Model Improvement for a Servovalve with Force Feedback and Back Pressure

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Abstract— Electro-hydraulic servovalve are usually an important part in mechatronic systems. The simplified governing equations for such systems already exist in literature. A comprehensive theoretical analysis of a two-stage electro hydraulic servo valve with a spool position feedback is carried out considering two main probable effects, underlap and back pressure. These analyses are based on fundamental laws of electromagnetism, fluid and general mechanics, and rectangular ports to simplify the equations. A detailed mathematical model of servovalve with circular ports (the real configuration) is developed to improve the accuracy of the model. Besides, the back pressure in the pilot region of the flapper nozzle servo valve is considered. Obtained simulation results are compared with the appropriate experimental data in literature through which the validity of the proposed model is confirmed with good accuracy.

The effects of the underlap spool and the back pressure on the performance, stability and response of the whole system are investigated through solving the governing equations in Matlab simulink. Moreover, the ways to get into better responses will be suggested.

Keywords— Servovalve, Flapper-Nozzle, Force feedback, underlap, Back pressure

I. INTRODUCTION

Electrohydraulic servovalves are used as interfaces between electrical devices and hydraulic systems. They are capable of converting low electrical inputs signals for precise movements of spools to control large-power on low-speed hydraulic actuators. Thus they are used extensively in numerical controlled machine tools, aircrafts, remotely controlled mechanisms, and in all kinds of off-highway and mobile applications.

So many works have been done to improve dynamic equations of servo valves. One of the most important references in this field is [1]. The principle of electrohydraulic system design is given and also functioning and design of spool valves, flapper-nozzle, torque motors, etc. is explained thoroughly.

Basically, two different approaches can be used to obtain linear mathematical model that describe the behavior of electrohydraulic servovalves. Either dynamic of the servovalve is neglected and the system is modeled with a first, second or even third-order transfer function, depending on the dynamic characteristic of a system that contains the servovalve. The values of time constants, undamped natural frequencies and damping ratios could be calculated from the experimentally determined servovalve frequency characteristics that could be found in a manufacturer's catalogues [2]. Manufacturers principally propose a third-order model for servovalve and a second-order model for the first stage flapper-nozzle [3].

The other approach implies theoretical or theoretical-experimental modeling and linearization about some characteristic working regime (the null position is frequently used) in order to obtain linear mathematical models. Nevertheless, certain phenomena or physical quantities that are considered to be of less importance are neglected. Researchers thereby propose higher order models presented in the form of transfer function or state-space equations [4, 5 and 6].

Many authors performed the non-linear analysis of various servovalve types and in this way gave contributions to their better understanding and a more realistic explanation of their behavior. In [7] authors made a special review on torque caused by electromagnetic forces, the deformation caused feedback spring and the deformation of the flapper. A non-linear mathematical model based on physical quantities was developed in [8]. This model includes nonlinear relations for the torque motor dynamics and a flow force on the flapper, fluid compressibility. The first stage control volume change due to a spool movement, the first stage leakage, and flow forces. In [9, 10] authors analyzed the torque motor dynamics and the elastic structure dynamics in detail.

In previous researches, all have supposed rectangular, ports for simplicity, but the circular ports are presented here to have more realistic configuration and improve the accuracy of the model.

In this work a precise modeling of a two-stage flapper-nozzle servo valve with force feedback is carried out considering back pressure in the pilot region and the results are validated by comparing with experimental results in literature.

Moreover, modeling of an under lap two-stage flapper-nozzle servo valve with force feedback is carried out. Besides the results are compared with the critically lapped valves and response improvement is discussed.

The back pressure effects are also illustrated, discussed and compared with the systems without back pressure. The back pressure was produced by constricting the drain port in pilot region.

II. Flow Forces on Flapper

It has been stated that the significant force on the flapper is that resulting from the static pressure acting on the nozzle area projected onto the flapper. The dynamic pressure also acts on the nozzle area projected onto the flapper, and it is of interest to include this effect. Consider the flapper-nozzle configuration in Fig. 1, in which the flapper is mounted on a torsion rod (torsion spring). Using Bernoulli's equation, we give the force F_1 by

$$F_1 = \left(P_1 + \tfrac{1}{2}\rho u_1^2\right)A_N \tag{1}$$

Where

$$\begin{aligned} u_1 &= \frac{Q_2}{A_N} = \frac{C_{df}\pi D_N\left(x_{f0}-x_f\right)\sqrt{(2/\rho)(P_1-P_b)}}{\pi D_N^2/4} \\ &= \frac{4C_{df}\left(x_{f0}-x_f\right)\sqrt{(2/\rho)(P_1-P_b)}}{D_N} \end{aligned} \tag{2}$$

Combining (1) and (2), we obtain

$$F_1 = \left[P_1 + \frac{16C_{df}^2\left(x_{f0}-x_f\right)^2(P_1-P_b)}{D_N^2}\right]A_N \tag{3}$$

Similar reasoning leads to the equation for F2. So the net force acting on the flapper is the difference of these forces.

$$F_1 - F_2 = (P_1-P_2)A_N + 4\pi C_{df}^2\begin{bmatrix}\left(x_{f0}-x_f\right)^2(P_1-P_b) - \\ \left(x_{f0}+x_f\right)^2(P_2-P_b)\end{bmatrix} \tag{4}$$

By approximating $P_1 \approx P_2 \approx P_s/2$, the steady-state value, we obtain the relation

$$\begin{aligned} F_1 - F_2 &= P_L A_N + 4\pi C_{df}^2 x_{f0}^2(P_1-P_2) + 4\pi C_{df}^2 x_f^2 P_L \\ &- 8\pi C_{df}^2 x_{f0}P_s x_f + 16\pi C_{df}^2 x_{f0} x_f P_b \end{aligned} \tag{5}$$

Where

$$P_L = P_1 - P_2 \tag{6}$$

$$P_s = P_1 + P_2 \tag{7}$$

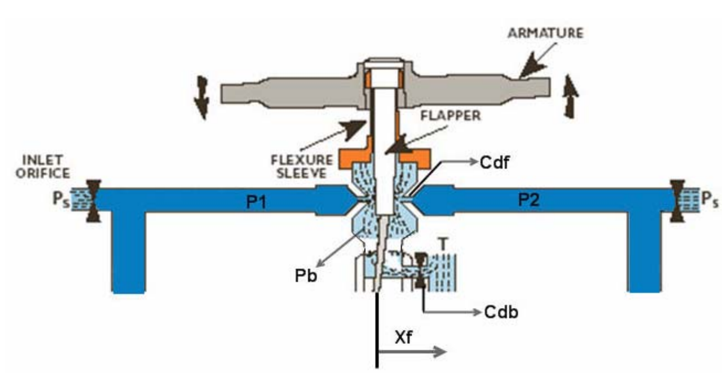

Figure1. Enlarged view of nozzles (Courtesy of Moog)

Good flapper valve design requires that $x_{f0}/D_N < (1/16)$ [1], which makes the second term negligible compared with the first term. The third term can also be neglected because $P_L \approx x_f \approx 0$ near null where most operation occurs. By using the equation $x_f = r\theta$ for small angles equation (4) can be approximated by

$$F_1 - F_2 = P_L A_N + \left(8\pi C_{df}^2 x_{f0}(2P_b - P_s)\right)r\theta \tag{8}$$

III. Flapper Motion Equations

Total torque developed on the armature due to electrical current input (T_d) together with (8) leads to.

$$T_d = K_t\,\Delta i + K_m\Theta \tag{9}$$

Applying Newton's second law to the armature

$$T_d = J_a\frac{d^2\theta}{dt^2} + B_a\frac{d\theta}{dt} + K_a\theta + T_L \tag{10}$$

Referring to Fig.2, the total effect of the cantilevered feedback spring at its end is $[(r+b)\theta + x_v]$

$$T_L = (F_1-F_2)\,r + (r+b)\,k_f[(r+b)\,\Theta + x_v] \tag{11}$$

Now, using equations (9), (10) and (11), the armature torque equation becomes

$$\begin{aligned} K_t\,\Delta i &= J_a s^2\theta + B_a s\theta + K_{am}\theta + rP_L A_N + \\ &(r+b)K_f\left[(r+b)\theta + x_v\right] \end{aligned} \tag{12}$$

Where $\quad K_{am} = K_a - K_m + 8\pi C_{df}^2\, r^2\, x_{f0}\,(2P_b - P_s)$ (13)

IV. Spool Equations

An improved model was developed with circular ports on the spool valve's sleeve and the dead zone created by the under lapped spool valve landings. Fig. 3 shows an overall schematic of the servovalve with an under lapped, non-symmetrical spool valve. A schematic of the spool valve landing and circular port in its neutral position is shown in Fig. 4.

The areas of A_e and A_s changes with spool position. These areas together with the areas of a partially uncovered circular orifice is calculated and presented below [11].

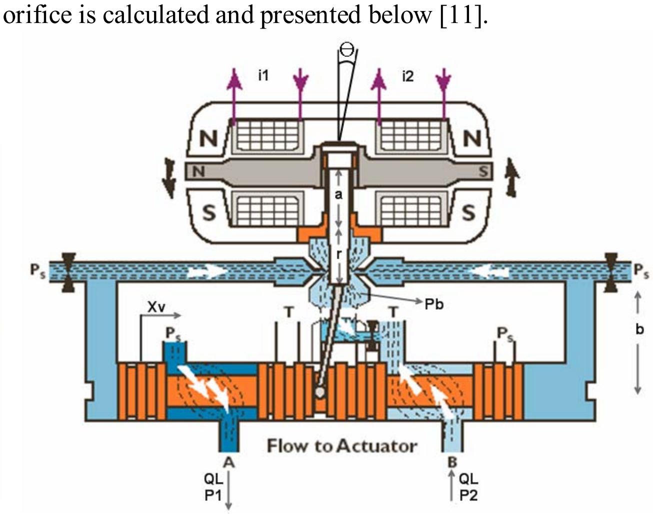

Figure2. Two-stage EHSV with force feedback (Courtesy of Moog)

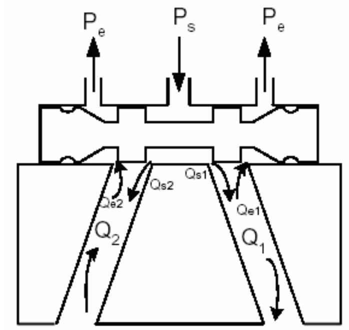

Figure 3.Under lapped, Non-symmetrical Spool valve

The calculations for circular orifices are significantly more complex than early works on rectangular orifices [13, 14, 15, 16, 18 and 19]. Equations (14) and (15) represent an approximation method for calculating the areas of the circular segment. The error in area calculation for opening angle between 0° to 150° is less than 0.1%. These equations are derived from equation 3 for a circular segment as in Fig. 3. Comparisons for several spool displacement values throughout the range are carried out using different methods.

$$A_s = \frac{2}{3}\left(h_s\sqrt{8.h_s.r - 4h_s^2}\right) + \frac{h_s^3}{2\sqrt{8h_s.r - 4h_s^2}} \quad (14)$$

$$A_e = \frac{2}{3}\left(h_e\sqrt{8.h_e.r - 4h_e^2}\right) + \frac{h_e^3}{2\sqrt{8h_e.r - 4h_e^2}} \quad (15)$$

$$A = \frac{r^2}{2}(\varphi - \sin(\varphi)) \quad (16)$$

Improvements to the model are based on an accepted modeling technique outlined by Merritt [1]. This model, like other accepted models was based on rectangular port geometries. Here model calculates a load flow rate, $\dot{Q}_L$ as the flow entering the piston chamber and ultimately displacing the servo valve's piston. Equation 4 shows load flow rate as a function of rectangular port area resulting from width (W) and spool valve displacement (X_v).

$$\dot{Q}_L = C_d.W.X_v\sqrt{\frac{1}{\rho}(P_s - P_L)} \quad (17)$$

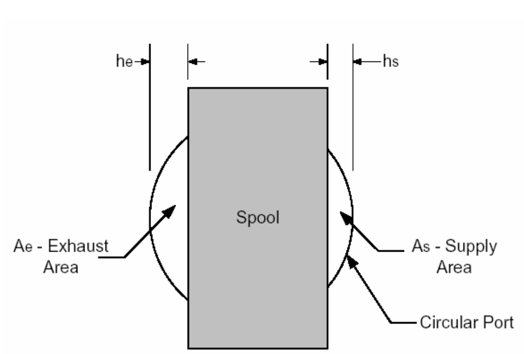

Figure 4.Undercut Spool valve Landing and Circular Port in neutral Position

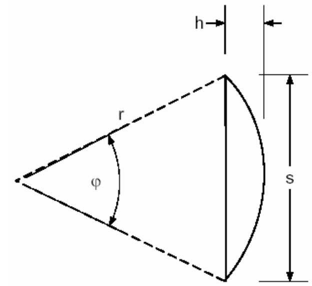

Figure 5.Circle Segment

Here the area term is replaced with a net area term which represents the summation of area for the under lapped circular port. The net area is the area available for flow into the piston chamber which is the difference between exhaust and supply area, Fig. 4. Net area is calculated using (14) and (15). The use of a net area indicates if the exhaust area is larger than the supply area, the fluid flow will simply circulate through the spool valve; see Fig. 6. However, as the supply area is larger than the exhaust area, a positive net flow is available. This positive net flow, or load flow, represents the fluid diverted to the piston chamber and available for piston displacement.

The supply, exhaust, and net areas are calculated for different spool valve displacement (-130µm<X_v<+130µm). These calculations can be seen graphically in Figure 6 where the net area is nearly a linear function of spool position. Therefore, the net area was curve fit as a linear function for flow rate into side 1 and 2 of the piston. The resulting linear equations, using base units of microns and square microns are shown in (18) and (19). Equation (18) represents fluid flow areas into chamber 1 of the servo valve's spool, resulting in positive spool displacement. Equation 6 is the fluid flow area into spool chamber 2, which would result in negative spool displacement.

$$A_{1\text{-net}} = +5264.7X_v - 502794 \quad (18)$$

$$A_{2\text{-net}} = -5264.7\,X_v - 502794 \quad (19)$$

Equation (4) can be modified by using the above relations,

$$\dot{Q}_L = C_d.A_{net}.\sqrt{\frac{1}{\rho}(P_s - P_L)} \quad (20)$$

This equation can be linearized about an operating point based on small perturbation technique. The operating point can be taken to be the point where the net flow area is zero at the dead zone transition point, i.e. the end of the dead zone. For the existing system this is around ±90 µm.

$$\dot{q}_L = \left.\frac{\delta \dot{Q}_L}{\delta X_v}\right|_a .X_v + \left.\frac{\delta \dot{Q}_L}{\delta P_L}\right|_a .P_L \quad (21)$$

$$\dot{q}_L = K_q\big|_a .X_v + K_c\big|_a .P_L \quad (22)$$

The flow gain, K_q, is the partial derivative of the load flow rate with respect to spool valve position, about the operating point.

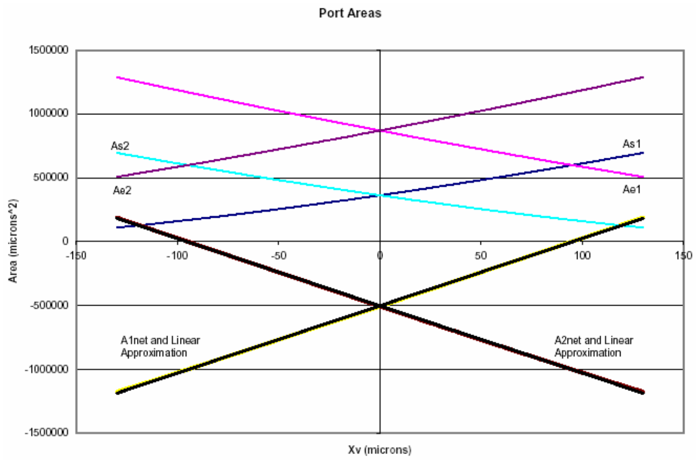

Figure 6.Port Area Calculations

The flow-pressure coefficient, K_c, is the partial derivative of the load flow rate with respect to load pressure about the same operating point.

$$K_q = C_d \cdot \sqrt{\frac{2}{\rho}\left(\frac{P_s - P_L|_a}{2}\right)} . \frac{dA_{net}}{dX_v} \tag{23}$$

$$K_c = \frac{-C_d \cdot A_{net}|_a}{2} . \sqrt{\frac{1}{\rho(P_s - P_L|_a)}} \tag{24}$$

The net areas are derived from (18) and (19). These equations represent flow area into the two sides of the spool. Inlet and outlet flows for either side of the valve are shown in Fig. 3. Sides 1 and 2 represent the two chambers of the servo valve's spool. For the present application, flow into chamber 1 results in a positive displacement of the spool, while flow into the chamber 2 results in negative displacement.

In K_q relation, the slope of the net area equations is the only contribution from the net area calculations. The derivatives from A_{net} equations reveal identical values with opposite sign. This sign is necessary to calculate the direction of flow into either chamber, dependent upon the value and sign of the spool valve's position. However, considering the incompressibility of the fluid, a positive flow into one chamber of the spool is identical to a flow out from the other chamber. Therefore, only one of the net flow area calculations is necessary to describe the displacement of the hydraulic piston as a function of the displacement of the spool valve. This simplification is valid because for a positive spool valve displacement, the net area increases allowing for flow into the positive chamber, resulting in positive displacement of the piston. For positive values of X_v, the calculated net flow area is for flow into chamber 1, resulting in positive displacement. For negative values of X_v, the calculation provides the net area for flow out of this chamber. The derivative of equation is needed for the calculation of flow gain, K_q. This is valid when the dead zone is considered. If both net area equations are negative, no net flow is entering either side of the spool, hence the dead zone. This causes nonlinearity in the system and it will be introduced in the description of the overall system transfer function.

The transfer function for an under lapped valve is [1].

$$x_p = \frac{\frac{K_q}{A_p} x_p - \frac{K_{ce}}{A_p^2}\left(1 + \frac{V_t}{4\beta_e K_{ce}} s\right) F_L}{s\left(\frac{s^2}{\omega_h^2} + \frac{2\delta_h}{\omega_h} s + 1\right)} \tag{25}$$

In two-stage electro hydraulic servovalve with force feedback, the spool acts as a piston load. Flow forces on the spool are spring type load together with spool inertia force. The transfer function for such a valve-piston combination is [1].

$$\frac{x_v}{x_e} = \frac{\frac{K_{qp}}{\omega_f A_v}}{\left(\frac{s}{\omega_f} + 1\right)\left(\frac{s^2}{\omega_{hp}^2} + \frac{2\delta_{hp}}{\omega_{hp}} s + 1\right)} \tag{26}$$

where

$$x_e = x_f - x_v \tag{27}$$

$$\omega_{hp} = \sqrt{\frac{2\beta_e A_v^2}{V_{vp} M_v}} \tag{28}$$

$$\delta_{hp} = \frac{\omega_{hp} K_{cp} M_v}{2 A_v^{\ 2}} \tag{29}$$

$$\omega_f = \frac{0.43 w P_s K_{cp}}{A_v^2} \tag{30}$$

V. Torque Motor

The voltage drop around the coil circuit can be written [1]

$$\mu e_g = \frac{1}{2}(R_c + r_p)\left(1 + \frac{s}{\omega_a}\right)\Delta i + K_a s \theta \tag{31}$$

VI. Block Diagram

Combining the previous equations results to a complete block diagram of the servovalve with force feedback where under lap and back pressure effects are considered. Fig.7 shows the final block diagram of an under lapped flapper-nozzle servo valve with force feedback.

$$G_a = \frac{\frac{2K_t}{R_c + r_p}}{1 + \frac{s}{\omega_a}} \tag{32}$$

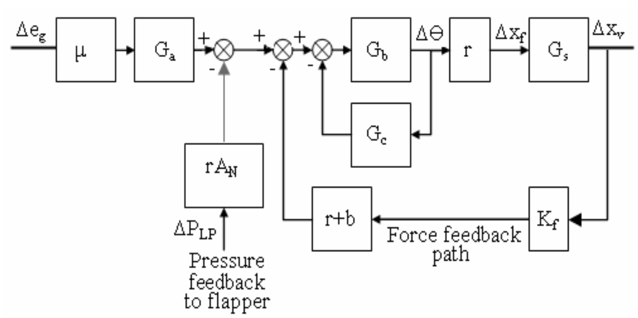

Figure 7.Block diagram of EHSV with force feedback

$$G_b = \frac{1}{J_a s^2 + B_a s + K_f (r+b)^2 + K_{am}} \quad (33)$$

$$G_c = \frac{(\frac{2K_t K_b}{R_c + r_p})s}{1 + \frac{s}{\omega_a}} \quad (34)$$

$$G(s) = \frac{x_v}{x_e} = \frac{\frac{K_q}{\omega_f A_v}}{(\frac{s}{\omega_f}+1)\left(\frac{s^2}{\omega_h^2} + \frac{2\delta_h}{\omega_h}s + 1\right)} \quad (35)$$

VII. Model Validation

Modeling of block diagram in Matlab, values derived from [1], [10], [18] and [19]. The results of the numerical analysis were compared with appropriate experimental results in order to verify the model. Aero Shell Fluid 4 which is a mineral oil was used as a working fluid. The experiment was performed at the constant ambient temperature of 20ºC. The supply pressure was 210 bar and the flow rate was 40 l/min in all cases.

In Fig. 8 the red line shows the experimental result and the blue dash shows the Matlab Simulation output. It is clear that the model represents the data reasonably well, but due to the linearization of several parameters, and especially the inability to capture the spool valve's dead zone, they do not precisely capture the nuances of the valve's movement. Furthermore, the transport delay associated with the under lapped spool valve in the actual system is not considered.

VIII. Comparing different conditions

A. Back pressure effects

In Fig. 9 the response of the valve by increasing the pressure inside the nozzle region is demonstrated. The solid (red) line shows the experimental results and the dashed (blue) shows the simulation results. Approximately coincidence of both plots lead to the same conclusion.

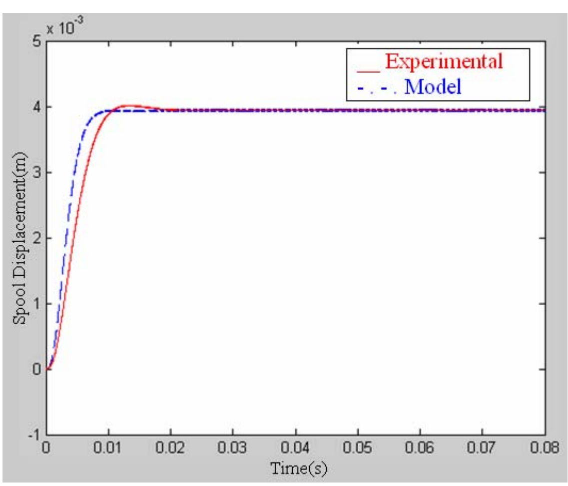

Figure8. Comparing model and experimental response (displacement of the spool) actuated with a step input neglecting the back pressure

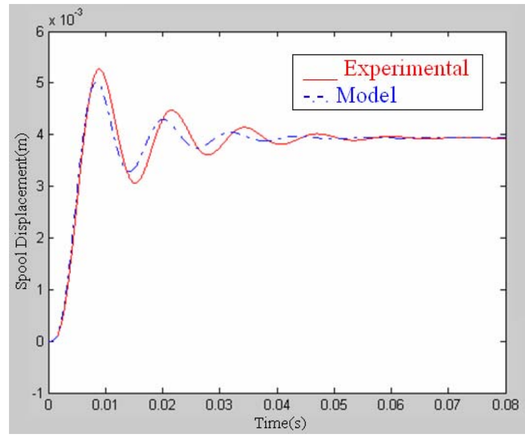

Figure 9.Displacement of spool actuated with a step input considering the back pressure

Comparing the Fig. 8 and 9 shows that even though, the pressure inside the pilot region has decreased the rise time, the settling time and overshoot of the whole system has increased.

As the most important parameter in controlling systems are having small settling time and small overshoot, increasing the pressure inside the pilot region must be avoided. The increasing of pressure is because of constricting the drain port. Moreover closing the drain port of the zone causes instability in the system.

So if oscillation in the response of a flapper-nozzle servo valve occurred, it can be the result of increasing the pressure inside the pilot region. The most important cause of increasing the pressure inside the pilot region is constricting the drain port. To prevent such happening, the easiest and the best way is better filtration. Using cleaner oil is the best prevention of entering contamination inside the valve body and also the pilot region.

B. Underlap effects

The response of a critically lapped and under lapped servo valve neglecting back pressure effects are shown in Fig 10.

It is clear that a spool which is under lapped increases the response time. As one of the most important characteristics of servo valve is their fast response, using an under lapped valve are not recommended in a fast response system.

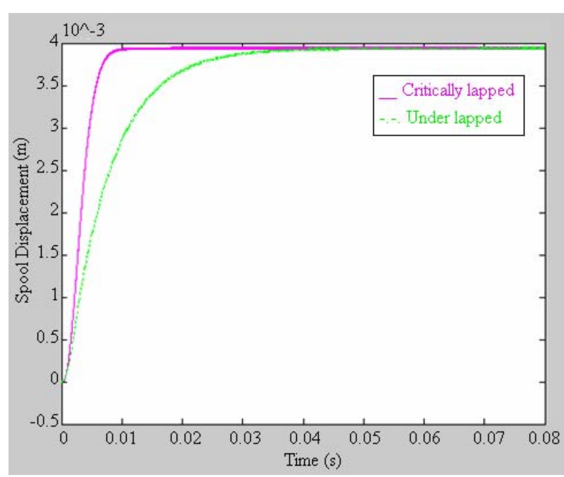

Figure10. Responses of servo valves neglecting back pressure

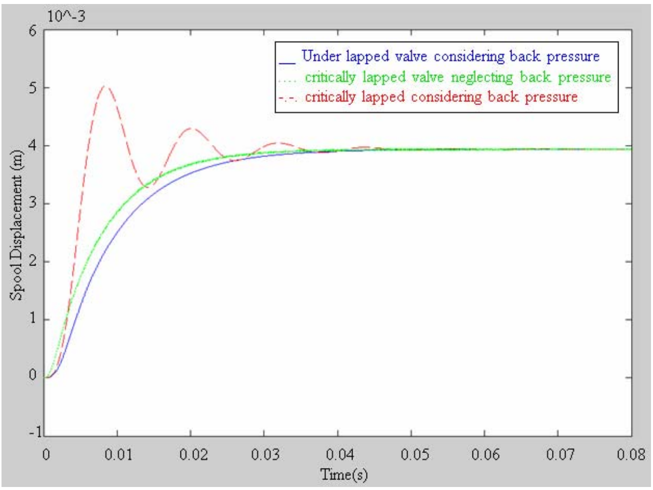

Figure11. Different valve response to a step input

In Fig. 11 three different responses to a step input are demonstrated. Solid line is the response of an under lapped valve with back pressure in the pilot region. Dotted line shows a critically lapped valve neglecting back pressure and dashed line represents the response of a critically lapped valve with back pressure in the pilot region.

These results show that the amount of underlap in the spool of a servo valve system would reduce the response time of the system considerably. Although the rise time of the valve with the back pressure is the lowest, the response of the under lapped valve is even slower than the critically lapped valve. As proved, the back pressure inside the pilot region also increases the response time of the servo valve. So to get into faster responses in the servo valve, the amount of under lap in spool and back pressure inside the pilot region have to be reduced.

Reducing this amount from under lap to zero lap, would increase the production costs considerably, but in some applications fast response is of vital importance and this would override the cost, but there is always no limitation in reducing the back pressure for the system.

IX. Conclusion

Using extensive theoretical analysis on functional parts of two-stage electrohydraulic servovalves with a spool position feedback, a more detailed set of dynamic equations considering the back pressure inside the pilot region is investigated. A new, improved model is developed incorporating both the circular ports associated with the spool valve's sleeve and the dead zone created by the underlapped spool valve landings which causes precise modeling of the valve. Such model primarily presents a useful design tool, since all its parameters are physical quantities. It could also be used as a fine base for the electrohydraulic servo systems design with the application of advanced control techniques.

The model was validated by comparing with empirical results. Different simulations for critically lapped and under lapped valve with considering and neglecting the back pressure is done in Matlab.

The responses indicate that having under lap in the spool increases the response of the servo valve and slows the system's response. Improvement could be possible by increasing the production precision to zero lap spools. The effect of increasing pressure in the pilot zone was oscillation (increasing overshoot and settling time) that is not acceptable in controlling systems. Avoiding this effect is possible by increasing the drain diameter and better filtration.

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