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DESIGN OF ROBUST POSITION/PRESSURE CONTROLLER FOR CYLINDER USING HYDRAULIC TRANSFORMER

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ABSTRACT

In this paper, the position/pressure control of a hydraulic cylinder using a hydraulic transformer is studied. A hydraulic transformer based on FFC pump/motor is used as a control component to drive the cylinder with a load horizontally. The two-degree-of-freedom (2DOF) design technique is applied to obtain the positioning controller with tracking performance and robustness for uncertainties while the structure of pressure controller is given by internal model principle to compensate stepwise disturbance. With these controllers the nonlinear friction torque, the leakage of the transformer and the load force acting on the cylinder can be compensated and the experimental results show that the sufficient accuracy of position and pressure are obtained.

KEY WORDS

CPS, Hydraulic transformer, Position control, Pressure control, 2-DOF

NOMENCLATURE

A	:	area of cylinder piston [m^2]
B_c	:	viscous damping coefficient of cylinder [kg/s]
B_{ht}	:	viscous damping coefficient of transformer [kg·m^2/s]
D	:	displacement [cc/rev]
J	:	moment of inertia [kg·m^2]
q_{le}	:	leakage flow [m^3/s]
F_L	:	load force [N]
m	:	mass of load and piston [kg]
p	:	pressure [MPa]

p_s : pressure of constant pressure source [MPa]

p_{1r} : desired pressure of load [MPa]

q : flow [m^3/s]

T : torque [N·m]

T_f : nonlinear friction torque [N·m]

y : cylinder displacement [mm]

y_r : Desired cylinder piston position[m]

ω : rotational speed [rad/s]

k : rate of spring [N/mm]

l : Initial distance from rod to spring[mm]

Subscripts

1 : cylinder head chamber

2 : cylinder rod chamber

A : variable unit of transformer

B : fixed unit of transformer

y : velocity controller

p : pressure controller

INTRODUCTION

The importance of energy saving in hydraulic power transmission system is increasing. Since the valve control strategy of hydraulic cylinder drive systems has relatively larger pressure loss, especially for large cylinders, a hydraulic cylinder drive with hydraulic transformer has been developed. The circuit consists of constant pressure power source, transformer and cylinder, as shown in Figure 1. The hydraulic transformer is constructed by combining a fixed displacement pump/motor and a variable displacement one which are connected by a driving shaft. The power is transmitted from the transformer to the cylinder without controlling the valve. The major advantage is that energy can be recovered and stored in accumulator in the common pressure line when the cylinder is being retracted under load [1]-[3]. The hydraulic transformer used in this paper consists of two FFC pump/motor. The Fluid Force Couple (FFC) pump/motor has high efficiency in wide operating range [4], [5].

In this research, we consider a design of robust positioning/pressure controller of hydraulic cylinder with transformer. Disturbances, *e.g.* nonlinear friction torque, leakage flow of transformer and load force, affect the performance of the positioning/pressure controller. These effects can not be compensated only by feedforward controller because it is difficult to describe them mathematically in detail a priori. Moreover, to accommodate some uncertainties on physical parameters, the feedback loop design is also used. Therefore in this paper, the 2DOF controller design method is introduced for positioning control, not only to compensate disturbance and parameter uncertainty but also to ensure the tracking performance for given drive pattern. 2 DOF method is now widely used in mechanical system control [6]-[9] and with is able to specify the performance quantitatively by adjusting controller parameters. And in pressure control, we design the controller via internal model principle to decide the structure. The cylinder is set horizontally and the load spring simulates a work in pressing machine. The proposed controllers are examined in experiment and evaluate its performance.

EXPERIMETAL DEVICE AND DRIVE PATTERN

Experimental device

The schematic diagram of the experimental system used in this research is shown in Figure 1. The transformer is connected to the head side of hydraulic cylinder. In this research the rod side of a cylinder is connected to the constant pressure system to realize heavy load. The cylinder displacement y and displacement of transformer D_A, the rotational speed of transformer ω and pressure in cylinder head chamber p_1 are taken into a controller.

The displacement of variable pump/motor is D_{Amax} =48.9[cc/rev] and the displacement of fixed pump/motor is D_B=19.6[cc/rev]. The diameter of hydraulic cylinder is 50[mm]. The rate of spring is k=362.6[N/mm]. The Initial distance from rod to spring l=157[mm].

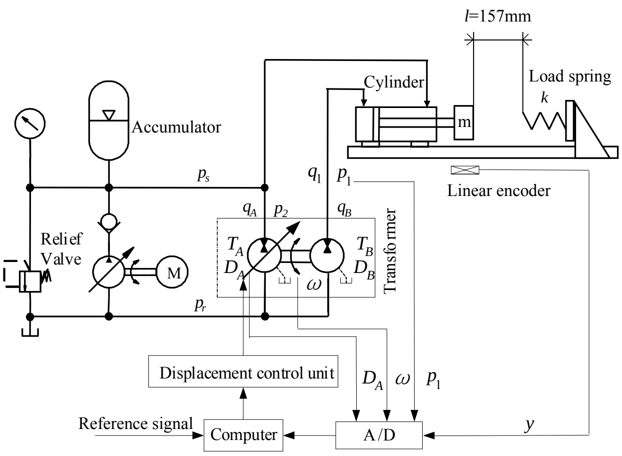

Figure 1 The experimental setup

Drive pattern

The drive pattern of the cylinder used in this research simulates the hydraulic pressing machines. Figure 2 shows a cylinder drive pattern used in this research. A load spring is supposed to be a work. In the cylinder position control, the reference step signal changes two phases and this implies an actual approaching motion of pressing machines. The cylinder velocity is controlled in the higher velocities before reaching the load spring. After touching load spring, the controller keeps lower cylinder velocity until the pressure exceeds a predetermined threshold. Then we switch from the position control to the pressure control as a pressing process.

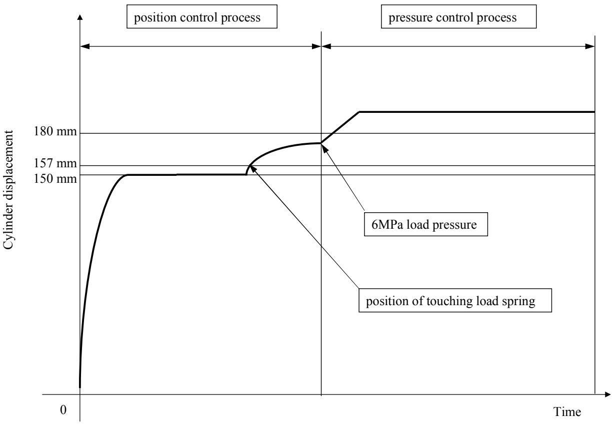

Figure 2 Cylinder drive pattern

MODELING

To derive the mathematical model of the system shown in Figure 1, we assume that the dynamics of piping are negligible and tank pressure p_r=0. The plant model can be reduced to lower dimension by ignoring the compressibility of the hydraulic oil. The leakage flow, the nonlinear friction torque and the load force are treated as disturbances. Then as the equation of motion of transformer, the flow rate and the cylinder motion, we obtain the following.

$$D_A p_s = J\dot{\omega} + B_{ht}\omega + D_B p_1 + T_f \quad (1)$$

$$q_1 = A_1 \dot{y} = \omega D_B - q_{le} \quad (2)$$

$$A_1 p_1 = m\ddot{y} + B_c\dot{y} + F_L = m\ddot{y} + B_c\dot{y} + A_2 p_2 + k(y-l) \quad (3)$$

$$F_L = \begin{cases} p_2 A_2, & \text{before contacting with spring} \\ p_2 A_2 + k(y-l), & \text{contacting with spring} \end{cases} \quad (4)$$

where l is the initial distance from cylinder to the load spring.

System model for positioning control

The cylinder displacement y is given as:

$$y(s) = G_y(s)\left(D_A(s) + d_y(s)\right) \quad (5)$$

where $G_y(s)$ is the transfer function of the nominal system from D_A to y, and $d_y(s)$ is a disturbance which includes nonlinear friction torque T_f, leakage flow q_{le} and load force F_L, and $G_y(s)$ is given as:

$$G_y(s) = \frac{y(s)}{D_A(s)} = \frac{K_y}{J_y s^2 + B_y s} \quad (6)$$

$$K_y = \frac{D_B p_s}{A_1}, \quad J_y = J + \frac{D_B^2 m}{A_1^2}, \quad B_y = B_{ht} \quad (7)$$

$$d_y(s) = -\frac{D_B}{A_1 p_s}F_L(s) - \frac{1}{p_s}T_f(s) - \frac{Js + B_{ht}}{D_B p_s}q_{le}(s) \quad (8)$$

$$F_L(s) = p_2 A_2 + k(y-l) \quad (9)$$

System model for pressure control

On the other hand, the pressure in the cylinder head chamber p_1 is expressed as:

$$p_1(s) = G_{P_1}(s)\left(D_A(s) + d_{p_1}(s)\right) \quad (10)$$

where G_{p1} (s) is a transfer function from D_A to p_1 and $d_{p1}(s)$ is a disturbance which includes nonlinear friction torque T_f, leakage flow q_{le} and load force F_L and G_{p1} (s) is given as:

$$G_{p_1}(s) = \frac{P_1(s)}{D_A(s)} = \frac{K_{p_1}}{a_1 s^2 + a_2 s + a_3} \quad (11)$$

$$K_{p_1} = D_B P_s(ms^2 + B_c s + k), \quad a_1 = A_1^2 J + D_B^2 m, \quad a_2 = A_1^2 B_{ht} + D_B^2 B_c \quad (12)$$

$$d_{p_1}(s) = \frac{A_1(Js + B_{ht})s}{a_1 s^2 + a_2 s + a_3}F(s) - \frac{D_B(ms^2 + B_c s + k)}{a_1 s^2 + a_2 s + a_3}T_f(s) - \frac{(ms^2 + B_c s + k)(Js + B_{ht})}{a_1 s^2 + a_2 s + a_3}q_{le}(s) \quad (13)$$

$$F_L(s) = p_2 A_2 + k(y-l) \quad (14)$$

CONTROLLER DESIGN

In this paper, cylinder velocity control and force control is considered. Figure 3 and Figure 4 show block diagrams of the position and the pressure control systems, respectively. For notational convenience, subscripts y and p_1 stand for the cylinder position control system and the pressure control system, respectively. We design each controller via 2-DOF and internal model principle.

Position controller system

Figure 3 shows the block diagram of feedback type 2-DOF control system for position control. In this figure, r is desired position, u control input, y controlled variable, d disturbance, w disturbance from parameters variation, n sensor noise.

According to the parameterization of 2-DOF control system, controllers are given as

$$C_A(s) = \frac{1}{G_y(s)} \cdot \frac{Q(s)}{1-Q(s)} \tag{15}$$

$$C_B(s) = \frac{1}{G_y(s)} \cdot \frac{G_M(s)}{1-G_M(s)} \cdot \frac{1}{1-Q(s)} \tag{16}$$

where $G_M(s)$ is a reference model and $Q(s)$ is a stabilizing filter.
The transfer function from desired value r to controlled variable y is given as

$$W_{ry}(s) = \frac{y(s)}{y_r(s)} = \frac{C_B(s)G_y(s)}{1 + C_A(s)G_y(s) + C_B(s)G_y(s)} = G_M(s) \tag{17}$$

The sensitivity function $S(s)$ is given as

$$S(s) = \frac{y(s)}{w(s)} = \frac{1}{1 + C_A(s)G_y(s) + C_B(s)G_y(s)} = (1-Q(s))(1-G_M(s)) \tag{18}$$

The complementary sensitivity function $T(s)$ is given as

$$T(s) = \frac{y(s)}{n(s)} = \frac{C_A(s)G_y(s) + C_B(s)G_y(s)}{1 + C_A(s)G_y(s) + C_B(s)G_y(s)} = 1 - S(s) \tag{19}$$

The disturbance response is given as

$$W_{dy}(s) = \frac{y(s)}{d(s)} = \frac{G(s)}{1 + C_A(s)G_y(s) + C_B(s)G_y(s)} = S(s)G_y(s) \tag{20}$$

From Eq.(17), W_{ry}(s) can be designed with G_M(s), and from Eq.(18), (19) and (20) S(s), T(s) and W_{dy}(s) can be designed with G_M(s) and $Q(s)$. *S(s)* is sensitivity index for disturbance rejection performance. And *T(s)* is an index for robust stability and shows the sensor noise rejection performance. According to mixed-sensitivity design of H-Infinity control theory, it is desired that S(s) is small in lower frequency domain while T(s) is small in higher frequency domain in practical application.
In the case of positioning control, the $G_M(s)$ is considered as a second order model:

$$G_M(s) = \frac{1}{t_r^2 s^2 + 2\zeta_r t_r s + 1} \tag{21}$$

where ω_r (=1/ t_r) and ζ_r are natural angular frequency and damping coefficient, respectively.

$Q(s)$ is given by the nominal model

$$Q(s) = \frac{3t_N s + 1}{t_N^3 s^3 + 3t_N^2 s^2 + 3t_N s + 1} \tag{22}$$

By substituting equations (21) and (22) into equation (15) and (16) and setting B_n=0, we have

$$C_A(s) = \frac{J_y}{K_y} \frac{3t_N s + 1}{t_N^3 s + 3t_N^2} \tag{23}$$

$$C_B(s) = \frac{J_y}{K_y} \frac{1}{s(t_r^2 s + 2\zeta_r t_r)} \frac{t_N^3 s^3 + 3t_N^2 s^2 + 3t_N s + 1}{t_N^3 s + 3t_N^2} \tag{24}$$

Before touching load spring, the controller parameters were selected as t_r＝0.44, ζ_r＝1.11 and t_N＝0.15 to ensure the velocity response considering the saturation characteristic of displacement of variable displacement pump/motor. The primal stepwise reference position is 150[mm] and second one 180[mm]. After touching the load spring, the controller parameters were selected as t_r＝1, ζ_r＝1 and t_N＝0.15 to push the load spring as slowly as possible.

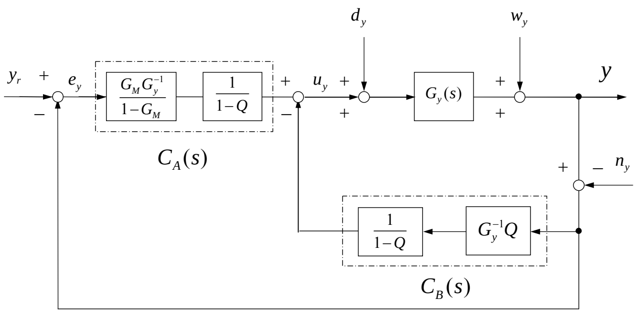

Figure 3 Cylinder position control system

Pressure control system

As the cylinder rod extends, the pressure p_1 increases. Therefore we switch the controller from position tracking to pressure tracking control imaging hydraulic pressing machines. Since the dynamics to be controlled is changed from Eqs. (6) to (11), the controller switching is made depending on the pressure threshold; when the threshold is exceeded, the controller is changed from position controller to pressure controller.
For given constant reference pressure, disturbance torque and leakage flow can be considered as constants. This implies that the controller should contain at least one integrator to track the reference pressure via internal model principle without steady state error. To consider a sufficient condition for the pressure controller, we define $G_{er}(s)$ and $G_{ed}(s)$ as transfer functions from reference $r(s)(=r_0/s)$ to error $e(s)$ and from input disturbance to error, respectively. Then, by applying the final value

theorem, the steady-state error e_∞ is given by

$$\begin{aligned} e_\infty &= \lim_{t\to\infty} e(t) \\ &= \lim_{s\to 0}\left\{G_{er}(s)r(s)+G_{ed}(s)d(s)\right\}\cdot s \\ &= \lim_{s\to 0}\left\{\frac{1}{1+G(s)C(s)}\frac{r_0}{s}-\frac{G(s)}{1+G(s)C(s)}\frac{d_0}{s}\right\}\cdot s \qquad (25) \\ &= \lim_{s\to 0}\frac{r_0-G(s)d_0}{1+G(s)C(s)} \end{aligned}$$

where d_0 is constant disturbance gain.
This result shows that a controller should contain at least one integrator to make steady state error reduce to zero.

The controller of the following structure is then given as

$$C_{p1} = k_p + \frac{k_i}{s} \qquad (26)$$

where k_p and k_i are PI control parameters to be determined.

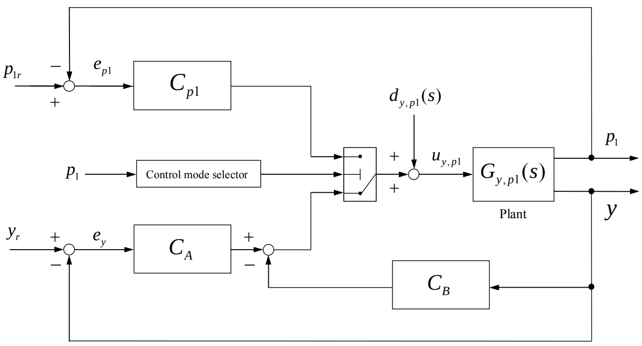

Figure 4 Positioning and pressure control system

EXPERIMETAL RESULT AND CONSIDERATION

Figure 5 shows the experimental results on pressure in cylinder head chamber p_1, cylinder displacement y and displacement of the variable unit D_A during position and pressure control. The constant system pressure is set to be 5[MPa] in all experimental results and the load mass m=60[kg]. It is observed that the cylinder tracks to the reference model without the overshoot both in positioning and pressuring control phase. Moreover, we can see some delay in initial response, about 0.3[s]. This can be considered as the influence of dead zone in transformer. In the position control before reaching the load spring, good position tracking performance to reference model is achieved, and the settling time is about 4 seconds. In position control after touching load spring, the disturbances of nonlinear friction torque and load force of acting on cylinder can be compensated and the settling time is about 0.5 seconds.

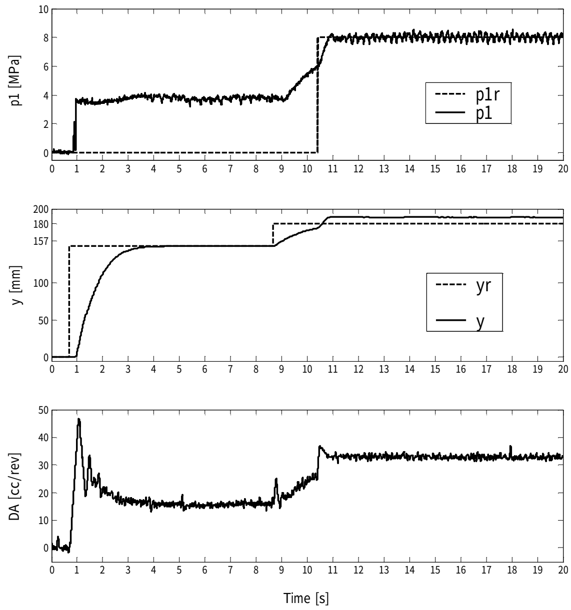

Figure 5 Experimental result

CONCLUSION

In this paper, the robust position/pressure control performance of cylinder using hydraulic transformer connected to constant pressure supply is examined. In positioning control, the feedback type 2-DOF controller is implemented and PI control system based on internal model principle is applied to pressure control. The experimental results show that the disturbances of nonlinear friction torque, the leakage of the transformer and load force of acting on cylinder can be compensated and the good positioning/pressuring accuracy is obtained. As a future work, we improve a speed of response by introducing nonlinear controller. Moreover the energy saving performance should be discussed.

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