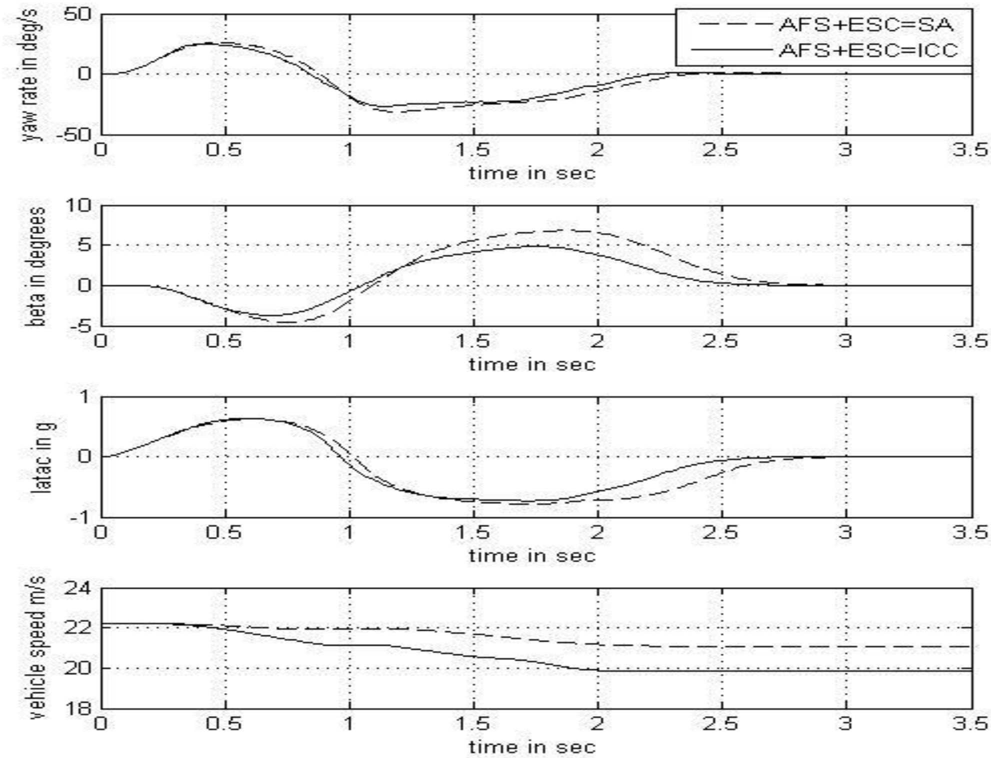

Fig 5.46: Performance of ICC (AFS+ESC) at high latac

5.4 Integration of ESC, AFS with VTD

Having integrated the AFS and the ESC systems, this section investigates the integration of VTD with the integrated controller developed in the previous section. From the standalone controller analysis in the earlier sections, the control authority of the AFS diminishes at the medium and the high latac regions and also less intrusive and providing driving fun at the less critical, low latac region. So the further integration strategy deactivates the AFS at the limits of low latac and considers the next two key stability control systems, VTD and ESC. Both VTD and ESC are effective in improving lateral handling of the vehicle at the medium latac zone, but the VTD limits the reduction in vehicle longitudinal speed compare to the more intrusive ESC. So the integrated control strategy activates only the AFS at the low latac and the VTD at medium latac. For the high and limit latac the ESC is activated.

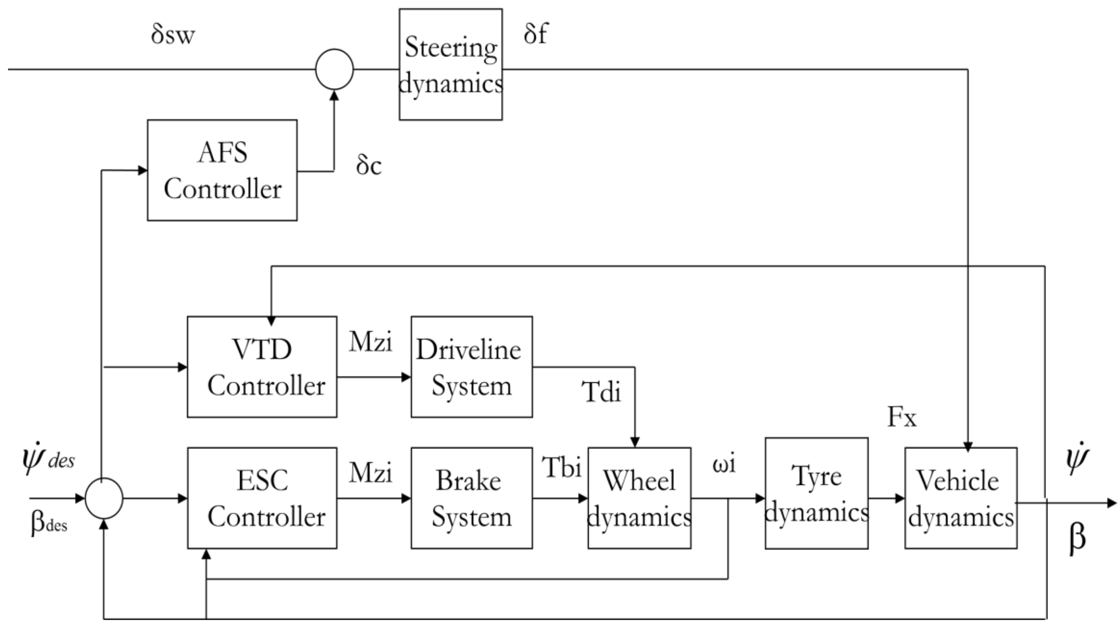

Fig 5.47: Schematic of AFS+ESC+VTD Standalone Controller

This integration strategy optimises the use of these three active chassis systems at the same time improves the vehicle handling without reducing the current vehicle performance, such as maintain or negligible effects of longitudinal vehicle speed. The figures 5.47 and 5.48 show the schematics of AFS, VTD and ESC controllers in standalone and integrated modes.

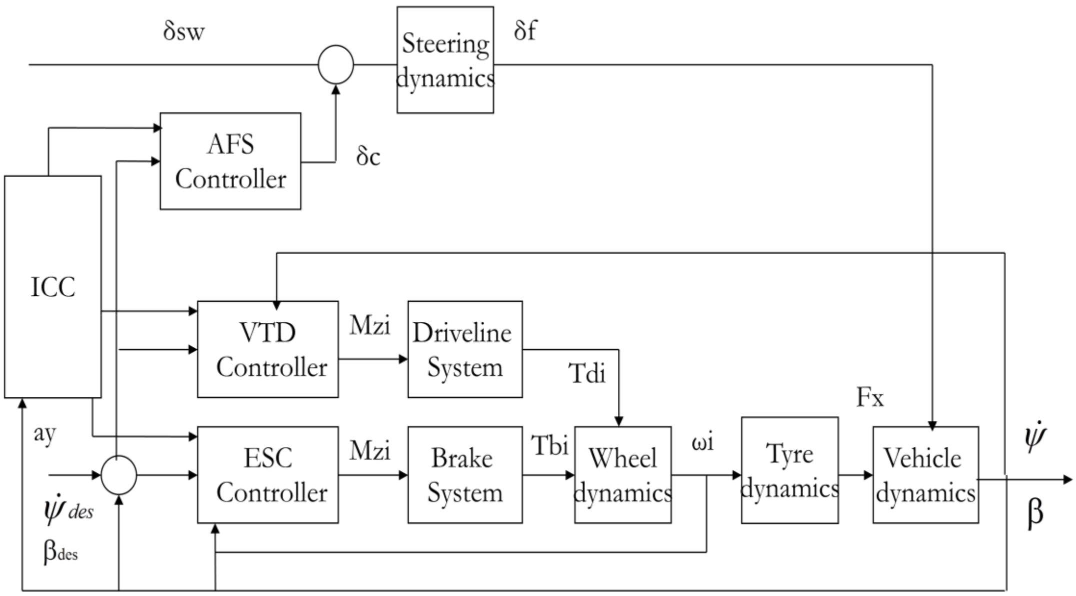

Fig 5.48: Schematic of AFS+ESC+VTD Integrated Controller (ICC)

The rule based integrated controller is enhanced to accommodate the necessary extra rules to integrate the VTD system to the existing integrated controller. From the figures 5.49 and 5.50, when the vehicle is operated in the medium and high latac regions, the integrated controller performs better than the standalone controller in improving the vehicle handling. Due to the activation of VTD and the deactivation of ESC in the medium latac zone of 0.3g to 0.6g, the exit speed of the manoeuvre is better than the AFS+ESC ONLY integrated controller system. In summary, the integrated controller (AFS+VTD+ESC) performs at par with the (AFS+ESC) integrated control system in the low, medium and high latacs and performs better than the standalone controller across the all latac regions. The exit speed of the manoeuvre with ICC is better in the medium latac range due to the activation of VTD.

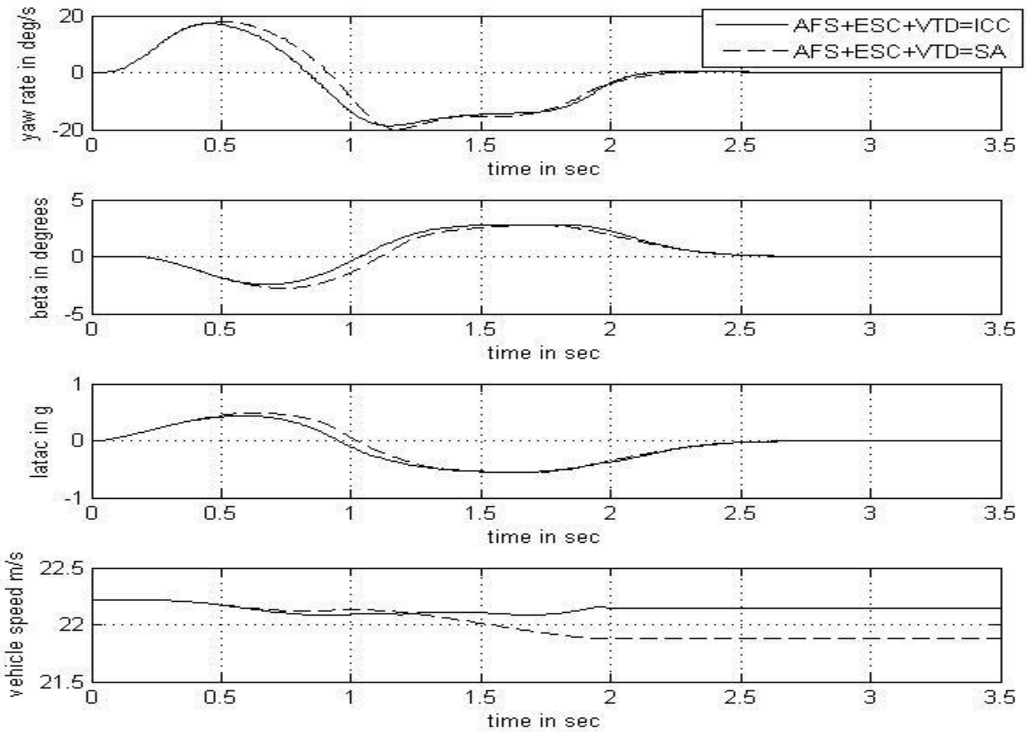

Fig 5.49: Performance of ICC (AFS+ESC+VTD) at medium latac

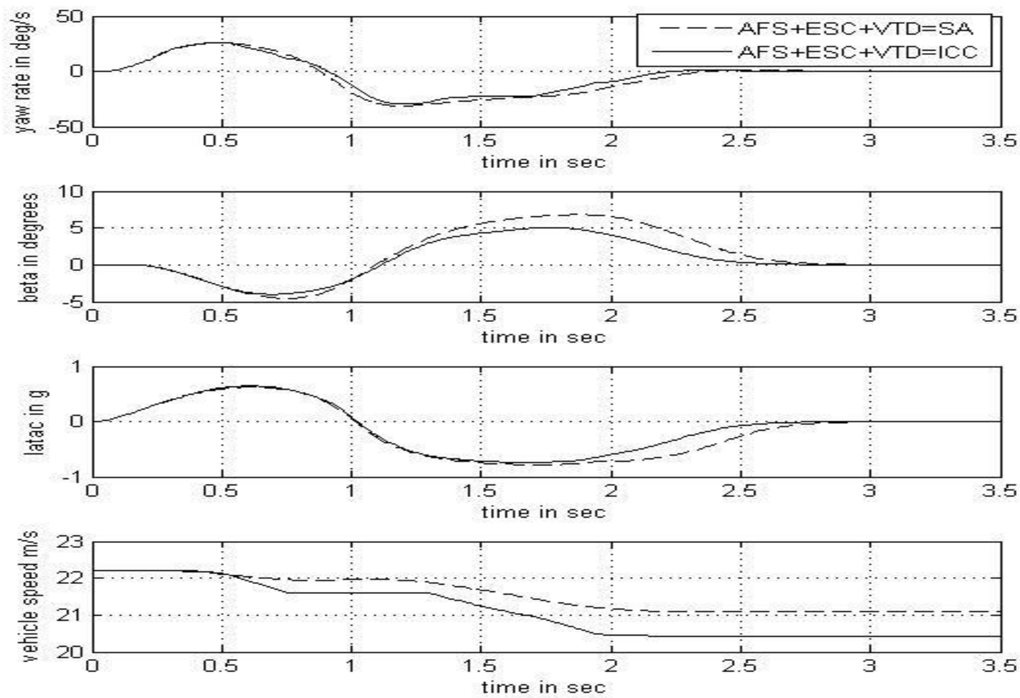

Fig 5.50: Performance of ICC (AFS+ESC+VTD) at high latac

5.5 Integration of ESC, AFS, VTD with NFC

From the individual chassis controller analysis the NFC controller has little or no effect at the low latac and a moderate effect on improving the vehicle handling in the medium latac region. Its ability to generate the extra tyre forces depends mainly on the amount of lateral and longitudinal load transfer. When NFC was activated in the previous chapter and in the earlier sections of this chapter, only the steering input was given to the vehicle. Hence the additional normal force on the wheels influenced only the lateral tyre forces. That too when these forces saturate then the additional load by NFC has little or no effect. But in the fully integrated controller mode, the corrective yaw moment is generated by the VTD and ESC in addition to the AFS. The effect of NFC on the longitudinal forces will add more influence on generating the corrective yaw moment. A schematic diagram of the AFS, ESC, VTD and NFC controllers in standalone manner is given on figure 5.51.

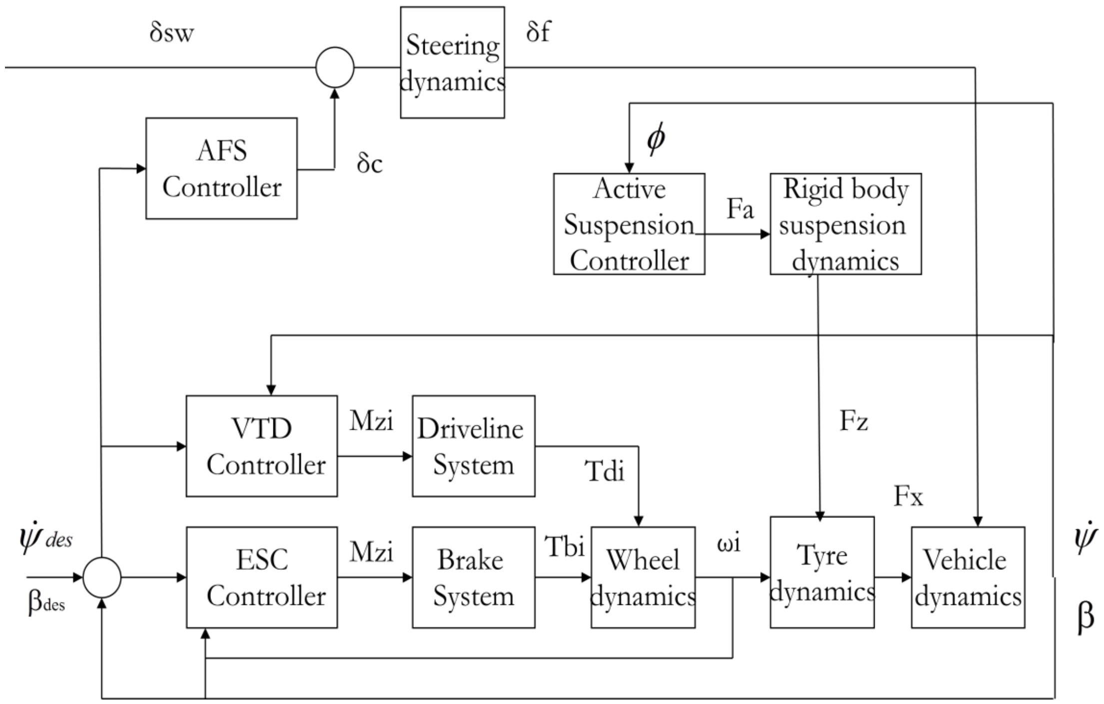

Fig 5.51: Schematic of AFS+ESC+VTD+NFC Standalone Controller

A further enhancement is made to the rule based integrated to accommodate the necessary rules to integrate the NFC system to the existing integrated controller. This fully integrated chassis controller (ICC), integrates the electronic stability control (ESC), active front steering (AFS), variable torque distribution (VTD) and suspension normal force control (NFC). This rule based ICC strategy provides the control authority to AFS at the low latac range, to VTD at medium latac range, to ESC at high and at limits and activates the NFC from medium latac onwards to optimise the generation of lateral and longitudinal tyre forces and to use the four active chassis systems effectively.

A schematic of the novel four systems ICC control strategy is given in figure 5.52.

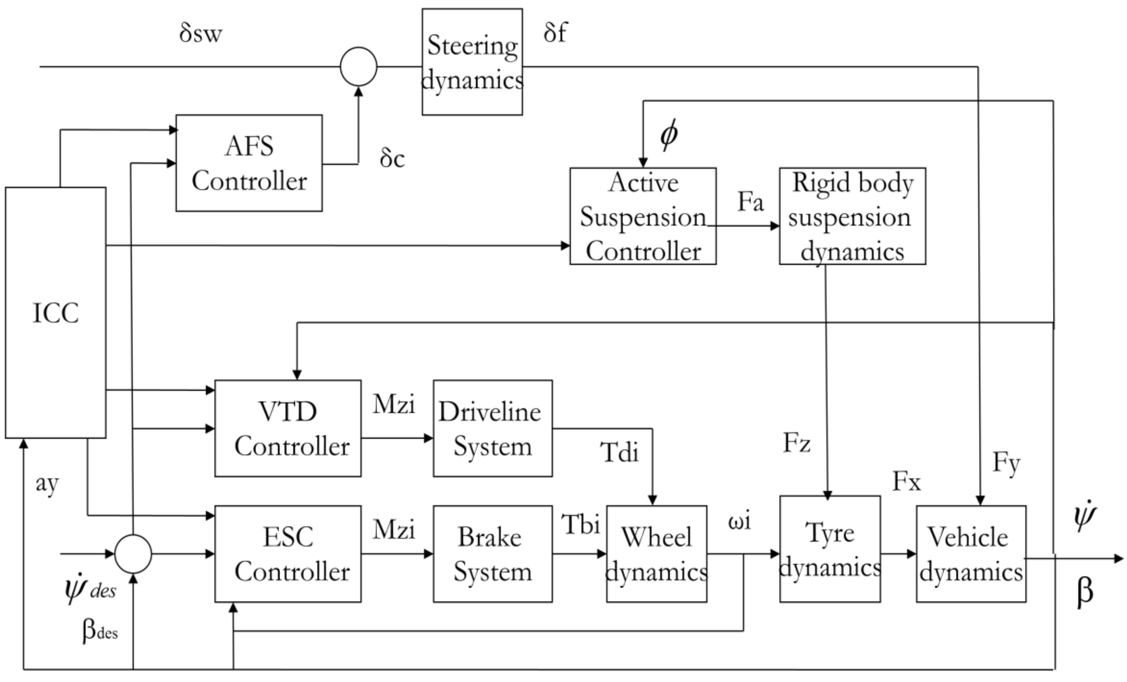

Fig 5.52: Schematic of AFS+ESC+VTD+NFC Integrated Controller (ICC)

From the figures 5.53 and 5.54, when the vehicle is operated in the medium and high latac regions, the integrated controller performs better than the standalone controller in improving the vehicle handling. In summary, the integrated controller (AFS+VTD+ESC+NFC) performs at par with the (AFS+ESC+VTD) integrated control system in the low latac region and performs better in the medium to high latac and at the limits.

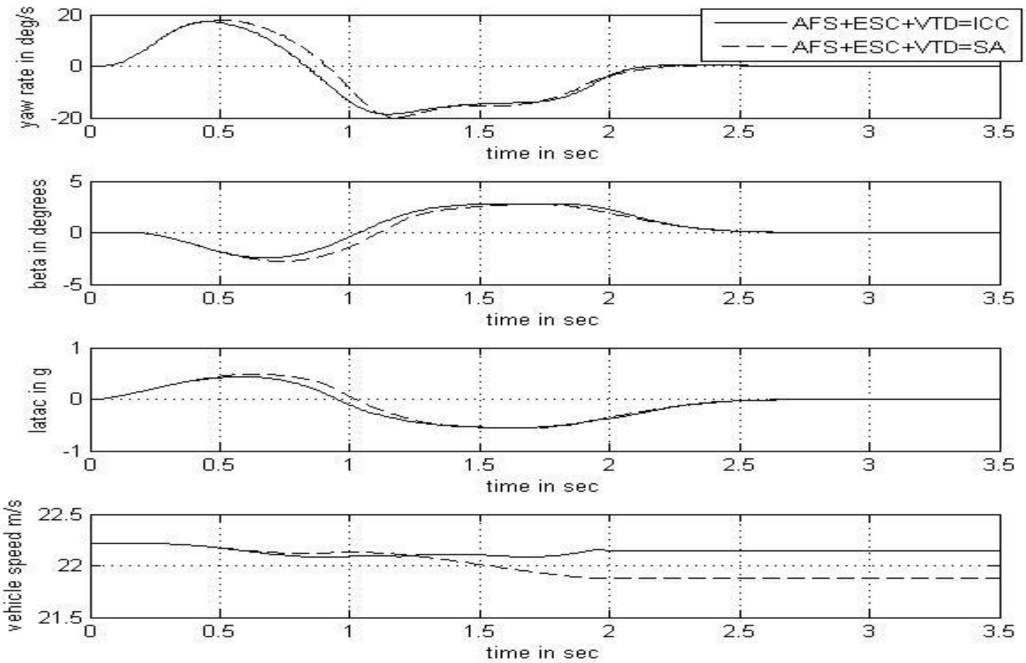

Fig 5.53: Performance of ICC (AFS+ESC+VTD+NFC) at medium latac

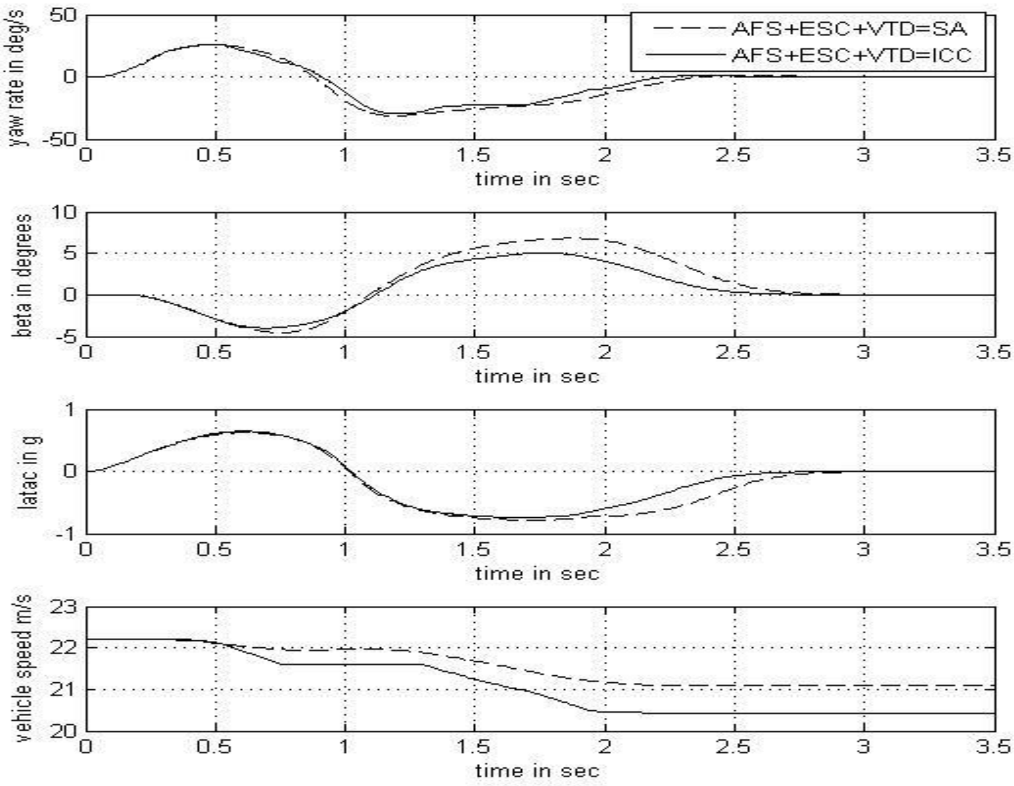

Fig 5.54: Performance of ICC (AFS+ESC+VTD+NFC) at high latac

5.6 Summary

This chapter discussed the integration of four active chassis control systems developed in the previous chapter to improve the current vehicle handling dynamics performance. It started with the analysis of individual active chassis control systems and established their control authorities on vehicle handling dynamics. Then it discussed the development of a rule based integrated chassis controller by starting the integration of electronic stability control and active front steering. After the successful integration of these two systems, the variable torque distribution system was integrated to further augment the handling performance. Finally the normal suspension force control system is added to produce this research goal of a fully integrated chassis controller.

Chapter 6

Conclusions and Recommendations

6.1. Results Summary and Conclusions

With reference to the aims and objectives of this thesis the following are achieved:

- Region of effectiveness or control authority of electronic stability control, active front steering, variable torque distribution are identified.
- Conditions of co-existence and avoiding potential conflicts among them are derived.
- Improvement in the vehicle dynamics behaviour through integration of the four active systems is achieved.
- A detailed non-linear vehicle model with all its necessary functional systems is developed to simulate the passive vehicle dynamics.
- A brief but useful study on the modelling principles of various tyre models used in the industry is conducted. A pacejka tyre model to represent the behaviour of tyres during the combined longitudinal and lateral slip conditions is developed.
- Development of a Matlab/Simulink based automotive toolbox with all the above mentioned mathematical models of vehicle systems is achieved.
- Detailed models of anti-lock brake system, electronic stability control, active front steering, traction control system, variable torque distribution and suspension are developed using simple fuzzy logic and PID control techniques.

- The non-linear passive vehicle dynamics model developed is validated against commercially available vehicle dynamics software, called CarSim.
- A detailed literature review about the four active chassis control systems was conducted to understand the different modelling and control strategies used to simulate these systems. Detailed models of hydraulic brakes, steering and suspension actuators are modelled. A simple first order dynamics is incorporated for variable torque distribution.
- The developed active systems are evaluated against their passive counterparts through whole vehicle simulations. The results prove that the active systems are effective in reducing the yaw-rate and side-slip angle against their passive counterparts.
- The control authority or regions of effectiveness for each of the four standalone chassis systems are identified and their performance boundaries are defined.
- The integration process is started with an analysis of two standalone active chassis systems (electronic stability control and active front steering), in a combined manner. From the simulation results, the conditions of coexistence and conflicts between them are understood.
- A novel rule based integrated control strategy is developed to make these two combined systems to functionally co-exist on a same vehicle without any conflicts in the performance on their own and the vehicle as a whole.
- A variable torque distribution system (VTD) is incorporated to augment the function of integrated chassis control system.

- The integration process was completed with the addition of a normal suspension force control (NFC) system on to the previously integrated system.
- The final rules of integration for these four systems are presented and proved that the final integrated controller is better in reducing yaw rate and sideslip angle compared to the standalone and combined systems.

6.2. Recommendations for future work

With reference to the reviewed literature the thesis has proved the possibility of integrating four active chassis systems, one from each key vehicle function. However due to some individual and academic reasons pertaining to the author certain details during the research were not considered in order to focus more on the thesis aims and objectives. Recommending them might form a possible venue for a future research that could make use of the automotive toolbox developed, which might give a head start to focus more on new objectives.

During the course of this research it was established that the four active systems from different vehicle function have the potential for integration. Each of these four functions have many active systems on their own to improve vehicle performance. For example, active steering control objective can be realised through many methods such as active front steering, active rear steering, four wheel steering etc. Similarly, active suspension employs various control strategies, such as continuously variable damping control, active roll control, roll moment distribution etc...A research literature is to be found that explains the possibility of integrating all the possible active systems within a vehicle function to provide comfort and handling. A feasibility study may be conducted to establish the potential of this research.

The future of active chassis control technology is bright and the number of electronic systems on modern vehicles is growing exponentially. Having analysed the possibility of integrating various vehicle dynamics control systems, researchers might move out of the vehicle dynamics domain and may look into the possibility of integrating with vehicle electronic systems such as communication and navigational systems, and even one step further to other vehicle systems to improve the group vehicle dynamics behaviour of a group of vehicles on highway.

Integration of all key vehicle systems may open a door to autonomous driving system such as auto-pilots in aeroplanes. Research to find out the rules of engagement between braking, steering, suspension and power train would be challenging under various driving conditions. However prior to that, another important element of any vehicle system is the driver. Starting to integrate the driver more into the function of vehicle systems may help to develop the knowledge required for integrating vehicle systems for autonomous driving.

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Appendix A

Brush Model Equations:

$$\frac{l_c}{l_t} = \frac{\mu F_z(1-s)}{2\sqrt{(C_s s)^2 + (C_\alpha \tan\alpha)^2}} \tag{A 2.1}$$

$$F_x = \frac{C_s s}{(1-s)} \tag{A 2.2}$$

$$F_y = \frac{C_\alpha \tan\alpha}{(1-s)} \tag{A 2.3}$$

$$F_x = \frac{\mu F_z C_s s}{\sqrt{(C_s s)^2 + (C_\alpha \tan\alpha)^2}}\left[1 - \frac{\mu F_z(1-s)}{4\sqrt{(C_s s)^2 + (C_\alpha \tan\alpha)^2}}\right] \tag{A 2.4}$$

$$F_y = \frac{\mu F_z C_\alpha \tan\alpha}{\sqrt{(C_s s)^2 + (C_\alpha \tan\alpha)^2}}\left[1 - \frac{\mu F_z(1-s)}{4\sqrt{(C_s s)^2 + (C_\alpha \tan\alpha)^2}}\right] \tag{A 2.5}$$

Dugoff Model Equations:

$$F_x = \frac{C_s s}{(1-s)} f(\lambda) \tag{A 2.6}$$

$$F_y = \frac{C_\alpha \tan\alpha}{(1-s)} f(\lambda) \tag{A 2.7}$$

where λ is given by

$$\lambda = \frac{\mu F_z(1-s)}{2\sqrt{(C_s s)^2 + (C_\alpha \tan\alpha)^2}}$$

and

$$f(\lambda) = (2-\lambda)\lambda \qquad \text{if } \lambda < 1 \tag{A 2.8}$$

$$f(\lambda) = 1 \qquad \text{if } \lambda >= 1 \qquad \text{(A 2.9)}$$

$$\mu = \mu_o (1 - \varepsilon U s) \qquad \text{(A 2.10)}$$

$$F_{xd} = \frac{C_s s}{(1-s)} \qquad \text{(A 2.11)}$$

$$F_{yd} = \frac{C_\alpha \tan \alpha}{(1-s)} \qquad \text{(A 2.12)}$$

$$R_1 = \mu_d F_z \qquad \text{(A 2.13)}$$

$$\mu_d = \sqrt{\mu_{bd\,2} + \mu_{sd}^{\ 2}}$$
$$\mu_{bd} = \frac{F_{xd}}{F_z} \qquad \text{(A 2.14)}$$
$$\mu_{sd} = \frac{F_{yd}}{F_z}$$

$$F_x = \frac{C_s \mu F_z}{\sqrt{C_s^{\ 2} + C_\alpha^{\ 2} \tan^2 \alpha}}$$
$$\qquad \text{(A 2.15)}$$
$$F_y = \frac{C_\alpha \tan \alpha \mu F_z}{\sqrt{C_s^{\ 2} + C_\alpha^{\ 2} \tan^2 \alpha}}$$

Vehicle Parameters

Basic Values	
Wheel base	2423mm
Track width front	1492mm
Track width rear	1426mm
Dynamic tyre radius	280mm
Total mass	1245kg
Distance front axle-centre of gravity	1100mm
Distance rear axle-centre of gravity	133mm
Centre of gravity	580mm

Moment of inertia in the centre of gravity	
Around the x-axis	335kg-m^2
Around the y-axis	1095kgm^2
Around the z-axis	1200kgm^2

Parameters of the tyre	
x01 Longitudinal Coefficient	-28.1983
x02 Longitudinal Coefficient	1124.52
x03 Longitudinal Coefficient	63.6611
x04 Longitudinal Coefficient	85.6943
x05 Longitudinal Coefficient	0.0740026
x06 Longitudinal Coefficient	-0.0717008
x07 Longitudinal Coefficient	0.7822
x08 Longitudinal Coefficient	-1.18694
y01 Longitudinal Coefficient	-43.6004
y02 Longitudinal Coefficient	1177.9
y03 Longitudinal Coefficient	965.218
y04 Longitudinal Coefficient	1.22727
y05 Longitudinal Coefficient	0.217334
y06 Longitudinal Coefficient	-0.0214168
y07 Longitudinal Coefficient	-0.0415905
y08 Longitudinal Coefficient	1.56238e^{-09}
y09 Longitudinal Coefficient	0
y11 Longitudinal Coefficient	0
y12 Longitudinal Coefficient	0
y13 Longitudinal Coefficient	0

	Spring rate [N/mm]
Spring rate front axle	22.8
Spring rate stabiliser front axle	24.0
Spring rate rear axle	19.4
Spring rate stabiliser rear axle	4.8

Appendix C

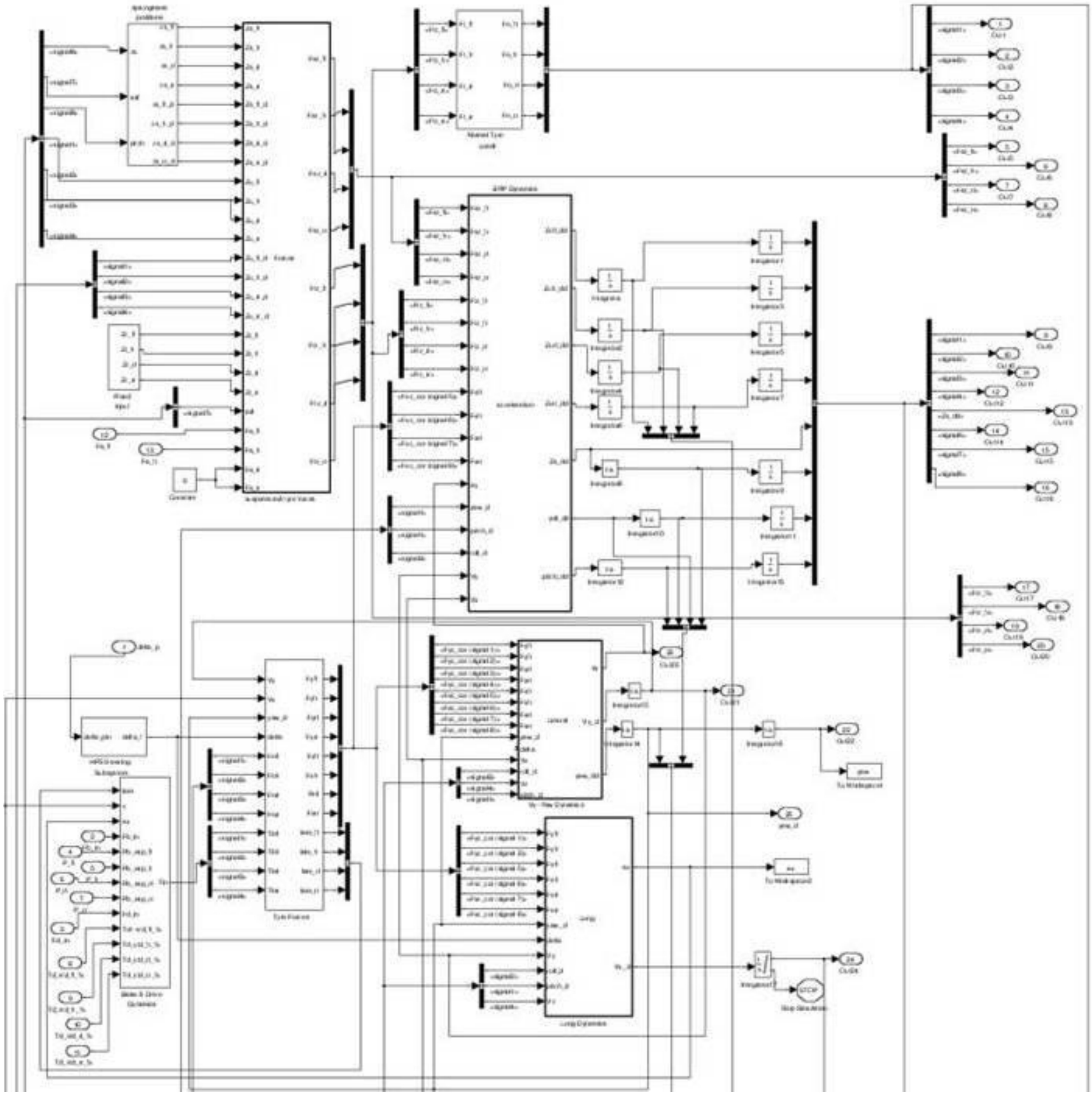

Fig. C1 Full Passive Vehicle Simulink Model

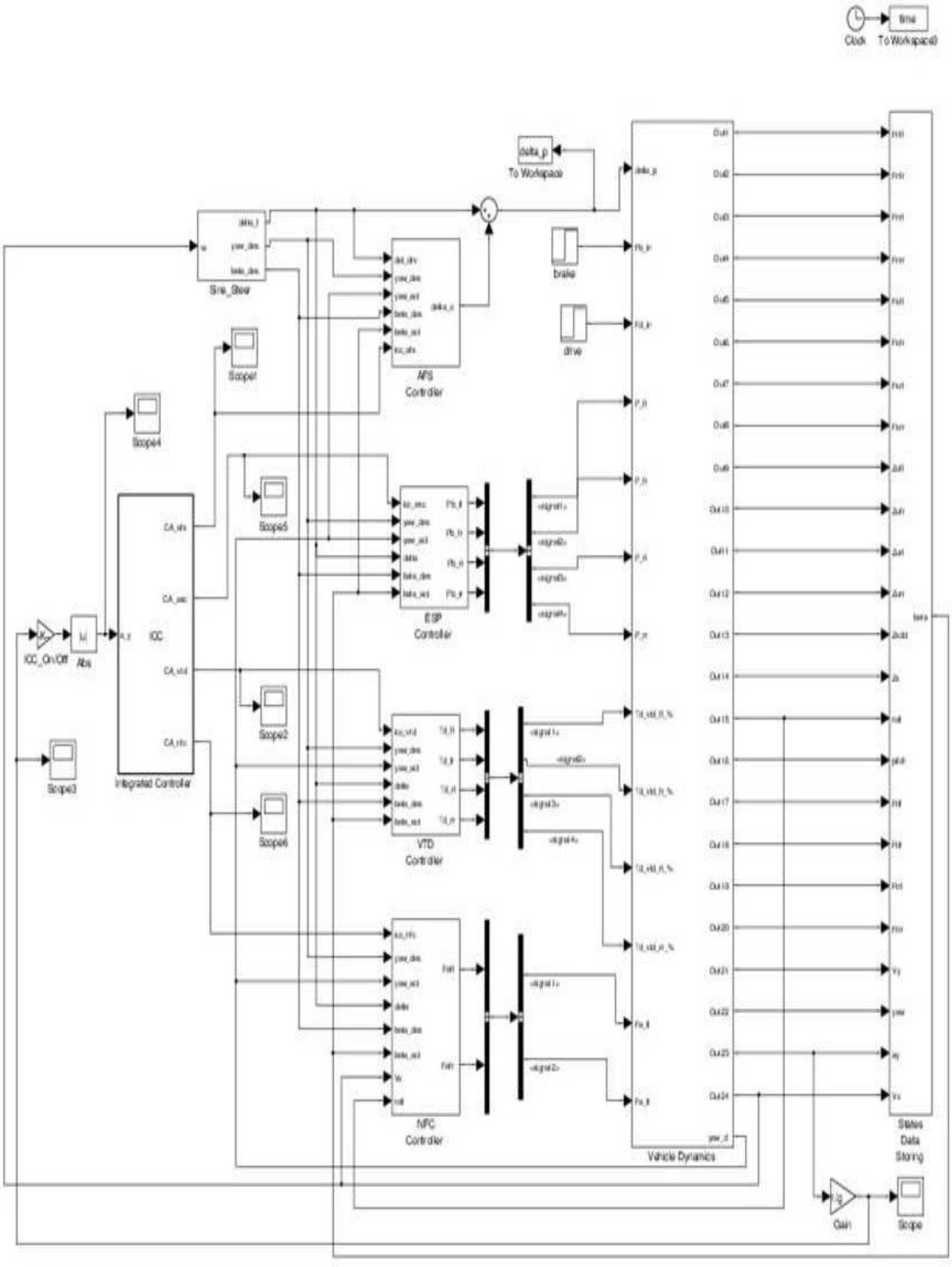

Fig. C2 Full Vehicle Simulink Model with 4 Systems Integrated Controller.

Appendix D

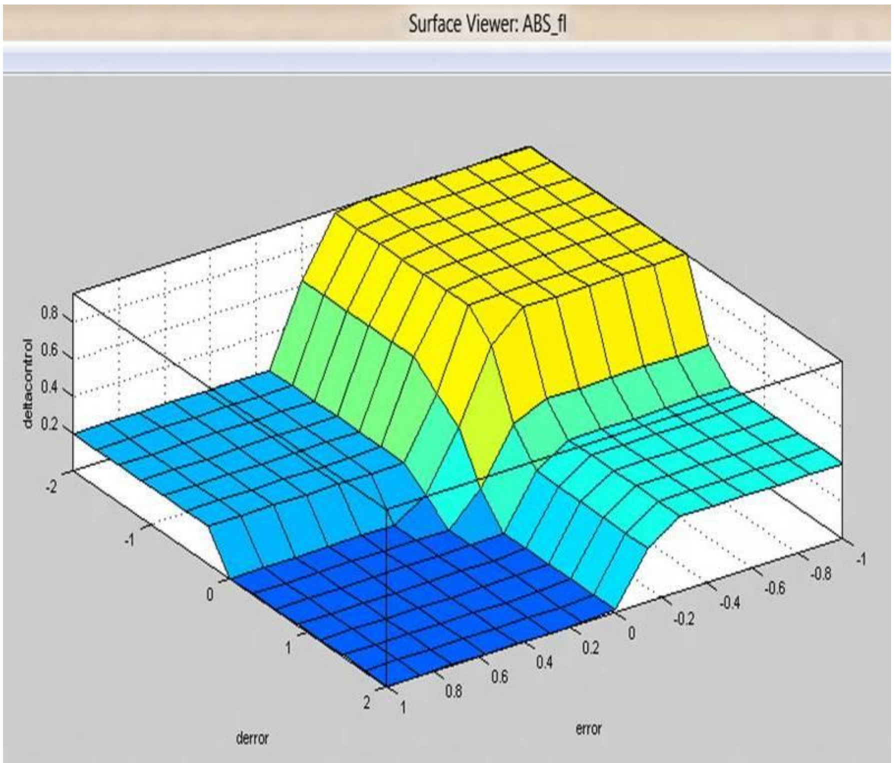

Fig. D1 Nonlinear Fuzzy Control Surface – Antilock Braking System

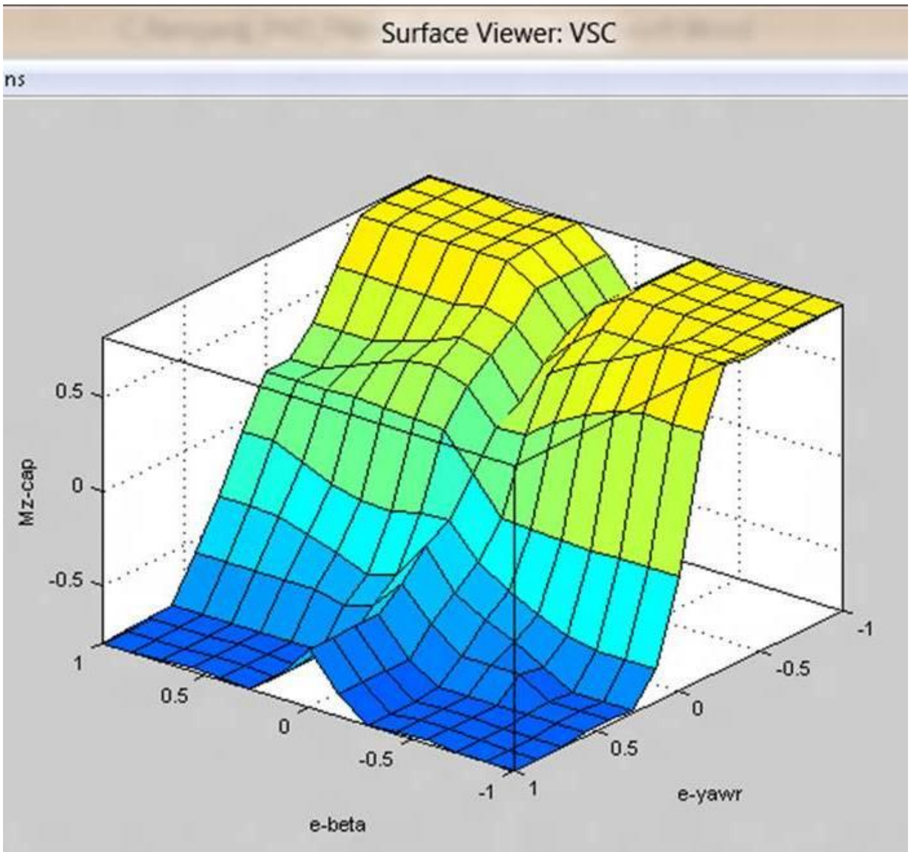

Fig. D2 Nonlinear Fuzzy Control Surface – Electronic Stability Control

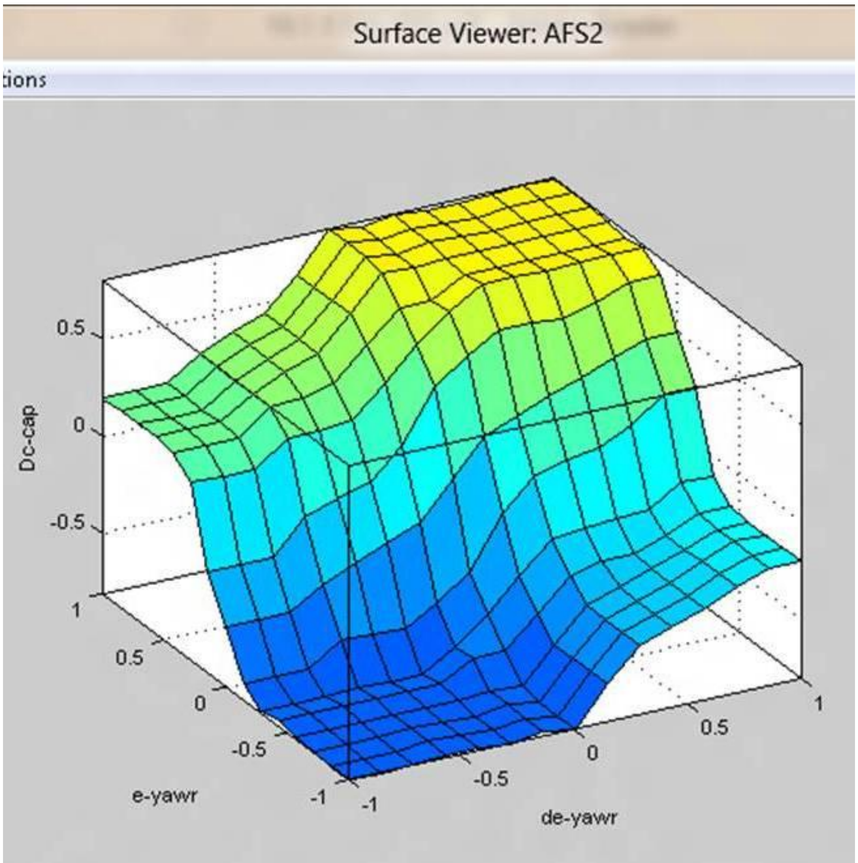

Fig. D3 Nonlinear Fuzzy Control Surface – Active Front Steering – Strategy 2

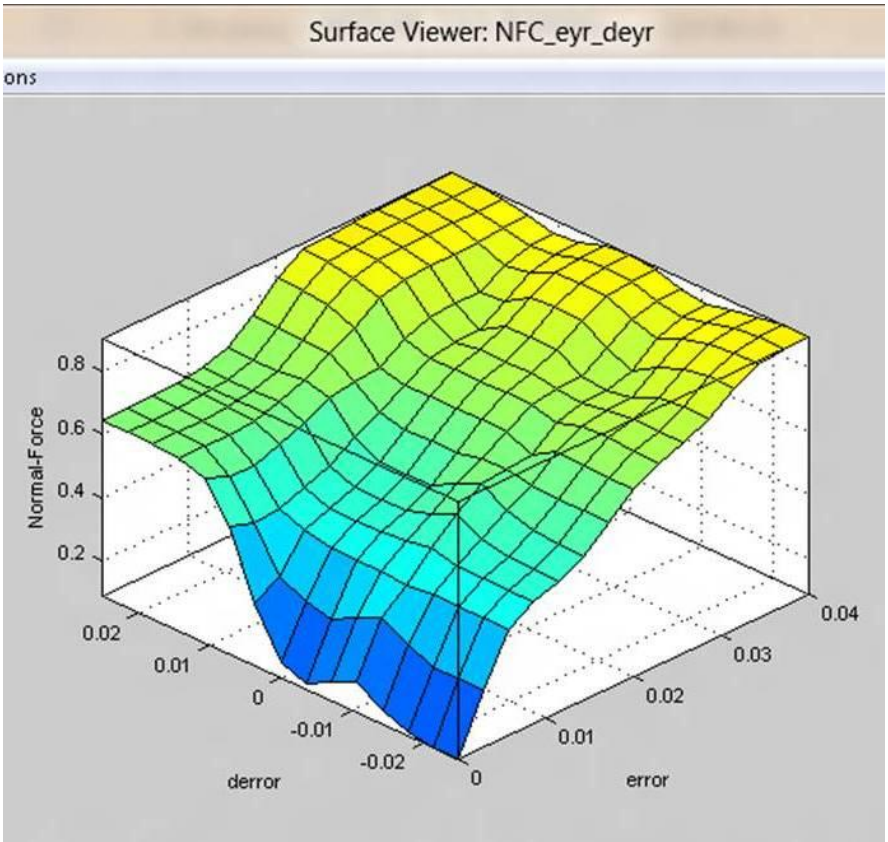

Fig. D4 Nonlinear Fuzzy Control Surface – Normal Force Control