High Precision Position Control of Electro-Hydraulic Servo System Based on Feed-Forward Compensation

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Abstract: The study is focused on an electro-hydraulic servo system which is a position control system. It is a non-minimum phase system when it was discretized with a certain sample time. To improve its tracking performance and extend its bandwidth, based on invariance principle, feed-forward compensation is developed by pole-zero placement theory for the system. The task is accomplished by transforming instable zero of the system into pole of the fitted closed-loop transfer function, forming the zero of feed-forward compensator and completing the compensation of the instable zero for the closed-loop system. The simulation and experimental results show the validity of the analytical results and the ability of the proposed algorithm to efficiently improve the system tracking performance and greatly extend system bandwidth.

Key words: Feed-forward compensation, invariance principle, non-minimum phase system, pole-zero placement, system bandwidth, tracking performance

INTRODUCTION

Hydraulic control systems are widely used in many main industrial fields, including aerospace, metallurgy, transportation, machine, marine technology, modern scientific experimental device and weapon control (Li, 1990; Yao et al., 2007). Compared with others types of drives, they have many distinct advantages, such as developing a comparatively small device with much larger torque, high precision, higher respond speed, higher stiffness and higher force-to-weight ratio (Yao et al., 2008).

As for the control algorithm aspects for electro-hydraulic servo system, classical PID controller has been employed in real applications. The regulation law of the conventional PID controller is effective for quite a number of plants, especially for linear time-invariant system. The control performance is dependent on the parameters of PID controller. However, its parameters are set under certain operation condition. Because nonlinearities and uncertainty usually occur in the electro-hydraulic servo system, when the classical PID controller is used for the system, it only gives mediocre control performance.

Adaptive position control based on Radial Basis Function (RBF) neural network and LQ controller was developed by Knohl and Unbehauen (2000) for a hydraulic system controlled by a 4/3 way proportional valve. Its structure is complex and it has a problem of computation burden, thus its real-time performance can not be guaranteed. Adaptive sliding control by Guan and Pan (2008a) and nonlinear adaptive robust control by Guan and Pan (2008b) were presented for electro-hydraulic system with nonlinear unknown parameters to improve the tracking performance. Self-tuning fuzzy PID controller in which the fuzzy controller was used to update the three parameters of the PID controller was developed by Wu et al. (2004). Its key issue is how to design the membership function and it needs to know the system model. Adaptive control scheme based on Popov criterion was proposed by Yao et al. (2006). But it has large computation. Adaline neural network using LMS adaptive filtering algorithm was proposed by Yao et al. (2007), whose task was accomplished by adjusting the weights of the network using LMS algorithm when phase delay existed between the input and its corresponding output, and the reference input was weighted in such a way that it makes the system output track the input efficiently, thus the weighted input signal was added to the control system such that the output phase delay was cancelled leaving the desired signal alone. To cancel amplitude attenuation and phase delay, Yao et al. (2011b) developed a network using normalized LMS adaptive filtering algorithm, whose weights were on-line updated according to the estimation error between the desired input and the weighted feedback, the updated weights thus were copied to the input correction. The two methods can greatly improve sinusoidal tracking performance. However, this method is only applicable to fixed-point sine wave. An adaptive tracking control with reference
model being 1 was presented by Wang et al. (2005) to achieve zero phase tracking. The initial value of integral adaptive law has great influence on the system output.

With the increase of test system requirements, it requires high performance both in amplitude frequency bandwidth and in phase frequency bandwidth, especially for large frequency bandwidth system, for example, electro-hydraulic servo shaking table, in which its frequency bandwidth may be required to be bigger than 100 Hz. Though three-variable control can extend system frequency bandwidth for an electro-hydraulic servo shaking table (Yao et al., 2010, 2011a), this improvement is limited by the servo valve frequency bandwidth and the hydraulic natural frequency. The methods can improve the system performance, especially the tracking performance, but it can not widen frequency bandwidth (Yao et al., 2006, 2007; Knohl and Unbehauen, 2000; Guan and Pan, 2008 a; Guan and Pan, 2008 b; Wu et al., 2004).

In the study, to improve the tracking performance of electro-hydraulic servo system and extend its bandwidth, based on invariance principle, feed-forward compensation is developed by pole-zero placement theory for the system. High precision position control is accomplished by pole-zero placement theory for the system. High precision position control is accomplished by transforming instable zero of the system into pole of the fitted closed-loop transfer function, forming the zero of feed-forward compensator and completing the compensation of the instable zero for the closed-loop system. The proposed control scheme is validated by the simulation and experimental results.

**SYSTEM DESCRIPTION**

The valve-controlled hydraulic actuator is used as a force amplifier. Very little force is forced to position the valve, but a large force is obtained. The hydraulic unit is relatively small, which makes its application very attractive. Figure 1 is the hydraulic system, which is comprised of a symmetric cylinder, a symmetric servo valve and a load force. The motion of the valve regulates the flow of oil to either side of the main cylinder. An input motion of a few thousands of an inch results in a large change of oil flow (John and Constantime, 1993). The resulting difference in pressure on the main piston causes motion of the output shaft. The oil flowing in is supplied by a hydraulic source that maintains a constant high supply pressure p_s, and the oil on the opposite side of the piston flows into the drain at low return pressure p_b.

The load-induced pressure p_L is the difference between the pressures on each side of the main piston:

\[ p_L = \frac{F_L}{A} = p_i - p_s \]

where \( A \) is the effective area of the cylinder piston, \( F_L \) the external force, and \( p_i, p_o \) are the input and output pressure of the cylinder, respectively.

The pressure drop \( \Delta p \) across the orifice is a function of the supply pressure \( p_s \) and the load-induced pressure \( p_L \). Since \( p_s \) is assumed to be constant, the flow equation for \( q \) is a function of valve displacement \( x_v \) and \( p_L \):

\[ q = f(x_v, p_L) \]

where, \( q \) is the rate of flow of hydraulic fluid through the valve. The differential rate of flow \( dq \), expressed in terms of partial derivatives, is:

\[ dq = \frac{\partial q}{\partial x_v} dx_v + \frac{\partial q}{\partial p_L} dp_L \]

If \( q, x_v \), and \( p_L \) are measured from zero values as reference points, and if the partial derivatives are constant at the values they have at zero, the integration of Eq. (2) gives the flow equation of fluid entering the main cylinder:

\[ q = \left( \frac{\partial q}{\partial x_v} \right)_o x_v + \left( \frac{\partial q}{\partial p_L} \right)_o p_L \]

\[ = K_q x_v - K_c p_L \]

where,

\[ K_q = \left( \frac{\partial q}{\partial x_v} \right)_o \]

is the flow-gain coefficient

\[ K_c = \left( \frac{\partial q}{\partial p_L} \right)_o \]

is flow-pressure coefficient

The oil flow can also be written as:

\[ q = \frac{q_{v_i} + q_{v_o}}{2} \]
where \( q_{v1}, q_{v2} \) is the input/output oil flow of the cylinder, respectively. The continuity equations of the cylinder are given as:

\[
\dot{p}_i = \frac{\beta}{V_i}(q_{v1} - A\dot{y}) \\
\dot{p}_e = \frac{\beta}{V_e}(A\dot{y} - q_{v2})
\]

where \( \beta \) is the bulk modulus of the hydraulic oil and \( \dot{y} \) is the velocity of the piston, \( \beta \) is defined as the ratio of incremental stress to incremental strain. Thus:

\[
\beta = \frac{\Delta p}{\Delta V / V}
\]

where \( V \) is the effective volume of fluid under compression.

From Eq. (1), (4), (5) and (6), the flow continuity equation of the cylinder is:

\[
q = A\dot{y} + \frac{V_i}{4\beta}\dot{p}_l
\]

where \( V_i \) is the total oil volume of the cylinder, \( V_i = V_i + V_o \).

The balance of the force at the sliding carriage is:

\[
Ap_\text{m}y + B\dot{y} + F_L = \frac{m\dot{y} + B\dot{y} + F_L}{4\beta}
\]

where \( m \) is the total mass of the piston and the load, \( B \) the equalized viscous damping coefficient, \( \dot{y} \) the acceleration of the piston.

Using Eq. (3), (7), (8) and applying the Laplace transformation to the resulting third-order differential equation results (Yao et al., 2007):

\[
Y = \frac{K_c X_y - K_i (1 + \frac{s}{\omega_h}) F_L}{s \left( \frac{4\beta t^2}{mV_i} + \frac{2\omega_h}{\omega_h^2 + s + 1} \right)}
\]

where

\[
\omega_h = \sqrt{\frac{4\beta t^2}{mV_i}} \\
\xi = \frac{K_c - \sqrt{\beta m}}{2A \sqrt{\frac{V_i}{V}}} \\
\omega_h = \frac{4\beta t^2}{V_i}
\]

**Feed-forward compensation:** In order to improve the control precision of random vibration PSD (Power Spectral Density) replication, Guan (2007) improved the correction algorithm of drive signals PSD and the generation of time domain drive signals in the classical algorithm. The drive signals PSD iteration algorithm with different step length in different frequency band was presented to raise the rate of convergence. The method of generating the drive signal in time domain by filtering a series of independent white noise with designed FIR (Finite Impulse Response) filter was presented. The FIR filter was designed by the Parks-McClellan algorithm with the information contained in the in the drive signal PSD. To avoid the time domain randomization procedure, the execution time of the algorithm was reduced. The improved PSD iteration algorithm can improve the control precision, but it has great computation burden, because it needs to off-line calculate FFT (Fast Fourier Transform) and IFFT (Inverse Fast Fourier Transform) in time domain and in frequency domain to generate the drive signal. What more, for a elastic load, its convergence may not be ensured.

For a servo control system, the feed-forward compensation scheme can extend the system frequency bandwidth and improve the system tracking performance. Generally, the feed-forward compensation scheme is based on the invariance principle (Lin and Tian, 2005). In other words, the feed-forward compensation loop is designed to be the inverse of the system closed-loop transfer function such that the transfer function of the compensated system is unity. However, when the closed-loop system is a non-minimum phase system, this method will face difficulty. In the invariance principle (Lee et al., 2001), the unstable zeros of a non-minimum phase system become the poles of the feed-forward compensation loop, leading to instability of the feed-forward compensation part. In real applications, with the increase of the sample frequency, discrete-time systems discretized from continuous systems will be non-minimum phase systems.

In this study, for such non-minimum phase systems, zeros are introduced into the feed-forward compensator to compensate the unstable zeros of the closed-loop system (Wang et al., 2005). Its basic theory is the pole-zero displacement theory.

The feed-forward compensation scheme can be described by Fig. 2. \( G_c(s) \) is the closed loop system transfer function including the controller, \( r \) the input command. Let \( G_c(z^{-1}) \) be a stable discrete-time system which is discretized from a continuous closed loop.
System, \( G_c(s) \) and \( G_c(z) \) has non-minimum phase zeros with unity static gain. Thus \( G(z) \) can be written as:

\[
G(z) = \frac{z^{-d}N_u(z^{-1})N_r(z^{-1})}{D(z^{-1})}
\]

(10)

where, \( N_u(z^{-1}) = n_0 + n_1z^{-1} + \ldots + n_5z^{-5}, n_0 \neq 0 \). \( N_u(z^{-1}) \) contains zeros of \( G(z) \) on the unit circle and instable zeros of \( G(z) \) beyond the unit circle, while \( N_r(z^{-1}) \) includes all stable zeros of \( G(z) \) within the unit circle.

**Theorem** (Liu, 2003; Xia and Meng, 1995): Let \( H(z) = N_u(z)N_r(z^{-1}) \) which meets:

\[
\angle H(e^{-j\omega}) = 0
\]

and

\[
\left| H(e^{-j\omega}) \right|^2 = \text{Re}^2[N_u(e^{-j\omega})] + \text{Im}^2[N_u(e^{-j\omega})] \forall \omega \in R
\]

where, \( T \) is the sample time. Thus the feed-forward compensator can be represented by Liu (2003):

\[
G_f(z^{-1}) = \frac{z^{-d}N_u(z^{-1})D(z^{-1})}{N_r(z^{-1})[N_u(1)]^2}
\]

(11)

Currently, the Proportional-Integral-Derivative (PID) algorithm is the most common control algorithm used in industry applications. In PID control, a feedback variable and a set-point should be specified. The feedback variable is the system parameter, which the designer wants to control, and the set-point is the desired value for the parameter. The PID controller generates a controller output, for example, valve position. The controller applies this output value to the system, which in turn drives the feedback variable toward the set-point value. The PID controller calculates the controller action as:

\[
u(t) = K_p\left[\int_0^t e(t)dt + T_i \frac{de(t)}{dt}\right]
\]

where \( e \) is the error signal obtained by comparing the set-point to the feedback variable. \( K_p \) is the controller gain. \( T_i \) is the integral time, which is also called the reset time. \( T_d \) is the derivative time or the rate time.

**Simulation results**: To validate the proposed feed-forward compensation scheme, the system was used for an electro-hydraulic servo shaking table is adopted as the plant (Yao et al., 2006). Its transfer function is:

\[
G_p(s) = \frac{1803158.187}{s^7 + 478555s + 24586.24}
\]

For real electro-hydraulic servo system, P controller is mostly applied. Before feed-forward compensation is applied, the system closed-loop bode plot is shown in Fig. 3 with proportional coefficient \( K_p = 0.3 \). From the figure, it can be seen that there are small -3 dB amplitude and -90° phase frequency bandwidth, especially for the amplitude frequency bandwidth. To improve the system frequency bandwidth and its tracking performance, the feed-forward compensation in Fig. 2 is applied.

During the simulation, system sample time is 1 ms. After it is discretized with the sample time, the closed-loop transfer function is:
The system closed-loop bode plot after feed-forward compensation is shown in Fig. 5. It can be seen that the system amplitude and phase bandwidth are all greatly enhanced. This is very useful for the applications where wide frequency bandwidth is required for electro-hydraulic servo system.

When the input is \(0.1 \sin(20\pi t)\) m, the system response is shown in Fig. 6. Figure 7 is the response corresponding to the command: \(0.1 \sin(16\pi t) + 0.05 \sin(30\pi t) + 0.02 \sin(40\pi t)\) m. From the two figures, the compensated system has good tracking performance.

**Experimental results:** The experimental setup of an electro-hydraulic servo system which is an asymmetric cylinder controlled by a MOOG D792 servo valve is used to validate the proposed control scheme. The system configuration is shown in Fig. 8. The hydraulic power supply provides a high supply pressure \(p_s\) of 5 MPa as shown in Table 1, which gives the system parameters. \(r\) is the command input, and \(y\) is the feedback displacement. The servo controller compares the feedback signal from sensor with the input to produce a command signal to drive the servo valve, which adjusts the flow of pressurized oil to move the hydraulic cylinder until the desired position is attained (Yao et al., 2011a).

![Fig. 5: System closed-loop bode plot after feed-forward compensation](image1)

![Fig. 6: System sinusoidal response](image2)

![Fig. 7: System sinusoidal response with hybrid frequencies](image3)

### Table 1: Experimental parameters

<table>
<thead>
<tr>
<th>Item</th>
<th>Value</th>
<th>Unit</th>
</tr>
</thead>
<tbody>
<tr>
<td>Cylinder stroke</td>
<td>1350</td>
<td>mm</td>
</tr>
<tr>
<td>Cylinder piston</td>
<td>125</td>
<td>mm</td>
</tr>
<tr>
<td>Cylinder rod</td>
<td>90</td>
<td>mm</td>
</tr>
<tr>
<td>Oil supply pressure</td>
<td>5</td>
<td>MPa</td>
</tr>
<tr>
<td>Equivalent external load</td>
<td>5500</td>
<td>kg</td>
</tr>
<tr>
<td>Sample time</td>
<td>2</td>
<td>ms</td>
</tr>
</tbody>
</table>

### Table 2: Sinusoidal response analysis

<table>
<thead>
<tr>
<th>Frequency (Hz)</th>
<th>Amplitude attenuation (dB)</th>
<th>Phase delay (deg)</th>
</tr>
</thead>
<tbody>
<tr>
<td>0.1</td>
<td>0</td>
<td>-5.616</td>
</tr>
<tr>
<td>0.5</td>
<td>-0.5691</td>
<td>-17.28</td>
</tr>
<tr>
<td>1.0</td>
<td>-0.6144</td>
<td>-21.6</td>
</tr>
<tr>
<td>1.7</td>
<td>-0.6600</td>
<td>-30.6</td>
</tr>
<tr>
<td>5.0</td>
<td>-3.7352</td>
<td>-64.8</td>
</tr>
</tbody>
</table>
Hydraulic power supply

Servo controller → Servo valve → Hydraulic cylinder → Load

Sensor

Fig. 8: System configuration of the electro-hydraulic servo system

Signal generator

0.18s + 1
0.06s + 1

K

D/A

D/A conditioner → Servo valve

A/D conditioner → Displacement sensor

Fig. 9: Block diagram of the control system

(a) Input frequency is 0.1 Hz

(b) Input frequency is 0.5 Hz
Fig. 10: Sinusoidal response at different frequencies

(c) Input frequency is 1 Hz

(d) Input frequency is 1.7 Hz

(e) Input frequency is 5 Hz

Fig. 10: Sinusoidal response at different frequencies
(a) Input frequency is 0.1 Hz

(b) Input frequency is 0.5 Hz

(c) Input frequency is 1 Hz
The control system is shown in Fig. 9. The signal generator is used to generate the command input. The saturation unit is to make the control signal in the range of \([-10, 10]\) V, which is also the volt range of D/A (digital to analog, D/A). The D/A and A/D (analog to digital, A/D) conditioner are used to condition signals for servo valve driving signal and for A/D board, respectively.

The sinusoidal responses are shown in Fig. 10, and its analysis is shown in Table 2, from which it can be seen clearly that with the increase of input frequency, the amplitude attenuation and the phase delay become more serious.

With the frequency response data in Table 2, the system closed-loop transfer function can be identified by the MATLAB command, invfreqs, which is useful in converting magnitude and phase data into transfer functions. The identified system closed-loop transfer function is:

\[
\frac{Y(s)}{R(s)} = \frac{-1482s + 7674}{s^4 + 0.0703s^3 + 1023s^2 - 23.64s + 4.78 \times 10^4}
\]

The proposed control scheme is then applied to the electro-hydraulic servo system, and its experimental results are shown in Fig. 11 plotted with the command and the feedback signals. From the results, it can be clearly seen that the feedback signal nearly coincides well with the command input at all five frequencies, thus the compensated system has good tracking performance, improving its frequency characteristics.

**CONCLUSION**

Though PID controller is widely used in electro-hydraulic servo system, because of its limitations, good tracking can not be achieved in every operation condition. To enhance system bandwidth and improve the tracking performance, feed-forward compensator, which is based on pole-zero placement theory, is developed. When the electro-hydraulic servo system is discretized, it becomes a non-minimum phase system. The zeros of the feed-forward compensator are obtained by transforming the system instable zeros to the poles of the fitted transfer function of the closed-loop system, compensating the
instable zeros of the closed-loop system, and high precision tracking performance is achieved.

For a real system, its amplitude and phase frequency characteristics can be established by sine sweep tests, thus the feed-forward compensator can be obtained based on the fit of the closed-loop system, so the proposed control scheme can be applied to real application.

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