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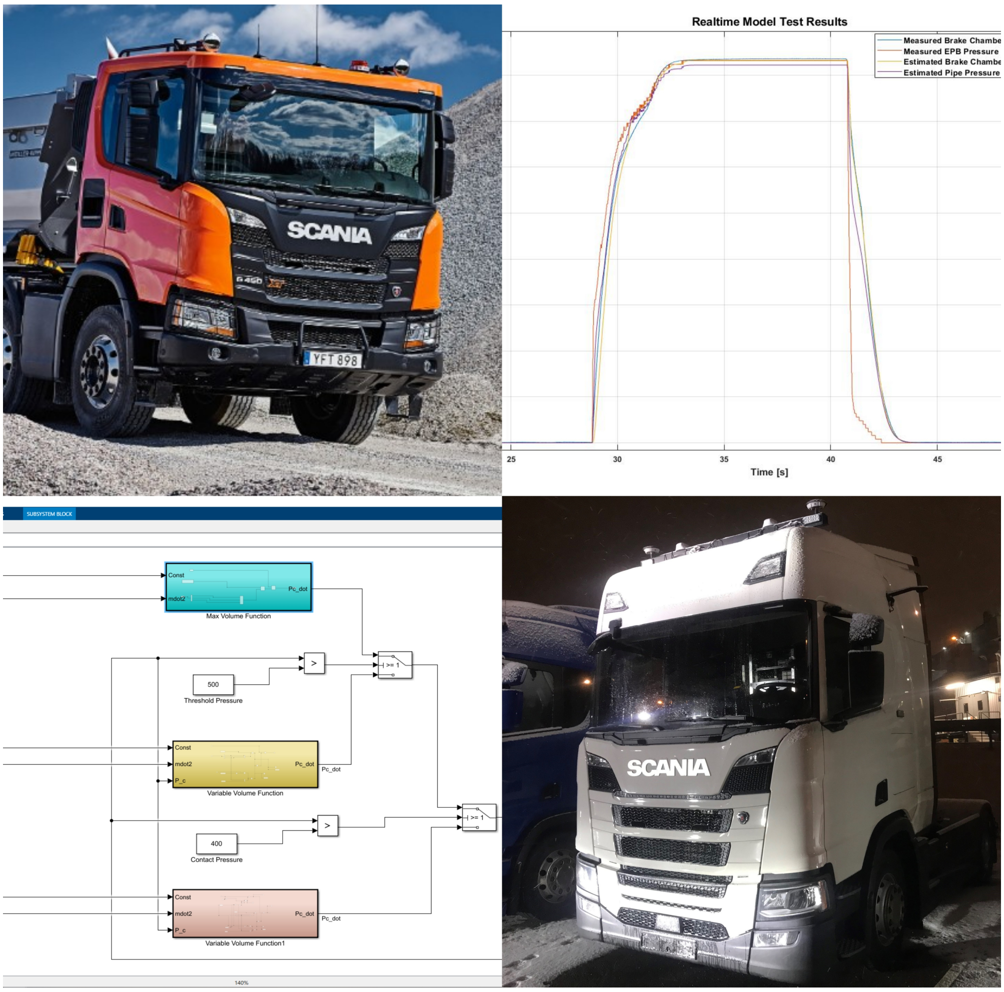

Modelling and simulation of electro-pneumatic parking brake system for real time estimation of pressure inside parking brake chamber.

Master's thesis in Automotive Engineering

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CHALMERS UNIVERSITY OF TECHNOLOGY
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Abstract

As a master thesis project, this research was conducted out at Scania CV AB's Brake Functions Department. This project is focused on the development of a model-based pressure estimator for electro-pneumatic parking brake chambers. The Electro-pneumatic Parking Brake System (EPB), which is now along with the Electronic Brake System (EBS), has a distinct pneumatic circuit dedicated entirely to the parking brake application; consequently, knowing the pressure inside the parking brake chamber is crucial if the parking brakes are to be engaged as backup brakes or autonomous stopping.
The thesis paper is concerned with the development of a mathematical model of an electro-pneumatic parking brake system based on the physical layout of the brake system. First two chapters of the report deals with the introduction and background of brake systems and it's components. Chapter 3 & 4 of the report deals with the methodology, approach, theory and the development of simulink model for each of the components present in the brake circuit. The developed model take the inputs from the EPB module and further estimates the pressure considering the physical parameters of all the components on the brake system.
Final part of the report deals with the implementation of model in real-time test scenario and comparison with the measured pressure followed by the conclusion and future work.

Keywords: Pressure Estimation, Electro-Pneumatic Braking System, Simulink, Pnuematic Actuators, Modelling and Simulation, Commercial Vehicle Brakes

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We would like to express our heartfelt gratitude to our examiner, Bengt Jacobson from Chalmers University of Technology, for believing in us and offering valuable inputs and helping us throughout the duration of thesis.
Finally, we would want to express our sincere gratitude to our family, friends, and relatives for their unconditional support throughout our master's program. They have been a big part of our lives since the beginning, motivating and inspiring us in different ways to help us achieve our goal. We would like to thank each other for not giving up during the difficult phases of the thesis work, for pushing each others boundaries, and for assisting each other in thinking outside the box when needed. Self belief and hard work will always earn you success.

Irfan Ahmed Khan Soudagar and Pranay Damodhar Tota, Gothenburg, February 2022

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Abbreviations

Mathematical notations

The following are the mathematical terms used in the report.

P Pressure
V Volume
m Mass of Air
R Universal gas constant
T Temperature
ρ Density of air
γ Ratio of specific heat
C_p Specific heat at constant pressure
C_v Specific heat at constant volume
$\dot{m}$ Mass flow rate
C_m Flow parameter of air
A_n Surface area of the nozzle
P_h High pressure supply
C_d Coefficient of discharge
P_l Low pressure supply
P_{CR} Critical pressure ratio
P_C Pressure in actuator cylinder
P_{atm} Atmospheric Pressure
P_{th} Threshold Pressure
P_{ct} Contact pressure
A_p Effective area of spring piston
M_p Mass of spring piston diaphragm
x_p Spring piston position
F_{Fb} Force from foundation brake
F_{kpl} Pre-load force of spring
F_{rs} Pre-load force of service brake return spring

SI units and radians are used throughout, unless stated.

1 Introduction

1.1 Project Background

Brakes are devices that use mechanical forces to stop or reduce speed of vehicle using the friction between the brake pads and the rotor or brake disc. Brakes on passenger vehicles require lesser forces while heavy commercial vehicles including buses gain higher kinetic energy while in motion due to their masses hence requires higher forces to bring them to stop of to reduce their speeds. Because the driver's pedal power is insufficient to perform this and translate the forces to longer distances in case of the heavy vehicles, This can be accomplished by using pneumatic brakes over hydraulic brakes where compressed air is used as the working fluid for the brake actuation.

There are numerous theories of brake systems which studies the characteristics of braking system such as brake force distribution, efficiency of brakes and so on. Pneumatic brakes have many advantages over the conventional hydraulic braking systems, they are capable of transmitting higher braking forces over longer distances and also it eases out the connection of the tractor to the trailer with a separate air control unit using series of valves in connection. These valves include some special ones to increase the flow of air as fast as possible to reduce the response time of the brake actuation which is the main disadvantages of pneumatic braking systems.

Pneumatic brakes have long response times this time lag is due to the complexity and the length of the piping in the system. Response time also called as brake application time is defined as the time from the first pedal movement to the first contact between brake pads and the rotor.

1.2 Motivation

The air is compressed and held in reservoirs before being supplied to each wheel via a set of valves that allow the air flow through the system move quickly. Despite this, after a small delay from the time the signal is received, the brakes are activated. The truck's braking performance is affected by the delay time in pressure response. The equation for estimating the time delay in pressure response is shown in [2]. However, a precise model is required to estimate the pressure in the brake chambers which will act as an aide to the control system by providing estimates of pressure to allow the Electronic Braking System (EBS) to supply the required amount of air to fulfil the demand and this will eventually reduce the time delay in brake actuation.

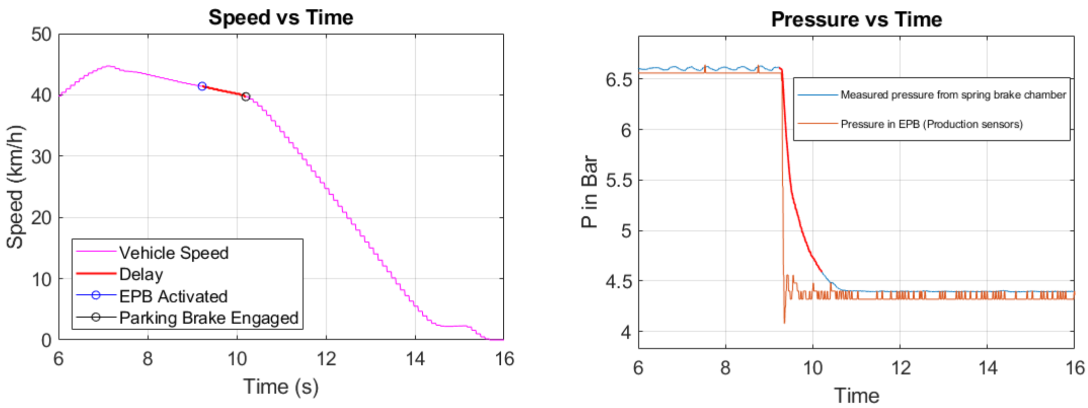

Figure 1.1: Plots representing Speed vs Time Pressure vs Time

The information in Fig.1.1 was gathered by driving a truck with pressure sensors in the braking chambers. The truck was driven at a speed of about 50 km/h on a straight, dry asphalt road with the Electronic Parking Brake (EPB) engaged. The request is sent to EPB in order to have pressure drop, as seen by the orange line in the fig. 1.1 it can be observed that pressure in the EPB drops immediately, but it takes some time for pressure in spring brake chamber (blue curve) to reach the desired value. This delay is estimated to be larger than 1 second. It can also be observed that the truck only begins to slow down/brake aggressively after a particular pressure in the brake chamber has been reached.
After performing the tests, it was concluded that there is a need of development of a model that estimates pressure inside the spring brake chamber, which will enable the system to signal the EPB to send requests to either stop or hold the system at a desired pressure.

1.3 Objectives

The preliminary research questions that could be answered by the end of the thesis work are:

1. What methods exist for estimating pressure inside the spring brake chamber in order to determine the brake response time?
2. Will the proposed approach handle the given technical challenge for the specified manoeuvres, and how well does it perform in dynamic tests?

To accomplish this, a estimator must be designed that will estimate pressure at the spring brake chamber. A controller can also be designed that will integrate the pressure estimator and the actuators for braking applications. Simulink will be used to model and simulate this arrangement, which will then be validated by using data from dynamic and static tests.

1.4 Limitations

1. Due to the complexity of components involved, the aim of the thesis is to model the system for the rear axle parking brakes of tractor only.

2. The pressure and time inputs will be taken from the EPB which is supplier owned component and will be subjected to the Simulink model.
3. Assumptions of certain physical parameters will be considered.
4. Only dry asphalt can be actively examined for braking manoeuvre data.

1.5 Deliverables

Following are the deliverable of this thesis:

1. Methods to estimate pressure inside the spring brake chamber of a truck
2. A mathematical model for the calculation of flow through nozzle, flow through pipe and the brake chambers.
3. Simulink model to estimate pressure by taking signals from the EPB.
4. Test results to verify the models.
5. Comparison of simulated data with the measured data and concluding the results.

2

Literature Review

2.1 Braking Theory

One of the most important systems in a vehicle is the braking system. It provides safe control of a vehicle in normal operation and brings the vehicle to a stop within a safe stopping distance. The main purpose of brakes is to slow down the vehicle, bring it to a complete stop and also keep it in a stable position when stopping on slopes [2]. A heavy truck brakes based on its functions are classified into service brakes, parking brakes and emergency brakes.
The general functions of the brakes are [2]:

1. Being able to slow down and stop the vehicle at the proper distance.
2. Be able to maintain speed when going downhill.
3. Be able to keep the vehicle in a fixed position when stopping on slopes.

2.1.1 Drum Brakes

The drum brake uses the friction of brake shoes or pads on a rotating cylinder, the brake drum. During normal drum brake application, the brake shoe presses against the inner part of the brake drum. Drum brakes are the most commonly used brakes on medium and heavy trucks and trailers for parking brakes.

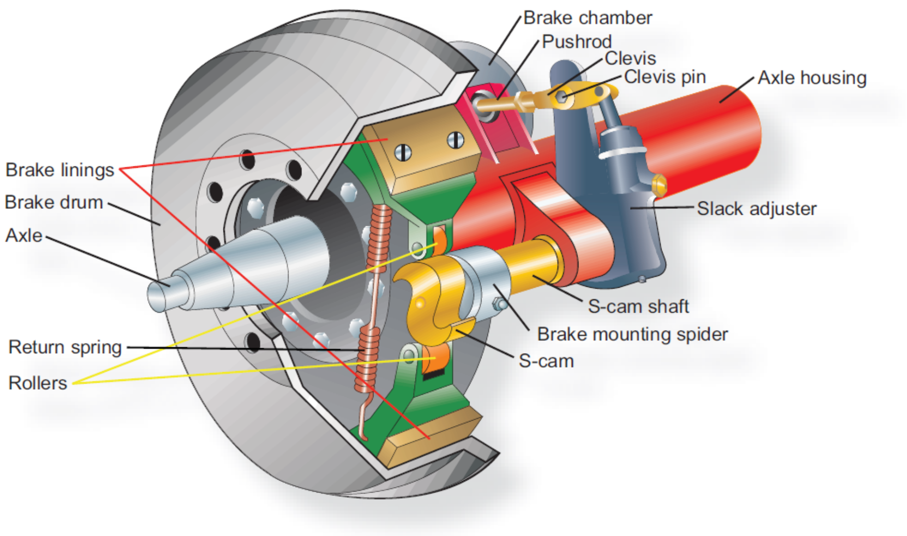

Figure 2.1: Drum Brakes [3]

About 90% of trucks that use air brakes are equipped with S-cam brakes. The S-cam brake uses the leading drag shoe design (picture to be added). The S-cam brake on a truck axle is shown in the fig. 2.1. This type of brake is actuated entirely mechanically by the rotation of an S-shaped cam within the brake drum, hence the name S-cam brake. A typical S-cam brake consists of the brake shoes, the S-cam, the slack adjuster, and an externally mounted brake chamber connected to the brake drum by a mechanical linkage. The rotation of the S-cam pushes the pins and rollers, forcing the brake shoes against the rotating drum and creating some force due to friction between the brake lining material and the brake shoes. The slack adjuster is used to transmit the mechanical force to the S-cam, which is connected to the brake chamber push rod, and also controls the adjustment of the gap between the brake lining and the brake drum.

2.1.2 Disc Brakes

The disc brake consists of a brake disc that rotates through the calliper where the brake pads are located as represented in the figure 2.2. When the brakes are applied, the pistons in the calliper push the brake pads against the rotor, creating a frictional force or braking torque that serves to slow or stop the vehicle. Unlike the S-cam in a typical drum brake, disc brakes use a power screw to help apply the same force to both sides of the disc. Disc brakes have an automatic slack adjuster in the calliper. The newer generations of heavy trucks equipped with pneumatic braking systems use disc brakes for both the service brake and the spring brake.

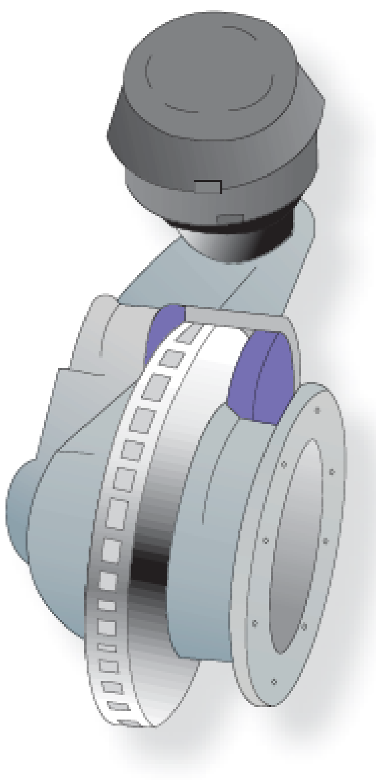

Figure 2.2: Disc Brakes [3]

2.1.3 Slack adjuster

Outside the brake chamber is the slack adjuster, which is connected to the push rod by a clevis and lever. When air pressure is applied, the diaphragm in the brake chamber pushes the push rod against the slack adjuster, this force depending on the air pressure and the area of the diaphragm. The main function of the slack adjuster is to adjust the clearance between the brake shoes on drum brakes and the brake pads on disc brakes [3]. This adjustment occurs as the brake pads wear with use. If the brakes have too much play, the brakes tend to become less effective as the stroke length of the piston rod increases.

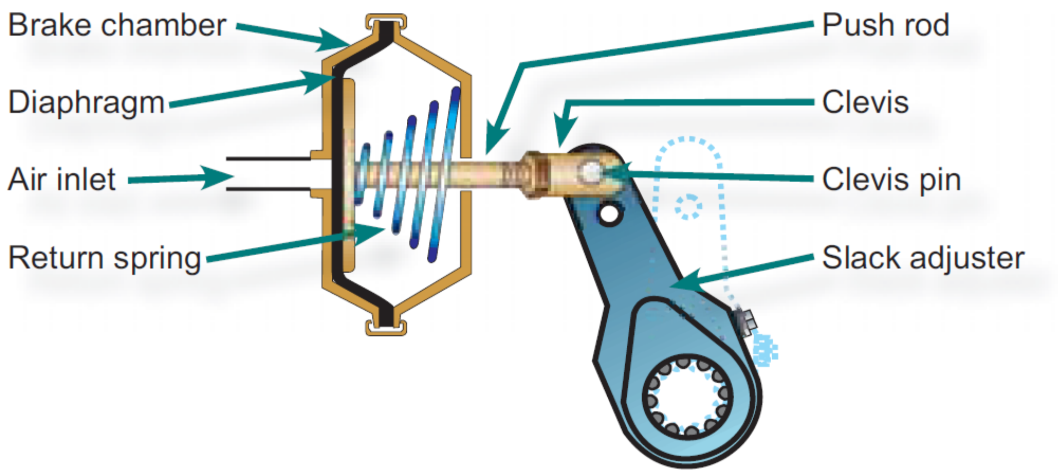

Figure 2.3: Slack Adjuster when brakes are released [3]

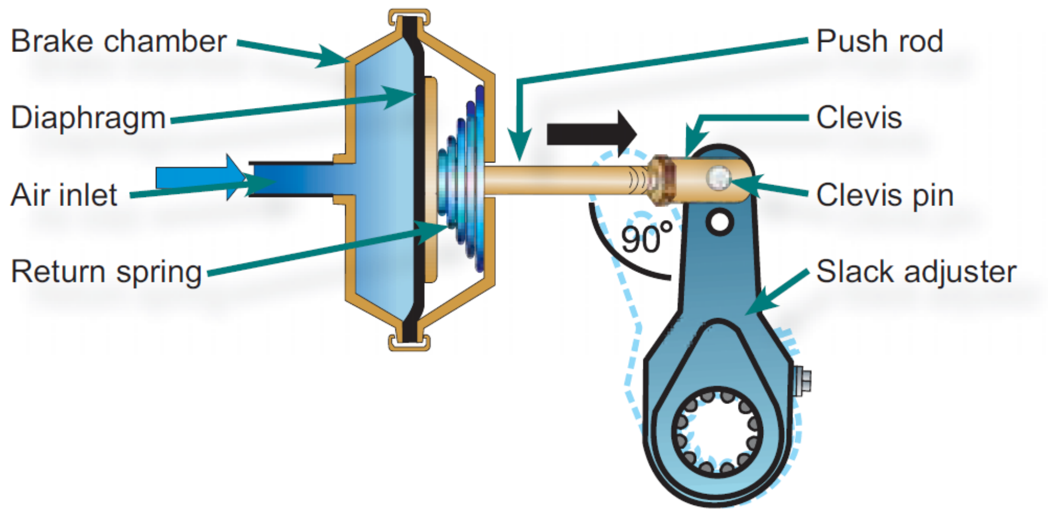

Figure 2.4: Slack Adjuster when brakes are applied [3]

2.2 Braking configuration in Heavy Vehicles

Unlike passenger cars which use hydraulic braking systems, heavy vehicles use air brakes that use compressed air as the working fluid. The force of the driver's brake pedal is used to adjust the air pressure in the brake chambers. Air is preferred over hydraulic fluid as the working fluid for several reasons, including the need to shop larger quantities of hydraulic fluid on board due to the larger number of axles in heavier vehicles. In the event of a leak in the pneumatic system, it will still respond with reduced power before the system fails completely. The only disadvantage of using air instead of hydraulic fluid in braking systems for heavy vehicles is that the response time of pneumatic systems is much longer than that of hydraulic systems, resulting in a delay in brake application.

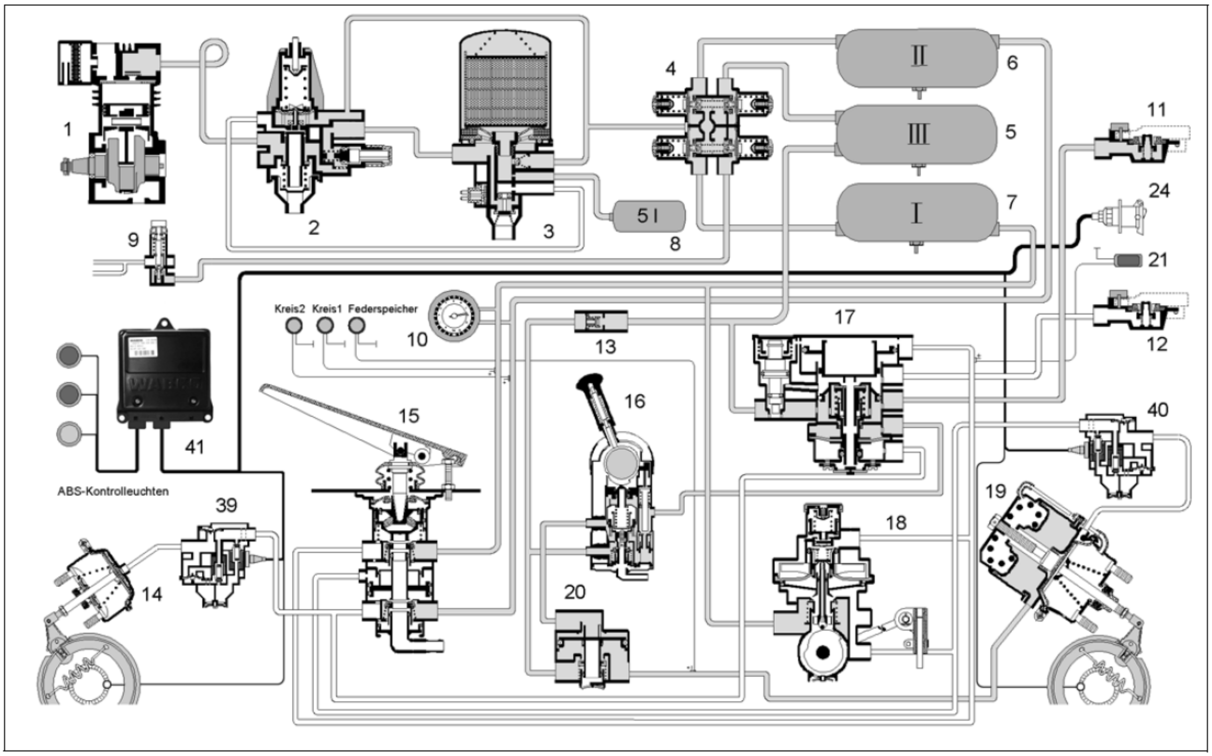

Figure 2.5: General layout of an air brake system in trucks [4]

As heavy vehicles use pneumatic braking systems that are now electronically controlled EBS, these systems have a sophisticated architecture with many control valves and filters. Some of the components are as follows:

1. Air filter
2. Air Dryer
3. Air compressor
4. Storage tanks or air reservoirs
5. Pressure control valves
6. Brake valves
7. Brake chambers

2.2.1 Working of an air brake system

To remove impurities and moisture, air is drawn in from the atmosphere, filtered, and dried. This air is then fed into the air compressor. The motor drives the compressor via a belt or gear drive. The air is compressed by the compressor and stored in air receivers. The compressed air is held within the confines of the reservoirs adjacent to the pressure control valves in the reservoir. When the foot brake module is actuated, the air from the reservoirs is directed into the brake chambers through a system of control valves and actuators, converting the pneumatic forces into braking torques. The high-pressure air from the reservoirs presses on a spring-loaded piston in the brake chamber, applying the brakes.

2.2.2 Parking Brakes/Spring brakes

For a robust parking brake system, spring parking brakes are installed on a vehicle with an air brake system. The brakes in the service brake system are applied by air pressure and retracted by springs, while the brakes in a spring parking brake system are applied by spring force and retracted by air pressure. Since the spring brake chambers are connected to the service brake chambers and operate through the same linkage, the effectiveness of the spring brake is dependent on the service brake setting.

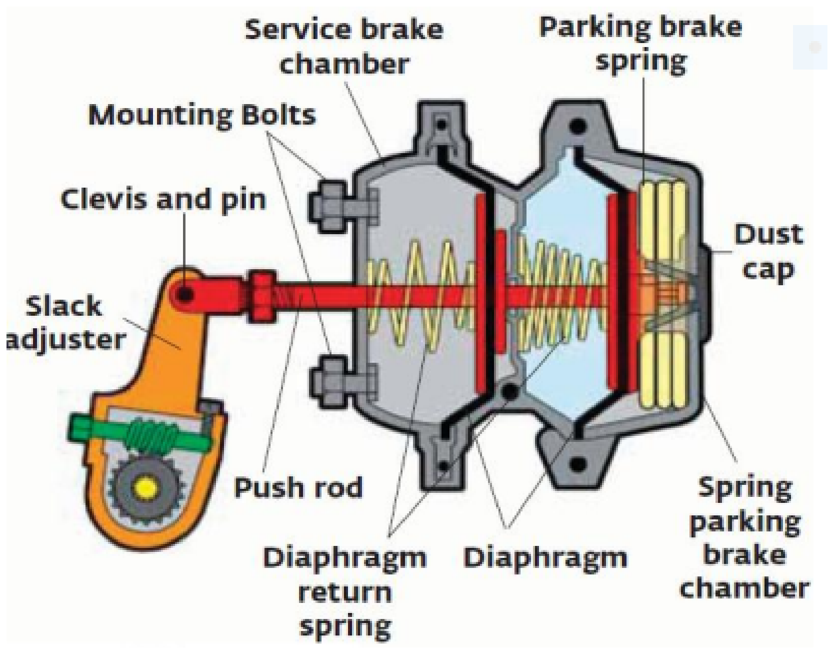

Figure 2.6: Parking brake disengaged, no service brake engaged

The cab parking brake switch, shown in the illustration 2.7 below, allows the driver to either release air from the parking brake spring accumulator to apply the brakes or pressurise the accumulator to release them. The parking brake system can be used as an emergency braking system if required. The parking brake is operated by a switch in the cab of the truck. A spring-loaded valve keeps the pressure in the brake chamber at about 8 to 9 bar when the brakes are released. Compressed air enters the brake chamber and is converted to mechanical work by the push-rod force, push rod is connected to a slack adjuster which in turn pushes the cam in case of a s-cam brake.

Figure 2.7: Parking brake switch [18]

A standard spring loaded brake chamber assembly is shown in figure 2.6. The chamber closest to the slack adjuster is the service brake chamber, while the other spring-loaded chamber is the spring brake chamber, used primarily for the parking brake and emergency brake functions. In one brake chamber, there is a diaphragm that assists the movement of the piston when air pressure is applied and pushes the push rod against the slack adjuster.

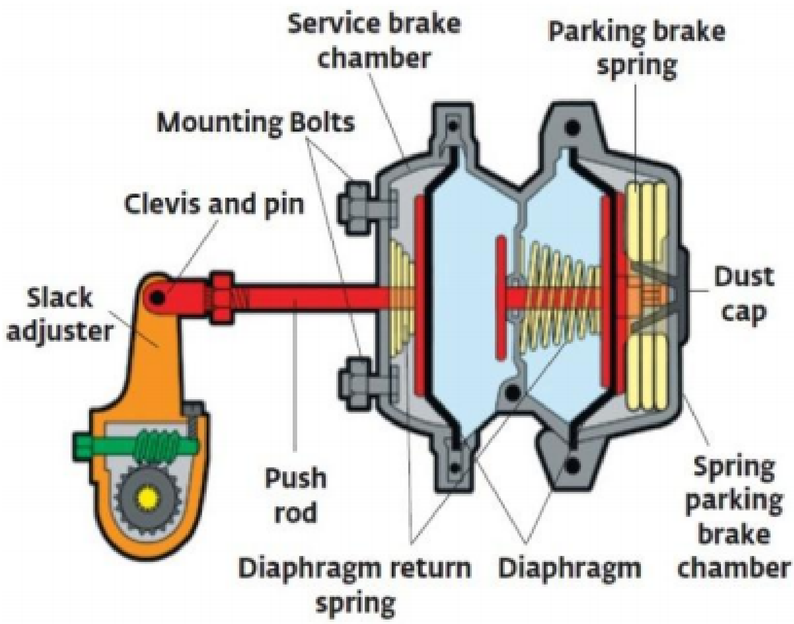

Figure 2.8: Parking brake disengaged, service brake engaged

When the brakes are applied, the spring is compressed or decompressed as needed. The operator can flow air into or out of the parking brake circuit to activate or deactivate the brakes through a control valve in the cab called the parking brake switch. When the air in the parking brake chamber is pressurised, the diaphragm compresses the spring, releasing the parking brake. When the air is released, the spring relaxes and the parking brakes are applied.

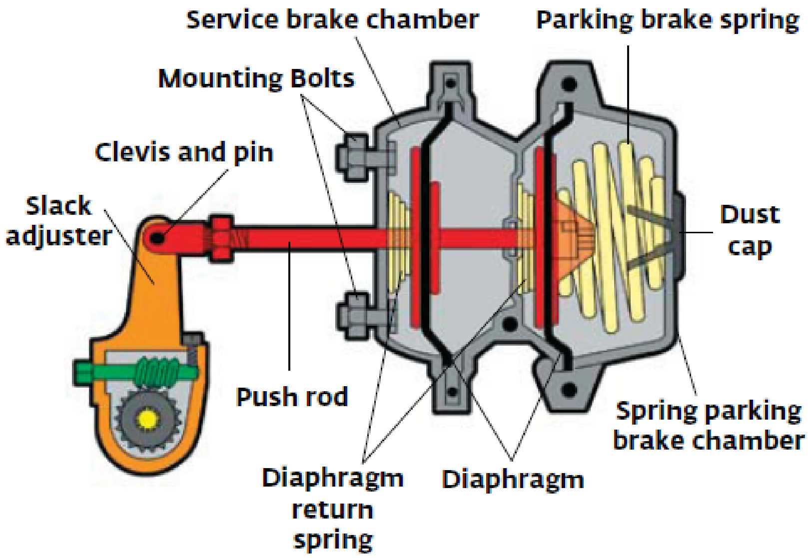

Figure 2.9: Parking brake engaged

2.3 Electronic Braking System (EBS)

The main disadvantage of a typical pneumatic braking system over a hydraulic braking system is that it takes longer to respond from the initial pressure requested by operating the brake pedals to the change in pressure in the brake chambers in the rear axles and trailers. This response time can range from 200 ms to 300 ms, which significantly affects the truck's braking performance. To solve this problem, air signals are replaced by electrical pulses via a Electronic Control Unit (ECU) that controls the braking system, known as the electronic braking system EBS [5].
The use of an electronic braking system EBS enables the optimisation of the driving and braking processes of commercial vehicles. The basic functions of the EBS include an electro-pneumatic brake, the Anti-Lock Braking System (ABS) and the Traction Control System (TCS). Electronic brake force adjustment in any braking situation can help reduce wear on connected components, increasing the overall economy of the braking system. It also makes it easier to monitor and maintain the braking system. The EBS electronic braking system works by sending an electronic control signal from the brake pedal sensors to the EBS control unit, which is then electronically processed and transmitted to the pressure control modules with almost no time delay. The pressure control modules pneumatically change the appropriate brake pressure in the wheel cylinders and simultaneously send sensor signals from the wheels to the EBS control unit. The traction between the tyres and the road can thus be optimally utilised.
In the event of a malfunction, the EBS resorts to a pure compressed air braking system

with two compressed air circuits. During braking, the pressure in the brake cylinders builds up uncontrollably, depending on the position of the brake pedal. By electronically and immediately activating the brakes, the pressure control modules enable the driver to experience a similarly direct braking sensation as in a passenger car.
The next subsections describes the working principle of individual modules involved in an air brake system.

2.3.1 Service Brake Module-SBM

The contact free sensor mechanism connects the Service Brake Module, also known as the Foot Brake Module, to the brake pedal. The driver depresses the foot brake module, it signals the EBS to apply the brakes. The pneumatic control of the module is based on brake pressures in both the rear and front axles. For commercial vehicles with front-mounted engines, rear axle brake pressure is lower than front axle brake pressure. When the EBS system is in backup mode, an unladen vehicle is more stable when the pressure to the rear axle is low, however in a loaded vehicle, the brake pedal stroke is increased. For commercial vehicles with rear mounted engines, brake pressure to the front axle is lower than to the rear axle for the same reason as stated above.

2.3.2 Trailer Control Module-TCM

It's a electro-pneumatic control valve that includes a relay valve, a filter, a muffler, offset shift solenoid valves, an inlet valve, an output valve, a pressure sensor, pneumatic connectors, and an electric connector. The offset shift valve shifts the trailer control pressure function to the front circuit braking function. During typical Trailer Control Module (TCM) operation, the EBS-ECU manages the trailer control pressure via valves, using data from pressure sensors. The offset valve is actuated, and the trailer control pressure is held as a function of the front circuit brake pressure. The TCM is regulated by the function of front circuit brake pressure and parking brake pressure due to a technical malfunction or failure in TCM/EBS-ECU. The offset shift valve is not engaged, and no trailer control pressure is held back [19].

2.3.3 Pressure Control Module-PCM

Pressure Control Module also known as Electro-Pneumatic Modulator The EBS-ECU communicates the necessary braking pressure to the Pressure Control Module (PCM) through brake-Controller Area Network (CAN). The pneumatic backup pressure controls the braking pressure when the PCM is in backup mode. It has a relay valve for each channel, a filter, a muffler, solenoid valves with backup, inlet and outlet valves, a pressure sensor, ECU, pneumatics, and electric connectors. It also functions as a panel point, sending the values of related signals (wheel speed and lining wear sensors) to the EBS-ECU through braking CAN. There are two PCM variants available: Single Channel (Basic version), which is used for a single axle and supports two Wheel Speed Sensors (WSS)/Lining Wear Sensors (LWS) with four electrical connectors, and Dual Channel (Full version), which is used for two axles and supports four WSS/LWS with eight electrical connectors.

2.3.4 Wheel Speed Sensor-WSS

It is a sensor that detects pulses from the wheel's pulse ring. The operating speed of the air gap is 3-9 km/h. If the air gap speed exceeds 9 km/h, the ABS warning bulb illuminates and a Diagnostics Code (DTC) is stored. The maximum frequency allowed by WSS is 1300Hz. It is determined by the number of teeth on the pulse ring and the spacing between the sensor and the pulse ring.

2.3.5 Lining Wear Sensor-LWS

It is a sensor that measures the entire thickness of a caliper's braking pads and disc. Calibration is usually performed during caliper assembly. It also measures the automated slack adjuster's adjustment.

2.4 Electronic Parking Brake System

2.4.1 Electronic Parking Brake Module-EPB

The EPB is a module that controls and actuates the valves necessary for the flow of air in the pneumatic parking brake circuit. The EPB is actuated by depressing the Parking brake switch inside the cab as shown in figure 2.7. Air pressure inside EPB module drops instantly after the driver requests for parking brake. But the system reaches the desired pressure after a brief delay.
The following figure represents the physical layout of the existing parking brake system at Scania along with the proposed estimator.

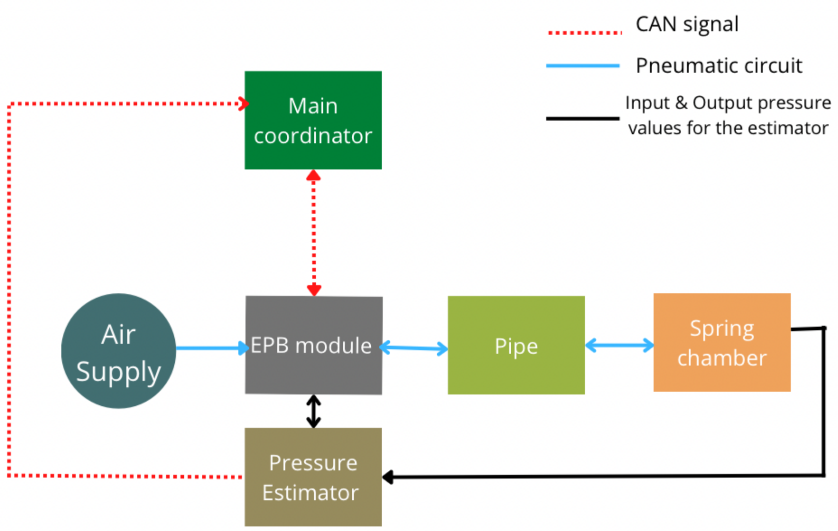

Figure 2.10: Block diagram of proposed estimator in EPB system circuit

When the parking brake is deactivated, the EPB module sends air from the secondary tank to the brake chamber, compressing the spring and releasing the parking brake. The diagram 2.11 illustrates the functioning of the parking brake circuit in the brake release mode.

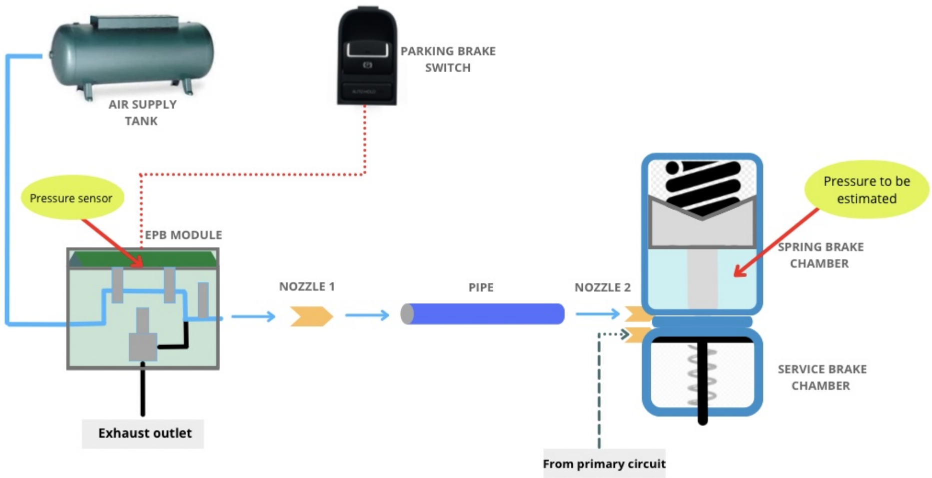

Figure 2.11: Schematic layout of parking brake system [Disengaged]

When the pressure inside the parking brake chamber is below the threshold pressure, the piston starts moving and engages the brake as the pressure decreases further more. The below 2.12 represents when the parking brake is in transition.

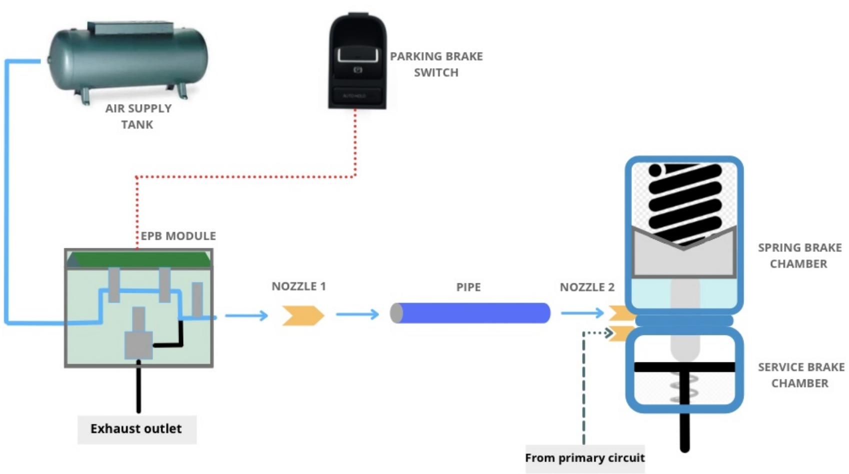

Figure 2.12: Schematic layout of parking brake system [Transition]

When the parking brake is actuated, the EPB module cuts off the air supply from the secondary tank to the brake chamber, causing the pressure in the brake chamber to drop, forcing the spring to decompress and apply the parking brake. The diagram 2.13 represents how the parking brake circuit works in the brake application mode.

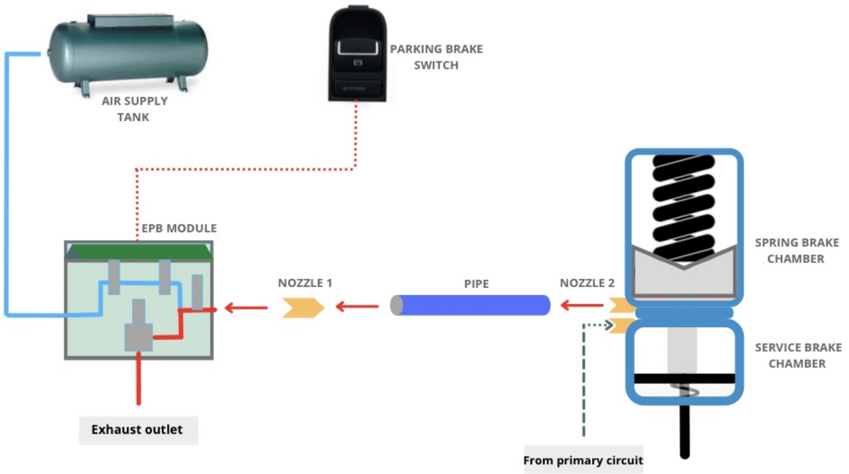

Figure 2.13: Schematic layout of parking brake system [Engaged]

3

Methods

In order to estimate the pressure inside the spring brake chamber, a model based approach is considered. The method used to develop a Simulink model of a pneumatic parking brake system derived from a physical operating system on a truck is explained in this chapter. The emphasis will be on modeling the sub-systems based on physical parameters.

3.1 Thesis Approach

The flowchart below depicts the approach we adopted for the thesis from beginning to end.

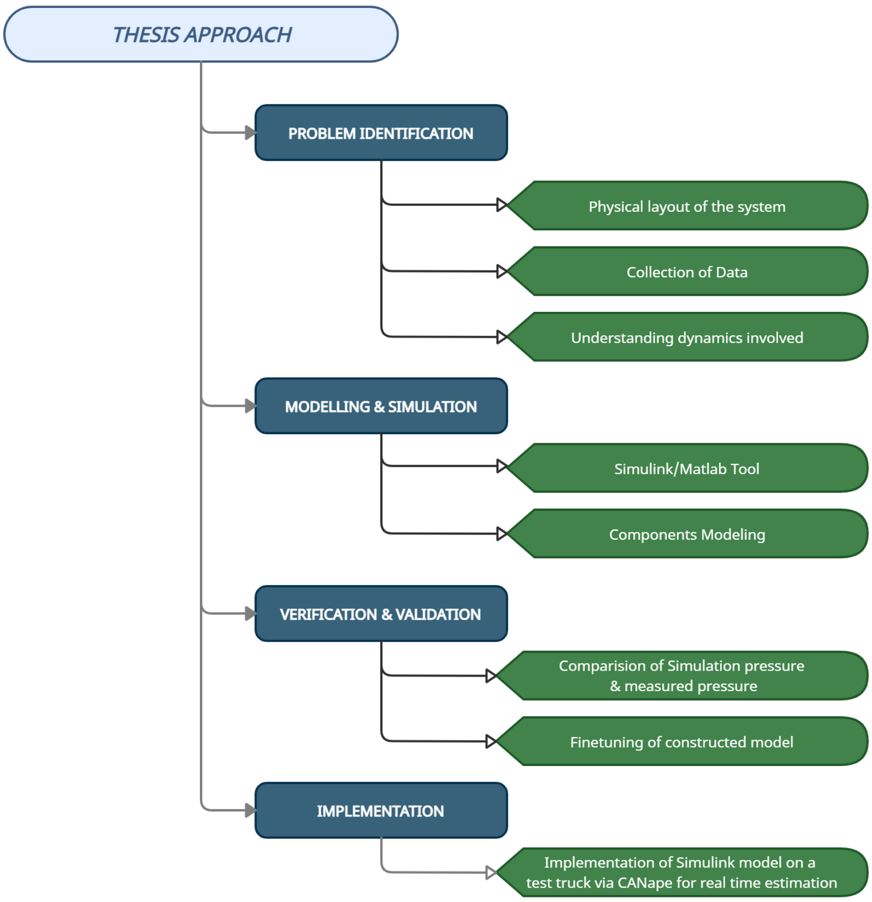

Figure 3.1: Flowchart of Thesis Approach

3.2 Development of Parking Brake System Model

We need to comprehend the system's theory in order to develop the simulink model of parking brake system. Due to the intricacy of the brake system components, it is important to formulate a simplified version of it. The simplified version will aid in estimating pressure inside the brake chambers. The following simplification of the components are made for the system modelling:

- The EPB module is attached to one end of the pipe via a connector, as seen in Fig. 2.11. As a result, in our system model, we will refer to the connector as Nozzle1.
- The pipe's other end is connected to a spring cylinder actuator through an elbow connector, which will be referred to as Nozzle2.
- To model the pipe in Simulink, length and cross section area of the pipe from nozzle 1 and 2 are considered.
- The spring brake cylinder is modelled considering the physics involved in it as shown in Fig.3.4.

Due to the sheer complexity of the EPB module and the number of solenoid valves involved inside, it is difficult to model valve opening and closing; thus, with the above implication, we are considering the pressure measured by the EPB module for further estimation of pressure inside the pipe and the spring brake cylinder.

3.3 Theory of air flow:

We model air as an ideal gas with constant temperature. The air flow process is also referred to as isentropic flow since no heat is transferred (negligible). Air flow modelling has traditionally relied on one-dimensional flow theory to properly depict air flow via restrictions (nozzle/orifice)[8].
The ideal gas law states that,

$$P \cdot V = m \cdot R \cdot T \tag{3.1}$$

where P is the pressure, V is the volume, m is the mass of air, R is the Universal gas constant amd T is the temperature.
We know that the density is the measure of mass per volume and is given by:

$$\rho = \frac{m}{V} \tag{3.2}$$

substituting equation (3.2) in equation (3.1) yields the density of a compressible fluid as a function of it's pressure and is given by

$$\rho = \frac{P}{R \cdot T} \tag{3.3}$$

Isentropic process of air flow is adiabatic and reversible in thermodynamics, the following relation for an ideal gas is considered [9][10]

$$PV^{\gamma} = constant \tag{3.4}$$

$$\gamma = \frac{C_p}{C_v} \tag{3.5}$$

where γ is the ratio of specific heat at constant pressure (C_p [J/K]) to the specific heat at constant volume (C_v [J/K]). The assumption for this ratio is considered to be constant[reference]

As we are considering the air is flowing through restriction which in this case is to be the nozzles, the mass flow rate through the nozzle is given by:

$$\dot{m} = C_d \cdot C_m \cdot A_n \cdot P_h \tag{3.6}$$

where $\dot{m}$ is the mass flow rate, C_m is the flow parameter of air, A_n is the surface area of the nozzle and P_h is the high pressure supply. Due to the uncertainty factors such as friction caused by a bend inside the nozzle etc, the coefficient of discharge C_d is taken into account. The value of C_d ranges from 0.8 to 0.95 depending on the complexity of the nozzle.
The nozzles are considered as cylinderical and the surface area of nozzle where the air acts is:

$$A_n = 2 \cdot \pi \cdot r \cdot l \tag{3.7}$$

where r is the radius and l is the length.

The flow parameter C_m is a function of the pressure ratio, critical pressure ratio and specific heat ratio. The critical pressure ratio is given as:

$$P_{CR} = \left(\frac{2}{\gamma + 1}\right)^{\frac{\gamma}{\gamma - 1}} \tag{3.8}$$

We already know that compressible fluids can never travel faster than sound. The choked flow state occurs when the speed of air flow equals the speed of sound. As a result, the choked flow occurs when

$$\frac{P_l}{P_h} = P_{CR} = 0.5283 \tag{3.9}$$

where P_l is low pressure supply, P_h is high pressure supply and P_{CR} is critical pressure ratio.

If the pressure ratio is greater than critical pressure ratio, then the flow parameter is given as:

$$C_m = \left(\frac{2\gamma}{RT(\gamma - 1)}\right)^{1/2} \cdot \left(\left(\frac{P_l}{P_h}\right)^{2/\gamma} - \left(\frac{P_l}{P_h}\right)^{\gamma+1/\gamma}\right)^{1/2} \tag{3.10}$$

If the pressure ratio is lower than or equal to the critical pressure ratio, then choked flow condition occurs and the flow parameter is given as:

$$C_m = \left(\frac{2\gamma}{RT(\gamma + 1)}\right)^{1/2} \cdot \left(\frac{2}{\gamma + 1}\right)^{\frac{1}{\gamma - 1}} \tag{3.11}$$

As a result, from the equation (3.7) to (3.11), the mass flow rate based on pressure differential through a restriction can be calculated as follows:

- Air supply phase
$$\dot{m} = C_d \cdot C_m \cdot A_n \cdot P_h \tag{3.12}$$
- Air release phase
$$\dot{m} = -C_d \cdot C_m \cdot A_n \cdot P_h \tag{3.13}$$

3.4 Pressure Dynamics

Since we are treating air to behave like an ideal gas, equation (3.1) can be written as:

$$m = \frac{P \cdot V}{R \cdot T} \tag{3.14}$$

As the one dimensional air flow is considered to be isentropic process, the pressure dynamics inside the brake chamber with respect to time is given by [9][10][15],

$$\dot{m} = \frac{1}{\gamma} \cdot \frac{\dot{P}V}{RT} + \frac{P\dot{V}}{RT} \tag{3.15}$$

The equations (3.12),(3.13) & (3.15) will serve as the foundation for our model in the thesis. The next section discusses more about the spring brake actuator dynamics

3.5 Spring Brake Actuator Dynamics

This section focuses more on the dynamics of the pneumatic actuator, Fig. 3.2 shows the test setup used for data measurement and validation purposes.

Figure 3.2: Test setup in truck Mjölner 4x2

The rear right wheel of a 4x2 truck was removed, and the production parking brake actuator was replaced with a type 24/30 parking brake actuator that had been modified with a force and displacement transducer attached on the pushrod. The pressure sensor was installed on a pipe near the brake actuator's inlet nozzle. The figure 3.3 gives a clear overview of measurement devices mounted in the setup.

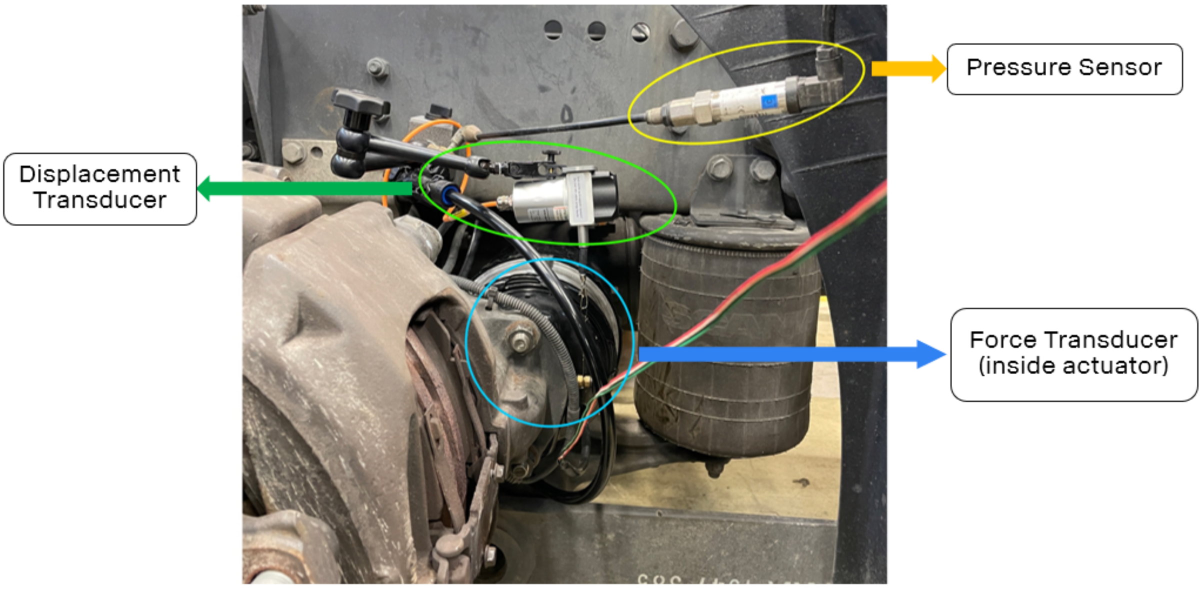

Figure 3.3: Mounting Position of measuring instruments

A force transducer is a device that converts a mechanical input force, such as a load, weight, tension, compression, or pressure, into another physical quantity, in this case an electrical output signal that can be measured. The electrical signal changes proportionally to the force applied to the force sensor. This transducer is connected to the push rod which gets reaction forces from the foundation brake part and the force exerted on the piston by the spring force [6].

A pressure transducer is a device that converts pressure input into the transducer into electrical signals. It is a strain gauge type transducer that detects elongation in the strain gauges located within the transducer's diaphragm. This pressure transducer is connected to the braking chamber through a T-Junction located 50cm away.[7].

A displacement transducer turns mechanical rectilinear motion or vibrations into electrical current, voltage, or signals. This transducer is connected to the piston within the brake chamber and is used to measure the distance travelled by the brake chamber piston.

The main purpose of testing using this configuration was to measure the stroke length of the spring piston at various pressures, as well as the change in force acting from the foundation brake during the air supply and release phases.

A free body diagram is constructed considering the dynamics involved in Type 24/30 spring brake actuator shown below:

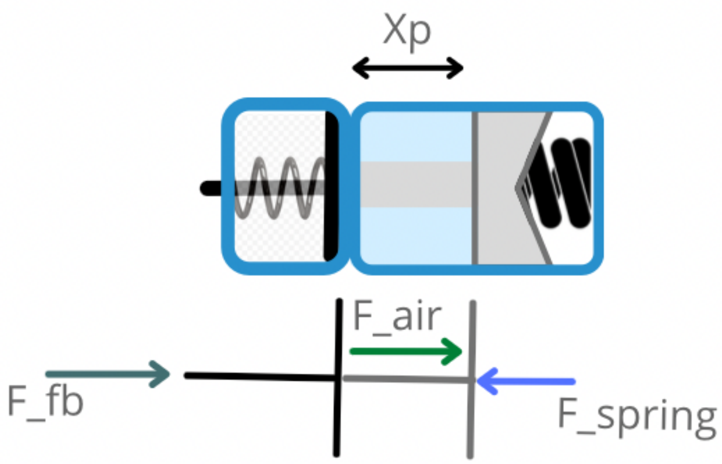

Figure 3.4: Free Body Diagram of Spring Brake Actuator

(Note: The model is valid as long as the contact push force between the two piston does not become negative. This can be carried out with validity check: $F_{spring} - F_{air} \geq 0$)

From the above figure,considering the force equilibrium,

$$F_{air} + F_{fb} = F_{spring} \tag{3.16}$$

The pneumatic force F_{air} is given by,

$$F_{air} = (P_c - P_{atm}) * A_p \tag{3.17}$$

The Spring force F_{air} is given by,

$$F_{spring} = F_{kpl} + M_p * (\frac{d^2 \cdot x_p}{dt^2}) + K_p * x_p \tag{3.18}$$

where P_c is the pressure in actuator cylinder, P_{atm} is the atmospheric pressure, A_p is the effective area of the spring piston M_p is the mass of the spring piston diaphragm, x_p is the piston position, F_{Fb} is the force from the foundation brake, F_{kpl} is the pre load force of spring piston, K_p is the spring constant of parking brake spring piston.
The equation of motion can be written as:

$$(P_c - P_{atm}) * A_p + F_{fb} - F_{kpl} = M_p * (\frac{d^2 \cdot x_p}{dt^2}) + K_p * x_p \tag{3.19}$$

We simplify the preceding equation for modeling the spring brake chamber by ignoring the interia of the spring piston. The service/primary brake portion's spring return force is between 200 and 600N during the entire stroke, hence it is included in the foundation brake force. As a result, the above equation can be expressed as follows:

$$x_p = \frac{(P_c - P_{atm}) * A_p + F_{fb} - F_{kpl}}{K_p} \tag{3.20}$$

Now, differentiating x_p with respect to time give the following equation:

$$\dot{x_p} = \frac{\dot{P_c} * A_p + \dot{F_{fb}}}{K_p} \tag{3.21}$$

The above the equation will be further used in the dynamic model in order to determine the position of piston at any instant of time.

Consider the variation of volume within the actuator as a function of pressure during the braking operation. When the electronic parking brake is applied, the pressure in the actuator cylinder begins to fall from its maximum to the desired pressure value. Given that the equilibrium point of the spring force and the combined opposed force (Pneumatic force + Foundation brake force) is at threshold pressure P_{th}, the diaphragm piston begins to move when the pressure in the actuator cylinder falls below the threshold pressure. When the pressure reaches the contact pressure P_{ct}, the force on the pushrod begins to increase as the pressure continues to fall. The diaphragm piston continues to move until the forces are balanced at the appropriate pressure level.

As a result, depending on the phase of operation, the volume can be separated into piecewise continuous flow function:

- if $P_c > P_{th}$ x_p=0,
$$V_c = V_{max} \tag{3.22}$$
- if $P_{ct} < P_c < P_{th}$ and $0 < x_p \leq x_{pct}$,
$$V_c = V_{max} - A_p \cdot x_p \tag{3.23}$$
- if $P_{ct} > P_c > P_{atm}$ and $x_{pct} < x_p \leq x_{pmax}$,
$$V_c = V_{ct} - A_p \cdot x_p \tag{3.24}$$

Now considering eqn (3.15) with these piecewise volume functions,

- if $P_c > P_{th}$
$$\dot{m} = \frac{1}{\gamma} \cdot \frac{\dot{P_c} V_{max}}{RT} \tag{3.25}$$
- if if $P_{ct} < P_c < P_{th}$ and $0 < x_p \leq x_{pct}$
$$\dot{m} = \frac{1}{\gamma} \cdot \frac{\dot{P_c} V_{max}}{RT} - \frac{P_c \dot{V_c}}{RT} \tag{3.26}$$

$$= \frac{1}{\gamma} \cdot \frac{\dot{P}_c V_{max}}{RT} - \frac{P_c A_p \dot{x_p}}{RT}$$

from equation (3.18), above equation can be written as

$$= \frac{1}{\gamma} \cdot \frac{\dot{P}_c V_{max}}{RT} - \frac{P_c A_p}{RT} \cdot \left(\frac{\dot{P}_c * A_p + \dot{F_{fb}}}{K_p} \right)$$

$$= \frac{1}{\gamma} \cdot \frac{\dot{P}_c V_{max}}{RT} - \frac{P_c A_p^2 \dot{P}_c}{RTK_p} - \frac{P_c A_p \dot{F_{fb}}}{RTK_p} \tag{3.27}$$

Since the $\dot{F_{fb}}$ is very low at this operating range, the term can be neglected

$$= \frac{1}{\gamma} \cdot \frac{\dot{P}_c V_{max}}{RT} - \frac{P_c A_p^2 \dot{P}_c}{RTK_p}$$

Therefore the final equation for this piecewise volume function at this operating range can be written as:

$$\dot{m} = \dot{P}_c \cdot \left(\frac{V_{max}}{\gamma RT} - \frac{P_c A_p^2}{RTK_p} \right) \tag{3.28}$$

- if $P_{ct} > P_c > P_{atm}$ and $x_{pct} < x_p \leq x_{pmax}$

  For this volume function equation (3.24) is considered, Since the foundation brake force $\dot{F_{fb}}$ starts acting gradually with decrease in pressure

$$= \frac{1}{\gamma} \cdot \frac{\dot{P}_c V_1}{RT} - \frac{P_c A_p^2 \dot{P}_c}{RTK_p} - \frac{P_c A_p \dot{F_{fb}}}{RTK_p}$$

  Rearranging the above equation we get:

$$\dot{m} = \dot{P}_c \cdot \left(\frac{V_1}{\gamma RT} - \frac{P_c A_p^2}{RTK_p} \right) - \frac{P_c A_p \dot{F_{fb}}}{RTK_p} \tag{3.29}$$

As a result, equations (3.22), (3.25), and (3.26) reflect first order non linear differential equations used to calculate the precise pressure inside the spring brake chamber.

From the above derivations, the following equations are grouped in the submodels which corresponds to the physical subsystem which is used in the Simulink implementation of the model.

- **Nozzle1 model**

  during air supply phase

$$\dot{m} = C_d \cdot C_m \cdot A_n \cdot P_{EPB} \tag{3.30}$$

  during air release phase

$$\dot{m} = -C_d \cdot C_m \cdot A_n \cdot P_{pipe} \tag{3.31}$$

- **Pipe model**

$$\dot{m} - \dot{m}_2 = \frac{1}{\gamma} \cdot \frac{\dot{P}_{pipe} \cdot V_{pipe}}{R \cdot T} \tag{3.32}$$

- **Nozzle2 model**

  during air supply phase

$$\dot{m}_2 = C_d \cdot C_m \cdot A_n \cdot P_{pipe} \tag{3.33}$$

  during air release phase

$$\dot{m}_2 = -C_d \cdot C_m \cdot A_n \cdot P_c \tag{3.34}$$

- **Spring brake cylinder model**

  if $P_c > P_{th}$ and x_p=0

$$\dot{m}_2 = \frac{1}{\gamma} \cdot \frac{\dot{P}_c V_{max}}{RT} \tag{3.35}$$

  if $P_{ct} < P_c < P_{th}$ and $0 < x_p \leq x_{pct}$

$$\dot{m}_2 = \dot{P}_c \cdot \left(\frac{V_{max}}{\gamma RT} - \frac{P_c A_p^2}{RTK_p} \right) \tag{3.36}$$

  if $P_{ct} > P_c > P_{atm}$ and $x_{pct} < x_p \leq x_{pmax}$

$$\dot{m}_2 = \dot{P}_c \cdot \left(\frac{V_1}{\gamma RT} - \frac{P_c A_p^2}{RTK_p} \right) - \frac{P_c A_p \dot{F}_{fb}}{RTK_p} \tag{3.37}$$

In the Simulink model the state variables are

- $\dot{m}$
- $\dot{m}_2$
- P_c
- F_{fb}

4

Model implementation in Simulink

This chapter will further explain the development of simulink model considering the physical components mentioned in the previous section. To determine the pressure inside the spring brake chamber, the model is developed for the supply phase and exhaust phase of the braking system at given operating conditions. The following figure represents the model layout of pneumatic parking brake system in Simulink.

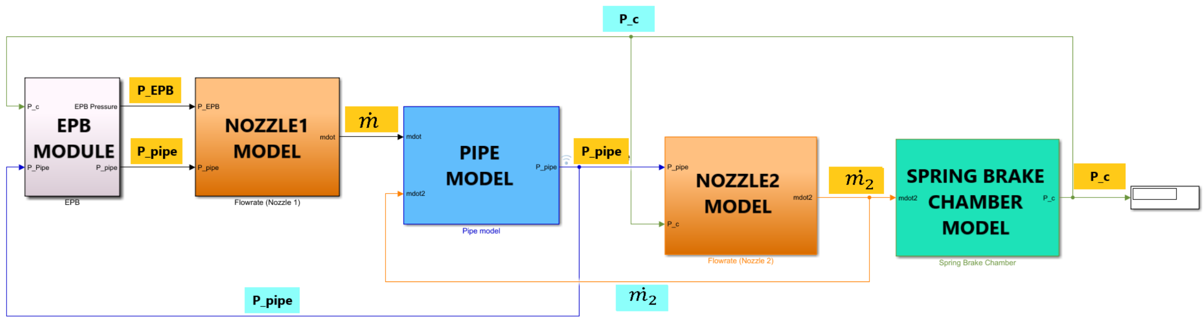

Figure 4.1: Simulink model layout

Since the system is built around nonlinear differential equations, the simulink model employs the ODE45 solver, which solves using the 'Runge Kutta' method with a predefined step size to precisely estimate the pressure. The functioning principle of the simplified component models is briefly outlined in the next subsections.

4.1 Nozzle 1

The figure 4.2 represents the block diagram of the nozzle1 model. The input to this block is the pressure from the EPB and the pressure inside the pipe model.

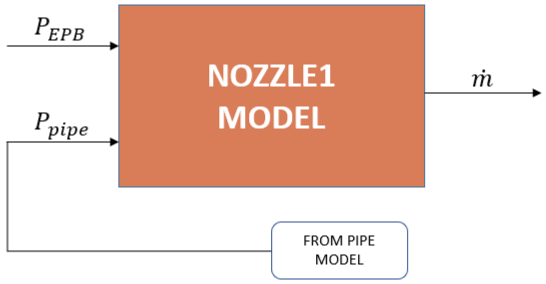

Figure 4.2: Nozzle1 model block representation

The simulink model for air flow through restriction was developed using eqns (3.6) to (3.13). The figure 4.3 represents the simulink model for the air flow through nozzle.

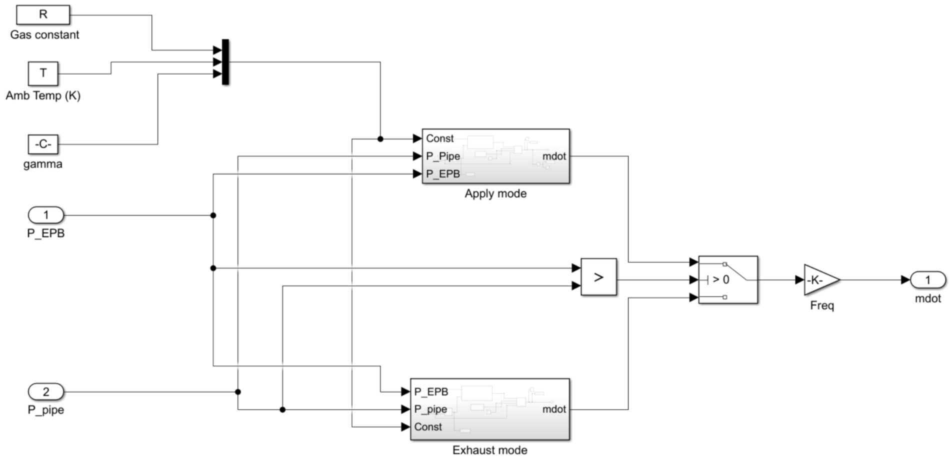

Figure 4.3: Nozzle1 Simulink model

The pressure from the EPB and the pressure in the pipe are the inputs to this simulink block. If the pressure in the EPB is greater than the pressure in the pipe, the mass flow rate increases until the pressure in the pipe equals the pressure in the EPB. This is described as supply phase. Similarly, if the pressure in EPB is lower than the pressure in pipe, the mass flow rate begins to decrease until the required EPB pressure in pipe is reached. This is referred as exhaust phase.

4.2 Pipe Model

The figure 4.4 represents the pipe model's block diagram. The mass flowrate entering the pipe ($\dot{m}$) and the mass flowrate leaving the pipe($\dot{m}_2$) are the inputs to this block.

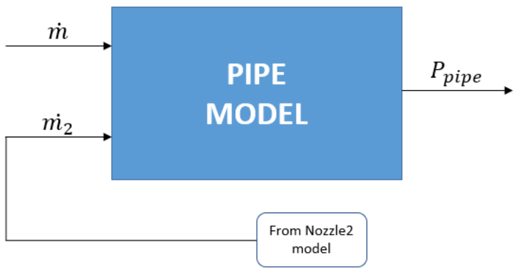

Figure 4.4: Pipe model block representation

The simulink model for pipe model was developed using eqns (3.15). The figure 4.5 represents the simulink model for the pipe.

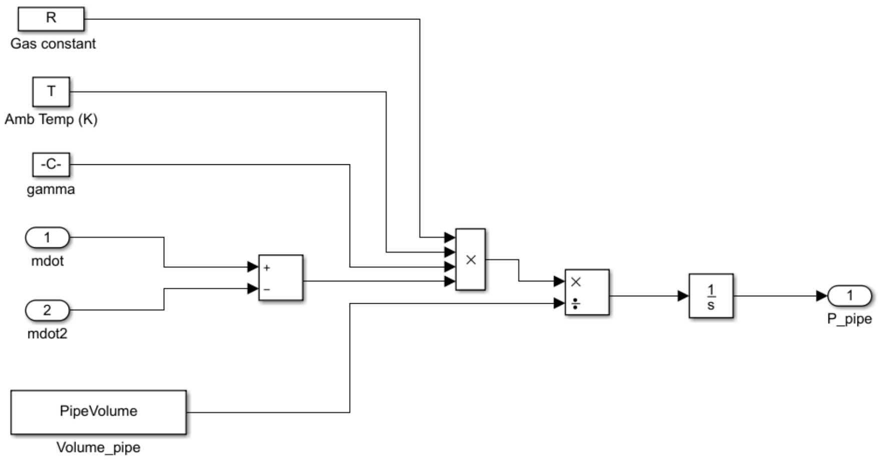

Figure 4.5: Pipe Simulink model

The volume of the pipe, as well as the constants, were defined within the model, and the chamber dynamics from eqn (3.15) were used to describe it. The resultant pressure change is integrated over time from the initial pressure given in the model.

4.3 Nozzle 2

The obtained pressure from the pipe model is given as input to the nozzle2 model along with the initial pressure inside the spring brake chamber represented in 4.6.

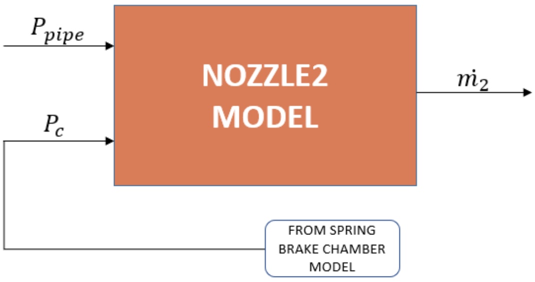

Figure 4.6: Nozzle2 model block respresentation

The figure 4.7 represents the simulink model for the air flow through nozzle, similar to nozzle1 model.

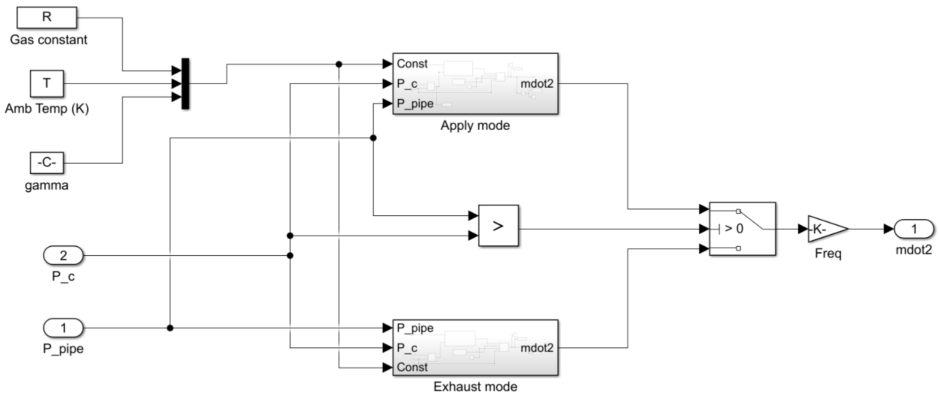

Figure 4.7: Nozzle2 Simulink model

The functioning principle for the nozzle2 model is the same as for the nozzle1 model. The main difference is that when the pressure in the pipe exceeds the pressure in the spring brake chamber, the mass flow rate increases until the pressure in the spring brake chamber equals the pressure in the pipe. Similarly, if the pressure in the pipe is less than the pressure in the spring brake chamber, the mass flow rate decreases until the required EPB pressure in the pipe and spring brake chamber is reached.

4.4 Spring-Brake Cylinder Model

The figure 4.8 represents the block diagram of the volume function of spring brake chamber. The inputs to this volume function is mass flow rate ($\dot{m}_2$) and the initial pressure inside the cylinder.

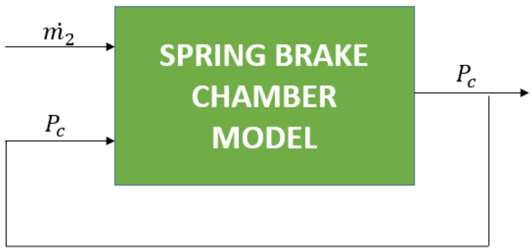

Figure 4.8: Spring brake volume function block representation

The simulink model for spring brake volume function model was developed using eqns (3.22),(3.25) & (3.26). The figure 4.5 represents the complexity of simulink model for the spring brake chamber.

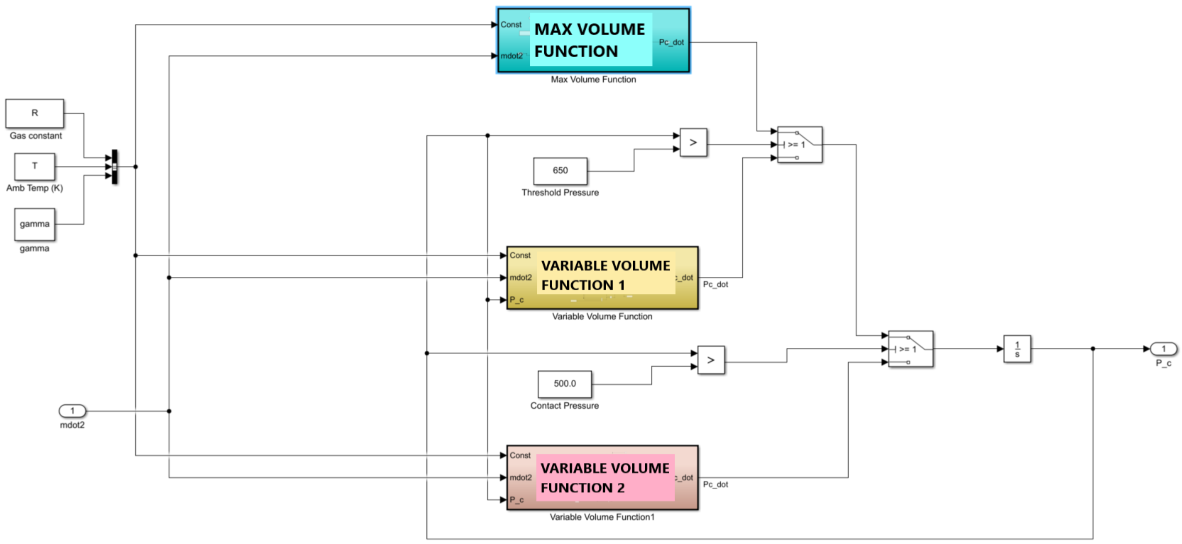

Figure 4.9: Spring cylinder simulink model

Section 3.4 provides a brief explanation of chamber dynamics as well as the derived equations (3.22), (3.25), and (3.26). The pressure derived from this model is utilized to determine the operation of the spring brake chamber. In the simulink, the cylinder pressure is initialized concurrently with the pipe pressure. The model predicts the pressure inside the pipe and the spring brake chamber using pressure inputs from the EPB.

5
Results

This section will focus on the verification and validation of the pressure estimator along with the measured pressure. It further more deals with the findings during the tests. The implementation of this model into the truck will be discussed at the end of this section.

5.1 Test Data Analysis

Several tests were carried out under various operating conditions to better understand the behaviour of pressure, force, and stroke inside the spring brake. The test setup is depicted in Figures 3.2 and 3.3. Using the test findings, the model was fine-tuned based on the spring brake dynamics in terms of pressure, force, and stroke.

5.1.1 Test case 1: Full pressure release

In this test case, a full drop in pressure i.e., from 8.5 bar to 0.5 bar was requested in EPB via Vector CANape tool. The figure 5.1 represents the pressure in EPB module and pressure inside the brake chamber as a function of time. The gradient of the slope changes as the pressure within the brake chamber reaches near to 5 bar, and then changes again when it reaches close to 4 bar.

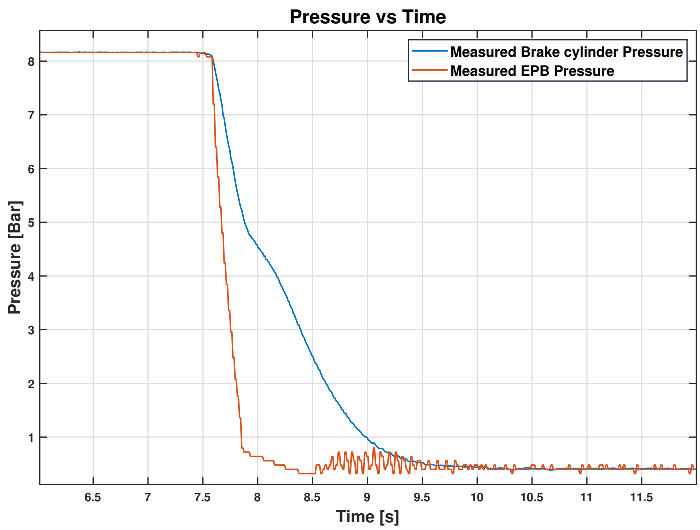

Figure 5.1: Pressure vs Time from 8.5 bar to 0.5 bar

The rationale for the slope's multiple shift in gradient from the above figure 5.1 must be considered. The figure below shows the pressure in the braking chamber as well as the piston stroke. The stroke starts to change only when the pressure inside the brake chamber drops to 5 bar. The slope of the stroke curve changes gradually between 4 - 5 bar and below 4 bar the stroke increases gradually but pressure drops rapidly.

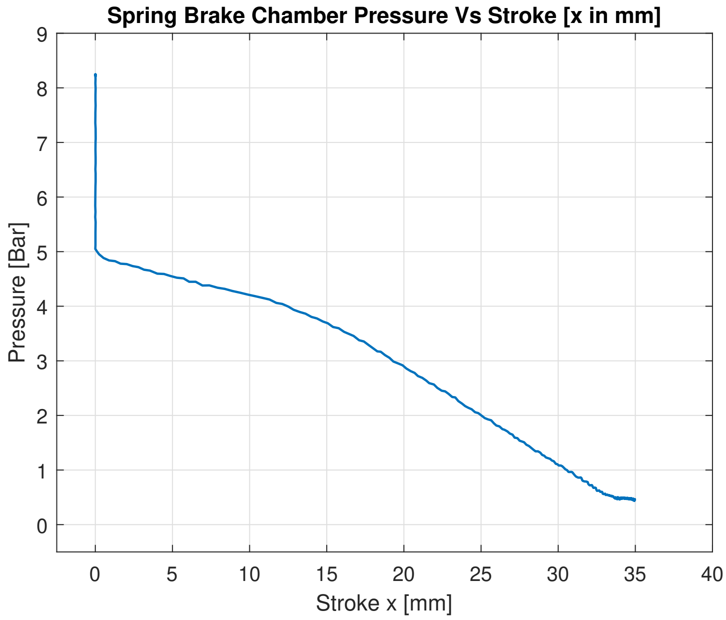

Figure 5.2: Pressure vs Stroke from 8.5 bar to 0.5 bar

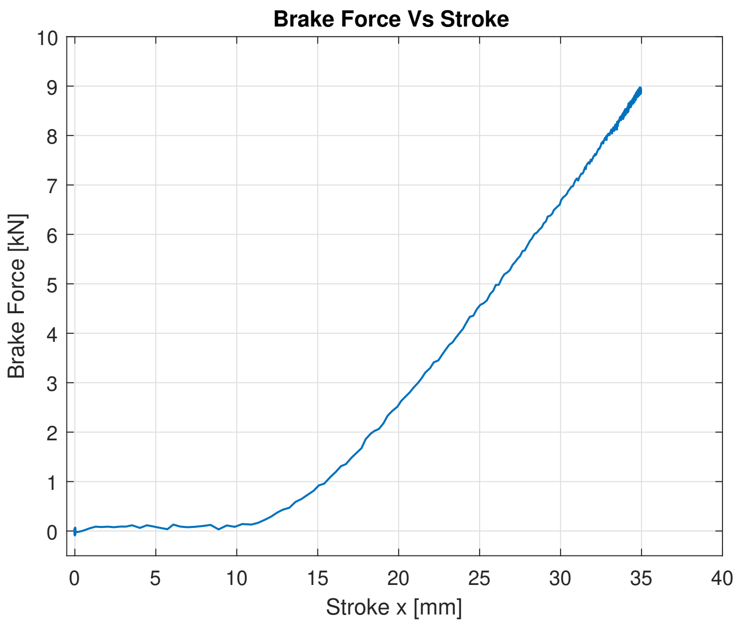

Figure 5.3: Brake Force vs Stroke from 8.5 bar to 0.5 bar

Figure 5.3 shows the brake force exerted on the piston along the stroke when the pressure drops from 8.5 bar to 0.5 bar. Until 15mm stroke length the brake force remains below

1kN. The brake force progressively grows when the piston stroke starts increasing after 15mm of stroke and reaches its maximum force at maximum stroke.

From figures 5.2 and 5.3, it is evident that when the pressure drops from 8.5 bar to 5 bar, the stroke does not change and there is no braking force acting on the piston. When the pressure decreases from 5 bar to 4 bar, the brake force increases progressively. This force overcomes the distance between the brake pads and the disc rotor and is the same as the return spring's pre-load force inside the service brake chamber.

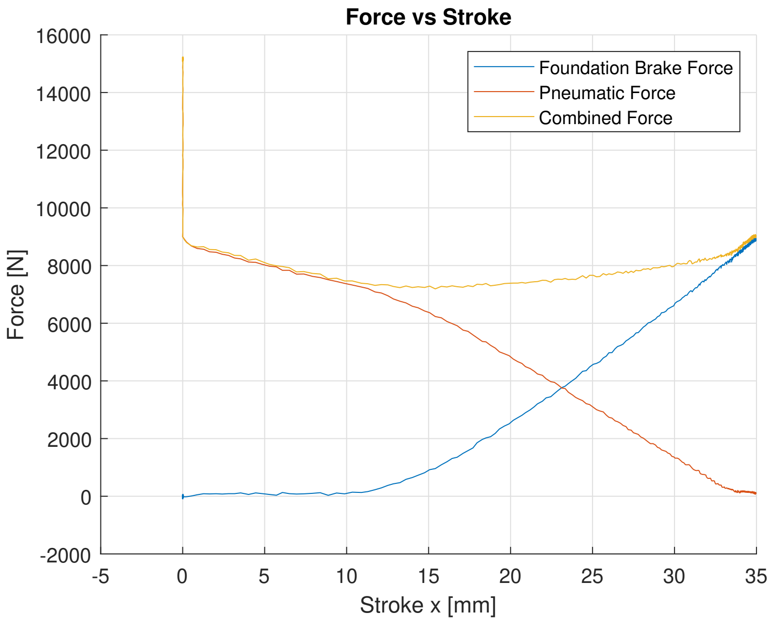

Figure 5.4: Force vs Stroke from 8.5 bar to 0.5 bar

Figure 5.4 shows the forces acting inside the brake chamber across the stroke when the pressure is requested from 8.5 bar to 0.5 bar. Blue line represents foundation brake force, red represents pneumatic force, and yellow represents the combined force, which is the combination of the brake and pneumatic forces. The stroke is maximum when the combined force is equal to springs pre-load force.

5.1.2 Test case 2: Partial pressure release

In this test case a partial drop of pressure from 8.5 bar to 3.5 bar was requested. It can be observed from the figure 5.5, the EPB pressure and measured pressure are plotted wrt time. The result of performing this test is to understand change in behaviour of the forces and pressure when a partial drop is requested.

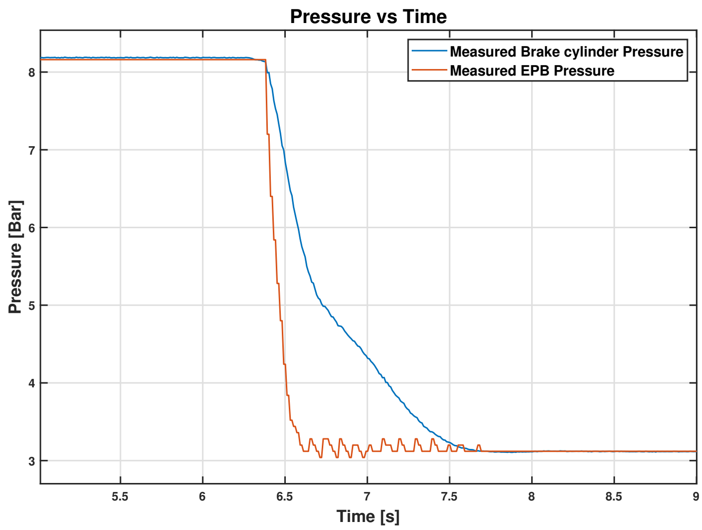

Figure 5.5: Pressure vs Time from 8.5 bar to 3.5 bar

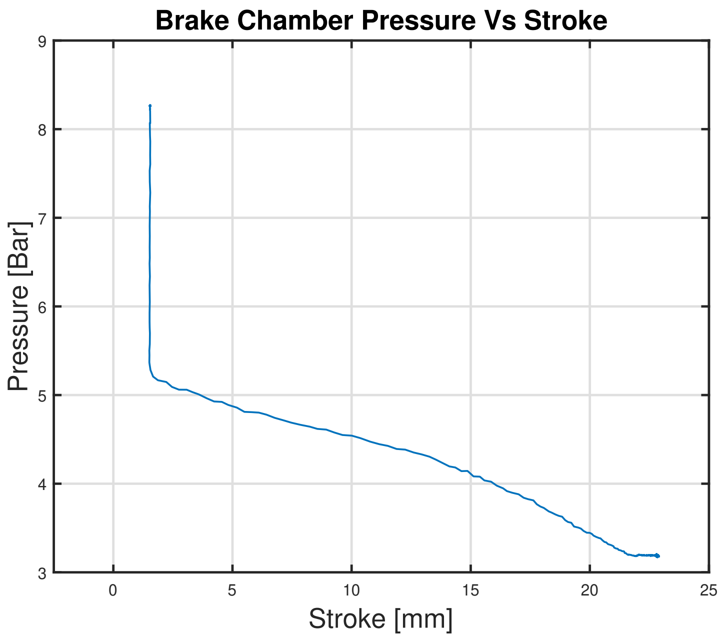

Figure 5.6: Pressure vs Stroke from 8.5 bar to 3.5 bar

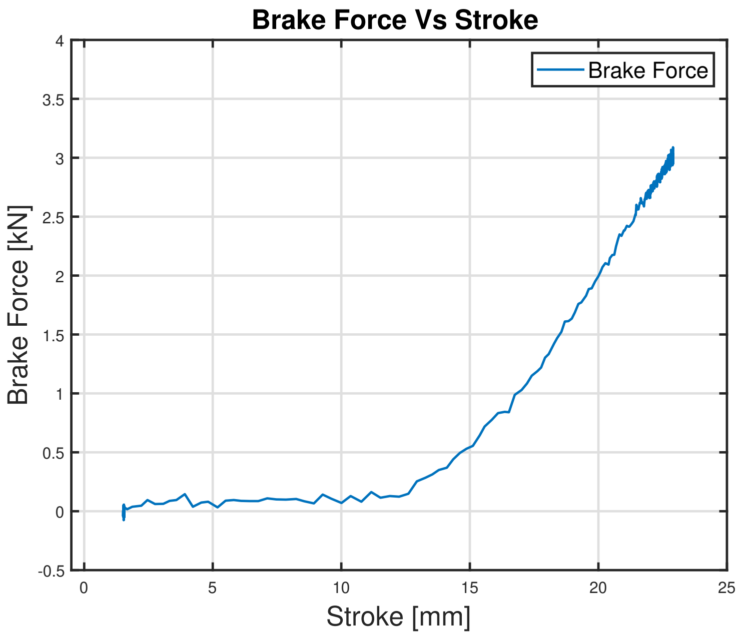

Figure 5.7: Brake Force vs Stroke from 8.5 bar to 3.5 bar

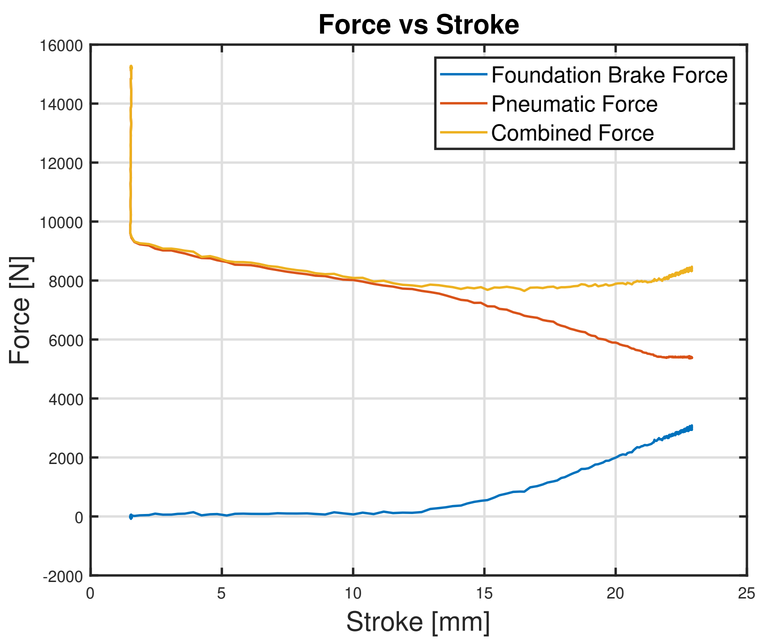

Figure 5.8: Force vs Stroke from 8.5 bar to 3.5 bar

From the figures in this subsection, the brake chamber dynamics are similar to the previous test case where a complete drop of pressures were requested and there was no higher deviation recorded.

5.1.3 Test case 3: Full pressure release and supply

In this test case a full pressure release and apply cycle was requested. This test gives an understanding of change in dynamics inside the brake chamber when the parking brake is applied and released. Fig. 5.9 represents the pressures inside the brake chamber plotted against time. The pressure release phase has a drop from 8.5bar to 0.5bar and the supply phase has a pressure increase of 5.5bar requested by the EPB.

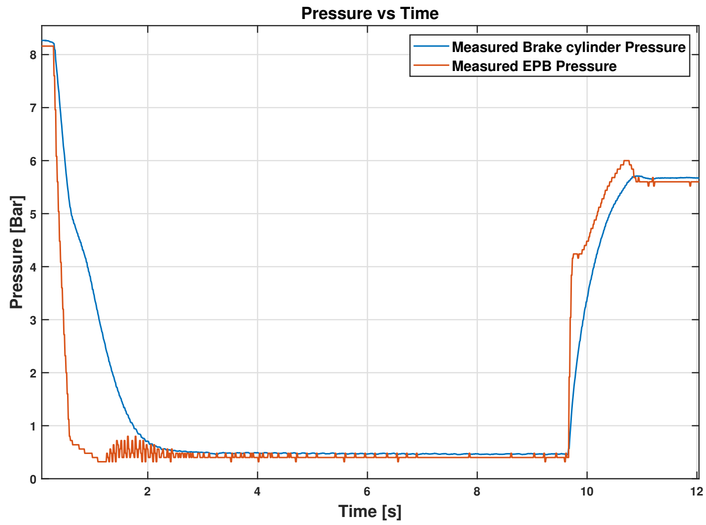

Figure 5.9: Pressure vs Time for complete parking brake cycle

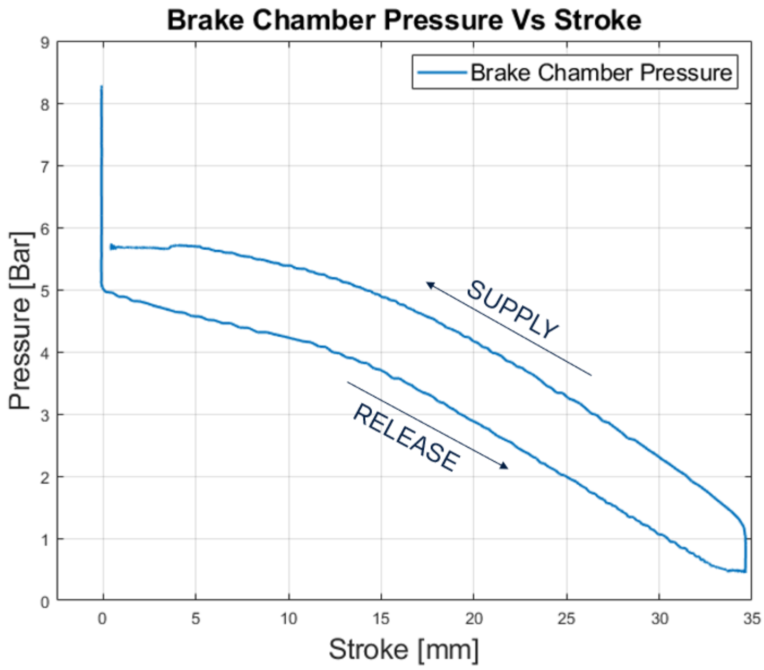

Figure 5.10: Pressure vs Stroke for complete parking brake cycle

Figure 5.10 shows pressure hysteresis occurring during the release and supply stages of parking brake application. The difference between the two curves is around 0.6 - 0.8 bar, showing that when air is supplied into the brake chamber, the piston begins to move backwards at approximately 0.6-0.8 bar greater than during the release phase.

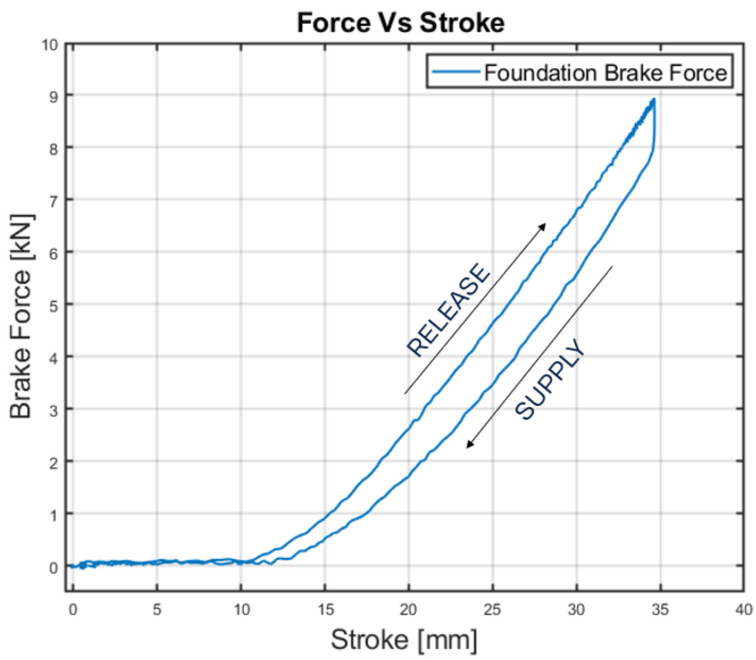

Figure 5.11: Brake force vs Stroke for complete parking brake cycle

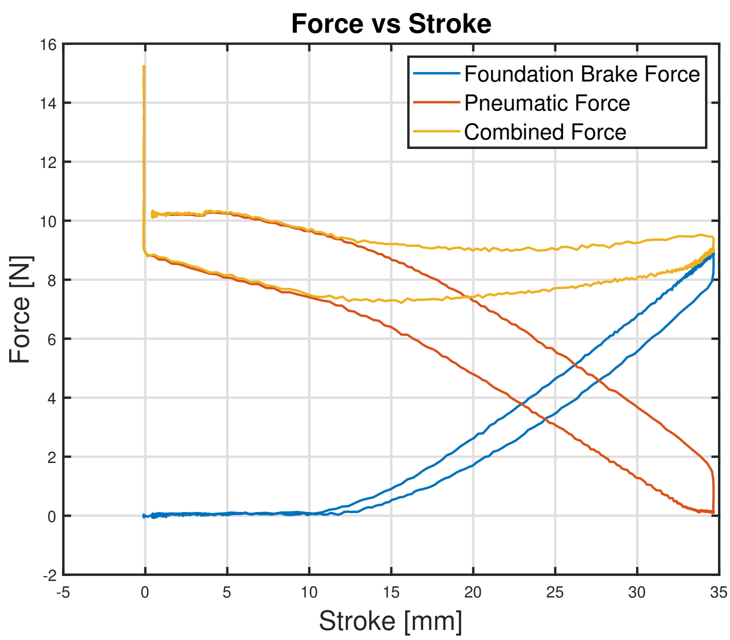

Figure 5.12: Force vs Stroke for complete parking brake cycle

Figure 5.11 represents the force hysteresis curves of the parking brake chamber during the release and supply phases.The force on the brake piston is lower during the supply phase than it is during the release phase. This is owing to the fact that during the supply phase, the force generated by the air on the brake piston is larger than the spring force, whereas during the release phase, the spring force is more than the pneumatic force. In figure 5.12, hysteresis of all the forces acting inside the brake chamber is illustrated.

5.2 Simulation Results

The simulink model was given initial values based on the data from the various test cases in order for it to predict the pressure as near to the measured pressure as possible. While estimating the stroke from the pressure inside the model is challenging, a lookup table is generated from the aforementioned test cases for the stroke at different pressures in order to estimate the foundation braking force inside the model.

5.2.1 Parameters from the Truck

The model requires numerical values on many parameters. The physical parameters used in the model for the estimation are shown in the table below.

S.no	Parameters	Notation	Values	Units
1	Surface area of nozzle 1	A_{n1}	**	m^2
2	Area of Spring piston	A_p	**	m^2
3	Surface area of nozzle 2	A_{n2}	**	m^2
4	Brake chamber spring rate	K_p	**	KN/m
5	Temperature	T_{amb}	300	K
6	Universal Gas Constant	R	287	J/Kg-K
7	Brake chamber max volume	V_{max}	**	m^3
8	Brake chamber initial volume	V_1	**	m^3
9	Specific heat ratio	γ	1.4	
10	Pipe diameter	d_p	**	m
11	Pipe length	l_p	**	m
12	Coefficient of Discharge	C_d	0.82	
13	Atmospheric Pressure	P_{atm}	101.3	kPa
14	Threshold Pressure	P_{th}	**	kPa
15	Contact Pressure	P_{ct}	**	kPa

Table 5.1: Parameters considered in the simulation (** Scania confidential values)

The simulink model was subjected to various pressure demands from the EPB while taking the aforementioned characteristics into account in order to determine how accurately the model behaves when compared to the measured pressure. The plots below show the Simulated pressure as well as the measured pressure for various pressure demands from the EPB.

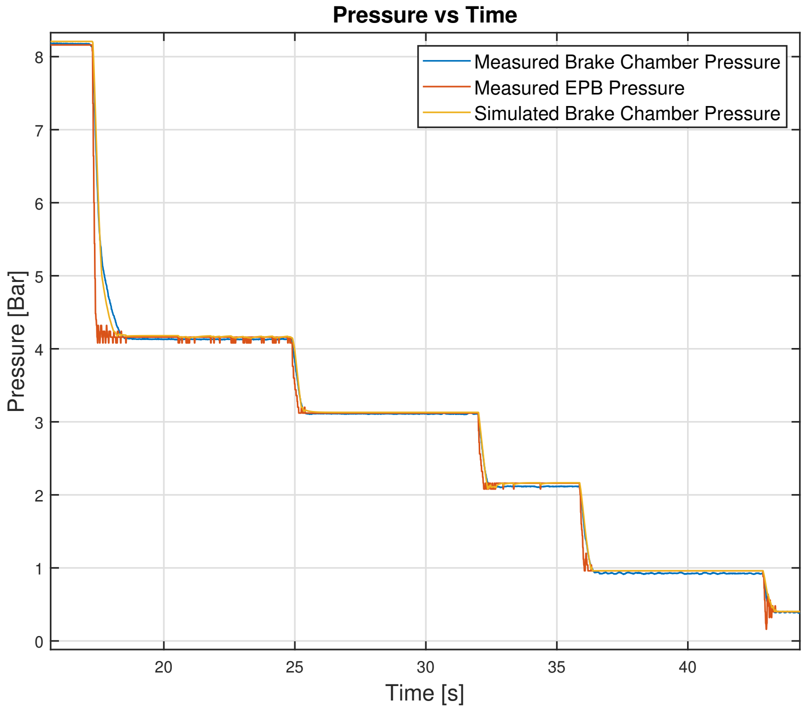

Figure 5.13: Simulated brake chamber pressure response along with measured brake chamber pressure via requested step response in EPB from 8.5 bar to 0.5 bar

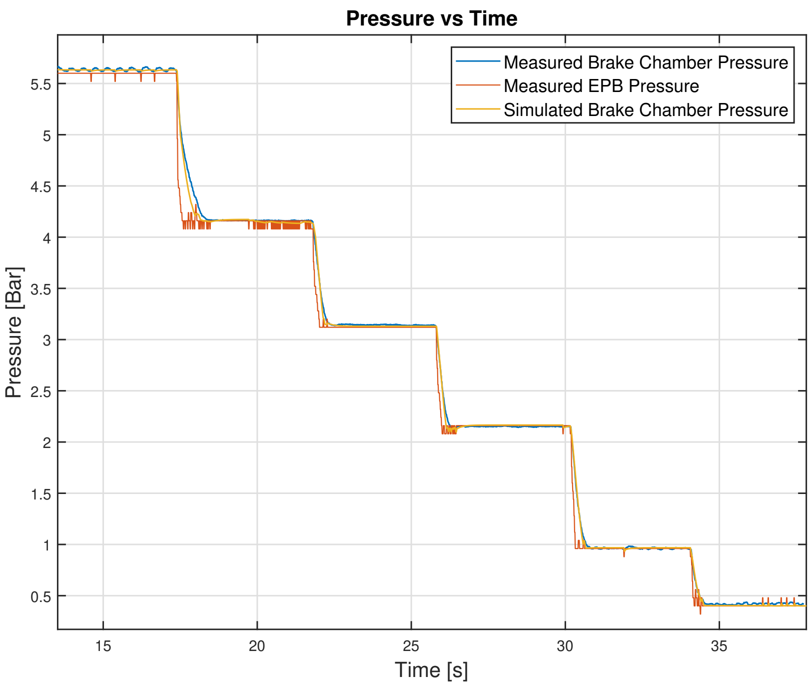

Figure 5.14: Simulated brake chamber pressure response along with measured brake chamber pressure via requested step response in EPB from 5.5 bar to 0.5 bar

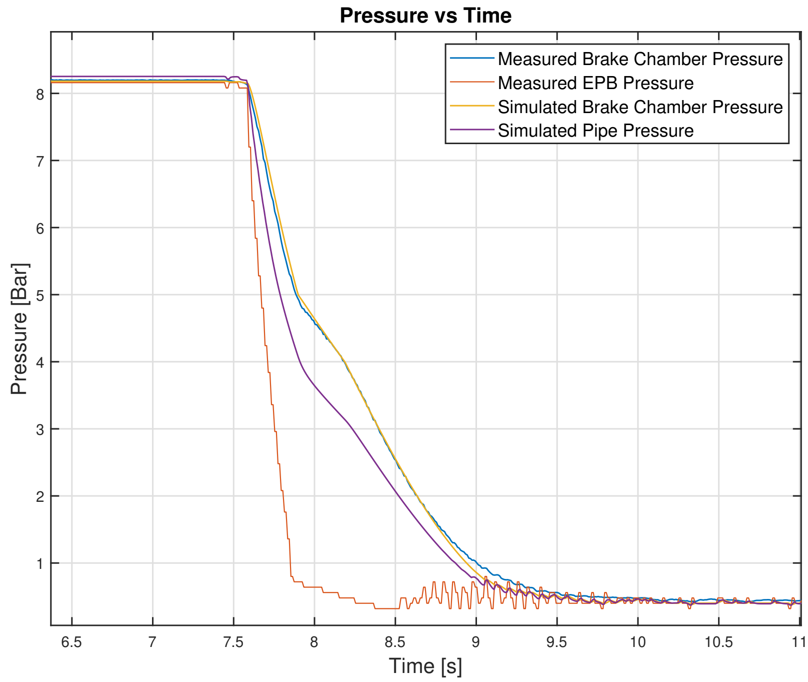

Figure 5.15: Simulated brake chamber pressure response along with measured brake chamber pressure via requested full drop in EPB from 8.5 bar to 0.5 bar

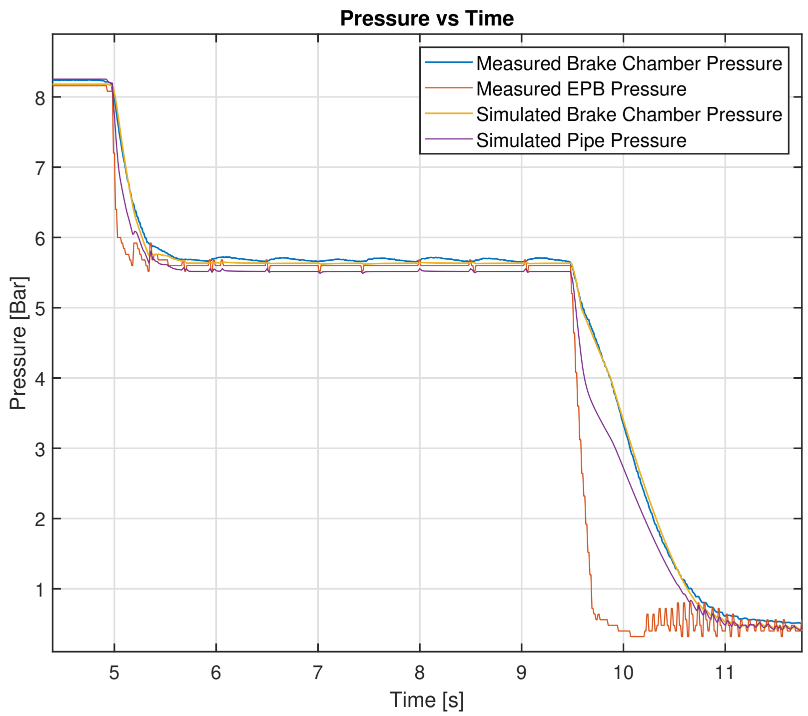

Figure 5.16: Simulated brake chamber pressure response along with measured brake chamber pressure via requested 2 step response in EPB from 8.5 bar to 5.5 bar & 5.5 bar to 0.5 bar

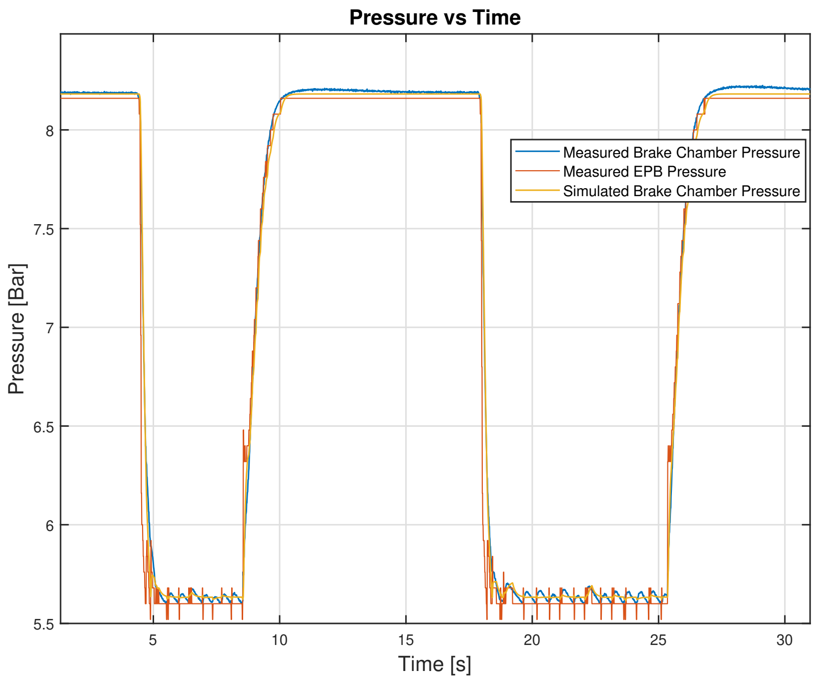

Figure 5.17: Simulated brake chamber pressure response along with measured brake chamber pressure via partial request cycle in EPB from 8.5 bar to 5.5 bar

The plots above show that the simulink model can estimate the pressure inside the brake chamber and inside the pipe using the truck's existing parameters. The simulink model is capable of handling any EPB pressure demands.

5.2.2 Connecting extra 4 meter pipe length in the pneumatic circuit

The objective of this modification was to test and validate how the model will react if any of the truck's physical parameters change. 4 meters of pipe were added to the truck's current arrangement for this test. With this modest alteration, similar tests to those mentioned above were performed. The plots below show the Simulated pressure response as well as measured pressure for various EPB requests.

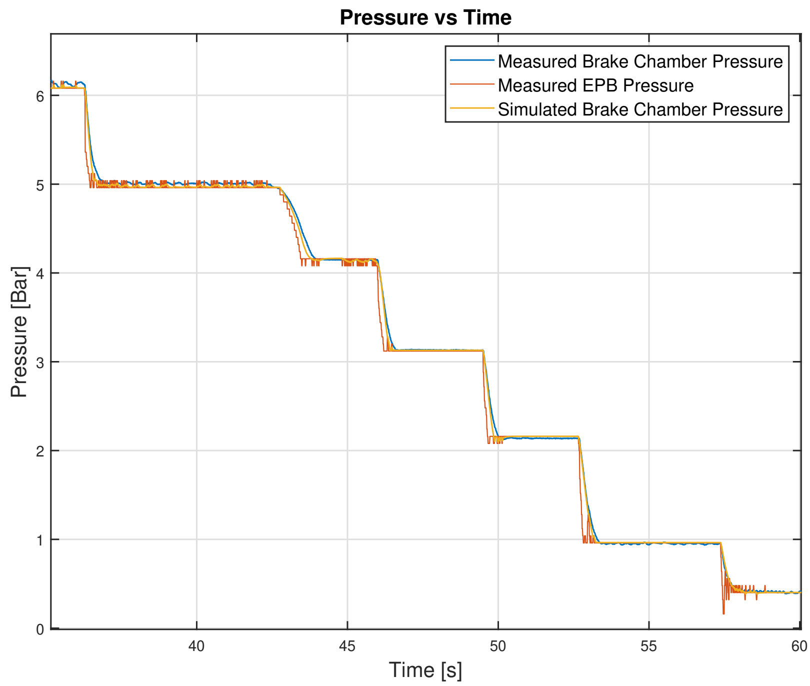

Figure 5.18: Simulated brake cylinder pressure response along with measured brake chamber pressure from 8.5 bar to 0.5 bar step response with extra 4 meter pipe length

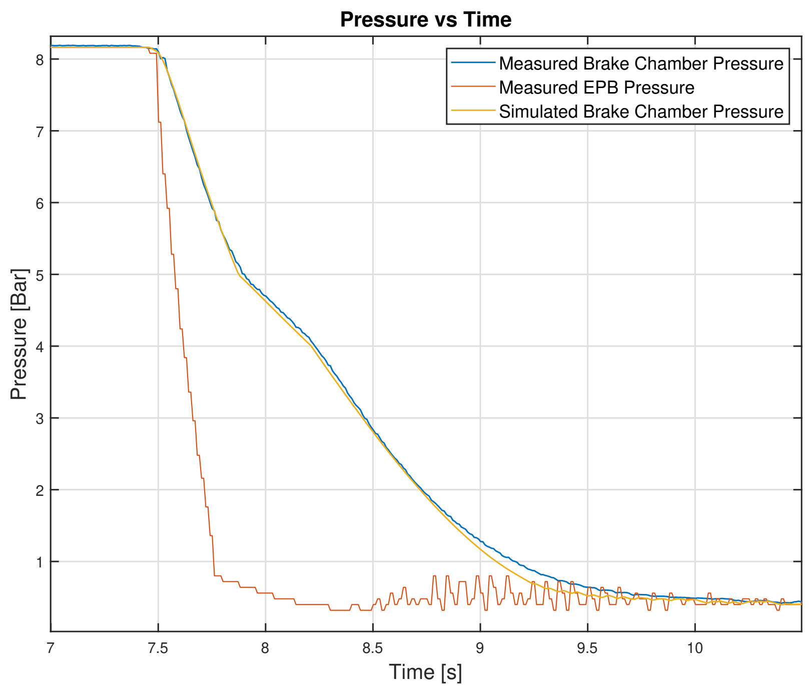

Figure 5.19: Simulated brake cylinder pressure response along with measured brake chamber pressure from 8.5 bar to 0.5 bar full drop response with extra 4 meter pipe length

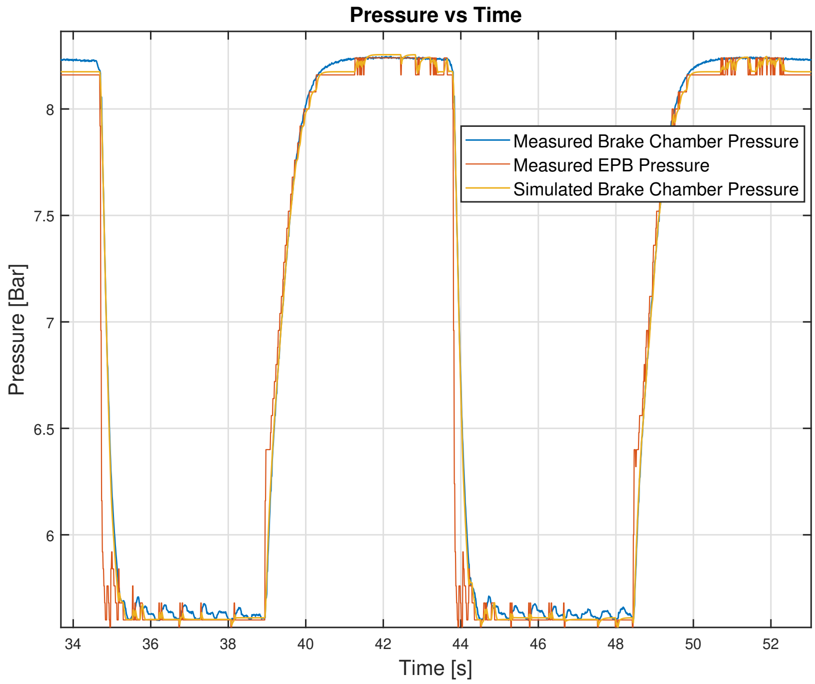

Figure 5.20: Simulated brake cylinder pressure response along with measured brake chamber pressure from 8.5 bar to 5.5 bar partial request response with extra 4 meter pipe length

The plots above clearly show that the model tends to estimate the pressure inside the brake chamber quite accurately with expected changes in physical parameters and with EPB pressure request. The simulated brake pressure is very close to the measured pressure.

5.3 Implementation Results

Following the satisfactory findings of the simulink model, the next step was to use this model as an estimator in a real vehicle. Vector tools were utilized to implement the model in the truck's control system.Input variable to the estimator was pressure from the EPB module. Below is the truck which was used to test the functionality of the developed model in a real time environment with the help of Canalyzer and CANape.

Figure 5.21: Scania 4x2 Truck used for realtime testing of the developed model

The results from the real time test are shown below. Figure 5.23 represents the pressure response in pipe and brake chamber along with measured pressure over time.

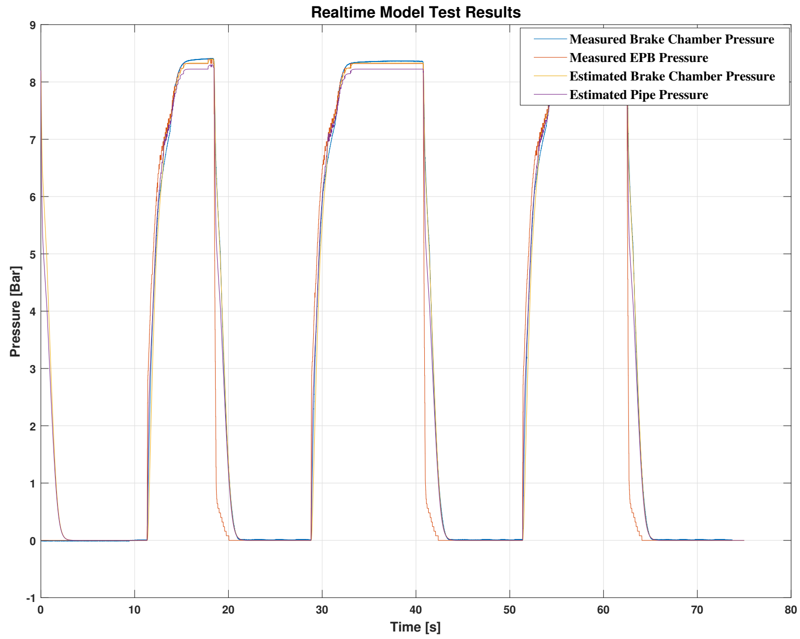

Figure 5.22: Real time estimation of brake chamber pressure response along with measured pressure

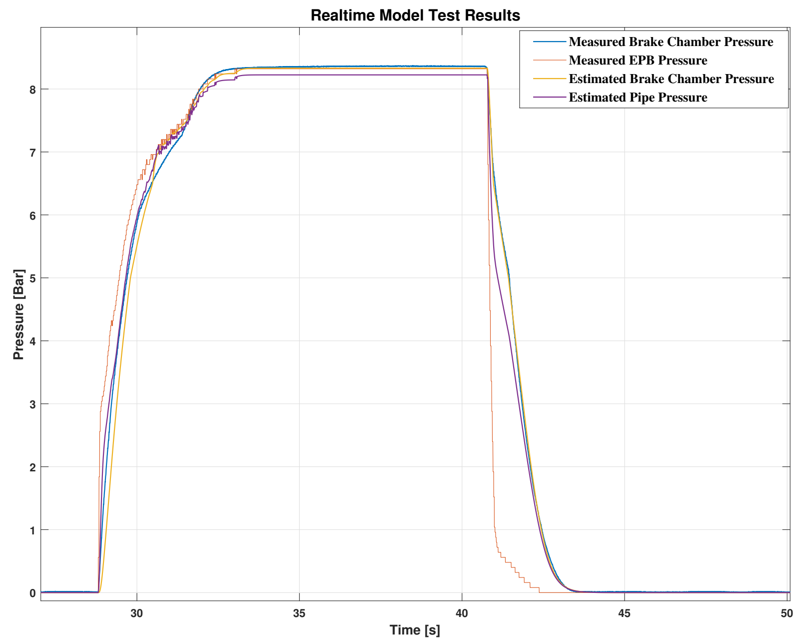

Figure 5.23: Real time simulation brake chamber pressure response along with measured pressure during complete cycle

Figure 5.24 illustrates the pressure response during the brake engaged portion from Figure 5.23. The figure indicates that the model can predict the pressure inside the brake chamber as closely as possible to the measured pressure.

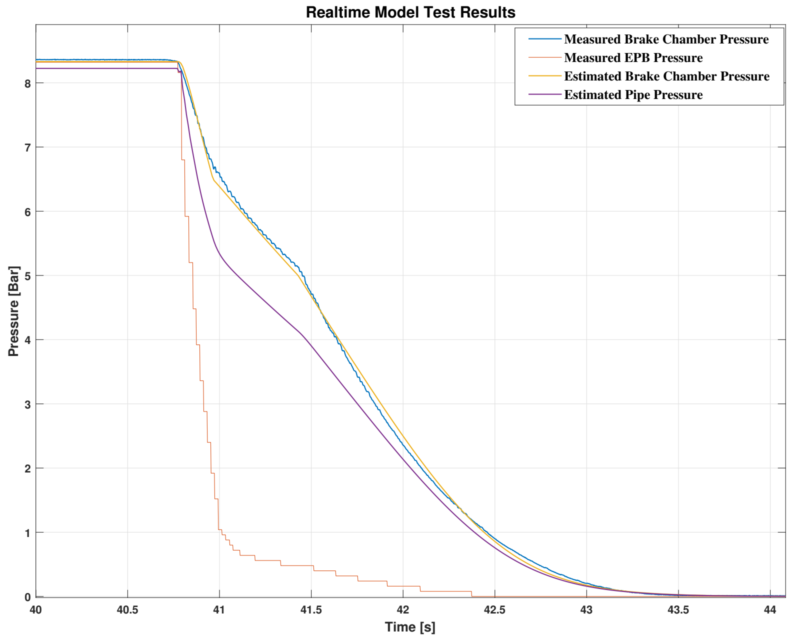

Figure 5.24: Pressure response when the parking brake is engaged

Figure 5.25 represents the pressure inside the brake chamber and pipe as well.

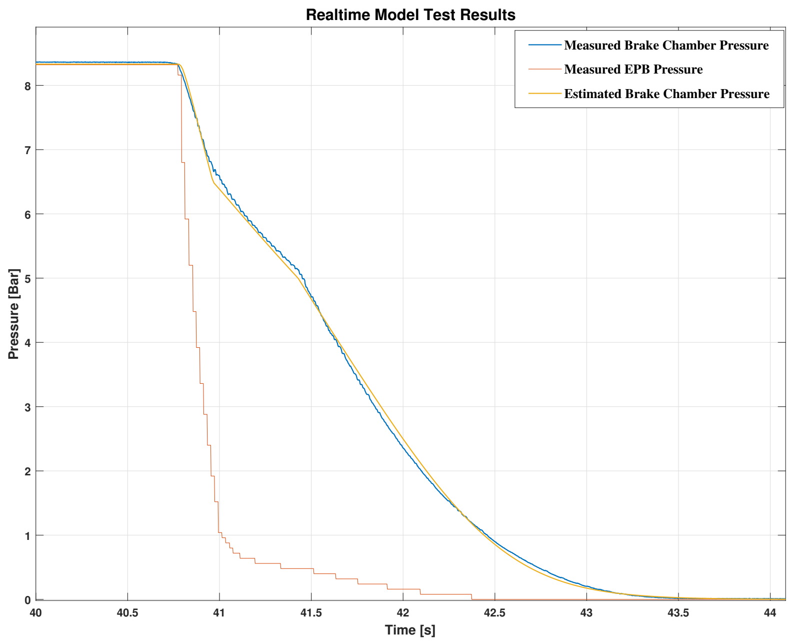

Figure 5.25: Pressure response inside brake chamber and pipe

Since the testing was done on two separate trucks, it was discovered that there is a minor difference for threshold pressure and contact pressure in both trucks. The figure 5.26 depicts the pressure response in both test trucks, Mjolner and Robben.

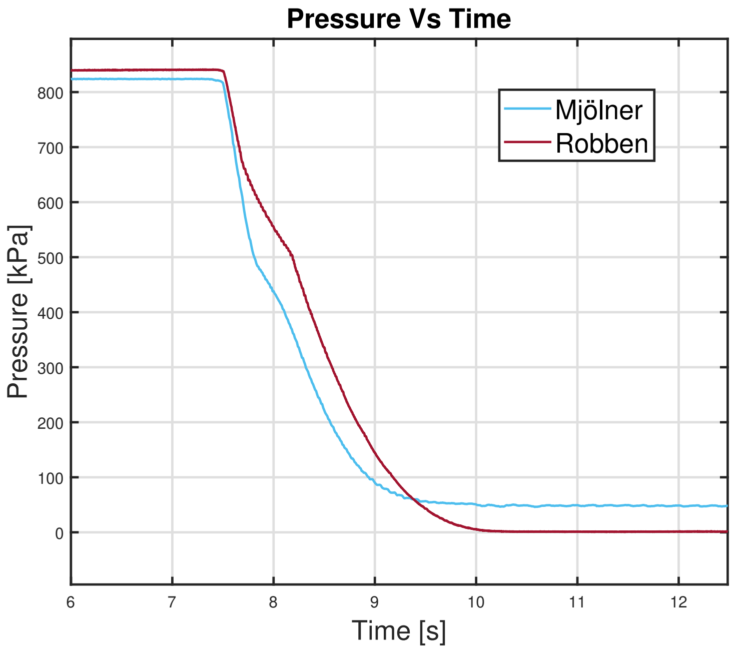

Figure 5.26: Comparision of pressure response for two seperate test trucks

The gradient of both plots changes at different pressure points, as shown in the figure above. Mjolner has a threshold pressure near 500 kPa and a contact pressure near 400 kPa, whereas Robben has a threshold pressure near 600 kPa and a contact pressure near 500 kPa. The cause for this disparity is that the pressure sensor in Mjolner was positioned on a pipe 1 metre distant from the brake chamber inlet nozzle, whereas the pressure sensor in Robben was mounted on the brake chamber nozzle. As a result, depending on the requirements, the model can be subjected to varying threshold pressures and contact pressures.

6

Conclusion

We began this research with two main goals: Investigating existing methods for estimating pressure inside brake cylinders, and how the proposed solution would manage to estimate the pressure in relevant vehicle operations.

This thesis presents a model-based approach for estimating pressure inside a parking brake chamber. This work proposes a mathematical model that takes into account all of the physical components involved in the parking brake circuit. The mathematical model described in this thesis serves as the framework for the development of a simulink model that uses pressure inputs from the Electronic Parking Brake Module to estimate the real-time pressure inside the parking brake cylinder.

By the end of the thesis we came to the following conclusions:

- The physical attributes were taken into account when developing subsystem models for each component in the layout parking brake circuit.

- Being a supplier component, the EPB was challenging to model on Simulink; as a result, the model took pressure inputs from the EPB for simulation and real-time testing.

- The plots in the results section shows that the developed model can estimate the pressure inside the brake chamber at various EPB demands quite accurately.

- To implement this model in different truck configuration, the physical parameters in the model has to be changed.

- Spring brake chamber comes in different sizes in different truck configuration, therefore the threshold pressure has to be determined before implementing this model on the truck.

- Due to the presence of a pre-loaded spring inside the brake chamber, it is essential to comprehend the forces acting at various pressure levels, which necessitates precise adjustment based on the spring's characteristics.

Future Work

This thesis work opens up an opportunity to carry out future work by improving the developed model in the following ways:

- Verification of developed model with different spring chamber sizes and different spring characteristics.
- Testing in different configurations and performing statistical analysis.
- Development of accurate force observer model in order to determine the change in volume.
- To get even more accuracy of the estimated pressure, estimating stroke length of the spring piston would be a good aid.
- Investigation into non-linear behaviour of the spring.
- Controller can be designed for the autonomous backup braking considering this model.
- Possibility of modelling in different tools such as Siemens AMESim, Ansys etc.

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A

Appendix

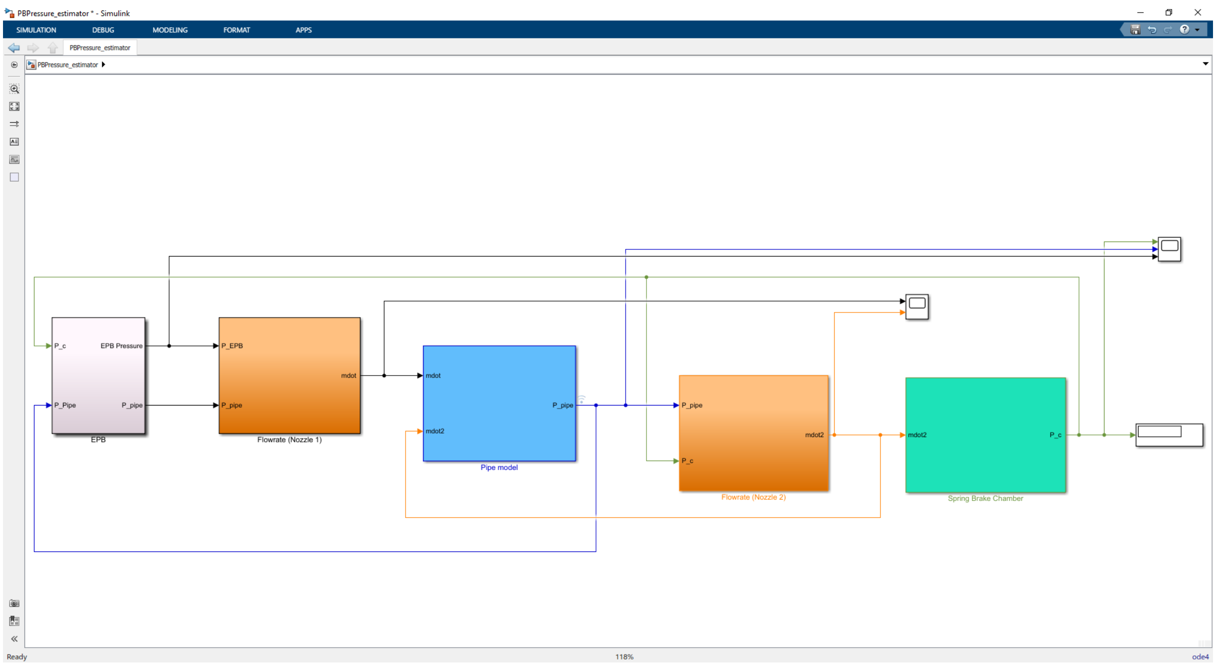

Figure A.1: Overview of simulink model

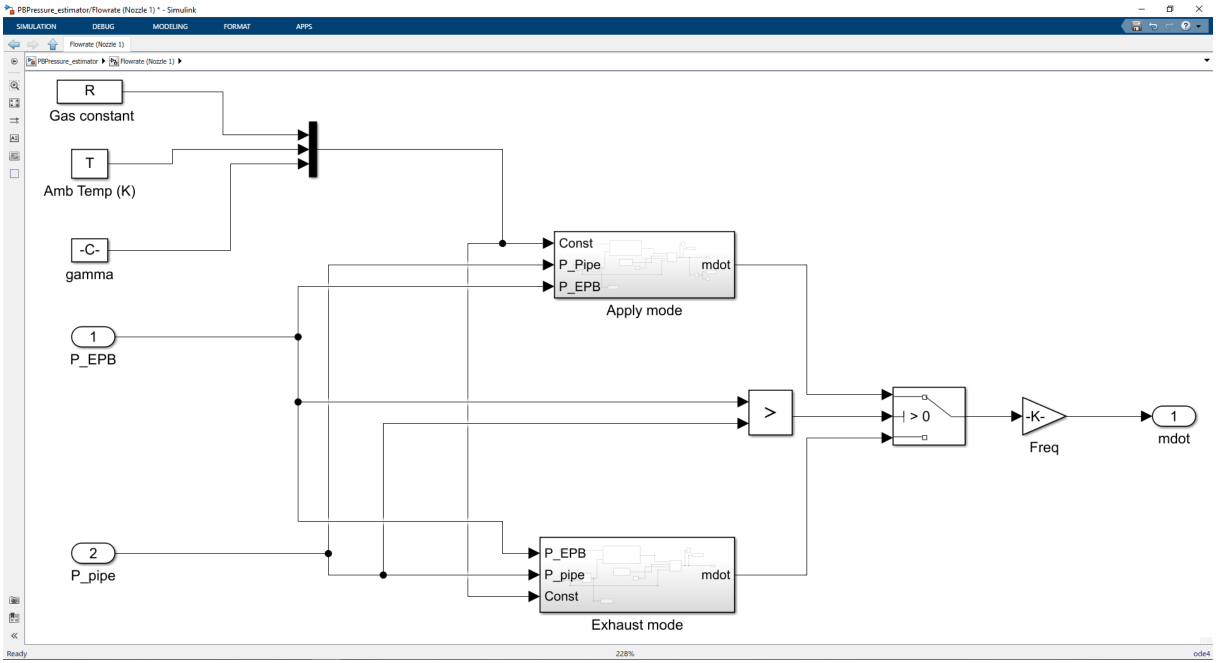

Figure A.2: Nozzle1 flowrate model

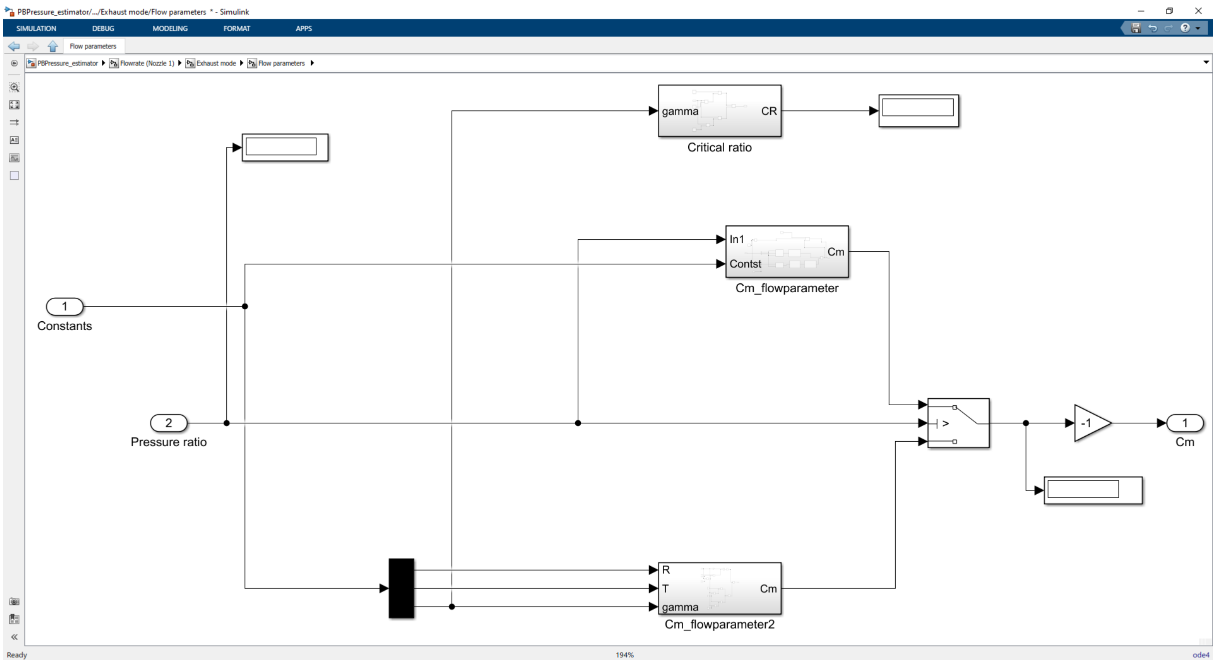

Figure A.3: Flowparameter in nozzle

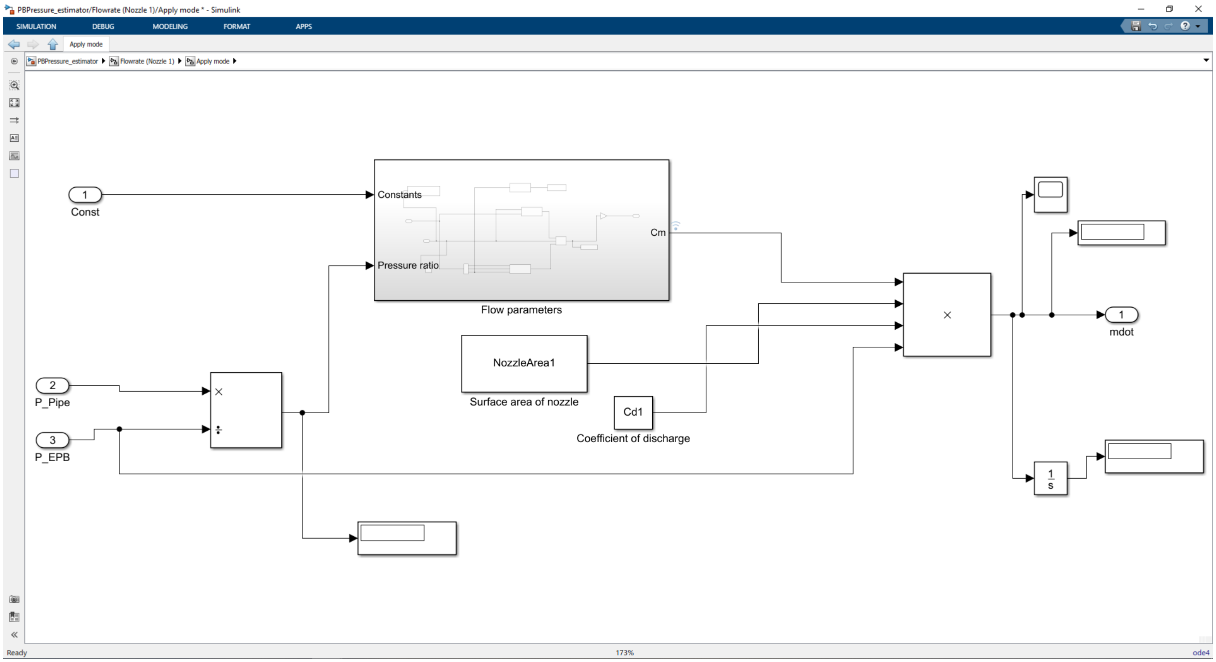

Figure A.4: Apply mode model

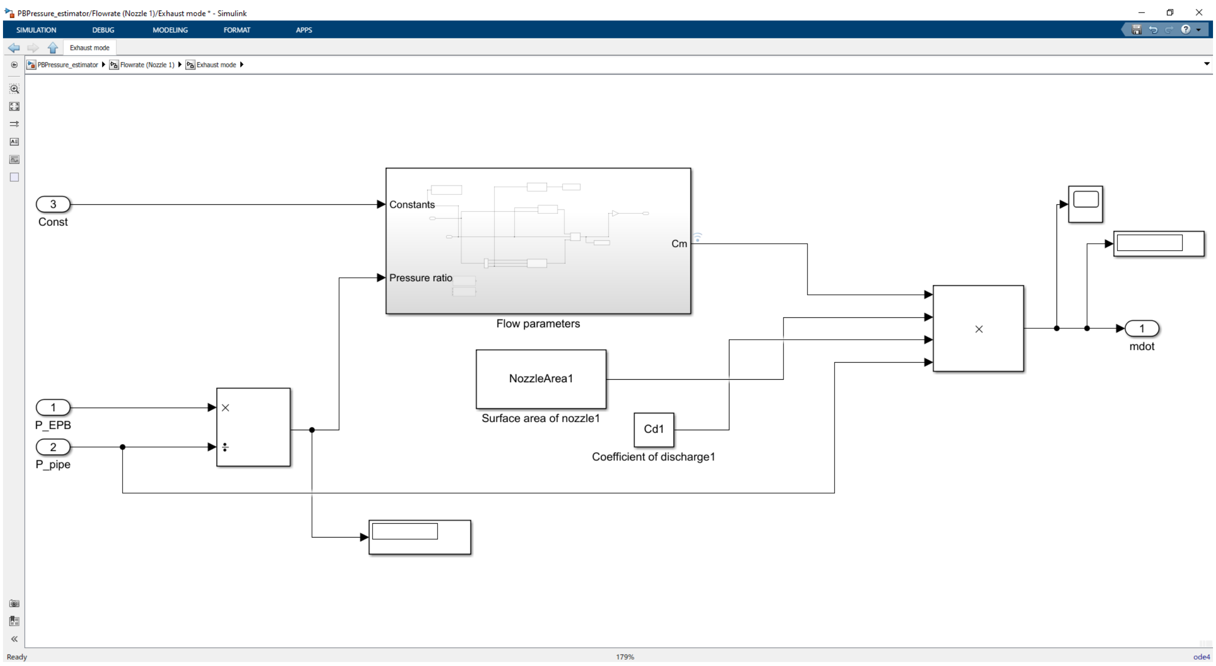

Figure A.5: Exhaust mode model

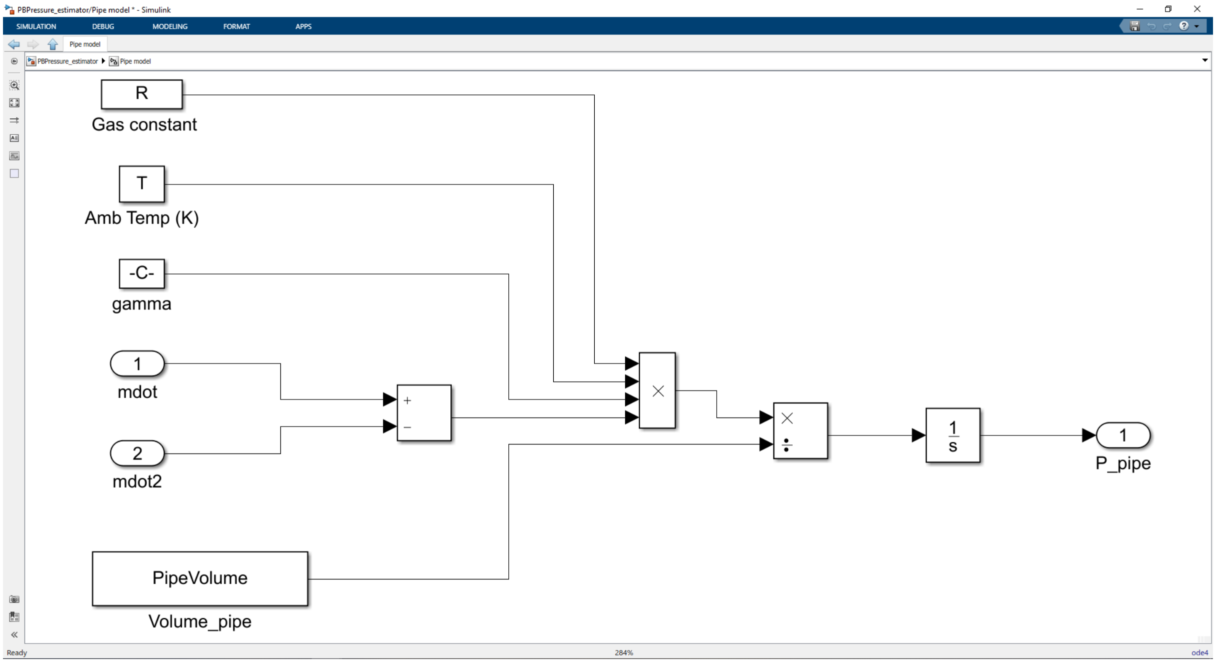

Figure A.6: Pipe model

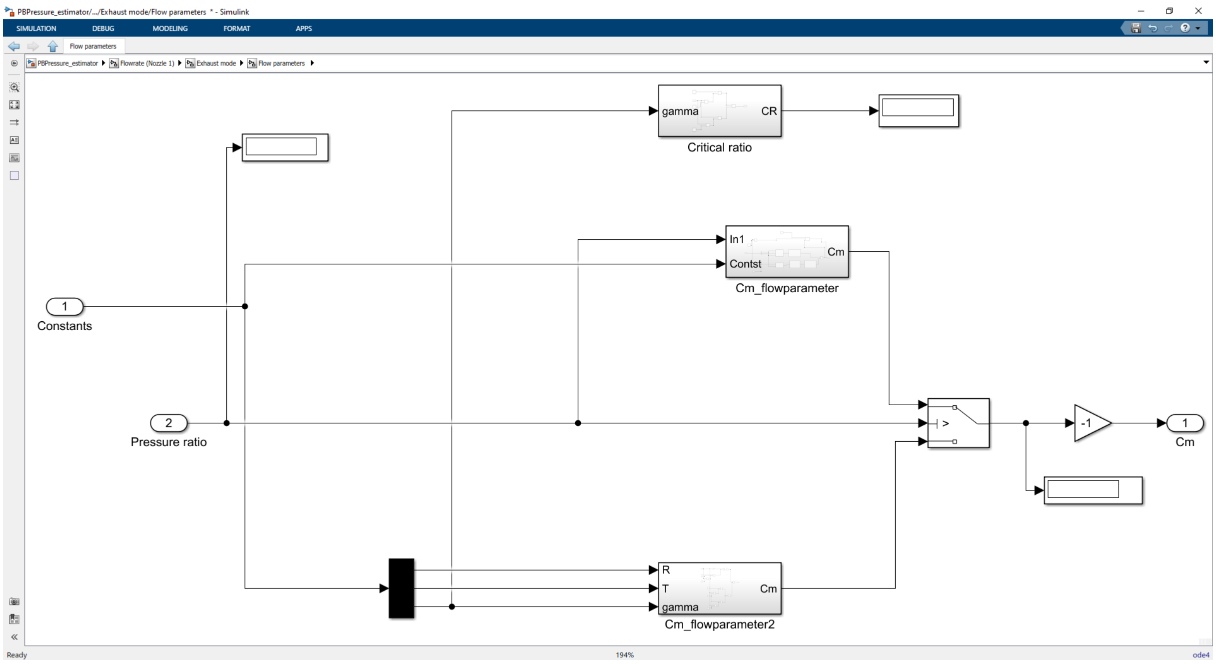

Figure A.7: Nozzle2 flowrate model

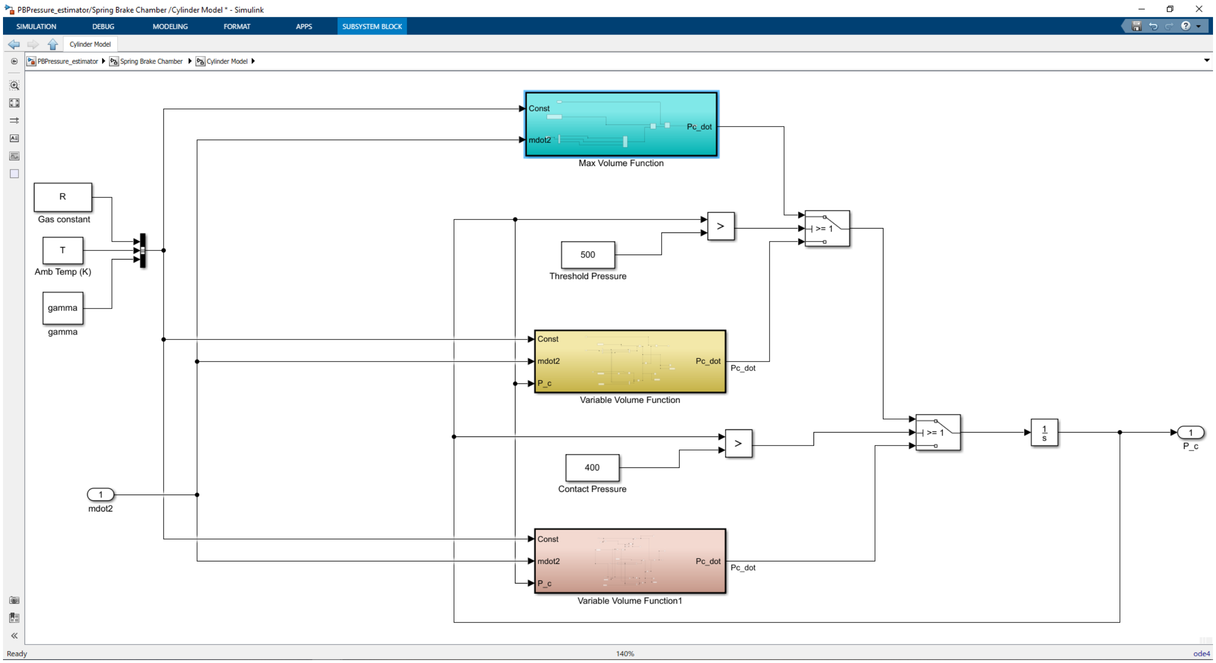

Figure A.8: Brake cylinder model

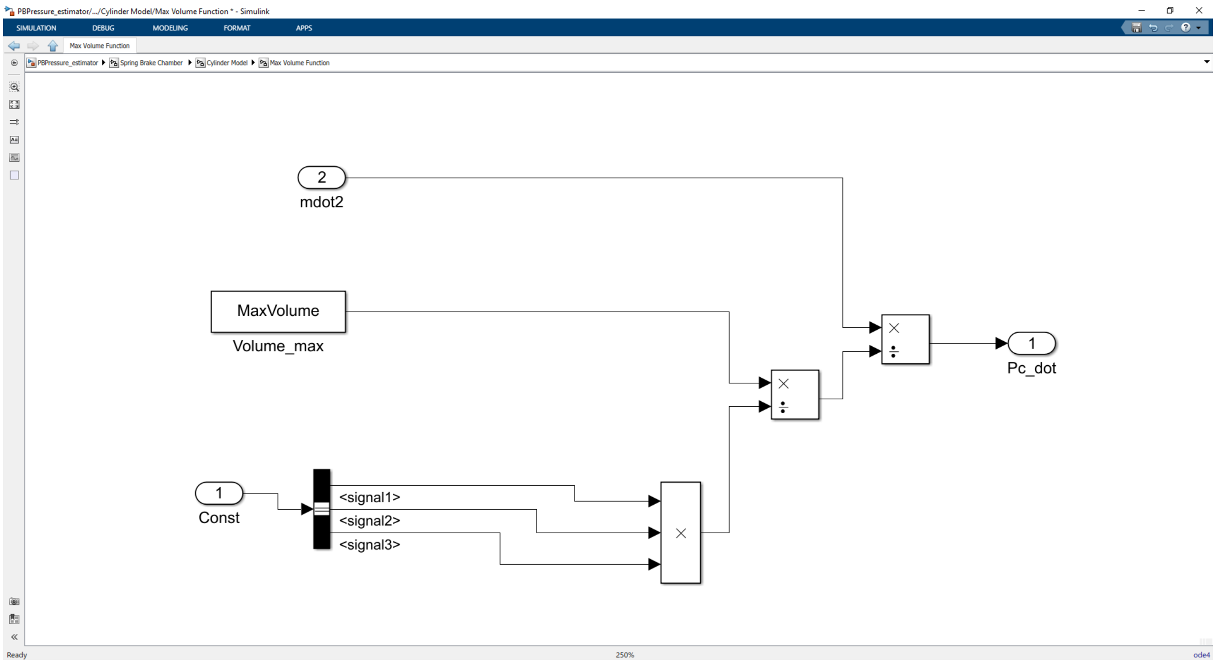

Figure A.9: Max volume function

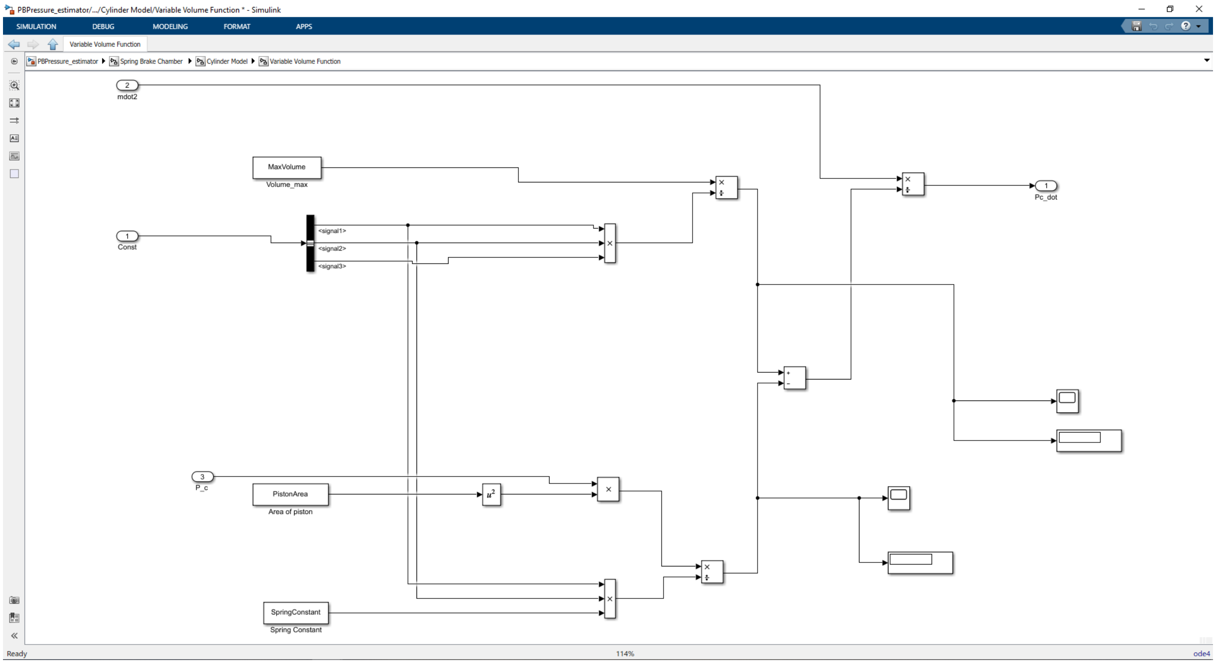

Figure A.10: Variable volume function

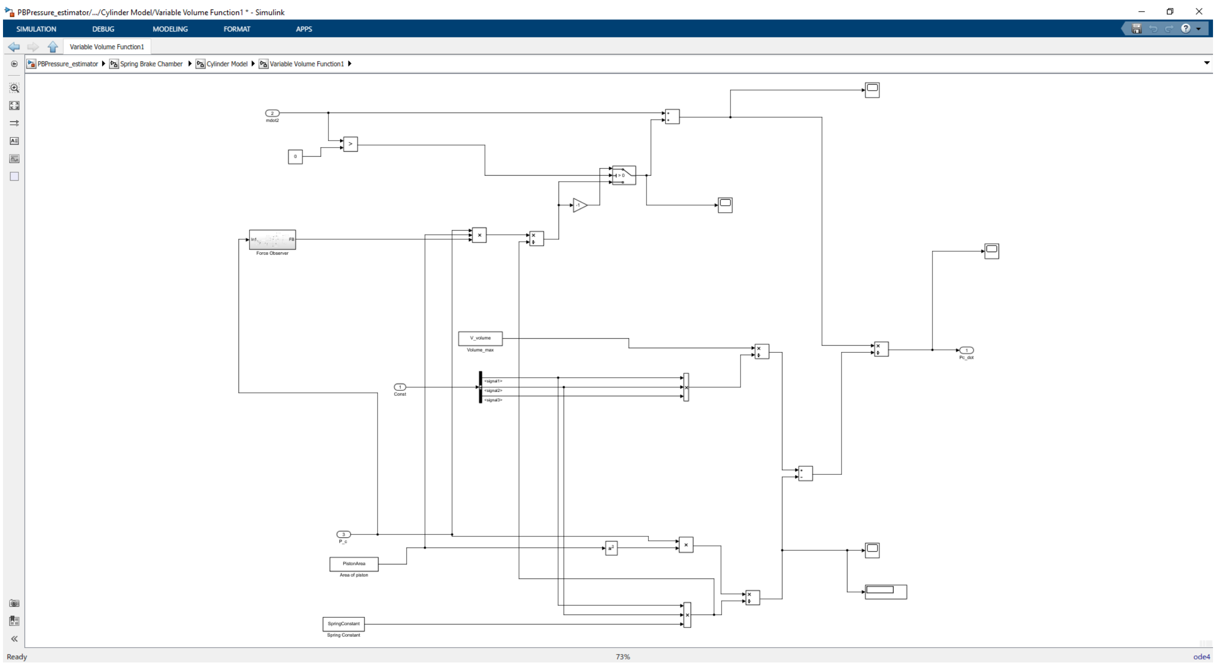

Figure A.11: Initial volume function

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