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Design and Validation of an Electro-Hydraulic Pressure-Control Valve and Closed-Loop Controller

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DESIGN AND VALIDATION OF AN ELECTRO-HYDRAULIC PRESSURE-CONTROL VALVE AND CLOSED-LOOP CONTROLLER

by

Jerry Boza

A dissertation submitted to the Graduate College in partial fulfillment of the requirements for the degree of Doctor of Philosophy
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DESIGN AND VALIDATION OF AN ELECTRO-HYDRAULIC PRESSURE-CONTROL VALVE WITH CLOSED-LOOP CONTROLLER

Jerry Boza, Ph.D.

Western Michigan University, 2016

Electro-hydraulic pressure-control valves are used in many applications, such as manufacturing equipment, agricultural machinery, and aircrafts to name a few. They are often used to actuate hydraulic clutches, such as those found in power shift transmissions. A traditional pressure-control valve with open-loop control algorithm is typically used in clutch applications. This scheme often results in inconsistent or undesirable system behavior due to the nature of open-loop control as well as the nonlinear system dynamics and uncertainties.

In this research two new electro-hydraulic pressure-control valves were designed in order to decouple the valve and control port (hydraulic) dynamics. This was achieved by removing the regulated pressure balancing force utilized in traditional pressure-control valves. Different closed-loop controllers were designed and tested in parallel in order to achieve the desired steady-state and dynamic regulated pressure response. A nonlinear dynamic model was developed for each valve then used to compare the performance characteristics of the valves. Linear analysis was performed and various control techniques were studied from classical PID control to modern optimal control. The model was also used to predict performance of the closed-loop controllers prior to experimental testing and to validate experimentally tuned controllers afterwards.
Prototype valves were fabricated in order to validate the model and to test the controller designs experimentally. Different valve and controller combinations were compared to a traditional pressure-control valve utilizing open-loop control through typical industry performance tests. This study found that a valve with a traditional pressure-control pilot and a main stage spool with no pressure balancing force, along with a gain scheduled PID controller, outperformed the traditional valve in all areas tested. This approach is also feasible within the existing infrastructure of most applications where the benchmark traditional valve is currently used.
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Next, I would like to thank several of my colleagues for helping me throughout the process, Josh Lambrix for his help designing the valve; Cody Sturgill for assistance with testing and implementing the controllers; and Jeff Huffman for his continued discussions and encouragement throughout the process.

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Jerry Boza
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### NOMENCLATURE

<table>
<thead>
<tr>
<th>Symbol</th>
<th>Description</th>
</tr>
</thead>
<tbody>
<tr>
<td>$A_d$</td>
<td>Damping area</td>
</tr>
<tr>
<td>$P_{a1}$</td>
<td>Damping pressure (opposing solenoid)</td>
</tr>
<tr>
<td>$P_{a2}$</td>
<td>Damping pressure (assisting solenoid)</td>
</tr>
<tr>
<td>$P_c$</td>
<td>Pilot control pressure</td>
</tr>
<tr>
<td>$P_s$</td>
<td>Pilot supply pressure</td>
</tr>
<tr>
<td>$A_n$</td>
<td>Nozzle area</td>
</tr>
<tr>
<td>$Q_{in}$</td>
<td>Flow into control volume</td>
</tr>
<tr>
<td>$Q_{out}$</td>
<td>Flow out of control volume</td>
</tr>
<tr>
<td>$c_{d_{in}}$</td>
<td>Discharge coefficient into volume</td>
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<tr>
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<td>$\rho$</td>
<td>Fluid density</td>
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<tr>
<td>$\mu$</td>
<td>Fluid dynamic viscosity</td>
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<td>$v$</td>
<td>Fluid velocity</td>
</tr>
<tr>
<td>$\beta_{fluid}$</td>
<td>Bulk modulus of fluid</td>
</tr>
<tr>
<td>$\beta_{air}$</td>
<td>Bulk modulus of air</td>
</tr>
<tr>
<td>$\beta_{eff}$</td>
<td>Effective fluid / air bulk modulus</td>
</tr>
<tr>
<td>$V_{air}$</td>
<td>Volume of air in a chamber</td>
</tr>
<tr>
<td>$V_{fluid}$</td>
<td>Volume of fluid in a chamber</td>
</tr>
<tr>
<td>$A_s$</td>
<td>Spool area (pilot side)</td>
</tr>
<tr>
<td>$A_{pb}$</td>
<td>Spool pressure balancing area</td>
</tr>
</tbody>
</table>
\( f_{fric} = \) Friction force
\( C = \) Main control pressure
\( P = \) Main supply pressure
\( \alpha = \) Transitional flow coefficient
\( \gamma = \) Transitional flow coefficient
\( h = \) Radial spool clearance
\( l = \) Engagement length
\( m_1 = \) Pilot mass
\( k_1 = \) Pilot spring rate
\( c_1 = \) Pilot damping coefficient
\( m_2 = \) Main stage mass
\( k_2 = \) Main stage spring rate
\( c_2 = \) Main stage damping coefficient
\( x = \) Displacement
\( V = \) Volume
\( f_{fric} = \) Friction force
CHAPTER 1

INTRODUCTION

1.1 Valve basics and applications

Hydraulic control valves can be divided into three basic categories: directional-control, flow-control, and pressure-control. Directional-control valves are used to connect and isolate hydraulic passages by simply opening and closing a flow path. Flow-control valves allow variable flow rate control to a component. Finally, pressure-control valves regulate variable pressure to a hydraulic component. Electromechanical solenoids are commonly used to actuate these hydraulic control valves, this allows for a flow or pressure for a corresponding input current. This research will focus on electro-hydraulic pressure-control valves.

Power shifting type transmissions have been used in agricultural tractors for over 50 years. These power shift transmissions utilize hydraulic clutches to transmit torque. Since their inception, continuous improvements have been made to increase number of speeds and full torque capability to maintain optimum engine speed and power match in order to improve fuel consumption, noise, and feel [1]. Recently these improvements have been made, in part, through the use of electro-hydraulic pressure-control valves to actuate the clutches. The mechanical elements of the transmission are well established and can be considered mature. So, although there are still improvements being made to these components, there is much more innovation still to come in the electronics and controls of these systems [2]. For each
shift there is typically an on-coming and off-going clutch, the timing of which is critical for shift quality. These clutches are often controlled by electro-hydraulic pressure-control valves in order to achieve this timing. Traditional pressure-control valves utilize pressure feedback on the main spool to achieve a commanded clutch pressure for a corresponding input current. One problem with this approach lies in the physics of the clutch being controlled. In order to actuate this clutch a moving volume must be filled and then pressure modulated, so in essence half of a shift sequence requires a flow-control valve and half requires a pressure-control valve. This is currently achieved using only a pressure-control valve. Another issue is the impact of the pressure balancing force acting on the spool. This balancing force significantly increases the sensitivity of the valve to disturbances in the control port, such as when the clutch is filled and suddenly stops moving. The clutches are typically actuated using open-loop feed forward control algorithms. This is not ideal due to uncertain, time-varying, nonlinear system parameters and the impact of pressure balancing force on valve performance. Both steady state and transient performance can vary significantly due to changes in oil temperature, air entrapment, pressure drop through passages, supply pressure and flow capacity, internal volumes, and performance variation of the valves and clutches.

A significant amount of hardware development is required to achieve desired performance. This occurs both at a system level and within the pressure-control valve actuating the clutch. Often times this requires the addition of expensive components to reduce response time, increase damping, reduce steady-state performance variation, and reduce sensitivity to variation and changes in system parameters. These improvements are constant and finite, and often do not completely address design goals. The system level
performance cannot be optimized through mechanical changes alone; therefore the need for a closed-loop controller exists. For example, the clutch volume is often isolated from the pressure balancing area on the main spool using an orifice in order to reduce the impact of clutch pressure dynamics on valve performance. This orifice is a fixed size and is not optimum for all operating conditions, so it is sized to balance performance requirements across the board. This can be addressed through closed-loop control.

There are many types of pressure-control valves. The benchmark for this research is an electro-hydraulic pilot operated 3/2 proportional pressure-control valve. Figure 1 shows a hydraulic schematic and physical model of the valve. The hydraulic porting is identified as follows: PT is the pilot tank line that directs pilot flow to the reservoir (atmospheric pressure); PS is the pilot supply line which is set upstream using a pressure regulating valve; P is the main stage pressure which is also set using an upstream regulating valve but has more flow capacity available (sometime PS and P are common); C is the main stage control line that communicates pressure / flow to the desired element (in this case a clutch); and T is the main stage tank line that directs main stage flow to the reservoir (atmospheric pressure). The solenoid is typically driven with closed-loop current control. When current is supplied to the solenoid the pilot builds pressure, this pressure acts on the main stage spool which directs flow to and from the clutch and also modulates pressure.
As current to the solenoid increases so does the force output. The armature is opposed by a mechanical spring and regulated pressure acting on a fixed area. As the solenoid force overcomes this mechanical spring rate and pressure balancing force the armature strokes and closes a variable orifice; as this variable orifice is closed off the pressure opposing the solenoid increases resulting in a proportional pressure vs current characteristic. This pilot pressure acts on a spool in the main stage of the valve, the main stage spool then strokes as pilot pressure increases. As the main stage spool strokes, pressure increases in the C port; this is all achieved through opening and closing variable orifices (flow areas). The pressure in the C port acts on a differential area on the spool, providing pressure a pressure balancing force. This pressure balancing force opposes the driving force, in this case the pilot control pressure, resulting in a consistent steady-state output pressure for a given input.
When actuating a clutch with a pressure-control valve there are generally three phases in an open-loop command scheme, the fill, the pressure modulation, and the emptying. During the fill phase the valve is actuated with full command for a short duration, this allows the main stage spool to shift over and provide flow to the clutch. Next, the command is stepped down to an intermediate pressure, below that required to engage the clutch plates. This command is held for a short period then ramped up to full command at various rates depending on the desired torque transfer; this is the pressure modulation phase where the main stage spool is in the center position. Finally, during the emptying phase the valve is de-energized so that the main spool shifts back and opens a path from clutch to tank. Figure 2 shows this process flow in simple block diagram form (a more comprehensive block diagram of the valve can be found in chapter 2). The red arrows show the process flow during each phase. Experimental data from a typical shift sequence is shown in Figure 3.

**Figure 2 Block Diagrams of Clutch Phases**
Development of this shift algorithm and coordination with the off-going clutch are required to achieve a smooth and efficient shift. The fill process is critical and is also a large source of uncertainty due to variables such as fluid temperature, valve and clutch characteristics, and line variations, making this process difficult to control [3]. These variables can lead to overfill or under fill of the on-coming clutch, both of which are undesirable. This is typically controlled through open-loop, event driven, feedback control, or some combination of them.

Valve performance variation is inherent in the design due to component and manufacturing variation. Attempts are made to minimize variation of the critical performance characteristics, such as the steady state pressure at specified currents, the pressure drop from P to C and from C to T at a specified flow, the response time, and the coil resistance. These are
controlled through dimensioning of component tolerances and precision assembly and adjust processes.

1.2 Literature review

The use of control systems for actuating clutches has continuously increased over the last several decades as electrohydraulic pressure-control valves are increasingly implemented into these systems. Electrohydraulic pressure-control valves have many advantages, one being their high power-to-weight ratio. Open-loop control is often used to actuate clutches with pressure-control valves for a variety of reasons, one being the difficulty in developing a closed-loop controller. The highly nonlinear dynamics, extreme variations in system stiffness, and unknown system parameters make controllability difficult. In this section, pertinent research is reviewed, focused on modeling the dynamics of the pressure-control valve, clutch and passages, and current control strategies for electrohydraulic applications.

1.2.1 Dynamics of pressure-control valves

The pressure-control valve is actuated by an electromechanical solenoid. Solenoids are a mature technology and have been used in various applications since the early 1900s. The characteristics, primarily force output, resistance, and inductance, of solenoids can be calculated using magnetic circuit concepts of magnetomotive force, reluctance, and magnetic flux [4]. These techniques as applied to actual solenoid design are well established and have been summarized by Roters [5]. The solenoid characteristics can also be established through solving the three dimensional Maxwell equations. This is best accomplished using a numerical software package due to several factors, such as, the nonlinearity of equations, geometric constraints, magnetic saturation, and eddy current effects. Both of these techniques can
require a significant amount of computational effort. Topcu et al showed that lookup tables could be used to simplify the numerical solutions of electromechanical actuators without loss of accuracy in dynamic simulations [6]. Use of lookup tables also allows for the incorporation of experimental data.

The spool position provides the actual pressure regulation, as it controls the flow area into and out of the control port. Some of the key spool characteristics include leakage, flow rate, and flow forces. Spool leakage for long annular lengths is fully developed, laminar, and follows Poiseuille’s law so it can be calculated analytically from the Navier-Stokes equations [7]. For large spool openings the flow is turbulent and can be calculated using the classic orifice equation. This equation is derived from the Bernoulli equation and incorporates a discharge coefficient, Cd, which is a loss factor [8]. Dong and Ueno showed that the discharge coefficient is a function of the Reynolds number for spool valves and that this could be determined numerically [9]. For high Reynolds number flow at smaller openings they discovered a reattached flow pattern causing the flow coefficient to increase, but only for flows with a Reynolds number less than a critical value. The numerical and experimental flow coefficient and flow force values matched well for several cases. The leakage and flow rate for the transitional spool opening, between fully developed laminar and fully open turbulent, is not as straightforward but is the most critical because this is where pressure modulation occurs. Ferreira et al. used a semi-empirical approach to calculate flow, pressure gain, and leakage [10]. Using a variable equation structure for the area that changed at a critical transition point they were able to match experimental data by tuning the parameters in the analytical equation.
They assumed that valve flow was always turbulent so that the short orifice equation with pseudo-section area functions could be used.

1.2.2 Dynamics of hydraulic conduit and clutch

The passage communicating the pressure-control valve and clutch is typically long and cylindrical as it must travel down a shaft. The pressure drop through this passage must be considered. This will depend on the flow rate, oil properties, geometry and surface, and flow type (laminar or turbulent) according to Munson et al. [7]. This passage can also be subject to hydraulic transients during the fill phase due to sudden changes in state. Deng et al. showed that these transients could be determined analytically for laminar pipeline flow [11]. They formulated the friction factor as a function of the Reynolds Number of the flow. This was accomplished using a separation of variables method, which matched well with the numerical solution obtained by the well-established Method of Characteristics. Taylor et al. developed a method to incorporate the known frequency dependent friction in hydraulic conduits into both the Finite Element Method and the Method of Characteristics [12]. This method was found accurate for both laminar and turbulent flow of incompressible liquids. Soumelidis et al. compared several numerical techniques for modeling hydraulic transients in pipelines [13]. They evaluated the method of characteristics (MOC), finite element method (FEM), transmission line method (TLM), and the rational polynomial transfer function approximation (RPTFA) method. They found the RPTFA model to be the most accurate (when an accurate solution could be obtained), but to also have significantly longer computation times. The MOC model provided the most accurate solutions in short computation times. The TLM model was the most accurate and efficient but was prone to integration problems. Finally, the FEM
method was least accurate and efficient but handled nonlinearities and varying parameters and time steps the best. In real systems, air is present in hydraulic oils to varying degrees and can have an impact on pressure transients. Jiang et al. incorporated the impact of air content and release into a model for pressure and flow transients [14]. The model utilized a genetic algorithm in order to identify the initial air bubble volume in the oil, as well as the air release and re-solution time constants which are unknown in real systems. Overall they were able to show good agreement with experimental data.

The clutch dynamics play a large role in overall system performance. Jiang et al. presented a clutch actuation model with various subsystem models all integrated [15]. They outlined the modeling of the master cylinder, which is actuated hydraulically. The characteristics of interest are the pressure acting on the cylinder, cylinder area, spring rate, friction coefficient, and opposing force. The spring rate consists of both the return spring and the diaphragm spring. The diaphragm spring is nonlinear and hard to represent so a lookup table with experimental data was used. A couple of areas that were not addressed were the flow coefficient, leakage, and fluid compressibility. The flow coefficient is dependent on the valve geometry and oil characteristics as discussed above. Lazar et al. incorporated oil compressibility into their model [16]. They showed analytically how the oil compressibility dictates the break frequency of the clutch chamber. They were able to match experimental data well and ultimately develop a predictive control scheme using the model. The effective compressibility must be considered, this is a combination of the air content, bulk modulus of the oil, and elasticity of the pressure vessel according to Manring [8].
1.2.3 Controls for electro-hydraulic systems

Due to the highly nonlinear dynamics of electro-hydraulic systems the classical control approach has been to linearize these dynamics and use constant gains in a feedback loop [17]. Since this approach is very limited, this has led to the synthesis of robust, adaptive, and predictive controllers. System nonlinearities and uncertainties in electrohydraulic servo systems have most recently been handled through robust control synthesis. Weng et al. developed a Lyapunov-based control algorithm for position control of a hydraulic cylinder using an electrohydraulic flow-control valve [19]. They incorporated flow vs. pressure nonlinearities as well as pressure chamber dynamics through a linear parameter varying (LPV) model. The closed loop cylinder position was asymptotically stable. Milic et al. presented a robust H-infinity state feedback controller for a similar system [17]. They designed a full state robust H-infinity observer to estimate internal states such as spool position. The valve and hydraulic dynamics were modeled and linearized. The linearized coefficients were modeled as parametric uncertainty in a linear fractional transform (LFT) framework. The integral of the signal error was introduced as a new state variable due to the steady state errors caused by disturbances which cannot be eliminated by state feedback gain. Finally they used the bounded real lemma (BRL) to obtain the H-infinity constraint; they guaranteed stability by finding a positive-definite matrix as a solution to the bilinear matrix inequality (BMI) using a general nonlinear transformation. The closed loop system showed good dynamic behavior with robustness to parameter uncertainty and external load disturbances of the cylinder both in nonlinear numerical simulations and in experimental trials.
A generalized approach for the design of predictive controllers for electrohydraulic systems was presented by Jadlovska and Jajcisin [20]. The control algorithm consists of two steps, predictor derivation and computing optimal sequence of control actions. Two input-output models were evaluated, an Auto Regressive Model with External Output (ARX), and a Controlled Auto Regressive Moving Average (CARIMA). The generalized algorithms were designed based on the models such that it was possible to compute the optimal control sequence by the receding horizon principle. They were able to control hydraulic flow between two chambers using both algorithms and found the CARIMA model to be preferable unless the system is noisy, then the ARX model is preferred.

1.2.4 Controls for electro-hydraulic actuated clutch

One difficulty in hydraulic clutch control is the sudden change in hydraulic characteristics. The transition from the fill phase to the pressure modulation phase is stiff and highly nonlinear as the clutch piston abruptly stops. The fill phase of the clutch is more suited for a flow-control valve, but once the clutch piston stops moving a pressure-control valve is required to proportionally transfer torque. Using two valves is not a realistic solution so a pressure-control valve is used for the entire sequence. The transition from the fill phase to the pressure modulation phase is difficult to control due to the sudden change in control port characteristics and the pressure feedback on the spool of the pressure-control valve.

Lazar et al. designed a predictive control scheme for a wet clutch actuated by an electro-hydraulic valve [16, 18]. They developed a CARIMA model for the input-output dynamics of the system, with supply voltage as the input and clutch piston displacement as the output. The solenoid force was modeled as a polynomial and current as a simple RL circuit. Linearized flow
equations with flow coefficients were modeled with all coefficient values for the system parameters determined experimentally. The objective function for the predictor was based on the minimization of tracking error balanced with the minimization of controller output. The dynamic model was accurate when compared to test data. The predictive controller was developed using the validated model and tested in simulations, not experimentally.

Dutta et al. devised a two-step strategy, called learning predictive control, consisting of an optimal reference trajectory to handle system nonlinearities and feedback predictive control to account for time varying system dynamics [21]. The goal was to achieve fast clutch engagement with minimum torque loss, which are conflicting requirements. This is also challenging due to the stiff nonlinearities in the system and lack of sensors. Due to the difficulty in developing the model, they determined a genetic algorithm based optimization would be ideal, but it is a feedforward scheme and not robust to disturbances or uncertainties. Therefore a model based predictive control scheme was also used to track reference pressure. System identification was used to model the filling phase, along with a variable delay based on the amount of oil in the pipeline and the temperature of the oil. Once the clutch piston is engaged they used a feedforward current signal that was optimized through several iterations instead of modeling the clutch due to the linear pressure vs. current relationship. Once the signal was optimized intermediate sensors were used for feedback. The feedback controller was also used to track the optimized profile as it changes due to wear, temperature, etc. The genetic algorithm was used to minimize engagement time and torque loss but was shown experimentally to not be robust. This was addressed with the feedback controller based on testing at two different temperatures.
Horn et al. designed a nonlinear feedforward controller using a flatness approach with a linear PD controller to stabilize the system [22]. They simplified the dynamic model by eliminating the dynamics of the valve piston because they are fast when compared to the clutch piston dynamics. They were able to show that the nonlinear system was flat but also required a PD controller because the open loop system was unstable. Clutch piston displacement was the output and they were able to show accurate trajectory tracking in experimental testing.

Song and Sun were able to design a sliding mode control to achieve robust control and avoid chattering, which is a well-known design issue with sliding mode control [23]. They constructed and validated an electro-hydraulically actuated clutch model and an observer to estimate clutch piston displacement. The design goals were to achieve fill, then provide smooth and precise torque control. They selected pressure feedback in the clutch chamber for 3 reasons, pressure is directly related to torque, it is difficult to package a displacement sensor in the clutch, and it is expensive to measure high resolution displacement in the required range. The pressure feedback approach differs from previous clutch engagement control schemes that require displacement feedback. Slip feedback was another option but cannot be used for the fill phase, while pressure feedback allows for both fill and torque control. They found it challenging to design a nonlinear robust controller due to the nonlinear 2nd order dynamics of the system and the fact that care must be taken in sliding mode control to avoid high gain chattering. Uncertainty bounds of pressure and flow dynamics were determined experimentally. A high slope saturation function and non-conservative uncertainty bounds were used to prevent chattering. A nonlinear observer for clutch displacement was
transformed into a linear observer design problem by incorporating the derivative of the pressure measurement. They showed that the sliding mode robust controller was able to track pressure and that the observer could be used to alleviate high gain demand and diagnose the clutch fill status.

1.3 Problem description

1.3.1 Problem statement

A traditional pressure-control valve with open-loop control algorithm is typically used in clutch applications. This scheme often results in inconsistent or undesirable system behavior due to the nature of open-loop control as well as the nonlinear system dynamics and uncertainties and therefore is the motivation for this research. In this study, a fundamentally different electro-hydraulic pressure-control valve and closed-loop controller are evaluated as an alternative to the traditional pressure-control valve with open-loop control. The new pressure-control valve will have no physical pressure force feedback, but rather measured pressure feedback (closed-loop control). The new valve is designed to reduce the impact of the control port pressure dynamics on the valve performance. The controller is developed in parallel with the valve to minimize steady state error, maintain stability without sacrificing response, and provide robustness to account for system uncertainty. A prototype valve with controller will be tested against an existing pressure-control valve. The difference between the traditional approach and proposed approach are illustrated with a simple block diagram in Figure 4.
1.3.2 Contribution of the present work

Closed loop pressure control systems with traditional pressure-control valves are common in industrial settings where the system parameters are consistent, transient response is not an issue, and only one setpoint is required. This is often achieved using simple PID control. Closed loop control traditional pressure-control valves is not typical in systems where the parameters vary significantly, the entire output range must be controlled, and the transient response is important; such as in a clutch for a Powershift transmission.

The present work is a new approach for closed loop pressure control in high speed, volatile applications such as clutch control. Hardware and software are developed in parallel through analytical modeling and nonlinear numerical simulation, linear analysis, and experimental validation. The main contributions of this research are:

- A comprehensive overview of traditional pressure-control valves and how they interact with systems they are used in, as well as the current state of hydraulic clutch control.
- Development of a mathematical model for various pressure-control valves with experimental validation
- Demonstration of the drawback of traditional pressure-control valves for closed-loop control through linear analysis
- Analysis of the influence of a hydraulic system on the performance of a pressure-control valve
- Evaluation of various control techniques and how they relate to different valve designs
- A novel valve design that can be controlled with simple linear techniques and outperform a traditional pressure-control in various industry driven areas of interest.
CHAPTER 2

MODELING AND DESIGN OF SUBSYSTEMS

2.1 Modeling approach and summary

In this section the various subsystems are identified and modeled individually. The system consists of an electro-hydraulic pressure-control pilot, a main stage spool, hydraulic passages, and a clutch simulator. Each subsystem is also compared to the traditional counterpart to highlight the differences and reasoning for the new approach.

The solenoid geometry, force output, resistance, and inductance of the solenoid for the proposed valve were determined numerically using ANSYS Maxwell electromagnetic field simulation software. The force output was converted to a lookup table as a function of armature position and current for improved implementation into dynamic simulations. The resistance and inductance were modeled as a simple RL circuit with uncertainty bounds for both parameters.

The pilot and main stage spools were modeled as Poiseuille flow when closed and using the classic orifice equation when open. During the transition from fully closed to open a semi-empirical analytical equation was developed. The flow forces were determined using the Reynolds Transport Theorem. Both of these subsystems of the valve were modeled as 2nd order systems with constant spring rate and viscous damping based on the oil properties and clearances of the spools.
The clutch simulator motion was also modeled as a 2\textsuperscript{nd} order system similar to the pilot and main stage spools. The return spring and clutch stiffness once engaged are nonlinear and were incorporated into the model as a lookup table. During pressure modulation of the clutch, the pressure rise rate equation was used; this is based on the volume and change in volume, effective bulk modulus of the clutch volume, and net flow rate. Finally a leak in the clutch was simulated using the classic orifice equation.

2.2 Traditional valve model development

In this section a dynamic model is developed for the traditional pressure-control valve. The pressure-control valve is actuated by an electromechanical solenoid. The characteristics, primarily force output, resistance, and inductance, of solenoids can be calculated using magnetic circuit concepts of magnetomotive force, reluctance, and magnetic flux [4]. These techniques as applied to actual solenoid design are well established and have been summarized by Roters [5]. The solenoid characteristics can also be established through solving the three dimensional Maxwell equations. This is best accomplished using a numerical software package due to several factors, such as, the nonlinearity of equations, geometric constraints, magnetic saturation, and eddy current effects. Both of these techniques can require a significant amount of computational effort. Topcu et al showed that lookup tables could be used to simplify the numerical solutions of electromechanical actuators without loss of accuracy in dynamic simulations [6]. Use of lookup tables also allows for the incorporation of experimental data.

2.2.1 Traditional pilot model

The equation of motion for the pilot armature / flapper assembly in the pilot can be derived from the forces shown in Figure 5. As illustrated in the cross section, the pilot consists
of a coil, working pole, suspension spring, armature / flapper assembly, nozzle, and orifice. There is also a confidential damping feature that is not shown.

![Diagram of a coil, working pole, suspension spring, armature / flapper assembly, nozzle, and orifice.]

**Figure 5 Traditional Pilot: (a) Cross Section and (b) Armature / Flapper Assembly Free Body Diagram**

The solenoid force is a function of current and armature position, the pressure force is the regulated pressure acting on flapper-nozzle area, the damping force is generated from a patented hydraulic damper (PAT# 6281772), and the spring force is from the suspension springs used to adjust the steady state performance of the solenoid. The solenoid can be treated as a simple RL circuit to determine the resulting current for a voltage input, the force characteristics are proprietary to FEMA Corporation and are not shown. The force versus current versus position is proprietary to FEMA Corporation. This results in the following equation of motion for the pilot stage:
\[ \ddot{x} = \frac{F(x, i)}{m} - \frac{kx}{m} - \frac{A_d(P_{d1} - P_{d2})}{m} - \frac{P_c A_n}{m} \]  

(1)

where \( k \) is the spring rate, \( m \) is the armature / flapper assembly mass, \( F(x, i) \) is the solenoid force, \( A_d \) is the damping area, \( P_{d1} \) is the damping pressure opposing the solenoid force, \( P_{d2} \) is the damping pressure assisting the solenoid force, \( P_c \) is the regulated pressure, and \( A_n \) is the nozzle feedback area that the regulated pressure acts against. The flapper position relative to the nozzle dictates the regulated pressure, as it controls the flow into and out of the control port volume. Figure 6 shows the pressure versus flapper-nozzle gap for a specific configuration. The cross section is also represented schematically, where \( P_s \) is the constant supply pressure, \( P_c \) is the regulated control pressure, and \( P_t \) is the tank pressure. In this configuration there is a fixed orifice between \( P_s \) and \( P_c \), and a variable orifice (the flapper-nozzle) between \( P_c \) and \( P_t \). As the flapper-nozzle gap closes down the variable orifice area decreases linearly and \( P_c \) pressure increases accordingly. This is the steady-state pressure versus position for the traditional pressure-control pilot.
The regulated pressure $P_c$ can be calculated for any configuration using the short orifice equation, derived from the Bernoulli equation, for each of the orifices and the pressure rise rate equation for the control port volume [8]. This results in the following system of equations:

$$Q_{in} = a_{in} cd_{in} \text{sign}(P_s - P_c) \frac{2}{\sqrt{\rho}} |P_s - P_c|$$  \hspace{1cm} (2)

$$Q_{out} = a_{out} cd_{out} \text{sign}(P_c - P_t) \frac{2}{\sqrt{\rho}} |P_c - P_t|$$  \hspace{1cm} (3)

$$\dot{P}_c = \frac{\beta_{eff}}{V_c} (Q_{in} - Q_{out} - \dot{V}_c)$$  \hspace{1cm} (4)

where $Q_{in}$, $a_{in}$, and $cd_{in}$ are the flow, area, and discharge coefficient for the fixed orifice (into the regulated pressure volume); $Q_{out}$, $a_{out}$, and $cd_{out}$ are the flow, area, and discharge coefficient for the variable orifice (out of the regulated pressure volume); $V_c$ is the volume of
the control port; \( \rho \) is the fluid density; and \( \beta_{eff} \) is the effective bulk modulus of the control port volume. The discharge coefficient of the orifice in Eq. (2) is dependent on the length to diameter ratio as well as the Reynolds’s number and can be determined experimentally [5]. The Reynolds’s number for flow through an orifice can be calculated as:

\[
Re = \frac{\rho v d_h}{\mu} = \frac{\rho Q d_o}{A \mu} = \frac{4 \rho Q}{\pi d_o \mu}
\]

where \( v \) is the fluid velocity, \( d_h \) is the hydraulic diameter, \( d_o \) is the orifice diameter, \( \mu \) is the dynamic viscosity of the fluid, \( Q \) is the flow rate, and \( A \) is the orifice area. This approach was extended to the flapper-nozzle as follows:

\[
Re = \frac{\rho v d_h}{\mu} = \frac{\rho Q d_h}{A \mu} = \frac{\rho Q \left( \frac{4 \pi d x}{2x + 2 \pi d_n} \right)}{\pi d x \mu} = \frac{\rho Q d_n}{2 \mu x (x + \pi d_n)}
\]

where \( d_n \) is the nozzle diameter and \( x \) is the flapper-nozzle gap. The effective bulk modulus can be calculated using the following relationship [4]:

\[
\frac{1}{\beta_{eff}} = \frac{1}{\beta_{fluid}} + \frac{V_{air}}{V_{fluid}} \cdot \frac{1}{\beta_{air}}
\]

where \( \beta_{fluid} \) is the bulk modulus of the fluid, \( V_{air} \) is the volume of air trapped in the fluid, \( V_{fluid} \) is the volume of fluid, and \( \beta_{air} \) is the bulk modulus of the air. In some instances modulus of the pressure vessel should also be considered, such as with a rubber hose, however in this instance the vessel is significantly stiffer and not included.

### 2.2.2 Traditional main stage model development

The equation of motion for the main stage spool can be derived from the forces shown in Figure 7. As illustrated in the cross section, the main stage consists of a spool, cartridge, and spring.
The pressure regulated by the pilot acts on the face indicated in Figure 7, the regulated pressure, acts on the differential area in the Pc port, the spring acts on the opposite end as the pilot pressure, the viscous force is a result of the spool / bore clearance, and finally a variable amount of mechanical friction is present due to hydraulic side loading and component tolerances. This results in the following equation of motion:

\[
\ddot{x} = \frac{P_c A_s}{m} - \frac{c \dot{x}}{m} - \frac{k x + f_{pl}}{m} - \frac{C A_{pb}}{m} - \frac{f_{fric}}{m} + \frac{f_{pl}}{m}
\]  

(8)

where \( A_s \) is the area of the spool that the regulated pilot pressure acts against, \( C \) is the regulated main stage pressure, \( A_{pb} \) is the regulated pressure balancing force area, \( f_{fric} \) is the
mechanical friction force, $c$ is the viscous damping coefficient, and $f_{pl}$ is the spring load at equilibrium. The viscous damping coefficient can be calculated as follows [24]:

$$c = \frac{\mu}{h} \pi d_s (l - x)$$

where $h$ is the radial clearance between the spool and bore, $d_s$ is the spool diameter, and $l$ is the engagement length at zero displacement. Equation 8 is only valid for blocked control ports with constant volume; therefore flow forces are not included.

The main stage spool is schematically similar to the flapper nozzle, except there are two variable orifices; therefore Eqs. (2) - (4) also apply to the main stage spool, except $a_{in}$ and $cd_{in}$ also vary as a function of spool displacement. The area versus stroke is more complicated for the spool than for the flapper-nozzle. When the spool is de-actuated, the P to C flow path is a long annulus and the C to T flow path is an open area; when the spool is fully actuated the opposite is true; and when the spool is regulating pressure to C there are short annulus paths from P to C and from C to T.

Spool leakage for long annular passages is fully developed, laminar, and follows Poiseuille’s law so it can be calculated analytically from the Navier-Stokes equations [7]. For large spool openings the flow is turbulent and can be calculated using the classic orifice equation. Dong and Ueno showed that the discharge coefficient is a function of the Reynolds number for spool valves and that this could be determined numerically [9]. The leakage and flow rate for the transitional spool opening, between fully developed laminar and fully open turbulent, is not as straight forward but is the most critical because this is where pressure modulation occurs. A semi-empirical approach to calculate flow, pressure gain, and leakage can be used to develop a variable equation structure for the transitional region by matching
experimental data and tuning the parameters in the analytical equation [10]. Equations (10) - (12) are used to calculate the annular, transitional, and open area regions of the flow versus displacement characteristic.

\[ Q_a = \frac{\pi (d_s + 2h)h^3(P - C)}{12\mu (x_{pc} - x)} \quad (10) \]

\[ Q_t = \alpha e^{rx} \text{sign}(P - C) \frac{2}{\sqrt{\rho}} |P - C| \quad (11) \]

\[ Q_o = \pi d_s x cd_{out} \text{sign}(P - C) \frac{2}{\sqrt{\rho}} |P - C| \quad (12) \]

where \( Q_a \) is the annulus flow, \( x_{pc} \) is the engagement length at zero displacement, \( Q_t \) is the transitional flow \( \alpha \) and \( \gamma \) are experimentally tuned coefficients, and \( Q_o \) is the open area flow. These three equations make up the variable structure equation for spool flow or can alternatively be used to develop a flow area versus displacement lookup table for computational purposes. This is illustrated graphically in Figure 8 for the P to C variable orifice in a typical spool.
2.2.3 Combining traditional pilot and main stage models

The final valve model can be represented as a system of first order differential equations by combining the models above. Due to the pressure balancing force on the pilot, an orifice must be placed between regulated pilot pressure and the main stage spool to damp the flow disturbances from the main stage. This can be modeled using Eq. (2) for the flow through the orifice and Eq. (4) for the pressure dynamics on each side of the orifice. Figure 9 shows a schematic representation of the valve. Combining the equations outlined above and converting to state space format yields the following system of differential equations:
\[
\begin{align*}
\dot{x}_1 &= x_2 \\
\dot{x}_2 &= -\frac{k_1 x_1}{m_1} - \frac{A_d (x_3 - x_4)}{m_1} - \frac{A_n x_6}{m_1} + \frac{F_1}{m_1} \\
\dot{x}_3 &= \frac{\beta_{\text{eff}}}{V_{3_o} - A_d x_1} [Q_{34} - Q_3 + A_d x_2] \\
\dot{x}_4 &= \frac{\beta_{\text{eff}}}{V_{4_o} + A_d x_1} [Q_{34} - A_d x_2] \\
\dot{x}_5 &= -\frac{R}{L} x_5 + \frac{1}{L} u \\
\dot{x}_6 &= \frac{\beta_{\text{eff}}}{V_6} [Q_{p6} - Q_6 - Q_{67}] \\
\dot{x}_7 &= \frac{\beta_{\text{eff}}}{V_{7_o} + A_s x_8} [Q_{67} - A_s x_9] \\
\dot{x}_8 &= x_9 \\
\dot{x}_9 &= -\frac{k_2 x_8}{m_2} - \frac{c_2 x_9}{m_2} - \frac{A_{fb} x_{10}}{m_2} + \frac{A_s x_7}{m_2} - \frac{f_{\text{fric}} + f_{\text{pl}}}{m_2} \\
\dot{x}_{10} &= \frac{\beta_{\text{eff}}}{V_{10}} [Q_{p10} - Q_{10}]
\end{align*}
\] 

where \( m_1 \) is the pilot armature / flapper mass, \( k_1 \) is the pilot spring rate, \( F_1 \) is the solenoid force output, \( m_2 \) is the main spool mass, \( k_2 \) is the main stage spring rate, \( c_2 \) is the main stage viscous damping coefficient, \( R \) is the solenoid resistance, \( L \) is the solenoid inductance, and the state variables are defined in Table 1. The subscripts on flow and volume refer to the associated state variables, for example \( Q_{67} \) is the flow between \( x_6 \) and \( x_7 \) pressure volumes.
Table 1 State Variable Description for Traditional Valve

<table>
<thead>
<tr>
<th>State Variable</th>
<th>Description of Variable</th>
</tr>
</thead>
<tbody>
<tr>
<td>$x_1$</td>
<td>Pilot armature / flapper assembly displacement</td>
</tr>
<tr>
<td>$x_2$</td>
<td>Pilot armature / flapper assembly velocity</td>
</tr>
<tr>
<td>$x_3$</td>
<td>Pilot damping pressure on nozzle side</td>
</tr>
<tr>
<td>$x_4$</td>
<td>Pilot damping pressure on solenoid side</td>
</tr>
<tr>
<td>$x_5$</td>
<td>Current in solenoid</td>
</tr>
<tr>
<td>$x_6$</td>
<td>Regulated pilot pressure on pilot side</td>
</tr>
<tr>
<td>$x_7$</td>
<td>Regulated pilot pressure on main spool side</td>
</tr>
<tr>
<td>$x_8$</td>
<td>Main spool displacement</td>
</tr>
<tr>
<td>$x_9$</td>
<td>Main spool velocity</td>
</tr>
<tr>
<td>$x_{10}$</td>
<td>Regulated main stage pressure</td>
</tr>
<tr>
<td>$u$</td>
<td>Voltage to the solenoid</td>
</tr>
<tr>
<td>$y$</td>
<td>Regulated main stage pressure</td>
</tr>
</tbody>
</table>
Several of the equations are coupled, such as the last two for example. The second to last equation is the equation of motion of the main spool and the last equation is the pressure response of the control volume. These are coupled through the spool position and regulated pressure. The physical meaning of this is important; it shows that the spool position impacts the pressure dynamics and the pressure dynamics impact the spool position. The result of this coupling is that the transient performance of the traditional valve is highly dependent on the properties of the control port volume. This is a key point that is revisited later; however this can also be illustrated through the block diagram of the valve. Due to the number of components this system is broken into three subsystems; the first subsystem is the pilot; the second subsystem is the damping chamber of the pilot; and the third subsystem is the main stage. Figure 10 shows the block diagram representation of the traditional pilot. The regulated pilot pressure balancing force is highlighted in red, this shows the interdependence of the regulated pressure and the spool dynamics. Also of note is the interaction of system states with the main stage and the damping chamber of the pilot valve.
Due to the dynamic characteristics of typical pressure-control valves it is often necessary to add hydraulic damping. This damping is often dependent on the spool dynamics and oil properties as described previously. Figure 11 shows a block diagram of the damping chamber for a typical pressure-control pilot. Studying Figure 10 and Figure 11 it is clear that the two are dependent on each other.
The final subsystem in the traditional pressure-control valve is the main stage, which regulates pressure to the desired hydraulic component. Figure 12 shows the block diagram representation of the traditional pressure-control valve main stage. Once again, the regulated pressure balancing force is highlighted in red. This block diagram is valid for a fixed, or blocked, control port. This best represents the characteristics of the valve by itself. However it is clear that the valve performance is dependent on the control port characteristics, and therefore dependent on the hydraulic component being controlled, demonstrating the objective of this study to eliminate the impact of the regulated pressure dynamics on the valve performance making it more conducive to future closed-loop control.
2.3 Traditional pilot with proposed main stage model development

The first proposed valve utilizes the same pilot as the traditional valve, but a different main stage spool with no pressure balancing force. Schematically this is very similar to the traditional valve minus the pressure balancing effect of C as shown in Figure 13.
The traditional and proposed spool function the same hydraulically, regulated pressure increases as C to T closes off and P to C opens up, however the pressure versus displacement characteristic is different. In the traditional spool, the pressure modulation occurs over a small range of the total spool displacement, while in the proposed spool the modulation occurs over a significantly larger range. Figure 14 shows the typical steady state pressure versus spool displacement for both spools.

![Figure 14 Traditional and Proposed Spool Pressure Versus Displacement](image)

As with the traditional spool, the pressure versus flow versus displacement characteristics can be determined using Eqs. (10) - (12). These can be used with Eq. (4) to calculate the regulated pressure dynamics. The equation of motion is similar to the traditional spool, minus the regulated pressure balancing force:
\[
\ddot{x} = \frac{P_c A_s}{m} - \frac{c \dot{x}}{m} - \frac{k \dot{x}}{m} - \frac{f_{fric} + f_{pl}}{m}
\]  

where the spring rate \( k \) is much higher than in the traditional valve since there is no pressure balancing force opposing the regulated pilot pressure. The resulting state space representation of the valve is also similar, minus the main stage pressure balancing force as shown below, with the same state variables outlined in Table 1:

\[
\begin{align*}
\dot{x}_1 &= x_2 \\
\dot{x}_2 &= -\frac{k_1 x_1}{m_1} - \frac{A_d (x_3 - x_4)}{m_1} - \frac{A_n x_6}{m_1} + \frac{F1}{m_1} \\
\dot{x}_3 &= \frac{\beta_{eff}}{V3_o - A_d x_1} [Q_{34} - Q_3 + A_d x_2] \\
\dot{x}_4 &= \frac{\beta_{eff}}{V4_o + A_d x_1} [Q_{34} - A_d x_2] \\
\dot{x}_5 &= -\frac{R}{L} \cdot x_5 + \frac{1}{L} \cdot u \\
\dot{x}_6 &= \frac{\beta_{eff}}{V6} [Q_{p6} - Q_6 - Q_{67}] \\
\dot{x}_7 &= \frac{\beta_{eff}}{V7_o + A_s x_8} [Q_{67} - A_s x_9] \\
\dot{x}_8 &= x_9 \\
\dot{x}_9 &= -\frac{k_2 x_8}{m_2} - \frac{c_2 x_9}{m_2} + \frac{A_s x_7}{m_2} - \frac{f_{fric} - f_{pl}}{m_2} \\
\dot{x}_{10} &= \frac{\beta_{eff}}{V10} [Q_{p10} - Q_{10}]
\end{align*}
\]

The pilot is identical to the traditional valve and the state variables of the main stage are as well. The block diagram is very similar to that of the traditional valve except for the
regulated pressure balancing force shown in red in Figure 12. This is shown in Figure 15, further illustrating the decoupling of the regulated pressure dynamics from the spool dynamics.

![Proposed Main Stage Block Diagram](image)

**Figure 15 Proposed Main Stage Block Diagram**

### 2.4 Proposed pilot with proposed main stage model development

The second proposed valve concept utilizes a pilot and main stage with no pressure balancing force. This valve is shown schematically in Figure 16. Notice that the pilot hydraulics are schematically equivalent to the main stage (except smaller in physical size). There also is no damping orifice between the pilot and main stage and no damping in the pilot. This is due to the elimination of the pressure balancing force on the pilot, which will be discussed in more detail later.
The model structure is similar to the previous two valves; however there are less state variables due to the elimination of the damping features. Also the equation of motion for the pilot takes the same form as the proposed main stage. The resulting state space representation for this valve is:

\[
\begin{align*}
\dot{x}_1 &= x_2 \\
\dot{x}_2 &= -\frac{k_1 x_1}{m_1} - \frac{c_1 x_2}{m_1} + \frac{F_1}{m_1} - f_{fric1} - f_{pl1} \\
\dot{x}_3 &= \frac{\beta_{eff}}{V_3} [Q_{p3} - Q_3 - A_s x_6] \\
\dot{x}_4 &= -\frac{R}{L} \cdot x_4 + \frac{1}{L} \cdot u \\
\dot{x}_5 &= x_6 \\
\dot{x}_6 &= -\frac{k_2 x_5}{m_2} - \frac{c_2 x_6}{m_2} + \frac{A_s x_3}{m_2} - f_{fric2} - f_{pl2} \\
\dot{x}_7 &= \frac{\beta_{eff}}{V_7} \cdot [Q_{p7} - Q_7]
\end{align*}
\]

where the variables are the same as before however the subscripts have been updated accordingly for the reduction of state variables. The state variables are summarized in Table 2 for this valve.
This valve concept is further decoupled, where the regulated pilot pressure, $x_3$, and the main spool dynamics, $x_5$ are the only coupled pressure and position states. There are no additional damping features required as a result of this, resulting in a more simple model with less system interdependence. Figure 17 is a block diagram of this new pilot, notice the reduction of states and interaction with pressure dynamics.
The main stage dynamics are identical to the valve presented in section 2.3, however the state variables are labeled different due the reduction in overall state variables. This is shown in Figure 18. This further illustrates the objective of this valve concept, as there is less interaction with the pressure dynamics, and therefore the system characteristics. Also notice that the damping orifice between the pilot and main stage is no longer necessary because the pilot spool dynamics are no longer impacted by the pressure dynamics in the head end of the main stage spool. The regulated pilot pressure dynamics are still dependent on the pilot spool position and oil characteristics however.
Figure 18 Proposed Main Stage Block Diagram
CHAPTER 3

NUMERICAL SIMULATIONS AND EXPERIMENTAL VALIDATION

3.1 Nonlinear numerical AMESim model

Differential equations for hydraulic pressure dynamics, such as Eq. (6), are typically stiff due to the low compressibility of the fluid [9]. The stiffness can also change drastically due to changes in volume or fluid aeration. This makes the solution of these equations difficult so a model was developed in AMESim for each valve. This software has different solution algorithms to handle such discontinuities and stiff equations allowing for faster simulations. The AMESim model for the proposed pilot with proposed main stage is shown in Figure 19 (the traditional valve model is shown in Figure 38 for reference). This model was used to develop the appropriate valve characteristics such as spring rates and spool geometry. Once an acceptable design was achieved the models were validated experimentally.
3.2 Baseline nonlinear simulations

The steady-state and transient regulated pressure response for the valves are compared in this section. These nonlinear simulations highlight some of the key performance differences between the valves. Figure 20 shows the steady-state pressure versus current characteristic for the valves under the conditions outlined in Table 3. Notice the traditional valve is more linear and there is more current span when compared to the proposed valves. This is important in open-loop applications and is an appealing characteristic of the traditional pressure-control valve, the steady state input-output relationship.
Table 3 Baseline Model Parameters

<table>
<thead>
<tr>
<th>Parameter</th>
<th>Value</th>
</tr>
</thead>
<tbody>
<tr>
<td>Supply pressure</td>
<td>20 bar</td>
</tr>
<tr>
<td>Control port volume</td>
<td>10 mL</td>
</tr>
<tr>
<td>Oil temperature</td>
<td>40°C</td>
</tr>
<tr>
<td>Oil density</td>
<td>865 kg/m³</td>
</tr>
<tr>
<td>Oil viscosity</td>
<td>79 cSt</td>
</tr>
<tr>
<td>Oil bulk modulus</td>
<td>14072 bar</td>
</tr>
<tr>
<td>Oil air content</td>
<td>1%</td>
</tr>
</tbody>
</table>

Figure 20 Steady-state Pressure Versus Current Comparison
The regulated pressure balancing force acting on the traditional spool is beneficial for the steady-state response as shown in Figure 20, however it can be detrimental to the transient response. This can be altered through design changes in the valve such as internal damping orifice sizes, solenoid force characteristics, mechanical spring rates, and spool geometry. However the dynamics will always change significantly under different operating conditions due to the interaction between the valve and the control port volume where pressure is regulated. Since the physical features that impact the transient response of the valve are fixed, they can only be discretely tuned for specific performance characteristics, under specific conditions, at specific operating points. So, for example, a damping orifice that can be sized to achieve a desired response in one system, may not achieve that response in a different system, or more importantly, in the same system under different operating conditions. Two of the most significant system parameters are oil temperature and aeration. This is illustrated through an example in Figure 21, which shows the step response of the traditional valve with different oil temperatures. This range of oil temperature is well within the variation of typical applications, therefore this performance variation can be problematic when trying to regulate pressure to a hydraulic component. In all step response simulations, the current to the solenoid was stepped on at 0.01s, in order to allow the model to stabilize prior to stepping the valves on.
The step response for the valve with the traditional pilot and proposed main stage under the same conditions is shown in Figure 22. Notice the transient response is more similar at the different oil temperatures, however the steady-state response changes (the same input current was supplied for both trials). The same is true for the proposed pilot with proposed main stage shown in Figure 23. Both responses are asymptotically stable, compared to the traditional valve which varies depending on the oil conditions. The steady-state response of the proposed valves varies significantly with oil temperature due to the lack of the pressure balancing force, whereas the traditional valve shows much less variation in steady-state response (when stable).
Figure 22 Traditional Pilot with Proposed Main Stage Valve Step Response at Different Temperatures

Figure 23 Proposed Pilot with Proposed Main Stage Valve Step Response at Different Temperatures
Oil temperature has a significant impact on valve performance as shown in Figures 21 - 23. Temperature is typically monitored at a single location in most hydraulic applications. The temperature will change throughout the hydraulic circuit based on many factors, including but not limited to: ambient temperature, flow rate through the different passages, usage of the different valves or actuators, pump reservoir and temperature control characteristics. This leads to uncertainty of the temperature of the oil in the volume where the pressure is regulated. The oil viscosity, density, and bulk modulus vary as a function of temperature, and the bulk modulus varies significantly with air content as shown in equation (7). These oil properties impact the pressure-flow characteristics of the various passages and also dictate the pressure dynamics, along with the bulk modulus, as shown in equation (4). This volume has the largest impact on the transient response of the traditional pressure control valve due to the pressure balancing force acting on the spool.

Another significant, and more uncertain, system characteristic is the aeration of the hydraulic fluid. Air can enter the hydraulic system through the pump’s inlet or through imperfections in seals, fittings or other unions. Air leaks in a hydraulic system can lead to aeration of the fluid, which drastically changes the fluid properties, primarily the bulk modulus and how it varies with pressure. The effective bulk modulus of a fluid can have a large impact on the pressure dynamics according to equation (4). The step response of the valves with varying levels of air in the oil was studied. Figure 24 shows the response of the traditional valve with 0% air and 1% air. No references were found documenting typical air percentages in hydraulic systems; however previous informal studies at FEMA and various customer applications have shown this to be typically 1% and up to 4% air in oil samples.
The transient response of the proposed valves were less sensitive to the same air variation than the traditional valve, however there was a larger time lag as air percentage increased due to the reduced effective bulk modulus of the regulated control volume. The traditional valve compensates for this lag due to the pressure balancing force, the spool continues to stroke until the force balance is satisfied allowing for more flow into the control volume resulting in a faster pressure rise. Figure 25 shows the step response of the traditional pilot with proposed main stage, and Figure 26 shows the same for the proposed pilot with proposed main stage. The increased lag in Figure 26 is a result of the lack of pressure balancing force on both pilot and main stage, when compared to the traditional pilot with main stage where the pilot compensates as discussed above.
Figure 25 Traditional Pilot with Proposed Main Stage Valve Step Response with Different Oil Aeration

Figure 26 Proposed Pilot with Proposed Main Stage Valve Step Response with Different Oil Aeration
3.3 Experimental validation of model

The baseline simulations supported the initial hypothesis that eliminating the pressure balancing force would reduce the sensitivity of the transient response to the regulated control volume characteristics and increase the sensitivity of the steady state response. This led to the assembly of prototypes in order to validate the model simulation results. The steady-state pressure versus current characteristic and 10bar step response were used to validate the accuracy of the model.

3.3.1 Experimental test setup

There are two basic components of the experimental setup, the hydraulic and the electrical. They hydraulic schematic is shown in Figure 27 with a picture of the setup in Figure 27. The hydraulic components are all labeled in Figure 28 except the pump is not shown.

Figure 27 Hydraulic Schematic
The electrical schematic for the test setup is shown in Figure 29. The current can be controlled to the valve through an external voltage source or through a potentiometer. This current can be measured with an Ammeter or as a voltage drop across a 1Ω resistor for data acquisition purposes. The actual current driver displayed as a black box in Figure 28 is shown schematically in Figure 30. The voltage input corresponds to a controlled current output to the valve.
The actual hardware is pictured in Figure 31. Everything is housed in the TN: 28080 box with switches to power the unit on or off and to toggle between external voltage control or the potentiometer on the box for manual control. The ammeter is shown in the background, and
not shown are the output for the voltage across the 1Ω resistor and the input for an external voltage source.

3.3.1 Experimental open loop response results

The steady-state and transient pressure response was measured for each valve and compared to the numerical model. Figure 32 shows the experimental and numerical steady-state pressure versus current for the traditional pressure-control valve. Since there is valve to valve variation, some of the model parameters had to be adjusted to match the experimental test piece. The initial airgap of the pilot solenoid and the position of the pilot flapper relative to the pilot nozzle were adjusted to achieve the proper slope of the pressure versus current curve. The main stage spring preload was adjusted to achieve the correct pressure calibration. Finally the magnetic hysteresis and spool friction were adjusted to achieve the correct separation between the rising and falling curves. In Figure 32, the right side of the curve is the pressure
output when current is ramped from 0 to 1A and the left side is when the current is ramped from 1A to 0A.

![Graph showing pressure versus current with two lines: one for increasing current and one for decreasing current.](image)

**Figure 32 Traditional Valve Pressure versus Current Validation**

The 10bar step response was also tested at different temperatures to validate the transient characteristics of the model. The percentage of air in the oil was not varied however because that is not a practical parameter to change or monitor in the test setup. Instead the experimental data was compared to the various air levels in the model as a means to estimate how much air was in the oil. Figure 33 shows the experimental step response of the traditional pressure control valve at different oil temperatures. These results match closely with the numerical step response shown in Figure 21. This indicates that the model dynamics are correct and that the experimental test stand oil has approximately 1% of air in it, which is consistent with previous tests discussed earlier.
The same performance characteristics were measured on the proposed valves and compared the corresponding models. Figure 34 shows the pressure versus current validation for the traditional pilot with proposed main stage. As with the traditional valve, the model was adjusted to match the experimental test piece.
Figure 35 shows the 10bar step response validation for the traditional pilot with proposed main stage. In Figure 35, the numerical and experimental data is shown; the numerical response was not displayed in Figure 33 for the traditional valve because it was too cluttered making it difficult to view the data. The experimental transient response matched the model well, further indicating that the test setup has approximately 1% of air in the test oil.
Finally, the proposed pilot with proposed main stage model was validated experimentally. Figure 36 shows the pressure versus current characteristic of the model compared to the experimental test piece and Figure 37 shows the step response comparison. Notice the steady-state pressure versus current characteristic is steeper than the traditional and the traditional pilot with proposed main stage, and the difference between increasing and decreasing characteristics is wider.
Figure 36 Proposed Pilot with Proposed Main Stage Pressure versus Current Validation

Figure 37 Proposed Pilot with Proposed Main Stage Step Response Validation
3.4 Open loop disturbance rejection simulations

Once the numerical models of the different valves were validated experimentally, some additional simulations were run to validate the impact of eliminating the pressure balancing force on external disturbance rejection. In a typical pressure-control application, external system disturbances often come in the form of flow into or out of the regulated control volume. This can be due to unexpected changes in the volume, such as a larger control spool or a clutch piston moving from external forces, or due to leakage into our out of the volume. This can be studied by modifying equation (4) to include a disturbance term as follows:

\[ \dot{p}_c = \frac{\beta_{eff}}{V_c} (Q_{in} - Q_{out} - V_c + \varphi_d) \quad (17) \]

where \( \varphi_d \) is an unknown external flow disturbance in the regulated control port volume. This is most significant in the main stage due to the relative size of the spool compared to hydraulic component to which pressure is regulated. This is also present in the pilot control volume, so external main stage flow disturbances are transmitted back to the pilot in the same manner. However, when the traditional pilot is combined with the proposed main stage spool, the external hydraulic disturbances are eliminated and therefore do not transmit back through the pilot. So, although the regulated pilot control volume is still moving, it is no longer impacted by the external flow disturbances discussed above, resulting in a more stable system with less sensitivity to the regulated control port. This was simulated in the AMESim model by introducing a flow disturbance to the main stage control port volume as shown in Figure 38 for the traditional pressure control valve model.
A 2LPM impulse flow disturbance was introduced to the main stage control port for both the traditional and proposed pressure-control valves under the system conditions outlined in Table 3. The regulated control pressure response to the impulse disturbance is shown in Figure 39. Both of the proposed valves have the same response because the main stage spool dynamics are not impacted by the regulated pressured dynamics, therefore the response is simply due to the regulated control port volume characteristics and the flow passages to and from this volume, which are the same for both valves. The traditional pressure-control spool dynamics are impacted by the regulated pressure dynamics and this is apparent in the impulse response.
3.5 Frequency response simulations

Frequency response was also evaluated using the experimentally validated model. This was evaluated through a manually constructed discrete frequency response plot using the nonlinear model. This method was used as an alternative to traditional Bode analysis due to the varying nonlinear system characteristics. This was accomplished by applying a sine wave voltage input (scaled by the DC gain at the operating point of interest) to the solenoid at discrete frequencies, measuring the amplitude and phase difference, and building a frequency response plot. An example of this is shown in Figure 40 for the proposed valve, where a 1 Hz sinusoidal voltage input with 0.05V amplitude (1bar equivalent output pressure using DC gain) was applied and regulated pressure was the output. Figure 41 shows the same response for a 10Hz sinusoidal input, notice the response is moving out of phase and starting to attenuate.
Figure 40 Nonlinear Simulation of 1Hz Sinusoidal Voltage Input and Pressure Output

Figure 41 Nonlinear Simulation of 10Hz Sinusoidal Voltage Input and Pressure Output
This process was completed at various frequencies up to 1000Hz. The amplitude and phase shift were calculated at each frequency. The amplitudes were converted to decibels and then compiled across the frequency range as shown in Figure 42 for the traditional and proposed valve. The phase shifts were converted to degrees and then compiled across the frequency range as shown in Figure 43.

The proposed valve has a higher gain at lower frequencies and damped system characteristics while the traditional valve shows resonance at 200Hz. Both valves have similar open-loop bandwidth frequency around 500Hz and phase crossover frequency between 150Hz and 200Hz, but the proposed valve has a higher slope on the high frequency asymptote.

![Figure 42 DiscreteMagnitude versus Frequency of Traditional and Proposed Valves](image-url)
Figure 43 Discrete Phase Angle versus Frequency of Traditional and Proposed Valve
CHAPTER 4

LINEAR ANALYSIS AND INITIAL CONTROLLER DESIGN

4.1 Model linearization

The ultimate goal of this research is to develop a pressure-control valve that is better suited for closed loop control than the traditional pressure-control valve, and then follow this up with the development of the controller. Due to the highly nonlinear dynamics of electro-hydraulic systems the typical control approach has been to linearize these dynamics at various operating points and design controller gains for closed loop control [17]. This approach is limited but allows for the use of a wide array of well-established linear control techniques and therefore it is a natural starting point prior to attempting more recent nonlinear control techniques. The different valves have already been modeled above as systems of coupled first-order differential equations of form

\[
\dot{x}(t) = h(x(t), u(t), t)
\]

in equations 13, 15, and 16. The valves can then be studied at different operating points as linear time invariant state-space systems:

\[
\dot{x}(t) = A x(t) + Bu(t)
\]

at various operating points where the A and B matrices are the Jacobians of \( h(x_o(t), u_o(t), t) \) with respect to \( x \) and \( u \) respectively [25]. The valve models were initially linearized at 10bar under the temperature and aeration conditions evaluated thus far. The dominant poles were then studied as the system parameters varied. Figure 44 shows the impact that these system
parameters have on the pole locations of the traditional valve. Notice that the poles migrate back and forth across the imaginary axis depending on the oil properties. Also there is no clear trend, for instance at 40°C the real part of the pole is positive for 0% air and negative for 1% air, but at 70°C the opposite is true. This matches the nonlinear simulation and experimental results, indicating the linearization is an accurate representation of the system at that operating point. This also further illustrates the sensitivity of the transient valve performance to the system parameters, and how difficult this is to generalize, and therefore design out across the system parameter variation range.

![Figure 44 Pole Migration of Traditional Valve](image)

The same analysis was completed on the traditional pilot with proposed main stage valve. The pole locations for this valve with the same system parameter variations are shown in Figure 45. Notice that the 2\textsuperscript{nd} order poles are less sensitive to the system parameters, and are
always stable. Also there is only one dominant second order pole in this system, compared to two in the traditional valve. The 1st order poles are impacted the most by the system parameter variation. This is also consistent with the nonlinear dynamic simulations and experimental results.

![Figure 45 Pole Migration of Traditional Pilot with Proposed Main Stage]

Finally, the proposed pilot with proposed main stage was analyzed in the same manner. As expected the 2nd order poles are even less sensitive to the system parameter variation, however the 1st order lag terms are more sensitive as shown in Figure 46. This is also consistent with the nonlinear simulation and experimental results. These results can be attributed to the elimination of the pressure-balancing force and are intuitively logical.
4.2 Initial controller evaluation

Since the valves were designed for improved controllability, the controller design approach was to start simple. The most common controller used in industry is the Proportional Integral and Derivative (PID) due to its long history and relative ease of tuning. This controller consists of proportional, integral, and derivative gains and can be represented functionally as:

\[ u(t) = K_p e(t) + K_i \int_0^t e(\tau) d\tau + K_d \frac{de(t)}{dt} \]  \hspace{1cm} (20)

Where \( K_p \) is the proportional gain, \( K_i \) is the integral gain, \( K_d \) is the derivative gain, \( u(t) \) is the controller output, and \( e(t) \) is the controller output (or error signal). This results in a basic single input single output (SISO) system with regulated pressure as the output and voltage as the input. This is shown in Figure 47, where the error signal (desired pressure, \( P_d \), minus the
measured regulated pressure, \( P_r \) is input to the PID controller, C1, which outputs a voltage scaled to a desired current through the current driver C2 (shown in Figure 29), which then drives the solenoid, G, to the regulated pressure. In this system, G, is the linearized valve at the desired operating point. This linear system can then be studied at various operating points of the system.

\[ \text{Figure 47 Pressure Feedback Block Diagram} \]

Developing the controller was an iterative process requiring the linearized system at the desired operating point, the nonlinear numerical model, and the experimental test setup. Until this point, the linear model, nonlinear model, and open loop experimental setup have all been described. The next section covers the experimental setup required for the closed loop controller development.

4.2.1 Closed loop experimental setup

The first approach to develop the controller was to tune the controller experimentally using the popular Ziegler Nichols method [26]. This required the experimental setup outlined in 3.3.1, a data acquisition system, and a computer with LabVIEW software. Figure 48 shows the components used for experimental closed loop control and how they interact. The controller logic was programmed in LabVIEW. A data acquisition device was used to communicate the desired input from the laptop to the current driver and the valve pressure response to the laptop. The current driver converts was required to drive current to the solenoid based on the
pre-scaled voltage signal from the data acquisition device. The data acquisition device was also used to read voltage from a pressure transducer connected to the control port volume of the valve. Finally a hydraulic test stand consisting of pump, motor, heat exchanger, and pressure reducing valve was used to supply pressure to the valve.

4.2.2 Initial controller experimental results

The Ziegler-Nichols tuning method yielded unacceptable results for the proposed valve with proposed main stage, and acceptable results for the traditional pilot with proposed main stage. There were two main issues encountered while attempting to tune the proposed pilot with proposed main stage, stability and steady-state tracking. In order to produce a stable response the controller had to be slowed down significantly, however the system was only able
to track the commanded pressure within 0.4 bar as shown in Figure 49. In this example, the system requires approximately 5.5 seconds stabilizing, the goal for this is less than 0.07 seconds. Also, the commanded pressure is 10 bar, but the actual pressure is 10.4 bar.

![Figure 49](image)

**Figure 49 Experimental Closed Loop Response of Proposed Pilot with Proposed Main Stage**

The traditional pilot with proposed main stage valve was experimentally tuned with relative ease using the Ziegler-Nichols method. The controller gains were then fine-tuned heuristically until the desired response was achieved. Figure 50 shows the resulting 10 bar step response of the traditional pilot with proposed main stage to that of the open loop step response of the traditional pressure-control valve. The closed response matches the rise time of the traditional valve with significantly less overshoot and settling time.
4.3 Revised controller evaluation

The initial controller evaluation indicated that a simple PID controller could be tuned to achieve the desired step response for the traditional pilot with proposed main stage valve but not for the proposed pilot valve. This led to further controller studies for the proposed pilot with proposed main stage valve. In parallel, the PID controller was developed further for the traditional pilot with proposed main stage valve.

4.3.1 Further classical control design

Several attempts at classical control were made with no success. The first was a root locus approach using a reduced order model from the linearized operating point. The linearized state space representation of the valve at the 10bar operating point is shown in Table 4, where
A is the 7x7 state matrix, B is the 7x1 input matrix, C is the 1x7 output matrix, and D is the null feedthrough matrix.

<table>
<thead>
<tr>
<th>Table 4 State Space Representation of Proposed Pilot with Proposed Main Stage at 10bar Linearization Point</th>
</tr>
</thead>
<tbody>
<tr>
<td><strong>A</strong></td>
</tr>
<tr>
<td>0 1 0 0 0 0 0</td>
</tr>
<tr>
<td>-3332495 -37.9316 1.677955 1782.219 0 0 0</td>
</tr>
<tr>
<td>12263925 -75.6442 0 62.72853 -7817.94 0</td>
</tr>
<tr>
<td>0 0 0 -1000 0 0 1</td>
</tr>
<tr>
<td>0 0 0 449.6978 0 -4913493 -167.265 5.125865</td>
</tr>
<tr>
<td>0 0 0 0 19596248 0 -615.848</td>
</tr>
<tr>
<td><strong>B</strong></td>
</tr>
<tr>
<td>0</td>
</tr>
<tr>
<td>0</td>
</tr>
<tr>
<td>0</td>
</tr>
<tr>
<td>0</td>
</tr>
<tr>
<td>0</td>
</tr>
<tr>
<td><strong>C</strong></td>
</tr>
<tr>
<td>0 0 0 0 0 0 1</td>
</tr>
<tr>
<td><strong>D</strong></td>
</tr>
<tr>
<td>0</td>
</tr>
</tbody>
</table>

This is a SISO system with voltage to the solenoid as an input and regulated main stage pressure as an output. This was then converted to a seventh order transfer function using the relationship:

\[
G(s) = \frac{Y(s)}{U(s)} = [C(sI - A)^{-1}B + D]
\]  

The resulting transfer function was then reduced to the following fourth order form using a balanced model realization technique to aid in the classical controller design process:

\[
G(s) = \frac{0.8284s^4 - 1971s^3 + 6.489e6s^2 - 1.531e10s + 1.977e13}{s^4 + 601.3s^3 + 3.378e6s^2 + 1.877e9s + 6.95e10}
\]  

(22)
The step response of this reduced order system was then compared to the original system and the nonlinear model to validate accuracy prior to designing the controller. Figure 51 shows this comparison, notice the linearized system and reduced order system match, however the nonlinear model exhibits some lag down below 2bar. This is due to the difference in valve dynamics and fluid properties at lower operating points so it does not show up in the linearized model, however near the 10bar operating point the dynamics are very similar indicating that the linearization at 10bar is an accurate representation of the valve at that operating point.

The reduced order model was then used to develop a controller by placing the dominant poles in order to meet transient design goals. The dominant poles were placed to achieve <10% overshoot, <0.05s rise time, and zero steady state error. The desired pole locations were estimated using the following expressions [27]:

The figure shows the step response comparison for the proposed pilot with the proposed main stage.
\[ M_p = e^{\left( \frac{\pi \zeta}{\sqrt{1-\zeta^2}} \right)} \]  
\[ t_r = \frac{1.8}{\omega_n} \]

where \( M_p \) is percent overshoot, \( t_r \) is rise time, \( \omega_n \) is the natural frequency, and \( \zeta \) is the damping ratio. This resulted in \( \omega_n = 36 \text{ rad/s} \) and \( \zeta = 0.59 \) resulting in a desired pole location of \(-21.24 \pm 29.1j\). To eliminate steady state error an integrator was also added. Once the desired transfer function was found through manipulation of the root-locus plot it was tuned in the nonlinear model resulting in the following compensator:

\[ C(s) = 0.135 \frac{1 + 0.1s}{s(1 + 0.001s)} \]

This compensator worked well in the nonlinear model with no friction, however once friction was added the design goals were not met. The 10bar step response from the simulation is shown in Figure 52 with and without friction in the valve. This erratic response was validated in the experimental test setup.
4.3.2 Internal model control

Internal model control (IMC) is a special case of classical control in which a reference model is introduced into the feedback loop as a way to estimate the effect of disturbances on the output of the system [28]. This structure can be derived by adding and subtracting the reference plant model into the feedback loop and manipulating the block diagram as shown in Figure 53, where $G_m$ is the plant model and $Q$ is the IMC compensator.
In this control scheme the ideal compensator is $Q = G_m^{-1}$ so that perfect control is achieved, if $G_m = G$. Since $Q$ must be a proper and stable system, all non-minimum phase elements in $G_m$ had to be factored out of this system. This resulted in the IMC compensator in equation 26. An acceptable step response was achieved for the linearized 10 bar operating point with this IMC compensator. Figure 54 shows the step response of the 10 bar linearized system. This controller was then implemented in the nonlinear model as shown in Figure 55, where the compensator, $Q$, the linearized model, $G_m$, and the nonlinear plant model $G$ are identified.

$$Q(s) = 0.0628 \frac{(1 + 0.00375)(1 + 0.000042s)}{s(1 + 0.0000044s)}$$  \hspace{1cm} (26)
Figure 54 Step Response of IMC Compensator with 10bar Linearized Model

Figure 55 IMC Compensator Integrated Into Nonlinear Model
In the nonlinear simulations the controller was not effective. The regulated pressure went straight to the maximum 20bar bound (supply pressure limit). This was due to error between the linearized model, $G_m$, and the actual plant dynamics as shown in Figure 56, for the 10bar step response with IMC compensator. This error was a result of the changing system characteristics between 0 and 10bar, as shown in the open loop step response in Figure 55. Lag terms were added between $Q$ and $G_m$ in an attempt to reduce the effect of these changing dynamics with no success. Since the controller was developed, it was also attempted on the experimental setup with the same result. Based on these results, no further attempts at classical control were made and an optimal state space approach was pursued.

![Figure 56 Error between the Nonlinear Model Regulated Pressure and Linearized Model Regulated Pressure](image)
4.3.3 Optimal control design

Once the classical control techniques were exhausted for the proposed pilot with proposed main stage, optimal control was evaluated. This approach was chosen due to the fact that acceptable compensators were designed rather easily, but they were not robust enough for the unknown friction disturbances. State space techniques allow for the use of the internal states, such as spool position in this instance, in order to develop the controller.

Since steady state tracking is important, the first step was to add an integrator to the system. Then a Linear Quadratic Regulator (LQR) could be designed for the augmented system [29]. Figure 57 shows the augmentation of integral action into the linearized state space system along with the full state feedback.

![Figure 57 Block Diagram for Full State Feedback with Integration Augmentation](image)

Adding the integrator transforms the control law to the form:

\[
 u(t) = -[K \quad -k_1] \begin{bmatrix} x(t) \\ \xi(t) \end{bmatrix}
\]

(27)

where \( r(t) \) is the reference input to be tracked by the output \( y(t) \), and therefore \( \xi(t) \) is the integral of the tracking error. The state equation for the system can then be written as:
\[
\begin{bmatrix}
\dot{x}(t) \\
\dot{\xi}(t)
\end{bmatrix} = \begin{bmatrix}
A - BK & Bk_1 \\
-C & 0
\end{bmatrix} \begin{bmatrix} x(t) \\ \xi(t) \end{bmatrix} + \begin{bmatrix} 0 \\ 1 \end{bmatrix} r(t)
\]

(28)

\[
y(t) = \begin{bmatrix} C & 0 \end{bmatrix} \begin{bmatrix} x(t) \\ \xi(t) \end{bmatrix}
\]

The LQR controller is an optimal control technique that provides a tradeoff between regulation performance and control effort. This is accomplished by minimizing the quadratic performance index:

\[
J = \frac{1}{2} \int_{t_0}^{t_f} [x^T(t)Qx(t) + u^T(t)Ru(t)]dt
\]

(29)

where \( Q \) and \( R \) are weighting matrices for the regulation performance and control effort respectively. The controller gains were calculated using the linearized 10bar model with various weighting matrices, finally resulting in the following controller gains for 10bar linearized operating point:

\[
\begin{bmatrix}
3.404 & 0.308 & 0.2671 & 0.9409 & 0.0026 & -0.1757 & 0.0081 & 7.560
\end{bmatrix}
\]

The integrator and LQR controller were then tested in the nonlinear numerical model. Figure 58 shows the model, with state measurements, controller gains, and integrator augmentation.
This controller yielded an acceptable response in the nonlinear model with and without friction. Figure 59 shows the step response with and without friction. Both simulations track to 10 bar, have no overshoot, and have a rise time less than 0.03 seconds. The friction has little impact on the response with this controller.
Unfortunately it is not feasible to measure all of the states in the physical system due to cost and complexity. The only measurable state (within reason) is the regulated main stage control pressure. Due to this limitation the unmeasurable states had to be reconstructed from the measured state using a Kalman Filter. The Kalman Filter is a probabilistic based observer, often combined with a LQR. The combination of these two is a Linear Quadratic Gaussian (LQG) controller. The Kalman Filter design process involves finding an optimal observer gain matrix, $L$, that minimizes the error covariance between the actual state and the estimated state in the presence of process and measurement noise [30]. The unknown states can then be estimated as $\hat{x}(t)$, and the outputs estimated as $\hat{y}(t)$, using the known state information, system model, control input, and observer gain matrix:
\[
\dot{x}(t) = (A - LC)\dot{x}(t) + Bu(t) + Ly(t)
\]
\[
\dot{y}(t) = C\dot{x}(t)
\]

These state estimates are then fed to the full state feedback controller, in this case the previously designed LQR compensator. This LQG controller with augmented integrator is illustrated in block diagram form in Figure 60.

![Block Diagram for LQG Controller with Integrator Augmentation](image)

Like with previously described controller designs, the optimal observer gain matrix was found for the linearized 10bar operating point of interest:

\[
\begin{bmatrix}
-2200.24 \\
-1136196.22 \\
380478.77 \\
2913.46 \\
3176.39 \\
543600.76 \\
99275.36
\end{bmatrix}
\]
The LQG controller with augmented integrator was then implemented into the nonlinear model as shown in Figure 61. This is similar to Figure 58; however only the regulated main stage pressured is measured and input into the Kalman Filter. The estimated state matrix is than input to the LQR controller and the augmented integrator remains unchanged.

The LQG controller exhibited acceptable performance in the nonlinear simulation with no friction. As with all of the classical controller designs, the performance degradation was significant once friction was introduced. The transient characteristics were erratic and the regulated pressure did not track the input signal as shown in Figure 62.
4.3.4 Summary of initial controller evaluation

The ultimate goal of this research was to develop a valve and controller in parallel, because traditional pressure-control valves are fundamentally unsuitable for closed loop control for reasons described throughout this paper. These initial controller development studies have not only demonstrated this objective but have also uncovered other issues in the hardware. One primary issue is the impact of friction, resulting in unacceptable performance with the classical linear controllers once introduced in the nonlinear system with uncertain system disturbances, in this case friction.

The proposed pilot with proposed main stage, as well as the traditional pilot with proposed main stage, were both shown to be improved hardware for closed loop control when compared to the traditional valve through linear analysis. However the proposed pilot had too
much friction, as illustrated in Figure 36, for classical control. Full state feedback control yielded acceptable performance, however the additional sensors required to monitor the states are not feasible for this application due to cost and complexity. Estimating these states resulted in similar performance issues as the classical controllers once uncertain friction was introduced. The traditional pilot with proposed main stage was able to achieve desired performance with simple classical techniques and only main stage regulated pressure feedback. This is because the traditional pilot dynamics are not dependent on the main stage control port characteristics, only the pilot stage control port characteristics, which are internal to the valve, are more consistent, and are not the dominant factor in the overall system response.

Based on these results, the traditional pilot with proposed main stage was further developed due to the simplicity of the controller and sensor requirements. The proposed pilot with proposed main stage valve demonstrated the objective of this research, developing the hardware and software in parallel, but required additional state measurements that were outside the scope of this project and not feasible from a cost and packaging standpoint. Therefore, from this point on the traditional pilot with proposed main stage was further evaluated using simple classical techniques, for reasons previously described, and the proposed pilot with proposed main stage was no longer evaluated due to the additional state information required to develop an acceptable closed loop step response.
CHAPTER 5

CONTROLLER DEVELOPMENT AND FURTHER EXPERIMENTAL TESTING

5.1 Experimental controller tuning under standard operating conditions

As outlined in chapter 4, both valve concepts illustrated the need to develop the valve hardware and controller software in parallel as opposed to the traditional approach of designing a controller for existing hardware. Both valve designs were better suited for closed loop control when compared to the traditional valve; however the proposed valve with proposed main stage had too much uncertainty, in the form of friction, when compared to the traditional pilot with proposed main stage. In this chapter the controller development for the final valve design is summarized, followed by experimental testing to compare the new valve with closed loop control and the traditional valve with open loop control.

Since the system is nonlinear and continuous, it had to be discretized into various operating points in order to develop controller gains using a linear technique. Figure 63 shows the operating points at which the controller gains were tuned. It was found that 2bar increments were adequate for acceptable response using PID control with gain scheduling.
The proportional, integral, and derivative gains from equation 20 were fine-tuned iteratively to achieve desired performance at each operating point. Figure 50 shows this at the nominal 10bar operating point used throughout the study. This approach was extended to the other operating points in order to develop the gain schedule in Table 5; Figure 64 illustrates this at 2bar.
Figure 64 Closed Loop 2bar Step Response Compared to Traditional Valve Open Loop Step Response

Table 5 PID Controller Gains

<table>
<thead>
<tr>
<th>Operating Point (bar)</th>
<th>Kp</th>
<th>Ki</th>
<th>Kd</th>
</tr>
</thead>
<tbody>
<tr>
<td>2</td>
<td>0.025</td>
<td>1.3021</td>
<td>0.00033</td>
</tr>
<tr>
<td>4</td>
<td>0.0161</td>
<td>0.8412</td>
<td>0.000196</td>
</tr>
<tr>
<td>6</td>
<td>0.0079</td>
<td>0.418</td>
<td>0.000087</td>
</tr>
<tr>
<td>8</td>
<td>0.0066</td>
<td>0.3537</td>
<td>0.000062</td>
</tr>
<tr>
<td>10</td>
<td>0.0061</td>
<td>0.328</td>
<td>0.00005</td>
</tr>
<tr>
<td>12</td>
<td>0.0061</td>
<td>0.3259</td>
<td>0.00005</td>
</tr>
<tr>
<td>14</td>
<td>0.006</td>
<td>0.3236</td>
<td>0.00051</td>
</tr>
<tr>
<td>16</td>
<td>0.0063</td>
<td>0.3488</td>
<td>0.00057</td>
</tr>
<tr>
<td>18</td>
<td>0.0068</td>
<td>0.3829</td>
<td>0.00064</td>
</tr>
<tr>
<td>20</td>
<td>0.0091</td>
<td>0.5159</td>
<td>0.00088</td>
</tr>
</tbody>
</table>
5.2 Validation of experimentally tuned controller in nonlinear model

Once the controller gains were found experimentally they were evaluated in the nonlinear model and compared to the experimental data. Figure 65 shows the nonlinear model with PID controller and gain schedules. The gain schedules were implemented using lookup tables with linear spline interpolation, so based on the commanded pressure the PID gains were updated. Discrete samplers were also added as shown. This was to account for the sampling rate of the experimental setup. Typical applications for pressure control valves of this type can read and write in approximately 5ms increments, so this was the read and write interval used.

Figure 65 Nonlinear Model with PID and Gain Scheduling

Figure 66 shows the resulting 10bar step response of this model compared to the experimental data. The rise time and overshoot are similar; however the settling response does not match as accurately. There are several factors that could contribute to this including signal noise and varying system dynamics. The model response was considered close enough to study the effect of gross changes in system characteristics on step response in the future.
5.3 Frequency domain analysis

Once the step response requirements were met, the system stability was studied in the frequency domain. To accomplish this, the same procedure outlined in section 3.5 was used. However this time the controller was included in the frequency response of the proposed valve. Figure 67 shows the magnitude versus frequency for the traditional valve and the proposed valve with controller and Figure 68 shows the phase versus frequency. The gain margin and phase margin were then calculated using the frequency response results and are summarized in Table 6.
Figure 67 Discrete Magnitude versus Frequency of Traditional and Proposed Valve with Controller

Figure 68 Discrete Phase versus Frequency of Traditional and Proposed Valve with Controller
Table 6 Frequency Response Results

<table>
<thead>
<tr>
<th>Characteristic</th>
<th>Traditional Valve</th>
<th>Proposed Valve with Controller</th>
</tr>
</thead>
<tbody>
<tr>
<td>Gain Crossover</td>
<td>520Hz</td>
<td>15Hz</td>
</tr>
<tr>
<td>Phase Crossover</td>
<td>190Hz</td>
<td>100Hz</td>
</tr>
<tr>
<td>Phase Margin</td>
<td>-150°</td>
<td>130°</td>
</tr>
<tr>
<td>Gain Margin</td>
<td>-25dB</td>
<td>12dB</td>
</tr>
</tbody>
</table>

5.4 Further experimental testing

This project was initiated in order to improve certain performance characteristics and how they vary with varying system parameters. There are many common tests used to compare pressure control valve performance. One of the objectives of this study was for the proposed valve to outperform the benchmark valve in some of these common tests that are often encountered, including:

- **Repeatability**: How consistently the valve achieves the commanded pressure
- **Variation**: The variation in pressure output for a given (or multiple) command(s) for the entire valve population
- **Hysteresis**: The difference in pressure at a given command depending on the direction or approach (increasing or decreasing current input)
- **Step response**: The time required to achieve a desired pressure
- **Leakage sensitivity**: Change in regulated pressure due to leakage from control port
- **Supply pressure sensitivity**: Change in regulated pressure due to changes in supply pressure
- **Temperature sensitivity**: Change in regulated pressure across temperature range
Testing was completed to compare the valves in the areas described above. The results are outlined in the following sections.

5.4.1 Repeatability and variation

Since open loop applications are inherently subject to output variation, the regulated pressure repeatability of a single valve and the variation from valve to valve are important characteristics. In this section the proposed valve is compared to the traditional valve in both of these areas. Each valve was stepped to achieve 10bar pressure (the traditional valve was stepped to the appropriate input current) 30 times each and the distributions were compared. Pressure was measured using a Sensotec TJE transducer with a +/- 0.1% of full scale accuracy. The results of this test indicate that the proposed valve is more repeatable than the traditional valve as shown in Figure 69. The transducer repeatability gates are also displayed, notice that the proposed valve is within the measurement capability of the transducer and the traditional valve has approximately three times the variation.
5.4.2 Hysteresis

The hysteresis of a valve is important for the same reasons described in 5.3.1 and can be quantified through several different methods. Figures 32, 34, and 36 show the typical hysteresis of the different valves when current is ramped from zero to maximum and back down at a set rate. The resulting pressure hysteresis is a combination of magnetic force hysteresis and various sources of friction. There can also be transient effects when ramping the input current due to hydraulic pressure dynamics and solenoid lag. To eliminate transient effects the valves were compared by stepping from 0bar to 10bar to 20bar to 10bar to 0bar in that order. This command profile for hysteresis testing is illustrated in Figure 70. The current signal is held for 2 second intervals to allow pressure to stabilize and eliminate transient effects.
The results are shown in Table 7 with corresponding command sequence from Figure 69. The traditional valve exhibited a 0.37bar difference when commanded to 10bar from different directions while the proposed valve was within gauge variation. This was expected based on the regulated pressure tracking of the proposed valve with closed loop controller.

<table>
<thead>
<tr>
<th>Command Sequence (from Figure 63)</th>
<th>Traditional Valve (bar)</th>
<th>Proposed Valve (bar)</th>
</tr>
</thead>
<tbody>
<tr>
<td>1</td>
<td>0</td>
<td>0</td>
</tr>
<tr>
<td>2</td>
<td>10.04</td>
<td>9.99</td>
</tr>
<tr>
<td>3</td>
<td>20</td>
<td>20</td>
</tr>
<tr>
<td>4</td>
<td>10.41</td>
<td>10</td>
</tr>
<tr>
<td>5</td>
<td>0</td>
<td>0</td>
</tr>
</tbody>
</table>

The magnitude of the hysteresis will vary for a given population. For the benchmark traditional valve, the maximum allowed is 0.7bar. Figure 71 shows the distribution of a large
production run of the traditional valve. For the proposed valve, a large population is expected to be within gage variation, similar to the repeatability study shown in Figure 69.

![Figure 71 Distribution of Traditional Valve Hysteresis](image)

**5.4.3 Step response**

The improved step response has already been demonstrated (see Figure 50 and Figure 64) as this was the focus of the research. However this is only the response from 0 to a target pressure. Another characteristic of interest for pressure-control valves is the step response from an intermediate command to another command. This can differ significantly from the full step response in the traditional pressure-control valve. Figure 72 shows the step response for both valves when stepped from 2bar to 10bar.
5.4.4 Sensitivity to system parameters

As discussed earlier throughout the paper, the traditional pressure-control valve performance characteristics are sensitive to changes in system parameters. In this section the steady state regulated control pressure is studied for varying system parameters. The valves were stepped to 10bar under nominal conditions and then pressure was measured as different system parameters were varied. The sensitivity to control port leakage, supply pressure, and temperature were compared.

Control port leakage is common in many hydraulic applications. For example, in a clutch, the regulated pressure is sealed to the tank reservoir through some type of rotary dynamic seal. These seals leak varying amounts and this leakage can change over time and under certain conditions. The regulated control pressure of the traditional valve is sensitive to
changes in leakage, which is not desirable. To test this, a leak was simulated using the setup show in Figure 27 and Figure 28, except the clutch was replaced with a needle valve. The valves were stepped to 10bar with no leakage, then the needle valve was opened until a target flow rate was achieved, at which point pressure was measured. Figure 73 shows the resulting change in pressure for both valves for a given flow, notice the traditional valve pressure drops with increasing flow but the proposed valve with controller remains at 10bar.

![Figure 73 Control pressure versus control port leakage](image)

The benchmark traditional pressure-control valve is also sensitive to the supply pressure. The regulated control pressure is proportionally related to the supply pressure. This was tested in the same setup except supply pressure was varied and is illustrated in Figure 74, once again the proposed valve with closed loop controller maintains 10bar regulated pressure.

100
Another system parameter of importance is the oil temperature. The oil properties change as a function of temperature which leads to changes in regulated control pressure. This was tested in a similar setup, shown in Figure 75. This setup is schematically the same, but it takes place in a temperature controlled thermal chamber, which also houses a large volume of oil (to stabilize oil temperature prior to testing).
Figure 75 Temperature sensitivity test setup

The control pressure was then measured at different oil temperatures and compared in Figure 76. This sensitivity can be erratic from valve to valve and at lower temperatures in the traditional valve due to the significant changes in oil properties; notice the change in slope at the -10°C data point. Once again, this is solved by the regulated pressure tracking of the closed-loop controller and de-sensitized valve design.
Figure 76 Control pressure versus oil temperature

5.4.5 Clutch shift simulation testing

One common issue in developing pressure-control valves is the disconnect between the individual valve performance and the resulting system performance. The performance characteristics shown in previous sections are applicable to system performance; however they are not a great indicator of system level performance in many cases. This is due to the coupled valve - system dynamics described previously. The ultimate test is how the valve performs in a simulated system, in this case a clutch control application is studied. This was tested using the clutch simulation sequence shown in Figure 3, in the test setup shown in Figure 28.

In typical clutch applications the valve is stepped full on for a period of time in order to fill the clutch prior to modulation, this is known as the fill phase. The duration of this fill phase is often times calibrated to account for variation in the passage geometries, valve performance,
and clutch characteristics. This is not a very robust process, often leading to over-filling or under-filling of the clutch resulting in variable shift quality. Another common issue is the transition between fill and pressure modulation phases, mainly stability issues as the forces come to a balance. This leads to longer hold times so that pressure can stabilize prior to the ramping phase. This issue also occurs between the hold and ramp of the pressure modulation phase, due to the amount of oil that needs to be displaced. All three of the characteristics are interdependent and a significant amount of development time is required to minimize the impact of these characteristics on overall shift quality. Figure 77 shows an experimental shift simulation trace with the traditional pressure-control valve with the undesirable performance characteristics indicated. In this trace, red is the current commanded to the valve and blue is the resulting pressure response.

Figure 77 Clutch shift simulation testing with traditional valve
The same shift scheme can be applied to the proposed valve except desired pressure is the input since the system is closed loop. Figure 78 shows the same experimental shift simulation as above but with the proposed valve and controller. Notice during the fill phase the regulated pressure tracks the desired pressure; this will lead to more consistent clutch fill on a given system and across a larger population of systems. Also the pressure tracks during the hold phase allowing this hold time to be reduced and allowing for a faster shift in an actual application. Finally the pressure also tracks better during the ramp phase; this would result in a “smooth” (no undesired acceleration or jerk experienced by the driver) in an actual application. One area of concern are the oscillations between the fill and hold phase of the simulation, however these did not seem to impact the areas of interest in limited testing so this was not explored further.

![Figure 78 Clutch shift simulation testing with proposed valve and controller](image-url)
6.1 Significance and summary

The motivation for this research was to improve pressure-control valve technology based on observing similar performance issues in different systems, primarily clutch control applications. These issues have been summarized throughout this dissertation. To this point, the solutions to these issues have been limited to hardware changes in the valve, which are difficult to execute in mature products, do not typically address the issue completely, and often require sacrifices in other areas of the design. Software solutions for these issues have failed due to the physics of pressure-control valves and interaction with the system; they are fundamentally not suited for closed loop control (especially in the volatile high speed applications in which they are typically used). These experiences lead to the development of new hardware in parallel to software in order to address the most common issues.

Hydraulic systems are highly nonlinear and sensitive to typical parameter variation. This is especially true in pressure-control valves because the valve dynamics are coupled to the hydraulic dynamics as discussed previously. This is the underlying reason for the previous challenges that could not be solved completely through hardware and also why closed loop control has failed in these applications.

6.2 Conclusion and contribution

In this research two valves were designed to be less sensitive to system parameters and therefore more suitable for closed loop control. These valves, along with a benchmark
traditional pressure-control valve, were modeled, experimentally validated, and studied extensively. The newly designed pressure-control valves were de-coupled from the system and had more consistent dynamics, but were not suitable for open loop applications due to the steady-state variation resulting from varying system parameters. The traditional valve had more consistent steady-state performance (as it was designed to do) but the dynamics were highly sensitive to typical system parameter variation.

The first proposed valve, which utilized no pressure balancing, was able to outperform all other valves (in the nonlinear model) using an LQR controller, but was not feasible because of the full state feedback that was required. The second proposed valve outperformed the traditional pressure-control valve using simple PID control with gain scheduling. This was determined using various industry tests for comparing pressure control valves, along with actual clutch shift simulations.

The objective of this research was achieved, as a new valve designed in parallel with a closed loop controller was able to address the concerns with the benchmark traditional pressure-control valve, and also outperform the valve in all areas studied. This valve and controller combination is also feasible for implementation within existing applications where the traditional pressure-control valve is currently used and provides much more design flexibility without the sacrifice of hardware changes for the end user due to the software tuning capabilities. The major contributions of this work can be summarized as follows:

- A comprehensive overview of traditional pressure-control valves and how they interact with systems they are used in, as well as the current state of hydraulic clutch control
• Development of a mathematical model for various pressure-control valves with experimental validation

• Demonstration of the drawback of traditional pressure-control valves for closed-loop control through linear analysis

• Analysis of the influence of a hydraulic system on the performance of a pressure-control valve

• Evaluation of various control techniques and how they relate to different valve designs

• A novel valve design that can be controlled with simple linear techniques and outperform a traditional pressure-control in various industry driven areas of interest.

6.3 Future work

This research was focused on developing a valve and controller that could outperform the benchmark traditional valve in typical industry tests, but was also feasible for implementation in actual systems. This was demonstrated through analysis and testing; however there is much more work to do in order to realize this potential in high volume applications.

The next step would be to study the robustness of this valve and controller across all of the different system parameters and comprehensively tune the controller gains across the parameters for desired dynamic performance. This would be followed by testing this on a larger sample of valves to incorporate more valve variation. Finally using these valves in actual systems and quantifying the improvement on the desired system level performance to validate.
Figure 79 Viscosity versus Temperature

Figure 80 Bulk Modulus versus Temperature
Figure 81 Density versus Temperature
REFERENCES


