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Robust control of the pressure in a control-cylinder with direct drive valve for the variable displacement axial piston pump

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Abstract: To achieve energy-efficient hydraulic systems, an accurate flowrate and pressure required by a load should be supplied to the systems. The discharge flow from the variable displacement swash-plate type axial piston pump is mainly determined by the angle of the swash-plate and it is adjusted by controlling the pressure in the control cylinder, which is a mechanical regulator for the swash-plate.

In this paper, a robust pressure control system of the control cylinder using a direct drive valve is proposed to control the discharge flow from the variable displacement swash-plate type axial piston pump precisely. In order to design a robust pressure control system, a mathematical model of a pressure control system is derived and system parameters are identified by using the signal-compression method. Also, the sliding mode controller is designed based on the identified mathematical model to guarantee the control performance of the pressure control system.

Experiments verified the satisfactory performance of the proposed pressure control system that uses a direct drive even with non-linearity and uncertainties such as flow characteristics, unknown discharge coefficient of the valve orifice, variation of the bulk-modulus, leakage, and disturbance induced by the pressure fluctuation in the pressure control system.

Keywords: variable displacement axial piston pump with the electric valve, control cylinder, robust pressure control, direct drive valve, sliding mode control

1 INTRODUCTION

In general, hydraulic systems that consist of a fixed displacement pump and a relief valve are widely used in industrial fields. In these systems, when the system pressure is higher than the setting pressure of the relief valve, surplus flow returns to the reservoir. Therefore, these kinds of hydraulic systems have comparatively low efficiency. On the other hand, the hydraulic systems that consist of a variable displacement swash-plate type axial piston pump with the built-in pressure regulator are much more efficient than conventional hydraulic systems. However, these systems can not actively adjust the discharge flow, that is, when the system pressure exceeds the setting pressure of the pump regulator, discharge flow is changed rapidly to the minimum discharge flow. These hydraulic systems are more energy-efficient than the conventional ones. However, these systems still have relatively low efficiency because they cannot actively adjust the discharge flow according to the load variation.

In order to improve the efficiency of hydraulic systems much further, load-sensing hydraulic servo systems consisting of a variable displacement swash-plate type axial piston pump with an electro-hydraulic valve, the LS valve, have been developed. The discharge flowrate of this pump can be actively adjusted by changing the swash plate angle according to load demands. The angle of the swash plate is determined by controlling the pressure of the control cylinder using an electro-hydraulic valve. Therefore,
to achieve extremely highly efficient load-sensing hydraulic servo systems, the pressure in the control cylinder should be controlled precisely, even under uncertainties such as the variation of effective bulk modulus and leakage from the control cylinder and disturbance caused by load pressure.

Lee employed a servo valve to control the pressure in the control cylinder and applied the model based control/loop transfer recovery technique as a robust control algorithm [1]. Akers and Lin also use a servo valve with optimal control theory to control the pressure in the control cylinder [2]. Although the servo valve can precisely adjust the pressure in the control cylinder, it is very expensive and complicated to use. Also, the maintenance problems are very severe. Therefore, the use of a servo valve for the control of pressure in the control cylinder is impractical. Chiang and Chien proposed a parallel control scheme for energy savings and velocity control of the hydraulic cylinder with the use of a proportional valve to control the swash-plate angle [3]. The proportional valve is comparatively highly resistant to contamination of the working fluid but has poor response performance under 40 Hz. In practice, it is not appropriate to use the proportional valve as a control valve to control the pressure in the control cylinder.

In this paper, the direct drive valve (DDV) which uses the permanent magnet linear motor is used to control the pressure in the control cylinder [4]. The DDV is a reasonable option to use for the control of the pressure in the control cylinder in comparison to the servo valve with respect to price. In particular, as the DDV drives the spool directly and does not use the nozzle-flapper system, it is a significant problem in the contamination of working fluid. Also, the response performance of DDV is in the same class as the servo valves. In addition, a pressure transducer and the feedback controller are inherently built in a DDV.

However, the experiments showed that transient and steady state responses are undesirable when the DDV is used for controlling the discharge flow of the variable displacement axial piston pump with load variation. Therefore, a precise and robust pressure control system is designed in this research.

In order to design a robust pressure control system, the pressure control system is modelled mathematically and unknown system parameters are identified. In this paper the focus was given to the robust control of the pressure in the control cylinder with simple mathematical model. The mathematical modelling was carried out as simply as possible to design the robust controller. Then a sliding mode controller is designed based on the identified mathematical model to achieve the desired performance and robustness. The performance and robustness of the pressure control system of a control cylinder based on the sliding mode control scheme is evaluated by experiments.

2 SYSTEM CONFIGURATION AND PROBLEM FORMULATION

In this section, the characteristics of the pressure control system of the control cylinder for variable displacement swash-plate type axial piston pump with the DDV are described. Figure 1 shows a

![Schematic diagram of the pressure control system of the control-cylinder](image-url)
schematic diagram of the pressure control system of a control cylinder. The pressure control system consists of an auxiliary hydraulic power unit, relief valve, which is required to set the system load, PC with data acquisition board, a variable displacement swash-plate type axial piston pump, and a DDV for controlling the pressure in the control cylinder. The supply pressure to the DDV is set as 3 MPa, which is adjusted by the relief valve installed on the auxiliary hydraulic power unit. The maximum discharge flow from the variable displacement swash-plate type axial piston pump is 28 l/min. The maximum value of the swash plate angle is 19°. The pressure in the control cylinder is fed back to the DDV by a pressure transducer inherently built in the DDV.

In order to verify the relationship between the pressure in the control cylinder and the swash plate angle, the discharge flow of the variable displacement swash-plate type axial piston pump is measured. Figure 2 shows the relationship between the pressure in the control cylinder and discharge flow from the pump. As shown in Fig. 2, the discharge flowrate from the variable displacement axial piston pump is decreased according to the increase of pressure in the control cylinder. This result shows that the angle of the swash plate is controlled by the pressure in the control cylinder.

In this system, although there are many factors that affect the dynamic response of the variable displacement axial piston pump, the only way to control the discharge flowrate is to control the pressure in the control cylinder. Therefore, to achieve the demanded discharge flowrate, the precise control of the pressure in the control cylinder is very important.

2.1 Control cylinder

The control cylinder is the mechanical regulator that is used to control the angle of the swash plate [4]. Figure 3 shows the various forces that act on the control cylinder. The main factors affecting the
moment of the swash plate are the pressure force acting on the pump piston and the inertia of the pump piston [5]. As shown in Fig. 3, the pressure force acting on the pump piston and the force due to the inertia of pump piston are directly transferred to the control cylinder by the swash plate. Also, the moment of the swash plate is affected by the load pressure [5–7]. The moment generated by the load pressure is transferred to the control cylinder by the swash plate. Therefore, the moment of the swash plate acts as disturbance to the pressure in the control cylinder.

Based on the fundamental studies on the bulk modulus, the bulk modulus of the working-fluid changes greatly in the low-pressure range as shown in Fig. 4 [8–10]. The pressure range in the control cylinder is comparatively low, between 0.3 to 1.2 MPa as shown in Fig. 2. Therefore, the bulk modulus in the control cylinder also changes significantly, and this change effects the hydraulic natural frequency. The hydraulic natural frequency also varies with stroke of the control cylinder because the total volume of the control cylinder is very small compared with the rest of the system. Accordingly, the change of the bulk modulus and the hydraulic natural frequency according to the stroke of the control cylinder can be considered as an uncertainty against the pressure control system of the control cylinder.

The mechanical regulators of the hydraulic pumps used today are quite energy consuming, and they are stabilized mainly through a leakage in the regulator [4]. In many pumps, energy losses in the regulator are greater than those in the rest of the pump. The total amount of leakage from a control cylinder in the variable displacement pump used in this research was also considerable, as shown in Fig. 5. Also, the amount of leakage from a control cylinder varies due to a stroke of the control cylinder. In the view point of control, the leakage can act as an

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**Fig. 4** Variation of effective bulk modulus according to pressure range [8, 11]

**Fig. 5** Leakage of control-cylinder

**Fig. 6** Schematic diagram of DDV [12]
uncertainty against the pressure control system of the control cylinder.

2.2 Direct drive valve

In this study, the DDV is used for the control of the pressure in the control cylinder. Figure 6 shows the cross-sectional view of the DDV and the specification of DDV is represented in Table 1.

The DDV consists of a linear motor, which is directly attached to a three-way function valve spool, the pressure-transducer, and the Amp. The DDV has an inherent closed-loop pressure controller \([12]\), and the load pressure is fed back to DDV by the pressure transducer. Therefore, inherent closed-loop feedback controller can compensate for any pressure drop and non-linear flow characteristic caused by the orifice to guarantee a linear output pressure of the DDV.

Figure 7 shows the experimental results which were obtained with use of only the inherent feedback controller built in the DDV. In the experiments, the load pressures were changed by using the relief valve, which set the maximum pressure of the pump to 0, 3.5, and 7 MPa. The applied reference pressures in the control cylinder were 0.4, 0.6, 0.8, and 1 MPa based on Fig. 2. All of the measured signals were acquired, sent, displayed, and saved by the MATLAB/Simulink Real-Time Windows Target which ran on a PC in the real time. As shown in Fig. 7, the high-frequency ripples of approximately 450 Hz appears in all experimental results. These high-frequency ripples were caused by the revolution of nine pistons of the variable displacement axial piston pump. In general, the ripple characteristic is an inherent characteristic of the measured pressure of a hydraulic system. As shown in Fig. 7, the steady state error and undesirable transient response such as the occurrence of an overshoot and irregular settling time due to the load pressure are observed. These undesirable characteristics are caused by the fixed control gain of the feedback controller implemented in DDV.

Accordingly, it is concluded that the feedback control system of DDV with a fixed control gain cannot effectively compensate system uncertainties and disturbances according to the change of the load pressure. As described in section 2.1, the change of

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<th>Table 1 Specification of DDV [12]</th>
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the bulk modulus and the hydraulic natural frequency according to the stroke of the control cylinder, leakage around the control cylinder, can be the possible system uncertainties, although it is not measured or estimated actually. It is hard to achieve a desirable pressure control performance with only the inherent feedback controller of the DDV. Therefore, a robust controller is essential to improve the performance and robustness of the pressure control system of the control cylinder.

3 SYSTEM MODELLING

In order to design a robust control system, the pressure control system of the control cylinder is modelled mathematically. In the current paper, the focus was given to the robust control of the pressure in the control cylinder with simple mathematical model. The mathematical modelling was carried out as simple as possible to design the robust controller readily.

Figure 8 shows a schematic diagram of the pressure control system of the control cylinder for mathematical modelling.

The dynamic response of a spool in the DDV can be simply described as a first-order system with time constant $T_1$ as follows

$$K_w v = \frac{dx_v}{dt} + C_1 x_v$$

(1)

where $K_w = K_v \frac{1}{T_1}$, $K_v$ is force constant of the linear motor, $v$ is input voltage into the linear motor of the DDV, $x_v$ is the spool displacement of the DDV, and $C_1 = \frac{1}{T_1}$ respectively.

The linearized orifice equation for the DDV with leakage can be represented as equation (2) [13]

$$Q_c = K_q x_v - K_c p_c$$

(2)

where $Q_c$ is flow through the control cylinder, $K_q$ and $K_c$ are the valve flow gain and valve flow-pressure coefficient of DDV respectively, and $p_c$ is the pressure in the control cylinder.

By applying the continuity equation to the control cylinder, equation (3) can be obtained

$$Q_c - C_{L1} p_c = A_c \frac{dx_c}{dt} + A_c x_{c0} \frac{dp_c}{\beta_e dt}$$

(3)

where $A_c$ is area of the control cylinder, $x_c$ is the displacement of the control cylinder, $x_{c0}$ is displacement of the control cylinder at operating point, $C_{L1}$ is leakage coefficient of the control cylinder, $V_c$ is the total volume of the control cylinder chamber, and $\beta_e$ is the effective bulk modulus of the working fluid.

For an elementary analysis, the displacement of the control cylinder can be related to the spool displacement of DDV by equation (4) [13]

$$A_c \frac{dx_c}{dt} = K_{eq} x_v$$

(4)

where $K_{eq}$ is the equivalent flow gain of the DDV and $K_{eq} < K_q$. 

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**Fig. 8** Schematic diagram of the pressure control system of control-cylinder
Based on equations (1) to (4), the block diagram for the pressure control system of the control cylinder with a built-in feedback controller can be written, as shown in Fig. 9.

From the block diagram of the pressure control system for the control cylinder, the transfer function can be derived as

\[
p_c(s) = \frac{K \omega_n^2}{s^2 + 2\zeta \omega_n s + \omega_n^2}
\]

where

\[
2\zeta \omega_n = \frac{\beta_e}{A_k x c_0} \left( C_1 (C_{L1} + K_c) + \frac{A_k x c_0}{\beta_e} \right)
\]

\[
\omega_n = \frac{\beta_e}{A_k x c_0} \left( C_1 (C_{L1} + K_c) + K_p (K_q - K_{eq}) \right)
\]

\[
K = \frac{K_p (K_q - K_{eq})}{C_1 (C_{L1} + K_c) + K_p (K_q - K_{eq})}
\]

In general, the unknown system parameters of the linear elements in a non-linear system can be identified by the signal compression method (SCM). In practice, hydraulic servo systems exhibit significant non-linear behaviour due to the variation of the orifice coefficient and uncertainties such as the line losses, leakages, and parameter variations of the working fluid [13]. Some of these non-linearities and uncertainties have to be neglected in the modelling process because the SCM can only identify the linear part of the system. The unknown parameters of the linearized mathematical model for pressure control system of the control cylinder in equation (3) are identified by the signal compression method [14].

The test signal that has the same amplitude up to 50 Hz in the frequency domain is applied to the pressure control system of the control cylinder by constructing a closed-loop P control system to obtain a more accurate equivalent impulse response for the precise identification of unknown parameters. Figure 10 shows comparison of the frequency response of the pressure control system between the closed-loop nominal model and the closed-loop actual system with P control.

From the identified nominal model for the closed-loop pressure control system with P control, the parameters of the pressure control system can be acquired by eliminating the effect of P controller mathematically.

After eliminating the effect of P controller mathematically, the values of the natural frequency \( \omega_n \), damping coefficient \( \zeta \), and system gain \( K \) of the pressure control system are finally identified as 15.2, 0.56, and 2.32 respectively.

The identified system parameters were verified on the time domain and the frequency domain. On the time domain, the step responses of the nominal model and actual system are compared as shown in Fig. 11. Even though the transient responses show some difference because of neglected non-linearity and an effect of zero which can exist in actual system, the settling time and steady state responses are comparatively well matched. The frequency domain validation was done by applying sinusoidal inputs of various frequencies such as 5, 10, 15, 18,
and 23 rad/s, with a bias of 1.0, and magnitude of 0.4. Figure 12 shows a comparison of the obtained bode-plots of the pressure control system based on the experiments and nominal model. The experimental result according to the various sinusoidal inputs is represented with tetragonal points and the result of the nominal model is represented with the solid line.

As shown in Fig. 12, the frequency response of the nominal model is close to that of the actual system. In the modelling process, it is difficult to satisfy time domain response characteristics such as the rising time, maximum overshoot, and settling time and frequency domain response characteristics such as DC gain and bandwidth simultaneously with the use of the simple second-order nominal model. Furthermore, the system bandwidth is related to the rising as well as settling time. Therefore, in this research, the main focus was given to the settling time and steady state response characteristic rather than the rising time. Accordingly, although transient responses show some difference compared with actual system, the simple second-order nominal model, which is identified by using SCM, was used in design of the sliding mode controller.

The validity of using this nominal model in the sliding mode controller design is verified through the control experiments in section 5.

4 DESIGN OF SLIDING MODE CONTROL SYSTEM

Even though the parameters of the pressure control system of the control cylinder are identified by SCM, the identified model can only represent the linear elements of the pressure control system. In addition, there still exist the uncertainties and disturbance in the pressure control system of the control cylinder. In order to achieve a robust control performance of the pressure control system of the control cylinder, the sliding mode control (SMC) scheme is suggested in this study.

Figure 13 shows the block diagram of the pressure control system of the control cylinder. As shown in Fig. 13, the SMC system is added to the inherent built-in feedback control system of the control cylinder with the DDV.

The sliding surface $s$ for the design of the SMC system is defined as

$$ s = \lambda e + \dot{e} $$

(6)

![Fig. 13](image)

**Fig. 13** Block diagram of control cylinder pressure control system with augmented SMC system
where, the tracking error of the pressure $e = p_{cd} - p_c$, and $\lambda$ is a positive design parameter. Then, the derivative of the sliding surface can be obtained by

$$\dot{s} = \lambda e + \dot{e} \quad (7)$$

When the state trajectories of the system reach the sliding surface, $s = 0$ and $\dot{s} = 0$. Hence it follows from equation (4) and $\dot{s} = 0$ that the equivalent control law in the sliding mode is given by equation (8) [15, 16]

$$u_{eq} = \frac{1}{K_{\omega_n^2}} \left( -\lambda \dot{p} + 2 \zeta \omega_n \dot{p} + w_n^2 p + D_s \right) \quad (8)$$

The semi-continuous SMC, which can solve the chattering problem of the control input, is designed by choosing the derivative of the switching surface as follows

$$\dot{s} = -Ds - K_s \text{sgn}(s) \quad (9)$$

Then, the sliding mode control law can be selected as

$$u = \frac{1}{K_{\omega_n^2}} \left( -\lambda \dot{p} + 2 \zeta \omega_n \dot{p} + w_n^2 p + D_s + K_s \text{sgn}(s) \right) \quad (10)$$

where, $D$ and $K_s$ are positive design parameters.

To analyse the reachability, the Lyapunov candidate is chosen as follows

$$V = \frac{1}{2} s^2 \quad (11)$$

The derivative of the Lyapunov function is expressed by equation (12)

$$\dot{V} = s \dot{s} = -Ds^2 - sK_s \text{sgn}(s) < 0 \quad (12)$$

This implies that the trajectory of the system is globally driven onto a specified switching surface $s = 0$ despite the uncertainties and external disturbance [15, 16].

### 5 CONTROL EXPERIMENTS AND DISCUSSION

In the experiments, three kinds of the load pressures – 0, 3.5, and 7 MPa – are applied by using a relief valve to set the system load. Also, the applied reference pressures in the control cylinder are 0.4, 0.6, 0.8, and 1 MPa. All of the measured signals are acquired, sent, displayed and saved by the MATLAB/Simulink Real-Time Windows Target which was run on a PC in real time. The high-frequency ripples of approximately 450 Hz also appeared in all experiments in this section.

Figure 14 shows the experimental results of the pressure control system of the control cylinder using the designed sliding mode controller. In the control
input of equation (10), a differential term of the pressure measured by the built-in pressure transducer of DDV is included. Therefore, it is necessary to use a first-order low-pass filter $G_f(s) = \frac{\tau}{\tau s + 1}$ to guarantee the stability of the control system for the differential term in equation (10), where $\tau$ is the time constant of the first-order low-pass filter.

Despite the system uncertainties neglected in the process of the mathematical modelling and the considerable amount of the leakage from the control cylinder, the SMC system has better performance and robustness than the feedback control system of DDV shown in Fig 7. Table 2 represents the comparison of the time domain performances between the feedback control system of DDV and the SMC system. As shown in Table 2, the settling time of the SMC system is approximately two times shorter than that of the feedback control system of DDV. Also, steady state error is twenty four times less than that of the feedback control system of DDV.

As shown in Fig. 11 and Fig. 14(c), the undesirable transient response occurs when the load pressure and reference pressure in the control cylinder is comparatively large. Even though the reason concern on the occurrence of this undesirable transient response is unknown, it can be conjecture that this phenomenon is due to feedback torque which act on the swash plate angle to reduce. Even in this case, the undesirable transient response is improved when the SMC is used.

### 6 CONCLUSION

To improve the efficiency of hydraulic servo systems with a variable displacement pump, it is necessary to control the displacement flow of the pump precisely. In the swash plate type variable displacement pump, the swash plate angle is determined by the pressure of the control cylinder. In order to achieve the high efficiency of hydraulic servo systems, the pressure in the control cylinder must therefore be controlled precisely.

The pressure control system of the control cylinder has significant uncertainties neglected in the process of the mathematical modelling and a considerable amount of the leakage through the gap between the sleeve and control cylinder. On the other hand, the DDV is more economical than the servo valve with respect to price and maintenance. In addition, the DDV has superior performance compared with the proportional valve. The DDV has an inherent feedback controller in the valve body. However, the pressure control system of the control cylinder using an inherent feedback controller does not yield desirable performance and robustness because the built-in feedback controller cannot compensate the disturbance force due to the torque of swash plate and load pressure. Therefore, a robust controller is required to realize a precise pressure control system of the control cylinder.

In order to design a robust control system, mathematical modelling was carried out based on theoretical analysis and the simple second-order nominal model was derived. Second, system parameters of the simple second-order nominal model were identified by the signal compression method.

In addition, as a robust controller, the sliding mode controller was designed with the use of the simple second-order nominal model, which is available to readily design the sliding mode controller. Experiments verified that the proposed control system yields satisfying performance and robustness against system uncertainties and disturbance due to the torque of the swash plate and load pressure. Therefore, it is identified that in the design process of sliding mode controller, the use of the simple second-order nominal model, which is identified by using SCM, is valid from the satisfying performance of experimental results.

### ACKNOWLEDGEMENT

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### APPENDIX

#### Notation

- $A_c$: pressurized area of control cylinder (m$^2$)
- $C_1$: reciprocal of time constant for valve spool dynamics
- $C_{L1}$: leakage coefficient of the control cylinder (m$^3$/s Pa)
- $K$: gain of transfer function for pressure control system
- $K_c$: valve flow-pressure coefficient of DDV (m$^3$/s Pa)
- $K_{eq}$: equivalent valve flow gain of DDV (m$^2$/s)
- $K_v$: force constant of the linear motor (N/V)
- $K_p$: proportional gain
- $K_q$: valve flow gain of DDV (m$^2$/s)
- $p_c$: pressure of the control cylinder (Pa)
- $p_{cd}$: desirable pressure of the control cylinder (Pa)
- $p_s$: supply pressure to DDV (Pa)
- $p_t$: return pressure from DDV (Pa)
- $Q_c$: flow through the cylinder (m$^3$/s)
- $v$: input voltage into the linear motor of DDV (V)
- $V_c$: total volume of the control cylinder (m$^3$)
- $w$: flow area gradient of DDV (m$^2$/m)
- $x_c$: displacement of control cylinder (m)
- $x_{c0}$: operating point for displacement of control cylinder (m)
- $x_v$: displacement of the spool of DDV (m)
- $\beta_e$: effective bulk modulus of the working fluid (Pa)
- $\rho$: density of the working fluid (kg m$^3$)
- $\zeta$: damping ratio of transfer function for pressure control system
- $\omega_n$: natural frequency of transfer function for pressure control system (rad/s)