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Discussion

Investigation of an Electrohydraulic Servovalve With Tuneable Return Pressure and Drain Orifice[1]

Hussein El-Gammal.[2] 1. In developing the equations of the system, equation (1) gives the flow rate to the actuator as:

$$q_1 = \frac{C_d A_v \sqrt{2(P_s - P_1)/\rho}}{Q_s} \tag{1}$$

This expression overlooks the flow rate through the flapper nozzles. In the attached diagram, which is an enlarged copy of Fig. 16, for a flapper deflection resulting in the spool position shown in a broken line, the flow rate to the actuator should be:

$$q_1 = \frac{C_d A_v \sqrt{2(P_s - P_1)/\rho}}{Q_s} + q_d \tag{2}$$

$$q_d = q_{n1} + q_{n2} \tag{3}$$

and

$$q_{n1,2} = \frac{C_c \Pi d_{n1,2} (y_0 \pm y) \sqrt{2(P_{n1,2} - P_d)/\rho}}{Q_s} \tag{4}$$

where,

P_d: pressure down-stream the flapper nozzles
$P_{n1,2}$: pressure up-stream flapper nozzles
q_d: the nondimensional flapper-nozzles flow rate
y: flapper deflection
y_0: flapper position at neutral or balanced position.

It seems to me that the inclusion of q_d in the analysis is necessary. The effects resulting from this on the transient response parameters (over shoot, settling time, and steady state speed) should in my opinion be investigated.

2. Equations (33) and (34) should include the effect of the piston movement i.e., $A\dot{x}$ and accordingly they should read:

$$\frac{dP_1}{dt} = \frac{\beta}{V_1}\left[Q_1 - A\dot{x} - C_1(P_1 - P_2)\right]$$

and

$$\frac{dP_2}{dt} = \frac{\beta}{V_2}\left[-Q_2 + A\dot{x} + C_1(P_1 - P_2)\right]$$

3. The flow rate through the drain orifice should be checked whether laminar or turbulent. Also conditions relating to the possible onset of cavitation in the return line and its components should be investigated.

4. I am particularly interested to know more about the dynamic behavior of the system in the frequency domain. I expect that the inclusion of a transportation lag due to the directional control on-off valve in addition to the increased volume of oil in the system will lead to deteriorating the frequency response characteristics of the servovalve. The system band width, phase and gain margins and stability criteria are particularly sensitive to the additional components and circuitry.

[1] By S. LeQuoc, R. M. H. Cheng, and A. Limaye published in the September 1987 issue of the JOURNAL OF DYNAMIC SYSTEMS, MEASUREMENT, AND CONTROL, Vol. 109, p. 276.

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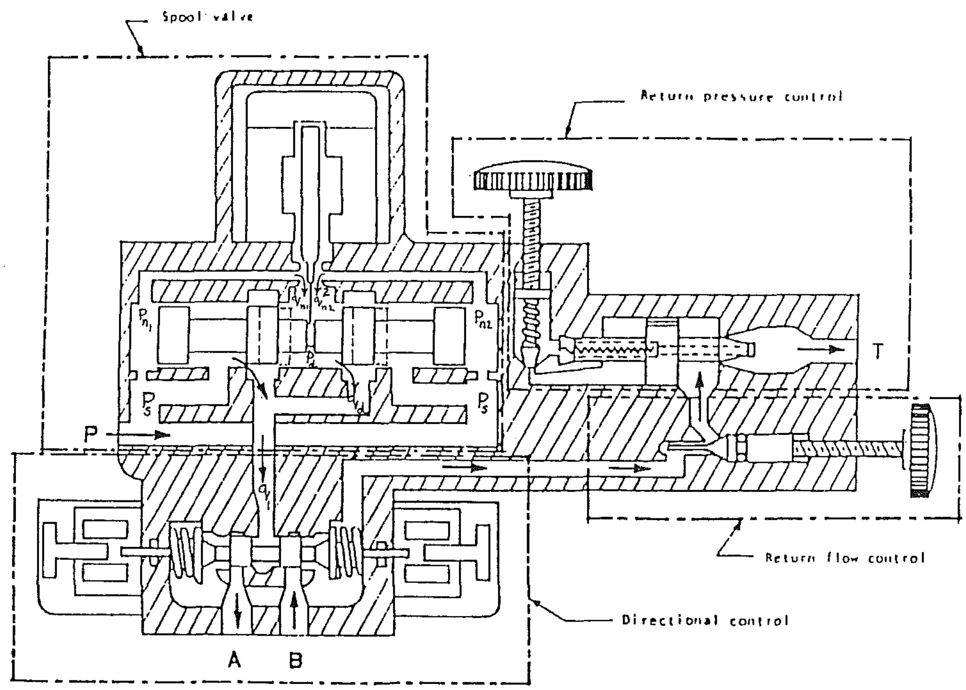

Fig. 16 Single block new servovalve configuration

Authors' Closure

In response to Dr. El-Gammal's discussion we offer the following comments:

1. The flapper-nozzles flow rate q_d mentioned in the discussion is normally referred to as "internal leakage" by servovalve manufacturer.

This leakage is generally maximum at the valve null, called "null leakage." The servovalve used in our experimental setup is Moog, model 760-103 which has a rated flow of 10 gpm with 1,000 psi supply pressure. The maximum internal leakage specified by the manufacturer is 0.17 gpm which represents less than 2 percent of the nominal rated flow. Therefore, the fact that q_d is not considered in our analysis seems very realistic.

2. In fact, equations (33) and (34) must include the effects of the piston movement, i.e., $A\dot{x}$, as suggested. The omission of this term is purely a typing mistake.

3. The calibration of the drain orifice at different openings indicates generally a turbulent flow.

4. The frequency response analysis of a servosystem with rotary actuator indicates that the resonance frequency of the new configuration has dropped by a factor of about $1/\sqrt{2}$ (Ref.: Paper #88-WA/DSC-52, 1988 ASME Winter Annual Meeting, Chicago, Nov. 27-Dec. 1, 1988). It is believed that this is mainly due to the fact that only one chamber's volume contributes on the oil spring because only one line is controlled by the harmonic orifice area input signal, while the other line has a fixed restrictor area.

Regarding the gain and phase margins, it is demonstrated that a proper tuning of the return pressure and drain orifice area would lead to a servosystem with higher amplitude ratio at low frequency while maintaining a lower peak amplitude in the resonance frequency range. This confirms the result obtained from the step input study which states that the new configuration can be tuned to obtain a high gain yet high damping system.