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PERFORMANCE OF AC SERVOMOTOR CONTROL ELECTRO-HYDRAULIC VALVE-LESS SERVOMECHANISM

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ABSTRACT

Valve-less control is a new trend of electro-hydraulic servomechanism. It is possible to position the servo-cylinder directly by controlling the fluid volume discharged from an AC-servomotor drive pump. This paper discusses performance of this system on the point of preciseness, responsibility and stability. It shows efficiency of the system at low power.

KEYWORDS

AC servomotor, Electro-hydraulic control, Valve-less servomechanism, Saving energy

NOMENCLATURE

A_H, A_R :Area of cylinder
α :Ratio of cylinder area $\frac{A_R}{A_H}$
C_{lp} :Inside leakage of pump
D_m :Discharge volume of pump
e :Deviation of control system
K_{ce} :Inside leakage of cylinder
K :Bulk modulus of oil
K_a :Amp gain
K_s :Servo amp constant
M :Mass
$\frac{d\phi}{dt}, N$:Pump revolution
p_1, p_2 :Circuit pressure
p_l :Equivalent pressure difference
q_1, q_2, q_3, q_4 :Circuit flow
V_H, V_R, V :Cylinder chamber volume
y :Cylinder displacement

INTRODUCTION

A servo-valve control system has features of high-speed responsibility and accuracy, but low efficiency of the system. Recent progress of both AC-servomotor and power electronic devices promote valve-less control of the hydraulic servomechanism for saving energy. This paper examines first the steady state position-control preciseness, step- and sinusoidal- responsibilities and next evaluates the power waste of a valve-less servomechanism energized by 1.3kW AC-servomotor driven pump. And a linear mathematical model of the mechanism is expressed.

HYDRAULIC CIRCUIT IN OPERATION

Figure1 shows a hydraulic circuit of the AC-Servomotor drive servomechanism, named as AC servo-pump.
The AC servo-pump discharges fluid to the cylinder in proportion to the electric command signal.

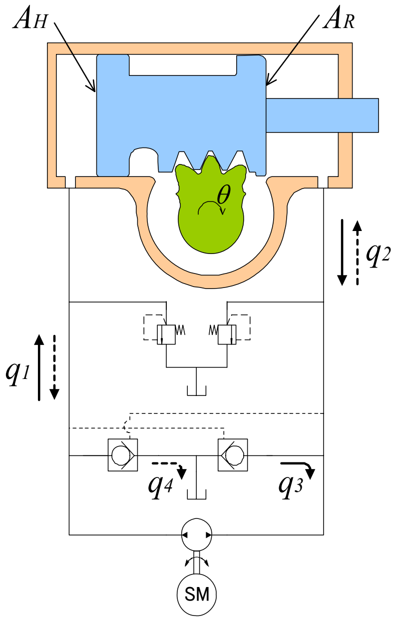

Figure 1 A schematic view of the servo cylinder with hydraulic circuit using AC servomotor

The hydraulic piston actuates a rotary-shaft through the sector gear.
This cylinder is a single-rod type and it is necessary to compensate discharged flow volume. When the cylinder piston moves right direction, flow rate q_2 is discharged from the rod-side chamber. On this case, the pump has to breathe the shortage flow-rate q_3 from reservoir. On the other hand, when the cylinder piston moves left, oil from piston head side chamber returns to the pump suction port, and then excessive flow rate q_4 discharge to reservoir. The rotary angle is measured by a potentiometer and its signal is feed-backed to the controller.

MODELING OF THE SYSTEM

The servo cylinder is a single-rod type, and then the displacement volume of the piston is different in right or left motion.
Flow continuity in the cylinder head chamber is

$$q_1 - C_{lp}(p_1 - p_2) = A_H \frac{dy}{dt} + \frac{V_H}{K}\frac{dp_1}{dt} \quad (1)$$

Excessive flow rate q4 has to be discharged to reservoir.

$$C_{lp}(p_1 - p_2) + A_R \frac{dy}{dt} = q_2 + \frac{V_R}{K}\frac{dp_2}{dt} \quad (2)$$

$$q_1 = \frac{D_m}{2\pi}\frac{d\phi}{dt} - Kce(|p_1 - p_2|) \quad (3)$$

$$q_3 = C_3 A\sqrt{p_2} \quad (4)$$

When high pressure oil from pump flows into the cylinder-rod chamber, the return flow rate from the cylinder head excesses to the pump flow rate, which is discharged to the reservoir through the pilot operated valve.

$$C_{lp}(p_1 - p_2) + A_R \frac{dy}{dt} = q_2 + \frac{V_R}{K}\frac{dp_2}{dt} \quad (5)$$

$$(q_1 - q_4) - C_{lp}(p_1 - p_2) = A_H \frac{dy}{dt} + \frac{V_H}{K}\frac{dp_1}{dt} \quad (6)$$

$$q_4 = C_4 A\sqrt{p_1} \quad (7)$$

$$q_4 = C_4 A\sqrt{p_1} \quad (8)$$

where C_{lp} is leakage coefficient of the cylinder, K is bulk modulus of oil, V_H , V_R are cylinder chamber volume with piping inner volume.
A motion equation of the actuator is expressed as follows.

$$A_H P_1 - A_R P_2 = M\frac{d^2y}{dt^2} + C\frac{dy}{dt} + F_D \quad (9)$$

The cylinder displacement y is transformed output angle θ through by sector gear.

$$\theta = Ly \quad (10)$$

Motion equations of AC servomotor and hydraulic pump describe as follows.

$$\frac{d\phi}{dt} = Ksei \quad (11)$$

$$e = \theta_0 - \theta \quad (12)$$

$$ei = K_{ae} \quad (13)$$

$$J_m\frac{d^2\phi}{dt^2} + B_1\frac{d\phi}{dt} = \tau \quad (14)$$

$$\tau = Jp\frac{d^2\phi}{dt^2} + B_2\frac{d\phi}{dt} + \frac{D_m}{2\pi}(p_1 - p_2) + F_P \quad (15)$$

where K_S is servomotor constant, J_m , J_P are moment of inertia servomotor and pump respectively, Dm is discharge volume of pump.

This system is not a linear system because hydraulic oil paths differ from by depending on the movement direction of the cylinder. Assuming to ignore terms of a little influence in the equations, we can get a linear system as shown next chapter.

A LINEAR MODELING OF THE SYSTEM

Both sides of the cylinder piston area are not the same, so that it cannot generally apply using usual pressure difference to simplify equations of the system. Substituting 'equivalent pressure difference p_l ' into the relevant equations, as a result, we can simplify the equations as follows.

$$\alpha = \frac{A_R}{A_H} \quad (16)$$

$$p_l = p_1 - \alpha p_2 \quad (17)$$

$$A_H(p_l) = M\frac{d^2 y}{dt^2} + C\frac{dy}{dt} + F_D \quad (18)$$

Assuming $q_l = q_1 \cong q_2$ at Eq. (5),(6)

Substituting Eq. (17) into Eq. (5) and (6), we obtain equations as follows.

$$q_l - C_{lp}(p_l) = A_H\frac{dy}{dt} + \frac{V}{2K}\frac{dp_l}{dt} \quad (20)$$

$$q_l = \frac{D_m}{2\pi}\frac{d\phi}{dt} - Kce(|p_l|) \quad (21)$$

$$\frac{d\phi}{dt} = K_s e_i \quad (22)$$

$$e = \theta_0 - \theta \quad (23)$$

$$e_i = K_a e \quad (24)$$

Block diagram of angle control system shows in Figure 2.

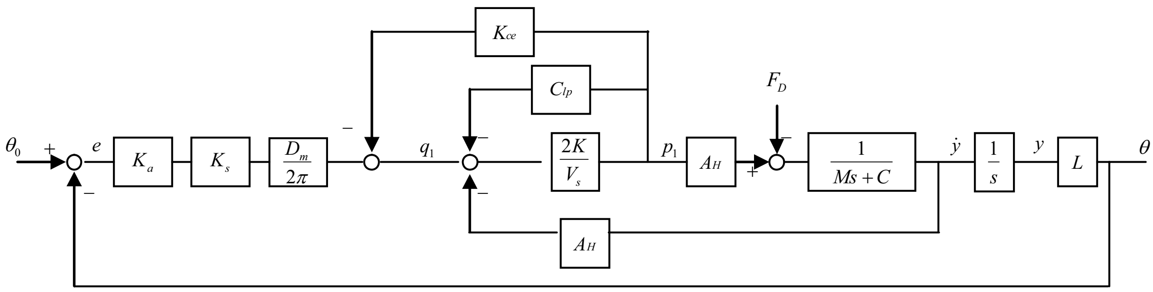

Figure 2 Block diagram of the angle control system using the valve-less servomechanism

We can obtain Eq. (25) by rearranging the block diagram shown in Figure 2.

$$\theta = \frac{KaKs\frac{D_m}{2\pi}A_H}{\frac{VM}{2K}s^2 + (KcM + \frac{VC}{2K})s + KcC + A_H{}^2}\frac{1}{s}L \cdot e \quad (25)$$

A block diagram of Eq. (25) is shown in Figure 3.

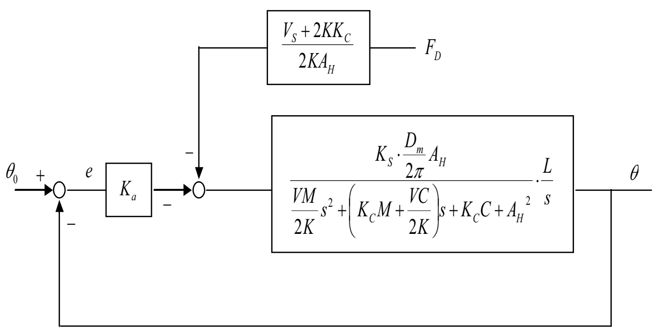

Figure 3 Block diagram of the valve-less servomechanism expressed by using Eq. (25)

Replacing new variables into each term of a fraction in Eq. (25), we can get simplified equation as shown in Eq. (26).

$$G(s)=\frac{\theta(s)}{e(s)}=\frac{kp}{(\frac{s^2}{\omega^2}+2\varsigma\frac{s}{\omega}+1)}\frac{1}{s} \qquad (26)$$

where

$$\omega=\sqrt{\frac{2KA_H{}^2}{VM}} \quad , \quad \zeta=\frac{Kc}{A_H}\sqrt{\frac{KM}{2V}}+\frac{C}{A_H}\sqrt{\frac{V}{2KM}} \quad ,$$

$$kp=\frac{KaKsD_m}{2\pi A_H}L$$

We evaluate the system stability of Eq. (26) by applying in Hurwitz criterion.

$$\frac{kp}{\omega}\langle 2\zeta \qquad (27)$$

Characteristic equation of the system has 3 orders of polynomial in s. We can get stability condition as shown in Eq. (27). The system has stability, if proper figures setting into K_a satisfy the condition of Eq. (27).

Evaluating of eq. (27), it is shown in $\omega=1{,}890$ **rad/s**, and $\frac{kp}{\omega}=0.016$. It means that the system is stable if ζ is more than 0.08.

EXPERIMENTAL RESULTS

A gain tuning of the system has only adjusted proportional gain. Test result of a static characteristics is shown in Figure 4, which has a good linearity without hysteresis.

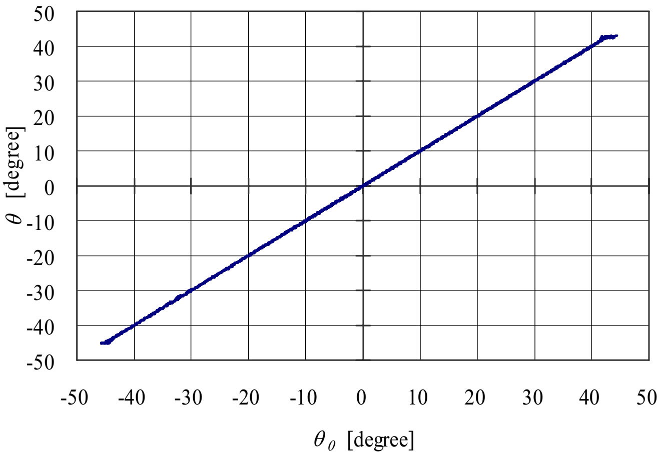

Figure 4 Static characteristic of command vs. output shaft angle

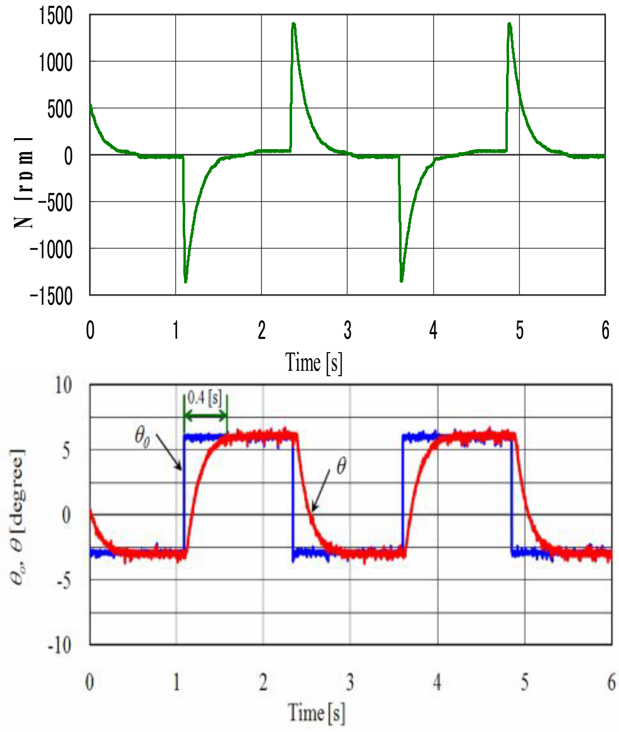

Figure 5 Step response of 10% command signal

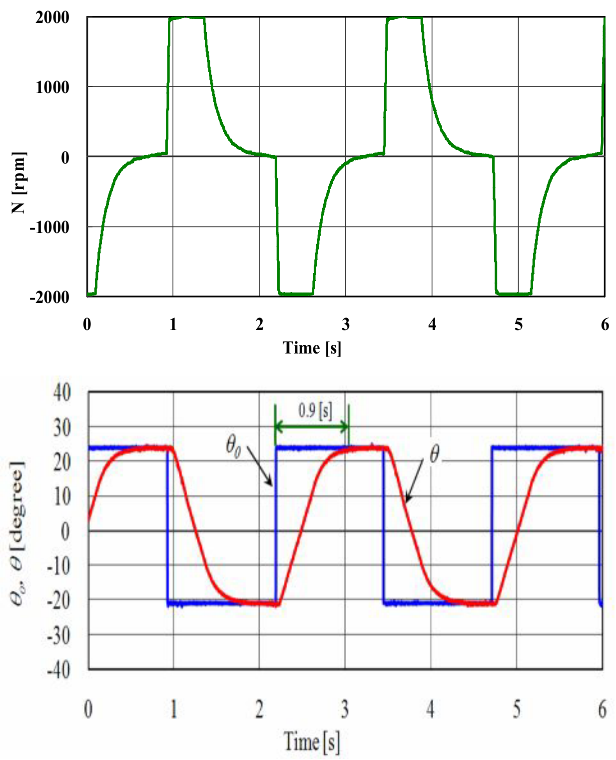

Figure 6 Step response of 50% command signal

Figure 5 and 6 show step responses of rotary angle θ and servo-motor rotational speed N at 10% and 50% amplitude, respectively. We can see that the delay at 10% amplitude is only 0.4s, while it is 0.9s at 50% due to the saturation of motor-speed at Nmax=2000rpm.

Bode diagram of the system is shown in Figure 7. The frequency response is expressed 1.2 Hz at -3dB.

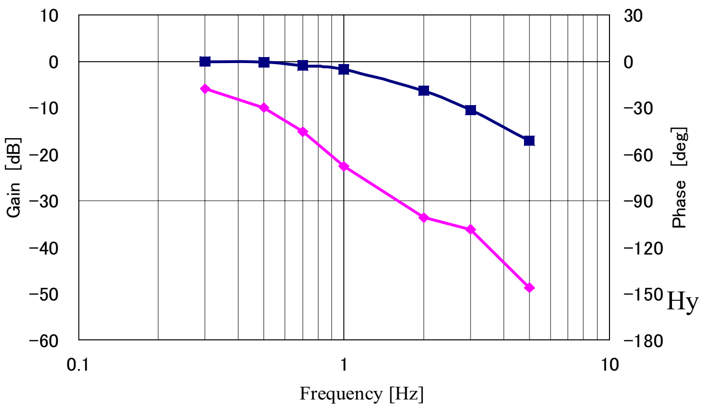

Figure 7 Bode diagram at 10% amplitude command

Experiments of efficiency have conducted under low inertia load, that sinusoidal command signal provides the system. Electric power consumption is measured by a wattmeter in supply line of AC 200V connecting with AC servo amp. The hydraulic output power are also measured.

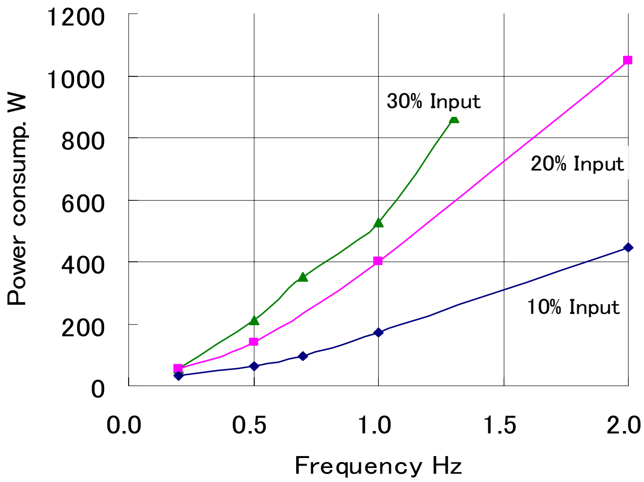

Figure 8 Measured electric power of the AC servo pump in sinusoidal motion of the piston with 10 to 30% amplitude

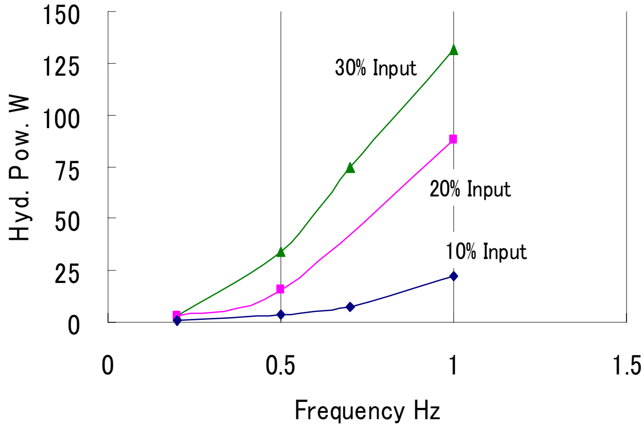

Figure 9 Measured hydraulic power of the AC servo pump in sinusoidal motion of the piston with 10 to 30% amplitude

Mass of the system assesses only 16.6kg. And force of mass inertia is low because of within a few Hz, it can neglect. Servo amp for 1.3kW AC servomotor consume about 40W in case of no operation.
We can realize that electric power consumptions increase in proportion to the frequency as shown in Figure 8.
Hydraulic output power shows in Figure 9 on measuring under the same condition of Figure 8, evaluate average hydraulic power from test results. Figure 10 shows efficiency η of the valve-less servo system, which is ratio of hydraulic output power and electric power consumption.

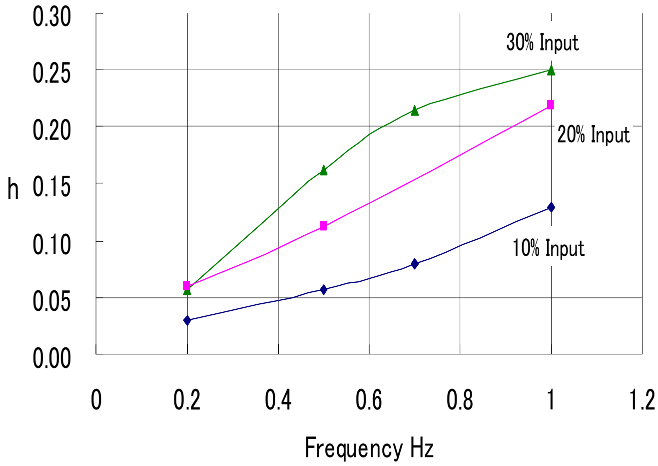

Figure 10 Efficiency of the valve-less servomechanism at low inertia load

On the other hand, we evaluate efficiency of ordinary servo system.
Assuming generating the same output hydraulic power shown in Figure 9, hydraulic power supply for ordinary servo system needs at least 2.4kW, which means flow rate 12l/min and 12MPa. Calculated efficiency ηs of the ordinary servo system, is shown in Figure 11.
Efficiency ηs of ordinary servo system attains only up to 5% under 30% amplitude at 1Hz, as a result.

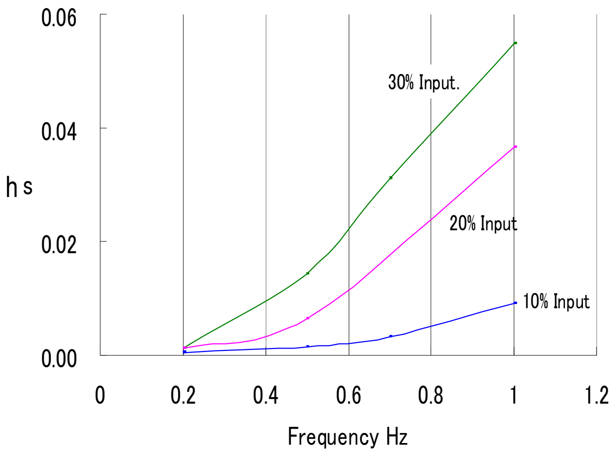

Figure 11 Efficiency of the valve-less servomechanism at low inertia load and Ps=12MPa

On the contrary, efficiency of the valve-less servomechanism is about 25% under 30% amplitude at 1Hz.
The efficiency of the valve-less servomechanism is obviously superior to ordinary servo system.
It shows that features of valve-less servomechanism and hydraulic rotational speed control.

CONCLUSION

This paper examines the control preciseness and response speed of the valve less servo actuator driven by a 1.3kW AC servo-pump. It controls a rack and pinion cylinder piston with adequate linearity and low hysteresis as shown in Figure 4, and positions the rotary actuator with 17kg mass within a short lag time of 0.4s under the range of maximum rotational speed of 2000rpm of the AC servo motor as shown in Figure 5. The efficiency of the valve-less control system at a 1Hz sinusoidal oscillation of the mass is about 25%, while the current valve control system is estimated only 5.5% as shown in Figures 10 and 11. The valve-less control system has potential to be applied not only to injection molding machinery but also to construction machinery, automobile auxiliary control systems and train suspensions.

ACKNOWLEGEMENT

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