STS09039
Examensarbete 20 p
Oktober 2009

Application Pressure for an Electronic Brake System

Estimation and implementation of a recursive filter

David Moody

UPPSALA
UNIVERSITET

Teknisk- naturvetenskaplig fakultet
UTH-enheten

Besöksadress:
Ångströmlaboratoriet
Lägerhyddsvägen 1
Hus 4, Plan 0

Postadress:
Box 536
751 21 Uppsala

Telefon:
018 – 471 30 03

Telefax:
018 – 471 30 00

Hemsida:
http://www.teknat.uu.se/student

Abstract

Application pressure for an Electronic Brake System: Estimation and implementation of a recursive filter

David Moody

Scania CV AB is one of the major heavy truck manufacturers in the world. Many of their trucks use a brake system that is electronic where the driver demands retardation and the brake power is generated pneumatically. The objective of this thesis is to re-search if it is possible to determine the pressure at which the brakes start giving a retard-ing force, the application pressure, for the Electronic Brake System (EBS) and if a pre-dictive estimation of the application pressure can be implemented.

Three different ways of determining the application pressure are presented in the thesis and a signal processing method where the mean of pre-braking acceleration is used for change detection gives the most reliable results. This method is then implemented in a recursive filter where a predictive estimation of the application pressure is developed. Due to the lack of precision in the signals the estimation does not give a result that can be used and until this is solved by Scania it is difficult to see how a good way of pre-dicting the application pressure can be developed.

Handledare: Daniel Frylmark
Ämnesgranskare: Alexander Medvedev
Examinator: Elisabet Andrésdóttir
ISSN: 1650-8319, STS09 039
Sponsor: Scania CV AB

Table of contents

Terminology

This Section explains some of the terms used in the thesis. Abbreviations and mathematical notations are also described. Further explanations are made in the sections where the terms are used more frequently.

Acceleration
The rate of change of velocity *(Light and Matter, 2008-04-23).*
Application Pressure
Pressure at which point a wheel starts to brake.
Brake Calliper
Device in a disc brake system with which the pads are applied to the brake disc (*Scania CV AB:2, 2008-01-18).*
Brake Chamber
Cylinder for the piston in a vacuum and compressed air brake system (*Scania CV AB:2, 2008-01-18).*
Brake Disc
Disc in the disc brake system, rotating with the wheel (*Scania CV AB:2, 2008-01-18).*
Brake Pads
Pad made of friction material which is applied to the brake disc when braking. Brake pads come in pairs - one pad on each side of the brake disc (*Scania CV AB:2, 2008-01-18).*
Brake Lever
Arm that transmits the cylinder's motion to the brake camshaft (*Scania CV AB:2, 2008-01-18).*
Clutch
Device which disconnects the transmission from the engine enabling the gears to be changed (*Scania CV AB:2, 2008-01-18).*
Deceleration
In this thesis it is considered equal to negative acceleration.
Diaphragm
A circular device in rubber that is attached on the inside periphery of the brake chamber. When the air pressure is large enough it expands and causes movement in the rest of the Air Disc Brake system *(Harju, 2008-01-28).*
Drive Shaft
A driveshaft, driving shaft, or Cardan shaft is a mechanical device for transferring power from the engine or motor to the point where useful work is applied *(Sveriges Trafikskolors Riksförbund, s 87-95).*
Driveline
Synonymous with powertrain.
Electronic Brake System (EBS)
Brake system that is controlled using electronic actuators and sensors *(Palmertz, 2003).*
Electronic Control Unit (ECU)
Unit for the controlling of electrically controlled functions or accessories, such as axle weight limiter, speed limiter, auxiliary heater, brake pressure etc (*Scania CV AB:2, 2008-01-18).*
Electro-Pneumatic Modulator (EPM)
The EPMs are electro-pneumatic actuators. The EPMs receive the demanded brake pressure via brake-CAN from the EBS-ECU and sends out the brake pressures *(Palmertz, 2003).*

Engine
Machine which converts a non-mechanical energy into a useful movement energy (*Scania CV AB:2, 2008-01-18).*

Exhaust brake
Valve in the exhaust pipe that reinforces engine brake by stopping up the exhaust flow (*Scania CV AB:2, 2008-01-18).*

Final drive
The final drive transfers the driving force from the propeller shaft to the drive shafts *(Sveriges Trafikskolors Riksförbund, s 87-95).*

Flywheel
Heavy cast iron machine element mounted on the crankshaft which levels power impulses of the pistons (*Scania CV AB:2, 2008-01-18).*

Foot Brake Module (FBM)
FBM is connected to the brake pedal in the pedal assembly. The FBM sends two electric analogue signals and one switch signal to the ECU *(Palmertz, 2003).*

Gear Ratio/Conversion Ratio
Ratio between the rotational speeds of connected shafts (*Scania CV AB:2, 2008-01-18).*

Powertrain
Totality of components making up the power transmission system of a motor vehicle (*Scania CV AB:2, 2008-01-18).*

Propeller Shaft
Shaft which transfers torque from the power source to the central gear (*Scania CV AB:2, 2008-01-18).*

Push Rod
Rod that transmits axial movement from one part of a system to another when in compression (*Scania CV AB:2, 2008-01-18).*

Retardation
Like deceleration, it is considered equal to negative acceleration.

Retarder
Additional braking device which supplements the ordinary service brake system to avoid the fatigue failure of the latter when in continuous use (*Scania CV AB:2, 2008-01-18).*

Transmission
Device which allows to change rotation speed of the engine down or up and thus to choose between gear ratios (*Scania CV AB:2, 2008-01-18)* .

Abbreviations

ADB	Air Disc Brake
CAN	Controller Area Network
EBS	Electronic Brake System
ECU	Electronic Control Unit
EPM	Electro Pneumatic Modulator
FBM	Foot Brake Module
WECO	Western Electric Company (rules)

Mathematical Notations

Symbol	**Description**
A_d	Area of the diaphragm in the Air Disc Brake.
A_{front}	Area of the front of the truck.
a	Generic term for acceleration.
a_{veh}	Acceleration of the vehicle.
α	Angle of slope inclination.
c	Spring damping constant
$c_{airdrag}$	Aerodynamic drag coefficient
$E(X)$	The expected value of random variable X.
F	Generic term for force.
F_{acc}	Term that groups the accelerating forces of the vehicle.
F_{air}	Force generated by the air pressure and diaphragm in the Air Disc Brake
$F_{airdrag}$	Force generated by aerodynamic drag.
F_{brake}	Braking force generated by the braking system.
F_d	Spring damping force.
F_{dec}	Term that groups the decelerating forces of the vehicle.
$F_{engbrake}$	Braking force generated by the engine brake.
F_f	Frictional force.
F_N	Normal force.
F_{piston}	Force generated by the brake piston in Air Disc Brake.
F_{res}	Resulting force of F_{air} and F_{rod}.
F_{rod}	Force holding the push rod in the Air Disc Brake in place.
F_{roll}	Roll resitance force generated by the contact between the tyres and the ground.
F_s	Generic term for spring force.
F_{slope}	Force generated due to road inclination.
F_w	Driving force generated by the powertrain.
f_c	Function that defines the torque coming from the clutch in the powertrain.
f_d	Function that defines the torque coming from the drive shaft in the power-train.

Symbol	Description
f_{fd}	Function that defines the torque coming from the final drive in the power-train.
f_p	Function that defines the torque coming from the propeller shaft in the powertrain.
f_r	Constant used in determining the roll resistance force.
f_t	Function that defines the torque coming from the transmission in the pow-ertrain.
g	Gravitational constant.
I	Generic term for inertia.
I_{eng}	Inertia of the engine.
I_w	Inertia of a wheel.
i_t	Transmission ratio.
i_{fd}	Final drive ratio.
k	Spring constant.
k_{std}	Standard deviation constant used in Chebyshev's inequality.
L	Generic term for angular momentum.
m	Generic term for mass.
m_{veh}	Mass of the vehicle.
μ_f	Frictional constant.
p_{air}	Air pressure
$p_{est}(t)$	Estimated pressure. Used in pressure estimation method.
$p_{real}(t)$	Actual pressure. Used in pressure estimation method.
r	Generic term for radius/length.
r_1	Length of brake lever.
r_2	Length of brake lever from where brake piston is connected.
$r(t)$	Residual of estimation and actual value.
r_w	Wheel radius.

Symbol	Description
ρ_{air}	Density of air.
σ	Standard deviation.
T	Generic term for torque.
T_{brake}	Brake torque.
T_c	Torque coming from the clutch in the powertrain.
T_d	Torque coming from the drive shaft in the powertrain.
T_{eng}	Torque coming from the engine in the powertrain.
T_{fd}	Torque coming from the final drive in the powertrain.
$T_{f:eng}$	Frictional torque in the engine.
$T_{f:fd}$	Frictional torque in the final drive.
$T_{f:t}$	Frictional torque in the transmission.
T_p	Torque coming from the propeller shaft in the powertrain.
T_t	Torque coming from the transmission in the powertrain.
T_w	Resulting torque on the wheels from the powertrain.
θ	Generic term for an angle.
θ_c	The angle of the clutch shaft generated by torsion.
θ_d	The angle of the drive shaft generated by torsion.
θ_{eng}	The angle of the shaft connected to the engine generated by torsion.
θ_{fd}	The angle of the final drive shaft generated by torsion.
θ_p	The angle of the propeller shaft generated by torsion.
θ_t	The angle of the shaft connected to the transmission generated by torsion.
θ_w	The angle of the shaft connected to the wheel generated by torsion.
$\dot{\theta}$	Generic term for angular velocity
$\dot{\theta}_c$	Angular velocity related to the clutch.
$\dot{\theta}_d$	Angular velocity related to the drive shaft.
$\dot{\theta}_{eng}$	Angular velocity related to the engine.
$\dot{\theta}_{fd}$	Angular velocity related to the final drive.
$\dot{\theta}_p$	Angular velocity related to the propeller shaft.
$\dot{\theta}_t$	Angular velocity related to the transmission.
$\dot{\theta}_w$	Angular velocity related to the wheel.

Symbol	Description
$\ddot{\theta}$	Generic term for angular acceleration.
$\ddot{\theta}_{eng}$	Angular acceleration related to the engine.
$Var(X)$	The variance of random variable X.
v	Generic term for velocity.
v_{pr}	Velocity of the push rod’s movement in the Air Disc Brake.
X	Indicated random variable.
$\overline{X}$	The mean of X.
x	Generic term for distance travelled by object.
x_{pr}	Distance travelled by push rod.
$\dot{x}_{pr}$	Velocity of the push rod written as a derivative of distance.
$y(t)$	Measured value in model based diagnosis.
$y_{est}(t)$	Estimated value in model based diagnosis.

Part I

1. Introduction

Scania CV AB is one of the major heavy truck manufacturers in the world and sells vehicles in over a hundred countries (*Scania CV AB:1, 2008-04-23*). The company builds most of the parts in-house and assembles the vehicles themselves. One of the available brake systems on Scania's trucks and buses is today an electronic system where the driver demands retardation instead of brake power. The actual brake power is generated pneumatically.

The control of the system depends on many different factors and one thing that makes it more difficult is that the exact pressure at which the brakes start producing brake force, the application pressure, is not known. At the moment this pressure is assumed to be constant for the brake components used in all vehicles, but tests have indicated that the application pressure could vary between vehicles and in the brake components of a specific vehicle. The value of the application pressure is used in several different functions that are used in the brake system and affect the brake performance of the vehicle. It would therefore be of interest for Scania to see if it is possible to make a better estimation of the application pressure than the constant that currently is used.

1.1 Objective

The objective of this thesis is to research if it is possible to determine the application pressure for the Electronic Brake System that Scania develops and if a predictive estimation of the application pressure can be implemented using the measurable signals that are provided by the onboard computer.

1.2 Disposition

In order to reach the goal that has been set, several different tasks were performed in different phases and these phases outline the disposition of the thesis.

Firstly the method explains in detail how the objective should be reached and by what means. Secondly a theoretic base is presented and it explains more about how a brake system works from different aspects and theory about how the task can be solved. Finally the empirical results of the thesis, which have been a product of the theory and method, are presented and from this an analysis of the results are made. This leads to conclusions and suggestions for developmental decisions that Scania can make in the future.

1.3 Confidentiality

Scania is a company that conducts research which the company uses to improve its products. In order to have a competitive edge compared with other manufacturers of similar products it is essential to protect the ideas and inventions that the company develops. Because this thesis is conducted as a part of the brake systems development at Scania certain aspects have to remain confidential. This means that for example specific values that have been determined during the development process will not be revealed.

Instead these were altered to constants or values that only show the relations of the measurements.

2. Method

The process for developing an estimation of the application pressure goes through a few different phases. The initial goal is to build up a model of the system that is going to be analyzed so that an understanding of the problem can be reached. This was done by interviewing the experts on the different parts of the system at Scania and by dismantling and studying the brakes in the workshop where repairs and modifications of the brakes take place. The procedure was fairly informal where the various experts on the different areas could be reached at the workplace and questions could be answered when they arose. More formal meeting sessions were also conducted where the brake system was explained more in detail, but this was done as a presentation of the system where questions could be asked freely throughout the session. This type of interaction and constant close proximity to people with expert competency within the area of brake systems lead to the understanding of the problem and how to approach it.

The next phase was to consider how the objectives of the thesis could be reached. This was deemed to be most appropriate to divide into two steps: first to develop a method that can find a point of deviation for each wheel and second to use this to develop a way to estimate the application pressure for each wheel. In order to find a suitable method for solving the first step, three different methods were tried and evaluated before the one that was considered the most suitable was implemented in the second step. The methods that were examined were considered due to their ability of implementation in the brake system and through discussions with experts at Scania and literature study.

Parallel with the development of the methods data was collected in order to test the methods and see how well they work.

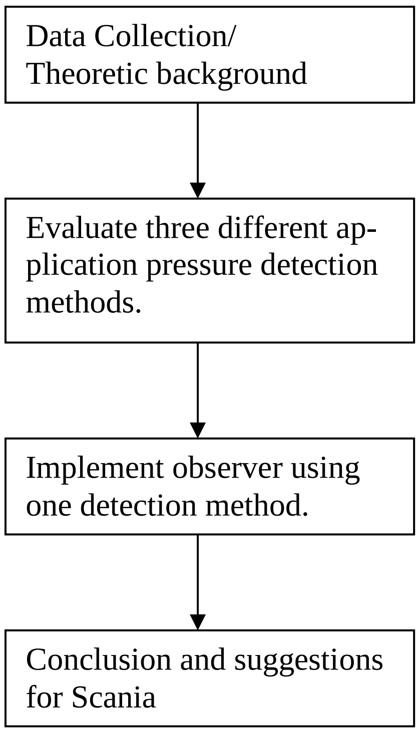

Figure 1.The figure shows the method that was used in the thesis. It displays the steps that were carried out and what order they followed.

2.1 Collecting data

The data collection process was conducted on the Scania testing grounds, a test circuit that simulates different types of roads, and on regular roads in the Södertälje and Stockholm area. The testing was performed on the truck Morbror, which can be seen in Figure 2. It had three wheel axles and the second axle was driven. The tag axle, the third axle, was not in contact with the ground when testing and therefore the truck only had four wheels that could brake. The truck carried weights on it to simulate real transporting scenarios. The brake system's electronic control unit (ECU), which controls the brake pressure, was connected to a laptop computer that made it possible to feed the wheels with different signals all depending on the type of test that was being conducted. Through the ECU the signals from different parts of the truck were collected and stored in a file on the computer. This meant that the signals were subjected to some filtering and were not raw, something that is discussed further in Section 6.3. The sampling frequency of the signals was set at 100 Hz. This level of accuracy was set after discussions with experienced testers at Scania and was considered to be appropriate for the study. A larger sampling frequency would mean that the data files would be too large to handle and tests would have to be shortened down.

When conducting the data collection some simplifications were made in order to reduce the number of variables that could influence the retardation. This made the distinction of the true application pressure easier to detect, but it was also important to be aware of that the reliability of the results was reduced. This was taken into consideration and the simplifications are presented in Section 2.1.1 to 2.1.6. The simplifications were not applied to all the data collection occasions and were minimized when possible.

2.1.1 Plane ground

The data collection was conducted on a regular asphalt road that did not have any hills or bumps of any greater magnitude. This was due to the fact that a hill would affect the retarding forces on the vehicle, explained further in Section 4.5.3, and bumps will also generate irregularities in the data that will make the detection harder. This isolates the retarding forces when braking occurs to only be dependent on aerodynamic drag and rolling resistance.

2.1.2 Constant velocity

As aerodynamic drag increases by the square of velocity when the truck moves, explained in Section 4.5.1, the velocity was also held at a constant before the brake pressure was applied. This reduced the disturbance from the aerodynamic drag on the retardation of the vehicle and made isolation of the forces that influence the vehicle less complicated.

2.1.3 No turning

When a vehicle turns the wheels on each side of it do not travel the same distance, because their radius when turning is different. The vehicle maintains the same speed but because the distance travelled is not the same, the velocity of the wheels is not the same. This lead to a disturbance in the retardation signal and in order to reduce this disturbance turning was minimized when conducting tests.

2.1.4 Free rolling

When a truck is driven forward a torque is generated in the powertrain, which gets transferred to the driven wheels of the truck. When taking the foot off the gas pedal and not applying any forward driving torque to the wheels the powertrain exerts a torque that generates a retarding force on the driven wheels, the engine brake as it is commonly known. The engine brake is explained further in Section 4.5.5. This affected the retardation and in certain data collection occurrences the retarding forces wanted to be isolated to reduce modelling and data errors. In order to reduce those influences the clutch was pushed down in order to disconnect the powertrain from the driven wheels and thus make the truck free rolling.

An assumption was made that there would be a certain time between the moment that the foot left the gas pedal until it pushed down the brake pedal. An empirical study showed that the fastest time was 0.7 seconds. One can therefore assume that the truck almost free rolled, if the forces from the powertrain were not included, before the brake pedal was pushed down. The free rolling simplification was therefore reasonable, even if it did not take into account the powertrain retardation.

2.1.5 Assumption that the ground is constant before braking

It was assumed that the ground, which the truck rolled on prior to braking had the same properties as when the truck started braking. That meant that the inclination, surface texture and material were the same during the 0.7 seconds before braking as they were after this period when the application pressure point was found.

2.1.6 Retarder and exhaust brake not included

The brake system in a truck does not only entail the wheel brakes. It also includes the retarder, the engine brake and the exhaust brake. The retarder is an extra braking device that is used to reduce the wear upon the main brake system. The engine brake uses the internal energy from the engine to slow down the vehicle. The exhaust brake complements the engine brake by reducing the exhaust and thereby braking the vehicle. In this thesis the retarder and exhaust brake will not be included when conducting tests and collecting data, but the engine brake is taken into account.

Figure 2. Morbror – the truck that was used for collecting the data used in the thesis.

2.2 Considerations for the Different Methods of Collecting Data

During the data collection process various different methods of collecting data were tried in order to see how the data responded to these modifications and if it could be used in the development process. In this section these different methods of data collection are presented.

2.2.1 Braking One Wheel

For purposes of detecting when the pressure was enough to start braking a vehicle the method of braking just one wheel was the easiest. This was because only one of the wheels demanded retardation and should therefore be the one that reacted first when the pressure was enough. The negative aspect of braking just one wheel was that it did not emulate an actual braking procedure when a person puts his or her foot on the brake pedal. It was however a good starting point for seeing how the acceleration of the vehicle responded to braking.

2.2.2 Braking All Wheels

When feeding all the wheels of the truck with the same pressure it created an environment that was closer to the true environment when braking. A wheel can start its retardation before the brakes have applied due to the fact that other brakes have applied to their wheels. This made it harder to detect when the pressure for a certain wheel was large enough for the brakes to apply and it demanded a greater precision from the methods of detection.

2.2.3 Braking One Axle

Braking one axle was another method that was used in this project and was somewhere in between the two previously explained ways of collecting data. It was more similar to braking one wheel as it just added one more wheel to the ones that were braked. But it has the advantage that it was possible to see how one wheel affected the other one's retardation if it braked first and made the tuning of the detection algorithm easier.

2.2.4 Altering the Application Pressure of the Wheels

A major concern when trying to detect when the brakes apply was the effect on the retardation of a wheel when another wheel has begun to brake. To extract data that showed this type of behaviour the properties of the brake chamber were altered so that the wheels on the front axle would need a larger pressure to begin to brake. This was done by shortening the length of the spring in the brake chamber by inserting discs that compressed it. In Section 3.2 the components of the Air Disc Brake are further explained. The intention was that when braking, the wheels on the drive axle would be caused to brake first and it would be possible to study how the wheels on the front axle behave when this happens.

Part II – Theory

In the second Part of this thesis the theoretical framework is presented. This Part will explain deeper how the brake system works and the theories that later are used in order to reach the objective.

3. What is EBS?

This section describes the Electronic Brake System (EBS) which is the type of braking system that will be able to use the more accurate representation of the application pressure. This will give the reader a better understanding of how the brakes work and therefore a better understanding of the rest of the thesis.

3.1 An Overview

The electronic brake system used by Scania consists of a few important parts that work together in order to get the vehicle to brake. These are the Electronic Control Unit (ECU), the Foot Brake Module (FBM), the Electro Pneumatic Modulators (EPM), air tanks and Air Disc Brake (ADB).Figure ***3*** shows an overview of a truck with the different parts marked out and the signals that are sent between them.

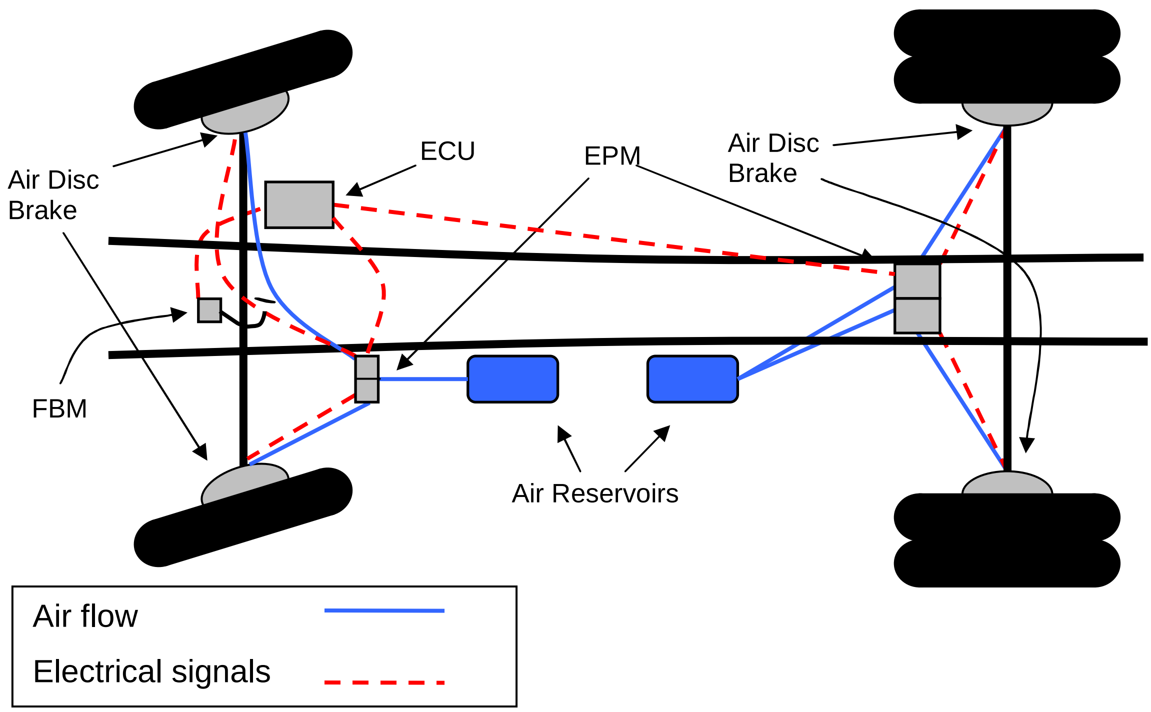

Figure 3. Schematic overview of the Electronic Brake System (EBS) in a truck. The red dotted lines show how the electrical signals are transmitted between the components and the blue lines show how the air flows from the reservoirs and is controlled from the EPMs.

The Scania EBS is a brake system that uses electrical impulses rather than hydraulics or pneumatics, which is common in other automotive braking systems, to control how much the vehicle should brake. The system is initially controlled from the FBM where the driver sends a signal demanding a certain retardation of the vehicle by pushing down the foot brake pedal. That signal is then processed by an ECU, which is a computer that translates the signal from the FBM to a certain pressure that should be applied to the brakes. This information about the pressures is sent to EPMs which apply the needed pressures by controlling the correct amount of air to the brake chambers in the pneumatic part of the system. The air is provided from air tanks located under the vehicle. Air is then sent at a certain pressure to the brakes which apply the force needed to brake the vehicle in the way the driver requested *(Palmertz, 2003).*

Unlike a hydraulic system in a car or a regular pneumatic system, which uses brake fluids or air to control the brake mechanics all the way from the pedal to the wheels, the force that has to be applied to the brake pedal is always the same in an EBS for the same retardation and not dependent on the weight of the vehicle *(Amberkar et al, 2000).*

3.2 The Air Disc Brake in the Electronic Brake System

The latter part of the system, the Air Disc Brake (ADB), was not electronic but pneumatical and mechanical. Pneumatics is the use of pressurized air to effect mechanical motion *(Wikipedia, 2007-11-08).* This part gives a more detailed view of the part of the EBS where the pressure influences the braking, which was pivotal for this thesis.

The Air Disc Brake could be divided into three parts: Brake Chamber, Brake Calliper and Friction Pair. Each of the three parts consisted of different components which are presented in Figure 4.

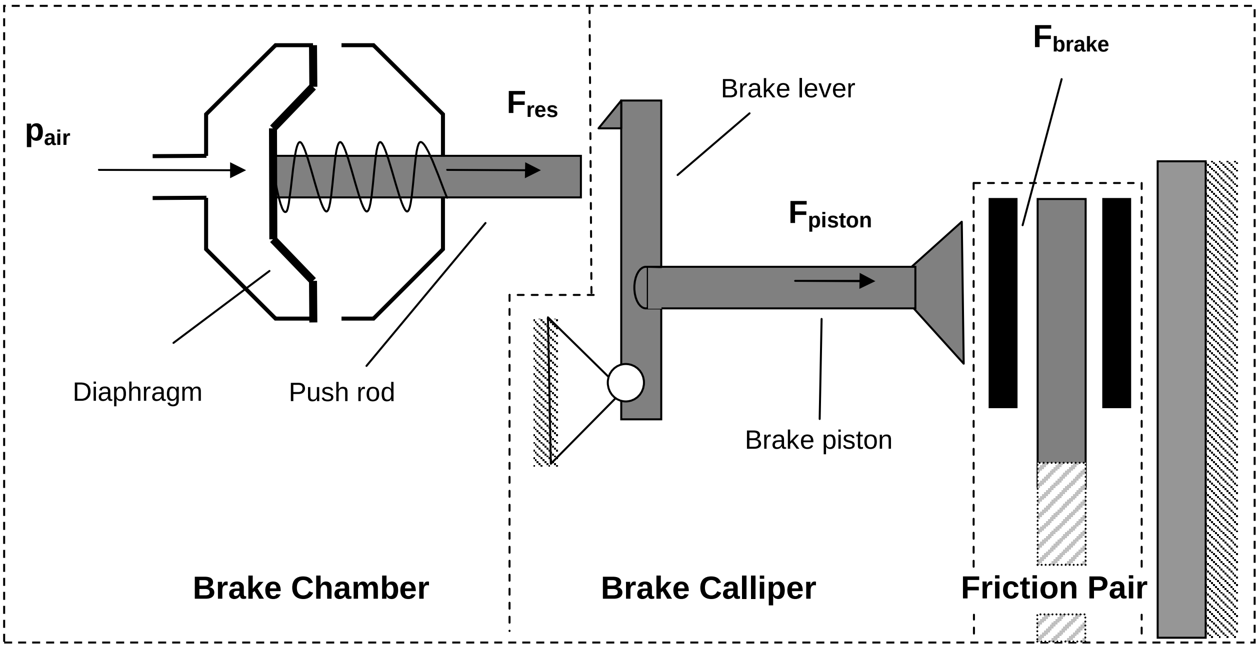

Figure 4. Schematic picture of the Air Disc Brake. The dashed lines mark the boundaries for the different parts of the system; Brake Chamber, Brake Calliper and Friction Pair (Based on Harju, 2008-01-28).

3.2.1 Brake Chamber

The brake chamber contained three important parts: diaphragm, return-spring and push rod. When the air pressure was fed from the air tanks and distributed from the EPM this made the diaphragm expand and a force made the push rod move forward. The push rod was fairly free moving but was controlled by a spring that was wrapped around it, which generated a force that had to be overcome by the force from the diaphragm if the push rod was going to be able to move *(Harju & Löfstrand, 2007-10-10).*

The dynamics that affected the brake chamber were primarily pressure and the movement of the push rod, which got its movement controlled by the return-spring wrapped around it. Pressure can be explained as the force applied on a unit area of a surface directed perpendicular to that surface *(Pytel & Kyusaalas, 1999a, p 401-405).* The surface was in this case the diaphragm in the brake chamber. The equation is written as

$$p_{air} = \frac{F_{air}}{A_d} \Leftrightarrow F_{air} = p_{air} A_d \,, \qquad (1)$$

where p_{air} was the pressure from the air tanks, F_{air} the force on the diaphragm and A_d the area of the diaphragm. The force that could move the push rod was therefore $F_{air,}$ but this had to overcome the force that held the bar in place before braking. The force F_{air} was not actually linear as it might appear because the area of the diaphragm changed when the force created by the pressure was large enough to make the push rod move. The area of the diaphragm could be written as a function of the length the push rod moved *(Harju & Löfstrand, 2007-10-10).* In this schematic representation this was simplified as it was not necessary with a more elaborate way of modelling.

A damped spring is a mechanical object that has different properties that also applied to the push rod in the brake chamber. The push rod itself was a body that did not act like a spring. However since it had a spring wrapped around it this caused its movement to act in a spring-like manner *(Harju, 2008-01-28).* A spring that is not fully stretched or compressed beyond its elastic limit obeys Hooke's law. The law says that the force with which the spring pushes back is linearly proportional to the distance from its equilibrium length and is written as

$$F_s = -kx, \qquad (2)$$

where k is the spring constant which is individual for each spring, x is the displacement vector with information about the distance and direction about the springs deformation and F_s is the resulting force vector which states the direction and magnitude of the force the spring exerts *(Pytel & Kyusaalas, 1999a, p 488).*

Damping is an effect that reduces the amplitude of oscillations of an oscillatory system. This also effects the spring and the push rod that its wrapped around. It is mathematically modelled as a force proportional with the velocity of the object but in the opposite direction

$$F_d = -cv, \qquad (3)$$

where c is the viscous damping coefficient, v is the velocity of the object and F_d is the resulting force vector. *(Pytel & Kyusaalas, 1999b, p 545-547)*

When combining Hooke's law with the damping abilities of a spring, a representation of the physical properties of the force holding the push rod in place, F_{rod} could be made

$$F_{rod} = F_s + F_d = -kx_{pr} - cv_{pr} = -kx_{pr} - c\dot{x}_{pr}\,. \tag{4}$$

This force had to be overcome by the force generated by air pressure to generate a resulting force, F_{res}, acting on the brake lever. The resulting force, F_{res}, was the force that was taken to the next part of ADB-system and is written as follows

$$F_{res} = F_{air} - F_{rod}. \tag{5}$$

3.2.2 Brake Calliper

The brake calliper had two main parts that influenced the braking of the vehicle: the brake lever and the brake piston. When the push rod moved it pushed with a force, F_{res}, on the brake lever in the brake calliper. The brake lever had the same sort of characteristics as a regular lever. The lever is an object that is used with an appropriate pivot point to multiply the mechanical force that can be applied to another object, as can be seen in Figure 5 *(Pytel & Kyusaalas, 1999a, p 33-86).*

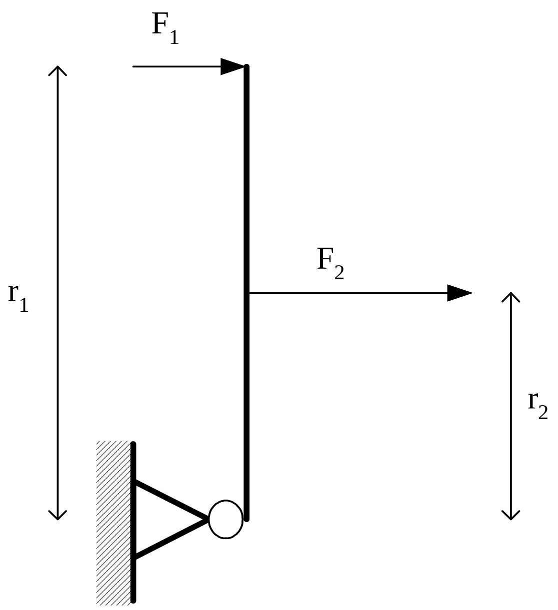

Figure 5. Explanation of how a lever works.

The equation is

$$F_1 r_1 = F_2 r_2 \Leftrightarrow F_{res} r_1 = F_{piston}\, r_2, \tag{6}$$

where F_1 is the force applied at one end of the lever at distance r_1 from the pivot point and F_2 is the force at the distance r_2 from the pivot point. This implies that if $r_1 > r_2$ then

$F_2 > F_1$ and this phenomenon is used by the brake lever in the brake calliper *(Pytel & Kyusaalas, 1999a, p 33-86).*

3.2.3 Friction Pair

The third part of the Air Disc Brake consisted of the two brake pads and the brake disc. In Figure 6 the friction pair can be viewed together with the other components in the electronic brake system. The brake piston pushed the brake pads, which came in contact with the brake disc, and frictional force caused the wheel to slow down.

The brake pads moved towards the brake disc with a force equal to F_{piston} and friction between the pads and disc was generated. Friction is the force between two surfaces in contact, which was what occurred in this case. The formula for frictional force is

$$F_f = \mu_f F_N = \mu_f F_{piston}, \qquad (7)$$

where μ_f is the frictional coefficient, which depends on the materials in contact, and F_N is the normal force, which in this case was F_{piston}. Heat was also generated, but since this was an ideal representation that gave a simplified view it was excluded *(Pytel & Kiusaalas, 1999b, p 303-306).*

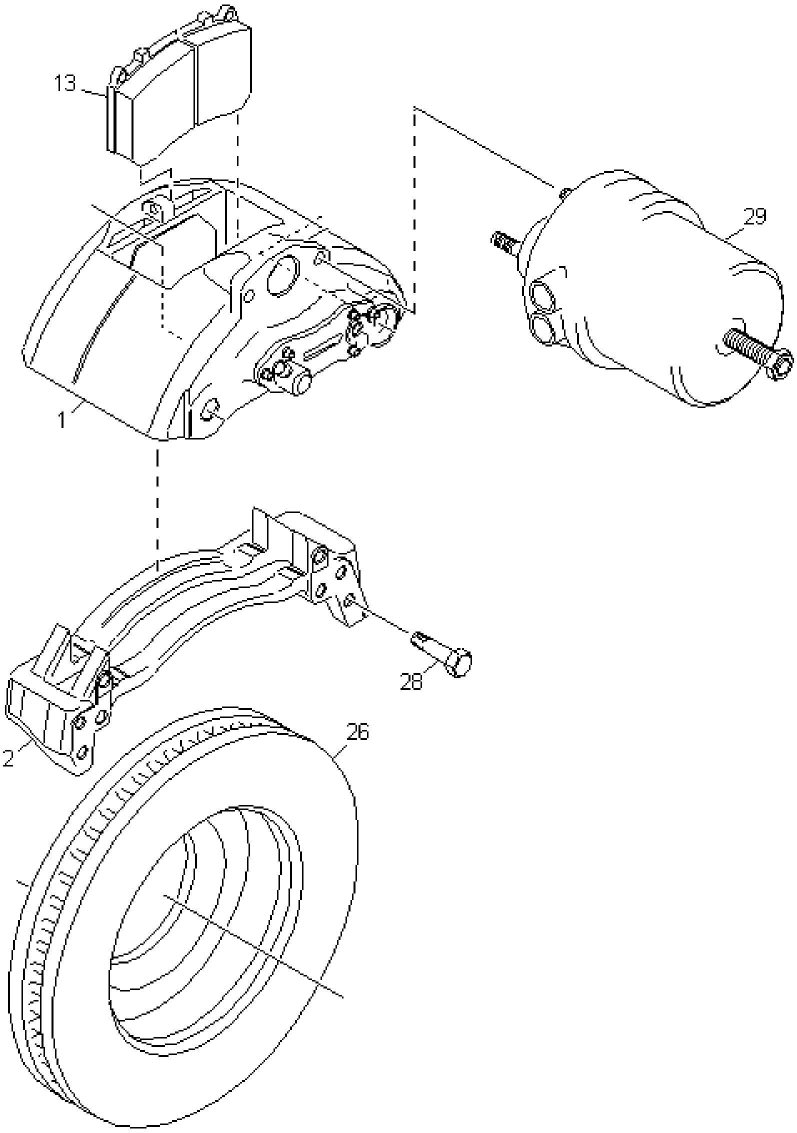

Figure 6. The figure shows some of the components in the air disc brake in the electronic brake system. The brake disc (26) and the brake pads (13), in this figure only one brake pad, make up the friction pair. The brake chamber(29)and the brake calliper (1) make up the other parts of the air disc brake.

3.2.4 Summary of the Air Disc Brake System

When all the above information was summarized the formula for the brake force looked as follows

$$F_{brake} = F_f = \mu_f \frac{r_1}{r_2}(p_{air}A_d - kx_{pr} - cx_{pr}) \,, \quad r_1 > r_2 \,. \tag{8}$$

The summary was simplified and losses due to friction or heat in the pneumatic system were excluded because they were not deemed necessary for the overall understanding of how the system works.

4. Wheel and Vehicle Dynamics

When causing a vehicle to move several different forces and other physical phenomena influence the direction and speed of the vehicle. The forces that were of interest in this thesis were the ones that were longitudinal and that drove the vehicle forward, the accelerating forces, and the forces that braked the vehicle, the decelerating forces. The forces that affect the vehicle laterally were for the sake of simplification not included.

In order to drive the vehicle forward the accelerating forces, $F_{acc,}$ had to be larger than the decelerating forces, F_{dec}. This gave a resulting force in the direction that the vehicle travelled that could be estimated with Newton's second law of motion:

$$ma = F_{acc} - F_{dec} \tag{9}$$

(Eriksson & Nielsen, 1998, p 109-111).

In Figure 8 these forces can be viewed. To fully understand how these forces are generated, some other physical phenomena have to be explained.

4.1 Moment of Inertia

Moment of inertia is analogous to mass but in rotational form. It is the inertia of a rotating body with respect to its rotation. The moment of inertia is defined as

$$I = mr^2, \tag{10}$$

where m is the mass of the object and r is the distance from the rotational axis to the point of mass *(Pytel & Kiusaalas, 1999b, p 347-351).*

4.2 Angular Velocity and Angular Acceleration

When a body moves in a circular motion it has an angular velocity. If the radius of the constant path it takes is r and the angle of the motion is θ then the velocity v of the body is

$$v = r\dot{\theta} \tag{11}$$

and the acceleration a that is tangent with the circle is

$$a = r\ddot{\theta}, \tag{12}$$

where $\dot{\theta}$ is the angular velocity and $\ddot{\theta}$ is the angular acceleration
(Pytel & Kiusaalas, 1999b, p 85-89).

4.3 Torque

Torque describes how a force can rotate a body around an axis. It is defined by

$$T = rF, \tag{13}$$

where T is the torque, r is the length from where the force, F, is perpendicular to the axis.

Torque is also the time derivative of angular momentum, L, which depends on the moment of inertia, I, and angular velocity and this gives the following relationship

$$T = \frac{dL}{dt} = \frac{I\dot{\theta}}{dt} = I\ddot{\theta} \tag{14}$$

(Pytel & Kiusaalas, 1999a, p 247-249, 344-346).

4.4 Accelerating Forces

The accelerating force, F_{acc}, is generated from the engine and through the driveline a forward driving motion of the vehicle can be obtained. In order to understand how F_{acc} is transferred from the engine to the wheels a schematic picture of the powertrain is presented in Figure 7. The powertrain consists of an engine, clutch, transmission, propeller shaft, final drive, drive shafts and wheels. The equations that explain how the driveline works are derived by using the generalized Newton's second law of motion:

$$F = ma. \tag{15}$$

Some of the equations that relate to the forces on the wheel are acquired by using the complete dynamics of the vehicle. This will mean that for instance the vehicle mass will be included in the equations describing the wheels *(Eriksson & Nielsen, 1998, p 109).*

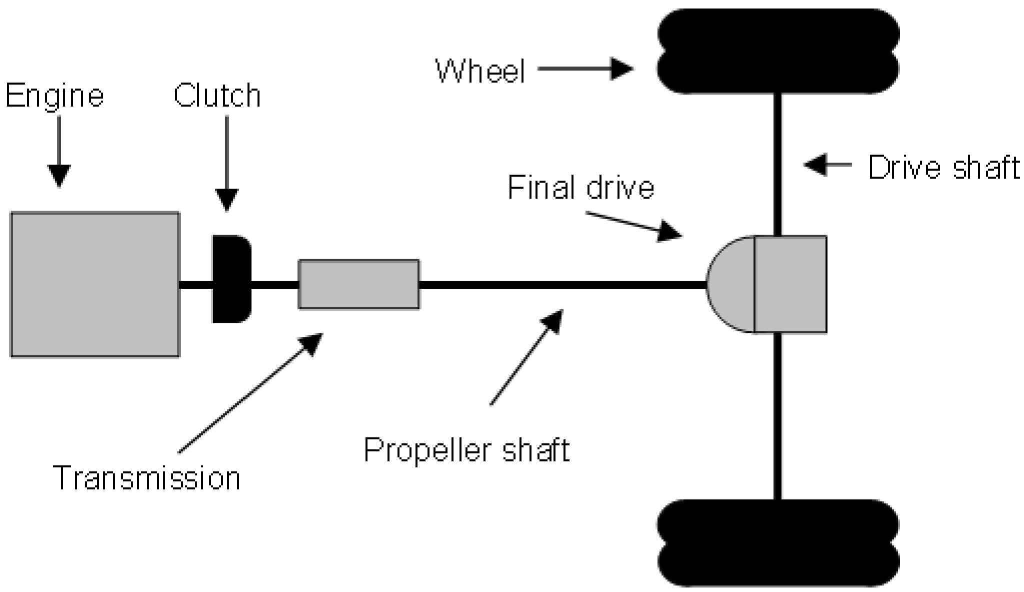

Figure 7. A rear-driven vehicular driveline (Based on Eriksson & Nielsen, 1998, p 110).

4.4.1 Engine

The outgoing torque from the engine is distinguished by the driving torque, T_{eng}, resulting from combustion, the engine's internal friction, $T_{f:eng}$, and the external load from the clutch, T_c. By using Newton's second law of motion the following equation is given

$$I_{eng}\ddot{\theta}_{eng} = T_{eng} - T_{f:eng} - T_c \,, \qquad (16)$$

where I_{eng} is the engine's mass moment of inertia and θ_{eng} is the angle of the flywheel, which is connected to the pistons in the engine *(Eriksson & Nielsen, 1998, p 111).*

4.4.2 Clutch

In vehicles which are equipped with manual transmission a clutch connects the flywheel of the engine with the transmission's input shaft. When the clutch is engaged, and the internal friction is assumed to be zero, $T_c = T_t$ is obtained where T_t is the torque of the transmission. The transmitted torque from the engine to the clutch and transmission is

$$T_c = T_t = f_c(\theta_{eng} - \theta_c, \dot{\theta}_{eng} - \dot{\theta}_c) \,, \qquad (17)$$

which means that it is a function, f_c, of the angular difference $\theta_{eng} - \theta_c$ and the angular velocity difference $\dot{\theta}_{eng} - \dot{\theta}_c$ over the clutch *(Eriksson & Nielsen, 1998, p 111).*

4.4.3 Transmission

A transmission consists of a set of gears and these each have a conversion ratio or gear ratio, i_t. The incoming torque is multiplied by this ratio, which is larger for the lower gears. This means that the engine has to make more revolutions for every revolution that the transmission makes and therefore more force is transferred for lower gears than for higher gears, which instead can get more revolutions from the transmission than from the engine. The relation between the input torque and the output torque of the transmission, T_p, is

$$T_p = f_t(T_t, T_{f:t}, \theta_c - \theta_t i_t, \dot{\theta}_c - \dot{\theta}_t i_t, i_t) \,, \qquad (18)$$

where f_t defines what the torque is a function of and $T_{f:t}$ is the internal friction of the transmission. The angular difference, $\theta_c - \theta_t i_t$, is considered due to the possibility of having torsional effects in the transmission *(Eriksson & Nielsen, 1998, p 111).*

4.4.4 Propeller shaft

The propeller shaft functions as the connector between the transmission and the final drive. When no friction is assumed, $T_p = T_{fd}$, the function, f_p, for the propeller shaft's torque is

$$T_p = T_{fd} = f_p(\theta_t - \theta_p, \dot{\theta}_t - \dot{\theta}_p) \,, \qquad (19)$$

where $\theta_t - \theta_p$ is the angular difference between the transmission and propeller shaft and $\dot{\theta}_t - \dot{\theta}_p$ is the difference in angular velocity *(Eriksson & Nielsen, 1998, p 111).*

4.4.5 Final drive

The final drive connects the propeller shaft with the drive shafts and has a conversion ratio, i_{fd}, in the same sense as in the transmission, but it only has one "gear" and the conversion is therefore constant. The output is as follows

$$T_d = f_{fd}(T_{fd}, T_{f:fd}, \theta_p - \theta_{fd} i_{fd}, \dot{\theta}_p - \dot{\theta}_{fd} i_{fd}, i_{fd}), \tag{20}$$

where f_{fd} defines what the final drive torque is a function of, T_d is the outgoing torque to the drive shaft, T_{fd} is the torque from the final drive, $T_{f:fd}$ the internal friction, $\theta_p - \theta_{fd} i_{fd}$ the angular difference and $\dot{\theta}_p - \dot{\theta}_{fd} i_{fd}$ the difference in angular velocity *(Eriksson & Nielsen, 1998, p 111).*

4.4.6 Drive shafts

With the drive shafts the wheels are connected to the final drive. In this case it is assumed that the wheel speed is the same for the two wheels and the drive shafts can be modelled as one shaft. If the vehicle turns and the speed is different for the two wheels, both drive shafts have to be modelled. With no friction assumed, $T_w = T_d$, the function of the torque, f_d, is

$$T_w = T_d = f_d(\theta_d - \theta_w, \dot{\theta}_d - \dot{\theta}_w), \tag{21}$$

where T_w is the resulting torque on the wheels and $\theta_d - \theta_w$ is the angular difference and $\dot{\theta}_d - \dot{\theta}_w$ is the difference in angular velocity *(Eriksson & Nielsen, 1998, p 111).*

4.4.7 Wheels

In Figure 8 the forces that act on the vehicle with mass m and speed v can be viewed. Using Newton's second law in the longitudinal direction gives

$$m_{veh} a_{veh} = F_w - F_{airdrag} - F_{roll} - F_{slope} - F_{engbrake} - F_{brake}, \tag{22}$$

where F_w is the driving force from the powertrain, m_{veh} the mass of the vehicle, a_{veh} the acceleration of the vehicle, $F_{airdrag}$ the aerodynamic drag resistance, F_{roll} the rolling resistance, F_{slope} the gravitational force due to the inclination of the road. $F_{engbrake}$ is the engine brake which brakes the vehicle when rolling and F_{brake} is the force from the brakes if they are used.

And with

$$F_w = \frac{T_w}{r_w}, \tag{23}$$

where r_w is the wheel radius an expression for the forces on the wheels can be gathered from equations (16) – (21) and by the following force expressions *(Eriksson & Nielsen, 1998, p 111).*

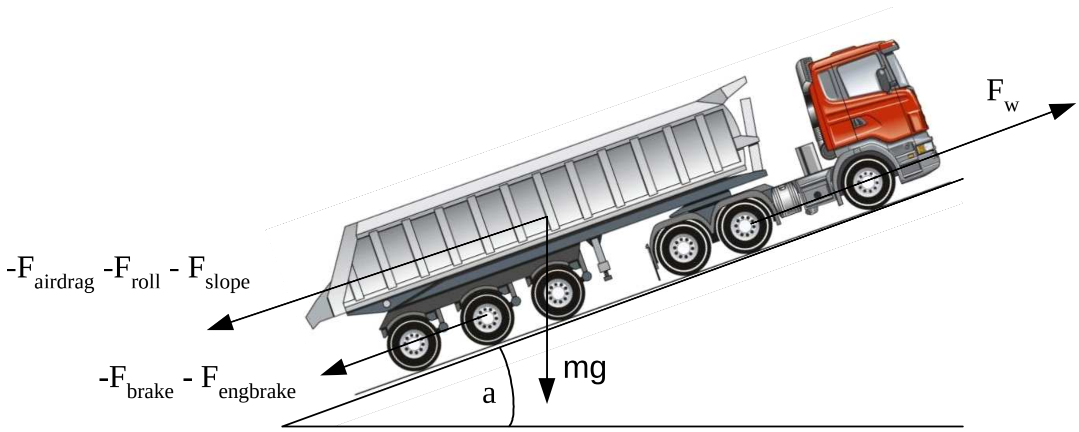

Figure 8. The figure illustrates the forces that affect the truck (Based on Eriksson & Nielsen, 1998, p 112).

4.5 Decelerating Forces

4.5.1 Aerodynamic drag resistance

Aerodynamic forces have an important impact on the motion of the truck. They can be divided into three components: drag, lift and side force. As they do not act on the centre of gravity they cause pitching, rolling and yawing moments on the vehicle (*Eriksson & Nielsen s 197).* As was explained in the beginning of the thesis a simplification was made which means that only the drag force will be of interest in the further studies as it works in the longitudinal direction. This force can be calculated as

$$F_{airdrag} = c_{airdrag} A_{front} \frac{\rho_{air}}{2} v^2 , \qquad (24)$$

where A_{front} is the front area of the vehicle, v is the velocity of the vehicle, ρ_{air} is the density of the air that the vehicle travels through and $c_{airdrag}$ is the drag coefficient (*Eriksson & Nielsen, 1998, p 197 - 198*).

4.5.2 Rolling resistance

The rolling resistance of the vehicle is caused by the deformation of the tyres. It depends on different factors such as the load on that tyre, tyre design, tyre pressure and velocity (*Eriksson & Nielsen, 1998, p 198*). The factors that have been considered in this case are the load on the tyre and velocity, the other ones have been considered constant in the tests made.

Modelling of the rolling resistance can be done with the equation

$$F_{roll} = f_r F_N = f_r m_{veh} g , \qquad (25)$$

where F_N is the normal force of the vehicle on the tyres and f_r is dependent on the above mentioned factors and can usually be found in the range 0.01-0.03 for a passenger car and less for a truck (*Eriksson & Nielsen, 1998, p 198*).

4.5.3 Slope inclination

The slope of the road has an effect on the motion of the vehicle. If the vehicle is driving uphill this will cause a decelerating force and the opposite will occur if the vehicle is going downhill due to gravitation. The weight of the vehicle and the slope α effect this force, where α = 0 is a flat road and α = 90° is equivalent of a wall and the equation is

$$F_{slope} = m_{veh} g \sin(\alpha) \tag{26}$$

(Eriksson & Nielsen, 1998, p 112).

4.5.4 Brakes

The force that the brakes have on the wheel is the generated in the way that was presented in Section 3.2 about the pneumatic part of the brake system and the equation is the same as in equation (8).

4.5.5 Engine brake

The engine brake has a decelerating effect on the vehicle and occurs when the driver is not pushing down the accelerator pedal but the engine is still producing energy in the opposite direction compared with when the driver pushes the accelerator pedal. This means that the engine generates torque on the wheels that works in the same way as was presented in equations (16) – (21), but in the opposite direction and the torque of the engine brake is equal to equation (21) and the force is the same as in equation (23). If the accelerator pedal is pushed down and the engine is exerting “positive” torque on the rest of the powertrain the force from the engine brake will be zero *(Lu & Hedrick, 2005).*

4.6 Summary of the forces acting on the vehicle

If the clutch, propeller shaft and drive shaft are assumed to be stiff, this will mean that from the engine to the wheel only the conversion ratios in the transmission and the final drive will be included in the calculation of the torque on the wheel. Stiffness means that the angular differences that occur between the components in the powertrain are not included in the expression. This will give a formula for the torque on the wheel due to the engine which is

$$T_w = i_t i_{fd} T_{eng} \tag{27}$$

and a force that is

$$F_w = \frac{T_w}{r_w} = \frac{i_t i_{fd} T_{eng}}{r_w} \tag{28}$$

When summarizing all the accelerating and decelerating forces this gives

$$m_{veh}a_{veh} = \frac{i_t i_{fd} T_{eng}}{r_w} - c_{airdrag} A_{front} \frac{\rho_{air}}{2} v^2 - f_r m_{veh} g - F_{engbrake} - F_{brake} \qquad (29)$$

$$\Leftrightarrow$$

$$a_{veh} = \frac{i_t i_{fd} T_{eng}}{r_w m_{veh}} - \frac{c_{airdrag} A_{front} \frac{\rho_{air}}{2} v^2}{m_{veh}} - f_r g - \frac{F_{engbrake}}{m_{veh}} - \frac{F_{brake}}{m_{veh}} \qquad (30)$$

(Eriksson & Nielsen, 1998, p 112 - 114).

5. Statistical theory/change-point analysis

This chapter presents some of the statistical methods that have been used in order to generate a solution for finding the application pressure. The theories presented deal with change detection in a data sequence, which means finding the point when the data starts to change, as shown in Figure 9. The reason for this being interesting is because when studying the data sequence of the acceleration curve there is always a short time span before braking that the vehicle rolls, the pre-braking phase which is defined in Section 7.3.1. During this time period the retardation of the vehicle will be fairly constant until the brakes start to take effect and at that point the application pressure will be able to be found.

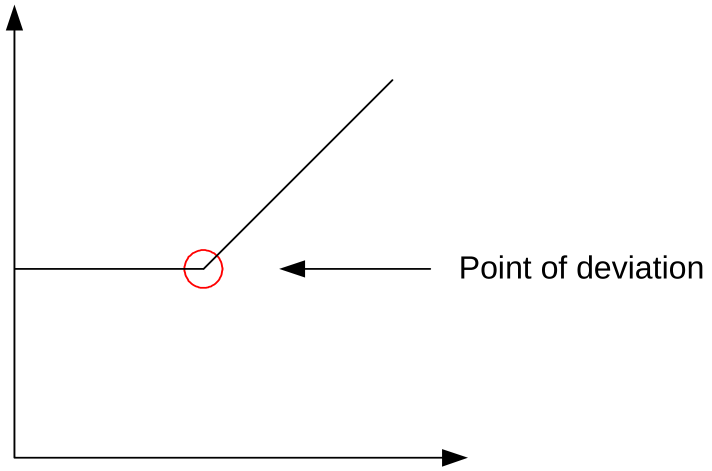

Figure 9. The red circle shows where the curve deviates and that indicates the point of deviation.

5.1 Statistical Terminology

This Section defines some of the terminology that will be used later in the thesis that is important to fully comprehend in order to understand how the methods used work.

5.1.1 Mean

The arithmetic mean is calculated by dividing the sum of all numbers in list with the number of items in the list *(About.com: Mathematics, 2008-01-30).* The formula for the mean is

$$\overline{X} = \sum \frac{X_i}{n}, \quad i = 1,\ldots,n, \tag{31}$$

where X is a vector of numbers, X_i is a number on the i:th position in the vector and n is the number of items in the vector.

5.1.2 Standard Deviation

The standard deviation measures the spread of the values in a data series from the mean of the values. The formula for the standard deviation is

$$\sigma = \sqrt{E((X - E(X))^2)} = \sqrt{E(X^2) - (E(X))^2} = \sqrt{Var(X)}\,, \tag{32}$$

where X is a random variable, E(X) is the expected value of X and Var(X) is the variance of X *(NIST/SEMATECH e-Handbook of Statistical Methods, 2008-01-13).*

5.1.3 Moving Average Filter

The moving average filter is used for calculating the mean of the n latest data points. When using the moving average filter the number of points, or window size, can be altered depending on how the data is going to be used. It also works as a low-pass filter to smooth noisy data. There are different types of moving average filters, but in this thesis the simple moving average filter has been used. It looks just like when calculating the mean as in equation 31, but it moves over the data series with a time window continuously calculating a new mean *(NIST/SEMATECH e-Handbook of Statistical Methods, 2008-01-13).*

5.2 Statistical Process Control

Statistical process control (SPC) is usually used in industry as a method of controlling the quality of the products that are manufactured. Quality is something that varies between products but when producing a specific product it is of the interest of the producer to make sure that the quality stays at a certain level. If the quality of an item that is produced varies from the rest of the production line this is of interest to detect and if it is faulty it will be separated from the rest of the produced items. And this is where SPC comes in. With SPC there is theoretic framework for detecting when data deviates from the expected or wanted value and this is something that can be applied not only in quality control but in other areas too as it is deviation that is the common ground *(Basseville & Nikiforov, p 23-44).* This was the reason for using SPC in the thesis.

Other possible methods to use to detect change are for example CUSUM and Likelihood-ratio test *(Schechtman et al, 2007).* Due to the limitations of the thesis these were not further investigated.

5.2.1 Shewhart Control Chart

The Shewhart control chart was developed as the first form of statistical method to determine quality in a process by detecting deviations. The control chart uses upper and lower control limits to detect whether a deviation has occurred. These control limits were set at three standard deviations below and above the mean of the data extracted from the process. The reason for choosing three standard deviations was that according to Chebyshev’s inequality that states that nearly all the values in any data sample are close to the mean value according to

$$P(|X - \overline{X}| \geq k_{std}\sigma) \leq \frac{1}{k_{std}^2} \tag{33}$$

and the Vysochanskii-Petunin inequality that refined Chebyshev’s inequality for a random variable with a finite variance in a unimodal distribution according to

$$P(|X - \overline{X}| \geq k_{std}\sigma) \leq \frac{4}{9k_{std}}\,, \quad k_{std} > \sqrt{\frac{8}{3}}\,, \tag{34}$$

where X is a random variable, k_{std} is the number of standard deviations and σ is the standard deviation *(Vysochanskij & Petunin, 1980).*

These inequalities state that the probability of a random variable deviating from the mean is less than 1/9 = 0.1 for Chebyshev's inequality and less than 4/81 = 0.05 for the Vysochanskij-Petunin inequality. Empirical studies also showed that less than one percent of the values were situated more than three standard deviations from the mean, which is due to the fact that for a normal distributed data sample about 0.003 of all values lay three standard deviations from the mean. The basis for the control chart can therefore be considered fairly conservative in comparison, but with the ability of a more general application *(NIST/SEMATECH e-Handbook of Statistical Methods, 2008-01-13).*

5.2.2 Western Electric Company Rules for Detecting Non-Random Behaviour

The Western Electric Company Rules (WECO) for detecting non-random behaviour were developed as a quality control mechanism at the Western Electric Company and have there basis in the Shewhart control chart. They are a set of rules that have been used to detect when a point deviates from the mean. If one of the rules is true, this means that a deviation can be said to occur. The rules are depicted in Figure 10 and are stated like this:

1) Any point deviating with three standard deviations from the mean
2) Two out of the last three points deviating from the mean by two standard deviations
3) Four out of the five last points deviating from the mean by one standard deviation
4) Nine consecutive points on the same side of the mean

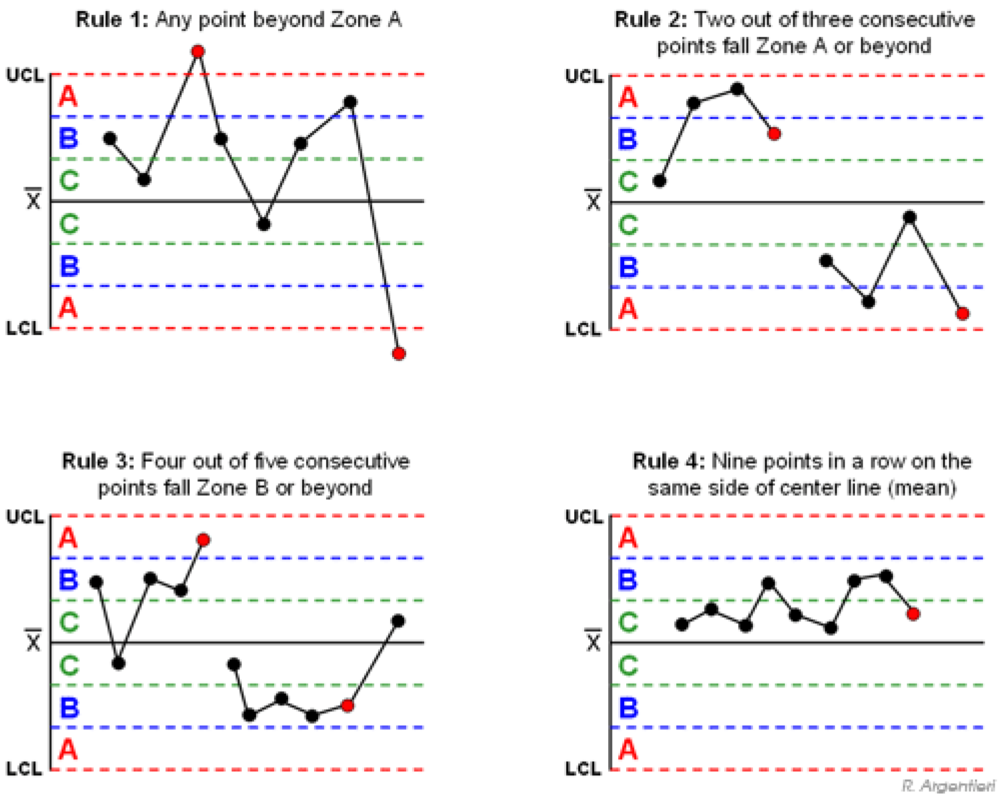

Figure 10. The graphs show the four Western Electric Company Rules and when a curve is considered to deviate, marked with red. The zones represent the distance from the mean where zone C is within one standard deviation of the mean, zone B is within two standard deviations and zone C is within three standard deviations. UCL is the Upper Control Limit and LCL is the Lower Control Limit (Argentieri, 2007-12-14).

The rules are based on probability, in the same sense that they were for the Shewhart control chart. For a normal distribution there is a probability of finding a point outside of $\pm 3\sigma$ of 0.3%, which is quite rare and this is an indication of a shift in the process. The similar case exists for the other rules that all have a probability of about 0.3% of occurring *(NIST/SEMATECH e-Handbook of Statistical Methods, 2008-01-13).*

5.2.2.1 False alarms

In the regular Shewhart control charts there exists a small risk of detecting false alarms, which are points that are not deviating due to non-random behaviour. For the Shewhart charts this happens on average every 371 points. When implementing the WECO rules this risk increases to a false alarm for every 91.75 points. This is something that the user of the WECO rules has to be aware of and something that is commented later in this thesis in the implementation chapter *(NIST/SEMATECH e-Handbook of Statistical Methods, 2008-01-13).*

Part III – Empirical studies

The theoretical groundwork has now been laid and in the third Part the empirical studies will commence. This Part will go through the methods used to reach the objective and how an estimation of the application pressure can be made.

6. Data considerations

This chapter addresses the data that was collected during the development process. It is central to understand what type of signals that have been used in order to grasp the researched methods and how they have been developed. The chapter also discusses the limitations that the data consists of and what kind of manipulations that were tried during the development process.

6.1 Signals

In this section the signals that were used for data collection are presented.

Velocity
The velocity of every wheel was used, as it was important for determining when the brakes apply to the wheels. The velocity also enabled a way to determine the acceleration, which is discussed further in Section 6.2.

Brake Pressure - Measured
The air pressure that was fed to the brake chamber was measured as it was crucial to know what the pressure actually was when the brakes had been determined to apply.

Brake Pressure - Requested
The requested brake pressure signal was the pressure signal that stated how much pressure actually was requested, either by pushing down the brake pedal or by computer in experimental environments. This was useful as a control to see if the measured pressure reached the value it was supposed to. This data was not used in the modelling, just as a reference point.

Clutch Pedal Position
The clutch pedal position indicated how far the clutch pedal had been pushed down at a certain time. This was an important indicator to see if the truck was free rolling or not, as the clutch released the powertrain from the gearbox which was connected to the wheels. This was used in the method for determining the application pressure that is presented in Section 7.3.1.

Brake Pedal Position
The brake pedal position worked in the same way as the clutch pedal position. The brake pedal position was crucial in determining the pre-braking phase, which is discussed further in Section 7.3.1.

Accelerator Pedal Position
The accelerator pedal position signal gave the position of the accelerator pedal. The signal was used to detect the pre-braking phase, defined in Section 7.3.1.

Time
Signal that showed the time in seconds for the data set. The signal was used in several different contexts, for example in estimating the acceleration, which is described in Section 6.2.

Loss of Torque
Loss of torque gave data on how much torque that had been lost due to for example friction and inertia in the driveline. The information given here was essential in determining the forces that affected the wheels from the driveline.

Use of Maximum Torque
The use of maximum torque indicated how much of the maximum torque was used by the engine. This was used together with the signal that determines the loss of torque from the engine when finding out the forces that were affecting the wheels.

6.2 Estimating Acceleration

A significant signal in detecting when the brakes apply to wheels was acceleration. The signal was used in the different methods that were tried for determining the application pressure, discussed in Chapter 7. However the acceleration of the individual wheels was not one of the signals that could be measured and had to be calculated using the signals that were present, the obvious one being velocity. As acceleration is equal to the derivative of velocity this was not too difficult but it required a method for making this estimation of the derivative.

When calculating the derivative of a set of data there were a number of different numerical and analytical methods that could be used. They offered different advantages but in this case a numerical method was chosen to be used. An observer would also have been possible to use but due to programming and implementation advantages with a numerical method this was considered to be the most appropriate.

6.2.1 Numerical Estimation of the Acceleration

The easiest way to calculate the derivative of a function f(x) is to use

$$f'(x) = \frac{f(x_i + h) - f(x_i)}{h}, \qquad (35)$$

where h is close to zero in order to make the approximation accurate *(Wolfram Mathworld:3, 2008-01-25).*

A more symmetrical derivative can be calculated using

$$f'(x) = \frac{f(x_i + h) - f(x_i - h)}{2h}, \qquad (36)$$

where h is close to zero *(Wolfram Mathworld:1, 2008-01-25).*

There were other approaches that included more points, but in this context the symmetric derivative was deemed the most suitable and of good enough accuracy. The step size h of the derivative was important when determining the approximation of the acceleration. As the data collected was discrete this meant that the step sizes had to have a value of at least one, which gave the most accurate acceleration estimation. However this also resulted in data that was noisy and difficult to study the behaviour of. Therefore a larger step size was used and through empirical tests this value was set at h = 50. This gave a smoother data sequence that was easier to interpret and use, but it also lost some of the information that a smaller step size would contain. This was a compromise that had to be considered when performing this type of operation.

It was also important to remember that the single that was being sampled had been pre-filtered during the data collection using a frequency of 100 Hz. This was taken into consideration during the following studies and was not considered as being a major problem.

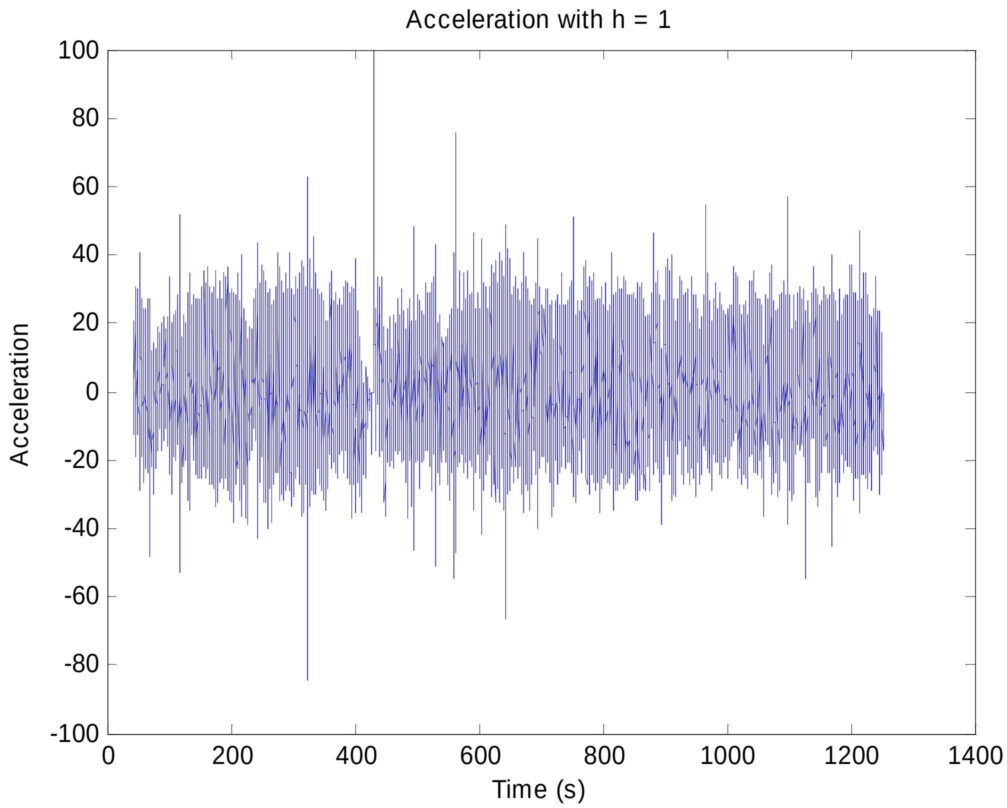

Figure 11. Plot of acceleration as a function of time with a step size of 1.

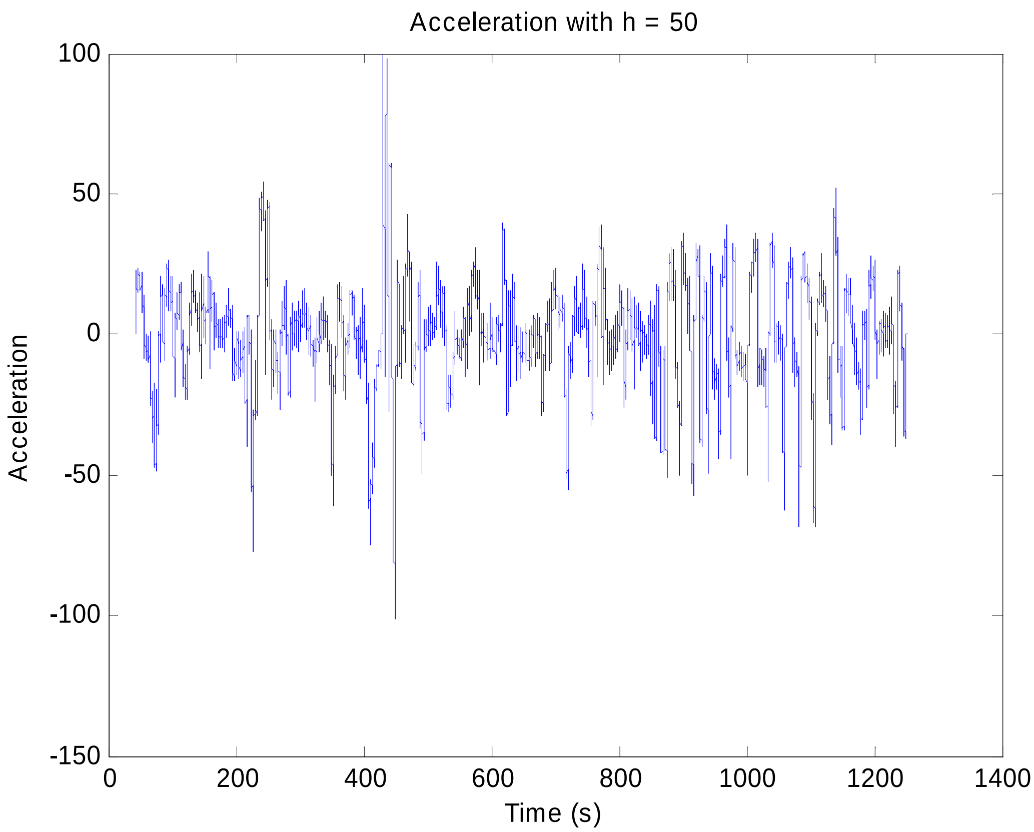

Figure 12. Plot of acceleration as a function of time with a step size of 50.

6.3 Signal noise

When conducting tests and gathering data that was going to be analyzed, signal noise was something that affected the collected information. Signal noise disturbs the actual signal and is not controllable or chosen. In many systems this signal can have a significant influence on the behaviour of the actual signal that is going to be studied *(Glad & Ljung, 2004, p 221).* The Electronic Brake System used both raw and filtered signals and in this case the filtered signals were collected. The reason was that these filtered signals were the ones used for the control of the vehicle by the EBS. Some noise was however still present and this could depend on numerous factors; the unevenness of the road, frictional noise in the driveline, fluctuations of the wind and instability in the measuring equipment are just a few examples.

It was important to be aware that noise existed in the signals that were studied and it could be dealt with in different ways. Filtering is a common method of reducing certain frequencies of the signal that the noise is presumed to have.

6.4 Filtering data

By filtering a signal, the noise that disturbs it can be reduced and the “true” signal can become more visible. This is done by only allowing certain frequencies to protrude. There are different forms of filters that have different functions. A low-pass filter only lets through signals with a frequency below a certain point, a high-pass filter is the opposite and band-pass filter lets the signals in a certain frequency band through *(All About Circuits, 2007-11-23).* The signals that were used in this study had been low-pass filtered before they were collected. This meant that further filtration of the signals was not considered necessary because as much information as possible wanted to be used. The exception was the acceleration signal that was extracted from the velocity signal, described in Section 6.2. In that case a low-pass filter was applied by extending the step size for the derivative calculation.

Further filtering tests were conducted where both a low-pass and band-stop Butterworth filter were applied to the acceleration signal, but the results showed that the unfiltered signal gave more reliable results. This is because if a filter was applied to a signal this will mean that a phase shift will be added to the signal and also that information might be lost.

6.5 Limitations in data

The filtering of signals meant that some information might be lost, as was explained in the previous Section. Filtering was however not the only reason for the loss of information. Many of the signals, for example the pedal position signals and the pressure signals, had data that was displayed by truncating with a floor function. To truncate with a floor function means that the function holds the same value until a threshold is exceeded and a new value is set *(Wolfram Mathworld:3, 2008-01-30).* A graphical description can be viewed in Figure 13. This meant that the signals did not display their actual value, but rather a value that was equal to or less than the actual value.

This was not crucial for the pedal position signals, but for the pressure signal this meant that the exact value of the pressure was not possible to obtain and this had an impact when determining the application pressure and making an observer.

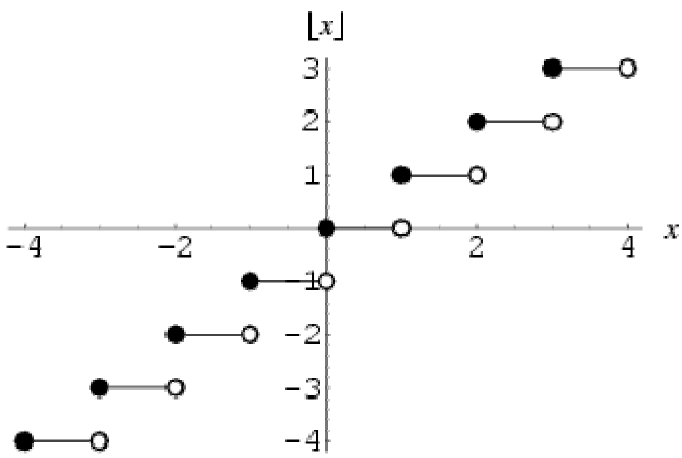

Figure 13. Explanation of a floor function. The floor function of x, $\lfloor x \rfloor$, remains at the same level until x reaches a value that is large enough and $\lfloor x \rfloor$ can change (Wolfram Mathworld:3, 2008-01-30). Translated to the pressure signal this will mean that the displayed pressure is a floor function of the actual pressure that will not show the exact value of the pressure.

6.5.1 Phase Shifts Between Signals

When dealing with the signals that had been filtered when they were collected it was important to be aware of the phase shifts that could be present. Depending on how much the signals had been filtered this would cause different phase shifts. Especially a phase shift between the pressure signal and the acceleration signal would have caused problems because these had to be in the same phase in order for the deviation detection to be accurate, which is discussed further in Section 7.3. This was studied and no phase shift was detected between the two signals.

7. Methods of determining the application pressure

There are several different ways to go about developing a method for determining the application pressure. Three different methods were tried, getting varying forms of accuracy. In this chapter the three methods will be presented and the results from these tests are analyzed. One of the methods has been implemented and further developed. Other ways of solving the task at hand are of course possible, but for this thesis the ones presented here were deemed most suitable.

The methods used can be put in two different categories; model-based and signal-based methods. Common for them is that the acceleration signal for the wheels is studied.

7.1 Using acceleration to determine application pressure

In order to study when the brakes had applied, the signals presented in Section 6 had to be used. There was no sensor that for instance detected contact between the brake pads and the brake disc. Therefore other signals that acted as a result of the contact had to be studied. The fact that the wheels' velocities and accelerations decreased when the brakes applied made these signals the most suitable to study. From these alternatives the acceleration was chosen as the most appropriate to use, due to the fact that it was easier to incorporate with physical expressions, such as Newton's Second Law, and it gave a more visible reaction when the brakes applied.

7.2 Model Based Method

The first approach was to build a model of the pneumatic system, breaking it down into smaller and easier pieces that could be described by physical properties, in the same way as was done in Section 3.2. The model was built around one wheel with its brakes.

To model the pneumatic system, the Matlab modelling and simulation extension Simulink was used. One consideration that had to be made was that the system contained several non-linearities, which could be hard to model. The system would for example act differently before the brakes had applied than after application was made. The components would move easier in the pneumatic system as the resistance was much less than when the brakes had applied.

7.2.1 Choosing a Model to Describe the System

The system could be described using two different models that were linear. One model to describe the system before the brake pads are in contact with the discs and one model to describe the system after the brake pads are in contact. This would avoid a non-linear description of the system, which would have been more difficult to model. It would also make it possible to represent in an accurate state space form and from there get an observer. Figure 14 shows schematically how the brake force was non-linear since it changed characteristics when the brake pads got in contact with the brake disc. When the characteristics of the acceleration were studied from graphs where the braking occurred, these linear attributes could be detected and therefore seemed reasonable to use in the method. *(Harju & Löfstrand, 2007-10-10)*

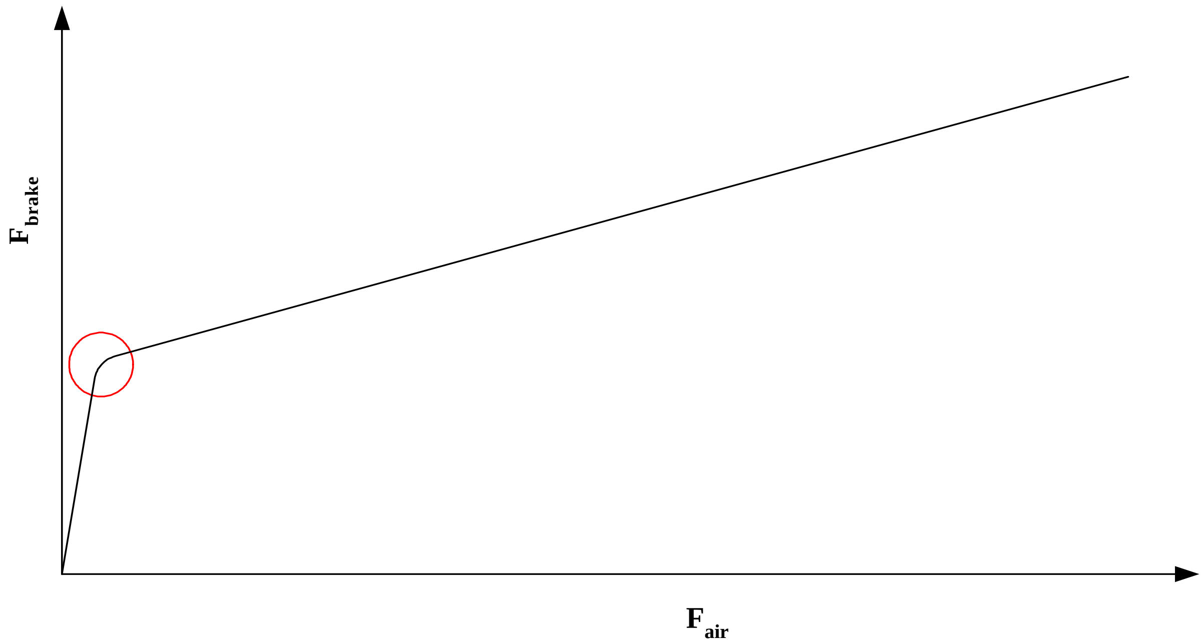

Figure 14. The graph shows how the force applied on the brakes, F_{brake},, changes when the brakes have applied, which the red circle indicates. Since the acceleration of the wheel depends on the brake force this means that the acceleration has the same nonlinear properties as the brake force (Based on Harju & Löfstrand, 2007-10-10).

From these two options it was chosen to model the system prior to contact between the brake pads and brake disc. State space representation of the acceleration when the brakes are not applied can be derived from equation (8) by going via torque and moment of inertia

$$F_{brake} = \mu_f \frac{r_1}{r_2}(p_{air} A_d - kx_{pr} - c\dot{x}_{pr}) \,, \qquad r_1 > r_2 \tag{37}$$

$$T_{brake} = r_w F_{brake} \tag{38}$$

$$a_w = \frac{T_{brake}}{I_w} = \frac{r_w \mu_f \frac{r_1}{r_2}(p_{air} A_d - kx_{pr} - c\dot{x}_{pr})}{I_w} \,, \qquad r_1 > r_2 \,, \tag{39}$$

where F_{brake} is the braking force, μ_f the frictional constant, r_1 the length of the brake lever, r_2 the length of the brake lever from where the brake piston was connected, p_{air} the air pressure, A_d the area of the diaphragm, k the spring constant, x_{pr} the length of the push rod, x'_{pr} the velocity of the push rod, T_{brake} the brake torque, r_w the wheel radius, a_w the wheel acceleration, I_w the wheel's moment of inertia.

7.2.2 Model Based Diagnosis

Model based diagnosis compares the model with the measured data in order to detect when a deviation between the two occurs. This means that it has to be possible to measure the data that is going to be modelled, in this case the velocity, and of course to be able to build an accurate model. In Figure 15 this is illustrated. By studying the residual, r(t), between the model that estimates the output, $y_{est}(t)$, and the data, y(t), according to equation (40) a detection can be made.

$$r(t) = y(t) - y_{est}(t) \tag{40}$$

If the model is accurate and there is no noise the residual will be zero. If a change or fault occurs the model will not be affected, but the measured output will change thus leading to a residual that will deviate from zero. The change can be detected and not isolated, but as change detection was what was needed to find the application pressure this method was appropriate *(Nyberg & Frisk, 2002, p 26).*

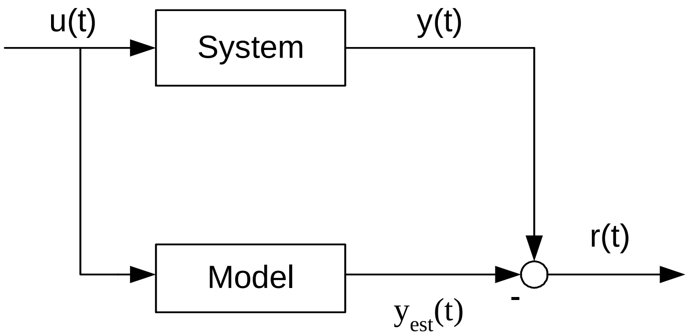

Figure 15. Description of model based diagnosis and the study of the residual between the measured output, y(t), and the estimated output, $y_{est}(t)$. The common input is u(t) (Nyberg & Frisk, 2002, p 26).

7.2.3 Results from Model Based Method

There were two purposes of building a model of the pneumatic system: The primary aim was to learn and get a proper understanding of the brake system and how it worked, both in a schematic form and through physically representing the pneumatic system. The secondary aim was to get a state space representation of the system and from that to be able to build an observer that would predict the application pressure. The latter being the purpose of the entire thesis.

The primary aim was hard to present any definitive results from, but the theoretic part of this thesis can be said to be the outcome of that purpose. During the process of understanding the system the physical properties of the system were derived and that has been the basis of the theoretic section about the pneumatic system and how it influenced the wheel when braking.

The secondary aim to use the physical representation to build an observer was less successful than intended. The model that was built to represent the system did not have the accuracy that was needed to be able to use any further and build an observer from and therefore the modelling of the pneumatic system was considered not being useful for this purpose. The reason for the model not being accurate enough could be that the physical representation was too schematic and missed properties that existed in the system, for example losses due to friction or materials acting in an unexpected way.

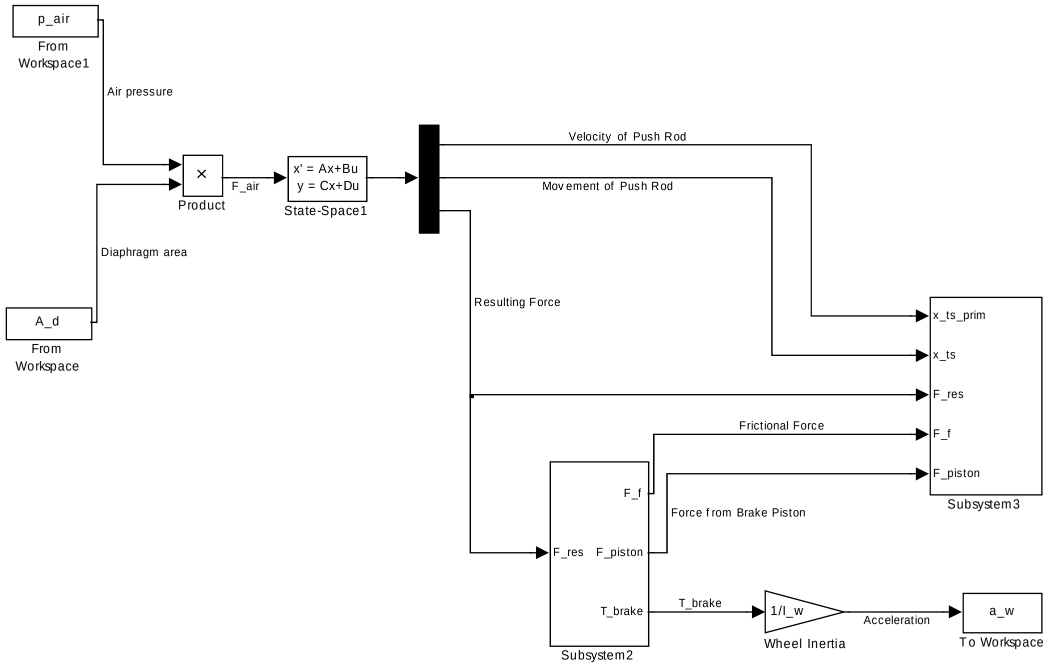

Figure 16. The figure shows the Simulink model that was used in the model based method for extracting the application pressure.

7.3 Signal Processing Methods

In the previous Section a method that based itself on modelling the physical properties of the system was presented. In this Section a different approach of estimating the application pressure will be made. Signal processing methods will be used. The signals from the Electronic Control Unit (ECU) in the truck were studied and used in order to find the application pressure. Two methods of using the signals are presented, but there are of course other ways to deal with the problem too.

The two methods deal with the problem in two different ways but have in common that they both use the acceleration signal, which was derived using the velocity signal from the ECU. By studying this signal and focusing on the data sequence before the brake pressure was above zero, this gave a level of all the decelerating forces in equation (22); $F_{airdrag}$, F_{roll} and $F_{engbrake}$, and this caused a negative acceleration that was at approximately the same level over the entire rolling sequence. When F_{brake} has reached a level that influences the deceleration of the vehicle this caused the curve to diverge from its previous steady level, as seen in Figure 9. At this point the pressure data could be studied and that was the application pressure of the vehicle.

7.3.1 Method one: Modelling the Pre-Braking Acceleration

This method approached the acceleration data from a modelling perspective. The data was modelled using the dynamics of the vehicle that were presented in Chapter 4. If the decelerating forces were correctly modelled using signals from the ECU this would have given a representation of the pre-braking resistance that was accurate to use for detecting when the data started deviating from it. The pre-braking resistance was modelled using the results from Chapter 4 and is defined later in this Section. The slightly modified equation is

$$-F_{prebrake} = -F_{airdrag} - F_{roll} - F_{engbrake} - F_{slope} = \\ = -c_{airdrag} A_{front} \frac{\rho_{air}}{2} v^2 - F_{roll} - F_{engbrake} - F_{slope} = m_{veh} a_{veh} \tag{41}$$

$$\Leftrightarrow$$

$$a_{veh} = \frac{-c_{airdrag} A_{front} \frac{\rho_{air}}{2}}{m_{veh}} - \frac{F_{roll}}{m_{veh}} - \frac{F_{engbrake}}{m_{veh}} - \frac{F_{slope}}{m_{veh}} \tag{42}$$

$$\Leftrightarrow$$

$$a_{veh} = \frac{-c_{airdrag} A_{front} \frac{\rho_{air}}{2}}{m_{veh}} - f_r g - \frac{i_t i_{fd} T_{eng}}{m_{veh} r_w} - g \sin(\alpha), \tag{43}$$

where a_{veh} was the pre-braking acceleration that worked as a reference value for the acceleration when the wheel started to brake.

7.3.1.1 Defining the Pre-Braking Phase

The pre-braking phase was defined as the phase before the brake pressure for a wheel was above a threshold pressure and when the vehicle did not accelerate. The threshold was significantly smaller than the estimated application pressure and the acceleration could be controlled by studying the position of the accelerator pedal. The pre-braking phase can be viewed upon as a rolling phase and if the clutch pedal was pushed down the vehicle was free rolling because the powertrain did not have any decelerating effects on the wheels. This phase always occurred before braking because of the time it took to move the foot from the accelerator pedal to the braking pedal. This time gap was enough to find a reference point to compare the acceleration to when the brakes had applied.

7.3.1.2 Determining the Parameter Values

In the representation of the pre-braking acceleration there were several parameters that had to be determined in order to adjust the model to the data and the conditions that the tests were conducted in.

$F_{airdrag}$, the aerodynamic drag resistance, contained a constant, $c_{airdrag}$, that depended on the aerodynamic properties of the object that travelled through the air. An object with $c_{airdrag} = 1$ brings all the air heading towards it to rest and spreads the air pressure equally on the surface *(Eriksson & Nielsen, 1998, p 197 – 198).* When designing a vehicle it is of interest to reduce the drag coefficient so as much air as possible is deflected to the side of the vehicle and the forward driving force of the vehicle becomes as large as possible. This reduces the consumption of energy. Through information from the Scania database it was possible to obtain an approximation of the drag coefficient and through tests the value was further adjusted. The same procedure accounted for the information about the front area of the vehicle. The weight of the truck, m_{veh}, was determined by weighing the truck before collecting the data. The density of the air was chosen to be 1.25 as it seemed reasonable judging by the temperature when the tests were performed *(The Engineering Toolbox, 2008-01-30).*

For the rolling resistance, F_{roll}, the mass of the vehicle, m_{veh}, was eliminated when determining the acceleration of the truck. This meant that only the rolling resistance coefficient, f_r, had to be chosen. For a car this value is usually in the range of 0.01 – 0.03, but for a truck the value is less as the wheels have different properties that cause less friction (*Eriksson & Nielsen, 2001 p 198*). From the Scania database and through testing a suitable value was extracted and applied.

When determining the decelerating force from the driveline, $F_{engbrake}$, the torque on the wheel had to be known. This could be done using two different methods. Firstly by using the conversion ratios for the transmission and final drive and multiplying this to the force from the engine and dividing with the wheel radius, just as it is stated in the formula. The second method was to use the internal signals that had information about the torque acting on the wheel and use these directly. The first way worked in theory, while the second one was more practical and faster to use in this context and that was why it was chosen.

The force that affected the retardation due to the slope, F_{slope}, could be determined by using an internal signal that calculated the slope of the road, α.

7.3.2 Results from Modelling the Pre-Braking Acceleration

The purpose of modelling the pre-braking acceleration was to get a reference value and detect when the acceleration deviated from it. It was therefore crucial to get a correct representation of the pre-braking acceleration before starting with the deviation detection. In Figure 17 the pre-braking model and the acceleration data before braking can be viewed, as well as the air pressure when braking.

The approximation of the acceleration was fairly good at certain sections, for example before and after the 1000 seconds mark. That kind of modelling would have made it possible to use the method for deviation detection. These data sequences were taken from a part of the collected data where the ground was plane and this meant that the influence from F_{slope} was zero, as $\sin(\alpha) = 0$ if $\alpha = 0$. If a different section was viewed, for example around 925 or 1100 seconds, it could be seen that the model did not fit the acceleration data in a very good way. The reason for this was that the data sequence was collected when the truck was driving in a hill and the signal used approximated the hill in a manner that lacked in precision. It was obvious that the model would be hard to implement if it could not handle hills, as it was quite common for hills to be present when driving on a road. In order for the modelling of the pre-braking acceleration to work, the force that came from the slope of the road had to be determined in a better way, otherwise it would have been of much less use to implement it.

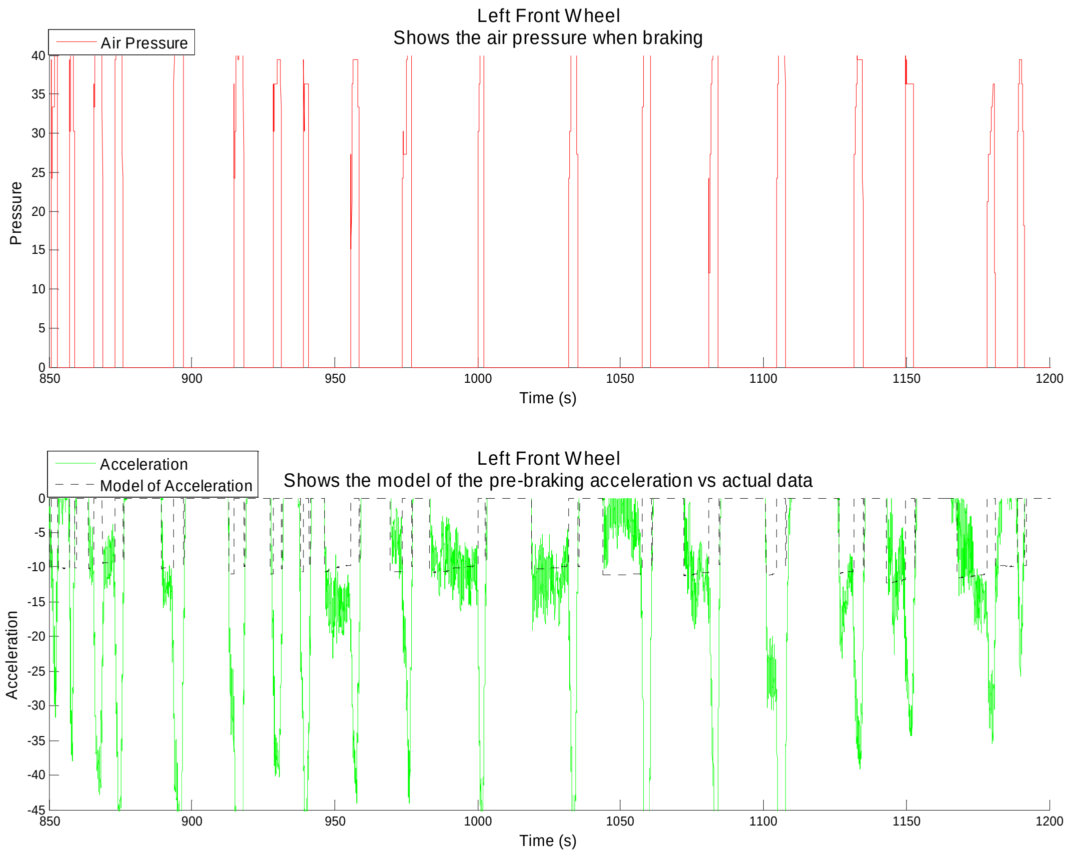

Figure 17. The top graph shows the air pressure when braking and the bottom shows the model of the rolling, pre-braking acceleration, dashed black line, compared with the acceleration data, solid green line. The model only emulates the sequence prior to braking.

7.3.3 Method two: Detecting when the Acceleration Deviates from the Mean of the Pre-Braking Acceleration

This was in many ways similar to the first signal processing method, but instead of using a model for the pre-braking acceleration, the mean of the acceleration before braking was calculated. This value was then used as reference value. When the acceleration deviated from the mean the brakes could be considered to have applied, given that the brake pressure was above the threshold value. This method was not dependent on any physical modelling but only used the data from the Electronic Control Unit and minimized the uncertainties that modelling could consist of. The major issues were determining the deviation from the mean and good way of calculating the mean.

7.3.3.1 Determining the Mean of the Pre-Braking Acceleration

To use this method, a way to calculate the mean in real time had to be chosen. If all the data needed was already accessible this would not have been necessary, but in this case it was necessary to be able to calculate the mean continuously. Therefore a moving average method was used. As described in Section 5.1, this meant that a suitable window size was chosen for the mean to be counted in and this was done throughout the pre-braking phase. The window size, if measured in time, should at least be smaller than the minimum brake response time, which is approximately 0.7 seconds for trucks according to empirical studies at Scania. To get the most accurate result to compare with, the window size should not be too large as there was a risk that unnecessary information would have been included in the determination of the mean. If the window size was too small the effect of using the mean would also have lost in significance as not enough values would influence the mean that should be used as a reference value. To determine the window size tests were conducted that indicated the appropriate size of the window.

7.3.4 Results from Determining the Mean of the Pre-Braking Acceleration

The problem with modeling the pre-braking acceleration was that handling hilly terrain was difficult to get accurate. The second signal processing method removed that problem because of the mean calculation of the pre-braking acceleration. In Figure 18 the reference value that was created by using the moving average algorithm of the pre-braking acceleration can be seen. The sequences are the same as for the first signal processing method and the accuracy in handling hills became greater when the reference value was set by the mean. It was therefore chosen to use this method in the further development of the application pressure estimation process.

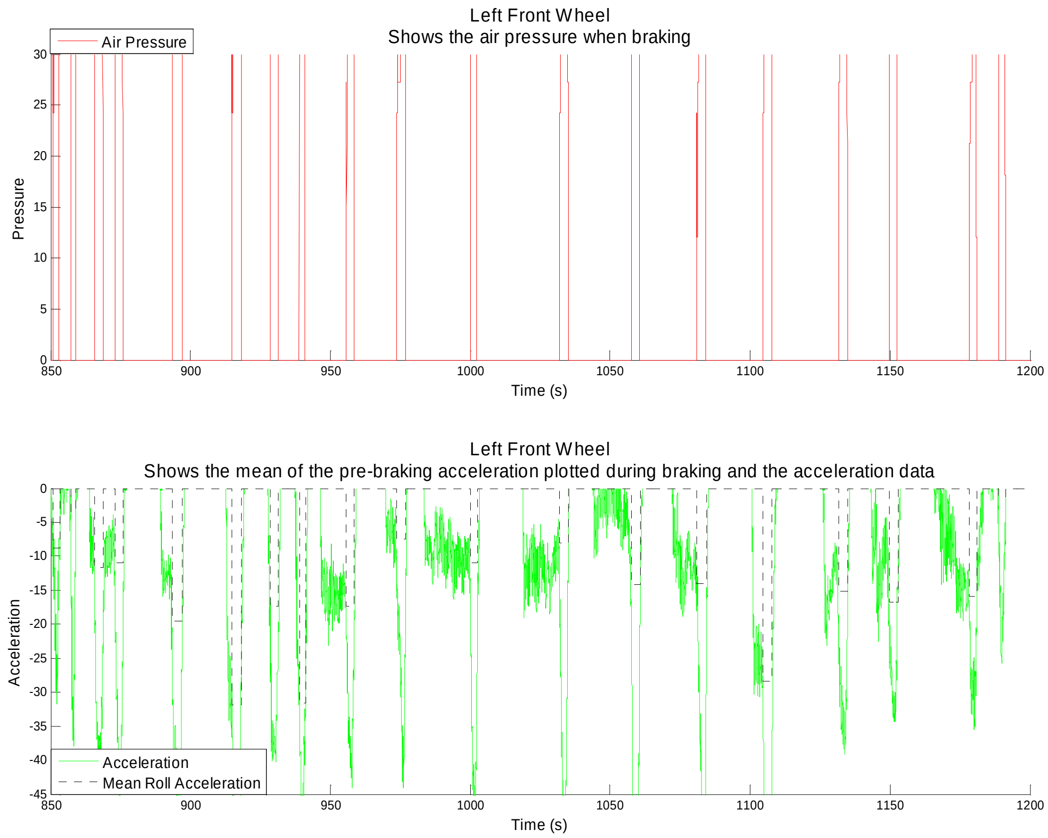

Figure 18. The top graph shows the air pressure when braking and the bottom graph shows the mean pre-braking acceleration, dashed black line, and the acceleration data, solid green line. The mean pre-braking acceleration is calculated during the pre-braking, but plotted during braking to make the comparison with the acceleration data easier.

7.4 Finding the Deviation from the Pre-Braking Acceleration

The WECO-rules were explained in Section 5.2.2 and they were a good starting point for trying to detect a deviation from the mean of a data sequence. The rules are usually used for detecting deviations in stationary processes and since that is not the case here it was interesting to see if the rules could be applied in a different context.

The implementation of the rules was done in Matlab. The criteria for a deviation to occur were that one of the rules came into play and the pressure was above zero. In Figure 19 the rules were applied and the point of deviation could be observed. An ocular overview of the curve indicates that the WECO-rules found the points at which the data deviated from the mean in each braking sequence in a satisfactory fashion.

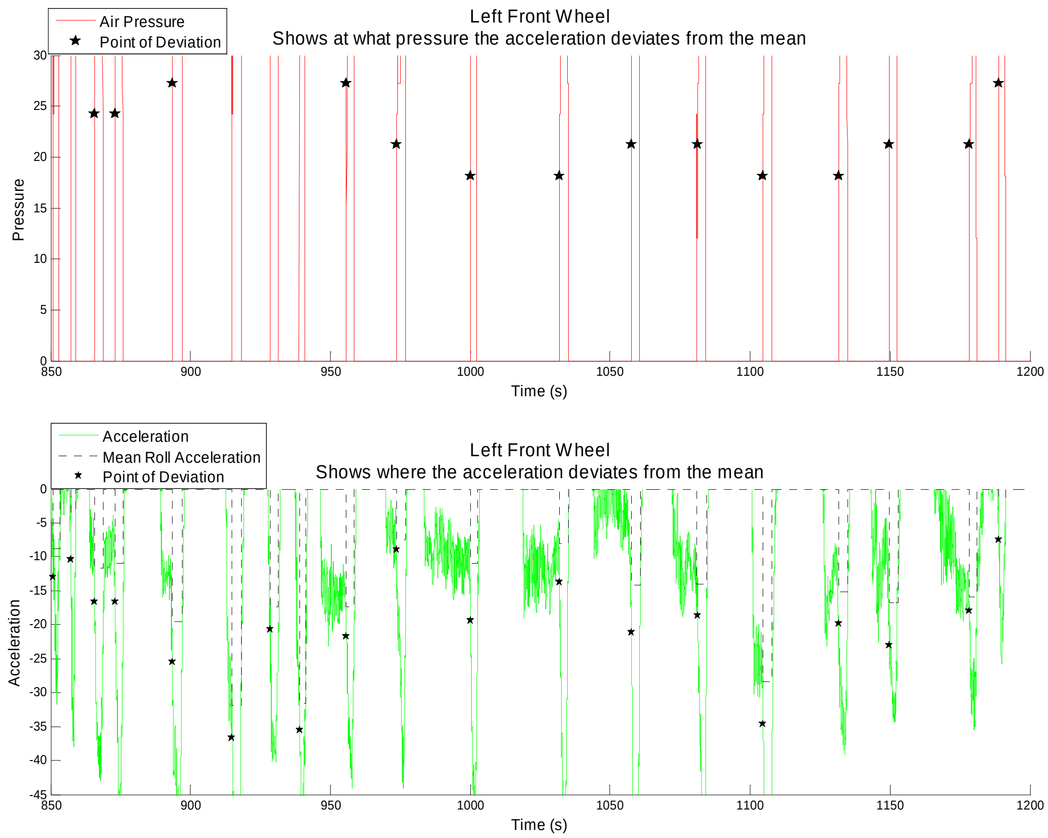

Figure 19. The top graph shows at what pressure the acceleration deviates from the mean, marked with black stars. The bottom graph shows the pre-braking mean acceleration, dashed black line, the acceleration data, solid green line, and the points of deviation from the pre-braking mean, marked with a black star. The deviation is calculated using the unmodified WECO-rules.

7.4.1 Modifying the Western Electric Company Rules

There was a risk for false alarms present when implementing the Western Electric Company rules, which was mentioned in Section 5.2.2. Therefore the implementation using the rules was modified through testing. Some of the rules were excluded and thereby the risk for false alarms occurring was decreased and the accuracy of the resulting deviation points was greater. Even if the accuracy of the results would be at a level of an error on every 91.75 points, or approximately one percent, this would be considered acceptable as the filter that is presented in Chapter 8 was constructed in a manner that made it less sensitive to random errors.

The Western Electric rules were developed to see at what point a curve started to deviate. In this context it was however necessary to investigate whether all the rules were necessary to use, if some were redundant or should be modified. In Section 5.2.2 the risk of detecting false alarms was discussed and that it increased if more rules were used. As false alarms should be avoided, tests were done with the goal of seeing if some of the rules could be modified or excluded in order to get a higher reliability for the chosen points.

Experimental studies were conducted and these showed that if the second and fourth rules were excluded, the points of deviation did not change and it was still possible to obtain an accurate result. The reliability of the results was also greater but it was also important to be aware of that since a few rules have been removed there might be a delay in detecting the correct point of deviation.

By using the WECO-rules and modifying them to suit the purpose of this thesis a simple and reliable method of determining the point of deviation and thereby the application pressure was developed. It was a virtual sensor that detected the point of deviation and was the starting point for developing an estimator of the pressure.

The previous methods that were discussed were crucial in understanding and testing what path to take in determining the application pressure, but they will from now on not be used when implementing an estimator of the application pressure.

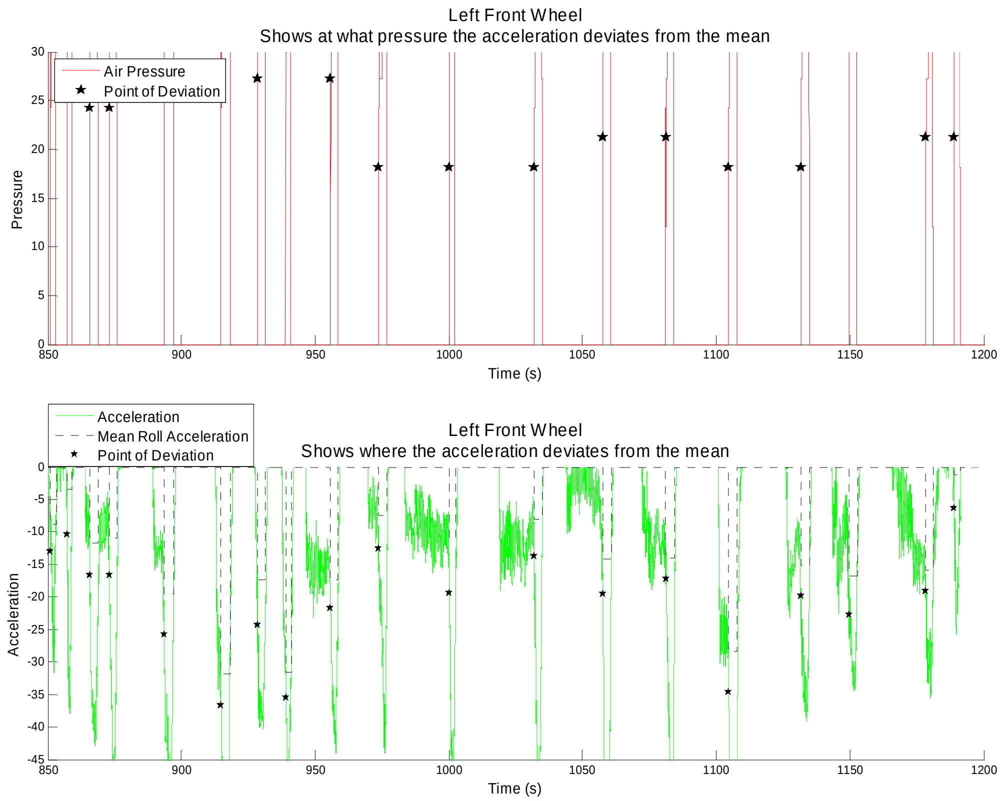

Figure 20. The top graph shows at what pressure the acceleration deviates from the mean, marked with black stars. The bottom graph shows the pre-braking mean acceleration, dashed black line, the acceleration data, solid green line, and the points of deviation from the pre-braking mean, marked with a black star. The deviation is calculated using the modified WECO-rules.

8. Implementing an Estimation of the Application Pressure

It was now possible to determine the application pressure by studying the acceleration data. The next step in the development process could begin: creating an estimator. This could be done in many different ways depending on what was going to be estimated and how the data was represented prior to this step. In this case a signal processing approach was chosen and the data was not modelled physically. This meant that for example constructing an observer would not be possible and other ways had to be considered.

8.1 Recursive Filter

The recursive filter used previous measured information to adjust the estimation of the application pressure. It started with an initial guess of the pressure and updated the estimation continuously by multiplying the difference between the previous measurement and the previous estimation by a scale factor. If the factor was large the estimation depended a great deal on every measurement. A small scale factor would only make small adjustments after every measurement and was not as sensitive for errors in the measurement. The formula was

$$p_{est}(t) = p_{est}(t-1) + k_d(p_{real}(t-1) - p_{est}(t-1)), \tag{44}$$

where p_{est} was the estimation of the pressure at time t, p_{real} was the measured pressure at time t and k_d was the multiplying factor that adjusted the estimation.

In order to design a recursive filter that makes a good prediction the initial guess, $p_{est}(0)$, and multiplying factor, k_d, have to be chosen. Because there already existed a constant estimation of the application pressure at Scania that value seemed like a reasonable initial guess. When determining the multiplying factor k_d this was solved through experimentation where sensitivity and speed of the estimation were weighted in order to make the estimation work in an adequate fashion.

8.2 Considerations when Implementing Recursive Filter

A concern when determining the application pressure and implementing a recursive filter was that of the dependence between the wheels. The velocity and acceleration of one wheel affected the velocity and acceleration of all the other wheels. Since all the wheels had separate Air Disc Brakes they could have different application pressures and therefore the time at which they started braking might not be the same. When determining the application pressure by studying the acceleration signal, this became important. If one wheel started to brake before all the other wheels, the other wheels would also slow down somewhat. This made it difficult to detect when a wheel actually started braking, because it might be influenced by another wheel that braked before it. The problem has been tried to be solved but not in a completely satisfactory manner. A solution that was developed was to study the different brake sequences and sort out which wheel brakes first each time. If this is done information about that wheel should at least be accurate since no other wheels could have affected its acceleration.

In Section 6.5 the limitations of the pressure signal were discussed. The signal was truncated and it did not contain the exact values of the pressure. This became a problem when the steps in the floor function were about the same size as the approximate spread of the application pressure that was used as a constant by Scania. It could also present a difficulty when using the recursive filter as the values that were collected would behave quite abruptly and the changes that adjust the pressure estimation will also be affected by this abrupt motion.

The positive thing with checking the deviation from the mean pre-braking acceleration was that it was easy to implement and that it gave an approximation of the application pressure. The definite pressure was hard to retrieve as the signal did not have enough precision, but it would generate a method that can be further developed in the future.

8.3 Results from the Recursive Filter Implementation

In Figure 21 the results of the observer implementation can be viewed. It is clearly visible that the estimation of the application pressure diverges from the measured application pressure. The reason for this was the truncated measurement of the application pressure. In order for the estimation to work properly and to be useful in the truck the signals have to be more accurate otherwise the estimation will lack in accuracy.

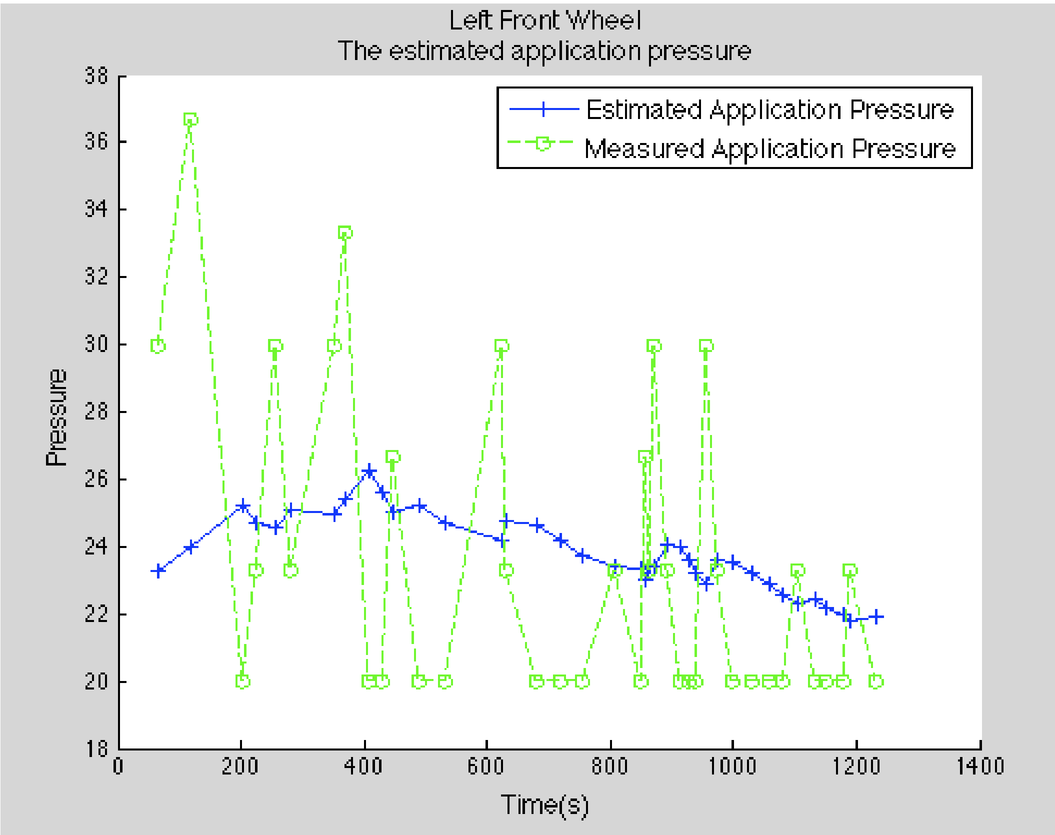

Figure 21. The graph shows the estimated application pressure in relation to the measured application pressure.

9. Conclusion

The purpose of this thesis was to see if it was possible to make a better and more precise approximation of the application pressure for the disc brakes in Scania's Electronic Brake System and implementing this solution. The objective was also to present different ways to address this problem and show which possible paths that are possible to take in order to estimate the application pressure.

In the thesis three different ways of determining the application pressure were presented; one model-based and two based on signal processing. When studying and developing the model-based approach essential information for understanding the system was generated and the nonlinear properties of the Air Disc Brake system were understood. That presented difficulties in modelling the acceleration of the wheel and therefore further paths were studied before a definite method was chosen.

Instead of modelling the mechanics of the brake system, a study of the acceleration signal was initiated. This would exclude the non-linearities of the brake system, as the results of the inner and outer effects on the wheels are studied directly. The first approach that was made tried to model the forces that affect the wheels acceleration in the sequence before braking and from this create a reference value that could be used to see if the actual signal would deviate from. The model that was created proved to be accurate in certain situations but lacked in precision when the braking occurred in hilly terrain when the signals from the vehicle's Electronic Control Unit were not accurate enough. The method did however prove useful in providing a way to find the application pressure point by using a reference value that the signal could be compared with. This was something that could be used in the further development when finding the application pressure.

The second signal processing approach used a different approach of finding the reference point for the acceleration to deviate from. By applying a moving average filter and calculating the mean acceleration continuously during the pre-braking phase, this was used to check when the acceleration signal would deviate and therefore when the brakes would apply. This method excluded all the difficulties of modelling outer and inner dynamics and just focused on the actual signal. By doing that different modelling errors could be ignored and a reliable way of determining the deviation was the only thing needed. For this purpose the WECO-rules were implemented. They defined a set of rules that determine when a curve deviates from its mean. They are primarily used for stationary processes, but were chosen here to see if they could be applied in a different situation as well. The implementation gave a reliable way of determining the application pressure points for the different wheels and this is the starting point for developing a method of estimating the application pressure.

The goal of the application pressure estimation was to present an estimated value of the application pressure before every braking sequence and update this value continuously. A recursive function fitted this description and was therefore implemented. The resulting estimation became difficult to use and lacked in precision due to the resolution of the pressure signal and the fact that the acceleration of the wheels depend on each other.

In conclusion a reliable method of determining the application pressure that was based on the acceleration signal was developed. The choice of using a numerical method for estimating the acceleration should in hindsight have been revised. More time should have been put into developing an observer that would have done the same calculations but in a more reliable fashion. This is something that should be taken into notice for further studies.

There were also issues regarding the resolution of the signals that were used in controlling the brakes and a reliable estimation was at this point not possible to implement. If an estimation is going to be possible to use, a change in these signals has to be made and then further studies would be useful for Scania. It is also necessary to develop a solution that can separate the effects on the acceleration that the wheels have on each other because otherwise the value of the application pressure found will not be of great significance. If the other methods that were discussed in the thesis should be possible to use an improvement of the signals that determine the slope of the road should be made.

9.1 Suggestions for Scania

Improve the resolution of the pressure signal and other signals.

Reduce or provide information about the phase shift between the signals.

A solution for separating the retardation on the different wheels is needed.

Improve the signals that have information about the slope of the road, for example using GPS.

9.2 Suggestions for further research

Design an accurate model for the brake pressure that can be used for the estimation.

Include some of the simplifications that were used in this study, for example turning in the model.

Use an observer when getting the acceleration signal.

Develop a method that gives the slope of the road, for example using GPS.

References

Printed

Amberkar, S., D'Ambrosio, J.G, Murray, B.T, Wysocki, J. & Czerny, B.J (2000), "A System-Safety Process For By-Wire Automotive Systems", SAE 2000 World Congress, Detroit, Michigan, March 6-9, 2000, SAE 200-01-1056

Sveriges Trafikskolors Riksförbund (2002)."Att köra lastbil/buss", Sveriges Trafikskolors Riksförbund, Sollentuna

Basseville, M. & Nikiforov, I.V. (1993), "Detection of Abrupt Changes: Theory and Application", Prentice-Hall, Englewood Cliffs

Eriksson, L. & Nielsen, L., (2001). "Course Material Vehicular Systems", Linköping Institute of Technology, Linköping

Glad, T. & Ljung, L., (2003). "Reglerteori: Flervariabla och olinjära metoder", Studentlitteratur, Lund

Glad, T. & Ljung, L., (2004). "Modellbygge och simulering", Studentlitteratur, Lund

Kiusaalas, J. & Pytel, A., (1999a). "Engineering Mechanics Statics", 2nd Ed., Brooks/Cole, Pacific Grove

Kiusaalas, J. & Pytel, A., (1999b). "Engineering Mechanics Dynamics", 2nd Ed., Brooks/Cole, Pacific Grove

Lu, X-Y & Hedrick, J. K., (2005). "Heavy-duty vehicle modelling and longitudinal control", Vehicle System Dynamics, 43:9, p653 – 669

Nyberg, M. & Frisk, E., (2002). "Diagnosis and Supervision of Technical Processes", Course Material Linköpings universitet, Linköping

Palmertz, J., (2003). "System Description Brake System EBS 5", Scania Technical Product Data Document, Södertälje

Schechtman, E., Bandner, G. & Meginy, S. (2007). "Detecting a change in a scale parameter – a combination of SPC and change point procedures", International Journal of Production Research, 45:23, p5535 – 5545

Vysochanskij, D F & Petunin, Y I (1980). "Justification of the 3σ rule for unimodal distributions", Theory of Probability and Mathematical Statistics vol. 21, p25-36

Electronic

About.com:Mathematics (2008). The Mean, http://math.about.com/library/weekly/aa020502a.htm (2008-01-30)

All About Circuits (2007), What is a filter?, http://www.allaboutcircuits.com/vol_2/chpt_8/1.html (2007-11-23)

Argentieri, R., (2006), Western Electric Rules, http://en.wikipedia.org/wiki/Western_Electric_rules (2007-12-14)

Light and Matter (2008). Acceleration and Free Fall, http://www.lightandmatter.com/html_books/1np/ch03/ch03.html#Section3.2 (2008-04-23)

NIST/SEMATECH e-Handbook of Statistical Methods (2006). http://www.itl.nist.gov/div898/handbook/ (2008-01-13)

Scania CV AB:1 (2008). Om Scania, http://www.scania.se/About_Scania (2008-04-23)

Scania CV AB:2 (2008). Lexicon, http://scania-lexicon.scania.se/termweb/app/termweb (2008-01-18)

The Engineering Toolbox (2008). Air Properties, http://www.engineeringtoolbox.com/air-properties-d_156.html (2008-01-30)

Wikipedia (2007). Pneumatic, http://en.wikipedia.org/wiki/Pneumatic (2007-11-08)

Wolfram Mathworld:1 (2008). Derivative, http://mathworld.wolfram.com/Derivative.html (2008-01-25)

Wolfram Mathworld:2 (2008). Floor Function, http://mathworld.wolfram.com/FloorFunction.html (2008-01-30)

Wolfram Mathworld:3 (2008). Numerical Differentiation, http://mathworld.wolfram.com/NumericalDifferentiation.html (2008-01-25)

Oral

Harju, R. & Löfstrand, M., Scania Brake Performance: Brake Callipers and Chambers, 2007-10-10

Harju, R., Scania Brake Performance: Air-Feeding System, 2008-01-28