Controller design for a nozzle-flapper type servo valve with electric position sensor

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Abstract. The control performance of hydraulic systems is basically influenced by the performance of the electro-hydraulic servo valve incorporated in a hydraulic control system. In this study, the authors propose a control design to improve the control performance of a servo valve with a non-contact eddy current type position sensor. A mathematical model for the valve is obtained through an experimental identification process. A PI-D control together with a feedforward (FF) control is applied to the valve. To further improve the dynamic response of the servo valve, an input shaping filter (ISF) is incorporated into the valve control system. Finally, the effectiveness of the proposed control system is verified experimentally.

Nomenclature

- $d_e$: disturbance,
- $K_p$, $K_I$ and $K_D$: proportional, integral and differential gain
- $p_s$, $p_t$: supply port pressure and tank port pressure
- $u$, $u_d$: control inputs shown in Figure 5
- $x_s$: spool position
- $x_{ref}$, $x_{ref}^*$: reference inputs shown in Figure 5

1. Introduction

Electro-hydraulic servo valves have been used for controlling hydraulic systems for a long time. The servo valve used in this study is a nozzle-flapper type two-stage servo valve [1]. The valve has a feedback spring mechanism for spool position feedback. The servo valve used in this study additionally has a non-contact type electric displacement sensor as shown in Figure 1. Electro-hydraulic servo valves with electric displacement sensor are currently available commercially [1].

In the case of servo valve with an electric position sensor, control system designers can actively intervene in the control action of the servo valve control loop, unlike in the case of the valve without an electric position sensor. However, the information on the control scheme of the valves with additional electric position sensor is not open to general hydraulic control engineers, and also the performance limitations of the valves are not known. These points are the motivation for this study.
Prior to the controller design for the servo valve, the mathematical model of the servo valve is obtained through a preliminary experiment of the frequency response characteristics of the valve without electric feedback control.

To achieve a robust reference following control while rejecting the effects of the disturbances, a reference following controller (including PI-D and feed-forward controller) is designed using the pole placement method and the zero placement method [2, 3]. Also, it is known that the input shaping filter (ISF) which is positioned outside the feedback control loop in control systems can modify the entire loop transfer function, without deteriorating the action of the control loop for disturbance rejection [4, 5]. So far, however, studies applying ISF for improving the control performance of hydraulic servo valves are hardly found. In this paper, therefore, the authors propose a control design using ISF for improving the control performance of the servo valve in the high-frequency range. The effectiveness of the proposed control design is verified experimentally.

2. Mathematical modeling of the servo valve

2.1. Overview of the servo valve controlled system

The experimental system consists of a servo valve, a servo-amplifier and a controller (personal computers). The servo valve is a nozzle-flapper type valve with two stage structure, which was manufactured by SG Servo Co. in Korea. Figure 1 shows the structure of the servo valve under study, where an eddy current type displacement sensor (Keyence AH-305) is installed. Figure 2 shows photographs of the key parts of the experimental system. Simulink Real-Time (SLRT) [6] is applied to perform control actions and collect experimental data on the controller (two personal computers). For the implementation of SLRT, the host PC and target PC are connected by a LAN cable.

Figure 1. Structure of the servo valve.

Figure 2. Photograph of the experimental system of the servo valve.
2.2. Modeling of the servo valve

In this study, a transfer function with standard second order form is obtained as Equation (1) through a preliminary experiment shown in Figure 3 using the servo valve, considering its' applicability to control system design. In the experiment, the supply current \( i = 0.4 \, i_r \sin \omega t \) \( (i_r = 40 \text{ mA}) \), the supply pressure is 70 bar.

\[
G_{v\text{-open}}(s) = \frac{\omega_n^2}{s^2 + 2\zeta\omega_n s + \omega_n^2} = \frac{250000}{s^2 + 900s + 250000}
\]

\[ \left[ \zeta \approx 0.9, \ \omega_n \approx 500 \right] \] (1)

![Figure 3. Bode diagram of the servo valve without electric position sensor. \( (i = 0.4 \, i_r \sin \omega t, i_r = 40 \text{ mA}) \)](image)

3. Controller design

Block diagram of the control system for the servo valve is shown in Figure 4. In the control system, proportional integral and derivative controller (PI-D), feedforward (FF), and input shaping filter (ISF) is applied to get a robust system in the servo valve. Design process for the controller is described in detail in the previous research paper of the corresponding author [3].

3.1. PI-D control

In Figure 4, the transfer function between the disturbance input \( U_d(s) \) and the spool position \( X_s(s) \) is

\[
\frac{X_s(s)}{U_d(s)} = \frac{G_{v\text{-open}}(s)}{1 + G_{v\text{-open}}(s)[G_{\text{PI}}(s)+G_{\text{FF}}(s)]}
\] (2)

By defining \( G_{\text{PI}}(s) = K_p + K_I / s \), \( G_{\text{FF}}(s) = K_D s \), and using equation (1) for \( G_{v\text{-open}}(s) \), \( \frac{X_s(s)}{U_d(s)} \) is described as
\[ X_s(s) = \frac{G_{v\text{-open}}(s)}{U_d(s)} = \frac{G_{v\text{-open}}(s)\left(G_{PI}(s)+G_D(s)\right)}{1+G_{v\text{-open}}(s)\left(G_{PI}(s)+G_D(s)\right)} \]

\[ = \frac{\omega_n^2s}{s^3 + \left(2\omega_n + \beta \omega_n^2k_p\right)s^2 + \left(\omega_n^2 + \beta \omega_n^2k_p\right)s + \beta \omega_n^2k_I} \]

\[ = \frac{250000s}{s^3 + (900 + 250000k_D)s^2 + \left(250000 + 250000k_D\right)s + 250000k_I}. \]

By placing the poles of Equation (3) to \([-1700 + 10j, -1700 - 10j, -255]\) so as to satisfy the design specifications on disturbance rejection, the PI-D gains are determined as

\[ K_p = 13.62, \quad K_i = 2601 \quad \text{and} \quad K_D = 0.0109 \quad (4) \]

**Figure 4.** Block diagram of the control system.

### 3.2. FF-PI-D control

From Figure 4, the transfer function \(X_s(s)/X_r(s)\) is described as

\[ \frac{X_s(s)}{X_r(s)} = \frac{\left(G_{PI}(s)+G_{FF}\right)G_{v\text{-open}}(s)}{1+\left(G_{PI}(s)+G_D(s)\right)G_{v\text{-open}}(s)} \]

Let’s describe \(G_{FF}(s)\) as

\[ G_{FF}(s) = c_1s + c_2. \]

From Equation (1), (4) and (6), we obtain the following equation.

\[ \frac{X_s(s)}{X_r(s)} = \left[\frac{13.62 + \frac{2601}{s}}{s^3 + 3625s^2 + 3.65 \times 10^6 s + 6.5 \times 10^6}\right]250000s \]

By applying the zero placement method [2], that is, by equating the numerator and the last three terms of the denominator of Equation (7), \(G_{FF}(s)\) can be determined to be

\[ G_{FF}(s) = 0.014s + 1. \]

Therefore, Equation (7) is described as
\[ \frac{X_s(s)}{X_r(s)} = \frac{3625s^2 + 3.65 \times 10^6 s + 6.5 \times 10^8}{s^3 + 3625s^2 + 3.65 \times 10^6 s + 6.5 \times 10^8} \] (9)

3.3. ISF-FF-PI-D control

The control performances of servo valves are generally shown by bandwidth in a Bode diagram. To reform the Bode diagram of an FF-PI-D controlled system into the desired form, in this study, an input shaping filter (ISF) is appended to the FF-PI-D controlled system designed already. The transfer function of the ISF adopted here has the form of a lead-lag compensator, and is described as

\[ G_{ISF}(s) = \frac{(T_s + 1)}{(T_s + 1)}. \] (10)

This ISF was designed so as to reform the Bode diagram of an FF-PI-D control system into the desired form. To the servo valve control system, \( G_{ISF}(s) \) was designed with \( T_1 = 6.36 \times 10^{-3} \), \( T_2 = 2.27 \times 10^{-3} \).

3.4. Low-pass filter

A digital low pass filter is designed as

\[ G_{filter}(s) = \frac{1}{(T_s + 1)}. \] (11)

where the value of \( T = 1/1257 \), so as the break frequency of the filter at -3 dB to be 200Hz.

4. Control performance of the servo valve

In the experiment, the reference input is 40% of the rated current input, the supply pressure \( (p_s) \) is 70 bar and the tank pressure \( (p_t) \) is 0 bar. To the system, a dither signal with 320 Hz and 1 mA amplitude is applied.

4.1. Oscillation phenomenon in step response

Figure 5 shows the experimental result of the step response of the open-loop system without the electric position feedback. The experimental result shows that the valve operates normally and the oscillation in the spool displacement with the magnitude of about due to dither signal is observed.

Figure 6 shows the experimental result of the step response of the system without the filter and with the filter under P control respectively. In the case of without filter, oscillation with big magnitude is observed in the control input \( u \) and output \( x \) signals. After applying the low pass filter, the oscillation could be moderated to the level of the signal in the open loop-loop control shown in Figure 5.

4.2. Frequency response

Figure 7 shows the experimental results of frequency response of the servo valve under the PI-D, FF-PID and the ISF-FF-PI-D control. The low pass filter with the break frequency of 200 Hz was applied to the system in order to avoid undesirable amplification of dither signal in the closed-loop control.

Then, appending FF control to the PI-D control corrected the gain deviation from 0 dB in the high frequency range and the phase delay in the high frequency range. Further adding the ISF.

When the open loop control and the PI-D control in Figure 6 are compared, it is noted that the PI-D control contributed to a substantial improvement in the band-width [from 60 Hz to 120 Hz], but caused excessive rise of the gain and the phase delay in the high frequency range.

Then, appending FF control to the PI-D control corrected the gain deviation from 0 dB in the high frequency range and the phase delay in the high frequency range. Further adding the ISF to the FF-PI-
D control contributed to the gain adjustment towards 0dB in the high frequency range without large damage of the phase angle.

The control performances in the various control schemes are summarized in Table 1.

**Table 1. Summary of the control performances of the servo valve.**

| Items                          | open loop | PI-D | FF-PI-D | ISF-FF-PI-D |
|-------------------------------|-----------|------|---------|-------------|
| rise time [ms]*               | 7         | 7    | 6       | 7           |
| Overshoot [%]*                | 0         | 8.03 | 4.95    | 6           |
| frequency [Hz] at −90° phase angle | 80       | 72   | 94      | 92          |

* Evaluated when \( x_{ref} = 0.4 x_e (1 - e^{-\tau}) \), \( \tau = 2 \text{ ms} \).

[\( \tau \) : the time for the response to attain 63.2% of the final value.]

**Figure 5.** Step response of the servo valve without electric position feedback signal (open-loop control).
Figure 6. Step response of the servo valve with electric position signal under closed loop control (with/without the digital filter).

Figure 7. Frequency response under the PI-D, FF-PID and ISF-FF-PI-D control in the servo valve.
5. Conclusions
1) Control performance of a nozzle-flapper type servo valve with/ without an electrical position sensor was investigated by experiments.
2) Dither signal in the closed-loop control of the servo valve may induce significant increase of oscillation amplitude. However, this symptom can be easily overcome by applying a digital filter.
3) Applying FF (feed-forward) & ISF (input shaping filter) control to the PI-D controlled system of the servo valve can remarkably improve the frequency response.

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