Design and Validation of a Brake Pressure Controller for a Hydraulically Actuated Braking System.

Master's Thesis

DC 2016.072

Neeraj Harale 0927267

MSc. Automotive Technology

Supervisors:
Dr. ir. Igo Besselink[1]
Dipl. Ing. Daniel Wegener[2]

Graduation Professor:
Prof. dr. Henk Nijmeijer[1]

[1]Eindhoven University of Technology
Department of Mechanical Engineering
Dynamics and Control group

[2] RWTH Aachen University
Institute for Automotive Engineering

September 27, 2016

Technische Universiteit
Eindhoven
University of Technology

Declaration concerning the TU/e Code of Scientific Conduct for the Master's thesis

I have read the TU/e Code of Scientific Conduct[i].

I hereby declare that my Master's thesis has been carried out in accordance with the rules of the TU/e Code of Scientific Conduct

Date

23-09-2016

Name

Neeraj Harale

ID-number

0927267

Signature

Submit the signed declaration to the student administration of your department.

[i] See: http://www.tue.nl/en/university/about-the-university/integrity/scientific-integrity/
The Netherlands Code of Conduct for Academic Practice of the VSNU can be found here also.
More information about scientific integrity is published on the websites of TU/e and VSNU

January 15 2016

Abstract

This research work focuses on the design and validation of a low level brake pressure controller for a hydraulically actuated braking system, considering the physical constraints imposed by the hydraulic lines and limitations imposed by the valve actuation characteristics. The majority of the current low level brake pressure controllers described in literature have not been implemented and tested, and are hardware specific. Furthermore, the majority of the research in the field of braking control design is focused on the high level algorithms. The commercially used low level brake pressure controllers are not published in technical literature for reasons of confidentiality. Another challenge for developing a low level brake pressure controller is the inherent actuator delay of the solenoid valves in the braking system due to their inertial properties.

Hence, to overcome the aforementioned shortcomings, a threshold based switching low level brake pressure controller has been designed, simulated and tested in real time through hardware in the loop tests on a braking system test bench. Closed loop simulations are performed with two different phase based switching algorithms and their results are analysed to select the algorithm with the best tracking performance. Subsequent improvements are made to the low level brake pressure controller model before performing hardware in the loop pressure tracking tests, with different reference pressure signals. The results of the hardware in the loop tests are analysed, to highlight the tracking performance of the low level brake pressure controller based on the threshold based switching algorithm. It is shown that the test results are in close agreement with the results of the closed loop simulations. Thus, the closed loop performance of the low level brake pressure controller has been validated successfully through hardware in the loop tests.

Contents

Nomenclature

Abbreviations

ABS	Anti-lock Braking System
BBW	Brake by Wire
CAN	Controller Area Network
ECU	Electronic Control Unit
EHB	Electro-Hydraulic Braking System
EMB	Electro-Mechanical Braking System
ESC	Electronic Stability Control
ESP	Electronic Stability Program
HAB	Hydraulically Actuated Braking System
HiL	Hardware in the Loop
HL	*Hinter Links* - Rear Left
HR	*Hinter Rechts* - Rear Right
PRBS	Pseudo Random Binary Signal
VL	*Vorne Links* - Front Left
VR	*Vorne Rechts* - Front Right

Symbols

Symbol	Description	Unit
P_{meas}	Measured pressure	bar
P_{ref}	Reference pressure	bar
P_{bound+}	Upper pressure threshold	bar
P_{bound-}	Lower pressure threshold	bar
e_{rel}	Relative pressure error	bar
e_{mean}	Mean Relative Pressure Error	bar
N	Number of sample data points	-
λ	Longitudinal slip ratio	-
V_x	Longitudinal vehicle velocity	m/s
V_{sx}	Longitudinal slip speed	m/s
r_e	Effective rolling radius of the tyre	m
ω	Angular velocity of the tyre	rad/s
J_w	Rotational inertia of the wheel	kgm^2
M_d	Driving torque provided by the engine	Nm
M_b	Braking torque	Nm
F_x	Longitudinal force acting on the tyre in the contact patch	N
F_z	Vertical force acting on the tyre	N

Chapter 1

Introduction

Advanced driver assistance systems play a crucial role in preventing road accidents by assisting the driver in handling safety critical as well as normal driving manoeuvres. The Antilock Braking System (ABS) is a good example of an advanced driver assistance system and ensures vehicle stability under varying road surface conditions by preventing wheel lock up and minimising the braking distance. Current commercial ABS systems employ complex heuristic rules and there is room for performance enhancements of ABS algorithms. The ABS control architecture consists of two individual controllers embedded in an electronic control unit. The high level ABS controller utilises wheel speed data from wheel speed sensors and wheel slip data from the wheel slip estimators, to provide reference brake pressure signals for each wheel brake cylinder to the low level brake pressure controller. This low level brake pressure controller then controls the valves and the internal pumps of the ABS system to ensure that the desired pressure value is achieved at each wheel brake cylinder. Thus, a robust and accurate low level brake pressure controller plays a key role in improving the overall performance of the braking system. Section 1.1 provides a brief background on the project followed by the problem definition in Section 1.2. The research objective and the steps to be taken in order to achieve the research objective are discussed in Section 1.3 and a brief outline of this report is provided in Section 1.4.

1.1 Project Background

The Institute for Automotive Engineering - Aachen (IKA) is currently developing the research vehicle SpeedE, as an open research and innovation platform for internal research and the automotive industry. The focus of the SpeedE research vehicle is on the innovative front suspension and steer by wire system, which is able to steer each wheel individually whilst attaining steering angles of up to 90°(Fig.1.1). The steering system uses two electric motors in combination with harmonic drive reduction gears integrated in the upper control arms of a double wishbone system. Also, the outer ball joints of the upper control arm are replaced by Cardan joints and the tie rods are eliminated. Further details regarding the kinematics and topology of the front axle of the SpeedE can be found in [1].

State of the art braking systems feature the possibility to adjust the brake pressure at each wheel individually which, besides the control of the longitudinal deceleration of a vehicle, also allows for the control of the vehicle's yaw acceleration. The current braking system of the SpeedE research vehicle comprises of two rudimentary sub-systems. The first sub-system consists of four wheel individual electro-hydraulic actuators. Each actuator consists of an electric motor which moves a piston by a spindle to generate hydraulic pressure. Based on the brake force demand, a simple PID controller is used to calculate the position of the spindle to reach the desired pressure [2]. In case of failure of the power supply, the actuator will not generate any brake pressure. The second sub-system, is a conventional hydraulic braking system without a brake booster, which acts as a fallback system in case of failure of the electro-hydraulic actuators. It is observed that the first subsystem i.e. the electro-hydraulic system, does not demonstrate decent braking performance in terms of maximum wheel brake cylinder pressures and desired stopping distances.

Hence, to ensure improved braking performance under all conditions, IKA is developing a novel partial brake by wire system. This would then replace the electromechanical actuators of the conventional brake by wire system with electrohydraulic brakes. In this layout, each individual wheel brake cylinder is actuated by a conventional Bosch ESP 8.0 Hydraulic Modulator (Section 3.2), which in turn necessitates the design and implementation of a robust low level brake pressure controller for precise control of the brake pressure in each individual wheel brake cylinder.

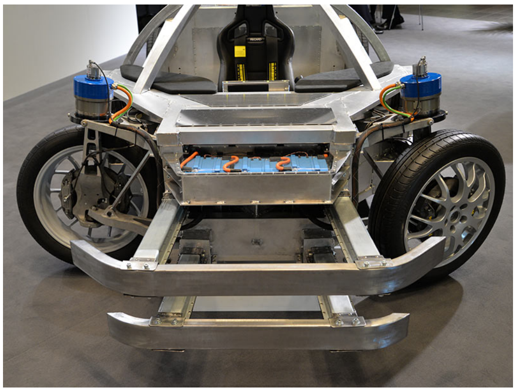


Figure 1.1: Front axle of the SpeedE research vehicle.

1.2 Problem Definition

As already mentioned in Section 1.1, the performance of the existing braking system installed in the SpeedE research vehicle is limited due to the implementation of a rudimentary low level brake pressure controller. This results in insufficient pressure build up at the wheel brake cylinders and longer stopping distances on a variety of road surfaces. Also, modern control algorithms for the high level ABS controller e.g. based on fuzzy logic and neural networks [3], cannot be tested physically due to the absence of a robust low level brake pressure controller. Thus, it is desired to develop a robust low level brake pressure controller, which can process the pressure requests from the high level ABS algorithms with high accuracy, for all road surface conditions and subsequently reduce the stopping distance of the SpeedE research vehicle.

The control strategy used in the low level brake pressure controller of the ESP 8.0 hydraulic modulator, manufactured by Bosch GmbH, is not described in literature for reasons of confidentiality, which adds to the challenge of designing a low level brake pressure controller. Furthermore, the results of tests published in literature are obtained by using different brake hardware setups.

In [4], continuous brake pressure control is obtained by utilising proportional linear servo valves, but such valves are not available in commercial hydraulic modulator hardware and hence continuous control cannot be implemented. Fuzzy model reference learning control, general genetic adaptive control and genetic model reference adaptive control have been implemented in [5] in order to investigate their ability to suppress the effects of process variations but, these methods have significant time delays and real time implementation of these control techniques has not been achieved. Quasi-continuous control by using sliding Pulse Width Modulation (PWM), investigated in [6], is a promising method to eliminate chattering due to the discrete operation of solenoid valves of the hydraulic modulator, but the robustness against varying road surface conditions has not been verified. Therefore, to overcome the limitations of other control approaches and considering the discrete ON/OFF nature of the valves used in the braking system, a switching control approach needs to be developed and evaluated. This low level brake pressure controller should be able to resolve the aforementioned problems in the performance of the braking system of the SpeedE research vehicle.

1.3 Research Objective and Plan

The main research objective of this master thesis is stated as, "*the design and validation of a low level brake pressure controller*". In order to achieve this research objective the stages of development are formulated as:

• Study of the hydraulic braking system to understand the system operation and dynamics.

• Familiarisation with the existing test bench, component level Simulink models and other HiL tools like Vector CANape and dSPACE Targetlink.

• Defining control and performance objectives for the robust low level brake pressure controller by analysing the reference brake pressure signals computed by the high level ABS controller.

• Reviewing technical literature to select a relevant control algorithm, that can be applied to the new low level brake pressure controller structure. The controller synthesis will be done in the MATLAB-Simulink environment.

• Validation of the closed loop performance in the MATLAB-Simulink environment. The closed loop will include the model of the low level brake pressure controller together with the plant model. The plant model, i.e. the hydraulic modulator with other peripheral components of the braking system has already been developed and validated through previous research at IKA [7].

• Hardware evaluation of the closed loop performance by utilising a braking system test bench.

The broader objective of this master thesis can be split into two parts. The first aim is to enable researchers at IKA to implement a partial brake by wire system for the SpeedE research vehicle using the Bosch ESP 8.0 hydraulic modulator. The second aim is to enable researchers at IKA to implement novel high level ABS and ESC algorithms on the test vehicle for field testing, instead of being limited only to algorithm simulations. In order to fulfill these objectives, it is imperative to design a robust low level brake pressure controller, that can precisely control the valves and the internal pumps of the hydraulic modulator to track any arbitrary reference pressure signal.

1.4 Thesis Overview

The findings of a literature review are summarised in Chapter 2. The hierarchical brake pressure controller architecture is explained, followed by a brief review of the state of the art in braking control design. Lastly, the different approaches for the design of a low level brake pressure controller, as adopted in literature, are reviewed.
Chapter 3 describes the characteristics of the braking system under consideration including its components and operating modes of the hydraulic modulator. Further, it introduces the reader to the braking system test bench, which will be used for validation of the designed low level brake pressure controller.
Chapter 4 focuses on the design of the low level brake pressure controller. The closed loop performance objectives are formulated and the physical constraints to be taken into consideration are discussed. The switching control architecture of the low level brake pressure controller, together with its limitations, is explained. The MATLAB-Simulink Model of the closed loop braking system, including the model of the hydraulic modulator and the model of the switching low level brake pressure controller are described. The setup, execution and results of closed loop simulations with two different types of switching brake pressure controllers are discussed in detail.
The real time implementation of the switching low level brake pressure controller and the interface with the braking system test bench, is explained in Chapter 5.
Chapter 6 analyses the experimental data obtained using the new low level brake pressure controller, and describes its validation procedure by comparing experimental data to simulation results.
The final chapter concludes the thesis by stating important conclusions and providing recommendations for future research.

Chapter 2

Literature Review

This chapter provides a concise overview of the research relevant to the field of brake pressure controller design and ABS control algorithms. Section 2.1 describes the hierarchical controller architecture implemented in current commercial ABS systems. The State-of-the-Art in braking control system design is discussed in Section 2.2. Section 2.3 discusses the publications related to brake pressure controller design methods.

2.1 Conventional Brake Pressure Controller Architecture

This section briefly describes the hierarchical structure of a classical brake pressure controller system, which is schematically depicted in Fig.2.1. The control architecture consists of two individual controllers embedded in an electronic control unit. The high level ABS (Anti-lock Braking System) controller utilises wheel speed data from wheel speed sensors, to compute a reference brake pressure signal which acts as an input to the low level brake pressure controller. The low level brake pressure controller then uses the hydraulic actuators to ensure that the desired pressure value is achieved at each wheel brake cylinder. The actual pressure in all four wheel brake cylinders is measured by pressure sensors and fed back to the low level brake pressure controller as an input. The performance of this low level brake pressure controller plays a key role in the overall performance of the braking system.

The control logic used by the high level controller to compute the reference pressure signal is either wheel deceleration based, wheel slip ratio based or a hybrid system combining both approaches. The majority of the ABS control literature deals with these high level controller algorithms [3], whereas the research work pertaining to the low level brake pressure controller is relatively limited. This is partly due to the fact that the design of the low level brake pressure controller is dependent on the hydraulic brake circuit to be controlled i.e. differences in the type of brake fluid or type of valves used in the system, can have an influence on the control methodology used to design the low level brake pressure controller.

2.2 State-of-the-Art - Braking Control Design

The increasing dependency of braking systems on electronics and control systems is depicted in Fig.2.2 [23]. This section describes in brief, the State-of-the-Art in the design of the high level ABS controller. The results available in scientific literature are invariably dependent upon the braking system under consideration. Standard ABS systems with conventional hydraulic actuators utilise rule based control logic due to the inherent limitation of the ON/OFF dynamics of the Hydraulically Actuated Braking (HAB) system. Electro-Hydraulic Braking systems (EHB) and Electro-Mechanical Braking systems (EMB) enable continuous brake pressure modulation and hence active braking control can be formulated as a classical regulation problem [8, 9, 10].The high level ABS controller utilises two output variables for regulation purposes: wheel deceleration and longitudinal slip. Traditionally, wheel deceleration is the controlled output used in ABS owing to its simplicity of measurement through wheel speed encoders. The major drawback of this method

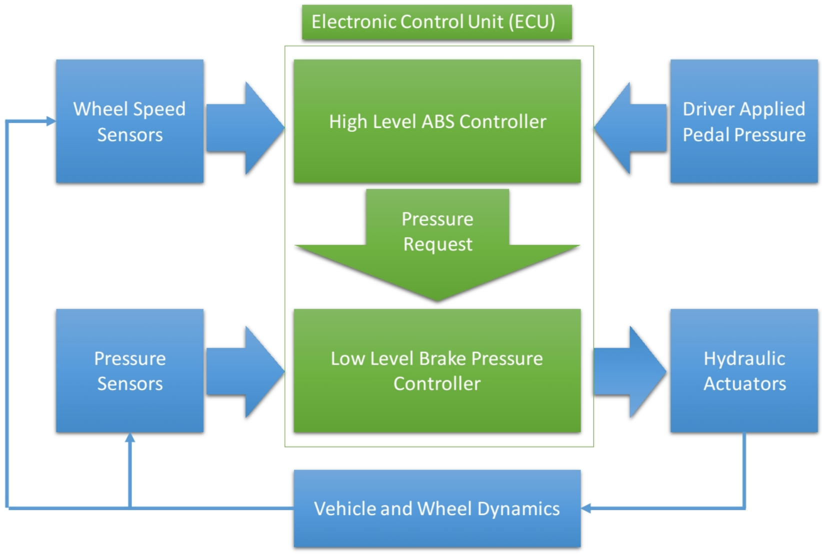


Figure 2.1: Schematic representation of the ABS control architecture.

is the dependency of the the control loop dynamics on the road surface conditions. Therefore, deceleration based control logic requires an online estimation of the road surface conditions [11, 12]. Furthermore, deceleration based control is usually implemented by utilising heuristic threshold based rules rather than using classical regulation loops [11, 12].

On the contrary, brake pressure regulation based on wheel longitudinal slip is easier to implement and is robust. The only drawback of this method is that, the estimation of the longitudinal vehicle speed is a prerequisite. The vehicle speed can directly be measured by expensive equipment for testing purposes only, but in production vehicles it has to be estimated by complex filtering and identification algorithms [13, 14, 15, 16]. The current trend in braking control research demonstrates a shift from threshold based control rules based on wheel deceleration, to longitudinal slip ratio control [17, 18, 19]. The major obstacle for longitudinal slip based brake pressure control is the sensitivity to poor slip estimation or measurement, which is critical at low speed. Another field of active research linked to braking system control is the estimation of tyre-road friction [20, 21, 22].

2.3 Low Level Brake Pressure Controller Design Methodologies

The research pertaining to the low level brake pressure controller is relatively limited, partly because the design of the low level brake pressure controller is dependent on the characteristics of the circuit to be controlled. Furthermore the control strategy of the low level brake pressure controller, designed by the inventor and supplier of the most commonly used ABS system (Bosch GmbH), has not been published for reasons of confidentiality.

Castillo et al. [4] proposed a continuous brake pressure control method employing a new brake system architecture with proportional linear servo valves and also developed an optimal controller for this architecture. The use of proportional servo valves allows continuous and accurate control of the brake pressure resulting in improved deceleration and reduced braking distances for varying road surface conditions. The authors have documented the results of both, simulations and experiments performed using a test vehicle, but the absence of linear proportional servo valves in commercial hydraulic modulator hardware is a major drawback for further implementation of this methodology.

Lennon et al. [5] implemented fuzzy model reference learning control, genetic adaptive control

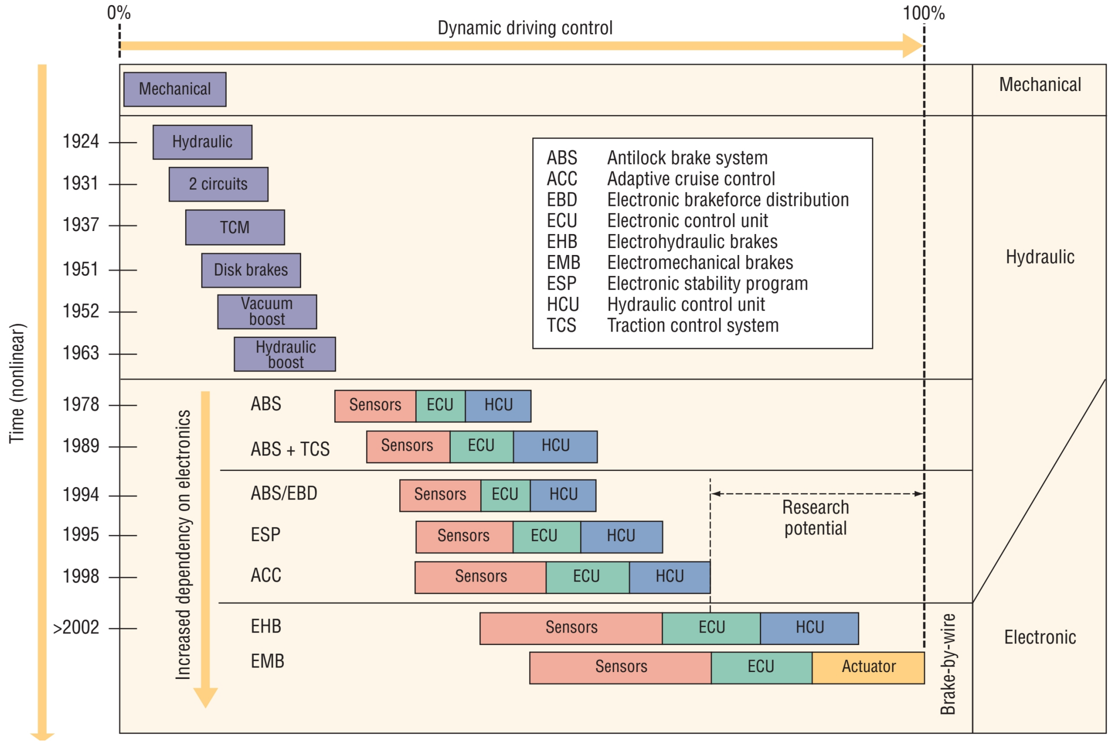


Figure 2.2: Evolution of Braking Systems [23].

and genetic model reference adaptive control for brake pressure control in order to investigate their ability to suppress the effects of process fluctuations and disturbances. In this research work, the input to the system is considered to be the brake torque requested by the driver and the output of the system i.e. the brake torque applied, is directly measured by sensors. Although the aforementioned control methods for brake pressure control demonstrate performance improvements in simulations, their computational complexity is a significant drawback for their real time implementation, which has not been realised so far.

Shah [25] proposed to optimize a standard brake pressure controller by utilising an adaptive control approach to enable the controller to adapt to variable conditions. The Author applied the partial update recursive least squares algorithm to design an adaptive brake pressure controller with varying coefficients. The drawback of this approach is the large variation in the evolution of coefficients during simulations with a highly dynamic reference signal. This may cause even a small error to destabilise the control system due to deviation from the optimal coefficient values.

2.4 Summary

To summarise the limited research work available on the design of a low level brake pressure controller, the following important conclusions can be drawn:

• The low level brake pressure control design approach should be suitable for implementation on the commercially available braking system hardware with discrete ON/OFF solenoid valves.

• The controller architecture should be independent of system variations and at the same time be viable for real time implementation.

• The low level brake pressure controller should track highly dynamic reference signals without loss of accuracy and stability.

Chapter 3

Braking System Characteristics

This chapter describes the important characteristics of the braking system, for which the low level brake pressure controller has to be designed. Section 3.1 provides a brief overview of a conventional ABS system followed in Section 3.2 by a detailed discussion on the components and operating modes of the hydraulic modulator. The braking system test bench, used for the validation of the brake pressure controller, is described in Section 3.3.

3.1 Overview of an ABS system

The goal of an anti-lock braking systems (ABS) is to maintain directional control and lateral stability of the vehicle whilst minimising the braking distances in safety critical manoeuvres. This is achieved by regulating the brake pressure in the wheel brake cylinders. The control strategies used for the brake pressure regulation have already been explained in Chapter 2 and this section provides a brief overview of the ABS system layout and components. Appendix A provides further details regarding the ABS system requirements and braked wheel dynamics.

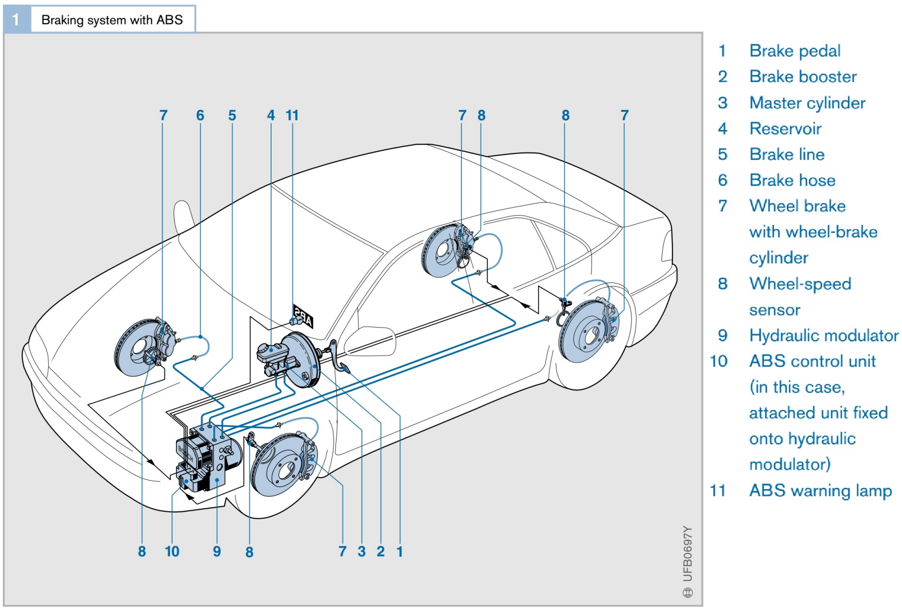


Figure 3.1: Layout of a braking system with ABS [23].

As depicted in Fig.3.1, a conventional ABS system consists of the following components of a conventional braking system:
• The brake pedal and brake booster (Fig.3.1, 1, 2).

• The master cylinder and reservoir (3, 4).
• The brake lines (5), hoses (6), brakes and wheel brake cylinders (7).
Apart from these standard braking components, the ABS system consists of the following additional components:
• Wheel speed sensors (8) - Wheel speed sensors detect the rotational speed of the wheels and these signals are transmitted to the ABS electronic control unit.
• ABS electronic control unit (ECU) (10) - The ABS electronic control unit includes the hierarchical control structure described in Section 2.1. The high level controller processes the wheel speed information and calculates the target reference pressure for each wheel brake cylinder, which is then tracked by the low level brake pressure controller by controlling the ON/OFF solenoid valves and the internal pumps of the hydraulic modulator.
• Hydraulic Modulator (9) - The hydraulic modulator will be explained in detail in the following section.

3.2 Hydraulic Modulator

This section describes the components, operating modes and the dynamics of the hydraulic modulator. The hydraulic modulator controls the hydraulic connection between the master cylinder and the wheel brake cylinders and therefore, is the most crucial component of a modern braking systems. It is controlled by the electronic control unit of the ABS system and utilises twelve solenoid valves and two internal positive displacement pumps to regulate the pressure of the wheel brake cylinders. Hydraulic modulators are classified into two distinct types, one for systems that only modulate driver applied brake pressure (ABS hydraulic modulators) and the other for systems that can build pressure automatically using the inbuilt positive displacement pumps (ESP hydraulic modulators). The hydraulic modulator utilised in this master thesis is an ESP hydraulic modulator and hence, it can build pressure, even in the absence of a driver applied brake pressure.

3.2.1 Components of the Hydraulic Modulator

An exploded view of the hydraulic modulator is depicted in Fig.3.2. The hydraulic modulator for ABS/ESP systems consists of a hydraulic block (Fig 3.2, 5), through which the entire hydraulic circuit, as depicted in Fig.3.3, is drilled. The brake fluid flow through the hydraulic circuit is controlled by actuating the discrete state (ON/OFF) solenoid valves (Fig.3.2, 4), by energising/de-energising the electromagnetic coils (Fig.3.2, 3). The input signals to the electromagnetic coils are provided by the electronic control unit mounted on the hydraulic block, described in Section 3.1. The hydraulic modulator utilises two positive displacement piston pumps (Fig 3.2, 7) to build up pressure in the wheel brake cylinders in the pressure build mode and to return the brake fluid from the wheel brake cylinders to the reservoir in the pressure release mode. The piston pumps are driven by a brushed DC motor, operating with a supply voltage of 13 Volts. The hydraulic modulator also incorporates two low pressure accumulators (Fig 3.2, 8) to ensure that in the pressure reduction mode, the excessive brake fluid is drained from the wheel brake cylinders as fast as possible. Further details regarding the construction and operation of the solenoid valves, pumps, accumulators and the electric motor can be found in [7].

3.2.2 The Hydraulic Circuit

The layout of hydraulic circuit drilled into the hydraulic block is depicted in Fig.3.3. It is divided into two brake circuits connected diagonally. One circuit is connected to the front right and rear left wheel brake cylinders and the other circuit connects to the front left and rear right wheel brake cylinders. The purpose of the solenoid valves provided in the hydraulic circuit is to connect/disconnect the master cylinder from the wheel brake cylinders based on the commands from the electronic control unit, in order to modulate the pressure in the wheel brake cylinders.

The solenoid valves used in this hydraulic circuit are two way ON/OFF type and they provide

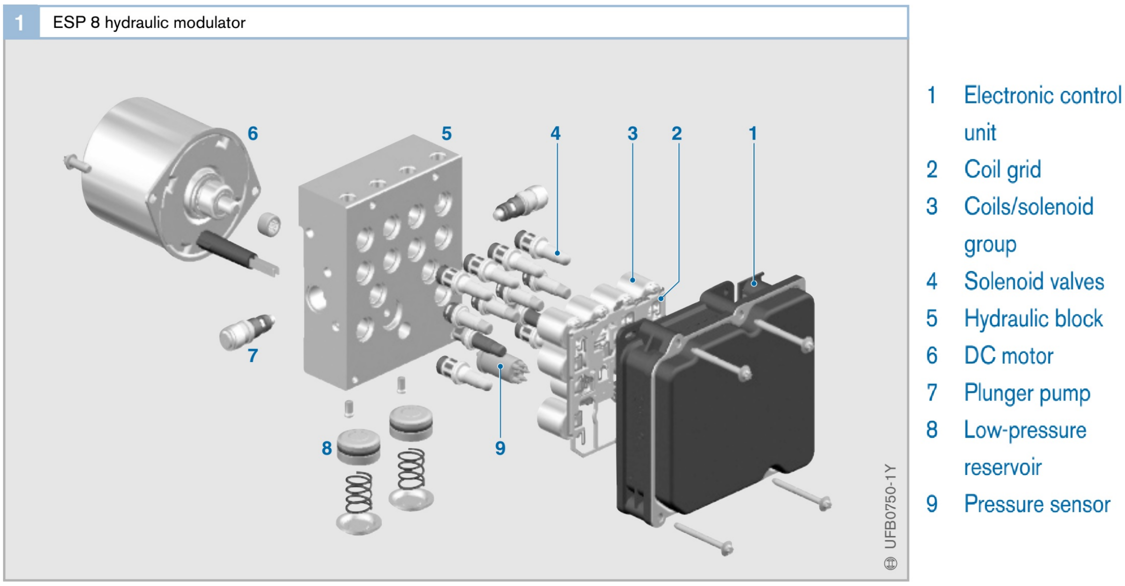


Figure 3.2: Components of the Hydraulic Modulator [23].

a high flow rate gain in spite of their small size, simple structure and low cost. The inherent drawback of these valves is their actuation delay and the resulting non-linear fluid flow behavior, which adversely affects the control accuracy and complicates the task of the brake pressure controller design. Therefore, Pulse Width Modulated (PWM) signals are used to control the valves in order to minimise the actuation delay and obtain linear flow characteristics for short valve cycle times.

The hydraulic circuit consists of twelve valves i.e. four inlet valves, four outlet valves, two switchover valves and two high pressure switching valves. The inlet and switchover valves are of normally open (NO) type to allow the pressure applied on the brake pedal to be transferred to the wheel brake cylinders. On the contrary, the outlet and the high pressure switching valves are of normally closed (NC) type to ensure that the pressure is not released from the wheel brake cylinder for normal braking. The switchover valves are used to disconnect the entire brake circuit from the master cylinder when necessary.

3.2.3 Brake Pressure Modulation Modes

This subsection describes the various modes of brake pressure modulation that the system can be in, during its operation. The term *mode* is used to describe the set of states (ON/OFF) of all the valves of the hydraulic modulator and the state (ON/OFF) of the electric motor which drives the piston pumps inside the hydraulic modulator.

It is necessary to highlight the difference between the operating modes of a standard ABS/ESP system and the operating modes of the braking system under consideration. As explained briefly in Section 1.1, it is desired to implement the braking system as a partial brake-by-wire system (BBW) and therefore, the braking system should be capable of generating brake pressure in the wheel brake cylinders even in the absence of the vacuum booster and the mechanical linkage between the brake pedal and the master cylinder. This will be achieved by utilising the positive displacement piston pumps inside the hydraulic modulator, which are driven by the DC electric motor. The state of each valve type for each mode, i.e. Pressure Hold, Pressure Build and Pressure Release for ABS, ESP and BBW operation, is listed in Fig.3.4.

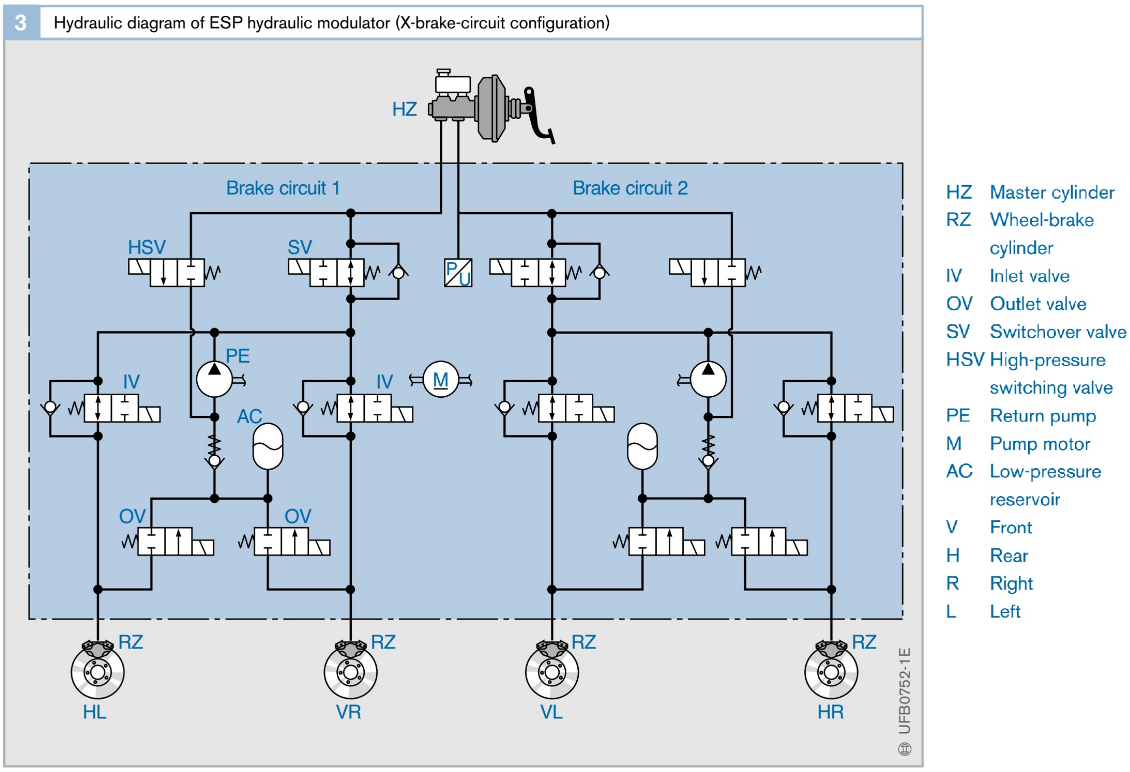


Figure 3.3: Hydraulic circuit diagram of the ESP Hydraulic Modulator [23].

	Mode	Inlet Valve (IV)	Outlet Valve (OV)	High Pressure Switching Valve (HSV)	Switchover Valve (SV)	Pump/ Motor
ABS Operation	Pressure Hold	Closed (Energised)	Closed (De-energised)	Closed (De-energised)	Open (De-energised)	OFF
	Pressure Build	Open (De-energised)	Closed (De-energised)	Closed (De-energised)	Open (De-energised)	OFF
	Pressure Release	Closed (Energised)	Open (Energised)	Closed (De-energised)	Open (De-energised)	ON
ESP Operation	Pressure Build	Open (De-energised)	Closed (De-energised)	Open (Energised)	Closed (Energised)	ON
BBW Operation	Pressure Hold	Closed (Energised)	Closed (De-energised)	Open (Energised)	Closed (Energised)	OFF
	Pressure Build	Open (De-energised)	Closed (De-energised)	Open (Energised)	Closed (Energised)	ON
	Pressure Release	Closed (Energised)	Open (Energised)	Closed (De-energised)	Open (De-energised)	ON

Figure 3.4: Brake pressure modulation modes of the hydraulic modulator.

The low level brake pressure controller is to be designed for the BBW operation and hence the functioning of the hydraulic modulator in this mode will be explained in further detail. The pressure hold, pressure build and pressure release modes for BBW operation are described below.

Pressure Hold Mode

The state of the hydraulic circuit in the pressure hold mode is depicted in Fig.3.5. The objective in this particular mode is to maintain the pressure in the wheel brake cylinder at a constant value. In order to achieve this, both the inlet and outlet valves are held closed i.e. the inlet valve solenoid is energised, as it is of normally open type and the outlet valve solenoid is de-energised, as it is of the normally closed type. The electric signals for energising and de-energising the solenoids are provided by the low level brake pressure controller embedded in the electronic control unit mounted on the hydraulic modulator. The high pressure switching valve and switchover valve solenoids are also energised resulting in them being in the open and closed state respectively, hence cutting off the supply of brake fluid from the brake fluid reservoir. The DC electric motor, indirectly the piston pumps, are in the OFF state and subsequently, the brake pressure is neither increased nor decreased. Therefore the combined effect of the valve and pump states in this mode leads to the wheel brake cylinder pressure being maintained at a constant value.

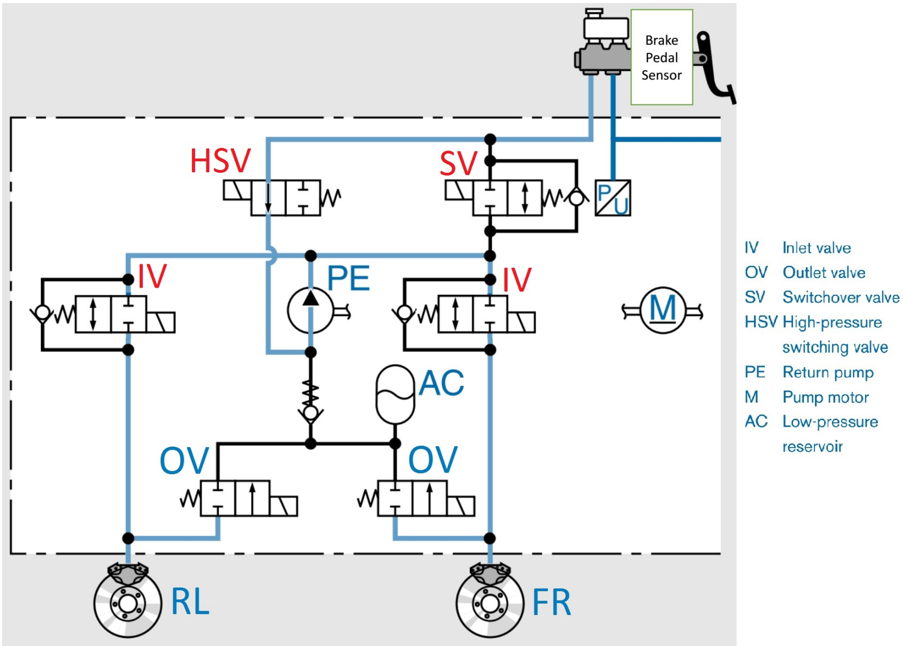


Figure 3.5: The hydraulic circuit in pressure hold mode.

Pressure Build Mode

The state of the hydraulic circuit and the path of brake fluid flow in the pressure build mode is depicted in Fig.3.6. The objective in this particular mode is to increase the pressure in the wheel brake cylinder, with minimum delay, to achieve the reference pressure that is computed by the high level brake pressure controller. The inlet valve is held in the open state by de-energising the inlet valve solenoid and the outlet valve solenoid is energised to keep it in the closed state, in order to avoid any decrease in wheel brake cylinder pressure. The high pressure switching valve and switchover valve solenoids are energised resulting in them being in the open and closed state respectively. The DC electric motor is switched ON, in order to drive the piston pumps and increase the pressure in the wheel brake cylinder. Subsequently, the brake fluid flows from the brake fluid reservoir through the high pressure switching valve into the suction side of the piston pump. It is then discharged by the pump, at high pressure, through the open inlet valve, to the wheel brake cylinder resulting in a subsequent rise in brake pressure. The wheel brake cylinder pressure is measured by pressure sensors and is transmitted to the low level brake pressure controller through a feedback loop, which then regulates the wheel brake cylinder pressure to the reference pressure value, which is requested by the high level controller. The regulation logic of the low level brake pressure controller will be explained in detail in Chapter 4.

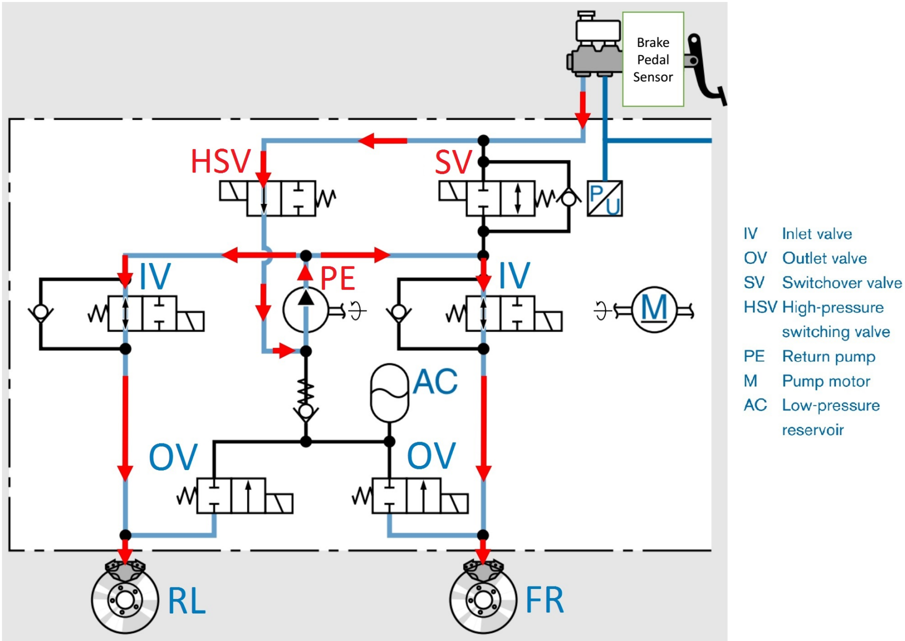


Figure 3.6: The hydraulic circuit in pressure build mode.

Pressure Release Mode

Fig.3.7 depicts the state of the hydraulic circuit and the path of brake fluid flow in the pressure release mode. The objective in this particular mode is to decrease the pressure in the wheel brake cylinder to the requested reference pressure, that is computed by the high level brake pressure controller. The inlet valve is held in the closed state by energising the inlet valve solenoid and the outlet valve solenoid is de-energised to hold it in the open state, to allow the fluid to flow from the wheel brake cylinder to the low pressure accumulator. The accumulator enables rapid release of the pressure in the wheel brake cylinder. The high pressure switching valve and switchover valve solenoids are de-energised resulting in them being in their default states i.e. closed and open state respectively. The DC electric motor is in the ON state in order to drive the piston pumps and decrease the pressure in the wheel brake cylinder. The positive displacement piston pump sucks the fluid from the wheel brake cylinder through the open outlet valve and it is discharged back to the brake fluid reservoir through the switchover valve. This entire process of pressure release has to be executed in within a short cycle time in order to track the reference pressure accurately and to avoid wheel lock-up for severe braking.

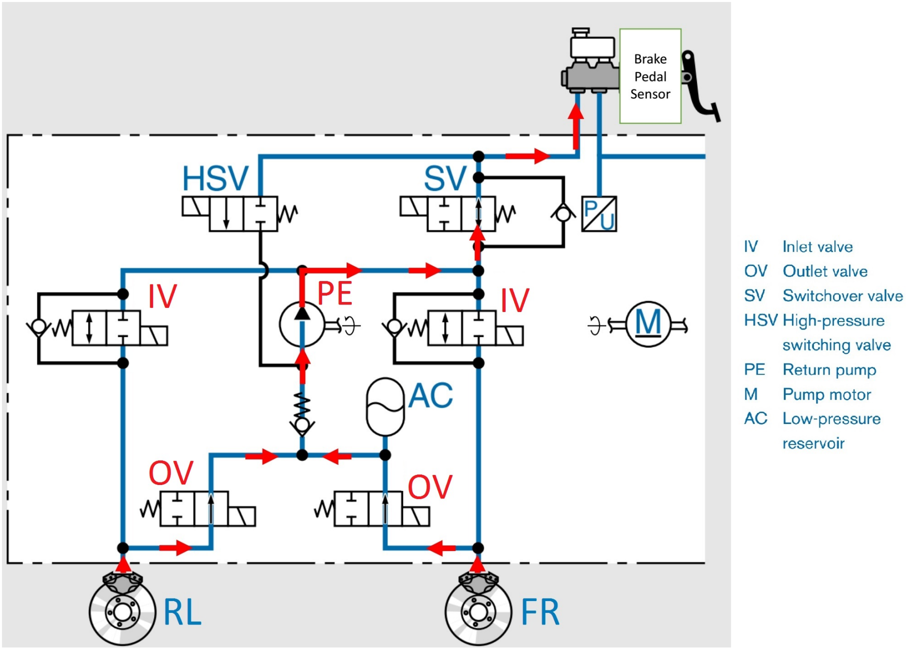


Figure 3.7: The hydraulic circuit in pressure release mode.

3.3 Braking System Test Bench

This section provides a brief introduction to the braking system test bench. This hardware is used for the validation of the self designed low level brake pressure controller. This test bench represents the entire braking system of a passenger car and is shown in Fig.3.8. It comprises of the hydraulic modulator from a Bosch ESP 8.0 system, together with the peripheral hardware components of the braking system. This test bench makes the hardware implementation of the designed low level brake pressure controller feasible. It also allows the validation of the closed loop simulation results of the low level brake pressure controller performed in the MATLAB-Simulink environment. The components of the test bench are briefly described below, for generic functional details of each component, reference is made to [7] :

• **1. Brake Pedal -** The brake pedal installed on the test bench has a mechanical ratio of 4:1, hence it amplifies the driver input force by a factor of 4 mechanically.

• **2. Brake Booster -** The pedal force is then transmitted and amplified further by a brake booster manufactured by Girling (Model no. LSC 115). It features a double diaphragm design that, provides a boost factor of 4.5.

• **3. Vacuum Pump -** The vacuum pump is required to create a pressure difference between the chambers of the brake booster. Due to the absence of an internal combustion engine in the setup, this vacuum pump is driven by a separate 12V DC electric motor.

• **4. Brake Fluid Reservoir -** It is used to store the brake fluid and is connected directly to the master cylinder. The fluid in the reservoir is maintained at atmospheric pressure by isolating it from the rest of the braking circuit under normal braking.

• **5. Master Cylinder -** The Master cylinder installed on the test bench has a tandem cylinder design, with two equal sized pistons, each with a diameter of 25.4 *mm*.

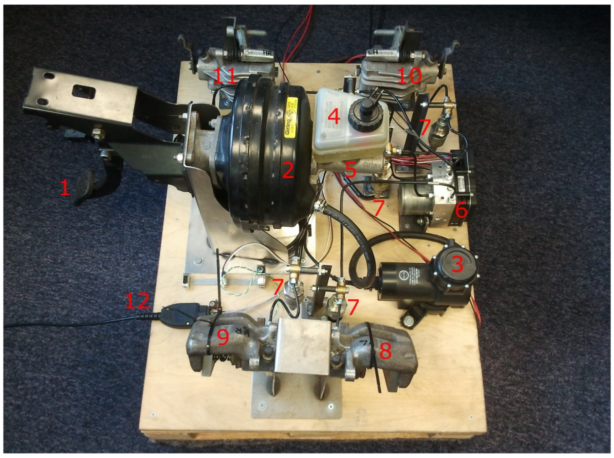
• **6. ESP 8.0 (Hydraulic modulator developed by Bosch) -** Described in detail in Section 3.2.

• **7. Pressure Sensors -** Six pressure sensors are installed on the test bench. Two of these are located in the brake lines between the master cylinder and the hydraulic modulator, to monitor the pressure at the outlets of the master cylinder. The other four are used to monitor the pressure at each wheel brake. These sensors are manufactured by Bosch-Rexroth and measure gauge pressure in the range of 0 bar to 210 bar.

• **8, 9, 10, 11. Brake Calipers -** The test bench is equipped with four single piston calipers which represent the disc brakes of a vehicle. Despite the differences in their mechanical design, all four calipers feature a similar wheel brake cylinder with a diameter of 38mm. To minimise the space requirements of the test workbench, disc rotors and brake pads are not installed within the setup. To ensure a pressure rise in the system, compact blocks of steel with adjustable spacers are inserted inside the calipers. These act as the disc rotor-brake pad interface and provide a stiff resistance to the translational motion of the piston. Details regarding the caliper assembly can be found in [7].

• **12. Controller Cable -** This is used to establish a connection with the electronic control unit of the ESP 8.0 system via the CAN (Controller Area Network) protocol.

• **Vector CANcaseXL -** This part is not depicted in Fig.3.8. This is an USB interface used for the exchange of data between the ECU of the ESP 8.0 system and the computer used for controller design and experimental data analysis. Its technical specifications are provided in Appendix B.



1. Brake pedal
2. Brake booster
3. Vacuum pump
4. Reservoir
5. Master cylinder
6. ESP 8.0
7. Pressure sensors
8. FL brake caliper
9. FR brake caliper
10. RL brake caliper
11. RR brake caliper
12. Controller cable

Figure 3.8: Braking system test bench representing the braking system of a passenger car [7].

3.4 Summary

In this chapter the characteristics of the braking system under consideration were described. A brief overview of a conventional ABS system was provided in Section 3.1. In Section 3.2, the hydraulic circuit of the hydraulic circuit was described and operating modes of the hydraulic modulator were explained. The state of each valve, the internal DC motor and the path of the brake fluid flow in each operating mode was depicted. In Section 3.3, the braking system test bench used for the validation of the low level brake pressure controller was described.

Chapter 4

Brake Pressure Controller Design

This chapter describes the closed loop performance objectives of the braking control system, the physical constraints to be considered and the control architecture employed in the design of the low level brake pressure controller. It also includes the closed loop simulations performed in the MATLAB Simulink environment with staircase type reference pressure signals. In this master thesis, the term "closed loop", implies the feedback loop comprising of the hydraulic modulator i.e. the plant to be controlled and the low level brake pressure controller. The purpose of the aforementioned simulations is to analyse the stability and the performance of the self designed low level brake pressure controller.
Section 4.1 describes the performance objectives of the closed loop system comprising of the hydraulic modulator and the low level brake pressure controller. The physical system constraints, which indirectly act as the limitations to the closed loop system bandwidth and play a crucial role in determining the control approach to be adopted, will be discussed in Section 4.2. Section 4.3 briefly explains the phased control approach implemented in the design of the low level brake pressure controller. Section 4.4 describes the MATLAB Simulink models of the hydraulic modulator and the low level brake pressure controller used for the closed loop simulations. The results of the closed loop simulations in the MATLAB Simulink environment, for two different types of switching controllers are discussed in Section 4.5. The improvements made to the low level brake pressure controller based on the analysis of the closed loop simulation results are described in Section 4.6.

4.1 Closed Loop Performance Objectives

The design objectives of the low level brake pressure controller are as follows:

• The reference pressure should be tracked with minimum settling time and delay.

• The reference pressure should be tracked with high accuracy without chattering (pressure oscillations), in the vicinity of the reference pressure value.

• The reference pressure signal used for the validation of the closed loop performance should resemble a typical signal from a high level ABS controller with both pressure build and pressure release steps.

The designed low level brake pressure controller should be ready for implementation in a brake-by-wire system and hence, it should be able to control the internal pumps of the hydraulic modulator to build-up pressure and empty the accumulator when required.

4.2 Physical System Constraints

The physical system of the hydraulic modulator together with the hydraulic lines and brake fluid have certain inherent limitations and constraints, which have to be considered during the synthesis of a low level brake pressure controller. These limitations are as follows:

• **Discrete ON/OFF valve control -** With the current hardware, proportional control of the valves using PWM (Pulse Width Modulation) is not possible and hence, only discrete ON/OFF control can be implemented, which results in the valves being either completely closed or completely open. This limitation rules out the possibility of continuous control of the hydraulic modulator.

• **Valve actuation delay -** The mechanical inertia of the valves combined with the lag in the feedback of the pressure signal from the pressure sensor, results in an unavoidable time delay during switching of the valve states from a 100% open to 100% closed state, and vice versa. Due to the difference in inertia between the inlet and outlet valves, a significant difference is observed between their closing times when de-energizing the solenoid. Their opening times however are approximately the same. Although, the delays cannot be measured directly through experiments on the test bench, they can be estimated from the validated simulation model of the hydraulic modulator from [7]. It is observed that, the closing time of the outlet valve is much greater as compared to its opening time, which has an adverse effect on the steady state pressure after the pressure release phase is executed. These actuation delay periods of the inlet and outlet valves adversely affect the bandwidth of the closed loop system as the control action has to be delayed to allow the valves to open/close completely.

• **Brake fluid inertia and hydraulic losses -** The inertia of the brake fluid and hydraulic losses in the brake lines cause further inevitable delays in the system response and limit the closed loop bandwidth of the braking system.

• **Limited pressure build-up rate -** The positive displacement pumps in the hydraulic modulator are only capable of pressure build-up rates of up to 300 bar/s. This implies that for a reference pressure step of approximately 100 bar, the internal pump will take a minimum of $0.3s$ in order to generate the demanded pressure from an initial state of 0 bar. This adds a further minimum delay to the response of the closed loop system in case of a pressure build-up phase and this delay can neither be eliminated nor reduced.

4.3 Control Architecture

As described in the system constraints, continuous control of the hydraulic modulator is impossible due to the discrete ON/OFF nature of the solenoid valves and therefore, switching control will be implemented. Through ABS literature [24], it is observed that the requests from the high level ABS controller can be of three different forms, i.e. to hold brake pressure at a constant value, to build up brake pressure and to release brake pressure. Based on this prior knowledge, the low level brake pressure controller is designed to be able to switch between three different modes, i.e. pressure hold mode, pressure build mode and pressure release mode, as shown in Fig.4.1. A switching scheme based on these three modes will be utilised to control the pressure in each individual wheel brake cylinder in accordance with the objectives mentioned in Section 4.1. Two such switching schemes are formulated and analysed in this section.

4.3.1 Absolute reference value based switching algorithm

Fig 4.2 depicts the switching logic of the absolute reference value based switching algorithm. In this algorithm, switching between the three modes, i.e. pressure hold, pressure build and pressure release mode, is based on the absolute value of the requested reference pressure. The values of the reference pressure and the measured wheel brake cylinder pressure are updated during each cycle and depending on these values, the respective mode is initiated. Each mode corresponds to predefined solenoid valve positions and pump state (ON/OFF). The details of individual valve states in each of the three modes have been described earlier in Section 3.2.3. The inherent advantage of this switching algorithm is its computational simplicity which is vital considering its real time application as the low level brake pressure controller.

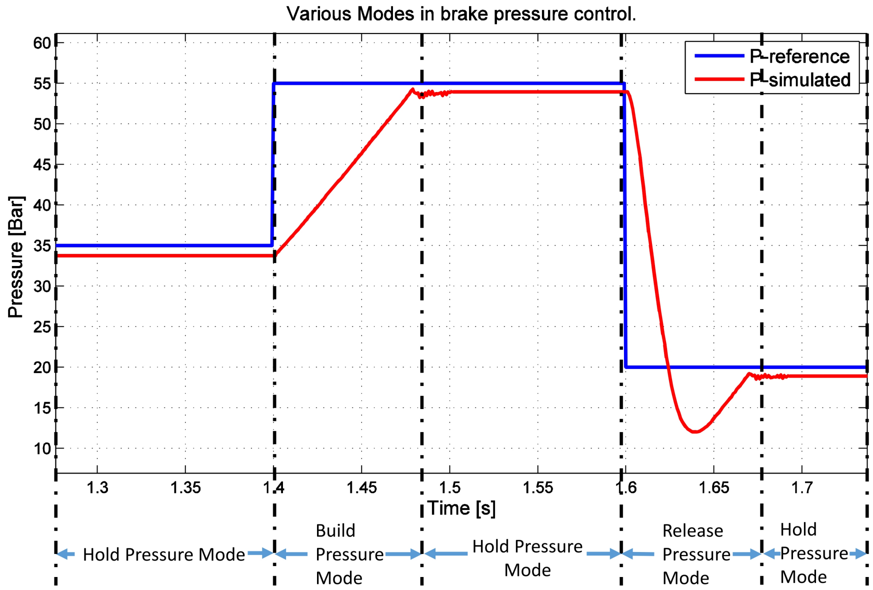


Figure 4.1: Modes in brake pressure controller design.

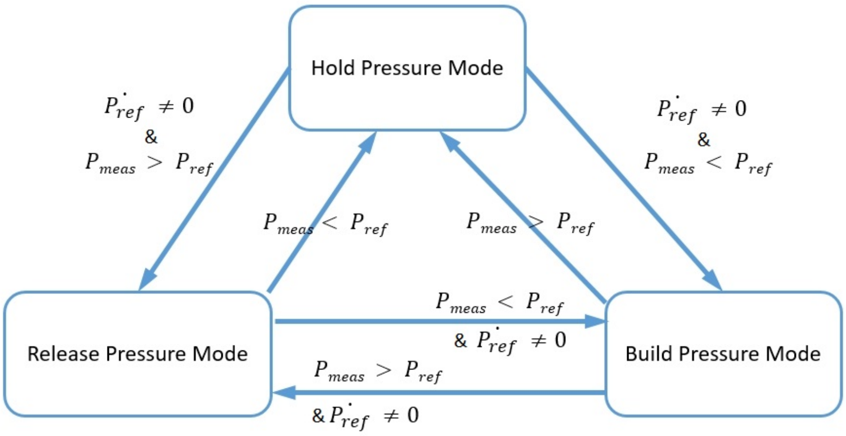


Figure 4.2: Absolute reference value based switching controller.

4.3.2 Threshold based switching algorithm

The threshold based switching algorithm utilises a reference pressure band created by offsetting the absolute reference pressure values to regulate the wheel brake cylinder pressure. This reference pressure band is bounded by a upper threshold P_{bound+} and a lower threshold P_{bound-} which are computed as follows:

$$P_{bound+} = P_{reference} + P_{offset}$$
$$P_{bound-} = P_{reference} - P_{offset}$$

The value of P_{offset}, which is a controller parameter, is to be chosen such that it does not adversely affect the tracking accuracy of the controller. At the same time it should provide a broad tolerance band to ensure that the system can reach a steady state when the reference pressure is not varying. The motivation behind adopting a threshold based algorithm is to avoid the pressure oscillations or chattering by providing a region of tolerance about the reference pressure value. Its switching logic is depicted in Fig.4.3.

The objective of the brake pressure controller is to regulate the pressure within these bounds

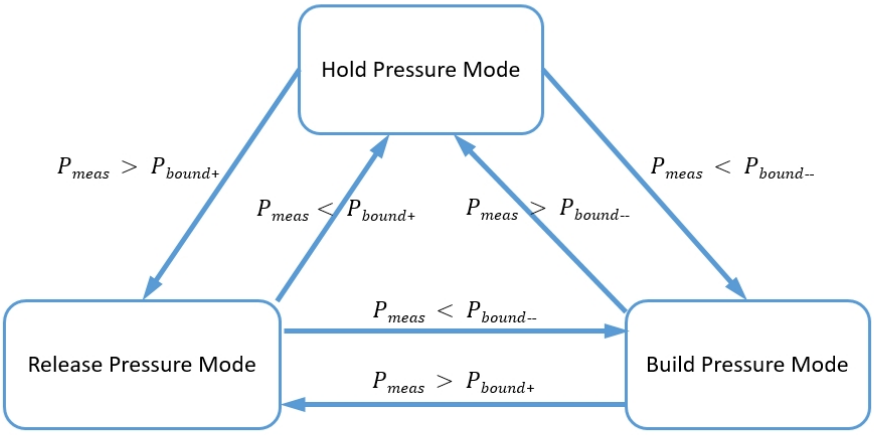


Figure 4.3: Threshold based switching controller.

to obtain optimum tracking performance with minimum pressure oscillations around the set reference value and minimum settling time. This threshold based switching logic can be summarised as:

- If $P_{measured}$ is within the defined thresholds - then Hold Pressure Mode.
- If $P_{measured}$ is below the lower threshold P_{bound-} - then Build Pressure Mode.
- If $P_{measured}$ is above the upper threshold P_{bound+} - then Release Pressure Mode.

Fig.4.4 depicts the tolerance band computed about the requested reference pressure trajectory with a P_{offset} value of 2 bar.

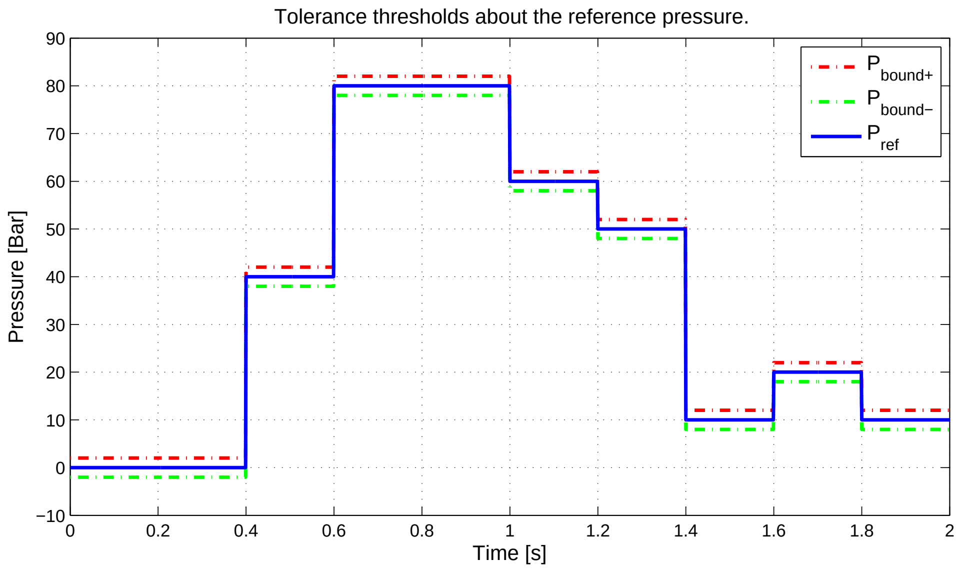


Figure 4.4: Tolerance thresholds about the reference pressure.

4.4 Model of the Closed Loop System

The MATLAB Simulink model of the closed loop system, comprising of the model of the hydraulic modulator and the model of the low level brake pressure controller, is depicted in Fig.4.6. For reduced simulation time and simplicity, only a single wheel brake cylinder is controlled in the closed loop simulations whereas, the other wheel brake cylinders are maintained in their default static state by closing the corresponding inlet and outlet valves. The simulation models of the hydraulic modulator and the low level brake pressure controller are treated individually in Section 4.4.1 and 4.4.2 respectively. A schematic representation of the closed loop system is provided in Fig.4.5. The model of the hydraulic modulator computes the pressure in each wheel brake cylinder, depending upon the inputs i.e. the state of each valve (ON/OFF) and the state of the DC motor. This wheel brake cylinder pressure is fed back as input to the low level brake pressure controller through a feedback loop, which then compares the simulated pressure signal to the set reference pressure signal and regulates the valves and the internal pumps of the hydraulic modulator model, in order to ensure accurate reference tracking.

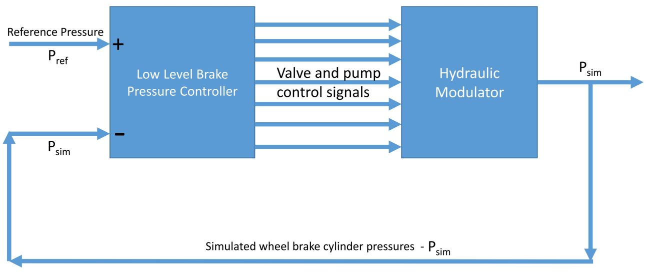


Figure 4.5: Schematic representation of the Closed Loop system.

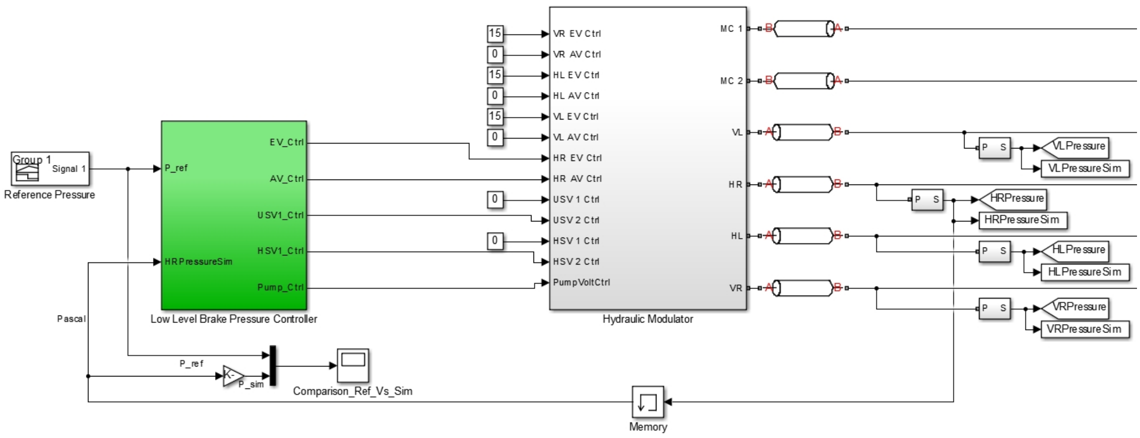


Figure 4.6: Model of the Closed Loop system.

4.4.1 Model of the Hydraulic Modulator

The MATLAB Simulink model of the hydraulic modulator, depicted in Fig.4.7, has been developed and experimentally validated by IKA [7]. It it utilised in this master thesis with minor modifications as the plant model in the closed feedback loop, which also includes the model of the low level brake pressure controller. It is essentially a subsystem with four SimHydraulics outputs, thirteen control inputs and two SimHydraulics input connections. The four output signals of the model are the wheel brake pressure signals and the two SimHydraulics input connections correspond to the master cylinder connections. The thirteen input signals correspond to the twelve solenoid valves in the hydraulic modulator block together with the control signal for the DC motor, which drives the two internal positive displacement pumps. The details regarding the physical modelling of each component of the hydraulic modulator, depicted in Fig.4.7, can be found in [7]. Fig.4.7 depicts the MATLAB Simulink model of the hydraulic modulator in greater detail, which is essentially a physical model of the hydraulic brake circuit depicted in Fig.3.3. Fig.4.8 depicts an example of the pressure response and pressure error from a model validation test performed in [7].

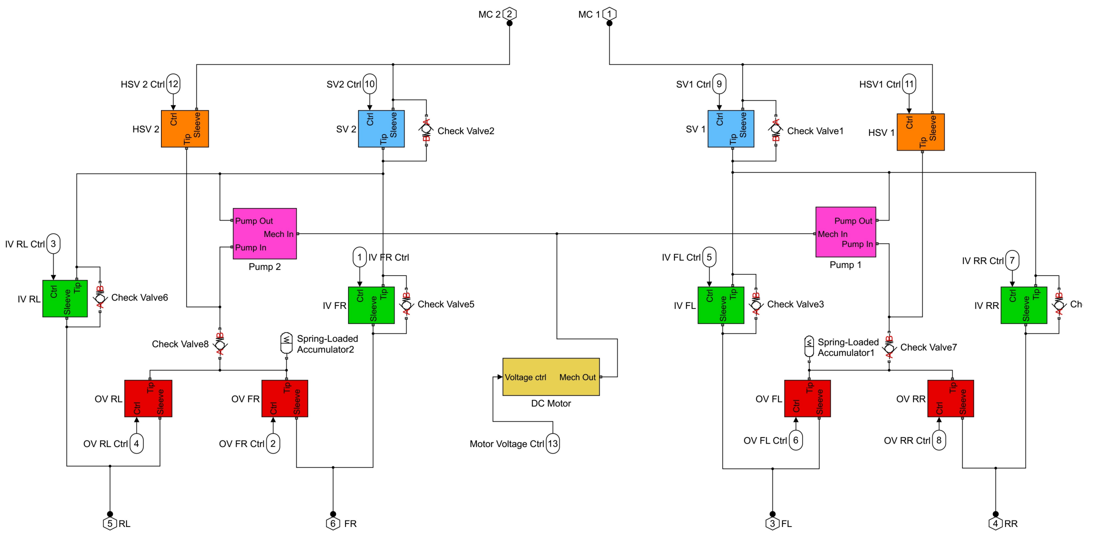


Figure 4.7: Model of the hydraulic modulator [7].

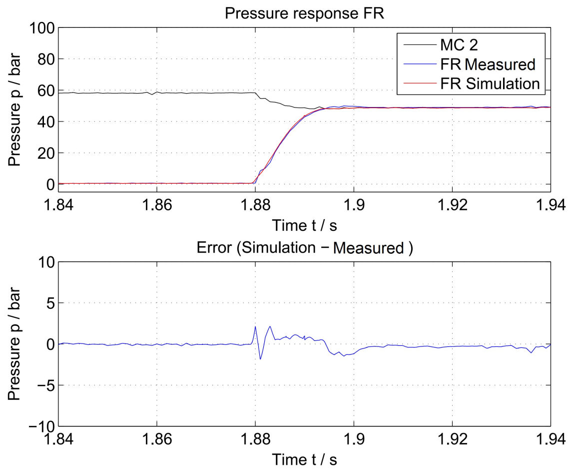


Figure 4.8: Example of a model validation test for the hydraulic modulator - pressure error [7].

4.4.2 Model of the Low Level Brake Pressure Controller

The model of the low level brake pressure controller for a single wheel brake cylinder, implemented as a Simulink subsystem, is depicted in Fig.4.9. The input signals to the model are the reference pressure signal and the simulated wheel brake cylinder pressure signal. The model outputs are the control signals for the solenoid of each of the four valves to be controlled for brake pressure modulation in the wheel brake cylinder, i.e. the inlet valve, outlet valve, high pressure switching valve and the switchover valve, together with the control signal for the DC motor in order to operate the internal pumps when required.

The control logic of the low level brake pressure controller is based on the threshold based switching algorithm described in Section 4.3.2. The simulated wheel brake cylinder pressure, which is the output of the hydraulic modulator model, is supplied as input to the brake pressure controller through a feedback loop with static scalar gain to apply a unit conversion from Pascal to bar. The reference pressure signal, which acts as an input to the brake pressure controller model, can be defined arbitrarily. This allows the brake pressure controller model to be connected to any high level ABS controller models or brake-by-wire models, which generate the reference brake pressure or demanded brake pressure.

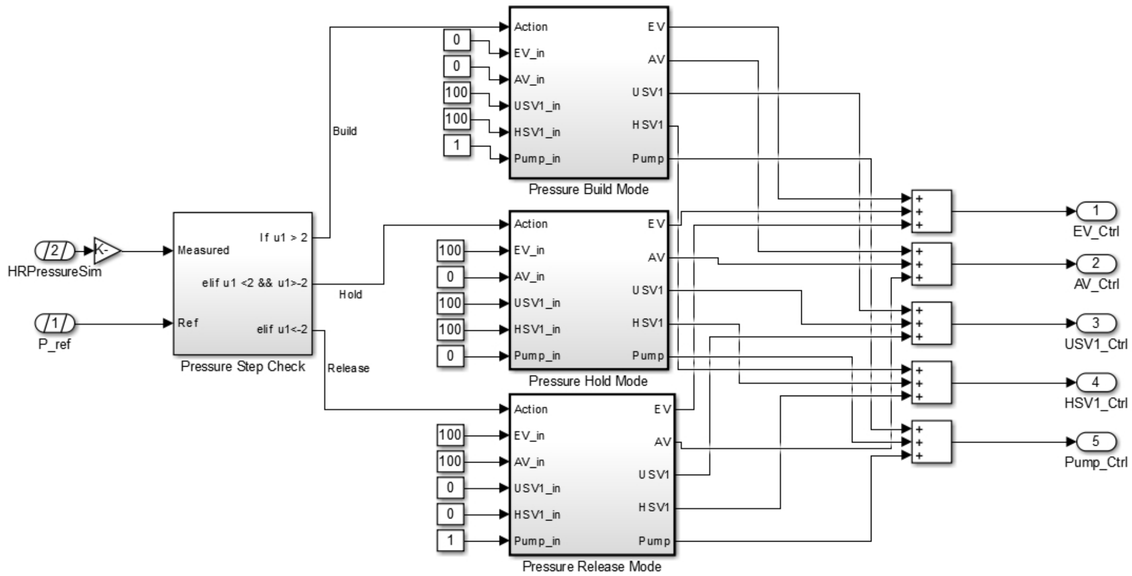


Figure 4.9: Model of the Low Level Brake Pressure Controller.

4.5 Closed Loop Simulation Results

In this section, the results from the closed loop simulations performed with the model described in Section 4.4 are analysed. The closed loop performance is studied while considering overshoot, oscillations about the reference value and the settling time. The absolute value based and the threshold based switching algorithms, discussed in Section 4.3, are both implemented in the MATLAB Simulink model of the low level brake pressure controller, and the closed loop performance of both algorithms is analysed in detail.

4.5.1 Absolute Reference Value Based Switching Algorithm

Fig.4.10 and 4.11 depict the tracking performance of the closed loop with the absolute reference value based switching controller. It is observed that the tracking performance of the absolute reference value based controller is acceptable considering its computational simplicity when compared with the controllers designed in [4] and [5]. It should be noted that the time scale of the depicted graph is from 0-2 seconds and hence, although the settling time appears to be too large, it is approximately 270 ms for a pressure build step of 80 bar. This settling time is limited by the relatively slow dynamics of the internal pump in the hydraulic modulator, which is only capable of pressure rise rates up to a maximum of 300 bar/s.

The major problem with the absolute reference value controller, as depicted in Fig.4.10, is the consistent chattering (pressure oscillations) about the reference pressure. These consistent pressure oscillations are undesirable as they do not allow the system to remain in a steady state, even when the reference pressure is constant and hence, the absolute reference value controller will not be implemented further. Furthermore, the consistent chattering will also reduce the life expectancy of the solenoid valves.

It is observed in the simulation results that, while switching from a pressure release mode to a pressure hold mode, the wheel brake cylinder pressure drops below the reference pressure for a considerable time duration in spite of the controller action. This is highlighted in green in Fig.4.10. This is attributed to the relatively large closing time of the outlet valve due to its inertia and is an inherent physical limitation of the braking system hardware.

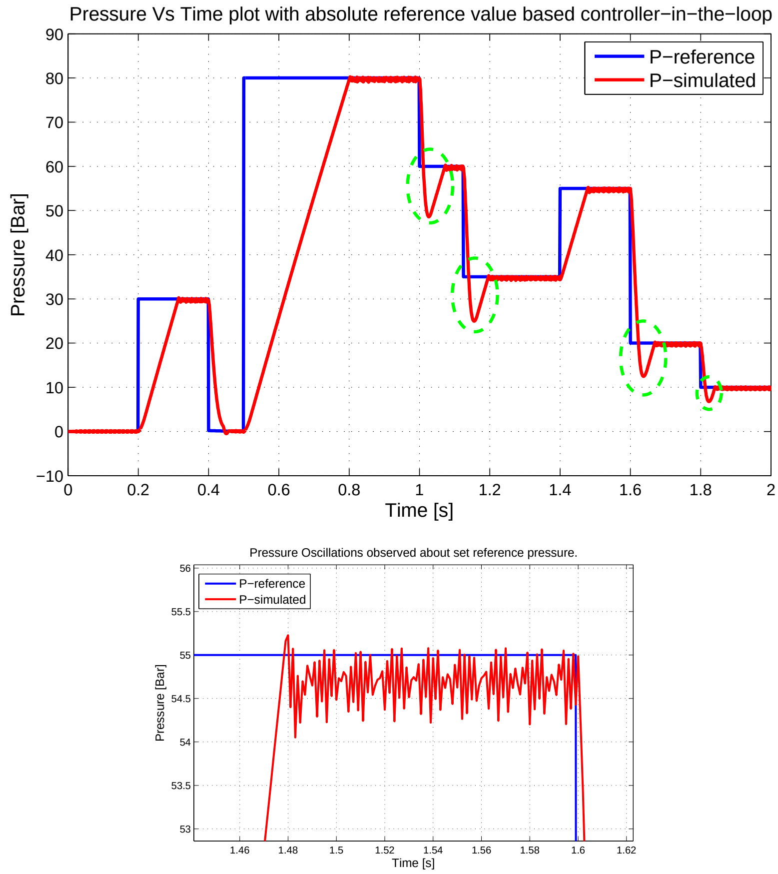


Figure 4.10: Tracking performance of the absolute reference value based controller.

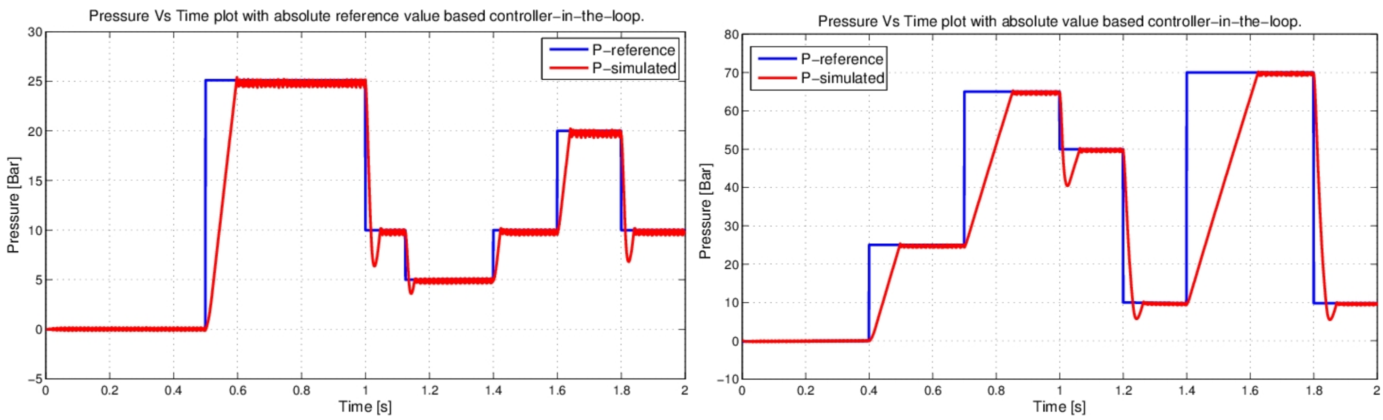
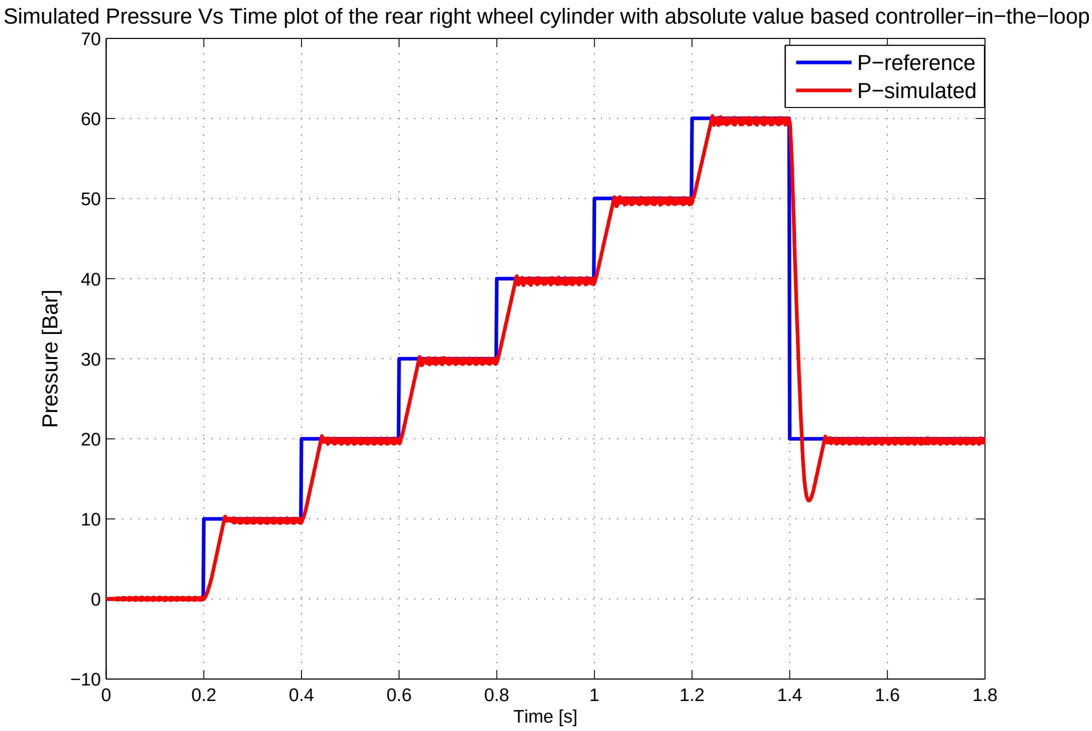


Figure 4.11: Additional illustrations of the tracking performance of the absolute reference value based controller.



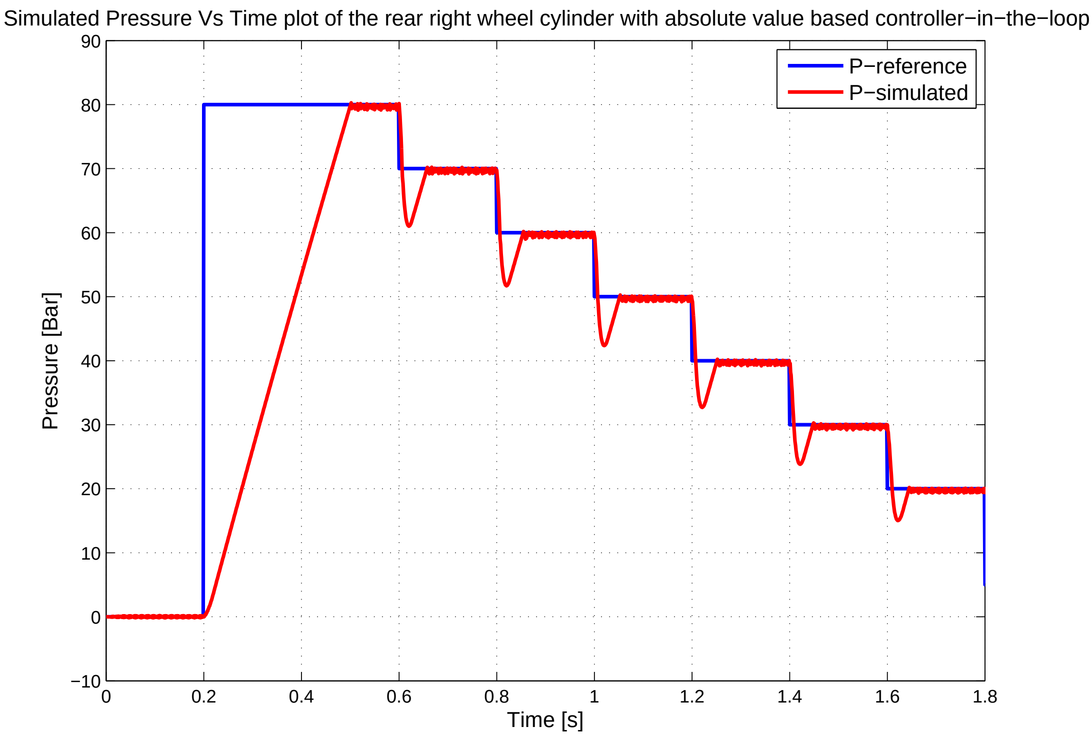


Figure 4.12: Tracking performance of the absolute reference value based controller.

Further simulations with small pressure rise and pressure release steps have been performed to analyse the closed loop performance, the results of which are depicted in Fig.4.12. As seen from Fig.4.12, even in the case of small pressure steps, the absolute value based controller exhibits undesirable chattering about the reference pressure value. The pressure drop when switching from a pressure release mode to pressure hold mode is also observed for small pressure release steps. In this case the pressure rise steps are of smaller magnitude as compared to the previous test and hence, the settling time is relatively less.

4.5.2 Threshold Based Switching Algorithm

Fig.4.13 and 4.14 depict the tracking performance of the of the closed loop with the threshold based controller. It is observed that, unlike the absolute reference value based controller, the threshold based controller does not cause pressure oscillations about the set reference pressure value and hence, the system reaches a steady state when the reference pressure is constant. The numerical value of the threshold is a controller parameter and is chosen as ± 2 bar. A higher value of the threshold adversely affects the tracking accuracy and its lower limit is constrained by pressure oscillations (chattering) which may lead to an unstable system. The minimum settling time, as mentioned in Section 4.1, is constrained due to the relatively slow dynamics of the internal pump (300 bar/s). The problem with the measured pressure dropping considerably below the reference pressure when switching from a pressure release mode to a pressure hold mode, as highlighted in Fig.4.13, due to the high outlet valve inertia, is also observed in the threshold based switching algorithm. Therefore, further investigation and modifications to the controller are necessary to resolve this persistent issue, despite the hardware limitations. This controller improvement based on the simulation results is described in the subsequent section.

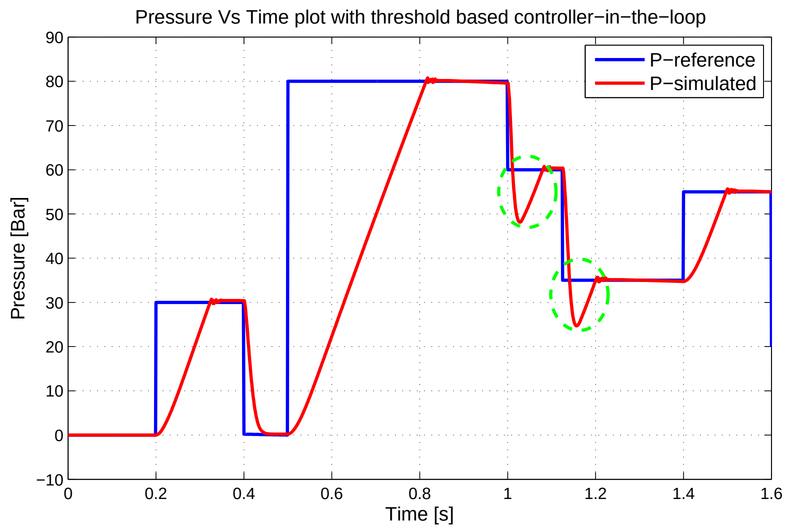


Figure 4.13: Tracking performance of the threshold based controller.

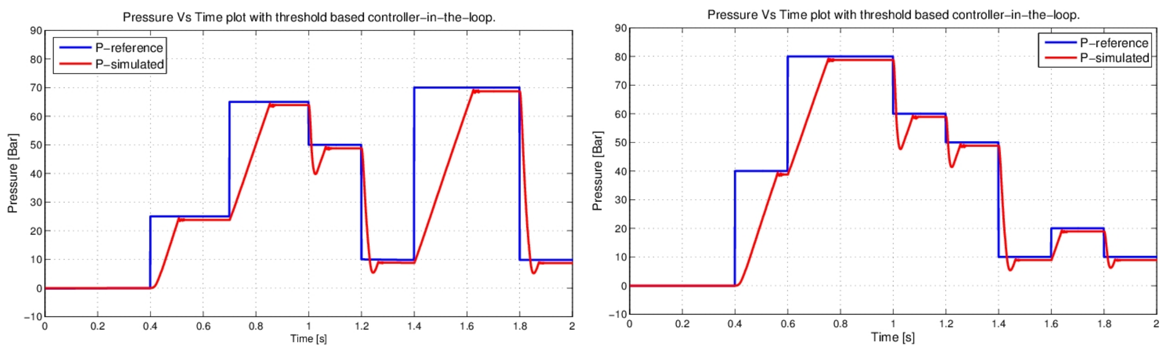


Figure 4.14: Additional illustrations of the tracking performance of the threshold based controller.

Simulations with small pressure rise and release steps have been performed and the results of these simulations are depicted in Fig.4.15. It is observed that the threshold based switching algorithm is devoid from the chattering problems encountered in the absolute value based switching algorithm, even for small pressure increase/decrease steps. The pressure drop issue when switching from the pressure release to the pressure hold mode is observed even in the case of small pressure release steps, as depicted in Fig.4.15. Results of closed loop simulations with highly dynamic reference pressure signals are provided in Appendix E.

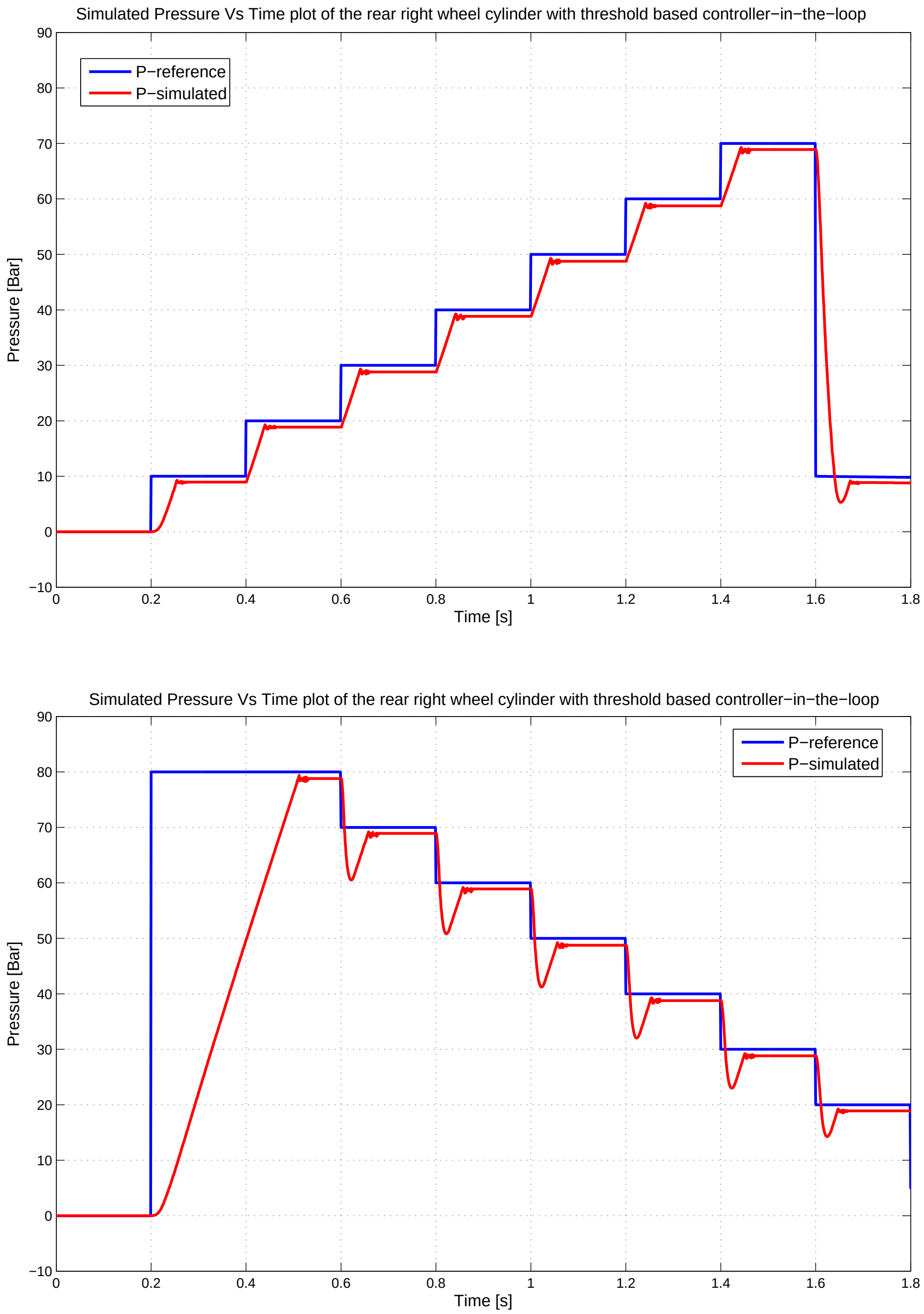


Figure 4.15: Tracking performance of the threshold based controller.

4.6 Controller Improvements based on Simulation Results

This section describes the modifications to the low level brake pressure controller in order to compensate for the pressure discrepancy highlighted in Section 4.5. This pressure discrepancy when switching from pressure release mode to pressure hold mode, is attributed to the high closing time of the outlet valve when de-energised, which is a result of its high inertia. The magnitude of this pressure discrepancy can be as high as 10-15 bar, which is unacceptable for brake pressure control. Thus, a solution to minimize the pressure discrepancy is described in this section. The fundamental notion is to increase the target reference pressure to be achieved after the pressure release mode, by a known magnitude, for a fixed duration of time. This time duration is numerically equal to the outlet valve closing time.

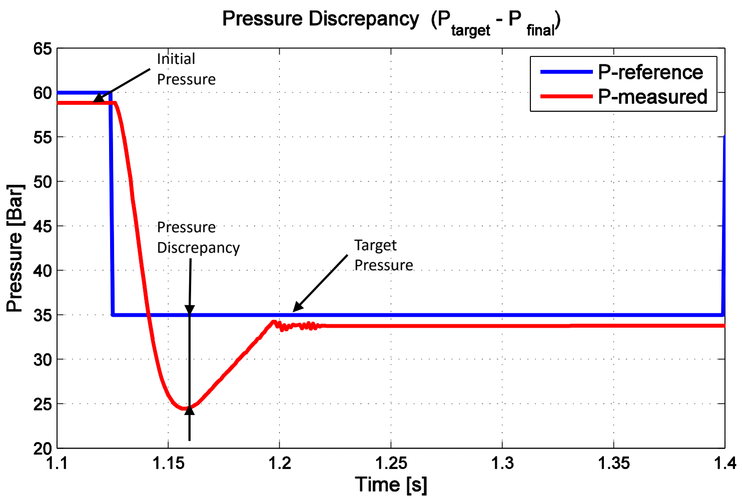


Figure 4.16: Pressure Discrepancy while switching from pressure release to pressure hold mode.

It is observed from the closed loop simulation results, that the magnitude of the pressure discrepancy depends upon the initial pressure before the outlet valve is opened and the target reference pressure after the pressure release mode is executed i.e. the outlet valve is closed, see Fig.4.16. This is attributed to the fluid flow through the outlet valve in the time span between the outlet valve closing signal and the actual closing of the outlet valve (actuation delay).

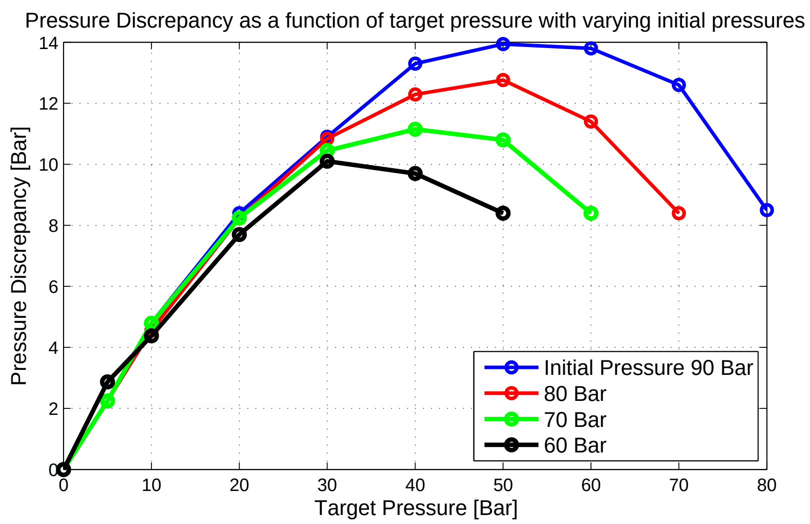


Figure 4.17: Pressure Discrepancy as a function of target pressure for various initial pressures.

Further investigation is performed to prove the strong correlation between the magnitude of the pressure discrepancy and the target reference pressure after the pressure release mode (termed as *target pressure* for simplification). The plot of the pressure discrepancy as a function of target pressure is depicted in Fig.4.17. Using this graph, a minimal magnitude of pressure discrepancy can be determined for each target pressure.

Thus, with knowledge on the magnitude of the pressure discrepancy as a function of target pressure, the pressure release mode can be modified by implementing a static map of the final expected pressure with respect to the target pressure, i.e. by computing a new reference pressure which compensates for the pressure discrepancy.

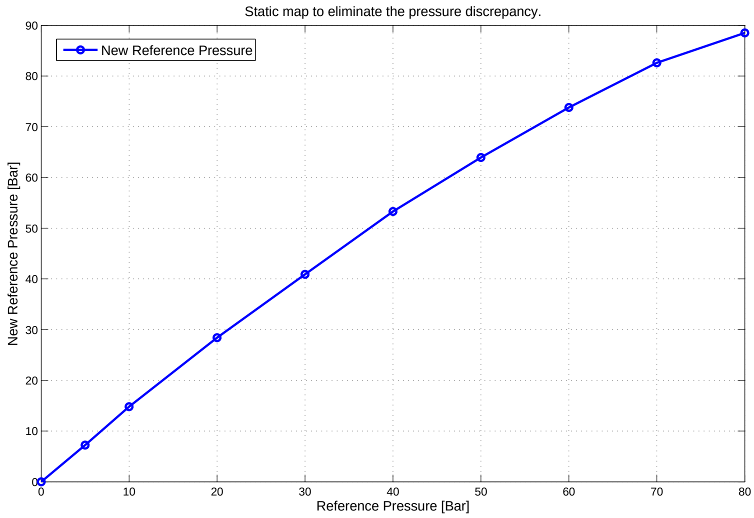


Figure 4.18: Static map of the new reference pressure as a function of original reference pressure or target pressure.

The condition for transition from the pressure release mode to the hold pressure mode ($P_{meas} < P_{bound+}$) is modified using the static map, i.e. $P_{meas} < P^*_{bound+}$, where P^*_{bound+} is based on the new reference pressure computed from the static map depicted in Fig.4.18. The new reference pressure is only implemented for a fixed time interval after the pressure release mode is triggered. This time interval is numerically equal to the closing time of the outlet valve. Once the outlet valve is completely closed, the condition for transition from the pressure release mode to the pressure hold mode is changed back to $P_{meas} < P_{bound+}$.
Since this pressure discrepancy is caused due to the high inertia or actuation delay of the outlet valve, it is only observed in the release pressure mode and hence, the controller logic for the build pressure mode is not altered. The result of a closed loop simulation performed after implementing the static map is depicted in Fig.4.19. As seen from Fig.4.19, the pressure discrepancy while switching from the pressure release mode to the pressure hold mode is suppressed up to a large extent and subsequently the pressure tracking accuracy is improved.

4.7 Summary

In this chapter, the closed loop performance objectives were formulated and the hardware constraints were explained. It was inferred that the hardware constraints would limit the closed loop performance of the system and the low level brake pressure controller would have to be designed considering these limitations. Subsequently, two switching algorithms were formulated, i.e. absolute reference value based switching algorithm and threshold based switching algorithm. Both low level brake pressure control algorithms were implemented as MATLAB Simulink models and were simulated in a closed loop together with the model of the hydraulic modulator. The MATLAB

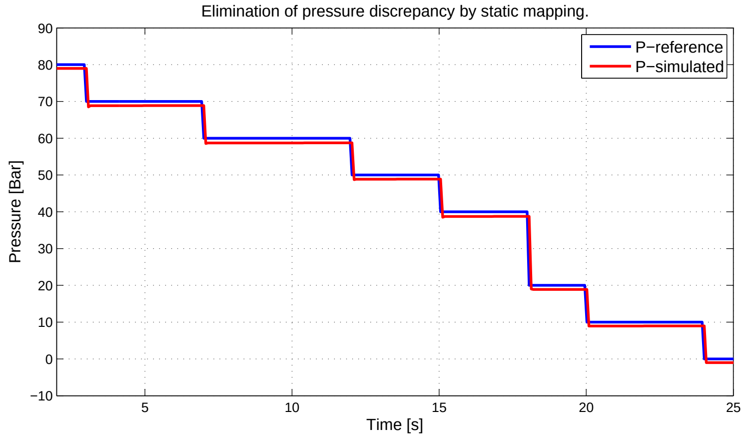


Figure 4.19: Result of simulation after implementing static map to eliminate the pressure discrepancy

Simulink model of the closed loop was explained in Section 4.4. The results of the aforementioned closed loop simulations were analysed in Section 4.5 and, it was concluded that the threshold based switching algorithm exhibits considerably better tracking performance as compared to the absolute reference value based switching algorithm. The absolute reference value based controller suffered from the problem of chattering about the set reference pressure value and hence was discarded. Furthermore, it was observed through closed loop simulations that both controllers suffered from the problem of a negative overshoot when switching from the pressure release to the pressure hold mode. Subsequently, this problem was resolved by implementing pressure compensation through a static map as explained in detail in Section 4.6.

Chapter 5

Real Time Controller Implementation

This chapter is focused on the real time implementation of the low level brake pressure controller on the braking system test bench introduced in Section 3.3. The purpose of this real time implementation is to validate the closed loop controller performance on the braking system hardware. Section 5.1 describes the interface with the braking system test bench and the steps involved in the process of implementing the low level brake pressure controller, developed in MATLAB Simulink, onto the electronic control unit of the hydraulic modulator.

5.1 Interface with the Braking System Test Bench

The real time implementation of the low level brake pressure controller necessitates a hardware and a software interface, between the electronic control unit of the hydraulic modulator and the computer containing the dSPACE Targetlink model of the control system. The software interface consists of the following tools:

• dSPACE Targetlink - Targetlink is a software system that generates production code (C language code) directly from MATLAB Simulink/Stateflow models. In this master thesis, it is used to convert the MATLAB Simulink model of the low level brake pressure controller into C language code, which can then be flashed onto the electronic control unit of the hydraulic modulator for experimental validation of the controller.

• Vector CANape - CANape is a software tool used for calibration of ECU's and recording measurement data. Data measured from the various sensors can be logged time synchronously with the ECU signals and can be represented in a custom built graphical environment. In this master thesis, CANape is implemented for two specific tasks. Primarily, it is used as a real time data logging and visualisation tool, to analyse and compare the trajectories of the set reference pressure and the actual wheel brake cylinder pressure for each wheel brake. Secondly, it is used to vary the reference pressure for each wheel brake cylinder independently. It is also used to control the solenoid valves and the internal DC motor of the hydraulic modulator manually (i.e. without the controller in the loop), when required. Additionally, it was also used for calibration of the wheel pressure sensors in real time as the model parameters i.e. the sensor scaling and offset, could be modified during runtime through the CANape GUI. The customised CANape interface developed for the analysis of data from the wheel brake pressure sensors and for manual control of the hydraulic modulator is depicted in Fig.5.1.

As depicted in Fig.5.1, the customised Vector CANape interface is divided into six windows, two of which are used for parameter settings and the other four for data visualisation and analysis. Window 1 (blue) is used for setting a reference pressure value for each individual wheel brake cylinder. This reference pressure can either be changed in real time through window 1 or programmed

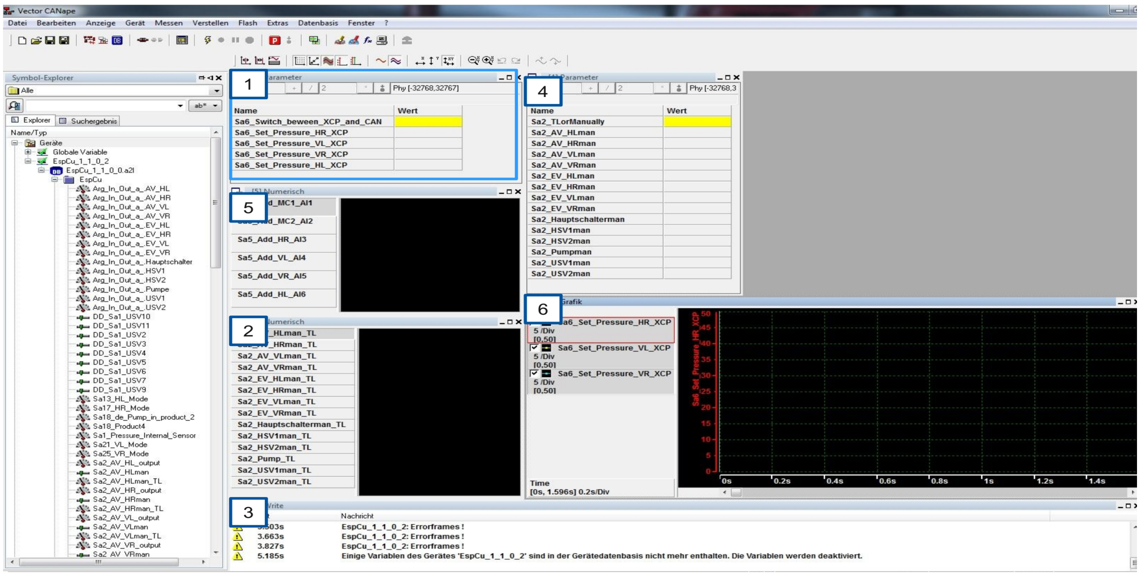


Figure 5.1: Vector CANape interface to the braking system test bench.

into the Targetlink model as a continuous input signal. The system can also switch between the two input methods through the first input signal in window 1. Window 4 enables the user to control each solenoid valve and the internal pumps of the hydraulic modulator manually (without the controller in the loop). This function is particularly useful for system diagnostics to ensure the proper operation of all the valves and the DC motor of the hydraulic modulator, before the system is operated with the controller in the loop. Window 2 displays the state of each valve and the DC motor in real time i.e. energised or de-energised and window 5 displays the real time value of the readings of the six pressure sensors installed on the braking system test bench. Window 6 is used for the visualisation of set pressure and wheel brake pressure signals from the pressure sensors respectively. All the data signals from the ECU and the wheel brake pressure sensors are stored in individual files which can be imported into MATLAB for further post processing and analysis. The start of each measurement session is triggered by the user and the data acquisition runs in a loop until the user terminates the session.

The hardware interface between the custom ECU of the hydraulic modulator and the computer containing the Targetlink model of the control system, consists of an Input/Output CAN to USB interface (Vector CANcase XL - Appendix B). This is used to establish a CAN protocol connection to the custom ECU. The control of the valves and the internal pumps of the hydraulic modulator is only possible via the custom ECU developed by IKA.

The steps in the real time implementation of the low level brake pressure controller are summarised in Fig.5.2. The MATLAB Simulink model of the low level brake pressure controller is adapted for the dSPACE Targetlink software system, which is subsequently used to generate C/Production code for real time implementation. This generated C code is then transferred to the custom ECU of the hydraulic modulator through an Input/Output CAN-USB interface. A standard computer with the Vector CANape software tool is used to measure and analyse the data from the pressure sensors.

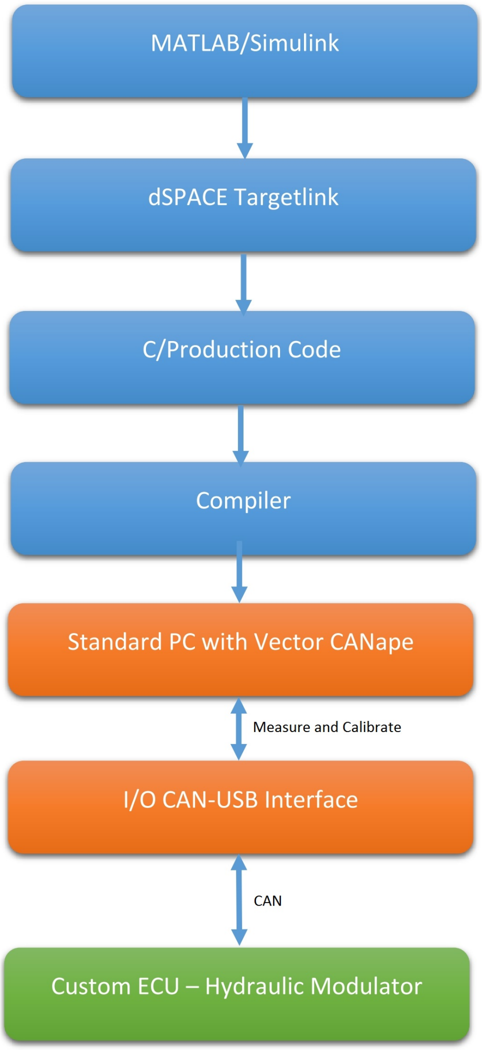


Figure 5.2: Steps involved in the real time implementation of the brake pressure controller.

Chapter 6

Experimental Results and Controller Validation

The control architecture, closed loop simulation results and real time implementation of the low level brake pressure controller has been explained in detail in the previous chapters. The focus of this chapter is on the analysis of experimental results and the closed loop performance of the braking system hardware with the new low level brake pressure controller. The experiments are performed with different reference pressure signals in order to validate the performance of the controller under varying braking scenarios. These experiments are indicated as "pressure tracking experiments" in this research work. The term "performance" in this context refers to the reference pressure tracking characteristics such as settling time, overshoot and maximum relative error. The MATLAB Simulink model of the low level brake pressure controller is adapted and integrated into a dSPACE Targetlink model for real time implementation, which is described in Section 6.1. Section 6.2 describes the problems encountered while performing experiments on the braking system test bench and the solutions adopted to overcome them. The validation of the closed loop simulation results with experimental data is discussed in Section 6.3.

6.1 Targetlink Model

This section describes the dSPACE Targetlink model of the low level brake pressure controller, which is essentially the MATLAB Simulink model of the low level brake pressure controllers for each wheel brake cylinder, with certain modifications and adaptations to make it suitable for real time implementation or Hardware in the Loop (HiL) tests. As explained in the previous section, dSPACE Targetlink is a software system that generates C language code from MATLAB Simulink models but, before this production code can be generated, the MATLAB Simulink model has to be modified to include Targetlink compatible blocks for efficient code generation.

The dSPACE Targetlink model used for the HiL tests on the braking system test bench is depicted in Fig.6.1. Model inputs are the four analog wheel brake cylinder pressure signals, two master cylinder pressure signals and the set reference pressure for each wheel brake cylinder. Analog signals from the wheel brake cylinder pressure sensors and the master cylinder pressure sensors, are initially scaled and offset by the corresponding subsystem. The scaling and offset parameters in the aforementioned subsystems can be altered in real time through the Vector CANape interface as explained in Section 6.1. The Targetlink model consists of 4 wheel brake pressure controller subsystems, as described in Section 5.1.2, one for each wheel brake cylinder. These subsystems utilise the reference pressure and actual wheel brake pressure signals to control the solenoid valves and the internal pumps of the hydraulic modulator, by implementing the threshold based switching algorithm explained in Chapter 4. The model outputs are the control signals for each of the twelve solenoid valves of the hydraulic modulator together with the control signal for the DC motor which drives the two internal piston pumps. Each block used in this particular model is optimised for Targetlink and the signals to be visualised in CANape, or parameters which have to be modified in real time, are specified in the Targetlink model settings. The *BlockTLorMan* subsystem is implemented in order to enable switching from Targetlink mode i.e. controller in the loop mode, to manual mode, in which the valves and the internal pumps can be controlled manually through the CANape interface (as explained in Section 6.1).

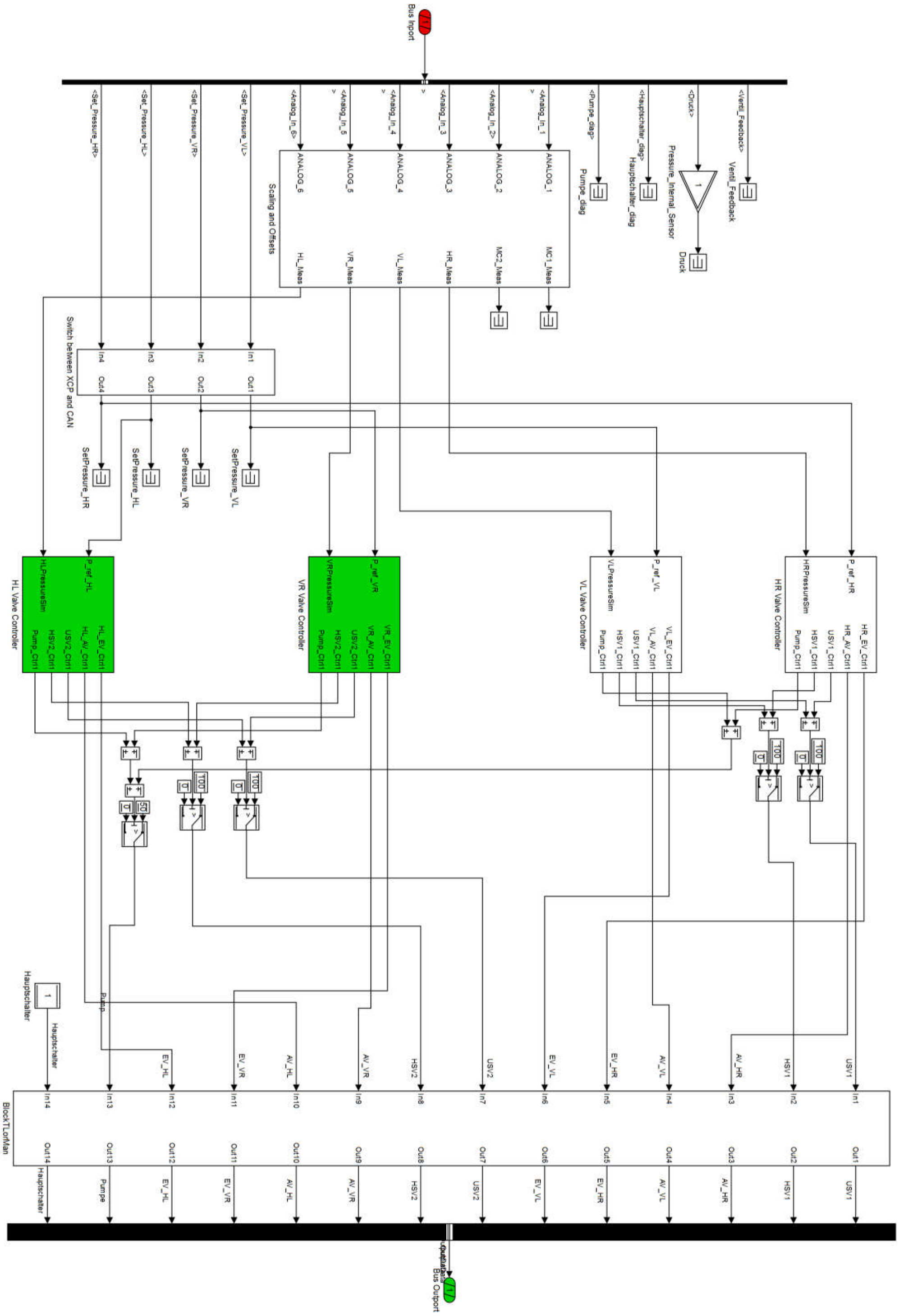


Figure 6.1: dSPACE Targetlink Model used for HiL tests.

6.2 Preliminary Experiments and Hardware Issues

This section is focused on the preliminary HiL experiments on the braking system test bench, the hardware issues encountered during experimentation and the solutions adopted to resolve these issues.

Closed loop pressure tracking experiments are performed on the braking system test bench by applying a staircase reference pressure signal to the rear left wheel brake cylinder, through the Vector CANape software interface. The low level brake pressure controller controls the valves and the internal pumps of the hydraulic modulator to achieve the requested reference pressure in the rear left wheel brake cylinder. The objective of performing these experiments is to analyse the closed loop tracking performance of the low level brake pressure controller. Fig.6.2 depicts the result of one such pressure tracking experiment with a staircase reference pressure signal consisting of pressure build steps.

The major discrepancy observed in successive instances of preliminary pressure tracking experiments, is highlighted in Fig.6.2. As depicted in the aforementioned figure, a sudden loss of wheel brake cylinder pressure is observed at the initiation of each pressure build mode i.e. when the magnitude of the set reference pressure increases. This discrepancy, observed repeatedly during preliminary HiL tests, could not be attributed to any particular hardware component initially. The pressure drop induces a significant time delay in the pressure build-up and results in large unacceptable deviations from the set reference pressure, as depicted by the pressure error plot in Fig.6.2. It also causes a significant and unwanted rise in pressure in the other wheel brake cylinder connected in the same braking circuit even when its set reference pressure is zero i.e the front right wheel brake cylinder, refer to Fig.6.3- VR-Measured. Furthermore, it leads to closed loop system instability for reference pressure steps of high magnitude, see Fig.6.3.

The pressure drop issue explained above has to be resolved, in order to ensure acceptable closed loop performance and stability. In order to address this issue, it is imperative to perform a root cause analysis. Initial tests with the braking system in manual mode, with the brake pedal and vacuum booster in operation, eliminated the possibility of any valve leakage or excessive wear of the pressure relief valves. Subsequently, the functioning and calibration of the wheel brake pressure sensors was re-examined to ensure that the pressure measurements are reliable.

The next step in the root cause analysis was to examine the electric circuit components of the ECU of the hydraulic modulator, for any voltage anomalies which might lead to incomplete valve opening/closing. This was performed by utilising a digital real time oscilloscope to observe the voltage across the solenoid valve terminals and the electrical switches on the ECU during the pressure tracking experiments, which are depicted in Fig.6.2. Ideally, the voltage across the solenoid valve terminals should remain constant at 13V, to hold the solenoid valves in their completely open/closed energised state when desired. Subsequently, it was observed that the voltage supplied to the terminals of the solenoid valves, in order to hold them in the completely open/closed energised state, exhibited a drastic drop at the instant the DC motor is energised to drive the internal pumps. This undesirable voltage drop for a short duration, results in an improper sealing between the valve and valve seat and allows the brake fluid to flow against the intended direction i.e. from the rear left wheel brake cylinder to the front right wheel brake cylinder, due to the incomplete closure of both inlet valves. This issue is resolved by utilising two separate power sources, one for the internal DC motor of the hydraulic modulator and the other for the ECU of the hydraulic modulator. This ensures that there is no undesirable voltage drop in the supply voltage of the solenoid valves and hence, the problem of the undesirable pressure drop was successfully eliminated.

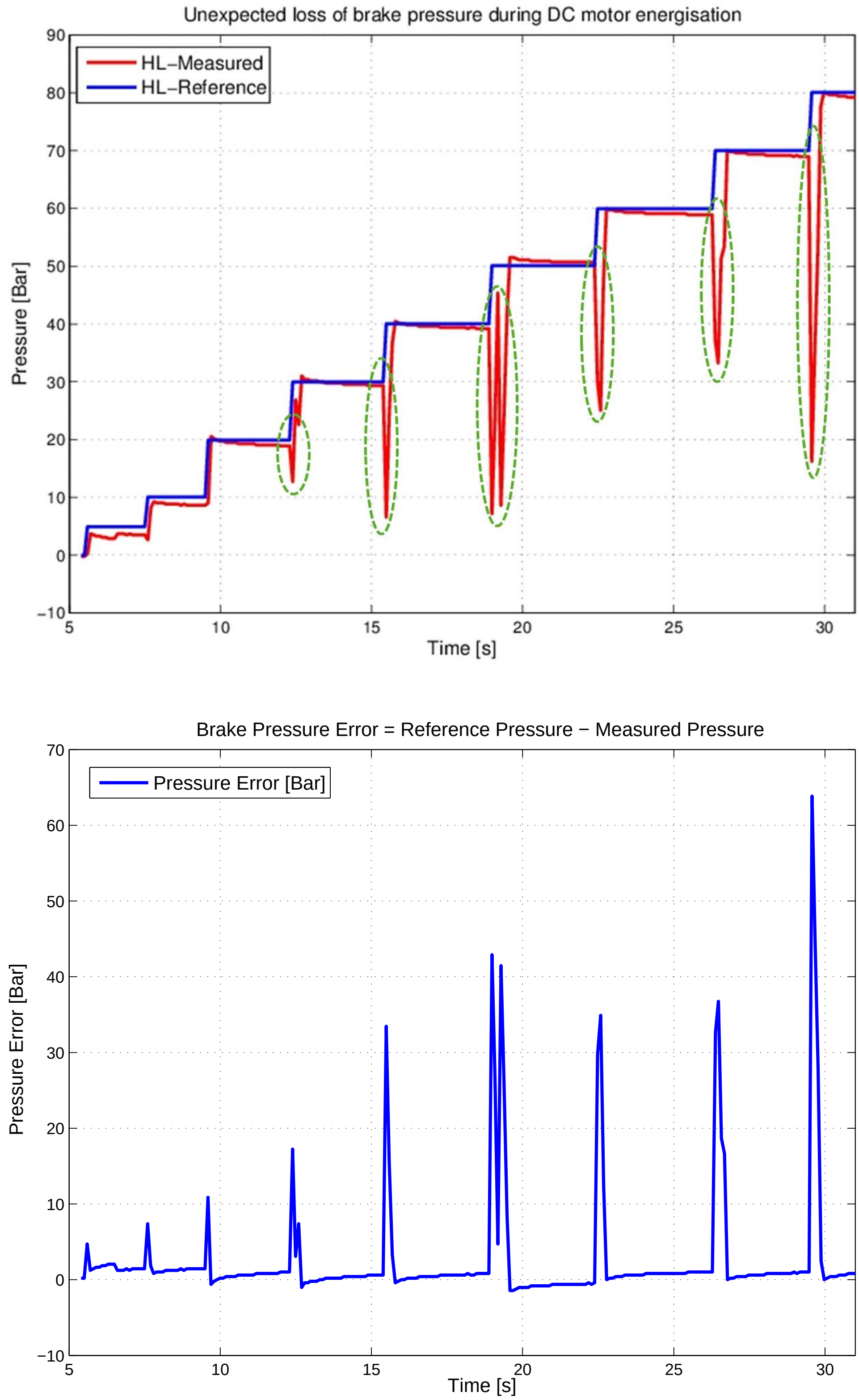


Figure 6.2: Sudden unexpected loss of brake pressure during DC motor energisation (HL-Rear Left).

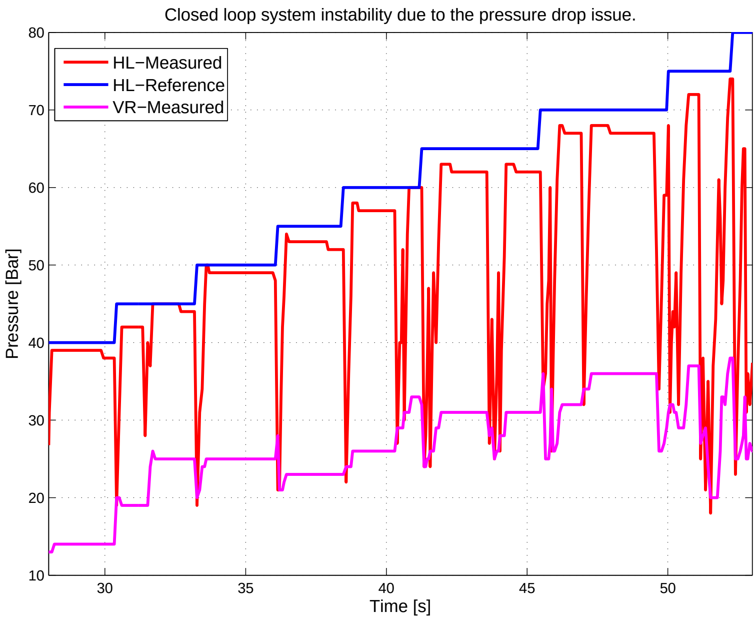


Figure 6.3: Closed loop system instability due to the pressure drop issue.

6.3 Controller Validation

This section describes the evaluation of the low level brake pressure controller, designed in Chapter 4, in a hardware in the loop environment. The ECU of the hydraulic modulator controls its solenoid valves and internal pumps, based on the threshold based switching algorithm of the low level brake pressure controller. The desired reference pressure trajectory to be tracked is communicated to the ECU through the Vector CANape interface described in Chapter 5. The wheel brake cylinder pressure signals are transmitted in real time from the wheel brake pressure sensors to the data acquisition system via the ECU. The control system runs at a sample time of 100 ms, which is limited by the volume of data that can be transmitted through the CAN bus hardware in real time. Furthermore, the controller parameters used in the simulations are also used for the pressure tracking experiments. The initial condition of all four wheel brake cylinders is set to 0 bar and the desired reference pressure trajectory to be tracked is communicated to the ECU of the hydraulic modulator in real time.

The pressure error is defined as the difference between the reference wheel brake cylinder pressure and the measured wheel brake cylinder pressure. The performance criteria used for controller validation are the mean relative pressure error and the standard deviation of the pressure error. The equations used for computing the mean relative error (%) and standard deviation (%) are stated below:

$$e_{rel} = \frac{P_{ref} - P_{meas}}{P_{meas}}.100\% \tag{6.1}$$

$$e_{mean} = \frac{\sum_{i=1}^{N} e_{rel(i)}}{N}.100\% \tag{6.2}$$

$$StandardDeviation = \sqrt{\frac{\sum_{i=1}^{N}(e_{rel}(i) - e_{mean})^2}{N}}.100\% \tag{6.3}$$

6.3.1 Staircase Reference Signal - Pressure Build Up

The closed loop performance of the low level brake pressure controller while tracking a staircase reference pressure signal with pressure build steps, is depicted in Fig.6.4. As seen from Fig.6.4, the problem with the undesirable pressure drop, as explained in Section 6.2, is now resolved and hence the tracking performance is improved considerably. The issue with the unexpected pressure rise in the other wheel brake cylinder connected in the same brake circuit, is also eliminated by utilising individual power sources, as explained in Section 6.2. Moreover, the reference pressure is tracked without any chattering or pressure oscillations about the reference pressure valve.

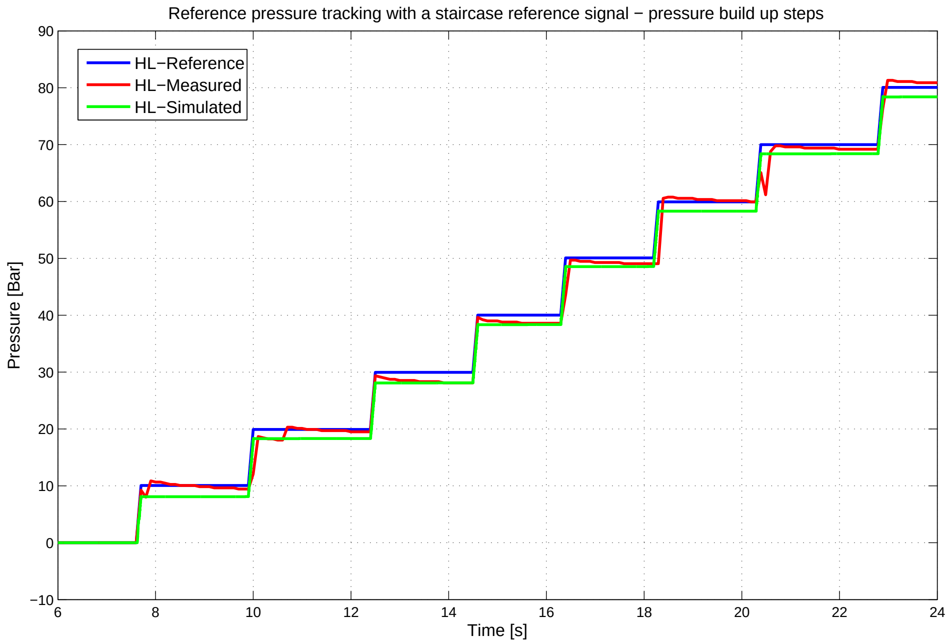


Figure 6.4: Closed loop controller performance for a staircase reference signal with pressure build up steps.

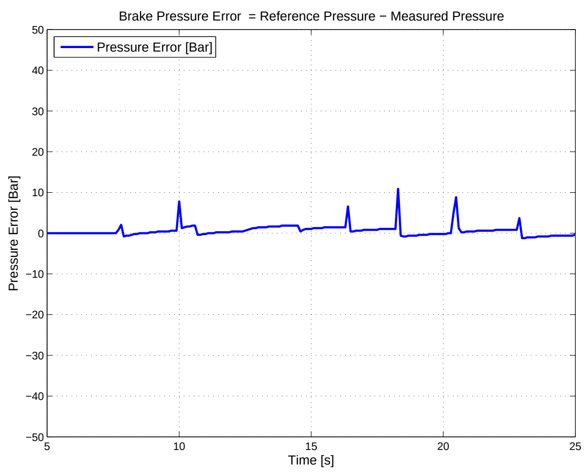


Figure 6.5: Brake pressure error as a function of time.

Fig.6.5 depicts the absolute error between the reference brake pressure and the measured wheel brake cylinder pressure, as a function of time. The steady state mean relative pressure error for this pressure tracking experiment is approximately 3%. The abrupt peaks in the absolute pressure error are instantaneous and are caused due to the relatively slow dynamics of the internal pump of the hydraulic modulator. As mentioned in Section 4.2, the internal pump is capable of a maximum pressure build rate of only 300 bar/s. Thus, there is an unavoidable delay before the reference pressure signal can be tracked accurately, when switching from the pressure hold to the pressure build mode. In spite of the hardware limitations, the actual pressure tracking performance is in close agreement with simulation results.

6.3.2 Staircase Reference Signal - Pressure Release Steps

This subsection describes the closed loop performance of the low level brake pressure controller while tracking a staircase reference pressure signal with pressure release steps. Fig.6.6 depicts the closed loop pressure tracking performance for a staircase type reference signal with pressure release steps.

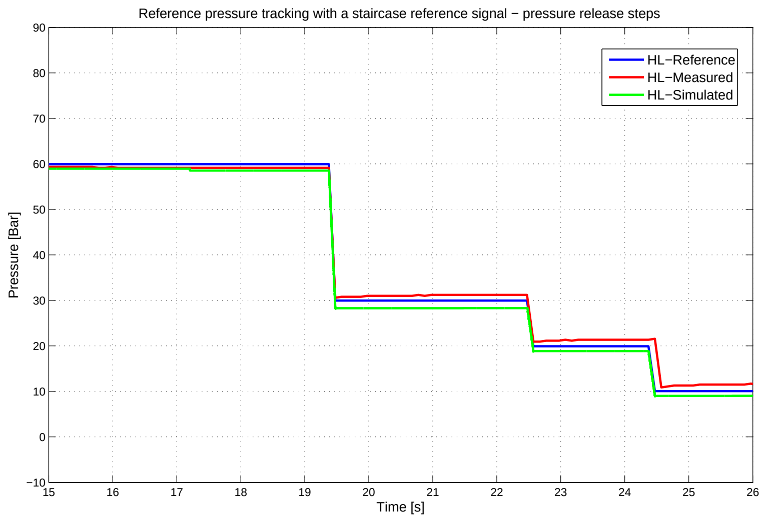


Figure 6.6: Closed loop controller performance for a staircase reference signal with pressure release steps.

The evolution of the brake pressure error as a function of time, for the pressure tracking experiment with a staircase reference signal with pressure release steps, is depicted in Fig.6.7. The magnitude of the mean relative pressure error for this pressure tracking experiment is approximately 7%.

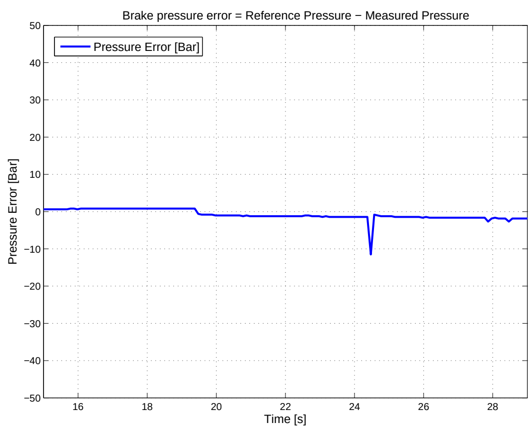


Figure 6.7: Brake pressure error as a function of time.

6.3.3 Reference Pressure Signal with Pressure Build and Release Steps

This subsection describes the results of a pressure tracking experiment with a reference pressure signal comprised of pressure build up followed by pressure hold and release steps. Fig.6.8 depicts the closed loop pressure tracking performance for such a staircase type reference signal, i.e. a comparison between the reference pressure trajectory, the simulated pressure trajectory and the measured pressure trajectory.

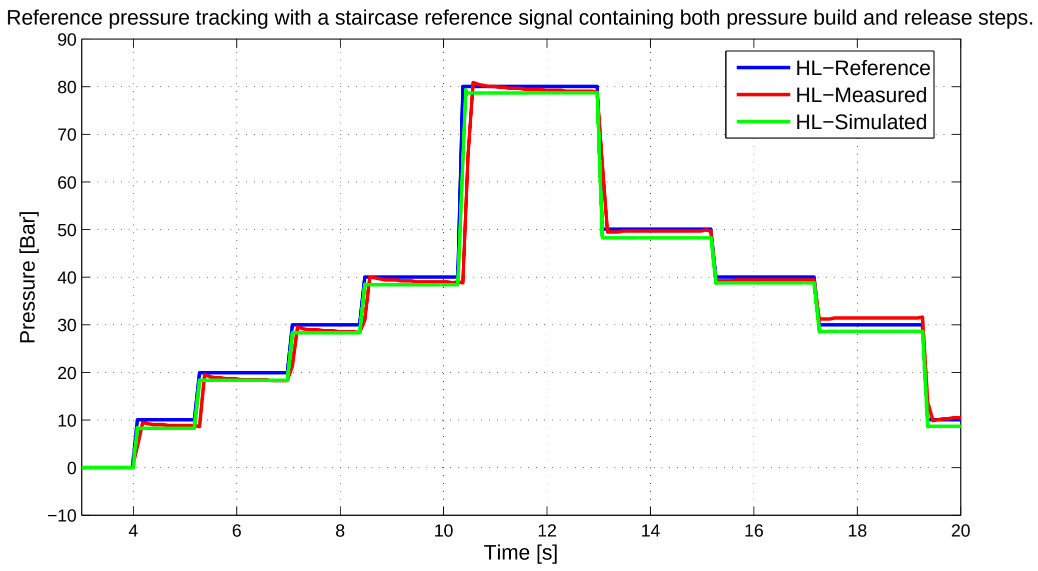


Figure 6.8: Closed loop controller performance with a staircase reference signal.

As seen from Fig.6.8, the measured pressure trajectory does not deviate significantly from the reference pressure trajectory and no chattering about the reference pressure is observed. The brake pressure error as a function of time, depicted in Fig.6.9, exhibits a mean relative error value of 4.99%.

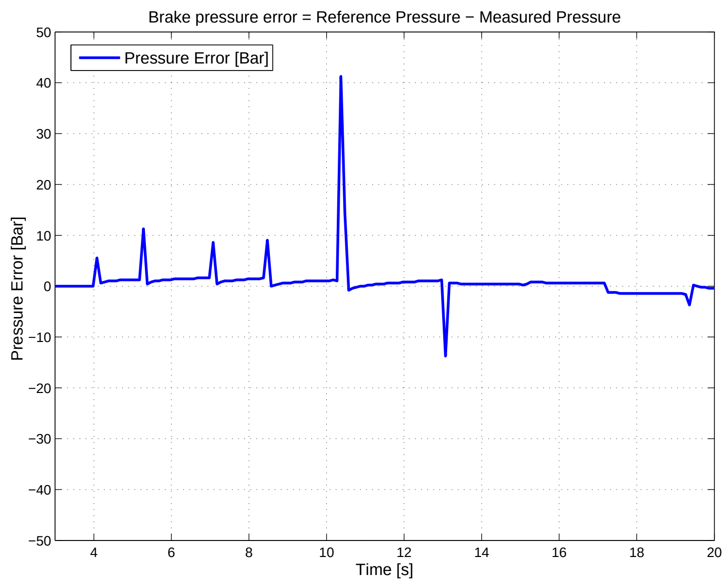


Figure 6.9: Brake pressure error as a function of time.

6.4 Summary

This section provides a concise summary of Chapter 6. An overview of the mean relative error and the standard deviation of the relative pressure error, is provided in Table 6.1. It should be noted that a specific tolerance value for the allowable brake pressure error is not available in braking control literature. Hence, it is difficult to judge the closed loop performance of the controller quantitatively.

Reference Signal	**Mean Relative Error** (%)	**Standard Deviation** (%)
Staircase Reference Signal, Build Steps	2.9449	4.0683
Staircase Reference Signal, Release Steps	6.9910	11.0043
Staircase Reference Signal, Build & Release Steps	4.9850	8.5192

Table 6.1: Closed loop performance - statistical analysis.

The dSPACE Targetlink model used for real time implementation of the low level brake pressure controller on the braking system hardware was described in Section 6.1. The issues faced during experimentation with the braking system test bench and the methods adopted to resolve them were described in Section 6.2. The problem with the undesirable pressure drop when switching from a pressure hold mode to a pressure build mode, was attributed to the large magnitude of starting current drawn by the DC motor which drives the internal pumps of the hydraulic modulator. In Section 6.3, an analysis of the results of the pressure tracking experiments with staircase type reference pressure signals was performed and the low level brake pressure controller was subsequently validated. The performance criteria used to quantify the pressure tracking error were described. It was observed that the mean relative pressure tracking error with the designed low level brake pressure controller is approximately 7%.

Chapter 7

Conclusions and Recommendations

This chapter outlines the conclusions drawn from the simulation, implementation and testing of the low level brake pressure controller. Furthermore, recommendations for future research are provided by the author based on the observations and findings of this research work.

7.1 Conclusions

The fundamental research objective of this master thesis can be stated as "*the design and validation of a low level brake pressure controller*". The individual observations and results from this research work hence contribute towards meeting this main research objective of designing and validating a low level brake pressure controller. Based on the analysis of the closed loop simulation and HiL test results, the following conclusions can be drawn with respect to the low level brake pressure controller:

• From closed loop simulation results, it is concluded that the performance of the threshold based switching algorithm is superior to that of the absolute reference value based switching algorithm. The absolute reference value based switching algorithm demonstrates significant undesirable oscillations about the set reference pressure value (chattering).

• The threshold based switching control algorithm implemented in the low level brake pressure controller demonstrates decent closed loop tracking performance without chattering, both in simulations and HiL tests on the braking system test bench. The maximum relative pressure tracking error equals approximately 7%. Considering the hardware constraints, it can be concluded that the controller performance is acceptable.

• The pressure discrepancy observed when switching to the pressure hold mode from the pressure release mode is resolved to a great extent by the static mapping compensation technique. This can be observed in the HiL test results.

• From the test results, it can be concluded that the problem with the sudden pressure drop while switching from the pressure hold mode to the pressure build mode is caused by the large magnitude of starting current drawn by the DC motor when energised. It is resolved successfully by using separate power sources for the internal DC motor and the ECU of the hydraulic modulator. For proper functioning, the ECU has to be connected to a constant voltage source without any fluctuations and hence, it cannot be connected in parallel to the internal DC motor of the hydraulic modulator.

• Lastly, as the closed loop simulation results are in fairly close agreement with the HiL test results, it can be concluded that the designed low level brake pressure controller has been implemented and validated. Although its tracking performance is acceptable, there is still room for improvement before it is implemented for its intended research applications, i.e. testing high level ABS and ESC algorithms and for developing a partial brake by wire system. Recommendations for such improvements to the brake pressure controller are provided in the subsequent section.

7.2 Recommendations for Future Research

This section lists the recommendations based on the findings and observations of the research work performed, to aid future research work:

• The most significant hurdle in achieving perfect reference pressure tracking is that it is impossible to implement continuous control of the brake pressure due to the limitations with the current braking system hardware, which possesses only discrete ON/OFF type solenoid valves. To enable continuous regulation of the brake pressure with increased accuracy, it is recommended to replace the ON/OFF solenoid valves of the hydraulic modulator with proportional servo valves after investigating the feasibility of doing so. This will allow the application of continuous control algorithms and hence, can ensure finer brake pressure regulation albeit at the cost of added complexity.

• The wheel brake pressure can be estimated by implementing state observers or identification algorithms, instead of measuring it with individual wheel brake pressure sensors. This could eliminate the feedback delay and possibly improve the response of the closed loop system.

• The low level brake pressure controller developed in this research work can be used to test high level ABS and ESP algorithms, such as the ones based on neural networks and linear quadratic regulators [17, 19], physically on the braking system test bench. While performing such tests, it is recommended to use short test cycles to prevent the solenoid valve coils and the ECU components from overheating.

• For further analysis, it is also recommended to perform road tests of the control system to investigate the effects of the voltage of the power source of the internal DC motor of the hydraulic modulator and its ECU, on the closed loop performance. This is based on the observations from the pressure tracking experiments which demonstrate that there is strong correlation between the voltage of the power source for the internal DC motor of the hydraulic modulator and its closed loop system performance.

• Since it is intended to implement a partial brake by wire system by using the low level brake pressure controller developed in this research work, the next step towards this objective could be the design and implementation of a brake pedal module, which translates the brake pedal displacement into the required/demand brake pressure signal.

• In this research work, the low level brake pressure controller has been tested with staircase type reference pressure signals. For further analysis, it is recommended to test the performance of the low level brake pressure controller on the braking system hardware with highly dynamic reference pressure signals e.g. sinusoidal reference signal.

Bibliography

[1] Hesse L., Schwarz B., Klein M., Eckstein L., 2014, "The Wheel Individually Steerable Front Axle of the Research Vehicle SpeedE, 5th International Munich Chassis Symposium, Springer Vieweg.

[2] Struth M., Forsthovel S., Eckstein L., 2015, "Functional Potential by Crosslinking Domains Optimized Recuperation for the SpeedE Research Vehicle, 24th Aachen Colloquium Automobile and Engine Technology 2015.

[3] Aly A., Zeidan S., Hamed A., Salem F., 2011, "Antilock-Braking Systems (ABS) Control: A Technical Review, Intelligent Control and Automation - Volume 2.

[4] Castillo J., Cabrera J., Guerra A., Simon A., 2015, "A Novel Electro-Hydraulic Brake System with Tire-Road Friction Estimation and Continuous Brake Pressure Control", IEEE Transactions on Industrial Electronics - Volume 63, Issue 3.

[5] Lennon K., Passino K., 1999, "Intelligent Control for Brake Systems", IEEE Transactions on control systems technology, Volume 7, No. 2.

[6] Shih M., Wu M., 2001, "Using the Sliding Mode PWM method in an Anti-lock Braking System", Asian Journal of Control, Volume 3, No. 3.

[7] Chaitas S., 2015, "Simulation and Validation of an ABS/ESP Hydraulic Modulator", Master Thesis Report, FH Aachen.

[8] Buckholtz K., 2002, "Reference input wheel slip tracking using sliding mode control", SAE Technical Paper 2002-01-0301.

[9] Drakunov S., Dix P., Ashrafi B., Ożgüner U., 1995, "ABS control using optimum search via sliding modes", Volume 3, IEEE Transactions on Control Systems Technology.

[10] Johansen T., Petersen I., Kalkkuhl J., Lüdemann J., 2003, "Gain-scheduled wheel slip control in automotive brake systems", Volume 11, IEEE Transactions on Control Systems Technology.

[11] "Antilock Brake System Review", 1992, SAE Technical Report J2246.

[12] Wellstead P., Pettit N., 1997, "Analysis and redesign of an antilock brake system controller", IEEE Proceedings on Control Theory and Applications 144, 413426.

[13] Jiang F., Gao Z., 2000, "An adaptive nonlinear filter approach to the vehicle velocity estimation for ABS, Proceedings of the 2000 IEEE International Conference on Control Application.

[14] Ray L., 1995, "Nonlinear state and tire force estimation for advanced vehicle control", IEEE Transactions on Control Systems Technology 3(1), 117124.

[15] Tanelli M., Savaresi S., Cantoni C., 2006, "Longitudinal vehicle speed estimation for traction and braking control systems", Proceedings of the 2006 Conference on Control and Applications, Munich, Germany.

[16] Kobayashi K., Cheok K., Watanabe K., 1996, "Estimation of absolute vehicle speed using fuzzy logic rule-based Kalman filter", Proceedings of the 1995 American Control Conference, Seattle.

[17] Johansen T., Petersen I., Kalkkuhl J., Lüdemann J., 2001, "Wheel slip control in ABS brakes using gain scheduled constrained LQR", Proceedings of the American Control Conference.

[18] Solyom S., Rantzer A., Lüdemann, J., 2004, "Synthesis of a model-based tire slip controller", Vehicle System Dynamics 41(6), 477511.

[19] Yu, J., 1997, "A robust adaptive wheel-slip controller for antilock brake system", Proceedings of the 36th IEEE Conference on Decision and Control.

[20] Breuer B., Eichorn U., Roth J., 1992, "Measurement of tyreroad friction ahead of the car and inside the tyre, Proceedings of the International Symposium on Advanced Vehicle Control (AVEC 92), pp. 347353.

[21] Gustafsson F., 1997, "Slip-based tire-road friction estimation", Automatica 33(6), 10871099.

[22] Müller M., Uchanski M., Hedrick K., 2003, "Estimation of the maximum tire-road coefficient", Journal of Dynamic Systems, Measurement and Control 125, 607 617.

[23] Savaresi M., Tanelli M., 2010, "Active Braking Control Systems Design for Vehicles", Springer Verlag - ISBN 978-1-84996-349-7.

[24] Reif K., 2014, "Brakes, Brake Control and Advanced Driver Assistance Systems", Springer Verlag - ISBN 978-3-658-03977-6.

[25] Shah S., 2016, "Design and Implementation of Adaptive Brake Pressure Controller", TU Delft Master Thesis.

[26] Kerst S., 2012, "Design and implementation of a novel force and brake pressure based Anti-lock Braking System", TU Delft Master Thesis.

[27] Paul v.d.Hof, 2012, "System Identification - Data Driven Modelling of Dynamic Systems", Lecture Notes - TU Eindhoven.

[28] Pacejka H., 2006, "Tyre and vehicle dynamics", Elsevier - ISBN 978-0-08-097016-5.

Appendix A

ABS System Requirements, Tyre Kinematics and Braked Wheel Dynamics

A.1 ABS system Requiements

An ABS system must satisfy a wide range of requirements associated with braking response and safety [24]:

• The main objective of an ABS system should be to maintain directional stability of the vehicle whilst minimising the braking distances on all types of road surface conditions.

• The braking control system must be capable of adapting to changing road surface conditions without a significant time delay.

• The ABS system must be capable of braking on mu-split surfaces whilst maintaining lateral stability of the vehicle.

• The system should be able to maintain directional stability even on highly uneven road surfaces irrespective of the braking force applied by the driver.

• The braking system should maintain the lateral stability of the vehicle during cornering under braking.

• The braking control system must be able to detect aquaplaning (tyres floating on a film of water on the road surface) and take necessary control action to maintain vehicle stability.

• The braking control system must be active at all vehicle speeds except those below 2.5 km/h, as the braking distance below such speeds is not critical.

• The braking control system must be equipped with a monitoring circuit which continuously monitors the correct functioning of the ABS system and warns the driver in case of system failure.

A.2 Longitudinal Tyre Kinematics

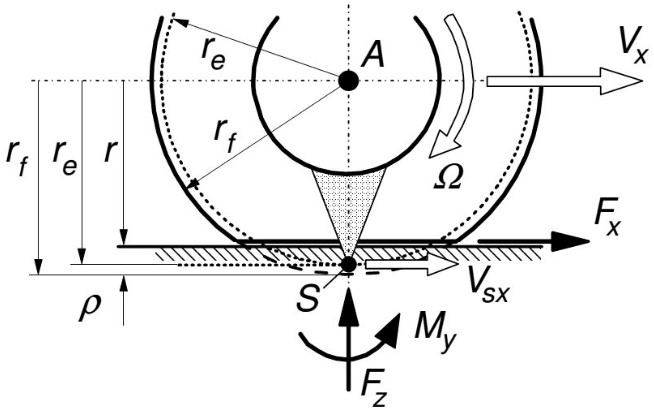


Figure A.1: Longitudinal slip velocity and slip ratio [28].

Longitudinal slip ratio(λ) is defined as the ratio of the longitudinal slip speed(V_{Sx}) to the absolute longitudinal velocity of the wheel(V_x) [23]:

$$\lambda = \frac{V_x - r_e w}{max(V_x, r_e w)} \tag{A.1}$$

where, V_x is the longitudinal velocity of the vehicle (forward velocity),
$V_{sx} = V_x - r_e w$ is the longitudinal slip speed,
r_e is the effective rolling radius of the tyre,
ω is the angular velocity of the tyre (also indicated with Ω).

• In this work, only straight line braking manoeuvres are considered, in which, the forward velocity V_x is greater than the circumferential velocity of the wheel ($r_e w$) and λ takes on a positive value, i.e. $\lambda \in [0, 1]$.

• When the wheel is locked, $r_e w = 0$, $V_{sx} = V_x$ and hence $\lambda = 1$.

• $\lambda = 0$ corresponds to a freely rolling wheel.

A.3 Braked Wheel Dynamics

The dynamics of a braked wheel can be described by a torque balance equation about the wheel centre. The equation of motion about the wheel centre equals:

$$J_w \dot{w} = M_d - M_b.sign(\omega) - r_e F_x \tag{A.2}$$

In the longitudinal direction, the equation of motion becomes:

$$m\dot{v} = F_x \tag{A.3}$$

where J_w is the rotational ineria of the wheel, M_d is the driving torque provided by the engine, M_b is the braking torque applied on the wheel, ω is the angular velocity of the wheel, r_e is the effective rolling radius of the tyre, v is the forward velocity of the vehicle, F_x is the longitudinal force acting on the tyre in the contact patch and F_z is the vertical force acting on the tyre.

To define the longitudinal slip dynamics, Eq. A.1 is differentiated with respect to time and for further analysis V_{sx} is taken as v:

$$\dot{\lambda} = \frac{-r_e \dot{\omega}}{v} + \frac{r_e \omega \dot{v}}{v^2} \tag{A.4}$$

Appendix B

Vector CANcaseXL

As explained in the product manual, the Vector CANcaseXL is a USB interface with a 32 bit 64MHz microcontroller with a ARM7 Core and two SJA1000 CAN controllers from Philips. It can process CAN messages with either 11-bit or 29-bit identifiers. It is capable of receiving and analysing remote frames without any limitations. The CANcaseXL can also detect and generate error frames on the CAN bus. Technical specifications are provided in Fig.B.1 below:

Housing	Robust metal housing
Display	Status display with three LEDs per channel
Channels	2x, independent
Transceiver	Pluggable CANpiggies or LINpiggies
CAN controller	2 Phillips SJA 1000
Microcontroller	ATMEL AT91R40008 32 bit 64 MHz
Max. baud rate	1 Mbit/s
Time stamp accuracy	1 μs
Error frame	Detecting and generating
PC interface	USB 1.1 and 2.0
Ext. Voltage supply (typ.)	7 V … 33 V
Current supply	by USB, external supply optional
Current consumption (without SD card)	90 mA (12 V) for CANcaseXL/log Typ. 30 mA for CANpiggy 251
Memory media	SD card (max. 2 GB)
Configuration	Plug & Play
Dimensions	approx. 105 mm x 85 mm x 32 mm
Weight	approx. 210 g without CANpiggies
Temperature range	Operating: -20...70 °C Shipping and storage: -40...85 °C
Relative humidity of ambient air	15 %...95 %, not condensation

Figure B.1: Technical specifications of the Vector CANcaseXL (From the product manual)

Appendix C

Hydraulic Symbols

Fig.C.1 below depicts the hydraulic symbols, used in diagrams throughout this report, with their conventional description.

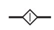
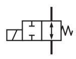
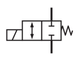
Description	Symbol
Universal hydraulic line	
Hydraulic line junction	
Hydraulic line restriction	
Non-return valve (check valve)	
Hydraulic filter	
2 way - 2 position solenoid valve, normally open	
2 way - 2 position Solenoid Valve, normally closed	
Generic hydraulic accumulator	
Hydraulic pump	

Figure C.1: Description of Hydraulic Symbols [7]

Appendix D

Identification of the Hydraulic Modulator Dynamics

In order to proceed with the design of a low level brake pressure controller, it is useful to have an insight into the dynamics of the hydraulic modulator. A physical MATLAB Simulink simulation model of the hydraulic modulator has already been developed and validated in previous research at IKA [7]. This is utilised to perform black box system identification experiments to identify the dynamics of the hydraulic modulator. The steps in the system identification experiment are depicted in Fig.D.1 below.

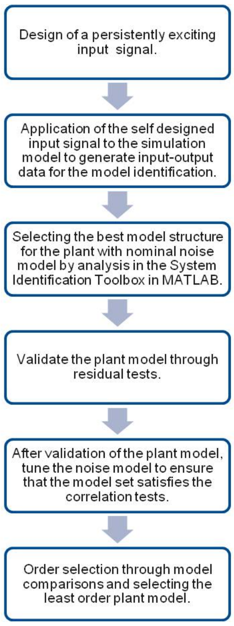


Figure D.1: System identification steps to identify the hydraulic modulator model.

Brief Explanation of the System Identification Method

The *Prediction Error method* of system identification is utilised to estimate models representing the dynamics of the hydraulic modulator. The schematic representation of the system identification setup is depicted in Fig.D.2 [27].

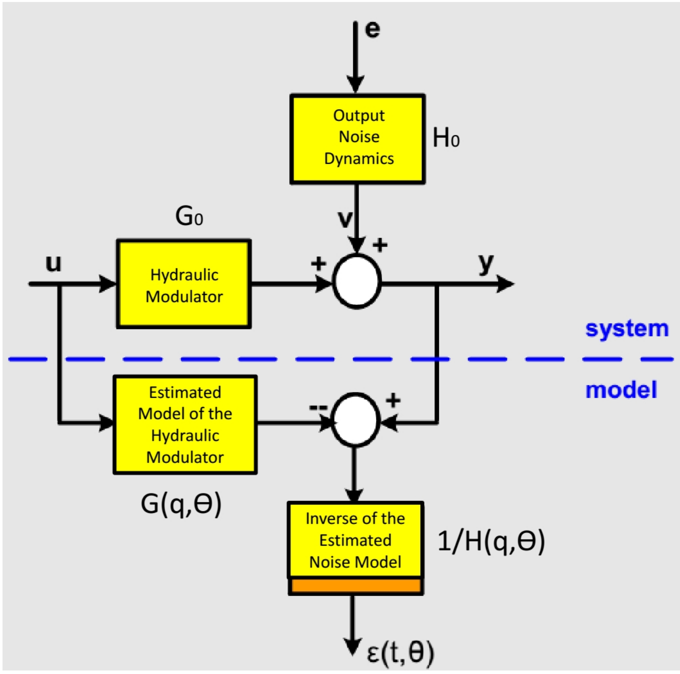


Figure D.2: System identification setup.

In Fig.D.2, $u(t)$ is the input signal to the valve (master cylinder pressure),
$y(t)$ is the output signal (wheel brake cylinder pressure),
$e(t)$ is the output noise/measurement noise,
$v(t)$ is the filtered output noise/measurement noise,
G_o is the Hydraulic Modulator (plant whose model is to be estimated),
H_o is the pre-filter applied to the output noise e.
θ is the vector of estimated model parameters.
$G(\theta)$ is the hypothesized (estimated) model of the Hydraulic Modulator,
$H(\theta)$ is the hypothesized (estimated) model of the output/measurement noise,
$\epsilon(t, \theta)$ is the prediction error.

The goal of the identification procedure is to identify models for the pressure build and release phases of the *Hydraulic Modulator*, that with its output predictions, predicts the measured output data best, given the measurement of the input and output data. Although, the hypothesized models cannot simulate the output data directly, they can predict the future outputs on the basis of past input and output data and therefore these predictions/estimates are used as a basis for identification. The prediction error is given by the following equation:

$$\epsilon(t, \theta) = y(t) - \hat{y}(t|t-1) \tag{D.1}$$

$$\epsilon(t, \theta) = H^{-1}(\theta)[y(t) - G(\theta)u(t)] \tag{D.2}$$

where $G(\theta)$, $H(\theta)$ reflects the hypothesized model,
$y(t)$, $u(t)$ is the output and input data from the data generating system,
$\hat{y}(t|t-1)$ is the one step ahead predictor output and is computed as:

$$\hat{y}(t|t-1) = H^{-1}(\theta)G(\theta)u(t) + [1 - H^{-1}(\theta)]y(t) \tag{D.3}$$

The identification criterion is the power of the prediction error estimated from a data sequence through the quadratic function (V_N):

$$V_N(\theta) = \frac{1}{N}[\sum_{i=1}^{n} \epsilon^2(t,\theta)] \tag{D.4}$$

The parameter estimation ($\hat{\theta_N}$) is then performed through minimising V_N:

$$\hat{\theta_N} = arg \;\; min_\theta(V_N(\theta)) \tag{D.5}$$

Details regarding the prediction error identification theory can be found in [27].

Design of a Persistently Exciting Input Signal

According to *System Identification Theory*, in order to obtain a consistent estimate $G(\theta)$ of the plant model (G_o) i.e. to minimise the prediction error $\epsilon(t,\theta)$, the input signal used to excite the system should satisfy the following characteristics:

• It should be periodic to reduce the variance of the estimated parameters (θ_N).

• It should be *Persistently Exciting* of order n i.e. the power spectral density of the input signal $\phi_u(w)$ has to be non-zero in n points in the interval $(-\pi,\pi]$ and $n > n_a + n_b$ (where n_a is the order of the denominator and n_b is the order of the numerator of the model structure of $G(\theta)$).

• The magnitude of the input signal should be bounded to avoid the excitation of high frequency non-linearities but at the same time, the power of the input signal should be maximum to reduce the variance of the parameter estimates.

Four types of input signals for identification were analysed and their characteristics are stated below:

White Noise Signal

• Has a flat frequency spectrum i.e. contains all frequencies uniformly.

• Provides uniform estimation at all frequencies but has a high crest factor (ratio of peak to RMS value of the signal).

Multisine Signal

• It is a combination of Sinusoids of different frequencies.

• Provides very good estimates of the transfer functions at only specific frequencies corresponding to the frequencies of the Sinusoids.

• But as the estimates at other frequencies are not available, the spectrum is discontinuous.

Random Binary Sequence (RBS) Signal

• A random binary sequence is generated by the following equation:

$$u(t) = c.sign[w(t)] \tag{D.6}$$

where $w(t)$ is a white stochastic process and c is a constant used in order to bound the magnitude.

• Although RBS has a low crest factor as desired, it does not offer proper control over the spectrum as the 'sign' operation distorts the spectrum of the input sequence.

Pseudo Random Binary Sequence (PRBS) Signal

• It has the lowest crest factor among all signals considered and possesses properties similar to white noise.

• Its frequency content can be varied by varying the clock sampling rate.

Thus, all the prerequisites of an input signal for identification are satisfied by a *Pseudo Random Binary Sequence* signal, which has maximal signal power under amplitude constraints. For system identification of the hydraulic modulator, a low frequency PRBS signal, depicted in Fig.D.3, is used to change the state of the valves. The valve opening and closing delays are known from previous experimentation and have also been considered in the design of the input signal for system identification. This PRBS input signal is used to excite the inlet and outlet valves asynchronously.

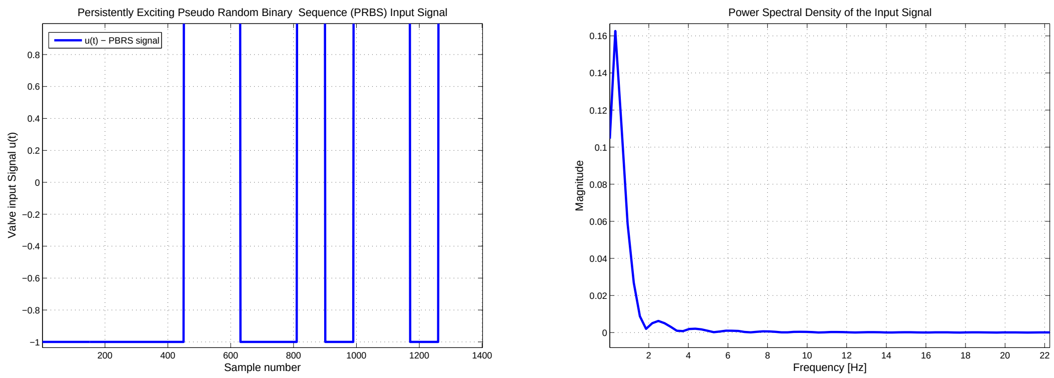


Figure D.3: The designed PRBS signal in the time domain (left) and in the frequency domain (right).

Generating Input-Output Data for System Identification

As mentioned earlier, the MATLAB Simulink model developed in previous research [7] is used to perform the black box system identification. The input signal designed in the previous sub-section above, is applied asynchronously to the inlet and outlet valve of the rear left wheel brake cylinder, in the Simulink model of the hydraulic modulator, to generate wheel brake cylinder pressure data (i.e. output data) for System Identification.

Fig.D.4 depicts the plots of rear wheel brake cylinder pressure, inlet valve displacement and outlet valve displacement with time, when a finite length PRBS signal is applied to the inlet and outlet valves. In case of the inlet valve, 0 mm displacement corresponds to the valve being completely open and 0.29 mm corresponds to the valve being completely closed. The negative sign on the y-axis is due to the Simulink model sign convention adopted. In case of the outlet valve, 0 mm displacement corresponds to the valve being completely closed and 0.29 mm corresponds to the valve being completely open. The master cylinder pressure is held constant at 160 bar and the PRBS signal is applied to the inlet and outlet valves asynchronously. As expected, in the time window where the inlet valve is closed and the outlet valve is open, the pressure in the wheel brake cylinder decreases and vice versa. The input data and output data i.e. the wheel brake cylinder pressure is then imported to the MATLAB System Identification toolbox to identify a parametric model with the least residual error.

As mentioned earlier, the goal of the system identification procedure is only to get some insight into dynamics of the hydraulic modulator. Two sets of input-output data generated from the system identification experiments on the simulation model are utilised, one for model identification and the other for model validation. Several models with different model structures were estimated iteratively using the MATLAB Simulink System Identification toolbox in order to compute a model which has minimal order, high accuracy and which satisfies the model residual tests of system identification [27]. The first and second order transfer function models demonstrated reasonably high accuracy as compared to other model structures with a level of fit percentage to validation data of approximately 80%. As seen from Fig.D.4, the dynamics of the hydraulic modulator closely resemble that of a first order system with delay. This observation provides a useful insight into the dynamics of the hydraulic modulator.

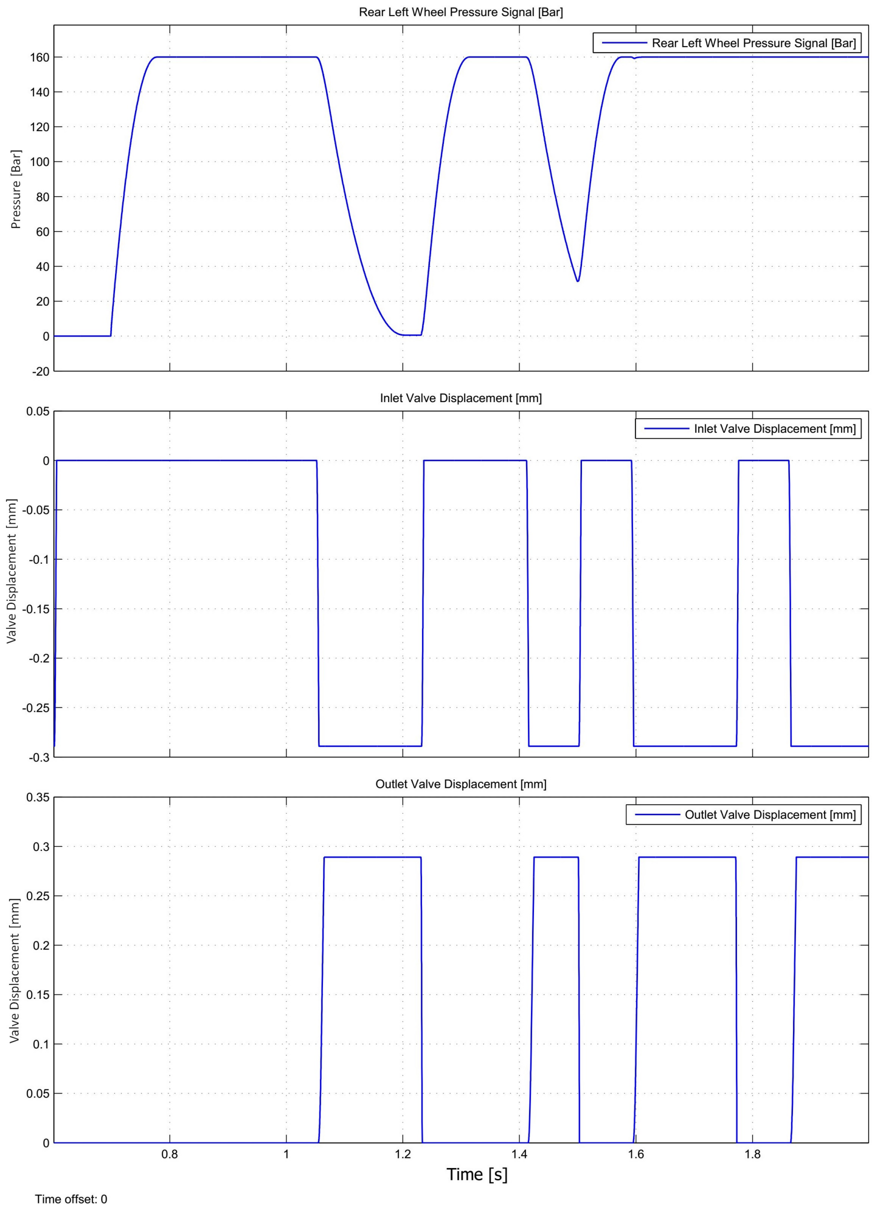


Figure D.4: Effect of the inlet/outlet valve displacement on the wheel brake cylinder pressure.

Appendix E

Simulation Results with Highly Dynamic Reference Pressure Signals

In this research work, the low level brake pressure controller has been tested only with staircase type reference pressure signals. In this chapter, simulation results with highly dynamic signals such as a sine wave and ramp signals with varying slopes have been presented. Due to hardware issues with the braking system test bench, HiL tests with these signals could not be performed. Fig.E.1 depicts the closed loop tracking performance of the threshold based low level brake pressure

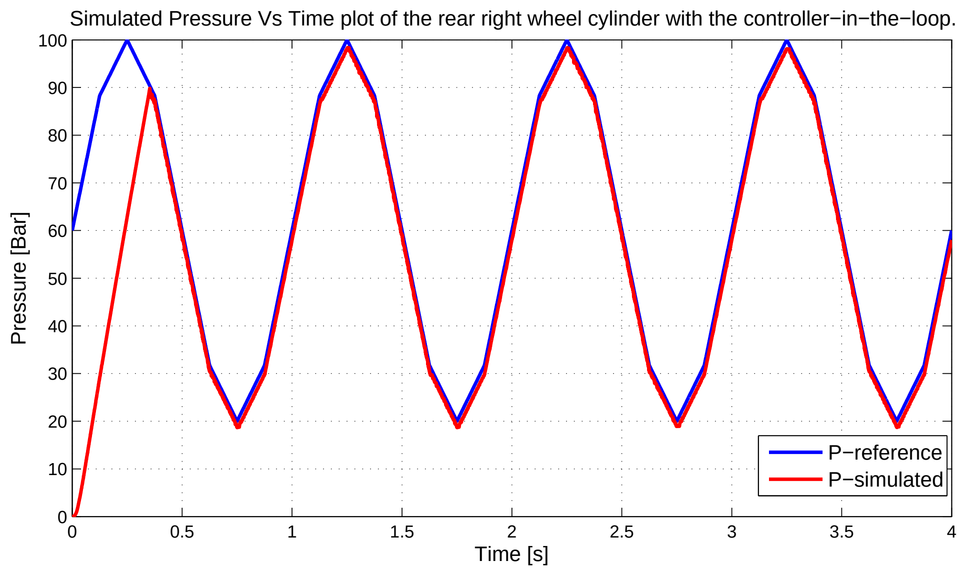


Figure E.1: Tracking performance of the low level brake pressure controller with a sinusoidal reference pressure signal.

controller with a sinusoidal reference pressure signal. This demonstrates the tracking performance of the controller for a highly dynamic reference pressure signal. It is observed that the overall tracking performance is satisfactory but, close to the crest and the trough of the sine wave the deviation of the simulated pressure trajectory from the reference pressure trajectory is relatively high.

Fig.E.2 depicts the tracking performance of the threshold based low level brake pressure controller with a reference pressure signal comprising of linear ramps. It is observed that the overall tracking performance with such a reference pressure signal is acceptable.

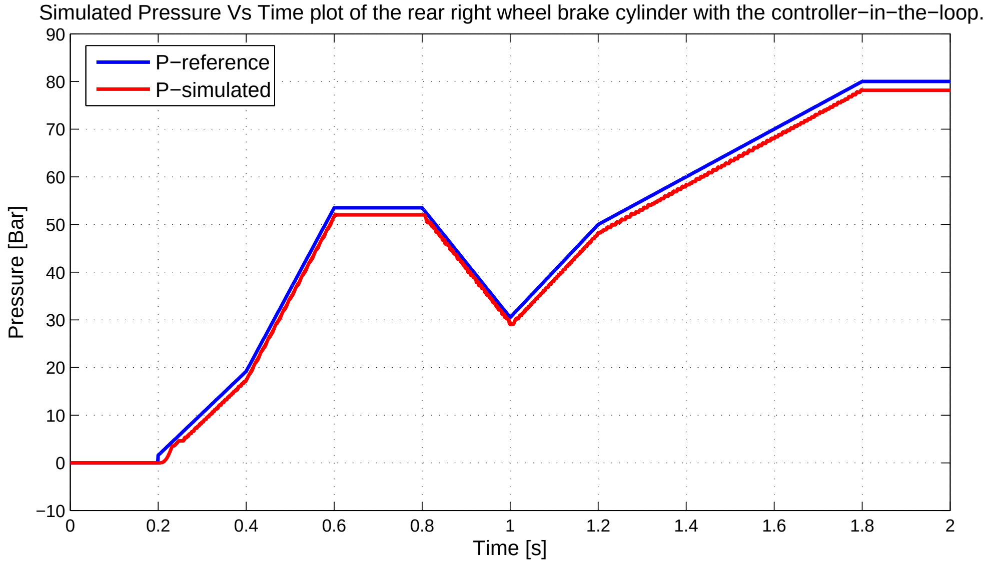


Figure E.2: Tracking performance of the low level brake pressure controller with a ramp type reference pressure signals.

The closed loop simulation results with highly dynamic reference signals, show that further HiL tests should be performed with such dynamic reference signals to verify the accuracy of the reference pressure tracking before its further implementation.