OPTIMUM PARAMETERS OF SPOOL VALVE FEEDBACK TO MINIMIZE SURGE PRESSURE

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ABSTRACT

This paper deals with the optimum design of the stiffness and damping coefficients of the spring and dashpot used in the feedback loop of a hydraulic servo-valve, to attain minimum surge pressure during the stop and reverse operations. The proportional element of the feedback represented by the lever ratio is predefined. The sate space equations are derived and two objective functions based on relative and absolute pressure rise are minimized. The applied minimization technique is based on a modified gradient method. The results show a better performance and less shocks to the optimum design of that dynamic parameters.

NOMENCLATURE

A	Effective area of the piston
a,b	Lengths of the proportional lever
C_d	Flow coefficient of the valve
C_1,C_2,C_3	Leakage coefficients
C	Damping coefficient
H	Dimensionless feedback gain
k	Spring stiffness
m	Mass
p_s	Supply pressure
p_1,p_2	Pressure in the actuator chambers
Q	Flow rate
t	Time
u	Input displacement signal
v	Velocity of the piston
V	Volume
W	Area gradient of the valve
x	Piston displacement
y	Spring displacement
Z	Displacement defined in equation 1
α	β_e/p_s
β_e	Effective bulk modulus of the fluid
ρ	Fluid density
ϕ	Objective function
av	Subscript indicating average
f	Subscript indicating final
max	Subscript indicating maximum
min	Subscript indicating minimum
r	Subscript indicating reference
o	Subscript indicating one step of optimization
oo	Subscript indicating more than one optimization step
*	Superscript indicating dimensionless value

INTRODUCTION

Hydraulic servomechanism, combines the versatility and the precision available from the measurement and signal processing technique with the rapid response and loading capacity of hydraulic cylinder drive. It has been widely used for position control of large such as antennas, air frame surfaces and machine tools.

The important factors affecting the dynamic performance and the steady state behaviour of hydraulic servomotors have received great attention of many investigators [1-12]. McCloy [1] developed an open loop model to study the response of a loaded hydraulic cylinder to step input of valve position.

He discussed the cavity formation, peak pressures, and the effect of cylinder position on system natural frequency. Velenius [2] applied a sensitivity analysis to investigate the influence of design parameters on the dynamics of electrohydraulic position control servos. Awad and EL-Gamal [3] studied the effect of the presence of an air bubble in a hydraulic servomotor on the dynamic response of a control loop. Baysec and Rees [4] described an efficient model of a servo-valve for use in digital computation. McClamroch [5] developed a mathematical theory for the multivariable displacement control of an arbitrary nonlinear flexible structure controlled by multiple electrohydraulic servo-actuators. Finney et al. [6] considered the practical application of sell-tuning control to an hydraulic cylinder drive.

The operation of hydraulic servomotors is characterized by stop and speed reversals in very short time.

This kind of operation always accompanied with high pressure surge and noise because of the transient response of fluid [11,12]. Adding more settling time by a smoothly changed signal is not feasible in practical engineering application. Yuwan et al. [9] considered an open loop hydraulic servomechanism and used the gradient method to find the optimal input control signal for speed conversion.

Since the hydraulic servomotor must have a feedback input, this paper is concerned with the minimization of the leak pressures during a short time speed conversion of a closed loop hydraulic servomotor. A dashpot and linear spring are connected to the actuator. The spring displacement is fedback to the valve through a proportional lever. The system state equations are derived in a dimensionless form. Then the optimum values of the dashpot damping coefficient and the spring stiffness are determined using a modified gradient optimization technique. The feasibility of the proposed system in practical applications is also discussed.

MATHEMATICAL MODEL

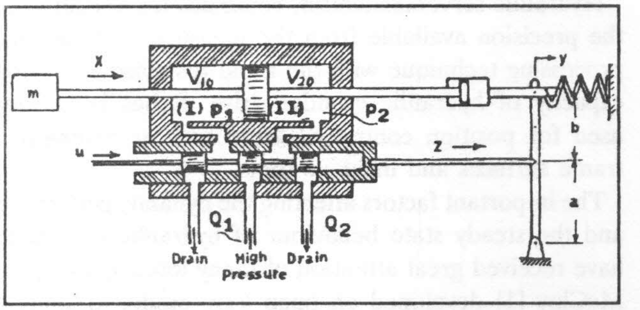

Figure 1. The Feedback Loop of a Hydraulic Servomotor with 5-ways spool valve.

The controlled hydraulic servomechanism under investigation is shown in Figure (1). The opening area of the valve orifice is a function of the input signal to the spool (u) and to the feedback signal from the lever,

$$Z = u-[a/(a+b)]y \tag{1}$$

It is assumed that the system is a lumped-parameter one, source pressure p_s is constant, return line pressure p_0 is zero, the pipes are short (i.e., the inertance and the friction can be neglected), and no cavitation is produced in the system.

Assuming that the servo-valve is critically centered and the orifices are matched and symmetrical, the flow rate to the actuator is given by,

$$Q_1 = C_d\, W\, Z[(2/\rho)(p_s-p_1)]^{\frac{1}{2}} \tag{2}$$

and

$$Q_2 = C_d\, W\, Z[(2/\rho)p_2]^{\frac{1}{2}} \tag{3}$$

where C_d is the flow coefficient of the valve and W is area gradient. Applying the continuity equation to the actuator chamber gives,

$$\sum Q_{in}-\sum Q_{out} = dv/dt-(V/\beta_e)dp/dt \tag{4}$$

$$Q_1-C_1(p_1-p_2)-C_2p_1 = Av+(V_{10}/\beta_e)dp_1/dt \tag{5}$$

$$C_3p_2+C_1(p_1-p_2)-Q_2 = -Av+(V_{20}/\beta_e)dp_2/dt \tag{6}$$

Substituting equations (2) and (3) into equation (4) and neglecting the leakage ($C_1=C_2=C_3=0$), result in

$$\dot{p}_1 = (\beta_e/V_{10})\{C_d\, W\, Z\, [(2/\rho)(p_s-p_2)]^{\frac{1}{2}}-Av\} \tag{7}$$

$$\dot{p}_2 = (\beta_e/V_{20})\{Av-C_d\, W\, Z[(2/\rho)p_2]^{\frac{1}{2}}\} \tag{8}$$

Considering the balance of the forces on the piston, the equation of motion is,

$$(p_1-p_2)A-C(dx/dt-dy/dt) = m(d^2x/dt^2) \tag{9}$$

and

$$C\{(dx/dt)-(dy/dt)\} = Ky \tag{10}$$

Since $v = (dx/dt)$, equations (9) and (10) can be written as follows,

$$\dot{y} = v-(k/C)y \tag{11}$$

and

$$\dot{v} = (A/m)(p_1-p_2)-(k/m)y \tag{12}$$

Define Z_r as the total spool valve displacement and the reference displacement x_r, time t_r and velocity v_r of the piston as,

$$x_r = V/A,\ t_r = [mV/(\beta_e A^2)]^{\frac{1}{2}},\ v_r = C_d W_r Z_r/A[2p_s]^{\frac{1}{2}}.$$

Introducing the dimensionless quantities

$$p^* = p/p_s,\ x^* = x/x_r,\ y_* = y/x_r,\ Z^* = Z/Z_r$$

$$t^* = t/t_r,\ v^* = v/v_r,\ \text{and}\ \alpha = \beta_0/p_s$$

and choosing the state variables as p_1^*, p_2^*, X^*, v^* and y^*, the state space equations can be expressed in a dimensionless form as,

$$p_1^{*\prime} = \alpha(V/V_{10})(v_r t_r/x_r)\{(W/W_r)Z^*[1-p_1^*]^{\frac{1}{2}}-v^*\} \quad (13)$$

$$p_2^{*\prime} = \alpha(V/V_{20})(v_r t_r/x_r)\{v^*-(W/W_r)Z^*[p_2^*]^{\frac{1}{2}}\} \quad (14)$$

$$X^{*\prime} = [(v_r t_r)/x_r]v^* \quad (15)$$

$$v^{*\prime} = [x_r/(v_r t_r)](1/\alpha)[(p_1^*-p_2^*)-k^*y^*] \quad (16)$$

$$y^{*\prime} = [(v_r t_r)/x_r][v^*-(k^*/C^*)y^*] \quad (17)$$

where $Z^* = u^*-Hy^*$, $k^* = (x_r k)/(p_s A)$, $C^* = (v_r C)/(p_s A)$, $H = a/(a+b)(u_r/Z_r)$ and $'$ denotes differentiation with respect to the time t^*.

ALGORITHM

1- Initial values for the parameters k^* and C^* are used
2- For zero initial value of the state vector, equations (13) to (17) are solved numerically using the fourth order Runge- Kutta technique with time increment of 0.05. The displacement signal to the spool value (u^*) is taken as $u^* = 1.0-\exp(-\epsilon t^*)$, $0 \leq t^* \leq t_0^*$, where ϵ is a factor controlling the speed of the exponential decay t_0^* is the time at witch the piston reaches the point $x^* = 0.5$.
3- A sudden change in the valve displacement and direction is applied at the time t^*, $u^* = u_2$, $t_0^* < t^* \leq t_f^*$, where u_2 is constant less than unity and t_f^* is the final time.
4- The average pressure in the two chambers is evaluated during the period $(t_f^* - t_0^*)$.
5- Two objective functions are considered

$$\phi_1 = \{[p_{1max}-p_{1av}]/p_{1av}\} + \{[p_{2max}-p_{2av}]/p_{2av}\}$$

$$\phi_2 = \left|[p_{1max}-p_{1av}]\right| + \left|[p_{2max}-p_{2av}]\right|$$

where p_{1max} and p_{2max} are the maximum pressure in the actuator chambers.
6- An optimization technique based on the modified gradient method [13] is used to minimize the above objective functions. It should be noticed that the change in the displacement always occurs at the time t_0^*, determined in step 2. The two sets of parameters realizing the minimum of ϕ_1 and ϕ_2 are considered the searched parameters.

DISCUSSION OF RESULTS

The descried algorithm was applied to a servo-valve with following conditions: $\alpha=50$, $v_r t_r/x_r=0.4$, $V/V_{10} = V/V_{20}=2.0$, $W/W_r=0.15$, $H=0.8$ and $\epsilon=10$.
Since the used minimization technique is unconstrained, therefore the obtained values of k^* and C^* should be checked from the point of view of practical application. Using an initial guess $k^*=2.8$ and $C^*=5.0$ and applying a sudden decrease in u^* from 1.0 to 0.5, then after one step of optimization the resulted values are $k^*=9.22$ and $C^*=4.37$. The objective function ϕ_1 is reduced from 1.176 to 0.956. More minimization of the objection function ϕ_1 (6 steps) reduced its value to 0.0131 and renders $k^*=208$ and $C^*=120$ which are not feasible. It can be noticed from Figure (2-a) and (2-b) that with the increase in the value of k^* and C^* the pressure surge and the sensitivity to any disturbance are decreased because the opening area of the valve orifice does not change. Also the high values of k^* and C^* are accompanied with a considerable decrease in the actuator velocity.

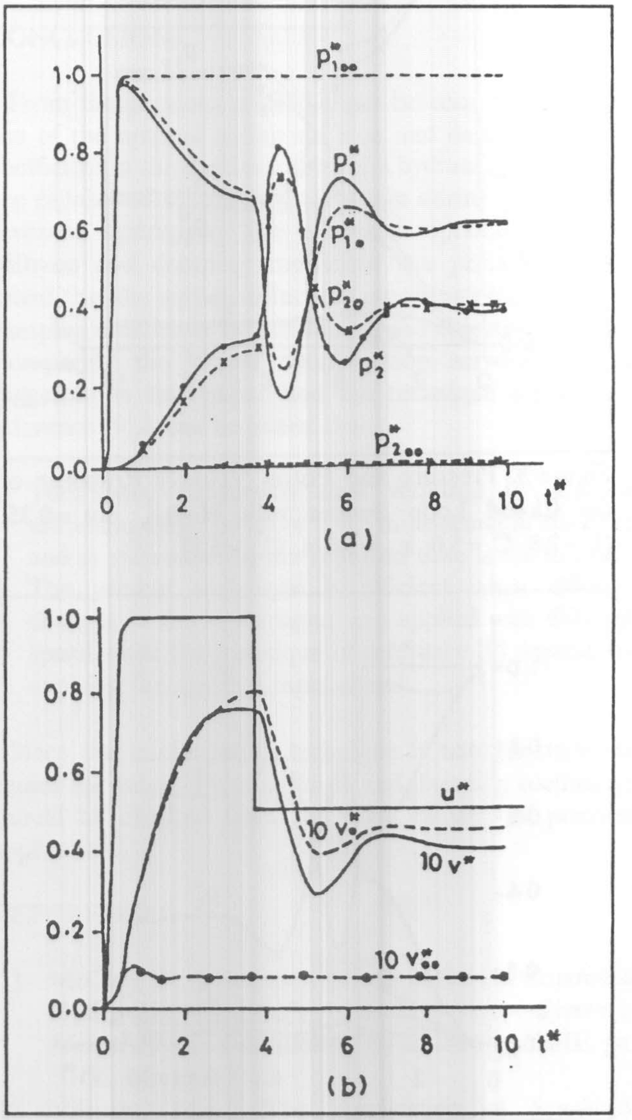

Figure 2. Pressure and Piston Velocity Response of the Closed Loop System with $H=.8$, $\Delta u^*=.5$ ($k^*=2.8$, $C^*=5$, $k^*_0=9.22$, $C^*_0=4.35$, $k^*_{\infty}=208$, $C^*_{\infty}=120$).

The amount and the speed of the input change affect the response greatly. As u^* decreased sharply from 1.0 to 0.75 and staring with the same initial guess ($k^*=2.8$ and $C^*=5.0$), the resulted optimal values are $k^*=5.474$ and $C^*=5.5009$ and the objective function ϕ is reduced from 0.427 to 0.3755 (compared to $\phi_1=0.956$ as u^* changed to 0.5). The results are illustrated in Figure (3). Meanwhile when u^* is changed from 0 at $t^*=0$ to 0.993 at $t^*=0.5$ (t=0.5 msec) it can be noticed from Figures (2) and (3) that the pressure surge is diminished.

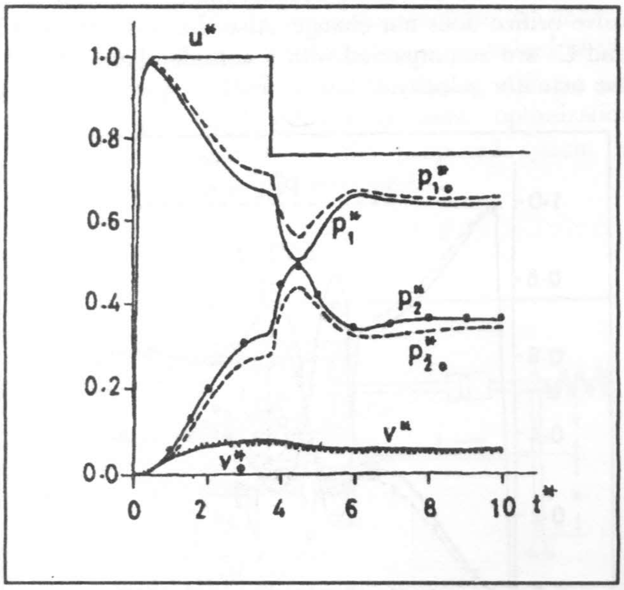

Figure 3. Pressure and Piston Velocity Response of the Closed Loop System with H=0.8, $\Delta u^*=0.25$, ($k^*=2.8$, $C^*=5.0$, $k^*_0=5.474$, $C^*_0=5.5$).

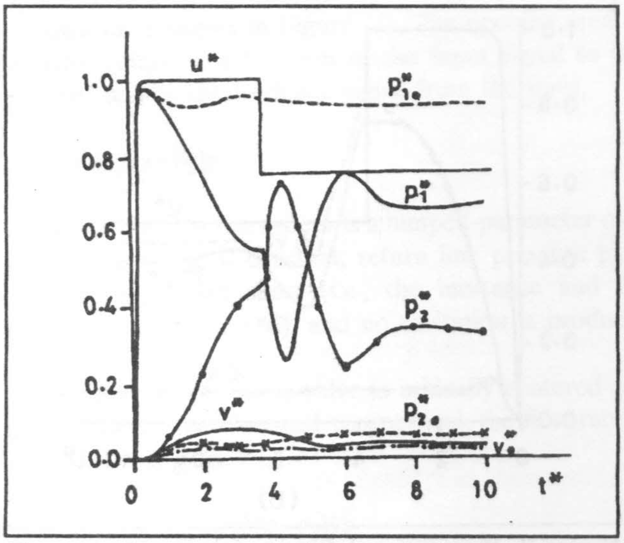

Figure 4. Pressure and Piston Velocity Response of the Closed Loop System with H=0.8, $\Delta u^*=0.25$, ($k^*=.8$, $C^*=10$, $k^*_0=22.804$, $C^*_0=32.83$).

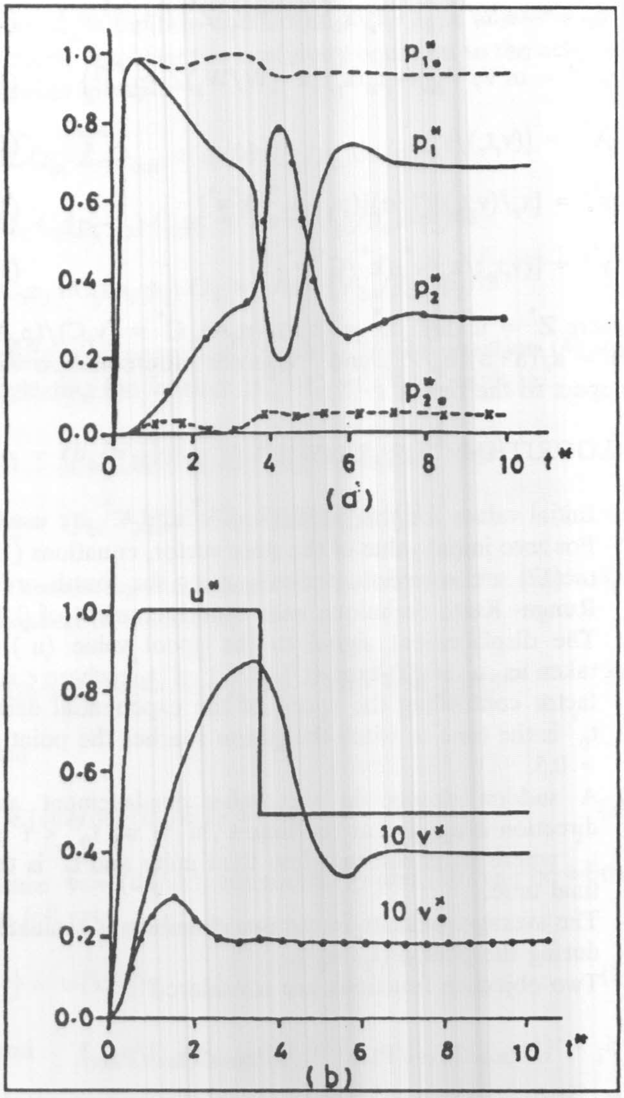

Figure 5. Pressure and Piston Velocity Response of the Open Loop System, $\Delta u^*=0.5$, ($k^*=0.8$, $C^*=10.0$, $k^*_0=30.47$, $C^*_0=47.5$).

As in all the cases of optimization, the obtained minimum values depend on the initial guess; starting with $k^*=0.8$ and $C^*=10$ and for a sudden change in u^* from 1.0 to 0.75, the optimum values are $k^*=22.804$, $C^*=32.833$ and ϕ_1 is decreased from 1.1227 to 0.03173 (Figure (4)).

Employing the servo-valve in both an open control loop (H=0.0) and a closed control loop (H=0.8), the system was solved for the same initial guess of $k^*=0.8$ and $C^*=10.0$ and an abrupt change in u^* from 1.0 to 0.5. It was found that the surge pressure in the case of the closed loop is higher than that in the open loop as shown as Figures (5-a) and (6-a).

Minimizing the objective function ϕ_1, the optimal values in both cases are reported in Table (1).

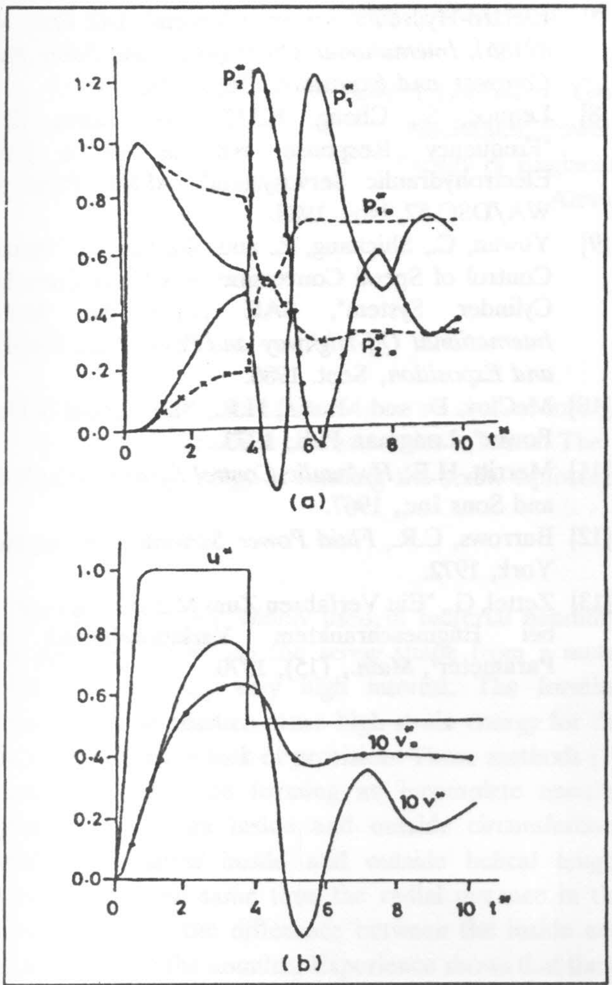

Figure 6. Pressure and Piston Velocity Response of the Closed Loop System, $\Delta u^*=0.5$, $H=0.8$, ($k^*=0.8$, $C^*=10.0$, $k^*_0=18.59$, $C^*_0=9.293$).

In the open loop, the obtained values of k^* and C^* are not practical and the actuator velocity does not change when the change in the input signal is applied at $t^*=3.6$ as shown in Figure (5-b). On the other hand, in the closed control loop, the optimal values of k^* and C^* are feasible. The actuator velocity is reduced from 0.067 to 0.04 in response to the change input signal u^* from 1.0 to 0.5.

	k^*	C^*	ϕ_1	k^*_o	C^*_o	ϕ_1
Without feedback	0.8	10	1.308	30.47	47.55	0.0726
Closed loop (H = 0.8)	0.8	10	2.717	18.59	9.293	0.74

Table 1. Performance of the System with and without Feedback.

Different forms of objective function are investigated. The results are shown in Table (2).

Objective function	k^*	C^*	ϕ	k^*_o	C^*_o	ϕ
ϕ_1	0.8	10	2.717	18.59	9.293	0.74
$\phi_1 + \left\lvert \frac{1}{(P_1 - P_2)} \right\rvert$	0.8	10	5.36	4318	309	0.7731
ϕ_2	0.8	10	1.395	1001.57	396	0.75

Table 2. Effect of Different Forms of Objective Functions.

CONCLUSIONS

From the previous results it can be concluded that the use of the optimal spring stiffness and dashpot damping coefficient in the feedback loop of a hydraulic servo-motor are capable of suppressing the surge during the stop and reversed operation. The obtained optimal values of stiffness and damping coefficient are practical. It was found that the spring stiffness is more important than the damping coefficient in minimizing the objective function. Comparing the closed control loop servo-mechanism suggested in this paper and the technique reported in reference [9],it can be noted that:

1- The speed conversion time required to attain a minimum surge pressure is 0.5 millisecond in this work and in reference [9] the reported time is 0.2 second.

2- The present technique is efficient when different changes in the input signal are applied with different speed while the technique of reference [9] depends on applying the optimal input signal.

Since the minimization technique is unconstrained the choice for value of the stiffness and damping coefficients should be checked from the point of view of practical application.

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