MODELING THE PNEUMATIC RELAY VALVE OF AN S-CAM AIR BRAKE SYSTEM

A Thesis

by

SHANKAR VILAYANNUR NATARAJAN

Submitted to the Office of Graduate Studies of
Texas A&M University
in partial fulfillment of the requirements for the degree of

MASTER OF SCIENCE

May 2005

Major Subject: Mechanical Engineering

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Approved as to style and content by:

K. R. Rajagopal
(Co-Chair of Committee)

D. V. A. H. G. Swaroop
(Co-Chair of Committee)

John Valasek
(Member)

Dennis O'Neal
(Head of Department)

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ABSTRACT

Modeling the Pneumatic Relay Valve of an S-Cam Air Brake System. (May 2005)
Shankar Vilayannur Natarajan, B.Tech, Indian Institute of Technology-Madras
Co-Chairs of Advisory Committee: Dr. K.R.Rajagopal
Dr. D.V.A.H.G.Swaroop

Statistics indicate that defects in brake system contribute significantly to fatal crashes involving commercial vehicles. Hence there is a need for developing preventive and active safety measures for assessing the performance of an air brake system in trucks.

Existing techniques for assessing the performance of brakes are infrastructure intensive, time and labor intensive. The premise of this thesis is that model-based diagnostic techniques can be employed to overcome these limitations of existing techniques. The design of a model-based diagnostic system requires the development and experimental corroboration of a mathematical model of the evolution of pressure in each brake chamber of a truck in response to the application of brake pedal input by the driver, when there are no faults or defects in the brake system.

This thesis is aimed at modeling and experimentally corroborating a subsystem of an air brake system, namely the pneumatic relay valve. The pneumatic relay valve takes a input signal from the primary delivery of a treadle valve and meters air from a storage reservoir to Type 30 rear brake chambers. A description of the development of the model, the experimental setup and corroborating experimental results are provided.

ACKNOWLEDGMENTS

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I would like to thank my brother for giving me the courage to face the hurdles in my research. Without him all this would not have been possible. I would finally like to thank my parents for their love and blessings. I am what I am because of them.

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INTRODUCTION

A. Background

Air brake systems are used in heavy vehicles such as trucks, tractors (with trailers attached), buses etc. It is indicated in [1] that most tractor- trailer vehicles with a gross vehicle weight rating over 19000 lb, single trucks with a gross vehicle weight rating over 31000 lb, transit and intercity buses and school buses are equipped with air brakes. It also indicates that more than 85 % of the commercial vehicles in the United States are currently equipped with S-Cam drum foundation brakes. The consequences of an accident involving commercial vehicles can be catastrophic because of their weight. Hence the brake system is very important from the view point of safety of commercial vehicles and other vehicles in traffic.

The performance at normal operating conditions of air brakes relies on proper maintenance procedures. In the absence of proper maintenance, the vehicle may not be able to develop the total braking force needed to bring the vehicle to a safe stop [1]. Brake lining replacement, checking for leaks and adjustment of push rod stroke are some of the important maintenance procedures that are to be followed [1] regularly. A lack of sufficient push rod stroke could cause a delay in brake application due to the time it takes for the extended travel needed for brake application. This could be very dangerous in emergency braking conditions. A loss of pressure due to leaks could cause a decrease in the steady state pressure in the brake chambers resulting in a smaller brake torque than what would be expected for a normal brake application without leaks.

The journal model is *IEEE Transactions on Automatic Control.*

The sensitivity of performance to maintenance procedures necessitates periodic inspection and enforcement of certain regulations for improving traffic safety levels. Federal Motor Vehicle Safety Standards (FMVSS 121) regulates the braking performance of commercial vehicles [2]. It has strict requirements on stopping distances, lateral vehicle stability etc for establishing proper braking. Even though most commercial vehicles carry out such maintenance inspections conforming to FMVSS 121, in the interest of public safety, government agencies such as the Office of Motor Carriers conduct enforcement inspections in accordance with the Commercial vehicle safety alliance (CVSA) guidelines.

Inspection techniques can be classified into two broad categories. They are performance based and visual based inspections [3]. Visual based inspection involves checking the brake linings, measuring push rod stroke and checking wear and tear in hoses. This involves manual checking, which could be time consuming considering the volume of commercial vehicle traffic on highways. It could become all the more difficult in vehicles with a low ground clearance. Inspection statistics between 1996 to 1999 [4] indicate that among the vehicle related violations, 29.3% of violations in intrastate carriers and 37.2% of those among interstate carriers were due to defects in brake system. Performance based inspection involves measurement of stopping distance, brake temperature, brake torque etc. Such a performance appraisal has certain advantages and disadvantages. Nonetheless, it is a useful way of studying the performance of air brakes.

Reference [5] describes the advantages and disadvantages of both performance and visual based inspections. For example, defects like chafed hoses do not show pronounced effect in performance based tests. In the past only a few cases of air leak violations have been detected by the conventional performance based testers namely roller dynamometer and flat plate testers. They only are limited to checking maximum deceleration and not the “lag” or “delay in brake response”. These tests deal with the magnitude of the eventual brake torque. If the leak causes substantial reduction of brake torque, then these tests can detect them. These tests are explained in greater detail in the following chapter. Some other defects such as contaminated linings, kinked air lines or valve spring defects which were not seen readily without dismantling the system, were detected by the performance based system since they had a pronounced effect on the brake force.

Reference [6] emphasizes on the need for a standardized, hand held diagnostic tool for improving the existing inspections. A model based, performance based diagnostic system could help in automating inspections and monitor the performance of the brake system periodically, sufficiently and swiftly. In addition to that, it could also be used to monitor the health of the brake system, which is important from the view point of traffic safety. Existing performance based diagnostic techniques use brake torque measurements (for example roller dynamometer,flat plate tester etc.). Our aim is to monitor the performance and check for faults in the brake system by studying the evolution of pressure with time in the brake chamber. Such a diagnostic system can complement existing diagnostic techniques since brake chamber pressure evolution is directly related to the brake torque and push rod stroke.

In the past, hydraulic brakes have been studied extensively and many models have been described in the literature. In most of the cases the authors have tried to predict the pressure transients in the wheel cylinder chamber with their models. Our intention is to work along similar lines for modeling pneumatic brake systems.

Most of the work related to the mechanical subsystem correlate pressure in the brake chamber with the torque output, brake pad temperature etc. In the air brake system literature (relevant to the pneumatic subsystem), [7] and [8] present a detailed description of the working of the treadle valve. Recently Acarman et al. [9] proposed a model for predicting the pressure transients in the brake chamber of a brake system equipped with antilock braking system(ABS). They used orifice flow equations to model the dynamics of air flow and included the dynamics of a modulator located downstream from the treadle valve. More recently Shankar et al. [7] proposed a model for predicting the pressure transients in the brake chamber of a brake system delivered by a treadle valve. The treadle valve is treated as a nozzle and the dynamics are simulated with the treadle valve plunger displacement (foot pedal displacement) as input.

This thesis deals with the development of a model of relay valve and the experimental corroboration of the same. An outline of the thesis is provided in the next subsection.

B. Outline of the thesis

The following is a brief outline of the chapters that follow.

Chapter II an introduction to air brake systems and describes all the relevant components. It also provides a brief survey of brake diagnostics as applied to commercial vehicles.

Chapter III presents a mathematical model for a relay valve. The various parts of the relay valve are described and relevant governing equations are presented. The assumptions for this model are stated and supported with explanations.

Chapter IV describes the experimental setup that was used to verify the model presented in Chapter III. A detailed explanation of the setup along with the computer interfacing with the setup is given in this chapter.

Chapter V presents the results of experiments and simulation. A discussion of results are also presented in this chapter.

Chapter VI presents sources of error and possible future work.

CHAPTER II

AIR BRAKE SYSTEM

A. The brake system

When the foot pedal in an air brake system is depressed, compressed air flows from the storage reservoir into the brake chambers through a series of valves. An air brake system can be thought of as the interconnection of pneumatic and mechanical subsystem. See Fig. 1 and Fig. 2.

A detailed description of these subsystems follow in the next section.

1. Mechanical subsystem

The mechanical subsystem consists of push rod, S-cam assembly and a slack adjuster. Fig. 2 shows a drum brake, S-cam along with the brake chamber and push rod. A sectional view of an S-cam foundation brake with the push rod and slack adjuster is shown in Fig. 3. When compressed air flows into the brake chambers it drives the push rod forward and this translation causes rotation of the slack adjuster. The slack adjuster rotates the S-cam. This makes the brake pads contact the brake drum thereby causing braking action due to friction.

Usual defects in this subsystem involves the stroke of a push rod increasing with time as a result of brake lining wear or due to the brake drum expansion at higher temperatures. As a result of this, the force output drastically reduces due to this increase in the push rod stroke. This state is usually referred to as “push rod going out of adjustment”. This is rectified using automatic slack adjusters that compensate for the loss in stroke.

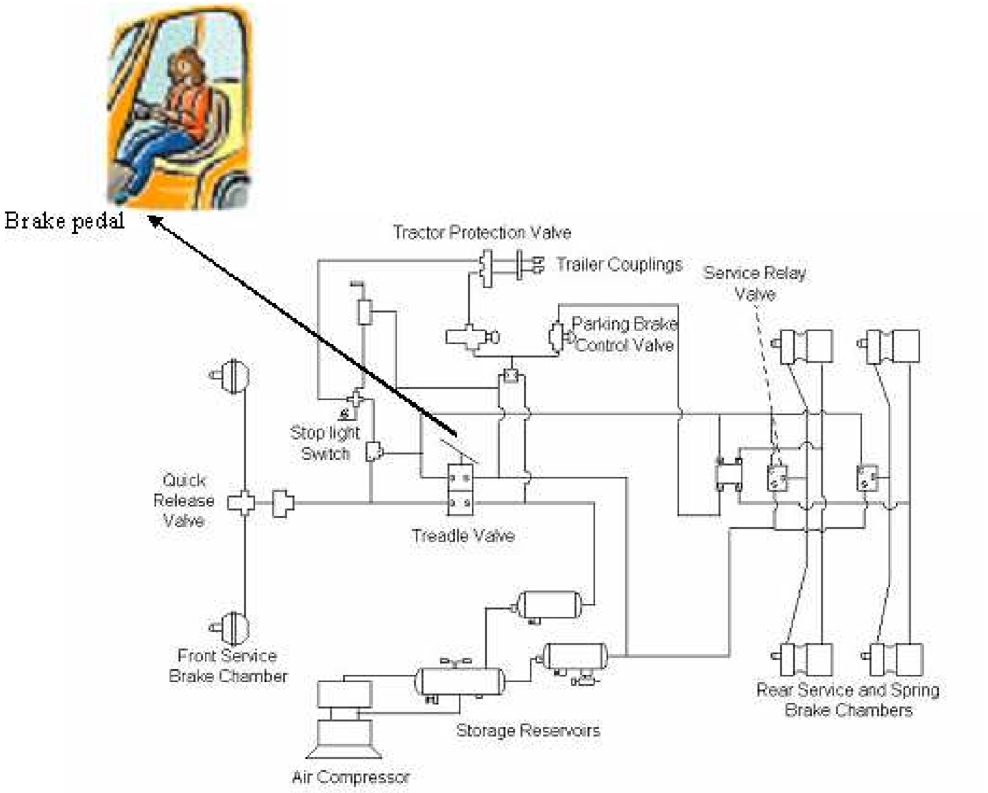

Fig. 1. Layout of a truck brake system

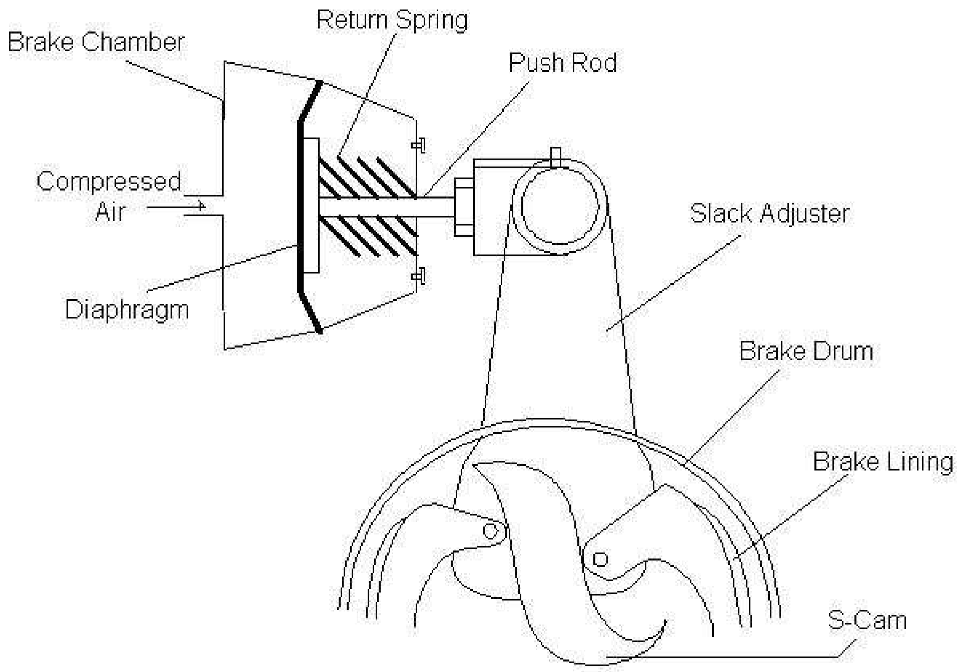

Fig. 2. Schematic of a drum brake along with the brake chamber and push rod

2. The pneumatic subsystem

The pneumatic subsystem consists of a treadle valve, a service relay valve, a quick release valve, an air compressor which charges a storage reservoir, front brake chambers and rear brake chambers just to name the most significant parts. Valves form a very important part of the pneumatic subsystem. The most important valves in a brake system are explained in the following sections.

a. Treadle valve

When the foot pedal in an air brake system is depressed, the treadle valve meters the flow of compressed air from the storage reservoir to the brake chambers. As a result, pressure of air in the brake chambers increases and causes the push rod to move and effects braking action in the mechanical subsystem as explained before. A

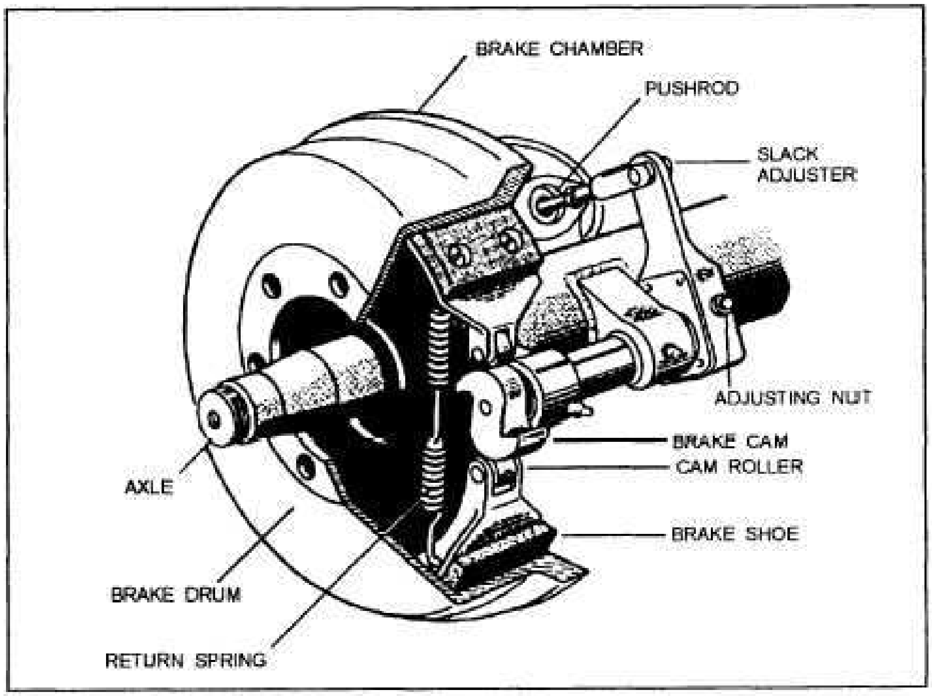

Fig. 3. An S-cam air brake system [10]

mathematical model for the treadle valve was presented in a recent paper by Shankar et al. [7]. A detailed explanation is presented in the same. This paper treats the treadle valve as a nozzle to develop the model.

When a brake application is made, due to the treadle valve being located close to the front axle, air takes a longer time to flow to the rear brake chambers causing a “brake lag” (delay in brake application) in the rear axle. Such a braking is highly undesirable since it causes premature braking in some axles. Hence, in order to have braking without a brake lag, a relay valve is used in proximity to the rear brake chambers to provide immediate braking (apply and exhaust) in the rear axles.

b. Relay valve

The primary circuit of the treadle valve controls the relay valve and the secondary circuit causes braking action at the front axle through a quick release valve. The relay valve has its own supply port and it gets its supply through a secondary storage reservoir. When air from the treadle valve reaches the relay valve, it opens a vent that connects the storage reservoir and the rear brake chambers [10]. This causes brake application at the rear axles. The exact mechanism of action of the relay valve is explained in greater detail in the next chapter, where a mathematical model of the valve is presented.

c. Quick release valve

As the name suggests, it is used for quickly exhausting air from the front brake chambers. When brakes are applied, air from the secondary circuit of the treadle valve flows into the quick release valve. The quick release valve has a diaphragm that gets pushed by the flow of the air. It seals the exhaust and air flows through the delivery port of the quick release and into the front brake chambers (refer to layout Fig. 1). When the pressure is no longer sufficient to hold the diaphragm sealing the exhaust, it detaches itself from the exhaust port. This causes air in the front brake chambers to quickly exhaust through the exhaust port of the quick release valve.

B. Existing diagnostic techniques and scope for improvement

In this section, a list of the existing performance based diagnostic schemes is presented. As mentioned before, performance based inspection involves measurement of stopping distance, brake temperature, brake torque etc. On Aug 9, 2002, the Federal Motor carrier Safety Administration (FMCSA) made a rule to allow inspection of commercial vehicles using performance based brake testing. Some commonly employed performance based testers are the roller dynamometer and flat plate plate testers.

In a flat plate tester, the test vehicle in motion is stopped by applying brakes. The brake forces generated are measured on each axle. In a roller dynamometer the wheels of the vehicle are rotated on rollers while the vehicle remains stationary. The resistance offered by the wheels to the rotation is a measure of the brake forces on that axle. This test does not indicate the lag or delay between the command (pedal input) to the braking action. Rather it only deals with the eventual magnitude of the brake torque.

Some existing onboard techniques give a warning when the push rod stroke exceeds a particular limit. They maybe done using transducers that transmit a signal to give a warning when the push rod stroke exceeds a limit or using stroke indicators (mounted on the push rod) to facilitate visual inspection. Brake pad wear indicators are also used to check the extent of wear on the brake pads.

The proper performance of air brake system depends on many parameters. It is to be noted that the push rod stroke is one of the parameters used to indicate the performance of an air brake system. It is well established that lack of sufficient push rod stroke affects the brake performance significantly. Since leakage in the air brake system increases brake lag time, and delays the braking, it is very important to include its role in developing performance based diagnostics along with the push rod stroke adjustment. It is possible that the push rod maybe in proper adjustment, but in the event of a leakage, sufficient braking force may not be generated. Hence it is important to develop a diagnostic system that can detect lack of sufficient push rod stroke as well as detect leaks. Since the pressure evolution in the brake chamber is a useful indicator of the presence of leaks in hoses and malfunctioning of the brake valves, our goal is to supplement the existing diagnostic schemes with a model based, performance based diagnostic system that makes heavy vehicle braking as safe as possible.

CHAPTER III

A MATHEMATICAL MODEL OF A RELAY VALVE

A. Working mechanism of a relay valve

For modeling a relay valve, it is important to understand how the parts are assembled and how the air is channelized through the various ports of the relay valve. Fig. 4 shows the various parts of a relay valve.

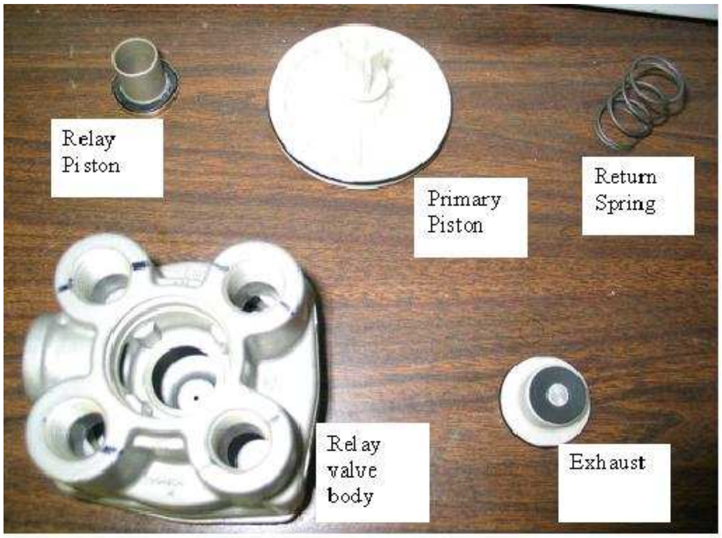

Fig. 4. Relay valve parts

The parts are assembled in the following way as shown in Fig. 5.

The functioning of a relay valve is similar to that of a treadle valve. Ref [7] explains the functioning of a relay valve and provides a mathematical model for the evolution of pressure transients as a function of the brake pedal displacement input. The application of brakes takes place in three phases, namely apply, hold and exhaust. Each of these phases is explained in the following sections.

Fig. 5. Relay valve parts getting assembled

1. Apply

When the brake pedal is depressed, the treadle valve meters air to the input port of the relay valve. This acts on the face of the primary piston of the relay valve as shown in Fig. 6.

Compressed air acts on this face of the primary piston

Fig. 6. Primary piston exposed to the inflow of compressed air

This causes the primary piston to move and come into contact with the relay piston. Till this point the inlet valve is closed to the supply pressure. During this phase there is no change in pressure in the brake chamber. Fig. 7 shows the position of the primary piston before it comes into contact with the relay piston. The relay piston is not shown seated so that the primary piston can be seen.

Fig. 7. The primary piston position before it comes into contact with relay piston(not shown for the sake of clarity)

As the pressure on the primary piston increases, it tries to overcome the preloads due to the relay valve return spring. The apply phase starts when the combined primary piston and relay piston start moving together (see Fig. 8 and Fig. 9).

During this phase the combined movement of the primary piston and the relay piston opens the inlet between the supply and the delivery ports with the exhaust port remaining closed. This causes compressed air from the storage reservoir to flow into the brake chambers. The inlet is clearly shown in Fig. 10.

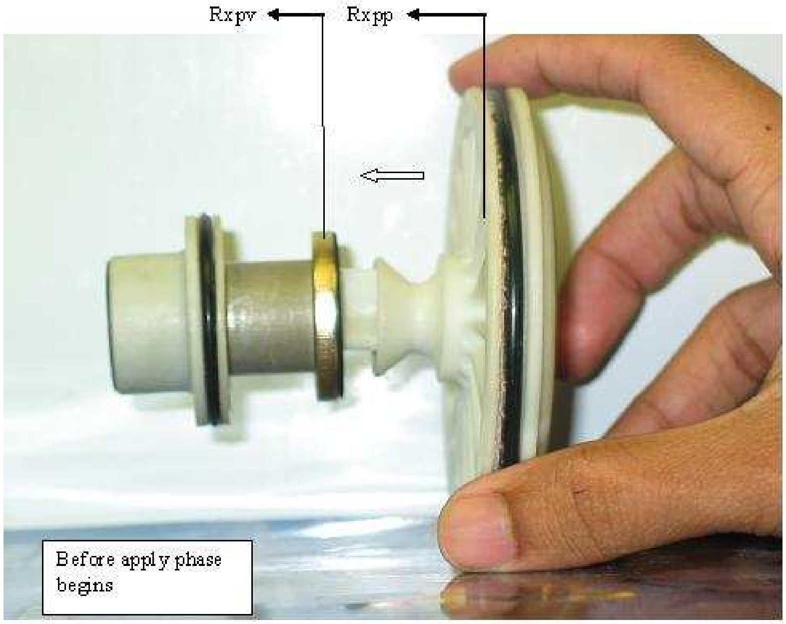

Fig. 8. The primary piston and relay piston before they come into contact

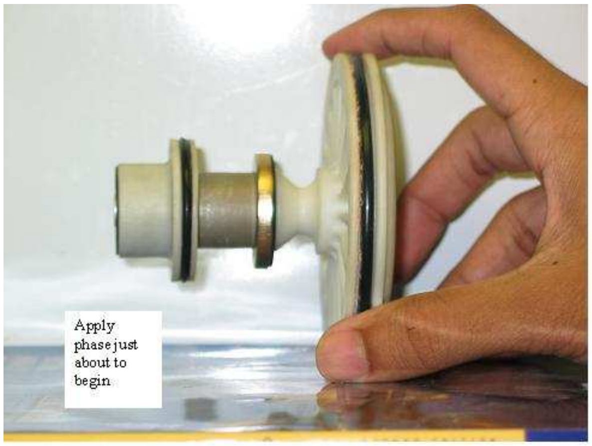

Fig. 9. The primary piston and relay piston after they come into contact

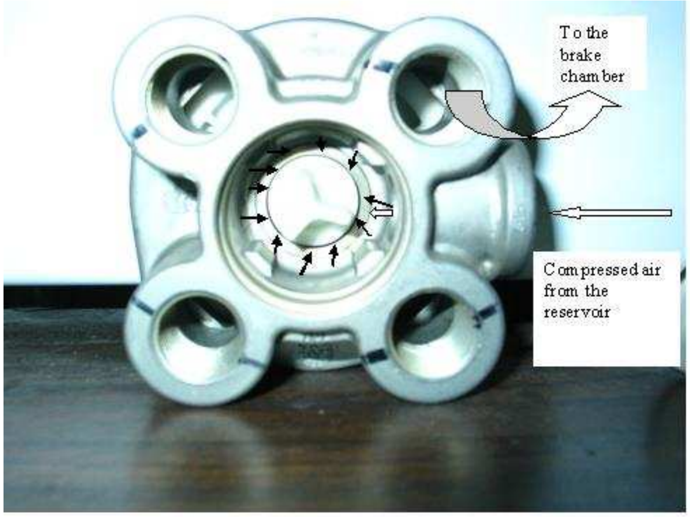

Fig. 10. The inlet and air flow from the supply to the delivery(again relay piston is not shown for the sake of viewing the inlet)

2. Hold

As the pressure in the brake chamber increases to a level where it balances the pedal input force, the inlet valve is closed and the relay piston moves back to its seat. The primary piston is still in contact with the relay piston. During this phase both the inlet and the exhaust valves are closed. This phase is called the hold phase. The pressure of air in the brake chamber reaches a steady value.

3. Exhaust

When the pedal is released, the applied force on the primary piston reduces, which causes the primary piston to break contact with the relay piston as shown in Fig. 11. This causes air in the brake chambers to get exhausted through the exhaust port. This is called the exhaust phase. Fig. 12 shows the exhaust valve opening and the flow of air from the brake chambers to the atmosphere.

It is therefore natural to expect that the governing equations for each phase will

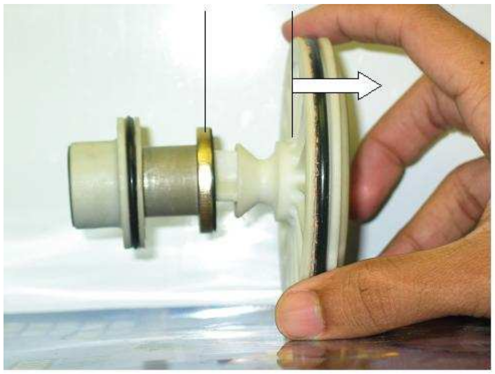

Fig. 11. Primary piston breaks off from the relay piston

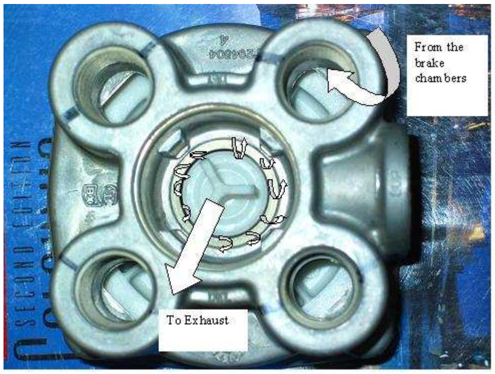

Fig. 12. Exhaust and flow of air from the brake chambers to the atmosphere

be different since the piston dynamics are different and flow of air changes direction as the phase changes from apply to exhaust.

B. Modeling assumptions

The model that is proposed was developed in [7] to model the treadle valve. The assumptions that were used in the model are as follows.

1. Viscous effects of air are neglected i.e. air is assumed to be perfectly elastic. This means that air behaves like a perfectly elastic spring that stores the energy given to it. This energy can completely recovered i.e. process is completely reversible.

2. The valve opening is considered as a nozzle. This is a reasonable assumption because the cross sectional area of the valve opening decreases monotonically to a minimum value and ratio of cross sectional area of the supply port of the relay valve to the cross sectional area of the valve opening is 15.4. So we can assume that the approach velocity is negligible and upstream properties can be assumed to be the stagnation properties [11].

3. Flow through the valve is assumed to be one dimensional and adiabatic.

4. Fluid properties are uniform at all the cross sections of the nozzle.

5. Compressibility effects in air in the hoses are assumed negligible. This is experimentally verified by taking Mach number measurements at the exit of the hoses. Since the values were less than 0.3 it is reasonable to assume that compressibility effects are negligible. This does not imply that rubber hoses are incompressible. It only implies that the process happens slow enough to keep the Mach number less than 0.3 [7].

6. The supply pressure acting on the valve piston is assumed to have no transients. The reservoir is so large that the any decrease in supply pressure due to the valve piston movement is replenished immediately by the reservoir.

C. Model

1. Piston dynamics

We can see from figure Fig. 8 and infer the conditions for the three phases in the brake application and release.

The three stages of operation of the relay valve operation can be described by the following relations:

- Apply Phase

$$Rx_{pp} > Rx_{pt} \tag{3.1}$$

- Hold Phase

$$Rx_{pp} = Rx_{pt} \tag{3.2}$$

- Exhaust Phase

$$Rx_{pp} < Rx_{pt} \tag{3.3}$$

where Rx_{pp} is the relay piston displacement, Rx_{pt} is the distance the primary piston needs to travel before it opens up the inlet valve opening and air starts to flow from the reservoir to the brake chambers.

Now, at any instant of time during the apply and hold phases,

$$Rx_{pv}(t) = Rx_{pp}(t) - Rx_{pt} \tag{3.4}$$

where Rx_{pv} is the valve gasket assembly displacement.

During the apply phase the equations of motion of the primary piston and the relay piston are as follows.

$$M_{pp}\left(\frac{d^2Rx_{pp}}{dt^2}\right) = P_{app}A_1 - P_{rd}A_1' - F_{reacn} - P_{atm}A_{hole} \tag{3.5}$$

$$M_{pv}\left(\frac{d^2Rx_{pv}}{dt^2}\right) = F_{reacn} - K_sRx_{pv} - F_{si} \tag{3.6}$$

where M_{pp} is the mass of the primary piston

M_{pv} is the mass of the relay piston and gasket assembly

P_{app} is the applied pressure input from the delivery port of the treadle valve to the control port of the relay valve

P_{rd} is the delivery pressure of the relay valve

F_{si} is the valve gasket spring preload

A_1 is that area of the primary piston on which the input pressure acts

F_{reacn} is the force that the relay piston exerts on the primary piston when they are in contact K_s is the relay gasket spring constant

A_1' is that area of the piston on which the relay delivery pressure acts

A_{hole} is that area of the piston on which the exhaust atmospheric pressure acts and this portion constantly sits on the exhaust and keeps it closed till the apply phase ends.

Adding the 2 equations in Eq. 3.5 and Eq. 3.6 and neglecting the masses of the piston and valve gasket assembly, we get

$$P_{app}A_1 - P_{rd}A_1' - P_{atm}A_{hole} - K_sRx_{pv} - F_{si} = 0 \tag{3.7}$$

Eq. 3.7 determines Rx_{pv} during the apply phase. Friction is neglected in all the equations.

During the exhaust phase, the valve gasket gets disconnected from the piston and thereafter the piston moves on its own due to pressure differences on either side of the piston. The valve gasket seats on the inlet and does not allow any more compressed air to move into the brake chambers. The equation governing the motion of the piston during exhaust is

$$M_{pp}\left(\frac{d^2Rx_{pp}}{dt^2}\right) = P_{rd}\acute{A}_1 - P_{app}A_1 \tag{3.8}$$

2. Nozzle dynamics and modeling the fluid flow

The model used by [7] is used to model the flow of compressed air through the relay valve. The brake chamber is treated as a control volume and mass balance is applied [7]. Fig. 13 shows a brake chamber.

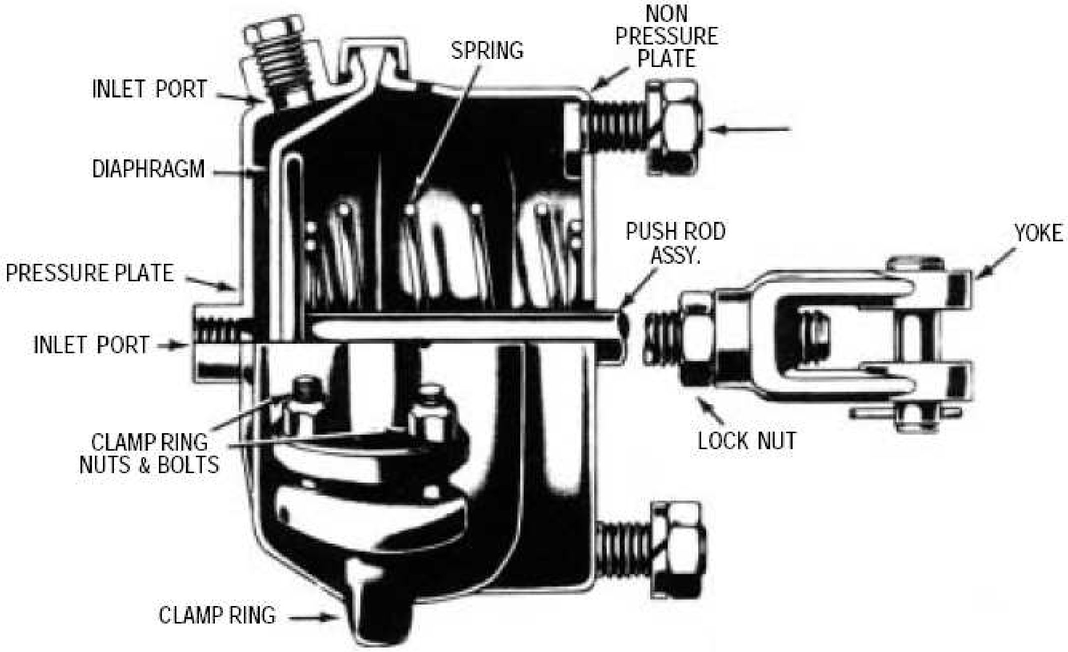

Fig. 13. Type 30 brake chamber [12]

Fig. 14 shows the control volume of the brake chamber.

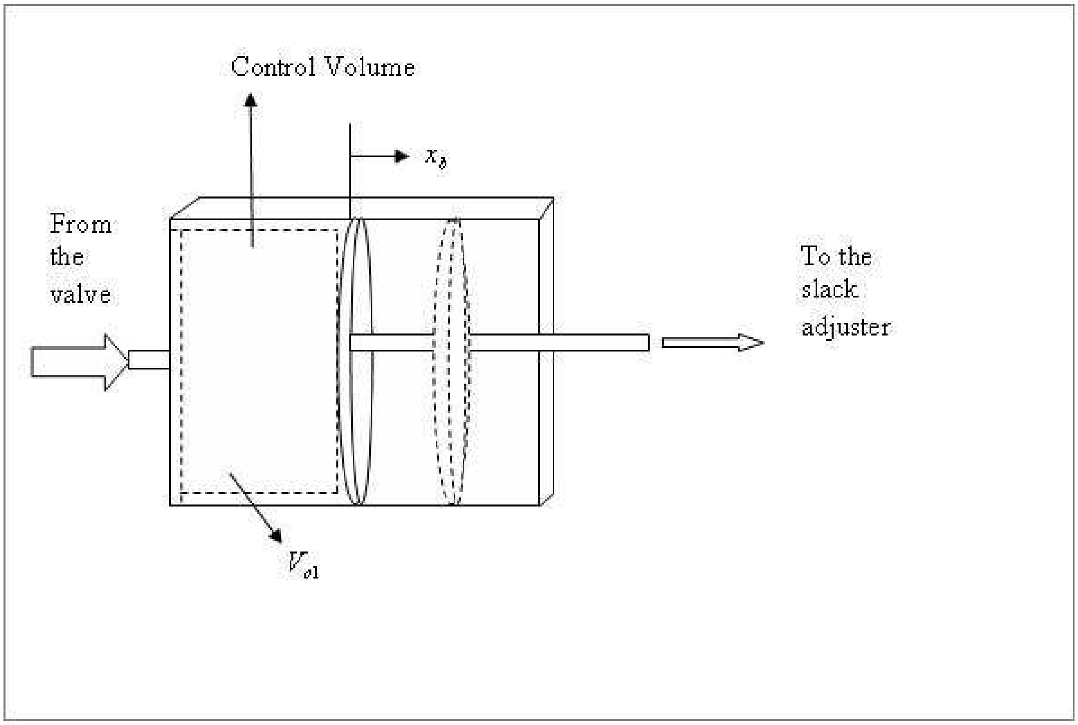

Fig. 14. Brake chamber control volume

The relay valve opening is modeled as a nozzle. The nozzle area is affected by the dynamics of the piston as follows:

$$\begin{aligned} A_p = &\ \ 2\pi R_{rp}(Rx_{pp} - Rx_{pt}) \ \ \text{during the apply phase} \qquad (3.9) \\ = &\ \ 2\pi R_{rp}(Rx_{pt} - Rx_{pp}) \ \ \text{during the exhaust phase} \qquad (3.10) \end{aligned}$$

where A_p is the nozzle area and Eq. 3.8 determines Rx_{pv} during the exhaust phase. The equations for the nozzle are presented below.

For the apply phase (refer to [7]):

$$\left(\frac{A_p C_D}{RT_b}\left(\left(\frac{2\gamma}{\gamma-1}\right)\frac{P_o}{\rho_o}\left|\left[1-\left(\frac{P_b}{P_o}\right)^{\left(\frac{\gamma-1}{\gamma}\right)}\right]\right|\right)^{\frac{1}{2}} sgn(P_o - P_b)\right) P_b = \begin{cases} \left(\dfrac{V_{o1}}{\gamma RT_b}\right)\dot{P}_b & \text{if } P_b < P_t \\ \left(\dfrac{V_b}{\gamma RT_b} + \dfrac{P_b A_b^2}{RT_b K_b}\right)\dot{P}_b & \text{if } 0 \leq x_b < x_{bmax} \\ \left(\dfrac{V_{o2}}{\gamma RT_b}\right)\dot{P}_b & \text{if } x_b = x_{bmax} \end{cases} \qquad (3.11)$$

$$V_b = \begin{cases} V_{o1} & \text{if } P_b < P_t \\ V_{o1} + A_b x_b & \text{if } 0 \leq x_b < x_{bmax} \\ V_{o2} & \text{if } x_b = x_{bmax} \end{cases} \qquad (3.12)$$

where V_{o1} is the initial volume of air in the control volume before the application of the brake, V_{o2} is the maximum volume of air in the control volume, A_b is the cross-sectional area of the brake chamber, x_b is the stroke of the brake chamber diaphragm, i.e., the stroke of the push rod, x_{bmax} is the maximum stroke of the push rod and P_t is the relay crack pressure. (See Fig. 14)

If the pressure ratio is less than a critical value choked flow conditions are assumed. The critical pressure ratio is given by Eq. 3.13.

$$\left(\frac{P_b}{P_o}\right)_{cr} = \left(\frac{2}{\gamma+1}\right)^{\left(\frac{\gamma}{\gamma-1}\right)} \tag{3.13}$$

The final equations from [7] are

$$\left(A_p C_D \left(\left(\frac{2\gamma}{\gamma-1}\right)\frac{1}{RT_o}\left|\left[\left(\frac{P_b}{P_o}\right)^{\left(\frac{2}{\gamma}\right)} - \left(\frac{P_b}{P_o}\right)^{\left(\frac{\gamma+1}{\gamma}\right)}\right]\right|\right)^{\frac{1}{2}} sgn(P_o - P_b)\right) P_o =$$
$$\begin{cases} \left(\dfrac{V_{o1} P_o^{\left(\frac{\gamma-1}{\gamma}\right)}}{\gamma R T_o P_b^{\left(\frac{\gamma-1}{\gamma}\right)}}\right)\dot{P}_b & \text{if } P_b < P_t \\ \left(\dfrac{V_b P_o^{\left(\frac{\gamma-1}{\gamma}\right)}}{\gamma R T_o P_b^{\left(\frac{\gamma-1}{\gamma}\right)}} + \dfrac{P_b^{\frac{1}{\gamma}} A_b^2 P_o^{\left(\frac{\gamma-1}{\gamma}\right)}}{RT_o K_b}\right)\dot{P}_b & \text{if } 0 \leq x_b < x_{bmax} \\ \left(\dfrac{V_{o2} P_o^{\left(\frac{\gamma-1}{\gamma}\right)}}{\gamma R T_o P_b^{\left(\frac{\gamma-1}{\gamma}\right)}}\right)\dot{P}_b & \text{if } x_b = x_{bmax} \end{cases} \tag{3.14}$$

Eq. 3.14 is the governing equation of pressure in the brake chamber. It needs to be solved together with equation Eq. 3.7 to get the pressure transients during the apply phase along with the condition that choked flow conditions occur when the pressure ratio is less than a critical value as in equation Eq. 3.13.

Similarly for the exhaust phase we use Eq. 3.15 in equation Eq. 3.11.

$$V_b = \begin{cases} V_{o2} & \text{if } x_{be} = 0 \\ V_{o2} - A_b x_{be} & \text{if } 0 \leq x_{be} < x_{bmax} \\ V_{o1} & \text{if } x_{be} = x_{bmax} \end{cases} \tag{3.15}$$

where x_{be} is the displacement of the brake chamber diaphragm from its equilibrium position at the start of the exhaust phase. We get similar equations governing the pressure transients in the exhaust phase. This needs to be solved along with equation Eq. 3.8.

D. Simulation

Equations 3.7, 3.8, 3.14, 3.15 were solved using the Euler method because it is reasonably simple and intuitive and gives fairly accurate solutions. Moreover the input is a low frequency brake application and hence the numerical noise can be expected to get damped. Hence it is reasonable to expect fairly accurate results for these brake applications. This is corroborated by the fact that experimental results match well with the simulations. The results are presented in the chapter V. The parameters used for the simulation are also provided in the appendix.

An experiment was set up to validate the model. The setting up of the experiment is explained in detail in the following chapter.

CHAPTER IV

EXPERIMENTAL CORROBORATION

A setup which was primarily designed for testing a treadle valve was upgraded to add a rear axle which mounts a relay valve and a wooden plate bolted to the front axle which mounts a quick release valve. The original setup and the upgraded setup are explained in the following sections.

A. Previously designed experimental setup

An experimental test bench which is the front axle of a tractor was used by Shankar et al. [7] in corroborating the results of their model for the treadle valve. A schematic is shown in Fig. 15. The treadle valve used was an E-7 dual circuit valve manufactured by Bendix.

A brief explanation of the relevance of the setup with the actual layout is as follows:- When the brakes are applied, the treadle valve, as explained before, meters air to the brake chambers mounted on the various axles. These brake chambers actuate the foundation brakes thereby causing braking action. In this experiment, the treadle valve was given measured displacement inputs. The pressure transients in the brake chambers were measured using pressure sensors. Compressed air supply to the treadle valve was provided by a compressor. The various components and the manufacturers are described below.

In the experiment built and described by Shankar et al. [7], the treadle delivers to a single brake chamber. The pressure transients in a single brake chamber (Type 20 of cross sectional area 20 in^2) was measured using pressure sensors mounted in a pitot tube assembly specially fabricated to suit the purpose (shown in Fig. 15). The equipment used and their manufacturers are as follows:

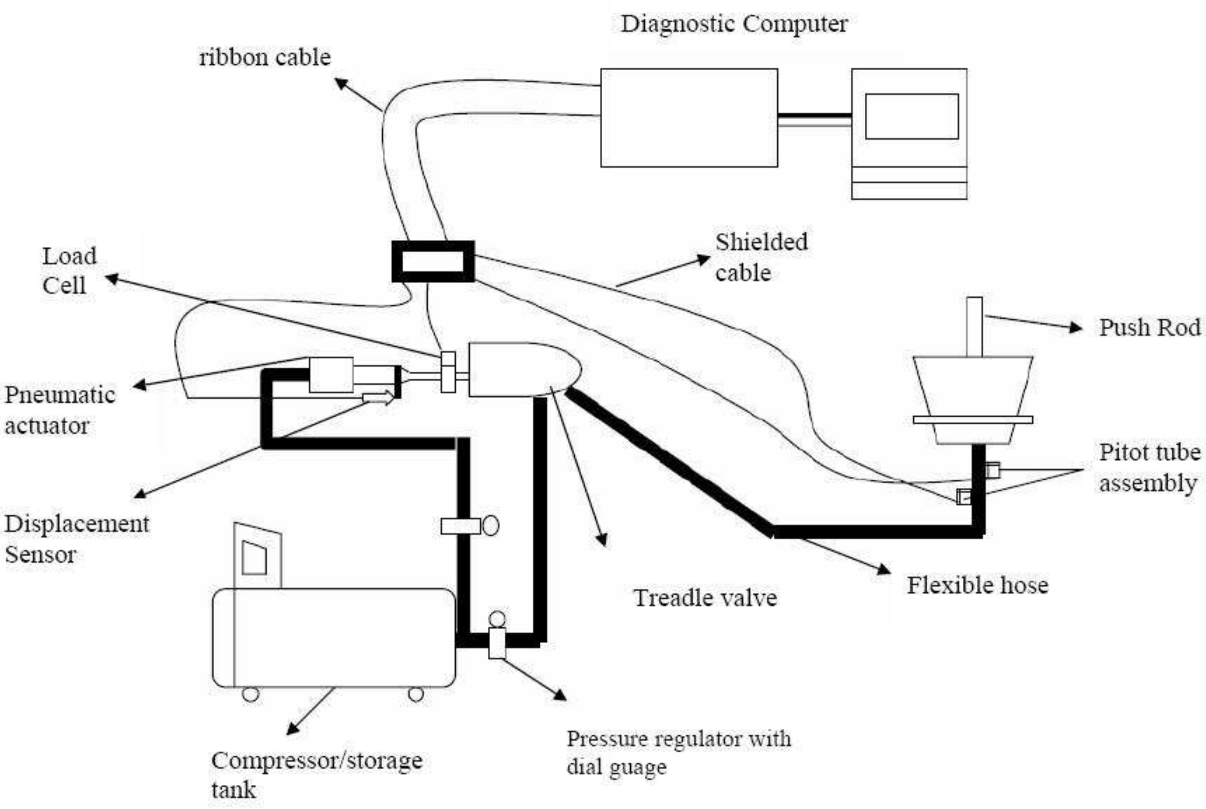

Fig. 15. Layout of the initial setup

- **Pressure regulator** - Manufactured by Omega Engineering, Model number PRG501-120.
- **Linear Potentiometer**(3" stroke)- Manufactured by Omega Engineering, Model number LP802-75.
- **Load cell**- Manufactured by Omega Engineering, Model number LC203-1K.
- **Load cell exciter 10V DC power source**-Manufactured by Omega Engineering, Model number DMD-465WB.
- **Pressure sensor**-Manufactured by Omega Engineering, Model number PX181-100G5V.

- **Pressure sensor excitation 24V DC power supply**-Manufactured by Omega Engineering, Model number U24Y101.
- **Data acquisition board**- National Instruments Model number PCI-1200

In addition to this shielded cables were used for connecting the sensors to a the connector block. the connector block was interfaced with the DAQ board through ribbon cables. The compressor used was provided by Campbell Hausfeld (Oil less compressor). Hoses for the pneumatic transmissions were bought from Mc-Master-Carr. A Matlab application program was written to record the data provided by the sensors.

Some pictures of the existing setup are shown in Fig. 16 and Fig. 17.

Fig. 16. A view of the experimental test bench [13]

B. Upgraded setup

The setup was upgraded to include a pneumatic relay valve and type 30 brake chambers. The relay valve actuates the brake chambers mounted on the rear axles. The

Fig. 17. Another view of the experimental setup([13])

purpose of these valves have been explained in chapter II.

The objective of this setup is to simulate the braking action at the rear axle. Hence a setup was developed using erector struts, brackets and gussets made of aluminium (manufactured by 80/20 Inc. The industrial erector set). This was built to mount the brake chambers and the brake valves. A picture of the setup is shown in Fig. 18.

The brake chambers are mounted in such a way that the push rod stroke can be adjusted to any desired maximum stroke. The relay valve is independently supplied by a secondary storage reservoir. The primary circuit of the treadle valve is connected to the control port of the relay. The relay valve delivers to the type 30 brake chambers mounted on the frame built for this purpose (which serves as the rear axle).

In addition to this a quick release valve is mounted on the original experimental test bench (which serves as the front axle) with the help of a wooden plate. The equipment used for measurement and instrumentation are the same as the previous

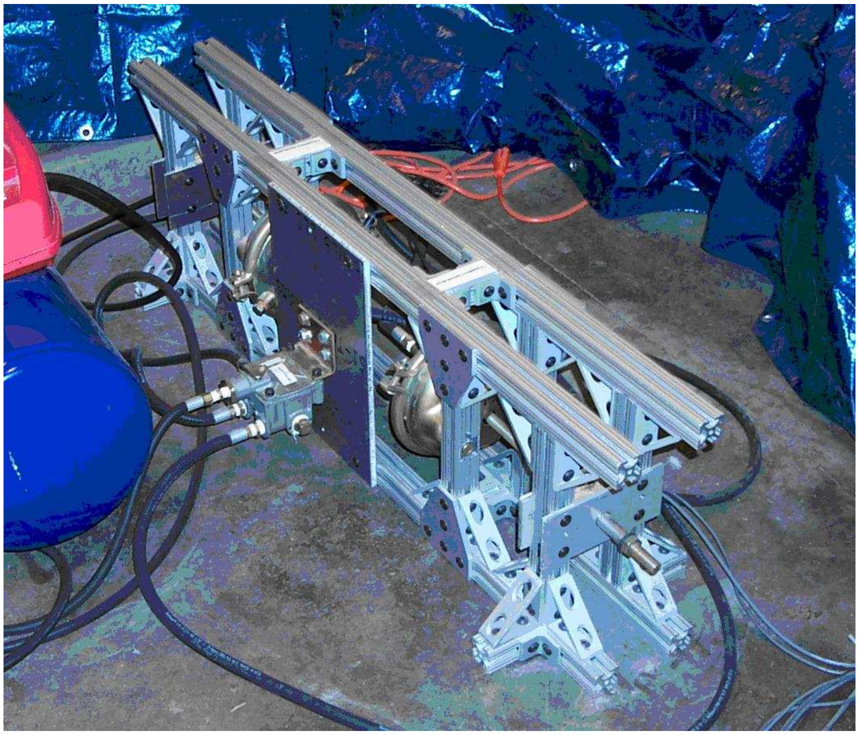

Fig. 18. The upgraded setup

setup. The new layout is shown in Fig. 19.

Data acquisition board used for the new setup was also upgraded to have more analog input channels since the setup has more sensors. The previously used DAQ board (used for the treadle valve testing) did not have enough channels for supporting more than five inputs.

Data was collected by giving the treadle valve displacement inputs through the actuator. The primary delivery of the treadle was connected to the control port of the relay and the relay delivery was connected to a single type 30 brake chamber. The primary delivery of the treadle which serves as the input to the relay valve was measured by another pressure sensor. This was done to calibrate the relay valve independently i.e. by treating it as a separate valve which takes pressure inputs and delivers pressure outputs to the brake chamber. The following chapter presents the

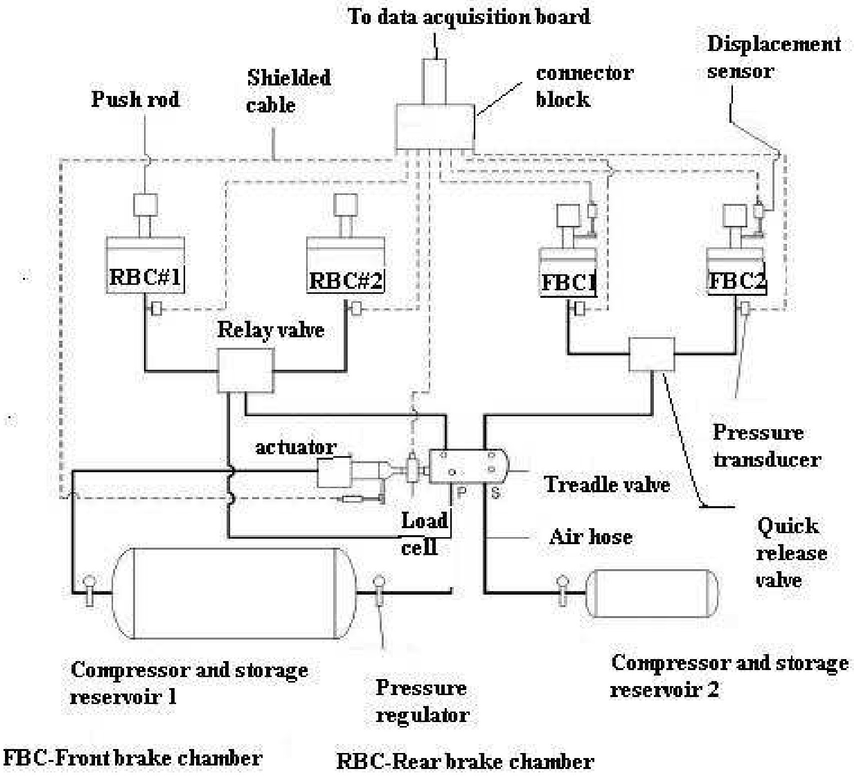

Fig. 19. A layout of the upgraded setup

experimental results and their correlation with the predictions of the model.

CHAPTER V

RESULTS AND DISCUSSION

In this section we will show that the experiments corroborate the model. Experiments were conducted at different supply pressures for different brake pedal inputs. The equations described in the chapter(IV) are solved and the results are presented. Equations (3.1), (3.2) and (3.3) were used to determine the start, end and switching inside the simulations.

The simulation for the apply phase starts when Rx_{pv} becomes greater than zero. The apply phase is terminated when Rx_{pv} becomes less than zero or when the brake chamber pressure. When Rx_{pv} becomes less than zero the exhaust phase begins. Many tests were conducted at various brake applications at various supply pressures. They are shown in the following figures. Both the model and the experimental results of pressure transients are plotted on the same plots to show the comparison. **All the pressure transients shown were measured in a single type 30 (rear axle) brake chamber.**

- event $= 1 \longrightarrow$ Phase before the primary piston comes into contact with the relay piston.
- event $= 2 \longrightarrow$ Phase when the the primary piston is in contact with the relay piston.
- event $= 3 \longrightarrow$ Phase when the primary piston and relay piston are moving together and apply phase has just started. Compressed air has started flowing into the brake chamber while the push rod has not started moving since brake chamber return spring exerts a preload on the push rod opposing the force due to the pressure of air flowing into the brake chamber.

- event = 4 ⟶ Phase when the force due to air flowing into the brake chamber has overcome the spring preloads and the push rod starts moving. The volume in the brake chamber increases and hence rate of increase of pressure is less.
- event = 5 ⟶ Phase when push rod stroke is a maximum and the volume of the brake chamber cannot increase any further. The primary piston is still in contact with the relay piston.
- event = 6 ⟶ Phase when the primary piston breaks off from the relay piston due to reduction in pressure input from the treadle valve. During this phase air from the brake chambers get exhausted through the exhaust port. The brake chamber volume is still a maximum.
- event = 7 ⟶ Phase when air is still getting exhausted from the brake chambers and the push rod is returning back to its natural position.
- event= 8 ⟶ Phase when air is still exhausting and push rod has returned to its natural position.

Clearly when event is 4, the push rod stroke is increasing. When the push rod stroke is increasing the rate of pressure increase is less owing to increase in volume of the brake chamber. This flattening of the pressure curve was predicted by Shankar et al. [7] for the treadle valve.

A. Test result plots

1. Brake application

Figs. 20 and 21 present plots showing the pressure transients, input pressure to the relay valve, event, push rod displacement and primary piston displacement. The various stages of brake application can be inferred from these plots. It is to be noted that the primary piston displacement increases without any increase in the pressure in the brake chamber till event=3. Thereafter any increase in the primary piston displacement occurs along with the relay piston, thereby causing the inlet to open and air to flow into the brake chamber.

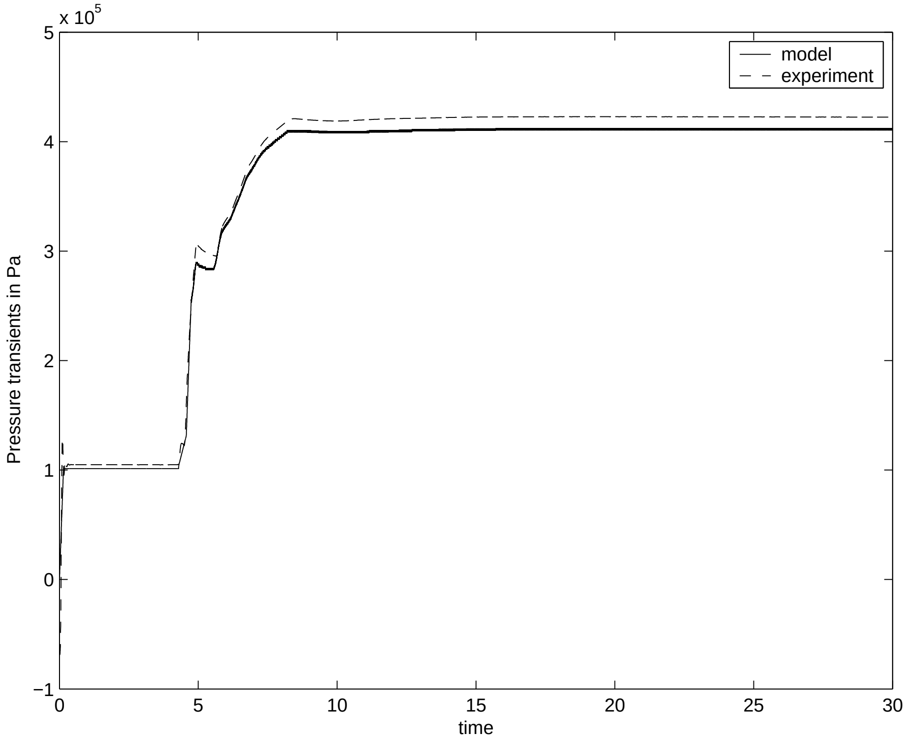

Fig. 20. Pressure transients for a supply pressure of 90 psi(model and experiment)

Another test result done for a supply pressure of 80 psi is shown below in Fig. 22 and Fig. 23.

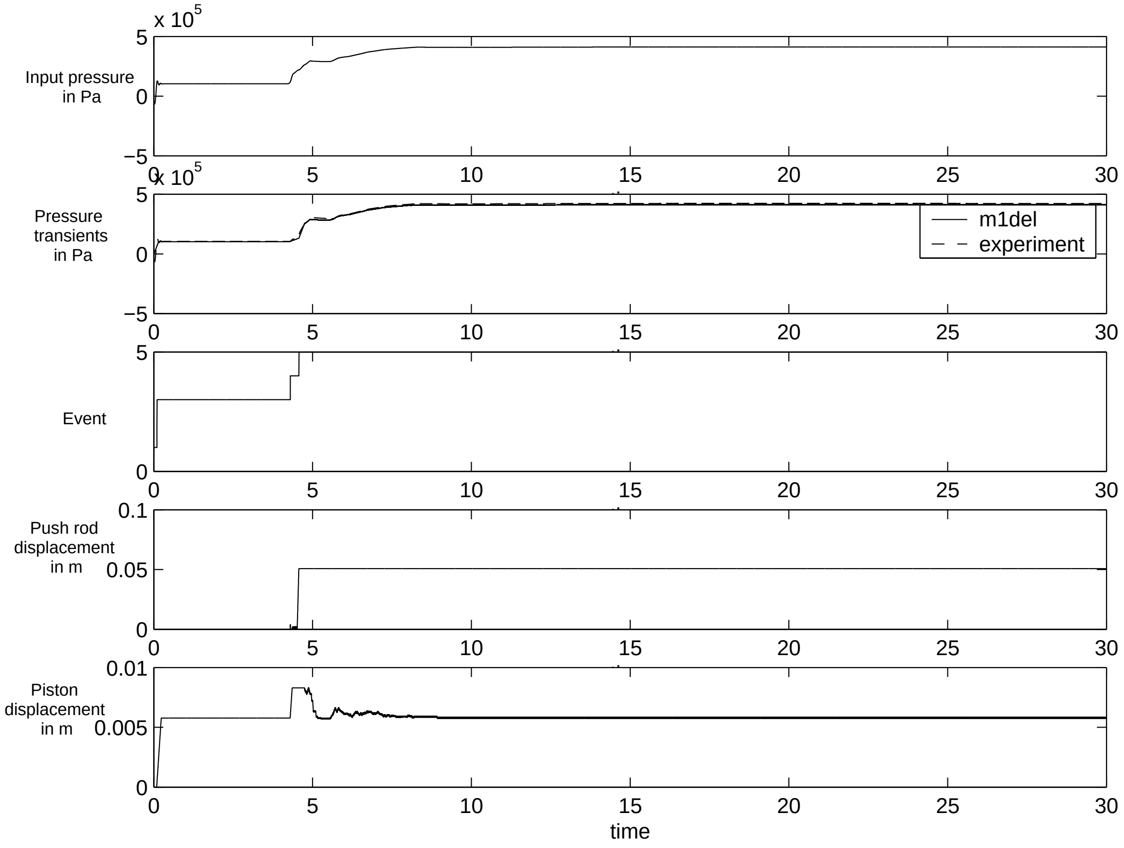

Fig. 21. Curves of all the variables at a supply pressure of 90 psi

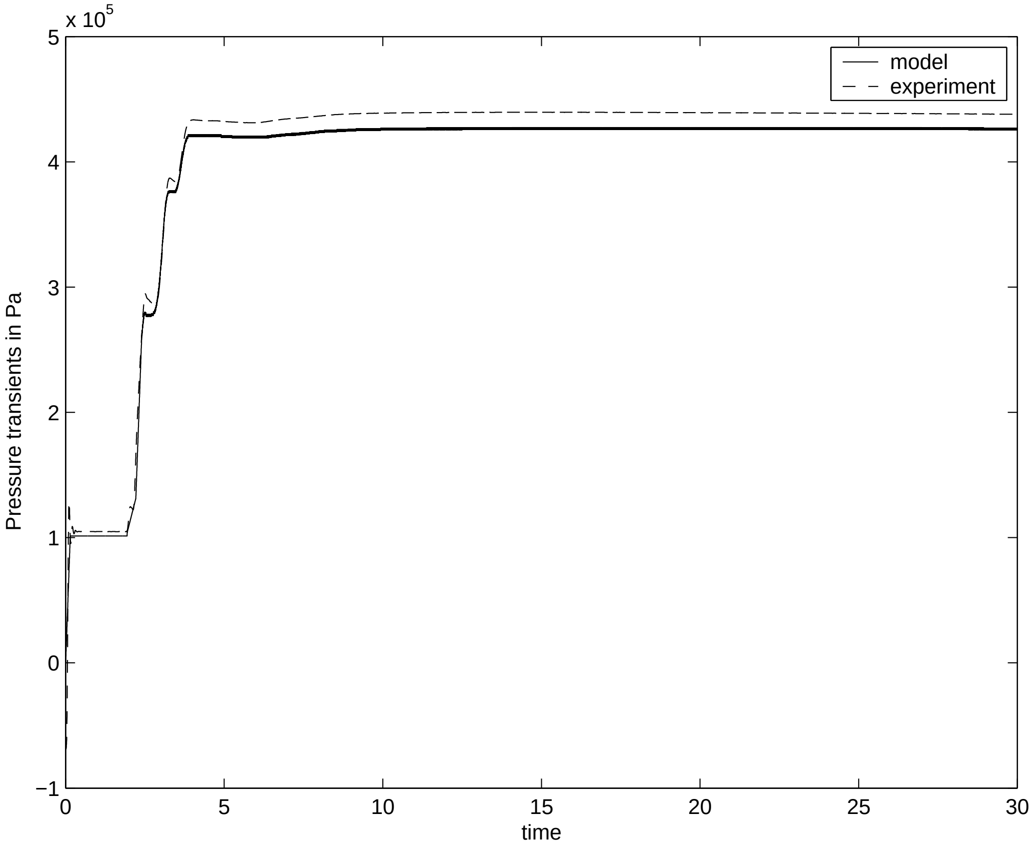

Fig. 22. Pressure transients for a supply pressure of 80 psi(model and experiment)

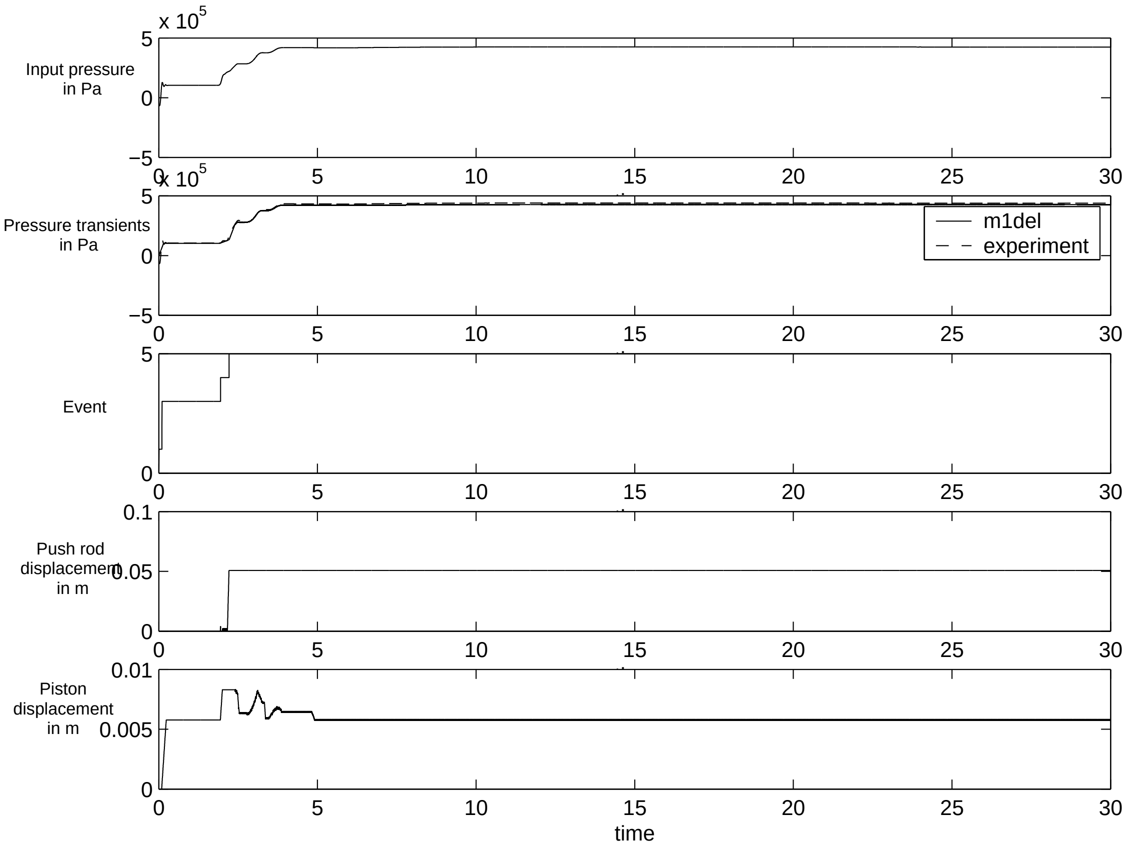

Fig. 23. Curves of all the variables at a supply pressure of 80 psi

2. Brake application and exhaust

Some tests were done with apply and exhaust. They are shown in Fig. 24. As explained before we can see that the event plots give us all the information about the various stages of brake application and exhaust. A test result showing brake application and exhaust with event plots at 70 psi are shown in in Fig. 25 and Fig. 26.

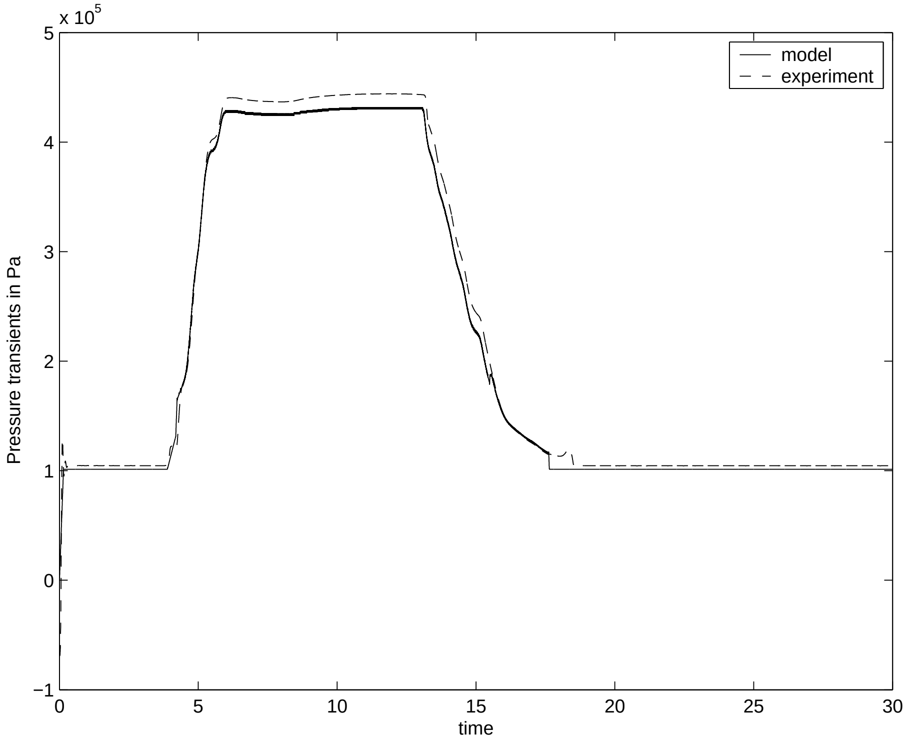

Fig. 24. Pressure transients for a brake application at a supply pressure of 80 psi(model and experiment)

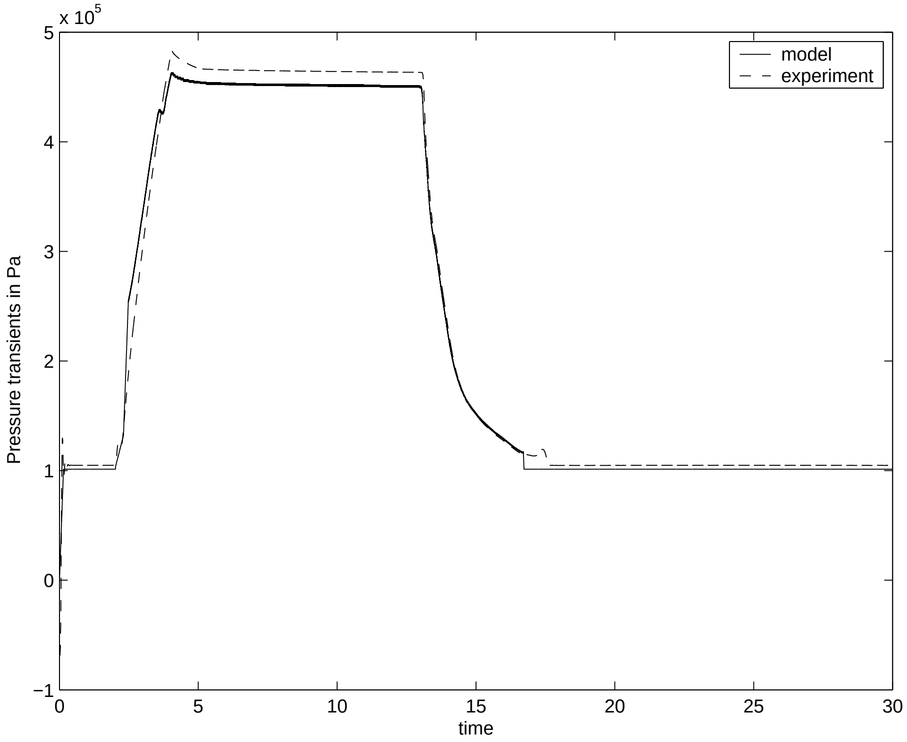

Fig. 25. Pressure transients for a brake application at a supply pressure of 70 psi(model and experiment)

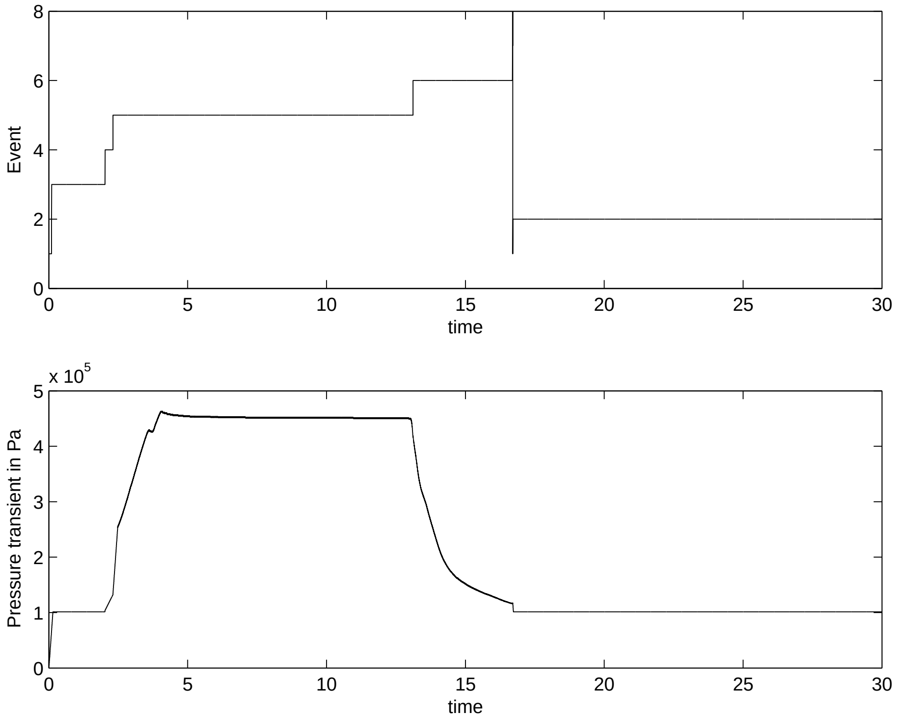

Fig. 26. Pressure transients for a brake application at a supply pressure of 70 psi with event plot

3. Cyclic brake application and exhaust

Some tests were conducted with a cyclic brake application as shown in Fig. 27 and Fig. 28. These plots are shown to emphasize on the fact that the model captures the complete cycle of brake application including cases where the brake application is partial.

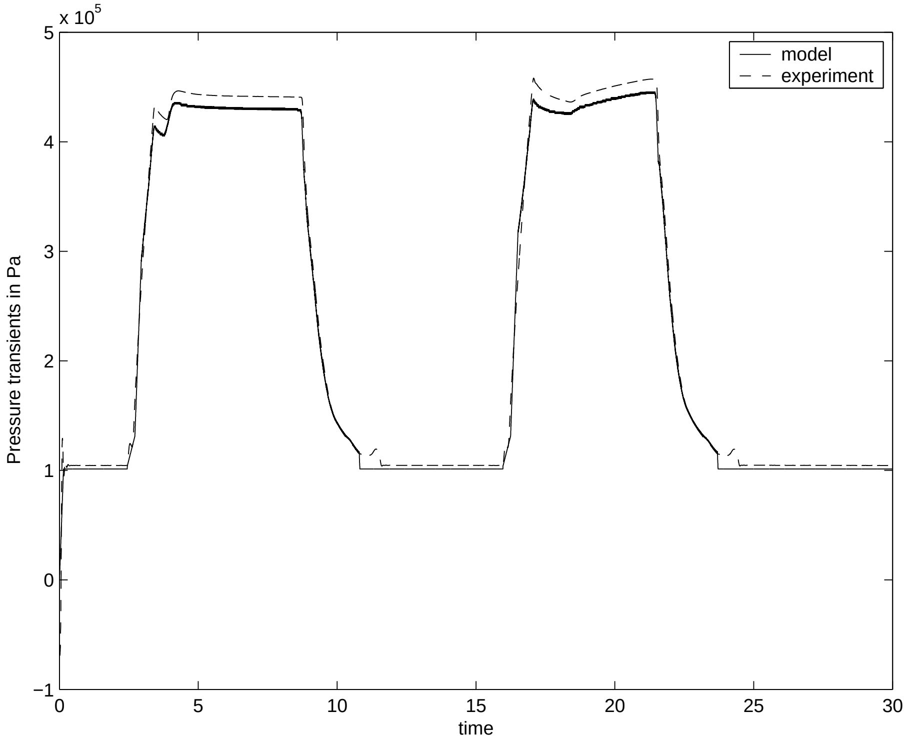

Fig. 27. Pressure transients for a cyclic brake application at a supply pressure of 70 psi(model and experiment)

Event plots corroborate the fact that the model clearly captures the various phases of brake application that were described in the previous chapters. The following chapter presents the sources of error and the scope for future work in the area of air brake diagnostics.

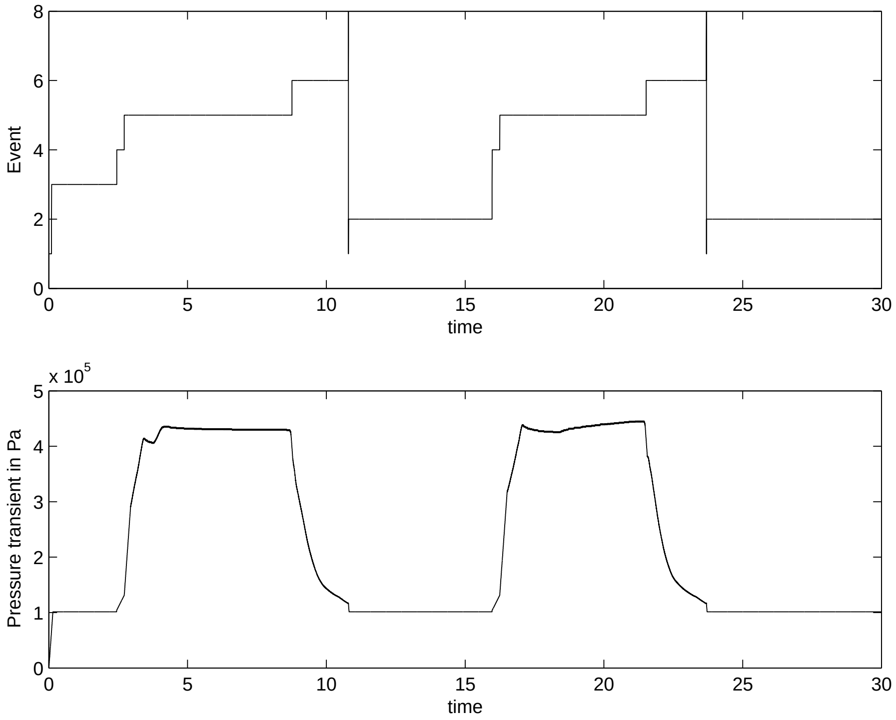

Fig. 28. Pressure transients for a cyclic brake application at a supply pressure of 70 psi with event plot

CHAPTER VI

SUMMARY AND FUTURE WORK

The figures shown in chapter (V) show that the model was able to predict the growth of pressure. The model captures the partial braking and cyclic brake applications at various supply pressures.

A. Sources of error

The model has a number of assumptions that were listed in Chapter III. In addition to this, there are assumptions and issues that deserve a mention and which could be a source of error. They are as follows:

- The valve spring and the brake chamber spring are assumed to behave like a linear spring in the range of operation.
- A lumped parameter approach has been used since a continuum approach is at the present intractable owing to its complexity. By taking such an approach, we convert the governing partial differential equations into ordinary differential equations for which known control techniques are available in literature and control schemes can be developed for tracking a desired pressure evolution in the brake chambers. But this might be a major source of error since we are neglecting spatial variations of pressure in hoses etc..
- The experimental setup for the rear axle does not capture the effects of the slack adjuster while it constrains the push rod stroke to the extent we desire. The setup may not capture the exact behavior of a rear axle. Nonetheless, it is useful in studying the dynamics of pressure evolution in the brake chamber.

- The sensors are assumed to have negligible noise.

B. Future work

Some possible future work related to this area are

- The relay valve was modeled for various pressure inputs delivered by the treadle valve. But the complete rear circuit is left to be simulated.
- A control algorithm for tracking a desired pressure evolution curve. This could be useful in automating the braking process.
- Prediction of faults like leakage in transmission lines resulting in loss of pressure, for the purpose of diagnostics and health monitoring.
- A numerical scheme to predict the push rod stroke.

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APPENDIX A

TABLE OF PARAMETERS

Parameter	Value
A_1	$0.0064\ m^2$
$A\acute{}_1$	$0.0062\ m^2$
A_{hole}	$0.0001704\ m^2$
K_s	$1478.5\ \dfrac{N}{m}$
F_{si}	$34.12\ N$
M_{pp}	$0.0758\ Kg$
K_{br}	$1751.1\ \dfrac{N}{m}$
$xbmax$	$0.0508\ m$
Ab	$0.0194\ m^2$
V_{o1}	$4.91{\times}10^{-4}\ m^3$
V_{o2}	$15{\times}10^{-4}\ m^3$
F_{kbi}	$144.55\ N$
R	$287\ \frac{J}{kgK}$
C_D	0.82
γ	1.4
T_o	$298\ K$
Rx_{pt}	$0.0058\ m$
R_{rp}	$0.0128\ m$

VITA

Shankar Vilayannur Natarajan was born in the city of Coimbatore, located in the Indian state of Tamilnadu. He was brought up in the city of Chennai and completed high school there. He obtained his B.Tech. in civil engineering from the Indian Institute of Technology-Madras. He joined the Mechanical Engineering Department at Texas A&M University in August 2002. Shankar Vilayannur Natarajan may be contacted through Prof. K. R. Rajagopal at the Department of Mechanical Engineering, Texas A&M University, College Station, TX 77843-3123.