MODELLING THE PRESSURE TRANSIENTS AND PUSHROD EXTENSION OF A MULTI-CHAMBER PNEUMATIC BRAKING SYSTEM

A Thesis

by

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ABSTRACT

The pneumatic braking system for a Tractor-Truck is a crucial element for safe operation. Creating a model of the transient pressure in the brake chambers and the pushrod extension can enable a better understanding of how the braking system works and further someone's ability to discover a malfunction in a braking system before a catastrophic failure occurs. A study done by FMCSA in July 2007 stated that brake problems were an associated factor in 29% of crashes with large trucks [15]. Previous studies have been completed on individual pieces of the braking system at Texas A&M. The purpose of this study is to verify those mathematical models when being used to monitor several brake chambers simultaneously. The experiment that has been performed, tracks the pressure in six brake chambers arranged in the "typical" tractor configuration. The results show a model that does a relatively good job of predicting the steady state of the system. However, the prediction of the transient dynamics of the pressure suffers from much error. Further exploration on this topic may be needed if the current model is to continue being used for monitoring the pressure transients in the system.

DEDICATION

To my parents, Rex and Darla Foster. For supporting my endeavors academic and otherwise.

ACKNOWLEDGMENTS

I would like to thank the committee for their support throughout this process. Their input on this process has been invaluable. It has been a privilege to work with Dr. Swaroop Darbha on this project. His input and guidance was essential for completing this project on time and in the midst of a global pandemic. His weekly meetings kept me diligent and guided me to completing a project that improved the knowledge of this project and furthered the progression of the development of a useful tool for pneumatic braking systems.

The input of the Dr. Pagilla and Dr. Song was very helpful in increasing the quality of this paper and ensuring the study can continue in a productive way with the next person to carry on the work. Their feedback on where this paper was unclear helped me create a more well-refined final product.

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NOMENCLATURE

γ	Air Ratio
x_{pt}	Valve Open Point
Ab_{20}	Type-20 Effective Area
Ab_{30}	Type-30 Effective Area
R	Gas Constant of Air
A_{pv}	Area of Primary Valve Piston
A_{pp}	Area of Primary Piston
K_b	Return Spring Constant for One Chamber
F_1	Pre-loaded Force Primary Valve
r_{pv}	Radius of the Primary Valve
x_{bmax}	Max Pushrod Stroke
C_d	Expansion Factor
A_{pv1}	Area of Back of Primary Valve Piston
K_{ss}	Stem Spring Constant
K_2	Combination of Valve Spring Constants
r_{pp}	Radius of Primary Piston
V_{or}	Volume of Signal Port
V_{o120}	Initial Volume of a Type 20 Brake Chamber
T_o	Temperature of Air
R_{rp}	Radius of Relay Piston
F_{si}	Pre-Loaded Force of Valve Gasket
A_{1s}	Area of Secondary Valve Piston
A_1	Area of Relay Piston
A_1'	Area of Backside of the Relay Piston
A_{hole}	Area of Exhaust Hole in Relay Valve
K_s	Relay Gasket Spring Constant
F_{kbi}	Pre-loaded Force of Single Chamber Return Spring
P_{atm}	Atmospheric Pressure
P_t	Breakout Pressure
K_3	Exhaust Combined Spring Constant
F_2	Exhaust Combined Pre-Loaded Force
R_{xpt}	Relay Valve Open Point
M_{pp}	Mass of Relay Primary Piston

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1. INTRODUCTION

1.1 Background

Pneumatic braking systems are implemented in many large commercial vehicles. According to the Commercial Vehicle Safety Alliance (CVSA), "out-of-adjustment brakes and brake-system violations combine to represent half of all out-of-service violations issued for commercial motor vehicles on the road" [6]. Collisions with commercial motor vehicles can be catastrophic due to their large weight and size. Regular sized vehicles and their occupants can be crushed in even seemingly small collisions with a large commercial vehicle. This is why brakes are one of the most important things to understand in these large vehicles. With the amount of mass carried in these vehicles, the stopping power of the brakes is paramount to roadway safety. That is why this study along with others seek to provide enough data to sufficiently create a model that could be implemented on semi-trucks to monitor the brakes for leaks and out of adjustment pushrods is necessary.

According to the IIHS, "loaded tractor-trailers take 20-40 percent farther than cars to stop" [13]. With the heavy weight carried by tractor-trailers these effects can be even more drastic when roads are wet or brakes are not properly maintained [13]. Many truck drivers drive beyond what is technically allowed and driver fatigue can be a contributing factor to some accidents with heavy vehicles. By creating a system to monitor and diagnose problems in heavy vehicle braking, at least one of these factors in many crashes can be prevented.

In January of 2021, the FMCSA posted an "Analysis of Variability in Heavy Truck Brake Systems" [11]. In this analysis it is shown that drum brakes are the brakes that have the longest stopping distance and higher variability than systems that contain disc brakes. This would indicate that the ability to monitor the condition of drum brakes is desirable due to the higher variability of drum brakes and longer stopping distances that make them less safe for road use. It is important to note that the use of drum brakes continues because the upfront cost of disc brakes is high enough to deter many trucking companies from implementing disc brakes. The disc brakes would also add weight to the vehicle which can affect the overall performance of the vehicle [12].

According to the Federal Motor Carrier Safety Administration in 29% of crashes with large trucks, braking problems were an associated cause in the crash. This made braking problems the most common associated cause in a crash [15]. The ability to diagnose issues with the braking system can greatly improve safety on the road around large commercial vehicles. The CVSA began a campaign called "Operation Airbrake" [6]. The campaign was originally started in Canada in 1998 with the goal of reducing the number of crashes caused by faulty braking systems. This is carried out by conducting more inspections and educating drivers and mechanics about all of the ways brakes can show early signs of failure [6][10].

Previous studies performed at Texas A&M University have sought to understand and model individual components of the pneumatic braking system. These models have typically been corroborated using partial setups of the braking system [1][2][3][4][5]. The corroborated models include the treadle valve, the relay valve, and their effect on the pressure transients and pushrod stroke of the braking system. The approach used with these models was a lumped-parameter model. Lumped-parameter models have many underlying assumptions that cause some difficulty in showing the true dynamics of a system. However, these models were proven to be useful in predicting the steady state values of the system and have been corroborated in the past for showing the dynamics of a smaller scale system.

1.2 Outline of Thesis

This paper seeks to combine the previously researched models for the prediction of the pressure transients and pushrod extension in a multi-chamber pneumatic system. A typical tractor braking system will be explained for comparison to the system being used in the lab. Each component's function will be explained before the models are presented. Then, the models that have been previously developed will be shown and explained. This is where assumptions used to create these models will also be listed. Any adaptations made to the models for chaining them together will also be explained. Finally, the data gathered to corroborate the model will be shown. The data is expected to slightly differ from the dynamics of the system. However, the steady state estimations are expected to be relatively accurate.

2. A TYPICAL AIR BRAKE SYSTEM

2.1 Overview

The goal of this study is to expand upon the mathematical models previously created by Subramanian and Natarajan to verify their usability when applied to a full braking system [1][2][3]. This makes it important to define what a normal air braking system looks like and what the experimental setup intends to replicate. Figure 2.1 below shows a schematic of a typical air brake system for the tractor portion of the semi truck [7]. This is the part of the truck we intend to replicate as the trailer brakes are normally an isolated system that function similar to the rear section of the tractor.

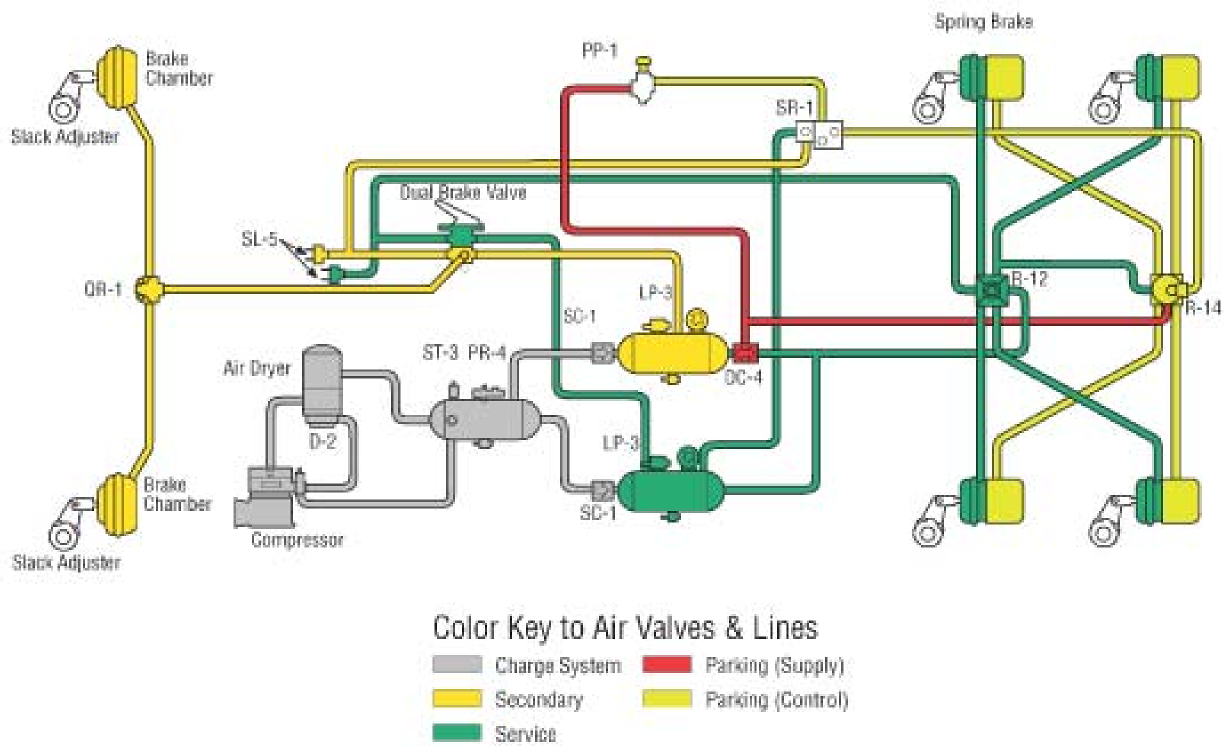

Figure 2.1: Typical Braking System [Reprinted from [7]]

The piece of this system being studied is the service and secondary brakes from the figure. Parking brakes are being ignored because they are essentially an on/off switch. The system consists of the treadle valve (dual brake valve in the figure), two front braking chambers connected to a quick exhaust valve, a relay valve, and four rear braking chambers connected to that relay valve.

2.2 Treadle Valve Primary Circuit

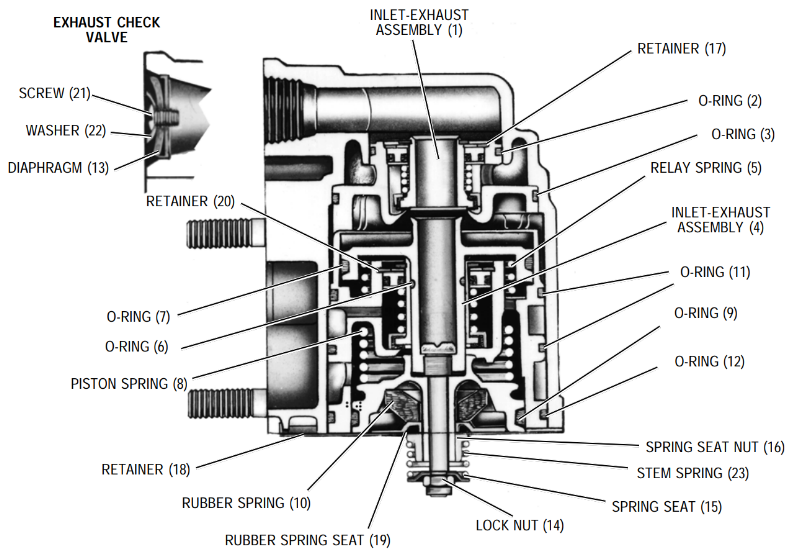

Figure 2.2: Treadle Valve Schematic [Reprinted from [8]]

The treadle valve's (Fig. 2.2) primary circuit controls the input to the relay valve based on an input from the driver through the brake pedal. The brake pedal is connected to the plunger on the treadle valve. Based on how far the pedal is depressed, the flow of air through the primary circuit of the treadle valve is changed. As pressure builds in the primary circuit the treadle valve will close off the flow of air by forcing the primary piston inside the valve back towards its original position. More information on how the treadle valve works including the equations to model it will be discussed later in this paper.

2.3 Relay Valve

As the treadle valve's primary circuit builds pressure, this acts as the signal pressure to the relay valve (Fig. 2.3). As the relay valve receives this signal pressure, the piston inside the relay valve

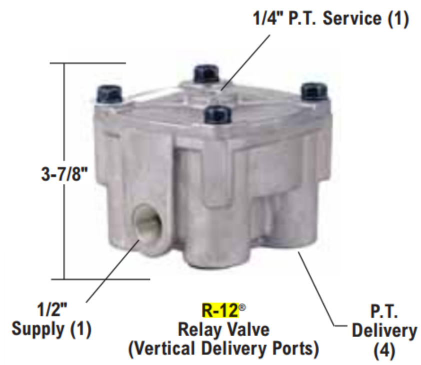

Figure 2.3: Relay Valve Ports [Reprinted from [9]]

opens the flow of air from the reservoir to the service brakes. The purpose of the relay valve is to reduce lag time in the braking system. If there was no relay valve and the lag was not compensated for in the system, the stopping distance of the truck would be increased, and the front brakes may wear faster due to the increased work required from them since the rear brakes would be late to be applied [14]. By only using a signal pressure the volume of air required to send through the treadle valve is significantly reduced. Then the high volume of air needed to fill the chambers can go through the relay valve and directly to the individual chambers.

2.4 Treadle Valve Secondary Circuit

The front brake chambers are controlled in a different manner. The secondary circuit of the treadle valve is used for the control of the front brake chambers. The secondary circuit of the treadle valve is essentially a relay valve attached to the primary circuit. As pressure builds in the primary circuit it pushes the piston in the secondary circuit back to open the flow for the secondary reservoir. The flow through this part of the valve will flow to the quick release valve until the pressure builds to close the flow, similar to the primary circuit.

2.5 Quick Release Valve

Once the flow of air has reached the quick release valve (Fig. 2.4) it closes the valve to allow flow to the two attached brake chambers. These chambers are smaller and closer to the treadle valve, thus allowing the flow through the treadle valve to not introduce excess lag in the system.

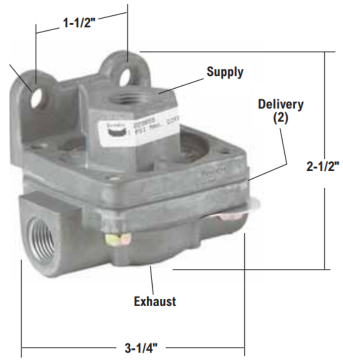

Figure 2.4: Quick Release Valve [Reprinted from [9]]

During the exhaust phase the quick release valve opens and increases the ability of the pneumatic system to exhaust air. This prevents lag in the releasing of the brakes so that when the driver takes their foot off of the brake pedal, the brakes do not stay on due to a bottleneck at the treadle valve. Without the quick release valve there would only be one exhaust port to to empty both treadle valve circuits. The quick release valve also serves the purpose of splitting the flow to the front brakes from one line to two lines.

Figures for each valve in this section were taken from associated manuals and catalogs for each of the parts [8][9].

3. EXPERIMENTAL SETUP

The goal of the experimental setup is to mimic the braking system detailed in the previous section. The system consists of just the valves and brake chambers of the braking system. This is because the addition of the slack adjuster and drum brakes adds in new variables that vary depending on the wear of the brakes. By modelling the simplest part of the system, we gain a better understanding of how the system should behave. When the rest of the braking system is attached the model can be trusted to be accurate and will require minimal calibration for application to a full braking system.

There are six total brake chambers in the experimental setup. Four (4) type 30 brake chambers represent the rear brake chambers (Fig. 3.1) and two (2) type 20 brake chambers represent the front brake chambers (Fig. 3.2). The brake chambers are all manufactured by Pro Trucking Parts. The brake "type" is a reference to the effective cross-sectional area of the chamber. Type 20 is for a 20 inch cross-sectional area and type 30 is a 30 inch cross-sectional area. This is an important variable to consider when modelling the system.

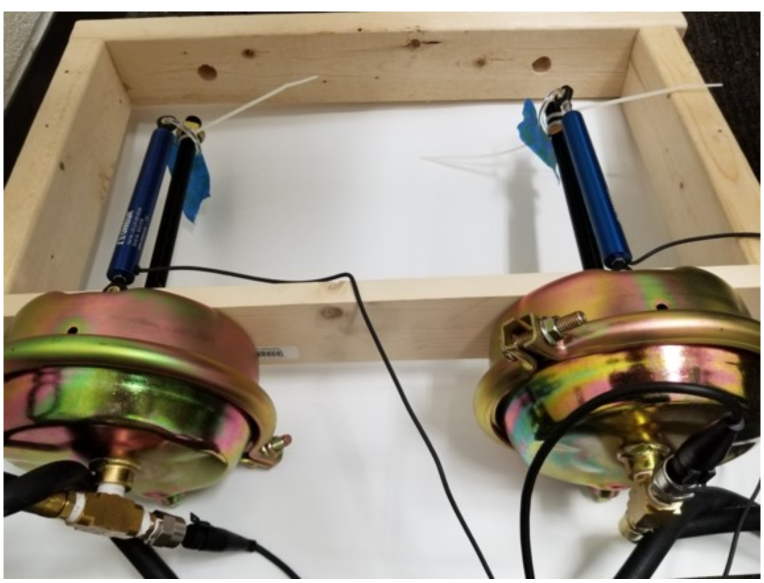

Figure 3.1: Wooden Frames and Rear Brake Setup

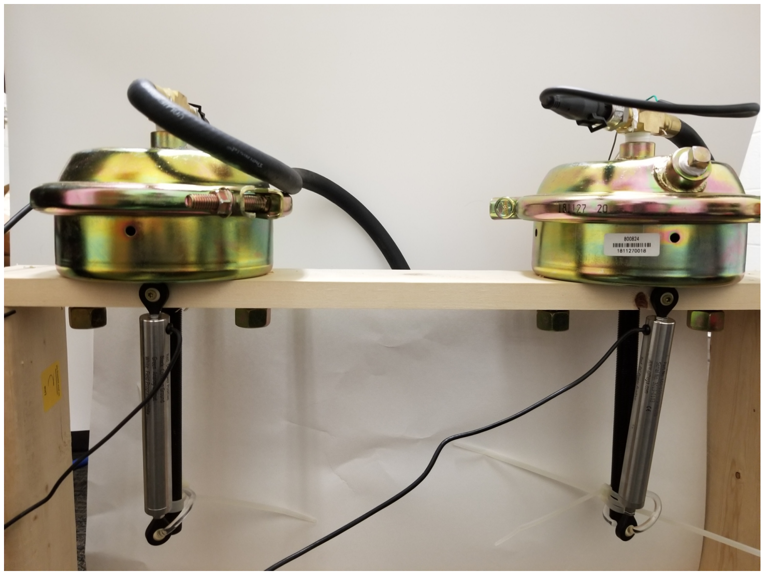

Figure 3.2: Front Brake Chambers

To keep the brake chambers off of the ground some simple wooden frames were built so that the chambers could be bolted to the frame. These frames are built using 2"x4" wood pieces and a 1"x4" piece to secure the brake chamber to the frame. The thinner cut of wood is necessary since the mounting bolts that come with the brake chambers themselves are not long enough to make it through the thickness of a 2"x4" piece of wood. This frame can also serve as a way of limiting the pushrod stroke. This effect could be achieved by adding the clevis to the pushrod. Depending on how far the clevis is screwed onto the pushrod, the limit of the pushrod extension changes. This is useful for further testing because the pushrod typically does not use the full extension. Over-extension of the pushrod is actually an indication of brake wear or a malfunction in the system.

There are two (2) air compressors/reservoirs included in the testing apparatus. There is a 15 gallon Campbell-Haussman compressor and a 20 gallon Husky air compressor. The 20 gallon air compressor is connected to the primary circuit with the rear brake chambers since it has the larger reservoir, and the rear portion of the system has more volume to fill. The front chambers are connected to the secondary circuit and the smaller air compressor. Both compressors are sufficiently large, so the pressure is assumed to remain constant in the reservoir during testing. A brass tee is necessary for the attachment of the compressor to the rear circuit. This is because in the system we are trying to mimic, the primary circuit, which provides the signal pressure to the relay valve, is supplied by the same reservoir that is connected to the supply port of the relay valve.

Each valve is connected as expected from the explanation of the typical braking system. The quick release valve (Fig. 3.3) is connected between the secondary circuit of the treadle valve (Fig. 3.4) and the front brake chambers. The relay valve (Fig. 3.5) is connected to the primary circuit of the treadle valve and the rear brake chambers. The treadle valve supply ports are used by each of the two air compressors.

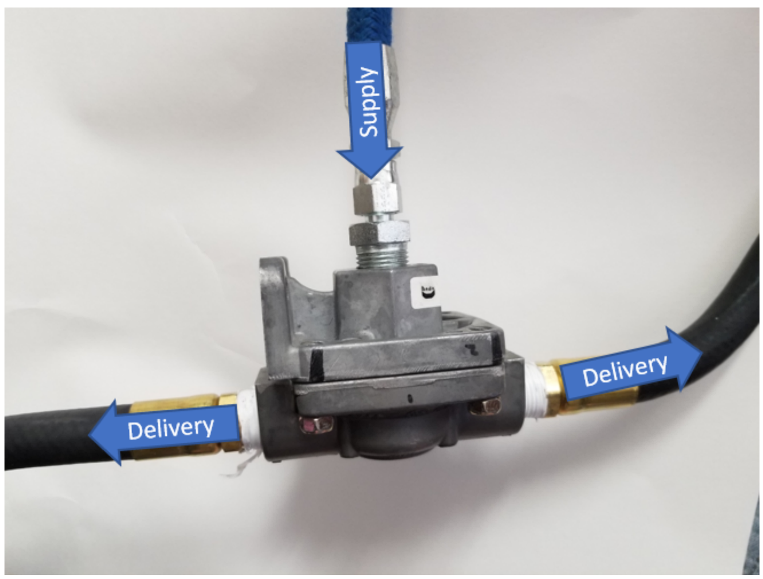

Figure 3.3: Quick Release Valve Connection

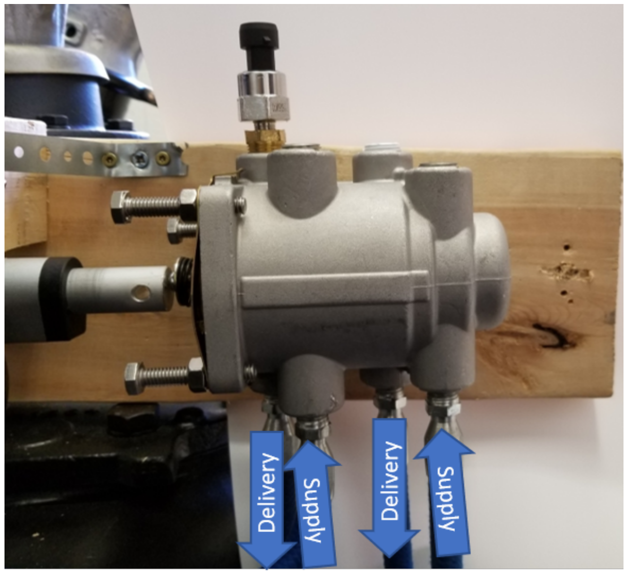

Figure 3.4: Treadle Valve

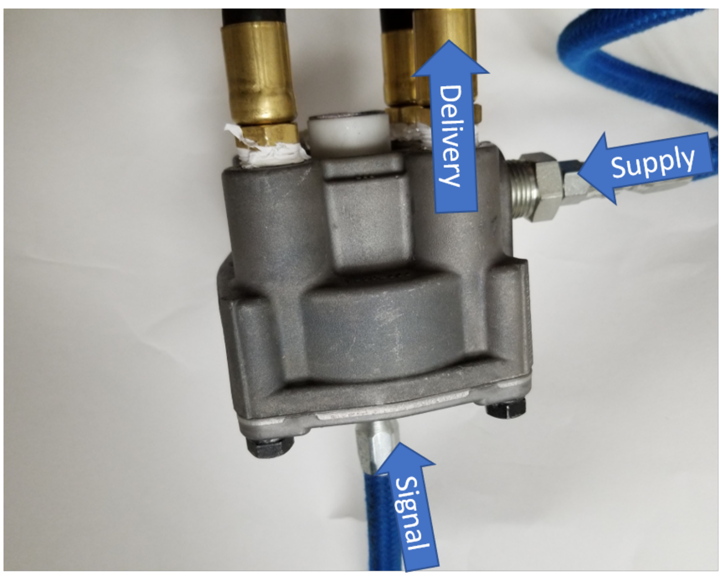

Figure 3.5: Relay Valve Connection

The sensor configuration is an important aspect of the system as well. It can be seen in the photos of the chambers that a brass tee was required to allow for the attachment of the air hose as well as the pressure sensor. The pressure gauge being located at the inlet/outlet allows for the monitoring of the pressure inside the individual chamber. The sensors for measuring the pushrod extension are pictured in the rear circuit photo. These sensors are fixed to the wooden frame using a wood screw and attached to the pushrod via a C-shaped plastic ring. The pressure sensors and the linear sensor inside the linear actuator are 5V sensors supplied with power by the NI-6211 DAQ and wired to share a common ground (Fig. 3.6). The linear displacement transducers that monitor the pushrod stroke are 12 V sensors that are supplied with power by an external power source (Fig. 3.7). The ground of this power source as well as the ground wire of all of the sensors are wired to a common ground on the DAQ. This should aid the accuracy of the measurement readings.

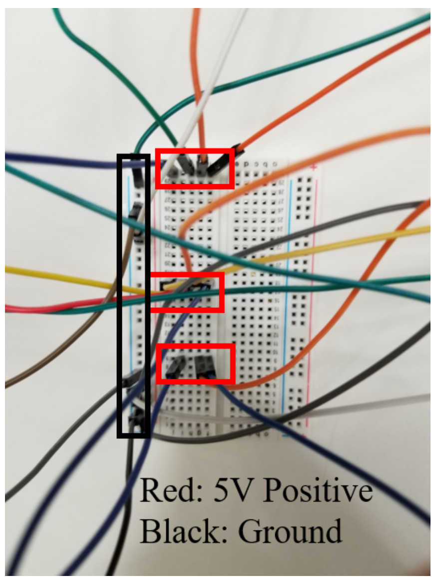

Figure 3.6: 5V Wiring Scheme

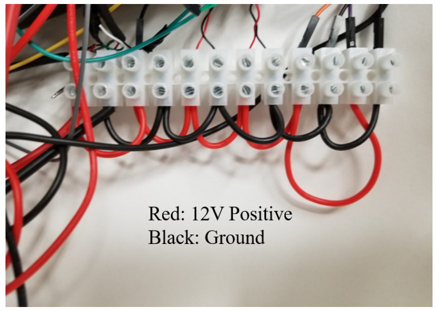

Figure 3.7: 12V Wiring Scheme

Table 3.1 shows a parts list that includes all of the model numbers for each component of the system.

Part	Brand	Model Number
E-7 Treadle Valve	Bendix	287411N
20 Gallon Air Compressor	Husky	C202H
15 Gallon Air Compressor	Campbell-Hausfeld	VX401100AJ
Quick Release Valve	Bendix	QR1
R-12 Relay Valve	Bendix	065104
Type 20 Brake Chamber	Pro Trucking Parts	800824
Type 30 Brake Chamber	Pro Trucking Parts	800826
Linear Potentiometer	Omega Engineering	LDI-119-100-A010A
Linear Potentiometer	Omega Engineering	LDI-119-75-A010A
Pressure Transducer	Autex	5823950559
12V Power Supply	Progressive Automations	LS200-12/L
Data Acquisition Board	National Instruments	NI-6211
Linear Actuator	Progressive Automations	PA-14P

Table 3.1: Parts List

4. THE MATHEMATICAL MODEL

4.1 Model Development

It is important to note that the model/equations as they are described in this section were developed and corroborated using a single chamber system by Subramanian and Natarajan [1][2][3]. These are brief descriptions of how each model works as it was developed. The conditions and new assumptions made for expanding the system to the new experimental setup are listed towards the end of the chapter. The implementation of the model in MATLAB Simulink is also discussed.

4.2 The Treadle Valve

The mathematical model of the treadle valve being used was developed by Subramanian in 2003 [1][2]. The important variable to be tracked is the position of the primary piston. For reference, a cross-sectional diagram of the treadle valve is included below (Fig. 4.1) [1].

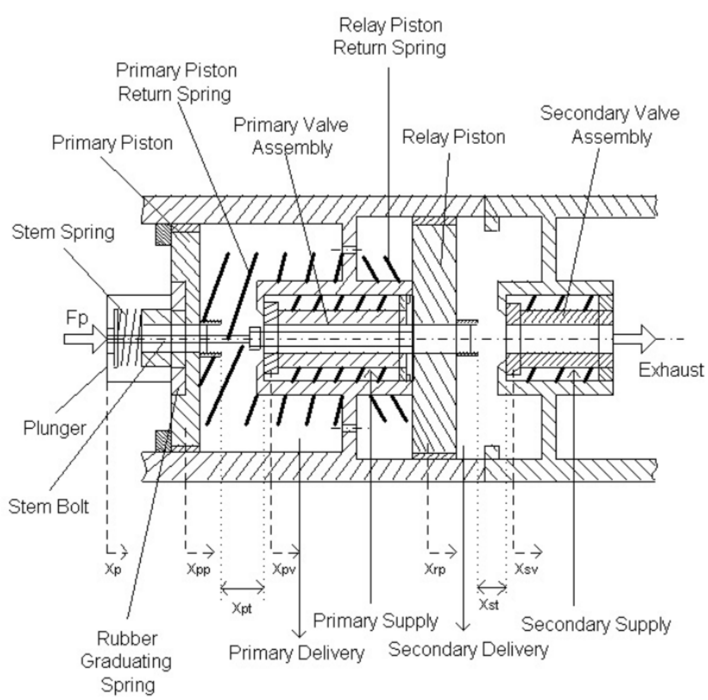

Figure 4.1: Cross-Sectional View of the Treadle Valve [Reprinted from [1]]

The position of the primary piston is what decides the opening area for the air to flow through for filling the signal port of the relay valve. The pressure in this section of the treadle valve, the primary circuit, also controls the secondary circuit of the treadle valve. The input to the system is the displacement of the primary plunger. The primary plunger is normally connected to the foot pedal in the truck. The primary plunger position is referred to as x_p. An important distinction for which equations to use when modelling the system is knowing what phase the system is in based on treadle valve position. The phases are the apply phase, where pressure is being added to the system, the hold phase, where the system is neither gaining, nor losing pressure, and the exhaust phase, where the system is losing pressure. The criterion used for determining what phase the system is in are as follows. For the apply phase:

$$x_{pp} > x_{pt} \tag{4.1}$$

For the hold phase:

$$x_{pp} = x_{pt} \tag{4.2}$$

For the exhaust phase:

$$x_{pp} < x_{pt} \tag{4.3}$$

where x_{pt} is the point at which the primary piston comes into contact with the primary valve piston. The equation for the position of the primary piston is as follows:

$$x_{pp} = \frac{K_{ss}x_p + F_{gs} + F_1 - P_{pd}(A_{pp} - A_{pv}) - P_{ps}A_{pv1} + P_{atm}A_{pp}}{K_2} \tag{4.4}$$

For the exhaust phase equation 4.5 is used.

$$x_{pp} = \frac{K_{ss}x_p + F_{gs} + F_2 - P_{pd}A_{pp} + P_{atm}A_{pp}}{K_3} \tag{4.5}$$

K_{ss} is the spring constant of the stem spring, F_1 is the pre-loaded force of the springs behind the primary piston, and F_{gs} is the force of the rubber graduating spring. The rubber graduating spring is a nonlinear spring with equations to follow in equation 4.6 [1][2]. P_{pd} is the pressure in the volume inside the treadle valve pressing back on the primary piston. This pressure is considered equal to the pressure in the signal port of the relay valve. P_{ps} is the supply pressure and multiplying it by A_{pv1} is its force on the back side of the opening for the air flow to the signal port.

$$F_{gs} = \begin{cases} m_1 x_{pd} + n_1 & \text{if } x_{pd} < l_1 \\ m_2 x_{pd} + n_2 & \text{if } l_1 \leq x_{pd} < l_2 \\ m_3 x_{pd} + n_3 & \text{if } l_2 \leq x_{pd} < l_3 \\ a_1 x_{pd}^2 + a_2 x_{pd} + a_3 & \text{if } l_3 \leq x_{pd} \end{cases} \tag{4.6}$$

Equation 4.6 lays out how the rubber graduating spring is modelled for the system. It's force equation changes based on how much compression there is between the plunger and the primary piston. x_{pd} is the difference between the plunger displacement and the primary piston displacement [1][2].

$$x_{pd} = x_p - x_{pp} \tag{4.7}$$

4.3 The Relay Valve

The equations of motion for the relay valve are different from those of the treadle valve. This model was developed by Natarajan [3]. The input to the relay valve is the pressure delivered by the primary circuit of the treadle valve. The signal pressure pushes on the primary piston of the relay valve. This opens the flow of air until the pressure inside the chambers pushes the primary valve and primary piston back to the hold position. When the treadle valve enters the exhaust phase, the signal pressure begins to drop and the primary piston opens the exhaust port. The exhaust equation is solved using a simple forces equation to solve for acceleration of the piston. This value is integrated twice to acquire the value for displacement [3]. This integration can be problematic due to the high force and low displacement in the system. Limits have been placed in the simulation to mitigate these problems, but further improvements to model the exhaust phase of the relay valve could warrant study. Determination of the current phase of the relay valve is determined by the position of the relay valve primary piston x_{Rpp}. For the apply phase:

$$x_{Rpp} > x_{Rpt} \tag{4.8}$$

For the hold phase:

$$x_{Rpp} = x_{Rpt} \tag{4.9}$$

For the exhaust phase:

$$x_{Rpp} < x_{Rpt} \tag{4.10}$$

Where x_{Rpt} is the position where the primary piston does not open the valve nor open the exhaust port. The position of the primary piston of the relay valve during the apply phase is modeled as follows:

$$x_{Rpv} = \frac{P_{app}A_1 - P_{pd}A_1' - P_{atm}A_{hole} - Fsi}{K_s} \tag{4.11}$$

Where x_{Rpv} is the displacement of the primary piston subtracted by x_{Rpt}.

For the Exhaust Phase:

$$\frac{d^2 x_{Rpp}}{dt^2} = \frac{P_{pd}A_1' - P_{app}A_1}{M_{pp}} \tag{4.12}$$

4.4 Transient Pressure Equations

The transient pressure equations calculate the rate of change of pressure at any time in the system. These values are then integrated to get the actual pressure in the system. A series of assumptions are made, and will be described later, to arrive at these equations. The most important observation to list upfront is that the valves are assumed to behave like nozzles.

These equations were developed by Subramanian and corroborated using a partial brake system setup [1][2]. These equations work for all phases of the system. The important distinction to make when using these equations for simulation purposes, is that when the system crosses into the exhaust phase, the supply pressure, P_o needs to be changed to P_{atm}. This is because the reservoirs are no longer providing air to the system. These equations are piece-wise because the volume inside the chamber is changing throughout the process of braking. Therefore, there is a period where the pressure cannot build as fast with an expanding volume. The equations for the transient pressure in the brake chambers are as follows:

$$f(P_b, A_p) = \begin{cases} \left(\frac{V_{o1} P_o^{\frac{\gamma-1}{\gamma}}}{\gamma R T_o P_b^{\frac{\gamma-1}{\gamma}}}\right) \dot{P}_b & \text{if } P_b < P_t \\ \\ \left(\frac{V_b P_o^{\frac{\gamma-1}{\gamma}}}{\gamma R T_o P_b^{\frac{\gamma-1}{\gamma}}} + \frac{P_b^{\frac{1}{\gamma}} A_b^2 P_o^{\frac{\gamma-1}{\gamma}}}{R T_o K_b}\right) \dot{P}_b & \text{if } P_b \geq P_t \text{ and } x_b < x_{bmax} \\ \\ \left(\frac{V_{o2} P_o^{\frac{\gamma-1}{\gamma}}}{\gamma R T_o P_b^{\frac{\gamma-1}{\gamma}}}\right) \dot{P}_b & \text{if } x_b = x_{bmax} \end{cases} \tag{4.13}$$

$$f(P_b, A_p) = \begin{cases} A_p C_d P_o \left(\left(\frac{2\gamma}{\gamma-1}\right) \frac{1}{RT_o} \left| \left[\left(\frac{P_b}{P_o}\right)^{\frac{2}{\gamma}} - \left(\frac{P_b}{P_o}\right)^{\frac{\gamma+1}{\gamma}} \right] \right| \right)^{\frac{1}{2}} sgn(P_o - P_b) \\ \text{if } \frac{P_b}{P_o} > \left(\frac{P_b}{P_o}\right)_{cr} \\ \\ A_p C_d P_o \left(\left(\frac{2\gamma}{\gamma+1}\right) \frac{1}{RT_o} \left(\frac{2}{\gamma+1}\right)^{\frac{2}{\gamma-1}} \right)^{\frac{1}{2}} sgn(P_o - P_b) \\ \text{if } \frac{P_b}{P_o} \leq \left(\frac{P_b}{P_o}\right)_{cr} \end{cases} \tag{4.14}$$

Where V_b is:

$$V_b = V_{o1} + A_b x_b \tag{4.15}$$

With x_b being the pushrod extension defined by:

$$x_b = \frac{(P_b - P_{atm})A_b - F_{kbi}}{K_b} \tag{4.16}$$

Setting these two sets of equations equal to each other and solving for $\dot{P}_b$ and integrating is the way the simulation handles solving the equations. It is important to note that these equations get slightly modified for the simulation since there are several different transient pressures being modeled. These equations are used in their entirety for the modelling of the brake chambers. However, for the modelling of the supply pressure only the first equation from equation 4.13 is used, as the volume change inside the relay valve is so small it is assumed to be negligible. A_p is where these equations use the models for each valve. A_p is the area of the opening that air is flowing through in each valve. The equation for A_p changes depending on which valve is being used since the radius of the opening changes. The equations for A_p are as follows:

For the Treadle Valve:

$$A_p = \begin{cases} 2\pi R_{pv}\big(x_{pp} - x_{pt}\big) & \text{Apply} \\ 2\pi R_{pv}\big(x_{pt} - x_{pp}\big) & \text{Exhaust} \end{cases} \tag{4.17}$$

For the Relay Valve:

$$A_p = \begin{cases} 2\pi R_{xpv}\big(x_{Rpp} - x_{Rpt}\big) & \text{Apply} \\ 2\pi R_{xpv}\big(x_{Rpt} - x_{Rpp}\big) & \text{Exhaust} \end{cases} \tag{4.18}$$

4.5 Implementing the Model

Important notes for the implementation of these equations into the model include how to handle the possibility of negative values. At the start of the simulation, the system is considered in the apply phase. However, the primary piston would be less than the the value of x_{pt}. If there is not a condition for A_p to be zero until reaching the crossover point, the system may never leave the "exhaust" phase, even though the system never built any pressure. A simple "if" statement is needed to correct this problem.

For reference some screenshots are included below to show how the simulation has to cycle back on itself for each time step. This simulation was built in MATLAB's Simulink and each large gray box is a subroutine for calculating the necessary variable to move to the next step.

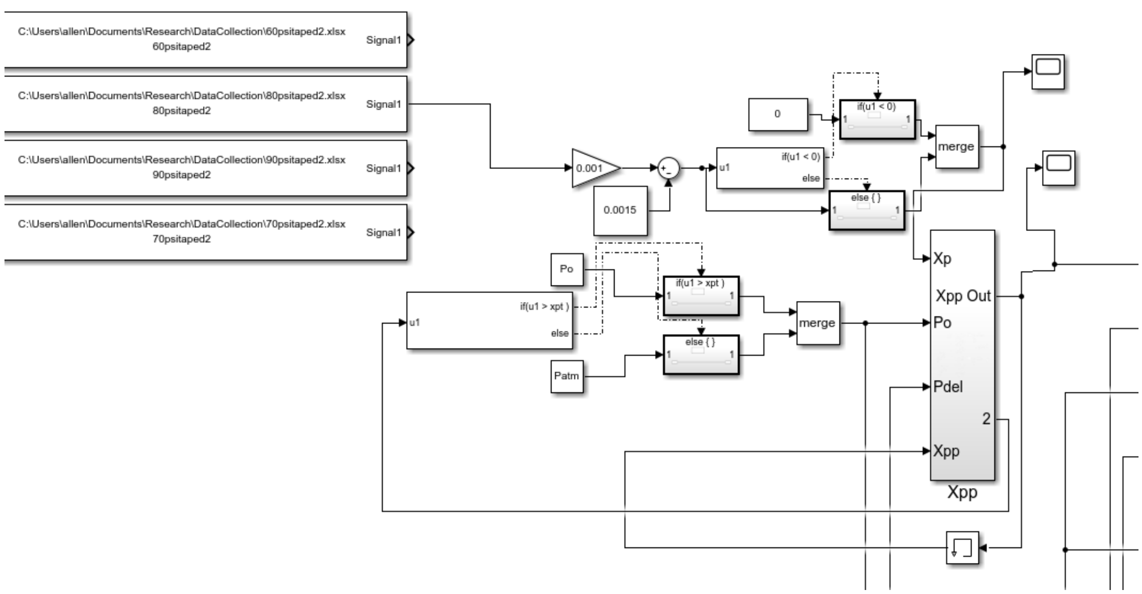

Figure 4.2: Input Modifications and Pressure Selection

In figure 4.2 it can be seen by the boxes on the left, that the input to the system is an excel sheet of the sensor data for the plunger displacement. The excel sheet has the voltage data converted to values in mm. It can then be seen that the simulation converts this to meters and subtracts a small amount from the data for calibration purposes. This calibrated data flows into the first subroutine which calculates the primary piston location using equation 4.4 and 4.5 depending on the phase of the system. Phase is determined by the location of the primary piston from the previous time step. It can also be seen that the phase changes the supply pressure through the if statement connected to P_o.

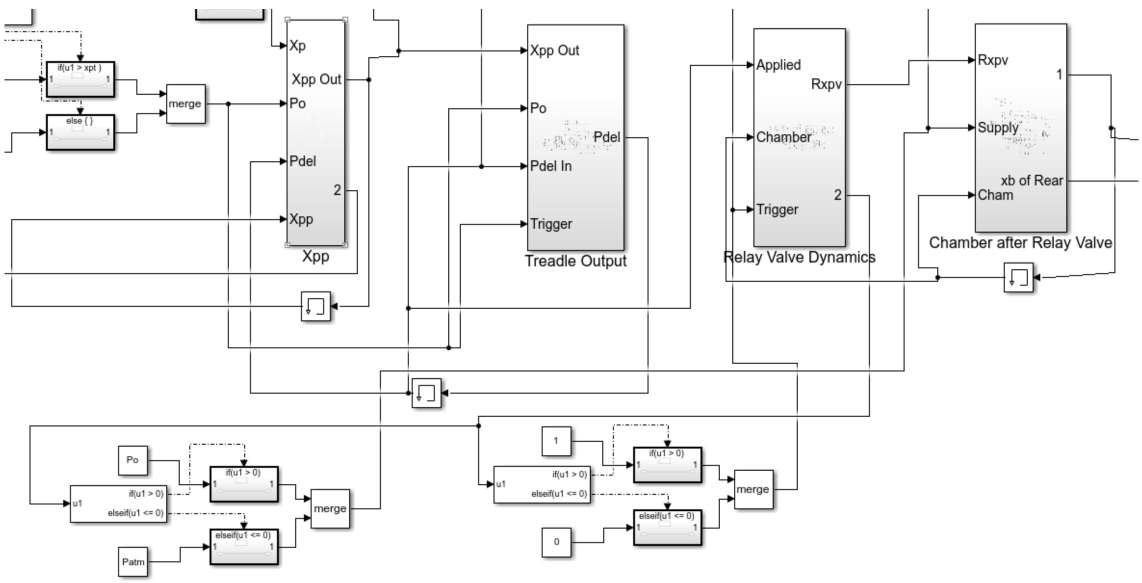

Figure 4.3: Rear Chamber Code with Switches

Looking at figure 4.3 the subroutines necessary to calculate the rear chamber estimations are portrayed. The calculation of x_{pp} is fed into the subroutine that calculates the pressure in the signal port of the relay valve. As previously mentioned, this calculation is different from the calculation for the chamber pressure as it only uses one stage. This result is then fed into the subroutine that calculates the primary piston position in the relay valve. This then feeds into the multi-stage calculation for the brake chamber pressure. Below the subroutines, switches for determining the supply pressure and a trigger are being used. The trigger resets the discrete integrator blocks inside the relay valve subroutine. If this reset did not occur the position of the relay valve primary piston could escape to $\inf$ as time increases. It is important to note that there may sometimes be an issue with the model not choosing the phase properly. This issue can be solved using a clock in Simulink and using time as a criteria for when each phase will be used. This does not change how the model uses the input data, rather it just makes sure that the model does not switch back and forth between phases due to noise in the data.

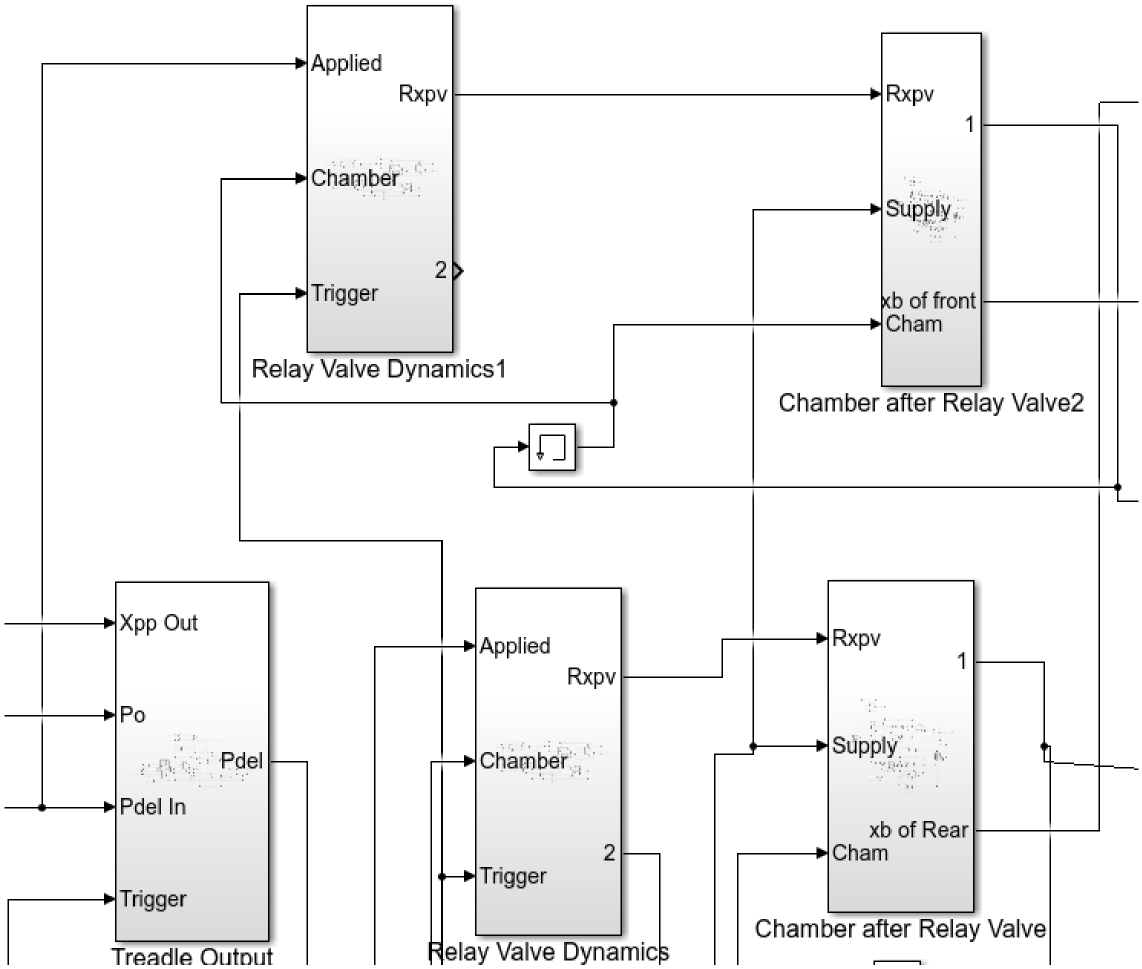

Figure 4.4: Front Chambers

Figure 4.4 shows the front brake chamber code running side-by-side with the rear chamber code. This uses the relay piston code to simulate the secondary circuit of the treadle valve. The area of the primary piston has been changed to represent the secondary piston area. All other values are assumed to be similar to the R-12 Relay Valve and are kept the same. All final outputs for pressure are fed to the scope that also takes input from the sensor data for comparison to the experimentally gathered values.

4.6 Assumptions Needed for the Model

Several assumptions have to be made in order for the model described to be accurate. Below is a list of each assumption that was made to arrive at this model [1][2].

1. A lumped parameter approach is acceptable when modelling the mechanical and air flow of the system. This has been corroborated previously.

2. Air is assumed to behave like an ideal gas. Thus, viscosity and frictional effects are not considered in the hoses. This has been corroborated previously.

3. The valve's expansion process is isentropic. The energy losses are accounted for using a discharge coefficient. This has been corroborated previously.

4. Air flow is considered to be adiabatic. This has been corroborated previously.

5. The reservoirs are considered sufficiently large as to not experience a pressure drop during the apply phase. This is corroborated by looking at the pressure in the reservoirs after the apply phase has taken place to make sure that the pressure before the regulator has not dropped below the desired value.

6. All of the chambers connected to the same valve will have the same pressure response in a leak-free system. This is corroborated by a plot of the pressure and pushrod stroke from all of the chambers connected to a relay valve.

7. Frictional losses in each hose are neglected. This was corroborated previously.

It is important to note regarding assumption 5, that the pressure in the reservoir does experience a slight drop. However, the reservoir is charged to a higher pressure than what is being used and flow is controlled by a regulator. The pressure behind the regulator does not drop below the desired pressure during the experiment so it is assumed that the flow through the regulator does not change. The important piece to remember when looking at this model is that, due to the assumption that a lumped-parameter approach is being used, the dynamics of the model are likely to slightly differ from the dynamics of the real-world system. However, this type of approach is supposed to be good for estimating the steady state value of the system and providing a rough estimation of the dynamics. Whether or not this holds true for the full brake chamber setup will be discussed in the next chapter, which describes the corroboration of the model

4.7 Modifications to the Model

For the purposes of this study there are some modifications made to each of the models in order to account for the chaining together of the models and adjust for the sensitivity of the system. The first change is to account for the amount of chambers attached to each relay valve. Since it is assumed that all chambers connected to the same valve have the same response, all chambers connected to one valve are modelled as a single larger chamber. To do this, the volume of the brake chamber is multiplied by the number of chambers attached to the valve. This makes the four chambers attached to the relay valve act as one chamber that is four times as large. The spring constant of the return spring in the chamber is also multiplied by the number of chambers connected to the valve to apply the proper resistance to the changing of the volume.

Another change to the model is to have a second relay valve code that accounts for the relay valve in the secondary circuit of the treadle valve. The area is adjusted from the original relay valve code to account for the different size of the piston. The input to this part of the code is the same pressure input as the original relay valve. This part of the code runs separate from the primary circuit and outputs the prediction for the smaller type 20 brake chambers connected to the secondary circuit through the quick release valve.

Figure 4.5 is a simplified flow chart representation of how the models are implemented in the code:

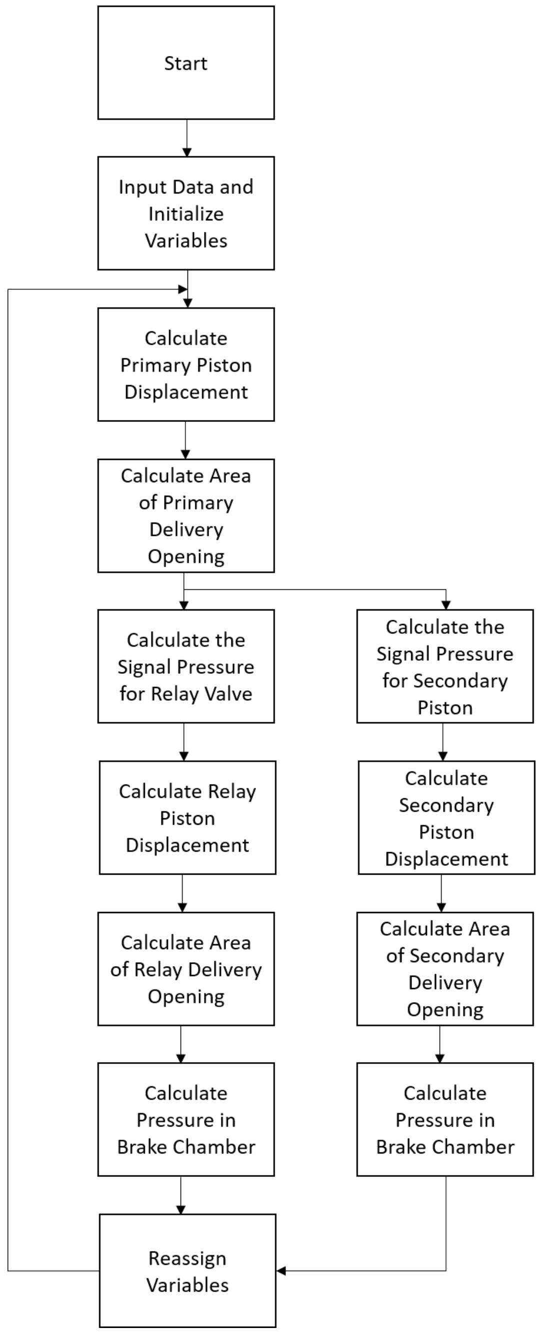

Figure 4.5: Flowchart for Code

5. CORROBORATION OF THE MODEL

5.1 Experimental Results

The corroboration of the model took place using six (6) brake chambers. Two (2) type 20 chambers and four (4) type 30 chambers were used. The type 30 chambers were attached to the relay valve and the type 20 chambers were attached to the secondary circuit of the treadle valve and a quick release valve as described in a previous section. The input data for each model is a graph of the linear actuator position. This input data was adjusted/calibrated to work with the model at 80 psig. All constants used to adjust the data were used across all of the data shown in this section. The constants for adjusting the input data are needed because of how sensitive the model is. In an effort to show how sensitive the model is and provide consistency the constants were the same for every data input. This explains why the data looks very good at 80 psig and the error grows for the surrounding pressures. The model was run in MATLAB's Simulink using a 0.0001 second time-step. When integration was necessary discrete-time integrators were used with a runge-kutta solver. All acquired data was also run through a low-pass filter that filtered out signal noise above 3 Hz.

The following four graphs (Fig. 5.1 to Fig 5.4) are made to show that chambers attached to the same valve build pressure and extend their pushrods at roughly the same rate. This was expected, but it is important to note as the assumption that all of the chambers connected to the same relay valve can be modelled as one larger chamber is an important assumption for modelling this system. With this evidence, the plots that will show the response of the system in comparison to the model will show data for a single brake chamber from the rear and a single chamber from the front of the system. This is to simplify the appearance of the graphs and make them more readable. For the rear chamber data there were two sensors malfunctioning and only three chambers are portrayed. However, this is considered enough to prove the idea that the chambers connected to the rear section all have the same profile. Some of the variation in the pushrod profiles comes from some slight movement in the attachments between the pushrod and the sensor.

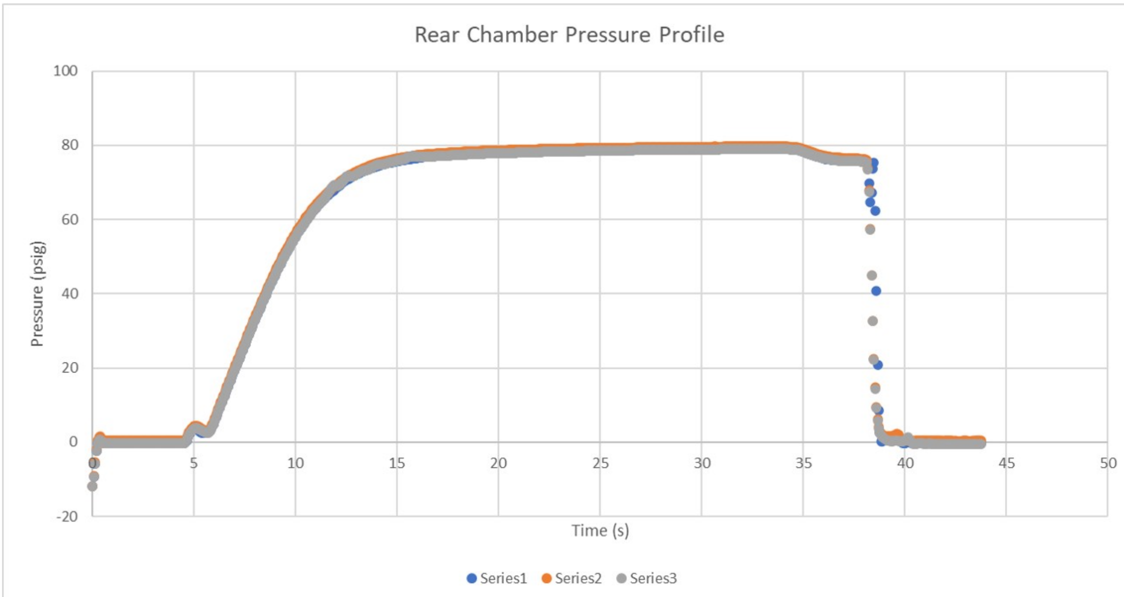

Figure 5.1: Rear Pressure Profile

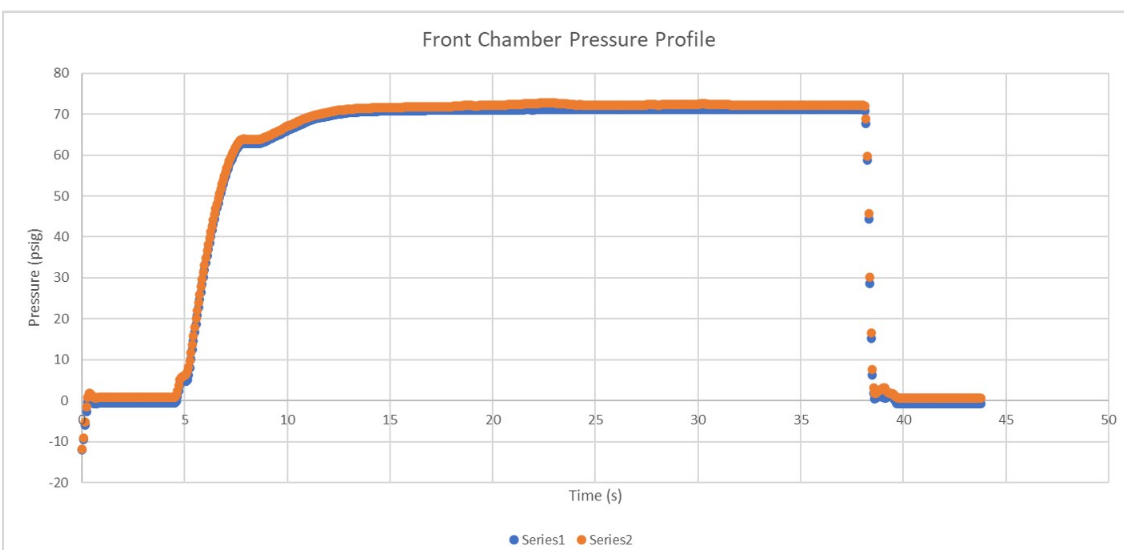

Figure 5.2: Front Pressure Profile

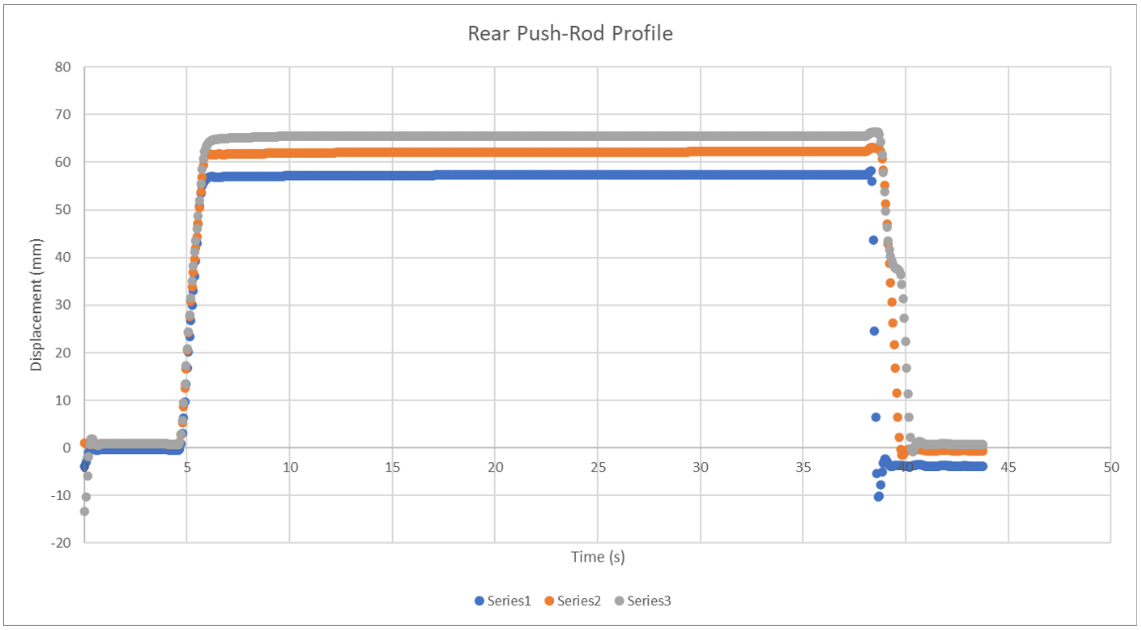

Figure 5.3: Rear Pushrod Profile

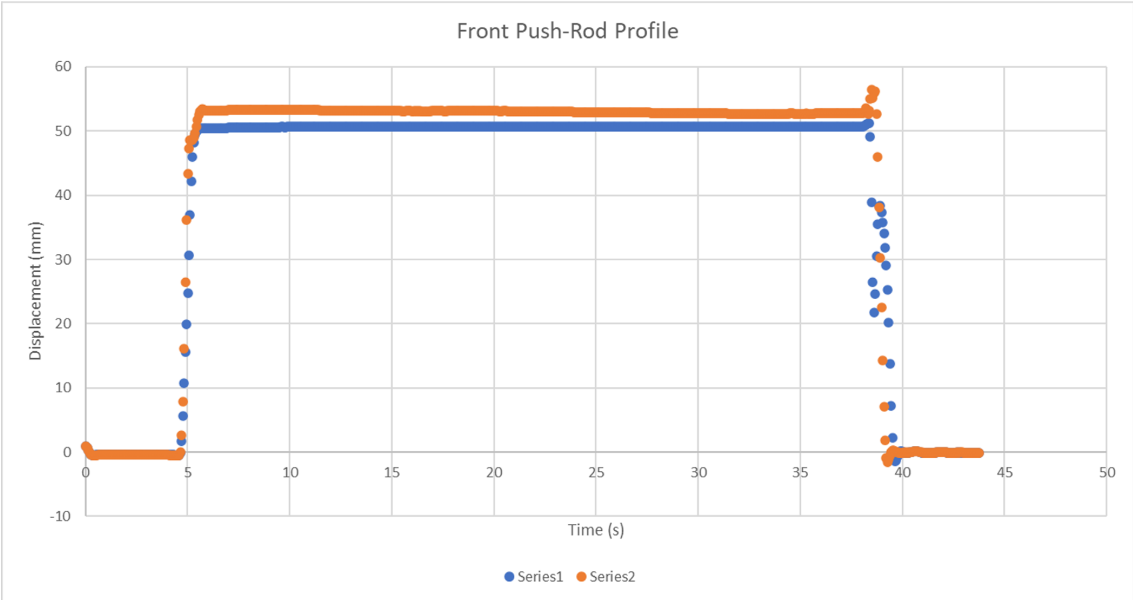

Figure 5.4: Front Pushrod Profile

The 80 psig results (Fig. 5.5 to Fig. 5.6) show that the model can effectively predict the steady-state value of the pressure in each of the chambers. The model is also able to show the split in pressures between the front and rear chambers. This is expected as each piece of the setup is connected to a different relay valve. Each valve will control pressure slightly different. The rise time of the model does not follow closely with the experimental data. There are several reasons why this may be happening that will be discussed at the end of this chapter as this is consistent throughout all tested pressures. The spike in the pressure at the beginning exists in most of the plots and is a result of the pressure building in the treadle valve and forcing the actuator back slightly. This results in a short exhaust right after building pressure. The motor was not released until the pressure was no longer exhausting under the steady state motor input to try and achieve what could be considered the "steady-state" pressure.

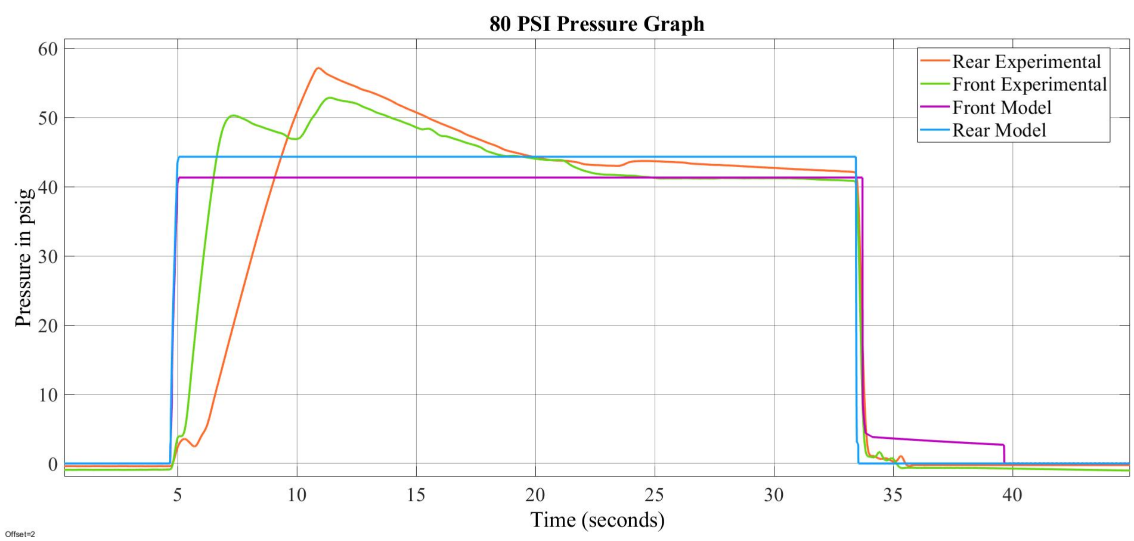

Figure 5.5: 80 psig Results

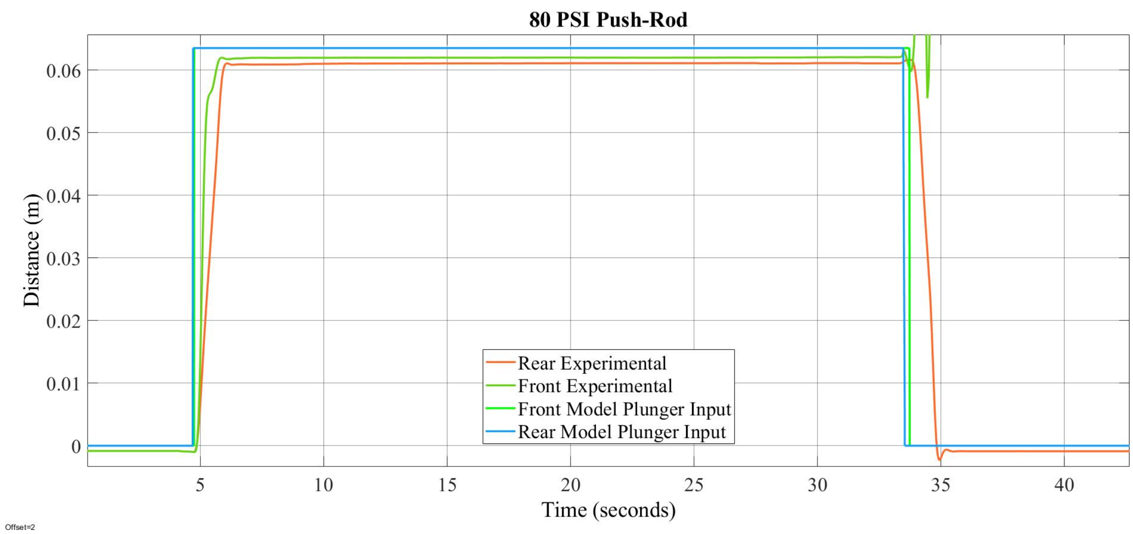

Figure 5.6: 80 psig Pushrod

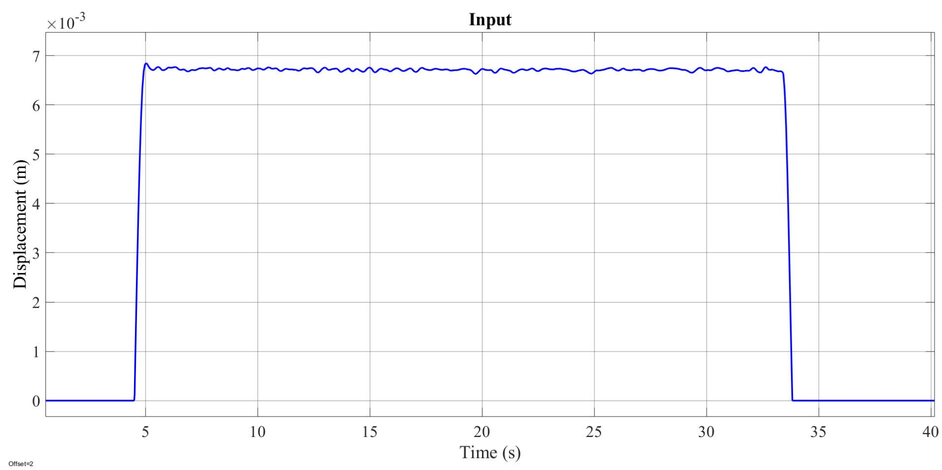

Figure 5.7: Input to the Model

The input data for the model is included above for reference (Fig. 5.7). The "noise" in the signal can be seen in the plot and will show up in some later graphs. The sensitivity of the model is so high that sometimes this limited noise can interfere with the model performance. Figure 5.8 shows a zoomed in view of the plot of the input. This is to show the rise time of the input. Figure

5.9 shows a zoomed in view of the rise dynamics of the model. This is important to show because the zoomed out view can appear as though the model is a step response. This is not the case. Upon closer inspection it can be seen that the volume change is being taken into account. This is evidence that the equations that take into account the volume change are still being used by the simulation.

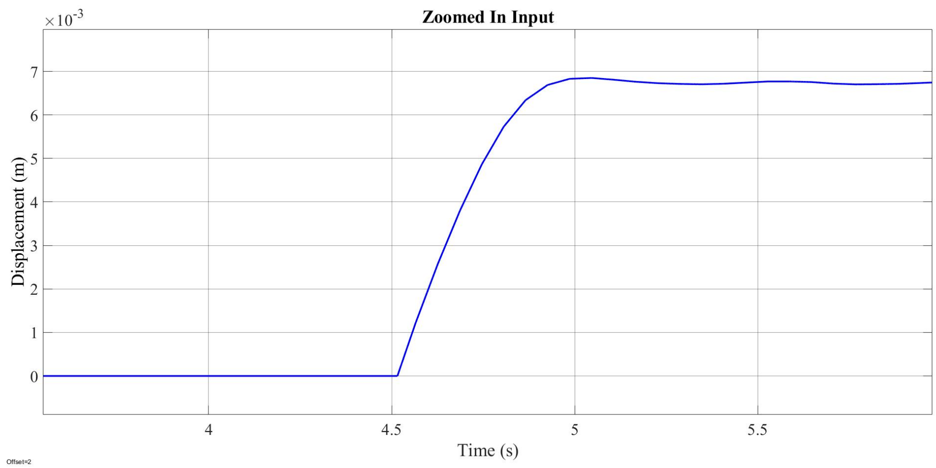

Figure 5.8: Zoomed In Input

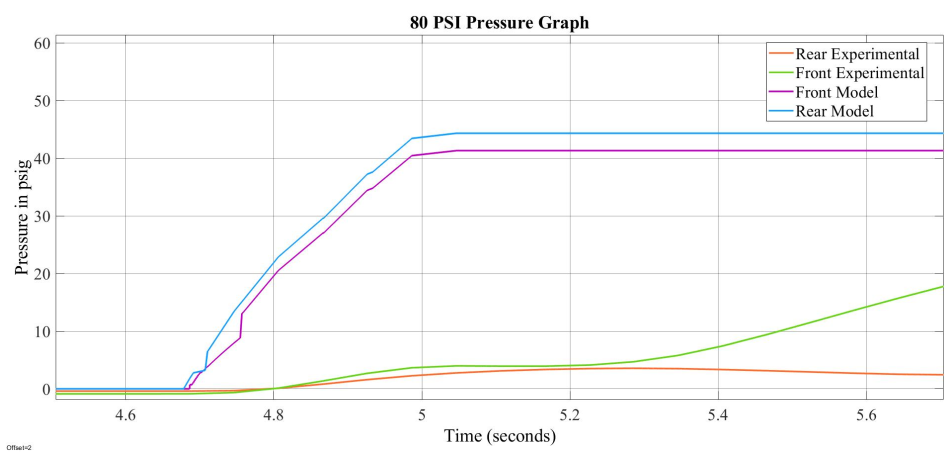

Figure 5.9: Zoomed In 80 psig Data

The 70 psig results (Fig. 5.10 to Fig. 5.11) show an overestimation from the model by about 10 psi. This appears to be a large discrepancy. However, it becomes important to note that according to Bowlin, most of the pressure possibilities using the treadle valve occur within a 2 mm window of displacement of the treadle valve [5]. This leaves very little room for error when estimating. When analyzing this data it was discovered that to change the 70 psig and 90 psig plots from their portrayed responses, to a response that looked more like the response in the 80 psig plot, took as little as a 0.25 mm change to the input data. This small of a change is very hard to account for. There are many ways an error this small could work its way into the system either physically, or just by the calibration and resolution of the linear transducer. For these reasons, a 10 psi difference is not likely to be an error in the model.

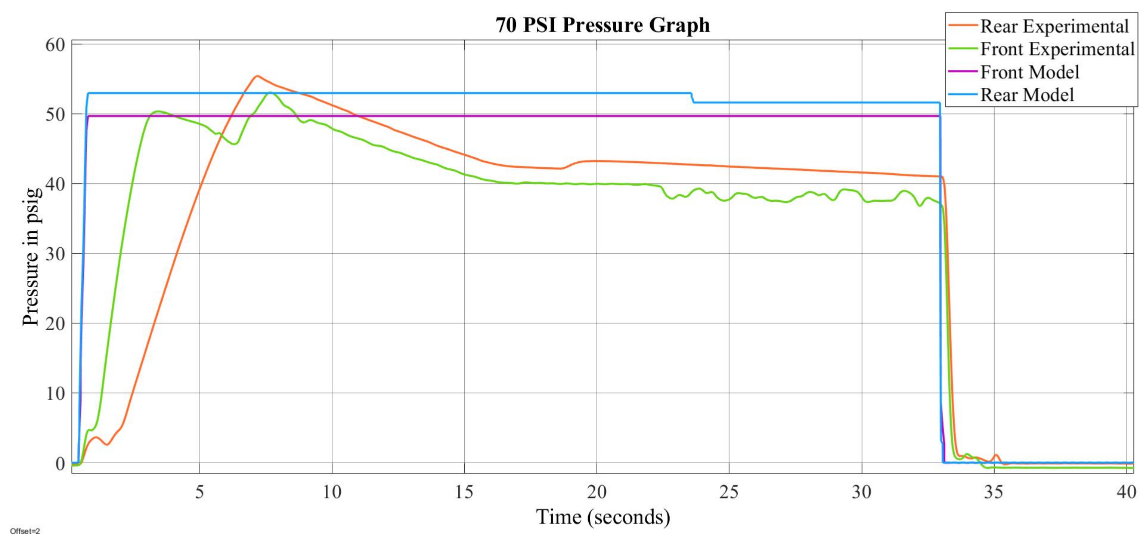

Figure 5.10: 70 psig Results

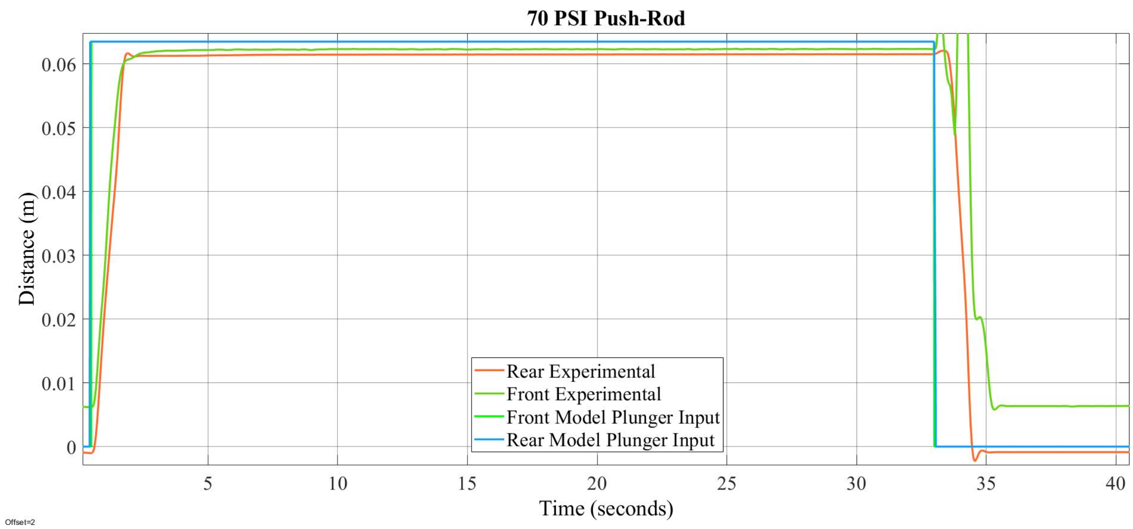

Figure 5.11: 70 psig Pushrod

The 90 psig results (Fig. 5.12 to Fig. 5.13) are similar to the 70 psig results but instead of overestimation, the model underestimates by about the same margin.

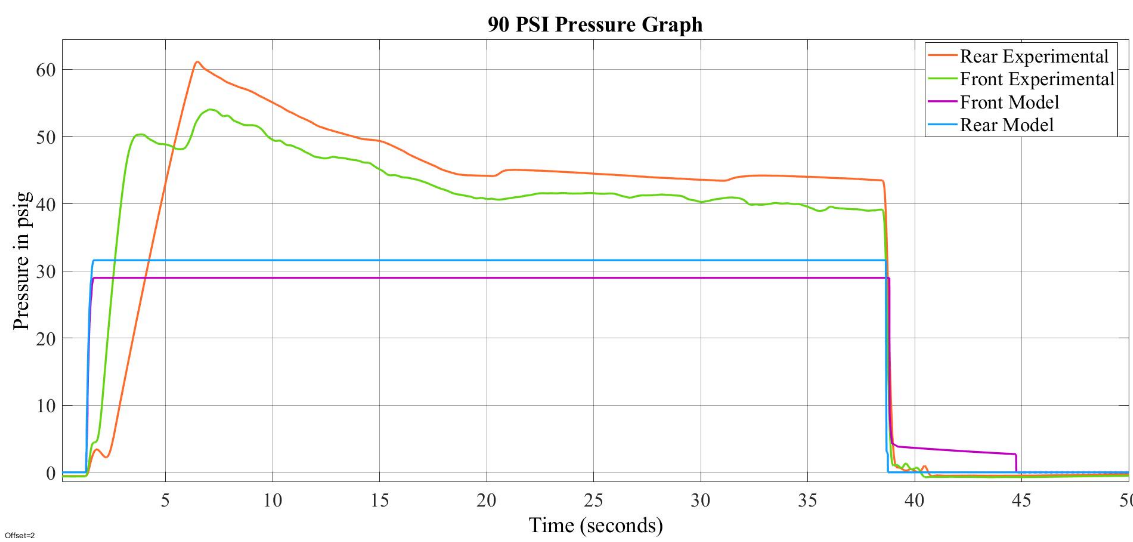

Figure 5.12: 90 psig Results

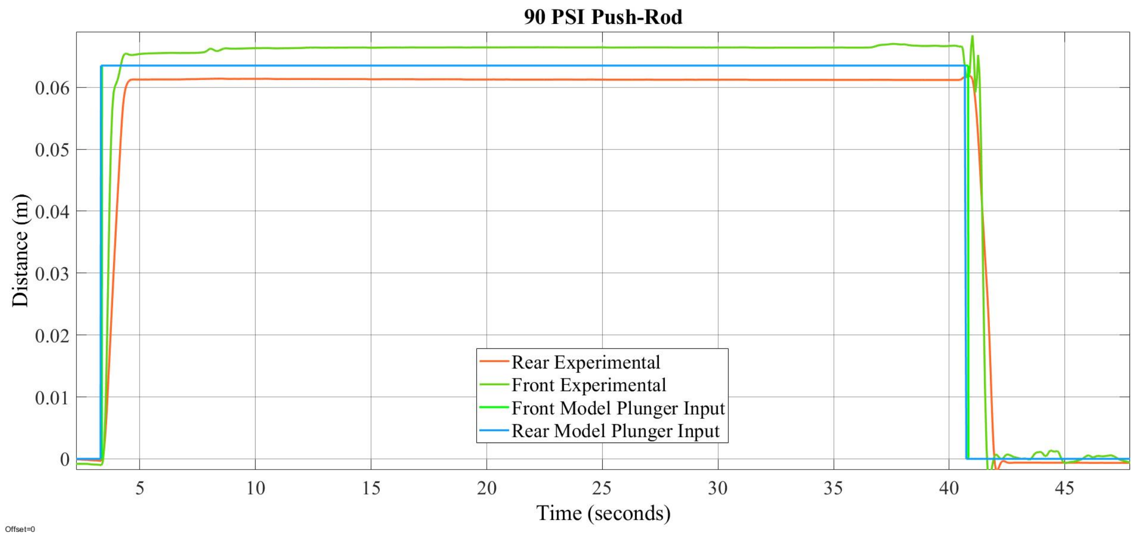

Figure 5.13: 90 psig Pushrod

The 60 psig results (Fig. 5.14 to Fig. 5.15) show a more similar response than 70 and 90 psig. This is likely because at this low of a pressure, the motor is able to open the valve wider and thus, more pressure can be built in the system. Therefore, the model reaches the maximum pressure and the experimental setup almost does the same.

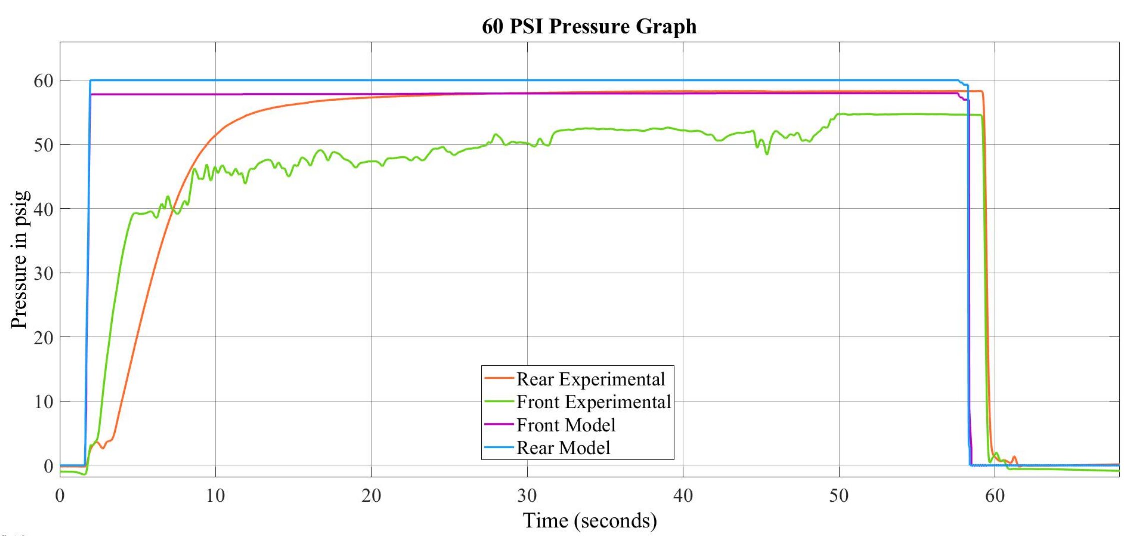

Figure 5.14: 60 psig Results

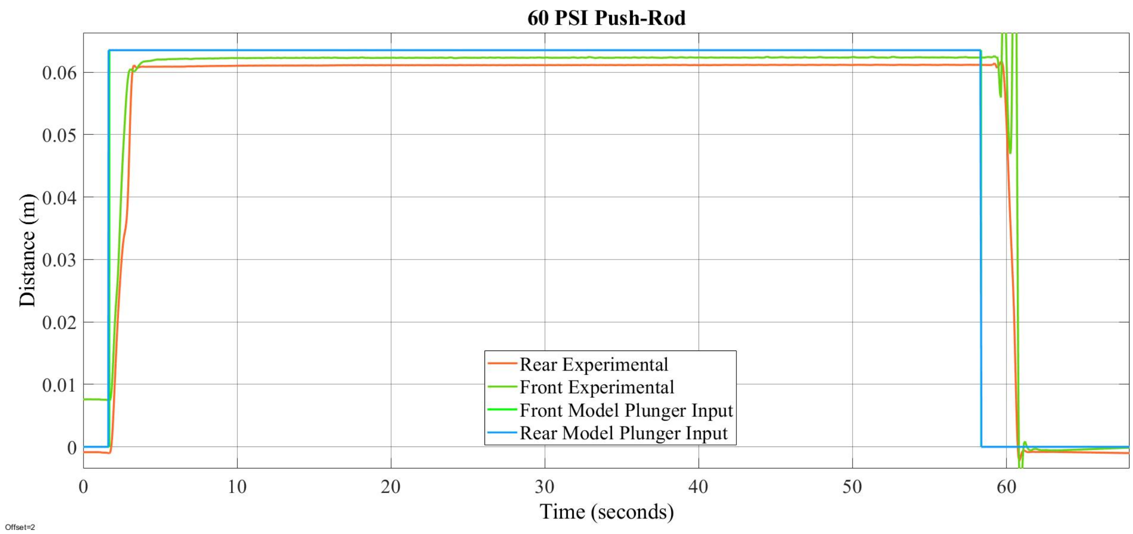

Figure 5.15: 60 psig Pushrod

For reference, the values used for each constant in the simulation are shown in table 5.1. Most of these values are gathered from past studies in order to keep the simulation as close to the original as possible. The most notable change is the maximum pushrod extension. This value changed to represent the pushrod extension without being attached to actual braking system. The values used for the graduated rubber spring are included in table 5.2.

5.2 Sources of Error

This section is to explain where error may come from in the system. Given the current state of the model being used, the sensitivity becomes an issue for future implementation into a diagnostic system. This is because the ability to predict the pressure in the system depending on treadle plunger displacement depends on the ability to measure distances less than a millimeter. Accuracy of calibration and resolution of sensors becomes very important at this scale. The sensor used to measure the displacement in the linear actuator for this experiment has an error of $\pm 5\%$. The documentation is unclear on how to translate this to a nominal distance error. However, it appears this error could be as large as 5 millimeters. Its believed based on the performance in testing that the error was not actually this large, however it is concerning given the sensitivity of the model. More error could result from the ability to mount the motor and treadle valve without any movement relative to each other when they come in contact. The mounting for the motor and treadle valve was upgraded several times throughout this study to improve the sturdiness of the system. While there was little movement that could be seen by the naked eye, no system is perfect and could add to the error. There is also some error in the pushrod experimental data. This is due to the mechanism that attaches the pushrod to the sensor. The movement appears upon the analysis of the data to have been kept to a minimum. However, in the future a better way of attaching the pushrod to the sensor should be devised. Finally, the rise time of the system needs more study. The simulation used for this paper was unable to accurately predict the rise dynamics of the system. This could be a result of an error in the code or an assumption that no longer holds true on a scaled up system.

γ	Air Ratio	1.4
x_{pt}	Valve Open Point	0.002286 m
Ab_{20}	Type-20 Effective Area	0.0129 m^2
Ab_{30}	Type-30 Effective Area	0.0194 m^2
R	Gas Constant of Air	287 $\frac{J}{KgK}$
A_{pv}	Area of Primary Valve Piston	0.0002032 m^2
A_{pp}	Area of Primary Piston	0.002371 m^2
K_b	Return Spring Constant for One Chamber	1167.454 $\frac{N}{m}$
F_1	Pre-loaded Force Primary Valve	-151.196 N
r_{pv}	Radius of the Primary Valve	0.0128 m
x_{bmax}	Max Pushrod Stroke	0.0635 m
C_d	Expansion Factor	0.82
A_{pv1}	Area of Back of Primary Valve Piston	0.0002026 m^2
K_{ss}	Stem Spring Constant	2846.545 $\frac{N}{m}$
K_2	Combination of Valve Spring Constants	9832.181 $\frac{N}{m}$
r_{pp}	Radius of Primary Piston	0.01232 m^2
V_{or}	Volume of Signal Port	1.28966e-5 m^3
V_{o120}	Initial Volume of a Type 20 Brake Chamber	1.639e-4 m^3
T_o	Temperature of Air	298 K
R_{rp}	Radius of Relay Piston	0.0128 m
F_{si}	Pre-Loaded Force of Valve Gasket	34.12 N
A_{1s}	Area of Secondary Valve Piston	0.003189 $\frac{J}{KgK}$
A_1	Area of Relay Piston	0.0064 m^2
A_1'	Area of Backside of the Relay Piston	0.0062 m^2
A_{hole}	Area of Exhaust Hole in Relay Valve	0.0001704 m^2
K_s	Relay Gasket Spring Constant	1478.5 $\frac{N}{m}$
F_{kbi}	Pre-loaded Force of Single Chamber Return Spring	355.54 N
P_{atm}	Atmospheric Pressure	101.356 KPa
P_t	Breakout Pressure	120 KPa
K_3	Exhaust Combined Spring Constant	6418.078 $\frac{N}{m}$
F_2	Exhaust Combined Pre-Loaded Force	-105.8624 N
R_{xpt}	Relay Valve Open Point	0.0058 m
M_{pp}	Mass of Relay Primary Piston	0.0758 Kg

Table 5.1: Values Used in the Simulation

m_1	Constant	155138.175 $\frac{N}{m}$
m_2	Constant	321320.874 $\frac{N}{m}$
m_3	Constant	678364.265 $\frac{N}{m}$
a_1	Constant	11536836074.281 $\frac{N}{m^2}$
a_2	Constant	-111089651.687 $\frac{N}{m}$
a_3	Constant	268522.758 N
n_1	Constant	76.68 N
n_2	Constant	-598.686 N
n_3	Constant	-2176.675 N
l_1	Distance	0.004064 m
l_2	Distance	0.00442 m
l_3	Distance	0.004851 m

Table 5.2: Graduated Spring Constants

6. CONCLUSIONS AND FUTURE WORK

The completion of this study portrays that the previous models that have been devised are likely sufficient for estimating the steady state response of the pneumatic braking system. However, the use of these models could be problematic in the future when implementing them into a diagnostic system. This is because of the sensitivity and variability in the system. The E-7 dual circuit treadle valve has a very small window for determining how much air pressure will build in the brake chambers. Therefore, it would not be unreasonable to assume that the upgrading of the measurements for plunger displacement or using other metrics to predict the system dynamics, such as force, could result in a more reliable model. It is also important to consider that the data gathered from this experiment confirmed that the dynamics of the system mimic that of what was found in previous studies in terms of plot shape. However, this data showed a lot of discrepancy in the rise time for the system. For this reason, it is recommended that some of the assumptions be revisited to improve the dynamic representation over time for the system. The expansion to six chambers may have caused some of the assumptions about neglecting energy loss in the hoses to be no longer valid. The results of the experiment are limited by the parts that were used. Given the limitations of the experiment, it is believed that it was reasonably accurate in determining steady state values for the system and the pushrod.

6.1 Future Work

1. Attach all six brake chambers to full brake assemblies. This would allow for the use of more calibration in the model and might improve the dynamic modeling

2. Test the system with a new foot valve. The foot valve currently being used is difficult to reliably predict

3. Use a stronger motor and a more accurate linear position sensor for the plunger displacement

4. Implement the controller developed by Bowlin [5] on the full setup

5. Revisit the assumptions made in the past to improve the rise time issue in the model

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