International Journal of Emerging Technology and Advanced Engineering

Volume 3, Special Issue 3: ICERTSD 2013, Feb 2013, pages 479-484

An ISO 9001:2008 certified Int. Journal, ISSN 2250-2459, available online at www.ijetae.com

A STUDY ON ELECTRO HYDRAULIC SERVOVALVE CONTROLLED BY A TWO SPOOL VALVE

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ABSTRACT

This paper establishes a non linear dynamic model of a two-spool electro hydraulic servovalve. It provides the ability to control flow into and out of valves independently by two separate spools. In this valve there is a flapper/pilot and the boost stage uses two spools instead of a single spool without any feedback wire. The feedback is obtained by pressure. It is likely reduce the valve cost substantially, give higher degree of adjustment and greater safety. In this paper dynamic behavior in respect of armature deflection, spool displacement, chamber pressure, discharge and displacement of the actuator is analyzed using Matlab/Simulink to see if such a low cost servovalve can provide acceptable performance.

Keywords: Servovalve, Separate spool, Simulation, Acceptable performance**.**

1. INTRODUCTION

In combination of electric and hydraulic device there must be a bridge element which closes the gap between those devices. This interface connection in control system is achieved by hydraulic servo valve. Such servo valve converts low power electrical signals into motion of a valve which in turn controls the flow and pressure to a hydraulic actuator. Probably, the electrical servo valve is one of the youngest of the standard hydraulic components. The vast majority of flow control servovalves in existence employ a double flapper nozzle pilot stage and a single spool boost stage. A stiff feedback spring is generally used to provide feedback from the boost stage to the pilot. These types of servo valves tend to be difficult to manufacture and expensive. A non conventional split spool valve provides the ability to control flow into and out of valves independently. It is less costly type of flow control servovalve utilizes a two-spool boost stage and a flapper control pilot. Because a feedback wire between the nozzle flapper pilot and boost stage is not needed, assembly is simplified. Moreover, precision machining of only two perfectly aligned control orifices instead of four control orifices makes it cheaper compared to a conventional four - way servovalve. The two spools in the boost stage are spring loaded and meter flow into and out of the valve separately. The main advantages of the two-spool, pressure control pilot design are: Ease of manufacturing, lower costs, higher degree of adjustment and Greater safety.

The rest of the paper is organized as follows. The basic operation of the valve is given in Section 2. The models for each subsystem are then developed. The models for the pilot and the boost stages are presented in Sections 3.1 and 3.2, respectively, the torque motor stage, armature dynamics and the chamber pressure dynamics are given in subsection A, B, C under pilot stage dynamics, load stage dynamics the output flow equations are given in Section 3.3 and 3.4 respectively. In Section 4 discussed the linearised Servovalve Model , Simulation issues are discussed in section 5. Results are shown in Section 6. Conclusions are given in Section 7. A table of nomenclature appears at the end of the paper.

2. OPERATION OF SPOOL VALVE

A schematic of the flow control servovalve using a two-spool boost stage, and pressure control pilot design is shown in Fig. 1.Mainly it has two distinct stages-a pilot stage and a boost stage. A Simple transition plate is between those stages. But those stages are connected through the chamber pressure. When the current applied on the electromagnetic torque motor, the armature creates a small angle with its initial position. As a result flapper moves to the right/left (depend on armature movements).Due to the flapper movements a differential pressure is arises on the chamber of the valve. This differential pilot pressure acts two ends of both spool and creates a spool displacement.

The steady-state displacements of spools are proportional to the differential pilot pressure, inversely to the total stiffness. A more detailed discussion of the operation of spool valve given with the help of fig 1.Suppose that the electrical current is the input to the coil of the torque motor at the top part of the valve. The current in the coil, together with the magnetic armature, generates a torque, which in turn rotates the armature and flapper in the clockwise motion about the pivot point (where the armature and the flapper intersect). Note that in Fig. 1, bold arrows correspond to the direction of fluid flow. As the flapper is displaced to the right (left),the nozzle opening on the right (left) decreases and the opening on the left (right) increases. This in turn raises P_a and lowers P_b (or vice versa) in the pressure chambers. The differential pressure ($P_a - P_b$) , therefore, has the effect of restoring the flapper to its neutral position.

The pilot pressures, P_a and P_b, act on the two ends of each spool. A positive (negative) differential pressure (P_a-P_b) causes both spools to move in the upward (downward) direction. As spools A and B move upward (downward), hydraulic oil is ported from the supply to port A (B) on one side, and from port B(A) into the tank on the other, creating flows Q_{L1} and Q_{L2} , respectively. As the spool displace, flows Q_x to and Q_y from the pilot stage are also created.

The regulation of the spool displacements (hence the flow rate) is achieved in two ways.

Primarily, displacements of the spools are resisted by the compression of the springs. Thus, in the steady state, the displacement of each spool would be proportional to the differential pilot pressure and inversely proportional to the spring stiffness.

Secondarily, the displaced fluid volume above and below the spools also tend to reduce the differential pressure. Therefore, the upward (downward) spool displacement and velocity tend to decrease (P_a - P_b). These effects in turn affect the flapper displacement and the differential pilot pressure. If the system is stable, the spool will reach an equilibrium displacement.

3. DYNAMICS MODEL

3.1 Pilot Stage Dynamics

There are three parts in Pilot Stage (A) Torque Motor Stage (B) Armature Stage and (C) Flapper Nozzle Stage or Chamber Pressure Dynamics.

(A)Torque Motor Stage

When current is made to flow through torque motor coils, armature ends become polarised. A torque is thus produced on the armature and it moves. Researchers mostly use a linear expression to calculate this torque:

$$T_d = K_i \Delta i + K_m \theta \tag{3.1}$$

Where values for K_i and K_m that depend on the connection of coils, are obtained experimentally.

$K_i = \frac{2aN\emptyset}{g}$ =Torque Motor Gain.

And $K_m = \frac{4a^2\emptyset^2}{g\mu_0 A_g}$=Torque Motor Stiffness.

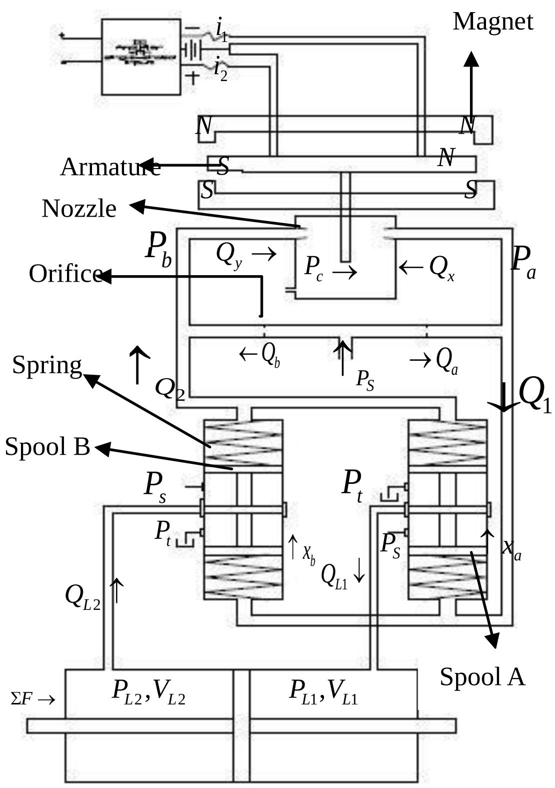


Fig-1: Schematic of the two spool flow control valve using a pressure control Pilot

B) Armature Dynamics

Under the influence of the torque caused by electromagnetic forces, an armature assembly (consisting of the armature and the flapper) moves, the armature assembly requires a flexure tube as an elastic support. The armature assembly and the flexure tube move and deform together. Although the movement of the armature assembly and the flexure tube are complex. Here the feedback wire is absent i.e. force feedback is not acting whether pressure acts as a feedback element. Appling Bernoulli Equation and Linear momentum equation in the right and left nozzle the flow force becomes

$$F_x = P_a.a_{n1} + 4.\pi.c_{qn}^2(x_{f0}^2 - 2.x_{f0}.x_f)(P_a - P_c) \quad ; [x_f^2 \ll 1] \tag{3.2}$$

$$F_y = P_b.a_{n2} + 4.\pi.c_{qn}^2(x_{f0}^2 + 2.x_{f0}.x_f)(P_b - P_c) \quad ; [x_f^2 \ll 1] \tag{3.3}$$

Torque due to asymmetric flow force

$$T_f = (P_a - P_b).a_n.r + 4.\pi.c_{qn}^2.x_{f0}^2(P_a - P_b)r - 8.\pi.c_{qn}^2.x_{f0}.r\theta.(P_a - P_b)r \quad (3.4)$$

Where, x_f= r.θ and $a_{n1} = a_{n2} = a_n$
Net torque on the torque motor

$$= T_d - T_f = J_a.\ddot{\theta} + B_a.\dot{\theta} + K_a.\theta$$

Replacing T_d and T_f we get

$$K_i.\Delta i = (K_a - K_m).\theta + (p_a - p_b)[a_n.r + 4.\pi.c_{qn}^2.x_{f0}^2.r - 8.\pi.c_{qn}^2 x_{fo} r^2.\theta] + J_a.\ddot{\theta} + B_a\dot{\theta} \quad (3.5)$$

(c)Flapper Nozzle Stage or Chamber Pressure Dynamics

Taking into the account that the effect of compressibility of the oil. The continuity equation in the nozzle chamber and the drain chamber is obtain as

For the right nozzle:

$$Q_1 = Q_a - Q_x = a_s.\dot{x}_a + a_s.\dot{x}_b + \frac{\forall_a}{\beta}.\dot{P}_a \quad (3.6)$$

For the left nozzle:

$$Q_2 = Q_y - Q_b = a_s.\dot{x}_a + a_s.\dot{x}_b \frac{\forall_b}{\beta}.\dot{P}_b \quad (3.7)$$

3.2. Boost Stage Dynamics

The model for the boost stage is developed next. We need to derive the dynamic equations that describe the two spools. We will analyze spool A in detail. The equations for spool B can be similarly derived. We assume the spools are designed to be critically centred. In Fig. 2, the spool is subjected to the differential pilot pressure, a force due to the centring spring, viscous friction and flow forces.

Equating forces on the spool yields the following:

$$m_s\ddot{x}_a = (P_a - P_b)a_s - 2k_s x_a - B_x\dot{x}_a - f_{xa} \quad (3.9)$$

Where RHS are forces due to the differential pressure, is the force due to the two centering springs of stiffness k_s , f_{xa} is the corresponds to the flow forces, and $B_x\dot{x}_a$ is the viscous damping force. In fluid flow force or hydraulic reaction force there are two types of forces- Steady state flow force and Transient flow force. Steady State Flow Force acting normal to the plain of fluid at the vena contracta , they depend on the flow rate and hence the spool displacement .Transient Flow Force, on the other hand, is the reactive forces associated with the acceleration of the fluid in the spool chamber. Thus it is depending on the rate of change of flow and the spool velocity. The flow force on the spool A

$$f_{xa} = \frac{c_q^2 \cos\theta w x_a}{c_c}(P_s - P_{L1}) + \rho.c_d w\sqrt{\frac{2}{\rho}}\left[\sqrt{(P_s - P_{L1})}\dot{x}_a - \frac{1}{2}\frac{x_a}{\sqrt{P_s - P_{L1}}}\dot{P_{L1}}\right]L_{s1} + m_{c\forall}\ddot{x_a} \quad (3.9a)$$

Similarly the dynamics for spool B can be similarly obtained:

$$m_s\ddot{x}_b = (P_a - P_b)a_s - 2k_s x_b - B_s\dot{x}_b - f_{xb} \quad (3.10)$$

And

$$f_{xb} = \frac{c_q^2 \cos\theta w x_b}{c_c}(P_s - P_{L2}) + \rho.c_d w\sqrt{\frac{2}{\rho}}\left[\sqrt{(P_s - P_{L2})}\dot{x}_b - \frac{1}{2}\frac{x_b}{\sqrt{P_s - P_{L2}}}\dot{P_{L2}}\right]L_{s2} + m_{c\forall}\ddot{x_b} \quad (3.10a)$$

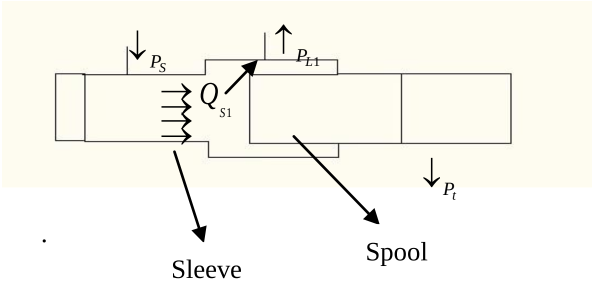


Fig-2 Flow paths when spool is displaced

3.3 Load Stage Dynamics

Assuming a symmetric critical- centre valve and a piston position with$L_1 = L_2 = L$, We find

$$\boldsymbol{C_q w x_a\sqrt{\frac{1}{\rho}}(P_s - P_L) = A\dot{y} + \frac{\forall_l}{2\beta}\dot{P}_L} \quad (3.11)$$

And

$$C_q w x_b\sqrt{\frac{1}{\rho}}(P_s - P_L) = A\dot{y} + \frac{\forall_l}{2\beta}\dot{P}_L \quad (3.12)$$

Where, $P_L = \frac{\sum F}{A} = \frac{M\ddot{y} + B\dot{y} + Ky + F_c(Sgn\dot{y})}{A}$ (3.13)

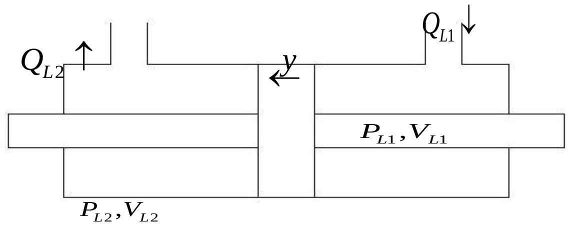


Fig-3 Load dynamics

3.4 Flow Equation Model

As the spools in the boost stage move, flow is either metered into or out of the valve through the orifice. Now for $x_a, x_b > 0$

$$Q_{L1} = c_q w x_a\sqrt{\frac{2}{\rho}(P_s - P_{L1})} \quad (3.14)$$

$$Q_{L2} = c_q w x_b\sqrt{\frac{2}{\rho}(P_{L2} - P_t)} \quad (3.15)$$

4. DERIVATION OF LINEARISED SERVOVALVE MODEL

From the mathematical modelling of two spool two stage electrohydraulic servovalve it is obvious that the electrohydraulic system is non-linear in nature. For present investigation the system is linerised from the non-linear mathematical model neglecting non-linearities like backlash, Coulomb friction and spool.

The coefficients of this derived model are expressed in terms of the system parameters so that servovalve performance can be improved by the prudent choice of the parameters.

The derivation of the linearised model is done with respect to an equilibrium state. The equilibrium state is derived for zero input i.e. input current Δi=0;

The complete linearised model is given as follow:

$$\Delta\ddot{\theta} = -C_1\Delta\theta - C_2\Delta\dot{\theta} - C_3\Delta P_a + C_4\Delta P_b + C_5\Delta i \quad (4.1)$$

$$\Delta\dot{P_a} = C_6\Delta\theta - C_7\Delta P_a - C_8\Delta\dot{x}_a - C_9\Delta\dot{x_b} \quad (4.2)$$

$$\Delta\dot{P_b} = -C_{10}\Delta\theta - C_{11}\Delta P_b + C_{12}\Delta\dot{x_a} + C_{13}\Delta\dot{x_b} \quad (4.3)$$

$$\Delta\ddot{x_a} = C_{14}\Delta P_a - C_{15}\Delta P_b - C_{16}\Delta x_a - C_{17}\Delta\dot{x_a} \quad (4.4)$$

$$\Delta\ddot{x_b} = C_{18}\Delta P_a - C_{19}\Delta P_b - C_{20}\Delta x_a - C_{21}\Delta\dot{x_a} \quad (4.5)$$

5. MATLAB / SIMULINK MODEL

The dynamic model for the torque motor stage,Eq. 4.1, spool dynamic Eq.4.4-4.5, chamber pressure dynamics Eq.4.2-4.3 and the load dynamics Eq. 3.11-3.13can be connected to each other, and to the hydraulic device, such as in fig 1 into a simulation model. The combined model will be capable of predicting the load flow into and out of the load ports, given the input of the input of the time trajectories of the electrical input current ,i, and two work port pressure $P_{L1}P_{L2}$ .In order to simplify the testing procedure, it will be convenient to assume that load flow in and out i.e. the valve is connected to volume conserving device (such as a double ended cylinder or loaded hydraulic servomotor).This allows us to specify only the load pressure $P_L = P_{L1} - P_{L2}$ instead of specifying P_{L1}, P_{L2} independently. To this end, we calculate $P_{L1} and\ P_{L2}\ given\ P_L$. $Equating$ $Q_a = Q_x$ We obtain-

$$P_{ao} = \frac{P_s}{1 + (\frac{C_{qn}}{C_{qo}})^2(\frac{4d_n x_{fo}}{d_o^2})^2} \quad and \quad P_{bo} = \frac{P_s}{1 + (\frac{C_{qn}}{C_{qo}})^2(\frac{4d_n x_{fo}}{d_o^2})^2}$$

In the fig4 that the operation of the two spool pressure control pilot valve with hydraulic actuator three feedback loop in design. One is principal feedback loop of the torque motor stage and the pressure dynamics. This feedback loop controls the differential pilot stage. Another loop involves the spool dynamics and pressure dynamics. And the last loop involves with load dynamics.

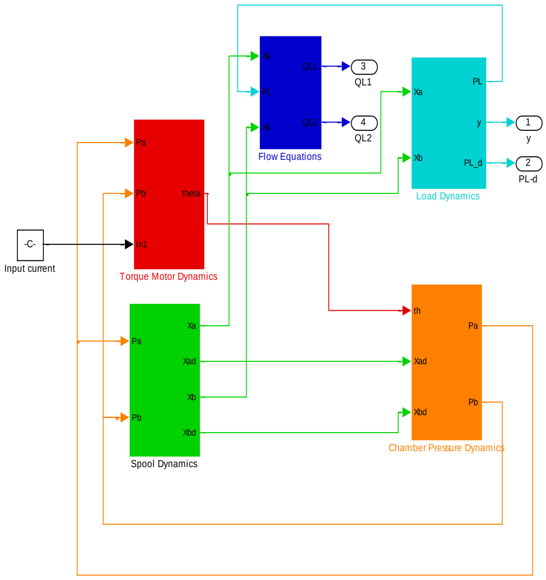


Fig-4, Simulink block diagram

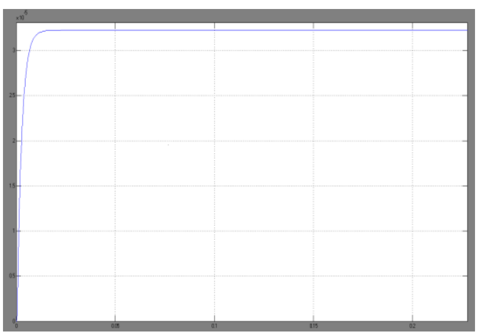
6. RESULT AND DISCUSSION

Matlab/simulink linearised simulation model for two spool split type EHSV with load hydraulic servomotor have been develop as shown in fig 4 using linearised dynamic equation 4.1 through 4.5. The configaration parameter of the simulink is variable step type,ode 15s solver. And using standard value of servovalve parameter.The steady-state load flow,for a rated current of 3mA and Ps=75bar, ω_b =10Hz, $\ddot{y}_{max}$=10 m/s^2 is obtained as $3.225e^{-005}$ m^3/min for mass of 1000 kg, as shown in fig M1.

Fig M2 through Fig M4 show servovalve spool displacement, spool velocity,actuator displacement, response respectively against unit step input of 3mA for mass load of 1000 kg only.All the responses match with physical behaviour of such a system. Also Fig M5 similar to the reference [6].

Fig M6 represent the actuator displacement at constant load aganist same unit step input and the result match with the physical behavior of physical system.

Load Flow



Time

FIG-M1,step response characteristics (Load flow vs. time)

Int. J Emerging Technology and Advanced Engineering
ISSN 2250-2459, *Volume 3, Special Issue 3: ICERTSD 2013, Feb 2013, pages 479-484*

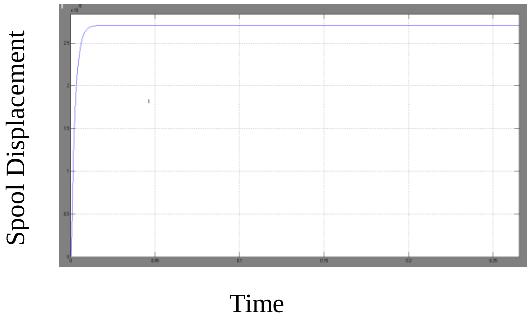


FIG-M2,step response characteristics(Spool Displacement vs Time)

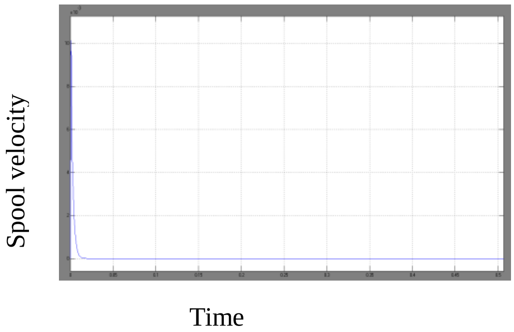


FIG-M3,step response characteristics(Spool velocity vs Time)

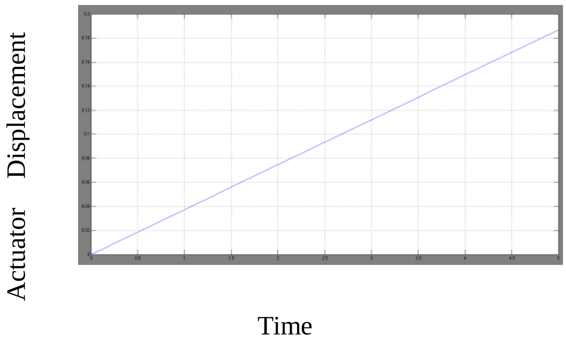


FIG-M4, Step response characteristics (Actuator Displacement vs Time)

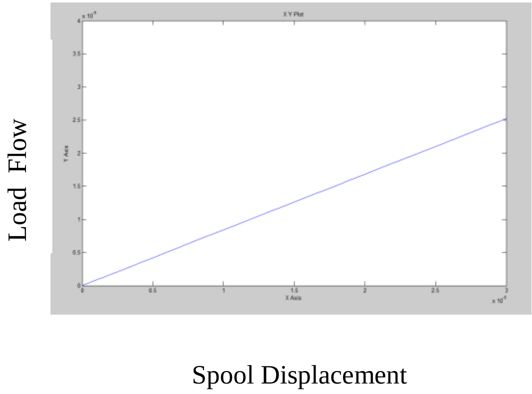


FIG-M5,Load Flow vs Spool Displacement

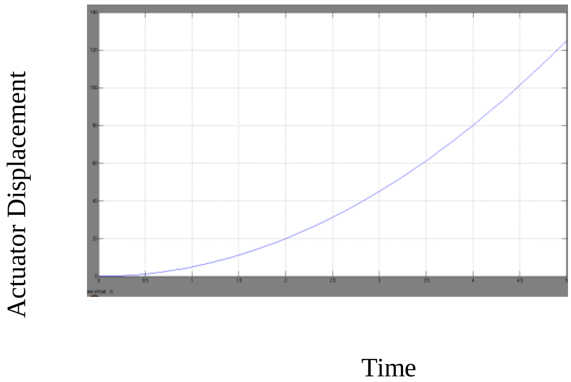


FIG-M6,step response (Actuator Displacement vs Time)

7. CONCLUSION

The mathematical model of the two spool split type two-stage ESHV under loaded condition has been develop. The model consist of the interconnection between the torque motor stage, spool dynamics, chamber pressure dynamics, output flow and load dynamics relationship. The model has been implemented using Matlab/Simulink. The proposed model can be use to predict performance and to provide insights for improving the design of the valve. Improved performances of this relatively inexpensive servovalve, either through improve physical design, or through the advance control, can potentially expand the use of electro hydraulics in cost constrained application.

NOMENCLATURE

a = half of armature length
a_n=Nozzle area
a_0=Discharge area of restrictor in pipe
a_s= Spool area
A_P=Cross section area of actuator
A_g=Air gap cross-section areas
b=Nozzle ball distance
B=Load damping co-efficient

B_a=armature viscous damping co-efficient
B_s=Spool viscous damping co-efficient
C_c=Coefficient of contraction
C_q=Discharge co-efficient
C_{qn}=Discharge Coefficient nozzle
C_{qo}=Discharge co-efficient of restrictor in pipe
d_n=Nozzle diameter
g =Air gap lengths at null
J_a= Mass moment of inertia of armature flapper
K_a=Armature-Flexture-Sleeve –Assembly Stiffness
K_m=Torque Motor Stiffness
K_s=Spring Constants of spool spring
K_i=Torque Motor gain
L_d=Length from work port to return port
L_{S1}=L_{S2}=L_P=Damping Length
L_s=Length From supply port to work port
$m_{c\forall}$=Hydraulic oil mass in the spool valve
m_s=Spool mass
M_P=Piston Mass
N=Number of turns
P_a=Pressure in right conduit
P_{ao}=Initial back pressure in the right nozzle
P_b= Pressure in left conduit
P_{bo}=Initial back pressure in the left nozzle
P_{ao}=Initial back pressure in the left nozzle
P_L=Load pressure
P_{L1}=Actuator chamber pressure at right side
P_{L2}=Actuator chamber pressure at left side
P_s=Supply pressure
Q_{L1}=Load flow at port A
Q_{L2}=Load flow at port B
r =Length from pivot to centre of nozzle
$\forall_a$=Volume of hydraulic oil in the left/lower chamber
$\forall_b$ =Volume of hydraulic oil in the right/upper chamber
w =Orifice area gradient
x_f=Flapper displacement
x_{fo}=Flapper to nozzle distance
$x_{s,a,b}$=spool displacement
y=Actuator piston displacement
$\dot{y}$=Actuator velocity
$\ddot{y}$=Actuator acceleration
θ_o=Armature null angular deflection
β=Bulk modulus of hydraulic oil
ρ=Hydraulic oil density
θ_f=Flow angel
Δi=input current
μ_o=Absolute permeability

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