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Design and implementation of a novel force and brake pressure based Anti-lock Braking System

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THE UNDERSIGNED HEREBY CERTIFY THAT THEY HAVE READ AND
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Design and implementation of a novel force and brake
pressure based Anti-lock Braking System

BY S.M.A.A. KERST

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Abstract

Road vehicles are nowadays equipped with numerous systems to improve safety, of which the Anti-lock Braking System (ABS) is one of the most, if not the most, important. The system ensures that during heavy braking both longitudinal and lateral friction are optimal, such that braking distance is minimized and steerability maintained.

Current commercial ABS systems are based on wheel deceleration control with a large set of heuristic control rules, which has led to complex systems that are, despite their good performance, hard to assess with respect to stability and performance. In opposition to that, in literature promising algorithms have been presented, based on clear mathematical background, that lack practical implementation. In this thesis, groundwork is performed to break with this latter.

A partnership between SKF and TU Delft has resulted in the acquisition of a BMW 5 series car, that during this thesis has been prepared for ABS field testing. The vehicle’s hydraulic circuit has been modified, and at this point last hand is laid on the implementation of state of the art force sensing bearings.

The vehicle preparation is an important part of the project, but the thesis also has a theoretical focus. The thesis describes the modeling and identification of the test vehicle’s hydraulic circuit, which has led to a highly realistic representation of the BMW’s hydraulic circuit dynamics. Based on this model, a brake pressure controller has been designed and optimized. This controller’s use is twofold, as on one hand it will be used for pressure control on the test vehicle, while on the other hand, in combination with the hydraulic circuit model, it is very useful for ABS algorithm simulation.

Furthermore a novel ABS algorithm is presented that makes full use of the force measuring capabilities of the vehicle. By using an unique combination of force measurement and brake pressure control, a new method to control slip is found. Due to the use of this new method, traditional ABS control difficulties as load transfer and road friction changes are overcome. The algorithm is designed such that it takes in account the capabilities and limitations of the test vehicle, and thus practical implementation is feasible.
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Chapter 1

Introduction

Modern cars are equipped with several systems to increase safety, which mainly can be divided in *active* and *passive safety systems*. Active safety systems as ESC (Electronics Stability Control) and ABS (Anti-lock Braking System) assist in prevention of a crash, where passive safety systems as for instance seat belts, airbags and the crumple zone try to protect occupants during a crash.

The Anti-lock braking system is one of the most important active safety system in road vehicles nowadays. On one hand it minimizes the braking distance, while on the other hand it also offers sufficient lateral friction to maintain steerability during heavy braking. This all is done by taking over control of braking pressure from the driver.

Figure 1.1: Early Anti lock braking tests (1973) by Bosch which resulted in the first series produced anti-lock braking system in 1978. Nowadays Bosch dominates the market with a share of almost 80% [5]. Picture from [1]

The first developments of anti lock braking date from about 100 years ago, when early developments began for trains, which was later followed by de-
sign and implementation of ABS systems on aircrafts in the '40s and '50s. Although the first road vehicle ABS systems were introduced in the early '60, it was much later, in 1978, that the modern era of ABS systems began with the introduction of the anti-blockier system (from which the actual name ABS arises) by Bosch. Development and implementation has come a long way since then, and currently ABS systems are already standard in all new passenger cars in the EU, the US and Japan [4],[2].

Current ABS systems found in road vehicles are based on wheel deceleration control with a large set of heuristic control rules, which have led to complex systems that are hard to assess with respect to stability and performance. And although the ABS has now been around for over 30 years, still improvements can be made [2]. They can be made simpler and could be improved with respect to braking performance and robustness to differing circumstances. Furthermore, new technological developments as electro mechanical braking and force sensing bearings offer new opportunities.

1.1 Motivation

The current commercial ABS systems show good results in practice whilst being based on large sets of heuristic control rules, which make them complex and hard to assess with respect to stability and performance. On the other hand in literature several promising algorithms, based on clear mathematical background, are proposed which lack practical implementation.

Algorithms such as for instance the 5-phase algorithm [9] and hybrid force based algorithm [2], show promising results in simulation both with respect to stability and performance. A partnership between the TU Delft and SKF has led to an automotive project in which preparations are made for field testing of those novel algorithms.

Figure 1.2: The BMW 5 series car, which is acquired by SKF and the TU Delft to, in future, perform novel ABS algorithm field tests
Besides the focus on the preparation for field testing, also novel SKF force sensing bearings will be implemented on the vehicle. These innovative sensors offer interesting new opportunities with respect to ABS control, as they offer new possibilities to tackle traditional problems regarding ABS systems. As the research field with respect to force measurements in ABS systems is quite new, only two algorithms have been published so far [2] [3], there is reason to believe that there are still numerous open possibilities.

1.2 Force sensing

The research field with respect to force measurements in ABS systems is reasonably new, as methods to acquire wheel forces are limited and still in development. At this moment there are two methods that have reached actual prototyping, which are an indirect measurement method using smart-tires [11], and a direct measuring method using force sensing bearings. In this project focus lies on the latter, as these will be installed on the BMW test vehicle by project partner SKF.

The force sensing bearings developed by SKF offer the ability to measure forces in all directions ($x$, $y$ and $z$), by measuring the deformations of the bearing due to loading. Accuracy appears to be good, as a RMS error of less than 2% and absolute maximal error of 10% is claimed.

![Figure 1.3: (a) and (b) show a schematic representation of the basic idea of measuring deformations. (c) shows an actual SKF load sensing bearing.](image)

1.2.1 Algorithms

Based on the prospect of wheel force measurements, researchers at the TU Delft have been working on methods to use this new sensory information to create better algorithms with respect to simplicity, braking distance and robustness to different circumstances. At this point this has led to two interesting published algorithms, which will be briefly discussed next.
Hybrid ABS Control Using Force Measurement [2]  This two phased algorithm, of which phase 1 ensures a decrease of slip and phase 2 an increase by the closed loop control of wheel deceleration, uses force measurements to detect whether slip is evolving towards or away from the friction peak. If the latter is the case, a phase switch is triggered as shown in figure 1.4. Figure 1.5 shows a more detailed image and explanation of the algorithm’s control logic.

![Figure 1.4](image1.png)

**Figure 1.4:** General trajectory of the algorithm visualized on the friction curve

![Figure 1.5](image2.png)

**Figure 1.5:** "In each phase, the acceleration is controlled in closed-loop at the desired level \( \dot{\omega}_{\text{ref}} \). The switching is based on force measurement, with \( F_n \) the normalized braking force, \( \hat{F}_n \) the local maximum over time of \( F_n \) and \( dF_n^+ \) and \( dF_n^- \) two tuning parameters. The inequalities at the beginning of the arrows indicate the switching condition; the equalities at the end indicate the reset assignments.” [2]

Although the simple layout of the algorithm, it is robust to both friction coefficient changes and load transfer phenomena. Furthermore results obtained with respect to braking performance are promising.

**On the performance increase of wheel deceleration control through force sensing** [3]  This algorithm, based on wheel deceleration, force measurement and brake torque control, uses the sensory information for a differ-
ent purpose than the algorithm presented previously. In this three phased control logic, force measurements are used to optimize the braking torque, whilst information of wheel deceleration is used to determine friction peak passing. Figure 1.6(a) shows the intended evaluation of torque over slip of the algorithm, where in 1.6(b) the three phased logic is presented.

Figure 1.6: (a) the intended evolution braking torque and slip and (b) state-flow diagram of the controller

The algorithm is shown to be robust to friction coefficient and load changes. However the assumption that brake torque can be controlled is doubtful.

1.2.2 Use of force measurements

Both of the previous ABS algorithms discussed used wheel force information as essential input for their controller. However, both controllers have a different purpose to the measurement of forces, as in the first algorithm discussed it was used for friction peak passing detection, whilst in the second algorithm it was used to optimize brake torque applied to the wheels. The fact that both algorithms found a different method to improve their control with respect to performance, stability and/or robustness by the use of force measurements feeds the thought that not all possibilities and combinations have been investigated yet, and that still more improvements can be made.

1.3 Research objectives

The research of this thesis covers a broad spectrum, running through the entire feedback loop of ABS control. Furthermore it comprises of both a practical and theoretical contribution.

On the practical side, the SKF/TU Delft test vehicle needs to be prepared to for field testing, which means that modifications to the vehicle need to be made, electronics need to be installed and tested, and software needs to
be written.

Furthermore the theoretical contributions includes the modeling and identification of the test vehicle’s hydraulic circuit using experiments performed after modification. Using this model a brake pressure controller will be synthesized and evaluated. Last but not least a force based algorithm will be designed that uses force information for both triggering and brake pressure control, thus making use of the full capabilities of force sensing. The design of the algorithm will be such that it could be implemented on the test vehicle when force sensing bearings are installed.

The following list sums up the contributions made and explains why these contributions are worth-full and/or innovative:

- The modification and preparation of the SKF/TU Delft test vehicle for ABS field testing is the main aim of the overall project. It will provide a useful test base to test algorithms, and will offer first hand experience and knowledge on the subject.

- The hydraulic circuit is modeled and identified in order to obtain a better insight in the behavior of the braking circuit and create a basis for brake controller design. Furthermore the highly realistic model obtained can be of high value for ABS algorithm simulation, as such a detailed validated model is currently not available in open literature.

- A brake pressure controller is designed and evaluated for both simulation and test vehicle, which is a useful asset for the test vehicle, but also a valuable part for ABS algorithm simulation, as current models usually rely on continuous actuation.

- The design and evaluation of a novel force based ABS algorithm which utilizes an innovative method to control wheel slip. Besides approaching the problem from a new point of view, which results in interesting advantages, also limitations of the test vehicle are taken in account, such that implementation is feasible.

The general aim of the thesis is to take vehicle dynamics research at the TU Delft a step further, such that field testing can be performed, and thus novel ABS algorithms, as designed in this thesis, can be evaluated on track.
1.4 Thesis outline

During the thesis both practical and theoretical work has been done, which in total overlap the entire feedback loop ABS control as shown in figure 1.7.

Figure 1.7: The control loop of ABS control

In chapter 2 the basic modeling of the vehicle dynamics and (force) measurements is described. Although no new theory is presented, this chapter is added as they are fundamental for simulation.

The chapter on modeling will be followed by three chapters related to actuation. First chapter 3 will describe the detailed modeling of the BMW’s hydraulic circuit. This is followed by the discussion of an identification method and actual identification of this hydraulic circuit model in chapter 4. The final chapter on actuation will discuss the design of a brake pressure controller for the BMW using the identified hydraulic circuit model.

Chapter 6 of the thesis will present a novel force based ABS algorithm, which is designed to work on the future test car. The chapter will explain a novel method on controlling the wheel slip, present the controller based on this, and discuss other aspects as for instance stability and robustness to changing circumstances.

The final chapter will give an overall conclusion on the work done, and present recommendations for future research.

Last but not least, details on the vehicle modification are summarized in appendix A.
Chapter 2

Modeling

To obtain insight in the quality of newly designed ABS algorithms, usually mathematical representations are used to evaluate the system dynamics. Different models exist for each part of the system dynamics (e.g. tire-wheel contact, vehicle representation and brake system) with varying realism and complexity. In this chapter the modeling of several basic aspect will be discussed. The brake system will not be part of this, as a new realistic brake circuit model will be presented in the following chapter.

In the section first to follow the tire-road contact will be discussed. Afterwards the single corner model will be discussed, which represents wheel and vehicle dynamics. The last section will focus on the addition of noise to the force signals, to improve simulation realism.

2.1 Tire-road contact forces

The tire-road contact forces can be decomposed in three orthogonal force vectors, $F_z$, $F_x$ and $F_y$, as shown in figure 2.1. $F_z$ is dependent on the load on the tire, where $F_x$ and $F_y$ are both dependent on several different factors as for instance road surface, tire pressure, vehicle load, longitudinal wheel slip, slip angle and camber angle. Longitudinal force $F_x$ allows the vehicle to brake, whilst lateral force $F_y$ ensures steerability.
During the study of ABS systems usually straight line braking is assumed, which simplifies modeling significantly without degrading quality of the representation. In straight line braking $F_y$ is logically assumed to be 0, whilst $F_x$ can be described by:

$$F_x = F_z \mu_x(\lambda)$$

(2.1)

where $\mu_x$ is a friction function dependent on longitudinal slip $\lambda$. This latter, longitudinal slip, is the normalized difference between vehicle velocity and the wheel linear velocity:

$$\lambda = \frac{v - \omega r}{v}$$

(2.2)

where $v [\text{m/s}]$ is the vehicle velocity, $\omega [\text{rad/s}]$ is the wheel speed and $r_{\text{w}} [\text{m}]$ is the wheel’s radius.

The friction function, $\mu_x(\lambda)$, is dependent on the contact characteristics between tire and road, which (logically) depends on the road surface, and can be described using several different models. In this study it is chosen to use the frequently employed burckhardt model, as its formulation is simple and it describes the $\mu - \lambda$ relation in a sufficient degree [7]. The burckhardt model can be described as:

$$\mu_x(\lambda, C_i) = (C_1(1 - e^{-C_2|\lambda|}) - C_3|\lambda|)e^{-C_4v}$$

(2.3)

where $C_i, i = 1, \ldots, 4$ are constants which determine the type of road surface, which can be obtained from literature. Figure 2.2 shows the trajectory of the friction - slip curve for several different road surfaces.
Although in the friction model slip and longitudinal friction are statically mapped, in reality this is not exactly true. When for instance slip changes stepwise, the longitudinal friction will undergo a transient leading to the new steady-state condition. Modeling of this phenomena is done by the so called relaxation dynamics, which can be described as:

\[
\dot{F}_{x\text{Act}} = \frac{1}{\tau}(F_x - F_{x\text{Act}}) \tag{2.4}
\]

where \(F_{\text{Act}}[N]\) is the acting friction force, \(F_x[N]\) is the force computed using a static friction model and \(\tau[s]\) is a variable and given by \(\tau = \frac{s_0}{\omega_r}\), in which \(s_0[l]\) is the relaxation length. Note that wheel speed is of influence in \(\tau\).

### 2.2 The single-corner model

The vehicle braking dynamics will be modeled using the single corner model. Although it’s simplicity, it still provides a sufficient description of the braking dynamics and it is thus very useful for design and early testing [10]. The model is based on the dynamics of a single wheel and quarter of the vehicles mass. Figure 6.1 shows the free body diagram on which the model is based, which is described by the following set of equations:

\[
J\ddot{\omega} = r_w F_x - T_b \tag{2.5}
\]

\[
m\ddot{v} = -F_x \tag{2.6}
\]

Where

- \(\omega[rad/s^2]\) = angular wheel acceleration
Figure 2.3: The single corner model

- $v[m/s^2]$ = longitudinal acceleration of the vehicle
- $T_b[Nm]$ = braking torque applied by the brakes
- $F_x[N]$ = longitudinal tire-road contact force
- $J[Kgm^2]$ = wheels moment of inertia
- $m[Kg]$ = quarter of the car’s mass
- $r_w[m]$ = wheels radius

2.3 Measurement of wheel forces

As discussed during the introduction, at this moment methods to obtain information with respect to wheel forces are brief and still in development. In this project focus lies on force measurements by the force sensing bearings developed by SKF, which claim to be able to measure wheel forces with an RMS error of less than 2% and absolute maximum error of 10%.

As the error levels are significant, and it is unsure what the error characteristics are, it is chosen to do a first estimate with respect to distortion of the signal by the addition of noise. It is chosen to add, to the original signal, a white noise with an RMS error of 2% with an absolute maximum of 10%. Furthermore, although no claims with respect to sampling frequency were made, it is chosen to reduce sampling frequency to 1 kHz. Figure 2.4 shows an example of the new force signal, in comparison to the original signal that doesn’t contain noise.
2.4 Conclusions

In this chapter several basic dynamics have been discussed, that will be used for testing of the novel force based ABS algorithm in chapter 6. It is chosen to describe wheel and vehicle dynamics by the single corner model, and use the Burckhardt static friction model in combination with wheel relaxations for tire-road friction description. Furthermore noise has been added to the wheel force signals which will be used by the algorithm, in order to increase simulation realism.
Chapter 3

Hydraulic circuit modeling

This chapter is dedicated to the modeling of the hydraulic circuit of an ABS and ESC equipped road vehicle. The layout of a BMW E60s hydraulic circuit is used as basis for the model design. This particular hydraulic circuit is a Hydraulic Actuated Braking (HAB) type circuit, which can be found in practically every commercial road vehicle.

In this chapter, first the Hydraulic Actuated Braking circuit will be discussed, and the operation of it will be explained. Then the modeling of all relevant components of the hydraulic system will be described. This is followed by the explanation of the complete Simulink model, and the chapter is concluded by a list of parameters which define the model and a brief conclusion of the work done. Identification of the model parameters, to create a good fit to reality, is performed in chapter 4.

3.1 Hydraulic Actuated Braking circuit

The main characteristic of the HAB type of braking system is that the driver’s applied force on the brake lever is directly transmitted to the brakes by a hydraulic circuit. Safety regulations require each HAB system to have two separate brake circuits, which often results in either a X or II configuration as shown in figure 3.1

![Figure 3.1: (a) II configuration (b) X configuration](image)

The BMW is equipped with the II type configuration, including a hydraulic
unit within the circuit for ABS and ESC actuation. In figure 3.2, the BMW’s front subsystem including the hydraulic unit is schematically represented. As the front and rear brake circuits have practically the same layout, it is unnecessary to discuss both, and thus only the the front subsystem will be discussed.

Figure 3.2: Overview of the front subsystem of the BMW’s hydraulic circuit

Looking at the figure, one can see the basic components of any HAB system, namely the brake lever, master cylinder, brake cylinders and the pipelines connecting them. Besides that, as mentioned before, the BMW’s hydraulic circuit also contains a hydraulic unit for ABS and ESC actuation, which consists of several valves, a pump and an accumulator. During normal operation none of the components of the hydraulic unit are energized or activated, and therefore brake fluid can move freely from master cylinder to brake cylinders. In this way the driver is in direct control of the brake pressure on both wheels.

When the ABS system notices that wheel lock is imminent, it uses the hydraulic unit to take over control of the brake pressures, in order to avoid wheel lock. By the use of the hydraulic unit, the ABS system is able to control the brake pressure in each brake line\(^1\), and thus brake, separately. For each brake (line) it has three brake pressure control actions by controlling the hold and release valves as shown in table 3.1 and discussed in the

\(^1\)In this report, the combination of pipeline and brake cylinder is called 'brake line'
successive list:

<table>
<thead>
<tr>
<th>Action</th>
<th>Hold Pressure valve</th>
<th>Release Pressure valve</th>
</tr>
</thead>
<tbody>
<tr>
<td>Hold pressure</td>
<td>Closed (energized)</td>
<td>Closed (de-energized)</td>
</tr>
<tr>
<td>Build pressure</td>
<td>Open (de-energized)</td>
<td>Closed (de-energized)</td>
</tr>
<tr>
<td>Dump pressure</td>
<td>Closed (energized)</td>
<td>Open (energized)</td>
</tr>
</tbody>
</table>

Table 3.1: Different states of the hydraulic unit

**Hold pressure**
By energizing the hold valve only, the pipeline and brake cylinder are ‘disconnected’ from the rest of the circuit, resulting in a constant pressure on the brake.

**Build pressure**
If none of the valves is energized, brake pressure will increase as the pipeline and brake cylinder are connected to the (high pressure) main chamber. The maximum pressure which can be obtained is logically the driver’s applied pressure. Although during ESC control pressure can be build by the pump, this is not applicable for ABS control as pressure increases too slow. Furthermore the necessary MBD and OS valves are not in use during conventional ABS control.

**Dump pressure**
By energizing both the hold and release valves, brake pressure will decrease as the pipeline and brake cylinder are disconnected from the (high pressure) main chamber, but connected to the (low pressure) accumulator.

As during each dump action the low pressure accumulator will fill, it needs to be emptied regularly to maintain good pressure decreasing performance. This is done by the pump within the hydraulic unit.

## 3.2 Modeling

In order to model the hydraulic circuit, the system has first been split into separate components. For each component an appropriate representation or model has been deducted based on literature [12],[13], [8], mathematical deductions and common sense. Afterwards all separate models have been combined towards one complete hydraulic circuit model. The following subsections will discuss the modeling of all individual components visible in figure 3.2, which are:

- Pipelines (section 3.2.1)
3.2.1 Pipelines

The pipelines within the circuit connect important parts of the hydraulic system. The modeling of it is based on three important characteristics:

- Fluid storage - The pipeline acts as a storage reservoir for brake fluid
- Inertia - The brake fluid within the line has a mass, and thus inertia
- Friction - Fluid flow is slowed down by friction

The three characteristics can be modeled by a hydraulic capacity, inductance and resistance respectively. In the following this modeling is explained:

**Hydraulic Capacity** Modeling fluid storage as a hydraulic capacity is quite common throughout literature [8], [13], and often the following equation is assumed:

\[ Q = \frac{V_0}{E} \cdot \dot{p} \]  

(3.1)

where

- \( Q [m^3/s] \) = the volumetric flow rate
- \( V_0 [m^3] \) = the initial volume
- \( E [N/m^2] \) = the bulk modulus of elasticity of the brake fluid
- \( \dot{p} [N/m^2s] \) = the pressure derivative

The equation states a proportional relation between fluid flow \( Q \) and pressure change \( \dot{p} \). This proportional relation \( \frac{V_0}{E} \) can be described with one factor, the hydraulic capacity:

\[ C_h = \frac{V_0}{E} \]  

(3.2)

The hydraulic capacity is far from constant, as the bulk modulus, which describes the oil’s elastic properties (and thus relation between pressure and
strain), is dependent on several factors as pressure, temperature and relative amount of trapped air. Furthermore the hydraulic capacity is also influenced by the elasticity of the container or pipeline. This phenomenon is often accounted for by introducing $E'$, as a replacement for the pure bulk modulus $E$, which then also takes in account the container or pipeline elasticity.

In the modeling of the hydraulic circuit of the BMW it is chosen to solely work with the hydraulic capacity factor $C_h$, and not to use $\frac{V_0}{E}$, as in the current setup it is practically impossible to determine the initial volume $V_0$ of the pipelines. The factor $C_h$ is represented by a fifth order pressure dependent polynomial as shown in equation 3.4. The values $c_1..5$ of the polynomial will be determined by identification experiments. The pipeline’s hydraulic capacity is thus represented by:

$$\dot{\rho} = \frac{Q}{C_h(p)}$$

(3.3)

where

$$C_h(p) = c_1 \cdot p^4 + c_2 \cdot p^3 + c_3 \cdot p^2 + c_4 \cdot p + c_5$$

(3.4)

Hydraulic inductance  Because the fluid has a considerable mass, acting forces will not instantaneous cause fluid movement. It will be, as any object with mass, accelerated according to Newton’s second law of physics, $F = m \cdot a$. Figure 3.3 shows how a volume element of fluid can be treated using Newton’s law.

![Figure 3.3: Forces on a volume-element in a pipeline](image)

The volume in the figure has a mass corresponding to $\rho \cdot V$, on which forces act from both left and right side. This can be described as:

$$A \cdot (p_1 - p_2) = \rho \cdot V \cdot \ddot{x}$$

(3.5)

substituting $\frac{Q}{A}$ for $\ddot{x}$ and $A \cdot l$ for $V$, one obtains:
\[ \dot{Q} = \frac{A \cdot (p_1 - p_2)}{\rho \cdot l} \]  

(3.6)

The factor \( \frac{\rho \cdot l}{A} \), which consists only of constants, can be seen as the combined hydraulic inductance constant \( L_h \), and thus one obtains the following description for the hydraulic inductance of the pipeline:

\[ \dot{Q} = \frac{(p_1 - p_2)}{L_h} \]  

(3.7)

**Hydraulic resistance**

When a fluid flow is present, the resistance in the pipeline will cause a pressure drop over the line, which in its turn will slow down the fluid flow. Assuming that the diameter of the pipeline doesn’t change, the following equation describes the pressure drop [13]:

\[ \Delta p_{\text{drop}} = \zeta \cdot \frac{\rho \cdot Q^2}{A^2} \]  

(3.8)

where \( \zeta \) is the dimensionless resistance characteristic. This characteristic is dependent on the Reynolds number and the diameter and length of the pipeline and can be described as follows:

\[ \zeta = \frac{c_1}{Re^{c_2}} \cdot \frac{l}{d} \]  

(3.9)

where \( c_1 \) and \( c_2 \) are constants and depend on the type of fluid flow. With the assumption that fluid flow within the system is laminar and isothermic, the values for \( c_1 \) and \( c_2 \) are 64 and 1 respectively. Furthermore the Reynolds number can be calculated by:

\[ Re = \frac{Q \cdot d}{A \cdot \nu} \]  

(3.10)

Combining eq. 3.8, 3.9 and 3.10 results in the following pressure drop due to pipeline resistance:

\[ \Delta p_{\text{drop}} = \frac{128 \cdot \rho \cdot \nu \cdot l}{d^4 \cdot \pi} \cdot Q \]  

(3.11)

As on the one hand it is fairly difficult to determine all the different individual parameters, and on the other hand all parameters are constant (neglecting temperature influences), it is chosen to create one hydraulic resistance factor \( R_h \) for each pipeline:

\[ \Delta p_{\text{drop}} = R_h \cdot Q \]  

(3.12)
Final pipeline model  The capacitance, inductance and resistance model equations are combined to resemble a single pipeline element. Within this element, in literature often called RLC element, the capacitance represents fluid storage, while inductance and resistance model the fluid inflow or outflow. The model’s wave propagation capability is improved by cascading several RLC elements as representation for one pipeline. Each RLC element can be described by:

\[ \dot{p} = \frac{Q}{C_h(p)} \]  
\[ \dot{Q} = \frac{(p_1 - p_2) - R_h \cdot Q}{L_h} \]  

3.2.2 Solenoid valves

The solenoid valves within the hydraulic unit offer the possibility to connect or disconnect different parts of the hydraulic circuit from each other. Whilst opening or when completely open, fluid will flow through the valve. When incompressible flow is assumed, this fluid flow, in literature, is usually modeled as:

\[ Q = \alpha \cdot A \cdot \sqrt{\frac{2}{\rho}} \cdot \sqrt{p_1 - p_2} \]  

where

- \( Q \) [m\(^3\)/s] = the volumetric flow rate through the valve
- \( \alpha \) = the (dimensionless) discharge coefficient
- \( A \) [m\(^2\)] = the valve’s opening surface
- \( \rho \) [kg/m\(^3\)] = the brake fluid’s density
- \( p_1 \) and \( p_2 \) [N/m\(^2\)] = the pressure on either side of the valve

The equation is basically a rewritten form of the Bernoulli equation in combination with a discharge coefficient \( \alpha \), which is a dimensionless number within the range of 0.6 to 1, that resembles flow decrease due to friction. As the current setup doesn’t allow the individual values to be determined, it is chosen to combine the constants \( \alpha \), \( A \) and \( \sqrt{2/\rho} \) to a single parameter \( Q_{nom} \) and multiply this by the percentage of opening of the solenoid valve, \( y \). This results in the following equation:

\[ Q = y \cdot Q_{nom} \cdot \sqrt{p_1 - p_2} \]
Dependant on the type of valve it is either completely open (MBD and hold valves) or closed (OS and release valves) in normal operation. In order to realistically model opening and closing, the valve is modeled as a mass-spring-damper system. In normal operation, dependent on the type of valve, the spring keeps it either open or closed. When the coil is energized, it will generate a force which will either close or open the valve. As the valve movement is normalized, it will range between 0 and 1. The following equation resembles the valve’s dynamics:

\[ m \cdot \ddot{y} = -F_{spring} + F_{coil} \cdot u + \begin{cases} -k \cdot y - c \cdot \dot{y} & \text{if } y < 0 \\ -k \cdot (y - 1) - c \cdot \dot{y} & \text{if } y > 1 \end{cases} \]  

(3.17)

where

- \( x[m] \) = is the valve’s position
- \( m[kg] \) = the valve’s moving mass
- \( F_{spring}[N] \) = the force applied by the spring (assumed constant)
- \( F_{coil}[N] \) = the force applied when coil is powered
- \( u \) = the control signal (on/off) (with time delay)
- \( k[N/m] \) = the wall stiffness
- \( c[N/ms] \) = the wall’s damping

### 3.2.3 Brake cylinder

The brake cylinder is a fluid reservoir whose volume may slightly change due to piston movement. As it stores fluid, it is represented as hydraulic capacitance, as described earlier in the brake line modeling. The piston movement, and thus volume change, is not explicitly modeled, but will be implicitly taken in account by the capacitance function. The brake cylinder model can be described as:

\[ \dot{p}_{cyl} = C_h(p) \cdot Q \]  

(3.18)

### 3.2.4 Main chamber

The hydraulic unit’s main chamber is, just like the brake cylinder, a fluid reservoir. Therefore it is modeled as a hydraulic capacity:

\[ \dot{p}_{mc} = C_h(p) \cdot Q \]  

(3.19)
3.2.5 Accumulator

In the vehicle’s hydraulic circuit a low pressure accumulator is added in order to allow the brake lines to dump fluid, and thereby decrease their pressure. As it’s main function is to store fluid, it can be modeled as a hydraulic capacity. Main difference in comparison to the brake lines and cylinder is that the volume of the accumulator increases significantly when the pressure increases, offering a relatively high fluid storage capacity. Figure 3.4 shows a schematic cross section view of an accumulator and a more schematic modeling representation, on which the accumulator modeling is based:

\[
\dot{p} = \frac{E'(p)}{V} \cdot Q
\]  (3.20)

Due to movement of the diaphragm, the volume of the accumulator will not remain constant, and thus:

\[
V = V_0 + A \cdot x
\]  (3.21)

The movement of the diaphragm, and thus the volume increase, is dependent on pressure, and can be deducted from the force equilibrium of the diaphragm:

\[
p \cdot A = k \cdot x
\]  (3.22)

The initial volume \( V_0 \) is assumed to be negligible compared to the influence of the volume increase due to spring compression, and thus one obtains:

\[
\dot{p} = \frac{E'(p) \cdot k}{A^2 \cdot p} \cdot Q
\]  (3.23)
Furthermore it is chosen to replace $\frac{E'(p)}{p}$ by a constant, as correct determination of it is hard due to the lack of knowledge about the amount of air trapped within the brake fluid, and probably unnecessary as its value is probably close to constant for the complete operating range of the accumulator as figure 3.5 shows.

![Figure 3.5: Results of $\frac{E'(p)}{p}$ for different percentages of trapped air within the brake fluid](image)

Furthermore, because $k$ and $A$ are both constants, they are replaced by one constant $E_{\text{acc}}$, that describes both. This results in the following final accumulator model:

$$\dot{p} = E_{\text{acc}} \cdot Q$$  \hspace{1cm} (3.24)

As the spring’s movement is limited, this accumulator model is only valid beneath a certain pressure of about ±7 bar. Although it is not necessary to model the accumulator’s behavior above this pressure, it is however useful to keep track of the filling rate of the accumulator. This latter has two reasons; first of all it tells whether the used representation is valid and second of all it can be used to determine when to actuate the pump. The filling rate is determined by the total fluid inflow minus outflow, $\int Q$, and will be tracked by the capacitance value $C_{\text{Acc}}$. The maximum amount of fluid the accumulator can accept until the spring movement is limited is described by $C_{\text{Accmax}}$. 

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Chapter 3. Hydraulic circuit modeling

3.2.6 Pump and motor

Figure 3.6: Schematic representation of an external gear pump

The pump can be used for both the ESC and ABS system. In the ESC system it is used to build up pressure in the brake lines, while for ABS purposes it is intended to empty the low pressure accumulator. The exact type of pump and motor used in the hydraulic unit is unknown, and therefore assumptions with respect to modeling have been made based on engineering judgment.

Most probably the pump itself is a fixed displacement pump (figure 3.6), which means that for each revolution a constant amount of fluid is pumped. This can be described as:

\[ Q_{\text{pump}} = V_{\text{nom}} \cdot \omega \]  

(3.25)

where

- \( Q_{\text{pump}} [m^3/s] \) = the volumetric flow rate
- \( V_{\text{nom}} [m^3/rad] \) = the volume of brake fluid pumped per radian rotation
- \( \omega [rad/s] \) = the angular velocity of the pump shaft

The angular velocity of the pump’s shaft is generated by an electro-motor. Modeling of this angular velocity is done by looking at the rotational dynamics of both pump and motor:

\[ \dot{\omega} \cdot I = u \cdot T_m - T_f \]  

(3.26)

where

- \( \dot{\omega} [rad/s^2] \) = is the angular acceleration
- \( I [kg \cdot m^2] \) = the moment of inertia of pump and motor combined
- \( u \) = the control signal (on/off) (with time delay)
- \( T_m [Nm] \) = the torque applied by the motor
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- $T_f [Nm] = \text{the torque applied on the shaft by friction}$

The torque applied by the motor is dependent on $\omega$, and probably shows characteristics similar to figure 3.7(a). Furthermore $T_f$ is the sum of frictions from several different causes as figure 3.7 describes.

![Figure 3.7: Several causes of power loss by a hydraulic pump [13]](image)

In the figure one can see that there are three causes for loss of power. It is assumed that these three power losses can be described by two friction functions $T_{f1}(\omega)$ and $T_{f2}(\omega, \Delta p)$. Figure 3.8(b) and (c) show the main characteristics of both functions.

![Figure 3.8: Motor (a) and friction functions (b & c)](image)

The complete pump and motor rotational dynamics can thus be described by:

$$\dot{\omega} \cdot I = u \cdot T_m(\omega) - T_{f1}(\omega) - T_{f2}(\omega, \Delta p) \quad (3.27)$$

With:

$$T_m(\omega) = -c_1 \cdot \omega + c_2 \quad (3.28)$$

$$T_{f1}(\omega) = c_3 \cdot \omega + c_4 \quad (3.29)$$
$T_{f2}(\omega, \Delta p) = c_5 \cdot \omega \cdot \Delta p + c_6 \quad (3.30)$

where $c_{1,6}$ are parameters which need to be determined during the identification phase.

### 3.2.7 Brake lever

The pressure applied by the brake lever and master cylinder on the hydraulic circuit depends on the driver’s force upon the lever. As the force applied is highly unpredictable, it is chosen to keep the force, and thus the pressure constant. The value of this force is an input of the model.

### 3.3 Simulink model

The following section will discuss the Simulink model of the hydraulic circuit. As the modeling equations have already been thoroughly discussed in previous section, this section only provides a brief view of the Simulink’s model structure.

#### 3.3.1 Overview

![Figure 3.9: Overview of the simulink model of the front subsystem](image)

Figure 3.9 shows an overview of the model. As can be seen, the model consists of:

- the brake lever, which represents the pressure applied by the master cylinder and driver
- the hydraulic unit, which contains all valves, pump, accumulator and main chamber
- the front left and front right pipeline
- the two brake cylinders
the controller, which generates the control inputs to the valves and pump.

It was chosen not to model the pipeline between brake lever and hydraulic unit, as identification of it is impossible in the current vehicle setup.

### 3.3.2 Hydraulic unit

![Diagram of the hydraulic unit](image)

Figure 3.10: Overview of the hydraulic unit within the front hydraulic sub-system

The hydraulic unit is the centerpiece of the hydraulic circuit. It connects all brake lines and the brake lever, and by opening and closing appropriate valves inside anti-lock braking control can be obtained. As visible in figure 3.10, the unit consists of:

- two hold valves, two release valves and the MBD valve. Although both MBD and OS valves are not used for ABS control, the MBD is modeled as it is useful for the identification experiments performed.
- the accumulator, offering the possibility for pressure decrease in the pipelines
- the main chamber, which is the connecting chamber between all three hydraulic connections
- the pump, offering the possibility to empty the accumulator
3.3.3 Brake lines and brake cylinder

The brake lines are, as discussed in previous section, modeled as several capacities in series. Fluid flow from one to another is modeled by inductance and resistance effects, thereby offering the opportunity to represent the inertia of the fluid and flow resistance. Both the left and right front brake lines are modeled as 5 capacities, connected by inductance and resistance effects. This is done in order to provide better wave propagation characteristics in the model. The end of each brake line is connected to a brake cylinder. In figure 3.11 only a part (2 capacities and inductance-resistance connectors) of the brake line is shown.

3.3.4 Parameters

The hydraulic circuit model now contains a large amount of parameters, shown in table 3.2, which determine the behavior of the model. In order to obtain a good correlation with the actual BMW’s hydraulic circuit, a parameter identification has to be performed.

<table>
<thead>
<tr>
<th>Component</th>
<th>Parameters and functions</th>
</tr>
</thead>
<tbody>
<tr>
<td>FL brake line</td>
<td>$C_h(p)$ $L_h$ $R_h$</td>
</tr>
<tr>
<td>FL brake cylinder</td>
<td>$C_h(p)$</td>
</tr>
<tr>
<td>FR brake line</td>
<td>$C_h(p)$ $L_h$ $R_h$</td>
</tr>
<tr>
<td>FR brake cylinder</td>
<td>$C_h(p)$</td>
</tr>
<tr>
<td>Main chamber</td>
<td>$C_h(p)$</td>
</tr>
<tr>
<td>Hold valves</td>
<td>$Q_{nom}$ $m$ $F_{spring}$ $F_{coil}$ $k$ $c$ $t_d$</td>
</tr>
<tr>
<td>Release valves</td>
<td>$Q_{nom}$ $m$ $F_{spring}$ $F_{coil}$ $k$ $c$ $t_d$</td>
</tr>
<tr>
<td>Accumulator</td>
<td>$E_{acc}$ $\int Q_{max}$</td>
</tr>
<tr>
<td>Pump and motor</td>
<td>$V_{nom}$ $I$ $T_m(\omega)$ $T_{r1}(\omega)$ $T_{r2}(\omega, \Delta p)$ $p$</td>
</tr>
<tr>
<td>Driver</td>
<td></td>
</tr>
</tbody>
</table>

Table 3.2: Parameters within the Simulink model
3.4 Conclusions

In this chapter a model has been deducted for the hydraulic circuit of a HAB type braking circuit, which is present in nearly every commercial road vehicle. Based on modeling of all individual components a detailed description of the hydraulic dynamics has been obtained. The model is implemented in Simulink, such that after the determination of parameters simulations can be performed easily. These parameters, which have all been summed up in table 3.2, will be identified in the following chapter, such that a detailed model of the BMW’s hydraulic circuit is obtained.
Chapter 4

Hydraulic circuit identification

In previous chapter the hydraulic circuit’s front subsystem of a ABS and ESC equipped BMW has been modeled. The chapter finished with table 3.2, which summarized all parameters within the model. This chapter is dedicated to determining these parameters, based on experiments performed on the BMW. In this chapter only the identification of the front subsystem will be explained and discussed. As the rear subsystem is practically the same, methods explained in this chapter are also valid for the rear’s subsystem identification.

This chapter will start with the description of the identification experiments, which are used to perform the identification. This is followed by the explanation of the identification methods which have been used to identify model parameters. The chapter finishes with an analysis and discussion of the quality of the obtained model.

4.1 Identification experiments

In this section the three identification experiments will be discussed. All three experiments have been performed numerous times using different experiment parameters, thus creating a large set of tests with which parameter determination is possible.

4.1.1 Experiment 1

This experiment offers insight in capacitance, inductance and resistance effects within the hydraulic system. Besides that, it also offers the possibility to observe the characteristics of the hold valves within the hydraulic unit.
The idea of the experiment is to let fluid flow from one brake line to another whilst being disconnected from the rest of the brake circuit. This is accomplished by closing the MBD valve, and by creating a pressure difference between the brake lines. The following list chronologically sums up the events during the experiment:

1. FL and FR brake lines are pressurized differently, by closing the lowest pressurized brake line’s hold valve.
   The MBD valve is closed in order to isolate the system
2. The hold valve is opened for a short period of time
3. Short waiting time, to let the system reach steady state
4. The hold valve is opened again (and kept open)
5. Waiting time, to let the system reach steady state

Figure 4.1 shows an example of the pressure over time and hold valve actuation scheme of a test of experiment type 1.

Figure 4.1: The pressure evaluation during experiment 1, for a certain set of initial pressures and opening time of the hold valve in phase 2

Within the experiment, the following experiment parameters are varied to observe pressure dependency and valve characteristics:

- Initial pressure in front left brake line
• Initial pressure in front right brake line
• Opening time of hold valve in phase 2

4.1.2 Experiment 2

The second experiment gives insight in inductance and resistance effects within the system. Furthermore release valve characteristics can be observed and the accumulator parameters can be determined. During this experiment, either the left or right brake line is subject of the experiment.

The basic idea of the experiment is to let a pressurized brake line dump fluid into the accumulator. The following list chronologically explains the experiment:

1. Brake line is pressurized, and hold valve is closed
2. The release valve is opened for a short period of time
3. Short waiting time, to let the system reach steady state
4. The release valve is opened again (and kept open)
5. Waiting time, to let the system reach steady state

Figure 4.2 shows an example of the pressure over time and release valve actuation scheme of a test from experiment 2.
Within the experiment, the following parameters were varied in order to determine pressure dependency and valve characteristics:

- Initial pressure
- Evaluation of pressure decrease in either left or right subsystem
- Opening time of the release valve in phase 2

To guarantee repeatability within the experiment, it is important to be sure that the accumulator is empty before starting the experiment. In order to achieve this, the pump should be activated for a short period of time before each experiment.

4.1.3 Experiment 3

The third experiment relates to identification of the pump’s parameters. The experiment uses the ESC valves within the hydraulic unit to pump brake fluid in either the left, right or both brake lines. The following list chronologically explains the experiment:

1. The hold valve of unused brake line is closed
2. The brake line is pressurized to about 5 to 10 bar
3. The MBD valve is closed and OS valve is opened
4. The OS valve of the rear subsystem is opened
Actuation of the pump for a certain time interval
Waiting time, to let the system reach steady state

Figure 4.3 shows an example of the pressure over time during the experiment.

Within the experiment, certain parameters were varied in order to obtain insight in the pump characteristics during different circumstances:

- Initial pressure
- Brake lines to pump in (left, right or both)
- Actuation time of the pump in phase 2

### 4.2 Identification methods

This section describes the identification of model parameters and functions using numerous tests of the previously described experiments. The order of the method discussion is similar to the order of the actual identification, as several parameters can only be determined by knowledge of other (earlier determined) model parameters.
4.2.1 Steady state characteristics

Observation of the modeling equations in chapter 3 shows that the relation of the steady state pressure between phase 1 and 5 in experiment 1 is solely dependent on the capacitive characteristics of the pipeline, brake cylinders and main chamber. As the pipelines and brake cylinders are inseparable, it is impossible to identify their capacitances individually. Therefore it is chosen to create a single capacitance function for each brake line during identification. The steady state characteristics of the system are thus now defined by $C_{FL}(p)$, $C_{FR}(p)$ and $C_{MC}(p)$.

In the current setup it is impossible to determine the absolute capacity of any of the components in the system. Therefore there is no absolute reference for identification. Capacities can thus only be identified in relation to each other in this system, and are therefore only valid in this particular model. Furthermore, due to the fact that the main chamber’s volume is much smaller than those of the pipelines and brake cylinders combined, it’s influence will be relatively small. This combined with the noise present in the measurements, makes correct determination of the capacitance factor $C_{MC}(p)$ close to impossible. Therefore it is chosen to set it to a small value, such that it doesn’t influence the system.

**Identification method** Figure 4.4 shows the main characteristics within experiment 1 that can be used to identify capacitance functions $C_{FL}(p)$ and $C_{FR}(p)$.

When starting from initial pressures $p_{0_{FL}}$ and $p_{0_{FR}}$ the capacitance functions determine $\Delta p_{FL}$ and $\Delta p_{FR}$, and thus the final pressure $p_{final}$. The final pressure $p_{final}$ is thus dependent on initial pressures $p_{0_{FL}}$ and $p_{0_{FR}}$.
and capacitive characteristics $C_{FR}(p)$ and $C_{FL}(p)$. By performing a simulation using similar initial conditions as during a real test, one can determine the quality of the simulation’s steady state characteristics by looking at the difference between simulated and experimental final pressure ($e_{p\text{final}}$).

As the capacitance function is pressure dependent, different tests have been performed with initial pressures ranging from ±10 to 90 bar. This allows identification of the capacitance functions over their entire operating range.

A correlating capacitance function $C_h(p)$ for both left and right subsystem is found by minimizing the sum of several test’s squared $e_{p\text{final}}$ values. The minimization of this cost function, which is described in equation 4.1, is done using the $fminsearch$ minimization method in Matlab. This is an unconstrained minimization method, which doesn’t make use of derivatives.

$$\text{Error} = \sum e_{p\text{final}}^2$$ (4.1)

It is chosen to square every individual error before summation, to emphasis minimization of larger errors.

### 4.2.2 Fluid dynamics

Both experiment 1 and 2 show clear pressure oscillations after valve closing in phase 2. These oscillations, which occur due to the fluid’s inertia, die out after a short period of time due to damping, which is caused by friction in the system. Within the model, inertia and friction are modeled as hydraulic inductance and hydraulic resistance respectively. Pressure oscillations will thus also occur within simulations, and are characterized by inductance and resistance parameters $L_{FL}$, $L_{FR}$, $R_{FL}$ and $R_{FR}$.

When modeling a hydraulic system using RLC elements, natural frequency and damping are defined by the parameters $C_h$, $I_h$ and $R_h$ [8]. As the capacitance parameter has already been determined, the inductance and resistance parameters remain to be tuned in order to obtain a good fit between simulation and experiment. Although both parameters influence both phenomena, natural frequency is mainly influenced by inductance $L_h$ whilst damping is primarily affected by resistance factor $R_h$. 
Identification method As the influence of resistance parameter $R_h$ on the natural frequency, as well as the influence of inductance parameter $L_h$ on the damping is quite small, it is chosen to separately approach optimization. Thus the inductance $L_h$ will be tuned in order to obtain a corresponding natural frequency, and the resistance $R_h$ will be optimized to obtain corresponding damping.

Natural frequency Figure 4.6 shows that natural frequency is only slightly influenced by pressure, and therefore any of the previously performed tests can be used to do perform inductance tuning. As figure 4.7 shows a higher inductance causes
more significant oscillations having a lower frequency with respect to a lower value of \( L_h \). The tuning is done manually using a trial and error method trying to match the period times of experimental and simulation results.

**Damping**
Looking at figure 4.6, one can see that for all four experiments the time needed for the oscillations to die out is quite the same. Again, any of the previously performed tests can thus be used to perform tuning. The resistance parameter tuning is done manually using a trial and error method in order to obtain damping which makes oscillations die out in a similar timespan as during the experiments. In figure 4.8 it is clearly visible how the resistance factor influences damping, whilst frequency of oscillations, as discussed before, remains constant.
4.2.3 Accumulator parameters

The two accumulator parameters $E_{\text{Acc}}$ and $C_{\text{Accmax}}$ can be determined using the experiment in which the accumulator is involved; experiment 2.

**Identification method** Observation of the modeling formula shows that the final pressure $p_{\text{final}}$ reached during the experiment (figure 4.9) is dependent on initial pressure $p_0$, capacitance functions $C_h(p)$ of either FL or FR brake line and accumulator parameter $E_{\text{Acc}}$. As the capacitance functions have already been determined, and initial pressure $p_0$ can be set, determination of $E_{\text{Acc}}$ using experiment 2 is possible.

![Figure 4.9: Important characteristics for the determination of accumulator parameters](image)

The tuning of $E_{\text{Acc}}$ is done manually using a trial and error method, by matching the final pressures of several tests to their corresponding simulations. It is chosen not to do an algorithm optimization, as by manual tuning a good correlation between experiment and simulation is quickly obtained.

Determination of the second accumulator parameter $C_{\text{Accmax}}$ is done by performing several tests of experiment 2, without emptying the accumulator in between each test. As long as $C_{\text{Accmax}}$ is not reached the accumulator will perform as normal, and pressure will decrease during phases 2 and 4. At the point the accumulator has completely filled however, pressure will not decrease anymore and $C_{\text{Accmax}}$ is found.
4.2.4 Valve parameters

The valves within the hydraulic system have a major influence in phases 2 and 4 in both experiment 1 and 2. Figure 4.10 shows the typical dynamic behavior of the system when opening and closing hold or release valves. The pressure transient is characterized by three phases, namely the opening dynamics, pure transient and closing dynamics. Although fluid inertia and resistance influence the entire transient, main characteristics are defined by the opening and closing dynamics of the valve, and the valve’s nominal fluid flow.

Figure 4.10: A closeup of the second phase of experiment 1, which shows the typical dynamic behavior of the system when opening and closing a valve.

Identification method  The identification of the valve parameters is done in four steps, using phase 2 of either experiment 1 or 2:

1. Identification of parameters influencing pure transient
2. Estimation of several unmeasurable solenoid valve parameters
3. Identification of parameters influencing opening dynamics
4. Identification of parameters influencing closing dynamics

As the hold and release valve have similar characteristics, the method of identification is similar. In this identification example, focus lies on the hold valves, and thus experiment 1.

Pure transient
The pure transient, which is the pressure transient when all valves are either completely open or completely closed, is influenced by the earlier determined...
inductance $L_h$ and resistance $R_h$ and the valve’s $Q_{nom}$. By the use of several different tests, having different valve opening times, a trial and error method is used to determine a $Q_{nom}$ which provides a correlating pressure derivative. Especially large valve opening times are useful during this identification method, as the pressure derivative can then be evaluated over a longer timespan.

**Solenoid valve parameters**
The valve’s mass $m$ and wall stiffness and damping $k$ and $c$ have no direct effect on the opening and closing dynamics of the valve and therefore cannot be determined by the current setup. The mass can be chosen freely, as in principle it only scales the dynamic equation. Wall stiffness and damping are chosen in such a way that the valve doesn’t chatter after closing or opening. The estimation of these variables is done manually, by trial and error.

**Opening dynamics**
The time from the moment of sending the opening valve command, till the moment the valve is completely open is regarded as the opening dynamics. It is influenced by the force that opens the hold valve (and closes a release valve), which is generated by a spring, and a signal time delay $t_d$. The spring force, which is modeled as a constant force $F_{spring}$, and $t_d$ are manually tuned, by comparing simulation and experimental results of the opening dynamics. Both the time delay and initial speed of the pressure increase are focused on.

**Closing dynamics**
The time from the moment of closing command till the moment the valve is completely closed is seen as the closing dynamics. The closing of a hold valve (and opening of release valve) in practice is done using a force generated magnetically using a permanent magnet and a coil, furthermore a small signal time delay, $t_d$, is present. Both force, $F_{coil}$, and time delay, $t_d$, are manually tuned by trial and error seeking for the best possible correlation between experiment and simulation closing dynamics.

**4.2.5 Pump parameters**
The last part of the hydraulic circuit identification is that of the pump. For a well correlating pump model, a correct combination of inertia $I$, motor torque function $T_m(\omega)$, and friction functions $T_{f1}(\omega)$ and $T_{f2}(\omega, \Delta p)$ needs to be determined. For this, experiment 3 is used.
Identification method  Identification of the parameters is done by looking at the correlation of pressure over time during several tests of experiment 3. The pressure difference between simulation and experiment over time of each test is integrated (error surface in figure 4.11), squared and summed to a total error value as equation 4.2 shows. Minimization of this error value results in the best possible combination of pump parameters.

\[
Error = \sum \int_{t=0}^{\bar{t}} |p_{simulation}(t) - p_{experiment}(t)|^2 dt \tag{4.2}
\]

where \(\bar{t}[s]\) is the time that pressure has reached steady state, thus \(\dot{p} = 0\).

Minimization is done using the \textit{fminsearch} function in Matlab, which is the same minimization method as used during the determination of the steady state variables.

4.3 Validation

In this section the quality of the model, including the identified parameters, will be analyzed and discussed. The quality will be analyzed using the identification experiments which have been introduced in section 4.1. First characteristics of each experiment will be reviewed thoroughly, after which a conclusion will reflect and discuss all results.
4.3.1 Experiment 1

In this subsection results of different tests of experiment 1 will be reviewed. Figure 4.12, 4.13 and 4.14 are basis for the discussion.

Figure 4.12: Pressure evaluation of phase 1, 2 and 3 of a fluid dump from FL to FR. Hold valve is opened at $t = 0$ s

Figure 4.13: Pressure evaluation of phase 1, 2 and 3 of a fluid dump from FR to FL. Hold valve is opened at $t = 0$ s
Chapter 4. Hydraulic circuit identification

Figure 4.14: Phase 4 and 5 of a pressure dump from FR to FL. Hold valve is opened at $t = 0\,\text{s}$

**Capacitance function** The quality of the capacitance function can be determined by looking at the pressure obtained in phase 5 of both simulation and experiment, as at that point the system has reached steady state and hold valve opening and closing has no effect anymore. The results of this comparison, based on 15 tests, is shown in the following table:

<table>
<thead>
<tr>
<th></th>
<th>Left brake line</th>
<th>Right brake line</th>
</tr>
</thead>
<tbody>
<tr>
<td>Average error [bar]</td>
<td>0.22</td>
<td>0.25</td>
</tr>
<tr>
<td>Max positive error [bar]</td>
<td>0.58</td>
<td>0.65</td>
</tr>
<tr>
<td>Max negative error [bar]</td>
<td>−0.32</td>
<td>−0.33</td>
</tr>
<tr>
<td>RMS error [bar]</td>
<td>0.33</td>
<td>0.36</td>
</tr>
</tbody>
</table>

Table 4.1: Results from comparison of experimental and simulated pressures in phase 5

The table shows that the final steady state pressure using the identified capacitance functions is simulated quite well. The average error is 0.22 and 0.25 bar for the front left and right brake line respectively, and the absolute maximum error found is only 0.65 bar. Furthermore the RMS error found is only 0.33 and 0.36 for respectively left and right brake line. Based on an operating range of about 0 to 100 bar, these results are very accurate.

**Opening dynamics** When valves are opened, it is generally seen that in simulation the time delay after which pressure increase and decrease starts is shorter. This especially counts for the high pressurized brake line, and can be well seen in the evaluation of FR brake pressure in figure 4.12. Although at first one could think this is related to the valve’s opening time delay, in this case also fluid inertia plays a major role, as pressure measurement is
performed far from the actual valve. Differences in time delay are in the order of 1 to 1.5 ms.

**Pressure transient** The pressure transient, and thus fluid flow when the valve is completely open, is quite similar with respect to experimental results. It is however hard to correctly determine from the pressure over time graphs, as pressure wave propagation disturbs the signal significantly. In overall it seems that simulation and experimental pressures tend to cross each other frequently over time, although one might say that simulation pressure change is slightly slower. This latter is best seen in the evaluation of the FR brake pressure in figure 4.12(a).

**Closing dynamics** The closing dynamics are mainly characterized by oscillations which result from wave propagation within the brake lines. The closing of the valves usually causes significant oscillations, which in the model is less present. The wave propagation itself will be discussed later. Furthermore it is hard to make any statement on time delays, as the oscillating pressure makes it impossible to read (in opposite to the opening dynamics). However, the following paragraph will give a more quantitative idea of the modeled closing dynamics’ quality, as it significantly influences the pressure obtained in phase 3.

**Obtained pressure is phase 3** With the assumption that the capacitance functions are correct to a degree discussed earlier, the pressure obtained in phase two is mainly dependent on the previously discussed opening dynamics, pressure transient and closing dynamics. This latter is true as opening and closing dynamics determine the opening time of the valve, and pure transient indicates the fluid flow when the valve is open. These combined results in the total fluid flow, which represents the pressure drop and increase of respectively the high and low pressurized brake lines. The difference in simulated and experimental pressure in phase 3 can thus be used to quantitatively reflect on the quality of the previous discussed. In table 4.2 results, based on 15 tests, of this comparison are shown.

<table>
<thead>
<tr>
<th></th>
<th>Left brake line</th>
<th>Right brake line</th>
</tr>
</thead>
<tbody>
<tr>
<td>Average error [bar]</td>
<td>0.07</td>
<td>−0.12</td>
</tr>
<tr>
<td>Max positive error [bar]</td>
<td>1.50</td>
<td>1.49</td>
</tr>
<tr>
<td>Max negative error [bar]</td>
<td>−1.09</td>
<td>−1.34</td>
</tr>
<tr>
<td>RMS error [bar]</td>
<td>0.80</td>
<td>0.78</td>
</tr>
</tbody>
</table>

Table 4.2: Results from comparison of experimental and simulated pressures in phase 3

From the table it can be seen that the errors are relatively small in compar-
ison to the absolute pressure values usually reached when performing ABS control. The average error is close to the noise level of the actual measured brake pressure, where the RMS error, which in this case is practically the same as the standard deviation, and thus gives an indication of the variance to be found, is also very small.

Based on the results in table 4.2 it can be said that although the discussed dynamics may have slightly different characteristics in simulation, the obtained pressure by them matches quite well.

**Wave propagation and pressure oscillations** The wave propagation within simulation is able to resemble certain basic properties of the real hydraulic circuit, but has difficulties resembling them properly. Although sometimes results are quite good, figure 4.12(b), usually differences are apparent. Usually the amplitude of oscillations is significantly smaller, and damping is less. The oscillation frequency of both left and right brake lines are quite reasonable, although there is usually a significant phase difference. Furthermore damping shows different characteristics, as during actual experiments oscillations magnitude seems to decrease quadratically, whilst in simulation this is linear. It is found the wave propagation before valve closing, and the actual timing of valve closing with respect to the pressure wave, has a major influence on the occurrence of pressure oscillations. This latter can well be seen in the evaluation of FR brake pressure in both graphs of figure 4.12.

### 4.3.2 experiment 2

This subsection will review simulation results obtained in experiment 2. Figure 4.15 and 4.16 provide a graphical view on the matter which will be discussed.
Figure 4.15: Pressure evaluation when dumping fluid from front left (a) and front right (b) brake lines (experiment 2 phase 1,2 and 3). Release valve is opened at $t = 0s$

![Pressure evaluation graph](image)

Figure 4.16: Pressure evaluation of phase 4 and 5 during experiment 2. Release valve is opened at $t = 0s$

![Pressure evaluation graph](image)

**Accumulator characteristics** Accumulator modeling quality can be analyzed by comparing simulated and experimental pressure obtained in phase 3. As at that point the system has reached an equilibrium, and pressure at that point is solely dependent on capacitance functions and accumulator parameter $E_{acc}$. The results from this comparison are summarized in table 4.3, which is based on 16 and 25 tests on the left and right brake line respectively.
Chapter 4. Hydraulic circuit identification

Table 4.3: Results from comparison of experimental and simulated pressures in phase 5

<table>
<thead>
<tr>
<th></th>
<th>Left brake line</th>
<th>Right brake line</th>
</tr>
</thead>
<tbody>
<tr>
<td>Average error [bar]</td>
<td>0.31</td>
<td>−0.28</td>
</tr>
<tr>
<td>Max positive error [bar]</td>
<td>0.80</td>
<td>0.06</td>
</tr>
<tr>
<td>Max negative error [bar]</td>
<td>0</td>
<td>−1.22</td>
</tr>
<tr>
<td>RMS error [bar]</td>
<td>0.43</td>
<td>0.39</td>
</tr>
</tbody>
</table>

The table shows that again both average and maximal errors are small. The RMS error shows that variance is quite small. The results furthermore show that simulations on the left brake line usually over predict pressure, whilst the right brake line generally underestimates the final pressure. Although again the absolute numbers are quite low, the errors are relatively larger, as the absolute final pressure is generally in the region of ±5 bar, as can be seen in figure 4.16.

Opening dynamics, pressure transient and closing dynamics The results during phase 2, which contain the opening dynamics, pressure transient and closing dynamics, are quite in line with the results observed during the analysis of experiment 1. Again pressure decreases start earlier, and the speed in which pressure decreases is quite similar with respect to experimental results. During experiments a significant pressure oscillation occurs when the release valve is closed, which is only slightly present in the model which is well visible in figure 4.15.

Obtained pressure in phase 3 A quantitative analysis of all dynamics discussed in previous paragraph can, for the same reasons as in experiment 1, be performed by looking at the simulation and experimental pressure differences is phase 3. Table 4.4 shows a summary of these results, which is are based on 16 and 25 tests for left and right brake line respectively.

<table>
<thead>
<tr>
<th></th>
<th>Left brake line</th>
<th>Right brake line</th>
</tr>
</thead>
<tbody>
<tr>
<td>Average error [bar]</td>
<td>−0.45</td>
<td>0.53</td>
</tr>
<tr>
<td>Max positive error [bar]</td>
<td>1.78</td>
<td>3.01</td>
</tr>
<tr>
<td>Max negative error [bar]</td>
<td>−1.79</td>
<td>−2.35</td>
</tr>
<tr>
<td>RMS error [bar]</td>
<td>0.96</td>
<td>1.46</td>
</tr>
</tbody>
</table>

Table 4.4: Results from comparison of experimental and simulated pressures in phase 3

The results show that errors are significantly larger than the other quantitative comparisons made before. It can be seen that the average error is about half a bar negative and positive for the front left and front right brake line,

50
and maximum errors are about ±2 for left and +3 and −2.5 for the right brake line. The RMS error is furthermore also significantly larger. Although errors are larger with respect to earlier comparisons, results are quite good in comparison to the total operating range of the system. The results show that errors are significantly larger than other quantitative comparisons. It can be seen that average error is about half a bar negative and positive for the front left and front right brake line, and errors are relatively large in both negative as positive direction. The RMS error is furthermore also significantly larger.

**Wave propagation and pressure oscillations**  With respect to wave propagation similar counts as discussed previously for experiment 1.

### 4.3.3 Experiment 3

In this subsection the last experiment will be reviewed, figure 4.17 is used as reference.

![Figure 4.17: Pressure over time when the pump is actuated (experiment 3)](image)

When evaluating the pressure increase due to a pump action, it is quite important to note that most of the pressure increase happens whilst the pump has already been turned off. Apparently the pump has a significant inertia, which causes it to run for a fairly long period of time once turned off.

When looking at the evaluation of pressure over time it can be seen that the pumping characteristics are well captured by the model. In figure 4.17 it can be seen that the first 100 ms is very well captured for both situations. Note that the initial decrease of pressure (at $t = 0.1s$) within the experiment is a measurement error. Furthermore it appears that when the pump is activated relatively long, simulation seems to slightly overestimate the amount of fluid pumped, which is clearly visible in figure 4.17(b). Although a good pressure increase over time is nice, a good correlation with respect to the final pressure reached is more significant. This latter is because the pump will be used for
accumulator emptying, and for that the absolute amount of fluid pumped is important. The following table shows a summary of the results when comparing final pressures of both simulation and experiment. The table has been build on 18 different tests.

<table>
<thead>
<tr>
<th></th>
<th>Left brake line</th>
</tr>
</thead>
<tbody>
<tr>
<td>Average error</td>
<td>0.42</td>
</tr>
<tr>
<td>Max positive error</td>
<td>1.85</td>
</tr>
<tr>
<td>Max negative error</td>
<td>-1.70</td>
</tr>
<tr>
<td>RMS error</td>
<td>0.99</td>
</tr>
</tbody>
</table>

Table 4.5: Difference in final pressure after pumping

Results from table 4.5 show that the average error is below half a bar, and that maximum error values of about ±2 are found. Considering the uncertainties present in the pump modeling and the significant pressure increases, these results are quite accurate. The table shows quite reasonable results, considering the amount of parameters and uncertainty present in the model. A maximum positive error of 1.84 and minimum error of -1.70 bar is acceptable, considering the previous and the fact that significant pressure increases take place.

4.3.4 Conclusions

Now all characteristics have been discussed and summarized, conclusions with respect to the quality of the hydraulic model will be made. These conclusions are based on the fact that the hydraulic circuit model will be used for brake pressure controller design, and ABS controller simulation. Main interest thus lies in the fact whether the model is a sufficient reflection of reality such that decent controller design and ABS testing can be performed.

In general it can be said that the steady state pressures obtained are accurate enough for the purpose at hand. The final pressures obtained in phase 5 of experiment 1 show that the capacitance functions describe both bulk modulus and volumes well. Furthermore pressures obtained in phase 3 are accurate, which means that timing of opening and closing of valves and pressure flow through valves is well modeled.

More closely looking at the pressure transients, it can be said that in overall the results from simulation and experiment are quite similar. The pressure gradient when valves are open is are quite alike, although it seems that the opening dynamics are a bit faster, something which is probably related to incorrect modeling of inertial effects.
With respect to this latter it can be stated that pressure wave propagation, and the resulting pressure oscillations are captured with less precision than other characteristics. Although in some tests the results look quite similar, most often the pressure oscillation magnitude is significantly less. Furthermore also the damping is noticeably different, as in simulation oscillation magnitude seems the decrease linear, whilst experiments show a more quadratic relation.

The reason for the differences in wave propagation and pressure oscillation effects can be sought in several areas; first of all the actual brake circuit has a different layout than the model, as the pressure sensors have not been mounted directly on the brake lines, but a t-split and relatively long cable have been used. Furthermore, logically, the inductance and resistance variables may have been incorrectly tuned, but also valve closing has significant effect on the creating of pressure waves. Differences in damping may be sought in the flexibility of the brake lines, which may affect damping significantly, and is not modeled.

The fact that the modeled oscillations are significantly less should be taken in account when testing for instance threshold based brake pressure controllers on the model. Furthermore in order to improve model behavior regarding this issue probably actual model changes are necessary, as by solely changing parameters other characteristics of the model degrade.

The accumulator quality is sufficient for the brake controlled design purpose. Although the pressure is usually slightly different, this does not have too much effect on the overall behavior of the system. The assumptions and simplifications made during modeling (sub-section 3.2.5) can thus be accepted.

The pump model is well able to resemble the volumetric flow during experimental tests, although initially during modeling numerous factors were uncertain. Although results for the test performed, which are all relatively short periods of pumping, are quite well, it is however not proven that the model is also valid under more ABS alike circumstances (Higher pressure, longer pumping periods). For now this is however impossible to test, as pressure measurement is limited at 100 bar.

All results discussed in this chapter are obtained by experiments performed under constant circumstances, as all test were performed on a still standing BMW at the intelligent automotive division of TU Delft. It is well possible that different circumstances will influence the system’s behavior. This is however at this point not accounted for. Main differences may arise when temperature changes, as the fluid’s bulk modulus is temperature dependent.
Furthermore battery voltage may influence pump power.
Chapter 5

Controller design

This chapter is focused on the design of a brake pressure controller for the TU Delft BMW. In previous chapters a hydraulic circuit was modeled and identified to obtain a good representation of reality. In this chapter this model is used to synthesize a brake pressure controller.

First several important system characteristics regarding brake pressure control will be discussed. This is followed by the discussion of the design of the pressure controller, which is done in two steps. First the valve controller design will be discussed, which is followed by the discussion on the design of an accumulator observer and pump control. In the last part the brake pressure controller is evaluated in simulation and vehicle tests, and the results obtained are discussed.

5.1 System characteristics

The hydraulic circuit has several important characteristics which should be taken in account when synthesizing the brake pressure controller. In this section, these characteristics are summarized.

5.1.1 Valve positions

As in the current setup there is no mean to measure valve currents, proportional control using PWM (Pulse Width Modulation) is impossible. Thus only on-off control is possible, which leads to either a completely open or completely closed valve. This lack of proportional control, combined with the time delays present, makes it practically impossible to have continuous brake pressure control. A phased control, using hold, release and build pressure stages in which valves are either completely open or closed, is thus most logical.
5.1.2 Actuation delays

An important factor with respect to brake pressure control is the delay occurring when switching valve state. This delay is composed of two factors: a pure time delay due to signal lag and amplification and a delay caused by the inertia of the valves’ moving mass. It is assumed that the pure time delay for each valve is of an equal length. Experimental results furthermore show that the delay caused by inertia is different for each type of valve (hold or release), and each action (energizing or de-energizing). The hold valves have a significantly shorter opening and closing time than the release valves, which results in good performance when switching between hold and build pressure phases. The release valves however react much slower, especially when de-energizing. Figure 5.1 shows the opening and closing times of both the hold and release valves in simulation. Although these results are not the actual experimental opening and closing times of the valves, they do give a good indication of the difference between hold and release valves, as model results match experimental results quite accurately.

![Simulated valve opening and closing times](image)

Figure 5.1: Simulated valve opening and closing times

From the figure it can be seen that the hold valve opens about 2 ms, and closes more than 10 ms faster than the release valve. Especially the closing of the release valve is thus much slower, which results in a significant difference between the pressure at the moment of valve closing and the final steady state pressure, as visible in figure 5.2. This difference will from now on be called the pressure mismatch.
Chapter 5. Controller design

Figure 5.2: Pressure over time when opening and closing of the hold and release valves respectively, starting with both valves closed.

The figure clearly shows that the pressure mismatch when switching from release to hold is much larger than from build to hold.

5.1.3 Fluid inertia

Besides the valves, also fluid inertia causes delays in the system, as time is needed to accelerate and move brake fluid. Due to this, pressure waves will propagate within the hydraulic circuit, which results in two important phenomena:

- Due to the wave propagation pressures may differ over the entire brake line, which causes measurements on a single point, as done in the current vehicle setup, to be unrepresentative for the complete brake line’s pressure.

- The pressure waves cause large pressure oscillations after valve closing, which may cause problems when using a threshold based control algorithm.

5.1.4 Accumulator characteristics

In order to decrease pressure in a brake line, fluid is dumped into the (low pressure) accumulator. This means that the accumulator fills up during each release phase. In order to maintain pressure dump performance, the accumulator should maintain at low pressure, which means that it needs to be emptied regularly. As in the current setup it is impossible to measure the pressure or state of the accumulator, a method should be thought of to keep track of accumulator filling such that a decent regulation of the pump can be designed to empty it.
5.1.5 Pump characteristics

For pump regulation, to keep the accumulator pressure low, it is very useful to have knowledge about the volume which is being moved by the pump. The pumped volume is dependent on several factors as pressure difference $\Delta p$, the motor’s rotational speed $\omega$ and battery voltage $V$. Figure 5.3 shows the pump’s volumetric flow over time for different values of $\Delta p$, obtained using the hydraulic circuit model.

![Graph showing pump's volumetric flow over time](image)

Figure 5.3: The pump’s volumetric flow for different values of $\Delta p$

From the figure it can be well seen that a larger $\Delta p$ results in a lower volumetric flow, while furthermore also the pump inertia plays a major role in the dynamics. This latter causes the gradual startup, and results in a gradually decreasing volumetric flow when the pump is turned off.

5.2 Valve controller

The brake pressure controller consists of two elements. On the one hand the valves have to be controlled to obtain a correct brake pressure on the wheel brakes, while on the other hand also the pump needs to be controlled, in order keep the accumulator’s pressure low enough to guarantee good pressure dump performance. In this section the valve controller will be discussed.

First the requirements with respect to the controller are determined, after which two possible control schemes are proposed, of which the best is selected. Afterwards a method to decrease pressure mismatches is proposed, which is then included in the final valve control scheme.
5.2.1 Requirements

The previously discussed characteristics and limitations of the hydraulic system should be taken into account during the valve controller synthesis. Besides the system’s characteristics, also the requirements of the ABS controller with respect to brake pressure control should be taken into account.

As the brake pressure controller will be used for the novel force based algorithm, which will be discussed in chapter 6, requirements have been taken from that algorithm. These requirements are:

1. No chattering should occur close to the reference pressure
2. When increasing pressure, the final pressure should at least be as high as the reference pressure \( p_{\text{final}} \geq p_{\text{ref}} \)
3. When decreasing pressure, the final pressure should at least be as low as the reference pressure \( p_{\text{final}} \leq p_{\text{ref}} \)

Besides the requirements of \( p_{\text{final}} \geq p_{\text{ref}} \) when increasing pressure and \( p_{\text{final}} \leq p_{\text{ref}} \) when decreasing pressure, it is desired that the final braking pressure \( p_{\text{final}} \) is as close as possible to the reference pressure \( p_{\text{ref}} \) in the shortest possible timespan.

5.2.2 Control schemes

Based on the limitations, characteristics and requirements discussed, several different brake pressure controllers can be designed. As mentioned before, continuous control is impossible in the current setup, and phased control is thus most logical. Three different phases, hold, release and build pressure, will be used to control the braking pressure, and switching phases should be done in such a way, that the requirements are met. Based on the three phased logic, two interesting switching scheme’s can be thought of:
Threshold based switching

The threshold based control scheme uses the pressure reference $p_{\text{ref}}$, obtained from the ABS algorithm, to create a reference region. This region, bounded by a lower and upper bound $p_{\text{bound}}^-$ and $p_{\text{bound}}^+$ respectively, is the controller’s objective. In general the following three rules apply:

- if $p_{\text{meas}}$ is within bounds $\implies$ Hold pressure
- if $p_{\text{meas}}$ is above bounds $\implies$ Release pressure
- if $p_{\text{meas}}$ is below bounds $\implies$ Build pressure

The bounds are created by offsetting the reference pressure $p_{\text{ref}}$ obtained from the ABS algorithm. This can be described as follows:

$$p_{\text{bound}}^+ = p_{\text{ref}} + p_{\text{offset}}^+$$
$$p_{\text{bound}}^- = p_{\text{ref}} - p_{\text{offset}}^-$$

The two offsets, $p_{\text{offset}}^+$ and $p_{\text{offset}}^-$, can be determined beforehand, or could be updated on-line, creating an adaptive bounding region.

Figure 5.4 shows the switching logic of the threshold based switching method.
Reference based switching
In this method, phase switching is based on changes of reference pressure. Each time the reference pressure is updated, the algorithm decides whether a different phase should be triggered based on the current measured and reference brake pressure. Furthermore, a switch to the hold pressure phase is made within the build or release phase whenever the measured brake pressure exceeds the reference pressure. Figure 5.5 shows the switching logic of the reference based switching method.

![Switching logic of the reference based switching method](image)

Figure 5.5: Switching logic of the reference based switching method

5.2.3 Control scheme selection
Both control schemes are able to control the brake pressure in a decent manner such that it can be used for brake pressure control for the ABS algorithm. However, during testing it was found that the reference based switching method is a more appropriate control method for the hydraulic circuit in question. Main reason for this is that the pressure oscillations, occurring when entering a hold phase, may cause chattering in the threshold based controller if the offsets are not large enough. Dynamic thresholding could offer a workaround for this problem, but will also increase controller complexity. It is therefore chosen to continue with the reference based switching method.

5.2.4 Mismatch compensation
As mentioned in subsection 5.1.2, actuation delays cause a difference between the pressure at the moment of switching and the final steady state pressure, the pressure mismatch. As the hold valves have a significantly
lower actuation delay, the problem is less influential when switching between build and hold phases. However, switching from a release to hold phase may cause large pressure mismatches, as the de-energizing phase of the release valve is relatively slow, as was visible in figure 5.1. The pressure mismatch may therefore become as high as 20 bar, which is unacceptable for brake pressure control used for the force based ABS algorithm. In this subsection a workaround is proposed, to minimize the pressure mismatch.

**Release pressure**  The pressure mismatch’s magnitude depends on the fluid flow \( Q \) at the time between the closing signal, and actual closing. This fluid flow \( Q \) on its turn is related brake line’s pressure derivative. The measured brake line’s pressure derivative could thus be an indication of the amount of mismatch to be expected. The volumetric flow \( Q \) is furthermore also related to the brake line pressure at closing, as the flow is caused by the pressure difference between accumulator and brake line. Figure 5.6 shows an simulation example of how mismatches can be determined with respect to switching pressure and pressure derivative.

![Figure 5.6: Method to determine the relation between switching pressure and derivative with respect to the pressure mismatch. The mismatch is defined as: \( p_{\text{switch}} - p_{\text{final}} \)](image)

Figure 5.7 shows two scatter plots, in which the pressure mismatch is shown in relation to (a) the pressure at which the valve is closed and (b) the pressure derivative at closing in simulation.
Chapter 5. Controller design

Figure 5.7: The relation between pressure mismatch and switching pressure (a) and the measured pressure derivative (b) when dumping pressure in simulation.

From the two scatter plots it’s clearly visible that the switching pressure has a stronger correlation with the pressure mismatch than the pressure derivative. Although the pressure derivative is driving the mismatch, the measured pressure derivative is not representative due to wave propagation as discussed in subsection 5.1.3. Furthermore it is practically impossible to filter out this effect, because of its low frequency. It is thus best to use the switching pressure to predict pressure mismatch.

Figure 5.8 shows the relation between switching pressure and pressure mismatch, but now also categorized by the decrease phase’s starting pressure.
From figure 5.8 it can be seen that the decrease phase’s initial pressure does not give any extra extra information with respect to the pressure mismatch as a wave shape is present on each individual initial pressure. This wave shape is related to the wave propagation within the circuit.

Requirements of the pressure controller state that when decreasing pressure, the final pressure should be smaller or equal to the reference pressure. As the accumulator filling percentage also influences the mismatch, figure 5.8 should be redrawn using the worst case scenario with respect to the accumulator. With the assumption that the accumulator will fill to a maximum of 75% during normal usage, one can redraw the figure by performing new simulations with a 75% filled accumulator:
Figure 5.9: The pressure mismatch related to the switching pressure starting with an 75% filled accumulator, categorized by initial pressure

From the figure it can be seen that a 75% filled accumulator causes the pressure mismatches to be slightly smaller. This is to be expected, as the driving pressure difference is smaller due to the higher accumulator pressure. Using this figure, a minimal pressure mismatch can be determined for each switching pressure, which is valid under all reasonable circumstances.

With the use of this minimal pressure mismatch knowledge, the hold phase trigger can be optimized, as there is knowledge of the pressure mismatch after closing. Using the minimal pressure mismatch graph, a static mapping can be created of the final expected pressure with respect to the switching pressure. This relation is visualized in figure 5.10.
The hold phase trigger, $p_{\text{meas}} < p_{\text{ref}}$, within the dump pressure phase can now be updated using the static mapping of figure 5.10: $p_{\text{meas}} < p_{\text{ref}}^*$. One problem of the use of this mapping arises when the pressure reference is only slightly smaller than the pressure present, as the switching pressure in that case might be higher than the measured pressure. This would mean a direct return from the release phase to the hold phase, and results in a pressure higher than the reference pressure $p_{\text{ref}}$, which is not in accordance to requirement 3 from subsection 5.2.1. This problem is avoided by using the static mapping triggering statement only after a certain time $t_{\text{wait}}$, which represents the opening time of the valve. The new switching condition now thus becomes:

$$\begin{cases} p_{\text{meas}} < p_{\text{ref}} & \text{if } t_{\text{release}} \leq t_{\text{wait}} \\ p_{\text{meas}} < p_{\text{ref}}^* & \text{if } t_{\text{release}} > t_{\text{wait}} \end{cases}$$ (5.2)

where

- $t_{\text{release}}[s] = $ the time since the start of the release phase

**Build pressure** The pressure mismatch when building pressure is, again, dependent on the pressure flow $Q$ at the time between the closing signal, and actual closing. As the hold valves are much faster than the release valves, this time window is smaller, and the mismatch is smaller. Figure 5.11 shows scatter plots relating the switching pressure and pressure derivative with respect to the pressure mismatch.
Chapter 5. Controller design

From the figure it can be seen both the switching pressure as the pressure derivative show no clear relation with respect to the pressure mismatch. Furthermore the initial pressure at which the build phase starts also doesn’t give any indication of the pressure mismatch, as can be seen in figure 5.12.

It can be seen that in general the pressure mismatch is smaller than in the release pressure stage. As the mismatch is small, and there is no clear method to predict it, it is chosen not to apply any mismatch compensation on the build phase.
5.2.5 Final control scheme

With the incorporation of the mismatch correction on the release phase, the control scheme becomes:

![Diagram of control scheme]

Figure 5.13: The final valve controller’s control scheme

For readability reasons the activation logic is not shown in the figure. The following activation logic is used:

- If Controller == On AND $p_{\text{meas}} < p_{\text{ref}}$ ⇒ Build pressure
- If Controller == On AND $p_{\text{meas}} > p_{\text{ref}}$ ⇒ Release pressure
- If Controller == On AND $p_{\text{meas}} == p_{\text{ref}}$ ⇒ Hold pressure

5.3 Pump controller

The second element of the brake pressure controller is the pump controller, which ensures that the accumulator’s pressure and filling maintains low, such that brake pressure dump performance is optimal. For a correct pump scheduling, an accumulator observer is designed based on the modeled hydraulic circuit. Based on that observer a simple pump control scheme is designed.
5.3.1 Accumulator observer

As direct measurement of the state of the accumulator is impossible in the current vehicle setup, an observer is synthesized to keep track of the accumulator’s state. This observer is based on the identified maximum amount of fluid the accumulator can accumulate, $C_{\text{Accmax}}$. Furthermore, the amount of fluid inflow is calculated by observation of the pressure decrease during release valve opening of either left and right brake lines and the fluid outflow is calculated using a pump model. The accumulator state is tracked as follows:

$$\dot{s} = \frac{Q_{FL} + Q_{FR} - Q_{Pump}}{C_{\text{Accmax}}}$$  (5.3)

where

- $\dot{s}[1/s] = \text{the accumulator state change rate}$
- $Q_{FL}[m^3/s] = \text{the fluid inflow through the FL release valve}$
- $Q_{FR}[m^3/s] = \text{the fluid inflow through the FR release valve}$
- $Q_{Pump}[m^3/s] = \text{the fluid outflow due to pumping}$
- $C_{\text{Accmax}}[m^3] = \text{the maximum fluid capacity}$

The accumulator state will vary from 0 to 1, where 0 means that the accumulator is empty, while 1 represents a full accumulator. In the following subsections the method of calculating $Q_{FL}$, $Q_{FR}$ and $Q_{Pump}$ will be more thoroughly explained.

5.3.2 Fluid flow through valves

Calculation of $Q_{FL}$ and $Q_{FR}$ is done using earlier obtained knowledge of the hydraulic system. The fluid flow is calculated using formulas 5.4 and 5.5, which use parameters determined earlier during identification:

$$Q_{FL} = \Delta p_{FL} \cdot C_{FL}(p)$$  (5.4)

$$Q_{FR} = \Delta p_{FR} \cdot C_{FR}(p)$$  (5.5)

where

- $\Delta p_{FL}$ and $\Delta p_{FR}[N/m^2] = \text{the pressure change since the last calculation}$
- $C_{FL}(p)$ and $C_{FR}(p)[m^5/N] = \text{the capacitance function of the left and right brake lines}$
Calculation of $Q_{FL}$ or $Q_{FR}$ is only done when the corresponding release valve is open; when closed the fluid flow $Q$ is set to zero. As the valves have a significant time delay, and the measured pressure is not always representative for the complete pipeline pressure, it is chosen to keep computing $Q_{FL}$ and $Q_{FR}$ using formulas 5.4 and 5.5 for a short time, $t_{close}$, after valve closing as shown in figure 5.14.

![Figure 5.14: Visualization of the timing of $Q_{FL}$ calculation](image)

### 5.3.3 Fluid flow trough pump

The fluid outflow due to the pump, $Q_{pump}$, is modeled using the pump characteristics discussed in subsection 5.1.5. Due to the many factors influencing the pump volumetric flow, it is practically impossible to make a good on-line prediction of it. Therefore it is chosen to make a more static prediction of its flow, based on the most tough circumstances, thus creating a model which underestimates the fluid outflow. In this way the accumulator’s actual state is always lower than the observed state, which ensures safe modeling.

$Q_{pump}$ while pumping is described by the following formula:

$$Q_{pump} = a \cdot e^{b \cdot t_{on}} + c \cdot e^{d \cdot t_{on}}$$  \hspace{1cm} (5.6)

where:

- $a$, $b$, $c$ and $d$ are constants
- $t_{on}[s]$ is the time the pump has been on

When turned off, the inertia of the pump causes fluid flow to gradually decrease to zero. This is described by the following formula:

$$Q_{pump} = Q_{final} - f \cdot t_{off}$$  \hspace{1cm} (5.7)

where:
• $Q_{\text{final}}$ is the final volumetric flow when the pump was on
• $f$ is a constant
• $t_{\text{on}}[s]$ is the time the pump has been off

Figure 5.15 shows how equation 5.6 and 5.7 describe the pump’s volumetric flow with respect to time. It can be seen that it quite accurately represents the worst case scenario from figure 5.15. All other scenarios where volumetric flow was higher are also vaguely shown in the figure. As the pump can only have a positive outflow, the value $Q_{\text{pump}}$ is saturated at 0, such that it can never be negative.

![Figure 5.15: Visualization of the estimated volumetric flow of the pump when turning on and off](image)

5.3.4 Control scheme

A simple two phase control scheme for the pump can now be constructed. It is chosen to start the pump when the accumulator is filled over 30%, and stop when the state has decreased to below 10%. The control scheme is shown in figure 5.16.
5.4 Controller results

The following section will show and discuss the results obtained using the valve and pump controller, together called the brake pressure controller, in both simulation and real vehicle tests. First the result of the controller in simulation is shown, which is followed by the results obtained from experiments on the BMW. Finally a wrap-up will be made, and conclusions will be drawn.

5.4.1 Simulation

This subsection will reflect on the quality of the brake pressure controller in simulation. First the build and release pressure results will be shown and briefly discussed. This is followed by a discussion of the controller during several phase switches, in which also the quality of the pump controller can be observed.

Build pressure Figure 5.17 shows the expected pressure mismatch when building pressure for several different initial pressures when a constant pressure of 95 bar is applied by the master cylinder.
Figure 5.17: Pressure mismatches for different initial pressures and reference pressures, when a constant pressure of 95 bar is applied by the master cylinder

The results from the figure are quite similar to those of figure 5.12, although the driver applied pressure is slightly higher. From this figure, and results from section 5.2, it can be concluded that the pressure mismatch in the build phase is quite unpredictable, but ranges between 0 and 5 bar.

The figure shows that in a few situations the pressure mismatch is below zero, which would mean that requirement 2 is not met. Although complying to this requirement could be simply achieved by adding a bias to the reference pressure, thus shifting the entire graph up, this is not done as the mismatches are small (< 0.5 bar) and the amount of occurrences is low.

**Release pressure** Figure 5.18 shows the expected pressure mismatch when releasing pressure from several initial pressures, when the initial accumulator state is 0 (empty).
Figure 5.18: Pressure mismatches for different initial pressure and reference pressures, when starting with an empty accumulator

The figure shows that using the compensation method the pressure mismatch is much smaller for most of the cases while requirement 3 is still met for all situations. Significant pressure mismatches are still found for situations in which only a small pressure decrease is requested. This latter is due to the waiting time introduced within the hold phase trigger, which ensures that no positive mismatches occur.

**Controller in the loop**  Figure 5.19 shows the pressure over time, if the reference pressure is changed several times.
Figure 5.19: Pressure over time, when a certain reference pressure is fed to the controller

The figure shows that the reference pressure is tracked quite well. The pressure mismatches found are in agreement to the earlier shown results from figures 5.17 and 5.18.

Figure 5.20 shows the state of the accumulator over time, and the actuation scheme of the pump. Both the real (blue) and predicted (green) accumulator state are shown.

Figure 5.20: Accumulator state and pump control over time, for the pressure control in figure 5.19
It can be seen that the accumulator state is predicted quite well. As intended, the prediction is slightly overestimated. Furthermore it can be seen that the accumulator’s state is kept low, due to the pump turning on after exceeding the 30% fill rate.

5.4.2 Experimental

In this subsection the quality of the brake pressure controller is evaluated on the hydraulic circuit of the BMW. It may be expected that results differ, as the model on which the controller is tuned doesn’t perfectly match reality. Again first the build and release pressure quality will be discussed, followed by a discussion of the controller during several phase switches. Differences with respect to the simulation results will be addressed and discussed. Note that exactly the same tuning parameters/mismatch compensation is used.

Build pressure Figure 5.21 shows the pressure mismatch when building pressure for several different initial pressures. During the experiments it is tried to maintain a constant master cylinder’s pressure of 95 bar. Furthermore, the mismatch is calculated when the system has reached steady state, and thus the pressure is constant.

![Figure 5.21: Pressure mismatches when building pressure for different pressure references and initial pressures](image)

The figure shows that, in opposition to the simulation results, in most of the cases the mismatch is lower than 0 bar. It is clearly visible that especially for larger pressure steps the mismatch is much lower than in simulation. Furthermore it can be observed (from the vertical lines in the figure) that when
performing similar test, different results are obtained. The significantly different results with respect to mismatches and repeatability in comparison to simulations are due to two important differences between reality and model. The first, and most influential difference, can be observed in figure 5.22.

![Figure 5.22: Evaluation of pressure over time when building pressure. (a) shows a small increase of pressure, while in (b) a larger pressure step is shown](image)

The figure clearly shows that for both cases initially the reference pressure is tracked quite accurately. However, when time passes, the pressure in (b) decreases significantly, resulting in a final pressure mismatch of several bar. This latter is most probably caused by the slow elasticity of the brake lines, which causes the volume to slowly increase, and pressure to slowly decrease, after the build pressure phase. This phenomena is logically more influential when making larger pressure steps. This thus explains the results obtained in figure 5.21.
The second difference with respect to simulation is the applied pressure from the master cylinder. In simulation this is assumed to be constant, while in reality this logically is heavily dependent on the driver. During the experiments it was tried to keep the applied pressure as constant as possible, but as the right brake line pressure (which is in direct connection with the master cylinder) in figure 5.22 shows, this still fluctuates significantly.

The influence of master cylinder pressure can be observed by performing build pressure experiments using different applied pressures. Results of these experiments are shown in figure 5.23.

![Figure 5.23: Pressure mismatches when building pressure, if different pressures are applied](image)

From the figure it can be concluded that the applied brake pressure only slightly influences the results obtained. Although a higher applied pressure results in a slightly more positive mismatch, differences are small in comparison to the overall tendency of the figure.

**Dump pressure** Figure 5.24 shows the pressure mismatch when releasing pressure, starting from different initial pressures and an initially empty accumulator. As for the build pressure paragraph, mismatch is calculated at steady state.
Figure 5.24: Pressure mismatches when releasing pressure for different reference pressures and initial pressures

From the figure it can be seen that the experimental pressure mismatch is (again) quite different with respect to the simulation results. Although for the smaller pressure steps the mismatch is negative, and thus requirement 3 is met, larger pressure steps result in a undesired positive mismatch. Again this problem is related to the slow elasticity of the brake lines as can be seen in figure 5.25.
Figure 5.25: Evaluation of pressure over time when releasing pressure. (a) shows a small pressure decrease, whilst (b) shows a much larger step. It is clearly visible that when the larger pressure step occurs, the pressure increases over time when the hold phase is active. It can be well seen that this effect is much smaller, and close to 0 bar, when performing a small pressure step.

**Controller in the loop** Figure 5.26 shows the pressure over time, if a similar reference pressure is fed to the controller as earlier in figure 5.19.
Figure 5.26: Pressure and pump actuation over time for a certain pressure reference

From the figure it can be seen that the reference pressure is tracked quite accurately, and the mismatches are quite small. The pressure mismatches in this controller in the loop case are even lower than discussed in previous paragraphs, as the timespan is shorter, and brake line elasticity has too little time to significantly influence the pressure. Furthermore it can be seen, by looking at the front right brake pressure, that the master cylinder’s pressure fluctuates heavily during the experiment. The pressure drops when pressure is build, and increases when the pump is actuated.

Although the pressure is tracked quite accurately, the earlier set requirements 2 and 3 are not always met when the controller is used during experiments on the BMW. This can be well seen in some different controller in the loop experiments shown in figure 5.27 and 5.28.
Figure 5.27: Pressure and pump actuation over time for a certain pressure reference

Figure 5.28: Pressure and pump actuation over time for a certain pressure reference
5.5 Conclusions

Now the controller is tested in simulation and on the BMW, conclusions can be drawn with respect to the quality of the controller. The following statements reflect to the brake pressure controller in general, and are thus valid for both simulation and experiments:

- Due to the use of the reference based switching logic, the wave propagation issue is coped with, and chattering of the controller is avoided

- As only on/off control is possible, the desire to switch pressure as fast as possible is granted. However, the lack of proportional control also results in difficulties with respect to pressure mismatches

- Due to different characteristics of the hold and release valves, mismatches obtained when building pressure are significantly smaller than when releasing pressure

- Proportional control could improve the quality significantly, especially for small pressure decrease steps

- Optimal performance is obtained if pressure is set stepwise and not continuous, as the pressure mismatch compensation may otherwise fail

The following conclusions can be drawn with respect to the controller in simulation:

- Due to the mismatch compensation, quality of releasing pressure has significantly improved

- The in subsection 5.2.1 stated requirements are met for almost all situations. Requirement 3 ($p_{final} \leq p_{ref}$ when decreasing pressure) in some cases is not, but as the amount of occurrences is low, and mismatches are small this shouldn’t lead to any problems when used in combination with the force based ABS algorithm of chapter 6.

- The controller performs close to optimal for the current limitations of the system. Progression could still be booked at the small pressure decrease steps, by the use of (a different) feed forward control (or as mentioned before; proportional control)

- Results show that the pump scheduling based on the accumulator observer works well. The observed accumulator state is quite accurate, and always over-predicting, which ensures safe modeling

With respect to experimental results the following statements can be made:
• It can be clearly seen that the simulation tuned controller does not behave properly with respect to valve control. Mismatches are larger, and requirements 2 and 3 are not met in many cases. This latter is mainly due to the fact that the slow flexibility of the brake lines is not modeled, and thus not taken in account in the simulation tuning.

• The controller should thus be re-tuned for use on the BMW. Besides that, also the build pressure phase should be improved by adding a mismatch compensation in order to avoid negative mismatches.

• Although there is no method to check the actual state of the accumulator, the fact that pressure decrease phases did not lock up due to a full accumulator, strengthens the belief that the observer and pump scheduling are correct.

• Pump scheduling is however only proven to be valid in the lab conditions in which all tests were performed. As battery voltage significantly influences pump power, which is currently kept constant by a battery charger, it is well possible that actual field tests need different scheduling.
Chapter 6

Algorithm

In this chapter a novel force and pressure control based ABS algorithm will be presented. In chapter 1 two recently at TU Delft designed force based algorithms were discussed, and it was concluded that both algorithms make use of different advantages of force sensing. In this chapter a controller that makes full use of the advantages of force sensing is synthesized.

This chapter will start with the description of the basic theory on which the algorithm is based, as the method to control the slip derivative, $\dot{\lambda}$, is different than other in literature described ABS algorithms. Based on this theory, a two phase controller is described in the section that follows. Afterwards filtering and parameter tuning will be discussed, and the overall performance of the controller will be reflected upon. Then a numerical stability analysis will show that the designed controller is able to ensure stable cycling around the friction peak such that optimal friction (both lateral and longitudinal) is obtained. Furthermore it will be shown that the algorithm is able to handle basic ABS issues as load transfer and road surface transitions. The concludes with a conclusion and discussion on the quality of the algorithm.
6.1 Algorithm basics

In this section the basic theory on which the algorithm is based will be discussed. Firstly it will be explained how $\dot{\omega}$ can be controlled using longitudinal force measurements in combination with brake pressure control. Then it will be shown that this $\dot{\omega}$ control can be used to have basic control on $\dot{\lambda}$. The chapter finishes with a discussion on the influence of parameter uncertainty on the theory presented, and concludes with a work around method for this.

6.1.1 $\dot{\omega}$ control using longitudinal force measurement & brake pressure control

The first step in the direction of the novel algorithm is $\dot{\omega}$ control using longitudinal force measurement and pressure control. Figure 6.1 shows the free body diagram of a vehicle’s wheel, and thus shows all the dynamic components that have influence on $\dot{\omega}$. The following mathematical deduction shows that $\dot{\omega}$ can be controlled by brake pressure control and longitudinal force measurement.

![Figure 6.1: FBD of braking wheel](image)

The rotational dynamics of the wheel can be described with:

$$J \dot{\omega} = r_w F_x - T_b$$  \hspace{1cm} (6.1)

where

- $J [Kgm^2]$ = moment of inertia of the wheel
• $\omega [rad/s^2]$ = angular wheel acceleration
• $r_w [m]$ = radius of wheel
• $F_x [N]$ = longitudinal tire-road contact force
• $T_b [Nm]$ = braking torque applied by the brakes

The braking torque applied to the wheel is a function of the braking pressure, and can be described by:

$$T_b = P_b \mu_b A_b r_b$$  \hspace{1cm} (6.2)$$

where
• $P_b [Pa]$ = the brake pressure
• $\mu_b [-]$ = friction coefficient between brake pad and brake disk
• $A_b [m^2]$ = brake pads surface
• $r [m]$ = brake pads distance from center

Now by combining equation 6.1 and 6.2 one obtains:

$$J \dot{\omega} = r_w F_x - P_b \mu_b A_b r_b$$  \hspace{1cm} (6.3)$$

With the assumption that $J, r_w, \mu_b, A_b$ and $r_b$ are all constant, positive and known, $\dot{\omega}$ is controllable if $F_x$ is measured and $P_b$ is controllable. This is better shown by rewriting 6.3 to:

$$\frac{J}{\mu_b A_b r_b} \dot{\omega} = \frac{r_w}{\mu_b A_b r_b} F_x - P_b$$  \hspace{1cm} (6.4)$$

and thus:

\[
\text{if } P_b < \frac{r_w}{\mu_b A_b r_b} F_x \implies \dot{\omega} > 0 \hspace{1cm} (6.5)\\
\text{if } P_b > \frac{r_w}{\mu_b A_b r_b} F_x \implies \dot{\omega} < 0 \hspace{1cm} (6.6)
\]

In figure 6.2, this relation is shown graphically.
Figure 6.2: By the use of equation 6.5 and 6.6, the plane is split in two sections where $\dot{\omega}$ is either increasing or decreasing

### 6.1.2 $\dot{\lambda}$ control using $\dot{\omega}$ control

In the previous section, it was shown that by the use of longitudinal force measurement and brake pressure control $\dot{\omega}$ can be controlled. In this section, it will be shown that this $\dot{\omega}$ control can be used to control $\dot{\lambda}$.

When deriving the slip equation 2.2 presented earlier in section 2, one obtains:

$$\dot{\lambda} = -\frac{r}{v} \dot{\omega} + \frac{r \omega}{v^2} \dot{\phi}$$  \hspace{1cm} (6.7)

which relates $\dot{\lambda}$, $\dot{\omega}$ and several other variables. It is evident that if:

$$\dot{\omega} > \frac{\omega}{v} \dot{\phi} \implies \dot{\lambda} < 0$$  \hspace{1cm} (6.8)

$$\dot{\omega} < \frac{\omega}{v} \dot{\phi} \implies \dot{\lambda} > 0$$  \hspace{1cm} (6.9)

The range in which $\frac{\omega}{v} \dot{\phi}$ will vary can be determined by looking at $\frac{\omega}{v}$ and $\dot{\phi}$ separately. Using equation 2.2, $\frac{\omega}{v}$ can be rewritten to a $\dot{\lambda}$ dependant equation:

$$\frac{\omega}{v} = \frac{1 - \lambda}{r}$$  \hspace{1cm} (6.10)

Furthermore, also $\dot{\phi}$ is $\dot{\lambda}$ dependant:

$$\dot{\phi} = -\mu(\lambda) \cdot g$$  \hspace{1cm} (6.11)

The latter is true with the assumption that all wheels have similar wheel slip. Although still $\lambda$ is unknown, a safe prediction of $\frac{\omega}{v} \dot{\phi}$ can be made by
using the maximum values that can occur. Plotting eq. 6.10 multiplied by 6.11 shows these maxima:

![Graphical representation of multiplication of equation 6.10 by 6.11 for a slip value of 0 to 1](image)

Figure 6.3: Graphical representation of multiplication of equation 6.10 by 6.11 for a slip value of 0 to 1

In figure 6.3 it can be seen that the road surface with the highest friction, dry asphalt, results in the largest negative wheel acceleration at which $\dot{\lambda}$ is zero (about $-33\,\text{rad/s}^2$). This if the wheel acceleration is lower than $-33\,\text{rad/s}^2$, it can be assumed that for any road type the wheel slip increases. Furthermore for all road types a wheel acceleration larger than 0 will result in a decrease of wheel slip. Thus if:

$$\dot{\omega} > 0 \implies \dot{\lambda} < 0 \quad (6.12)$$

$$\dot{\omega} < -33 \implies \dot{\lambda} > 0 \quad (6.13)$$

Now combining equation 6.5, 6.6, 6.12 and 6.13 results in:

$$p_b < \frac{r_w}{\mu_b A_b r_b} F_x \implies \dot{\omega} > 0 \implies \dot{\lambda} < 0 \quad (6.14)$$

$$p_b > \frac{r_w \cdot F_x + 32 \cdot J}{\mu_b A_b r_b} \implies \dot{\omega} < -32 \implies \dot{\lambda} > 0 \quad (6.15)$$

which shows that the longitudinal force measurement combined with brake pressure control offers $\dot{\lambda}$ control. Figure 6.7 shows this control graphically.
6.1.3 Influence of parameter uncertainty

In the previous sections it was assumed that $J, r_w, \mu_b, A_b$ and $r_b$ are all constant, positive and known. Although the second statement is true, the first and last are not. This section will briefly discuss all variables, to determine in what degree they can be estimated and how large their variance will be. Finally the influence of this variability will taken in account, such that the previous presented theory is valid for the entire range of variance.

The wheel’s inertia $J$ can be assumed to be constant. Although tire wear could be influential, this most probably is not the case. The exact value of $J$ could thus be determined quite accurately on beforehand.

Wheel radius $r_w$ is not constant over time, and is difficult to measure. Main factors that influence the radius are wheel loading and tire pressure. Furthermore, exact measurement is difficult as $r_w$ is not simply the height of the axle with respect to the road. Although not going into depth regarding this issue, figure 6.5, which is taken from [14], clearly shows this. Although it is hard to exactly measure, a estimate of the order of $r_w$ can be made.
The friction coefficient between brake disk and brake pad, $\mu_b$, varies significantly, and its measurement is practically impossible on a moving vehicle. The friction coefficient is dependent on several variables, such as applied pressure, rotational speed and brake disk and pad temperature. The influence of temperature is most significant, as temperature increases significantly due to the energy dissipation. Depending on the materials used, influence of temperature varies significantly. This is clearly visible in figure 6.6, taken from [6], which compares different types friction materials with respect to friction coefficient and temperature change.

From the figure it can be seen that $\mu_b$ varies significantly, a fact that should be taken in account when using the theory presented earlier.

Both brake pad surface $A_b$ and its distance to the wheel’s center $r_b$ are constant and can be determined quite accurately on beforehand. These param-
eters can thus be assumed to be known and constant.

In practice it is probably easiest not to measure each variable individually, but instead, to determine the relation between brake pressure and longitudinal force directly using a special setup. However, the variability that can be found is logically still dependent on the variability of the individual components.

Now looking at the equations that allow $\dot{\lambda}$ control, equation 6.14 and 6.15, which will be used in the novel ABS algorithm, one can see that all parameters are either multiplied or divided by each other. As this is the case, it is possible to simplify the parameter variety by setting all variables but one constant. It is thus chosen to let $\mu_b$ vary, and let $J, r_w, A_b$ and $r_b$ constant. For the following of this report, it is assumed that $\mu_b$ is a value between 0.475 and 0.675. Because of this uncertainty in the value of $\mu_b$, relations 6.14 and 6.14 and figure 6.7 lose their validity and should be updated using the bounds of the friction coefficient $\mu_{b_{min}}$ (0.475) and $\mu_{b_{max}}$ (0.675). The new relations and figure now become:

\begin{align*}
  p_b &< \frac{r_w}{\mu_{b_{min}} A_b r_b} F_x \implies \dot{\lambda} < 0 \quad (6.16) \\
  p_b &> \frac{r_w \cdot F_x + 32 \cdot J}{\mu_{b_{max}} A_b r_b} \implies \ddot{\omega} < -32 \implies \dot{\lambda} > 0 \quad (6.17)
\end{align*}

Figure 6.7: Graph of the relation between longitudinal force, brake pressure and $\dot{\lambda}$ when taking in account parameter uncertainties

The new relations and figure now contain a much larger 'grey' area, in which it is unsure whether slip increases or decreases, due to the uncertainty of parameters.
6.2 Control logic

As discussed in chapter 1, the two main objectives of an ABS system are to limit longitudinal slip in order to maintain steerability, and following to this, also optimize braking force. In order to do so, the algorithm tries to maintain wheel slip close to the point of maximum friction, \( \lambda^* \).

In this section the control logic of the novel ABS algorithm is presented. The algorithm uses the previously presented method of \( \dot{\lambda} \) control in combination with force measurements to robustly cycle around optimum slip \( \lambda^* \). Firstly the basic idea of the algorithm will be discussed, followed by a more detailed description of the phases and triggering conditions. Then the activation logic will also be discussed, concluding with the presentation of the final control scheme.

6.2.1 Two phased control

The algorithm consists of two phases and a phase triggering mechanism. Within each phase the braking pressure is set in such a way that wheel slip either increases or decreases. In phase 1, the brake pressure is set such that wheel slip decreases (\( \dot{\lambda} < 0 \)) whilst the objective of phase 2 is exactly the opposite (\( \dot{\lambda} > 0 \)). A triggering mechanism, based on force sensing, ensures that phase 1 is triggered when slip has passed friction peak \( \lambda^* \) in positive direction (thus when \( \dot{\lambda} > 0 \)), and phase 2 is triggered when the friction peak is passed in negative direction (when \( \dot{\lambda} < 0 \)). The basic idea of this control is shown in figure 6.8

![Figure 6.8: General idea of the two phase controller](image)

The method to ensure a decrease or increase of slip during phase 1 and 2, as well as the phase triggering mechanism will be discussed in the following subsections.
6.2.2 Decreasing wheel slip

In previous sections it was shown that by measurement of the longitudinal force, a brake pressure reference could be determined that guaranteed a decrease of slip. Therefore in order to decrease wheel slip, brake pressure will be controlled such that this relation is met. Besides this relation, there are other requirements to the control in order to guarantee correct functioning of the system.

First of all, due limitations of the hydraulic system, the pressure reference should be changed stepwise in order to obtain optimal performance. Furthermore, pressure should be kept as high as possible, such that switching to a high pressure stage is performed in a minimal time window. Based on the requirements discussed, the following pressure reference control is formulated:

\[
p_{ref} = \frac{r_w}{\mu_{b\text{max}} A_b r_b} \cdot F_{xmin} - \Delta_p
\]

(6.18)

where \(p_{ref}[N/m^2]\) is the pressure reference fed to the brake pressure controller, \(\Delta_p[N]\) is a predetermined pressure offset tuned during simulation and \(F_{xmin}[N]\) is an indication for the minimal longitudinal force encountered during the entire phase. This latter variable is updated according the following scheme:

\[
\begin{align*}
\text{On Phase Entering} & \\
F_{xmin} &= F_x - \Delta F_-
\end{align*}
\]

(6.19)

\[
\text{During Phase} \quad \text{if} \quad F_x \leq F_{xmin} \implies F_{xmin} = F_x - \Delta F_-
\]

where \(F_x[N]\) is the current measured longitudinal force and \(\Delta F_-[N]\) is a predetermined force offset to ensure stepwise changing of the reference pressure.

The presented pressure reference complies to all requirements stated. First of all, due to the use of the basics of relation 6.16 and subtraction of \(\Delta_p\), pressure is set such that a decreasing wheel slip is ensured. Furthermore the pressure is changed stepwise by the use of the fictive \(F_{xmin}\) value, and pressure is kept high by subtracting a tunable variable \(\Delta_p\).

6.2.3 Increasing wheel slip

As for the decrease of wheel slip, in previous sections a relation between longitudinal force and brake pressure was deducted such that a increase of slip is guaranteed. Thus by controlling the brake pressure, and measurements of
the longitudinal force a positive $\dot{\lambda}$ can be ensured when relation 6.17 is met.

The limitation of the hydraulic system still stands, and thus a method of stepwise reference changing should be used. Furthermore it is important to set braking pressure as low as possible, such that switching to a low pressure stage is as quick as possible. Based on requirements stated, the following pressure reference control is formulated:

$$p_{ref} = \frac{32 \cdot J + r_w \cdot F_{xmin}}{\mu_{bmax} A_b r_b} + \Delta p$$

(6.20)

where $p_{ref}[N/m^2]$ is the pressure reference input for the pressure controller, $\Delta p[N]$ is a predetermined pressure offset tuned during simulation and $F_{xmax}[N]$ is an constantly updated indication for the maximum longitudinal force, which is updated according the following scheme:

**On Phase Entering**

$$F_{xmax} = F_{xmaxprev} + \Delta F_+$$

(6.21)

**During Phase**

if $F_x \geq F_{xmax} \implies F_{xmax} = F_x + \Delta F_+$

where $F_x[N]$ is the current measured longitudinal force, $F_{xmaxprev}[N]$ is the maximum longitudinal force encountered in previous phase and $\Delta F_+ [N]$ is a predetermined force step, to ensure stepwise changing of the reference pressure.

All the requirements are met by the proposed pressure reference setting, as first of all pressure is indeed higher than the relation stated in equation 6.17, second of all pressure is indeed changed by steps due to the $F_{xmax}$ updating method, and finally pressure is kept low by the tunable variable $\Delta p$. Figure 6.9, shows that the pressure indeed is changed stepwise, and that the controller is able to handle this correctly.
6.2.4 Phase switching

As could be seen in figure 6.8, phase switching should occur when the slip value is moving away from the optimal friction $\lambda^*$. For both phases this is characterized by a decrease in longitudinal force, as the friction peak is then being moved away from.

As we assume that all wheel forces are measured ($x$, $y$ and $z$), the measured longitudinal force can best be divided by the vertical force to obtain an indication of wheel friction, as in that case only force decrease due to slip change, and not load transfer, is measured. When this value is thus negative, it is certain that slip is moving away from $\lambda^*$.

For robust phase triggering furthermore also knowledge of the acting brake pressure is necessary; although brake pressure references in phase 1 and 2 guarantee a decreasing and increasing slip respectively, actual braking pressure may differ. Therefore for phase triggering it is checked whether the acting braking pressure indeed guarantees a decrease or increase of wheel slip during phase 1 and 2 respectively. This results in the following triggering conditions:

\[
\text{Trigger Phase 1} \quad \text{if} \quad p_b > \frac{r_w F_x + 32 J}{\mu_{\min} A_{rb}} \quad \text{AND} \quad \dot{\mu} < 0 \quad (6.22)
\]

\[
\text{Trigger Phase 2} \quad \text{if} \quad p_b < \frac{r_w A_{rb} F_x}{\mu_{\max} A_{rb}} \quad \text{AND} \quad \dot{\mu} < 0 \quad (6.23)
\]

As both the longitudinal and vertical force measurement contain noise, and thus $\mu$ is even more noisy, the signal needs to be filtered in order to obtain
Chapter 6. Algorithm

A correct estimation. This filtering, and its optimization is discussed in section 6.3.

6.2.5 Activation logic

The activation logic of the ABS system is similar to the triggering condition from phase 2 to 1. If the driver applied pressure is such that relation 6.17 is fulfilled and thus slip is increasing whilst a decrease in friction is detected, the slip has past the optimum, and thus the ABS system should trigger on to phase 1. Thus:

\[
\text{ABS On if } p_b > \frac{r_w F_x + 32 \cdot J}{\mu_{\text{min}} A_0 r_b} \text{ AND } \dot{\mu} < 0 \quad (6.24)
\]

6.2.6 Control scheme

Figure 6.10 shows the an overview of the complete control scheme of the novel algorithm.

![Figure 6.10: The two phased control scheme](image)

6.3 Filtering

As discussed in chapter 2, noise has been added to the measured force signals in order to increase simulation realism. The force signals therefore need to
be filtered, before being fed to the controller. This section is mainly focused on the filtering method to obtain $\dot{\mu}$, as it is most challenging and essential for correct functioning of the controller.

### 6.3.1 $\dot{\mu}$ filter

Obtaining a decent $\dot{\mu}$ signal is the most challenging filtering work within the controller. This is due to both the high noise level and the importance of low time delays. As by taking the derivative high frequency signals (which in this case is noise) are amplified, it is chosen to filter the $\dot{\mu}$ signal by a second order low pass filter. Hereby the high frequencies are attenuated, whilst the low frequency components maintain their phase lead. Essentially, a bandpass filter is thus obtained. Figure 6.11 shows the bode plot of the filter, of which the cutoff frequency determination, in order to minimize time delays, will be discussed next.

![Bode plot of filter to obtain $\dot{\mu}$](image)

**Figure 6.11**: Bode plot of filter to obtain $\dot{\mu}$

### 6.3.2 $\dot{\mu}$ filter tuning

The filter’s cutoff frequency will be determined experimentally using signals which represent actual friction signals that can be expected during ABS operation on three different road surfaces. In figure 6.12 one of the three signals used for cutoff frequency determination is shown.
The figure shows the evaluation of slip over time, the noisy friction measurement and the filtered result. As the filter is only focused on whether friction is decreasing, and not on the derivative’s magnitude, a threshold is used for detection. Whenever the filtered value exceeds this (negative) threshold, the filter is triggered and $\dot{\mu}$ is assumed negative. Determination of the best cutoff frequency and appropriate threshold is done based on three requirements:

1. The filter should not trigger when friction is constant or increasing (false trigger)

2. The filter should trigger during each significant friction decrease, as none of the friction drops may be unnoticed

3. Decreases in friction should be detected as fast as possible

Using the signals all these requirements can be evaluated on the three different road types, and therefore the best filter configuration can be found. The procedure performed is simple; for each cutoff frequency in the range of 10 to 200 rad/s the smallest (negative) threshold is determined that results in no false triggers within the first second (requirement 1). For this combination of cutoff frequency and threshold the average time delay is calculated, and the unnoticed friction drops are counted. All combinations that result in unnoticed friction drops are rejected (requirement 2). Figure 6.13 shows the results of the procedure.
From the figure it can be seen that very low (< 25) and higher (> 110) cutoff frequencies result in unnoticed friction drops (red), and therefore these results are rejected. Furthermore lowest time delays, and thus best results, are obtained at filtering cutoff frequencies between 80 and 100 rad/s. The best combination found was a filtering cutoff frequency of 90 rad/s, combined with a threshold of $-1.19$.

### 6.3.3 False triggering due signal lag

Due to filtering of the $\dot{\mu}$ signal, in order to attenuate higher frequencies, the signal will (logically) have a delay with respect to the original derivative signal. This delay may cause problems with respect to phase triggering, which is best explained using figure 6.14:

---

**Figure 6.13: Results of filter determination**

**Figure 6.14: Time delay between actual derivative and filtered derivative**
Chapter 6. Algorithm

The figure shows both the noiseless friction derivative, and the derivative obtained by the filtering method. It can be clearly seen that the result from the filter is shifted in time. Although this does not directly cause problems with the initial friction decrease detection, it may cause false triggering in the subsequent phase. This may happen if in that next phase pressure reaches the phase triggering criterion within the time that the filter still indicates a friction decrease due to time delay.

As this false triggering is highly unwanted, it is prevented by filtering the brake pressure that is used to detect phase triggering (conditions 6.22 and 6.23) with a similar second order low pass filter as the friction derivative. Hereby a similar lag is put on the pressure measurement, and so the issue discussed is avoided.

6.3.4 Longitudinal force filter

The longitudinal force measurements used to set brake pressure references also needs to be filtered, to attenuate noise within the signal. As the pressure is set stepwise, and only lowest and highest longitudinal force are tracked during phase 1 and 2 respectively, some noise is not problematic. Besides that the phase lag preferably should be as short as possible. It is chosen to, again, use a second order low pass filter. The crossover frequency is determined by engineering judgment, such that it phase lag is relatively low and most significant noise is attenuated.

6.4 Tuning of controller parameters

Besides the tuning parameters of the filter, three other parameters, namely $\Delta p$, $\Delta F_+$, and $\Delta F_-$, need tuning in order to obtain optimal results. In this section these parameters will be discussed, to obtain better insights of their influence.

6.4.1 $\Delta p$ tuning

$\Delta p$ ensures that relation 6.16 and 6.17 are satisfied during respectively phase 1 and 2, and thereby decrease and increase of slip is guaranteed. Furthermore due to this, also the pressure statements of both phase trigger conditions (equation 6.22 and 6.22) are satisfied.

The $\Delta p$ value is thus in theory, when using an ideal pressure controller necessary to comply to the discussed relations. However, as the brake pressure controller used shows significant pressure mismatches, pressure conditions are usually satisfied automatically. This latter can be seen in figure 6.15,
where final pressure is significantly higher when building, and significantly lower when releasing pressure.

\[ \Delta p \]

Figure 6.15: It is clearly visible that pressure mismatches are significant

As this inaccuracy thus in general automatically ensures fulfillment of the pressure relations, the \( \Delta p \)'s necessity in this case is lower. However, as mismatches in some specific cases may be slightly negative when building pressure, it is definitely needed to ensure stability in all cases. Ideally the \( \Delta p \) value is as low as possible, as thereby the slip cycle is most tight, which is clearly seen in figure 6.16

\[ \Delta p \]

Figure 6.16: It can be seen that a higher \( \Delta p \) results in larger cycles, and thus decreased performance

Although in simulations a \( \Delta p \) of zero did not cause any problems, theoretically they could arise due to the negative pressure mismatches in the build phase. As these negative errors are about \(-0.5\) bar at maximum, a \( \Delta p \) of \(0.5\) bar should suffice to ensure stability, and obtain best possible results.
6.4.2 \( \Delta F_- \) tuning

The \( \Delta F_- \) value is of influence during phase 1, and determines the step size of \( F_{x_{\text{min}}} \), which is used as input for the pressure reference setting (equation 6.19). Indirectly, \( \Delta F_- \) thus influences the reference pressure’s step size, which is clearly seen in figure 6.17.

![Figure 6.17: Influence of \( \Delta F_- \) on the pressure reference setting by the controller](image)

Left two graphs clearly show the pressure steps made, which are the result of the longitudinal force decreasing below the \( F_{x_{\text{min}}} \) references during the beginning of phase 1. Longitudinal friction decreases during the beginning of the phase as braking pressure is still too high. The tuning of the \( \Delta F_- \) should be done taking in account two important facts:

1. For optimal brake pressure controller performance it is best to set the reference pressure in one step, as mismatch compensations works best then. A larger \( \Delta F_- \) in this sense is thus better (figure 6.18)

2. Once \( \Delta F_- \) is large enough to ensure a single step reference, enlarging \( \Delta F_- \) will only result in a lower pressure setting, which causes an increase in algorithm cycle size (figure 6.19)
It can thus be seen that both a too small and too large $\Delta F_-$ results in suboptimal performance. The ideal value of $\Delta F_-$ is determined by the pressure decrease during the initial stage of phase one, which is dependent on several characters as road surface, vehicle speed and initial pressure of stage 1. A more thorough investigation in all these relations is needed to determine the ideal value for $\Delta F_-$. Most probably this value should be made
speed dependent to obtain best results.

### 6.4.3 $\Delta F_+$ tuning

The parameter $\Delta F_+$ is mainly influential when changes in road friction or loading occur which result in a larger $F_{\text{max}}$. It’s influence is best explained using figure 6.20, which shows the evaluation of pressure over time for several values of $\Delta F_+$, when the friction peak is higher than earlier measured ($F_{\text{max prev}}$).

**Figure 6.20:** Influence of $\Delta F_+$ on the pressure reference setting when the maximal longitudinal friction is higher than expected

From the figure it can be seen that in all three cases the pressure reference builds up in steps, which size is determined by $\Delta F_+$. Pressure mismatches are of minor concern in this issue, as first of all pressure mismatches while building are relatively small, and second of all the issue only affects those cycles in which road friction changes occur, or significant load transfers take place. The following two issues however are of significance when determining $\Delta F_+$:

1. The value of $\Delta F_+$ influences the speed at which the algorithm can adapt to new, higher, maximal longitudinal forces. A higher parameter value ensures faster adaption.

2. A larger value of $\Delta F_+$ degrades the algorithm performance in unchanging conditions (figure 6.21)
From the figure it is clearly seen that the limit cycle increases when the parameter $\Delta F_+$ increases. A trade off should thus be made with respect to the speed of adaption to new maximal longitudinal force and overall performance.

### 6.5 Algorithm performance

In this subsection the overall algorithm performance will be presented. As no intensive parameter tuning has been performed, results are not optimal. However, they do give a decent indication with respect to the algorithms performance. Figure 6.22 shows evaluation of brake pressure, slip and vehicle and wheel speed over time when a hard brake maneuver is initiated by the driver at an initial speed of 140 km/h.
From the figure it can be seen that the algorithm is activated when the friction peak has been passed. Brake pressure is quickly decreased and the algorithms continuous cycling between phase 1 and 2, to ensure slip cycling around the optimal slip. In the first 2.5 seconds this cycling is tight, later the cycles become larger due to the decreasing vehicle speed, and therefore increasing speed of the dynamics. The algorithm behaves as supposed, and slip cycling looks fine. Furthermore the figure shows that indeed speed influences the pressure step setting in phase one (subsection 6.4.2), as at low speeds an extra step is necessary. In figure 6.23, the first second of braking is shown in a longitudinal force - pressure plane.
The figure shows evaluation of brake pressure in relation to the longitudinal force. The dotted lines are the relations that ensure that wheel slip either decreases or increases. It can be seen that the algorithm controls brake pressure, such that either the the system is in the left ($\lambda < 0$) or right plane ($\lambda > 0$). In combination with phase triggering when longitudinal friction decreases, this ensures that slip cycles around the optimum.

6.6 Stability analysis

In this section it will be numerically proven that the presented control logic ensures convergence to a limit cycle, such that wheel slip cycles around the optimum and thus longitudinal brake force is maximized and lateral friction and stability are ensured. First the conditions on which the simulations have been performed will be discussed. This is followed by discussion of the limit cycle. Then the effect of the initial state of the system with respect to obtaining the stable limit cycle will be treated. Furthermore different road surfaces, as well as the influence parameter variability of $\mu_b$ will be discussed. This all is followed by the investigation of road surface changes and vertical load changes during ABS operation. It will be shown that due to the nature of the control logic, both phenomena are coped with well.

6.6.1 Analysis conditions

The stability analysis will be performed numerically using the in previous chapters presented hydraulic circuit and brake pressure controller. Furthermore only a single wheel will be discussed, as all wheels are controlled individually, and thus only one wheel needs to be considered for proving stability. Chassis speed is set constant and wheel relaxation dynamics will
be taken in account. Furthermore noise and thus filtering is added to increase realism. Initially vertical load and road surface will be set constant during simulation, but in subsection 6.6.4 it will be shown that the control logic is capable of coping with variations of them. Furthermore also $\mu_b$ is set constant, to a value in between minimum and maximum estimate $\mu_{bmin}$ and $\mu_{bmax}$. In subsection 6.6.4, influence of its variance will be treated separately.

### 6.6.2 Limit cycle

As the model contains a significant amount of disturbances, it is logical that the limit cycle varies slightly each time. Figure 6.24 and 6.25 show the evaluation of brake pressure and slip over time for a certain set of initial conditions on a wet asphalt and cobblestone road respectively.

Figure 6.24: Evaluation of the system over time in the slip - pressure plane for a wet asphalt road
In both figures it can be seen that the trajectory converges to a bounded region in state space which encircles the friction peak, the limit cycle. The size of the bounded region is dependent on both external conditions and controller tuning. As discussed before, each cycle is different due to external disturbances, but it can be clearly seen that a limit cycle exists.

### 6.6.3 Limit cycle stability

In order to determine the stability of the system it is interesting to examine what effect perturbations or different initial conditions have with respect to obtaining the limit cycle. A numerical analysis with respect to this is performed, by varying the following initial conditions:

- Slip (from 0 to 1, in steps of 0.1)
- Brake pressure (from ±5 to ±60 bar, in steps of ±5 bar)
- Controller phase (1 or 2)

The entire set of initial conditions is tested on both a cobblestone and wet asphalt road surface. For each simulation within the set, it is evaluated whether at the end of simulation the system has converged to the limit cycle or that the system evolved elsewhere. This latter is obviously unwanted, and controller is in that case thus not able to handle those specific initial conditions. Figure 6.26 and 6.27 show the results with respect to obtaining the limit cycle for all initial conditions discussed, for two different road types.
Figure 6.26: Limit cycle convergence for different initial conditions on a wet asphalt road

Figure 6.27: Limit cycle convergence for different initial conditions on a cobblestone road

In both figures, green dots represent the initial conditions which converge to the limit cycle and red dots imply initial conditions which do not converge and thus end up in either full wheel lock ($\lambda = 1$) or almost zero slip. It can be seen that the controller is able to achieve a stable limit cycle from a large set of initial conditions. For both road types, unstable points can be found at similar points in the pressure - slip graphs. In phase 1 these points lie on the 0 slip axis, while in phase 2 these points are located close to and at maximal slip. The reason that these initial conditions do not converge to the intended cycling is related to the phase triggering mechanism of the controller, which is based on the detection of a friction decrease. In both sets of initial conditions no significant friction decrease will be measured, and thus the controller will get stuck in it’s initial phase.

As all instable points are far from the actual operation range of the ABS system, it can be concluded that the controller in all reasonable situation results in a stable cycling around the optimal slip.
6.6.4 Robustness to other influences

The stability analysis discussed so far is based on the assumptions that $\mu_b$ is exactly in between estimates, the road surface doesn’t change and vertical loading maintains constant. In the following the influence of changes in these parameters is discussed.

Effects of changing $\mu_b$ The theory presented in section 6.1.1, 6.1.2 and 6.1.3, which concludes with a relation between brake pressure and longitudinal force such that the slip derivative can be set, is valid as long as the parameter variability lies between the expected bounds. Thus as long as the actual $\mu_b$ lies between the expected $\mu_{bmin}$ and $\mu_{bmax}$, the theory is valid, and pressure control of the control algorithm is correct. The latter was validated by performing a similar stability analysis as in previous section, this however will not be discussed, as it does not add any valuable information to this discussion. However, although variance of $\mu_b$ does not influence stability, it does influence characteristics of the limit cycle. The limit cycle differences will be discussed by the use of figure 6.28, which shows simulation results for 3 different values of $\mu_b$. These three values are both minimum and maximum predicted friction values $\mu_{bmin}$ and $\mu_{bmax}$, and the average of both $\mu_{bavg}$.

![Figure 6.28: Evaluation of slip and controller phase over time when brake friction $\mu_b$ changes](image)

Before discussing the differences in results, it is useful to reflect on the physical meaning of changing $\mu_b$. The controller parameters during this study
are not altered, and thus the settings of the brake pressure control remain similar. However, due to the fact that the brake friction parameter $\mu_b$ has changed, the braking torque applied to the wheels is significantly changed. A low $\mu_b$ logically means a lower braking torque, whilst a high $\mu_b$ means a higher braking torque. During the design of the controller it is ensured that for the entire predicted range of $\mu_b$, braking torques are low enough to decrease slip in phase 1, and high enough to increase slip in phase 2. However, changing $\mu_b$ will alter the margins, which is clearly visible in the results.

In figure 6.28 it can be seen that the evaluation of slip over time is significantly influenced by a changing brake friction coefficient. For a low $\mu_b$, slip remains more close to 0 while as $\mu_b$ increases the slip tends to move to higher slip values. Furthermore it can be clearly seen that the length of phases is significantly influenced; for a low $\mu_b$, phase 2 is longest, whilst at a higher $\mu_b$, phase 1 is clearly most present. These changes result from from the earlier discussed change in braking torque, and are best explained by comparing the evaluation of phase 2 during a low and high $\mu_b$ situation:

For a low $\mu_b$, the braking torque is relatively low, whilst a high $\mu_b$ means a much larger braking torque. Due to this, slip will increase much faster for the high $\mu_b$ situation. If now the friction peak is passed, the algorithm should be triggered to phase 1 to decrease slip again. As shown in table 6.1, triggering time delay decreases for higher $\mu_b$ situations, while the time needed to decrease braking pressure such that slip will decrease again ($t_d$ actuation) increases. It can be seen that the total time delay is quite similar for all friction situations, but as the slip derivative in the high friction situation is higher, a larger overshoot is obtained.

<table>
<thead>
<tr>
<th>$\mu_b$</th>
<th>$t_d$ trigger [s]</th>
<th>$t_d$ actuation [s]</th>
<th>$t_d$ total [s]</th>
</tr>
</thead>
<tbody>
<tr>
<td>Low</td>
<td>0.0664</td>
<td>0.0207</td>
<td>0.0871</td>
</tr>
<tr>
<td>Middle</td>
<td>0.0548</td>
<td>0.0310</td>
<td>0.0858</td>
</tr>
<tr>
<td>High</td>
<td>0.0462</td>
<td>0.0407</td>
<td>0.0869</td>
</tr>
</tbody>
</table>

Table 6.1: Time delays when switching from phase 2 to 1

For the phase switch from 2 to 1, exactly the opposite holds. Although in this case both triggering and actuation times are quite similar, as can be seen from table 6.2.
Although the results in this situation are quite clear, especially for switching from phase 2 to 1, it is not guaranteed that different road surfaces, filter tuning and other influences will provide similar results. For this, more thorough research would be necessary.

Furthermore if the value of \( \mu_b \) is set such that it falls outside the predicted region, or more natural, the predicted region is incorrectly set, stability cannot be guaranteed as the pressure control does not ensure decrease and increase of slip during phase 1 and 2 respectively. The limit cycle convergence in general does not directly fail when limits are passed, as both \( \Delta p \) and brake pressure controller add some margin, but logically the situation should be avoided.

Effects of different road surfaces Due to the nature of the controller different road types have no influence on the behavior of the controller, as long as the road characteristic shows a sufficient decrease in friction in both outward directions of the friction curve. This latter is due to the triggering nature of the algorithm, which is based on a measurement of friction decrease. The theory that ensures decrease and increase of slip in phase 1 and 2 respectively is independent on the type of road surface, and therefore no problems are to be expected when the road surfaces changes during braking. False triggering may however occur when large steps in measured friction take place, as it may take time for the filter to reinitialize.

In previous sections, it was already shown that a limit cycle can be obtained on both asphalt and a cobblestone road. In this part it will be shown through examples that transitions of road surfaces during control can also be coped with by the controller.

Figure 6.29 and 6.30 show a road surface change from wet asphalt to cobblestone and cobblestone to wet asphalt respectively.
Figure 6.29: Evaluation of slip and pressure over time, when after 1 second a road transition from wet asphalt to cobblestone takes place.
In both figures it can be seen that the controller is able to cope with the road transition. The transition from wet asphalt to cobblestone is smooth, and it can be seen that brake pressure is increased stepwise, as a higher friction is measured. In the transition from cobblestone to wet asphalt, a large peak in longitudinal slip can be observed. This is due to the large stepwise decrease of friction when the road surface changes, which causes the filter to falsely trigger phase 2 after about 1.1 second. This issue is related to the filter delay. Although false triggering occurs, it can be seen that the controller corrects itself, and finally a new limit cycle is obtained.

**Effects of load variations** Due to the design of the controller, vertical load variations which could result from load transfer or road irregularities do not influence the performance of the controller. Both the triggering as the method to obtain an slip increase or decrease are designed such that vertical load is of no influence.
The triggering condition is unaffected by vertical load, as friction, which is used as input, is independent on vertical load. Furthermore it can be seen that the pressure setting criteria in both phase 1 and 2 are independent on the vertical force. Indirectly pressure setting is influenced however, as longitudinal friction is related to the vertical loading on the wheel. This should furthermore not cause any problems, as can be seen in figure 6.31, which shows an example of a situation in which vertical load increases significantly (40%) during braking.

Figure 6.31: Evaluation of pressure and slip over time, when between 0.75 and 1.25 seconds vertical load is increased with 40%

From the figure it can be seen that the controller has no problem dealing with the increasing load. The pressure references increase, as longitudinal force increases due to the higher vertical force.

Note that in this simulation it is assumed that the vertical load has a linear influence on the dynamics. However, reality is different, as for instance saturation occurs at very high forces.

6.7 Conclusions and discussion

In this section the proposed algorithm will be discussed with respect to different aspects. First a discussion on the algorithm basics will take place, then modeling realism will be discussed followed by a discussion of algorithm results. Finally several future fields of interests regarding the algorithm will be summed up.
Algorithm basics  In this chapter a complete new method to tackle problems related to ABS control is presented. In opposition to the current ABS algorithms presented in literature, this novel algorithm does not use wheel speed measurements. Instead, it uses a combination of force measurements and pressure control, which has not been presented before. Based on this exotic measurement and control combination a control theorem is deducted, which is able to obtain optimal braking performance on a wide variety of road surfaces. Furthermore traditional problems of load transfer and road friction changes are automatically coped with, due to the nature of the controller.

The basic theory which is presented to obtain an increasing or decreasing wheel slip, and the method of friction peak passing detection can be used to design a variety of different controllers. In the controller designed in this chapter, it was chosen to take in account limitations of the SKF / TU Delft test vehicle, such that implementation is more feasible.

Although in this chapter it is proven that the novel ABS algorithm is able to offer robust control of the wheel slip, such that optimal longitudinal and lateral friction is obtained, the algorithm presented here should be mainly seen as a proof of concept. Parameters have not been optimally tuned and furthermore some important assumptions have been made, that still need validation in actual field testing. Most important factor regarding this is the measurement of forces, for which sensors are still in development. In the following paragraph the realism of the modeling done will be more thoroughly discussed.

Modeling realism  Most of the dynamics models used in this chapter, and presented in chapter 2, are taken from the traditional vehicle dynamics which is used in numerous vehicle braking related literary documents. Description of the wheel dynamics using the quarter corner model and tire to road contact using the burckhardt model and relaxation dynamics is well accepted. Furthermore, in order to increase realism, brake pressure dynamics are represented by the hydraulic circuit model presented in chapter 3, whilst brake pressure control is performed by the controller presented in previous chapter. This all leads to a relatively high realism vehicle description.

The basic theory used within the control logic is based on knowledge of certain parameters that link longitudinal force and braking pressure. At this moment it is assumed that these parameters, and their variance, is known. The measurement of the individual parameters, or perhaps easier, the entire relation, is in practice possible. However, probably variance found on commercial vehicles is much larger than assumed in this chapter, as it is chosen
to assume a relative flat disk brake friction coefficient based on figure 6.6, that is unlikely to be found in commercial vehicles.

The last significant remark with respect to the realism of the simulation is the assumed quality of the force signals. Although a significant amount of noise has been added and sampling is limited, the signal is likely still idealized with respect to actual force measurement by the SKF force sensing bearings. A more detailed specification of the sensors is needed to improve simulation realism.

Although both parameter variance and force measurements should be further investigated, the estimates made are reasonable, and are based on literature. In the following part the quality of the actual algorithm results will be discussed.

**Algorithm results** The functioning of the algorithm is proven by a numerical stability analysis, which shows that for all reasonable conditions a stable limit cycle is obtained. The algorithm causes the slip to cycle around the optimal friction $\lambda^*$, even when wheel loading changes or road surface transitions take place. The performance obtained is probably not optimal, as not all parameters have been optimized thoroughly. Their behavior however is reflected upon, to obtain a first insight in their influences.

Although it has been numerically shown and discussed that the algorithm is stable, and robust to several influences. A more analytical reflection on these should still be performed. A more thorough research should also be done with respect to changes of brake friction $\mu_b$, to investigate it’s relation to the limit cycle and phase lengths.

The filtering at this point is quite optimal for the signals present at this moment, but as discussed before, more knowledge of the force signals is needed to have a better understanding of what filtering is necessary. At this point the filters are again mostly a proof of concept, that the controller is able to cope with non-ideal signals.

**Future fields of interest** Already several interesting issues have been mentioned that should be solved, or could be interesting to improve algorithm quality. The following lists sums up the complete list of interests:

- At this point the algorithm is based on discontinuous control of the braking pressure. Continuous control could improve the quality of the algorithm
• The relation between longitudinal force and braking pressure should be more thoroughly investigated, to have a better idea of its variance. Furthermore it should be investigated whether for instance phase lengths (figure 6.28) could help determine its relation on-line.

• Analytical proof of stability, and robustness to external influences.

• What is the quality of the actual force signal, that is obtained by the force sensing bearings?
Chapter 7

Conclusions

This last chapter of the thesis will give a summary of the most important conclusions that resulted from the research, and will finalize with a set of recommendations for future research.

In chapter 3 and 4 a detailed model of a hydraulic circuit has been build and identified. The obtained model, which is based on a BMW 5 series hydraulic circuit, shows high correlation with experimental results, and is therefore very useful for brake circuit simulation purposes.

The hydraulic circuit model obtained was used for synthesizing a brake pressure controller for the BMW. In order to cope with system characteristics and limitations an intelligent reference based switching logic has been designed. The controller is tested in both simulation and experiments, and results show that, considering system characteristics and limitations, the method designed offers fine pressure control. For optimal performance on the actual test vehicle the controller should first be fine tuned. Furthermore quality of pressure control could significantly improve if proportional valve control was possible.

The hydraulic circuit and pressure controller presented offer a more realistic model of pressure dynamics and control than currently available in open literature, and can thereby improve ABS simulation realism significantly. As it takes in account the actual limitations of HAB circuits, it can be very useful for quality evaluation of ABS algorithms before field testing, or in later design stages.

In the last part of the thesis a force and pressure control based ABS algorithm, that is based on an innovative method to control wheel slip, is presented. Simulations using the brake circuit and pressure controller earlier deduced show that the new algorithm optimizes wheel slip, and a conver-
gence to a stable limit cycle is obtained for all common initial conditions. Furthermore it is shown that the algorithm is robust to road friction changes, brake friction changes and load variations. As the ABS controller is designed to work with electronics and hydraulic system present in the BMW, practical implementation is feasible.

In parallel to the theoretical work, the SKF/TU Delft test vehicle has been prepared for field testing by the implementation and testing of new sensors and electronics. At this point last hand is being laid on the implementation of force sensing bearings by SKF, after which innovative field testing can take place.

7.1 Recommendations for future research

Although a huge step has been made in the project, still a lot of interesting issues are up ahead. The following list sums up the most logical next steps within the project:

Brake friction coefficient
The quality of the novel algorithm is heavily dependent on the accuracy in which the brake friction coefficient can be determined. The test vehicle can be used to obtain a deeper knowledge on this matter. Furthermore methods for on-line determination or updating, by for instance using phase length information, should be investigated.

Force sensing bearings
After the implementation of the force sensing bearings in August, field testing should take place in order to obtain a better insight in the quality of the force signals. This knowledge will be of great value for the quality assessment of current force based ABS algorithms.

Proportional valve control
At this moment only on/off valve control is possible, which is a significant limitation upon the quality of brake pressure control. It is assumed that the hold valves offer the possibility to be controlled by PWM signals, however at this moment the setup misses the necessary electronics to do this properly, as current measurement is needed.

Brake controller tuning
In order to have optimal brake pressure control on the test vehicle, tuning on the vehicle itself should take place.

Re-identification of the front subsystem
Due to reasons unknown, the front subsystem’s characteristics have changed significantly since a leakage of one of the brake lines. Therefore the parameter identification should be (partly) performed again. Furthermore also the rear could be identified, although this has no high priority.
Bibliography


Appendices
Appendix A

Vehicle modifications

In order to be able to perform actual field tests for novel ABS algorithm experimentation, a BMW 5 series car has been acquired by the intelligent automotive division of TU Delft and SKF. This car has been modified, such that full control can be gained over original actuators in the hydraulic circuit of the car. Furthermore new sensors have been added and important signals with respect to the vehicle state are tapped. In this appendix a brief overview of the modifications is given, and the current setup is explained.

Figure A.1: The SKF/TU Delft BMW 5 test vehicle
A.1 Autobox

The center of all control and data acquisition within the test vehicle is a dSpace AutoBox. This computer, which has modular set of hardware boards to acquire and send signals, is positioned in the trunk of the car. It is connected to a laptop via a PCMCIA connection, with which programs (simulink models) can be loaded to the AutoBox, or real time data can sent (and logged) to the computer. The choice of hardware boards is based on the inputs and outputs necessary, and can be changed if needed. Figure A.4, in the overview subsection, shows which board are currently installed.

A.2 Hydraulic actuation

Connecting to the vehicle’s hydraulic components In order to perform ABS testing, control over the hydraulic circuit is needed. This basically means that control of the hydraulic actuators, which are located in the DSC (Dynamic Stability Control) unit of the car, should be acquired. In order to do so, the DSC unit has been reverse engineered. Due to this reverse engineering, a connection to all valves, the pump and the pressure sensor is obtained. Figure A.2 shows the basic layout of the DSC unit, and shows how reverse engineering has resulted in connections to the hydraulic unit which contains all hydraulic actuators.

![Figure A.2: Schematic view on how the hydraulic unit is controlled. The original cable has been led to a dummy DSC unit, such that errors in the CAN bus are avoided](image-url)
**Controlling the hydraulic actuators**  The control over the hydraulic actuators is possible by the use of the AutoBox computer. The signals generated by this computer are sent to the valve interface and power electronics, in which they are amplified. After amplification the signal is sent to the hydraulic unit, where valves and pump are actuated. The valve interface and power electronics are necessary, as the AutoBox’s signals have a too low voltage and current is limited.

**Dummy module**  As simply disconnecting the DSC unit would cause errors and thus problems on the CAN bus of the vehicle, it is chosen to equip the car with a second (dummy) DSC unit. Original cabling will be led to this dummy unit, which is not connected to the hydraulic circuit. Thereby the BMW electronics are in the presumption that no changes have been made.

![Diagram of DSC unit and valve interface](image)  
*Figure A.3: The position of the DSC unit and valve interface. The component to connect to the hydraulic unit is clearly visible. The dummy DSC unit is positioned underneath the valve interface.*

**A.3 Wheel speed measurement**

In order to obtain wheel speed information, the signal wiring from wheel encoder to DSC has been tapped. As the signal is current based, and the AutoBox only reads voltage based signals, conversion is needed. This signal conversion is performed under the hood of the car by the wheelspeed conditioner, from where they are also amplified and sent to the analog interface in the trunk. In the analog interface high frequency noise is filtered by squaring the signal, after which the signal is led to the AutoBox. The signal is
acquired and converted to a frequency by a PWM input on a DS4002 board.

### A.4 Brake pressure measurement

Brake pressure measurement is performed by four brake pressure sensors, one for each brake, which have been installed on the vehicle. These sensors, which have a measuring range of 0 to 100 bar, are connected by using a T-split bold at the brake calipers. Sensor signals are fed via the analog interface to the DS2003 board of the AutoBox. In this board A/D conversion takes place such that pressure can be read.

### A.5 CAN bus readout

As the CAN (Controller Area Network) busses of the vehicle contain valuable information with respect to the state of the vehicle, it is decided to read out the powertrain and chassis CAN busses. This is simply done by tapping off the CAN signal, and connecting it to the DS4302 CAN interface board.
A.6 Overview of electronics

Figure A.4 shows an overview of all components previously discussed, and shows their data connections.

Figure A.4: An overview off all modified and added electronics within the BMW test vehicle
A.7 Power wiring

Figure A.5 shows the power wiring to all electronics implemented.

![Diagram of power wiring]

From the figure it can be seen that all electronics are controlled from the inside of the vehicle, by the use of four switches:

- One switch turns on and off the entire added electric circuit (purple)
- One switch is used to turn on and off the battery charger (yellow)
- All electronics related to data acquisition and the AutoBox are switched by a third switch (green)
• The valve interface, and thus the electronic box that ensures actuation, is switched by the last switch. For safety reasons an emergency button, located on the dashboard, is placed in series with this switch.

A.8 Interfaces

The switches discussed before are positioned in the center armrest panel of the vehicle, the panel is shown in figure A.6. Furthermore figure A.7 shows the dashboard panel, which contains the emergency button and a LED display, which shows the activated electronics.

Figure A.6: The control panel with switches which is positioned in the center armrest

Figure A.7: The dashboard panel with indication LEDs and the emergency button to stop actuation