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HYDRAULIC SERVO SYSTEM USING A FEEDBACK LINEARIZATION CONTROLLER AND DISTURBANCE OBSERVER - SENSITIVITY OF SYSTEM PARAMETERS -

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ABSTRACT

In this study, the control performance of a hydraulic servo system with a feedback linearization compensator is investigated. The focus of this study is set on the quantitative investigation of the effects(sensitivities) of disturbances and system parameters' variation on control performances of the hydraulic control system. Finally, verifies the efficacy of a disturbance observer to overcome the control performances deterioration due to system parameters' variations, disturbances in the control system.

KEY WORDS

Hydraulic Servo System, Feedback Linearization, Disturbance Observer, Parameters Sensitivity

INTRODUCTION

There are many nonlinearities in hydraulic servo systems like nonlinear pressure-flow characteristics in valves, hysteresis and null point drift in valves, nonlinear driving force from asymmetric cylinder. If hydraulic servo systems are controlled using linear controllers, which are most common so far, it is not easy to achieve satisfactory control performances, as the linear controllers have to be dimensioned conservatively to ensure stability.

As a countermeasure to overcome this difficulty due to hydraulic systems' nonlinearities, applications of feedback linearization controllers to hydraulic control systems has been tried[1-4]. But the research works with the controllers were not fully satisfactory in most cases, because the researchers did not consider the effects of disturbances and parameters' variations in systems.

This study applies state feedback controllers incorporating a feedback linearization compensator to a hydraulic servo system. This study focuses on the effects of system parameters' variations, disturbances on the control performances of the hydraulic servo system with a feedback linearization compensator. Finally, considers the applicability of a disturbance observer to overcome the control performances deterioration due to system parameters' variations, disturbances in the control system with a feedback linearization compensator.

MODELING THE OBJECT HYDRAULIC SYSTEM

System description

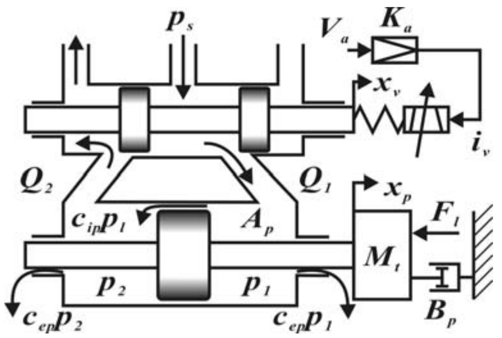


Figure 1 Overview of the hydraulic system considered

Fig. 1 represents the object hydraulic servo system in this study. The main parts of the system are an electro-hydraulic servo-valve, a servo-cylinder and a mass(an inertia load). In the figure, p_s : supply pressure, p_1 and p_2: pressure inside the cylinder, Q_1 and Q_2: flowrate in the servo valve, i_v: electric current in the servo valve.

Basic equations

Flowrate Q_l in the servo valve is described as

$$Q_l = K_{sv} i_v \sqrt{p_s - \frac{i_v}{|i_v|} p_l} \tag{1}$$

where $K_{sv}\left(= Q_{ro} / \left(i_{vr}\sqrt{p_s}\right)\right)$ is a proportional constant, Q_{ro} is flowrate when $i_v = i_{vr}$ and $p_l = 0$, that is the rated current under no load condition. Eq. (1) is effective when the flow in the valve is in steady state.

Considering the piston staying in the mid point of the symmetric cylinder, the continuity equation in the cylinder is given by:

$$Q_l = A_p \frac{dx_p}{dt} + C_{tp} pl + \frac{V_t}{4\beta_c}\frac{dp_l}{dt} \tag{2}$$

where A_p : piston area, x_p ; piston displacement, C_{tp} : leakage coefficient in the cylinder, β_e : effective bulk modulus of oil in the cylinder, V_t : total volume of oil in both chamber of the cylinder.

The equation of motion of the combined body of the piston and the load is shown as

$$A_p p_l = M_t \frac{d^2 x_p}{dt^2} + B_p \frac{dx_p}{dt} + F_l \tag{3}$$

where M_t : mass of the combined body, B_p : viscous frictional coefficient, F_l : external force to the piston.

The spool position in the servo valve is described as

$$x_v \cong k_v\, i_v \tag{4}$$

FEEDBACK LINEARIZATION - STATE FEEDBACK CONTROLLER (FL-SFC)

This section describes the design procedure applying a feedback linearization technique to the object system to overcome the nonlinerities of the system. Differentiating the Eq. (3) with respect to time yields:

$$\frac{d^3 x_p}{dt^3} = \frac{A_p}{M_t}\frac{dp_l}{dt} - \frac{B_p}{M_t}\frac{d^2 x_p}{dt^2} \tag{5}$$

By substituting Eq. (2) and (3) to Eq. (5), we obtain the following equation.

$$\frac{d^3 x_p}{dt^3} = \frac{4A_p\beta_e}{M_t V_t} Q_l - \left(\frac{4{A_p}^2\beta_e}{M_t V_t} - \left(\frac{B_p}{M_t}\right)^2\right)\frac{dx_p}{dt} - \left(\frac{4A_p C_{tp}\beta_e}{M_t V_t} + \frac{A_p B_p}{{M_t}^2}\right) p_l + \frac{B_p}{{M_t}^2} F_l \tag{6}$$

Eq. (6) is rearranged as Eq. (7), by separating terms including Q_l and terms having no relation with Q_l.

$$\frac{d^3 x_p}{dt^3} = f\left(\frac{dx_p}{dt}, p_l, F_l\right) + B\, Q_l \tag{7}$$

with

$$f(\frac{dx_p}{dt}, p_l, F_l) = -\left(\frac{4{A_p}^2\beta_e}{M_t V_t} - \left(\frac{B_p}{M_t}\right)^2\right)\frac{dx_p}{dt} - \left(\frac{4A_p C_{tp}\beta_e}{M_t V_t} + \frac{A_p B_p}{{M_t}^2}\right) p_l + \frac{B_p}{{M_t}^2} F_l \tag{8}$$

$$B = \frac{4A_p\beta_e}{M_t V_t} \tag{9}$$

The non-linearities can be canceled out by using the following equation with regard to $\hat{Q}_l$.

$$\hat{Q}_l = \left\{\phi - f\left(\frac{dx_p}{dt}, p_l, F_l\right)\right\} \Big/ B \tag{10}$$

where, $\hat{Q}_l$: calculated flowrate, ϕ : control input of feedback linearized system.

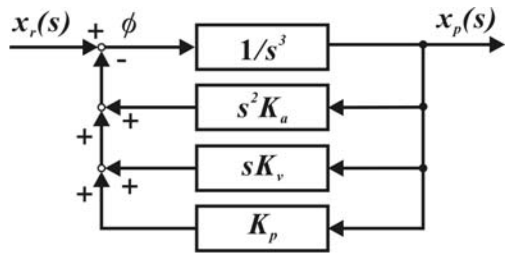


Figure 2 Block diagram for the linearized system using FL-SFC

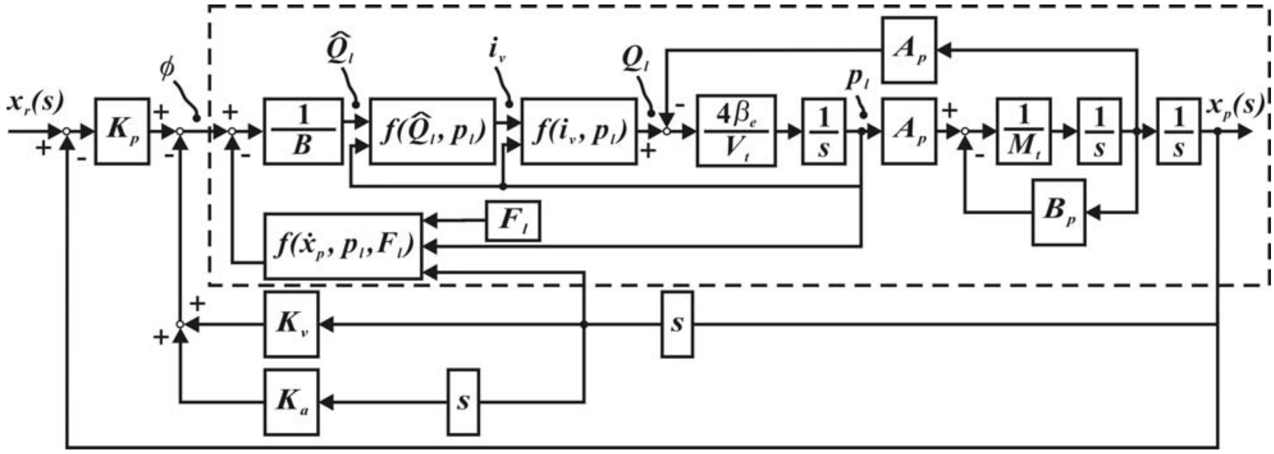


Figure 3 Block diagram of the hydraulic control system using the FL-SFC [the block surrounded with dashed line can be simplified as by the feedback linearization]

Thus, if the flowrate computed by the Eq. (10) is supplied to the system continuously, the condition $d^3x_p/dt^3 = \phi$ can be satisfied. Thereby, a linerized relationship between ϕ and x_p is obtained. Then, a state feedback controller can be applied to the system, as shown in Fig. 2. And also the representative poles in the control system can be placed at the predetermined positions by adjusting the closed loop control gains.

From Fig. 2, a transfer function shown as Eq. (11) is obtained, which is the nominal model of the control system with the feedback linearization compensator.

$$\frac{x_p(s)}{x_r(s)} = H_n(s) = \frac{K_p}{s^3 + K_a s^2 + K_v s + K_p} \qquad (11)$$

Fig. 3 shows the block diagram of the control system with the FL-SFC. The state feedback control gains $\mathbf{K}$ are computed by placing the representative poles at $-38 \pm 24i$ ($\omega_n \doteq 45$ rad/s, $\zeta \doteq 0.85$), and other pole is placed at -38×5 on the real axis.

SENSITIVITY ANALYSIS OF THE SYSTEM WITH THE FEEDBACK LINEARIZATION COMPENSATOR

The object hydraulic control system

Fig. 4 shows the photo of the object hydraulic control system. Physical parameters values are listed in Table 1.

Sensitivity of the compensated system

Fig. 5 shows a simplified block diagram of the block surrounded with dashed line in Fig. 3. $G_p(s)$ and $G_c(s)$ in the figure are the control object and the feedback linearization compensator part respectively. Also, ① and ② depict null point drift in the control valve, and external disturbances.

Fig. 6 shows the frequency response characteristics of the subsystem described in Fig. 5 under two different physical situations; one situation is when external force of 1000 N is applied, the other is when null point drift of the control valve of +2% is applied. Data in Fig. 6 were obtained from simulations by the help of the Control Design™, MATLAB/SIMULINK®. Results in Fig. 6 shows that the linearized model ($1/s^3$) can be affected in a large scale by the external force and valve null point drift.

Table 2 shows the variation of poles of the object system in Fig. 3 under the variation of system parameters β_e, B_p and M_t. The data were given from simulations using MATLAB/SIMULINK®, and

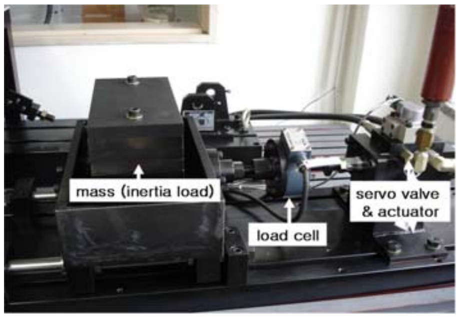


Figure 4 Photo. of the experimental equipment

Table 1 Pysical parameters values in the object system

$A_p\,[\mathrm{m^2}]: 0.00094$, $C_{tp}\left[\frac{\mathrm{m^3}/s}{\mathrm{N/m^2}}\right]: 0$, $B_t\left[\frac{\mathrm{N}}{\mathrm{m/s}}\right]: 15000$,
$\beta_e\left[\frac{\mathrm{N}}{\mathrm{m^2}}\right]: 1.4\times10^9$, $F_l\,[\mathrm{N}]: 0$, $V_t\,[\mathrm{m^3}]: 6\times10^{-4}$,
$K_a\left[\frac{\mathrm{mA}}{\mathrm{V}}\right]: 6$, $K_{LT}\left[\frac{\mathrm{V}}{\mathrm{m}}\right]: 200$, $M_t\,[\mathrm{kg}]: 118$, $i_{vr}\,[\mathrm{mA}]: 15$,
$K_{sv}\left[\frac{\mathrm{m^3}/s}{\mathrm{mA}\sqrt{\mathrm{N/m^2}}}\right]: 8.0\times10^{-9}$, $\zeta_v: 0.84$, $\omega_v\,[\mathrm{rad/s}]: 760$,
$p_s\,[\mathrm{bar}]: 35$

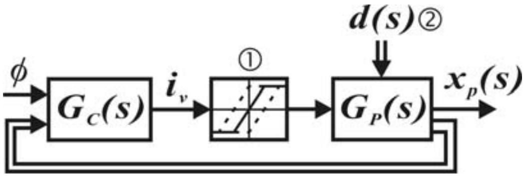


Figure 5 Simplified block diagram of the block surrounded with dashed line in Fig. 3

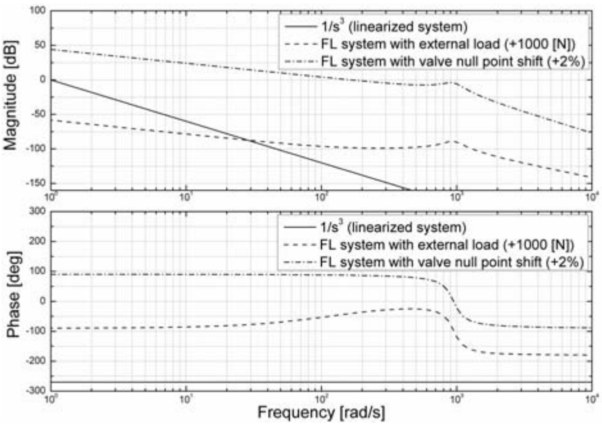


Figure 6 Frequency characteristics of the block surrounded with dashed line in Fig. 3 with external force, with valve null point shift

Table 2 Poles of the object system shown in Fig. 3 under the variation of system parameters

	Pole 1	Pole 2	Pole 3
No Variation	-38+24i	-38-24i	-191
β_e : -50%	-31+30i	-31-30i	-104
β_e : +50%	-40+21i	-40-21i	-290
B_p : -50%	-102+43i	-102-43i	-32
B_p : +50%	-26+30i	-26-30i	-247
M_t : -90%	-21+23i	-21-23i	-3990
M_t : +90%	-22+85i	-22-85i	-27

the transfer functions between Q_l and i_v, and between i_v and Q_l were substituted as '1'. We can have a hint from the data in Table 2 on the fact that the control performances of the system shown in Fig. 3 might be affected severely under the variation of system parameters β_e, B_p and M_t.

DESIGNING A DISTURBANCE OBSERVER

In this section, we will design a Feedback Linearization – State Feedback Controller with Disturbance Observer (FL-SFC-DOB) for the hydraulic servo system. Fig. 7 shows a system with the disturbance observer. $H(s)$ in the figure shows the control system including FL-SFC,

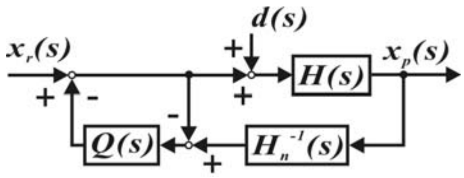


Figure 7 Application of a disturbance observer to the control system shown in Fig. 3

and $H_n(s)$ is the nominal model described by Eq. (11). $Q(s)$ is a filter for disturbance compensation, and $d(s)$ describes external disturbances or disturbance equivalence of system parameters' variation.

In this study, Umeno's method[5] was applied to design $Q(s)$. $Q(s)$ for the system was obtained as Eq. (12) considering that the relative order between the numerator and the denominator of the system transfer function is 3.

$$Q(s) = \frac{\alpha^3}{(s+\alpha)^3} \qquad (12)$$

where α is a cut-off frequency.

RESULTS OF EXPERIMENT AND SIMULATION

Experiments were done using the experimental system shown in Fig. 4. The parameters values of the experimental system are given in Table 1. In all the experiments of this study, p_s is set to be 35 bar. For realizing digital control and signal measurements, a PC and MATLAB/RTWT[6] were used.

The results when「FL-SFC」applied

Fig. 8 shows the experimental and simulated results when a step input signal(0→10 mm) is given to the system with FL-SFC. In addition, responses of C-SFC(the Conventional State Feedback Controller applied to the hydraulic servo system) were included in the same figure to evaluate the results of FL-SFC objectively.

In designing C-SFC, a flow equation linearized in the operating point of the servo valve was used. Controller gains for C-SFC were obtained from pole placement method. The representative poles for the

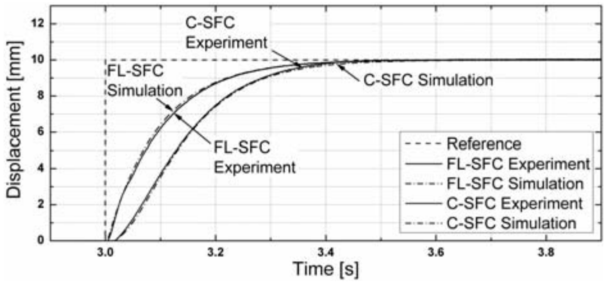


Figure 8 Experimental and simulated results of the object system with C-SFC or with FL-SFC

C-SFC were placed at same positions as ones for FL-SFC(designed in section 3) so as to enable an objective comparison of control performances of the systems with the FL-SFC and the C-SFC. And other poles were placed at -38×5 on the real axis.

Fig. 8 shows the validity of the mathematical model of the control systems used in this study. In this figure, FL-SFC shows better response compared to C-SFC by reducing 17.1% in settling time(±2% basis).

The results when「FL-SFC-DOB」applied

Under external load

Fig. 10 shows the step responses of the control system under external load(Fig. 9), 1000N. When FL-SFC was applied(Fig. 10 (a)), 0.4 mm steady state error and undershoot response appeared. But, with FL-SFC-DOB application, the effects of the external load could be rejected clearly.

Under null point drift in the control valve

In general, allowable limit of null point drift in servo valves is said to be ±2%. With the allowable limit value of null point drift, the control performances of the control system were investigated. The null point drift in

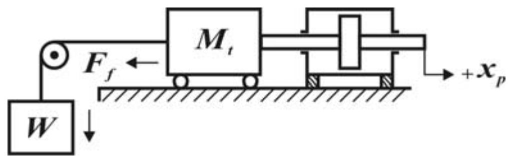
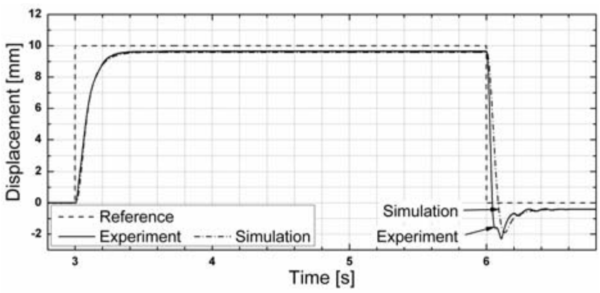
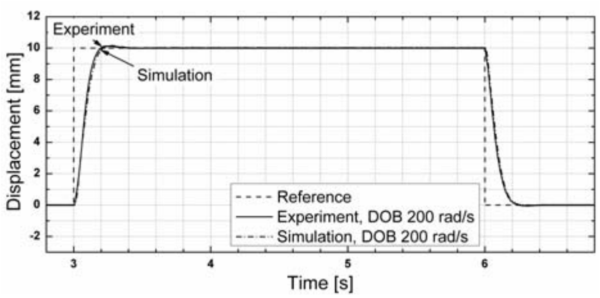


Figure 9 Simplified schematic diagram of the load system in the object system



(a) when FL-SFC applied

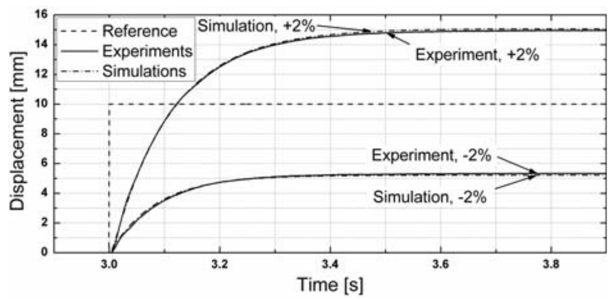


(b) when FL-SFC-DOB applied

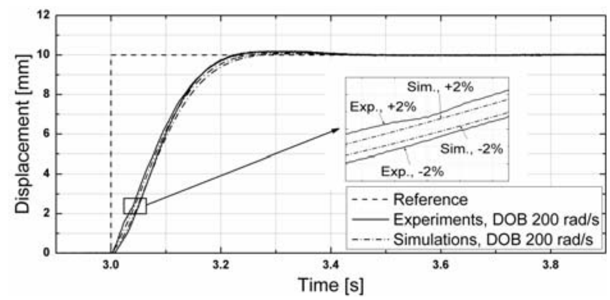
Figure 10 Experimental and simulated results of the object system with external load, 1000 N

the servo valve was realized equivalently by shifting null point of the servo amplifier output.

Fig. 11 shows the step responses of the control system under null drift, ±2%. When FL-SFC was applied(Fig. 11 (a)), great steady state errors appeared, which was anticipated result by referring to Fig. 5. But,

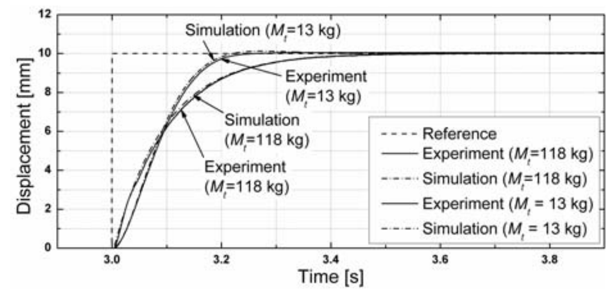


(a) when FL-SFC applied

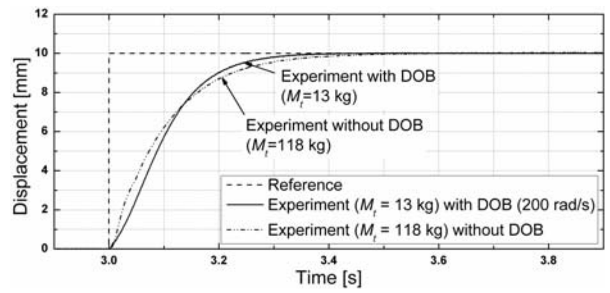


(b) when FL-SFC-DOB applied

Figure 11 Experimental and Simulated results of the object system under valve null point shift with +2%, -2%

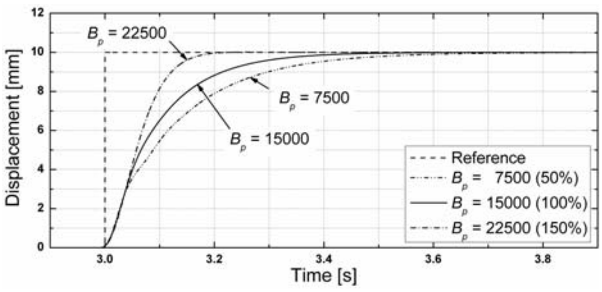


(a) when FL-SFC applied

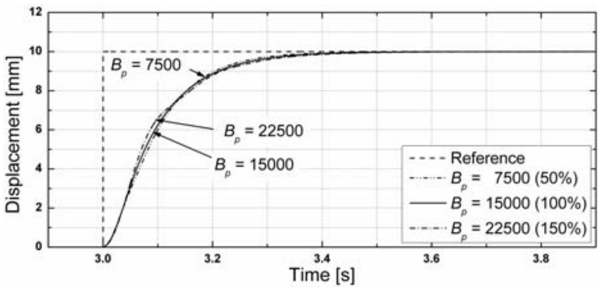


(b) when FL-SFC-DOB applied

Figure 12 Experimental and simulated results of the object system under M_t variation

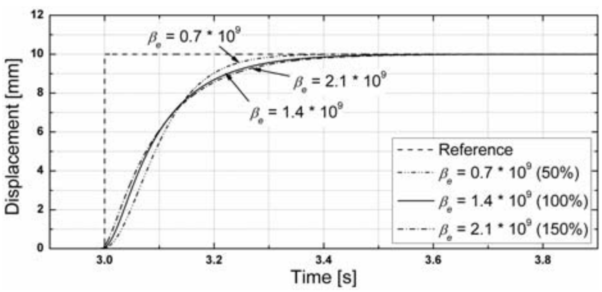


(a) when FL-SFC applied

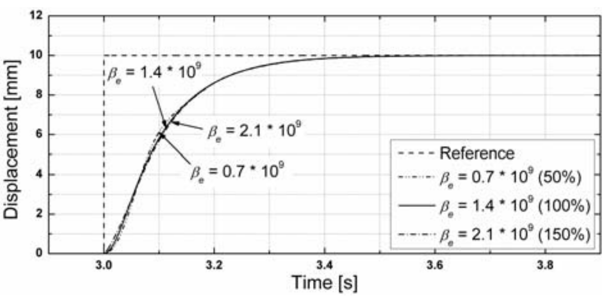


(b) when FL-SFC-DOB applied

Figure 13 Simulated results of the object system under B_p variation



(a) when FL-SFC applied



(b) when FL-SFC-DOB applied

Figure 14 Simulated results of the object system under β_e variation

with FL-SFC-DOB application, the effects of the null point drift could be removed and zero steady state error realized.

Under the variations of M_t, B_p and β_e

The effects of the variations of the representative physical parameters M_t, B_p and β_e on the control performances of the control system were investigated. Step responses of the control system under the variation of M_t, B_p and β_e were shown in Fig. 12, 13 and 15. When FL-SFC was applied, change in control performances appeared according to the variation of M_t, B_p and β_e. And, it was ascertained that, with FL-SFC-DOB application, the change in the control performances under the variation of M_t, B_p and β_e could be moderated in a great deal.

CONCLUSIONS

In this study, the control performance of a hydraulic servo system with a feedback linearization compensator was investigated. The focus of this study was set on thequantitative investigation of the effects(sensitivities) of disturbances and system parameters’ variation on control performances of the hydraulic control system. Finally, verified the efficacy of a disturbance observer to overcome the control performances deteriorations due to system parameters' variations, disturbances in the control system.

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