DYNAMIC SIMULATION OF TWO-STAGE ELECTROHYDRAULIC SERVOVALVE

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1. Introduction

Servovalves are an important component of all electrohydraulic systems because they present the interface between electrical signals corresponding to a desired motion and force and the power elements such as actuators and motors. Due to the complexity and nonlinear behavior of servovalves and requirements for precise control of electrohydraulic systems, there is a need for a mathematical model that will give a good correlation between theoretical and actual responses over a wide range of operating conditions.

2. Principle of operation

Principle operation of a two-stage electrohydraulic servovalve with flapper-nozzle first stage and feedback spring, which is the commonest type of two-stage electrohydraulic servovalve, can be explained by the figure 1. The input signal is an electric current and the output signal is the oil volume flow through the valve. When a constant input current is applied to the torque motor, the combination armature-flapper will rotate over

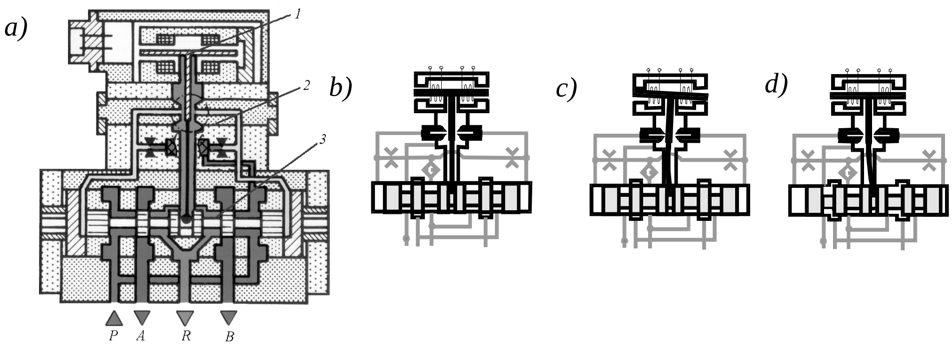

Figure 1. Two-stage electrohydraulic servovalve
a) scheme: 1 – torque motor, 2 – first stage (flapper-nozzle valve), 3 – second stage (spool valve), b) neutral position, c) flapper deflection, d) desired position

a small angle leading to a flapper deflection. This will result in a pressure difference over the spool causing the spool to slide. The spool position is fed back to the flapper by feedback spring so that equilibrium will be reached between the pressure difference over the spool, the torque of the torque motor and the feedback spring force (and the flow forces). The spool displacement will open and close different ports of the valve, leading to a flow between the pressure supply unit and valve control ports.

3. The mathematical model

A mathematical model of a two-stage electrohydraulic flow control servovalve is developed in this section. For that purpose, knowledge of fundamental laws of fluid mechanics, general mechanics and electronics is necessary. Applying these laws, relatively complex system of nonlinear differential equations is obtained:

$$\mathbf{A}\cdot i = \mathbf{C}\cdot\ddot{\theta}+\mathbf{D}\cdot\dot{\theta}+\mathbf{B}\cdot\theta+\mathbf{E}\cdot y+\mathbf{F}\cdot p_d+\mathbf{G}\cdot\theta^2\cdot p_d-\mathbf{H}\cdot\theta\cdot(p_{1L}+p_{1D}), \qquad (1)$$

$$p_d = p_{1L}-p_{1D}, \qquad (2)$$

$$\mathbf{I}\cdot\sqrt{p_s-p_{1L}} = \mathbf{J}\cdot\sqrt{p_{1L}-p_r}-\mathbf{K}\cdot\theta\cdot\sqrt{p_{1L}-p_r}+\mathbf{L}\cdot\dot{y}+\mathbf{M}\cdot\dot{p}_{1L}+\mathbf{N}\cdot p_d, \qquad \text{(3-a)}$$

$$\mathbf{I}\cdot\sqrt{p_s-p_{1D}} = \mathbf{J}\cdot\sqrt{p_{1D}-p_r}+\mathbf{K}\cdot\theta\cdot\sqrt{p_{1D}-p_r}-\mathbf{L}\cdot\dot{y}+\mathbf{M}\cdot\dot{p}_{1D}+\mathbf{N}\cdot p_d, \qquad \text{(3-b)}$$

$$\mathbf{T}\cdot p_d = \mathbf{P}\cdot\ddot{y}+\mathbf{X}\cdot\dot{y}+\mathbf{R}\cdot y+\mathbf{S}\cdot\mathrm{sgn}(\dot{y})+\mathbf{U}\cdot\theta, \qquad (4)$$

$$Q_{SV} = \mathbf{W}\cdot y, \qquad (5)$$

with three known quantities: input current – i, tank pressure – p_r and supply pressure - p_s, and six unknown quantities that should be determined: angle of flapper deflection – θ, spool displacement – y, pressure difference over the piston – p_d, pressure at left (right) nozzle and left (right) side of the spool – p_{1L} (p_{1D}) and volume flow through servovalve – Q_{SV}. In equations (1)-(5) bold letters denote:

$$\mathbf{A}=k_i,\ \mathbf{B}=(K_k-k_m+K_{ops}\cdot(r+b)\cdot(r+b)),\ \mathbf{C}=J_k,\ \ \mathbf{D}=b_k,\ \mathbf{E}=K_{ops}\cdot(r+b),$$

$$\mathbf{F}=r\cdot\frac{d_m^2\cdot\pi}{4}+4\cdot K_m^2\cdot\pi\cdot r\cdot l^2,\ \mathbf{G}=4\cdot K_m^2\cdot\pi\cdot r^3,\ \ \mathbf{H}=8\cdot K_m^2\cdot\pi\cdot r^2\cdot l;\ \mathbf{I}=\sigma_0,$$

$$\mathbf{J}=\sigma_{m0},\ \mathbf{K}=\sigma_{m0}\cdot\frac{r}{l},\ \mathbf{L}=A_k,\ \mathbf{M}=\frac{V_{t1}}{2\cdot\beta},\ \mathbf{N}=K_L;\mathbf{T}=A_k,\ \mathbf{P}=m_k,$$

$$\mathbf{X}=k_{vtr}+(l_{ot2}-l_{ot1})\cdot K_{ot}\cdot f\cdot\sqrt{\rho\cdot(p_s-\Delta p_L-p_r)},\ \mathbf{U}=K_{ops}\cdot(r+b),\ \mathbf{S}=|F_{str}|,$$

$$\mathbf{R}=2\cdot K_{ot}^2\cdot f\cdot\cos\alpha\cdot(p_s-\Delta p_L-p_r)+K_{ops},\ \mathbf{W}=K_{ot}\cdot f\cdot\sqrt{\frac{2}{\rho}}\cdot\sqrt{\frac{p_s-p_r-\Delta p_L}{2}},$$

where physical quantities from right side of this equations are presented in table 1. Values are taken from available literature (see detailed in [2]).

3. Numerical simulation

Considering the fact that obtained system of nonlinear differential equations is very difficult for analytical solving, numerical solution was made using software package MATLAB/ SYMULINK. Numerical integration was performed using Runge-Kutta and Adams-Gear algorithms and obtained results were nearly identical. Integration parameters were: integration interval 0 ÷ 0.2 s, minimal step size: $5\cdot10^{-7}$ s, maximal step size - 0.1 s and tolerance 10^{-6}. Time dependencies of physical quantities that determine dynamic characteristics of two-stage servovalve are presented in figures 2-6.

Symbol	*Physical quantities*	*Value*	*Unit*
α	Flow angle in and out of spool compartment	69	^{o}
b	Feedback spring length	12.7	mm
β	Hydraulic fluid bulk modulus	$1.1 \cdot 10^9$	Pa
b_k	Armature damping coefficient	$1 \cdot 10^{-2}$	N·m·s/rad
d_k	Spool diameter	16	mm
d_m	Nozzle diameter	2.5	mm
d_o	Orifice diameter	0.71	mm
f	Spool port width	4.06	cm
$\lvert F_{str} \rvert$	Coulomb friction force on the spool	0	N
J_k	Armature moment of inertia	$5 \cdot 10^{-6}$	N·m/s^2
k_i	Torque motor torque constant	1.14	N·m/A
K_k-k_m	Difference of flexture tube stiffness and torque motor magnetical stiffness	0	N·m/rad
K_L	Spool leakage coefficient	$6.56 \cdot 10^{-12}$	m^3/(s·Pa)
K_m	Flapper-nozzle configuration discharge coefficient	0.6	-
K_o	Flow through the ports discharge coefficient	0.67	-
K_{ops}	Feedback spring stiffness	$1.24 \cdot 10^4$	N/m
k_{vtr}	Spool viscous coefficient	3.32	N·s/m
l	Equilibrium flapper deflection	76.2	μm
l_{ot1}, l_{ot2}	Port distance (P-L), (L-R)	50	mm
m_k	Spool mass	25.2	g
p_r	Tank pressure	1	bar
r	Flapper length	19	mm
ρ	Hydraulic fluid density	850	kg /m^3
V_{t1}	Connection channel + spool compartments volume	$6.44 \cdot 10^{-6}$	m^3

Table 1.

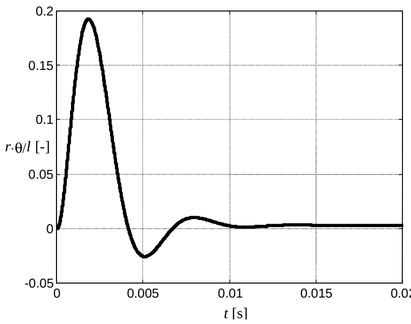

Figure 2. Relative flapper displacement

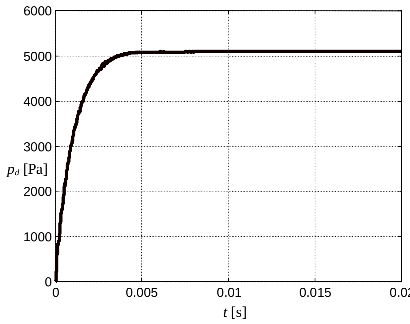

Figure 3. Pressure difference over the piston

4. Conclusion

This mathematical model is not only intended for analysis of existing servovalves. Primary it can play a significant role in new servovalve design by obtaining optimal physical quantities that can bring desired servovalve characteristics.

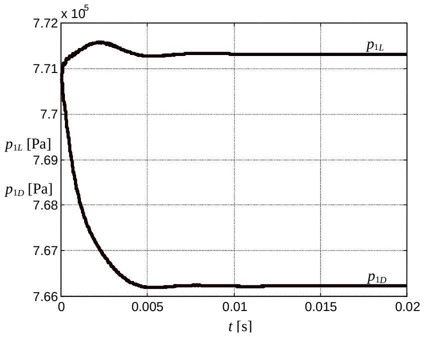

Figure 4. Pressure at left (right) nozzle

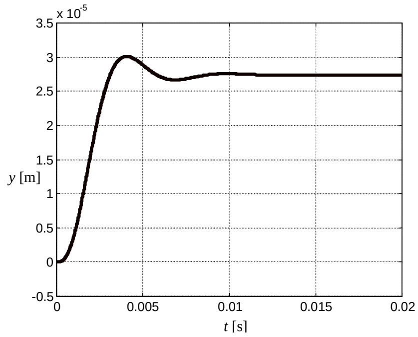

Figure 5. Spool displacement

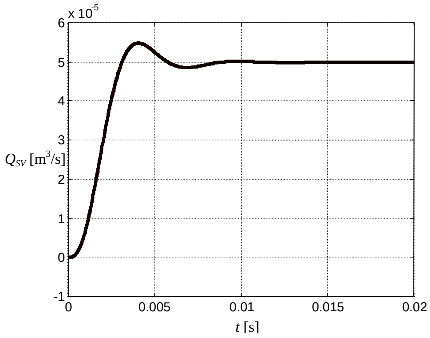

Figure 6. Volume flow through servovalve

This work should be understood as the first and the most difficult step in investigation of servovalve dynamics. Future research must comprise experimental verification of obtained results and possible model correction.

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