COMPARATIVE STUDY OF POSITION CONTROL WITH 2-WAY AND 3-WAY ON/OFF ELECTROHYDRAULIC VALVES

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Abstract

A comparative study of system-level performance is given for on/off valve control of a hydraulic cylinder. A dynamic model was created and simulated to develop and evaluate innovative control strategies specific to the use of on/off valves. Cylinder position control was investigated using a configuration consisting of two 3-way on/off valves and a configuration with four 2-way on/off valves. Controllers were developed to minimize the influence of on/off valve performance on the cylinder position responses. All controllers and valve configurations were experimentally tested and compared with the simulation results. A sensitivity analysis was performed by varying the supply pressure and external load applied to the cylinder. It was determined that successful performance to sine wave and step inputs can be achieved with the control theories presented and that the four 2-way on/off valve configuration provides superior position performance.

Keywords: position control, on/off valves, 2-way valve, 3-way valve, comparative study

1 Introduction

The most common method for controlling actuator position with fluid power equipment is by the use of variable resistive orifices. High pressure fluid is metered through a variable orifice which is controlled by the user and regulates the flow into the actuator, thereby controlling its inherent position. Although this provides excellent control, by metering the high pressure fluid it is energetically inefficient. Fluid power energy is converted into heat energy by metering inside of a valve and is not recovered. Due to this disadvantage, alternative methods for controlling position have been investigated (Scheidl et al., 2000) including methods that do not use metering (Ivantysynova, 2005).

Research has shown that accurate position control of hydraulic actuators can be accomplished with the use of multiple on/off valves (Van Varseveld and Bone, 1997; Ahn and Yokota, 2005; Paul et al., 1994; Liu et al., 2000; Gentile et al., 2002; Scheidl et al., 2000). An advantage of using on/off valves for position control is that either state of the on/off valve is a low power consuming state (Li et al., 2005). By strategically timing pulse durations consisting of the valve being fully open or fully closed, the throttling losses associated with traditional proportional resistance control are reduced.

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Various configurations and valve types have been used to achieve position control. Researchers emphasize the advantages of decoupled orifice openings by implementing multiple on/off valves and provide design tradeoffs associated to each configuration and valve type (Gao et al., 2008; Manhartsgruber et al., 2005; Liu and Yao, 2002).

Most research with on/off valve position control show strong dependence of the actuator response to the performance of the on/off valve (Van Varseveld et al., 1997; Branson et al., 2008). Therefore, for desired performance of the actuator position, a high quality, high-cost on/off valve must be used.

It is the focus of this research to investigate innovative control strategies that minimize the dependence of the actuator response to the performance of the on/off valve. Control theories such as pulse width modulation (PWM) are not used because the success of these strategies rely significantly on the on/off valve characteristics.

In addition to the controller development for position control, a comparative study between four 2-way valve and two 3-way valve configurations is given. Design tradeoffs and comparisons in actuator performance based on responses to step inputs and sinusoidal inputs are given for systems operating at low pressures

(< 60 bar). Conclusions are made for the best controller and a discussion of configuration comparisons is given.

2 Physical Configurations

The actuator used throughout this study was a double acting, double ended hydraulic cylinder. The hydraulic power source consisted of a fixed displacement pump with a relief valve which maintained a constant supply pressure to the system. The controlling valves were Parker Hannifin on/off valves. These valves are 3-way, 2 position valves with a spring bias to connect the work port to the tank port of the valve in the normal, non-powered state. To maintain consistency in valve characteristics and performance, the four 2-way valve configuration used the same 3-way valves but with one port blocked, causing the valve to act as a 2-way valve. The configurations investigated for position control are shown in Fig. 1 and Fig. 2.

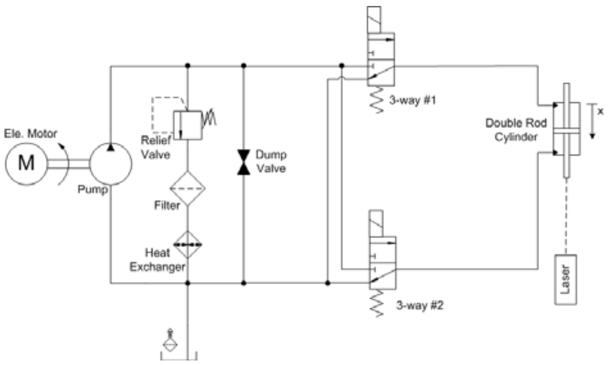

Fig. 1: *Two 3-way valve configuration*

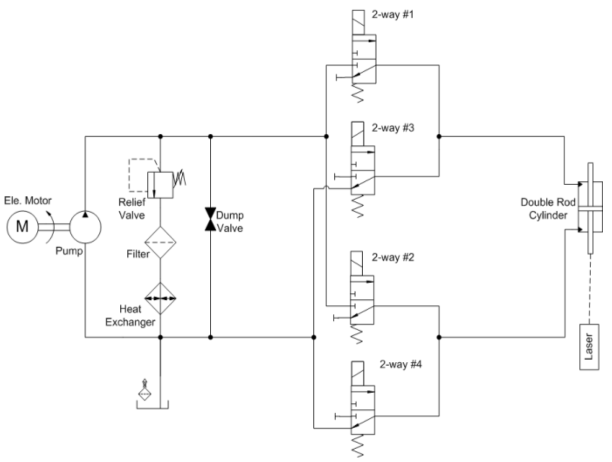

Fig. 2: *Four 2-way valve configuration*

The valves in Fig. 1 and Fig. 2 are shown in their normal, non-powered states. The four 2-way valve configuration provides independent control of both supply and tank pressures to either chamber of the cylinder. Notice that the 3-way configuration only allows either supply pressure or tank pressure to be exposed to either cylinder chamber. This leads to the configuration not being able to hold external loads by hydraulically locking the cylinder position.

2.1 On/Off Valves

The objective of the controllers in this research is to minimize the effects of the on/off valve characteristics on the actuator performance. The on/off valves are operated with no specialized driving circuitry and are operated at the manufacturer's specifications. The valves are rated at 500 mA DC continuous current. Flow testing of the valves indicated that the valve orifice areas and associated valve response times for the P-A orifice and A-T orifice are asymmetric.

2.2 Actuator

The actuator used for testing is a Parker Hannifin double rod, double acting hydraulic cylinder. The displacement of the cylinder was measured using a Keyence LK-G82/LK-GD500 Laser with 12 ms averaging for noise filtering. Cylinder position was the only feedback provided to the controller. The cylinder has a 26.7 mm stroke with a 38.1 mm bore and 25.4 mm diameter rods.

2.3 Hydraulic Power Supply and Lines

A Parker Hannifin Hydraulic Power Unit (H1 8.1NS3) is used to provide the hydraulic flow source. The power unit supplies 30.7 l/min flow with a maximum pressure of 125 bar. A manual pressure relief valve (Parker Hannifin RP600SF) is used to adjust the supply pressure to the desired setting. Hydraulic transmission lines connecting the supply to the valves, cylinder to the valves, and the reservoir to the valves are 10 mm diameter SAE100R2AT steel reinforced rubber hoses.

2.4 System Operational States

The two 3-way valve configuration with both valves in either the powered or normal, non-powered states allow the cylinder to float. In reference to Fig. 1, powering valve #1 causes extension of the cylinder as chamber A becomes pressurized and chamber B is connected to tank pressure through valve #2. Valve #2 is powered and valve #1 is turned off to accomplish retraction of the cylinder. Table 1 shows the operational states for the two 3-way valve configuration.

Table 1: Two 3-way valve operational states

Valve No.	Cylinder Float	Cylinder Extend	Cylinder Retract
#1	OFF	ON	OFF
#2	OFF	OFF	ON

The four 2-way valve configuration shows more flexibility as each orifice is independently controlled. Various "centre positions" are available with this configuration such as open centre, float centre, and closed centre. The controllers investigated utilize only the closed centre position. The operational states for the four 2-way valve configuration are given in Table 2.

Table 2: Four 2-way valve operational states

Valve No.	Cylinder Hold	Cylinder Extend	Cylinder Retract
#1	OFF	ON	OFF
#2	OFF	OFF	ON
#3	ON	ON	OFF
#4	ON	OFF	ON

Note that the four 2-way valve configuration can operate as the two 3-way valve configuration. In this scenario, valves #1 and #3 actuate together and valve #2 and #4 actuate together.

3 Simulation

A simulation model was developed in MATLAB Simulink® utilizing the Simscape™ toolbox (a power flow modelling toolbox). The simulation was used to test, tune, and develop the controllers investigated in this research.

3.1 Simulation Assumptions

The simulation assumes that the components are ideal with no internal friction or volumetric inefficiencies. It is assumed in the simulation that the pressure drop from the flow source to the cylinder and from the cylinder to the reservoir is due to a simplified lumped pressure drop model following Eq. 1.

$$\Delta p = k_1 q + \text{sign}(q) k_2 q^2 \tag{1}$$

Experimental validation of the model provides a pressure drop vs. flow curve for each P-A orifice path and A-T orifice path of each cylinder chamber. A quadratic curve fit is given for the pressure drop vs. flow curve for each orifice providing constants k_1 and k_2. k_1 corresponds to a linear resistance and is assumed to account for all pressure drops in the lines, manifolds, and fittings that are linearly proportional to the flow. The k_2 term is used to calculate the effective orifice areas of the on/off valves. It is assumed that the on/off valve orifices provide the turbulent portion of the pressure drop. The effective orifice areas of the valves are determined from the k_2 coefficients following Eq. 2.

$$A_{\text{veff}} = \frac{1}{C_{\text{d}}} \sqrt{\frac{\rho}{2k_2}} \tag{2}$$

Because the valve orifices are assumed to contribute only to the turbulent portion of the pressure drop to and from the hydraulic cylinder, the orifice equation for the valves are:

$$q_{\text{val}} = C_{\text{d}} A_{\text{veff}} \sqrt{\frac{2}{\rho}|p|} \cdot \text{sign}(p) \tag{3}$$

The cylinder is modelled with no internal leakage but does account for friction, mass, and end stop characteristics. The friction of the cylinder is assumed to follow Eq. 4 which is developed from the Stribeck friction curve.

$$F_{\text{f}} = \text{sign}(\dot{x})\left(F_{\text{coul}} + \left(F_{\text{brk}} - F_{\text{coul}}\right) e^{-c_{\text{vf}}|\dot{x}|}\right) + F_{\text{v}} \tag{4}$$

The end stops that limit the stroke of the cylinder are modelled as a combined stiff spring and damper. The mass of the cylinder piston and attached rods are accounted for in the model as well.

It is assumed in the model that the valve characteristics can be captured with a phase lag in combination with a slew rate to limit the transition time. The simulation controller command and the signal to an individual valve is shown in Fig. 3.

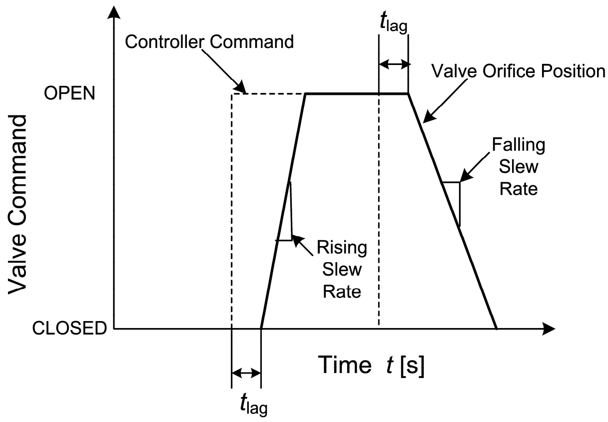

Fig. 3: *Simulation valve characteristic model*

3.2 Simulation Model

The simplified simulation model of the two 3-way valve configuration is shown in Fig. 4.

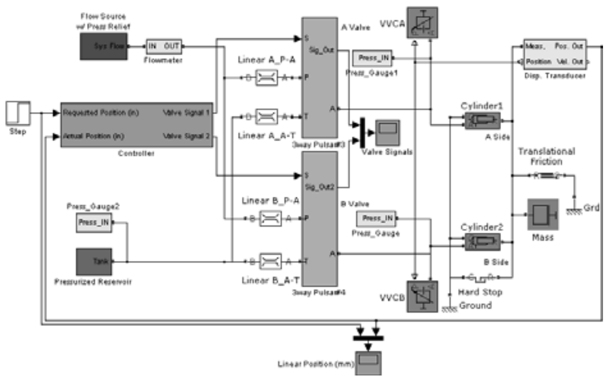

Fig. 4: *Two 3-way configuration simulation model*

Simulation parameters used to test, tune, and develop the controllers for each configuration are shown in Table 3 (Long, 2009).

Table 3: Simulation parameters

Parameter	Value	Unit
Syst. Supply Pressure	variable w/ experiment	bar
Syst. Tank Pressure	variable w/ experiment	bar
Oil Viscosity	variable w/ experiment	cSt
Syst. Supply Flow	30.7	l/min
Oil Density	870	kg/m^3
System Bulk Modulus	6200	bar
Air Entrapment in Oil	0.03	V_g/V_l
Cyl. Piston Area	633	mm^2
Cyl. Piston Mass	953	g
Cyl. Chamber Dead Vol.	8390	mm^3
Cyl. Stroke	26.7	mm
End Stop Spring Constant	$1.00\text{x}10^{12}$	N/m
End Stop Damping Coeff.	$1.50\text{x}10^{7}$	N/(m/s)
Cyl. Breakaway Friction	225	N
Cyl. Viscous Friction Coeff.	2170	N/(m/s)
Cyl. Coulomb Friction	201	N

Lines and trapped volumes are modelled to account for fluid compressibility and follows Eq. 5.

$$V_\mathrm{f} = V_\mathrm{c} + \frac{V_\mathrm{c}}{E} p_\mathrm{sys} \tag{5}$$

The fluid in the contained volume (V_f) is variable with respect to the geometric volume (V_c), system bulk modulus (E), and system pressure (p). The bulk modulus is modified to capture cavitation effects in the model and follows Eq. 6.

$$E = E_1 \frac{1+\alpha\left(\dfrac{p_\mathrm{a}}{p_\mathrm{a}+p_\mathrm{sys}}\right)^{1/\mathrm{n}}}{1+\alpha\dfrac{p_\mathrm{a}^{1/\mathrm{n}}}{n\left(p_\mathrm{a}+p_\mathrm{sys}\right)^{\mathrm{n}+1/\mathrm{n}}}E_1} \tag{6}$$

The pressures inside the cylinder chambers follow the pressure build up equation. Because the cylinder is assumed to have no internal leakage, no leakage flow term is included in the pressure build up equation of Eq. 7.

$$\frac{dp_\mathrm{cyl}}{dt} = \frac{E}{(V_0 + A_\mathrm{cyl}x)}\left(q_\mathrm{i} - A_\mathrm{cyl}\dot{x}\right) \tag{7}$$

4 Controller Development

A bang-bang type control is utilized as the baseline controller. The bang-bang controller works with an error band about the commanded position which determines the action of the valve(s). If the actuator position is outside of the error band, corrective action is taken by the valve(s) to return the position inside the error band. Once the actuator position is inside the error band, the valve(s) take no further action to promote movement of the actuator. This control theory has been used by other researchers and has shown successful results with use of position control utilizing on/off valves (Beachley, 1998; Liu, 2000). The bang-bang controller is implemented in both the 2-way valve configuration and the 3-way valve configuration. For the 3-way configuration, once the actuator position enters inside the error band, both valves are turned off to their normal, non-powered states connecting both chambers of the cylinder to tank pressure. With the 2-way configuration, once the position of the cylinder is inside the error band the controller commands a "closed centre" position with valves #1 and #2 being turned off and valves #3 and #4 being powered to achieve hydraulically locked cylinder chambers. Since all controllers relied on bang-bang, or on/off control, comparisons were focused on accuracy (minimal limit cycles measured by mean error), and not on overall system stability.

4.1 3-way Configuration Controllers

Specific to the 3-way configuration as shown in Fig. 1, two controllers are developed in addition to the bang-bang controller.

A controller that utilizes fuzzy logic is implemented. Fuzzy logic is different from other controllers in that it is not based on a physical model of the system, but instead is based on heuristic knowledge of the system following a set of governing rules determined from the controller designer. Because of the nonlinear characteristics of hydraulic components and the complexity of accurately capturing responses due to fluid compressibility, capacitance, and inertia, fuzzy logic is an interesting design to consider for this application. The controller rules are shown in Table 4.

Table 4: Fuzzy logic controller rules

Rule No.	Rule Statement	Weight
#1	If error is NegLarge, then output is MoveBack	1.0
#2	If error is Negative, then output is MoveBack	0.75
#3	If error is SmallError, then output is NoChange	1.0
#4	If error is Positive, then output is MoveFwd	0.75
#5	If error is PosLarge, then output is MoveFwd	1.0

Fuzzy logic has been applied to many applications by past researchers (Mauer, 1995; Chen, 2004), but the unique feature of the controller developed here is the way the controller output is interpreted and communicated to the digital on/off valves. Because the fuzzy logic controller is analog in nature, a discretizing controller is implemented into the design. The discretizing controller interprets the fuzzy logic output and then converts the analog output into digital commands to the on/off valves. There are three distinct modes of operation for the on/off valves with the fuzzy logic controller. The valves cause the cylinder to fully extend or retract, the valves turn completely off, or they achieve a state of repeated pulses. The extend, retract, and float conditions are given in Table 1. The repeating pulsing mode consists of repeated sequences of one valve being turned on for 15 ms then turned off for 25 ms. Valve #1 is pulsed to extend the cylinder, while valve #2 is pulsed to retract the cylinder.

The pulsed mode of the fuzzy logic controller allows the cylinder to coast to the desired position. The 15 ms on pulse initiates cylinder movement, then the 25 ms hold allows the inertia of the cylinder moving mass to be utilized. When the cylinder is moving due to its inertia, fluid is being drawn from the tank, thereby showing efficiency gains as the translational momentum is captured and utilized by the control strategy.

Another controller developed provides a variable phase lag between the valves as they continually switch states. The magnitude of phase lag between the valves for transition is proportional to the position error of the cylinder. For no error between the command and the actuator position, the valves switch states together in phase. Based on the magnitude of the error, one valve lags the other in transition from state to state. This allows for very small durations of time where one cylinder chamber is pressurized and the other is at tank pressure. The pressure differences in the time domain are less than the switching speeds of the valves since it is only the difference (out of phase switching) between the switched pairs of valves, not the actual switching speed of the valves themselves. The operational function of the controller is shown in Fig. 5.

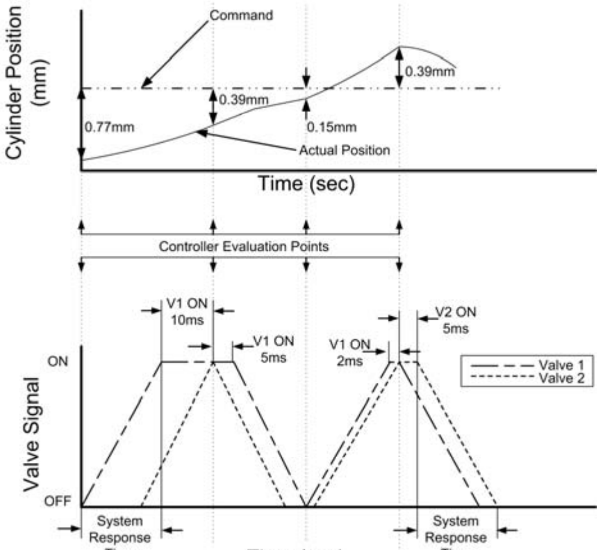

Fig. 5: *Phasing controller operation*

Notice that for small errors, the time which one valve is in an ON state is smaller than the transition response time for the system. It was assumed by the controller that the time duration from command input to actual cylinder movement is 15 ms, or the system response is 15 ms. This provides an avenue for which low performing valves can be utilized to achieve small pulse durations to the actuator that are shorter than the response times of the specific valves used. Meeting the objectives of the controller design, the performance of the position response is decoupled from the performance of the on/off valves.

The ability to command small pulse durations to the actuator does come with a design trade off. As the valves spend a significant portion of the cycle time in transition between the states, the fluid entering and exiting the cylinder chambers is being metered through smaller orifices than what the case would be if the valves were in their normal "on" or "off" positions. Therefore this controller exhibits increased metering losses, but has the advantage of better controllability as the controller provides a sense of proportionality.

4.2 2-way Configuration Controllers

In addition to the bang-bang controller, a pulsing controller was developed for the 2-way valve configuration. The pulsing controller takes advantage of four independently controlled orifices specific to the system configuration.

The pulsing controller operates in three modes. The first is a simple bang-bang control. Another mode is a hydraulically locked position where the actuator is not allowed to move. The third mode consists of the valves causing the cylinder chambers to be compressed and decompressed utilizing the compressibility of the hydraulic fluid.

Once the controller enters the pulsing mode, one chamber is charged to supply pressure, while the other remains hydraulically locked. After a determined time in this state, the valves revert to the closed centre position where the fluid in both cylinder chambers is trapped. Once the pressures are equalized in the cylinder chambers due to movement of the cylinder piston, the opposite chamber is exhausted to tank pressure. This decompresses one chamber causing cylinder movement as the pressures in the cylinder chambers equalize. The pulsing mode is shown in Fig. 6 for extending the cylinder.

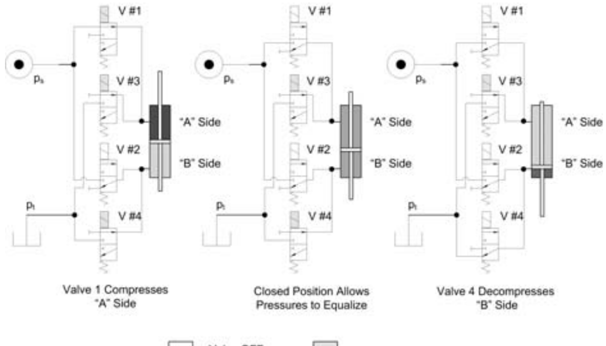

Fig. 6: *Pulsing controller – pulsing mode operation*

When the controller is in the pulsing mode, the only movement of the cylinder is due to the compressibility effects of the hydraulic fluid. Therefore the system bulk modulus, cylinder chamber volumes, and operating pressures all affect the magnitude of cylinder travel per pulsing cycle.

Because of the compression and decompression of the cylinder chambers, small steps in position can be achieved which leads to finer accuracy and lower steady state errors.

5 Performance Results

To evaluate the performance of the controllers, testing was conducted with 7 mm and 20 mm step inputs as well as 0.4 Hz and 2.0 Hz sinusoid inputs with 6.5 mm amplitude. Testing parameters are given in Table 5.

Table 5: Performance testing parameters

Parameter	Value	Unit
System Supply Pressure	43	bar
System Supply Flow	31	l/min
System Tank Pressure	3.5	bar
Oil Temperature	45	°C
Oil Viscousity	26	cSt

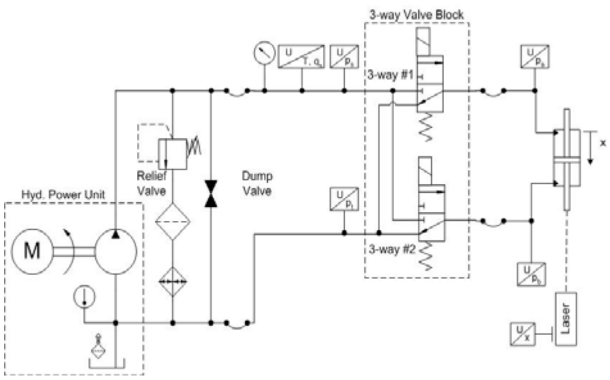

Fig. 7: *Performance testing schematic*

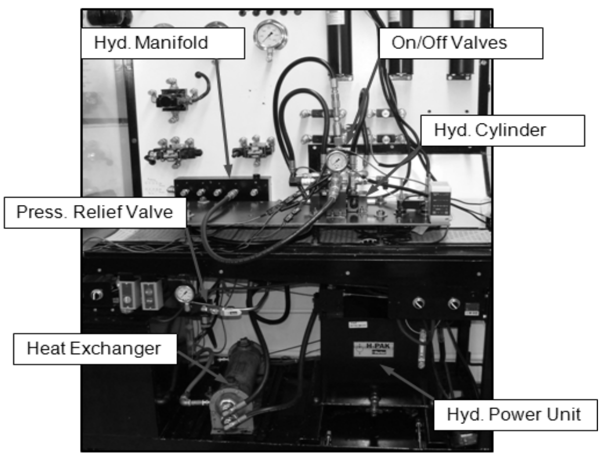

Fig. 8: *Experimental testing hardware*

The testing schematic for evaluating the controllers is given in Fig. 7. Note that Fig. 7 is shown for the 3-way valve configuration. For the 2-way configuration, the two 3-way valves in Fig. 7 are replaced by the four 2-way valves shown in Fig. 2. The testing hardware is shown in Fig. 8.

5.1 3-way Valve Configuration Results

Response plots for each of the 3-way configuration controllers are given below for step inputs. The position of the cylinder was measured with the Keyence non-contacting laser.

The results for both the 7 mm and 20 mm step inputs show similar results between the controllers. The 7 mm step input is shown here for the bang-bang controller (Fig. 9), fuzzy logic controller (Fig. 10), and phasing controller (Fig. 11).

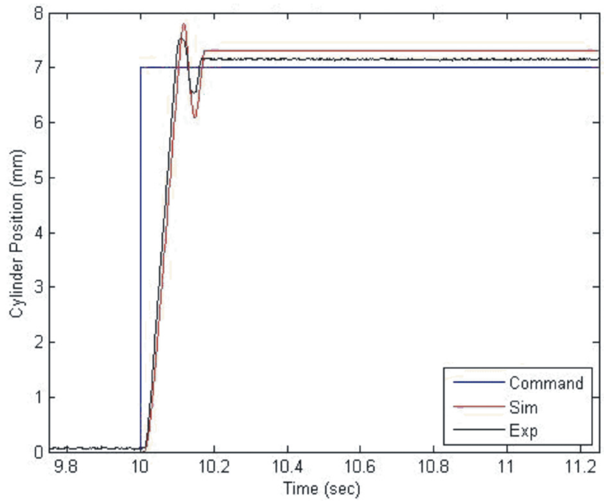

Fig. 9: *Bang-bang controller response – 7 mm step*

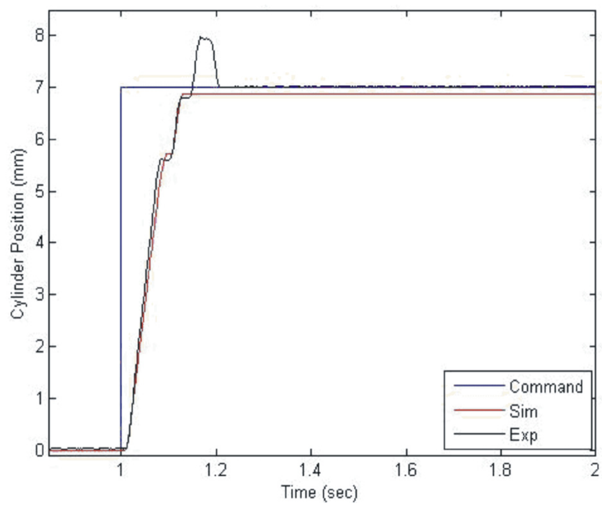

Fig. 10: *Fuzzy logic controller response – 7 mm step*

Notice from Fig. 9 that the presence of the error band of the bang-bang controller is evident in the response due to the steady state error when the valves return to their normal, non-powered states. The error band for the 3-way bang-bang controller that produced the best response performances was ± 0.40 mm.

The fuzzy logic controller shows steps in the response that correspond to the 15 ms pulse on, 25 ms hold mode of the controller. Notice that there is significant overshoot, but the steady state error is small. Data in Table 6 and Table 7 also shows the same response trends.

The phasing controller shows similar overshoot as the bang-bang controller, but achieves nearly zero average steady state error. Because this controller does not exhibit a mode where the valves turn off, the controller is always adjusting to any error between the response and the command. This is indicative in Fig. 11 from the small amplitude oscillations about the steady state command.

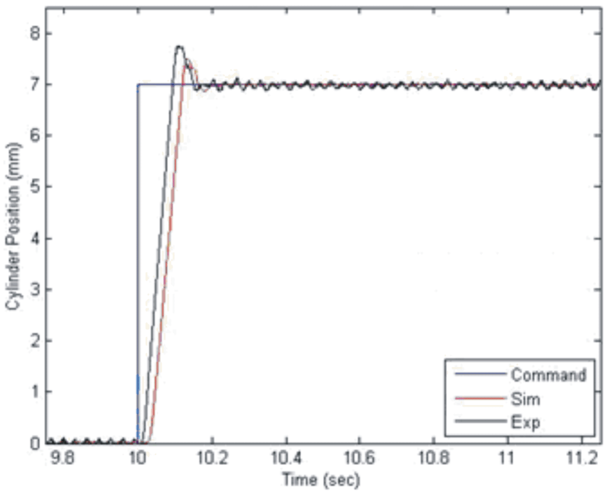

Fig. 11: *Phasing controller response – 7 mm step*

Comparative results for the 3-way controllers are given in Table 6 and Table 7 for 7 mm and 20 mm step inputs, respectively. The responses are compared using experimental data with root mean square of error (RMSE), steady state error (SSE), percent overshoot (% O.S.), maximum values of the response peak, rise time for the response to reach the command value, and the control effort of the valve during the response which is measured by average pulse per second.

Table 6: 3-way controller 7 mm step responses

	3-way Controllers		
Parameter	**Bang-bang**	**Phasing**	**Fuzzy Logic**
RMSE [mm]	0.98	0.91	1.05
SSE [mm]	-0.33	0.09	-0.06
% O. S. [%]	8.38	5.31	20.0
Max Value [mm]	7.68	8.17	7.85
Rise Time [ms]	86.0	87.0	153
Cntl. Efft. [pul/s]	1.50	30.0	1.00

Table 7: 3-way controller 20 mm step responses

	3-way Controllers		
Parameter	**Bang-bang**	**Phasing**	**Fuzzy Logic**
RMSE [mm]	4.07	4.27	4.35
SSE [mm]	0.16	0.06	-0.01
% O. S. [%]	1.40	7.84	12.2
Max Value [mm]	20.7	23.5	20.9
Rise Time [ms]	225	241	308
Cntl. Efft. [pul/s]	1.00	27.0	1.50

To evaluate how the controllers respond to continuingly changing inputs, a sinusoid input is commanded to the controller. Tests were conducted with 0.4 Hz and 2.0 Hz sinusoid inputs. Responses are shown below for the 0.4 Hz sinusoid input.

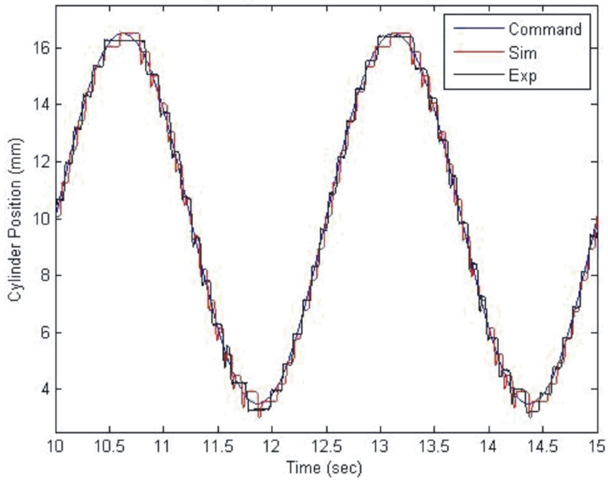

Fig. 12: *Bang-bang controller response – 0.4 Hz sinusoid*

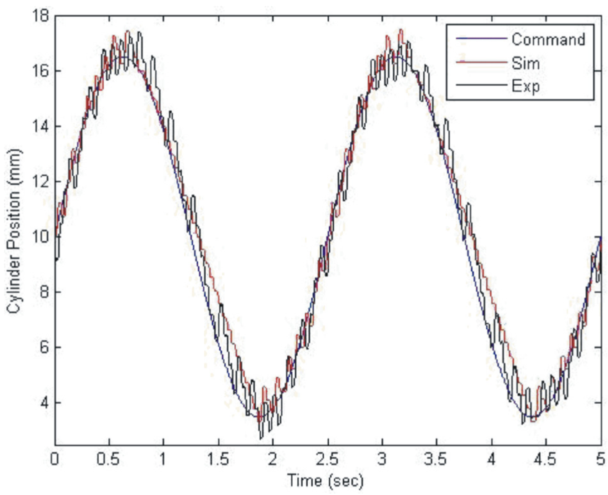

Fig. 13: *Fuzzy logic controller response – 0.4 Hz sinusoid*

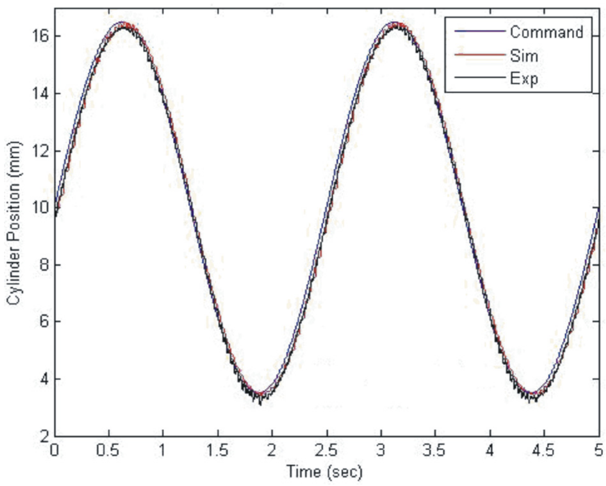

Fig. 14: *Phasing controller response – 0.4 Hz sinusoid*

The presence of the error bands consistent with the bang-bang and fuzzy logic controllers are visible in Fig. 12 and Fig. 13. The stair step response is characteristic of the bang-bang controller as documented in other studies (Beachley, 1998).

The phasing controller shows excellent tracking of the command with a smooth position profile. This smoothness of the response is due to the controller not using an error band. The controller is always correcting for any error as the phase lag between the valves for transition from state to state is directly proportional to the magnitude of the error.

Responses to the 2.0 Hz sinusoid input show similar results between all controllers. The bang-bang controller shows nice agreement to the command with no response phase lag. The fuzzy logic and phasing controllers show responses with slight phase lag, and a small decrease in wave amplitude.

Responses to both the 0.4 Hz and 2.0 Hz sinusoid inputs are compared in Table 8 and Table 9, respectively. For sinusoid inputs, the responses are compared with RMSE and average control effort.

Table 8: 3-way controller 0.4 Hz sinusoid responses

	3-way Controllers		
Parameter	**Bang-bang**	**Phasing**	**Fuzzy Logic**
RMSE [mm]	0.33	0.23	0.75
Cntl. Efft. [pul/s]	8.93	29.4	14.0

Table 9: 3-way controller 2.0 Hz sinusoid responses

	3-way Controllers		
Parameter	**Bang-bang**	**Phasing**	**Fuzzy Logic**
RMSE [mm]	1.25	2.68	3.76
Cntl. Efft. [pul/s]	6.03	13.0	9.50

5.2 2-way Valve Configuration Results

The pulsing and bang-bang controllers that were developed for the 2-way configuration were compared with the same inputs as the 3-way configuration controllers.

As done previously, the 7 mm and 20 mm step inputs are compared using the RMSE, SSE, % O.S., maximum value achieved, rise time, and averaged control effort. Responses for the bang-bang and pulsing controllers are shown for the 7 mm step input in Fig. 15 and Fig. 16, respectively.

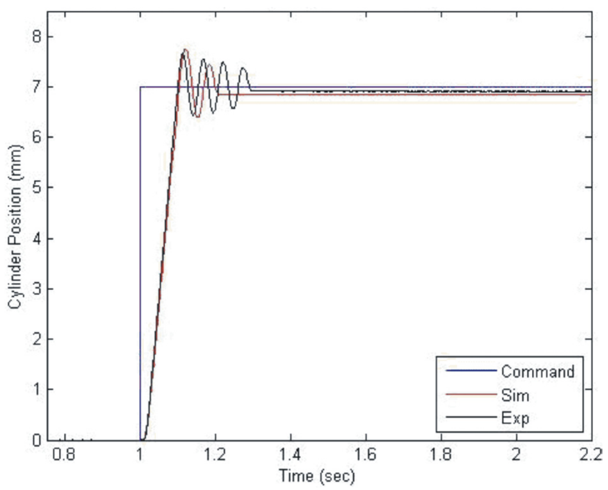

Fig. 15: *Bang-bang controller response – 7 mm step*

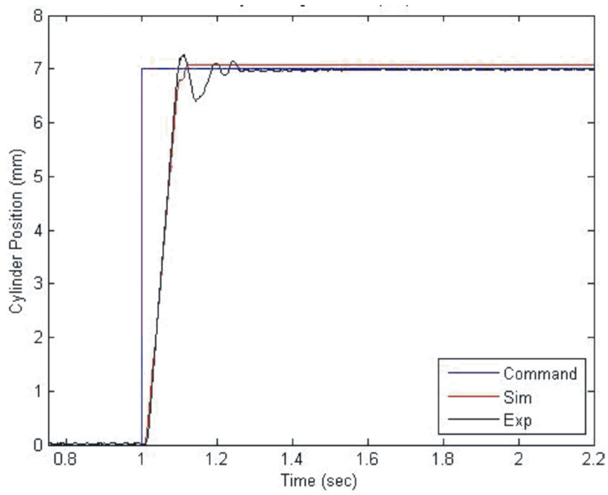

Fig. 16: *Pulsing controller response – 7 mm step*

The bang-bang controller used an error band of ± 0.35 mm for the 2-way configuration. Notice the bang-bang controller shows more oscillation before reaching a steady state condition as compared to the pulsing controller. Additionally, the steady state error for the pulsing controller is significantly less than that of the bang-bang controller.

Responses to the 20 mm step input are similar to the 7 mm step responses. Tests show that the bang-bang controller exhibits three to four oscillation cycles before reaching a steady state condition. The pulsing controller shows one to three oscillations, but at lower amplitude, before reaching a steady state condition.

Table 10 and Table 11 provide a comparison of the controllers for the step inputs. Notice that the pulsing controller shows better performance for all parameters except control effort.

Table 10: 2-way controller 7 mm step responses

	2-way Controllers	
Parameter	**Bang-bang**	**Pulsing**
RMSE [mm]	1.03	1.02
SSE [mm]	-0.10	0.01
% O. S. [%]	12.1	10.7
Max Value [mm]	7.66	7.35
Rise Time [ms]	99.5	97.0
Cntl. Efft. [pul/s]	2.0	5.0

Table 11: 2-way controller 20 mm step responses

	2-way Controllers	
Parameter	**Bang-bang**	**Pulsing**
RMSE [mm]	4.39	4.26
SSE [mm]	-0.25	0.00
% O. S. [%]	3.52	2.14
Max Value [mm]	20.6	20.4
Rise Time [ms]	260	248
Cntl. Efft. [pul/s]	1.5	1.5

It is worth noting from Table 10 and Table 11 that the pulsing controller achieves a very small steady state error. Because of the compression/decompression control theory that is used in the pulsing controller, small steps in position can be accomplished which lead to finer accuracy and lower steady state errors.

The 2-way controllers were evaluated with sinusoid inputs. The responses to a 0.4 Hz sinusoid input with 6.5 mm amplitude is shown in Fig. 17 and Fig. 18 for the bang-bang and pulsing controllers, respectively.

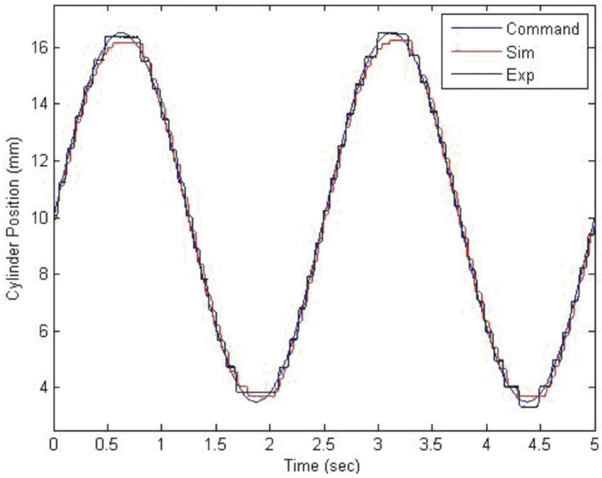

Fig. 17: *Bang-bang controller response – 0.4 Hz sinusoid*

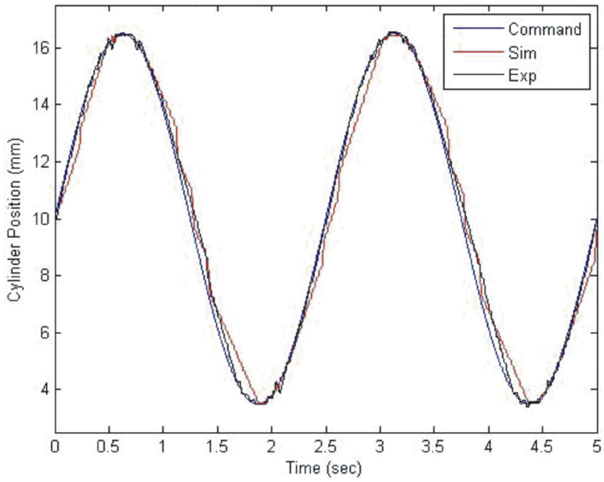

Fig. 18: *Pulsing controller response – 0.4 Hz sinusoid*

Like the 3-way bang-bang controller, the 2-way bang-bang controller shows stair step response due to the presence of the error band. The pulsing controller shows smoother position profile, but some deviation from the command on the falling side of the commanded input.

The responses of both controllers to the 2.0 Hz sinusoid input are similar. The bang-bang controller follows the command nicely, while the pulsing controller shows a slight phase lag on the falling side of the command. The phase lag in the pulsing controller response is due to an averaged slower cylinder velocity when the controller is in the pulsing mode. Therefore with faster sinusoid inputs, the response lags due to the compression/decompression cylinder velocities.

Responses to the 0.4 Hz and 2.0 Hz sinusoid inputs are compared in Table 12 and Table 13. The sinusoid inputs are compared with RMSE and average control effort.

Table 12: 2-way controller 0.4 Hz sinusoid responses

	2-way Controllers	
Parameter	**Bang-bang**	**Pulsing**
RMSE [mm]	0.24	0.16
Cntl. Efft. [pul/s]	7.5	13

Table 13: 2-way controller 2.0 Hz sinusoid responses

	2-way Controllers	
Parameter	**Bang-bang**	**Pulsing**
RMSE [mm]	0.76	1.61
Cntl. Efft. [pul/s]	6.0	9.5

The pulsing controller generally shows slightly higher control effort values than the bang-bang controller. Notice that due to the phase lag with the 2.0 Hz sinusoid command, the pulsing controller shows a higher RMSE value.

6 Sensitivity Analysis

A sensitivity analysis was conducted to understand variations in supply pressure and external loading and how they impact the system response characteristics for each controller. Normal operating conditions as shown in the performance results section are 43 bar supply pressure with no external load. Sensitivity analysis for supply pressure included varying the supply pressure to 24 bar and 58 bar respectively. Adjustment of the pressure relief valve shown in Fig. 7 provided the variable supply pressures to the valves and hydraulic cylinder.

In general for each controller, lower supply pressures cause the controller to show lower overshoots for step inputs and smaller RMSE values for sinusoid inputs. For those controllers that show phase lag in their responses to a 2.0 Hz sinusoidal input, increased supply pressure causes lower RMSE values. This is due to higher cylinder velocities which allow the controllers to adjust to error changes given the 2.0 Hz sinusoid input. Shown in Fig. 19, comparative results to supply pressure variation are given for the 2-way configuration controllers for a 20 mm step input. Notice the direct relation of maximum value with increased supply pressure, but an inverse relationship of RMSE values with supply pressure for the controllers. Fig. 19 also shows the pulsing controller having significantly smaller steady state errors for all testing conditions.

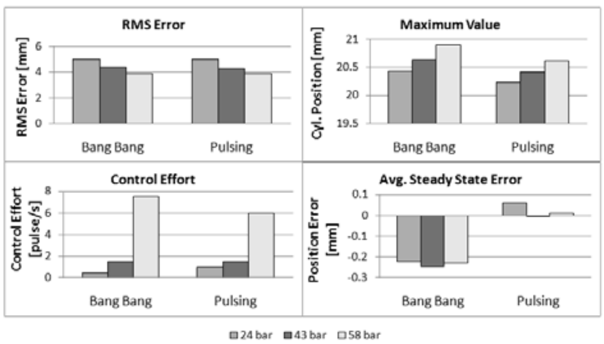

Fig. 19: *20mm step – supply pressure sensitivity analysis*

External loading was evaluated by applying a constant static load to the hydraulic cylinder. The load applied was 10 %, 25 %, and 50 % of the maximum hydraulic force produced at the cylinder with the hydraulic system operating at 43 bar. All external loads were applied in the direction to cause cylinder extension.

The external loading device consisted of a pneumatic cylinder mechanically attached to the hydraulic cylinder. The constant static load was determined by the pneumatic supply pressure in the cap end of the cylinder. To achieve the 10 %, 25 %, and 50 % loading conditions, the pneumatic supply pressure was 131 kPa, 324 kPa, and 648 kPa, respectively. The pneumatic cylinder has an effective area of 2027 mm^2.

The application of the loading device provides the system with additional translational friction and moving mass. These characteristics are lumped together to provide general external loading situations. The external loading device is shown in Fig. 20.

Fig. 20: *External loading device – sensitivity analysis*

For increased external load, the performances of each controller decreased. Step responses show increased maximum values for increases in external loading, while sinusoid inputs show increased RMSE values with increased external loads. It should be noted that because of the 2-way configuration having the advantage of hydraulically locking the cylinder displacement, the effects of increases in external loading are minimized. Once the position of the cylinder causes a closed centre configuration of the valves, the only movement of the cylinder is due to compressibility of the fluid in the chamber volumes and any leakage through the valves.

Fig. 21 gives an example of the external loading sensitivity analysis for the 3-way configuration controllers with a 0.4 Hz sinusoid input command.

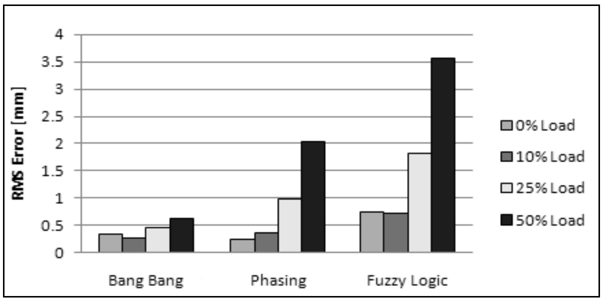

Fig. 21: *0.4 Hz sinusoid – external load sensitivity analysis*

7 Discussion

7.1 3-way Valve Configuration Controllers

When focusing on the 3-way valve configuration, position control is difficult because there are essentially three operating states: cylinder extension, cylinder retraction, or cylinder float. Therefore holding a position is difficult because the float position is not capable of holding load or resisting external forces without action from the valves.

When comparing between each of the 3-way configuration controllers, each has their own benefits and drawbacks. After examining the performance testing and the sensitivity analysis results, the best controller for the 3-way configuration is the phasing controller.

The phasing controller uses a variable phase lag between the valves during transition between the two states of the on/off valves. Because the error determines the magnitude of phase lag from the controller, position responses show small steady state errors for step inputs and small RMSE values for sinusoid inputs. Theoretically, when the error is zero, there would be no phase lag between the valves, thus causing no net movement of the cylinder. Not only does the phasing controller achieve small steady state errors, but also has smooth position profiles for continually moving commands. As can be seen in Fig. 14, the phasing controller shows smooth position profile which leads to reduced cylinder position chatter.

The disadvantages of the phasing controller are found with the control effort of the on/off valves. Since the controller is always adjusting to error, the valves are continually being switched. This reduces the working lives of the on/off valves and, depending on the valve type used, can introduce significant noise to the hydraulic system.

7.2 2-way Valve Configuration Controllers

When investigating the controllers for the 2-way valve configuration, it can be concluded that this configuration allows more flexibility and control theories than the 3-way configuration. As mentioned earlier, the flexibility of the 2-way configuration allows for any control theory developed for the 3-way configuration to be utilized in the 2-way configuration. Additionally, multiple "centre" position can be achieved with the same hardware for the 2-way configuration.

Given the results from the performance testing and the sensitivity analysis, the best performing controller for the 2-way configuration is the pulsing controller. By utilizing the compressibility effects of the hydraulic fluid, small steps in position can be accomplished which lead to finer accuracy and small steady state errors. Responses to step inputs show small overshoots, quick settling times, and low steady state errors. Low frequency sinusoid inputs show smooth profiles with better RMSE values than the bang-bang baseline controller.

The disadvantages associated with the pulsing controller are the control effort and compressibility losses. The pulsing controller shows slightly higher control effort values than the baseline bang-bang controller.

This leads to reduced valve life which may be a significant problem for some applications. Because the fluid in the cylinder chambers is continually being compressed and decompressed, compressibility losses may be significant for certain applications.

7.3 3-way and 2-way Configuration Comparison

Strictly based on the performance characteristics of the best controllers for each configuration, the 2-way valve configuration proves to be superior. Steady state errors are smaller for all step inputs and RMSE values are smaller for sinusoid inputs. The advantage of the controller to allow for a hydraulically locked cylinder position proves to be a significant benefit for the testing conditions investigated in this research.

Even though the 2-way configuration shows better position control performance than the 3-way configuration, it may not be appropriate for all applications. The 2-way configuration has more complexity, cost, and electrical controlling effort. As the 2-way configuration uses four on/off valves, the cost and complexity can nearly double that of the 3-way configuration.

If some applications require simple, low-cost position control with moderate performance, then the 3-way configuration with the phasing controller would be a great fit. On the other hand, if an application requires a fail-safe hydraulically locked position and increased performance, then the 2-way configuration should be used.

Therefore, the determination of which configuration and control strategy that is used for a particular application should be made based on the performance, cost, complexity, and electrical effort required for the application. Additionally, the design tradeoffs associated with each configuration and controller should be investigated and a sound engineering decision should be made between use of 3-way or 2-way on/off valve configurations as well as a comparison with proportional or servo valve configurations.

8 Conclusion

For this research, innovative control strategies have been developed for use with 3-way and 2-way on/off valve control of a hydraulic cylinder. The controllers look to minimize the dependence of the system response to the performance characteristics of the on/off valves. This allows for various on/off valves to be used and especially those that are low cost and lower performance.

The controller that was determined to show the best performance for the 3-way configuration was the phasing controller. The controller allows for small pulse durations to be accomplished by allowing a variable phase lag between the two on/off valves which is proportional to the error of the cylinder position and command.

The best performing 2-way configuration controller was determined to be the pulsing controller. The pulsing controller allows for small steps in position and utilizes the compressibility effects of the hydraulic fluid as an advantage to the control theory.

When comparing the 3-way and 2-way configuration with the best controllers for each configuration, it was determined that the 2-way configuration shows superior position control performance. The ability of the 2-way configuration to achieve multiple centre positions, hydraulically lock the cylinder position, and act as the 3-way configuration gives the 2-way configuration an advantages in flexibility and controllability.

It is recommended that the application determine which configuration is used. As each has its own benefits and drawbacks, the design tradeoffs of the configurations need to be weighed for the desired performance, cost, and complexity of the given application.

Overall, on/off valve position control can be utilized and achieve successful results. The controllers developed minimize the effects of the on/off valve performance on the system position responses. This allows for low cost position control and possible efficiency gains as the throttling losses are minimized.

Nomenclature

α	gas to liquid volume ratio	[--]
A_{cyl}	cylinder area	[m^2]
A_{eff}	effective orifice area	[m^2]
A-T	work port to tank port orifice	[--]
C_d	valve discharge coefficient	[--]
c_{vf}	viscous friction coefficient	[s/m]
E	system bulk modulus	[Pa]
E_l	pure liquid bulk modulus	[Pa]
F_{brk}	cyl. breakaway friction force	[N]
F_{coul}	cyl. coulomb friction force	[N]
F_f	combined cyl. friction force	[N]
F_v	cyl. viscous friction force	[N]
k_1	linear pressure coefficient	[bar/(m^3/s)]
k_2	quadratic pressure coefficient	[bar/(m^3/s)^2]
n	gas specific heat ratio	[--]
ρ	fluid density	[kg/m^3]
p	valve pressure differential	[Pa]
p_a	atmospheric pressure	[Pa]
p_{cyl}	cylinder pressure	[Pa]
Δp	pressure differential	[bar]
P-A	pressure to work port orifice	[--]
p_{sys}	system pressure	[Pa]
q	cylinder flow rate	[m^3/s]
q_i	cyl. chamber flow rate	[m^3/s]
q_{val}	on/off valve flow rate	[m^3/s]
V_c	geometric fluid volume	[m^3]
V_f	fluid volume	[m^3]
V_o	cyl. chamber volume (@x=0)	[m^3]
$\dot{x}$	cylinder velocity	[m/s]

References

Ahn, K. and **Yokota, S.** 2005. Intelligent Switching Control of Pneumatic Actuator Using On/Off Solenoid Valves. *Mechatronics*, Vol. 15, pp. 683-702.

Beachley, N. H., **Fronczak, F. J.**, **Lucier, P. E.**, **Lumkes, J.** and **Sosnowski, T.** 1998. Pump/Motor Displacement Control Using High-Speed ON/OFF Valve. *SAE International Off-Highway and Powerplant Congress and Exposition*.

Branson, D. T., **Lumkes, J.**, **Wattananithiporn, K.** and **Fronczak, F. J.** 2008. Simulated and Experimental Results for a Hydraulic Actuator Controlled by Two High-Speed ON/OFF Solenoid Valve. *International Journal of Fluid Power*. Vol. 9 (2), pp. 47-56.

Chen Y., **Huang, Z.** and **Liu, S.** 2004. Automobile Active Suspension System with Fuzzy Control. *J. Cent. South Univ. Technol.* Vol. 11 no. 2, pp. 206-209.

Gao, Y., **Gu, Y.** and **Zhang, Q.** 2008. Study on the Integrated Programmable Multifunction E/H Valve. *National Conference of Fluid Power*. Vol. 11 no. 4, pp. 307-311.

Gentile, A., **Giannoccaro, N. I.** and **Reina, G.** 2002. Experimental Tests on Position Control of a Pneumatic Actuator Using ON/OFF Solenoid Valves. *IEEE ICIT'02*, pp. 555-559.

Ivantysynova, M. and **Grabbel, J.** 2005**.** An Investigation of Swash Plate Control Concepts for Displacement Controlled Actuators. *International Journal of Fluid Power*. Vol. 6 no. 2, pp. 19-36.

Li, P. Y., **Li, C. Y.** and **Chase, T. R.** 2005. Software Enabled Variable Displacement Pumps. *Proceedings of ASME-IMEC'05*.

Liu, S., Zhou, J., and **Zhu, M.,** 2000. Optimal Control of Hydraulic Position System Employing High Speed ON/OFF Solenoid Valve. *J. Cent. Univ. Technol.* Vol. 7 no. 1, pp. 46-48.

Liu, S. and **B. Yao.** 2002. Energy-saving control of single-rod hydraulic cylinders with programmable valves and improved working mode selection. *SAE Transactions – Journal of Commercial Vehicle*, SAE 2002-01-1343, pp. 51-61.

Long, G. 2009**.** Comparison Study of Position Control with 2-way and 3-way High Speed ON/OFF Electrohydraulic Valves. Purdue University. Master of Science in Engineering Thesis. pp. 1-169.

Manhartsgruber, B., **Mikota, G.** and **Scheidl, R.** 2005. Modelling of a Switching Control Hydraulic System. *Mathematical and Computer Modelling Of Dynamical Systems*. 11 no. 3, pp. 329-344.

Mauer, G. F. 1995. A Fuzzy Logic Controller for an ABS Braking System. *IEEE Transactions on Fuzzy Systems.* Vol. 3 no. 4, pp. 381-388.

Paul, A. K., **Mishra, J. K.** and **Radke, M. G.** 1994. Reduced Order Sliding Mode Control for Pneumatic Actuator. *IEEE Transactions on Control Systems Technology.* Vol. 2 no. 3, pp. 271-276.

Scheidl, R., **Garstenauer, M.** and **Manhartsgruber B.** 2000. Switching Type Control of Hydraulic Drives - a Promising Perspective for Advanced Actuation in Agricultural Machinery. Society of Automotive Engineers, Warrendale, 37-47, 9-2000, ISBN: 0-7680-0650-3.

Van Varseveld, R. B. and **Bone, G. M.** 1997. Accurate Position Control of a Pneumatic Actuator Using ON/OFF Solenoid Valves. *IEEE/ASME Transactions on Mechatronics.* Vol. 2 no. 3, pp. 195-204.

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