Coventry University

DOCTOR OF PHILOSOPHY

A simple and low cost anti-lock braking system control method using in-wheel force sensor and wheel angular speed sensor

Li, Chen

Award date:
2017

Awarding institution:
Coventry University

Link to publication

General rights
Copyright and moral rights for the publications made accessible in the public portal are retained by the authors and/or other copyright owners and it is a condition of accessing publications that users recognise and abide by the legal requirements associated with these rights.

• Users may download and print one copy of this thesis for personal non-commercial research or study
• This thesis cannot be reproduced or quoted extensively from without first obtaining permission from the copyright holder(s)
• You may not further distribute the material or use it for any profit-making activity or commercial gain
• You may freely distribute the URL identifying the publication in the public portal

Take down policy
If you believe that this document breaches copyright please contact us providing details, and we will remove access to the work immediately and investigate your claim.

Download date: 04. Jul. 2025

A Simple and Low Cost Anti-Lock Braking System Control Method Using In-Wheel Force Sensor and Wheel Angular Speed Sensor

Li, C.

Submitted version deposited in Coventry University's Institutional Repository

Original citation:
Li, C. (2017) *A Simple and Low Cost Anti-Lock Braking System Control Method Using In-Wheel Force Sensor and Wheel Angular Speed Sensor.* Unpublished PhD Thesis. Coventry: Coventry University

Copyright © and Moral Rights are retained by the author. A copy can be downloaded for personal non-commercial research or study, without prior permission or charge. This item cannot be reproduced or quoted extensively from without first obtaining permission in writing from the copyright holder(s). The content must not be changed in any way or sold commercially in any format or medium without the formal permission of the copyright holders.

Some materials have been removed from this thesis due to third party. Pages where material has been removed are clearly marked in the electronic version. The unabridged version of the thesis can be viewed at the Lanchester Library, Coventry University.

A Simple and Low Cost Anti-Lock Braking System Control Method Using In-Wheel Force Sensor and Wheel Angular Speed Sensor

By

Chen Li

Doctoral Thesis

2017

A Simple and Low Cost Anti-Lock Braking System Control Method Using In-Wheel Force Sensor and Wheel Angular Speed Sensor

By

Chen Li

A thesis submitted in partial fulfilment of the University's requirements for the award of Doctor of Philosophy

12th January 2017

© by Chen Li

Abstract

The ABS (Anti-lock Braking System) is an active safety system that is designed for emergency braking situations. In an emergency braking scenario, the ABS instructs the disk-pad braking force to achieve the maximum available tyre-road braking force without locking the wheels. The maximum available tyre-road braking force helps to achieve the optimal braking distance, while the rotating wheels allow the vehicle to retain directional control capability, which allows the driver to avoid dangerous obstacles during an emergency braking scenario. This research has delivered a new and novel approach to ABS design, which could be developed at a low cost in a way which will benefit specialist and niche vehicle manufacturers alike. The proposed ABS control method combines the control logic from both theory-based ABS and commercialised ABS. Therefore, it is more practical compared to the theory-based ABS and less complex compared to a commercialised ABS. The control method only has two control phases with simple decrease, hold, and increase control actions. The proposed ABS control method uses representable tyre-road braking force data from an in-wheel-hub force measurement sensor as well as wheel angular acceleration data from a wheel angular speed sensor as control references. It uses the detected peak tyre-road braking force and its relative predefined drop percentage as control activation and control phase alternation triggers. It uses wheel angular acceleration to identify the control phase and implement the correct control actions. Zero wheel angular acceleration is used to trigger the hold control action in the first control phase, while wheel angular acceleration is used as an aid to increase the accuracy of the in-wheel-hub force sensor. An ADAMS full vehicle model based on a Subaru Impreza and a Simulink ABS control logic model have been used to establish a co-simulation environment to test the performance of the proposed ABS control method using high, low friction and split-mu road surfaces. The co-simulation results demonstrate that the proposed novel ABS control method satisfies the ABS control target, and its control results are similar to commercialised ABS.

Acknowledgements

First of all, I would like to express my very great appreciation to Professor Mike Blundell for his valuable and constructive guidance during this research work. I would like to thank Dr. Gary Wood and Dr. Yuri Vershinin for their great support. I would like to thank Professor Keith Burnham, Professor Peter White, and Mr Bill Dunn for their valuable advices on this research work.

I would also like to thank my parents. They are always supporting and encouraging me with their best wishes.

Finally, I would like to thank my wife Zhu Lin. She is always there cheering me up and stand by me through the good times and bad.

Contents

List of Figures

List of Tables

Notation

Chapter 2

V_w	Longitudinal wheel velocity in the tyre plane
W_ω	Wheel angular velocity
R	Effective rolling radius of tyre
F_x	Tyre longitudinal force
λ σ	Tyre longitudinal slip ratio
μ	Road friction coefficient
A	Drop percentage after optimal slip ratio
B	Drop percentage before optimal slip ratio
V	The longitudinal velocity of the vehicle
I_t	The total moment of inertia of the wheel
m_t	The total mass of quarter vehicle
m_{vs}	The total sprung mass of the vehicle
m_w	Wheel mass
T_b	The braking torque
F_z	The vertical force
h_{cg}	The height of vehicle gravity centre
l	Wheel base
x	The vehicle longitudinal travel distance
D	Peak value factor of magic formula
C	Shape factor of magic formula used for stretching
B	Stiffness factor of magic formula
E	Curvature facto of magic formula
S_v	Vertical shift factor of magic formula
S_h	Horizontal shift factor of magic formula
Y	Tyre forces and moment representation of magic formula
F_y	Lateral force
α	Slip angle
C_i	Tyre longitudinal stiffness of Dugoff tyre model
C_α	Tyre cornering stiffness of Dugoff tyre model

Chapter 3

F_p	The force transmitted by brake piston
F_b	The force between brake disk and brake pad
A	The area of brake piston
n	The number of brake pad
T_b	The braking torque
P	The brake pressure
μ	The friction coefficient between brake pad and disk

F_{d-p} Disk-pad braking force
F_{t-r} Tyre-road braking force
F_t Weight transfer force
R_d Disk pad to wheel center distance
V_i Vehicle initial speed
V_f Vehicle final speed
S Vehicle total travel distance
g Earth gravity
$F_{t\text{-}r\,Max}$ Peak tyre-road braking force
T Time

Chapter 4

$\dot{\omega}_{ref}^{+}$ Desired reference wheel angular acceleration in control phases 1

$\dot{\omega}_{ref}^{-}$ Desired reference wheel angular acceleration in control phases 2

1. Introduction

1.1. Background

The ABS (Anti-lock Braking System) is an active safety system that is designed for emergency vehicle stop situations. During an emergency stop, the ABS allows the driver to apply the optimal disk-pad braking force without locking the wheels. The optimal disk-pad brake force helps to achieve the maximum available tyre-road braking force, so that the optimal brake distance can be achieved. At the same time, rotating wheels allow the vehicle to retain directional control capability so the driver can steer away from dangerous objects during an emergency stop.

Due to consumer safety concerns and the introduction of new vehicle safety legislation, ABS will soon be more widely adopted on vehicles, including those in the niche vehicle sectors. At present, the most used ABS on production vehicles is supplied by Bosch. The company charges automotive manufacturers millions of pounds for licencing and the installation of Bosch patented ABS. For the automotive giants who manufacture huge numbers of vehicles annually, the spread cost is affordable. But for some niche vehicle companies which only manufacture small number of vehicles, the price is not feasible. Accordingly, a newly patented ABS with low implementation cost would be welcomed by not only niche vehicle companies, but also the automotive giants. The development of a new ABS with low implementation cost which is based on new sensor technology is the main objective of this PhD. The multibody dynamics software ADAMS (Automotive Dynamic Analysis of Mechanical Systems) is used to develop a dynamic full vehicle model, while MATLAB Simulink is used to develop the ABS control logic model. ADAMS-Simulink co-simulation is used to integrate vehicle and control logic models for the simulation of test scenarios.

1.2. Structure of thesis

This PhD thesis consists of seven chapters. Chapter 1 gives a brief introduction to anti-lock braking systems and provides an overview of the structure of this thesis. Chapter 2 reviews essential ABS knowledge, as well as current developments in ABS from both academic and industry aspects. Conclusions are drawn on the basis of these reviews in relation to the direction of the proposed ABS research. Chapter 3 introduces the full Subaru vehicle model used in this study and developed in ADAMS. Vehicle validation results are then presented. Proposed ABS related experimental test results are also presented and incorporated into the guidelines to establish the new ABS control logic. Chapter 4 combines the analysis of commercialised ABS algorithms with the latest academically developed ABS, along with the basic disk-pad braking force and tyre-road braking force interaction properties to form the details of the new ABS control logic. Chapter 5 presents the Simulink based control logic model which was developed based on the guidelines and details established previously. Chapter 6 presents the ADAMS-Simulink co-simulation test procedures and results, along with an analysis of the results, are also presented. Chapter 7 provides a conclusion to the thesis in its entirety, provides recommendations and clarifies future work.

1.3. Aims & Objectives

The aim of this research is to develop a novel low cost ABS control method and to investigate the potential usage, intended primarily for use by the niche vehicle sector.
In order to meet this aim the following will be undertaken:

- Carry out a literature review into existing ABS algorithms and implementation strategies.
- Research brake design, tyre behaviours and tyre-road friction characteristics.
- Develop and validate a demonstration simulation model in ADAMS based on a Subaru Impreza research vehicle at Coventry University.
- Develop a new low-cost ABS design and control algorithm based on advanced sensor technologies.
- Carry out vehicle/ABS co-simulation to demonstrate the effectiveness of the ABS control method, simulating a range of braking test scenarios.
- Interpret the results and provide a conclusion on the potential for exploitation and implementation.

2. Literature Review

2.1. ABS (Anti-lock Braking System) Overview

The commonly used ABS system consists of four individual units. The Electronic Control Unit (ECU), the Sensor Unit (SU), the Hydraulic Unit (HU), and the conventional brake system unit.

This item has been removed due to 3rd Party Copyright.This item has been removed due to 3rd Party Copyright. The unabridged version of the thesis can be found in the Lanchester Library, Coventry University

This item has been removed due to 3rd Party Copyright.This item has been removed due to 3rd Party Copyright. The unabridged version of the thesis can be found in the Lanchester Library, Coventry University

Figure 2.1 Generic brake system layout and Bosch ABS component (Bosch website)

As seen in Figure 2.1, the conventional brake system unit includes parts such as the brake pedal (1), which is used as the driver input mechanism; the vacuum booster (2), which is used to amplify the driver input by using the vacuum produced pressure difference (the vacuum is produced by engine); and the master cylinder (3), which is used to pressurise and route brake fluid. The pressurised brake fluid is usually split between the front wheel brake channel and the rear wheels' brake channel. The split ratio is determined by the brake proportioning valve configuration. The brake fluid reservoir (4) is used to store brake fluid, while the brake unit (5), which is either in disk-pad configuration or in drum-shoe configuration, applies the braking force to the wheel.

The Electronic Control Unit (ECU) is a microprocessor system that implements the ABS control algorithm and the related data process. The black part which is attached to the Bosch ABS hardware shown in Figure 2.1 contains the ECU.

The SU may consist of different types of sensors, but the most common is the wheel angular speed sensor which is usually an electromagnetic sensor. Other than the wheel angular speed sensor, the SU could also consist of vehicle accelerometer and the tyre force sensor.

As seen in Figure 2.2, the HU mainly consists of electronically activated solenoid valves which are used to modulate brake pressure. A low-pressure accumulator is used to temporarily store brake fluid from the release solenoid. The electric brake fluid pump is used to transfer brake fluid from the low-pressure accumulator to the master cylinder as well as supply the solenoid to guarantee sufficient brake fluid supply to the pressure modulator. The high-pressure damper is used to dampen the high-pressure brake fluid supplied by the electric pump for the purposes of smoothing the brake fluid supply to the master cylinder.

This item has been removed due to 3rd Party Copyright.This item has been removed due to 3rd Party Copyright. The unabridged version of the thesis can be found in the Lanchester Library, Coventry University

This item has been removed due to 3rd Party Copyright.This item has been removed due to 3rd Party Copyright. The unabridged version of the thesis can be found in the Lanchester Library, Coventry University

Figure 2.2 Solenoid valves (Hyundai Motor company ABS training document)

The operational process of the ABS is as follow: First, the ABS ECU receives signals from the directly connected sensors and through the CAN (Controller Area Network) bus; secondly, the ABS ECU processes and analyses the received signals; then the ABS ECU generates adequate power signals (voltage or current) to activate the HU according to the sensors, other related signals analysis results and the relevant control algorithm; finally, the modulated brake pressure is applied via the conventional brake unit.

The Solenoid layout configuration usually includes a dual positional solenoid 2/2 which only has two operation states; one is active and the other is deactivated. There are two related positions, one is open and the other is closed. In the active state, power is supplied to the solenoid, which could be in either the open position or the closed position depending on the ABS hydraulic route design. In the deactivated state, the power supply is cut from the solenoid, which could also be in either the open position or the closed position, depending again on the ABS hydraulic route design. These dual positional configurations of the solenoids (in either active or deactivated positions) can be employed to achieve a different control topology. As seen in Figure 2.3, the brake fluid inlet in solenoid A is open in the deactivated state and closed in the active state. The outlet in solenoid C is closed in the deactivated state and open in the active state. If brake pressure increase is desired, the inlet solenoid is put in a deactivated state in the open position, and the outlet solenoid is put in a deactivated state in the closed position. If brake pressure hold is desired the inlet solenoid is put in an active state in the closed position, and the outlet solenoid is put in a deactivated state in the closed position. If brake pressure decrease is desired, the inlet solenoid is put in an active state in the closed position, and the outlet solenoid is put in an active state in the open position.

This item has been removed due to 3rd Party Copyright.This item has been removed due to 3rd Party Copyright. The unabridged version of the thesis can be found in the Lanchester Library, Coventry University

Figure 2.3 Basic ABS layout (The Clemson University Vehicular Electronics Laboratory website)

Application delay could present in electro-mechanical hydraulic execution units, which could cause ABS control delays. Delay can be on both apply and release phases. They represent the periods that the output pressure increases or decreases to a desired level (Terry et al., 2002). These delays must be considered in the ABS controller design and the ABS performance simulation.

2.2. ABS theory

The maximum available negative vehicle longitudinal acceleration and adequate steering ability are ABS control objectives. The maximum available negative vehicle longitudinal acceleration is governed by the maximum longitudinal tyre-road braking force which is generated by the tyre at the tyre-road contact patch. The steering ability is governed by the lateral force generated by the tyre at the tyre-road contact patch. Both the longitudinal tyre force and the lateral tyre force are related to the tyre longitudinal slip ratio.

2.2.1. Longitudinal slip ratio

The longitudinal tyre slip ratio is defined as the difference between the longitudinal wheel velocity in the tyre plane and the wheel peripheral velocity. The equation for the longitudinal slip ratio is:

$$slip = \frac{V_{w-\omega \cdot R}}{V_w} \qquad \textbf{(2.1)}$$

where

V_w= Longitudinal wheel velocity in the tyre plane

W = Wheel rotational velocity

R = Effective rolling radius of tyre

The equation is ostensibly somewhat simple, but it in practice achieving the desired effect is very complex. That is because the longitudinal wheel velocity in the tyre plane and the effective rolling radius of the tyre are difficult to be measured directly and accurately. This is due to sensor limitations, the non-linearity involved in vehicle dynamics, tyre dynamics and the unpredictability of the environment during a braking manoeuvre. When braking, the wheel peripheral velocity cannot be used to represent the longitudinal wheel velocity due to the slip between the tyre and the road. The situation is exacerbated when the vehicle is turning or travelling on a low friction road surface and/or when an uneven road causes unexpected tyre deformation. Therefore, a lot of research has focused on estimation methods regarding longitudinal wheel velocity and effective tyre rolling radius. Estimations can achieve a high level of accuracy in predictable situations, especially regarding effective tyre rolling radius (Shannon et al., 2001). But the limitations of these estimations could still cause some instability in ABS control in unpredictable scenarios (during emergency braking), so accurate and consistent measurement methods are desired and remain the focus of intensive research.

2.2.2. Longitudinal tyre braking force

The typical relationship between the longitudinal tyre slip ratio and longitudinal tyre braking force is shown in Figure 2.4. As can be seen, initially, the longitudinal tyre braking force increases when the slip ratio increases; then, when the slip ratio reached certain level, the longitudinal tyre braking force approaches its maximum value. This slip ratio level is relatively low (between 15% and 30%) and is called the optimal slip ratio. It is a very important ABS control parameter. Once the optimal slip ratio is reached, the longitudinal tyre braking force starts to decrease. This relationship is very important as it is widely used in different ABS control methods which will be discussed later. In addition to the longitudinal tyre slip ratio, the tyre vertical force and the tyre-road friction coefficient also have strong effects on the longitudinal tyre braking force.

This item has been removed due to 3rd Party Copyright.This item has been removed due to 3rd Party Copyright. The unabridged version of the thesis can be found in the Lanchester Library, Coventry University

Figure 2.4 Longitudinal tyre braking force to slip ratio A (Damiano et al. 2009)

As seen in Figure 2.5, the region on the left of the optimal slip ratio is called the stable brake region and the region on the right of the optimal slip ratio is called the unstable region. Within the stable region, the maximum longitudinal tyre braking force can be achieved, and the wheel rotates allowing steering ability to be retained. Within the unstable region, the wheel locks up very quickly, the tyre braking capability is reduced and steering ability is compromised. Consequently, the ABS control algorithm is needed to limit the slip ratio within the stable region, maintain maximum tyre braking capability and retain directional control.

This item has been removed due to 3rd Party Copyright.This item has been removed due to 3rd Party Copyright. The unabridged version of the thesis can be found in the Lanchester Library, Coventry University

Figure 2.5 Longitudinal tyre braking force to slip ratio B (Damiano et al. 2009)

The tyre longitudinal braking force has an approximately linear relationship with the longitudinal slip ratio within the stable region of the tyre. Beyond the stable region, the linear relationship does not exist and the dynamics of nonlinear tyre behaviour takes effect (Hossein et al., 2010). As it is assumed that ABS control could maintain tyre longitudinal braking force within the linear region, the linear tyre longitudinal braking force and slip ratio relationship could be used as ABS control reference (Shannon et al., 2001).

2.2.3. Road surface friction coefficient

There is another important factor that must be considered in the slip ratio study; the road surface friction coefficient. The relationship between wheel slip ratio and the friction coefficient is shown in Figure 2.6. As is seen, the optimal slip ratio is different on high and low friction coefficient surfaces; the maximum available longitudinal friction coefficient is achieved at the optimal slip ratio; the lateral force generation capacity decreases with the increase in the slip ratio because the lateral friction coefficient decreases. This estimation method is commonly used to ascertain the friction coefficient.

Direct measurement methods are becoming more popular, and some of these methods will be introduced later in this chapter.

This item has been removed due to 3rd Party Copyright.This item has been removed due to 3rd Party Copyright. The unabridged version of the thesis can be found in the Lanchester Library, Coventry University

Figure 2.6 Longitudinal and lateral coefficient of friction versus slip ratio curve (Kazemi et al. 2005)

2.3. ABS control methods

In the literature review, two designs of the ABS control method are mostly employed. One is based on logic switching, which is in turn based on wheel angular acceleration information; the other is tyre longitudinal slip ratio regulation based. Nonlinearity and model uncertainties are considered difficulties in regard to ABS control method development.

2.3.1. Slip ratio control method

In regard to the slip ratio control method, the ABS modulates the brake pressure to maintain the slip ratio (located between B and A as shown in Figure 2.7) in order to keep the longitudinal tyre braking force within the maximum range without locking the wheel. The slip ratio based method can adopt a simple increase-hold-decrease brake pressure control approach. The method works even when no maximum tyre forces occur, and there are no periodic oscillations in the brake pressure control. However, this method is based on linearization arguments, so the stability analysis is only available locally and the method might fail with respect to the unstable region of the tyre braking force (Mathieu et al., 2010).

This item has been removed due to 3rd Party Copyright.This item has been removed due to 3rd Party Copyright. The unabridged version of the thesis can be found in the Lanchester Library, Coventry University

Figure 2.7 Slip ratio control interval (Damiano et al. 2009)

The slip ratio control interval, which is the period between A and B as seen in Figure 2.7, was discussed by Damiano et al. (2009) and influences the ABS performance. The interval should not be too large, so as to permit intensive control (in order to shorten the braking distance); however, it is difficult for ABS to maintain a small interval, especially on a low friction surface, mainly due to delays in the hydraulic unit. A compromise should be reached to achieve the best average performance.

As mentioned previously, the slip ratio cannot be measured directly as it is calculated using the slip ratio equation (equation 2.1). As seen in the slip ratio equation, to calculate the optimal slip ratio the related longitudinal wheel velocity and effective tyre rolling radius must be estimated, or accurately measured, as must the tyre-road friction coefficient. Between the longitudinal wheel velocity and the tyre-road friction coefficient the latter influences the optimal slip ratio the most. They are usually estimated or measured at the same time in the slip ratio control. A typical slip ratio based control process block diagram is shown in Figure 2.8. It represents the most common theoretical ABS control logic flow.

This item has been removed due to 3rd Party Copyright.This item has been removed due to 3rd Party Copyright. The unabridged version of the thesis can be found in the Lanchester Library, Coventry University

Figure 2.8 Slip ratio based ABS control process block diagram (Damiano et al. 2009)

2.3.1.1. Slip ratio control related longitudinal wheel velocity and tyre-road friction coefficient estimation and measurement method

Damiano et al. (2009) proposed an effect based tyre-road friction estimator which uses the wheel angular deceleration to identify the tyre/road friction coefficient, but it can only estimate high and low levels of friction (it cannot provide more detailed friction estimates).

They also proposed a longitudinal vehicle speed estimation method based on the four-wheel angular speed measurements. A speed variation method is proposed in which the speed variation is defined as the difference between the measured wheel speed and the estimated vehicle speed. The speed variation is proportional to the friction coefficient. The friction coefficient can be obtained from the friction estimator to calculate the speed variation; then the speed variation is added to the measured wheel speed to determine the vehicle speed. The vehicle longitudinal acceleration signal is also used as a security parameter to aid in determining the longitudinal vehicle speed. This accounts for instability caused by sudden grip change, which is considered a limitation to the pure slip control method.

Sahin et al. (2010) proposed a friction coefficient estimator that only uses vehicle longitudinal acceleration as an input. The linear tyre longitudinal braking force and the slip ratio relationship are employed in this estimator, while the longitudinal acceleration (a function of braking force) and the change in vehicle vertical load while braking are ignored. The friction coefficient could be found by interpreting the relationship between the vertical load, braking force and friction coefficient. This method will encounter some difficulties if the vehicle does not travel in a straight line and/or is characterised by non-linear vehicle and tyre properties, as the longitudinal acceleration measurement is inaccurate without calibration.

The same paper also proposed a longitudinal vehicle speed estimation method based on the wheel angular speed and vehicle longitudinal acceleration measurements. This method uses the polluted vehicle longitudinal acceleration and wheel angular speed to estimate the vehicle longitudinal speed. Sahin et al. (2010) state that vehicle longitudinal acceleration can aid the pure longitudinal slip vehicle reference speed estimation. In this regard a signal processing tool, the Kalman filter, was introduced and it will be discussed later.

Sahin et al. (2010) also proposed the direct calculation method which uses vehicle longitudinal and lateral acceleration, yaw rate and steering angle measurements from the on-board sensors as inputs to calculate the longitudinal wheel velocity. The sensors used to measure the required data are not ordinarily used with ABS, but if the ABS is integrated with other active safety systems, like ESP, this method could become a reality on production vehicles.

M.Tanelli et al. (2006) proposed the same longitudinal speed estimation method as Sahin et al. (2010). The method is based on the measurement of the four wheels' angular speed and longitudinal acceleration. This method uses the average longitudinal speed of the four wheels and the two non-driving wheels (derived from their respective angular speeds), as well as the filtered signal from the longitudinal acceleration sensor as inputs. The test results were positive and the performance of the proposed longitudinal speed estimation method was close to the performance of the commercialised ABS control method.

Shannon et al. (2001) proposed the use of a Global Positioning System (GPS) to measure a vehicle's absolute speed. Shannon et al. (2001) discussed the GPS measurement latency but not the measurement accuracy (although it was referred to). They suggested that a longitudinal accelerometer can be a significant aid to the GPS in terms of measuring latency and accuracy.

Sami (2010) introduced the use of both conventional vehicle dynamic sensors and conceptually new environment sensors to measure the tyre-road friction coefficient. Conventional vehicle dynamic sensors include wheel angular speed measurement sensors, an Inertial Measurement Unit (IMU) - for accelerations and rotations, brake pressure sensors, and steering angle sensors. The conceptually new environment sensors include a road eye sensor, a LUX laser scanner, a Correvit ground speed camera, an IcOR polarization camera, and an APOLLO/FRICTI@N project tyre sensor. The LUX laser scanner can provide true ground speed and vehicle heading direction if a static object is identifiable, but the measurement cannot be performed if no visible static object can be identified. This research offers a positive appraisal of the sensors currently under development, although the costly sensors are only available for circuit test vehicles at this time.

Sami (2010) also introduced the friction estimation method based on dynamical friction models, such as the LuGre model, that includes hysteresis effects, velocity dependence and so on. Within the model, road conditions are based on a single parameter, so friction can even be estimated using only wheel angular velocity (although this is only the case with regards to slip). Deng et al. (2006) proposed the use of the LuGre dynamic friction model in the friction coefficient estimation process for their proposed ABS control method as well.

Eichhorn et al. (1992) proposed to mount a microphone in the wheel box to measure the noise emitted by the tyre in order to estimate the friction coefficient, but tyre noise is affected by many factors, such as road surface texture, so it only has limited uses.

2.3.1.2. Slip ratio control related effective rolling radius measurement or estimation

Shannon et al. (2001) proposed an effective rolling radius estimation method that uses both the GPS measurements, the wheel speed sensor measurements and a simple least-squares regression technique.

Sami (2010) introduced an optical tyre deformation sensor with APOLO/FRI@TION. This sensor could also be modified to measure the effective tyre radius.

Bulent (1999) examined the effectiveness of a geometry based tyre rolling radius calculation method and a tyre rolling radius calculation formula introduced by Dunlop Tyres. Both calculation methods present similar results with less than 2% difference.

2.3.2. Wheel angular acceleration based control

Mathieu et al. (2010) proposed a two-phase ABS control method which is wheel angular acceleration based. It measures and controls the wheel angular acceleration in a closed loop.

A wheel angular acceleration based control algorithm is simple and robust. It can keep the wheel slip in the neighbourhood of its optimal point without explicitly using the value of the optimal set point. Since the algorithm eliminates the optimal slip ratio estimation, estimations or measurements of vehicle velocity, tyre/road friction coefficient and effective tyre rolling radius are no longer necessary. These enhance the performance of the control method regarding handling changes in vehicle velocity, the tyre-road friction coefficient and the effective tyre rolling radius. This also allows the algorithm to cope with measurement noise. Some of the recent research even uses the wheel angular deceleration thresholds to track the optimal slip ratio by analysing the stability of its limit cycles.

Nonetheless, the algorithm is always based on heuristic arguments. The tuning of the thresholds could be difficult (Kiencke et al., 2000) in the absence of accurate mathematical representation or an absolute linear relationship. The conventional wheel angular acceleration based control method uses wheel angular acceleration for both maximum tyre-road braking force detection and control, and this may cause conflict.

2.3.3. Control algorithms other than pure slip ratio or pure angular acceleration based

2.3.3.1. Bosch ABS

The Bosch ABS is based on wheel angular acceleration thresholds and a slip ratio threshold. A complex rule is used to determine the brake pressure modulation (Terry et al., 2002). The Bosch algorithm has eight control phases, as seen in Figure 2.9, and the wheel angular acceleration thresholds represent the boundaries for most control phases. In the second control phase, the slip ratio threshold is set, then, from the start of phase 3 to phase 8, the current estimated slip ratio is compared with the previously determined optimal slip ratio. If the current slip ratio exceeds the optimal slip ratio, all other control phases are bypassed and the brake pressure decrease phase is active. This is called an adaptive learning function, which makes the algorithm robust in the event of a change in road friction. Figure 2.9 shows the Bosch ABS brake pressure control cycle.

This item has been removed due to 3rd Party Copyright.This item has been removed due to 3rd Party Copyright. The unabridged version of the thesis can be found in the Lanchester Library, Coventry University

Figure 2.9 Bosch ABS control algorithm explanations (Bosch, 2004)

Bosch added a complicated slip ratio threshold in its control method for assurance. When the wheel experiences high inertia, the tyre-road friction coefficient is low and change in brake pressure is delayed, as in the case of cautious initial braking on black ice, for example. In such cases the wheel could lock without any response from the pure wheel angular acceleration switching threshold based control method. The slip ratio threshold is designed to aid the ABS control in such exceptional situations in order to guarantee an overall satisfactory ABS performance (Bosch, 2004).

The Bosch ABS is very robust in handling the uncertainties and nonlinearities involved in emergency braking situations. But its performance is limited because its oscillatory nature can cause undesired and noticeable vibrations. This complexity limits system integration flexibility and performance (Petersen, 2003).

2.3.3.2. Van der Jaght et al. proposed rules based control method

Bulent (1999) introduced an ABS control method based on a theory proposed by Van der Jaght et al (1989). This control method shares the same philosophy as the Bosch ABS; it uses a very complex rule to modulate the brake pressure. The set thresholds are wheel angular acceleration and the difference between wheel peripheral speed and wheel linear speed. The control method has seven states and five events. The set thresholds are used to switch between the control phases. The disadvantage of this control method is its complexity.

2.3.3.3. Combined wheel angular acceleration and wheel longitudinal slip ratio

Mathieu et al. (2010a) proposed an ABS control method based on both wheel angular acceleration and longitudinal slip ratio control. The cascaded control approach is unlisted in the control method, so it is not as complex as the Bosch ABS. This control method can handle the non-linear nature of ABS and provides good performance.

2.3.3.4. Two phase, tyre longitudinal braking force phase switch

Mathieu et al. (2010b) proposed a hybrid, two-phase ABS control algorithm based on wheel angular acceleration thresholds. This control method measures and controls the wheel angular acceleration in a closed loop. The tyre longitudinal braking force measurement from the SFK (a bearing manufacturer) Load Sensing Hub Bearing Units (LSHBU) is used to switch control phases and calculate related reference thresholds. This is based on the relationship between tyre longitudinal braking force and slip ratio.

The maximum longitudinal tyre braking force occurs at the optimal slip ratio. Before reaching the optimal slip ratio, the longitudinal tyre braking force increases with the increase of the slip ratio. After reaching the optimal slip ratio, the longitudinal tyre braking force starts to decrease. Therefore, when the measured longitudinal tyre braking force starts to decrease in an emergency brake situation, it could be assumed that its maximum braking force has been passed. On the basis of this relationship between the longitudinal tyre braking force and the slip ratio, it can be concluded that the optimal slip ratio is passed, which triggers the switch of the control phase to decrease brake pressure.

The proposed hybrid control method is robust enough to deal with the uncertainties of the ABS control, which are caused by the absence of a slip ratio estimation. It uses a longitudinal tyre braking force measurement as a control phase switch, so it is more accurate than a control method based on conventional wheel angular acceleration thresholds. Accordingly, the conflict between detection and control is resolved. This control method is simple, so it is easy to tune the control parameters to improve its performance.

Nonetheless, this control method has two limitations. Firstly, the longitudinal tyre braking force cannot be held at its maximum level as drop in the brake force is necessary to switch the control phase; secondly, there is no clear maximum tyre-road friction in a limited slip event (e.g. on snow or ice where the control phase cannot be switched).

2.3.4. Rear wheel brake control

In order to achieve better vehicle handling stability when braking, the rear wheel control strategy should be configured slightly differently from that of the front wheels. While braking and cornering, a high lateral force generation with regards to the rear wheels - especially for the wheel on the outside of the bend is desired to enhance cornering performance. Accordingly, the brake control strategy should keep the rear wheel slip ratio at a somewhat low level (Bosch, 2004). In regard to the control strategy design for the rear wheel, the brake pressure distribution bias between the front and rear wheel should also be considered.

2.3.5. Controller technique

In addition to the thresholds based logic controller used on the Bosch ABS, fuzzy logic and PID controllers are used for ABS. Sahin et al. (2010) compares the error elimination performance of a fuzzy logic controller and a PID controller on an ABS.

The fuzzy logic controller was fed with an error with regards to the estimated and the reference slip ratio and the wheel angular acceleration. The controller controls the brake pressure to eliminate the error. The wheel angular acceleration information is used as an insurance mechanism in regard to the controller. When the wheel angular deceleration reaches a predefined threshold, the controller fully reduces the braking pressure irrespective of the slip ratio. An extensive trial and error process is expected in the development of a fuzzy controller.

The PID controller was fed with an error in regard to the estimated and reference slip ratio only. The control parameters’ determinations are not based on an analytical approach. They are determined based on intensive simulations and experiments, along with previous experience.

It was concluded that both types of controllers were able to track the slip ratio values. The fuzzy logic controller presents better performance regarding control accuracy. The PID controller presents good performance regarding brake pressure overshoot elimination at the initial braking stage.

2.4. Sensing technology

2.4.1. In-wheel acceleration sensor

Hee et al. (2010) introduced a tri-axial acceleration sensor that fits in the wheel hub to measure the wheel lateral acceleration. This acceleration is then converted into tyre lateral force through simple algebraic equations. The tyre lateral force is used to calculate the vehicle body slip angle which is a control parameter used in ESC (Electronic Stability Control).

The three-axial acceleration sensor is designed by MEMS technology. It is integrated with the wheel angular speed sensor, and it is small and robust. Figure 2.10 shows a picture of the sensor.

This item has been removed due to 3rd Party Copyright.This item has been removed due to 3rd Party Copyright. The unabridged version of the thesis can be found in the Lanchester Library, Coventry University

Figure 2.10 Integrated wheel speed sensor and accelerometer (Hee et al. 2010)

Hee et al. (2010) also devised a signal conditioner. It uses a second-order low pass filter and a first-order high pass filter to condition the noisy signal. This research did not specifically refer to the possible delay caused by the signal conditioner. The effectiveness of this method is ascertained in a road test. The in-wheel six-component wheel force transducer measured lateral force is compared to the wheel lateral acceleration converted lateral force. The two share the same tendency, but a small difference between the peak values is noticed. A numerical simulation is carried out based on the converted lateral force in the test. The results show that the use of a wheel lateral acceleration converted lateral force can enhance the performance of the original ESC. Hee et al. (2010) also conclude that the wheel hub acceleration occurs about 0.2s earlier than the vehicle body acceleration.

2.4.2. Smart wheel

The information regarding the forces and moments acting at the tyre/road contact patch is very important for vehicle dynamic related control systems. But the information is too complex to be accurately measured, so the estimation-based control systems gain popularity. The direct forces and moment measurement methods are under intensive research, as the accurately measured forces and moments will significantly improve the performance and reduce the complexity of the vehicle handling related control systems. Smart wheel is the name of a wheel that can measure tyre forces and moments with integrated sensors. The smart wheel could be used to validate mathematical tyre models. Most of the smart wheels incorporate wireless data transmission and non-battery powered supply technologies that could experience difficulties in real driving conditions.

Gobbi et al. (2010) present a lightweight smart wheel which can be used to measure the three forces and three moments acting at the tyre-road contact patch. The wheel has three supporting spokes. Strain gauges are fitted on top of the spokes to detect any deformation in them, which is then converted into data on forces and moments. The gauge sensor uses a non-battery powered power source and the measured signal is transmitted wirelessly. This wheel is used in a laboratory environment and is currently too expensive to use on production vehicles.

Ryosuke et al. (2006) proposed a smart wheel to measure the forces acting on the tyre-road contact patch directly. This concept involves planting metal wires on a small area in the tyre belt which act as electrodes to connect with the steel wires within the tyre carcass. The capacitance change between two adjacent steel wires and the resistance change within the wires while the tyre is deforming are used to determine the force information.

Compared to the inner tyre surface and wheel rim embedded strain sensor, the proposed sensor naturally lay within the tyre belt. This approach could improve the precision of the measurements, as it could overcome the hysteresis between the tyre belt and the inner tyre surface and sidewall (this hysteresis can reduce the precision of the inner tyre surface and wheel rim measurement). Compared to some inner tyre surface embedded strain sensors, another advantage of the proposed sensor is its lower levels of stiffness.

If the strain sensor has higher levels of stiffness than the tyre rubber, it could influence the original tyre deformation and the forces acting on the tyre.

Sami (2010) introduced the APOLLO/FRICTI@N project tyre forces sensor. The sensor is an optical position sensor that uses a position sensitive detector to detect the movement of the LED (as seen in Figure 2.11). The movement of the LED represents the motion and the deformation of the inner tyre surface, and the inner tyre surface deformation is brought about by the forces acting on the tyre. The tyre sensor is theoretically capable of carrying out continuous measurements on the tyre-road contact patch. However, according to Ryosuke et al. (2006), the accuracy and the durability of the sensor could be compromised due to the harsh working environment it would be exposed to.

This item has been removed due to 3rd Party Copyright.This item has been removed due to 3rd Party Copyright. The unabridged version of the thesis can be found in the Lanchester Library, Coventry University

Figure 2.11 APOLLO/FRICTI@N project tyre forces sensor (Sami, 2010)

2.4.3. Niche Force sensor

Tyre-road braking torque representable in wheel force measurement sensor

The FlexiForce sensor developed by Tekscan pushes force measurement to a new practical level. It offers a great opportunity to develop a low cost in-wheel force sensor. The FlexiForce force sensor operates in a very similar way to a strain gauge. The main difference is that the strain gauge deforms with attached element while the The FlexiForce force sensor does not. The sensor acts as a force sensing resistor. As seen in Figure 2.12, the sensor has a very high resistance when it is unloaded. The resistance decreases once a force is applied. The resistance can be measured by an electronic circuit. Both the force versus resistance and force versus conductance characteristics of the sensor are supplied, and they can be used for accuracy calibration.

This item has been removed due to 3rd Party Copyright.This item has been removed due to 3rd Party Copyright. The unabridged version of the thesis can be found in the Lanchester Library, Coventry University

Figure 2.12 Resistance and conductance to force (Tekscan website)

A seen in Figure 2.13, the sensor is constructed of two layers of substrate (polyester) film. A conductive material (silver) is applied to each layer followed by a layer of pressure-sensitive ink. The two layers of substrate are laminated together with adhesive. The silver circle on top of the pressure-sensitive ink is the active sensing area. Silver extends from the sensing area to the connectors at the other end of the sensor forming the conductive leads. The sensor is terminated with male square pins allowing it to be easily incorporated into a circuit. The two outer pins of the connector are active and the centre pin is inactive. The sensor is ultra-thin and very flexible. Its high-temp model (HT201) can withstand up to 200 degrees centigrade and operate in a 4448N force range. Accordingly, it could easily be integrated into a wheel hub in the way seen in Figure 2.14. The current price is $65 which is tantamount to £41 for a pack of 4.

This item has been removed due to 3rd Party Copyright.This item has been removed due to 3rd Party Copyright. The unabridged version of the thesis can be found in the Lanchester Library, Coventry University

This item has been removed due to 3rd Party Copyright.This item has been removed due to 3rd Party Copyright. The unabridged version of the thesis can be found in the Lanchester Library, Coventry University

Figure 2.13 Force sensor brake down

Figure 2.14 In wheel sensor (Tekscan website)

One way to integrate the FlexiForce sensor into an application is to incorporate it into a force-to-voltage circuit. A means of calibration must then be established to convert the output into the appropriate engineering units. Depending on the setup, an adjustment could then be performed to increase or decrease the sensitivity of the force sensor. Figure 2.15 shows a typical calibrated sensor response to applied force.

This item has been removed due to 3rd Party Copyright.This item has been removed due to 3rd Party Copyright. The unabridged version of the thesis can be found in the Lanchester Library, Coventry University

Figure 2.15 Force to sensor output characteristic (Tekscan website)

2.4.4. Environmental Sensors

The environmental sensors are usually based on optics, acoustics and radio frequency. They use the changes in the signal reflectance, polarization and absorption properties caused by the environment. They are usually used to measure the road friction, object distance, rainfall and so on. One popular environmental sensor used on production vehicles is a rain sensor. It is used to detect rain drops and activates the wiper automatically.

A Swedish company called Sensice developed a novel and cheap ice detector based on an infrared spectroscopy. The detector is able to detect dry and wet conditions, ice, black ice and ice/sleet covered with a layer of water.

Audi developed the Audi Road Vision system which uses a laser and an infrared spectroscopy to identify road surface conditions. It can even detect gravel.

Sami (2010) gave the environmental sensors currently under development an extensive review. The main objective of Sami's (2010) was to use the environmental sensors and conventional vehicle dynamic sensors to measure the tyre/road friction. The environmental sensors, including the Road Eye sensor, the LUX laser scanner, the Correvit ground speed camera and the ICOR polarization camera were presented.

A Road Eye sensor is used to detect different road surface conditions and classify them as dry, wet, snow and ice. It is based on absorption spectroscopy which measures the absorption of infrared light at wavelengths between 1320 and 1570 nm. Every road surface condition occupies a specific position on the absorption spectrum, and the Road Eye can classify each one based on an analysis of the reflected wavelengths. The wavelengths are produced by two laser diodes, and active lighting ensures the sensor works in darkness. Nonetheless, this sensor encounters difficulties in distinguishing between water and ice, as their chemical properties are similar.

The LUX laser scanner operates by sending short laser pulses and measuring the time taken for the reflected light to return to the sensor. Then the travel time is used to identify the distance to an object. The sensor could provide true ground speed and vehicle heading when a static object is identified. It uses multi-echo technology, and by splitting each laser pulse into four layers it can filter out reflections coming from rain, dirt and fog.

The VTTís IcOR polarization camera combines a stereo camera with polarization filters. This camera system limits visible spectrum detection bandwidth to below 950 nm, as a system capable of higher bandwidth is considered too expensive to implement on production vehicles. The sensor can be used as an aid to the Road Eye sensor.

As the environmental sensor can estimate the tyre/road friction in front of the vehicle, it can provide the control system with an initial friction measurement that conventional vehicle dynamic sensors cannot. This initial friction measurement can help to prevent the brake pressure control from overshooting at the beginning of the control process.

2.4.5. Sensor fusion

The sensor fusion technique is standard in avionic navigation systems. This technique makes the best use of multi sensors, so it is usually more accurate and efficient than a single sensor. As more and more sensors are used for different vehicle control systems, sensor fusion is increasingly being employed with regards to on-road vehicle control.

Jorge et al. (2010) introduced the fusion level, which is defined as the level at which different sensor measurements are fused. There is a feature level and a full information level, and depending on the fusion level there are two basic algorithm architectures. They are centralized and decentralized. If the sensors' measurements are fused at the feature level, the algorithm has a centralized architecture and the sensors' measurements are fused just after the raw data processing stage when the full measurements are not yet available. Sometimes, this fusion level may not be able to provide the key information before the measurements of all the sensors involved are available. If the sensors' measurements are fused at full information level, the algorithm will have a decentralized architecture. In regard to this architecture the sensors' measurements are fused after the full measurements of each sensor is available. The advantage of this is that the optimal performance of an individual sensor could contribute to the overall system performance, and the degradation of an individual sensor performance does not affect the performance of the other sensors.

Sami (2010) used the sensor data fusion approach for friction measurement. The information comes from environmental sensors, vehicle dynamic sensors, and tyre sensors. The data fusion architecture is decentralized, each sensor catalogue has its own data fusion unit, and the fusion unit which makes the final decision fuses this information to initialise road friction estimations.

Fredrik et al. (2001) used sensor fusion to estimate vehicle velocity by combining the measurements of the wheel speed sensor and the vehicle longitudinal accelerometer. Vehicle acceleration is considered a good complement to the wheel speed signals when non-driven wheel speed signals are not available. This approach gives relatively accurate longitudinal wheel velocity compared to an estimation provided exclusively by the angular speed sensor.

Fredrik et al. (2001) also posited that a vertical accelerometer be used to compensate for a non-horizontal vehicle position (this causes bias regarding the longitudinal accelerometer measurement). Due to regular accelerometers having bias and the unsuitability of scale factors for continuous integration, some other related information could be involved to help overcome those limitations.

Sensor data fusion underpins active safety system interaction. More and more active safety systems are being developed and implemented on modern vehicles. Different active safety systems may use different types of sensors. Efficient integration of these sensors could help to improve the overall system performance, simplify the overall system structure and reduce the overall system cost.

2.4.6. Sensor data process

Due to the presence of nonlinear vehicle dynamics while driving, the raw data from the vehicle dynamic sensors, like the wheel speed sensors for example, usually contain measurement noise. The polluted data cannot be used directly by the ABS. A Kalman filter is a useful tool that can be used to attenuate measurement noise (Fredrik et al., 2001).

Deng et al. (2006) explain that the Kalman filter attenuates measurement noise by giving an accurate estimation of the measurement offset. Recently, the Extended Kalman Filter has been widely used as a nonlinear system state estimation tool. Nonetheless, it requires large computational effort, and it is of limited use when the Jacobians or Hessians matrixes are insufficient to calculate the non-linear system. Still, the Unscented Kalman Filter can overcome the limitations of the Extended Kalman Filter without an increase in computational effort. A comparison of the simulation results suggests that the Unscented Kalman Filter could offer a more accurate estimation than the Extended Kalman Filter.

2.5. ABS modelling and simulation

2.5.1. Computer modelling and simulation

The stability of the ABS performance is very critical. It is influenced by many factors, especially by the suspension system and tyre characteristics. In order to develop an ABS for commercial vehicular use (using a conventional development method) a prototype vehicle must be produced first. Time consuming and costly experimental track testing must then be carried out to establish the control thresholds and validate the performance of the system. Accordingly, the whole development process is long and expensive.

Modern computer modelling and simulation software are the solutions to the stated problems. Simulation software allow an individual system or the whole vehicle to be modelled accurately, even before the production of a prototype vehicle. The ABS can also be modelled and integrated in the vehicle model. The performance can be tested during the track test simulation process. These software are now being used in the development of new vehicles by most automobile companies. After many years of development these software deliver much more accuracy in terms of measuring vehicle performance.

2.5.2. Vehicle model and tyre model

In order to simulate the ABS performance, a proper vehicle model and tyre model must be established and integrated. The performance of the vehicle and the tyre model directly affect the accuracy of the simulation results.

Due to the non-linearity involved in the ABS control process a non-linear dynamic vehicle model and tyre model more accurately simulate ABS performance. The non-linear dynamic vehicle model usually includes the effect of weight transfer during braking, the effects of aerodynamic forces, and other vehicle dynamic effects such as yaw and roll. A non-linear dynamic tyre model should be able to represent the relationship between nonlinear longitudinal tyre braking and the slip ratio, the effect of the vertical load, the rolling resistance, as well as the friction coefficient on the tyre longitudinal braking force. The interaction between the tyre longitudinal braking force and the lateral force while the wheel is turning should also be included in the tyre model.

In most of the literature, straight line braking simulations that neglect any cornering effects were carried out to reduce complexity. In order to simulate the braking and cornering manoeuvres a comprehensive vehicle and tyre model must be used. However, the balance between simplicity and performance should always be considered during modelling.

2.5.2.1. Matlab Vehicle models

The Matlab based vehicle model uses formulas that represent motions and interactions as the inputs. The representation of real vehicle motion in different situations could require very complex equations. The model introduced below is a simple example. The model could be used as a base model if Matlab is used in regard to the simulation software.

Hossein et al. (2010) introduced a simplified continuous non-linear vehicle dynamics model as seen in Figure 2.16. The model has proven capable of representing the essential features of the vehicle when braking while driving in a straight line. If combined braking and cornering simulation needs to be carried out in future research, the motion equations for yaw and roll effects should be included.

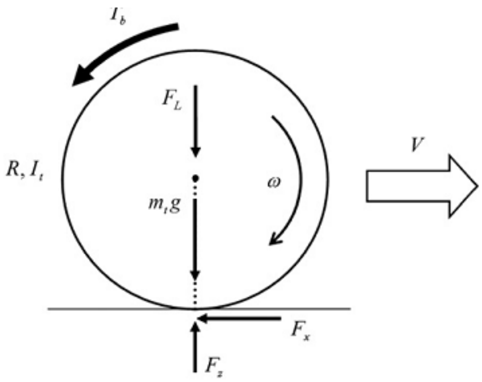


Figure 2.16 Wheel free body diagram during braking

The equations used to describe the model are listed below and the parameter definitions are shown in table 2.1. As seen in equation 2.8, the longitudinal force Fx is described as a function of the longitudinal slip.

$$\dot{V} = \frac{-F_x}{m_t} \tag{2.2}$$

$$\dot{\omega} = \frac{1}{I_t}\left(RF_x - T_b\right) \tag{2.3}$$

$$m_t = \frac{1}{4}m_{vs} + m_w \tag{2.4}$$

$$F_z = m_t g - \frac{m_{vs}h_{cg}}{2l}\ddot{x} = m_t g - F_L \tag{2.5}$$

$$\lambda = \frac{V - R\omega}{V} \tag{2.6}$$

$$\dot{\lambda} = \frac{\dot{V}(1-\lambda) - R\dot{\omega}}{V} \tag{2.7}$$

$$\dot{\lambda} = -\frac{1}{V}\left[\frac{F_x}{m_t}(1-\lambda) + \frac{R^2 F_x}{I_t}\right] + \left(\frac{R}{VI_t}\right)T_b \tag{2.8}$$

V	The longitudinal velocity of the vehicle.	T_b	The braking torque.
ω	The angular velocity of the wheel.	F_z	The vertical force includes both static force and load transfer force due to brake.
I_t	The total moment of inertia of the wheel.	h_{cg}	The height of vehicle gravity centre
m_t	The total mass of quarter vehicle.	l	Wheel base.
m_{vs}	The total sprung mass of the vehicle.	x	The vehicle longitudinal travel distance.
m_w	Wheel mass.	λ	Slip ratio
Fx	Longitudinal tyre force	R	wheel radius

Table 2.1 Quarter vehicle model parameters definitions

Due to the non-linear property of individual vehicle components like suspension, tyres have a great influence on ABS performance, and it is very complex to model such properties in Matlab. Accordingly, the ADAMS (Automatic Dynamic Analysis of Mechanical Systems) model, which is discussed below, is used in this research.

2.5.2.2. ADAMS Vehicle models

ADAMS modelling is used to understand the behaviour of systems containing rigid or flexible parts connected by joints and undergoing large displacement motion (Blundell, 1991). It provides a good performance in vehicle applications, especially in a suspension dynamic simulation.

Bulent (1999) proposed using ADAMS to model a whole vehicle with a detailed suspension system, along with a FIALA tyre model, to simulate the performance of the ABS control method. A FORTRAN subroutine was used to write the ABS control method code. The FORTRAN code is then integrated with the ADAMS vehicle model.

In order to establish an ADAMS vehicle model, static rig measurements are required to measure the vehicle and suspension parameters such as geometry, mass, and the spring-damper properties. This data was then used to calculate other immeasurable or variable vehicle and suspension properties, such as inertia and mass centre. A flat bed tyre test machine can be used to collect data from the tyre model (such as the correspondence between slip and the generation of peak tyre braking force on different road conditions) and it forms an important part of the ABS control method.

The FORTRAN ABS control method code and the method used to implement the control code in the ADAMS vehicle model provide good references for the research in this thesis. Establishing the static rig measurements is a very time consuming process. As this thesis focuses more on the development of a new ABS control algorithm, existing vehicle data for ADAMS modelling is used. A comprehensive tyre model, such as the Magic Formula based tyre model, is used in this research.

The ADAMS vehicle model used in this research is based on a Subaru Impreza. The full vehicle suspension data is available and access to other relevant data is open.

The first stage of the modelling process involves modelling the front suspension and rear suspension individually. The models presented in the ADAMS/View are shown in Figure 2.17 and 2.18. These two suspension models can be aligned to form half the vehicle.

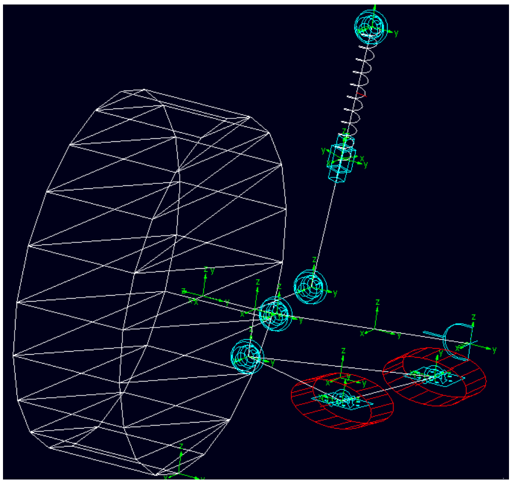


Figure 2.17 Front suspension model

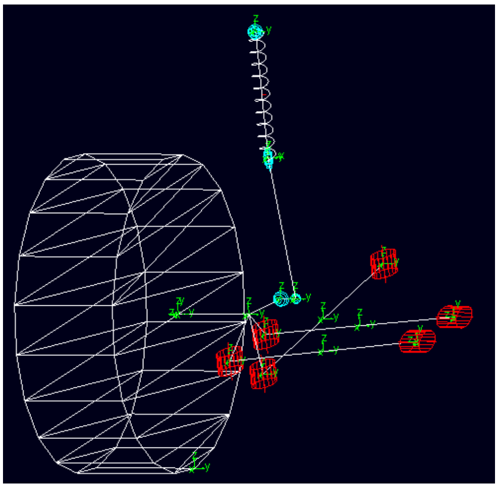


Figure 2.18 Rear suspension model

In the second stage the mirror method is used to model the other half of the vehicle, and the four suspesion models are connected by a vehicle body to form a full vehicle model. In the third stage the steering wheel assembly is added to the front wheel hubs, so the front wheels turn according to steering wheel input. In the fourth stage an anti-roll bar is added as it is fitted on the original vehicle and it influences the vehicle dynamics. Finally, a MICHELIN FIALA tyre model and test track are added to the model externally. The full vehicle model viewed from the top is shown in Figure 2.19. The vehicle body graphic has not been represented here, but it exists and is recogeniseable by the software.

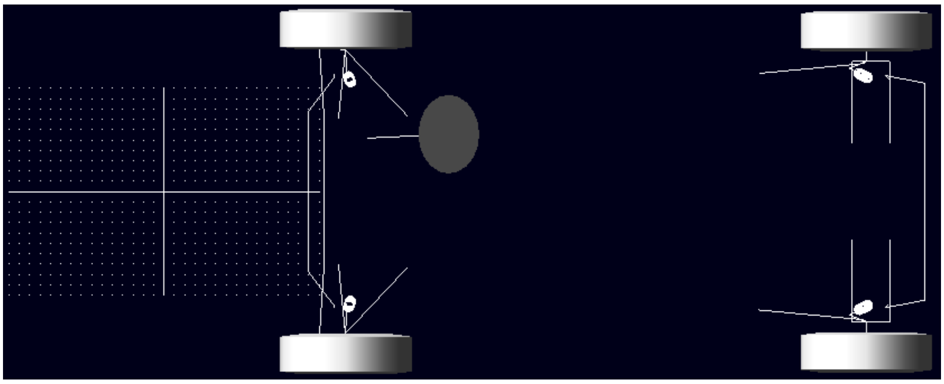


Figure 2.19 Full vehicle model

The model can simulate different vehicle braking manoeuvres such as straight line braking and combined slip braking. The FIALA tyre model was later replaced by a non-linear dynamic tyre model. A test track with different surface friction coefficients was also considered, and the method used to integrate the Matlab Simulink control model was developed.

The tyre model generally relates the tyre longitudinal braking force to the slip ratio depending on the vertical load force, the tyre-road friction coefficient, the tyre characteristics and other factors such as camber and slip angle. There are basically two types of the tyre model; the empirical tyre model and the physical tyre model.

2.5.2.3. Empirical Tyre models

The empirical tyre model is based on measured tyre parameters. The measurements can be carried out using a tyre test rig as seen in Figure 2.20.

Figure 2.20 Tyre test rig

The Magic Formula tyre model - The Magic Formula tyre model is an empirical model. As seen in Figure 2.21, the tyre force and moment curves resemble a sine function wave which has been modified by introducing an arctangent function to ‘stretch’ the slip ratio values on the x-axis. As seen in the Magic Formula (equation 2.9), a series of coefficients are used to fit the measured data on the curve. This provides a functional approximation for lateral force, aligning moment and longitudinal force (Pacejka, 1993).

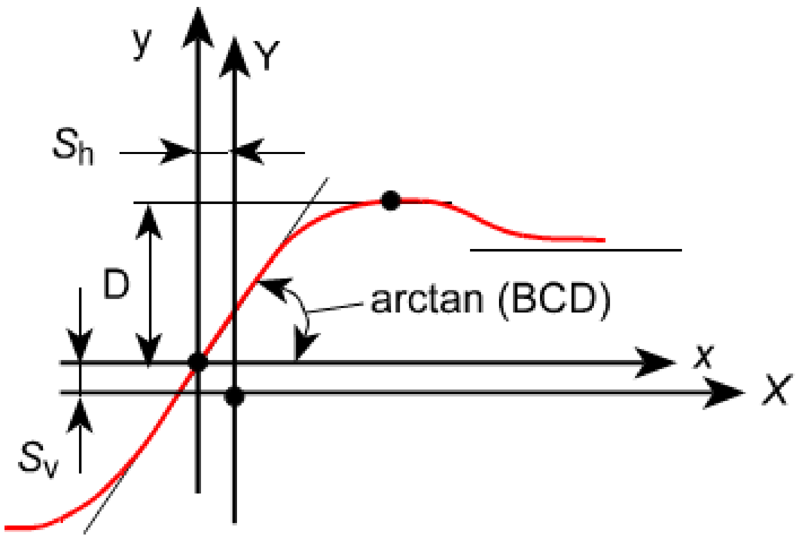


Figure 2.21 Coefficients used in the Magic Formula tyre model

$$Y(x) = D \sin[C \arctan \{ Bx - E (Bx - \arctan(Bx))\}] \quad \textbf{(2.9)}$$

Where $Y(X) = y(x) + Sv$, and $x = X + Sh$.

In equation 2.9, x is the slip angle or longitudinal wheel slip ratio. D is the peak value factor, C is the shape factor used for stretching, B is a stiffness factor, B, C and D are the slope at zero slip, and E is a curvature factor. B, C, D and E are coefficients which depend on vertical force and camber angle. Here, Y could represent the longitudinal force, lateral force or aligning moment. S_v is the vertical shift factor. S_h is the horizontal shift factor.

The tyre longitudinal braking force is a function of the slip ratio. It requires 11 coefficients. The lateral force is a function of the slip angle. It requires 14 coefficients. The aligning moment is as a function of the slip angle and requires 18 coefficients. These coefficients are obtained from steady state testing. The lateral force and aligning moment are measured without any effect due to longitudinal braking force. The longitudinal braking force is measured without any effect of cornering (Blundell, 2004). Further effort to extend the model to combine cornering and braking have been made, and a detailed description of this process is given in a research paper published by Pacejka and Bakker (1993).

Although large numbers of coefficients are required in the Magic Formula, once these coefficients are found, the formula is very representative. The Magic Formula tyre model requires low computational effort, but it is hard to include temperature effect and tyre wear effect within the model. Research is being carried out to overcome these shortcomings.

FIALA tyre model- The FIALA tyre model only requires 10 parameters which relate directly to the physical properties of the tyre. In terms of striking a balance between simplicity and accuracy, the FIALA tyre model could be a good choice for a simulation with lower levels of accuracy. It is also an ADAMS default tyre model.

There are several disadvantages to the FIALA tyre model. The camber angle effects on the lateral force and the aligning moment are not included. A combined cornering and braking or fraction situation cannot be represented. The aligning moment turns to zero at large slip angles, and the tyre asymmetry (conicity and ply-steer) means that small values of lateral force and an aligning moment at a zero slip angle cannot be represented. In addition, cornering stiffness variations are not considered (Bulent, 1999).

The Harty tyre model - The Harty tyre model only requires a low number of tyre property parameters to establish a reasonable representable tyre model. In this tyre model, the camber angle effect is considered and the combined slip effect can also be represented.

The Dugoff tyre model - The Dugoff tyre model is based on a friction ellipse. The model can represent the non-linear tyre characteristics and the interaction between tyre longitudinal and lateral forces (Hossein et al. 2010).

The Dugoff tyre model represents the longitudinal force as F_x the lateral force as F_y with the slip ratio λ, slip angle α, and vertical force F_z. C_i is the longitudinal stiffness and C_α is the cornering stiffness. The parameter f(S) is defined as shown in the following equations:

$$F_x = \frac{C_i \lambda}{1-\lambda} f(S) \quad \textbf{(2.10)}$$

$$F_y = \frac{C_\alpha tan\alpha}{1-\lambda} f(S) \quad \textbf{(2.11)}$$

$$\mathrm{f(S) = S(2 - S), if\ S < 1} \quad \textbf{(2.12)}$$

$$\mathrm{f(S) = 1, if\ S > 1} \quad \textbf{(2.13)}$$

$$\mathrm{S} = \frac{\mu F_z\left[1-\varepsilon_r\sqrt{\lambda^2+tan^2\alpha}\right](1-\lambda)}{2\sqrt{C_i^2\lambda^2+C_\alpha^2 tan^2\alpha}} \quad \textbf{(2.14)}$$

2.5.2.4. Physical tyre model

A physical tyre model represents the physical tyre forces involved in the tyre-road interaction using a mathematical model. A simple example is to consider the tyre as an idealised radial array of elements (such as brush elements). These elements represent given physical properties, such as elasticity in different directions and the constraints resulting from interactivity with adjacent elements.

FTire model - The FTire (Flexible ring tyre) tyre model is a nonlinear in and out of plane tyre simulation model and it can be used for vehicle ride comfort, vehicle durability, and other vehicle dynamic related simulations on even or non-even roads.

In the FTire model, as seen in Figure 2.22, a flexible and extensible tyre ring-belt is used to represent the physical processes involved in the tyre-road contact patch. The ring/belt carries bending stiffness formed by distributed dynamical stiffness in the radial, tangential, and lateral direction on the rim. The ring-belt is approximated by a finite number of ring/belt elements, and the elements are connected to each other by stiff springs and bending stiffness, both in and out plane. On each of the ring/belt elements, there are numbers of mass-less 'tread blocks' which carry nonlinear stiffness and damping properties in the radial, tangential, and lateral directions. The tyre forces and moments are calculated by integrating the forces in the elastic foundation of the belt (Michael Gipser, 2000).

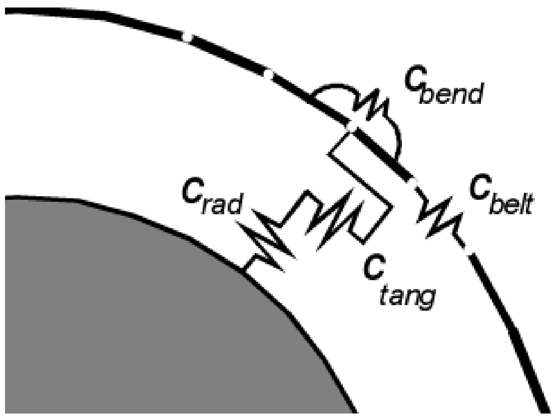
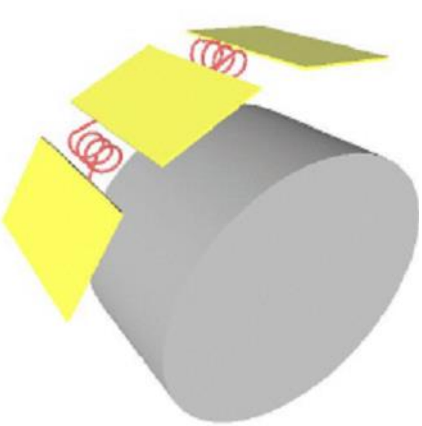


Figure 2.22 Simplified in-plane FTire model

The in-plane bending stiffness is represented by a torsional spring in the lateral axis as seen in Figure 2.23. And the out-plane bending stiffness is represented by a torsional spring in the radial axis as seen in Figure 2.24.



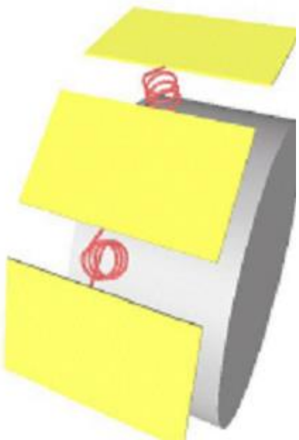


Figure 2.23 In-plane mechanism   Figure 2.24 Out-plane mechanism

The FTire tyre model uses the static and modal tyre properties to establish the model, so the data collection is relatively easy. Nonetheless, the model requires a large amount of tyre parameters which makes the data collection process complex and time-consuming. The parameters required in the FTire tyre model can be found in the "Using the FTire Tyre Model" guidance book (Ftire website). An example of FTire model based simulation is seen in Figure 2.25, which shows the results of a tyre running over very short longitudinal wave-length and sharp-edged obstacles. Compared to the real test data, the simulation results demonstrate sufficient accuracy.

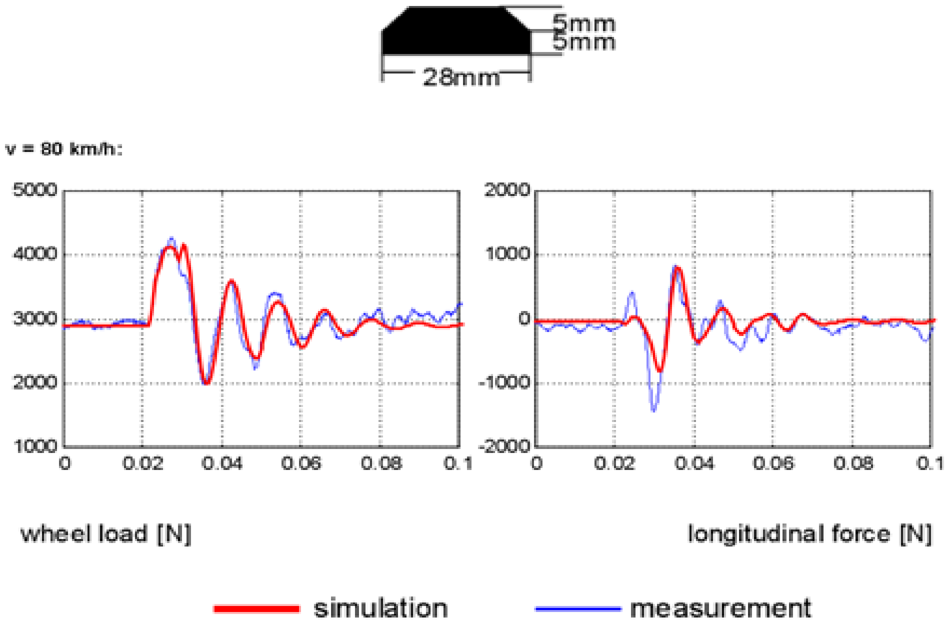


Figure 2.25 Validation of FTire model

Sharp tyre model - The Sharp tyre model is also called a Multi-radial-spoke tyre model. In the model, the tyre is assumed to consist of many radial spokes. The spokes can deform in the radial, lateral and circumferential directions, so stiffness is assigned to each direction. These deformations are not influenced by the position of neighbouring spokes.

The vertical load, slip angle and camber angle, along with the deformation stiffness are used to calculate the directional force on each spoke. If the measured force on one spoke does not exceed the sliding limit force, the calculation will proceed to next spoke. If the sliding limit force is exceeded, the model allows the single spoke to slide until the force equilibrium is reached, then the calculation goes on. The measured directional forces are added together to ascertain the total number of directional forces acting on the tyre (Sharp, 1999).

Only seven physically meaningful tyre parameters are required to establish this tyre model, but it could only be used for low accuracy modelling. Once again, the compromise between simplicity and accuracy should be considered when using this tyre model.

2.6. Hardware in the loop (HiL) simulation

HiL is at a higher stage of development than pure computer simulation. Hardware in the loop simulation (HiLS) is an effective tool to evaluate the design and performance of a newly developed vehicle subsystem like ABS (Kimbrough et al., 1996). The HiLS couples the computer model with physical hardware in real-time. With regards to the HiL system, real ABS hardware can interact with the software vehicle model, which permits complex phenomena to be investigated for the implementation of ABS in a vehicle without testing on a real vehicle. This saves ABS development time and reduces costs while at the same time facilitating greater flexibility.

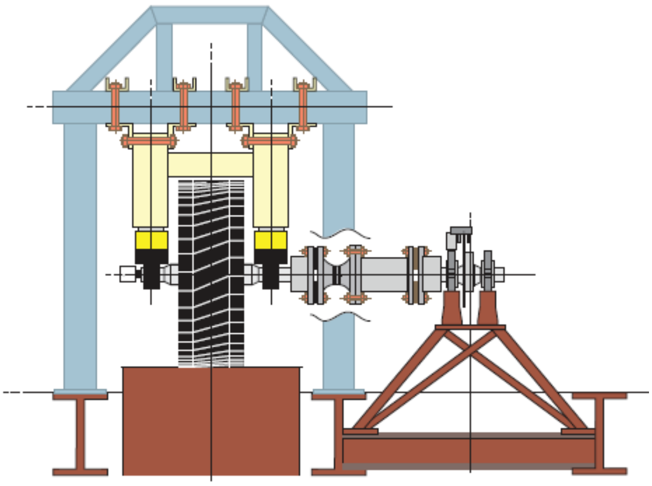


Figure 2.26 HIL ABS test rig

Damiano et al. (2009) proposed a hardware in the loop ABS test bench, as seen in Figure 2.26. A dSPACE simulator is used to generate a control signal and transmit it to the actual brake system. The feedback to the simulator is provided via signals from the on-board sensors. To use the dSPACE system, Matlab/Simulink is usually used for vehicle modelling.

2.7. Conclusion from literature review

Due to the non-linear nature of vehicle and tyre dynamics, most vehicle longitudinal speed estimation methods may not be sufficiently accurate in different circumstances. A vehicle longitudinal acceleration sensor can provide a good aid to the vehicle longitudinal speed estimation. Due to the limitations of sensor technology, the direct measurement of vehicle longitudinal speed remains difficult. Niche sensors used to measure vehicle longitudinal speed, like radar and GPS, are too expensive to implement on production vehicles.

Control parameters and a control phase switch mechanism other than vehicle longitudinal speed or slip ratio should be established. In a straight-line braking situation, the wheel longitudinal deceleration and wheel angular acceleration against time plots can be studied to determine their relationship at optimal slip ratio. The wheel angular acceleration curve may be affected by the effective tyre rolling radius, so the wheel longitudinal deceleration and wheel angular acceleration relationship may not be robust enough to function successfully as a control phase switch mechanism. The wheel longitudinal deceleration rate and the wheel angular acceleration rate relationship could form the basis of a solution as they would allow the effective tyre rolling radius to be eliminated from the process. The linear relationship between tyre longitudinal braking force and the slip ratio within the stable brake region could be used in the analysis of the relationship between the wheel longitudinal deceleration rate and the wheel angular deceleration rate.

The rule based control approaches, such as the Bosch ABS, are too complex. But the so-called hybrid control method that based on the combination of the slip ratio and the wheel angular acceleration control rule has great advantages. As the hybrid control phases switch control method is simple and robust it will be the focus of future research. The hybrid, two-phase ABS control method proposed by Mathieu et al. (2010) is promising. It uses wheel angular acceleration as a control parameter. It also uses measured tyre longitudinal force as the control phases switch mechanism, but this approach might be impractical as the smart wheel used to measure the force is expensive and hard to implement on a production vehicle. It is not desirable to fit two or four heavy wheels that are equipped with various types of electronic components on a standard road car. The smart wheel also experiences difficulties with wireless data transmission, sensor power supply and sensor durability, as large amounts of force are experienced by the sensor. Finally, if this control switch method is implemented, a low friction surface ABS control method would have to be developed to overcome the aforementioned limitations.

More and more advanced sensors are now available to measure the vehicle dynamics and the environment. Sensor fusion is a good method which effectively integrates the measurements from different sensors, along with Kalman based filters that clean the polluted measurement signals. But sensor implementation on a production vehicle must follow the optimal cost to performance ratio rule. An advanced but expensive sensor should not be used, unless it offers significant performance improvement.

Nonlinear dynamic vehicle and tyre models could arguably be good substitutes for a real vehicle experiment test, especially in complex vehicle manoeuvre simulations. But the balance between simplicity and accuracy should be considered in the model selection. In the developed ADAMS vehicle model, due to the need for real test representative tyre road interaction during complex vehicle manoeuvres, the non-linear dynamic Magic Formula tyre model is implemented.

As summary, a hybrid ABS control method that uses wheel angular acceleration and any possible in-wheel sensor measurement information is going to be proposed in this research. The proposed ABS should be less complex than the commercialised ABS, and it should be more practical than the pure theoretical ABS. A representable ADAMS vehicle model with a Matlab Simulink control model that represents the hybrid control method is going to be used to test the proposed ABS control method.

3. Vehicle Modelling & Simulation Set Up

The proposed method to develop and test an ABS control method in this research is simulation based. In order to establish an appropriate simulation environment, a representative vehicle model that can simulate the complex tyre and road interaction in parallel with vehicle body, suspension system and tyre interaction must be established. The vehicle model must include a detailed brake disk and pad implantation model. The emergency braking environment must also be established.

Analyses of the physical and mathematical aspects of basic braking and an emergency braking scenario are used to establish these models. Initial simulations based on the established models are also carried out to achieve a thorough simulation environment for the purposes of this research.

3.1. ADAMS Model Simulation

3.1.1. The Vehicle Model

The vehicle model used in this research is based on a Subaru Impreza. The model was first built in the code-based dataset file (adm) format and the Fiala tyre model was implemented. But due to the limitations described in the previous literature review, the more complex Magic Formula tyre model is required to simulate the tyre slip effect and the generation of related forces. The implementation of the Magic Formula tyre model in the dataset file (adm) format model is very difficult due to the large amount of data it involves. Accordingly, the vehicle model was rebuilt in command file (cmd) format to implement the Magic Formula tyre model. Figure 3.1 provides a detailed view of the model.

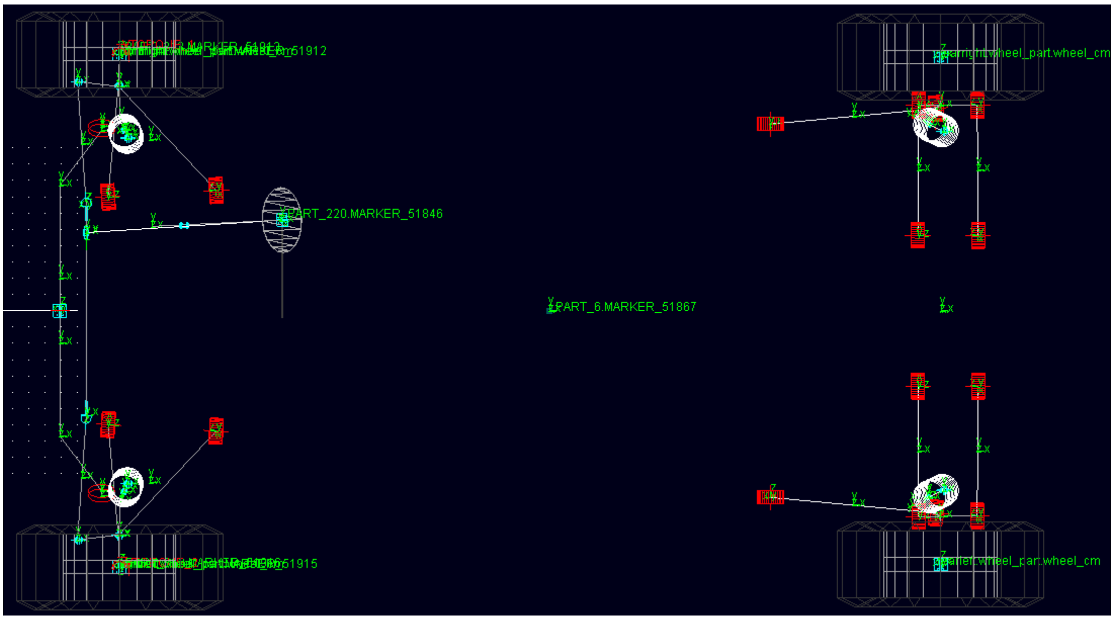


Figure 3.1 Full ADAMS vehicle model

3.1.2. Model Validation

The model validation data is from a real track test experiment performed with a Coventry University research vehicle as seen in Figure 3.2. It is a Subaru Impreza, which is the base car of the ADAMS model used in this research. The test experiment is designed to ascertain the influence of vehicle mass distribution and inertial properties on handling and stability. The tests were performed using three individual masses that were attached to different locations on the test vehicle. A change in the distribution of the three masses alters the vehicle dynamic index values without changing the location of the centre of mass or the total mass of the vehicle. The main purpose of this test is to test the yaw moment in regard to the inertia influence on the transient-state handling of a vehicle. The steering wheel angle is the input and the yaw velocity is the output.

Figure 3.2 Coventry University track test vehicle

The first set of the data is collected at a speed of 40mph and a 0.91 dynamic index with various steering input. Figure 3.3 shows the real experiment test data plot. Figure 3.4 shows the ADAMS model simulated test data plot. As shown, the simulated test data has a 1 second delay compared to the real test data. The intention is to settle the vehicle model's dynamic behaviour during the initial stage of simulation. It is also noticeable that although the simulated test data plot follows the main trend of the real test data plot, the simulated data plot is smoother. This is due to absence of external disturbances and road imperfections in the simulation model.

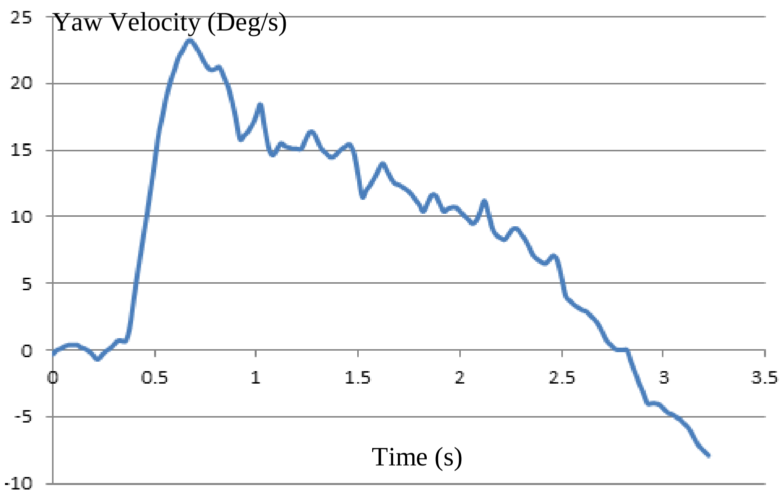


Figure 3. 3 40MPH Track test data

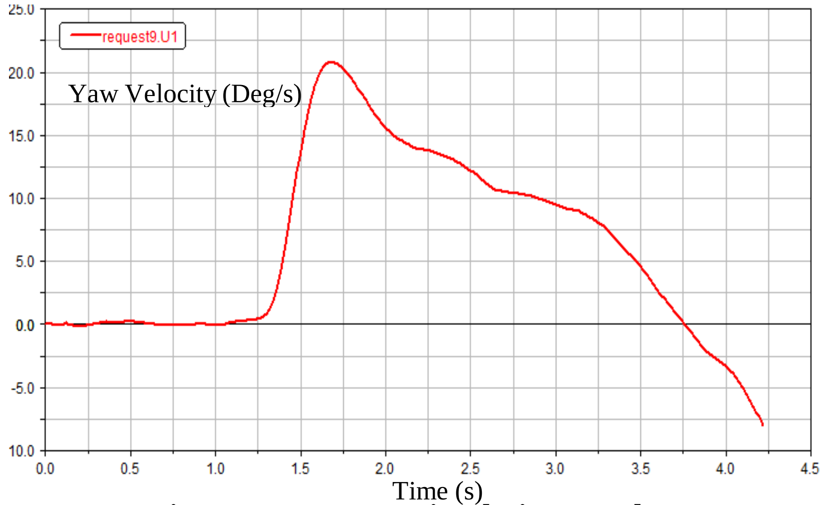


Figure 3.4 40MPH Simulation test data

The second set of the data is collected at a 60mph vehicle speed and a 0.91 dynamic index with various steering input. Figure 3.5 shows the real experimental test data plot. Figure 3.6 shows the ADAMS model simulated test data plot. As can be seen, the simulated test data plot follows the main trend of the real test data plot with slight value offset.

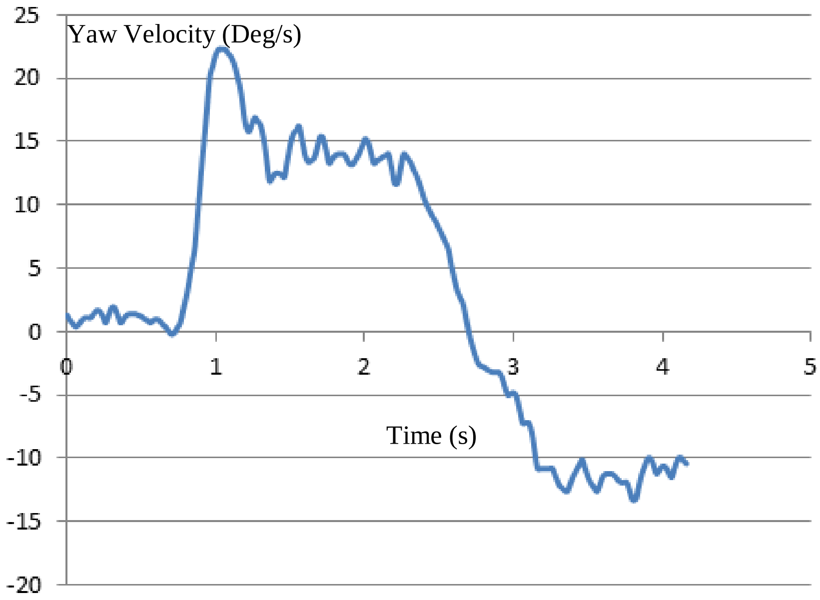


Figure 3. 5 60MPH Track test data

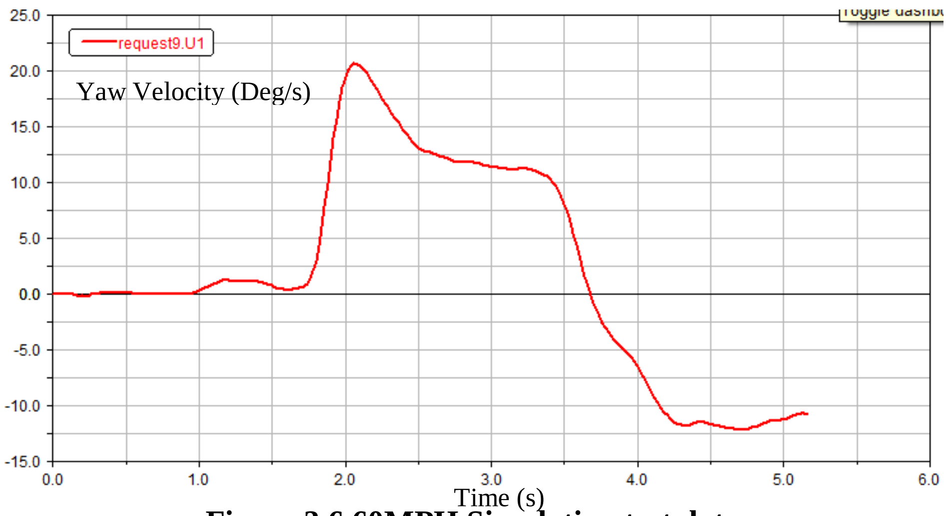


Figure 3.6 60MPH Simulation test data

By comparing the two sets of real and simulated test data, it can be concluded that the Subaru Impreza ADAMS model used in this research can be used to represent the dynamic behaviour of the real vehicle.

3.2. Simulation Set Up

3.2.1. Braking Physics

During the braking process, there are two important braking forces. The first braking force is generated by the brake system, which is in turn generated between the brake disk and the brake pad and is called the disk-pad braking force. The second braking force is a transmitted force whereby the disk-pad braking force is transmitted to the road surface through the tyre. It is called tyre-road braking force.

3.2.1.1. Disk-pad braking force physics

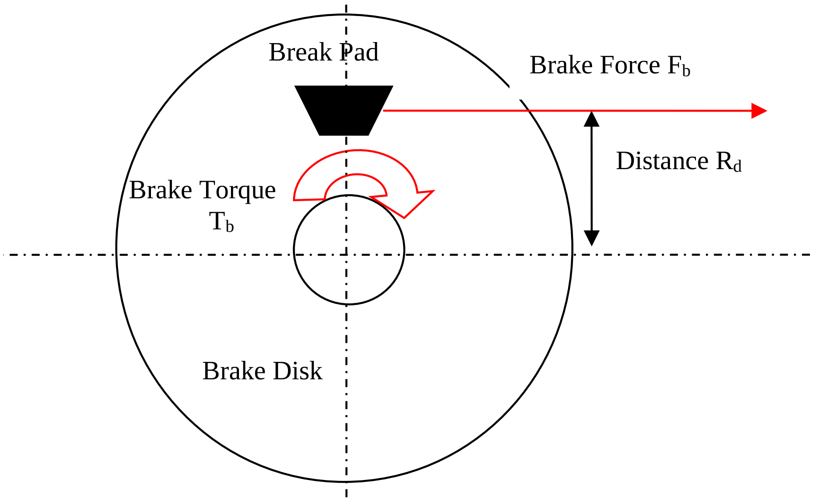


Figure3.7 Brake force physics

As seen in Figure 3.7, the force transmitted to the brake pad by the brake piston is represented by equation3.1. The frictional force between the brake disk and the brake pad is represented by equation3.2. The brake torque is represented by equation3.3. Table 3.1 explains the parameters presented in the equations.

$$F_p = P*A \tag{3.1}$$

$$F_b=F_p*\mu*n \tag{3.2}$$

$$T_b = F_b *R_d \tag{3.3}$$

F_p	The force transmitted by brake piston.	T_b	The braking torque.
F_b	The force between brake disk and brake pad.	P	The brake pressure.
A	The area of brake piston.	μ	The friction coefficient between brake pad and disk
n	The number of brake pad	R_d	Brake pad to wheel center distance

Table 3.1 Disk-pad braking force physics related parameters

The brake system converts the kinetic energy of the vehicle to thermal energy (heat) via the brake disk, brake pads and other related parts. Most of the generated heat dissipates into the environment. Some of the excess heat that cannot be dissipated will build up, and the resulting increase in heat will influence the disk-pad braking performance.

3.2.1.2. Tyre-road braking force physics

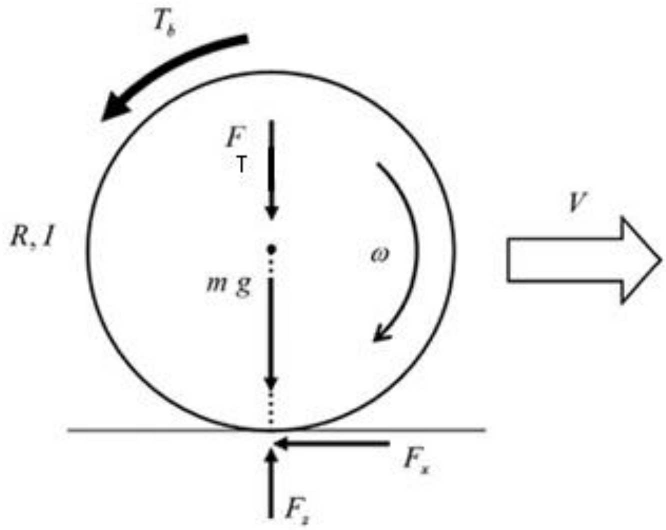


Figure 3.8 Wheel free body diagram during braking

As seen in Figure 3.8, F_x is the braking force between the tyre and the road surface, and it is a frictional force. The equation 3.4 is the friction equation. The value of F_x mainly depends on the tyre-road friction coefficient μ and the tyre vertical load forceF_z. There are static and sliding frictional forces. For the same tyre on the same road surface the static tyre-road friction force is generally higher than the sliding tyre-road friction force. During the braking process, the tyre-road friction transitions into the total sliding friction (wheel fully locked) from the static friction (when disk-pad braking force is just applied). The general ABS controlled slip ratio range is around 20 percent, so the tyre-road friction can be considered static friction during an effective ABS control process. For the same tyre on the same road surface, the tyre-road friction coefficient is generally considered fixed, although the Bosch brake system handbook indicates that vehicle speed change can slightly influence the tyre-road friction coefficient (Bosch 2009).

$$F_x = \mu F_z \tag{3.4}$$

The vertical force F_z is another influencing factor on tyre-road friction force. When analysing the dynamic performance of the vehicle braking process, there are two forces included in F_z. There is the normal vertical load force mg and the extra vertical load force caused by the vehicle weight transfer, which is represented by equation 3.5 (Single wheel). Table 3.2 explains the parameters presented in the equation. During a normal vehicle braking process, weight transfer force is added to the normal vertical force on the front axle, and weight transfer force is subtracted from the normal vertical force on the rear axle. During hard braking, the weight transfer force can cause a significant change in the vertical load force and thus influence the tyre-road braking force.

$$F_T = \frac{m \times h_{cg} \times a}{2L} \tag{3.5}$$

V	The longitudinal velocity of the vehicle.	T_b	The braking torque.
ω	The angular velocity of the wheel.	F_z	The vertical force includes both static force and load transfer force due to brake.
I	The total moment of inertia of the wheel.	h_{cg}	The height of vehicle gravity centre
m	The total mass of quarter vehicle.	L	Wheel base.
a	Wheel longitudinal acceleration.	λ	Slip ratio
Fx	Longitudinal tyre-road braking force	R	wheel radius

Table 3.2 Tyre-road braking force physics related parameters

3.2.2. Analysis Based on Basic Physics

As stated previously, the tyre-road braking force depends on its vertical force, the road surface friction coefficient and the slip ratio. During initial hard braking, the tyre-road braking force and the disk-pad braking force increase simultaneously, but there is a point beyond which the tyre-road braking force cannot increase any further. Regardless, the disk-pad braking force continues to increase, the two forces resulting torques are shown in Figure 3.9. This tyre-road braking force behaviour is usually related to the tyre slip ratio as seen in Figure 3.10. The maximum tyre-road braking force is generated at the optimal slip ratio. After passing the optimal slip ratio, the tyre-road braking force starts to decrease and the wheels lock up quickly. At the same time, the tyre lateral force generation capability decreases continually, so the steering performances are compromised. Disk-pad braking force is usually designed with a higher capability than variable tyre-road braking forces in order to cope with various braking situations.

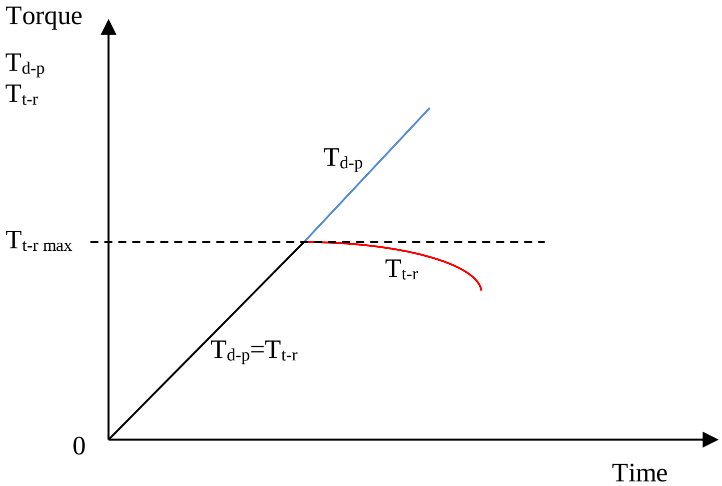


Figure 3.9 The two braking forces resulted torque characteristic

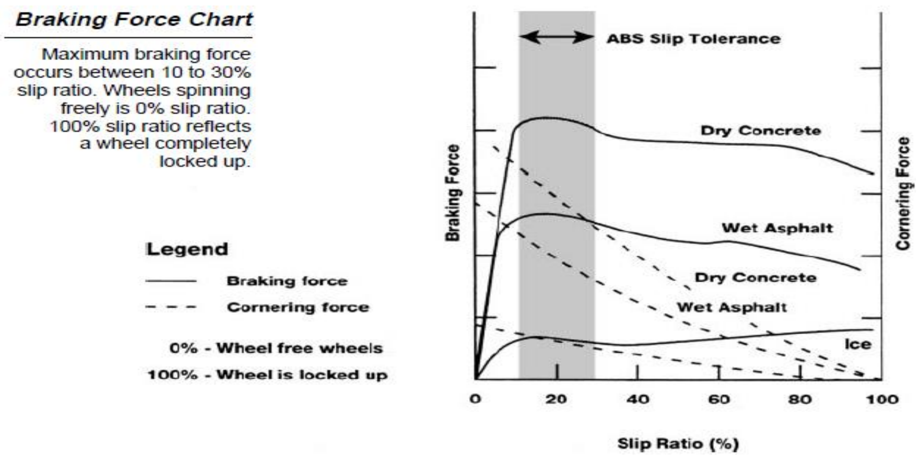


Figure 3.10 Tyre-road braking force characteristics on different road surfaces

The basic principle of ABS control is the control of the slip ratio. The purpose of the slip ratio control is to modulate disk-pad braking force in order to keep tyre-road braking force within its maximum range; and within that range the tyre lateral force is also kept within a functional range. Accordingly, the vehicle can expect the best brake performance while maintaining steering ability.

In accordance with Newton's Second Law equation (equation 3.6), where F is the tyre-road braking force, a is the wheel longitudinal acceleration, and m is the mass the single wheel braking force must stop. In the formula, the tyre-road braking force is related to the wheel longitudinal acceleration. The relationship between them would be uncomplicated if the m were constant, but weight transfer during braking will shift the mass.

$$F = ma \tag{3.6}$$

When considering the basics of weight transfer, the transferred weight is mainly from the vehicle body (Sprung mass). The total load vertical force F_z acting on the two front wheels is presented in equation 3.7, the weight transfer F_T is presented in equation 3.8, and the total vertical load force equation is converted into equation 3.9. The tyre-road braking force F_x is represented by equation 3.10, and along with the mass equation 3.11 and equation 3.9, the tyre-road braking force is converted into equation 3.12. If the total vehicle mass m_v, earth gravity g, height of centre gravity h_{cg} and vehicle track L are considered constant. And when the front wheel longitudinal acceleration shares the same direction as the tyre road braking force, the wheel longitudinal acceleration should follow the same trace as tyre road braking force.

$$F_z = \frac{1}{2} m_v g + F_T \tag{3.7}$$

$$F_T = \frac{m_v \times h_{cg} \times a}{L} \tag{3.8}$$

$$F_z = \frac{1}{2} m_v g + \frac{m_v \times h_{cg} \times a}{2L} \tag{3.9}$$

$$F_x = m_v a \tag{3.10}$$

$$m_v = \frac{F_z}{g} \tag{3.11}$$

$$F_x = (\frac{1}{2} m_v + \frac{m_v \times h_{cg} \times a}{2Lg}) a \tag{3.12}$$

3.2.3. Brake Manoeuvring Simulation

3.2.3.1. Braking Time

As described in the European Union vehicle safety regulations, the vehicle braking system actuation time, which is the time it takes to reach 10% of the maximum designed braking pressure should be less than 0.2 seconds, and 75% of the maximum designed braking pressure should be reached within 0.4 seconds, and should not take more than 0.6 seconds.

The braking time in this research means the period from the first effective disk-pad braking force application until the maximum designed braking force is reached. The braking time varies in different situations and with regards to different physical brake system designs. The typical braking time range found in the references will be used as a guideline for the simulation.

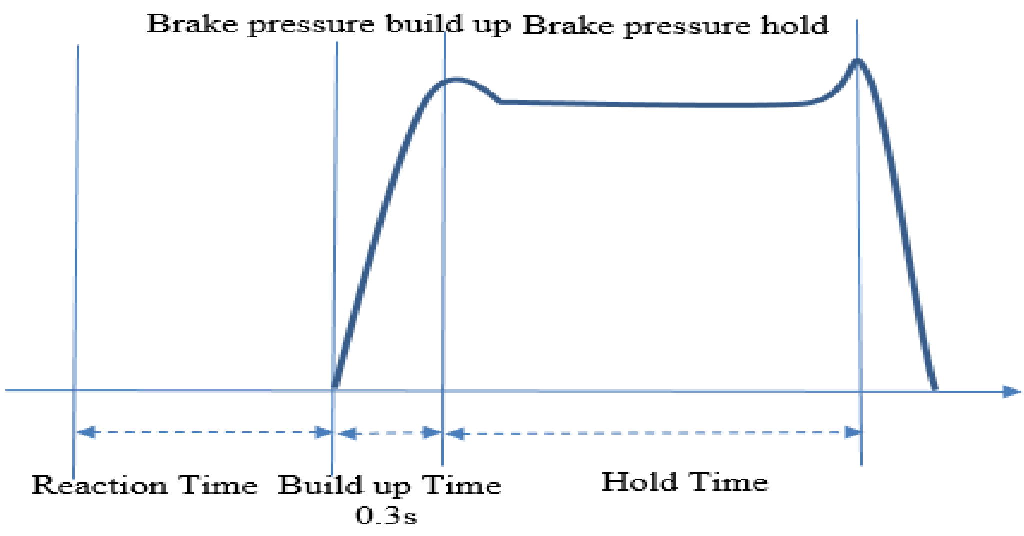


Figure 3.11 Driver brake time

As seen in Figure 3.11, the brake time splits into three phases. Phase one is the reaction time, which includes eye detecting time, brain reacting time, and foot force implementation time. The second phase is the braking force build up. The third phase is the maximum braking force holding time. This research focuses on braking force interaction in phase two. This facilitated the discovery that the maximum designed braking force can be reached in 0.3 seconds during emergency braking.

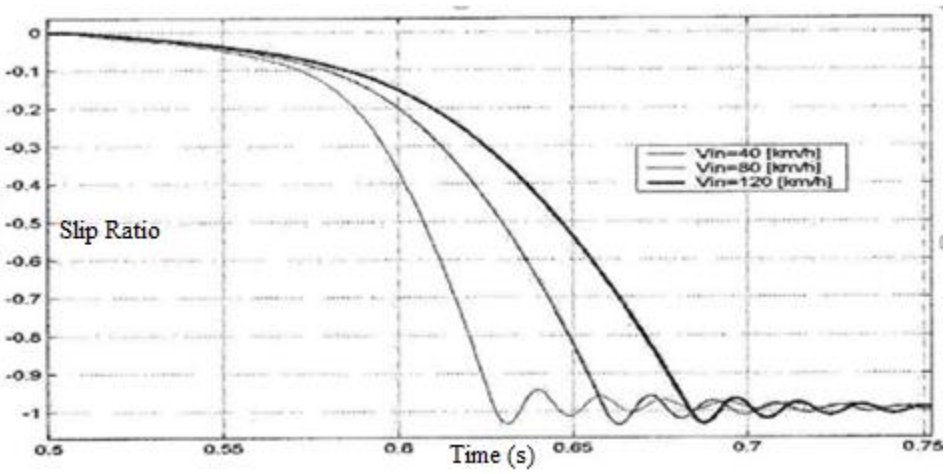


Figure 3.12 Typical slip ratio to time curve under various speeds

Figure 3.12 shows a typical slip ratio to time curve during the emergency braking process at different vehicle longitudinal speeds. As shown, in a typical emergency braking process when initial vehicle speed is 80km/h, the typical optimal slip ratio (10%-25%) could be reached in 0.56 to 0.63 seconds.

Based on all the above information the simulation disk-pad braking force build up time range is set between 0.3-0.6 seconds, but this will be adjusted to meet the purpose of the test during the simulation.

3.2.3.2. Braking force

According to the braking force versus slip ratio theory when the optimal slip ratio is passed the tyre-road braking force should start to reduce. This means that a further increase in the disk-pad braking force after the optimal slip ratio is reached will cause a drop in the tyre-road braking force. As mentioned in the literature review, this drop action could be used to switch the control phase in ABS control.

During braking, the natural interaction between the tyre and the road is very complex, but simulating this complex interaction properly is a key goal of this research. Therefore, a real scenario representative disk-pad braking force simulation is required. The origin of the disk-pad braking force acts between the wheel knuckle and the wheel, and its final output format is a torque. Accordingly, in order to achieve a natural interaction between the tyre and the ground, a rotational torque, which acts between the wheel knuckle and the wheel centre is used to simulate disk-pad braking torque.

Based on Newton's Second Law equation as seen in equation 3.6, $\boldsymbol{F}$ is the braking force required in order to stop the test vehicle travelling at a certain speed within a predefined distance; $\boldsymbol{m}$ is the total kerb weight of the test vehicle, and $\boldsymbol{a}$ is the vehicle acceleration, which is negative when braking. The kerb weight of the vehicle, $\boldsymbol{m}$ is 1071.2kg, which was calculated using the ADAMS model.

Figures 3.13 and 3.14 indicate typical braking distances during emergency braking when a vehicle is travelling at a specified speed in both dry and wet road conditions. If the vehicle initial speed $\boldsymbol{V_i}$ is derived from the two figures, the final speed $\boldsymbol{V_f}$ will be zero as the vehicle stops, and the braking distance $\boldsymbol{S}$ is derived from the two figures. The equation 3.12 which describes the relationship between vehicle speed, braking distance and longitudinal acceleration, can be used to calculate the longitudinal acceleration a. The converted equation is 3.13. If the chosen initial speed $\boldsymbol{V_i}$ is 60km/h, which equals 16.67m/s, the braking distance on a dry road $\boldsymbol{S}$ is 21.4m (only the braking distance is of interest as the simulation focuses on the braking process). The average longitudinal deceleration $\boldsymbol{a}$ is 6.49m/s^2.

$$\boldsymbol{V_f}^2 = \boldsymbol{V_i}^2 + 2\boldsymbol{aS} \tag{3.13}$$

$$\boldsymbol{a} = \frac{\boldsymbol{V_f}^2 - \boldsymbol{V_i}^2}{2\boldsymbol{S}} \tag{3.14}$$

Speed (km/h)	Reaction distance (m)	Braking distance (m)	Total stoppingg distance (m)
30	5.5	5.3	10.8
50	9.2	14.8	24.0
60	11.0	21.4	32.4
80	14.7	38.0	52.7
100	18.3	59.4	77.7
120	22	85.5	107.5

Source Transport Research Laboratory, UK, 2007, © Road Safety Authority, 2007

Figure 3.13 Dry road braking distance

Speed (km/h)	Reaction distance (m)	Braking distance (m)	Total stoppingg distance (m)
30 km/h	5.5	9.4	14.9
50 km/h	9.2	26.1	35.2
60 km/h	11.0	37.5	48.5
80 km/h	14.7	66.7	81.4
100 km/h	18.3	104.3	122.6
120 km/h	22	150.2	172.2

Source Transport Research Laboratory, UK, 2007, © Road Safety Authority, 2007

Figure 3.14 Wet road braking distance

When substituting ***a*** and ***m*** in equation 3.6, ***F*** equals 6952N. This is the total braking force required between the tyre and road surface to bring the vehicle to a full halt when travelling at 60km/h, within a distance of 21.4m on a dry road.

The tyre-road braking force then needs to be split between the front and rear axles in a ratio that is due to the weight transfer resulting from the braking manoeuver. In the simulation, a typical passenger vehicle dry road braking force distribution ratio of 60% at the front, and 40% at the rear is used. In this case, the front axle braking force is 4171N and the rear axle braking force is 2781N. When distributed between the two wheels on each axle, each front wheel is subjected to a braking force of 2085.5N, and each rear wheel is subjected to a braking force of 1390.5N.

Due to the nature of factors like braking force distribution and vehicle weight transfer, which impact during a manoeuver, the disk-pad braking force generated tyre-road braking forces are different on the front and rear axle. In order to simplify the simulation process and to study each of the wheel's forces, speed, acceleration and slip ratio during braking, a full braking force will be applied on one axle at a time. The wheels on the same axle are assumed to share the same braking force properties. Individual wheel forces and the acceleration interaction is of interest in this research.

As the data provided in the above references is representative of typical test data, the simulation in this research may not exactly represent the same test environment and vehicle properties. Therefore, the calculated braking force could only be used as guidance. The trial and error method should be employed to ascertain the proper disk-pad braking force, which can bring the tyre-road braking force up to its peak value within the simulation.

3.2.3.3. Braking torque

The full tyre-road braking force required on the front axle is 6952N, which divided by two wheels means that the braking force on each front wheel is 3476N. To convert the tyre-road braking force to the braking torque, which is applied to the centre of the wheel, the braking force is multiplied by the rolling radius of the wheel $\boldsymbol{R}$. In the ADAMS tyre model data, the unloaded tyre radius is 0.3169m, but the rolling radius is slightly different. To establish the rolling radius, the vehicle model was taken for a 5 second test run, and the average rolling radius was found to be 0.307m. Accordingly, the braking torque applied to the centre of the wheel is $3476 \times 0.307 = 1067\text{Nm}$, and this torque could be used as guidance for the minimum disk-pad braking torque applied to the wheel centre by the brake system. As discussed previously, the disk-pad braking torque must have a greater capability than the tyre-road braking torque generation capability of a tyre. Based on the trial and error method and considering the calculated guidance disk-pad braking torque, 1050Nm will be used as start disk-pad braking torque, then 50Nm will be added to each new simulation set up to ascertain the influence of each different maximum disk-pad braking torque on the tyre-road braking torque. This will facilitate an over braking scenario for ABS activation.

3.2.4. The Simulation Test Establishments

As discussed previously, the simulation disk-pad braking torque build up time will be set between 0.6s and 0.3s. Each new simulation set up adds a 0.1s increment to the test braking time, and in every simulation set up several different disk-pad braking torques are tested. The centre wheel disk-pad braking torque will be set around 1050Nm as a minimum, then 50Nm will be added to each new simulation set up, while the braking time will remain the same. The maximum test disk-pad braking torque will be decided during the test.

During the first 1 second of the simulation no braking torque is applied in order to allow the model's initial dynamic motion to settle. Therefore, a clear difference can be seen in the data trace once the braking torque is applied.

In the first simulation, where the disk-pad braking torque build up time is set to 0.6s the trial and error method is used to discover if the calculated guidance disk-pad braking torque is sufficient. The initial test disk-pad braking torque is lowered to 950Nm to get a better overview of how different disk-pad braking torques influence the tyre-road braking force. The test results show that at a 1050Nm disk-pad braking torque or below, the tyre-road braking force trace does not show any significant drop action This indicates there is no over braking, which can trigger ABS control. So in a further test, 1050Nm will be confirmed as the minimum applied disk-pad braking torque. There is another force, which is in the same orientation as the tyre-road braking force, but on the wheel hub's revolute joint (which represents wheel bearing on a real vehicle) is of interest in this research in terms of establishing how accurately it could represent tyre-road braking force trace. Accordingly, the revolute joint force trace is taken during simulation.

Table 3.3 and 3.4 summarise maximum tyre-road braking force, optimal slip ratio, and the maximum wheel revolute joint force of the Subaru vehicle model during straight line emergency braking on a dry road (with the maximum disk-pad braking torque and its build up time as established above).

Simulation results data

Braking Time / Braking Torque	1.9-2.5s (0.6s)	2.0-2.5s (0.5s)
1050Nm	2.484s→ F=3315.9529N Slip Ratio = 23.12% 2.484s Revolute joint force reaches max: 3301.6631N	2.493s→ F=3319.5234N Slip Ratio = 22.75% 2.494s Revolute joint force reaches max: 3305.9522N
1100Nm	2.448s→ F=3323.1114N Slip Ratio = 25.58% 2.448s Revolute joint force reaches max: 3309.8805N	2.457s→ F=3327.1842N Slip Ratio = 23.72% 2.458s Revolute joint force reaches max: 3313.7639N
1150Nm	2.421s→ F=3326.9076N Slip Ratio = 26.9% 2.421s Revolute joint force reaches max: 3313.6061N	2.439s→ F=3332.7916N Slip Ratio = 26.46% 2.439s Revolute joint force reaches max: 3319.8015N
1200Nm	2.394s→ F=3330.6258N Slip Ratio = 25.71% 2.394s Revolute joint force reaches max: 3318.1225N	2.421s→ F=3337.0556N Slip Ratio = 27.42% 2.421s Revolute joint force reaches max: 3324.9239N
1250Nm	2.376s→ F=3333.1541N Slip Ratio = 26.35% 2.376s Revolute joint force reaches max: 3320.9901N	2.403s→ F=3340.9393N Slip Ratio = 26.98% 2.403s Revolute joint force reaches max: 3329.2016N
1300Nm	2.358s→ F=3335.0216N Slip Ratio = 25.82% 2.358s Revolute joint force reaches max: 3323.2501N	2.385s→ F=3342.9307N Slip Ratio = 25.21% 2.385s Revolute joint force reaches max: 3332.0293N

Table 3.3 Tyre-road braking force characteristic under different disk-pad braking torque A

Braking Time / Braking Torque	2.1-2.5s (0.4s)	2.2-2.5s (0.3s)
1050Nm	2.502s→ F=3326.8143N Slip Ratio = 22.01% 2.529s Revolute joint force reaches max: 3314.6522N	2.52s→ F=3354.9934N Slip Ratio = 21.63% 2.538s Revolute joint force reaches max: 3343.6633N
1100Nm	2.475s→ F=3340.7213N Slip Ratio = 24.14% 2.475s Revolute joint force reaches max: 3328.6503N	2.493s→ F=3376.6554N Slip Ratio = 24.28% 2.52s Revolute joint force reaches max: 3365.5569N
1150Nm	2.457s→ F=3349.3965N Slip Ratio = 25.71% 2.457s Revolute joint force reaches max: 3337.7735N	2.475s→ F=3384.8881N Slip Ratio = 25.04% 2.484s Revolute joint force reaches max: 3374.3561N
1200Nm	2.439s→ F=3355.238N Slip Ratio = 25.27% 2.439s Revolute joint force reaches max: 3343.8539N	2.466s→ F=3391.6801N Slip Ratio = 27.67% 2.466s Revolute joint force reaches max: 3381.2531N
1250Nm	2.43s→ F=3361.1896N Slip Ratio = 27.68% 2.43s Revolute joint force reaches max: 3350.3531N	2.457s→ F=3395.982N Slip Ratio = 29.17% 2.457s Revolute joint force reaches max: 3386.6861N
1300Nm	2.421s→ F=3365.3482N Slip Ratio = 29.29% 2.421s Revolute joint force reaches max: 3354.8215N	2.448s→ F=3397.8897N Slip Ratio = 29.86% 2.448s Revolute joint force reaches max: 3388.6736N

Table 3.4 Tyre-road braking force characteristic under different disk-pad braking torque B

3.2.5. Data Analysis

When maximum disk-pad braking torque is sufficient to let tyre-road braking force reach its peak, the shorter the braking time and the larger the peak tyre-road braking force as seen in tables 3.3 and 3.4.This is since the shorter the disk-pad braking torque build up time, the quicker the peak tyre-road braking force is reached, which leads to larger weight transfer. Therefore, the wheel vertical load force increases, leading to an increase in peak tyre-road braking force. It is noticeable that when the disk-pad braking torque build up time decreases by around 50% the peak tyre-road braking force only increases by about 1.2%(1050Nm), 1.6%(1100Nm), 1.7%(1150Nm), 1.8%(1200Nm), 1.9%(1250Nm), and 1.9%(1300Nm). This demonstrates the influence of braking time change on peak tyre-road braking force.

When the maximum disk-pad braking torque build up time remains the same, the larger the disk-pad braking torque and the larger the peak tyre-road braking force. This is also caused by the increase in the wheel vertical load force (in turn explained by the larger weight transfer). It is noticeable that when disk-pad braking torque increases by about 23.8%, the peak tyre-road braking force only increases by about 0.6%(0.6s), 0.7%(0.5s), 1.2%(0.4s), and 1.3%(0.3s). This demonstrates the influence of disk-pad braking torque change on peak tyre-road braking force.

Based on the above information, it could be concluded that the shorter the maximum disk-pad braking torque build up time and the larger the maximum disk-pad braking torque, the larger the peak tyre-road braking force. But the difference in the overall peak tyre-road braking force is relatively small.

Although the peak revolute joint force and the peak longitudinal tyre-road braking force are always reached at the same time, there is a noticeable difference between the two forces. The shorter the disk-pad braking torque build up time or the larger the applied disk-pad braking torque, the smaller the difference between the two forces. This could be because the effects of the bushes connecting the wheel knuckle to the vehicle body have been minimised. A low disk-pad braking torque coupled with a short braking time may not change the motion of the wheels fast enough due to wheel inertia and hysteresis. Either by increasing disk-pad braking torque or by decreasing braking time to a certain level, or by changing both, the difference between peak revolute joint force and peak longitudinal tyre-road braking force decrease can be achieved. In the real brake system design, the disk-pad braking torque generation capability should always be sufficient.

In an overview of the simulation results seen in tables 3.3 and 3.4, all the data shows that the revolute joint force reaches its peak value at the same time as the tyre-road longitudinal braking force reaches its peak value. But there is a noticeable difference between the two values of the respective forces. Accordingly, the force measurement within the revolute joint could represent the trend of the real tyre-road longitudinal braking force, but it may not fully represent the value of the tyre-road longitudinal braking force.

3.2.6. Simulation Data with Regards to Various Vehicle Speeds

The influences of different test vehicle speeds on the ABS control characteristics has been discussed in much of the literature. Therefore, simulations with different test vehicle speeds are carried out to see how the various speeds influence the peak tyre-road longitudinal braking force and the peak longitudinal revolute joint force characteristic. A typical 0.4s disk-pad braking torque build up time and a 1200Nm disk-pad braking torque are used as non-change parameters in the simulations. The 85km/h, 75km/h, 55km/h, and 45km/h test simulations are carried out and the results data is shown in table 3.5.

1200Nm @ 0.4s	A
85Km/h	2.448s→ Max force F=3362.6745N 2.448s Revolute joint force reaches max: 3352.2256N
75Km/h	2.448s→ Max force F=3358.6026N 2.448s Revolute joint force reaches max: 3347.6001N
65Km/h	2.439s→ Max force F=3355.238N 2.439s Revolute joint force reaches max: 3343.8539N
55Km/h	2.439s→ Max force F=3352.6287N 2.439s Revolute joint force reaches max: 3340.2353N
45Km/h	2.439s→ Max force F=3346.4081N 2.439s Revolute joint force reaches max: 3332.7332N
35Km/h	2.431s→ Max force F=3343.5567N 2.430s Revolute joint force reaches max: 3330.1576N

Table 3.5 Tyre-road braking force characteristic under initial vehicle speed

As shown, the peak tyre-road braking force change caused by the differences in vehicle speed is not significant. In conclusion, the peak revolute joint force and the peak tyre longitudinal braking force are reached at the same time at different vehicle test speeds. A noticeable difference between the two forces presents in all simulations. The lower the testing speed, the bigger the difference in force.

Consequently, the force measurement within the wheel hub at the revolute joint which connects the wheel and wheel knuckle could be useful in the development of a new ABS control method. Although there is an offset between the measured force and the real tyre-road braking force, it has been shown to represent the trend of the tyre-road braking force. If an adequate control method is implemented to adopt the property of the measured revolute joint force, a less complex ABS control method with a high level of performance could be achieved.

3.3. Conclusion

This chapter established a validated ADAMS vehicle model for ABS control method development test simulation. A range of model parameters are established and tested to stimulate various emergency braking scenarios which the tested ABS control method is then exposed to. The wheel hub revolute joint longitudinal force measurement is proved to be a good representation of tyre-road braking force in different simulations. The ABS control method proposed in the next chapter can implement this force measurement to improve the control efficiency.

4. Control Method Development

In order to develop a practical ABS control method, the commercialised ABS methods (Bosch and Wabco) are analysed and their advantages are made use of. Neither very detailed instructions on how these methods work nor analyse of them is found in any of the literature. In this research an analysis of these instructions is provided based on extensive understanding of brake physics and tyre road interactions.

4.1. Analysis of Bosch ABS control method

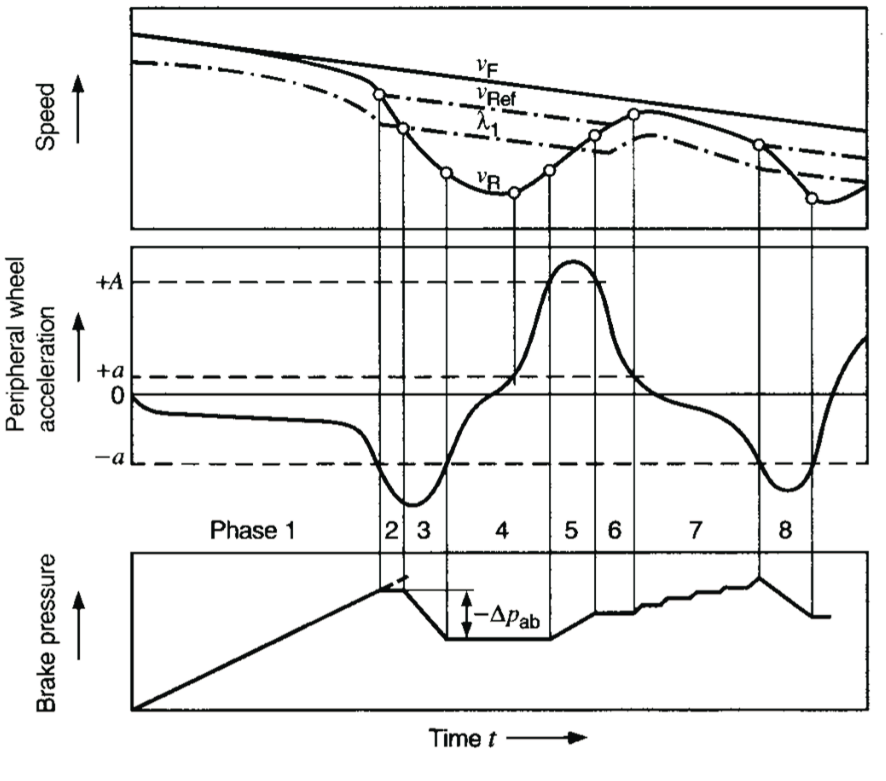


Figure 4.1 Bosch ABS control characteristics

As seen in Figure 4-1, during initial braking, a short sharp drop at the beginning of the wheel angular acceleration curve diagram can be observed. The sharp drop then settles to a relatively stable value, although it varies slightly from time to time within the stable brake zone. This deceleration indicates a delay between the application of the disk-pad braking torque and the effective generation of tyre-road braking torque. This delay is due to the tyre hysteresis that the tyre needs time to generate forces physically, as shear stress builds up between the tyre rubber in the contact patch and the road surface. At the same time, the brake disk and pad work on a shorter time-scale and more efficiently build up a higher force. While the brake process continues, the disk-pad braking torque increases continuously within the stable braking zone before the optimal slip ratio is reached, and the tyre-road braking torque increases at the same rate as the disk-pad braking torque, maintaining balance between them. However, as mentioned previously, there is a lag between the two braking torques during their build-up process. This difference is caused by the lag, and it is presented as a negative wheel angular acceleration. The absolute value of negative wheel acceleration is relatively constant, although there is a slight increase over time. The increase is more obvious when the tyre-road braking torque is approaching the optimal slip ratio. This is because the rate of the increase in the tyre-road braking torque decreases when approaching the optimal slip ratio. Meanwhile, the rate of the increase of the disk-pad braking torque is maintained. After the optimal slip ratio is reached, the tyre-road braking torque can no longer rise, irrespective of how much further the disk-pad braking torque increases as seen in Figure 4.2. At this moment, the relatively stable balance between the two braking torques is broken and a sharper increase in the absolute value of the negative wheel acceleration is observed in phase 1 (Figure 4.1). As the difference between the two braking torques increases further, the absolute value of the negative wheel acceleration increases further. This further increase is due to the combined effect of the changes in the two braking torques (i.e. while one is increasing the other is decreasing). The rate at which the absolute value of the negative wheel angular acceleration increases depends on the individual rate of the change in the two braking torques. Theoretically, the moment that the balance between the two braking torques is broken, the absolute value of the negative wheel angular acceleration will suddenly increase sharply. At this point, it can be concluded that the optimal slip ratio has then been reached.

In reality however, due to the measurement noise, nonlinear dynamics of the tyre and brake system, and the lack of accurate tyre-road braking torque information, the optimal slip ratio cannot be confirmed until other related parameter thresholds are reached. The Bosch ABS withholds this confirmation until point '-a', which is indicated in the middle graph in Figure 4.1. This is based on previous tuning experience and an understanding of the specified test vehicle's tyre-road dynamics, which is obtained in experimental tests. This is one of the most difficult tuning points to establish. In the Bosch text book, it is stated that the optimal slip ratio represented by wheel angular acceleration point '-a' might be reached within the stable brake zone. That could mean three things.

Firstly, it requires initial accuracy determination. So during its first control cycle, in the second control phase, there is a brake pressure hold action to confirm that the tyre-road braking torque is in the currently determined state (i.e. decreasing). Secondly, this could mean that on a medium to low friction coefficient surface, even before the optimal slip ratio is reached, the tyre-road braking torque has reached a similar value to its value at the optimal slip ratio point. Thirdly, it could mean that the disk-pad brake efficiency increases due to the increase in temperature during braking, which causes disk-pad braking torque increases much faster than the tyre-road braking torque, especially when the tyre-road braking torque is approaching its peak value. If the tyre-road braking torque measurement is available, then the drop in the tyre-road braking torque can confirm that the unstable braking zone has been reached. The optimal slip ratio confirmation by holding disk-pad braking torque in Bosch ABS can be saved so that the peak tyre-road brake torque can be maintained more efficiently.

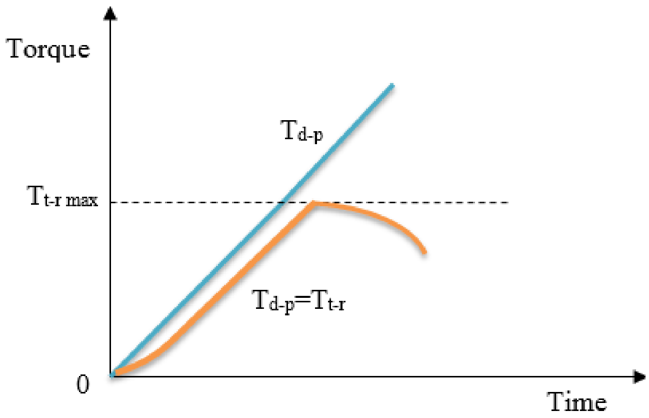


Figure 4.2 Disk-pad and tyre-road braking torque interaction

The Bosch ABS control method has to determine the reference wheel peripheral speed and reference wheel peripheral speed decrease slope (gradient) in the first control cycle and then use them as further control references in the next ABS control cycle. Once the full ABS control kicks in, the wheel angular speed sensor measured data may not be used to determine the reference wheel peripheral speed, which represents the wheel peripheral speed when optimal slip ratio is maintained while braking. In the Bosch text book (Robert Bosch GmbH, 2009), it states the reference peripheral speed is measured at the beginning of the ABS control phase, before the ABS control fully kicks in. It uses the measurements of two diagonally opposed wheels, and the faster one is used as a reference speed. The measurement time can be an uncertainty. It is stated that this is at the beginning of ABS control before the ABS control fully kicks in, but it is not known if this is at 1 second or 2 seconds.

However, there is a predefined parameter; the optimal slip ratio representative wheel angular acceleration point. The correct understanding is that when the predefined optimal slip ratio representative wheel angular acceleration point is reached, the ECU takes the measurements of the wheel angular speed sensor from the two diagonally opposed wheels and chooses the faster one as the reference speed. Accordingly, during the initial ABS control cycle, at the negative wheel angular acceleration point '-a', which represents optimal slip ratio, the currently determined wheel peripheral speed is defined as the current reference wheel peripheral speed. Due to the nature of braking, the reference wheel peripheral speed will decrease. Therefore, the Bosch ABS has to determine the rate of the decrease in the reference wheel peripheral speed in ideal brake conditions, in which the exact optimal slip ratio is maintained, as seen in Figure 4.3.

As discussed previously, due to the difficulties involved in accurately determining the optimal slip ratio, the Bosch ABS delays its confirmation of it, and at this point the reference wheel peripheral speed is determined by the value of the wheel peripheral speed. Since the optimal slip ratio representative '-a' has been delayed, the decrease slope in the corresponding wheel peripheral speed (negative wheel angular acceleration) may steepen, so it may not be able to represent the decrease slope in the reference wheel peripheral speed accurately. Other complex prediction methods have been used to compensate for the offset. The Bosch ABS algorithm determines the decrease slope by analysing the relative logical relationships extracted in experiment tests (Robert Bosch GmbH, 2009), which make it extremely difficult to tune.

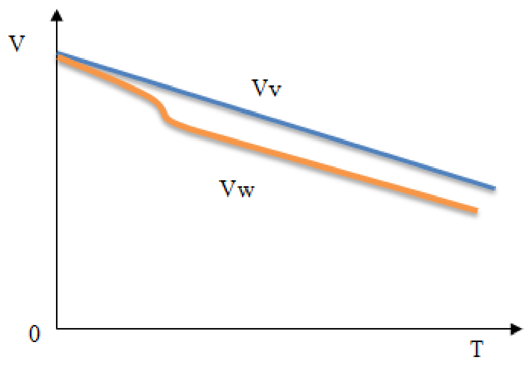


Figure 4.3 Vehicle and wheel speed characteristics under ideal ABS control

During control phase 2 of Bosch ABS, as seen in Figure 4.1, it uses the reference peripheral speed and measured wheel angular speed to calculate the slip ratio threshold λ1 while holding the disk-pad braking torque to validate the accuracy of the optimal slip ratio representative negative wheel angular acceleration point, '-a', it is also to confirm the optimal slip ratio point has been passed. Due to the combination of the threshold λ1 validation procedure and the fact that the Bosch ABS already delays the optimal slip ratio confirmation point, further effort must be taken to reduce the offset effect, so the optimal slip ratio can be restored more efficiently. This could cause instability within the control cycle in some fast peak tyre-road braking torque recovery demand circumstances.

In the Bosch ABS, after λ1 is passed, the disk-pad braking torque is decreased in order to restore the optimal slip ratio. Since the absolute value of the negative wheel angular acceleration continues to increase, the brake state enters an unstable zone. The wheel tends to lock very quickly, so the disk-pad braking torque is decreased very significantly to bring the wheel angular acceleration into positive territory. This facilitates an increase in the wheel peripheral speed to avoid the risk of the wheel locking and brings the brake state back into a stable zone to restore the peak tyre-road braking torque. From the slip ratio equation, it is known that the increase in the difference between the wheel longitudinal speed and the wheel peripheral speed calculated from the wheel angular speed causes the increase in the slip ratio. Therefore, after the optimal slip ratio point is reached a further increase in the difference between these two speeds pushes the slip ratio further from its optimal value. As long as the difference between the two braking torques causes the absolute value of the negative wheel angular acceleration to increase, the difference between the two speeds will continually increase until the wheel locks, unless negative wheel angular acceleration is reduced to a lower level than negative wheel longitudinal acceleration (and this state is maintained for a certain time).

However, the most efficient and safest way of restoring the optimal slip ratio is to bring the wheel angular acceleration to a positive value quickly, allowing the difference between the two speeds to be reduced quickly and efficiently. As mentioned previously during the disk-pad braking torque decrease action, the disk-pad braking torque responds more efficiently towards the control action than the tyre-road braking torque. This is supported by the data from the wheel angular acceleration curve of the Bosch ABS during the disk-pad braking torque hold action in phase 4. This means that even when the disk-pad braking torque is reduced below the peak tyre-road braking torque point, the tyre-road braking torque itself still has to wait for a certain period to get back to the level. If the ABS control wants to bring the tyre-road braking torque to its peak point quickly, it has to reduce the disk-pad braking torque to a lower level than the peak tyre-road braking torque point.

During the first decrease control phase of the Bosch ABS, the disk-pad braking torque is decreased by 'ΔP_{ab}' calculated torque, and then it is held at the level. It is noticeable that after the second control cycle, the level of the disk-pad braking torque decrease is less than 'ΔP_{ab}' which is because the calculation phase for the optimal slip ratio confirmation point $\lambda 1$ is unnecessary. Since the disk-pad braking torque decrease is very significant, as soon as the negative wheel angular acceleration value increases back to '-a', the disk-pad braking torque is held without a further decrease. As seen in Figure 4.4, even in the hold state, the disk-pad braking torque has been decreased to a relatively low level, which allows the tyre-road braking torque to switch to its increase state. While the disk-pad braking torque is in its hold state and the tyre-road braking torque is in its increase state, the difference between the two braking torques becomes smaller. As such, the absolute value of the negative wheel angular acceleration continuously decreases, then it changes to positive.

As discussed, the positive angular acceleration can decrease the difference between the two speeds and lead back towards the optimal slip ratio. A positive angular acceleration means that the disk-pad braking torque level is below the tyre-road braking torque level, as seen in Figure 4.5. It is usually considered ideal to maintain the hold state of the disk-pad braking torque to further increase the positive wheel acceleration until the optimal slip ratio is achieved. But due to a possible initial delay in the disk-pad braking torque hold action and a delay in the tyre dynamic reaction to the control input, even if the disk-pad braking torque were in the hold state, it could have already experienced an over-decrease. Therefore, the hold state continues until ‘A’ on the wheel angular acceleration curve is reached. The ‘A’ point is the upper limit of the positive wheel angular acceleration. It represents the pre-defined threshold of the difference between the two braking torques when approaching the optimal slip ratio.

The rate in the decrease of the disk-pad braking torque is an important tuning parameter. If it is set to decrease as quickly as the Bosch ABS, or even faster, the hold action start point should not be any greater than the negative wheel angular acceleration ‘-a’ in control phase 3. If, at the same rate of decrease, the hold state is set to be active until zero-wheel angular acceleration or even a positive wheel angular acceleration, it could compromise control performance due to an over-decrease in disk-pad braking torque. If the disk-pad braking torque is set to decrease more slowly than the Bosch ABS, there is a greater risk that the wheel will lock, unless the optimal slip ratio point is detected earlier and the ABS control action kicks in sooner. After reaching ‘A’, the Bosch ABS increases the disk-pad braking torque even before the next optimal slip ratio is reached, which is due to the need to avoid a loss in tyre-road braking torque. This is because the disk-pad braking torque has been decreased too much initially. If the tyre-road braking torque information were available, better control performance could be achieved.

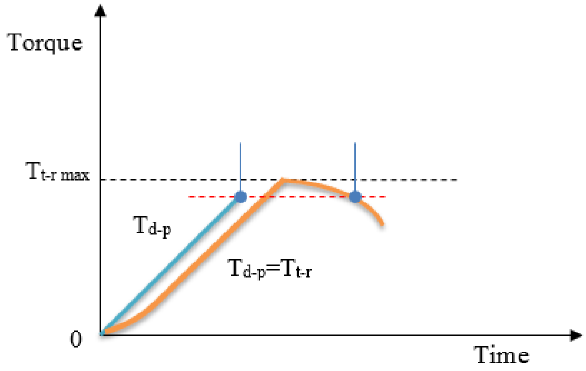


Figure 4.4 Disk-pad braking torque control show case A

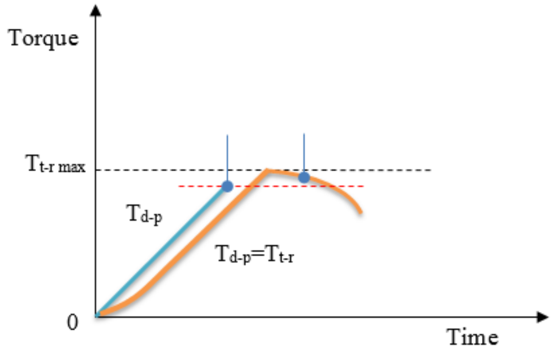


Figure 4.5 Disk-pad braking torque control show case B

Bosch ABS holds the disk-pad braking torque again in phase 6 to confirm what is happening to tyre-road braking torque. While the wheel positive angular acceleration is continuously decreasing, it can be confirmed that the peak tyre-road braking torque has been passed, so the Bosch ABS starts to increase the disk-pad braking torque in a continuous increase-hold pattern. The purpose of this control action is to reduce suspension vibration, and this is only after ABS initialisation.

There are some interesting properties shown in the Bosch ABS control curves. These properties are shown in control phases 2, 5, and 8. The explanation of these properties includes a consideration of the typical change in the status of the longitudinal tyre-road braking force, and the properties of the interaction of the two braking torques during ABS control. The properties shown in phases 2 and 8 are the same, even while the disk-pad braking torque is being held or decreased. There is still a short increase in the absolute value of negative wheel angular acceleration, which happens when the level of the disk-pad braking torque is greater than the level of the tyre-road braking torque. Theoretically in phases 2 and 8, as soon as the disk-pad braking torque starts to hold or decrease, the tyre-road braking force should react to hold or increase. Since the disk-pad braking torque is one of the main generators of tyre-road braking torque, a change in the tyre-road braking torque should correspond with a change in the disk-pad braking torque, as long as the vehicle longitudinal inertia force is considered steady.

However, due to this correspondence, combined with the physical braking system actuation delay and the physical tyre dynamics limitation, the lag in the tyre-road braking torque reaction to the disk-pad braking torque change is resulted. The content of this lag depends on how fast the disk-pad brake torque is changing and its interaction with the vehicle longitudinal inertia force and tyre properties. In the case of phases 2 and 8, the disk-pad braking torque starts to decrease when the tyre-road braking torque is decreasing and the level of the disk-pad braking torque is much greater than the level of tyre-road braking torque. At the beginning of the disk-pad braking torque decrease action, due to the previously discussed reasons, tyre-road braking torque continues to decrease. If the rate of the decrease in disk-pad braking torque is lower than the rate of the decrease in tyre-road braking torque, then the decrease in both does not necessarily result in decreased absolute wheel angular acceleration value. The tyre-road braking torque change dynamics must be identified in order to define the disk-pad braking torque application. This reaction time lag is obvious when the tyre-road braking torque is in process of changing its state. After the reaction time lag, tyre-road braking torque starts to increase and the absolute value of the negative wheel angular acceleration starts to decrease.

As seen in control phase 5, in which the disk-pad braking torque is increasing, there is a short increase in the wheel angular acceleration before it starts to decrease. This happens when the level of the disk-pad braking torque is below that of the tyre-road braking torque and the latter is in a state of increase. This could be explained by the delay in the build-up of the disk-pad braking torque. Accordingly, tyre-road braking torque increases at a higher rate than the disk-pad braking torque. This could also arise from a delay in the response of the tyre-road braking torque or an over-decrease in the disk-pad braking torque. Therefore, it appears that tyre-road braking torque cannot react immediately to the disk-pad braking torque. Although, some research has criticised the accuracy of this phenomenon, it arguably failed to consider the tyre-road braking torque reaction state and the effect of the disk-pad braking torque change rate.

There are two reasons for describing the aforementioned properties above. One is that many theoretical based ABS control methods ignore the reaction time lag of both braking torques, especially when the change in the disk-pad braking torque is resulting in state change of tyre-road braking torque. The second relates to how Bosch ABS defines the rate of the change in the disk-pad braking torque. This definition, must be established based on a good understanding of the physical tyre and ABS dynamics. This is possible by carefully studying the interactions between the disk-pad braking torque, the tyre-road braking torque, the vehicle longitudinal inertia force, and the tyre vertical force.

4.2. Analysis of Wabco ABS control method

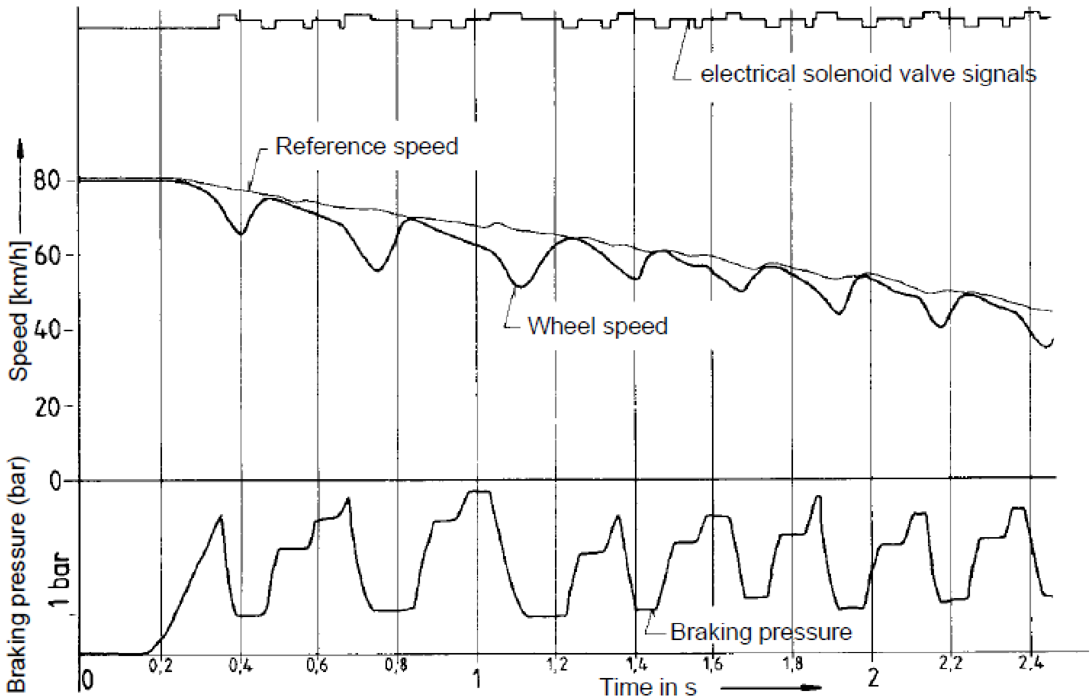


Figure 4.6 Wabco ABS control characteristic

In the Wabco ABS control cycle, shown in Figure 4.6, the braking pressure decreases immediately after a sudden drop in wheel speed, which indicates a sudden increase in the absolute value of the negative wheel angular acceleration. As discussed, typically, in ABS control phase 1, the tyre-road braking torque increases with the disk-pad braking force torque. Due to tyre hysteresis there is always a delay in the tyre-road braking torque build-up process. As such, there is a difference between the tyre-road braking torque and the disk-pad braking torque. The difference increases slightly within the stable braking zone. Negative wheel angular acceleration results from the difference between the two and indicates that the disk-pad braking torque has a higher value. Once the peak tyre-road braking torque is passed the brake state enters the unstable zone. The tyre-road braking torque decreases as the disk-pad braking torque increases. The difference between them suddenly increases at a significant rate. Consequently, the absolute value of the negative wheel angular acceleration increases sharply, and it can be concluded that the Wabco ABS uses this sudden increase which could represent the peak tyre-road braking torque, as the disk-pad braking torque decrease action trigger in control phase 1. The sudden increase in the absolute value of the negative wheel angular acceleration is predefined and was based on test data and tuning experience.

After the disk-pad braking torque decrease in control phase 1, the disk-pad braking torque hold control is triggered shortly before or when the negative wheel angular acceleration reaches zero. In this stage, the tyre-road braking torque is below its peak value and it increases while the disk-pad braking torque is held. Initially, however, if tyre hysteresis is considered there will be a delay in the response of the tyre-road braking torque to the change in disk-pad braking torque. Considering the highly efficient way in which disk-pad braking torque decreases, when it drops to the same level as the tyre-road braking torque, the latter is just building up an increase. While the disk-pad braking torque is held, the tyre-road braking torque continues to increase, and therefore the positive wheel angular acceleration also increases (as long as positive wheel angular acceleration is present the wheel peripheral speed increases). The rate of the increase is represented by the value of the wheel angular acceleration. When the tyre-road braking torque is approaching its peak, the rate of its increase slows. Once the peak is reached, the value of the positive wheel angular acceleration stops increasing and starts decreasing, and if the wheel angular acceleration is still positive, the wheel peripheral speed will still increase.

However, due to the decrease in the positive wheel angular acceleration, the increase in the wheel peripheral speed slows. When the tyre-road braking torque drops to the same level as the holding disk-pad braking torque, the wheel angular acceleration hits zero. At this point, the peak tyre-road braking torque has already been passed and this is when the Wabco ABS starts to increase the disk-pad braking torque again. However, as there is no simple tyre-road braking torque measurement method, the Wabco ABS increases the disk-pad braking torque for a short while and then holds it to observe the wheel peripheral speed and the wheel angular acceleration. This is done to observe the tyre-road braking torque trace in control phase 1 indirectly. This increase and hold action may go through several cycles to achieve the next peak tyre-road braking torque. Once a sudden increase in the absolute value of the negative wheel angular acceleration is detected again, Wabco ABS enters the disk-pad braking torque decrease control phase again.

4.3. Wheel revolute joint force measurement representable local tyre-road braking force

As seen in Figure 4.7, the solid red curve represents the trace of the longitudinal force of the revolute joint in the ADAMS model, which is used to attach the wheel to the wheel knuckle. The blue curve represents the tyre longitudinal force. As shown, the two forces have broadly the same trace, but the revolute joint longitudinal force amplitude is slightly lower than the tyre longitudinal force during the braking process. This indicates that the force measurement within the wheel bearing of a real vehicle could be used to represent the tyre-road braking force.

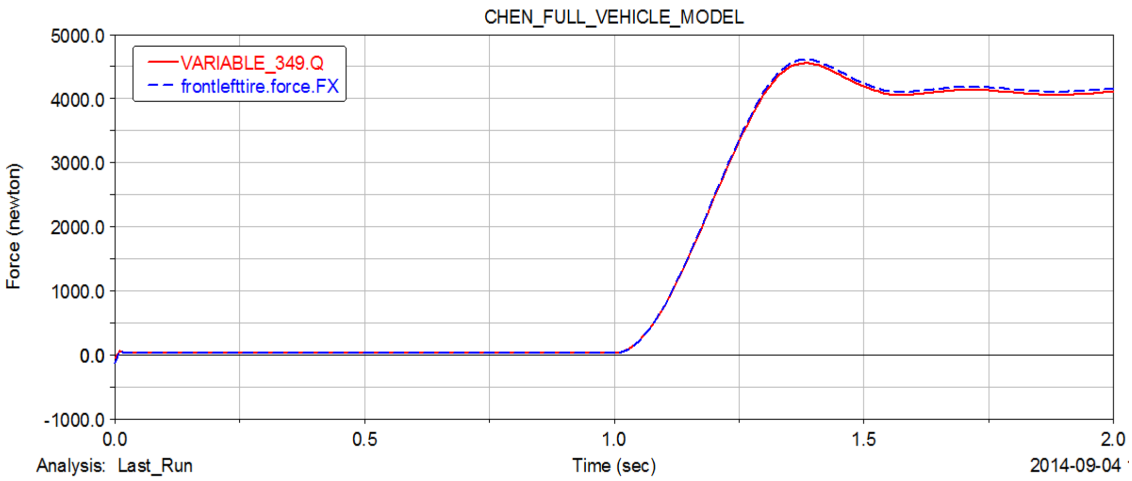


Figure 4.7 Tyre and revolute joint force characteristic during emergency braking

Mathieu Gerard (2010) proposed using a tyre force measurement bearing developed by a company called SKF. The force measurement bearing monitors bearing deformation, as seen in Figure 4.8, and the deformation is then converted into the true tyre force through a sophisticated look up table. This tyre force measurement method is complex, and it requires a large amount of very accurate data mapping of the bearing characteristics. Characteristic bias with regards to each bearing due to manufacture machine accuracy, material bias, and so on could also cause difficulties in determining the accurate value of the tyre forces. The SKF force measurement bearing is claimed to be up to 90% accuracy. Mathieu Gerard proposed using the measured tyre force as a critical parameter in the mathematical based ABS control method, but the nonlinearity of each individual parameter in the proposed mathematical equations, such as the vertical tyre force, the brake disk and pad friction coefficient, as well as the measurement of the tyre longitudinal force (less than 100% accurate), could compromise the calculated control disk-pad braking torque, which can result in compromised performance of the ABS control method.

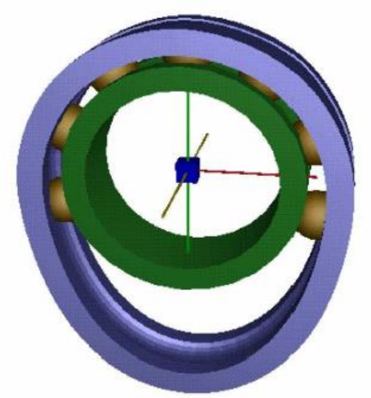


Figure 4.8 Bearing deformation simulation

However, based on all the above analysis, the force measured within the wheel hub sensor (which might be a good representation of tyre-road braking force) could be used as an important control reference parameter, not as an exact value but as a trace. Practically, if the wheel bearing longitudinal force calibration can be carried out to compensate its offset to tyre-road braking force, the exact wheel bearing longitudinal force value could be used.

4.4. Discussion and inspiration

As guidance to develop the disk-pad braking torque control, the heuristic rule based and the theory based ABS control methods are analysed. The heuristic rule based ABS control methods that define many different non-related thresholds are defined and tuned based on a large number of experiments which cost millions of pounds. The parameter definition and tuning methods are based on complex rules and comprehensive experiences. The theory based ABS control methods use full mathematical methods, along with many assumptions, to compensate for their practical limitations (or practical offsets). Learning from both these types of ABS control methods, and understanding how disk-pad braking torque is controlled, is arguably a reasonable path to follow to achieve the goal of this research.

4.4.1. Basic control pattern

The fundamental principle of ABS control is to modulate the disk-pad braking torque (to achieve peak tyre-road braking torque). All the popular ABS control methods follow similar basic control patterns, as seen in Figure 4.9. The control action ensures the slip ratio cycles around the optimal slip ratio and within a limited range. Accordingly, the tyre-road braking torque can be maintained around its peak range and the tyre-road lateral force can be kept at an adequate level. The second phase of the control process is when the slip ratio heads toward the optimal slip ratio while disk-pad braking torque is increasing. The slip ratio then passes the optimal slip ratio and enters the unstable braking zone. It continues further into the unstable braking zone until a predefined threshold is reached. The first control phase is then activated. In the second control phase, disk-pad braking torque decreases to bring the slip ratio back to the optimal slip ratio. These two control phases are cyclical. Each individual ABS control method uses its own thresholds and adds extra rules and control actions on the base pattern to establish its own control pattern.

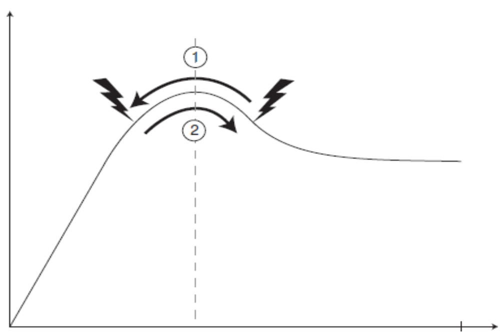


Figure 4.9 Control phase switch trigger

In the ABS control method proposed in this thesis, the same basic control pattern is followed. The optimal slip ratio representative peak tyre-road braking torque is directly detected by the in-wheel force sensor, so a process which determines the optimal slip ratio point is unnecessary. This process of detection is continuous, therefore there is also no need to determine the reference wheel peripheral speed or its decrease slope. The slip ratio threshold $\lambda 1$ validation procedure can also be removed from the list. Hence, precious time can be saved in regard to the entire control process.

4.4.2. Definition of control cycle range

It is necessary to define the control cycle range and establish the predefined threshold used to trigger each different control phase. In practice, ABS control cannot always maintain the slip ratio at its exact optimal point. The control cycle is a range. Under some circumstances, even after the optimal slip ratio is passed, the tyre-road braking torque is still maintained at a similar value within a certain range of the slip ratio. Therefore, even if the optimal slip ratio is confirmed, in order to make the maximum use of the available tyre-road braking torque, a delay in the control action is desirable. This is demonstrated in the typical longitudinal tyre-road braking force to slip ratio diagram in Figure 4.10. On the curves, the slope is sharper on the left-hand side of the optimal slip ratio than the right-hand side. If the same time interval is taken on these curves, the tyre-road braking torque drops further on the left-hand side than the right-hand side, so range B is preferable to range A. There are two other important reasons for allowing the slip ratio to pass its optimal point; firstly, to reduce the sensor measurement noise effect, and secondly to avoid control difficulties caused by the very tight control cycle. With regards to the latter, the control actuator delay and controlled plant (tyre) reaction delay could lead to instability. Further increases in the slip ratio after the optimal point is reached are desired in practice, however, they must be retained within a limited range.

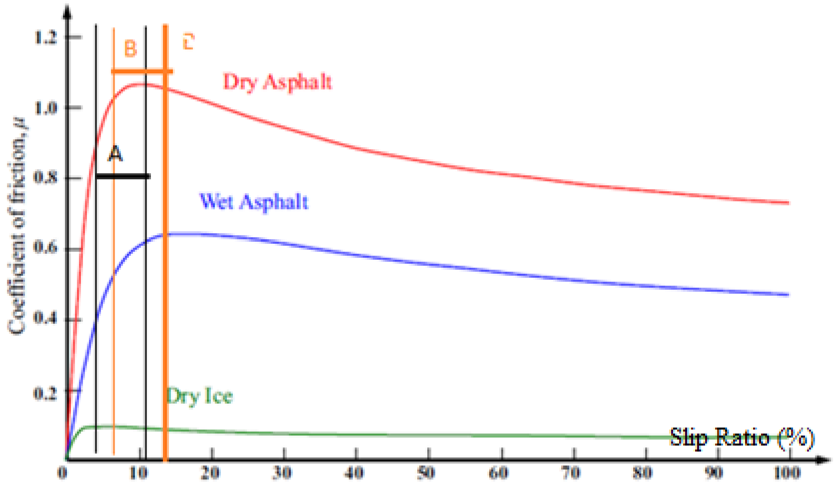


Figure 4.10 Tyre-road braking force characteristic analysis beyond optimal slip

In the proposed ABS control method in this thesis, control range B is implemented. Therefore, the drop in the percentage of the peak tyre-road braking torque on both sides of the peak point is used to define the control cycle range. The position of range B on the curve suggests that the drop in the two percentage values should be similar. These two values are two tuning parameters, which shall be determined in a future simulation.

4.4.3. Wheel angular acceleration control reference

During the first control phase of the proposed ABS control method (when the peak tyre-road braking torque is passed) while the slip ratio and disk-pad braking torque are increasing, the tyre-road braking torque is decreasing. It is desirable to quickly and effectively decrease the disk-pad braking torque in order to change the decrease state of the tyre-road braking torque and allow it to increase back to its peak value. But a reference parameter is required to determine when to stop the decrease in the disk-pad braking torque. The wheel angular acceleration can be used as the reference parameter, but the effects brought about by the greater build-up efficiency of the disk-pad braking torque over the tyre-road braking torque must be considered. As discussed previously, in the first control phase of Bosch ABS the disk-pad braking torque decreases so fast that the tyre-road braking torque cannot react immediately (due to the resulting changes in tyre hysteresis). A predefined wheel angular acceleration reference point which considers such delays can be used to limit the extent of the decrease in disk-pad braking torque. Consequently, the wheel angular acceleration reference point in addition to a rules based ABS control method are proposed on this research.

4.4.3.1. An explanation of wheel angular acceleration control based on physics

Both the Bosch and Wabco ABS introductions state that the wheel angular acceleration represents the difference between the tyre-road braking torque and the disk-pad braking torque. The change in the behaviour of the wheel angular acceleration is used to observe the behaviour of the tyre-road braking torque when the disk-pad braking torque is in the hold state. For example, when the disk-pad braking torque is being held, if the positive wheel angular acceleration is increasing, it can be concluded that the tyre-road braking torque is also increasing (if the change in the disk-pad friction coefficient caused by the change in temperature is isolated as a variable).

The change in the behaviour of the wheel angular acceleration is also used to observe the interaction between the two braking torques when the current state of both braking torques is confirmed by the ABS control phase. For example, in the slip ratio increase control phase (when the disk-pad braking torque is increasing), if wheel angular acceleration stays at a similar value, it can be concluded that the tyre-road braking torque is increasing at the same rate as the disk-pad braking torque. And if the absolute value of wheel angular acceleration is slowly increasing, it can be concluded that the tyre-road braking torque is increasing with the disk-pad braking torque increase, but at a lower rate. The relationship between the wheel angular acceleration represented disk-pad braking torque and the tyre-road braking torque is accurate only when both braking torques are in their true state. Tyre hysteresis causes a false tyre-road braking torque state and the change in the brake disk and pad friction coefficient (due to temperature change) causes a false disk-pad braking torque state. These effects must be considered in the development of a new ABS control method, especially for rules based control.

But it is always true that wheel angular acceleration accurately represents the difference between the disk-pad braking torque and the tyre-road braking torque, and this can be used as an important reference to decide how the disk-pad braking torque should be controlled. As the disk-pad braking torque is the resistance torque which acts against rotating tyres, and the tyre-road brake torque is the propulsion torque which acts against rotating tyre, when the disk-pad braking torque is greater than the tyre-road braking torque the wheel angular acceleration is negative (deceleration). When the disk-pad braking torque is lower than tyre-road braking torque, the wheel angular acceleration is positive. The greater the difference between the two the greater the absolute wheel angular acceleration is and vice versa. When the disk-pad braking torque is equal to tyre-road braking torque, the wheel angular acceleration is zero.

4.4.3.2. An explanation of the wheel angular acceleration control based on a mathematical model

From a mathematical perspective, based on the derivative of the wheel slip ratio equation, as seen in equations 4.1 and 4.2, it can be concluded that by controlling the wheel angular acceleration trend (increase or decrease), the tyre slip ratio trend can be controlled indirectly. This is in addition to the previously discussed theory, in which it was stated a change in wheel angular acceleration can influence the slip ratio resulting from the interaction between wheel peripheral speed and wheel longitudinal speed.

$$\dot{\lambda} = \frac{-r\dot{\omega}_{ref}^{+}v + r\omega\dot{v}}{v^2} < 0 \qquad \textbf{(4.1)}$$

$$\dot{\lambda} = \frac{-r\dot{\omega}_{ref}^{-}v + r\omega\dot{v}}{v^2} > 0 \qquad \textbf{(4.2)}$$

In the equation, the $\dot{\omega}_{ref}^{+}$ and $\dot{\omega}_{ref}^{-}$are the desired levels of wheel angular acceleration in two different control phases. During the braking phase the condition $0 < r\omega < v$ should be taken into account. In order to fulfil the inequality in the above equations 4.1 and 4.2, the following conditions as shown in equations 4.3 and 4.4 should be met: The $\dot{v}_{min}$ here means the possible maximum longitudinal deceleration of the test vehicle, with a typical value of 1.2g.

$$\dot{\omega}_{ref}^{+} > 0 \quad \textbf{(4.3)} \qquad \dot{\omega}_{ref}^{-} < \frac{\dot{v}_{min}}{r} \quad \textbf{(4.4)}$$

From the above analysis, it could be concluded that in the slip ratio decrease control phase, disk-pad braking torque must be decreased in order to bring wheel angular acceleration to positive. Therefore, the optimal slip ratio can be brought back. In the proposed ABS control method in this research, the wheel angular acceleration value occurring at the moment peak tyre-road braking torque is achieved is a very important control reference. The reference value can be used to identify each of the two different control phases, and ensure the correct control actions.

4.4.4. A detailed explanation of the interaction between the two braking torques

The delayed response of the tyre-road braking torque (caused by tyre hysteresis) to the change in disk-pad braking torque, as well as the difference in change efficiency between the two braking torques, are considered in the proposed control method. As seen in Figure 4.11, the generation mechanisms of the tyre-road braking force requires vehicle forward inertia (i.e. the force maintaining the forward motion of the wheel regardless of whether it is rotating or not); the disk-pad braking force (wheel rotational motion resistance force); the vertical tyre load force (tyre contact force with the road surface); and friction property, which guarantees the friction force (the generation of tyre-road braking force when there is slide action between tyre and road). When the disk-pad braking force is applied the rotational motion of the wheel slows down. Vehicle inertia tries to maintain a forward motion so that sliding occurs between tyre and road. The vertical tyre force helps to maintain the physical contact between tyre and road, so the friction force generated corresponds to the amount of slide action. On the same friction coefficient road surface, the peak tyre-road braking torque should be within a close range throughout the whole ABS control process. The largest influence on the peak tyre-road braking torque difference is the change in wheel vertical force due to weight transfer. But once the ABS control action kicks in, the weight transfer should be roughly same, so the change in the value of wheel vertical force is relatively stable.

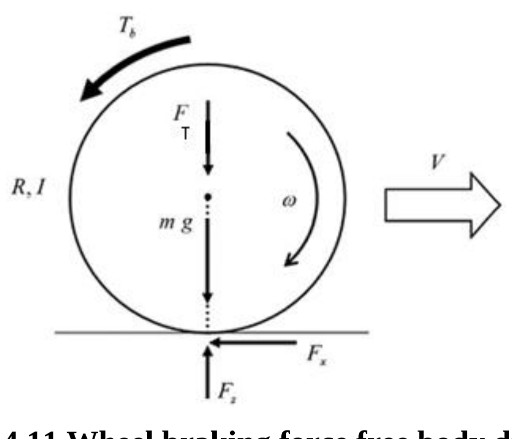


Figure 4.11 Wheel braking force free body diagram

4.4.5. Detailed discussion of disk-pad braking torque decrease control

In the first control phase of the proposed ABS control method, the desired range of the decrease in disk-pad braking torque is between x and y, as seen in Figure 4.12. If the disk-pad braking torque is decreased to a point above x, it could resist the effective increase in tyre-road braking torque or even stop it from increasing back to its peak value. If disk-pad braking torque is decreased to a point below y, the decrease could be excessive and it could take longer to increase in the next control phase (pressure build-up). If disk-pad braking torque is decreased to a point between x and y, depending on how much further the tyre-road braking torque decreases below its peak value and how long the excessive negative wheel angular acceleration lasts (this is to identify how much further the wheel peripheral speed drops below its reference value), the decrease in the value of the set disk-pad braking torque may vary, which causes complexity in the development of the control method.

In order to slowly modulate the disk-pad braking torque to lead and match the change in the tyre-road braking torque until the latter reaches its peak value, the disk-pad braking torque control must be very accurate. Due to the uncertainties and nonlinearity involved in the ABS control process, it is very difficult to achieve mathematical closed loop point-to-point control. But by considering the combined effect of tyre hysteresis and faster disk-pad braking torque change, a point ‘a’ on the tyre-road braking torque curve (as shown in Figure 4.12) can be defined in order to simplify the control method. That means the disk-pad braking torque is capable to decrease to the current tyre-road braking torque level (‘a’) fast enough, while the response of tyre-road braking torque to ABS control is delayed, so it still remains very close to ‘a’. At that very moment, wheel angular acceleration becomes zero. The disk-pad braking torque then enters the hold state until the next peak tyre-road braking torque is reached. The ‘a’ is the drop in percentage of peak tyre-road braking force, and it must be able to maintain the optimum balance between reducing the measurement noise effect, adapting the physical ABS control system capability and achieving good control performance.

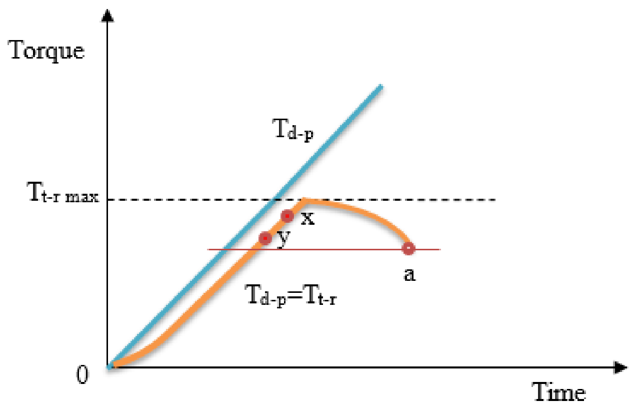


Figure 4.12 Disk-pad braking torque control show case C

4.4.6. Discussion and conclusion of overall control method

If only the revolute joint force measurement is used as a control parameter in the proposed ABS control method, it is sufficient to detect the peak tyre-road braking force and the related drop in percentage thresholds for control phase alternation points. But if disk-pad braking torque is controlled according to the predefined peak tyre-road braking torque drop percentage thresholds without consideration of other control reference parameters, the thresholds may not be always accurate (due to the many uncertainties involved), which can cause an issue that by the time the thresholds are reached the disk-pad braking torque may have increased or decreased to such an extent that the expected control performance cannot be achieved and ABS control failure results. The solution to the problem is to schedule the predefined drop in the percentage of the tyre-road braking force thresholds for different road surface friction coefficients in different braking conditions, which requires implementing sophisticated braking conditions clarification logic. However, this increases the complexity of the control algorithm significantly and renders it unable to cope with the uncertainties and non-linearity involved in the ABS control process.

If only wheel angular acceleration is used as control parameter, the difference between the disk-pad braking torque and the tyre-road braking torque can be accurately monitored. But if there is a combination of tyre hysteresis, a delay in control actuation, and a lack of tyre-road braking torque information, even if the status of the disk-pad braking torque can be confirmed, it might not be possible to confirm if the control action is correct. Extra control logics and control phases have to be added in order to confirm the real reaction of the tyre-road braking torque to the change in the disk-pad braking torque. Other parameters such as the estimated reference wheel longitudinal speed and the slip ratio must be used to ensure the control is adequate and accurate. The Bosch and Wabco ABS fall into this category in that a significant number of experiments must be carried out to calibrate and schedule different control parameters along with the complex control logic, a process which is necessary to achieve good ABS control performance.

Full scale and accurate calibration mapping of the different reference wheel angular accelerations and the relative disk-pad braking torque control is required to cope with every control point in various control conditions. Any mismatch or missing control point clarification on the mapping could compromise performance and cause control instability. In the case of a change in brake efficiency (caused by a change in temperature between the brake disk and pad), extra logic and control action delay must be implemented to cope with a possible misjudgement of the correct control action.

A control method based on the preceding discussion which uses wheel revolute joint force measurement to represent tyre-road braking force for its local peak detection and its drop percentage thresholds as the control phase alternation trigger, as well as predefined wheel angular acceleration reference values that determine different control actions, will be implemented in this research. The control method is rules based with wheel angular acceleration control based on a mathematical model.

4.4.7. Detailed description of proposed ABS control method

The proposed control method will be exposed to an emergency braking situation in which the vehicle is travelling over a speed threshold of over 5 miles per hour and the driver applies the brake heavily. This could be determined by brake pedal application to driver brake demand mapping, which is a look up table containing predefined brake pedal travel distance and travel speed to braking torque demand. Once the measured representative local tyre-road braking torque reaches its local peak value and starts to decrease, the ABS control enters a ‘ready’ state. The wheel angular acceleration is recorded at this moment for future control reference. Once the measured representative local tyre-road braking torque drops to a predefined level (a percentage of the detected peak value), the ABS control enters a ‘go’ state. During the ‘go’ state the ABS control decreases the disk-pad braking torque by closing the inlet solenoid and opening the outlet solenoid. The wheel angular acceleration is used as one of the control references and a control reaction observation parameter. Once a new control phase has been triggered, the previously recorded wheel angular acceleration value is used to identify which control phase the ABS control is in. When this value is negative, it can be concluded that ABS control is in a disk-pad braking torque increase phase. ABS control should enter a disk-pad braking torque decrease phase. When it is positive it can be concluded that ABS control is in a disk-pad braking torque decrease phase, and then ABS control should enter a disk-pad braking torque increase phase. In control phase one, the disk-pad braking torque decreases until wheel angular acceleration increases to zero. Then the disk-pad braking torque enters a ‘hold’ state, both inlet and outlet solenoids are closed, and the wheel angular acceleration and measured local representative tyre-road braking torque are used to observe the effect of the disk-pad braking torque control on the tyre-road braking torque. Once the measured representative local tyre-road braking torque drops to the predefined level (as a percentage of the newly detected peak value), and the reference wheel angular acceleration value determined control phase is confirmed, ABS control increases disk-pad braking torque by opening the inlet solenoid and closing the outlet solenoid.

To ensure that the correct solenoid is chosen, the priority is to confirm that the solenoid actuated disk-pad braking torque change capacity is greater than the ordinary brake system’s maximum (in an emergency brake situation) change capacity, especially when in a disk-pad braking torque decrease control phase. A similar property can be found in both Bosch and Wabco ABS, where in an emergency brake situation the disk-pad braking torque decreases at its full capacity. In the proposed ABS control method, the disk-pad braking torque decreases at the system’s maximum capacity.

4.5. Low friction surface control method

4.5.1. Estimation of road surface friction conditions

As mentioned in the literature review, the hybrid ABS control method has limitations on low friction coefficient surfaces. As seen in Figure 4.13 there is no clear peak on the tyre-road braking torque curve. Accordingly, the road surface friction condition should be estimated first to overcome this limitation. The road surface friction condition estimator does not have to be able to identify the exact friction coefficient, but it should at least identify the low friction surface with no obvious peak tyre-road braking torque characteristics in order to implement a different control strategy. If a well calibrated look-up table of peak tyre-road braking force values on different friction coefficient surfaces can be established, there is potential for the estimator to identify the road surface friction coefficient.

This item has been removed due to 3rd Party Copyright.This item has been removed due to 3rd Party Copyright. The unabridged version of the thesis can be found in the Lanchester Library, Coventry University

Figure 4.13 Longitudinal coefficient of friction μ and lateral force versus slip ratio curve (Kazemi et al., 2005).

4.5.2. A discussion of low friction surface ABS control method

As seen in Figure 4.13, on a low friction coefficient surface the tyre-road braking torque reaches its peak value and stays at a relatively constant level once it passes the point indicated by the red dot, which is called the peak split point. The peak split point is the point which separates the tyre-road braking force's increase slope and zero (or negative) slope. The lateral force which can be generated by the tyre becomes the main control target beyond this point. The peak split point is an important control parameter that could be used in low friction surface detection and control. One possible peak split point detection method is to identify a moment that the tyre-road braking torque has been at zero or in a negative slope for a predefined time period. Once the peak split point is detected, a relative ABS control action is triggered. But with this detection method, precious control time could be wasted, which is critical, especially on a low friction surface. Moreover, this method may have a problem detecting low friction surfaces with a very small positive tyre-road braking torque slope. The adopted low friction surface detection method will be described and discussed in the next chapter.

As tyre-road braking torque remains steady before reverting to the peak split point (irrespective of change in disk-pad braking torque), wheel angular acceleration could be more accurate in monitoring disk-pad braking torque behaviour. The effect of tyre hysteresis effect could be amplified on a low friction surface. Overall tyre-road braking torque is in its low range and its overall change rate is low. While disk-pad braking torque remains at its full change capacity, it is more sensitive, so a moderate control method should be adopted to avoid over control due to inaccurate tyre-road braking torque state observation. Regular small steps in change should be a feature of it and a hold action should be added to identify the true tyre-road braking torque trend.

One suggestion regarding low friction ABS control is to decrease disk-pad braking torque to the same level as the steady peak tyre-road braking torque so wheel angular acceleration reaches zero. If there were no decrease in tyre-road braking torque, which would prevent the detection of the peak split point after a predetermined period of time (that would provide sufficient time for it to overcome tyre hysteresis and react to the change in the disk-pad braking torque). In such a scenario, a further decrease in the disk-pad braking torque is required. The decrease control should be in a short pulse format, and it would remain on until tyre-road brake force decreases, which would allow the detection of the peak split point.

Another suggestion is that once the peak split point is detected (when disk-pad braking torque is increasing), the ABS control decreases disk-pad braking torque immediately until a pre-defined positive angular acceleration is achieved. Then ABS control would enter a hold action. The pre-defined positive angular acceleration should represent disk-pad braking torque just below the steady state peak tyre-road braking torque. The wheel angular acceleration should also be relatively steady during the disk-pad braking torque hold action. The brake hysteresis causes less trouble in this control phase, as the tyre-road braking torque remains at a steady value after the peak split point. The tyre hysteresis could cause problems while disk-pad braking torque is holding and tyre-road braking torque is reverting to the peak split point. Therefore, the true tyre-road braking torque change trend may not be observed (which is the case when the regular disk-pad braking torque hold action is implemented). A sudden decrease in wheel angular acceleration could indicate the presence of the peak split point in this control phase. Negative angular acceleration is expected in next control phase as soon as the peak split point is detected and disk-pad braking torque enters an increase state.

The first suggestion is advantageous in regard to maintaining the peak tyre-road braking torque, but it could result in the lateral force control target being missed. The second suggestion may result in a slight loss in available tyre-road braking torque, but it could achieve efficient lateral force control.

4.6. A discussion of split-mu surface control

The split-mu ABS control is a higher-level control that runs by Electronic Stability Program (ESP). The split-mu control uses the surface friction estimation method during a braking event to clarify the road surface friction on both the left and the right sides of the vehicle. The ABS control keeps the low friction side under the control of the low friction ABS control method. On the high friction side, with the aid of a yaw rate sensor and a steering wheel angle sensor from ESP, the driver steering intention can be extracted from the steering wheel angle and its change speed; and the current vehicle yaw motion can be measured by the yaw rate sensor. Then the disk-pad braking torque is decreased in order to reduce the tyre-road braking torque so that it can match the low friction surface side’s tyre-road braking torque according to the comparison between driver’s demanded and the vehicle’s yaw motion. This control method could reduce the unintended vehicle yaw motion to a safe level, so the vehicle stays controllable by the standards of an average driver.

Apart from the surface friction estimation method, ABS control can compare the value of the angular acceleration of the four wheels before the ABS control kicks in to identify the split-mu condition. If the angular acceleration value of the left side wheels is different from those on the right side, and the difference is over a pre-defined threshold, then a split-mu brake condition could be identified. As such, it can be concluded that the side with the higher absolute wheel angular acceleration value is on a low friction surface and that it should be under the control of the low friction ABS control method. The side with the lower absolute wheel angular acceleration value is on a high friction surface and it should be controlled to match the value of tyre-road braking torque on the low friction side.

4.7. Overview of the implementation of different ABS control methods in different braking situations

There are a number of specific brake situations that the ABS must accommodate:

- All four wheels are on a medium to high friction surface - Normal ABS.
- All four wheels are on a low friction surface - Low friction surface ABS.
- Left and right split: Two wheels on one side of the vehicle are on a medium to high friction surface and the other two wheels are on a low friction surface – Split-mu ABS low friction ABS on low friction side, which decreases disk-pad braking torque on high friction side to match disk-pad braking torque on low friction side (to restore reasonable yaw rate).
- Front and back split: Two wheels are on a medium to high friction surface and the other two wheels are on a low friction surface - Normal ABS for medium to high friction surface, low friction ABS for low friction surface.
- One wheel is on a medium to high friction surface and the other three wheels are on low friction surface – Split-mu ABS.
- One wheel is on a low friction surface and the other three wheels are on a medium to high surface - Low friction ABS for low friction surface, normal ABS for medium to high friction surface.

4.8. Conclusion

This chapter established a frame and detailed the contents of an ABS control method based on an understanding of basic property of tyre forces generation mechanism and accompanying scenarios of ABS controlled disk-pad braking torque and tyre-road braking torque interaction (extracted from extensive analyses of commercialised ABS control methods and related basic physics). A high friction, a low friction and a split-mu ABS control methods are proposed and analysed. The detailed Simulink model of the proposed control methods are presented in next chapter.

5. ABS Control Logic Development & Performance Analysis

5.1. Control logic model development

Based on the control method development discussion and summarisations in chapter 4, a detailed Matlab Simulink control model is developed. This is shown in Figure 5.1. Due to the complex nature of the control model, a clear full page version of the overall model Figure is shown in landscape format and can be found in Appendix 1. A detailed explanation of the individual subsystem of the control model is described and discussed in this chapter.

5.1.1. Description and discussion of overall control logic model

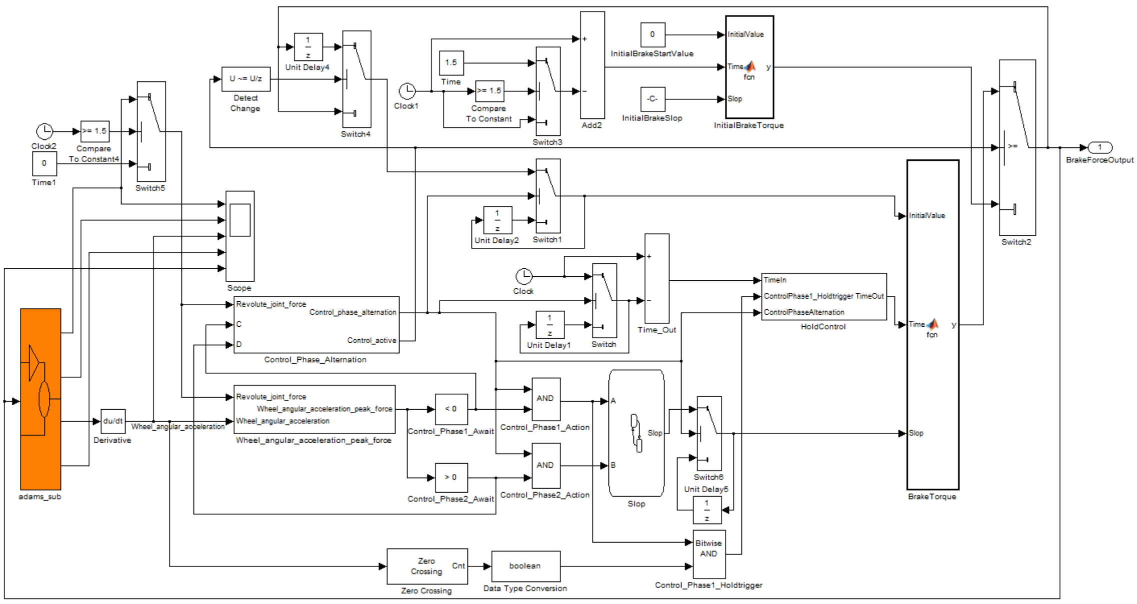


Figure 5.1 Main ABS control logic model

Once the brake pedal is heavily and quickly applied at a vehicle speed of over 5mph, the ABS is initiated. The first stage is to detect if the vehicle is travelling on a low or a high friction coefficient road surface. The ABS then engages a low or high friction coefficient surface control method accordingly. Following this, throughout the rest of the ABS control process, low and high friction surface detection is on. If the road surface friction coefficient changes the ABS changes its control logic, and the low and high friction coefficient road surface detection logic is at a dominant level. If the low-high friction surface detection logic detects that the two left hand wheels are on a different road surface to the two right hand wheels, then a split-mu situation is identified and the ESP split-mu control logic is activated. The lower level ABS control executes the actual disk-pad braking torque control.

On a high to medium friction surface, the control logic detects local peak tyre-road braking force, and at the same time it records both the peak tyre-road braking force and the wheel angular acceleration values. The recorded peak tyre-road braking force is broadcast immediately to establish its peak value drop percentage trigger. This is both an assurance factor and a control performance influence factor. Before the tyre-road braking force peak value drop percentage criteria has been met, the ABS control maintains its current control action, so no activation or control phase alternation will be triggered. The recorded wheel angular acceleration value will be held and the previous reference value will not be replaced. Once the peak value drop percentage criteria has been met, the ABS control activation or control phase alternation is triggered. The recorded reference wheel angular acceleration value is broadcast to the control logic in order to decide which control phase it should enter and to load the control action logic accordingly.

Once the ABS control activation is triggered, the ABS control enters control phase 1. When in control phase 1, the ABS decreases disk-pad braking torque and observes the current measured wheel angular acceleration. When the wheel angular acceleration reaches zero the disk-pad braking torque decrease action is switched off and instead the hold action is switched on. While disk-pad braking torque is in the hold state the control logic waits for the next peak tyre-road braking force detection and its pre-defined drop percentage control phase alternation trigger.

Once the next local peak tyre-road braking force is detected, the peak value is broadcast to establish the peak value drop percentage trigger. The wheel angular acceleration values captured at that moment are recorded but not broadcast. Once the pre-defined peak tyre-road braking force drop percentage trigger is detected, the recorded wheel angular acceleration value is broadcast to the control logic to decide the next control phase action. At this point, the ABS enters control phase 2, the disk-pad braking torque hold action is switched off, and instead, the increase control action is switched on. The

ABS increases disk-pad braking torque until the next control phase alternation trigger is observed.

In control phase 2, two peak value drop percentage values could be set. One is at a higher percentage level and the other one is at a lower percentage level. If a sudden sharp increase in the absolute value of negative wheel angular acceleration is detected before both pre-defined peak value drop percentage values, then once the drop percentage value at the higher level is reached the control phase alternation can be triggered. If the drop percentage value at the higher level is reached first, then a sudden sharp increase in the absolute value of negative wheel angular acceleration is detected (before the drop percentage value at the lower level is reached), and the control phase alternation can be triggered. If both drop percentage values are reached before a sudden sharp increase in the absolute value of negative wheel angular acceleration is detected, the control phase alternation can be triggered after the lower percentage is reached. Once the control phase alternation is triggered, the ABS control enters control phase 1 again.

These control phases are cyclical, so the control actions are switched on and off during the ABS control process. When the brake pedal is released or the vehicle is at a standstill the whole ABS control process is deactivated and the brake control system reverts to pedal only mode.

5.1.2. Peak tyre-road braking force detection

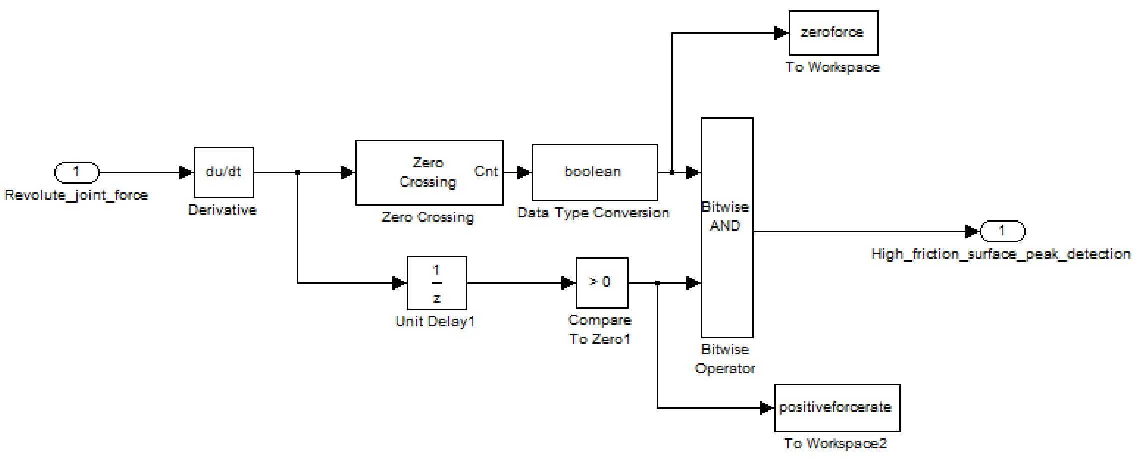


Figure 5.2 Peak tyre-road braking force detection logic model

The tyre-road braking force signal under ABS control resembles a wave shape, and the same sign is present throughout the effective ABS control process. These are based on typical ideal ABS control cycles. The initial logic is to compare the current time step measured value to a previous time step measured value. If the current value is smaller than the previous value, the local peak tyre-road braking force is assumed to have been reached. But it is difficult for this logic to identify a peak value on a negative slope on the tyre-road braking force curve as the nature of the negative slope means that the current value is smaller than the previous value. It might also be difficult for the logic to cope when the tyre-road braking force value remains the same sometime after passing the optimal slip ratio. In this case, the ABS control active process will be delayed so the risk of a wheel lock will increase.

The implemented peak detection logic is used to determine when the slope (derivative) of the tyre-road braking force curve reaches zero. The zero slope detection method is used, but the waves are characterised by upper and lower peaks, and only the upper peak is desired. Considering the nature of the curve, the control logic requires extra logic to identify whether the previous slope was positive. If it is true that there is a zero slope at the current step and a positive slope at the previous step the tyre-road braking force peak is assumed to have been detected. The peak tyre-road braking value is then recorded and broadcast until the next peak value is detected. During the implementation of the peak detection logic, the local peak tyre-road braking force is not fixed. Instead, it updates every time a peak value is detected throughout the ABS control activation period. This gives the logic more ability to cope with changes and is more adaptive for real vehicle implementation. The Simulink model of the proposed peak tyre-road braking force detection logic as described above and can be seen in Figure 5.2.

5.1.3. Control Phase alternation trigger

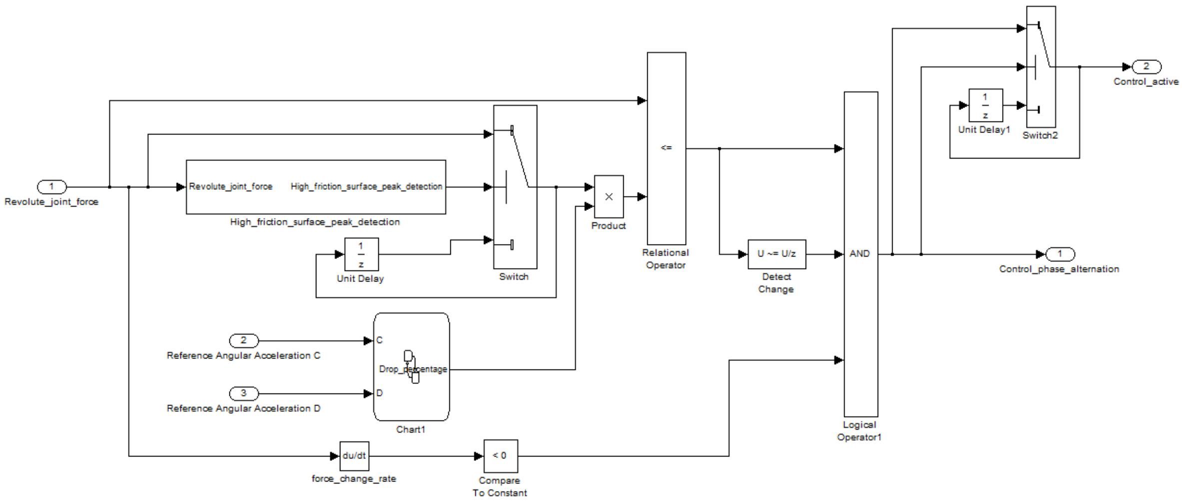


Figure 5.3 Control phase switch trigger logic model

As discussed previously, due to the measurement noise and the vehicle and tyre dynamic nonlinearity, the peak tyre-road braking force detection may not be completely accurate. An assurance factor must be used to ensure the accuracy of the peak tyre-road braking force detection. ABS control action activation or control phase alternation will not materialise until the assurance factor is satisfied.

The assurance factor used in this ABS control logic is based on a previous analysis of the tyre-road braking force curve during emergency braking. On a medium to high friction coefficient road surface, after reaching the peak value at optimal slip ratio the tyre-road braking force starts to decrease. A predefined drop in the percentage of the peak value is used as the assurance factor. This means that once the peak value has been reached, and the current measured tyre-road braking force drops to a pre-defined percentage of the detected local peak value, the ABS control activation or control phase alternation is triggered. In the event that the next detected peak tyre-road braking force is either at the same, or an even greater level than the current peak value, then the current pre-defined drop percentage of the peak value detection will be false within the same control phase (before the next peak is detected). Therefore, the true negative slope logic and the trigger state change logic are added to confirm that the next control phase is only activated once the next peak value is detected and its drop percentage trigger is active again. The Simulink model of the proposed control phase switch trigger logic (described above) can be seen in Figure 5.3.

Ideally, the assurance factor can use just a single point value in a normal simulation environment. But in reality, due to the measurement noise and sensor measurement limitations, a group of calibrated values within a very tight range may have to be used.

5.1.4. Disk-pad braking torque control

5.1.4.1. Disk-pad braking torque hold action in control phase 1

In addition to the assurance factor which assures the correct ABS control activation and control phase alternation in both control phases, the drop percentage is also a very important control action justification factor for the proposed ABS control method, which has a great influence on the ABS control performance. Based on the discussion of control action development in control phase 1, the disk-pad braking torque decreases at its full capability until it reaches the same level as the tyre-road braking torque. This happens when the negative wheel angular acceleration reaches zero, making the control method more simple. At this point, the disk-pad braking torque decrease action stops and its hold action is on until the next control phase alternation is triggered. As discussed previously, when the ABS control is on the disk-pad braking torque changes much faster than the tyre-road braking torque. This means that even when the disk-pad braking torque decreases to the current tyre-road braking torque level and holds its value, the tyre-road braking torque may only have just started to react to the change in the disk-pad braking torque. It takes some time for the tyre-road braking torque to fully achieve its true status (i.e. to be in accordance with the current level of disk-pad braking torque).

If an adequate peak tyre-road braking force drop percentage value is adopted, when the disk-pad braking torque quickly decreases to the same level and holds its position, the current disk-pad braking torque level is sufficient to revert tyre-road braking torque back to its peak level again in a more efficient way during control phase 1. At the same time, if the current disk-pad braking torque level in the hold position is not much below the peak tyre-road braking torque, a faster peak tyre-road braking torque recovery could be expected in control phase 2.

As per the conclusion of the previous discussion, in control phase 1, the decrease in the level of disk-pad braking torque could be justified if the tyre-road braking torque is at a pre-defined peak tyre-road braking force drop percentage. However, in such a scenario, the disk-pad braking torque must decrease quickly enough to avoid the tyre-road braking torque increasing more significantly when the disk-pad braking torque decrease control is triggered in control phase 1. The zero wheel angular acceleration hold action trigger could be implemented in order to simplify the ABS control logic and make it more efficient. So when choosing ABS hardware the ABS controlled disk-pad braking torque change capability must be higher than the tested tyre's tyre-road braking torque reaction capability. The hold control action trigger logic model for control phase 1 can be seen in Figure 5.4.

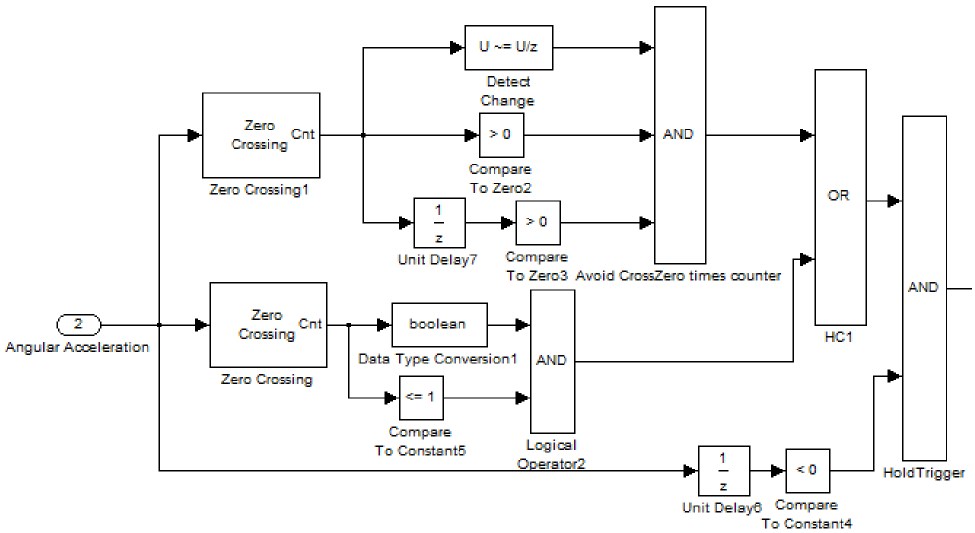


Figure 5.4 Hold control action trigger for control phase 1

5.1.4.2. Comparison between proposed ABS and Bosch ABS, with further discussion on Bosch ABS

The proposed ABS control method can use this zero wheel angular acceleration hold action because the proposed ABS algorithm uses the measured wheel revolute joint force. This offers a good representation of the tyre-road braking force and it can observe the tyre-road braking force conditions more efficiently than the Bosch and Wabco ABS. This means that it could make an earlier control decision. In the Bosch and Wabco ABS, due to the extra logics and determination processes that are required to confirm the status of the tyre-road braking force, the control action decision is delayed and the tyre-road braking force decreases further than its local peak value (considering the fact that the change in disk-pad braking torque is faster than the tyre-road braking torque in control phase 1). If the Bosch ABS quickly decreases the disk-pad braking torque to the same level as the excessively reduced tyre-road braking torque and holds its value, tyre-road braking torque will revert to its local peak value. However, the level of the disk-pad braking torque will be well below that of the peak tyre-road braking torque, so it will take more time and effort for the ABS to recover the tyre-road braking torque to its peak again in control phase 2.

To be specified with previous discussion. In the same disk-pad braking torque decrease and hold control phase, the Bosch ABS uses pre-defined parameters, for example the wheel angular acceleration to ensure effective control performance. But even if the reference wheel angular acceleration is used the disk-pad braking torque could still be decreased too much. To prevent this, the Bosch ABS has to include another pre-defined wheel angular acceleration value to identify the over-decrease and to increase the disk-pad braking torque before reaching next control phase. This would allow the Bosch ABS to regain effective control performance, but this will also cause a further delay in the detection of the next peak tyre-road braking force due to the simultaneous change in both braking torque. At this point, the only observer for the Bosch ABS (wheel angular acceleration) is insufficient in identifying the peak tyre-road braking force. Therefore, the Bosch ABS has to hold the disk-pad braking torque again to observe the status of the tyre-road braking torque via the wheel angular acceleration. As the proposed ABS control method provides a more accurate observation of the status of the tyre-road braking force, the ABS control activation and control phase alternation could be triggered earlier and more efficiently.

5.1.4.3. Disk-pad braking torque control actions model

In the ABS control simulation test the disk-pad braking torque must firstly be brought to a level where the reacting tyre-road torque force reaches its peak value and starts to decrease. Once the ABS control activation or control phase alternation is triggered the representative revolute joint braking force is set as a start reference value for the new control phase. The wheel angular acceleration value is recorded when the peak tyre-road braking force is detected and it is used as a reference to decide which control phase it should be in and which control action is adequate for the current control phase.

As discussed previously, during ABS activation negative wheel angular acceleration indicates it is in control phase 2, but it should enter control phase 1, and disk-pad braking torque decrease control action is required when the next control phase alternation is triggered. Positive wheel angular acceleration indicates that ABS is in control phase 1, but it should have entered control phase 2 and the disk-pad braking torque increase control action is required when new control phase alternation is triggered. The functions described above are translated in the Simulink model as seen in Figure 5.5.

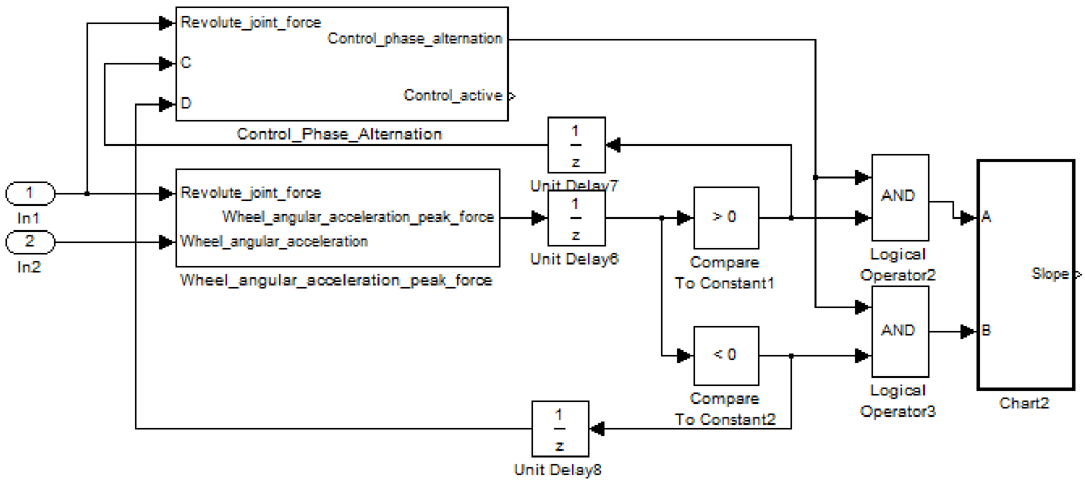


Figure 5.5 Angular acceleration reference determined control action

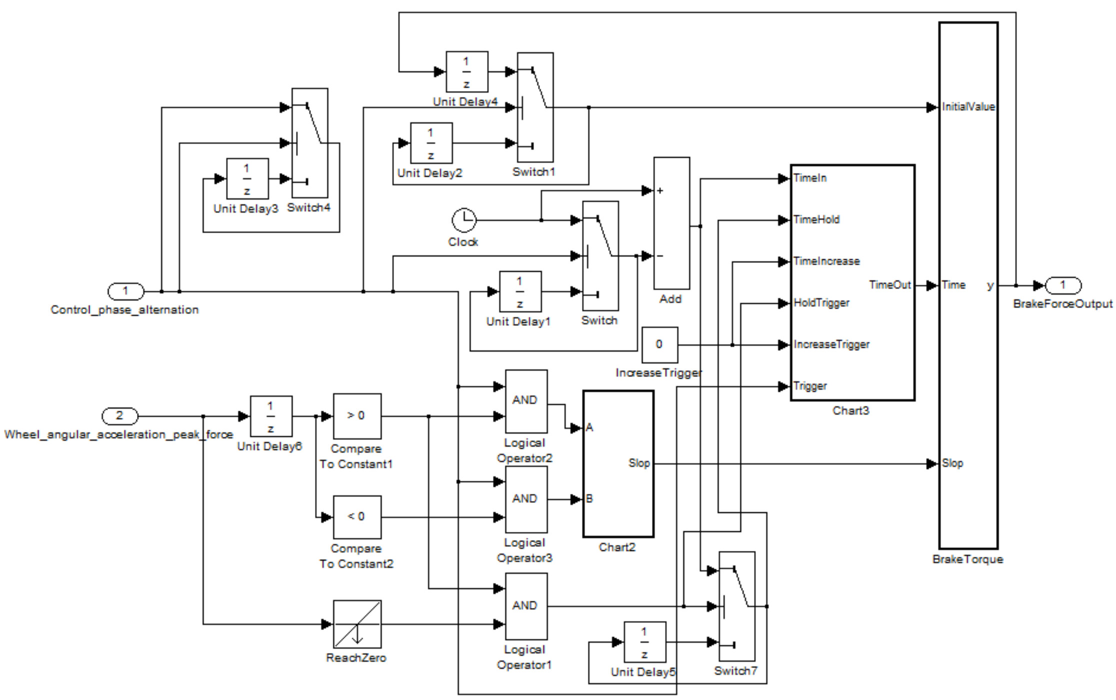


Figure 5.6 Disk-pad braking torque control action model

As seen in Figure 5.6, the slope function is used to simulate the ABS disk-pad braking torque application control. It requires the following parameters: Initial value, slope, and start time. Both the initial value and the start time are taken when the ABS control activation or control phase alternation are triggered. The disk-pad braking torque application slope is a negative value when the ABS is in control phase 1, and it is a positive value when ABS is in control phase 2. The slope value is dependent on the ABS hydraulic control unit's specification, and the increase in the disk-pad braking torque slope usually differs from the disk-pad braking torque decrease slope. The decrease in the disk-pad braking torque decrease is usually faster than the increase in the disk-pad braking torque. During ABS control phase 1, zero wheel angular acceleration is used as a disk-pad braking torque hold trigger. In the slope function, within the same control phase, the initial value and slope can be considered as constant; the only dynamic value is the time. If the wheel angular acceleration changes from a negative value to zero in ABS control phase 1, the time value should be held without any change, so the disk-pad braking torque can hold its current value.

But the potential for the decrease in the initial disk-pad braking torque being insufficient to bring the tyre-road braking torque back to its peak value must be considered, and a further decrease in the disk-pad braking torque may be required. In this case, the initial disk-pad braking torque value could be set to its value in the hold state and the time should be reset to start from the point when the increase request is active. Otherwise, the initial value stays the same as it is when the initial disk-pad braking torque increase control is triggered. When the hold action is triggered the time is recorded and held without being updated. When the increase request is active again in control phase 1, the overall time will be set to start from the moment a request for an increase is triggered plus the previously recorded time when the hold action is triggered. The latter method is arguably better as there is only one changing parameter. The hold and decrease control action logic model can be seen in Figure 5.7, and Figure 5.8 shows the logic flow regarding the decisions to hold and decrease control.

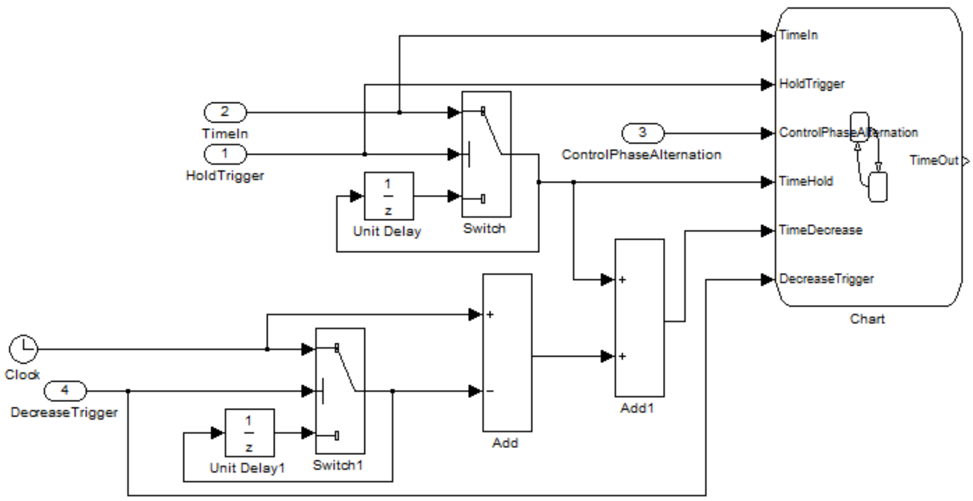


Figure 5.7 Hold and decrease control in control phase 1

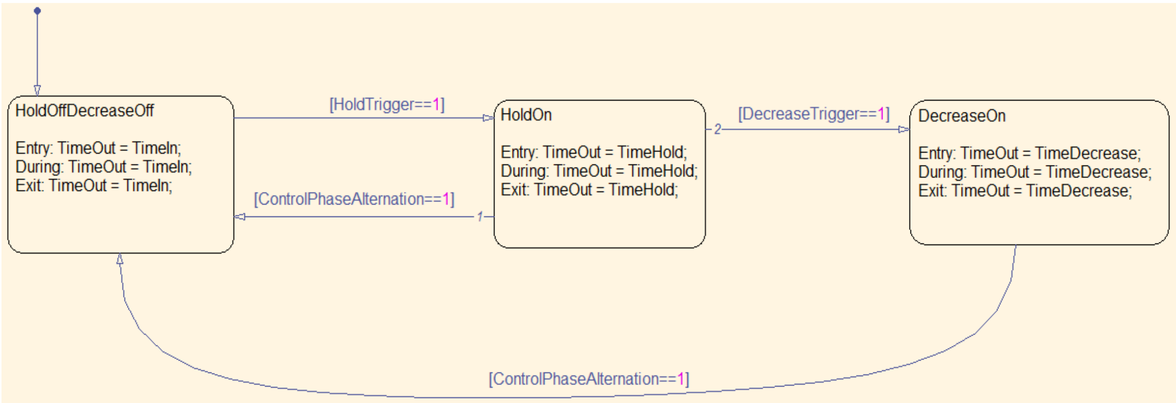


Figure 5.8 State flow within hold and decrease trigger in control phase 1

5.1.4.4. Control phase 2 triggering and disk-pad braking torque control action

The implemented control phase 2 triggering logic is slightly complex compares to the control phase 1 triggering logic, as in this case, the rate of decrease in the tyre-road braking torque after reaching its peak is higher than in control phase 1 triggering case. In order to avoid over decrease in peak tyre-road braking force, the two levels (high and low) drop percentages in peak tyre-road braking force theory proposed at the beginning of this chapter, as well as the sudden sharp wheel angular acceleration change control phase alternation trigger method, are used as a control alternation trigger to improve the control efficiency. This logic is implemented only after ABS activation and once the control phase 1 control action is finished.

As previously discussed, the tyre-road braking force has a sharper increase slope on the left-hand side than on the right-hand side of the optimal slip ratio. This means that the tyre-road braking torque may react faster to the change in the disk-pad braking torque when it is on the left-hand side. In the Bosch ABS the disk-pad braking torque increase control has been divided into many short increase and hold steps. The purpose of this is to observe the status of the tyre-road braking torque and reduce suspension vibration. In the Wabco ABS there are disk-pad braking torque hold actions which, via the wheel angular acceleration, are mainly used to observe the response of the tyre-road braking torque and its status. Once the tyre-road braking force can be monitored directly, the hold actions in control phase 2 could be removed if the delay caused by tyre hysteresis is well understood.

Also, due to the nature of the brake pressure build-up and its releasing processes, the brake pressure build-up process usually takes longer than the brake pressure release process with the same set of ABS components in the same braking environment. Accordingly, the difference in the rate of change between the disk-pad braking torque and the tyre-road braking torque is smaller in the brake pressure build-up phase, and in turn the difference between the two braking torques increases slowly. Therefore, in control phase 2, the tyre-road braking torque increases with the disk-pad braking torque, and the tyre-road braking torque follows the increase in disk-pad braking torque closely. Accordingly, the braking action in control phase 2 is not as complex as in control phase 1, it is designed to keep the disk-pad braking torque increasing until the next pre-defined peak tyre-road braking force drop percentage control phase alternation trigger is active. The same slope function is used to simulate the disk-pad braking torque increase, and the slope value is positive.

5.1.5. Low friction coefficient surface detection

As discussed previously, the logic used to identify a low friction surface is based on the measurements of longitudinal tyre-road braking force on a low friction coefficient surface taken in an experiment. The logic detects a low friction coefficient surface before the tyre-road braking force reaches its peak range. When the tyre-road braking force derivative (change rate) drops below a pre-defined threshold, (while the tyre-road braking force value is below a pre-defined threshold and the same state lasts for a pre-defined time step) then the logic could conclude that a low friction coefficient surface has been detected. The discussed low friction surface detection model is seen in Figure 5.9. The pre-defined tyre-road braking force derivative threshold should be greater than zero, as a zero derivative usually represents a local peak force value.

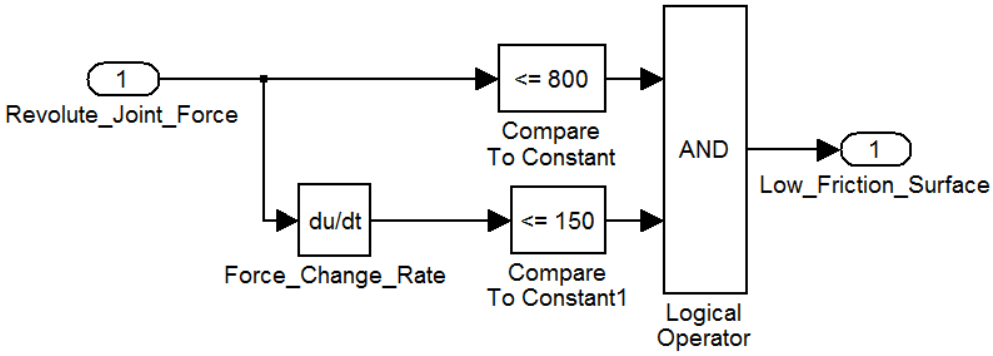


Figure 5.9 Low friction surface detection logic model

The corresponding tyre-road braking force threshold is determined by a summary of experimental test data which represents the overall average tyre-road braking force value when its derivative drops below the specified level. The logic is not used to clarify road surface friction in detail; it is just employed to distinguish between high and low friction surfaces. If a detailed look-up table with tyre-road braking force value at its specified derivative level, under different friction coefficients road surfaces can be established via the in-wheel force sensor measurements in the experimental test, the exact road friction coefficients could be identified. In the current simulation environment, the difference between high and low surface is significant enough to avoid misjudgement, and the pre-defined continuous status identification threshold will not be implemented. In practice, as the peak tyre-road braking force representative wheel bearing longitudinal force is used for the force measurement, calibration must be carried out to compensate for the offset of the two.

5.1.6. Low friction coefficient surface control

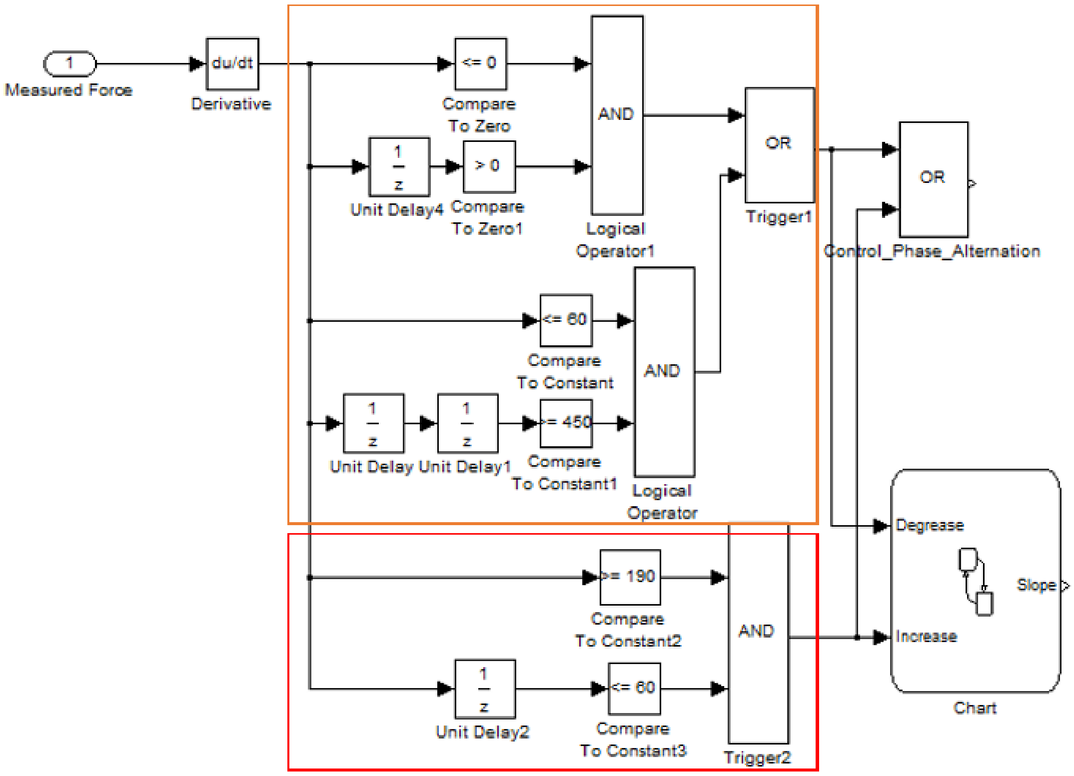


Figure 5.10 Peak split point detection and related control action logic model

Once it has been identified that the vehicle is travelling on a low friction coefficient surface when ABS is active, low friction coefficient surface control is engaged. The first step is to identify the peak split point. This is the point when the tyre-road braking force stops increasing, or even starts to decrease. Due to the nature of the tyre-road braking force curve on a low friction coefficient surface, the peak value drop percentage method used to switch the disk-pad braking torque control phases cannot be used. After reaching its peak value the tyre-road braking force always remains the same or within a similar range. The peak split point identification logic, as seen within orange frame of Figure 5.10, detects the zero or negative tyre-road braking force derivative on an increase slope. In parallel, the logic detects sudden significant drop of tyre-road braking force derivative, which happens when approaching the peak split point, the drop in values is extracted from low friction coefficient surface brake experiment tests in practice. If any of these conditions is met, then the peak tyre-road braking force can be confirmed and the peak split point is identified. In real vehicle implementation, the tyre-road braking force recorded when the peak tyre-road braking force was first detected can be used as a second assurance factor. A range of measured peak tyre-road braking force values, also extracted from low friction coefficient surface brake experiment tests, are compared to the recorded force value. If the recorded force value is within the reference force value range, a peak tyre-road braking force can be confirmed. If all these conditions are satisfied, then the ABS control action is active or a control action alternation is triggered.

Once the peak force and peak split point are detected, it is very important to bring the disk-pad braking torque down quickly. On a low friction coefficient surface the wheel locks condition very quickly. In this disk-pad braking torque decrease control phase, due to the tyre-road braking torque reacting slowly to the change in disk-pad braking torque (even more slowly than on high friction surfaces) and the available tyre-road braking torque being very limited, the preferred control action is to decrease the disk-pad braking torque to a level just below peak tyre-road braking torque. Wheel angular acceleration can be used as a control reference. If the wheel angular acceleration reaches zero before the peak split point is detected again, it can be concluded that the disk-pad braking torque has been decreased close to the peak split point and it should enter hold action. This hold control action trigger model is the same as the one used in high friction ABS control, which can be seen in Figure 5.4. The ABS control logic then waits for the tyre-road braking torque to react to the change in disk-pad braking torque. Once the peak split point is detected again and the tyre-road braking force drops to a pre-defined percentage after the peak split point, the disk-pad braking torque increase control is on, and the control logic enters control phase 2.

The peak split point detection method used in control phase 2 is different from that employed in control phase 1. It uses a current tyre-road braking force derivative which is higher than the pre-defined threshold, and a previous tyre-road braking force derivative which is lower than the pre-defined threshold to detect sudden significant increase of tyre-road braking force derivative. This proposed control model can be seen within red frame of Figure 5.10. In real vehicle implementation, during this control phase, due to the same slow response of the tyre-road braking torque to the change in disk-pad braking torque, many short pulses of the increase and hold control action are preferred in order to avoid over-increasing the disk-pad braking torque. Once the next peak split point is detected, the control logic enters control phase 1 again.

5.1.7. Split-mu control logic

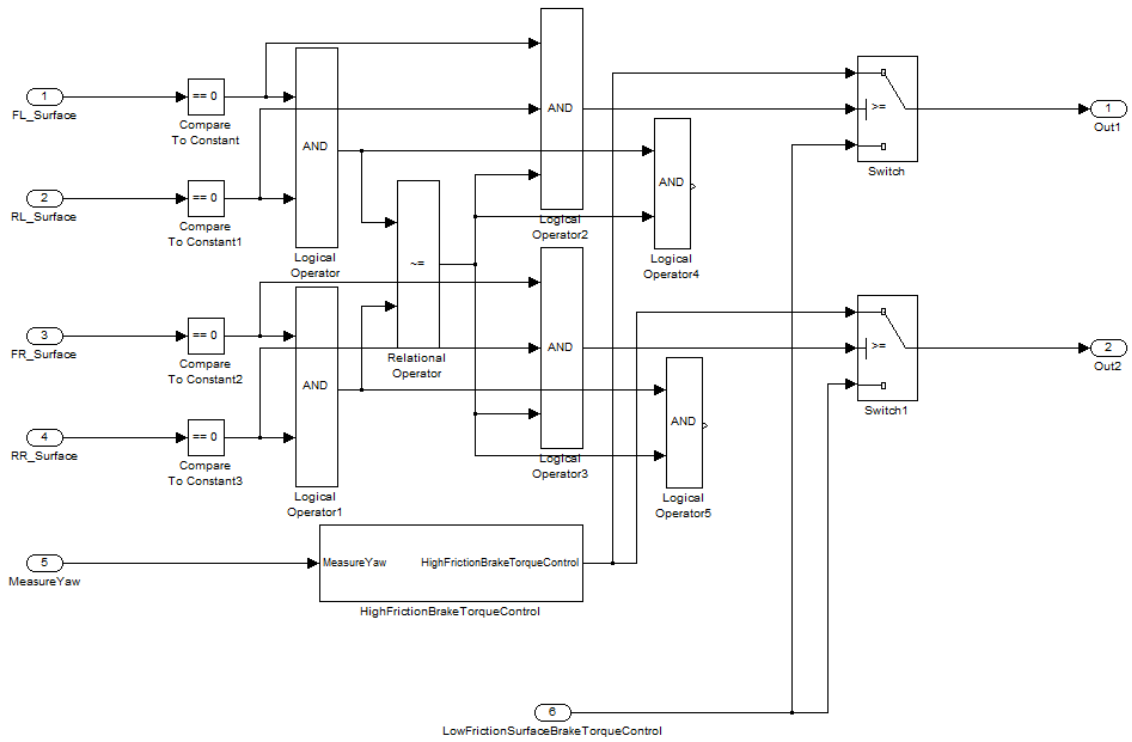


Figure 5.11 Split-mu detection logic model

The split-mu detection logic model, as seen in Figure 5.11, was developed based on the split mu detection method referred to in the discussion at the end of chapter 4. The wheels on a low friction surface are independently controlled using low friction ABS control logic. As seen in Figure 5.12, the wheels on a high friction surface are controlled using a PID controller that regulates disk-pad braking torque to maintain a vehicle yaw rate within a pre-defined reference safety limit. The controlled disk-pad braking torque value is usually higher than the peak tyre-road braking torque on a low friction surface.

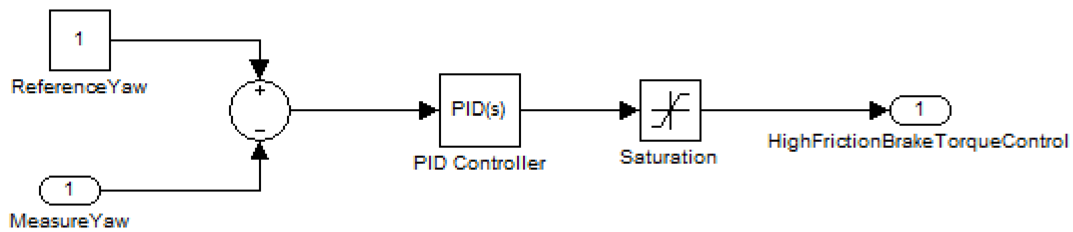


Figure 5.12 PID controller for high friction surface control of split Mu control logic

5.1.8. Conclusion

In this chapter, the entire proposed ABS control logic model is split into many subsystems with different functionalities, and all their functionalities are then translated in Simulink models. These individual sub-models are efficiently connected to each other to form the whole Simulink ABS control model. A high, a low friction and a split-mu control methods were proposed. The detail of the ABS control activation and the control phase alternation trigger differ slightly depending on whether the control method is high or low friction. High friction control uses a pre-defined peak tyre-road braking force drop percentage, while low friction control uses the peak split point. This is due to the difference between the natural tyre-road braking force curves on the two different types of surfaces, and the fact that the very limited amount of available tyre-road braking force on low friction surfaces does not allow much more change in disk-pad braking torque once local peak tyre-road braking force is reached. The control actions of the two control methods are very similar, the only difference is the amount of change in overall disk-pad braking torque is much smaller with regards to the low friction control method. The simulation results of the control models will be presented and discussed in the next chapter.

6. Results Analysis

In this chapter, an emergency brake scenario in which the vehicle is travelling at 80km/h is simulated. The simulation results of the high friction ABS control method, low friction ABS control method, and split-mu ESP and ABS control method are presented and discussed.

6.1. High friction ABS control logic results

In total there are five sets of simulation results with different sets of input parameters. In general, the simulation results can be described as follows: The ‘a’ picture in Figures 6.1, 6.2, 6.3, 6.4, and 6.5 shows the revolute joint longitudinal force time history while the ABS is active. In all the test results it can be seen that the revolute joint longitudinal force is maintained around its peak range. The up and down slide action can be clearly seen, which is the reaction of the tyre-road braking force resulting from the ABS controlled disk-pad braking torque. The slide reaction shows that the ABS control logic tries to keep the tyre-road braking force around its peak range, but due to actuators and ABS control method efficiency limitations, the exact peak tyre-road braking force value cannot be maintained all the time.

The up and down slide action matches the expected response of the proposed ABS control method’s tyre-road braking force, and the control logic maintains tyre-road braking force around its peak range. Each up and down slide motion usually represents one control phase, or half a control cycle. Although each up and down slide motion looks similar, the disk-pad braking torque control actions are different for the two neighbouring up and down slide motions. During ABS control process, the peak tyre-road braking force will not be the same but it should remain similar within a tight range unless the road surface friction changes significantly.

The ‘b’ picture in Figures 6.1, 6.2, 6.3, 6.4, and 6.5 shows the wheel angular acceleration time history. As can be seen, it moves up and down and the zero axis represent a reference middle point. This is also an expected action as acceleration and deceleration indicates that the wheel rotational motion is maintained in parallel to the peak tyre-road braking force range (which is maintained by the control method). From the revolute joint longitudinal force time history, it can be seen at start of each control phase that an immediate effective disk-pad braking torque control cannot change tyre-road braking torque states in the same efficient way. This is represented by a fast change in wheel angular acceleration and a slow change in the tyre-road braking force, as seen in the figures. The control actions are balanced, as there is a similar level of change in wheel angular acceleration on both sides of the zero axis in each control cycle.

The ‘c’ picture in Figures 6.1, 6.2, 6.3, 6.4, and 6.5 shows the wheel longitudinal speed time history. As can be seen, it decreases through time at a relatively stable rate. The observed noise is due to the change in the tyre-road braking force which is the dominant force slowing down the wheel from moving forward. In all these simulation results the wheel longitudinal speed value reduced by more than half within 2 seconds. If these results are compared to the reference braking time data in chapter 2, the performance of this ABS control method is superior.

The ‘d’ picture in Figures 6.1, 6.2, 6.3, 6.4, and 6.5 shows the wheel angular speed time history. It can be seen that it never reaches zero while the vehicle is moving. This directly proves that the wheel rotational motion was maintained during the ABS control process and that accordingly the wheel has not been locked.

The ABS control action can be observed in its time history curve. When wheel angular speed has been decreased too much compared to relative wheel longitudinal speed, the ABS control increases it again, and its overall average curve trace follows the curve trace of the wheel longitudinal speed. Similar traces can be observed between the curves in the simulation results and the Bosch and Wabco ABS control curves, which indicates correct basic ABS control actions. It is important to use the commercialised ABS as a benchmark, not because they set the standard ABS control pattern, but because they follow the natural behaviour of the tyre and adopt a reasonable pattern of control actions. Moreover, the commercialised ABS considers all the practical limitations, which result in an effective brake system and a tyre interaction that slows the vehicle down without locking the wheels.

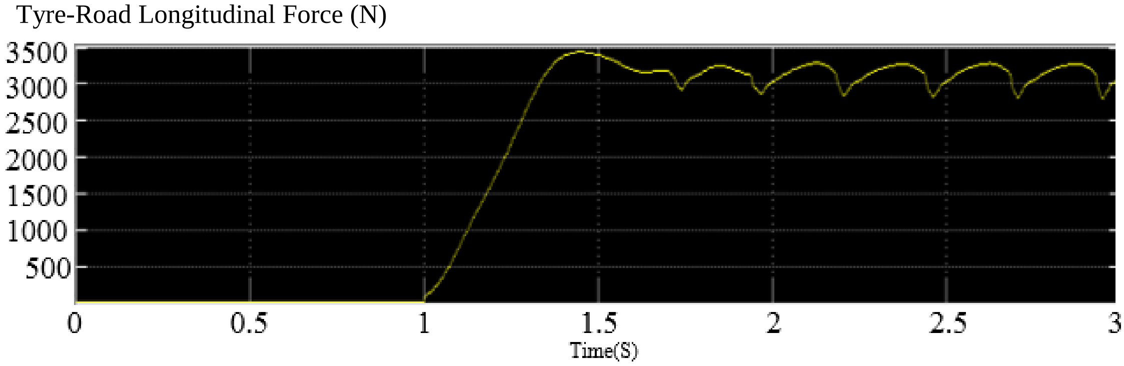
6.1.1. Analysis of the simulation results data

Table 6.1 as seen below is the summary of the key control input parameters in each simulation.

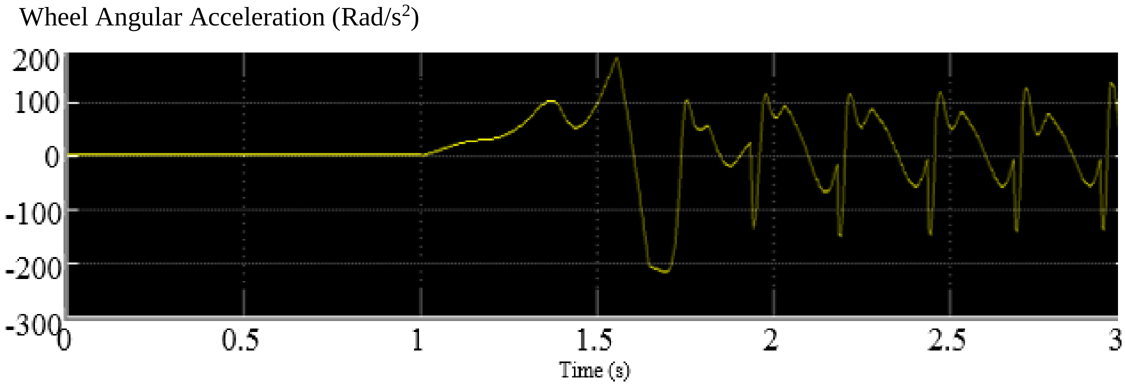
	Disk-pad braking torque decrease rate (Nmm/s)	Disk-pad braking torque increase rate (Nmm/s)	Phase 1 Percentage Drop (%)	Phase 2 Percentage Drop (%)
Parameter Set 1	-2,444,444	1,111,111	97.8	97.3
Parameter Set 2	-2,687,500	1,187,500	97.8	97.3
Parameter Set 3	-12,500,000	5,500,000	97.1	96.5
Parameter Set 4	-12,766,666	4,766,666	98.9	98
Parameter Set 5	-12,766,666	4,766,666	98.9	99.1

Table 6.1 Sets of the simulation control input parameters

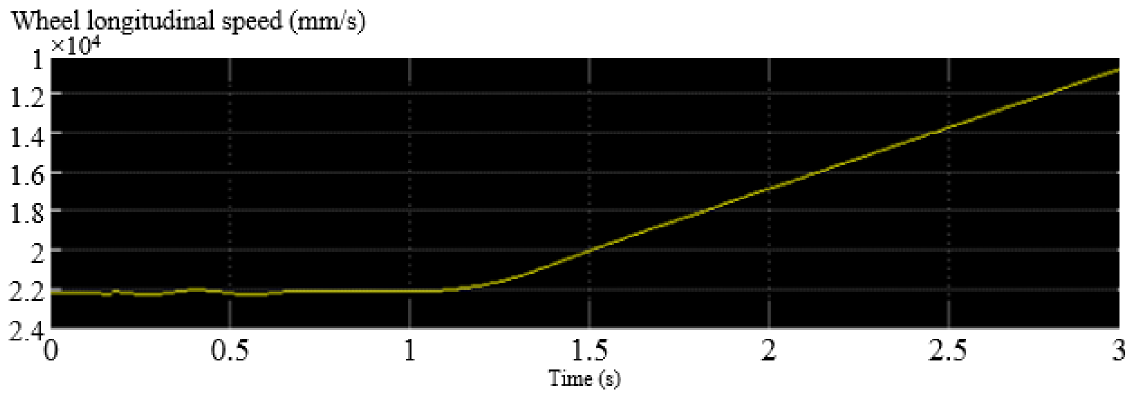
6.1.1.1. Simulation 1 with control input parameter set 1



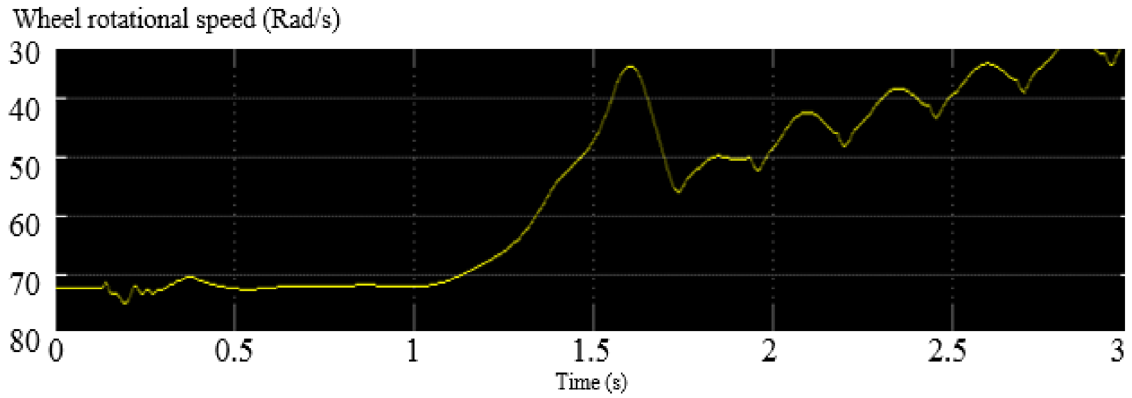
(a)



(b)



(c)

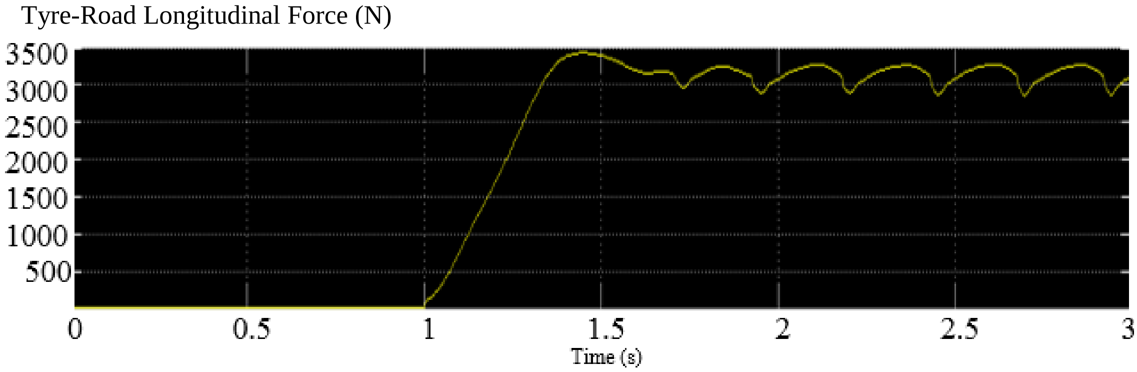


(d)

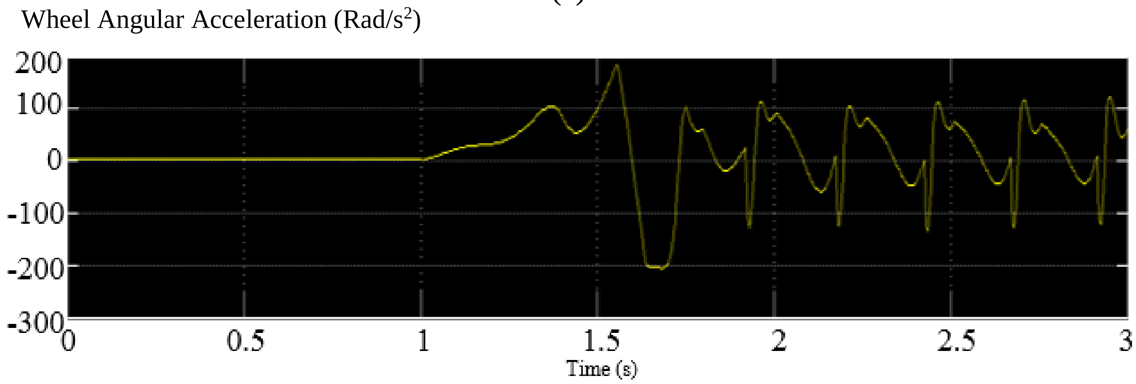
Figure 6.1 ABS control simulation results set 1

In simulation 1, the rate of the decrease in disk-pad braking torque in control phase 1 is -2,444,444Nmm/s and its rate of increase in control phase 2 is 1,111,111 Nmm/s. The percentage drop is 97.8% in control phase 1, and 97.3% in control phase 2. Overall, as seen in Figure 6.1, the initial increase in disk-pad braking torque was too high, and at this point the wheel angular speed was decreased to a very low level compared to the wheel longitudinal speed. This indicates that with this control input parameter set it takes longer for the ABS control logic to activate and decrease the disk-pad braking torque efficiently, especially in the initial stage.

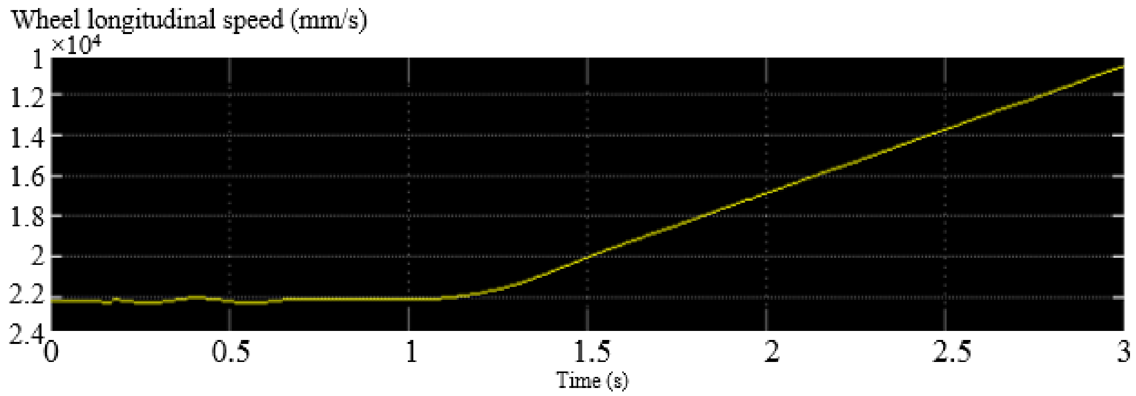
6.1.1.2. Simulation 2 with control input parameter set 2



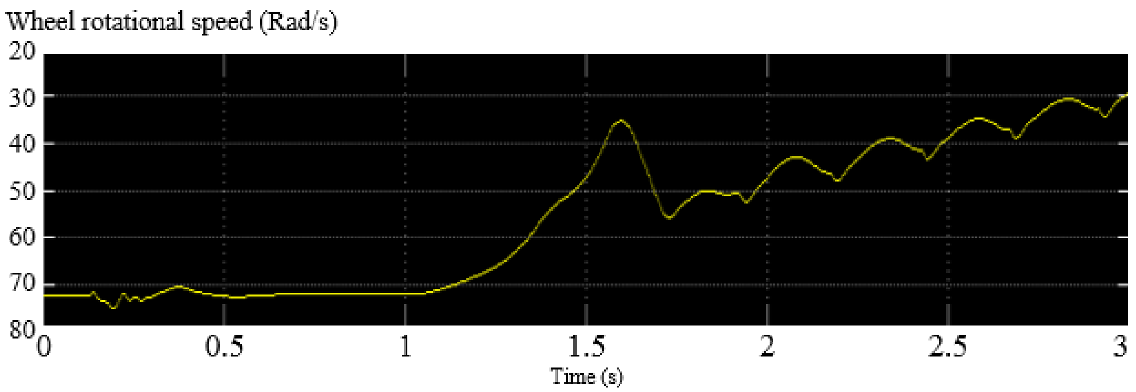
(a)



(b)



(c)



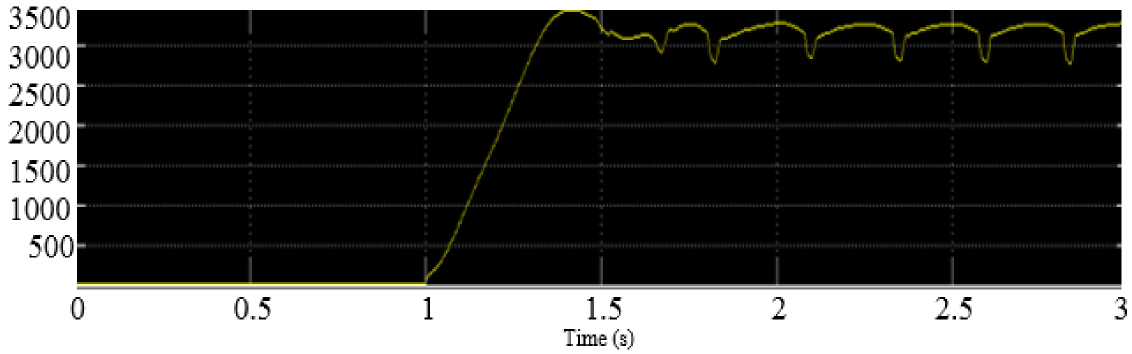
(d)

Figure 6.2 ABS control simulation results set 2

In simulation 2, the rate of the decrease in disk-pad braking torque in control phase 1 is -2,687,500 Nmm/s, while its rate of increase in control phase 2 is 1,187,500 Nmm/s. The drop percentage is 97.8% in control phase 1 and 97.3% in control phase 2. The rates of increase and decrease in disk-pad braking torque are all slightly higher than in the precious simulation. Compared to simulation 1, it can be seen that the difference in control performance is small. A higher rate of decrease is expected to lower the initial over-increase in disk-pad braking torque. The roughness of the wheel angular acceleration curve and the longitudinal revolute joint force curve indicate that the control action needs to be smoothened.

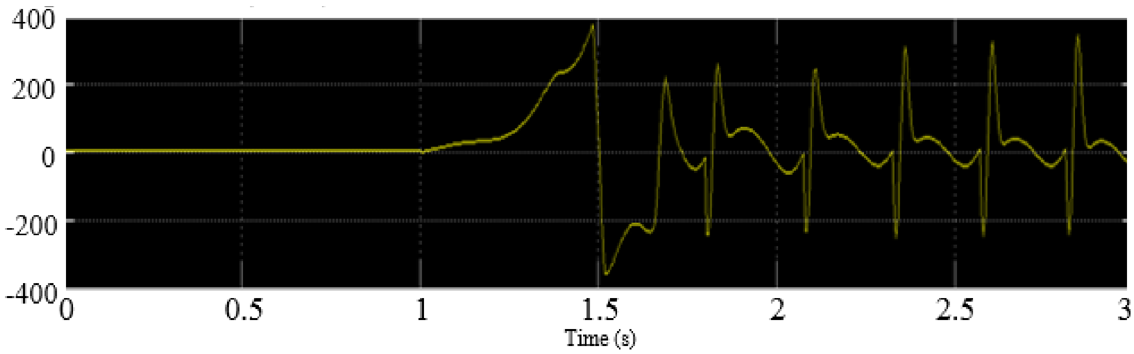
6.1.1.3. Simulation 3 with control input parameter set 3

Tyre-Road Longitudinal Force (N)

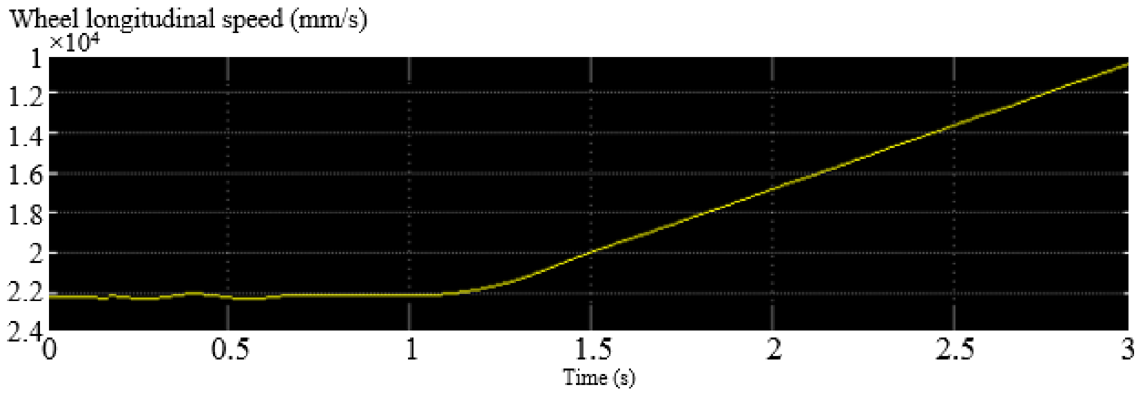


(a)

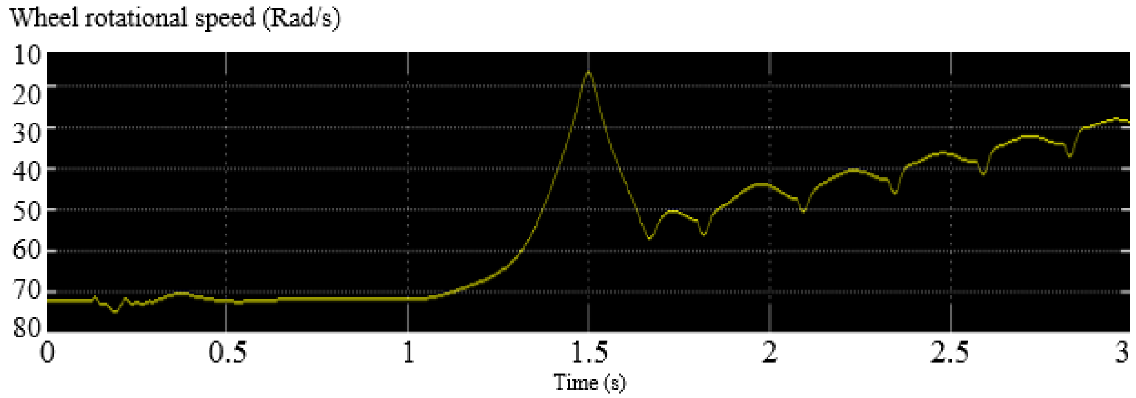
Wheel Angular Acceleration (Rad/s^2)



(b)



(c)

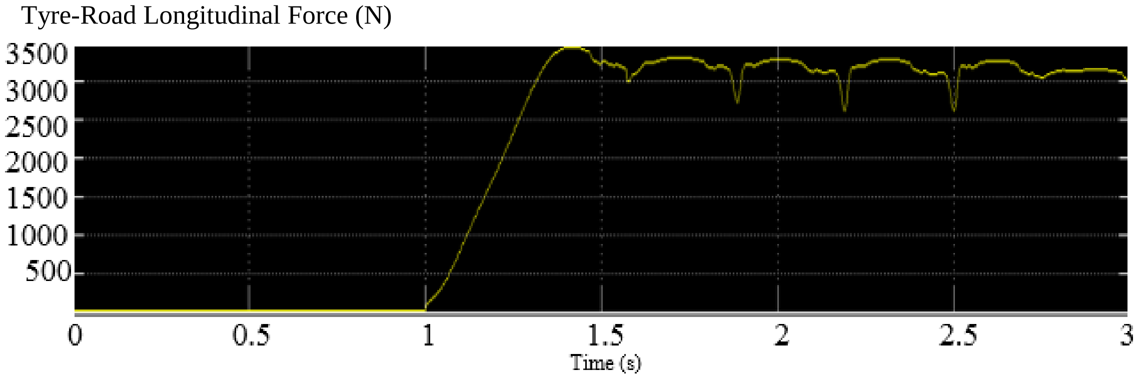


(d)

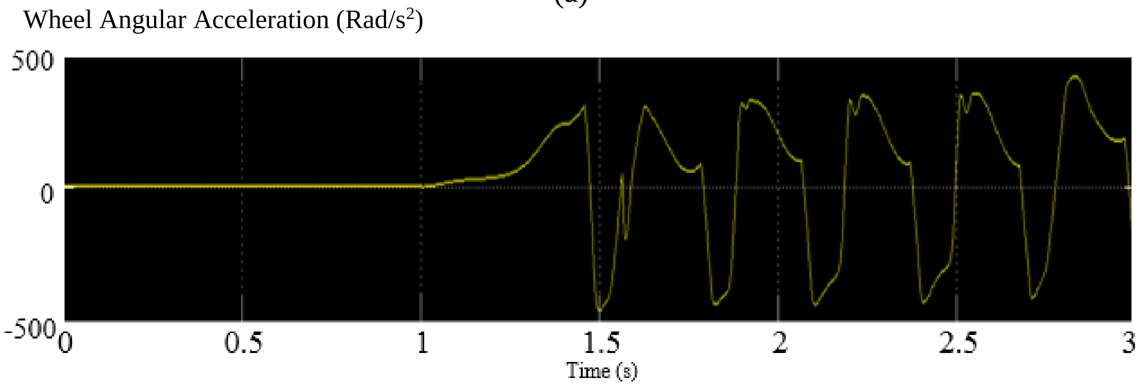
Figure 6.3 ABS control simulation results set 3

In simulation 3, the rate of the decrease in disk-pad braking torque in control phase 1 is -12,500,000Nmm/s. Its rate of increase in control phase 2 is 5,500,000Nmm/s. Compared to the first two simulations, both the rates of decrease and increase in the disk-pad braking torque are significantly higher. The drop percentage in phase 1 has fallen to 97.1%, and in phase 2 it has fallen to 96.5%. As seen in Figure 6.3, the control reaction is even volatile. The initial increase in disk-pad braking torque is excessive and the wheels are almost locked. This rough behaviour is due to the reduced drop percentage which delays the activation of the ABS control, even with a greater decrease in the rate of the disk-pad braking torque. This highlights the importance of the ABS control activation and phase alternation trigger tuning. Further simulations are required to study other related issues. But it can be concluded that the performance of the ABS disk-pad braking torque control is more sensitive to changes in drop percentage.

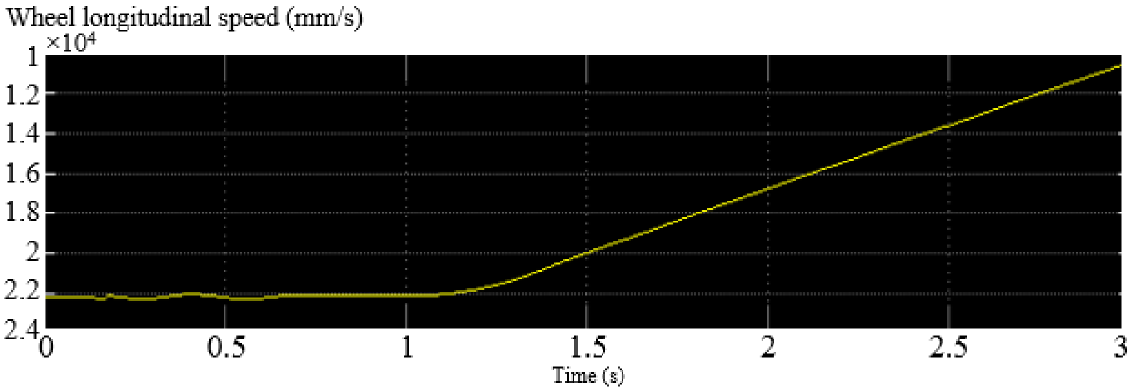
6.1.1.4. Simulation 4 with control input parameter set 4



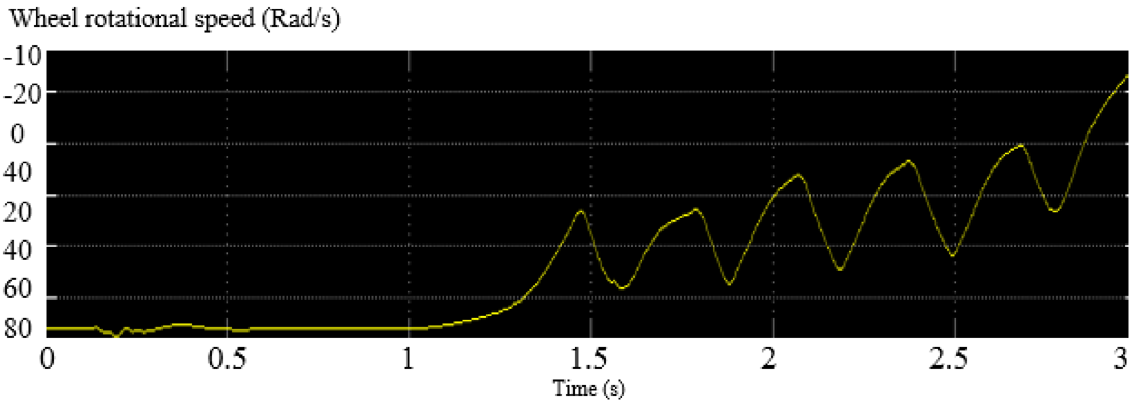
(a)



(b)



(c)

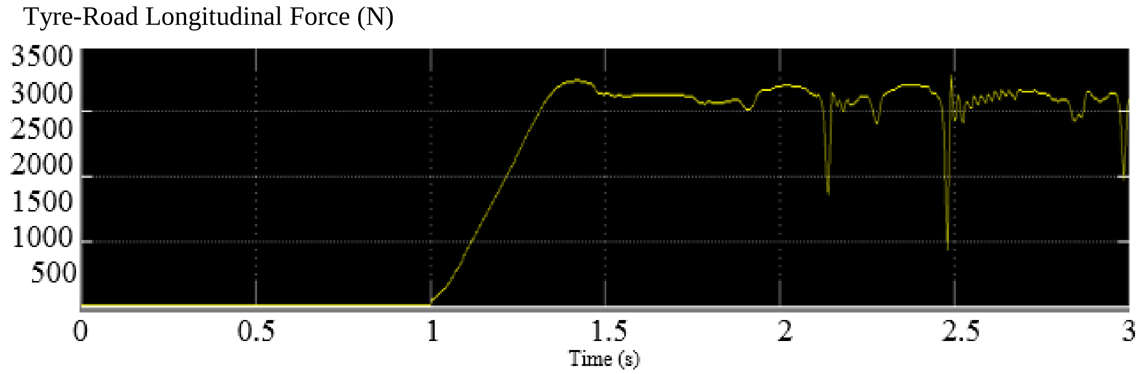


(d)

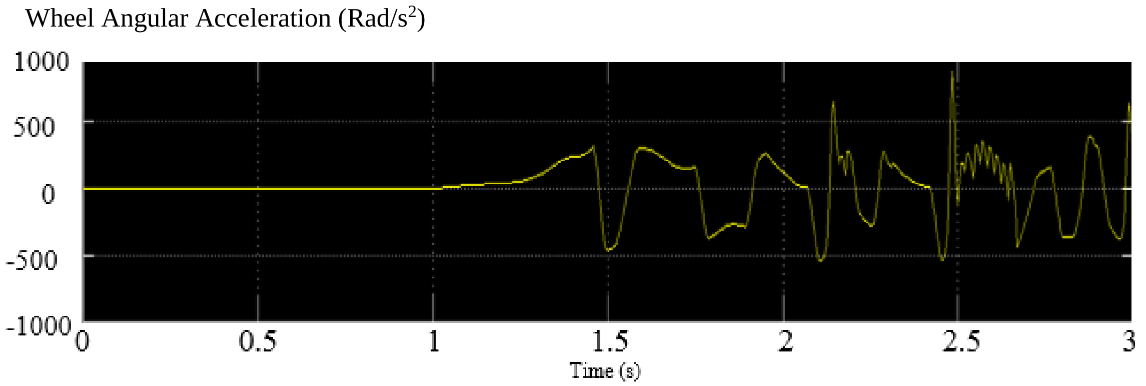
Figure 6.4 ABS control simulation results set 4

In simulation 4, the rate of the decrease in disk-pad braking torque in control phase 1 is -12,766,666Nmm/s. Its rate of increase in control phase 2 is 4,766,666Nmm/s. Compared to the last simulation, both the rates of decrease and increase in the disk-pad braking torque are slightly higher. The drop percentage in phase 1 has increased to 98.9% and to 98% in control phase 2. As seen in Figure 6.4, all the result curves are smoother compared to the last three simulations, especially the wheel angular acceleration curve, which indicates a smoother control action. But in the last 0.2 seconds of the wheel angular speed curve the ABS failed to meet its control target. The wheel angular speed reaches zero while the wheel longitudinal speed is still above a certain level. On the other hand, it can be seen that the wheel angular speed was brought back to a reasonable level in control phase 1, which indicates a good disk-pad braking torque decrease control. Considering that this control input parameter set provides smooth control, the failure could be caused by a lower rate of increase in the disk-pad braking torque in control phase 2 than in control phase 1, and a low drop percentage in control phase 2.

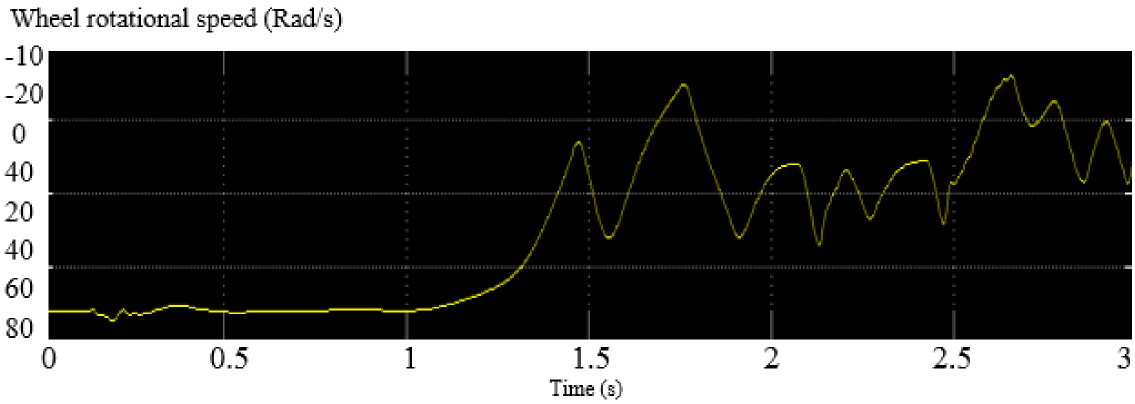
6.1.1.5. Simulation 5 with control input parameter set 5



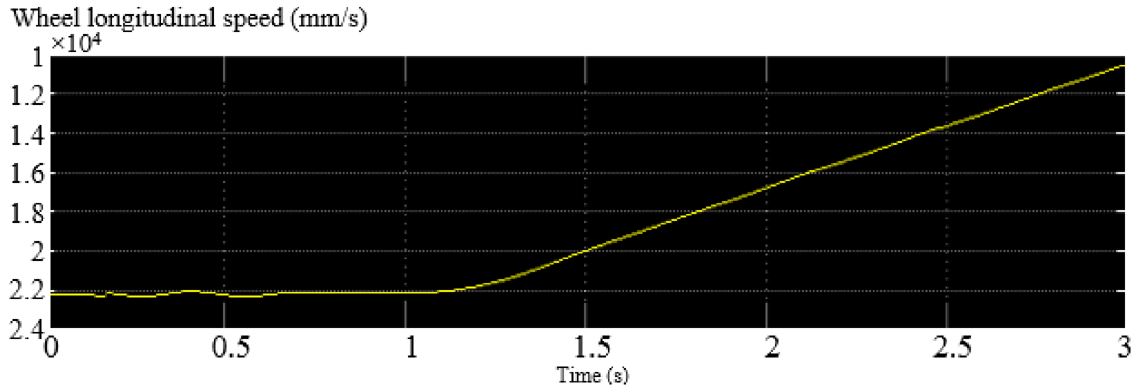
(a)



(b)



(c)



(d)

Figure 6.5 ABS control simulation results set 5

In simulation 5, the rate of the decrease in disk-pad braking torque in control phase 1 is -12,766,666Nmm/s. Its rate of increase in control phase 2 is 4,766,666Nmm/s. Both the rates of decrease and increase in the disk-pad braking torque remain the same as in the last parameter setting. The drop percentage in phase 1 is 98.9% but it has increased to 99.1% in control phase 2. As seen in Figure 6.5, control instability could be observed on the longitudinal revolute joint force curve and the wheel angular acceleration curve. This could be caused by the high drop percentage value in control phase 2. Once the control cycle tightens, the delay in the change of the tyre-road braking force (caused by tyre hysteresis) might not be fully observed by the ABS control method before the ABS control actions are implemented.

The measured longitudinal revolute joint force curve oscillation is explained by the attempts of the ABS control actions to match the delayed response in tyre-road braking force during the initial state of its change in status. The zero wheel angular speed indicates that the overall increase in disk-pad braking torque is excessive in control phase 2. Considering the results of the analysis from simulation 4 (which found a low rate in the increase of disk-pad braking torque in control phase 2), in this simulation, only the drop percentage was increased, the rate of increase in disk-pad braking torque stays the same as in simulation 4, but the simulation analysis results indicate the increase in disk-pad braking torque was excessive. Therefore, it can be concluded that the balance between the change in the rate of disk-pad braking torque and the drop percentage must be well understood and tuned. If attempts are made to improve one without considering the other, optimal control performance is hard to achieve.

6.1.2. Discussion of simulation results

The performance of the ABS control method relies on both pre-defined drop percentage and the disk-pad braking torque change rate, and consequently it is sensitive to changes in both. The higher the drop percentage value, the more quickly the ABS control method can take action and the shorter the ABS control cycle. But the limitations of the force measurement sensor and tyre reaction hysteresis must be considered. A high drop percentage value does not necessarily mean reliable and smooth ABS control; instead it can cause unstable control. This is explained by a combination of limitations in control method efficiency and resonant vehicle dynamic behaviours, especially regarding tyres. In order to reduce the excessive initial decrease in wheel angular speed in the ABS activation phase, a higher rate of decrease in disk-pad braking torque is necessary. But if the rate of decrease is too high it could make the whole control process rougher.

During the simulation tests, with regards to the first three control input parameter sets, very rough wheel angular acceleration curves can be observed. Compared to control input parameter set 4, it can be concluded that the rate of increase in disk-pad braking torque is insufficient (in simulation 4 a much higher rate of increase is implemented). Based on the related wheel angular acceleration curves, an adequately high rate of increase in braking torque could help to smoothen the control process in phase 2. If the rate of increase in disk-pad braking torque is low, it will revert the tyre-road braking torque back to its peak, but the process is slow, so the vehicle or tyres could react unexpectedly. Based on the analyses of all the simulation results, it can be concluded that a well-balanced tuning between drop percentage and the rate of change in disk-pad braking torque, as well as an understanding of the influence of the interaction on the control process (based on the interaction between physical tyre and ABS control), can help to achieve efficient control actions. Preventing wheel lock is always the priority of the whole ABS control logic and this cannot be compromised, even if the resulting control process is rough and could cause discomfort to passengers. No control logic and control input parameter set is perfect, but a well-balanced logic and control input parameter set which makes compromises between the key ABS control performance requirements can help to achieve reliable and smooth control in many different emergency brake situations. The tuning of real vehicle test and control parameters is a very important process in establishing balance in the implementation of the ABS control logic due to the non-linearity and uncertainty involved in real emergency braking scenarios.

6.2. Low friction coefficient surface ABS control method results

The simulation test environment is set for emergency braking on a low friction coefficient surface with a 0.2 friction coefficient and an initial 80km/h vehicle speed. The low friction ABS control method is implemented. Both the predictive method proposed for peak split point confirmation, and the drop percentage method, which is similar to the one used in high friction ABS control method, are used, as some decrease in tyre-road braking force (after it reaches its peak point) can be observed during the simulation. The combined effect of the two control action triggering methods improves the overall efficiency of the low friction ABS control method. Therefore, the drop percentage value can be set at a relatively high level.

6.2.1. Simulation 1 with control input parameter set 1

In simulation 1, the low friction ABS control method did not implement the disk-pad braking torque hold action while the disk-pad braking torque was decreasing (during which the zero angular acceleration point is detected). Accordingly, the disk-pad braking torque decreases until it is confirmed that the peak split point has been passed. The purpose here is to compare the control performance of this aggressive control method with the balanced control method used in simulation 2. The rate of the decrease in disk-pad braking torque in control phase 1 is -200000Nmm/s. Its rate of increase in control phase 2 is 120000Nmm/s. The drop percentage in phase 1 is 99% but it increases to 98.5% in control phase 2.

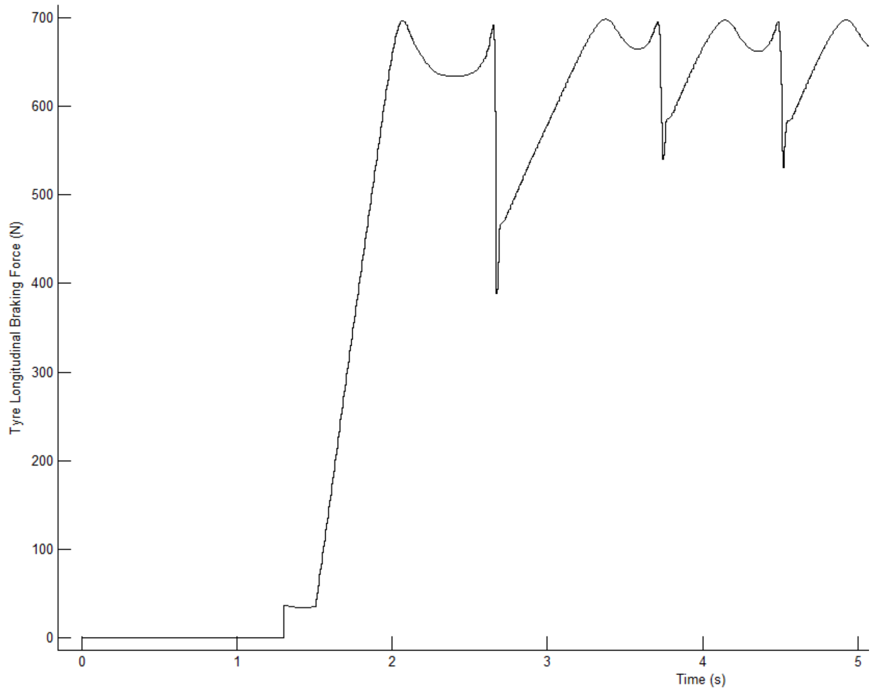


Figure 6.6 Low friction set 1 tyre longitudinal braking force

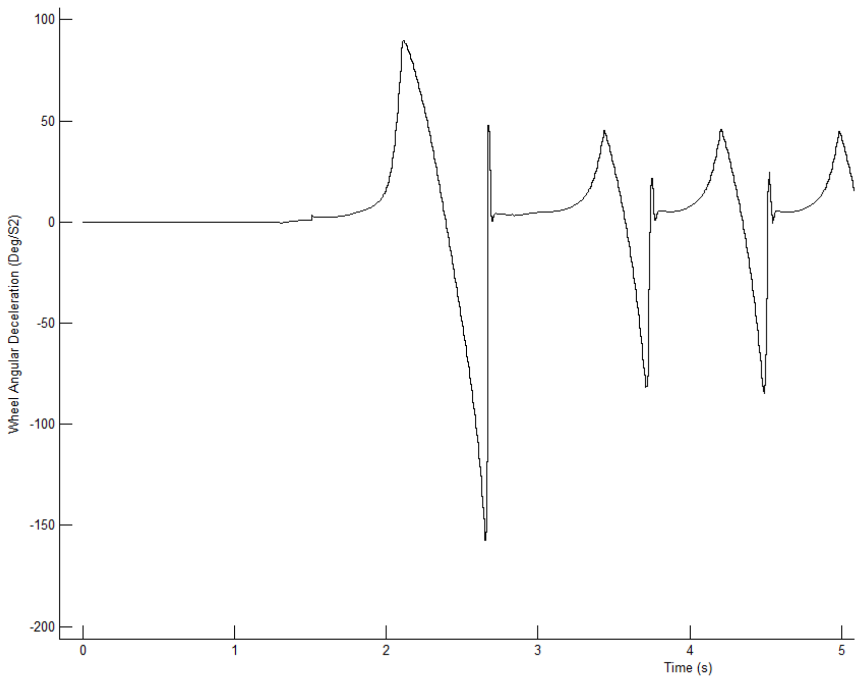


Figure 6.7 Low friction set 1 wheel angular deceleration

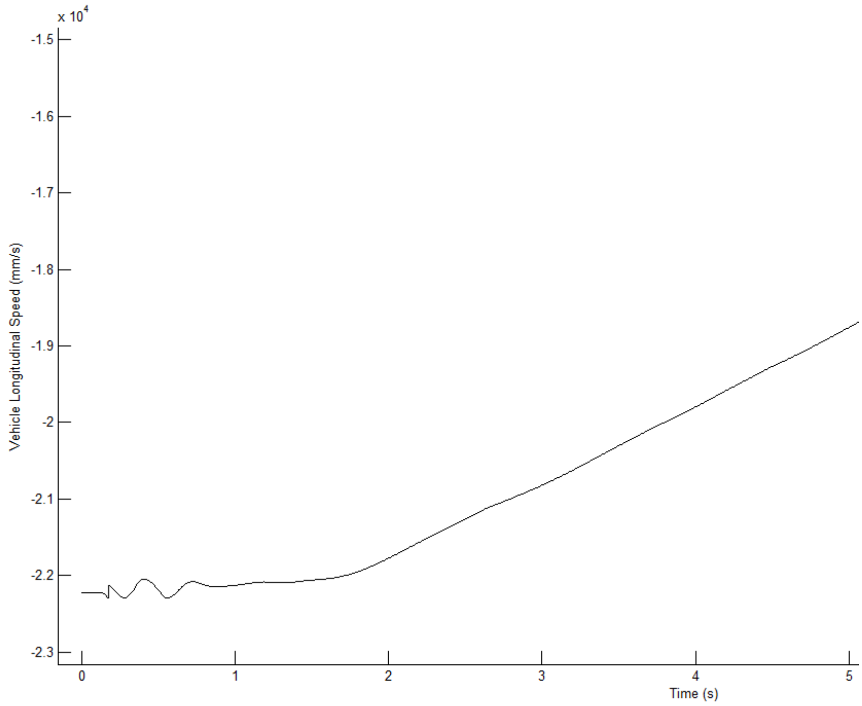


Figure 6.8 Low friction set 1 vehicle longitudinal speed

It can be clearly seen that the available peak tyre-road braking force on a low friction surface is very limited compared to on a high friction surface. It drops to around 700N, which is around one fifth of the peak force (around 3500N) on a high friction surface. It also becomes more sensitive to the change in disk-pad braking torque, especially when the ABS control first kicks in. As seen in Figure 6.6, tyre longitudinal braking force drops below 400N, which causes a significant change in wheel angular deceleration, which can be seen in Figure 6.7. Although it stabilises later, the ABS control decreases the tyre-road braking force to below around 550N on average, when it is in the braking torque decrease phase. As seen in Figure 6.8, due mainly to the overall low levels of peak tyre-road braking force and the inefficient use of the available tyre-road braking force, the vehicle speed slows considerably compared to on a high friction surface. After 3.5 seconds of ABS control, it only drops to 1580mm/s. As the available peak tyre-road braking force on a low friction surface is very limited, the loss of precious available tyre-road braking force must be minimised, and at the same time the risk of wheel lock must be eliminated. Accordingly, an improved ABS control with a braking torque hold action is implemented and tested next.

6.2.2. Simulation 2 with control input parameter set 2, with hold control action

In simulation 2, the rate of decrease in disk-pad braking torque in control phase 1 is -200000Nmm/s. The rate of increase in control phase 2 is 120000Nmm/s. The drop percentage in phase 1 is 99% but it decreases to 98.5% in control phase 2. All these control input parameters are the same as in simulation 1, but the zero angular acceleration triggered disk-pad braking torque hold action is added during the disk-pad braking torque decrease control phase.

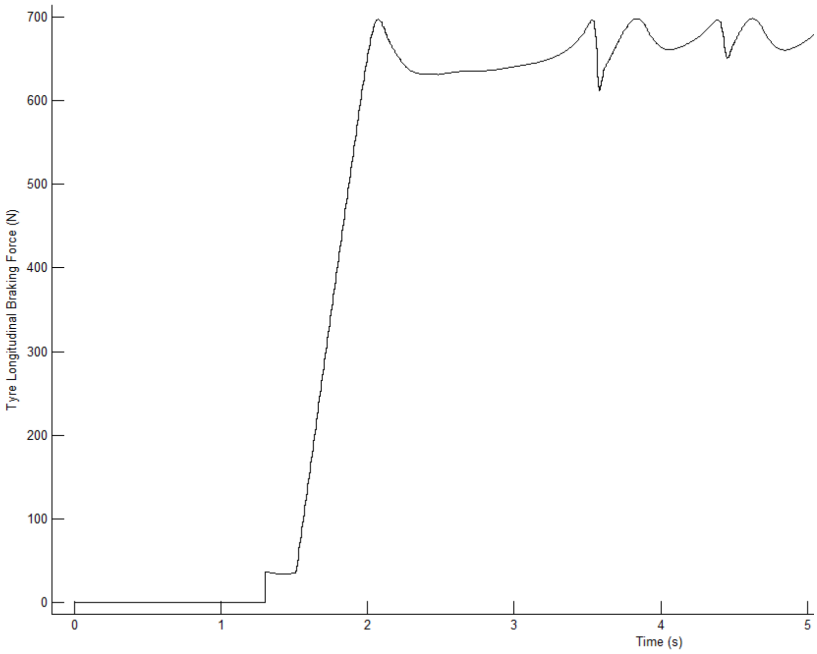


Figure 6.9 Low friction set 2 tyre longitudinal braking force

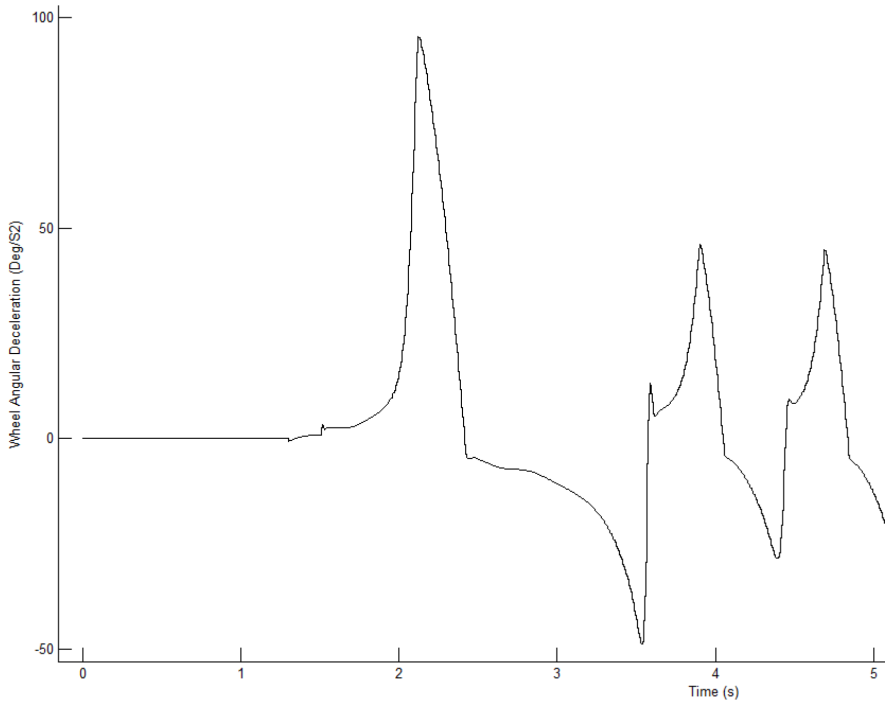


Figure 6.10 Low friction set 2 wheel angular deceleration

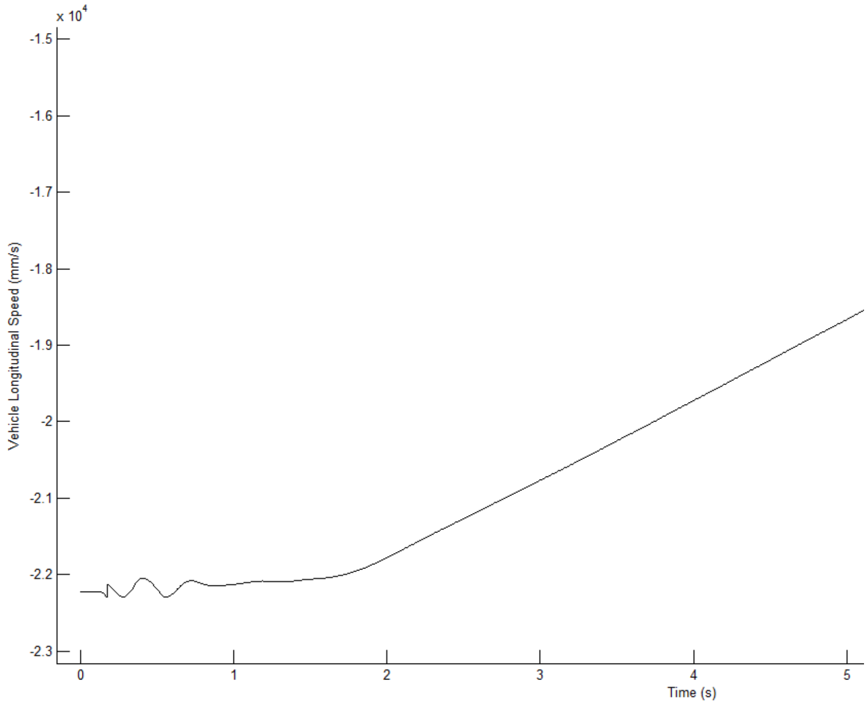


Figure 6.11 Low friction set 2 vehicle longitudinal speed

As seen in Figure 6.9, after the adoption of the hold action, the initial steep tyre-road braking force drop is fixed, so is the significant change in the wheel angular deceleration that can be seen in Figure 6.10. The average drop in tyre-road braking force was limited to above 600N. The control cycle is longer, so the proposed control action is more achievable for low level ABS control actuator. The control reaction is more stable, and the average use of available tyre-road braking force is slightly better than the ABS control without the braking torque hold action, so the vehicle speed slows down slightly more within the same time interval as seen in Figure 6.11.

6.2.3. Simulation 3 with control input parameter set 3

In simulation 3, the rate of decrease in disk-pad braking torque in control phase 1 is changed to -220000Nmm/s. Its rate of increase in control phase 2 is 120000Nmm/s. The drop percentage in phase 1 is increased to 99.5%, and it increases to 99% in control phase 2.

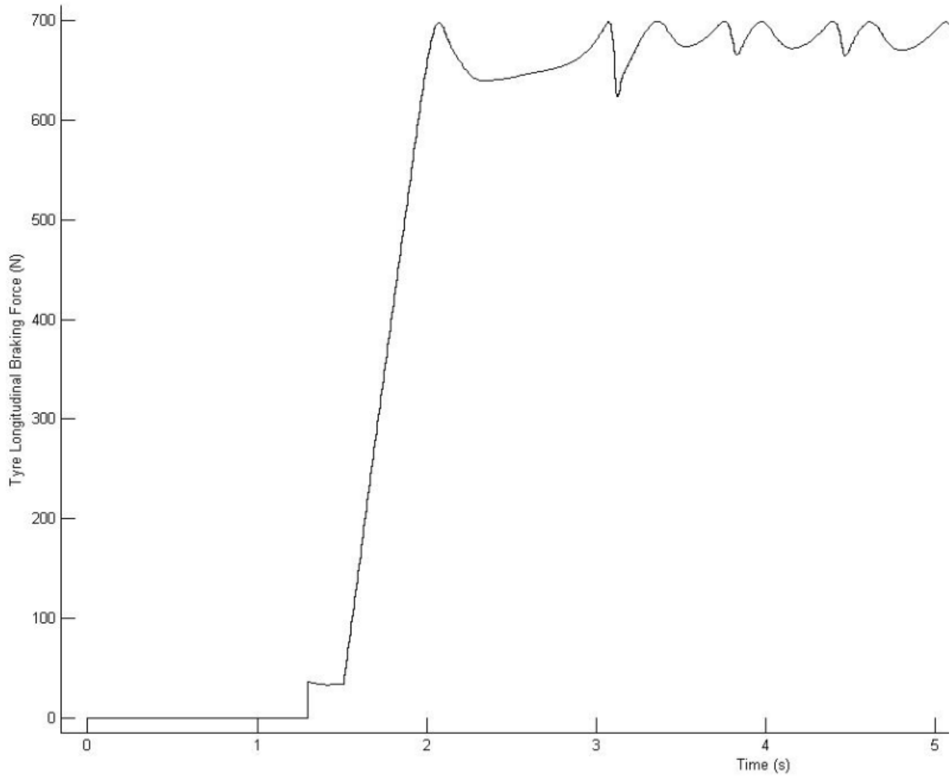


Figure 6.12 Low friction set 3 tyre longitudinal braking force

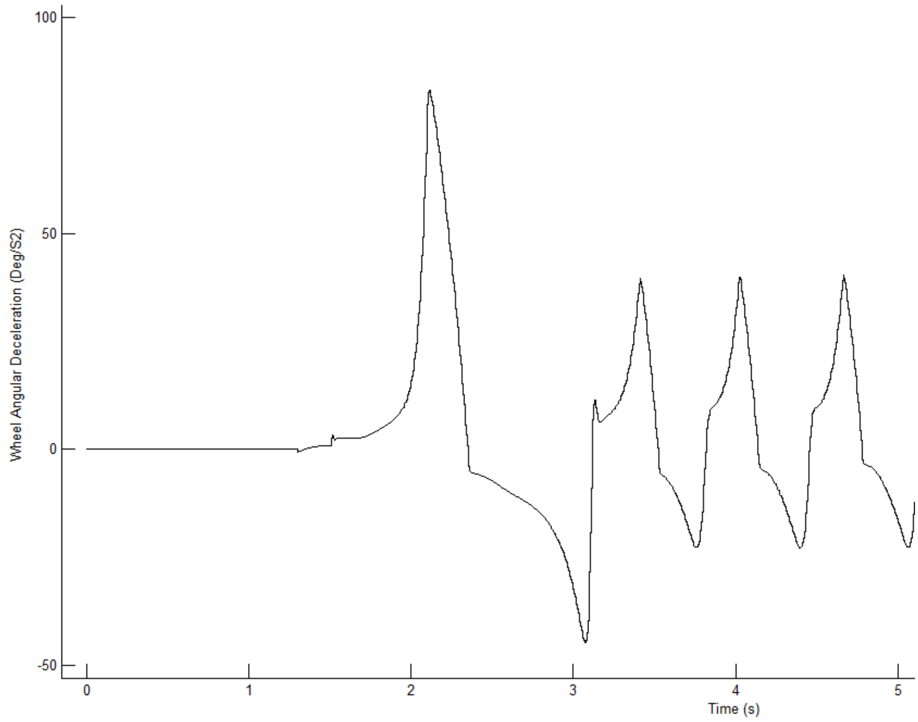


Figure 6.13 Low friction set 3 wheel angular deceleration

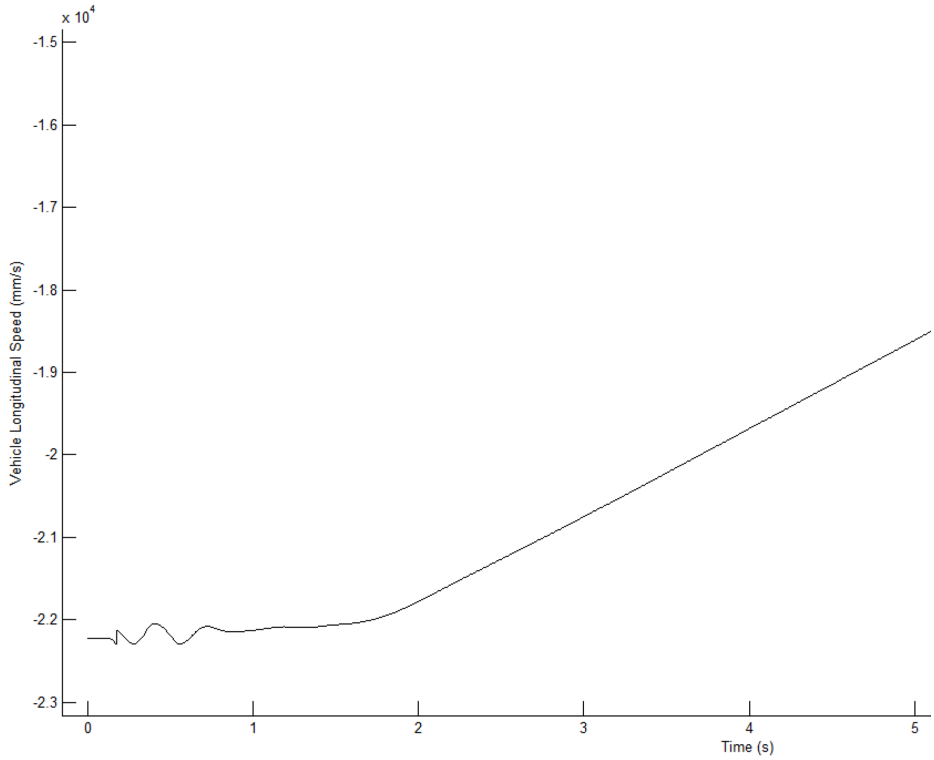


Figure 6.14 Low friction set 3 vehicle longitudinal speed

This represents further tuning of the low friction ABS control with braking torque hold action. On the one hand, the control cycle is even tighter as both drop percentages are advanced, which results in a faster disk-pad braking torque decrease action. On the other hand, a decent level of overall stability in the control actions is maintained due to the efficiency of the control method and the healthy tuning of the control input parameters. This can be observed in Figure 6.13 that the overall change in wheel angular acceleration and its change overshoot are reduced, which indicates an even smoother control process. All these improvements result in overall better use of available tyre-road braking force, which can be observed in Figure 6.12. Accordingly, the vehicle speed slows down even more, as seen in Figure 6.14.

6.2.4. Low friction ABS conclusion

The overall performance of the proposed low friction ABS control is satisfactory. The tyre longitudinal braking force is cycled around its peak range, which indicates the efficient use of available tyre-road braking force. Therefore, a possible shorter braking distance could be achieved. It can be seen that the rotational wheel is cycled between deceleration and acceleration, which indicates the wheel rotating motion is under controlled manner, so the tyre's lateral force generation capability is maintained. It can also be seen that the disk-pad braking torque control is smooth and stable, which makes it easier for the ABS actuator to handle, so better ABS stability and durability could be achieved.

6.3. Split-mu ABS control method results

The simulation test environment is set for an emergency braking situation on a split-mu road surface with an initial 80km/h vehicle speed. It is a low friction road surface with a 0.2 friction coefficient on the left-hand side of vehicle and a high friction road surface with a 1 friction coefficient on the right-hand side. The low friction ABS control method is implemented on the left-hand side of the vehicle, while the disk-pad braking torque matching control method is implemented on the right-hand side. This ensures that the generated tyre-road braking force matches the tyre-road braking force which can be generated on the low friction surface side in order to avoid unwanted yaw motion which causes a conflict between the direction of the vehicle and the driver's intended trajectory. The inputs in the low friction ABS control are as follows: The rate of decrease in disk-pad braking torque in control phase 1 is -220000Nmm/s. Its rate of increase in control phase 2 is 120000Nmm/s. The drop percentage in phase 1 is 99.5%, and it is 99% in control phase 2.

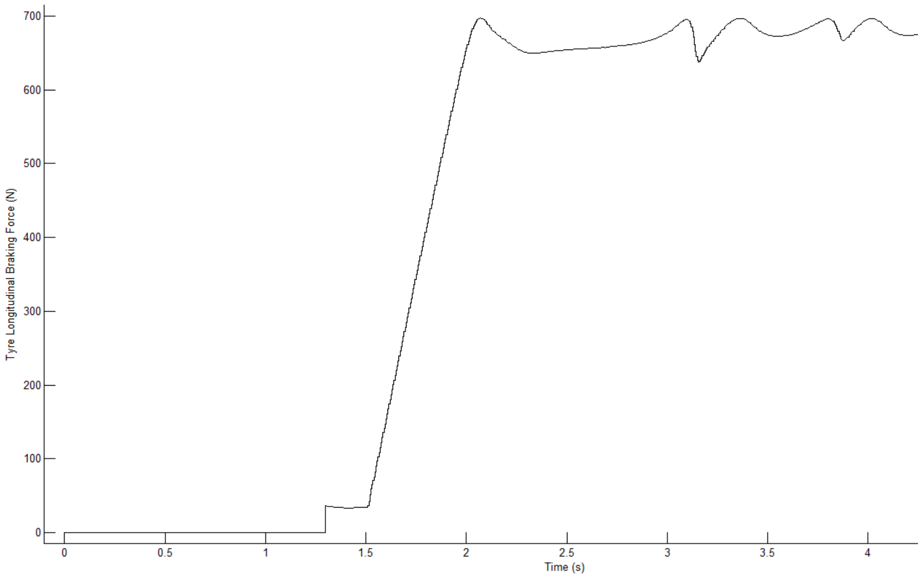


Figure 6.15 Split-mu tyre longitudinal braking force on low friction surface

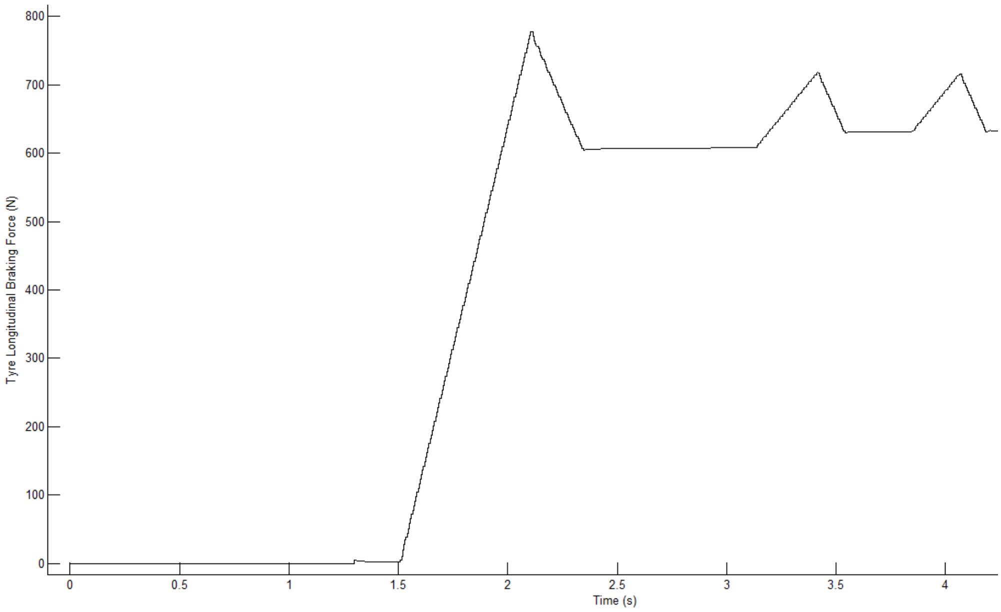


Figure 6.16 Split-mu tyre longitudinal braking force on high friction surface

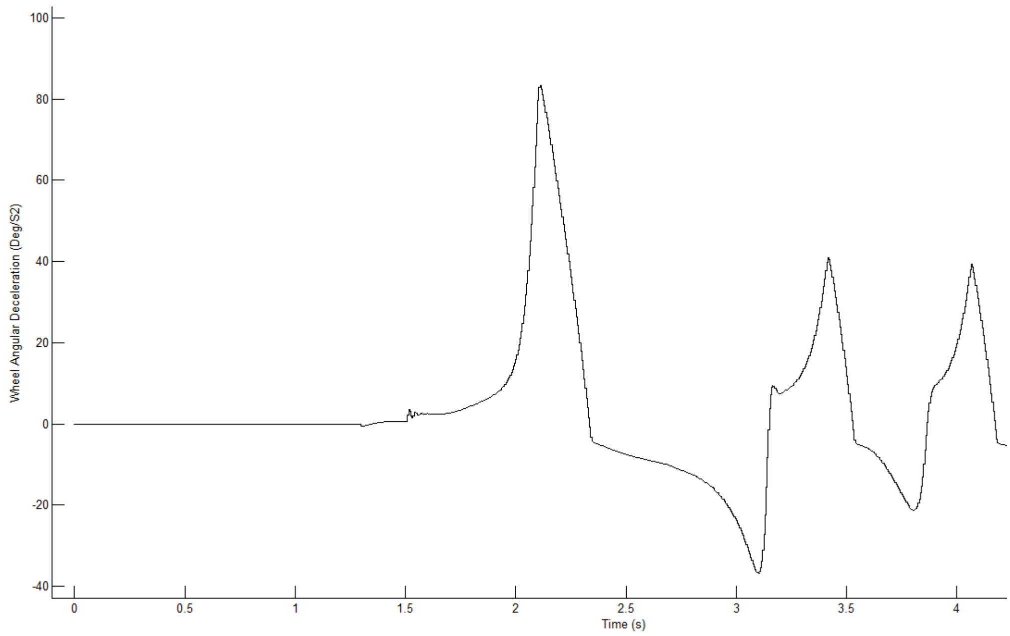


Figure 6.17 Split-mu wheel angular deceleration on low friction surface

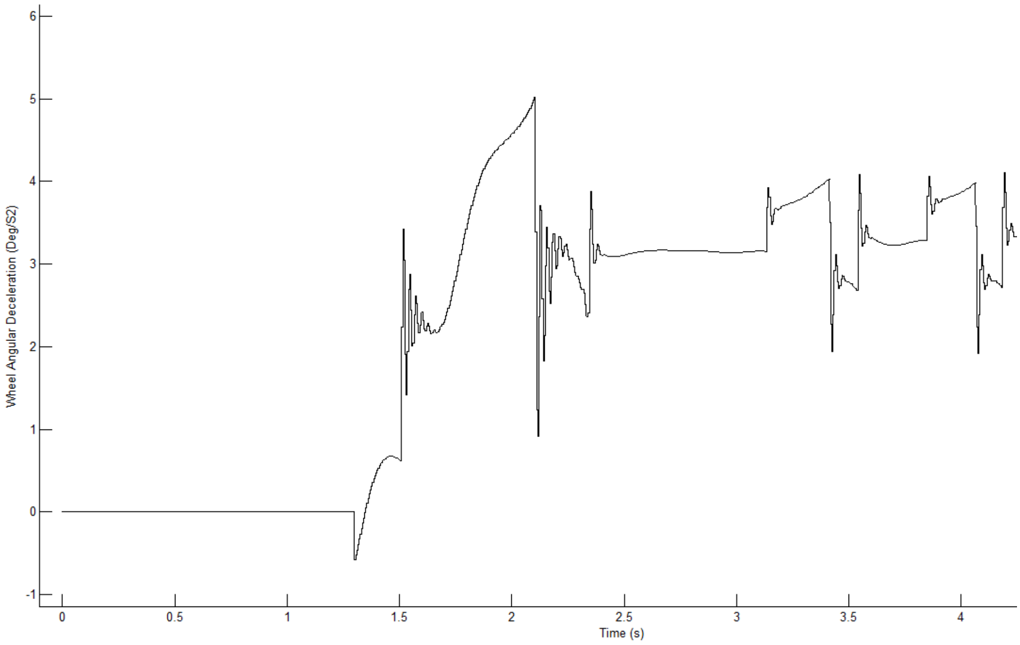


Figure 6.18 Split-mu wheel angular deceleration on high friction surface

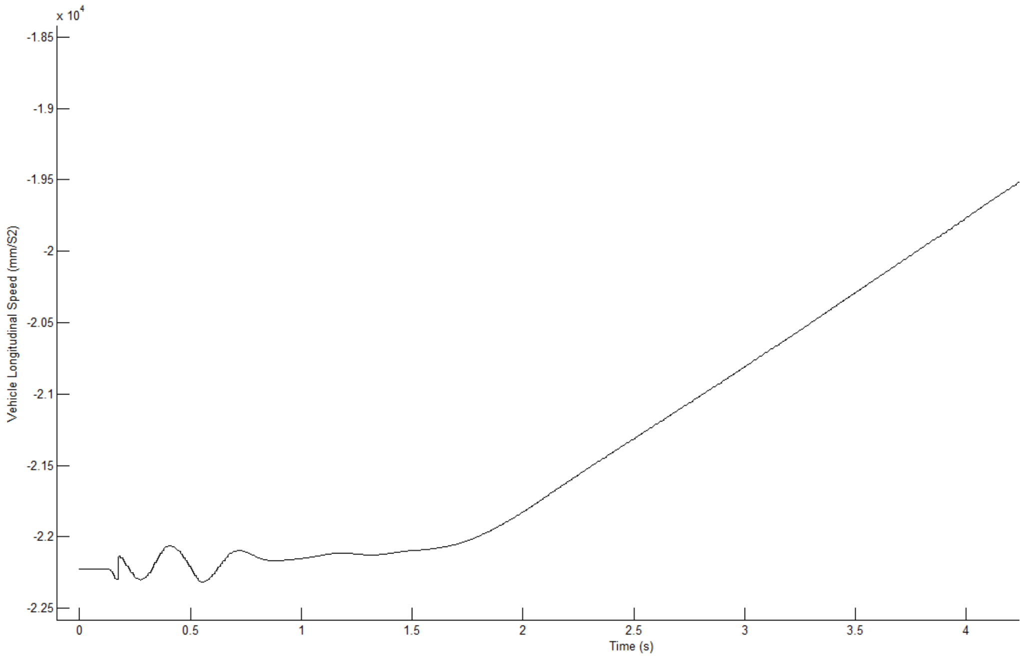


Figure 6.19 Split-mu vehicle longitudinal speed on low friction surface

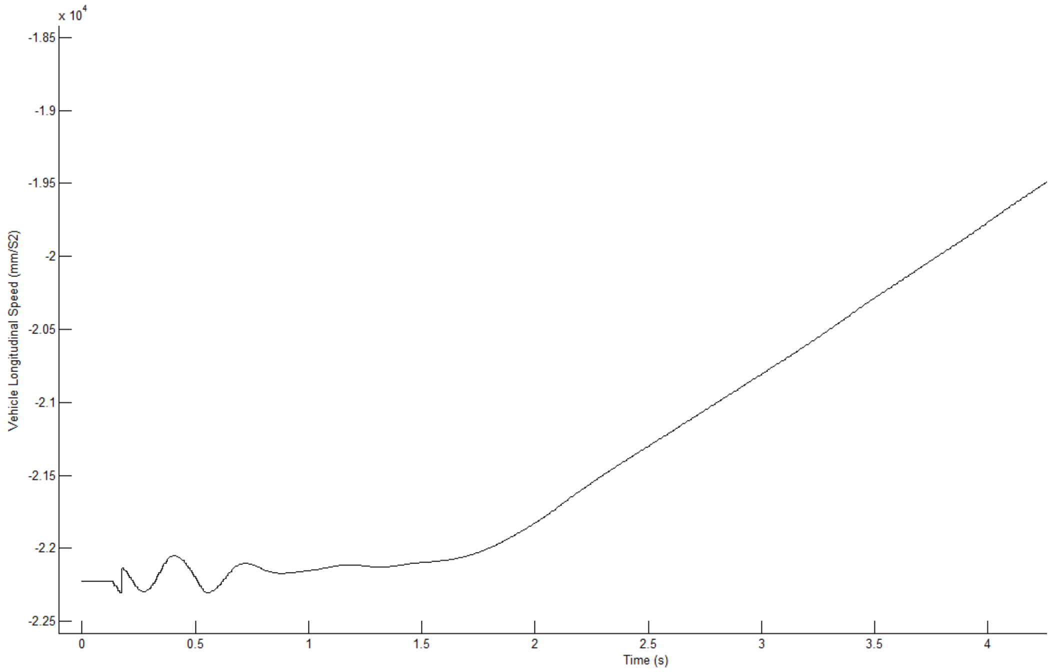


Figure 6.20 Split-mu vehicle longitudinal speed on high friction surface

As seen in Figure 6.15, on a low friction surface, the ABS control tries to maintain the tyre-road braking force at around its peak value, which is only around 700N. As seen in Figure 6.16, on a high friction surface, ABS control tries to keep the tyre-road braking force in a similar range as it would be on a low friction surface, instead of maintaining it at its true peak value (around 3500N). Therefore, the tyre-road braking force is roughly balanced between the left-hand side and the right-hand side, and the vehicle moves in a straight line without any steering wheel input.

As seen in Figure 6.17, on a low friction surface, the wheel angular deceleration moves up and down with the 0 axis as its middle line. This indicates full ABS control action, which decelerates and accelerates to avoid wheel lock. As seen in Figure 6.18, on a high friction surface, the wheel angular deceleration also moves up and down, but it is always in a deceleration state and at a lower level than that on a low friction side. This indicates that ABS control has not sufficiently slowed the wheels down. In the same graph, a lot of noise can been seen, which is caused by some control instability. Therefore, further control calibration is required.

As seen in Figure 6.19 and Figure 6.20, vehicle longitudinal speed slows down at a similar rate on both low and high friction surfaces. This guarantees the synchronised motion on both sides of the vehicle, so the vehicle moves in a straight line, which is one of the main targets of the split-mu control.

6.3.1. Split-mu ABS conclusion

All the above simulation results prove the base line of the effectiveness of the developed split-mu control method. As expected the stopping distance is increased by about 90% compared with braking on a completely dry road, but the split-mu control method allows the vehicle to decelerate without the wheels locking, with optimum tyre-road braking force and with stabilisation of the yaw moment acting on the vehicle. During the simulation only a small vehicle yaw angle presented. The split-mu control is operated by ESP, therefore further control performance improvement on the high friction side should be targeted with full ESP capability considered.

6.4. Overall discussion

6.4.1. Big data and reverse engineering

To develop a practical control method for an ABS vehicle safety system, especially when tyre behaviour and other highly nonlinear parameters are involved, pure mathematical model based control methods can be only used to establish initial control set up guidance. Even very complex mathematical models can only guarantee a certain level of accuracy. In real ABS control method implementation, a rules based control method, or a combination of mathematical models and a rules based control method, are superior.

Other than simulation, it is essential to implement the developed control method on a real vehicle, and carry out related experiment tests. To achieve good control performance, actual performance outputs should be compared with expected performance outputs, bias should be identified, and the established control method should be recalibrated. As well as being used for recalibration purposes, the real vehicle experiment test can even be used to reverse engineer the basic control method. The idea is to use real vehicle experiment tests results to establish an overall behavioural map of the test vehicle. The experiment tests must cover the test vehicle's behaviours under different braking situations as thoroughly as possible. The data of the test results is then analysed, which can enhance knowledge of how the controlled plant is behaving during the expected control actions. The control method can then be redeveloped based on the data from the analysis of the understood behaviours. The more data collected, and the more efficient the data analysis method used, the more efficient and accurate a control method can be developed.

6.4.2. Further improvement

On both high and medium friction road surfaces the high friction surface control logic performs satisfactorily. But during the control input parameters tuning process some compromises must be made to enable a balanced control logic to perform equally well on both high and medium friction surfaces with only one set of control input parameters.

If the control logic can distinguish between high and medium friction surfaces, control input parameters can be tuned individually so even better control performance on the specified friction road surface could be achieved.

The low friction surface detection logic could be extended to distinguish between high and medium friction surfaces. As discussed previously, it is a matter of extracting another set of tyre-road braking force derivative threshold and the corresponding tyre-road braking force value threshold from the data of the experimental test results. It could share the same tyre-road braking force derivative threshold as the low friction surface detection logic, albeit with a different tyre-road braking force value threshold.

7. Conclusions and Future Work

7.1. Conclusions

In this research, a new ABS control method has been developed based on reviews of the commercialised and theoretical ABS control methods, an extensive understanding of the physics of tyre force generation mechanisms, the disk-pad braking force generation mechanism and the interaction between the two in a braking situation. In addition, with the aid of the proposed in-wheel representative tyre forces measurement sensor that uses an ultra-thin and flexible force sensor within the wheel hub, the new ABS control method features the practicality of the commercialised ABS, but with reduced complexity. Therefore, a reduction in the implementation costs is possible. A Simulink control model had been developed based on the developed control method. A Subaru Impreza ADAMS vehicle model has been developed and validated to simulate real braking situations. A co-simulation environment has also been established between the Simulink control model and the ADAMS vehicle model to test the performance efficiency of the developed ABS control method. The overall performance of the developed ABS control method is satisfactory.

Following this research, it can be concluded that, considering current technological and implementation costs, there is no simple and straight forward solution for ABS control method development. All current commercialised ABS control methods need to be subject to analysis and understand the data from related real vehicle experimental tests to adapt their algorithms to different types and models of vehicles in different emergency braking scenarios. The tuning of the control input parameters and calibration are also required to get the adapted control algorithm to perform efficiently in all circumstances. Real vehicle testing and tuning requires man-power, effort, time, related expertise and money, in order to achieve satisfactory ABS control performance. Adding more ABS control related information from different kinds of sensors will help to improve the accuracy and efficiency of ABS control, as well as reduce the cost and complexity of implementation. This research has followed this direction to seek alternatives to the current commercialised ABS control methods, while considering the practical implementation in a real vehicle.

The ABS control method developed in this research is based on an understanding of the fundamentals of an emergency braking scenario and the basic functionality of ABS. The function of ABS is to maintain wheel rotational motion during emergency braking while the vehicle is still moving. The ABS control target is to keep the tyre-road braking force at its peak range while keeping the tyre-road lateral force capability within a sufficient range, so that the overall brake distance could be shortened and steering ability can be maintained. Fortunately, during emergency braking on a medium to high friction coefficient road surface, when tyre-road braking force is maintained around its peak range the tyre-road lateral force generation capability remains within a sufficient range by default. ABS can then just focus on maintaining the tyre-road braking force at its peak range. Fundamental theory based ABS control methods use slip ratio as a control reference, but in reality, it remains difficult to acquire the information that is used to calculate the slip ratio. At the same time, complex tyre experimental tests must be carried out to determine the optimal slip ratio for different braking situations, for example, with regards to braking on different friction coefficient road surfaces, at different initial vehicle speeds, with different tyre vertical load forces, and so on. The slip ratio control method must be able to identify the exact type of braking situation, so it can establish the appropriate optimal slip ratio from the pre-scheduled slip ratio mapping. This presents as another difficulty since a misjudgment could cause overall control failure. Commercialised ABS, which is based on control rules, as well as the majority of academic research oriented ABS, which is based on mathematical models, have been developed to achieve the same control results.

The commercialised ABS control methods from Bosch and Wabco use sophisticated logic that is based on information extracted from a large quantity of data from reverse engineered experimental test results, along with ABS related theories. They use wheel angular acceleration as an observer to ascertain the relationship between disk-pad braking torque and tyre-road braking torque. For example, once disk-pad braking torque is held constant, the trend of tyre-road braking torque could be observed. They also use angular acceleration as a control reference to switch between different control phases. Extra logic which adopts other control input parameters, such as reference wheel peripheral speed and an estimation of its rate of decrease, and an estimation of the reference slip ratio etc. must be used to aid the control phase alternation decision. Their goal is to achieve an ABS control target in a practical way, by using a large amount of real vehicle test data, sophisticatedly calibrated control input parameters, along with complex logics to overcome practical issues encountered during different emergency braking scenarios.

Some of the complex control logic and control references used in the commercialised ABS control methods are not required in the proposed ABS, for example the reference slip ratio, so the overall control method is simpler. This makes tuning and implementation easier. Control decision time is therefore saved which allows the overall control method to deliver faster control decisions, so more efficient control performance is achieved. Also, the need for the related experimental tests which are required to determine and calibrate different control input parameters used in this extra logic is reduced. This leads to an easier and faster implementation process, so cost saving is achieved. Finally, the dependency on wheel angular acceleration as the only direct main control reference, and the inaccuracy and uncertainties introduced by the dynamic changes involved in ABS related components in real ABS working conditions can be overcome by using more directly measured information.

It is difficult for a purely mathematical model based ABS control algorithm to include all the nonlinearities and uncertainties involved in an emergency braking scenario, even when the model is very complex with many extra functionality models added. Even then, if the behavioural interaction between these extra complex functionality models and the main model are not fully understood, the whole model becomes inflexible. It was established that within purely mathematical model based ABS control methods there are always assumptions that some of the important parameters are known, but in reality these parameters are very hard to measure. This is the main reason commercialised ABS control methods are mainly rules based instead of mathematical models based.

Compared to a purely mathematical model based ABS control method, the ABS control method developed in this thesis is not characterized by unrealistic assumptions; second, it does not rely heavily on mathematical models which in turn depend on a large number of highly accurate measurements to be representative, so it is more practical to implement. Once the new ABS control method is tuned and adapted on a vehicle in a real vehicle experiment test based development process, it has the capability and flexibility to cope with changes and uncertainties in different real braking situations.

The ABS control method proposed in this thesis shares the same control philosophy as the basic theory based ABS, which is to maintain optimal wheel slip ratio. In implementation, the new ABS control method uses the peak longitudinal tyre-road braking force as a basic control reference. The longitudinal tyre-road braking force is representable by an in-wheel force sensor measurement, which is used to detect the point at which the local peak tyre-road braking force is reached and to identify its drop in percentage after it has reached its peak point. This information is used to decide when to activate ABS control or the alternate ABS control phase. Learning from commercialised ABS and mathematical model based ABS control methods, the new ABS control method adopted wheel angular acceleration as another control reference. The wheel angular acceleration is used to identify control phases and decide which disk-pad braking torque control action the control method is going to implement in the identified control phase.

The new ABS control method then uses appropriate control actions learnt from theoretical ABS control method and those from the commercialised ABS control methods. These represent the natural and practical control actions and deliver good control performance. Finally, the new ABS control method includes extra control logics that do not require complex input parameters to aid the response to the changes in the braking environment involved in real emergency braking situations. Accordingly, the rules based control action switch method, in combination with the representative tyre-road braking force measurement, and the wheel angular acceleration mathematical model resulting from the tyre-road and disk-pad braking torques makes the control method efficient enough to handle the nonlinearities and uncertainties involved in a real ABS control process.

The proposed in-wheel tyre-road braking force measurement sensor integrates a commercialised ultra-thin and flexible printed circuit force sensor into the space between the outside wall of the wheel bearing and the inside wall of the wheel hub. The individual force sensor can be placed at the front, back, and top of the outside wall of the wheel bearing to measure the tyre acceleration force, braking force, and vertical load force. In the proposed ABS control method, the exact tyre-road braking force measurement is not required, only a representation of the tyre-road braking force is necessary. This means that it does not necessarily have to match the exact value of the measured target signal, but it must represent its trend and rate of change. The representative measurement will increase, decrease, or hold whenever the original signal does the same, and its rate of change must also match that of the original signal. This reduces the performance requirements of the proposed sensor. This is in contrast to the current complex force measurement sensors that use sophisticated wheel bearing deformation mapping to extract the tyre-road braking force information (with an error rate up to 20%).

This approach can also be contrasted to the method of planting metal wires inside the tyre carcass to measure tyre deformation. The measurements are acquired by using the electrical capacitance change between the wires, and the tyre deformation is then used to extract tyre forces. In conclusion, the proposed tyre-road braking force representative in-wheel force measurement sensor is more promising. It would be cheaper to manufacture, easier to implement, and the measurement method is straight forward, so it would generate accurate measurements. To further improve the performance of the measurement of sensor, an automated recalibration logic could be implemented and activated every time the brake pedal is pressed.

A detailed comparison between commercialised ABS and the proposed ABS is shown in table 7.1.

Bosch represented ABS	**Propose new ABS**
8 control phases	3 control phases
Reference slip ratio required	Reference slip ratio not required
Reference wheel peripheral speed and related decrease slope determination required	Reference wheel peripheral speed and related decrease slope determination not required
Detailed tyre-road friction coefficient determination required	Only high and low tyre-road friction coefficient determination required
Specified reference wheel angular acceleration map required	Specified reference wheel angular acceleration map not required
Wheel speed sensor only	Wheel speed sensor and in-wheel force sensor

Table 7.1 Comparison between commercialised ABS and the proposed ABS

In conclusion, this work has delivered a novel approach to ABS control method design. The output is a feasible system which could be developed at a low cost. It is envisaged that this work will benefit specialist and niche vehicle manufacturers where the cost of commercial ABS puts it beyond reach. The practical development and implementation of the proposed ABS is achievable by following the programme described in the following section.

7.2. Future Work

To develop and implement a new ABS control method on a real vehicle is not a one-man job; there is a lot work involved. The current research is intended to establish a frame-work and a simulation environment in which an ABS control method can go through an initial test and secure validation. An efficient control method should take into account all the dynamic properties of the tyre, the physical ABS brake system, and the suspension system. The control method should also have different control logics to cope with different braking situations, such as different road surfaces, brake disk and pad friction coefficient changes due to temperature change, and other unpredictable situations. A large number of experimental tests and an analysis of the results are necessary to tune and fit the control method for it to be ready for implementation on the target vehicle.

The mechanical and hydraulic brake system properties, such as pressure build-up or release delay, have not been taken into account in the simulation model presented here. Further work could include the ABS hydraulic system model in the simulation.

The implementation of the ABS control method on a real vehicle is an important aspect of potential future research. In regard to implementing the developed ABS control method on a real vehicle, the first stage in the progress requires the collection of the target vehicle's basic statistics, including its vehicle mass, mass distribution, tyre dimensions, brake disk and pad dimensions, friction coefficient (including changes related to temperature), brake force distribution ratio and so on. With this information some fundamental calculations can be carried out to establish a basic set-up of the necessary ABS control input parameters for the developed control method. The second stage is to carry out tyre testing on different friction coefficient surfaces with different vertical loads, to collect the related tyre-road braking force data. The collected data is then analyzed to establish some other ABS control related parameters, including the average available tyre-road braking force on high medium and low friction surfaces, as well as the maximum percentage drop in the peak tyre-road braking force that would still maintain a sufficient tyre-road braking force and lateral force (which guarantee basic ABS functionality).

A method could be developed to analyse the data from these tests efficiently. Such a method could also be used to improve the test procedures plan. A systematic method could be developed by combining both the test procedure plan and the test data analysis. Lastly, all the analyzed data from the tests can be used to identify the ABS hardware specification necessary to meet the proposed ABS control method's requirements for disk-pad braking torque and tyre-road braking torque interaction. When choosing ABS hardware it is very important to strike a good balance between the hardware capability, the tyre-road braking torque capability, and the ABS control method operational capability which regulates the disk-pad braking torque to achieve the desired tyre-road braking torque interaction. Once the real vehicle test environment is established, the initially developed ABS control method can be tested in real time. The real action control performance data is then analyzed to recalibrate any offset control input parameters before being added to the control method for retesting until a stable control performance is achieved.

ABS is fundamental to a vehicle safety system. Traction control is the opposite of ABS; it brakes the spinning wheels while the vehicle accelerates; but its control goal is similar to ABS, in that it seeks to maximize longitudinal tyre-road force while maintaining the lateral tyre-road force capacity. ESP is developed based on ABS and traction control but at a higher level. It requires extra sensors, including a steering wheel angle sensor, a yaw rate sensor and a lateral acceleration sensor. ESP has the capability to build up disk-pad braking pressure independent of the brake pedal position, which could be used in an ABS to improve its disk-pad braking torque reapplication efficiency. Other than the above three systems, more advanced vehicle safety systems at different levels have been developed recently, including adaptive load control, roll over mitigation control, adaptive cruise control and so on.

The well-established integration of these vehicle safety systems could allow all the data from the sensors' measurements to be shared through an efficient communication layout between all individual systems. This leads to high accuracy, efficient control performance and less complexity regarding to the implementation of the ABS control method. For example, ABS can use the yaw rate sensor and steering wheel angle sensor to identify a split mu braking situation, as well as to adjust the control action according to the difference between the reference and measured yaw rate. All these require an effective higher level controller that could coordinate different individual systems to work as one to adapt to different emergency situations. The development of such a high-level controller should be an important feature of future research.

References

Sahin, M. and Samim Ünlüsoy, Y. (2010) 'Design and simulation of an ABS for an integrated active safety system for road vehicles', *Int. J. Vehicle Design*, Vol. 52, Nos. 1/2/3/4, 64–81.

Damiano Capra, Nicolò D'Alfio, Andrea Morgando and Alessandro Vigliani, (2009) 'Experimental Test of Vehicle Longitudinal Velocity and Road Friction Estimation for ABS System', SAE paper 2009-01-0428.

Fredrik Gustafsson, Stefan Ahlqvist, Urban Forssell, and Niclas Persson, (2001) 'Sensor fusion for accurate computation of yaw rate and absolute velocity', SAE 2001-01-1064.

Mathieu Gerard, Matteo Corno, Michel Verhaegen, and Edward Holweg, (2010) 'Two-phase Anti-lock Braking System using Force Measurement', *AVEC 10*, Coventry University.

Bulent Ozdalyan, (1999) *The integration of tyre, suspension and ABS models into vehicle braking simulations*. PhD Thesis. Coventry University.

Mathieu Gerard, Antonio Loria, William Pasillas-Lepine, and Michel Verhaegen, (2010) 'Design and experimental validation of a cascaded wheel slip control strategy', *AVEC 10*, Coventry University.

Terry D. Day and Dydney G. Roberts, (2002) 'A simulation model for vehicle braking systems fitted with ABS', SAE paper 2002-01-0559.

Robert Bosch GmbH, (2004) *Bosch Automotive Handbook 6th edition*. Wiley-Blackwell.

Gobbi, M., Guarneri, P., Mastinu, G., and Rocca, G. (2010) 'A smart wheel for improving the active safety of road vehicles', *AVEC 10*, Coventry University.

Jorge Sans Sangorrin, Jan Sparbert, Ulrike Ahlrichs and Wolfgang Branz, (2010) ‘Sensor data fusion for active safety systems’, SAE paper, 2010-01-2332.

Ryosuke Matsuzaki, and Akira Todoroki, (2006) ‘Passive wireless strain monitoring of actual tyre using capacitance–resistance change and multiple spectral features’, *Sensors and Actuators*, A 126 277–286.

Sami Koskinen, (2010) *Sensor data fusion based estimation of tyre/road friction to enhance collision avoidance*. PhD thesis. Tampere University of Technology.

Hossein Mirzaeinejad, and Mehdi Mirzae, (2010) ‘A novel method for non-linear control of wheel slip in anti-lock braking systems’, *Control Engineering Practice 18,* 918–926.

Deng Kun, Li Kaijun, and Xia Qunsheng, (2006) ‘Application of Unscented Kalman Filter for the state estimation of Anti-lock Braking System’, *Vehicular Electronics and Safety*, ICVES 2006, IEEE International Conference.

Shannon L. Miller, Brett Youngberg, Alex Millie, Patrick Schweizer, J. Christian Gerdes, (2001) ‘Calculating longitudinal wheel slip and tyre parameters using GPS velocity’, *Proceedings of the American control conference Arlington,* VA June 25-27.

Tanelli, M., Savaresi, S., and Cantoni, C. (2006) ‘Longitudinal vehicle speed estimation for traction and braking control systems’, *Proceedings of 2006 IEEE International Conference on Control Applications*.

Eichhorn, U., and Roth, J. (1992) ‘Prediction and Monitoring of Tyre/Road Friction’, *Proceeding of XXIV FISITA Congress*, London.

Kiencke, U., and Nielsen, L. (2000) *Automotive Control Systems*. Spring-Verlag.

Van, der, Jagt, P., Pacejka, H.B., and Savkoor, A.R.(1989) ‘Influence of tyre and suspension dynamics on the braking performance of an anti-lock system on uneven roads', *Proceedings of 2nd International EAEC conference on New Developments in Powertrain and Chassis Engineering,* Strasbourg, France.

Blundell, M., and Harty, D. (2004*) The multibody systems approach to vehicle dynamics*. Elsevier.

Pacejka, H. B., Bakker, E. (1993) 'The magic formula tyre model, supplement Vehicle system dynamics', Vol 21:1-28.

Gipser, M. (1999) 'FTire, a New Fast Tire for Ride Comfort Simulations', *International ADAMS User's Conference*, Berlin, Germany

Crolla, D.A., and Whitehead, J.P. (2003) 'Vehicle Dynamics, Control and Suspensions', University of Leeds, Leeds.

Kimbrough, S., and Datla, K. (1996) 'Effective means for implementing wheel slip control without a groundspeed sensor'. *In: Proceedings of the 14th IAVSD Symposium on the Dynamics of Vehicles on Roads and Tracks, in Vehicle System Dynamics*, vol. 25, p. 327–39.

Hee Young Jo, Jae Min Yun, Sang Ho Lee, Jun Hong Park, and Jung Yun Kim, (2010) 'Measurement Method of the Tyre Lateral Force for the Use of Chassis Control System', *AVEC 10*, Coventry University.

Bosch Website
<http://de.Bosch-automotive.com> [Accessed Jan.2012]

Clemson University Vehicular Electronics Laboratory website
<http://www.cvel.clemson.edu/auto/systems/braking.html> [Accessed Jan.2012]

EU vehicle safety regulation
<http://eur-lex.europa.eu/legal-content/EN/TXT/?uri=CELEX:32015R0068> [Accessed Jan.2012]

Ftyre website
<http://cosin.eu> [Accessed Jan.2012]

Hyundai Motor company ABS training document
<http://www.myshared.ru/slide/885201> [Accessed Jan.2012]

Tekscan website force sensor
<www.tekscan.com> [Accessed May.2012]

Transport research lab UK
<http://www.rulesoftheroad.ie/index.html> [Accessed Jan.2012]